AN INVESTIGATION OF INTERMITTENT FEEDING OF A HYDRAULIC CYLINDER

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PREFACE

Although the study of hydraulics is many centuries old, the field of fluid power is just a young science. It has only been since World War II that the use of fluid power has come to the foreground as a control mechanism. Prior to that time, fluid power was used principally as a convenient method of moving large masses or for generating large forces. This study is the result of looking for a new and more positive method of positioning the piston within a hydraulic cylinder.

My deepest appreciation is due Professor E. C. Fitch for his competent advice, his constructive criticism, and often needed encouragement. I wish to thank Dean M. R. Lohmann, Dr. Clark Dunn, Professor R. E. Venn, and Dr. J. H. Boggs for making it possible for me to do this study. To Professor B. S. Davenport and Mr. George Cooper, I am grateful for their aid and advice during the construction of the testing apparatus. I am indebted to Professor C. M. Leonard for his constructive criticism of my thesis. My gratitude is due Mrs. Mildred Avery for efficient and capable services as my typist during the preparation of this thesis. Last but not least, to Wanda and my girls I can only say thanks.

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CHAPTER I

INTRODUCTION

The field of fluid power first started in the eighteenth century when men first realized the advantages of using a fluid to move rather large masses easily and quickly. Then shortly after the First World War, the pioneers in this field realized the advantages of using oil over that of using water as the fluid in the fluid power systems. The modern day presses employ large rams to transfer large masses and many of these presses are capable of exerting forces in the neighborhood of 50,000 tons. So today when some people think in terms of fluid power, they think only of the large forces which some fluid power systems are capable of producing, not realizing the advantages of fluid power systems as a control system.

In its search to find better and easier ways to accomplish the necessary positioning, drilling, cutting, and rotating of a given object, industry is turning more and more to fluid power not only as a means of moving large masses but also as a means of producing the precision control necessary in the modern day industry. Pippenger (9) stated that a quick summary might reflect the fact that power to any device that must quickly change direction, go around corners, or be infinitely controllable at low cost, fits into the category of the fluid power industry.

The positioning of the cutting table, the rotating of a turntable, plus many other aspects in this line are all required on the modern machine tools in order to achieve the maximum benefits which are available in the era of automation. The successful operation of an electric-arc furnace depends upon the proper positioning of the heavy electrodes inside the furnace. (12). In the future, and as the nuclear reactor becomes more commonplace, the use of fluid power as a source of control as well as a means of applying large forces will become a more universal practice. Rice (10) stated that the advantages of using a fluid powered system on a nuclear reactor were the facts that there was a minimum of system inertia to be overcome, that use of fluid power gave fast scram action, and that an operator had extreme flexibility in shim rates in controlling the reactor rods. George (5) stated that there was no apparent radiation damage to the hydraulic fluid or to any of the hydraulic components in the nuclear reactor in operation at Los Alamos scientific laboratory.

The need to position an object quickly, economically, and accurately suggested a need for making this investigation. Some of the problems encountered during this investigation came about due to the fact that the author tried to limit the circuit to only one system, a fluid power system with no electrical nor mechanical controls involved.

CHAPTER II

PREVIOUS INVESTIGATIONS

No evidence was found that any work had been published relative to the proposed investigation. Several papers have been published which, in general, represent possible solutions, either to the positioning of a slave cylinder or to pressure sequencing for obtaining pilot signals. In any case, there was no paper uncovered which dealt with both of these problems at the same time.

One of the earliest attempts to position a slave cylinder was made in 1932, when Morey (8) positioned a cylinder by means of a rachet and pawl to obtain intermittent feeding of the cylinder. There were several other attempts to do this by means of mechanical linkages such as tying a rack on the piston rod of the slave cylinder which was driving another rack attached to a smaller cylinder through a pinion gear. As the piston in the smaller cylinder moved, fluid was displaced from one side of the piston through a fluid motor and then back to the other end of the small piston. The position of the slave cylinder was indicated by a pointer attached to the shaft of the fluid motor.

Probably one of the most successful methods of positioning a slave cylinder is by means of limit switches and solenoid operated directional control valves. This involves placing the limit switches at known desired locations and as the piston rod hits one of the limit switches, the travel of the piston in the slave cylinder will stop. However, this method generally relies on fixed locations and does not provide the flexibility needed in a modern fluid power circuit according to A. Dürr. (3).

J. C. Cotner (2) reported another means of positioning a cylinder by a combination of cam-operated pilot-size directional control valves. As a dog on the rod of the cylinder hits a cam, the direction of flow through the pilot-size valve was reversed thereby stopping the cylinder's travel. Again the main limitation of this type of arrangement was the fixed positions necessary which allowed very little flexibility in changing the stop position.

If size and weight limitations are necessary in positioning of a cylinder such as would be the case in the aircraft or the guided-missile field, the only logical method at the present time would require the use of servo-motors and valves. This method is very successful; however, in 1949, Buller and Ford (1) reported that this type of control had several disadvantages in the guided-missile field. These disadvantages were the following: (1) time lag in response time, (2) non-proportional displacement, and (3) the limited mechanical movements and electrical quantities.

As to the second part of the problem, that of using pressure sequencing for obtaining pilot signals, Hadekel (6), a consulting engineer in England, used a special valve basically consisting of two sequence valves built into a hydraulically operated two-position, four-way, directional control valve to obtain the cylinder reversal by pressure sequencing. At the present time, this special valve combination is not available commercially in the United States. The thinking of the engineers in the United States is along the lines of main line or power sequencing rather than pilot sequencing. Both Stewart (11) and Kurzweil (7) reported mainline sequencing operations, which let the pressure in one part of the system build up to the desired level, then allowing the sequence valve to open permitting all the flow to go to another cylinder. This has the disadvantage of requiring each sequence valve to be set at a higher pressure than the previous one so as to be capable of passing the full volume of the pump.

CHAPTER III

STATEMENT OF PROBLEM

This investigation is two-fold. The first objective was to determine whether it was possible to position a slave cylinder accurately some place between the limits of travel of the piston by using an intermittent feeding process and by employing commercially available components in an open circuit. The second objective was to determine whether or not it was possible to cause the reversal of a cylinder by letting the pressure build up at the end of its stroke and sending a signal through a pressure sequence valve to the directional control valve, thereby reversing the direction of flow through the valve and causing the cylinder to reverse its direction of travel.

It is possible to accomplish the positioning of the cylinder by using a serve-valve or by using two cylinders tied together in a closed loop so that any movement of one cylinder would cause the other cylinder to move the same distance. The first method, using a servo-valve, requires a feedback which involves an additional system that is in many cases impractical. The second method limits the amount of load that can be imposed on the slave cylinder to the amount of load that it would take to move the master cylinder to the desired position. This arrangement also limits the application of this method. After careful study of the problems, it was believed that the positioning of a slave cylinder could be accomplished by using a positive-displacement valve to feed a slave cylinder. The intermittent feeding of the slave cylinder caused by the movement of the positive-displacement valve would cause the piston rod in the slave cylinder to move out in constant increments. By stopping the positive-displacement valve after a given number of cycles, it would be possible to stop the slave cylinder at any desired position. The only limitations on loads that could be moved by the slave cylinder would be those caused by the pressure and volume capabilities of the pump and the net area of the slave cylinder which was available.

To prevent the introduction of another system, such as a mechanical or electrical system, into the problem, it was decided to try to obtain the reversal of the positive-displacement valve by using pressure sequence valves. At the end of each stroke of the positive-displacement valve, the pressure in the system would increase thereby causing the pressure sequence valve to open. This allowed the fluid to pass through the valve and to one side of the directional control valve. The direction of the fluid flow through the valve would be reversed causing the piston in the positive-displacement valve to reverse its direction of travel.

Due to many uncertainties of the characteristics of most of the hydraulic components, it was believed that the solution of these problems could best be obtained from an experimental approach rather than an analytical solution. Therefore, the author in this investigation approached the solution to these problems through an experimental research.

CHAPTER IV

ANALYSIS OF CIRCUIT OPERATION

Definition of Components

In order to analyze adequately the problem at hand, one must have a complete understanding of the hydraulic components used in the circuit and their purposes for being used. Not only will each of the components be identified as to their characteristics and functions; but there also will be included a graphical representation of each component as approved by the American Standards Association.

A relief valve is a safety valve used to maintain the pressure in the circuit below the desired maximum pressure and is adjustable over a wide pressure range. As soon as the pressure in the system reaches this limiting pressure, the relief valve opens and allows the excess volume of fluid to bypass the circuit and return directly to the reservoir. The main purpose of the relief valve is to prohibit the pump from being subjected to excessively high pressures. See Fig. 1.

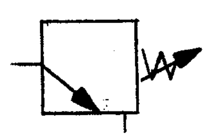


Fig. 1. Relief Valve.

A sequence valve is a pressure control valve. It is used to control the order of flow to the various parts of the system by requiring the pressure at the inlet port to reach the desired pressure level before the sequence valve will open thereby permitting fluid to pass through the valve. Full pressure is then available at the outlet port only as long as the inlet control pressure remains above the spring setting of the valve. This spring setting is adjustable over a wide range of pressure settings. For this investigation the sequence valve was used to obtain pilot signals. See Fig. 2.

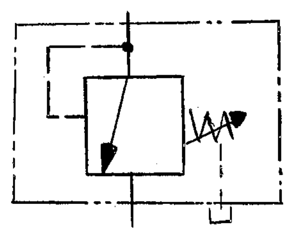


Fig. 2. Sequence Valve.

A counterbalance valve is also a pressure control valve which permits free flow in one direction and restricted flow in the other direction. In one direction, flow through the valve is blocked until the back pressure in the system reaches the desired level, at which time the valve opens allowing fluid to flow. This flow continues until the pressure in the system drops below a given minimum value at which time the valve closes. In the opposite direction through the valve, the flow is not restricted in any way. Usually the back pressure is variable over a large range by adjusting the spring setting of the valve. The valve was used to simulate a given load on the cylinder. See Fig. 3.

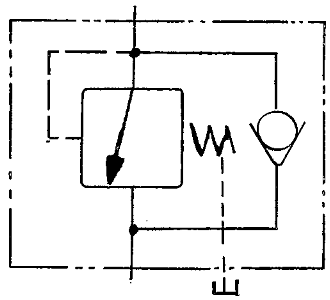


Fig. 3. Counterbalance Valve.

A check valve is used to permit flow in one direction and to block the flow in the opposite 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.

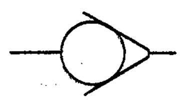


Fig. 4. Check Valve.

A pilot check valve is essentially a check valve with the additional provisions for allowing reverse flow when desired. It is a free flowing valve in one direction while it has blocked flow in the other direction until such time as pilot pressure is placed on the ball thereby unseating the ball and permitting free flow in the reversed direction. See Fig. 5.

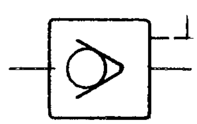


Fig. 5. Pilot Check Valve.

A directional control valve is used for the specific purpose of directing the flow of fluids in hydraulic systems to cylinders, to fluid motors, or to other valves. The movement of the valve spool may be accomplished either manually, electrically, by applying pilot pressure to the spool end, or by a combination of any of these methods. When the graphical symbol for the valve is used in the circuit, it is shown in the neutral position. These valves may be either two-way, three-way, four-way, or even more, meaning the number of main line connections in the valve; however, in this investigation only four-way valves were used. Usually these valves are either two-position or three-position valves. When the four-way, three-position valve is in the extreme left position, the pressure port is open to cylinder part A, while cylinder port B is open to the tank port. When this valve is in the extreme right position, the pressure port is then connected to cylinder port B, while cylinder port A is connected to the tank port. In the neutral position of a four-way, three-position valve, the various ports may be interconnected in twelve different ways by using different spools. Two combinations of four-way, three-position valves were used in this investigation: (1) in the neutral position in one combination all four ports were blocked; and (2) in the other combination only the pressure port was blocked with the rest of the ports being interconnected. Figure 6 shows the graphical symbol for a pilot operated, four-way, two-position directional control valve. Figure 7 is the graphical symbol for a pilot-operated,

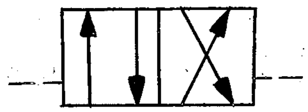


Fig. 6. Directional Control Valve.

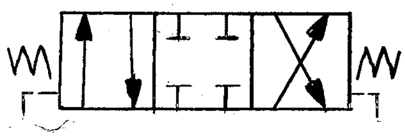


Fig. 7. Directional Control Valve.

spring-centered, four-way, three-position valve which has all four ports blocked in its neutral position. The graphical symbol for a manually-operated, four-way, three-position directional control valve which has all four ports blocked when in the neutral position is shown in Fig. 8. Figure 9 represents the same type of directional

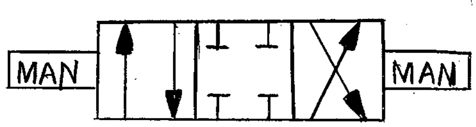


Fig. 8. Directional Control Valve.

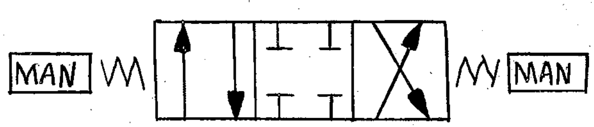


Fig. 9. Directional Control Valve.

valve as is illustrated by Fig. 8 only with the addition of spring centered neutral position; whenever the control handle is released, the valve returns to the neutral position. The graphical symbol for a solenoid-operated, spring-centered, four-way, three-position directional control valve which has the pressure port blocked and the three other ports connected together when the valve is in its neutral position is illustrated by Fig. 10.

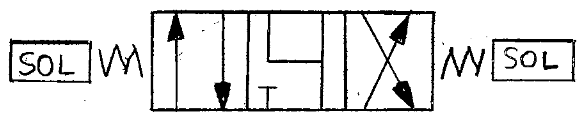


Fig. 10. Directional Control Valve.

A gate valve is an on- or off-type of shut-off valve. This valve is manually operated and its graphical symbol is illustrated by Fig. 11.

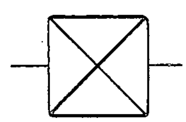


Fig. 11. Gate Valve

A needle valve is a manually-operated, non-pressure compensated, volume regulated type of valve. It has a long tapered point that permits a very gradual opening and closing of the passage. The graphical symbol for a needle valve is illustrated by Fig. 12.

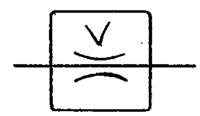


Fig. 12. Needle Valve.

Figure 13 illustrates the graphical symbol for a pump unit which is driven by an electric motor through a variable speed reducer. The pump is a fixed-displacement gear type and becomes a variable volume pump by varying the speed of the input shaft.

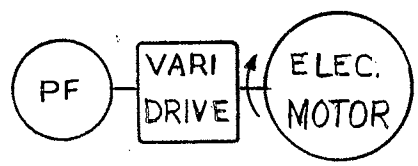


Fig. 13. Pump Unit.

In Fig. 14 a fixed displacement pump is illustrated directly coupled to an electric motor.

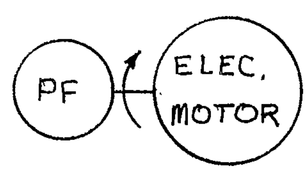


Fig. 14. Pump Unit.

A positive-displacement valve is a constant-volume valve. The purpose of this valve was to provide a source of a constant volume of fluid to the circuit. The graphical symbol for this valve is shown in Fig. 15. Since this type of valve is unavailable on the

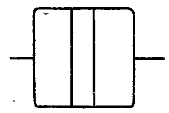
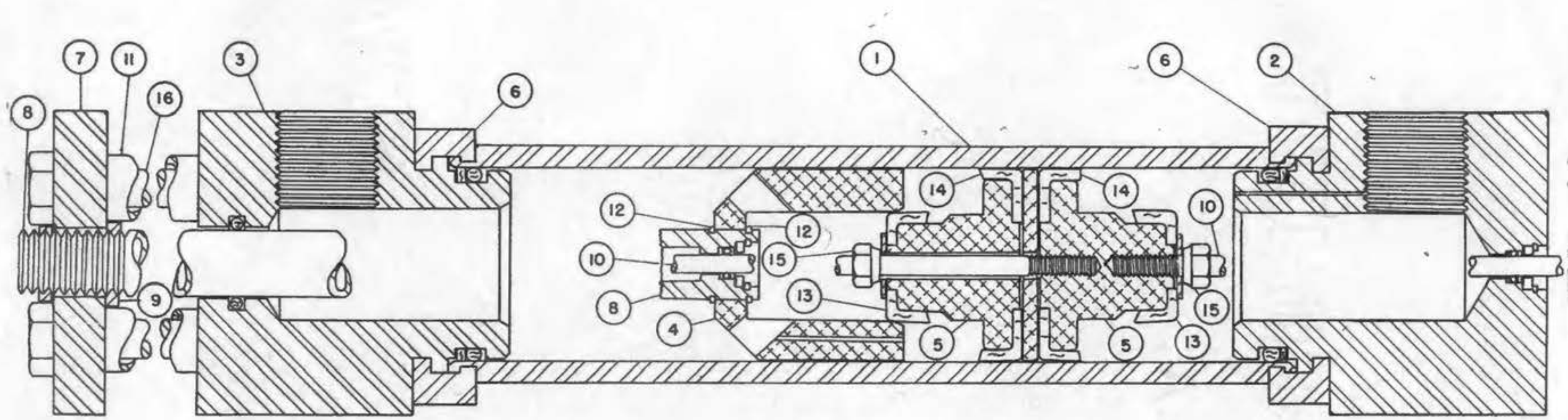


Fig. 15. Positive Displacement Valve.

commercial market at the present time, it was necessary to design and build a valve which would fulfill these requirements. This writer designed the valve and the Research Apparatus Development Laboratory of the College of Engineering at Oklahoma State University manufactured the valve. Basically speaking, the valve consisted of a free-floating piston inside a cylindrical tube. Fluid flow through one port would move the piston against an adjustable stop. Then, by reversing the flow through the valve, the piston would be forced against the other stop and as this movement occurred a constant volume of fluid was forced out ahead of the piston. Then, again reversing the flow, another volume of fluid was forced out ahead of the piston. Since one of the stops was initially adjustable by a screw arrangement, the volume of fluid which was displaced ahead of the free floating piston could be varied. The one-quarter inch rods which were extended out from both ends of the valve were for the purpose of giving a visual check as to the position of the free-floating piston inside the cylinder. As previously stated, the positive-displacement valve is unavailable on the commercial market; therefore, a cross sectional view of the valve is included as shown in Fig. 16.



NO.	PART NAME	NO.	PART NAME
1	CYLINDER TUBE	9	LOCK NUTS
2	CYLINDER END "A"	10	PUSH RODS
3	CYLINDER END "B"	11	COMPRESSION TUBE
4	ADJUSTABLE STOP	12	RETAINING RINGS
5	FREE-FLOATING PISTON	13	CUSHIONING CUPS
6	END CLAMPS	14	PACKING CUPS
7	BACK STOP	15	LOCK NUTS
8	ADJUSTING ROD	16	STUD BOLTS

FIGURE 16 CROSS-SECTIONAL VIEW OF THE POSITIVE-DISPLACEMENT VALVE

An accumulator is an energy-storage container. The purpose of the accumulator for this investigation was to absorb surges or shock waves created in the hydraulic system by the sudden closing of the directional control valve. It is directly analogous to a spring in a mechanical system. Its graphical symbol is shown in Fig. 17.

Fig. 17. Accumulator.

To dissipate the heat generated in the oil-powered circuit, a heat exchanger was used. By circulating the oil through the heat exchanger, the temperature of the oil in the system was maintained at a constant level thereby keeping the properties of the oil within certain limits. In this investigation a two-pass condenser was used with the oil going to the shell side and with the water going through the tubes. The graphical symbol of the heat exchanger is shown in Fig. 18.

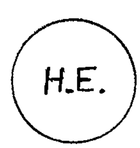


Fig. 18. Heat Exchanger.

The graphical symbol for an oil reservoir which is vented to the atmosphere is illustrated in Fig. 19, and Fig. 20 illustrates the graphical symbol for a strainer. The graphical symbol for a pressure gage is shown in Fig. 21, and the graphical symbol for a double-acting, single rod cylinder is illustrated in Fig. 22.

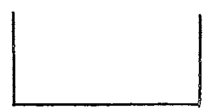


Fig. 19. Oil Reservoir.

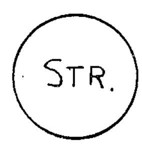


Fig. 20. Strainer.

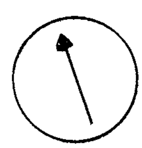


Fig. 21. Pressure Gage.

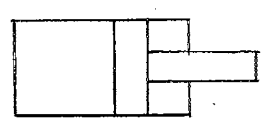


Fig. 22. Cylinder.

The Integrated Circuit

The analysis of the problem can best be discussed by dividing the complete hydraulic system into three major circuits. The first circuit discussed will be the pump unit and its related equipment; the second circuit will be the positive-displacement valve unit and its related equipment; and the third circuit will be the slave cylinder unit and its related equipment.

A graphical representation of the complete fluid power system is illustrated in Fig. 23, and the physical arrangement of the original system is shown in Fig. 24.

Pump Circuit

The pump circuit can be further divided into two major units: (1) the main pump unit, and (2) the recirculating unit. Before the fluid entered the pump it passed through one of three strainers to keep the oil in the system relatively clean. Foreign contaminants are extremely harmful to hydraulic valves and pumps. The fluid then flowed through the gear pump and into the system. The flow rate into the system was regulated by varying the speed of the pump by means of a vari-drive unit. The pressure of the system was measured by a gage installed in the discharge line of the pump. To protect the pump from being subjected to excessive pressures a relief valve was installed on the downstream side of the pump.

The second unit in the pump circuit was the recirculating unit. It was installed for the purpose of keeping the fluid in

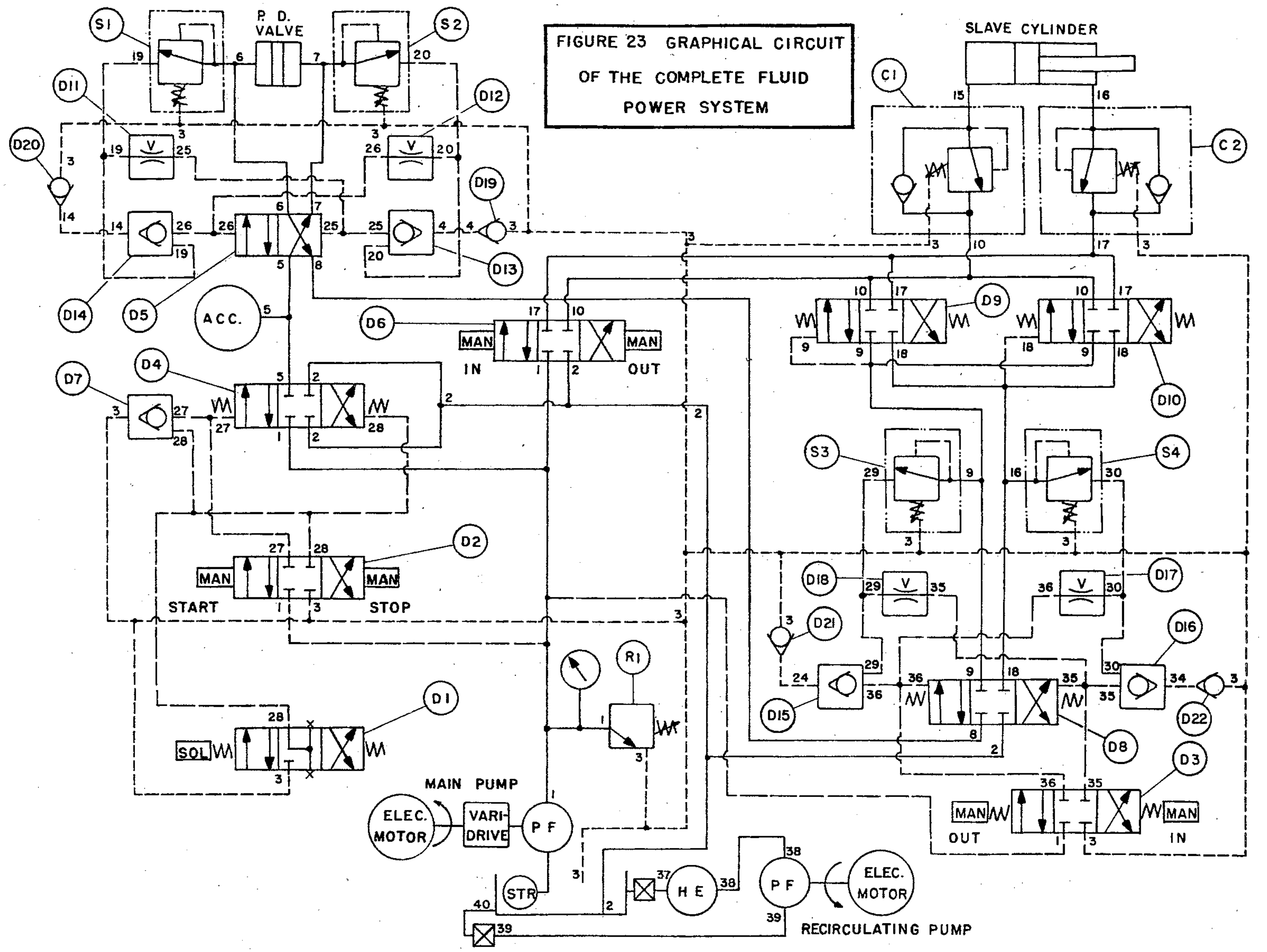


FIGURE 23 GRAPHICAL CIRCUIT OF THE COMPLETE FLUID POWER SYSTEM

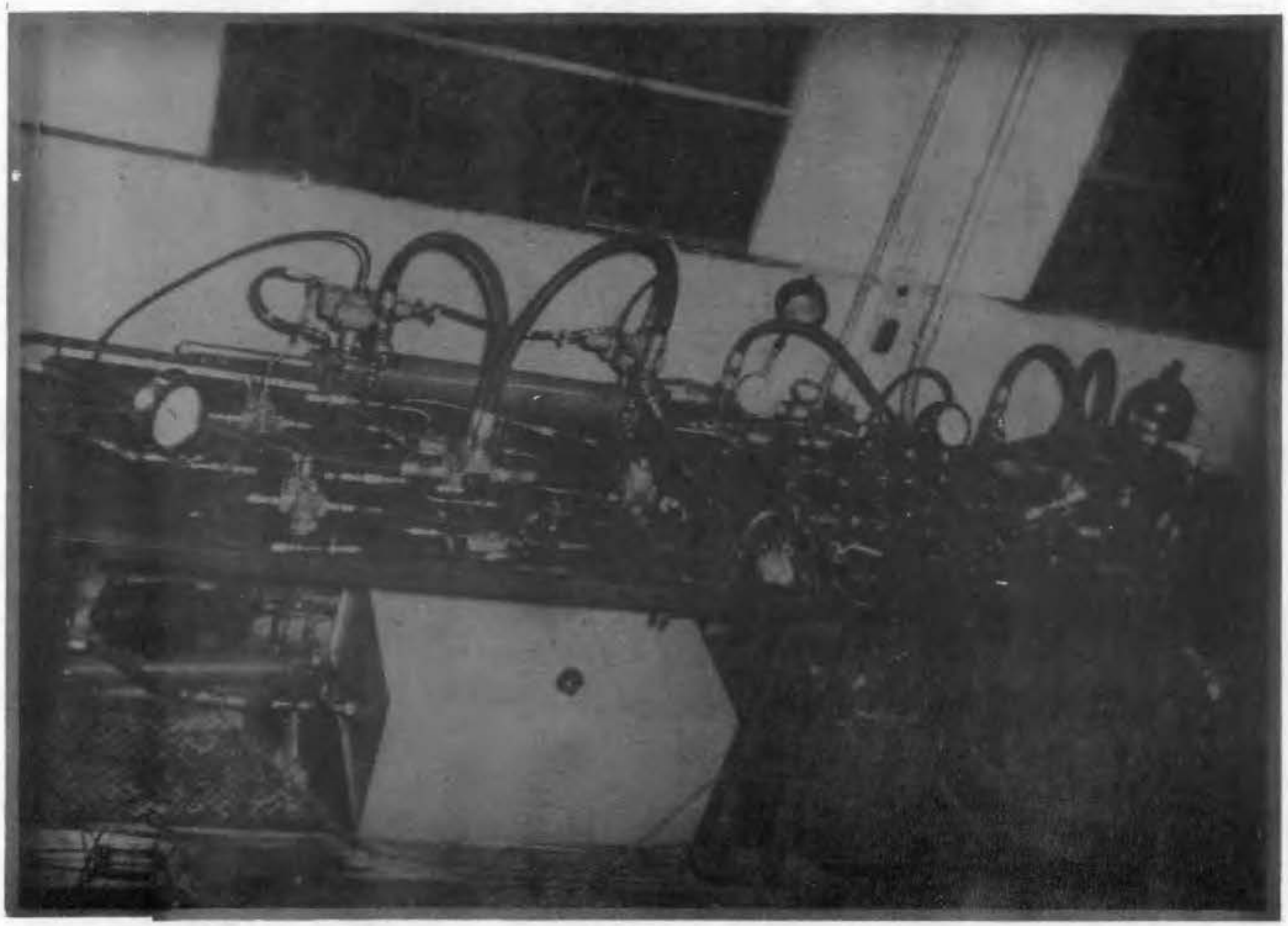


Fig. 24 Physical Arrangement of the Original System.

the system reservoir at a constant temperature. Fluid was taken from the reservoir by the recirculating pump and then returned to the reservoir through a two-pass heat exchanger.

Positive-Displacement Valve Circuit

Fluid flow into the positive-displacement valve circuit was blocked by directional control valve, D-4, until such time that pilot pressure was applied from the manually operated directional control valve, D-2. When valve, D-2, was shifted to the extreme left position, position No. 1, pilot pressure was applied to the left side of directional control valve, D-4, which shifted its spool to allow fluid to enter the circuit. The fluid then entered the accumulator and also the directional control valve, D-5. As shown in the circuit diagram, this valve was in the right hand position which allowed fluid to go to the right side of the free-floating piston inside the positive-displacement valve. This caused the piston to move to the left. The fluid displaced by the piston on the other side of the valve went through directional control valve, D-5, and then to the slave cylinder circuit.

When the free floating piston reached the limit of its travel to the left, the fluid pressure increased causing sequence valve, S-2, to open. This permitted the signal fluid to travel through pilot line No. 20. The pilot signal fluid energized pilot check valve, D-13, which opened it, thus allowing the pilot signal from the right side of directional control valve, D-5, to go to the tank.

The pilot signal continued until it passed through the needle valve, D-12, and then to the left side of directional control valve, D-5. As the pilot pressure built up on the left side of the spool, the spool shifted. This action changed the direction of flow of the fluid through the valve. The fluid then entered the left side of the positive-displacement valve where the same cycle of events took place using a similar series of valves -- S-1, D-14, D-11, and D-5.

This automatic cycling of the positive-displacement valve continued until the spool in directional control valve, D-4, was shifted to the right. This was accomplished by moving the spool of manual valve, D-2, from the extreme left position to the extreme right position, position No. 2, which sent a pilot signal to the right side of directional valve, D-4, and at the same time tanked the pilot signal to the left side of the directional valve, D-4. As soon as the spool in directional valve, D-4, was shifted to the right, the flow of the fluid was diverted from the positive-displacement valve circuit directly to the reservoir thus stopping the automatic cycling operation.

If it were desired to position the rod on the slave cylinder manually, it was necessary to energize the solenoid on directional control valve, D-1, which caused the valve spool to move to the extreme left. This tanked the pilot signal to the right side of directional control valve, D-4, and the valve spool returned to the neutral position where all ports through the valve were blocked. By de-energizing the solenoid of valve, D-1, the spool returned to the neutral position.

Slave Cylinder Circuit

The control of the slave cylinder circuit was provided by directional control valve, D-3. When this valve was moved into the left position, pilot pressure was transmitted to the left of directional control valve, D-8, which permitted the piston in the slave cylinder to move out. When this valve, D-3, was positioned to the right, pilot pressure was sent to the right side of valve, D-8, and any pilot pressure to the left side of valve, D-8, was tanked which permitted the piston in the slave cylinder to move in.

In starting the circuit, valve D-3 was positioned to the right so that pilot pressure was exerted on the right side of valve D-8. This movement of the spool in valve D-8 directed the fluid from the positive-displacement valve to line 18 and tanked the fluid in line 9. After the fluid entered line 18, nothing happened in the slave cylinder circuit until the fluid pressure was great enough to overcome the spring resistance in valve D-10. At this time the spool of valve, D-10, was positioned to the left. This movement of the spool permitted the fluid to flow through the valve, D-10, on through the unrestricted passage in pressure control valve, C-2, and into the rod side of the slave cylinder. The cycling operation of the positive-displacement valve intermittently delivered a constant volume of fluid to the slave circuit and caused the piston in the slave cylinder to move in constant increments.

The fluid on the blank side of the slave cylinder was restrained by pressure control valve, C-1, until the pressure of the fluid on the blank side of the slave cylinder was high enough to overcome the spring setting on the valve, C-1. In this manner the slave cylinder was effectively loaded to a simulated load condition. After the fluid had passed through the valve, C-1, it passed through the directional control valve, D-1, and then through valve, D-8. The fluid was tanked after it had passed through valve, D-8.

When the rod in the slave cylinder reached the end of the stroke, pressure in the system increased rapidly until such time as the pressure in the system was large enough to overcome the pressure setting of sequence valve, S-4. Then full line pressure was available at the outlet port of the sequence valve, S-4, and fluid was able to enter pilot line 30. The pilot signal went to pilot check valve, D-16, energized the valve and tanked the pilot signal from the right side of directional control valve, D-8. The pilot signal continued until the signal passed through the needle valve, D-17, and on the the left side of directional control valve, D-8. As the pilot pressure was applied to the left side of the valve, D-8, the spool shifted and the direction of flow through the directional control valve, D-8, was reversed. Then the flow of the fluid was directed to the blank side of the slave cylinder where the same sequence of events took place only using a similar series of valves--D-9, C-1, C-2, S-3, D-18, and D-15.

This sequence of events continued until the flow of the fluid coming from the positive-displacement valve circuit was stopped.

A check valve was installed on the drain side of each pilot check valve in such a way that any fluid signal would be tanked through the pilot check valve. It was installed to eliminate any possibility of an extraneous signal passing back through the unrestrained passage in the pilot check valve and causing the pilot operated directional control valve to shift unexpectedly.

In order to accurately simulate the load conditions on both sides of the slave cylinder, it was necessary to provide a direct source of fluid from the main pump unit to the slave cylinder so the piston in the slave cylinder could be run in or out to the end of its stroke, or positioned somewhere between the limits of its stroke. One means of providing this direct path was by the addition of manual directional control valve, D-6, which bypassed everything in the slave cylinder circuit with the exception of the counterbalance valves.

Size Selection

F. R. Ward (13) stated that in the design of the Consolidated Edison Reactor it was necessary to move a load 4 feet in 1/3 seconds and then decelerate to a cushioned stop. In order to accomplish this, a velocity in excess of 12 feet per second would be required. After checking existing available equipment, it was decided to use a Reeves Vari-drive unit driving a Commercial gear type pump in conjunction with a 2-inch diameter cylinder which has an 18-inch stroke. At the rated speed of 1500 rpm, the pump was capable of discharging 32 gallons per minute which gave the piston of the slave cylinder a velocity of 3.3 feet per second. Even though these two velocities were not comparable, it was felt that this 3.3 feet per second velocity would serve as a starting point for this investigation. At a later time in the investigation, the 2-inch diameter cylinder was exchanged for a 3-inch diameter cylinder with a 24-inch stroke. In order to prevent excessive pressure drop through the connecting lines and through the valves, it was decided to keep the fluid flow through the circuit in the laminar region which would allow a maximum fluid velocity somewhat less than 20 feet per second.

With a flow rate of 32 gallons per minute through the system, the fluid in a 1-inch standard pipe would have a velocity of approximately 14.6 feet per second while in a 3/4-inch line it would have a velocity of approximately 23.5 feet per second. Since the maximum capacity of the pump as used with the Reeves Vari-drive unit was 32 gallons per minute and since the maximum range of this investigation was limited by the pump capacity, this investigator decided that the main pressure lines would be 1-inch and the main pressure hydraulic valve would also be of 1-inch size. Since the sequence valves, needle valves, and the pilot check valves would only be used for pilot signals, it was decided that 1/4-inch valves would be sufficiently large to pass the required flow. It was also decided that all pilot lines would be 1/4-inch and that all branch drain lines and pilot return lines would be 3/8-inch to eliminate the possibility of excessive back pressure. The directional control valves used strictly for control of the pilot signals were selected from available hydraulic control valves of 1/4- and 3/8-inch size.

After investigating the cost of 1-inch pilot check valves, it was decided to substitute 1-inch four-way, three-position directional control valves, D-9 and D-10, in place of 1-inch pilot check valves. During the experimental investigation, certain eccentricities of the circuit appeared which required a thorough investigation of the characteristics of the directional control valves that were used in place of pilot checks. Upon investigating these characteristics, it was found that at static pressures above 700 psig these directional control valves had excessive leakage due to the clearances in the valve and they could not be used in place of pilot check valves. After these undesirable characteristics of the directional control valves were revealed, it was necessary to eliminate the provisions for manually positioning the slave cylinder by removing from the circuit directional control valves, D-6, D-9, and D-10, and their related lines.

Revised Circuit

At a later period during this investigation, the difference in the pressure ranges of the commercially available sequence valves could not be adjusted far enough apart to permit automatic operation of both the positive-displacement valve circuit and the slave cylinder circuit. Since the positive-displacement valve as designed by the author required some major modifications, it was then decided to determine the effects of different load conditions on the sequencing time and operation of the slave cylinder. The circuit was revised as indicated in Fig. 25. The valves and components were not changed and were for the same purpose as they were in the original circuit with the exception of directional control valve, D-8, which was converted to a four-way, two-position directional control valve with no springs to center it to the neutral position.

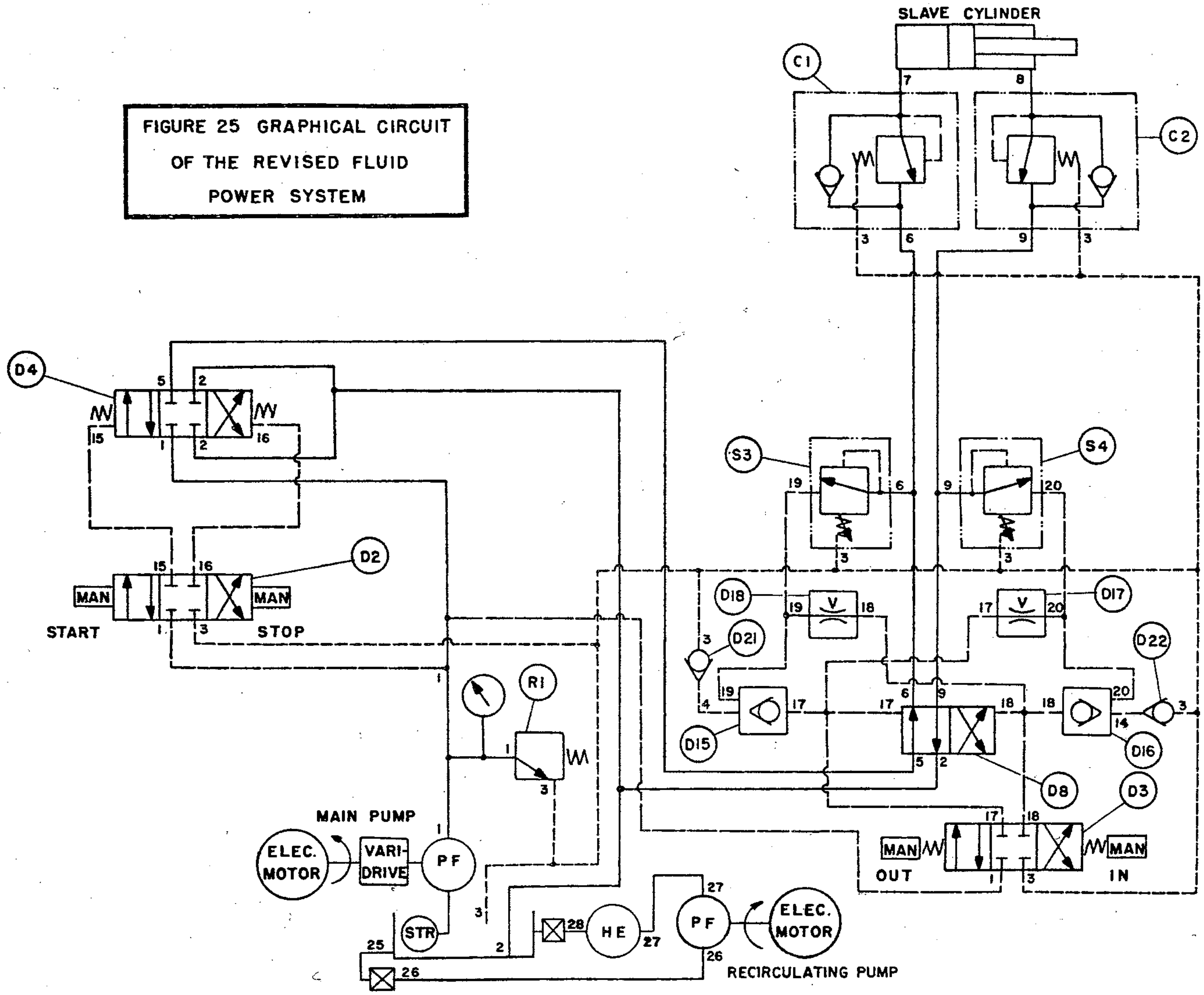


FIGURE 25 GRAPHICAL CIRCUIT OF THE REVISED FLUID POWER SYSTEM

CHAPTER V

TESTING PROCEDURES

Before any test was started, a quick look at the entire system was required in order to be sure that there were no disconnected lines in the circuits. Next, the directional control valve, D-2, was checked to be certain that it was in the No. 2 position so that the pump would not be loaded when it started. The cooling water was turned into the heat exchanger and then the recirculating pump was started. After both of these units were running, the main pump was started. Before proceeding any further, a careful inspection of the relief valve was made in order to be certain that it was in good operating condition.

Directional control valve, D-2, was then moved to the No. 1 position which allowed fluid to enter the positive-displacement circuit. At this time, all actions in the circuit were manually controlled and the noise from the circuit was very obnoxious until all the air was removed from the lines. First it was necessary to bleed as much air as possible from the system by opening pressure gage connections or breaking lines at the high places in the circuit. When bleeding the lines no longer reduced the amount of air in the circuit, the system was run until all the air was entrained in the oil. The entrained air was separated from the oil in the reservoir. After the air was eliminated from the circuit, the circuit noise was at a normal level. After the air was initially eliminated from the circuit, the above procedure was not required again until a line had been disconnected.

The desired load condition was introduced into the system by adjusting the pressure setting of the counterbalance valves. This was accomplished by adjusting the spring settings of the valves until the desired back pressure of the system was reached which was required to open the counterbalance valves.

Each system, positive-displacement valve circuit and slave cylinder circuit, was then isolated and the sequence valves in each individual circuit were adjusted until the circuit was operating automatically. Then the two circuits were tied together and attempts were made to readjust the sequence valves in the positive-displacement valve circuit until the positive-displacement valve would again operate automatically in conjunction with the slave cylinder circuit. However, due to the similarities of the pressure range adjustments of all four sequence valves, it was impossible to get the pressure setting of the sequence valves in the positive-displacement valve circuit set high enough above the sequence valves in the slave cylinder circuit to maintain a continuous automatic operation of both circuits.

Before the main pump was stopped it was again necessary to move the directional control valve, D-2, into the No. 2 position in order for the pump to be in the unloaded position when the circuit was again started.

After the circuit was revised as indicated on Fig. 25, the startup procedure for the slave cylinder circuit was just the same as above. After the desired load condition was imposed on the system, by adjusting the counterbalance valves, and after the sequence valves were checked in order to have automatic operation, a series of runs was made at the different pump speeds. The time for a complete cycle of the piston in the slave cylinder was determined at the various flow rates. New load conditions were imposed and the test was repeated. These tests were continued until the capacity of the driver of the main pump was reached.

CHAPTER VI

DISCUSSION OF RESULTS

Even though the investigator could not make the original circuit work properly and give the desired results, there were enough indications displayed by the circuit that with a few modifications made on the various components, such as directional control valves, pilot checks, and positive-displacement valve, the circuit could be made to work in the expected manner.

In order to eliminate some of the troubles that were encountered in the circuit operation, it was found that the pilot operated directional control valves should have the following features built into the valves or added to the valves. First, there should be some type of stroke and choke adjustment to the spool of the valves to limit the stroke length of the spool and also to control the velocity and acceleration at which the spool starts and stops. These features either should be built into the valve when the valve is commercially obtained or should be built into the valve after obtaining it. Second, there should be a built in feature which would tend to eliminate the tendency of the spool to drift from the desired position. This may be partially accomplished by using a friction drag mechanism, such as an O-ring installed at each end of the spool between the tank port passage and the end of the spool. The use of an O-ring in this position actually accomplishes two things. It creates a friction surface which must be overcome anytime the spool's position is shifted; also, it tends to eliminate any leakage past the tank port passage to the pilot end of the spool thereby giving positive assurance that there will be no unnecessary pressure build-up in the pilot caps which may cause the spool to shift positions.

As stated previously, the characteristics of these pilot operated directional control valves were such that they could not be substituted for pilot check valves. Any time fluid was applied to the pressure port there was some leakage by the spool to the cylinder ports and for this reason there was not a positive shut-off of the fluid flow.

The last feature that the pilot operated directional control valves should have incorporated for this type of circuit is a hydraulic or mechanical mechanism to insure that the spool would shift only when sufficient pressure was applied to the pilot end of the spool to enable the spool to travel through a complete stroke. This mechanism would eliminate any tendency of the spool to stop in its neutral position. Adding all the above features to the pilot operated directional control valves would be required in order for them to have the desired characteristics necessary in this type of circuit.

The pilot check valves which were obtained for this circuit also had to have several modifications made to them before they could be used satisfactorily in the circuit. When the valves were received from the factory, a small orifice sized hole had been drilled to the pilot piston. This small orifice restricted the flow to and from the pilot piston thereby giving the pilot check valve a slow response time both in unseating the ball check and also in allowing the ball check to reseat when pressure was removed from the pilot piston. A larger hole, approximately 1/4-inch, was drilled in place of the small orifice which speeded up the response time considerably.

The normal use of a pilot check valve is such that there usually is operating pressure on the restricted flow side of the pilot check valve which would help the spring to seat the ball check. In the normal application of pilot check valves, a certain amount of leakage could occur before the ball check is seated, and it will do little, if any, harm. However, in the usage that the pilot check valves were placed during this test, any leakage past the ball check would go directly to the tank. This leakage past the pilot check valves would cause the pilot signal to become lost entirely by letting it go directly to the tank or it would slow the response time of the pilot check beyond the desired maximum limit. This undesirable leakage was overcome in the pilot check valve by adding a much stronger spring which only resulted in raising the necessary pilot pressure to unseat the ball check valve.

In order to insure that fluid in the drain lines would not return through the pilot check valves and would not go to the pilot side of the pilot operated directional control valves, a check valve was installed on the downstream side of the pilot operated check valve. This not only permitted the pilot operated check valve to dump any signal to the tank but it also eliminated any possibility of the fluid in the drain lines returning back through the pilot check valves and giving any extraneous signal to the directional control valve.

As would be expected any time a new type of valve is built, the original model of the positive-displacement valve had several features which were undesirable. The positive-displacement valve as designed by the author had several features which should be modified in order to improve the characteristics of the valve. One thing that should be checked when using any manufactured cylinder tube is the eccentricity of the tube. If the cylinder tube is out of round, the seals used on the piston will have a very short life and will give considerable amount of trouble during their life. The moving seals used in the positive-displacement valve were low pressure hydraulic seals; therefore their life was naturally short. In order to continue using the positive-displacement valve a source of high pressure seals will have to be found. Another source of trouble which should be corrected was the fact that the 1/4-inch position rods would get bent causing the rod to hang up slightly in the opening and to strip the threads out of the aluminum piston. Most of this trouble could be corrected by making the free floating piston from steel rather than from aluminum.

Originally the positive-displacement valve had an adjustable stop on the inside of the tube. By adjusting the position of the stop, the volume of fluid which would be displaced by each movement of the free floating piston could be varied. Initially this adjustable stop was added in order that the volume could be varied to change the increments by which the slave cylinder would be advanced. After considerable amount of trouble was encountered by the adjusting stop coming loose from the adjusting rod because the retainer rings were of insufficient size, this variable-volume feature was removed from the positive-displacement valve.

This involved removing both the positioning rod and adjusting rod from that side of the free floating piston and inserting a small plug where the adjusting rod had previously gone through cylinder end "B".

One other feature which should be given considerable amount of thought before modifying the positive-displacement valve is a different method of cushioning the free floating piston. This author used rubberized fabric piston cups to provide the necessary cushioning at each end of the stroke. However, in general, this did not prove to be a satisfactory method of performing the necessary cushioning because of the short life of these rubberized fabric cups. One suggested method of providing the necessary cushioning action is by the use of belleville washers on each end of the free floating piston. It is absolutely necessary to provide some means of decelerating the free floating piston rapidly at the end of its stroke which the belleville washer could accomplish. Another method could be the use of a conventional cylinder type cushioning method.

During the course of the experimental investigation of this circuit several of the needle valves were removed from the circuit in order to ascertain just what effects the needle valve had upon the circuit. The circuit seemed to operate just the same when the needle valves were in the circuit or out of the circuit. In this type of circuit the needle valves proved to be unnecessary.

In order to achieve a uniform velocity of the piston in both directions of travel, the single rod slave cylinder should be replaced by a double-rod cylinder. This substitution would give uniform loading to the pump in both directions of travel and would permit higher flow rates to be obtained. Also, the double-rod cylinder should be cushioned on both ends to provide a means for giving a rapid shockless deceleration of the piston when it has reached the end of its stroke.

One of the more serious problems encountered during the experimental investigation of this type is the problem of air in the circuit. Any time air is present in the circuit, the circuit will not give a true indication of its natural characteristics. The positioning of the piston in the slave cylinder will be mushy; that is, the piston will float in the cylinder and will not move a definite amount and then stop as would be the case with no air in the circuit. Any time a line is broken in the circuit air enters the circuit. When air gets into the system, the circuit will not give accurate positions and will be noisy.

In a circuit analysis of this general nature, it would be wise to incorporate into the system a means for isolating each individual circuit from the rest of the circuit. In this way, each circuit will have a separate connection for line pressure and for draining the system directly to the tank. This could be accomplished in the original circuit by adding a line between lines 5 and 8 which contain a gate valve. This new line would necessitate adding another gate valve in line 5 downstream from the tie-in. By closing the gate valve in line 5 and opening the gate valve in the new line, the positive-displacement valve circuit would be bypassed entirely thus directing the flow to the slave cylinder circuit. Then any necessary adjustments could be made in the slave cylinder circuit without disturbing the positive-displacement valve circuit. The positive-displacement circuit could be isolated and run directly to the tank by adding a line between lines 8 and 2 which contain a gate valve and by adding a gate valve in line 8 downstream from this new tie-in. This would permit any necessary adjustments in the positive-displacement valve circuit without disturbing the slave cylinder circuit. No lines would have to be broken which would let air into the circuit when adjusting the individual circuits.

One particular characteristic of the sequence valves that were used in this circuit was that for a given operating pressure there was only one adjustment that could be made to the spring setting of the valve. If too much force was exerted against the spool in the sequence valve, the sequence valve would fail to open before the main relief valve would open. If the spring setting of the sequence valve was set too low, the sequence valve would open prematurely before the slave cylinder reached the end of its stroke. There seemed to be an optimum pressure above the normal operating pressure at which the sequence valve would operate. When working with one set of sequence valves, there were three main pressure levels which must be sufficiently separated to insure three distinct operations. These three levels were the normal operating pressure level, the sequencing pressure level, and the relief valve pressure level. If two sets of sequence valves are used, then there would have to be four distinct pressure levels in the circuit, one for each set of sequence valves, one for the normal operating pressure, and one for the relief valve setting. The commercially available sequence valves seem to be too pressure sensitive to accomplish the purpose of providing pilot pressure signals.

It was finally necessary to omit the positive-displacement valve circuit due to the failure of this investigator to get both the positive-displacement valve circuit and the slave cylinder circuit operating automatically together. After the positive-displacement valve circuit was omitted, it was decided to study the effects of different load conditions on the slave cylinder's ability to continue reversing itself automatically.

The counterbalance valves were adjusted to different pressures in order to impose approximately the same load on the cylinder on both directions of travel of the piston. These different pressures were necessary because of the differences in the net area of the piston available to the fluid pressure.

After the load conditions on the slave cylinder were established, the pump speed was varied over its entire speed range which in turn varied the flow rate through the system as plotted in Fig. 26. Referring to Table I, the average cycle time was the average of the time of one complete cycle of the piston. This cycle time was the time required for the piston to travel to the end of its stroke, reverse itself, return to the other extreme position of its stroke, and reverse itself again. A minimum of five readings for each time was taken at each pump speed. The maximum error in reading the time was plus or minus 2.3 percent from the average reading. The average error in reading the time was plus or minus 0.8 percent from the average reading. In the future to eliminate the human error in determining

TABLE I

OBSERVED DATA

August 10 and 11, 1957

Run No.	Pump Speed	Average Cycle Time	Oil Temp.	System Pressure		Counterbalance Valve Pressure	
				Out	In	#1	#2
	rpm	sec.	°F	psi	psi	psi	psi
1	292	13.62	85	250	300	120	260
2	398	9.85	86	300	300		
3	502	7.84	86	350	375		
4	599	6.59	89	350	375		
5	698	5.56	90	375	360		
6	798	4.91	91	400	400		
7	899	4.39	93	450	450		
8	1003	3.92	95	500	500		
9	1101	3.58	98	520	550		
10	1199	3.29	101	550	600		
11	1300*	*	*	*	*	120	260
12	289	14.57	89	430	560	300	380
13	495	8.18	90	480	625		
14	700	5.71	92	550	700		
15	899	4.43	95	575	750		
16	1002	4.03	99	620	800	300	380
17	288	14.95	91	450	630	350	450
18	500	8.10	92	500	650		
22	582	6.90	85	520	720		
19	700	5.74	93	550	720		
23	802	5.05	88	530	730		
20	903	4.43	97	600	800		
21	997	4.06	92	600	800		
24	1100*	*	*	*	*	350	450
25	287	15.13	95	530	670	390	490
26	503	8.09	95	520	700		
27	702	5.78	95	600	780		
28	808	5.11	96	630	620		
29	902	4.51	100	620	820	390	490

*The runs at these conditions were unstable; therefore, no readings were obtained.

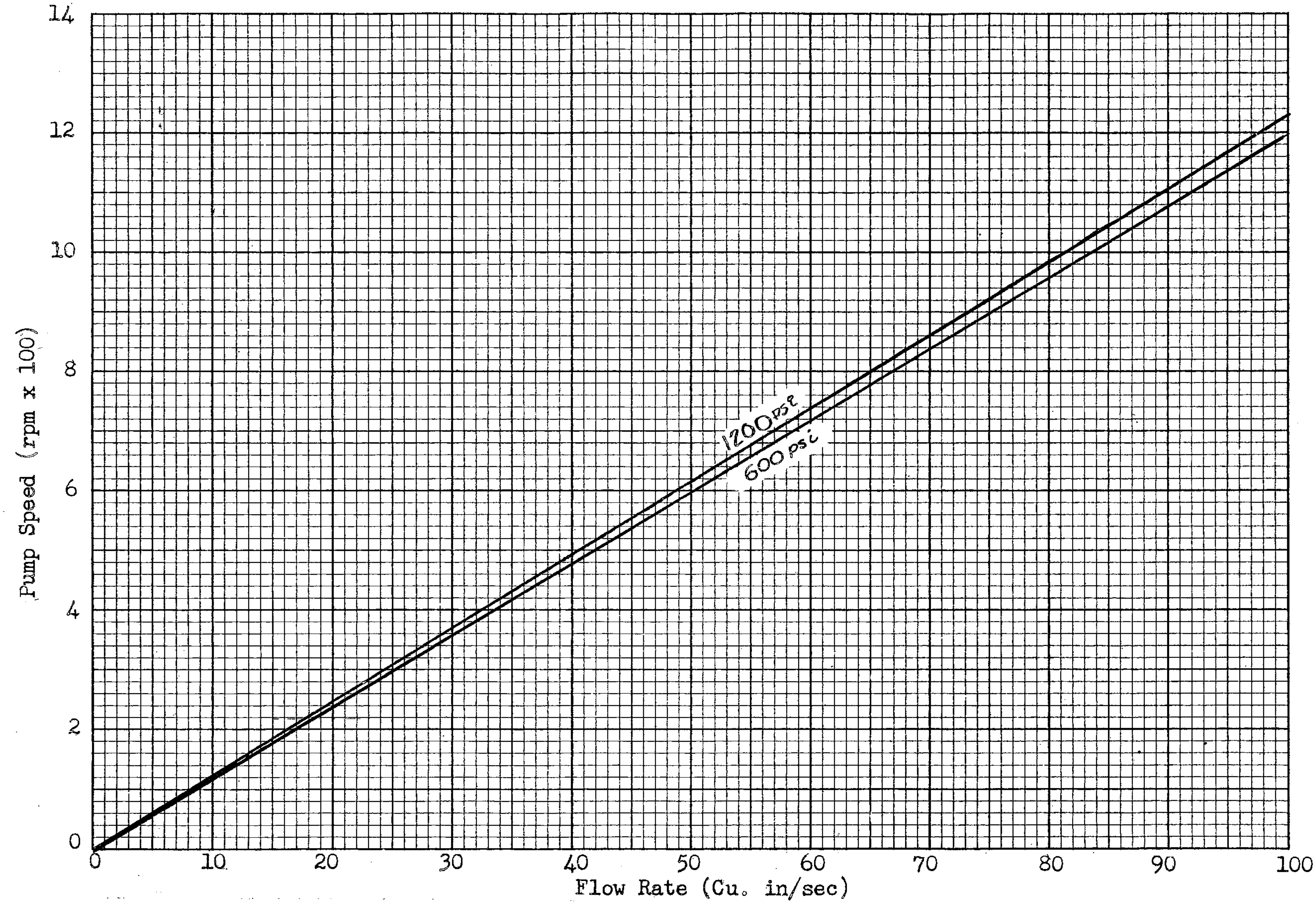


Fig. 26. Performance Curves for Commercial Gear Pump.

cycle time, an electric eye should be installed into the timing circuit.

The reading for oil temperature was the temperature of the reservoir oil at the end of each run. As noted from Table I, these readings varied between 90° and 100°F which were fairly constant. This constant temperature caused the physical properties of the oil to remain constant.

The system pressure readings were the readings of the system pressure when the piston was going out to the end of the stroke and the system pressure when the piston was going into the end of the stroke. These readings were taken from a pressure gage and probably were not closer than plus or minus 10 percent. Either a calibrated strain gage or a calibrated pressure pick-up should be used to accurately determine the system pressure.

The load conditions of the slave cylinder were changed and again the average cycle time was determined at the various flow rates. A total of four different sets of load contitions was established for the slave cylinder. In Fig. 27, the cycle time for the slave cylinder versus the pump speed was plotted. At all pump speeds with the exception of the low speed, 290 rpm, all the points fell on or near the curve represented by the following equation:

$$t = 4000\ N^{-1}$$

where: t is cycle time in seconds, and
N is the pump speed in rpm.

Also on the same figure, the curve for the theoretical cycle time for the slave cylinder was plotted assuming an instantaneous reversal of the slave cylinder. The equation representing this theoretical cycle time can be written as follows:

$$t = 3670 N^{-1}$$

From these two curves, one can draw the conclusions that the effects of load conditions upon the automatic reversal of the slave cylinder is negligible. When the pump was rotating at 800 rpm and when the sequence pressure was 1000 psi, the actual cycle time of the slave cylinder was 5 seconds and the theoretical cycle time of the slave cylinder was 4.588 seconds. This leaves 0.412 seconds to be accounted for. From the equation developed by Ernst (4), the pressure build up time for sequencing operation on the blank end of the cylinder would be 0.023 seconds and on the rod end of the cylinder would be 0.020 seconds. The flow time necessary to shift the directional control valve at the above conditions would be 0.031 seconds for each reversal operation. Since there were two reversal operations per cycle, the total accountable time would be 0.105 seconds which would leave 0.307 seconds per cycle or approximately 0.150 seconds per reversal operation unaccountable.

A dual beam oscilloscope was used in conjunction with a crystal type pressure pick-up to enable this investigator to study the transient pressure conditions of the fluid during sequencing operations. However, no method was available to trigger the oscilloscope in order to obtain pictures of this transient pressure condition. Due to the short duration sequencing time, it was impossible for this writer to do any more than merely observe the pressure build up characteristics of the fluid. In addition, there was no calibration curve available for the crystal pick-up so absolute values for sequencing pressures could not be obtained.

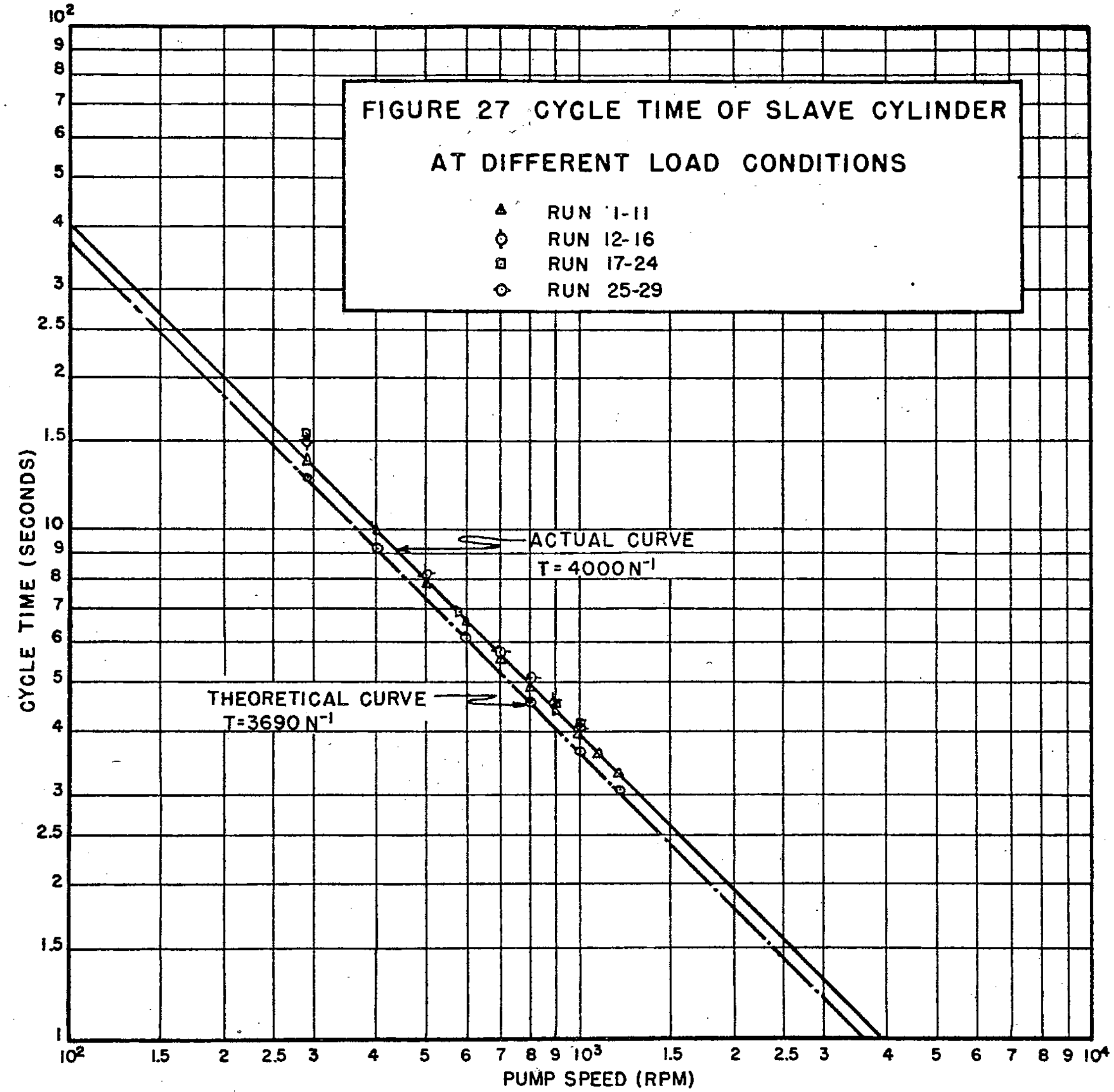


FIGURE 27 CYCLE TIME OF SLAVE CYLINDER
AT DIFFERENT LOAD CONDITIONS
RUN 1-11
RUN 12-16
RUN 17-24
RUN 25-29
ACTUAL CURVE
T=4000N⁻¹
THEORETICAL CURVE
T=3690 N⁻¹
CYCLE TIME (SECONDS)
PUMP SPEED (RPM)

CHAPTER VII

SUMMARY

The objective of this investigation was twofold. The first objective was to determine whether it was possible to accurately position a slave cylinder some place between the limits of travel of the piston by using intermittent feeding of the cylinder and by employing commercially available components in an open circuit. The second objective of this investigation was to determine whether or not it was feasible to cause the reversal of a cylinder by pressure sequencing at the end of the piston stroke and create a signal that would be sent to a directional control valve. This signal would cause the direction of flow through the directional control valve to be reversed and thus would cause the cylinder to reverse its direction of travel.

The circuit as a whole appears able to satisfy the first part of this investigation. However, due to similarities of the pressure ranges of the sequence valves, this investigator could not obtain two distinct levels of signals for the sequencing operations. Either a modification of the existing sequence valves would be required or a new sequence valve should be designed that would give the different pressure levels necessary for distinct operations of the system as a whole.

The second part of the investigation was definitely satisfactory to this author. Pressure sequencing is a good means for causing the reversal of a cylinder. One thing which was noted from this investigation was the fact that the load conditions on the slave cylinder had little, if any, effect upon the ability of the pressure sequence valve to cause the reversal of the slave cylinder. When more than one set of sequencing valves is used in a circuit, each set of pressure sequence valves must have a separate and distinct pressure level of operation which is higher than the normal operating pressure of the system and yet lower than the relief valve pressure.

CHAPTER VIII

RECOMMENDATIONS FOR FUTURE RESEARCH

From the evidence revealed during this investigation it is the belief of this writer that there should be continued research collinear with these objectives. There are several ramifications which should be investigated before this method of positioning is either released for commercial development or abandoned as too cumbersome. These ideas are discussed in the order in which this author believes they should be investigated.

The first effort should be focused toward developing a new and different type of sequence valve. This valve could be similar to the one sketched by Hadekel (6) in his book. From the schematic representation of this valve as shown in Fig. 28, it can be seen that the use of this type of valve would eliminate several components from the original circuit. It would eliminate not only the commercial sequence valve used in the circuit, but also it would eliminate the pilot check valve, the needle valve, and the interconnecting lines between these components. It should speed up the operational time of this circuit since the signal to the other side of the pilot directional control valve is already tanked and

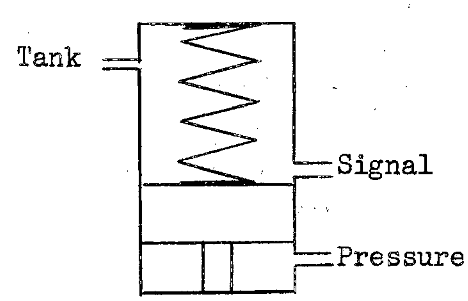


Fig. 28. Sequence Valve

it would not be necessary to wait until that signal is tanked before the spool in the directional control valve is shifted thereby reversing the direction of flow through the valve. The sequence valves which are available commercially in the United States have characteristics more desirable for power hydraulics rather than control hydraulics. It also appears that this new signal valve should be less sensitive to pressure variations than are the commercially available sequence valves. It is believed that the signal valve would have a much wider range of sequencing pressures which would be highly desirable.

After a new signal valve has been designed and built, the whole system as discussed in this investigation should be reconnected and the experimental work repeated.

The second effort should be an investigation of the necessary sequencing pressure level above operating conditions which is necessary to insure efficient operation of a pilot signal circuit. It is extremely important to know just what will be the necessary sequencing pressure above operating pressure for a given flow rate and pressure. A thorough analysis would be required of the characteristics of commercially available sequence valves as well as the new signal valve discussed previously in this chapter. After this experimental study was completed, an analytical study could be undertaken. From the combination of the experimental investigation and the analytical study, it should then be possible to predict a reasonable sequencing pressure necessary to insure efficient and dependable operation of a pilot valve.

Another effort on this line of thinking should be a complete experimental study of the effects of one set of sequence valves upon another set of sequence valves in order to determine just what pressure difference is required for each set of valves. After this information is obtained an analytical study should be made using at least three sets of sequence valves. A correlating experimental study should be undertaken to obtain information that will assist designers to estimate the cycle time for a given circuit.

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APPENDIX A

APPARATUS AND EQUIPMENT

Pump Circuit:

1. Fairbanks Morse Pump and Electric Motor Unit: rated, 1750 rpm; figure no. 5500; serial number, 2926; pump suction 1 1/4 inches; pump discharge, 1 inch.

2. 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.

3. 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.

4. 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.

5. Reservoir; manufactured by Mechanical Engineering Laboratory; size 30 inches by 34 inches by 17 1/4 inches; capacity, approximately 75 gallons; compartments, 4.

6. Marvel Sump Type Filter: two required; model no. C-1-10; capacity 10 gallons per minute.

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

8. Gresen Relief Valve: model no. J-50; port size, 1/2 inch; pressure range 500 to 1500 psi.

9. Texaco Regal Hydraulic Oil: 65 gallons; type AZ RO; viscosity range 140 to 150 Saybolt Universal Seconds at 100°F.

Positive Displacement Valve Circuit:

1. Directional Control Valve, (*D-2): Manufacturer, Republic; model no. 8141-1/4 H; type: manually operated, four-way, three-position, all ports blocked in the neutral position; port size, 1/4 inch.

2. Directional Control Valve, (*D-4): 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.

3. Directional Control Valve, (*D-5): 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.

4. Positive-Displacement Valve: Designed by this investigator and build by Research Apparatus Development Laboratory, College of Engineering, Oklahoma State University; port size, 1 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.

6. Pilot Check Valve, (*D-13 and D-14): Manufacturer, Fluid Controls; model no. 25200-2; port size, 1/4 inch.

*Indicates the number assigned to this valve in Fig. 23.

7. Needle Valve, (*D-11 and D-12): Manufacturer, P & C Co.: model no. 1040A; port size, 1/4 inch.

8. Check Valve, (*D-19 and D-20): Manufacturer, unknown; model no. 527-6D; port size, 3/8 male tube connection.

9. Accumulator: Manufacturer, Greer; model no. 30A-1A; capacity, 1 gallon; type, hydro-pneumatic, bag; port size, 1 1/4 inch.

Slave Cylinder Circuit:

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

2. Directional Control Valve, (*D-6): Manufacturer, Logansport Machine Co; model no. 4095A 3/4; type: manually operated, four-way, three-position all ports blocked in neutral position; port size, 3/4 inch.

3. Directional Control Valve, (*D-8, D-9, and D-10): 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. Counterbalance Valve, (*C-1 and C-2): 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.

5. Sequence Valve, (*S-3 and S-4): 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. 23.

6. Pilot Check Valve, (*D-15 and D-16): Manufacturer, Fluid Controls; model no. 25200-2; port size, 1/4 inch.

7. Needle Valve, (*D-17 and D-18): Manufacturer, P & C Co.; model no. 1040A; port size, 1/4 inch.

8. Check Valve, (*D-21 and D-22): Manufacturer, unknown; model no. 527-6D; port size, 3/8 inch male tube connections.

9. Cylinder: Manufacturer, Logansport Machine Co.; model no. 11020; size, 3 inch bore and 24 inch stroke; type, non-cushioned; maximum operating pressure, 1500 psi; port size, 3/4 inch.

10. Directional Control Valve, (*D-1): Manufacturer, Denison Engineering Co.; model no DD-011-358-CK; type, solenoid operated, four-way, three-position, spring centered, pressure port blocked and all other ports intra-connected in the neutral position; port size, 1/4 inch.

Instruments:

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

2. Pressure Gauge: Manufacturer, Crosby; pressure range, 0 to 3000 psi.

3. Pressure Gauge: Manufacturer, Ashcroft; pressure range, 0 to 2000 psi.

4. Pressure Gauge: Manufacturer, Ashcroft; pressure range, 0 to 5000 psi.

5. Pressure Gauge: Manufacturer, Marsh; pressure range, 0 to 1000 psi.

6. Oscillograph: Manufacturer, Allen B. DuMont Lab.; model, dual-beam cathode-ray; type, 322; serial no. 9178.

7. Audio Oscillator: Manufacturer, Hewlett-Packard Co.; model, 200B; Serial 41096.

8. Pressure Pickup: Manufacturer, Commercial Research Laboratory; model, Cox quartz pressure element, type 3; serial no. 1328.

*Indicates the number assigned to this valve in Fig. 23.

APPENDIX B

SAMPLE CALCULATIONS

To determine the time required to shift the spool of a directional control valve, one must know the time that fluid takes to build up the necessary sequencing pressure and the time that fluid takes to flow through the sequence valve to the directional control valve. If the flow rate is 65 cubic inches per second, and if the sequence pressure is 1000 psi, the solution to the above problem would be as follows:

1. The time required to build up to the sequence pressure of 1000 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 system pressure (psi)

At 1000 psi pressure and 100°F, $A = 4.38 \times 10^{-6}$ and $B = 5.65 \times 10^{-11}$.

$$C = \frac{4.38 \times 10^{-6} - 2(5.65 \times 10^{-11})1000}{1 + 4.38 \times 10^{-6} - 5.65 \times 10^{-11}(1000)^2}$$

$$C = 4.267 \times 10^{-6}$$

V_1 = Volume of related lines = 174 in.^{3}

V_2 = Volume of blank end of cylinder = 170 in.^{3}

V_b = Total volume = 344 in.^{3}

V_1 = Volume of related lines = 174 in.^{3}

V_3 = Volume of rod end of cylinder = 134 in.^{3}

V_r = Total volume = 308 in.^{3}

Pressure build up time blank end of cylinder

$$t = \frac{4.267 \times 10^{-6} \times 344 \times 1000}{65} = 0.023 \text{ seconds}$$

Pressure build up time, rod end of cylinder

$$t = \frac{4.267 \times 10^{-6} \times 308 \times 1000}{65} = 0.020$$

Total pressure build up time per cycle

$$t = 0.023 + 0.020 = 0.046 \text{ seconds}$$

2. Determine the time required to shift the valve after pressure is built up. Assume 4.0 cubic inches of line volume and spool volume per cycle.

$$t = \frac{V}{Q} = \frac{4.0}{65} = 0.062 \text{ seconds}$$

VITA

Alva Gene Comer

Candidate for the Degree of

Master of Science

Thesis: AN INVESTIGATION OF INTERMITTENT FEEDING OF A HYDRAULIC CYLINDER

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