A COMPUTER SIMULATION FOR THE DESIGN OF PERCUSSIVE HYDRAULIC DRILLS

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A dissertation submitted to the Faculty of Engineering, University of the Witwatersrand, Johannesburg in fulfilment of the requirements for the degree of Master of Science in Engineering.

I declare that this dissertation is my own, unaided work. It is submitted for the Degree of Master of Science in Engineering in the University of the Witwatersrand, Johannesburg. It has not been submitted before for any degree or examination in any other University.

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(Signature of Candidate)

28th day of December 1990

ABSTRACT

This research project was undertaken to develop a computer model for a hydraulic percussive drill that could be used as an aid to design. Because a saleable drill was required in a relatively short time a high water based fluid was used initially. The water drill was not properly developed before this dissertation was completed.

The computer program was developed in Basic using a Hewlett Packard 9816 microcomputer. It was used in the design of a prototype drill to run on 5/95 fluid. The drill was made and tested under controlled laboratory conditions and then in field conditions underground in a mine. In the first tests the machine was instrumented to measure pressures and displacements. These parameters were digitized and fed into an IBM compatible PC. Performance curves were plotted and compared with those plotted using the model.

It was found that the percussion frequency and blow energies of 55 Hz and 74 Joules predicted by the model were very close to the values of 53 Hz and 76 Joules recorded on the prototype. The shapes of the pressure curves although close, differed at the striking point.

Unfortunately some difficulties were caused by the model being on the Hewlett Packard and the recording system being on the IBM. The two systems are not compatible. The model programs will in future be written for the IBM.

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LIST OF SYMBOLS

QUANTITY	SYMBOL	UNIT
Constant pressure piston area	A_1	m^2
Variable pressure piston area	A_2	m^2
Constant area valve bias	A_{v1}	m^2
Variable area valve kicker area	A_{v2}	m^2
Piston stroke	s	m
Valve stroke	s_v	m
Valve opening area	A_v	m^2
Piston mass	M_p	kg
Valve mass	Mv	kg
Piston velocity	v	m/s
Piston initial velocity	v	m/s
Piston acceleration	a	m/s
Valve initial velocity	v	m/s
Valve final velocity	v	m/s
Valve acceleration	a_v	m/s
Fluid bulk modulus	K	MPa
Fluid density (mass density)	w	kg/m
Pressure in chamber 1	p_1	MPa
Pressure in chamber 2	p_2	MPa
Time	t	s
Supply pressure	p_s	MPa
Rotation motor speed	N	rev/min
Energy per blow	E	Joules
Frequency	f	Hz
Force	F	N
Coefficient of friction	μ	
Efficiency	η	
Angle	α	deg
Area of surface	A	m^2
Breadth	b	m
Diameter	d	m
Distance	s	m
Height	h	m
Length	l	m
Radius	r	m
Volume	V	m^3
Flow rate	Q	l/s

QUANTITY	SYMBOL	UNIT
Viscosity, kinematic	ν	mm/s
Viscosity, dynamic	n	g/ms
Reynolds number	Re	
Youngs modulus	E	GPa
Shear modulus	G	GPa
Torque	T	Nm
Power	P	kW

1 INTRODUCTION

1.1 Review of Current Hydraulic Drills

Drilling by rotary percussive means has long been accepted as one of the most efficient ways to drill blast holes in hard rock. The method of delivering the blows has changed over the years. Initially the drill steel was struck by a sledgehammer swung by the miner. In between each blow, he or his assistant, turned the drill steel to position the bit on an unfractured part of the rock at the bottom of the hole. The introduction of the pneumatically driven drill or jackhammer was a major improvement. This synchronized the percussion and rotation actions in one air driven tool. Drilling rates were considerably increased.

The pneumatic drill originally designed at the end of the last century has been gradually improved. It has, by careful development and innovative design, been made into a sophisticated, reliable unit that will withstand the daily use and misuse, found in a mining environment. Using compressed air as its power source means that it is intrinsically safe to operate. This is particularly important when used where inflammable gases such as methane may be present. The relatively low pressure air is usually supplied to the drill at pressures ranging from 400 kPa to 700 kPa. This requires a single air supply hose, usually made from a lightweight polypropelene material. In addition a small hose is required to supply flushing water to suppress the dust formed when drilling. No air return hose is required as the air is exhausted to atmosphere.

Air as a power source suffers from the disadvantages however, that it is compressible and, at the compressor, heat is generated and taken out of the air by intercoolers. This is energy lost. The reticulation system used to transmit air from surface to the stope faces underground is complex in deep level mines. This results in large pressure drops when peak usage of drills occurs. This is made worse by any air leaks in the system. The larger the system the greater the possibility of leaks. The overall efficiency of the pneumatic system is therefore very low. During drilling, pressures at the drills sometimes drop to 400 kPa or less and as drill performance is proportional to pressure to the power 1.5, any drop in dynamic pressure represents a large drop in output. This can be critical in maintaining drilling rates and output of ore in a mine.

In addition to the system weaknesses the drill itself has some shortcomings. It´s lubrication system is usually rather crude in that the oil or grease is introduced to the drill in the supply air. By the time the exhaust air leaves the drill all of the lubricant has not been deposited in the drill. As a result some comes out with the exhaust along with condensed water to form a foggy cloud around the drill and operator. This is without doubt a health hazard. In addition the expanding air emits a high level of noise which necessitates the use of hearing protection by the operator and anyone nearby. Any attempt by the designer of the drill to attenuate the noise will result in some sacrifice of performance. This may vary from 5 to 7 percent, depending on the design and efficiency of the silencer used.

Over the last ten to fifteen years hydraulic percussive drills have been introduced into mining and quarrying applications. These use mineral oil as the working fluid. They are usually the large mounted type of rock drill called a drifter, with a percussive power output of between 7 and 15 kW. The rotation mechanism is also run on oil and can be operated independently from the percussion. When compared with their pneumatic counterparts they are more efficient, quieter and have more power built into them for a given size. A list of such drills currently available showing power and size is shown in Appendix A, on page 62. In the more developed countries most drilling is now done with these hydraulic units mounted on mechanized equipment. Where the vehicles are rubber tyred this method is now commonly called trackless mining. It usually results in a more efficient and less labour intensive mining operation. It is however more capital intensive and less flexible than the pneumatic jackleg operation it replaced.

One of the last strongholds of the pneumatic drill is in the hand held or thrust leg mounted applications in narrow reef stoping conditions, particularly here in the gold mines of South Africa. These mines have for a long time relied on this form of drilling. Although there has been a move in recent years to introduce heavy mechanized equipment into some mines, a lot are still too deep and difficult to convert to the trackless type of mining. These mines will continue to use small jackhammers for some time to come.

The feasibility of using small hydraulic percussive drills to overcome the weaknesses of pneumatic drills was investigated by the Chamber of Mines Research Division some time ago. They have, in collaboration with drill manufacturers, demonstrated that it is possible to achieve acceptable reliability and performance under controlled conditions. The use of hydraulic jackhammers with their higher overall efficiency would result in much greater productivity. The use of oil as the working fluid has however several problems associated with it. In a stoping environment the hydraulic jackhammer would be subjected to the same rough usage as its pneumatic counterpart. It would in the normal drilling cycle be dragged from hole to hole. This would cause abrasive wear on the supply and return hoses. The incidence of hose failure would be expected to be high. The consequent leakage of oil into the working area would cause very slippery and dangerous conditions. Extra leakage will also occur when drills are disconnected from the hoses at the end of the shift prior to blasting or to replace a drill requiring servicing. Any oil leakage or loss in the total system, such as damaged barrels in transit, will be costly. Mineral oil is not bio-degradable and adversely affects the efficiency of the reduction process in gold mines.

There are many good reasons then for not using oil as the working fluid. These include: its high cost; effect on the mining environment when spilt; and relatively high viscosity which causes high pressure losses in the reticulation system. The fluid chosen by the Chamber of Mines was initially a mixture of 5 percent synthetic oil concentrate and 95 percent water, commonly called 5/95. Drills have been running successfully on such fluids for the last ten years or so. This fluid when deposited on the footwall of a gold mine through leakage or spillage is bio-degradeable. It does not materially affect the ore reduction process as does mineral oil. In more recent years only 2 percent of oil concentrate is used in some applications resulting in a 2/98 fluid. The next step is of course to run a machine on water alone with no additives to improve lubricity, or inhibit corrosion. The type of machine developed by the Chamber of Mines in collaboration with Ingersoll Rand is one solution to the problems posed by the use of water based fluids. There are certainly many more design solutions to the same problem. Some may provide a more efficient drill which will ultimately be suitable for use on water. One of the reasons for this project is to investigate the possibilities of using a computer simulation to design a drill.

1.2 Water Drill Parameters

The operating parameters of any drill will to a large extent depend on the fluid to be used. Water as a fluid is not normally regarded as a good choice because of its physical properties. Thus the change of fluid from oil to water imposes several constraints on the design of the drill, the main problems being:-

* Low viscosity
* Low lubricity
* Corrosion
* Erosion
* Cavitation
* Temperature limitations

1.2.1 Low viscosity

Water has a low viscosity, typically 0.663 centistokes at 40°C compared with 20 centistokes at the same temperature for a typical hydraulic mineral oil. Comparative viscosities at various temperatures are shown in Figs 1 and 2 on page 104. Low fluid viscosity has advantages and disadvantages in hydraulic drill design. One of the advantages that can be used is the lower fluid friction in piping to and from the drill and in the internal passages of the drill. This will result in lower pressure drops in these areas and thus less power loss. The drill and system should therefore be more efficient and will allow fluid transport over longer distances than oil. Offset against this is the possibility of greater internal and external leakage because of the lower viscosity. Unless a careful approach to the design in terms of small clearances or sealed junctions is used then these leakages will result in a low volumetric and overall efficiency.

The two factors must be considered carefully early in the design stage if the one is to be maximized and the other minimized. In any event the potential is there for the water or 5/95 drill to be more efficient in overall terms than the oil hydraulic unit.

1.2.2 Low lubricity

Hydraulic mineral oils have a high degree of lubricity built into them. Additives are also included which assist in preventing metal to metal contact between moving parts of working mechanisms. These additives, in addition to the higher viscosities of 20 to 30 centistokes mean that under normal circumstances using conventional materials seizure should not occur. In contrast water, as such, has none of these advantages. Material compatibility and detail design will therefore play an important part in the success and reliability of any parts in a water driven machine with relative motion between components. Similarly the choice of suitable rotary and reciprocating seals is important to restrict wear of these components to acceptable levels, so that long periods between maintenance can be achieved.

1.2.3 Corrosion

Mineral oils also have corrosion inhibiting additives. These go some way to preventing internal corrosion in machinery using such fluids. In addition the possibility and almost certainty of some external leakage during operation, however small, ensures that the external parts of the drill are covered by at least some small amount of oil. This usually reduces external corrosion, even in a mining environment, to acceptable proportions. Water again has no such properties.

1.2.4 Erosion

Water, because of its low viscosity and the possibility of entrained solids, will cause erosion problems. These will be more severe at areas of high fluid velocity and sharp or weak metal edges. Careful design to keep velocities low and changes of section to a minimum will be required.

1.2.5 Cavitation

Water has a relatively high vapour pressure compared to mineral oils. This creates the possibility of cavitation damage to parts of the drill where poor design allows rapid changes of pressure and velocity in internal ports and ducting. Even the change from 5/95 to water can significantly increase cavitation damage.

1.2.6 Temperature limitations

Water as is well known has only a limited temperature range over which it can operate as a liquid. At the lower end it will freeze at sea level at 0°C and at the higher end it will boil at 100°C. Both these limits will vary with external conditions such as pressure and chemical additives to the water. They do however restrict the operating range of water fluids to less than 100°C if trouble free drill running is to be expected.

1.3 Overall Objectives

Because of time constraints one of the original objectives of this project, to design and built a water drill, was postponed. The drill was designed to run on a high water based fluid. This will not affect the validity of the computer model for the drill. It does mean however that while the low viscosity, cavitation and erosion features of the fluid had to be considered in the design, low lubricity and corrosion were not such a problem. In effect these were put off until another time. The design was therefore easier in these areas. Where concessions were made in the design to allow for the difference between water and 5/95 later in the dissertation the reasons will be discussed. As a result the overall objectives were modified and are as follows :-

* To develop a computer model for a hydraulic percussive drill.

* Using this model to design a drill using the model that will run on a high water based fluid such as 5/95 or 2/98.

* To manufacture and build such a drill.

* To develop and test the drill, then use the test data to verify the computer model.

* To compare the test data with the computer model and modify the model where necessary.

* To do underground drilling reliability trials with the drill in a normal mining environment.

As already stated the computer programme developed for an emulsion or 5/95 drill should still be valid for an oil or water drill if the fluid parameters such as viscosity, density and bulk modulus are changed.

The design of the emulsion drill was made easier than that for water because corrosion and low lubricity were no longer problems.

2 DESIGN STUDY

2.1 Description of Oil Hydraulic Drills

Before commencing the design study for a drill using water or 5/95, it will be appropriate to discuss the operation of a typical oil hydraulic drill. This will provide a starting point for the design. It will also serve to highlight the areas in an oil drill which would cause problems for use with a low viscosity fluid. All hydraulic percussive drills currently available consist of two main parts. These are the percussion element and rotation mechanism. In almost all machines these parts operate independently. The rifle bar rotation used in many pneumatic drills is not normally used in hydraulic machines. The main reason for this is the relatively short stroke used in hydraulic drills when compared to pneumatic ones. A typical stroke would be 35mm compared with 50 to 86mm stroke for a pneumatic drill. The short stroke of the hydraulic drill is normally the result of constructional constraints. The pneumatic drill piston only has to seal on the main head diameter and the forward shank. This is done by maintaining close clearances between the piston, cylinder and front cylinder washer liner. Some leakage in these areas, even if it is external, does not cause any real problems. Air as an external leakage to atmosphere is not readily seen unless the leakage is felt by the operator. The noise of the leakage is usually lost in the general machanical and exhaust noise of the drill itself. On a hydraulic drill any leakage, particularly external leakage is serious. The hydraulic piston is therefore made up of three main sections as shown in Fig 3 on page 105. These sections are a forward section, a centre section and a tail. Both forward and rear sections have to be sealed against external leakage by some sort of positive pressure loaded seal. The centre section seals against port to port leakage ,ie, high to low pressure internal leakage. It may therefore consist of only a labyrinth type seal relying on small clearances over a relatively long length to keep the leakage value to an acceptably low level. It does not have to form an absolute seal. Thus with three basic sections to the piston instead of two as for the pneumatic drill, a short stroke is essential if the overall piston length and hence the machine length, is to be kept within reasonable limits.

2.1.1 Percussive mechanism

Various percussive cycles are used in drills currently available. One of the most popular has the impact piston operated by a three way directional control valve. The valve is pilot operated by the position of the piston. This type of cycle is used on many of the drills listed in Appendix A on page 62.

Fig 3 on page 105 shows a typical oil hydraulic drill. Figs 4a to 4d on pages 106 and 107 show diagramatically the percussion cycle of the drill. The piston consists of the three main diameters as described. This results in two operating areas A_1 and A_2, typically $A_2 > A_1$. Oil pressure is applied continually to A_1 and intermittently to A_2. Hence when applied only to A_1 in the diagram the piston moves to the left or away from the drill steel. When pressure is applied to both areas the net force on the piston is to the right causing it to accelerate towards the drill steel.

The pressure is alternately applied to A_2 and vented from A_2 by a distributor valve as shown. The valve in this particular design is a sleeve valve in the cylinder body around the rear portion of the piston. It has two operating or pilot areas A_{v1} and A_{v2}. In this case $A_{v2} = 2A_{v1}$. When the piston is moving towards the right on the impact stroke, as shown in Fig 4a, both pilot areas of the valve are under pressure and the valve is held hard over to the right because $A_{v2} > A_{v1}$. Under this condition the high pressure port of the valve is open to the operating chamber of the piston and area A_2.

The piston accelerates and strikes the drill steel as shown in Fig 4a. Just before it strikes the steel a metering edge on the piston connects the kicker port 5 to the tank port 6 which vents the kicker chamber and the valve area A_{v2} to low pressure. The valve now under the action of high pressure on area A_{v2} moves across to exhaust the piston chamber 2 and area A_2 to tank. The piston then starts to move to the left under the action of fluid pressure on the piston area A_1. As it moves to the left or rearward, the rear edge of the piston metering land covers the kicker port. As the piston moves further back the front edge of the metering land uncovers the kicker port to high pressure as in

Fig 4b. The valve then moves to the right , the high pressure port is opened to the piston operating chamber and area A_2. The piston now deccelerates under the action of the net hydraulic force acting to the right. It comes to a controlled stop at the rearmost position as shown in Fig 4c ready to start again. Fig 4d shows the piston on the forward stroke just before impact as the rear edge of the piston metering land opens the valve kicker port to tank.

The striking face of the drill steel is maintained in the correct position by means of a collar on the drill steel. This collar may be an upset forged collar on the drill steel itself. It may however be a bonded or crimped-on rubber collar. This is the more usual system used with light jackhammers in South African mines. The positioning of the drill body against the collar is achieved by thrusting the drill against it. If the thrust is insufficient or if the drillsteel is not in place whilst the machine is running then the piston of the impact mechanism would tend to run on and hit part of the drill body. In the drawings in Fig 4 on page 106 and 107 the machine illustrated shows a collar on the piston which in an overtravel condition enters a matching bore in the cylinder. This traps a volume of oil as a hydraulic dashpot, generating high pressure which will act on the collar area A_3. This generates a deccelerating force to bring the piston to a controlled stop before metal to metal contact occurs.

2.1.2 Rotation mechanism

The rotation on nearly all oil hydraulic drills is independent of the percussion as already stated. The rotation usually consists of a hydraulic motor driving the drill chuck or shank adapter via a reduction gearbox. The motor will normally be a gear type unit. It can be a standard external gear type, internal gear or gerotor type. During normal drilling the torque required from the motor will be approximately 10 percent of the maximum designed torque. This allows a large safety factor for the higher torques required in long holes or with badly fractured rock. The rotation speed will depend on the indexing angle necessary for rock breakage. This will depend on rock type, bit diameter and design. Button bits for example will normally require a different rotation speed to cross or chisel bits.

2.1.3 Percussion and rotation relationship

The control of the motor speed and hence the drill string rotation speed may be manual or automatic. If manual then the operator can adjust the rotation to suit the bit diameter and type of rock being drilled. He can monitor the rotation pressure to give an indication if the drill bit is getting stuck or not. If the pressure rises towards the maximum value then he can reduce the drill string feed force, the percussive power or both. This will reduce the tendency to jam the bit. As soon as the pressure drops to normal he can increase percussive power and thrust to continue at optimum penetration again.

This method can be simulated hydraulically in an automatic system. The rotation motor pressure can be sensed and interconnected to the feed pressure. Automatic feed reduction will result as imminent bit jamming is sensed. This frees the drill operator to concentrate on other things. Automatic drill retract can be arranged at the end of the hole.

2.2 Critical Design Areas

2.2.1 Fluid parameters

In paragraph 1.2 the water drill parameters were discussed. Most of these related to the fluid. A more detailed look at the fluid properties will now show some of the problems associated with the use of water as the working fluid. The main parameter which affects the design is the viscosity. When using water or 5/95 fluid the operating viscosity will be in the region of 0.7 cs to 1 cs depending on the fluid temperature. This is some three to five percent that of mineral oil usually used in hydraulic equipment such as pumps, motors or hydraulic drills.

The other parameter is lubricity. The use of water means that conventional materials are unlikely to be successful.

2.2.2 Designing for low viscosity

Low viscosity will mean high internal and external leakage unless careful attention is paid to effective sealing. External leakage from the percussion piston can be taken care of by the use of Shamban type seals. These comprise a filled PTFE or solid UHMWPE ring energized by a backing 'O' ring. This gives a low friction seal-to-piston interface with minimal leakage. The maximum rubbing speed just before impact of up to 9 to 10 m/sec is rather high for this type of seal, but no other type is available for this sort of application.

In a water or 5/95 drill the space before the seals can no longer be vented to low pressure. With an oil drill this is acceptable if the bearing before the seal is used as a labarynth type seal itself. The leakage through the close clearance bearing can be calculated using the formula:-

$Q=15708dpc^3/60l\nu w$ (l/s) Guillon (Ref No 7) (2.1)

It can be seen that if the viscosity of the oil and 5/95 are in the ratio of 20 cs to 0.663 cs or 30:1 then for equivalent diameters and clearances the bearing must be 30 times longer when using the lower viscosity fluid. As the oil machine uses a bearing length of 20 mm this means 600 mm for the 5/95 drill. This is obviously totally unacceptable.

The seals therefore have to be subjected to the full system pressure. This usually means higher wear rates because of the higher seal-to-piston contact pressure. It will also mean higher friction forces on the piston and hence lower striking velocities and frequency of operation.

The leakage on the centre section of the piston will also be higher because of the low viscosity. Leakage at this area will be very high if the low pressure port shown in the oil machine is included for venting the valve kicker signal to tank. If there is no low pressure port then the leakage from high to low pressure will only occur over approximately half the cycle time. That is when the operating chamber is exhausting and is at low pressure. During the other part of the cycle its pressure is much the same as the constant pressure chamber. On both the piston and valve operating diameters leakage must be kept to a minimum to maintain an acceptable overall efficiency. On the piston centre section the use of a seal would reduce leakage and allow a greater clearance than if a labyrinth type seal were used. In order to do this however no metering edges or holes on the piston must cross the sealing component. This means that a different operating cycle to the normal oil hydraulic drill must be used. The difference in such a cycle will be in the method of triggering the valve from the piston position. Such triggering should cause little or no leakage and as low a power requirement as possible.

2.2.3 Valve triggering

The standard type of hydraulic percussive cycle described earlier is shown in Fig 4 on pages 106 and 107. It can be seen that valve triggering is done by the signalling edges on the central part of the piston. It is therefore extremely difficult to arrange for a seal at this part of the piston because of the sharp edges. The radial clearance of the piston centre section in the housing must be small in order to maintain a low leakage from high pressure to low pressure and to the valve kicker port.

2.2.4 Types of valve

The operating valve is one of the most important parts of the percussive mechanism. It must change over completely in less than 4 ms. It must effect a seal so that the port-to-port leakage after changeover is low enough to maintain an acceptable overall volumetric efficiency. It must not consume too much power to move it either way. It must also respond as soon as the machine throttle valve allows the operating fluid into the machine under pressure. There must be no dead spot in the cycle where the machine is switched on by opening the throttle valve and the drill does not start.

Three types of valve were considered:

(a) Spool valve

This is the most commonly used type for changeover applications. It is relatively simple and easy to manufacture with the right kind of equipment. Careful control needs to be exercised however on the bore size, roundness, straightness and concentricity. The axial position of the metering edge needs to be maintained with some accuracy if the overlap to underlap conditions are to be controlled. Too much overlap will result in excessively high and low pressure pulses in the rear working chamber. This can cause broken side rods and casings. Too much overlap results in excessive leakage during changeover of the valve in mid stroke. Low volumetric efficiency will occur along with the possibility of valve instability.

Several types of spool valve that can be used in various applications are shown in Figs 5, 6 and 7, on pages 108 and 109. The three way valves are for use in the standard type of percussive drill cycles with differential piston areas. One under constant pressure and the other under variable pressure. The four way valve is for use in the more complex arrangement where both piston areas are subjected to alternating pressure.

An absolute seal with this type of valve is not possible. An acceptable seal can however be achieved by close clearance and adequate overlap of mating edges when closed. The smaller the valve diameter the lower the leakage rate for a given overlap. See Fig 8 on page 109 for spool valve leakage characteristics.

(b) Poppet valve

This type of valve can in theory give zero leakage when closed. In practice some leakage may result, but should be extremely small with valves in good condition. Unfortunately for this type of valve to form a changeover function then several valves are needed in combination. For the arrangements normally accepted in spool valves, some possible poppet or check valve combinations are shown in Fig 9 on page 110. In diagrammatic form it can be seen that a minimum of two valves are required up to a maximum of four. This may not seem too complicated until the necessary pilot operation of these valves is considered. Here the problem is both hydraulic and mechanical. It can result in a fairly complex and also bulky piece of equipment. In a piece of mining machinery which has to be simple and light such as a handheld jackhammer, the complexity may not justify the higher volumetric efficiency.

(c) Shear seal valve

This type of valve may be rotary or reciprocating. It is essentially a face seal type of valve. It may have some compliant facing to ensure better sealing between the faces. Careful design is required to ensure that proper hydraulic balancing is achieved throughout the valve travel. If this is not achieved the result will be high leakage or high friction in operation. An example of this type of valve is shown in Fig 10 on page 110

2.3 Analysis of Possible Cycles

2.3.1 Hydraulic spring cycle

The cycle shown in Fig 11 on page 111 illustrates that used by the Ingersoll Rand WFO 35 drill already in production. As can be seen each diameter has its own seal of a material such as UHMWPE. The clearance between the piston and its housing can be quite large in this case, thus making manufacturing relatively easy and cheap.

Operation is as follows: chamber 3 contains a trapped volume of fluid when the piston is fully forward. In practice there is some make-up system to replace fluid lost through leakage. The two piston diameters entering the chamber are very slightly different. Thus as the piston moves backward under the action of fluid pressure on area A_1 the volume in chamber 3 is reduced. The fluid is compressed and the pressure rises depending on the Bulk Modulus of the the fluid used. The pressure in chamber 3 is therefore approximately proportional to the axial position of the piston in the housing. It is used to move the valve against the constant pressure force on A_{v1}. The system will be reliable as long as the chamber is effectively full of fluid when the piston is fully forward. Some variation of signal will take place with variation of fluid temperature, ie, the higher the temperature the greater will be the chamber pressure. At high temperatures the valve will be triggered early on the return stroke. Variations of this type of system can be developed using mechanical or gas type springs.

2.3.2 Motor driven rotary valve

This cycle is that used by SIG the Swiss company in their range of hydraulic rockdrills. The valve is in the form of a sleeve around the operating chamber of the piston. The sleeve, as shown in Fig 12 on page 111, has a series of radial holes in it which coincide with ports in the cylinder body as the sleeve rotates. These ports are connected to high and low pressure. The sleeve is driven in a rotary motion by a gear reduction mechanism operating from the rotation motor. Thus as the sleeve rotates the holes in it connect the piston chamber alternately with high and low pressure. This alternating pressure drives the piston backwards and forwards against a fixed high pressure acting on the forward smaller area of the piston.

This mechanism results in a very simple piston design as no triggering annulli are necessary on the piston. It does however mean some mechanical complexity is involved to transfer the drive from the forward mounted motor to the valve at the correct ratio for percussion. Also the valve itself is rather large in diameter to fit around the piston. Although this is not a problem with oil operation as on the SIG machines, on water this could result in high leakage rates.

2.3.3 Motor driven reciprocating valve
(Patent No:- SA/85/6786)

An alternative to driving the valve in a rotary motion from the rotation mechanism is to reciprocate it. Here the valve is driven backwards and forwards by a cam and constant pressure loading of the valve onto the cam. The drive from the rotation is less difficult than in the previous case and the valve can be made smaller in diameter. The valve leakage can thus be more controlled making it more suitable for water.

The valve supplies pressure to and from chamber 2 as shown in Fig 13 on page 112. No signal is therefore required from the piston at all. A central seal can be easily incorporated in this design to prevent port-to-port leakage in the unit. Seals at the ends of the piston prevent external leakage. Generous clearance consistent with seal performance can be allowed between the piston and housing at the three diameters. Only three sealing areas are necessary compared with the four required on the hydraulic spring system. Total seal friction will then be less.

Problem areas anticipated with this design are as follows:-

* Wear at the cam valve interface.
* Wear at the cam bearings.
* A reduction of rotation motor speed due to torque demands may lead to overstroking of the piston with consequent damage to the backhead of the machine.
* Practical design problems of packaging the motor and valve with the mechanical cam interface. This will result in added complication compared with other cycles.

2.3.4 Motor driven piston

In this unit a hydraulic motor drives a conventional slider crank mechanism with piston. This piston then drives a free impact piston by means of air compressed between them. In effect this is like a pneumatic drill, with its own built-in compressor driven by a hydraulic motor as shown in Fig 14 on page 112. The hydraulic motor would, by means of a suitable gearing, provide rotation power for the drill string. This mechanism would be similar to that used in some electrically driven jackhammers and paving breakers, namely Kango, Bosch and AEG.

An additional method would be for the crank driven piston to operate the free piston by a water or oil interface. This would require the use of an accumulator in this interface circuit. The accumulator would simulate the air interface so that some form of energy storage could take place between the crank driven piston and the free impact piston. This will ensure that the impact piston still accelerates while the crank piston is deccelerating at the end of its stroke.

The cycle suffers from the weakness of the previous cycle, that the rotation and percussion are interconnected. In practical mining applications this may be a limitation. It does have some advantages however, one of which is the possibility of a more uniform input flow from the pump or power source. Pressure losses in the inlet and outlet hoses will be less cyclic and possibly lower in value. The smoother flows will certainly ensure longer life of hoses and fittings. Other advantages include of course the fact that the valve operating signal does not have to be picked up from the piston.

2.3.5 Orifice bleed valve cycle

(Patent No:- SA/85/2266)

Another type of cycle which is similar to the original standard type of hydraulic cycle is the orifice bleed cycle. Here the valve signal is triggered initially by the forward metering edge of the piston uncovering the valve kicker port as shown in Fig 15 on page 113. This occurs on the return stroke of the piston just as in the standard type of oil hydraulic drill cycle. On the forward stroke the front metering edge of the piston crosses the valve kicker port in the cylinder. As there is no rear triggering edge on the piston, the valve trigger port is closed off except for the orifice bleed to tank. Hence as there is no feed in, the orifice allows fluid to leak away and the kicker pressure will decay letting the valve move across. The orifice is sized to allow the changeover of the valve to occur just after the piston strikes the drill steel.

The cycle although somewhat wasteful of fluid, is independent of the rotation mechanism. It can therefore give the power required whatever happens to the rotation motor speed or pressure.

By experiment and careful sizing of the orifice and the valve operating pistons the cycle efficiency can be optimised. The leakage across the orifice is still less than that which would occur across the piston with the double kicker hydraulic type valve. A seal may be used on the centre section as shown in the diagram. Excess leakage to the kicker port when covered by the piston land is kept low by using one or two holes and no annulus. A close clearance between the piston and housing is however necessary to prevent excessive leakage to the kicker port or ports.

2.3.6 Choosing the cycle

From the range of cycles discussed two were tried. The unit described in 2.3.3 and shown in Fig 13 on page 112, was chosen initially as the one showing most promise as a 5/95 and then water cycle. At that time the cycle in 2.3.5 had not been thought of. It had the potential advantage that the necessary piston design was simple. It could be run in simple bearings, possibly plastic, with large clearances. This would make it easy to manufacture and maintain. Piston diameter concentricity tolerances need not be too tight.

It was decided to use a simple unbalanced three way spool valve design similar to that shown in Fig 5 on page 108. It had pressure, tank and cylinder ports and was hydraulically biased onto the operating cam by means of small piston subjected to supply pressure. Concentricity between the two diameters was essential. With careful manufacture this did not seem unattainable. The cam was an eccentric with a hardened roller running on it using a plastic bearing. A preliminary prototype was made and tested. Although the initial mock-up drill showed promise, in both the penetration rates achieved, flow required and potential reliability perceived, the anticipated sensitivity to valve timing and motor speed soon became evident.

The orifice bleed cycle described in 2.3.5 was proposed at this time and seemed to offer a quicker and more acceptable solution. Its independence from the rotation part of the drill was seen as a big advantage. The higher potential leakage was a limitation but not an insurmountable problem. By careful detail design the leakage could be reduced to acceptable levels. The motor driven valve unit was shelved and work was started on the orifice bleed unit.

It was seen that at a later date when time permitted, the cam operated valve principle should be reassessed. The potential of this mechanism was seen when the percussive mechanism, including valve, was run on water for an hour or so with no apparent ill effects

2.3.7 Piston design

The design of the piston will depend on which type of operating cycle is used. One thing will be common however and that is the need for some kind of damping at the stroke ends. Damping on the forward or impact stroke is essential to prevent overstroking and impacting of the piston into the housing if the drill is run without the drill steel or inadequately thrusted. It is optional on the return stroke, but may be employed as a safety measure in case the valve malfunctions. In the case when the valve is driven by the rotation motor through a cam, it may be again essential. If the rotation motor stops so will the valve operation. Under such circumstances the valve may not operate in time, or not at all on the return stroke, with resultant damage to the housing if no effective damping chamber is used. It is however important that the damping areas are positioned well clear of the normal working stroke range of the piston. If this is not achieved then some restriction on piston movement will be introduced by the damping with a resultant decrease in machine performance. Although this may seem small at the time, an accumulation of these sort of losses can be significant. Attention to detailed design features is important for good overall performance.

3 THE COMPUTER MODEL

3.1 The Empirical Method

Before looking in detail at the equations of motion necessary to form a computer model let us first look at methods used initially with hydraulic drills. A lot of design work on hydraulic drills was done using an empirical method. The work of Pfleider and Lacabanne and later Hustrulid et al on pneumatic drills could have been used on hydraulics. Some modification of the constants would have been necessary, but satisfactory overall results would have been achieved. This was certainly the case at Seco in the early work on the HS600 oil hydraulic drill.

Using this method we have :-

$$E=C_1sAp \tag{3.1}$$

where
$A = (A_2-A_1)$

$$f=C_2(sA/pM_p)^{0.5} \tag{3.2}$$

where
p = supply press p_s
s = nominal stroke
M_p = piston mass
C_1 = Energy constant
C_2 = Frequency constant

Then if the total percussive power is P in kilowatts we have :-

$$P=Ef$$

or

$$P=C_1C_2s^{0.5}M_p^{-0.5}p^{1.5}A^{1.5} \tag{3.3}$$

These are exactly the same as the equations established for pneumatic machines where a range of values for C_1 and C_2 were given. The most suitable values for pneumatic machines designed by the author have been $C_1=0.46$ and $C_2=0.26$.

When dealing with hydraulic machines the best values for these constants have been found to be $C_1=0.6$ and $C_2=0.27$. The equations for the energy per blow and frequency were established using the fundamental mechanical parameters of the machine such as piston mass, stroke and effective area. The constants C_1 and C_2 allow for pressure drops in the system and for friction losses. In other words they are correction factors for the unknowns in the drill, which until the advent of computers would have been almost impossible to calculate with any degree of accuracy. When designing machines of similar geometry, ie, ratio of A_2 to A_1 and similar striking velocity, fairly accurate correlation of energy per blow and frequency can be expected once C_1 and C_2 have been established. Changes in the fundamental ratios however cannot be done without affecting the accuracy of the calculated power output. New values of constants C_1 and C_2 would then need to be estimated. Such parameters as valve timing, valve overlap, accumulator volume, pressure variations, case reaction have to be established by experimentation. This can be very time consuming and hence costly.

Another limitation of the empirical method is that it gives the conditions at the moment of impact. That is the blow energy and hence the piston velocity and machine frequency. It cannot provide values for intermediate piston velocities at any point on the piston stroke. Similarly it will give no information regarding the return stroke velocities. The effect of system variables such as feed and return hose length, static head and fluid viscosity cannot be calculated.

What is immediately evident however is that within reason:-

(a) The energy per blow is independent of the piston mass. Thus as the mass is decreased the velocity increases to compensate and so keeps a constant energy.

(b) The input pressure and effective piston area are the most significant factors in percussive drill design. The overall output depends largely on these parameters as shown in equation (3.3). The power output varies as the pressure and piston area to the power 1.5. Thus if either are doubled the power increases by a factor of 2.83.

(c) The piston mass and stroke have only a minor effect on the overall drill performance. The power output varies as the square root of the stroke. It also varies as the inverse of the square root of the piston mass. Therefore changes in these variables give a minimal change in output power.

It can be used with limited effect as a design aid and applied using a simple calculator. This in conjunction with calculations on valve triggering points and flows can result in a machine which will function, but requires an involved development programme on expensive prototypes. Optimum performance is rarely achieved because the necessary time and money for development is not available.

3.2 System Losses

As discussed above, the empirical approach can be used to give a reasonably accurate estimate of performance. Energy per blow and frequency calculated in such a manner will be usable, but some previous experimental work on a similar drill layout will have been necessary. Any changes to that basic layout will render the empirical constants useless.

In order to circumvent the shortcomings of the previous method, it was decided to write a computer program in Hewlett Packard Basic. This language was chosen for several reasons. It is a powerful basic language with good graphics facilities that is very suitable for engineering work. It is relatively easy to learn and an HP9816 computer was readily available.

Before commencing the program, equations of motion for the main parts of the drill were necessary. These were of course the piston, the valve and the accumulator. In order to do this with some degree of accuracy the various losses that would be incurred in the system had to be known or estimated.

3.2.1 Summary of possible losses in the system

Before formulating the equations of motion for such a system as shown in Fig 15 on page 113, it will be best to consider what possible losses can occur. These can be listed as follows:-

1. Fluid friction, orifice and bend losses in the high pressure feed hose from the supply pump to the drill plus any adjustments for static head differences.

2. Hydraulic losses due to sudden enlargements in the internal ducting of the drill, for example, where conduits enter chambers 1, 2 and the accumulator.

3. Hydraulic losses due to sudden contraction in the internal ducting of the drill, similar to 2 where conduits exit from chambers 1, 2 and the accumulator.

4. Hydraulic losses at bends and elbows.

5. Pipe friction losses in internal conduits

6. Fluid inertia losses which add to the effective mass of the piston and valve.

7. Orifice losses due to throttling which occur mainly at the valve.

8. Piston friction losses. These can be separated into three types :-

(a) Seal friction due to elastomeric external seals.
(b) Viscous friction due to shearing of the fluid film.
(c) Mechanical friction losses from metal-to-metal contact.

9. Pressure losses due to the discharge of the accumulator when the fluid flow demanded by the piston exceeds the inflow to the drill from the supply pump. This shortfall is made up by the fluid stored in the accumulator and the pressure falls accordingly.

10 Friction at the accumulator piston and seals.

11 Accumulator piston mass

12 Compressibility effects of the fluid, casings and hoses.

13 Fluid friction, orifice and bend losses in the return hoses to the tank plus any adjustments for static head differences.

Appendices B and C list hydraulic losses in greater detail.

3.2.2 Fluid losses in the pressure line

The losses in the supply line will be mainly fluid friction losses. Any orifice or bend losses will be small in comparison. They are also very specific to the way the hydraulic system is piped up. In trying to account for every elbow, connection and fitting the computer simulation will become too complex and slow. The objective is to provide a model which will be easy to use and provide a useful design tool if standard hydraulic design procedures are used. Therefore only the length and diameter of the feed hose are considered necessary. The length and diameter of a typical hose would be 20m and 13mm. The equations used for these calculations are set out in Appendix B on page 65.

When simplified for use in the model the pressure drop in the line is:-

$$p_d = Klwv^2/2d \qquad (3.4)$$

Here $K = 0.025$ for practical purposes in industrial hosing, see Guillon (Ref No 7).

3.2.3 Internal leakage losses

All of the equations for a drill will be modified by the inclusion of flow losses and additions due to internal leakage. The calculation of such leakage can and should be included in the program. It will be done in a similar step by step method to the other calculations. At best however it can only be used as a guide and must be compared with final machine flows and sub-assembly leakage tests for accuracy. The problem with leakage calculations is the number of potential leakage points, the dependence on temperature, and of course viscosity. Clearance is also very important particularly in laminar flow leakage through small annular gaps. Here the leakage varies as the cube of the clearance. A small error in clearance or the presence of ovality, lack of parallelism and concentricity will have a great effect on the calculated leakage.

3.2.4 Orifice losses in the valve

The pressure losses at the valve are similar to orifice losses. The equation used for this is shown below and is that shown in more detail in Appendix C on page 67.

$$p_d = 2.68wv^2/2 \qquad (3.5)$$

It is assumed that the coefficient of discharge is the same for flow into and out of the working chamber 2 of the piston shown in Fig 4 on pages 106 and 107. It is also assumed that the coefficient is the same for all flows. This is a simplification but is justified.

3.2.5 Piston Friction Losses

The losses due to the external seals at either end of the piston are as shown in detail in Appendix D on page 69. They are as shown below:-

$$Fs = (p_1F_{S1} + p_2F_{S2}) \qquad (3.6)$$

The viscous shear losses for the piston are as follows:-

$$Fv = 0.6v \qquad (3.7)$$

The detailed calculations for this are shown in Appendix D on page 69.

Although in addition to the friction calculated above metal-to-metal friction must occur, no reliable method of calculating this was found. It was therefore neglected.

3.2.6 The Influence of the accumulator

The supply pressure at the entrance to the drill is p_s. Fluid is either supplied by a constant volume pump or by a variable displacement pressure compensated pump. If a constant volume pump is used then the pressure p_s at the entrance to the drill will be set by the characteristics of the drill. These are determined by such parameters as piston mass, stroke and operating areas. It will also be set by external factors such as the length and diameters of the inlet and exhaust hoses. Pressure variations at the pump and entrance to the drill will take place during the percussion cycle. These will depend on the position and velocity of the piston at that moment during the cycle. The pump under these conditions will try to maintain a constant flow.

If a constant pressure pump is used then it will try to maintain a constant pressure throughout the drill cycle. The mechanical inertia of the pump compensating mechanism is assumed to be too large for it to respond to pressure variations in the drill cycle. (The frequency of the drill cycle being of the order of 50 Hz or more.) It can only respond to the average flow demand over several cycles maintaining a constant average pressure.

An accumulator is necessary in the supply side of the drill to absorb flow when the drill demand is low and supplement pump flow when the drill demand is high. Usually the initial charge pressure is set at some proportion of the supply pressure. The charge pressure would be Rp_s where R is some ratio, typically 0.33.

The initial gas or total volume of the accumulator = V_A

The interface membrane between the charging gas, nitrogen, and the fluid will normally be rubber or some other flexible medium. Its mass is considered insignificant relative to the forces involved and has been neglected. If a piston type accumulator were used then the mass of the piston and the friction involved may be significant and should be included.

Now consider the flow conservation equation at the accumulator over a small increment of time dt. If V_A is the initial gas volume in the accumulator before the drill is started and hydraulic pressure applied to the other side of the diaphragm then the equation for the hydraulic fluid entering the accumulator is

$$V_{Af} = V_{Afo} + (1000Q/60 - A_1U)dt \qquad (3.8)$$

The flow into or out of the accumulator is then

$$Q_A = V_{Af}/dt \qquad (3.9)$$

The pressure in the accumulator then becomes

$$p_A = Rp_s(V_A/(V_A - V_{Af}))^n \qquad (3.10)$$

3.2.7 Drill string strain effects

When subjected to the impact of the piston, the drill string will deflect. This will occur according to Hookes Law and through buckling as a strut.

Consider the piston striking the end of the drill steel at a velocity of v m/s and let the longitudinal deflection be x. Then if P_o is the equivalent static load to cause a deflection x in the drill steel we have :

The loss in kinetic energy of the piston = The gain in strain energy of the drill steel.

$$1/2Mv^2 = 1/2Px$$

But

$$x = P_o l/AE$$

$$1/2Mv^2 = 1/2P_o l/AE$$

$P_o = (Mv^2AE/l)^{0.5}$

The equivalent static load can be calculated from the impact energy of the piston. The deflection of the steel can be calculated by using

$x = P_o l/AE$

$x = l/AE(Mv^2AE/l)^{0.5}$

Therefore when considering impact only.

$$x = (Mv^2l/AE)^{0.5} \tag{3.11}$$

Note that the Euler crippling load for a strut hinged at both ends is

$P_e = \pi^2EI/l^2$

When considering the combined effects of the piston impact and the continual drill thrust force P_F we have :-

$P_T = P_i + P_F$

$$x = l/AE((Mv^2AE/l)^{0.5} + P_F) \tag{3.12}$$

It will be seen later however that the value of P_F is small in relation to P_i. It will also be seen that for a drill steel of average length of 1.4 m such as used in stoping, the impact load P_i exceeds the Euler crippling load by a considerable amount.

3.2.8 Fluid losses in the return line

The losses in the return line are similar to those incurred in the pressure line. They are again treated as mainly fluid friction losses. The equations used will be those shown in Appendix B on page 65.

$$P_d = Klwv^2/2d \tag{3.4}$$

Where K = 0.025 as used before.

Losses in the return line are particularly significant when related to drill efficiency. High instantaneous flow rates can be recorded because of the relatively large operating area A_2 on the piston. In addition because of this large area any pressure increase through excessive back pressure during exhaust will cause large forces on the piston. These forces will act against the motion of the piston, causing loss of energy on the return stroke. Usually a larger diameter hose is used on the return side of the drill to the feed side for this reason.

3.2.9 Compressibilty losses

These will occur throughout the system where there are pressure changes in the fluid. The equation for flow due to fluid compression is :-

$$Q_c = (V/B)dp/dt \qquad (3.13)$$

Where
V = Fluid volume in the chamber concerned
B = The fluid bulk modulus
dp/dt = Rate of change of pressure

The value used is that given by Merritt (ref No 14) as a practical value for normal industrial circuits, that is 689MPa.

3.3 Forming the Equations

Consider the drill shown in Fig 3 on page 105 as an example. Let us use the convention that force, displacement, velocity and acceleration to the right are positive. To the right in this case being towards the drill steel or point of impact.

The following equations can be written for the system shown in Fig 4 on page 106.

Equation of motion of the piston by Newtons Second Law is:-

$F = M_p dv/dt$

This becomes :-

$$p_2A_2 - p_1A_1 + F_s + F_v = M_p dv/dt \qquad (3.14)$$

F_s = the seal friction force on the piston

F_v = the fluid viscous friction force on the piston

Equation (3.14) becomes :-

$$dv/dt = (p_2A_2-p_1A_1+F_s+F_v)/M_p \qquad (3.15)$$

This equation can be used to determine the velocity and displacement of the piston over small increments of time using the steady state equation,

$v = u+(dv/dt)dt$

Therefore the general equation for the piston velocity becomes:-

$$v = u + dt(p_2A_2-p_1A_1+F_s+F_v)/M_p \qquad (3.16)$$

Where u is the initial velocity at the beginning of a small increment of time dt and v the final velocity.

It is assumed that if dt is sufficiently small then dv is approximately constant over that period of time. The velocity can then be calculated incrementally and the piston displacement can be calculated using the equation :-

$$s = s_o + (v+u)dt/2 \qquad (3.17)$$

Where s_o is the initial displacement at the beginning of the small increment of time dt and s the final displacement.

The pressures p_1 and p_2 applied to the areas A_1 and A_2 will be modified, ie, increased or decreased by the pressure losses in the system.

In order to analyse the orifice delay cycle, such that the equations can be written, it is probably best to consider its operation in four phases. These are shown in Figs 16, 17, 18 and 19 on pages 115 and 116. Illustrated clearly in these diagrams are the piston and valve positions and the pressures in the various operating chambers and conduits. Also shown are the main areas of fluid leakage. As will be shown later leakage plays an important part in the dynamics of the hydraulic drill. It is very difficult and in fact almost impossible, to provide a computer model that is so general that it may be used for any type of drill. The model produced in this excercise is no different. It is made to suit the standard type of hydraulic cycle used on most drills currently in use. It has been adapted to suit the orifice delay type cycle used in the HD30 drill.

In order to be fairly general therefore some assumptions have been made. These are:-

* The supply hose from the hydraulic pump is of uniform section with no bends, elbows or fittings etc.

* The pressure in chamber 1, p_1 is the same as the hydraulic pressure at the accumulator p_A.

* The dimensions of the internal ports of the drill are sufficiently large in diameter and short in length that the pressure drops through them are small in comparison to the losses at the valve and in the feed and return hoses.

* The return hose to the tank is of uniform section with no bends or fittings (as for the feed hose.)

The model was produced to be of use in the design and performance prediction of drills to be used practically in the mining industry. It's performance prediction accuracy in both energy per blow, frequency and flow requirements is therefore aimed at being within 5% of that required. In order to achieve this, the program can accept the normal variables that may be necessary or desirable in a machine. The following parameters can be changed in the program and a "what if?" design situation can be evaluated. The parameters are :-

* Piston mass M_p
* Piston operating areas A_1 and A_2
* Piston clearance D_{pc}
* Valve triggering point S_1
* Valve mass M_v
* Valve operating areas A_{v1} and A_{v2}
* Bleed orifice diameter A_o
* Valve diameter D_v
* Valve stroke S_{vt}
* Valve clearance D_{vc}
* Drill inlet and chamber 1 pressure p_1
* Accumulator volume V_A
* Accumulator charge pressure ratio R
* High pressure supply hose diameter and length D_s, L_s
* Low pressure return hose diameter and length D_r, L_r
* Rotation motor flow Q_r
* Pump constant flow Q_p

3.3.1 Phase 1

If phase 1 is now considered it can be seen from Fig 16, on page 115, that the piston is starting on the return stroke from the impact position. Pressure is acting on the area A_1 in chamber 1. A flow Q_p at a pressure of p_p is being supplied by a fixed volume pump through the feed hose of length L_s and diameter D_s. The piston starting from rest will acquire a velocity v which is by the convention described earlier, negative.

The general equations of motion for the piston derived earlier can be applied here.

$$v = u+dt(p_2A_2-p_1A_1+p_2F_{s2}+p_1F_{s1}+F_v)/M_p \qquad (3.16)$$

$$s = s_o+(v+u)dt/2 \qquad (3.17)$$

In this case because the velocity of motion is negative all the positive terms in the brackets oppose the motion. This is as it should be because these terms include the opposing pressure p_2 on area A_2 in chamber 2. It also includes the seal friction terms p_1F_{s1} and p_2F_{s2} for the seals at the outboard ends of chambers 1 and 2. An additional term for fluid shear F_v is added. This will include fluid shear and friction at the centre and both ends of the piston. As will become apparent friction is a very significant parameter in this analysis.

These equations are the basic equations relating to piston motion. Within them some of the parameters will be modified by the ongoing dynamics of the system. p_2 will be modified by the restriction at the spool valve to the fluid exhausting from chamber 2 and by the pressure p_4, downstream of the valve in the return hose cause by fluid friction in the hose. Any additional flow to that displaced by the piston will have a significant effect on the pressure drops in these areas. These additional flows will be leakage at the piston, Q_{p1}, the valve, Q_{v1}, the orifice, Q_{o1}, and the additional flow of the rotation motor, Q_{rot} which exhausts into the common return line. Let us then consider the flow equations for the drill :-

Total flow in return hose = Sum of all flows from the drill

$$Q_r = Q_{p1}+A_2v+Q_{v1}+Q_{rot}+Q_o \tag{3.18}$$

The flow through the valve is

$$Q_v = A_2v+Q_{p1}+Q_{v1} \tag{3.19}$$

Using the formula for the pressure drop across the valve from Appendix C, on page 67, we have:-

$$p_2-p_r = 2,6w(Q_v/A_v)^2/2 \tag{3.20}$$

and for the pressure losses in the return hose

$$p_r = 0,026L_rw(Q_r/A_r)^2/2D \tag{3.21}$$

In order to calculate Q_r and Q_v the leakage flows Q_{v1} and Q_o must be determined. The rotation flow is one which will be known for the particular drill to be used and will be entered along with the other drill parameters before running the program. In the case of the HD30 Q_{rot} is 13,4 l/min and remains fairly constant over the pressure range as shown in the flow control curve in Fig 28 on page 121.

Q_o is a straight orifice leakage flow. The orifice in this case is not a true sharp edged orifice but a short tube. The coefficient of discharge would be appproximately 0,816 as shown in Merritt (Ref No 14).

$$Q_o = C_d A_o (2p/w)^{0,5}$$

$$Q_o = 0,816 A_o (2(p_1 - p_r)/w)^{0,5} \qquad (3.22)$$

The leakages Q_{p1} and Q_{v1} through the piston and valve clearances are treated as simply laminar flow between concentric cylinders. It was assumed that because there is relative motion between the parts that they would be approximately concentric.

$$Q = 15708Dpc^3/60lvw \ (l/s) \qquad (2.1)$$

The value for p_2 now becomes:-

$$p_2 = p_3 + p_4$$

$$p_2 = 1,34w(Qp/Av)^2 + 0,026L_r w(Qr/Qr)^2/2D \qquad (3.23)$$

The input percussive flow equation for phase 1 is as follows:-

$$Q_{p1} = A_1 v + Q_{p1} + Q_a + Q_{v1} \qquad (3.24)$$

The output percussive flow equation for phase 1 is :-

$$Q_{p2} = A_2 v + Q_{p1} + Q_{v1} \qquad (3.25)$$

The equation of motion for the valve in this phase is similar to that for the piston. The valve will in the early part of this phase be held over to the right so that chamber 2 is open to the return line. Operating area A_{v1} will be subjected to pressure p_1 and area A_{v2} to pressure p_r. This will continue until the piston moves far enough for the edge on area A_1 to uncover the kicker port to the valve operating area A_{v2}. The valve will then start to move over.

The valve equations are:-

$$v_v = u_v+dt(p_1A_{v1}-p_rA_{v2})/M_v \qquad (3.26)$$

This will be the case until the kicker port is uncovered. Once the port is uncovered then the equation becomes :-

$$v_v = u_v+dt(p_1A_{v1}-p_1A_{v2})/M_v \qquad (3.27)$$

$$s_v = s_{vo}+(v_v+u_v)dt/2 \qquad (3.28)$$

The reaction on the drill casing at this point will be equal to the net force acting on the piston which is:-

$$F= p_2A_1-p_1A_1+p_2F_{s2}+p_1F_{s1}+F_v \qquad (3.29)$$

This reaction force is towards the drill steel and rock and will be fed into the drill steel through the collar on the steel. It will not be seen as a reaction force against the thrust mechanism.

3.3.2 Phase 2

This phase, shown in Fig 17 on page 115, starts as soon as the spool valve changes past the midpoint to direct high pressure fluid to chamber 2. The piston itself is still travelling backwards away from the impact position. Its velocity is therefore negative. The acceleration in this case will now be positive as the net force on the piston will be to the right.

The equation of motion is then:-

$$v = u+dt(p_2A_2-p_1A_1+p_2F_{s2}+p_1F_{s1}+F_v)/M_p \qquad (3.16)$$

$$s = s_o+(v+u)dt/2 \qquad (3.17)$$

These are the same as for phase 1 but the velocity v is becoming less negative as the piston slows down ready to accelerate forward on the impact stroke. The chamber 2 is now connected via the valve to the main high pressure system and not the low pressure return. The pressure loss across the valve will cause p_2 to be higher than the supply pressure to the drill. p_2 then becomes:-

$$p_2 = p_1+p_3 \tag{3.31}$$

$$p_3 = 1{,}34w(Q_v/A_v)^2 \tag{3.32}$$

$$Q_v = A_2v+Q_{p1}-Q_{v1} \tag{3.33}$$

$$Q_{v1} = 15708Dv(p_2-p_1)c^3/60lvw$$

$$Q_{p1} = 15708Dp(p_2-p_1)c^3/60lvw$$

The reaction on the drill casing at this point will again be equal to the net force acting on the piston.

$$F = p_2A_1-p_1A_1+p_2F_{s2}+p_1F_{s1}+F_v$$

This force is directed at the thrust mechanism and must be cumulatively calculated over the whole cycle to give the net force required by the drill during drilling.

3.3.3 Phase 3

As the cycle moves into phase 3 as shown in Fig 18, on page 107, the only change is in the velocity, which now becomes positive. The forces acting on the piston are the same as phase 1 except for those relating to friction which will now change sign.
The equation of motion now becomes:-

$$v = u+dt(p_2A_2-p_1A_1-p_2F_{s2}-p_1F_{s1}-F_v)/M_p \tag{3.40}$$

$$s = s_o+(v+u)dt/2 \tag{3.17}$$

The case reaction being:-

$$F_r = (p_2A_2-p_1A_1-p_2F_{s2}-p_1F_{s1}-F_v) \tag{3.41}$$

Now p_2 becomes lower in value to p_1 because the pressure drop across the valve acts to reduce it.

$p_2 = p_1 - p_3$

The valve will start to change over when the piston forward metering edge has passed the S_1 stroke position and covered the kicker port. The equation of motion of the valve is :-

$$v_v = u_v + dt(p_1 A_{v1} - ((A_{2v} v_v / C_d A_o)^2 w/2 + p_r))/M_v$$

$$s_v = s_{vo} + (v_v + u_v)dt/2$$

The leakage flow equations will be the same as in phase 2.

3.3.4 Phase 4

The cycle now passes into phase 4 shown in Fig 19 on page 107. At this point the piston either strikes the drill steel when drilling or runs into the cushion if no drill steel is present. First consider the drilling mode. At the point of impact the piston hits the steel and gets a reactive force from it. This force slows the piston down to zero forward velocity and the residual spring in the steel starts to accelerate it back on the return stroke. While this takes place the valve changes over to vent the chamber 2 to the return hose. This starts as soon as the piston has closed off the valve trigger port on the forward stroke.

The equations now become:-

$$v = u + dt(p_2 A_2 - p_1 A_1 - p_2 F_{s2} - p_1 F_{s1} - F_v - S(s-35))/M_p$$

$$s = s_o + (v+u)dt/2$$

With the piston running into the cushion, the resistance term used for the drill steel will be changed to that for the cushion area A_3. This is as follows :-

$$Q = 15708pD_3c^3/601wv$$

But

$Q = A_3 v_p$

Thus

$p = A_3 v_p 601wv/15708D_3c^3$

$F_c = pA_3$

$F_c = A_3^2 v_p 601wv/15708D_3c^3$

The equation for velocity then becomes :-

$$v = u+dt(p_2A_2-p_2A_{12}-p_1F_{s1}-F_v-F_c)/M_p \tag{3.44}$$

The displacement equation will be the same as before. In both cases the piston, whether on the steel or cushion, comes to a stop. It then starts another cycle by repeating phases 1,2,3 and 4.

4.0 DESIGN

The following preliminary design specification was formulated to cover the type of drill for use in underground narrow reef stopes in South Africa.

4.1 Specification

Drill mass	30 kg
Operating pressure	16 mPa
Maximum back pressure	2 mPa
Normal back pressure	1.2 mPa
Energy per blow	80 Joules
Frequency	50 Hz
Power output	4 kW
Percussive and rotation flow	55 l/min
Drill thrust	1.8 kN
Rotation torque max.	80 Nm
Rotation speed	250 rpm
Rotation power max.	3 kW
Rotation flow	13.5 l/min

4.1.1 Percussion

Initially the cam driven valve cycle was chosen for percussion. A prototype was built and run. The results of drilling tests done on this drill are shown in Appendix B on page 65. Initial valve timing was estimated from the predicted velocity displacement curve. The performance of the unit on the first runs was slightly less than the required performance. Efforts to change the timing of the valve to improve the performance resulted in a drop in output. It was discovered that the cycle was very sensitive to valve timing. Just a small change resulted in a big change in penetration rate. No improvement could be achieved by changing the timing either way.

Because of this and the necessity with such a system of having percussion and rotation mechanically linked, no further work was done on this cycle. Work was transferred to the orifice delay cycle which offered the potential advantages listed earlier.

4.1.2 Piston assembly

Using the computer program in conjunction with a design layout, a piston mass of 2 kg was the result. The ratio between A_1 and A_2 on the piston was adjusted to give the required energy per blow and frequency. As can be seen in Fig 4 on page 106, the smaller the value of A_1 the higher will be the overall cycle efficiency. The ratios of A_2 to A_1 should be kept as large as possible. Unfortunately as the ratio increases so does the sensitivity to return line back pressure. Because A_1 is relatively small the return force is low. Any opposing force such as friction or return line pressure acting on area A_2 will reduce the nett return force on the piston considerably. Seal friction and in fact any friction will therefore increase the back pressure sensitivity of the drill.

The use of a return line accumulator built into, or close to the outlet port of the drill will help to reduce high and low pressure peaks. This will raise the effective back pressure at the drill that can be tolerated. It was however decided not to use a built-in low pressure accumulator because of increasing the machine mass.

The piston was designed using the diameters d_1, d_2 and d_3.2

This gave

$A_1 = (d_2^2-d_1^2)/4$

$A_1 = (37.15^2-36^2)/4 = 66.069$ mm^2

$A_2 = (d_2^2-d_3^2)/4$

$A_2 = (37.15^2-32^2)/4 = 279.698$ mm^2

In order to ensure that the piston does not strike metal-to-metal in the mating sleeve when not drilling, a damping collar is incorporated. This collar was designed at 44 mm diameter. A corresponding damping chamber is designed in the bronze sleeve. The radial clearance of 0.001mm and damping chamber length of 11 mm ensure that the piston is brought to rest in a controlled manner if the drill steel striking face is not kept in the correct position. This could occur if the drill is underthrusted or if the machine is run for any reason without a drill steel.

A damping chamber is also included on the rearstroke of the piston as a safeguard. This will only be operative if the valve malfunctions through the ingress of dirt or foreign matter. Because the return velocity of the piston is less than half that of the forward velocity the energy will be less than a quarter of the impact energy if the valve does not function at all. The length of this damping chamber can therefore be much shorter than the one on the forward stroke.

The damping effect of the forward chamber can be seen in Fig 4 on page 106, when the drill is run at the full pressure of 16 MPa with no drill steel in place.

It can be seen that under these conditions the piston stops some 2mm before the end of the damping chamber. Clearances between the damping collar on the piston and the mating chamber are critical. Also important is the maintenance of concentricity on the piston collar and damping chamber diameter. Clearance between the piston diameter of 37.15mm and the sleeve forward of the damping chamber is also critical.

A case hardening material EN 39B was chosen for the piston because of its high mechanical strength and shock resistance, combined with a surface hardness of some 60 HRc. This when used with a case depth of some 2mm gives good resistance to cupping and impact wear on the striking face. The diameters were chrome plated to resist corrosion and keep the seal friction and wear low.

The mating sleeve was made in a PB1 phosphor bronze as this is a good bearing combination with the chrome plated hard piston. The hydraulic seals used were initially standard seals called Stepseals supplied by Shamban in a filled PTFE material. Later UHMWPE seals were used. The seals were discussed earlier in the section on seal friction.

The valve kicker ports in the sleeve were kept to two 3mm diameter opposing holes. This was done instead of the usual annulus, to reduce the amount of leakage down the sleeve from high pressure to low pressure. Hence the leakage from the high pressure chamber to the kicker ports is kept to a minimum. Two ports were used to try and balance the hydraulic forces on the piston and reduce side loading.

Initially balance grooves were not used on the piston as a central seal in the sleeve was used. Unfortunately the extra friction caused by such a seal gave starting problems and erratic running at low operating pressures.

4.1.3 Valve assembly

The valve assembly and its effective operation is one of the major components in a percussive rockdrill. The design chosen was a pressure biased valve that was kicked against the pressure bias by a kicker pin of twice the effective area of the bias pin. In effect the operation is similar to that of the piston except that the area ratios are 2:1 instead of the 5:1 of the piston.

The valve type used was a three way spool valve. A total travel of 6 mm was used giving a symmetrical 3 mm opening each way. The opening of 3 mm on the 20 mm diameter chosen was not necessary for adequate flow. A 3 mm overlap was however chosen to give adequate sealing lands in either direction.

A simple single diameter valve was chosen to run in a replaceable sleeve shrunk into the cylinder body. The two operating areas were designed in the form of small pistons. The constant pressure bias piston ran in a bore machined in the low pressure end of the spool. The larger area kicker piston ran in a bore machined in a small separate kicker cylinder. This was fitted into the cylinder coaxially with and at one end of the valve sleeve.

The single diameter spool and sleeve with open ends lent themselves to easy manufacture, using grinding or honing techniques. Some problems were initially experienced however with the small diameter bias and kicker piston bores. Both bores were in hardened parts and really required to be internally ground.

A size to size condition was aimed at with the spool operating head length and the matching annulus in the sleeve. This was intended to give a zero overlap or underlap condition. As a result in practice on extremes of tolerance the fit varied from small underlap to small overlap. A problem with this was that as the spool metering land crossed the sleeve annulus in the underlap condition it could catch slightly in the annulus. This resulted in very rapid rounding of the metering edges with the low viscosity fluid used. On similar valves run on oil this wear did not occur until after several hundred hours of running. The solution to this was to introduce three small support lands either side of the metering diameter.

This length of land allowed for some wear on the spool before leakage became excessive. The spool diametral clearance chosen was 0.01 mm. The leakage either way when the spool was closed would be :-

$Q = 15708\ pdc^3/lvw$

If

$p = 16mPa$

$d = 20mm$

$l = 3mm$

$c = 0.01mm$

$v = 0.663cs$

$w = 1$

Then

$Q = 2.53\ l/min$

The kicker and bias pin diameters were taken from the computer program results. The delay orifice diameter was also taken from trial runs with the program.

No seals were used anywhere on the valve in the interests of low friction and low hysteresis. The rear end land on the valve was made relatively long and was made with pressure balance grooves. This kept the leakage values low and ensured coaxial running while reducing the tendency to hydraulic lock.

The following check was done on the dynamic forces on the valve during operation.

4.1.4 Accumulator

Of the three well known types of accumulator, diaphragm, bag and piston, the piston unit was chosen as the most suitable for this particular drill. The reasons for its choice were mainly size and ease of fitting into the profile of the drill. The piston was made from aluminium and hollowed out to keep its mass low. Two step seals and an AQ type Shamban seal were used as shown in Fig 21 on page 117. A high surface finish was maintained in the bore of the accumulator cylinder in the rockdrill body.

The advantages of the piston type were its small diameter profile for a given capacity. This was in contrast to the relatively flat large diameter of the diaphragm type. This would have been difficult to fit easily into the drill without a large bulge and ugly retaining bolts on the side of the drill

The disadvantages of the piston unit is its non-positive seal, which can cause leakage and loss of performance, and the mass of its piston. The piston mass gives the unit inertia which will be seen as pressure peaks in the operating pressure line. The diaphragm unit has no moving seals and therefore unless it is punctured it will not leak. Its nitrile rubber or polyurethane construction combined with small movement, means low effective inertia and hence low pressure peaks and losses.

The narrow bag type unit shown in Fig 22 on page 118, was tried as an alternative to the piston unit. It was designed as a direct cartridge type replacement. It functioned well but suffered from splitting at various times. Small design changes like thickness and form were varied but failures still occurred. This type of accumulator was shelved through lack of development time. If time is available later, further work on this unit could provide a more reliable unit than the piston type.

4.2 Rotation

4.2.1 Torque requirement

The maximum rotational power requirement was estimated from equivalent pneumatic drills already in use. A torque of 80Nm was estimated as that required at the chuck for a drill with a percussive output of 4kW. In rock of the quartzite or norite variety a ratio of percussive frequency to rotation speed of 11 or 12:1 is acceptable from experience. Thus for a percussive frequency of 50 Hz a good approximation for the rotation speed would be 250rpm. Hence for rotation power P_r we have:-

$$P_r = 2\pi NT/(60x1000)$$
$$= 2\pi x250x80(60x1000)$$
$$= 2.09kW$$

Four types of rotation motor were considered for this application. These were the gerotor,vane,ratchet and gear motor. They are shown in Figs 23,24,25 and 26 on pages 119 and 120.

The ratchet motor could be built around the chuck to drive the drill steel directly as shown. It has some limitations however which resulted in it not being used. These were its intermittent operation, large overall diameter and low hydraulic efficiency. Its mechanical and volumetric efficiency is quite high, but because of its intermittent operation a high proportion of its power is wasted resetting the mechanism.

The gerotor, vane and gear motors would be best suited driving the chuck through a reduction gearbox. They would therefore have to be fitted in close to the chuck, preferably below the drill. Of the three the gear motor had a lower profile and so would result in a neater package. It was therefore the unit that was chosen to drive through a 2:1 reduction ratio.

$$pqN/60 = 2.09x1000/0.80$$

$q = 19.59cm^3/rev$ if the mechanical efficiency is 80%

If the volumetric efficiency drops to 65% at full pressure then rotation speed will drop to 250x0.65 = 150 rev/min

A standard gear motor of 19.66cm^3 capacity was used.

4.2.2 Volumetric efficiency

High volumetric efficiency of the rotation motor is desirable. This is not easy with a low viscosity fluid like water. On a gear motor there are several areas where fluid can leak from high to low pressure. These are across the gear teeth and across the faces of the gears. The leakage across the tips of the teeth can be reduced by an accurate fit of the teeth in the body cavity. In addition the hydraulic separating forces cause the teeth to be pressed against the casing thus reducing the clearance through which fluid can leak.

Leakage across the faces of the gears can be reduced by a side plate or plates which can be pressure loaded onto the sides of the gears. The loading is usually proportional to the operating pressure of the motor. These plates are called balance plates. These plates are made of bronze or some other bearing material. The hydraulic pressure is contained at the back of the plate by means of a spectacle shaped seal. This keeps the balancing portion of the high pressure from leaking to low pressure. The degree of balance or the balance force can be set at the design stage. The larger the balance area the larger the force. As the motor in this stoping drill was intended to be unidirectional the balancing was much easier than with a reversible unit.

The two plates, both thrust and balance must be a close fit in the body of the motor. If this is not so then excessive leakage will occur around the outside of the two plates.

In addition to the volumetric efficiency, the mechanical efficiency must be high. For this reason every attempt is made to reduce friction. Needle roller bearings are used in the 5/95 fluid drill. These have given good life in service even with the oil content at 5%. Tests have in fact been done with an oil content down to 2%, with no apparent ill effects. Steel gears running with bronze balance and thrust plates give a good bearing combination. The shaft seal is a PTFE type arrangement to minimize power loss in this area.

Even with all these precautions the overall efficiency is only around 50%. Much more work needs to be done to improve this. It should be more like 80% to be a viable proposition.

4.3 Percussion and Rotation Control

The control of percussion and rotation is carried out by a single manual control valve at the rear of the drill. This valve operates in a machined bore in the backhead of the drill. When the operator rotates this valve fluid is fed into the drill from the main supply hose. It then passes down the main feed duct to both percussion and rotation. It gains access to the percussion through the distributor valve which is driven by the piston. To get to the rotation motor it passes through a simple pressure compensated flow control valve. This valve allows a maximum of 13.4 l/min of fluid to go to the motor. At no load this drives the motor at approximately 250 rev/min. At full load because of the low volumetric efficiency this drops to 150 rev/min or so. The balance of flow is allowed to go to the percussion mechanism.

The flow passed by the valve can be changed by shims under the control spring. A stronger spring giving more flow and vice versa. This valve and its characteristic curve can be seen in Figs 31 and 32 on page 114. The unit is in the form of an easily removable cartridge in case of a malfunction. A simple pressure compensated restrictive type flow control valve was chosen to control the rotation flow only. It had to pass the 13.4 l/min for the rotation motor. A bypass flow control valve would have had to pass the total machine flow of 60 l/min or so. It would therefore have been more than four times the size.

5 TESTS

5.1 Test Facility

5.1.1 Test rig

The test rig, shown in Fig 31 on page 124, comprised a main frame, rail track, feedbeam, feed carriage and thrust cylinder. The whole rig ran on rails transversely in front of a block of Norite granite. The main frame of the rig had four vertical columns in which screw jacks were situated. These screw jacks were synchronized and driven by a hydraulic motor via a chain and four equal sprockets attached to the top of each screw jack. The screw jacks ran in bearings in the main frame and through captive nuts in the supplementary raisable frame. Thus as the hydraulic motor was operated the supplementary frame was raised or lowered in the main frame.

A feed beam was bolted onto the supplementary frame. This beam was perpendicular to the face of the Norite block. On this feed beam was mounted the drill carriage on plain slide bearings. The carriage could be fed towards, or retracted from the rock by means of a 50 mm bore hydraulic cylinder.

The hydraulic drill to be tested was mounted on this feed carriage. The drill steel projected from the front of the drill and was supported at the front of the feed beam by a collaring support or steady. Water was fed into and down the drill steel by means of an external water box in front of the drill. The water exited from the drill steel at the bit end as flushing water. It served to flush out rock chips and suppress any rock dust created during drilling. A 51 mm diameter threaded cross bit was used in conjunction with a 32 mm diameter drill steel. This size was necessary so that the water box could be used. In normal stope drilling a 22 mm hexagonal steel would be used with a 30 to 38mm diameter chisel bit.

5.1.2 Instrumentation

The instrumentation used on the rockdrill consisted of a linear variable displacement transducer and pressure transducers. The displacement transducer was used to measure the position of the piston relative to the drill body. The core of the transducer was attached to the piston and the body to the rear of the rockdrill housing. The signal from this was fed, through an amplifier and signal conditioning unit, to an analogue to digital converter in a P.C.

The pressure transducers were of the piezo-electric type. Five were used in the following positions; behind the piston, in front of the piston, at the valve kicker port, at the machine inlet, at the machine exhaust port and in the thrust cylinder.

A turbine flow meter with a digital readout was used in the pressure feed line to the drill. Thus inlet flow could be monitored at all times. Temperature gauges were fitted in the inlet and return lines to and from the drill.

5.2 Test Procedure

5.2.1 Calibration

The LVDT attached to the piston was used to measure its position relative to the drill casing. On the HD 30, the drill being tested, the overall stroke was 50mm. This comprised 35mm nominal stroke plus 15mm overstroke into the cushion area. The actual values varied slightly from machine to machine depending on the build up of manufacturing tolerences.

With the LVDT movable inner probe securely attached to the piston, the piston was pushed back into the cylinder body as far as it would go. The output value of the transducer was set to zero at this point. A short trace was taken against the time base. This gave a short straight line reading zero stroke. A vernier depth gauge was used to measure the position of the piston at this point. Thus the datum vernier reading was obtained.

The piston was then moved forward 50mm as shown on the vernier. A short trace was taken on the computer. The output from the transducer was set so that the computer read 50mm. By careful adjustment the zero and maximum displacement levels were set.

Calibration was continued by moving the piston back to zero and then forward in 2mm steps measured on the vernier. For each step a short trace was recorded on the computer. From this information a calibration curve of transducer output against measured piston displacement was plotted. This curve should be a straight line. The slight variation from the ideal was calibrated out by using the calibration curve in the computer calculations.

The pressure transducers were calibrated using a Budenburg dead weight tester. Each transducer was connected to the tester. At zero pressure the output shown on the computer was set to zero. The pressure for each transducer was set to the maximum value on the tester. The reading on the computer was then set using the gain and amplitude setting provided on the pre-amp. Once this was done the transducer was taken through its operating range in 2MPa steps on the tester. For each setting a short calibration line was recorded on the computer. After this a calibration curve of computer readout against dead weight tester reading was plotted. This was done for each pressure transducer.

The flow meter was calibrated by running it at set flows of 5 l/min to 80 l/min in 5 l/min steps for one minute time intervals into a calibrated container. This was done on the 5/95 fluid to be used in the development tests. The temperature was controlled at 30 deg Celsius, which was to be the temperature at which the tests were to be run. Both oil and 5/95 were used as the fluid for this test.

The rotation motor dynamometer consisted of a Whichita air brake connected via a strain guage torque transducer to the rotation motor. The dynamometer was calibrated throughout its range 0 to 100 Nm. This was done using a 1 metre torque arm and standard SABS weights. The dead weight torque was increased in 5 Nm increments. The digital readout torquemeter reading was recorded for each increment. A calibration curve of readout versus dead weight torque was plotted.

5.2.2 Drilling

Prior to drilling tests had been done to ascertain the optimum thrust levels necessary for maximum penetration. Set points at regular intervals on the thrust penetration curve were chosen. Thrust levels at these points were selected. Starting at the lowest thrust, the thrust was set. For an inlet pressure of 16 MPa and return line pressure at the drill of 0.5 MPa a hole was started slowly. The drill was then stopped but the thrust was kept on. The drill was then started and as soon as stable conditions were achieved and LVDT traces were taken on the PC. The drill was then switched off.

This procedure was repeated for a whole range of thrust levels from 1000 N to 3500 N. In addition a run with no drill steel in was done for each pressure. The piston under this condition was running in the damping cushion.

5.2.3 Recording

The drill was started so that both percussion and rotation were operating at approximately half power. The feed was then operated to bring the drill bit into contact with the rock. A short hole of about 60mm was drilled while setting the feed thrust to the correct level for the test. The drill was stopped with the feed force still applied. With the computer on and all the transducers connected the drill was then started at full power. As soon as the drill was performing at steady conditions recording was started. When one set of data was taken the drill was stopped. test conditions such as thrust were changed and the drill was started for more data to be recorded. This was continued until the hole could be drilled no deeper and then another hole was started. Usually only a short length of hole of say 100mm was required for each data recording. Hence a metre hole was sufficient for nearly ten points.

5.2.4 Check calibration

With any experimental and test equipment it is important to calibrate before testing and then to check the calibration afterwards. This ensures that if there has been a change of calibration it is discovered so that a retest can be done. With the sort of high freqency high impact equipment being tested here a check calibration is particularly important.

The LVDT was checked in a similar manner to the original calibration. A check was also done that the LVDT probe and casing were still securely attached to the piston and drill casing respectively. Any relative movement in either place would prove detrimental to the accuracy of the results taken.

5.3 Friction Tests

The friction characteristics of the percussion piston seals will have a major influence on the computer model. As has been discussed the seal manufacturers catalogue was used as a starting point to calculate the seal friction forces acting upon the piston. It was decided, however, to confirm these figures by doing some simple test work on the seals.

A simple test rig was designed and manufactured. This is shown in Fig 46 on page 132. It consisted of a vertical pressure cylinder with replaceable caps at either end. These were identical and contained piston bearings and grooves for the seals to be tested. Alternative caps could be used to try different sized pistons and seals. The piston was a simple rod made of case hardened steel similar to the rockdrill piston. A large diameter cap was bolted to the top of the piston to prevent it passing out of the bottom of the cylinder. A weight carrier was attached to the lower end of the piston. A workable travel of 150mm was allowed.

The cylinder was made with a high pressure supply port in one side. Pressure was provided by a small hydraulic pump with a relief valve and pressure gauge. Care was taken after each assembly of the rig to see that all air was bled out before the test was begun. The pressure capability of the pump was 20MPa, so covering the normal range of seal pressures to be used. The complete assembly was mounted on a bench in the vertical position ready for test.

5.3.1 Procedure

Initially the rig was assembled with the UHMWPE seals to be tested. The pressure hose was attached and pressure was brought up to the maximum value to ensure that the seals were fully settled in their grooves. The pressure was then dropped to 2MPa with the piston in its uppermost position. Weights were placed on the carrier until the piston just started to fall downwards. The amount of weight on the carrier and the pressure applied to the seals was recorded. The total mass of the piston and carrier had previously been checked.

The procedure was repeated in 2MPa increments up to 20MPa. The pressure was then reduced in 2MPa increments and the load to move the piston recorded. This was done using 32mm and 36mm diameter pistons.

A plot of load against pressure was then made for one seal. This can be seen in Fig 47 on page 132. The seals used were the standard Shamban stepseal with type 46 material and the UHMWPE seals locally made by Sealrite.

6 RESULTS

6.1 Presentation

The results are presented both in tabular and graphical form. The graphical representation allows a convenient comparison with the graphical output from the computer model. To make this easier the same scales for axes have been used where possible. Some smoothing was necessary and where this was done the unsmoothed curves are also shown. A Fourier algorithm was used in smoothing as this was seen to give best results in de Sousa et al (Ref No 5).

6.2 Accuracy

Although every effort has been made to ensure optimum accuracy the very nature of the project will have introduced certain errors. The development and testing of percussive rockdrills whether air or hydraulically driven imposes very high shock loads on any instrumentation used. Efforts were made to isolate the pressure transducers from both hydraulic and mechanical shocks. The transducers were not screwed directly into the drill housing. They were connected by short lengths of hose to the appropriate tappings in the drill cylinder. The hoses isolated the transducers to some extent from the very high transverse vibrations of the drill body during drilling. In addition the volume of fluid in the hose provided some attenuation of hydraulic peaks which could damage the instrument. In order to provide further protection snubbers were fitted in the end of the transducers.

Both methods of protection for the pressure transducers would also have affected the values of pressure read by these instruments during operation. It must be assumed that a true recording of the conditions in the drill chambers was therefore not received by the computer. The extent of this deviation is not known but is assumed to be small enough to not materially affect the results obtained.

The fluid flow figures recorded are the average values taken over several cycles. The turbine flow meter used was not capable of picking up and recording the variations of flow over a cycle or sequence of cycles. What effect the pressure and flow variations had on the accuracy of the turbine is not clear. Fluid flow was however not one of the parameters requiring great accuracy. It was used merely as a guide to the overall efficiency of the drill.

6.3 Comparison with Computer Model

When comparing the actual experimental results with those simulated on the computer, then the two most important curves are the displacement-time and velocity-displacement curves. These can be of great use in designing new equipment as they present the machines characteristics in such a way that drill power output in terms of frequency and blow energy can readily be obtained.

The displacement-time curves, Figs 33 and 34 on page 125, show the cycle time and overall stroke. It can be seen that while drilling at optimum thrust the computer and test results are close. A cycle time of 18ms was predicted against a measured value of 19ms. The piston travels from 6mm to 41mm in the simulation compared to 6mm to 42mm in the test. Some discontinuity is seen in the simulated displacement curve as the piston approaches the rear of its stroke. This point coincides with the valve changeover and can be seen more clearly in the velocity-time curves on Figs 35 and 36. Here the changeover in the simulation is very abrupt compared to the test curve. It is probably due to lower compressibility and leakage losses on the computer compared to the test model. In practice the valve edges would be slightly rounded and not sharp, thus increasing leakage at the changeoever point.

Although the velocity-time and velocity-displacement curves are rather more angular than the test curves, the values of impact velocity and maximum return velocity are close. They are 8.6m/s and 8.4m/s for computer and test and 4.3m/s for both on return. The return part of the velocity curves appears to correlate better than the forward stroke part. The time for return is almost the same at 10.8ms for the simulation and 11ms for the test curve. The main difference is on the impact stroke at 7ms and 8ms for simulation and test respectively.

It would appear from both the model and test figures that the piston runs partially into the forward cushion volume, which starts at 39mm, when striking the steel. This will obviously cause some piston energy to be lost in the cushion and hence not passed to the steel. In future drill designs the cushion chamber should be ideally moved further forward to see if more power can be developed. The valve changeover is also perhaps a little early and so may contribute to some loss of energy on impact.

One of the problems with the orifice delay type cycle is that a change in orifice diameter or kicker port position usually affects both kicker directions of the valve.

The supply pressure curves, Figs 39 and 40 on page 128 do not compare very favourably at all. All that can be said is that for each cycle both figures two short curves with a sharp lower point. The shape of the two short curves is not the same however. The pressure in Fig 39 does not rise high enough from the point of impact to the valve changeover point compared to the tes results. Conversely it rises higher than the test results in Fig 40 from valve changeover on the return stroke to impact on the forward stroke. Obviously more work is neede here to match the simulation to, actual test results.

The chamber presures in Figs 41 and 42 on page 129 are similar in overall shape but differ in detail. The same problems with the simulation will surely affect this curve.

Fig 43 on page 130 shows the valve changeover in the computer simulation. Unfortunately no transducer could be fitted to the actual valve to give a comparison.

Fig 45 on page 131 shows actual penetration curves of the drill built to the computer simulation. The data taken off the drill was taken at the optimum thrust figure on the 16MPa curve.

7 SUMMARY AND CONCLUSIONS

7.1 Summary

The experimental work and formulation of a computer model for the hydraulic drill were done as part of ongoing development work to produce a practical mining drill. This drill would be an addition to the existing range of pneumatic drills manufactured and marketed by The Steel Engineering Company Ltd. Because of this and the necessary time constraints inherent with such commercially driven exercises an essentially practical solution was sought to a practical problem. The limitations which were perceived with the test equipment had to be accepted because of the environment in which the work was done.

The digitized graphs taken from the prototype were very close in shape to the computer model graphs. The velocity-displacement graphs in Figs 37 and 38 on page 127, show close similarity in the major points such as valve changeover, piston rearmost position, striking point and piston overstroke. Similarly maximum velocities of forward and return strokes are within 2 and 3% respectively.

It can be seen that even when properly thrusted to give optimum penetration the combination of drill string deflection, bit penetration into the rock and drill case deflection allows quite a large piston overstroke. The overstroke is almost as large in fact as when no drill steel is present. The frequency with the steel present is much higher than without, ie, 53 Hz instead of 47 Hz. This is partially due to the spring effect of the drill steel and the drilling system giving a high acceleration to the piston at the start of the return stroke. This can be seen also in the slope of the curve at the start of the return stroke curve, when running on the steel. The slope at this point when running on the cushion is not so large. It can be seen that the high value of seal friction incorporated in the computer model is necessary to match the actual prototype. When the indicated power of the prototype is plotted against actual power the losses through seal friction are again seen to be significant. The values of friction from the prototype are very close to the values used in the computer model. For more details on this see

Appendix B.

When considering the thrust necessary for optimum performance from the computer model the negative part of the curve can be ignored. Only the positive part of the curve was considered. This proved a useful starting point for setting the thrust value of the prototype. The experimental arrangement for applying and measuring thrust was not sufficiently accurate. Piston and seal friction, carriage and guide friction were included in the pressure reading recorded by the transducer.

7.2 Discussions and Accuracy

Although efforts were made to ensure accurate recording of all of the variables measured, some were less accurate than others. The pressure transducers were positioned so that vibration had a minimum effect on their readings. This will have to some extent reduced the accuracy of their readings by introducing some damping due to the extra volume of fluid and by the flexibility of the hosing. This unfortunately was necessary to ensure the life of the equipment for the duration of the test.

7.3 Future Work

Because of time constraints very little work was actually done on the drill using water as the driving fluid. Just some short endurance runs were done to indicate the areas which would need development if the HD30 drill were to be considered for use on water. These results indicated, as would have been anticipated, that the percussion element would run on water with some relatively small changes to cater for the lower lubricity and higher corrosivity of that fluid. The gear rotation motor would on the other hand require a lot of fundamental development to run on water.

Areas on the rotation motor that would certainly have high wear rates on water would be the shaft roller bearings and the side plates, both thrust and balance.

It would be expected in any case that the balance factor on the plates should be changed for water or any low viscosity fluid. In order to ensure adequate efficiency on water then very careful selection of materials will be essential. This must be combined with a design approach which is more suited to water. It is no secret that the drill used as the model in this dissertation was used because it was commercially expedient to do so at the time. A more fundamental research and design approach should however be taken for a straight water unit. The reduction of friction is a most important factor in this design process, both from the mechanical efficiency aspect and of course the wear problems.

Both percussion and rotation parts of the drill therefore require a considerable amount of work to ensure reliable efficient running on water. The two mechanisms described in this document as used on the HD 30 and its prototype may not be the most suitable for water drills. Further research and development work may result in improved cycles and mechanisms which are more appropriate to water technology.

The seals used on the percussive piston currently give an adequate performance and reliability. In order to keep leakage to a low almost zero level the clamping force on the piston is high. It has been seen previously that high wear of both the seal and piston diameters can occur. Much more work needs to be done on the design of high pressure high speed reciprocating seals. The simple test rig used on the seal work gave very useful results, but was obtained too late in the program to be of full benefit.

Improved seals may come from the use of better or different materials. The major improvement will however come from improved design with efforts made to provide partial hydraulic balance as is the case with rotary face seals. This will reduce the contact forces between the rubbing part of the seal and the moving piston or rod. With water an absolute seal is not always necessary. This could result in lower contact forces being required. The problems of back pressure in the return line from the drill discussed in 5.2. can be largely overcome in a water drill. If the water were not used in a closed system but exhausted on to the floor then no return line back to tank would be required. This would reduce the number of hydraulic hoses to the drill and keep the return line pressure to a maximum of the atmospheric pressure. The only pressure drop then would be in the high pressure feed line. The effective pressure across the drill would be higher and hence the efficiency would be higher.

Currently the solution to corrosion protection for 5/95 and certainly water drills in mines is to use stainless steels wherever possible. This results in costly, difficult to obtain materials being used. Cheaper more cost effective solutions may be available using coatings or plastics in some areas. Materials science is advancing rapidly and cheaper materials and more reliable coatings may soon be available.

The conditions under impact when the drill piston is deccelerated rapidly upon impact with the drill steel were not fully recorded and investigated in this work. Better understanding of these conditions could result in a more efficient drill and drill string combination. A lot of work has been done on stresses in drill steels and the action of the bit on the rock. Very little appears to have been done on the interface between the drill piston and the drill steel.

The accumulator is one of the weak points in any hydraulic drill whether on oil or water. There is a need for a more compact reliable accumulator where gas loss is not a problem.

For a hand held drill the reaction force must be a minimum. More work could be done on reducing this force and thus the force to be applied by the operator. Similarly the very nature of this force is intermittent and impulsive. This can over a period of time as already mentioned cause the so called " White Finger Desease" or Reynards Desease. More work on the drill reaction characteristics is necessary to combat this. In addition damping characteristics of operating handles and mountings can be improved to provide better attenuation.

Appendix A

List of Hydraulic Drill Manufacturers

Manufacturer Type	Weight kg	Impact Nominal kW	Rate Hz	Rotation Rev/Min	Torque max cont Nm	Hydraulic pressure bar	Drill Rod size mm
Atlas Copco, Stockholm, Sweden.							
COP 1022HD	51	5.5	50	0-300	120	140	H22 Int.
COP 1025HD	52	5.5	50	0-300	120	185	H25 Int.
COP 1028HD	52	5.5	50	0-300	120	185	R28 Ext.
COP 1032HD	110	7.5	40-53	1-120	125	200	R,H28/R/H32
COP 1032HB	112	8	50	0-240	200	210	R28/R32
COP 1238HF	150	14	100-105	0-200	700	250	R38
COP 1238ME	151	15	40-60	0-100	1000	250	R32/R38/T38
COP 1238LP	150	12-18	48-80	0-100/300	430/1000	120-240	R32/R38/Ext
COP 1440	151	20	60-70	0-300	500	250	T,R38 Ext
COP 1550	160	18	35-48	0-100/200	700/1000	230	T45/S51 Ext
Boart UK, Sheffield, UK.							
HD-65	69	6	60-70	0-250	130	170	R22/25/32
HD-75	75	6	60-70	0-250	160	170	R22/25/32
HD-80	70	8	60-70	0-250	80	170	R22/25/32
HD-125	125	10	60-70	0-550	325	170	R25/28/32/38
HD-150	150	14	40-60	0-200	425	170	R25/28/32/38
HD-160	155	17	60-70	0-200	425	170	R25/28/32/38
Bohler, Kapfenberg, Austria.							
HS 416S	80	8	60-66	0-450	300	170	R25/28/32/38
HS 432S	115	9	51-59	0-320	400	170	R28/32/38/B38
HS 448	155	14	43-58	0-206/320	600/800	175	R32/38/B38/45
HS 464	260	15	39-42	0-200	900	180	B38/B45/B51

Manufacturer Type	Weight kg	Impact Nominal kW	Impact Rate Hz	Rotation Rev/Min	Torque max cont Nm	Hydraulic pressure bar	Drill Rod size mm
Eimco-Secoma, Meziieu Cedex, France.							
8							
Hydrastar 200	105	12	40-66	220	350	175	R25/R28/R32
Hydrastar 300	130	15	40-50	220	350	175	R28/R32
Furukawa Rockdrilling Sales Co Ltd,Tokyo.							
HD75, HD135, HD300 hydraulic rockdrills							
KMMB(Krupp, Markmann & Moll) Essen, F.R.Germany.							
HB 101	240	11,5	30	0-140	950	150-170	T38/45
HB 103	270	11,5	30	0-110	4000	150-170	T38/45
HB 106	270	11,5	30	0-110	4000	150-170	T45/55
HB 105	370	11,5	30	0-40/80	3-6000	150/170	T45/55
Montabert,SA, Saint Priest, France.							
HC 40	80	12	47/62	0-120/310	430	130	28/32
HC 80	132	18	47/60	0-120/220	800	130	38
HC 120	158	21	34/40/52	0-60/160	12500/1400	140	38/45
Perard Torque Tension(PTT)Ltd, Grange Mill Lane, Sheffield.							
RD 500	32	5	83	140-300	65	172	22/25/28/32/38
RD 2500	85-115	3,5/9,5	58/67	300-550	280-300	172	25/28/32
RD 3000	125	7	63	150-480	230	172	25/38
RD 400	155	13	58	150-380	290	172	T/R38

Manufacturer Type	Weight kg	Impact Nominal kW	Rate Hz	Rotation Rev/Min	Torque max cont Nm	Hydraulic pressure bar	Drill Rod size mm
Salzgitter Maschinenbau GbmH, F.R.G.							
HA 3015	90	13.4	58	0-280	430	150-180	29mm holes +
HA 4025	150	14.5-17.5	58	0-280	220	150-180	43mm holes +
SIG Swiss Industrial Co.(Construction and Mining Division)							
HBM 40	75	5.5	57/70	0-450	350	175	22/25/32
HBM 50	75	5.5	47/63	0-450	350	175	22/25/32
HBM 60	75	7.5	57/70	0-450	350	175	22/25/32
HBM 100	140	12	42-100	0-550	430	175	25/32/38
HBM 150 2	140	15	50/57/63	0-450/300	430/700	175	32/38
Tamrock Group, Tampella Ltd, Finland							
HH 50	27	2	63	200-300	50	100-120	19/22
HE 110	38	3	64	0-300	70	100-120	19
HE 122	39	4	68	0-300	100	120-140	22
HE 322	40	5	85	0-350	95	120-150	22
HE 425	108	10	53	0-300	75-120	75-120	25/28
HLR 438	110	14	60	0-300	160-240	75-165	32/38
HL 538	120	15	62	0-300	240	75-170	32/38
HL 500	130	15	60	0-300	245	90-170	32/38
HL 500 Super	130	22	68	0-300	245	140-250	32/38
HL 645/600	180	16	53	0-200	470	120-170	32/38/45
HL 850	255/295	18	38	0-150	700/1800	90-130	51/76-102
HL 1000	290/300	22	37	0-125	1340	90-130	51/76-102
HL 2000	750	40	36	0-75	2600	90-140	127
HL 4000	1300	70	30	0-60	3900	90-140	152/165

Appendix B

Friction Losses in Pipes

1 The D´Arcy Formula:-

$H_f = 4flv^2/2gd$ thus

$\Delta p = 4flwv^2/2d$

2 Manning Formula

$v = (1/n)m^{0.66}i^{0.5}$ in SI units

n is a roughness coefficient increasing with the roughness of the pipe from 0.009 for glass to 0.022 for dirty cast iron pipe.

3 Hazen Williams Formula

$v = 0.82Cm^{0.6}i^{0.54}$ in SI units

Where C is a coefficient varying from 140 for very smooth pipes to 80 or less for old iron pipes in bad condition.

Note:- Both the Manning and Hazen Williams formulae are more practical than the D´Arcy formula.

For round pipes $m = d/4$

i = hydraulic gradient or loss of head per unit length

$i = H_f/l$

The Manning formula becomes:-

$\Delta p = wgl(vn/m^{0.66})^2$

The Hazen Williams formula becomes:-

$\Delta p = wgl(v/0.82Cm^{0.6})^{1.852}$

4 Turbulent flow

An alternative expression, given in Guillon (Ref 7), can be used for hydraulic systems where the flow is mainly turbulent. That is when the Reynolds number exceeds 2500.

$$\Delta p = 0.025 lwv^2/2D$$

This can be used without much error if the internal surfaces of the pipes and conduits are in good condition. It is also easier to use in the computer program.

Appendix C

Fluid Losses

(a) Orifice losses in the valve

The two reference books used gave slightly different values for the losses at the spool control valves.

The first reference 'Hydraulic Control Systems' by H E Merritt (Ref 14) quotes a useable coefficient of discharge $C_d = 0.611$ for three way spool valves of the type used is the hydraulic drill.

$Q = C_d A(2p/w)^{0.5}$
$\Delta p = (Q/(C_d A))^2 w/2$
$\Delta p = (v/C_d)^2 w/2$

$\Delta p = 2.68wv^2/2$

In the second reference 'Hydraulic Servo Systems analysis and design' by M Guillon (Ref 7) the following expression was used :-

$\Delta p = \mathcal{E}wv^2/2$ where $\mathcal{E} = 1.6$ for control spool valves

$\Delta p = 1.6wv^2/2$

The value here is somewhat smaller than the constant quoted by Merrit. The larger value of 2.68 was used in the computer simulation.

(b) Losses due to sudden enlargement

$\Delta p = Kwv^2/2$ where $K = (1-(d_1/d_2))^2 = (1-A_1/A_2)^2$

Thus
$\Delta p = (1-A_1/A_2)^2 wv^2/2$

(c) Losses due to sudden contraction

Here $\Delta p = \mathcal{E} w v^2/2$ where $\mathcal{E} = (1/\beta - 1)^2$

For a square edged contraction $\beta = 0.61$ to 0.65

Thus $\mathcal{E} = 0.40$ to 0.30

Then use 0.35 as an average value.

$\Delta p = 0.35 w v^2/2$

(d) Losses due to bends and elbows

$\Delta p = \mathcal{E} w v^2/2$ where $\mathcal{E} = 1$ for an elbow

$\Delta p = w v^2/2$

Appendix D

Seal Friction

The seals used on the reciprocating part of the percussive section of this drill are plastic seals with a nitrile rubber '0' ring to provide loading. These seals, of the Shamban type, give low friction and high wear resistance. They are relieved on the inner diameter so that they are partially pressure balanced. Approximately 40% of the seal is balanced in this way.

The Turcite material used as the rubbing material between the '0' ring and the reciprocating piston has an average coefficient of friction of 0.036 for the velocities used in the drill. For the two piston diameters the friction loss per seal is as follows:-

For the 32mm diameter seal

Width over which net pressure acts = 0.6x4.2mm
Groove diameter = 43mm
Rod diameter = 32mm

$$F_{s1} = \mu N$$
$$= 0.036 \times \pi \times 32 \times 0.6 \times 4.2 \times p_1$$
$$= 9.12 p_1$$

If p_1 = 16MPa then F_{s1} = 145.9 N

For two seals this gives a total friction force of 291.8 N at the constant pressure end of the piston. As the pressure varies along the stroke this will not be strictly true, but is sufficiently accurate to be used for the friction values in the computer simulation.

Similarly for the 36mm diameter seal

Width over which net pressure acts = 0.6x4.2mm
Groove diameter = 47mm
Rod diameter = 36mm

$F_{s2} = \mu N$

$= 0.036 \times \pi \times 36 \times 0.6 \times 4.2 \times p_2$

$= 10.26 p_2$

If p_2 = 16MPa then F_{s2} = 164.16 N
For two seals the total friction at the 36mm dia end of the piston will be 328.32 N

Thus when both A_1 and A_2 are pressurised the total friction force will be of the order of 620.12 N

When only A_1 is pressurised the force will be of the order of 291.8 N.

As the hydraulic forces in either direction will be nominally of the order of 1057N and 3418N respectively, then losses of 292N and 620N are high in proportion. Seal friction is therefore an area of great importance in drill design.

The values taken from the test done on a 36mm diameter piston are shown in Fig 47 on page 132. They correlate well with the calculated ones. Any seal redesign which results in less friction will give significant increases in performance. This will be seen in higher frequency and higher blow energy.

In a water drill some external leakage will be acceptable as long as the overall drill efficiency is not reduced too much. However as mentioned before a slight increase in leakage may result in a significant reduction in seal friction. Thus the loss in volumetric effiiciency may be more than offset by the increase in mechanical and hence overall efficiency.

Appendix E

Instrumentation Specifications

LVDT signal conditioner

Make	RDP Electronics
Model	D7M
Oscillator output	5 V rms @ 5 kHz at 100mA max
Output	+10 V @ 10 mA
Linearity	0,1% full scale
Noise	Typically +25 mV at full rated output

Strain gauge bridge conditioner

Make	EMI
Model	SE994
Input	10 V
Gain accuracy	<1%
Output frequency response	dc to 50 kHz within +0,2dB and -3dB
Noise	<3 V rms reference to input. Bandwidth 10Hz to 10kHz

Piston LVDT

Make	RDP Electronics
Supplier	Bell and Howell
Model number	D5/2000
Linear range	+50 mm
Linearity	0,12% full range
Frequency	5 kHz

Appendix F

Rock Properties

The properties of typical norite rock were taken from Brink(1979). Although Brink's samples were taken from the Belfast district the values should be fairly close to the rock used which came from the Brits area.

Triaxial compression strngth (MPa)	493 728 953
Uniaxial compressive strength (MPa)	292
Uniaxial tensile strength (MPa)	17,6
Density (kg/m^3)	2945
Porosity	0,3%
Modulus of elasticity	100
Poisson's ratio	0,24

Appendix G Computer Program Listing (HD30S)

```
10     !RE-STORE"HD30S"
20     REM "THIS PROGRAM IS CALLED HD30S AND IS A HYDRAULIC DRILL PE
OGRAM FOR A DRILL RUNNING ON THE STEEL"
30     GINIT
40     PLOTTER IS 3,"INTERNAL"
50     GRAPHICS ON
60     GOTO 690
70     CSIZE 6
80     MOVE 4,90
90     LABEL "HYDRAULIC DRILL PERFORMANCE PROGRAM"
100    CSIZE 4
110    MOVE 2,80
120    LABEL "This Program calculates the performance characteristic
130    MOVE 2,75
140    LABEL "of a hydraulic drill operating on oil,5/95 or water."
150    MOVE 2,70
160    LABEL "It then plots all relevent curves on request,as well a
170    MOVE 2,65
180    LABEL "listing the calculated results."
190    MOVE 2,60
200    LABEL "The required curves are selected from the menu.When th
210    MOVE 2,55
220    LABEL "next curve is needed,the menu is recalled by pressing"
230    MOVE 2,50
240    LABEL "the continue button."
250    MOVE 2,40
260    LABEL "To use the program please press the continue button."
270    PAUSE
280    GINIT
290    GRAPHICS ON
300    MOVE 4,90
310    LABEL "DRILL LAYOUT USING TWO VALVE TRIGGER PORTS"
320    MOVE 50,45
330    DRAW 20,45
340    DRAW 20,60
350    DRAW 50,60
360    DRAW 50,65
370    DRAW 55,65
380    DRAW 55,40
390    DRAW 50,40
400    DRAW 50,65
410    MOVE 55,62
420    DRAW 65,62
430    DRAW 65,43
440    DRAW 55,43
450    PAUSE
460    GOTO 690
470    !-----------------------------------------------------------
480    !          DRILL DATA INPUT
```

```
490   !------------------------------------------------------------
500   INPUT "ENTER VALUE OF AREA,A1",A1
510   INPUT "ENTER VALUE OF AREA,A2",A2
520   INPUT "ENTER VALUE OF PUMP SUPPLY PRESSURE,P",P
530   INPUT "ENTER VALUE OF PISTON MASS,M",M
540   INPUT "ENTER VALUE OF PISTON SEAL FRICTION,Fs",Fs
550   INPUT "ENTER VALUE OF VALVE MASS,Mv",Mv
560   INPUT "ENTER VALUE OF STROKE AT VALVE TRIGGER POINT,Si",Si
570   INPUT "ENTER VALUE OF VALVE DIAMETER,Dv",Dv
580   INPUT "ENTER VALUE OF VALVE DIAMETRAL CLEARANCE,Dvc",Dvc
590   INPUT "ENTER VALUE OF VALVE STROKE,Svt",Svt
600   INPUT "ENTER VALUE OF VALVE AREA 1,Av1",Av1
610   INPUT "ENTER VALUE OF VALVE AREA 2,Av2",Av2
620   INPUT "ENTER VALE OF MAXIMUM PUMP FLOW,Qp",Qp
630   INPUT "ENTER VALUE OF ACCUMULATOR VOLUME,Va",Va
640   INPUT "ENTER VALUE OF FLUID SPECIFIC MASS,M2",M2
650   GOTO 1010
660   !------------------------------------------------------------
670   !                          DATA
680   !------------------------------------------------------------
690   A1=66.0696
700   A2=279.698
710   M=2
720   Fs1=9.12
730   Fs2=10.26
740   Mv=.192
750   Si=20
760   Dv=20
770   Dvc=.015
780   Dp=37.15
790   Dpc=.025
800   Svt=6
810   Av1=28.27
820   Av2=50.26
830   Q=57
840   Va=20
850   M2=1000
860   Do=1.3
870   Nnt=4
880   B=1000
890   Vc=41.6
900   !------------------------------------------------------------
910   !                       MOTOR DATA
920   !------------------------------------------------------------
930   Am2=283.53
940   Am1=38.48
950   Mpr=.0566
960   Id=.00707969
970   Ir=.0001013
980   R=25
990   Trm=10
1000  !------------------------------------------------------------
1010  !                      STANDARD DATA
1020  !------------------------------------------------------------
1030  Ti=.0001
1040  Fvf=0
```

```
1610  WINDOW 0,50,-8,12
1620  AXES 1,1,0,0,5,2,3
1630  CLIP OFF
1640  LDIR 0
1650  CSIZE 2.5,.5
1660  LORG 6
1670  FOR I=0 TO 50 STEP 5
1680  MOVE I,-.2
1690  LABEL USING "#,K";I
1700  NEXT I
1710  LORG 8
1720  FOR I=-8 TO 12 STEP 2
1730  MOVE -.5,I
1740  LABEL USING "#,DDD";I
1750  NEXT I
1760  CLIP ON
1770  MOVE Y,X
1780  GOTO 6840
1790  !------------------------------------------------------------------
1800  !                 VELOCITY-DISPLACEMENT TO PLOTTER
1810  !------------------------------------------------------------------
1820  INPUT "Do you want graph in upper or lower part of paper(U/L
1830  IF P$="U" THEN Pp=0
1840  IF P$="L" THEN Pp=65
1850  GINIT
1860  GRAPHICS ON
1870  PLOTTER IS 705,"HPGL"
1880  DEG
1890  LDIR 90
1900  MOVE (Pp+65),30
1910  CSIZE 2.5
1920  LABEL "Fig 1 Piston Velocity v Displacement"
1930  MOVE (Pp+60),50
1940  CSIZE 2
1950  LABEL "Displacement (mm)"
1960  LDIR 180
1970  MOVE (Pp+42),22
1980  LABEL "Velocity (m/s)"
1990  VIEWPORT (Pp+9),(Pp+55),28,92
2000  FRAME
2010  WINDOW 12,-8,0,50
2020  AXES 1,1,0,0,2,5,1
2030  CLIP OFF
2040  CSIZE 2,.5
2050  LDIR 90
2060  LORG 9
2070  FOR I=0 TO 50 STEP 5
2080  MOVE -8,I
2090  LABEL USING "#,DDD";I
2100  NEXT I
2110  LDIR 90
2120  CSIZE 2
2130  LORG 6
2140  FOR I=-8 TO 12 STEP 2
2150  MOVE I,-3
2160  LABEL USING "#,K";I
```

```
2170  NEXT I
2180  MOVE Y,X
2190  CLIP ON
2200  GOTO 6840
2210  !----------------------------------------------------------------
2220  !                  VELOCITY-TIME CURVE
2230  !----------------------------------------------------------------
2240  INPUT "Do you want curve plotted on the screen or plotter (S
2250  IF A$="S" THEN 2270
2260  IF A$="P" THEN 2590
2270  GINIT
2280  Nk=3
2290  MOVE 30,7
2300  LABEL "VELOCITY TIME CURVE"
2310  MOVE 30,7
2320  LABEL "Time T*.0001 (SECS)"
2330  DEG
2340  LDIR 90
2350  CSIZE 5
2360  MOVE 7,27
2370  LABEL "Velocity (M/sec)"
2380  VIEWPORT 10,120,15,90
2390  FRAME
2400  WINDOW 0,20,-8,12
2410  AXES 1,1,0,0,5,2,3
2420  CLIP OFF
2430  LDIR 0
2440  CSIZE 2.5,.5
2450  LORG 6
2460  FOR I=0 TO 20 STEP 5
2470  MOVE I,-.2
2480  LABEL USING "#,K";I
2490  NEXT I
2500  LORG 8
2510  FOR I=-8 TO 12 STEP 2
2520  MOVE -.5,I
2530  LABEL USING "#,DDD";I
2540  NEXT I
2550  CLIP ON
2560  MOVE 0,0
2570  GOTO 6840
2580  !----------------------------------------------------------------
2590  !                VELOCITY-TIME TO PLOTTER
2600  !----------------------------------------------------------------
2610  INPUT "Do you want graph in upper or lower part of paper(U/L
2620  IF P$="U" THEN Pp=0
2630  IF P$="L" THEN Pp=69
2640  GINIT
2650  Nk=3
2660  GRAPHICS ON
2670  GOTO 2690
2680  PLOTTER IS 705,"HPGL"
2690  DEG
2700  LDIR 90
2710  MOVE (Pp+64),35
2720  CSIZE 2.5
```

```
2730  LABEL "Fig 35 Piston velocity v time(computer)"
2740  MOVE (Pp+59),50
2750  CSIZE 2
2760  LABEL "Time (ms)"
2770  LDIR 180
2780  MOVE (Pp+41),22
2790  LABEL "Velocity (m/s)"
2800  VIEWPORT (Pp+8),(Pp+54),28,92
2810  FRAME
2820  WINDOW 12,-8,0,20
2830  AXES 1,1,0,0,2,5,1
2840  CLIP OFF
2850  CSIZE 2,.5
2860  LDIR 90
2870  LORG 9
2880  FOR I=0 TO 20 STEP 5
2890  MOVE -8,I
2900  LABEL USING "#,DDD";I
2910  NEXT I
2920  LDIR 90
2930  CSIZE 2
2940  LORG 6
2950  FOR I=-8 TO 12 STEP 2
2960  MOVE I,-1
2970  LABEL USING "#,K";I
2980  NEXT I
2990  MOVE 0,0
3000  CLIP ON
3010  GOTO 6840
3020  !----------------------------------------------------------------------
3030  !                  PRESSURE-DISPLACEMENT CURVE
3040  !----------------------------------------------------------------------
3050  Nk=1
3060  MOVE 21,95
3070  LABEL "P2 PRESSURE DISPLACEMENT CURVE"
3080  MOVE 25,7
3090  LABEL "Piston Displacement S(mm)"
3100  DEG
3110  LDIR 90
3120  CSIZE 5
3130  MOVE 6,27
3140  LABEL "Pressure (MPa)"
3150  VIEWPORT 10,120,15,90
3160  FRAME
3170  WINDOW 0,50,0,20
3180  AXES 1,1,0,0,5,5,3
3190  CLIP OFF
3200  LDIR 0
3210  CSIZE 2.5,.5
3220  LORG 6
3230  FOR I=0 TO 50 STEP 5
3240  MOVE I,-.2
3250  LABEL USING "#,K";I
3260  NEXT I
3270  LORG 8
3280  FOR I=0 TO 20 STEP 5
```

```
3290  MOVE -.5,I
3300  LABEL USING "#,DD";I
3310  NEXT I
3320  MOVE 0,0  .
3330  CLIP ON
3340  GOTO 6840
3350  !----------------------------------------------------------------
3360  !             PRESSURE-TIME CURVE
3370  !----------------------------------------------------------------
3380  Nk=2
3390  MOVE 25,95
3400  LABEL "P2 PRESSURE-TIME CURVE"
3410  MOVE 30,7
3420  LABEL "Time T*.0001 (Secs)"
3430  DEG
3440  LDIR 90
3450  CSIZE 5
3460  MOVE 7,27
3470  LABEL "Pressure (MPa)"
3480  VIEWPORT 10,120,15,90
3490  FRAME
3500  WINDOW 0,130,0,20
3510  AXES 5,1,0,0,4,5,3
3520  CLIP OFF
3530  LDIR 0
3540  CSIZE 2.5,.5
3550  LORG 6
3560  FOR I=0 TO 130 STEP 10
3570  MOVE I,-.2
3580  LABEL USING "#,K";I
3590  NEXT I
3600  LORG 8
3610  FOR I=0 TO 20 STEP 5
3620  MOVE -.5,I
3630  LABEL USING "#,DD";I
3640  NEXT I
3650  MOVE 0,0
3660  CLIP ON
3670  GOTO 6840
3680  !----------------------------------------------------------------
3690  !             FORCE-DISPLACEMENT CURVE
3700  !----------------------------------------------------------------
3710  Nk=3
3720  MOVE 25,95
3730  LABEL "FORCE-DISPLACEMENT CURVE"
3740  MOVE 30,7
3750  LABEL "Piston Displacement(mm)"
3760  DEG
3770  LDIR 90
3780  CSIZE 5
3790  MOVE 7,27
3800  LABEL "Piston Force(kN)"
3810  VIEWPORT 10,120,15,90
3820  FRAME
3830  WINDOW 0,50,0,12
3840  AXES 1,1,0,0,5,2,3
```

```
3850  CLIP OFF
3860  LDIR 0
3870  CSIZE 2.5,.5
3880  LORG 6
3890  FOR I=0 TO 50 STEP 5
3900  MOVE I,-.2
3910  LABEL USING "#,K";I
3920  NEXT I
3930  LORG 8
3940  FOR I=0 TO 12 STEP 2
3950  MOVE -.5,I
3960  LABEL USING "#,DD";I
3970  NEXT I
3980  MOVE 0,0
3990  CLIP ON
4000  GOTO 6840
4010  !-----------------------------------------------------------------
4020  !            CONSTANT FORCE-DISPLACEMENT CURVE
4030  !-----------------------------------------------------------------
4040  Nk=4
4050  MOVE 25,95
4060  LABEL "P1 FORCE-DISPLACEMENT CURVE"
4070  MOVE 25,7
4080  LABEL "Piston Displacement S(mm)"
4090  DEG
4100  LDIR 90
4110  CSIZE 5
4120  MOVE 7,27
4130  LABEL "Piston Force (kN)"
4140  VIEWPORT 10,120,15,90
4150  FRAME
4160  WINDOW 0,50,0,12
4170  AXES 1,1,0,0,5,2,3
4180  CLIP OFF
4190  LDIR 0
4200  CSIZE 2.5,.5
4210  LORG 6
4220  FOR I=0 TO 50 STEP 5
4230  MOVE I,-.2
4240  LABEL USING "#,K";I
4250  NEXT I
4260  LORG 8
4270  FOR I=0 TO 12 STEP 2
4280  MOVE -.5,I
4290  LABEL USING "#,DD";I
4300  NEXT I
4310  MOVE 0,0
4320  CLIP ON
4330  GOTO 6840
4340  !-----------------------------------------------------------------
4350  !            P1-PRESSURE-TIME CURVE
4360  !-----------------------------------------------------------------
4370  Nk=5
4380  MOVE 25,95
4390  LABEL "P1 PRESSURE-TIME CURVE"
4400  MOVE 30,7
```

```
4410  LABEL "Time T*.0001(Secs)"
4420  DEG
4430  LDIR 90
4440  CSIZE 5
4450  MOVE 7,27
4460  LABEL "Pressure P1(MPa)"
4470  VIEWPORT 10,120,15,90
4480  FRAME
4490  WINDOW 0,80,0,20
4500  AXES 5,1,0,0,4,5,3
4510  CLIP OFF
4520  LDIR 0
4530  CSIZE 2.5,.5
4540  LORG 6
4550  FOR I=0 TO 80 STEP 10
4560  MOVE I,-.2
4570  LABEL USING "#,K";I
4580  NEXT I
4590  LORG 8
4600  FOR I=0 TO 20 STEP 5
4610  MOVE -.5,I
4620  LABEL USING "#,DD";I
4630  NEXT I
4640  MOVE 0,0
4650  CLIP ON
4660  GOTO 6840
4670  !------------------------------------------------------------
4680  !             PISTON DISPLACEMENT-TIME CURVE
4690  !------------------------------------------------------------
4700  INPUT "Do you want curve plotted on the screen or plotter (S/
4710  IF A$="S" THEN 4730
4720  IF A$="P" THEN 5050
4730  GINIT
4740  Nk=6
4750  MOVE 25,95
4760  LABEL "PISTON DISPLACEMENT-TIME CURVE"
4770  MOVE 30,7
4780  LABEL "Time (ms)"
4790  DEG
4800  LDIR 90
4810  CSIZE 5
4820  MOVE 7,15
4830  LABEL "Piston Displacement (mm)"
4840  VIEWPORT 10,120,15,90
4850  FRAME
4860  WINDOW 0,130,0,50
4870  AXES 5,1,0,0,2,5,3
4880  CLIP OFF
4890  LDIR 0
4900  CSIZE 2.5,.5
4910  LORG 6
4920  FOR I=0 TO 130 STEP 10
4930  MOVE I,-10.2
4940  LABEL USING "#,K";I
4950  NEXT I
4960  LORG 8
```

```
4970  FOR I=0 TO 50 STEP 5
4980  MOVE -.5,I
4990  LABEL USING "#,DDD";I
5000  NEXT I
5010  MOVE 0,0
5020  CLIP ON
5030  GOTO 6840
5040  !-------------------------------------------------------------
5050  !            PISTON DISPLACEMENT-TIME PLOT
5060  !-------------------------------------------------------------
5070  INPUT "Do you want graph in upper or lower part of paper(U/L)
5080  IF P$="U" THEN Pp=0
5090  IF P$="L" THEN Pp=65
5100  GINIT
5110  Nk=6
5120  GRAPHICS ON
5130  PLOTTER IS 705,"HPGL"
5140  DEG
5150  LDIR 90
5160  MOVE (Pp+65),30
5170  CSIZE 2.5
5180  LABEL "Fig 33 Piston displacement v time(computer)"
5190  MOVE (Pp+60),50
5200  CSIZE 2
5210  LABEL "Time (ms)"
5220  LDIR 180
5230  MOVE (Pp+42),22
5240  LABEL "Displacement (mm)"
5250  VIEWPORT (Pp+9),(Pp+55),28,92
5260  FRAME
5270  WINDOW 50,0,0,80
5280  AXES 1,1,0,0,5,5,1
5290  CLIP OFF
5300  CSIZE 2,.5
5310  LDIR 90
5320  LORG 9
5330  FOR I=0 TO 80 STEP 5
5340  MOVE 0,I
5350  LABEL USING "#,DDD";I
5360  NEXT I
5370  LDIR 90
5380  CSIZE 2
5390  LORG 6
5400  FOR I=0 TO 50 STEP 5
5410  MOVE I,-3
5420  LABEL USING "#,K";I
5430  NEXT I
5440  MOVE Y,0
5450  CLIP ON
5460  GOTO 6840
5470  !-------------------------------------------------------------
5480  !            VALVE DISPLACEMENT-TIME CURVE"
5490  !-------------------------------------------------------------
5500  Nk=7
5510  MOVE 25,95
5520  LABEL "VALVE DISPLACEMENT-TIME CURVE"
```

```
5530  MOVE 30,7
5540  LABEL "Time (ms)"
5550  DEG
5560  LDIR 90
5570  CSIZE 5
5580  MOVE 7,17
5590  LABEL "Valve Displacement (mm)"
5600  VIEWPORT 10,120,15,90
5610  FRAME
5620  WINDOW 0,130,-5,15
5630  AXES 5,1,0,-5,2,5,3
5640  CLIP OFF
5650  LDIR 0
5660  CSIZE 2.5,.5
5670  LORG 6
5680  FOR I=0 TO 130 STEP 10
5690  MOVE I,-5.2
5700  LABEL USING "#,K";I
5710  NEXT I
5720  LORG 8
5730  FOR I=-5 TO 1520 STEP 5
5740  MOVE -.5,I
5750  LABEL USING "#,DDDD";I
5760  NEXT I
5770  MOVE 0,0
5780  CLIP ON
5790  GOTO 6840
5800  !-----------------------------------------------------------------------
5810  !            CASE REACTION -TIME CURVE
5820  !-----------------------------------------------------------------------
5830  Nk=8
5840  MOVE 25,95
5850  LABEL "CASE REACTION-TIME CURVE"
5860  MOVE 30,7
5870  LABEL "Time T*.0001(Secs)"
5880  DEG
5890  LDIR 90
5900  CSIZE 5
5910  MOVE 7,27
5920  LABEL "Reaction Force(kN)"
5930  VIEWPORT 10,120,15,90
5940  FRAME
5950  WINDOW 0,130,-3,9
5960  AXES 5,1,0,0,5,1,3
5970  CLIP OFF
5980  LDIR 0
5990  CSIZE 2.5,.5
6000  LORG 6
6010  FOR I=0 TO 130 STEP 10
6020  MOVE I,-.2
6030  LABEL USING "#,K";I
6040  NEXT I
6050  LORG 8
6060  FOR I=-3 TO 9 STEP 1
6070  MOVE -.5,I
6080  LABEL USING "#,DD";I
```

```
6090  NEXT I
6100  MOVE 0,0
6110  CLIP ON
6120  GOTO 6840
6130  !-----------------------------------------------------------
6140  !            Motor-Torque-Speed Curve
6150  !-----------------------------------------------------------
6160   INPUT "Do you want curve plotted on the screen or plotter(S/
6170   IF A$="S" THEN 6190
6180   IF A$="P" THEN 1910
6190   GINIT
6200   Nk=1
6210   MOVE 21,95
6220   LABEL "Motor Torque v Speed"
6230   MOVE 35,6
6240   LABEL "Motor Speed rpm"
6250   DEG
6260   LDIR 90
6270   CSIZE 5
6280   MOVE 6,37
6290   LABEL "Torque Nm"
6300   VIEWPORT 10,120,15,90
6310   FRAME
6320   WINDOW 0,600,0,50
6330   AXES 10,1,0,0,50,5,3
6340   CLIP OFF
6350   LDIR 0
6360   CSIZE 2.5,.5
6370   LORG 6
6380   FOR I=0 TO 600 STEP 50
6390   MOVE I,-1
6400   LABEL USING "#,K";I
6410   NEXT I
6420   LORG 8
6430   FOR I=0 TO 50 STEP 5
6440   MOVE -.5,I
6450   LABEL USING "#,DD";I
6460   NEXT I
6470   MOVE 0,0
6480   CLIP ON
6490   GOTO 6840
6500  !-----------------------------------------------------------
6510  !             Motor Speed v Time Curve
6520  !-----------------------------------------------------------
6530  Nk=2
6540  MOVE 25,95
6550  LABEL "Motor Speed v Time"
6560  MOVE 30,7
6570  LABEL "Time (ms)"
6580  DEG
6590  LDIR 90
6600  CSIZE 5
6610  MOVE 3,35
6620  LABEL "Speed (rpm)"
6630  VIEWPORT 10,120,15,90
6640  FRAME
```

```
6650  WINDOW 0,200,0,2000
6660  AXES 5,1,0,0,100,500,3
6670  CLIP OFF
6680  LDIR 0
6690  CSIZE 2.5,.5
6700  LORG 6
6710  FOR I=0 TO 200 STEP 10
6720  MOVE I,-.2
6730  LABEL USING "#,K";I
6740  NEXT I
6750  LORG 8
6760  FOR I=0 TO 2000 STEP 500
6770  MOVE -.5,I
6780  LABEL USING "#,DDDD";I
6790  NEXT I
6800  MOVE 0,0
6810  CLIP ON
6820  GOTO 6840
6830  !------------------------------------------------------------------------
6840  !             Impact Mechanism
6850  ! ----------------------------------------------------------------------
6860  ALPHA ON
6870  IF Nn>Nnt THEN GOTO 7030
6880  GOTO 6900
6890  Ts=Tt
6900  V=0
6910  U=0
6920  Sv=0
6930  Svo=0
6940  So=35
6950  Vv=0
6960  Uv=0
6970  Pt=0
6980  Pto=0
6990  Ft=0
7000  Fto=0
7010  Qrot=13.4
7020  PEN 1
7030  Ts=Tt
7040  GOTO 7060
7050  Nn=Nnt
7060  FOR T=Ts TO .55 STEP Ti
7070  Vct=Vc+(3.715^2-3.2^2)*S*3.1416/40
7080  Sf=2600
7090  Ss=33
7100  Av=3.1416*Dv*(Svt/2+.3-Sv)
7110  Pa=Pac*(Va/(Va-Vaf))^N
7120  P1=Pa
7130  P=(Q/67)^2+P1
7140                        P2t=P2*Ti+P2to
7150                        P2to=P2t
7160                        P1t=P1*Ti+P1to
7170                        P1to=P1t
7180                        Pt=P*Ti+Pto
7190                        Pto=Pt
7200  Qo1=((.816*Ao*60)/1000)*(2*(P1-Po)*1000)^.5
```

```
7210              IF Sv>Svt/2 THEN S4=S
7220              IF Sv>Svt/2 THEN GOTO 7240
7230  GOTO 7260
7240             Av=3.1416*Dv*(Sv+.3-Svt/2)
7250              GOTO 7350
7260  Qv1=15708*Dv*(P1-P2)*Dvc^3/(.67*(Svt/2-Sv+.5))
7270  Qp1=15708*Dp*(P1-P2)*Dpc^3/(.67*(S+14))
7280  Vaf=Vafo+(1000*(Q-Qrot-Q1)/60+A1*U)*Ti
7290  P3=C*M2*(A2*U/Av)^2/2000000
7300  P4=K*Lr*M2/(2000*Dr)*((-A2*U+1000*(Qrot+Q1)/60)/Ar)^2
7310  IF Mm=-1 THEN P4=0
7320  P5=A2*B*(U-Uo)*Ti/Vct
7330  P2=Mm*P3+P4-P5
7340  GOTO 7470
7350              Qv1=15708*Dv*(P2-Po)*Dvc^3/(.67*(Sv+.5-Svt/2))
7360              Qp1=15708*Dp*(P1-P2)*Dpc^3/(.67*(S+14))
7370              Vaf=Vafo+(1000*(Q-Qrot-Q1)/60-(A2-A1)*U)*Ti
7380              P3=C*M2*(A2*U/Av)^2/2000000
7390              P4=K*Lr*M2/(2000*Dr)*(((Qrot+Q1)/60)/Ar)^2
7400              P5=A2*B*(U-Uo)*Ti/Vct
7410              P2=P1+Mm*P3-P5
7420          Fdr2=P2*Am2
7430          Fdr1=P1*Am1
7440          Wfr=Wor+Ti*(P2*Am2-P1*Am1)/(Mpr*R/1000+Ir*1000/R)
7450          Angr=Angro+(Wfr+Wor)*.5*Ti
7460          Nfr=Wfr*30/3.1416
7470  IF S<33 THEN GOTO 7530
7480  IF V<=0 THEN Sf=260
7490  IF V<=0 THEN Ss=33
7500  IF V<0 THEN Mm=1
7510  V=U+Ti*(P2*A2-P1*A1+Mm*(P2*Fs2+P1*Fs1)-Sf*(S-Ss))/M
7520  GOTO 7560
7530  V=U+Ti*(P2*A2-P1*A1+Mm*(P2*Fs2+P1*Fs1))/M
7540  IF V>0 THEN Mm=-1
7550  IF V<0 THEN Mm=1
7560  Sc=Vct*(P2-P2o)*1000/(B*A2)
7570  IF S>35 THEN Sc=0
7580  S=So+(V+U)*.5*Ti*1000-Sc
7590          Fd2=P2*Am2
7600          Fd1=P1*Am1+Trm*1000/(.7*R)
7610          IF Ang>.275 THEN GOTO 7660
7620          IF Fd1>=Fd2 THEN GOTO 7660
7630          Wfd=Wod+Ti*(P2*Am2-P1*Am1-Trm*1000/(.7*R))/(Mpr*R/100
7640          Ang=Ango+(Wfd+Wod)*.5*Ti
7650          GOTO 7680
7660          Wfd=Wod+Ti*(-Trm*1000/(.7*R))/(Id*1000/R)
7670          IF Wfd<0 THEN Wfd=0
7680          Nfd=Wfd*30/3.1416
7690  Q1=Qv1+Qp1+Qo1
7700  IF Nn<1 THEN GOTO 7710
7710  IF So<35 AND S>35 THEN V1=V
7720  IF S>34.9 THEN Sb=S
7730  IF U>0 AND V<0 THEN GOTO 7750
7740  GOTO 7890
7750                      Nn=Nn+1
7760                      Ang=0
```

```
7770                        Ango=0
7780                      T2=T
7790                      Tc=T2-T1
7800                        P1t2=P1t
7810                        P1t3=P1t2-P1t1
7820                        P1m=P1t3/Tc
7830                      Pt2=Pt
7840                      Pt3=Pt2-Pt1
7850                      Pm=Pt3/Tc
7860                        P2t2=P2t
7870                        P2t3=P2t2-P2t1
7880                        P2m=P2t3/Tc
7890  IF S>Si THEN GOTO 7930
7900  IF Sv>=Svt THEN GOTO 7960
7910  GOSUB 8960
7920  GOTO 7960
7930  IF Sv=0 THEN GOTO 7970
7940  GOSUB 9010
7950  IF Sv<0 THEN Sv=0
7960  IF Sv<0 THEN Vv=0
7970  Uv=Vv
7980  Vafo=Vaf
7990  Uo=U
8000  U=V
8010  P2o=P2
8020  So=S
8030  Wor=Wfr
8040  Wod=Wfd
8050  Ango=Ang
8060  T1=T2
8070  Pt1=Pt2
8080  P1t1=P1t2
8090  IF Nk=0 THEN GOSUB 9070
8100  IF Nk=1 THEN GOSUB 9210
8110  IF Nk=2 THEN GOSUB 9330
8120  IF Nk=3 THEN GOSUB 9410
8130  IF Nk=4 THEN GOSUB 9580
8140  IF Nk=5 THEN GOSUB 9540
8150  IF Nk=6 THEN GOSUB 9620
8160  IF Nk=7 THEN GOSUB 9700
8170  IF Nk=8 THEN GOSUB 9290
8180  DRAW X,Y
8190  IF Nn>Nnt THEN GOTO 8220
8200  NEXT T
8210  !------------------------------------
8220  !            RESULTS
8230  !-------------------------------------
8240  F=1/Tc
8250  E=.5*M*V1^2
8260  H1=(F*E)/1000
8270  H2=Q*P1/60
8280  IF Nk=8 THEN DRAW X,0
8290  IF Nk=2 THEN DRAW X,0
8300  IF Nk=3 THEN DRAW X,0
8310  IF Nk=6 THEN Ts=T
8320  IF Nk=6 THEN GOTO 7050
```

```
8330  INPUT "          Press the continue button to display menu for
,C
8340  IF C THEN GOTO 8360
8350  PAUSE
8360  GOTO 1180
8370  ALPHA ON
8380  GCLEAR                              ! CLEARS THE CRT OF GRAPHICS.
8390  PRINT "                                                        "
8400  PRINT "                                                      "
8410  PRINT "                                                       "
8420  PRINT "                                                        "
8430  PRINT "                                                       "
8440  PRINT "                                                        "
8450  PRINT "                                                       "
8460  PRINT "PAGE 1.    INPUT DATA"
8470  PRINT "PAGE 1.    INPUT DATA"
8480  PRINT "                                                        "
8490  PRINT "PISTON AREA A1 (sq.mm)=                              ",A1
8500  PRINT "PISTON AREA A2 (sq.mm)=                              ",A2
8510  PRINT "PUMP SUPPLY PRESSURE P (MPa)=                        ",Pm
8520  PRINT "AVERAGE DRILL INPUT PRESSURE Ps (MPa)=               ",P1m
8530  PRINT "PISTON MASS M (Kg)=                                  ",M
8540  PRINT "VALUE OF STROKE AT VALVE TRIGGER POINT (mm)=",Si
8550  PRINT "VALUE OF STROKE AT VALVE CHANGEOVER (mm)=",S4
8560  PRINT "PHASE 1 TIME(secs)=                                  ",T1
8570  PRINT "PHASE 2 TIME(secs)=                                  ",T2
8580  PRINT "CYCLE TIME (secs)=                                   ",Tc
8590  PRINT "                                                        "
8600  PRINT "                                                        "
8610  PRINT "                                                         "
8620  PRINT "                                                        "
8630  PRINT "                                                        "
8640  PRINT "                                                        "
8650  PRINT "For Page 2 please  press continue button."
8660  PAUSE
8670  GCLEAR
8680  PRINT "                                                      "
8690  PRINT "                                                      "
8700  PRINT "                                                       "
8710  PRINT "                                                      "
8720  PRINT "                                                      "
8730  PRINT "                                                      "
8740  PRINT "PAGE 2.   CALCULATED DATA"
8750  PRINT "
8760  PRINT "STROKE LENGTH (mm)=                                  ",(35-S2)
8770  PRINT "IMPACT VELOCITY (M/sec)=                             ",V1
8780  PRINT "FREQUENCY (Hz)=                                      ",F
8790  PRINT "ENERGY PER BLOW (Joules)=                            ",E
8800  PRINT "POWER OUTPUT(Kw)=                                    ",H1
8810  PRINT "THEORETICAL FLOW TO OPERATE VALVE(l/min)=",Qv
8820  PRINT "Total Oil Flow To The Drill (l/min)=                 ",Q
8830  PRINT "THEORETICAL POWER TO OPERATE VALVE(Kw)=              ",Hv
8840  PRINT "THEORETICAL HYDRAULIC POWER INPUT(Kw) =              ",H2
8850  PRINT "THEORETICAL EFFICIENCY (%)=                          ",E2
8860  PRINT "THEORETICAL DRILL THRUST (Newtons)=                  ",Fc
8870  PRINT "                                                        "
8880  PRINT "                                                       "
```

```
8890 PRINT "
8900 PRINT "
8910 PRINT "
8920 PAUSE
8930 GCLEAR
8940 GOTO 1180
8950 STOP
8960 Vv=Uv+Ti*(Pa*(Av2-Av1))/Mv
8970 Sv=Svo+(Vv+Uv)*.5*Ti*1000
8980 Uv=Vv
8990 Svo=Sv
9000 RETURN
9010 Vv=Uv+(Ti/Mv)*(P1*Av1-(M2*Av2/2000000)*(Av2*Uv/(.41*Ao))^2)
9020 Sv=Svo-(Vv+Uv)*.5*Ti*1000
9030 Uv=Vv
9040 Svo=Sv
9050 RETURN
9060 STOP
9070 IF A$="P" THEN GOTO 9140
9080 X=S
9090 IF Np=1 THEN GOTO 9120
9100 Y=V
9110 RETURN
9120 Y=V
9130 RETURN
9140 Y=S
9150 IF Np=1 THEN GOTO 9180
9160 X=V
9170 RETURN
9180 X=-V
9190 RETURN
9200 STOP
9210 IF Nkn=2 THEN GOTO 9250
9220 X=S
9230 Y=P2
9240 RETURN
9250 X=Nrm
9260 Y=Trm
9270 RETURN
9280 STOP
9290 X=T*1000
9300 Y=Fr/1000
9310 RETURN
9320 STOP
9330 IF Nkn=2 THEN GOTO 9370
9340 X=T*1000
9350 Y=P2
9360 RETURN
9370 X=T*1000
9380 Y=Nfd
9390 RETURN
9400 STOP
9410 IF Nkn=2 THEN GOTO 9450
9420 X=S
9430 Y=Fa2/1000
9440 RETURN
```

```
9450  IF A$="P" THEN GOTO 9500
9460  X=T*1000
9470  Y=V
9480  RETURN
9490  STOP
9500  X=V
9510  Y=T*1000
9520  RETURN
9530  STOP
9540  X=T*1000
9550  Y=P1
9560  RETURN
9570  STOP
9580  X=S
9590  Y=Fa1/1000
9600  RETURN
9610  STOP
9620  IF A$="P" THEN GOTO 9660
9630  X=T*1000
9640  Y=S
9650  RETURN
9660  X=S
9670  Y=T*1000
9680  RETURN
9690  STOP
9700  X=T*1000
9710  Y=Sv
9720  RETURN
9730  END
```

Appendix H Computer Program Listing (Plotprog)

```
10 GCLEAR
20    REM "THIS PROGRAM IS CALLED PLOTPROG"
30    ALLOCATE T(800),S(800),V(800),P1(800),P2(800),Pk(800),Po(800)
40    GCLEAR
50    PRINT "This is a program for entering data to be plotted. Out|
on the screen or on the plotter as required."
60    ON KEY 0 LABEL "Input Disp-Time" GOTO 2380
70    ON KEY 1 LABEL "Input Kicker-Time" GOTO 2410
80    ON KEY 2 LABEL "Plot Pk Time" GOTO 2700
90    ON KEY 3 LABEL "Vel-Dis-Screen" GOTO 980
100   ON KEY 4 LABEL "Vel-Dis-Plot" GOTO 1150
110   ON KEY 5 LABEL "Dis-Tim-Screen" GOTO 470
120   ON KEY 6 LABEL "Dis-Tim-Plot" GOTO 590
130   ON KEY 7 LABEL "PRINT P2" GOTO 290
140   ON KEY 8 LABEL "Menu 2" GOTO 170
150   ON KEY 9 LABEL "Vel-Tim-Curve" GOTO 750
160   GOTO 60
170   OFF KEY
180   ON KEY 0 LABEL "Input P1-Time" GOTO 1320
190   ON KEY 1 LABEL "Input P2-Time" GOTO 1840
200   ON KEY 2 LABEL "Write to-Disc" GOTO 4360
210   ON KEY 3 LABEL "Read from Disc" GOTO 3990
220   ON KEY 4 LABEL "P1-Time-Screen" GOTO 1450
230   ON KEY 5 LABEL "P1-Time-Plot" GOTO 1610
240   ON KEY 6 LABEL "P2-Time-Screen" GOTO 1970
250   ON KEY 7 LABEL "P2-Time-Plot" GOTO 2190
260   ON KEY 8 LABEL "Menu 1" GOTO 60
270   ON KEY 9 LABEL "Abort" GOTO 6760
280   GOTO 180
290   FOR C=1 TO N
300   PRINT C,T(C),P2(C)
310   NEXT C
320   GOTO 60
330   !-----------------------------------------------------------
340   !              Enter Displacement Time Data
350   !-----------------------------------------------------------
360   INPUT " ENTER VALUE OF N",N
370   INPUT "ENTER VALUE OF Ti",Ti
380   A$="S"
390   FOR C=1 TO N
400   INPUT "ENTER DATAPOINT VALUES",S(C)
410   T(C)=Ti*C
420   PRINT "                         ",C,T(C),S(C)
430   NEXT C
440   PAUSE
450   GOTO 60
460   !-----------------------------------------------------------
470   !              Draw Dis Time Curve on Screen
480   !-----------------------------------------------------------
```

```
490  GOSUB 3640
500  M=0
510  FOR C=1 TO N
520  DRAW (T(C)+M*T(N))*1000*(1/.88),S(C)
530  NEXT C
540  M=M+1
550  IF M>3 THEN GOTO 570
560  GOTO 510
570  GOTO 60
580  !-------------------------------------------------------------
590  !              Plot Displacement Time Curve
600  !-------------------------------------------------------------
610  GINIT
620  GRAPHICS ON
630  PLOTTER IS 705,"HPGL"
640  GOSUB 5140
650  M=0
660  FOR C=1 TO N
670  IF C>1 THEN GOTO 680
680  DRAW S(C),(T(C)+M*T(N))*1000*(1/.88)
690  NEXT C
700  M=M+1
710  IF M>3 THEN GOTO 730
720  GOTO 660
730  GOTO 60
740  !-------------------------------------------------------------
750  !              Draw Velocity Time Curve
760  !-------------------------------------------------------------
770  INPUT "Do you want data from memory? (Y/N)",M$
780  IF M$="Y" THEN GOSUB 3990
790  PAUSE
800  INPUT "Do you want graph on screen or plotter?(S/P)",B$
810  IF B$="S" THEN GOTO 830
820  IF B$="P" THEN GOTO 860
830  PLOTTER IS CRT,"INTERNAL"
840  GOSUB 3280
850  GOTO 880
860  PLOTTER IS 705,"HPGL"
870  GOSUB 1120
880  M=0
890  FOR C=1 TO N
900  V(C)=.88*((S(C)-S(C-1))/.075)
910  DRAW T(C),V(C)
920  NEXT C
930  M=M+1
940  IF M>3 THEN GOTO 960
950  GOTO 890
960  GOTO 180
970  !-------------------------------------------------------------
980  !              Draw Velocity Displacement Curve on Screen
990  !-------------------------------------------------------------
1000 GINIT
1010 GCLEAR
1020 GRAPHICS ON
1030 PLOTTER IS 3,"INTERNAL"
1040 GOSUB 2940
```

```
1050  FOR C=1 TO N
1060  IF C>1 THEN GOTO 1100
1070  V(C)=0
1080  MOVE S(C),V(C)
1090  GOTO 1110
1100  V(C)=.88*((S(C)-S(C-1))/.075)
1110  DRAW S(C),V(C)
1120  NEXT C
1130  GOTO 60
1140  !---------------------------------------------------------------
1150  !             Velocity Displacement Plot
1160  !---------------------------------------------------------------
1170  GCLEAR
1180  GINIT
1190  GRAPHICS ON
1200  PLOTTER IS 705,"HPGL"
1210  GOSUB 4780
1220  FOR C=1 TO N
1230  IF C>1 THEN GOTO 1270
1240  V(C)=0
1250  MOVE V(C),S(C)
1260  GOTO 1280
1270  V(C)=.88*((S(C)-S(C-1))/.075)
1280  DRAW V(C),S(C)
1290  NEXT C
1300  GOTO 60
1310  !---------------------------------------------------------------
1320  !             Enter P1 Time Data
1330  !---------------------------------------------------------------
1340  INPUT " ENTER VALUE OF N",N
1350  INPUT "ENTER VALUE OF Ti",Ti
1360  A$="P1"
1370  FOR C=1 TO N
1380  INPUT "ENTER DATAPOINT VALUES",P1(C)
1390  T(C)=Ti*C
1400  PRINT "                          ",C,T(C),P1(C)
1410  NEXT C
1420  PAUSE
1430  GOTO 180
1440  !---------------------------------------------------------------
1450  !             Draw Supply Pressure Time Curve on Screen
1460  !---------------------------------------------------------------
1470  A$="P1"
1480  INPUT "Do you want data from memory (Y/N)?",M$
1490  IF M$="Y" THEN GOSUB 3990
1500  PAUSE
1510  GOSUB 5500
1520  M=0
1530  FOR C=1 TO N
1540  DRAW (T(C)+M*T(N))*1000*(1/8.8),P1(C)
1550  NEXT C
1560  M=M+1
1570  IF M>3 THEN GOTO 1590
1580  GOTO 1530
1590  GOTO 180
1600  !---------------------------------------------------------------
```

```
1610  !              Plot Supply Pressure Time Curve
1620  !-------------------------------------------------------------
1630  A$="P1"
1640  INPUT "Do you want data from memory (Y/N)?",M$
1650  IF M$="Y" THEN GOSUB 3990
1660  PAUSE
1670  INPUT "Do you want graph in upper or lower part of paper(U/L)
1680  IF P$="U" THEN Pp=0
1690  IF P$="L" THEN Pp=65
1700  GINIT
1710  GRAPHICS ON
1720  PLOTTER IS 705,"HPGL"
1730  GOSUB 5970
1740  M=0
1750  FOR C=1 TO N
1760  IF C>1 THEN GOTO 1770
1770  DRAW P1(C),(T(C)+M*T(N))*1000*(1/8.8)
1780  NEXT C
1790  M=M+1
1800  IF M>3 THEN GOTO 1820
1810  GOTO 1750
1820  GOTO 170
1830  !-------------------------------------------------------------
1840  !              Enter P2 Time Data
1850  !-------------------------------------------------------------
1860  INPUT " ENTER VALUE OF N",N
1870  INPUT "ENTER VALUE OF Ti",Ti
1880  A$="P2"
1890  FOR C=1 TO N
1900  INPUT "ENTER DATAPOINT VALUES",P2(C)
1910  T(C)=Ti*C
1920  PRINT "                              ",C,T(C),P2(C)
1930  NEXT C
1940  PAUSE
1950  GOTO 180
1960  !-------------------------------------------------------------
1970  !              Draw Chamber Pressure Time Curve on Screen
1980  !-------------------------------------------------------------
1990  A$="P2"
2000  INPUT "Do you want data from memory? (Y/N)",M$
2010  IF M$="Y" THEN GOSUB 3990
2020  PAUSE
2030  INPUT "Do you want graph in upper or lower part of paper(U/L)
2040  IF P$="U" THEN Pp=0
2050  IF P$="L" THEN Pp=65
2060  GINIT
2070  GRAPHICS ON
2080  PLOTTER IS CRT,"INTERNAL"
2090  GOSUB 5500
2100  M=0
2110  FOR C=1 TO N
2120  DRAW (T(C)+M*T(N))*1000*(1/.88),P2(C)
2130  NEXT C
2140  M=M+1
2150  IF M>3 THEN GOTO 2170
2160  GOTO 2110
```

```
2170  GOTO 180
2180  !---------------------------------------------------------------
2190  !           Plot Chamber Pressure Time Curve
2200  !---------------------------------------------------------------
2210  A$="P2"
2220  INPUT "Do you want data from memory (Y/N)?",M$
2230  IF M$="Y" THEN GOSUB 3990
2240  PAUSE
2250  INPUT "Do you want graph in upper or lower part of paper(U/L
2260  IF P$="U" THEN Pp=0
2270  IF P$="L" THEN Pp=65
2280  GINIT
2290  GRAPHICS ON
2300  PLOTTER IS 705,"HPGL"
2310  GOSUB 6400
2320  M=0
2330  FOR C=1 TO N
2340  IF C>1 THEN GOTO 2350
2350  DRAW P2(C),(T(C)+M*T(N))*1000*(1/.88)
2360  NEXT C
2370  M=M+1
2380  IF M>3 THEN GOTO 2920
2390  GOTO 2330
2400  !---------------------------------------------------------------
2410  !           Enter Kicker Pressure Time Data
2420  !---------------------------------------------------------------
2430  INPUT " ENTER VALUE OF N",N
2440  INPUT "ENTER VALUE OF Ti",Ti
2450  A$="Pk"
2460  FOR C=1 TO N
2470  INPUT "ENTER DATAPOINT VALUES",Pk(C)
2480  T(C)=Ti*C
2490  PRINT "                             ",C,T(C),Pk(C)
2500  NEXT C
2510  PAUSE
2520  GOTO 180
2530  !---------------------------------------------------------------
2540  !           Draw Kicker Pressure Time Curve on Screen
2550  !---------------------------------------------------------------
2560  A$="Pk"
2570  INPUT "Do you want data from memory (Y/N)?",M$
2580  IF M$="Y" THEN GOSUB 3990
2590  PAUSE
2600  GOSUB 5500
2610  M=0
2620  FOR C=1 TO N
2630  DRAW (T(C)+M*T(N))*1000*(1/.88),Pk(C)
2640  NEXT C
2650  M=M+1
2660  IF M>3 THEN GOTO 2680
2670  GOTO 2620
2680  GOTO 180
2690  !---------------------------------------------------------------
2700  !           Plot Kicker Pressure Time Curve
2710  !---------------------------------------------------------------
2720  A$="Pk"
```

```
2730  INPUT "Do you want data from memory (Y/N)?",M$
2740  IF M$="Y" THEN GOSUB 3990
2750  PAUSE
2760  INPUT "Do you want graph in upper or lower part of paper(U/L)
2770  IF P$="U" THEN Pp=0
2780  IF P$="L" THEN Pp=65
2790  GINIT
2800  GRAPHICS ON
2810  PLOTTER IS 705,"HPGL"
2820  GOSUB 5970
2830  M=0
2840  FOR C=1 TO N
2850  IF C>1 THEN GOTO 2860
2860  DRAW Pk(C),(T(C)+M*T(N))*1000*(1/.88)
2870  NEXT C
2880  M=M+1
2890  IF M>3 THEN GOTO 2910
2900  GOTO 2840
2910  GOTO 170
2920  !-------------------------------------------------------------
2930  !            Velocity Displacement Axes on Screen
2940  !-------------------------------------------------------------
2950  GINIT
2960  GCLEAR
2970  GRAPHICS ON
2980  ! PLOTTER IS 3,"INTERNAL"
2990  MOVE 25,95
3000  LABEL "VELOCITY DISPLACEMENT CURVE"
3010  MOVE 25,7
3020  LABEL "Piston Displacement S(mm)"
3030  DEG
3040  LDIR 90
3050  CSIZE 5
3060  MOVE 7,27
3070  LABEL "Velocity (M/sec)"
3080  VIEWPORT 10,120,15,90
3090  FRAME
3100  WINDOW 0,50,-8,12
3110  AXES 1,1,0,0,5,2,3
3120  CLIP OFF
3130  LDIR 0
3140  CSIZE 2.5,.5
3150  LORG 6
3160  FOR I=0 TO 50 STEP 5
3170  MOVE I,-.2
3180  LABEL USING "#,K";I
3190  NEXT I
3200  LORG 8
3210  FOR I=-8 TO 12 STEP 2
3220  MOVE -.5,I
3230  LABEL USING "#,DDD";I
3240  NEXT I
3250  CLIP ON
3260  RETURN
3270  !-------------------------------------------------------------
3280  !              Velocity v Time Axes Subroutine
```

```
3290  !----------------------------------------------------------------
3300  DEG
3310  LDIR 90
3320  MOVE (Pp+65),30
3330  CSIZE 2.5
3340  LABEL "Fig Nn Velocity versus Time"
3350  MOVE (Pp+60),50
3360  CSIZE 2
3370  LABEL "Time (ms)"
3380  LDIR 180
3390  MOVE (Pp+42),22
3400  CSIZE 2
3410  LABEL "Velocity (M/sec)"
3420  VIEWPORT (Pp+12),(Pp+58),28,92
3430  FRAME
3440  WINDOW 10,-5,0,20
3450  AXES 1,1,0,0,5,5,1
3460  CLIP OFF
3470  CSIZE 2.5
3480  LDIR 90
3490  LORG 9
3500  FOR I=0 TO 20 STEP 5
3510  MOVE 0,I
3520  LABEL USING "#,DDD";I
3530  NEXT I
3540  LDIR 90
3550  CSIZE 2
3560  LORG 6
3570  FOR I=-5 TO 10 STEP 5
3580  MOVE I,-3
3590  NEXT I
3600  CLIP ON
3610  MOVE 18,0
3620  RETURN
3630  !----------------------------------------------------------------
3640  !             Displacement v Time Axes Subroutine
3650  !----------------------------------------------------------------
3660  GINIT
3670  GCLEAR
3680  GRAPHICS ON
3690  PLOTTER IS 3,"INTERNAL"
3700  MOVE 19,95
3710  LABEL "PISTON DISPLACEMENT-TIME CURVE"
3720  MOVE 30,7
3730  LABEL "Time  (ms)"
3740  DEG
3750  LDIR 90
3760  CSIZE 5
3770  MOVE 7,15
3780  LABEL "Piston Displacement (mm)"
3790  VIEWPORT 10,120,15,90
3800  FRAME
3810  WINDOW 0,40,0,50
3820  AXES 5,1,0,0,2,5,3
3830  CLIP OFF
3840  LDIR 0
```

```
3850  CSIZE 2.5,.5
3860  LORG 6
3870  FOR I=0 TO 40 STEP 5
3880  MOVE I,-1
3890  LABEL USING "#,K";I
3900  NEXT I
3910  LORG 8
3920  FOR I=0 TO 50 STEP 5
3930  MOVE -.5,I
3940  LABEL USING "#,DDD";I
3950  NEXT I
3960  CLIP ON
3970  MOVE 0,35
3980  RETURN
3990  !------------------------------------------------------------
4000  !            READ DATA FROM DISC
4010  !------------------------------------------------------------
4020 Disc_read:
4030  GCLEAR
4040  OFF KEY
4050  PRINTER IS CRT
4060  INPUT "NAME OF DATA FILE",File_name$
4070  Concat$=File_name$&"_1"
4080  ASSIGN @Path TO Concat$
4090  ENTER @Path;N
4100  MASS STORAGE IS ":HP9122,700,0"
4110  ASSIGN @Path TO File_name$
4120  PRINT N," VALUE READ"
4130  IF A$="P1" THEN GOTO 4210
4140  IF A$="P2" THEN GOTO 4260
4150  IF A$="Pk" THEN GOTO 4310
4160  FOR C=1 TO N
4170  ENTER @Path;T(C),S(C)
4180  PRINT C,T(C),S(C)
4190  NEXT C
4200  GOTO 60
4210    FOR C=1 TO N
4220    ENTER @Path;T(C),P1(C)
4230    PRINT C,T(C),P1(C)
4240    NEXT C
4250    GOTO 60
4260  FOR C=1 TO N
4270  ENTER @Path;T(C),P2(C)
4280  PRINT C,T(C),P2(C)
4290  NEXT C
4300  GOTO 60
4310    FOR C=1 TO N
4320    ENTER @Path;T(C),Pk(C)
4330    PRINT C,T(C),Pk(C)
4340    NEXT C
4350    RETURN
4360  !------------------------------------------------------------
4370  !            WRITE DATA TO DISC
4380  !------------------------------------------------------------
4390 Disc_store:
4400  GCLEAR
```

```
4410  PRINTER IS CRT
4420  GOTO 4440
4430  PRINT "INCORRECT FILE NAME, TYPE CORRECT NAME"
4440  INPUT "NAME OF DATA FILE",File_name$
4450  ON ERROR GOTO 4430
4460  Concat$=File_name$&"_1"
4470  CREATE BDAT Concat$,1,8
4480  ASSIGN @Path TO Concat$
4490  OUTPUT @Path;N
4500  CREATE BDAT File_name$,400,16
4510  ASSIGN @Path TO File_name$
4520  PRINT "WRITING ALL DATA TO DISC"
4530  PRINT N," VALUE"
4540  IF A$="P1" THEN GOTO 4620
4550  IF A$="P2" THEN GOTO 4670
4560  IF A$="Pk" THEN GOTO 4720
4570  FOR C=1 TO N
4580  PRINT C,T(C),S(C)
4590  OUTPUT @Path;T(C),S(C)
4600  NEXT C
4610  GOTO 60
4620    FOR C=1 TO N
4630    PRINT C,T(C),P1(C)
4640    OUTPUT @Path;T(C),P1(C)
4650    NEXT C
4660  GOTO 170
4670  FOR C=1 TO N
4680  PRINT C,T(C),P2(C)
4690  OUTPUT @Path;T(C),P2(C)
4700  NEXT C
4710  GOTO 170
4720    FOR C=1 TO N
4730    PRINT C,T(C),Pk(C)
4740    OUTPUT @Path;T(C),Pk(C)
4750    NEXT C
4760    GOTO 170
4770  !-------------------------------------------------------------
4780  !                 Velocity Displacement Plot
4790  !-------------------------------------------------------------
4800  DEG
4810  LDIR 90
4820  MOVE 65,30
4830  CSIZE 2.5
4840  LABEL "Fig 1 Piston Velocity versus Displacement
4850  MOVE 60,50
4860  CSIZE 2
4870  LABEL "Displacement (mm)"
4880  LDIR 180
4890  MOVE 42,22
4900  LABEL "Velocity (m/sec)"
4910  VIEWPORT 9,55,28,92
4920  FRAME
4930  WINDOW 12,-8,0,50
4940  AXES 1,1,0,0,2,5,1
4950  CLIP OFF
4960  CSIZE 2,.5
```

```
4970  LDIR 90
4980  LORG 9
4990  FOR I=0 TO 50 STEP 5
5000  MOVE -8,I
5010  LABEL USING "#,DDD";I
5020  NEXT I
5030  LDIR 90
5040  CSIZE 2
5050  LORG 6
5060  FOR I=-8 TO 12 STEP 2
5070  MOVE I,-3
5080  LABEL USING "#,K";I
5090  NEXT I
5100  MOVE 40,0
5110  CLIP ON
5120  RETURN
5130  !-----------------------------------------------------------
5140  !                    Displacement Time Plot
5150  !-----------------------------------------------------------
5160  DEG
5170  LDIR 90
5180  MOVE 130,30
5190  CSIZE 2.5
5200  LABEL "Fig 2 Piston Displacement versus Time "
5210  MOVE 125,50
5220  CSIZE 2
5230  LABEL "Time (ms)
5240  LDIR 180
5250  MOVE 107,22
5260  CSIZE 2
5270  LABEL "Displacement (mm)
5280  VIEWPORT 74,120,28,92
5290  FRAME
5300  WINDOW 50,0,0,80
5310  AXES 1,1,0,0,5,5,1
5320  CLIP OFF
5330  CSIZE 2,.5
5340  LDIR 90
5350  LORG 9
5360  FOR I=0 TO 80 STEP 5
5370  MOVE 0,I
5380  LABEL USING "#,DDD";I
5390  NEXT I
5400  LDIR 90
5410  CSIZE 2
5420  LORG 6
5430  FOR I=0 TO 50 STEP 5
5440  MOVE I,-3
5450  LABEL USING "#,K";I
5460  NEXT I
5470  CLIP ON
5480  RETURN
5490  !-----------------------------------------------------------
5500  !               Pressure v Time Axes Subroutine
5510  !-----------------------------------------------------------
5520  GINIT
```

```
5530  GCLEAR
5540  GRAPHICS ON
5550  PLOTTER IS 3,"INTERNAL"
5560  MOVE 25,95
5570  IF A$="P2" THEN GOTO 5600
5580  LABEL "SUPPLY PRESSURE-TIME CURVE"
5590  GOTO 5610
5600  LABEL "CHAMBER PRESSURE-TIME CURVE"
5610  MOVE 48,7
5620  LABEL "Time (ms)"
5630  DEG
5640  LDIR 90
5650  CSIZE 5
5660  MOVE 7,20
5670  IF A$="P2" THEN GOTO 5700
5680  LABEL "Supply Pressure (MPa)"
5690  GOTO 5710
5700  LABEL "Chamber Pressure (MPa)"
5710  VIEWPORT 10,120,15,90
5720  FRAME
5730  WINDOW 0,80,0,20
5740  AXES 5,1,0,0,4,5,3
5750  CLIP OFF
5760  LDIR 0
5770  CSIZE 2.5,.5
5780  LORG 6
5790  FOR I=0 TO 80 STEP 5
5800  MOVE I,-.2
5810  LABEL USING "#,K";I
5820  NEXT I
5830  LORG 8
5840  FOR I=0 TO 20 STEP 5
5850  MOVE -.5,I
5860  LABEL USING "#,DDD";I
5870  NEXT I
5880  LORG 6
5890  FOR I=-8 TO 12 STEP 2
5900  MOVE I,-3
5910  LABEL USING "#,K";I
5920  NEXT I
5930  MOVE 40,0
5940  CLIP ON
5950  RETURN
5960  !-----------------------------------------------------------------
5970  !                     Supply Pressure P1 Time Plot Axes
5980  !-----------------------------------------------------------------
5990  DEG
6000  LDIR 90
6010  MOVE (Pp+65),30
6020  CSIZE 2.5
6030  IF A$="Pk" THEN GOTO 6060
6040  LABEL "Fig   Supply Pressure versus Time "
6050  GOTO 6070
6060  LABEL "Fig   Kicker Pressure versus Time. "
6070  MOVE (Pp+60),50
6080  CSIZE 2
```

```
6090  LABEL "Time (ms)
6100  LDIR 180
6110  MOVE (Pp+42),22
6120  CSIZE 2
6130  IF A$="Pk" THEN GOTO 6160
6140  LABEL "Supply Pressure (MPa)
6150  GOTO 6170
6160  LABEL "Kicker Pressure (MPa)
6170  VIEWPORT (Pp+9),(Pp+55),28,92
6180  FRAME
6190  WINDOW 20,0,0,80
6200  AXES 1,1,0,0,5,5,1
6210  CLIP OFF
6220  CSIZE 2,.5
6230  LDIR 90
6240  LORG 9
6250  FOR I=0 TO 80 STEP 5
6260  MOVE 0,I
6270  LABEL USING "#,DDD";I
6280  NEXT I
6290  LDIR 90
6300  CSIZE 2
6310  LORG 6
6320  FOR I=0 TO 20 STEP 5
6330  MOVE I,-3
6340  LABEL USING "#,K";I
6350  NEXT I
6360  CLIP ON
6370  MOVE 18,0
6380  RETURN
6390  !---------------------------------------------------------------
6400  !                   Chamber Pressure Time Plot Axes
6410  !---------------------------------------------------------------
6420  DEG
6430  LDIR 90
6440  MOVE (Pp+65),30
6450  CSIZE 2.5
6460  LABEL "Fig   Chamber Pressure versus Time "
6470  MOVE (Pp+60),50
6480  CSIZE 2
6490  LABEL "Time (ms) "
6500  LDIR 180
6510  MOVE (Pp+42),22
6520  CSIZE 2
6530  LABEL "Chamber Pressure (MPa) "
6540  VIEWPORT (Pp+12),(Pp+58),28,92
6550  FRAME
6560  WINDOW 20,0,0,80
6570  AXES 1,1,0,0,5,5,1
6580  CLIP OFF
6590  CSIZE 2,.5
6600  LDIR 90
6610  LORG 9
6620  FOR I=0 TO 80 STEP 5
6630  MOVE 0,I
6640  LABEL USING "#,DDD";I
```

```
6650  NEXT I
6660  LDIR 90
6670  CSIZE 2
6680  LORG 6
6690  FOR I=0 TO 20 STEP 5
6700  MOVE I,-3
6710  LABEL USING "#,K";I
6720  NEXT I
6730  CLIP ON
6740  MOVE 18,0
6750  RETURN
6760  GCLEAR
6770  END
```

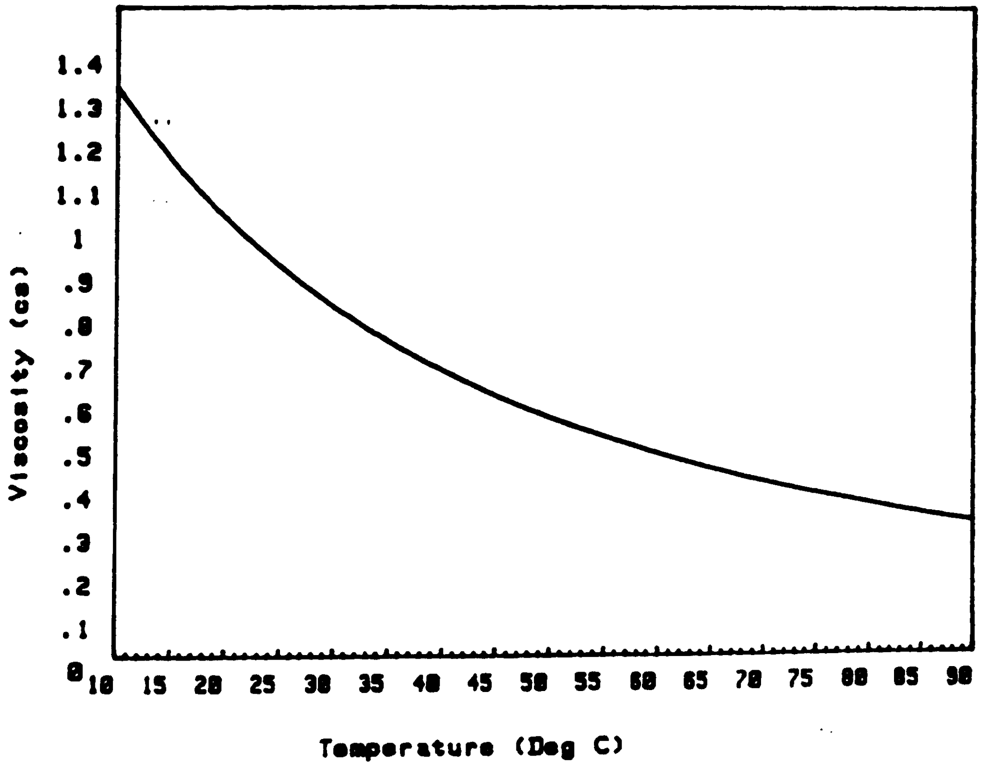

Fig 1 Water Viscosity versus Temperature

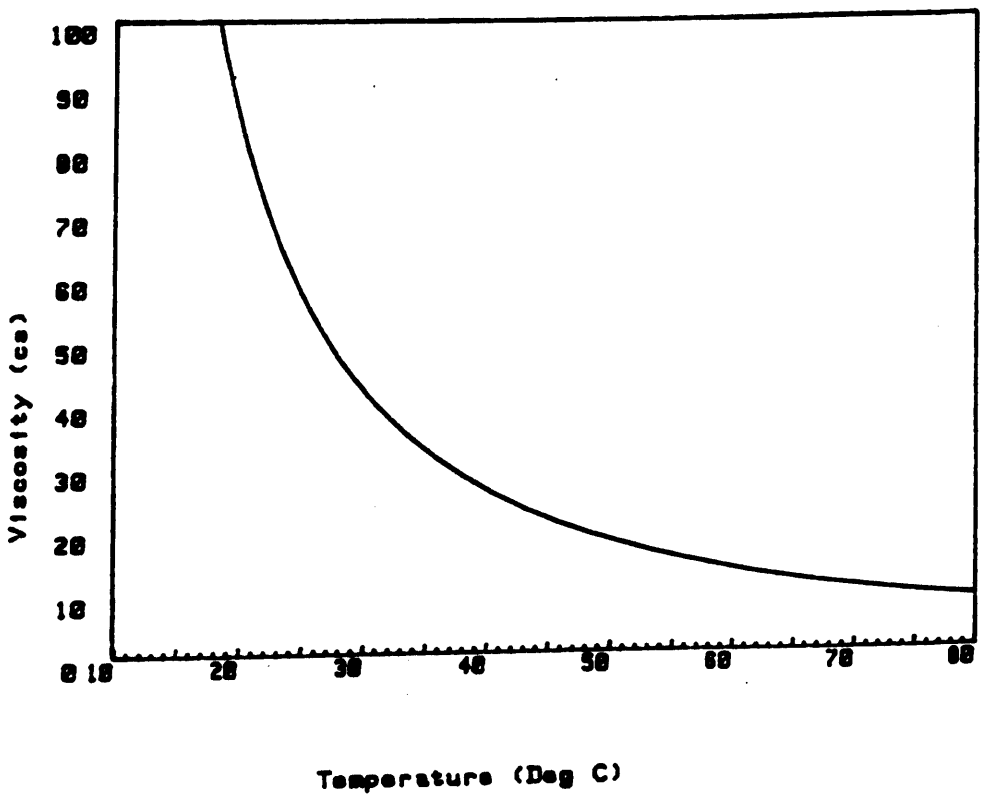

Fig 2 Oil Viscosity versus Temperature

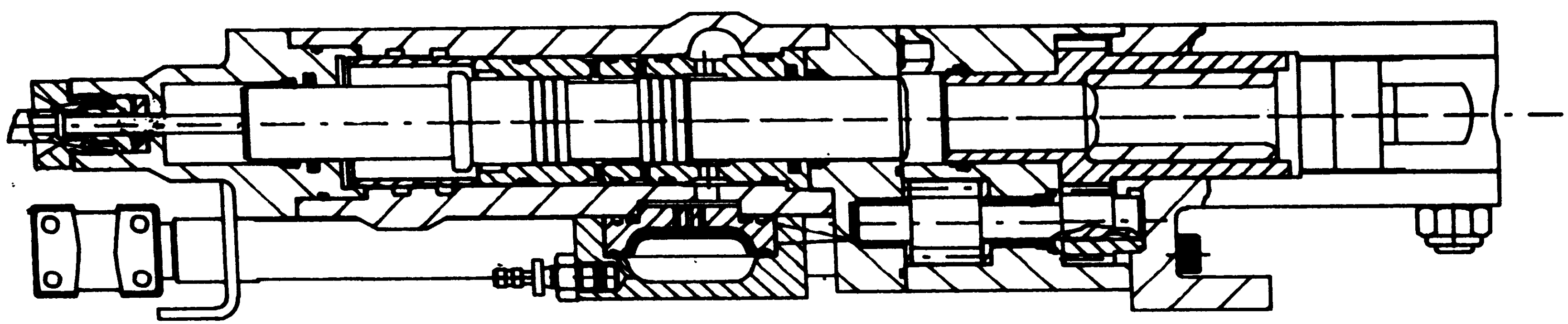

Fig 3 Diagram of oil hydraulic drill

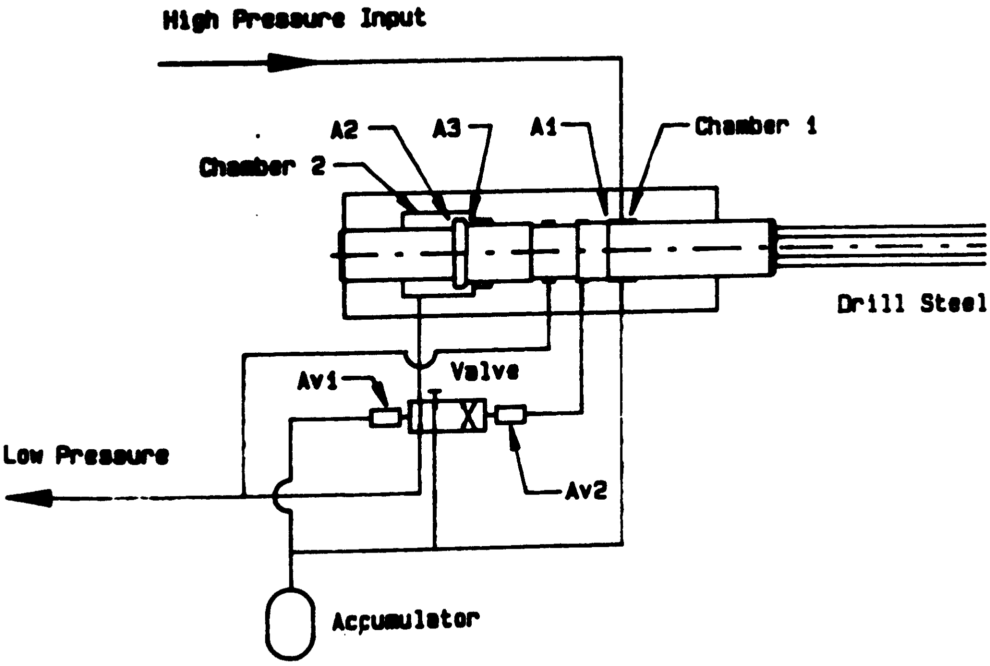

Fig 4 (a) Piston Impacts Drill Steel

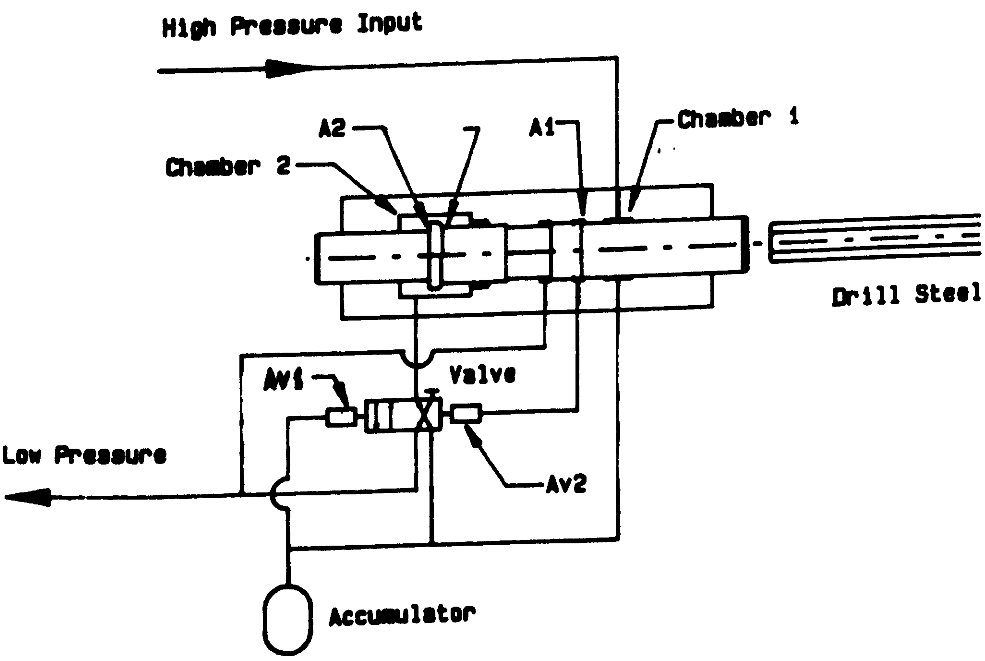

Fig 4 (b) Valve Kicked on Return Stroke

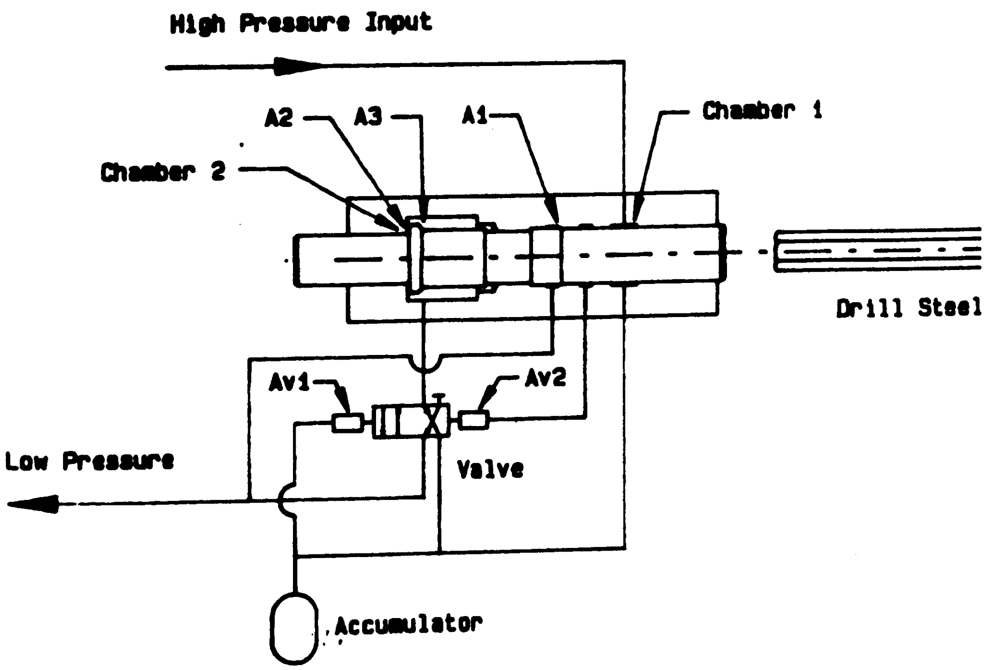

Fig 4(c) Piston Stops at Back of Stroke

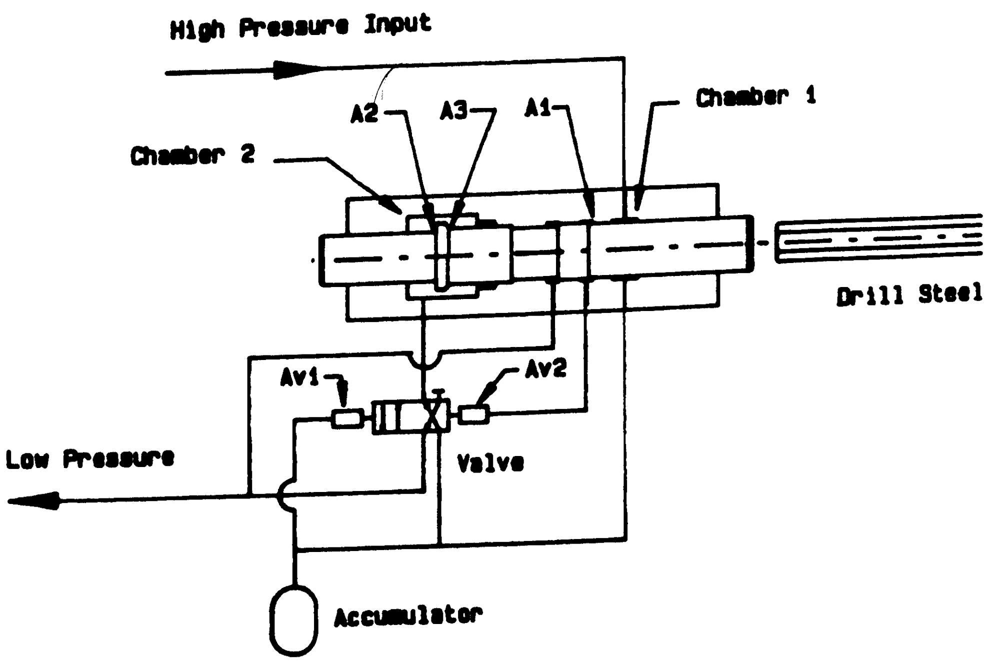

Fig 4(d) Valve Kicked on Forward Stroke

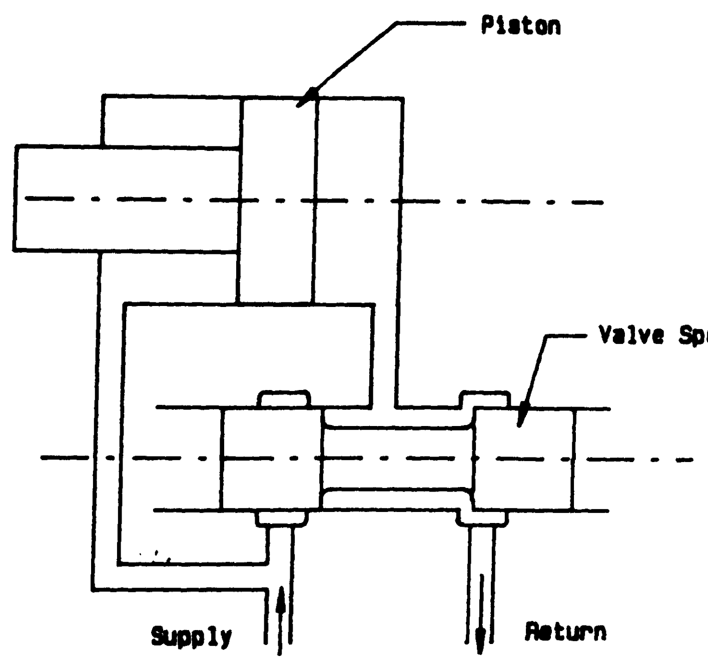

Fig 5 Balanced Three Way Spool Valve

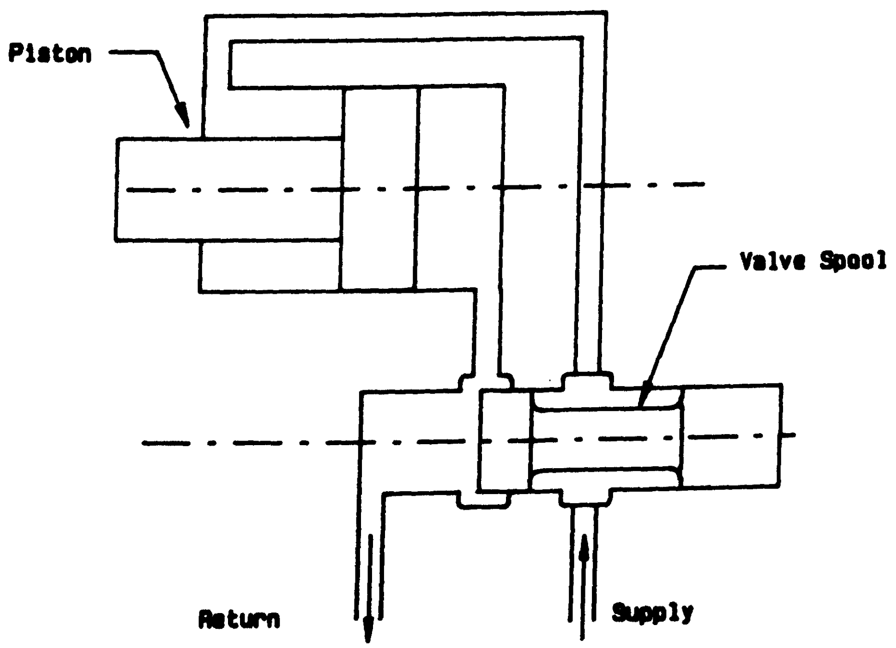

Fig 6 Unbalanced Three Way Valve

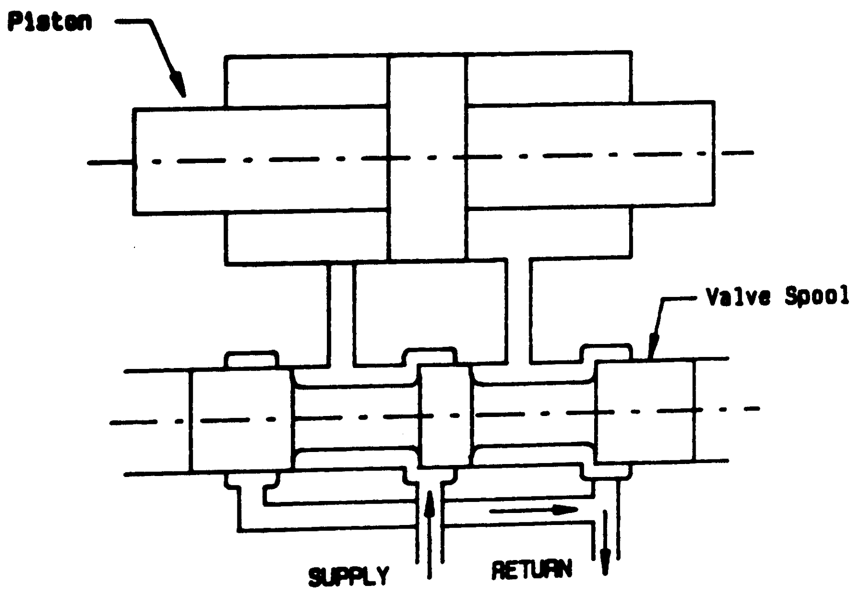

Fig 7 Balanced four way spool valve

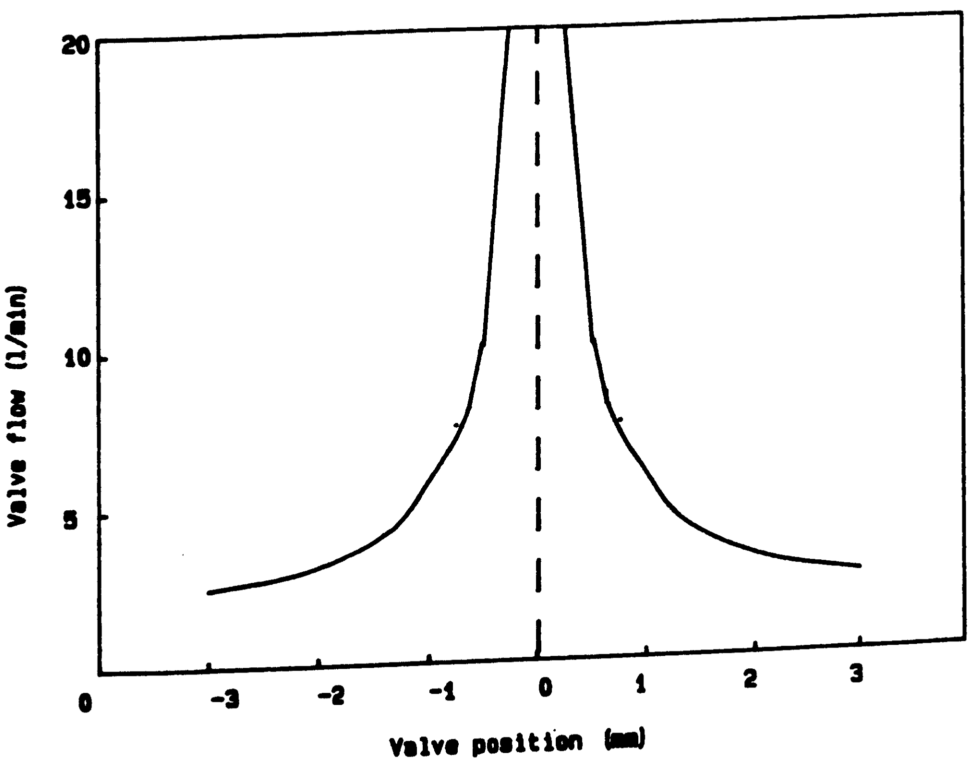

Fig 8 Spool valve characteristics

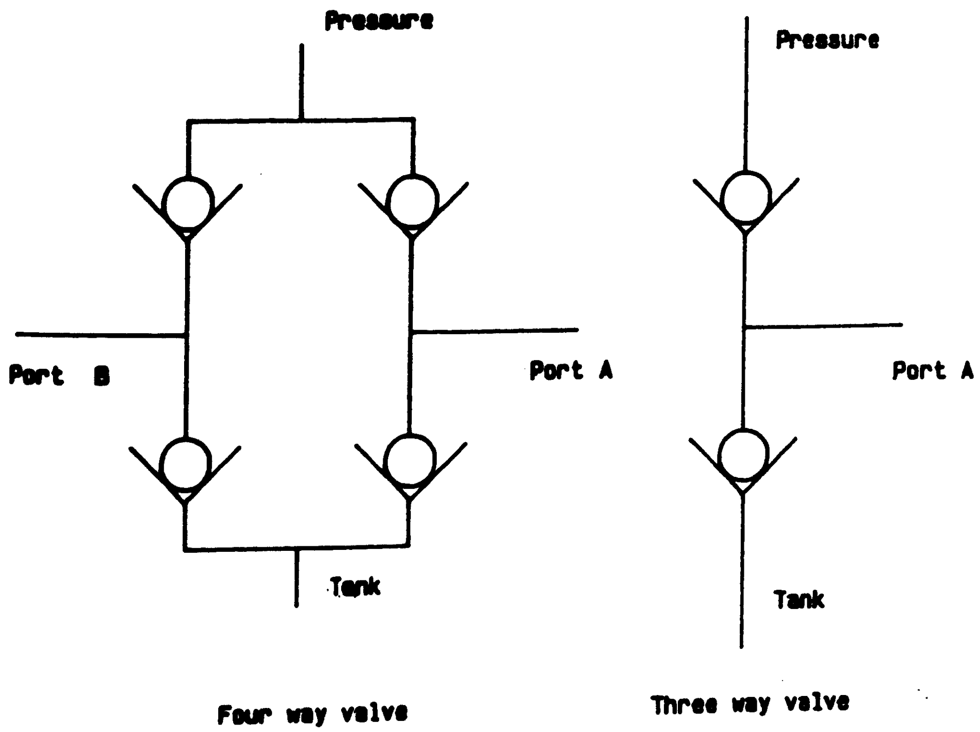

Fig 9 Poppet valve arrangements

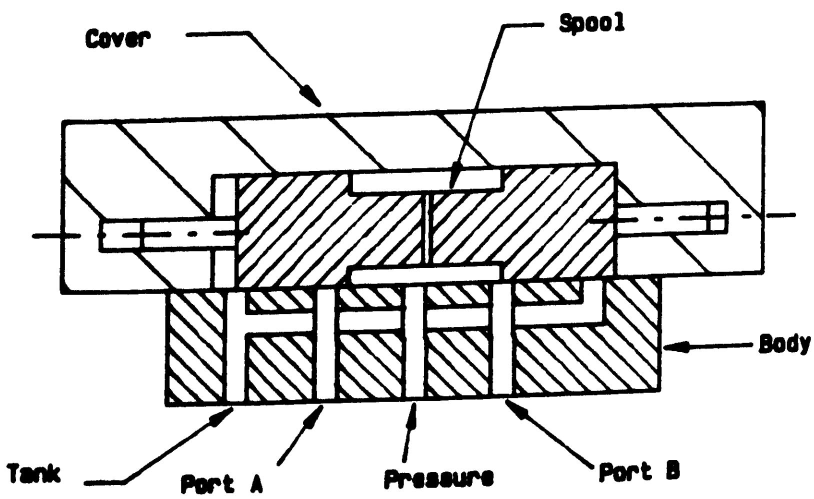

Fig 10 Shear seal valve

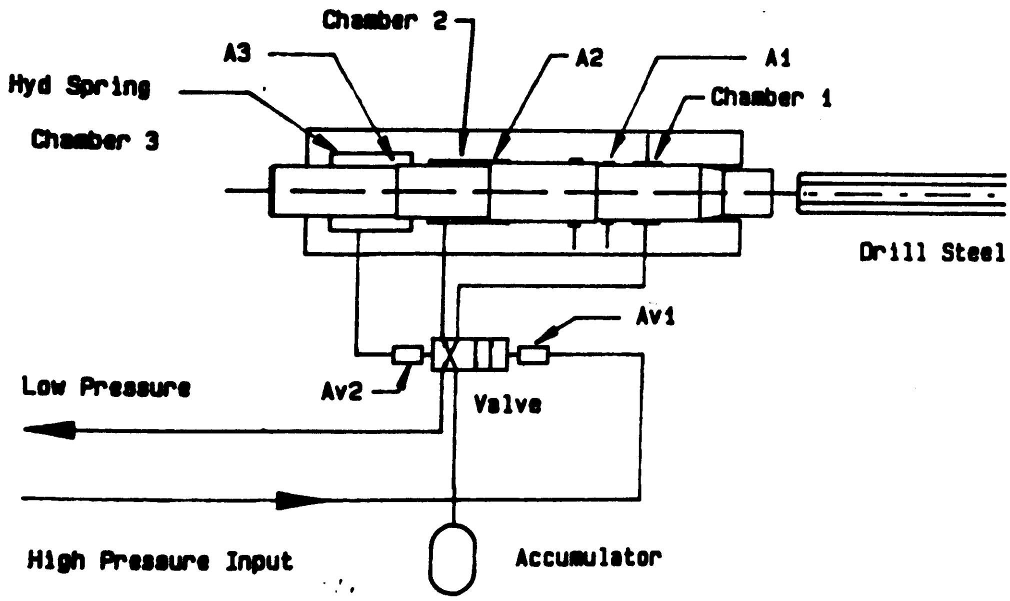

Fig 11 Hydraulic Spring Percussive Cycle

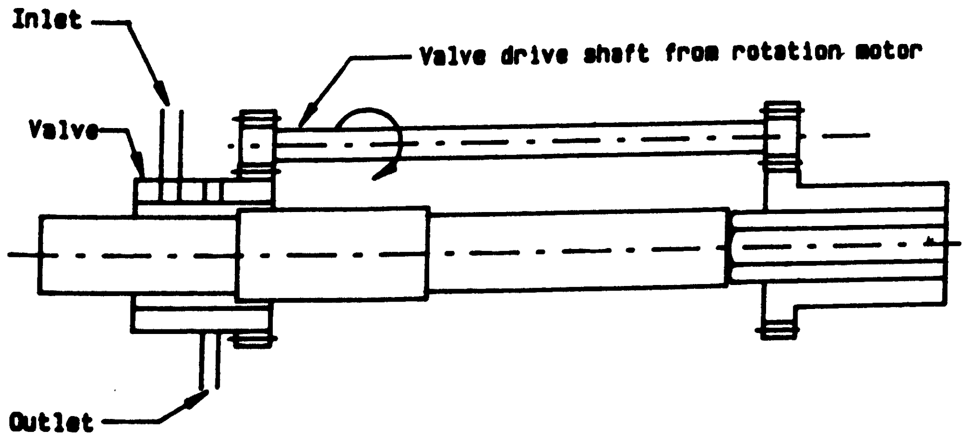

Fig 12 Rotary valve percussive cycle

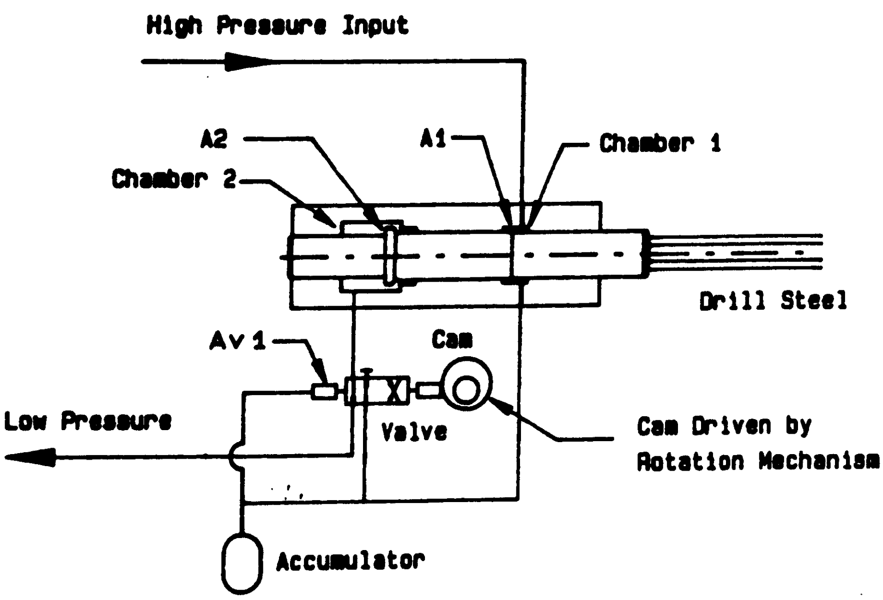

Fig 13 Cam driven valve cycle

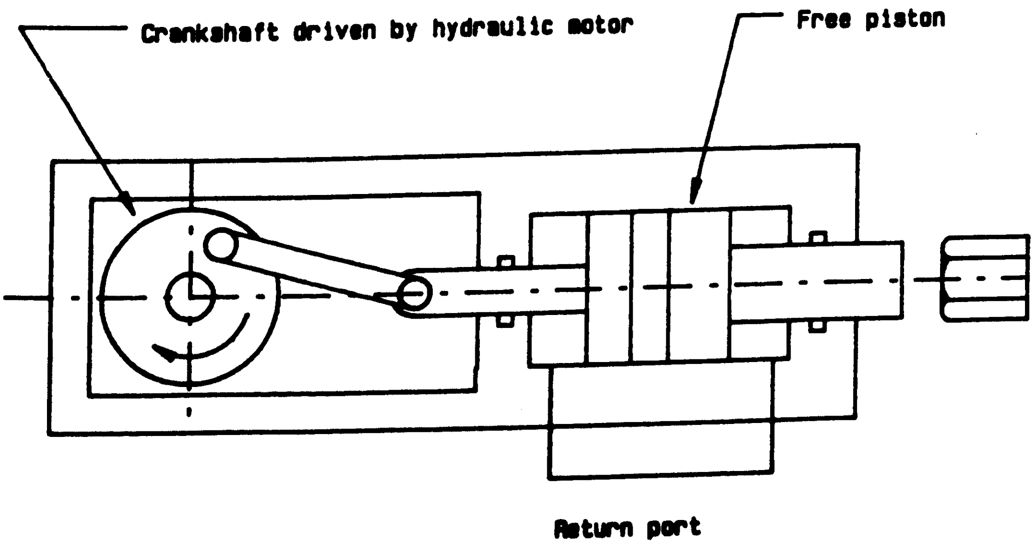

Fig 14 Motor driven piston cycle

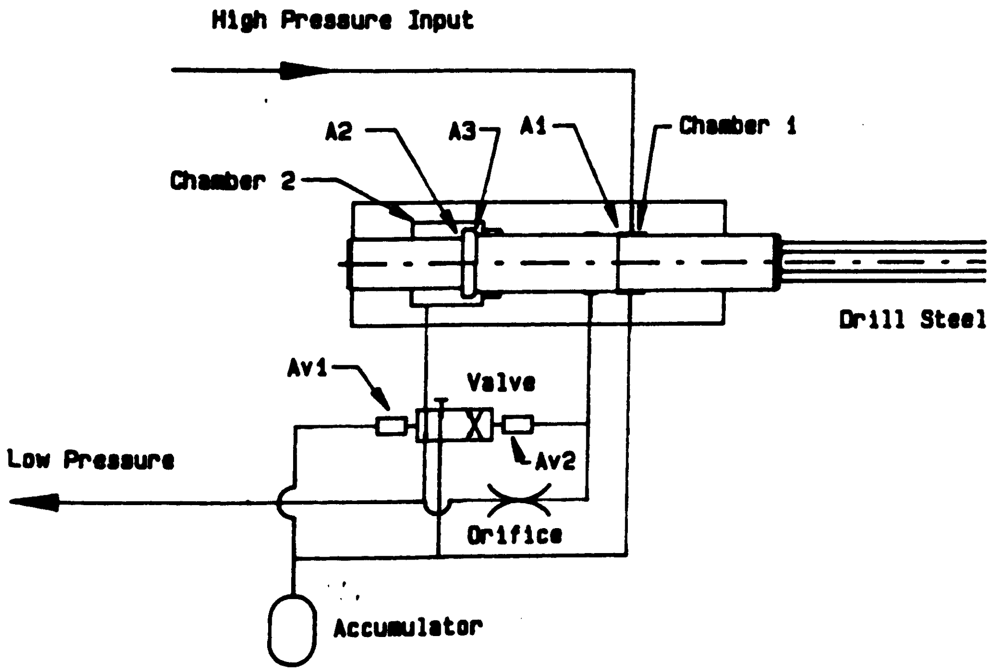

Fig 15(a) Piston impacts drill steel

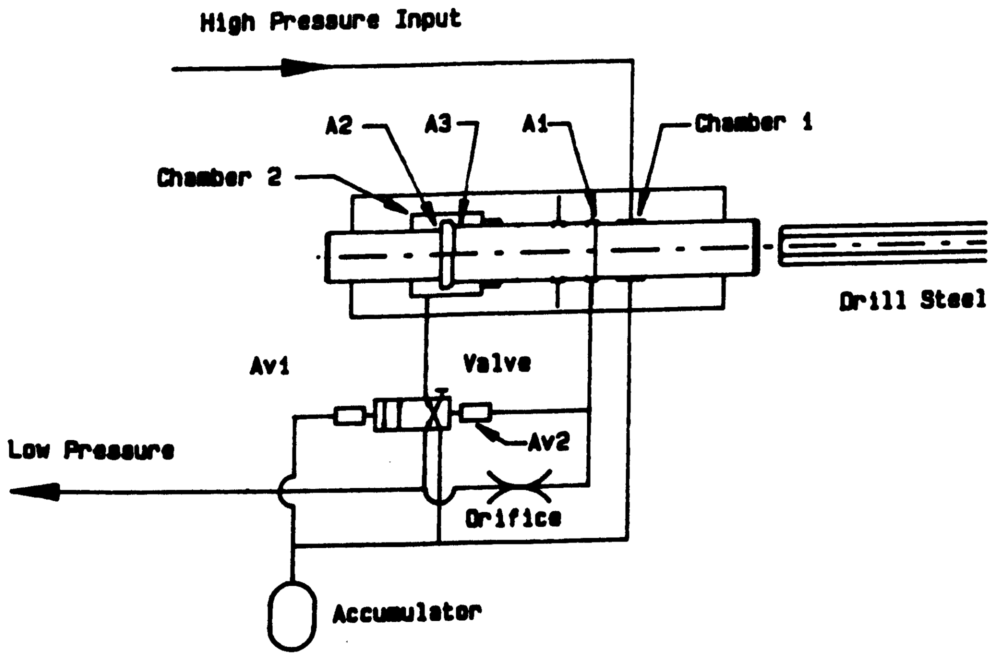

Fig 15(b) Valve kicked on return stroke

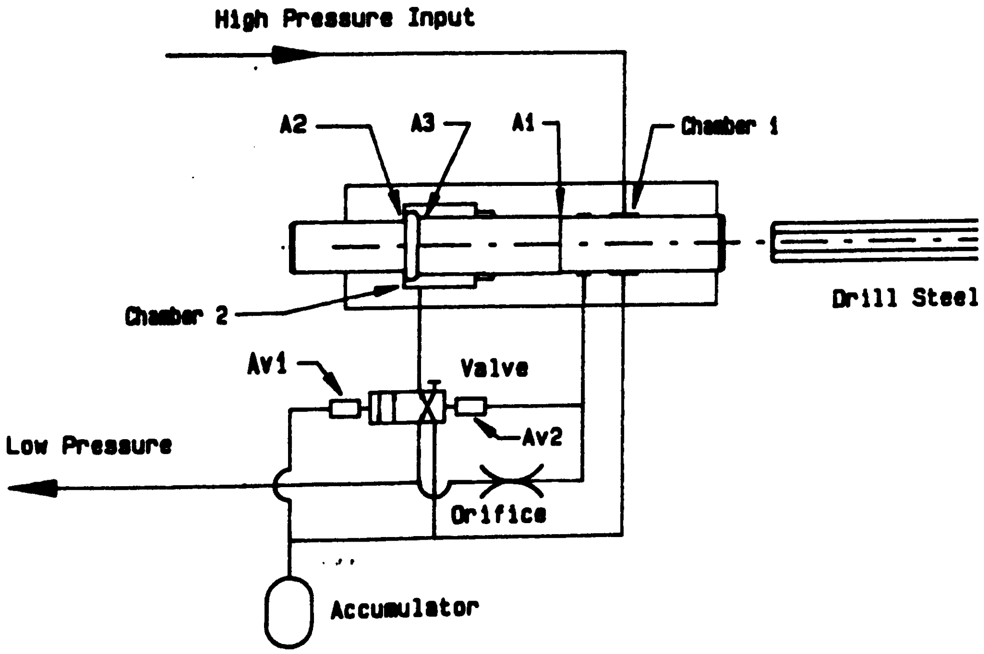

Fig 15(c) Piston stops at back of stroke

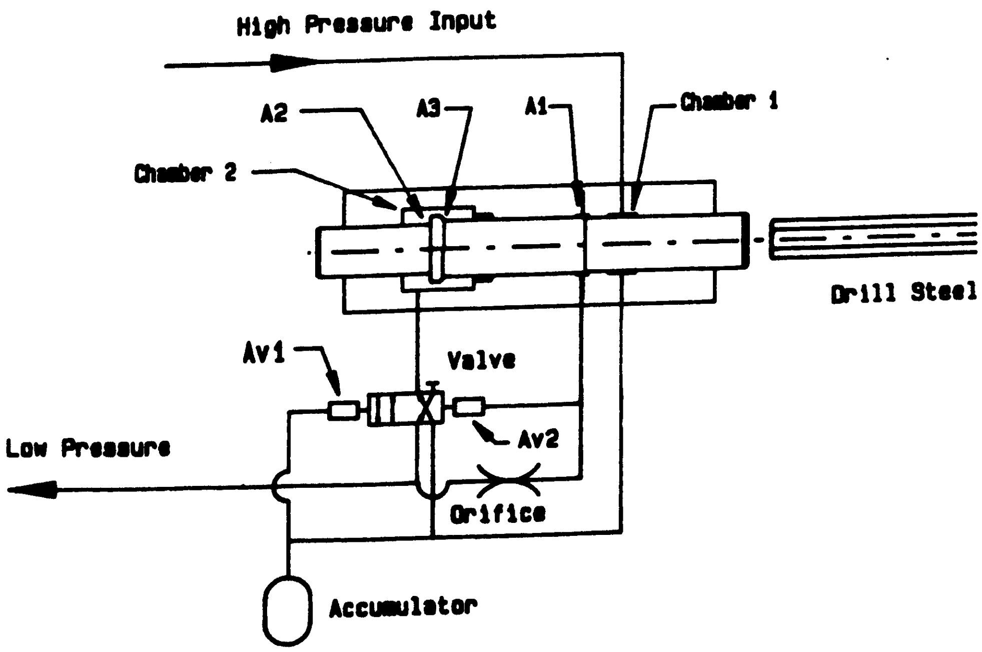

Fig 15(d) Valve kicked on forward stroke

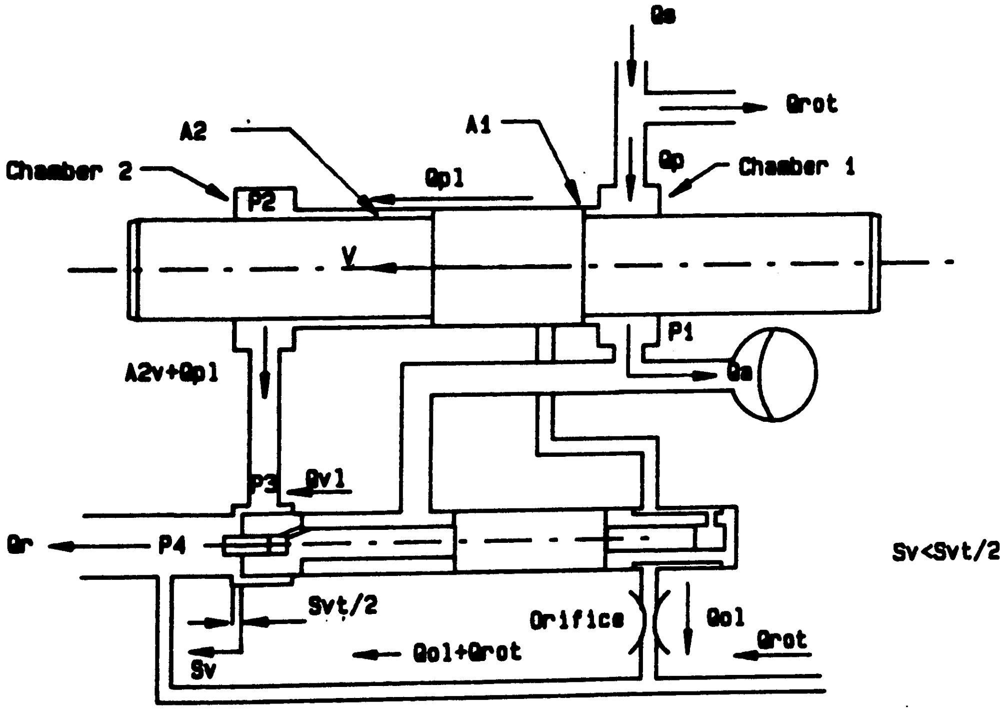

Fig 16 Operation: Phase 1

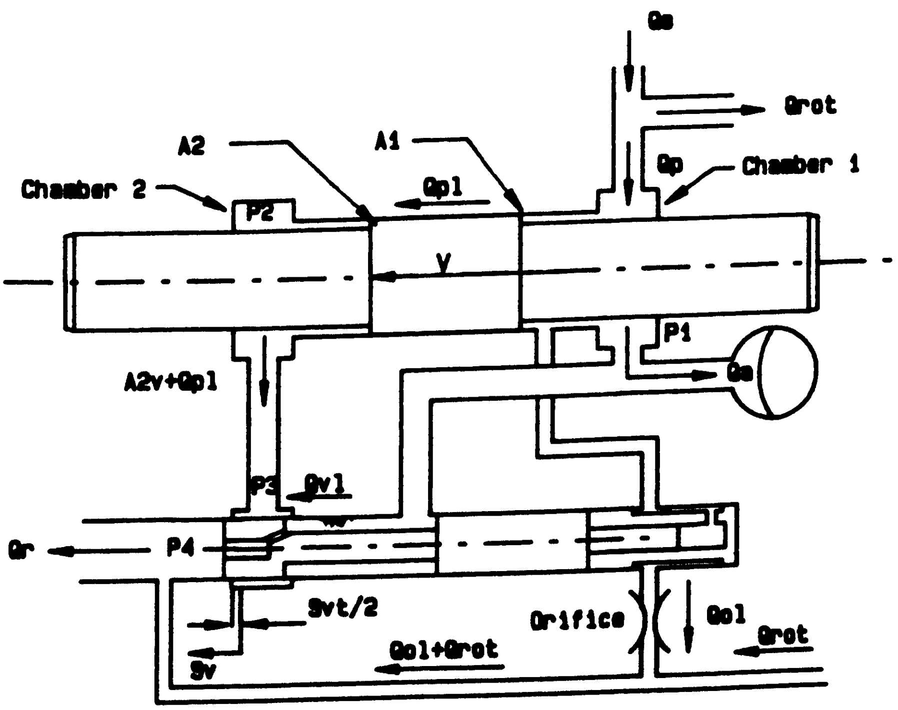

Fig 17 Operation: Phase 2

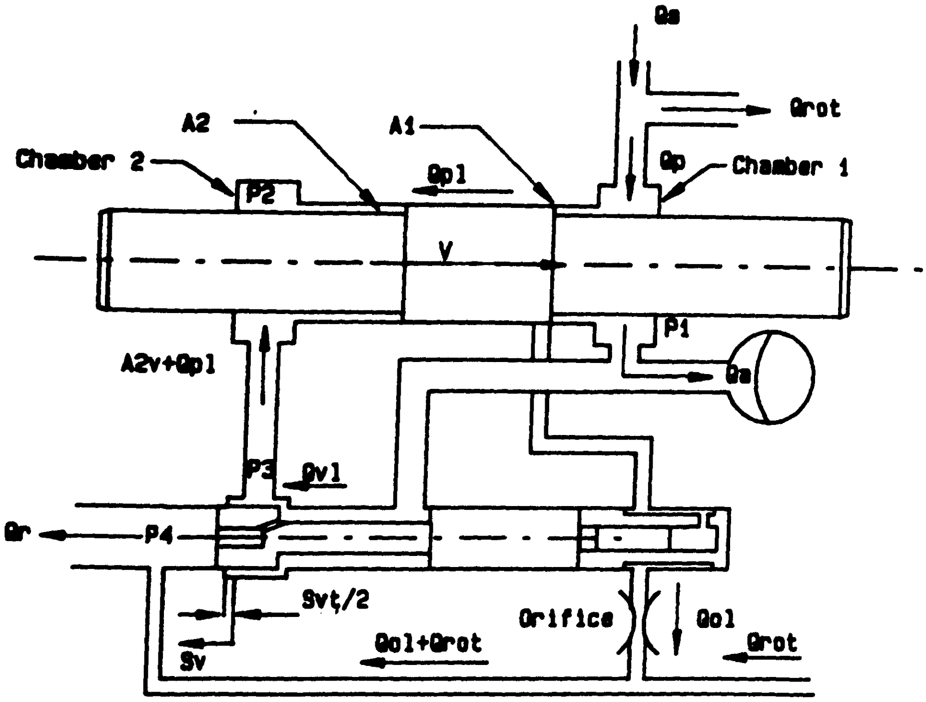

Fig 18 Operation: Phase 3

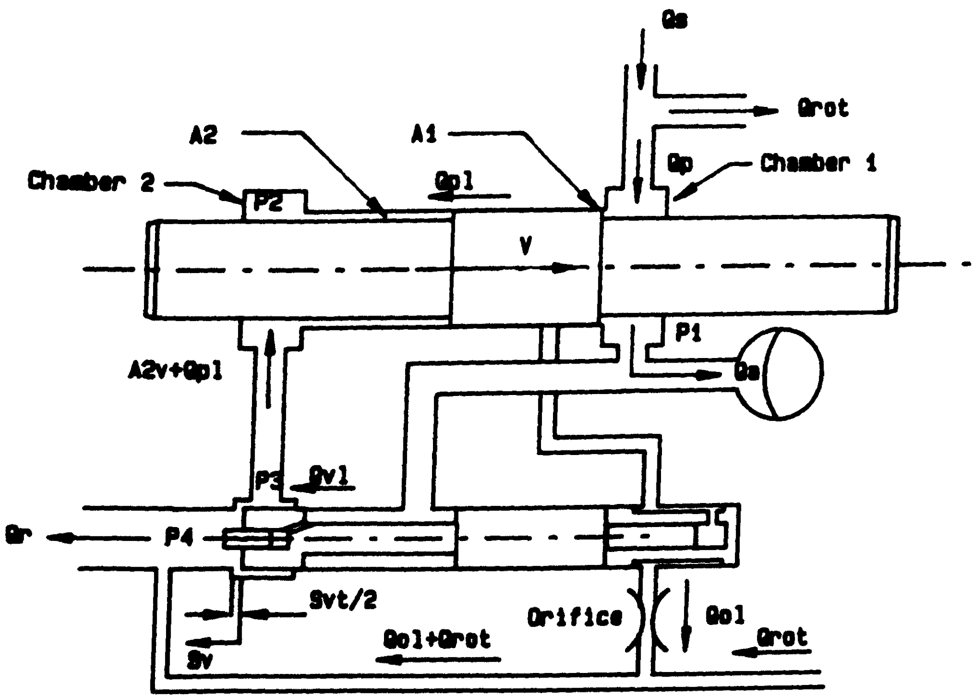

Fig 19 Operation: Phase 4

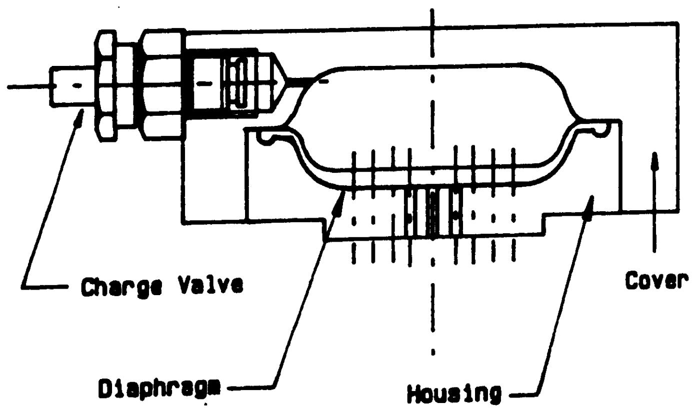

Fig 20 Diaphragm type accumulator

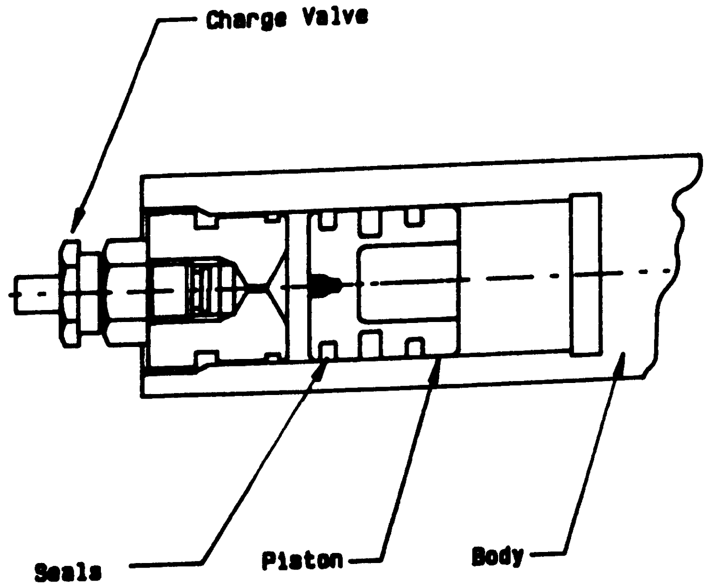

Fig 21 Piston type accumulator

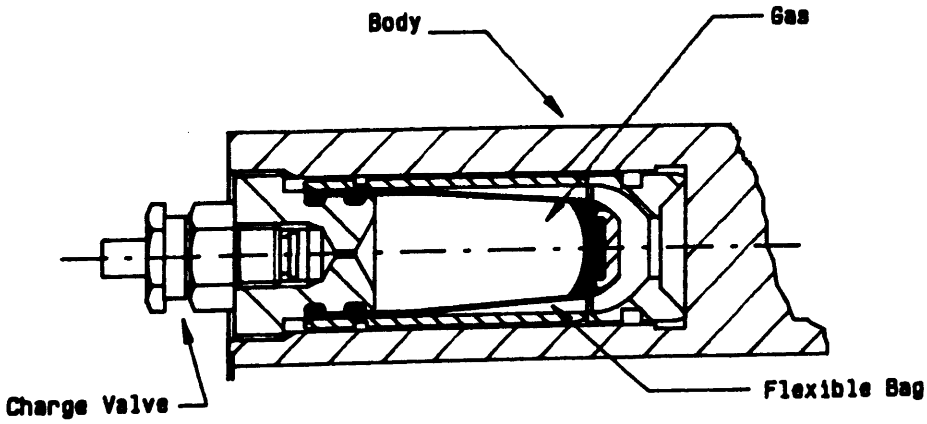

Fig. 22 Bag type accumulator

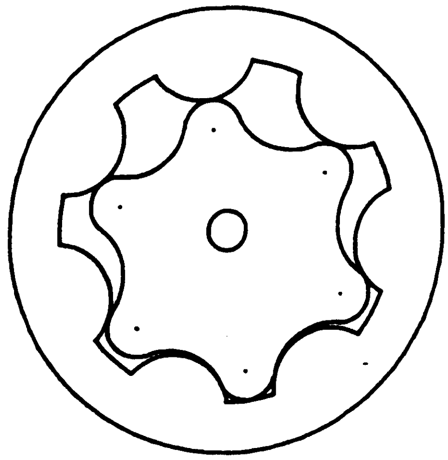

Fig 23 Gerotor type rotation mechanism

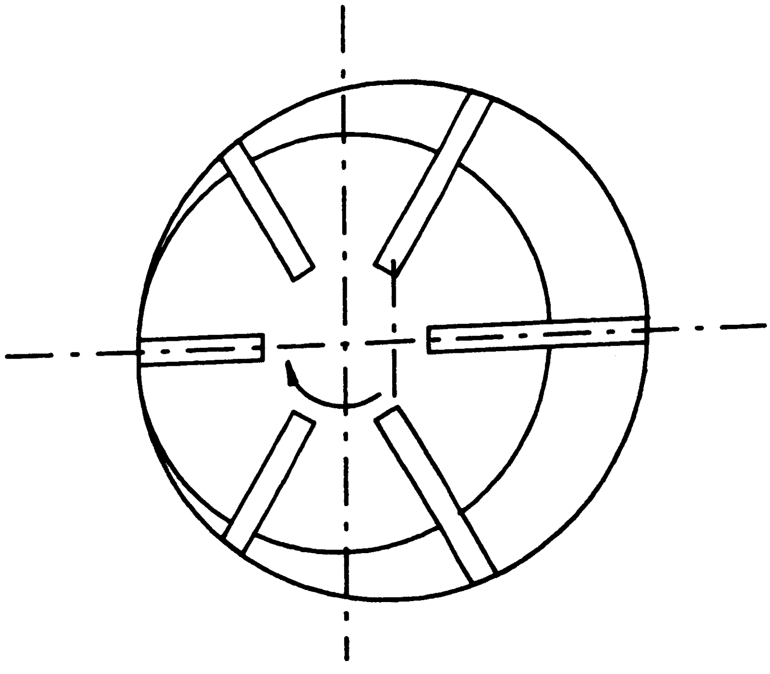

Fig 24 Vane type motor

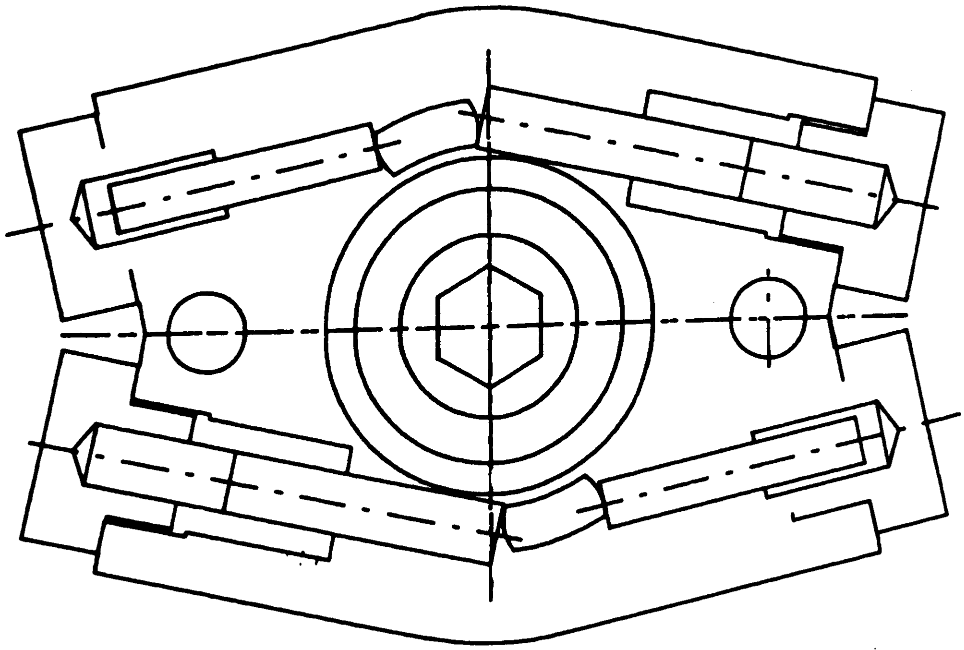

Fig 25 Ratchet type rotation motor

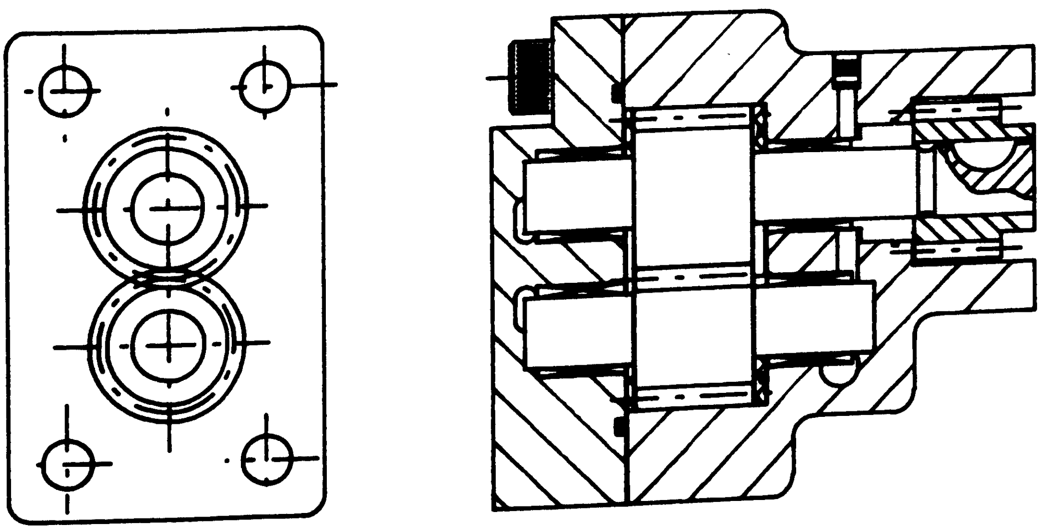

Fig 26 Gear rotation motor

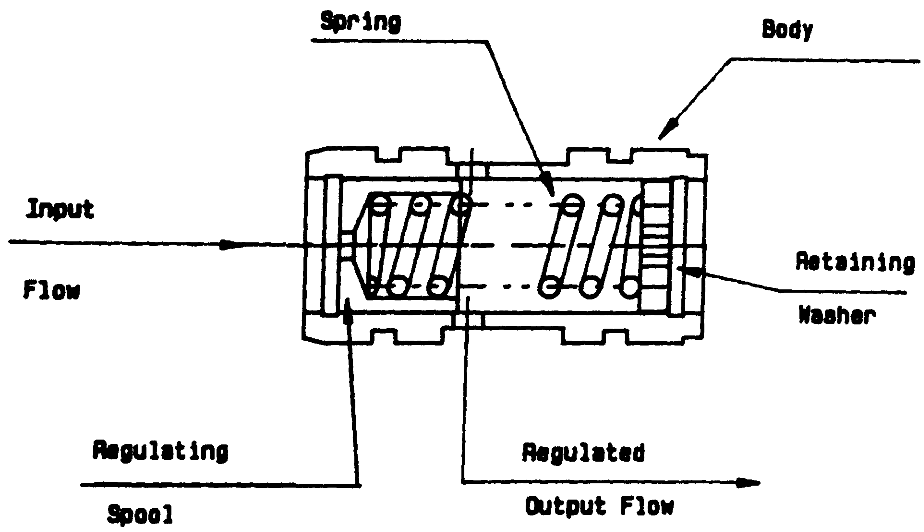

Fig 27 Flow control assembly

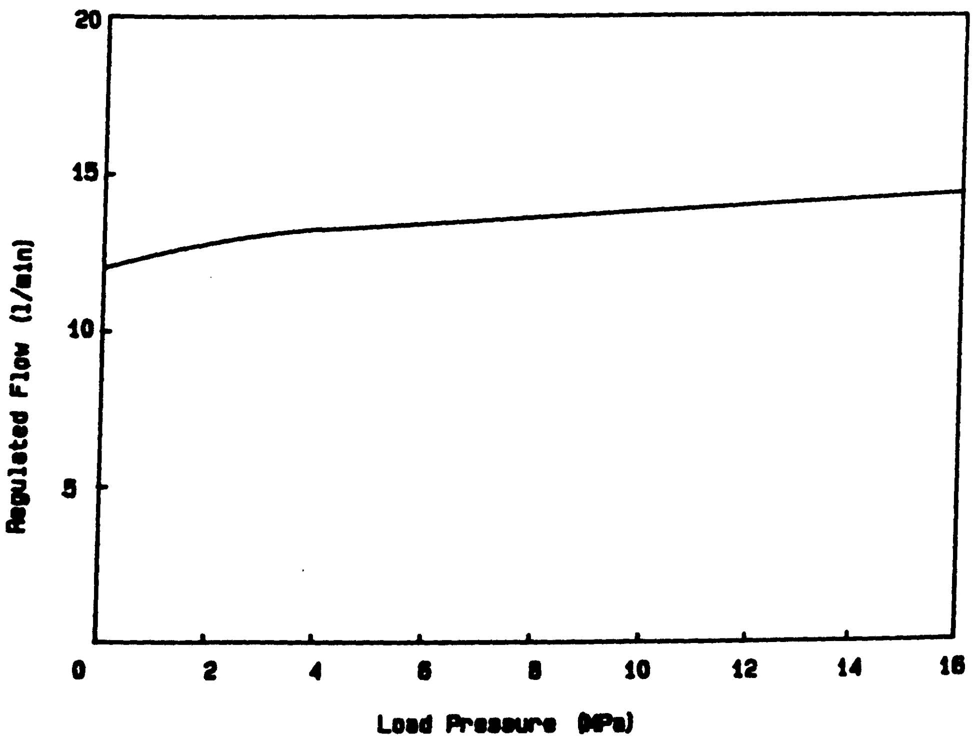

Fig 28 Characteristic curve of flow control valve

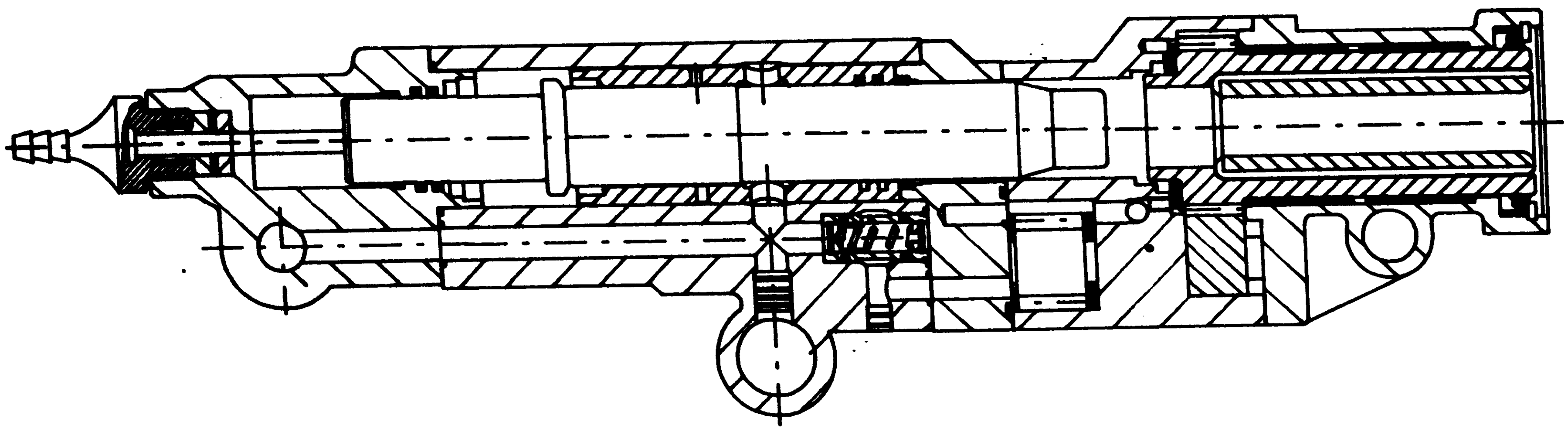

Fig 29 Assembly drawing of final HD30

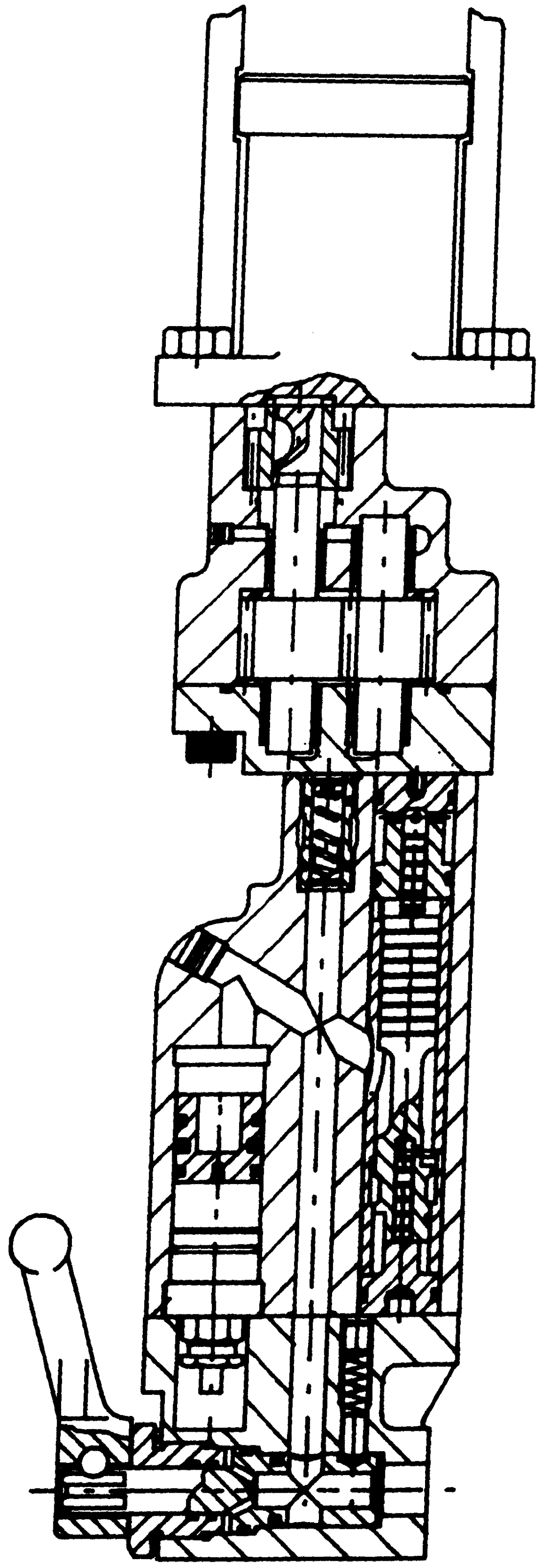

Fig 30 Section through HD30

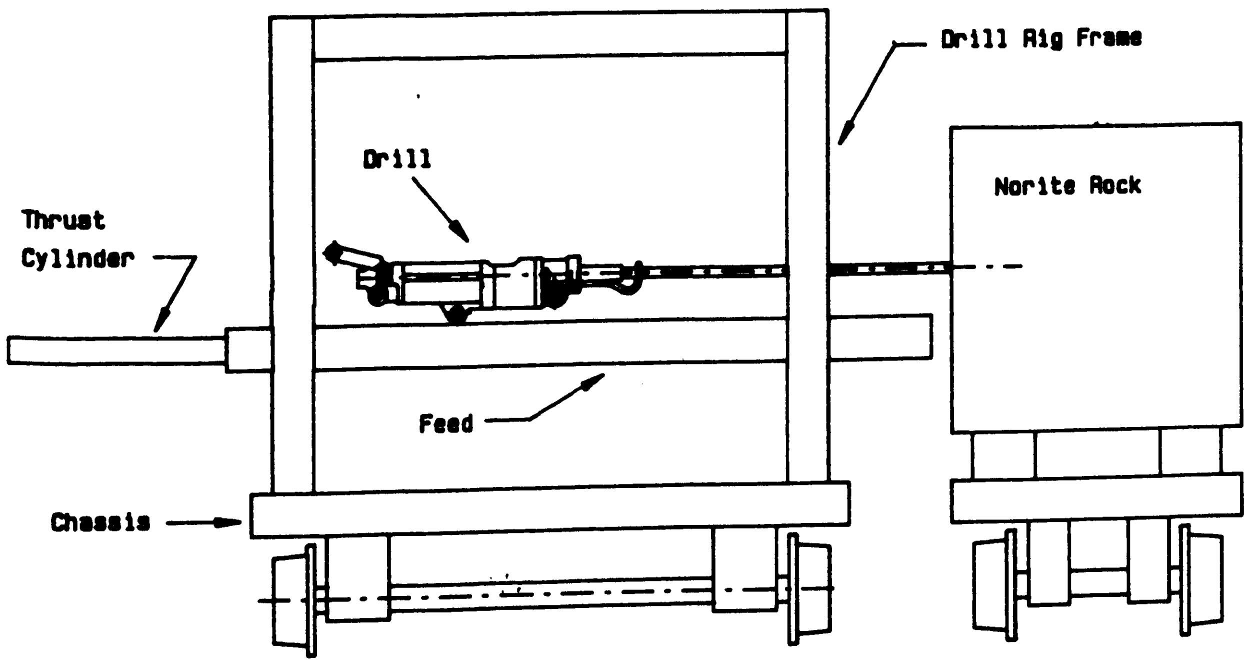

Fig 31 Schematic of test rig

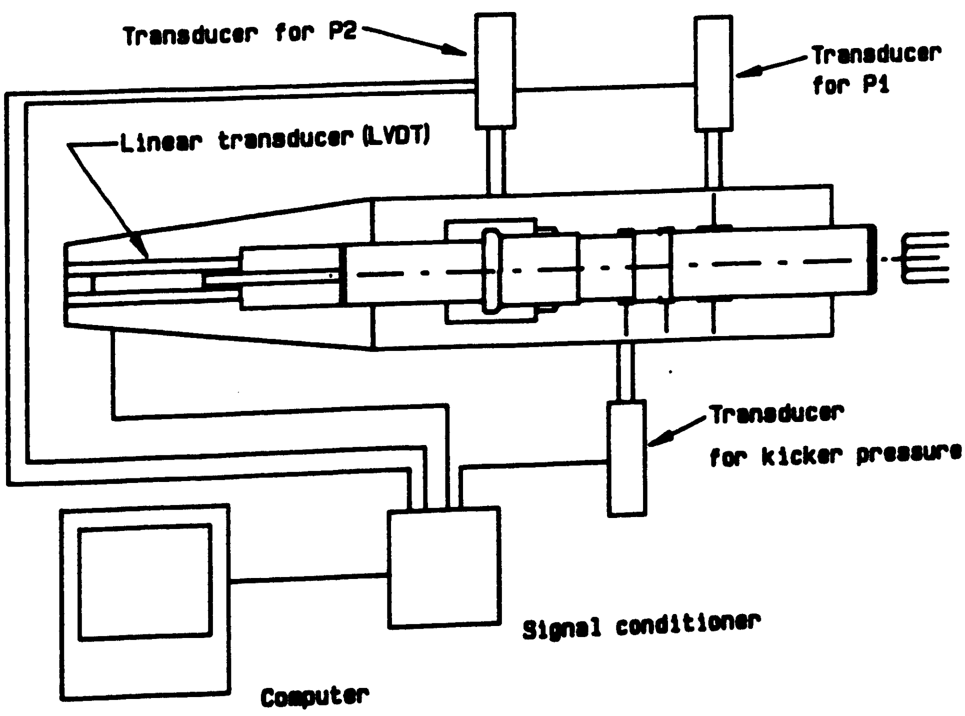

Fig 32 Layout of recorder and transducers

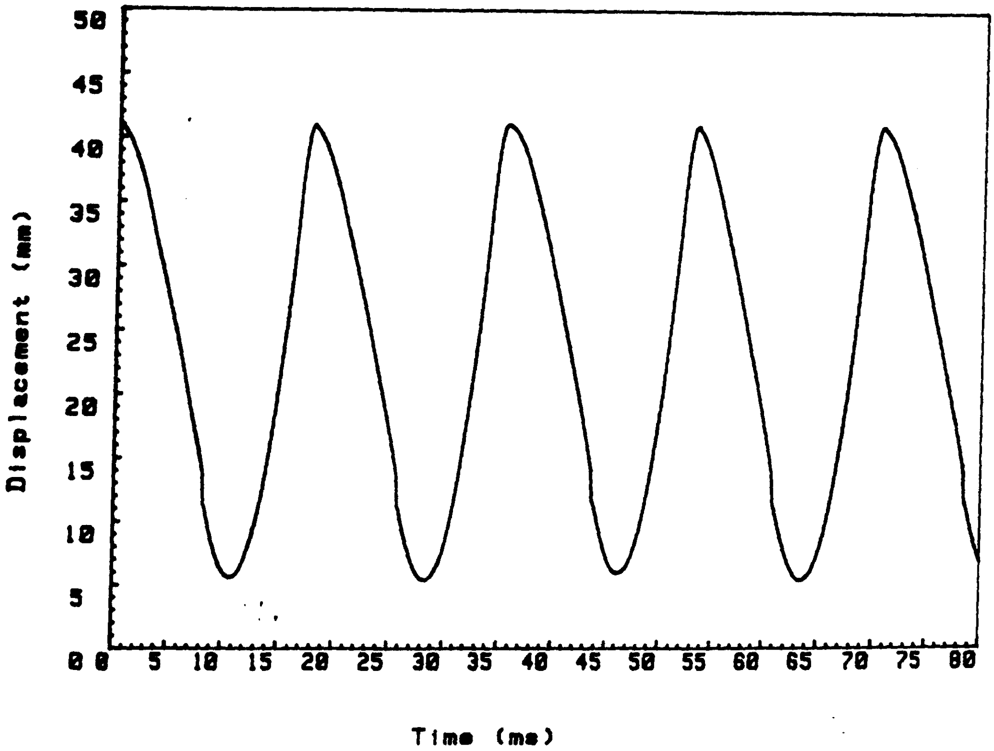

Fig 33 Piston displacement v time(computer)

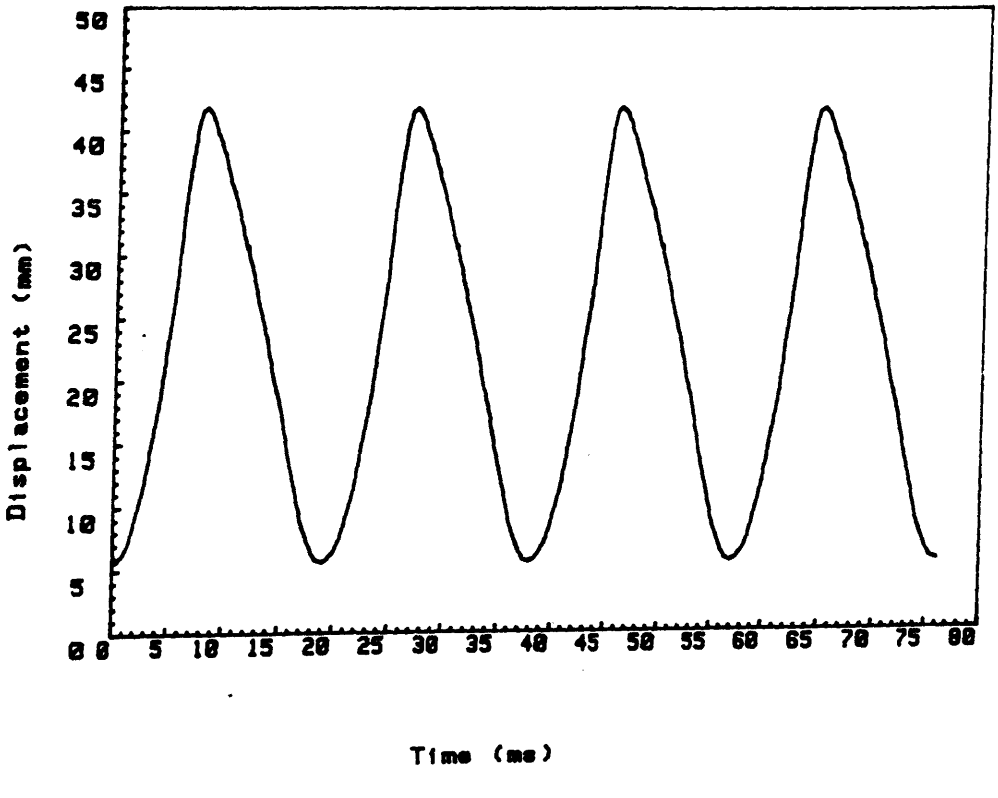

Fig 34 Piston displacement v time(test)

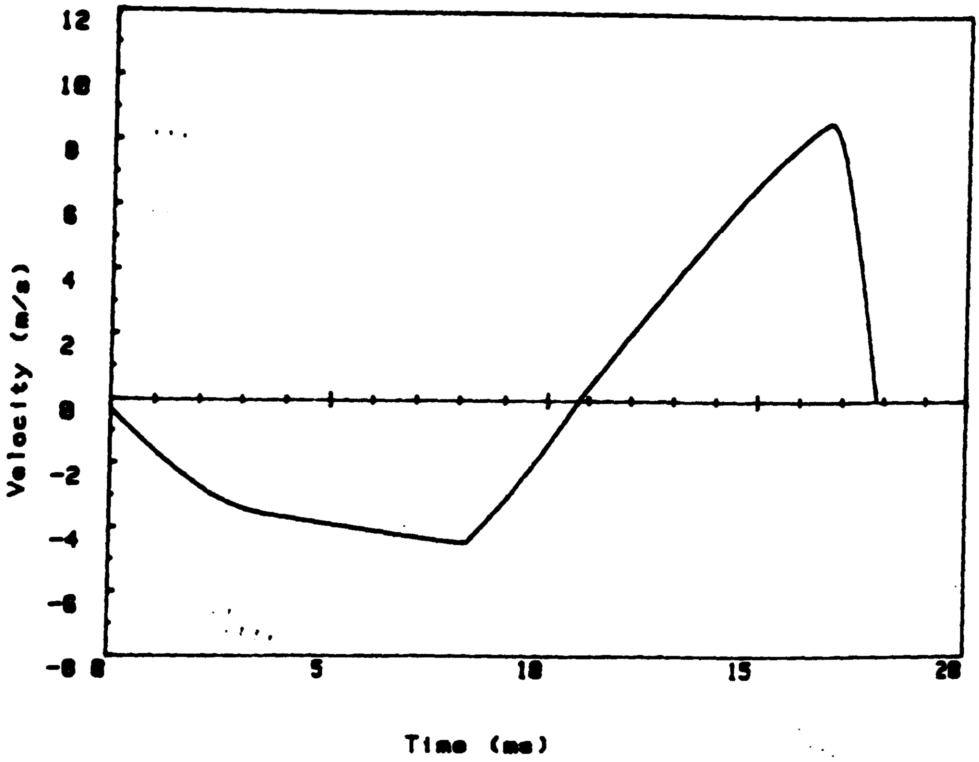

Fig 35 Piston velocity v time(computer)

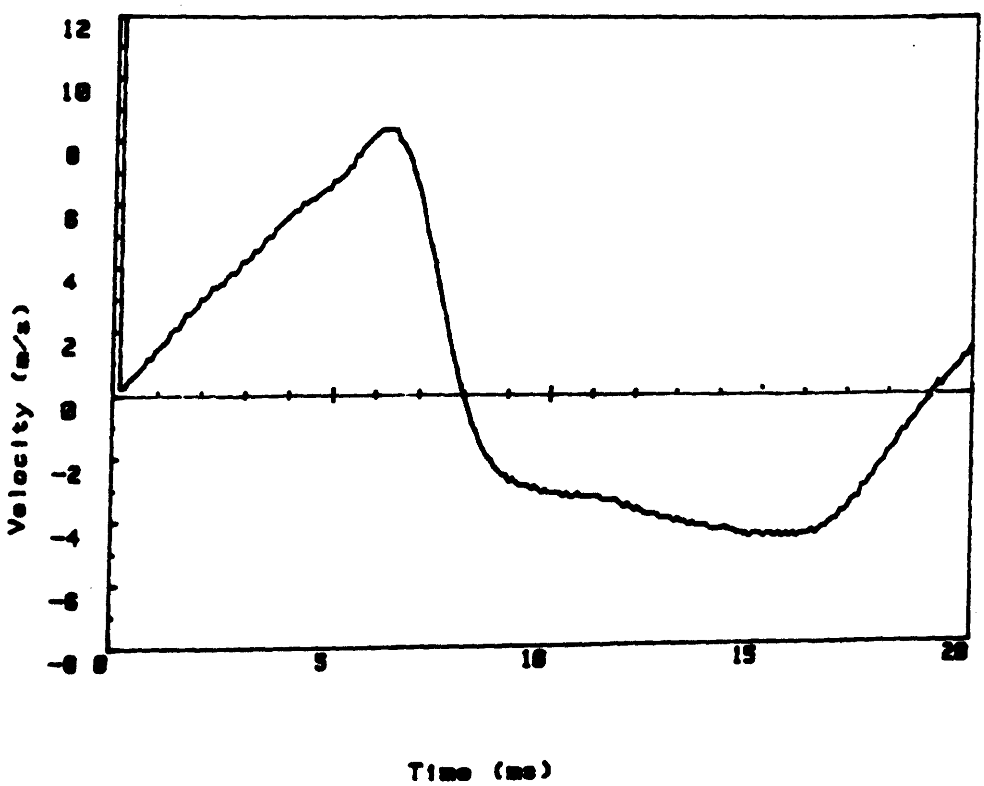

Fig 36 Piston velocity v time(test)

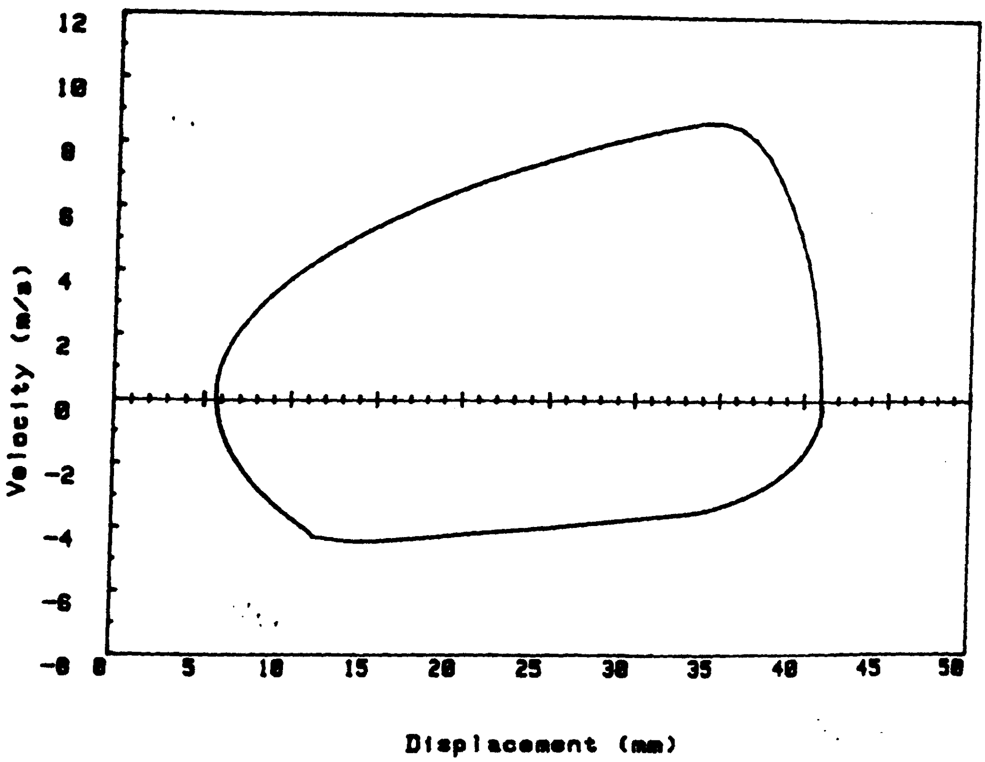

Fig 37 Piston velocity v displacement(computer)

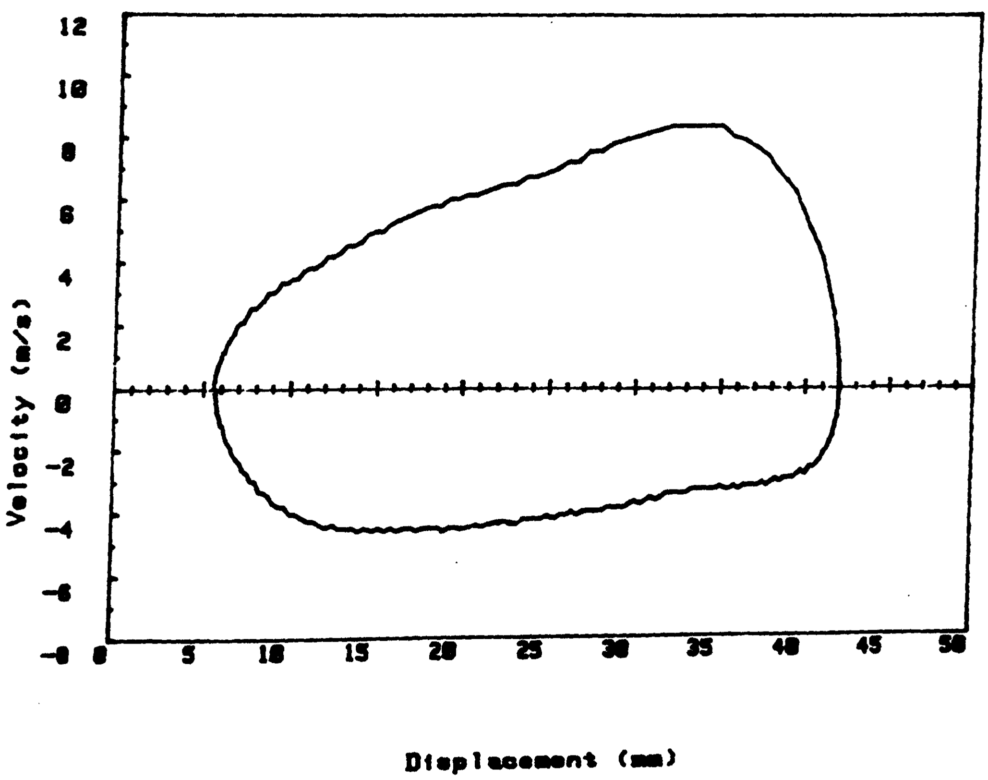

Fig 38 Piston velocity v displacement(test)

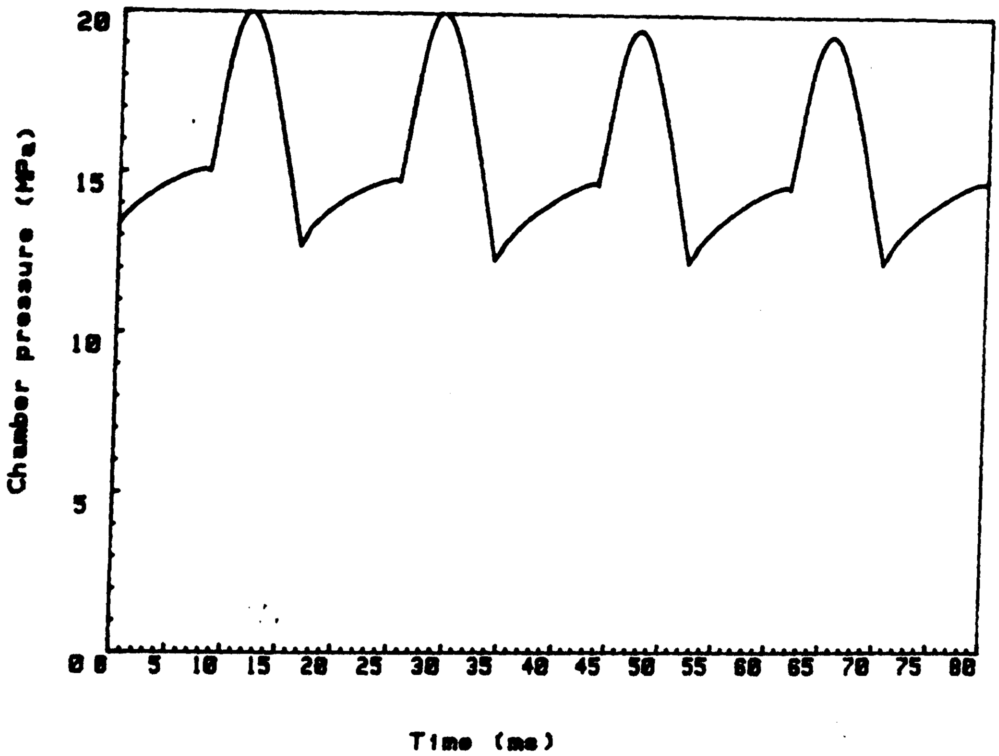

Fig 39 Supply pressure v time(computer)

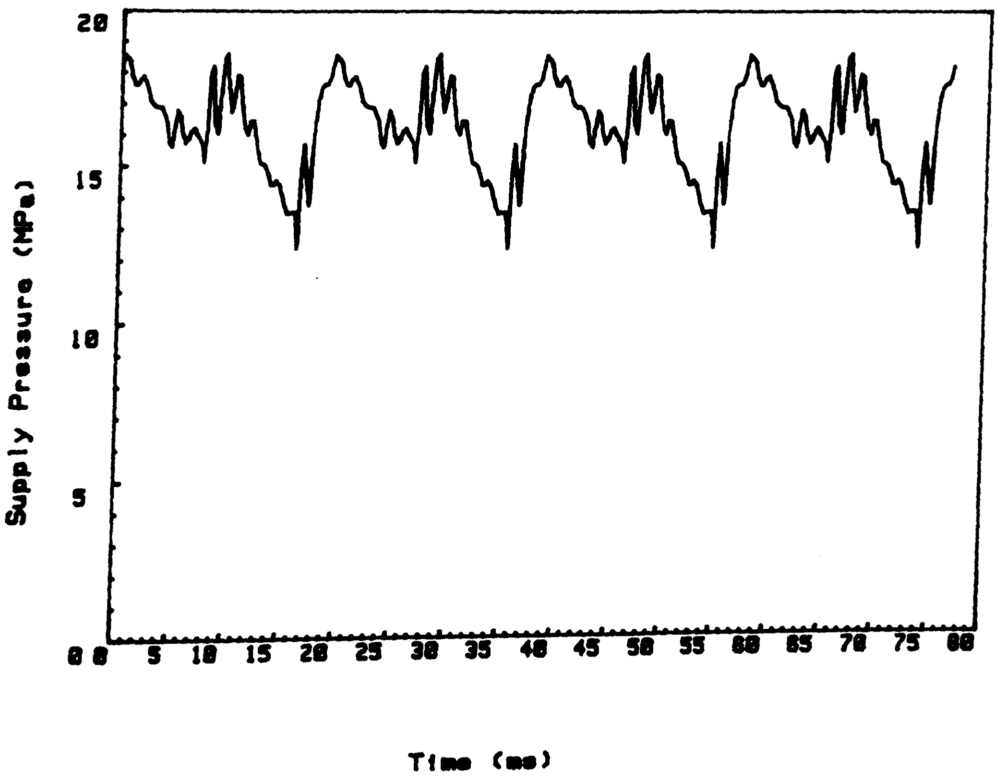

Fig 40 Supply pressure v time(test)

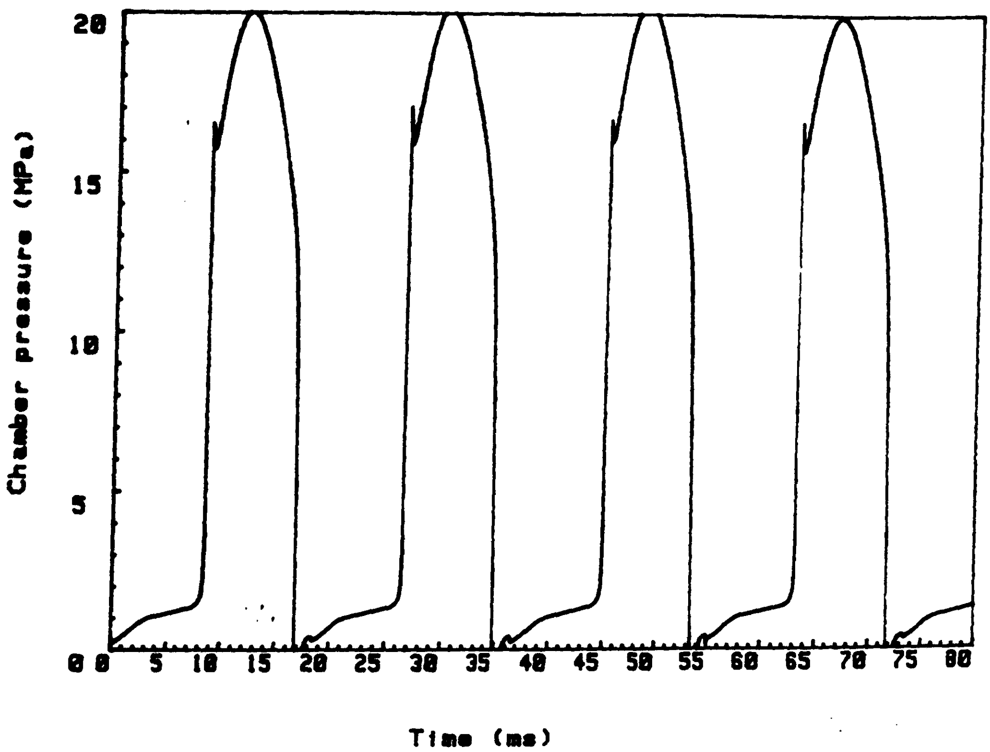

Fig 41 Chamber pressure v time(computer)

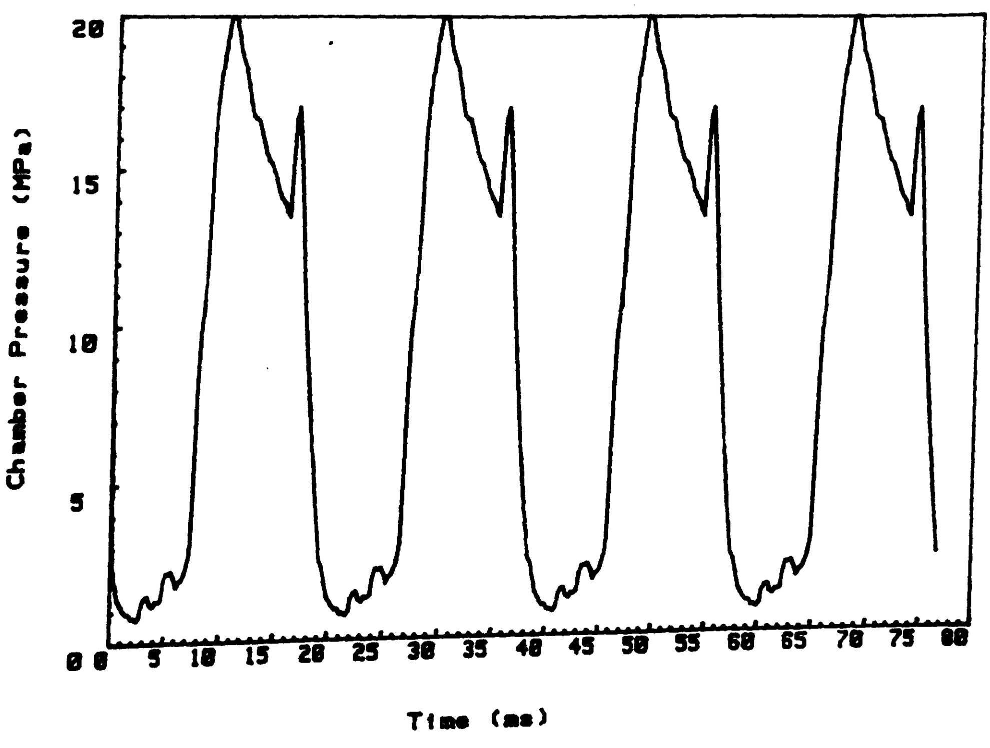

Fig 42 Chamber pressure v time(test)

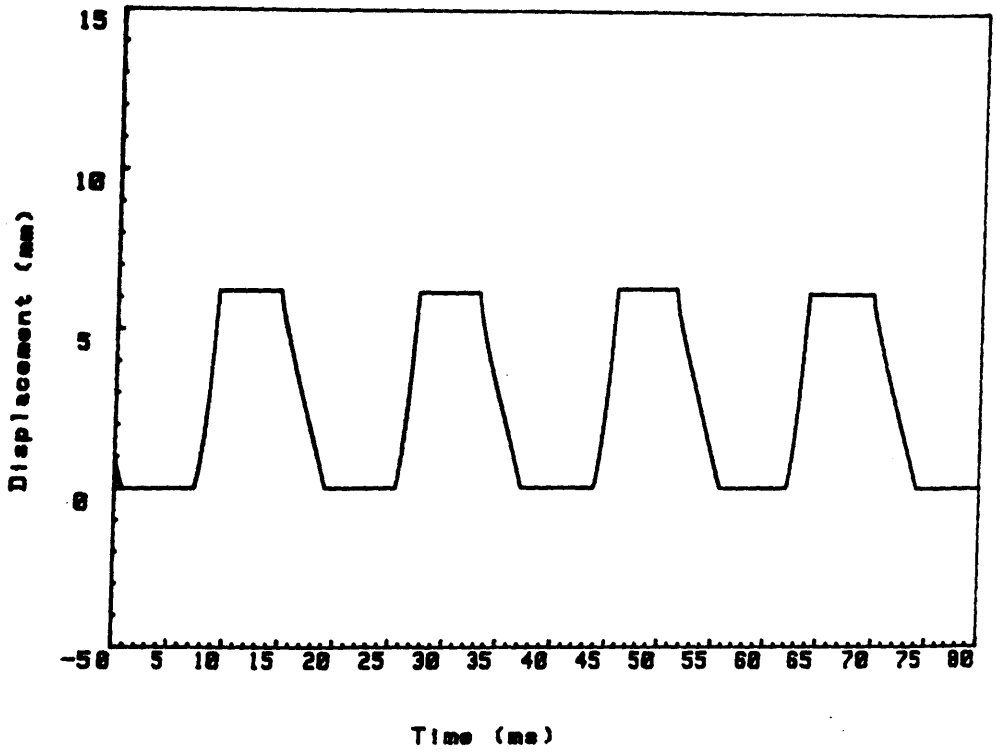

Fig 43 Valve displacement v time(computer)

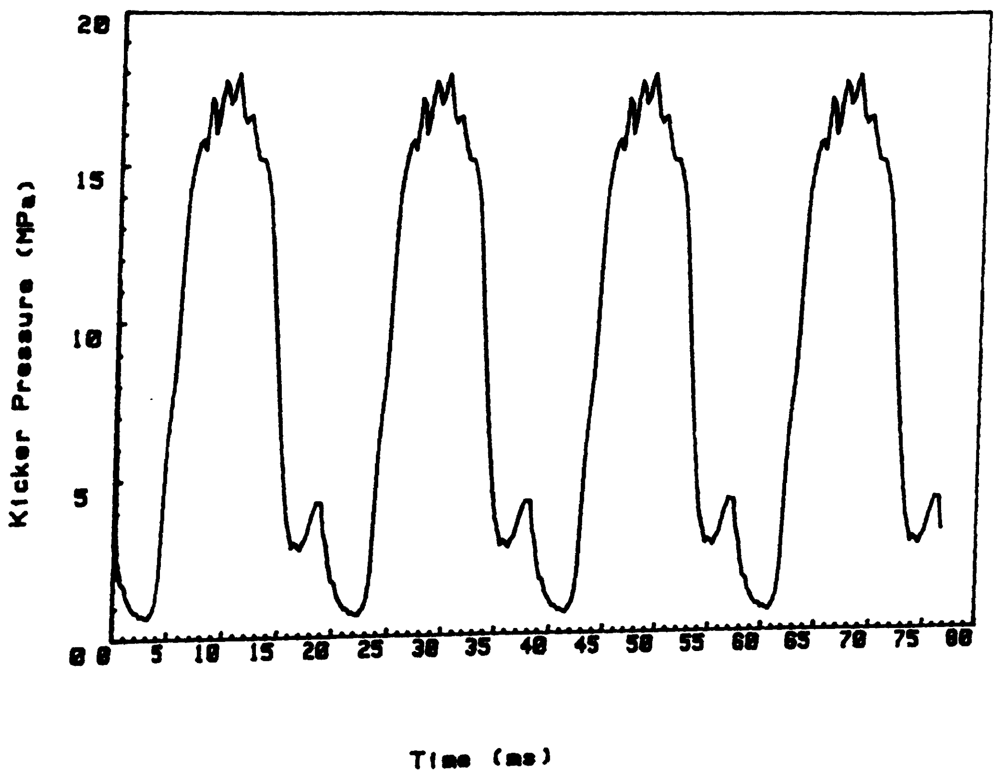

Fig 44 Kicker pressure v time(test)

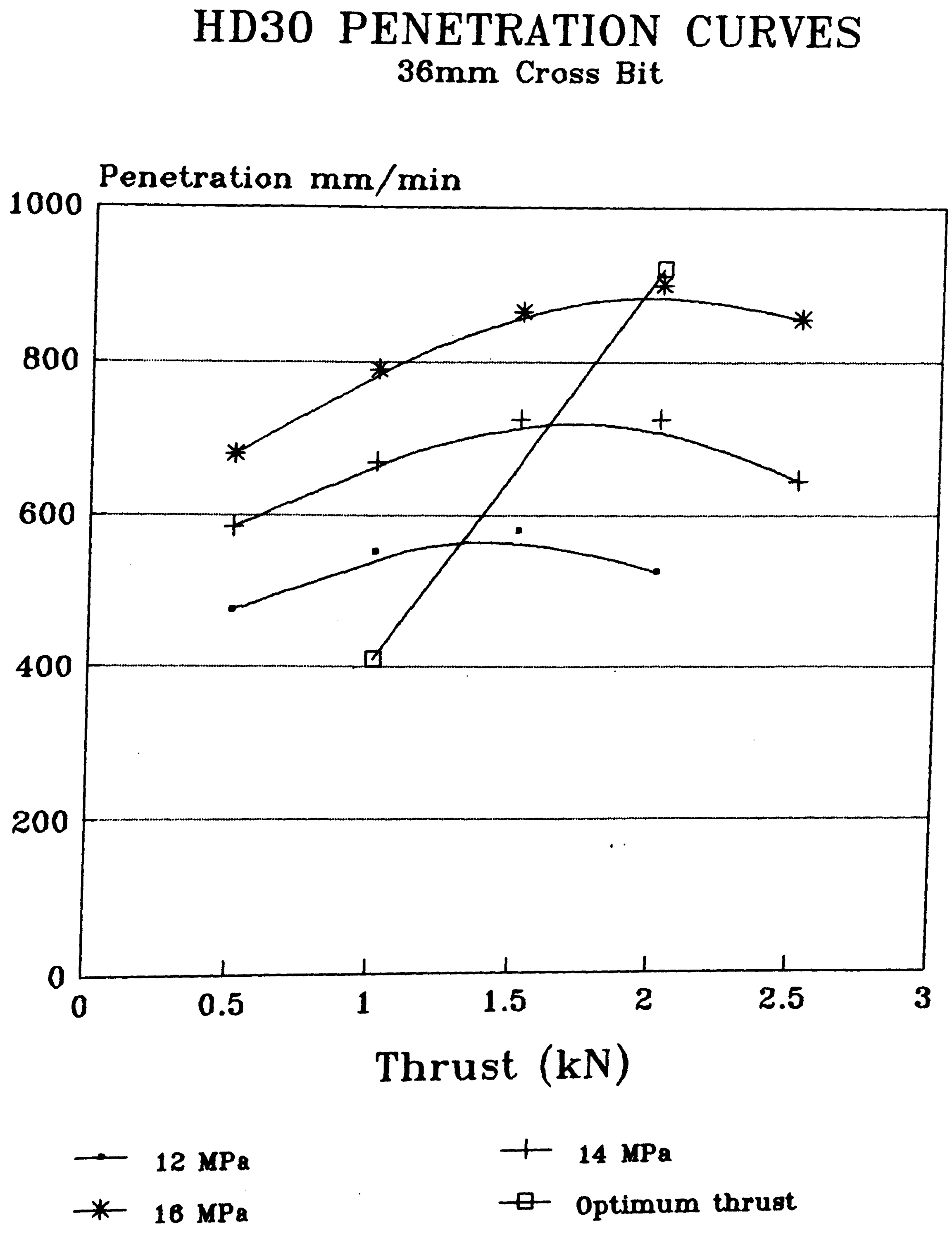

Fig 45 Drill penetration v thrust (test)

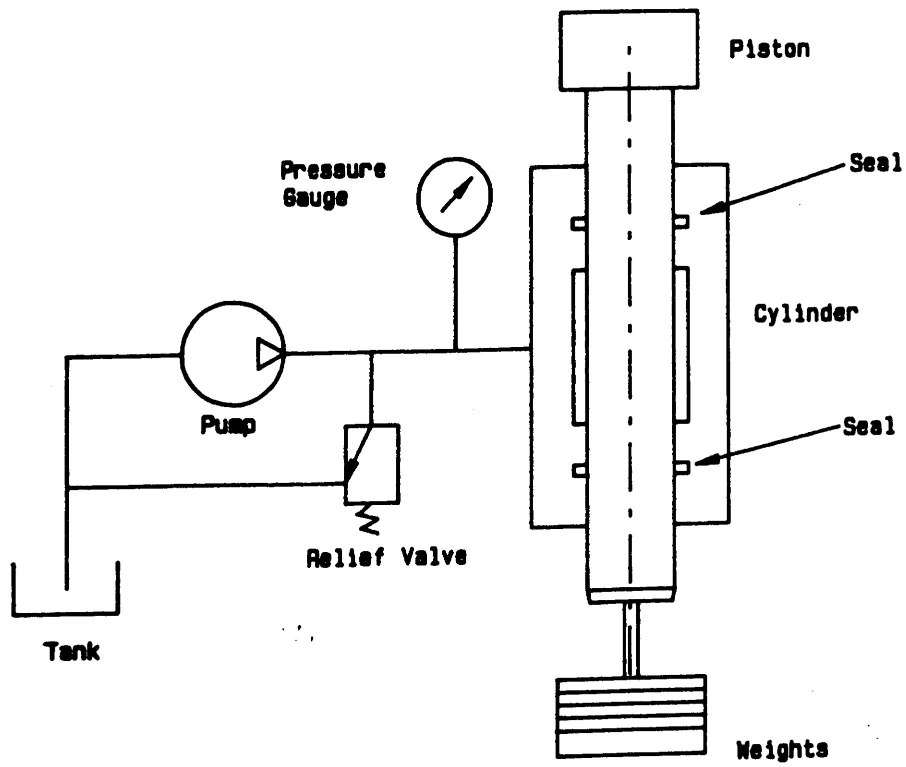

Fig 46 Friction test rig

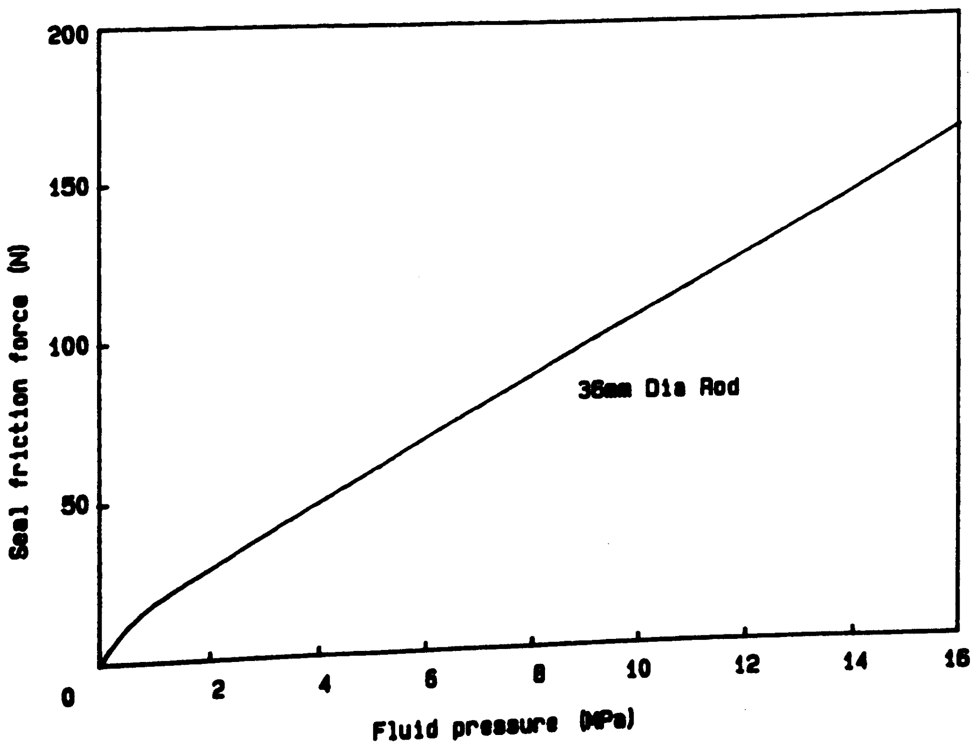

Fig 47 Seal friction v pressure

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