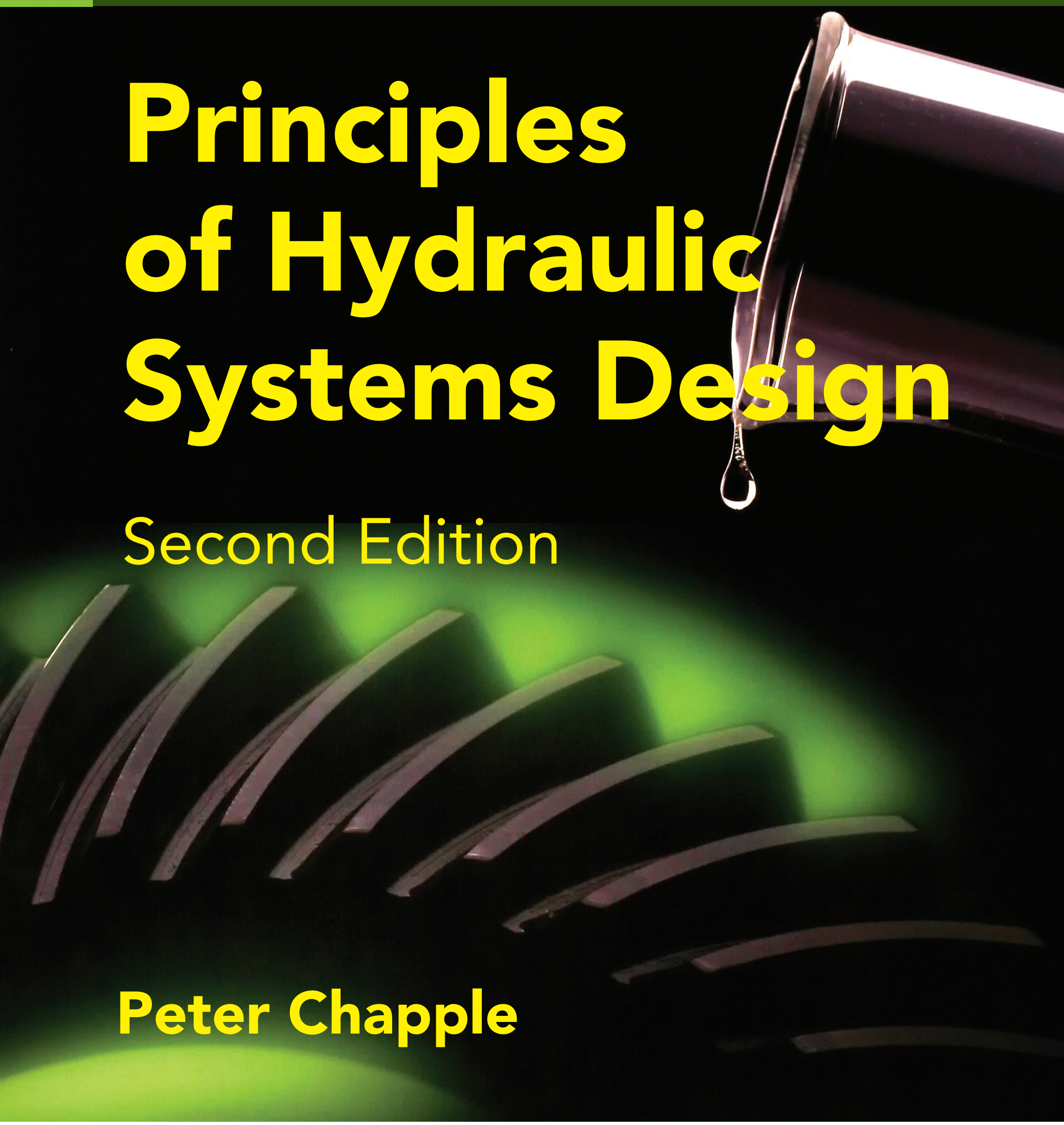
FLUID MECHANICS COLLECTION

Principles of Hydraulic Systems Design

Second Edition



Peter Chapple

MOMENTUM PRESS
ENGINEERING

PRINCIPLES OF HYDRAULIC SYSTEMS DESIGN

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Abstract

This is the second edition of the book which was published in 2002. Fluid power systems are manufactured by many organizations for a very wide range of applications, which often embody differing arrangements of components to fulfill a given task. Hydraulic components are manufactured to provide the control functions required for the operation of a wide range of systems and applications. This second edition of the book is structured so as to give an understanding of:

- The basic types of components, their operational principles and the estimation of their performance in a variety of applications. Component manufacturer diagrams are included to aid the understanding of the mechanical principles involved. This second edition has added a description of the digital control methods for independently operated valves for pumps and motors.
- A resume of the flow processes that occur in hydraulic components.
- A review of the modeling process for the efficiency of pumps and motors. This second edition includes an analysis for estimating the mechanical loss in a typical hydraulic motor.
- The way in which circuits can be arranged using available components to provide a range of functional system outputs. This includes the analysis and design of closed loop control systems and some applications.
- The analytical methods that are used in system design and the prediction of steady state and dynamic performance in a range of applications. This second edition deals more extensively with the analysis of hydraulic circuits for different types of hydrostatic power transmission systems and their application.
- This second edition also includes a description of the use of international standards in the design and management of hydraulic systems.

KEYWORDS

closed loop control systems, contamination control filters, digital displacement control of pumps and motors, flow and pressure control valves,

flow loss, fluid bulk modulus, fluid compressibility, fluid pressure loss, fluid viscosity, hydraulic circuits, hydraulic cylinders, hydraulic filters, hydraulic pumps and motors, hydraulic seals, hydrostatic transmissions, international standards, mechanical efficiency, mechanical power loss, oil coolers, oil flow process, pressurized gas accumulators, proportional valves, pump and motor displacement control, volumetric efficiency, volumetric power loss

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- Danfoss
- Bosch Rexroth
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- Sun Hydraulics
- Pall
- Parker
- Eaton
- Mactaggart, Scott
- Hallite

Introduction to Second Edition

The first edition of this book was intended to provide knowledge of hydraulic components, their operating characteristics and available circuit arrangements to assist in the design of hydraulic systems for a range of applications. For some circuits it is important to predict the dynamic performance of the overall system particularly where closed loop control is being employed. For this purpose and to provide a general understanding of how parameters vary transiently analytical methods are developed that provide techniques for studying the dynamics of systems. In this second edition these features have been retained and expanded upon so as to evaluate the performance of some hydrostatic transmission examples.

In the period since the first edition there has been a considerable expansion in the use of digital computers for control purposes which has been accompanied by developments in the instrumentation field and use of electromechanical valves. This technology often involves closed loop control methods for example, the control of motor speed in a hydrostatic transmission. This raises issues of dynamic performance and stability, areas that are expanded in this second edition.

It is useful to have an understanding of the mechanical losses in pumps and motors and a section has been added to model frictional processes in a typical hydrostatic motor which shows the effects of friction on torque and mechanical efficiency.

Independently operated valves for pumps and motors are not a new technological development but their use is becoming more apparent because of the options that are available for displacement control. The advantages of this technology are that conventional distributor valves and displacement change mechanisms are avoided which eliminates their frictional and volumetric losses and consequently results in increased efficiency particularly at reduced displacement. The design aspects of servo systems to provide this important feature are discussed.

There has been a considerable development of industrial standards particularly in the fields of the determination of dangerous failure possibilities and system safety. These standards are referred to where applicable and discussed in the final chapter on systems management. Relevant standards are listed in the Appendix.

CHAPTER 1

Hydraulic Power Transmission and Its Control

1.1 INTRODUCTION

Hydraulic fluid power is one of the oldest forms of power transmission which, despite the period of rapid growth of electric power generation, became accepted for driving a wide range of machines because of the inherent advantages that it has over other available forms of power transmission.

The increased reliability and life that resulted from the introduction of oil-based fluids and nitrile rubber sealing elements created a rapid growth in the use of fluid power transmission systems for a large variety of machine applications. Some of the advantages that hydraulic power has over other transmission mediums are summarized:

- The equipment designer is released from the dimensional limitations that are imposed by conventional gears and drive shafts.
- Stepless speed control can be obtained with relatively little increase in circuit complexity.
- The high ratio of power to mass allows fast response and a low installed weight at the point of application.
- The available output force is independent of operating speed. Stalled loads can be maintained for indefinite periods.

The introduction of electronic control into fluid power has created scope for its use in a wide range of machine applications particularly when operation by computers or programmable logic controllers (PLCs) is required.

Electronic devices have improved the accuracy of control using closed loop control techniques in many applications that have traditionally been served by hydromechanical open loop systems.

Whether or not fluid power transmission is adopted in a particular application depends on a number of features that would require consideration for a comparative study to be made if different types of power transmission are to be evaluated.

1.2 FLUID POWER SYSTEM DESIGN

There are broad categories of the types of fluid power systems in normal use for which there is a range of available components for any chosen system. The type of circuit employed often depends on company practice or user choice and, as a consequence, this often has an important influence on the components selected for the system. However, there are technical aspects that can be used to evaluate the performance of systems, which the designer needs to be aware of in order to provide some influence in the process of selecting both the type of circuit to be used and the components.

1.2.1 COMPONENT SELECTION

Circuits can be arranged in various ways using alternative components to provide a system for any given application. Additionally, different component designs are available to perform a specific function and because of this and their influence on circuit design, the component selection process does not easily lend itself to a discrete synthesized approach, as it requires knowledge of the:

- Range of hydraulic components that are available.
- Operating characteristics of the components and their use in circuits and control systems.
- Available types of hydraulic circuits.
- Analytical methods for determining the system performance to meet the machine specification.

1.2.2 CIRCUIT SELECTION

Generally speaking, the type of circuit that is chosen for a given application depends on a number of factors that include:

- First cost
- Weight
- Ease of maintenance
- Operating cost
- Machine duty cycle

1.2.3 SYSTEM DESIGN PROCESS

The major activities involved in the design process can be summarized as follows:

- Evaluate the machine specification and determine the type of hydraulic system to be used.
- Establish the types and sizes of the major hydraulic components.
- Select an appropriate design of the hydraulic circuit.
- Carry out a performance analysis of the system and determine its ability to meet the machine specification.

This process, or parts of it, may need to be repeated as the final design is evolved.

1.3 CONTENTS OF THE COURSE BOOK

Bearing in mind the foregoing comments, this book has been arranged to provide background knowledge for the design of fluid power systems. The contents include:

- Descriptions of major hydraulic components and circuits and their performance characteristics.
- Methods for analyzing the flow in pipes and components and flow forces on valves.
- The modeling of the efficiency of pumps and motors.
- Techniques for the design and analysis of control systems.
- Methods for the analysis of system performance.

This book is, therefore, aimed at providing the required background knowledge in the design of hydraulic fluid power systems and their application in a wide range of engineering equipment and machines.

CHAPTER 2

HYDROSTATIC PUMPS AND MOTORS

2.1 INTRODUCTION

Power transmission pumps in fluid power systems are usually hydrostatic or positive displacement units, which convert mechanical power into fluid power, the most common types being gear pumps, vane pumps, and piston pumps. In these pumps, fluid is transferred through the machine in discrete volumes, for example, a gear tooth cavity. The pump size and speed determine the fluid flow rate.

Hydrostatic pumps are sources of flow so that when they are connected to a hydraulic motor, the outlet pressure will rise so the flow can cause the motor to rotate against the load torque. Hydrostatic motors convert fluid power into mechanical power so rotation of the output shaft can take place against an opposing torque load. Generally speaking, pumps can be run as motors but a number of factors influence this possibility, some of which are:

- Not all pumps are reversible in the direction of rotation because of their internal and external sealing arrangements.
- Pumps are designed to operate at relatively high speeds and can be inefficient at low speeds particularly while starting.
- Motor applications often require a significant shaft side load capacity. Pump-rotating components are generally not designed to carry such shaft side loads and consequently cannot be directly coupled to the output drive where side loading exists.

This chapter is concerned with describing the operating principles of hydrostatic units, some aspects involved in their selection, and the determination and presentation of their performance characteristics.

2.2 MAJOR ASPECTS IN THE SELECTION OF PUMPS AND MOTORS

The selection of pumps is determined by a number of factors, which need to be considered by the user. These factors include:

- Cost
- Pressure ripple and noise
- Suction performance
- Contaminant sensitivity
- Speed
- Weight
- Fixed or variable displacement
- Maximum pressure and flow, or power
- Fluid type

2.3 TYPES OF PUMPS AND MOTORS

The mechanical principle that is chosen in the design of high-pressure positive displacement pumps and motors, which include those using pistons, vanes, and various gear arrangements, depends on a number of factors. These include the operating speed and pressure, the type of fluid, and the requirement for providing variable displacement control.

Pumps normally operate at a constant speed (e.g., driven by electric motor) although in some situations (e.g., those driven by an internal combustion engine, as found typically in mobile applications) the speed will vary over a small range. However, for motors it is normally required to operate at varying speeds including starting from rest (e.g., winch drives), and this aspect is reflected in the design of some available types.

Positive displacement machines are quite distinct from those using rotodynamic principles, which are often used for the transfer of fluid at relatively high flow rate and low pressures. Positive displacement units operate at relatively low flow rates and high pressure and normally can only be used with fluids having good lubricating properties. However, there are machines that can be used with fire-resistant fluids and pure water.

2.3.1 FIXED DISPLACEMENT UNITS

2.3.1.1 External Gear Pumps and Motors

In many applications, for operation at pressures up to 250 bar, external gear pumps and motors are used extensively because of their simplicity, low cost, good suction performance, low contamination sensitivity, and relatively low weight. In applications requiring low noise, vane or internal gear pumps are often used.

Essentially, the unit consists of two meshing gear pinions, mounted in bearings and contained in a housing or body as shown in Figure 2.1. As the pinions are rotated, oil is trapped in the spaces between the gear teeth and the housing and carried around from the pump inlet to its outlet port when the trapped volume is discharged by the action of the gears meshing together.

Torque is required at the input shaft at a level dependent on the outlet pressure acting on the gear teeth. When supplied with high-pressure flow, the unit acts as a motor by providing torque to drive the load on the output shaft.

Some of the outlet fluid is transferred back to the low pressure side by way of small leakage flows through the:

- Clearance space between the teeth and the housing.
- Shaft bearing clearances.
- Clearance between the gear faces and the side plates in the housing. Most gear units have pressure loaded side plates to minimize this leakage.

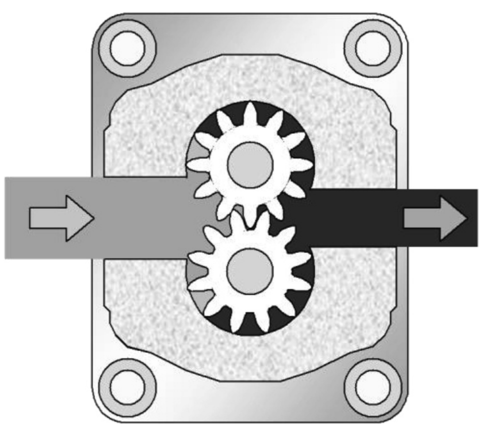

Figure 2.1. External gear pump and motor.
Source: Courtesy Eaton.

The design of the unit is aimed to minimize flow losses as they reduce its efficiency, particularly when using fluids of low viscosity such as with some water-based fluids**.** The geometric capacity, or displacement, cannot be varied so their displacement is fixed. For a given gear form, the manufacturer can produce pumps of different displacements by using different gear widths.

Standard types operate at speeds of 1,000 to 3,000 RPM and at pressures up to 250 bar, but higher speeds and pressures are available. Powers range from one to over 100 kW. The efficiency of gear units has been raised during recent years, with peak overall efficiencies of 90 percent or above.

2.3.1.2 Internal Gear Pumps

Internal gear pumps, as shown in Figure 2.2, have an internal gear driven by the input shaft and an external gear, which rotates around its own center and driven by the internal gear. By means of the separator element, both gears transmit fluid from the pump inlet to the outlet. This pump creates a low noise level that favors it for some applications although its pressure capability is about the same as that of the external gear pump.

2.3.1.3 Vane Pumps and Motors

The vane pump and motor consists of a rotor, carrying a number of sliding vanes, rotating in a circular housing. With the rotor being eccentric to the casing, oil is transmitted in the vane spaces across the pump from the suction to the discharge port.

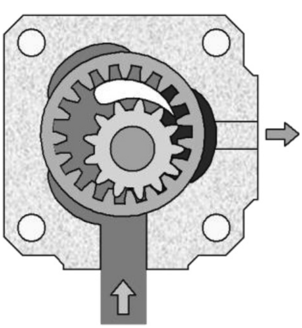

Figure 2.2. Internal gear pump.
Source: Courtesy Eaton.

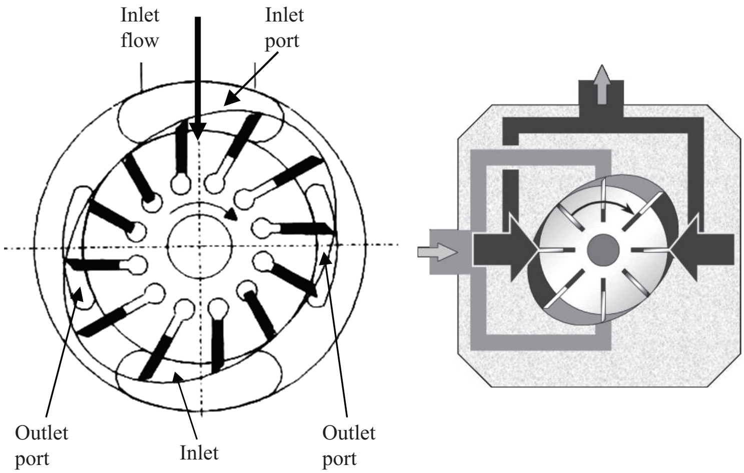

Figure 2.3. Balanced vane pump.
Source: Courtesy Eaton.

The vanes are acted on by centrifugal force when the unit is rotating, but in order to reduce leakage at the tips it is common practice to pressure load them (by supplying discharge pressure to the base of the vane slots). In some motors, the vanes have springs to force them against the housing which provides a force in addition to that created by the pressure. As with the gear unit, control of the clearances at the sides of the rotor assembly is most important.

The balanced design in Figure 2.3 eliminates pressure loading on the bearings and uses an "elliptical" vane track with the vanes moving in and out twice each revolution. There are diametrically opposed suction ports and discharge ports as shown in Figure 2.3, and these are connected together in the cast body. This pump is only available as fixed displacement.

Vane pumps are inherently more complex than gear pumps, they contain a greater number of components and are, therefore, more expensive. However, vane pumps operate at much lower noise levels than gear pumps and their cost can be offset against their good serviceability, which is not available with gear-type pumps.

2.4 VARIABLE DISPLACEMENT UNIT

2.4.1 VANE PUMPS

Variable displacement vane pumps are available as shown in Figure 2.4, where the center of the rotating vane block can be moved in relation to the

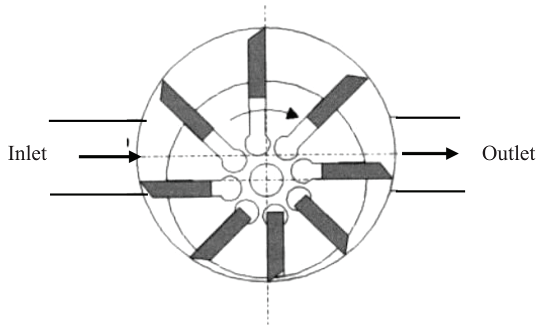

Figure 2.4. Variable displacement vane pump.

center of the housing. Unlike the balanced vane unit of Figure 2.3, these are single acting and, as a consequence, there is an unbalanced pressure force on the rotor so that the bearing size has to be increased in order to obtain adequate life.

2.4.2 PISTON PUMPS AND MOTORS

Piston units operate at higher efficiencies than gear and vane units and are used for high-pressure applications with hydraulic oil or fire-resistant fluids. Several types of piston pumps are available that use different design approaches, and these include those having axial and radial piston arrangements.

The majority of piston pumps and motors are of the axial variety, in which several cylinders are grouped in a block around a main axis with their axes parallel as shown in Figure 2.5, which has variable displacement capability. The pressure force from the pistons is transferred to the angled swash plate by lubricated slippers that are mounted onto the pistons with a ball coupling. Rotation of the cylinder block causes the pistons to oscillate in their cylinders by the action of the swash plate, which provides the conversion between the piston pressure force and shaft torque.

Varying the swash plate angle allows the displacement to be changed over the full range from zero to maximum. The swash plate angular position can be arranged to vary either side of the zero displacement position so that flow reversal is obtained. This is referred to as overcenter control.

The piston cylinders are alternately connected to the high- and low-pressure connections by a plate valve between the cylinder block and the port connection housing. Figure 2.6 shows a pump port plate with the circular slots, which are connected to the pump inlet and outlet ports.

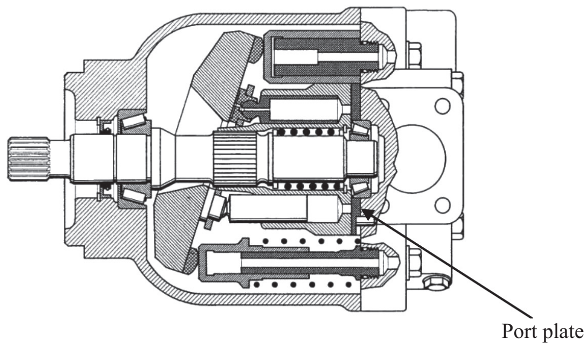

Figure 2.5. Axial piston variable displacement pump and motor. *Source*: Courtesy Eaton.

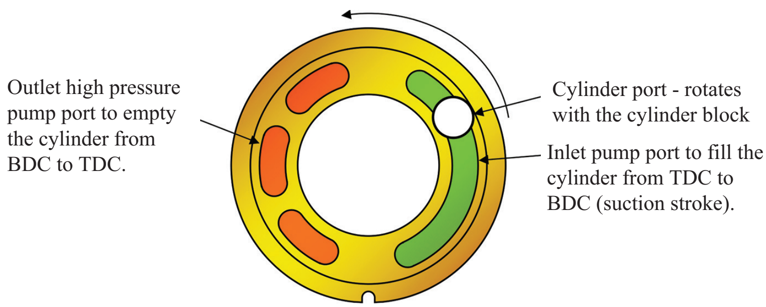

Figure 2.6. Diagram of a pump port plate.

The port plate installed into a swash plate pump is shown in Figure 2.5 between the rotating cylinder block and the stationary port block. The cylinder block is forced against the port plate to minimize leakage. The high pressure slot in the port plate is shown in Figure 2.6 to be in three parts for improved strength and stiffness, but in motors that can rotate in either direction and with high pressures on either of the two ports, both sets of slots have an identical arrangement.

Figure 2.7 shows a fixed displacement bent axis type of axial piston unit whereby the ball-ended pistons are located in the output shaft. During rotation of the shaft there will be a rotating sliding action in the ball joint, and possibly, between the piston and the cylinder. Each cylinder is connected successively to the high- and low-pressure ports by a similar valve to that used in the swash plate units. For this motor, the port plate surface is spherical in shape, which provides a location for the rotating cylinder block.

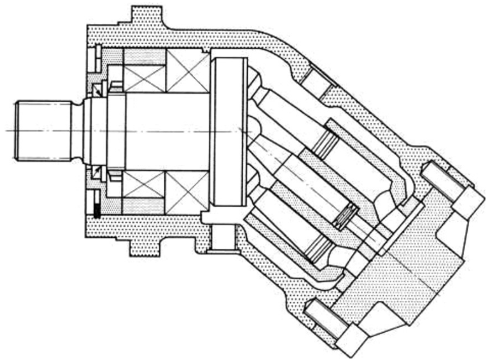

Figure 2.7. Bent axis type axial piston motor.
Source: Courtesy Bosch Rexroth.

In variable displacement units, a mechanism is used to vary the tilt angle of the cylinder block from zero to maximum that, if required, can provide overcenter operatio n to give reverse flow.

There are two types of radial piston pumps; those in which the cylinder block rotates about a stationary pintle valve and those with a stationary cylinder block in which the pistons are operated by a rotating eccentric or cam.

Many standard piston units of recent design operate at pressures of up to 450 bar. A wide range of types are available up to powers of 100 kW, although a number of manufacturers provide units having powers up to 300 kW with some available at powers of 1,000 kW. Peak overall efficiencies in the region of 90 percent are usually obtained. The price of piston units varies from manufacturer to manufacturer but may be as much as ten times the price of a gear pump of similar power.

Some pumps cannot draw the inlet fluid directly from the reservoir and may require boosting from a separate pump, often of the external gear type, that can accept low inlet pressures. However, for open loop circuits, variants are available that do not require separate boosting of the inlet, which can be connected directly to the reservoir. These aspects also apply to motors that operate as pumps in hydrostatic systems when regeneration occurs (e.g., winch and vehicle drive applications).

Variable displacement pumps provide a range of control methods, which include pressure compensation, load-sensing and torque, or power, limiting devices. Pump displacement controls incorporating electrohydraulic valves are also available.

In addition to their use to control outlet flow, variable displacement pumps can provide considerable increase in overall system efficiency with the additional benefits of reduced heat generation and operating cost. This reduction in operating cost can show an overall reduction in the total lifetime cost of the machine.

2.5 EQUATIONS FOR PUMPS AND MOTORS

2.5.1 FLOW AND SPEED RELATIONSHIP

For the ideal machine with no leakage, the displacement of the machine, D, and its speed of rotation, ω, determines the flow rate Q.

Thus
$$Q = D\omega \tag{2.1}$$

where

D is the volumetric displacement [$m^3\ rad^{-1}$]
ω is the rotational speed [$rad\ sec^{-1}$]

For pumps that are driven by electric motors, the speed is often constant. However for motors, the speed depends on the level of the supplied flow:

Thus
$$\omega = \frac{Q}{D} \tag{2.2}$$

2.5.1.1 Volumetric Efficiency

The internal flow leakage in pumps and motors affects the relationship between flow and speed and is taken into account by the use of the volumetric efficiency (η_v).

Thus for pumps, Equation 2.1 becomes

$$Q = \eta_v \omega D \tag{2.3}$$

And for motors Equation 2.2 becomes

$$\omega = \eta_v \frac{Q}{D} \tag{2.4}$$

The volumetric efficiency varies with the fluid viscosity, pressure, and rotating speed as discussed in more detail in Chapter 8. Manufacturers will usually give values for the volumetric efficiency for operation under specified conditions.

2.5.2 TORQUE AND PRESSURE RELATIONSHIP

For the ideal machine, the mechanical power is entirely converted to fluid power:

$$\text{Power} = T\omega = PQ \tag{2.5}$$

where

T is the torque [Nm]
P is the differential pressure [N m^{-2}]

From Equation 2.5 we get

$$T = \frac{QP}{\omega} \text{ which from Equation 2.2 gives } T = PD \tag{2.6}$$

Thus the ideal torque is proportional to the pressure for a given displacement. In a pump, this is the input torque required from the prime mover, and for a motor, it is the output torque available from the motor shaft.

2.5.2.1 Mechanical Efficiency

The presence of friction between the moving parts creates mechanical losses that are represented by the mechanical efficiency (η_m). Thus:

For pumps, the required input torque is given by

$$T = \frac{PD}{\eta_m} \tag{2.7}$$

And for motors, the output torque is given by

$$T = \eta_m PD \tag{2.8}$$

The mechanical efficiency, as for the volumetric efficiency, will vary with the fluid viscosity, pressure, and rotating speed as discussed in more detail in Chapter 8.

The power input, H, to a pump is

$$H = \frac{PQ}{\eta_m \eta_v} \tag{2.9}$$

The power output from a motor is

$$H = \eta_m \eta_v PQ \tag{2.10}$$

The total efficiency of both units is therefore

$$\eta_T = \eta_m \eta_v$$

Figures 2.8 and 2.9 show how the measured performance of pumps and motors is presented for use with a particular fluid at a particular viscosity. For the pump, it can be seen that the flow output reduces with the output pressure at constant speed because of the effect of the increasing leakage flow loss.

For the motor, the output torque varies with increasing speed at constant pressure as a result of the variation in the mechanical efficiency. The theoretical analysis given in Chapter 8 shows how the efficiencies are related to the system parameters, which enables the performance for operation under other conditions to be predicted.

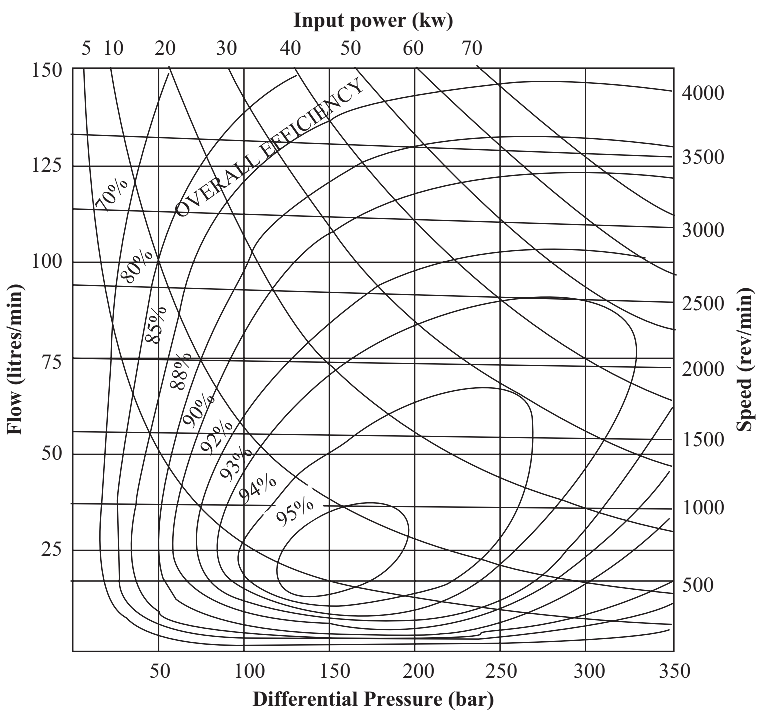

Figure 2.8. Pump performance characteristics.

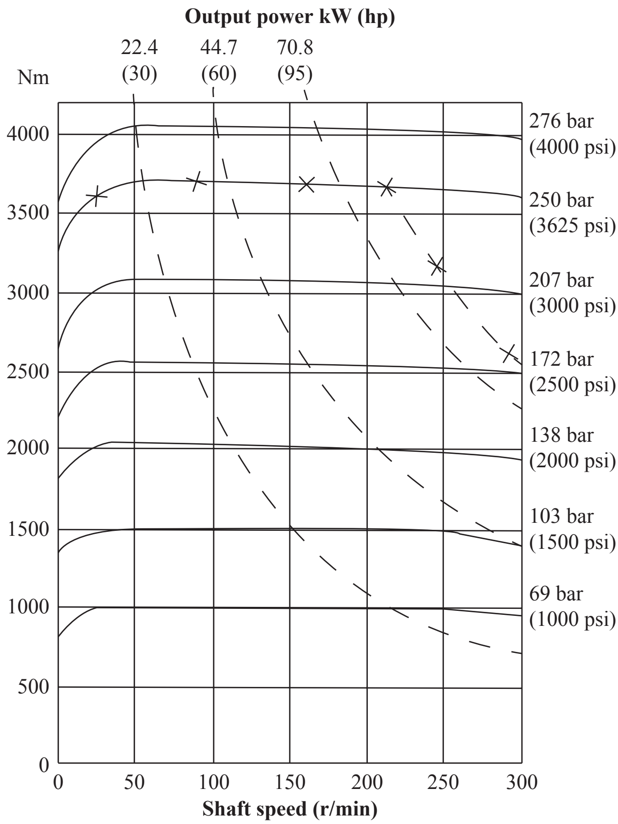

Figure 2.9. Motor performance characteristics.

2.5.3 PUMP SELECTION PARAMETERS

The process involved in the selection of a suitable pump for a given application depends on many parameters, some of which were summarized in Section 2.2. As a consequence, a generalization is not possible but some major features can be identified as shown in Table 2.1.

In most cases, manufacturer preference and the experience of the machine designer usually dictate the type of pump that is used in applications, but in some circumstances, it may be necessary to evaluate different types of pumps in new applications or where significant changes are required.

2.6 LOW SPEED MOTORS

As described in the Introduction, in principle, most pumps can be operated as motors. However, pumps are low torque high-speed devices and

Table 2.1. Comparison of pump types

Gear pump (fixed displacement)	**External type**	**Internal type**	**Other features**
	• Low cost • Low contaminant sensitivity • Compact, low weight • Good suction performance • 250 cm^3 rev^{-1}, 250 bar	• Low noise • Low contaminant sensitivity • 250 cm^3 rev^{-1}, 250 bar	In-line assembly for multipump units
Vane	**Fixed displacement**	**Variable displacement**	
	• Low noise • Good serviceability • 200 cm^3 rev^{-1}, 280 bar	• Low noise • Low cost • Good serviceability • Displacement controls • 100 cm^3 rev^{-1}, 160 bar	In-line assembly for multipump units
Piston	**Fixed and variable displacement**		
	• High efficiency • Good serviceability • Wide range of displacement controls • Up to 1,000 cm^3 rev^{-1}, 350/400 bar		• Integral boost pump and multipump assemblies (not bent axis) • Can use most types in hydrostatic transmissions

generally require the use of reduction gearboxes in order to provide an output drive at increased torque and reduced speed. The availability of low cost multistage gearboxes enables a wide range of ratios to be offered to suit different applications.

In many applications, the drive operating speed (e.g., winches, vehicles) is variable in the range from zero to a few hundred revolutions per minute. For this type of application, specialist low speed motors are available that have higher operating efficiencies at low speeds. In many cases, a low speed motor can be selected that avoids the necessity of employing a gearbox and has sufficient bearing capacity to support side loads on the output shaft.

The range of available low speed motors encompasses a number of different design concepts that include radial piston eccentric and cam, axial piston, and those using the Gerotor principle. The motor displacement for a given application is dependent on the required torque and operating pressure. However, the type of motor to be used is a function of a number of variables that include maximum speed, torque and pressure, shaft side load, duty cycle, fixed or variable displacement, weight, and cost.

The selection criteria for hydrostatic motors are different from those that apply to pumps because the required output parameter is torque that needs to be available over a wide speed range. This, in most cases, extends from zero and may require significant operating periods at speeds below 10 rev/min. Motors are often also required to operate in both directions.

2.6.1 TYPES OF LOW SPEED MOTORS

2.6.1.1 Radial Piston Motors

Figures 2.10 and 2.11 show two different types of radial piston motors, that of Figure 2.10 using an eccentric that gives one piston stroke per revolution while the cam unit of Figure 2.11 creates several piston strokes per revolution.

The eccentric motor shown is typical of several designs that are available for converting the piston force into output torque at the drive shaft. In the motor shown, the valve successively connects each cylinder to the supply and returns for successive half revolutions of the shaft.

These motors generally operate at a maximum continuous pressure of 250/300 bar with displacements up to 30 L/rev.

The motor displacement can be altered between two levels by pressurizing pistons that control the position of the eccentric (not shown in Figure 2.10). In some motors, the displacement can be controlled continuously in a closed loop control of motor speed, inlet pressure, or motor flow.

The particular type of radial piston cam motor shown in Figure 2.11 operates by transferring the pressure force on the piston, which is directed radially outwards, onto the cam by the use of rolling element bearings attached to the piston. Here the distributor valve plate, shown in the figure, connects each piston to the high-pressure port when the piston is moving outwards thus creating a torque on the cylindrical cam. When the piston is

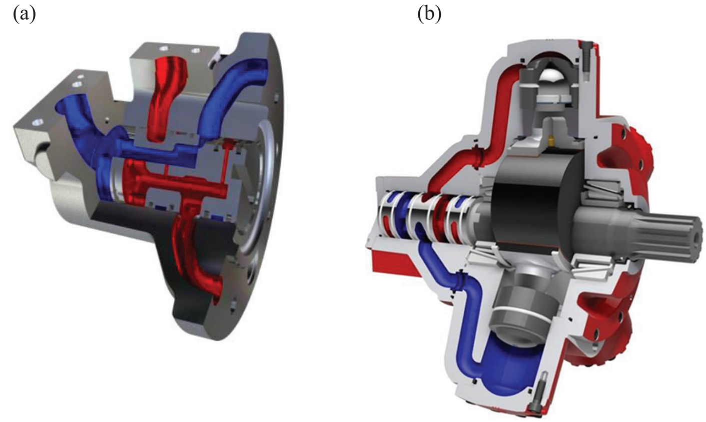

Figure 2.10. Radial piston eccentric motor (a) valve housing (b) complete motor assembly.
Source: Courtesy Kawasaki Precision machinery.

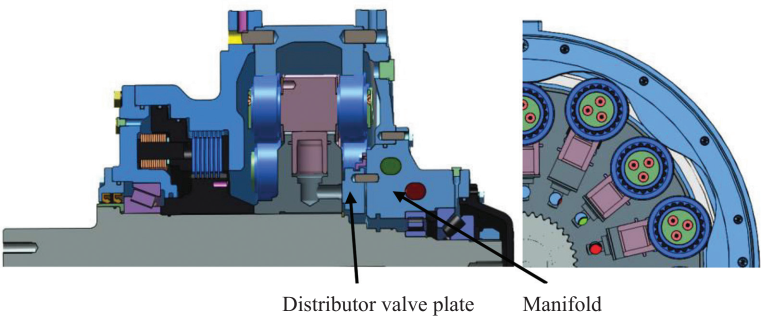

Figure 2.11. Cam-type hydraulic motor.
Source: Courtesy MacTaggart Scott.

moving inwards the valve connects the cylinder to the low-pressure port thus allowing fluid to be passed back to the return line.

Radial piston cam motors generally operate at continuous pressures up to 350 bar with displacements up to, and beyond, 250 L/rev. Some versions can be operated with two selectable displacements by short-circuiting some of the cylinders to low pressure. Cam motors can be arranged to provide low levels of torque ripple and have a high ratio of output torque to the motor mass (see Figure 2.14).

In the Gerotor, or orbit-type motor, as shown in Figure 2.12, the rotor center rotates in an orbit as the rotor rolls in contact with the lobes created by the fixed rollers attached to the outer housing. This action causes the

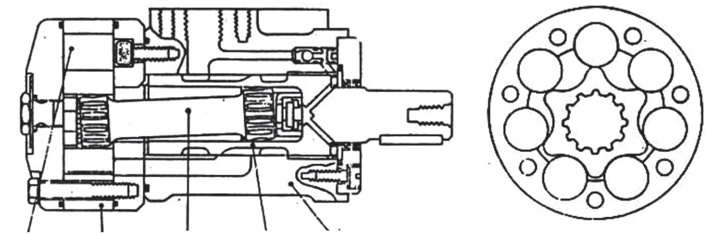

Figure 2.12. Gerotor, or orbit-type low speed high torque motor.
Source: Courtesy Danfoss.

volumes trapped between the several contacts to vary with rotation of the shaft.

The rotating valve connects increasing volumes to the high-pressure port and reducing volumes to the low-pressure port, creating a torque on the output shaft and a continuous rotation in accordance with the supply flow.

Orbit motors have a high ratio of torque to motor mass but operate at lower pressures than the radial piston motors and are usually employed for medium pressure, low power applications. Typically motor displacements are up to 1,600 cm^3/rev at pressures up to 175/200 bar.

2.7 DIGITAL VALVES FOR PUMPS AND MOTORS

A new technology [2] has created a line of research into the development of independently operated valves for radial piston machines, as shown in Figure 2.13. This motor is designed to carry out tests on the valves and their methods of control. The valves are switched on and off to connect the pump and motor cylinder alternately to either low or high pressure.

In order to consider how the valves might be operated it is necessary to examine the forces that will act on the valves during normal operation. This is described in Sections 2.7.1 and 2.7.2.

2.7.1 VALVE OPENING TO HIGH PRESSURE SUPPLY

For a motor, it is required for flow to pass from the high-pressure supply to fill the cylinder when the shaft rotates from the top dead center (TDC) to the bottom dead center (BDC) positions. Referring to Figure 2.13 it can

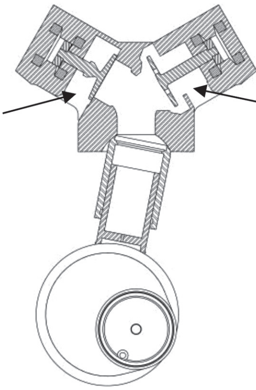

Figure 2.13. Digital control valves for a radial piston motor.
Source: Courtesy Diinef.

be seen that there will be a pressure force acting to close the high-pressure valve (HPV). Thus, to open the valve a force of this magnitude has to be provided in the opposite direction.

Now the movement of the piston, as it approaches TDC with both valves closed, will compress the fluid thus creating a pressure force in the direction to open the HPV. This action could be utilized so that after the valve is opened in this way the valve could be kept in the open position by, for example, a solenoid or hydraulic latch that is operated by a signal from a shaft position sensor. Thus, as the piston moves from TDC to BDC the cylinder will be filled with high-pressure oil from the supply through the opened valve.

Alternatively, such a force on the valve could be obtained, for example, from a direct acting solenoid or from a hydraulic actuator.

2.7.2 VALVE OPENING TO LOW PRESSURE

Following from Section 2.7.1 when the shaft reaches the BDC position it is required to close the HPV and open the low-pressure valve (LPV). If the HPV is closed just before BDC the cylinder pressure will reduce and when this pressure is lower than that at the low-pressure port the LPV will be opened when it can be latched into the open position.

For the unit to act as a pump a similar analysis can be followed and the design of an appropriate system could provide the latching cycle as described in this analysis.

2.7.3 BENEFITS OF USING DIGITAL VALVES

Pumps and motors using separate valves to provide the flow function are sometimes referred to as digital pumps or motors and examples of their use in systems are described in [3]. The major benefits that are provided by using digital valves in place of the mechanical flow distribution and variable displacement control systems that are used in the various designs of "conventional" pumps and motors can be summarized as below.

2.7.3.1 Improved Part Load Efficiency

In any pump or motor when operating at say, constant pressure and speed, the magnitude of the losses is substantially unaffected by a change in displacement. This means that when reducing the displacement from maximum, the losses, as a proportion of the total output, will increase thus reducing the efficiency.

For example, for a motor with efficiency of 95 percent at the maximum displacement, if the displacement is reduced to 50 percent the efficiency reduces to 90 percent in the event that the losses remain the same. Similarly for a displacement of 25 percent, the efficiency will be 80 percent. In a conventional machine, there are friction and leakage losses in the mechanical distribution valve, which can create a large proportion of the losses. In the digital unit there is a pressure loss in the inlet and outlet valves but because there is adequate space around the periphery of the cylinders of a radial piston unit, there is no restriction on the size of the valves. Thus, the loss in a digital unit is largely due to the torque-producing mechanism, which, can be a relatively small proportion of the total loss.

This indicates an advantage of digitally controlled valves, which can provide greater efficiency due to the elimination of the mechanical flow distribution valve used in conventional motors.

2.7.3.2 Ability to Provide a Wide Range of Displacement Control Functions

The operation of digital valves has the flexibility to provide a displacement range that is not constrained by the limitations that exist in the mechanical displacement in conventional motors.

In multicylinder hydraulic motors, a reduction in displacement can be achieved by switching off a cylinder by connecting it to low pressure all the time. The operating procedure described in Sections 2.7.1 and 2.7.3 suggests that it is only possible for a valve to be operated at either TDC or BDC.

However, there are alternative possibilities that will allow variable displacement, so that:

i. During the motoring stroke the piston moves towards the eccentric when the HPV is open and the LPV is closed, the latch on the HPV can be released. This will cause the HPV to close and following the reduction in the cylinder pressure releasing the LPV latch will cause it to open.
ii. During the exhaust stroke with the piston moving away from the eccentric releasing the latch on the LPV will cause it to close and following the increase in pressure to a sufficient level will cause the HPV to open.

The net effect of these switching operations is to reduce the proportion of the stroke that is at high pressure thus reducing the torque output from the motor.

An alternative operation of the valves, which allows switching of the valves at different shaft positions by using a pressure switching circuit, is discussed in Section 7.7.6.

2.8 SOME GENERAL CONSIDERATIONS

To avoid motor malfunction, the system must be capable of working with the range of loads and speeds, both steady and transient, that the application demands. Some general points to be considered are as follows:

- Displacement control—The operation of motor displacement variation has to be carefully considered for a given application as the pressure and speed levels are determined by the ratios of $\frac{T}{D}$ and $\frac{Q}{D}$ so that for given values of torque and flow, reducing the displacement increases both the pressure and the speed.
- Motor creep—A motor under load (e.g., winch) with closed ports will rotate slowly backwards (creep) because of internal leakage. For this reason, braking systems are required to prevent this motion, for example, excavator slew drives, winches.
- Torque ripple—This refers to variations in the shaft torque while rotating. It is only of significance at speeds where the ripple

frequency is in the range of zero to slightly higher than the natural frequency of the hydraulic system.

- Backpressure—Some motors cannot operate for significant periods of time with high backpressure (i.e., working as a pump in meter-out mode).
- Boost pressure—Some motors and pumps require a certain level of boost pressure in order to keep the mechanical parts together when operating at high speed. This is particularly important, for example, during over-running conditions (e.g., winch systems) when cavitation can occur.
- Motor drain—The external drain from the motor should allow completely free flow in order to avoid excessive pressure levels occurring in the crankcase or sump.
- With the availability of variable speed electric motors having good efficiency levels there is a growing trend for their use in providing a variable output flow from fixed displacement pumps. The control systems of the electric drives can provide a basic simple flow control or closed loop control of various output parameters (e.g., speed, pressure).

2.9 COMPARISON OF MOTOR PERFORMANCE CHARACTERISTICS

As can be seen from the foregoing discussion, there is a large variety of motor types that can be used for any given application that includes high speed low torque (HSLT) and low speed high torque (LSHT) units. Table 2.2 gives a summary of the nominal maximum displacement and torque for these types of motors.

The mainstream of pumps and motors are in the displacement range up to 1 L/rev. High speed units tend to be of the axial piston type (swash plate and bent axis) whose maximum speeds vary with displacement as shown in Figure 2.14. Also shown in Figure 2.14 is the performance of radial piston units, which in the main are LSHT motors that as seen in Table 2.2 have displacements in the range up to 380 L/rev.

For units of the same generic design where the principle dimensions have the same scale factor, a_s applied to obtain different displacements it is seen that if the limit to performance and life is, say, a maximum sliding velocity U then it can be stated:

Table 2.2. Motor data

Type	Max displacement ($cm^3\ rev^{-1}$)	Max continuous pressure (bar)	Max torque (Nm)
External gear	250	250	1,000
Vane	350	250	1,400
Axial piston swash plate	1,000	350	5,570
Orbit	1,600	210	4,500
Radial piston eccentric	35,000	300	170,000
Radial piston cam	380,000	350	2×10^6

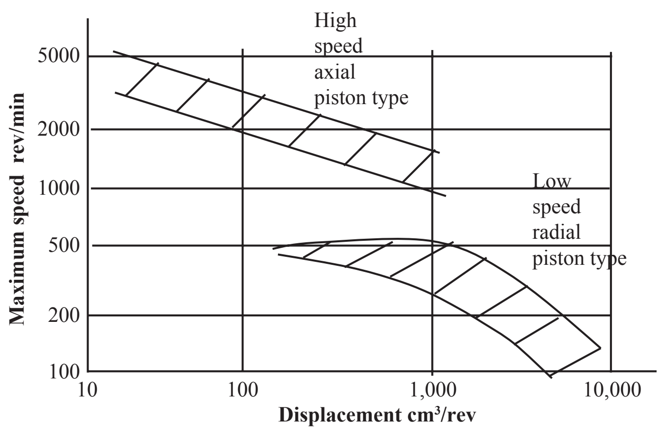

Figure 2.14. Typical maximum speeds for motors of various types.

$$U \propto \omega R \propto \omega a_S$$

where

R is a dimension that relates to displacement (e.g., piston diameter) and ω is the rotational speed.

Now as the displacement, *D*, varies with a_s^3 then for a fixed maximum velocity *U* it can be stated from the foregoing:

$$\omega a_S = \omega D^{1/3} \propto U = \text{constant}\,(C) \tag{2.11}$$

This variation in the rotational speed of the HSLT units with displacement, shown in Figure 2.14 for a range of such motors and pumps, and

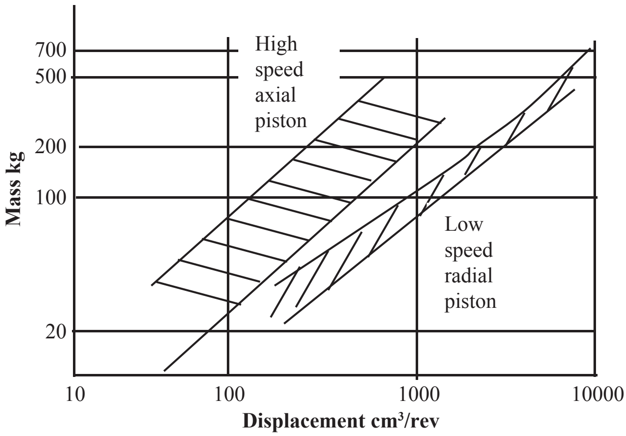

Figure 2.15. Typical mass values for a range of motor types.

the slope of the variation is seen to generally follow the relationship of Equation 2.11.

High speed units can, in general, be used for low speed drives by using a gear reducer ratio of between 7 and 20:1, which will provide HSLT motors with torque levels that are similar to those obtained from LSHT motors. This can be provided by a single-stage epicyclic gearbox (max. ratio of 7:1) and a single ratio spur gear set or a two-stage epicyclic gear unit or a combination of both.

For the LSHT types of motor, which in the main are a range of radial piston units this speed relationship is quite different.

From a power perspective point of view the maximum power that can be obtained currently from an HSLT unit would be in the region of 800 to 900 kW (1 L/rev operating at 1,500 rev/min) whereas the same power from an LSHT unit would require, for example, a radial piston cam motor of around 50 L/rev operating at 30 rev/min. However, the selection of an appropriate unit for a given application depends on a number of parameters such as speed, life and reliability, size, shape, weight, and so on.

The mass of a combined drive unit can be generally similar to those of LSHT motors. The space envelope of these units is, however, generally longer and smaller in diameter than that of radial piston motors and may be a disadvantage in some applications such as vehicle drives and some industrial systems. However, for winch drives the gearbox can be installed inside the winch drum, which provides an optimal solution to system design in some applications.

When compared directly on the basis of displacement LSHT motors generally are of lower mass, as shown in Figure 2.15, thus giving higher values of specific torque. Typically, this parameter ranges from 5 to 20 Nm/kg for HSLT motors and 40 to 200 Nm/kg for LSHT motors. However, LSHT motors generally have lower values of specific power with values up to 3 kW/kg when compared to 8 kW/kg for HSLT motors.

There are a wide range of motor drive units from which to select for a given application but the choice may require the consideration of cost (capital and operating), machine design aspects such as its control, servicing, and maintenance, and user preference which is often based on previous experience.

SUMMARY

There is a wide range of hydrostatic pumps and motors available in the market and the purpose of this chapter is to describe the operating principles and features of the most commonly used types. The formulae that are used for determining the performance of pumps and motors are presented and some of the major parameters that can be used as a basis for comparison are outlined as a background for the selection process. However, because of the wide variety of the types of units that are available it is impossible to generalize on the selection process in any given application.

Commonly, machine builders and users have preferences for particular types of pumps and motors that are based on experience with particular applications, which are determined by factors such as the system function, its control, servicing aspects, environmental features, life expectancy, duty cycle, and type of fluid to be used. The designer needs to be aware of the relative performance of the different types and how this knowledge can be utilized in the selection process to suit a particular application.

CHAPTER 3

HYDRAULIC CYLINDERS

3.1 INTRODUCTION

This chapter is concerned with:

- The construction, mounting methods, and cushioning of hydraulic cylinders.
- The construction of the major types of rotary actuators and the sealing methods.

3.2 HYDRAULIC CYLINDERS (LINEAR ACTUATORS)

Hydraulic cylinders convert flow into linear movement and pressure into force by employing a piston that slides inside a cylinder. The construction of a typical double-acting actuator can be seen in Figure 3.1 that shows the use of appropriate sliding seals for the piston and rod components. The double-acting feature to provide operation in both directions is not always required as in some applications (e.g., fork lift truck systems) the force from the load is used for retraction.

Actuators are also available that have a rod at both ends so that the piston areas may be equal in both directions of movement.

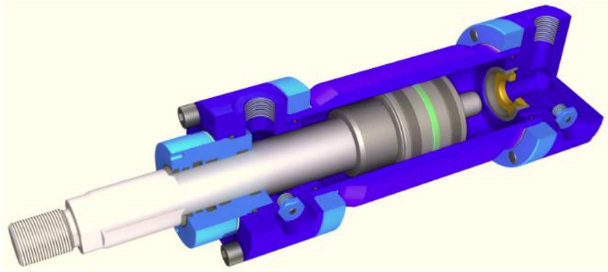

Figure 3.1. Typical double-acting actuator.
Source: Courtesy Parker.

3.3 PRINCIPLE FEATURES

3.3.1 END COVERS

The basic methods of attaching the end covers, or caps, to the cylinder of hydraulic actuators include:

a. Screwed to the cylinder
b. Tie rods as shown in Figure 3.2
c. Welded
d. Mill Type cylinders where the end caps are bolted to a flange on the barrel commonly used for heavy-duty industrial applications such as steelworks
e. A combination of (a) and (d) as in Figure 3.1

3.3.2 MOUNTING METHODS

A wide variety of actuators and rod end-mounting methods are available to suit the requirements of different applications.

Figure 3.2 shows a flange mounting that is part of the front end cap for locating the actuator rigidly to the machine frame.

The trunnion mounting shown in Figure 3.3a provides a pivoting action in one plane whereas the mounting shown in Figure 3.3b uses a spherical bearing, which allows pivotal movements in any plane. For some position transducers, it is necessary to drill a hole into the piston rod.

As discussed, the mounting style used has a significant influence on the actuator strength, and the various alternative methods are shown in Figures 3.4a–d.

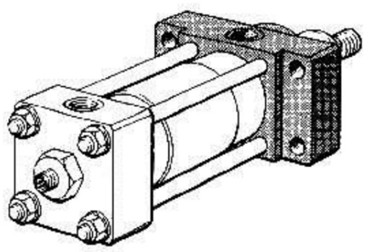

Figure 3.2. Actuator of tie rod construction.
Source: Courtesy Bosch Rexroth.

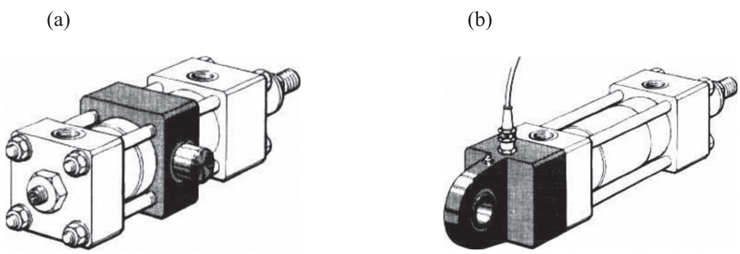

Figure 3.3. Actuator mountings (a) with position transducer (b).
Source: Courtesy Bosch Rexroth.

For working against pushing load forces, the actuator acts as a strut for which the Euler failure criteria are applied according to the method of mounting the strut. The Euler buckling load, F_E, is given by

$$F_E = \frac{S_F \pi^2 EI}{L^2}$$

where

E = Young's modulus

I = Second moment of area $= \dfrac{\pi d^4}{64}$

d = Road diameter

L = Rod length

S_F = Actuator strength factor

Thus, it is seen that for a given strength factor the buckling load varies proportionally with the fourth power of diameter and inversely with the length squared.

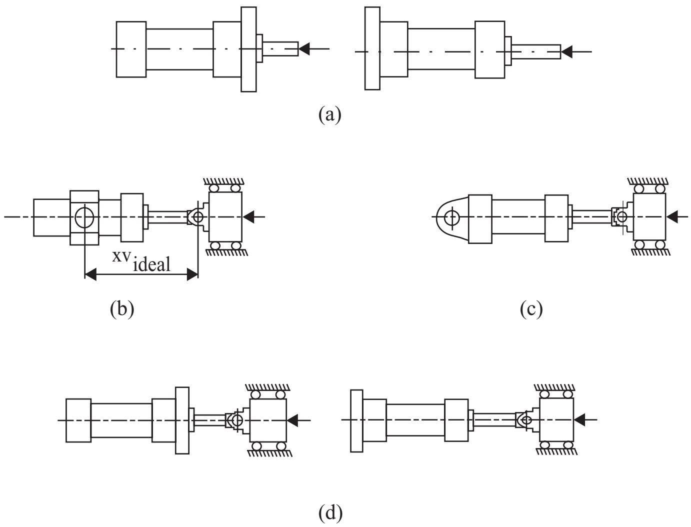

Figure 3.4. Actuator mounting styles (a) Actuator flange mounted to front or rear end cover with the rod end unguided (b) Actuator trunnion mounted with the rod end guided (c) Spherical coupling with the rod end guided (d) Actuator flange mounted with the rod end guided.
Source: Courtesy Bosch Rexroth.

The values for the strength factor, S_F, that apply to the mounting styles in Figure 3.4 are:

i. Fixed actuator mounting with unconstrained rod end (as a) $S_F = 0.25$
ii. Actuator and rod attached by free pivots but with constrained rod end (as c) $S_F = 1$
iii. Fixed actuator mounting with constrained rod end (as d) $S_F = 2$

Actuator manufacturers usually give the maximum capability of actuators with the mounting style in terms of maximum extension at a given actuator piston pressure.

A typical example is given in Table 3.1 for an actuator of 50 mm diameter at 100 bar pressure.

3.3.3 SEALS

The cutaway drawing of the cylinder in Figure 3.5 shows the position of typical sealing components. The function of the bearings is to guide the piston through the cylinder bore and the rod through the gland. They also carry any side loads between the moving parts of the cylinder. The rod

Table 3.1. Maximum piston rod extension (refer to Figure 3.4)

	Maximum piston rod extension (mm)			
d (mm)	Case a	Case b	Case c	Case d
28	390	610	500	1,260
36	690	1,120	730	1,690

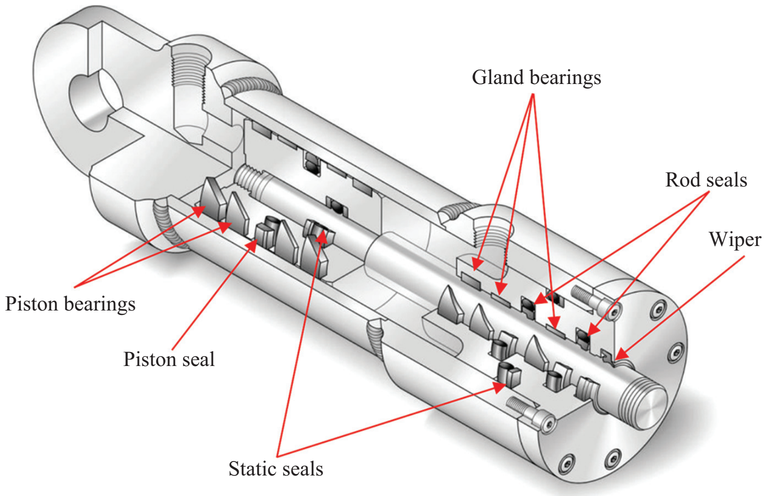

Figure 3.5. Section view of a typical cylinder showing the various seals. *Source*: Courtesy Hallite.

(or gland) seals retain the hydraulic fluid within the cylinder and the main function of the wiper is to prevent dirt and contamination from entering the cylinder. Some wiper profiles have also been developed to provide a secondary gland sealing function. The piston seal shown in the drawing is double acting—it separates the fluid in the full bore and annulus cylinder chambers. The static seals prevent leakage through the gaps between the rod and the piston and the outside of the gland and the cylinder barrel.

Many different reciprocating seal profiles have been developed over the years [4] for different cylinder applications and a considerable number of sealing materials are available to work with the wide range of possible hydraulic fluids (see Appendix 6). For systems using mineral oil-based hydraulic fluids (90 percent of applications), the most common rod seal is currently made from polyurethane, either as a U-ring profile or a polyurethane shell with a nitrile rubber (NBR) insert. The coaxial O-ring energized rod seal profile shown in the cylinder drawing

(Figure 3.5) generally has a filled polytetrafluoroethylene (PTFE) face in contact with the rod and is used in applications where low breakout friction is particularly important. Wipers may be made of NBR but again the use of polyurethane is more common. Typically, double-acting piston seals may be of the coaxial type as shown in the drawing, with faces made of various materials including filled PTFE and thermoplastic elastomers, or they may be of a five part construction with a NBR dynamic sealing element, supported by anti-extrusion rings and bearing rings. This five part seal profile fits in a T-shaped groove. The static seals are generally NBR O-rings, with antiextrusion rings usually fitted for working pressures of 100 bar and above.

Further information on all aspects of sealing can be found in [5].

In practice, many piston rods are subjected to lateral as well as axial loads. These loads must not be permitted to cause piston misalignment inside the cylinder, so to reduce this problem, the seal housing fitted to the end cover may be provided with a bearing bush to carry side loads on the rod. Lateral loads can also put excessive loads onto the cylinder rod bearing.

3.3.4 POSITION TRANSDUCERS AND LIMIT AND PROXIMITY SWITCHES

Many manufacturers include the option of position switches or position transducers and limit and proximity switches in the actuator assembly. This avoids having to fit an external transducer that, in many applications, can be exposed to the possibility of damage. The methods used for position indication vary but usually incorporate transducers involving no physical contact (e.g., inductive and ultrasonic systems), giving digital or analogue output. These usually require a hole to be drilled in the piston rod.

3.3.5 TELESCOPIC CYLINDERS

A diagram of a typical telescopic cylinder is shown in Figure 3.6, the main feature of which is to use multiple pistons of short stroke, which together will provide a greater extension than would be available from a single piston cylinder. These tend to be special purpose designs because of the variety of applications that require large movements, for example, truck-tipping mechanisms.

For extending the actuator against a given load initially, normally the largest piston moves first and the pressure required to start the extension

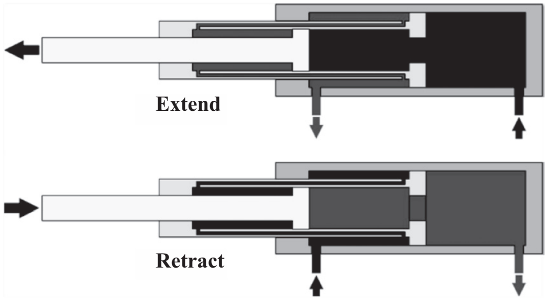

Figure 3.6. Telescopic cylinder.

will depend on the area of the first and largest piston. Referring to Figure 3.6 the force F is given by $F = P_1A_1$.

For the second piston to extend the required pressure P_2 is therefore $P_2 = \dfrac{F}{A_2}$ so that as $A_1 > A_2$ and $P_2 > P_1$.

This neglects any force that results from the return line pressure.

To retract the actuator, the pressure is applied to the annuli of both pistons which will determine the pulling force capability of the cylinder. For applications such as tipping trucks, the force only acts in one direction so that the circuit need not provide high pressure to the piston annuli. However, a restrictive circuit on the high pressure side is needed in order to provide the necessary control action.

There can be more pistons in the cylinder in order to increase the amount of extension. However, the telescopic cylinder is acting as a strut so the force will place constraints on the overall design of the cylinder and the maximum pressure that can be permitted.

3.4 ACTUATOR SELECTION

3.4.1 ACTUATOR FORCE

For the maximum load force (stall force), the system pressure and actuator size may be determined. Factors to consider when choosing system pressure are the mounting style (as discussed in Section 3.3.2), duty cycle, utilization, performance, reliability, and cost.

For an ideal actuator,

$$\text{Force} = \text{Pressure} \times \text{Area}.$$

However, in practice various losses must be allowed for as described:

i. There will normally be pressure on both sides of the piston so the net force, F, is given by $F = P_H A_H - P_L A_L$
The suffixes H and L refer to the high and low pressure sides. The pressure on the low pressure side may be caused by pressure losses in the outlet flow, which will reduce the available force at a given inlet pressure.

ii. The force available at the load is reduced by friction. Frictional forces are difficult to predict, but could be of the order of 20 percent of the load under operating conditions and higher for starting conditions.

iii. An allowance should be made for the efficiency of any mechanical linkages or gears connected to the output.

iv. In valve-controlled systems (e.g., meter-in), the inlet pressure will reduce with increasing actuator velocity. Thus, for systems in which the force is varying, the velocity may vary as a consequence.

In applications where the load is unguided, transverse loads may be acting on the cylinder rod. A stop tube, or spacer, is sometimes fitted to reduce the stroke in such instances but in any case if such loads are to be expected on the cylinder the application should be discussed with the manufacturer.

3.4.2 CUSHIONING

To retard inertial loads and increase the fatigue life of cylinders, some form of internal cushioning is often used. An example of cylinder cushioning can be seen in Figure 3.7.

When the actuator outlet flow is directed through the restrictor, the pressure drop generated will create a backpressure on the actuator, thus causing it to be retarded. The restrictor must be sized such that the maximum pressure, which occurs when the plunger first blocks the normal outlet port, does not exceed the safe value for the actuator.

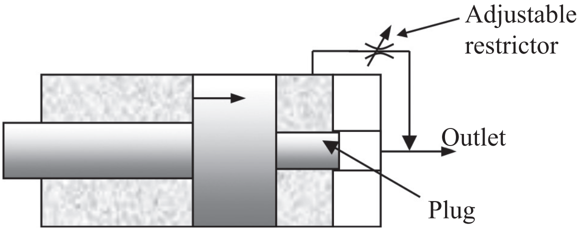

Figure 3.7. Actuator cushioning.

For simple inertial loads, with no other forces acting, the actuator velocity decays exponentially, as does the actuator outlet pressure. This can be shown by simple analysis assuming an incompressible fluid and neglecting friction. Thus from Newton's Law we have:

Inertial force:

$$m\frac{\mathrm{d}^2X}{\mathrm{d}t^2} = m\frac{\mathrm{d}U}{\mathrm{d}t} = P_C A_C \tag{3.1}$$

where X is the movement of the actuator from the commencement of cushioning.

Flow:

$$Q_R = C_D A_R \sqrt{\frac{2}{\rho} P_C}$$

$$\therefore P_C = \frac{Q_R^2}{C_D^2 A_R^2}\frac{\rho}{2} \tag{3.2}$$

Actuator:

$$Q_C = Q_R = A_C U \tag{3.3}$$

Now:

$$\frac{\mathrm{d}U}{\mathrm{d}t} = \frac{\mathrm{d}U}{\mathrm{d}X}\frac{\mathrm{d}X}{\mathrm{d}t} = U\frac{\mathrm{d}U}{\mathrm{d}X}$$

so we get from Equations 3.1 to 3.3:

$$P_C = \frac{U^2 A_C^2}{C_D^2 A_R^2}\frac{\acute{A}}{2} = \frac{mU}{A_C}\frac{\mathrm{d}U}{\mathrm{d}X} \tag{3.4}$$

And:

$$\int_{U_m}^{U} \frac{\mathrm{d}U}{U} = \frac{\rho A_C^3}{2C_D^2 A_R^2 m}\int_0^X \mathrm{d}X = \frac{C}{m}\int_0^X \mathrm{d}X \tag{3.5}$$

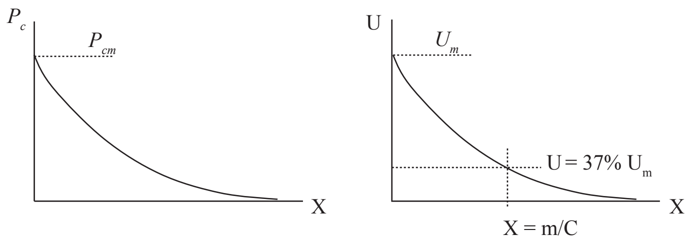

Figure 3.8. Velocity and pressure variation.

The solution of Equation 3.5 gives:

$$U = U_m \exp\left(-\frac{CX}{m}\right) \tag{3.6}$$

The velocity and pressure variations with the distance, *X*, which the actuator has moved after cushioning has commenced are shown in Figure 3.8. At the start of the cushioning, the pressure rises to a maximum value, P_{Cm}, when the flow is a maximum. For a given mass and initial velocity, the maximum cushion pressure is determined by the size of the adjustable restrictor. This also determines the distance that is required for the actuator velocity to reduce to an acceptable value.

It is usual that P_C max should not exceed 350 bar, which is a normal fatigue pressure rating for 10^6 actuator cycles. The change in the pressure, and velocity, will be slightly modified by the effect of the fluid compressibility but in most systems this effect will be small and the cushion performance can be calculated using the equations.

Some cushioning systems employ a long tapered plunger that maintains a higher mean pressure throughout the cushioning stroke and, consequently, reduces the cushioning distance. In others, the plunger has stepped diameters to give almost the same effect. The performance of these cushion methods is usually given in manufacturer's literature in terms of the energy that is to be absorbed and the pressure level on the supply side of the actuator.

The shortest cushion length would be obtained from one that creates a constant pressure at the maximum permissible value. A comparison with a tapered cushion shows that the cushion length can be reduced by around 30 percent.

3.5 ROTARY ACTUATORS

Rotary actuators are designed specifically to provide a limited angle of rotation. These are distinct from hydraulic motors and as a consequence are of simple design as no timing valve is necessary.

3.5.1 ACTUATOR TYPES AND CAPACITY RANGE

Three types of rotary actuators include rack and pinion, vane, and helical screw, which are used to produce a limited range of output rotary shaft

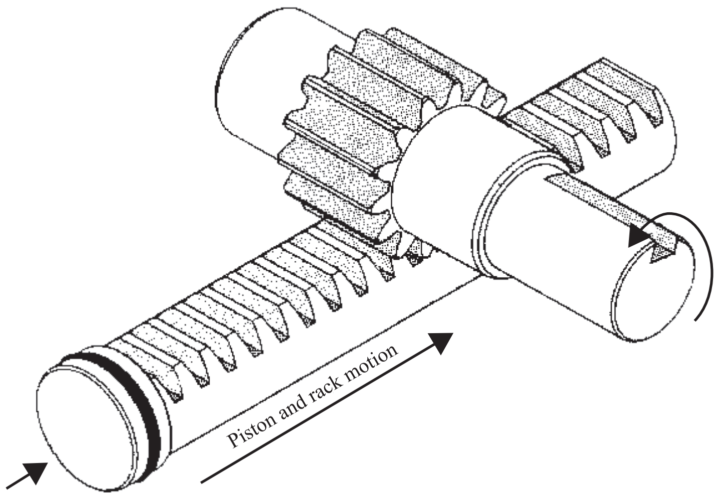

Figure 3.9. Rack and pinion rotary actuator.

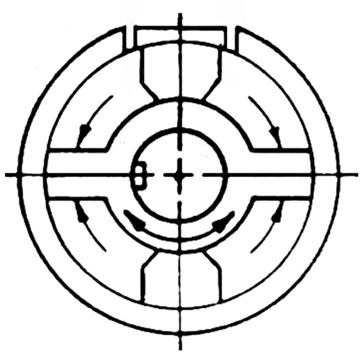

Figure 3.10. Vane rotary actuator.

Table 3.2. Summary of typical rotary actuator performance

Type	Angle range	Torque (Nm)
Rack and pinion	>360°	42,000
Vane	<280°	22,000
Helical (not shown)	<420°	26,000

angles. The basic construction of the gear and vane types is shown diagrammatically in Figures 3.9 and 3.10. A performance summary is given in Table 3.2.

3.5.2 APPLICATIONS

Rotary actuators are used for the following applications:

- Steering systems
- Gate valves
- Boom slew of backhoe
- Manipulator drive
- Tunneling machine
- Container handling

Most of the actuators will carry side loads and can usually be supplied with position indication, cushioning valves, mechanical stops, and a variety of shaft attachment features.

SUMMARY

As discussed in Chapter 2, the flow output from a pump can be used to drive a motor where the rotary speed of the motor is determined by its displacement. Hydrostatic motors are a class of actuators in that they convert flow and pressure into velocity and torque on a continuous basis, but they can also be used in some applications where only limited movement is required.

However, rotary actuators (semirotary actuators or rotary actuators with a limited swivel angle) are available that have a limited rotation angle and offer a reduction in cost because of their relative mechanical simplicity. These can also avoid the need for a holding brake because of their low leakage. Hydraulic cylinders are normally used for providing linear motion, which can have virtually zero leakage so that, with blocked ports, stationary loads will be held indefinitely. The use of a rack and pinion gear drive allows the linear movement to be converted to rotary motion whilst retaining the stalled characteristics of the hydraulic cylinder.

Hydraulic cylinders (linear actuators) are extensively used in all of the major engineering fields and the system designer needs to be aware of the different types of construction that are available, the various mounting methods, and the influence that these have on their load carrying characteristics.

CHAPTER 4

Pressure Control Valves

4.1 INTRODUCTION

Major types of valves that are used to control the pressure level in hydraulic systems include:

- Relief valves for limiting the maximum system pressure.
- Reducing valves for limiting the pressure in parts of a circuit at a lower level than in the supply system.
- Load control valves to control the motion of an actuator or motor under the action of overrunning, or negative, forces.

The type that is employed in a given application depends on the particular requirements and system specification.

4.2 RELIEF VALVES

Relief valves are the most commonly used pressure control valve as they are required in all systems to prevent the generation of excessive pressures. In many systems they are used in combination with a pump to provide a source of flow at constant pressure.

4.2.1 SINGLE-STAGE RELIEF VALVE

Single-stage relief valves, as with all valves, can use either a piston or a spool that is opened by a pressure force against a preloaded spring as shown in Figure 4.1 together with the ISO symbol that is used to represent it in a circuit diagram. On opening, the valve allows some of the supply flow to be passed back to tank thus limiting the maximum pressure in the supply. The valve needs to be sized such that all of the supply flow can be returned to tank at a supply pressure that does not exceed the maximum desired level.

Single-stage relief valves are the simplest and lowest cost valves. Considering the diagram of a poppet valve in Figure 4.2, the valve will start to open at its "*cracking pressure*" when the force on its face due to the inlet pressure is equal to the spring preload.

As the pressure increases above this value, the valve opens progressively thus allowing flow to pass through the valve. This feature is referred to as the "*pressure over-ride*."

The rate of the spring will determine the relationship between the valve displacement and the inlet pressure. Minimizing the free length of the spring requires the spring to have a high stiffness. However, the higher the stiffness the greater will be the pressure over-ride. There will be some hysteresis due to friction between the components that will result in the

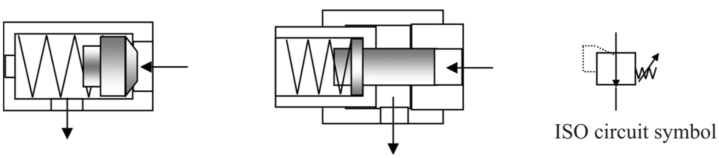

Figure 4.1. Poppet and piston-type valves.

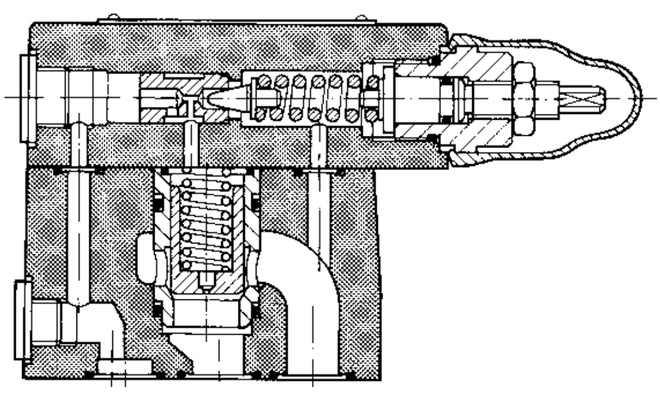

Figure 4.2. Two-stage poppet type relief valve.
Source: Courtesy Eaton.

pressure being slightly higher when the valve is being opened than when it is being closed.

As discussed in Chapter 8, there is an additional force that arises from the increase in the momentum of the fluid as it passes through the valve opening, which acts in the direction to close the valve. This force acts as a spring, the rate of which is added to that of the mechanical spring and its effect is to increase the pressure over-ride.

4.2.2 TWO-STAGE RELIEF VALVES

The two-stage valve shown in Figure 4.2 uses a spring-loaded pilot poppet (pilot relief valve) to sense the pressure level in the supply at A. When this pressure causes the pilot relief valve to open, the flow through the balancing orifice creates a pressure drop across the main valve poppet that has a spring preload in the region of 2 bar. This causes flow from the supply to be returned to tank at a controlled level of the supply pressure.

Two-stage valves have a much reduced pressure over-ride compared to single-stage valves because the main spring is not required to be preloaded to the controlled pressure level. This allows a reduced spring rate to be used that reduces the pressure over-ride. This can be of advantage where it is required to control the supply pressure within close limits.

The pilot relief valve can be isolated from the main valve. There can be more than one pilot valve, which can be set at different pressures so that the connection of any one will operate the relief at the respective pilot set pressure.

Figure 4.3 shows a typical cartridge type of two-stage relief valve. Cartridge valves are available for installation in individual housings, sandwich mounting (stacking), and in special manifold blocks.

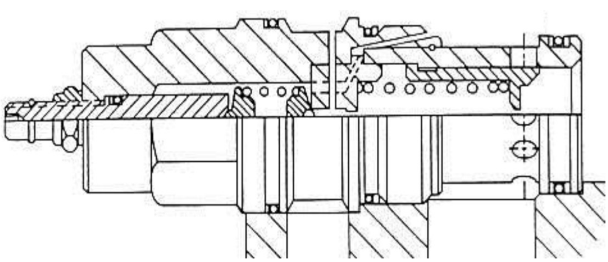

Figure 4.3. Two-stage cartridge relief valve.
Source: Courtesy Sun Hydraulics.

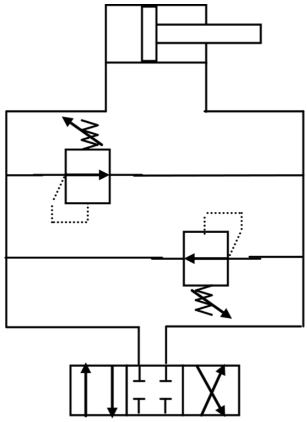

Figure 4.4. Dual relief valves in actuator circuit.

Relief valves are also available with electric control using proportional solenoids to provide the necessary force in place of a mechanical spring. A major advantage of this type of valve is the ability to adjust the pressure setting from an appropriate voltage-controlled amplifier.

Dual crossline relief valves, as shown in Figure 4.4 for limiting the pressure on both sides of a linear actuator, are available in a single casing.

4.3 PRESSURE REDUCING VALVES

A pressure reducing valve is used to provide a sub-circuit with a supply of fluid at a pressure, which is less than the pressure in the main circuit. Figure 4.5 shows a schematic diagram.

The downstream pressure P_r acts on the first-stage, spring loaded, poppet valve and when this valve is open, the resultant flow through the orifice drilled through the spool valve creates a pressure differential between the two ends of the spool. This moves the spool against a spring so that the spool throttles the flow between the supply and service ports. If the downstream pressure rises above the required level, the spool moves to increase the throttling action (and vice versa).

The reducing valve in Figure 4.6 has a screw adjustment to change the set pressure.

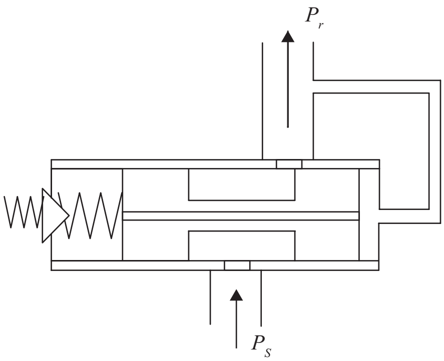

Figure 4.5. Reducing valve operating principles.

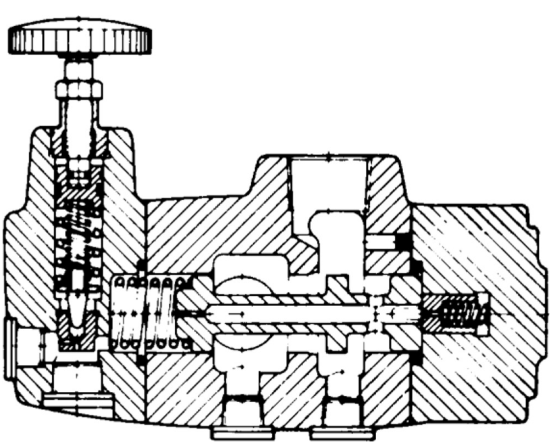

Figure 4.6. Reducing valve.
Source: Courtesy Eaton.

4.4 COUNTERBALANCE VALVES

For actuators, or motors, under the action of negative forces (e.g., pulling forces during extension) it is necessary to provide a restriction in the flow outlet in order to create a resisting, or back, pressure.

A cartridge type of load control valve of the counterbalance type is shown in Figure 4.7. The operation of the valve opening is controlled by both pressures. The ratio of the areas exposed to the pressures P_1 and P_2 can be selected, this usually being in the range 3:1 and 10:1.

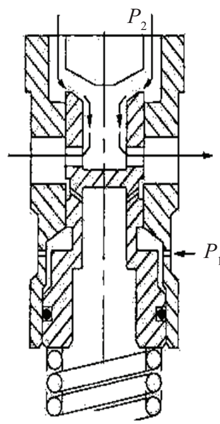

Figure 4.7. Cartridge type of counterbalance load control valve.

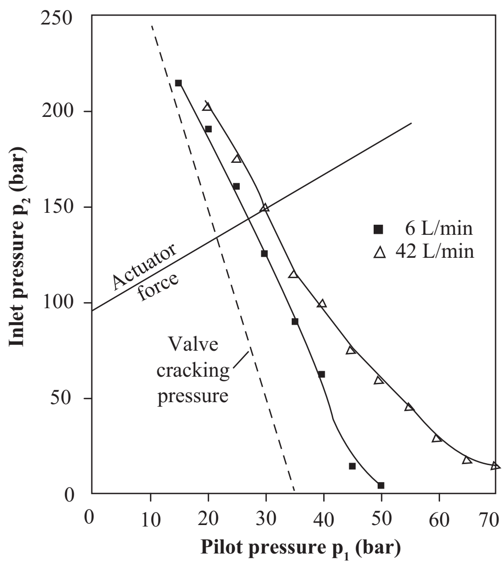

Figure 4.8. Flow characteristics of the counterbalance valve in Figure 4.7.

A typical operating characteristic of the valve is shown in Figure 4.8 for the cracking pressure and for two flows. The flow through the valve depends on its opening and the pressure P_2 so as P_2 reduces, the valve opening has to be increased, which requires P_1 to have a higher value over that for the valve just cracking open. This effect can be seen in Figure 4.8.

The maximum value of P_2, which is set by preloading the spring with a screw adjustment, is chosen to provide a maximum safe pressure for the actuator and occurs when P_1 is zero. For the actuator, the variation of P_2 with P_1 depends on its area ratio, which has a minimum value of unity for an equal area cylinder.

For a given force on the actuator, the pressure relationship for extension of the actuator is shown in Figure 4.8 superimposed onto the valve characteristics, the point of intersection providing the operating conditions of the system. Standard cylinders normally have a maximum area ratio of around 2 but some applications use actuators having ratios considerably higher than this. Clearly, this has an influence on the operating pressures when under the control of a counterbalance valve.

As seen from Figure 4.9, the pilot pressure, P_1, is obtained from the pump outlet to the actuator inlet when the actuator is extending. If the actuator flow exceeds that of the pump, P_1 will reduce and close the valve. This will cause the actuator outlet pressure, P_2, to increase and thus reduce the actuator speed until the pressures create an opening of the valve that provides equality of the actuator and pump flows and also provides a pressure force that is equal to the load force. This condition is shown in Figure 4.8 where the actuator force line crosses the valve operating line for a given pump flow rate.

The dynamic performance of counterbalance valve systems, which operate as closed loop systems because of the use of pressure feedback to control the valve position, is complex and can result in oscillatory motion of the load. For large systems, it is advisable to carry out a simulation of the system in order to avoid this problem.

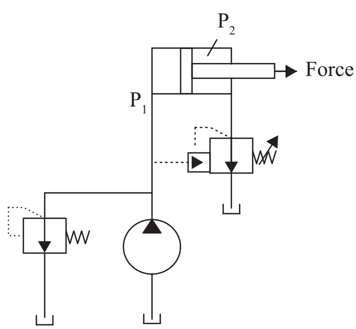

Figure 4.9. Extending actuator controlled by a counterbalance valve with a pulling load.

The operating pressures can be estimated for a given valve and actuator using the area ratios for these components and the maximum pressure setting, P_S, so, considering a valve having an area ratio of *b,* the pressures to just crack the valve open are given by:

$$P_1 = \frac{P_S - P_2}{b} \tag{4.1}$$

For the actuator, the value of P_2 when $P_1 = 0$ is $P_2 = P_L = \dfrac{\text{Force } F}{\text{Annulus Area}}$.
For the force balance of the actuator, the pressure P_2 will increase with changes in P_1. Thus, for an actuator area ratio of α:

$$P_2 = P_L + a\,P_1 \text{ or } P_1 = \frac{(P_2 - P_L)}{a}$$

Thus referring to Figure 4.10, the value of the operating pressure P_{20} can be obtained from equating the two values of P_1:

$$\frac{P_S - P_{20}}{b} = \frac{(P_{20} - P_L)}{a} \tag{4.2}$$

Therefore

$$\therefore P_2 = \frac{P_1 + P_S\left(a/b\right)}{1 + \left(a/b\right)} \tag{4.3}$$

For a very high valve area ratio, $a \to \infty, P_{20} \to P_L$

Thus for P_S = 200 bar, P_L = 100 bar, α = 2 and b = 10 gives P_{10} = 8.3 bar and P_{20} = 117 bar.

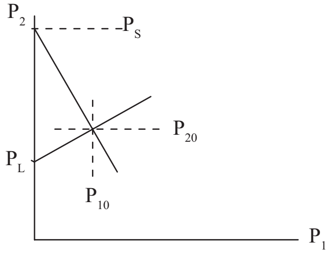

Figure 4.10. Operating pressures.

It is seen from Figure 4.8 that the pressures will change with the flow from the pump—the amount of this change being dependent on the area ratio (a) of the valve. The value of the pressure determined from Equation 4.3 is that for the valve just cracking open. Consequently, for different pump flows, the operating pressures have to change to open the valve a sufficient amount to pass a given flow. The variation of the pressures with flow can be solved by using similar techniques to those applied in Section 8.6.1.2 for the variation of pressure and flow in a simple check valve.

SUMMARY

Generally, the pressure level in hydraulic systems will vary so as to provide the required torque or force from an actuator in order to drive an external load from the particular application. During start and stopping situations and when the load is varying transiently, the pressure may exceed the maximum safe value for the system. In many systems, several actuators will be driven by a single pump and, in addition to limiting the supply pressure, it may be necessary to reduce the pressure level supplied to individual services. There are many different types of valves on the market and this chapter describes some of these and the operating principles involved.

CHAPTER 5

Flow Control Valves

5.1 INTRODUCTION

The control of flow is broadly divided into major types that include:

- Directional control
- Simple restrictor
- Pressure compensated
- Open center and bypass

The control of flow is a major feature of hydraulic systems and there are a variety of methods available for this, which can be used with both fixed and variable displacement pumps. This chapter is concerned with the basic aspects of flow control valves and their characteristics that are of importance in the design and selection of hydraulic circuits.

5.2 DIRECTIONAL CONTROL VALVE

Spool-type valves provide the major method of controlling the direction of flow and are used extensively for many circuit functions that are discussed in the chapter concerned with circuit design. Figure 5.1 shows the basic features of this type of DCV, which connects one of the two outlet ports (B) to the supply port (P) and the other (A) port to the tank or return line with movement of the spool from the central position.

The valve of Figure 5.1 has all the ports closed in the center position as represented by the ISO symbol; however, as discussed in Chapter 7, other configurations having a range of spool options and port connections

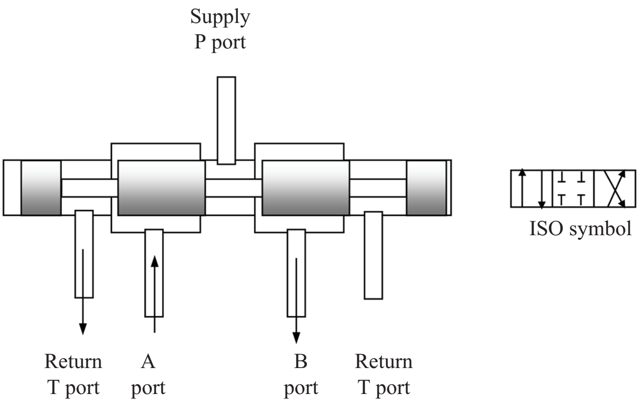

Figure 5.1. Four-way directional control valve (DCV).

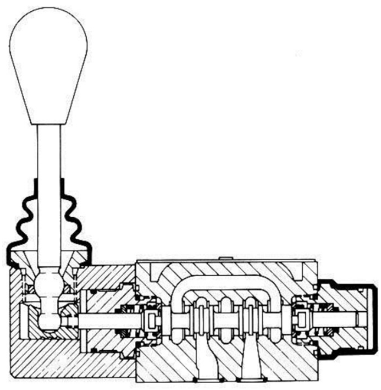

Figure 5.2. Manually operated DCV.
Source: Courtesy Eaton.

are available in order to perform a range of alternative circuit functions. For holding the valve in given positions, they can be spring centered and have detents that engage at particular spool displacements.

The valve can be positioned by a direct manual control, as shown in Figure 5.2, where the input lever is connected by a spherical coupling to the end of the spool. Other means of positioning the spool are available including hydraulic or pneumatic pilot signals, direct force control from an electric solenoid or indirectly from a solenoid using a hydraulic amplifier (e.g., electrohydraulic servo valve).

Pilot control has a distinct advantage over mechanical operation because the signal can be derived some distance from the valve itself. The electrically operated proportional valve, shown in Figure 5.3, has high

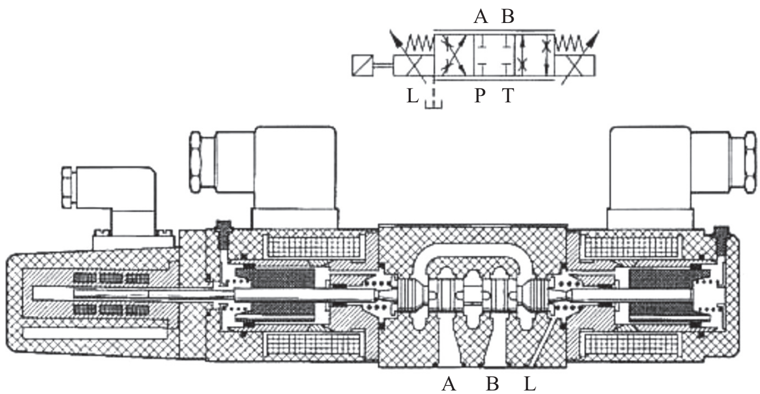

Figure 5.3. Proportional control valve.
Source: Courtesy Eaton.

accuracy and resolution, and these valves are used extensively in a wide range of applications for both open and closed loop control of actuator position.

The response time of proportional valves is dependent on the type and size of the valve and is quoted in the manufacturer's technical literature. The rate at which these valves open and close can be adjusted in the amplifier in order to minimize the magnitude of pressure shocks in the system. This is particularly applicable to loads that have a large mass or inertia.

Monitoring signals are available so that failures can be detected and the valve put into a system safe position. The maximum flow is limited by the effect of flow forces, discussed in Chapter 8, which oppose the force from the solenoid. It is important to observe the manufacturer's recommendation on contamination control, as fluid-borne particles are the major cause of failure and unreliable operation of control valves including solenoid burnout in AC valves.

5.3 RESTRICTOR VALVE

Spool valves, particularly the proportional type, are used extensively for restrictive control of actuators as discussed in Chapter 7 on circuit design. A simple method of flow control can be obtained using a restrictor for meter-in, meter-out, and bypass circuits. An adjustable type is shown in Figure 5.4 where the position of the tapered needle is altered by the screw. Pressure is created at the valve inlet in order to pass the flow through the tapered orifice.

Restrictors create a pressure loss in the flow as described in Chapter 8 for orifices and are used with a controlled pressure source. The flow

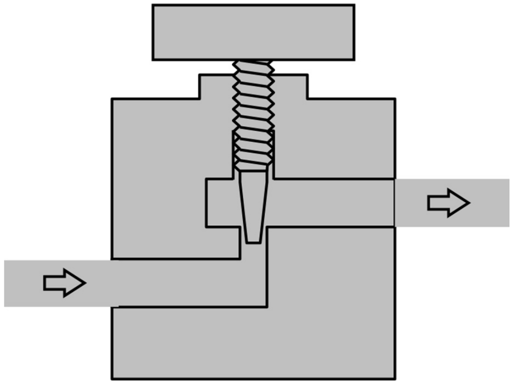

Figure 5.4. Adjustable restrictor valve.
Source: Courtesy Eaton.

through the valve will vary with the square root of the available pressure drop and the adjustment allows for obtaining the desired flow in an application.

The restrictor valve is normally pre-set but variable control can be obtained by positioning the spool in DCVs to give a range of openings from closed to fully open. Proportional valves provide very accurate control of spool position and are often used to vary the restriction in the inlet and outlet flow paths to an actuator as described in Chapter 7.

In situations where the pressure drop is not constant, the flow will vary as a consequence. To avoid this problem, pressure compensated valves can be used as described in the next section.

5.4 PRESSURE COMPENSATED VALVE

Pressure compensated valves use an adjustable restrictor together with an additional valve that opens and closes in order to maintain a constant pressure drop across the restrictor.

Figure 5.5 shows this type of valve whereby as flow passes from the A to the B ports, the pressure drop ($P_2 - P_3$) across the manually adjustable rotary restrictor creates a force on the spool against that of the spring. If the force from the pressure drop exceeds the value set by the spring, the spool valve closes and further restricts the outlet flow to the B port. With the inlet being supplied from a controlled pressure source, P_1, the supply flow will be reduced in order to maintain this pressure at a constant value. Either a pressure relief valve or a pressure compensated pump could be used to perform this action of controlling pressure P_1.

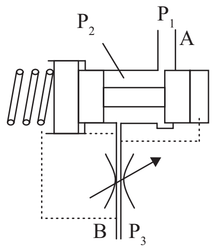

Figure 5.5. Pressure compensated valve.

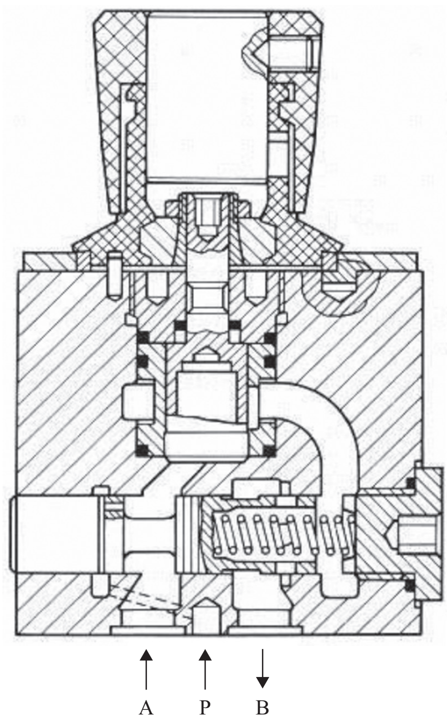

Figure 5.6. Pressure compensated flow control valve.
Source: Courtesy Eaton.

Thus, the flow is kept constant with changes in the outlet pressure, which provides a flow that, unlike the simple restrictor, is independent of the outlet, or load, pressure. This valve is a closed loop control in that the pressure drop is fed back onto the spool to control its position. However, the flow itself is controlled by the pressure supply control system. A typical pressure compensated valve is shown in Figure 5.6, the variable restriction being set by the control screw on the top of the valve.

The control method employed in the pressure compensated valve is used extensively in many other types of valves for a variety of hydraulic circuit functions as discussed in Chapter 7.

5.5 CENTRAL BYPASS VALVE

Central bypass valves provide a combination of directional control and restrictive metering where the metered flow is bypassed directly to the tank, or return line.

Figure 5.7 shows the construction of a typical bypass valve having three spools for operating three functions from a single pump. With the spools in the neutral position, the pump flow passes through the open centers and as a spool is displaced it creates an increasing restriction to the pump flow thus raising the pressure.

Progressive displacement of the spool eventually opens a service port to the pump pressure, but flow will only pass to this service if the pump pressure is high enough to work against a loaded actuator that is connected to the outlet. Figure 5.8 shows the valve disposed from the center position so that the “A” port is connected to the supply and the “B” port to return.

The load check valve is fitted so that it will open when the pump pressure is slightly higher than that required to move the actuator, or motor connected to either of the outlet ports. A number of circuit configurations are available with this valve some of which are described in Chapter 7 on circuit design.

The spool movement also connects the other service port to the return line so that four-way control of actuators is provided. The major advantages of this valve are its simple construction and the generation of pump pressures that are only slightly higher than the maximum outlet pressure so minimizing energy losses. The flow will vary with the load pressure from the interaction that arises when two or more functions are operated simultaneously, the level of which varies with the relative pressures of the valve outlets. This variation can be useful in some equipment such as, for example, in a digger where such variation provides the operator with a “feel” that something hard in the ground has been struck—this can be useful when digging close to drains or cables.

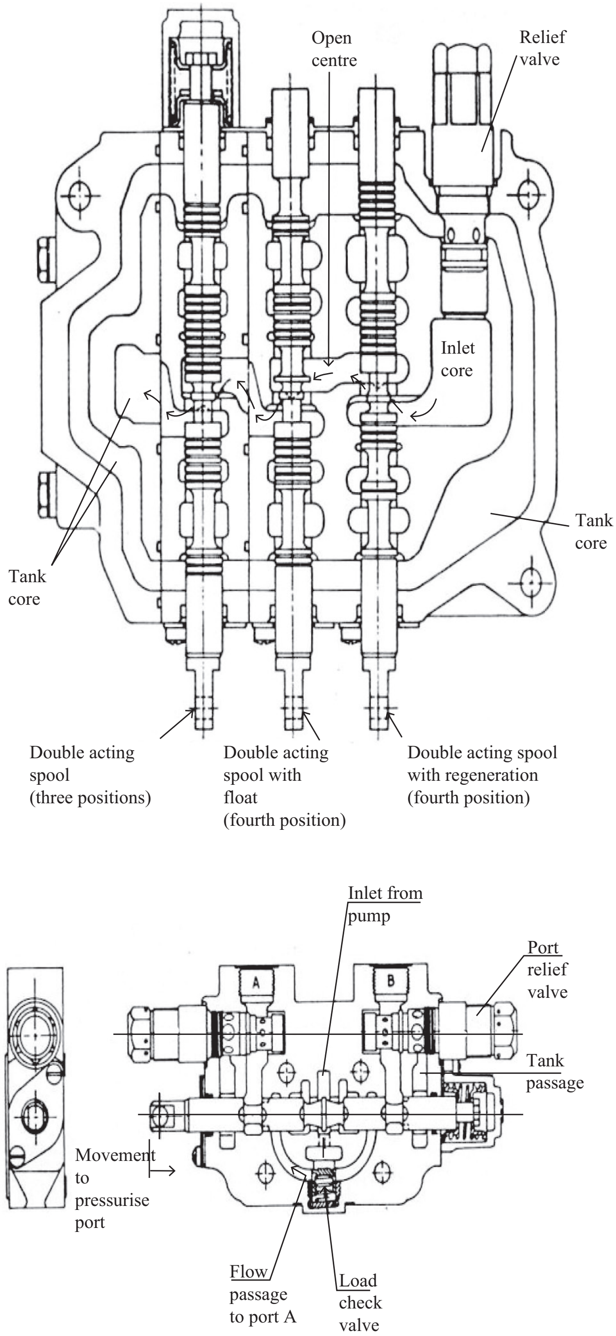

Figure 5.7. Central bypass valve.
Source: Courtesy Parker.

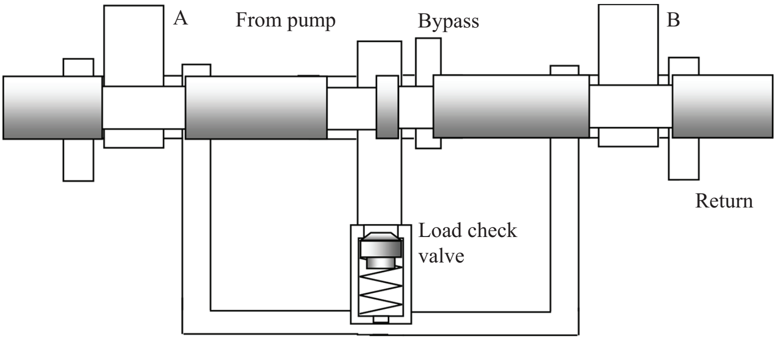

Figure 5.8. Bypass valve connecting the pump flow to port A and return to port B.

SUMMARY

The flow from hydraulic pumps can be fixed or variable depending on the type of pump being used. For fixed displacement pump systems, the simplest form of flow control is obtained by generating a level of pump pressure that causes operation of the relief valve, but this method can be extremely inefficient. Alternative valve systems can be used to improve this situation particularly when operating several services from a single pump. The type of circuit used in this case will depend on whether the pump is fixed or variable displacement. Some of the available circuit options are described in Chapter 7 that use valve types that are described in this chapter.

CHAPTER 6

Ancillary Equipment

6.1 INTRODUCTION

This chapter is concerned with describing the function and relevant performance aspects of the following components:

- Accumulators
- Filters
- Coolers
- Reservoirs

6.2 ACCUMULATORS

6.2.1 TYPES

Accumulators are widely used in fluid power systems as a means of storing energy. Although weight- and spring-loaded types are sometimes used, those employing a pressurized gas are preferred because of their compactness and superior performance.

There are two main types of pressurized-gas-filled accumulators: those using piston and those using collapsible bladders.

As shown in Figure 6.1, the fluid is separated from the pressurized gas by either a bladder, Figure 6.1a, or a piston, Figure 6.1b. The gas is usually nitrogen that is supplied via the gas valve with a pre-charge pressure that is determined by the pressure range required by the application.

Sealing of the piston is obviously important, and there can be friction between the piston and the cylinder that can affect the liquid-pressure

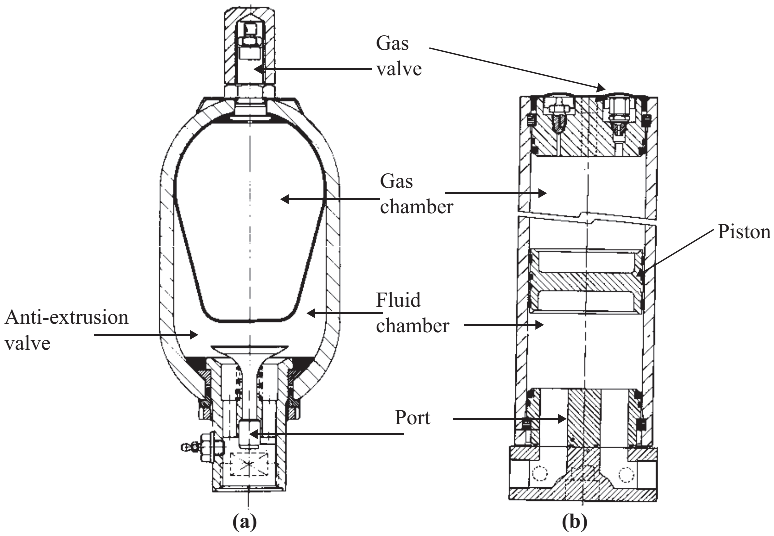

Figure 6.1. Accumulators: (a) bladder type and (b) typical piston type.
Source: Figure 6.1a courtesy Fawcett Christie.

level. This problem does not arise with the bladder types, and extra gas can be added by the use of separate storage gas containers.

Accumulators are typically used for the following purposes:

i. The supplementation of pump flow to meet high transient flow demands
ii. Emergency supply
iii. Leakage compensation
iv. Shock alleviation
v. Compensation required for volume changes due to temperature or pressure
vi. Simple suspension elements
vii. Pulsation absorption

Legislation on the use of gas-filled vessels requires certain maintenance procedures to be carried out, which are described in Section 12.6.

6.2.2 PERFORMANCE

The accumulator is initially charged to a pressure P_0 that is set at a level lower than the minimum operating pressure P_1. The pressure of the gas will vary with changes in the volume, but the relationship between these parameters will depend on the amount of heat transferred to the surroundings. It is usual to assume a polytropic expansion index for the gas, the value of which depends on the operating times and the duty cycle.

Referring to Figure 6.2, the gas states are defined as follows:

Pre-charge: pressure P_0 and volume V_0
Minimum operating: pressure P_1 and volume V_1
Maximum operating: pressure P_2 and volume V_2.

For a gas having a mass *m*, an absolute temperature *T*, and a polytropic index *n*, the universal gas laws for a perfect gas give

$$PV^n = \text{constant} \tag{6.1}$$

$$PV = mRT \tag{6.2}$$

R = universal gas constant.

The accumulator is connected to an appropriate point in the hydraulic system such that when the pressure falls the gas will expand and deliver a volume of fluid ΔV to the hydraulic system, thus maintaining its pressure. The maximum volume is given by

$$\Delta V = V_1 - V_2 \tag{6.3}$$

Using a polytropic index, n_1, for compression from V_0 to V_2, for the period when the fluid pressure increases to its maximum value, Equation 6.1 gives

$$P_0 V_0^{n_1} = P_2 V_2^{n_1}$$

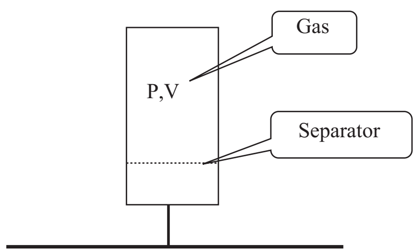

Figure 6.2. Accumulator pressure.

Thus

$$V_2 = \left(\frac{P_0}{P_2}\right)^{1/n_1} V_0 \tag{6.4}$$

Also, for the gas expansion from V_2 to V_1 with a polytropic index of n_2

$$V_1 = \left(\frac{P_2}{P_1}\right)^{1/n_2} V_2 \tag{6.5}$$

Equations 6.4 and 6.5 with Equation 6.3 give

$$\Delta V = V_0 \left(\frac{P_0}{P_2}\right)^{1/n_1} \left\{\left(\frac{P_2}{P_1}\right)^{1/n_2} - 1\right\} \tag{6.6}$$

The values of the polytropic indices cannot be accurately predicted, and it is usual to take the value of n_1 as 1 (isothermal) and n_2 as γ for the gas, where this value is obtained for the expected operating temperature and pressure.

Thus Equation 6.6 yields

$$V_0 = \frac{\left(\frac{P_2}{P_0}\right)}{\left\{\left(\frac{P_2}{P_1}\right)^{1/\gamma} - 1\right\}} \Delta V \tag{6.7}$$

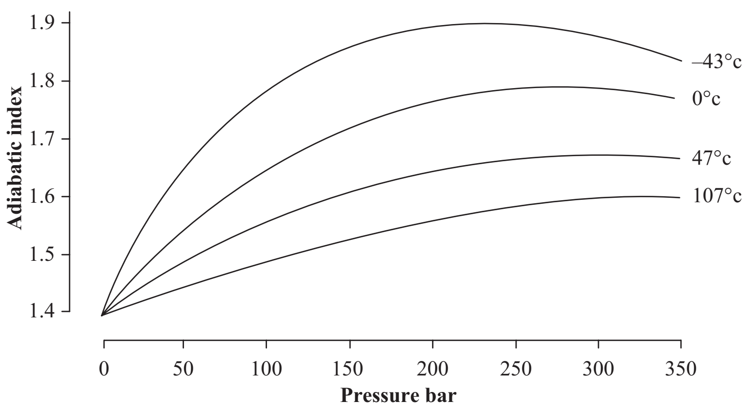

Figure 6.3. Variation in adiabatic index with pressure and temperature for nitrogen.

This equation gives a conservative value for most applications. In certain cases, for example, high or low temperatures, it may be necessary to apply a correction factor and, in those situations, information should be obtained from the manufacturer.

The value for γ can be obtained from Figure 6.3 that applies to real gases and should be used in accumulator sizing calculations.

6.3 CONTAMINATION CONTROL

6.3.1 COMPONENTS

The selection of inadequate filters or poor maintenance procedures can cause excessive contamination levels that may result in the unreliable operation and breakdown of hydraulic components. Filtration systems should, therefore, be designed such that the fluid cleanliness level is better than that specified by the component manufacturers.

The loaded contact regions are shown in Figures 6.4 and 6.5, where metal particles are generated in pumps and also where particles in the incoming fluid will accelerate the wear process. The clearances in pumps, motors, and valves are of the order of a few microns, and it is, therefore, essential that the particles in the fluid are maintained at a size that is appropriate for the prevention of wear in the clearances. Filters in a system clearly have to have a rating that protects the smallest clearance in the system.

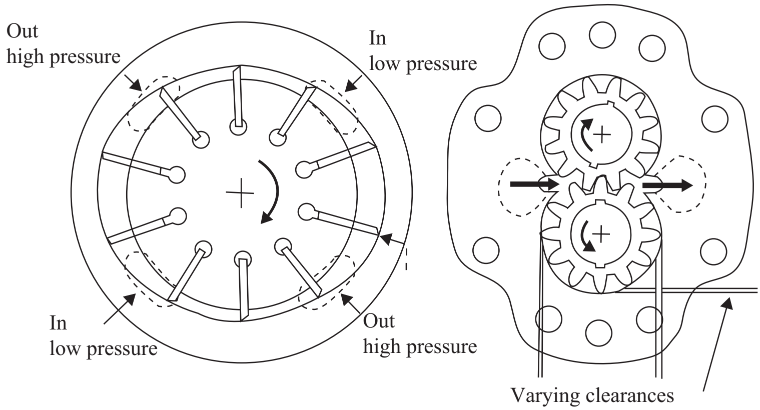

Figure 6.4. Important contamination aspects in vane and gear pumps.
Source: Courtesy Eaton.

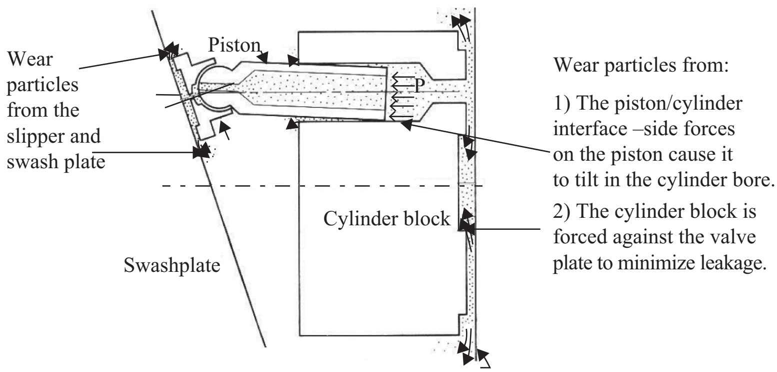

Figure 6.5. Wear particle generation in piston pumps.
Source: Courtesy Eaton.

6.3.2 FILTERS

The main features of a replaceable element high-pressure filter are shown in Figure 6.6. As the fluid passes radially inward through the element, contaminant is trapped in the material. With time, the pressure drop across the filter will increase at a rate that is dependent on the fluid condition and, eventually, this will cause the bypass valve to open, thus passing contaminated fluid directly into the system. However, the pressure drop can be monitored either mechanically or by electronic methods to give an early warning of bypassing, and this aspect is an important feature in a properly maintained system.

A major problem associated with filtration is that its effect cannot be seen because of the small size of particles that can cause poor system reliability and component failure. So, it is important that monitoring of the filter condition is carried out on a regular basis. Sampling techniques and the measurement of the contaminant concentration provide an improved basis for monitoring the condition of the hydraulic system. Of these, online monitoring is the most cost-effective method of achieving this.

The performance of a filter is based on its ability to trap particles, which is defined by its beta ratio, β, that is obtained from an internationally accepted laboratory-based test method, which is discussed in [6]. There are a number of standards that are listed in the Appendix (11, 12, 13, and 14).

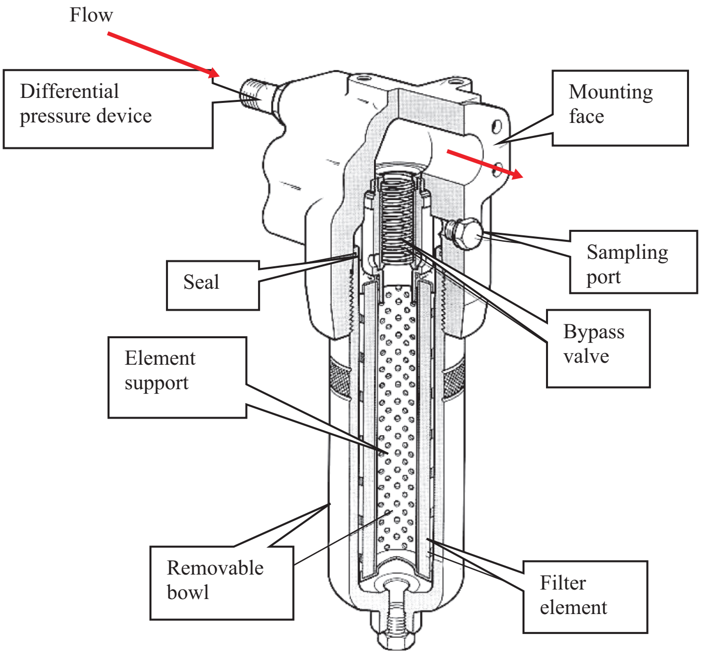

Figure 6.6. High-pressure filter.
Source: Courtesy Pall.

The beta ratio is defined as

$$\beta_x = \frac{\text{No. of particles} > \text{particle size } 'x' \text{ upstream}}{\text{No. of particles} > \text{particle size } 'x' \text{ downstream}}$$

The filtration media of most modern hydraulic filters consist of fibrous material, usually filaments of glass, and this gives the filter a different performance with size, as seen in Figure 6.7, which shows the performance of different grades of media when tested using the Multi-pass test. The rating of a filter is obtained from this data and is the micron size (x) where a stated β value is attained, usually $\beta_x = 200$ or $\beta_x = 1{,}000$. Note that calling the rating “absolute” for these filters has been discredited due to the statistical nature of particle removal.

The performance of the filter has an immediate effect on the cleanliness level of the fluid downstream of the filter, and the higher the beta ratio the cleaner the fluid, as seen in Figure 6.7.

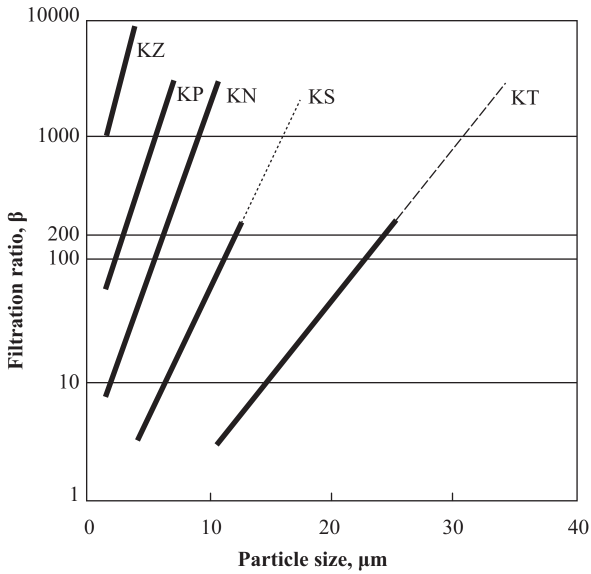

Figure 6.7. Beta ratio for filters.
Source: Courtesy Pall.

Filters are selected on the basis of achieving the desired contamination levels and having sufficient contaminant holding capacity to maintain the required contamination levels under the worst envisaged circumstances. Various selection methods are available from different filter manufacturers, but for consistency, the process developed by the British Fluid Power Association (BFPA) (Appendix 6) has been developed into two ISO standards (Appendix 12 and 13). Their use is recommended, the majority of which are based on an absolute filter rating at a given β ratio.

In general terms, the downstream fluid quality varies as the reciprocal of the filter beta ratio, as represented by the chart in Figure 6.8.

Contaminant levels are denoted by an ISO code that is related to the numbers of particles of sizes greater than 4, 6, and 14 microns, respectively; if the analysis method is using an automatic particle counter, but if it is by microscope, the ISO 4406 Code is at –/5/15 microns. This is shown in Figure 6.9.

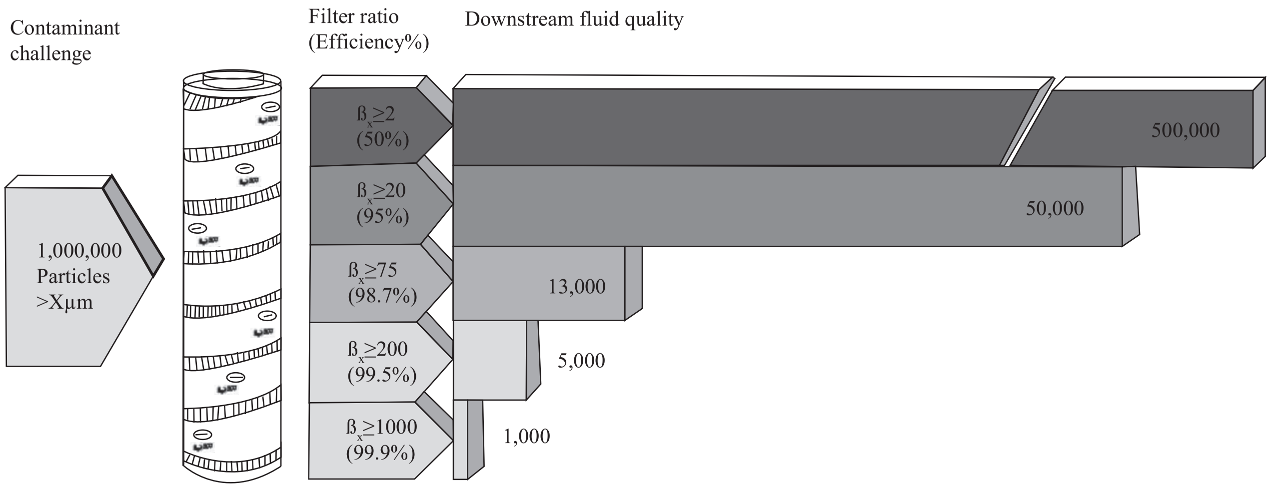

Figure 6.8. Downstream fluid quality and the beta ratio.
Source: Courtesy Pall.

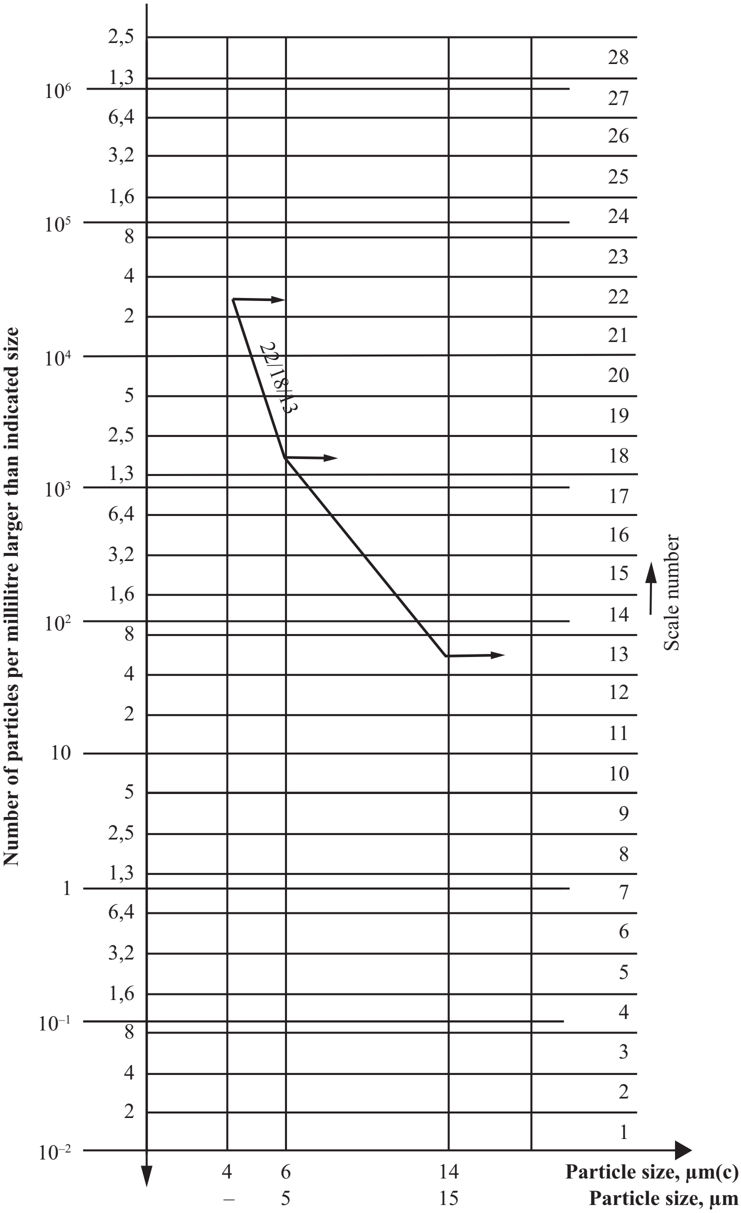

Figure 6.9. ISO 4406 standard code for contamination levels.

6.4 COOLERS

Heat is generated in the fluid in hydraulic systems because of losses in pipes, fittings, and, particularly, in control valves, where the rate of heat dissipation can be of the same order of the power being produced at the system output. The temperature created in the fluid will depend on the system duty cycle and its environment, as natural heat convection from pipes and reservoirs is not at a very significant level.

In industrial systems, the fluid temperature is usually around 50°C to 60°C, and in mobile equipment, this can be as high as 80°C. The condition of most hydraulic oils is significantly affected by operation at high temperatures, which will shorten the life of the oil and reduce its viscosity to unacceptable levels.

Hydraulic component manufacturers specify the viscosity range to be used, and in the main these will call for a minimum value of around 20 cSt, although some will operate satisfactorily at 10 cSt and less. It is important to realize that the volumetric efficiency of pumps and motors is significantly affected by operation with low-viscosity fluids, which will cause a further increase in the heat load.

It is therefore necessary to estimate the amount of heat generation in order to establish the size of cooler that will be required to maintain a satisfactory fluid temperature.

6.4.1 COOLER TYPES

Coolers use either air or water as the cooling fluid. In water coolers, the water flows through the tubes and the oil across the tubes, the latter guided in its flow path through the shell by baffle plates. There are two common constructions: In the first, the tubes are arranged in a U-bundle with a single tube sheet, and in the second two tube-sheets are used in a straight tubing arrangement.

The maximum oil pressure that the cooler can be subjected to is limited by the shell; a typical figure would lie in the range 15 to 30 bar. The pressure drop associated with the oil flow through the cooler is usually small, of the order of 1 bar. Water coolers are more compact than those using air and, provided that an adequate supply of cool water is available, they are less sensitive to environmental conditions. In some cases, it may be necessary to fit a strainer at the cooler inlet in order to prevent blockage of the water flow.

Air coolers using fans to create the necessary airflow are of a lighter construction than water coolers but are larger and sensitive to the environmental conditions, which needs to be considered. Air coolers are mostly used for mobile applications and usually can only work with oil pressures up to around 7 bar.

For both types of coolers, automatic temperature controls are available, which use thermostats to control either the water inlet flow or the speed of the fan.

6.4.2 THERMODYNAMIC ASPECTS

The heat flow into the oil is equal to the power loss due to the pressure drop in the oil flow. Thus we get:

$$C_P \Delta T \frac{\mathrm{d}m}{\mathrm{d}t} = Q \Delta P$$

As $\frac{\mathrm{d}m}{\mathrm{d}t} = \rho Q$, we have $\Delta T = \frac{\Delta P}{\rho C_P}$

where

$\frac{\mathrm{d}m}{\mathrm{d}t}$ = mass flow rate	kg s^{-1}
C_p = specific heat	2.1 kJ kg^{-1} K^{-1} (typical value)
T = temperature	K
ρ = density	870 kg m^{-3} (typical value)
ΔP = pressure loss	N m^{-2}
Q = flow	m^3 s^{-1}

Therefore, for a system with a 100 bar pressure loss, the temperature rise will be

$$\Delta T = \frac{\Delta P}{\rho\, C_p} = 5.5\ °C$$

The input power, $W_0 = Q \Delta P \therefore \Delta T = \frac{W_o}{\rho\, C_p\, Q}$

where W_0 is the power input given in Watts.

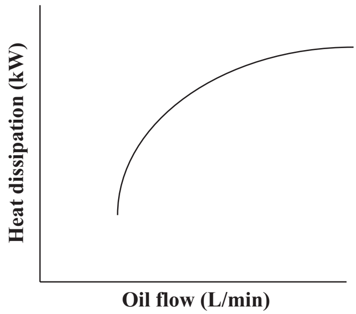

Figure 6.10. Cooler performance characteristics.

6.4.3 COOLER CHARACTERISTICS

The cooling characteristics are usually presented in the form shown in Figure 6.10, which will apply for a particular value of inlet temperature difference. For different values, a correction factor is applied to suit the application.

6.5 RESERVOIRS

If possible, the reservoir design should be such that any entrained air is released before the fluid is passed to the system inlet. A baffle can be used to increase the flow circulation and hence improve the air release. It also reduces the fluid movement due to motion of the reservoir itself.

Passing the return flow through a diffuser in the reservoir reduces the fluid velocity and, by directing the fluid away from the bottom of the reservoir, reduces the re-entrainment of solid contaminant and water from the bottom of the reservoir. Transient changes in the fluid level and the release of air require the fitment of a breather. This must contain a filter that is sized to the minimum requirements of the system. It is preferable that the fluid is filtered during topping-up or replenishing.

SUMMARY

Ancillary equipment basically includes those components or subsystems that are not directly involved in the major functions of the circuit. In many applications, accumulators provide a supplementary flow source that can be used to meet high transient flow demands, compensation for leakage, and absorption for pulsation and shock situations. These employ a volume of pressurized gas, usually nitrogen, which can be used to displace a fixed volume of hydraulic fluid as and when required.

All circuits require the fluid to be filtered, as contaminant particles in the fluid are the biggest cause of unreliability and failure of components and systems. These particles enter the system from the environment and are also generated by the wear process such as the one that occurs in pumps and motors.

Inefficiencies in pumps, motors, and actuators generate heat, which is absorbed by the fluid. It is necessary to be able to estimate the rate of such heat generation in order to install a cooler for the fluid if necessary. The fluid reservoir needs to be designed so as to enable absorbed air to be released and minimize the possibility of contaminant particles re-entering the system.

CHAPTER 7

Circuit Design

7.1 INTRODUCTION

To a very large degree, the main function of hydraulic circuits is to control the flow to one or several actuators as required by the application. There are, however, a variety of methods for controlling flow, some of which act indirectly by using pressure as the controlling parameter.

The circuits discussed in this chapter include the following:

- Directional control and valve configurations
- Velocity controls with constant supply pressure
- Velocity controls with load sensing
- Variable displacement pump controls
- Hydrostatic transmissions
- Load control
- Contamination control

7.2 PRESSURE AND FLOW

Hydraulic systems provide flow from the pump that is directed to one or more actuators (motors) at a pressure level that satisfies the highest demand. Where a single output is being driven, the pump pressure will float to the level demanded by the load. However, even for such simple systems, the method that is employed to provide variable flow needs to be evaluated in order to ensure that best efficiency is obtained. In circuits with multiple outputs, this aspect can be more difficult to evaluate.

For operation at pressures and flows that are lower than the required maximum values, the efficiency of the system will depend on the type of

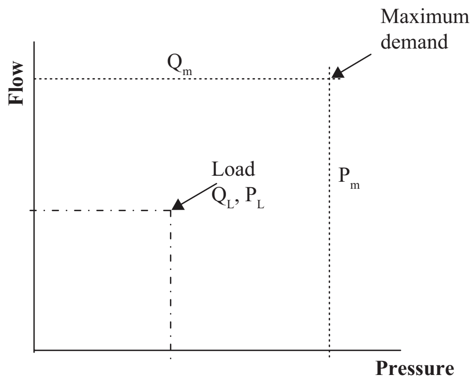

Figure 7.1. Flow and pressure variation.

pump being used (i.e., fixed or variable displacement). This can be represented diagrammatically as in Figure 7.1.

For fixed displacement pump systems, it is clear from Figure 7.1 that excess pump flow will have to be returned to the reservoir so that the power required by the pump is greater than that being supplied to the load. The level of inefficiency incurred is dependent on the ratio between the pressure required by the load and that at the pump outlet which can be controlled at the maximum level by the relief valve or at lower pressures by various types of bypass valves.

For variable displacement pumps, the generation of excess flow can be avoided. However, the level of pump pressure will depend on the method that is used for controlling the displacement, but clearly there is scope for achieving much higher efficiencies than with fixed displacement pumps.

Each of these control methods will require a particular circuit design employing components that have been described in the previous chapters.

7.3 DIRECTIONAL CONTROL

Valves used for controlling the direction of the flow can be put into fixed positions for this purpose, but many types are frequently used in a continuously variable mode, where they introduce a restriction into the flow path.

7.3.1 TWO-POSITION VALVES

A four-way valve with two positions for changing direction of the flow to and from an actuator is shown in Figure 7.2. For supply flow, Q, the actuator velocities will be

$$\text{Extend } U_E = \frac{Q}{A_P};\ \text{Retract } U_R = \frac{Q}{A_A}$$

Here, the actuator areas are A_P for the piston and A_A for the annulus or rod end of the actuator. Hence, $U_R > U_E$ as $A_P > A_R$.

Any external forces (F) that are acting on the actuator rod must be in opposition to the direction of motion. For reversing force applications it will be necessary to apply restrictor control, which will be discussed later in the chapter. These forces will create a supply pressure that is $= \dfrac{F}{A_P}$ or $\dfrac{F}{A_A}$.

Three-way valves are used in applications where only one side of the actuator needs a connection from the supply. A typical example for this is the operation of the lift mechanism on a fork lift truck, as shown in Figure 7.3, where the actuator is lowered under the action of the weight.

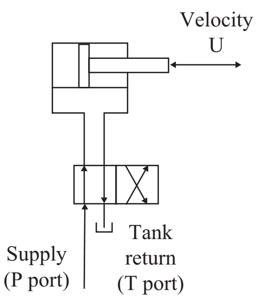

Figure 7.2. Two-position four-way valve.

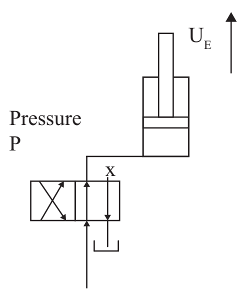

Figure 7.3. Two-position three-way valve.

7.3.2 THREE-POSITION VALVES

Three-position valves have a third, central position that can be connected in different configurations. These variants are described.

7.3.2.1 Closed Center Valves

Closed center valves block all of the four ports as shown in Figure 7.4. This prevents the actuator from moving under the action of any forces on the actuator. The supply flow port is also blocked, which may require some means of limiting the supply pressure unless other valves are being supplied from the same source. The limitation of the supply pressure can be made by appropriate pump controls or by a relief valve.

7.3.2.2 Tandem Center Valves

Tandem center valves block the actuator ports, but the supply is returned to the tank at low pressure as shown in Figure 7.5. If other valves are being supplied from the same source, this type of valve may not be used—unless connected in series.

7.3.2.3 Open Center Valves

Open center valves connect all of the four ports to the tank so that the supply and the actuator pressure are at low pressure as shown in Figure 7.6. This allows the actuator to be free to move under the action of any external forces.

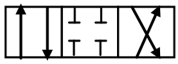

Figure 7.4. Closed center valves.

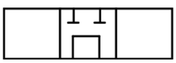

Figure 7.5. Tandem center valves.

Figure 7.6. Open center valves.

Figure 7.7. Open centre valve connected to tank.

Where it is necessary to block the supply flow, the configuration shown in Figure 7.7 can be used.

7.4 LOAD HOLDING VALVES

The radial clearance between the valve and its housing of spool valves is carefully controlled in the manufacturing process to levels of around 2 micron. The leakage through this space, even at high pressures, is small, but for applications where it is essential that the actuator remains in the selected position for long periods of time (e.g., crane jibs where any movement would be unacceptable), valves having metal-to-metal contact have to be used.

Check valves usually employ metal-to-metal contact, but they are only open in one direction under the action of the flow into the valve. For their use in actuator circuits, it is necessary that they are open in both directions as required by the directional control valve (DCV). This function can be obtained from a pilot-operated check valve (POCV) that uses a control pressure to open the valve against reverse flow.

Figure 7.8 shows a typical POCV whereby a pilot pressure is applied onto the piston to force open the ball check valve to allow flow to pass from port 1 to port 2 when the check valve would normally be closed. The ratio of the piston and valve seat areas has to be chosen so that the available pilot pressure can provide sufficient force to open the valve against the pressure on port 1.

The use of a POCV is shown in Figure 7.9, where the external force on the actuator is acting in the extend direction. With the DCV in the center position, the check valve will be closed because the pilot is connected to the tank return line that is at low pressure. Opening the DCV so as to extend the actuator causes the piston side pressure, now connected to the supply, to increase.

When this pressure reaches the level at which the check valve is opened against the pressure generated on the rod side of the actuator by the load force, the actuator will extend. The ratio of the pilot and ball seat diameters needs to be such that the pressure areas cause the POCV to be fully open against the annulus pressure. If the pilot pressure is insufficient to open the valve because of an intensified pressure at the check valve inlet

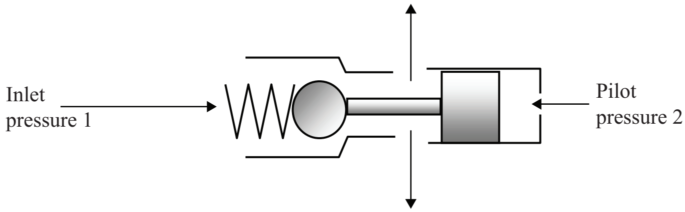

Figure 7.8. Pilot-operated check valve.

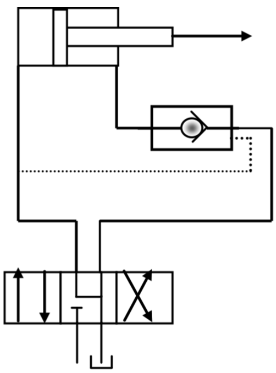

Figure 7.9. Actuator circuit using a POCV.

from the actuator annulus and/or back pressure on the POCV outlet due to restriction in the DCV, oscillatory motion can result.

7.5 VELOCITY CONTROL

The velocity of actuators can be controlled by using a number of different methods. In principle, various methods can be employed for both linear and rotary actuators or motors, but in some cases it may be necessary to refer to the manufacturer's literature for guidance.

7.5.1 METER-IN CONTROL

Meter-in control refers to the use of a flow control at the inlet to an actuator for use with actuators against which the load is in opposition to the direction of movement.

For a meter-in circuit that uses a simple adjustable restrictor valve, selection of the DCV to create extension of the actuator will cause flow to pass through the restrictor into the piston end of the actuator. The required piston pressure, P_p, will depend on the opposing force on the actuator rod. With a fixed displacement pump delivering a constant flow, excess flow from the pump will be returned to the tank by the relief valve at its set pressure, P_{Smax}. Consequently, the available pressure drop, $P_{Smax}–P_p$, will determine the flow delivered to the actuator for a given restrictor opening.

With this system the flow, and hence the actuator velocity, will vary with the load force. For systems where such velocity variations are undesirable, a pressure-compensated flow control valve (PCFCV) can be used. This valve will maintain a constant delivery flow, provided that the pressure drop is greater than its minimum controlled level that is usually in the region of 10 to 15 bar.

Figure 7.10 shows a typical system in which the flow control is bypassed with a check valve for reverse operation of the actuator. If the load force varies considerably during operation, there will be transient changes in actuator velocity at a level that depends on the mass of the load.

For example, when the load force suddenly reduces, the piston pressure will reduce but at a rate that is dependent on the fluid volume and its compressibility and the mass of the load. During the period that the pressure is greater than the required new value, the actuator will accelerate and, as it does so, the piston pressure will fall. The pressure can then fall below the new level and deceleration results and damped oscillations can occur as shown in Figure 7.11.

In some situations the mass of the load can be such as to cause problems of cavitation and overrunning, because the pressure falls transiently to a level at which absorbed air is released. If the pressure falls low

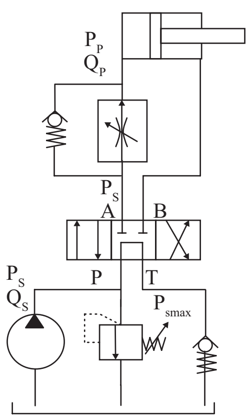

Figure 7.10. Meter-in control for actuator extension.

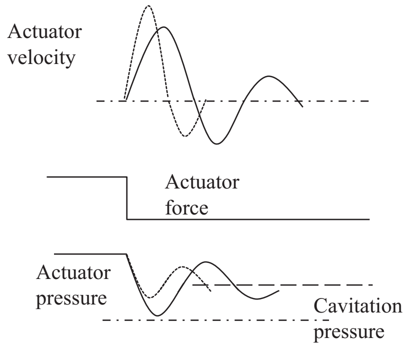

Figure 7.11. Pressure and velocity variations with meter-in control.

enough, the fluid will vaporize. Both of these phenomena are referred to as cavitation, and noisy operation and damage to the components can be the result.

The changes represented by dotted lines in Figure 7.11 are for a low-inertia load that creates a lower magnitude of the pressure oscillations and, hence, reduces the possibility of cavitation.

A check valve having a spring cracking pressure that is high enough to suppress cavitation is sometimes used as shown in Figure 7.11, but this has the disadvantage of increasing the pump pressure and thus reducing the efficiency and increasing the heating effect on the fluid.

7.5.2 METER-OUT CONTROL

For overrunning load forces and/or those with a large mass, meter-out control is used where the actuator outlet flow during its extension passes through the restrictor or PCFCV as shown in the circuit of Figure 7.12.

The flow control operates by controlling the actuator outlet pressure at the level required to oppose the forces exerted on the actuator by the load and by the piston pressure, which is the same as that of the pump. This prevents cavitation from occurring during transient changes arising from load force variations or due to forces that act in the same direction as the movement (i.e., pulling forces).

This system can, however, cause high annulus pressures to occur from the intensification of the piston pressure together with the pressure created by pulling forces. Further, when compared to meter-in, the rod and piston seals have to be capable of withstanding high pressures that may require a higher cost actuator to be used.

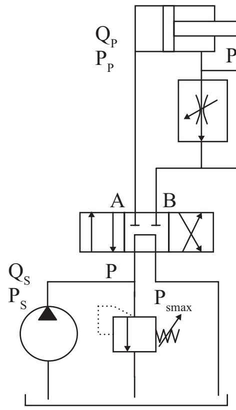

Figure 7.12. Meter-out control.

7.5.3 BLEED-OFF CONTROL

For the fixed displacement pump system shown in Figure 7.13, excess flow is bled off from the supply so that the pump pressure is now at the same level as that required at the actuator piston.

Bleed-off control is therefore more efficient than meter-in and meter-out because of the lower pump pressure. However, as for meter-in, it cannot be used with pulling loads and, moreover, it can only be used to control one actuator at a time from the pump. This is in contrast to meter-in and meter-out where several actuators can be supplied by a single pump as shown in Figure 7.14.

Meter-in and meter-out controls can be supplied from a variable displacement pump that is operated with a constant pressure control (pressure compensated), which reduces the power wastage that is inherent with a fixed displacement pump. This is demonstrated by making a comparison of the efficiencies as follows:

For meter-in control the power efficiency, $\eta = \dfrac{Q_P P_P}{Q_S P_S}$.

For a pressure-compensated pump the power efficiency, $\eta = \dfrac{Q_P P_P}{Q_P P_S} = \dfrac{P_P}{P_S}$ as $Q_S = Q_P$.

Thus, referring to Figure 7.1, the pump flow is always equal to that of the load, Q_L. The pump is still capable of achieving the maximum demand, which is referred to as the "corner power" of the pump. The fixed displacement pump operates at this rating continuously because of the use of the relief valve to control the flow to the actuator.

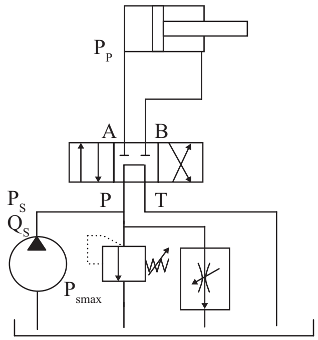

Figure 7.13. Bleed-off control.

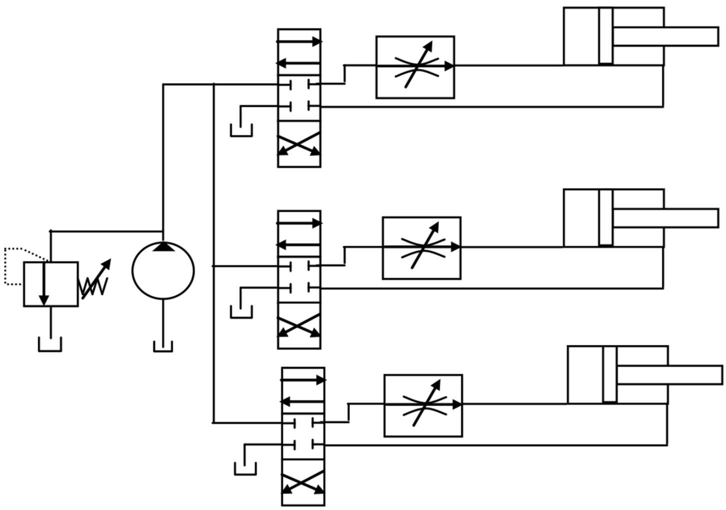

Figure 7.14. Multiple actuator circuit with meter-in control.

The flow control methods described in this section are usually preset in a system that is being used on a continuous basis such as for a production machine (e.g., injection moulding), where possibly the operations are being carried out sequentially. It would normally be expected that the duration of, say, actuator movement is small in relation to the overall cycle time so that the power losses are relatively small. Where a continuously variable flow control is required, alternative components and circuits need to be considered.

7.5.4 FOUR-WAY VALVE RESTRICTIVE CONTROL

Four-way valves can be used to control the velocity of actuators by introducing both meter-in and meter-out restrictions into the flow path as shown in Figure 7.15.

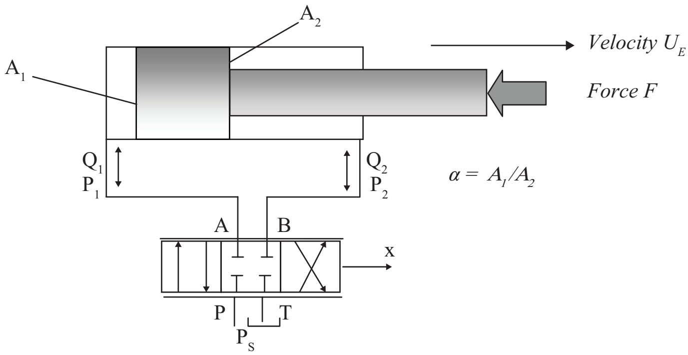

Figure 7.15. Four-way valve.

The valve position is fully variable and can be controlled by the following:

- Direct lever manual input
- Hydraulic pilot operated from an input lever or joystick
- Proportional solenoid

For proper design of the circuit and appropriate component selection, it is necessary to analyze the system in order to determine the actuator pressures as a function of the actuator load force. In this system, the pressure drop across each valve land needs to be considered as a function of the flow and the valve position or opening.

7.5.4.1 Analysis of the Valve/Actuator System

7.5.4.1.1 Actuator Extending

Valve flow characteristics

For the parameters shown in Figure 7.15, the valve flows are given by

$$Q_1 = K_1\, x \sqrt{P_s - P_1}$$

$$K_1 = R_1 \sqrt{\frac{2}{\rho}}$$

where

R_1 = effective metering area coefficient.

For zero pressure in the return line

$$Q_2 = K_2 x \sqrt{P_2}$$

$$K_2 = R_2 \sqrt{\frac{2}{\rho}}$$

where
R_2 = effective metering area coefficient (e.g.,= $C_Q \pi d$ for annular ports). This may be different to R_1 in some valves.

Also

$$U_E = \frac{Q_1}{A_1} = \frac{Q_2}{A_2}$$

These equations give

$$P_S - P_1 = \left(\frac{A_1}{A_2}\frac{K_2}{K_1}\right)^2 P_2 = \left(\frac{a}{R_S}\right)^2 P_2$$

The ratio A1/A2 is given by α.

For a valve spool that has symmetrical metering, $R_S = 1$.

Then

$$P_S - P_1 = a^2 P_2 \tag{7.1}$$

Equation 7.1 relates the pressures P_1 and P_2 for the valve connected to the unequal area actuator as represented in Figure 7.16. The flow from the annulus is lower than that to the piston because of its smaller area, which results in a lower pressure drop in the valve.

Actuator force

$$P_1 A_1 - P_2 A_2 = F$$

$$\therefore P_1 = \frac{P_2}{a} + \frac{F}{A_1} \tag{7.2}$$

Equation 7.2 relates the actuator pressures for a given actuator force with a positive force that acts in the opposing direction to the extending movement of the actuator. This relationship is shown in Figure 7.17, and

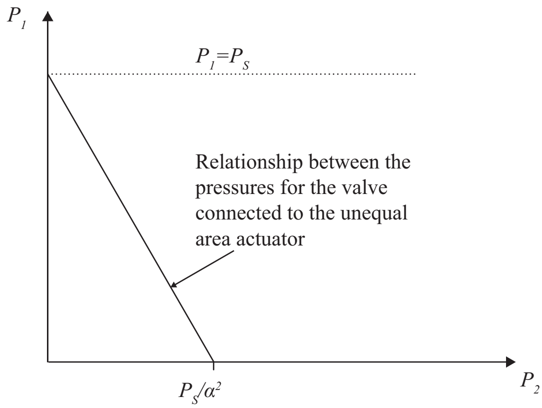

Figure 7.16. Valve pressures during extension.

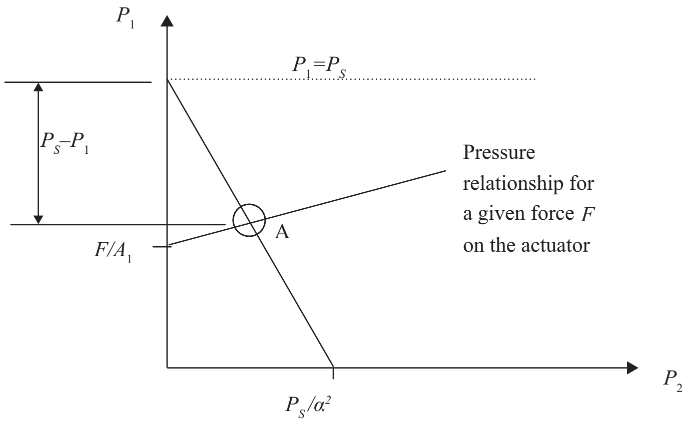

Figure 7.17. Interaction between the flow and the force characteristics during extension.

the pressures for the valve controlling the actuator are determined by the intersection of the lines for the actuator and the valve at point A.

The values of these pressures can be obtained from Equations 7.1 and 7.2 for the valve and actuator. The force acting on the actuator can be in either direction.

Using a force ratio defined by

$$R = \frac{F}{P_s\ A_1}$$

we get

$$\frac{P_1}{P_s} = \frac{(1 + R a^3)}{(1 + a^3)} \tag{7.3}$$

and

$$\frac{P_2}{P_s} = \frac{a\ (1 - R)}{(1 + a^3)} \tag{7.4}$$

Variations of the force will change the position of the line in Figure 7.17 that represents the actuator and, hence, change the position of the point of intersection, *A*, that will change the pressure levels according to Equations 7.3 and 7.4.

For a value of $R = 1$, the force will be the maximum available and is the stall force for the system. For a given application, the actuator area is chosen to provide the desired stall force at the chosen supply pressure. This will then determine the actuator pressures that are obtained for any other value of the force.

7.5.4.1.2 Actuator Retracting

For retraction of the actuator the valve position is reversed, the annulus now being connected to the supply and the piston to the return line. Following the same method for a symmetrical valve ($K_1 = K_2$) as for the actuator extension, we get

$$Q_2 = K_1 x \sqrt{(P_S - P_2)}$$

and

$$Q_1 = K_1 x \sqrt{P_1}$$

and as

$$Q_1 = a\,Q_2$$

we have

$$P_2 = P_S - \frac{P_1}{a^2} \tag{7.5}$$

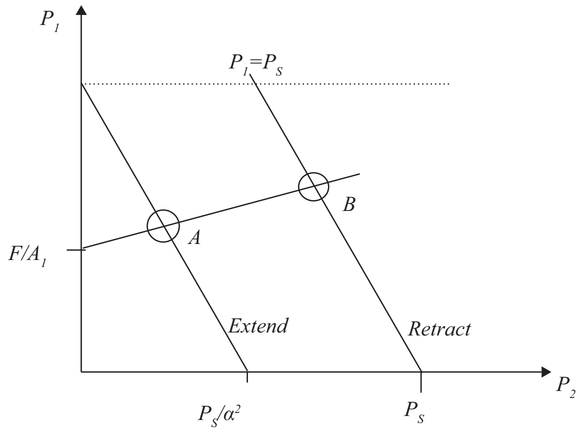

Figure 7.18. Actuator retracting.

This relationship between P_1 and P_2 for the retracting actuator is shown in Figure 7.18, where point *B* is the operating condition for retraction of the actuator. It can be seen that when the valve is reversed there is a pressure change between points *A* and *B*.

The equations for the pressures for actuator retraction are

$$\frac{P_2}{P_s} = \frac{a^3 - a\ R}{1 + a^3} \tag{7.6}$$

$$\frac{P_1}{P_s} = \frac{a^2(1 + a\,R)}{(1 + a^3)} \tag{7.7}$$

The actuator velocity is determined from the flow equation for the valve using the appropriate values for the pressures P_1 and P_2. The selection of a valve having the necessary capacity is determined from the maximum required velocity condition.

It should be noted that neither of the pressures can be less than zero; so, for example, during extension, the maximum value of the force ratio, *R*, that is permissible in order to avoid cavitation of the flow to the piston is given by

$$\frac{P_1}{P_S} = 0 = \frac{1 + Ra^3}{1 + a^3}$$

$$\therefore R > -\frac{1}{a^3}\ \textit{to prevent cavitation}$$

for which condition

$$P_2 = P_S a \frac{\left(1+ {}^{1}\!/_{a^3}\right)}{(1+a^3)}$$

$$\therefore P_2 = \frac{P_S}{a^2}$$

7.5.4.2 Valve Sizing

The valve size needs to be selected such as to provide the flow required for the specified velocity and force conditions. In general, it is desirable to operate the system close to its condition of maximum efficiency, which can be obtained by considering the power transfer process.

For simplicity, consider an equal area actuator for which the power transmitted to the load is given by

$$\text{Power}, E = P_m Q_m$$

$$\therefore \; E = \left(P_{s\max} - 2\Delta P_V\right) Q_m$$

Here, P_m is the load pressure difference, Q_m is the actuator flow, and ΔP_v is the valve pressure drop across each land. From the orifice equation, Q_m can be expressed as

$$Q_m = C_q A \sqrt{\frac{2\,\Delta P_v}{\rho}}$$

$$\therefore \; \Delta P_V = \frac{1}{2}\left(\frac{\rho}{C_q^2 A^2}\right) Q_m^2 = k\,Q_m^2$$

where A = the valve orifice area.

Hence

$$E = (P_{s\max} - 2\Delta P_V) = (\,P_{s\max} - 2kQ_m^2)Q_m$$

For maximum power at the load

$$\frac{dE}{dQ_m} = 0 = P_{S\max} - 6kQ_m^2$$

and $\therefore kQ_m^2 = \frac{P_{s\max}}{6}$, giving the total valve pressure drop $2\Delta P_V = \frac{P_{s\max}}{3}$.

Therefore, the load pressure drop for maximum power, $P_m = P_{s\max} - 2\Delta P_V \therefore P_m = \frac{2}{3} P_{S\max}$.

Based on this analysis, an approximation is often applied from the maximum power condition to give the best combination of valve and actuator sizes for a given supply pressure such that the force at maximum power is chosen to be two-thirds of the stall thrust.

The actuator size and the supply pressure can be selected to provide this stall thrust, and the valve size is then determined on the basis of the maximum required value of $\frac{Q}{\sqrt{\Delta P_V}}$.

Valves are rated by determining the flow at a total fixed pressure drop across both ports and for a given valve position. Thus the flow at any other pressure drop is given by

$$Q = Q_{\mathrm{R}} \sqrt{\frac{\Delta P}{\Delta P_{\mathrm{R}}}}$$

where

Q_{R} = rated flow
ΔP_{R} = rated pressure drop

ΔP_{R} is sometimes given as the pressure drop through only one of the metering lands.

7.5.4.3 Valves with Nonsymmetrical Metering

The use of valves in which the metering is nonsymmetrical, normally by machining metering notches of different shapes, provides two major advantages over symmetrical valves which are as follows:

- The possibility to increase the maximum negative or overrunning force during extension (meter-out control).
- The avoidance of the pressure change during reversal of the valve—as seen from Figure 7.19, the valve characteristic is a single line for both directions of movement.

The dotted line in Figure 7.19 shows the characteristics where the valve metering ratio, R_{S}, has the same value as the actuator area ratio, α.

Figure 7.20 shows the output velocity U plotted against the force ratio R in comparison with the particular case of an equal area ratio actuator for which $\alpha = 1$. The equal area actuator has a symmetrical characteristic and is often used in servo systems for this reason. The dotted line shows the effect of asymmetrical metering where the valve-metering ratio is the same as the area ratio of the actuator.

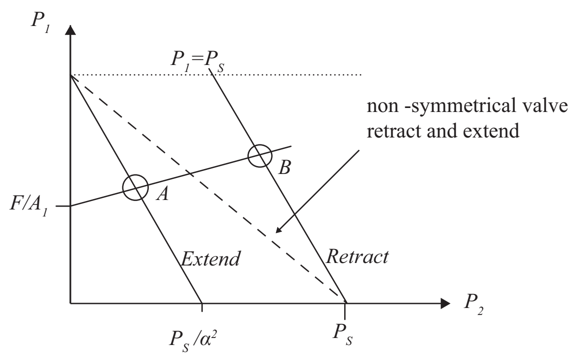

Figure 7.19. Nonsymmetrical valve metering.

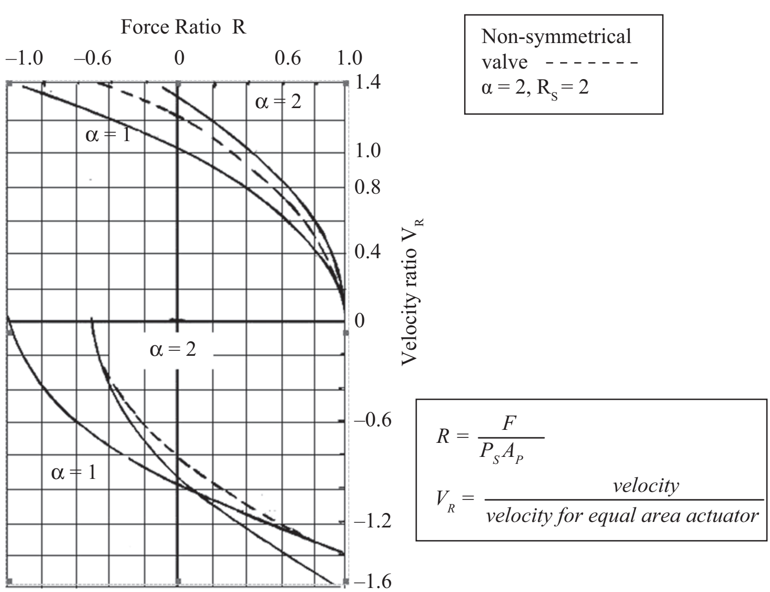

Figure 7.20. Load locus of velocity ratio against force ratio.

7.5.5 BYPASS CONTROL WITH FIXED DISPLACEMENT PUMPS

7.5.5.1 Open Center Valves

Open center valves that use a central bypass combine the use of bleed-off control with the directional control function. A circuit is shown in Figure 7.21 for two valves connected in series. Operation of a valve creates an outlet flow from the valve, the magnitude of which will depend on the valve opening and the required outlet pressure as shown graphically in Figure 7.22. When the valve is moved to an extreme position on either side, the central bypass will be closed and all of the pump flow passed to the outlet.

On starting a loaded actuator, when the valve is opened the pump pressure will rise to a level that is determined by the amount of opening.

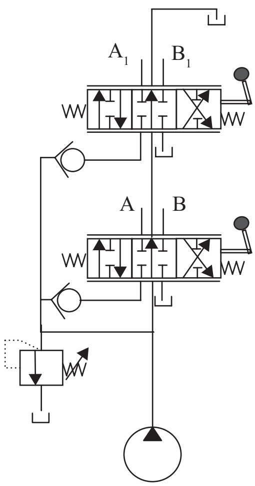

Figure 7.21. Central bypass valves in series.

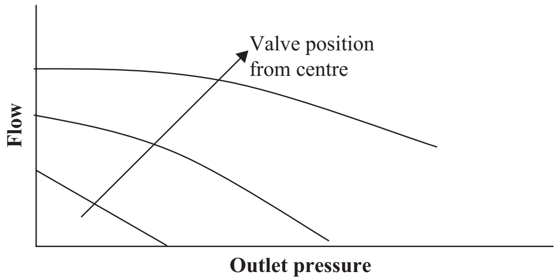

Figure 7.22. Bypass valve characteristics.

The load check valves are placed in the circuit to prevent the actuator reversing should the pump pressure be less than that required by the actuator. The relief valve is required to protect the pump and system from excessive pressures.

The valve is load sensitive in that, for a given valve position, the flow reduces with increases in the outlet pressure. When both valves are operated simultaneously, there will be interaction causing the flows to vary with changes in either of the outlet pressures.

Machining a notch in an overlapped valve land often forms the valve metering area. The typical valve configuration in Figure 7.23 for three valve spool positions shows this feature in the central bypass land. By using different numbers of notches and/or their sizes different metering characteristics can be obtained from the same size of valve spool and also differential metering related to the selected flow direction, which includes meter-out control for negative loads.

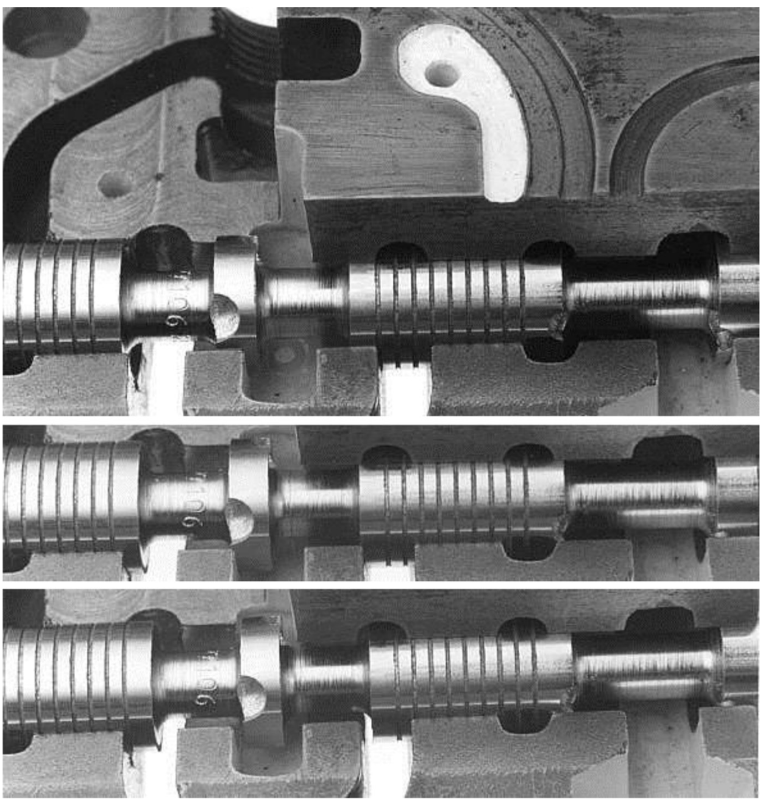

Figure 7.23. Central bypass valve with notched metering edges in three positions.

7.5.5.2 Closed Center Valves with Load Sensing and Pressure Compensation

Closed center valves can be used with fixed displacement pumps whereby excess flow is bypassed from the pump output under the action of a spring loaded valve that senses the pressure drop between the pump and the load pressure. This creates a pump pressure that is about 20 bar above that of the load that provides the efficiency benefits from bleed-off control. This also provides pressure compensation and thus avoids the load sensitivity obtained from central bypass valves.

For use in circuits where several actuators are to be controlled, valves are available in which load sensing can be incorporated to operate the bypass valve using a pilot signal from the highest load pressure.

Such a circuit is shown in Figure 7.24 in which individual pressure compensators are included to maintain constant flow through the valve that is supplying the actuator that is at the lowest pressure. The highest load pressure

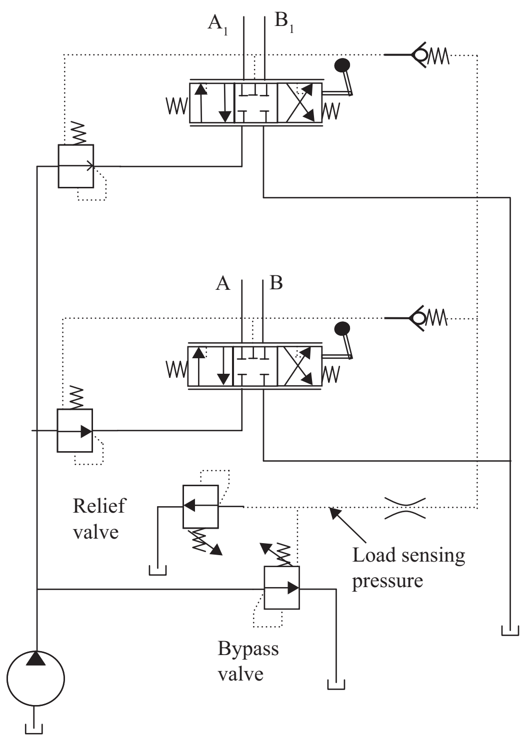

Figure 7.24. Pressure-compensated bypass valve with load sensing.

is sensed by the check valves and passed to the bypass-regulating valve, the opening of which is such as to deliver the flow required by both valves.

The valve can be used to supply several actuators and, provided the pump flow is adequate, the flow to any selected service will not vary with changes in the required outlet pressure, that is, it is not load sensitive. When the load sensing pilot pressure exceeds the setting of the relief valve, it will open and the resulting flow will depress the pilot pressure on the bypass valve causing it to open so that the system flow is reduced and, hence the pressure.

7.6 VARIABLE DISPLACEMENT PUMP CONTROL

There is a range of controls available for variable displacement pumps that are provided by most manufacturers to suit various application requirements. In general, the pump displacement mechanism is operated by a hydromechanical servo that usually has the possibility of accepting an electrical input. In some cases, the displacement is sensed by an electric transducer for closed loop control, which would normally be referred to as an electrohydraulic control system.

7.6.1 LOAD SENSING

For load-sensing control, the valve shown in Figure 7.24 can be used where the pilot signal is supplied to a valve that operates the pump displacement mechanism. Pressure limiting is usually incorporated so as to limit the maximum pump pressure. In applications where it is required to limit the input torque and power to the pump in order to prevent stalling of the prime mover, the servo system will, for a constant pump speed, provide a constant output power.

The circuit shown in Figure 7.25 controls the displacement to maintain the difference between the pump and the load sense pilot pressures. If the force from the pressure difference on valve B is higher than that set by the spring, the pump displacement is reduced by the actuator until it has fallen to the correct value. Valve A is the pressure compensator that senses pump pressure, which, if it is too high, causes the pump displacement to be reduced in order to reduce the pressure.

7.6.2 POWER CONTROL

For the control of the pump power, Figure 7.26 shows a circuit that is used by Eaton for this purpose. This incorporates a pressure feedback from

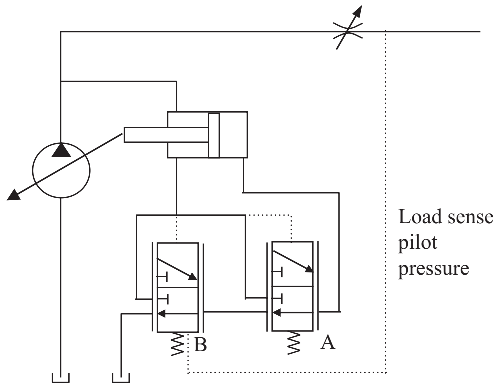

Figure 7.25. Variable displacement pump pressure limiting and load sensing control.

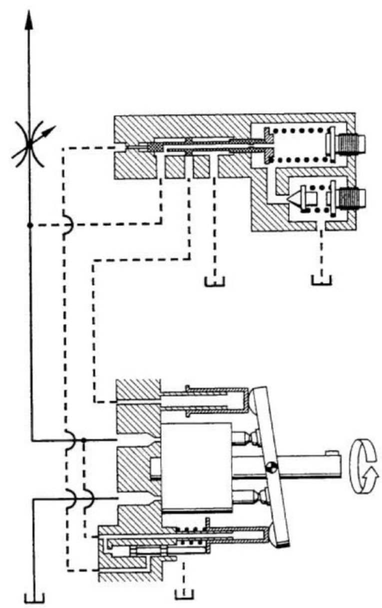

Figure 7.26. Constant power control.
Source: Courtesy Eaton.

the hydraulic potentiometer that is varied by the pump displacement, the force from which is opposed by that created by the pump outlet pressure as can be seen from Figure 7.26. If the pump pressure increases to too high a value, its displacement is reduced by the action of the spool valve to a lower level that corresponds to that demanded for a constant power characteristic.

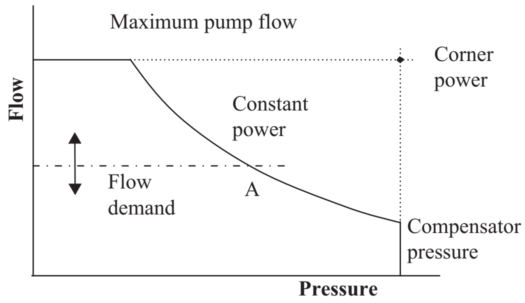

Figure 7.27. Pump operating characteristics.

Figure 7.27 shows the operating envelope for a pump that has all three controls so that for a given flow demand, determined by the valve setting, the maximum available pressure will correspond to that at point A. As the flow demand is altered, the maximum available pressure will be controlled to the constant power line. Thus, point A traverses the maximum power line, the maximum pressure being limited by the compensator setting.

Should the pump outlet pressure exceed this value the compensator will reduce the pump stroke and, hence, the flow. This control is able to reduce the pump displacement to zero. With a blocked outlet port, the pump stroke reduces to a level when the output flow is just sufficient to make up the pump leakage at the set pressure.

7.6.3 ACCUMULATOR CHARGING

Accumulators are frequently used in circuits to provide a supplementary flow in applications where the demanded flow varies in a cyclic manner. In this situation, the accumulator can provide flow at a higher level than that available from the pump thus allowing a reduced capacity pump to be used.

This operation requires a circuit to:

- Increase the pump flow when the accumulator requires recharging.
- Reduce the pump flow when there is sufficient fluid volume in the accumulator.

Figure 7.28 shows a circuit that is used to control the pump displacement to satisfy these requirements.

After a period when the accumulator has discharged its volume and the unloading valve has closed, the pump displacement will increase and

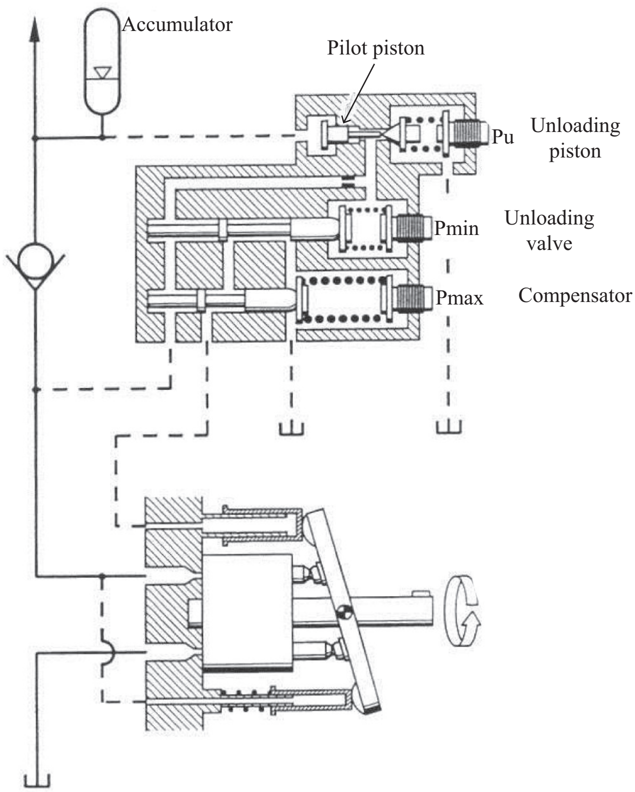

Figure 7.28. Accumulator charging circuit.
Source: Courtesy Eaton.

charge the accumulator. When the pump pressure reaches the value, P_U, set by the unloading pilot spring pre-load, the poppet will open. The associated flow through the restrictor will cause the unloading valve to move against its spring and pressurize the pump stroke piston to reduce the pump displacement.

The accumulator pressure will keep the unloading pilot poppet open because of the force on the pilot piston. The closed circuit check valve will maintain pressure in the accumulator circuit whilst the pump flow has reduced to zero. When the accumulator is required to discharge flow to the system, its outlet pressure will reduce until it has reached the level that causes the unloading pilot to close. The area of the pilot piston is greater than that of the poppet sensing area and, consequently, the level of pressure required to close the poppet can be typically 15 percent lower than that required by the pump pressure to open it.

7.7 HYDROSTATIC TRANSMISSIONS

7.7.1 PUMP CONTROLLED SYSTEMS (PRIMARY CONTROL)

Hydrostatic transmissions connect the actuator directly to the supply pump without using any valves for restrictive metering, the control of velocity being made by the displacement of the pump, and, in the case of motors, additionally by the motor displacement. The flow from the actuator is returned to the pump inlet thus avoiding the need for a large capacity boost pump.

The pressure level rises to that required to drive the actuator against the load. Consequently, the pump output flow can only be used to drive a single actuator or multiple actuators that are constrained to move at the same velocity (e.g., coupled motors and actuators attached rigidly to the same moving component)

Figure 7.29 shows the circuit for a hydrostatic transmission used to drive a hydraulic motor that includes:

- The provision of boost flow to make up for the external losses from the pump and motor. The check valves connect the boost input to the low pressure (unloaded) side of the loop. This applies for both the pump driving the motor and, for overrunning conditions, when the motor is driving the pump (e.g., winch lowering).

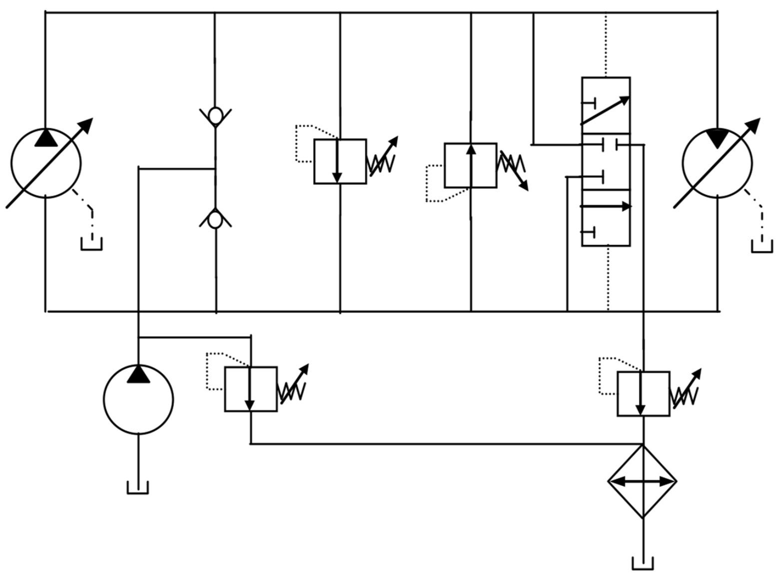

Figure 7.29. Rotary hydrostatic transmission circuit.

- Crossline relief valves to prevent excessive pressures. The flow is passed to the low pressure side in order to maintain the flow into the pump inlet.
- The extraction of fluid from the loop using a purge valve to provide increased cooling. This flow needs to be controlled as it has to be made up from the boost flow.
- Variable motor displacement control for systems requiring a higher speed range at reduced torque.

7.7.2 MOTOR BRAKE CIRCUIT

The operation of spring loaded brakes is incorporated into the hydraulic system (e.g., winches, swing drives) by directing system pressure to the brake actuator as shown in Figure 7.30.

The pressure required to release the brake actuators needs to be higher than the boost pressure. Often it is necessary to fit a reducing valve to limit the maximum pressure at the actuators if they have a lower rated pressure than the system.

7.7.3 LINEAR ACTUATOR TRANSMISSIONS

For linear actuator systems using equal area actuators, the circuit is similar in principle to that for rotary systems as can be seen from Figure 7.31, which shows a basic circuit.

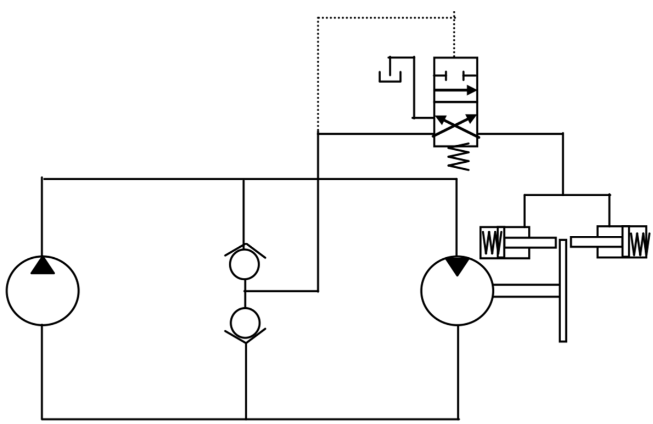

Figure 7.30. Motor brake circuit.

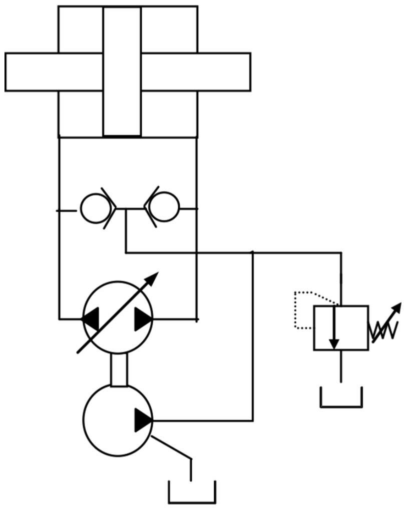

Figure 7.31. Linear actuator hydrostatic transmission circuit.

7.7.4 MOTOR CONTROLLED SYSTEMS (SECONDARY CONTROL)

Secondary control systems [7, 8] operate at a constant supply pressure that can be provided by a pressure-compensated pump, the motor displacement being controlled so as to maintain constant speed in a closed loop system as shown in Figure 7.32.

The use of secondary control provides some advantages over the conventional hydrostatic system which include:

- The storage of energy in the accumulator from regenerative (e.g., overrunning) loads.
- Accuracy and dynamic performance.
- Use with multiple motors (ring main systems).

7.7.5 DIGITAL OPERATION OF THE VALVES OF PUMPS AND MOTORS

The application of simple on–off valves for pumps and motors is discussed in Section 2.7. Using the method for operating the valves described in Section 2.7 for the radial piston motor of Figure 2.12 (shown here as Figure 7.33), cylinder pressures are used to open or close the valves with

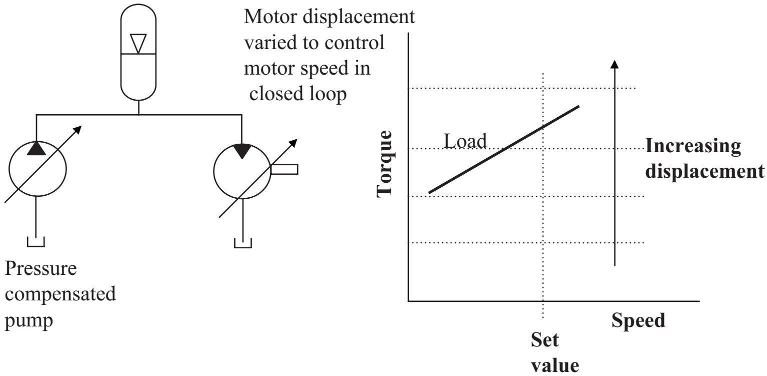

Figure 7.32. Secondary control system.

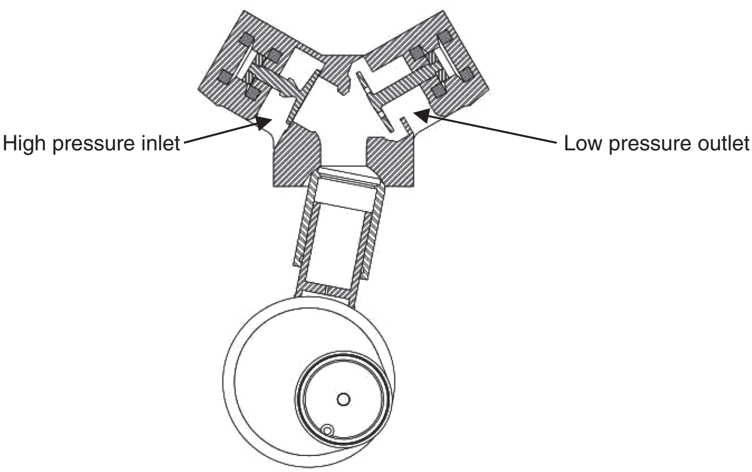

Figure 7.33. Radial piston motor with digitally operated valves (as in Figure 2.13).
Source: Courtesy Diinef.

signals from the shaft position used to lock the valves in the final position. In this section, an operating procedure is considered that allows the movement of the valves at any shaft angle as desired [9].

The poppet valves shown in Figure 7.34 are used instead of the radial piston motor valves shown in Figure 7.33 and work in conjunction with the control circuit shown in Figure 7.35, the function of which is described in the following:

7.7.5.1 Motor

a. Flow from high pressure into the cylinder during the motor power stroke

At TDC following the exhaust stroke from BDC, the low pressure valve will close when the pilot valve (Figure 7.35) has changed position under the action of a signal from the shaft position sensor and consequently put high pressure onto the low pressure valve pilot area A_3 (Figure 7.34). The timing signal can be set to give the best motor performance in terms of, for example, minimizing pressure and flow spikes. So the low pressure valve could be set to close before TDC, which will create an increase in cylinder pressure. This will reduce the rate at which flow enters the cylinder when the high pressure valve is opened thus reducing noise from possible pressure fluctuations.

The pilot pressure to the high pressure valve is changed from high to low by the action of the pilot valve. The area A_2, which is connected to the high pressure, will cause the high pressure valve to be opened when the pilot pressure falls to low pressure.

b. Flow from the piston exhaust stroke to low pressure

It is required to take the cylinder flow out of the low pressure port from BDC to TDC by closing the high pressure poppet valve and opening the low pressure valve. If the high pressure valve is closed before BDC, the cylinder pressure will reduce the amplitude of flow and pressure spikes passing to the low pressure line.

c. Displacement control

As has been mentioned in Section 2.7, the motor displacement can be reduced by connecting a cylinder or cylinders to the low pressure line continuously thus reducing the overall motor torque and power. For finer control of displacement, it is possible with this active pilot control to delay the opening of the high pressure valve until after the

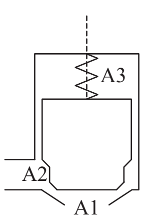

Figure 7.34. Poppet valve for cylinders.
Source: Courtesy Diinef.

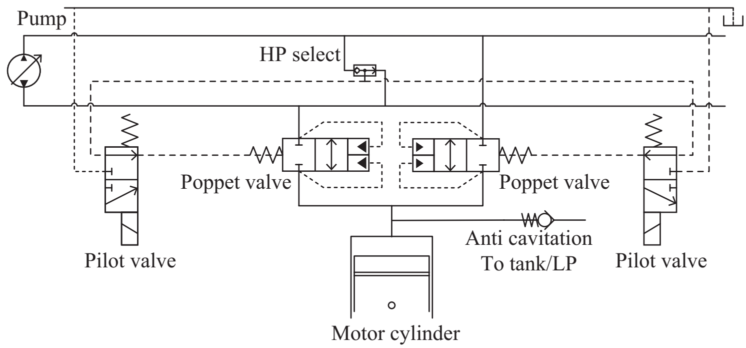

Figure 7.35. Pilot circuit for active control of the motor cylinder valves. *Source*: Courtesy Diinef.

TDC position thus proportionately reducing the torque from that cylinder. Using this method means that the cylinder torque can be infinitely variable between maximum and zero. The low pressure line has an anti-cavitation valve so that additional flow can be passed directly to the cylinder from the tank or low pressure line should the cylinder pressure fall to a low level.

It is also possible to open the low pressure valve before the BDC position to give the same effect in reducing the torque.

7.7.6 EXAMPLE APPLICATION OF A HYDROSTATIC TRANSMISSION FOR POWER TRANSMISSION IN WIND TURBINES

In wind turbines it is necessary to convert the power from the low speed turbine into electric power from the generator at speeds up to 1,500 rpm. Conventionally, this is performed either by speed increasing gearboxes having ratios in the range 25 to 100 or by other types of mechanical drives. Various forms of hydraulic power transmissions are available, and this example describes a particular type on which some development work has been carried out using the circuit shown in Figure 7.36 in which the pump is directly connected to the turbine. In the design proposed by Chapdrive AS [10], the pump is integrated into the turbine bearing assembly.

In the circuit, the load (turbine torque) drives the pump and the turbine speed is controlled by altering the motor displacement, provided the generator is connected to the grid and there is sufficient torque from the generator. Aside from starting up when the generator speed is increased

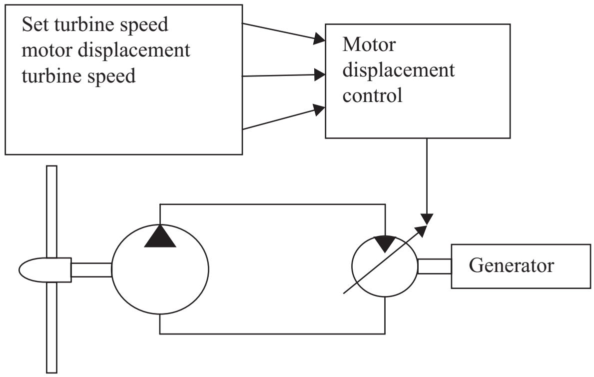

Figure 7.36. Wind turbine hydrostatic power transmission system.
Source: Courtesy Chapdrive AS.

from zero, the generator speed can be considered to be constant if the generator is of the synchronous type.

The main benefits of hydrostatic transmissions over most of the mechanical drives can be summarized as follows:

1. Relatively higher ratio of power over weight.
2. Variable speed control of the turbine by changing the motor displacement [10, 11].
3. The direct connection of synchronous generators to the grid, which avoids the requirement for a frequency converter and power amplifier.

The major aspect of the hydrostatic transmission for use in wind turbines is reflected mostly by the large rotary inertia of the turbine rotor, which could be expected to have a considerable effect on the dynamic performance of the application. The pressure level is proportional to the turbine torque, but the control method used in the operational circuit is similar to the secondary control circuit in Section 7.7.4.

The dynamic performance of this system is analyzed in Section 10.9.3 with a worked example in Section 11.20 that presents performance calculations for the wind turbine circuit.

7.8 PILOT OPERATED VALVE CIRCUITS

The pilot operation of valves has a wide range of application for the functional control in many machines and three control valve functions are described in this section.

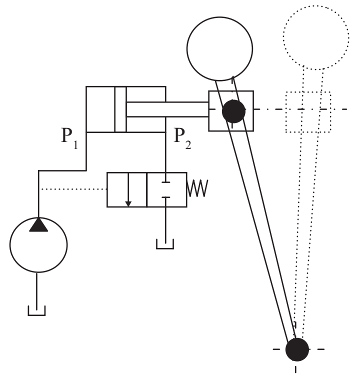

Figure 7.37. Load control circuit using counterbalance valves.

7.8.1 LOAD CONTROL VALVES

In actuator systems when working with overrunning loads (e.g., crane booms, winches), it may be necessary to provide meter-out control in order to protect the system. Counterbalance valves are used for this purpose, as they require inlet pressure to the actuator to cause them to open. These act as closed-loop systems and the dynamic behavior during operation is complex and can often create oscillations of the load movement.

The circuit shown in Figure 7.37 is for operating a load that goes over center and thus reversing the direction of the force acting on the actuator. As this force becomes negative it will initially cause the actuator velocity to increase, which will reduce the inlet pressure P_1. When this pressure falls below the set value of the valve, it will close so causing the actuator outlet pressure, P_2, to increase until it is at a sufficient level to resist the load force.

7.8.2 PUMP UNLOADING CIRCUIT

The double pump system shown in Figure 7.38 uses a pilot operated valve to connect one of the two pumps to the tank when the load pressure exceeds the value set by the spring. The outlet pressure from this pump will then be zero when the check valve will close and prevent flow from the high-pressure pump returning to the tank.

7.8.3 SEQUENCE CONTROL

The circuit given in Figure 7.39 shows how pilot operated valves can be used to sequentially control the extensions of the two actuators. When one has reached the end of its stroke, the increase in pump pressure that follows opens the sequential control valve thus allowing the second actuator to move.

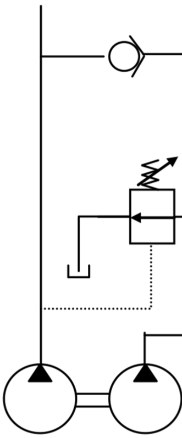

Figure 7.38. Double pump system with unloading valve.

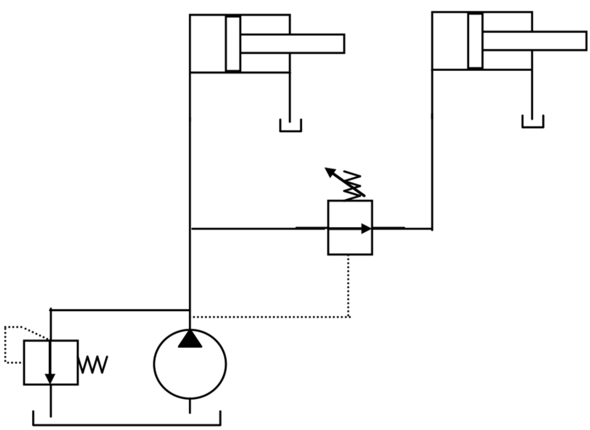

Figure 7.39. Sequence valve used for operation in a press circuit.

7.9 CONTAMINATION CONTROL

Filters can be incorporated into hydraulic circuits in a number of ways, some of which are described in this section. Two basic points in the selection of a circuit depend on where the filter(s) are to be situated (high or low pressure) and the use of a filter bypass. The filter circuit shown in Figure 7.40 uses a high-pressure filter with a bypass check valve so that if the filter starts to become blocked, fluid can still be supplied to the system.

In situations where it is imperative that sensitive components are protected from contaminated fluid then the alternative approach is to not use a bypass. The system relief valve protects the pump from overpressures and, at the same time, the security of the filter housing. It is essential that in systems not employing a bypass the filter condition is monitored carefully.

It is recommended that filters are not sited in areas of high vibration and, if possible, to put them in positions where the flow is constant. Both of these issues relate to the retention of contaminant in the filter that could otherwise become free to pass through the filter.

In order to maintain a constant flow through the filter, the relief valve can be placed downstream of the filter as shown in Figure 7.41a.

Figure 7.41b shows a low-pressure filter circuit that provides a cheaper alternative to high-pressure types. These filters are often of the spin-on type, which protect the reservoir and pump inlet from particles generated in the system but do not protect the system from particles generated in the pump. Spin-on filters can be sensitive to flow transients and pressure shocks.

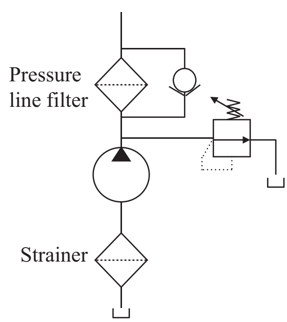

Figure 7.40. High pressure filter circuits.

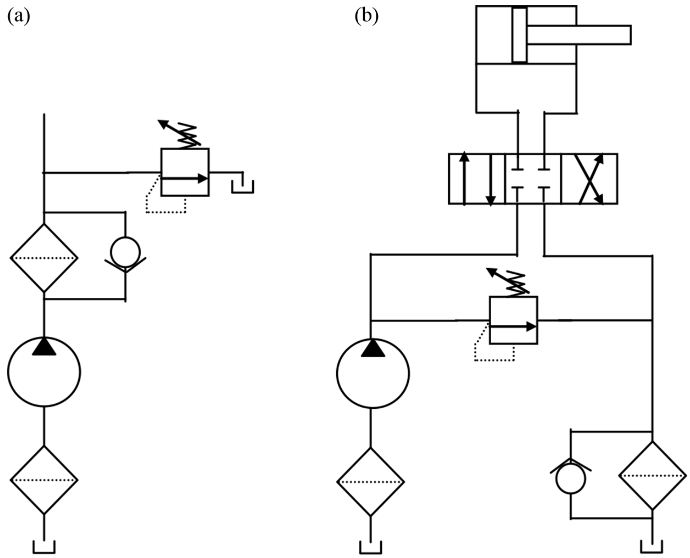

Figure 7.41. High and low pressure filter circuits.

The reservoir can be a major source of contamination, and suction filters provide protection to the pump. However, because of the pressure loss in filters these can usually only provide filtration at levels of around 75 microns, the maximum allowable pressure loss being of the order of 0.2 bar. Higher pressure losses will cause aeration and cavitation of the fluid, and subsequent damage to the pump.

Off-line filtration can be used to circulate fluid from the reservoir on a continual basis. This system does not protect the components from contaminants created by a component failure but it does protect the hydraulic system on a long-term basis. A pump is fitted that can also be used to top up, or fill, the reservoir with fluid that is pre-filtered by the off-line system that circulates the fluid. The reservoir breather should also be fitted with a filter that has the same rating as that of the main system.

The breather caps for the reservoir pass air in and out of the reservoir particularly in systems having a number of single-ended actuators. The breather should, therefore, be fitted with a filter that has the same level of filtration as the main system in order to prevent the ingress of large particulate contaminants.

SUMMARY

The selection of hydraulic components for use in a given application is determined by their ability to meet the required specification within the desired cost framework. A variety of components can be arranged to fulfill a given function by using different circuit configurations as the fluid power system designer has the freedom, within the constraints set by the preferences of the machine builder and/or the user, to select components of their choice.

This freedom makes it difficult to summarize circuit design; however, the designer needs to be able to justify the circuit on the basis of technical considerations. This chapter therefore describes and, where applicable, evaluates a variety of circuit options that can be used for the range of functions generally encountered in the application of fluid power systems.

CHAPTER 8

FLOW PROCESSES IN HYDRAULIC SYSTEMS

8.1 INTRODUCTION

This chapter is concerned with the analytical methods for the evaluation of:

- Pressure losses in pipes and small clearances.
- Pressure/flow characteristics of restrictors.
- Fluid momentum forces on valves.

8.2 FLUID PROPERTIES

Fluid density, ρ, is the mass per unit volume and has units of kg m^{-3}. The variation in the density of most hydraulic fluids with temperature is relatively small for the normal operating temperature usually encountered in hydraulic systems. For hydraulic oils the value can be taken as **870 kg/m^3**.

Fluid viscosity
The viscosity of the fluid can be referred to by:

i. **Dynamic viscosity, μ**, for which the units are **Ns m^{-2}**.
   The technical literature often uses other units. Typically, **Poise (P)** where **1 P = 0.1 Ns m^{-2}** and the **centipoise (cP)** where **1 cP = 10^{-3} Ns m^{-2}**.
ii. **Kinematic viscosity ν**. This is more commonly quoted in technical literature and data sheets and is equal to the dynamic viscosity divided by the density.

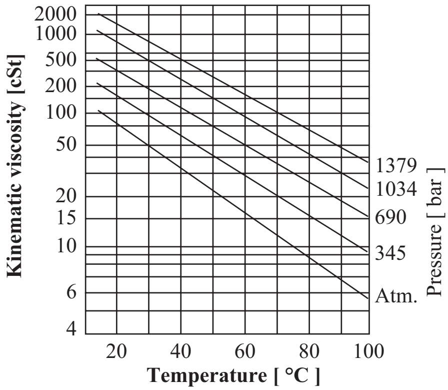

Figure 8.1. Oil viscosity variation with pressure and temperature.

The units for kinematic viscosity are $\mathbf{m^2\,s^{-1}}$ but it is usually quoted in **centi Stoke (cSt)**

$$\mathbf{1\ cSt = 10^{-6}\ m^2\ s^{-1}.}$$

The kinematic viscosity of a typical hydraulic oil varies with temperature and pressure as shown in Figure 8.1, and it can be seen that the viscosity varies considerably with changes in fluid operating temperature.

8.3 FLOW IN PIPES

The effect that viscosity has on the flow through pipes results in a pressure loss that is determined by the Reynolds number, the length to diameter ratio of the pipe, and the dynamic velocity head of the fluid. This pressure loss results in a loss of energy that is dissipated as heat in the fluid that may require cooling in some situations. Consequently, the loss of efficiency thus created has to be evaluated in order to establish the appropriate size of pipes, fittings, and valves and the overall influence of the machine duty cycle on the fluid temperature.

The pressure loss in pump inlet, or suction, pipes is especially important as the flow is usually supplied from a reservoir at atmospheric pressure. This means, that in order to avoid cavitation and aeration of the fluid, the pressure loss must not exceed 0.2 bar in most cases and in some instances it may be necessary to boost the pump inlet.

Air is normally absorbed in the fluid in the reservoir at atmospheric pressure and aeration occurs when the pressure falls below this value. When the fluid pressure falls to its vapor pressure, vapor bubbles begin to form. The problems caused by both of these processes, referred to as cavitation, are:

- Surface damage to the components
- Noise
- Damage and possible failure in pumps and motors
- Loss of control

The flow equation for the pressure loss, Δp, in a pipe of diameter d is given by:

$$\Delta p = 4f\frac{L}{d}\frac{1}{2}\rho u_m^2$$

This equation is derived in the Appendix (Equation 8A5) for laminar flow where, f is the friction factor which, for laminar flow, is given by:

$$f = \frac{16}{R_e}$$

where R_e is the Reynolds number which $= \dfrac{\rho u_m d}{\mu} = \dfrac{u_m d}{\upsilon}$ and $u_m = \dfrac{4Q}{\pi d^2}$.

It is usual to employ the equation to determine the pressure loss in a pipe because, for a Reynolds number that is greater than 2,000, the flow becomes turbulent and the velocity distribution is no longer parabolic as for laminar flow. For turbulent flow, the friction factor, f, has a different relationship with the Reynolds number and it is usual to obtain its value from the Moody chart shown in Figure 8.2.

Numerical example

Pipe flow $Q = 72\text{L}/\min = \dfrac{72}{6\times10^4} = 1.2\times10^{-3}\,\text{m}^3/\text{s}$

Pipe diameter $d = 25\text{mm}$; area $A = \pi\times\dfrac{(25\times10^{-3})^2}{4} = 4.91\times10^{-4}\,\text{m}^2$

Flow velocity $u_m = \dfrac{Q}{A} = \dfrac{1.2\times10^{-3}}{4.91\times10^{-4}} = 2.44\text{m/s}$; $R_e = \dfrac{u_m d}{\nu}$

Consider values for the fluid viscosity of 70 and 20 cSt (i.e., 70×10^{-6} and $20\times10^{-6}\,\text{m}^2\,\text{s}^{-1}$)

For the higher viscosity, the value of R_e is 871, which is laminar when $f = \dfrac{16}{871} = 0.018$ (this value can also be obtained from Figure 8.2).

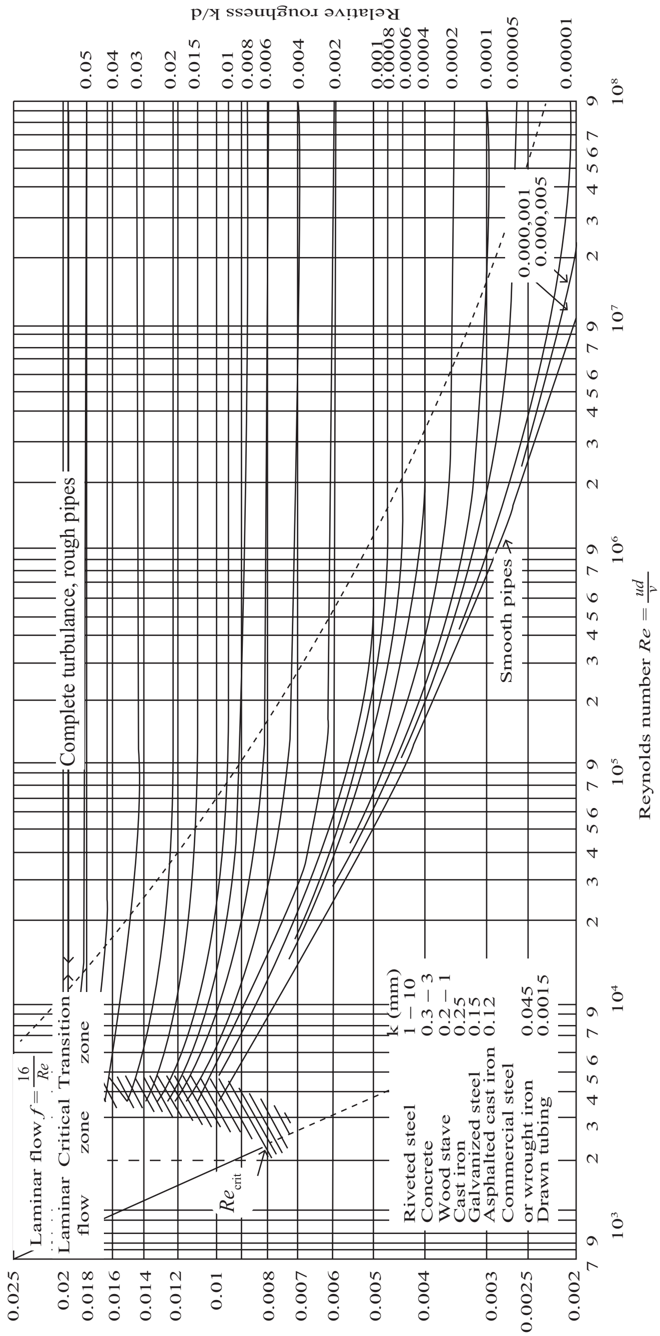

Figure 8.2. Moody chart for determining the friction factor.

For the lower viscosity, R_e = 3,050 giving a value of f = 0.011 from Figure 8.2.

Considering a 10 m pipe length gives $4\frac{L}{d}\frac{1}{2}\rho u_m^2 = 4.14\times10^6$.

The pressure losses are 0.75 and 0.46 bar, which would be considered low for flow in the high pressure side of a hydraulic system but would be unacceptable for the inlet supply to a pump.

8.4 LAMINAR FLOW IN PARALLEL LEAKAGE SPACES

For fully developed steady laminar flow, the velocity distribution with y across the gap between parallel plates is parabolic assuming that there is no transverse flow across the width of the flow path. In Figure 8.3, y is the vertical distance from the center line in the element of fluid having a width w and length L. The analysis for this situation is similar to that used for the pipe in the Appendix where the total viscous force acting on the rectangular fluid element is given by:

$$\mu\frac{\mathrm{d}u}{\mathrm{d}y}wL = -\Delta Pwy$$

$$\therefore -\int_{u_1}^{0}\mathrm{d}u = \frac{\Delta P}{\mu L}\int_0^h \mathrm{d}y$$

This gives for the maximum velocity, u_1, at the center:

$$u_1 = \frac{\Delta P}{\mu L}\frac{h^2}{2} \tag{8.1}$$

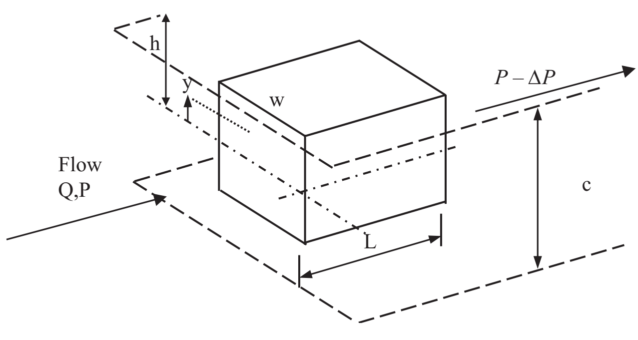

Figure 8.3. Flow between parallel plates.

and

$$u = \frac{\Delta P}{2\mu L}(h^2 - y^2) \tag{8.2}$$

From Equation 8.2 the flow through the parallel path is given by:

$$Q = \int_0^h 2wudy = \frac{w\Delta P}{\mu L}\left[h^2 y - \frac{y^3}{3}\right]_0^h = \frac{c^3 w}{12\mu L}\frac{\Delta P}{L} \tag{8.3}$$

Equation 8.3 is used to determine the leakage flow through small clearances between hydraulic components (e.g., spool valves, pump pistons, and seals).

8.5 ORIFICE FLOW

For flow through a sharp-edged orifice, as shown in Figure 8.4, the maximum velocity and lowest pressure are at the *vena contracta* that occurs downstream of the orifice itself, the area of the *vena contracta* being smaller than that of the orifice. Assuming that the upstream velocity is very low in relation to that in the orifice, the maximum velocity is obtained from the Bernoulli energy equation thus:

$$u = \sqrt{\frac{2\,(P_1 - P_2)}{\rho}} \tag{8.4}$$

Flow, Q, through an orifice can be related to the area and the velocity. Generally:

$$Q = uA_o \tag{8.5}$$

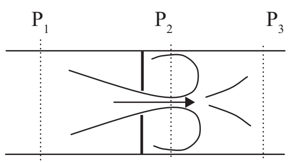

Figure 8.4. Sharp-edged orifice.

where A_o = orifice area.

Equation 8.5 gives the theoretical flow to which the discharge coefficient, C_d, is applied as in Equation 8.6 to give:

$$Q = C_d A_o \sqrt{\frac{2(P_1 - P_2)}{\rho}} \tag{8.6}$$

The pressure level, P_2, is not easily measured and it is usual to relate the flow through the orifice to the pressure drop from inlet to outlet and use a flow coefficient C_q.

$$Q = C_q A_o \sqrt{\frac{2(P_1 - P_3)}{\rho}}$$

Note that usually the orifice area is very much smaller than the upstream vand downstream areas in which case:

$$C_q = C_d$$

For orifice Reynolds numbers higher than around 2,000, the value of C_d for the sharp-edged orifice tends to be 0.62 but for valves having different geometrical configurations the value will depend on the particular shape. A typical variation of C_q with Reynolds number is shown in Figure 8.5.

8.6 VALVE FORCE ANALYSIS

For flow passing through valve openings, the change in velocity is often high and the associated change in momentum has to be created by pressure forces in the fluid. The fluid generally enters the valve at a low velocity so that for a particle of fluid having a mass *m*, and an exit velocity *u*, the change in the momentum is given by *mu*.

The force required to create this change in momentum is given by:

$$f = m\frac{\mathrm{d}u}{\mathrm{d}t}$$

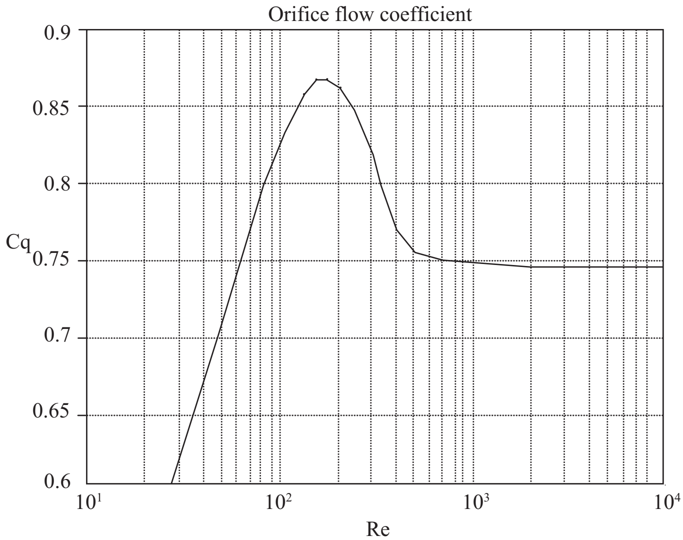

Figure 8.5. Flow coefficient variation with Reynolds number.

Now the mass is related to both the flow Q and the time period dt so that $m = \rho Q \mathrm{d}t$. This gives for the rate of change of momentum:

$$f = \rho Q \mathrm{d}t \frac{\mathrm{d}u}{\mathrm{d}t} = \rho Q u$$

where u is the change in velocity.

8.6.1 POPPET VALVES

8.6.1.1 Momentum Force

A force analysis for the poppet valve shown in Figure 8.6a needs to consider the pressure force on the valve face that will vary with the velocity of the fluid in the small valve opening.

With the valve in the closed position:

$$\text{force} = P_1 A_1$$
$$A_1 = \frac{\pi d^2}{4}$$

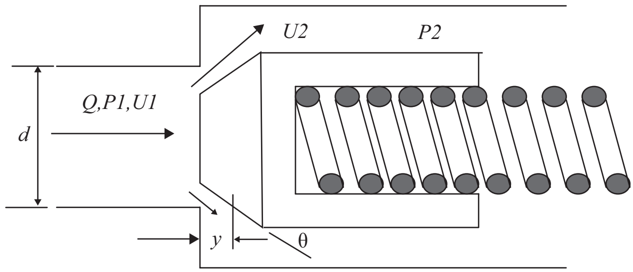

Figure 8.6a. Single-stage poppet-type relief valve.

For the valve open as in Figure 8.6b:

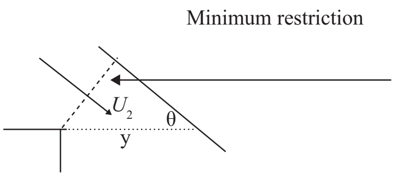

Figure 8.6b. Valve in open position.

There is a reduction in the pressure as the velocity increases through the valve and increases the fluid momentum. The pressure force on the valve, which will depend on the pressure distribution on the valve face, is expressed by:

$$F = \int pdA$$

The upstream velocity will be low and hence the rate of increase in the axial momentum of the fluid through the valve is given by $\rho QU_2 \cos\theta$. The pressure force on the fluid that is required to increase the fluid momentum is given by:

$$P_1A_1 - \int pdA = \rho QU_2 \cos\theta$$

Assuming that the downstream pressure P_2 is zero, the pressure force on the face of the valve must equal the spring force. Thus:

$$\int p dA = P_1 A_1 - \rho Q U_2 \cos\theta = C + ky \tag{8.7}$$

where C is the spring compression force when the valve is closed, k is the spring stiffness, and $A_1 = \pi d^2/4$.

The term $\rho Q U_2 \cos\theta$ is referred to as the flow force (or Bernoulli force), which acts in the direction to close the valve and is additive to the force from the spring. As a consequence of this, in order to increase the flow through the valve, the upstream pressure force $P_1 A_1$ has to increase.

8.6.1.2 Valve Flow

The restriction created by the valve reduces the pressure of the fluid. Assuming that P_2 is zero the valve flow equation gives:

$$Q = C_Q \pi d y \sin\theta \sqrt{\frac{2}{\rho} P_1}$$

$$\therefore y = \frac{Q}{C_Q \pi d \sin\theta \sqrt{\frac{2}{\rho} P_1}} \tag{8.8}$$

also

$$U_2 = \sqrt{\frac{2}{\rho} P_1} \tag{8.9}$$

8.6.1.3 Valve Pressure/Flow Characteristics

Combining Equations 8.7, 8.8, and 8.9 gives:

$$P_1 = \frac{C}{A_1} + K_1 \frac{Q}{\sqrt{P_1}} + K_2 Q \sqrt{P_1} \tag{8.10}$$

where:

$$K_1 = \frac{k}{A_1 \pi d C_Q \sin\theta \sqrt{\frac{2}{\rho}}}$$

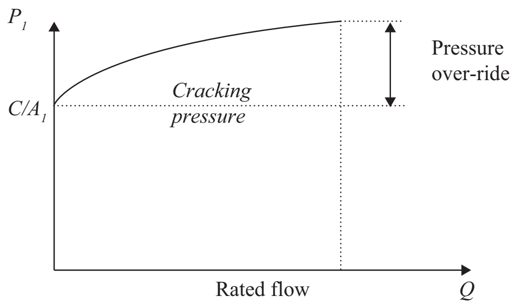

Figure 8.7. Valve pressure flow characteristic from Equation 8.11.

and

$$K_2 = \frac{\cos\theta}{A_1}\sqrt{2\rho}$$

$$\therefore Q = \frac{P_1 - {}^{C}\!/\!_{A_1}}{\frac{K_1}{\sqrt{P_1}} + K_2\sqrt{P_1}} \tag{8.11}$$

It is seen from Equation 8.11 that the force coefficient K_2 (due to the flow force) has an additive effect to the force coefficient K_1 that arises from the spring stiffness. Increasing the values of these coefficients reduces the level of flow that the valve will pass for a given upstream pressure P_1. The valve flow to pressure characteristics obtained from Equation 8.11 are shown typically in Figure 8.7 where the zero flow intercept is referred to as the "cracking pressure".

8.6.2 SPOOL VALVES

In spool valves, there is a peripheral flow of fluid through the annulus formed by the valve land and the port.

The flow from the high pressure inlet to the spool valve in Figure 8.8 creates a high velocity in the outlet metering annulus so that there is a resulting force on the spool tending to close the valve. This arises from the difference between the pressure force on the spool land on the left side, which is due to the inlet pressure, P_S, and that on the right hand spool

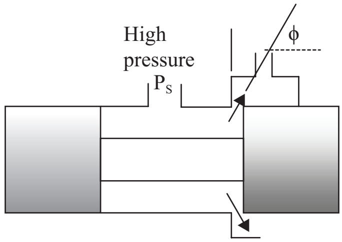

Figure 8.8. Spool valve.

land, which is less because the pressure is reducing radially outwards due to the increasing fluid velocity. This pressure force is equal to the axial component of the momentum change so for a velocity *U*, the momentum force is given by:

$$\rho Q U \cos\varphi$$

The flow through the restriction is given by:

$$Q = A_0 U = C_Q A_0 \sqrt{\frac{2P_S}{\rho}}$$

where A_0 is the valve opening area = πdx and x is the valve displacement.

Thus, the momentum force = $2\pi d C_Q \times P_S \cos\varphi$. The angle φ is usually taken as 69° (von Mises) but this can vary as a function of the valve clearance and other dimensional parameters [13].

SUMMARY

Pascal's law for a fluid at rest states that the pressure at any point, neglecting head effects, is the same at any other point in the fluid. Generally speaking, in hydraulic systems, this law can be applied and the term "hydrostatic" refers to this condition. However, when the fluid is moving this may not apply locally due to the effects of the fluid viscosity, which creates energy losses that increase with the velocity. The fluid velocity also influences the pressure forces that act on components.

It is important that the system designer is aware of the background to the pressure/flow relationship in pipes, as there can be significant energy losses if their diameter is too small for the particular application. This relationship is based on the original theoretical and laboratory work carried out by Reynolds, and an analysis of the pressure losses for piping is given in the Appendix.

For laminar flow, the pressure loss is inversely proportional to the fluid viscosity, but this effect diminishes as the flow becomes more turbulent for Reynolds numbers >2,000 when it tends to be proportional to the fluid velocity squared. The leakage of flow through small clearances such as those found around the pistons in pumps and motors can be similarly analyzed to provide a basis for evaluating the variation in their volumetric efficiency with the fluid viscosity, pressure, and rotational speed.

The pressure loss that occurs with flow in restrictions provides a principal method of controlling flow in hydraulic systems. For turbulent flow, the flow through a spool valve is found to vary proportionally with its opening and with the square root of the pressure drop across the spool. This process is similarly important to the operation of pressure control valves such as relief valves and those used in many control circuits.

The level of the pressure force that acts on valves is strongly affected by the change in the momentum of the fluid as it passes through the valve. This process is difficult to analyze and is frequently evaluated using computational fluid dynamics (CFD) software. A simplified analytical method is discussed in this chapter, which provides a basic understanding of the process that can be used to estimate these forces for some design purposes.

APPENDIX

This appendix provides the background to the equations that are used for determining the relationship between flow and pressure loss in pipes [13].

For fully developed, steady laminar flow, the theoretical velocity distribution is parabolic, which is verified by experiment. Consider a

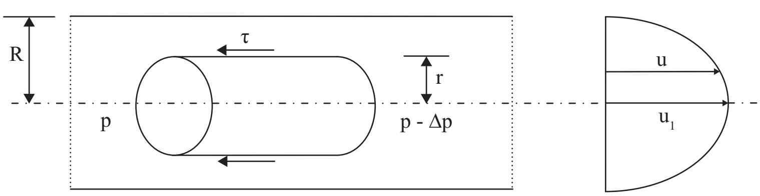

Figure 8A.1. Flow through a pipe.

cylindrical element of fluid of radius r and length L. The total viscous force acting on the cylindrical element is given by:

$$2\pi rL\tau = -2\pi rL\mu \frac{du}{dr}$$
$$\mu = fluid\ dynamic\ viscosity$$
$$\tau = shear\ stress\ in\ the\ fluid$$

For steady motion of the fluid (i.e., no change in the velocity):

$$\pi r^2 \Delta p = -2\pi rL\mu \frac{du}{dr}$$

and:

$$-\int_{u_1}^{0} du = \frac{\Delta p}{2\mu L}\int_{0}^{R} rdr$$

The velocity is zero when $r = R$ and is a maximum when $r = 0$. The solution of the integrals gives:

$$u_1 = \frac{\Delta p}{4\mu L} R^2 \qquad (8A1)$$

For any radius r in the range $0 \le r \le R$ where $u_1 \ge u \ge 0$ the velocity distribution is given by:

$$u = \frac{\Delta p}{4\mu L}(R^2 - r^2) \qquad (8A2)$$

The total volume flow Q is given by:

$$Q = \int_{0}^{R} 2\pi rudr = \frac{\Delta p}{8\mu L}\pi R^4$$

The mean velocity, u_m, is given by:

$$u_m = \frac{Q}{\pi R^2} = \frac{\Delta p}{8\mu L} R^2 = \frac{u_1}{2} \qquad (8A3)$$

Therefore:

$$\Delta p = \frac{8\mu u_m L}{R^2} = \frac{32\mu u_m L}{d^2} \tag{8A4}$$

It is normal to relate this to the Reynolds number R_e:

$$R_e = \frac{\rho u_m d}{\mu}$$

By rearranging Equation 8A4 we get:

$$\Delta p = \frac{32\mu u_m}{d} \times \frac{L}{d} \times \frac{\rho u_m}{\rho u_m} = 4\left(\frac{16\mu}{\rho u_m d}\right) \times \frac{L}{d} \times \frac{1}{2}\rho u_m^2 = 4f\,\frac{L}{d}\frac{1}{2}\rho u_m^2 \tag{8A5}$$

The parameter, *f*, is referred to as the friction factor, and for laminar steady flow, it is given by $\frac{16}{R_e}$.

The term $\frac{1}{2}\rho u_m^2$ is referred to as the dynamic or velocity head.

CHAPTER 9

Operating Efficiencies of Pumps and Motors

9.1 INTRODUCTION

The leakage losses in pumps and motors are derived analytically using the method developed by Wilson [14] that is based on the laminar flow equations in Chapter 8, which produces a simple model for the volumetric efficiency. The mechanical losses are based on assuming that viscous forces between moving components are proportional to the viscosity and the relative velocity and that there is a coulomb friction force that is proportional to the pressure.

9.2 MECHANICAL AND VOLUMETRIC EFFICIENCY

The power loss mechanism is the same for both pumps and motors. However, the efficiency of pumps and motors is slightly different because of the way the losses affect the power flow in the units. In pumps, to achieve a given output power in the fluid, the power losses have to be added to the theoretical power at the input shaft so that as the overall efficiency reduces, the input power has to be increased. In motors, the power losses detract from the input fluid power and the result is that the same level of losses gives a different efficiency.

Denoting the volumetric, or leakage, flow loss by Q_S and the mechanical or frictional torque loss by T_m gives the efficiencies as:

For a pump:

$$\eta_v = \frac{Q_t - Q_S}{Q_t} = 1 - \frac{Q_S}{Q_t} \tag{9.1}$$

$$\eta_m = \frac{T_t}{T_t + T_m} = \frac{1}{1 + {T_m}/{T_t}} \tag{9.2}$$

And for motors:

$$\eta_v = \frac{Q_t}{Q_t + Q_S} = \frac{1}{1 + {Q_S}/{Q_t}} \tag{9.3}$$

$$\eta_m = \frac{T_t - T_m}{T_t} = 1 - \frac{T_m}{T_t} \tag{9.4}$$

In these equations, the ideal flow is denoted by Q_t, and the ideal torque by T_t. For either unit, the overall efficiency is given by:

$$\eta_o = \eta_v \, \eta_m$$

These losses can be determined from test results but in order to establish their variation over the range of operating conditions a theoretical approach is required such as that described in the next section.

9.3 ANALYSIS OF THE LOSSES

9.3.1 *THEORETICAL PERFORMANCE*

For the ideal pump, or motor, with no losses, the ideal or theoretical flow, Q_t depends only on the pump geometric capacity, *D,* and its rotational speed, ω, hence:

$$Q_t = D\omega$$

Equating the mechanical power with the fluid power gives:

$$T_t\,\omega = P\,Q_t$$

Hence, the ideal or theoretical torque T_t depends only on the displacement and the differential pressure across the unit.

Thus:

$$T_t = D\,P$$

At the system design stage, the theoretical equations can be used but when units have been selected actual efficiency values can be obtained from the technical literature.

Most mathematical models for pump and motor steady state performance are based on the work of W.E. Wilson [13].

9.3.2 VOLUMETRIC FLOW LOSS

The volumetric flow loss arises from leakage through the various clearance spaces between the moving components in the unit. The degree of these losses will vary between the different types of units, but in this analysis, the overall leakage is assumed to be dependent on the fluid pressure and viscosity as obtained in Section 8.4.

Thus, the laminar flow equation for flow through a slot gives:

$$Q_S = \frac{h^3}{12\mu} w \frac{dp}{dx}$$

Here μ is the fluid dynamic viscosity. If the clearance, h, is assumed to be constant, for a given unit this equation can be reduced to:

$$Q_S = C_1 \frac{P}{\mu}$$

And the ratio,

$$\frac{Q_S}{Q_t} = \frac{C_1}{D}\left(\frac{P}{\mu\omega}\right) \tag{9.5}$$

Thus, it is seen that the flow ratio, $\frac{Q_S}{Q_t}$, is a function of a nondimensional parameter, $\frac{P}{\mu\omega}$, that will tend to be zero for low pressures, high speeds, and fluids with a high viscosity, that is, the leakage will be zero. Whilst the value of C_1 can be obtained from test data, these results usually tend to have significant levels of scatter because:

- The clearances are not likely to be constant during the rotating cycle, for example, a piston can be eccentrically disposed and/or tilted in a cylinder bore (e.g., for line contact conditions the leakage increases 150 percent over that when the piston is concentric within the cylinder).
- The component sliding velocities are not constant during the rotating cycle.
- The fluid viscosity will vary through the leakage path due to frictional heating and pressure.
- There may be a variation in the leakage with the rotational speed.

In many unit types, the leakage loss has been shown to vary as $P^{1.5}$. In the above equations, P is usually taken as the difference between the inlet and outlet pressures for a pump or motor. However, in piston units much of the leakage is to the case drain, which may not be at the same pressure as the pump inlet or motor outlet.

9.4 MECHANICAL LOSS

Mechanical losses occur as a result of viscous and coulomb friction. Viscous friction arises from fluid shear in the clearance spaces that result from the relative speed of the various moving components and is referred to as speed-dependent friction. Viscous friction torque is expressed by:

$$T_v = C_2 \, \mu \, \omega$$

And the torque ratio,

$$\frac{T_v}{T_t} = \frac{C_2}{D}\left[\frac{\mu \, \omega}{P}\right] \tag{9.6}$$

This torque ratio is a function of a nondimensional parameter, $\frac{\mu\omega}{P}$, that is the inverse of that for the flow ratio. The value of the coefficient

C_2 can be obtained from tests, but the results will also be dependent on the same parameters identified as being likely to have an influence on C_1.

Coulomb friction torque is proportional to load and independent of speed:

$$T_f = C_3 P \tag{9.7}$$

and the torque ratio:

$$\frac{T_f}{T_t} = \frac{C_3}{D} \tag{9.8}$$

The value of the coefficient C_3 depends on the frictional conditions between the various sliding components. In some gear units, the coulomb friction torque loss can vary with $P^{1.5}$, and in all types of units, the relative frictional loss increases at low speeds.

The total mechanical loss,

$$T_m = T_V + T_f \tag{9.9}$$

9.5 UNIT EFFICIENCY

9.5.1 VOLUMETRIC EFFICIENCY

For pumps, the variation in the volumetric efficiency with the nondimensional parameter $\dfrac{P}{\mu\omega}$ can be obtained by substituting Equation 9.5 in 9.1, which gives:

$$\eta_v = \frac{Q_t - Q_S}{Q_t} = 1 - C_s\left[\frac{P}{\mu\,\omega}\right] \tag{9.10}$$

where $C_S = \dfrac{C_1}{D}$

Here, it is seen that as the pressure is increased and the speed and fluid viscosity are reduced, the volumetric efficiency is reduced. Pumps are tested under given operating conditions and Equation 9.10 can be used to estimate the volumetric efficiency under different operating conditions.

9.5.2 MECHANICAL EFFICIENCY

The mechanical efficiency can be obtained similarly by substituting Equations 9.4, 9.7, and 9.9 into Equation 9.2 to give:

$$\eta_m = \frac{T_t}{T_t + T_V + T_f} = \frac{1}{1 + C_f + C_v\left[\frac{\mu\,\omega}{P}\right]} \tag{9.11}$$

where $C_f = \frac{C_3}{D}$ and $C_V = \frac{C_2}{D}$

The mechanical efficiency from Equation 9.11 is seen to vary in the opposite manner with pressure, speed, and viscosity to the volumetric efficiency.

9.5.3 OVERALL EFFICIENCY

The overall efficiency, η0, which is the product of the volumetric and mechanical efficiencies will vary with the nondimensional parameter as shown in Figure 9.1.

Figure 9.1 shows that pumps (and motors) have an optimum operating condition, which is a combination of the pressure, speed, and fluid viscosity. This condition can be theoretically determined by considering the total power loss in the unit, which is given by:

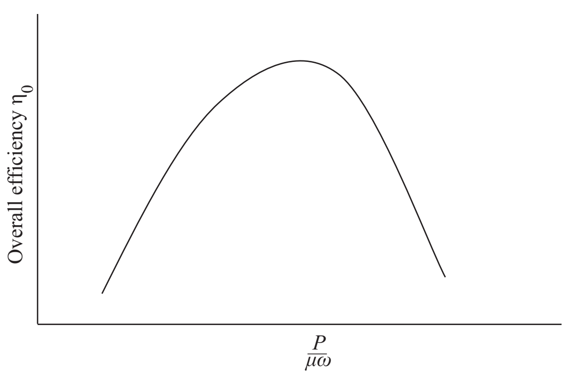

Figure 9.1. Variation in overall efficiency.

$$\frac{\text{Power loss}}{\text{Ideal power}} = C_s\left[\frac{\Delta P}{\mu\,\omega}\right] + C_f + C_v\left[\frac{\mu\,\omega}{\Delta P}\right]$$

Differentiating with respect to $\left[\frac{\mu\omega}{\Delta P}\right]$ and equating the expression to zero gives the minimum power loss when:

$$-C_s\left[\frac{\mu\,\omega}{\Delta P}\right]^{-2} + C_v = 0 \qquad \text{or} \qquad \frac{\mu\,\omega}{\Delta P} = \sqrt{\frac{C_s}{C_v}}$$

At this condition, the slip and viscous friction losses are equal. This suggests that for maximum overall efficiency, the volumetric losses and mechanical losses should be of similar magnitude; alternatively that η_v and η_m should be similar.

This method has wide applications in optimizing designs and operating conditions.

9.6 MODELING THE MECHANICAL LOSSES IN A TYPICAL HYDRAULIC PUMP AND MOTOR

9.6.1 ASSUMPTIONS IN THE GENERALISED MODEL

The efficiency models developed in the previous sections were based on generalized relationships between leakage and friction and the unit pressure and rotational speed. The major assumptions for these analytical methods are that:

1. The clearances between the sliding components are constant and do not vary with either speed or pressure.
2. The relationships between the frictional forces, the applied forces, and the speed are fixed and independent of the angular position of the shaft.

In an actual unit, these assumptions do not apply because of geometrical variations and hydrodynamic effects between the leakage faces and distortion of these faces due to the prevailing pressure and mechanical forces. It is also necessary, if possible, to consider the transfer of heat that is generated due to friction in the fluid. Whilst these are simple statements, their resolution is far from simple having been the subject of considerable research that continues and which is outside the remit to this book although references will be given for those who wish to investigate further. Where possible, the variations in efficiency are taken from test results, but from a design point of view, it is desirable to be able to model the flow and torque losses as best as possible.

9.6.2 ANALYSIS OF A RADIAL PISTON MOTOR

Looking at the various types of units that are described in Chapter 2 it will be realized that their geometry will cause both the flow and torque levels to vary as the shaft rotates. The degree of this variation will depend on the mechanical principle that is used in any particular design.

As an example, the theoretical torque for a radial piston eccentric motor, shown diagrammatically in Figure 9.2, which is of similar design principle to that shown in Figure 2.13, is derived in order to show how major parameters vary with the shaft angle. This analysis also uses a simplified friction model and its effect on mechanical efficiency.

The outer concentric cylinder of this motor is mounted on a spherical bearing, which is pressure loaded to maintain contact between the stationary and oscillating components. Thus, as the shaft rotates the eccentric center O_2 rotates around O_1 causing the inner cylinder to slide out of the outer cylinder, which creates the displacement of the motor by taking in fluid through the port in the outer bearing.

This analysis is mainly concerned with the level of friction that is created in the outer bearing and the effect that this has on the friction between the slipper and the eccentric surfaces. In an ideal motor, the slipper and eccentric faces are kept apart by the hydrostatic pressure between them from the fluid that is supplied by the slipper supply orifice from the main pressure in the cylinders.

In conventional motors, the cylinder port is connected to a rotating valve as seen in the various motor diagrams in Chapter 2. Such valve arrangements involve mechanical and flow leakage losses but if a digital valve system as shown in Figure 2.13 is used to supply the flow during one half of the shaft rotation such losses can be largely avoided.

Referring to Figure 9.2 for θ = 0 when the inner cylinder is at top dead center (TDC) the length $L_C = L_D - R_E = L_0$ when the stroke is zero. As X increases to a maximum of $2R_E$ at $\theta = 180°$. Note that L_D is the center distance between O_1 and O_3.

For intermediate positions of the shaft, the displacement X of the inner cylinder is given by:

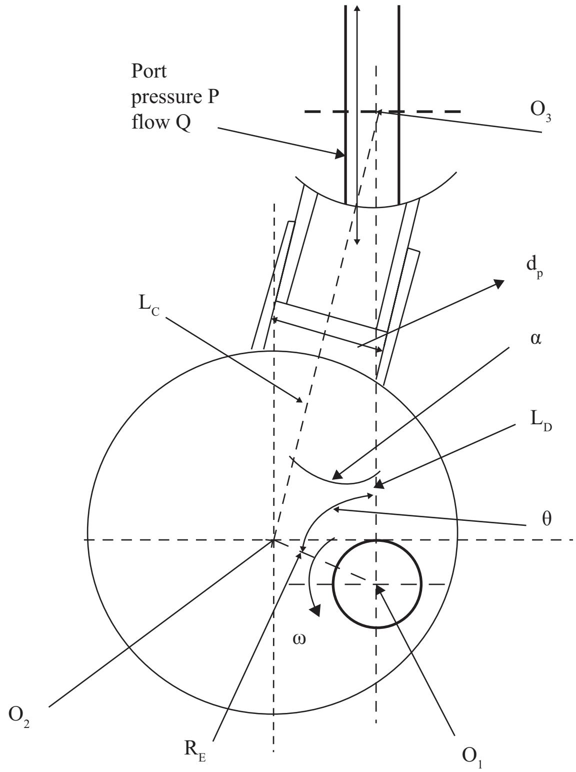

Figure 9.2. Radial piston eccentric-type motor.

$$\alpha = \sin^{-1}\left(\frac{R_E \sin\theta}{L_C}\right); \; X = \frac{L_D - R_E \cos\theta}{\cos\alpha} - L_0 \tag{9.12}$$

Also $L_C = X + L_0$

For a cylinder radius R_p, the pressure force is:

$$F_p = \pi R_P^2 P \tag{9.13}$$

This gives the ideal torque on the motor shaft of:

$$T_I = F_P R_E \sin(\theta + \alpha) \tag{9.14}$$

The mean torque is given by $T_{CM} = PD_C$, where D_C is the cylinder displacement for one shaft revolution.

Thus, the variation in the ideal torque with θ can be derived from Equation 9.14. The total output torque from a motor or the input torque to a pump is obtained by adding all the individual cylinder torques together over the respective shaft angles.

9.6.3 HYDROSTATIC BEARING

For the hydrostatic slipper in Figure 9.3, the slipper annulus outer and inner radii are R_S and R_1, respectively. The total separating force between the slipper and the eccentric (Appendix 9A1), F_S, is given by:

$$F_S = \pi P_P \left(\frac{R_S^2 - R_1^2}{2\ln\left(R_S / R_1\right)} \right)$$

Thus, for the slipper pressure force to be equal to the cylinder pressure force, that is, $F_S = F_P$ we get for the hydrostatic slipper pressure:

$$P_P = 2P\ln\left({}^{R_S}\!/_{R_1} \right) \frac{R_P^2}{R_S^2 - R_1^2}. \tag{9.15}$$

Hence, for a given pressure P the value of the slipper pocket pressure, P_P, can be obtained for given bearing dimensions. For laminar flow through the restrictor of length L_P and diameter d_o the flow, Q_O, can be determined (see Chapter 8), which will be equal to the slipper flow Q_S (derived in Appendix 9A1). Thus we get:

$$Q_O = \frac{\pi d_o^4 (P - P_P)}{128 \mu L_P} = Q_S = \frac{2\,\pi h^3 P_P}{12\mu \ln\left({}^{R_S}\!/_{R_1} \right)} \tag{9.16}$$

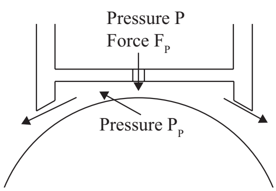

Figure 9.3. Slipper hydrostatic bearing.

The hydrostatic bearing flow, Q_S, is a leakage loss, which is added to any other leakages in the motor in order to determine the volumetric efficiency. Other sources of leakage include the outer bearing and the sealing between the two cylinders.

Equation 9.16 gives for the clearance, h:

$$h^3 = \frac{6d_o^4 \ln\left({R_S}/{R_1} \right)}{128L_P}\left(\frac{P}{P_P} - 1 \right) \tag{9.17}$$

Thus, the steady state bearing clearance is uniquely defined by the bearing dimensions.

9.6.4 FRICTION IN THE SLIDING CONTACTS

The effect of friction in the sliding surfaces is needed to determine the mechanical efficiency. Friction will occur in the:

1. Outer cylinder bearing.
2. Interface between the two cylinders.
3. Slipper bearing due to viscous resistance of the oil film and any contact between the two surfaces.

9.6.5 OUTER BEARING FRICTION

As described in Section 9.6.2, the outer cylinder is forced against the bearing surface, which creates a friction torque, T_{FC}. When the shaft rotates, the cylinder assembly oscillates so that there will be a reaction torque on the cylinder assembly. This creates a reaction torque on the slipper, which will cause a tilt between the slipper and the eccentric in the clearance space, which is shown in Figure 9.4.

If it is assumed that the flow in the clearance space is two dimensional, the pressure at any radius will be constant. As a result, there will be contact between the slipper and the eccentric with a resulting contact force F_{C1} on the cylinder.

Often a spring is placed inside the cylinders in order to ensure that contact is maintained should transient low values of the cylinder pressure occur. These can arise during high rotational speed conditions when there are large centripetal forces on the oscillating cylinder.

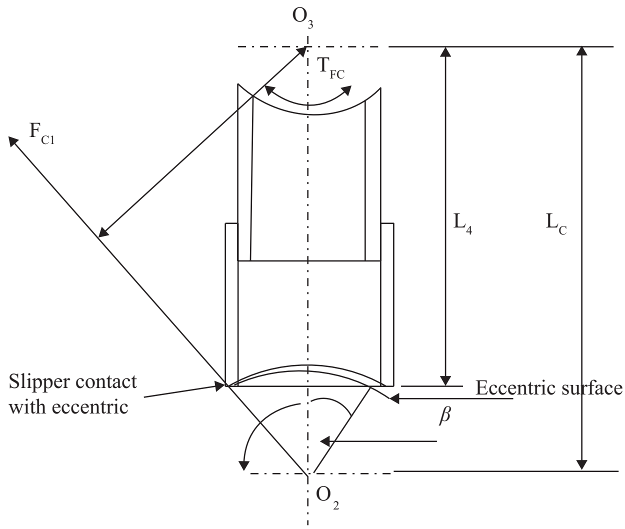

Figure 9.4. The effect of outer bearing friction on the slipper.

9.6.6 HYDROSTATIC BEARING LEAKAGE FLOW

In order to obtain the hydrostatic bearing pressure, P_P, it is necessary to determine the slipper leakage flow for the tilt condition. As a simplification, it is assumed that in the tilted condition there is no variation in the circumferential pressure in the area of a given slipper radius.

For simplicity, the bearing geometry in Figure 9.5 is shown as a flat surface. In Appendix 9A.2, the equation that relates the slipper flow as a function of the pocket pressure and the outer clearance with the slipper edge touching the surface of the eccentric (i.e., $h_4 = 0$) is derived.

Restating Equation 9A2.6 for the slipper flow:

$$Q_S = \frac{B}{I_{RT}} P_P$$

where $B = \dfrac{\pi^2 h_1^3}{24\mu}$ and I_{RT} is a function of the slipper radii given by Equation 9A2.5.

Thus, it is seen that in conjunction with Equation 9.16 for the slipper orifice flow Q_0 the pocket pressure will vary with the clearance h_1 as will

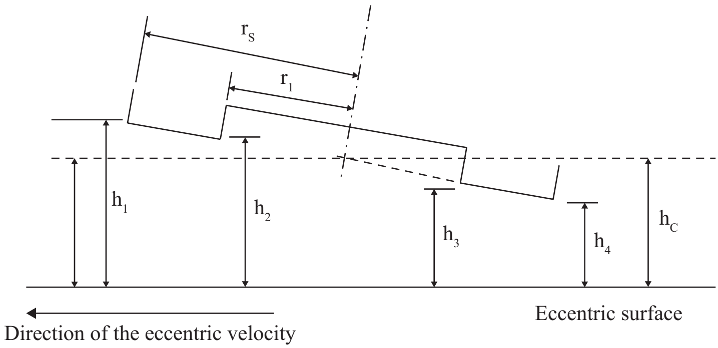

Figure 9.5. Cross-sectional view of the hydrostatic slipper.

the bearing pressure force, F_S. Now that the effect of friction in the outer bearing has been introduced as the slipper edge force F_{C1} a balance of the slipper radial forces will establish a slipper tilt that is created by the edge clearance h_1 and the slipper pocket pressure P_P. This will require an iterative method, where h_1 is varied until the equilibrium radial force condition is obtained.

For the slipper, the hydrostatic bearing maintains equality between the separating and hold-down pressure forces by ideally maintaining a separation between the faces as described in Section 9.6.3. Because of the friction torque in the outer bearing it is assumed that with the slipper edge in contact with the eccentric (as shown in Figure 9.4), the contact force, F_{C1}, will be such as to oppose the frictional resistance torque T_{FC}.

9.6.7 SLIPPER AND ECCENTRIC INTERFACE

For the slipper, as it tilts the slipper pressure will change, as will the flow, as described in Section 9.6.6 such as to create equilibrium of the radial forces whilst providing the necessary contact force F_{C1}.

The conditions between the two surfaces are complex because of the effect of the eccentric surface velocity and the hydrodynamic flow between the surfaces, which will depend on the rotational speed of the motor, the pressure, and deformation of the sliding faces. The solution of equations for this situation is outside the scope of this book but some research papers on the subject can be found in [15, 16]. For low-speed motors, the analysis can be simplified with the friction being determined by using a friction coefficient f_S [17, 18].

Thus, for example, from Figure 9.4, for $0 < \theta < 90°$ resolving moments about O_3 the equation for the contact force is given by:

$$\therefore F_{C1} = \frac{T_{FC}}{L_C \sin\beta + f_S (L_4 - R_S \sin\beta)\cos\beta} \tag{9.18}$$

The contact force F_{C1} on the eccentric causes a friction torque loss, which is here derived by using the friction coefficient f_S. The net shaft torque from the slipper is therefore the sum of the effect of the friction force between the slipper and the eccentric, the pressure force on the slipper plus the effect of the contact force about the eccentric center. Thus, the output torque T_C from the shaft is:

$$T_C = F_S R_E \sin(\theta + \alpha) - F_{C1} f_S (R_D + R_E \cos\lambda) + F_{C1} R_E \sin\lambda \tag{9.19}$$

where the angle λ is given by:

$\lambda = \theta + \alpha - \beta$ for shaft angles in the range $0 < \theta < 90°$.

For $90° < \varphi < 180°$ $\lambda = \theta + \alpha - \beta$ and the contact force is denoted as F_{C2}.

The cylinder pressure force, F_P, is reduced by the friction forces in the interface of the two cylinders, which will be affected by the overlap length because of the side loads that are generated by the moment due to the outer bearing friction torque.

The total torque loss that results from the outer bearing friction will therefore be the difference between the ideal and the actual torque, which includes the friction torque loss. For one cylinder of swept volume D_C the mean torque $T_{CM} = PD_C$ and so the proportion of the torque loss to mean torque can be obtained.

Because of the oscillatory motion of the outer cylinder, the friction torque will change direction. Thus, the point of contact between the slipper and the eccentric will be on the trailing edge for $0° < \theta < \theta_M$ and on the leading edge for $\theta_M < \theta < 180°$, where θ_M is the shaft angle at which the cylinder angle α is at its maximum value.

9.6.8 MECHANICAL EFFICIENCY

From a design perspective, it is important to be able to determine how the mechanical efficiency of a motor is affected by the levels of friction in the sliding faces and the motor geometry. Important design aspects include:

1. The ratio between the outer bearing separating pressure force and the total pressure hold down force on the outer cylinder.
2. The minimum engagement length or overlap between the two cylinders at maximum stroke. This is influenced by the clearance between the two components.
3. The piston diameter and eccentricity for a given displacement. As the piston size is increased, the stroke R_E will be reduced for a given displacement. The maximum cylinder angle α_M will also reduce and it is required to find the value of the ratio R_P/R_E that gives maximum mechanical efficiency within the given constraints on the geometry and the physical size of the motor.
4. The materials and surface treatments of the sliding surfaces.

The analytical method described here is essentially for static or low-speed conditions and considers neither the inertia forces of the components, which increase as the square of the speed or viscous effects between sliding components, which increase proportionately with the speed for a given clearance between the components.

9.7 COMMENTS

The early work in this chapter was concerned with developing generalized models for efficiency in order to show how overall efficiency varies with the nondimensional parameter $\frac{\mu\omega}{\Delta P}$. In general, the mechanical loss is based on pressure- and speed-dependent terms. The analysis of the motor dealt with in Section 9.6.1 is only concerned with static losses, which vary proportionately with the pressure level. Speed-dependent friction would be concerned with viscous effects in small clearances where there is relative motion, for example, slipper/eccentric and cylinder interfaces.

Whilst the use of a friction coefficient is a simplification, it is nonetheless useful in obtaining a value for the mechanical loss and for optimizing the geometry of the unit so as to minimize the loss whilst also giving the opportunity to investigate any loading effects on the components. As has been mentioned, a full analysis requires the use of hydromechanical flow and finite element analyses, which would be applied to evaluating the motor performance over its operational speed range. The dynamic effects of the inertia of the oscillating components could also be included in the model.

Clearly, the values of the friction coefficients can be different from each other and will, in reality, vary during the rotation of the shaft. Also, the various clearances between the components will not be the same in a given clearance space (e.g., piston in a cylinder bore can be tilted). All of these variations will naturally have a significant effect on the torque losses so that overall these can only be determined from experimental test results.

In conventional motors, the shaft-driven flow distribution valve introduces viscous torque and pressure losses that will increase with the rotational speed. There will also be leakage losses through the various clearances. For motors using digitally operated valves, as in Figure 2.13, for each cylinder, such losses are considerably reduced, which results in an improvement of its overall efficiency.

SUMMARY

The performance of pumps and motors is subject to the effect of torque and leakage losses that determine their overall efficiency. Manufacturers' technical literature will usually show the efficiency variation over the range of operating conditions for a particular value of the fluid viscosity. For the use of the unit with a fluid having a different viscosity from that quoted requires knowledge of the way in which the mechanical and volumetric losses arise. The processes involved can be modeled using the analytical methods described in this chapter, which also provides a background for their implementation in computer database and simulation systems.

APPENDIX

9A.1 FLOW THROUGH AN ANNULAR SPACE

From Chapter 8, Equation 8.3

$$Q = -\frac{h^3 2\pi r}{12\mu}\frac{dp}{dr} \tag{9A1.1}$$

$$\int_P^p dp = (p - P) = -\frac{12\,\mu Q}{2\,\pi h^3}\int_R^{R_S}\frac{dr}{r} = -\frac{12\,\mu Q}{2\,\pi h^3}\ln\left(\frac{R_S}{R_1}\right) \tag{9A1.2}$$

For $p = 0$ at the outer boundary we get:

$$Q = -\frac{2\,\pi h^3 P}{12\,\mu\ln\left(\frac{R_S}{R_1}\right)} \tag{9A1.3}$$

Also Equations 9A1.1 and 9A1.3 give:

$$\frac{P}{\ln\left(\frac{R_S}{R_1}\right)} = -r\frac{dp}{dr}\;\therefore\;\frac{1}{P}(P - p) = \frac{\ln\left(\frac{r}{R_1}\right)}{\ln\left(\frac{R_S}{R_1}\right)}\text{ or }p = P\left[1 - \frac{\ln\left(\frac{r}{R_1}\right)}{\ln\left(\frac{R_S}{R_1}\right)}\right] \tag{9A1.4}$$

The pressure force between the bearing faces is:

$$F = 2\,\pi\int_{R_1}^{R_S} p\,r dr = 2\,\pi\,P\left[\int_{R_1}^{R_S} r dr - \frac{1}{\ln\left(\frac{R_S}{R_1}\right)}\int_{R_1}^{R_S} r\ln\left(\frac{r}{R_1}\right)dr\right]$$

$$\therefore F = 2\,\pi P\left[\frac{(R_S^2 - R_1^2)}{2} - \frac{1}{\ln\left(\frac{R_S}{R_1}\right)}\left\{\frac{(R_S^2)}{2}\ln R_S - \frac{R_1^2}{2}\ln R_1\right.\right.$$

$$\left.\left.-\frac{1}{2}\ln R_1(R_S^2 - R_1^2) - \frac{1}{4}(R_S^2 - R_1^2)\right\}\right]$$

$$F = \pi P\left[(R_S^2 - R_1^2) - \frac{R_S^2 \ln\left(\dfrac{R_S}{R_1}\right) - \dfrac{1}{2}(R_S^2 - R_1^2)}{\ln\left(\dfrac{R_S}{R_1}\right)} \right]$$

$$= \pi P\left[\frac{(R_S^2 - R_1^2)}{2\ln\left(\dfrac{R_S}{R_1}\right)} - R_1^2 \right] \qquad (9A1.5)$$

Note: $\int r \ln r dr = \frac{1}{2}\int (\ln r) d(r)^2 = \frac{r^2}{2}\ln r - \frac{r^2}{4} + C$

9A.2 HYDROSTATIC FLOW THROUGH A TILTED SLIPPER

The hydrostatic flow is passing through an annular space $rd\varphi$ in width, where the angle φ is that around the circumference. This gives the flow equation:

$$dq = -\frac{2\pi}{12\mu}\left(\frac{rdp}{dr}\right)h^3 d\phi \qquad (9A2.1)$$

Referring to Figure 9.5 we get:

$$h = \left((R_s - r\cos\varphi)a_s + h_4\right)$$

If we consider the situation when the edge of the slipper is touching the eccentric then:

$$h_4 = 0; \quad a_s = \frac{h_1}{2R_S}$$

This assumes that the flow is only radially outwards (i.e., hydrostatic two-dimensional flow), which means that the pressure is the same on any radius—that is, there is no circumferential flow.

This will give from Equation 9A2.1:

$$Q_S = -\frac{4\,\pi}{12\mu}\left(\frac{rdp}{dr}\right)\int_0^{\pi}(R_S a_S - r a_S \cos\varphi)^3 d\,\varphi$$

$$= -\frac{4\pi a_S^3}{12\mu}\left(\frac{rdp}{dr}\right)\int_0^{\pi}(R_S a_S - r a_S \cos\varphi)^3 d\,\varphi \tag{9A2.2}$$

Now:

$$(R_S - r\cos\varphi)^3 = R_S^3 - 3R_S^2\, r\cos\varphi + 3R_S r^2 \cos\varphi^2 - r^3\cos\varphi^3$$

On substituting into Equation 9A2.2 this gives:

$$Q_S = -\frac{4a_S^3\,\pi^2}{12\mu}\left(\frac{rdp}{dr}\right)\left(R_S^3 + \frac{1}{2}R_S r^2\right) \tag{9A2.3}$$

$$\int_{R_1}^{R_S}\frac{dr}{r\left(1+\dfrac{r^2}{2R_S^2}\right)} = -\frac{B}{Q_S}\int_{P_P}^{0} dp = \frac{B}{Q_S}P_P \tag{9A2.4}$$

where: $B = \dfrac{\pi^2 a_S^3 R_S^3}{3\mu} = \dfrac{\pi^2 h_1^3}{24\mu}$ (on substituting for a_s^3). From Equation 9A2.4 we get:

$$\int_{R}^{R_S}\frac{dr}{r\left(1+\dfrac{r^2}{2R_S^2}\right)} = \ln\left(\frac{R_S}{R_1}\right) - \frac{1}{2}\ln(2R_S^2 + R_1^2) + \frac{1}{2}\ln(2R_S^2 + R_1^2)$$

$$= \ln\left(\frac{R_S}{R_1}\right) + \frac{1}{2}\left[\ln\left\{\frac{2R_S^2 + R_1^2}{3R_S^2}\right\}\right] = I_{RT} \tag{9A2.5}$$

Equations 9A2.4 and 9A2.5 give for the flow

$$Q_S = \frac{B}{I_{RT}}P_P \tag{9A2.6}$$

CHAPTER 10

CONTROL SYSTEM DESIGN

10.1 INTRODUCTION

Hydraulic valves are extensively employed in the closed-loop position of actuators so as to obtain good steady state accuracy and speed of response. The open-loop velocity control of linear actuators was discussed in Chapter 7, where the flow through the valve, for constant supply pressure, was shown to depend upon the valve opening and the pressure drop across the opening. Thus, under specified pressure conditions, the valve can be used as a flow source when connected to an actuator, the velocity of which will be proportional to the valve opening.

The dynamic performance of a valve actuator system is developed in which it is assumed that the mass of the moving components and the load force are both zero. The effect of these assumptions is to eliminate the influence of the fluid compressibility so that the system response is first order.

The inclusion of the mass into the analysis creates a third-order system because of the effect of the fluid compressibility and, as a consequence of this, the value of the system gain has to be determined in order to obtain satisfactory stability margins for the system.

10.2 SIMPLE VALVE ACTUATOR CONTROL

10.2.1 OPEN-LOOP SYSTEM

The valve actuator circuit in Figure 10.1 has an equal area, or double ended, actuator that is frequently used for closed-loop control systems because of its symmetry with regard to the hydraulic force and velocity in both directions of movement. The force characteristics for the system can

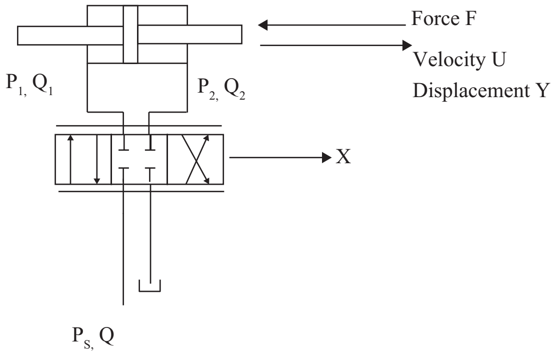

Figure 10.1. Valve actuator circuit.

be obtained from the analysis in Chapter 7, Equations 7.3 to 7.6, which with the area ratio α = 1 gives

Extend

$$\frac{P_1}{P_s} = \frac{(1+R\alpha^3)}{(1+\alpha^3)}$$

$$\therefore P_1 = \frac{P_S}{2}(1+R) = \frac{P_S}{2} + \frac{F}{2A}$$

$$P_1 = \frac{P_S}{2} \; for \; F = 0$$

$$\frac{P_2}{P_s} = \frac{\alpha(1-R)}{(1+\alpha^3)}$$

$$\therefore P_2 = \frac{P_S}{2}(1-R) = \frac{P_S}{2} - \frac{F}{2A}$$

$$P_2 = \frac{P_S}{2} \; for \; F = 0$$

Retract

$$\frac{P_1}{P_s} = \frac{\alpha^2(1+\alpha R)}{(1+\alpha^3)}$$

$$\therefore P_1 = \frac{P_S}{2} + \frac{F}{2A}$$

$$P_1 = \frac{P_S}{2} \; for \; F = 0$$

$$\frac{P_2}{P_s} = \frac{\alpha^3 - \alpha R}{1+\alpha^3}$$

$$\therefore P_2 = \frac{P_S}{2} - \frac{F}{2A}$$

$$P_2 = \frac{P_S}{2} \; for \; F = 0$$

The force F is positive for the direction shown in Figure 10.1. For the equal area actuator, the valve flows are the same on each side of the valve and the pressure differences $(P_s - P_1)$ and P_2, respectively, will therefore be the same.

For zero force, $P_1 = P_2 = P_S/2$ that, as can be seen from Figure 10.2, is the pressure at which the flow characteristics intersect.

The flow through the valve is given by

$$Q_{1,2} = Q = C_Q \pi dX \sqrt{\frac{2}{\rho}\frac{P_S}{2}}$$

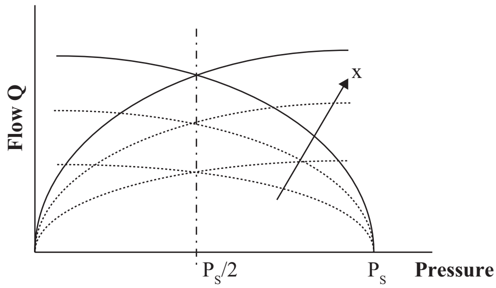

Figure 10.2. Valve characteristics.

For the simple system, assume that the moving components have negligible inertia so that, as a consequence, the actuator pressures will remain constant during transient changes caused by displacement of the valve.

Thus

$$Q = K_Q \; X \tag{10.1}$$

where

$$K_Q = C_Q \pi d \sqrt{\frac{P_S}{\rho}} \; \mathrm{m}^2/\mathrm{s}$$

And, for an actuator area *A,* the actuator velocity *U* is

$$U = \frac{Q}{A} = \frac{K_Q \; X}{A} \tag{10.2}$$

$$\text{As } U = \frac{\mathrm{d}Y}{\mathrm{d}t}, \text{ then } Y = \frac{K_Q}{A} \int X \,\mathrm{d}t \tag{10.3}$$

The actuator displacement is formed by the integral of the valve opening and, as a consequence, it is referred to as an integrator. A step opening of the valve will, therefore, cause the actuator to move at a constant velocity, stopping when the valve is closed. This describes the open-loop performance of the system.

The time response of the actuator, or integrator, to a step change in the valve position is shown in Figure 10.3.

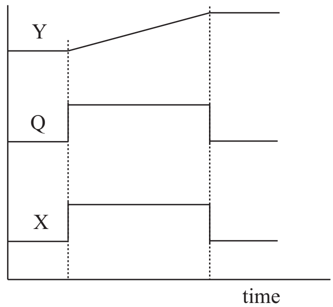

Figure 10.3. Open-loop time response.

10.2.2 CLOSED-LOOP SYSTEM

Closed-loop systems operate by comparing the output position with an input demand signal, such systems being referred to as feedback control. In electrohydraulically operated valve actuator systems, the input signal is a voltage and the output position is fed back as a voltage signal from a position transducer. The comparison of the two signals produces a voltage difference referred to as the error signal that is amplified as a current signal and supplied to the electrohydraulic valve. In this analysis, the relationship between the input and output is obtained from the following equations.

The valve position is given by

$$X = K_A K_V (V_i - K_T Y) \tag{10.4}$$

K_T = position transducer gain, V/m
K_V = valve gain, m/A
K_A = amplifier gain, A/V
V_i = input signal, V

Equations 10.2 to 10.4 give

$$\frac{\mathrm{d}Y}{\mathrm{d}t} = \frac{K_Q}{A} K_A K_V (V_i - K_T Y)$$

It is usual to express $\frac{\mathrm{d}}{\mathrm{d}t}$ by the Laplace operator "*s*" allowing this equation to be treated algebraically:

Thus

$$\left(K_T + \frac{A}{K_Q K_A K_V} s \right) Y = V_i$$

This gives

$$\frac{Y}{V_i} = \frac{1/K_T}{(1+Ts)} \tag{10.5}$$

This is a first-order differential equation where the time constant $T = \dfrac{A}{K_A K_Q K_V K_T}$, which has the units of time (s).

The system can be represented as a block diagram in Figure 10.4.

This diagram can be simplified.

Note that electrohydraulic manufacturer's literature will usually specify the valve performance in terms of flow for a given input current at a specific valve pressure drop, which will enable the gain product $K_V K_Q$ to be obtained.

The transfer function for the relationship between y and θ_i can be obtained from Figure 10.5, which is given by

$$\frac{y}{V_i} = \frac{\dfrac{K_1}{As}}{1+\dfrac{K_1 K_2}{As}} = \frac{\dfrac{1}{K_2}}{1+Ts} \quad where\ T = \frac{A}{K_1 K_2} \tag{10.6}$$

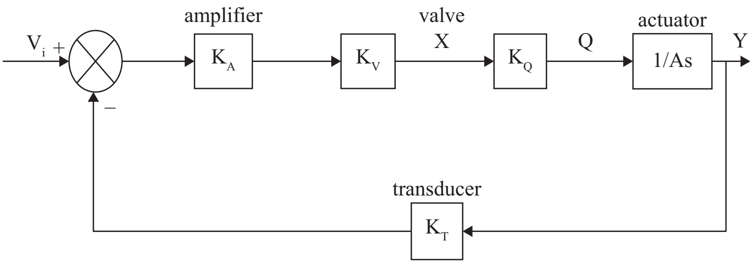

Figure 10.4. System block diagram.

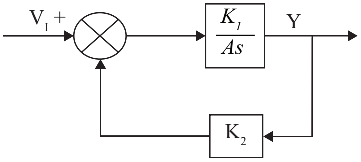

Figure 10.5. Simplified block diagram.

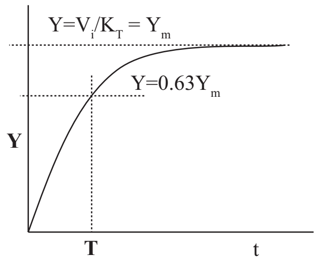

Figure 10.6. Step response.

Substituting for K_1 and K_2 in Equation 10.6 gives the same result as in Equation 10.5.

10.2.3 SYSTEM RESPONSE

The time response from Equation 10.5 can be obtained for a range of input signals by referring to a dictionary of standard inverse Laplace transforms. The response to a step input gives the solution

$$Y = \frac{V_i}{K_T}\left(1 - e^{-t/T}\right) \tag{10.7}$$

This exponential variation of Y with time reaches 63.2 percent of the final value in one time constant as shown in Figure 10.6.

10.3 FLUID COMPRESSIBILITY

10.3.1 BULK MODULUS

The compressibility of fluids is expressed by the bulk modulus, β, which varies with pressure and, as a consequence, either average or local values have to be used. However, the bulk modulus is also considerably affected by absorbed air in the fluid and also by the use of hoses. As a consequence, the value used in the analysis has to be considered carefully.

Bulk modulus is defined as $\beta = -\dfrac{P}{\Delta V / V}$ where the term $\dfrac{\Delta V}{V}$ is the volumetric strain in the fluid. For hydraulic oil, the value of β is usually taken as 1.8×10^9 N m^{-2} but this can reduce by 30 to 50 percent in some circumstances.

10.3.2 HYDRAULIC STIFFNESS

With both ports blocked as in Figure 10.7 and with an initial pressure level that will prevent negative pressures from arising we get:

Pressure changes from the actuator force

$$\Delta F = \Delta P_1 A_1 - \Delta P_2 A_2 \tag{10.8}$$

From the bulk modulus

$$\Delta V_1 = -V_1 \frac{\Delta P_1}{\beta} \text{ and } \Delta V_2 = -V_2 \frac{\Delta P_2}{\beta} \tag{10.9}$$

Also

$$\Delta V_1 = -A_1 \Delta X \text{ and } \Delta V_2 = A_2 \Delta X \tag{10.10}$$

Equations 10.8, 10.9, and 10.10 give

$$\frac{\Delta F}{\Delta X} = \beta \left(\frac{A_1^2}{V_1} + \frac{A_2^2}{V_2} \right) \tag{10.11}$$

For an equal area actuator with $A_1 = A_2 = A$ and with the piston in the central position with $V_1 = V_2 = V$, Equation 10.11 gives for the hydraulic stiffness

$$\frac{\Delta F}{\Delta X} = 2\frac{\beta A^2}{V} = K_H \tag{10.12}$$

The bulk modulus has two significant effects on the dynamic performance of closed-loop hydraulic systems in that:

- The hydraulic stiffness gives rise to a hydraulic natural frequency

$$\omega_n = \sqrt{\frac{stiffness}{mass}} = \sqrt{\frac{K_H}{m}} \ rad/s$$

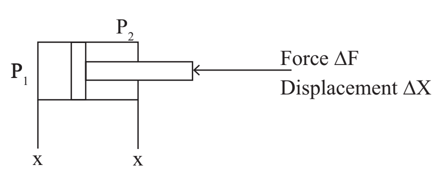

Figure 10.7. Fluid compressibility.

- The compressibility creates transient delays in volumes where $\frac{\mathrm{d}P}{\mathrm{d}t} = \frac{\beta}{V}\frac{\mathrm{d}V}{\mathrm{d}t} = \frac{\beta}{V}Q_C$ (Q_C is the difference between the inlet and outlet flow to the volume V).

10.4 VALVE ACTUATOR DYNAMIC RESPONSE INCLUDING COMPRESSIBILITY EFFECTS

10.4.1 VALVE FLOW

The simple system model assumed that the inertial mass of the moving components was zero so that the actuator pressures remained constant during transients. In the presence of inertial load mass, the pressures will have to change from the steady state values in order to change the velocity of the actuator and these changes will need to take account of the fluid compressibility in relation to the system flows.

The analysis in Section 10.2.1 can be applied to systems where the force is not zero by determining the steady state actuator pressures. For the equal area actuator during extension as in Figure 10.1 these are

$$P_1 = \frac{P_S}{2} + \frac{F}{2A} = \frac{P_S}{2} + \frac{P_m}{2} \tag{10.13}$$

and

$$P_2 = \frac{P_S}{2} - \frac{F}{2A} = \frac{P_S}{2} - \frac{P_m}{2} \tag{10.14}$$

The pressure P_m is the load pressure and it is seen that it is divided equally in its effect on P_1 and P_2, which will affect the flows also equally as shown in Figure 10.8.

The load force will affect the flow coefficient K_Q in Equation 10.1 because the valve pressure differences become

$$P_S - P_1 = \frac{1}{2}(P_S - P_m) = P_2 \tag{10.15}$$

Equation 10.1 can be restated as

$$Q = C_Q \pi dX\sqrt{\left(\frac{P_S - P_m}{\rho}\right)} = K_Q X\sqrt{(1 - \frac{P_m}{P_S})} \tag{10.16}$$

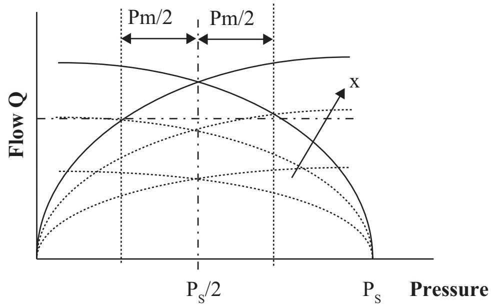

Figure 10.8. The effect of load pressure on the valve characteristics.

For $P_m = 0$ (i.e., zero force) Equation 10.16 reverts to Equation 10.1. The load pressure ratio P_m/P_S varies in the range +1 to −1 (pushing to pulling) and from Equation 10.16 it is seen that the maximum flow is $\sqrt{2}\,K_Q X$.

For a constant value of the load force, the valve flow gain with valve displacement is from Equation 10.15

$$\frac{Q}{X} = \left(\frac{\partial Q}{\partial x}\right)_{P_m} = K_Q\sqrt{\left(1 - \frac{P_m}{P_S}\right)}$$

To develop the equations for the flow through each side of the valve with changes in the pressures P_1 and P_2 it is necessary to apply the method of small perturbations to the valve flow equations.

For the flow entering the actuator we have

$$Q_1 = C_Q \pi d X \sqrt{\frac{2}{\rho}(P_S - P_1)} = KX\sqrt{(P_S - P_1)} \qquad (10.17)$$

$$K = C_Q \pi d \sqrt{\frac{2}{\rho}}$$

Thus, the flow $Q_1 = f(X, P_1)$ and changes in the flow for small changes in X and P_1 are given by

$$\Delta Q_1 = q_1 = \frac{\partial Q_1}{\partial X}x + \frac{\partial Q_1}{\partial P_1}p_1 \qquad (10.18)$$

The lower case letters denote small changes in the variables.

From Equation 10.17 we have

$$\frac{\partial Q_1}{\partial P_1} = \frac{KX}{2\sqrt{(P_S - P_1)}}(-1) \; and \; \frac{\partial Q_1}{\partial X} = K\sqrt{(P_S - P_1)} \tag{10.19}$$

Similarly for the flow leaving the actuator

$$Q_2 = C_Q \pi dX \sqrt{\frac{2}{\rho} P_2} = KX\sqrt{P_2}$$

$$Q_2 = f(X, P_2),\; \Delta Q_2 = q_2 = \frac{\partial Q_2}{\partial X} x + \frac{\partial Q_2}{\partial P_2} p_2 \tag{10.20}$$

$$\frac{\partial Q_2}{\partial P_2} = \frac{KX}{2\sqrt{P_2}} \; and \; \frac{\partial Q_2}{\partial X} = K\sqrt{P_2} \tag{10.21}$$

Thus, it is seen that the values of the flow coefficients used to express the changes in the flows with changes in X and with the pressures are dependent on the steady state values of the pressures. This is why the equations are only applicable for small changes about a steady state condition.

The symmetry of the valve flow characteristics arising from the use of an equal area actuator results in the following:

$$\frac{\partial Q_1}{\partial X} = \frac{\partial Q_2}{\partial X} = C_x \tag{10.22}$$

and

$$\frac{\partial Q_1}{\partial P_1} = -\frac{\partial Q_2}{\partial P_2} = -C_p \tag{10.23}$$

Thus, the changes in flow are given by

$$q_1 = C_x x - C_p p_1 \tag{10.24}$$

and

$$q_2 = C_x x + C_p p_2 \tag{10.25}$$

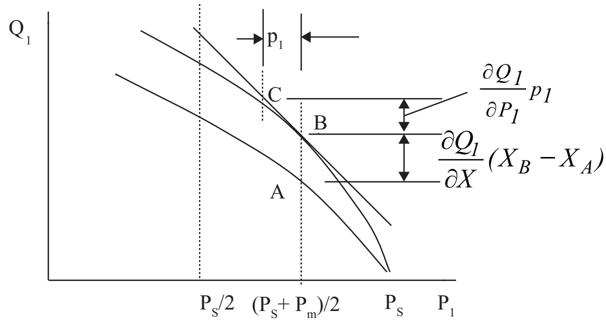

Figure 10.9. Valve flow coefficients.

Figure 10.9 gives a representation of Equation 10.24, where p_1 is the change in pressure away from the steady state value of $(P_S + P_m)/2$. For reasons of symmetry, the changes in p_1 and p_2 will be equal in magnitude and opposite in sign, that is, $p_1 = -p_2$.

10.4.2 ACTUATOR FLOWS

The flow between the valve and the actuator on both sides of the piston will be subject to compressibility effects as described in Section 10.3.

For an actuator velocity, u, the inflow is given by

$$q_1 = C_x x - C_p p_1 = Au + \frac{V}{\beta}\frac{dp_1}{dt} = Au + \frac{V}{\beta} s p_1$$

$$\therefore p_1 = \frac{C_x x - Au}{C_p + \frac{V}{\beta} s} \tag{10.26}$$

and

$$q_2 = C_x x + C_p p_2 = Au - \frac{V}{\beta} s p_2$$

$$\therefore p_2 = \frac{Au - C_x x}{C_p + \frac{V}{\beta} s} \tag{10.27}$$

The analysis is for equal volumes on either side of the piston, which would apply to the piston in the mid-position for equal pipe volumes on both sides. It is seen from Equations 10.26 and 10.27 that $p_1 = -p_2$ because of the symmetry created by the equal area actuator.

10.4.3 ACTUATOR FORCE

This analysis is concerned with small changes about a steady state operating condition for an external force that is assumed to be constant so the analysis will only need to consider changes in force due to inertia and friction. Friction is usually considered to compose of coulomb, which is assumed to be unchanging, and a speed-dependent element that is assumed to vary proportionally with the velocity. In practice, the level of coulomb friction is known to vary considerably but in an unpredictable manner and this effect, consequently, is not included in this analysis.

Thus, changes in the forces can be given by

$$(p_1 - p_2)A = m\frac{du}{dt} + C_f u = msu + C_f u \tag{10.28}$$

Substituting Equations 10.26 and 10.27 into 10.28 gives

$$2AC_x x = 2A^2 u + \left(C_p + \frac{V}{\beta}s\right)\left(ms + C_f\right)u$$

$$\frac{C_x}{A}x = \left(1 + \frac{C_p C_f}{2A^2} + \left(\frac{C_p m}{2A^2} + \frac{C_f V}{2A^2\beta}\right)s + \frac{mV}{2A^2\beta}s^2\right)u$$

The transfer function between u and x is of the form

$$\frac{u}{x} = \frac{C_x/A}{\left(1 + 2\frac{\zeta}{\omega_n}s + \frac{1}{\omega_n^2}s^2\right)} \tag{10.29}$$

where

$$\omega_n = \sqrt{\frac{2\beta A^2}{Vm}} \tag{10.30}$$

and

$$\zeta = \frac{C_p}{2A}\sqrt{\frac{m\beta}{2V}} + \frac{C_f}{2A}\sqrt{\frac{V}{2m\beta}} \tag{10.31}$$

This has assumed that the $\frac{C_f C_p}{2A^2}$ term is negligibly small.

10.4.4 COMMENTS

Generally, it is found that the value of ζ predicted by Equation 10.31 is lower than in practical systems, some of the reasons for this include:

- The value of C_f is usually difficult to predict with any accuracy, and actuator friction can vary considerably, particularly at low velocities.
- The electrohydraulic valve has been assumed to have an instantaneous response. In many applications the valve has a much higher response than the hydraulic natural frequency of the system but nonlinearities at the null position, such as friction, leakage, and coil hysteresis, can affect the system performance.
- The value of C_p, for a given load pressure, varies over the flow range from zero at the valve null position to a maximum when the valve is fully open but in the analysis this has to be taken as a constant. In fact, valves will leak over a range of valve positions around the null point as specified by the pressure gain quoted by the manufacturer, which affects the system response and steady state accuracy. This is discussed in Section 10.6.3.1.

10.4.5 ACTUATOR POSITION

Changes in actuator displacement are the integral of velocity that can be written as

$$y = \frac{u}{s}$$

Thus, from Equation 10.29 we get

$$\frac{y}{x} = \frac{C_x / A}{s\left(1 + 2\frac{\zeta}{\omega_n}s + \frac{1}{\omega_n^2}s^2\right)} \tag{10.32}$$

Equation 10.32 is the open-loop transfer function that relates changes in actuator displacement to small changes in the valve position with constant external force acting on the actuator rod. This is a third-order equation that can give rise to instability in a closed loop.

The effect of the load mass has been to add a second-order equation to the first-order system obtained in Equations 10.2 and 10.3 because of the hydraulic system stiffness created by the fluid compressibility. The value of the flow coefficient C_x is the same as that of the flow coefficient K_Q used in Section 10.2.1 for the simple system when the actuator is unloaded and $P_1 = P_2$.

10.4.6 VALVE SELECTION

Valves are usually rated on their flow, Q_R, with both outlet ports connected together at an overall pressure drop of 70 bar (ΔP_R). The flow at any other pressure difference is given by

$$Q = Q_R\sqrt{\frac{\Delta P}{\Delta P_R}}$$

Thus, for the analysis the overall pressure difference in the valve is $2(P_S - P_1)$ *or* $2P_2$, where these pressures determine the load force acting on the actuator rod. The valve size needs to be a given amount larger than the maximum required flow at the maximum valve input current.

10.4.7 PRESSURE SHOCK CONTROL IN OPEN-LOOP SYSTEMS

In the open-loop operation of valve actuator systems, fast opening of valves can cause considerable pressure shocks, which may result in noise, excessive vibration of the load and, possibly, of the machine itself. Such shocks can be reduced by ramping the valve (open and/or closed) over a short period of time.

The variation in actuator displacement and pressure can be analyzed by using the linearized Equations 10.26 and 10.27 with the valve displacement changing with time at a constant rate (ramp). A comparison of the responses with reduced rates of change in valve movement will show that the amplitude of the pressure shocks will be reduced. The actuator position will also be reduced but not significantly, as the ramping time is usually of the order of 10 to 50 ms. An example of the effect of the valve ramping time is given in Section 11.13.8.

10.5 FREQUENCY RESPONSE

The stability of a closed-loop system can be obtained by examining the frequency response of the open-loop system. For the simple system the open loop is an integrator, and in cases where the natural hydraulic frequency is high, the second-order term in Equation 10.32 can be neglected.

10.5.1 SIMPLE ACTUATOR

When supplying flow into an actuator as a sine wave, the actuator displacement will also be a sine wave but displaced by 90° phase shift as shown in Figure 10.10. This is because for positive flow the actuator moves in one direction, returning to the starting point during the negative flow period.

Thus, during the time period t_1 the fluid volume entering the actuator will be

$$V = \int_0^{t_1} \frac{Q_A}{2} \sin \omega t dt = \frac{Q_A}{\omega} \quad for\, t_1 = \frac{\pi}{\omega}$$

For this volume, the amplitude of the actuator displacement is

$$Y_A = \frac{V}{A} = \frac{Q_A}{A\omega} \tag{10.33}$$

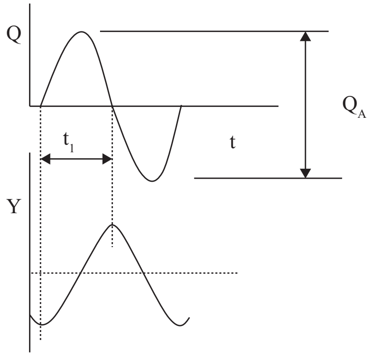

Figure 10.10. Sinusoidal flow variations and actuator displacement.

The actuator displacement is therefore inversely proportional to the frequency with a phase lag of 90°. Mathematically, this is obtained by replacing the "*s*" term with $j\omega$ in the transfer function. From Equation 10.33 for the actuator this gives

$$\frac{y}{x} = \frac{K_Q}{Aj\omega}; \quad \left|\frac{y}{x}\right| = \frac{K_Q}{A\omega} \;\&\; \varphi = -90^0 \tag{10.34}$$

It is preferable to plot the frequency response as a Bode diagram that uses logarithmic axes so that Equation 10.34 becomes

$$20\log R = 20\log\left(\frac{K_Q}{A}\right) - 20\log\omega \tag{10.35}$$

$R = \left|\frac{y}{x}\right|$ and $20\log R$ is the amplitude in decibels (dB).

Equation 10.35 is a straight line having a slope of −20 dB/decade that crosses the 0 dB axis when $\omega = K_Q/A$.

10.5.2 VALVE ACTUATOR SYSTEM

The frequency response of the open-loop transfer function of Equation 10.32 can be obtained by replacing the "*s*" terms with $j\omega$, which gives for the amplitude and phase angle

$$R = \frac{C_x/A}{\omega\sqrt{\left(1-\left(\frac{\omega}{\omega_n}\right)^2\right)^2 + \left(2\zeta\frac{\omega}{\omega_n}\right)^2}} \tag{10.36}$$

$$20\log R = 20\log\left(\frac{C_x}{A}\right) - 20\log\omega - 20\log\left(\left(1-\left(\frac{\omega}{\omega_n}\right)^2\right)^2 + \left(2\zeta\frac{\omega}{\omega_n}\right)^2\right)^{0.5} \tag{10.37}$$

$$\phi = -\tan^{-1}\left(\frac{2\zeta\frac{\omega}{\omega_n}}{(1-(\frac{\omega}{\omega_n})^2)}\right) - 90^0 \tag{10.38}$$

The variations in the amplitude ratio and phase angle with frequency are shown in Figure 10.11.

The amplitude ratio of Equation 10.37 can be simplified by approximating the value of the second-order term for $\omega < \omega_n$ and for $\omega > \omega_n$.

Thus, for $\omega < \omega_n$:

$$20\log R \to 20\log\left(\frac{C_x}{A}\right) - 20\log\omega$$

because the powers of $\frac{\omega}{\omega_n}$ << 1 so that the second-order term in the denominator tends to be unity.

And for $\omega > \omega_n$:

$$20\log R \to 20\log\left(\frac{C_x}{A}\right) - 20\log\omega - 40\log\left(\frac{\omega}{\omega_n}\right)$$

when the second-order term in the denominator tends to be $\frac{\omega}{\omega_n}$. This is a straight line having a slope of −60 dB/decade.

These straight-line approximations, or asymptotes, are shown as dotted line in Figure 10.11, and the actual response deviates from these in a small range of frequencies around the natural frequency.

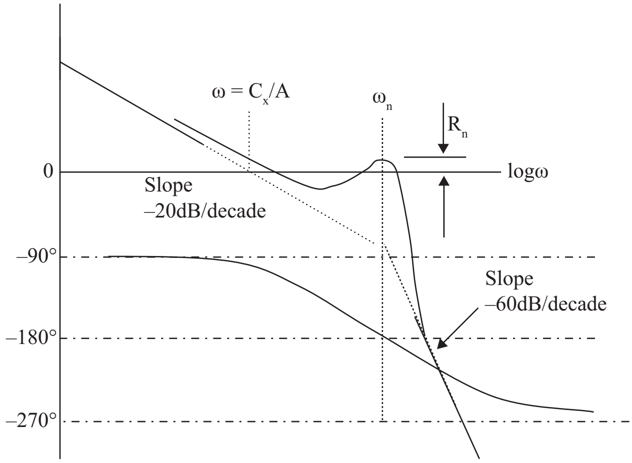

Figure 10.11. Bode plot for valve actuator open-loop transfer function.

From Equation 10.36, at the natural frequency:

- the amplitude ratio $R_n = \dfrac{C_x / A}{2\zeta\omega_n}$ (10.39)
- the phase angle is -180°.

10.6 STABILITY OF THE CLOSED-LOOP POSITION CONTROL SYSTEM

10.6.1 STABILITY CRITERION

For stability of the closed-loop system, the open-loop response transfer function must have an amplitude ratio that is less than unity when the phase lag is 180°. Thus, for the electrohydraulic position control system, it will be necessary to set the value of the power amplifier gain to a value that gives an open-loop amplitude ratio that is less than unity. In order to provide some margin it is usual to design the system so that the gain is 0.5, or −6 dB. The design should also aim to achieve a phase lag of less than around 140° when the amplitude ratio is unity (0 dB), that is, a phase margin of at least 40°.

10.6.2 SYSTEM DESIGN

The system frequency response analysis in Section 10.5.2 has shown that −180° phase lag occurs at the natural frequency when the amplitude ratio of the valve actuator system is $\dfrac{C_x / A}{2\zeta\omega_n}$. The block diagram shown in Figure 10.4 is modified in Figure 10.12 to include the effect of the fluid compressibility using the transfer function for the valve actuator system from Equation 10.32.

From Equation 10.39, which gives the amplitude ratio at the natural frequency when $\varphi = 180°$, a satisfactory stability margin for the system in Figure 10.12 requires that

$$\frac{C_x K_A K_V K_T}{2\zeta\omega_n A} = \frac{K_L}{2\zeta\omega_n} \le \frac{1}{2} \tag{10.40}$$

where K_L is the open-loop gain which $= K_A K_V K_T \dfrac{C_X}{A}$.

The value of the system open-loop gain required for stability can be obtained from Equation 10.40, where it is seen that for this to be satisfied $K_L \le \zeta\omega_n$. Typically, taking a value for ζ of 0.2 means that K_L needs to be

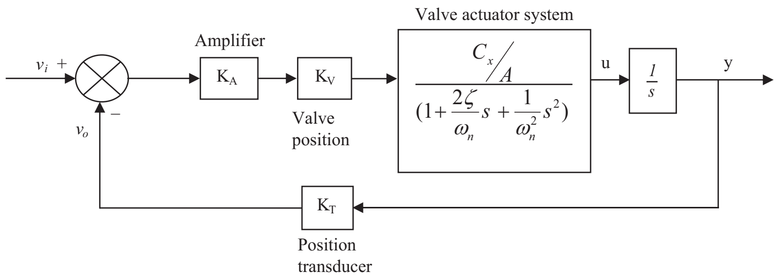

Figure 10.12. Electrohydraulic position control system block diagram.

$\leq \dfrac{\omega_n}{5}$. Consequently, having determined the hydraulic natural frequency enables an initial value of the system gain to be obtained. Adjustment of the amplifier gain K_A can be used to alter the system gain so as to provide a level that satisfies the stability criteria.

The valve gain, K_V, can be established by selecting one having the appropriate flow rating to achieve the required actuator velocity and having the desired frequency response. The valve performance is normally given in terms of the flow variation with input current or voltage at a given pressure drop so that the valve flow gain = $K_V C_X$ {(m^3/s)/A or (m^3/s)/V}.

In this analysis, the valve has been assumed to respond instantaneously. The valve dynamic performance is usually described by its frequency response and if this produces an insignificant phase lag at the system hydraulic natural frequency then the valve can be treated as a steady state gain. The valve manufacturer will show how the valve phase lag and amplitude ratio vary with frequency and this information can be used if necessary in the system Bode plot.

The feedback transducer gain, K_T, is usually selected to have a voltage range of ±10V for the required actuator movements about the mid-position. Thus, knowing the required valve input current enables an amplifier gain, K_A, to be selected that will give the value of K_L that provides the necessary gain and phase margins.

10.6.3 STEADY STATE ACCURACY

For the position control system in Figure 10.12, the closed-loop transfer function is given by

$$\frac{v_0}{v_i} = \frac{K_L}{K_L + s\left(1 + \dfrac{2\zeta}{\omega_n}s + \dfrac{1}{\omega_n^2}s^2\right)} \tag{10.41}$$

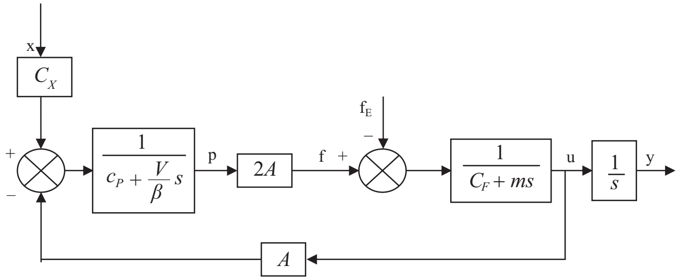

Figure 10.13. Valve actuator block diagram.

In the steady state, the "s" terms are zero and from Equation 10.41 we get $v_0 = v_i$ or $y = \dfrac{v_i}{K_T}$. The error signal, $v_i - v_0 = 0$ because the control valve has to be closed for the actuator to be at rest. Mathematically, there needs to be a zero input to the integral action of actuator velocity to displacement for steady state conditions to be obtained.

The linear analysis used to develop the open-loop transfer function excluded nonlinear effects such as leakage through the valve at the null position, electrical and mechanical hysteresis in the valve, and actuator friction forces (coulomb and stiction). All of these affect the steady state accuracy.

The block diagram in Figure 10.13 shows pressure and force from Equations 10.26, 10.27, and 10.28 for an equal area actuator with the piston in the central position when the perturbation pressure $p = p_1 = -p_2$. The diagram also includes changes in the external load force, f_E, as input to the system. For $f_E = 0$, the transfer function for y to changes in x is the same as that given in Equation 10.32 and shown in Figure 10.12.

For the closed-loop position system of Figure 10.12 the effect of changes in external force can be evaluated from Figure 10.13 by considering, for example, a fixed input voltage when v_I will be zero. It must be remembered that the system transfer function applies only for *changes* in the variables about a steady state condition where the initial values are known.

As has been described, the frequency response of the linearized system allows the system stability to be evaluated. However, it is necessary to examine the effects that the nonlinear elements have on the steady state accuracy of the system.

10.6.3.1 Valve Leakage

The linearization of the valve characteristics enabled the small perturbation technique to be used to obtain the variation in flow for small changes in pressure as represented by the flow coefficient C_P. For the ideal valve that closes completely (i.e., $x = 0$) in the null position the value of C_P is zero, which would cause very low damping and probably require a low system gain to obtain stability of the closed-loop system.

Conversely, for high load forces when the value of $(P_S - P_1)$ approaches zero the value of C_P will approach infinity. These variations in C_P are a major limitation of the linearization technique but in reality the valve will leak at the null position and it may even be deliberately underlapped to improve its flow characteristics. Consequently, the effect of leakage through the valve for positions close to the null can be evaluated by treating the valve as underlapped as shown in Figure 10.14.

Underlapping results in the port width being greater than the width of the spool land so that some fluid passes from the port to the return (Q_{R1} in Figure 10.13) for $0 < X < X_L$. When $X > X_L$, the valve behaves as a zero lap valve.

The flows through the valve can be analyzed as follows

$$Q_1 = Q_{S1} - Q_{R1} = K(X + X_L)\sqrt{P_S - P_1} - K(X_L - X)\sqrt{P_1} \tag{10.42}$$

now

$$C_{X1} = \frac{\partial Q_1}{\partial X} = K\sqrt{P_s - P_1} + K\sqrt{P_1}$$
$$for\ P_1 = {P_S}/{2},\ C_{X1} = 2K\sqrt{\frac{P_S}{2}} \tag{10.43}$$

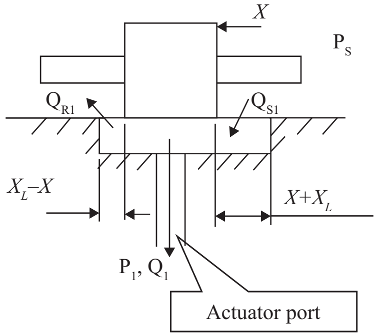

Figure 10.14. Valve underlap.

and

$$C_{p1} = \frac{\partial Q_1}{\partial P_1} = -\frac{K(X + X_L)}{2\sqrt{P_s - P_1}} - \frac{K(X_L - X)}{2\sqrt{P_1}}$$

$$for\ P_1 = {P_S}/{2}\ and\ X = 0,\ C_{p1} = \frac{-K}{\sqrt{P_S/2}} X_L \qquad (10.44)$$

Equation 10.42 can be used to obtain the variation of P_1 with X for zero port flow (e.g., ports blocked). The same approach can be used to obtain the variation of P_2 with X hence giving the change in actuator load pressure as the valve is moved through the leakage range as shown in Figure 10.15.

This information is obtained from tests carried out by the valve manufacturer and is referred to as the pressure gain. It can be seen, therefore, that in the closed-loop position control system, variations in the load pressure will require changes in the valve position in order to provide the requisite actuator load pressure. This will result in $v_i \neq v_0$, which means that the load force will affect the steady state accuracy.

The perturbation analysis can be applied to the underlapped valve in order to provide an indication of the effect on steady state performance.

Now for the valve: $q_1 = C_{X1}x + C_{P1}p$. For steady state conditions, $q_1 = 0$ so $C_{x1}x = -C_{p1}p_1$
And from Equations 10.43 and 10.44

$$p_1 = -\frac{C_{x1}}{C_{p1}} x = \frac{P_S}{X_L} x \qquad (10.45)$$

{for $X = 0$ (null position) and $P_1 = \dfrac{P_S}{2}$ (zero load force)}.

From symmetry of the valve characteristics, the change in the pressure P_2 is

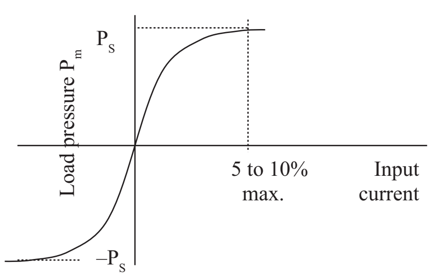

Figure 10.15. Valve pressure gain.

$$p_1 = p = -p_2 \tag{10.46}$$

Consequently, the change in the actuator force, f, is given by

$$f = 2pA \tag{10.47}$$

Substituting Equation 10.45 into 10.47 gives

$$f = 2AP_S \frac{x}{X_L} \tag{10.48}$$

The block diagram in Figure 10.16 shows the parameters involved in the steady state operation of the closed-loop position system when the actuator force change, f, must be equal to the change in the external force, f_E. The change in the actuator force is created by a change in the output position, Y. For zero changes in the input command signal (i.e., $v_i = 0$) the corresponding change in x is given by

$$x = -K_T K_A K_V y \tag{10.49}$$

Thus, from Equation 10.48 we get

$$f = f_E = -2AP_S \frac{K_T K_A K_V y}{X_L}$$

Or the system stiffness is

$$\frac{f_E}{y} = -\frac{2AP_S K_T K_A K_V}{X_L} \tag{10.50}$$

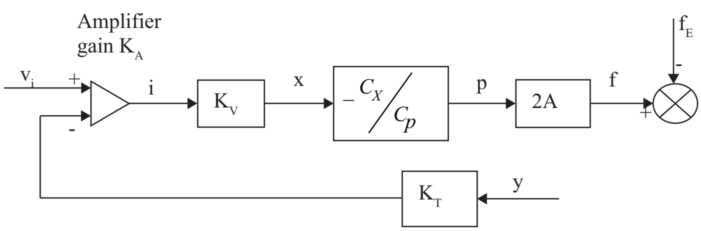

Figure 10.16. Block diagram of steady state conditions.

This linear analysis is useful to illustrate the effect of system parameters on the system stiffness, but is rather idealized in that it takes no account of valve hysteresis. Because of the linearization, the value of the stiffness only applies to small changes in X and the pressures P_1 and P_2 about any steady-state operating condition although the valve characteristics are very linear in the underlapped region.

This effect is referred to as the stiffness of the closed-loop system and to increase this, and consequently reduce the steady state error, the system gain needs to be increased. However, increases in the system gain are limited by the available gain margin and additional methods are sometimes required to avoid this problem.

10.6.3.2 Valve Hysteresis

The magnetic hysteresis in the valve solenoids and friction in the components will affect the valve position for a given input current as shown in Figure 10.17. As in Section 10.6.3.1, increasing the system gain will reduce the error voltage required to provide the necessary valve current to hold the valve in the appropriate position.

10.7 THE IMPROVEMENT OF CLOSED-LOOP SYSTEM PERFORMANCE

In addition to controlling actuator position, electrohydraulic systems are used to control output variables that include:

- Actuator velocity
- Actuator force

10.7.1 POSITION CONTROL

As was shown in Section 10.6, the steady state accuracy of a position control system is limited by the nonlinear characteristics inherent in

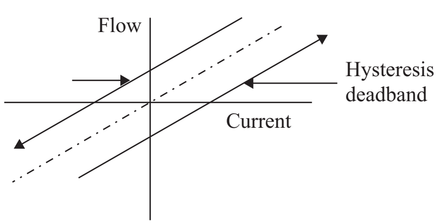

Figure 10.17. Valve hysteresis.

the components. Increasing the system open-loop gain can reduce these effects but this action is limited by stability considerations. The same limitation naturally arises if it is desired to increase the system response by increasing the system gain.

To improve steady state accuracy and/or system dynamic performance, compensation networks can be applied to the electronic control system and some of the available methods will be described.

10.7.2 VELOCITY CONTROL

The use of a velocity transducer to provide a feedback signal in the place of actuator position will provide a velocity control system. The closed-loop transfer function obtained from Figure 10.12 when using velocity feedback with a velocity transducer gain, K_T, is given by

$$\frac{u}{v_i} = \frac{K_F}{\left(1 + \frac{2\zeta}{\omega_n} s + \frac{1}{\omega_n^2} s^2\right) + K_L} \tag{10.51}$$

where

$$K_F = K_A K_V \frac{C_X}{A} = \frac{K_L}{K_T} \tag{10.52}$$

In the steady state we get

$$\frac{u}{v_i} = \frac{K_F}{1 + K_L} = \frac{K_L}{K_T(1 + K_L)}$$

If the error voltage is zero, $u = \frac{v_i}{K_T}$ but this can be obtained only if $K_L >> 1$. This is because the valve has to be open in order to pass the necessary flow. As the open-loop gain is increased, the required error voltage to produce a given valve displacement will be reduced.

In this system, which is second order, an integrating amplifier can be placed in the forward path so that the closed-loop transfer function of Equation 10.51 becomes

$$\frac{u}{v_i} = \frac{K_F}{s\left(1 + \frac{2\zeta}{\omega_n} s + \frac{1}{\omega_n^2} s^2\right) + K_L} \tag{10.53}$$

For steady state conditions, following a step input in v_I, the voltage error signal supplied to the integrator will be zero, the integrator output having the level appropriate to provide the required valve flow. Putting the "*s*" terms to zero in Equation 10.53 produces the same result. This system is now of third order, and therefore its stability margin will have to be determined in the same manner as that used for the position control.

10.7.3 PRESSURE CONTROL

In systems where it is required to control the actuator force, a pressure transducer can be used to provide a feedback signal so that the system is closed loop on actuator pressure instead of its position. This will provide actuator force control so that the actuator will be displaced such that the force is kept constant. Clearly, this process will be affected by the characteristics of the load. This type of control is frequently used in material testing machines where it is desired to vary the force in a controlled manner and the actuator displacement will then depend on the mechanical stiffness of the test component.

From Figure 10.13, the block diagram for the pressure control system is shown in Figure 10.18 where the pressure is fed back by the pressure transducer having a gain K_T.

From Figure 10.18 with $f_E = 0$ (constant load force), we get

$$\frac{p}{v} = \frac{K_A K_V C_X (C_f + ms)}{2A^2 + \left(C_P + \frac{V}{\beta}s\right)(C_f + ms)} \tag{10.54}$$

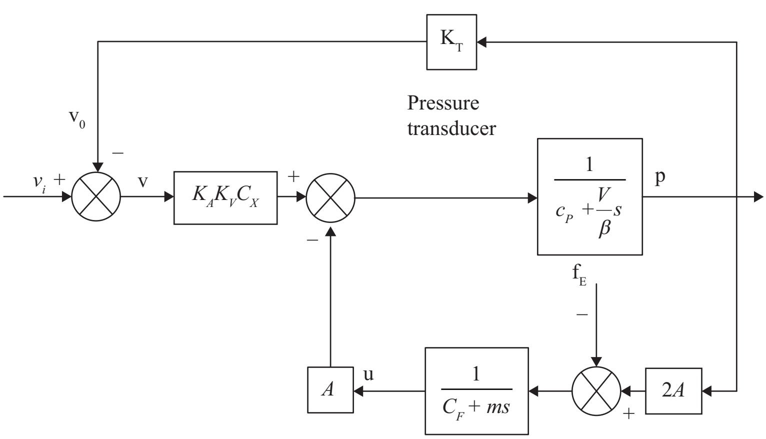

Figure 10.18. Pressure control.

For the closed-loop system the transfer function is given by

$$\frac{p}{v_i} = \frac{K_A K_V C_X (C_f + ms)}{2A^2 + \left(C_P + \frac{V}{\beta}s\right)(C_f + ms) + K_A K_V C_X K_T (C_f + ms)} \tag{10.55}$$

The steady state gain is given by

$$\frac{p}{v_i} = \frac{K_A K_V C_X C_f}{2A^2 + C_P C_f + K_A K_V C_X C_f K_T} \tag{10.56}$$

For this situation, where the load force is constant, the actuator is moving at a velocity such that the viscous friction (coefficient C_f) absorbs the actuator force from the pressure increase. Clearly, if this coefficient is zero then the actuator will be uncontrolled because there will not be any increase in the pressure. Pressure control can be combined with position control so that, in a press system, it will only be operational when the pressing action takes place.

In the fatigue test systems mentioned above the load force will increase with actuator displacement due to the component mechanical stiffness and the steady state open-loop gain will be

$$\frac{p}{v_i} = \frac{K_A K_V C_X}{C_P + K_A K_V C_X K_T} \tag{10.57}$$

For this situation, the actuator displacement will increase to a level such that the increased pressure force equals the load force when the actuator velocity, u, will be zero. As with the position control, this steady state gain will vary due to the system nonlinear characteristics and improvements can be obtained by the use of integral plus proportional control.

10.8 COMPENSATION TECHNIQUES

The objective of most compensation systems is to provide a high gain at low frequency in order to minimize steady state errors whilst modifying the system frequency response so as to obtain the requisite gain and phase margins. Some of the available techniques are described.

10.8.1 INTEGRAL PLUS PROPORTIONAL COMPENSATION

The transfer function for this element is given by

$$K_p + \frac{K_I}{s} = \frac{K_p s + K_I}{s} = \frac{K_I}{s}\left(1 + \frac{K_P}{K_I}s\right) \tag{10.58}$$

For frequency inputs the amplitude ratio and phase change are given by

$$R = \frac{K_I\sqrt{1+\left(\frac{K_P}{K_I}\omega\right)^2}}{\omega} \tag{10.59}$$

$$\varphi = -90° + \tan^{-1}\left(\frac{K_p\,\omega}{K_I}\right) \tag{10.60}$$

when

$$\omega >> \frac{K_1}{K_P}; R \rightarrow K_P$$

The frequency response is shown in Figure 10.19 where at low frequencies (when $\omega \rightarrow 0$), $R \rightarrow \frac{K_1}{\omega}$, which increases with reducing

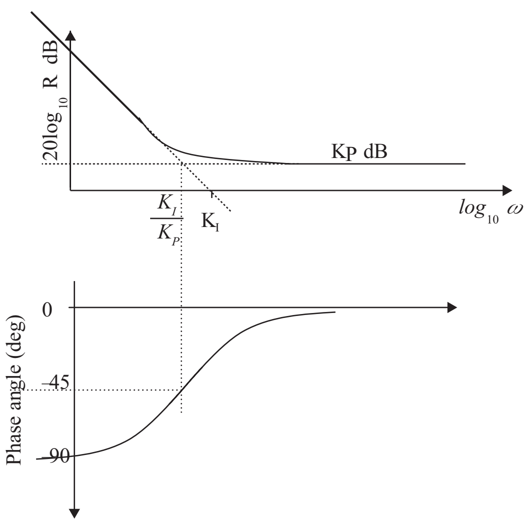

Figure 10.19. Frequency response for integral plus proportional control.

frequency (i.e., has the characteristics of an integrator), so eliminating steady state errors. The ratio $\frac{K_1}{K_P}$ needs to be selected in relation to the hydraulic natural frequency and the system frequency response as shown in Figure 10.11.

If this ratio is too low the system response will be slow and if it is too high it will not have the required effect. The system gains can be selected so as to provide the requisite gain and phase margins.

10.8.2 PROPORTIONAL PLUS DERIVATIVE CONTROL

The transfer function for this compensator is given by

$$K_P + K_D s = K_P\left(1 + \frac{K_D}{K_P}s\right) \tag{10.61}$$

This has a frequency response that is given by

$$R = K_P\sqrt{\left(1 + \left(\frac{K_D}{K_P}\omega\right)^2\right)} \tag{10.62}$$

$$\varphi = \tan^{-1}\left(\frac{K_D}{K_P}\omega\right) \tag{10.63}$$

As can be seen from Figure 10.20, this compensator provides a phase advance characteristic in that the phase angle is positive, which will reduce the overall phase lag in the system. However, the amplitude ratio also increases and is zero at low frequencies so there is no effect on the steady state gain. A major problem with derivative control is that it will amplify noisy input signals and for this reason it is often not used. It is more common to use a phase advance compensator, which is described in the next section.

10.8.3 PHASE ADVANCE COMPENSATION

As an alternative to derivative control phase advance can be provided by a transfer function of the form

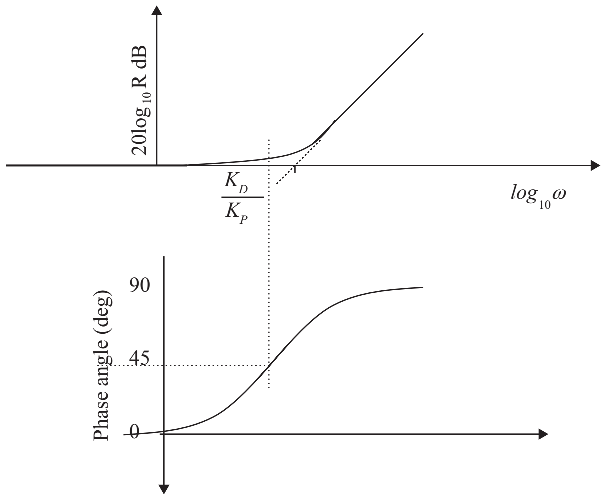

Figure 10.20. Frequency response for proportional plus derivative control.

$$\frac{1+\alpha\tau s}{1+\tau s} \tag{10.64}$$

By putting α > 1 the magnitude of the phase angle of the numerator is always greater than that of the denominator and the overall phase angle is zero at high and low frequencies.

Thus

$$R = \sqrt{\frac{1+(\alpha\tau\omega)^2}{1+(\tau\omega)^2}} \tag{10.65}$$

and

$$\varphi = \tan^{-1}(\alpha\tau\omega) - \tan^{-1}(\tau\omega) \tag{10.66}$$

As shown in Figure 10.21, the phase advance system provides maximum phase advance at a frequency that is determined by the selection of the values of the coefficients α and τ so as to stabilize an unstable system and/or increase the natural frequency hence improving its response.

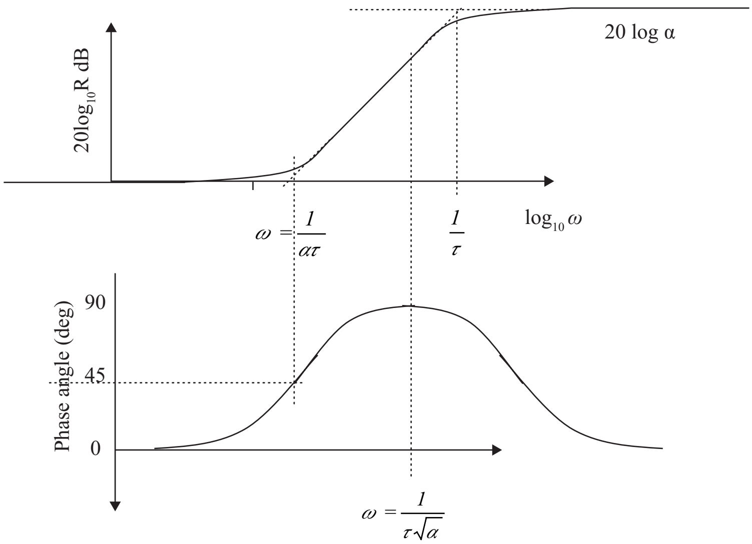

Figure 10.21. Phase advance frequency response.

10.8.4 PROPORTIONAL, INTEGRAL, AND DERIVATIVE CONTROL

There is a general approach to compensation, which is referred to as proportional, integral, and derivative (PID) compensation in the forward path, or PID. This can be represented by the following transfer function

$$K_P + \frac{K_I}{s} + K_D\left(\frac{1+\alpha\tau s}{1+\tau s}\right)$$

The frequency response of the compensator can be modified by adjusting the values of:

1. The proportional gain coefficient K_P.
2. The integral gain coefficient K_I.
3. The derivative gain coefficient K_D.
4. The time constant τ and phase advance term α.

The electronic control amplifiers for proportional and servo valves usually contain an adjustable PID system. The application of PID, because of the large combination of adjustable coefficients that are available, is difficult to discuss in a generalized fashion and it is often "tuned" after it has been installed in the system.

10.8.5 PRESSURE FEEDBACK

Pressure feedback has been discussed as a means for providing force control but it can also be fed back inside a position control loop for increasing the damping ratio of the open-loop system as shown in Figure 10.22. Obtaining the open-loop transfer function can show this effect. Thus

$$p = \left(\frac{1}{C_P + \frac{V}{\beta} s} \right) \left[(i - K_{PF}\, p) K_V C_X - Au \right]$$

Rearranging this equation gives

$$\left[\left(C_P + \frac{V}{\beta} s \right) + K_{PF} K_V C_X \right] p = K_V C_X i - Au \tag{10.67}$$

Also

$$\frac{u}{p} = \frac{2A}{C_f + ms} \tag{10.68}$$

Substituting p from Equation 10.68 into Equation 10.67 gives

$$\left[\left(C_P + \frac{V}{\beta} s \right) + K_{PF} K_V C_X \right] \frac{(C_f + ms)}{2A} u = K_V C_X i - Au$$

And so

$$\left(\frac{mV}{2A^2 \beta} s^2 + \left(\frac{mC_P}{2A^2} + \frac{C_f V}{2A^2} + \frac{K_{PF} K_V C_X m}{2A^2} \right) s + 1 \right) u = \frac{K_V C_X}{A} i \tag{10.69}$$

The term $\dfrac{C_f C_P}{2A^2}$ has been neglected as being <<1.

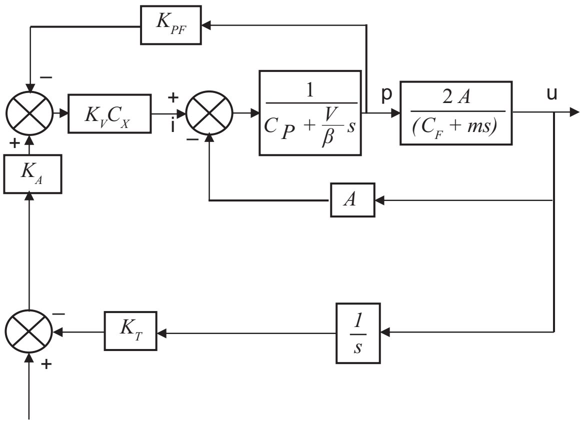

Figure 10.22. Pressure feedback control ($f_E = 0$).

On comparing Equation 10.69 with that for the damping factor in Equation 10.31 it is seen that the pressure feedback has introduced the term $\frac{K_{PF}K_V C_X m}{2A^2}$, which increases the value of the damping factor and thus reduces the resonance at the natural frequency in the Bode plot of Figure 10.11. This can provide a considerable benefit in some systems where the damping factor is low (typically <0.2), which may allow the system gain to be increased whilst maintaining an adequate stability margin.

A problem that can arise with pressure feedback is that it reduces the system stiffness and, consequently, increases the steady state error as discussed in Section 10.6.3.1. The use of a high pass filter can prevent this problem as it excludes all signals below a set frequency. This will, therefore, only pass the feedback pressure transducer signal when transient changes occur and not when the system is in the steady state.

10.9 HYDROSTATIC SYSTEMS

Different types of hydrostatic transmissions are presented in Section 7.7 and this section is concerned with the evaluation of their static and dynamic performance. Such analyses are necessary in order to avoid stability problems and provide information for the design of appropriate control systems.

10.9.1 PUMP-CONTROLLED HYDROSTATIC SYSTEMS

A circuit for a hydrostatic transmission is discussed in Section 7.7.1 for which a typical diagram is shown in Figure 7.29. The following analysis develops the equations for the dynamic changes for this system driving an inertial load.

Nomenclature

Q_m = motor flow — Ω_m = motor speed
Q_P = pump flow — Ω_P = pump speed
C_f = speed dependent friction coefficient
V = pressurized volume — J = rotary inertia
β = bulk modulus — T_M = motor torque
P = system pressure — T_L = load torque
C_L = leakage coefficient
D_m = motor displacement — D_p = pump displacement

10.9.1.1 Flow

Motor flow = pump flow – compressibility flow – leakage

$$Q_m = Q_p - \frac{V}{\beta}\frac{\mathrm{d}P}{\mathrm{d}t} - C_L P = \Omega_m D_m \tag{10.70}$$

For small changes about a steady state operating condition

$$\therefore q_p = D_m\ \omega_m + \left(\frac{V}{\beta}S + C_L\right)p \tag{10.71}$$

where the parameters in lower case letters denote small perturbations. For changes in the pump flow, that is, $q_p = 0$, Equation 10.71 gives

$$p = \left(\frac{q_p - \omega_m D_m}{C_L + \frac{V}{\beta}s}\right) \tag{10.72}$$

This shows that an increase in pump flow will increase the high pressure at a rate that is determined by the response of the motor speed to the change in the pressure, that is, the motor torque, which is discussed in the next section.

10.9.1.2 Torque

$$T_m = j\frac{d\Omega_m}{dt} + C_f\Omega_m + T_L = PD_m \text{ ideal motor torque} \tag{10.73}$$

For changes about a steady state operating condition

$$\Delta T_m - \Delta T_L = (J_s + C_f)\ \omega_m; \quad \Delta T_m = pD_m \tag{10.74}$$

Equations 10.72 and 10.74 are used to draw the block diagram shown in Figure 10.23, which represents the dynamic effect of changes in load torque on rotational speed.

From Figure 10.23 for the case of $\Delta T_L = 0$, the dynamic equation for the response of motor speed to changes in the pump flow is given by Open-loop transfer function

$$A = \frac{D_m^2}{(C_f + Js)\left(C_L + \frac{V}{\beta}s\right)}$$

$$\frac{\omega_m}{q_p} = \frac{1}{D_m}\left\{\frac{A}{1+A}\right\} = \frac{1}{D_m}\left\{\frac{D_m^2}{D_m^2 + (C_f + Js)\left(C_L + \frac{V}{\beta}s\right)}\right\}$$

$$\frac{\omega_m}{q_p} = \frac{1}{D_m k_1\left(1 + \frac{(JC_L + C_f)s + \frac{VJ}{\beta}s^2}{k_1 D_m^2}\right)} \tag{10.75}$$

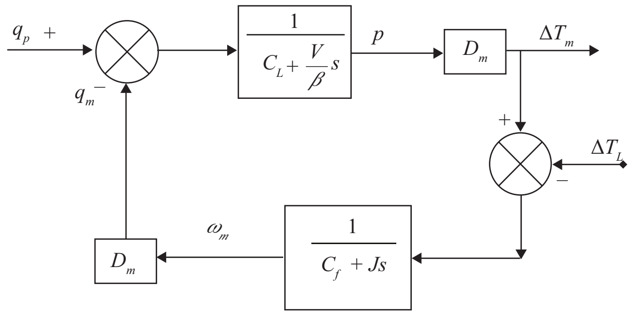

Figure 10.23. Hydrostatic system block diagram.

A second-order system of the standard form is given by

$\dfrac{1}{1+\dfrac{2\varsigma}{\omega_n}s+\dfrac{1}{\omega_n^2}s^2}$; ω_n = natural frequency, ζ = damping factor.

For Equation 10.75

$$k_1 = 1 + \frac{C_f C_L}{D_m^2} \tag{10.76}$$

From Equation 10.75 the steady state gain

$$K_{SW} = \frac{1}{k_1 D_m} = \frac{1}{D_m\left(1+\dfrac{C_f C_L}{D_m^2}\right)} \tag{10.77}$$

damping factor $$\zeta = \frac{1}{2\sqrt{k_1}D_m}\left\{C_f\sqrt{\frac{V}{J\beta}} + C_L\sqrt{\frac{J\beta}{V}}\right\} \tag{10.78}$$

natural frequency $$\omega_n = \sqrt{\frac{\beta D_m^2\ k_1}{VJ}} \tag{10.79}$$

This linearized model of the hydrostatic transmission with the assumed variations in the frictional resisting force and the leakage gives a second-order response of the motor speed to changes in pump flow. The steady state gain of Equation 10.77 shows that the motor speed will be less than the ideal value of $1/D_m$ because of the leakage and friction coefficients. Control of the pump flow provides a variable motor speed including its reversal with an over center pump.

If it is desired to evaluate the response of motor speed to changes in load torque ΔT_L the transfer function can be derived from Figure 10.23 for $q_p = 0$,which gives

$$\frac{\omega_m}{\Delta T_L} = -\frac{\left(C_L + \frac{V}{\beta}s\right)}{\left(D_m^2 + C_f C_L\right) + \left(JC_L + C_f \frac{V}{\beta}\right)s + \frac{VJ}{\beta}s^2} \tag{10.80}$$

The denominator is of second order with the same values for ω_n and ζ as for the response to a change in input flow. The steady state gain is

$$K_{SS} = -\frac{C_L}{D_m^2 k_1} \tag{10.81}$$

Equation 10.80 shows that for an increase in load torque the motor speed is reduced because of increased leakage that arises from the associated increase in pressure. Changes in oil temperature will have associated changes in the leakage coefficient and possibly in the value of the friction coefficient. As a consequence, it can be difficult to be precise about the motor speed over a range of operating conditions.

10.9.2 MOTOR CONTROL SYSTEMS (SECONDARY CONTROL)

As described in Chapter 7.4, a variable displacement pump operating at a constant pressure can provide hydrostatic transmission to one or more motors having variable displacement each of which is controlled in a closed loop with the motor speed. The dynamic variation in the controlled motor speeds to changes in load torque with the pump supply pressure, *P,* kept constant can be modeled as follows:

10.9.2.1 Torque

The motor output torque is given by

$$T_m = J\frac{d\Omega_m}{dt} + C_f\Omega_m + T_L = PD_m \text{ ideal motor torque} \tag{10.82}$$

For small changes about a steady state operating condition we get

$$\therefore P\Delta D_m = Js\ \omega_m + C_f\ \omega_m + \Delta T_L \tag{10.83}$$

10.9.2.2 Displacement Controller

The motor displacement is controlled in response to a speed error signal. With a speed transducer gain K_F and a controller gain K_G, a selected input demand speed of Ω_i will create an input voltage $K_F\Omega_i$ to the controller. The steady state motor displacement is given by

$$D = D_{\max} \text{ for } \Omega_j \geq \Omega_m \text{ and } D = D_{\max} + K_G K_F(\Omega_j - \Omega_m)$$

$$\text{for } \Omega_m \geq \Omega_j$$

The motor displacement has a minimum value of $D_{\min}$.

The corresponding motor torques are

$$T_{\max} = PD_{\max} \text{ and } T_{\min} = PD_{\min}$$

For $\Delta\Omega_i = 0$ and assuming that the displacement controller has a first-order response with a time constant T_C, changes in displacement with changes in motor speed are given by

$$\Delta D_m = -\frac{K_G K_F}{1 + T_C^S}\ \omega_m \tag{10.84}$$

Thus, for this proportional controller the motor displacement reduces with increases in the speed above the set value Ω_i. For the pump controller maintaining the pressure at a constant level this will cause the motor torque to reduce with motor speed and this is shown in the control diagram

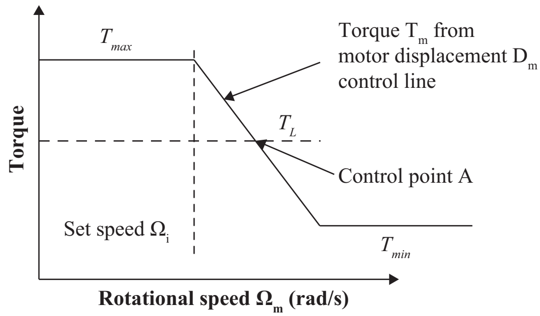

Figure 10.24. Steady state torque diagram of the load and motor control.

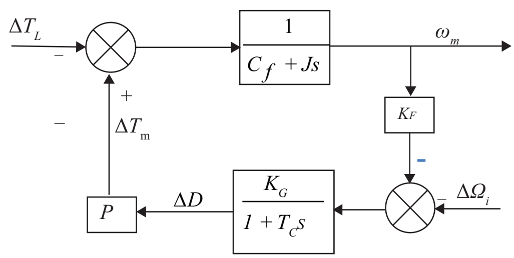

Figure 10.25. Block diagram for secondary control system.

in Figure 10.24. The dynamic equations express the variation in motor speed with changes in load torque, T_L, about the values at the operating point A.

From Figure 10.24, it is seen that the motor control will act where its operating line crosses that of the load at point A.

The block diagram for this system that is based on the dynamic Equations 10.83 and 10.84 is shown in Figure 10.25.

10.9.2.3 Closed Loop

For no change in the control speed setting, $\Delta\Omega_i = 0$. Figure 10.25 gives

$$\frac{\omega_m}{\Delta T_L} = -\frac{(1 + T_C s)}{(PK_G K_F + C_f) + (T_C C_f + J)s + JT_C s^2}$$

$$= -\frac{1}{\left(PK_GK_F + C_f\right)}\left\{\frac{\left(1+T_Cs\right)}{1+\dfrac{\left(T_CC_f + J\right)}{\left(PK_GK_F + C_f\right)}s + \dfrac{JT_C}{\left(PK_GK_F + C_f\right)}s^2}\right\} \tag{10.85}$$

The steady state gain

$$K_{ST} = -\frac{1}{\left(PK_GK_F + C_f\right)} \tag{10.86}$$

The gain K_{ST} is negative which, referring to Figure 10.25, is because an increase in load torque will reduce the motor speed from the point A.

$$\omega_n^2 = \frac{(PK_GK_F + C_f)}{JT_C} \tag{10.87}$$

and

$$\zeta = \frac{(J + C_fT_C)}{2\sqrt{JT_C(PK_GK_F + C_f)}} \tag{10.88}$$

Thus, it is seen that when the load torque increases with this proportional control there will be a steady state error in that the motor controlling speed will be higher than the set value as shown in Figure 10.24, that is, the speed at point A is greater than the set value Ω_i.The speed error can be reduced by increasing the gain term K_G. For this controller the use of compensation elements such as PID as discussed in Section 10.8 can be applied so that this steady state error can be substantially reduced if required whilst providing a stable system. Unlike the hydrostatic system in Section 10.9.1 these system parameters are independent of the fluid compressibility and viscosity, the fluid volume, and the motor displacement.

10.9.3 HYDROSTATIC POWER TRANSMISSION FOR A WIND TURBINE

Hydrostatic transmission systems are applied extensively in a wide range of industrial machines using either primary or secondary control technology. However, there are some applications where the type of transmission does not fit into either of these two categories. An example of this is the circuit that has been developed for transmitting the power in a wind turbine [10] from the turbine to the generator as discussed in Section 7.7.5 using the circuit in Figure 7.33.

For this system, it is necessary to carry out dynamic studies in order to evaluate a suitable control system for satisfactory operation of the wind turbine over the range of operating conditions. The start-up of the wind turbine is initiated by changing the blade pitch, which is controlled so as to maintain the generator speed at the required level (synchronous speed) for connection of the generator to the grid. Once connection has been established the rotational speed of the generator can be considered to be effectively fixed because of the high stiffness of the generator torque with respect to its speed.

Over the range of wind speeds the wind turbine speed can be varied in order to maximize its efficiency. This is done by altering the hydraulic motor displacement, which will alter the flow demand from the pump whilst the generator speed remains substantially unchanged. During steady state conditions, the high pressure is created by the turbine torque. On reducing the motor displacement there will be a transient increase in this pressure due to the turbine rotary inertia, which will increase the hydraulic resisting pump torque and consequently the turbine/pump speed will reduce. Steady state conditions will eventually be established when the turbine speed has reduced to give the appropriate value of pump output flow.

The main benefits of hydrostatic transmissions over most of the mechanical drives can be summarized as they:

- Have a relatively higher power to weight ratio.
- Allow stepless variable speed control of the turbine.
- Avoid the requirement for a frequency converter and power amplifier so that generators can be connected directly to the electric grid.

There are a number of circuit solutions that can be applied to the wind turbine and that chosen here has been evaluated thoroughly and tested in two turbines [10, 11]. The main limitation to building such turbines is that currently there are no pumps and motors available of sufficient power capacity. The overall efficiency level of conventional pumps and motors in hydrostatic transmissions is lower than that of most mechanical transmissions. Using digital valve operation in pumps and motors, as described in Section 2.7, satisfactory levels of efficiency could be achieved.

In order to evaluate the process, it is necessary to determine the transient response of turbine speed to changes in motor displacement and suitable parameters for stable operation of the control system.

The flow from the pump provides the flow required by the motor plus that for leakage and the fluid compressibility which gives

$$D_m\Omega_m + \frac{V}{\beta}\frac{\mathrm{d}P}{\mathrm{d}t} = D_p\Omega_p - C_L P, \tag{10.89}$$

Ω_m is the motor speed, Ω_p is the pump speed.

This equation can be linearized in a steady state operating condition. Changes in parameters are denoted by lower case letters. The Laplace operator "s" is used to represent changes in parameters with time. Thus equation gives

$$p = \frac{(D_p\omega_p - D_m\omega_m - \Omega_m d_m)}{(C_L + \frac{V}{\beta}s)} \tag{10.90}$$

The turbine torque, T_W, is a function of the turbine speed, Ω_p for a given wind speed. This has to provide torque to drive the pump and the inertia during speed changes. Thus

$$T_w = f(\Omega_p) = PD_p + J\frac{\mathrm{d}\Omega_p}{\mathrm{d}t} \tag{10.91}$$

And linearizing gives

$$\Delta T_w = pD_p = \Delta f(\Omega_p) - Js\omega_p = \frac{\partial T_w}{\partial \Omega_p}\omega_p - Js\omega_p = C_T\omega_p - Js\omega_p$$

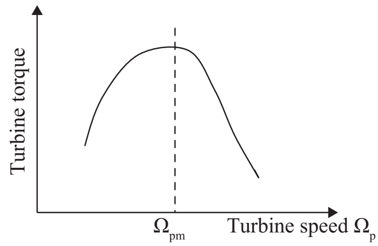

Figure 10.26. The relationship between turbine torque and speed at a given wind speed.

Here, C_T is the local slope of turbine torque to speed characteristic, which is typically as shown in Figure 10.26. This gives for changes in the pressure

$$p = \frac{(C_T - Js)\omega_p}{D_p} \tag{10.92}$$

Using the two equations for the pressure change p gives for changes in turbine speed

$$\omega_p = \frac{1}{D_p}\frac{(D_m\omega_m + \Omega_m \mathrm{d}_m)}{((1-a)+(b-c)s+\mathrm{d}s^2)} \tag{10.93}$$

where

$$a = \frac{C_L C_T}{D_p^2};\ b = \frac{C_L J}{D_p^2};\ c = \frac{VC_T}{\beta D_p^2};\ d = \frac{JV}{\beta D_p^2} = \frac{1}{\omega_n^2} \quad (\omega_n = \text{hydraulic natural frequency})$$

An induction generator is directly driven by the motor and its torque, T_G, is created by the difference between the shaft speed and the grid synchronous speed. The relationship between the ideal motor torque, T_M, the motor speed, and the pressure is

$$T_m = PD_m = (J_m + J_G)\frac{d\Omega_m}{dt} + C_m\Omega_m + T_G \ \text{ and } \ T_G = K_{mg}(\Omega_m - \Omega_{ms}) \tag{10.94}$$

where

Ω_{ms} = synchronous frequency,

J_m and J_G are the motor and generator inertias, and

K_{mg} is the gain of the generator.

Synchronous frequency is taken as the grid frequency.

Using typical values it is generally found that the generator gain K_{mg} is high so that for changes in the operating conditions the resulting changes in the motor speed are small enough to be neglected in Equation 10.93 (the generator always operates at near synchronous speed). Also, the value of "a" is found to be << 1 so that making these assumptions the variation in turbine speed with changes in motor displacement is a second-order equation which, from Equation 10.93, has a damping factor ζ

$$\zeta = \frac{1}{2}\left(\frac{C_L J_p}{D_p^2} - \frac{VC_T}{\beta D_p^2}\right)\omega_n = \frac{1}{2D_p}\left[C_L\sqrt{\frac{J_p\beta}{V}} - C_T\sqrt{\frac{V}{J_p\beta}}\right] \tag{10.95}$$

The first term is the normal hydraulic damping factor and so this is modified by a term containing the coefficient C_T. It is necessary for stability that ζ is positive and for this to be so we get

$$C_T \le C_L\left[\frac{J_p\beta}{V}\right] \tag{10.96}$$

From Figure 10.26 for speeds greater than the rated value, Ω_{mg}, the value of C_T is negative, which increases the value of the damping factor. However, for turbine speeds less than Ω_{mg}, C_T is positive which will reduce the damping factor with increasing wind speed so it would be possible that excessive speed oscillations could occur if C_T exceeds the value in Equation 10.96.

10.9.3.1 Turbine Speed Control

The use of a proportional closed-loop turbine speed control can be applied by reducing the motor displacement for speeds greater than the set value ω_{ps} as shown by the control line in Figure 10.27.

Also seen in Figure 10.27 is the relationship between motor displacement and turbine speed for a fixed motor speed. The dotted line shows the speed increase due to increased leakage with pressure hence with the wind

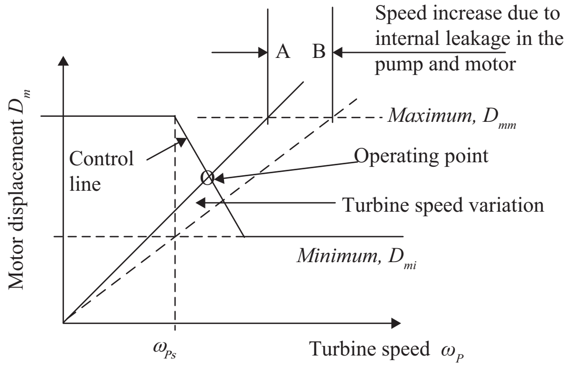

Figure 10.27. Relationship between motor displacement and turbine speed and closed-loop control for a fixed speed of the motor [9, 10].

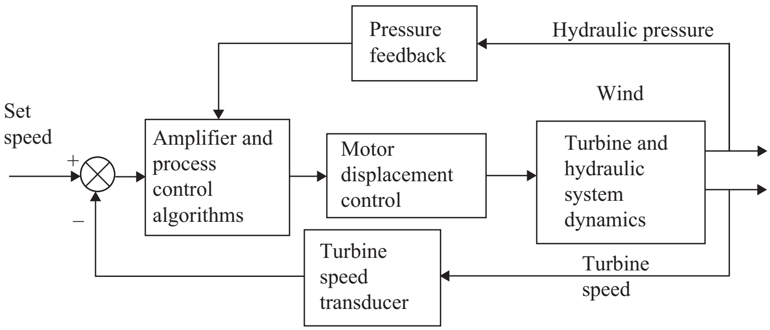

Figure 10.28. Turbine speed control system block diagram.

speed. The closed-loop condition arises at the intersection of the control and turbine speed lines.

The block diagram of the system is shown in Figure 10.28, which represents the processes in the various components with a closed-loop control of turbine speed. The block diagram also shows the addition of pressure feedback, which can be used to increase the damping of the control loop.

Improvements in the dynamic performance of the control can be obtained from PID compensation, which can be set to provide sufficient damping to minimize unwanted oscillations in speed.

Digital control methods that have been used for controlling the turbine are discussed in [10].

10.10 SYSTEM FREQUENCY RESPONSE TESTS

As has been described, frequency response analysis can be used for the design of the electrohydraulic closed-loop system. For testing a physical system, it is clear that if the gain is set at too high a value the closed-loop system will be unstable, consequently, during commissioning the gain should be set at a low value so that frequency response tests can be carried out.

Frequency response tests on closed-loop systems can provide an open-loop Bode plot by the use of a Nichols chart that converts closed to open loop and open to closed-loop performance. However, the problem with closed-loop testing is that the input signal to the valve will vary with the frequency and it is normally desired to keep this constant because of nonlinearities in the valve.

Open-loop tests can be carried out by removing the feedback signal but the main difficulties involved in this are:

- The actuator position may drift slightly if there is any bias in the frequency input signal and eventually the actuator may hit the end stops.
- At low frequencies the actuator movement may be longer than the available stroke when it will be necessary to reduce the input signal amplitude at low frequencies. At high frequencies it may be necessary to increase the input signal in order to obtain actuator displacement amplitudes that can be measured with sufficient accuracy. This will, however, mean that the input amplitude to the valve is not constant for all frequencies.

The actuator mountings may introduce a low stiffness in the mechanical system (natural frequency ω_m) including the rod connection to the load, which can have an effect on the system dynamic performance. The natural frequency, ω_S, of the composite system is

$$\omega_S = \frac{\omega_n \omega_m}{\sqrt{\omega_n^2 + \omega_m^2}} = \frac{\omega_n}{\sqrt{1 + \left(\frac{\omega_n}{\omega_m}\right)^2}} \tag{10.97}$$

This can affect the system stability, which may depend on the location of the position transducer, that is, actuator rod to actuator cylinder or load to machine base. It can be seen from Equation 10.97 that the system natural frequency, ω_S, is significantly reduced if the two frequencies are equal or when $\omega_m < \omega_n$ and, consequently, it is recommended that the mechanical stiffness should be around 5 to 10 times that of the hydraulic system.

10.11 PUMP-CONTROLLED SYSTEMS

The use of valve control creates restrictive losses in hydraulic systems that are dissipated in the fluid as heat and can lead to low power transmission efficiencies. Low system efficiency increases the size of the prime mover and that of the cooling system. The advantage of the method over other forms of control is its compactness, high natural frequency, and flexibility.

However, as an alternative, the pump can be directly connected to the actuator as a hydrostatic system as shown in Figure 10.29 thus improving the power transmission efficiency to a theoretical level of 100 percent. The pump displacement is controlled by an electrohydraulic servo valve that operates in the closed-loop system in the same way as in the valve controlled system, that is, from the position voltage error signal.

Figure 10.23 shows the hydraulic circuit for a variable displacement pump driving an equal area actuator. As with rotary hydrostatic

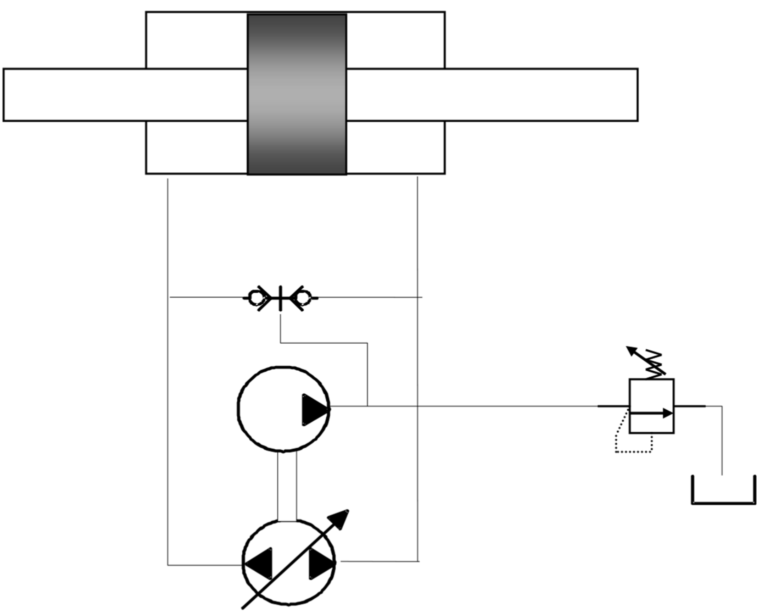

Figure 10.29. Actuator control using a variable displacement pump.

systems, a boost make-up flow is required to compensate for leakage losses in the pump. In this system, therefore, only one side of the actuator is pressurized, if the load force changes direction, the opposite side of the pump becomes pressurized and the appropriate boost check valve is opened.

Pump-controlled systems reduce the cooling requirement and the running cost and in some systems can be an attractive alternative to valve control. However, there are some major disadvantages that include:

a. The pump has to be dedicated to a single actuator or a mechanically grouped set of actuators. This can increase the capital cost in some installations but this must be balanced against reduced operating costs.
b. The pressurized volumes are likely to be larger than those in valve-controlled systems that will reduce the hydraulic natural frequency. However, in many actuator systems, the volume of the actuator dominates the hydraulic stiffness.
c. The frequency response of the pump displacement controller may be lower than that of an electrohydraulic control valve, which could reduce the available gain margin in the closed-loop system. However, taking into consideration point (b) concerning the hydraulic natural frequency this may not be a problem.

The transfer function of the hydraulic system in that controlled by a pump is similar to that given in Equation 10.32 for the servo valve control where C_x is the flow gain of the pump for changes in the pump displacement controller position x.

SUMMARY

Hydraulic power is used extensively in closed-loop control systems because the high force to mass ratio provides a relatively high natural frequency and fast response. Electrohydraulic control allows flexibility in terms of the range of parameters that can be controlled (e.g., position, velocity, pressure, and force) and also has the capability of incorporating elements that assist in improving the steady state and dynamic accuracy and the stability margin of a system.

This chapter develops the equations that describe the dynamic performance of actuator control systems, which are used to determine their frequency response and the stability margins so that appropriate components can be selected. Because of nonlinear characteristics, the analysis is based on the small perturbation technique, which enables the dynamic performance to be predicted. However, predicting the steady state performance has to take account of the various nonlinearities and some compensation methods for improving this and the dynamic performance are discussed.

The system can be approximated to a first-order response by assuming that the load has a negligible mass so that, having selected an appropriate actuator and the valve size to provide the necessary flow and actuator velocity, the time constant can be determined. The effect of load mass introduces a second-order mass/spring/damper into the open-loop transfer function having a natural frequency that is a function of the fluid bulk modulus, the inertial mass, the actuator area, and the compressed volume. This frequency enables an approximate value of the system gain to be determined using a simple design procedure that assumes a value for the damping ratio.

The transfer function obtained from the linearized equations provides a good estimate of the system performance for small changes but for high accuracy and/or fast response compensation circuits have to be incorporated into the electronic control. These can be made from either electronic components or can be incorporated into computer software where appropriate.

CHAPTER 11

PERFORMANCE ANALYSIS

11.1 INTRODUCTION

This chapter is devoted to the examination of case studies in order to illustrate the various aspects that are involved in the analytical design of fluid power systems and the evaluation of component performance.

11.2 METER-IN CONTROL

It is required to select an appropriate valve for an actuator having a 50-mm piston and 28-mm rod diameter that is supplied from a fixed displacement pump of 35 L min^{-1} capacity with a relief valve setting of 150 bar. The required actuator velocity is 0.2 m s^{-1} with an actuator force of 10 kN.

$$\text{Piston area}\left(A_{\mathrm{P}}\right)=\frac{\pi\, 0.05^2}{4}=1.96\times 10^{-3}\ \mathrm{m}^2$$

$$\text{Pump flow}\ (Q_{\mathrm{P}})=\frac{35}{1000}\frac{1}{60}=5.8\times 10^{-4}\ \mathrm{m}^3\mathrm{s}^{-1}$$

$$\text{Max. velocity}\ (U_{\mathrm{e}})=\frac{Q_{\mathrm{P}}}{A_{\mathrm{P}}}=0.3\ \mathrm{m\,s}^{-1}$$

For an actuator force of 10 kN, the piston pressure (P_{p}) will be given by:

$$P_{\mathrm{p}}=\frac{10\times 10^3}{1.96\times 10^{-3}}=51\times 10^5\,\mathrm{N\,m}^{-2}=51\ \text{bar}$$

For the relief valve set at 150 bar, the pressure drop across the restrictor (P_v) will be:

$$P_v = 150 - 51 = 99 \text{ bar}$$

For an actuator velocity of 0.2 m s^{-1}:

The supply flow

$$Q_S = \frac{0.2}{0.3} \times 35 = 23.3 \text{ L min}^{-1} = 23.3 \times \frac{10^{-3}}{60} = 3.9 \times 10^{-4} \text{m}^3 \text{ s}^{-1}$$

Re-arranging the orifice equation, the restrictor area (A) can be found from:

$$Q_s = C_q A \sqrt{\frac{2 P_v}{\rho}}$$

$$\therefore A = \frac{Q_s}{C_q \sqrt{\frac{2 P_v}{\rho}}}$$

Therefore, assuming that the flow coefficient is 0.65 and the fluid density is 870 kg m^{-3}, the restrictor area, A, is given by:

$$A = \frac{3.9 \times 10^{-4}}{0.65 \sqrt{\frac{2 \times 99 \times 10^5}{870}}} = 4 \times 10^{-6} \text{m}^2$$

This is equivalent to a restriction diameter of 2.25 mm. Alternatively, obtain a restrictor valve of an appropriate size from manufacturers' literature.

For a restrictor pressure drop of 7 bar, which applies to the flow characteristics of an available adjustable restrictor valve given in Table 11.1 at a flow of 23.3 L min^{-1} at 99 bar, the restrictor will pass a flow of:

$$Q = 23.3 \sqrt{\frac{7}{99}} = 6.2 \text{ L min}^{-1} \text{ at } 7 \text{ bar}$$

From Table 11.1 it can be seen that the restrictor setting requires approximately three turns.

Table 11.1. Restrictor valve characteristics

Turns	**Flow (L min^{-1})**
0	0
1	0.9
2	2.8
3	5.6
4	8.5
5	10.5

11.2.1 THE EFFECT OF LOAD FORCE CHANGES

If the force increases to 20 kN, the required load pressure increases to 102 bar, thus reducing the available restrictor pressure drop to 48 bar. Hence the flow will reduce accordingly.

Thus:

$$Q_s = 23.3\sqrt{\frac{48}{99}} = 16.2 \text{ L min}^{-1}$$

For this flow, the actuator velocity will be 0.14 ms^{-1}.

11.3 VALVE CONTROL OF A SINGLE-ENDED ACTUATOR

A single-ended actuator is to operate against a load that causes a pulling force on the actuator rod. For the given data determine the suitability of a proportional valve for controlling the actuator to give a retract velocity of 0.25 m s^{-1}. Also, calculate the velocity when extending and the maximum pulling force capability of the actuator.

11.3.1 DATA

Piston diameter	50 mm
Rod diameter	25 mm
Supply pressure	75 bar
Pulling force	3,050 N
Valve rated flow	40 L min^{-1}

Valve rated pressure drop/metering land 35 bar
Piston area = 1.96×10^{-3} m^2
Annulus area = 1.47×10^{-3} m^2
Area ratio $\alpha = 1.33$

11.3.2 ACTUATOR RETRACTION

For this application, with a four-way valve as shown in circuit Figure 7.15 the force is negative and from Equations 7.6 and 7.7, Section 7.5.4.1.2, from Figure 11.1, for the actuator retracting we get:

$$P_1 = P_S \alpha^2 \left(\frac{1 + \alpha R}{1 + \alpha^3} \right)$$

$$\frac{P_2}{P_s} = \frac{\alpha \left(\alpha^2 - R \right)}{\left(1 + \alpha^3 \right)}$$

Stall force $F_S = P_S A_p = 14700\,\mathrm{N}$ and so for $F = -3{,}050$ N we get $R = -0.21$. This gives:

$P_1 = 28.5$ bar, $P_2 = 59$ bar, and the valve pressure drop from the supply to the annulus during retraction is $\Delta P_{S2} = 75 - 59 = 16\,\mathrm{bar}$

$$Q_2 = 0.25 \times A_2 = 0.25 \times 1.47 \times 10^{-3}\,\mathrm{m^3\,s^{-1}} = 3.7 \times 10^{-4}\,\mathrm{m^3\,s^{-1}} = 22\,\mathrm{L\,min^{-1}}$$

Now for the valve, the rated flow at maximum opening is $Q_R = k_V \sqrt{\Delta P_R}$
For the actuator

$$\frac{Q_2}{\sqrt{\Delta P_{S2}}} = \frac{3.7 \times 10^{-4}}{\sqrt{16 \times 10^5}} = 2.93 \times 10^{-7} \frac{\mathrm{m^3/s}}{\sqrt{\mathrm{N/m^2}}}$$

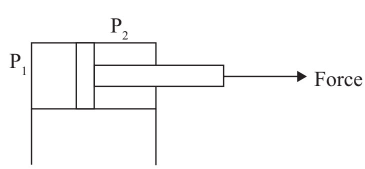

Figure 11.1. Single-ended actuator.

The valve data gives a value of:

$$\frac{Q_R}{\sqrt{\Delta P_R}} = \frac{40}{6\times10^4\times\sqrt{35\times10^5}} = 3.56\times10^{-7}\ \frac{m^3/s}{\sqrt{N/m^2}} = k_{\mathrm{V}}$$

The valve is therefore adequate to provide the required retract velocity.

11.3.3 ACTUATOR EXTENSION

From Section 7.5.4.1.2, using Equations 7.3 and 7.4, the pressures are given by:

$$P_1 = P_S\left(\frac{1+Ra^3}{1+a^3}\right)$$

$$P_2 = P_S a\left(\frac{1-R}{1+a^3}\right)$$

For $R = -0.21$, these give:

$$P_2 = 36\ \text{bar}\ \textit{and}\ P_1 = 11.3\ \text{bar}\ \therefore\ \Delta P_{s1} = 36\ \text{bar}$$

The ratio of the extend and retract velocities is given by:

$$\frac{U_{\mathrm{R}}}{U_{\mathrm{E}}} = \sqrt{\frac{(P_S-P_2)_{\text{retract}}}{(P_2)_{\text{extend}}}} = \sqrt{\frac{16}{36}} = 0.67\ \text{ and }\ U_{\mathrm{E}} = 0.38\,\text{m s}^{-1}$$

The maximum force capability during extension is that which avoids cavitation of the piston end of the actuator. This is given by:

$$R = -\frac{1}{a^3} = -\frac{1}{1{,}33^3} = -0.425$$

The maximum force, $F_{\max} = -0.425\times14700 = -6247.5\,\text{N}$

11.4 WINCH APPLICATION

11.4.1 LIFTING THE LOAD

For the winch shown in Figure 11.2

Winch drum torque $$T_D = \frac{M\,g\,r}{\eta_d}$$

Motor torque $$T_m = \frac{T_d}{R\,\eta_R} = \frac{M\,g\,r}{R\,\eta_R\,\eta_d}$$

Motor pressure $$P = \frac{T_m}{D\,\eta_m} = \frac{M\,g\,r}{R\,D\,\eta_m\,\eta_R\,\eta_d}$$

η_m = motor mechanical efficiency
η_R = reduction gearbox mechanical efficiency
η_D = winch drum mechanical efficiency

11.4.2 LOWERING THE LOAD

$$T_D = M\,g\,r\,\eta_D \qquad T_m = \frac{T_d\,\eta_R}{R} = \frac{M\,g\,r\,\eta_R\,\eta_D}{R}$$

$$P = \frac{T_m\,\eta_m}{D} = \frac{M\,g\,r\,\eta_m\,\eta_R\,\eta_D}{R\,D}$$

$$\therefore\ \frac{P_{up}}{P_{down}} = \frac{1}{(\eta_m\,\eta_R\,\eta_D)^2} = \frac{1}{\eta_T^{\,2}}$$

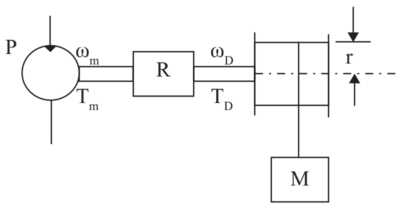

Figure 11.2. Winch driven by motor and reduction gearbox.

11.4.3 NUMERICAL VALUES

Data

$r = 0.25$ m
$M = 2.5$ tonnes
$D = 574 \text{ cm}^3 \text{ rev}^{-1} = 9.1 \times 10^{-5} \text{ m}^3 \text{ rad}^{-1}$
$R = 5{:}1$ (reduction)
$\eta_m = 0.78$ (starting), 0.92 (running)
$\eta_R = \eta_d = 0.94$

Ideal pressure

$$P = \frac{M\,g\,r}{R\,D} = \frac{2500 \times 9.81 \times 0.25}{5 \times 9.1 \times 10^{-5}} = 133 \text{ bar}$$

Therefore, during start-up from rest:

$\eta_T = 0.69$
$P_{up} = 193$ bar and $P_{down} = 92$ bar

And when operating at speed:

$\eta_T = 0.81$
$P_{up} = 164$ bar and $P_{down} = 108$ bar

11.5 HYDRAULIC MOTOR FOR DRIVING A WINCH

A winch is to be driven by a hydraulic motor, and in order to provide a wide range of operating speeds at the maximum supply flow, the motor displacement can be set at either a maximum or minimum value. These displacement values (D_{max} and D_{min}) are pre-set by mechanical stops in the

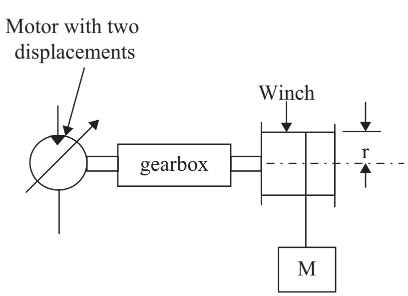

Figure 11.3. Hydraulic winch with gearbox.

motor, but their level can be chosen from within the range shown in the data. The winch operator makes the selection of the two displacements by changing the position of a control valve in the hydraulic circuit.

The mechanical transmission has a speed-reducing gearbox, and to design the system, it is required to select an appropriate motor from the two sizes given in the data and determine the reduction ratio of the gearbox that will provide the specified performance.

For the gear box driven winch shown in Figure 11.3

1. Calculate the gearbox ratio that is required for each motor to give the maximum winch torque when the motor is operating at a selected pressure.
2. Determine the motor speeds from (1) that are required to give the maximum cable speed of 50 m min^{-1} and select a suitable motor from the data.
3. Calculate the flow that is required to drive the selected motor when in maximum displacement at the speed that produces a cable speed of 15 m min^{-1}.
4. Using the flow from (3) determine the required minimum motor displacement that will provide the maximum cable speed of 50 m min^{-1}.
5. For this flow, calculate the motor speed for an oil viscosity of 10 cSt. This needs to account for the effect of oil viscosity on the volumetric efficiency.

Motors (maximum pressure for continuous operation = 300 bar**).**
Using the data from Table 11.2 gives:

Winch drum diameter	0.5 m
Gearbox mechanical efficiency (η_G)	92%
Maximum winch load	200 kN
Maximum cable speed at maximum load	15 m min^{-1}
Maximum cable speed	50 m min^{-1}
Motor mechanical efficiency at (a) D_{max}, (b) $D < D_{max}$ (η_H)	95%, 90%

Table 11.2. Motor data

Motor displacement (D_{max}/D_{min}), cm^3 rev^{-1}	55/11	80/17
Maximum speed at D_{max}, rev min^{-1}	4,200	3,750
Maximum speed for $D < D_{max}$, rev min^{-1}	6,300	5,600

Motor volumetric efficiency for
$D_{\max}$ (35 cSt oil viscosity) (η_{VH}) 95%
Motor volumetric efficiency for $D < D_{\max}$
(35 cSt oil viscosity) (η_{VL}) 90%

11.5.1 GEARBOX RATIO

Choose the maximum operating pressure of 300 bar.

Winch drum torque =
load force × drum radius = $200 \times 10^3 \times \dfrac{0.5}{2} = 50 \times 10^3\,Nm$
Motor torque at 300 bar = $PD_{\max}\,\eta_H \eta_G$

$$= D_{\max} \times \frac{10^{-6}}{2\pi} \times 300 \times 10^5 \times \eta_H(0.95) \times \eta_G(0.92) = 4.17 D_{\max}\,Nm$$

Required reduction gearbox ratio (n) $\dfrac{winch\ torque}{motor\ torque} = \dfrac{T_D}{T_m} = \dfrac{50 \times 10^3}{4.17 D_{\max}}$

This gives values of n required for the two types of motors of 218 and 150.

11.5.2 MOTOR SELECTION

Cable speed $\pi dN = \dfrac{\pi \times 0.5 N_m}{n} = U_{\max} = 50 m/\min$ (N_m = motor speed)

Motor speed $N_m = \dfrac{50n}{0.5\pi}$, which gives motor speeds of 6,939 and 4,775 rev $\min^{-1}$, respectively.

Select the larger motor, as its speed is less than the maximum allowable value given in the data.

11.5.3 FLOW REQUIRED

Motor speed for a cable speed of 15 m $\min^{-1}$ $\dfrac{15}{0.5\pi} \times 150 = 1432\ rev/\min$

The flow, Q, required for this speed

$$= \frac{N_m D_{\max}}{\eta_{VH}} = \frac{1432 \times 80 \times 10^{-3}}{0.95} = 121 L/\min$$

11.5.4 MINIMUM MOTOR DISPLACEMENT

For the flow given in Section 11.5.3, the minimum motor displacement required to operate the winch at a cable speed of 50 m $\min^{-1}$ is given by:

$$D_{\min} = \frac{Q}{N_m} \times \eta_{VL} = \frac{121 \times 10^3 \times 0.9}{4775} = 23\ cm^3 / rev$$

11.5.5 MAXIMUM MOTOR SPEED WITH OIL HAVING A VISCOSITY OF 10 CST

At minimum displacement, the volumetric efficiency of 90 percent was quoted for an oil viscosity of 35 cSt, which represents a leakage flow of 10 percent. This leakage will increase as the oil viscosity reduces so that here it will be $= 10\% \times \frac{35}{10} = 35\%$ and the volumetric efficiency will therefore be 65 percent.

11.6 HYDRAULIC SYSTEM FOR GANTRY CRANE

A hydraulically powered gantry crane, shown in Figure.11.4, is to lift loads up to 300 tonnes. It is required to design the hydraulic systems for driving both the winch and the crane wheels.

Details of the installation are given below:

a. Crane Winch (two Drums)

- Total load = 300 tonnes

Two pulley blocks each having four pulleys to provide eight lengths of cable for each of the two lifting hooks.

- Maximum lift = 10 m
- Winch drum diameter = 600 mm
- Lift speed (variable) (max) = 0.02 ms^{-1}

Each of the two winch drums is to be powered separately, and they are required to be held stationary in any position.

b. Wheel Drive

- The vehicle has eight wheels that are grouped together into units having two wheels each. In these units, the two wheels are connected by a chain drive, with one of the wheels being driven by

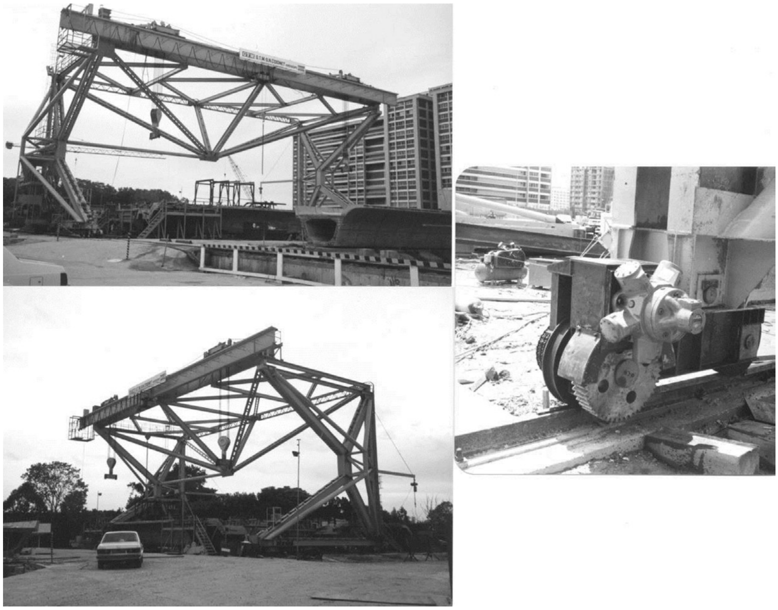

Figure 11.4. Gantry crane.

a hydraulic motor (i.e., four motors in total). There is no requirement for brakes on these wheels.

- Supply pipe lengths are 30 m on one side and 80 m on the other.
- As far as possible the torque on each side must be within 10 percent of each other.
- Braking will be carried out by reducing the pump supply and means must be provided to prevent both cavitation and excessive pressures.
- Gear reduction ratio between wheels and motor = 2.5:1
- Wheel drive system efficiency = 80%
- Maximum wheel torque at low speed = 4,250 Nm
- Maximum wheel torque at high speed = 2,120 Nm
- Wheel diameter = 0.46 m
- Maximum speed = 30 m min^{-1}
- Minimum speed = 1.2 m min^{-1}
- Maximum pressure = 250 bar

11.6.1 GANTRY CRANE

Load = 300 tonne = 3,000 kN.

Eight cables (falls) and two pulley blocks.

Table 11.3. Cable information

Cable diameter (mm)	Minimum breaking force (kN)
35	785
40	1,000
48	1,460

Cable tension $= \dfrac{3000}{2\times 8} = 187.5\ kN$

Allow 10 percent for friction 206.3kN

Choose cable of 40 mm diameter from Table 11.3

For one cable layer, the winch drum torque

$$= \left(\frac{0.6+0.04}{2}\right)\times 206.3\times 10^3 = 66,000\ Nm$$

The winches can be driven by either low or high speed motors and the reduction gearbox ratio needs to be selected accordingly.

a. Low speed motor

Choose 10:1 reduction gearbox and allow 8 percent for mechanical losses

$$\text{The motor torque } = \frac{66,000}{10}\times\frac{1}{0.92} = 7,176\ Nm$$

For 80 percent motor efficiency at 250 bar, the motor displacement is given by:

$$Motor\ torque\ T = PD\eta_m$$

$$\therefore D = \frac{7,176}{250\times 10^5\times 0.8} = 3.6\times 10^{-4}\, m^3/rad$$

$$= 2254\ (cm^3/rev)\ or\ 2.254(L/rev)$$

There are four pulleys and eight lengths of cable on each side which give a maximum cable speed of: $0.02\times 8 = 0.16\ m/s$

$\therefore$The winch drum speed $= \dfrac{0.16\times 60}{\pi\times 0.64} = 4.78\ rev/\min$ and the motor speed will be 47.8 rev min^{-1}.

b. High-speed motor

For a high-speed motor, a gearbox ratio of 100:1 can be used which, using the same gearbox efficiency as in (a), will require a motor having a displacement of 225 $cm^3\ rev^{-1}$ operating at a speed of 478 $rev\ min^{-1}$.

c. Motor flow

For both the high- and low-speed motor drives, the motor flows will be the same. Thus, assuming a motor volumetric efficiency of 90 percent, the flow required for each winch is:

$$\frac{speed \times displacement}{volumetric\ efficiency} = \frac{478 \times 0.225}{0.9} = 119.5\, L/\min$$

The required maximum pump displacement for supplying both winches, assuming a volumetric efficiency of 95 percent and a drive speed of 1,500 $rev\ min^{-1}$, will be:

$$\frac{119.5 \times 10^3}{1500 \times 0.95} \times 2 = 167.7\ cm^3/rev$$

For a mechanical efficiency of 95 percent for the pump, the required input power is:

$$= \frac{2 \times 119.5 \times 250 \times 10^5}{60 \times 10^3 \times 10^3 \times 0.95} = 104.8\ kw$$

The total output power from the winches is:

$$3 \times 10^6 \times 0.02 \times 10^{-3}\, kw = 60\ kw$$

Appropriate winch motors and supply pump can be selected from manufacturers' literature, and the assumed values of the mechanical and volumetric efficiencies used in the analysis can be checked.

11.6.2 WHEEL DRIVE

For a torque at each wheel of 4,250 Nm, two wheels require 8,500 Nm torque. Thus for a gearbox reduction ratio of 2.5:1, the required motor torque is given by:

$$\frac{8500}{2.5 \times 0.8} = 4250Nm$$ (80 percent wheel drive mechanical efficiency).

At a vehicle speed of 1.2 m min^{-1} (U) with 0.46 m wheel diameter, $\pi DN = U$.

Hence

$$N = \frac{U}{\pi D} = \frac{1.2}{\pi \times 0.46} = 0.83\ rev/\min$$

As for the winch drive, low- or high-speed motors can be selected for the wheel drive. By way of example, a dual displacement low-speed motor is considered here, which gives the performance outlined in Table 11.4.

The maximum total nominal flow (two motors each side) = $2 \times 38.3 \times 2 = 153.2\, L/\min$. Assuming that the volumetric efficiency of the motors is 90 percent and 95 percent for the pump, the maximum pump flow is:

$$\frac{153.2}{0.9 \times 0.95} = 179 L/\min$$

For 1,500 rev min^{-1} pump speed, the required pump displacement is:

$$D_p = \frac{179 \times 10^3}{1500} = 119.5\ cm^3/rev$$

As for the winch drive, the pump and motors having displacements that are closest to those required can be selected from manufacturers' literature.

Table 11.4. Motor performance

U (m min^{-1})	1.2	30.0
Wheel speed (rev min^{-1})	0.83	20.75
Motor speed (rev min^{-1})	2.08	51.9
Motor D_m (cm^3 rev^{-1})	1,476	738
Nominal flow (L min^{-1})	3.06	38.3
Motor torque (Nm)	4,250	2,120
Pressure (bar)	215	214

11.6.3 PIPE SIZES

For motor flows of 80 L min^{-1} to each side.

The pressure loss $\Delta_p = 4f\frac{L}{d}\frac{\rho}{2}U^2$

f—Friction factor from Moody diagram (Chapter 8).

For the pipe, the value of Reynolds No. $R_e = \frac{Ud}{\upsilon}$

Taking a value of fluid viscosity of 40 cSt ($4\times10^{-5}\,m^2/s$) and length L of the longest side of 80 m gives:

$$Q = \frac{\pi d^2}{4}U \quad \therefore \Delta p = 8\times(4f)\frac{L}{d^5}\frac{\rho}{\pi^2}Q^2$$

The effect of the pipe diameter on the pressure loss can be seen from Table 11.5.

A circuit that will provide the necessary functions is shown in Figure 11.5. This contains the following major functions:

11.6.4 CRANE WINCH

- Pilot operated check valve to protect from hose failures.
- Cross line relief valves.
- Brake control by:
  - o Selection of the directional control valve (DCV).
  - o High pressure in the circuit.
- Counterbalance valves for lowering the load.

11.6.5 WHEEL DRIVE

- Cross line relief valves.
- Brake valves.

Table 11.5. Pipe pressure loss

d **(mm)**	**Re**	***4f***	Δ_p**(bar)**	***U*** **(m s^{-1})**
20	2,122	0.03	9.4	4.2
25	1,700	0.038	3.9	2.7
30	1,415	0.043	1.8	1.9

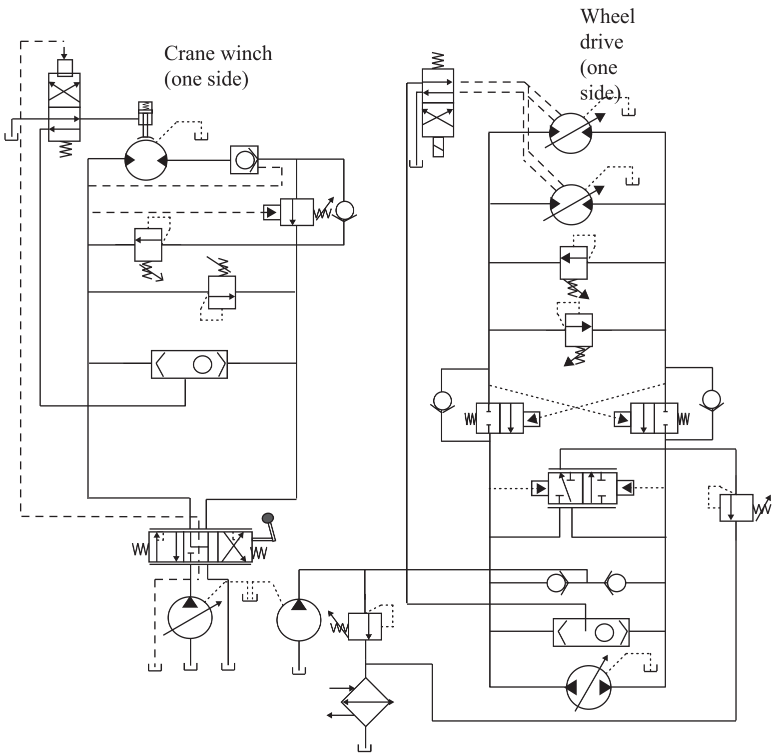

Figure 11.5. Circuit diagram.

- Purge valve to extract flow for cooling from the low pressure side of the circuit (drain flows may also be passed to the cooler inlet).
- Motor displacement selection valve.
- Boost pump to make up drain flows from the pump and motors.

11.7 PRESSURE LOSSES

The evaluation of the pressure losses in pipes was described in Chapter 8, and a typical problem area for the application of this analysis is the determination of appropriate pipe sizes for the inlet, or suction, flow to pumps directly from a reservoir. Frequently there are valves and pipe fittings in such circuits and standard charts can be used to give the dynamic head losses that will arise. This example is concerned with the evaluation of a typical pump application.

The connections to a pump inlet from the supply tank consist of the following components:

- A pipe, descending vertically having a length of 1 m and an internal diameter of 25 mm.
- A gate valve, which is fully opened.
- A standard sweep elbow having an internal diameter of 25 mm.
- A horizontal pipe having a length of 3 m and an internal diameter of 25 mm.

The supply tank is not pressurized.

The pump is required to operate with a fluid temperature range from 20°C to 60°C. Using the given data and the graphs in Chapter 8 calculate the pressure at the pump inlet for fluid temperatures of 20°C and 60°C. Pipe inlet effects can be ignored.

The pump requires an inlet pressure of at least 0.9 bar absolute (−0.1 bar gauge) to operate satisfactorily. Show how the system can be modified to achieve this.

11.7.1 DATA

Pump displacement	48 cm^3 rev^{-1}
Pump speed	1,500 rev min^{-1}

11.7.2 PRESSURE LOSS AT 20°C

The pump flow $Q = 48\times10^{-6}\times\dfrac{1500}{60} = 1.2\times10^{-3}m^3/s$

Pipe area (d = 25 mm) $A = \dfrac{\pi\times25^2\times10^{-6}}{4} = 4.91\times10^{-4}m^2$

The mean velocity in the pipe $u = \dfrac{1.2\times10^{-3}}{4.91\times10^{-4}} = 2.44m/s$

At 20°C, the fluid viscosity from Figure 8.1, Chapter 8, is 85 cSt.

The Reynolds No. $R_e = \dfrac{ud}{\upsilon} = \dfrac{2.44\times0.025}{85\times10^{-6}} = 717$. Therefore, the flow is laminar for which the friction coefficient $f = \dfrac{16}{R_e} = 0.022$.

From the chart in Figure 11.6, the fluid resistance of fittings is given as an equivalent length of pipe for which the pressure loss is calculated in the normal manner for a pipe. For the circuit details given in the data, we get the following equivalent pipe lengths for the various fittings from Figure 11.6:

- Five pipe diameters for the fully opened gate valve.
- 32 pipe diameters for the standard sweep elbow.

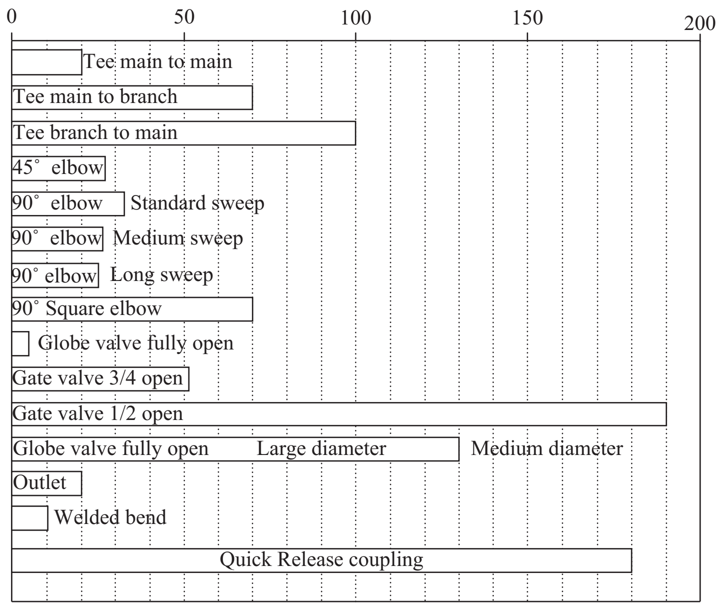

Figure 11.6. Fluid resistance in fittings.

The equivalent length, L, of the fittings is therefore $(5 + 32)d$, giving an $\frac{L}{d}$ ratio of 37 which is added to that for the pipe system.

The total pressure loss:

$$= 4f\frac{L}{d}\frac{\rho u^2}{2} = 4\times 0.022\times\left(\frac{4}{0.025}+37\right)\frac{875\times 2.44^2}{2}$$

$$= 44900\,pa \ \ (= 5.1\times 4f\ bar)$$

The pressure increase from the vertical height of the supply tank = $\rho gh =$ $870\times 9.81\times 1 = 8580\,pa$

The pump inlet pressure = (8,580 – 44,900) pa = −0.36 bar (gauge).

11.7.3 PRESSURE LOSS AT 60°C

At a fluid temperature of 60°C, the viscosity is 15 cSt for which the Reynolds No. is $717\times\frac{85}{15} = 4100$. Therefore, the flow is turbulent, for which, value of the friction factor f obtained from the graph in Figure 8.2, Chapter 8, is approximately 0.01. For this condition, the friction pressure loss is:

$$5.1 \times 0.04 = 0.204\,bar$$

The inlet pressure will therefore be: $-0.204 + 0.086 = -0.118$ bar (gauge).

11.7.4 PUMP REQUIREMENTS

Both of the calculated inlet pressures are lower than those recommended, and in order to reduce the pressure loss, the pipe diameter can be increased. Increasing the pipe diameter to 35 mm will reduce the pressure loss for the 20°C condition to the value given as follows:

New velocity $= 2.44 \times \left(\frac{25}{35}\right)^2 = 1.24 m/s$

This gives a new Reynolds Number $R_e = \frac{ud}{\upsilon} = \frac{1.24 \times 0.035}{85 \times 10^{-6}} = 510$

$\therefore f = \frac{16}{510} = 0.031$ for which the pressure loss will be:

$$4f\frac{L}{d}\frac{\rho u^2}{2} = 4 \times 0.031 \times \left(\frac{4}{0.035} + 37\right)\frac{875 \times 1.24^2}{2} = 12620\,pa$$

The inlet pressure will now be 0.086 – 0.126 = −0.04 bar (gauge), which is more than the minimum allowable and therefore satisfactory.

11.8 SINGLE STAGE RELIEF VALVE

For a single stage relief valve, having the given data, calculate the variation of the flow through the valve and the valve opening with pressure. The spring preload is set to give a cracking pressure of 200 bar.

11.8.1 DATA

Valve seat diameter d	12.5 mm
Valve poppet seat angle ϕ	45°
Spring stiffness k	100 N mm^{-1}
Valve flow coefficient C_Q	0.8

Table 11.6. Valve flow characteristics

***P* (bar)**	***Q* ($m^3\ s^{-1}$)**	***Q* ($L\ min^{-1}$)**	***X* (mm)**
210	7.9×10^{-4}	47	0.17
220	1.55×10^{-3}	93	0.31
250	3.7×10^{-3}	222	0.70

For a direct acting, poppet type of relief valve, the equation for the variation in flow with pressure including the effect of the flow force is, from Chapter 8, Equation 8.11:

$$Q = \frac{P - C/A}{K_1/\sqrt{P} + K_2\sqrt{P}}$$

For the data, the values of the constants are:

$$K_1 = \frac{k}{A\pi dC_Q \sin\phi\sqrt{\frac{2}{\rho}}} = 7.7\times10^{11}$$

$$K_2 = \frac{\cos\phi}{A}\sqrt{2\rho} = 2.4\times10^5$$

$$X = \frac{Q}{\pi dC_Q\sqrt{\frac{2P}{\rho}}\sin\phi}$$

$$\phi = 45^0$$

Results for this analysis are given in Table 11.6

11.9 SIMPLE ACTUATOR CUSHION

Using the analysis given in Chapter 3.4.2, the performance of an actuator cushion using a simple orifice or adjustable restrictor valve can be determined for the actuator application having the given data. The peak actuator pressure needs to be limited to 300 bar that occurs at the commencement of cushioning.

11.9.1 DATA

Actuator piston diameter	140 mm
Cushion plug diameter	43 mm
Mass	40,000 kg
Initial velocity	0.3 ms^{-1}

From Chapter 3.4.2, the velocity variation is given by:

$$U = U_m \exp\left(-\frac{CX}{m}\right)$$

$$\text{where}\, C = \frac{\rho A_C^3}{2C_D^2 A_R^2}$$

For a maximum pressure of 300 bar, the restrictor size is given by:

$$A_R = \frac{Q_m}{C_D}\sqrt{\frac{\rho}{2P_C}}$$

$$A_C = \frac{\pi}{4}(d_P^2 - d_C^2) = 13.9\times10^{-3}\, m^2$$

$$Q_m = A_C U_m = 0.3\times13.9\times10^{-3} = 4.2\times10^{-3}\, m^3 s^{-1}\ \ (251\, L/\min)$$

and, taking a value for C_D of 0.65 gives:

$$A_R = \frac{4.2\times10^{-3}}{0.65}\sqrt{\frac{870}{2\times3\times10^7}} = 24.6\times10^{-6}\, m^2\ (5.6\ mm\ diam.\ hole)$$

Now:

$$U = U_m \exp(-\frac{CX}{m})$$

$$\text{where}\ \ C = \frac{\rho A_C^3}{2C_D^2 A_R^2} = 4.6\times10^6$$

$$\frac{m}{C} = \frac{40000}{4.6\times10^6}\times10^3\ \ mm = 8.6\ mm$$

$$\therefore U = 0.3\exp\left[-\frac{x}{8.6}\right] m/s$$

The velocity will reduce to 37 percent of its initial value (0.11 ms^{-1}) in 8.6 mm.

11.10 CENTRAL BYPASS VALVE

Figure 11.7 shows a weight-loaded actuator that is operated by a central bypass type of open center valve.

The flow characteristics for central bypass valves are usually given in manufacturers' literature. However, an analysis of these valves provides a good example of the use of variable restrictions for the control of flow and how the valve characteristics are obtained.

Figure 11.8 shows the construction of a complete central bypass valve and the basic valve dimensions, where the distance L refers to the initial opening of the bypass and the distance H refers to the overlap of the metering to the outlet port.

The system circuit can be reduced to that shown in Figure 11.9 where the variable restrictions are the bypass and outlet port metering openings that are altered by movement of the valve spool, that of the bypass reducing as the spool is progressively moved from the central position.

As the valve is moved through distance H, the pump pressure is increased because of the reduced bypass opening. For movements greater than H, flow can pass to the outlet port provided, as discussed in Chapter 5, the pump pressure can open the load check valve against the pressure required to operate the actuator.

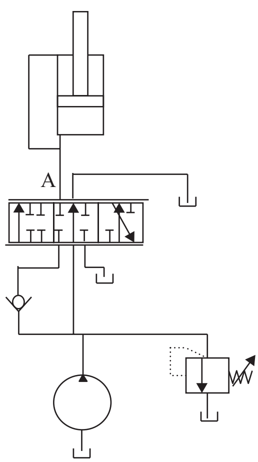

Figure 11.7. Weight loaded system.

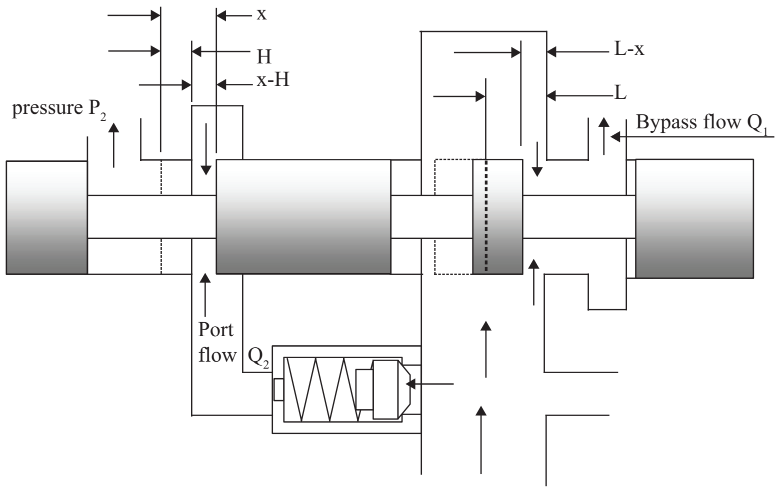

Figure 11.8. Valve dimensions.

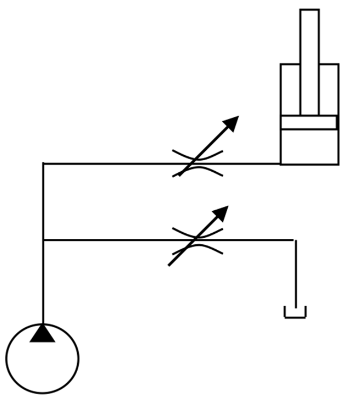

Figure 11.9. Equivalent circuit for actuator extension.

11.10.1 FLOW ANALYSIS

Assuming that the outlet pressure from the central bypass is small (i.e., zero pressure losses to the tank) we get:

Bypass flow
$$Q_1 = K(L - x)\sqrt{P_S} \tag{11.1}$$

Port flow
$$Q_2 = K(x - H)\sqrt{(P_S - P_2)} \tag{11.2}$$

Pump flow $$Q_P = Q_1 + Q_2 \tag{11.3}$$

Note that for $x < H$, Q_2 is zero because the valve is positioned in the overlap region. The flow constant $K = C_Q \pi d \sqrt{\frac{2}{\rho}}$

We can obtain solutions to these equations for a given valve position and load pressure, P_2, which would give the pump pressure and hence the values of Q_1 and Q_2. However, carrying out this process is complicated because of the square root of pressure. An alternative method can be applied that uses two known extreme conditions to establish the range of values of Q_2 for any given valve opening, x.

11.10.1.1 Condition 1

Condition 1 establishes the pump pressure that is created when all the pump flow is passed through the bypass. Thus from Equation 11.1 we have:

$$P_S = \left[\frac{Q_1}{K(L-x)}\right]^2 = \left[\frac{Q_P}{K(L-x)}\right]^2 \tag{11.4}$$

Equation 11.4 determines the maximum pump pressure that is available for a given value of x, which establishes the valve movement that is required in order to obtain flow at the valve outlet for a given load pressure, P_2. On further movement of the valve beyond this point, flow will pass to the outlet.

11.10.1.2 Condition 2

The maximum available outlet flow, Q_{2max}, will occur with zero load pressure, and this condition can be obtained from Equations 11.1 to 11.3 with $P_2 = 0$.

Thus, Equations 11.1 and 11.2 give:

$$P_S = \left[\frac{Q_{2\max}}{K(x-H)}\right]^2 = \left[\frac{Q_1}{K(L-x)}\right]^2$$

Or: $Q_{2\max} = \left(\frac{x-H}{L-x}\right) Q_1$ which into Equation 11.3 gives:

$$Q_{2\max} = \left(\frac{x-H}{L-H}\right)Q_P \tag{11.5}$$

Equation 11.5 shows that the maximum possible outlet flow varies linearly with x, for zero outlet pressure, when $x > H$ (i.e., the valve has moved out of the overlap region) reaching its maximum value (pump flow) when the central bypass is fully closed (i.e., $x = L$).

11.10.1.3 Practical Example

For the system shown in Figure 11.5, the valve and actuator dimensional data are given as follows:

11.10.1.3.1 Data

Pump flow	= 400 L min^{-1}
Valve spool diameter	= 12 mm
Central bypass opening for valve in central position	= 6 mm
Valve overlap at port A	= 2 mm
Actuator rod diameter	= 70 mm
Actuator piston diameter	= 100 mm

11.10.2 VALVE CHARACTERISTICS

Consider the extension of the actuator for lifting the weight load. We can apply conditions 11.1 and 11.2 in order to establish the operating flow range of the valve at different positions.

Thus we have:

$$K = C_Q \pi d \sqrt{\frac{2}{\rho}} = 0.65 \times \pi \times 12 \times 10^{-3} \sqrt{\frac{2}{870}} = 1.18 \times 10^{-3}\, m^3 / s / (N / m^2) \tag{11.6}$$

For condition 2 from Equation 11.5 it is seen that, for zero load pressure, the outlet flow, Q_2, varies linearly with valve movements from $x = 2$ mm to $x = 6$ mm.

For condition 1, the variation of pump pressure for zero outlet flow is obtained from Equation 11.4. Thus:

$$P_S = \left(\frac{Q_P}{K(L-x)}\right)^2 = \left(\frac{400}{1.18\times10^{-3}\times6\times10^4\times(6-x)\times10^{-3}}\right)^2\times10^{-5}\,bar$$

$$= \frac{319}{(6-x)^2}\,bar \qquad (11.7)$$

Values from Equation 11.7 are shown in Table 11.7 where it can be seen that the pump pressure increases rapidly as x nears the closing position of the bypass. If the valve is moved rapidly to the full open position (bypass fully closed), excessive transient pump pressures could occur. In order to avoid this and when there is a blocked outlet port or an excessive actuator load force, a relief valve needs to be fitted as shown in Chapter 5, Figure 5.6.

11.10.3 ACTUATOR FORCE

The actuator has a piston diameter of 100 mm and a rod diameter of 70 mm and for the connections shown in Figure 11.5 (regenerative), the effective area is that of the rod. This is because the supply pressure acts on both the piston and annulus areas, and the net flow into the actuator is due to the volume displaced by the rod area only.

The rod area $= \frac{\pi}{4}\times(100^2-70^2)\times10^{-6} = 4\times10^{-3}\,m^2$

The actuator weight load capacity can be determined at a given valve position from Table 11.7, the value chosen being dependent on the maximum required velocity.

Thus for a load pressure of 35.4 bar (corresponding to the maximum possible at x = 3 mm), the weight load that can be lifted is:

$$F = 4\times10^{-3}\times35.4\times10^5 = 14160\,N$$

Table 11.7. Valve data

Valve position x (mm)	**Maximum flow for $P_2 = 0$ Q_{2max} (L min^{-1})**	**Pump pressure for $Q_1 = Q_P$ P_P (bar)**
0	0	8.9
1	0	12.8
2	0	19.9
3	100	35.4
4	200	80.0
5	300	319.0
6	400	∞

11.10.4 VALVE OPERATION

The valve will need to be opened 3 mm in order to generate the necessary pump pressure of 35.4 bar. The flow that is available at valve positions greater than 3 mm can be determined by the following method.

$x = 4$ mm

$L - H = 4$ mm and $x - H = 2$ mm.

Thus, from Equation 11.5, $Q_{2max} = 200$ L min^{-1} (valve at mid open position).

From Table 11.7, the maximum pump pressure for $x = 4$ mm is 80 bar and the pressure for other bypass flows can be obtained from Equation 11.7 thus:

$$P_S = 80 \times \left(\frac{Q_1}{400}\right)^2 bar \tag{11.8}$$

And from Equation 11.2,

$$P_S - P_2 = \left(\frac{Q_2}{K(x - H)}\right)^2 \tag{11.9}$$

Thus by selecting values of Q_2 between 0 and 200 L min^{-1}, values of P_S and P_2 can be obtained from Equations 11.8 and 11.9, and these are shown in Table 11.8.

For $x = 5$ mm

$Q_{2max} = 300$ L min^{-1}

Applying the same method as for $x = 4$ mm gives the values shown in Table 11.9

From Figure 11.10 it is seen that for the weight load of 14,160 N, the change of flow with valve position is reasonably linear for the 3 mm of valve movement required for zero to maximum flow. If a heavier weight load is lifted, say that for 80 bar pressure, the valve will have to be opened 4 mm before the start of actuator movement.

Table 11.8. Valve flow/pressure characteristic, $x = 4$ mm

Q_2 (L min^{-1})	Q_1 (L min^{-1})	P_S (bar)	$P_S - P_2$ (bar)	P_2 (bar)
200	200	20	20	0
150	250	31.3	11.3	20
100	300	45	5	40
50	350	61.3	1.3	60
0	400	80	0	80

Table 11.9. Valve flow/pressure characteristic, $x = 5$ mm

Q_2 (L min^{-1})	Q_1 (L min^{-1})	P_S (bar)	$P_S - P_2$ (bar)	P_2 (bar)
300	100	20	20	0
200	200	80	8.9	71.1
100	300	180	2.2	177.8
0	400	319	0	319

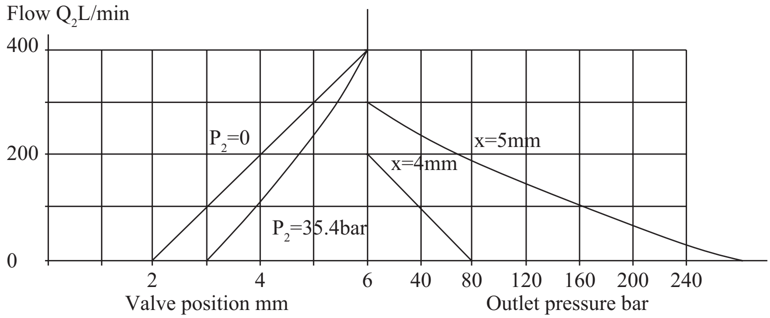

Figure 11.10. Valve performance characteristics.

For lowering the load, the control of flow is made by metering the outlet flow to the return, the valve spool being arranged so that pump flow is bypassed to the return for this position.

11.11 PUMP AND MOTOR EFFICIENCIES

a. The pistons of an oil-hydraulic pump are 20 mm in diameter and 25 mm in effective length. At the rated performance, the mean piston velocity is 1 ms^{-1} and the pump discharge pressure is 200 bar. Estimate the radial clearance between the piston and the cylinder bore that will give optimum overall efficiency. For this application, the fluid kinematic viscosity, v, is 35 cSt and the fluid density is 870 kg m^{-3}. The dynamic viscosity $\mu = v \times \rho = 35 \times 10^{-6} \times 870 = 0.03 Nsm^{-2}$.

For a leakage flow of Q_S with a pump pressure of P, the leakage power loss is given by: $Q_S P = \left(\dfrac{h^3}{12\mu} \times \dfrac{P}{x} \times 2\pi R \right) \times \dfrac{P}{2}$ (piston pressurized for half the time).

Thus inserting values:

$$\frac{h^3}{12 \times 0.03} \times \frac{2 \times 10^7}{25 \times 10^{-3}} \times 2\pi \times \frac{20}{2} \times 10^{-3} \times \frac{2 \times 10^7}{2} = 1.4 \times 10^{15} h^3 W.$$

The frictional power loss is caused by the relative velocity between the piston and the cylinder in the clearance space acting on the surface areas. This creates a shear stress in the fluid that acts over the surface area.

The viscous shear stress, τ, in the fluid is given by:

$$\tau = \mu \frac{\partial u}{\partial y} \qquad \text{Couette flow.}$$

Thus $\tau = \mu \frac{U}{h}$ and the power loss is given by:

$$\tau\, AU = \left(\mu \frac{U}{h}\right)(2\pi RL)\, U$$

$$= \frac{1}{h} \times 0.03 \times 1 \times 2\pi \times \frac{20 \times 10^{-3}}{2} \times 25 \times 10^{-3} = 4.71 \times 10^{-5} \times \frac{1}{h}\, W$$

$$(for\ \ U = 1 m/s)$$

Adding the viscous and leakage power losses and differentiating with respect to h will give a minimum when this is zero. Thus we have:

$$3 \times 1.4 \times 10^{15} h^2 - 4.71 \times 10^{-5} \times \frac{1}{h^2} = 0$$

$$h^4 = 1.12 \times 10^{-20} \qquad h = 10.3 \mu m$$

b. A pump, which runs at 1,450 rev min^{-1}, operating with an oil kinematic viscosity of 35 cSt has the following loss coefficients:

Slip coefficient $C_s = 1.2 \times 10^{-8}$
Coulomb friction coefficient $C_f = 0.03$
Viscous friction coefficient $C_v = 1.31 \times 10^5$

Estimate the maximum overall efficiency of the pump and the corresponding discharge pressure.

From Section 9.5.3, the maximum efficiency is obtained approximately when:

$$C_S \left(\frac{P}{\mu\omega}\right) = C_V \left(\frac{\mu\omega}{P}\right)$$

$$P^2 = \frac{C_V}{C_S}(\mu\omega)^2 = \frac{1.3 \times 10^5}{1.2 \times 10^{-8}} \left[\frac{0.03 \times 1450 \times 2\pi}{60}\right]^2$$

$$P^2 = 2.25\times10^{14} \qquad P = 150\ bar$$

Slip loss = viscous friction loss = $C_S\left(\dfrac{P}{\mu\omega}\right)$ = 0.04.

From Equations 9.10 and 9.11 the overall efficiency =

$$\frac{1-0.04}{1+0.03+0.04} = 0.9$$

11.12 CONTROL SYSTEM DESIGN

From Equation 10.36, a valve/actuator system has an open-loop system frequency response relating the actuator position to the valve spool position as follows:

$$R = \left|\frac{y}{x}\right| = \frac{k}{\omega\sqrt{\left[1-\left(\dfrac{\omega}{\omega_n}\right)^2\right]^2+\left[\dfrac{2\zeta\omega}{\omega_n}\right]^2}}$$

11.12.1 DATA

Actuator

Supply pressure	210 bar
Actuator: piston area	10^{-3} m^2
Total volume (V_T) of trapped oil	0.3 L

For the actuator piston in the center, the volume (V) each side is 0.15 L

Fluid bulk modulus	1.4×10^9 N m^{-2}

Valve

Flow coefficient C_x	0.96 m^2 s^{-1}
Flow coefficient C_p	$7.42\times10^{-12}(m^3/s)/(N/m^2)$

Load

Mass	250 kg
Viscous friction	16 kN (m s^{-1})$^{-1}$

Natural frequency

$$\omega_n = \left[\frac{2A^2B}{MV}\right]^{1/2} \text{ or } \left[\frac{4A^2B}{MV_T}\right]^{1/2}$$

$$\omega_n = \left(\frac{2\times(10^{-3})^2\times1.4\times10^9}{250\times0.15\times10^{-3}}\right)^{0.5} = 273\, rad/s$$

Damping factor

$$\frac{2\zeta}{\omega_n} = \frac{C_f V}{2\beta A^2} + \frac{MC_P}{2A^2}$$

$$\zeta = \left(\frac{16\times10^3\times0.15\times10^{-3}}{1.4\times10^9} + (7.42\times10^{-12}\times250)\right)\left(\frac{273}{2\times2\times10^{-6}}\right) = 0.243$$

11.12.2 SYSTEM GAIN

The gain, *k*, is $= \frac{C_x}{A} = \frac{0.96}{(1\times10^{-2})} = 960\, s^{-1}$

Hence, the amplitude ratio and phase angle can be found by substituting different values of ω into the following equations:

$$R = \frac{960}{\omega\sqrt{\left[1-\left(\frac{\omega}{273}\right)^2\right]^2 + \left[\frac{0.486\,\omega}{273}\right]^2}} \quad \text{and} \quad \varphi = -90 - \tan^{-1}\left[\frac{\frac{0.486\,\omega}{273}}{1-\left(\frac{\omega}{273}\right)^2}\right]$$

The resulting Bode plot in Figure 11.11 shows that the system would be unstable if it were operated in a closed-loop. Therefore, the system must be stabilized by reducing the gain, *k*. (In practice, this could be achieved by reducing the valve flow gain.)

Thus, the amplitude ratio when $\varphi = -180$ (i.e., $\omega = w_n$) is 17.2 dB $20\log(\frac{k}{2\zeta\omega_n})$. Therefore, to provide a stability margin of 6 dB, *k* has to be reduced by:

$$17.2 + 6 = 23.2 dB$$

$$\therefore 20\log 960 - 23.2 = 20\log k$$

$$\frac{23.2}{20} = 1.16 = \log\frac{960}{k}$$

$$\frac{960}{k} = 10^{1.16} = 14.45$$

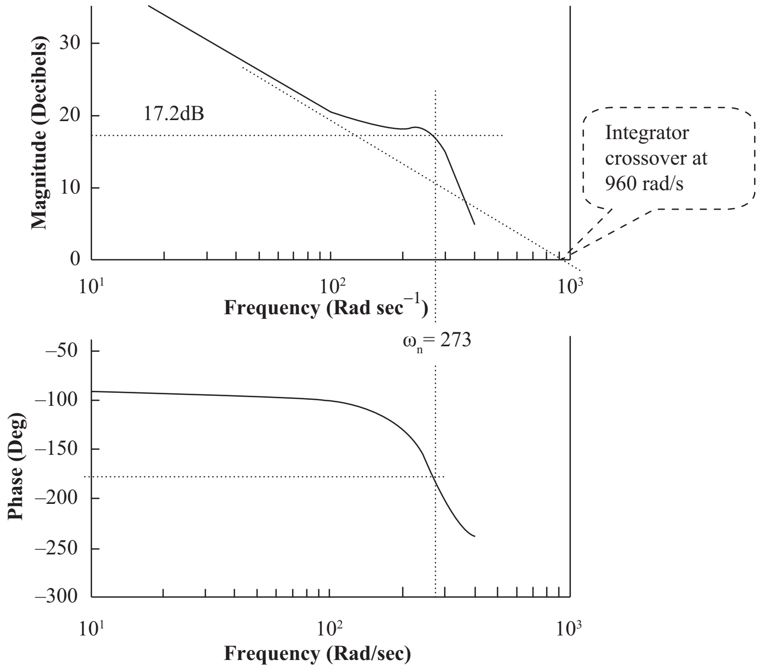

Figure 11.11. Open-loop frequency response.

Hence, the value of *k* to give an adequate stability margin must be:

$$k = \frac{960}{14.45} = 66.5\,rad\,s^{-1}$$

In the system as represented in Figure 11.11, there will be a power amplifier to deliver current to the valve. This amplifier will have an adjustment facility that will allow the gain to be set at the required value.

11.13 HYDRAULIC SYSTEM FOR INJECTION MOULDING MACHINE

A hydraulic system is required to operate an injection-moulding machine that has a movable platen using two hydraulic cylinders for the mould and an injection screw feed of granular material through the heater. Design data is given below:

11.13.1 SYSTEM DATA

- Maximum operating pressure = 150 bar
- Platen clamping force (to be pre-set) = 150,000 N (max)
- Platen movement, l = 50 mm
- Time required for platen movement (t_C) = 0.1 s
- Clamping time = 1.2 s
- Time for injection of material (t_I) = 0.1 s
- Feed screw torque = 1,500 Nm
- Feed screw operating speed = 200 rev min^{-1}
- Mass of platens = 250 kg
- Injection screw operates after the platens have closed

11.13.2 INJECTION-MOULDING MACHINE

For the diagram of an injection moulding machine shown in Figure 11.12 it is required to select appropriate sizes for the hydraulic motor and the clamping actuators.

11.13.3 ACTUATOR

Required clamping force = 150,000 N

Total actuator area, $A = \dfrac{150000}{150 \times 10^5}$ for 150 bar supply pressure

$$\therefore A = 10^{-2}\, m^2$$

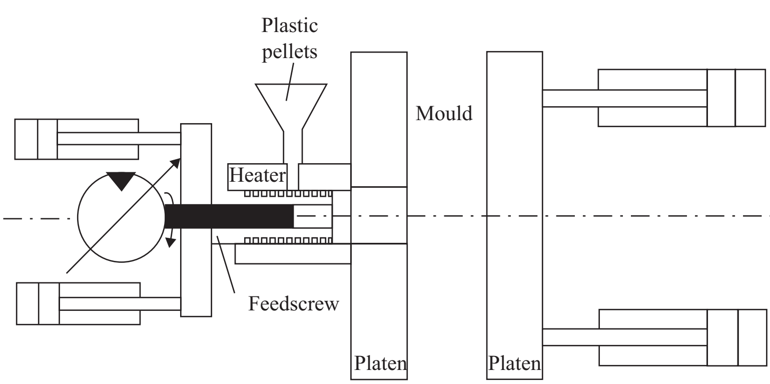

Figure 11.12. Injection-moulding machine schematic diagram.

This area is shared between the two cylinders, therefore, the flow required to move two cylinders:

$$Q_A = \left(\frac{Al}{t_C}\right) = \frac{10^{-2} \times 50 \times 10^{-3} \times 60 \times 10^{3}}{0.1} \quad \therefore Q_A = 300\ l/min$$

11.13.4 MOTOR

The motor displacement required,

$$D_m = \frac{T_m}{P} = \frac{1500 \times 2\pi \times 10^{3}}{150 \times 10^{5} \times 0.8} = 0.785\, L/rev$$

for a mechanical starting efficiency of 80 percent.

The motor flow, $Q_m = 0.785 \times 200 = 157 L/\min$.

11.13.5 VOLUME REQUIRED/CYCLE

Motor volume displaced/cycle,

$$V_m = \frac{D_m \times N \times t_i}{60} = \frac{0.785 \times 200 \times 0.1}{60} = 0.26L\,.$$

Actuator volume displaced/cycle, $V_A = 2 \times (A \times l) = 2 \times 10^{-2} \times 50 \times 10^{-3}$ $= 1L$ (assuming the same actuator area for retract and extend).

Total volume displaced/cycle = 1.26 L and for a cycle time of 1.4 s, the average flow, Q_C is: $Q_C = \frac{1.26}{1.4} \times 60 = 54 L/\min$. This is the average flow usage for the circuit, which has to be provided by the pump.

11.13.6 ACCUMULATOR

From Equation 6.7, Chapter 6, the accumulator capacity, V_0, required to provide the fluid volume, ΔV, is:

$$V_0 = \frac{P_2}{P_0} \frac{\Delta V}{\left[\left(\frac{P_2}{P_1}\right)^{1/\gamma} - 1\right]}$$

Taking P_1 = 100 bar, P_0 = 0.9 bar, P_1 = 90 bar, and P_2 = 150 bar. For 1.26 L total volume required in 0.3 s we get for the accumulator volume required:

$$\Delta V = 1.26 - \frac{0.3Q_A}{60} = 0.99L \text{, which gives } V_0 = 6.3L \text{ for } \gamma = 1.75$$

The value for γ has been taken from Chapter 6, Figure 6.3, for the application pressure level and the lowest likely temperature.

11.13.7 CIRCUIT

A basic circuit for this application is shown in Figure 11.13. Proportional valves would be used for the three flow control functions, their opening being set to give the desired flow in meter-in or meter-out as appropriate. A single valve could be used to operate the two actuators, if preferred, as they are mechanically linked together by the platen.

The selected accumulator needs to be capable of providing the maximum flow of 300 L min^{-1} for closing the platens. The use of an accumulator has provided a low-cost solution to this problem as it employs a fixed displacement pump, the size of which would have to be slightly greater

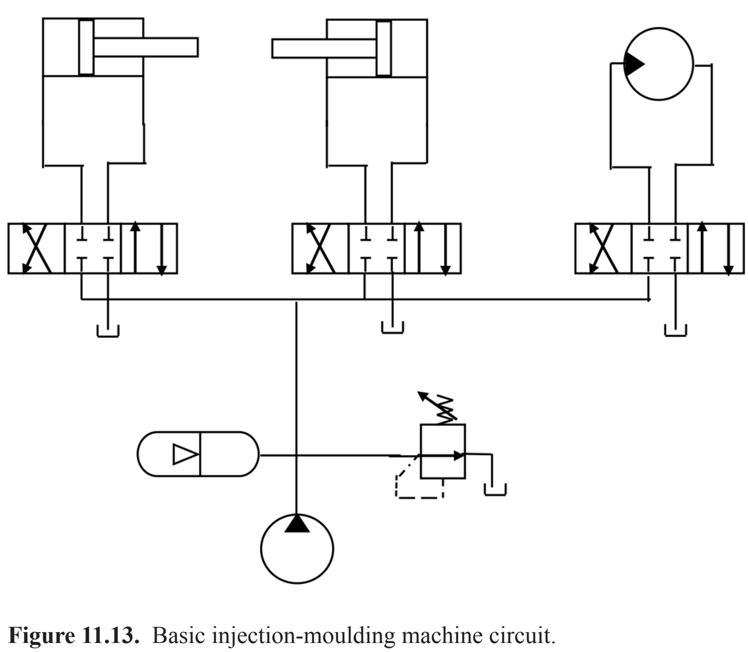

Figure 11.13. Basic injection-moulding machine circuit.

than the calculated value in order to account for the volumetric efficiency of the pump and motor.

11.13.8 THE PREVENTION OF PRESSURE SHOCKS

The use of controlled opening of the valve to reduce pressure shocks was discussed in Section 10.4.7, and in this example, the response of pressure and actuator displacement to step and ramp valve movements have been evaluated using a computer simulation of the system. The results are shown in Figure 11.14 for a step change and a 19-ms ramp change in valve movement.

It can be seen that the peak pressure amplitude has been reduced from 60 to 10 bar, which represents an 83 percent reduction in the acceleration force imparted to the actuator and, consequently to the machine itself. Neither of these pressures is in any way excessive but, as can be seen from the actuator displacement, high acceleration (hence, high pressure) is not required to achieve the velocity for sufficient time to move the actuator the specified distance. Indeed, only a very low pressure is needed to maintain the actuator velocity constant against a friction force that has been assumed to be proportional to velocity.

The natural frequency of the hydraulic system can be obtained from the following data: $A = 10^{-2} m^2$, $m = 250 kg$, viscous damping coefficient = 3,000 N (m s^{-1})$^{-1}$, $\beta = 1.4 \times 10^9 N/m^2$, Volume = 10^{-3} m^3

$$\therefore \omega_n = \sqrt{\frac{\beta A^2}{Vm}} = 750 rad/s = 119 Hz.$$

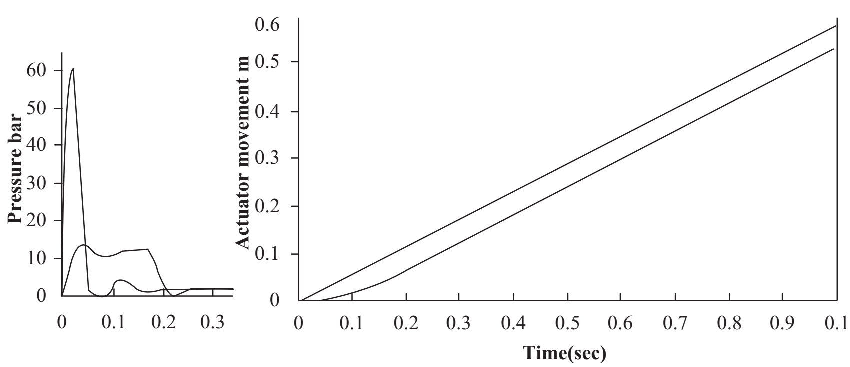

Figure 11.14. Variation of actuator pressure and displacement.

Thus $$t_C = \frac{1}{119} = 0.0084s.$$

The ratio of the ramp time of 19 ms to the time for one cycle, t_C is 2.3, which gives a guide to setting the valve opening time to reduce pressure shocks.

From Figure 11.14 it is seen that for the valve ramp case, the actuator takes about 10 ms longer to reach the specified extension of 50 mm. The use of a proportional valve to control the flow would allow the ramp time to be adjusted on installation of the system to reduce low-pressure shocks during operation.

11.14 OIL COOLING

The hydraulic circuit shown in Figure 11.15 is used to provide the feed of a circular saw and employs a meter-out pressure compensated flow control valve with a fixed displacement pump. It is required to select an appropriate cooler from the performance data in Figure 11.16. Assume that there is no heat loss to the surroundings.

11.14.1 SYSTEM DATA

Pump flow Q_S $= 50 \text{ L min}^{-1}$
Relief valve pressure P_S $= 200$ bar
Actuator area ratio a_R $= 1.33$
Pressure losses between pump
and actuator (both sides) ΔP_L = 20 bar = pressure loss for valve in central position
Open center valve pressure loss = 20 bar
Cooler performance conditions:
Oil to water inlet temperature difference $= \Delta T_{ow}$ for rated cooling power W_R

Performance at other temperature differences $W = W_R \dfrac{\Delta T_{ow}}{40} kW$

Water flow $= 1.4 \text{ L min}^{-1} \text{ kW}^{-1}$
Specific heat of water C_P $= 2{,}100 \text{ J kg}^{-1}\,°\text{C}^{-1}$

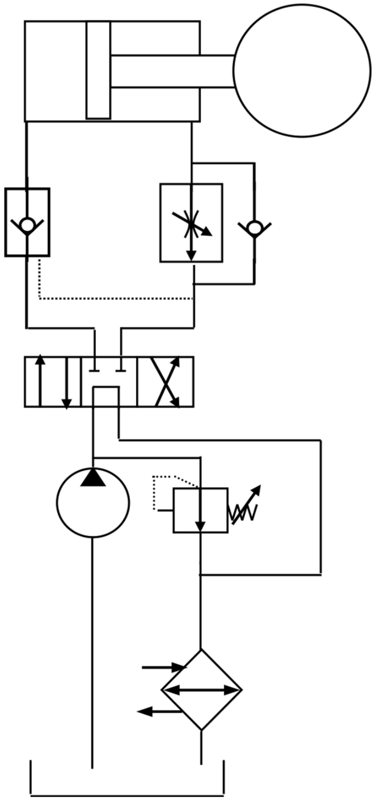

Figure 11.15. Hydraulic circuit.

11.14.2 DUTY CYCLE

Actuator extend

Duration t_E	= 120 s
Piston flow Q_E	= 20 L min^{-1}
Relief valve flow Q_V	= 30 L min^{-1}
Actuator return flow Q_R	= 15 L min^{-1}
Actuator retract	
Duration t_R	= 36 s
Annulus flow Q_A	= 50 L min^{-1}
Piston flow Q_P	= 66.7 L min^{-1}
Idle condition	
Duration t_I	= 60 s
Flow Q_S	= 50 L min^{-1}

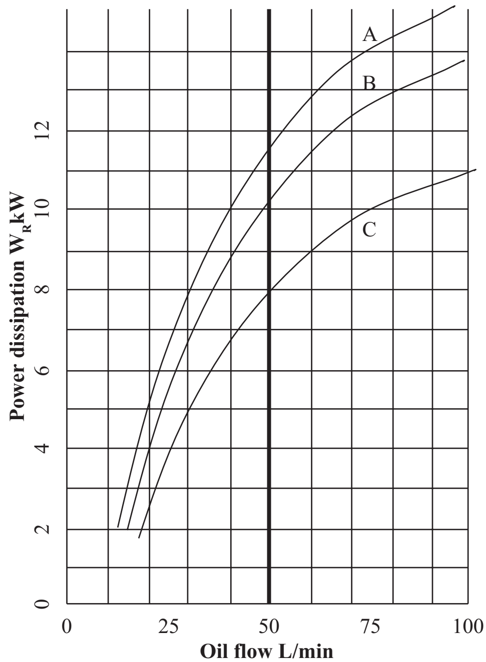

Figure 11.16. Cooler performance characteristics.

11.14.3 HEAT GENERATED

a. Actuator extension

The actuator extends under the control of the meter-out pressure compensated flow control valve so that the pressure created at the actuator outlet (annulus end) will rise to provide a force that satisfies the force balance on the actuator piston. The reaction force from the saw will normally be in opposition so the highest annulus pressure will occur when this force is zero.

For this situation, the annulus pressure $P_A = P_P \times a_R = 200 \times 1.33$ = 240 *bar*. This pressure will be dissipated as heat in the flow control valve and the pipes as will the pipe pressure loss in the inlet flow from the pump.

Energy dissipated in heat $\dfrac{Q(L/\min)}{60 \times 1000} \times P(bar) \times 10^5 \times t(s) = 1.67Q$

Energy dissipated in the inlet flow $H_E = 1.67 Q_E \Delta P_L t_E = 1.67 \times 20 \times 20 \times 120 = 80kJ$

Energy dissipated in the return flow $H_R = 1.67 Q_R P_A t_E = 720kJ$

Energy dissipated in the relief valve flow $H_V = 1.67 Q_V P_S t_E = 1200kJ$

b. Heat generated during retraction of the actuator
Energy dissipated in the inlet flow $H_E = 1.67 Q_A \Delta P_L t_R = 60kJ$
Energy dissipated in the return flow $H_R = 1.67 Q_P \Delta P_L t_R = 80kJ$

c. Heat generated during idle period
Energy dissipated through open center valve $H_I = 1.67 Q_S \Delta P_L t_I = 100kJ$

d. Total heat generation and cooler selection
The total heat generation = 2,240 kJ

$$\text{Average power dissipation } = \frac{2240}{216} = 10.4kW$$

For this power to be dissipated, it is necessary to select an appropriate cooler from Figure 11.16. The heat dissipated by the coolers is based on a difference between the oil and water inlet temperatures of 40°C and a water flow of 1.4 L min^{-1} kW^{-1}. The mean oil inlet temperature will need, therefore, to rise so as to create a sufficient temperature difference for the required power dissipation.

The power dissipation (W) at any other temperature difference is given by: $W = W_R \frac{(T_I - T_W)}{40}$ where W_R is the power dissipation at an oil flow of 50 L min^{-1}, T_I is the oil inlet temperature, and T_W is the water inlet temperature. For coolers A and B, the power dissipation at different oil inlet temperatures is shown in Table 11.10 for a water inlet temperature of 20°C.

The variation in the cooler power of Table 11.10 can be used to obtain the operating oil inlet temperature as shown in Figure 11.17. To dissipate 10.4 kW, cooler A will have to operate at an inlet temperature of 58°C and for cooler B, 63°C will be required. The reservoir temperature can be determined from:

Table 11.10. Power dissipated in the cooler

	Power dissipation W (kW)	
Oil inlet temperature T_I, °C	**Cooler A (W_R = 11.5 kW)**	**Cooler B (W_R = 10.1 kW)**
50	8.6	7.6
55	10	8.8
60	11.5	10.1

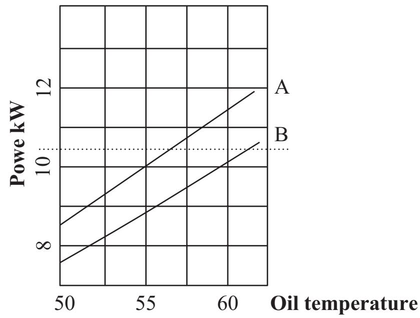

Figure 11.17. Oil inlet temperature.

$$W = \rho C_P Q \Delta T_C$$

$$\therefore \Delta T_C = \frac{W}{\rho C_P Q} = \frac{10400 \times 60 \times 1000}{870 \times 2100 \times 50} = 6.8^{\circ}\text{C}$$

Thus the reservoir temperatures will be, respectively, 51.2°C and 56.2°C. The required water flow = $1.4 \times 10.4 = 14.6L/\min$

11.14.4 HEAT LOSS TO THE SURROUNDINGS

The heat loss to the surroundings is difficult to quantify because of the wide range of heat transfer coefficients that can apply. If we consider a cube shaped reservoir having a capacity of 150 L with the dimensions of 0.6 m, the surface area will be 2.16 m^2.

The power dissipation due to convection is given by $W = UA\Delta T_A$ where U is the heat transfer coefficient that can have values in the range 2 to 25 W m^{-2}°C^{-1}. An effective area, A, of 2 m^2 with a mean tank temperature, T_T, of, say, 50°C, and an ambient temperature, T_A, of 20°C would provide heat dissipation in the range:

$$W = 2 \times 2 \times 30 = 120W \ \textit{to} \ 25 \times 2 \times 30 = 1.5kW$$

Consequently, heat dissipation to the environment can be quite significant but unreliable because of its variation with ambient conditions. It is normal to incorporate a thermostat into the cooling system that operates a flow control valve in the water supply so that the effects of ambient temperature and load force variations on the oil temperature can be reduced.

11.14.5 PUMP EFFICIENCY

The efficiency of the pump will also create a heat load in the oil. Thus for a pump overall efficiency of 90 percent, the heat generated in the pump due to this inefficiency is:

$$\left(\frac{1}{\eta}-1\right)\times\frac{Q(L/\min)P(bar)\times 10^5}{60\times 1000\times 1000}=0.11\frac{QP}{600}kW=1.83kW$$

Using a variable displacement pressure compensated pump will reduce the heat generated in the pump but more importantly, will reduce the heat generated in the relief valve because this flow will be eliminated thus approximately halving the heat dissipation. The size of the cooler could therefore be reduced by approximately 50 percent as can the required water flow.

11.15 VEHICLE TRANSMISSION

A track driven earth-moving vehicle uses a hydrostatic motor in each caterpillar track drive, the hydraulic power being supplied from a variable capacity pump driven at constant speed by the engine.

For the vehicle climbing an incline of 20 degrees, after making suitable allowance for losses, estimate:

a. The maximum vehicle speed.
b. The fluid pressure and flow in the transmission.
c. The effect on the transmission efficiency when the fluid viscosity reduces to 20 cSt.

Data:

Maximum engine power	250 kW
Hydraulic motor capacity	2,000 $cm^3\ rev^{-1}$
Motor gearbox ratio	2:1 reduction
Relief valve setting	280 bar
Vehicle wheel diameter	0.55 m
Vehicle weight	270 kN
Friction (speed independent)	10 kN
(speed dependent)	4 kN $(m\ s^{-1})^{-1}$
Track drive efficiency	90%
Motor/gearbox mechanical efficiency	92%
Pump/motor volumetric efficiency	95% (at 35 cSt viscosity)
Pump mechanical efficiency	95%

a. Vehicle speed

Total load force at the vehicle tracks $= 270 \sin 20° + 10 + (4\,U)\ kN$

The maximum vehicle speed may be limited by the available power, the system pressure, or the system flow rate. In this case, the maximum speed is limited by the power available.

The overall efficiency = $0.90 \times 0.92 \times 0.95 \times 0.95 \times 0.95 = 0.71$.

The output power at the tracks = $0.71 \times 250 = 178\, kW$

Hence the maximum vehicle speed whilst climbing the 20° gradient is given by:

$$178 = (270 \sin 20^{\circ} + 10 + 4U)U\ \ kW$$

$$\therefore 4U^2 + 102.3U - 178 = 0$$

For which

$$U = \frac{-102.3 \pm \sqrt{102.3^2 + 2.9 \times 10^3}}{8} = 1.64\, ms^{-1}$$

b. Pressure and flow

For the maximum vehicle velocity, the maximum load force is given by:

Max force at the tracks = $270 \sin 20^{\circ} + 10 + (4 \times 1.64) = 109 kN$

The motor torque is given by:

$$Torque = \frac{Max\,Force}{2 \times 2} \times Wheel\ radius$$

(i.e., two motors and a gearbox reduction ratio of 2:1).

$$\therefore T = \frac{109}{4} \times \frac{0.55}{2} = 7.5\, kNm$$

For this torque, the ideal system pressure is:

$$P_{ideal} = \frac{T}{D} = \frac{7.5 \times 10^3}{\left(\frac{2000 \times 10^{-6}}{2\pi}\right)} = 236\ bar$$

And for the motor/gearbox mechanical efficiency of 92 percent, and the track drive efficiency of 90 percent, the maximum working pressure is:

$$P = \frac{236}{0.90 \times 0.92} = 285\ bar$$

For the vehicle speed of 1.64 ms^{-1}, the ideal flow rate for all the motors is given by:

$$Q_{ideal} = 4(DN) = 4\left[2000x10^{-3}\left(\frac{1.64 \times 60}{\pi\ 0.55}\right)\right] = 455\ L\ \min^{-1}$$

Therefore, allowing for the motor volumetric efficiency, the maximum flow required at the motor inlet ports is:

$$Q = \frac{455}{0.95} = 479\, L / \min$$

Power check:
Hydraulic power

$$= P \times Q = 285 \times 10^{5} \times \frac{479 \times 10^{-3}}{60} \times \frac{1}{10^{3}} = 227.5 kW; \quad This\ is\ Power = \frac{PQ}{600}$$

Pump input power $= \dfrac{227.5}{0.95 \times 0.95} = 252 kW$

c. The effect of fluid viscosity

Motor $\eta_V = 0.95\ at\ 35\, cSt$

Now $\eta_V = \dfrac{Q_t - Q_s}{Q_t} = 1 - \dfrac{Q_s}{Q_t}\ \therefore \dfrac{Q_s}{Q_t} = 0.05$ (refer to Chapter 9)

For a fluid viscosity of 20 cSt $\dfrac{Q_s}{Q_t} = 0.05 \times \dfrac{35}{20} = 0.0875$

$$\therefore \eta_V = 91.25\%$$

Assuming that the pump volumetric efficiency will change by the same proportion with viscosity as the motor.

Overall efficiency at 35 cSt = 71%. At 20 cSt, the overall efficiency

$$= 71 \times \left(\frac{91.25}{95}\right)^2 = 65.5\%$$

d. Pump controls

The methods of controlling the flow and pressure from the variable displacement pump discussed in Chapter 7 can be applied to this transmission system. It is required to control:

- Maximum pressure.
- Pump torque. At a constant pump speed, this would limit the input power to the pump.
- Pump output flow.

Figure 7.27 of Chapter 7 is reproduced above as Figure 11.18, which portrays the three parameters that are to be controlled. Maximum pump pressure is controlled by the compensator shown in Figure 11.19 (reproduced from Figure 7.25, Chapter 7) that reduces the pump displacement should any service demand a pressure that is higher than that set by the

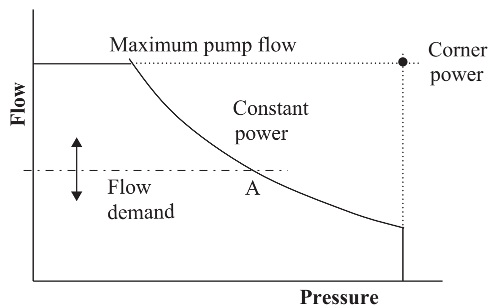

Figure 11.18. Pump control strategy.

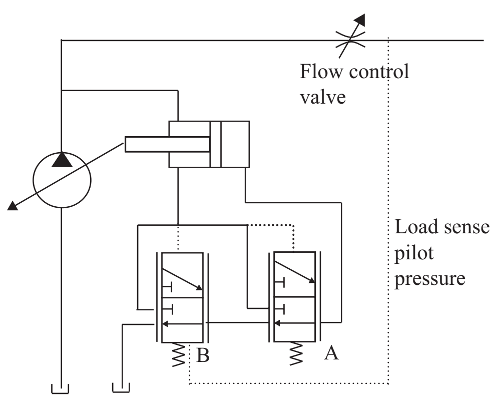

Figure 11.19. Variable displacement pump with pressure compensation and flow control.

compensator (valve A). This control is capable of reducing the net output flow to zero if necessary (stalled condition).

The pressure drop across the selected valve opening will control the pump displacement to maintain constant flow to the service (valve B). Referring to Figure 11.18, this sets the flow at point "A" which will be limited by either:

- The pressure compensator, as described above or
- The setting of the pump power (torque) control shown in Figure 11.20 (reproduced from Figure 7.26, Chapter 7).

The operation of the power control, which is described in Chapter 7, is achieved by sensing a pressure signal that varies with the pump displacement. This pressure provides a feedback to the pump displacement control valve that will maintain the displacement so that the pump output power is constant under changing load pressure conditions. An analytical method for evaluating the steady state performance of the control is given at the end of this section.

For the track drive with the vehicle traveling on flat ground against low resistance, maximum speed will be obtained because the required pressure will be reduced in relation to that required when moving up the 20° gradient.

Referring to Figure 11.18, the flow of 479 L min^{-1}, (b), will be selected by the flow control valve, the pump displacement being controlled by the pressure drop valve B in Figure 11.19. The pressure of 285 bar that is required

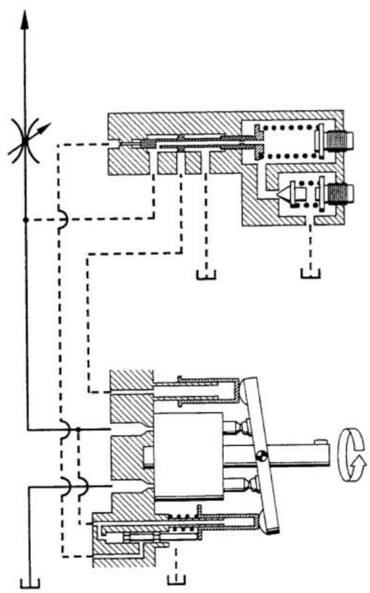

Figure 11.20. Pump power (torque) control.

for the 20° incline may be limited by the constant power control (point A at, say, 15 percent lower pressure than the pump compensator setting).

With a displacement increase of, say, 150 percent (2.5:1), the vehicle will be capable of traveling at $2.5 \times 1.64 = 4.1$ ms^{-1} (14.8 km h^{-1}). The higher flow will be selected by the flow control valve, the pressure being limited by the constant power control should the pressure increase above the corresponding value for the selected flow. Maximum speed will naturally be determined by the maximum displacement of the pump.

11.15.1 STEADY STATE PUMP POWER CONTROL ANALYSIS (FIGURE 11.20)

11.15.1.1 Displacement Control Valve

The steady state valve forces are given by:

$$P_S a_1 + P_P a_2 = C + KX$$

For a spring stiffness K, preload C, and pressure sensing areas a_1 and a_2, X is the valve movement away from the closed position. P_P and P_S are, respectively, the pump outlet pressure and the signal pressure from the displacement hydraulic potentiometer as in the diagram below.

For $X = 0$ in steady state conditions, the force balance of the valve gives:

$$P_S = \frac{C}{a_1} - P_P \frac{a_2}{a_1} \tag{11.1}$$

This gives a relationship between the two pressures for the valve to be held in a closed position ($X = 0$).

11.15.1.2 Displacement-Sensing Hydraulic Potentiometer

The length of the restriction A (L_1) is reduced with increasing displacement and the length of B (L_2) increases. Considering the flow through the pressure signal hydraulic potentiometer and assuming laminar flow (i.e., $Q \propto \dfrac{\Delta P}{L}$) gives:

$$Q = \frac{K_1(P_P - P_S)}{L_1} = \frac{K_1 P_S}{L_2} \tag{11.2}$$

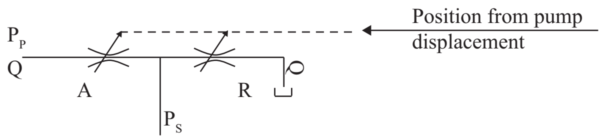

Figure 11.21. Hydraulic potentiometer.

Now $L_1 + L_2 + L$ (total length of flow path). (11.3)

$L_1 = 0$ when pump displacement $D = D_{max}$

$L_2 = 0$ when pump displacement $D = 0$

For Figure 11.21 choose the geometry such that:

$$L_1 = L\left(1 - \frac{D}{D_{max}}\right) \text{ and } L_2 = L\frac{D}{D_{max}} \tag{11.4}$$

Equation 11.2 gives:

$$P_S = \frac{P_P}{1 + L_1 / L_2} \tag{11.5}$$

Then, with Equation 11.4

$$P_S = \frac{D}{D_{max}} P_P \tag{11.6}$$

Equations 11.1 and 11.6 give:

$$P_S = \frac{D}{D_{max}} P_P = \frac{C}{a_1} - \frac{a_2}{a_1} P_P \text{, so } P_P\left(\frac{D}{D_{max}} + \frac{a_2}{a_1}\right) = \frac{C}{a_1} \tag{11.7}$$

Equation 11.7 shows that in the steady state, increases in the pump pressure P_P will be associated with reductions in the pump displacement D at a level that depends on the valve sensing area ratio to give an approximate constant power with the pump operating at constant speed.

11.16 PUMP CONTROL APPLICATIONS

11.16.1 APPLICATION OF PUMP UNLOADER VALVE TO VEHICLE CRUSHER/REFUSE MACHINES

A pump-unloading valve can be used as shown in Figure 11.22 so that high speed can be obtained from both pumps together. When the force

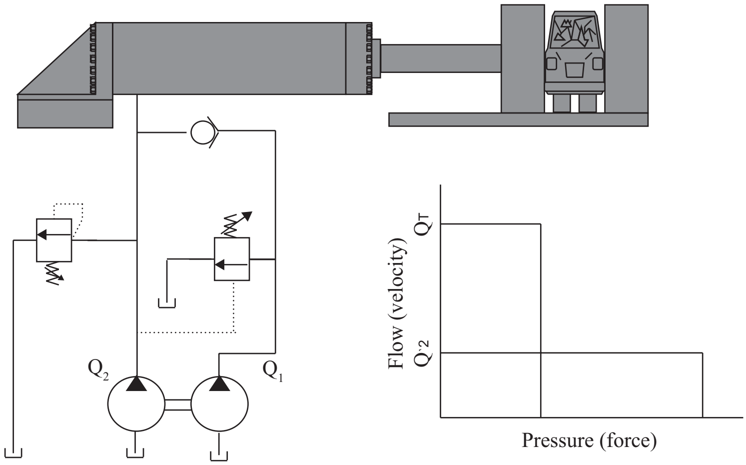

Figure 11.22. Vehicle crusher unloading pump circuit.

increases so that the pressure rises to a level that is higher than the setting of the unloading valve, the valve opens and allows flow from one pump to be bypassed at low pressure to tank. The check valve separates the two circuits.

To limit the maximum power, the set pressures for the unloading and relief valves need to be in the ratio:

$$\frac{P_2}{P_1} = \frac{Q_T}{Q_2} = \frac{Q_1 + Q_2}{Q_2} = 1 + \frac{Q_1}{Q_2}$$

$$\text{Thus for } Q_2 = \frac{Q_1}{2};\ \frac{P_2}{P_1} = 3$$

Triple pumps can provide three pressure/flow ranges by the use of two unloading valves set at different pressures.

11.16.2 APPLICATION OF PUMP CONTROLS TO BENDING MACHINES

Operation of the bending machine, shown in Figure 11.23, requires the following actions to be carried out:

- Rotate drive rollers (hydraulic motor)
- Operate cutter

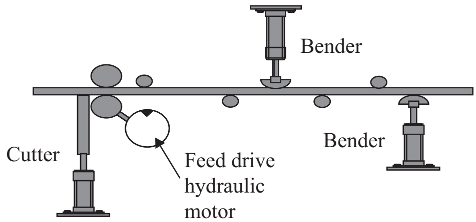

Figure 11.23. Bending machine schematic.

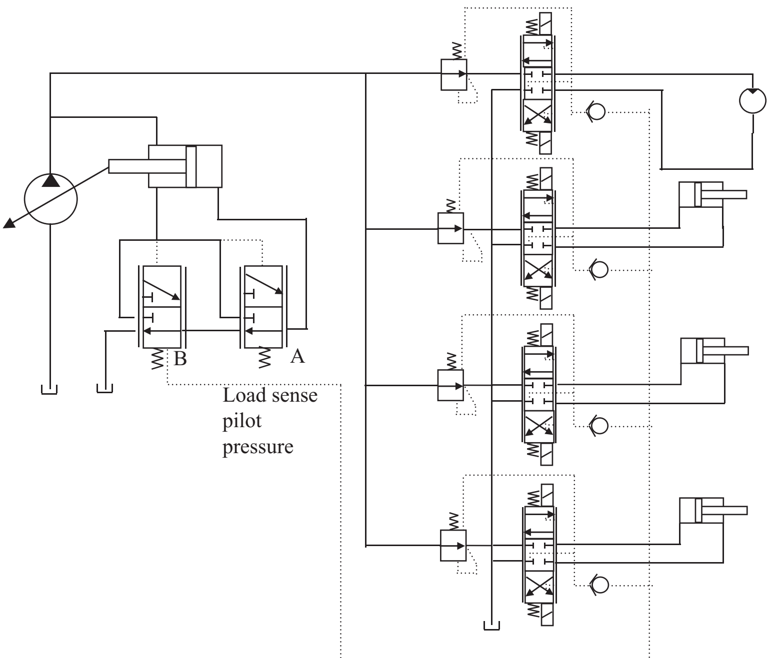

Figure 11.24. Bending machine circuit.

- Operate two bender actuators
- Push finished piece into a skip

The circuit in Figure 11.24 controls the flow to each service by maintaining the pump pressure at a level set by valve B from the highest load pressure. The directional control valves set the flow required for each service with the individual flow compensators maintaining constant pressure drop across each valve.

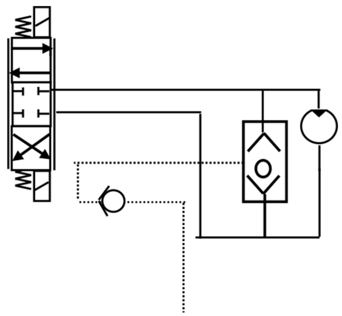

Figure 11.25. Alternative load sensing system.

Note:

A shuttle valve is often used to select the highest actuator or motor pressure for the load sense signal as shown in Figure 11.22. In Figure 11.25 a shuttle valve is used to provide an alternative load sensing system.

11.17 APPLICATION OF COMPENSATION TECHNIQUES

It is required to improve the performance of the electrohydraulically operated crane mechanism shown in Figure 11.26, which is controlled by closed-loop position feedback. The compensation techniques discussed in Chapter 10 will be used to demonstrate the application of these methods.

11.17.1 STEADY STATE ACCURACY

A major problem experienced by the crane is the effect of changes in the external forces on the steady state position due to valve leakage when it is in the null position. A result obtained from a simulation of the system is shown in Figure 11.27, which has a feedback transducer gain of 20 V m^{-1}. This shows that for the change in the force there will be an error of about 1.5 mm in the crane position for the valve having leakage at the null position (underlap).

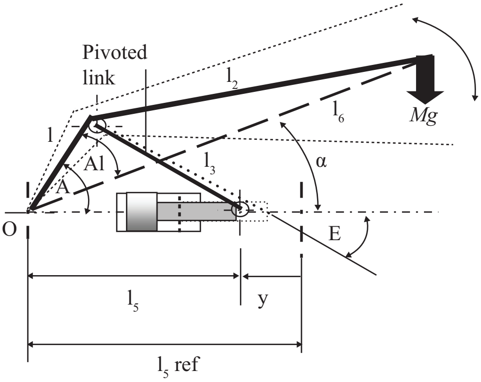

Figure 11.26. Hydraulically operated crane.

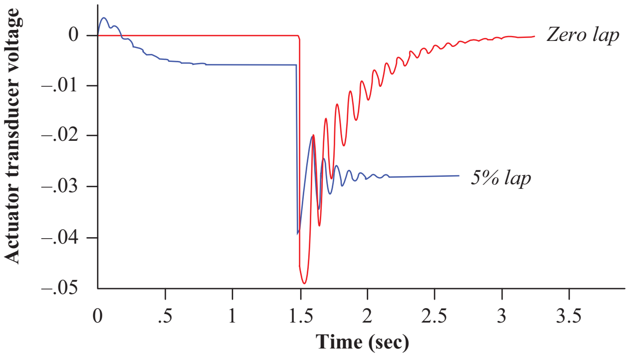

Figure 11.27. The effect of changes in external forces on steady state errors.

11.17.2 PROPORTIONAL PLUS INTEGRAL COMPENSATION

Figure 11.28 shows how the use of integral plus proportional (P + I) compensation removes the steady state error that arose in the original system with the application of external forces. The choice of coefficients for the compensator depends on their effectiveness and the stability of the overall system.

The second method of Ziegler–Nichols for selecting the gains is based on the value of the loop gain, K_M, that will cause continuous oscillations of the system and the frequency of the oscillations, f_n. Continuous oscillations occur at a natural frequency of 10 Hz with an open loop gain value, K_M, of 6.

The P + I compensator has the transfer function given by:

$$K_P\left(1+\frac{s}{T_I}\right)$$

where the coefficient values by the Ziegler–Nichols method are given by:

$K_P = 0.45K_M$ and $T_I = 0.83T_P$ and T_P = periodic time of oscillations = $1/f_n$.

Figure 11.29 shows the system step response with different values of the integral coefficient and it can be seen that the value has an important effect on the overshoot and successive oscillations.

The values obtained using the Ziegler–Nichols method give K_P = 2.7 and T_I = 0.083 and a transfer function:

$$\frac{2.7}{s}(s+12.5) \tag{11.8}$$

$$\text{Or } = \frac{2.7}{s}(s+a) \text{ (for the uncompensated system } a = 0\text{)}$$

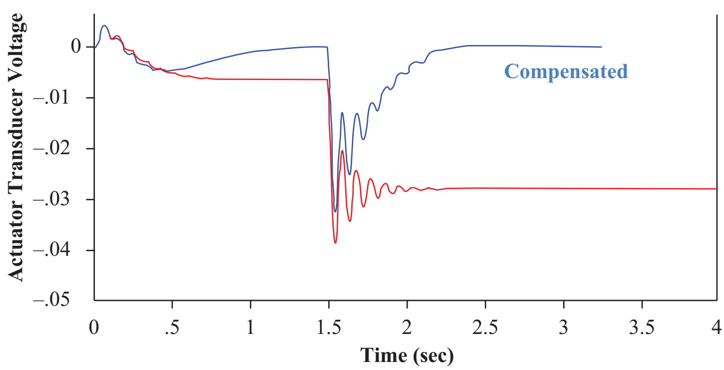

Figure 11.28. The effect of adding proportional plus integral compensation.

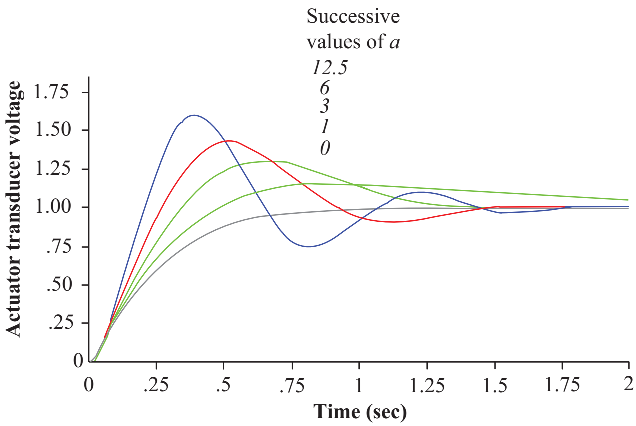

Figure 11.29. Step response with the P + I compensator.

These values are those used in the simulation, the results of which are shown in Figure 11.29.

11.17.3 PROPORTIONAL PLUS INTEGRAL PLUS DERIVATIVE COMPENSATION

The transfer function for the Proportional Plus Integral Plus Derivative (PID*)* compensator can be written:

$$G_C = K_P\left(1 + \frac{1}{T_p s} + T_D s\right) \tag{11.9}$$

The Ziegler–Nichols second method proposes the following values of the PID coefficients:

- $K_P = 0.6 \times$ the proportional gain K_M that just causes the system to oscillate
- $T_P = 0.5 \times$ the time for one cycle of the oscillations $= \dfrac{0.5}{f_n}$
- $T_D = 0.13 \times$ the time for one cycle of the oscillations $= \dfrac{0.13}{f_n}$

Applying the values for the crane linearized simulation with the data from Section 11.17.2 gives a transfer function for the PID compensator:

$$G_C = 3.6(1 + \frac{1}{0.05s} + 0.013s) = \frac{3.6}{s}(0.013s^2 + s + 20) \tag{11.10}$$

For the system the natural hydraulic frequency is 10 Hz with a damping factor of approximately 0.6. Using these values gives a linearized transfer function of the system:

$$K_P \frac{1}{(1+0.002s+0.00025s^2)} \tag{11.11}$$

The value of K_P is set in the controller amplifier, which for applying the PID compensator will be 3.6.

The response of the system using Equation 11.11 is shown in Figure 11.30 with the compensator in Equation 11.10 and a modified compensator, which is seen to provide an improved response.

11.17.4 LOAD PRESSURE FEEDBACK

The application of load pressure feedback was shown in Section 10.8.5 to improve the damping coefficient of the closed-loop hydraulic system. The effect of increasing the damping ratio by 10 times to around 0.6 allows the system gain to be increased by more than 100 percent as shown in Figure 11.31. The use of the compensators has created oscillations in the closed loop depending on the values of the coefficients that are chosen. However, the compensators are put into the system to reduce the steady state error created by the null position leakage of the valve.

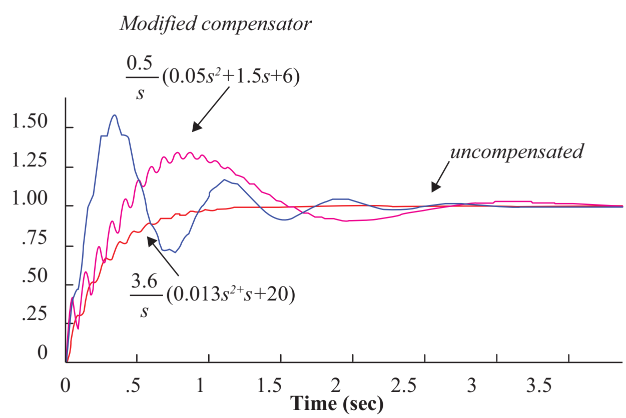

Figure 11.30. Dynamic responses using a *PID* compensator.

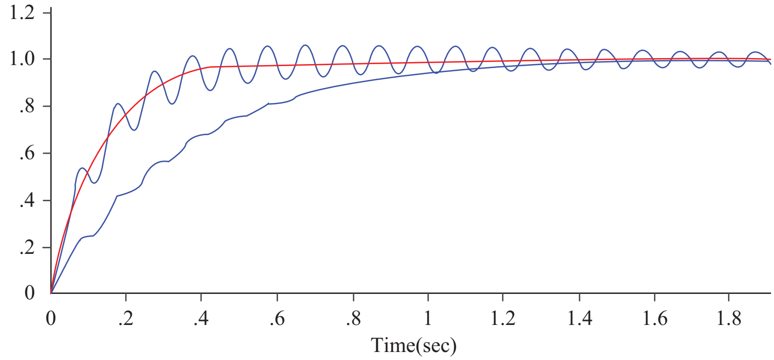

Figure 11.31. The application of load pressure feedback.

The increased damping ratio not only is seen to allow the use of an increase in the system gain but also eliminates oscillations even at the higher gain value.

11.17.5 CONCLUDING REMARKS

The results using the Ziegler–Nichols values for the coefficients give excessive oscillations for both the P + I and PID compensators, and changes to these values provided considerable improvements. Both of these methods provide the elimination of steady state errors and improved response but optimization to reduce oscillations requires a trial and error approach, which is greatly assisted by the use of simulation techniques.

This emphasizes the considerable choice that is available for setting these gains in that stability is obtained but the damping is not clearly defined. In most systems, the values of the coefficients are determined by trial and error methods.

Other methods are also available mostly using digital techniques, which include observer, pole placement, sliding mode control, and other forms of robust control.

11.18 HYDROSTATIC TRANSMISSION EXAMPLE

The analyses in Section 10.9.1 for primary (pump control) hydrostatic transmission system show that for the simple system model that is used, the basic transfer function is of second order.

This worked example considers a given load application for the transfer function that relates changes in motor speed to changes in load torque.

11.18.1 COMPONENT DETAILS

Maximum motor displacement = 250 cm^3 rev^{-1} = 4×10^{-5} m^3 rad^{-1}
Supply pressure, P = 250 bar
Oil bulk modulus $\beta = 1.2 \times 10^9$ N m^{-2}
Viscous friction coefficient for the transmission $C_f = 2Nm/(rad/s)$

For the motor operating at 1,500 rev min^{-1}, taking the volumetric efficiencies for the supply pump and motor as 95 percent gives a total leakage flow of:

Leakage flow $Q_L = 375 \times 0.1 = 37.5$ L min^{-1}

Hence the leakage coefficient $C_L = \dfrac{37.5}{60 \times 10^3 \times 250 \times 10^5} = 25 \times 10^{-12}$ m^3 s^{-1} (N m^{-2})$^{-1}$

For a pipe diameter of 30 mm, taking a pipe length of 10 m together with an allowance for the pump and motor volumes gives a total pressurized volume of: $V = 7.25$ L

11.18.2 SYSTEM ANALYSIS

The analysis in Section 10.9.1 derived the equations for a linearized transfer function of the system parameters. In particular, it is important to know how the motor speed will vary to changes in the torque load on the motor shaft. The transfer function is;

$$\frac{\omega_m}{\Delta T_L} = -\frac{\left(C_L + \dfrac{V}{\beta}s\right)}{\left(D_m^2 + C_f C_L\right) + \left(JC_L + C_f \dfrac{V}{\beta}\right)s + \dfrac{VJ}{\beta}s^2}$$

Damping factor $\zeta = \dfrac{1}{2\sqrt{k_1 D_m^2}}\left\{C_f\sqrt{\dfrac{V}{J\beta}} + C_L\sqrt{\dfrac{J\beta}{V}}\right\}$

Steady state gain $K_{SS} = -\dfrac{C_L}{D_m^2 k_1}$

Natural frequency $\omega_n = \sqrt{\dfrac{\beta D_m^2 k_1}{VJ}}$

11.18.3 CALCULATED VALUES

Table 11.11. Dynamic performance parameters

Load inertia J (kg m^2)	Motor displacement (%)	Natural frequency ω_n (rad s^{-1})	Damping ratio, ζ	Steady state gain K_S (rad s^{-1} Nm^{-1})
5	100	7.4	0.31	0.015
5	50	3.9	0.62	0.061
10	100	5.2	0.42	0.015
10	50	2.7	0.84	0.061

11.18.4 SUMMARY

As seen from the equation for the damping factor, changes in inertia *J* and system volume *V* have opposite effects, the relative magnitude of which depends on the relative values of the friction and leakage coefficients. The values for the damping factor indicate that the leakage coefficient term has a considerably greater influence on the value of ζ than does that of the friction coefficient term.

System volume also has an influence on the value of the damping coefficient and from the previous comment increases in the pipe length could result in unsatisfactory values of ζ.

Motor speed will vary with load torque because of the influence of friction and leakage. If greater accuracy of speed is required it may be necessary to investigate a closed loop control such as that discussed in Section 11.19.2.

11.19 EXAMPLE OF A SECONDARY CONTROL SYSTEM

The hydrostatic transmission having secondary control in which the motor displacement is controlled in a closed-loop system to maintain a constant motor speed as discussed in Section 10.9.2.

11.19.1 COMPONENT DETAILS

The system component dimensions are taken to be the same as those in Section 11.18.

The motor displacement controller transfer function is taken here to be first order having a time constant T_C of 0.05 s. It is probable that it will be of higher than first order but it is likely that such higher order terms would have a negligible effect on the overall response.

Speed feedback gain $K_F = 0.1$ V (rad $s^{-1})^{-1}$,
Motor displacement controller gain $K_G = 1\times10^{-4} m^3 / V$
Load friction coefficient $C_L = 2Nm/\left(rad/s\right)$.

11.19.2 SYSTEM ANALYSIS

The analysis in Section 10.9.2 derived the equations for a linearized transfer function of the system parameters. As this is a closed-loop control on motor speed, it is important to investigate the effect of load torque on the motor speed. From Section 10.9.2.3, the transfer function for speed changes in relation to changes in torque is:

$$\frac{\omega}{\Delta T_L} = -\frac{\left(1+T_C s\right)}{\left(PK_G K_F + C_f\right) + \left(T_C C_f + J\right)s + JT_C s^2}$$

Steady state gain $K_S = -\dfrac{1}{\left(PK_G K_F + C_f\right)}$

Natural frequency $\omega_n^2 = \dfrac{\left(PK_G K_F + C_f\right)}{JT_C}$

Damping factor $\zeta = \dfrac{\left(J + C_f T_C\right)}{2\sqrt{JT_C\left(PK_G K_F + C_f\right)}}$

11.19.3 CALCULATED VALUES

For two values of the load inertia of 5 and 10 kg m^2, the calculated values for the above parameters are shown in Table 11.12. The secondary control is not influenced by variations in the hydraulic system parameters.

Table 11.12. Dynamic performance parameters

Load inertia J, kg m^2	Natural frequency ω_n (rad s^{-1})	Damping ratio ζ	Steady state gain K_S (rad s^{-1} Nm^{-1})
5	32	0.32	0.004
10	22.5	0.45	0.004

11.19.4 SUMMARY

The steady state gain K_S results in speed variations with changes in the load torque. By the appropriate use of compensators such as those described in Section 10.8 and other digital techniques, this speed error can be minimized. In this secondary control system, the control parameters are not functions of the hydraulic system design because the supply pressure is being maintained constant by the pump control.

For ring main systems such as for ships and mobile drives, the application of secondary control has the following benefits in relation to conventional hydraulic systems:

- Energy regeneration is possible using an accumulator for energy storage in the hydraulic side of the system.
- The dynamic response and stability of a motor speed controller can be set up by the use of appropriate techniques.
- In multi-motor ring main systems, the use of multi-pumps can be arranged to achieve a high level of the mean operating efficiency and reliability of the system.
- Multi-function output systems can allow an overall reduction in weight in some cases in relation to individual hydrostatic transmissions by reducing the number of pumps in the system.
- The possibility of being able to hold a loaded motor stationary by means of its displacement control can be advantageous in, for example, winch systems.
- The closed-loop control of speed avoids speed changes that occur in hydrostatic transmission systems due to leakage and regulation of the prime mover after changes from motoring to pumping (e.g., mobile and winch drives).

11.20 WIND TURBINE HYDROSTATIC TRANSMISSION SYSTEM EXAMPLE

This example shows how the hydrostatic system, described in Section 10.9.3, can be used for transmitting the power from a wind turbine to the generator and providing speed control of the turbine over the range of operating conditions.

11.20.1 COMPONENT DETAILS

Data for the 225 kW turbine

Table 11.13. Hydrostatic transmission parameters

Maximum turbine speed (rev min^{-1})	**Pump displace-ment (L rev^{-1})**	**Generator speed (rev min^{-1})**	**Maximum motor dis-placement (L rev^{-1})**	**Maximum output power (kW)**	**Turbine inertia (Kgm^2)**	**System leakage coefficient, (m^3 s^{-1}) (N m^{-2})**
45	11.3	1,000	0.5	225	30,000	3.6×10^{-11}

Oil bulk modulus $\beta = 1 \times 10^9$ N m^{-2}

For the 225 kW turbine, the pump and motor sizes were selected as shown in Table 11.13.

11.20.2 SYSTEM ANALYSIS

For the assumption that the generator speed remains unchanged during transient changes from Equation 10.93, the transfer function for changes in turbine speed to changes in motor displacement is given by:

$$\omega_p = \frac{1}{D_p} \frac{\Omega_m d_m}{((1-a)+(b-c)s+ds^2)}$$

The damping factor and natural frequency for the second order equivalent transfer function for changes in turbine speed, ωp, to changes in motor displacement, d_m, is given by Equations 10.93 and 10.95:

$$\zeta = \frac{1}{2}\left(\frac{C_L J_p}{D_p^2} - \frac{VC_T}{\beta D_p^2}\right)\omega_n = \frac{1}{2D_p}\left[C_L\sqrt{\frac{J_p\beta}{V}} - C_T\sqrt{\frac{V}{J_p\beta}}\right]; \qquad \omega_n^2 = \frac{\beta D_P^2}{VJ}$$

11.20.3 CALCULATED VALUES

The maximum value for $\zeta > 0$ of the coefficient C_T (the slope of the turbine torque/speed characteristic) is given from Equation 10.77:

$$C_T \le C_L\left[\frac{J_p\beta}{V}\right]$$

Using the values for the turbine transmission gives this maximum value of C_T to be 9,640 Nm (rad s^{-1})$^{-1}$. Figure 11.32 shows the torque/speed relationship for this turbine at two wind speeds and blade pitch angles. For speeds in the region of 45 rev min^{-1}, which is the operating speed at these wind speeds, the slope of the curve is approximately the same as the critical value at a wind speed of 13 ms^{-1}. This is confirmed by the oscillatory variation of turbine speed in Figure 11.33 predicted by a nonlinear simulation.

As described in Section 10.9.3.1, the use of a proportional closed-loop turbine speed control with feedback of the hydraulic pressure as shown in the control block diagram in Figure 10.28 will provide high damping and accurate control of the turbine speed.

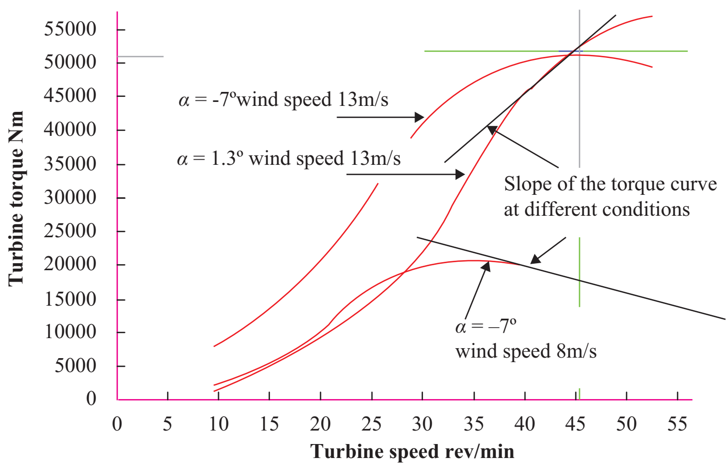

Figure 11.32. Turbine torque variation with turbine speed at different wind speeds and turbine blade pitch angle α.

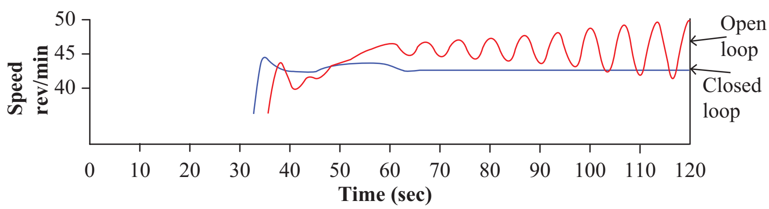

Figure 11.33. Nonlinear simulation of turbine speed to change of wind speed at 50 s.

Figure 11.33 shows the simulated response of turbine speed to a change in wind speed, where it can be seen that the closed-loop control has eliminated the speed oscillations that arise in open loop with fixed motor displacement. Thus, the development of an effective closed-loop control of turbine speed has made it possible to consider the approach that can be taken for varying the turbine speed so as to obtain optimum performance over the full range of operating conditions. For small changes in turbine speed following the change in wind speed, the linear analysis has, as described, used the tangential slopes to local points on the curved line of the turbine torque/speed characteristic. It is found that the linear method predicts the same response as the nonlinear simulation result shown in Figure 11.33.

The linearized approach is a very powerful method for investigating the dynamic behavior of systems and developing an appropriate design of the control system.

CHAPTER 12

Systems Management

12.1 INTRODUCTION

The book has concentrated on describing the operating principles of hydraulic components, how they are used in hydraulic systems, together with the analytical methods that can be used to determine the steady state and dynamic performance of the system.

The management of a hydraulic system is concerned with all aspects from the initial design stage to its final disposal after completing a useful life. During the design phase, the components and circuits that are selected for the system need to provide the performance that is required to meet the machine specification and have the necessary life to meet the expected duty cycle under the given environmental conditions.

For a given application, there will be a range of different types of system operating principles that could be employed, and it is necessary for the designer to select the system that is most suitable for the application. Key aspects would include efficiency, suitable life expectation, the system environment, control method, and so on.

The designer needs to have some knowledge of standards that relate to hydraulic components and systems. These could be national standards such as British Standards (BS), International Standards (ISO), or European Standards (EN). Some of these standards are listed in the Appendix and discussed later in this chapter. There are around 300 BS, ISO, and EN standards that are listed in the British Fluid Power Association (BFPA) *Engineers Data Book* mentioned in the Appendix. Obviously, different countries may have their own standards, but the intention is that eventually there will only be ISO standards. For example, many BS standards are equivalent or identical to ISO standards, but not all of them have been converted yet. Some of these will be presented to show their relationship to the design process.

It is important to ensure that the system is properly installed and commissioned and that appropriate maintenance and operational procedures are put into place by the machine builder/user.

Ultimately, the major aspect of management is to achieve the most economic cost over the total lifetime of the equipment. This does not refer only to the cost of purchase because a higher cost system may provide a lower running cost so that in its lifetime the total cost may be less than that obtained with a system having a lower purchase cost. Reliability is a major issue because the level of this will reflect on the maintenance of the system and its operating cost in relation to the cost of machine "downtime".

It may be preferable in some circumstances to use condition-monitoring techniques to determine loss of performance and predict component failure. This may also be required where a high level of safety is demanded by the specification, and often it may be necessary to perform a fault analysis for the system. All of these aspects will need to be considered as part of the systems management process.

The management of hydraulic systems is an important feature that needs to be considered during the design process Clearly, this involves a number of factors, which include the complexity of the system, the machine duty cycle, its environment, and the level of reliability required by the machine builder/user.

12.2 ASPECTS OF SYSTEMS MANAGEMENT

In general, systems management will need to:

1. Ensure that the proposed system will meet the machine specification.
2. Carry out a fault or failure analysis of the system.
3. Determine the reliability of the system using information from point 2. The new European Machine Directive (2006/42/EC) states that machine safety is evaluated according to its reliability. This is specified in the standard EN ISO 13849.
4. Establish that the procured components/systems achieve the specified levels of cleanliness and are protected prior to, and during, installation from the ingression of contaminants.
5. Check that the system has been installed correctly, flushed and filled to the correct level, and commissioned appropriately.

6. Determine the type of maintenance procedure (i.e., preventative or corrective) to be used.
7. Establish the training of personnel to a technical level of competence in order to achieve reliable and economic operation of the system and deal with problems that arise during its use.

12.3 SYSTEMS MANAGEMENT OBJECTIVES

The necessity and level of systems management needs to be determined. For example, systems used for research test rigs, complex manufacturing machines, and auxiliary drives on mobile plant will require quite different approaches. The objectives of the management system will, therefore, need to be defined. Likely objectives would include those concerned with the following:

- The minimization of operating costs
- The maintenance of end-product quality
- The maximization of system reliability
- The assurance of a high level of safety

The achievement of these objectives will be strongly related to the design of the system in relation to the machine duty cycle and its environment. To achieve high reliability, it may be considered appropriate to measure major parameters that would include pressures, temperatures, valve positions, pump and motor displacements, and flows for condition-monitoring purposes. This would allow predictive maintenance procedures to be used, which can avoid machine downtime due to component failures. It has been shown that reductions in the total cost of systems in their lifetime can be upward of 10 percent by the use of condition-monitoring methods.

12.4 SYSTEM CLEANLINESS

The premature failure of components and unreliable operation of hydraulic systems is often the result of inadequate contamination control, the incorporation of which should be part of the system design process. During installation of the system, the levels of particulate contamination should be monitored using a particle counter, and the system should be operated until these levels have reduced to the specified value. This process may require the filter to be changed in order that contaminants do not pass into sensitive components. When the contaminant level reaches a satisfactory level, it is often stated that the system should be drained, a new filter fitted, and the system filled with new oil. However, the flushing process also cleans the fluid, which in many instances may well be cleaner than new unflushed oil, so it could be preferred not to replace the oil. It is up to the user to decide which option to choose.

Ideally, the achievement of optimum system life and reliability is obtained by co-operation of the various manufacturers and suppliers involved in its manufacture and operation, This is represented in Figure 12.1. The application of such a total cleanliness control program will reduce the failure rate for systems as shown by the well-known "bathtub" curve in Figure 12.2. This has particular effect on the number of failures at the beginning and end of the product life and consequently the overall life.

12.5 FAULT ANALYSIS

The evaluation of circuit faults needs to examine all of the system operating modes and states. However, circuit drawings are often complex, they are only able to show one operating state, and, generally, they do not show component sizes, operating conditions, and ratings.

Two commonly employed fault-finding methods include Fault Tree Analysis (FTA) and Failure Modes Effects Analysis (FMEA).

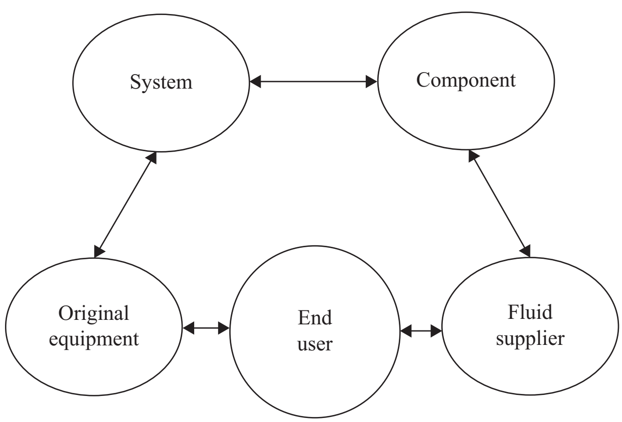

Figure 12.1. The complete partnership.

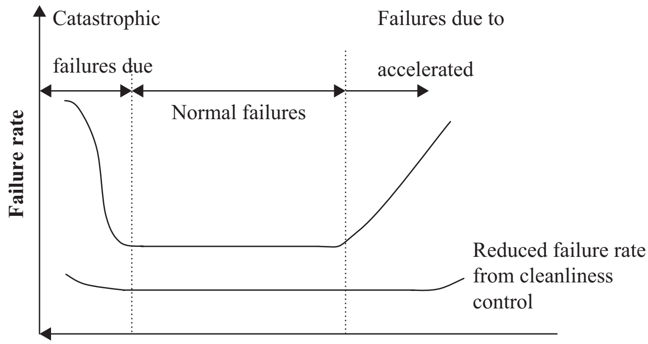

Figure 12.2. The bathtub life curve.

12.5.1 FAULT TREE ANALYSIS

Fault trees are drawn to show possible causes of major malfunctions of the system, and they usually deal only with those failures that are considered to be most likely to occur, events of low probability being excluded.

This approach can for example be applied to the actuator circuit shown in Figure 12.3 for considering all the possible failures that will prevent movement of the actuator. Most of the causes of this failure are single events, which would include the following:

- Directional control valve (DCV) not opening **OR**
- No pump flow (e.g., drive motor failed, inlet suction filter blocked) **OR**
- Relief valve failed to open (e.g., broken spring) **OR**
- Load force greater than that available from the actuator **OR**
- Insufficient fluid in the reservoir

Some faults require double failures to occur such as blockage of the return line filter **AND** the bypass valve jammed closed. This is normally an extremely unlikely situation, but in some systems where safety interlocks are used, these types of failures might need to be considered. A typical example is the circuits used for the control of hydraulic presses.

Thus a fault tree can be drawn by following the circuit from the component associated with the top event along the lines that lead either to the supply (pressure) or the return (tank); the top event itself may be linked to more basic faults. The event statements are linked through logic gates. OR gates require only one input to be available before the output event occurs.

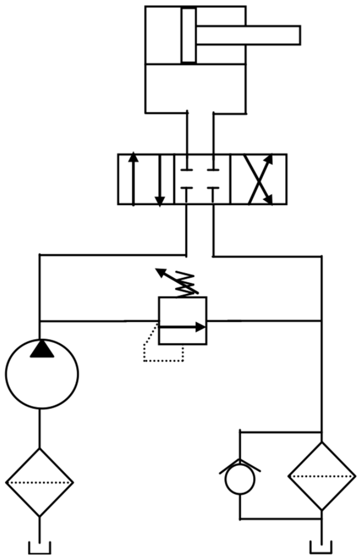

Figure 12.3. Valve actuator system.

AND gates, on the other hand, require both input events to have occurred before the output event can happen.

In the simplest circuits, only OR gates are required, linking alternative fault events; for example, there may be no flow from a directional control valve due to no input flow, OR failure of the pilot signal selecting the valve open position, OR valve jammed shut. In circuits with inbuilt redundancy, AND gates are also required; for example, there is no flow in a specified line when there is no flow from the main pump supply AND no flow from the auxiliary pump supply.

12.5.2 FAILURE MODES EFFECTS ANALYSIS

The fault tree analysis works from the top down in that a key malfunction is selected first and failures that can cause this are then established. The FMEA method works from the bottom up in that it determines the effect on the system of every failure mode of every component. This is a long and tedious procedure that is very suited to computerized methods.

An advantage that this method has over the FTA is that all failures are identified, some of which may have been missed in the FTA. However, identifying the relative importance of the failures is not part of the process, and to reduce the amount of work involved the Pareto method is often used, wherein only major malfunctions are considered, as in the FTA.

When carrying out these analyses, the type of fluid, the effects of the environment, and the possibility of operator error must also be included.

If failure probability data is available—usually in terms of mean time between failures (MTBF)—the reliability of the system can be established. However, such data for commercially available components often does not exist, whereas this approach can probably be applied to military projects and military/civil aircraft, where it is necessary to perform a quantitative analysis of the system reliability.

12.6 STANDARDS

Some relevant standards for hydraulic systems and components are listed in the Appendix to the book. The subject of standards has been referred to in the Summary and Step 3 in Section 12.2, which give a background to the Safety Standard ISO 13849. To meet the requirements of the safety standard, it is necessary for an estimate of the mean time to dangerous failure (MTTFd) to be carried out.

ISO standard 10771-1 is concerned with the testing of components so as to establish the number of cycles that are achieved without causing a fatigue failure of the component at a given maximum pressure. ISO standard 10771-2 TR is a technical report that presents a method of establishing a pressure rating for the component based on the fatigue test result. Thus, if this information is available, some estimate can be made of the expected life in a given application.

It is necessary that all systems are safe, and for this they have to comply with the relevant legislation. In Europe, systems have to comply with the Essential Health and Safety Requirements (EHSRs) of the Machinery Directive. A system can be seen to comply with the EHSRs if it can be shown that it has been designed and constructed to a relevant standard such as ISO 4413 (Appendix 15). However, some machinery which is deemed to be particularly dangerous, referred to as Annex IV machines (e.g., man lifts, presses, saws, plastics machinery, etc.), has to either comply with more rigorous (Type C) standards or be independently approved by a notified body. This generally applies to any machine whereby the consequences of an unexpected breakdown are considered to be dangerous.

The control of contamination is of particular importance because component and system failures are frequently caused by contamination in the fluid where proper contamination control and fluid monitoring have not been used. Hence there are three ISO standards and a guidelines document on this subject in the Appendix. Clearly, the contamination level could be severely influenced by the type of environment of the application.

For example, mining machinery and excavators will be more exposed to particles in the air than machinery in a closed industrial application. The designer therefore has to decide where the filters are to be fitted into the circuit, a subject which is discussed in Section 7.9, and the filter rating is discussed in Section 6.3.

For the design of mountings for cylinders and pumps/motors, there are standards in the Appendix that specify the dimensions for a range of component sizes. There are also standards in the Appendix that relate to the general performance testing of components.

As described in Section 3.3.1, seals are of considerable importance, particularly with regard to hydraulic cylinders, and the guideline BFPA/P112:2013 (Appendix 6) provides information on the types of seals that are available and the different types of fluids that can be used with the range of seal materials.

There is global legislation for the manufacture and use of accumulators, which requires manufacturers to comply with the "Essential Safety Requirements" within the directive and the harmonized standard BS EN 14359 (Appendix 9). This may require an accumulator and its associated controls to be CE-marked, which represents conformance with the Pressure Equipment Directive 97/23/EC.

Any component that is defined by the Machinery Directive as being a "safety component" also has to have a CE mark. One definition of such a component is that if the system would be functional without the component its only purpose is therefore to provide protection in the event of a failure when it will therefore be regarded as a 'safety component'.

The designer needs to be aware of these various standards and determine which, if any, relate to the application, the environment that it will be working in, and the life of the more critical components.

12.7 GENERAL

Systems management involves some or all of the activities that have been discussed, which approach is used depends on the requirements of the machine to which it is being applied. It may be considered on the basis of cost and/or safety to use condition-monitoring techniques for managing the operation of the machine.

The following parameters are those that can be measured and can be used to monitor the system for faults:

- Pressure
- Flow

- Temperature
- Torque
- Energy consumption
- Contamination
- Vibration
- Acoustic emission
- Noise
- Speed
- Position (valve, actuator, etc.)

Thus, for example, using this range of parameters in the wind turbine hydrostatic power transmission, described in Section 7.7.6 condition-monitoring purposes would help to detect if there is a problem in the hydraulic motor, by comparing the pressure drop across the motor and the measured torque. This approach could be utilized for other major components in the system.

The most appropriate monitoring is determined by undertaking a careful examination of the failure modes and the available measurement methods. The priority methods would be those that provide information on the most likely failures, and then, depending on the importance of the system, others can be incorporated if necessary.

Appendix

RELEVANT STANDARDS AND GUIDELINES FOR MANUFACTURERS AND USERS OF HYDRAULIC COMPONENTS AND SYSTEMS.

British Fluid Power Association (BFPA) address: Cheriton House, Cromwell Park, Chipping Norton, Oxfordshire, OX7 5SR, UK. www.bfpa.co.uk

BSI address: British Standards Institution, 389 Chiswick High Road, London, W4 4AL

ISO address: ISO Central Secretariat 1, ch. de la Voie-Creuse, P.O. Box 56, CH-1211 Geneva 20

The BFPA Engineer's Data book contains a comprehensive list of International standards (ISO), British standards (BS), and European standards (EN) that are relevant to hydraulic components and systems. The book can be purchased from the BFPA address as above.

ISO; EN AND BS STANDARDS AND BFPA GUIDELINES

Pump and Motor Performance Testing

1. ISO 4392-1, 2, 3:2002 HFP—Determination of characteristics of motors—Part 1: At constant low speed and constant pressure, Part 2: Start ability, Part 3: At constant flow
2. ISO 4409:2007 HFP—Positive displacement pumps, motors, and integral transmissions—Methods of testing and presenting basic steady state performance

Mounting Dimensions for Hydraulic Cylinders

3. ISO 6020-1 Mounting dimensions single rod 160 bar—Medium series

4. ISO 6020-2 Mounting dimensions single rod 160 bar—Compact series
5. BS 3018-2 Metric series pump and motor flanges

Seals

6. BFPA/P112:2013 "Guidelines on seals for fluid power applications including care and handling" British Fluid Power Association

Valve testing

7. ISO 6403:1988 HFP—Valves for controlling flow and pressure—Test methods

Accumulators

8. BFPA publication P54 entitled "Guidelines to the Pressure Systems Safety Regulations 2000 and their application to gas—Loaded accumulators"
9. BS EN 14359 Unfired pressure vessels—Gas loaded accumulators for fluid power applications

Contamination Control and Filtration

10. ISO 16889 Hydraulic fluid power—Filters—Multi-pass method for evaluating filtration performance
11. BFPA P5 Guidelines to contamination control in hydraulic fluid power systems
12. ISO TS 12669 Hydraulic fluid power—Guidelines for determining the required cleanliness level (RCL) of a system, (in draft)
13. ISO TR 15640 Hydraulic fluid power—Contamination control—General principles and guidelines for selection and application of hydraulic filters

System Safety

14. ISO 13849—Safety related parts of control systems

15. ISO 4413—General rules and safety requirements for systems and their components

Reliability Testing and Pressure Rating

16. ISO/CD 10771-1 HFP—Fatigue pressure testing of metal pressure containing envelopes—Part 1: Test method
17. ISO 10771-2 TR Hydraulic fluid power—Fatigue pressure testing of metal pressure containing envelopes—Part 2: Rating method

USEFUL WEBSITES

http://www.atos.com/espanol/technical_tables/espanol/P007.pdf
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Fluid power systems are manufactured by many organizations for a very wide range of applications, embodying different arrangements of components to fulfill a given task. Hydraulic components are manufactured to provide the control functions required for the operation of a wide range of systems and applications.

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- A resume of the flow processes that occur in hydraulic components.
- A review of the modeling process for the efficiency of pumps and motors.

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Peter Chapple received a BSc in mechanical engineering from Bristol University, 1960, an MSc from Princeton University in 1962, and a PhD from Bath University Fluid Power Centre in 1991. From 1960 he was a systems engineer on the Harrier aircraft development and later he held the position of technical director at Staffa Motors. As part-time professor in fluid power in Norway he was involved in tests on two wind turbines using hydrostatic systems to transfer the turbine power. He is a convener in ISO and also chairman of a BFPA technical committee.

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