AN EXPERIMENTAL INVESTIGATION
OF FLUID SIGNAL CREATION

By

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PREFACE

For many years, hydraulics has been used as a means of transmitting and multiplying forces in machines, while its capability as a control medium has been overlooked. This control feature has been neglected to such an extent, that it is common practice to use manual, mechanical, or electrical devices for controlling hydraulic machinery. This author has therefore set out to verify that hydraulically-produced pressure signals can be created which are capable of controlling the mechanical motions of machines.

My sincerest appreciation is due Professor E. C. Fitch for his constant advice and invaluable aid throughout the course of this study. I am indebted to Professor A. G. Comer and Professor B. S. Davenport for their competent advice. I wish to thank Dr. J. H. Boggs and Dr. Clark Dunn for making this research project possible. My sincere thanks are extended to Mr. John McCandless and Mr. George Cooper for their assistance in constructing the testing apparatus. I am also indebted to Professor C. M. Leonard for his suggestions and advice in writing this thesis.

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CHAPTER I

INTRODUCTION

The word "hydraulics" is commonly associated with the field of civil engineering and is usually understood to concern such matters as pipeline design, the flow, erosion, and damming of rivers, and the flotation of ships. This investigation, however, is concerned with none of these but instead with the branch of hydraulic engineering which applies the principles of hydraulics as a means of operating and controlling the mechanical motions of various types of machinery. In this investigation, hydraulic power is used as a means of multiplying, transmitting, and controlling forces in machines.

Hydraulics has been used as a method of transmitting and multiplying forces in machines for many years while the control concept of hydraulics is relatively new. Common applications of hydraulic force transmission and multiplication are present in such machinery as lathes, grinders, milling machines, hydraulic lifts, and hydraulic presses. Here, however, the hydraulic system has usually been controlled by some other means, such as manual, mechanical, or electrical devices. Published evidence indicates that very little work has been done in utilizing hydraulics as an automatic control system. If the hydraulic-force-producing system could also provide its own method of control, the mechanical or electrical system could be eliminated.

Control of an all-hydraulic system can be accomplished by pressure, or pilot, signals. These pressure, or pilot, signals can be created by a pressure-sequence valve, which opens at a predetermined pressure level to supply fluid which is capable of controlling various hydraulic components. This fluid which passes through the sequence valve can be used to energize a directional-control valve which causes operations in other parts of the circuit to take place.

A series of operations such as clamping, drilling, grinding, milling, and unclamping a manufactured part can be performed automatically by selecting a system of pressure-sequence valves which work in conjunction with other components such as cylinders, directional-control valves, and fluid motors.

If an operation is to be timed with respect to some other operation or if it is to take place in some complicated sequence, a "brain circuit" can be developed to handle this problem.

A hydraulic brain circuit is analogous to an electric relay circuit in that it can perform the more complex operations of timing and counting. It is important that the hydraulic brain circuit receives dependable signals, just as the electric brain circuit is dependent upon definite signals in its operation. Without reliable pilot signals, the whole concept of complete hydraulically-controlled machinery is worthless.

For this reason, this investigator has set out to verify that dependable and reliable pressure signals can be produced. The system chosen for this investigation was a fluid-signal controlled automatic reversing cylinder circuit.

CHAPTER II

PREVIOUS INVESTIGATIONS

Prior to this investigation, only one paper was found which indicated that there has been any work accomplished in utilizing pressure sequencing to obtain pilot signals. This paper, written by A. G. Comer (1), investigated the positioning of a slave cylinder by intermittent hydraulic feeding. This system was controlled entirely by means of pilot signals obtained by pressure sequencing. Comer also studied a pilot-controlled automatic reversing circuit which was similar to the one analyzed in this investigation. In the circuit which he investigated, he found that hydraulic sequence valves were pressure sensitive and only one distinct sequence pressure setting could cause automatic cylinder reversal.

Two English engineers, R. Hadekel (5) and H. G. Conway (2), have used pressure sequencing for obtaining pilot signals. Both authors have used a special valve consisting basically of two sequence valves built into a hydraulically operated, two-position, four-way, directional-control valve to obtain automatic cylinder reversal by means of pressure sequencing. At the present time, this special valve combination is not commercially available in the United States.

Several American engineers have used pressure sequencing operations. J. C. Cotner (3) described five hydraulic machines with utilized pressure sequencing and H. L. Stewart (7) reported on the sequential operation of hydraulic cylinders. R. Kurzweil (6) explained the operation of sequence controls to slide manufactured parts to a conveyor. The operations which these men reported, however, were mainline or power sequencing, rather than pilot sequencing. In mainline sequencing, pressure in one part of the system is allowed to build up to a desired level, then the sequence valve opens, which permits all the flow to enter another portion of the circuit. This type of sequencing requires that each successive sequence valve be set at progressively higher pressures, thus limiting the number of sequential operations by the maximum pressure available. Therefore, it can be seen that many times very high pressures are necessary if the control feature of mainline sequencing is to be utilized. This system also requires large valve sizes, since the entire pump volume must pass through each of the sequence valves.

In addition to mainline sequential control, cam-operated and electric controlled systems have been used to obtain sequential action in hydraulic machinery. Both Cotner and Stewart reported the use of cam-operated valves to obtain cylinder reversal. In this method, a dog on the rod of the cylinder hits a cam-operated, pilot-size, directional-control valve, which reverses the direction of flow through this valve and thereby reverses the cylinder's travel. Stewart also described an electric system of sequencing which utilized limit switches and solenoid-operated, directional-control valves. In this case, the cylinder piston rod strikes a limit switch which allows electric current to energize the solenoid and close the valve. When the valve closes, fluid flow is blocked and the cylinder piston stops.

Based on published information, it can be seen that very little work has been done in utilizing pressure sequencing to obtain pilot signals for control purposes. In the past, the general practice has been to use other systems, therefore it was felt that an investigation on this subject would be a definite contribution to the field of hydraulics.

CHAPTER III

STATEMENT OF PROBLEM

This investigation was carried out under Engineering Research Project No. 152, and its objective was the determination of the effectiveness and dependability of using fluid signals for controlling an automatic reversing circuit. It was decided that this cylinder reversal should be produced by permitting the fluid pressure in the cylinder to build up at the end of the cylinder stroke until the pressure sequence valve opened to send a signal to the directional-control valve, thereby reversing the direction of flow through the valve and causing the cylinder piston to reverse its direction of travel.

Since the principal component in producing pressure signals is a pressure sequence valve, it was believed that a study should be conducted to determine the range of pressure settings required to produce reliable pilot signals. Also, in order to simulate actual operating conditions, it was decided to use a variety of pump speeds and various back pressure levels to load the cylinder.

In order to limit this analysis, only commercially available hydraulic components were used in the system. Since the characteristics of these commercial components were unknown, an experimental investigation was deemed necessary.

CHAPTER IV

ANALYSIS OF CIRCUIT OPERATION

Definition of Components

In order to adequately analyze the problem at hand, the function of of each hydraulic component in the circuit should be understood. Therefore each hydraulic component will be described in terms of its characteristics and functions. Since hydraulic circuits are generally illustrated in graphical form, a graphical representation of each component as proposed by the American Standard Association is shown.

A relief valve serves to establish the maximum pressure in a hydraulic circuit. This valve is adjustable over a wide range of pressures and its primary function in the circuit is to protect the pump from being subjected to excessively high pressures. In operation, the relief valve opens when the system reaches this limiting pressure and allows the excess volume of fluid to by-pass the circuit and return directly to the reservoir. See Fig. 1.

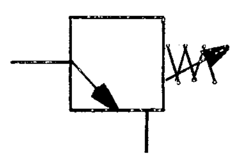


Fig. 1. Relief Valve.

A sequence valve is a pressure control valve which controls the order of flow to various parts of a system. This valve requires that the pressure at the inlet port reach the desired level before the sequence valve will open and permit fluid passage through the valve. As long as the inlet-control pressure remains greater than the spring setting of the valve, full pressure is available at the outlet port. This valve is adjustable over a wide range of pressure settings and its primary function in the system being analyzed was to create pilot signals. See Fig. 2.

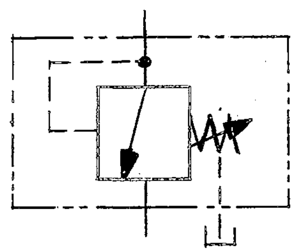


Fig. 2. Sequence Valve.

A counterbalance valve is also a pressure-control valve which permits flow at a desired minimum-pressure level in one direction and free flow in the other direction. Fluid flow in one direction through the valve is blocked until the desired level of back pressure is attained, at which time the valve opens to permit fluid passage. Flow continues through this valve until the back pressure of the system drops below a given minimum value, at which time the valve closes. Flow through the counterbalance valve in the opposite direction is unrestricted. This valve is capable of maintaining a large range of back pressure values. Its primary function in this investigation was to simulate a given load on the cylinder. See Fig. 3.

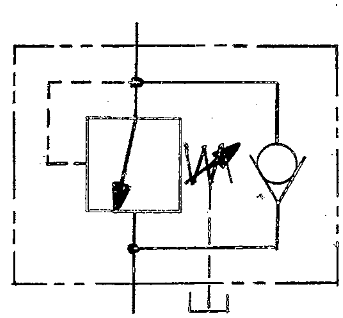


Fig. 3. Counterbalance Valve.

A check valve is used to block fluid flow in one direction while permitting free flow in another direction. This valve consists of a closing device such as a swinging disc, spring-seating disc, or a spring-loaded ball to block the flow. See Fig. 4.

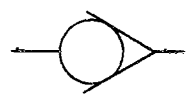


Fig. 4. Check Valve.

A pilot check valve is essentially a check valve which has a provision for reverse flow when desired. This valve always permits free flow in one direction while flow in the opposite direction is blocked until pilot pressure is applied to a control port which unseats the ball and permits free flow in this direction. See Fig. 5.

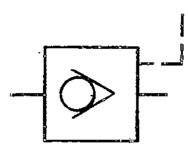


Fig. 5. Pilot Check Valve.

A directional-control valve is used to direct the flow of hydraulic fluid to various parts of the system. In operation, this valve requires some type of motive force to move the spool which in turn changes the direction of flow. The motive force required to shift the spool may be a mechanical force, an electrically derived force, or a hydraulic force. These valves may have several main-line connections. Usually these valves have either two or three positions. When graphically representing this valve, it is shown in the neutral position. When the three-position, four-way valve is in the extreme left position, the pressure port is open to cylinder port A, while cylinder port B is open to the tank port. In the extreme right position, the pressure port is open to cylinder port B and cylinder port A is open to the tank port. In the center position of a three-position, four-way valve, the four ports may be blocked or the cylinder ports may be open to tank while the pressure port is blocked. Although a neutral position exists in a four-way, two-position valve, the spool of the valve is either in the extreme right or extreme left position unless neutralizing forces act on both sides of the spool. Fig. 6 shows the graphical symbol for a pilot-operated, four-way, three-position, spring-centered, directional-control valve. Spring-centering causes the valve to return to the neutral position when there is no pilot pressure on either side of the sliding spool. Fig. 7(a) represents a manually-operated, four-way, three-position, directional-control valve with the four ports blocked in the center position and Fig. 7(b) represents the same valve with the pressure port blocked and the cylinder ports open to tank in the center position. Fig. 8 shows the graphical symbol for a pilot-operated, four-way, two-position, directional-control valve.

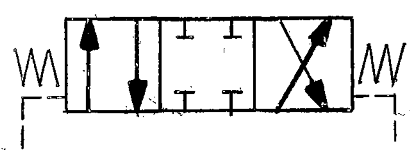
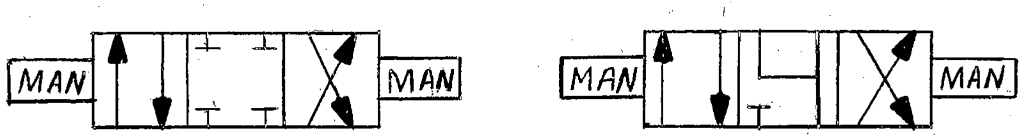


Fig. 6. Directional-control Valve.



(a) (b)

Fig. 7. Directional-control Valve.

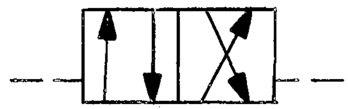


Fig. 8. Directional-control Valve.

A pump unit, driven by an electric motor through a variable-speed reducer, is graphically represented in Fig. 9. The pump is a fixed-displacement gear type pump and becomes a variable volume pump by changing the speed of the variable-speed reducer.

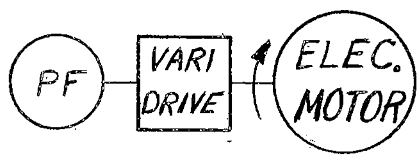


Fig. 9. Pump Unit.

A fixed-displacement pump which is directly coupled to an electric motor is illustrated in Fig. 10.

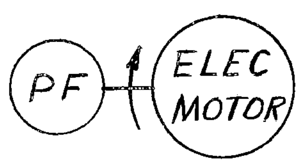


Fig. 10. Pump Unit.

A heat exchanger is used to dissipate the heat generated in the oil-powered circuit. Oil is circulated through the heat exchanger to maintain the oil temperature at some constant level so that the properties of the oil can be maintained within certain limits. In this investigation, a two-pass, shell-and-tube type condenser was used with water passing through the tubes and oil going to the shell side. Fig. 11 shows the graphical symbol for a heat exchanger.

Fig. 11. Heat Exchanger.

A gate valve is an on- or off-type of shutoff valve. This valve is manually operated and it is graphically represented in Fig. 12.

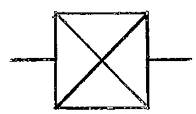


Fig. 12. Gate Valve.

A strainer consists of a small-sized screen which collects any foreign particles which may be present in the hydraulic fluid. Strainers are usually located at the reservoir in both the return and suction lines. The graphical symbol for a strainer is given in Fig. 13.

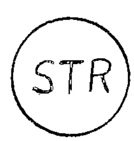


Fig. 13. Strainer.

The graphical symbol for an oil reservoir which is vented to the atmosphere is illustrated in Fig. 14 and a pressure gage is shown in Fig. 15. A double-acting, double-rod, cushioned cylinder is shown in Fig. 16.

Fig. 14. Oil Reservoir.

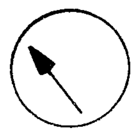


Fig. 15. Pressure Gage.

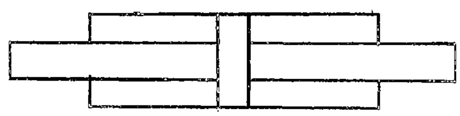


Fig. 16. Cylinder.

The Integrated Circuit

In order to more easily understand the complete hydraulic system, the system will be divided into two major circuits. The pump unit and its related equipment comprise one circuit while the cylinder unit and its related components and lines comprise the other circuit.

The complete fluid power system is graphically represented in Fig. 17.

Pump Circuit

The pump circuit itself was composed of two parts, these being:

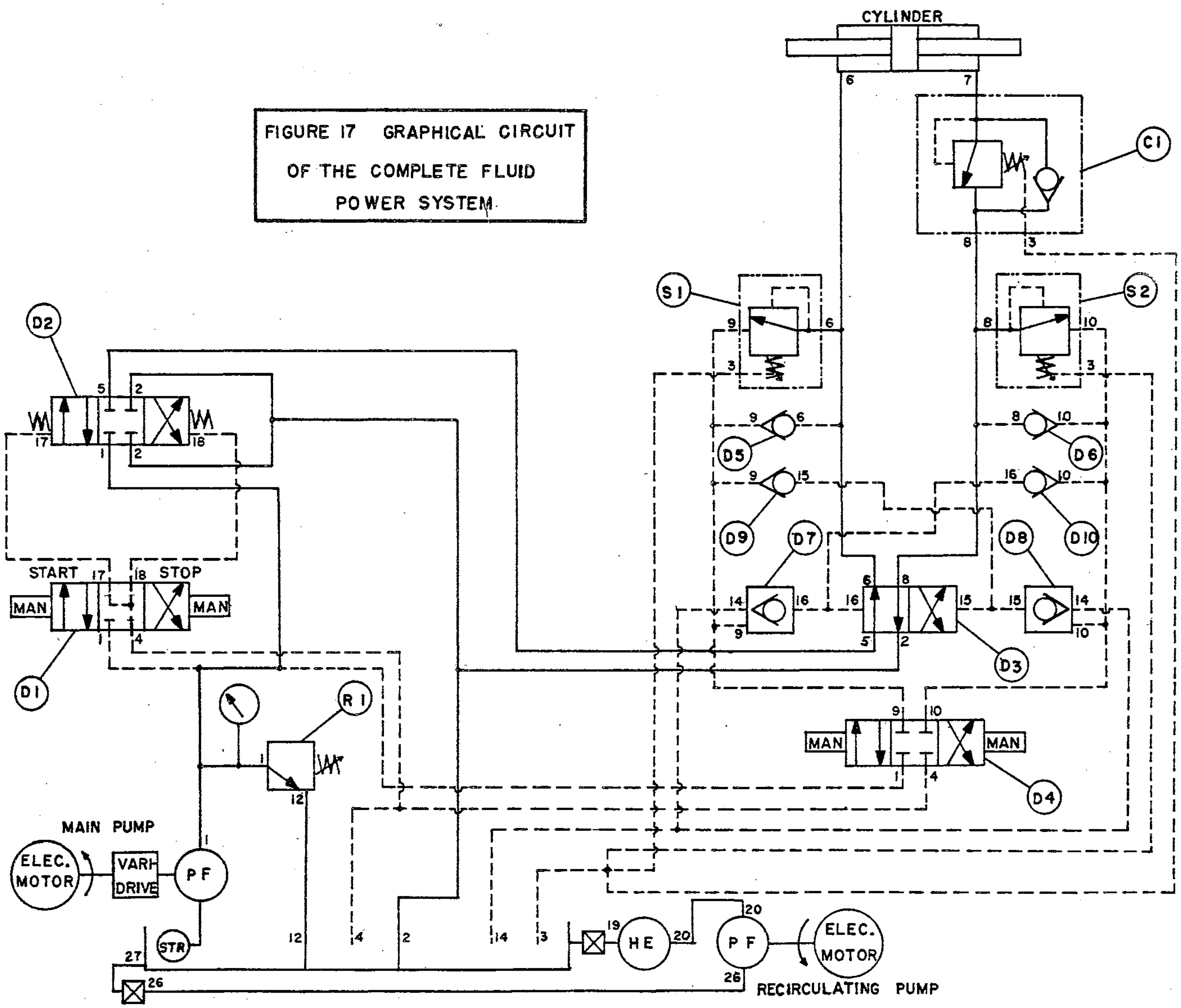


FIGURE 17 GRAPHICAL CIRCUIT OF THE COMPLETE FLUID POWER SYSTEM

(1) the main pump unit, and (2) the recirculating unit. In order to keep the oil in the system relatively clean, the fluid was passed through one of three strainers before entering the pump. Clean oil is a necessity since contaminants are extremely harmful to hydraulic valves and pumps. After the fluid passed through the strainer, it flowed through the gear pump and into the system. The pump was driven by means of a vari-drive unit which made possible a wide range of flow rates by varying the speed of the pump. A relief valve on the downstream side of the pump protected the pump from being subjected to excessive pressures. A gage in the discharge line was installed to measure the pressure of the system at this station.

The recirculating unit was used for the purpose of maintaining the temperature of the hydraulic fluid at a constant value. Fluid was taken from the reservoir and circulated through a two-pass heat exchanger and then returned to the reservoir.

Cylinder Circuit

Fluid flow into the cylinder circuit was blocked by directional-control valve D-2 (Fig. 17) until the manually-operated directional-control valve D-1 was shifted to the extreme left position, which in turn applied pilot pressure to the left side of valve D-2. Fluid then entered the directional-control valve D-3 and could flow into either the left or right line into the cylinder depending on the spool position of valve D-3. If flow entered the left line, the right line would be open to tank and fluid under pressure would flow into the left side of the cylinder. The fluid displaced on the right side of the cylinder would pass in the restricted direction through the counterbalance valve C-1, through the directional-control valve D-3, and then return to tank. If flow entered the right line, the left line would be open to tank and fluid under pressure would flow in the unrestricted direction through the counterbalance valve C-1 and into the right side of the cylinder, while the fluid displaced on the left side of the cylinder would pass through the valve D-3 and go to tank.

If the directional-control valve D-3 were in the extreme left position, as shown in Fig. 17, flow continued into the left side of the cylinder until the cylinder rod reached the limit of its travel to the right, at which position the fluid pressure increased, causing the sequence valve S-1 to open. This permitted signal fluid to travel through the pilot line to energize the pilot check valve D-7 which opened it, thus allowing the pilot signal from the left side of the directional-control valve D-3 to return to tank. The pilot signal continued until it passed through check valve D-9 and then to the right side of directional-control valve D-3. When pilot pressure built up on the right side of the spool, the spool shifted. Fluid now flowed into the right line and fluid in the left line went to tank. With the left line opened to tank, the pilot pressure locked in the pilot line above the check valve D-9 then went to tank through check valve D-5. Flow into the right side of the cylinder continued until the cylinder rod reached the limit of its travel to the left, at which time the increased fluid pressure caused the sequence valve S-2 to open and the same cycle of events took place using a similar series of valves--D-8, D-10, D-3, and D-6.

This automatic cycling of the cylinder continued until the spool in the directional-control valve D-2 was shifted to the extreme right position. This was accomplished by manually shifting the directional valve

D-1 to the extreme right position which caused the pilot signal to energize the right side of valve D-2. This shifted the spool into the extreme right position and at the same time tanked the pilot signal on the left side of the directional-control valve D-2. As soon as the spool in directional valve D-2 was shifted to the right, the flow of fluid was diverted from the cylinder circuit directly to the reservoir, thus stopping the automatic cycling action.

To position the cylinder rod manually into the extreme right position, it was necessary to position the manually-operated directional-control valve D-4 into the extreme right position. This directed the pilot signal to the left side of directional-control valve D-3 and tanked the right side of valve D-3. The extreme left position of the cylinder rod could be obtained manually by shifting the directional-control valve D-4 into the extreme right position which directed the pilot signal to the right side of valve D-3.

Revised Circuit

The hydraulic system under investigation should operate without the presence of check valves D-5, D-6, D-9, and D-10, in the circuit. However, it was found that when these valves were absent from the circuit, it was impossible to obtain reliable pilot signals because the pilot check valves were required to tank a larger volume of fluid in a very short period of time. By placing check valves D-9 and D-10 in the circuit, pilot check valves D-7 and D-8 were not required to tank the fluid in the pilot line beyond check valves D-9 and D-10. Check valve D-5 now tanked the fluid in the pilot line between sequence valve S-1 and check valve D-9 while check valve D-6 tanked the fluid in the pilot line between sequence valve S-2 and check valve D-10. Check valves D-9 and D-10 also helped to maintain pressure on the directional-control valve in case there was any fluid leakage past the control port of pilot check valves D-7 and D-8.

Line Size Selection

Since high fluid velocities cause excessive pressure losses in valves and connecting lines, the velocity consideration was the determining factor in the selection of lines and components. If the pump were operating at a capacity of 22 gallons per minute (gpm), the corresponding fluid velocity would be 23.0 feet per second (fps) in a 3/4-in. hose and 11.8 fps in a 1-in. hose. This lower velocity would maintain the fluid flow in the laminar region, so 1-in. directional-control valves, counterbalance valves, and main pressure lines were selected for this investigation.

Since the sequence valves, pilot check valves, and check valves would be used only for pilot signals, it was decided that 1/4-in. valves would be sufficiently large to pass the required flow. It was also decided that all branch lines and pilot return lines would be 3/8-in. to eliminate the possibility of excessive back pressure. The directional-control valves used strictly for control of pilot signals were 3/8-in. size.

Test Stand

In order to conserve floor space and at the same time provide a large area for the mounting of components, a vertical test stand was constructed. The test stand consisted of a steel-pipe frame which supported a 6 X 8 ft. steel grating. The stand was mounted on casters for maneuverability and was equipped with brakes which held the stand rigid when testing was being performed. Integrated into this test stand was a drip-pan which extended the entire length of the stand and had a width of 2 ft. Its purpose was to catch any oil which might drip when circuit connections were broken.

Three pipe manifolds were incorporated on the vertical test stand which served to distribute and return fluid to the hydraulic circuit through a number of openings in each manifold. The upper manifold consisted of $1\frac{1}{4}$-in. high-pressure pipe which directed fluid to the circuit. The center manifold was a $1\frac{1}{2}$-in. low-pressure line to which were connected drain and pilot return lines for the purpose of returning the fluid in these lines to the reservoir. The lower manifold was a 2-in. low-pressure line which returned main line fluid to the reservoir. See Fig. 18.

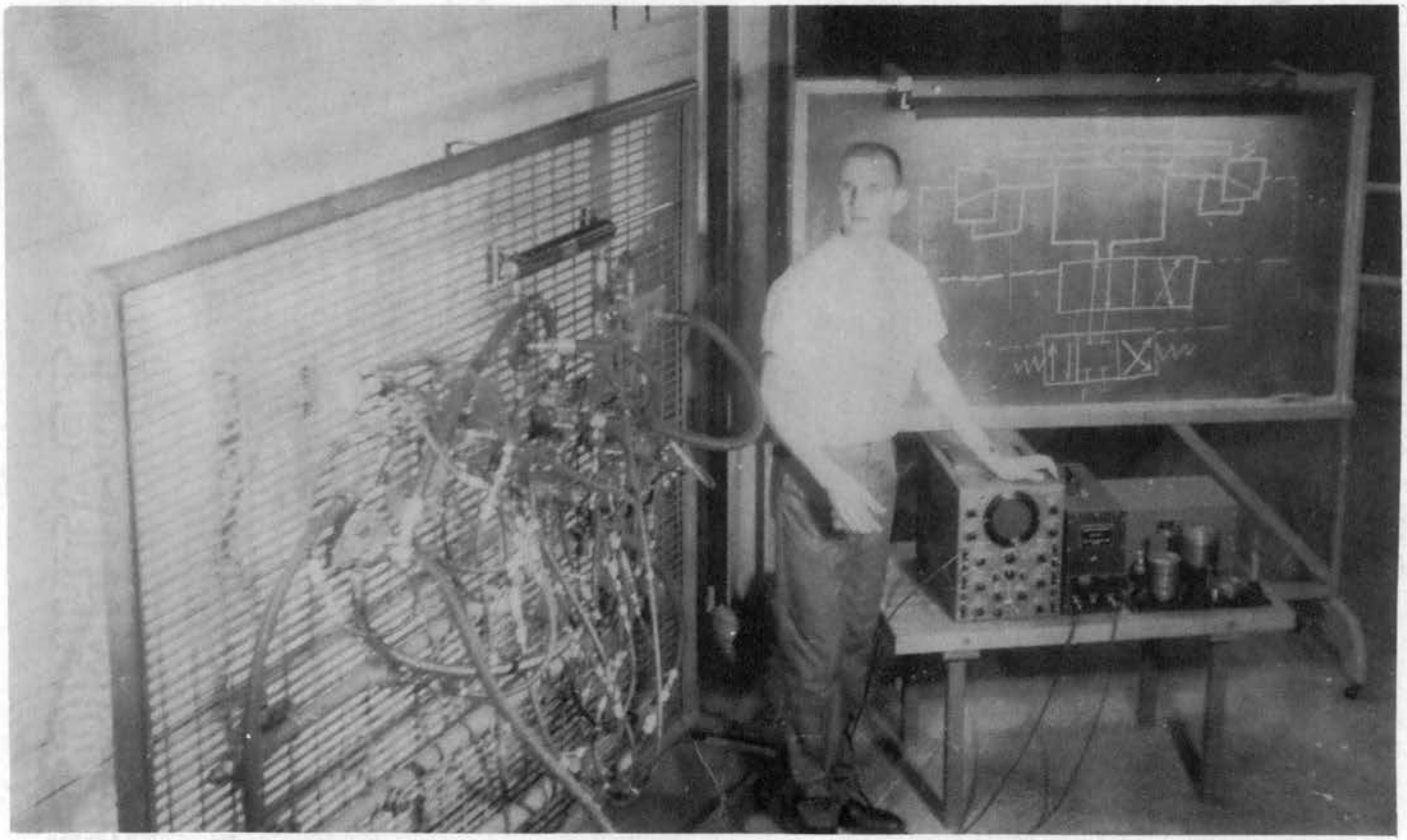


Fig. 18 Physical Arrangement of the Testing Apparatus.

CHAPTER V

TESTING PROCEDURES

In making test runs, the three variables considered were cycle rate, loading conditions, and sequence pressure conditions. The cycle rate was varied by the pump speed which governed the volume of fluid flow. Loading conditions were obtained by causing the cylinder to operate against a back pressure established by the setting of the counterbalance valve, C-1. Because only one counterbalance valve was used, it was possible to load only one side of the cylinder. Maximum pressure conditions were regulated by the relief-valve setting and the sequence valve setting controlled the sequence pressure level.

Before beginning a test run, the pressure pickup was connected to the pressure gage port of sequence valve S-2. The system was then checked for disconnected lines and loose connections. Next, the directional-control valve D-1 was checked to be certain that it was in the extreme right position so that the pump would not be under a load condition when it was started. The valve was opened which allowed cooling water to enter the heat exchanger and then the recirculating pump was started. With both of these units operating, the main pump was started. The directional-control valve D-1 was then manually shifted into the middle position which opened both control ports of D-1 to tank. This caused the directional-control valve D-2 to shift into the blocked center position. Valve D-2 was held in this position for a short interval of time while the setting of the relief valve R-1 was being checked.

Directional-control valve D-1 was shifted into the extreme left position which caused the pilot pressure to shift the valve D-2 into the left position. This allowed fluid to enter the cylinder through the directional-control valves D-2 and D-3. After this action took place, operations in the circuit became automatic if sequence valves S-1 and S-2 were set properly. The circuit continued to operate automatically until the valve D-1 was shifted into the extreme right position which caused directional-control valve D-2 to shift into the extreme right position.

If automatic action stopped while the sequence valves were being adjusted, the valve was reset and directional-control valve D-4 was shifted into either the extreme right or extreme left position to start automatic action again. As the cylinder piston started moving, valve D-4 was returned to the neutral position.

After the counterbalance valve and sequence valves were adjusted to the desired pressure levels, a series of runs was made at different pump speeds. The number of complete cycles per minute was recorded and the image magnitudes produced on the oscilloscope screen by the pressure pickup were measured during each run.

When each series of runs was completed, the pump was stopped and the pressure pickup was removed from the sequence valve and calibrated to determine the actual sequence pressures and back pressures. A dead weight tester was used as the standard in calibrating the pressure pickup. The pressure pickup was again installed and the pump was started. New load conditions were imposed upon the cylinder and the test was repeated. These tests were run at pump speeds between 300 and 1000 rpm.

When all of the data on cylinder operation was complete, the pump was calibrated to determine the flow rates occurring at various pump speeds. The flow rate was determined at pressures of 0, 300, 600, 900, and 1200 psi at pump speeds between 300 and 1500 rpm.

CHAPTER VI

DISCUSSION OF RESULTS

This investigator found that it was possible to create reliable pilot signals by pressure sequencing. Even though it was necessary to modify the original circuit in order to obtain consistent fluid signals, these modifications were minor and no specially designed components were added to the system.

The pilot check valves which were obtained for this investigation were not specifically designed for pilot signal operation so it was necessary to modify them. The spring which seated the ball check was too light to provide fast action and a positive seal for the pilot fluid. Therefore a much stronger spring was added to the valve because leakage and slow response time could easily cause the signal to become lost by letting it go to tank. The end result of this modification was that a higher pressure would be necessary to unseat the ball check. The pilot check valves were manufactured with a small-sized inlet to the pilot piston also. The small hole served to restrict fluid flow into and out of the piston cylinder when pilot pressure was applied. A larger hole of approximately 1/4-in. diameter was drilled in place of the smaller hole so that the response time for seating and unseating the ball check was reduced.

Before attempting to determine the operating characteristics of the circuit investigated in this study, a reversing circuit was constructed which utilized a single-rod non-cushioned cylinder, and a circuit similar to that of Fig. 17 without check valves D-5, D-6, D-9, and D-10. In this circuit, another counterbalance valve was installed at the left inlet port of the cylinder. Here it was found that the cylinder would operate automatically if the sequence valve setting was at one distinct pressure level which was above the cylinder operating pressure but below the relief valve setting.

This circuit was then modified by installing a double-rod, cushioned cylinder and removing the counterbalance valves from both cylinder inlet ports and placing one counterbalance valve at the tank port of directional-control valve D-3. It was impossible to make this circuit operate properly. Automatic reversing was obtained on each end of the cylinder but it was never possible to adjust the two sequence valve settings to obtain automatic reversal on both ends of the cylinder at the same time.

The position of the counterbalance valve at the tank port of the cylinder now affected the operation of the directional valve D-3. Whenever the cylinder failed to reverse automatically, the valve D-3 would hesitate for an instant, then gradually the spool would shift into the desired position and reverse the direction of the cylinder piston's travel. This shifting of the spool was definitely produced by some element other than the pressure signal. Fluid was leaking past the spool into the pilot pressure caps and causing the spool to shift its position.

The counterbalance valve was removed from the tank port of the directional-control valve D-3 and placed at the right inlet port of the double-rod, cushioned cylinder. This circuit was the originally-test circuit with the exception that a cushioned, double-rod cylinder and only one counterbalance valve were used. Since it was impossible to obtain automatic cylinder reversal, the pilot return line from the pilot check valve was disconnected to determine the amount of fluid which composed the fluid signal. Upon observation, it was found that the actual volume of the fluid signal was much less than the anticipated amount. It was believed that this condition was responsible for the circuit's failure to operate automatically.

Since the volume of fluid in a pressure signal was quite small, it was necessary that a sufficient fluid surge accompany the signal in order to shift the directional-control valve into the required position. The action of the cushioned cylinder could not provide this necessary surge which the non-cushioned cylinder was capable of providing. When this condition existed, two factors might be responsible for the system's failure to operate. The first factor was the loss of the pressure signal as the spool of the directional-control valve D-3 reached the center position. This would cause the spool to stop, thereby blocking fluid flow into the cylinder and stopping the automatic cylinder reversal. The other factor influencing this operation was the failure of the tank port of the pilot check valves D-7 and D-8 to be closed when pilot pressure was sent to the control ports of directional-control valve D-2.

To overcome this difficulty, check valves D-5, D-6, D-9, and D-10 were added to the system as shown in Fig. 17. Check valves D-9 and D-10 prevented the pressure signal from being lost while check valves D-5 and D-6 made it possible to insure that the tank port of the pilot check valves was closed in advance of the time when the signal was sent into the pilot pressure line. This modification made certain that fluid pressure would be held upon the pilot control port of the directional-control valve until it was tanked by the pilot check valve. It was also possible to tank the undesired signal in two stages rather than in a signal operation. This meant that the pilot check valve had less fluid to tank and therefore could perform this operation in the required time.

It is also important to note that when the four check valves along with the two pilot check valves were used to tank the pilot pressure signal, it was necessary to have one of the main pressure lines feeding the cylinder open to tank. This was necessary if check valves D-5 and D-6 were to perform their tanking operations successfully. Also this situation required that the counterbalance valve C-1 be placed at the cylinder rather than at the tank port of the directional-control valve D-3. Locating the counterbalance valve at the cylinder caused back pressure on only one side of the cylinder, while locating the counterbalance valve at the tank port of the directional-control valve would cause a back pressure on both sides of the cylinder. In practice, cylinders are generally loaded only in one direction of travel, so this loading would simulate many actual applications of fluid power systems.

The cylinder used in this system was a double-rod cylinder so that an uniform piston velocity occurred in both directions of travel. This cylinder was cushioned on both ends, thus greatly decreasing the impact which normally occurs in non-cushioned cylinders. A 14-in. stroke and a 2-in. bore were used so that an analysis of high-speed reversing circuits could be achieved.

There seemed to be no adverse effects in the cylinder operation due to the entrainment of air in the system. Because of the height of cylinder with respect to other components in the system, air collected in the cylinder. Since the larger lines capable of tanking this air were connected to the cylinder, it was possible to remove the accumulated air by cycling the cylinder piston a few times. This system was so effective in removing air that a vacuum was observed when the pressure pickup was removed from the system. This vacuum was caused by the deaeration effect produced by fluid flow in the return line when tanking operations were taking place.

An objectionable feature of this system was the extremely noisy condition existing in the directional-control valve when the spool was shifted. This noise was due to the impact of the spool with the end caps. These impacts were of such magnitude that the end caps showed indentations where they had been hit by the spool. This condition could be eliminated if some type of adjustment could be made to stop the travel of the spool before reaching the end caps of this valve.

The sequence valves which were used in this circuit produced dependable pressure signals when they were set above a pressure level which was somewhat higher than the operating pressure. The cylinder would cycle automatically as long as these conditions existed. If the sequence pressure was set below this definite pressure level, the sequence valve opened prematurely before the cylinder piston reached the end of its stroke. This minimum sequence pressure level increased with increasing flow rates. It was also observed that if the flow rate remained constant while the operating pressure increased by changing the counterbalance valve setting, the minimum sequence pressure increased. The upper limit on the sequence pressure level was the setting of relief valve R-1. Between these lower and upper limits on the sequence pressure, there existed a range of pressures, rather than a single pressure level, which would create reliable pressure signals.

Runs were made with a variety of loads on the cylinder. These loads were: (1) a no-load condition, (2) a back pressure of 300 psi corresponding to a force of 800 pounds on the cylinder, and (3) a back pressure of 475 psi or a force of 1250 pounds.

At various flow rates, the cycles per minute of the cylinder piston were determined from the observed data in Table I. Table II gives the cycles per minute and the corresponding flow rates used to plot the performance curves for this investigation. In a complete cycle, the distance travelled was equal to twice the cylinder stroke. In measuring the cycle rate, it was necessary to include the time for two cylinder reversals as well as the time required to travel twice the length of the cylinder. This cycle rate was determined by counting the number of cycles taking place in a definite period of time. A timer was used to record the time while the cycles were counted manually. A Chrono-Tachometer was used to obtain the pump speed at the same time the cycle rate was being determined.

The back pressure and sequence pressure were measured by means of a pressure pickup and an oscilloscope. The pressure pickup employed a disc which deflected when pressure was applied to it. A capacitive type element sensed these deflections and transformed them into signals which were received by a pressuregraph. The pressuregraph converted these signals into voltages which were placed across the vertical plates of a cathode-ray oscilloscope. By adjusting the vertical gain control of the oscilloscope and the output control of the pressuregraph, the height of the band formed on the oscilloscope screen indicated the magnitude of the pressures exerted upon the pressure pickup. The actual pressure corresponding to a certain band height was determined by calibration

TABLE I

OBSERVED DATA

March 31, 1958

Run No.	Back Pressure psi	Sequence Pressure psi	No. of Cycles	Time, min	No. of Pump Rev.
1	0	600	25	1.100	417
2	0	800	50	1.299	722
3	0	900	50	1.170	817
4	0	1000	70	1.348	1148
5	0	1150	70	1.165	1169
6	0	925	25	1.078	425
7	0	1000	40	1.210	667
8	0	1200	50	1.202	835
9	0	1300	70	1.434	1208
10	0	1400	70	1.329	1329
11	0	550	50	2.467	771
12	0	750	50	1.545	769
13	0	500	50	1.085	756
14	0	600	100	1.696	1518
15	0	650	100	1.414	1559
16	0	1175	25	1.434	418
17	0	950	50	1.700	823
18	0	1150	50	1.213	840
19	0	1300	50	1.062	954
20	0	1350	60	1.173	1287
21	300	550	25	1.067	417
22	300	750	40	1.184	662
23	300	850	50	1.177	823
24	300	950	70	1.362	1160
25	300	1100	80	1.357	1364
26	300	750	22	0.965	390
27	300	800	40	1.252	687
28	300	950	50	1.212	851
29	300	1100	70	1.433	1214
30	300	1300	70	1.301	1309
31	475	800	40	1.255	678
32	475	950	50	1.222	840
33	475	1100	70	1.432	1202
34	475	1300	70	1.321	1320

TABLE II

PERFORMANCE DATA

Run No.	Pump Speed, rpm	Flow Rate, gpm	Cycle Rate, cpm
1	379	8.1	22.7
2	555	11.9	33.4
3	698	15.0	42.7
4	852	18.3	51.9
5	1003	21.5	60.1
6	394	8.4	23.2
7	551	11.8	33.1
8	695	14.9	41.6
9	842	18.1	48.8
10	1000	21.5	52.7
11	313	6.7	20.3
12	498	10.7	32.4
13	697	15.0	46.1
14	895	19.2	59.0
15	1103	23.6	70.7
16	292	6.2	17.4
17	484	10.4	29.4
18	693	14.8	41.2
19	898	19.3	47.1
20	1097	23.6	51.2
21	391	8.0	23.4
22	559	11.6	33.8
23	699	14.6	42.5
24	852	17.9	51.4
25	1005	21.2	59.0
26	404	8.3	22.8
27	549	11.4	31.9
28	702	14.7	41.3
29	847	17.8	48.8
30	1006	21.2	53.8
31	540	11.0	31.9
32	687	14.2	40.9
33	839	17.4	48.9
34	999	20.8	53.0

using a dead weight tester. Since the pressure image appearing on the screen of the oscilloscope was of very short duration, the error in each reading was in the range of 50 psi.

The pressure pickup made it possible to obtain a clear picture of the pressure occurring in the cylinder. Fig. 19 is a sketch of a typical pressure-time diagram for conditions existing in the cylinder. An extremely high pressure surge occurred in the cylinder when fluid was directed into the inlet port. The pressure surge was greater than the sequence pressure level and was caused by the necessity of the fluid to accelerate the mass of the piston and piston rod to the velocity determined by the flow rate. The cylinder pressure then dropped abruptly to the working pressure necessary to push the piston to the end of the cylinder. As the cylinder piston approached the end of the cylinder, pressure gradually built up to the sequence pressure level due to the cushioning effect. The pressure sequence valve opened to send a pilot signal to the directional-control valve which reversed the direction of piston travel. Pressure then dropped abruptly to the back pressure level determined by the counterbalance valve setting and the cylinder piston returned to the opposite end of its stroke.

In making test runs to determine the characteristics of the system, the sequence valves were adjusted to a particular setting which would operate through the entire range of flow rates. The first setting was at a level which required the minimum sequence pressure at the lower flow rates. The next setting was at some higher level which was the maximum sequence pressure at the higher flow rates. Fig. 20 and Fig. 21 indicate the cycle rate versus flow rate for these two settings. A no-load condition was used in Fig. 20 while a 800-pound load was used in

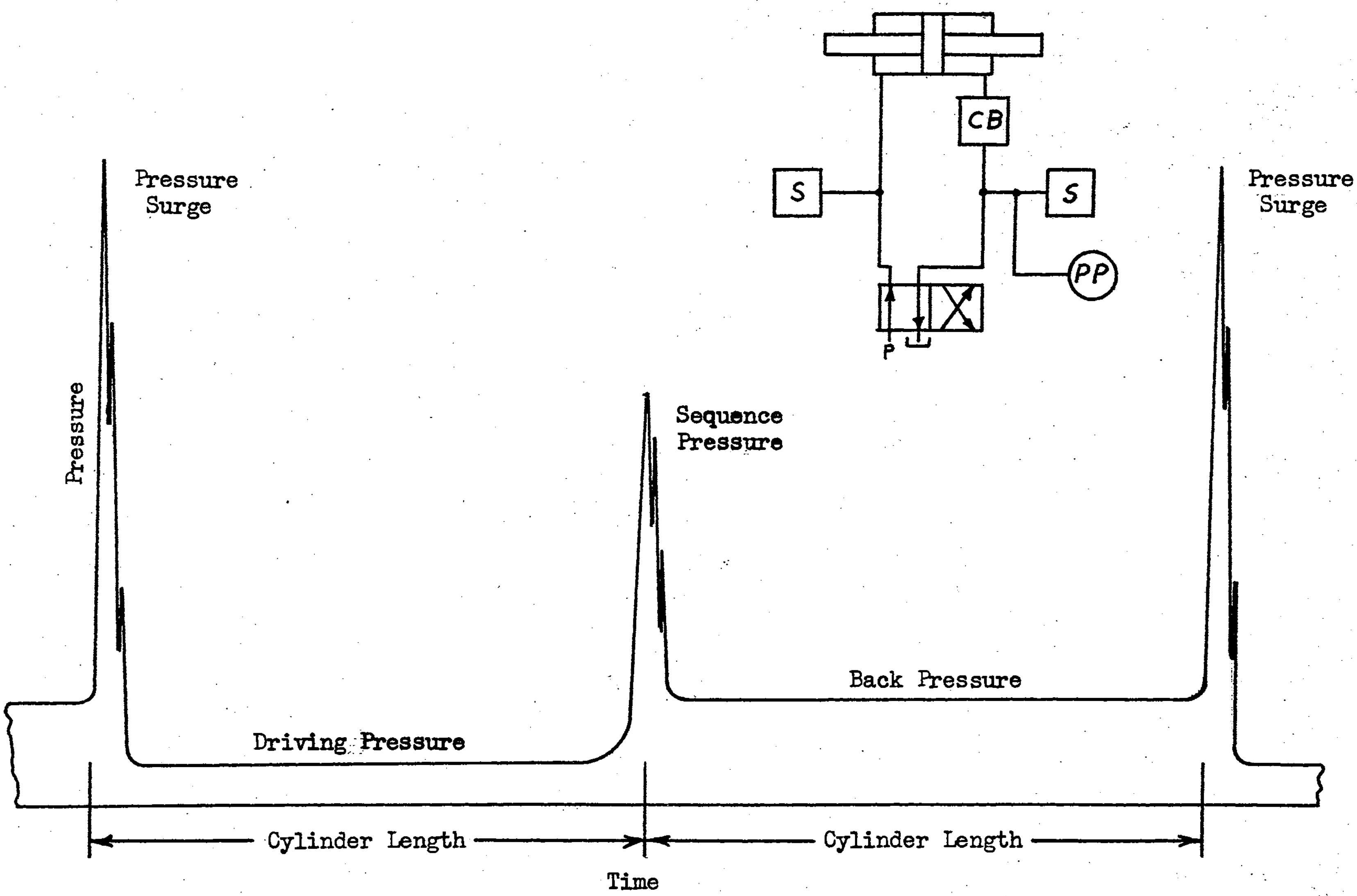


Fig. 19. Pressure-Time Diagram

Fig. 21. To illustrate more clearly the effects of load conditions, a series of runs was made in which the only variable was the back pressure. This information is given in Fig. 22.

A series of runs was made to determine the cycle rate versus flow rate when the sequence valves were adjusted at each flow rate to obtain the maximum and minimum sequence pressure levels. This information was also plotted on Fig. 20. It was found that at the minimum sequence pressure level, the performance curve was exactly the same as the theoretical cycle rate which assumed instantaneous reversal of the cylinder. Physically this is impossible because a small time interval of 0.017 seconds is required for each reversal of the cylinder. It therefore must be concluded that the piston was not travelling the entire length of the cylinder and therefore the theoretical volume of fluid was not entering the cylinder. The curves in Fig. 22 indicate that the effects of load conditions upon automatic cylinder reversal are negligible. Load conditions could be varied, and as long as the sequence valve setting was unchanged, the loading conditions had no appreciable effect. In this investigation, however, the sequence valve setting did have a pronounced effect on the system. For a given flow rate, the cycle rate decreased as the sequence pressure was increased. This condition can be partly attributed to the fact that higher sequence settings require a rapid pressure build up at the end of the cylinder stroke, which introduces pressure surges of sufficient magnitude to open the pressure relief valve. This relief valve opening causes a portion of the fluid to bypass the circuit and return to the reservoir. In this investigation it was impossible to eliminate this condition because of the maximum pressure rating of the pump. Pressure surges of over 2000 psi were

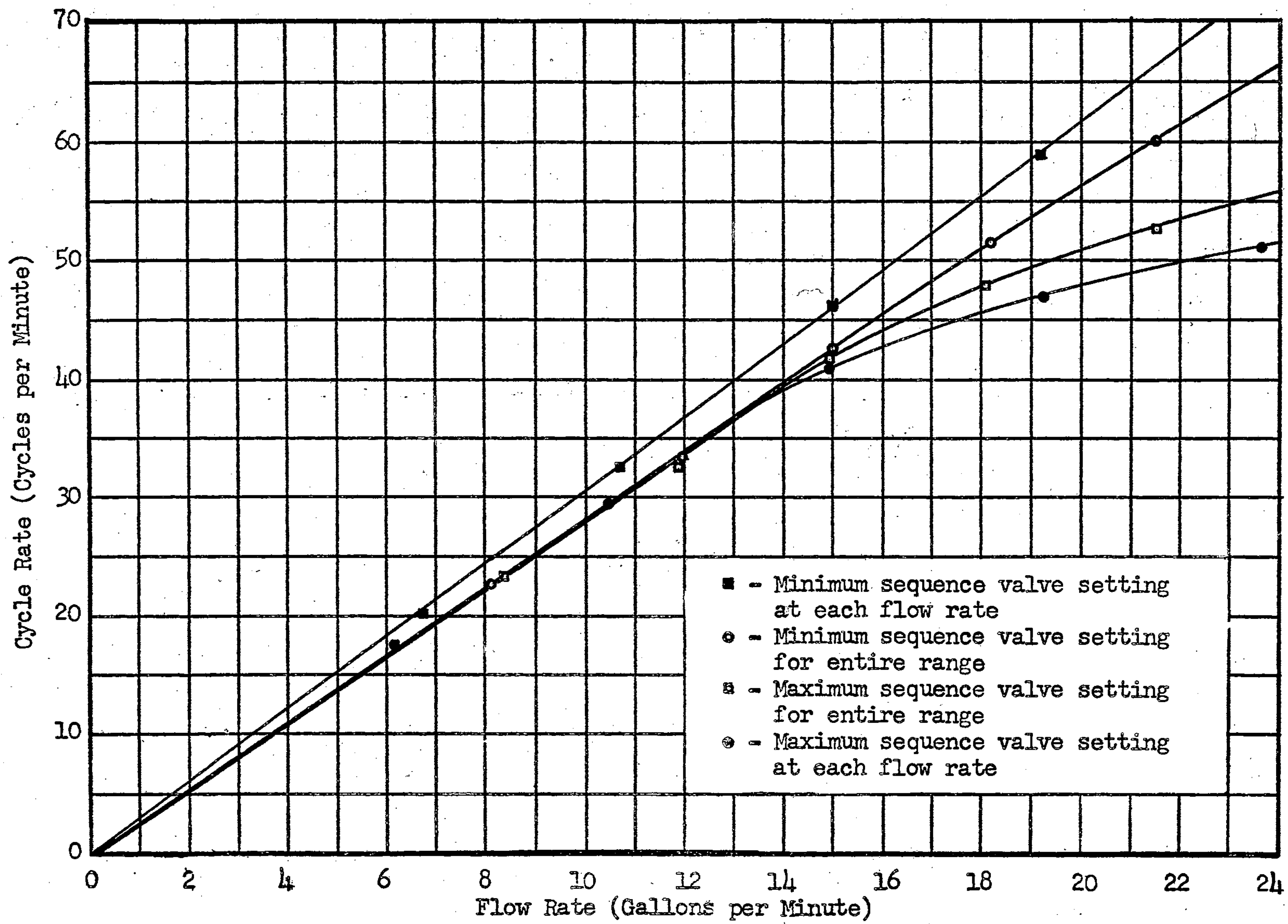


Fig. 20. Cycle Rate at No-load Conditions.

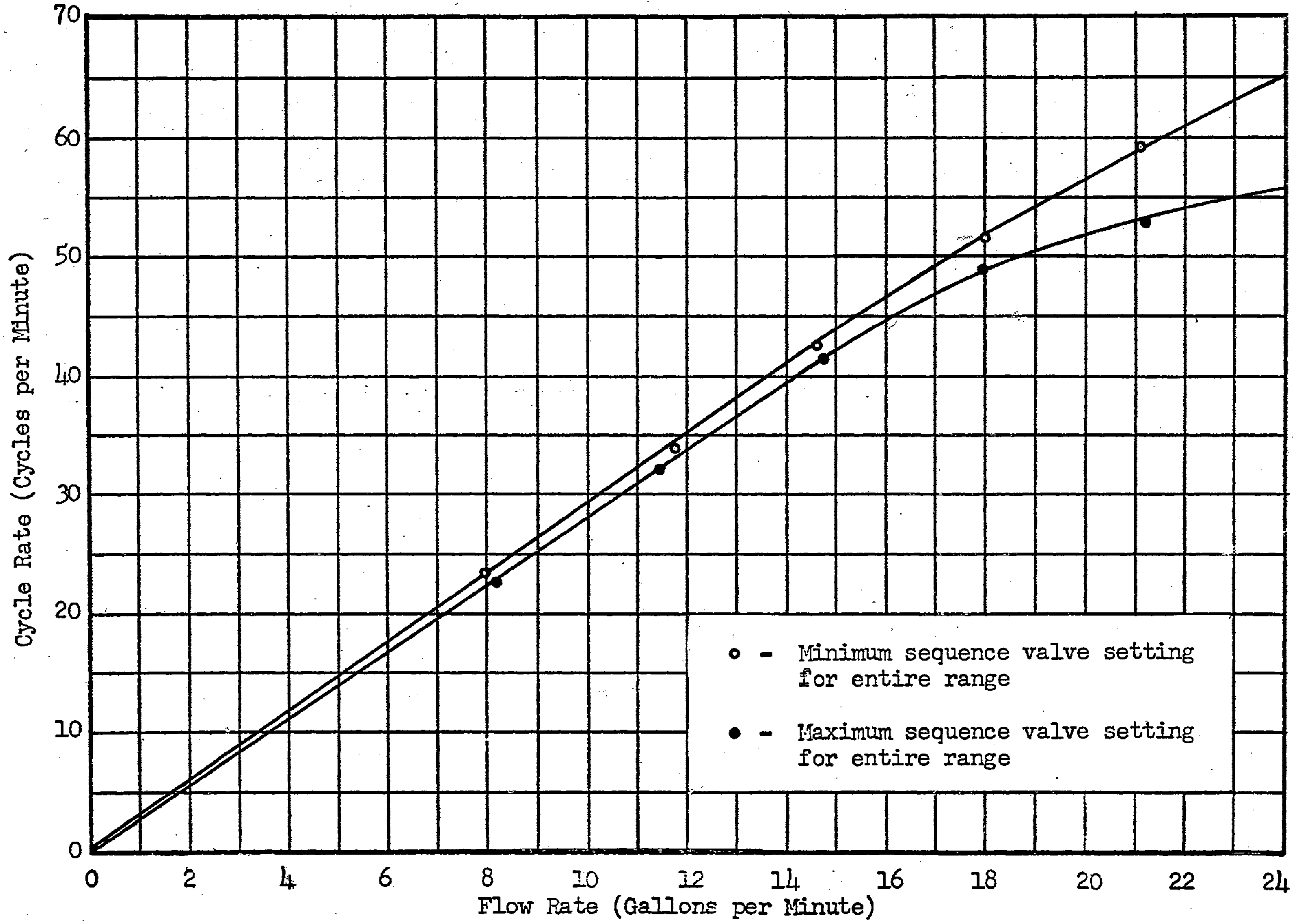


Fig. 21. Cycle Rate with 800 lb. Load on Cylinder.

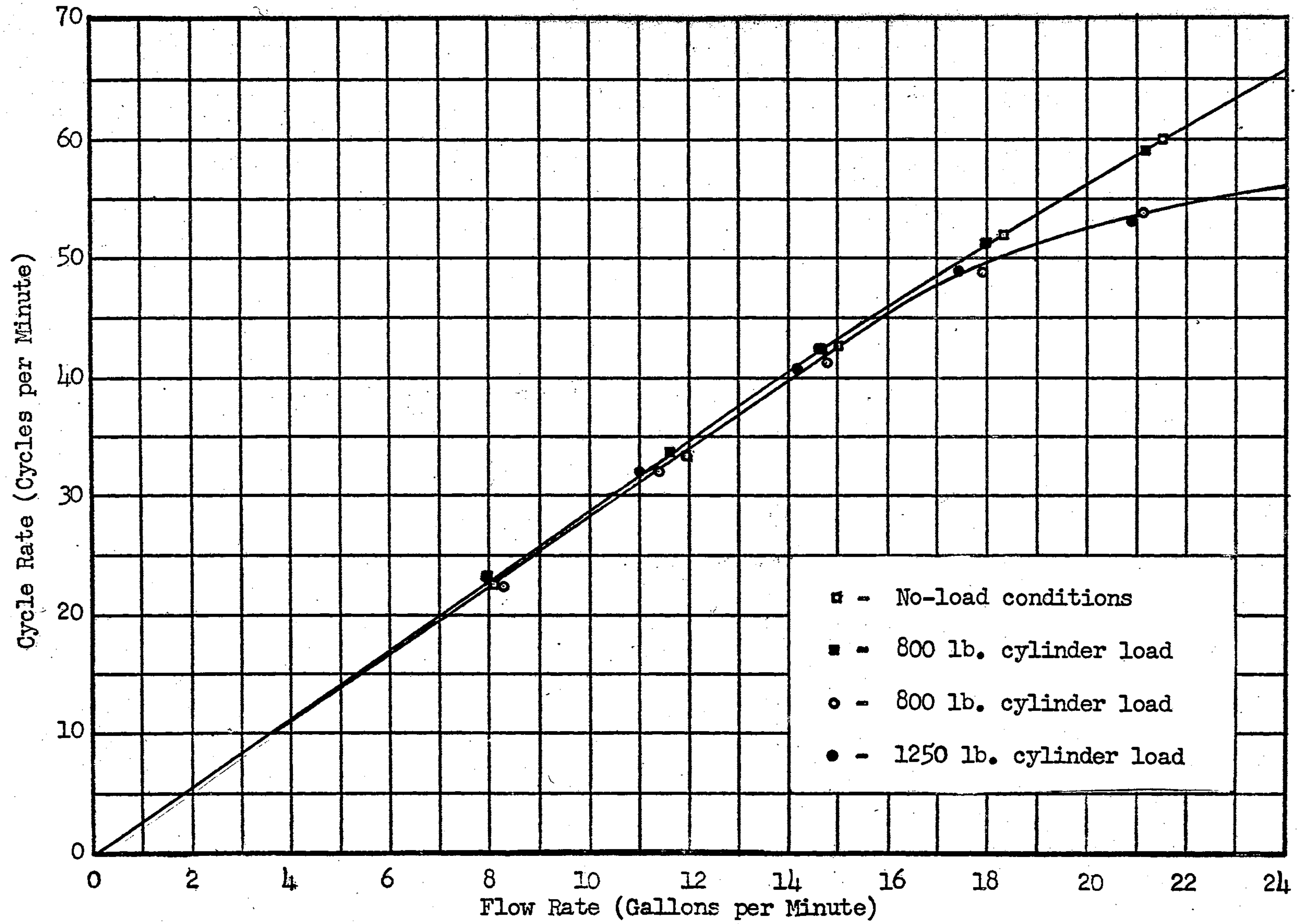


Fig. 22. Cycle Rate at Different Loading Conditions with Fixed Sequence Valve Settings.

encountered while the relief valve setting was limited to 1500 psi.

Sequence pressure also had an effect upon the cycle rate since the hydraulic fluid is slightly compressible. When a flow rate of 15 gpm and a sequence pressure of 1200 psi were encountered, the actual time for one cycle was 1.429 seconds and the theoretical cycle time was 1.308 seconds. This leaves 0.121 seconds for which an accounting must be made. From the equation developed by Dow and Fink (4), the pressure build up time for sequencing operations on each end of the cylinder would be 0.040 seconds. The flow time necessary to shift the direction control valve at the above conditions would be 0.017 seconds for each reversal operation. Since there were two reversal operations per cycle, the total accountable time would be 0.114 seconds per cycle or approximately 0.007 seconds per cycle unaccountable. This very accurate correlation between the actual and theoretical values of cycle time indicated that the relief valve was not opening when the above conditions existed.

The oil temperature was maintained between 75° and 85°F for all flow rates. These temperatures were fairly constant which helped to maintain the physical properties of the oil at a constant level.

CHAPTER VII

SUMMARY

The objective of this investigation was to determine the effectiveness and dependability of fluid signals for controlling automatic reversing circuits. Automatic cycling of a cylinder was produced by pressure sequencing at the end of the piston stroke, which created a signal that was sent to a directional-control valve. This signal caused the direction of flow through the valve to be reversed, thereby causing the cylinder piston to reverse its direction of travel.

This circuit indicated that effective and dependable pressure signals could be produced which would cause automatic reversal of a hydraulic cylinder. A wide range of sequence pressure valve settings rather than one definite sequence pressure level can produce reliable pilot signals.

It was found that loading conditions on the cylinder had negligible effect upon the cycle rate of the ability of the pressure sequence valve to cause reversal of the cylinder. The sequence valve setting, however, affected both the cycle rate and the ability of the sequence valves to produce reliable signals. If pressure signals are to be produced, the sequence valve pressure setting must be above a definite value which is higher than the normal operating pressure of the system and lower than the relief valve setting. The higher sequence valve settings decrease the cycle rate because: (1) high pressure surges result which open the relief valve, and (2) more time is required to build up to the sequence pressure level due to the slight compressibility of hydraulic fluid.

CHAPTER VIII

RECOMMENDATIONS FOR FUTURE REASEARCH

Although this investigator found that automatic cylinder reversal could be achieved by utilizing pressure sequencing to create pilot signals, several modifications could be made to the system under investigation which might improve its operating characteristics. These characteristics would include the cycle rate and the minimum sequence pressure level.

It was felt that the four check valves D-5, D-6, D-9, and D-10 were necessary only because the pilot check valves were unable to operate effectively. Even with these check valves in the system, the minimum sequence pressure seemed to be unnecessarily higher than the operating pressure. This condition could be attributed to the inability of the pilot check valves to tank the undesired signal quickly, thereby causing the sequence pressure to build up to the higher level.

A new system should therefore be developed and tested which utilizes a commercially manufactured pilot check valve with some configuration other than that shown in Fig. 25, which was used in this investigation. If this new pilot check valve would operate effectively, then the standard check valves would not be necessary.

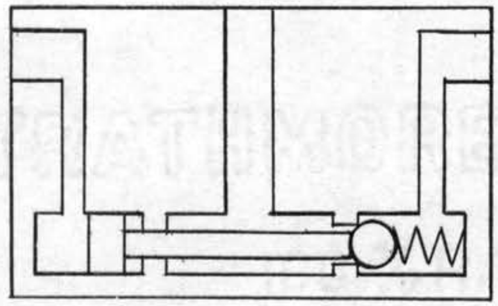


Fig. 23. Pilot Check Valve.

Another possible solution to this problem would be to utilize a system which does not require pilot check valves in its operation. Fig. 24 illustrates a system of this type. Here the undesired signal is tanked when one of the main pressure lines is open to tank. A hydraulic detent is also used which prevents the spool of the directional-control valve from drifting after the pilot signal is tanked. The hydraulic detent causes the system pressure to operate on a small plunger which contacts the spool of the directional-control valve. This plunger operates on an area which is separate from the pilot pressure area of the directional-control valve. A small area is utilized in a hydraulic detent because pilot pressure which shifts the directional-control valve must also overcome the pressure force existing on the plunger.

It is suggested that an investigation should be made to determine the characteristics of the system illustrated in Fig. 24. This system is composed entirely of commercial components with the exception of the hydraulic detent. The hydraulic detent modification requires that the four-way directional-control valve must be machined and new pressure caps must be added which contain the plunger assembly for the hydraulic detent. This system should be capable of operating at a minimum sequence pressure which is lower than the values obtained in the system under investigation. It should also be noted that due to the arrangement of the components, a counterbalance valve cannot be placed at the tank port of the directional-control valve to load both sides of the cylinder.

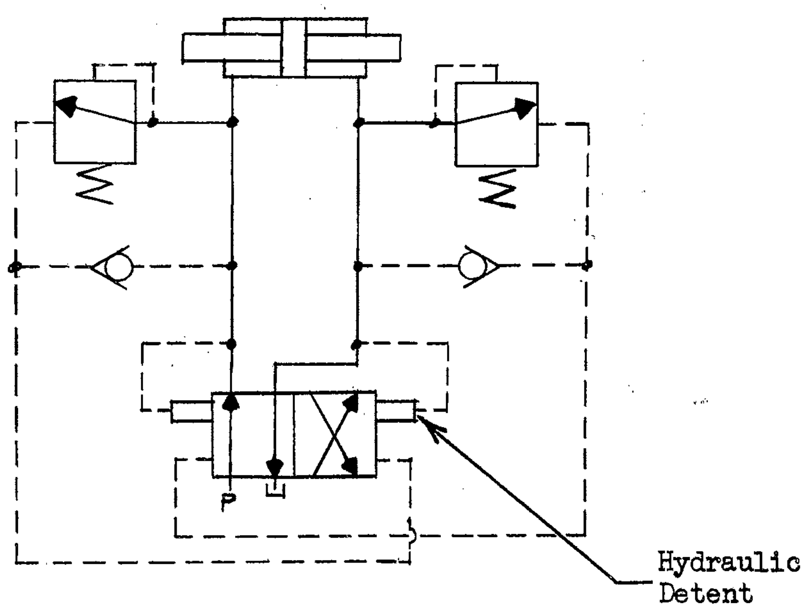


Fig. 24. Automatic Reversing Circuit.

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APPENDIX A

APPARATUS AND EQUIPMENT

Pump Circuit:

1. Louis Allen Electric Motor: rated, 1150 rpm; 3/4 horsepower; 220-440 volts, 3 phase, 60 cycles; serial no. 1812413.
2. Yale and Town Variable Displacement Pump: type A-6; serial no. 18467; 1 1/4 inch ports.
3. Young Heat Exchanger: maximum working pressure, 75 psi; model no., 67219; serial no., L 11187; number of passes, two; water inlet and outlet through the tubes, 1 inch; oil inlet and outlet to the shell side, 1 inch.
4. Reeves Vari-Speed Moto Drive Unit: size, 6281-C-12; gear ratio, 1.54 to 1; maximum output speed, 1500 rpm; minimum output speed, 250 rpm; electric motor, Robbins and Myers: rated, 1150 rpm; 15 horsepower; voltage, 220-440 volts, 3 phase, 60 cycles.
5. Commercial Gear Pump: model no. PD 322 BEEL 206; maximum pressure, 1500 psi; rotation, clockwise or counterclockwise; gear size, 2 inches; maximum capacity, 50 gallons per minute at 1000 psi; pump suction, 1 inch; pump discharge, 3/4 inch.
6. Reservoir: manufactured by Mechanical Engineering Laboratory; size, 30 inches by 34 inches by 17 1/4 inches; capacity, approximately 75 gallons; compartments, 4.
7. Marvel Sump Type Filter: two required; model no. C-1-10; capacity, 10 gallons per minute.

8. Capital Suction Line Filter: model no. 10M100; capacity, 10 gallons per minute.

9. Fluid Controls Pilot Type Relief Valve: part no. 1500-6-6; port size, 3/4 inch; pressure range, 50 to 2000 psi.

10. Texaco Regal Hydraulic Oil: 65 gallons; type, AZRO; viscosity range, 140 to 150 Saybolt Universal Seconds at 100°F; specific gravity, 0.868 at 80°F.

Cylinder Circuit:

1. Directional-control Valve, (*D-1): Manufacturer, Bendix Pacific Division; model no. 411620; type, manually-operated, four-way, three-position, pressure port blocked and cylinder ports open to tank in the neutral position; port size, 3/8 inch.

2. Directional-control Valve, (*D-4): Manufacturer, Electrol, Inc.; model no. 185-8FP; type, manually-operated, four-way, three-position, all ports blocked in the neutral position; port size, 3/8 inch.

3. Directional-control Valve, (*D-2): Manufacturer, Double A Products Co., model no. DA-180-C-SC; type, pilot-operated, four-way, three-position, spring-centered, all ports blocked in the neutral position; port size, 1 inch; pilot connections, 1/4 inch.

4. Directional-control Valve, (*D-3): Manufacturer, Double A Products Co., model no. DA-180-C; type, pilot-operated, four-way, two-position; port size, 1 inch; pilot connections, 1/4 inch.

5. Sequence Valve, (*S-1 and S-2): Manufacturer, Double A Products Co., model no. UA-165-C; type, internal pilot and external drain; pressure range, 500 to 1000 psi; port size, 3/8 inch.

*Indicates the number assigned to this valve in Fig. 17.

6. Counterbalance Valve, (*C-1): Manufacturer, Double A Products Co.; model no. SA-180-B; type, internal pilot and external drain; pressure range, 250 to 500 psi; port size, 1 inch.

7. Cylinder: Manufacturer, Carter Controls, Inc.; serial no. C-62032; size, 2 inch bore and 14 inch stroke, 3/4 inch rod, type, cushioned, double rod; maximum operating pressure, 1500 psi; port size, 3/8 inch.

8. Pilot Check Valve, (*D-7 and D-8): Manufacturer, Fluid Controls; model no. 25200-2; port size, 1/4 inch.

9. Check Valve, (*D-5, D-6, D-9, and D-10): Manufacturer, unknown; model no. 527-6D; port size, 3/8 inch male tube connections.

Instruments:

1. Chrono-Tachometer: Manufacturer, Standard Electric Time Co., model, CM.

2. Pressure Pickup: Manufacturer, Electro Products Labs; model no. 3700-A Electro Pressuregraph; maximum pressure, 2000 psi with 0.062 inch disc.

3. Oscillograph: Manufacturer, Allen B. Dumont Lab.; model, dual-beam cathode-ray; type, 322; serial no. 9X78.

4. A. C. Voltage Regulator: Manufacturer, Sorenson and Co., Inc.; model no. 1000-S; serial no. 1-3239; input voltage, 95-130 v.; output voltage, 115 v.; rating, 1 kva.

5. Gage Tester: Manufacturer, Ashcroft Division, Manning, Maxwell, and Moore, Inc.; type, 1313A; serial no. 2-50.

*Indicates the number assigned to this valve in Fig. 17.

APPENDIX B

SAMPLE CALCULATIONS

In analyzing the cycle rate of an automatic reversing circuit, it is necessary to know the time required to compress the hydraulic fluid to the sequence pressure level. Also it is necessary to determine the time required for fluid to flow from the sequence valve to the directional-control valve. A solution to this problem at a flow rate of 15 gpm and a sequence pressure of 1300 psi would be as follows:

1. The time necessary to compress the fluid to a sequence pressure level of 1300 psi is

$$t = \frac{C\ V\ P}{Q}$$

where t is time in seconds

C is a constant (see the equation below)

V is the total volume of fluid to be compressed (cu. in.)

P is the sequence pressure (psi)

Q is the flow rate in cubic inches per second

$$C = \frac{A - 2\ B\ P}{1 + A\ P - B\ P^2}$$

where C is the constant used in above equation

A is a constant depending upon pressure and temperature

B is a constant depending upon pressure and temperature

P is the sequence pressure level (psi)

At 1300 psi pressure and 100°F, $A = 4.38 \times 10^{-6}$ and $B = 5.65 \times 10^{-11}$.

$$C = \frac{4.38 \times 10^{-6} - 2(5.65 \times 10^{-11})1300}{1 + 4.38 \times 10^{-6} - 5.65 \times 10^{-11}(1300)}$$

$$C = 4.227 \times 10^{-6}$$

V_L = Volume of related lines = 508 in.^{3}

V_C = Volume of cylinder = 38 in.^{3}

V_T = Total volume = 546 in.^{3}

$$Q = 15 \frac{\text{gal.}}{\text{min.}} \times 231 \frac{\text{in.}^3}{\text{gal.}} \times \frac{\text{min.}}{60 \text{ sec.}} = 57.75 \text{ cu. in. per sec.}$$

$$t = \frac{4.227 \times 10^{-6} \times 546 \times 1300}{57.75} = 0.040 \text{ seconds for each reversal}$$

Total pressure build up time per cycle

$$t = 2 \times 0.040 = 0.080 \text{ seconds}$$

2. If a line volume of 1 cubic inch is assumed, the time required for the pilot signal, which was produced by the sequence valve, to energize the directional-control valve would be

$$t = \frac{V}{Q} = \frac{1.0}{57.75} = 0.017 \text{ seconds}$$

VITA

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Candidate for the Degree of

Master of Science

Thesis: AN EXPERIMENTAL INVESTIGATION OF FLUID SIGNAL CREATION

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