"POSITION CONTROLLED DISC VALVE"

by

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Summary

Recent developments of electro-hydraulic disc valves at Surrey University have shown that with a careful balance between the hydraulic and magnetic forces, this type of valve can be used as a digital or proportional device. As the valve is simpler in construction and involves very few critical dimension compared with a servo-valve, the sensitivity to contamination is considerably reduced. The dynamic response of the valve is fast due to utilising high electro-magnetic and fluid forces for actuation.

The research described in this thesis is an extension of earlier work by Yuksel and Usman to improve electro-hydraulic disc valves by applying closed-loop position or pressure control to the disc. From an investigation of an unbalanced single disc valve, it was found that using position feedback can help to stabilise the disc under varying load conditions. A special differential capacitive transducer to measure the disc position was designed and constructed and was found to perform satisfactorily. As the pressure-flow characteristic of the valve can be varied by controlling the disc position, the function of the valve is similar to an electrically controlled variable orifice. Various modular configurations are proposed to perform more complicated control functions.

In the final part of the research, a double disc valve is described for used in an application study to control the damping characteristic of a modified vehicle shock absorber. Initially, the valve was designed for closed-loop position control due to the non-linear hydraulic and magnetic forces. Results show that the valve can be controlled to generate the required range of damping force and has adequate dynamic performance with a response time in the range of 10 to 30 msec. However, tests using direct pressure control were also carried out. Preliminary results indicate that
pressure feedback is preferable to position feedback and that by using lead compensation together with a proportional plus integral controller, stable operation is possible.
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<td>Distance between disc and chamber face</td>
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<td>$h_{1,2}$</td>
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<tr>
<td>$i_c$</td>
<td>Magnetic coil current</td>
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</table>
\[ \dot{i}_c \] Rate of change of coil current

\[ \Delta i_c \] Change of coil current

\[ i_t \] Insulator thickness

\[ I_b \] Base current

\[ I_c \] Collector(coil) current

\[ I_e \] Emitter current

\[ K_a \] Current to voltage gain at coil amplifier

\[ K_f \] Position transducer gain

\[ K_i \] Integral gain

\[ K_p \] Proportional gain

\[ K_s \] Spring constant

\[ l_i \] Length of magnetic flux path

\[ L_c \] Magnetic coil inductance

\[ \dot{L}_c \] Rate of change of coil inductance

\[ L_{n1,2} \] Lengths of supply and discharge nozzles

\[ m \] Mass of disc

\[ N \] Number of turns in magnetic coil

\[ P_c \] Chamber pressure

\[ P_d \] Discharge pressure

\[ P_{dn} \] Discharge nozzle pressure

\[ P_r \] Radial pressure

\[ P_s \] Supply pressure

\[ P_{sn} \] Supply nozzle pressure

\[ P_{sr,dr} \] Radial pressures at radius \( r \) on supply and discharge sides

\[ Q \] Oil flow rate

\[ \dot{Q} \] Rate of change of oil flow rate

\[ r, \theta, z \] Vector components in cylindrical co-ordinates

\[ r_{t1,2} \] Transducer rings thicknesses

\[ r_g \] Radial gap between disc and insulated transducer rings

\[ R_{1,2} \] Bridge arm resistances
\( R_c \)  
Coil resistance  
\( R_e \)  
Emitter resistance  
\( RE \)  
Reynolds number  
\( R_{f1,f2} \)  
Resistances of transducer filter network  
\( R_i \)  
Radius of supply and discharge nozzles  
\( R_{iv} \)  
Ratio of inertia to viscous pressure  
\( R_o \)  
Disc radius  
\( s \)  
Laplace operator  
\( sm \)  
Specified disc movement in dynamic test  
\( SHG \)  
Supply side holding gap  
\( T_o \)  
Time constant at magnetic coil  
\( TD \)  
Total disc travel  
\( u, v, w \)  
Velocity components in cylindrical co-ordinates  
\( \Delta V_a \)  
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\( V_b \)  
Base voltage  
\( V_{BE} \)  
Base-emitter voltage  
\( V_{cc} \)  
Collector voltage  
\( V_{CE} \)  
Collector-emitter voltage  
\( V_f \)  
Feedback voltage from position transducer  
\( V_{ib} \)  
Bridge supply voltage  
\( V_{ob,ob'} \)  
Bridge output voltages  
\( w \)  
Bridge supply frequency  
\( x_{1,2} \)  
Disc positions as referred to the supply and discharge nozzles
\(x_{1,2}\)  Disc velocities as referred to the supply and discharge nozzles
\(\dot{x}_1\)  Disc acceleration
\(\Delta x_1\)  Disc displacement
\(x_p\)  Initial steady-state disc position
\(x_g\)  Distance between disc and magnetic coil
\(\tau\)  Shear stress
\(\mu\)  Viscosity
\(\varepsilon\)  Permittivity
\(\rho\)  Oil density
\(\beta\)  Transistor gain
\(\mu_0 \cdot \mu_g\)  Magnetic permeability
Nomenclature for Double Disc Valve

Aa  Cross-sectional area of annulus (nozzle 1)
Aco Cross-sectional area of compression chamber
An1,2 Cross-sectional areas of nozzle 1, 2
Are Cross-sectional area of rebound chamber
B Damping coefficient
Ca,b Capacitances of lead compensator
Cd Discharge coefficient
Cdi Inward flow discharge coefficient
Cdo1,2 Outward flow discharge coefficients at nozzle 1, 2
dn Nozzle diameter
dn1,2 Diameters of nozzle 1, 2
dr Diameter of push rod
Dpis Diameter of shock absorber piston head
Drod Diameter of shock absorber piston rod
e Error signal
F1fr Fluid flow force on right-hand surface of disc 1
F2fl,r Fluid flow forces on left and right-hand surfaces of disc 2
F1mr Momentum force on right-hand surface of disc 1
F2ml,r Momentum forces on left and right-hand surfaces of disc 2
Fn1,2 Total forces acting on disc 1, 2
Fnet Net hydraulic force
F1sl,r Hydrostatic forces on left and right-hand surfaces of disc 1
F2sl,r Hydrostatic forces on left and right-hand surfaces of disc 2
Fspring Spring force
h1,12,33,44 Distances between disc and chamber faces on disc 1, 2 chambers
HG Holding gap
Ka Current to voltage gain at coil amplifier
$K_f$ Position transducer gain

$K_i$ Integral gain

$K_p$ Proportional gain

$K_{pq}$ Rate of change of pressure drop with respect to flow rate under constant disc position

$K_{px}$ Rate of change of pressure drop with respect to disc displacement under constant flow rate

$K_s$ Spring constant

$L_c$ Coil inductance

$L_n_{1,2}$ Lengths of nozzle 1,2

$LL(s)$ Lead-lag compensator transfer function

$m$ Mass of disc and push rod

$M_r$ Multiplier

$P_{1,2}$ Pressures at nozzle and edge of disc

$P_{a,b}$ Left and right-hand side annulus pressures at nozzle 1

$P_d$ Discharge pressure

$P_{n2}$ Pressure within nozzle 2

$P_s$ Supply pressure

$Q_{1,2}$ Flows across nozzle 1,2

$Q_{in}$ Total input flow rate

$Q_{mech}$ Oil flow through mechanical valve

$R_{a,b}$ Resistances of lead compensator

$R_c$ Coil resistance

$RE_{1,2}$ Reynolds numbers at nozzle 1,2

$R_{n1,2}$ Radii of nozzle 1,2

$R_0$ Disc radius

$s$ Laplace operator

$T_c$ Time constant at magnetic coil

$TD$ Total disc travel

$\Delta v_a$ Voltage input to coil amplifier

$x_d$ Disc position
$x$  \hspace{1cm} \text{Disc position from nozzle 2}

$x_\varnothing$  \hspace{1cm} \text{Initial steady-state disc position}

$x_{1,2}$  \hspace{1cm} \text{Positions of disc 1,2 from nozzle 1}

$x_{\text{gap} \varnothing}$  \hspace{1cm} \text{Gap between disc 1 and magnetic coil}

$x_{\text{gap}1,2}$  \hspace{1cm} \text{Minimum gaps between nozzle 1 and disc 1,2}

$\mu$  \hspace{1cm} \text{Viscosity}

$\rho$  \hspace{1cm} \text{Oil density}
Chapter 1

Introduction
1. Introduction

The function of any hydraulic control device is to control one or more of the following parameters: direction, velocity, acceleration, deceleration or force of an actuator with either linear or rotary output. This control function can be achieved in a number of different ways, depending on the system specification, the type of control device used and most importantly, the cost.

Modern industrial hydraulic control valves can be broadly divided into two groups: the digital valve group and the analog valve group as shown in Figure 1.1. In the digital group, the valve functions as a logic switch and by using one or more control signals the output is changed from one state to another. For example, a wall-attachment device or a turbulence amplifier shown in Figure 1.2 is a no moving part device in which the change of state is achieved entirely by a fluid interaction of the supply and control jet fluid signals. However, as some of these devices require a constant use of fluid power and tend to have large quiescent consumption at high operating pressure, they are used more in the transmission and manipulation of information than in the transmission of power[1,2].

In an attempt to reduce the power consumption, there are fluidic devices developed with moving parts such as the ones shown in Figure 1.3. The moving parts commonly used are in the form of a spool, a diaphragm, a ball, a disc or a free-foil. Although the valve construction is made slightly more complicated, power is consumed only in the transition period or when the output is changed from one state to another. As the response of this type of valve can be very fast, it has attracted a lot of research interest which consequently widened the application of digital valves. For example, the successful development of a pneumatic servo by Goldstein using the
pulse-width-modulation technique with floating disks\cite{3}. The floating disks were used both for modulation of the pneumatic input signal to the system and for switching of fluid power to and from an actuator. Also, in a recent study by Mansfeld, single or two-stage ball valves operated indirectly from a bistable torquer were used as digital control elements for an electro-hydraulic servo actuator\cite{4}.

However, by introducing a small moving part, the valve has been made less reliable than a pure fluid device described earlier due to mechanical wear and possible fatigue failure. On the other hand, the control of the moving part is found to be quite difficult and in many cases could only be achieved hydraulically. As a result, this type of devices is sometimes not flexible enough to be integrated with other system components.

The third type of digital valve is quite similar in principle to the second one just described except that it is usually larger and is actuated differently. This is known commonly as a directional valve and is used extensively in slow-acting industrial systems. Figure 1.4 shows a manually-operated directional valve of this type used to change a cylinder rod from one state to another. Another common method of controlling directional valve is by a solenoid as shown in Figure 1.5. In this case, although the control of the valve is more flexible and interaction with other system components can be done more easily, the valve construction has been made more complicated and costly. Also, as the force generated from a solenoid is limited, the response of the valve is relatively slow.

In the analog group, the valves can be sub-divided into two main types: the servo-valve and the proportional valve. The input to these valves is usually in the form of a low power electrical signal. The servo-valve is developed primarily for proportional operation and is a well-developed,
sophisticated control device used particularly in such complicated operations as flow or pressure control. Some characteristics of a servo-valve are: high pressure amplification, minimum hysteresis and high dynamic capabilities. The servo-valves are sometimes used in multi-stages to increase the pressure or flow amplification. Figure 1.6 shows an example of a two-stage servo-valve. The hydraulic output from the first stage is used to operate the second stage, which has a higher pressure-flow capacity and the output from which passes to the controlled equipment.

However, despite all these desirable characteristics, the servo-valve has the disadvantages of being expensive and being contamination sensitive. For example, in the process of ensuring a high pressure amplification, the radial clearance between the valve spool and the sleeve is made very accurate and small, usually down to ten microns. With such dimensional requirements, the mechanical part can be damaged quite easily and wear could be significant. Therefore, special material treatment is usually involved in extending the operating life of the spool assembly[5]. Also, in order to stop the spool from being jammed by the oil contaminants, very fine filtering and good system maintenance are needed. Thus manufacturing and running costs of electro-hydraulic servo systems are high, a factor which must be weighed against their operating performance.

The proportional valve is one of the more recent developments in the analog valve group. It is called a proportional device because the output condition is varied in relationship with the input demand signal. It is different from a servo-valve by the use of a cheaper external feedback and lower cost electrical actuation. The proportional valve as shown in Figure 1.7 can be described as a spin-off from the accumulated experience of solenoid-operated directional valve design. It has been developed with the
objective of simplifying the servo-valve construction while retaining some of its dynamic performance.

The proportional valves are commonly operated by force-controlled feedback or stroke-controlled feedback[6]. In force-controlled feedback, the movement of the mechanical part is resisted by a constant force generated at a proportional solenoid. The force generated from this specially-made solenoid can remain constant over a small range of armature movement. The current in the solenoid is fixed via a current feedback loop which keeps the actuating force constant. In the other type of feedback, the position of the mechanical moving part is chosen as the controlled variable. The actuating force is adjusted continuously to keep the mechanical moving part at the desired position regardless of any change in force condition.

The proportional valve has the advantage of being simpler in construction, therefore leading to a lower cost, but is not as dynamically responsive as a conventional servo-valve[7]. However, the proportional valves can certainly be used in a variety of industrial applications. For example, in speed control system such as for winches and in earth-moving equipment for remote operation of diggers.

Apart from the digital and analog valves described above, there is a continuing interest in developing valves with simpler construction and better dynamic performance. Most of the approaches have been to use a moving part similar to the one used in the digital valves but incorporating an electric actuator to improve control flexibility. For example, the use of an electro-magnetic coil acting on a permanent magnet to control the flow from a nozzle in a four-way on-off pneumatic valve by Taft[8] or the control of a three-way valve using a solenoid acting directly on a circular disc by Pointout[9] or some other recent similar developments such as by
Nakajima et al who developed a three-way valve based on the use of electro-magnetic force acting against a coil spring on a movable block{10}, by Yuksel who formed a four-way valve by using two nozzles acting in opposition on a circular disc and two electro-magnetic coils{11}, and by Taft et al who controlled the position of a ball between two nozzles by supplying pulse-width-modulated current to two coils{12}.

As the moving part is acted upon directly by the electric actuator, this type of valve should prove to be more attractive in terms of a better dynamic performance than conventional switching valves. On the other hand, it can also be shown that by carefully balancing the electro-hydraulic force on the moving part, a proportional device can also be configured{13}. There is no doubt that these valves are simpler in construction and therefore should be economically and technically attractive compared to a proportional solenoid-operated valve.

The major activity of this research is to build on earlier work at Surrey University on the disc valve design of Yuksel and Usman and consider whether improved performance can be obtained by using disc position feedback within a valve.

This research thesis is basically divided into two parts. In the first part, the construction of a position-controlled single disc valve is described. This is followed by some studies of the hydraulic and magnetic theories to be applied in an electro-hydraulic disc valve. Static and dynamic simulation programs are then developed for predicting the valve performance. A study of the position control stability is also made which results in a better understanding on the disc control mechanism and helps to define the type of control function required.
In the second part, the research is directed towards developing a disc valve to be used for a specified application. A position-controlled double disc valve, having a similar operating principle to the single disc valve, is developed. This double disc valve is designed to be used in a vehicle shock absorber to electrically modulate the shock absorber damping characteristic. In calculating the nozzle size, the experimental results obtained from the single disc valve are used. The theoretical study and control analysis are made in basically the same way as in the single disc valve. However, because instability is shown to occur in open-loop, the use of closed-loop control and a compensation network is found to be necessary. Testing of the double disc valve as a variable orifice in a modified shock absorber is also performed and a comparison of the test results and the desired characteristics is also made.
Figure 1.1 General Classification of Hydraulic Control Valves
Figure 1.2 No-Moving Part Fluidic Device

a). Wall-attachment Device

b). Turbulence Amplifier
a). Spool-type Moving Part

b). Ball-type Moving Part

c). Free Foil-type Moving Part

Figure 1.3 Moving Part Fluidic Device
Figure 1.4 Manually-operated Directional Valve

Figure 1.5 Solenoid-operated Directional Valve
Figure 1.6  Servo-valve (with mechanical feedback)

Figure 1.7  Proportional Valve
In the following chapters, the development of a single disc valve is described. The single disc valve is designed to be a two-port flow control device. The flow through the valve is controlled by the position of a circular disc which is modulated by magnetic force generated from an electro-magnetic coil. The disc position is monitored by a position transducer, the output signal of which is also used as feedback signal for closed-loop position control function. A study of the hydraulic and magnetic behaviour is made and the results are used to predict the static and dynamic valve characteristics. The valve is tested under steady-state and dynamic conditions and the position control loop is also examined by applying a pressure step change to simulate load disturbances.
Chapter 2

The Single Disc Valve
2. The Single Disc Valve

2.1 Operating Principle

The operating principle of a single disc valve is illustrated in Figure 2.1. The valve is designed to be used as a flow control valve and to achieve this, the position of a circular disc is varied from a fixed nozzle to provide the desired flow modulation. As it is shown in the figure, the hydraulic oil enters the valve chamber via a supply tube and leaves through a discharge tube. A circular disc is placed between the two tubes and as a result, two annular orifices are formed. Because of the two orifices present, the valve can be chosen to operate from the supply orifice to the disc travel mid-point or from the mid-point to the discharge orifice. This is because when the disc moves from left to right in the figure, the flow resistance decreases to a minimum at about the centre of the disc travel and then increases again until the discharge nozzle is closed.

The initial disc position is determined by the balance of a hydraulic force acting from left to right and a resistive spring force acting in the opposite direction. To change the disc position, current is supplied to a magnetic coil. The magnetic force generated acts in the same direction as the hydraulic force and as a result, the combined magneto-hydraulic force pulls the disc closer to the discharge nozzle. As the disc position is altered, the orifice areas are also altered, thus leading to a change of flow across the valve.

The single disc valve described above can be used as a two-port flow control valve or it may be used in a modular configuration to perform a similar function to a conventional spool valve. The function of a single disc valve is analogous to a spool land in a spool valve in which the oil
flow is controlled by the opening or closing of the orifice. For example, in Figure 2.2a, a three-way valve is formed by connecting two single disc valves together or alternatively a four-way valve is made by using two three-way valves together as shown in Figure 2.2b. Although such a valve has not been constructed, it has considerable potential for flexible changing of valve functions.

2.2 Design and Construction

The construction of the single disc valve is shown in Figure 2.3. Figure 2.4 shows the valve's components in the unassembled condition. The basic components consist of a circular disc(1) and two nozzles axially aligned. When hydraulic oil emerges from the supply nozzle(2), it impinges on the front surface of the disc and flows radially outward to a collection chamber. The oil flow is restricted by the gap formed between the disc and the nozzle nose. This gap is varied by the disc position and is limited to less than one-fourth of the nozzle diameter to ensure that the pressure drop across the valve is not dominated by the tube diametrical resistance.

While oil is flowing across the front disc surface, a hydraulic force which tends to push the disc away from the supply nozzle is generated. The main function of the wavy washer(3) is to provide a resistive spring force to stop the disc from moving forward. The steady-state disc position is established by the balance of this spring force and the hydraulic force.

On leaving the front disc surface, the oil is routed to the other side of the disc, where the oil flow direction is changed to radially inward. The
discharge nozzle(4), which is shown on the right hand side of the valve chamber, sets up another restriction with the disc before allowing the oil to flow out. The presence of this second restriction increases the pressure drop which in turn causes the chamber pressure to be also increased and thereby creating a useful resisting force against the forward force. As the magnitude of the net hydraulic force acting from left to right has much been reduced, the selection of the wavy washer and the selection for an appropriate drive for changing the disc position are made considerably easier.

To move the disc, an electro-magnetic coil(5) is used. This coil is placed at a location where the magnetic force generated pulls the disc closer to the discharge nozzle. Among the various types of electro-magnets developed in recent years, the flat-faced armature type is found to be the one most suitable to use in these valves. It has a high force-short stroke characteristic(14), which is advantageous for an adjacent flat disc actuation. However, for this type of electro-magnet, the attractive force decreases rapidly as the armature is placed further away from the magnetic core. Therefore, to avoid any excessive current in the coil wires, the magnetic coil must never be placed too far away from the disc.

The electro-magnet presently constructed is based on a uniform cross-sectional criteria(13). The high permeability Remko iron is selected as the material for the magnetic core and the disc. Because of the weak mechanical strength of the Remko iron, a hard metal insert is threaded into the centre of the disc to prevent any change of disc thickness as a result of impact with the hard metal nozzles. The disc travel can be changed by a change of disc thickness. Both the disc and the core are heat treated before use so that their best magnetic characteristic lost in machining can be recovered.
As shown in Figure 2.3, the supply and discharge nozzles are not flush to the chamber wall. There are two reasons for this. The first is to create a thin land condition so that when the disc is moved to the nozzle, a good positive contact can be achieved. Ideally a sharp nozzle would be the most desirable because the valve can be assured of being closed properly. However, a sharp nozzle is very difficult to realise in practice, especially in the single disc valve where impact by the disc frequently occurs. The stops(6) that are shown in figure are intended to reduce this impact force.

The second reason of maintaining a gap between the disc and the chamber wall is to allow hydraulic oil to get into both sides of the disc so that a hydrostatic balance can be established. The high hydrostatic force produced on one side of the disc must be cancelled by an equal and opposite force or the disc can never be pulled back from the supply nozzle. In the next chapter, it will be shown that this gap also has an influential effect on the pressure-flow characteristic and the hydraulic force produced. Also because of the presence of this gap on the discharge side, the disc is located further away from the core face so as to reduce the magnetic force generated.

In order to be able to monitor the disc position and to provide a signal for subsequent disc control, a capacitive type position transducer(7) has also been developed. The position of the disc relative to the two fixed nozzles is indicated by the electrical signal generated from the transducer. The transducer is shown to be constructed from two circular rings which are insulated from the disc by a dielectric material(PVDF). The position signal is derived from the capacitive effect generated across the disc and the circular rings. Since the disc is required to move freely
across the chamber, a radial gap is shown to exist between the disc and the insulated rings. This gap must be dimensioned so that no jamming occurs between the circumferential chamber wall and the disc in the valve chamber[13].

For the position transducer to function, either the disc or the transducer rings must be connected to a higher potential so that when electric energy is applied, the desired capacitive effect is generated. In the present application, the transducer rings have been selected to be the higher potential electrode, and the disc is selected as ground. The selection is for simplicity reason so as to avoid the trouble of insulating the disc from other valve components. Because of positive spring compression, the disc is naturally grounded by the wavy washer.

All the components described above are housed in two aluminium end bodies, namely the transducer end body(8) and the coil end body(9). Figure 2.5 shows the construction of these two end bodies in assembly with other components. As their names imply, the position transducer is attached to the transducer end body and the wires carrying the signal of the disc position are terminated at a connector which is linked subsequently to an electronic demodulation circuit. Similarly, the electric supply to the magnetic coil is made to the valve through another connector located at the coil end body.

Oil enters the valve through the base of the centre body(10) using a CETOP port configuration. Cross drillings in the body connect to annular grooves in the two end bodies thereby connecting supply and return to the two nozzles. The use of these annular grooves simplifies the valve assembly as no specified orientation for installing the two bodies is required.
The following is a list of some of the valve's components dimensions referred to in Figure 2.3:

(1) Disc: diameter = 31.65 mm,
    thickness = 4.0 mm.

(2) Supply nozzle: inner diameter = 2.54 mm,
    outer diameter = 3.05 mm.

(3) Wavy washer: outer diameter = 15.8 mm,
    inner diameter = 10.5 mm,
    thickness = 0.3 mm,
    free length = 1.78 mm.

(4) Discharge nozzle: inner diameter = 2.54 mm,
    outer diameter = 3.05 mm.

(5) Electro-magnetic coil: refer to Figure 3.7 for dimensions.

(7) Transducer rings: refer to Figure 2.6 for dimensions.

(11) Chamber ring: thickness = 4.4 mm.

2.3 Position Transducer

The function of a position transducer is to monitor the disc position so that a study of the single disc valve characteristic can be made and also to provide a feedback signal which can be used for control purpose. In the beginning stage of the development, a study was made to find out which type of position transducer would be the most appropriate one to use. The following is a list of factors considered in the study.

a) The use of any transducer must not affect the free floating disc nature.
b) Any force exerted on the disc by the transducer must be minimal.

c) The transducer output must be relatively insensitive to any change of fluid property, particularly temperature.

d) The valve construction must remain simple and the transducer must be cost effective.

The principal methods in use for measuring displacements may be classified as mechanical, optical, acoustical and electrical. The use of a mechanical method is discarded because the free floating disc feature is violated. Optical sensors which require an optical path to operate are also not too appropriate, since the transmission of the light into the hydraulic oil would be made rather complicated and difficult. Sealing would also be a problem. Ultrasonics have been used on many occasions for position measurement, but to provide a good resolution in a high viscosity fluid medium, the design of a suitable wave generator would be very complicated and costly[15]. There are many methods for displacement measurement, but when the response rate, the ease of signal processing and the cost are considered, electrical methods have a clear advantage in this type of application.

Among the various types of transducers developed under the electrical category, the inductive and capacitive types are the two most commonly used. They require no contact to be made to the object measured. Therefore, they are both suitable to use as far as factor (a) above is concerned. Study of the excitation forces by Hugill[16] shows that the force from the transducer is so small that it can be neglected as compared with the other forces on the disc. When there is no other magnetic field present, an inductive type transducer tends to be more stable in reaction to temperature changes, unlike the capacitive type in which the signal
accuracy can be badly affected. In terms of transducer construction, the capacitive type is more compact and the manufacture of it is far simpler than its inductive counterpart. With the above considerations in mind, a capacitive transducer using a push-pull method for temperature compensation was developed.

The construction of the transducer in use is shown in Figure 2.6. The capacitive effect is generated across the radial gap between the rings and the disc after connections to the electric source are made. The generated capacitance in ring 1 in simplified form may be given as

$$C_1 = 2\pi \varepsilon (r_1 - x_1)/\ln(D_{tr}/D_d)$$

Equation (2.1) assumes that the effect from the radial gap, $rg$, is insignificantly small compared with the one from the dielectric material. It is shown that the capacitance increases as the disc displacement decreases. The objective of putting two rings together is to allow the capacitance to increase on one ring and decrease on the other while the disc is traversing across the chamber. Such an arrangement is normally referred to as a push-pull arrangement. If these two changing capacitances are subtracted from each other, the temperature change effects can be minimised and an increase of transducer sensitivity can also be made.

There are various methods to measure the changing capacitance signals[17-19]. The one that is found to be the most convenient to use is illustrated in Figure 2.7. The following two examples illustrate how these circuits are applied with push-pull sensors for position measurement:

Neubert[20] in his analysis on the same measuring circuit given in Figure 2.7a showed a constant sensitivity being independent of the exciting frequency could be achieved if a perfectly coupled inductance ratio arms was used for the two impedances $Z_1$ and $Z_2$. If the input and output
connections shown in Figure 2.7a are rearranged to 2.7b, which was used by Morgan and Brown[21] in their development for a displacement transducer, then the resulting bridge could be made insensitive to temperature.

The measuring circuit being used for the single disc valve is not quite the same as the two mentioned. As discussed earlier, the disc is grounded for convenience. Therefore, a bridge similar to Figure 2.7b cannot be applied. When the circuit in Figure 2.7a is used, the impedances $Z_1$ and $Z_2$ are substituted instead by two resistors. The reason for this is to enable the transducer operating point to be varied, if needed, by varying the magnitude of the resistors. The complete transducer circuit is shown in Figure 2.8. The high frequency signal from each capacitor arm is connected to an input buffer. Subtraction between the two signals is immediately followed by a differential amplifier. An output signal which combines the changing capacitive effect from both measuring arms is obtained and passed to a peak detector for subsequent demodulation. If the cable capacitance and stray capacitance are ignored, the equation for the output signal will be given as

$$V_{ob} = \frac{1}{jw.C_1.R_1} - \frac{1}{jw.C_2.R_2}.V_{ib}$$

$$= \frac{jw.V_{ib}.(C_2.R_2 - C_1.R_1)}{(1 - w^2.C_1.C_2.R_1.R_2 + jw.(C_1.R_1 + C_2.R_2))} \quad (2.2)$$

Equation (2.2) shows an expression that looks rather different from the ones given by Neubert or Morgan. As in the review made by Hugill[16], the advantage of using push-pull sensors is not only that the sensitivity will be increased, the transducer output, if arranged properly, can also be made temperature insensitive. The output signal given by equation (2.2) shows the dependence on temperature still exists unless the term \( (1 - \)
\(w^2C_1C_2R_1R_2\) at the denominator disappears. If the term does disappear and if the two resistors are equal, equation (2.2) may be reduced to

\[V_{ob} = \frac{dt - 2rt - it + 2x}{dt - it}V_{ib} \tag{2.3}\]

since \(C_1 = 2\pi\varepsilon_0\) and

\[C_2 = 2\pi\varepsilon_0\frac{x}{ln(D_t/D_d)}\]

The dielectric constant, a temperature dependable parameter, is now eliminated from equation (2.3).

Reducing \((1 - w^2C_1C_2R_1R_2)\) to zero can be made in two possible ways. They are by varying the arm resistance or by changing the supply source frequency. Since a waveform generator is being used in the transducer circuit, adjusting the supply frequency will therefore be more convenient. While the output signal is monitored, the frequency is adjusted until a maximum output occurs. This maximum output can be treated as an indication of a minimum \((1 - w^2C_1C_2R_1R_2)\). Figure 2.9 shows the result of the frequency calibration. The optimum frequency was found to occur at around 13.5 KHz. It is noticed that the derived condition for temperature insensitivity is applicable to only one particular disc position. However, since the capacitance change in the moving disc is relatively small, the change of magnitude in \((1 - w^2C_1C_2R_1R_2)\) will be rather insignificant and the 'insensitive' condition may perhaps be extendible to other disc positions.

Before the position transducer was applied in the single disc valve, a test model shown in Figure 2.10 was developed. The model was designed with approximately the same dimension as that of the valve. The moving disc which was attached to a micrometer was turned to move up and down across the transducer rings. The calibration of the test model given in Figure
2.11 is shown to be quite linear except when the disc had moved above the upper ring.

In changing it to DC form, the signal coming out from the differential amplifier is passed to a demodulator. This demodulator is shown in Figure 2.8 to be composed by a peak detector[22] and an active low-pass filter. The correct use of the RC network in the peak detector depends on the magnitudes of the carrier frequency and the modulation frequency. The time constant should be large as far as the carrier signal is concerned and should be small by comparison with the modulation frequency[23]. The carrier frequency component of the signal are filtered off at the output of the demodulator.

The use of a peak detector is not as popular as a phase-sensitive detector[19]. The disadvantage of the former is that the output signal has no sign change when the detected object crosses the centre of the sensors. In other words, the output signal will have the same magnitude if the disc is located equally away from the centre line on either side of the rings. Points A and B in Figure 2.11 is an example to show this unchanged magnitude. To avoid any misinterpretation of disc position, the two transducer rings shown in Figure 2.6 are made with different thicknesses. The purpose of this is to ensure that the capacitance on Ring 2 is never increased to more than the minimum capacitance on Ring 1 and when the disc moves from left to right, the output signal varies in one direction only. This is similar to limiting the transducer operation to line A or line B in Figure 2.11. The use of different resistances for \( R_1 \) and \( R_2 \) can also produce the same effect except that the output linearity will be slightly affected.
Figure 2.12 shows the set-up for calibrating the position transducer mounted in the single disc valve. The result of the calibration with resistors $R_1$ and $R_2$ equal to 200 kΩ is shown in Figure 2.13. It was found that the electrical signal changed in one direction only as the disc was moving away from the supply nozzle. However, the output signal was shown to be non-linear. This is because the coil end body was removed in order to attach the disc to a micrometer, the stray capacitances on both sensors were out of balance and therefore the output voltage was not linear.

By substituting $j\omega$ in equation (2.2) with the Laplace operator $s$, the signal appearing at the bridge output is

$$V_{ob}(s) = \frac{V_{ib}(s)s(C_2 - C_1)}{(1 + (C_1 + C_2)s + C_1C_2R_1R_2s^2)} \quad (2.4)$$

where $V_{ib}(s)$ is the input voltage applied across the bridge,

- $C_1 = 2\pi \varepsilon_0 (\rho_1 - x_1(s))/\ln(D_{tr}/D_d)$ and
- $C_2 = 2\pi \varepsilon_0 (x_1(s) + dt - rt_1 - it)/\ln(D_{tr}/D_d)$.

If the transducer is initially set to the optimum operating condition, i.e. by setting $(1 + s^2C_1C_2R_1R_2)$ to zero, then equation (2.4) is reduced to

$$V_{ob}(s) = \frac{V_{ib}(s)(C_2 - C_1)}{(C_2 + C_1)}$$

$$= \frac{V_{ib}(s)(dt - 2rt_1 - it + 2x_1(s))}{(dt - it)} \quad (2.5)$$

If the movement made by the disc is $\Delta x(s)$, then the new output voltage is

$$V_{ob}'(s) = \frac{V_{ib}(s)(dt - 2rt_1 - it + 2x_1(s) + 2\Delta x(s))}{(dt - it)}$$

since $C_1' = 2\pi \varepsilon_0 (\rho_1 - x_1(s) - \Delta x(s))/\ln(D_{tr}/D_d)$ and
\[ C'_2 = 2\pi \xi_1(x(s) + \Delta x(s) + dt - rt_1 - it)/\ln(D_{tr}/D_d). \]

Therefore, the change in output voltage is

\[ \Delta V_{ob}(s) = V_{ob}^\prime(s) - V_{ob}(s) = \frac{2.V_{ib}(s) \Delta x(s)}{dt - it} \]  

(2.6)

As this signal will be demodulated at the peak detector, it is assumed that there is no other signal except the DC component of the signal is present at the output of the demodulator. If the transducer gain is \( K_f \), the position signal after the demodulator will be given as

\[ \Delta V_f(s) = \frac{K_f \Delta x(s)}{(1 + (C_{f1}R_{f1} + C_{f2}R_{f2})s + C_{f1}C_{f2}R_{f1}R_{f2}s^2)} \]  

(2.7)

where \( C_{f1}, C_{f2}, R_{f1} \), and \( R_{f2} \) are the capacitances and resistances of the filter network.

Theoretically, the transducer gain could be evaluated from equation (2.6). However, because of stray capacitive effect, the calculated value could never be accurate in practice. Therefore, the determination of the transducer gain was made instead from the actual transducer. The denominator of equation (2.7) represents the transfer function of the active filter[23]. This transfer function is combined with the transducer gain to ensure that any response problem as a result of filtering will be identified in the position control loop. The active filter has been designed to operate at 150Hz cut-off frequency.
Figure 2.1  Simplified Construction of a Single Disc Valve

TD = Total Disc Travel

dt = Disc Thickness
Figure 2.2a). Simulated 3-Way Valve

Figure 2.2b). Simulated 4-Way Valve
Figure 2.3 General Assembly of a Single Disc Valve

1 - Circular Disc  
2 - Supply Nozzle  
3 - Wavy Washer  
4 - Discharge Nozzle  
5 - Electro-magnetic Coil  
6 - Stops  
7 - Transducer Rings  
8 - Transducer End Body  
9 - Coil End Body  
10 - Centre Body  
11 - Chamber Ring  
12 - 'O' Ring
Figure 2.4 Unassembled Single Disc Valve

Figure 2.5 Half-Assembled Single Disc Valve
Figure 2.6 Position Transducer Construction Diagram

Dielectric Material

Disc

View A

Dd = 31.65 mm
TD = 0.4
dt = 4.0
rt1 = 2.5
rt2 = 1.5
it = 0.5
g = 0.05
rt = 1.0

Discharge Nozzle

Supply Nozzle

Figure 2.6 Position Transducer Construction Diagram
Figure 2.7 Bridge Circuit for Measuring Capacitance Change in a Push-Pull Type Transducer
Figure 2.8 - Electronic Construction of Position Transducer
Figure 2.9 Optimum Frequency Calibration of Position Transducer
Figure 2.10 Construction of a Position Transducer Test Model

- $d_t = 4.0$ mm
- $r_{t1} = 2.7$
- $r_{t2} = 3.0$
- $i_t = 0.35$
- $D_d = 31.0$
- $rg = 0.25$
- $r_{it} = 0.5$
\( \Delta \) - Upward movement
\( \gamma \) - Downward movement

Voltage \( V_{ib} = 12 \text{ V p-p at 5 KHz} \)

\[ Z_1, Z_2 = 900 \, \Omega \]

Figure 2.11 Position Transducer Test Model

Output Characteristic
1. Right-hand movement
2. Left-hand movement

Voltage (V): 0.5, 0.4, 0.3, 0.2, 0.1, 0.0

Disc Displacement (mm): 0.0, 0.1, 0.2, 0.3, 0.4, 0.5

Figure 2.12 Position Transducer Calibration Set-up

Figure 2.13 Calibration of Position Transducer
Chapter 3

Theoretical Study of a Single Disc Valve
3. Theoretical Study of a Single Disc Valve

The operation of the single disc valve relies on a complex interaction among the hydraulic force, the electro-magnetic force and the spring force. The prediction of the valve performance is therefore not possible unless sufficient information is available to describe the characteristics of these forces. In this chapter, the flow mechanism within the single disc valve is studied and the equations for determining the hydraulic and magnetic forces are also derived.

3.1 Approximated Pressure-Flow Relationship

The construction of the single disc valve may be described in simple terms as two similar orifices connected together in series. Figure 3.1 shows the arrangement of disc and nozzles in the disc valve. When expressed in mathematical form, the oil flow rate across the valve is given as

\[ Q = C_{ds} \pi d_n x_1 \sqrt{\frac{2 \cdot (P_s - P_c)}{\rho}} \]  
\[ = C_{dd} \pi d_n x_2 \sqrt{\frac{2 \cdot (P_c - P_d)}{\rho}} \]  

It is assumed that the oil flow across the radial gap is zero and the flow on each side of the disc is represented by an orifice where the oil flow rate is related to the pressure drops by an assumed discharge coefficient. However, the flow on a disc surface should actually be described as three steps as shown in Figure 3.2:

a) a flow contraction at the nozzle,
b) a flow separation from the chamber face and
c) a fully attached, laminar flow.
The calculation of the true discharge coefficient in region (a) requires a careful and precise measurement of pressure at the nozzle outlet. This can be done quite conveniently in a scale-up model but will be very difficult as in the case of a single disc valve where the nozzle land is less than 0.3 mm and the flow condition could be easily disturbed. Equation (3.1) is thus used mainly for calculation purpose and for comparison with other experimental works in which a similar definition for the discharge coefficients was used{13,25,26}.

Rewriting equation (3.1) gives

\[
\frac{2\,(P_S - P_d)}{\rho} = Q^2\left(\frac{1}{(C_{ds}\pi.dn.x_1)^2} + \frac{1}{(C_{dd}\pi.dn.x_2)^2}\right)
\]

\[
= \left(\frac{Q}{\pi.dn.x_1}\right)^2\left(\frac{1}{C_{ds}^2} + \frac{x_1^2}{(C_{dd}x_2)^2}\right)
\]

(3.2)

It is shown in equation (3.2) that the pressure-flow characteristic of the single disc valve is a function of both the discharge coefficient and disc displacement. A lot of interest had been shown in studying the radial flow across two parallel plates with a centre nozzle supply[25-28]. All these studies revealed that the discharge coefficient varied considerably with the geometry of the orifice and the development of any empirical formulae would be quite an impossible task. As discussed by Lichtarowicz{26}, at least two types of flows appear in a nozzle-flapper configuration, it is therefore useful to first identify which type of flow is taking place before attempting any prediction of the flow behaviour at the single disc valve.

In the results reported by Lichtarowicz{26}, when the flapper or disc is
close to the nozzle, the width of the nozzle land has an equally important influence on the flow behaviour as the gap. The first type of flow is described as the one in which the land-to-gap ratio is very much less than 1. Under this condition, the behaviour of the discharge coefficient is similar to a short orifice. Because the discharge coefficient is relatively more stable at higher Reynolds Numbers, this type of flow is normally used in the design of flapper valves[26].

The second type of flow is similar to that occurring in the disc valve and is associated with a land-to-gap ratio of greater than 1. Meginns[27], in his observation of flow between two parallel discs, found that the flow was laminar when the Reynolds number was small. As the Reynolds Number increased, the flow became separated from the channel wall near the edge of the nozzle and re-attached as the fluid was continuously moving out. Further increasing the Reynolds Number resulted in greater separation and the flow was becoming more turbulent. The same results were observed by Duggins[25] who had also plotted the relationship between the discharge coefficient and the Reynolds Number. It is noted that the relationship curves varied considerably with the orifice geometry, i.e. the land-to-gap ratio.

If equation (3.2) is rewritten in terms of the Reynolds Number, which is defined as

\[ \text{RE} = \frac{\rho Q}{\mu a \mu_T \cdot d_n} \]

then the pressure drop across the valve may be written as

\[ 2 \cdot p_s (P_s - P_d) = \text{RE}^2 \cdot \mu^2 \cdot \left( \frac{1}{C_{d_s} x_1} \right)^2 + \frac{1}{\left( C_{d_d} x_2 \right)^2} \]

(3.4)
If the relationship between the discharge coefficient and the Reynolds number is available, equation (3.4) may be used to solve for the discharge coefficients in the single disc valve.

Differentiating equation (3.2) with respect to \( x_1 \) and equating it to zero for condition of maximum flow, the result becomes

\[
0 = RE^2 \mu^2 \left( -\frac{2}{C_{ds}^2 x_1^3} + \frac{2}{C_{dd}^2 x_2^3} \right) \\
\frac{1}{C_{ds}^2 x_1^3} = \frac{1}{C_{dd}^2 x_2^3}
\]

Equation (3.5) suggests that the maximum flow will occur at the centre of the travel if the discharge coefficients at the supply and discharge orifices are similar. However, because the inward and outward flows have each got a different pressure drop characteristic, it can be deduced that the position for maximum flow will never occur at the centre of the travel.

In estimating the fluid flow across the single disc valve, the relationship between the discharge coefficient and the Reynolds number found by Usman[13] is used. The reason for this is the single disc valve has a similar orifice geometry to the experimental model used by Usman. Therefore, the discharge coefficients derived should be quite accurate, although it is noted that a slight change of orifice dimensions could cause a substantial change in the discharge coefficient.

The fluid flow rate is calculated by a computer program which is written based on equation (3.4). The flow diagram of the program is given in Figure 3.3. The disc position is shown to be used as the second input variable after the pressure drop. Figure 3.4 is an example of the flow rate calculated. This calculated flow rate is found not only useful in the prediction of the output flow characteristic, it is also one of the essential parameters in the later study of the hydraulic force acting on
3.2 Steady-State Hydraulic Force

The hydraulic force considered is constituted by the hydrostatic force and the hydraulic flow force. The hydrostatic force exists whenever pressurized oil is supplied to the valve. Its magnitude is unaffected by the fluid dynamics and is varied only by the pressure drop and the area to which the fluid pressure is applied. The hydraulic flow force however, although having a more significant effect on the disc, does not exist when the fluid is stationary. The existence of this force is due to a pressure drop when oil is flowing over the disc surface and is caused mainly by viscous and inertia effects.

The derivation of a hydraulic force equation begins with finding the pressure distribution on the disc. The fundamental Navier-Stokes equation in cylindrical form, for the radial component only, is\(^\text{(29)}\)

\[
\rho \left( \frac{\partial u}{\partial t} + u \left( \frac{\partial u}{\partial r} \right) + v \frac{v}{r} + w \frac{\partial u}{\partial z} \right) = - \frac{\partial P_r}{\partial r} + \mu \left( \frac{1}{r} \frac{\partial \left( \frac{r u}{\partial r} \right)}{\partial r} \right) + \frac{\partial^2 u}{\partial z^2} \]

\( \text{If the tangential velocity is zero and the gravitational force is not considered, equation (3.6) may be simplified as} \)

\[
\rho \left( \frac{\partial u}{\partial t} + u \left( \frac{\partial u}{\partial r} \right) + w \frac{\partial u}{\partial z} \right) = - \frac{\partial P_r}{\partial r} + \mu \left( \frac{1}{r} \frac{\partial \left( \frac{r u}{\partial r} \right)}{\partial r} \right) + \frac{\partial^2 u}{\partial z^2} \]

The pressure distribution is indicated as a function of radius \( r \), channel
width z and time t. However, it can be shown that the change of pressure in the axial direction is insignificant and the pressure distribution will be a function of radius only under steady-state conditions\cite{30}. When the disc is stationary, the axial velocity is zero. The final version of equation (3.7) therefore becomes

$$\rho u_r \left( \frac{\partial u}{\partial r} \right) = -\frac{\partial p_r}{\partial r} + \mu \left( \frac{\partial}{\partial r} \left( \frac{1}{r} \frac{\partial (r u)}{\partial r} \right) \right) + \frac{\partial^2 u}{\partial z^2}$$  \hspace{1cm} (3.8)

Many studies had been made in finding a solution for the pressure distribution in radial flow\cite{31-36}. Some of these solutions were shown to be in good agreement with the experimental data. Appendix A1 describes the derivation of one of these solutions. This solution as is given below was found to be the closest to the experimental results among the others:

$$\frac{\partial p_r}{\partial r} = \pm \frac{6 \mu Q}{\pi r h^3} + \frac{27 \rho Q^2}{70 \pi^2 h^2 r^3}$$  \hspace{1cm} (3.9)

Equation (3.9) is not an exact solution to the differential equation given in (3.8). It is obtained as a result of a first-order integration. The order of integration may be increased if a better representation of the inertia term is required, but the improvement in accuracy is unlikely to be significant\cite{37}.

The first term on the right hand side of equation (3.9) is the pressure drop due to viscous friction. The plus and minus signs are used for the inward and outward flows respectively\cite{27,36}. The second term represents the inertia effects as it takes the velocity change into account when the fluid is flowing across the disc surface. This inertia term is usually not considered in the design of hydrostatic bearings\cite{38,39} because from experience, the viscous term alone is shown to be adequate in describing the pressure distribution and for calculating the resulting bearing load.
However, if the magnitudes of the viscous and inertia effects are compared as in equation (3.10) below, such an assumption may be true only when the ratio \( R_{iv} \) is very much less than 1.

\[
R_{iv} = \frac{\text{inertia}}{\text{viscous}} = \left( \frac{27 \rho Q^2}{70 \pi^2 h^2 r^3} \right) \left( \frac{\pi r h^3}{6 \mu Q} \right) = \frac{27 \rho Q h}{420 \pi \mu r^2} = \frac{27 h}{210. \text{RE}(\frac{r}{h})}
\]

(3.10)

The ratio \( R_{iv} \) may be made very small by either reducing the Reynolds Number or choosing a smaller term for \((h/r)\) or both. A typical value for \((h/r)\) in a thrust bearing is \(0.5 \times 10^{-3}\) for a bearing load of 4 MN/m². Figure 3.5 is an example of the ratio \( R_{iv} \) plotted against radius. The \( R_{iv} \) values were calculated by using the flow rate given in Figure 3.4. The magnitude of the inertia term is shown to be rather small as the radius is increased. The prediction of pressure distribution without including the inertia term in those regions, say \( r/R_o > 0.2 \), is quite acceptable. But as the radius is reduced, the inertia term is shown to have a stronger effect on the pressure distribution than the viscous term.

There have been a number of investigations to verify the validity of equation (3.9) [28,33,36]. The results showed that at small radius, the predicted pressure without taking any inertia effect into account was much higher than the experimental value. The use of equation (3.9) was shown to be reasonably accurate under laminar flow conditions, but when the Reynolds Number was increased, the onset of separated flow and turbulence would
require a different equation to be used. However, because of the force limitation imposed by the magnetic coil, the single disc valve will preferably be used for low pressure and low flow rate applications, say, less than 70 bar and less than 10 l/min. In this case, equation (3.9) may be applicable as the Reynolds Number would be lower than 2,000, which was suggested by Moller as the transition point between laminar and turbulent flows. Also, the inclusion of the inertia effect should help to improve the accuracy of the hydraulic flow force estimated.

If equation (3.9) is integrated, the pressure on the supply side will be given as

\[ P_{sr} = P_c + \frac{6 \mu Q}{\pi h^3} \ln\left(\frac{R_o}{r}\right) + \frac{27 \rho Q^2}{140 \pi^2 h^2} \left(\frac{1}{R_o^2} - \frac{1}{r^2}\right) \]  

(3.11)

Similarly, the pressure on the discharge side will be

\[ P_{dr} = P_c - \frac{6 \mu Q}{\pi h^3} \ln\left(\frac{R_o}{r}\right) + \frac{27 \rho Q^2}{140 \pi^2 h^2} \left(\frac{1}{R_o^2} - \frac{1}{r^2}\right) \]  

(3.12)

As the radial pressure distribution is available, the hydraulic force acting on the disc can be determined. If the inside of the valve is divided into two control volumes as shown in Figure 3.6, then the forces acting on the left hand disc surface will include a hydrostatic force, a momentum force and a hydraulic flow force{5}.

For the left-hand control volume:

The hydrostatic force \( F_{S1} \) is given by

\[ F_{S1} = P_{sn} \cdot A_n \]

where \( P_{sn} = P_s \cdot \frac{8 \mu Ln_1 \cdot Q}{\pi R_1^4} \)  

(3.13)
The theoretical study of a single disc valve

The momentum force $F_{m1}$ is

$$F_{m1} = \frac{\rho Q^2}{An} \tag{3.14}$$

and the fluid flow force $F_{sf1}$ is

$$F_{sf1} = P_0 \cdot \pi \cdot (R_0^2 - R_1^2) + K_1 \cdot Q/h_1^3 + K_2 \cdot Q^2/h_1^2$$

where $h_1 = x_1 + \text{SHG}$

$$K_1 = 3 \cdot \mu \cdot (R_0^2 - R_1^2) \cdot (2 \cdot \ln(R_0/R_1) + 1) \quad \text{and}$$

$$K_2 = \frac{27 \cdot \rho}{140 \cdot \pi} \cdot \left(\frac{(R_0^2 - R_1^2)/R_0^2 - 2 \cdot \ln(R_0/R_1)}{2}ight) \tag{3.15}$$

The total hydraulic force on the left face of the disc is therefore given as

$$F_1 = F_{s1} + F_{m1} + F_{sf1} \tag{3.16}$$

By similar arguments, the hydraulic forces acting on the right surface of the disc are given as:

$$F_{s2} = P_{dn} \cdot An$$

where $P_{dn} = P_d + \frac{8 \cdot \mu \cdot \ln_2 \cdot Q}{\pi \cdot R_1^4} \tag{3.17}$

$$F_{m2} = \frac{\rho Q^2}{An} \tag{3.18}$$

$$F_{sf2} = P_0 \cdot \pi \cdot (R_0^2 - R_1^2) - K_1 \cdot Q/h_2^3 + K_2 \cdot Q^2/h_2^2 \tag{3.19}$$

where $h_2 = x_2 + \text{DHG}$ and

$K_1$ and $K_2$ are defined the same as in equation (3.15) above.
The total force acting on the right hand surface is

\[ F_2 = F_{s2} + F_{m2} + F_{sf2} \]  \hspace{1cm} (3.20)

This latter force is acting in opposition to the one generated on the left-hand surface. Therefore, the net force that drives the disc towards the discharge nozzle is given as

\[ F_{hyd} = F_1 - F_2 \]  \hspace{1cm} (3.21)

Equation (3.21) implicitly contains the forces due to hydrostatic pressure and due to fluid flow. If the flow rate and the channel width are known, the resultant hydraulic force can be found. In the previous section, the flow rate was estimated by applying the relationship between the discharge coefficients and the Reynolds Number. If these flow rates are substituted into equation (3.21), it provides an estimate of the hydraulic forces. Once the hydraulic forces are found, the size of the magnetic coil as well as the wavy washer stiffness can be decided.

### 3.3 Dynamic Hydraulic Force

When the disc is moving across the chamber, the Navier-Stokes equation given in equation (3.7) above can no longer be simplified. The axial velocity \( w \) is now represented by the disc velocity and a change of radial velocity with respect to time \( \frac{du}{dt} \) should also be expected. The pressure instead of just being a function of radius as it is in the steady-state case will also be a function of time. Again, the pressure change in the axial direction is ignored as compared with the pressure change in the radial direction.
Numerous efforts had been made to find a solution for the dynamic pressure distribution given by equation (3.7) \cite{40-45}. However, most of these approaches were based on squeezing a thin film of fluid between two parallel discs. The solutions derived, although proven to be accurate by experiments, are unlikely to be relevant to the single disc valve. The reason for this is that in addition to the squeezing flow in the single disc valve, there is also a source flow at the supply side and a sink flow at the discharge side. The only solution which is shown to be appropriate is the one obtained by Warinner and Pearson\cite{45}. In their outward flow model, a source flow similar to the one in the single disc valve had also been included. The pressure distribution was found as

$$\frac{\partial P_{sr}}{\partial r} = \frac{6\mu r x_1}{h_1^3} - \frac{6\mu Q}{\pi r h_1^3} - \frac{3\rho (Q - \pi r^2 x_1^2)}{5\pi r h_1} + \frac{27\rho Q^2}{70\pi^2 h_1^2 r^3}$$

$$+ \frac{24\rho Q x_1}{35\pi r h_1^2} - \frac{15\rho r x_1^2}{14 h_1^2}$$

Equation (3.22) is derived with the use of

$$\int_0^{h_1} \frac{1}{2\pi r dz} = Q - \pi r^2 x_1^2$$

as the continuity equation, where $x_1$ is positive when the disc is moving toward the discharge nozzle.

In deriving a similar pressure distribution equation for the inward flow, the Navier-Stokes equation of (3.7) is modified to

$$\rho (\partial u/\partial t - u(\partial u/\partial r) + w(\partial u/\partial z)) = \partial P_{sr}/\partial r$$

$$- \mu ((1/r(\partial (r u)/\partial r))/\partial r) - \partial^2 u/\partial z^2)$$

The derivation of equation (3.24) can be found in Appendix A2.
The theoretical study of a single disc valve is based on the continuity equation used is

\[ \int_{0}^{h_2} u_2 \pi r dr = Q + \pi r^2 x_2 \]  

(3.25)

where \( x_2 \) is negative for the disc moving towards the discharge nozzle.

Applying the iteration method, the inward flow pressure distribution can be shown as (Appendix A3)

\[ \frac{\partial P_{dr}}{\partial r} = \frac{6 \mu Q x_2}{h_2^3} + \frac{6 \mu Q}{\pi r h_2^3} + \frac{3 \rho (Q + \pi r^2 x_2)}{5 \pi r h_2} + \frac{27 \rho Q^2}{70 \pi^2 h_2^2 r^3} \]

\[ - \frac{24 \rho Q x_2}{35 \pi r h_2^2} - \frac{15 \rho r x_2^2}{14 h_2^2} \]  

(3.26)

Integrating equations (3.22) and (3.26) and putting \( x_1 = -x_2 \), the pressures on both the supply and discharge sides are found as

\[ P_{sr} = P_c + \frac{6 \mu Q}{\pi h_1^3} \ln \left( \frac{R_0}{r} \right) - \frac{3 \mu x_1 (R_0^2 - r^2)}{h_1^3} \]

\[ - \frac{24 \rho Q x_1}{35 \pi h_1^2} \ln \left( \frac{R_0}{r} \right) + \frac{15 \rho x_1^2 (R_0^2 - r^2)}{28 h_1^2} \]

\[ + \frac{3 \rho Q}{5 \pi h_1} \ln \left( \frac{R_0}{r} \right) - \frac{3 \rho x_1}{10 h_1} (R_0^2 - r^2) \]

\[ + \frac{27 \rho Q^2}{140 \pi^2 h_1^2} \left( \frac{R_0^2}{r^2} - 1 \right) \]  

(3.27)

\[ P_{dr} = P_c - \frac{6 \mu Q}{\pi h_2^3} \ln \left( \frac{R_0}{r} \right) + \frac{3 \mu x_1 (R_0^2 - r^2)}{h_2^3} \]

\[ - \frac{24 \rho Q x_1}{35 \pi h_2^2} \ln \left( \frac{R_0}{r} \right) + \frac{15 \rho x_1^2 (R_0^2 - r^2)}{28 h_2^2} \]

\[ - \frac{3 \rho Q}{5 \pi h_2} \ln \left( \frac{R_0}{r} \right) + \frac{3 \rho x_1}{10 h_2} (R_0^2 - r^2) \]
If the disc is stationary, i.e. when $x_1 = x_2 = 0$, then the two equations above will become the same as equations (3.11) and (3.12).

By integrating equations (3.27) and (3.28), the dynamic hydraulic force expressed in terms of disc velocity can be found as

$$F_{df1} = P_0 \pi \left( R_o^2 - R_i^2 \right) + \frac{6 \mu K_3 Q}{h_1^3} x_1 - \frac{3 \mu K_4}{h_1^3} x_1 - \frac{24 K_3 \rho Q x_1}{35 h_1^2}$$
$$+ \frac{15 K_4 P x_1^2}{28 h_1^2} + \frac{3 K_3 P}{5 h_1} - \frac{3 K_4 P}{10 h_1} + \frac{27 \rho K_5 Q^2}{140 \pi h_1^2}$$

(3.29)

$$F_{df2} = P_0 \pi \left( R_o^2 - R_i^2 \right) - \frac{6 \mu K_3 Q}{h_2^3} x_1 + \frac{3 \mu K_4}{h_2^3} x_1 - \frac{24 K_3 \rho Q x_1}{35 h_2^2}$$
$$+ \frac{15 K_4 P x_1^2}{28 h_2^2} + \frac{3 K_3 P}{5 h_2} + \frac{3 K_4 P x_1}{10 h_2} + \frac{27 \rho K_5 Q^2}{140 \pi h_2^2}$$

(3.30)

where

$$K_3 = \left( R_o^2 - R_i^2 \left( 2 \ln \left( \frac{R_o}{R_i} \right) + 1 \right) \right) / 2$$

$$K_4 = \pi \left( R_o^2 - R_i^2 \right)^2 / 2$$

$$K_5 = \left( \left( R_o^2 - R_i^2 \right) / R_o^2 - 2 \ln \left( R_o / R_i \right) \right)$$

These two equations represent the dynamic flow force in radial outward and radial inward directions. In a later section, a dynamic simulation program will be presented and it will see how these equations are applied to predict the dynamic disc response.

In addition to the hydraulic flow force described above, there is another force also introduced by the movement of the disc. This force is caused by
the frictional effect developed between the disc and the insulated transducer rings. As there is no restriction for the oil to go from the left-hand side chamber to the right-hand side, it is assumed that the pressures at the edges on both sides of the disc are equal. The shear stress across the radial gap is therefore given as

\[ \tau = \frac{\mu \cdot \dot{x}_1}{rg} \]  

The induced frictional force is

\[ F_{frn} = \tau \cdot \pi \cdot D_d \cdot \dot{x} \]

\[ = \frac{\mu \cdot \pi \cdot D_d \cdot \dot{x} \cdot \dot{x}_1}{rg} \]

### 3.4 Electro-magnetic Force

The equations for finding the magnetic force are well established and may be derived as the follows:

\[ F_{mag} \cdot \dot{x}_g = H_g \cdot B_g \cdot \text{volume/2} \]  

\[ \text{work done} \quad \text{energy input} \]

Since \( H_g = \frac{B_g}{(\mu_0 \cdot \mu_g)} \) and volume = \( A_c \cdot x_g \), therefore \( F_{mag} \) may also be written as

\[ F_{mag} = B_g^2 \cdot A_c / (\mu_0 \cdot \mu_g) \]  

Equation (3.34) is commonly used for plunger type magnets. For flat-faced armature type magnets, the equation is slightly modified to take into account the two working surfaces available. The magnetic force is finally
The above equation suggests that the magnetic force can be evaluated if the flux density across the gap is known. The calculation of flux density is generally made by analysing the magnetic circuit of the magnetic coil considered. The circuit shown in Figure 3.7 is the one to use in a flat-faced armature magnet. The equation for the balance of magnetomotive force in the circuit is

\[ N.I_c = H_g.(2.x_g) + \sum H_i.l_i \]

\[ = 2.B_g.x_g/(\mu_0\mu_g) + \sum H_i.l_i \]  (3.36)

where \( I_c \) is the steady-state coil current.

The left-hand term of equation (3.36) is the input magnetomotive force to the electric winding. This magnetomotive force is converted into magnetic energy, which is taken up partly by the gap and partly by the magnetic core as shown by the two terms given on the right-hand side. If the relationships between the flux density at the gap and the flux densities at other parts of the magnetic circuit are known and the magnetization characteristic of the magnetic material is available, then for a given input current, the magnetic flux density, \( B_g \), at the gap can be found by using equation (3.36).

Roters{14} in his study of electro-magnetic devices had made a comprehensive study of how the flux densities in a magnetic coil could be related with each other. He had also developed some empirical formulas to show the effects of the fringing and leakage fluxes. The following is a list of the suggested relationships and the equations which would enable
the permeances to be found.

\[
\phi_g = \frac{P_u \phi_y}{(P_a + P_1)} \quad (3.37)
\]

\[
\phi_a = \frac{P_a \phi_y}{(P_a + P_1)} \quad (3.38)
\]

\[
\phi_1 = \frac{P_1 \phi_y}{(P_a + P_1)} \quad (3.39)
\]

Inner pole fringing permeance

\[
P_{f1} = 3.26.\mu_0.\mu_g.\pi.g.r_2 + 4.\mu_0.\mu_g.\pi.g.r_2 \ln((r_3 - r_2)/(\pi.x_g)) \quad (3.40)
\]

Outer pole fringing permeance

\[
P_{f0} = 1.63.\mu_0.\mu_g.r_2 + 2.\mu_0.\mu_g.r_2 \ln(1 + r_2/x_g)
+ 3.26.\mu_0.\mu_g.r_2 + 4.\mu_0.\mu_g.r_2 \ln((r_3 - r_2)/(\pi.x_g)) \quad (3.41)
\]

Useful permeance at working gap

\[
P_1 = \pi.\mu_0.\mu_g.(r_2^2 - r_1^2)/x_g \quad (3.42)
\]

Total permeance between pole cores by way of armature

\[
P_a = 1/(1/(P_1 + P_{f1}) + 1/(P_1 + P_{f0})) \quad (3.43)
\]

Total useful permeance

\[
P_u = \pi.\mu_0.\mu_g.(r_2^2 - r_1^2)/(2.x_g) \quad (3.44)
\]

Leakage permeance between pole cores

\[
P_1 = \mu_0.\mu_g.(1.57.H.(r_3 + r_2)/(r_3 - r_2)
- (r_3 + r_2).(1 - \pi.x_g/(r_3 - r_2))/2) \quad (3.45)
\]

It is shown in equations (3.37), (3.38) and (3.39) that the magnetic fluxes at other parts of the magnetic circuit are all expressed in terms of the flux at yoke. Therefore, if the cross-sectional area of the core is uniform, any yoke flux density assumed will enable the flux densities at other parts of the core to be found.
A computer program has been written in Fortran to calculate the magnetic force. The flow diagram of the program is shown in Figure 3.8. Both the current and the disc displacement are used as the input variables. The program begins with the calculation of the output magnetomotive force on the right hand side of equation (3.36) by assuming a small yoke flux density. This calculated magnetomotive force is compared with the input magnetomotive force. If the former one is larger, the computation stops. If it is smaller, the computation is iterated with a higher yoke flux density until the condition for termination is satisfied. The flux density across the gap is therefore found and is substituted into equation (3.35) to calculate the magnetic force. Figure 3.9 shows a typical force-displacement relationship calculated by the software program. It is noted that the magnetic force decreases rapidly as the armature moves away from the core face.

The above sections have reviewed the hydraulic and the electro-magnetic aspects of the single disc valve. It has described how the flow rate may be estimated by using the experimental discharge coefficients and how the net hydraulic force under laminar conditions can be found. The assumption that the flow at the disc surface is entirely laminar is subjected to argument. The problem is due to the flow separation as illustrated in Figure 3.2. This separated flow could have a considerable effect on the hydraulic force estimated. It was studied by Keith that the effect of flow separation varied directly with the Reynolds number[46]. If the Reynolds number was sufficiently high, say, at 2,000 as suggested by Moller, the flow condition could become entirely turbulent. However, because the single disc valve operates at low Reynolds numbers, the effect of separation on the hydraulic force is assumed to be negligible. In any case, the area of the disc which
might be affected by turbulent flow is small compared with the area that is covered by laminar flow.

In the following sections, a simulation study of the static and dynamic characteristics of a single disc valve is made. The steady-state and dynamic force equations developed before are used to calculate the force interaction thus allowing the change of disc position in the valve chamber to be estimated.

### 3.5 Software Simulation

The software simulation is developed to predict the static and dynamic characteristics of a single disc valve using the equations derived in the previous sections. The use of the steady-state program is to ensure that the forces acting on the disc are correctly balanced and the magnetic coil develops enough force to modulate the disc position. For the dynamic program, it is aimed to predict the response of the disc in reaction to a step change of demanded position under open-loop condition.

#### 3.5.1 Steady-state Program

In this program, the steady-state hydraulic and magnetic forces are calculated. Figure 3.10 shows the force calculation by assuming different disc positions and different coil currents. The primary objective of the program is to obtain the combined hydraulic-magnetic force and plot it against the disc displacement so that a correct wavy washer stiffness could be selected and the maximum current required to operate the valve successfully could be estimated.
In Section 3.2, the steady-state hydraulic force was found to be a function of pressure drop, oil flow rate and disc position. It was also shown that if the pressure drop is constant, the oil flow rate can be estimated from the experimental discharge coefficients. The first calculation in the steady-state program is to calculate the oil flow rate. Based on the estimated flow rate and the corresponding disc position, the hydraulic force is found by applying equation (3.21). Curve A in Figure 3.11 shows a typical result of the calculated hydraulic forces plotted non-dimensionally against the disc displacement at a pressure drop of 42.0 bar. The valve dimensions are based on the ones given in Section 2.2.

Similarly, by using equations (3.35) and (3.36), the magnetic forces at any specified current can be evaluated. Curves B1, B2 and B3 in Figure 3.11 are the results of the calculated magnetic forces plotted against the disc displacement using different coil currents. The dimensions of the magnetic coil are based on the ones shown in Figure 3.7a.

As the hydraulic and magnetic forces are both acting in the same direction, the total force acting on the disc is obtained by adding the two forces together. Curve C shown in Figure 3.11 is the combined magneto-hydraulic force by adding curve A and curve B together.

The selection of a correct wavy washer is governed by two factors. Firstly, the initial disc position must not exceed the centre of the disc travel over the range of pressure drops considered. This is to ensure that the disc travel is not reduced too much so as to affect the maximum flow. Secondly, the maximum spring force produced must not be higher than the combined magneto-hydraulic force otherwise the valve will never be closed.
even with the maximum current applied. Curve D of Figure 3.11 shows the force-displacement of a particular wavy washer. The intersection between this curve and curve A represents the initial steady-state disc position. This initial position can be altered by a different washer stiffness or a different pre-compression of the wavy washer.

When the magnetic coil is energised, the steady-state operating points are determined by the intersections between the combined magneto-hydraulic force and the spring force. These intersections are shown as points M and N in Figure 3.11. The maximum current to be required is roughly estimated by checking the intersecting point right at the end of the disc travel. If the calculating procedures described above are repeated at different pressure drops, then the new maximum current required can be found and the washer stiffness can also be selected. For pressure drop up to 70 bar, the maximum current required was found to be around 1.5 A at a wavy washer stiffness of 140 N/mm.

3.5.2 Dynamic Program

To simulate the dynamic response of the disc, a solution to the following set of first-order differential equations is required:

\[
\begin{align*}
\dot{z} &= x_1 \\
\dot{z} &= x_1 \\
\dot{z} &= (F_{\text{hyd}} + F_{\text{mag}} - F_{\text{sprg}})/m \\
Q &= (dQ/dx_1)\dot{x}_1 \\
\dot{i}_C &= (V_a - i_C(R_C + (dL_C/dx_1)\dot{x}_1))/L_C \\
\dot{L}_C &= (dL_C/dx_1)\dot{x}_1
\end{align*}
\]

These five first-order differential equations are shown to be inter-related
with each other. For example, the magnetic force in equation (3.47) will not be available unless the coil current in equation (3.49) is solved. A similar dynamic analysis for solenoid-operated hydraulic valves was made by Korn and Simpson\cite{47} who used an analog computer method to solve for the differential equations. In this research study, numerical methods are used instead for finding the solutions of the above differential equations.

The calculation of hydraulic force is based on the use of equations (3.29) and (3.30) in Section 3.3 to take into account the additional viscous and inertia effects caused by the disc movement. The magnetic force is found in the same manner as in the steady-state case except that the current is now a function of the disc velocity as well as the coil inductance.

In simplifying the computing procedure, the oil flow rate is assumed to vary as a parabolic function of disc displacement rather than calculated with the discharge coefficients. From the results of the experiment to be presented in a later chapter, it is found that the shape of curve showing the change of oil flow rate with disc displacement can be roughly represented by a parabolic function. The oil flow rate is thus written as

\[ Q = 4.\frac{Q_{\text{max}}}{TD}(\frac{x_1}{TD})(1 - (\frac{x_1}{TD})) \]  

(3.51)

where TD is the total disc travel and \( Q_{\text{max}} \) is the maximum flow rate occurring at mid-travel.

Differentiating equation (3.51) w.r.t. \( x_1 \),

\[ \frac{dQ}{dx_1} = 4.\frac{Q_{\text{max}}}{TD}(1 - (2\frac{x_1}{TD}))/TD \]  

(3.52)

Since the maximum flow rate varies with the pressure drop, this parameter is entered as one of the input variables to the dynamic program. The integration method used is the Runge-Kutta-Merson method readily available
from the University's PRIME computing NAG library. Figure 3.12 shows a
typical result of the program calculation after a step change of voltage is
applied across the magnetic coil. The steady-state current determined by
the voltage step is shown to be 1.2 A, although the maximum current to
completely close off the discharge nozzle is 1.5 A. The initial conditions
for all variables given in equations (3.46) to (3.50) are set to zero
except that the initial disc position and the flow rate are determined by
the balance of the hydraulic and the spring forces.
Total Disc Travel:
\[ TD = x_1 + x_2 \]

Figure 3.1 Simplified Internal Layout of a
Single Disc Valve

Figure 3.2 Typical Radial Flow between
Parallel Plates at High
Reynolds Number
Start

Enter Pressure Drop

Do J = 1, 11
\[ x_1/TD = 0.1 \times (J - 1) \]

Begin Iteration

Increment RE

Determine \( C_{ds}, C_{dd} \) with assumed Reynolds No.

Calculate Reynolds No. with Eqn. 3.4

Tolerance satisfy?

Yes

Calculate Flow Rate

No

\[ J > 11? \]

Next J

Yes

Stop

Figure 3.3 Flow Diagram for Flow Rate Estimation
Theoretical Study of a Single Disc Valve

$P_s - P_d = 42.0$ bar

$dn = 2.54$ mm

$TD = 0.4$ mm

Figure 3.4 Typical Estimated Flow Rate by Discharge Coefficient
Theoretical Study of a Single Disc Valve

Figure 3.5 Comparison of Viscous and Inertia Effect for Radial Flow
Figure 3.6 Simplified Valve Interior Diagram
Figure 3.7 a) Construction Details of a Flat-Faced Armature Magnet

Figure 3.7 b) Magnetic Circuit Diagram

\[ d_1 = 5.0 \text{ mm} \]
\[ d_2 = 12.5 \]
\[ d_3 = 22.8 \]
\[ d_4 = 25.6 \]
\[ dt = 4.0 \]
\[ H = 13.0 \]
Start

Do N = 1,5
I_c = 0.2 * N * I_{max}

Calculate Input mmf

Do J = 1, 20
x_g/TD = 0.05 * J

Do K = 1, 100
B_y = 0.02 * K

Calculate Output mmf at Magnetic Coil

Is mmf Input > Output?

Yes
  Calculate Magnetic Force

No
  Next K

Next J

Next N

Stop

Figure 3.8 Flow Diagram for Magnetic Force Calculation
Theoretical Study of a Single Disc Valve

Current:  
1 - 0.3 A  
2 - 0.6  
3 - 0.9  
4 - 1.2  
5 - 1.5

Magnetic Force (N)

Operating Range in Single Disc Valve

Material: Remko Iron

Disc Position (mm)

Figure 3.9 Variation of Magnetic Force with Disc Position at Constant Coil Current
Figure 3.10 Steady-State Force Estimation Flow Diagram

Start

Enter Pressure Drop

Do N = 1, 5
$I_c = 0.2 * N * I_{max}$

Do J = 1, 21
$x_1 / TD = 0.05 * (J - 1)$

Calculate Flow Rate

Calculate Magnetic Force

Calculate Hydraulic Force

Combine $F_{hyd}$ and $F_{mag}$

Next J

Next N

Stop
Theoretical Study of a Single Disc Valve

Steady-State Force

\[ \frac{F}{\Delta P \times An} \]

A - Hydraulic Force
B - Magnetic Force
( at 20, 60 and 100% Maximum Current)
C - Combined Hydraulic-Magnetic Force
D - Spring Force
\( x_0 \) - Initial Position

Figure 3.11 Estimated Steady-State Force Characteristics
Figure 3.12 Estimated Dynamic Response based on a Step Voltage Change across the Electromagnetic Coil.
Chapter 4

Single Disc Valve Control Analysis
4. Single Disc Valve Control Analysis

In the last chapter, it is shown that the stability of the single disc valve can be analysed by graphical methods, i.e. by plotting out the steady-state force-displacement diagrams and finding the intersection between the spring force and the combined hydraulic and magnetic force. The graphical method, however, can only be used when the system is open-loop stable. On the other hand, the information given shows only the steady-state valve condition. The development of the open-loop dynamic simulation program has brought the analysis a step forward by showing the response of the disc due to a sudden change of input signal. Such results of the dynamic response are useful in terms of knowing the switching time of the disc from one position to another.

However, to actually predict the stability of the valve under closed-loop control involves the analysis of complex non-linear equations. It may be possible to develop the dynamic simulation program for checking the valve stability but the difficulty still remains in identifying which parameter changes improve the valve stability and which have a detrimental effect. In view of this, the analysis of the single disc valve under closed-loop control has been undertaken using linearisation techniques.

4.1 Open-Loop Analysis

Figure 4.1 shows the change of hydraulic and magnetic forces with disc displacement. Assuming that the valve is initially stable at point 0. If the pressure drop across the valve is constant, the following relationships are derived:

\[ \Delta F_{\text{hyd}} = -K_h \Delta x_1 \]  (4.1)
\[ \Delta F_{\text{mag}} = K_m \Delta i_c + K_x \Delta x_1 \quad (4.2) \]
\[ \Delta F_{\text{sprg}} = K_s \Delta x_1 \quad (4.3) \]

where \( K_h = \frac{\partial F_{\text{hyd}}}{\partial x_1} \) at point 0,

\[ K_m = \left. \frac{\partial F_{\text{mag}}}{\partial i_c} \right|_{i_2=i_0} \quad \text{and} \]
\[ K_x = \left. \frac{\partial F_{\text{mag}}}{\partial x_1} \right|_{i_2=i_0}. \]

Introducing the Laplace operator \( s \) the force balance equation is given as

\[ \Delta F_{\text{hyd}} + \Delta F_{\text{mag}} - \Delta F_{\text{sprg}} = (m \cdot s^2 + B \cdot s).\Delta x_1(s) \quad (4.4) \]

where \( B \) is the damping coefficient due to the hydraulic effect.

It is shown in Chapter 3 that the dynamic hydraulic force is composed by the steady-state force and the forces due to the disc velocity and acceleration. This is described in either equation (3.29) or (3.30). If the damping of the single disc valve is derived from these latter forces, as they are acting in opposition to the disc movement, it will then be possible to estimate the damping coefficient by using the dynamic hydraulic force equation. It can be shown that among all the forces caused by the disc velocity and acceleration, the one due to viscosity has the most dominant effect. Therefore, if all the other forces are ignored, the following relationship may be established:

\[ B = 3 \mu K \cdot \left( \frac{1}{h_1^3} + \frac{1}{h_2^3} \right) \quad (4.5) \]

where \( K = \pi (R_o^2 - R_i^2)^2 / 2 \) and \( h_1 \) and \( h_2 \) are the clearances between the disc surfaces and the chamber faces at the supply and discharge sides respectively.
Figure 4.2 shows a typical change of the damping coefficient $B$ with disc position. It is shown that because of a smaller holding gap used at the supply side, a stronger resistive force is generated as the disc approaches the supply nozzle and therefore a higher damping coefficient is experienced.

The current-voltage relationship at the magnetic coil is formulated as follows. If $Δv_a$ is the voltage entered to the coil amplifier, which has a current to voltage gain of $K_a$, then the voltage drop across the magnetic coil is

$$Δv_c(s) = K_a R_c Δv_a(s) = (R_c + sL_c) Δi_c(s)$$

i.e. $K_a R_c Δv_a(s) = (R_c + sL_c) Δi_c(s)$

The transfer function of the combined coil amplifier and magnetic coil is therefore given as

$$\frac{Δi_c(s)}{Δv_a(s)} = \frac{K_a R_c}{R_c + sL_c} = \frac{K_a}{T_c s + 1}$$

where $T_c = L_c/R_c$.

For $R_c = 15Ω$ and $L_c = 5 \text{ mH}$ by measurement, $T_c$ is found to be 0.333 msec. Figure 4.3 shows the relationship between the coil current and the input voltage at the coil amplifier. $K_a$ is found to be about 0.6 A/V.
Substituting equations (4.1), (4.2) and (4.3) into equation (4.4) and re-arranging

\[ K_m \Delta i_c = (m.s^2 + B.s + (K_s + K_h - K_x)) \Delta x_1(s) \]  

(4.7)

Then by combining equations (4.6) and (4.7),

\[ \frac{K_m K_a \Delta v_a(s)}{T_c s + 1} = (m.s^2 + B.s + (K_s + K_h - K_x)) \Delta x_1(s) \]

The open-loop transfer function is therefore given as

\[ G(s) = \frac{\Delta x_1(s)}{\Delta v_a(s)} = \frac{K_m K_a}{(T_c s + 1)(m.s^2 + B.s + (K_s + K_h - K_x))} \]  

(4.8)

The block diagram shown in Figure 4.4 represents this transfer function together with the terms to generate the flow. Equation (4.8) immediately defines the condition for open-loop stability. This condition is that \((K_s + K_h - K_x)\) must be positive so that no positive pole exists in the right-hand s-plane.

### 4.2 Electronic Controller

In the control of disc movement, the current is supplied to the magnetic coil via a coil driver. The construction of this coil driver is shown in Figure 4.5. It consists of a power transistor and an input potentiometer. When the voltage at the output of the potentiometer is higher than the transistor biasing voltage, the transistor is turned on and current is
conducted to the magnetic coil. The current flow increases with the biasing voltage until the transistor saturates, where further increase in the biasing voltage causes no further increase in the current flow. The mathematical relationship between the transistor input voltage and the output coil current may be derived as follows:

\[ I_c = \beta \cdot I_b \]  
\[ I_e = I_c + I_b \]  
\[ V_b = I_b \cdot R_b + V_{BE} + I_e \cdot R_e \]  
\[ V_{cc} = I_c \cdot R_c + V_{CE} + I_e \cdot R_e \]

Substituting \( I_c = \beta \cdot I_b \) into \( I_e = I_c + I_b \),

\[ I_e = (1 + \frac{1}{\beta}) \cdot I_c \]  

Replacing \( I_b \) and \( I_e \) in \( V_b = I_b \cdot R_b + V_{BE} + (1 + \beta) \cdot I_c \cdot R_e \) with \( I_c \),

\[ V_b = V_{BE} + \frac{R_b}{\beta} + \frac{R_e}{\beta} \cdot I_c \]  

or

\[ I_c = \frac{V_b - V_{BE}}{\left( \frac{R_b}{\beta} + \frac{R_e}{\beta} \right)} \]  

Voltage \( V_{cc} \) may be found by combining equations \( I_e = I_c + I_b \), \( V_{cc} = I_c \cdot R_c + V_{CE} + I_e \cdot R_e \) and \( V_b = V_{BE} + \frac{R_b}{\beta} + \frac{R_e}{\beta} \cdot I_c \).

The coil driver described above can never be used for good control of disc position. The reason for this is any change in transistor characteristic or coil resistance due to temperature can cause a change in coil current. Therefore, the magnetic force generated is likely to drift with temperature. A solution to this is to apply current feedback\(^{[48]} \) to the coil driver as shown in Figure 4.6 so that the coil current can be held constant regardless of any change due to temperature.
However, in implementing position control, the use of current feedback is not quite adequate. This is because the disc, although being driven by a constant magnetic force, may shift in position due to load disturbances and for this reason a special controller is required. In order to eliminate the steady-state position error, a proportional-plus-integral (PI) controller as shown in Figure 4.7 is constructed. The output of the controller is connected to the current feedback circuit shown in Figure 4.6.

To avoid any saturation at the integrator or in turn at the PI controller, the integrator output is clamped down by two zener diodes. The cut-off voltage of these two diodes has been selected such that it is high enough to produce the necessary current required by the magnetic coil. The maximum current to be generated for the present application is 1.5 A. It may be noticed that passing a negative signal to the coil driver would produce no effect on the disc. The negative signal is treated the same as a zero signal which stops the transistor from biasing.

4.3 Closed-Loop Analysis

For the valve operating in closed-loop, the linearised model in Figure 4.4 is extended as shown in Figure 4.8. A PI controller is included in the circuit to generate the necessary voltage required by the magnetic coil. The disc position signal is feedback to a summing junction where it is compared with the demanded position to produce the error signal. The forward transfer function of the system is thus modified as

\[ G'(s) = \frac{\Delta x_1(s)}{\Delta e(s)} \]
\[ G'(s) = \frac{K_mK_e(K_p + K_i/s)}{(T_0.s + 1).(m.s^2 + B.s + (K_s + K_h - K_x))} \]

\[ = \frac{K_iK_mK_e(K_{pi}.s + 1)}{s.(T_0.s + 1).(m.s^2 + B.s + (K_s + K_h - K_x))} \quad (4.15) \]

where \( K_{pi} = K_p/K_i \).

Given that the transfer function of the error signal is

\[ e(s) = \frac{1}{v_i(s) \cdot 1 + G'(s)\cdot H(s)} \]

it can be shown that the controller integrator is an essential element for zero steady-state position error (Appendix A6).

It was shown in Chapter 2 that the transfer function of the position transducer is

\[ H(s) = \frac{\Delta v_f(s)}{\Delta x_1(s)} \]

\[ = \frac{K_f}{(1 + (C_{f2}.R_{f1} + C_{f2}.R_{f2}).s + C_{f1}.C_{f2}.R_{f1}.R_{f2}.s^2)} \quad (4.16) \]

By combining equations (4.15) and (4.16), the closed-loop transfer function will be given as

\[ \frac{C(s)}{R(s)} = \frac{G'(s)}{1 + G'(s)\cdot H(s)} \]

\[ = \frac{K_{op} (K_{pi}.s + 1).(E.s^2 + D.s + 1)}{s.(T_0.s + 1).(m.s^2 + B.s + (K_s + K_h - K_x)) (E.s^2 + D.s + 1) + K_{op} (K_{pi}.s + 1)} \]

(4.17)
where \( K_{op} = K_i.K_a.K_m \),
\[ K_{cp} = K_i^2.K_a.K_m.K_f, \]
\[ D = C_{f2}.R_{f1} + C_{f2}.R_{f2} \]
and \( E = C_{f1}.C_{f2}.R_{f1}.R_{f2}. \)

If the following data is used in the analysis,

\[
\begin{align*}
  m &= 10 \text{ g}, \\
  K_s &= 140 \text{ N/mm (Appendix A4)}, \\
  K_a &= 0.6 \text{ A/V}, \\
  K_f &= 5 \text{ v/mm}, \\
  K_m &= 50 \text{ N/A (Appendix A5),} \\
  B &= 100 \text{ N.sec/m}, \\
  T_c &= 0.333 \text{ msec}, \\
  K_x &= 50 \text{ N/mm}, \\
  K_i &= 100 \text{ N/mm}, \\
  D &= 1.41 \times 10^{-3} \text{ sec and } E = 4.44 \times 10^{-7} \text{ sec}^2,
\end{align*}
\]

then on substituting into equation (4.17), the closed-loop transfer function becomes

\[
C(s) = \frac{K_{op}.(K_{pi}.s + 1).(3.78 \times 10^{-7}.s^2 + 1.26 \times 10^{-3}.s + 1)}{R(s)}
\]

\[
D = 1.41 \times 10^{-3} \text{ sec and } E = 4.44 \times 10^{-7} \text{ sec}^2,
\]

then on substituting into equation (4.17), the closed-loop transfer function becomes

\[
C(s) = \frac{K_{op}.(K_{pi}.s + 1).(3.78 \times 10^{-7}.s^2 + 1.26 \times 10^{-3}.s + 1)}{(1.48 \times 10^{-12}.s^6 + 2.394 \times 10^{-8}.s^5 + 1.3695 \times 10^{-4}.s^4 + 0.35796.s^3 + 4.3117 \times 10^2.s^2 + 1.9 \times 10^5.s) + K_{cp}.(s.K_{pi} + 1)}
\]

The characteristic equation is given as

\[
(1.48 \times 10^{-12}.s^6 + 2.394 \times 10^{-8}.s^5 + 1.3695 \times 10^{-4}.s^4 + 0.35796.s^3 + 4.3117 \times 10^2.s^2 + 1.9 \times 10^5.s) + K_{cp}.(s.K_{pi} + 1) = 0
\]

Figure 4.9 shows the root locus plot of equation (4.18). It is noticed that the system remains stable in closed-loop until the loop gain \( K_{op} \) is increased to more than \( 1.4 \times 10^8 \). Since the magnitude of \( K_{pi} \) is chosen to be 5 msec, the maximum proportional and integral gains are thus evaluated as

\[
K_{cp} = 1.4 \times 10^8
\]
\[ = K_i.K_a.K_m.K_f \]
i.e. \( K_i = 933.5 \text{ sec}^{-1} \).
For $K_p = K_{p1}K_1$, therefore

$$K_p = 4.67.$$  

In Figure 4.9, the second and third poles from the right are the two dominant poles which significantly affect the natural frequency of the single disc valve. If these two poles can be placed further away from the origin, the dynamic response of the disc valve could be improved. From the characteristic equation given in (4.19), it is found that these two poles are in fact the roots of the active low-pass filter used in the transducer circuit. In other words, if the dynamic response of the valve is to be improved, the frequency bandwidth of the low-pass filter must be expanded. On the other hand, the natural frequency may also be modified by increasing the damping coefficient and the gradient of the hydraulic force $K_n$ through the use of a smaller holding gap.

It is shown in the above analysis that the stability of the single disc valve operating in closed-loop can be investigated by applying the linearisation method. The method also helps to define what values of proportional gain and integral gain at the PI controller shall be used in order to obtain the desirable response from the disc while at the same time not to introduce any instability to the system. As it is shown in Figure 4.7, the proportional gain can be selected between 0.2 and 7.0, whereas the integral gain can be adjusted from 100 to 2000 sec$^{-1}$. This adequately covers the range required from the root locus stability analysis predictions.
Figure 4.1 Variation of Forces with Disc Displacement
Estimated Damping Coefficient

TD = 0.4 mm
SHG = 0.1 mm
DHG = 0.4 mm

Figure 4.2 Variation of Estimated Damping Coefficient with Disc Displacement (x1/TD)
Figure 4.3 Calibration of Coil Amplifier

Coil Current $I_c$ (A)

Input Voltage increase

Input Voltage decrease

$K_a = \frac{\Delta I_c}{\Delta V_a}$
$F_{hyd} = K_d \cdot \Delta p + (-K_h) \cdot \Delta x$

$Q = K_q \cdot \Delta p + C_x \cdot \Delta x$

$F_{mag} = K_m \cdot \Delta i_c + K_x \cdot \Delta x$

---

**Figure 4.4** Linearised Open-Loop Block Diagram
Figure 4.5 Coil Driver Circuit Diagram

Figure 4.6 Coil Driver with Current Feedback
Figure 4.7 Circuit Diagram of PI Controller
Figure 4.8: Linearised Closed-Loop Block Diagram

Note: \[ H(s) = \frac{K_p}{1 + (C_{f1}R_{f1}C_{f2}R_{f2})s + (C_{f1}R_{f1}C_{f2}R_{f2}s^2)} \]
Figure 4.9 Root Locus Plot of a Closed-Loop Position-Controlled Single Disc Valve
Chapter 5

Experiments with a Single Disc Valve
5. **Experiments with a Single Disc Valve**

Steady-state and dynamic tests have been performed on a single disc valve under closed-loop control of disc position. In the steady-state test, the disc was held at a constant position by the balance of the hydraulic force, the magnetic force and the spring force. The change of oil flow rates with the change of disc positions was investigated. For the dynamic test, changes of disc position was used to obtain the transient response. The step input conditions changed leading to the change of disc position included the voltage applied to the magnetic coil and the pressure drop across the valve.

5.1 **Steady-State Test**

The experimental set-up for the steady-state test is illustrated in Figure 5.1. The same set-up but in schematic form is shown in Figure 5.2. The supply pressure was set to approximately 120 bar and then regulated down to the desired level. Pressure transducers and gauges were connected before and after the valve and at the valve body where the chamber pressure could be measured. The oil flow rate was recorded by two different range turbine meters installed downstream from the valve. The electrical supply to the magnetic coil was monitored via the resistor Re connected in series with the coil shown in Figure 4.6. Two different settings: 10 N and 20 N for the washer pre-compression were used to allow tests to be carried out on the valve in both halves of the disc travel.

Throughout the experiment, the signal from each of the sensors was measured by a data acquisition system. Stored data was then transmitted to the University's PRIME computer for further modification and processing. During
the test, the disc was moved towards the discharge nozzle and then returned to its initial position to test for hysteresis while the pressure drop across the valve was unchanged.

The steady-state test result of the single disc valve under closed-loop position control is shown in Figure 5.3. The controller settings for the proportional and integral gains were 2 and $300 \text{ sec}^{-1}$ respectively. Because of the two orifices present, the oil flow rate is shown to rise to maximum as the disc was moving away from the supply nozzle and then gradually decreased when the disc was approaching the discharge nozzle. The relationship between the currents required at the magnetic coil and the flow rates is shown in Figure 5.4. It is noticed that a smaller current was used as the disc was moving back towards the supply nozzle. The resulting current hysteresis seems to suggest that the net hydraulic force acting on the disc was unequal as the direction of disc movement was reversed. A detailed discussion on the hydraulic force will be given later.

The flow-displacement characteristic shown in Figure 5.3 is a combination of the data collected from the two different pre-compression settings used. The initial steady-state position was determined by the spring pre-load and stiffness. The other boundary on the characteristic was when the magnetic force could not be further increased as the current was at a maximum. This occurred in the low pressure drop case when the disc was still some way from the coil face and that the electro-magnetic force on the disc was not high despite being in the maximum current condition. It is therefore important that the spring is carefully selected to ensure that it covers the range of pressure drops and disc movement over which the valve will be operated. The software simulation described in Chapter 3 was thus developed for this purpose.
Figures 5.5 and 5.6 show the pressure drops at the supply and discharge sides plotted against the disc displacement respectively under constant pressure drops across the valve. The supply side pressure drop was found to decrease by the increasing disc movement, whereas the discharge side pressure drop increased because of the domination by the discharge orifice. If a vertical line representing a constant disc position is drawn on Figures 5.3, 5.5 and 5.6, the relationship between the discharge coefficient and the Reynolds number for a fixed width channel can then be established. According to the results reported by Lichtarowicz[26], the discharge coefficient of a long orifice rises with the Reynolds number until it reaches a maximum. The discharge coefficient which was calculated from equation (3.1) indicated that the flow characteristic at the single disc valve was very similar to a long orifice. Figures 5.7 and 5.8 show the calculated discharge coefficients plotted against the Reynolds number at the supply and discharge sides respectively. The effect of varying the channel width is also found to be the same as reported by Duggins[25], in which he showed the discharge coefficient was higher as the channel width was smaller.

It is noticed that the discharge coefficient at the discharge side was smaller and was less affected by the disc position. The smaller magnitude is believed to be caused by the effect of inward flow. It has been shown in Chapter 3 that the pressure drop for inward flow, given by equation (3.12) is higher because of the combined viscous and inertia effects. Therefore, the discharge coefficient should be smaller when the flow rate and the channel width are the same as they are on the supply side. The minute change in discharge coefficient is suspected to be due to the use of a wider holding gap at the discharge side. When the holding gap is wide, the change in disc position is comparatively small and therefore the change in
Experiments with a Single Disc Valve

discharge coefficient will be less.

Figures 5.9 and 5.10 show the change of discharge coefficients with the Reynolds numbers at the supply and discharge sides respectively when the pressure drops across the valve were constant. It is found that a few discharge coefficients calculated are greater than one. The cause of this is due to the definition of the discharge coefficient used. In Chapter 3, the orifice area is defined as the gap between the nozzle and the disc times the perimeter of the nozzle and the pressure drop is taken as the one from the centre of the nozzle to the edge of the disc. This definition is clearly not the same as the one commonly used for orifice, where the gap, the cross-sectional area and the downstream pressure are all defined at the venturi contraction.

Although the same definition as equation (3.1) was also used by other researchers, their calculated discharge coefficients could be lower than the one described here. Usually the orifice gap is formed by two flat, parallel surfaces when equation (3.1) is applied. In the single disc valve, the nozzles are however made to be higher than the chamber walls to ensure that the valve can be closed properly. As 98% of the flow channel has a total width equal to the sum of the disc position and the holding gap, the use of disc position alone as defined above is likely to result in a higher discharge coefficient.

In studying the hydraulic force acting on the disc, the magnetic coil was not energised. Figure 5.11 shows the change of disc position caused by the hydraulic force as the pressure drop across the valve was increased and decreased. It was found that the disc position hysteresis was significant as illustrated by selecting a vertical line at a pressure drop of 16 bar. The corresponding flow rate across the valve is shown in Figure 5.12. If
the two disc positions found in Figure 5.11 are superimposed on Figure 5.12, two different flow rates are obtained. In Figure 5.3, it is shown that when the disc position \( \frac{x_1}{TD} \) changes from 0.26 to 0.37 at pressure drop equal to 16 bar, the flow rate changes about \( 0.7 \times 10^{-5} \) m\(^3\)/s, less than one-half of the change in Figure 5.12. As the pressure transducers and the turbine meter were calibrated properly, the difference in flow rates seems to suggest that the measured position signal was incorrect and the disc was actually at a location different from the signal indicated.

From the experimental data, a check may be made on the validity of the hydraulic force equations derived in Chapter 3. If the measured pressure drop, flow rate and disc position given in Figures 5.11 and 5.12 are substituted into equation (3.21), the estimated net hydraulic force acting on the disc can be calculated and is shown in Figure 5.13. It is found that in some cases the hydraulic forces are almost 50\% higher than the spring force being shown as line A as the pressure drop is increased in the hysteresis loop. As the pressure drop decreases, the two forces are comparable. The cause of the difference could either be due to the incorrect position signal measured as described above or the limitation of the laminar force equations or both.

In the flow model developed by Yuksel[49], it was suggested that a separation region existed between the inner nozzle edge and an assumed re-attachment radius, which was then followed by laminar flow up to the edge of the disc. The pressure distribution within the separation region was estimated by an orifice equation which enabled the pressure at the venturi radius to be found. Figure 5.14 shows the results of using the separation model comparing with the results of Figure 5.13. It is found that the changes of hydraulic force between the separation and the laminar models
are quite similar in form but the estimated force magnitudes of the former model are even higher, suggesting that separation is not the mechanism for the flow hysteresis.

From the above force comparison, it seems that to find out the limitation of the laminar force equation, a better test set-up is needed so that the effect of separation and disc misalignment can also be investigated. It can be shown that if the measured disc position given above is 25% different, the 50% error in force comparison will disappear. An accurate position signal is thus an essential requirement in the hydraulic force study. However, because of the free floating disc nature, such position accuracy is very difficult to achieve and since the position transducer is developed to measure very fine disc displacement, any tilting occurring is likely to corrupt the position signal. On the other hand, equation (3.21) is derived based on two parallel surfaces. If the disc is misaligned, the estimated hydraulic force could be inaccurate and keeping the flow channel parallel may be necessary. However, as the single disc valve does not conform to this, the validation of the laminar force equations is considered as unsuccessful.

The relationship among the flow rate, the disc displacement and the pressure drop is shown in Figure 5.15. The flow rate is found to increase by both the disc displacement and the pressure drop. However, as the maximum flow rate was reached, which is indicated by curve A in the figure, the increase of the disc displacement further resulted in a lower maximum flow. Although the curves were shown to be parabolic, the assumption for a laminar flow along the parallel channel could still be valid as it is suspected that the parabolic shape was caused by the pressure drop due to the change of flow direction from axial to radial and the flow contraction at the nozzle nose.
5.2 Dynamic Test

The study of the single disc valve dynamic performance was made under two different test conditions. In the first test which was performed in open-loop, a demand voltage step was applied across the magnetic coil under constant pressure drop conditions. The disc being constantly monitored by the position transducer was moved to the next specified position by the resulting change of magnetic force. In the second test, where the single disc valve was operated under closed-loop position control, a pressure step change was applied by a servo-valve across the disc valve at a constant demanded position so as to study the ability of the single disc valve to correct for load changes.

5.2.1 Voltage Step Test

The hydraulic set-up for the voltage step test is shown in Figure 5.16. It is similar to the one used in the steady-state test except that the two turbine meters were disconnected. During the test, the pressure drop across the valve was adjusted to different levels. Any change in the disc response due to the different pressure drops applied was then examined. The voltage step used in the test was generated from a potentiometer used as a potential divider. This potentiometer was separated from the coil driver by a switch as shown in Figure 5.17. Before the step was applied, the coil driver was connected through the switch to an input potentiometer which set the initial disc position. If the voltage was zero, the initial disc position would be determined by the balance of the hydraulic force and spring force.

A second potentiometer, connected when the switched was active, determined
the final voltage applied to the coil. Each test was recorded by the data acquisition system sampling at one millisecond time interval. Variables measured were the valve pressures, the disc position and the coil current.

Figure 5.18 shows one of the results obtained. The disc was moving in a direction towards the discharge nozzle. As the discharge orifice area was reduced, the supply pressure was shown to be affected by the disc movement. The supply pressure was forced to move up as the demand for oil was suddenly decreased. The chamber pressure was shown to change at about the same rate as the disc. This seems to suggest that the compressibility effect was not too significant. The chamber pressure was also found to be affected by the change in supply pressure which was probably a function of the gas accumulator characteristics.

The coil current is shown to grow at a faster rate than the disc movement. The rate of current growth, as was described by equation (3.49), is a function of the coil inductance and disc velocity. When the inductance is low, the current rises more rapidly and the magnetic force produced moves the disc more quickly to the next position. The zero initial current indicates that the magnetic coil had not been driven prior to the voltage step was applied. Figure 5.19 is an example showing that the disc was initially driven to a different position by the magnetic coil. The effect of the voltage step in this case was to increase the coil current to a higher level.

The following table summarises the response times of the disc obtained under different test conditions. The response time is defined as the time taken to move from 10% to 90% of the final movement.
In the table above, the response of the disc was shown to be affected by the pressure drop applied. In tests 2 and 8, for example, the disc was started at about the same position but a shorter response time was found for the higher pressure drop case. The shorter response time is believed to be caused by the increased flow rate across the valve. As the flow rate was increased, a stronger hydraulic flow force was produced. When this hydraulic force was added to the magnetic force, the disc was accelerated faster towards the discharge nozzle.

For a constant pressure drop, the response time was found to be affected by the position at which the disc was placed. The response time was shown to be longer when the disc was moving at the same distance but in a region closer to the discharge nozzle. These results can be found between tests 1 and 2, 4 and 5, and 10 and 11. The cause of this longer response may be explained as a result of a slower current growth. The current growth was determined by the time constant of the magnetic circuit. When the disc was placed closer to the discharge nozzle, the coil inductance, which gives the time constant, was increased by the reduced working gap. Therefore, the rate of current rise was reduced. The other reason for this slower response could be a result of change in hydraulic force. In the third right-hand
The induced hydraulic force is shown to vary directly with the disc velocity and inversely with the width of the channel between the disc and the chamber wall at the discharge side. If the channel width was smaller, a higher resisting force would be produced and could therefore reduce the speed of the disc.

Interestingly, there was not a single case of disc position overshoot observed during the course of the experiment. The cause of this may be explained as a combination of a reduced hydraulic driving force and an increased resisting force as the disc is moving across the chamber. The decrease of driving force can be described as a result of reduced outward flow in the radial direction. When the disc moves towards the discharge nozzle, the volume at the supply side increases. Most of the oil flows to the opposite side via the radial channel, but a small quantity of oil is used to fill up the extra volume created by the disc movement. This reduction in oil flow results in a smaller pressure drop and therefore a smaller driving force. However, on the discharge side, a resisting force caused by the squeezing of the oil is generated. This is similar to the damping effect in a hydrostatic bearing. In order to discharge the extra oil flow caused by the movement of the disc, the pressure drop is consequently increased and a stronger resisting force results.

The last column in the result table is the estimated response time obtained from the software program given in Chapter 3. The calculations of the disc response are based on the same pressure drop and the same specified movements. Figures 5.20 and 5.21 are the comparisons of two experimental and theoretical responses. The theoretical predictions are shown to be quite accurate. The theoretical response times given in the result table are all shown to be higher than the experimental values. The temporary
increase of supply pressure due to the dynamic response limitation of the accumulator may be a reason for causing the difference.

5.2.2 Pressure Step Test

The hydraulic set-up for the pressure step test is illustrated in Figure 5.22. Figure 5.23 shows the same set-up but in schematic form. The connection of the single disc valve to the hydraulic lines is shown to be made via the servo-valve. Although the servo-valve used was capable of reversing the flow direction across the disc valve, it was used only uni-directionally so that it always supplied oil into the disc valve supply nozzle.

To produce the pressure step, the same electronic circuit was used to generate the voltage step change. In this case, instead of being applied to the coil driver, the voltage was applied to the servo-valve amplifier. When the switch was activated, the amplifier reacted by moving the spool of the servo-valve to a different position. The change of spool position subsequently changed the pressure drops at the inlet and outlet ports. Since the disc valve was connected in series, a pressure step across the valve therefore occurred.

As soon as the pressure step occurred, there was a transient change of disc position before the position feedback control returned it to its original position. The same data acquisition system was used to record the results of the experiments. Figures 5.24 and 5.25 are two of the results obtained with the pressure step increased in one case and decreased in the other.
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Although the voltage at the servo-valve amplifier adjusted very quickly, the chamber pressure at the disc valve changed much more slowly. The rate of change was similar to that of the disc position. The cause of this was probably due to the time constants of the magnetic coil and the integrator in the PI controller. The latter determines the rate at which voltage is supplied to the coil amplifier in order to adjust for the change of pressure drop.

In the results shown in Figure 5.24, the disc was held in position by the electro-magnetic force before the pressure step was applied. As the single disc valve was operated in closed-loop, when the disc was forced to move away from the equilibrium position, the current initially in the coil was reduced by an error signal presented at the electronic controller. The smaller magnetic force then produced allowed the wavy washer to move the disc back towards the supply nozzle. As the disc continued to move back and the error signal continued to diminish, the current eventually dropped to a level which was just enough to hold the disc at its original position. The results in Figure 5.25 can be described in the same way. The disc, instead of moving forward, was moving in an opposite direction towards the supply nozzle. The current was rising to increase the magnetic force in compensation for the loss of hydraulic force caused by the decreased pressure drop.

The pressure test in the single disc valve was performed in two groups. In the first group, the pressure step applied to the valve was varied, but the disc was set to the same initial position before the step was applied. The results of this group are summarised in Figures 5.26 and 5.27. It is noticed that the amount of disc displacement varied directly with the size of pressure step applied. From the results given, the restoring time is
found to vary between 25-30 ms. This restoring time cannot be used as a measure of the single disc valve dynamic performance. They are not like the results in the open-loop voltage step test where the response times are obtained without any controller action.

The disc displacement was found to be greater when the pressure change increased than when it decreased. The cause of the difference was possibly due to the use of a wider holding gap at the discharge side where the dynamic flow force was comparatively small. On the other hand, it is shown in Figure 3.11 that the hydraulic force has a negative slope over part of the disc travel. Therefore, when the disc was forcing back to the supply nozzle as a result of a lower pressure drop applied, a stronger resisting force should be expected.

In the second group of the pressure step test, the pressure step change was kept constant while the disc was placed at different initial positions in the chamber. The results are summarised in Figures 5.28 and 5.29. It is noticed that the disc displacements were getting smaller as the disc was placed closer to the discharge nozzle. This can be interpreted as an increase of damping force as the disc is placed nearer to the chamber surface. This may be explained from equation (3.29) or (3.30) in which it shows the resistive hydraulic force increases as the channel width is reduced.
Figure 5.1 Steady-State Test Hydraulic Set-up
Figure 5.2 Schematic Diagram for Steady-State Test Set-up

1 - Tank
2 - Booster Pump
3 - Filter (also for items 12 and 23)
4 - Relief Valve (also for item 6)
5 - Main Pump
6 - Pressure Gauge (also for items 11, 14, 16, 18 and 22)
7 - Temperature Gauge
8 - Pressure Reducing Station
9 - Directional Valve
10 - Accumulator
11 - Pressure Transmitter (also for items 17 and 19)
20 - Turbine Meter
21 - Check Valve
24 - Cooler
25 - Single Disc Valve
Flow Rate
(x10^-4 m^3/s)

Pressure Drop: Preload 1 Preload 2
Δ,∇ - 4.5 bar +,× - 26.5 bar
+,× - 8.5 □,○ - 30.0
□,○ - 11.0 ○,* - 36.5
0,* - 16.0  Δ,∇ - 23.5 TD = 0.4 mm

Figure 5.3 Steady-State Test Results
at Constant Pressure Drop
Flow Rate
\( \frac{Q}{Q_{\text{max}}} \)

Pressure Drop:
\( \Delta, \nabla \) - 8.5 bar
\( +, \times \) - 26.5

Current (Ic/I_{\text{max}})

Figure 5.4 Typical Flowrate - Current Hysteresis at Constant Pressure Drop
Figure 5.7 Variation of Supply Side Discharge Coefficient with Reynolds Number at Constant Disc Position

Figure 5.8 Variation of Discharge Side Discharge Coefficient with Reynolds Number at Constant Disc Position
Figure 5.11 Change of Disc Position due to Change of Pressure Drop across Valve

Figure 5.12 Change of Flow Rate due to Change of Disc Displacement
Experiments with a single disc valve

Figure 5.13 Variation of Estimated Hydraulic Force with Disc Displacement

Figure 5.14 Comparison of the Estimated Hydraulic Forces between Laminar and Separated Flow Models
Figure 5.15. Variation of Flow Rate with Pressure Drop at Constant Disc Position

$\frac{x_1}{TD} : \triangle - 0.2 \quad \times - 0.5 \quad \nabla - 0.3 \quad \square - 0.6 \quad + - 0.4 \quad \Diamond - 0.7$

Flow Rate (l/min)

Pressure Drop (bar)
Figure 5.16 Schematic Diagram for Dynamic Voltage Step Test Set-up

1 - Tank
2 - Booster Pump
3 - Filter (also for items 12 and 23)
4 - Relief Valve (also for item 6)
5 - Main Pump
7 - Pressure Gauge (also for items 11, 14, 16, 18 and 22)
8 - Temperature Gauge
9 - Pressure Reducing Station
10 - Directional Valve
13 - Accumulator
15 - Pressure Transmitter (also for items 17 and 19)
20 - Single Disc Valve
21 - Cooler
Experiments with a Single Disc Valve

Figure 5.17 Electrical Set-up for Dynamic Step Test
Figure 5.20 Comparison of Dynamic Response of Test No. 2

Figure 5.21 Comparison of Dynamic Response of Test No. 8
Experiments with a Single Disc Valve

Figure 5.22 Pressure Step Test Hydraulic Set-up
Figure 5.23 Schematic Diagram for Dynamic Pressure Step Test Set-up

1 - Tank
2 - Booster Pump
3 - Filter (also for items 12 and 23)
4 - Relief Valve (also for item 6)
5 - Main Pump
6 - Pressure Reducing Station
7 - Pressure Gauge (also for items 11, 17, 19, 21 and 22)
8 - Temperature Gauge
9 - Directional Valve
10 - Accumulator
11 - Pressure Transmitter (also for items 16, 18 and 20)
12 - Servo Valve
13 - Single Disc Valve
14 - Cooler
Figure 5.24
Typical Pressure Step Test
Result with Pressure Step Increasing

Figure 5.25
Typical Pressure Step Test
Result with Pressure Step Decreasing
**Figure 5.26** Summary of Disc Response at Different Increasing Pressure Steps under Constant Disc Position

**Figure 5.27** Summary of Disc Response at Different Decreasing Pressure Steps under Constant Disc Position
Figure 5.28 Summary of Disc Response at Different Disc Positions under Constant Increasing Pressure Step

Figure 5.29 Summary of Disc Response at Different Disc Positions under Constant Decreasing Pressure Step
Chapter 6

Summary of the Single Disc Valve
6. Summary of the Single Disc Valve

The advantage of a capacitive type position transducer is its simplicity in construction and low cost. However, because of the relatively small change of capacitance that occurs in a single disc valve, the measured signal has to be transmitted carefully in order to avoid any loss of sensitivity and noise problems. The most undesirable effect is the drifting of the signal with temperature. The problem may be eliminated by incorporating a temperature compensation circuit in the early stage of the transducer design. It has been shown in many previous studies that using push-pull compensation techniques can be used and at the same time improve the transducer sensitivity.

In addition to the temperature problem, the accuracy of the measured signal can also be affected by the orientation of the disc itself. Because of the high resolution of the position transducer, any misalignment as the disc moves across the chamber could cause an error in the position signal measured. Both of these problems affect the position transducer directly and hence reduce the performance of the closed-loop control system.

The use of discharge coefficient to simplify the calculation between flow rate and pressure drop should prove to be useful. In many previous studies, the discharge coefficient was shown to be a function of gap length and Reynolds number. The estimation of flow rate can therefore be made if the discharge coefficient to be used is expressed in terms of the gap length and the Reynolds number. From the results of Figures 5.7 and 5.8, it was found that the discharge coefficient increased as the gap length was reduced. To develop an empirical formulae, the following method was used:

The data points shown in Figures 5.7 and 5.8 were curve-fitted by the
numerical Chebyshev method and the results of which are shown in Figures 6.1 and 6.2 respectively. Because of the limited data, only one set of data points for each outward and inward flows was curve-fitted. These are the $x_1/TD = 0.4$ for the outward flow and the $0.6$ for the inward flow. These fitted curves were used as the general relationships between the discharge coefficient and the Reynolds number. The other curves shown in the same figures were obtained from these fitted curves and a second-order function given below. This second-order function was developed to modify the general discharge coefficient so that it was made to be a function of disc displacement as well as the Reynolds number.

$$C_{ds} = F\{x_1/\text{TD}, \text{RE}\} = (a.(x_1/\text{TD}) + b.(x_1/\text{TD})^2 + c).f\{\text{RE}\} \quad (6.1)$$

$$C_{dd} = G\{x_1/\text{TD}, \text{RE}\} = (a.(x_1/\text{TD}) + b.(x_1/\text{TD})^2 + c).g\{\text{RE}\} \quad (6.2)$$

where $f\{\text{RE}\}$ and $g\{\text{RE}\}$ are the fitted curves for the outward and inward flows respectively.

The coefficients $a$, $b$ and $c$ were found to be

<table>
<thead>
<tr>
<th></th>
<th>a</th>
<th>b</th>
<th>c</th>
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<tbody>
<tr>
<td>Inward</td>
<td>-0.03</td>
<td>-0.30</td>
<td>1.13</td>
</tr>
<tr>
<td>Outward</td>
<td>-1.60</td>
<td>0.82</td>
<td>1.51</td>
</tr>
</tbody>
</table>

It can be shown that when $x_1/\text{TD}$ is equal to 0.4 in equation (6.1) and 0.6 in equation (6.2), the second-order functions become unity.
With the empirical formulas developed above, a comparison of the experimental results and the estimated flow rates using the discharge coefficients derived was made. In Figure 6.3, the estimated flow rates are shown to be quite accurate except when the pressure drop across the valve is high. However, in Figure 6.4 it is found that the actual discharge coefficient is greater than the one estimated by the parabolic function as the orifice gap is decreased. Thus inaccuracy in the estimated discharge coefficients near the dotted line region shown in Figure 6.3 would be expected.

From the above discussion, it is found that the flow across a disc valve can possibly be estimated by using a simplified discharge coefficient approach. However, to account for the increase of discharge coefficient at small gap lengths, a parabolic function is clearly not adequate. To rectify the problem, more experimental work is needed so that the pressure-flow characteristic in those regions can be explored and the empirical formulae modified. Unfortunately, because of a high hydraulic force present which tends to push the disc away from the supply nozzle, such pressure-flow characteristic cannot be made available from the single disc valve developed.

The study of hydraulic force based on an assumed laminar flow model was not quite successfully. Firstly, the measured position signal needs to be accurate because for only 25% error in disc position, the estimated hydraulic force can be changed by as much as 50% as discussed before. However, to accurately measure the disc position is extremely difficult because the disc is allowed to float freely inside the valve chamber.

Secondly, using laminar flow model is not exactly correct. The presence of a separation region found in many previous studies should also be included.
Although the separation model suggested by Yuksel was tried, the calculated hydraulic force was shown to be even higher than the one obtained by the laminar model. It is suspected that either the venturi and the re-attachment radii were assumed too large or the modeling of separation using an orifice flow approach was inappropriate. As the laminar model is seemed to be able to give a closer estimate, it is suggested that if the local Reynolds number is not higher than 2,000 as described by Moller for transition to turbulent flow, the effect of separation on the estimated hydraulic force could be assumed negligible. On the other hand, from the comparison of the dynamic test results, the estimated disc response was shown to be in good agreement with the experiment. Thus using the laminar force equations may be considered as adequate for general design calculation.

In the dynamic simulation model, the disc is shown to be dampened by a similar mechanism found in a hydrostatic thrust bearing. The oil between the disc and the chamber wall is squeezed as the disc is moving, thereby increasing the pressure drop and producing the damping force. It is found that as the damping force is dominated by the viscous effect, an estimation of the damping coefficient can be made and a possible change of disc response may be achieved by varying the width of the flow channel. It is suspected that because of the laminar equations used, the dynamic simulation model would have the same limitation as the steady-state model and would become inaccurate when the local Reynolds number is raised to 2,000 or higher.

The study of the disc valve stability by the linearisation method has proven to be quite successful. The formulation of an open-loop transfer function helps to determine how the control of disc position may be
Summary of the Single Disc Valve

achieved and through the plotting of the root locus, the type of controller to be required for closed-loop function is also defined. From the result of the analysis, an integrator is found to be essential in order to eliminate the closed-loop steady-state position error. This is also indicated in the pressure step test where the disc was held at the same position despite the change of hydraulic condition. It is noticed that the frequency response of the position transducer used and the magnitude of the magnetic force gradient, \( K_x \), have the most important effect on the dynamic valve performance. The response of a closed-loop system is known to be no better than the response of its feedback component. Therefore, further work on improving the position transducer design is definitely needed if the dynamic performance of the valve is to be improved further.

Although the linearisation method has been proven to be useful, the result obtained could be inaccurate due to the assumptions made. For example, the damping coefficient is assumed to be constant, which in practice would vary with disc position. Also the hydraulic and magnetic force gradients are sensitive to the operating point chosen and stability may not be achieved unless worst case analysis is adopted, making the desired response even more difficult to obtain. Therefore, the use of linearisation method should be limited to checking the valve control stability and finding a suitable controller for closed-loop applications.

In summing up the above discussion, the single disc valve developed can basically satisfy the function of a two-port flow control valve. Because of the method in use for balancing the hydraulic force, the operating range of the valve is shown to be limited by the stiffness of the wavy washer chosen. However, despite the limitation on pressure drop, the single disc valve has been shown to be very fast-reacting. This might be an advantage over conventional solenoid-operated valves which are relatively slow due to
a heavier mass and a less powerful electro-magnetic actuation. Because the valve has a simpler construction and is less sensitive to oil contaminants, by combining more than one single disc valves together as discussed in Chapter 2, it may prove to be an attractive alternative to conventional 3-way or 4-way spool valves.
Figure 6.3 Comparison of Estimated and Actual Flow Rates under Constant Pressure Drops

Figure 6.4 Comparison of Simulated and Actual Discharge Coefficients at Constant Disc Position
SECTION II

An Application Study of Position-Controlled Disc Valve

In the first part of this thesis, the static and dynamic performance of a single disc valve were investigated. The results of the tests demonstrate that by implementing position control on the disc, an electrically modulated variable orifice can be made. The development of the single disc valve has provided some useful information in the understanding of the complex flow behaviour in the disc chamber. For example, the limitation of the laminar theory, the discharge coefficients on both outward and inward radial flows and the dynamic response of the disc. With this information, the design of alternative disc valves for a particular application becomes possible. In this second part of the thesis, the development of a double disc valve designed to operate with a vehicle shock absorber is described. It is aimed at varying the damping characteristic of the shock absorber through the electrical operation of the disc valve.
Chapter 7

A Vehicle Suspension System Application
7. A Vehicle Suspension System Application

The function of any vehicle suspension system is to minimise the vertical movement which will otherwise be transmitted through the wheel to the vehicle body due to the irregularities of the road surface. How this vertical movement is attenuated varies with the system components used. In this chapter, three different suspension systems are described and the use of a disc valve in a semi-active system is considered. The requirements for the disc valve to be developed so as to achieve the desired functions are also given.

7.1 Vehicle Suspension System

A vehicle suspension system will normally consist of a main spring and a damper, which is referred to as the shock absorber. The construction of a conventional vehicle shock absorber is shown in Figure 7.1. It is similar to a double-acting hydraulic cylinder except it is constructed with two concentric tubes and is activated passively. The outer tube(1) acts as a reservoir which holds the excess oil displaced from the compression chamber(2) when the absorber is moving downward.

As the absorber is compressed, the oil moves freely through the piston head(3) via a non-return valve(7) into the rebound chamber(4). The excess oil is pushed across a foot valve(5), which is essentially a mechanical spring valve, attached to the base of the inner tube(6) into the reservoir. The oil pressure within the compression chamber increases and varies with the absorber velocity. The magnitude of this induced pressure is governed by the pressure-flow characteristic of the foot valve used. Since the piston area in the compression chamber side is larger, a force acting in
opposition to the compression is generated and hence the required damping effect is produced.

In the extended stroke, the oil in the rebound chamber is compressed and forced to flow into the compression chamber via a piston valve(8) attached to the piston head. Some of the oil in the reservoir is sucked back into the compression chamber via another non-return valve(9) at the base to make up the difference in displaced volumes between the rebound and the compression chambers. While this is happening, a pressure drop is developed across the two chambers and a force acting against the shock absorber motion is generated. The force-velocity characteristic described may be summarised as Figure 7.2.

The curve shown in Figure 7.2 is due to the effect of the foot valve and the piston valve, which in this case is composed by a bleed orifice and a spring-loaded blow-off valve[50]. At low velocity, the blow-off valve is inoperative. The resistive force is produced entirely by the pressure drop at the bleed orifice. As the pressure is increased by the increasing velocity, the blow-off valve is gradually opened, resulting in a smaller pressure change but a larger volume flow across the two chambers. It is shown that the force produced in the compression cycle is smaller than that of the rebound cycle. This is aimed at limiting the upward jolting forces fed into the vehicle body during axle movements, whereas on the rebound side, similar axle induced downward forces are limited by the fact that the wheel leaves the ground[51].

Figure 7.3 shows the normal suspension elements of a single wheel vehicle model. There are two ranges of frequencies, primarily due to the difference in weights, to be dealt with on a vehicle. The natural frequency of the sprung mass(M) is generally in the order of 1 to 1.5Hz, whereas the natural
frequency of the unsprung mass \( m \) is 8 to 12 Hz. The unsprung mass attached to the lower end of the shock absorber can achieve velocities over the full range of the diagram as shown in Figure 7.2, when excited by severe road disturbances. The sprung mass however, being some ten to twenty times heavier, can only achieve much lower maximum velocities, as indicated by the central band of the diagram. To ensure that the frequency components are effectively attenuated, the damping coefficient in the shock absorber must be carefully chosen.

The control of the sprung mass under resonant condition will normally require a high damping coefficient. However, if this same level of damping is applied across the whole range of velocities, the unsprung mass will be more controlled than is necessary and uncomfortable jolting at the sprung mass will be experienced\(^{[51]}\). To correct for this, the damping coefficient is shown to reduce progressively as the velocity increases, thus providing the typical characteristic shape of a steep slope at low velocity progressing to a shallow one at high velocity. This also explains why the mechanical valve is constructed with a bleed orifice and a blow-off valve in the early part of this description. The idea is to produce a changeable damping characteristic so that the two different frequency range components can be individually isolated.

The suspension system described above is classified as a passive type suspension in which all the power generated from the vehicle movement is dissipated as heat\(^{[52]}\). The developed damping characteristic may be proven to be effective under one particular operating condition. There are however many other factors in addition to the road irregularities that need to be considered. For example, the change in vehicle load, the accelerating and braking effects and the effect of changing roll and pitch angles due to
turning[53]. The improvement in riding comfort is thus hampered by the lack of flexibility to cope with such a wide range of operating conditions. In addition to all these external factors, the non-linear and hysteresis behaviours occurring within the shock absorber must also not be overlooked. These latter effects, studied by Hall and Gill[54], could make the prediction of the ride and handling even more difficult.

Figure 7.4 shows the arrangement of an active suspension system operated by a servo-actuator in a position control loop. The great advantage of the active systems is their capacity to adapt to many different conditions under which they are operating[52]. However, the active systems are seldom used in road vehicles, mainly because of the high cost and power requirements involved in the additional equipment required. A compromise between cost and riding comfort was found to be possible following the introduction of a semi-active suspension system[55].

Such a semi-active system, as illustrated in Figure 7.5, has essentially the same sensor arrangement as the active system except that the servo-valve and the actuator are replaced by a rapidly adjustable damper. The pressure drop across the damper chambers is adjusted by responding to the velocity change at the suspension unit. Studies of the semi-active systems have shown that the riding performance can be made as good as the active system provided that a proper control of the damping characteristic is adopted[56,57]. What is then needed in the semi-active system is a mechanism which will enable the damper coefficient to change rapidly. This mechanism may be realised as a variable orifice fitted into the shock absorber. Figure 7.6 shows a modified view of the force-velocity characteristic. Two curves are shown to represent the upper and lower limits of the damping characteristic. The function of the variable orifice is to change the damping characteristic from one limit to another through
two or more steps or continuously.

In the next section, the requirements for a double disc valve are described. This double disc valve is designed to be a variable orifice by using a similar operating principle to a single disc valve. The pressure-flow characteristic is varied electrically via an electro-magnetic coil and is aimed at satisfying the requirement for adjustable damping in a vehicle shock absorber. The integration of the double disc valve with a modified conventional shock absorber is also presented.

### 7.2 A Disc Valve Application in a Vehicle Suspension System

The design and development of a double disc valve is aimed at producing a low-cost device which will function as an electrically modulated variable orifice suitable for use in a semi-active suspension system. It has been demonstrated in Section I that a single disc valve has the characteristic that the flow resistance can be varied continuously by altering the disc position. Therefore, by using the same principle, a similar disc valve is considered, which is designed to provide a pressure-flow characteristic to satisfy the force-velocity relationship required by a vehicle shock absorber.

The disc valve to be developed has to satisfy four basic requirements:

a) to satisfy the upper and lower force-velocity relationships,

b) to be capable of modifying electrically the force-velocity relationship continuously between the two limits,

c) to be fail-safe and
d) to be fast reacting.

The required force-velocity relationship is given in Table 1 below. The data is basically divided into two groups, representing the upper and lower limits of the force-velocity relationship. Those flow rates and pressures shown in the table form the design criteria for the new valve.

<table>
<thead>
<tr>
<th>Piston velocity (m/s)</th>
<th>Load (N)</th>
<th>Flow rate (l/min)</th>
<th>Pressure (bar)</th>
<th>Load (N)</th>
<th>Flow rate (l/min)</th>
<th>Pressure (bar)</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Lower Limit</strong></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>0.05</td>
<td>50.0</td>
<td>0.93</td>
<td>1.6</td>
<td>54.0</td>
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<td>155.8</td>
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<td>17.83</td>
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Table 1 - Specified Force-Velocity Relationship

The pressure and the flow rate given in Table 1 are calculated at a piston diameter of 25.4mm and a rod diameter of 15.9mm. This size of shock
absorber is particularly chosen for simplifying the design of the disc valve. When plotted out in Figure 7.7, it is shown that the pressure-flow characteristics between the compression and the rebound cycles are very similar to each other. As a result of this, if the construction of the conventional absorber is modified, only one valve is needed to control the movement of the shock absorber in both directions.

The modification required to a conventional shock absorber is shown schematically in Figure 7.8. The oil is restricted to flow through the damping valve in one direction, although the shock absorber motion may change from compression to rebound or vice versa. The required damping characteristic is developed regardless of which direction the absorber is moving. Of the three orifices connected in parallel, two of them are derived from a double disc valve, the construction of which will be described in a later chapter. The third orifice is a mechanical valve used in conventional shock absorbers, the function of which is to provide a pressure-flow characteristic which will satisfy the upper limit of the force-velocity requirements.

The reason for using a mechanical valve in parallel with the disc valve is to simplify the disc valve construction. When the double disc valve is not energised, the pressure drop across the shock absorber is generated mainly across the mechanical valve. If a high stiffness, spring is chosen for the mechanical valve, a pressure-flow characteristic similar to the one specified for the upper force limit can be produced. As the disc valve is energised, part of the flow passes through the mechanical valve and part through the disc valve and the pressure drop across the shock absorber is consequently reduced. The pressure drop is reduced further until the disc valve is fully opened. If the minimum pressure drop across the valve assembly is lower than or equal to the lower force limit, then by
controlling the disc position in the disc valve, a change of pressure drop from the upper to the lower limit can be made.

The second important requirement which the disc valve has to satisfy is to be fail-safe. The valve has to be made in such a way that should there be an electric failure, the shock absorber reverts to the upper force limit condition. This produces a hard, but acceptable, ride condition for the car.

The function of the control valve is to adjust the damping characteristic of the shock absorber by responding to an input signal, which can be the differential pressure across the shock absorber, the piston velocity etc. The response of the valve must therefore be well above the maximum natural frequency of the vehicle and a value of 50 Hz is thought to be normally sufficient for shock absorber applications[53]. Previous work on disc valves indicates that their dynamic performance is sufficient for this purpose.

From the criteria discussed above, the single disc valve is not entirely suitable for this application. To meet the fail-safe condition, the wavy washer stiffness and pre-load would have to be high and it is doubtful if the coil would provide enough power to move the disc away from the supply nozzle. Even if it did, the range of operating pressure drops would be very limited. Accordingly, a new design has been developed using two discs with position feedback and this will be described in the next chapter.
A Vehicle Suspension System Application

1 - Outer tube
2 - Compression chamber
3 - Piston head
4 - Rebound chamber
5 - Foot valve
6 - Inner tube
7 - Non-return valve
8 - Piston valve
9 - Non-return valve

Figure 7.1 A Conventional Shock Absorber

Figure 7.2 Force - Velocity Characteristic
Figure 7.3 Passive Suspension System

Figure 7.4 Active Suspension System
Figure 7.5 Semi-Active Suspension System

Figure 7.6 Variable Force-Velocity Characteristic
Figure 7.7 Specified Pressure-Flow Characteristic

Figure 7.8 Modified Shock Absorber with Control Valves connected
Chapter 8

Design and Construction of the Double Disc Valve
8. **Design and Construction of the Double Disc Valve**

8.1 **Operating Principle**

The operating principle of the double disc valve is illustrated in Figure 8.1. The main function of the valve is to change the damping characteristic of a shock absorber from the upper force limit to the lower limit. To satisfy the upper force limit, a high flow resistance is required. This implies that the disc valve may be designed to nearly or completely close off flow, allowing the pressure drop to be generated essentially by flow through the mechanical valve when the electrical signal is zero. However, to balance the fluid flow force within the double disc valve, it has been found advantageous to allow a small bleed flow through the disc valve. This can be allowed for by a suitable spring setting in the mechanical valve.

Before any current is applied, disc 1(1) on the left-hand side of Figure 8.1a is pushed to the right by the presence of a higher hydraulic force on the left-hand disc surface. A small gap is shown between the right-hand surface of disc 1 and nozzle 1(2). This small gap allows a small bleed flow across the right-hand disc surface and has an effect of lowering the fluid pressure on this surface. A precise control of the gap is needed because any bleed flow across it considerably affects the pressure drop generated across the mechanical valve as discussed above.

The holding force keeping disc 1 to the right is also transmitted to the right-hand disc 2(3) via a push rod(4) located at the centre of the nozzle body(5). As nozzle 2(6) is closed firmly by disc 2 due to the force transmitted, oil can only flow through the valve via the small gap at disc 1. The function of disc 1 may be seen as a fail-safe mechanism in
which a closing fluid force is developed by the bleed flow to keep nozzle 2 in a closed condition. The other function that disc 1 performs is to balance the hydraulic forces acting on disc 2. A lower current can then be used to modulate the disc position in its operating range.

When current is applied to electro-magnetic coil(7), it produces an attraction force which pulls disc 1 to the left. Since the net force keeping disc 2 at nozzle 2 has been reduced, disc 2 moves away from nozzle 2. As nozzle 2 is opened, the main flow of oil occurs across the right-hand surface of disc 2 and increases in magnitude as the gap between disc 2 and nozzle 2 increases. The maximum oil flow occurs when the disc assembly reaches the end of the travel. The two variable orifices which are joined together to form a single output port as shown in Figure 7.8 thus represent the bleed and main flow conditions through nozzle 1 and 2 respectively.

A small gap, $x_{gap}$, is shown between disc 1 and the coil face in Figure 8.1b as the disc reaches the end of the travel. This small gap is used to generate a hydrostatic force on the left-hand surface of disc 1, which can help to push the disc away from the coil after the current is withdrawn. It also serves to limit the maximum electro-magnetic force that can be generated and the magnetic hysteresis.

The double disc valve is designed in such a way that the shock absorber damping characteristic is adjusted by controlling the disc between the two limiting positions. However, using the disc position as the controlling signal to vary the valve characteristic is not the only strategy that can be adopted to produce the desired damping characteristic for the vehicle shock absorber. Another possibility is to use the differential pressure across the damper piston to measure the damping force directly. These two
types of control methods will be tested individually although position control forms the main theme of the study as it also provides valuable insight into the valve operation by monitoring the disc position.

8.2 Design Calculations

To calculate the sizes of nozzle 1 and 2, the following orifice equation is used. This equation is the same as equation (3.1) used in the single disc valve and is given here mainly for calculation purpose.

\[
Q = C_d \cdot \pi \cdot d_n \cdot x_d \sqrt{\frac{2 \cdot (P_1 - P_2)}{\rho}}
\]  

(8.1)

where \(x_d\) is the disc position from the nozzle and \(P_1\) and \(P_2\) are the pressures at the nozzle and at the edge of the disc respectively. Flow is outward when \(P_1 > P_2\) and inward when \(P_1 < P_2\).

In Chapter 7, it is shown that when the shock absorber is at the upper force limit, which occurs when nozzle 2 in the double disc valve is closed, the required pressure drop at maximum flow rate of 27.78 l/min is 155.8 bar. This pressure drop is reduced to 38.9 bar when the shock absorber is changed to the lower force limit or when nozzle 2 is fully opened. Because this pressure-flow data represents the two limiting conditions, it is used as the design criteria for calculating the nozzle sizes.

Before any calculation is made, the unknown discharge coefficient is estimated. As the orifice geometries of the single and double disc valves are quite similar to each other, the discharge coefficients obtained from
the single disc valve experiments are used. In Chapter 6, two empirical formulas are developed and the discharge coefficients are expressed in terms of the Reynolds number and disc position. To calculate the nozzle sizes, these empirical formulas together with equation (8.1) are used.

When the diameter of nozzle 1 is calculated, the pressure-flow data at the upper force limit is used and because nozzle 2 is closed under this condition, it is assumed that oil flows to the mechanical valve and nozzle 1 of the double disc valve only. Since the two valves are connected in parallel, the following equations with reference to Figure 8.2 apply:

\[ x = 0, \quad Q_2 = 0. \]

\[ Q_{in} = Q_{mech} + Q_1 \quad (8.2) \]

\[ P_s - P_d = M_f \cdot h(Q_{mech}) \quad (8.3) \]

\[ P_s - P_a = \frac{\rho (\frac{Q_1}{2 C_{d1} \cdot \pi \cdot dn_1 \cdot x_1})^2}{P_s - P_a} = \frac{\rho (\frac{Q_1}{2 C_{d1} \cdot \pi \cdot dn_1 \cdot x_1})^2}{P_s - P_a} \quad (8.4) \]

\[ P_a - P_b = \frac{96 \cdot \mu \cdot ln_1 \cdot Q_1}{\pi \cdot dn_1 \cdot (dn_1 - d_f)^2} \quad (8.5) \]

\[ P_b - P_d = \frac{\rho (\frac{Q_1}{2 C_{d0} \cdot \pi \cdot dn_1 \cdot x_2})^2}{P_b - P_d} \quad (8.6) \]

where \( Q_1 \) and \( Q_{mech} \) are the flows in nozzle 1 and the mechanical valve respectively,

\( M_f \) is the empirical constant for flow through the mechanical valve,

\( h(Q_{mech}) \) is the curved-fitted function of the mechanical valve pressure-flow characteristic which is the same as curve A shown in Figure 7.7 and
$C_{di}$ and $C_{do}$ are the discharge coefficients for the inward and outward flows respectively.

Combining equations (8.4), (8.5) and (8.6),

$$P_s - P_d = \frac{\rho}{2} \left( \frac{Q_1}{C_{di} \cdot \pi \cdot d_{n1} \cdot x_1} \right)^2 + \frac{96 \cdot \mu \cdot L_{n1} \cdot Q_1}{\pi \cdot d_{n1} \cdot (d_{n1} - d_\mu)^3}$$

$$+ \frac{\rho}{2} \left( \frac{Q_1}{C_{do} \cdot \pi \cdot d_{n1} \cdot x_2} \right)^2$$

Putting $x_1 = x_{gap1}$ and $x_2 = x_{gap2} + TD$ and applying equations (6.1) and (6.2), the discharge coefficients become

$$C_{di} = F_1 \{ x_1/TD, RE_1 \}$$

$$= (a \cdot (x_{gap1}/TD) + b \cdot (x_{gap1}/TD)^2 + c) \cdot g(\text{RE}_1)$$

(8.8)

$$C_{do} = F_2 \{ x_2/TD, RE_1 \}$$

$$= (a \cdot (1 + x_{gap2}/TD) + b \cdot (1 + x_{gap2}/TD)^2 + c) \cdot f(\text{RE}_1)$$

$$= (a + b + c) \cdot f(\text{RE}_1)$$

(8.9)

where $x_{gap2}$ is defined as the minimum gap between the left-hand surface of disc 2 and nozzle 1 when nozzle 2 is fully opened. $x_{gap2}$ must be made greater than or at least equal to $(x_{gap1} + TD)$ to ensure that the flow at nozzle 1 is not reduced. However, because of the parabolic function used, it is assumed that $x/TD$ has a limiting value of 1.0, i.e. the discharge coefficient for $x/TD > 1$ is the same as when $x/TD = 1$.

The Reynolds number is defined as

$$\text{RE}_1 = \frac{\rho \cdot Q_1}{\mu \cdot \pi \cdot d_{n1}}$$

(8.10)
If $M_f$ in equation (8.3) is 1.2, i.e. by assuming that the pressure drop at the mechanical valve is 20% higher, then by trial-and-error the flow at nozzle 1 can be estimated. This estimated flow is substituted into equations (8.7) to (8.10) to calculate the diameter of nozzle 1. If $x_{gap1}$ is 0.05 mm and the diameter of the push rod is 2.5 mm and $L_{n1} = 12.0$ mm, the nozzle 1 diameter can be found to be approximately 3.2 mm. The disc travel is limited to 0.4 mm to prevent the disc from being too far away from the magnetic coil. $Q_{in}$ and $(P_s - P_d)$ are used as 27.78 l/min and 155.8 bar respectively under this condition.

When the diameter of nozzle 2 is calculated, the pressure-flow data at the lower force limit is used and some of the equations above are modified. This is to include the flow at nozzle 2 when it is at the full open condition.

\[ Q_{in} = Q_{mech} + Q_1 + Q_2 \]  
\[ (8.11) \]

\[ P_{n2} - P_d = \frac{\rho Q_2}{2 C_{do2} \pi d_{n2} x} \]  
\[ (8.12) \]

where $P_{n2} = P_s = \frac{8 \mu L_{n2} Q_2}{\pi R_{n2}^4}$

\[ C_{do2} = \mathcal{F}(x/TD, \text{RE}_2) \]
\[ = (a \cdot (x/TD) + b \cdot (x/TD)^2 + c) \cdot \mathcal{F}(\text{RE}_2) \]  
\[ (8.13) \]

Because nozzle 2 is fully opened, $x$ is equal to $TD$. Therefore,

\[ P_s - P_d = \frac{\rho Q_2}{2 C_{do2} \pi d_{n2} TD} + \frac{8 \mu L_{n2} Q_2}{\pi R_{n2}^4} \]  
\[ (8.14) \]

\[ C_{do2} = (a + b + c) \cdot \mathcal{F}(\text{RE}_2) \]  
\[ (8.15) \]
Design and Construction of the Double Disc Valve

\[ \text{RE}_2 = \frac{\rho Q_2}{\mu \pi d n_2} \]  
\[ \text{(8.16)} \]

\[ C_{d1} = F_1 \left( \frac{x_1}{TD} \ , \ \text{RE}_1 \right) \]
\[ = (a + b + c)g\left( \frac{(TD + x_{gap1})}{TD} \right) \]
\[ = (a + b + c)g\left( \text{RE}_1 \right) \]
\[ \text{(8.17)} \]

\[ C_{do1} = F_2 \left( \frac{x_2}{TD} \ , \ \text{RE}_1 \right) \]
\[ = (a + b + c)f\left( \frac{x_{gap2}}{TD} \right) \]
\[ = (a + b + c)f\left( \text{RE}_1 \right) \quad \text{for} \ x_{gap2} \geq x_{gap1} + TD. \]
\[ \text{(8.18)} \]

With \( Q_{in} = 27.78 \, \text{l/min} \) and \( (P_s - P_d) = 38.6 \, \text{bar} \) and by using equations \( (8.3) \), \( (8.7) \), \( (8.10) \), \( (8.11) \) and \( (8.14) \) to \( (8.18) \), the diameter of nozzle 2 is found to be approximately 3.4 mm.

In this section, it is shown that for a given orifice gap, \( x_{gap1} \), the diameters of nozzle 1 and 2 can be found by referring to the pressure-flow data at the lower and upper force limits. These calculated nozzle diameters will be modified when a different orifice gap is used. However, the orifice gap is correctly chosen only when the requirement for a positive right-hand force is also satisfied. It is shown in Figure 8.1 that the discs are pulled to the left by a magnetic coil. A positive right-hand force is therefore required to act against the magnetic force so that when the magnetic force is removed, the discs are pushed back to close off nozzle 2.

In the next section, the construction of a double disc valve is described and then an estimation of the double disc valve pressure-flow characteristics is made. By calculating the net hydraulic force acting on the discs, it is found that a positive right-hand force can be generated when the calculated nozzle diameters and the 0.05 mm orifice gap are used.
8.3 Construction of the Valve

A general assembly drawing of the double disc valve is shown in Figure 8.3. Figure 8.4 shows the valve's components in the unassembled condition and a half-assembled double disc valve is shown in Figure 8.5. Based on the calculations made in the previous section, the followings are some of the valve's components dimensions referred to in Figures 8.1 and 8.3:

1. **Disc 1**: diameter = 25.0 mm  
   thickness = 3.0 mm
2. **Nozzle 1**: inner diameter = 3.2 mm  
   outer diameter = 3.7 mm  
   length = 12.0 mm
3. **Disc 2**: diameter = 25.0 mm  
   thickness = 3.0 mm
4. **Push rod**: diameter = 2.5 mm  
   length = 12.85 mm
5. **Nozzle 2**: inner diameter = 3.4 mm  
   outer diameter = 3.9 mm  
   length = 12.5 mm
6. Orifice holding gap = 0.5 mm

\[ x_{\text{gap1}} = 0.4 \text{ mm}, \quad x_{\text{gap1}} = 0.05 \text{ mm}, \quad x_{\text{gap2}} = 0.45 \text{ mm} \quad \text{and} \quad TD = 0.4 \text{ mm}. \]

In addition to the valve components described in section 8.1, it is shown in Figure 8.3 that to the left of disc 1, a coil spring(9) is used to increase the net right-hand force. The amount of force increase can be adjusted directly by increasing the spring pre-compression. In order to control the position of the disc assembly, a non-contacting inductive eddy-current position transducer(10) is used and is placed at the right-hand
side of the valve to measure the position of disc 2 as it moves away from nozzle 2. Since the net force produced at any one side of the valve is transmitted to the other side via the push rod, any movement of disc 2 is normally the same as the movement of disc 1. The function of the access screw(11) is to enable the calibration of the disc travel to be made by using a depth micrometer.

Appendix A7 shows the specification of the position transducer, which in general is better than the capacitive type as it is not so temperature sensitive[59]. Also it is found to be easier to integrate into the valve without causing any substantial increase in the overall valve dimensions. It will be seen in a later chapter that because of the non-linear characteristic generated by the magnetic coil, the position transducer is necessary to achieve stable position control in the double disc valve.

The material of the valve is mainly aluminium. The valve construction is similar to a single disc valve in which the magnetic coil and the position transducer are also attached to two end bodies, which are then combined with the centre body to form a cartridge design.

8.4 The Mechanical Valve

The mechanical valve to be used in the present application can either be a foot valve or a piston valve supplied by the Armstrong Patents Co. Ltd. as shown in Figure 8.6, depending on which pressure-flow characteristic is more desirable. This foot or piston valve is normally used in a 25.4 mm diameter inner tube shock absorbers. Figure 8.7 shows the construction of these valves and their functions in an Armstrong shock absorber. The piston
valve is used to restrict the oil flow from the rebound chamber into the compression chamber as the shock absorber is moving upward and the foot valve is used to control the excess oil passing from the compression chamber into the reservoir as the piston is moving downward. The pressure-flow characteristics of the valves have already been described by curves A and B in Figure 7.7 respectively.

To test the prototype shock absorber, a valve body is required to hold the mechanical valve and mount on the side of a conventional shock absorber to provide uni-directional flow irrespective of the piston direction. Also, it is found to be more convenient for analysis purpose if the pressure-flow characteristic of the mechanical valve can be identified independently before it is tested with the double disc valve and the shock absorber. Figure 8.8 shows the sectional view of a mechanical valve body designed. The port configuration is also a CETOP configuration, thus allowing the mechanical valve to be tested with the same hydraulic set-up as the double disc valve. The internal dimension of the valve body has been made in such a way that either a foot valve or a piston valve can be installed. Figure 8.9 shows the mechanical valve in the unassembled condition.

8.5 The Construction of a Valve Mounting Block

It is shown in Figure 7.8 that the uni-directional oil flow from the rebound chamber is passed to the double disc valve and the mechanical valve before returning to the reservoir. A connection between the shock absorber and the valve assembly is therefore needed. Figure 8.10 shows the use of a mounting block connecting to a shock absorber via two 12 mm diameter pipes.
Both the mechanical valve and the double disc valve are mounted on the mounting block and as the shock absorber operates, the oil from the rebound chamber flows through the valves before it goes to the reservoir. Two transducer ports are provided to enable the upstream and downstream pressures to be measured and can also be used as a study of the shock absorber damping force generated.

The material of the block is mild steel and the manifold surface is ground to provide a good seal for the valves. The extended pipes are welded to the block which is yoke mounted to the shock absorber. As the prototype shock absorber was intended to provide functional study of the valves, no attempt was made in this design to minimise the mounting block or valves.
Figure 8.1 Simplified Construction of a Double Disc Valve
Figure 8.2 Simplified Internal Layout of a Double Disc Valve
Figure 8.3 General Assembly of a Double Disc Valve
Figure 8.4 Unassembled Double Disc Valve

Figure 8.5 Half-Assembled Double Disc Valve
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Figure 8.6 Mechanical Valve Component

1 - Piston Valve Coil Spring
2 - Foot Valve Coil Spring
3 - Foot Valve Orifice
4, 6 - Non-Return Valve
5 - Piston Orifice

Figure 8.7 Sectional View of an Armstrong Shock Absorber

- OIL FLOW
- REBOUND BLOW OFF
- REBOUND LEAK
- COMPRESSION BLOW OFF
- COMPRESSION LEAK
Design and Construction of the Double Disc Valve

1 - Foot or Piston Valve
2 - Valve Body
3 - End Cap
4 - 'O' Ring

Figure 8.8 General Assembly of a Mechanical Valve

Figure 8.9 Unassembled Mechanical Valve
Figure 8.10 Connection of Valve Mounting Block to Shock Absorber
Chapter 9

Theoretical Study of a Double Disc Valve
9. Theoretical Study of a Double Disc Valve

In this chapter, a simulation based on the theoretical ideas used for the single disc valve is developed to predict the steady-state pressure-flow characteristics in the shock absorber as the disc position is varied over its operating range. The steady-state hydraulic force is also calculated to check if the size of the magnetic coil is correctly chosen. Linearisation analysis is then followed and through which the type of controller to be required in order to achieve stable position control is defined.

9.1 Approximated Pressure-Flow Characteristics

To simulate the valve characteristics, a steady-state fluid flow analysis software program was developed. Figure 9.1 shows the flow diagram of the program written in which it is assumed that a mechanical valve is connected in parallel externally and has a pressure-flow characteristic slightly higher than the specified upper limit. The reason for this higher characteristic is to compensate for the bleed flow across nozzle 1 when the magnetic coil is not energised. Thus, when nozzle 2 is not opened, the pressure-flow characteristic of the combined disc valve and mechanical valve should be the same as the upper limit given in Figure 7.7.

As the flow diagram illustrates, the pressure drop across the disc valve is predicted by assuming that a small percentage of the total flow, which is entered as the input variable, passes through the mechanical valve. If the calculated pressure drop due to this small percentage of flow is less than the one found across the disc valve, which will be passing the rest of the total flow, then a higher percentage flow is assumed at the mechanical valve. The trial-and-error solution is continued until the pressure drops
across both the disc valve and the mechanical valve are approximately equal.

Figure 9.2 shows the estimated flow rates across the disc valve and the mechanical valve. The total input flow rate in this example calculation is 27.78 l/min. The flow across the disc valve is shown to increase progressively as nozzle 2 is gradually opened, while the flow across the mechanical valve is diminishing. In Figure 9.3, the pressure-flow relationship is given for different disc positions. When nozzle 2 is closed, this corresponds to the zero disc position and the combined pressure-flow characteristic is found to be quite close to the specified upper limit as is shown by the comparison made in Figure 9.4. As the disc assembly moves towards the magnetic coil, the pressure drop across the disc valve is reduced and becomes a minimum when it reaches the end of the travel. Although the estimated minimum flow resistance is shown to be slightly lower than the specified lower limits, particularly during the rebound cycle, this condition could be remedied by moving back the disc assembly using the magnetic force to increase the flow resistance.

The pressure-flow characteristics described above were calculated by using the following dimensions:

Nozzle diameters: \( d_{n1} = 3.2 \text{ mm}, \quad d_{n2} = 3.4 \text{ mm}. \)

Orifice gaps: \( x_{gap1} = 0.05 \text{ mm}, \quad x_{gap2} = 0.45 \text{ mm}. \)

Disc travel: \( TD = 0.4 \text{ mm}. \)

Rod size: \( \text{diameter} = 2.5 \text{ mm}. \)

The same discharge coefficients as described by equations (8.8), (8.9) and (8.13) were used.
9.2 Steady-State Hydraulic Force

The estimate of the hydraulic force is made to ensure that the disc position is varied by the magnetic force generated. Figure 9.5 shows the forces acting on disc 1, disc 2 and the push rod. The function of the magnetic coil is to pull the discs to the left. This requires a right-hand force to push the disc assembly back to the right when the magnetic attraction is removed. In other words, the requirement for the net force, $F_{\text{net}}$, shown in Figure 9.5, must be positive at any disc position. The following is a list of force equations used to calculate the net hydraulic force in the simulation program. The form of the equations is identical to those for the single disc valve although the resultant forces acting on the disc assembly are different.

Disc 1:

Force acting on the left-hand surface

$$F_{1\text{sl}} = p_s \pi R_0^2$$  \hspace{1cm} (9.1)

$$F_{\text{sprg}} = K_s (x_s + x)$$  \hspace{1cm} (9.2)

The spring force given in equation (9.2) is derived from a coil spring located at the left-hand side of disc 1 as shown in Figure 8.3. $x_s$ is the spring pre-compression.

Force acting on the right-hand surface

$$F_{1\text{sr}} = p_a A_a$$  \hspace{1cm} (9.3)

$$F_{1\text{mr}} = \frac{\rho \cdot Q^2}{A_a}$$  \hspace{1cm} (9.4)
Theoretical Study of a Double Disc Valve

$F_{1fr} = P_s \cdot \pi (R_o^2 - R_n^2) - K_1 \cdot Q / h_{22}^3 + K_2 \cdot Q^2 / h_{22}^2$

where $h_{22} = x_{gap1} + HG + x$,

\[ K_1 = 3.\mu\cdot(R_o^2 - R_n^2) \cdot (2\ln(R_o/R_n) + 1) \] and

\[ K_2 = \frac{27.\rho \cdot (R_o^2 - R_n^2)}{70.\pi} \cdot \left( \ln \left( \frac{R_o}{R_n} \right) - 1 \right) \] 

(9.5)

Total right-hand force on disc 1:

$F_{1n} = F_{1sl} + F_{sprg} - F_{1sr} - F_{1mr} - F_{1fr}$

\[ = P_s \cdot A_n + F_{sprg} - P_a \cdot A_a - \rho \cdot Q^2 / A_a + K_1 \cdot Q / h_{22}^3 - K_2 \cdot Q^2 / h_{22}^2 \] 

(9.6)

Disc 2:

Force acting on the left-hand surface

$F_{2sl} = P_b \cdot A_a$ 

(9.7)

$F_{2ml} = \frac{\rho \cdot Q^2}{A_a}$

(9.8)

$F_{2f1} = P_d \cdot \pi (R_o^2 - R_n^2) + K_1 \cdot Q / h_{33}^3 + K_2 \cdot Q^2 / h_{33}^2$

(9.9)

where $h_{33} = x_{gap2} + HG - x$ and

$K_1$ and $K_2$ are defined the same as equation (9.5) above.

Force acting on the right-hand surface

$F_{2sr} = P_n_2 \cdot A_n_2$

(9.10)

where

\[ P_n_2 = P_s - \frac{8.\mu \cdot Ln_2 \cdot Q_2}{\pi \cdot Rn_4} \]

$F_{2mr} = \frac{\rho \cdot Q^2}{A_n_2}$

(9.11)
Theoretical Study of a Double Disc Valve

\[ F_{2fr} = P_d \pi \left( R_o^2 - R_n^2 \right) + K_1' Q/h_{44}^3 + K_2' Q^2/h_{44}^2 \]

where \( h_{44} = \Delta G + x \),

\[ K_1' = 3 \mu (R_o^2 - R_n^2 \cdot (2 \ln(R_o/R_n) + 1)) \]

and

\[ K_2' = \frac{27 \rho^2}{70 \pi} \left( \frac{R_o^2 - R_n^2}{2 R_o^2} \right) - \ln \left( \frac{R_o}{R_n} \right) \]  

(9.12)

Total left-hand force at disc 2:

\[ F_{2n} = F_{2sr} + F_{2mr} + F_{2fr} - F_{2sl} - F_{2ml} - F_{2f1} \]

\[ = P_n^2 A_n^2 + \frac{Q^2}{A_n^2} + K_1' Q/h_{44}^3 + K_2' Q^2/h_{44}^2 - P_b A_a - \frac{Q^2}{A_a} - K_1' Q/h_{33}^3 - K_2' Q^2/h_{33}^2 - P_d \pi (R_n^2 - R_1^2) \]  

(9.13)

The net hydraulic force pushing the disc assembly to the right is therefore given as

\[ F_{net} = F_{1n} - F_{2n} \]  

(9.14)

Figure 9.6 shows the results of the net hydraulic force calculated. The pressure drops used were 20, 50 and 100 bar. In the shock absorber application, the maximum hydraulic force is estimated as curve A shown in the figure, which is found by using the maximum flow rate of 27.78 l/min across the parallel valve assembly at maximum shock absorber velocity. For larger flow rates or pressure drops, a bigger magnetic coil will be required as the hydraulic force will become too strong for the magnetic force to handle. It is shown that as a result of using disc 1, the bleed flow fluid forces generated have effectively cancelled the forces generated at disc 2 and the net hydraulic force acting on the disc assembly is found to be rather small. This also implies that a low current is sufficient to open nozzle 2. However, the increase of magnetic force as the disc assembly moves closer to the coil creates a problem in controlling the disc.
position. Unless the slope of the combined hydraulic and spring force is steeper than that of the magnetic force in the position control region, the disc assembly is switched rapidly to the end of the travel with a slight increase of current in the magnetic coil. In the next section, the control of disc position is described in more detail. It is shown that the position control problem can be eliminated by adopting closed-loop operation.

9.3. Double Disc Valve Control Analysis

It has been shown in the case of the single disc valve that the stability of the valve can be studied effectively by the linearization method. A similar analysis is therefore applied on the double disc valve, although it will be seen later that the control condition are somewhat different. The linearized model of the double disc valve is shown in Figure 9.7. It has the same elements as the one shown for the single disc valve except that the hydraulic part of the valve has been ignored. The hydraulic force as studied in the last chapter is found to be quite small as compared with the other forces and is therefore not considered in the analysis for reasons of simplicity. The open-loop transfer function for the double disc valve is therefore

\[
G(s) = \frac{\Delta x(s)}{\Delta v_a(s)} = \frac{K_a K_m}{m s^2 + B s + (K_s - K_x)} \cdot (T_c s + 1)
\]  

(9.15)

It is shown in equation (9.15) that the double disc valve will only be stable if the condition \(K_s > K_x\) is satisfied. However, \(K_x\) is a variable quantity, which will increase in magnitude as the disc gets closer to the
magnetic coil. Therefore, if the double disc valve is to be stable in open-loop, the spring stiffness $K_s$ must be larger than the worst case magnetic stiffness. Increasing the spring stiffness is unfortunately not a desirable solution. The coil current must also increase at large disc displacement in order to overcome the increased spring force. This increase of coil current causes further problems such as the increase of temperature in the coil and the need for a higher voltage supply.

While stable operation of the disc does not seem to be possible in open-loop, it can be stabilised by closing the loop with position feedback. The block diagram of the closed-loop control system is shown in Figure 9.8, which is similar to the one developed for the single disc valve. The error signal is passed to a PI controller, where a driving voltage for the coil amplifier is generated. The modified open-loop transfer function is

$$G(s) = \frac{K_i K_a K_m (s/K_{pi} + 1)}{s (T_c s + 1) (m s^2 + B s + (K_s - K_x))} \tag{9.16}$$

where $K_{pi} = K_p/K_i$.

If the frequency response of the position transducer is well above the operating range of valve frequency considered, then for simplicity

$$H(s) = K_f \tag{9.17}$$

Combining equations (9.16) and (9.17),

$$G(s)H(s) = \frac{K_i K_a K_m K_f (K_{pi} s + 1)}{s (T_c s + 1) (m s^2 + B s + (K_s - K_x))} \tag{9.18}$$
The closed-loop transfer function therefore becomes

\[
\begin{align*}
\frac{C(s)}{R(s)} &= \frac{G(s)}{1 + G(s)H(s)} = \frac{K_{op}(K_{pi}s + 1)}{s(T_0s + 1)(m_0s^2 + B_0s + (K_s - K_x)) + K_{op}(K_{pi}s + 1)}
\end{align*}
\]

where \(K_{op} = K_i K_a K_m\) and
\(K_{op} = K_{op} K_{f} \)  \( (9.19) \)

Expanding the denominator of equation (9.19), the characteristic equation of the closed-loop system is

\[
\begin{align*}
m_T c s^4 + (m + B_T c)s^3 + (B + (K_s - K_x)T_c)s^2 + ((K_s - K_x) + K_{op} K_{pi})s + K_{op} &= 0 \\
(9.20)
\end{align*}
\]

Applying Routh's stability criterion,

\[
\begin{align*}
\begin{array}{cccc}
s^4 & m_T c & (B + (K_s - K_x)T_c) & K_{op} \\
s^3 & (m + B_T c) & ((K_s - K_x) + K_{op} K_{pi}) \\
s^2 & B + \frac{B(K_s - K_x)T_c - m_T c K_{op} K_{pi}}{m + B_T c} & K_{op} \\
s^1 & ((K_s - K_x) + K_{op} K_{pi}) - \frac{K_{op} (m + B_T c)^2}{B(m + B_T c) + B(K_s - K_x)T_c^2 - m_T c K_{op} K_{pi}} \\
s^0 & K_{op}
\end{array}
\end{align*}
\]

The damping coefficient \(B\) used in the equations above may be estimated by the same method used in the single disc valve, i.e. by assuming that the
viscosity term has the most dominant effect. Since two more disc surfaces are involved in the double disc valve, the damping coefficient as referred to Figure 9.9 is derived as

\[
B = 3\mu \left( \frac{C_1}{h_{11}} + \frac{C_2}{h_{22}} + \frac{C_3}{h_{33}} + \frac{C_4}{h_{44}} \right)
\]

where 
\[
C_1 = \pi R_0^4 \quad \text{and} \quad C_2 = \pi (R_0^2 - R_1^2)^2 \nonumber \\
C_3 = \pi (R_0^2 - R_2^2)^2.
\] (9.21)

Figure 9.10 shows the variation of the damping coefficient calculated with equation (9.21).

If the following data is used for the double disc valve,

\[
m = 20 \, \text{g}, \quad K_a = 12.0 \, \text{N/mm}, \quad K_x = 150 \, \text{N/mm}, \\
K_a = 0.6 \, \text{A/V}, \quad K_m = 50 \, \text{N/A}, \quad T_c = 0.333 \, \text{msec}, \\
K_f = 8 \, \text{V/mm}(\text{Appendix A7}), \quad B = 40.0 \, \text{N.sec/m}, \quad K_{pi} = 5 \, \text{msec},
\]

then by substituting into the Routh's matrix, the result is

\[
s^4 \quad 6.67 \times 10^{-6} - 6.0 \quad 240 \times 10^3 K_i
\]
\[
s^3 \quad 33.3 \times 10^{-3} (-138 \times 10^3 + 240 \times 10^3 K_p)
\]
\[
s^2 \quad (21.6 - 48 K_p) \quad 240 \times 10^3 K_i
\]
\[
s^1 \quad (-138 \times 10^3 + 240 \times 10^3 K_p - \frac{266 K_i}{0.72 - 1.6 K_p})
\]
\[
s^0 \quad 240 \times 10^3 K_i
\]
For the system to be stable, all coefficients in the first column of the Routh's matrix must be positive[60]. This leads to the following two limiting conditions:

\[ 21.6 - 48K_p > 0 \quad \text{and} \quad \]
\[ -138 \times 10^3 + 240 \times 10^3K_p - \frac{266K_i}{0.72 - 1.6K_p} > 0 \]  
(9.22) (9.23)

Equation (9.22) shows that \( K_p \) must never be greater than 0.45 or the system will go unstable. However, in equation (9.23), it shows that \( K_p \) must at least be greater than 0.45 if the left-hand side of the equation is to be kept positive. The analysis thus suggests that if the slope \( K_x \) is 150 N/mm or higher, for example, the use of a PI controller will not be adequate to achieve a stable control of disc position.

Figure 9.11 shows the root locus of the same control system. It is shown that the root locus is affected considerably by the location of the positive pole. The system could be made conditionally stable by properly selecting the gain of the PI controller, assuming that the positive pole is not too far away from the origin or the slope \( K_x \) is relatively small. For example, when \( K_x \) equals to 50 N/mm, the system is stable when the proportional gain is adjusted to between 0.22 and 0.56 and the corresponding integral gain 44 and 112 sec^{-1} respectively.

As a result of the above analysis, it is concluded that the stability of the system cannot be satisfied by the PI controller alone. To make the system stable, the serial compensation method may be used. It can be shown that the system locus can be made to shift to the left-hand s-plane if one or more zero's are added to the control circuit[61]. In other words, the
stability of the system can be improved if lead compensation is used. Figure 9.12 shows the modified root locus after a lead compensator is added to the open-loop transfer function. From the figure, the system is found to be stable when the loop gain is increased to a value between $2.4 \times 10^8$ and $8 \times 10^8$. The location of the added zero was found by trial-and-error using the same root-locus program and is shown to be equal to 1000 sec.

If the associated pole of the lead compensator is chosen to be ten times greater than the zero, then the compensator transfer function is given as

$$LL(s) = \frac{(s + 1000)}{(s + 10000)}$$

Expressing the time constants in terms of resistance and capacitance, equation (9.24) becomes

$$LL(s) = \frac{1}{(s + \frac{1}{R_a C_a})} \cdot \frac{1}{(s + \frac{1}{R_b C_b})}$$

$$= \frac{1}{R_b (R_a + \frac{1}{s C_a})} \cdot \frac{1}{R_a (R_b + \frac{1}{s C_b})}$$

If $R_a = R_b = 10K\Omega$, then $C_a$ and $C_b$ will be found to be equal to 0.1\mu F and 0.01\mu F respectively. The construction of the lead compensator is shown in Figure 9.13 and is connected in series with the PI controller as illustrated in Figure 9.14.
The resulting open-loop transfer function therefore becomes

\[
G(s)H(s) = \frac{K_i K_a K_m K_f (K_p i s + 1)(s + 1000)}{s (s T_c + 1)(s + 10000)(m s^2 + b s + (K_s - K_x))}
\]

(9.26)

In studying the damping characteristic of the closed-loop system, it is found that because the two root loci closer to the imaginary axis are moving in opposite direction as the loop gain is increased, the maximum damping ratio achievable is 0.3, which occurs when the closed-loop poles from these two loci and the origin are colinear. The optimum loop gain under this condition is found to be \(3.2 \times 10^8\). The system will be more oscillatory if the loop gain is increased or decreased beyond this optimum value.

If the damping ratio of the closed-loop system is specified as 0.3, then the proportional and integral gains at the PI controller can be found by solving

\[
3.2 \times 10^8 = K_i K_a K_m K_f
\]

i.e. \(K_i = 1333 \text{ sec}^{-1}\).

Since \(K_{pi} = K_p/K_i = 5 \text{ msec}\), therefore \(K_p = 6.67\).

The above analysis suggests that the control of disc position in the double disc valve can be achieved by operating the valve in closed-loop together with a lead compensator and a properly chosen loop gain. In the next chapter, tests on the double disc valve under closed-loop position control are performed, which is then followed with some discussions.
Figure 9.1 Flow Diagram of Double Disc Valve Steady-State Simulation
Figure 9.2 Estimated Flow Rates across Double Disc Valve and Mechanical Valve at Constant Input Flow Rate

Figure 9.3 Estimated Pressure Drop across Valve Assembly at Constant Input Flow Rate
Figure 9.4 Comparison of Pressure-Flow Characteristics
2D m CM JD O 5° - aoai CL CN 03 i > -0 V \ <\ a■ o □ m-aZ > csI / ) CM (vj a. 2 z - o C3 L O 453x673 ■ a T3 □ OJ 455x559 « / ) L D s O O T C T 453x461 _ 481x587 Constant Pressure Drop

Figure 9.5 Force Components in a Double Disc Valve

Pressure Drop: Δ - 20.0 bar
υ - 50.0
+ - 150.0
Net Hydraulic Force (N) 1, 2 and 3 - Magnetic Force at 0.2, 0.4 and 0.6 A respectively

Figure 9.6 Estimated Net Hydraulic Force at Constant Pressure Drop

Maximum Net Hydraulic Force at Maximum Shock Absorber Velocity

Disc Displacement (x/TD)
Figure 9.7 Linearised Open-Loop Block Diagram

Figure 9.8 Linearised Closed-Loop Block Diagram
Figure 9.9: Definition of Channel Width in Double Disc Valve

Figure 9.10: Variation of Estimated Damping Coefficient with Disc Displacement (x/TD)

Estimated Damping Coefficient $B$ (Ns/m):

- TD = 0.4 mm

Disc Displacement ($x/TD$):

- 0.2
- 0.4
- 0.6
- 0.8
- 1.0
$K_x = 150 \text{ N/mm}$

Figure 9.11 Root Locus Plot with PI Controller only

$K_x = 50 \text{ N/mm}$

$K_{cp} = 26.7 \times 10^6$

$K_{cp} = 10.7 \times 10^6$
Figure 9.12 Root Locus Plot with Lead Compensator used

Figure 9.13 Construction of Lead Compensator
Figure 9.14. Linearised Closed-Loop System with Compensator Connected
Chapter 10

Experiments with the Double Disc Valve
10. Experiments with the Double Disc Valve

10.1 Steady-state Test

The steady-state test set-up for the double disc valve is shown in Figure 10.1. The schematic diagram for Figure 10.1 is the same as the one for the single disc valve. The objective of the steady-state test is to study the pressure-flow characteristics while the position of the disc assembly is varied. In particular, the two important characteristics, which correspond to the close and full open conditions of nozzle 2, determine whether the upper and lower shock absorber limits described in Chapter 7 are satisfied.

The overall steady-state test was divided into two parts. The first part of the test was performed by entering no input signal to the valve controller. The pressure drop was increased gradually and the change of flow rate was recorded. The disc position was also monitored so that any change of disc position caused by the increase of pressure drop could be detected. As no current entered the magnetic coil, nozzle 2 should remain held closed by the positive net force and the valve should exhibit a high flow resistance due to the bleed flow alone.

During the test, there was no change of disc position observed as the pressure drop was varied. Figure 10.2 shows the test results obtained. The simulated pressure-flow characteristic is also drawn for comparison and is derived by assuming that an orifice characteristic can be obtained from the mechanical valve and when the two valves are operated together, the combined pressure-flow characteristic will be similar to curve A shown in Figure 9.4.
From Figure 10.2, it is found that the double disc valve passes slightly more bleed flow than is expected, particularly at higher pressure drops. The cause of this is suspected to be due to a lower discharge coefficient estimated by the empirical formulas for short orifice gap as discussed in Chapter 6. The decrease of flow resistance means that when the two valves are operated together under constant flow conditions, the overall pressure drop generated will be lowered. However, because the spring stiffness of the mechanical valve can be adjusted, it should be possible to maintain the upper pressure drop limit by using a slightly stiffer spring to reduce the flow passing through the mechanical valve.

In the second part of the test, the magnetic coil was activated so that the disc assembly was moved by the magnetic force from one side of the chamber to the other while the pressure drop across the valve was constant. The electronic controller was the same as the one used for the single disc valve except that a lead compensator was added in series with the PI controller as discussed in the last chapter. The test was conducted in closed-loop and the demanded position was entered to the controller via an input potentiometer. The error signal was produced by comparing the demanded signal with the signal fed back from the position transducer. The proportional and integral gains were set to 1.5 and 100 sec\(^{-1}\) respectively.

Figure 10.3 shows the change of pressure-flow characteristics under different disc positions. Each of the curves shown represents a different pressure drop applied across the valve. The disc assembly was driven in one complete cycle to test for hysteresis, which was found to be insignificant. The oil flow rate under different pressure drop conditions was shown not surprisingly to be a maximum when nozzle 2 was fully opened. In Figure 10.4, the simulated pressure-flow characteristic is plotted for comparison with the experimental data. The simulation results predict slightly lower
flow rates although the agreement is felt to be acceptable.

From the above results, it was demonstrated that the control of disc position in the double disc valve could be achieved by applying closed-loop position feedback. However, small fluctuations were observed when disc 1 was pulled to the end of the travel, i.e. when the left-hand side of nozzle 1 was fully opened. The cause of this fluctuation was suspected to be initiated by an increasing flow force at nozzle 2. At pressure drops below 20 bar, there was no sign of fluctuation throughout the disc travel. As the pressure drop applied was increased, disc 2 began to fluctuate but only at the fully open condition. Ideally a larger force appears to be required on the right-hand surface of disc 2 to keep the disc in contact with the push rod, i.e. the total left-hand force $F_{2n}$ in equation (9.13) should always be positive. The fluctuation signal seemed to suggest that the forces more or less cancelled each other out in this condition. Also, as it was shown in Figure 9.6 that when the disc is at full displacement, the hydraulic forces seem to be approximately constant and therefore the disc displacement is very sensitive to small current change in the coil. Possibly the dynamics of the position control loop are not fast enough to dampen the fluctuations under these conditions.

There were attempts made to eliminate the fluctuations. The first attempt was to lower the fluid flow force at the left-hand surface of disc 2 by reducing the bleed flow at nozzle 1. A larger diameter pin was used for this purpose. It was aimed at increasing the pressure drop along the annulus and thereby reducing the flow at nozzle 1. However, the modification produced a worse force balance condition as the use of maximum current in the magnetic coil was shown to be incapable of moving disc 2 to the left. The result seems to suggest that as the bleed flow was reduced,
the force acting on the left-hand surface of disc 2 was increased thereby stopping disc 2 from moving forward. The inability to move the disc assembly could be explained as a substantial increase of hydrostatic force on the left-hand end of the push rod. The loss of disc control could easily occur under these conditions as disc 1 could separate from the push rod due to the attraction from the magnetic coil. The position feedback would indicate no change as the transducer only monitors the position of disc 2.

In the above experiment, it was shown that when the bleed flow at nozzle 1 was reduced, a different flow condition was created which completely altered the force balancing at the disc assembly. It indicated that the bleed flow at nozzle 1 should not be reduced in order to maintain a low hydrostatic pressure at the annulus. The second experiment to eliminate the disc fluctuation was to reduce the hydrostatic pressure in the annulus by fitting a longer rod to reduce the restriction at the right-hand end of nozzle 1. However, this second attempt was also not successful as same type of fluctuation was observed when the valve was tested at 20 bar or above. It is suspected that because the flow restriction was reduced, the fluid flow force at the left-hand surface of disc 2 increased and the result was similar to the earlier condition in which the hydraulic forces on both sides of disc 2 were more or less balanced.

In the two experiments, attempts were made to reduce the force on the left-hand surface of disc 2 to eliminate fluctuations. Unfortunately, neither of these attempts was shown to be successful. An increase of force at the right-hand surface of disc 2 may be viewed as another possible alternative to increase the force \( F_{2n} \). But to implement this, an increase of nozzle diameter would be required so that a higher hydrostatic force and a lower negative fluid flow force would appear on the right-hand surface of disc 2. It was felt that to eliminate fluctuations while involving the least change
in component size, one possible alternative was to bolt the two discs to the push rod. The result of this modification proved to be very successful. No more fluctuation was detected when the disc assembly moved to the fully open position. This suggested that the position transducer was effectively monitoring the position of both discs and feedback control was restored even though the flow forces were small and varying.

10.2 Dynamic Test

The dynamic tests of the double disc valve were performed using a similar hydraulic set-up to that for the single disc valve. A voltage step representing the change of demanded disc position was applied to the valve controller. The resulting transient response was monitored; this included the pressure drop across the valve, the disc position and the coil current. The same controller setting as used in the steady-state tests was employed. Figures 10.5 and 10.6 are two typical dynamic test results showing that nozzle 2 was switched to open and close respectively. It is shown in Figure 10.5 that the coil current was increased initially by the applied voltage and decreased as the disc moved. The decrease of coil current was caused by the negative voltage generated at the lead compensator which also functions as a differentiator. This negative voltage varied in proportion to the disc velocity and would become positive when the disc was moving in the opposite direction. As the lead compensator had an effect of reducing the disc overshooting, the response of the valve could possibly be improved by a careful selection of the compensator gain.
It is noticed that the response of the disc tended to be more oscillatory than in the single disc valve. The cause of this can be explained by the root locus diagram given in Figure 9.12. Because the two root loci were moving in opposite direction as the loop gain was increased, the maximum damping ratio occurred when the closed-loop poles of the two opposite-moving loci and the origin were colinear. This maximum damping ratio was found to be 0.3 as the closed-loop gain was increased to \(3.2 \times 10^8\) in the analysis. However, although the valve may still be stable, further increase or decrease of loop gain can cause the damping ratio to reduce further and consequently the disc will be more oscillatory.

In Table 1 below, a summary of the dynamic test results under closed-loop position control is given. It is found that the response time of the disc varied between 10 and 30 msec.

<table>
<thead>
<tr>
<th>Pressure drop (bar)</th>
<th>Test No.</th>
<th>Initial Position ((x_0/T_D))</th>
<th>Use of Initial Current</th>
<th>Specified Movement ((s_m/T_D))</th>
<th>Steady-State Time (msec)</th>
</tr>
</thead>
<tbody>
<tr>
<td>15.7</td>
<td>1</td>
<td>0.00</td>
<td>no</td>
<td>0.25</td>
<td>45.0</td>
</tr>
<tr>
<td>15.7</td>
<td>2</td>
<td>0.25</td>
<td>yes</td>
<td>0.25</td>
<td>23.7</td>
</tr>
<tr>
<td>15.7</td>
<td>3</td>
<td>0.50</td>
<td>yes</td>
<td>0.25</td>
<td>16.2</td>
</tr>
<tr>
<td>15.7</td>
<td>4</td>
<td>0.00</td>
<td>no</td>
<td>0.50</td>
<td>26.2</td>
</tr>
<tr>
<td>15.7</td>
<td>5</td>
<td>0.25</td>
<td>yes</td>
<td>0.50</td>
<td>18.7</td>
</tr>
<tr>
<td>15.7</td>
<td>6</td>
<td>0.00</td>
<td>no</td>
<td>0.75</td>
<td>31.0</td>
</tr>
<tr>
<td>15.7</td>
<td>7</td>
<td>0.00</td>
<td>no</td>
<td>1.00</td>
<td>32.5</td>
</tr>
<tr>
<td>15.7</td>
<td>8*</td>
<td>0.75</td>
<td>yes</td>
<td>0.50</td>
<td>28.0</td>
</tr>
<tr>
<td>25.4</td>
<td>9</td>
<td>0.00</td>
<td>no</td>
<td>0.75</td>
<td>26.2</td>
</tr>
<tr>
<td>25.4</td>
<td>10</td>
<td>0.00</td>
<td>no</td>
<td>1.00</td>
<td>28.7</td>
</tr>
<tr>
<td>25.4</td>
<td>11*</td>
<td>0.75</td>
<td>yes</td>
<td>0.50</td>
<td>23.7</td>
</tr>
</tbody>
</table>

* Reverse switching. Disc moved towards nozzle 2.

Table 1 - Summary of the dynamic tests performed.

From the results in Table 1, the following points can be made:

1) The valve response was faster when the disc was located further away from nozzle 2. This is shown in tests 2 and 3 in Figure 10.7 and tests 4
and 5 in the table. The reason for this is suspected to be due to a
decrease of damping force as the gap between the disc and the chamber wall
was increased. However, in the case of test 1, the long response time was
also caused by the time taken for the current to rise to a minimum level
before sufficient magnetic force was generated.

2) It is shown that the response time was shorter when a larger error
signal was used. This is indicated by tests 1 and 4 in Figure 10.8 and
tests 2 and 5 in the table. Before the disc moved, the current rose more
rapidly in test 4 than in test 1. The result of this reveals that the
charging time in the integrator was shortened by the larger error signal
used. However, between tests 2 and 5, the increase of error signal had made
the travel of the disc in the positive velocity direction longer and, since
a negative voltage was generated in proportion to this velocity signal in
the lead compensator, the current was shown to be reduced to a much lower
level, causing a faster return of the disc to the desired position.

3) The response time was shorter when the valve was operated at a higher
pressure drop. This is shown in tests 6 and 9 in Figure 10.9 and tests 7
and 10 and tests 8 and 11 in the table. In test 9 the disc was shown to be
more stable and move more rapidly than in test 6 when a higher pressure
drop was used. The change of disc response seems to suggest that the net
hydraulic force had increased, which agrees with the result in Figure 9.6
where the estimated net hydraulic force was higher when a higher pressure
drop was used across the valve.

4) The switching of the disc was faster from left to right than in the
opposite direction. This is shown in tests 5 and 8 in Figure 10.10. As the
difference in response time was found to occur after the disc overshoot,
the magnetic force is suspected to have caused this difference. Because the
Experiments with the Double Disc Valve

The disc in test 5 was in a position closer to the coil after switching and since the magnetic force gradient there was greater, any change of coil current around that region was likely to generate a bigger change in magnetic force and therefore the disc would be expected to respond faster.

All the dynamic tests described above were based on changing the disc position while keeping the pressure drop constant. If the pressure drop is chosen as the controlled variable, a change of disc position will occur when there is a change of flow rate through the valve. As the function of the valve in the shock absorber application is similar to a pressure control device, therefore it should be capable of generating the required damping force regardless of any change of oil flow due to the change of shock absorber velocity. The response of the valve under the variable flow conditions occurring in a shock absorber is described in the next section.

10.3 Experiments with the Double Disc Valve and a Shock Absorber

This section describes the tests carried out on a modified shock absorber with the double disc valve and the mechanical valve mounted on the valve block. In Chapter 7, the construction of the shock absorber test assembly was described, a complete assembly of which is shown in Figure 10.11. The use of the test assembly is to simulate the actual working conditions so that the dynamic performance of the double disc valve under variable flow condition can be studied. Unlike the previous set-up, the uni-directional oil flow through the valve is generated by the reciprocating motion of the shock absorber. The function of the disc valve is to vary the flow resistance so that the pressure drop across the valves is altered and
therefore a change in the shock absorber damping force is made.

10.3.1 Step Position Change

The experimental set-up for the shock absorber test is shown in Figure 10.12. The test assembly was mounted on an INSTRON loading machine(1), which was operated through a programmable electronic controller(2). The body of the shock absorber assembly(3) was cycled up and down according to the frequency selected at the controller. To calculate the velocity of the shock absorber, the following equation may be used:

\[ \text{Velocity} = \text{Frequency} \times \text{Twice Stroke Length} \] (10.1)

This equation is applied only when the shock absorber is programmed at constant velocity, which is equivalent to selecting a triangular wave for the time and displacement from the controller panel.

During the tests, the stroke length was set to 80 mm, which is 20 mm less than the maximum stroke of the shock absorber. Therefore, if the shock absorber is cycled at 3 Hz, the velocity will be 0.48 m/s and in calculating the total flow rate across the valves, the following cross-sectional areas are used:

- Compression cycle: \( A_{co} = \pi \cdot D_{rod}^2 / 4 \)
- Rebound cycle: \( A_{re} = \pi \cdot (D_{pis}^2 - D_{rod}^2) / 4 \)

With the rod and piston diameters equal to 15.9 mm and 25.8 mm, the cross-sectional areas for the compression and rebound cycles are found to be 1.98 \( \times 10^{-4} \) mm\(^2\) and 3.24 \( \times 10^{-4} \) mm\(^2\) respectively.
The measured signals included the upstream and downstream pressures of the parallel valves, the position of the disc and the coil current. Figures 10.13 and 10.14 are typical test results with the double disc valve being un-energised and fully energised respectively. The positive velocity signal indicated that the shock absorber was in the compression cycle and negative in the rebound cycle. When the disc valve was not energised, the pressure drop generated was higher in the rebound cycle than in the compression cycle. This is because the valves were passing more oil in the rebound cycle due to a larger cross-sectional area. As the disc valve was energised, the pressure drops fell to a low level. The change of pressure drop suggests that the shock absorber damping force was adjusted correctly as a result of the disc position change in the valve chamber.

During the dynamic test, the disc position was switched from the hard to soft damping condition and vice versa while the shock absorber was cycled at constant velocity. The double disc valve was operated under closed-loop position control and was triggered via a microswitch(4) located at the INSTRON machine to synchronise the switching point. Figure 10.15 shows the wiring diagram for this synchronisation. It is shown that the double disc valve can be triggered either in the compression or the rebound cycle, depending on the wires connection at the switch. The 5 V output from the triggering and delay circuit was not generated until the reset switch was connected to a 5 V source. This ensured that the voltage step applied to the valve controller was produced only when a voltage pulse was generated and after the data acquisition system was triggered.

Figure 10.16 shows the transient response of the shock absorber assembly as the disc was switched from zero to the full travel position, i.e. from hard to soft damping condition. It is shown that the coil current did not rise instantaneously after the triggering signal. This was because the disc
position signal had not been set to zero prior to the step input change and as a result the integrator was charged to negative saturation due to the negative error present. Because the position signal was sensitive to disc alignment, the voltage sometimes fluctuated. The positive position offset was to avoid any current flowing to the coil when the demanded signal was zero. As the step input was applied, although the error signal changed to a positive value, the current did not flow to the magnetic coil until the integrator was positively charged. This delay of current response is not shown in Figure 10.17 or 10.18 because the coil was already in operation and the disc was held at a specified initial position.

In Figure 10.17, the demanded position signal occurred while the shock absorber was moving in the compression stroke, whereas in Figure 10.18 the triggering occurred as the shock absorber was in the rebound cycle. From the two results, the change of upstream pressure generally followed the change of disc position. As nozzle 2 was opened, the upstream pressure dropped and the damping force at the shock absorber was correspondingly reduced. Conversely, the damping force was increased when nozzle 2 was shut. Although the downstream pressure was quite noisy, it was low enough to be assumed as atmospheric. In Table 2 below, the response time of the disc and the upstream pressure are summarised. The comparison of the upstream pressure with the estimated pressure is also given. The estimated pressure, which is given in Figure A8.3 in Appendix A8, is found by using the same steady-state simulation model developed in Chapter 9 except that the pressure-flow characteristic assumed for the mechanical valve was replaced by the experimental result shown in Figure A8.2.
**Experiments with the Double Disc Valve**

<table>
<thead>
<tr>
<th>Shock absorber operating frequency (Hz)</th>
<th>3</th>
<th>2</th>
<th>1</th>
</tr>
</thead>
<tbody>
<tr>
<td>Change of disc position from 0 (nozzle 2 closed) to 0.4 mm (nozzle 2 fully opened)</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Upstream Pressure before change (bar)</td>
<td>18.0 (27.7)</td>
<td>11.7 (21.1)</td>
<td>5.5 (11.6)</td>
</tr>
<tr>
<td></td>
<td>+28.1 (33.4)</td>
<td>18.7 (28.2)</td>
<td>9.1 (17.1)</td>
</tr>
<tr>
<td>Pressure after change (bar)</td>
<td>3.7 (2.5)</td>
<td>2.3 (1.3)</td>
<td>N/A (0.5)</td>
</tr>
<tr>
<td></td>
<td>+6.3 (4.7)</td>
<td>3.3 (2.5)</td>
<td>1.6 (0.9)</td>
</tr>
<tr>
<td>Response time (ms)</td>
<td>24.0</td>
<td>28.0</td>
<td>22.0</td>
</tr>
<tr>
<td>Change of disc position from 0.4 mm to 0</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Pressure before change (bar)</td>
<td>3.0 (2.5)</td>
<td>2.0 (1.3)</td>
<td>N/A (0.5)</td>
</tr>
<tr>
<td></td>
<td>+5.9 (4.7)</td>
<td>3.5 (2.5)</td>
<td>1.2 (1.9)</td>
</tr>
<tr>
<td>Pressure after change (bar)</td>
<td>16.2 (27.7)</td>
<td>11.7 (21.1)</td>
<td>5.6 (11.6)</td>
</tr>
<tr>
<td></td>
<td>+26.5 (33.4)</td>
<td>19.5 (28.2)</td>
<td>9.0 (17.1)</td>
</tr>
<tr>
<td>Response time (ms)</td>
<td>32.0</td>
<td>32.0</td>
<td>30.0</td>
</tr>
<tr>
<td>Change of disc position from 0.1 to 0.3 mm</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Pressure before change (bar)</td>
<td>6.3 (7.4)</td>
<td>3.7 (4.3)</td>
<td>1.9 (2.0)</td>
</tr>
<tr>
<td></td>
<td>+12.9 (13.5)</td>
<td>6.6 (7.4)</td>
<td>2.7 (3.2)</td>
</tr>
<tr>
<td>Pressure after change (bar)</td>
<td>3.7 (3.1)</td>
<td>2.3 (1.7)</td>
<td>0.6 (0.7)</td>
</tr>
<tr>
<td></td>
<td>+6.2 (6.0)</td>
<td>3.9 (3.2)</td>
<td>1.4 (1.2)</td>
</tr>
<tr>
<td>Response time (ms)</td>
<td>14.0</td>
<td>18.0</td>
<td>15.0</td>
</tr>
<tr>
<td>Change of disc position from 0.3 to 0.1 mm</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Pressure before change (bar)</td>
<td>3.3 (3.1)</td>
<td>2.3 (1.7)</td>
<td>0.8 (0.7)</td>
</tr>
<tr>
<td></td>
<td>+6.9 (6.0)</td>
<td>3.7 (3.2)</td>
<td>1.6 (1.2)</td>
</tr>
<tr>
<td>Pressure after change (bar)</td>
<td>7.7 (7.4)</td>
<td>3.7 (4.3)</td>
<td>1.6 (2.0)</td>
</tr>
<tr>
<td></td>
<td>+13.3 (13.5)</td>
<td>6.6 (7.4)</td>
<td>2.4 (3.2)</td>
</tr>
<tr>
<td>Response time (ms)</td>
<td>26.0</td>
<td>28.0</td>
<td>25.0</td>
</tr>
</tbody>
</table>

* Upstream pressure during compression cycle
+ Upstream pressure during rebound cycle
Readings in brackets are the estimated pressure drop

**Table 2 - Summary of dynamic tests with the shock absorber**
All the results in Table 2 were obtained with the valve triggered in the compression cycle. Similar experiments but with the valve triggered in the rebound cycle had also been performed, the results of which were found to be similar to the ones given in Table 2. The response times of the disc under variable flow condition are similar to the results described in the last section and are shown to vary between 10 and 30 msec.

The comparison between the actual and predicted pressure drop when nozzle 2 was closed is shown to be far from satisfactory. The error could be due to using an underestimated discharge coefficient in the steady-state simulation model described in Section 10.1. However, as difficulty was experienced in filling the shock absorber with the shock absorber oil, the presence of air bubbles inside the shock absorber chambers may also be responsible for the lower pressure drop generated. In Figure 10.19a, the pressure generated in the compression cycle was not increased until the volume of the compression chamber was reduced by half. The delay of pressure rise suggests that an air expansion might have occurred in the compression chamber during the rebound cycle. The lower upstream pressures could be an indication of the collapse of the air bubbles, which tended to reduce the volume of oil passing through the valves.

### 10.3.2 Upstream Pressure Control

In the above results, it is shown that by changing the disc position a different pressure drop could be generated across the shock absorber test assembly. To control the magnitude of the pressure drop, the disc position must be able to change automatically in order to react to the changing operating conditions. In Figure 10.20 the block diagram of the closed-loop pressure control system is illustrated. The upstream pressure is shown to
be used as the controlled variable and the feedback signal is connected to
the summing junction in such a way that to increase the upstream pressure,
a negative error signal is needed. The reason for this is that a higher
pressure drop is generated as the disc displacement becomes smaller. The
negative gradient shown in Figure 10.21 may also be used to explain this.
The graph represents calculated data which has been cross-plotted to
provide pressure drop variation with disc displacement for various constant
velocity conditions using the steady-state simulation.

In general, the generated pressure drop is a function of disc displacement
and oil flow rate. Therefore, the following linearised relationship may be
established:

\[ P = K_{px} \Delta x + K_{pq} \Delta Q \]  \hspace{1cm} (10.2)

where

\[ K_{px} = \frac{\partial P}{\partial x} \quad \text{and} \quad K_{pq} = \frac{\partial P}{\partial Q}. \]

If the shock absorber is operating at constant velocity, then \( K_{pq} \) is zero
and the pressure drop is related to the disc displacement by the
coefficient \( K_{px} \) as shown in Figure 10.20. The coefficient \( K_{px} \) may be
estimated by referring to the gradient shown in Figure 10.21. In the
present study, point 0 is assumed to be the steady-state operating point,
which was chosen for the reason that from the previous results, the
greatest pressure change seems to occur between the 0 and the 0.1 mm disc
positions. Thus \( K_{px} \) is found to be \(-240\) bar/mm. If the linearised model
described in Chapter 9 is valid, then for a pressure transducer with \( K_f \)
equal to 16 bar/V, it can be shown that the valve could be stable when the
same controller developed for the position control is used. However, it is
noted that because the pressure gradient varies considerably with the disc
displacement and the shock absorber velocity, modification on the
controller might be required so as to maintain stability over the entire operating range.

Figure 10.22 is a typical result of the valve operated under closed-loop pressure control with the shock absorber being subjected to step changes of velocity. No synchronising switches were needed for these tests. It is shown that the disc was held further away from nozzle 2 in the rebound cycle than in the compression cycle. This is because a higher flow rate was generated in the rebound cycle and in order to maintain the same upstream pressure as the compression cycle, the valve must be set softer. The more oscillation shown in the rebound cycle could be due to the reduction of loop gain. As the displacement was larger in the rebound cycle, the pressure gradient was lower, therefore the loop gain was reduced.

For the tests illustrated in Figure 10.23, the shock absorber was switched at constant velocity as before but instead of a constant upstream pressure demanded signal, a 10 Hz sinusoidal input was used. The purpose of the test was to study the response of the valve under practical control conditions. In this test, the movement of the car body was assumed to be represented by the 1 Hz velocity signal while the 10 Hz signal was assumed to be the oscillation of the wheel due to the irregularities of the road surface. Ideally, to attenuate the wheel disturbance before it is transmitted to the car body as the wheel leaves the ground, a correct damping force should be generated at the same frequency as the disturbance signal. The test should show the ability of the valve to provide continuous correction under rapid dynamic conditions.

From the result of Figure 10.23, the pressure feedback loop was found to operate satisfactorily. The input signal in the graph has been inverted to
facilitate comparison with the generated pressure signal which is shown to be in opposite phase to the position signal.

In Figure 10.24, the constant velocity signal was replaced by a sinusoidal signal and the test was found to be not quite so successful. The disc valve is shown to have failed to control the magnitude of the generated pressure as indicated by points A and B in the figure. The cause of this was due to the negative saturation of the controller integrator. It was found that when the velocity signal was low, the coil current had to be zero in order to keep nozzle 2 closed. However, although the valve was at maximum flow resistance, the generated pressure was still not high enough and as a result a negative error signal was produced, which after a short time saturated the integrator to the negative limit. As the shock absorber velocity increased, more flow was generated, forcing the pressure to rise before the integrator action could charge from negative saturation. This problem would probably not arise if the shock absorber did not have air inclusion and valve deficiencies so that higher pressure could be generated by the valve.

The results from a different testing condition are shown in Figure 10.25. The shock absorber was set to cycle at 10 Hz frequency while the input signal was changed to 1 Hz. Initially, the generated pressure was the same as the input signal, but as the magnitude of the input signal decreased, it reached to a minimum level where a further decrease in signal magnitude caused no further decrease in generated pressure. This minimum level was equivalent to the minimum pressure drop at the disc valve when nozzle 2 was fully opened and is marked in the figure by lines A and B for the compression and rebound cycles.
In the last test, the constant velocity was replaced by a sinusoidal velocity. Figure 10.26 shows that the same result as described above occurred as the input signal fell below the minimum level. However, the controlled pressure was found to be higher than the demanded signal as in the other part of the cycle. The cause of this is thought to be due to the change of shock absorber velocity and the disc valve had not been fast enough to bring the pressure under control as the flow rate was changing. However, it should be remembered that in the pressure control analysis, it had been assumed that $K_{pq}$ was zero, which is true only when the shock absorber velocity is constant. Therefore, for variable velocities, a detailed analysis taking the change of flow rate into account should be made as there may be a need to change the valve controller under these conditions.
Figure 10.1 Steady-State Test Hydraulic Set-up
Experiments with the Double Disc Valve

Figure 10.2 Double Disc Valve Pressure - Bleed Flow Characteristic with Nozzle2 closed
Total Flow Rate (l/min)

Pressure Drop:

\[ \Delta p = 9.0 \text{ bar} \]
\[ +,x = 14.5 \]
\[ o,0 = 20.0 \]
\[ o,\times = 25.8 \]
\[ \Delta p = 31.3 \]

**Figure 10.3** Steady-State Test Result at Constant Pressure Drop

\[ \Delta = 20.0 \text{ bar} \]
\[ \triangledown = 31.3 \]

Total Flow Rate (l/min)

Simulated Characteristics at 20.0 and 31.3 bar respectively

**Figure 10.4** Comparison of Pressure-Flow Characteristics at Constant Pressure Drop
Pressure Drop : 15.7 bar

1 - Triggering Signal
2 - Disc Position
3 - Coil Current
4 - Supply Pressure
5 - Discharge Pressure

$x_0 = 0.2 \text{ mm} \quad x_f = 0.3 \text{ mm}$

Figure 10.5 Typical Voltage Step Test with Disc Position increasing at Constant Pressure Drop

Pressure Drop : 15.7 bar

1 - Triggering Signal
2 - Disc Position
3 - Supply Pressure
4 - Coil Current
5 - Discharge Pressure

$x_0 = 0.3 \text{ mm} \quad x_f = 0.1 \text{ mm}$

Figure 10.6 Typical Voltage Step Test with Disc Position decreasing at Constant Pressure Drop
Figure 10.7

Dynamic Response of Test 2

Position Change: 0.1 to 0.2 mm
Pressure Drop: 15.7 bar

Time ($\times 10^{-2}$ s)

Voltage (V)

23.7 ms

Dynamic Response of Test 3

Position Change: 0.2 to 0.3 mm
Pressure Drop: 15.7 bar

Time ($\times 10^{-2}$ s)

Voltage (V)

16.2 ms

$xf$

$x_i$

$f_i$
Dynamic Response of Test 6

Figure 10.9

Dynamic Response of Test 9
Figure 10.11 Complete Valve-Shock Absorber Assembly
Figure 10.12 Shock Absorber Test Set-up

INSTRON Controller
Machine
Disc Valve Controller
Data Acquisition System
Figure 10.12 Shock Absorber Test Set-up

Shock Absorber Body (3)
Mechanical Valve
Double Disc Valve

Microswitch (4)
Detail (A)
**Figure 10.13** Typical Shock Absorber Test

Result with Nozzle2 closed

**Figure 10.14** Typical Shock Absorber Test

Result with Nozzle2 fully opened
Figure 10.15 Electrical Set-up for Shock Absorber Voltage Step Test

Figure 10.16 Typical Shock Absorber Voltage Step Test Result

1 - Disc Position (from 0 to 0.4mm)
2 - Triggering Signal
3 - Coil Current
4 - Upstream Pressure
5 - Downstream Pressure

Stroke Length: 80mm
Frequency: 2Hz
Pressure Scale: 8 bar/V
1 - Triggering Signal
2 - Disc Position (from 0.3 to 0.1mm)
3 - Coil Current
4 - Upstream Pressure
5 - Downstream Pressure
6 - Shock Absorber Velocity

Figure 10.17 Typical Voltage Step Test with Disc Position decreasing

Enlargement of Figure 10.17
1 - Triggering Signal
2 - Disc Position (from 0.1 to 0.3 mm)
3 - Shock Absorber Velocity
4 - Coil Current
5 - Upstream Pressure
6 - Downstream Pressure

Voltage (V)

Stroke Length: 80 mm
Frequency: 2 Hz

Enlargement of Figure 10.18
Figure 10.19 Dynamic Response of Shock Absorber at 5Hz Constant Velocity

a. Nozzle 2 closed
b. Nozzle 2 fully opened

1 - Shock Absorber Velocity
2 - Disc Position
3 - Upstream Pressure
4 - Downstream Pressure

Voltage (V)
Figure 10.20  Linearised Pressure Feedback Block Diagram
Experiments with the Double Disc Valve

Pressure Drop (bar)

\[ \Delta (1, 2, 4, 6) - \text{Rebound Cycle at 5, 4, 3, 2 Hz Frequency} \]

\[ \nabla (3, 5, 7, 8) - \text{Compression Cycle at 5, 4, 3, 2 Hz Frequency} \]

Constant velocity at 80 mm stroke length; TD = 0.4 mm

\[ K_{px} = \frac{\Delta P}{\Delta x} = -240 \text{ bar/mm} \]

Figure 10.21 Variation of Pressure Drop generated with Disc Displacement at Constant Shock Absorber Velocity
Stroke Length: 80 mm  
Frequency: 2 Hz

Figure 10.22 Typical Constant Pressure Control Test Result
Figure 10.23 a) Variation of Upstream Pressure at Constant Shock Absorber Velocity

Figure 10.23 b)
Figure 10.24a) Variation of Upstream Pressure at Sinusoidal Shock Absorber Velocity

Figure 10.24b)
Figure 10.25a). Variation of Upstream Pressure
at High Frequency Constant
Shock Absorber Velocity

Figure 10.25b)
**Figure 10.26a**. Variation of Upstream Pressure at High Frequency Sinusoidal Shock Absorber Velocity

**Figure 10.26b**. Nozzle2 fully opened
Chapter 11

Summary of the Double Disc Valve
11. Summary of the Double Disc Valve

Through using the simplified discharge coefficient approach, a double disc valve was designed to meet specified pressure-flow requirements. The assumed discharge coefficient was known to contain inaccuracies as the actual flow condition was a typical combination of separated flow and re-attached laminar flow as discussed before. However, the estimated flow rate using the orifices calculated was found to be quite accurate except the bleed flow, which occurred when nozzle 2 was closed, was lower than the experimental values. The cause of the lower flow rate was shown to be due to the inadequacy of the parabolic function fit used as described in Figure 6.3. It is believed that by improving the empirical formulas, the estimation can be made more accurate. However, it is noted that the empirical formulas are applicable only to a similar valve geometry; such as a similar gap-to-land ratio and a similar nozzle-to-disc diametrical ratio. Further work on validating the discharge coefficient approach and the empirical formulas is desirable.

By connecting a lead compensator to the valve control circuit, the position control instability shown in open-loop was eliminated. The linearisation method and the subsequent root locus plot had proven to be capable of showing how the control stability might be achieved. Because of the inherent dynamic characteristic of the valve controller, the step response of the valve tended to be more oscillatory than in the single disc valve. As the valve was conditionally stable, instability could occur if the loop gain was not within the lower and the upper limits. The limitation of the linearisation method described in the single disc valve summary also applies.
The function of the valve as a variable orifice in a shock absorber was generally satisfactory. The generated pressure drop was shown to be controllable by changing the disc position. Because of a lower pressure-flow characteristic in the mechanical valve, the comparison of the generated and theoretical pressure drops was not possible. However, using the state-steady model and the experimental characteristic of the mechanical valve, comparison of the test results with the estimated pressure drops was made. It was found that the lower pressure drop generated might not be due entirely to the inaccuracy of the steady-state model. The presence of air bubbles coupled with a slow-acting recuperation valve could be a major factor in reducing the oil flow and causing the lower pressure drop.

From the results of the switching tests performed in the shock absorber, the dynamic performance of the disc valve was shown to be quite similar to that obtained under constant pressure drop conditions when testing the valve by itself. The response time is believed to be dominated by the controller settings as such factors as air bubbles, slow-acting mechanical valve, etc were not present in the latter tests.

Instead of being a flow control valve, the double disc valve could also be operated as a pressure control device. The use of pressure feedback enabled the damping force of the shock absorber to be adjusted directly by the demanded signal. From the results of Figure 10.22, it seems that the disc valve was fast enough for the shock absorber application as the valve responded basically at the same frequency as the input signal.

As the shock absorber can be turned into a variable damper by implementing pressure feedback in the disc valve, the use of position transducer will be unnecessary in future designs. With the elimination of the position
transducer, the valve construction can be simplified considerably and is more available to miniaturisation. However, the use of position transducer will still be essential if the valve is to be used as a flow control valve due to the non-linear magnetic force characteristic and the small spring constant.
Chapter 12

Conclusions and Future Work Recommendations
12. Conclusions and Future Work Recommendations

12.1 Conclusions

This thesis is concerned with the development of two position-controlled disc valves. Earlier work at Surrey University concentrated on single and double disc valves which can be operated as proportional or digital devices without position feedback. This was because an inherent balance could always be made between the hydraulic and electro-magnetic forces. In the new valves, because of a different design adopted, stable operation is only possible through position or pressure feedback control.

The use of position feedback in hydraulic valves is quite common. For example, the control of spool position in a high quality electro-hydraulic servo-valve or more recently in solenoid actuated directional valves. The advantage of incorporating position feedback is the valve's controllable element can be held at the same position regardless of any load change or hydraulic disturbances. With the advance of transducer technology made in recent years, very accurate, cheap devices should now be incorporated into a variety of valves. However, most of these devices, such as the LVDT, are still relatively expensive and bulky and therefore not suitable for disc valve control.

In the case of the single disc valve, a special differential capacitive transducer was designed and worked satisfactory, although it added to the complexity of the valve from a manufacturing point of view. For the double disc valve, a proprietary inductive transducer was used with success, although it was too bulky and expensive for a production valve. Thus, in general, further thought needs to be given to transducer design if this type of disc valves is to be manufactured economically. In the application
to shock absorbers, pressure feedback was shown to be attractive so that problem of choosing a pressure transducer is minimised as a variety of cheap, rugged devices are readily available and do not need to be integrated into the valve.

The most probable use of a position-controlled single disc valve is to operate it as a two-port flow control device. It was shown that by adjusting the disc position, a corresponding change of flow rate could be made. This change of flow rate could be used directly to control, say, the speed of a hydraulic motor or as a pilot device in a two-stage hydraulic valve. However, by connecting more than one single disc valves together as described in Chapter 2, more complicated functions such as the ones performed by a 3-way or a 4-way valve may also be implemented. Because the connected disc valves are energised independently, a variety of control functions are possible under micro-processor control.

The making of a variable damper by connecting a double disc valve to a modified conventional shock absorber was shown to be quite successful. Despite the non-linearities of the hydraulic and electro-magnetic forces generated, the shock absorber damping force was adjusted successfully by changing the disc position inside the valve chamber. Although the double disc valve was originally designed as a position-controlled valve, it was found that the control of damping force could be achieved more conveniently through a pressure feedback loop. However, useful results were obtained only when the shock absorber was cycled in the 1 to 3 Hz frequency range. The presence of air bubbles in the shock absorber chambers and the incorrect operation of a mechanical non-return valve derated the performance of the prototype unit but the initial tests are sufficiently promising to warrant a more detailed investigation.
The response of the valve to step demand changes was in the range of 10 to 30 msec which was adequate to provide body vibration isolation from road noise with this type of damper. More work is required on the exact form of controller and compensator design for pressure feedback control in order to reduce the effect of non-linearities in the valve. Here, the linearisation technique together with a root-locus analysis gave valuable insight into the valve stability and response.

12.2 Future Work Recommendations

The inaccuracy of the estimated flow rate as the disc approached the nozzle had been shown to be caused by the values of the discharge coefficients used. It seems that the error incurred could be avoided had a better polynomial function fit been derived. To improve the estimation accuracy, further validation of the discharge coefficient approach and a better experimental model for finding the discharge coefficients are desirable. Furthermore, the effect of increasing the nozzle-to-disc diametrial ratio should be investigated as it would help to provide the information required for reducing the valve size.

As a result of attempting to predict the hydraulic force, it was found that more experimental work in this area was needed. In the single disc valve model, it was not possible to identify the limitation of the laminar theory. The difficulty was compounded by the questionable interpretation of the position signal which was shown to have a considerable influence on the hydraulic force calculated. This was because the disc position might be measured in a tilted condition. A more realistic approach may be to study
the hydraulic force acting on a tilted disc, both experimentally and theoretically, and use the estimated maximum value as the basic design criteria.

The effect of flow separation for the supply nozzle has yet to be investigated. Although it may not be too significant on a large disc, the understanding of how separation occurs in a disc valve and what the radial pressure distribution is like in general would be very useful from a theoretical point of view.

The damping coefficient assumed in the linearisation analysis needs to be verified. As the linearisation method has proven to be useful, improving the damping coefficient accuracy could ensure a better controller design. On the other hand, studying the damping coefficient experimentally could help to validate the dynamic equations derived.

Despite the noise and temperature problems, the capacitive position transducer developed may still be an attractive low-cost alternative to use in a disc valve requiring position feedback control. Although the temperature problem was shown to have been minimised by the push-pull arrangement, further study of the transducer stability under adverse temperature condition is needed. The transducer dynamic response, which is currently bounded by the frequency bandwidth of the demodulation circuit, is another area that needs further investigation. Operating the transducer at higher bridge frequency may be advantageous as far as increasing the frequency bandwidth is concerned, but the increase of the cable capacitance effect may change the problem into a transmission problem.
The successful control of damping force in a shock absorber demonstrates that a position-controlled disc valve may be turned into a pressure control valve by applying a similar linearisation analysis. However, it should be remembered that the coefficient, $k_{pq}$, was ignored in the analysis and the valve was still stable despite the use of a variable shock absorber velocity. This was because a low velocity change was implemented in the test and therefore the effect of $k_{pq}$ was not significant. To ensure that unstable operation will not occur, the effect of $k_{pq}$ must also be included in any future analysis. A re-formulation of the closed-loop transfer function is necessary if the disc valve is to be used as a pressure control valve in the shock absorber application or in other pressure control applications.
Bibliography
Bibliography


6. Rexroth Hydraulic Valves Data Sheets


58. Armstrong Shock Absorber Specification


Appendices
Appendix A1

Derivation of Steady-state Pressure Distribution

From equation (3.7), the simplified Navier-Stokes equation is

\[ \rho u \frac{\partial u}{\partial r} = - \frac{\partial P}{\partial r} + \mu \left( \frac{1}{r} \left( \frac{\partial (r u)}{\partial r} \right) \right) + \frac{\partial^2 u}{\partial z^2} \]  

(1)

The continuity equation in integral form is

\[ Q = \int_0^h 2\pi r u \, dz \]  

(2)

To find a velocity which will satisfy both equations (1) and (2), the iterative method is used. If the left hand term in equation (1) is assumed zero, then equation (1) will become

\[ \frac{\partial^2 u}{\partial z^2} = \frac{1}{\mu} \frac{\partial P}{\partial r} \]  

(3)

where \( u_0 \) denotes the first assumed velocity in the iteration process.

Integrating equation (3) twice,

\[ u_0 = \frac{1}{\mu} \frac{\partial P}{\partial r} \frac{z^2}{2} + C_1 z + C_2 \]

where \( C_1 \) and \( C_2 \) are the integration constants.

Applying the boundary conditions, where \( u_0 = 0 \) at \( z = 0 \) and \( h \),

\[ u_0 = \frac{1}{2\mu} \frac{\partial P}{\partial r} \left( z^2 - h z \right) \]  

(4)

Substituting (4) into (2) and integrating,

\[ Q = - \frac{\partial P}{\partial r} \pi r h^3 / (6\mu) \]  

(5)
Combining (4) and (5) and eliminating \( \frac{\partial P}{\partial r} \),

\[ u_0 = -3Q(z^2 - hz)/(\pi rh^3) \]  

(6)

This \( u_0 \) is the first assumed velocity in the iteration.

Substituting (6) into (1),

\[ \mu \frac{\partial^2 u}{\partial z^2} = \frac{\partial P}{\partial r} + \frac{9Q^2}{(\pi rh^6r^3)}(2hz^3 - h^2z^2 - z^4) \hspace{1cm} (7) \]

where \( u_1 \) is the resulting velocity after the first iteration.

Integrating equation (7) twice,

\[ \mu u_1 = \frac{\partial P}{\partial r} \left( \frac{z^2}{2} \right) + \frac{9Q^2}{(\pi rh^6r^3)} \left( \frac{h^2z}{10} - \frac{h^2z^4}{12} - \frac{z^6}{30} + \frac{h^5z}{60} \right) + C_1z + C_2 \]  

(8)

Applying the same boundary conditions,

\[ \mu u_1 = \left( \frac{\partial P}{\partial r} \right) \left( \frac{z^2 - hz}{2} \right) + \frac{9Q^2}{(\pi rh^6r^3)} \left( \frac{h^2z^5}{10} - \frac{h^2z^4}{12} - \frac{z^6}{30} + \frac{h^5z}{60} \right) \]  

(9)

Substituting (9) into (2) and integrating,

\[ \mu Q/(2\pi r) = - \left( \frac{\partial P}{\partial r} \right) h^3/12 + 27/840Q^2h^7/(\pi^2h^6r^3) \quad \text{or} \quad \frac{\partial P}{\partial r} = -6\mu Q/(\pi rh^3) + 27/70Q^2/(\pi^2h^6r^3) \]  

(10)

The pressure distribution given by equation (10) is resulted from a first-order iteration. A second- or higher-order iteration can be implemented by the same procedure. However, analysis shows that the magnitude change in the inertia term is very small when a higher-order iteration is used, but the pressure distribution expression will surely be more complicated.
Appendix A2

Inward Flow Momentum Equation

Referring to Figure A2.1, the sum of forces acting on the fluid element is

\[ F_{\text{sum}} = (\sigma_r - (\sigma_r/\partial r).dr).r.d\theta.dz - \sigma_r.(r + dr).d\theta.dz \]
\[ + (\tau_{r\theta} + (\partial \tau_{r\theta}/\partial \theta).d\theta - \tau_{r\theta}).dr.dz \]
\[ + (\tau_{zr} + (\partial \tau_{zr}/\partial z).dz - \tau_{zr}).((r + dr)^2 - r^2).d\theta/2 \]
\[ + 2.\sigma_\theta.d\theta.dz.dr/2 \]
\[ = (\sigma_r/\partial r).dr.d\theta.dz.r - \sigma_r.dr.d\theta.dz + (\partial \tau_{r\theta}/\partial \theta).d\theta.dr.dz \]
\[ + (\partial \tau_{zr}/\partial z).dz.d\theta.(2.r.dr + dr^2)/2 \]
\[ + \sigma_\theta.d\theta.dz.dr \]  

(1)

It can be shown that for constant density and viscosity, the normal and shear stresses at the fluid element are\[29]:

\[ \sigma_r = - P + 2.\mu.(\partial u/\partial r) \]  

(2)

\[ \sigma_\theta = - P + 2.\mu.((\partial v/\partial \theta)/r + u/r) \]  

(3)

\[ \tau_{r\theta} = \mu.(r.(\partial v/r)/\partial r) + (\partial u/\partial \theta)/r \]  

(4)

\[ \tau_{zr} = \mu.((\partial u/\partial z) + (\partial w/\partial r)) \]  

(5)

Therefore

\[ \partial \sigma_r/\partial r = - \partial P/\partial r + 2.\mu.(\partial^2 u/\partial r^2) \]  

(6)

\[ \partial \tau_{r\theta}/\partial \theta = \mu.((\partial^2 v/(\partial \theta.\partial r)) - \partial(v/r)/\partial \theta + (\partial^2 u/\partial \theta^2)/r) \]  

(7)

\[ \partial \tau_{zr}/\partial z = \mu.((\partial^2 u/\partial z^2 + \partial^2 w/(\partial z.\partial r)) \]  

(8)

Substituting (2) to (8) into (1),

\[ F_{\text{sum}} = (\partial P/\partial r - 2.\mu.(\partial^2 u/\partial r^2)).r - 2.\mu.(\partial u/\partial r) + \mu.((\partial^2 v/(\partial \theta.\partial r)) \]
\[ + (\mu/r).((\partial^2 u/\partial \theta^2) + \mu.r.(\partial^2 u/\partial z^2 + \partial^2 w/(\partial z.\partial r)) \]
\[ + (\mu/r).((\partial v/\partial \theta) + 2.\mu.u/r \]  

(9)
The continuity equation is derived as follows:

\[
(u - (\partial u/\partial r).dr).r.d\theta.dz - u.(r + dr).d\theta.dz +
\]
\[
(w + (\partial w/\partial z).dz) - w).((r + dr)^2d\theta - r^2d\theta) +
\]
\[
((v + (\partial v/\partial \theta).d\theta) - v).dr.dz + \partial(\rho.r.d\theta.dr.dz)/\partial t = 0
\]

Simplifying the above equation and differentiating w.r.t. \( r \),

\[- \partial u/\partial r - u/r + (\partial v/\partial \theta)/r + \partial w/\partial z = 0\]
\[\partial^2 u/\partial r^2 + (\partial u/\partial r)/r - u/r^2 + (\partial v/\partial \theta)/r^2 =\]
\[\partial^2 w/(\partial z.\partial r) + (1/r).((\partial^2 v/(\partial \theta.\partial r)))\]

(10)

Combining (9) and (10),

\[F_{sum} = \partial p/\partial r - \mu.(\partial^2 u/\partial r^2) - (\mu/r).(\partial u/\partial r) + \mu.u/r^2 +\]
\[+ (\mu/r^2).(\partial^2 u/\partial \theta^2) + \mu.(\partial^2 u/\partial z^2) + 2.\mu.(\partial v/\partial \theta)/r^2\]

(11)

Replacing \( F_{sum} \) with the total acceleration of the fluid element and assuming that the gravitational force is negligible, the final Navier-Stokes equation for radial inward flow is

\[\partial u/\partial t - u.(\partial u/\partial r) + v.(\partial u/\partial \theta)/r + v^2/r + w.(\partial u/\partial z) = \partial p/\partial r -\]
\[- \mu.(\partial(\partial u/\partial r)/r)/\partial r - (\partial^2 u/\partial \theta^2)/r^2 - 2.(\partial v/\partial \theta)/r^2 - \partial^2 u/\partial z^2)\]
Figure A 2.1 Fluid Flow Element in Radial Direction
Appendix A3

Derivation of Dynamic Pressure Distribution

From equation (3.24), the Navier-Stoke equation for radial inward flow is

\[ \rho \left( \frac{\partial u}{\partial t} - u \frac{\partial u}{\partial r} + w \frac{\partial u}{\partial z} \right) = \frac{\partial p}{\partial r} - \mu \left( \frac{1}{r} \frac{\partial}{\partial r} \left( r \frac{\partial u}{\partial r} \right) \right) - \frac{\partial^2 u}{\partial z^2} \]  

(1)

The inward flow continuity equation with zero tangential velocity in differential form is

\[- \frac{\partial (ru)}{\partial r} + \frac{\partial (rw)}{\partial z} = 0 \]  

(2)

The continuity equation as shown in Figure A3.1 is

\[ Q = \int_0^{h_2} u \cdot 2 \pi r \cdot dz - \pi r^2 x_2 \]  

(3)

The first assumed radial velocity \( u_\phi \) is found by applying the same method used in the steady-state in which the left hand term of equation (1) is put equal to zero and the result is

\[ u_\phi = 3 \cdot (Q + \pi r^2 x_2) \cdot (z \cdot h_2 - z^2) / (\pi r \cdot h_2^3) \]  

(4)

Combining equations (2) and (4), the first assumed axial velocity is given as

\[ w_\phi = - x_2 \cdot (3 \cdot z^2 \cdot h_2 - 2 \cdot z^3) / h_2^3 \]  

(5)
Substituting these velocities into (1),

\[ \mu \frac{\partial^2 u_1}{\partial z^2} = - \frac{\partial P}{\partial r} + 3 \rho (\dot{Q} + \pi r^2 \ddot{x}_2)(z_h^2 - z^2)/\left(\pi r h^3\right) - 9 \rho (- \frac{Q^2}{r^2} + \frac{\pi^2 r^2 \dot{x}_2^2}{2})(z_h^2 - 2 z^3 h^2 + z^4)/\left(\pi^2 r h^6\right) + 3 \rho (\dot{Q} + \pi r^2 \dot{x}_2) \dot{x}_2 (6 z_h^2 - 8 z^3 h^2 + 4 z^4 - 2 z_h^3)/(\pi r h^6) \]  

(6)

Integrating equation (6) twice and applying the boundary conditions of \( u_1 = 0 \) when \( z = 0 \) and \( h^2 \),

\[ \mu u_1 = - \left( \frac{\partial P}{\partial r} \right) (z^2 - z h^2)/2 + 3 \rho (\dot{Q} + \pi r^2 \ddot{x}_2)(z_h^3 h^2/2 - z^4/4 - z_h^3 h^3/4)/\left(\pi r h^3\right) + 9 \rho (\frac{Q^2}{r^2} - \frac{\pi^2 r^2 \dot{x}_2^2}{2})(z^4 h^2/12 - z^5 h^2/10 + z^6/30 - z h^5/60)/\left(\pi^2 r h^6\right) + 3 \rho (\dot{Q} + \pi r^2 \dot{x}_2) \dot{x}_2 (z^4 h^2/2 - 2 z^5 h^2/5 - z^3 h^3)/3 + 2 z^6/15 + h^5 z/10)/\left(\pi r h^6\right) \]  

(7)

Substituting equation (7) into (3) and integrating,

\[ \frac{\partial P}{\partial r} = 6 \mu Q/\left(\pi r h^3\right) + 6 \mu r \dot{x}_2/h^3 + 3 \rho \dot{Q}/\left(5 \pi r h^2\right) + 3 \rho \ddot{x}_2/(5 h^2) + 27 \rho Q^2/\left(70 \pi^2 r^3 h^2\right) - 24 \rho \dot{Q} \dot{x}_2/(35 \pi r h^2) - 15 \rho \ddot{x}_2^2/(14 h^2) \]  

(8)
Disc Velocity $x_1$

\[ Q_r = \int_0^h 2\pi r^2 u^* dz \]

\[ Q_x = \pi r^2 x_1^* \]

Figure A 3.1 Flow Control Volume
Appendix A4 Calibration of Wavy Washer
Appendix A5 Calibration of Magnetic Force at Constant Disc Position

Disc Position:
- Δ - 0.4 mm
- ▼ - 0.5
- + - 0.6

Magnet Force (N)

Coil Current (A)
Appendix A6

Steady-state Error Check

The error transfer function is defined as

\[ \frac{e(s)}{V_i(s)} = \frac{1}{1 + G(s)H(s)} \]  \hspace{1cm} (1)

From equations (2.7) and (4.15),

\[ G(s) = \frac{K_i K_m K_a (K_p s + K_i)}{s(m s^2 + B s + K)(s T_c + 1)} \] \hspace{1cm} (2)

where \( K = K_s + K_h - K_x \), and

\[ H(s) = \frac{K_f}{(1 + (C_{f_2} R_{f_1} + C_{f_2} R_{f_2}) s + C_{f_1} C_{f_2} R_{f_1} R_{f_2} s^2)} \] \hspace{1cm} (3)

Substituting (2) and (3) into (1),

\[ \frac{e(s)}{V_i(s)} = \frac{1}{1 + \frac{K_i K_m K_a (K_p s + K_i)}{s(m s^2 + B s + K)(s T_c + 1)(1 + D s + E s^2)}} \] \hspace{1cm} (4)

where \( D = C_{f_2} R_{f_1} + C_{f_2} R_{f_2} \)

\( E = C_{f_1} C_{f_2} R_{f_1} R_{f_2} \).
For a step input voltage change, the output response is

\[ e(s) = \frac{V_i}{s \left( 1 + \frac{K_i \cdot K_m \cdot K_a \cdot (K_p \cdot s + K_i)}{s \cdot (m \cdot s^2 + B \cdot s + K) \cdot (s \cdot T_c + 1) \cdot (1 + D \cdot s + E \cdot s^2)} \right)} \]

Applying the final value theorem, the steady-state position error is

\[ \text{ess} = \lim_{s \to 0} s \cdot e(s) \]

\[ = \lim_{s \to 0} s \cdot \frac{V_i}{s \left( 1 + \frac{K_i \cdot K_m \cdot K_a \cdot (K_p \cdot s + K_i)}{s \cdot (m \cdot s^2 + B \cdot s + K) \cdot (s \cdot T_c + 1) \cdot (1 + D \cdot s + E \cdot s^2)} \right)} \]

\[ = 0. \]
### Technical data

**Typical long-term drift of sensor**
- up to middle of measuring range (20°C) — 0.1%/month

**Temperature drift at middle of measuring range** — 0.05%/°C

**Permissible ambient temperature**
- for sensor and cable — 20°C up to 125°C (up to 200°C for a short period)

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<th>A 7</th>
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<td>27</td>
<td>36</td>
<td>mm</td>
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(The values are typical for steel. The values for other materials may be different)
Technical data

Signal conditioning electronics I-W-A/OLIE with sensors I-W-A/A4 to A68

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<td>Power supply:</td>
<td>± 15 VDC/50 mA</td>
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<td>Dimensions:</td>
<td>European format 100 x 160 mm with front panel 7 E/3U</td>
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<tr>
<td>Output voltage:</td>
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<tr>
<td>Output current:</td>
<td>0 - 20 mA (changeover to 4 - 20 mA possible), max. load 500 Ohm</td>
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<td>1% of measuring range at maximum</td>
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<td>Initial non-linear distortion:</td>
<td>According to sensor calibration curve</td>
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<td>Resolution:</td>
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<td>Working temperature of electronics:</td>
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<td>Typ. drift (20°C) per month at centre of measuring range:</td>
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DORNIER

L-W-A/Kalibrierung

UA
10
9
8
7
6
5
4
3
2
1
0

Volt

0
0.25
0.5
0.75
12.5
125m

L-W-A/A-4 Nr. 12854
L-W-A/01.1-4 Nr. 12856
Empfindlichkeit: 18 V/min
Medium: ST 37
Datum: 12.10.86

mit 2m Kabel
Appendix A8

Calibration of the Mechanical Valve

The mechanical valve was tested under the same steady-state set-up used by the disc valves. The flow rate across the valve was measured while the upstream valve pressure was increased and decreased progressively to check for hydraulic hysteresis. Figures A8.1 and A8.2 are the results of the pressure-flow characteristics obtained from a foot valve and a piston valve respectively.

It is shown that the mechanical valves under test are too soft as compared with the specified upper force limit, which is also the minimum requirement for the mechanical valve as described in Chapter 7. This implies that if the shock absorber is to generate the same level of force as specified, a stiffer spring is required in the mechanical valve.

With the results in Figure A8.2, the pressure-flow characteristic of the combined disc valve and mechanical valve is estimated and is shown in Figure A8.3. This estimated characteristic is obtained by curve-fitting the experimental results of the piston valve and substituting it into the steady-state simulation program. In a later test on the shock absorber assembly, the result in Figure A8.3 are used to compare with the pressure drops generated across the parallel valve assembly while the shock absorber is cycled at different velocities.
Figure A8.1 Comparison of Pressure - Flow Characteristics of a Foot Valve

Figure A8.2 Comparison of Pressure - Flow Characteristics of a Piston Valve
Pressure Drop (bar)

Nozzle 2 closed

$x/TD = 0.0$

0.2

0.4

0.6

0.8

1.0

Nozzle 2 fully opened

Flow Rate (l/min)

Figure A8.3 Estimated Pressure-Flow Characteristic of a Double Disc Valve and a Mechanical Valve in parallel