An investigation into miniature hydraulic actuation and control techniques for use on high-speed reciprocating mechanisms

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AN INVESTIGATION INTO MINIATURE HYDRAULIC ACTUATION AND CONTROL TECHNIQUES FOR USE ON HIGH SPEED RECIPROCATING MECHANISMS

by

DAVID STUART CAMERON B.Sc, C.Eng, M.I.MechE

A Doctoral Thesis

Submitted in partial fulfilment of the requirements for the award of Doctor of Philosophy of Loughborough University of Technology

October 1979

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SYNOPSIS

The research presented in this thesis relates to an actuation and control concept whereby miniature hydraulic mechanisms are sequenced by rotary spool valves. The objectives of the work were to advance the technology, further develop the control technique and improve the engineering of the associated hardware, to enable the concept to be applied to high speed reciprocating motions used in manipulative machinery. This necessitated improved and predictable actuator and valve performance to overcome deficiencies apparent in previous hardware.

The research was concerned initially with investigations into the performance limitations of previous actuator and valve designs as the basic requisite for uprating and extending their performance. Based on theoretical considerations and the results of these investigations, improved actuator and valve designs were conceived, analysed, manufactured and tested. Theoretical equations of motion of mechanisms were derived for a generalised actuator/valve circuit and correlated with the measured performance levels of various prototype mechanisms.

The detailed theoretical and practical investigations yielded the appropriate technology for the correct application of the actuator/valve concepts to reciprocating mechanisms. The basic circuit elements were subsequently incorporated into the design of an interlock sewing machine to suit a specific industrial application. The visual evidence of consistent and reliable stitch formation at the predicted performance levels, is offered as vindication of the overall actuation/control concept and as evidence that fluid power can compete with and offer advantages over, solid member mechanisms.

For ease of comprehension the research has been presented in five main sections.

SECTION 1: Miniature Actuators - design concept and analysis; resulting from design investigations into the operating deficiencies
of previous actuator designs, two new concepts of miniature actuators are presented. One design being a single acting module and the other being a double-acting version. Both designs incorporate a novel method of hydraulic cushioning, have no friction seals and are capable of compound movements. The various operating modes that can be obtained from the actuators are outlined together with their general operating characteristics. An analysis to obtain theoretical motion of different mechanism configurations when reciprocated by the actuators is presented. The derived equations of motion are analysed, their practical application discussed and the relative importance of the system parameters identified.

SECTION 2: Practical Mechanism Performance. Prototype mechanisms based on the actuating concepts outlined in SECTION 1 are tested to determine the accuracy of theoretical performance predictions and consistency of the operating characteristics. The essential points that must be observed to obtain accurate correlation between theoretical predictions and measured performance are 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, is presented.

SECTION 3: The Design and Analysis of Rotary Valves. 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 selection facility and improved performance is presented. An analysis of theoretical valve losses to minimise leakage and drag losses against specific operating parameters is also presented.

SECTION 4: The Testing of Rotary Valves. Prototype valves designed and balanced by methods outlined in SECTION 3 are tested and compared with previous valve designs and theoretical predictions.
The validity of the design is confirmed on the sequencing of both single actuator and multi-actuator circuits. The compactness of the new design is demonstrated by a small valve module having a restricted pulse selection facility.

SECTION 5: A Mattress Sewing Machine. Using the valve and actuator concepts, the design and testing of a lockstitch sewing machine is presented as an example of the flexibility and predictability of the system elements. Also illustrated are hydraulically actuated needle and looper mechanisms which form part of a candlewick tufting machine currently under development at the University.
ACKNOWLEDGEMENTS

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INTRODUCTION

Engineers within the manufacturing and process industries have certain preconceived ideas regarding the use of hydrostatic power systems. Whilst it may be appreciated that fluid power has certain advantages for reciprocating mechanisms, its use is usually restricted to the more expensive, larger and slower applications. Similarly mechanisms which operate at the other end of a financial and performance spectrum, that is simple high speed mechanisms, are generally held to belong to solid member devices such as cam and linkage mechanisms. However, previous research at the University of Technology has demonstrated that the exclusion of fluid power from the field of high speed reciprocating mechanisms is not necessarily valid. In support of this claim, a circular weft knitting machine was successfully developed whereby individual needles were powered by miniature hydraulic actuators. To obtain the correct displacement profile suitable for knitting, the control and sequencing of individual needles was obtained by a rotary spool valve.

It was considered by the researchers that this basic actuator/valve concept, although conceived initially to meet a specific knitting application, had considerable and perhaps better potential in the general field of high speed manipulative machinery. It is possible to identify numerous examples of where a linear reciprocating motion is obtained, via a tortuous succession of cams and linkages, from a rotary input. It is on this very type of application, that fluid power can compete with its solid member counterpart.

The final orientation of the linear motion and the routing of the power source through conduits, are matters of convenience and not of kinematics. These basic advantages of fluid power are recognised by machine designers and manufacturers but for applications which demand fast response times and correctly phased motions between associated mechanisms, solid member devices are used almost exclusively. It is believed that the underlying reasons for this do not relate to any inherent deficiencies in the use of fluid power, but rather that fluid power technology has been concentrated in other areas. Conse-
quently little or no hydraulic hardware, expertise or examples relating to this application of fluid power, is available for technical consideration.

The knitting machine developed at the University of Technology, is an example where fast response times were required and where the phasing of individual mechanisms (needles) was critical. In this application, fluid power demonstrated its capability to produce higher performance levels than those possible using conventional mechanical machines. It was not claimed by the researchers that anything revolutionary had been invented, the fact that linear rams were used by Bramah in the first hydraulic machine and rotary spool valves were used by the Romans would deny any such claim. However, their combined use, to achieve the high speed synchronised movement of 96 knitting needles, was indeed unique and indicated that it was possible to attain the other end of the performance and cost spectrum.

The actuators and valves which were developed for the knitting machine application, although adequate to demonstrate the feasibility of the basic concept, were known to have certain operating deficiencies. However, owing to sponsorship restrictions, and the necessity of producing prototype machines, these deficiencies were not satisfactorily eliminated in the early research. Whilst this was considered to be acceptable as a short-term solution, it was known that the basic circuit elements would need refining and the technology would need to be rigorously investigated before the concept could be applied to commercial machines or to similar mechanisms. Despite these limitations, the success of the knitting machine generated additional sponsorship by:

b) Matramatic Company Limited.
c) The University of Technology.

This sponsorship enabled the author to undertake a second stage of research having the objectives of advancing the technology, further developing the control technique and improving the engineering of
This thesis therefore represents a combination between a "complement" and a "follow on" to the earlier research\(^1,2\). On this occasion the research was not unduly hampered by sponsorship restrictions and had the benefit of hindsight. The author is therefore indebted on these counts.

It was considered at the outset, that the ultimate success of the research would depend on how the basic actuator/valve concept would be received and applied by ordinary engineers. It was believed that, to aid its acceptance as a viable alternative to mechanical systems, the inherent simplicity of the concept should be matched by the final simplicity of both the hardware and the technology. Theoretical and design work was therefore carried out with this in mind. In general, analytical work was followed by an appraisal of the physical consequences of theoretical predictions and coupled with practical recommendations and design procedures. There is naturally a limit to how far a simplistic attitude may be taken before the idea becomes self defeating. It is hoped that a reasonable compromise between overwhelming the concept with unnecessary sophistication and perhaps omitting pertinent factors, has been realised. The level of knowledge assumed necessary for the design function to be effective was a reasonable competence in hydraulic/mechanical engineering. It was felt that, for example, having included dynamic viscosity as part of an overall "viscous damping coefficient" and indicated the effect of changes of the coefficient on system performance, fluid viscosity/temperature effects would be known and would not therefore require further elaboration.

For convenience and ease of comprehension, the thesis is presented in six main sections.

Section 1 deals with the design concept and theoretical analysis of miniature cushioned actuators.

Section 2 relates to some of the practical tests carried out on actuators to establish their functional and dynamic fidelity against design and theoretical expectations.

Section 3 presents a new design concept and analysis of a rotary spool valve plus a technique for applying it to multi-actuator circuits.
Section 4 relates to tests carried out on rotary valves to establish the correlation between measured and theoretical performance.

Section 5 presents, as a specific example, the design and development of a mattress sewing machine which embodied the hardware and techniques developed. Also illustrated are hydraulically actuated mechanisms for the needle and looper control of a candlewick tufting machine.

Section 6 summarises the main conclusions.

Appendices have been used to include certain analytical and detail work believed will be of use in understanding and applying the concepts incorporated in the main body of the thesis. It is hoped that this leaves the bulk of the thesis uncluttered of detail. Similarly, experimental results have been incorporated in a separate compendium at the end of the thesis. Where experimental traces have been included these are copies of original U.V. traces in the author's possession.

Although the thesis is presented in discrete areas, mainly for the sake of clarity, it is to be appreciated the research did not proceed along such clearly defined paths. It is the nature of research and development that much 'leapfrogging' and sidetracking occurs which, if presented fully and chronologically, would only bore and confuse the reader. There has by necessity therefore, been much pruning to leave only what is considered to be the potentially useful information. The basis for assessing "usefulness" was whether the information would enable the non-specialist mechanism designer to understand and apply successfully the actuation and control concept. The information necessary to achieve this will depend not only upon the particular individual and the extent of his subject knowledge, but also the requirements of the mechanism to be controlled. Inevitably, the assessment of usefulness was personal, however, as the author's subject knowledge was limited at the onset of the research, it is believed that the information personally required would be typical of other designers. It remains for others to judge objectively whether the author has been successful on this account.
SECTION ONE

MINIATURE ACTUATORS: DESIGN CONCEPT AND ANALYSIS

1.1 Introductory Summary

For a hydraulic actuator to replace successfully solid member mechanisms which operate with response times measured in milliseconds rather than seconds, it is essential that inherent system losses are minimised. Furthermore, as the basic actuator/valve system is not conservative, it is also necessary that power wastage, by incorrect actuator sizing for a particular mechanism, be avoided. These basic considerations and others relating to the overall system performance imply that the dynamic behaviour of hydraulically powered mechanisms should be known at the onset of design. This section relates to the design of miniature cushioned actuators and their predicted dynamic behaviour when forming part of a rotary valve/actuator circuit. Although some of the information presented is particular to the actuators developed for this research, the majority will be applicable to a variety of actuator types.

The need for an improved actuator design was highlighted by the deficiencies of the actuators used in the knitting machine developed by Garside. Although these deficiencies are given in more detail later, the principal shortcomings of the actuators were that:

a) They had no form of controlled deceleration.

b) They were fitted with high pressure friction seals which leaked severely after a limited number of cycles.

Initial investigations were therefore carried out to find suitable solutions to these problems within an economic and performance specification believed to be necessary for the ultimate success of the research.

A novel method of achieving full hydraulic cushioning of miniature actuators was conceived by the inclusion of what has been described as an auxiliary piston or sleeve. The cushioning
concept was incorporated into designs of single-acting and double-acting actuators. These actuators also had clearance seals to replace the friction seals used in previous designs. Clearance seals having been considered to offer the best compromise between the opposing demands of low leakage and low wear rates/ friction losses. Both single-acting and double-acting actuators are capable of compound movements by using the auxiliary piston or sleeve as an independent actuator. However, single-acting units were in general preferred because of their manufacturing simplicity. It was also considered that, although the double-acting design was not suitable for actuators of piston sizes less than 5 mm, the single-acting version could operate satisfactorily with a piston size as low as 1 mm.

Following the identification of the various actuator operating modes plus the necessary control port switching sequences, the differential equations of motion of a typical actuator/valve circuit are presented. To simplify the subsequent analysis, the differential equations were linearised so that results could be presented in the form of equations of motion containing only five main variables, namely: Supply Pressure, Actuator Area, Apparent Mechanism Mass, Viscous Damping Coefficient and Spring Force. This simplification was subsequently verified by checking "analytical" solutions against iterative solutions obtained from a digital computer.

Equations of motion relating to double-acting, single-acting spring return and cushioning motions are presented. These equations were in turn analysed to obtain a more complete understanding of the relevance of the performance limiting variables plus the practical implications and subsequent use of the equations.

1.2 Research Objectives

Following an initial period spent in becoming familiar with the earlier research and evaluating the prototype hardware, a series of collaborative meetings were held between Dr J D Garside, Mr T P Priestley and the author. The previous work was exhaustively discussed, its limitations identified and the future research objectives established. The objectives of further work on miniature
actuators and the reasoning related to them is given below:

1. To improve the dynamic performance of the actuator:

   It was known that a weak point existed between the actuator piston rod thread and the attached mechanism (Fig. 1.01) which limited the operating life of the actuator. The principal cause of failures was attributable to the lack of any controlled mechanism deceleration towards the end of actuator travel. The lack of controlled deceleration resulted in high induced forces in the piston rod thread which was of insufficient area to limit the stress to a satisfactory level. This in turn limited the operating speeds and/or the maximum mechanism mass which could be successfully operated by the actuator. (Appendix 1.1). In addition the actuator exhibited:

   a) Assymetric dynamic profiles between the inward and outward movement attributable to the area difference of the double acting pistons (see 1.6.2).
   b) Overshoot of the displacement step sequence.
   c) Instability of the mechanism during dwell periods caused by the open hydraulic stop ports.
   d) Power wastage through open hydraulic stop ports during dwell periods.

2. To reduce manufacturing difficulties:

   The basic design of two stepped diameter pistons mounted in series caused:

   a) Concentricity problems between individual components.
   b) Slenderness of the main actuator.
   c) A high length to diameter ratio of the main bore.

   These factors created manufacturing difficulties and resulted in some sticking of the actuators at lower operating pressures.

3. To conduct a theoretical analysis of the performance characteristics of the actuator when controlled via a pulse generating rotary valve:

   Early test actuators and subsequent prototype machinery designs were based on practical experience only. The aims of a theoretical
Mechanism

Piston rod

End cap

O Ring seal

Main actuator

Hydraulic stop

Tuck probe

Tuck probe hydraulic stop

End plug and retaining plate

Displacement

Knit amplitude

Tuck amplitude

Revolutions of rotary valve

Port number

Pressure switching sequence

--- Pressure

--- Exhaust

FIG 1,01 FINAL TEST ACTUATOR (GARSIDE 1970)
analysis of the dynamic performance of the actuators were to
provide information relating to:

a) The maximum operating speeds of mechanisms.
b) The induced vibrations to associated machinery.
c) The power input requirements.
d) The correct sizing of actuators to a particular duty.
e) The identification and relevance of the performance limiting
variables of typical mechanisms.
f) The induced stresses within a system.
g) The correct phasing of mechanisms in multi-actuator circuits.

4. To investigate alternative seal designs:

Owing to space limitations, use had been made of 'O' rings
for the high pressure sealing of the piston rod. These seals had
the following disadvantages.

a) 10-25 bar was required to overcome the static friction induced
by their compression.
b) The seal exhibited a slip-stick phenomena.
c) The seal friction reduced the maximum cycling rate of the
actuator.
d) The seal friction represented a significant heat source and
caused the actuators to run hot.
e) The combined effect of high rubbing velocities and of generated
heat caused a rapid deterioration in the effectiveness of
the seal. The seals began to leak after 3-4 hours running.

5. To determine the accuracy of theoretically predicted mechanism
performance:

This was to be carried out by a prototype build and test pro-
gramme of single and multi-actuator circuits.
1.3 Actuator Design

1.3.1 Design Specification

In addition to overcoming the limitations outlined in Section 1.2 consideration of possible applications of miniature mechanisms led to the formulation of the following design specification for the actuator.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Specification</th>
</tr>
</thead>
<tbody>
<tr>
<td>Diameter</td>
<td>The design concept was to be suitable for actuator sizes within the range 2-10 mm piston diameter.</td>
</tr>
<tr>
<td>Stroke</td>
<td>Up to a maximum of 8 times the piston diameter.</td>
</tr>
<tr>
<td>Life expectancy</td>
<td>Minimum of 100 million cycles.</td>
</tr>
<tr>
<td>Cost</td>
<td>Designed for low unit cost - the actuator had to be capable of being manufactured using conventional machine tools.</td>
</tr>
<tr>
<td>Dynamic performance</td>
<td>The actuator must have:</td>
</tr>
<tr>
<td></td>
<td>a) Controlled deceleration.</td>
</tr>
<tr>
<td></td>
<td>b) Be capable of simple reciprocation.</td>
</tr>
<tr>
<td></td>
<td>c) Be capable of compound reciprocation.</td>
</tr>
<tr>
<td></td>
<td>d) Have operating speeds comparable with or better than previous actuator designs.</td>
</tr>
</tbody>
</table>

1.3.2 Pre-Design Investigation - Seals and Cushioning

The two principal problems which had not been successfully overcome in the earlier research were those of effective high pressure seals and the controlled deceleration of the actuator. The first requirement of a revised actuator design was to investigate these problems in depth.

Seals: The effective sealing of reciprocating shafts is a difficult problem even on applications where space limitations do not exist or where dynamic performance and cost are of secondary importance. Dr Garside had followed the conventional use of '0' rings for the sealing of the piston rod that is adopted by many
manufacturers of hydraulic/pneumatic cylinders.

The results of a somewhat negative and frustrating investigation into seal types\textsuperscript{3,4} indicated that the viable alternatives to the '0' ring were few. For most simple seal types some leakage was to be expected and to reduce this leakage to a minimal level at higher operating pressures, required a high degree of design and manufacturing sophistication. Design studies indicated that to achieve the required sophistication would be costly and further, that for most known seal types, inherent friction would have a detrimental effect on the dynamic performance of an actuator. Thus it was considered that as it was not possible to satisfy the opposing demands of low leakage and low wear rates/friction losses, a compromise solution was necessary.

Of the simple seal types which were investigated, the one which offered the best compromise in the design of miniature actuators was the "labyrinth" or "clearance" seal. These seals have no physical contact between adjacent sliding members but rely on a small clearance between members to limit the resultant leakage to an acceptable level. The advantages of removing friction seals were considered to be:

i) Improved actuator response time.
ii) Simplification of analysis and design.
iii) Elimination of a heat source.
iv) Simplification of manufacture.

The disadvantage of labyrinth seals was that, although the predicted leakage was low, it would be significant and the fluid would still require containing and draining back to its source. However, the practical problems of achieving this were considered to be substantially reduced as the leaked fluid would be at a pressure only nominally above atmospheric pressure. A small amount of leakage was not considered necessarily to be a disadvantage. During its containment it was believed that the leaked fluid could be usefully employed in the lubrication of ancillary mechanisms.
This might obviate the necessity of additional lubricating systems often used on more complex machinery. This technique is sometimes used on hydraulically actuated machine tools where slideways and bearings are lubricated by the leakage from hydraulic components. Similarly no attempt is made to seal the individual pistons embodied in some piston pump/motor designs.

Following the practical verification that the leakage rates predicted by design studies were realistic (Appendix 1.2) it was decided that clearance seals would be used in the designs of future actuators.

Hydraulic cushioning: The usual technique for decelerating a hydraulic cylinder is by means of a dashpot and needle valve, see Figure 1.02. Towards the end of the stroke a spigot on the piston enters a bore of nominally the same diameter. This effectively blocks the passage of oil to the exhaust port and the flow of oil is diverted through a smaller passage. This smaller passage usually contains a needle valve or similar device for controlling the final speed of the cylinder. To prevent the oil flow from being restricted when the direction of cylinder motion is reversed, it is necessary to fit a small check valve as indicated. This design arrangement was considered impracticable for the sizes of actuators envisaged. Physical limitations prevented the design from being used on cylinders below 20 mm bore diameter.

In conventional cylinder designs the final actuator impact speed is a function of the clearance between the cylinder bore and the spigot and also the orifice area of the by-pass passage. In operation, the needle valve would be adjusted until the desired amount of cushioning is achieved and then locked in position. Functionally, thereafter, the needle valve could be replaced by a fixed orifice/restriction to achieve the same degree of cushioning.

An alternative solution to this conventional design was conceived by combining the functions of the check valve and needle valve into the inclusion and operation of what is described as an "auxiliary piston", see Figure 1.03. By use of the auxiliary piston it was considered possible to obtain a cushioned motion in one direction whilst having an unrestricted reverse motion. It was believed
FIG 1.02 CONVENTIONAL CUSHIONING OF A HYDRAULIC CYLINDER

FIG 1.03 COMBINED CHECK VALVE AND AUXILIARY PISTON
that hydraulic cushioning would be achieved if, when the main piston closed port 2, a volume of oil could be trapped between the main and auxiliary pistons. Once port 2 had been closed, further movement of the main piston would then only be possible either by compressing the entrained fluid or, by fluid leaking past the auxiliary and main pistons into the supply port. The movement possible by the compression of the fluid was however considered insignificant in view of the small volumes envisaged. It was believed that by controlling the dimensions of the pistons these could be made into fixed restrictions, tailored to achieve full hydraulic cushioning.

To obtain an unrestricted reverse motion of the actuator it was envisaged that the auxiliary piston would operate as a check valve as follows: with the actuator fully retracted, and hence port 2 closed, with pressure applied to the supply port, the flow of oil to the rear of the main piston would be severely restricted. However, pressure would be applied simultaneously to the rear of the auxiliary piston. If the effects of the mass of the auxiliary piston and the force of the check spring are low, the nominal applied pressure would then be transmitted to the entrained fluid and hence to the rear of the main piston. The actuator would therefore move out with an unrestricted motion. As actuator movement occurs and port 2 is cleared, the auxiliary piston would then be subjected to nominally equal pressures at either end. It was originally envisaged that its movement would be arrested and reversed by the force exerted by the check spring.

In addition to enabling a cushioned motion to be obtained, it was considered that the auxiliary piston could also act as an actuator in its own right. By separating the supply ports 1 and 2 in Figure 1.03 the design would become the equivalent to two single-acting pistons in series and therefore compound actuator movements could be obtained by pulsing the pistons independently.

1.3.3 Design Concept

The concept of an auxiliary piston was adapted for the rod end of a double acting cylinder and a cushioned unit was designed. See Figure 1.04. To simplify manufacture and to obtain compound actuator movements, the check spring behind the auxiliary piston
FIG 1.04 A DOUBLE ACTING CYLINDER WITH HYDRAULIC CUSHIONING
was omitted and the supply ports separated. It was believed that the design had much to commend it and that it would be capable of producing the same dynamic profiles as previous actuator designs. However, in common with such designs, the existence of the piston rod bearing/labyrinth seal resulted in an unnecessary increase in the overall diameter of the actuator. This would have precluded its use on close pitching applications. Also in common with existing actuator designs it was believed that the unit:

a) Would not be suitable for very small actuators. In view of the resultant diameter of the piston rod, it was felt that the practical limit of this design was for an actuator having a main bore of 5 mm diameter and a piston rod diameter of 3 mm.

b) Would suffer from concentricity errors between the piston rod and both the rod bearing and the auxiliary sleeve.

c) Would produce asymmetric dynamic profiles due to the difference in pressure areas between the piston end and rod end.

Attempts to improve on this design for a double-acting cylinder were unsuccessful and eventually single-acting actuators were considered as a possible alternative.

A single acting actuator had been rejected by Garside as being unsuitable for his knitting application but because of its inherent simplicity it was reconsidered. Its principal advantage was considered to be the removal of a stepped piston and the concentricity problems that this eliminated. The cushioned single acting unit which was envisaged, can be seen in Figure 1.05. The actuator consisted simply of a rod in a housing counterbored to accept the auxiliary piston and then sealed using an end plug. If necessary the counterbore could be eccentric without affecting the actuator performance. The manufacture of such a unit was considered to be extremely simple, as both pistons could utilise standard parts (e.g. dowels, needle rollers, ground stock bar etc) and the housing bore produced by conventional machining techniques. Furthermore such a unit could be used either on its own, with a return spring, or
FIG 1.05  SINGLE ACTING ACTUATOR WITH HYDRAULIC CUSHIONING

Note: \[ \frac{A_1(x - \bar{x})d^2}{A_1 + A_2} \geq x_\ast d^2 \geq \bar{x}d^2 \]
opposed modules made to form a double-acting mechanism. To illustrate a double acting arrangement a design based on two opposed modules was prepared. (Figure 1.06). In this design the actuators are shown contained inside a split casing which houses a square slide. The slide being used for attaching ancillary mechanisms such as the schematic lever illustrated and lubricated by the oil leakage from the actuators.

After consideration it was decided that single acting units were capable of meeting all the operating requirements and that initial prototype mechanisms based on these should be built and tested. It was appreciated that there would be occasions when the double acting design illustrated in Figure 1.04 would prove advantageous. It was proposed therefore that this design would be evaluated at a later date. Functionally the two designs are identical and where specific reference is given to one concept, this will, in general, be applicable to the other.

1.4 Actuator Operating Modes

Each single acting unit (or 'half' of the double-acting cylinder) has two control ports which, when connected to a suitable valve, can have three states:

1) Connected to the main hydraulic supply.
2) Connected to the system exhaust.
3) Blanked off thereby permitting no flow.

Of the nine possible combinations, inspection will show that only 3 of these will cause movement when the actuator is fully retracted as in Figure 1.05(a), the remaining combinations either prevent movement or are necessary when actuator motion is reversed. The combinations which cause movement and the general operating characteristics which were envisaged are outlined below.

1.4.1 Basic Reciprocation

Pressure applied simultaneously to ports 1 and 2: the actuator will commence to move out with an unrestricted movement. The pressure of the entrained fluid between the auxiliary and the main piston will
FIG 1.06 A DOUBLE ACTING MECHANISM USING SINGLE ACTING MODULES
be nominally the system pressure. As movement commences the auxiliary and main pistons will separate owing to the difference in the swept areas.

As the main piston clears port 2, the oil flow will switch from the entrained fluid to the mains supply from port 2. The auxiliary piston will then cease to have any significant effect on the subsequent movement of the actuator. With the increase in velocity of the main piston and the consequential increase in the viscous losses between the valve and the actuator, there will be a reduction in the pressure behind the main piston ($p_2$ in Figure 1.05(b)). This will cause a pressure difference across the auxiliary piston and result in its continued movement outward.

As the movement of the main piston and its oil column is arrested, the auxiliary piston will act as a hydraulic spring. This will help to reduce the effects of the oil momentum in the supply conduit, i.e. it will help to reduce transient pressure surges and fluid hammer. The final position of the auxiliary piston will be indeterminate as it will then be subjected to nominally equal pressures on either end.

1.4.2 Forward Uncontrolled Stepped Motion

Pressure applied to Port 1; Port 2 connected to exhaust: the actuator will commence to move out as 1.4.1 above. As the main piston clears port 2 the entrained fluid will be connected to exhaust and the pressure behind the main piston will be relieved. The subsequent motion of the actuator will then be a function of the resistance to motion and the kinetic energy that the system has attained. This switching combination would therefore result in an uncontrolled step movement similar to that obtained by the Garside actuator. Port 2 used in this manner being the equivalent of the hydraulic stop port.

On a mechanism with a low resistance to motion, a desirable characteristic for low response times, considerable further movement of the actuator may result. If the motion is opposed, for example by a spring, then deceleration may be rapid and actuator movement could cease when port 2 is only partially open. Provided that the system resistance is not too great the auxiliary piston
will continue to move forward until arrested by the housing recess. Its influence on the subsequent actuator motion would therefore depend on the particular system and the physical dimensions of the auxiliary piston. For the design illustrated the spigot on the leading edge of the auxiliary piston can enter the main piston bore. This will enable the auxiliary piston to act as a support and prevent any backward movement of the main piston, or depending on the opposing forces, cause further movement of the actuator.

1.4.3 Forward Controlled Stepped Motion

Pressure applied to Port 1, Port 2 blanked off: The actuator will commence to move out as above, movement will continue until the entrained fluid is displaced and the auxiliary piston is prevented from further movement by the housing recess. The movement of the main actuator will therefore depend on the physical dimensions of the auxiliary piston and could be any value up to its maximum displacement.

When operating in this mode the auxiliary piston is analogous to a miniature hydraulic accumulator capable of supplying a fixed volume of oil to the main piston. In operation, the auxiliary piston moves proportionately less than the main piston. This feature could be used on high speed mechanisms where the resultant decrease in viscous losses may result in higher response times. For example, if the diameter of the auxiliary piston were twice the diameter of the main piston, then the viscous losses between the actuator and its control valve would be one fifth of those, assuming the fluid were fed direct to the rear of the main piston.

When this mode is used to obtain a step movement, the entrained fluid will act as the retarding medium and prevent overshoot. Deceleration control would be obtained by suitably dimensioning the spigot on the leading edge of the auxiliary piston.

1.4.4 Requirements for Effective Cushioning

In general, to obtain the correct cushioning characteristic on the return movement of the actuator, it is necessary that the auxiliary piston is fully back when the main piston reaches port 2.
This can be assured by maintaining pressure at port 2 whilst exhausting port 1 during the outward movement. During the return movement the exhaust pressure will ensure that its position will be maintained. If the auxiliary piston is not fully retracted, when the main piston closes port 2, then the cushioning effect could be partially or even completely negated.

1.4.5 Controlled Reverse Stepped Motion

A further envisaged use of the auxiliary piston was for it to provide a stop during the return movement of the actuator by maintaining pressure at port 1. The auxiliary piston would then act as a hydraulically sprung stop and could be used either to limit possible mechanism overshoot or to provide an additional discrete stop. If this facility was used it was noted that the movement would need to be uncushioned, i.e. it must be possible for fluid to flow via port 2 to the entrained volume.

Some of the displacement profiles which were envisaged, using the basic module, can be seen in Figures 1.07 and 1.08. Also shown are the necessary pressure/exhaust switching sequences to achieve these motions. In the case of the single acting mechanism, sequence 1.4.3 above was omitted, as it was considered that overshoot would be minimal when the motion is opposed by a spring.

1.5 Theoretical Analysis

1.5.1 Introduction to Analysis

The research was biased towards the practical application of miniature hydraulic actuators to industrial equipment. The theoretical analysis also had a practical bias and was aimed at enabling the non-specialist mechanism designer to use and understand the techniques and equipment which evolved. The objectives of the analysis were:

1) To provide an accurate prediction of the dynamic equations of motion of hydraulically actuated mechanisms.

2) To enable the mechanism designer to understand the relative importance of the performance limiting variables.
displacement

(a) forward step
(b) double step
(c) backward step
(d) basic reciprocation

revolutions of valve

port 1
port 2

pressure
exhaust
control point

FIG 1.07 OPERATING MODES — SINGLE ACTING SPRING RETURN MECH
FIG 108 OPERATING MODES — DOUBLE ACTING MECHANISM
FIG 1.09 SCHEMATIC HYDRAULIC CIRCUIT - THEORETICAL ANALYSIS

pump a reservoir

rotary valve

D = pipe bore
L = pipe length

single acting module

spring S (N/m)

displacement
To achieve these objectives, the analysis made use of several fundamental fluid equations and mathematical techniques. It was believed that this approach, in combination with the physical simplicity of the basic elements would result in a high degree of accuracy in the predictions. The validity of the analysis was to be checked by a subsequent prototype build and test programme.

1.5.2 Schematic Circuit - System Losses

A typical circuit which was analysed can be seen in Figure 1.09. The oil supply from a constant delivery pump is fed via a rotary valve to the actuating mechanisms. Single-acting modules are shown operating on a slide mounted in oil lubricated bearings, and which also carries a return spring. As the valve is rotated, the actuators will be alternately connected between the system pressure and exhaust. Under the action of the applied pressure the mechanism will reciprocate, the resultant dynamic profiles of the mechanism will then be dependent upon the boundary conditions imposed and the losses generated within the system. For the circuit shown the losses were divided into three areas:

1) Pipe losses.
2) Mechanism losses.
3) Valve losses.

In addition there would be losses in the main hydraulic power pack, these were not considered.

1. Pipe losses: The imposed boundary conditions (Section 1.5.3) enabled the friction losses to be calculated using the Hagan-Poiseuille formula:

\[ \Delta p = \frac{32 \mu L V}{D^2} = k \dot{x} = \frac{32 \mu L d^2}{D^4} \dot{x} \]  

In addition there would be a loss in pressure at each change in pipe section and direction. The empirical formula that, a
sharp right angle bend causes a pressure loss equal to the dynamic pressure, was used throughout i.e.

$$\Delta p = \frac{\rho V^2}{2} = \frac{\rho}{2} \left( \frac{q}{D} \right)^2 x^2$$

(2)

2. **Mechanism Losses**: There will be a resistance to motion in each part of the mechanisms. For analytical purposes these were restricted to the drag force imparted as adjacent lubricated surfaces moved with a relative velocity. For the mechanism illustrated, two such examples are:

a) The pistons moving in their housing.
b) The slide moving in its bearings.

A constant velocity profile was assumed between members which enabled the viscous stress and subsequent drag force to be calculated by applying the basic definition of viscosity. This force was then expressed in terms of the pressure required at the actuator to overcome it. It can be shown that in the case of (a) above that:

$$\Delta p = \frac{4}{h_d} \frac{1}{u} \mu \dot{x} = k_a \ddot{x}$$

(3)

and similarly for (b) above that

$$\Delta p = \frac{4}{d^2 h} \frac{G}{d} \mu \dot{V} = k_m \ddot{V}$$

(4)

3. **Valve Losses**: As part of the test programme on rotary valves the pressure drop across the valve versus flow through the valve was obtained. As recommended in BS 4062 the characteristic was expressed in terms of a dimensionless coefficient $k_*$ such that:

$$\Delta p = \frac{\rho}{2} k_* V^2$$

(5)
1.5.3 Generalised Equations of Motion

The dynamic equations of motion were compiled with the following boundary conditions:

a) The actuator is controlled by a rotary valve which generates square pressure pulses of a constant amplitude throughout the displacement of the actuator.

b) Compressibility and leakage flows throughout the system are ignored, both effects being considered insignificant due to the small entrained volumes and leakage flows involved.

c) No work is done by the actuator so that all available force is used to accelerate the system.

d) The fluid flow is laminar and hence all viscous losses are proportional to velocity.

e) Momentum losses due to changes in pipe direction and also the rotary valve losses are small, can be linearised over the operating range of the mechanism and made proportional to velocity.

f) Gravitational forces are insignificant compared with operating forces.

g) All interconnecting pipework is in the form of smooth bore circular conduits, pipe runs between equipment to be continuous and of uniform cross section.

h) Changes in dynamic viscosity due to pressure are ignored.

i) Mechanical friction losses are insignificant compared with operating forces.

j) All fluid and parts of the mechanism are at rest at the commencement of pulse initiation.

By applying Newton's second law of motion and the continuity equation to the schematic circuit, Figure 1.09, the following equations of motion can be written:
If $p_1$ is the nominal system pressure $P$ and the exhaust pressure $p_8$ is atmospheric then the above equations can be combined to give an overall equation of motion thus:

$$\dot{P} = \frac{\ddot{X}}{A} \left[ m_1 \left( \frac{d}{D_1} \right)^4 + m_2 \left( \frac{d}{D_2} \right)^4 + \cdots \right] + \dot{\ddot{X}} \left[ 2 k_a + k_m + k_1 \left( \frac{d}{D_1} \right)^4 + \cdots \right]$$

$$+ \rho \frac{\ddot{X}^2}{2} \left[ 2 k_* + N_1 \left( \frac{d}{D_1} \right)^4 + N_2 \left( \frac{d}{D_2} \right)^4 + \cdots \right] + \frac{S}{A} \ddot{X}$$

i.e. the general form of the equation is:

$$\frac{M}{A} \dddot{X} + k \ddot{X} + c \dot{X}^2 + \frac{S}{A} \ddot{X} - P = 0$$

(1) $p_1 - p_2 = \left( \frac{m_1}{A} \left( \frac{d}{D_1} \right)^4 \right) \ddot{X} + k_1 \left( \frac{d}{D_1} \right)^2 \dot{X} + \rho \frac{N_1}{2} \left( \frac{d}{D_1} \right)^4 \dot{X}^2$

(2) $p_2 - p_3 = \rho \frac{k_*}{2} \dot{X}^2$

(3) $p_3 - p_4 = \frac{m_2}{A} \left( \frac{d}{D_2} \right)^4 \ddot{X} + k_2 \left( \frac{d}{D_2} \right)^2 \dot{X} + \rho \frac{N_2}{2} \left( \frac{d}{D_2} \right)^4 \dot{X}^2$

(4) $p_4 - p_5 = \frac{m}{A} \ddot{X} + (2 k_a + k_m) \dot{X} + \frac{S}{A} \ddot{X}$

(5) $p_5 - p_6 = \frac{m_3}{A} \left( \frac{d}{D_3} \right)^4 \ddot{X} + k_3 \left( \frac{d}{D_3} \right)^2 \dot{X} + \rho \frac{N_3}{2} \left( \frac{d}{D_3} \right)^4 \dot{X}^2$

(6) $p_6 - p_7 = \rho \frac{k_*}{2} \dot{X}^2$

(7) $p_7 - p_8 = \frac{m_4}{A} \left( \frac{d}{D_4} \right)^4 \ddot{X} + k_4 \left( \frac{d}{D_4} \right)^2 \dot{X} + \rho \frac{N_4}{2} \left( \frac{d}{D_4} \right)^4 \dot{X}^2$. 

(9) $\frac{M}{A} \dddot{X} + k \ddot{X} + c \dot{X}^2 + \frac{S}{A} \ddot{X} - P = 0$
This non-linear differential equation is not amenable to analytic solution, but it can be solved using an iterative technique on a digital computer. However, in view of the practical bias of the theoretical analysis, it was decided to combine and linearise the coefficients $k$ and $c$ into a single coefficient $K$ proportional to velocity only. This enabled the differential equation to be solved analytically and the results presented in the form of equations of motion i.e. displacement/velocity/acceleration versus time relationships. In view of the relatively low losses proportional to the square of the velocity, the resulting error was considered to be minimal. The differential equation was therefore reduced to the linear form

$$\frac{M}{A} \ddot{x} + K \dot{x} + \frac{S}{A} x - P = 0$$  \hspace{1cm} (10)$$

where:

1) 'M' is defined as the APPARENT MECHANISM MASS. This was adopted to simplify the analysis and the presentation of the theoretical equations of motion. This mass term included:
   a) The actual mechanism mass to be reciprocated ($m$).
   b) The apparent mass of fluid in the circuit to be analysed.

To relate the acceleration of the fluid in the supply conduits to the actuator acceleration it was necessary to introduce the ratio $(D/d)^4$. The actual value of the oil mass must be multiplied by the ratio $(D/d)^4$ to obtain its apparent mass, e.g.

$$M = m + m_1 (\frac{D}{d_1})^4 + m_2 (\frac{D}{d_2})^4 + \ldots$$  \hspace{1cm} (11)$$

It was noted that in view of the fourth power relationship between $d$ and $D$ the apparent oil mass could have a significant effect on the dynamic performance of the mechanisms, particularly in a system where the rotary valve is not located very close to the actuator i.e. where the entrained fluid mass $x (\frac{d}{D})^4$ becomes significant compared with the combined mass of the actuator and mechanism.
2) 'A' is the effective area of the particular actuator operating the mechanism. In the case of a double-acting cylinder of the type illustrated in Figure 1.04 this would be a different value between forward and reverse movements.

3) 'P' is the full (nominal gauge) system pressure.

4) 'K' is defined as the LINEARISED VISCOUS DAMPING COEFFICIENT of the mechanism to be analysed. This term included the contribution of both the viscous and momentum coefficients k and c where these terms were the respective sums of the individual viscous and momentum losses thus:

\[ k = k_1 + k_2 + k_3 + \ldots \]  \hspace{1cm} (12)

\[ c = c_1 + c_2 + c_3 + \ldots \]  \hspace{1cm} (13)

As in the case of the apparent oil mass terms account needed to be taken of the ratio between the oil velocity in the conduits and the mechanism velocity. (Equations 1 to 5 in Section 1.5.2).

1.5.4 Determination of Linearised Viscous Damping Coefficient K

From elementary dynamic considerations, the maximum mechanism velocity will be obtained when all the mechanism and circuit force resistances equal the input force. For a circuit where a double-acting mechanism of the types illustrated in Figures 1.04 and 1.06 has an unopposed external force this can be expressed as follows:

\[ k \ddot{x} + c \dot{x}^2 = P \]  \hspace{1cm} (1)

The maximum mechanism velocity \( \dot{x} \) of the mechanism can be found from the above expression and used to obtain a coefficient \( K \) proportional to velocity only, i.e.
For the case of a single-acting spring return mechanism the determination of a linearised viscous damping coefficient is complicated by the existence of the spring. The maximum mechanism velocity is indeterminate analytically as the equation equating forces reduces to:

\[ k \dot{x} + c \dot{x}^2 = P = K \dot{x} \quad (2) \]

To obtain analytical solutions to equation (10) in Section 1.5.3, it was decided to take an estimated value for \( K \) based on the value of the viscous coefficient \( k \). The analysis of various circuits using double-acting mechanisms indicated generally that the momentum coefficient \( c \) would be small when compared with the viscous coefficient, typically in the order of 20%. For single acting spring return mechanisms the linearised viscous coefficient \( K \) was therefore taken as being \( 1.2k \).

1.5.5 Validity of Theoretical Approach

The general solution of the linearised differential equation of motion (equation (10) Section 1.5.3) was solved by the so named "complementary function and particular integral technique". The equations of motion that were obtained were then used to predict the theoretical performance of different mechanism configurations. These "linearised" solutions were themselves checked by using iterative solutions for the same mechanism configurations and boundary conditions.

The iterative solutions confirm that the "linearised" equations of motion resulted in insignificant errors in the theoretical dynamic profiles of double-acting mechanisms. For single-acting spring return mechanisms with disproportionately high momentum losses it was felt that some justification existed for an iterative solution to be applied. An example of the iterative technique used can be found in Appendix 5.3. In general however the linearised equations yielded accurate correlation with the iterative solutions.
In the forthcoming analyses, the differential equation of motion relating to the particular mechanism type is presented initially and is followed by the derived equations of motion. The validity of the equations can be checked by use of successive differentiation and by consideration of elementary dynamics. For ease of comprehension and application the analysis was separated into three areas:

i) The initial movement of a double-acting mechanism up to the hydraulic cushion.

ii) Single acting spring return mechanisms.

iii) Hydraulic cushioning.

1.6 Double Acting Mechanism - Initial Travel to Hydraulic Cushion

1.6.1 Equations of Motion

Using the basic configuration shown in Figure 1.06, but omitting the compression spring, the general differential equation of motion is:

\[
\frac{M}{A} \ddot{x} + K \dot{x} - P = 0
\]  

(1)

the derived equations of motion were as follows:

\[
\text{MECHANISM} \quad \text{DISPLACEMENT (m)} \quad x = \frac{PM}{K^2 A} (e^{-KAt/M} - 1) + \frac{Pt}{K}
\]  

(2)

\[
\text{MECHANISM} \quad \text{VELOCITY (m/sec)} \quad \dot{x} = \frac{P}{K} (1 - e^{-KAt/M})
\]  

(3)

\[
\text{MECHANISM} \quad \text{ACCELERATION (m/sec}^2) \quad \ddot{x} = \frac{PAe^{-KAt/M}}{M}
\]  

(4)
Examples of typical displacement and velocity profiles are shown in Figures 1.10 and 1.11. It was noted that theoretically, the velocity curve increases to a maximum as $t \to \infty$ and that when maximum velocity is reached the above equations reduce to:

- \text{mechanism displacement} \quad x = \frac{P}{K} (t - \frac{M}{KA}) \quad (5)

- \text{maximum mechanism velocity} \quad \frac{P}{K} \quad (6)

For practical purposes a mechanism was said to \text{flow saturate}, and that the mechanism had reached maximum velocity, when $KA/t > 3$ as the term $e^{-KA/t}$ becomes decreasingly significant beyond this point. The \text{acceleration time} was therefore defined as being equal to $3M/KA$. Using this definition the following relationships were derived:

- \text{Acceleration displacement of mechanism} \quad = \frac{2MP}{KA^2} \quad (7)

- \text{Average mechanism velocity over the acceleration displacement} \quad = \frac{2P}{3K} \quad (8)

- \text{The Acceleration time of the mechanism} \quad = \frac{3M}{KA} \quad (9)
displacement $x' = \frac{p}{K}$

$2m \frac{p}{K^2A}$

$\frac{m}{KA}$

$\frac{x}{K}[t - \frac{m}{KA}]$

FIG 1.10 DISPLACEMENT PROFILE - INITIAL TRAVEL OF A DOUBLE ACTING MECHANISM

velocity $\dot{x}$

$\frac{p}{K}$

$2\frac{p}{3K}$

$\dot{x} = \frac{p}{K}[1 - e^{-\frac{KAt}{m}}]$

FIG 1.11 VELOCITY PROFILE - INITIAL TRAVEL OF A DOUBLE ACTING MECHANISM
1.6.2 Manipulation of the Variables

The variables P, K, A and M are all capable of being manipulated to achieve different mechanism response times. The variables are not independent and affect the equations of motion and hence mechanism response times differently. It was considered necessary that each variable be examined independently and its affect on mechanism performance put into perspective. It was felt that this would enable a better appreciation of the relative importance of the variables to be obtained.

The mean about which changes in a variable were gauged was the previously defined "acceleration displacement" of a mechanism. The selection of this "mean" has an advantage when a mechanism designer is confronted with the task of the initial selection of system parameters to suit a particular operating specification (Section 2.8). The advantage being that the reduced equations 7, 8 and 9 in Section 1.6.1 enable basic mechanism data to be translated into the necessary hydraulic data.

The technique adopted was to assume that a particular mechanism had a displacement equal to the acceleration displacement and consequently had a theoretical response time equal to $3M_{f}KA$. Each variable was then scaled independently by means of a coefficient 'Z' within the range 0.1 to 10 times its initial value whilst keeping other physical parameters constant. When a change in one of the variables did not result in an increase in the acceleration displacement it was possible to calculate the new theoretical response time 't*' by use of the expressions given below. These relationships were derived from the analysis of the resultant displacement-time curves of the mechanism (Figures 1.12 to 1.15).

1. Increase in Area ... $Z$ greater than 1 ... Figure 1.12:

$$t_{*} = \frac{m}{KA} \left(2Z + \frac{1}{Z^{2}}\right)$$  \hspace{1cm} (1)

2. Decrease in Mass ... $Z$ less than 1 ... Figure 1.13:

$$t_{*} = \frac{m}{KA} (2 + Z)$$  \hspace{1cm} (2)
FIG 1.12 EFFECT OF INCREASE IN ACTUATOR AREA ON RESPONSE TIME OF A D.A. MECHANISM

FIG 1.13 EFFECT OF CHANGE IN MASS ON THE RESPONSE TIME OF A D.A. MECHANISM
FIG 1.14 EFFECT OF INCREASE IN VISCOSITY COEFFICIENT ON RESPONSE TIME OF A D.A. MECHANISM

FIG 1.15 EFFECT OF DECREASE IN PRESSURE ON RESPONSE TIME OF A D.A. MECHANISM
FIG 1.16 MANIPULATION OF VARIABLES ABOUT THE ACCn. DISPLACEMENT OF A D.A. MECHANISM
3. Increase in Viscous Coefficient ... Z greater than 1 ... Figure 1.14:

\[ t_\star + \frac{m}{KA} (2Z + \frac{1}{Z}) \quad (3) \]

4. Decrease in Pressure ... Z less than 1 ... Figure 1.15:

\[ t_\star = (1 + \frac{2}{Z}) \quad (4) \]

For a change in variable which resulted in an increase in the acceleration displacement use was made of equation 2 in Section 1.6.1. In both cases however the following assumptions were made.

a) That the change in mass by the increase/decrease in the other variables is insignificant compared with the overall mechanism mass.

b) That the linearised viscous damping coefficient K is directly proportional to A. This assumes that the viscous drag loss of the pistons in their housings and the momentum losses are small compared with the pipe losses.

A series of results was obtained for each of the variables, these are given in Table 1 and shown plotted in Figure 1.16. Consideration of the analysis and results indicated the following:

1) That the effect on mechanism response time of increase in the actuator area and viscous coefficient above their mean values are comparable. Proportionate increases in the mechanism mass result in a smaller increase in the mechanism response time than either the actuator area or viscous coefficient.

2) That the effect of reductions in the mechanism mass and the viscous coefficient below this mean value are comparable. Both variables have the same limiting value of reducing the mechanism response time to 2M/KA, i.e. by a maximum of one third.
For the case of a mechanism having no viscous losses, the response time would reduce to that obtained by applying elementary inviscid equations of motion to the mechanism, i.e. to assume that the mechanism had a constant acceleration of PA/M for the duration of mechanism travel. Similarly, for the case of an (imaginary) mechanism having no mass to accelerate, the mechanism would move throughout its travel at a constant velocity proportional to the viscous damping coefficient.

3) That over a limited range, the effect of changes in actuator area on the response time are minimal. Consideration of the area plot in Figure 1.16, showed that, over the scaling range $Z = 0.6$ to $Z = 1.0$ changes in the theoretical response time would be approximately 4%. It was deduced from this that it was theoretically feasible to design a double-acting cylinder of the type shown in Figure 1.04 to have nominally the same response times in both directions. However, this would only be possible provided that the areas of both the piston and the rod ends of the actuator were designed within the scaling range $Z = 0.65$ to $Z = 1.0$. This conclusion led to a qualification to one of the objections raised regarding double-acting cylinders (Section 1.2), i.e. that: a double-acting cylinder will result in asymmetric dynamic profiles between inward and outward movements due to area differences. This statement should be qualified to acknowledge the possibility of symmetry provided that the actuator is applied and designed correctly.

4) That the optimum actuator area to achieve the fastest mechanism response time about the mean for constant mass and pressure was with a scaling factor of nominally 0.8. A corollary of this was that: for a mechanism designed with a displacement equal to its acceleration displacement, calculated areas should be reduced by 20%. This would have the twofold advantage of increased response times for a lower power input.
5) That fluid temperature/viscosity changes will have less effect if a mechanism operates within its acceleration displacement.

6) That the system pressure is the only independent variable. Although changes in the system pressure do not alter the acceleration time of a mechanism, these changes have the most significance on the overall mechanism response time.

1.7 Single Acting Spring Return Mechanism

1.7.1 Equations of Motion

For the basic configuration shown in Figure 1.09 but with only one module the differential equations of motion for free reciprocation are as follows:

a) Outward movement:

\[ M \ddot{x} + K A \dot{x} + S x = P A - S B \]  \hspace{1cm} (1)

b) Return movement:

\[ M \ddot{x} + K A \dot{x} + S x = S(B + \overline{x}) \]  \hspace{1cm} (2)

The form of these equations are similar and were recognised as being analogous to a spring/mass/damper system response to step excitation. In the case of the outward movement the step input force \( F = P A - S B \) and for the return movement \( F = S(B + \overline{x}) \). For dynamic similarity between the two movements it will be apparent that:

\[ P A = S(2B + \overline{x}) \]  \hspace{1cm} (3)

There may be occasions when either a faster or slower response is required in one direction, this would be achieved by varying the initial compression \( B \).

The mathematical analysis is the same in both cases and the general solution of equations (1) and (2) has three forms depending on whether the roots of the auxiliary equation are real, equal or complex. The auxiliary equation being:
\[ M a^2 + K A a + S = 0 \]

which has roots \( \frac{-K A}{2 M} \pm \sqrt{\frac{(K A)^2}{2 M^2} - \frac{S}{M}} \) \( (4) \)

**Case 1:** Real roots \( a \) and \( b \) \( \ldots \quad (K A / 2 M)^2 > S / M \) (overdamped)

\[ a = \frac{-K A}{2 M} + \sqrt{\frac{(K A)^2}{2 M^2} - \frac{S}{M}} \]

\[ b = \frac{-K A}{2 M} - \sqrt{\frac{(K A)^2}{2 M^2} - \frac{S}{M}} \]

\[ F = \frac{(P A - S B)}{S} \text{ or } F = B + \bar{x} \]

**MECHANISM**

**DISPLACEMENT** \( (m) \)

\[ x = F \left[ \frac{a e^{b t} - b e^{a t}}{b - a} + 1 \right] \] \( (5) \)

**MECHANISM**

**VELOCITY** \( (m/sec) \)

\[ \dot{x} = F \left[ \frac{a b}{b - a} (e^{b t} - e^{a t}) \right] \] \( (6) \)

**MECHANISM**

**ACCELERATION** \( (m/sec^2) \)

\[ \ddot{x} = F \left[ \frac{a b^2 e^{b t} - a^2 b e^{a t}}{b - a} \right] \] \( (7) \)

**Case 2:** Equal roots \( a \) \( \ldots \quad (K A / 2 M)^2 = S / M \) (critically damped)

\[ a = \frac{K A}{2 M} = \sqrt{\frac{S}{M}} \]

\[ F = \frac{P A - S B}{S} \text{ or } F = B + \bar{x} \]
MECHANISM
DISPLACEMENT (m) \[ x = F [1 - e^{-at}(1 + at)] \] (8)

MECHANISM
VELOCITY (m/sec) \[ \dot{x} = F a^2 e^{-at} \] (9)

MECHANISM
ACCELERATION (m/sec²) \[ \ddot{x} = F a e^{-at}(1 - at) \] (10)

Case 3: Complex roots \( v + i\omega \), \((KA/2M)^2 < S/M\)

\[ v = \frac{KA}{2M} \]

\[ \omega = \frac{S}{M} - \left(\frac{KA}{2M}\right)^2 \]

\[ F = \frac{PA - SB}{S} \text{ or } F = B + \bar{x} \]

MECHANISM
DISPLACEMENT (m) \[ x = F [1 - e^{-vt}\left(\frac{v}{\omega} \sin\sqrt{\omega} t + \cos\sqrt{\omega} t\right)] \] (11)

MECHANISM
VELOCITY (m/sec) \[ \dot{x} = F \left(\frac{\omega + v^2}{\sqrt{\omega}}\right) e^{-vt} \sin\sqrt{\omega} t \] (12)

MECHANISM
ACCELERATION (m/sec²) \[ \ddot{x} = F \left(\frac{\omega + v^2}{\sqrt{\omega}}\right) [e^{-vt}(\sqrt{\omega} \cos\sqrt{\omega} t - v \sin\sqrt{\omega} t)] \] (13)
1.7.2 Use of Equations

The same freedom of manipulation of the variables that was possible for double-acting mechanisms cannot be obtained in a single-acting mechanism. The fact that the mechanism motion is opposed by a spring must be constantly borne in mind. The presence of a varying force not only imposes a limit on the maximum displacement obtainable but the spring stiffness introduces an additional variable.

The analysis of spring return mechanisms showed that the principal factor controlling their successful application related to the boundaries imposed by the spring design. In general it was assumed that the return motion would be achieved by a helically coiled compression spring. Theoretically it is possible to design a spring suitable for almost any mechanism. In practice, however, design and manufacturing limitations of the spring and its physical size would govern the applicability of spring return mechanism. Within the limits imposed by the spring design, it was considered that satisfactory dynamic performance was subsequently related to the amount of viscous damping. Curves for a lightly damped, overdamped and a critically damped response are shown in Figure 1.17.

Regardless of the degree of system damping it will, in general, be necessary for a mechanism to have a residual force at either end of mechanism travel to maintain positional accuracy during dwell periods. This would be achieved in one direction by the initial spring compression and by a residual hydraulic force in the other. The residual hydraulic force implies that the outward mechanism would be terminated by a physical stop before the theoretical maximum displacement is reached. As the outward motion will be uncushioned the impact speed of the mechanism against such a stop is a significant design factor.

For lightly damped systems, as the mechanism velocity will increase throughout the mechanism travel, the impact speed will be greater than the average mechanism velocity. As the viscous damping is increased, up to and beyond the critical value, the impact speed against a physical stop will be a function of the amount of damping and the residual hydraulic force allocated for mechanism stability.
FIG 1,17 VELOCITY & DISPLACEMENT PROFILES OF A S.A. SPRING RETURN MECHANISM
By selecting the correct system parameters, this impact velocity can be designed to be lower than either the "peak" or "average" velocities. After design investigations it was considered that the dynamic profiles of critically damped mechanisms offered the best compromise between reduced impact velocities and a fast response time, with a realistic residual hydraulic force. Consideration of the critically damped dynamic profiles (Figure 1.17) showed that with a residual hydraulic force of 0.2F, the impact velocity would be only 40% of the peak value. In terms of the system parameters the mechanism response time would be nominally 6M/KA. Thus it was decided therefore than wherever practical spring return mechanisms should be designed to have critical viscous damping and a residual hydraulic force of 0.2F.

1.8 Hydraulic Cushioning

1.8.1 General Considerations

Refer to Figure 1.18. Consider that the main piston is travelling in the positive direction shown under the section of an external source (e.g. by its complementary piston not shown), and that the auxiliary piston is butting against the end stop. The only restriction to flow of the entrained oil out of the supply port 2 will be the normal flow losses due to exhaust pipe and orifice restrictions. As further movement occurs, the leading edge of the main piston will start to close the port and cause an increase in the exhaust pressure. When the supply port is fully closed, a certain volume of oil will then be trapped between the pistons. Further movement of the actuator will then be possible only by:

a) compressing the entrained fluid,
b) fluid leaking past the main piston and out of port 2,
c) fluid leaking past the auxiliary piston and out of port 1.

For the purposes of analysing the cushioning characteristics of the mechanism both the compressibility of the entrained fluid and the transient effect of the closing of port 2 were ignored. The analysis
FIG 1.18 HYDRAULIC CUSHION

FIG 1.19 VELOCITY PROFILE - HYDRAULIC CUSHIONING OF A D.A. MECHANISM
therefore assumed that the deceleration of the mechanism was
dependent upon the flow of oil past the auxiliary and main pis­
tons. Apart from dimensional differences the analytical treat­
ment of these was the same.

1.8.2 Flow Past Auxiliary Piston and Main Actuator

The radial clearance between the pistons and their respective
bores is small in comparison with their diameters, this enabled
the pistons to be unwrapped for analytical purposes and one dimen­sional parallel flat plate theory to be applied. The governing
equation regarding flow can be expressed:

\[ V = \frac{\Delta p h^2}{12 \mu L} \]  

(1)

where:

- \( V \) = mean velocity of flow between plates
- \( \Delta p \) = pressure difference across section
- \( L \) = length of section
- \( h \) = clearance between plates.

The velocity of the actuator can be found by applying the above
formula together with the continuity equation. It can be shown
for the auxiliary piston that:

\[ \dot{x} = \frac{p_i d_4 h^3}{3 \mu d^2 l} \]  

(2)

where:

- \( p_i \) is the induced pressure difference.

In the case of the main piston, the length \( L \) in equation 1 was
taken as being equal to a constant, and that:

\[ \dot{x} = \frac{p_i h^3}{3d \mu L} \]  

(3)
To obtain the resultant mechanism velocity equations 2 and 3 require to be added together. Although the induced pressure $p_i$ will be the same in both equations the same may not be true for the radial clearance $h$. For analytical purposes the relationship between pressure and velocity was therefore put in the form

$$(c_1 + c_2) p_i = \dot{x} \quad (4)$$

where $c_1$ is obtained from equation (2) and $c_2$ from equation (3).

1.8.3 Equations of Motion

Assume that at the port shut off position the actuator is travelling with a velocity of $V_i$ defined as the "CUSHION IMPACT VELOCITY". This velocity would be calculated from the equations of motion derived in Sections 1.6 and 1.7. Applying Newton's 2nd Law of Motion and using the convention for positive values adopted in Figure 1.18, the differential equation of motion will be as follows:

$$\frac{M}{A} \ddot{x} + C \dot{x} - P = 0 \quad (5)$$

The solution of which is identical to that obtained in Section 1.6 but with different boundary conditions. The equations of motion can be shown to be:

$${\text{MECHANISM DISPLACEMENT (M)}} \quad x = \frac{M}{CA} (V_i - \frac{P}{C}) \left[1 - e^{-\frac{CAt}{M}}\right] + \frac{P}{C} t \quad (6)$$

$${\text{MECHANISM VELOCITY (M/sec)}} \quad \dot{x} = V_i e^{-\frac{CAt}{M}} + \frac{P}{C} \left(1 - e^{-\frac{CAt}{M}}\right) \quad (7)$$

$${\text{MECHANISM ACCELERATION (M/sec)}} \quad \ddot{x} = e^{-\frac{CAt}{M}} \frac{PA}{M} \left(\frac{\dot{x}}{M} - \frac{CA}{M} \cdot V_i\right) \quad (8)$$
Where \( x \) here is the displacement after the port shut off is reached and \( C \) is defined as the "overall cushioning coefficient" and includes:

1) The flow past the main piston \( c_1 \).
2) The flow past the auxiliary piston \( c_2 \).
3) The viscous mechanism losses \( k \).

Such that:

\[
C = \left( \frac{1}{c_1 + c_2} \right) + k \tag{9}
\]

1.8.4 Use of Cushioning Equations

The equations are all exponential in form and a typical velocity profile can be seen in Figure 1.19. Theoretically the constant velocity term \( P/C \) would be obtained as \( t \to \infty \) however for evaluation purposes as per 1.6.1 a mechanism was defined as being fully cushioned when the exponential term

\[
(1 - e^{-\frac{CAt}{M}}) = 0.95 \text{ i.e. when } e^{-\frac{CAt}{M}} = 0.05.
\]

This simplification enabled the cushioning time to be expressed as follows:

\[
t = \frac{3M}{CA} = A \text{ constant} \tag{10}
\]

The following relationships regarding the cushioning characteristics were derived by substituting equation (10) into equations (6) and (7) and putting \( (1 - e^{-\frac{CAt}{M}}) = 1 \).

\[a) \text{ minimum length of cushion } = \frac{M}{CA} \left( V_i + \frac{2P}{C} \right) \tag{11} \]

\[b) \text{ final impact velocity of mechanism } = \frac{P}{C} \tag{12} \]
These relationships were used to obtain specific information, the factors which were considered to be of importance were:

1) The maximum induced pressure $p_i$.
2) The final impact velocity.
3) The cushioning coefficient $C$.

1. Maximum induced pressure $p_i$: It was apparent that as the cushioning characteristics are reactive, the greater the shut off velocity and mass, the greater would be the induced pressure. The maximum value will occur at the port shut off point i.e. when $t = 0$ in equation (8). To limit the magnitude of the theoretically induced pressure it was considered that this should not be allowed to exceed 500 bar. This figure being within the fatigue and yield stress levels of common engineering materials.

Now as the maximum deceleration at $t = 0$ ... $= \frac{A}{M} (P - CV_i)$

and the maximum acceleration of a mechanism $= \frac{PA}{M}$

The maximum induced pressure was expressed as a ratio of the supply pressure $P$. Similarly the maximum deceleration was expressed in terms of the maximum acceleration and hence it was deduced that:

$$p_i = CV_i - P$$  \hspace{1cm} (13)

Equation (13) was then used to find the maximum dashpot coefficient $C_{\text{max}}$ necessary to contain the induced pressure, within the selected limit, i.e.:

$$C_{\text{max}} = \frac{(p_i + P)}{V_i}$$  \hspace{1cm} (14)

This equation was in turn substituted into equation (11) to establish the associated minimum cushion length:
2. Final impact velocity: The impact velocity that a particular mechanism can tolerate will be a function of the kinetic energy at impact and the ability of the mechanism to absorb this without any detrimental effects. As this relates specifically to a particular design it was difficult to establish any firm guidelines. However, numerical examples based on the maximum induced pressure criterion indicated that final impact velocities could be made extremely low. This would result in very severe cushioning, but it was doubted whether such severity would be required in practice. A more realistic approach was considered to be:

a) Either to reduce the impact velocity by a certain percentage, for example 80%, or

b) To limit the final impact velocity to figure within the permissible stress limit of the mechanism, for example, as per Appendix 1.1.

The practical implications that would result could then be assessed and any necessary adjustments then made.

3. The cushioning coefficient \( C \): The analysis ignored:

i) the effect of component eccentricity

ii) the pressure/viscosity and temperature/viscosity effect of the entrained fluid

iii) expansion/contraction effects due to the induced pressure.

These factors will combine to cast doubt on the accuracy of the theoretical value of \( C \). In addition, the cubic relationship of the radial clearance imposes a heavy responsibility on the accuracy of machining of the parts involved. To account for intangibles it was anticipated that an actuator would need to be designed slightly overcushioned and suitably modified following practical tests.
To facilitate this approach and after consideration of the operating modes of the actuator, it was felt that the auxiliary piston should be viewed as a fixed restriction. The radial clearance of the auxiliary piston would then be dictated by its required running clearance only. The desired cushioning coefficient would be obtained by means of a cushioning land on the main piston (see Figure 1.18). The design of the basic module is such that the main piston can easily be removed and modified. To limit cushioning adjustments only to the main piston also helps to reduce cushioning errors caused by component eccentricity as the piston running clearance will be small in comparison with the land clearance. For example, the calculated value of the cushioning land for the prototype test rig was 50 microns and the piston running clearance less than 10 microns.
SECTION TWO

PRACTICAL MECHANISM PERFORMANCE

2.1 Introductory Summary

The manufacture and testing of the actuator concepts outlined in Section 1 took place over a prolonged period. This section deals with selected tests which illustrate the salient characteristics which must be considered when using the actuators in a mechanism design. The majority of information relates to mechanisms reciprocated by single-acting units as these were preferred to their double-acting counterpart. No evidence was found, however, of any significant difference between the dynamic behaviour of double-acting or opposed single-acting actuators. Specific information relating to dynamic and operating tests of opposed single-acting actuators can be applied directly to their double-acting counterpart.

The mechanism illustrated in Figure 1.06 was designed, manufactured and tested to establish whether it functioned as initially conceived. Tests confirmed that the auxiliary piston enabled a cushioned movement to be obtained in one direction whilst allowing an unrestricted reverse motion. It was also confirmed that the auxiliary piston could be sequenced as an independent actuator and hence enable "compound" or stepped motions to be obtained. The actuating concepts were then embodied into a more complex test rig designed to simulate the mechanism motions of a chain-stitch sewing machine. This rig, which had hydraulically actuated mechanisms for the needle, needle looper and looper cross-feed mechanisms, was sequenced by a rotary spool valve (Figure 2.01). The actuators and mechanisms which comprise this rig were used as the basic models for assessment. Whilst the dynamic results indicated in this section are specific to this rig, information obtained from the mattress sewing machine (Section 5) and other sundry mechanisms has been incorporated into the general observations and recommendations.
The objective of conducting dynamic tests was to establish the fidelity of the operating and dynamic characteristics against theoretical expectations. It was considered that paramount amongst these was the accuracy of theoretical mechanism response times. Thus considerable effort was spent in understanding and eliminating as far as possible, the cause(s) of any disparity between measured and theoretical response times. No accuracy level was pre-envisaged, it was considered that, provided the error was low and due allowance could be made in the design to nullify its effect, then this would be acceptable. In general, theoretical mechanism performances were calculated using the analytical approach outlined in Section 1.5 and used design data only. Thus it was hoped that results would be representative of a true practical mechanism. The results of these tests yielded considerable information of how particular mechanisms performed against design expectations. Thus this was the inverse of what would normally be required in practice, where a mechanism design must be tailored to achieve a particular operating specification. Therefore, based on accumulated test experience and theoretical considerations, a simple design procedure is presented to enable actuators to be sized correctly to meet a particular operating specification.

The remainder of the Section relates to the use of the actuators to obtain stepped motion and effective hydraulic cushioning. Whilst the information presented is mainly qualitative, its inclusion will enable a greater appreciation to be obtained of the flexibility and limitations of the actuating concept.

2.2 Instrumentation

All pressure measurements in the dynamic testing of the actuators were taken using pressure transducers. The type used were S.E. Laboratories variable reluctance transducers, these give good linearity with little hysteresis. The measurement of mechanism displacement was taken via a linear induction transducer with a soft iron core mounted directly onto either the mechanism slide or the piston rod.
The transducers were used in conjunction with a carrier system and a S.E. Laboratories Ultra-Violet galvanometer recorder. This unit directs a high intensity U.V. light beam onto sensitised paper via a rotatable coil and mirror assembly. The principal advantage of using this instrument was the simultaneous display of several independent readings and in the provision of an immediate and (relatively) permanent record of the results. The traces could be analysed graphically and an overall picture obtained of what happened during the test. The inertial effect of the mirror/coil assembly was evidenced by an apparent overshoot as the mechanism decelerated. At various times use was also made of a recording oscilloscope but the definition of the traces was poor and results obtained using this instrument have not been included.

At the commencement of each test the pressure transducer was calibrated using a Budenburg dead weight pressure tester and the displacement transducer calibrated using a dead stop and slip gauges.

2.3 Basic Reciprocation Using Single Acting Units

2.3.1 Test Procedure - Test 1

The prototype mechanism shown in Figure 1.06 was connected to run in the simple reciprocating mode, i.e. with ports 1 and 2 coupled together and likewise ports 3 and 4. The actuators were designed to give an uncushioned movement in the forward direction and a severely cushioned return movement. The mechanism was connected to a rotary valve so that each actuator received a pressure pulse of $155^\circ$ duration, but with a phase angle of $130^\circ$ between pulse initiation. This overlap of pulses was deliberate and was designed to demonstrate the combined effect of oil compressibility and hose flexing, see Figure 2.02.

A pressure transducer was fitted at the inlet to the rotary valve and the core of a displacement transducer connected to the square sliding member of the mechanism. With the mechanism mounted horizontally both transducers were calibrated and connected to the
pump capacity 23 l/min  
Max. pressure 83 bar  
Valve speed 570 r.p.m  
pipe data:  
<table>
<thead>
<tr>
<th>No.</th>
<th>bore (mm)</th>
<th>length (m)</th>
<th>90° bends</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>12</td>
<td>2.0</td>
<td>4</td>
</tr>
<tr>
<td>2</td>
<td>6.25</td>
<td>0.85</td>
<td>2</td>
</tr>
<tr>
<td>3</td>
<td>6.25</td>
<td>1.40</td>
<td>3</td>
</tr>
<tr>
<td>4</td>
<td>3.0</td>
<td>0.4</td>
<td>11</td>
</tr>
<tr>
<td>5</td>
<td>3.0</td>
<td>0.5</td>
<td>13</td>
</tr>
<tr>
<td>6</td>
<td>4.72</td>
<td>1.5</td>
<td>2</td>
</tr>
</tbody>
</table>

Hydraulic Accumulator  
(Test 2 only)  
Pressure Transducer  
Mechanism displacement  
forward  
Retract  
Port No. 1, 2  
3, 4  

25° Overlap of pressure pulses  
--- pressure  
..... exhaust.
recorder to give a simultaneous read-out of results. Traces of the supply pressure and mechanism displacement were then taken at various nominal pressures between 14 and 69 bar. The traces were then analysed and a table of results compiled (Table 2). The response time of the mechanism against nominal supply pressure is shown plotted in Figure 2.07, also shown plotted is the theoretically predicted response curve (Appendix 2.1).

2.3.2 Supply Pressure Fluctuation

Consideration of the traces, copies of which are shown in Figures 2.03 - 2.06, showed that considerable fluctuation of the supply pressure occurred.

The start of each pulse from the rotary valve was however clearly visible, this being evidenced by a serious drop in the supply pressure which lasted for several milliseconds. The pressure continued to fluctuate throughout subsequent mechanism travel and also during dwell periods. These circuit pressure fluctuations were attributed to the principle of operation of the pressure relief valve.

The response of a relief valve is a function of its detail design but notwithstanding, it is still subject to the same mechanical laws as the remainder of the circuit. A typical relief valve consists of a spring loaded mechanism either directly or indirectly varying a control orifice. When circuit pressure is too high, caused by an over supply from the hydraulic pump, the orifice opens to allow a sufficient quantity of fluid to be diverted back to tank whilst maintaining the circuit pressure at the nominally set value. The particular pressure at which this occurs being determined by the amount of pre-loading on the controlling spring. For this test circuit, during 90% of the time the whole of the fluid supply from the fixed displacement pump was being diverted over the relief valve, for the remaining time the maximum fluid demand did not exceed 15% of the available pump capacity. The low inertia of the mechanisms being tested was such that when supply pressure was connected to them by the switching effect of the rotary valve, the hydraulic power supply was feeding alternately very low and very high load resistance conditions.
transient overpressure

severe cushioning

with accumulator 69 bar

exponential decay

without accumulator 69 bar

FIG 2.03 DISPLACEMENT & PRESSURE TRACES
FIG 2.04 DISPLACEMENT & PRESSURE TRACES

transient overpressure

with accumulator 55bar

without accumulator 55bar
transient overpressure

with accumulator 41.4 bar

without accumulator 41.4 bar

1 m/s paper feed

FIG 2.05 DISPLACEMENT & PRESSURE TRACES
FIG 2.06  DISPLACEMENT & PRESSURE TRACES

with accumulator
27.6 bar

without accumulator
27.6 bar
The pressure traces clearly indicated the resulting response of the relief valve, which is characteristic of an underdamped second order system to an impulse excitation. The pressure to the valve and hence to the actuator was not therefore the steady nominal value assumed in the theoretical analysis. The pressure oscillated about the nominal pressure setting following the start of each pulse and although some decay of the oscillations was apparent, the relief valve did not settle down before the next pulse was initiated. Table 2 also gives the maximum and minimum pressures which occurred when the mechanism was cycling.

2.3.3 Mechanism Response Times

Consideration of Figure 2.07 will show there was a significant difference in the response times between the forward and return movements. This was caused by the different pressure/exhaust sequences applied to the two movements by the rotary valve. The theoretical analysis had assumed that the effects of oil compressibility were negligible but previous experience had shown that this was subject to a certain qualification. To facilitate rig changes the actuators were only piped in rigid tubing up to a manifold block mounted on the base plate. Between the manifold block and the rotary valve, use was made of flexible hoses. When connected to the exhaust areas of the rotary valve these hoses relaxed and some additional oil, over and above the nominal swept volume of the actuator, was discharged to exhaust. When the hose was subsequently reconnected to the main supply pressure, the initial flow from the valve therefore had the dual role of replacing this oil and of supplying the actuator. This resulted in a slight delay or lag between the pulse initiation and the commencement of movement. For this particular test the delay was approximately 2 milliseconds. This phenomena was not discernible on other tests when the actuator and valve were connected using rigid conduits throughout. To overcome this effect on the return movement the pulses were therefore overlapped by 25° rotation of the valve bobbin. During this overlap the actuator was trapped between two opposing pressure
FIG 2.07 MECHANISM RESPONSE VERSUS SUPPLY PRESSURE—TEST ONE
pulses and the actuator could only return when the forward pulse was connected to exhaust. This "overlapping technique" resulted in a faster return movement because it tended to nullify the effects of both the supply pressure drop and hose flexing.

Although the general pressure versus time profile for both movements were similar to the theoretically predicted curve, a significant error resulted. In view of the recorded pressure fluctuations this was not unexpected. The error of the return movement within the pressure range 40-69 bar was however less than 10%.

The results of the return movement indicated that a constant error in the supply pressure of approximately 8.6 bar still existed. The practical curve fitted a displacement equation of:

\[ x = \frac{M}{K^2A} (P - 8.6) \times 10^5 \left[ e^{-KA/m} - 1 \right] + \frac{(P - 8.6)}{K} \times 10^5 t \]

where \( P \) is the pressure in bar (70 > \( P \) > 8.6).

The response curve of the forward movement exhibited a similar pressure error but with an additional constant delay of 4 milliseconds. The discrepancies were attributed therefore to the loss in supply pressure due to the relief valve response when the pulse was initiated and also the effect of hose flexing.

2.3.4 Circuit Modifications - Test 2

The experience of severe pressure drops and fluctuations in the supply pressure had been encountered when commissioning the mattress sewing machine (Section 5). The technique which had been adopted to minimise this was the inclusion in the circuit of a hydraulic accumulator. This had been placed between the main power pack and the rotary valve. This modification resulted in a considerable improvement in the constancy of the supply pressure to the mechanism and a consequential increase in accuracy between theoretical predictions and practical results.
This approach was also tried on the prototype rig and a second series of tests carried out with an accumulator in the circuit. Copies of the traces obtained are also shown in Figures 2.03 - 3.06 for comparison with the earlier test and their analysis is given in Table 3. The mechanism response times versus the nominal supply pressure are shown plotted in Figure 2.08.

Consideration of the pressure traces showed that there was an improvement in the pressure fluctuations. The accumulator succeeded in damping the fluctuations sufficiently so that at pulse initiation the pressure had settled to its nominal value. However there was still a substantial pressure drop when each pulse was initiated and some pressure fluctuations during mechanism movement. The effect on the mechanism response was to produce parallel curves to those obtained in the previous test. In the case of the forward movement the curve was displaced by 5.2 bar and 1.8 milliseconds and the return movement by 3.45 bar and 1.5 milliseconds. The mechanism response times for both tests are shown plotted, together with the theoretical curve, on Figure 2.09.

Consideration of Figure 2.09 will show that the combined effect of the hydraulic accumulator and the overlapping of the pressure pulse resulted in a marginally faster response time than that predicted theoretically at pressures above 27 bar. Consideration of the corresponding pressure traces showed that above this value, the pressure to the actuator exceeded the nominal circuit pressure setting. This was caused by the coincidence of a high point in the pressure fluctuation cycle and the commencement of movement of the actuator. At a supply pressure of 27.6 bar the pressure remained sensibly constant throughout the mechanism travel. This resulted in very close agreement between theoretically predicted and measured performance.

Similarly, if the average measured supply pressure to the rotary valve is plotted against measured response time, a more accurate correlation is also achieved.
FIG 2.08 MECHANISM RESPONSE VERSUS SUPPLY PRESSURE TEST TWO
FIG 2.09  COMPARATIVE RESULTS TESTS 1&2 – ERROR ANALYSIS
2.4 Stepped Motion Using Single Acting Units

The prototype double-acting mechanism was capable of compound or stepped movements in either direction. The pressure/exhaust switching sequence to achieve these motions together with the general operating characteristics are outlined in Section 1.4. As considerable test data on the response time of mechanisms operating at full stroke had been obtained, it was considered unnecessary to repeat this on the reduced step heights. The objective of these tests was limited to establishing the accuracy and repeatability of the step height at different supply pressures. The tests which were carried out and the resulting characteristics of each operating mode were as follows.

2.4.1 Uncontrolled Stepped Motion - Test 1

The mechanism was connected to the rotary valve so that it received the pressure/exhaust switching sequence shown in Figure 2.10. With the valve rotating at constant speed, displacement traces were obtained at circuit supply pressures varied between 7 and 45 bar.

The traces indicated that considerable overshoot of the step height occurred which increased as the supply pressure was increased. Figure 2.10 shows a plot of the average overshoot as a function of supply pressure and also an illustration of the typical displacement profile that resulted. The displacement profiles also indicated that the mechanism moved out to a maximum value but subsequently retracted slightly during the dwell period. At a particular operating pressure and temperature, the repeatability of the maximum step height was found to be within 0.5 mm.

The test was then repeated with a compression spring opposing the motion. The particular spring chosen enabled full motion to be obtained with a supply pressure of approximately 25 bar. The test was run at a nominal pressure of 40 bar. The spring completely eliminated any mechanism overshoot without any discernible unevenness of the displacement profile.
(a) Displacement profiles & port sequencing

(b) Mechanism overshoot against supply pressure

FIG 2.10 COMPOUND RECIPROcation TEST 1
2.4.2 Controlled Step Height - Test 2

The mechanism was connected to the rotary valve to give the pressure/exhaust sequence shown in Figure 2.11. Also shown in Figure 2.11 are illustrations of the theoretical and practical displacement profiles that were obtained. Displacement traces were obtained with circuit supply pressures varied between 15 and 70 bar and with the rotary valve running at constant speed.

The traces indicated that apart from a small transient overshoot of the nominal step height, the fidelity of the displacement profile was accurate and repeatable over the whole pressure range of the test. The transient overshoot had a constant maximum value of 1.3 mm and was found to be independent of the supply pressure. This overshoot was traced to the physical dimensions of the auxiliary piston and its relationship with control port 3. The auxiliary piston had been designed with a spigot on its leading edge so that it could enter the main bore (see Figure 1.05). The leading edge of the spigot overlapped the control port by the amount of transient overshoot, namely 1.3 mm.

In operation, pressure applied to ports 1 and 2 caused the main piston to move out until its complementary main piston hit the opposing auxiliary piston. This auxiliary piston then acted as a sprung stop and only allowed the complementary piston to move an amount sufficient to close port 3. After port 3 had been fully closed, the mechanism was decelerated rapidly with no discernible further movement of the mechanism. The mechanism was subsequently returned to the nominal step height due to the area difference between the main piston and the opposing auxiliary piston.

It is believed that this overshoot would have been completely eliminated by reducing the length of spigot on the auxiliary pistons by an appropriate amount. As this would have entailed re-machining the actuator housings and manufacturing new auxiliary pistons this was not confirmed.
FIG 2.11 DISPLACEMENT PROFILES & PORT SEQUENCING COMPOUND RECIPROCATION

FIG 2.12 COMPOUND RECIPROCATION TEST 3
2.4.3 Controlled Step Height - Test 3

The mechanism was connected to the rotary valve to give the pressure/exhaust sequence shown in Figure 2.12. Also shown are illustrations of the theoretical displacement and a typical practical profile obtained. A series of traces were obtained for circuit supply pressures varied between 15 and 45 bar.

The traces confirmed that as expected the actuator moved out initially to a distance proportional to the swept volume of the auxiliary piston. However, when the movement of the auxiliary piston was arrested by its housing, the actuator continued to move out at a reduced velocity. This further movement being caused by the fluid leaking past the auxiliary piston and hence to the main piston. This leakage had been overlooked when considering the operating modes of the mechanism. The tests indicated however, that unless deliberate measures were taken to nullify it, this continued outward movement of the actuator was inevitable.

One technique which was successfully employed to nullify this leakage was by controlling the duration of the pressure pulse to the auxiliary piston. By using the theoretical response time for the step height at a pressure of 30 bar (10 milliseconds) the speed of the rotary valve was increased to 1500 r.p.m. At this speed the 90° pulse length controlling the auxiliary piston was effective for 10 milliseconds. The mechanism movement then attenuated slightly at this valve speed but a consistent step height was obtained.

The supply pressure was subsequently increased to 40 bar and the full step height was consistently and accurately obtained. This technique, although successful, was considered limited in its application to only single speed mechanisms. Although the mechanisms themselves have a fixed response time there would be occasions such as the mattress sewing machine (Section 5) where it would not be practical to have a single speed drive to the rotary valve.

The only other solution which was considered possible, but which was not pursued, entailed making the leading edge of the auxiliary piston into a valve seat. The edges of the auxiliary piston and the inside of the housing had been chamfered to increase the contact
area between the parts and help prevent impact burrs from forming. This lowered the contact pressure between the auxiliary piston and the housing below that theoretically necessary to prevent leakage. To have reduced this contact area would have meant re-machining the parts and lapping the surfaces to ensure a good seat. As the lapping/machining operations implied accurate concentricity between parts this would have negated one of the advantages of the actuator design, its ability to function correctly with considerable eccentricity.

Although the characteristics of the operating mode were not entirely as envisaged, the resultant displacement profile was considered to be of some use. The general characteristic of a rapid advance followed by a slow feed is commonly used in machine tools. Although no immediate use for the feature was envisaged, it remained an unexpected yet welcome possible operating feature of the basic modules.

2.5 Hydraulic Cushioning Tests

The prototype mechanism was tested initially with a severely cushioned motion in one direction and an uncushioned motion in the other. This was achieved by suitably machining the ends of the main piston. See Figure 1.05. To obtain the severely cushioned motion, the end of the piston was relieved so that the supply port was fully closed within 1 mm of the end of travel. For the uncushioned movement this relief was extended so that the supply port remained open throughout the cushioning stroke.

The mechanism was then tested up to a maximum pressure of 69 bar to establish its ability to withstand the extremes of cushioning. The displacement trace illustrated in Figure 2.03 is an example of the two extremes. The constant reduced velocity characteristic during the cushioned movement is apparent but the transition between the high and low velocities is masked by the overshoot of the recording instrument.
The mechanism demonstrated its ability to withstand both the severely induced impact stresses and the high induced fluid pressures without any discernible damage to the individual parts. There was however a distinct difference between the impact noise produced at either end. The cushioned motion producing a dull thud and the uncushioned movement an unhealthy sounding and hard metallic ring. Whilst appreciating that subjective judgements should where possible be avoided, an aural assessment was used extensively on subsequent cushioning tests. A mechanism was judged to be insufficiently cushioned if the final impact noise was discernibly metallic.

Using the aural criterion, a series of tests were carried out on the uncushioned movement to establish the maximum final impact velocity considered to be acceptable. These tests made use of the theoretical velocity profiles of the mechanism given in Appendix 2.1. By manipulating the system pressure and displacement, the mechanism was made to impact at a known velocity and the resultant noise assessed. These tests indicated that provided the impact speed did not exceed (nominally) 1 metre per second, the impact noise was not discernibly metallic. It was appreciated however that this figure related only to the mechanism tested and that other threshold figures might be obtained with other mechanism configurations and system parameters.

The prototype mechanism was then analysed to optimise the cushioning length when reciprocated at a supply pressure of 0.69 bar. The effect of different cushion lengths on the induced fluid pressure, together with the theoretical clearance of the cushioning land, the final impact velocity and the resultant cushioning time, was calculated. The analysis, together with a summary of results is given in Appendix 2.1. The analysis showed that to comply with the induced pressure criterion indicated in Section 1.8 the minimum cushion length was 2 mm. Further, to limit the final impact velocity within the 1 m/sec limit established by the aural assessment, the cushioning length should not exceed 5 mm. It was therefore decided that a cushioning length of 3 mm would satisfy both criteria whilst still allowing a reasonable tolerance on the accuracy of machining the cushioning land.
A series of cushioning tests were then carried out with actuators having the calculated clearances. These tests indicated however that at pressures above 50 bar the impact noise became discernibly metallic, therefore the mechanism was stripped down to inspect the component parts. This inspection indicated that the running clearances between the pistons and their respective housings were greater than had been allowed both in the detail design and subsequently the theoretical analysis. Owing to the physical size of the housing bores it was not possible to obtain the clearances accurately by direct measurement. It was decided that to obtain these clearances indirectly by ascertaining the flow restriction past each leakage path would be advantageous. It would enable the "datum" cushioning coefficients of individual parts to be obtained directly and the radial clearance associated with these coefficients to be calculated.

The mechanism was cushioned to maximum severity in both directions over the last 3 mm of movement and tested again. To restrict the flow past the main piston during the outward movement, the supply ports to the auxiliary pistons were permanently connected to exhaust. Displacement traces were obtained at various supply pressures between 15 and 69 bar, a copy of a relevant trace can be seen in Figure 2.13.

The traces clearly indicated the constant velocity characteristic during the cushioning movement and also the restricted flow to each actuator during (its) outward movement. The traces also enabled a disparity between the two auxiliary pistons to be observed, this being evidenced by the different constant velocities during cushioning at either end. Similarly a disparity between the restricted flows past the main pistons was also apparent. The traces also indicated two unexpected features:

1. That no evidence existed of any transient characteristics between the cushion impact velocity and the constant reduced cushioned velocity. Transient effects were evident however on the outward motion of each actuator.

2. That the actuators moved out faster than their corresponding cushioned velocities.
restricted oil feed to main actuator when auxiliary piston is made inoperative

cushion velocity

8,78 mm/sec

cushion velocity

25 mm/sec

FIG 2.13 CUSHIONING TRACE OF PROTOTYPE D.A.MECHANISM — SUPPLY PORTS 2 & 3 ONLY CONNECTED TO PRESSURE (27,6 bar)
It has been anticipated that as during the cushioning movement, the entrained oil had two possible leakage paths, i.e. past both the auxiliary and main pistons, the cushioning velocities would be greater than the outward velocities. However during outward movement of the actuator, the oil supply had only the leakage path past the main piston to cause mechanism travel. Furthermore transient effects were also anticipated due to:

a) The main piston progressively closing the supply port.

b) The main piston progressively increasing the cushioning restriction as it entered deeper into the cushion.

It was concluded that during the cushioning movement little or no leakage occurred past the main piston. It appeared that the induced cushioning pressure forced the main piston to fully close its supply port so that only the oil flow past the auxiliary piston was significant. When the supply pressure was subsequently connected to the main piston, it was lifted clear of the supply port and oil flow past the piston then occurred, transient effects were then apparent. It was therefore assumed for the analysis of the displacement traces that during the cushioning movements, the leakage flow was due entirely to the clearance between the auxiliary pistons and their housings. Conversely, during outward movement of the actuator the leakage flow was entirely past the main piston and further, that the piston was forced fully eccentric.

From the analysis of the test data, the clearance of one of the auxiliary pistons was calculated to be 38 microns and the other 28 microns. Similarly the main piston clearances were estimated to be 20 microns and 15 microns. All theoretical cushioning calculations had however been based on a radial clearance of 10 microns. These relatively large radial clearances were attributed to manufacturing errors. A subsequent investigation revealed, for example, that a 7/32" (5.556 mm) reamer had been used to finish the auxiliary piston bores instead of the 5.50 mm called for on the detail design drawings.

These manufacturing errors were therefore taken into account, the main pistons ground with a reduced radial clearance and the mechanism retested. The modifications resulted in satisfactory cushioning
without discernible impact noise at the maximum supply pressure of 69 bar. The theoretical analysis of the mechanism had also indicated that the cushioning would increase the overall response time by 1 millisecond and modify the displacement profile as shown in Figure 1 of Appendix 2.1. Both the increase in response time and the modified displacement profile were confirmed by the tests. However, owing to the lack of suitable instrumentation no direct information relating to the actual final impact velocity was obtained.

Up to this juncture all cushioning tests had been carried out with the mechanism horizontal and having the pressure/exhaust sequence shown in Figure 1.06. This sequence being designed to ensure that the auxiliary piston was driven fully back prior to the commencement of the return movement and hence that hydraulic cushioning would be assured. A further series of tests were therefore carried out with the mechanism mounted vertically and supply ports 1/2 and 3/4 coupled together such that each auxiliary and main piston combination received the same pressure pulse. During such a control sequence the position of the auxiliary piston at the commencement of the actuator return movement is indeterminate (see Section 1.4). It was hoped however that during the actuator return movement the auxiliary piston would be driven fully back by the generated exhaust pressure before the commencement of the cushioning stroke.

These tests confirmed that the previous cushioning characteristics were maintained, there was no discernible difference either between the previously obtained displacement traces or in the impact noise that resulted. For simple reciprocating mechanisms it was envisaged that in future the control ports 1/3 and 3/4 could be combined and thereby enable fewer pipe runs to be made between the control valve and the mechanism.
2.5.1 Geometrical Requirements for Consistent Cushioning

The control technique of coupling together the supply ports to the auxiliary and main pistons for basic reciprocating motion was found to be subject to certain restrictions. These restrictions were highlighted when cushioning problems were experienced on the prototype candlewick tufting mechanism (Section 5.8). Problems were encountered on the looper mechanism which had been designed to have a cushion length of 4 mm at either end of a total actuator displacement of 10 mm. The problem was one of erratic and inconsistent cushioning in one direction only of mechanism travel.

The problem resulted through the Author not taking cognisance of his own guidelines whereby the auxiliary piston must be fully retracted when the main piston closes its supply port (1.4.4). With the above control sequence the detail design of the auxiliary piston must ensure that it is driven back by the 'exhaust' oil flow generated by the actuator. This in turn implies that the geometry of the respective parts ensures that this is feasible. A subsequent check on the looper mechanism details showed that cushioning could not be guaranteed, due to incorrect detail design. Thus it was felt necessary that the essential relationships between the main and auxiliary pistons should be stated explicitly to avoid a repetition of the same mistake. Although the equations presented are specific to single-acting actuators, the general principles will apply to their double-acting counterpart.

When operating in this control mode, the oil flow generated by the reverse motion of the actuator has two possible paths:

a) directly out of the main piston supply port 2;

b) into the chamber occupied by the auxiliary piston displacing oil out of port 1.

If gravitational and inertial forces are low, the problem is analogous to the oil flow being exhausted out of 2 ports. In this analogy, the auxiliary piston could be replaced by an equivalent column of oil. The essential problem is to guarantee that the
necessary quantity of oil is displaced out of port 1 before the main piston closes port 2, such that the auxiliary piston is carried or driven back by this oil flow. With the assumptions that the auxiliary piston is a free sliding fit and that the orifice discharge coefficients are sensibly the same, the oil displaced by the actuator Q will divide according to the area ratios of the ports, i.e.

\[
\text{Volume displaced from port 1} = \frac{Q A_1}{A_1 + A_2}
\]

\[
\text{" " " " port 2} = \frac{Q A_2}{A_1 + A_2}
\]

where 'Q' can be expressed simply in terms of the swept volume of the main piston:

\[
Q = \frac{\pi}{4} d^2 (x - \bar{x})
\]

Similarly, the maximum amount of oil required to be displaced by the auxiliary piston, assuming that it is fully forward at the commencement of the return travel, can also be expressed by:

\[
Q_1 = \frac{\pi}{4} x_0 d^2
\]

Therefore to ensure that sufficient oil is displaced out of port 1, the parts should be dimensioned so that

\[
Q_1 > \frac{\pi}{4} d^2 x_0
\]

i.e.

\[
\frac{A_1}{A_1 + A_2} d^2 (x - \bar{x}) > x_0 d^2
\]

The above relationship can be used to check whether full cushioning is feasible, however the parts must also be suitably dimensioned
to ensure that an unrestricted forward motion is also obtained. It can be shown that this will be ensured provided also that the following relationship is also satisfied:

\[ x_d^2 > \frac{x}{d^2} \]  \hspace{1cm} (2)

### 2.6 Double-Acting Cylinder

The concept of an auxiliary piston to achieve hydraulic cushioning was applied to the rod end of a double-acting cylinder (Figure 1.04). It was envisaged that such an actuator design would function in a similar manner to two opposed single-acting modules. In view of their inherent simplicity and reliability, single acting modules were preferred for the majority of mechanisms investigated in this research. However, there will be applications where it will be advantageous to utilise a double-acting actuator and consequently that the design concept illustrated should be verified. The actuator shown in Figure 2.14 was therefore designed and manufactured. This unit incorporated a small spool valve designed to interrupt the oil flow to ports 1 and 2.

When the actuator was assembled it became apparent that the concentricity problems envisaged at its conception were realised in practice. At the extreme outward travel of the piston, the actuator was found to seize. This seizing was attributed to a build up of eccentricity errors between:

i) The piston and piston rod
ii) The two housing bores
iii) The O.D. and I.D. of the auxiliary sleeve
iv) The O.D. and I.D. of the end bearing.

The actuator had been designed so that the radial clearances between the various parts were normal running clearances, i.e. approximately 10 microns. The cushioning at the rod end being achieved by means of a cushioning land on the leading edge of the piston and by treating the auxiliary sleeve as a fixed restriction. It was concluded however
actuator data
maximum stroke ............ 30 mm
main piston dia .......... 6 mm
auxiliary piston dia ...... 8 mm
piston rod dia ............ 4 mm

FIG 2.14 DOUBLE ACTING CYLINDER WITH BUILT IN SEQUENCING VALVE
that this cushioning approach was not practical.

So that the actuator could be checked for its other functional characteristics, it was assembled and lapped in a drilling machine in an attempt to overcome the seizing problem. This lapping operation was partially successful and enabled the actuator to be reciprocated provided that the supply pressure was greater than 15 bar. The actuator was therefore connected into the test circuit used for checking the single-acting mechanisms and sequenced by a small valve module (Section 4) to run in a basic reciprocation mode.

Displacement traces obtained showed no evidence of any cushioning at the piston rod end thus it was concluded that this had been destroyed by the lapping operation. The traces also showed that an appreciable delay occurred between the initiation of the pressure pulse to retract the piston and the commencement of actuator movement. This delay was eventually traced to a form of mechanical/hydraulic locking which occurred between the piston and the actuator housing. The retract movement had been obtained by applying system pressure simultaneously to ports 3 and 4. It appeared that pressure applied to port 3 induced frictional forces between the piston and the housing which could not be overcome by the force exerted via the auxiliary sleeve. The piston end of the actuator however, functioned well, and its characteristics were identical to those obtained on single acting modules.

In view of the lack of cushioning at the rod end and also the locking feature it was decided to redesign the actuator. To reduce the effects of eccentricity errors, the cushioning concept at the rod end was modified. For the modified design, the flow of the entrained oil past the piston and out of port 3 was treated as a fixed restriction and the cushioning land eliminated. The cushioning characteristics were to be obtained by adjusting the O.D. of the auxiliary sleeve instead of the piston. This modification increased the radial clearance between the sleeve and the housing considerably (from 10 to 50 microns for the actuator shown) and hence the design would be more tolerant of eccentricity errors. To reduce the locking problem, the piston diameter was recessed opposite supply port 3 when the actuator is fully extended. This modification was
designed to balance the hydraulic pressure applied to the piston at the commencement of the retract movement. In addition the chamfers at the ends of the auxiliary sleeve and the end bearing were replaced by machined steps. The built-in sequencing valve was also omitted as the design had proved to be ineffective.

A second actuator incorporating the above design modifications was therefore manufactured and assembled. Following verification that actuator movement was free in both directions, the unit was incorporated into the test rig and displacement traces obtained. These traces showed that the actuator response was good in both directions and that no locking was apparent. As in the case of mechanisms actuated by single-acting modules, the actuator would reciprocate freely at extremely low operating pressures. The cushioning at the rod end was however only very light. Using the aural criterion outlined in Section 2.5 the actuator was considered to be inadequately cushioned at pressures above 10 bar.

The actuator was then subjected to a series of qualitative tests to check its various modes of operation. These tests confirmed that the unit functioned in a similar manner to two opposed single-acting modules and that compound movements could be obtained in both directions. It was concluded therefore that apart from being undercushioned, the design appeared to function well. To verify the cushioning at the rod end, additional auxiliary sleeves having different outside diameters were manufactured. These auxiliary sleeves were designed to increase progressively the severity of hydraulic cushioning. Subsequent tests demonstrated the ability of the actuator to achieve the same cushioning characteristics in both directions of motion. It was concluded therefore that the design modifications were successful and that the concept could be used as the basis for a range of double-acting actuators (Figure 2.16).

2.7 Requirements for Predictable System Performance

The objective of carrying out dynamic tests was to establish the fidelity of the operating and dynamic characteristics against theoretical expectations. It was considered that paramount amongst these was the accuracy of theoretical mechanism response times.
FIG 2.15 A DESIGN OF A CUSHIONED ACTUATOR FOR BASIC RECIPROCATING MOTION
Considerable test data was obtained from operating a variety of different mechanisms under different system parameters. In view of space limitations only one or two selected tests have been included in this thesis which illustrate the salient points to be considered and allowed for in hydraulic mechanism design. The tests have not been included because they flatter the practical results, on the contrary they were amongst the least accurate results obtained. The reason for their selection was that they represent very light mechanisms with fast response times and which operated within their "acceleration displacement" (Section 1.6). Thus the effect of supply pressure fluctuations on mechanism performance was very apparent. The other operating extreme of a relatively heavy mechanism operating under flow saturated conditions was encountered on the needle mechanism of the mattress sewing machine. Again pressure fluctuations were apparent but their effect on subsequent mechanism performance was minimal. Rather than present optimistic performance forecasts that are subject to particular operating conditions, the author would prefer a more conservative estimate that could probably be bettered. With this proviso, the detailed testing of various actuator/mechanism configurations led to the following general observations.

1) That the analysis of valve/actuator circuits of the type outlined in this thesis can yield accurate mechanism predictions provided that a constant input supply pressure is maintained.

2) That the proprietary gear pumps and pressure controlling/relief valves used in this research, were unable to maintain a constant supply pressure on circuits where high cyclic flow demands were encountered. Use of these units alone resulted in significant supply pressure fluctuations, particularly on circuits where the mechanisms had a low impedance.

3) Pressure fluctuations are felt by all mechanisms in a particular circuit. These fluctuations can cause significant errors in the predicted response times. The amplitude of the pressure fluctuations increased with increased pressure.

4) Pressure fluctuations can be damped but not eliminated entirely,
by the inclusion of a "bladder type" hydraulic accumulator in the main pressure inlet to the rotary valve.

5) Compressibility effects cannot be ignored on circuits which incorporate flexible hoses between the actuators and the rotary valve. An average delay of two milliseconds was encountered between the commencement of actuator movement and the initiation of the pressure pulse on circuits which utilised flexible hoses. Compressibility effects were not in general apparent on circuits which employed rigid conduits.

6) On circuits where dwell periods occur between mechanism movements the combined effects of compressibility and pressure fluctuations can be offset by overlapping the pressure pulses between movements.

7) The accuracy of mechanism response time predictions were in general unsatisfactory at supply pressures below 20 bar. The most accurate results were obtained at supply pressures between 30 and 50 bar.

8) For mechanisms that are operated at supply pressure between 30 and 50 bar and designed take cognisance of the foregoing, can anticipate mechanism performance predictions to be within 15% of measured performance.

9) For mechanisms which require an accuracy level better than 85%, it will be necessary to resort to "tuning" techniques after a mechanism has been incorporated into the control circuit. Tuning techniques might include operating a mechanism at a slightly higher pressure to reduce response times or conversely, deliberately increasing flow losses to increase response times.

2.8 The Selection of System Parameters

2.8.1 Introduction

The results of dynamic tests carried out on actuators and prototype mechanisms indicated that it is possible to obtain predictable response times for known system parameters. The equations of motion given in Section 1 can therefore be used to decide the various para-
meters. However, to simplify the presentation of the equations, the factors which influence mechanism performance were combined to show only:

i) System pressure ...................... P
ii) Actuator area ...................... A
iii) Linearised viscous damping coefficient, K
iv) Apparent mechanism mass .............. M
v) Spring rate .......................... S

Of these variables only the system pressure is independent, the other variables being interrelated via the area ratio between the actuator and its supply conduit. This interrelationship imposes a discipline when designing mechanisms using the actuation and control technique outlined in this thesis. Thus it was considered necessary to develop a general procedure to assist the design selection of the system parameters. The procedure developed subsequent to the testing of actuators, together with the underlying reasons, is given below.

2.8.2 Mechanism Specification

At its conception each mechanism will have certain constraints imposed on it. In general terms these can be expressed as "the necessity to move a particular mass a certain distance in a specified time", i.e. the fundamental identification of:

a) The mechanism response time .......... t
b) The actuator displacement .......... x
c) The (approx) mechanism mass .......... m

Although the theoretical analysis assumes that all available actuator force is used to accelerate the mechanism, this will only be valid in a limited number of cases. In the more general case, a mechanism will be required to perform some work during its movement,
an example being the needle mechanism of the mattress sewing machine outlined in Section 5. Therefore an estimate of any external mechanism resistance felt by the actuator should also be made.

d) **Actuator resistance/work** ........... R

### 2.8.3 The Power Source

The first step in the design procedure should be to select the desired nominal operating pressure of the system. Whilst the maximum pressure will be dictated by the availability of suitable pump units, the working pressure should be conservatively rated in terms of the maximum available because:

a) *As the system pressure is the only independent variable, it is considered to be the best means of making adjustments to the final mechanism performance. That some final mechanism "tuning" may be necessary after practical results have been obtained should be recognised and allowed for at the design stage.*

b) *Unless deliberate steps are taken to ensure otherwise, pressure drops will occur in the supply conduits as the valve switches over from exhaust to pressure. This will result in a slower mechanism response time than that predicted. To compensate for this loss in pressure, it may be necessary to operate the mechanism at a higher nominal pressure than that originally envisaged. Practical experience gained on the mechanisms tested in this research indicates that pressure drops of up to 15 bar and lasting for 10 milliseconds, can occur even on circuits which employ accumulators to smooth out cyclic flow demands.*

Having established the nominal system pressure, consideration should be given to the siting of the principal circuit elements, viz. the power source, the rotary valve and the mechanism, and to establish the nominal conduit lengths in the circuit. Finally, consideration should be given to hydraulic fluid to be used in the power transmission and to any temperature limits imposed by either the
associated equipment or the process/mechanism to be actuated. Condensing the foregoing points, the following parameters should be established:

i) Nominal system pressure \( P \)

ii) Pipe lengths \( l \)

iii) Oil viscosity \( \mu \)

iv) Oil mass \( m_2 \)

2.8.4 Actuator Area

In addition to depending upon the selection of the parameters outlined above, the size of the required actuator will depend on the type of actuating concept used, e.g.

i) Mechanisms using opposed single-acting modules

ii) Double-acting actuator

iii) Single-acting spring return mechanisms.

The nominal actuator areas found to be necessary for these basic mechanism configurations can be calculated from the following equations:

i) **Opposed single-acting modules:**

\[
PA = \frac{3.5 \times M}{t^2} + R
\]  

(1)

ii) **Double-acting cylinder**

(a) Rod End \( PA = \frac{3 \times M}{t^2} + R \)  

(2)

(b) Piston End \( PA = \frac{4 \times M}{t^2} + R \)  

(3)
iii) Single-acting spring return mechanisms

\[
PA = \frac{14 \times M}{t^2} + R \tag{4}
\]

where \( M \) = mechanism mass + 4 times oil mass.

These equations were derived from the initial assumption that the mechanism displacement \( x \) equals the acceleration displacement defined in Section 1.5. This assumption enables the unknown viscous loss coefficient \( K \) to be eliminated and the actuator area to be calculated from the established parameters. The actuator area is then reduced by a nominal 25% as the theoretical analysis given in Section 1 indicates that the area is unnecessarily large (see Figure 1.16). The equations should be used to estimate the required actuator area and adjusted upwards to the nearest nominal actuator diameter.

2.8.5. Viscous Damping Coefficient \( K \)

The remaining unknown variable is the combined viscous damping coefficient \( K \). The maximum value of \( K \) can be found from the expression:

double-acting mechanism \( K_{\text{max}} = \frac{2pt}{3x} \) \tag{1}

spring-return mechanism \( K_{\text{max}} = \frac{3pt}{7x} \)

where \( p \) is the system pressure available to accelerate the apparent mechanism mass \( M \) i.e.

\[
p = P - \frac{R}{A} \tag{2}
\]

The linearised viscous coefficient includes the combined sum of

i) The mechanism viscous losses

ii) Viscous pipe losses
ii) Linearised momentum losses - valves and fittings.

Realistic estimates of these individual loss terms cannot be made until a design has been finalised. However it is necessary to establish the relative dominance of the individual terms of the coefficients early in the design stage so that supply conduits can be correctly matched to suit a particular mechanism. In the absence of a more accurate estimate of the proportions, the average values for the mechanisms analysed in this research can be used.

i) Mechanism losses 0.2K

ii) Viscous pipe losses 0.6K

iii) Momentum losses 0.2K.

The apportioning of the pipe losses in terms of the maximum overall viscous coefficient enables the bore size(s) of the conduits to be calculated. Using the Hagen-Poiseuille formula given in Section 1.5.2 and the assumption that the pipe losses equal 0.6K, the bore size $D$ of a conduit of length $l$ can be calculated from the equation:

$$D^4 = \frac{128 \mu l A}{0.6K} \quad (3)$$

As $D$ is related to the other variables by its fourth power, the resultant error from the incorrect apportionment of the pipe losses is likely to be small. It is recommended that the bore size calculated using the above equation be rounded up to the nearest nominal conduit size.

2.8.6 Summary of Procedure

It is believed that the foregoing procedure will enable realistic estimates of the system parameters to be made early in a mechanism design, with the minimum of detailed calculations. However, as a design proceeds and more detailed knowledge becomes available, the effect on mechanism performance of any design changes should be monitored. It is believed that Figure 1.16 will be of use to ascertain
the effect of any deviations in the original design estimates in addition to being of use in establishing possible courses of any necessary remedial action. It is further recommended that individual mechanism data sheets, similar to those shown in various appendices, are compiled as a mechanism design proceeds.

The procedure can be summarised for a mechanism using single-acting modules and where pipe losses are apportioned as in 2.8.5. as follows:

1. ESTABLISH: Mechanism response time $t = \text{seconds}$
   Mechanism displacement $x = \text{metres}$
   Approx. mechanism mass $m_1 = \text{kilogrammes}$
   External work/resistance $R = \text{Newtons}$

2. DECIDE: Nominal system pressure $P = \text{N/m}^2$
   Pipe lengths in conduits $l = \text{metres}$
   Oil viscosity $\mu = \text{N sec/m}^2$
   Oil mass $m_2 = \text{kilogrammes}$

3. CALCULATE: Mechanism mass $M = m_1 + 4m_2$
   Actuator area (e.g) $PA = \frac{3.5 \times M}{t^2} + R$
   Accelerating pressure $p = P - \frac{R}{A}$
   Maximum viscous coefficient $K = \frac{2pt}{3x}$
   Conduit bore $D^4 = \frac{128 \mu L A}{0.6K}$

2.9 Spring Return Mechanisms

The formulae given in the preceding section relating to actuator area and viscous coefficient were derived on the assumption that the mechanism:

a) has critical viscous damping
b) is dynamically symmetrical
c) has a residual hydraulic force of 0.2F.

As outlined in Section 1.7 the above design constraints are considered to optimise the opposing criteria of low mechanism response times and a low final impact velocity in the outward direction. As an aid to establishing the viability of these mechanisms the following relationships have also been derived:

\[
\text{Spring stiffness} = \frac{2pA}{3x}
\]

\[
\text{Outward impact velocity} = 56\% \text{ average mechanism velocity}
\]

\[
\text{Peak mechanism velocity} = 138\% \text{ average mechanism velocity}
\]

It is considered essential that the spring characteristics are checked at the outset to ensure that a suitable spring can be obtained. Further, that the proposed mechanism can tolerate the outward impact velocity.
SECTION THREE

THE DESIGN AND ANALYSIS OF PULSE GENERATING ROTARY VALVES

3.1 Introductory Summary

The heart of the actuating and sequencing concept is the rotary valve. Functionally this device connects each actuator port to either system "pressure" or "exhaust". In operation the valve must expedite this function at any rotational speed up to a required maximum value. This maximum being dependent upon the dynamic capability of the actuators it sequences. Furthermore, the valve performance should be as efficient as possible, be compatible with the prime mover and be independent of the circuit's system pressure. This latter proviso being considered desirable so that the actuator/valve concept might be extended in the future to include variable speed/variable pressure systems.

The earliest prototype valves developed by Garside were simple arrangements and only minimal consideration was given to the technical and practical problems associated with their design and operating characteristics. As a consequence, these valves suffered from excessive starting and running torques caused by imbalanced hydrostatic forces acting on the valve bobbin. The valve bobbin was later redesigned to incorporate hydro-static balancing pads. This modification succeeded in reducing torque levels significantly and the valve concept then remained unchanged throughout the remainder of the earlier research. Despite this improvement, the starting and low speed driving torques did not agree with theoretical predictions and were still considered to be too high. The necessity of reducing still further these torque levels and overcoming certain of the valve's operating deficiencies, prompted a further design investigation by the author.

Initial efforts were concentrated in understanding the divergence between theoretically predicted and measured torque levels of a reference valve. This divergence was subsequently attributed to residual hydro-static imbalance causing the valve bobbin to act as a hydrodynamically lubricated loaded journal. A new valve concept,
aimed principally at reducing viscous drag losses and hydro-static imbalance, then evolved. The principal valve feature of sculpturing away the bobbin surface to reduce drag losses and also control the pressure distribution was embodied into the design of a "Universal valve". This valve had a greatly increased pulse selection facility over previous designs, essential to give design flexibility of hydraulic actuator mechanisms. In addition, the valve was made inherently self-balanced so that the necessity of additional hydro-static balancing pads was eliminated.

Following the design concept, the valve was analysed to identify its pertinent losses and operating characteristics. It was concluded that the valve’s flow loss coefficient \( k_* \) (as defined in B.S. 4062) would need to be established by practical tests. However, it was considered that the other losses of internal leakage and viscous drag could be determined theoretically and their combined value minimised to suit predetermined system parameters. Essentially the technique presented involves calculating the correct radial clearance between the valve bobbin and its casing to achieve minimised combined losses. To establish compatibility between the valve and the actuators it sequences during port opening, the transient effect of uncovering a circular supply port was also investigated. A method of identifying compatibility by means of a disproportionately high orifice restriction is presented. Finally, so that the valve may be used to obtain correctly sequenced circuits, the design procedure adopted by the author throughout the research is included for reference purposes.

3.1.1 Previous Rotary Valve Design

The final concept of valve developed by Garside can be seen in Figure 3.01. The valve is comprised of an outer casing into which the power supply and mechanism supply ports are tapped and a closely fitting inner bobbin held in place by end plates; The bobbin has a series of slots machined in its surface which are connected by means of radial and axial drillings to either:

a) A central pressure supply annulus, or
b) An axial exhaust port.
In addition, at the extremities of the bobbin, the hydrostatic balancing pads are connected to the main pressure supply by small drillings.

In operation, the valve would be connected into a hydraulic circuit of the type shown in Figure 2.09. When the valve bobbin is rotated, the mechanism take-off ports in the outer casing will be exposed sequentially to the pressure and exhaust areas. Provided these ports are connected correctly to various linear actuators, a series of sequential movements will be obtained. The phase relationship between the actuator movements being determined by the radial position of the mechanism supply ports relative to their associated pressure/exhaust areas.

3.2 Research Objectives

The success of the above design over the earliest prototype valves is clearly evidenced in the work expedited by Garside. However limitations of the valves practical performance are also indicated and formed the basis of this additional study. The majority of the valve's faults were identified by Dr Garside and would in the normal course of events have been tackled by him, but at this juncture Dr Garside ceased full-time research at Loughborough University of Technology. The remaining limitations became evident when the valve was used on various machinery development.

1. To restrict the magnitude of the imbalance forces, near optimum positioning of the pressure/exhaust areas was necessary prior to applying the computer programme. If this was not done, the size of the compensating pads became disproportionately large and a bulky valve resulted.

2. At rotational speeds below 200 r.p.m. there was a large divergence between theoretically predicted driving torques and the measured torque levels.

3. The valves exhibited a high breakout torque characteristic.
4. The valve was inflexible in application. Each rotary valve was custom designed, balanced and manufactured to suit a particular hydraulic circuit and control sequence. Sequential circuit changes could not be made without disturbing the balancing. This was a source of frustration when the valve was utilised in machine developments and where sequential changes are found to be necessary.

5. The valves were physically large, the valve design did not permit its overall dimensions to be reduced significantly because:

a) The minimum bobbin diameter was dependent on making provision for: the axial exhaust hole, the cross drillings and the pressure/exhaust slots.

b) The length of the bobbin was determined by the pulse generating section plus the hydrostatic balancing pads.

These limitations in the valve's performance formed the basis of a further design study of pulse generating rotary valves, the principal aims of the work being:

i) To improve the low speed characteristics of the rotary valve, in particular to reduce the starting torque significantly.

ii) To design a more universal valve, suitable for a wide range of multi-actuator circuits.

iii) To make the valve inherently self-balancing and so remove the necessity of additional compensating pads.

iv) To make a more compact design of valve.

v) To determine the valve's operating characteristics and to optimise these wherever possible, in particular the following were considered pertinent:

a) The pressure drop across the valve versus flow rate through the valve

b) The bobbin driving torque versus rotational speed

c) The internal valve leakage versus supply pressure
d) The bobbin starting torque versus supply pressure.

Condensing the above, the overall objective became to design a self-balanced, universal, pulse generating rotary valve.

3.3 Pre-Design Investigation

The design work on a new valve commenced with an investigation into the starting characteristics of a computer balanced reference valve. Before any refinements to the valve concept could be introduced it was a necessary pre-requisite that the divergence between the theoretical and practical torque levels at breakout and low rotational speeds be understood.

3.3.1 Previous Balancing Technique

This text is a précis of the technique developed by Garside and is included for the sake of completeness and reference purposes. In order to compensate the out of balance forces and moments acting on the bobbin it was necessary that the pressure distribution over the bobbin surface be known. The assumption was made that the pulse generating section of the rotary valve, with its pattern of pressure and exhaust slots, was analogous to two dimensional hydrodynamic flow with various combinations of:

1) high pressure sources,
2) low pressure exhaust sinks,
3) certain definable boundary conditions.

With such an assumption the pressure distribution over the bobbin surface could be found from the solution of the Laplace equation:

$$\frac{d^2 p}{dx^2} + \frac{d^2 p}{dy^2} = 0$$

However for this equation to yield accurate results the following assumptions were also made.
1) The fluid is Newtonian.
2) The fluid is incompressible.
3) Flow is laminar.
4) The fluid film is so thin that the pressure remains constant across the depth, i.e. \( \frac{dp}{dz} = 0 \).
5) No slip occurs between the fluid and the bearing surfaces.
6) The viscosity of the film is uniform throughout.
7) The effects of elastic and thermal distortion are neglected.
8) Inertia and body forces are negligible.
9) The bobbin axis is parallel to the axis of the casing.
10) The bobbin axis is concentric with the axis of the casing.
11) The curvature of the bobbin surface is large compared to the film thickness so that the film may be unwrapped for analysis.

The solution of the Laplace equation was achieved by using finite difference theory and an over-relaxation technique on a digital computer. The pulse generating section of the valve was unwrapped, the developed area split into a grid pattern, the known boundary conditions inserted and the mean pressure at each of the grid points computed. By multiplying these values by the mesh area, the magnitude and direction of the force exerted at a particular point was found. The computer program summated these results and found the magnitude and direction of the imbalance forces acting on the bobbin. The program then computed the required size and radial position of the hydro-static pads necessary to compensate these forces and balance the bobbin.

### 3.3.2 Starting Characteristics of Reference Valve

A rotary valve designed and balanced using the technique outlined in Section 3.3.1, was taken for assessment and reference purposes and the bobbin breakout torque at different supply pressures obtained. Results were obtained with the power supply connected but with the mechanism take-off ports blanked off. Torque readings were obtained
by applying static loads to a moment arm attached to the bobbin input shaft. The results obtained are given in Table 4 and shown visually in Figure 3.03. Dynamic results were obtained using a variable speed drive and an electronic torque indicator.

Consideration of the results and the detail construction of the valve led to the following observations:

1) The starting characteristics were typical of those normally obtained by a hydro-dynamically, oil lubricated loaded journal bearing: at standstill the shaft rests on the bearing surface, when it starts to revolve the shaft moves up the bore until, at an equilibrium position, the shaft is supported by a wedge of fluid lubricant (See Figure 3.08). The moving surfaces at equilibrium are then held apart by the pressure generated within the fluid film. Bearings are designed so that under normal operating conditions the hydraulic pressure build up carries the rotating shaft and preserves the fluid film between the bearing surfaces. This results in extremely low wear and friction properties.

During start up, before a fluid film has been established, some contact of the metal surfaces is unavoidable. In the case of the particular valve tested this metal to metal contact could be felt when rotating the valve by hand. Consideration of Figure 3.02 indicated that the equilibrium speed of the valve bobbin was approximately 200 r.p.m.

2) The journal load at standstill was attributable to the residual hydrostatic imbalance acting on the bobbin: it was believed that some residual imbalance was inherent in the balancing technique because of:

a) Round off errors in the execution of the computer program.

b) The mesh size employed in the computer program; in order to restrict the computer time and the core capacity to be utilised, a mesh size of 5 mm was used. At points of high pressure gradient therefore, some inaccuracy was possible.
**FIG 3.02** DRIVING TORQUE VERSUS SPEED OF GARSIDE VALVE

**FIG 3.03** BREAKOUT TORQUE VERSUS SUPPLY PRESSURE OF GARSIDE VALVE
c) Assumptions 9 and 10 made in Section 3.3.1 were not valid. Consideration of the detail design, Figure 3.01, showed that the bobbin was in principle only supported by one bearing at the input drive end. For the particular valve tested this bearing was found to be redundant as the clearance between the shaft and its journal exceeded the clearance between the bobbin diameter and its casing. The bobbin was free therefore to find its own path with regard to concentricity and axial alignment with the bobbin casing. This eliminated one of the simplification factors in computing the pressure distribution.

3) The bobbin breakout torque could be used to obtain an indication of the imbalance forces acting on the bobbin by use of the elementary relationship:

\[ \text{Torque} = F \cdot \varnothing \cdot R \]

By assuming an average value for the friction coefficient \( \varnothing \) of 0.25, between the bobbin and the casing, it was estimated that the valve tested had a residual imbalance of 40 Newtons per bar of supply pressure.

4) A typical pulse generating section with its plurality of pressure sources and exhaust sinks would have large areas of high and low pressure as adjacent sources and sinks blend together. This was verified by modifying the computer program for the valve tested to print out the pressure distribution over the bobbin surface. See Figure 3.04. The extent and variability of these areas would however be dependent upon:

a) The source and sink geometry of a particular valve.

b) The variability of the radial clearance between the bobbin and its casing.

c) The superposition of the hydro-dynamic lubrication pressure distribution plus the two dimensional viscous flow calculated by the computer programme.
Bobbin pressure & exhaust boundaries

High pressure area

Low pressure area

Medium pressure area

FIG. 3.04 PRESSURE DISTRIBUTION OVER BOBBIN SURFACE OF A TYPICAL GARSIDE VALVE
3.3.3 Preliminary Conclusions

These observations led to the following conclusions:

1) The theory and balancing technique developed by Garside was basically sound but a certain residual imbalance was to be expected.

2) The bobbin acted as a hydro-dynamically lubricated loaded journal bearing, the journal load being propagated by the residual hydro-static imbalance.

3) The breakout torque and the critical rotational speed were an indication of the amount of residual imbalance.

4) In order to overcome the breakout torque characteristics either:
   a) The balancing technique needed to be improved, or
   b) The residual imbalance needed to be carried more efficiently e.g. by mounting the bobbin in low friction bearings.

5) The pressure distribution over the reference valve bobbin was a dependent variable. An improved valve would result if the pressure distribution was controlled more rigorously by the valve design.

3.4 A New Concept of Rotary Valve

It is impossible to transmit satisfactorily to the written word, the thought processes which contribute to the ultimate selection of a particular design. For this reason no attempt has been made to trace the tortuous path by which the particular valve concept presented here evolved, or to record the various inversions of valve geometry which were considered. The design is the one which appeared the most technically and practically elegant based on:

1) the experiences of earlier research,
2) the investigations into the reference valve,
3) consideration of the theoretical valve losses,
4) the research objectives.

3.4.1 General Description of the Valve

The design of the valve can be seen in Figure 3.05, consideration of which will show that the valve has the same basic construction as previous valves and is comprised of:

a) An outer casing into which are tapped the main hydraulic inlet, the various mechanism supply ports and two valve exhaust ports.

b) A rotatable inner bobbin, a detailed description of which is given in the next section.

c) Two end plates or caps.

d) Two light duty ball bearings.

e) Two shaft lip seals.

For the design illustrated the bobbin has been extended to produce a through shaft and is carried by ball bearings pressed into either end of the outer casing. The bobbin and its bearings are constrained axially by end plates into which are mounted the shaft lip seals.

The main hydraulic supply enters the valve casing and is distributed to the various mechanism supply ports tapped into the pulse generating section of the valve. The valve has two exhaust ports situated at either end of the valve, either one or both of these ports are utilised for the return of the hydraulic fluid to the main supply source.

Functionally the valve concept is similar to previous designs, as the valve bobbin is rotated each of the mechanism supply ports will be exposed to alternate pressure and exhaust areas which can be utilised for the sequential control of hydraulic mechanisms. The bobbin is fundamentally different to previous designs, both in terms of its balancing concept and its internal fluid distribution.
FIG 3.05 UNIVERSAL SELF BALANCED PULSE GENERATING ROTARY VALVE
2.4.2 Bobbin Concept

The bobbin and the development of the pulse generating section can be seen in Figures 3.06 and 3.07. Essentially some 80% of the valve bobbin surface has been sculptured away to leave a series of narrow lands, slightly wider than the mechanism supply orifices. These lands enclose five areas, two of which are connected to the supply pressure source and the remaining three areas to the valve exhausts. These areas represent slices of a trapezoidal shaped pulse generating section which extends over the whole width of the bobbin between the two exhaust annuli. The axial and radial dimensions of the slices have been calculated to produce an inherently hydro-statically balanced bobbin.

One of the research objectives was to make a more universal valve, suitable for a wider range of applications. The theoretical dynamic analysis of reciprocating mechanisms using the techniques outlined in Sections 1 and 2 will indicate the optimum pulse length required for a particular mechanism movement. This pulse length could be (theoretically) any nominal value between 0 and 360° of the bobbin rotation. Experience, with the testing of the prototype hydraulic actuators and the analysis of previous prototype hardware, indicated a pressure pulse length selection of between 60° and 300° was more practical. Thus it was considered desirable that the universal valve should have such a pulse selection facility and hence the pulse generating section of the valve should be a trapezoid as indicated in Figure 3.09.

By selecting the correct axial dimension 'x' from a datum it would be theoretically possible to select any desired pressure pulse length within the maximum and minimum limits. For this valve concept 50% of the bobbin surface is subjected to the system supply pressure, which if left as shown would result in considerable out of balance forces acting on the bobbin and casing. The valve was made inherently balanced in the following manner.
Centre of pressure

Static balance $F_1 + F_2 = F_3 + F_4$
Moment balance $F_1 \cdot a + F_4 \cdot d = F_2 \cdot b + F_3 \cdot c$

FIG 3.06 BALANCING OF PROTOTYPE VALVE

Separating lands
Centres of pressure
Axis of symmetry
High pressure areas

FIG 3.07 DEVELOPMENT OF VALVE BOBBIN
FIG 3.08 HYDRODYNAMIC LUBRICATION OF A LOADED JOURNAL BEARING

FIG 3.09 SCHEMATIC BOBBIN DEVELOPMENT OF PULSE GENERATING SECTION
3.4.3 Balancing Technique

It will be appreciated that for the trapezoidal pressure area chosen there are two axes of symmetry:

1) Centre line axis: as the section is symmetrical along this axis resolved forces in one direction will cancel each other out, Figure 3.10. Therefore when balancing the bobbin only resolved forces \( P \Delta A \cos \theta \) need to be considered.

2) As the trapezoid section exceeds 180° certain areas become inherently self balancing as per (1) above and the effective area which requires balancing is reduced. By selecting the pressure area to be 60° - 300°, two smaller trapezoid areas result, with an axis of symmetry connecting them. See Figure 3.11.

From a balancing viewpoint this reduced effective high pressure zone needed to be manipulated to achieve moment and static balance without the inclusion of compensating pads. The trapezoid areas were therefore subdivided into four parts and manipulated radially, see Figure 3.06, i.e. for static balance:

\[
F_1 + F_3 = F_2 + F_4
\]

for moment balance

\[
F_1a + F_4d = F_2b + F_3c
\]

where \( F_{1234} \) is the total resolved force of areas 1234 respectively and \( a, b, c, d \) are the dimensions of the centres of pressure of areas 1234 respectively from the \( \phi \) axis of symmetry. The resultant schematic development in its balanced format is as shown in Figure 3.12.

The balancing of the schematic development did not in principle present any serious difficulty. In reality the division between the high and low pressure areas must be achieved by means of separating lands. The lands still represented a significant proportion of the bobbin surface and their balancing needed to be included. By stipulating a constant width of separating land and assuming that, due to kinematic constraints, the clearance across each of the lands
FIG 3.10 SECTION THROUGH ROTARY VALVE

FIG 3.11 EFFECTIVE BALANCING AREAS

FIG 3.12 SCHEMATIC BALANCING LAYOUT
is constant, their inclusion into the overall moment and static balance of the bobbin could be achieved in two ways.

**METHOD 1:** To assume that the schematic development extended to the centre line of each separating land.

**METHOD 2:** To balance each land and pressure area separately.

While it was considered desirable to achieve perfect hydrostatic balance, it was felt unlikely that either of the techniques would achieve this. All that could reasonably be expected was that the out of balance forces were sufficiently low that the efficient functioning of the valve would not be affected. The accuracy of balance required of the valve will depend upon various factors, i.e.

a) The maximum system pressure at which the valve would be expected to operate - any inherent hydrostatic imbalance being directly proportional to pressure.

b) The drive characteristics of the prime mover to the rotary valve - the torque versus speed characteristics of the prime mover would need to be compatible with those of the valve and vice versa.

c) The life and duty required from the valve - the valve imbalance would be reacted by the end journal bearings and the lower these are loaded, the longer the life of the valve.

For the reference valve tested, some 2100 mm² of the bobbin surface was utilised for high pressure pulse generation. At a system pressure of 35 bar this could have resulted in a maximum imbalance of 7350 Newtons and an estimated maximum breakout torque of 45 Nm. The measured breakout torque at this pressure was 4.0 Nm and hence the valve was defined as being 41/45 fully balanced, i.e. approximately 91.5% (Garside estimated his error at 10% ref. 2). For the new concept of valve bobbin the high pressure area for a comparable sized valve was increased some five times. Consequently, in order not to exceed the actual breakout torque figures of the
reference valve, the accuracy of balancing needed to be considerably improved. The aim set was to achieve a 50% actual reduction in breakout torque of a valve having plain journals, thus the balancing would need to be 10 times more accurate than the computer technique.

3.4.4 Internal Flow Characteristics

The other major departure from the previous design concepts was in the method of internal fluid distribution from the main inlet to the various mechanism supply ports. In previous valves the fluid flow was directed to individual pressure sources via internal axial and radial drillings. In the universal valve design there are only two large, symmetrical, transverse drillings which connect one side of the valve bobbin to the other. The flow from the main hydraulic supply inlet to mechanism ports fed from areas 2, 3 and 4 will occur peripherally round the bobbin. Area 1 will be supplied via area 2 through one of the transverse drillings. In a similar manner the "exhaust area" enclosed by pressure areas 2, 3 and 4 will feed the right hand exhaust annulus by the other transverse drilling, whilst in the remaining exhaust areas the flow will be peripheral to either one or both of the exhaust annuli.

The high pressure zones form a small irregular shaped internal reservoir and fluid transfer within the valve is free to find its own most efficient path from the inlet supply port to the various mechanism ports. It was believed that this would result in lower pressure versus flow losses than previous designs. The removal of the axial mechanism drillings and in particular the large bore axial exhaust port had two other advantages:

i) It enabled the valve to be designed with a greater permutation of input drives, e.g. drives from either end, through shafted drive, male or female drives.

ii) On small installations with only a limited number of mechanism supply ports, the bobbin diameter could be significantly reduced and a reduction in both the bobbin drag losses and the valve internal leakage obtained (Section 3.5).
3.5 **Theoretical Valve Losses**

Functionally the task of the rotary valve is to switch supply pressure and exhaust on and off various mechanism supply ports. System losses are propagated in achieving this function which must be predictable at the design stage and compatible with the capabilities of the associated hardware. For the design of rotary valve presented here these losses are generated in two distinct areas:

i) The power required to drive the rotary valve bobbin.

ii) The fluid transmission losses through the valve.

3.5.1 **Valve Bobbin Drive**

The power required to drive the rotary valve were divided for analytical consideration into two parts:

a) The power required to overcome mechanical friction losses propagated by the oil seals and the end bearings: one of the basic design aims of the rotary valve was that it should be inherently self-balanced and that subsequently the bearing reactions and theoretical friction losses are nil. As stated in Section 3.4.3 this was considered unlikely in practice even in the static (non-rotating) condition without resorting to fine balancing techniques analogous to those used to solve conventional mechanical balancing problems. When the valve is used as part of a practical system and oil is flowing through it, pressure gradients may arise throughout the pulse generating sections which could disturb the overall bobbin balance, this "dynamic imbalance" would need to be reacted by the end bearings. It was inherent in the basic valve design that these dynamically induced out of the balanced forces could not be calculated. The fluid flow path is indeterminate as it will be constantly changing with the angular position of the valve bobbin. The end bearings must therefore carry the residual static hydrodynamic imbalance of the bobbin and superimposed on this, the dynamically induced imbalance. It was envisaged that valves would incorporate rolling element bearings having a friction
value \( \phi \) of approximately 0.0015. Provided, therefore, that
the magnitude of the imbalance was within the design capa-
bilities of the bearings selected, the resultant torque due
to bearing friction was considered negligible.

The conventional lip seals fitted at either end of the rotary
valve will cause a small drag force to be generated. The
characteristic of this type of seal is one of a very low,
almost constant drag force over a wide range of peripheral
speeds, i.e. \( T = \text{constant} \). Therefore the power required to
overcome the seal friction = \( \text{constant} \times \text{velocity} \).

b) The power required to overcome the viscous shear losses
between the bobbin and the casing: If it is assumed that
the bobbin is constrained to run concentrically, the torque
required to overcome the drag force between the separating
lands and the outer casing can be found from the elementary
expression:

\[
T = \frac{v}{h} \mu \times \text{Area} \times R
\]

and the power required to overcome this torque:

\[
\text{power } w = \frac{v^2 \text{ Area}}{h} \mu.
\]

The area can be found by multiplying the land width by the
total developed length of the various lands.

3.5.2 Fluid Transmission Losses

The fluid flow losses through the valve were divided for
analytical purposes into two parts:

1) Internal leakage flow: The high pressure areas are separated
from the exhaust areas by relatively narrow lands. If it is assumed
that the bobbin is constrained to be concentric to the outer
casing, the theoretical leakage can be estimated from the expre-
ssion:
\[ Q = \frac{p \cdot h^3 \cdot L}{12 \mu b} \]

where: \( L \) = length of separating lands.
\( b \) = land width.

and the resultant power loss

\[ \text{power loss} \ w = \frac{p^2 \cdot h^3 \cdot L}{12 \mu b} \]

these expressions being derived from basic flat plate theory, the clearance ‘\( h \)’ being very small compared with the bobbin radius enabling the valve to be "unwrapped" for analytical purposes.

2) Internal flow restrictions: The principal flow losses will occur:

a) At the main hydraulic fluid entry to the valve supply annulus.

b) The internal flow losses as the fluid is distributed to and from the various pressure and exhaust areas.

c) The orifice losses of the mechanism supply ports.

To obtain theoretical estimates of these losses it is necessary to know how the fluid velocity distribution around the periphery of the valve. This would be difficult to establish for even a stationary valve bobbin and considerably more so under the dynamic condition whereby mechanism supply ports are constantly opening and closing. The conclusion was drawn, therefore, that the internal flow losses were indeterminate analytically and that only practical tests would yield an accurate relationship of the valves pressure drop versus flow characteristics.

3.5.3 Combined Losses

The principal losses directly governable by the valves metrology and geometry were considered to be:
a) The viscous drag losses between the bobbin and the outer casing.

b) The internal leakage across the separating lands,

and that the combined power loss of these terms could be expressed as follows:

\[ \text{Power loss } w = \frac{p^2 L^3 h^3}{12 \mu b} + \frac{\mu v^2 \text{Area}}{h} \]

Consideration of the above equation indicated that owing to the inverse relationship of the variables 'h' and 'μ' plus the squared terms of pressure and velocity, a rotary valve cannot be designed for minimum losses to suit all operating conditions. However for a particular application certain of the variables in the above expression will be determined by a general dynamic analysis of the related mechanisms, these would normally include:

i) The required system pressure.

ii) The normal running speed of the valve.

iii) The viscosity of the oil to be used.

iv) The physical dimensions of the valve.

The inclusion of these terms into the above expression will reduce it to the form:

\[ \text{Power loss } w = A h^3 + \frac{B}{h} \]

\[ \frac{d(w)}{dh} = 3A h^2 - \frac{2B}{h^2} \]

from which the optimum clearance h for a particular set of design parameters can be obtained.

It was noted that as the leakage power loss is dominated by the radial clearance h, a 10% deviation could result in errors between theoretical and practical results of 33%. Similarly it is known that
shaft eccentricity has a marked effect on the accuracy of leakage flow rates, at maximum eccentricity the practical leakage could be 2.5 times the theoretical computed values. Theoretical calculations indicated that even under the most adverse conditions, the resultant valve leakage would represent only a small percentage of the overall system losses. Similarly it can be shown that the viscous drag losses are dominated by the valve diameter and that the power loss is proportional to $D^2$.

3.5.4 Valve Port Transient Flow Characteristics

The overall flow characteristics of the rotary valve were to be found by practical tests. The pressure drop across the valve would be incorporated into the dynamic analysis of an actuator/valve circuit by means of a loss coefficient $k_\ell$ (Section 1.5). This coefficient would, however, only relate to a fully open mechanism supply port. In practice each port will require a finite time to open and if this represents a high proportion of a particular mechanism's response time, then some performance down-rating may occur. Thus it was considered necessary to establish a method of assessing compatibility between the flow capability of the valve and the mechanism flow demand during port opening.

If compressibility flow is ignored, a mechanism actuator will be subjected to full system pressure as soon as the valve port starts to open. Thus the equations of motion derived in Section 1 will apply, i.e. $t = 0$ when the valve port starts to open. Application of the theoretical equations of motion will enable the calculation of:

a) The mechanism flow requirement at any time 't' from commencement of travel (velocity $v$ time relationship).

b) The rate at which flow demand varies with time (acceleration $v$ time relationship).

In a similar manner a relationship between valve port orifice area and time can be established for any particular valve rotational speed and by correlating the mechanism flow demand against the flow capability of the valve, a measure of compatibility can be obtained.
A measure of compatibility using this approach will be a disproportionately high orifice pressure drop denoting orifice restriction.

**Circular Valve Port Opening Characteristics**

The opening characteristics were established by analysing the change in orifice area 'a' as a dividing land passes across the supply port, see Figure 3.13.

\[
\frac{da}{dx} = \text{chord length 'L'} = 2 \sqrt{xD - x^2}
\]

\[
= 0 \quad \text{at } x = 0 \text{ and } x > D
\]

\[
= D \quad \text{at } x = D/2
\]

\[
= \frac{\pi D}{4} \quad \text{(average)}
\]

Now the flow \( Q_v \) through a fully open valve can be expressed by the elementary relationship

\[
Q_v = a \sqrt{\frac{2\Delta p}{\rho k_\star}}
\]

Now if \( k_\star \) is assumed to be constant during the port opening, the change in the flow capability of the valve orifice can be expressed as follows:

\[
\frac{dQ_v}{dt} = \frac{da}{dx} \cdot \frac{dx}{dt} \cdot \frac{dQ_v}{da}
\]

\[
= \frac{da}{dx} \omega R \cos \gamma \sqrt{\frac{2\Delta p}{\rho k_\star}}
\]
**FIG 3.13 CIRCULAR PORT GEOMETRY**

- **Trailing edge of separating land**
- **Orifice area**
- **Bobbin axis**
- **V = \omega r**
- **\( L / 2 \)**
- **X**
- **6X**
- **% Area**
- **100**
- **50**
- **0**
- **R, \theta mm**
- **da/dx = \pi d/4**
- **Port fully open**
In a similar manner, the rate of change of the mechanism flow demand \( Q_m \) can be expressed:

\[
\frac{dQ_m}{dt} = X A = \frac{PA^2}{M} @ t = 0 \text{ (maximum)}
\]

Now if the flow characteristics \( Q_v \) and \( Q_m \) and their derivatives are compared, the following will be deduced:

1) At \( t = 0 \) the actual flow demand and capability are compatible i.e. both equal zero but that:
   a) \( \frac{dQ_m}{dt} \) is at a maximum at \( t = 0 \)
   b) \( \frac{dQ_v}{dt} = 0 \) at \( t = 0 \) because \( \frac{da}{dt} = 0 \)

Therefore at the port cracking position, the change in mechanism demand will always exceed the flow capability of circular ports.

2) \( \frac{da}{dx} \) changes rapidly, having a maximum value of \( D \) and an average value of \( D/4 \). An indication of the compatibility of the mechanism demand and the valve port opening characteristic can be made if this average figure of \( \frac{da}{dx} \) is taken and used to equate the maximum mechanism demand and average flow capability, i.e.

putting \( \frac{dQ_m}{dt} = \frac{dQ_v}{dt} \)

\[
\frac{PA^2}{M} = \frac{\pi D}{4} \omega R \cos \gamma \sqrt{2\Delta p}
\]

Now for predetermined values of the variables \( P, A, M, R, D, \rho \) and \( \cos \gamma \) the above expression is of the form:
where $\beta$ is an overall valve constant. For a particular value of $\Delta p$ a minimum rotational speed can be calculated above which value no valve orifice restriction should be apparent. Conversely at a particular valve rotational speed, $\Delta p$ can be calculated and compared with the main supply pressure. Should $\Delta p$ represent a significant proportion of the main supply pressure, then some mechanism performance downrating may result. An example of this approach is given in Appendix 2.1.

3) $\frac{da}{dx}$ can be controlled by the mechanism orifice geometry. The above analysis relates to a circular orifice and which yields incompatibility at $t = 0$. If square or trapezoidal shaped orifices are used $\frac{da}{dx}$ will be a constant and the approach outlined above would be more theoretically exact. It is debatable whether in practical terms the additional complication of non-circular orifices is worth a significant pay off in terms of fidelity of dynamic predictability. For the practical systems investigated to date the response times of the mechanisms were in the range of 5-50 milliseconds, with port opening times of less than 1 millisecond to reach the average value of $\frac{da}{dx}$. The likelihood of this initial opening restriction being physically apparent is considered remote. There may be a case for special orifices when ultra high speed response is of importance, for example, on the knitting machine applications investigated by Garside (Ref. 2) when mechanism response times in the order of 2 milliseconds might be required.

The valve port closing characteristics were not considered for two reasons:

a) For the majority of applications the nominal pulse length selected for a particular movement will exceed the actual movement pulse length requirement. The movement will have terminated earlier, but supply pressure will need to be maintained to a particular supply port to prevent the mechanism drifting or
becoming unstable whilst other mechanisms are being operated.

b) For those applications which require a basic reciprocating movement at maximum speed, i.e. when the pressure pulse equals the mechanism response time, the control technique is such that the fluid is cut off at the mechanism port before the valve port orifice closes.

3.6 The Application of the Valve to a Sequential Circuit

The universal valve concept was aimed at accommodating a wide range of circuits by having a pulse selection facility between 60° and 300° of bobbin rotation. The valve casing was designed initially with only the main supply inlet and exhaust ports shown. The mechanism supply ports being added to suit the particular circuit. To ensure that correct mechanism operating sequences were obtained, a simple design technique was used to establish the correct coordinates of the mechanism ports. This involved the use of a schematic development of the pulse generating section prepared from the design calculations. To illustrate the technique, the circuit used for testing the prototype mechanisms, shown in Figure 2.01, is given as an example.

The initial step in any circuit is to ascertain the desired sequence of motions of the mechanisms involved. The test rig shown in Figure 2.01 was designed to simulate the mechanical motions of a chain stitch sewing machine. This rig involved 3 separate mechanisms and ten mechanism supply ports. The displacement profiles of the mechanisms (the needle, looper, and cross-feed), together with the associated pressure/exhaust switching sequence are shown in Figure 3.14. From this was compiled a table of the pulse lengths and phase angles required at each mechanism port. The design technique involved the translation of this information into machining coordinates of the valve casing. To obtain the correct coordinates, use is made of the schematic layout of the pulse generating areas of the valve, Figure 3.15. This figure gives the axial dimension from the centre line of the valve bobbin and casing for different pulse
FIG. 3.14 PRESSURE SWITCHING REQUIREMENTS OF PROTOTYPE RIG

<table>
<thead>
<tr>
<th>Port No</th>
<th>Pulse Length (°)</th>
<th>Phase Angle (°)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>140</td>
<td>40</td>
</tr>
<tr>
<td>2</td>
<td>80</td>
<td>160</td>
</tr>
<tr>
<td>3</td>
<td>80</td>
<td>320</td>
</tr>
<tr>
<td>4</td>
<td>140</td>
<td>220</td>
</tr>
<tr>
<td>5</td>
<td>120</td>
<td>90</td>
</tr>
<tr>
<td>6</td>
<td>170</td>
<td>80</td>
</tr>
<tr>
<td>7</td>
<td>170</td>
<td>250</td>
</tr>
<tr>
<td>8</td>
<td>100</td>
<td>250</td>
</tr>
<tr>
<td>9</td>
<td>140</td>
<td>180</td>
</tr>
<tr>
<td>10</td>
<td>140</td>
<td>320</td>
</tr>
</tbody>
</table>
FIG 3.15 SCHEMATIC LAYOUT OF PROTOTYPE VALVE PULSE GENERATING AREAS.
FIG. 3.16 MACHINING CO-ORDINATES OF VALVE CASING FOR PROTOTYPE RIG

<table>
<thead>
<tr>
<th>port</th>
<th>X (mm)</th>
<th>θ (°)</th>
<th>Tap &amp; Drill thro'</th>
<th>port</th>
<th>X (mm)</th>
<th>θ (°)</th>
<th>Tap &amp; Drill thro'</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>+26.5</td>
<td>287</td>
<td>(\frac{3}{16})ENOTS ø3</td>
<td>8</td>
<td>+49</td>
<td></td>
<td>(\frac{3}{16})ENOTS ø3</td>
</tr>
<tr>
<td>2</td>
<td>+59</td>
<td>200</td>
<td></td>
<td>9</td>
<td>+26.5</td>
<td>67</td>
<td></td>
</tr>
<tr>
<td>3</td>
<td>+59</td>
<td>0</td>
<td></td>
<td>10</td>
<td>+26.5</td>
<td>207</td>
<td></td>
</tr>
<tr>
<td>4</td>
<td>+26.5</td>
<td>107</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>5</td>
<td>+36.5</td>
<td>328</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>6</td>
<td>-11.5</td>
<td>175</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>7</td>
<td>-11.5</td>
<td>345</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>
lengths. For example, Port 1 required a pulse length of 1400°, this occurs at 39 - (6 x 2.5) mm from the \( \theta \) i.e. 24 mm. This dimension can then be added to a valve casing machining chart similar to that shown in Figure 3.16.

To establish the radial coordinate, the required phase angle must be measured from the start of the pulse initiation line, in the direction of the valve rotation. The resulting radial position \( \theta \) can be read directly from Figure 3.15 and also entered onto the machining chart. The resolution of \( \theta \) is given in movements of 100°, however, interpolation by eye will give a greater accuracy.

This procedure is adopted for each of the mechanism supply ports until the circuit is completed. As each port position is established its position on the schematic layout should be indicated by an identification number as shown. The actual coordinate must then be enclosed by an area which indicates the minimum distance that another port may be placed next to it. This "minimum approach" distance being governed by size of pipe fittings to be used.
4.1 Introductory Summary

It was believed that the valve concept outlined in Section 3 would result in a more versatile unit and further, that such a valve could be made self-balancing. To substantiate the design concept and to ascertain the valve's operating characteristics necessitated the manufacture and testing of prototypes. Although the original concept envisaged the use of rolling element bearings to carry the valve bobbin, it was decided that initial prototypes should utilise plain journals. To specify correctly sized bearings it is necessary to know the duty required of them. As outlined in Section 3.2 the breakout torque of a bobbin having plain bearings can be used to estimate the magnitude of the unbalanced forces. It was intended that practical results obtained from valves having plain bearings would be used to specify the correct size of bearings on subsequent units.

The theoretical analysis of the valve also indicated the advantage of having small bobbin diameters. Although the valve concept enabled small bobbin diameters to be utilised, the decision was taken that the initial prototypes should be made with the same bobbin diameter as the reference valve. This would enable comparisons between the two valve concepts to be made on a one to one basis. Work proceeded initially on the design of a valve having a bobbin diameter of 2.000" and a pressure pulse facility of between 600° and 3000° of bobbin rotation. To enable comparison between the accuracy obtainable from the two balancing techniques outlined in Section 3, two valve bobbins were manufactured to fit a common outer casing. The detail design data and précis of the design procedure adopted is given separately in Appendix 4.1.

A series of detailed tests were carried out to establish the validity of the "universal" valve design concept and to quantify the pertinent valve characteristics. The results of these tests and their comparison with the reference valve are outlined. The experience gained by using the first prototypes for actuator tests
and for the mattress sewing machine control led subsequently to the concept of a small valve module. This valve module was based on the same balancing and design idea as the "universal" valve but had a limited pulse selection facility. Additional tests were then carried out on this valve module to confirm and quantify the advantages of small diameter valve bobbins.

4.2 Manufacture of Valve Bobbin

The bobbin was manufactured by producing a blank, with the central inlet annulus and the exhaust annuli accurately machined in its periphery. The bobbin blank was then held in a dividing head on a conventional milling machine and the coordinate pilot holes calculated in Appendix 4.1 machined in the critical areas of the bobbin geometry.

A paper development of the bobbin land geometry was then glued to the surface of the pulse generating section using the pilot holes and annuli as datums. It was then a relatively simple task, if somewhat tedious, to machine the bobbin surface away up to the reference holes. The paper development being used as a rough guide to ensure that helizes were generated in the correct manner.

Despite this procedure some difficulties were encountered in the manufacture of the bobbin balanced by method 1. The original datum holes were incorrectly placed and the resulting valve geometry was marginally in error on all areas. This was not discovered until manufacture was nearing completion: rather than scrap the bobbin, it was finished off. It was felt that the resulting valve geometry could be measured, its theoretical imbalance calculated and subsequently compared with practical results. The bobbin balanced by method 2, however, proceeded smoothly and eventually both bobbins were ground to suit the same outer casing.

4.3 Valve Casing Manufacture

The valve casing is essentially an accurately machined thick tube. Ideally the part required fine boring and the final accuracy
and surface finish obtained by grinding. However, owing to a lack of an internal grinding facility, the basic blank was first turned, the various supply ports drilled and the bore finished off by honing. The honing was achieved by attaching a tool to the spindle of a radial drill. Some difficulty was encountered in producing an accurate bore, the hone having a tendency to follow any irregularities in the original machining. The final accuracy that was attained resulted in the bore having an ovality of 10 microns (0.0004") and a taper of 12.5 microns (0.0005").

4.4 Valve Testing

The design of the new valve had yielded certain theoretical results regarding the projected performance of the valve. To validate the analysis and to obtain practical data of the valve's operational characteristics, a series of tests were devised. A test rig was prepared, Figure 4.01 whereby the valve was driven by a variable speed mechanical drive through an electronic torque indicator. The main hydraulic supply to the valve was from a gear pump and tank unit. For details of the equipment used, together with a sample calibration of the various instruments, refer to Appendix 4.2. The tests which were carried out, together with a description of the test procedure adopted, were as follows:

4.4.1 Breakout Torque Versus Supply Pressure

Before coupling the variable speed mechanical drive to the valve, a moment arm was attached to one of the input shafts of the valve bobbin. The mechanism supply ports were blanked off and the valve was subjected to supply pressures varied between 0 and 69 bar. The breakout torque at different angular positions of the valve bobbin was obtained by loading the moment arm using a force balance. A note was taken of the force required to cause the initial turning motion against a particular supply pressure. From a series of initial tests it was concluded:

1) That the valve had a clearly defined tight zone which extended for some 90° of the bobbin rotation. The torque figures obtained
Fig 4.01
Valve Test Rig
over this zone were significantly higher than at all other valve positions.

2) That at a nominal angular position of the bobbin, at a particular supply pressure, there was a significant scatter of results which increased as the supply pressure was increased.

3) That away from the tight zone the breakout torque was independent of angular position of the valve bobbin and the direction of the applied moment.

The valve was stripped down and the cause of the tight zone investigated. Using conventional toolroom measuring equipment, the bobbin and casing were rechecked for ovality and taper. Results showed that subsequent to the final honing operation the casing had collapsed slightly and its ovality had increased. It was the opinion of the manufacturing staff that, as both the bobbin and the casing were to their nominal finished size, it was impractical to reduce the ovality or diametral variation. Despite the existence of the tight zone, the initial quantitative results of the valve were encouraging and the decision was taken to continue to test the valve. The valve was reassembled and the following data collected at supply pressures varied between 0 and 69 bar.

a) The mean breakout torque over the tight zone.

b) The average maximum breakout torque at positions away from the tight zone.

c) The average minimum breakout torque away from the tight zone.

d) The mean breakout torque of the unbalanced prototype bobbin.

The results obtained are given in Table 6 and are shown plotted in Figures 4.02 and 4.03; for comparative purposes the maximum and minimum breakout torques of the computer balanced reference valve are also shown plotted in Figure 4.03.
FIG 4.02 BREAKOUT TORQUE VERSUS SUPPLY PRESSURE OF PROTOTYPE VALVE 1
FIG 4.03 COMPARISON OF BREAKOUT TORQUE VERSUS SUPPLY PRESSURE CHARACTERISTICS
4.4.2 The Driving Torque Versus Rotational Speed of an Unloaded Valve

The mechanical drive was connected to the rotary valve and without any supply pressure connected, but with the valve full of oil the following tests were undertaken:

1) The driving torque for various bobbin rotational speeds at 20°C.
2) The driving torque for various bobbin rotational speeds with one of the valve end plates and seals removed at 20°C.
3) The driving torque at various bobbin rotational speeds at an oil and valve temperature of 45°C.

The results obtained are given in Table 5 and are shown plotted in Figure 4.04. There was no discernible difference between the gradients of the drive torques of tests 1 and 2 therefore only one set of data is presented.

4.4.3 The Effect of Static Supply Pressure on the Driving Torque Characteristics

The rotary valve was connected to the main hydraulic supply, the mechanism ports blanked off and the valve rotated at a constant speed of 500 r.p.m. The main hydraulic supply to the valve was varied between 0 and 69 bar and the corresponding driving torque noted. This was repeated at various rotational speeds between 100 and 500 r.p.m. The results of the two extreme speeds are given in Table 7 and are shown plotted in Figure 4.05.

4.4.4 The Effect of Pulsating Flow Through the Valve on the Driving Torque Characteristics

The prototype test rig (Section 2) test circuit was connected to the rotary valve. At different rotational speeds the supply pressure to the valve was varied between 0 and 55 bar, and a note of the resultant driving torque taken. As the test circuit was connected to the rotary valve using nylon pipe (rated nominally at 30 bar) courage evaporated at 55 bar. The results are given in Table 8 and are shown plotted in Figure 4.06.
Theoretical prototype results

Oil temperature
20°C

Oil temperature
45°C

Practical result of valve module

FIG 4.04 DRIVING TORQUE VERSUS SPEED OF UNLOADED VALVE
FIG 4.05 EFFECT OF SUPPLY PRESSURE ON THE DRIVING TORQUE OF VALVE - PORTS BLANKED OFF
FIG 4.06 EFFECT OF SUPPLY PRESSURE ON DRIVING TORQUE OF VALVE MECHANISMS CONNECTED
4.4.5 Valve Internal Leakage

With the mechanism supply ports blanked off the valve was subjected to supply pressures between 0 and 69 bar. The valve leakage was collected from the exhaust ports over a period of one minute and weighed, a note of the mean oil temperature being taken at each test. An initial set of results was obtained with the valve bobbin stationary and a second set with the bobbin rotating at 100 r.p.m. The results obtained are given in Table 9 and are shown plotted in Figure 4.07.

4.4.6 Valve Internal Flow Losses

One of the mechanism supply ports was connected via a flow meter to the supply tank. A differential pressure transducer was connected across the entry of the main supply inlet to the valve and the mechanism outlet port. By the use of a separate restricting valve the flow rate through the valve was varied between 0 and 10 litres/min and the resultant pressure drop across the valve obtained. This test was carried out with the valve bobbin stationary and the supply port fully open, the results obtained, are given in Table 10 and shown plotted in Figure 4.08.

4.4.7 Mode of Valve Imbalance

To identify the mode of imbalance, the valve was stripped down and inspected for wear points which would indicate contact between the bobbin journals and the casing. However, as these points were not readily discernible, the lands and journals were stained using gun metal and the valve reassembled. The valve was then subjected to a static supply pressure of 55 bar and rotated at approximately 30 r.p.m. After running for several minutes the valve was stripped down and reinspected. This time the contact points were clearly visible and a reproduction of results obtained is shown in Figure 4.09.
Mass flow (g/m) vs. Supply pressure bar

- Reference valve
- \( \times \) denotes theoretical leakage at max'm bobbin eccentricity
- Theoretical leakage ~ concentric bobbin

**FIG 4.07 VALVE LEAKAGE VERSUS SUPPLY PRESSURE OF PROTOTYPE VALVE**
FIG 4.08 PRESSURE DROP VERSUS FLOW CHARACTERISTIC OF PROTOTYPE VALVE
FIG 4.09  RUBBING TRACE OF PROTOTYPE VALVE BALANCING METHOD TWO
4.4.8 Valve metrology

The detailed measurements of the valve casing and the bobbin were obtained with the aid of the Department of Engineering Production of Loughborough University of Technology, who possessed sophisticated measuring equipment. The results obtained are given in Table 12 and given pictorially in Figure 4.10.

4.5 Analysis of Results: Prototype 1

For the purposes of comparison between practical performance and theoretical predictions, all theoretical results quoted in this section have been based on the measured mean diametral clearance between the bobbin and the valve casing. The clearance was obtained by reading the clearance between the nominal bobbin diameter and the casing at 10 mm intervals along the valve using Figure 4.10 and obtaining the average value. This was established as being 0.000835" and hence for the theoretical analyses 'h' is taken as 0.000417 (10.6 microns) this value being slightly lower than that used for the design calculations in Appendix 4.1.

4.5.1 Cause of Tight Zone on the Prototype Valve

The existence of a tight zone in the prototype valve had a significant effect on its performance. It was necessary therefore to understand its cause before any conclusions could be drawn from the various tests. Consideration of the valve's metrology showed that the casing had considerable ovality and diametral variation, and that the bobbin was slightly tapered and eccentric.

The intention of the prototype design was for the imbalance load to be carried by the plain journals, these being at the positions shown dotted in Figure 4.10. If the bobbin is placed inside the casing and the journals are resting on the bottom surface of the casing, it should lie at an angle \( \alpha_1 \). Owing to the slight taper of the bobbin the top surface will however lie at an angle \( \alpha_2 \). On to this figure has been superimposed the nominal diameter and the
FIG 4.10  METROLOGY OF FIRST PROTOTYPE VALVE
eccentricity zone of the valve bobbin. It can be seen that this figure indicated the probability of a zone of interference between the minor diameter of ovality of the casing and the bobbin at 120 mm from end 1. The foregoing assumed that there is no bowing of the valve casing, the existence of bow will exacerbate the condition and increase the interference zone.

In practice, the situation was not as described. Owing to the interference zone the bobbin was constrained to lie at some intermediate angle with journal 2 lifted clear from the casing. This was verified by the rubbing traces obtained from the valve, Figure 4.09, which indicated that the offending zone was caused by points 'a' and 'd' passing across the minimum diameter of ovality. These points correlated both on axial and radial position, with the known data. The rubbing trace also indicated that journal 2 was virtually redundant, only the slightest contact being made between it and the casing. Owing to the mode of imbalance, the residual imbalance load which should have been carried by journal 2 was in reality carried by the separating land enclosing area 1.

4.5.2 Mode of Valve Imbalance

The rubbing trace, Figure 4.09, in addition to verifying the cause of the valve's tight zone was also used to indicate the mode of the residual imbalance. Consideration of the trace showed that the journal at end 1 was carrying the greatest load and that the other major load bearing area was the separating land enclosing area 1. From a knowledge of the radial positions of these contact points it was possible to deduce that the bobbin had both residual moment and static imbalance. The only combination of moment and static imbalance which satisfied these known reactions was:

1) \[ F_1 + F_3 > F_2 + F_4 \]

2) \[ M_{23} > M_{14} \]

provided also that the static imbalance journal reactions were lower
than the moment imbalance reactions.

Referring to Table 16, it will be noted that this valve bobbin, which was balanced by method 2 from data obtained by method 1, (Section 3.4.3) would produce a residual imbalance:

1) \( F_2 + F_4 > F_1 + F_3 \)

2) \( M_{14} > M_{23} \)

and that the dominating residual force was caused by moment imbalance. Steps were therefore taken to increase \( F_1 + F_3 \) and \( M_{23} \). The practical results indicated that the valve had been marginally overcompensated.

### 4.5.3 Accuracy of Balancing Method 2

Consideration of Figure 4.03 will show that away from the tight zone the prototype was considerably better balanced than the reference valve. One of the principal objectives was to obtain a reduction in actual breakout torque of 50%. If the mean torques at a supply pressure of 55 bar are compared it will be found that the breakout torque of the reference valve was 7.75 Nm compared with 1.2 Nm for the new concept, a reduction of 84.5% in absolute terms. The true comparison is however made when the decrease in torque due to the residual hydrostatic imbalance only, is considered. Contained in the figure of 7.75 Nm for the reference valve was approximately 0.2 for seal friction, whilst seal friction in the prototype was 0.4 Nm. The torque due to residual hydrostatic imbalance was therefore 7.55 for the reference valve and 0.8 for the prototype, a reduction of 89.4%.

As indicated in Section 3.3.2 the accuracy of the balancing method can be estimated from the elementary expression:

\[ T = F \phi R \]

If \( \phi \) is assumed to be 0.25 at \( P = 69 \) bar, the above equation indicated
that the combined journal reactions were approximately 190N. At 69 bar the bobbin was subjected to total load of approximately 70,000N and the mean accuracy of balance was taken to be 99.73% compared with the estimated 91.5% for the reference valve.

Checking balancing method 1 by method 2, Appendix 4.1, indicated that the former technique would have a residual imbalance of 110N per unit pressure. At a supply pressure of 69 bar this would have resulted in a residual imbalance load of 750 Newtons. As stated in Section 4.5.2 results indicated that method 2 had over-compensated by 190N (25%). It was not possible to determine whether this was attributable to the actual technique or to manufacturing errors. Putting the error into perspective, a residual imbalance load of 190N @ P = 69 bar represented an area of 5 mm², insignificant when compared with the total area of the high pressure zones and the separating lands.

4.5.4 Driving Torque Characteristics

i) Unloaded Valve

Consideration of the driving torque versus speed characteristics of an unloaded valve, Figure 4.04 will indicate that good linear plots were obtained in all the test runs. This was in agreement with the theoretical analysis which indicated that \( \frac{dT}{dV} \) = a constant for a given oil viscosity. Theoretical torque speed relationships based on the average clearance of the prototype valve at various operating temperatures are also plotted on Figure 4.04. It will be noted that this resulted in close agreement at an oil temperature of 20°C but was slightly less accurate at 45°C. It is believed that the disparity at the higher temperature was attributable to inaccurate temperature measurements of the oil at the valve.

Owing to physical limitations the temperature of the oil was measured at the supply tank of the hydraulic power pack and not directly at the valve. For the initial test carried out at the ambient temperature this was acceptable. As the temperature of the oil was induced to rise, natural heat losses between the valve and atmos-
phere will occur and the accuracy of the oil temperature measurement will become more suspect. In absolute terms the disparity between the practical and theoretical results was small. The tests did show quite clearly the dominating influence of fluid viscosity and hence the oil temperature on the drive torque characteristics.

As stated in Section 4.4.2 no appreciable variation in the drive torque of the valve was recorded with one end plate removed. Some small shear losses between the end plates and the end faces of the plain journals had been expected but these could not be detected. It was also verified that the lip seals exhibited the constant torque v speed characteristic claimed by their manufacturers.

ii) Valve subjected to static pressure

Consideration of Figure 4.05 will indicate that when the valve was subjected to a static supply pressure:

a) there was minimal effect on the drive torque characteristic at a rotational speed of 500 r.p.m.

b) considerable fluctuating torque was registered at 100 r.p.m. These fluctuations steadily worsened as the pressure was increased.

The two plots show the extremes in performance levels obtained, between the range 100-500 r.p.m. - the characteristics gradually changed. At 400 r.p.m. the drive torque exhibited a small fluctuation and a small rise as the pressure was increased, at 300 r.p.m. an increased fluctuation etc. It was established that the peak torques coincided exactly with the known radial position of the tight zone. It is believed that the reasons for the transition between the two plots was attributable to the characteristics of hydrodynamic lubrication.

Under conditions of hydrodynamic lubrication the valve bobbin will be supported on a film of oil around the journals and the offending land will be lifted clear. However as the speed is reduced and/or the imbalance load is increased the interference zone becomes apparent and causes the fluctuating torque. The limit of this trend would be for the torque to fluctuate between the maximum
and minimum torques established in 4.4.1.

The limiting trend was verified by an additional test with the valve rotating at 20 r.p.m. At this speed the supply pressure was steadily increased from zero to 45 bar. The results described a similar torque/pressure envelope to that encountered when measuring the breakout torque (Figure 4.02). At pressures higher than 45 bar the mechanical drive unit stalled when the maximum torque was encountered. The stalled position was also used to confirm the angular position of the tight zone and the accuracy of the torques measured using the moment arm (Section 4.4.1).

iii) Valve subjected to pulsating flow

Consideration of Figure 4.06 will indicate that the inverse of the results obtained for the static supply pressure test was obtained for pulsating flow. At the lower driving speeds the valve exhibited little change in the driving torque at various supply pressures but as the speed was increased the torque rose significantly at supply pressures above 35 bar. At these higher speeds and pressures the mechanical drive unit was unable to maintain a constant speed and considerable droop was experienced. Some torque fluctuations were experienced at the higher speeds and pressures, but these were less severe than the fluctuations experienced with the static supply pressure. Despite a knowledge of the valve metrology and of the previous test results, no satisfactory explanation could be deduced for this phenomena. It was resolved therefore to reserve judgement until it could be established whether this characteristic was peculiar to the particular valve (i.e. in view of the tight zone) or was a feature of the design in general.

4.5.5 Valve Internal Leakage

Consideration of the practical results showed that there was an insignificant difference in leakage levels between the valve rotating at 100 r.p.m. and when stationary, therefore, only one curve was constructed from the data, this being plotted on Figure 4.07. The reference valve was also tested for its internal leakage but at supply
pressures above 15 bar the leakage was so severe that these tests were discontinued, those results that were obtained are also shown plotted on Figure 4.07.

The theoretical leakage of the prototype valve at each of the test supply pressures was calculated based on the mean radial clearance of the valve, and these are given in Table 11. An allowance was made in the calculations for the change in oil viscosity due to temperature rise and for the cylinder expansion due to the supply pressure. The table gives two sets of theoretical results.

1) Assuming that the valve bobbin and casing were concentric, the theoretical leakage was calculated using the equation given in 3.5.1.

2) Assuming maximum eccentricity, it can be shown that under full eccentricity between two circular surfaces that the theoretical leakage calculated by 1 above will be increased 2.5 times.

Comparison between the data will indicate that close agreement between the practical results and the theoretical results at maximum eccentricity was achieved. This was not unexpected as previous tests had indicated that at standstill the valve bobbin was eccentric due to the residual hydrostatic imbalance. The results also indicated that at 100 r.p.m. the bobbin was still running at almost maximum eccentricity. That such a high leakage rate was encountered when testing the reference valve was surprising, however a subsequent check on the physical dimensions of the valve showed that the valve had a mean radial clearance some 3 times that of the new valve. Theoretically therefore the leakage rate of the valve would approach \(3^3\) that of the prototype and could not therefore form the basis of a realistic comparison.

### 4.5.6 Pressure Drop Versus Supply Pressure

The theoretical analysis indicated that, although no quantitative assessment can be made of the internal flow losses of the rotary valve, the relationship between the pressure drop across the valve
and the oil velocity through a mechanism supply port was of the form:

\[ \Delta p \propto (\delta V)^2 \]

Having obtained a plot of \( \Delta p \) against \( Q \), the resultant curve was used to construct a plot of \( \Delta p \) against \( Q^2 \). See Figure 4.08. It will be seen that \( \Delta p \) against \( Q^2 \) has resulted in the linear relationship predicted by the theory. B.S. 4062 "The Testing of Oil Hydraulic Control Valves", recommends that the pressure loss should be stated in the form of a dimensionless coefficient \( k^* \) such that

\[ p = k^* \frac{\rho V^2}{2} \]

where in this application is the oil velocity through the mechanism supply port. By use of Figure 4.08 the value of \( k^* \) was found to be 1.2.

4.5.7 Second Prototype

Although the results obtained from the first prototype were in the main excellent, the existence of the tight zone not only detracted from the valve's overall performance, but provided a convenient scapegoat to be blamed for certain of the valve's practical results. It was decided that a second valve casing should be manufactured, this time with the bore internally ground and that certain of the tests should be repeated to assess the real significance of the tight zone.

A new valve casing was manufactured and although the casing was slightly bell mouthed (Table 13) the valve metrology was considerably better than that obtained previously. The bore was ground nominally 0.125 mm smaller than the first casing and the valve bobbin then reground to suit this smaller size. To ensure that the end journals would carry the imbalance loads these were individually ground to produce a running clearance of 0.017 mm at each journal. The pulse
generating section of the bobbin was then relieved a further 0.025 mm.

The valve was assembled and the tests outlined in 4.4.1, 4.4.2, 4.4.3 and 4.4.4 were repeated with the second prototype. The results obtained were as follows:

1) **Breakout Torque:** the new casing eliminated the tight zone completely and the breakout torque was found to be independent of both direction of rotation and angular position of the valve bobbin. The general level of results were consistent with those obtained with the first valve but with less scatter.

2) **Driving Torque:**
   a) Unloaded Valve: The results obtained were consistent with those obtained previously (Section 4.4.2) good linear plots being obtained.
   b) Valve subjected to static pressure and pulsating flow: With the new casing there was no discernible difference between the static and pulsating flow conditions. The general level of driving torques were consistent with those obtained for the valve in an unloaded state. From this it was concluded that the driving torque characteristics experienced with the previous valve casing were caused entirely by the existence of the tight zone. For practical purposes the torque versus speed relationship of the new casing can be taken as identical to that shown in Figure 4.04.

4.6 **Valve Module**

4.6.1 **Concept and Advantages**

It is believed that the testing of the self-balanced universal valve had validated the original design concept. In addition to the specific tests outlined in the previous sections the valve was used for:
i) The development of a mattress sewing machine (Section 5).

ii) The dynamic testing of miniature mechanisms (Section 2).

iii) The development of a chain-stitch sewing machine.

For each of the above applications it was possible to utilise the one standard bobbin and by providing a suitably machined casing, to obtain the desired sequence of motions of the various mechanisms. The use of the valve on these projects, together with design studies of other applications did however indicate certain restrictions in its application. Although the mattress sewing machine is covered fully in the next section, these restrictions can best be illustrated by highlighting two difficulties which were encountered in the development of this rig.

1) To optimise the sewing sequence it was found necessary to make minor alterations in the phase relationship of the mechanisms. This necessitated the machining of additional supply ports adjacent to the originally selected positions either to advance or to retard the initiation of a mechanism in relation to the rotary sewing hook. Owing to the physical size of the pipe fittings used, the closest angular pitch that could be made was 18⁰. In view of the 2:1 drive relationship between the hook and the valve, the minimum phase adjustment in relation to the hook which could be made was therefore 36⁰. Although the performance of the rig was highly satisfactory this was despite, rather than because of, the inability to make small phase angle changes between the various mechanisms.

2) The dominating speed limiting variable of the sewing machine was the magnitude of the viscous pipe losses between the valve and the actuators. To minimise these losses it is desirable that the valve be placed close to the mechanisms it controls in order to reduce the lengths of conduits and the mass of the enclosed fluid. Although on the sewing rig it is possible to reduce the viscous losses considerably, this may not be the case on more complex machinery where individual mechanisms may be widely spaced. Under such circumstances the valve would
require to be placed in a compromise position with some resultant performance downrating.

In addition to these limitations the work on the valve had not demonstrated one of the design concepts principal advantages, namely compactness of design. So the idea evolved for what may be considered to be the complete antithesis of the original universal valve concept - that of a small valve module having a restricted pulse selection facility. What was envisaged was a module incorporating the same design and balancing principles as the larger valve but which would be tailored to suit a particular mechanism. These modules could be either placed adjacent to the mechanism(s) they controlled or even built directly into the main actuator housing (see Figures 5.17 to 5.20). It was not anticipated that this concept would replace the original one but would complement it. The advantages of having such a concept were considered to be:

1) A reduction in the valve's power losses due to its smaller size (Section 3.5.3).

2) A much increased circuit flexibility. For example it would be possible to have modules running at different rotational speeds from one prime mover. This would enable more complex sequencing movements to be obtained (e.g. patterning on textile machinery).

3) The effects of dynamically induced imbalance would be reduced. Each module would contain its own bearings, inlet supply and exhaust and hence spurious pressure fluctuations induced by associated mechanisms would be reduced.

4) Phase adjustments between mechanisms could be made independently by rotating valve bobbins relative to one another.

5) Response time adjustments for individual mechanisms could be made independent of one another. By having individual modules with their own supply pressure inlets, these pressures can be individually adjusted to optimise the performance of a particular mechanism.
6) Flexibility of drive arrangements. The modules could be mounted in either a series or parallel configuration from the prime mover. When used in a series (end to end) configuration the interface seals may be removed and the drag losses reduced even further.

7) Ease of circuit modifications. Individual modules could be removed and modified without disturbing the remainder of the control circuit.

8) Reduction in manufacturing difficulties. As the pulse generating areas on the bobbin would be small, so the accuracy of balancing and manufacture could be relaxed. Similarly the outer casing would be shorter and the bore manufacture made easier.

4.6.2 Design and Manufacture

To prove and test the valve module concept, a unit suitable for sequencing a small double-acting mechanism was designed. This entailed the valve having a pulse selection facility of 90° and 180° of bobbin rotation and a bobbin diameter of 20 mm. The unit was designed using Method 1 (Section 3.4.3) and the various calculations and theoretical predictions are given in Appendix 4.2. The valve casing was manufactured to accommodate 16 mechanism take off ports, the port numbering, phase angles and pulse lengths obtainable from the valve are given in Figure 4.11.

4.6.3 Testing of Valve Module

A test rig was prepared whereby the valve module was driven via a 3 stage reduction gear train by a small variable speed motor. The valve was then used to sequence the double-acting mechanism tested in Section 2. As the larger valve prototype had been subjected to exhaustive tests which had indicated a close correlation between practical results and theoretical predictions, it was considered
Casing development

A~Main hydraulic inlet
B~Exhaust ports
1~16~Mechanism ports

Pressure
Exhaust
Start of pressure pulse

Pulse selection facility

FIG 4.11 VALVE MODULE CASING
unnecessary to repeat some of these on the smaller unit. The principal design criteria of the valve were however tested, these were:

1) The breakout torque at maximum system pressure
2) The valve's internal leakage at maximum system pressure
3) The driving torque versus rotational speed
4) The effects of pulsating flow on the driving torque
5) The valve's internal flow losses.

The results of these tests are given below;

1) Breakout torque at 35 bar: The valve module exhibited a similar tight zone as a result of manufacturing errors to the first universal valve (4.5.1). Throughout this zone the valve bobbin again was not being supported by the end journals and thus the torque levels encountered over this zone were not considered to be representative of the valve's true characteristics. All torque measurements were therefore taken away from this zone and results were obtained using a simple moment arm technique.

The moment arm consisted of a short length of 5 mm square bar fitted into a driving slot in the valve bobbin. By measuring the amount that this bar was moved off-centre so an estimate of the torque required to overcome mechanical friction was obtained. With the main hydraulic supply disconnected and without lip seals the mechanical stiction torque was estimated to be 0.008 Nm. When pressure was applied to the valve the breakout torque could not be measured as the bobbin appeared to be almost perfectly balanced. It was concluded therefore that the breakout torque due to hydrostatic imbalance was negligible and that the breakout torque of the valve could be viewed as being entirely that necessary to overcome the friction induced by the particular lip-seals used.

2) Valve internal leakage at 35 bar: With the valve bobbin stationary and at an oil temperature of 25°C the power loss due to internal leakage was found to be 9 watts. This figure was
slightly lower than that predicted by the theoretical analysis.

3) **Driving torque versus rotational speed:** The power consumed by the electric motor was measured with and without the bobbin connected. The results obtained are shown plotted for comparative purposes on Figure 4.04. The results indicated that, for example, the power required to drive the valve at 800 r.p.m. was 9 watts. This power was not all consumed in overcoming the viscous drag losses for included in this figure was the driving gear resistance (3 stage reduction from prime mover). In view of the extremely low power levels it was considered unnecessary to attempt to subdivide between the gear losses and valve losses.

4) **The effect of pulsating flow on the driving torque:** The valve was connected to the chain-stitch sewing rig to obtain the sequence of operations shown in Figure 4.01. The supply pressure to the valve was varied between 0-40 bar. At a particular speed there was no discernible change in the power consumed by the prime mover regardless of the applied supply pressure. It was again concluded that the valve's driving torque was independent of the main supply pressure.
SECTION FIVE

A HYDRAULIC INTERLOCK SEWING MACHINE

5.1 Introduction

Giltspur Precision Products Limited, a company engaged in the business of producing mattress cover sewing machines, sponsored an investigation into the feasibility of a hydraulically powered interlock sewing head. It was this company's opinion that, provided such a unit could be designed and incorporated into their machines, certain specific advantages, not possible using a conventional mechanical machine, would be realised. This sponsorship had resulted through earlier work carried out by J D Garside who had demonstrated the feasibility of producing an interlock stitch by using miniature hydraulic actuators sequenced by a rotary valve.

The sewing head produced by Garside was based on a small domestc sewing machine and used an actuating concept similar to that shown in Figure 1.01. As a result of the performance limitations of this actuator design, the sewing head was considered to be unsuitable for uprating to meet the physically larger and more arduous duty of sewing mattress covers. Investigations were therefore held in abeyance until an actuator capable of meeting the required duty, had been designed. When the initial qualitative assessment of the actuating and rotary valve concepts outlined in Sections 1 and 3 were encouraging, it was decided that the mattress cover sewing head would provide a suitable and realistic vehicle by which to fully assess them. Work proceeded therefore on the design, building and testing of a hydraulically powered sewing head using these valve and actuator concepts. Although reference is made throughout this section to stitch formation and the success of the project from the sponsor's viewpoint was judged by the accuracy and consistency of stitch formation, the sewing aspects have been minimised. The reasons for this are twofold. The first reason is that the overall machine concept and the detailed problems encountered on stitch formation will be of interest to only a minority. These aspects have been the subject of a paper published by Mr T P Priestley and the author and interested parties are referred to reference 5.
The second reason is a question of emphasis, to the author, the project was a hydraulic test rig to validate the theory and equipment which had been developed. Success from the author's viewpoint was therefore to be judged by:

a) The correlation between theoretically predicted and measured system performance.
b) The flexibility, adaptability and durability of the system elements.

Although the design problems posed in this project and the solutions presented are related to the specific objective of achieving high speed stitch formation, the system is representative of the many applications of rotary valves and actuators which can be envisaged. It was believed that if the above features could be demonstrated and confirmed, then the mattress cover sewing machine would represent a worked example of one application of hydraulic mechanisms to high speed manipulative machinery.

5.2 Analysis of Stitch Formation by Mechanical Sewing Machine

5.2.1 The Stitch

The stitch to be formed for mattress sewing is designated as type 301 in B.S. 3870, 1965. The stitch is more commonly known as "lockstitch" and is used extensively in industrial and domestic sewing machines, see Figure 5.01. The stitch is formed with two threads; one needle thread 'A' and one bobbin thread 'B'. A loop of thread 'A' passes through the material to be stitched and is interlaced with thread 'B'. Thread 'A' is subsequently passed back so that the interlacing comes midway between the surfaces of the material, or materials being sewn.

5.2.2 Mechanical Sewing Machines

The essential elements of a typical lockstitch sewing machine are as follows:

i) a needle mechanism
FIG 5.01 INTERLOCK STITCH
ii) a needle thread control mechanism

iii) a material feed mechanism

iv) a material stabilising mechanism

v) a bobbin thread mechanism

The heart of a modern machine is the bobbin assembly and to form a correctly set stitch, the other mechanisms must maintain certain definite instantaneous position relationships with the bobbin mechanism. In current machines, the bobbin thread is carried in a rotary hook which requires two revolutions for each stitch formed, one revolution to handle the needle thread and one revolution while the machine is feeding.

To obtain a more complete understanding of the problems to be overcome in utilising hydraulic actuators for stitch formation, the mechanical sewing machine utilised by the sponsor was analysed. The displacement profiles of the mechanisms in relation to the bobbin hook can be seen in Figure 5.02. To differentiate between essential and non-essential positional relationships with the bobbin mechanism, the problems encountered on stitch formation using mechanical machines were also investigated.

5.3 Functional Requirements of Sewing Machine Elements

Certain constraints on the phasing and amplitude requirements of the machine elements in relation to the bobbin hook position are necessary for correct stitch formation. As a result of the investigation into stitch formation problems, these were considered to be as follows.

5.3.1 The Needle Mechanism

The overall amplitude of the needle displacement is controlled by:

a) The height that the needle must retract clear from the working surface in order to allow free movement of the sewing head, on the material being stitched.
FIG 5.02 DISPLACEMENT PROFILES OF MECHANICAL SEWING MACHINE ELEMENTS
b) The depth that the needle must penetrate through the material. This is controlled by the bobbin position relative to the working surface. The most important phase relationship between the needle and bobbin hook is at $\theta = 360^\circ$. At this point, the needle must have descended to full depth and started to retract, in doing so a loop of needle thread is formed at the back of the needle. This loop will be caught by the rotating bobbin hook at the initiation of a new stitch being formed. The size of this loop is of importance for stitching efficiency.

c) Insufficient needle retraction will result in a small loop being formed which could cause the bobbin hook to miss a stitch.

d) Too much needle retraction will cause a large loop to be formed which could collapse, or twist, out of the path of the bobbin hook and again cause a missed stitch.

5.3.2 Needle Thread Control Mechanism

The amplitude required is a function of the bobbin design. Once the needle thread loop has been picked up by the bobbin hook, there is a rapid demand for thread as the needle loop is expanded and passed over the bobbin. The following points were considered essential for correct functioning.

a) As the needle descends, slack must be introduced to the needle thread otherwise thread will be drawn prematurely from the needle thread supply.

b) The rate of introducing slack must be in advance of demand. On a mechanical sewing machine, the drawing back of the needle thread commences when the bobbin hook is at $\theta = 540^\circ$. However, the rate that the needle thread is withdrawn is also important.

c) If the needle thread is drawn up too quickly, it will cause it to catch on the bobbin casing.
d) If the needle thread is drawn up too slowly, the bobbin hook can catch the thread a second time; this is known as double-looping.

5.3.3 Material Stabilising Mechanism

This is commonly known as the "presser foot" and its function is to stabilise and support the material during needle penetration. The constraints on this mechanism are:

a) The amplitude of movement required is dependent upon the thickness of the material to be stitched. In the particular case of mattress covers this is a composite consisting typically of cover fabric, sponge rubber, horse hair and a backing fabric.

b) The presser foot must be down whilst the needle is in the material.

c) The presser foot must not compress the mattress cover too severely as to do so will cause excessive needle heating and resistance to motion.

5.4 Hydraulic Machine Concept

5.4.1 Actuation and Control: Basic Idea

The design concept of the hydraulic sewing machine was to actuate hydraulically, the needle mechanism, the needle control mechanism and the presser foot, whilst retaining a proven, mechanically driven, rotary hook assembly. The sewing head was to form part of an automated mattress sewing machine and hence the material feed mechanism was not required.

The principal hydraulic control problem was to establish and maintain the essential phase relationships between the mechanically driven rotating bobbin assembly, in particular conditions 5.3.1(c); 5.3.1(d); 5.3.2(c) and 5.3.2(d) stated above, needed to be met over
the entire speed range of the machine. This would be from 10 to 1000 stitches per minute.

A hydraulics systems designer has a variety of techniques available for solving such a problem, e.g. closed or open loop systems, pressure compensated systems, automatic feedback systems, automatic advance and retard mechanisms. Each of these systems has certain advantages and disadvantages, in terms of cost, complexity and performance. The decision was taken to make the power source, actuating devices and control method as simple as possible, e.g. to utilise a small gear pump for the main oil supply and to obtain the necessary sequence control by simple pressure and exhaust switching using the concept of valve outlined in Section 3. Such a decision was based on the need for a low cost mechanism, requiring comparatively low precision manufacturing techniques.

5.4.2 Idealised Hydraulic Circuit

The decision to use a simple hydraulic circuit posed a problem in establishing the necessary phase relationships between the hydraulic mechanisms and the mechanically driven bobbin, over the range of 10-1000 stitches per minute. For a control circuit of the type outlined in Section 1.5.3 each mechanism will have a constant displacement versus time profile for fixed system parameters. Thus, the use of simple reciprocating movements was not considered to be feasible. A solution to the problem was possible if the movements of the actuators could be made in discrete steps, during the critical parts of the cycle by use of compound actuators. The investigations into sewing problems using mechanical machines indicated that only the upward movement of the needle mechanism and the thread take-up mechanism needed to be discretised. Further, that although the needle mechanism required a controlled step height the needle thread control mechanism could be an uncontrolled step movement. Figure 5.03 shows a comparison between the proposed hydraulic and mechanical sewing machine displacement profiles.
NEEDLE YARN TAKE-UP MECHANISM

FIG 5.03 COMPARATIVE DISPLACEMENT PROFILES OF MECHANICAL AND HYDRAULIC SEWING MACHINE
5.5 Design of the Prototype Sewing Machine

5.5.1 General Considerations

In order to minimise the financial cost of the development programme, use of certain existing hydraulic equipment was necessary. This inevitably led to some equipment mismatching on the test rig and imposed limitations on the maximum performance obtainable. However, this was considered acceptable since one of the primary aims of the investigations was to determine the possible accuracy of hydraulic mechanism performance predictions. If good correlation between theoretically predicted and measured performance could be achieved, then performance uprating would be a comparatively straightforward design and development procedure. The bobbin assembly and other standard parts from a Singer Model 133K18 interlock sewing machine were incorporated into the hydraulic sewing machine. The standard equipment which was used in the prototype machine is given in Appendix 5.1.

5.5.2 The Needle Mechanism

This mechanism required a movement amplitude of 50 mm with a displacement step of 3 mm on retraction, necessary for correct loop formation. In terms of the required thrust levels, tests on mattress covers indicated that a force of approximately 70N was required to drive the needle through the material when the presser foot was stabilising the material. Design calculations based on the maximum operating pressure of 50 bar obtainable from the power pack, the desired maximum operating speed of the sewing head and the thrust required for needle penetration, indicated that an actuator diameter of 8 mm would be suitable. Utilising this size of actuator would necessitate a pressure of 15 bar for the needle to penetrate the material and leave 35 bar for accelerating the mechanism.

The design of needle mechanism which was envisaged can be seen in Figure 5.04. This figure shows the needle located and clamped to a needle bar which is a free running fit in a support bearing. The needle bar is connected via a double knuckle joint to a square
FIG 5.04 NEEDLE MECHANISM

sealing plug
needle slide
main actuator

actuator housing

supply ports

enlarged view of auxiliary piston

needle bar

needle clamp and thread eyelet

needle
slide which can be reciprocated by two opposed single-acting actuators. Also shown in Figure 5.04 is an enlargement of a modified design of auxiliary piston which was used. Functionally it was envisaged that the new design would operate in a similar manner to the original piston concept outlined in Section 1.4.

The original concept was for an auxiliary piston having a larger diameter than the main actuator it controlled with the oil feed to it being direct to the rear of the piston. For the new design, the auxiliary piston was the same diameter as the main actuator. The oil supply to the rear of the auxiliary piston was achieved via an annular recess and small radial drillings into an axial hole drilled from the rear of the piston. For the new design it was envisaged that the auxiliary piston would act as its own spool valve and its movement subsequently arrested as the rear land closed the supply port.

5.5.3 Needle Yarn Take-Up Mechanism

This mechanism required a total movement of 75 mm with an uncontrolled step on the return movement. The mechanical sewing machine take-up lever has an amplitude of 95 mm however, by having a modified needle displacement profile this figure was considered unnecessarily large and that it could be reduced to 75 mm. It was decided that it would be advantageous to obtain this amplitude through a simple lever which would enable a shorter stroke actuator to be utilised. The mechanism which was envisaged can be seen in Figure 5.05.

The mechanism consisted of two opposed single-acting modules operating on a separate sliding member. A pivot pin on the sliding member acted as the fulcrum for a thread take-up lever which was connected at one end to the swinging link arm. The other end of the lever was guided in a slot and had an eye through which the needle thread was passed.

The resistance to movement of the mechanism was restricted to the extreme upward/return movement of the lever, when a stitch was being set and yarn was drawn from the supply cone. Tests indicated that this would represent an actuator force of 20N and that a 5 mm
FIG 5.05 NEEDLE THREAD TAKE-UP MECHANISM
diameter actuator would be compatible with the requirements of the mechanism.

5.5.4 The Presser Foot Mechanism

The presser foot mechanism required a displacement amplitude of 13 mm. The resistance to motion of the mechanism was limited to the compression of the mattress cover material. No precise value could be placed on this resistance since it was dependent upon the shape and size of the presser foot and the particular characteristics of the material to be stitched.

The decision was taken to operate this mechanism using a single-acting spring return mechanism. It was felt that this mechanism afforded an opportunity to verify the theoretical analysis of this actuation technique, on a relatively uncritical machine element. The mechanism which was envisaged can be seen in Figure 5.06. The presser foot from a mechanical sewing machine was attached to a guide bar and prevented from rotating by a guide and guide pin. The guide bar ran in a support bush and was spring loaded in the return (off) direction. The forward movement of the mechanism was obtained by a 10 mm diameter single-acting actuator.

5.5.5 General Construction of Sewing Rig

The needle mechanism, presser foot and thread take-up control mechanism were contained in a split housing on which were mounted the necessary thread tensioning devices, eyelets and the needle yarn supply cone. See Figures 5.07 and 5.08. The leakage from the various mechanisms was collected in the bottom of the housing and gravity fed back to the supply tank. No attempt was made to control the leakage past the two mechanism support bushes, the assessment of any necessary sealing devices was to be made later.

The sewing head was supported by a tubular framework to simulate the large throat depths used on mattress sewing machines. The framework also carried a working surface for conducting manual sewing tests (see Figure 5.09). The support frame was bolted to a base plate
FIG 5.06 PRESSER FOOT MECHANISM
Fig 5.07 - Inside the Hydraulic Sewing Head Needle & Presser Foot Mechanisms
Fig 5.08
Inside the Hydraulic Sewing Head - Needle Yarn Take-Up Mechanism
Fig 5.09
The Sewing Machine - Front View
Fig 5.10
The Sewing Machine
Rear View
onto which was also mounted a pulse generating valve. The mechanical sewing bobbin was mounted on to a drive shaft assembly located from, and running parallel to, the rotary valve. The sewing bobbin drive shaft was coupled to the rotary valve by a 2:1 positive drive and the drive to the rotary valve achieved via a flexible coupling from a variable speed hydraulic pump/motor set, see Figure 5.10.

5.5.6 Theoretical Machine Performance

Estimates were made of the individual mechanism response times by the theoretical approach outlined in Section 1.5 and using data obtained from both the prototype test rig and the detail design drawings (Appendix 5.2). This analysis indicated that the response time for needle penetration movement controlled the maximum operating speed obtainable from the sewing head. The maximum cycling rate of the rig, for the stipulated system parameters, was estimated to be 450 stitches/min. At sewing speeds greater than this figure, the analysis indicated that the needle movement would attenuate. The theoretical displacement profiles of the sewing head mechanisms at the maximum and minimum stitching rates can be seen in Figure 5.11.

A flow demand curve of the sewing head at 450 stitches/min was constructed using the equations derived in Appendix 5.2, this can be seen in Figure 5.12. The peak demand, when the presser foot mechanism had an unopposed downward movement, was found to be approximately 15 litres/min. When the sewing head stitched mattress covers and the motion of the presser foot was opposed by the material, it was estimated that the peak demand would be reduced to approximately 9 litres/min. The average flow demand of the rig over the operating cycle at the maximum stitching rate of 450 per minute on mattress covers was found to be approximately 3.2 litres/min. This figure represented at the maximum supply pressure of 50 bar, an average power consumption of 270 watts.
FIG 5.11 DISPLACEMENT PROFILES OF HYDRAULIC SEWING MACHINE ELEMENTS AT 450 S.P.M. (SUPPLY PRESSURE = 50 bar)

- yarn take-up
- needle
- presser foot
- position of bobbin hook
FIG 5.12 FLOW DEMAND OF SEWING MACHINE AT 450 s.p.m (pressure = 50 bar)

- profile with unopposed presser foot motion
- modified profile with presser foot operating against mattress covers

average flow 3.16 l/min
5.6 Sewing Rig Development

5.6.1 Initial Tests and Modifications

The sewing rig was assembled and the rotary valve piped, with the exception of the hydraulic accumulator as indicated in Figure 5.13. The sewing bobbin and the rotary valve were rotated by hand and the movements of the individual actuators were checked to confirm the correct sequence.

It became apparent that the new design of auxiliary piston fitted to the needle actuators did not function as had been anticipated. It transpired that the rear land of the piston failed effectively to shut off the supply of oil from the supply port to the rear of the piston. This resulted in the auxiliary piston inching forward until it came under the influence of the supply port to the main actuator, whereupon it proceeded to reciprocate as a loose addendum to the main actuator. The needle actuators were therefore modified by counter-boring the housing to accept a 9 mm diameter auxiliary piston and the rig re-assembled.

The modification was successful and when the correct sequence of operations was confirmed, stitching tests were conducted by rotating the valve by hand. Following confirmation that the hydraulic mechanism correctly formed stitches, the hydrostatic drive was connected to the rotary valve and the stitching rate was increased to approximately 100 per minute. The stitches were still being correctly formed but yarn tensioning was unsatisfactory and stitching runs were limited to about 50 stitches before random yarn breakage occurred.

a) Incorrectly tensioned stitches were traced to the incorrect sequencing of the take-up arm and particularly discernible at very slow speeds. Inspection of the original hydraulic sequence showed that a delay of 90° of bobbin rotation existed between the needle actuation and introduction of slack to the needle thread. This resulted in the needle drawing thread from the system prematurely, instead of the take-up arm later in the cycle. To obviate this fault, the sequence was altered so that the slack
FIG 5,13 SCHEMATIC CIRCUIT AND RIG DATA OF PROTOTYPE INTERLOCK SEWING MACHINE

<table>
<thead>
<tr>
<th>port</th>
<th>function</th>
<th>$D_{(mm)}$</th>
<th>$L_{(m)}$</th>
<th>$N$</th>
<th>$\Theta^{\circ}$</th>
<th>$\phi$^{\circ}</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>presser foot advance</td>
<td>4.37</td>
<td>1.587</td>
<td>6</td>
<td>190</td>
<td>90</td>
</tr>
<tr>
<td>2</td>
<td>needle advance</td>
<td>1.943</td>
<td>1.994</td>
<td>9</td>
<td>60</td>
<td>90</td>
</tr>
<tr>
<td>3</td>
<td>needle advance</td>
<td>1.994</td>
<td>1.994</td>
<td>11</td>
<td>120</td>
<td>90</td>
</tr>
<tr>
<td>4</td>
<td>needle thread loop</td>
<td>1.511</td>
<td>1.511</td>
<td>11</td>
<td>70</td>
<td>330</td>
</tr>
<tr>
<td>5</td>
<td>needle retract</td>
<td>1.765</td>
<td>1.765</td>
<td>8</td>
<td>210</td>
<td>390</td>
</tr>
<tr>
<td>6</td>
<td>complete take-up</td>
<td>3.05</td>
<td>1.791</td>
<td>10</td>
<td>90</td>
<td>630</td>
</tr>
<tr>
<td>7</td>
<td>commence take-up</td>
<td>1.752</td>
<td>1.752</td>
<td>9</td>
<td>70</td>
<td>570</td>
</tr>
<tr>
<td>8</td>
<td>introduce yarn slack</td>
<td>1.943</td>
<td>1.943</td>
<td>11</td>
<td>90</td>
<td>90</td>
</tr>
<tr>
<td>9</td>
<td>introduce yarn slack</td>
<td>1.841</td>
<td>1.841</td>
<td>7</td>
<td>90</td>
<td>90</td>
</tr>
</tbody>
</table>
yarn condition was initiated earlier. Alteration of the phasing of the pulse used to actuate the yarn take-up was achieved by drilling a new mechanism supply port in the valve casing and re-piping.

b) Random yarn breakage: when the machine was "chaining" i.e. producing the correct stitch formation, but without fabric, snatching of the thread could be felt every cycle, until eventually this was so severe as to cause yarn breakage. This was attributed to the incompatible motion between the thread take-up mechanism and the sewing bobbin.

The original take-up sequence was designed as a compound movement, initiated when the sewing bobbin was at 540° and completed at 600°. Cycling the rig slowly showed that the final thread take-up was occurring too early in the cycle. This mechanism was converted to a simple reciprocating movement by coupling the main actuator and auxiliary piston ports together (ports 6 and 7) and then the whole hydraulic sequence advanced so that these ports were pressurised when the sewing bobbin was at 630°.

5.6.2 High Speed Stitching Tests

The design modifications referred to in 5.6.1 resulted in an immediate improvement in the quality and speed of stitch formation. The machine produced good quality stitching up to a maximum of 300 per minute. Attempts to increase the stitching rate beyond this figure resulted in:

i) failure of the sewing bobbin to pick up the needle loop

ii) random needle yarn breakage.

The machine, however, was observed to have an ideal cruising speed of 250 stitches/min at the maximum system pressure of 50 bar. This figure was substantially below the theoretical design prediction of 450 per minute.

A series of tests were conducted to find the maximum operating speed of the rig, based on the attenuation speed of the needle mechanism, so that these could be compared with the theoretical predic-
tions. This is shown in Figure 5.14. Two high speed films were taken of the rig operating at 180 and 300 stitches per minute to enable a study to be made of the actuator motions during stitch formation. These films were taken at 1000 frames per second with the machine sewing thick cardboard. The thrust for the needle to penetrate thick cardboard was minimal compared with the 15 bar required on the mattress cover material. The sewing rig was therefore run at a supply pressure of 35 bar to compensate for the disparity.

Reference to Figure 5.14 shows that the practical performance (without hydraulic accumulator) fell substantially below that predicted theoretically and that the needle displacement was attenuating, when cycling at 300 stitches/min at 35 bar. This was substantiated by the high speed film, which also showed how fortuitous stitch formation was at 300 stitches/min. Needle attenuation occurred at the designed loop pick up height, but no loop was formed. However, due to the scallop at the rear of the needle, the bobbin hook successfully picked up the needle yarn and a stitch was formed. The failure of the bobbin to pick up the needle thread on occasions was therefore attributed to the needle attenuation. Random needle yarn breakage was also attributed to the needle attenuation, as no thread loop was formed, the sewing bobbin hook would on occasions pierce the yarn and cause the yarn to break.

The analysis of the high speed film also indicated:

a) That at a stitching rate of 180 per minute, when needle attenuation did not occur, that the needle mechanism exhibited a slight overshoot at the loop step height before settling to its nominally designed height of 3 mm. This was attributed to the kinetic energy attained by the mechanism during this movement which was subsequently dissipated and returned by the gravitational forces.

b) That an overlap of pressure pulses occurred between the needle descent and the loop formation step. The needle descent had been designed for operation with a 120° pulse length from the rotary valve and timed to finish coincident with the start of the needle loop pulse. However, a subsequent check showed that
FIG 5.14 PRESSURE-SPEED CHARACTERISTICS OF SEWING MACHINE
the needle was receiving a pulse of 128° duration and was the cause of the overlap. This overlap resulted in a delay of 8° of valve rotation in the initiation of needle loop formation.

c) The only movement whose response time correlated with the theoretically predicted performance was that of the presser foot return. The theoretically predicted response for this movement was 30 milliseconds and the measured performance was found to be 28 milliseconds.

5.6.3 Effects of Transient Supply Pressure Fluctuations

A series of pressure traces were taken at a rotary valve control port to establish the cause of the disparity between the predicted and the measured results. A relevant pressure trace is shown in Figure 5.15.

The theory outlined in Section 1.5 assumed that square wave pressure pulses of a magnitude equal to the supply pressure are applied to the actuator. However, reference to Figure 5.15 shows that several pressure drops occurred over the periods of the actuator flow demand. By extrapolation from the pressure traces it was possible to detect the cause of the pressure drops. It was established that these drops occurred as various movements were initiated throughout the sewing cycle, e.g. needle descent etc. The duration of these pressure drops varied between 10 and 50 milliseconds.

It was evident that the pump/relief valve combination was unable to meet the very high instantaneous flow demands during the very fast actuating parts of the displacement-time cycle. Thus, to maintain a sharp pressure pulse of the necessary amplitude additional flow was required at peak demand periods. Hence, a bag type accumulator was incorporated in the main supply line from the pump outlet to rotary valve inlet. Supply pressure traces for the power pack running both without and with the accumulator were taken and are shown in Figure 5.16. With the accumulator, a supply pressure having only small perturbations about the nominal supply pressure was obtained, at the point of peak demand, i.e. when the motions of all three mechanisms were operated simultaneously. Observations of needle motion
supply pressure (bar)

valve exhaust pressure

valve rotating at 60 r.p.m.

FIG 5.15  130° PRESSURE PULSE WITHOUT HYDRAULIC ACCUMULATOR
supply pressure (bar)

with accumulator

without accumulator

FIG 5.16 COMPARATIVE VALVE INLET PRESSURES OF SEWING MACHINE WITH AND WITHOUT AN HYDRAULIC ACCUMULATOR IN CIRCUIT
attenuation were taken at different speeds and supply pressures, to verify improved performance. Inherent accumulator characteristics limited the range of operating pressure between 20 and 40 bar. This performance is shown on the same graph as the previous results in Figure 5.13 and resulted in close agreement between predicted and practical results. Further sewing tests were then carried out with the accumulator in circuit, to determine any further improvement in stitching rates. However, these were only marginal, a rate of 380 stitches/min was reached before double looping occurred, this was attributed to attenuation of the take-up mechanism.

The start of the take-up mechanism motion had been retarded, to overcome the low speed characteristics. However, this limited the total time available for the take-up mechanism operating to 90° of the sewing bobbin rotation, one-eighth of the cycle period. This corresponded to initiation of the take-up mechanism with the bobbin at 630° and the loop drawn clear of the bobbin at 720°. Analysis of the high speed film had indicated the likelihood of this problem, since at a cycling rate of 300 stitches/min the loop was cleared at a bobbin angle of 710°, i.e. with only 10° of the cycle remaining. The bobbin and timing were altered relative to the complete hydraulic sequences, so that the take-up and all other motions were initiated earlier, with the bobbin at 610°. This resulted in a further marginal improvement enabling 400 stitches/min to be achieved, with occasional runs of 430 stitches/min, at higher oil temperatures with reduced viscous damping. This method of retiming was not regarded as satisfactory since secondary problems with the needle movement could occasionally result in failure to pick up the needle loop.

5.6.4 Leakage and Case Drainage

The major source of oil leakage from the mechanism case was through the take-up lever slot. This was due to oil from the take-up actuator running along the oscillating arm and out of the casing. An oil splash guard, clamped to the take-up arm, was designed to throw oil from the arm. This modification substantially reduced the amount of oil allowed to escape but was not completely effective. Owing to
space limitations imposed by the original detail design it was considered impracticable to introduce additional design modifications.

The two other potential sources of oil leakage, i.e. past the support bushes of the needle and presser foot mechanisms proved to be insignificant. Some minor leakage would occur over the duration of a prolonged stitching trial but the fitting of additional sealing devices was considered to be unnecessary on the prototype rig.

Gravity drainage of leakage oil from the actuators was found to be inadequate when the rig was cycling at maximum speed and was attributed to the fan effect of the reciprocating lever, causing a slight pressure depression in the casing. Slight pressurisation of the casing was incorporated to overcome the fan effect, this modification, in conjunction with an increase in the diameter of the drain line, proved to be effective.

5.6.5 Final Stitching Tests

To enable further investigation at higher stitching speeds, the hydraulic sequence was modified so that the thread take-up motion was a compound movement initiated as shown in Figure 5.11. The actuator pistons were ground to the geometry required, to achieve correct cushioning at a design pressure of 35 bar. All previous tests were carried out with the pistons slightly undercushioned, to ensure full movement at the expense of slight impact noise.

At the maximum operating pressure of 50 bar, the prototype rig mechanism stitched consistently, on the combination mattress cover materials, at speeds up to 450 stitches/min. This maximum speed correlated well with the maximum dynamic performance figure, predicted in Appendix 5.2, for the specific actuators, system pressure and hydraulic circuit used. The practical sewing performance achieved was specific only to the set of system parameters of this rig and did not represent the maximum stitching capability attainable using the basic actuators. For example, on less difficult material, like thick corrugated cardboard, the maximum stitching rate for the same system pressure was about 580 stitches/min. The figure of 580 stitches/min again correlated well with the maximum theoretical running speed of the needle mechanism with an unopposed motion.
5.7 Uprating Sewing Machine Performance

5.7.1 General Considerations

The maximum stitching speed of the hydraulic sewing head was substantially below the capability of the mechanical machine used in the production of mattress covers. To realise the advantages envisaged by the sponsor there was a fundamental requirement that the hydraulic machine should be capable of operating at maximum speeds approaching those obtained by conventional mechanical machines. The maximum stitching rate for this type of machine is 1000 per minute, although for much of the duty cycle, the running speed is in the range 600 to 900 stitches per minute.

Having obtained a good correlation between the theoretical machine performance and that obtained in practice, the uprating of the sewing head was considered to be a straightforward design and development exercise. The specific information obtained during stitching trials relating to the formation of stitches using hydraulic mechanisms, together with the detailed knowledge of the sewing rig performance, indicated that an improved design capable of exceeding 1000 stitches per minute was feasible. However, owing to financial limitations, major design and structural changes were not possible and recommendations for the uprating of the machine performance were limited to a design exercise only.

5.7.2 Modifications to Prototype Rig

i) Reduced viscous losses:

The major factor which limited the maximum stitching rate of the prototype machine resulted through the mismatching of the conduit bore sizes in relation to the actuator diameters (the d:D ratio). This resulted in unacceptably high viscous damping and a consequential loss in machine performance. If the conduit bore sizes were larger, an increase in machine performance to a level comparable to conventional machines was predicted (Appendix 5.3). It was calculated that the stitching rate of the machine when sewing mattress covers would increase from 450 to 880 stitches per minute by the incorporation of the following modifications:
a) Increasing conduits with 4.37 mm bore to 6.25 mm bore.
b) Increasing conduits with 3.00 mm bore to 4.37 mm bore.
c) Enlarging main inlet and exhaust bores to/from rotary valve to 12.5 mm.

ii) Higher operating temperature:

To minimise the warming up period of the sewing machine, stitching trials were generally carried out with an oil temperature between 20°C and 30°C. For the particular hydraulic oil used, the viscosity of the oil at 25°C is 0.05 N sec/m². However for a production machine an operating temperature of 45°C was considered acceptable. The oil viscosity at 45°C is 0.025 N sec/m² and would result in a higher machine performance.

Theoretical predictions based on this lower value of oil viscosity indicated that maximum operating speed of the prototype machine would increase from 450 to 525 stitches per minute. Similarly for a machine having the increased conduit sizes outlined above, the corresponding speed increase predicted would be from 880 to 930 stitches per minute.

iii) Modified hydraulic sequence

Consideration of the hydraulic switching sequence and the displacement profiles shown in Figure 5.11 indicated that a dwell period occurred between the completion of the needle yarn take-up movement and the initiation of the needle descent. Experience gained through stitching trials showed that this dwell was unnecessary and that the initiation of the needle descent could be advanced. This would allow a greater proportion of the overall machine cycle time to be allocated for the needle descent, i.e. from 120° to 160° of valve rotation.

If the pulse length allowed for the needle descent were increased to 160° of valve rotation, the maximum operating speed of the uprated rig would increase from 880 to 1175 stitches per minute at an oil temperature of 25°C. At the higher oil temperature of 45°C the theoretical machine performance would be 1240 stitches per minute. The
theoretical machine performance given in Appendix 5.3 indicated that the response times of the needle yarn and presser foot mechanism would be compatible at an elevated performance.

5.7.3 Analytical Design Based on Design Procedure

The design procedure outlined in Section 2 was aimed at providing the correct sizes of actuators and their conduits to be used in the design of a particular mechanism. This procedure was not developed until after the detailed testing of actuators and the mattress sewing machine had been completed. This procedure was subsequently used to check the initial selection of actuator sizes used on the sewing rig. A comparison between the rig changes recommended in 5.7.2 and those indicated by the design procedure is given below:

<table>
<thead>
<tr>
<th>MECHANISM</th>
<th>PROTOTYPE</th>
<th>DESIGN PROCEDURE</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>d(mm)</td>
<td>D(mm)</td>
</tr>
<tr>
<td>Needle Mechanism</td>
<td>8</td>
<td>6.25</td>
</tr>
<tr>
<td>Thread Mechanism</td>
<td>5</td>
<td>4.37</td>
</tr>
<tr>
<td>Presser Foot</td>
<td>10</td>
<td>6.25</td>
</tr>
<tr>
<td>Average Power Consumption</td>
<td>580 watts</td>
<td>420 watts</td>
</tr>
</tbody>
</table>

The design selection did not assume any information relating to the sewing rig which was obtained subsequent to the design stage. This was deliberate so that the validity of the procedure could be checked against a known mechanism. To achieve the target stitching rate of 1000 per minute, necessitates that the needle advance should be accomplished in 26.7 milliseconds. The theoretical needle performance based on the final measured mechanism data and the system parameters derived by the design procedure, was calculated to be 26.0 milliseconds.
5.8 Candlewick Tufting Machine Mechanisms

Although financial limitations prevented any major structural changes being made to the sewing machine, additional sponsorship was obtained to design a prototype candlewick tufting machine. Such a machine has similar mechanisms to a chain-stitch sewing machine, i.e., a needle and looper mechanism, but does not require a separate yarn take-up mechanism. In terms of the required machine duty, this had to be significantly higher than that required for the sewing machine. The prototype had to be capable of attaining running speeds of 2000 stitches per minute with a needle amplitude of 50 mm, a duty cycle some four times that obtained on the sewing machine.

The project presented an ideal opportunity of incorporating the actuating and control concepts which had been developed subsequent to the sewing machine project. Also, unlike previous work, the requirement that a specified performance figure needed to be obtained could form a realistic test of the effectiveness of design techniques which had been derived. The project was started some time after the Author ceased full-time research, thus personal involvement was limited to the design and analysis of the hydraulic mechanisms only. Furthermore, the design work ran concurrently with the preparation of this thesis. At the time of writing, only the manufacture of the hydraulic mechanisms and limited qualitative performance assessments have been made. The detailed testing will take place when the mechanisms can be correctly embodied into the overall machine framework. However, such tests that have been carried out confirm the design principles and indicate that the mechanisms will achieve the required duty at the design operating pressure.

5.8.1 Mechanism Designs

Following verification of the design concept of the double-acting cylinder shown in Figure 2.14, it was decided that a through piston rod version would be ideal for the needle mechanism. Although the tufting machine does not require a separate yarn take-up mechanism, yarn feed is achieved on mechanical machines by an extension to the needle bar. Thus, one end of the double rod actuator could be connected
to the needle bar and the other end utilised for the yarn feed. The use of opposed single-acting modules was however retained for the looper mechanism.

In order to achieve the required duty of 2000 stithes per minute, it was decided that the needle and looper mechanisms should be sequenced by individual rotary valve modules embodied adjacent to their respective actuators. This design approach was considered to have the advantage of minimum circuit impedances to each mechanism plus the ability of phase adjustment between the mechanisms (see Section 4.6). The mechanism designs together with their associated components can be seen in Figures 5.17 to 5.20.

5.8.2 Qualitative Tests

It had been anticipated that of the two hydraulic mechanisms, the looper design, by utilising both well proven techniques, would present fewer problems than the needle mechanism. However, the performance of the two mechanisms under test showed the needle actuator to be superior, in having a quiet well cushioned response. At the outset the looper mechanism suffered from inconsistent cushioning in one direction of travel. Subsequent checks revealed that this was caused by incorrect geometry of the auxiliary piston.

Suitable modifications were put in hand so that the auxiliary piston complied with the design constraint now explicitly outlined in Section 2.5.1. On the other hand the large double-acting cylinder used on the needle mechanism functioned well from the outset.
Fig 5.17
Tufting Machine
Needle Mechanism
Fig 5.19
General view of Looper Mechanism
Fig 5.20
Looper Mechanism Components
Previous research at the University of Technology has indicated that miniature linear hydraulic actuators sequenced by rotary spool valves can be used to replace solid member mechanisms on high speed reciprocating mechanisms. This was demonstrated by the design and development of a hydraulically powered circular weft knitting machine and a small interlock sewing machine. It was considered that this basic actuator/valve concept had considerable and perhaps better potential in the more general field of high speed manipulating machinery rather than the specific textile examples. However, the actuator and valves which had been developed for the specific applications were known to have performance limitations which needed to be overcome before the concept could be expanded. Further, prototype machinery had been designed without any rigorous attempt to identify theoretical mechanism performance or to correlate this with measured performance. As a consequence, machines were designed on an empirical basis. Thus it was also considered essential that actuator/valve circuits would need to be analysed in terms of the fundamental system parameters of mechanism and oil mass, damping, pressure, speed of response, so that machine performance could be calculable at the design stage.

The overall objective of this research was therefore to advance the technology, further develop the dynamic control technique and improve the engineering of linear actuators sequenced by rotary spool valves. This necessitated investigations into existing equipment designs to identify and minimise their performance limitations. As a result of these investigations, new designs of linear actuators and rotary valves were conceived, analysed, manufactured and tested. The equipment designs were then embodied into a prototype mattress sewing machine to assess their predictability, flexibility and durability when forming part of a total system design.

It is believed that through compliance with the specific equipment objectives as evidenced by detailed test programmes, the over-
all objective has been realised. Also the visual evidence of consistent and reliable stitch formation by the mattress sewing machine further vindicates the individual design concepts and the overall control technique. The sewing machine further demonstrates that fluid power can compete with and offer advantages over solid member mechanisms in this and a number of similar manipulative machines.

6.1 Actuators and Control

6.1.1 Seals

As a result of an investigation into sealing methods, it was considered that the viable alternatives to the '0' ring seals used in earlier designs were few and that a compromise solution was necessary. Of the simple seal types which were investigated, the one which was considered to offer the best compromise between the opposing demands of low leakage and low wear rates/friction losses was the "labyrinth" or "clearance" seal. It was believed that although such seals would leak, the leaked fluid would be at nominally atmospheric pressure and that consequently the technical problems associated with its containment would be reduced. Further, that as an actuator would be the prime mover of a "reciprocating mechanism", the leaked fluid could be used to lubricate ancillary parts during its containment.

Clearance seals were therefore used exclusively for the actuators developed during this research. The anticipated advantages of:

i) Improved actuator response times
ii) Simplification of analysis and design
iii) Elimination of a heat source
iv) Simplification of manufacture
v) Lubrication of ancillary mechanisms.

were realised in practice and more than compensated for the inherent increased fluid leakage rates over previous designs. There may be
occasions when it is impracticable to incorporate/encapsulate an actuator as part of an overall mechanism design and a self-contained, independent actuator is required. Provided that the subsequent operating deficiencies are considered acceptable, the designs of actuators presented herein could be extended to include friction seals such as '0' rings. It is, however, recommended that these are fitted in series with clearance seals and a separate drainage line is provided between them as shown in Figure 2.14. The compression of the '0' ring need then only be sufficient to overcome the low pressure leakage and not the system pressure. This idea was tried on a double-acting actuator and proved effective in principle, but no quantitative data relating to the effect on actuator performance, both in terms of duty rates and life expectancy was obtained.

6.1.2 Hydraulic Cushioning

It is considered fundamental that high performance linear actuators should have a controllable deceleration characteristic to limit the stress and noise levels generated in their usage. Previous miniature actuators designed at the University had achieved only partial deceleration control by means of hydraulic stop ports, designed to relieve the driving pressure towards the end of actuator travel. This design concept was found to be satisfactory only for actuators used to reciprocate knitting needles. When attempts were made either to increase the connected mass or to increase the size of actuator for more arduous duties, the severity of the final impact resulted in the failure of the piston rod connecting thread and an unacceptable noise level. Such dynamic problems are not unique and are overcome in proprietary large diameter actuators by well proven "cushioning" techniques. However, the conventional forms of hydraulic cushioning were considered impracticable for the actuator sizes required for this research.

A solution was conceived to the problem of providing effective hydraulic cushioning of miniature actuators by the incorporation of what is described as an "auxiliary piston" or "sleeve" in series with the main piston. This device combined the two main requirements
of a flow restrictor necessary to achieve cushioning in one direction and a check valve to allow unrestricted reverse motion. The essential cushioning concept involves trapping a small volume of fluid between the auxiliary piston/sleeve and the main piston towards the end of actuator travel. By the suitable dimensioning of the parts the escape of this entrained fluid is used to control the deceleration characteristic. The basic concept has been confirmed on both single-acting and double-acting actuators of various dimensions. Whilst in principle both actuator designs have the same cushioning concept, practical considerations have shown that

a) Cushioning control of single-acting units should be obtained by treating the auxiliary piston as a fixed restriction and providing a cushioning land on the main piston.

b) On double-acting actuators, the main piston should be treated as a fixed restriction and cushioning control obtained by suitable dimensioning of the auxiliary sleeve. This results in a sufficient radial clearance between the auxiliary sleeve and the actuator housing to minimise inherent eccentricity problems with double-acting actuators. It was also found necessary to groove the main piston opposite its control port to eliminate any tendency of the actuator to seize during reverse motion.

The original cushioning control sequence entailed the provision of separate supply ports for the main piston and the auxiliary piston/sleeve. Thus four control ports were considered necessary to achieve double-acting motion. The control sequence envisaged initiating pressure pulses to associated ports simultaneously but making the pressure pulse to the auxiliary piston/sleeve of a shorter duration. This had been adopted to ensure that the auxiliary piston/sleeve was driven back under system pressure before the commencement of the actuator return movement. Subsequent cushioning tests have shown that, subject to certain design constraints, the separation of the control ports and their pressure pulses is not essential. It was found that the oil supply could be combined such that only two supply ports
were necessary to achieve double-acting motion. This technique
relies implicitly on the auxiliary piston/sleeve being suitably
"free" so that it is carried or driven back by the oil flow
generated by the main piston. Any tendency of the part to stick
during this return motion could negate the cushioning. The or-
ginal control sequence has the advantage of a more positive action
albeit at the expense of a greater control complexity.

As part of theoretical actuator performance, the cushioning
concept was analysed and equations of motion have been derived.
These equations were in turn studied for their application and
practical limitations. This study indicated that the cushioning
characteristics were reactive in that they depend upon the impact
velocity of the mechanism on the trapped volume of oil and the
overall mass to be decelerated. For this reason the cushioning
characteristic can only be optimised for one particular set of
fixed operating parameters, variance about these parameters will
result in either overcushioning or undercushioning. The factors
which must be considered when establishing theoretical cushioning
requirements are:

c) The maximum induced cushioning pressure: the induced stresses
in any connective device will be related to the peak decelera-
tion characteristic of the actuator, this in turn is related
to the induced cushioning pressure. Practical tests have shown
that severe cushioning can be achieved however it is recommended
that the theoretical induced pressure should be limited to 500
bar, a figure within the fatigue and yield stress levels of
common engineering materials.

d) The final impact velocity: the final impact velocity that a
mechanism can tolerate is a function of the kinetic energy at
impact and the ability of the actuator and its associated mecha-
nism to absorb this without detrimental effects. This depends on
the particular design and as such no firm guidelines can be
given. In addition to this mechanical strength criterion it is
felt that an aural assessment, whilst being subjective, is also
valid. Tests on the particular actuators used in dynamic tests
indicated that should the final impact speed exceed 1 m/sec then the impact noise was discernibly metallic and did not engender confidence. Again it should be appreciated that this will depend on the particular design but in the absence of more specific values, it is recommended that the figure of 1 m/sec should not be exceeded.

e) Manufacturing tolerances: the cushioning characteristic is related to the cube of the radial clearance between the actuator parts and the housing and as such is heavily dependent upon manufacturing tolerances. Practical experience demonstrated that it was prudent to deliberately design an actuator to be overcushioned and then to reduce progressively the cushioning until the requisite amount was achieved.

6.1.3 Compound or Stepped Motions

The design concept of an auxiliary piston or sleeve in series with the main actuator enables "compound" or "stepped" actuator movements to be obtained. This is achieved by treating the auxiliary pistons or sleeves either as independent actuators or as hydraulically sprung stops. It is therefore possible for a double-acting mechanism to have up to three discrete movements in either direction of travel and a single-acting unit to have two discrete movements. This facility will undoubtedly contribute to design flexibility and extend the range of applications that can benefit from the overall actuating and control concept. Detailed tests on the different operating modes and control sequences have confirmed that the following movements are possible.

a) An uncontrolled forward stepped movement: by using the auxiliary piston/sleeve as a single acting actuator, the main piston can be caused to move out until the pressure induced in the fluid trapped between the parts is relieved by the opening of the main piston supply port (Section 2.4.1). Tests confirmed that in this mode the actuator performed as though controlled via a hydraulic stop and that consequently overshoot of the nominal step height occurred. This overshoot was found to be dependent upon both the supply pressure and the resistance to motion of the mechanism.
It was found that by opposing mechanism motion, for example, by a spring, this overshoot could be eliminated. Thus it is considered that this operating mode is eminently suited for single acting spring return mechanisms where stepped motions are required. On double-acting mechanisms overshoot cannot be ignored and due allowance must be made for it in the detail design. Despite this reservation the operating mode was used successfully on the yarn take-up mechanism of the mattress sewing machine.

b) A controlled stepped motion by using the opposing auxiliary piston as a sprung stop (Section 2.4.2): this motion has only been demonstrated on single-acting actuators, although conceptually the author believes that it can be applied on their double-acting counterparts. Tests on opposed single-acting units confirmed that the step height could be made independent of supply pressure. A small transient overshoot of the nominal step height occurred due, it is believed, to the geometry of the parts. This overshoot could have been eliminated by arranging for the step heights to have been coincident with the main piston port shut off.

c) A rapid advance followed by a reduced constant velocity movement: this motion was encountered when testing single-acting units for a postulated controlled step height. By blanking off the supply ports to the main piston, pressure applied to the auxiliary piston/sleeve causes the actuator to move out with an unrestricted movement until the auxiliary piston is arrested by the actuator housing. The main piston then continues to move out at a reduced constant velocity dependent upon the oil flow (leakage) past the auxiliary piston. The actuator continues to move out at this reduced velocity until either it achieves full displacement or encounters the opposing cushion whereupon a further velocity reduction may occur. Thus it is believed that in addition to achieving up to three discrete "steps" the actuating concept can achieve up to three different velocity profiles in each direction. Although a variable velocity profile was not required and hence demonstrated in this research, it is believed
that this feature will also contribute to the overall design flexibility of the actuation/control concept.

6.1.4 Different Actuator Arrangements and Geometries

The majority of detailed information presented in this thesis relates to single-acting units: in principle the design, incorporating control techniques, of both single-acting and double-acting actuators are the same. Examples of all the design inversions illustrated and referred to in this work have been built and tested, either as individual units or incorporated into particular mechanisms. Single-acting units were, in the main, preferred because of their inherent simplicity, robustness and compactness. The practical experience gained by testing all types of configurations has justified this initial preference. From the outset these units have proven to be reliable, insensitive to eccentricity errors and easy to manufacture. The designs of all double-acting actuators are prone to eccentricity errors and the designs presented herein are no exception, although it is contended that these have been minimised. Such manufacturing difficulties that were encountered were attributable to the absence of the precision tooling necessary for close tolerance work. In the final analysis it will probably be the particular mechanism design which will determine the type of actuator best suited for the application. It is hoped that sufficient information has been provided to enable others to select the appropriate actuator for a particular duty, both in terms of its configuration and size. Some general observations with regard to the basic actuator types follow.

1) Double-acting actuators: these are arguably the most common of conventional, large bore hydraulic cylinders. However, when reduced in scale to the size of actuators investigated, care must be taken in their application. Despite the ability to achieve hydraulic cushioning it is considered that this type of actuator should have a piston rod diameter of at least 3 mm. At sizes below this figure, the inherent weakness of the connecting thread and the susceptibility of the rod to buckling, impose
significant limits on the mass which can be reciprocated and the severity of cushioning (Appendix 1.1). The largest actuator built and tested was a through rod, equal area actuator and was used for needle reciprocation on a prototype candlewick tufting machine. The actuator required a main piston diameter of 9 mm, a stroke of 50 mm and a duty cycle in excess of 2000 per minute. Such tests that have been carried out have confirmed the overall validity of the design and cushioning concept. No reason can be envisaged why the basic design could not be extended to include actuators which have piston diameters of 20 mm or more.

2) Single-acting - opposed pair: a considerable number of double-acting mechanisms were tested using opposed single-acting modules. These varied in size from actuators having main piston diameters of between 3 and 8 mm and having stroke lengths of between 10 and 50 mm. It is considered that this range of actuators can be extended to have actuators with bore sizes of between 1 and 20 mm and stroke lengths of up to 8 times the bore. Whilst a preference is expressed for single-acting modules for previously stated technical reasons, it is admitted that they have certain disadvantages in terms of kinematic design and mechanism layout. Whilst it can be shown that single-acting modules are diametrically more compact than their double-acting equivalent, they generally occupy more total space. Also for the designs illustrated herein the fact that the thrust exerted is offset from the axis of motion of the associated slide, induces undesirable moment reactions. Double-acting units can be arranged more easily to ensure that the thrust exerted is coincident with the axis. Despite these reservations, all mechanisms which embodied single-acting actuators have functioned well.

3) Single-acting-spring return: this actuating concept was demonstrated on the mattress sewing machine presser foot mechanism. The mechanism functioned well, indeed for a considerable time only the measured response of the spring return movement agreed with
theoretical predictions, the spring being unaffected by the vagaries of pressure fluctuations. Within constraints imposed by the spring design and the outward impact velocity, it is believed that this actuator is a viable alternative to both double-acting and opposed single-acting actuators for constant pressure/speed mechanisms.

The outward impact velocity can be controlled by varying the viscous damping in the actuator circuit. It is recommended that this damping should be designed near the critical value so that a reasonable compromise is achieved between obtaining fast mechanism response and a reduced impact velocity. The impact velocity in the outward direction would then be reduced to nominally 40% of peak mechanism velocity and 60% of the average velocity. In the reverse motion generated by the spring force, the actuator can be designed to have full hydraulic cushioning and hence the impact velocity controlled in the normal manner.

6.1.5 Theoretical Mechanism Performance

A mechanism designer must have confidence in both the hardware and technology incorporated in his designs. At the commencement of this research the technology relating to theoretical dynamic performance of the actuator and control technique was not available for technical consideration. Prototype mechanisms had been built on a semi-empirical basis and no attempt had been made to correlate theoretical and measured performance. Thus it was considered a primary objective to analyse typical actuator/valve circuits to obtain theoretical mechanism performance and also to identify the relative importance of the system parameters. The validity of the analysis was then to be checked against the measured performance of different mechanism configurations. It was not an objective that analyses should be theoretically pedantic, but that they could enable realistic estimates of mechanism performance to be made by non-specialist mechanism designers.
The theoretical dynamic behaviour of double-acting mechanisms, spring-return mechanisms and hydraulic cushioning was identified by linearising their differential equations of motion. This technique enabled displacement, velocity and acceleration versus time relationships to be derived in terms of the main parameters, system pressure, viscous coefficient, actuator area, apparent mass and spring stiffness. The validity of the linearising technique was subsequently confirmed by checking the derived equations against iterative solutions obtained on a digital computer.

Prototype mechanisms were then analysed and tested to obtain correlation between theory and practice. These tests have indicated that the major cause of disparity did not relate to any significant shortcomings in the analytical technique, but to practical limitations imposed by the test equipment used. One of the basic analytical assumptions that the rotary valve would generate pressure pulses of a constant magnitude, was not realisable in practice.

Supply pressure to the rotary valve was regulated by proprietary pressure relief valves which proved unable to cope with high cyclic flow demands. Thus when an actuator port was connected via the rotary valve to the main fluid supply, pressure fluctuations occurred which affected subsequent mechanism performance. Depending on the actuator control sequence, these fluctuations could result in either a faster or slower mechanism performance than that predicted. The problem resulted in the main from the inherent relief valve characteristics insofar that their response time was generally higher than the mechanisms they were controlling. It was found that these supply pressure fluctuations could be suppressed, but not eliminated, by the inclusion of a bladder type hydraulic accumulator in the supply line to the rotary valve. This was found to be entirely satisfactory on the prototype mattress sewing machine where mechanism response times were themselves fairly high, but less satisfactory on mechanisms having low response times. However, the essential problem remains that the inherent characteristics of the proprietary relief valves used are inadequate for low impedance mechanisms. It is believed that to minimise further these fluctuations, it will be necessary to develop pressure relief valves with improved dynamic performance in terms of short rise time, low overshoot and minimum oscillation.
Despite problems associated with pressure fluctuations, for mechanisms which operated at supply pressures of between 30 and 50 bar, performance predictions within 15% of measured values could be realised. For applications which require a better accuracy level, it will be necessary to resort to "tuning" techniques when the mechanisms are embodied into the control circuit. Such tuning techniques might include operating a mechanism at a slightly higher pressure in order to reduce response times, or conversely by increasing the circuit impedances to increase response times. Thus, as a general recommendation, it is considered prudent to select a pump unit capable of working at least 10 bar above the nominally selected system pressure. If pressure control can be improved, a consequential increase in the accuracy of performance predictions will be realised. For particular tests where the pressure remained sensibly constant throughout the actuator travel, accuracy levels in the order of 95% were realised. However, in order to achieve either of these accuracy levels, cognisance must be taken of the detailed guidelines given in Section 2.7.

A design procedure based on theoretical considerations has been derived which enables the essential system parameters to be selected without an in-depth study. This procedure was used subsequently to ascertain the theoretical design data necessary to update the prototype mattress sewing machine to a required value of 1000 stitches per minute. This showed that both the needle and presser foot actuator were unnecessarily large and also indicated the correct sizes of their associated supply conduits. Owing to financial restrictions the recommended modifications were not incorporated and hence the validity of the procedure could not be confirmed. However, the author is confident that the procedure will yield satisfactory results and has used it for the hydraulic mechanism design of a candlewick tufting machine. This is an on-going development and although detailed assessment must wait until the mechanisms are embodied into the overall machine framework, initial tests confirm its effectiveness.
6.2 Rotary Valves

A new design concept of rotary valve has been developed which it is believed overcomes operating limitations identified in earlier valves. The valve evolved following investigations into earlier designs and by consideration of theoretical losses. The principal valve feature of sculpturing away the bobbin surface to reduce viscous drag losses and control the pressure distribution has been embodied in the design of a "universal" valve and also individual valve modules. The detailed tests and use of these valves on prototype machinery have confirmed that they are ideally suited to achieve high speed sequenced movements of linear actuators.

6.2.1 Valve Bobbin Balancing

The rotary valves developed by Garside were balanced by hydrostatic pads at either end of the valve bobbin. The sizes and positions of these pads were calculated by means of a computer programme. Practical tests on a reference valve balanced by this technique showed that its accuracy of balance was 91.5% of the hydrostatic forces, a figure in agreement with Garside's own estimate of a 10% error. Investigations showed that some error was to be expected (Section 3.3).

It was believed that by sculpturing away the bobbin surface to leave only narrow separating lands between pressure and exhaust areas, the hydrostatic forces could be controlled more accurately and further, that the valve could be made inherently self-balancing. Thus it was envisaged that the accuracy of such a balancing concept would reduce to that obtainable in the calculation and manufacture of a series of simple geometric shapes. Two methods of balancing the pressure areas and separating lands have been investigated:

a) The first method entailed splitting each pressure area and land into a series of smaller areas, compiling a balancing table and modifying individual areas to achieve self-balance. It was believed that this method whilst being tedious, would be rigorous. The large universal valve was balanced by this
method and subsequent tests indicated that a balance of 99.7% of the hydrostatic forces acting on the valve bobbin had been obtained.

b) The second method assumed that each pressure area extended to the mid-point of an adjacent separating land, thus lands did not need to be individually balanced. A large universal valve was also balanced using this technique to enable direct comparisons to be made. Unfortunately, owing to manufacturing errors, this valve bobbin was incorrectly machined and direct comparison was not possible. The bobbin balancing, however, was checked using the first technique which indicated that an imbalance of less than 1% would have resulted. As this method was the simpler it was decided to use it for the designs of subsequent valves.

A small valve module was balanced using this technique and its breakout torque calculated on the basis of a 1% error. Subsequent tests showed however that near perfect hydrostatic balance had been achieved. Similar results were also obtained on the valves used in the candlewick tufting machine mechanisms. Thus, it is concluded that this balancing technique is ideally suited for the size of valves envisaged and that a balancing accuracy of at least 99% can be anticipated.

6.2.2 Starting and Running Torque

Investigations into the starting and low speed characteristics of the computer balanced reference valve showed that these were typical of a hydrodynamically lubricated loaded journal. It was concluded that to reduce the breakout torque either:

i) The valve bobbin balanced needed to be improved and so reduce the residual hydrostatic imbalance, or

ii) The residual imbalance needed to be carried more efficiently at low speeds.

The universal valve had a bobbin of the same diameter as the reference
valve but a greater proportion of its surface area was subjected to high pressure areas. In view of this increased area and the uncertainty as to the amount that the overall accuracy of balancing could be improved, led, initially, to the selection of rolling element bearings to carry the valve bobbin. However, to measure valve imbalance initial prototypes were manufactured with plain journal bearings. It was believed that this would enable the bearings to be correctly sized and also enable direct comparison with the reference valve to be made.

As a result of the increase in the overall accuracy of balancing the breakout torque of the universal valve attributable to imbalance was only one tenth that of the reference valve. In practical terms this represents, at its design pressure of 35 bar, a breakout torque of 0.4 Nm. This figure is comparable with the torque necessary to overcome the friction force induced by the shaft lip seals. For the smaller valve bobbins the breakout torque was so low that it could not be detected with the equipment available. Having attained such low breakout torques it is now considered unnecessary, on the size of valves used in this research, to fit rolling element bearings as originally envisaged.

In addition to carrying the residual hydrostatic imbalance, the journal bearings have to withstand the effects of any dynamically induced imbalance caused by the pulsating flows generated by the valve. The dynamic testing of the valves demonstrated that plain journal can carry any such imbalance without any significant effect on the drive torque. At low operating speeds and for larger diameter valves, the combined effects of residual hydrostatic and dynamically induced imbalance may be outside the efficient load carrying capacity of hydrodynamically lubricated plain journals. Under these conditions it may be necessary to fit rolling element bearings, however, only more operating experience with the valves can identify the limits of when it would be advantageous to fit them.

Tests confirmed that the valve driving torque was related only to the viscous drag force generated between the bobbin and the casing. From elementary considerations it can be shown that the driving torque
necessary to overcome this force is related to the square of the bobbin diameter and also to the "sheared" surface area between the bobbin and casing. Thus to minimise the driving torque valve bobbins should be as small as is practicable and have the minimum of sheared area. Both of these conditions have been realised in the valve design presented herein. For the large universal valve the power consumption was estimated to be 25 watts at 500 r.p.m. The smallest valve tested was one having a 20 mm diameter bobbin and which required a theoretical power consumption to overcome viscous drag losses of less than 4 watts at 1000 r.p.m.

Within limitations imposed by both fluid flow and mechanical strength criteria no reason can be envisaged why the valves cannot be further miniaturised. Having attained such low breakout and running torques the valve input drive can also be suitably miniaturised, and need not be limited to the conventional motor drives used in this research.

6.2.3 Minimised Leakage and Drag Losses

During investigations into theoretical valve losses it was concluded that only the leakage and viscous drag losses could be controlled by the valve's geometry and metrology. Further, that whilst it was not possible to satisfy the opposing demands of low leakage and low drag losses, it was possible to minimise their combined value for one set of predetermined system parameters. The technique involved the calculation of the optimum radial clearance between the valve bobbin and the outer casing using the equations given in Section 3.5.3. The derived value of clearance then being used to obtain estimates of both the valve's internal leakage and viscous drag losses/driving torque for the specified operating conditions.

Quantitatively the theoretical predictions of the combined losses were small, and the correct order of magnitude of the predictions was verified by the subsequent testing of prototypes.
i) The calculated combined loss of the prototype valve having a 50 mm diameter bobbin rotating at 500 r.p.m. and at a supply pressure of 35 bar, based on the measured valve metrology, was predicted to be 56 watts. The subsequent measured loss was found to be 59 watts.

ii) The predicted combined loss of the prototype valve having a 20 mm diameter bobbin rotating at 1000 r.p.m., with a supply pressure of 35 bar, was 15 watts. The subsequent measured loss was found to be less than 20 watts.

It was concluded, therefore, that the equations relating to the viscous drag and leakage losses given in Section 3.5 yielded realistic estimates, provided:

a) That for a valve having plain journal bearings, the rotational speed is greater than that required to achieve full hydrodynamic lubrication of the bearings. This was at speeds greater than 30 r.p.m. and 10 r.p.m. for the valves having 50 and 20 mm bobbin diameters respectively. If the valves are rotated at speeds lower than these, the driving torque will approach the measured breakout torque of the valve at the particular operating pressure.

b) That an allowance is made to overcome the drag force exerted by any lip seals fitted to the drive shaft. This drag force was verified as being sensibly constant over the speed range of the valves tested (0-1000 r.p.m.) and was found to be 0.4 Nm and 0.15 Nm for the 50 and 20 mm bobbins respectively.

c) That, in the calculation of the valve's internal leakage, an allowance is made for the expansion of the valve casing due to pressure and the possible eccentricity of the valve bobbin.

6.2.4 Internal Flow Losses

The other major departure from earlier valve designs was the method of fluid distribution within the valve. In previous designs
this had been achieved by means of internal axial and radial drillings, for the new concept fluid flow occurs mainly peripherally over the bobbin surface. The high pressure zones form small irregular shaped internal reservoirs and fluid transfer within the valve is free to find its own most efficient path from the inlet supply port to the various mechanism ports. Thus it was believed that the pressure versus flow losses would be low. Experimentally it was established that the pressure drop across the valve (i.e. from the main hydraulic supply inlet to a mechanism supply port) was related to the square of the flow of fluid through the valve. In accordance with B.S. 4062 the pressure drop was stated in the form of dimensionless coefficient $k_*$ such that

$$p = k_* \rho \frac{v^2}{2}$$

where $v$ is the oil velocity (m/sec) through the mechanism supply port. The experimental value of $k_*$ was found to be 1.2 for the 50 mm diameter bobbin and 1.1 for the 20 mm diameter bobbin.

### 6.2.5 Universal Valve Concept

One of the research objectives was to develop a universal self-balanced valve which could be used on a wide range of multi-actuator circuits. Thus first prototypes of the new valve concept were designed to provide (theoretically) an infinite pulse selection facility between 60° and 300° of bobbin rotation. In practice complete universatility was not possible, physical restraints imposed by the separating lands and the proximity of adjacent mechanism ports, restricted the actual flexibility of the valve. Complete freedom of port positioning could only be applied to the selection of the initial mechanism port. Once this initial port position had been established, it precluded a certain area for the selection of subsequent mechanism ports. For the first prototypes the closest radial position of an adjacent port was 18° and 10 mm in an axial direction. With hindsight it is believed that a better valve arrangement would have resulted if a duplicate pressure area of 90° to 270° spaced either side of the supply annulus, had been selected. This would
have minimised the port positioning problems and also have provided a more practical pulse selection range.

Despite the above limitations, the universal valve was used successfully on three different circuits having from 5 to 9 actuators per circuit and with actuator diameters varying in size from three to ten millimetres, i.e.

a) The prototype mattress sewing machine (Section 5).

b) The prototype actuator test circuit (Section 2).

c) A two needle, three thread, chain stitch sewing machine.

For each of these circuits, it was possible to utilise the one standard bobbin and by providing a suitably machined outer casing, to obtain the desired sequence of actuator movements. To facilitate the application of the valve and the machining of the casing to suit a particular circuit, a simple graphical design procedure was adopted (Section 3.6) which enabled the correct position of a mechanism supply port to be established. This technique can be used to detail complicated sequential circuits in a matter of a few minutes. Thus it is considered that the "universal" valve concept has demonstrated its validity and has proven to be of considerable use in the development of prototype machinery.

6.2.6 Integral Valve Modules

The universal valve bobbin made and tested was sized to enable direct comparison with the reference valve and using a valve body of similar dimensions, thus one of the primary design features of the universal valve geometry, namely compactness of design, was not demonstrated. Also in view of the high viscous losses encountered on the mattress sewing machine and the inability to make small phase adjustments between mechanism movements, so the idea of a valve module evolved. What had been envisaged was a small valve having a pulse selection facility tailored to meet the control requirements of particular mechanisms. Such valves could be placed adjacent to the mechanisms they sequenced or even embodied into the housings of
the actuator(s). It was considered that such a valve would complement the "universal" concept rather than replace it. The design and control advantages that are envisaged are outlined in Section 4.6.

A valve module having a bobbin diameter of 20 mm and a length of 60 mm was manufactured and tested. The theoretically predicted losses were confirmed by practical tests and the valve then used to sequence six independent actuators without any detrimental characteristics being detected. The use of small independent valves was preferred for the hydraulic mechanism designs of a candlewick tufting machine. The high duty requirement of 2000 stitches per minute in combination with the relative simplicity of the mechanism control was considered ideally suited for an integral valve design. The two principal mechanisms, that of the needle and looper control, required minimum viscous losses to achieve the duty cycle, needed to be phased together correctly to achieve loop formation but each mechanism required only simple reciprocating movements. The subsequent detail design proved to be straightforward and resulted in neatly packaged mechanisms. Such performance tests that have been carried out on these mechanisms have confirmed both mechanism design objective and the validity of the valve module as an alternative to the universal valve.

6.3 Prototype Mechanism Designs

The fundamental objective of this research was to elevate the basic actuation and control technique to a level of predictability and reliability such that it could be offered as a viable alternative to solid member mechanisms. Thus the majority of the research presented herein was directed towards ensuring that the designs of both the actuators and valves are capable of achieving and withstanding high duty cycles. It is believed that the designs illustrated have the essential characteristics to enable the overall objective to be realised. Considerable work remains to refine the designs in terms of their detailed construction and the materials and manufacturing techniques which would ultimately be used. This form of
routine detail design, whilst being essential in terms of the ultimate success of the equipment, was of secondary importance to the author. Prototype equipment was generally designed with ease of in-house manufacture in mind and used materials/equipment readily available from the departmental stores.

The actuator and valve designs have been incorporated in two industrially sponsored projects, one a large industrial mattress sewing machine and the second a candlewick tufting machine. Whilst the major emphasis of the work on these projects has been the design and proving of mechanisms for particular duties, the work has also been aimed at developing techniques for the design of mechanisms suitable for a wide range of applications. In addition to these textile machines there are other fields where machinery design and performance is limited by the solid member mechanism. For example machines for packaging, automatic component handling and assembly, machine tool controls, contain continuous motions operating in phase related sequences. The developed theoretical analysis and design techniques established for using hydraulic actuators and rotary valve combinations could in the future be applied to improve machine design and performance of many machines in the above categories. It is hoped that the visual evidence of the performance of these prototype mechanisms will encourage others to consider and apply this use of fluid power.

6.3.1 Interlock Mattress Sewing Machine

The design and development of the lock stitch sewing machine has demonstrated in specific terms, that a group of seal-less cushioned actuators, forming part of a composite mechanism, with a mechanically driven rotating member, can be controlled by a single rotary valve for lock stitch formation. This stitch formation has been carried out in composite mattress cover material. Technical evidence indicates that the stitching speeds achieved can be uprated to levels competitive with the equivalent duty mechanical machines. In addition to demonstrating essential sewing characteristics in terms of fault free stitching runs, the ability of the mechanism to sew on and off the needle, correct stitch tensioning etc, the following functional and design advantages were achieved in practice by
the prototype rig.

i) Increased starting and stopping speeds, by independent power switching on and off individual actuators without stopping the main rotary drive.

ii) Independent selection of motions.

iii) Correct amplitude of motion achieved for individual actuators without mechanical constraints of the actuators.

iv) True straight line motion and individually controlled dwells obtained.

v) Achievement of extended throat depth and tortuous routing of power transmitted.

vi) Very large stitches up to 25 mm achieved.

vii) Compact upper sewing mechanism produced.

The success of the prototype sewing rig, as the initial prerequisite of a development programme having the ultimate aim of producing a commercially viable production machine, is evidenced by the quality and consistency of the stitching produced. A number of refinements and improvements are necessary to transform the prototype design presented to a practicable working system. However as the transition from the first prototype to final design stage can be both protracted and expensive these factors combined to prevent any refinements being incorporated.

In general terms the sewing rig represents an example of the flexibility, durability and predictability of the actuators and rotary valve concept. To the author the project represented a challenging hybrid mechanical/hydraulic mechanism which, if designed correctly, would produce stitches. What has been demonstrated is that a design solution to an exacting process was facilitated by applying the basic actuation and control technique. The relative merit of hydraulic actuation techniques versus established mechanical systems for this application was for others to decide. It would be naive to expect such a limited research project suddenly to transform traditional designs, especially those which have been developed
over a period in excess of 100 years such as a sewing machine. It is hoped that this consideration and the acknowledged ignorance of the author of any "sewing technology" will prevent cynics pointing to prototype machine deficiencies to the detriment of the inherent advantages of the actuation and control technique.

6.3.2 Candlewick Tufting Machine

This project was started after the author had ceased full-time research and personal involvement has been limited to the design and analysis of the hydraulic mechanisms only. The overall design concept includes two features that are not readily obtainable on mechanical tufting machines, that of programmable patterning and the "running on" of the looper mechanism with the needle retracted at the end of a stitching run. On the envisaged hydraulic machine these can be achieved simply by preventing needle descent at the appropriate times by the engagement of needle latch device. Thus by superimposing a simple on/off switching to the latch mechanism onto the basic hydraulic control, both of these features can be obtained without decoupling the mechanical drives. The project also presented an ideal opportunity for demonstrating and proving some of the actuating and control techniques that had been developed subsequent to the mattress sewing machine project. These included using the design procedure to establish the appropriate hydraulic system parameters, developing and testing a double-rod/double-acting actuator and sequencing the mechanisms using integral valve modules. Also unlike previous work where dynamic performance was to a certain extent consequential, the project is committed to a specified performance duty.

At the time of preparing this thesis the project has still some way to go before the overall concept can be verified. To date only the manufacture of the hydraulic mechanisms and limited qualitative tests on their performance have been made. Their detailed testing will only be possible when the individual mechanisms are embodied in the overall machine frame, this work is however outside the author's control. Such hydraulic tests that have been possible confirm the design principles and indicate that each individual mechanism will achieve its necessary dynamic performance.
APPENDIX 1.1

DYNAMICALLY INDUCED STRESSES IN A SCREWED PISTON ROD

The connective device between the piston rod of a hydraulic ram and its associated mechanism is commonly achieved by a screw thread. This has the advantage of providing a spatial adjustment. It was known that a weak point existed on the Garside actuator between the piston rod and the mechanism. It was felt necessary therefore to analyse this connective means in order to establish its limits of application. The system can be represented as shown in Figure 1. The stresses induced in the thread can be estimated by considering the energy transfer.

As the mechanism is decelerated the kinetic energy of the mechanism and the rod will need to be dissipated, this will be done at the expense of an increase in the strain energy of the piston rod, i.e.

\[
\frac{mv^2}{2} = \sigma \cdot a \cdot \sigma \cdot \frac{d}{E}
\]

\[
\sigma = \frac{2v}{d} \sqrt{\frac{mE}{\pi^2}}
\]

In addition there will be a stress concentration generated at the root of the thread. As the stress concentration factor for this application is normally taken as two, the induced stresses were therefore estimated from the equation:

\[
\sigma = \frac{4v}{d} \sqrt{\frac{mE}{\pi^2}}
\]

Using this approach the stresses induced in the Garside actuator for various mechanism masses and impact velocities were calculated and are shown plotted on Figure 2. Consideration of these results indicated that the design is only suitable for applications where relatively low masses and low impact velocities will be encountered.
1 STRESS INDUCED IN A SCREWED PISTON ROD

\[
\sigma = \frac{4V}{d} \sqrt{\frac{mE}{\pi l}}
\]

2 PERMISSIBLE IMPACT VELOCITY OF GARSDIE ACTUATOR FOR VARIOUS MECHANISM MASSES

\[
\sigma \times 10^{-8} \text{ N/m}^2
\]

- \(m = 20 \text{ gm}\)
- \(m = 5 \text{ gm}\)
- \(m = 2 \text{ gm}\)

average u.t.s. of steel

\[
l = 50 \text{ mm} \\
d = 1.68 \text{ mm}
\]
APPENDIX 1.2

LEAKAGE LOSSES FROM A SINGLE-ACTING ACTUATOR

It was assumed that for a concentric and co-axial piston inside its housing the flow between it and the main housing bore is analogous to the "Hagen-Poiseuille" flow in parallel sided slots. If the main piston is prevented from moving when subjected to a system pressure, the leakage flow can be estimated from the relationship:

\[
\text{Flow } Q = \frac{p h^3 \pi d}{12 \mu l}
\]

Similarly the general dynamic situation, when there is a relative velocity between the walls, is analogous to the so-called "Couette flow". It can be shown that the velocity profile between the piston and its housing is a function of \( \lambda \) where:

\[
\lambda = \frac{h^2}{2 \mu x} \left( - \frac{dp}{dx} \right)
\]

and where the velocity profile can be found from the equation

\[
\text{Velocity} = \frac{\dot{x} y}{h} \left[ 1 + \lambda (1 - \frac{y}{h}) \right]
\]

Inspection of these equations indicated that when the pressure is decreasing in the direction of motion (\( \lambda > 0 \)) the velocity profile is always positive. These conditions would exist as the piston moves out. As the direction of motion is reversed and \( \lambda < 0 \) then reverse flow can occur near to the fixed wall. See Figure 1. The net flow of oil from the actuator will be the algebraic sum of the individual flows using the above equations.

For the particular case of a dynamically symmetrical mechanism operating at its maximum cycling rates and having a displacement equal to its "acceleration displacement" (see Section 1.6) the solution of these equations was simplified by making the following assumptions.

1) That \( \dot{x} \) is the average velocity over the particular movement.
2) That the pressure gradient \( dp/dx \) is constant over the stroke of the piston. When system pressure is applied via the rotary
FIG 1  VELOCITY DISTRIBUTION IN PARALLEL SIDED SLOT — COUETTE FLOW

FIG 2  LEAKAGE FROM A DYNAMICALLY SYMMETRICAL D.A. MECHANISM
valve to the piston the pressure gradient will be p/l. As movement occurs there will be:
i) a decrease in pressure due to a build up of system losses
ii) a reduction in the length of engagement of the piston inside the housing.

These factors will combine to keep the pressure gradient approximately constant during the outward movement of the actuator. However during the inward movement the "exhaust" pressure gradient commences at zero and increases to a value of p/2l as the exhaust at the maximum velocity is reached. The average value of dp/dx during the inward movement was therefore taken to be equal to p/4l.

With these assumptions λ will be constant and the leakage can be calculated using the expression:

\[ Q = \frac{\pi d}{2h} \left( \lambda_0 - \lambda_i \right) \int_0^h \left( y - \frac{y^2}{h} \right) dy \]  

(5)

\[ Q = \frac{\pi dxh}{12} \left( \lambda_0 - \lambda_i \right) \]  

(6)

Substituting λ₀ and λ₁

\[ Q = \frac{5 h^3 p \pi d}{96 \mu l} \]  

(7)

Thus this analysis indicated that the leakage from the actuator under maximum cycling would be 0.625 times the static leakage estimated using equation (1).

Both equations 1 and 7 however were derived assuming that the main actuator would be concentric and parallel inside the housing bore. In practice, inevitable side loads will cause the actuator to run both tilted and eccentric. It is known that at full actuator eccentricity the static leakage would be 2.5 times greater than that predicted using equation 1. For estimating purposes it was consi-
dered that the dynamic equation should also be similarly modified. Therefore for estimating the leakage from an actuator operating at its maximum cycling rate the following equation was used:

\[ Q = \frac{\rho h^3 \pi d}{8 \mu L} \]  
(8)
APPENDIX 2.1

THEORETICAL PERFORMANCE OF TEST MECHANISM RECIPROCATED BY OPPOSED SINGLE-ACTING ACTUATORS

1. Design Data

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Stroke</td>
<td>20 mm</td>
</tr>
<tr>
<td>Main piston diameter</td>
<td>4 mm</td>
</tr>
<tr>
<td>Maximum cushion length</td>
<td>7.5 mm</td>
</tr>
<tr>
<td>Auxiliary piston diameter</td>
<td>5.5 mm</td>
</tr>
<tr>
<td>Fluid</td>
<td>Shell Tellus 27</td>
</tr>
<tr>
<td>Dynamic viscosity @ 250°C</td>
<td>0.06 N sec/m²</td>
</tr>
<tr>
<td>Fluid density ρ</td>
<td>877 kg/m³</td>
</tr>
<tr>
<td>Overall apparent mass of mechanism</td>
<td>0.07 kg</td>
</tr>
<tr>
<td>Maximum supply pressure</td>
<td>69 bar</td>
</tr>
<tr>
<td>Conduit 1 bore D</td>
<td>12 mm</td>
</tr>
<tr>
<td>&quot; length</td>
<td>2 m</td>
</tr>
<tr>
<td>&quot; 90° bends N</td>
<td>4</td>
</tr>
<tr>
<td>Conduits 2 &amp; 3 bore D</td>
<td>6.25 mm</td>
</tr>
<tr>
<td>&quot; length</td>
<td>2.6 m</td>
</tr>
<tr>
<td>&quot; 90° bends</td>
<td>6</td>
</tr>
<tr>
<td>Conduits 4 &amp; 5 bore D</td>
<td>3 mm</td>
</tr>
<tr>
<td>(combined)</td>
<td></td>
</tr>
<tr>
<td>&quot; length</td>
<td>0.9 m</td>
</tr>
<tr>
<td>&quot; 90° bends</td>
<td>24</td>
</tr>
<tr>
<td>Conduit 6 bore D</td>
<td>4.72 mm</td>
</tr>
<tr>
<td>&quot; length L</td>
<td>1.5 m</td>
</tr>
<tr>
<td>&quot; 90° bends N</td>
<td>3</td>
</tr>
</tbody>
</table>

2. Viscous Coefficients

a) Connecting pipes: Using equation 1, Section 1.5.2 the viscous coefficients of the connecting pipes based on the design data were calculated to be as follows:

<table>
<thead>
<tr>
<th>Pipe</th>
<th>Coefficient</th>
</tr>
</thead>
<tbody>
<tr>
<td>pipe 1</td>
<td>insignificant</td>
</tr>
<tr>
<td>pipes 2 &amp; 3</td>
<td>$1.027 \times 10^6 , \mu , \dot{x}$</td>
</tr>
<tr>
<td>pipes 4 &amp; 5</td>
<td>$5.689 \times 10^6 , \mu , \dot{x}$</td>
</tr>
<tr>
<td>pipe 6</td>
<td>$1.540 \times 10^6 , \mu , \dot{x}$</td>
</tr>
</tbody>
</table>
b) *Mechanism and piston drag:* Using equations 3 & 4 Section 1.5.2 and using design data relating to Figure 1.06, the viscous drag coefficients were calculated to be as follows:

- piston drag \( \cdots \cdot 2.777 \times 10^6 \mu \dot{x} \)
- mechanism drag \( \cdots \cdot 3.82 \times 10^6 \mu \dot{x} \)

Substituting for \( \mu \), the sum of the coefficients proportional to velocity was therefore taken to be

\[
k = 8.91 \times 10^5 \text{ N sec/m}^3
\]

3) **Momentum Coefficient**

Using equations 2 and 5, Section 1.5.2, the momentum coefficients of the rotary valve and the connecting pipework were calculated to be as follows:

- pipe 1 \( \cdots \cdot \text{insignificant} \)
- pipes 2 & 3 \( \cdots \cdot 368 \dot{x}^2 \)
- pipes 4 & 5 \( \cdots \cdot 33260 \dot{x}^2 \)
- pipe 6 \( \cdots \cdot 678 \dot{x}^2 \)
- rotary valve \( \cdots \cdot 684 \dot{x}^2 \)

The total of coefficients proportional to \( (\text{velocity})^2 \) was therefore taken to be:

\[
c = 3.499 \times 10^4 \text{ kg/m}^3
\]

4) **Equations of Motion**

As outlined in Section 1.5.4, to solve the generalised differential equation analytically, the viscous and momentum coefficients are linearised over the acceleration displacement of the mechanism. Constant velocity will be obtained when the combined viscous losses and momentum losses equal the supply pressure.
at a supply pressure of 69 bar (1000 lbf/in²)

\[ 6.9 \times 10^6 = 8.91 \times 10^5 \dot{x} + 3.499 \times 10^4 \ddot{x}^2 \]

\[ \dot{x} = 6.223 \text{ m/sec} \]

The theoretical equations of motion show that the constant velocity will be \( \frac{p}{K} \) where \( K \) is the combined coefficient, i.e. when

\[ \frac{p}{K} = 6.223 \text{ m/sec} \]

\[ K = 1.1087 \times 10^6 \text{ N sec/m}^3 \]

Using this value of \( K \) the equations of motion at 69 bar were calculated to be:

Displacement (mm) \( x = 31.26 (e^{0.199t} - 1) + 6.223t \)

Velocity (m/sec) \( \dot{x} = 6.223 (1 - e^{0.199t}) \)

where \( t \) is in milliseconds in the above equations.

In a similar manner the equations of motion at different supply pressures were derived, e.g.

a) When \( P = 55 \text{ bar (800 lbf/in²)} \)

Displacement (mm) \( x = 26.84 (e^{0.192t} - 1) + 5.1526t \)

Velocity (m/sec) \( \dot{x} = 5.1526 (1 - e^{0.192t}) \)

b) When \( P = 41.4 \text{ bar (600 lbf/in²)} \)

Displacement (mm) \( x = 21.697 (e^{0.185t} - 1) + 4.014t \)

Velocity (m/sec) \( \dot{x} = 4.014 (1 - e^{0.185t}) \)
c) When $P = 27.6$ bar ($400$ lbf/in$^2$)

Displacement (mm) $x = 15.77 (e^{0.177} - 1) + 2.7916t$

Velocity (m/sec) $\dot{x} = 2.7916 (1 - e^{0.177t})$

d) When $P = 13.8$ bar ($200$ lbf/in$^2$)

Displacement (mm) $x = 8.67 (e^{0.169t} - 1) + 1.4646t$

Velocity (m/sec) $\dot{x} = 1.4646 (1 - e^{0.169t})$

Using the above equations the displacement and velocity profiles for different supply profiles were calculated and are shown plotted in Figure 1.

5) **Hydraulic Cushioning**

The prototype mechanism had a maximum cushion length of 7.5 mm. To select the optimum cushioning length the effect of different cushioning lengths with the mechanism operating at the maximum supply pressure was investigated.

The length of cushion associated with a particular value of cushioning coefficient $C$ was obtained using equation 11, Section 1.8.4, i.e.

$$\text{Cushion length} = \frac{M}{CA} \left( V_i + \frac{2P}{C} \right)$$

The value of $V_i$ at a particular cushion length and supply pressure was found directly from the velocity profile (Figure 1) and used to calculate $C$. As outlined in Section 1.8 the cushioning coefficient combined the coefficients $c_1$, $c_2$ and $K$ in the relationship:

$$c = \frac{1}{c_1 + c_2 + K}$$
THEORETICAL DISPLACEMENT AND VELOCITY PROFILES OF PROTOTYPE D.A. MECHANISM
where \( K \) is the linearised coefficient calculated in (3) above, \( c_1 \) is the coefficient of the auxiliary piston and \( c_2 \) the coefficient of the cushioning land on the main actuator. The auxiliary piston was treated as a fixed orifice and its cushioning coefficient, based on an average radial clearance of 10 microns, calculated to be \( c_1 = 2.75 \times 10^{-10} \). With values obtained for \( C, c_1 \) and \( K, c_2 \) together with its associated value of radial clearance was then calculated, e.g. for a cushion length of 1 mm and at a supply pressure of 69 bar (1000 lbf/in\(^2\))

\[
V_1 = 4.625 \quad \ldots \quad \text{Figure 1, Section 1.8}
\]

\[
C = 28.46 \times 10^6 \quad \ldots \quad \text{Equation 11, Section 1.8}
\]

\[
K = 1.1087 \times 10^6 \quad \ldots \quad \text{Section 3 above}
\]

\[
c_1 = 2.75 \times 10^{-10} \quad \ldots \quad \text{Equation 2, Section 1.8}
\]

\[
c_2 = 3.6286 \times 10^{-8} \quad \ldots \quad \text{Equation 9, Section 1.8}
\]

\[
h = 29.6 \text{ microns} \quad \ldots \quad \text{Equation 3, Section 1.8}
\]

Using the above procedure, values of 'h' corresponding to various nominal cushioning lengths were calculated and are shown plotted on Figure 2. The value of the cushioning coefficient was also used to calculate the maximum induced pressure and the final impact velocity of the mechanism, these are shown plotted in Figure 3. A summary of the calculated results is given in Table 24, which also indicates the cushioning time and total mechanism response time for various cushion lengths.

After consideration of Figures 2 and 3 it was decided that a cushion length of 3 mm would be ideal for the prototype mechanism. To aim for a shorter cushioning length would increase the induced pressure beyond the recommended 500 bar. Conversely, longer cushioning lengths would have resulted in relatively high final impact velocities and the cushioning become increasingly more susceptible to errors in the diameter of the cushioning land.
FIG 2 LAND CLEARANCE VERSUS CUSHION LENGTH OF PROTOTYPE D.A. MECHANISM

FIG 3 INDUCED PRESSURE AND FINAL IMPACT VELOCITY V CUSHION LENGTH OF D.A. MECHANISM
6) **Rotary Valve Port Opening Characteristics**

As outlined in Section 3.5.4, a measure of the rotary valve and mechanism compatibility can be made using the equation

\[
\frac{PA^2}{M} = \frac{\pi D}{4} \cdot \omega R \cos \gamma \sqrt{\frac{2\Delta p}{\rho k_*}}
\]

Using the relevant data from Section 4 and at a rotary valve speed of 570 r.p.m. the orifice pressure loss at 69 bar was calculated to be 0.14 bar. The port is fully open when \( t = 2.64 \) milliseconds.
APPENDIX 3.1

ROTARY VALVE BALANCE EQUATIONS

Summary

1) Trapezoidal area: coordinates $x_1 \theta_1; x_2 \theta_2$

$$F = \frac{PWR}{\pi} (\cos \theta_1 - \cos \theta_2) \quad (1)$$

$$\bar{X} = \frac{1}{2\pi} \frac{L (\sin \theta_2 - \sin \theta_1) + x_1 \cos \theta_1 - x_2 \cos \theta_2}{(\cos \theta_1 - \cos \theta_2)} \quad (2)$$

2) Rectangular area: coordinates $\theta, X$ (width) $\bar{X}$

$$F = 2 PXR \sin \theta \quad (3)$$

3) Helical lands (pairs): coordinates $\theta_1$ and $\theta_2$

$$F = \frac{PWL}{2\pi \cos \phi} (\sin \theta_2 - \sin \theta_1) \quad (4)$$

4) Straight normal lands: coordinates $\theta$

$$F = PWR \sin \theta \quad (5)$$

5) Straight parallel lands (pairs): coordinates $l, \theta$

$$F = PWL \cos \theta \quad (6)$$

$F =$ total resolved force (Newtons)
$R =$ radius of bobbin (m)
$L =$ lead (m)
$P =$ supply pressure N/m²
$\bar{X} =$ centre of pressure (m)
$W =$ width of separating land (m)
$l =$ length of parallel land (m)
parallel land  normal land  helical land  rectangular area  trapezoidal area

BALANCING METHOD 2

FIG 1
Balancing Equations (refer to Figure 1)

1) **Trapezoidal Area**

a) Static force equation:

Force on elemental area $\delta A = P.R.\delta \theta.\delta x$

From symmetry $\sum P \sin \theta \delta \theta \delta x = 0$

\[
\sum P \cos \theta R \delta \theta \delta x = 2 \int_{x_1}^{x_2} \int_0^\theta P R \cos \theta \delta x \delta \theta = \text{Total resolved force of trapezoidal area}
\]

Now $L$ is the lead and is defined as being the axial dimension required to generate one revolution of helix i.e. $2\pi$ radians. Therefore the angular extremities of the trapezoidal area and may be found from the expression

\[
\theta_{\text{rad}} = \frac{2\pi x}{L}
\]

". by solving the double integral.

The total resolved force on the trapezoidal area

\[
= \int_{x_1}^{x_2} 2 R P \sin \theta \delta x.
\]

\[
= \int_{x_1}^{x_2} 2 R P \sin \frac{2\pi x}{L} \delta x
\]

\[
= \frac{P R L}{\pi} \left| \cos \frac{2\pi x}{L} \right|_{x_1}^{x_2}
\]

Vertical force $F = \frac{P R L}{\pi} (\cos \theta_1 - \cos \theta_2)$ \hspace{1cm} (1)

where $\theta_1, \theta_2$ correspond to the angular measurement of the developed trapezoid at $x_1$ and $x_2$ respectively.
b) In a similar manner moments may be taken about \( x_0 \) in order to determine the centre of pressure of the trapezoidal area.

\[
\text{Total moment} = 2 \ PR \int_{x_1}^{x_2} \int_{0}^{\theta} \cos \theta \, x \, \theta \, dx
\]

\[
= 2 \ PR \int_{x_1}^{x_2} \sin \frac{2\pi x}{L} \, x \, dx
\]

Integrating by parts

\[
\text{Total moment} = \frac{PR}{\pi} \left| \frac{L}{2} \sin \frac{2\pi x}{L} - x \cos \frac{2\pi x}{L} \right|_{x_1}^{x_2}
\]

\[
= \frac{PR}{\pi} \left| \frac{L}{2\pi} (\sin \theta_2 - \sin \theta_1) + x_1 \cos \theta_1 - x_2 \cos \theta_2 \right|
\]

The centre of pressure of the developed trapezoid can now be found by dividing equation 2 by equation 1

\[
\bar{x} = \frac{L}{2\pi} \frac{(\sin \theta_2 - \sin \theta_1) + x_1 \cos \theta_1 - x_2 \cos \theta_2}{\cos \theta_1 - \cos \theta_2}
\]

(3)

2) **Balancing of Rectangular Area**

\[
\sum p \, \delta A \, \sin \theta = 0
\]

\[
\sum p \, \delta A \, \cos \theta = 2 \int_{0}^{\theta} PXR \, \cos \theta \, d\theta
\]

\[
\bar{x} \text{ total resolved force on rectangular area} = 2 \ PXR \sin \theta
\]

(4)

\( \bar{x} \) may be found directly from detail drawing of rotary valve.
3) **Land Balancing**

With narrow lands separating the high and low pressure areas, the pressure gradient across them will be constant when taken normal to the land. Each of the lands separating the high and low pressure zones may be split into various smaller elements for the purposes of analysis.

a) **Helical lands.**

b) **Curved lands normal to the axis of symmetry.**

c) **Curved lands parallel to the axis of symmetry.**

With the pressure varying linearly across the land, i.e. having a triangular pressure distribution, the centre of pressure will occur \( \frac{W}{3} \) from the high pressure side of the land and there will be an average pressure of \( \frac{P}{2} \).

The locus of the centre of pressure of each individual portion of the separating lands will therefore form the basis of the land balancing equations.

a) **Helical lands:**

As an axis of symmetry of each trapezoid area exists, all helical lands will be considered in matched pairs i.e. resolved forces in one direction still oppose and cancel each other and that only resolved vertical forces need be considered.

\[
\begin{align*}
\text{Force on elemental area} & = \frac{PW}{2} \, dy \\
\text{Resolved force} & = \frac{PW}{2} \cos \theta \, dy \\
\text{Total resolved force on land} & = \int_{\theta_1}^{\theta_2} \frac{PW}{2} \cos \theta \, dy \\
\text{Now} \quad \frac{dx}{dy} = \cos \phi; \quad \frac{d\theta}{dx} = \frac{2\pi}{L} \quad \therefore \quad \frac{dy}{d\theta} = \frac{L}{2\pi} \cos \phi
\end{align*}
\]

\[
\therefore \text{Total resolved force on land} \ F = \int_{\theta_1}^{\theta_2} \frac{PW}{4\pi \cos \phi} \cos \theta \, d\theta
\]
total resolved force of a pair of lands

\[ F = \frac{PWL}{2\pi \cos \phi} (\sin \theta_2 - \sin \theta_1) \]  \hspace{1cm} (5)

In order to find the centre of pressure of the land in relation to \( x_0 \), the rigorous method would be to adopt the same technique as in 1(b) above. However, as the magnitude of the forces and moments are small compared with the main pressure areas, sufficient accuracy will result by assuming that the centre of pressure occurs midway along the land.

\[ \bar{x} = \frac{x_1 + x_2}{2} \]  \hspace{1cm} (6)

b) Straight lands normal to the axis of symmetry:

Force on elemental area = \( \frac{PWR}{2} \delta \theta \)

By symmetry \( \sum P \frac{WR}{2} \sin \theta \delta \theta = 0 \)

Total force = \( 2 \int_0^\theta P \frac{WR}{2} \cos \theta \delta \theta \)

\[ F = PWR \sin \theta \]  \hspace{1cm} (7)

\( \bar{x} \) will be found directly from detail drawing of rotary valve bobbin.

c) Straight lands (pair) parallel to axis of symmetry:

By inspection \( F = PWl \cos \theta \)
APPENDIX 4.1
DESIGN CALCULATIONS OF UNIVERSAL ROTARY VALVE

1) Selection of Helix Angle

The trapezoidal areas were to be generated on a conventional universal milling machine with a dividing head coupled to the main lead spindle of the table. The choice of angle $\phi$ (Figure 1) therefore had to be compatible with helices obtainable using standard change gears supplied with the machine. Using such a machining arrangement the angle $\phi$ can be found from the following relationship

$$\tan \phi = \frac{\pi D}{L}$$

where:

- $L$ is the lead (mm)
- $D$ is the bobbin diameter (mm)

The standard leads obtainable from the milling machine were variable between 16 and 4060, hence $\phi$ could have been any angle between $84^\circ$ and $2^\circ$ for a 50 mm diameter bobbin. A series of approximate calculations were made to determine the axial dimension 'a' of the smallest slice of the trapezoid using the equations given in Appendix 3.1, for various values of lead. It was felt that this width 'a' should be wide enough to accommodate two adjacent mechanism ports and needed to be approximately 20 mm wide. From these approximate calculations it was resolved to use a lead of 14.583 (inches).

2) Balancing of Valve Bobbin: Method 1

Data

- $\phi = 23.30865^\circ$
- $L = 14.583 \times 25.4$ mm
- $R = 25.4$ mm
- $\theta^\circ = \frac{360x}{L}$
- $\theta_{\text{rads}} = \frac{2\pi x}{L}$
FIG 1  SCHEMATIC BALANCING METHOD 1

FIG 2  SCHEMATIC BALANCING METHOD 2
a) Design Method:

From balancing concept and valve specification the following can be immediately calculated:

\[
\begin{align*}
\text{Area 1} & \quad \theta_1 = 30^\circ \quad \therefore x_1 = 30.868 \\
\text{Area 2} & \quad \theta_2 = 90^\circ \quad \therefore x_2 = 92.604
\end{align*}
\]

To obtain \( x_2 \theta_2 \) for area 1 the following procedure was adopted.

i) The resolved force of the total trapezoidal area on one side of the \( \varphi \) axis of symmetry was calculated using coordinates \( \theta_1 = 30^\circ \) and \( \theta_2 = 90^\circ \) and applying equation 1 in Appendix 3.1.

ii) The centre of pressure \( \overline{x} \) from \( x_0 \) was calculated using equation 2 in Appendix 3.1.

iii) The moment of the total area about the \( \varphi \) axis of symmetry of the valve bobbin was then calculated using the results from (i) and (ii) above.

iv) An estimate of the dimension 'a' of the trapezoidal slice (area 1) was made as follows:

Consideration of Figure 1 indicated that the ratios of the centres of pressure would be approximately 2:1 from the bobbin centre line and consequently for moment balance that the area ratio should be the inverse of this, i.e. 1:2. Therefore a reasonable first approximation was made that:

\[
a_1 = \frac{92.604 - 30.868}{3}
\]

\[
a_1 = 20.549
\]

and that the first approximation for \( x_2 \) of area 1 would be
\[ x_2 = 30.868 + 20.549 \]

i.e.  \[ x_2 = 51.4 \]

\[ \theta_2 = 50^\circ \]

v) (i), (ii) and (iii) above were then repeated to establish the moment of this area about the centre line of the valve bobbin. For theoretical hydrostatic balance the moment should be half the value calculated in (iii) above. The error between these values was designated as \( \delta M \).

vi) The dimension ‘a’ was then adjusted based on the comparison between the required moment and the value calculated above. An estimate of the required amount to shift ‘a’ (i.e. \( \delta a \)) was made by applying equation 3 in Appendix 3.1, where \( \delta a = X \) and \( b = \bar{X} \), i.e. \( M = b \cdot \delta a \cdot 2 \cdot PR \sin \theta \)

\[ \delta a = \frac{\delta M}{b \cdot 2 \cdot PR \sin \theta} \]

The above procedure was repeated until the desired accuracy of ‘a’ was obtained. In view of machining accuracy of the milling machine this was taken to 0.05 mm.

b) Calculations:

1. Total resolved force acting vertically on total area

\[ F = P \frac{RL}{\pi} \cos 30^\circ \]

2. To find \( \bar{X} \) of total area from \( x_0 \)

\[ \bar{X} = \frac{L}{2\pi} \left( 1 - \sin 30 \right) + \frac{(30.868 \times \cos 30)}{\cos 30} - (0) \]
\[ \bar{x} = \frac{29.477 + 26.732}{0.866} \]

\[ \bar{x} = 64.905 \]

3. Calculate moment from \( \bar{x} \) of bobbin, \( \bar{x} \) from \( \bar{x} = 92.604 + 4.5 - 64.905 \)

\[ \bar{x} = 32.199 \]

4. Moment about \( \bar{x} \) = 32.199 \( \cos 30^\circ \) \( \frac{P}{\pi} \cdot RL \)

Moment = 27.885 \( \frac{P}{\pi} \cdot RL \)

\[ \therefore \text{Half moment} = 13.9426 \frac{P}{\pi} \cdot RL \]

5a. To calculate force on Area 1 with coordinates

\[ x_1 = 30.868 \quad \theta_1 = 30^\circ \]
\[ x_2 = 51.4 \quad \theta_2 = 50^\circ \]

Force = \( P \frac{RL}{\pi} \) (\( \cos 30^\circ - 50^\circ \))

Force = 0.2232 \( P \frac{RL}{\pi} \)

5b. To find \( \bar{x} \) of Area 1 from \( x_0 \) with above coordinates:

\[ x = \frac{L}{2} \cdot \frac{(\sin 50 - \sin 30) + (30.868 \times \cos 30) - (\cos 50 \times 51.4)}{(\cos 30 - \cos 50)} \]

\[ \bar{x} = 42.006 \]
5c. Moment of Area 1 about \( \theta \) of bobbin

\[
M = 92.604 + 4.5 - 42.006 \left(0.2232 \frac{P}{RL} \right)
\]

\[
M = 12.2998 \frac{P}{RL} \quad \text{(i.e. too small)}
\]

6. Now \( \delta M_1 = (13.9425 - 12.2998) \frac{P}{RL} \)

\[
\delta M_1 = 1.6427 \frac{P}{RL}
\]

\[
\delta a_1 = \frac{1.6427 \frac{P}{RL}}{b_2 \cdot P \cdot R \cdot \sin \theta}
\]

\[
\delta a_1 = \frac{1.6427 L}{2 \cdot b \cdot \sin \theta}
\]

Now \( \theta = 50^\circ \) and \( b = 92.604 + 4.5 - x_2 \)

\[
b = 45.704
\]

\[
\delta a_1 = \frac{1.6427 \times 14.583 \times 25.4}{2 \times 45.704 \times x_2 \times \sin 50}
\]

\[
a_1 = 2.765 \text{ mm}
\]

A revised estimate of \( x_2 \) for area 1 is therefore \( x_2 = 2.765 + 51.400 \).

\[
x_2 = 54.174 \text{ mm}
\]

\[
\theta_2 = 52.651^\circ
\]

7. Repeating 4 to 8 above with new coordinates:

a) Force \( = 0.2593 \frac{P}{RL} \)
b) $\bar{X} = 43.397$

c) $M = 13.929 P \frac{RL}{\pi}$

d) $\delta M_2 = 0.0135 P \frac{RL}{\pi}$

e) $\delta a_2 = 0.023 \text{ mm}$

f) $\theta_2 = 52.672 \ (52^0 - 40')$

From the above it was noted that the iterations converged monotonically and rapidly; no further iterations were deemed necessary as $\delta a_3 = 10^{-4} \text{ mm}$ by inspection.

As the effective trapezoid pulse generating section was symmetrical (Section 3) about the centre line of the bobbin, the above coordinates apply also to the right hand side of the bobbin.

c) Machining coordinates:

It was then necessary to translate the above information into machining coordinates $x$ and $\theta$ of the valve body. Consider Figure 3, the above procedure had established points $a$, $b$ on area 1 and hence their counterparts on areas 2, 3 and 4. It was decided to use a 6 mm diameter end mill for the sculpturing and hence $x_1 = 5.50 \text{ mm}$. From consideration of the geometry (Figure 3) it can be seen that:

- $pq = 5.5 \sec \phi$
- $pq = 5.988 \text{ mm}$

and
- $qr = 5.5 \tan \phi$
- $qr = 2.369 \text{ mm}$

$. \ . \ rp = 5.998 - 2.369$
- $rp = 3.628 \text{ mm}$

$. \ . \ \text{Angular coordinate of } rp = \theta_{rp} = 8.1845^0$
- $= 80^0 - 10'$
Extended perimeter of high pressure area

FIG 3 DESIGN GEOMETRY ~ BALANCING METHOD ONE

Parallel land  Helical land

Normal land

FIG 4 DESIGN GEOMETRY ~ BALANCING METHOD TWO
Machining coordinates of $p$

\[
x = x_1 + 5.5 = 30.868 + 5.5
\]

\[
x = 36.368
\]

\[
\theta = 30^\circ - (80 - 10')
\]

\[
\theta = 210 - 50'
\]

Coordinates from bobbin

\[
x_1 = 60.736
\]

\[
x_4 = 37.407
\]

\[
\theta_1 = 158 - 10
\]

\[
\theta_4 = 440 - 30
\]

Similarly, the machining coordinates of all other relevant datums were found and the results are shown tabulated in Figure 5.

3. **Balancing of Valve Bobbin: Method 2**

Using the data obtained by Method 1 it was then possible to split the various trapezoidal areas into sections and discrete lengths of separating lands as shown in Figure 2. By applying the equations given in Appendix 3.1, Method 1 was checked using Method 2. The summary of the results is given in Table 15. From consideration of these results it was seen that for a unit pressure of 1N/mm²:

a) For static forces only:

\[
F_2 + F_4 > F_1 + F_3
\]

the disparity between them being 89 Newtons per unit pressure

b) For moment balance:

\[
M_{14} > M_{23}
\]
Machining co-ordinates

<table>
<thead>
<tr>
<th>Ref</th>
<th>X mm</th>
<th>θ degrees</th>
<th>Ref</th>
<th>X mm</th>
<th>θ degrees</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>60,736</td>
<td>131°-9'</td>
<td>4</td>
<td>37,407</td>
<td>44°-30'</td>
</tr>
<tr>
<td>2</td>
<td>48,407</td>
<td>146°-10'</td>
<td>5</td>
<td>48,407</td>
<td>146°-10'</td>
</tr>
<tr>
<td>3</td>
<td>37,407</td>
<td>44°-30'</td>
<td>6</td>
<td>60,736</td>
<td>158°-10'</td>
</tr>
</tbody>
</table>

Note: - Use 6,00 mm diameter cutter
LEAD = 14.583 LAND WIDTH = 5,00 mm ± 0,10

FIG 5 MACHINING CO-ORDINATES OF SELF BALANCED ROTARY VALVE ~ METHOD 1
the disparity between them being 8847 N mm for a unit pressure.

For the prototype bobbin, the static and moment imbalance was to be reacted by plain journals positioned 160 mm apart. Method 2 therefore indicated that Method 1 would, per unit pressure:

i) Produce the journal reactions shown in Figure 5.

ii) Result in a breakout torque of 0.6875 Nm.

iii) Result in an accuracy of balancing of 0.9%.

The results obtained by analysing Method 1 were then used to improve the accuracy of balancing. From consideration of the balancing summary it was seen that in order for the imbalance to be reduced $F_3$ needed to be increased. If, for example, the land in Figure 2, was pushed back to increase $F_3$ the only area to be affected would be number 5. Reference to Table 15 will show that this area was almost inherently self balanced and that such a move would be effective.

This was checked and a new coordinate for land '1' established. Calculations showed that the imbalance then reduced to:

1) A zero static imbalance
2) A reduction in moment imbalance to 4800 N mm per unit pressure with $M_{14}$ still greater than $M_{23}$.

This remaining residual moment imbalance was eliminated by stretching the valve slightly in the axial direction. As $F_2 + F_3 = 3 (F_1 + F_4)$ any increase in the axial dimensions of the respective centres of pressure would result in a proportionately larger increase in $M_{23}$ than in $M_{14}$.

The final coordinates of the valve are given in Figure 6 and the summary of the balancing is given in Table 16.
MACHINING COORDINATES OF SELF BALANCED ROTARY VALVE—METHOD 2

<table>
<thead>
<tr>
<th>Ref.</th>
<th>X mm</th>
<th>θ degrees</th>
<th>Ref.</th>
<th>X mm</th>
<th>θ degrees</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>62,236</td>
<td>131.9</td>
<td>4</td>
<td>38,529</td>
<td>45.52</td>
</tr>
<tr>
<td>2</td>
<td>50,929</td>
<td>145.00</td>
<td>5</td>
<td>50,229</td>
<td>145.28</td>
</tr>
<tr>
<td>3</td>
<td>39,229</td>
<td>45.10</td>
<td>6</td>
<td>63,236</td>
<td>158.10</td>
</tr>
</tbody>
</table>

Note: Use 6.00 mm diameter cutter
LEAD = 14.583    LAND WIDTH = 5.00 mm ± 0.10

FIG. 6
4) **Theoretical Valve Losses**

The rotary valve prototypes were optimised for minimised power losses based on the following data:

Supply pressure $P$ = 35 bar

Maximum valve speed = 500 r.p.m.

Oil viscosity $\mu$ = 0.05 kg/m sec

Length of separating lands = 0.75m

Width of separating lands = $5 \times 10^{-3}$ m

Bobbin diameter = $50.8 \times 10^{-3}$m

a) Maximum peripheral velocity:

$$V = \frac{50}{6} \times \pi \times 50.8 \times 10^{-3} \text{ m/sec}$$

$$V = 1.33 \text{ m/sec}$$

b) To find the optimum clearance 'h' for minimised power loss the equation derived in Section 3.5.3 was applied:

Power loss $w = \frac{3.5^2 \times 10^{12} \times 0.75 \times h^3 + 0.05 \times 1.33^2 \times 5 \times 0.75 \times 10^{-3}}{60 \times 0.05 \times 10^{-3}}$ h

$$w = 3.0625 \times 10^{15} h^3 + \frac{3.3167 \times 10^{-4}}{h}$$

$$\frac{dw}{dh} = 9.1875 \times 10^{15} h^2 - \frac{3.3167 \times 10^{-4}}{h^2}$$

\[ \therefore \text{putting} \ \frac{dw}{dh} = 0 \ \text{for maxima and minima} \]

$$h'' \equiv \frac{3.3167}{9.1875} \times 10^{-19}$$

$$h = 1.38 \times 10^{-5} \text{m}$$
c) Valve leakage loss:

\[
\text{Valve leakage} = \frac{3.52 \times 10^{12} \times 1.38^3 \times 10^{-15} \times 0.75}{60 \times 0.05 \times 10^{-3}}
\]

\[
= 8.03 \text{ watts}
\]

d) Viscous drag loss:

\[
\text{Power loss} = \frac{0.05 \times 1.33^2 \times 5 \times 10^{-3} \times 0.75}{1.38 \times 10^{-5}}
\]

\[
= 24 \text{ watts}
\]

Therefore the total theoretical losses were estimated to be 32 watts and the driving torque at 500 r.p.m. to be 0.457 Nm.

5) Valve Casing Stresses

The optimum value of 'h' calculated for minimised losses was at a supply pressure of 35 bar. At such a pressure the outer casing will expand and distort due to the asymmetry of the pressure distribution contained within it. The exact analysis of the stress and strain distributions in the outer casing would be complex even in the static condition and would be infinitely more difficult to analyse when the bobbin is rotating. An indication of the casing expansion and the stress levels induced was obtained by simplifying the problem to one of a thick cylinder subjected to a constant internal pressure. It was then possible to apply the standard Lamé equations from elementary stress analysis theory, i.e.

\[
\delta h = \frac{R}{E} \left( \sigma_c + \nu P \right)
\]
where \( \sigma_c \) is the circumferential stress,
and \( E \) for cast iron = 9.6 \times 10^{10} \text{ N/m}^2
\( \nu \) = Poisson's ratio = 0.26.

\( \sigma_c \) at \( R = 25 \times 10^{-3} \text{ m} \) is at a maximum and can be found from the expression:
\[
\sigma_c = \frac{(d_2^2 + d_1^2)}{(d_2^2 - d_1^2)} \cdot P
\]
\[
= \frac{37.5^2 + 25^2}{37.5^2 - 25^2} \cdot P
\]
\( \sigma_c = 2.6 \cdot P \).

\[
\therefore \quad h = \frac{R}{E} \cdot P \cdot (2.6 + 0.26)
\]
\[
\delta h = \frac{2.86 \cdot R \cdot P}{E}
\]

\[
\therefore \quad \frac{\delta h}{9.6 \times 10^{10}} = \frac{2.86 \times 35 \times 10^5 \times 25 \times 10^{-3}}{9.6 \times 10^{10}}
\]
\( \delta h = 2.61 \times 10^{-6} \text{ m} \)

Considering the schematic pressure distribution over the bobbin surface it will be seen that this approximation was valid for one side of the valve, i.e. for areas 3 and 4 where the pressure areas are greater than 180° but that for areas 1 and 2 such a simplification would be inaccurate. In addition the theory assumes that end
6. **Bobbin Stresses**

The bobbin stresses can be analysed more accurately than the cylinder stresses. If the variation in the dynamic pressure is ignored, the pressure distribution over the bobbin surface when oil is passing through the valve, and the forces acting on the bobbin will be the same in both the static and dynamic condition. In the static condition the bobbin is analogous to a loaded beam of circular cross-section with superimposed radial stresses in the pulse generating areas.

An accurate bending moment diagram for the bobbin can be constructed from the bobbin balancing summary given in Table 15 and hence the magnitude of the induced stresses and strains derived. The resulting bending moment diagram will be similar to that shown in Figure 7.

The maximum bending moment will occur at the centres of pressure of $F_2$ and $F_3$ and will be of magnitude $F_1 (b - a)$ where $(b - a)$ can be established by varying equations 1 and 2 in Appendix 3.1. This figure was found to be

\[
\begin{align*}
b &= 56.06 \\
a &= 22 \\
(b - a) &= 24 \text{ mm}
\end{align*}
\]

Now $F_1 = 800 P$ where $P$ is supply pressure in N/mm$^2$.

\[
\text{Maximum bending moment} = 800 \times 25 \text{ P N mm.}
\]

Hence at a supply pressure of 35 bar the maximum bending moment = 67.2 Nm.

The maximum bending stresses were then calculated from the elementary theory of loaded beams i.e.
NB  $p = \text{pressure}\ (N/mm^2)$

bending moment diagram

FIG. 5  BEARING REACTIONS AND BENDING MOMENT DIAGRAM OF VALVE BOBBIN
The value of $I$ was taken for the circular core of the bobbin. 

$I$ for a circular beam is given by the expression:

$$I = \frac{\pi d^4}{64} \quad \text{where } d \text{ the core diameter is } 37 \text{ mm.}$$

\[
\therefore \quad \sigma_b = \frac{67.2 \times 32}{d^3} \quad \text{i.e. at } y = \frac{d}{2}
\]

\[
\sigma_b = 13.5 \text{ N/mm}^2
\]

This indicated that the maximum bending stress would be approximately four times the supply pressure, well within the yield strength of the material (mild steel). It was noted that the separating lands would carry a significant proportion of the bending moment in addition to withstanding the direct pressure of the entrained fluid.

The lands were therefore specified to be generated with a spherically ended cutter, the advantages for so doing being:

a) to reduce the stress concentration factor between the root of the land and the core of the bobbin.

b) to distribute the stresses more evenly across the lands.

However these cutters were not in stock and consequently the prototypes were manufactured with sharp internal corners. Approximate calculations indicated that despite this the lands would be strong enough to withstand the combined effects of the internal pressure and the bending stress.
APPENDIX 4.2
TEST EQUIPMENT - ROTARY VALVE

Hydraulic Power Pack: Vickers Gear pump/tank unit
Maximum working pressure = 1200 lbf/in²
Maximum delivery = 5 gals/min

Valve Input Drive: 0.5 DHP 'Kopp' variable speed drive

Flow Meters: GEC Rotometers, 2100 series

Torque Transducer: Westland Aircraft Ltd.
(Saunders Roe Division)
Indicator TM-GR
Transducer Type TT2/4/AB
Ser. No. 52/20702

Pressure Transducer: S.E. Laboratories Ltd.
Carrier System SE 511
U.V. Recorder Type 2600/12
Transducer 0-200 psi Ser. No. 49080

Temperature Measurement: Fieldon Thermocouple

Pressure Calibration Rig: Budenberg
Calibration of Westland torque transducer TT2/4/AB

Calibration of pressure transducer Ser. No. 49080
APPENDIX 4.3

DESIGN CALCULATIONS OF VALVE MODULE

DESIGN PARAMETERS

Supply pressure $P = 35$ bar  
Valve running speed $= 1000$ r.p.m.  
Oil viscosity $\mu = 0.05$ kg/m.sec  
Bobbin diameter $d = 20$ mm  
Land width $b = 3$ mm  
Total land length $L = bwd$

a) Theoretical Valve Losses

Power losses $= \frac{P^2h^3}{12\mu b} + \frac{V^2bL}{h}$  

Peripheral velocity $V = 1.048$ m/sec

Power loss $w = \frac{3.5^2 \times 10^{12} \times 0.377 \times h^3 + 0.05 \times (1.048)^2 \times 3 \times 10^{-3} \times 0.377}{12 \times 0.05 \times 3 \times 10^{-3}}$

$= 2.56 \times 10^{15} h^3 + \frac{6.2}{h} \times 10^{-5}$

$\frac{dw}{dh} = 10^{15} \times 7.68h^2 - \frac{6.2 \times 10^{-4}}{h^2} = 0$ for max or min

$\therefore h = \frac{6.2}{0.768} \times 10^{-20}$

$h = 1.69 \times 10^{-5}$m (approximately 0.00066")

$\therefore$ diametral clearance = $3.38 \times 10^{-5}$m (0.0013")
Viscous drag loss = \( 0.05 \times \frac{(1.048)^2 \times 3 \times 10^{-3}}{1.69 \times 10^{-5}} \times 0.377 \)

\[ = 3.67 \text{ watts} \]

Driving torque \( J = \frac{3.67 \times 60}{2 \times 1000} \)

Torque = 0.035 Nm

Internal leakage loss = \( 2.56 \times 10^{15} \times (1.69)^3 \times 10^{-15} \)

Leakage = 12.35 watts

\( \therefore \) total power losses approximately 15 watts.

b) Valve Balancing Method 1: Section 3.4.3

See Figure 1.

Areas 1 and 4 are 90\(^\circ\) pulses - total effective width

\[ = 4.5 + 3.0 = 7.5 \text{ mm} \]

\( \therefore \) using equations in Appendix 3.1

Moment of areas 1 and 4 = \( \frac{7.5 \times 18}{\sqrt{2}} \) = 95.5 units

Moment of land \( a \) = \( F \bar{x} \)

\[ = 1.5 \times 4 \]

\[ = 6 \text{ units} \]

\( \therefore \) total moment of land 'a' + Area 1 or 4 = 101.5 units

Areas 2 and 3 to balance moments are 180\(^\circ\) pulses.

\[ \bar{x} = 3 + \frac{x}{2} \text{ and moment} = x \left( 3 + \frac{x}{2} \right) = 101.5 \]

\( \therefore \) \( x^2 + 6x - 203 = 0 \)
FIG 1
DESIGN GEOMETRY 90° and 180° VALVE MODULE ~ BALANCING METHOD 1

FIG 2
TRANSVERSE DRILLING OF BOBBIN
c) **Estimated Breakout Torque**

Based on the accuracy level obtained with the large prototype balanced by Method 1 of nominally 1% and using the expression

\[ T = F \cdot R \]

\[ = P \times \text{Area} \times 0.01 \times 0.25 \times 10^{-2} \]

From the detail drawings the total area of the high pressure zones are 722 mm\(^2\) for areas 2 and 3 and 236 mm\(^2\) for areas 1 and 4. The estimated breakout torque was therefore calculated to be 0.084 Nm at a supply pressure of 35 bar.
APPENDIX 5.1

TEST EQUIPMENT - MATTRESS SEWING MACHINE

Hydraulic Power Pack: Pratt Hydraulics Gear pump/tank unit
Maximum pressure 700 lbf/in²
Maximum delivery 2.5 galls/min
Serial No. 880275

Relief Valve: Pratt Hydraulics type VR4-1000-RA-SC/01
Serial No. 79325

Variable Speed Drive: Sunstrand Hydraulics motor Model 15-3007
Serial No. OZ-PA 22581; pump model No. 15-2016 Serial No. 04-PO-21973

Hydraulic Accumulator: Fawcet

Temperature Measurement: Fieldon Thermocouple
Serial No. 470

Pressure Transducer: S.E. Laboratories Ltd.
Carrier System SE511
U.V. Recorder Type 2600/12
Transducer 0-1000 lbf/in² Serial No. 56025
APPENDIX 5.2
THEORETICAL SEWING MACHINE PERFORMANCE

Based on physical data obtained from the hydraulic sewing machine, and the detail design drawings, each mechanism was analysed to obtain its theoretical equations of motion. These equations of motion were used in conjunction with the hydraulic pressure/exhaust switching sequence, to indicate which mechanism movement imposed the upper limit on the machine's performance. This involved the identification and estimation of all the possible performance limiting factors and applying the equations of motion given in Section 1.5.

Tables 18, 19 and 20 give a breakdown of the appropriate design data and a summary of the variables affecting mechanism performance. The general pipe data and hydraulic circuit used in the analysis is given in Figure 5.13. The following general design parameters were used in the analysis.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>System pressure P</td>
<td>50 bar</td>
</tr>
<tr>
<td>Fluid</td>
<td>Shell Tellus 27</td>
</tr>
<tr>
<td>Operating temperature</td>
<td>25°C</td>
</tr>
<tr>
<td>Fluid viscosity @ 25°C</td>
<td>0.05N sec/m²</td>
</tr>
<tr>
<td>Average running clearance</td>
<td>20 microns</td>
</tr>
</tbody>
</table>

1) The Needle Mechanism

a) The combined viscous and damping coefficient K:

The needle mechanism will flow saturate when:

\[ k\ddot{x} + c\dot{x}^2 - P = 0 \]

For the dynamic analysis P was taken as 35 bar to allow 15 bar for the needle penetration of the materials to be sewn. From table 19:

\[ 2.1 \dot{x} + 0.1336 \dot{x}^2 - 3.5 = 0 \]
the saturation velocity of the needle mechanism
= 1.5197 m/sec.
As the saturation velocity is equal to \( P/K \), the combined
viscous damping coefficient was calculated to be

\[ K = 2.303 \times 10^6 \text{ N sec/m}^2 \]

b) Dynamic equations of motion:
Using the value of \( K \) calculated above and the rig data
the theoretical equations of motion for the needle mecha­
nism were found to be:

1. Needle displacement (m) = 0.0166 \( (e^{-91t} - 1) \) + 1.5197t
   \( (1) \)
2. Needle velocity (m/sec) = 1.5197 \( (1 - e^{-91t}) \)
   \( (2) \)
3. Needle accln. (m/sec\(^2\)) = 138.5 \( e^{-91t} \)
   \( (3) \)

where time \( t \) is in seconds.

c) Mechanism response times:
Using equation 1 above the response times of the consti­
tuent parts of the needle movement were calculated to be:

<table>
<thead>
<tr>
<th>Movement</th>
<th>Time</th>
</tr>
</thead>
<tbody>
<tr>
<td>Needle advance</td>
<td>43.8 ms</td>
</tr>
<tr>
<td>Throw thread loop</td>
<td>7.2 ms</td>
</tr>
<tr>
<td>Needle retract</td>
<td>41.8 ms</td>
</tr>
</tbody>
</table>

d) Sewing machine maximum running speed:
The response times calculated above, in conjunction with
the proportion of the overall cycle time allotted for their
operation, were used to determine the maximum running speed
of the sewing machine.

<table>
<thead>
<tr>
<th>Movement</th>
<th>Speed</th>
</tr>
</thead>
<tbody>
<tr>
<td>Needle advance</td>
<td>120° pulse = 456 stitches/min</td>
</tr>
<tr>
<td>Throw thread loop</td>
<td>300° pulse = 690 stitches/min</td>
</tr>
<tr>
<td>Needle retract</td>
<td>1650° pulse = 660 stitches/min</td>
</tr>
</tbody>
</table>
2) The Needle Yarn Take-up Mechanism

a) The combined viscous and damping coefficient $K$:
For this mechanism the full system pressure of 50 bar was available to accelerate the mechanism. Using the rig data the actuator saturation velocity was found from the equation

$$2.4\dot{x} + 0.0742x^2 - 5 = 0$$

\[.\] actuator actuation velocity $= 1.964 \text{ m/sec},$

\[.\] combined viscous damping coefficient $K = 2.545 \times 10^6 \text{ N sec/m}^2.$

b) Dynamic equations of motion:
Using the value of $K$ calculated above the theoretical equations of motion of the take-up actuator were found to be

1) Actuator displacement (m) $= 0.020 \left(e^{-98t}-1\right) + 1.964t$
(4)
2) Actuator velocity (m/sec) $= 1.964 \left(1-e^{-98t}\right)$
(5)
3) Actuator accln. (m/sec$^2$) $= 192 e^{-98t}$
(6)

c) Mechanism response times:
Using equation 4 above the response times were calculated to be:

<table>
<thead>
<tr>
<th>Event</th>
<th>Time (milliseconds)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Introduce slack</td>
<td>23</td>
</tr>
<tr>
<td>Return step</td>
<td>8.2</td>
</tr>
<tr>
<td>Completion of return</td>
<td>14.8</td>
</tr>
<tr>
<td>Full return</td>
<td>23</td>
</tr>
</tbody>
</table>

d) Maximum cycling rates based on the above response times:

<table>
<thead>
<tr>
<th>Event</th>
<th>Cycles/min</th>
</tr>
</thead>
<tbody>
<tr>
<td>Introduce slack</td>
<td>$120^\circ$ pulse = 870 stitches/min</td>
</tr>
<tr>
<td>Return step</td>
<td>$30^\circ$ pulse = 610 stitches/min</td>
</tr>
</tbody>
</table>
Completion of return ..... 60° pulse = 675 stitches/min
Full return ..... 90° pulse = 650 stitches/min

3. The Presser Foot Mechanism

The analysis of this machine was complicated by the characteristics of the material it compressed. The material was a composite which exhibited a non-linear spring characteristic. The presser foot was therefore designed to produce asymmetric profiles between the forward and return movements when unresisted by the material, i.e. with a fast outward movement and a relatively slow return. When operating against the resistance of the mattress cover material, the material resilience would tend to balance the dynamic profiles by retarding the outward movement and assisting the return movement. The analysis of the presser foot mechanism did not account for the non-linearity of the material to be stitched, consequently the response times calculated were viewed as being respectively the maximum and minimum values of the return and outward movements.

a) Spring return movement:

For the return movement, neither the viscous coefficient or mass terms relating to the main inlet pipe needed to be considered. Using a value of the viscous coefficient of $24.5 \times 10^6$ and an overall mechanism mass of 1.12 kg, the equations of motion were calculated to be:

Displacement (mm) = $23 \left[1-e^{-43t}(1.237 \sin 34.7t + \cos 34.7t)\right]$  
(7)

Velocity (m/sec) = $20.2 \, e^{-43t} \sin 34.7t$  
(8)

Acceleration (m/sec²) = $2.02[\, e^{-43t}(34.7 \cos 34.7t - 43 \sin 34.7t)]$  
(9)

where $t$ is seconds.
b) Forward movement of presser foot:

For this movement the viscous coefficient and mass term relating to the exhaust pipe were ignored. Using a value if viscous coefficient of $18.23 \mu \times 10^6$ and an overall mechanism mass of 1.05 kg, the equations of motion were calculated to be

Displacement (mm) = $105[1-e^{-34t} (0.7456 \sin 45.73t + \cos 45.73t)]$  

Velocity (m/sec) = $7.469 e^{-34t} \sin 45.73t$  

Accln. (m/sec²) = $7.469[e^{-34t} (45.73 \cos 45.73t - 34 \sin 45.73t)]$

(10)  
(11)  
(12)

c) Mechanism response times:

Using equations 7 and 10 the response times of the presser foot mechanism were found to be:

- Presser foot advance ....... 10 milliseconds
- Presser foot return ....... 30 milliseconds

These response times would indicate that the sewing rates would be:

- Presser foot advance - 120° pulse = 2000 stitches/min (Max)
- Presser foot return - 125° (equiv) = 695 stitches/min (Min)

The mean response, to allow for the spring effect of the mattress cover material was taken to be 20 milliseconds for both the forward and return movements and that the mechanism could accommodate stitching rates of approximately 1000 stitches/min.

4) Maximum Sewing Speed of Prototype Rig

The theoretical analysis of the various mechanism movements indicated that the downward movement of the needle mechanism imposed the upper limit of machine performance. At speeds greater than 450 stitches per minute, needle attenuation would occur.
5) **Flow Demand of Rig**

At the maximum speed of 450 per minute and using the velocity equations derived for each mechanism (equations 2, 5 and 11 above) the flow demand per stitch cycle was constructed. Figure 5.12 shows the demand curve constructed using the data tabulated below for two conditions.

i) Assuming that the presser foot motion had unopposed motion i.e. that equation 11 above could be applied directly.

ii) Assuming that the presser foot motion was opposed by the mattress covers and that its response time was extended to 20 milliseconds.

Data: \( t \) in milliseconds; \( Q = \text{cc's per minute} \)

<table>
<thead>
<tr>
<th>( t )</th>
<th>Needle 'Q'</th>
<th>Take up 'Q'</th>
<th>Presser foot 'Q'</th>
</tr>
</thead>
<tbody>
<tr>
<td>5</td>
<td>1673</td>
<td>895</td>
<td>6691</td>
</tr>
<tr>
<td>10</td>
<td>2744</td>
<td>1449</td>
<td>11026</td>
</tr>
<tr>
<td>15</td>
<td>3407</td>
<td>1790</td>
<td></td>
</tr>
<tr>
<td>20</td>
<td>3830</td>
<td>1990</td>
<td></td>
</tr>
<tr>
<td>25</td>
<td>4101</td>
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<td>30</td>
<td>4262</td>
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</tr>
<tr>
<td>40</td>
<td>4463</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>
APPENDIX 5.3

UPRATED SEWING MACHINE PERFORMANCE

To improve the response times of the individual mechanisms, it was proposed that the following changes should be made to the prototype sewing rig.

a) To increase the bore of the main inlet and exhaust pipes to the rotary valve from 6.25 mm to 4.37 mm respectively to 12.5 mm.

b) To increase the bore size of conduits between the rotary valve and the needle mechanism from 4.37 mm to 6.25 mm.

c) To increase the bore size of conduits between the rotary valve and the needle yarn take-up mechanism from 3 mm to 4.37 mm.

d) To increase the bore size of the conduit between the presser foot mechanism and the rotary valve from 4.37 mm to 6.25 mm.

e) To reduce the contract area of the needle and needle yarn mechanism slides.

Tables 21, 22 and 23 give a breakdown of the design data for inclusion in the theoretical analysis of each mechanism.

a) The needle mechanism equations of motion:

Displacement (m) = 0.0966 (e^{-54.3t} - 1) + 5.248t \quad (1)

Velocity (m/sec) = 5.248 (1 - e^{-54.3t}) \quad (2)

Accln. (m/sec^2) = 285 e^{-54.3t} \quad (3)

The response time of the needle mechanism using equation 1 above was calculated to be 22.75 milliseconds and hence the maximum sewing speed of the rig would be 880 stitches/min.

b) The needle yarn take-up mechanism - equations of motion:

Displacement (m) = 0.06176 (e^{-100t} - 1) + 6.156t \quad (4)

Velocity (m/sec) = 6.156 (1 - e^{-100t}) \quad (5)
\[ Accln. \ (m/sec^2) = 616 \ e^{-100t} \]  

Using equation 4 above, the response time of the needle yarn take-up mechanism was calculated to be 10.6 milliseconds and that based on this figure the sewing machine was capable of speeds up to 1180 stitches/min.

c) Presser foot mechanism - equations of motion: iterative solution:

The inclusion of the momentum coefficients into the theoretical equations of motion of a single-acting spring return mechanism cannot be achieved in the same manner as that adopted for ordinary double-acting mechanisms. Where it is apparent that the momentum coefficients will be significant in the analysis of a single-acting mechanism, an iterative solution of the equations of motion is recommended.

From the application of Newton's second law of motion, and using the notation adopted for a single-acting spring return mechanism:

i) Forward movement:

\[ \text{Force} = PA - S (B + x) - KA \dot{x} - \alpha A \ddot{x}^2 = M \ddot{x} \]  

ii) Spring return movement

\[ \text{Force} = S (B + \overline{x}) - Sx - KA \dot{x} - \alpha A \ddot{x}^2 = M \ddot{x} \]  

If over a small time interval \( \delta t \) the acceleration \( \dot{x} \) is assumed constant, then the following equations can be applied:

\[ \dot{x}_n = \dot{x}_{n-1} + \ddot{x}_{n-1} \delta t \]  

\[ x_n = x_{n-1} + \dot{x}_{n-1} \delta t + \frac{1}{2} \ddot{x}_n (\delta t)^2 \]  

\[ x_n = \frac{A}{M} (P - \frac{SB}{A} - \frac{S}{A} x_{n-1} - K \dot{x}_{n-1} - \alpha \dot{x}^2_{n-1}) \]  

or \[ = \frac{S}{M} (B + \overline{x}) - \frac{S}{M} x_{n-1} - \frac{KA}{M} \dot{x}_{n-1} - \frac{\alpha}{M} \ddot{x}^2_{n-1} \]
where equation 11 relates to the forward movement of 12 to the spring return.

Inserting the boundary condition that \( V_0 \) and \( x_0 = 0 \), the above equations can be used to construct the appropriate dynamic profiles. Using the data given in Table 24, the acceleration equations of the forward and spring return movements of the presser foot were calculated to be:

\[ a) \text{ } \text{Forward movement} \]

\[ \ddot{x}_n = 1.577 (360 - 6.684\dot{x}_{n-1} - 19.83\ddot{x}_{n-1} - 1.8665\dot{x}_{n-1}^2) \]  
(13)

Using equation 13 and equations 9 and 10 the following data was calculated:

<table>
<thead>
<tr>
<th>time t (milliseconds)</th>
<th>1</th>
<th>2</th>
<th>3</th>
<th>4</th>
<th>5</th>
<th>6</th>
<th>7</th>
</tr>
</thead>
<tbody>
<tr>
<td>accln. (m/sec²)</td>
<td>.567</td>
<td>.546</td>
<td>.517</td>
<td>.482</td>
<td>.442</td>
<td>.398</td>
<td>.350</td>
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<tr>
<td>velocity (m/sec)</td>
<td>.567</td>
<td>1.113</td>
<td>1.63</td>
<td>2.112</td>
<td>2.554</td>
<td>2.952</td>
<td>3.303</td>
</tr>
<tr>
<td>displacement (mm)</td>
<td>.285</td>
<td>1.125</td>
<td>2.5</td>
<td>4.37</td>
<td>6.70</td>
<td>9.45</td>
<td>12.6</td>
</tr>
</tbody>
</table>

The response time of the presser foot advance was therefore estimated to be 7.2 milliseconds.

\[ b) \text{ } \text{Spring return} \]

\[ \ddot{x}_n = 357 - 10.5x_{n-1} - 31.14\dot{x}_{n-1} - 2.932\dot{x}_n^2 \]  
(14)

Using equations 14 and equations 9 and 10 the following data was calculated:

<table>
<thead>
<tr>
<th>time t (milliseconds)</th>
<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>
</tr>
</thead>
<tbody>
<tr>
<td>x (m/sec²)</td>
<td>.357</td>
<td>.343</td>
<td>.326</td>
<td>.305</td>
<td>.281</td>
<td>.254</td>
<td>.226</td>
<td>.196</td>
<td>.164</td>
</tr>
<tr>
<td>( \dot{x} ) (m/sec)</td>
<td>.357</td>
<td>.700</td>
<td>1.026</td>
<td>1.331</td>
<td>1.612</td>
<td>1.866</td>
<td>2.092</td>
<td>2.288</td>
<td>2.442</td>
</tr>
<tr>
<td>( \ddot{x} ) (mm)</td>
<td>.179</td>
<td>.71</td>
<td>1.57</td>
<td>2.75</td>
<td>4.22</td>
<td>5.96</td>
<td>7.94</td>
<td>10.13</td>
<td>12.5</td>
</tr>
</tbody>
</table>

The response time for the presser foot return was therefore estimated to be 9.25 milliseconds.
MATTRESS SEWING MACHINE OPERATING INSTRUCTIONS

1) The needle thread should be laced as shown in Figure 1.

2) Check that the sewing bobbin has sufficient thread and that it is projecting through the needle hole in the throat plate. The machine is not capable of single cycling therefore, if necessary, the bobbin thread must be threaded through the throat plate by hand before placing the bobbin assembly into the rotary hook.

3) Check that the rig isolating valve is in the off position. Switch on the hydraulic supply to the sewing machine and set the relief valve to the following settings:
   a) 30 bar for stitching cardboard or light fabrics.
   b) 40 bar for stitching mattress cover composites.

4) Place materials to be sewn under the needle whilst ensuring that both the needle and bobbin thread ends are placed in the direction of the material feed.

5) Switch the hydraulic supply to the sewing rig using the isolation valve. The actuators will move accordingly.

6) Ensure that the drive to the rotary valve is set at low speed and switch on the hydrostatic transmission. The valve will then commence to rotate and stitches will be formed. The speed of the rotary valve may be increased up to the attenuation speed of the needle mechanism. Attenuation will be evidenced by the occurrence of missed stitches.

General Notes

a) The presser foot is clamped to the rectangular section of its guide by means of a cap head screw. The presser foot can be set at two discrete heights:
   i) The lowest position for stitching thin materials.
   ii) The highest position for stitching mattress composites.
FIG 1  NEEDLE THREADING DIAGRAM
b) The mechanical sewing bobbin assembly is clamped to its drive shaft by three symmetrically placed screws. In the event of a thread seizure it is possible for the bobbin assembly to rotate on its shaft and result in the incorrect timing between the hook point and the hydraulic sequence. The correct timing can be set as follows:

i) Set the main supply pressure to 10 bar and switch on isolating valve to sewing machine.

ii) Rotate the rotary valve bobbin by hand until the needle thread take-up mechanism starts to move.

iii) The needle thread take-up mechanism has a compound movement, continue to rotate the valve until the second stage is initiated.

iv) Set the point of the sewing bobbin hook at the 3 O'clock position when viewed looking at the bobbin assembly.
REFERENCES


TABLE 1
Fig. 1.16

MANIPULATION OF VARIABLES ABOUT THE ACCELERATION DISPLACEMENT OF A DOUBLE-ACTING MECHANISM

<table>
<thead>
<tr>
<th>Scaling Factor Z</th>
<th>Mechanism Response Time x KA / M</th>
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<tr>
<td></td>
<td>Mass</td>
</tr>
<tr>
<td>10</td>
<td>7</td>
</tr>
<tr>
<td>9</td>
<td>6.66</td>
</tr>
<tr>
<td>8</td>
<td>6.4</td>
</tr>
<tr>
<td>7</td>
<td>6.02</td>
</tr>
<tr>
<td>6</td>
<td>5.61</td>
</tr>
<tr>
<td>5</td>
<td>5.2</td>
</tr>
<tr>
<td>4</td>
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<td>.2</td>
<td>2.2</td>
</tr>
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<td>.1</td>
<td>2.1</td>
</tr>
<tr>
<td>Nominal Pressure (Bar)</td>
<td>Practical Results - Forward Movement</td>
</tr>
<tr>
<td>------------------------</td>
<td>-------------------------------------</td>
</tr>
<tr>
<td></td>
<td>Theoretical Results</td>
</tr>
<tr>
<td></td>
<td>Response time (milli-secs)</td>
</tr>
<tr>
<td></td>
<td>Pressure 1bf/in²</td>
</tr>
<tr>
<td></td>
<td>Min</td>
</tr>
<tr>
<td>13.8</td>
<td>200</td>
</tr>
<tr>
<td>27.6</td>
<td>400</td>
</tr>
<tr>
<td>41.4</td>
<td>600</td>
</tr>
<tr>
<td>55.2</td>
<td>800</td>
</tr>
<tr>
<td>69</td>
<td>1000</td>
</tr>
</tbody>
</table>

**TABLE 2**  
ANALYSIS OF PRESSURE AND DISPLACEMENT TRACES - PROTOTYPE TEST RIG - TEST ONE

Fig. 2.07
<table>
<thead>
<tr>
<th>Nominal Pressure</th>
<th>Response time (milliseconds)</th>
<th>Pressure (lbf/in²)</th>
<th>Error (%)</th>
<th>Response time (milliseconds)</th>
<th>Pressure (lbf/in²)</th>
<th>Error (%)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Bar</td>
<td>lbf/in²</td>
<td>Min</td>
<td>Max</td>
<td>Avge.</td>
<td>Min</td>
<td>Max</td>
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<tr>
<td>20.7</td>
<td>300</td>
<td>13</td>
<td>285</td>
<td>310</td>
<td>290</td>
<td>20</td>
</tr>
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<td>27.6</td>
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<td>10.95</td>
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<td>440</td>
<td>390</td>
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<td>41.4</td>
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<td>8.51</td>
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<td>670</td>
<td>600</td>
<td>12</td>
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<td>55.2</td>
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<td>69</td>
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<td>1310</td>
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**TABLE 3**

*Fig. 2.08*

ANALYSIS OF PRESSURE AND DISPLACEMENT TRACES - PROTOTYPE TEST RIG - TEST 2
### TABLE 4

**Fig. 4.03**

**BREAKOUT TORQUE OF COMPUTER BALANCED ROTARY VALVE BOBBIN VERSUS SUPPLY PRESSURE**

<table>
<thead>
<tr>
<th>Pressure P Bar</th>
<th>Torque (Max) 1bf/in² Nm</th>
<th>Torque (Min) 1bf/in² Nm</th>
</tr>
</thead>
<tbody>
<tr>
<td>0</td>
<td>0.15 1.33</td>
<td>0.15 1.33</td>
</tr>
<tr>
<td>6.9</td>
<td>0.65 5.75</td>
<td>0.50 4.42</td>
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<td>13.8</td>
<td>1.30 11.3</td>
<td>1.10 9.75</td>
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<td>20.7</td>
<td>2.05 18.1</td>
<td>1.65 14.5</td>
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<tr>
<td>27.6</td>
<td>3.00 26.6</td>
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<td>47.3</td>
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<td>55.2</td>
<td>9.5 84</td>
<td>6.50 57.5</td>
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</table>

### TABLE 5

**Fig. 4.04**

**DRIVING TORQUE OF UNIVERSAL VALVE VERSUS ROTATIONAL SPEED - NO APPLIED PRESSURE**

<table>
<thead>
<tr>
<th>Valve Speed r.p.m</th>
<th>Torque T = 20°C 1bf/in</th>
<th>Torque T = 20°C Nm</th>
<th>Torque T = 45°C 1bf/in</th>
<th>Torque T = 45°C Nm</th>
</tr>
</thead>
<tbody>
<tr>
<td>100</td>
<td>3.6 0.407</td>
<td>2.8 0.316</td>
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<td>3.75 0.424</td>
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<tr>
<td>200</td>
<td>5.7 0.651</td>
<td>3.75 0.424</td>
<td>7.7 0.868</td>
<td>4.55 0.514</td>
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<tr>
<td>300</td>
<td>7.7 0.868</td>
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<td>400</td>
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<td>5.5 0.622</td>
<td>11.4 1.29</td>
<td>6.4 0.723</td>
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<tr>
<td>500</td>
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<td>13 1.47</td>
<td>7.3 0.825</td>
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<td>13 1.47</td>
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TABLE 6
Figs. 4.02 and 4.03
BREAKOUT TORQUE OF UNIVERSAL VALVE AGAINST SUPPLY PRESSURE -
BALANCE METHOD 2

<table>
<thead>
<tr>
<th>Supply Pressure</th>
<th>Torque Max.</th>
<th>Torque Min.</th>
<th>Tight Spot on Valve</th>
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<tr>
<td></td>
<td>Nm</td>
<td>lbf.in</td>
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TABLE 7
Fig. 4.05
VALVE DRIVING TORQUE VERSUS STATIC SUPPLY PRESSURE

<table>
<thead>
<tr>
<th>Pressure</th>
<th>Valve Speed 500 r.p.m</th>
<th>Valve Speed 100 r.p.m.</th>
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<tbody>
<tr>
<td>lbf/in²</td>
<td>Bar</td>
<td>Nm</td>
</tr>
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<td>100</td>
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<td>600</td>
<td>41.4</td>
<td>1.13</td>
</tr>
<tr>
<td>700</td>
<td>47.7</td>
<td>1.15</td>
</tr>
<tr>
<td>800</td>
<td>55.5</td>
<td>1.17</td>
</tr>
<tr>
<td>900</td>
<td>62.1</td>
<td>1.18</td>
</tr>
<tr>
<td>1000</td>
<td>69</td>
<td>1.2</td>
</tr>
</tbody>
</table>
TABLE 8
Fig. 4.06
VALVE DRIVING TORQUE VERSUS SUPPLY PRESSURE - PULSATING FLOW

<table>
<thead>
<tr>
<th>Valve Speed r.p.m</th>
<th>Nom</th>
<th>Driving Torque N.m.</th>
<th>Valve speed @ 55 bar (r.p.m)</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td>Max</td>
<td>Min</td>
</tr>
<tr>
<td>100</td>
<td>.41</td>
<td>.41</td>
<td>.41</td>
</tr>
<tr>
<td>200</td>
<td>.68</td>
<td>.68</td>
<td>.68</td>
</tr>
<tr>
<td>300</td>
<td>.85</td>
<td>.85</td>
<td>.85</td>
</tr>
<tr>
<td>400</td>
<td>.98</td>
<td>.98</td>
<td>1.01</td>
</tr>
<tr>
<td>500</td>
<td>1.09</td>
<td>1.09</td>
<td>1.22</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Supply Pressure lbf/in²</th>
<th>200</th>
<th>400</th>
<th>600</th>
<th>800</th>
</tr>
</thead>
<tbody>
<tr>
<td>Bar</td>
<td>13.8</td>
<td>27.6</td>
<td>41.4</td>
<td>55.2</td>
</tr>
</tbody>
</table>

TABLE 9
Fig. 4.07
INTERNAL LEAKAGE OF UNIVERSAL VALVE AGAINST SUPPLY PRESSURE

<table>
<thead>
<tr>
<th>Supply Pressure Bar</th>
<th>Oil Temp. °C</th>
<th>Mass Flow grammes/min Stationary</th>
<th>100 r.p.m.</th>
</tr>
</thead>
<tbody>
<tr>
<td>6.9</td>
<td>20</td>
<td>26</td>
<td>25</td>
</tr>
<tr>
<td>13.8</td>
<td>22</td>
<td>60</td>
<td>56</td>
</tr>
<tr>
<td>20.7</td>
<td>24</td>
<td>90</td>
<td>86</td>
</tr>
<tr>
<td>27.6</td>
<td>25</td>
<td>148</td>
<td>138</td>
</tr>
<tr>
<td>34.5</td>
<td>27</td>
<td>180</td>
<td>180</td>
</tr>
<tr>
<td>41.4</td>
<td>28</td>
<td>233</td>
<td>235</td>
</tr>
<tr>
<td>47.3</td>
<td>29</td>
<td>306</td>
<td>300</td>
</tr>
<tr>
<td>55.2</td>
<td>30</td>
<td>387</td>
<td>390</td>
</tr>
<tr>
<td>62.1</td>
<td>32</td>
<td>512</td>
<td>508</td>
</tr>
<tr>
<td>69</td>
<td>34</td>
<td>646</td>
<td>640</td>
</tr>
</tbody>
</table>
### TABLE 10
Fig. 4.08
PRESSURE DROP ACROSS UNIVERSAL VALVE AGAINST FLOW RATE

<table>
<thead>
<tr>
<th>Flow litres/min</th>
<th>U.V. Deflection mm</th>
<th>Pressure Drop lbf/in²</th>
<th>Bar</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>0.5</td>
<td>0.25</td>
<td>0.017</td>
</tr>
<tr>
<td>2</td>
<td>1.5</td>
<td>0.6</td>
<td>0.041</td>
</tr>
<tr>
<td>3</td>
<td>3.5</td>
<td>1.2</td>
<td>0.082</td>
</tr>
<tr>
<td>4</td>
<td>6.5</td>
<td>2.5</td>
<td>0.17</td>
</tr>
<tr>
<td>5</td>
<td>8.2</td>
<td>3.0</td>
<td>0.204</td>
</tr>
<tr>
<td>6</td>
<td>11.5</td>
<td>4.25</td>
<td>0.306</td>
</tr>
<tr>
<td>7</td>
<td>14.5</td>
<td>5.5</td>
<td>0.374</td>
</tr>
<tr>
<td>8</td>
<td>18.5</td>
<td>7.0</td>
<td>0.476</td>
</tr>
<tr>
<td>9</td>
<td>23</td>
<td>8.6</td>
<td>0.585</td>
</tr>
<tr>
<td>10</td>
<td>28</td>
<td>10.5</td>
<td>0.714</td>
</tr>
</tbody>
</table>

### TABLE 11
Fig. 4.07
THEORETICAL LEAKAGE OF UNIVERSAL VALVE BASED ON VALVE METROLOGY

<table>
<thead>
<tr>
<th>Supply Bar</th>
<th>Pressure lbf/in²</th>
<th>Oil Temp °C</th>
<th>Dyn. Viscosity N.sec/m²</th>
<th>Mass Flow gms/min Conc. Ecct. c</th>
</tr>
</thead>
<tbody>
<tr>
<td>6.9</td>
<td>100</td>
<td>20</td>
<td>.070</td>
<td>9.35</td>
</tr>
<tr>
<td>13.8</td>
<td>200</td>
<td>22</td>
<td>.066</td>
<td>21.1</td>
</tr>
<tr>
<td>20.7</td>
<td>300</td>
<td>24</td>
<td>.063</td>
<td>33.4</td>
</tr>
<tr>
<td>27.6</td>
<td>400</td>
<td>25</td>
<td>.059</td>
<td>52.2</td>
</tr>
<tr>
<td>34.5</td>
<td>500</td>
<td>27</td>
<td>.055</td>
<td>72</td>
</tr>
<tr>
<td>41.4</td>
<td>600</td>
<td>28</td>
<td>.052</td>
<td>98</td>
</tr>
<tr>
<td>47.3</td>
<td>700</td>
<td>29</td>
<td>.050</td>
<td>181</td>
</tr>
<tr>
<td>55.2</td>
<td>800</td>
<td>30</td>
<td>.048</td>
<td>156</td>
</tr>
<tr>
<td>62.1</td>
<td>900</td>
<td>32</td>
<td>.044</td>
<td>208</td>
</tr>
<tr>
<td>69</td>
<td>1000</td>
<td>34</td>
<td>.040</td>
<td>260</td>
</tr>
</tbody>
</table>
TABLE 12  
Fig. 4.10  
METROLOGY OF PROTOTYPE 1 VALVE CASING AND BOBBIN  

a) Valve Casing Data  

<table>
<thead>
<tr>
<th>Dimension from end 1 (mm)</th>
<th>Major diameter (ins)</th>
<th>Talyrond Max</th>
<th>Talyrond Min</th>
<th>Ovality (ins)</th>
</tr>
</thead>
<tbody>
<tr>
<td>3</td>
<td>1.9976</td>
<td>98</td>
<td>60</td>
<td>0.00038</td>
</tr>
<tr>
<td>19</td>
<td>1.9976</td>
<td>86</td>
<td>63</td>
<td>0.00045</td>
</tr>
<tr>
<td>30</td>
<td>1.9976</td>
<td>91</td>
<td>54</td>
<td>0.00070</td>
</tr>
<tr>
<td>37</td>
<td>1.9974</td>
<td>94</td>
<td>58</td>
<td>0.00071</td>
</tr>
<tr>
<td>56</td>
<td>1.9971</td>
<td>88</td>
<td>67</td>
<td>0.00031</td>
</tr>
<tr>
<td>62</td>
<td>1.9969</td>
<td>87</td>
<td>67</td>
<td>0.00031</td>
</tr>
<tr>
<td>78</td>
<td>1.9982</td>
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<td></td>
</tr>
<tr>
<td>81</td>
<td>1.9982</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>108</td>
<td>1.9982</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>127</td>
<td>1.9982</td>
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<td></td>
</tr>
<tr>
<td>133</td>
<td>1.9982</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>150</td>
<td>1.9982</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>160</td>
<td>1.9982</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>170</td>
<td>1.9982</td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

b) Valve Bobbin Data  

Diameter of end 1 = 1.9958 ins  
Diameter of end 2 = 1.9956 ins  
Maximum eccentricity = ±0.00015 ins
### TABLE 13
METROLOGY OF SECOND PROTOTYPE UNIVERSAL VALVE CASING

<table>
<thead>
<tr>
<th>Dimension from end 1 (in)</th>
<th>Mean diameter (in)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>1.9885</td>
</tr>
<tr>
<td>2</td>
<td>1.9884</td>
</tr>
<tr>
<td>3</td>
<td>1.9885</td>
</tr>
<tr>
<td>4</td>
<td>1.9885</td>
</tr>
<tr>
<td>5</td>
<td>1.9885</td>
</tr>
<tr>
<td>6</td>
<td>1.9888</td>
</tr>
</tbody>
</table>

Note: All measurements were taken in the same plane; mean ovality = 0.0002 in

### TABLE 14
DRIVING TORQUE OF SECOND PROTOTYPE UNIVERSAL VALVE - \( T = 25^\circ C \)

<table>
<thead>
<tr>
<th>Valve Speed r.p.m.</th>
<th>Driving Torque</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>lbf.in</td>
</tr>
<tr>
<td>100</td>
<td>3.30</td>
</tr>
<tr>
<td>200</td>
<td>4.50</td>
</tr>
<tr>
<td>300</td>
<td>5.90</td>
</tr>
<tr>
<td>400</td>
<td>7.35</td>
</tr>
<tr>
<td>500</td>
<td>8.60</td>
</tr>
<tr>
<td>600</td>
<td>9.90</td>
</tr>
</tbody>
</table>

Note: The valve was subjected to both static pressure and pulsating flow without any discernible change in driving torque.
### TABLE 15

**Balancing Summary of Prototype Universal Valve: Balance Method 1**

**Checked by Method 2**

<table>
<thead>
<tr>
<th>Area</th>
<th>Coordinates</th>
<th>Sign</th>
<th>Force N</th>
<th>X mm</th>
<th>Moment Nmm</th>
<th>Sign</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>$\theta_1 = 26.29 \ \theta_2 = 44.76$</td>
<td>+</td>
<td>556</td>
<td>53.56</td>
<td>29770</td>
<td>+</td>
</tr>
<tr>
<td>2</td>
<td>$\theta_1 = 49.62 \ \theta_2 = 86.28$</td>
<td>-</td>
<td>1646</td>
<td>21.65</td>
<td>35640</td>
<td>-</td>
</tr>
<tr>
<td>3</td>
<td>$\theta = 94 \ W = 9$</td>
<td>+</td>
<td>490</td>
<td>9.32</td>
<td>4569</td>
<td>-</td>
</tr>
<tr>
<td>4</td>
<td>$\theta_1 = 94 \ \theta_2 = 121.92$</td>
<td>+</td>
<td>1286</td>
<td>27</td>
<td>34785</td>
<td>-</td>
</tr>
<tr>
<td>5</td>
<td>$\theta_1 = 56 \ \theta_2 = 129 \ W = 5$</td>
<td>+</td>
<td>10</td>
<td>42.9</td>
<td>430</td>
<td>-</td>
</tr>
<tr>
<td>6</td>
<td>$\theta_1 = 128.76 \ W = 5$</td>
<td>-</td>
<td>195</td>
<td>47.8</td>
<td>9332</td>
<td>+</td>
</tr>
<tr>
<td>7</td>
<td>$\theta_1 = 128.76 \ \theta_2 = 138$</td>
<td>-</td>
<td>433</td>
<td>53.4</td>
<td>23135</td>
<td>+</td>
</tr>
<tr>
<td>8</td>
<td>$\theta = 138 \ W = 4.5$</td>
<td>-</td>
<td>167</td>
<td>61.4</td>
<td>10249</td>
<td>+</td>
</tr>
<tr>
<td>a</td>
<td>$\theta = 28.44$</td>
<td>+</td>
<td>61</td>
<td>65.4</td>
<td>4026</td>
<td>+</td>
</tr>
<tr>
<td>b</td>
<td>$\theta_2 = 45.72 \ \theta_1 = 28.44$</td>
<td>+</td>
<td>75</td>
<td>57</td>
<td>4290</td>
<td>+</td>
</tr>
<tr>
<td>c</td>
<td>$\theta = 45.72 \ \theta_1 = 5$</td>
<td>+</td>
<td>23</td>
<td>46.4</td>
<td>1050</td>
<td>+</td>
</tr>
<tr>
<td>d</td>
<td>$\theta = 45.72$</td>
<td>+</td>
<td>91</td>
<td>43.14</td>
<td>3940</td>
<td>+</td>
</tr>
<tr>
<td>e</td>
<td>$\theta = 51.75$</td>
<td>-</td>
<td>100</td>
<td>42.1</td>
<td>4193</td>
<td>-</td>
</tr>
<tr>
<td>f</td>
<td>$\theta_2 = 91.75 \ \theta_1 = 51.75$</td>
<td>-</td>
<td>67</td>
<td>29.27</td>
<td>1970</td>
<td>-</td>
</tr>
<tr>
<td>g</td>
<td>$\theta = 88.7$</td>
<td>+</td>
<td>126</td>
<td>6.16</td>
<td>779</td>
<td>+</td>
</tr>
<tr>
<td>h</td>
<td>$\theta = 86$</td>
<td>-</td>
<td>126</td>
<td>6.16</td>
<td>779</td>
<td>+</td>
</tr>
<tr>
<td>i</td>
<td>$\theta = 86 \ L = 3$</td>
<td>-</td>
<td>4</td>
<td>9</td>
<td>36</td>
<td>+</td>
</tr>
<tr>
<td>j</td>
<td>$\theta_2 = 86 \ \theta_1 = 53.75$</td>
<td>-</td>
<td>67</td>
<td>29.27</td>
<td>1970</td>
<td>+</td>
</tr>
<tr>
<td>k</td>
<td>$\theta = 53.75$</td>
<td>-</td>
<td>100</td>
<td>43.73</td>
<td>4370</td>
<td>+</td>
</tr>
<tr>
<td>l</td>
<td>$\theta = 46$</td>
<td>+</td>
<td>91</td>
<td>42.07</td>
<td>3828</td>
<td>-</td>
</tr>
<tr>
<td>m</td>
<td>$\theta = 46 \ L = 8$</td>
<td>+</td>
<td>23</td>
<td>45.33</td>
<td>1042</td>
<td>-</td>
</tr>
<tr>
<td>n</td>
<td>$\theta_2 = 46 \ \theta_1 = 36.89$</td>
<td>+</td>
<td>40</td>
<td>53.735</td>
<td>2150</td>
<td>-</td>
</tr>
<tr>
<td>o</td>
<td>$\theta = 36.89 \ \theta_1 = 7$</td>
<td>+</td>
<td>20</td>
<td>62.4</td>
<td>1243</td>
<td>-</td>
</tr>
<tr>
<td>p</td>
<td>$\theta = 141.5$</td>
<td>-</td>
<td>76</td>
<td>65.4</td>
<td>4970</td>
<td>+</td>
</tr>
</tbody>
</table>

Balance: $-89 \ \text{N}$

**Static imbalance**: $F_2 + F_4 > F_1 + F_3 \text{ by } 89 \ \text{N}$

**Moment imbalance**: $M_{14} > M_{23} \text{ by } 8847 \ \text{N mm}$

**Bearing reactions**: $53 \pm \frac{89}{2} \ \text{N/unit pressure}$
TABLE 16  
BALANCING SUMMARY OF BOBBIN BALANCED BY METHOD 2

<table>
<thead>
<tr>
<th>Area</th>
<th>Coordinates</th>
<th>Sign</th>
<th>Force</th>
<th>( \bar{x} )</th>
<th>Moment</th>
<th>Sign</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>( \theta_1 = 26.3^\circ ), ( \theta_2 = 44.8^\circ )</td>
<td>+</td>
<td>545</td>
<td>57.5</td>
<td>31300</td>
<td>+</td>
</tr>
<tr>
<td>2</td>
<td>( \theta_1 = 50^\circ ), ( \theta_2 = 86^\circ )</td>
<td>-</td>
<td>1720</td>
<td>23.1</td>
<td>39800</td>
<td>-</td>
</tr>
<tr>
<td>3</td>
<td>( \theta = 90^\circ ), ( W = 9.62 )</td>
<td>+</td>
<td>485</td>
<td>9.31</td>
<td>4550</td>
<td>-</td>
</tr>
<tr>
<td>4</td>
<td>( \theta_1 = 90.74^\circ ), ( \theta_2 = 118.84^\circ )</td>
<td>+</td>
<td>1435</td>
<td>28.5</td>
<td>40950</td>
<td>-</td>
</tr>
<tr>
<td>5</td>
<td>( \theta_1 = 127^\circ ), ( \theta_2 = 60 )</td>
<td>+</td>
<td>7</td>
<td>45.5</td>
<td>320</td>
<td>-</td>
</tr>
<tr>
<td>6</td>
<td>( \theta = 127^\circ ), ( W = 5.26 )</td>
<td>-</td>
<td>218</td>
<td>50</td>
<td>10900</td>
<td>+</td>
</tr>
<tr>
<td>7</td>
<td>( \theta_1 = 127^\circ ), ( \theta_2 = 137.4 )</td>
<td>-</td>
<td>415</td>
<td>57.9</td>
<td>24000</td>
<td>+</td>
</tr>
<tr>
<td>8</td>
<td>( \theta = 137.4^\circ ), ( W = 3 )</td>
<td>-</td>
<td>113</td>
<td>64.7</td>
<td>7300</td>
<td>+</td>
</tr>
<tr>
<td>a</td>
<td>( \theta = 29.84^\circ )</td>
<td>+</td>
<td>64</td>
<td>67.9</td>
<td>4310</td>
<td>+</td>
</tr>
<tr>
<td>b</td>
<td>( \theta_1 = 30^\circ ), ( \theta_2 = 45 )</td>
<td>+</td>
<td>66</td>
<td>59</td>
<td>3910</td>
<td>+</td>
</tr>
<tr>
<td>c</td>
<td>( \theta = 450^\circ ), ( l = 4 )</td>
<td>+</td>
<td>14</td>
<td>47</td>
<td>660</td>
<td>+</td>
</tr>
<tr>
<td>d</td>
<td>( \theta = 45^\circ )</td>
<td>+</td>
<td>90</td>
<td>45.6</td>
<td>4090</td>
<td>+</td>
</tr>
<tr>
<td>e</td>
<td>( \theta = 53.9^\circ )</td>
<td>-</td>
<td>103</td>
<td>43.2</td>
<td>4420</td>
<td>-</td>
</tr>
<tr>
<td>f</td>
<td>( \theta_1 = 53.9^\circ ), ( \theta_2 = 90^\circ )</td>
<td>-</td>
<td>62</td>
<td>31.5</td>
<td>1960</td>
<td>-</td>
</tr>
<tr>
<td>g</td>
<td>( \theta = 90^\circ )</td>
<td>+</td>
<td>126</td>
<td>6.16</td>
<td>780</td>
<td>+</td>
</tr>
<tr>
<td>h</td>
<td>( \theta = 90^\circ )</td>
<td>-</td>
<td>126</td>
<td>6.16</td>
<td>780</td>
<td>+</td>
</tr>
<tr>
<td>i</td>
<td>( \theta = 88^\circ ), ( l = 7 )</td>
<td>-</td>
<td>2</td>
<td>10</td>
<td>20</td>
<td>+</td>
</tr>
<tr>
<td>j</td>
<td>( \theta_1 = 54.5^\circ ), ( \theta_2 = 88^\circ )</td>
<td>-</td>
<td>64</td>
<td>35</td>
<td>2240</td>
<td>+</td>
</tr>
<tr>
<td>k</td>
<td>( \theta = 54.5^\circ )</td>
<td>-</td>
<td>96</td>
<td>45.5</td>
<td>4360</td>
<td>+</td>
</tr>
<tr>
<td>l</td>
<td>( \theta = 49^\circ )</td>
<td>+</td>
<td>98</td>
<td>44.6</td>
<td>4360</td>
<td>-</td>
</tr>
<tr>
<td>m</td>
<td>( \theta = 49^\circ ), ( l = 7.5 )</td>
<td>+</td>
<td>24</td>
<td>49.5</td>
<td>1190</td>
<td>-</td>
</tr>
<tr>
<td>n</td>
<td>( \theta_2 = 49^\circ ), ( \theta_1 = 37^\circ )</td>
<td>+</td>
<td>36</td>
<td>58.5</td>
<td>2110</td>
<td>-</td>
</tr>
<tr>
<td>o</td>
<td>( \theta = 37^\circ ), ( l = 5 )</td>
<td>+</td>
<td>19</td>
<td>65.5</td>
<td>1240</td>
<td>-</td>
</tr>
<tr>
<td>p</td>
<td>( \theta = 143^\circ )</td>
<td>-</td>
<td>80</td>
<td>67.9</td>
<td>5390</td>
<td>+</td>
</tr>
</tbody>
</table>

Balance \( F_1 + F_3 > F_2 + F_4 \) by 10N

\[ M_{23} > M_{14} \] by 900 N mm

Bearing reactions = 5 ± 5.6 N
TABLE 17
BALANCING SUMMARY OF INCORRECTLY MACHINED BOBBIN

<table>
<thead>
<tr>
<th>Area/Land</th>
<th>Data</th>
<th>±</th>
<th>Force N</th>
<th>Moment Nmm</th>
<th>( \overline{x} )</th>
<th>±</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>( \theta_1 = 26.24 ) ( \theta_2 = 48.1 )</td>
<td>+</td>
<td>683.47</td>
<td>37864</td>
<td>55.4</td>
<td>+</td>
</tr>
<tr>
<td>2</td>
<td>( \theta_1 = 54.42 ) ( \theta_2 = 87.46 )</td>
<td>-</td>
<td>1603</td>
<td>33623</td>
<td>21</td>
<td>-</td>
</tr>
<tr>
<td>3</td>
<td>( \theta_1 = 94^\circ ) ( W = 9.5 )</td>
<td>+</td>
<td>480.7</td>
<td>4444</td>
<td>9.25</td>
<td>-</td>
</tr>
<tr>
<td>4</td>
<td>( \theta_1 = 94.26 ) ( \theta_2 = 1190 )</td>
<td>+</td>
<td>1224.6</td>
<td>32452</td>
<td>26.5</td>
<td>-</td>
</tr>
<tr>
<td>5</td>
<td>( W = 45 ) ( \theta_1 = 60 ) ( \theta_2 = 126 )</td>
<td>+</td>
<td>13</td>
<td>546</td>
<td>42</td>
<td>-</td>
</tr>
<tr>
<td>6</td>
<td>( \theta = 126 ) ( W = 6 )</td>
<td>-</td>
<td>246.1</td>
<td>11641</td>
<td>47.3</td>
<td>+</td>
</tr>
<tr>
<td>7</td>
<td>( \theta_1 = 126 ) ( \theta_2 = 137.5 ) ( x_1 = 133.5 ) ( x_2 = 245.5 ) ( \theta_1 = 129^\circ )</td>
<td>-</td>
<td>446</td>
<td>25196</td>
<td>56.5</td>
<td>+</td>
</tr>
<tr>
<td>8</td>
<td>( W = 5 ) ( \theta = 137.5 )</td>
<td>-</td>
<td>171.26</td>
<td>11132</td>
<td>65</td>
<td>+</td>
</tr>
<tr>
<td>a</td>
<td>( \theta = 29^\circ )</td>
<td>+</td>
<td>61.5</td>
<td>4271</td>
<td>69.5</td>
<td>+</td>
</tr>
<tr>
<td>b</td>
<td>( \theta_2 = 49 ) ( \theta_1 = 29 )</td>
<td>+</td>
<td>86.6</td>
<td>5068</td>
<td>58.5</td>
<td>+</td>
</tr>
<tr>
<td>c</td>
<td>( \theta = 49 ) ( \theta_1 = 49 )</td>
<td>+</td>
<td>6.56</td>
<td>295</td>
<td>45</td>
<td>+</td>
</tr>
<tr>
<td>d</td>
<td>( \theta = 49 )</td>
<td>+</td>
<td>95.6</td>
<td>4162</td>
<td>43.5</td>
<td>+</td>
</tr>
<tr>
<td>e</td>
<td>( \theta = 56 )</td>
<td>-</td>
<td>105</td>
<td>4287</td>
<td>40.8</td>
<td>-</td>
</tr>
<tr>
<td>f</td>
<td>( \theta_2 = 87 ) ( \theta_1 = 56 )</td>
<td>-</td>
<td>54.5</td>
<td>1197</td>
<td>22</td>
<td>-</td>
</tr>
<tr>
<td>g</td>
<td>( \theta = 93 )</td>
<td>+</td>
<td>126.6</td>
<td>780</td>
<td>6.16</td>
<td>+</td>
</tr>
<tr>
<td>h</td>
<td>( \theta = 82.7^\circ )</td>
<td>-</td>
<td>125.7</td>
<td>775</td>
<td>6.17</td>
<td>+</td>
</tr>
<tr>
<td>i</td>
<td>( \theta = 82.7 )</td>
<td>-</td>
<td>5.08</td>
<td>47</td>
<td>9.25</td>
<td>+</td>
</tr>
<tr>
<td>j</td>
<td>( \theta_2 = 82.7 ) ( \theta_1 = 53.6 )</td>
<td>-</td>
<td>60.1</td>
<td>1740</td>
<td>29</td>
<td>+</td>
</tr>
<tr>
<td>k</td>
<td>( \theta = 53.6 )</td>
<td>-</td>
<td>102</td>
<td>4335</td>
<td>42.5</td>
<td>+</td>
</tr>
<tr>
<td>l</td>
<td>( \theta = 49^\circ )</td>
<td>+</td>
<td>95.6</td>
<td>3970</td>
<td>41.5</td>
<td>-</td>
</tr>
<tr>
<td>m</td>
<td>( \theta = 49 ) ( L = 8 )</td>
<td>+</td>
<td>26.24</td>
<td>1207</td>
<td>46</td>
<td>-</td>
</tr>
<tr>
<td>n</td>
<td>( \theta_2 = 49 ) ( \theta_1 = 38 )</td>
<td>+</td>
<td>44.6</td>
<td>2522</td>
<td>56.5</td>
<td>-</td>
</tr>
<tr>
<td>o</td>
<td>( \theta = 38 ) ( L = 7 )</td>
<td>+</td>
<td>27.58</td>
<td>1820</td>
<td>66</td>
<td>-</td>
</tr>
<tr>
<td>p</td>
<td>( \theta = 142 )</td>
<td>-</td>
<td>78</td>
<td>5384</td>
<td>69</td>
<td>+</td>
</tr>
<tr>
<td>Balance</td>
<td>-</td>
<td>24</td>
<td>26622</td>
<td>+</td>
<td>+</td>
<td>+</td>
</tr>
</tbody>
</table>

Static imbalance \( F_1 + F_3 > F_2 + F_4 \) by 24 N
Moment imbalance \( M_{14} > M_{23} \) by 26622 N mm
Bearing reactions = 12 ± 166 N/unit pressure
TABLE 18
SUMMARY OF NEEDLE MECHANISM DATA OF PROTOTYPE RIG

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Actuator diameter</td>
<td>( d = 8 \text{ mm} )</td>
</tr>
<tr>
<td>Combined engagement length of actuators</td>
<td>( L = 100 \text{ mm} )</td>
</tr>
<tr>
<td>Contact area of needle slide</td>
<td>( G = 3750 \text{ mm}^2 )</td>
</tr>
<tr>
<td>Contact area of support bush</td>
<td>( G = 850 \text{ mm}^2 )</td>
</tr>
</tbody>
</table>

Mechanism Masses

1. Actuators, slide, needle bar etc. \( m_1 = 275 \text{ gm} \)
2. Pipes - valve to actuators \( m_2 = 554 \text{ gm} \)
3. Inlet pipe to rotary valve \( m_3 = 145 \text{ gm} \)
4. Exhaust pipe from rotary valve \( m_4 = 295 \text{ gm} \)
5. Total overall mechanism mass \( m = 1.270 \text{ kg} \)

Viscous Coefficients \((\text{N.Sec/m}^3) \times 10^{-6}\)

1. Needle slide \( k_1 = 3.73 \mu \times 10^6 \)
2. Actuators \( k_2 = 2.5 \mu \times 10^6 \)
3. Support bush \( k_3 = 0.85 \mu \times 10^6 \)
4. Pipes - valve to actuators \( k_4 = 21 \mu \times 10^6 \)
5. Inlet pipe to rotary valve \( k_5 = 2.68 \mu \times 10^6 \)
6. Exhaust pipe \( k_6 = 11.23 \mu \times 10^6 \)

Total viscous coefficient \( k = 42 \mu \times 10^6 \)

Momentum Coefficients

1. Rotary valve \( c_1 = 1050 \)
2. Pipes - valves to actuators \( c_2 = 98500 \)
3. Inlet pipe \( c_3 = 9460 \)
4. Exhaust pipe \( c_4 = 24600 \)

Total momentum coefficient \( c = 133600 \)
### TABLE 19
### SUMMARY OF NEEDLE YARN TAKE-UP MECHANISM DATA OF PROTOTYPE SEWING RIG

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Actuator diameter</td>
<td>$d = 5 \text{ mm}$</td>
</tr>
<tr>
<td>Maximum actuator displacement</td>
<td>$x = 25 \text{ mm}$</td>
</tr>
<tr>
<td>Combined engagement length of actuators</td>
<td>$l = 75 \text{ mm}$</td>
</tr>
<tr>
<td>Contact area of slideway</td>
<td>$G = 1350 \text{ mm}^2$</td>
</tr>
</tbody>
</table>

**Mechanism Masses**

1. Actuators, slides, lever, pin, etc. $m_1 = 65 \text{ gm}$
2. Pipes - valves/actuators $m_2 = 375 \text{ gm}$
3. Inlet pipe to rotary valve $m_3 = 25 \text{ gm}$
4. Exhaust pipe from rotary valve $m_4 = 45 \text{ gm}$

Total overall mechanism mass $M = 0.51 \text{ kg}$

**Viscous Coefficients** (N.Sec/m³)

1. Slideway $k_1 = 3.44 \mu \times 10^6$
2. Pipes - valve/actuators $k_2 = 34 \mu \times 10^6$
3. Actuators $k_3 = 3 \mu \times 10^6$
4. Inlet pipe to rotary valve $k_4 = 0.03 \mu \times 10^6$
5. Exhaust pipe from rotary valve $k_5 = 4.39 \mu \times 10^6$

Total viscous coefficient $k = 48 \mu \times 10^6$

**Momentum Coefficients**

1. Rotary valve $c_1 = 525$
2. Pipes - valves/actuators $c_2 = 71000$
3. Inlet pipe to rotary valve $c_3 = 1437$
4. Exhaust pipe from rotary valve $c_4 = 751$

Total momentum coefficient $c = 73675$
<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Actuator diameter</td>
<td>( d = 10 \text{ mm} )</td>
</tr>
<tr>
<td>Maximum displacement of actuator</td>
<td>( x = 13 \text{ mm} )</td>
</tr>
<tr>
<td>Average engagement length of actuator</td>
<td>( l = 40 \text{ mm} )</td>
</tr>
<tr>
<td>Spring stiffness</td>
<td>( S = 3416 \text{ N/m} )</td>
</tr>
<tr>
<td>Initial spring compression</td>
<td>( B = 10 \text{ mm} )</td>
</tr>
<tr>
<td>Contact area of support bush</td>
<td>( G = 1070 \text{ mm}^2 )</td>
</tr>
</tbody>
</table>

**Mechanism Masses**

1. Piston, spring, slide bar, presser foot etc. \( m_1 = 130 \text{ gm} \)
2. Pipe - valve/actuator \( m_2 = 570 \text{ gm} \)
3. Inlet pipe to rotary valve \( m_3 = 350 \text{ gm} \)
4. Exhaust pipe from rotary valve \( m_4 = 420 \text{ gm} \)
5. Total overall mechanism mass out \( M = 1.05 \text{ kg} \)
6. Total overall mechanism mass in \( M = 1.12 \text{ kg} \)

**Viscous Coefficients (N.Sec/m³)**

1. Actuator \( k_1 = 0.8\mu \times 10^5 \)
2. Support bush \( k_2 = 0.34\mu \times 10^5 \)
3. Pipe - valve/actuator \( k_3 = 13.92\mu \times 10^6 \)
4. Inlet pipe to rotary valve \( k_4 = 4.2\mu \times 10^6 \)
5. Exhaust pipe from rotary valve \( k_5 = 10.55\mu \times 10^6 \)

Total viscous coefficient out \( (k_4 + k_1 + \ldots) \) \( k = 18.23\mu \times 10^6 \)

Total viscous coefficient in \( (k_5 + k_1 + \ldots) \) \( k = 24.5\mu \times 10^6 \)

**Momentum Coefficients**

1. Rotary valve \( c_1 = 1050 \)
2. Pipe - valve/actuator \( c_2 = 72143 \)
3. Inlet pipe \( c_3 = 22990 \)
4. Exhaust pipe \( c_4 = 60119 \)

Total momentum coefficient \( c = 1.563 \times 10^5 \)
TABLE 21
SUMMARY OF NEEDLE MECHANISM DATA OF UPRATED SEWING MACHINE

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Actuator diameter d</td>
<td>8 mm</td>
</tr>
<tr>
<td>Combined engagement length of actuators</td>
<td>100 mm</td>
</tr>
<tr>
<td>Contact area of needle slide G</td>
<td>3750 mm²</td>
</tr>
<tr>
<td>Contact area of support bush G</td>
<td>850 mm²</td>
</tr>
<tr>
<td>Mechanism Masses</td>
<td></td>
</tr>
<tr>
<td>1. Actuators, slide, needle bar etc.</td>
<td>275 gm</td>
</tr>
<tr>
<td>2. Pipes - valve/actuators</td>
<td>270 gm</td>
</tr>
<tr>
<td>3. Inlet pipe to rotary valve</td>
<td>36 gm</td>
</tr>
<tr>
<td>4. Exhaust pipe from rotary valve</td>
<td>36 gm</td>
</tr>
<tr>
<td>Total overall mechanism mass M</td>
<td>0.617 kg</td>
</tr>
<tr>
<td>Viscous Coefficients (N.Sec/m³)</td>
<td></td>
</tr>
<tr>
<td>1. Needle slide</td>
<td>2μ x 10⁶</td>
</tr>
<tr>
<td>2. Actuators</td>
<td>2.5μ x 10⁶</td>
</tr>
<tr>
<td>3. Support bush</td>
<td>0.85μ x 10⁶</td>
</tr>
<tr>
<td>4. Pipe - valve/actuators</td>
<td>5μ x 10⁶</td>
</tr>
<tr>
<td>5. Inlet pipe to rotary valve</td>
<td>1.67μ x 10⁶</td>
</tr>
<tr>
<td>6. Exhaust pipe from rotary valve</td>
<td>1.67μ x 10⁶</td>
</tr>
<tr>
<td>Total viscous coefficient k</td>
<td>10.68μ x 10⁶</td>
</tr>
<tr>
<td>Momentum Coefficients</td>
<td></td>
</tr>
<tr>
<td>1. Rotary valve</td>
<td>1050</td>
</tr>
<tr>
<td>2. Pipe - valve/actuators</td>
<td>23541</td>
</tr>
<tr>
<td>3. Inlet pipe to rotary valve</td>
<td>367</td>
</tr>
<tr>
<td>4. Exhaust pipe from rotary valve</td>
<td>367</td>
</tr>
<tr>
<td>Total momentum coefficient c</td>
<td>25326</td>
</tr>
</tbody>
</table>
### TABLE 22
**SUMMARY OF NEEDLE YARN TAKE-UP MECHANISM OF UPRATED SEWING MACHINE**

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Actuator diameter</td>
<td>( d = 5 \text{ mm} )</td>
</tr>
<tr>
<td>Maximum actuator displacement</td>
<td>( x = 25 \text{ mm} )</td>
</tr>
<tr>
<td>Combined engagement length of actuators</td>
<td>( \varepsilon = 75 \text{ mm} )</td>
</tr>
<tr>
<td>Contact area of slideway</td>
<td>( G = 1350 \text{ mm} )</td>
</tr>
</tbody>
</table>

#### Mechanism Masses
1. Actuators, slide, lever, pins etc.           | \( m_1 = 65 \text{ gm} \) |
2. Pipes - actuators/valve                       | \( m_2 = 84 \text{ gm} \) |
3. Inlet pipe to rotary valve                    | \( m_3 = 5.5 \text{ gm} \) |
4. Exhaust pipe from rotary valve                | \( m_4 = 5.5 \text{ gm} \) |
5. Total overall mechanism mass                 | \( M = 0.16 \text{ kg} \) |

#### Viscous Coefficients (N.Sec/m³)
1. Slideway                                      | \( k_1 = 3\mu \times 10^6 \) |
2. Pipe - valve/actuators                        | \( k_2 = 8.25\mu \times 10^6 \) |
3. Inlet pipe to rotary valve                    | \( k_3 = \text{Insignificant} \) |
4. Actuators                                    | \( k_4 = 3\mu \times 10^6 \) |
5. Exhaust pipe from rotary valve                | \( k_5 = \text{Insignificant} \) |
6. Total viscous coefficient                    | \( k = 14.25\mu \times 10^6 \) |

#### Momentum Coefficients
1. Rotary valve                                 | \( c_1 = 1050 \) |
2. Pipes - valve/actuators                       | \( c_2 = 15029 \) |
3. Inlet pipe to rotary valve                    | \( c_3 = 56 \) |
4. Exhaust pipe from rotary valve                | \( c_4 = 56 \) |
5. Total momentum coefficient                    | \( c = 16191 \) |
### TABLE 23
**SUMMARY OF PRESSER FOOT MECHANISM DATA OF UPRATED SEWING MACHINE**

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Actuator diameter</td>
<td>d = 10 mm</td>
</tr>
<tr>
<td>Average engagement length of piston</td>
<td>e = 40 mm</td>
</tr>
<tr>
<td>Maximum displacement of actuator</td>
<td>( \bar{x} = 13 , \text{mm} )</td>
</tr>
<tr>
<td>Spring stiffness</td>
<td>( S = 5250 , \text{N/m} )</td>
</tr>
<tr>
<td>Initial spring compression</td>
<td>( B = 21 , \text{mm} )</td>
</tr>
<tr>
<td>Contact area of support bush</td>
<td>( G = 1070 , \text{mm}^2 )</td>
</tr>
</tbody>
</table>

#### Mechanism Masses
1. Piston, spring, slide bar, presser foot etc.  
   \[ m_1 = 130 \, \text{gm} \]
2. Pipe-valve/actuator                         
   \[ m_2 = 280 \, \text{gm} \]
3. Inlet pipe to rotary valve                  
   \[ m_3 = 88 \, \text{gm} \]
4. Exhaust pipe from rotary valve              
   \[ m_4 = 88 \, \text{gm} \]
5. Total overall mechanism mass In/Out         
   \[ M = 0.498 \, \text{kg} \]

#### Viscous Coefficients
1. Actuator                                    
   \[ k_1 = 0.8 \mu \times 10^5 \]
2. Support bush                                 
   \[ k_2 = 0.34 \mu \times 10^5 \]
3. Pipe-valve/actuator                         
   \[ k_3 = 3.32 \mu \times 10^6 \]
4. Inlet pipe to rotary valve                   
   \[ k_4 = 0.262 \mu \times 10^6 \]
5. Exhaust pipe from rotary valve              
   \[ k_5 = 0.262 \mu \times 10^6 \]
6. Total viscous coefficient                   
   \[ k = 3.966 \mu \times 10^6 \]

#### Momentum Coefficients
1. Rotary valve                                
   \[ c_1 = 525 \]
2. Pipe valve to actuator                       
   \[ c_2 = 17242 \]
3. Inlet pipe to rotary valve                   
   \[ c_3 = 898 \]
4. Exhaust pipe from rotary valve               
   \[ c_4 = 895 \]
5. Total momentum coefficient In/Out            
   \[ c = 18665 \]
<table>
<thead>
<tr>
<th>Cushion length (mm)</th>
<th>Impact velocity $v_i$ m/sec</th>
<th>Cushioning coefficient $c \times 10^6$</th>
<th>Radial clearance $h$ microns</th>
<th>Induced pressure $p_i$ (bar)</th>
<th>Final impact velocity $v_i$ m/sec</th>
<th>Cushioning time millisecs</th>
<th>Total mechanism response time millisecs</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>4.625</td>
<td>28.46</td>
<td>30</td>
<td>1250</td>
<td>0.242</td>
<td>0.62</td>
<td>7.37</td>
</tr>
<tr>
<td>2</td>
<td>4.55</td>
<td>15.2</td>
<td>47</td>
<td>622</td>
<td>0.453</td>
<td>1.14</td>
<td>7.64</td>
</tr>
<tr>
<td>3</td>
<td>4.47</td>
<td>10.69</td>
<td>58</td>
<td>409</td>
<td>0.645</td>
<td>1.61</td>
<td>7.86</td>
</tr>
<tr>
<td>4</td>
<td>4.38</td>
<td>8.30</td>
<td>63</td>
<td>294</td>
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