1992

A pulse-width-modulated flow-control valve

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TITLE:

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DATE: May 31, 1992

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BY
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A Thesis
Presented to the Graduate Committee
of Lehigh University
in Candidacy for the Degree of
Master of Science
in
Mechanical Engineering

Lehigh University
June 1992

Certificate of Approval

Approved and recommended for acceptance as a thesis in partial fulfillment of the requirements for the degree of Master of Science in Mechanical Engineering

5/12/92
date

Professor in charge

Department Chairman

TO MY PARENTS

Acknowledgments

Acknowledgments should be written in large letters to Professor M. Abo-Zeid who is the reason which made this opportunity possible for me.

I would like to extend my appreciation and thanks to Professor F. T. Brown for the quality of his supervision and expertise. To say that I would not have been able to undertake this work without his help is an understatement.

I would also like to thank Professor D. O. Rockwell for his generous support during hard times.

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Abstract

Standard speed control fluid power systems are more or less dissipative due to the inherent behavior of standard flow control valves. This dissipated energy becomes more significant at low load flow demand and most significant at zero load flow demand.

This thesis introduces a new flow control valve to save energy in fluid power flow control systems which use fixed-displacement pumps. A preliminary study of a valve system shows that controlling a fixed displacement pump by this valve would have it act like the much more expensive controlled-displacement pump. The system can be compared to basic bleed-off control of a fixed displacement pump.

The bond graph technique was implemented to facilitate modeling of the system. Computer simulations were carried out to predict the effect of different valve parameters on its performance to help optimizing the control system. Simulation results predict attractive pressure-flow characteristics and significant energy saving over virtually the entire operating region. The model used is fairly

accurate, although it does not address the vibrations problem.

A system similar to the one proposed in this thesis was designed and constructed to validate simulation results. Problems with oil contamination frustrated the collection of a reliable set of experimental results in the available time. The experimental results are expected to be in another publication.

Chapter 1

Introduction

Fluid power flow control systems which use fixed displacement pumps have to use inherently dissipative means to control their flows. Interest in energy efficiency has resulted in the use of load- sensing systems which substitute relatively expensive variable-displacement pumps for fixed-displacement pumps. The possibility of controlling a fixed- displacement pump to make it act more like a variable-displacement pump motivated this research represented herein.

This thesis introduces a newly designed valve, called a pulse-width-modulated flow control valve, to be incorporated into flow control systems which use fixed displacement pumps. The valve hydraulic resistance is, mechanically, periodically switched from virtually zero to virtually infinite. A modulation of the duration of these two values of the valve hydraulic resistance is imposed to achieve the required flow control.

In a proposed bleed-off flow control system, the unwanted portion of the pump flow is periodically bled to the tank under virtually zero pressure drop. Simulation results shows that using the pulse-width-modulated flow control valve instead of the usual bleed-off valve would significantly cut back on the energy losses in bleed-off flow control systems and have a fixed displacement pump act like a controlled-displacement pump.

The great reduction in the pump pressure around zero flow demand would, at least partially, compensate for the threat to working life resulting from the continuous fluctuations in the pump pressure.

Chapter 2 presents a background and fundamental concepts regarding the energy saving issue in flow control fluid power systems. It discusses briefly the standard fluid flow control systems from the energy point of view.

Chapter 3 presents a comprehensive analysis to a proposed system, a pulse-width-modulated bleed off flow control system. Effect of various valve parameters are investigated to help optimizing the control system.

Chapter 4 presents a design of an experimental model similar to the system analyzed in chapter 3. The system is ready for testing apart from possibly continuing problems with fluid contamination which jammed the rotor and the check valve.

Chapter 2

Background and fundamental concepts

2.1 Classical flow control fluid power systems

There are a variety of classical flow control fluid power systems. The more important among them are summarized to establish a figure of merit for the sake of comparison.

2-1-1 Meter-in and meter-out systems

Meter-in and meter-out flow control systems are alike except that the flow control valve is located upstream of the load in the meter-in systems, while it's located downstream of the load in the meter-out systems to help overrunning loads. Fig. 2.1 shows the concept of meter-in and meter-out flow control systems. The pump pressure and flow are both constant, so the pump power also is constant, and the relief valve is always open to bypass the unwanted portion of the pump flow. Assuming a square law orifice flow control valve, the load flow is

$$Q_L = C_d A \sqrt{\frac{2(P_P - P_L)}{\rho}} \tag{2.1}$$

where C_d is the orifice flow coefficient(taken to be .7), A is the valve effective area, P_P is the pump pressure, P_L is the load pressure and ρ is the oil density. The energy efficiency of the system is defined as the ratio between the load power and the pump output power:

$$\eta = \frac{P_L Q_L}{P_P Q_P}. \tag{2.2}$$

Equations 2.1 and 2.2 give the pressure-flow characteristics and efficiency map shown in Fig. 2.2. The map reveals a poor efficiency whenever the load power is low. This is because the pump power is constant.

2-1-2 Bleed-off systems

Fig. 2.3 shows the concept of bleed-off flow control systems. The efficiency of the bleed-off systems are higher than that of the meter-in since the pump pressure is equal to the load pressure nevertheless, the efficiency is poor whenever the load flow is small. The load flow rate is

$$Q_L = Q_P - Q_{fcv}, \tag{2.3}$$

where Q_P is the pump flow rate. Q_{fcv} is the oil flow rate through the flow control valve:

$$Q_{fcv} = C_d A \sqrt{\frac{2P_P}{\rho}}. \tag{2.4}$$

The efficiency is equal to the ratio between the load flow and the pump flow:

$$\eta = \frac{Q_L}{Q_P}. \tag{2.5}$$

These equations are used to draw the pressure-flow characteristics and efficiency map, shown in Fig. 2.4, assuming a constant pump flow rate.

Meter-in and meter-out flow control systems have the advantage of providing positive flow control because the flow control valve directly controls the load flow, but they are inherently inefficient because the pump works against the sum of the load and the flow control valve resistances. On the other hand, bleed-off flow control systems lack such a direct control because the flow control valve in this case controls the rate of bleed flow.unless the pump characteristic is ideal or is known accurately, or unless a feedback control is implemented, any deviation from the assumed pump characteristic will produce an error in the load flow rate.

2-1-3 Pressure-centering pump with meter-in flow control

In these systems the pump adjusts its flow automatically to match the system demand so as to keep the operating pressure constant. Fig. 2.5 shows the concept of pressure-centering pump used in a meter-in system.

Equation (2.1) applies. Since the entire pump flow reaches the load, equation (2.2) becomes

$$\eta = \frac{P_L}{P_P}. \tag{2.6}$$

These equations give the pressure-flow characteristics and efficiency map shown in Fig. 2.6. Despite the higher cost of this system , its efficiency is comparable to that of the bleed-off system, although the details are different.

2.2 Can the efficiency of a flow control system be improved despite using a fixed displacement pump?.

To answer that question, the energy wasted in the flow control systems should be examined. This energy is dissipated in the flow control valves, that includes the relief valves. The energy dissipated in a valve is

$$E_{dfcv}=\int R_{fcv}Q^2{}_{fcv}dt, \tag{2.7}$$

where R_{fcv} is the valve hydraulic resistance. Equation (2.7) says that there are two parameters constituting the energy loss: the oil flow rate through the valve and the valve hydraulic resistance to the flow.

In systems using fixed displacement pumps, the pump capacity is generally larger than the load demand. Flow to the load and/or the bleed therefore must be metered by valves. From equation (2.7), however, the only way to achieve small dissipation in a valve is to have either small flow or small resistance (which means small pressure drop); but this clearly is impossible in the steady-state. The question now becomes: is there some way of using unsteady characteristics, that is of cycling the resistance of the valve?. An answer is the pulse-width-modulated flow control valve.

2.3 A pulse-width-modulated flow control valve (PWMFCV)

A schematic drawing of the porting members of the pulse-width-modulated flow control valve developed in this thesis is given in Fig. 2.7. The fluid enters a hole drilled into the end of a shaft. This shaft is rotated from an external source; in practice it could be attached to the pump shaft. The fluid passes out through two radial diamond-shaped holes, except when prevented from doing so by a non-rotating sleeve (called the modulation sleeve). The sleeve has four helical porting surfaces with a $\pm$ 45° helix angles to match the four angles of the edges of the diamond-shaped holes in the shaft. The holes are large enough to pass all the pump flow with a negligible pressure drop.

As the shaft rotates the resistance to the flow is switched back and forth from virtually zero to virtually infinite so that the fluid is either passed through a negligible resistance or is shut off. The axial position of the modulation sleeve modulates the periods of times for the maximum and the minimum hydraulic resistances. This position could be controlled from a feedback signal to achieve a desired flow through the valve.

Obviously, the abruptness with which the valve switches between the two resistances and the valve dynamic performance are critical factors in the success of this device; the closer the valve hydraulic resistance resembles a square wave and the smaller the compliance of the fluid under compression, the larger the energy saving. These matters are discussed in more detail later on.

A pressure fluctuation upstream of the valve is inherent to its energy-saving behavior. This pressure fluctuation behavior precludes the definition of a meaningful pressure-flow valve characteristic. Therefore to investigate the merit of the PWMFCV as an energy saving-flow control valve it has to be incorporated into a system. More than one such proposed system is outlined in this thesis; one of these is called the pulse-width-modulated bleed-off flow control system.

2.4 Pulse-width-modulated bleed-off flow control system

This system, conceptualized in Fig. 2.8, is a bleed-off flow control system in the sense that it controls the bleed flow but, it replaces the traditional dissipative flow control valve with a PWMFCV and it uses a check valve to prevent back-flow from the load to the tank when the pump pressure is smaller than the load pressure.

The result is a significant energy saving over virtually the entire operating range. On the other hand , the fluctuation in the pump pressure which is inherent to the saving of energy in any system which uses the PWMFCV also produces undesirable vibration and possibly wear to the pump. The accompanying fluctuation in the load flow however, can be attenuated by an accumulator. A compromise choice of the accumulator size should smooth the load speed and keep a reasonable hydraulic natural frequency at the same time.

This system is studied analytically and experimentally in the following chapters to get a reliable judgement about the PWMFCV. However, for now, Fig. 2.9 shows a typical time simulation history for this system with the valve modulation sleeve at two different positions. As the figure indicates, increasing the modulation factor (X) results in increasing the duration of the minimum hydraulic resistance (τ_{min}) and decreasing the duration of the maximum hydraulic resistance (τ_{max}), leading to a decrease of the mean load flow rate.

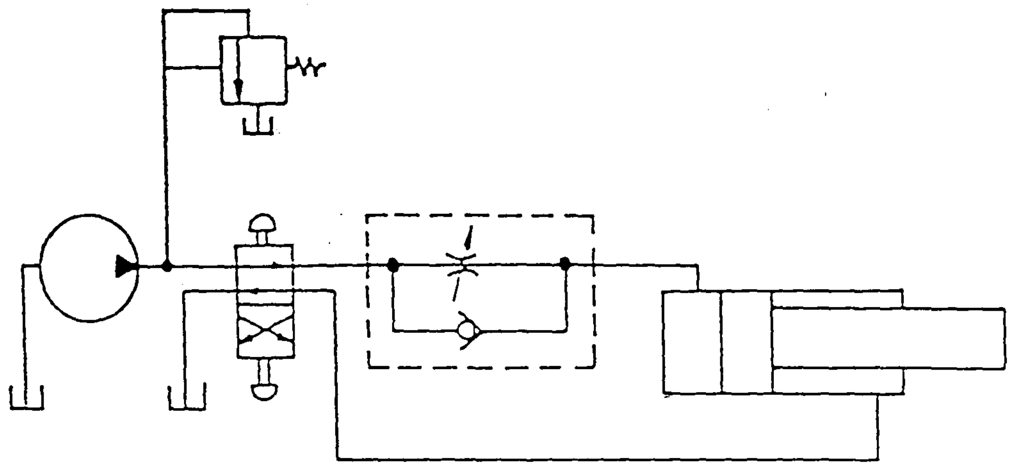


Fig. 2.1a Meter-In Control of a
Fixed-Displacement Pump

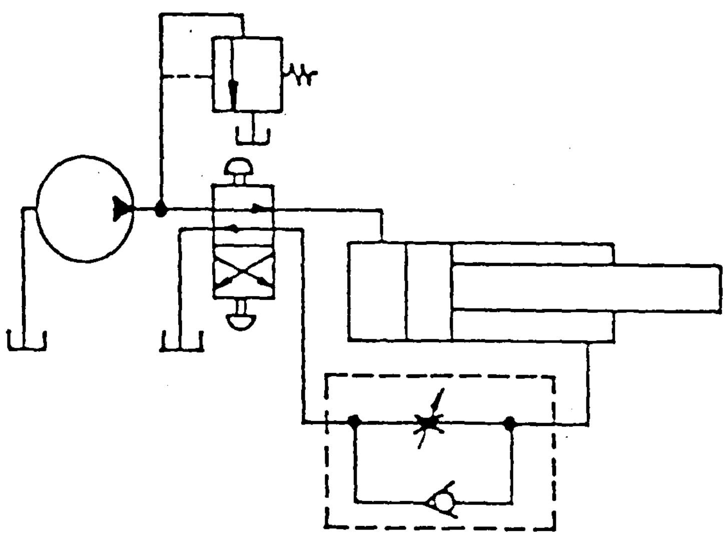


Fig. 2.1b Meter-Out Control of a
Fixed-Displacement Pump

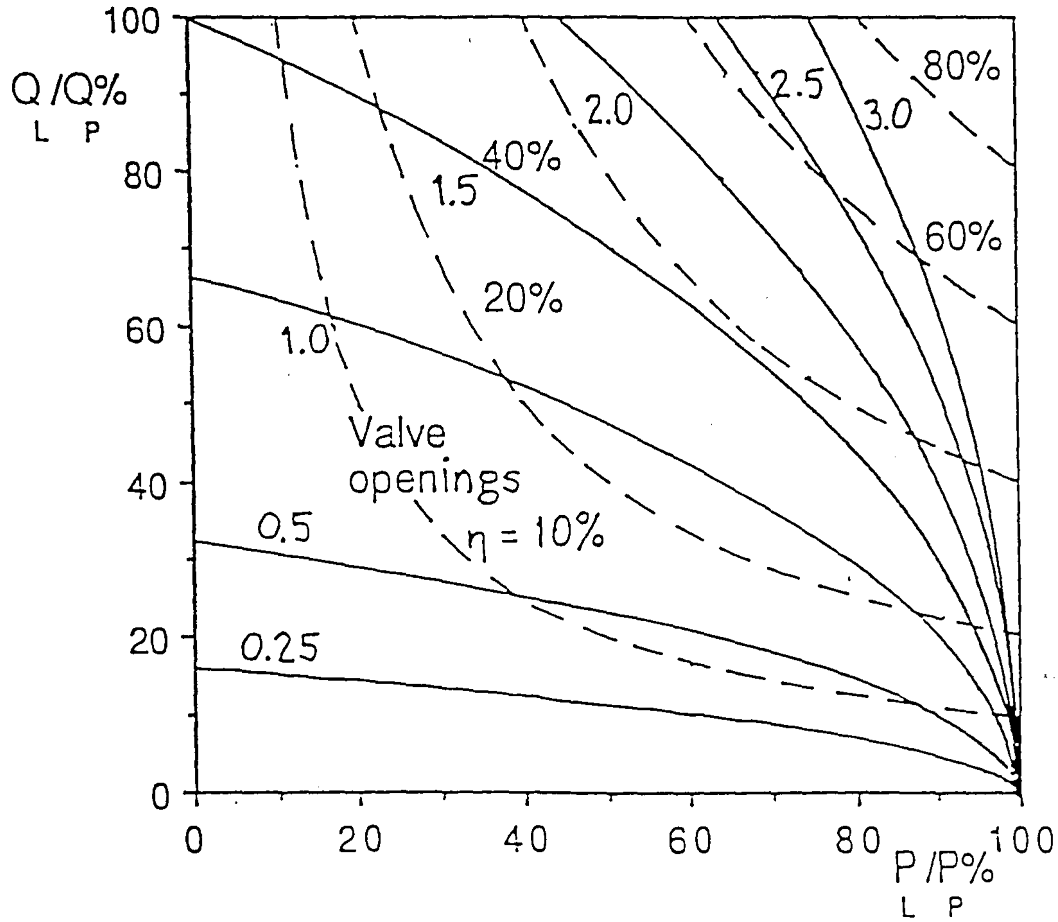


Fig. 2.2a Ideal Meter-In-Control of a Fixed Displacement Pump

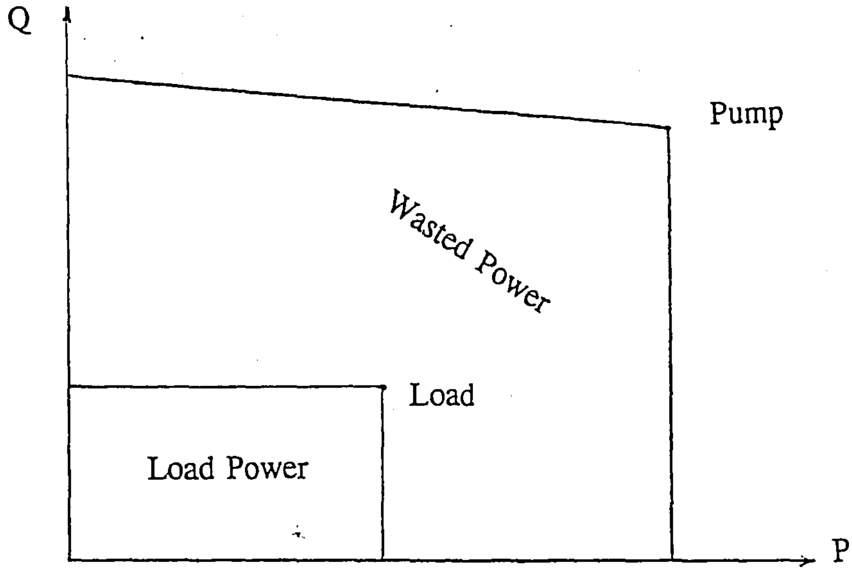


Fig. 2.2b Power Map for a
Meter-In System

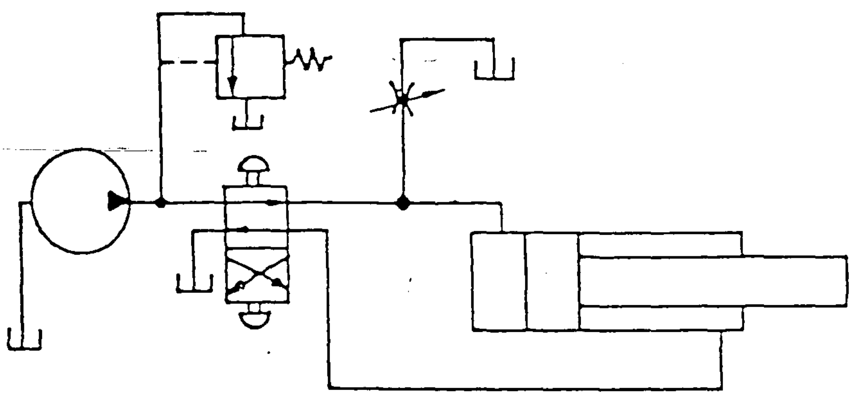


Fig. 2.3 Concept of Bleed-Off Control
of a Fixed-displacement Pump

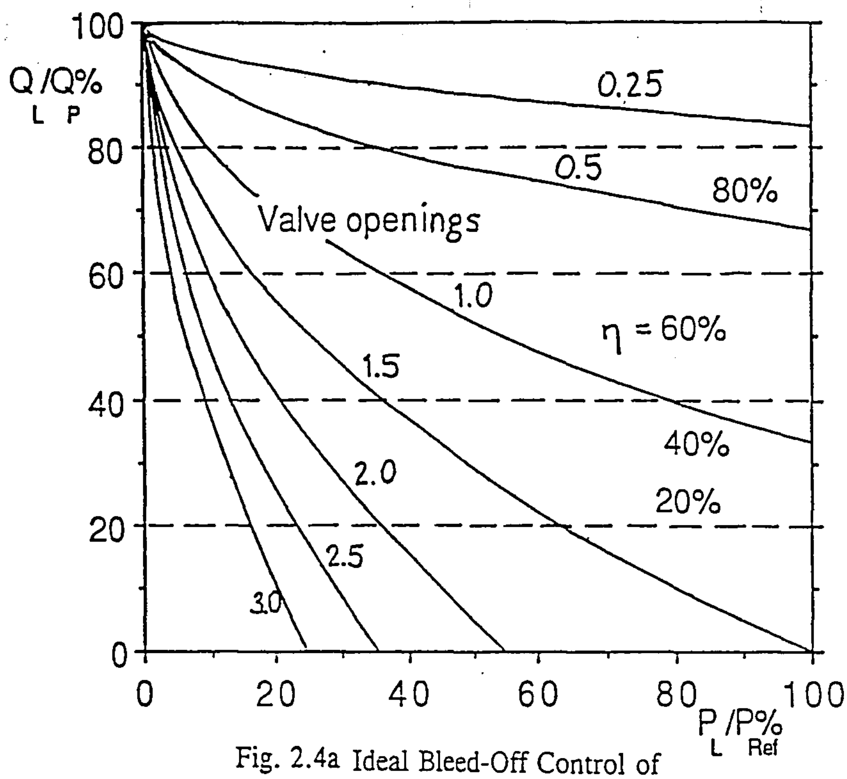


Fig. 2.4a Ideal Bleed-Off Control of a Fixed Displacement Pump

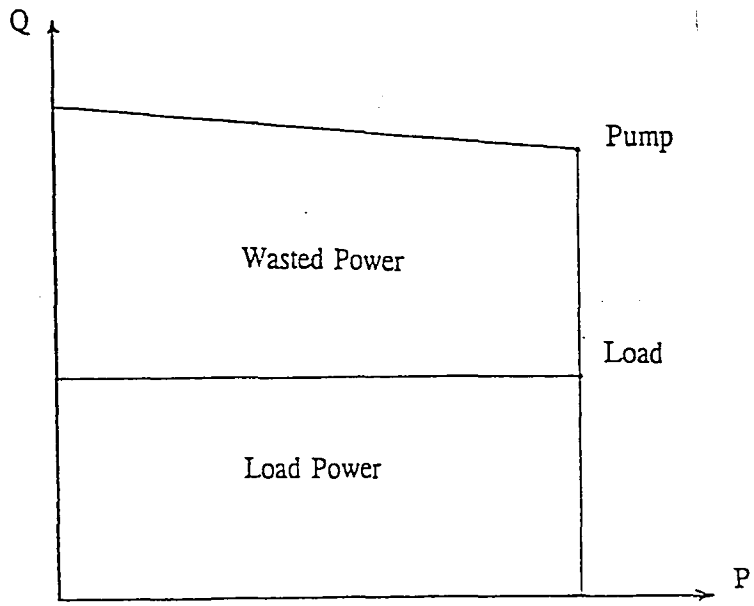


Fig. 2.4b Power Map for a Bleed-Off System

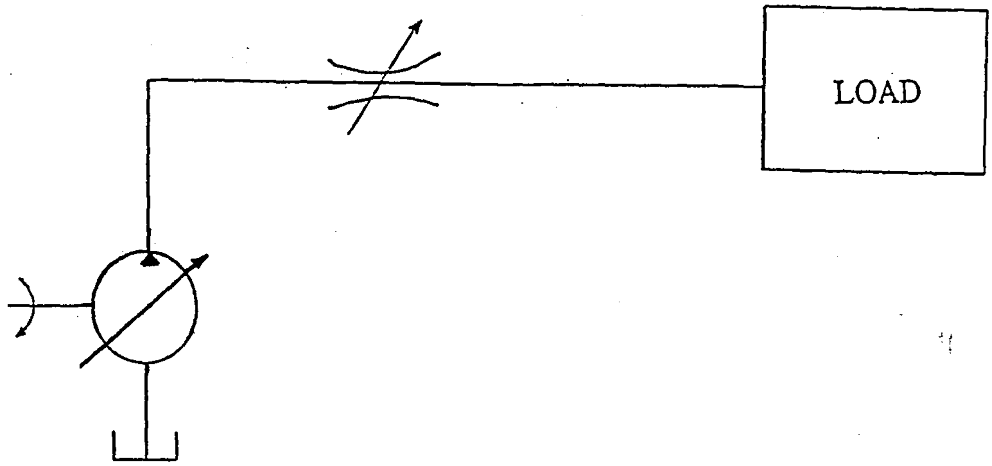


Fig. 2.5 Concept of Pressure-Centering Pump
used in a Meter-In System

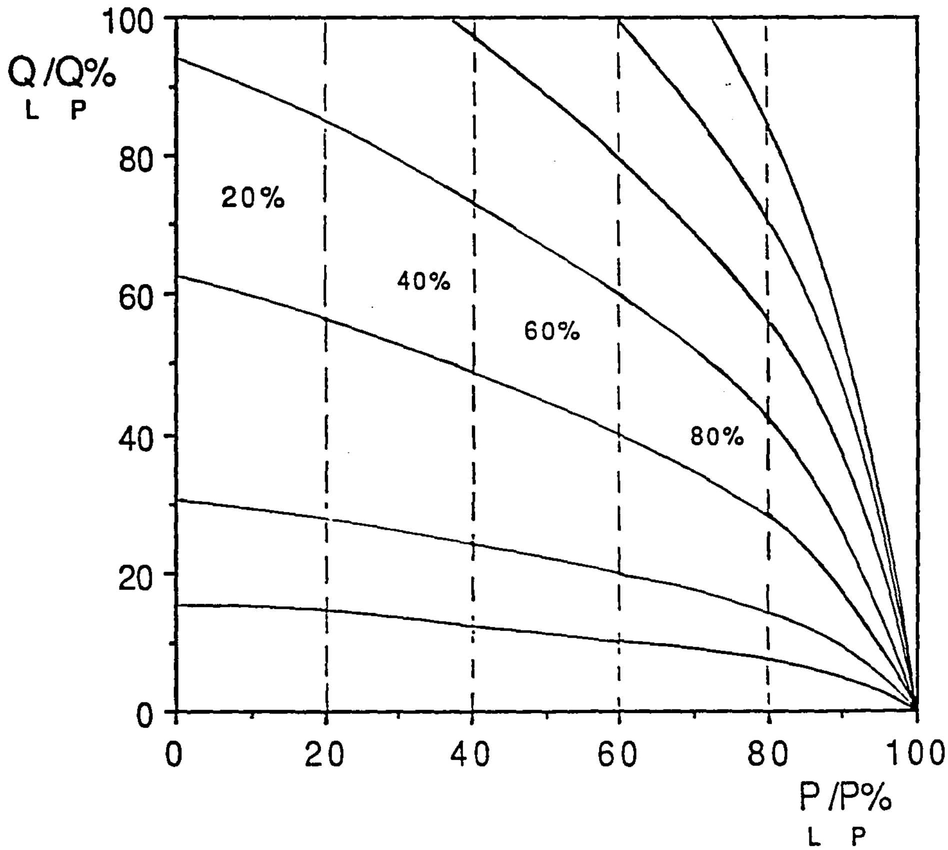


Fig. 2.6a Ideal Meter-In Control of a
Pressure-Centering Pump

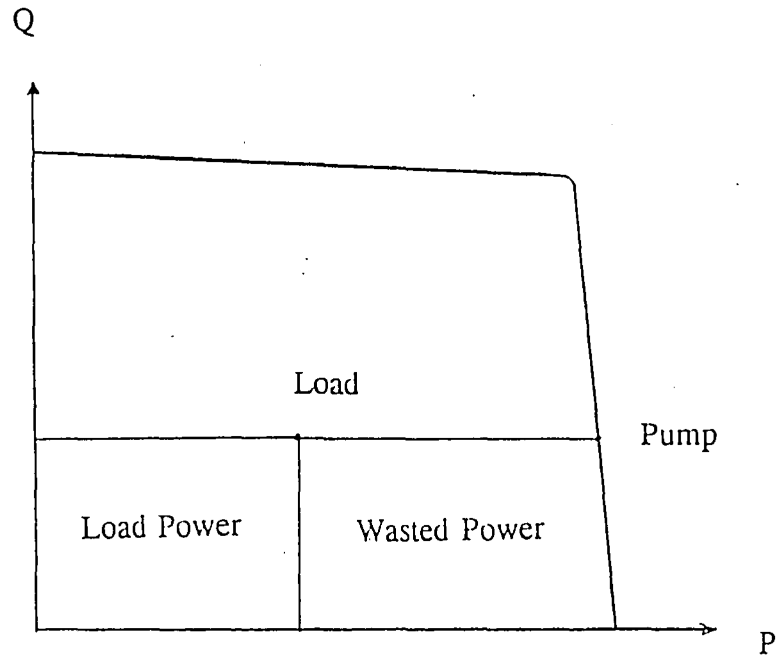


Fig. 2.6b Power Map for a Pressure-Centering Pump Meter-In System

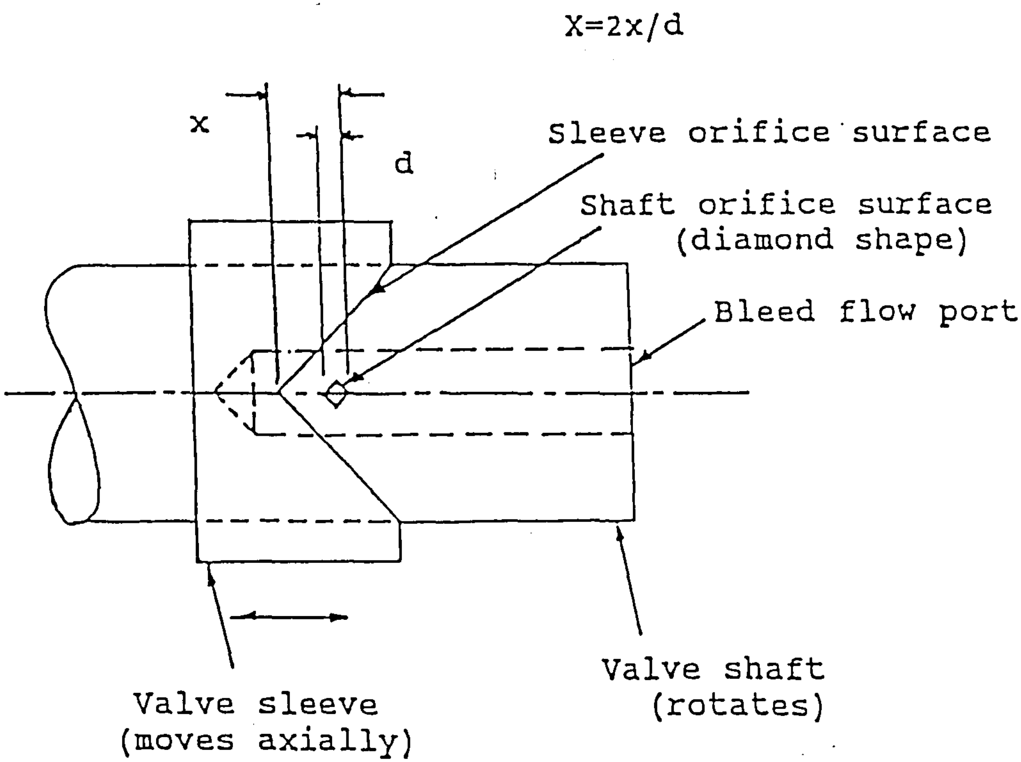


Fig. 2.7 Schematic Drawing of the Porting Members

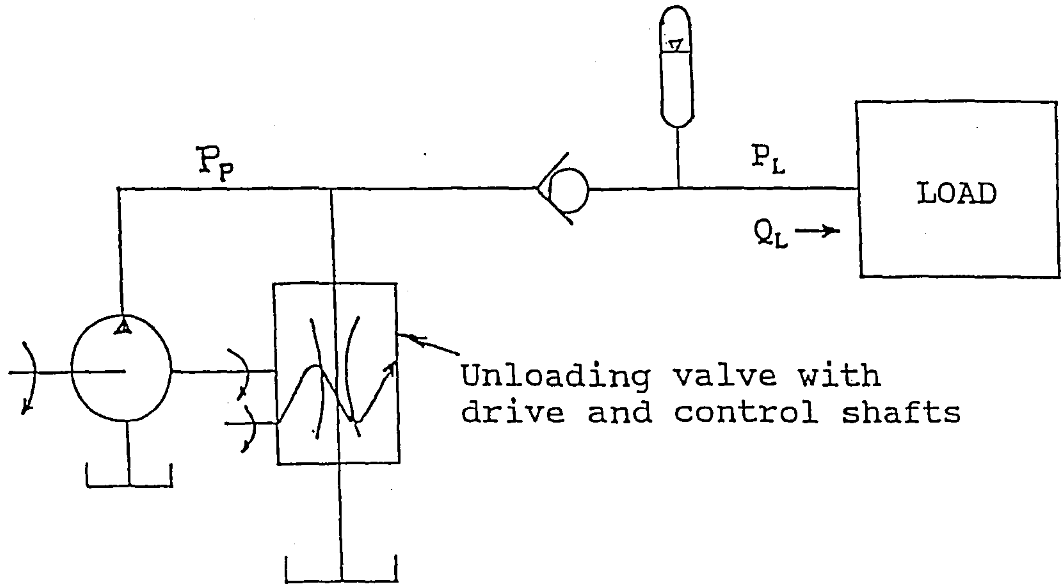


Fig. 2.8 Concept of a Flow Control System Based on a Pulse-Width-Modulated Unloading Valve

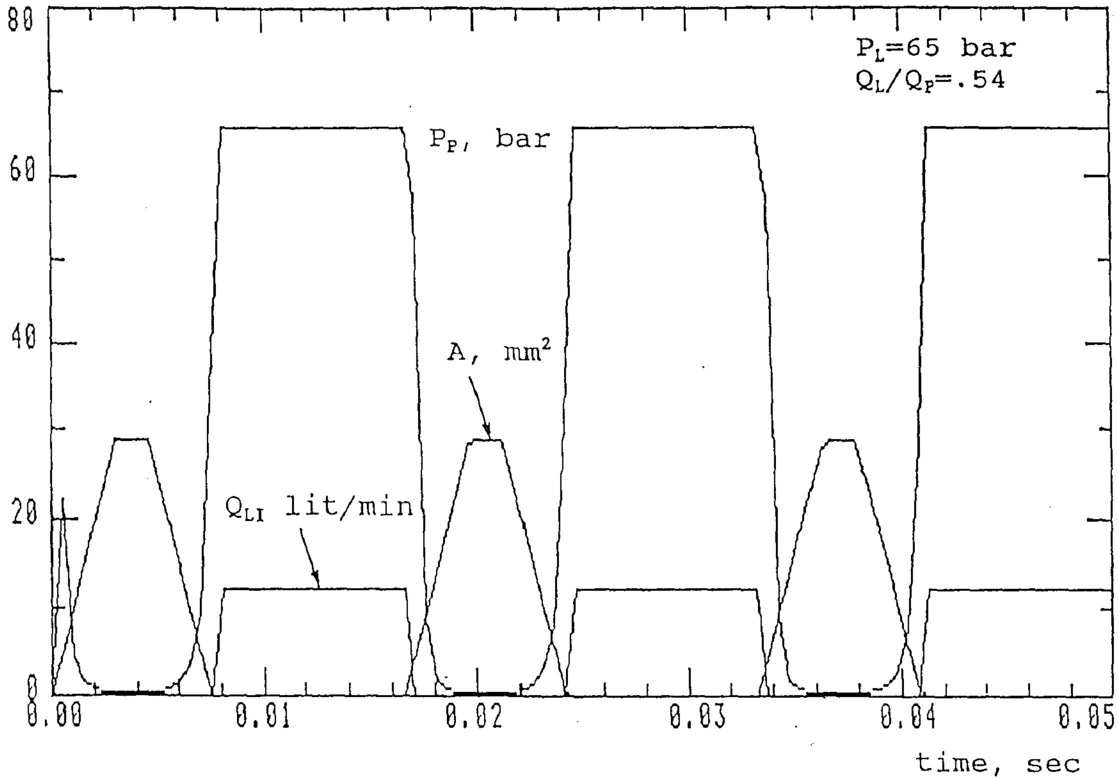


Fig. 2.9a Typical simulation results, X=2.5

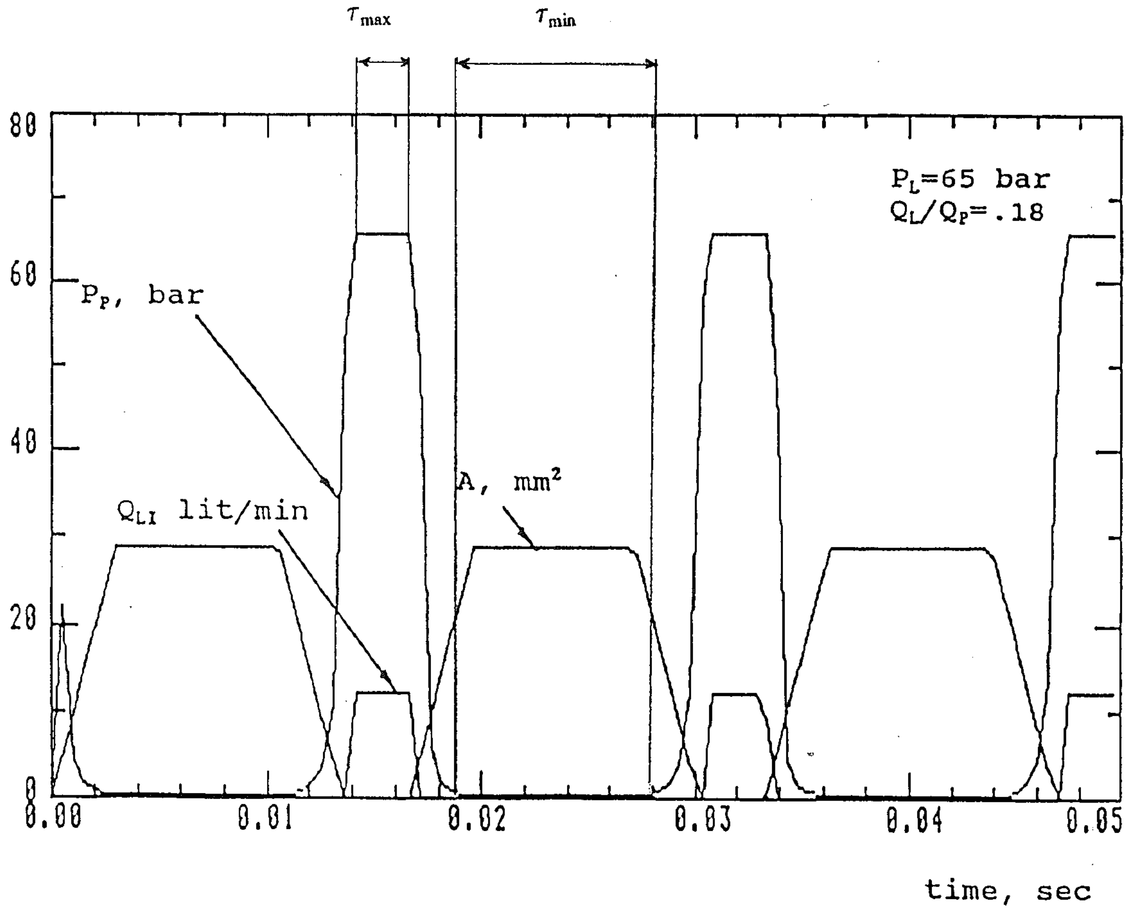


Fig. 2.9b Typical simulation results, X=4.5

Chapter 3

Modeling and simulation of a pulse-width -modulated bleed-off flow control system

A mathematical model is developed to carry out a computer simulation for the system. The bond graph technique is implemented to facilitate modeling of the system. A variable-step fourth-order Runge-Kuuta technique is used to integrate the differential equation.

The system under study (shown in Fig. 2.8) assumes a 19.05 mm (3/4 inch) valve shaft diameter D and 5.81 mm diamond side length W. Specifying the shaft diameter means specifying the travelling stroke of the modulation sleeve (L)

$$L=\frac{\pi D\tan\phi}{4} \tag{3.1}$$

where ϕ is the angle of the modulation sleeve helical shaped-edges which happens to be 45°. Justifications for the choice of those parameters will be given later.

3.1 Modeling the system

A pump of 6.6 cc displacement per revolution is used. The flow from the pump Q_P is taken to be constant. The pressure in the pump body and valve chamber P_P is assumed to be uniform, which implies that this region is relatively compact. The volume of this region (V) is taken to be equal to the pump displacement per revolution plus the volume of oil inside the tubes which are connecting the pump to both the check valve and the PWMFCV. The tubes are taken to be 12.7 mm (1/2 inch) size and 12 inches long which means that V is about 26 cc. The effective bulk modules of elasticity (β) is taken to be 1.5 X 10^9 Pa so, the compliance of this region is:

$$C = V/\beta = 1.73 \text{ mm}^3/\text{bar}.$$

The resistance of the orifice is assumed to be the same as for steady-flow of an inviscid incompressible fluid through an ideal orifice with time varying area A(t) and a flow coefficient C_d. The resistance of the check valve $R_C(t)$ is taken to be infinite when the driving pressure drop is negative, and a constant R_{C0} when the driving pressure drop is positive. These assumptions can be improved to give small corrections.

The ratio 2x/d defines the modulation factor X, where $d=W\sqrt{2}$. With the aid of Fig. 3.1, the following set of equations for A(t) can be recognized:

For $X \geq 2$

$$0 \leq t \leq \frac{W\sqrt{2}}{v} \quad , \quad A=2\left(W\frac{vt}{\sqrt{2}}\right) \tag{3.2}$$

$$\frac{W\sqrt{2}}{v} \leq t \leq \frac{2x-W\sqrt{2}}{v} \quad , \quad A=2W^2 \tag{3.3}$$

$$\frac{2x-W\sqrt{2}}{v} \leq t \leq \frac{2x-W\sqrt{2}}{v}+\frac{W\sqrt{2}}{v} \quad , \quad A=2W^2-2\left(W\frac{vt_{r1}}{\sqrt{2}}\right) \; , \tag{3.4}$$

$$t_{r1}=t-\frac{2x-W\sqrt{2}}{v}$$

$$\frac{2x}{v} \leq t \leq T \quad , \quad A=0 \tag{3.5}$$

where t is the time, v is the valve shaft linear velocity and T is the period of the cycle. For $1 \le X \le 2$,

$$0 \le t \le \frac{2x - W\sqrt{2}}{v} \quad , \quad A = 2\left(W\frac{vt}{\sqrt{2}}\right) \tag{3.6}$$

$$\frac{2x - W\sqrt{2}}{v} \le t \le \frac{W\sqrt{2}}{v} \quad , \quad A = 2\left[\left(W - \frac{vt_{r1}}{\sqrt{2}}\right)\left(\left(x - \frac{W}{\sqrt{2}}\right)\sqrt{2} + \frac{vt_{r1}}{\sqrt{2}}\right)\right] \tag{3.7}$$

$$\frac{W\sqrt{2}}{v} \le t \le 2\frac{W\sqrt{2}}{v} \quad , \quad A = 2\left[W\left(\left(x - \frac{W}{\sqrt{2}}\right)\sqrt{2}\right) - W\frac{vt_{r2}}{\sqrt{2}}\right] \ , \tag{3.8}$$

$$t_{r2} = t - \frac{W\sqrt{2}}{v}$$

$$\frac{W\sqrt{2}}{v} \le t \le T \quad , \quad A = 0 \ . \tag{3.9}$$

Finally, for $X \le 1$:

$$0 \le t \le \frac{2x}{v} \quad , \quad A = 2\left[\left(x\sqrt{2} - \frac{vt}{\sqrt{2}}\right)\left(\frac{vt}{\sqrt{2}}\right)\right] \tag{3.10}$$

$$\frac{2x}{v} \le t \le T \quad , \quad A=0 \ . \tag{3.11}$$

These equations can be simplified as follows.

For $X \ge 2$:

$$0 \le t \le \frac{W\sqrt{2}}{v} \quad , \quad A=\sqrt{2}Wvt \tag{3.12}$$

$$\frac{W\sqrt{2}}{v} \le t \le \frac{2x-W\sqrt{2}}{v} \quad , \quad A=2W^2 \tag{3.13}$$

$$\frac{2x-W\sqrt{2}}{v} \le t \le \frac{2x}{v} \quad , \quad A=\sqrt{2}W(2x-vt) \tag{3.14}$$

$$\frac{2x}{v} \le t \le T \quad , \quad A=0 \tag{3.15}$$

For $1 \le X \le 2$:

$$0 \le t \le \frac{2x-W\sqrt{2}}{v} \quad , \quad A=\sqrt{2}Wvt \tag{3.16}$$

$$\frac{2x-W\sqrt{2}}{v} \leq t \leq \frac{W\sqrt{2}}{v} \quad , \quad A=(2x-vt)\,vt \tag{3.17}$$

$$\frac{W\sqrt{2}}{v} \leq t \leq \frac{2x}{v} \quad , \quad A=\sqrt{2}W(2x-vt) \tag{3.18}$$

$$\frac{2x}{v} \leq t \leq T \quad , \quad A=0 \ . \tag{3.19}$$

Finally, for $X \leq 1$:

$$0 \leq t \leq \frac{2x}{v} \quad , \quad A=vt(2x-vt) \tag{3.20}$$

$$\frac{2x}{v} \leq t \leq T \quad , \quad A=0 \ . \tag{3.21}$$

A schematic of the simple model used and its bond graph are shown in Fig. 3.2. The model assumes that the pressure upstream of the load P_L is constant. In practice this could be achieved by using a suitable accumulator immediately upstream of the load. The system differential equation can, easily, be written by checking Fig. 3.2:

$$\frac{dV_C}{dt}=Q_P-c_dA(t)\sqrt{2p_P/\rho}-(P_P-P_L)/R_c(t) \tag{3.22}$$

where ρ is the oil density and V_C is the volumetric compression that the fluid of volume V experiences when its pressure is higher than zero for which $P_P=V_C/C$.

the efficiency η is

$$\eta=\frac{P_L\int Q_{LI}dt}{Q_P\int P_P dt} \tag{3.23}$$

where Q_{LI} is the instantaneous load flow.

3.2 Simulation Results

Simulations were carried out assuming the following approximation for the valve:

$C_d=0.7$

$R_{c0}=0.05$ bar/liter/min

$Q_P=12$ liter/min

V=26 cc

ρ=800 Kg/m^3

3.2-1 System characteristics and efficiency map

A first simulation series was run for a cycle period at 1/60 second, which corresponds to 1800 rpm with two cycles per revolution. Two typical cases are shown in Fig. 3.3. Equilibrium cycling is reached during the first cycle. Therefore all the time averaged results are based on the second and third cycles only.

Fig. 3.4 shows a normalized load flow versus a normalized sleeve position relation for different load pressures. The relation is linear over virtually the entire range. Cross plots of each relation, in addition to the efficiency related to every one of them, are given in Fig. 3.5. The efficiency of this system is clearly superior to those of the bleed-off and meter-in systems over virtually the entire region of operation. This improvement is most significant in the region with small load flow. Another observation is that each position of the sleeve virtually specifies the output flow, much like a variable displacement pump.

The radical variation in the load flow at very small load pressures, particularly for large modulation factors, is due to the square law relationship between the flow through the valve and the pressure drop. The rate of change of flow through a square law orifice, with respect to the pressure, tends to infinity as the pressure drop tends to zero. The reduction accompanying the decrease in the modulation factor is natural. As the modulation factor approaches zero it eliminates the effect of the PWMFCV on the bleed-off system and it drives the pulse-width-modulated bleed-off system towards the traditional bleed-off system; when $Q_L/Q_P = 1$, or the modulation factor X=0, both systems are the same.

3.2-2 Compliance effect

A second series of simulations was run to predict how sensitive the system efficiency is to changes in the compliance of the volume V. Simulations were carried out for a system with twice as much compliance as the original system; the result is shown in Fig. 3.6. Increasing the compliance can be understood to increase the system time constant, slowing down the switching of the pump pressure.

This in turn means more flow is bled to the tank with a pressure drop higher than the possible minimum. Fortunately, doubling the compliance of the volume under compression from the nominal expected volume does not hurt the efficiency much. Thus the system is not very sensitive to the compliance, the energy dissipated during the switching process is not large and the technique of switching the valve hydraulic resistance inherently saves energy. This conclusion could be verified from Fig. 3.7 which shows typical time simulation results for tow similar systems with different compliances

3.2-3 valve shaft speed effect

In a third series of simulations the relation between the shaft speed and the efficiency was investigated by holding the load pressure constant at 65 bar, while different shaft speeds each of several modulation factors were considered. Fig. 3.8 shows typical time simulation results for two similar systems with different shaft speeds and the summary of results is presented in Fig. 3.9. Not surprisingly, the high speed asymptote of the efficiency of the pulse-width-modulated bleed-off flow control system should approach that of the standard bleed-off flow control system. That is because if the shaft speed is infinitely fast, the period of the cycle is zero or in other words there is no cycling for the PWMFCV hydraulic resistance and so there is no difference between the standard bleed-off flow control system and the pulse-width-modulated bleed-off flow control system.

The best shaft speed involves a trade-off between the smoother load flow associated with high shaft speeds and the higher efficiency associated with low shaft speeds.

The reason why increasing the shaft speed decreases the efficiency can be viewed from two perspectives. First, increasing the shaft speed decreases the cycle period and consequently decreases the period of the minimum hydraulic resistance per cycle (τ_{min}). This means that the inefficient switching process for the PWMFCV hydraulic resistance has to be done more times for the same desired mean load flow. Second, like any other control system, this system has a time constant; at rapid shaft speeds the system will not have enough time to fully respond to the input signal.

The reduction in the amplitude of the load flow fluctuation accompanying an increase in the PWMFCV shaft speed results from the associated higher frequency for the noise signal exciting the low pass filter-system through the valve orifice .

3.2-4 Shaft diameter effect

The PWMFCV shaft diameter in the original proposed system is 19.05 mm (3/4 inch). This small size was motivated mainly by the expected need for a compact design. To evaluate the effect of the shaft diameter on the efficiency, simulations were carried out for a system similar to the original proposed system but with a shaft diameter of 38.1 mm (1.5 inch). Fig. 3.10 shows the result; Increasing the shaft diameter increase the efficiency of the system substantially for the low load flow region and slightly for the rest of the operating region. Again, this result makes sense because increasing the shaft diameter means increasing the speed with which the hydraulic resistance switches and slightly increasing the span of its minimum value per cycle (τ_{min}), both of which help the efficiency. The disadvantages of increasing the shaft diameter are first, a slowing down of the response to a feedback signal due to increased inertia of the modulation sleeve , and second, a less compact design.

3.3 Valve configuration

From the foregoing discussions, it is clear that fast opening and closing of the valve is of interest. There are many possible shapes for the orifices and porting surfaces. The author believes nevertheless, that the combination of diamond shaped orifices and 45° helical porting surfaces is the optimal choice for fast switching of the valve hydraulic resistance. This problem can be reduced to the well known optimization problem of minimizing the circumference of a known rectangular area; The solution is the square.

The small shaft diameter size was motivated by the expected need for a compact design. The area of the orifices was chosen big enough to bypass all the pump flow, taken to be 12 L / min, with an 0.3 bar pressure drop assuming an 0.7 orifice flow coefficient.

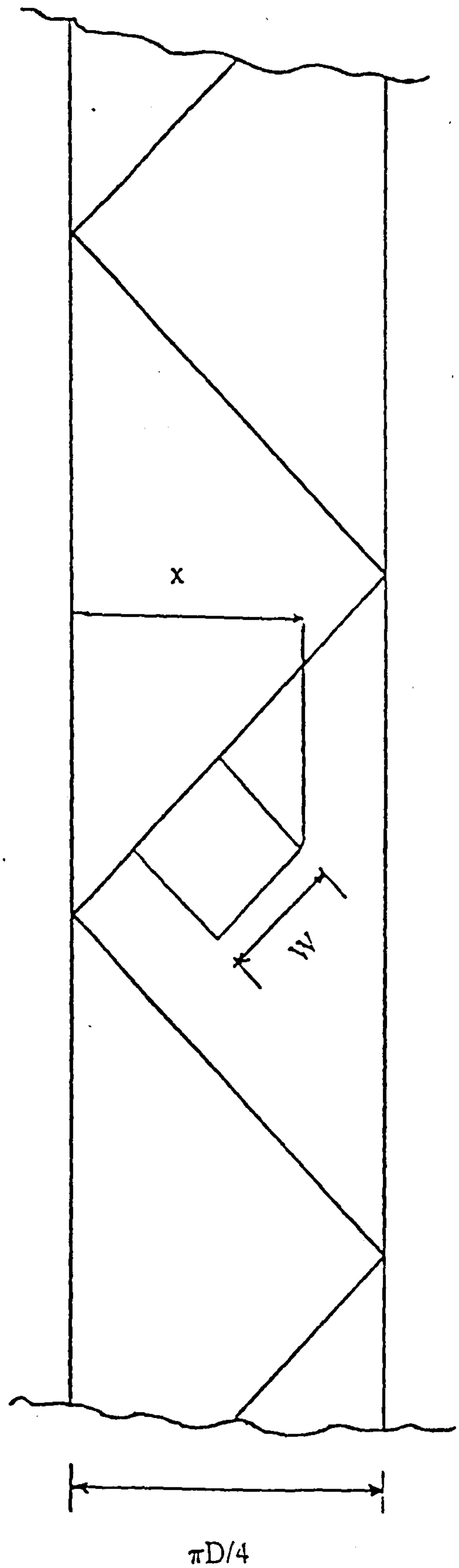


Fig. 3.1a Projection of the sleeve and orifice surfaces, X ⟩ 2

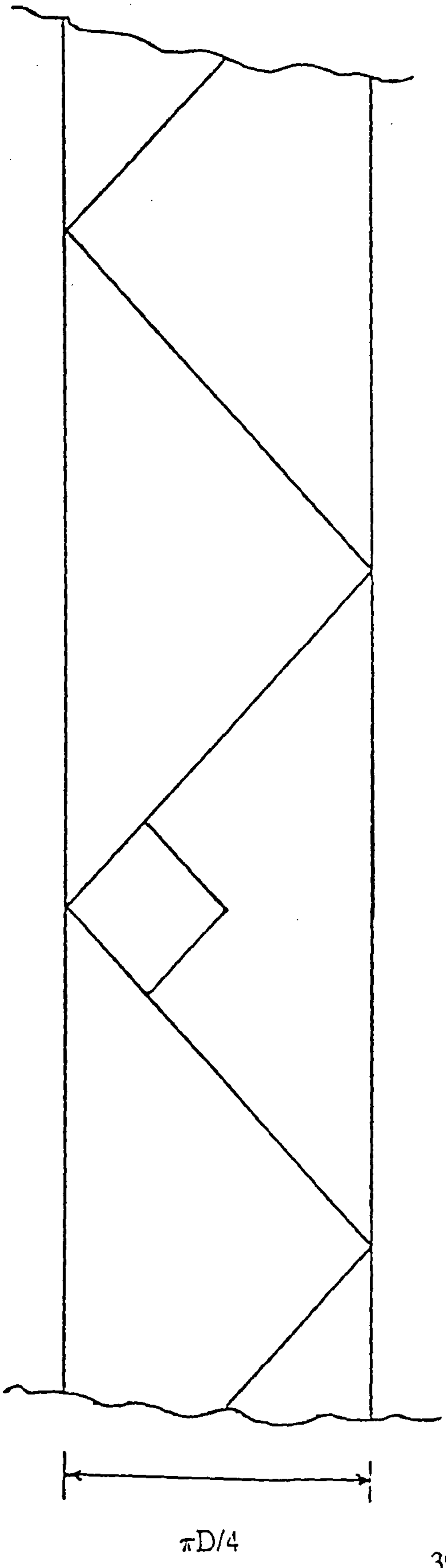
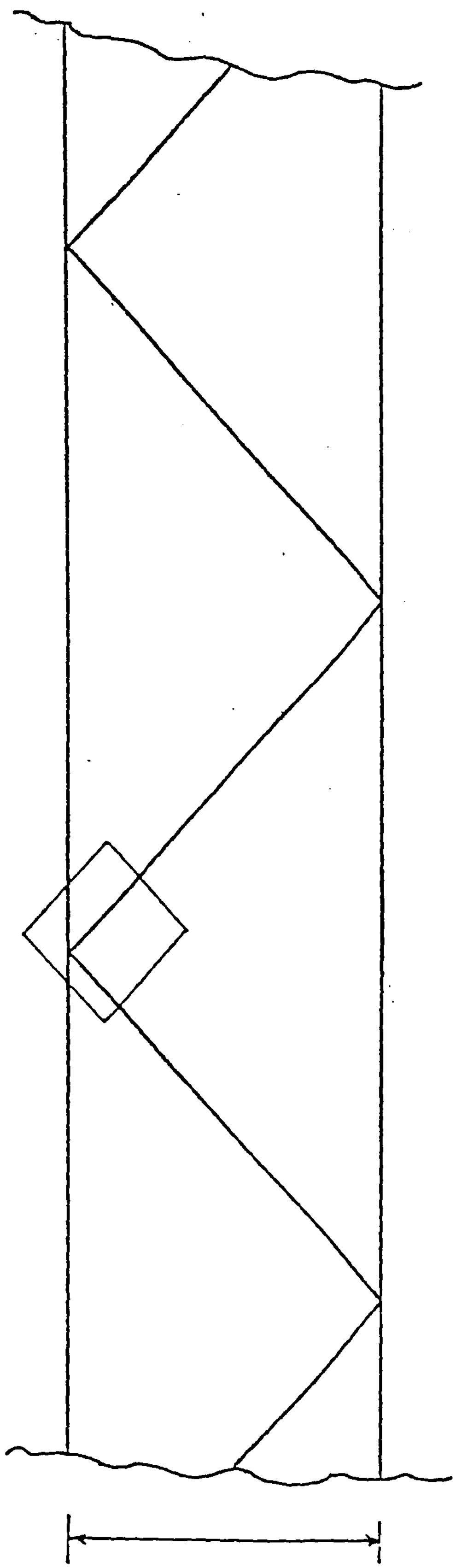


Fig. 3.1b Projection of the sleeve and orifice surfaces, $X = 2$



$\pi D/4$

Fig. 3.1c Projection of the sleeve and orifice surfaces, $1 < X < 2$

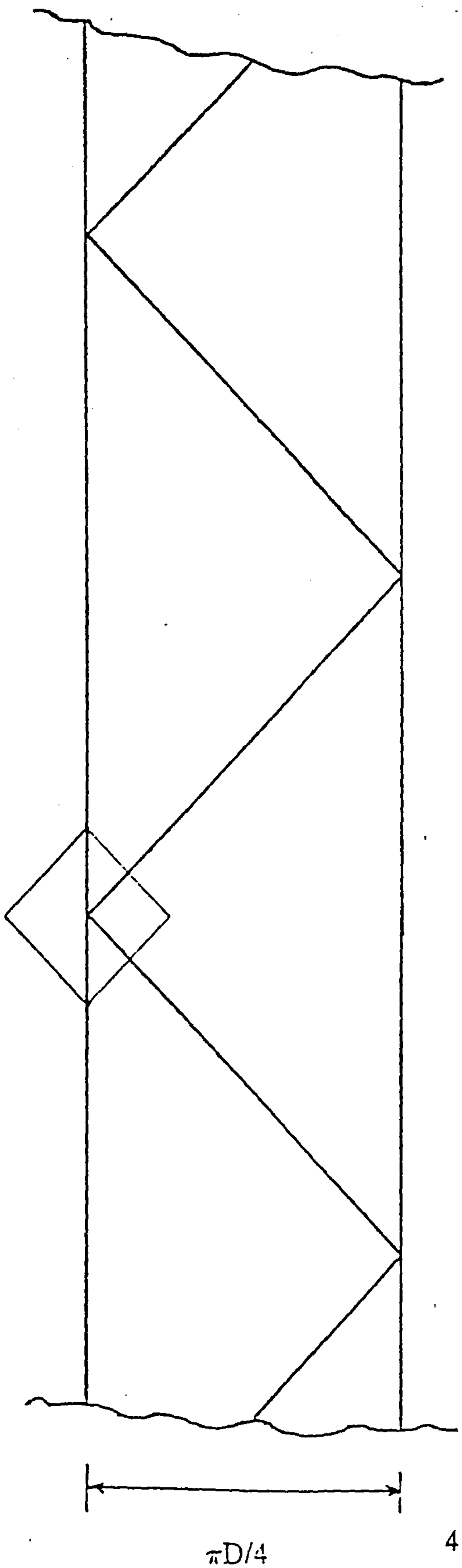


Fig. 3.1d Projection of the sleeve and orifice surfaces, $X = 1$

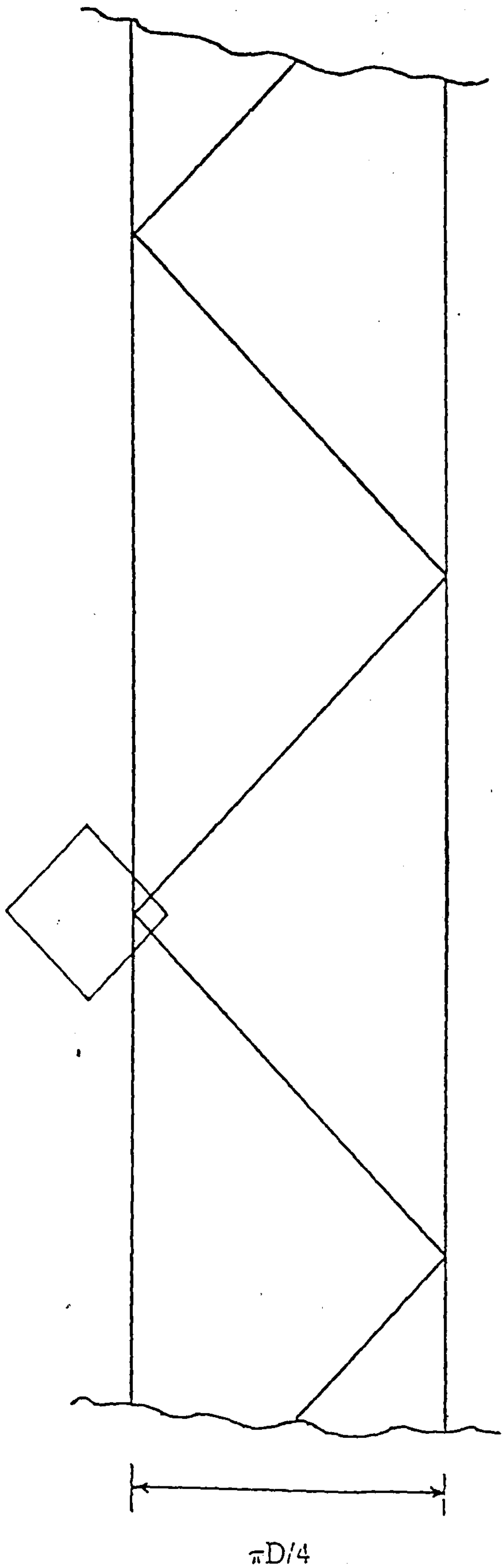


Fig. 3.1e Projection of the sleeve and orifice surfaces, $X < 1$

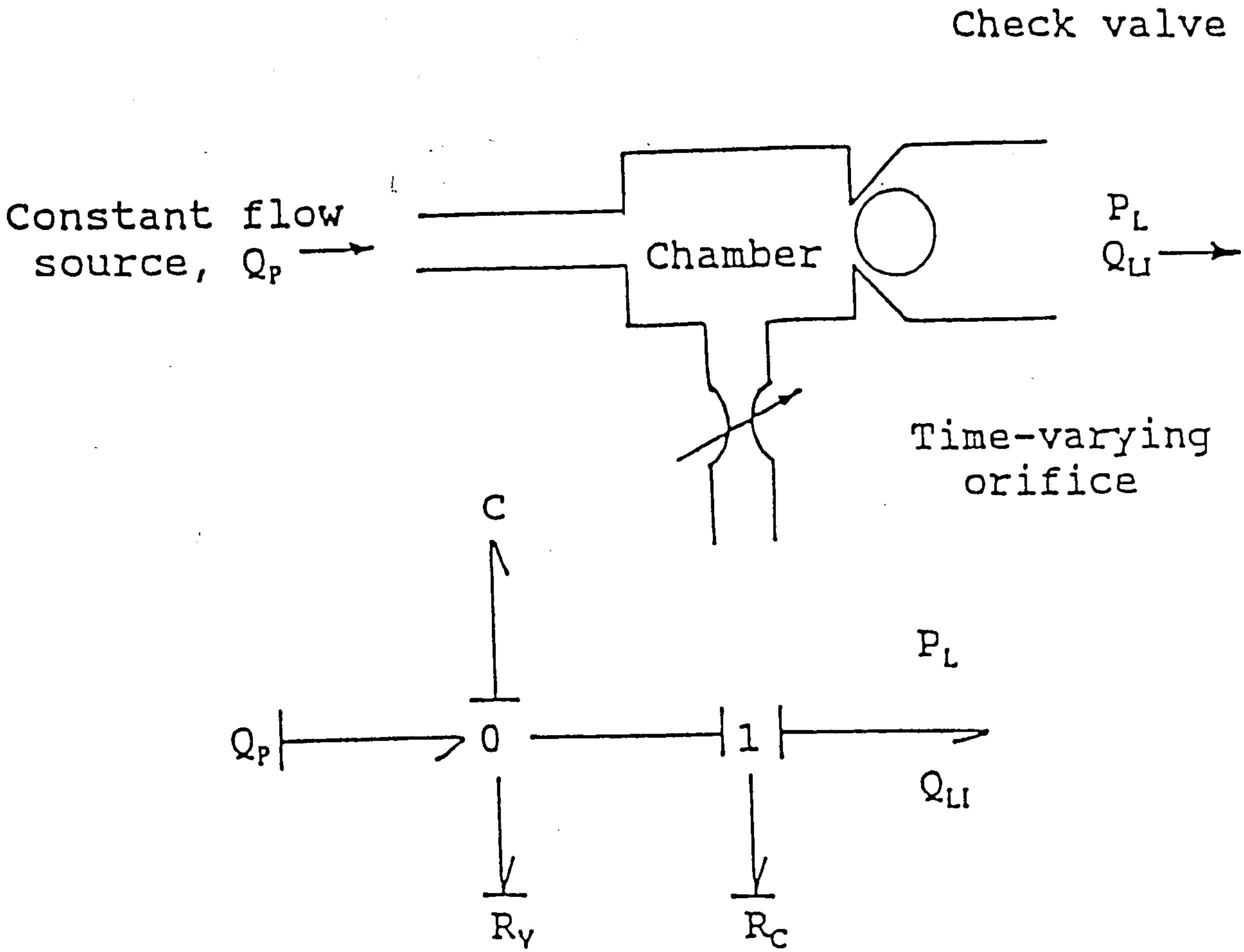
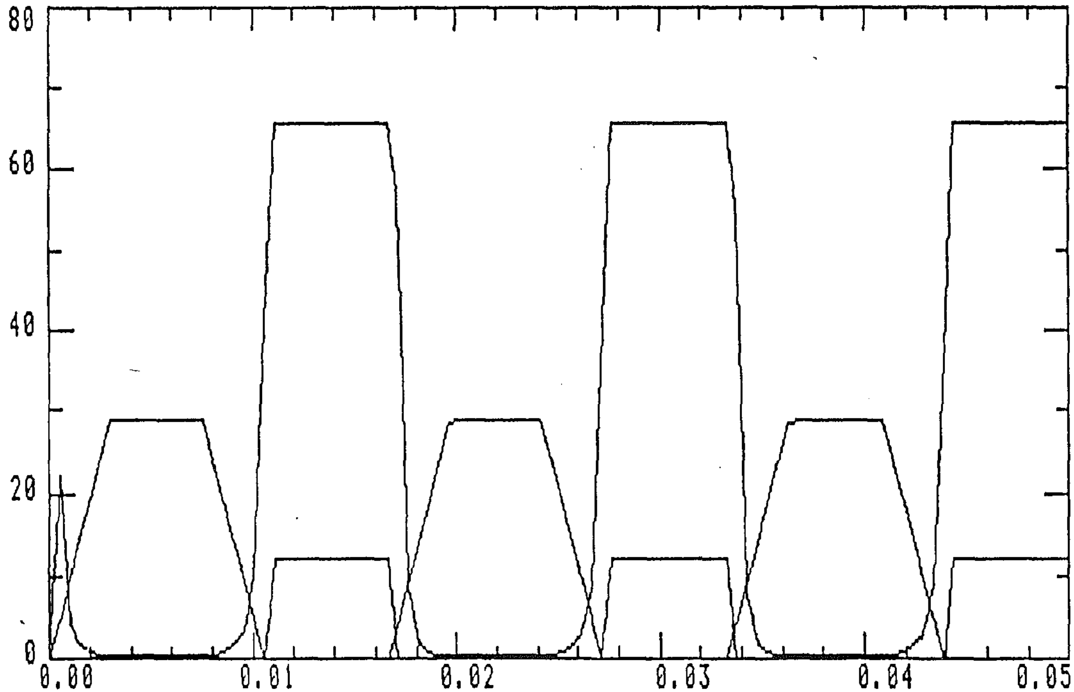


Fig. 3.2 Analytical model of the system



P_L = 65 bar

C = 1.73 mm^3 / bar

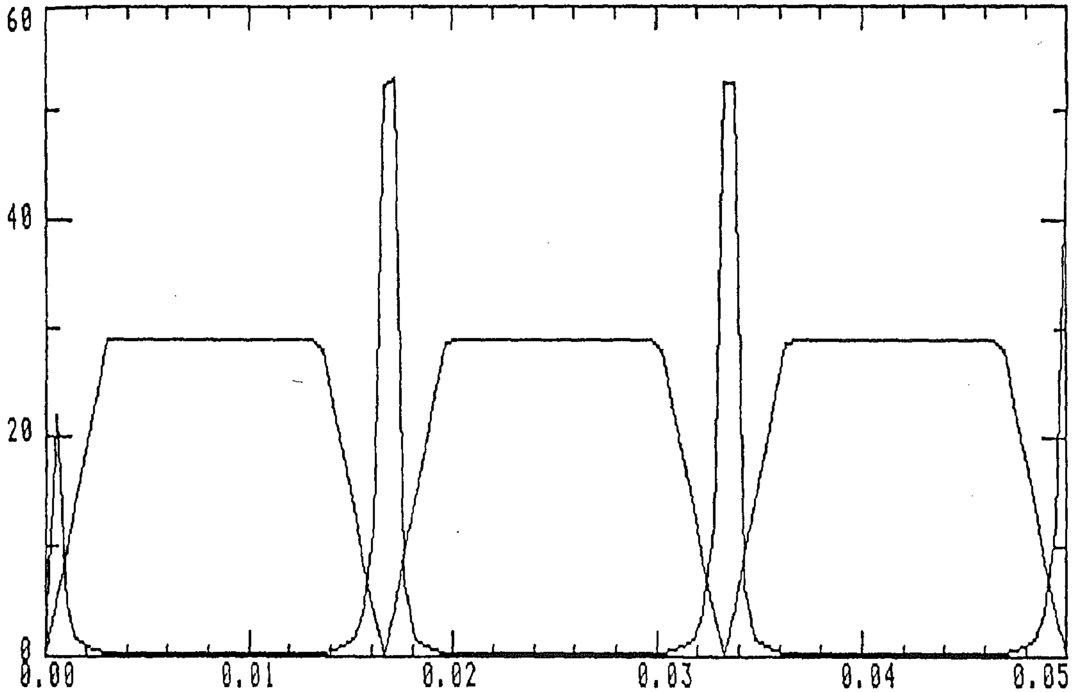
1800 R.P.M

D = 19.05 mm

X = 3.5

Q_L/Q_P % = 36.23 %

Fig. 3.3a Typical time simulation results



P_L = 65 bar

C = 1.73 mm^3 / bar

1800 R.P.M

D = 19.05 mm

X = 5.5

Q_L/Q_P % = .23 %

Fig. 3.3b Typical time simulation results

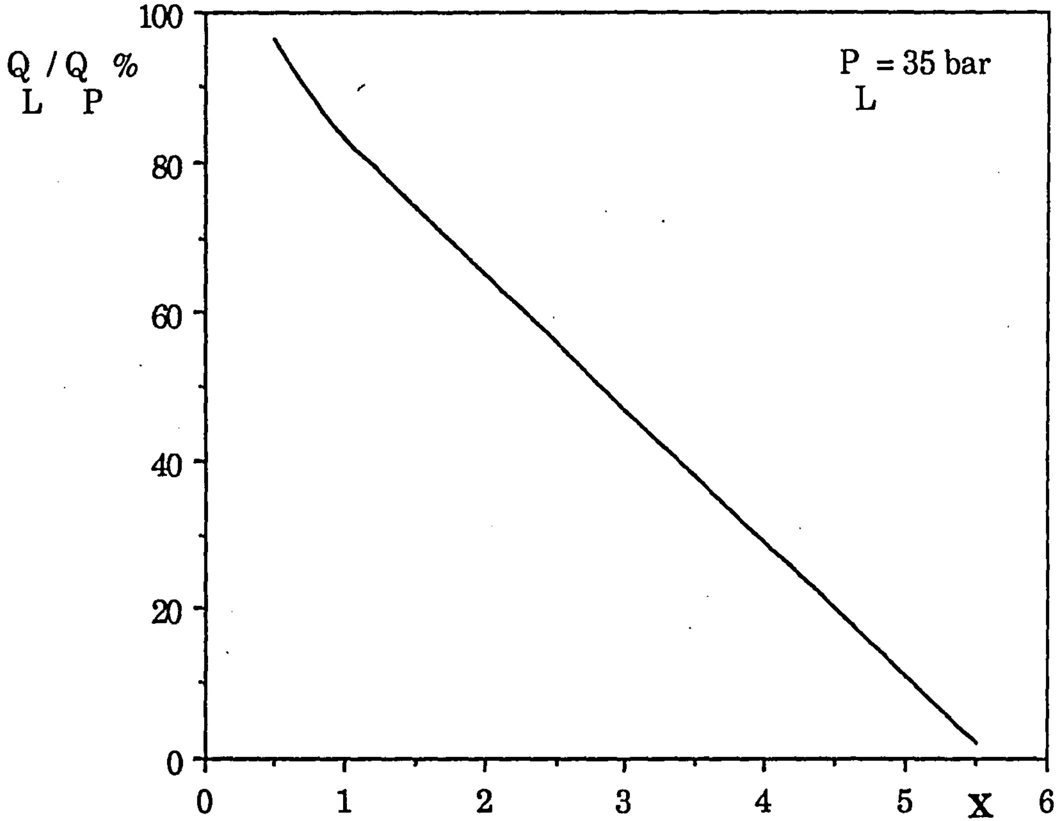


Fig. 3.4a Normalized load flow versus normalized sleeve position

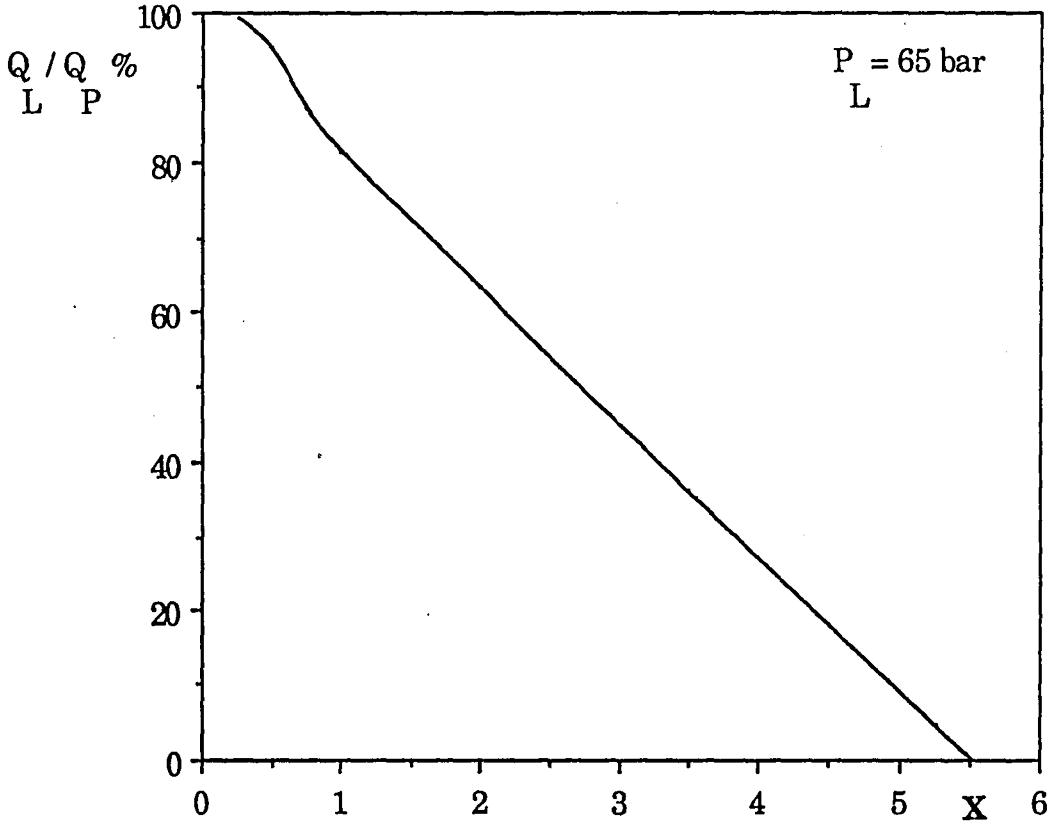


Fig. 3.4b Normalized load flow versus
normalized sleeve position

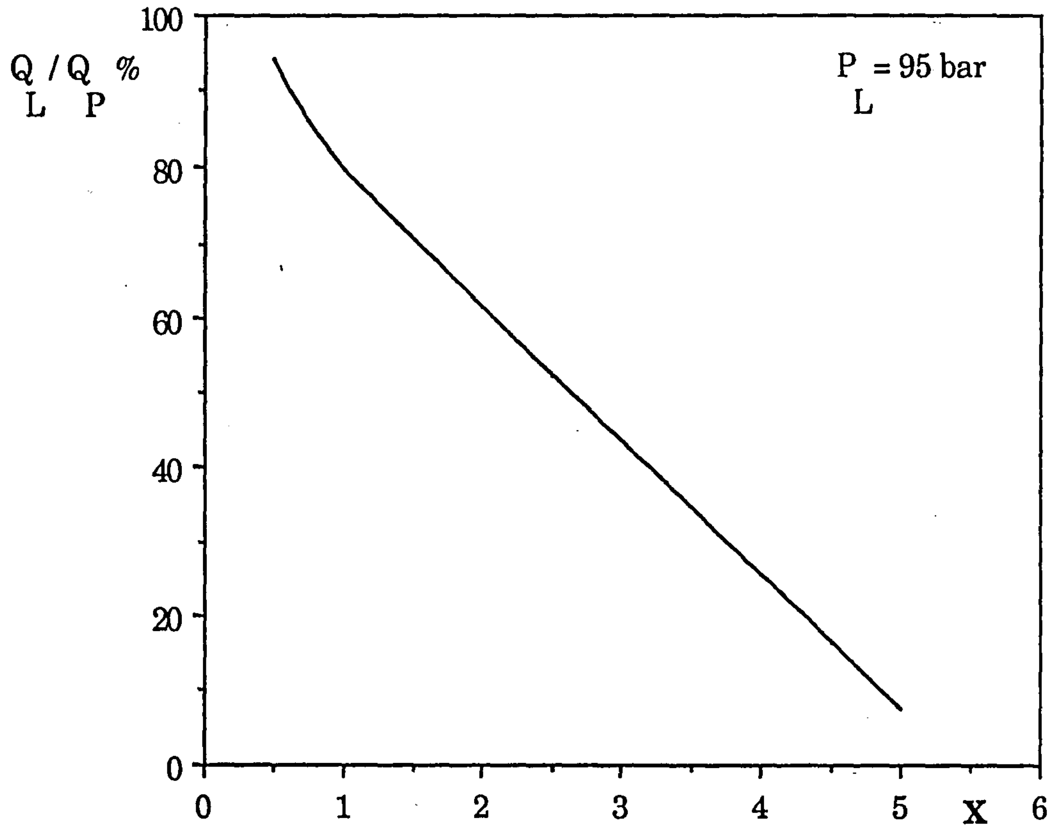


Fig. 3.4c Normalized load flow versus normalized sleeve position

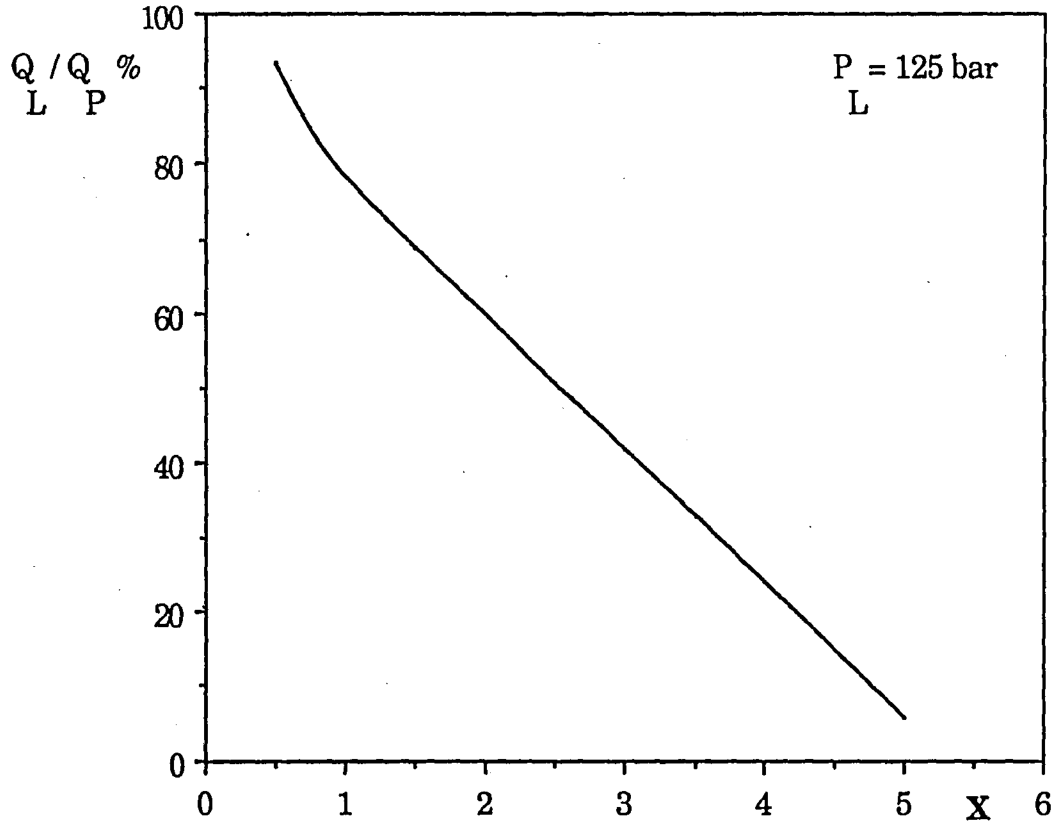


Fig. 3.4d Normalized load flow versus normalized sleeve position

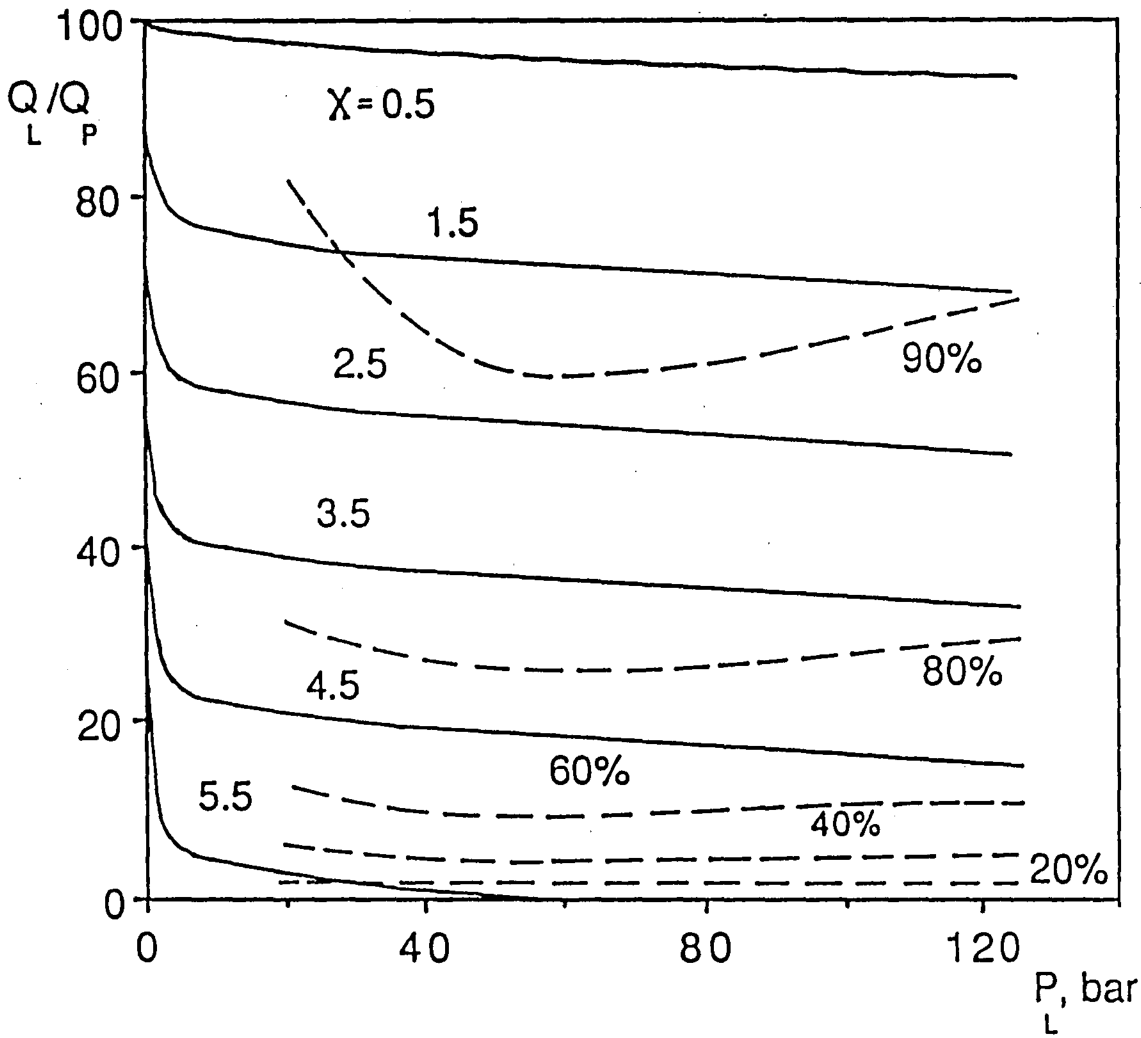


Fig. 3.5 Summary of results of simulations

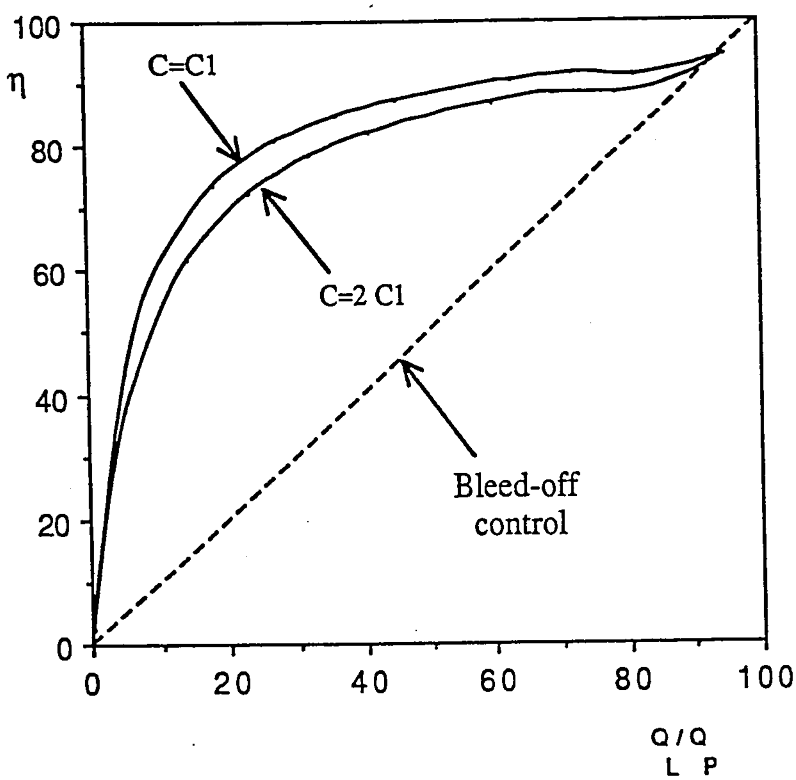
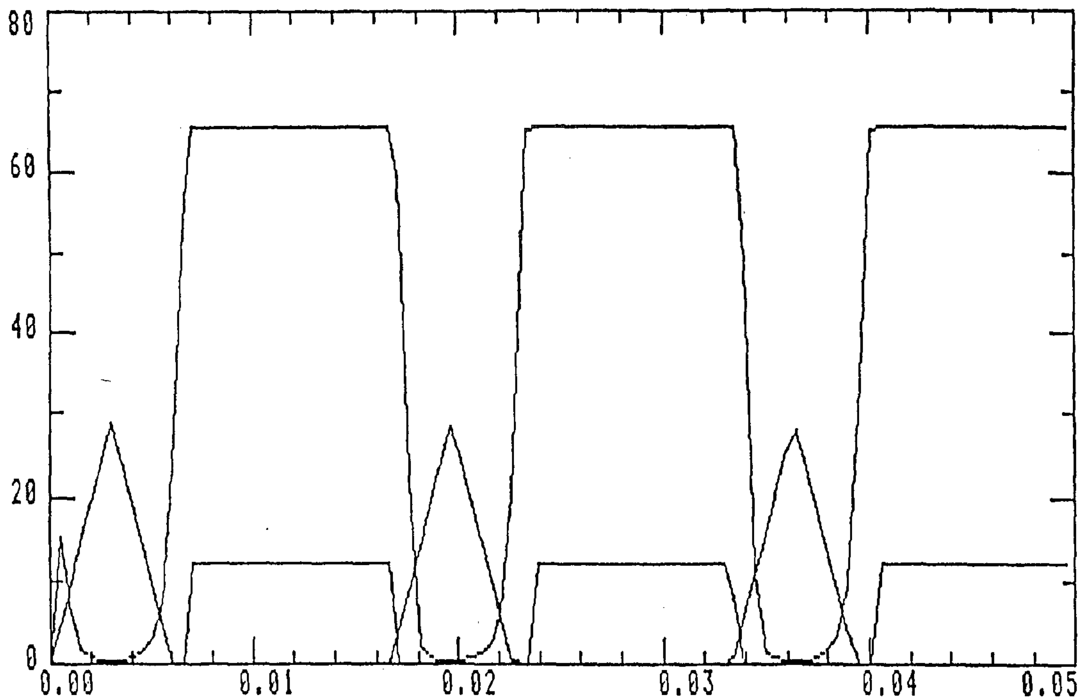


Fig. 3.6 Effect of the compliance on the efficiency



P_L = 65 bar

C = 3.46 mm^3 / bar

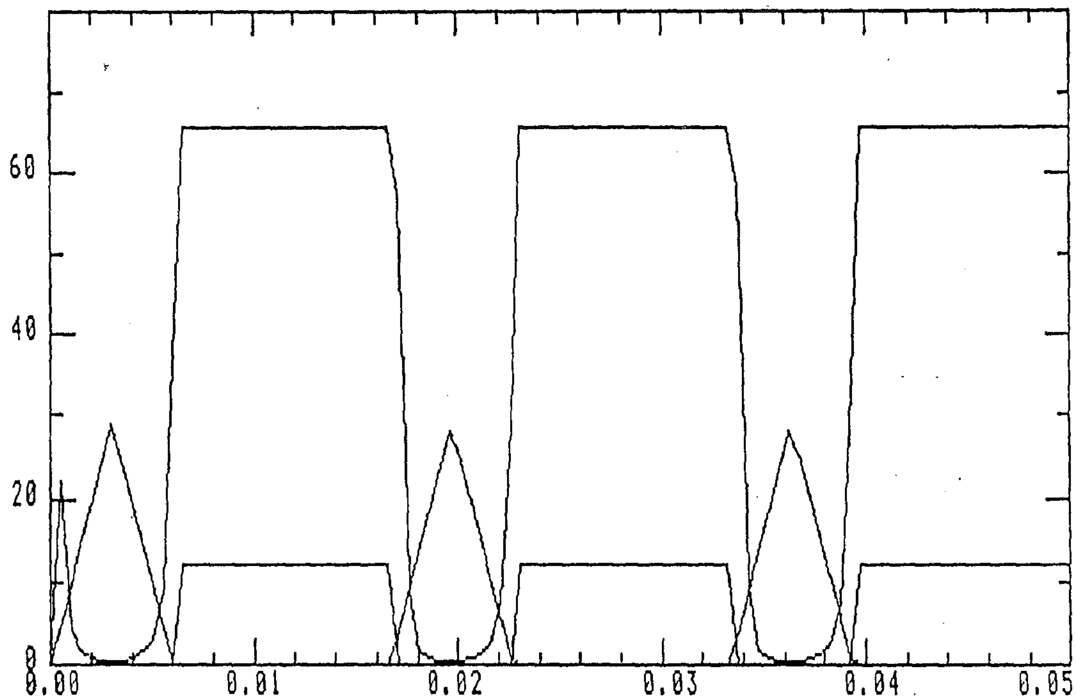
1800 R.P.M

D = 19.05 mm

X = 2

Q_L/Q_P % = 60.32 %

Fig. 3.7a Typical time simulation results



P_L = 65 bar

C = 1.73 mm^3 / bar

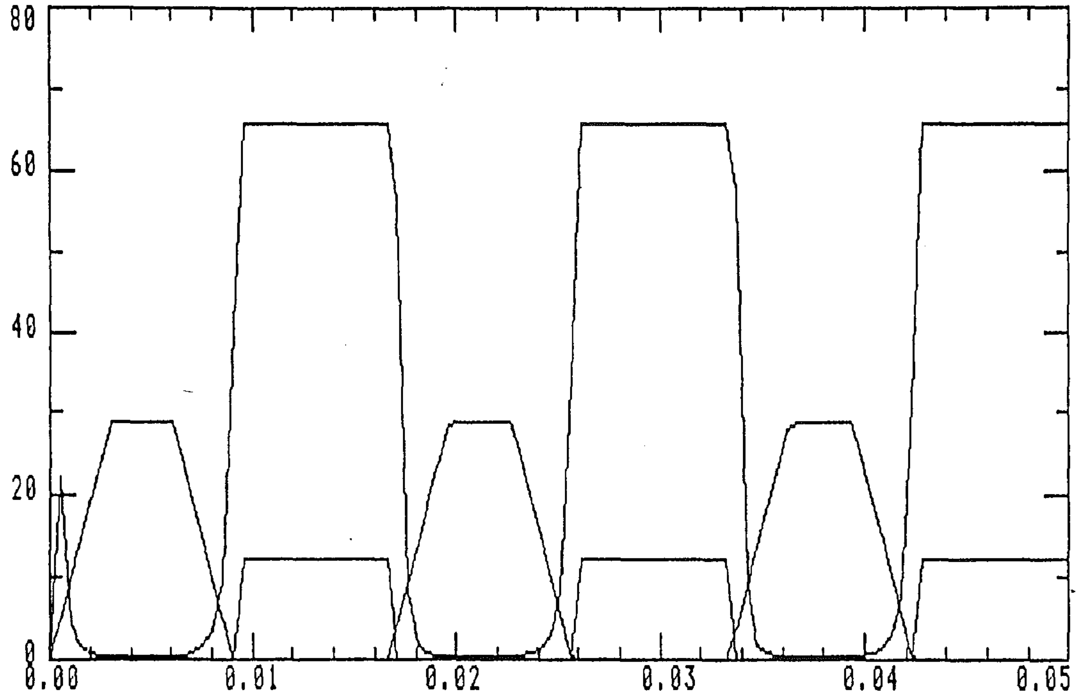
1800 R.P.M

D = 19.05 mm

X = 2

Q_L/Q_P % = 63.23 %

Fig. 3.7b Typical time simulation results



P_L = 65 bar

C = 1.73 mm^3 / bar

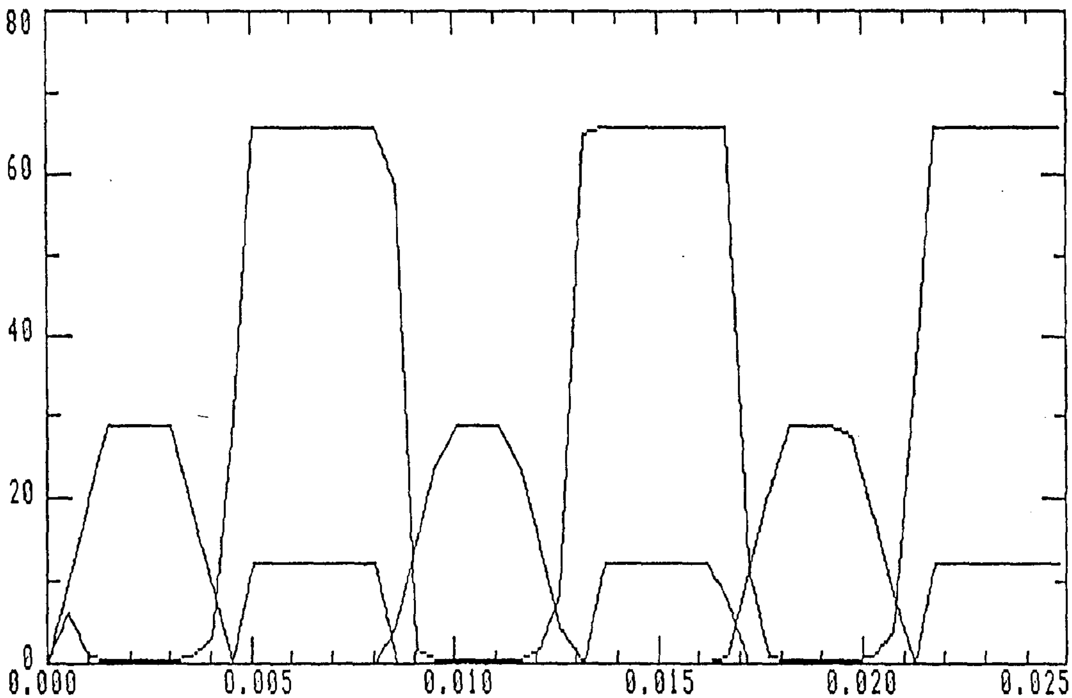
1800 R.P.M

D = 19.05 mm

X = 3

Q_L/Q_P % = 45.23 %

Fig. 3.8a Typical time simulation results



P_L = 65 bar

C = 1.73 mm^3 / bar

3600 R.P.M

D = 19.05 mm

X = 3

Q_L/Q_P % = 42.32 %

Fig. 3.8b Typical time simulation results

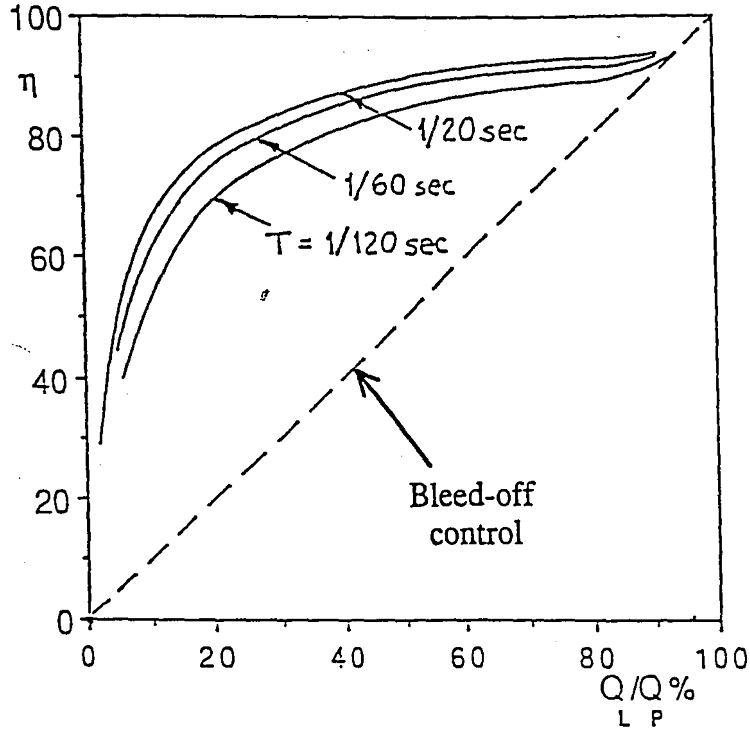


Fig. 3.9 Effect of valve shaft speed on the efficiency

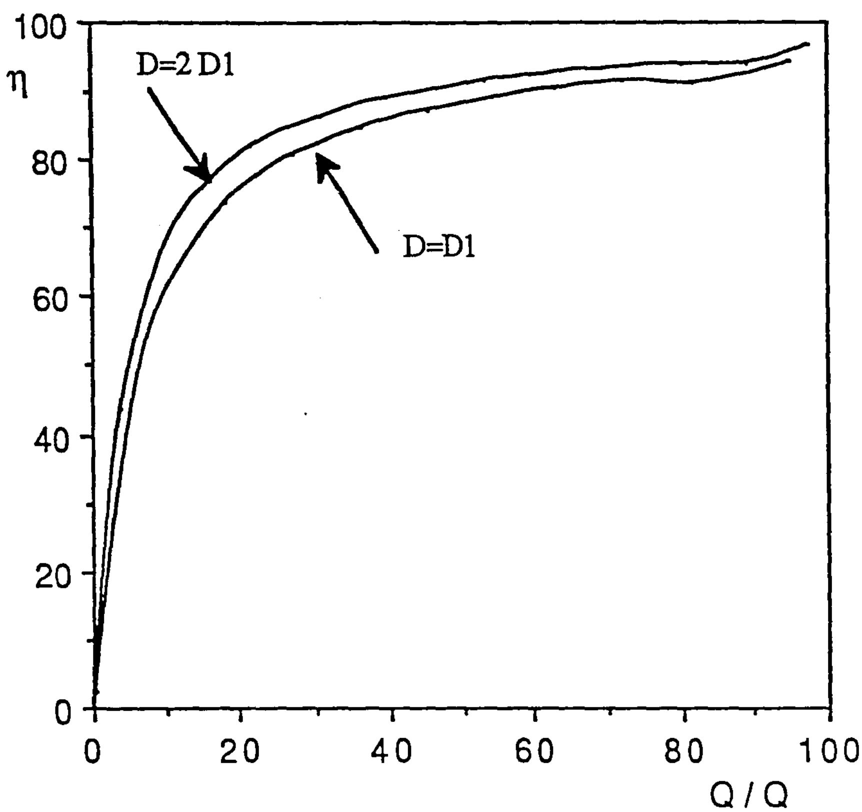


Fig.3.10 Effect of valve shaft diamter on the efficiency

Chapter 4

Model design

The test model is designed for simple testing and instrumentation. In practice, a compact design is necessary. The principal parts of the model are shown in figures 4.1-4.4. An overall schematic of the test apparatus and instrumentation is shown in Fig. 4.5. A separate variable speed motor was used to rotate the valve shaft which would be attached to the pump shaft instead in a practical system. The modulation sleeve is chosen small enough to reduce its inertia for a faster control system.

As previously mentioned oil contamination has thus far frustrated the experimental work.

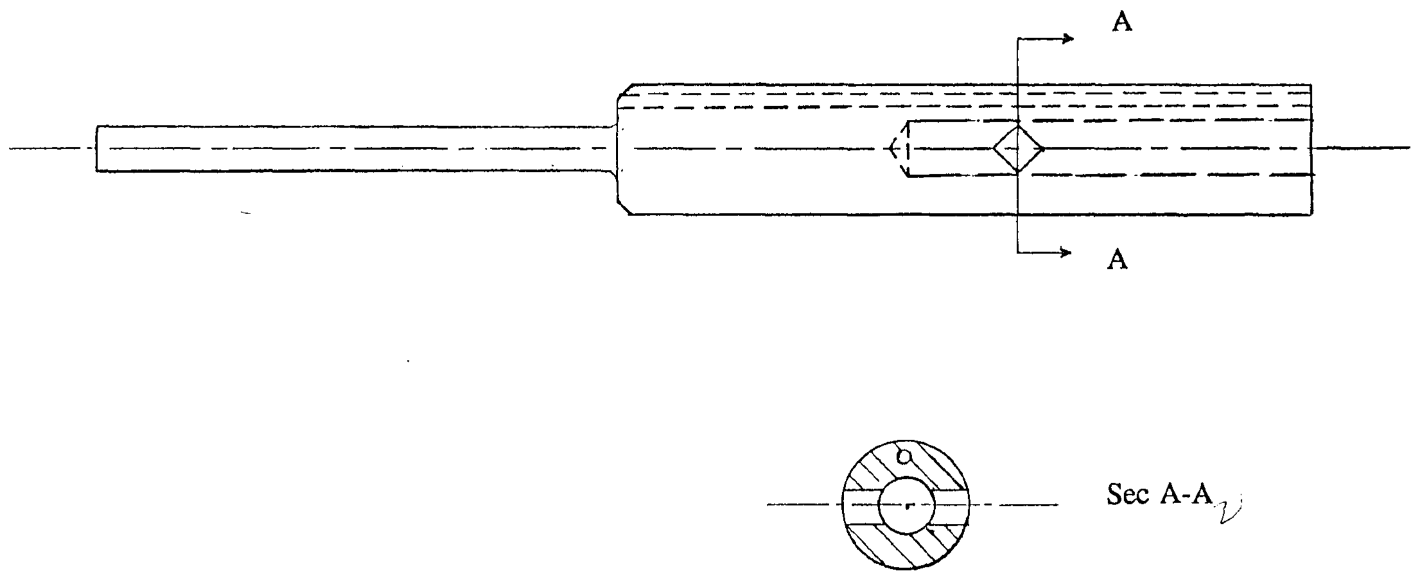


Fig. 4.1 Projection view for the flow shaft, scale 1:1

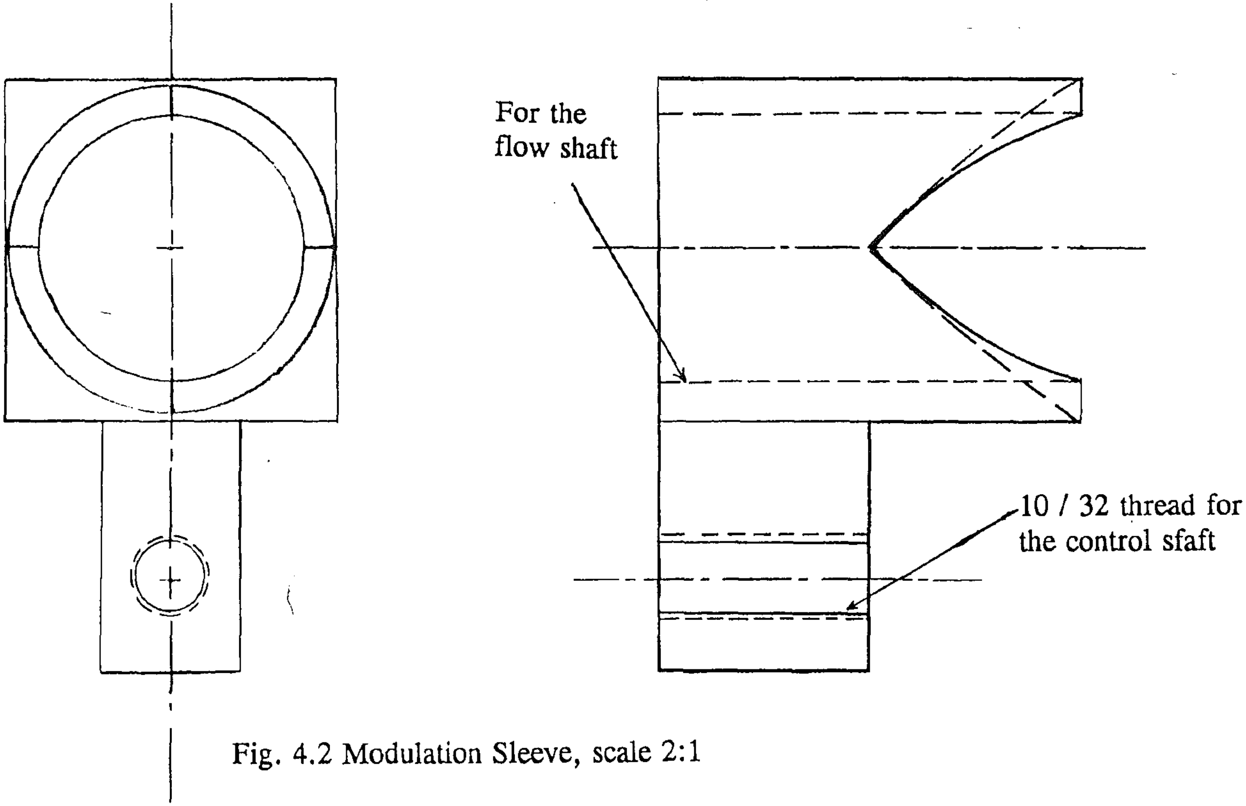


Fig. 4.2 Modulation Sleeve, scale 2:1

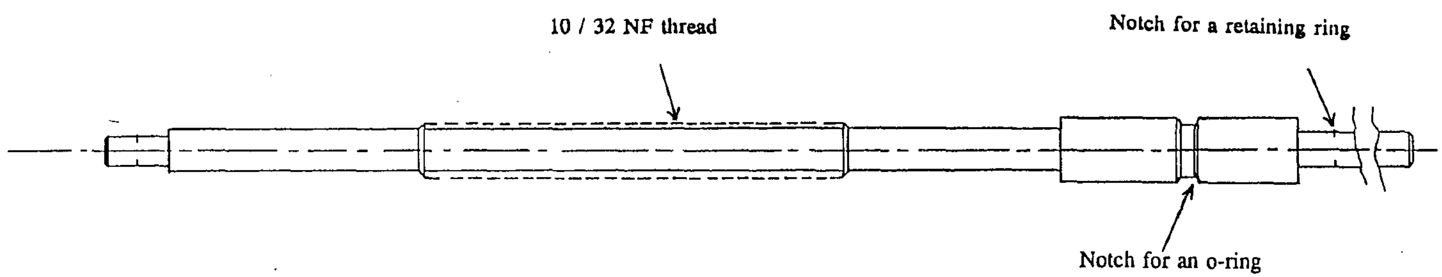


Fig. 4.3 Control shaft, scale 1..5 :1

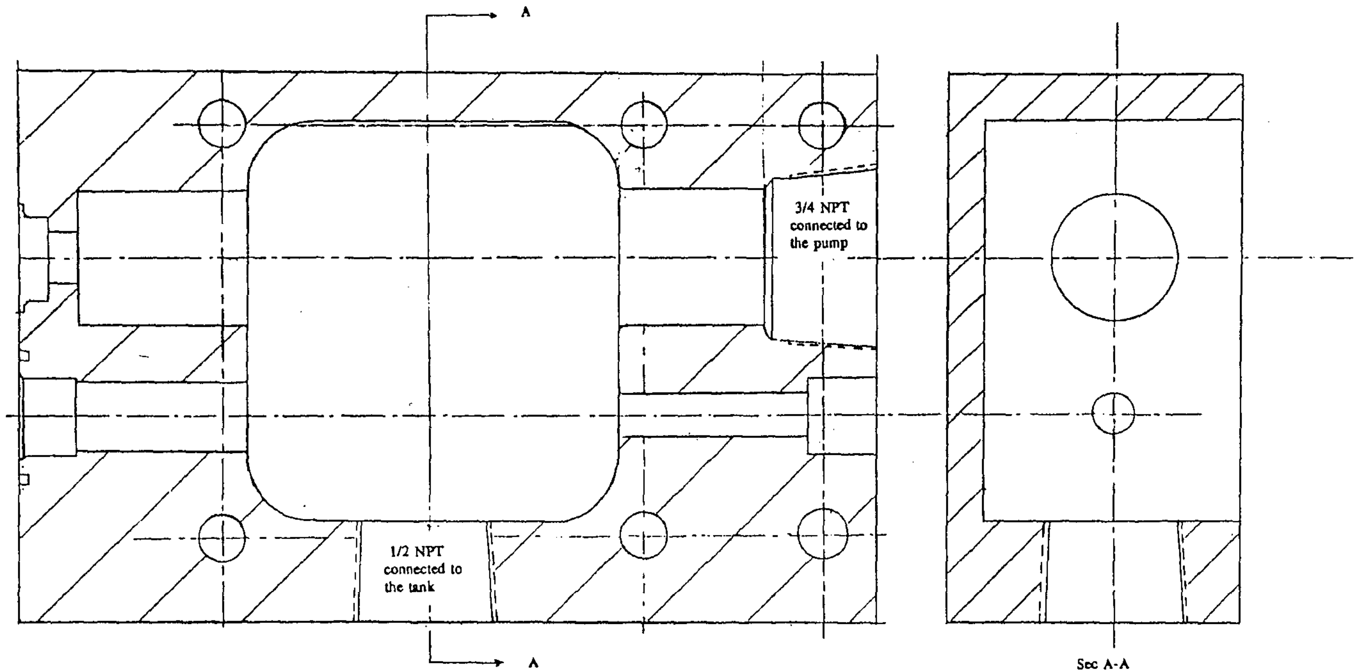


Fig. 4.4 Valve body scale 1:1

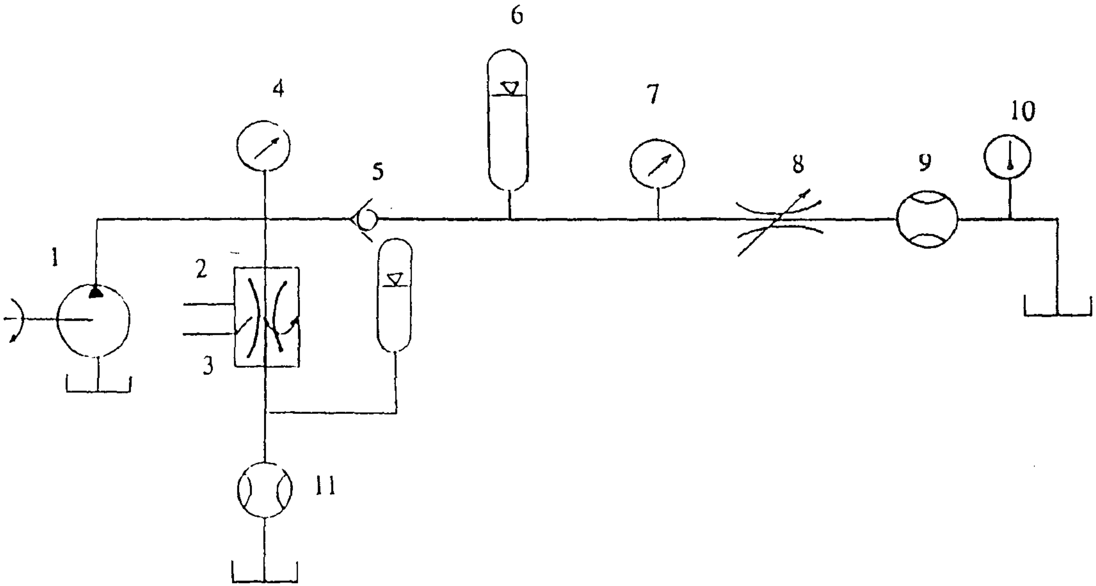


Fig. 4.5 Schematic of the Experiment Set-Up

1- A 13.32 Lit / min gear pump

2- The control shaft, driven manually in the test model

3- The flow shaft, driven by a variable speed DC Motor

4- A piezoelectric pressure transducer mounted integrally into the housing of the check valve

5- A hydraulic sealing check valve

6- A 250 CC accumulator

7- A burden tube pressure gauge

8- A flow control needle valve

9 & 11- Turbine flow meters

10- A thermometer

11- A 125 CC accumulator

Biography

Ahmed Abdelaziz Omara was born on June 7, 1964 in Cairo, Egypt. He graduated with honors from Zagazieg University with a Bachelor of Science in Mechanical Engineering in May 1987. Then he worked for the industry, where he developed a special interest in control of fluid power systems, before he joined Water Research Center. He started his graduate studies at Lehigh University in fall semester of 1990.

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