THE POSITIONING OF A HYDRAULIC SLAVE CYLINDER WITHOUT FEEDBACK

By

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PREFACE

The modern industrial trend of automation in industry has imposed an urgent demand for the development of more powerful, accurate, and automatic positioning devices. Although there are many well known mechanical and electrical methods of positioning, hydraulic positioning devices are becoming increasingly more popular with machine designers. This recent increase in the approbation of oil hydraulic power is due to the fact that a confined fluid is one of the most versatile means of modifying motion and transmitting power known today. This investigation is a result of searching for a new, inexpensive, and versatile method of positioning a hydraulic slave cylinder.

My deepest appreciation is due Professor E. C. Fitch for his invaluable assistance and competent counsel throughout the course of this study. I wish to thank Dr. Clark Dunn and Dr. J. H. Boggs for making this research project possible. My sincere gratitude is extended to Professor B. S. Davenport, Mr. John McCandless and Mr. George Cooper for their aid in the construction of the testing apparatus. I am indebted to Professor C. M. Leonard for his suggestions and advice in the writing of this thesis. I am particularly grateful to my wife, Joan, for her encouragement and capable services, as typist, during the formulation of this thesis.

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CHAPTER I

INTRODUCTION

Positioning devices have played an important role in the great industrial growth of the United States. Practically every machine manufactured is dependent upon some form of positioning device for the accurate performance of its prescribed duties. Thus, as the modern industrial trend of automation is reflected by a demand for larger, more precise, and more automatic machines, there is a corresponding demand for more powerful, accurate, and automatic positioning devices.

Although there are many well known mechanical and electrical methods of positioning, it is not surprising that hydraulic positioning devices are becoming more and more popular with machine designers. This rather recent increase in the popularity of oil hydraulic power is largely due to the fact that a confined fluid is one of the most versatile means of modifying motion and transmitting power known today. Vickers (9) states that "No other medium combines the same degree of positiveness, accuracy, and flexibility, maintaining the ability to transmit a maximum of power in a minimum of bulk and weight."

A great many methods have been proposed for the positioning of hydraulic cylinders. Some of these methods are limited in their capabilities and provide only a single positioning station in each direction of the slave cylinder's travel. A physical rearrangement of the limiting devices on the slave cylinder is required in order to change the positioning stations of the cylinder. Other methods which have been proposed are more versatile because they provide an infinite number of possible positioning stations with one physical arrangement of the components. In these methods, the positioning station is determined by merely selecting a setting on a control which can be remote from the slave cylinder. The latter method of positioning is often referred to as a hydraulic servo system and it requires some form of position feedback which transmits information about the location of the slave cylinder rod to the directional control valve. Although the hydraulic servo system is very versatile it is also quite expensive because it usually requires a fine precision control valve and sometimes necessitates an elaborate feedback mechanism.

A. G. Comer (1) has suggested that a hydraulic slave cylinder could be positioned accurately by metering a fixed amount of fluid to the cylinder. This could be accomplished by the use of a positive displacement fluid metering device. A positioning circuit of this type could approach the versatility of the hydraulic servo system and yet would be less expensive since it would not require a position feedback mechanism or a precision control valve.

If such a circuit could be developed, it would have almost unlimited application in industry and would be particularly valuable in installations where environmental conditions necessitate an awkward feedback system. For example, a hydraulic intermittent feeding circuit could be applied as the positioning device for: (1) the plow blade on farm tractors, (2) the probe in atomic reactors, (3) the parts being machined on automatic machine tools, and (4) the flaps on an airplane.

This investigation was inspired by an industrial demand for the development of a practical hydraulic positioning circuit similar to the one described by Comer.

CHAPTER II

PREVIOUS INVESTIGATIONS

There are a great many different methods which are used to position slave cylinders. One of the more primitive methods was reported in 1932, when C. Morey (7) produced intermittent displacement of a cylinder using a rachet-and-pawl device. Another method which has been used to position slave cylinders consists of connecting two double-acting cylinders in series and forming a closed loop. One cylinder serves as a slave cylinder while the other serves as a variable displacement pump. Although this arrangement does provide positive positioning of the slave cylinder, the disadvantage is that the force available at the slave cylinder is limited by the amount of force which can be applied to the rod of the pump cylinder.

J. C. Cotner (3) reported positioning a slave cylinder using pilot-size, cam-operated valves and pilot-operated control valves. The cam-operated valves were located in such a manner that they were shifted by a dog attached to the rod of the cylinder thereby directing the flow of fluid through the cam valve to the appropriate port of a pilot-operated directional control valve. Another similar method of positioning a slave cylinder is accomplished by the use of limit switches and solenoid-operated control valves. As in the above example, a dog, attached to the rod of the cylinder, trips the limit mechanism which is, in this case, an electrical switch. Electrical current is directed in this manner to the appropriate connection on the directional control valve. The disadvantage of the preceding examples is that their positioning capabilities are limited to one setting in each direction.

The use of electro-hydraulic servo valves provides one of the most versatile methods of positioning a slave cylinder. This method has almost unlimited positioning flexibility and has the added advantage of requiring only a minimum of bulk and weight. The servo valve arrangement is frequently employed in the aircraft industry as a control surface actuating device for airplanes and guided missiles. The requirement of fine manufacturing precision and the incorporation of an elaborate feedback device, however, make this method very expensive.

R. Kurzweil (5) and H. L. Stewart (8) have reported the sequencial extension or retraction of several cylinders using pressure sequence valves to divert main-line fluid flow. Fluid was directed by a control valve to the first actuator of a succession. After this actuator had completed its stroke, a pressure rise developed and the fluid was then diverted, by means of a pressure sequence valve, to the next consecutive actuator. The positioning of the slave cylinders in this case was limited to either a "bottom" position or a position where an external resistance sufficient to open the sequence valves was encountered.

H. G. Conway (2), an English engineer, reported the positioning of a slave cylinder using a pilot-operated directional control valve containing a mechanical detent device ("stay-put" or holding device). Fluid was directed through the directional control valve to one side of the cylinder and the appropriate pilot port of the directional control valve. Flow continued in this manner until the cylinder reached the "bottom" position, at which time a sufficient pressure level was attained to overcome the detent mechanism on the valve spool and the valve was shifted. Another English engineer, R. Hadekel (4), reported a similar scheme for the positioning of a slave cylinder. Hadekel utilized a control valve configuration which essentially consisted of two pressure sequence valves incorporated within the design of a pilot-operated directional control valve. As in the above example, the directional control valve operation was governed by the pressure level. In this case, however, the directional control valve was shifted at the time sufficient pressure was generated to open the sequence valve, while, in Conway's method, control was exercised when the pressure level exceeded the resistance imposed by the mechanical detent device. Neither of the composite directional control valves used by Hadekel or Conway is commercially available in the United States.

W. R. Matthews (6) described a cylinder reversing circuit using commercially available pressure sequence valves, pilot-operated direction control valves, and pilot-operated check valves. Fluid was diverted through the directional control valve to one side of the slave cylinder. Flow continued in this fashion until sufficient pressure was generated to open a pressure sequence valve which in turn directed the fluid flow or pilot signal to the appropriate pilot port of the directional control valve thereby shifting the valve spool. Flow through the directional control valve was then reversed,and the cylinder moved in the opposite direction until such time as it reached the end of its stroke when the cylinder was reversed in the same manner as outlined above. Check valves were used to "lock in" the pilot signal acting on one side of the directional control valve spool while pilot-operated check valves were used to release this signal when the opposite signal was received. This arrangement of check valves and pilot-operated check valves prevented any premature drifting of the directional control valve spool.

A. G. Comer (1) reported an effort to obtain intermittent displacement of a slave cylinder using a combination of two cylinder reversing circuits. Each reversing circuit consisted of commercially available components and was similar to the one described by Matthews in the previous example. Exhaust or discharge fluid from one cylinder-reversing circuit was utilized as the fluid source of the second reversing circuit. Intermittent hydraulic feeding of the second cylinder or slave cylinder was accomplished in this manner. Comer stated, however, that he was unable to maintain continuous operation of both cylinder-reversing circuits because the operating pressure of the commercially available pressure sequence valves was not adjustable over a wide enough range to accommodate two distinct pressure levels.

CHAPTER III

STATEMENT OF PROBLEM

The objectives of this investigation were to construct and analyze an experimental hydraulic positioning circuit for a slave cylinder which did not require feedback. The circuit was intended to be of practical value in that the slave cylinder was required to perform while being subjected to loads very similar to what might be expected in actual working conditions. Little effort was made, however, to incorporate the features of several basic hydraulic components into one composite design as is frequently done in a practical industrial installation. This separation of basic components facilitated the study of each portion of the circuit. The control of the metering fluid to the automatically-operated slave cylinder was accomplished by the use of pressure-generated pilot signals. Since pressure variations were utilized in the operation of the circuit, pressure profiles were obtained at key points and used as a basis for the circuit analysis.

CHAPTER IV

COMPONENT SELECTION AND DEVELOPMENT

Since the design of the positioning circuit did not include any consideration for time-displacement performance of the slave cylinder, the size selection of the components was determined principally by the size of stock components available in the Oklahoma State University fluid power laboratory. Standard commercially-available hydraulic components were used in the circuit design for this investigation wherever possible. A description of each of the standard components used in the circuit and its corresponding A.S.A. symbolic representation is listed in Appendix A. In some cases, it was necessary to develop specially designed hydraulic components to perform functions that could not be fulfilled by standard components. The following material is a description of the design and operation of these special hydraulic components.

Signal Valve

The signal valve is essentially a pilot-operated, spring-offset, three-way, two-position, directional control valve which contains a provision for varying the pilot pressure required to shift the valve spool. An assembly drawing of the signal valve is shown in Fig. 1. When the valve is in the spring-offset position, free flow is allowed from port A through the valve to the tank port while the pressure port is blocked. Flow continues in this manner until sufficient pilot pressure is attained

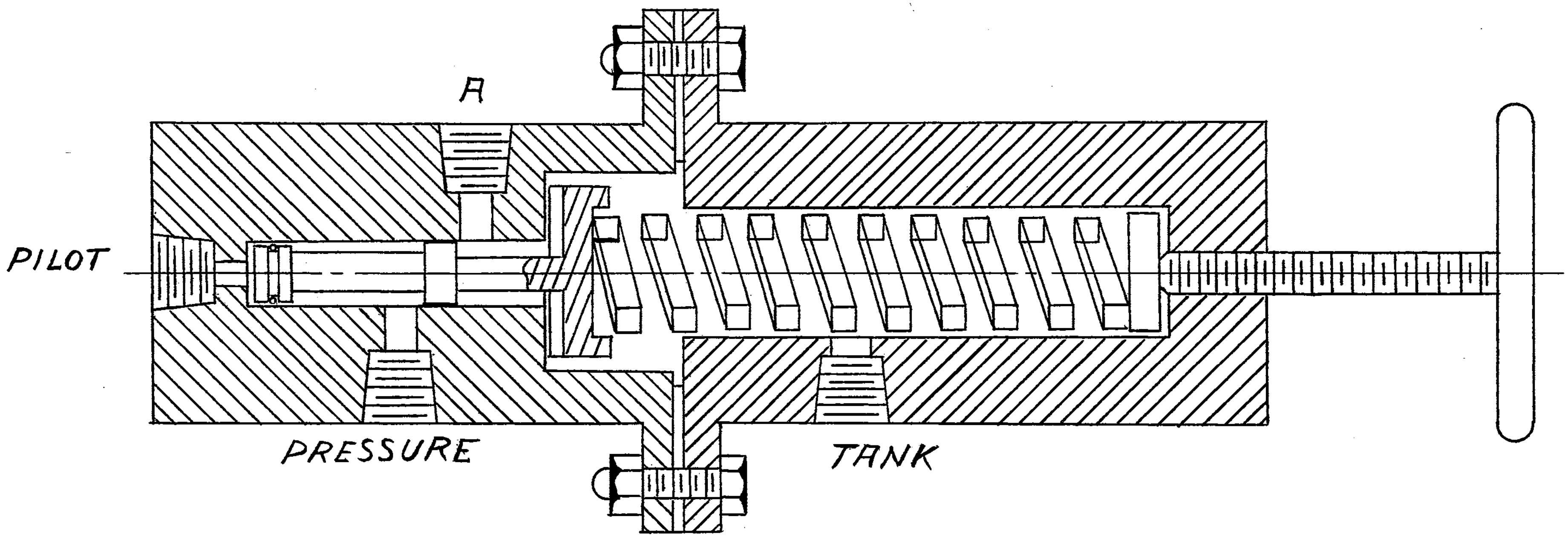


Fig. 1. Signal Valve Assembly Drawing.

to overcome the force imposed upon the spool by the spring at which time the spool shifts and pressure is allowed to flow through the valve to port A and the tank port is blocked. It should be noted that the tank port also serves as an internal drain for the valve. The pilot pressure required for the operation of the valve is adjustable from 500 to 1,250 psi. The entire valve was designed by the investigator and built by the Research Apparatus Development Laboratory, College of Engineering, Oklahoma State University. Figure 2 is the symbolic representation for the hydraulic signal valve.

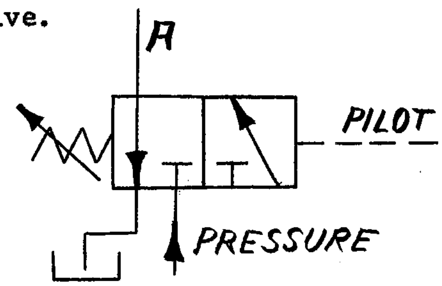


Fig. 2. Signal Valve.

Pilot-Operated Directional Control Valve with Hydraulic Detent

The addition of a hydraulic detent device to a standard pilot-operated, four-way, two-position directional control valve was accomplished by a mere design modification of the valve spool and end caps. An assembly drawing of this valve after design modifications is shown in Fig. 3. The valve was modified to provide two different areas, pilot and detent, upon which fluid pressure could be applied to control the position of the valve spool. The pilot area is approximately ten times greater than the detent area. Figure 4 is the symbolic representation for the pilot-operated, hydraulic detent, four-way, two-position, directional control valve as it was used in this investigation. The valve spool is retained in either of its two extreme positions by the

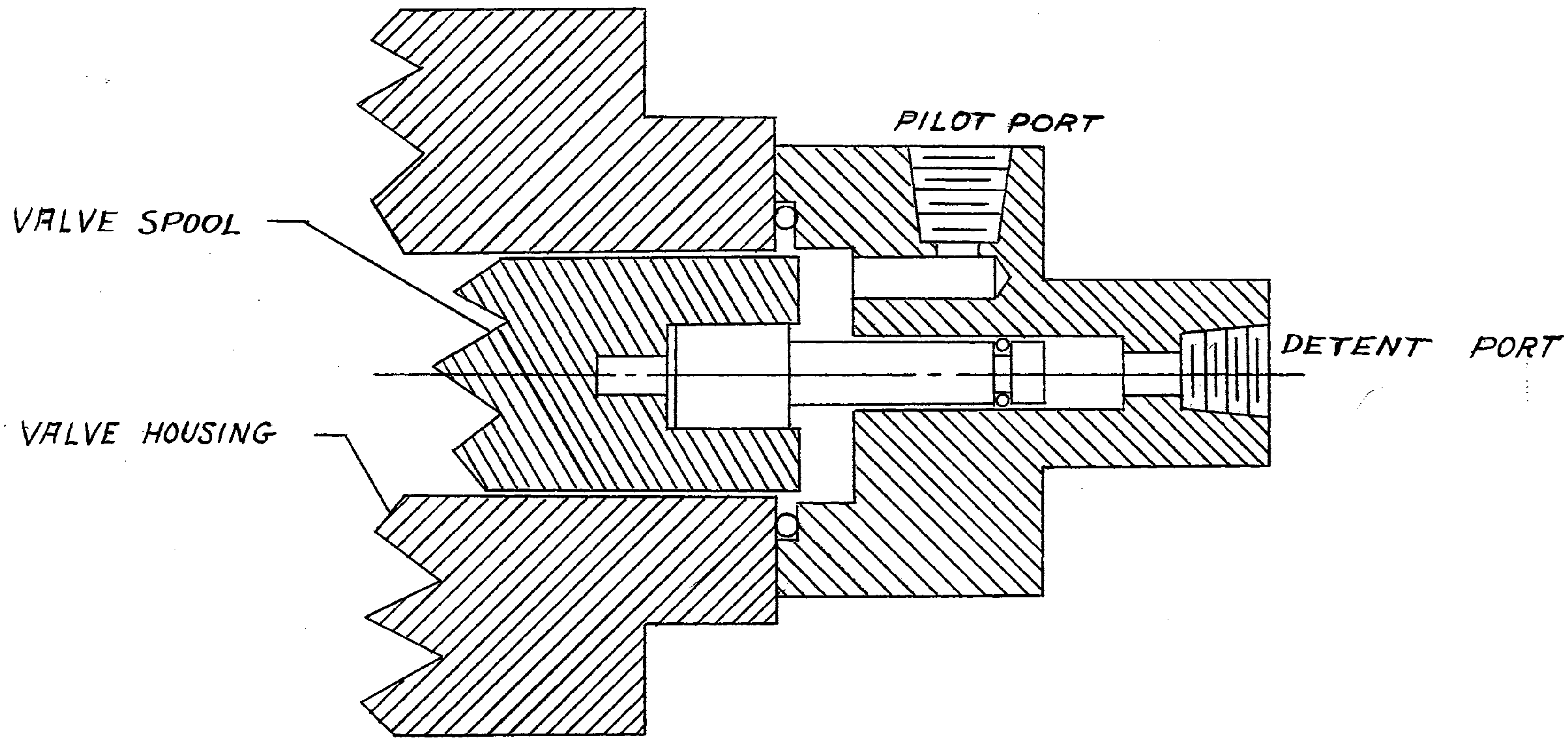


Fig. 3. Pilot-Operated Directional Control Valve with Hydraulic Detent.

application of main-line fluid pressure on the appropriate detent area. When signal fluid is permitted to act upon the pilot area which opposes the active detent area, the ensuing unbalanced force shifts the valve spool.

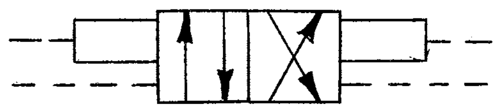


Fig. 4. A.S.A. Symbol for Pilot-Operated Directional Control Valve with Hydraulic Detent.

Cylinder with Hydraulic Cushion

Hydraulic cushioning was provided on a standard Boeing canopy actuating cylinder by the addition of two press-fitted sleeves to the piston rod. (See Fig. 6.) Figure 5 is the symbolic representation for a double acting, double-rod, cushioned type, hydraulic cylinder used in this investigation.

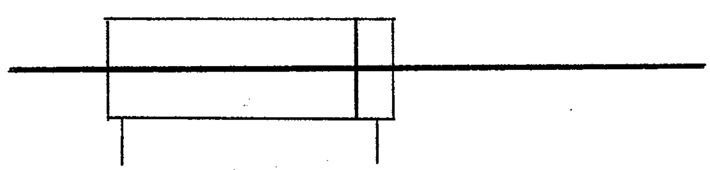


Fig. 5. A.S.A. Symbol for Double-Acting, Double-Rod, Cushioned Type, Hydraulic Cylinder.

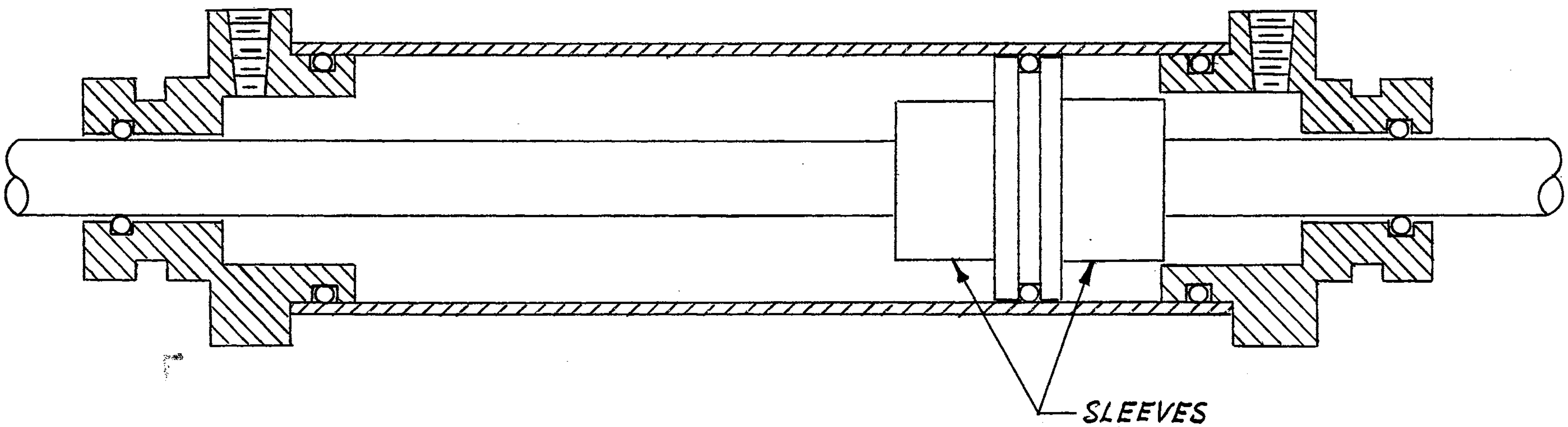


Fig. 6. Double-Acting, Double-Rod, Cushioned Type, Hydraulic Cylinder.

CHAPTER V

CIRCUIT DESIGN

The positioning circuit used in this investigation was a modified form of the hydraulic intermittent feeding circuit described by A. G. Comer (1). An analysis of the complete positioning circuit design can best be discussed by separating the integrated circuit into three major divisions: (1) the fluid power source; (2) the positive-displacement valve circuit; and (3) the slave cylinder circuit.

A description of the fluid power source and its related equipment is presented in Appendix B, Fluid Power Test Stand Facilities. A graphical representation of the complete positioning circuit is illustrated in Fig. 7, and the physical arrangement of this system is presented in Fig. 8.

Positive-Displacement Valve Circuit

Main-line pressure was unloaded manually into the reservoir by by-pass valve D-2 when it was desired to "idle" the circuit. Selective operation of the positive-displacement valve circuit was accomplished by closing gate valves G-2 and G-3 and opening gate valves G-1 and G-4. Main-line pressure was directed through directional control valve D-4 to the left port of the positive-displacement valve. Flow continued in this fashion until the piston of the positive-displacement valve "bottomed out" in an extreme right position at which time sufficient pressure was

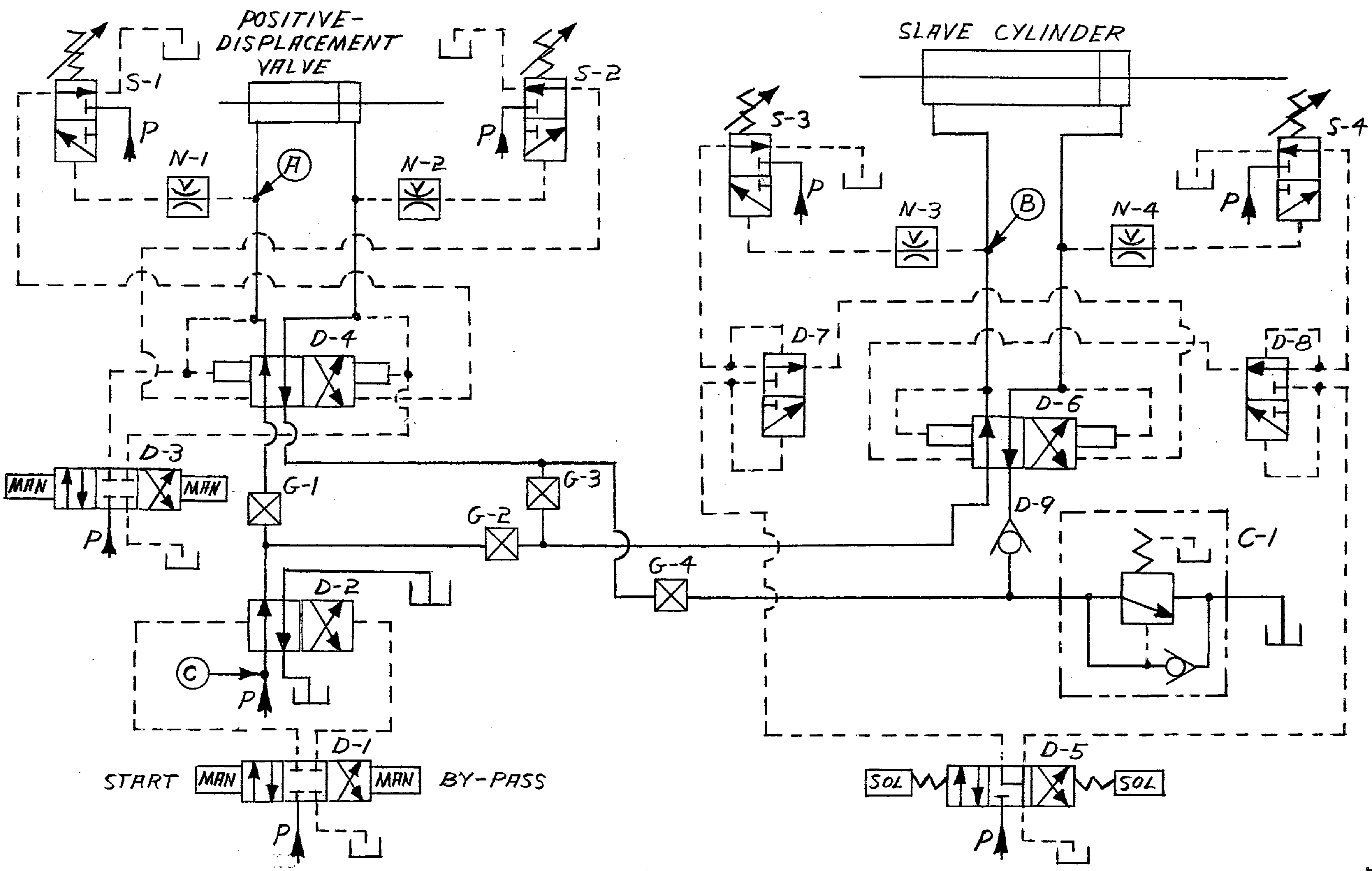


Fig. 7. Graphical Representation of the Complete Positioning Circuit.

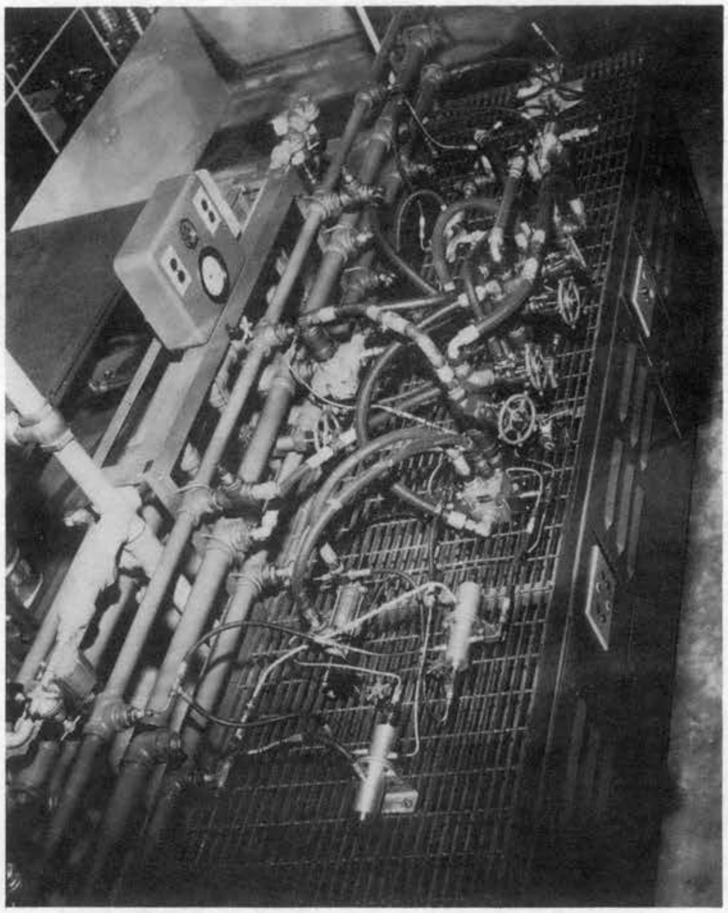


Fig. 8. Physical Arrangement of the
Complete Positioning Circuit

generated to overcome the spring setting of signal valve S-1. Pilot flow was then allowed to pass through signal valve S-1 until it shifted the directional control valve D-4. The direction of travel of the positive-displacement valve was then reversed, main line pressure was relieved, and signal valve S-1 returned to the spring-offset position. The return half of the positive-displacement valve cycle was achieved in a similar manner by the use of signal valve S-2.

Automatic and continuous cycling of the positive-displacement valve was effected in the above manner and constant volume increments of fluid were metered through directional control valve D-4, gate valve G-4, and counterbalance valve C-1. The pilot-size directional control valve D-3 was used only to provide a quick and convenient means for relocating the spool of valve D-4 in the event it should "hang up" in the "blocked" intermediate position.

Slave Cylinder Circuit

Selective operation of the slave cylinder circuit was achieved by closing gate valves G-1, G-3, and G-4 and opening gate valve G-2. Automatic and continuous cycling of the slave cylinder between its two extreme positions was furnished by a circuit design similar to the one described in the positive-displacement valve circuit. The pilot-size, solenoid-operated, directional control valve D-5 and shuttle valves D-7 and D-8 were incorporated in the circuit to provide selective directional control of the slave cylinder during its intermediate range of travel.

The Integrated Positioning Circuit

The coordination of the positive-displacement valve circuit and the slave cylinder circuit was accomplished by closing gate valves G-2 and G-4 and opening gate valves G-1 and G-3. This unification of the two component circuits resulted in a cyclic incremental advance of the slave cylinder.

CHAPTER VI

TESTING PROCEDURES

A routine safety check of the entire system was required before any test was started in order to make certain that there were no disconnected or open hydraulic lines and that adequate clearance space was provided for the reciprocating cylinders. After this was accomplished, the fluid power source was prepared for circuit operation. (See Appendix B.) The reservoir fluid level was checked, the heat exchanger was put into operation, the main pump was started, and the relief valve was adjusted to the desired pressure setting.

Upon completion of the preparation of the fluid power source, the positive-displacement valve circuit and the slave-cylinder circuit were isolated and adjusted separately to provide continuous and automatic cycling. Care was taken, in each circuit, to regulate both signal valves for the same operating characteristics. The slave cylinder circuit was adjusted to obtain smooth and cyclic operation with a minimum of pressure rise while the positive-displacement valve circuit was adjusted to obtain smooth and cyclic operation with a maximum of pressure rise. It was hoped that by thus spreading the operating pressures, the combined system could then maintain two distinct levels of control.

Next, the two component circuits were combined to form the integrated positioning circuit, and pressure profiles were obtained at points A, B, and C by the use of a capacitance-type pressure pickup and an oscilloscope. (See Fig. 7.) It was found that although the sweep of the oscilloscope could by synchronized to the low cyclic frequency of the system, the retention time of the screen was not sufficient to sustain a continuous trace of the pressure variation. Therefore, in order to obtain a plot of the pressure profiles, it was necessary to perform a slow and tedious point-by-point transcription from the moving trace on the oscilloscope screen to a sheet of graph paper. Also, it was observed that the pressure pickup and its associated apparatus displayed a very noticeable tendency to drift out of calibration during the test. It was, therefore, necessary to frequently calibrate the pressure pickup device while obtaining each pressure profile. Even then, it was estimated that the accuracy of the plots, thus obtained, was limited to $\pm$ 100 psi. (See Figs. 9, 10, and 11.)

CHAPTER VII

ANALYSIS OF EXPERIMENTAL CIRCUIT

Although it was observed that the operation of the integrated positioning circuit did provide the desired incremental displacement of the slave cylinder, it was noticed that the performance of the circuit was not dependable. It then became apparent that the potential accomplishment of this positioning circuit was restrained by inconsistent control and that any further development of the circuit was contingent upon a thorough comprehension of the control devices employed in this investigation. It was, therefore, concluded by the author that the entire analysis should be devoted to a fundamental understanding of the basic control mechanisms used in the experimental circuit.

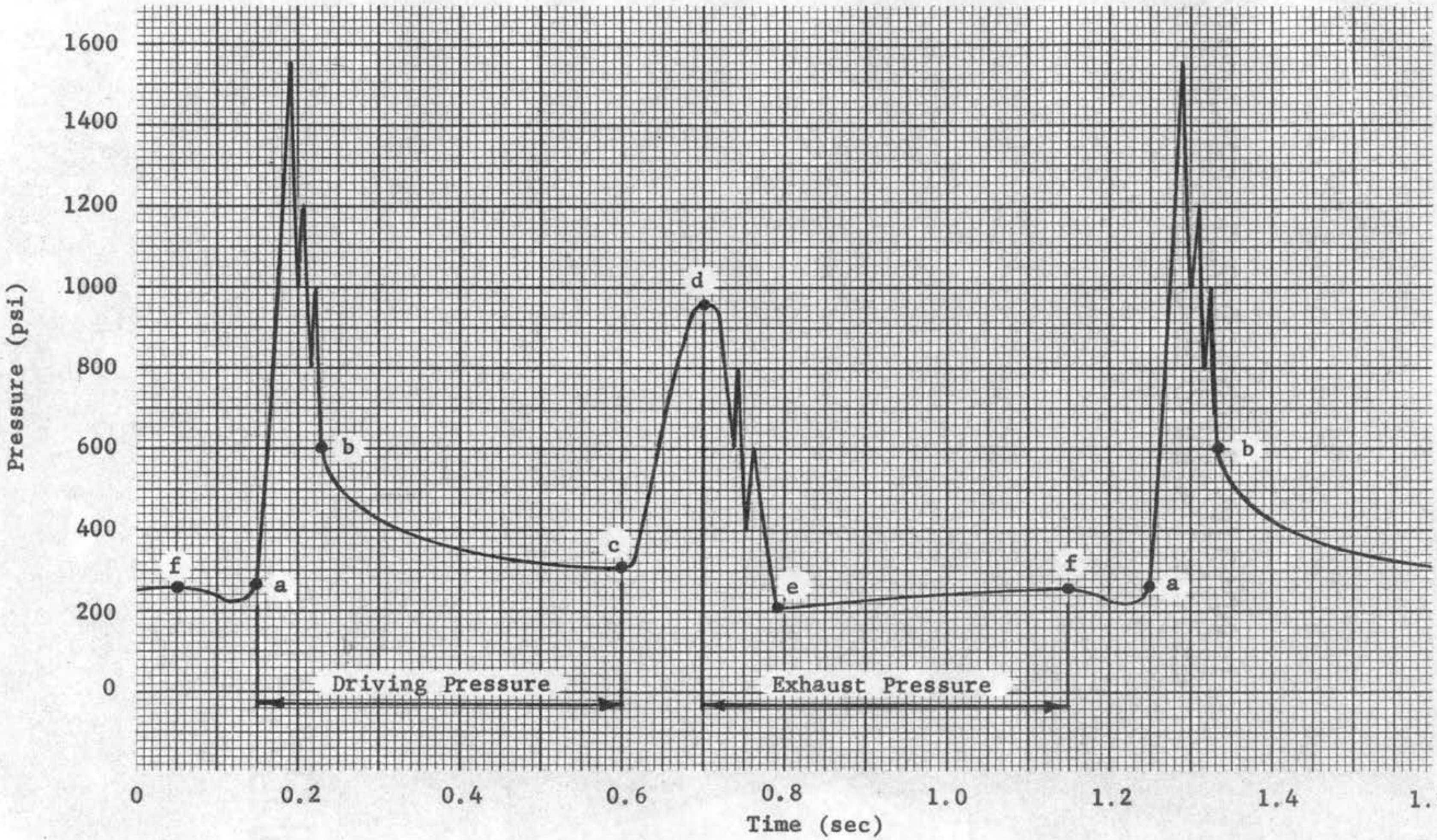
The successful automatic control of the positive-displacement valve circuit and the slave cylinder circuit was dependent upon the creation of reliable pressure-generated pilot signals. Furthermore, in order to coordinate these two component systems into a dependable integrated positioning circuit, it was necessary to establish two distinct pressure levels at which pilot signals were produced.

Since the control of the combined circuit was largely dependent upon pressure variations within the system, pressure profiles were obtained at three key points and used as a basis for the circuit analysis. These three pressure profiles are presented and discussed separately in the following order: (1) the positive-displacement valve pressure profile; (2) the slave cylinder pressure profile; and (3) the main-line pressure profile.

Positive-Displacement Valve Pressure Profile

The positive-displacement valve pressure profile was obtained from pressure tap A (see Fig. 7) and is illustrated in Fig. 9. The portion of the curve between points a and c was a trace of the pressure variation that occurred at tap A while the positive-displacement valve was traveling from left to right. This segment of the profile is referred to as the driving pressure. The portion of the curve between points d and f was a trace of the pressure variation that occurred at tap A while the positive-displacement valve was traveling from right to left. This segment of the profile is referred to as the exhaust pressure. The intervals between points f and a and points c and d were reversing time lags that occurred at each end of the positive-displacement valve stroke.

At point a, the beginning of the driving pressure half cycle, the directional control valve had just shifted and main-line pressure was turned into the left port of the positive-displacement valve. However, main-line pressure then contained a surge which had been induced by the "bottoming" of the positive-displacement valve during the previous half cycle. Thus, the pressure rose rapidly from point a to take on the value of the main-line surge and then, as the inertia of the positive-displacement valve piston began to yield under the tremendous force, the surge pressure dropped abruptly to point b. As the acceleration of the mechanical and fluid parts of the system decreased, the driving pressure continued to decline until the piston of the positive-displacement valve "bottomed" (point c). A pressure surge was then generated and the directional



Relief Valve Setting: 850 psi
Flow Rate: 11.5 gpm

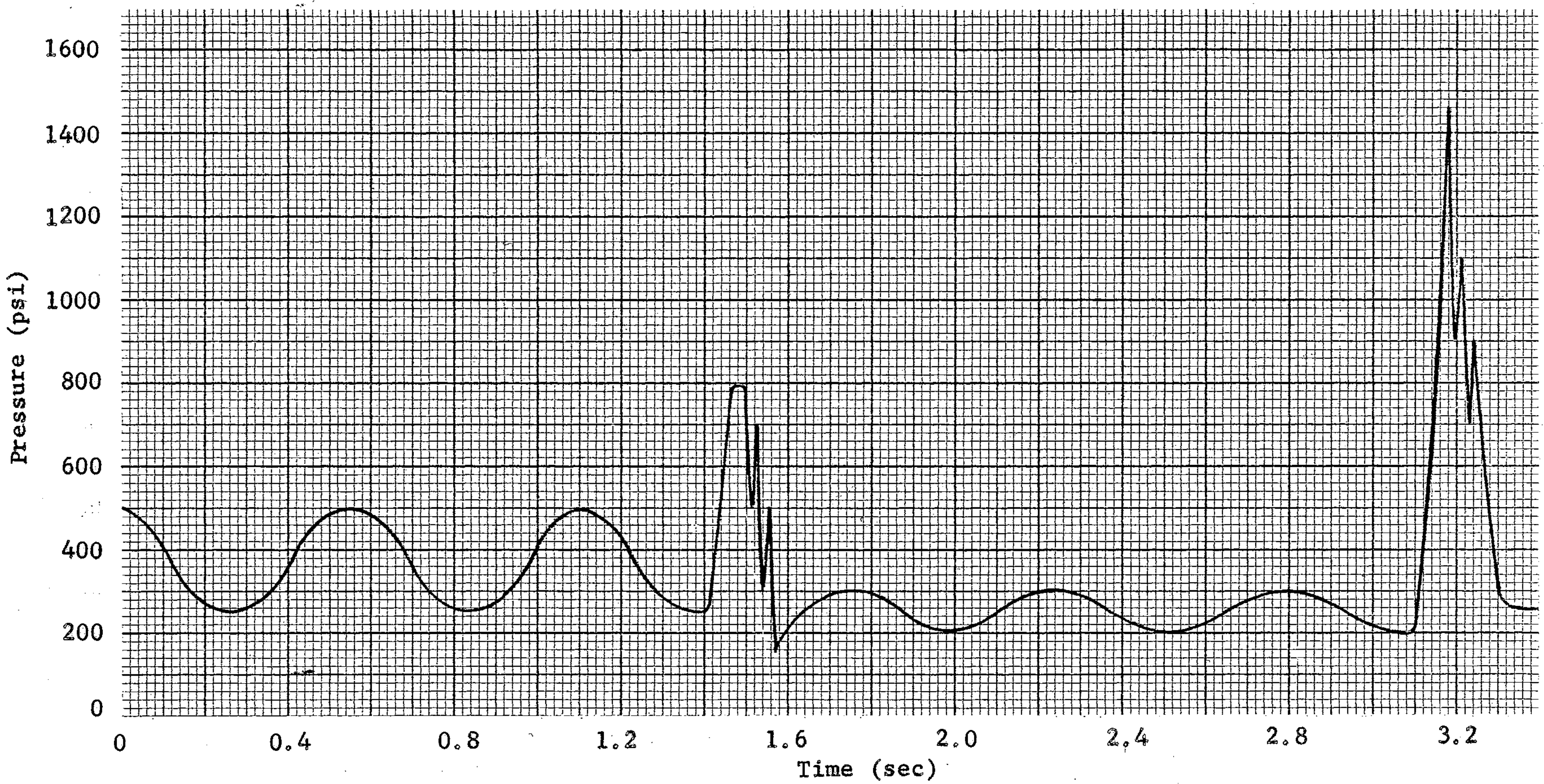
Fig. 9. Positive-Displacement Valve Pressure Profile.

control valve was shifted, subjecting the left port of the positive-displacement valve to exhaust pressure (point d). Then, the momentum of the exhaust fluid pulled down the pressure level sharply to point e. As the deceleration in the exhaust fluid changed to acceleration, the pressure increased until the piston of the positive-displacement valve "bottomed" in the opposite direction (point f). Exhaust fluid was again decelerated and the pressure was reduced until the directional control valve shifted once more, subjecting the left side of the positive-displacement valve to the driving pressure (point a).

It should be noted that two strong pressure surges occurred during the driving pressure half cycle. One of these surges occurred at the beginning (points a to b) and the other at the end (points c to d). However, in order to achieve automatic and continuous cycling of the circuit, it was necessary to make signal valve S-1 sensitive only to the surge which occurred at the end of the driving pressure stroke. This was accomplished by the addition of needle valve N-1. Although the pressure surge from a to b had a higher peak value than that of the pressure surge from c to d, it was not sustained for a sufficient duration of time to allow the passage of enough fluid through the needle valve to operate signal valve S-1.

Slave Cylinder Pressure Profile

The slave cylinder pressure profile was obtained from pressure tap B (see Fig. 7) and is illustrated in Fig. 10. The portion of this profile which occurred near each end of the slave cylinder's stroke had the same characteristic shape as did Fig. 9. Therefore, the behavior of the pressure variation in these areas can be explained by the same reasoning



Relief Valve Setting: 850 psi
Flow Rate: 11.5 gpm

Fig. 10. Slave Cylinder Pressure Profile.

as was presented above.

The portions of the slave cylinder's pressure profile which corresponded to the cylinder's intermediate range of travel were generated while the positive-displacement valve was making several strokes. Therefore, these segments of the curve appeared as a series of damped positive-displacement valve exhaust cycles.

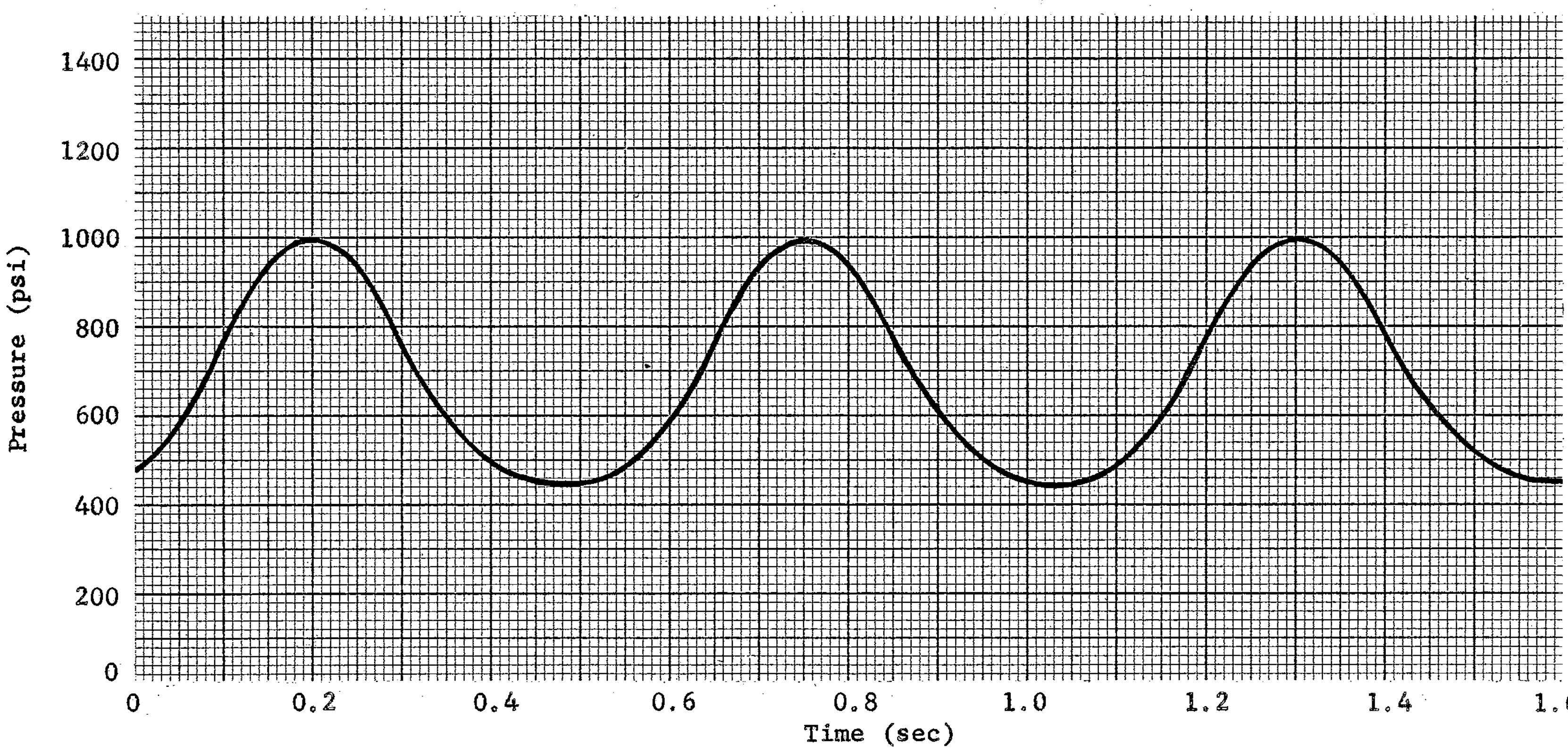
Main-Line Pressure Profile

The main-line pressure profile was obtained from pressure tap C (see Fig. 7) and is illustrated in Fig. 11. Since main-line pressure supplied fluid to the positive-displacement valve, this profile appeared as a series of damped positive-displacement valve driving cycles.

Conclusions

It was found that since undesirable surges were present within the system, the pressure-operated signal valves, alone, were not capable of generating pilot signals which could be used for the dependable control of the positioning circuit. Hence, it was necessary to combine flow regulating devices with the signal valves before automatic control could be achieved.

Also, it was found that the pressure profiles, taken at two separate points in the same line, were not identical in shape. This phenomenon has been observed before by E. J. Waller (10). Waller discovered that cyclic pressure variations may change in amplitude, shape, and phase angle from one end of a closed conduit to the other. Consequently, it was concluded that the operation of the signal valves was partially dependent upon the physical location of their hydraulic pilot connections in a given pressure line.



Relief Valve Setting: 850 psi
Flow Rate: 11.5 gpm

Fig. 11. Main Line Pressure Profile.

CHAPTER VIII

SUMMARY

The objectives of this investigation were to construct and analyze an experimental hydraulic positioning circuit for a slave cylinder which did not require feedback. The circuit was to be capable of operating under a "loaded" slave cylinder condition.

The circuit which was designed for this investigation was a modified form of the hydraulic intermittent feeding circuit described by A. G. Comer (1). Hydraulic fluid was metered to the slave cylinder, in fixed increments, by the use of a positive-displacement metering valve. Pressure-generated pilot signals were used to control the flow of the metering fluid. No provision was made in the experimental circuit to selectively achieve and retain definite positions within the slave cylinder's intermediate range of travel. Only the fundamental and basic principles of a positive-displacement metering circuit were studied in this investigation. Since pressure variations were utilized for the control of the circuit, pressure profiles were obtained at key points and were used as a basis for the circuit analysis.

The circuit which is diagrammed in Fig. 7 proved to be operative under simulated "load" conditions. Moreover, it was observed that the intermittent feeding of hydraulic fluid provided by the positive-displacement valve did produce an incremental advance of the slave cylinder. The pressure-derived control of the positioning circuit, however, was not dependable. The circuit analysis proved that it was necessary to combine flow regulating devices with the pressure-operated signal valves before automatic control of the positioning circuit could be achieved.

CHAPTER IX

RECOMMENDATIONS FOR FUTURE RESEARCH

It is the belief of the writer that the basic concept of a positive-displacement type positioning circuit is sound and worthy of further consideration. However, in the author's opinion, the pressure-derived method of control which was used in this investigation was inherently faulty and inadequate. Therefore, it is recommended that the positive-displacement valve circuit which was used in this study be deleted and replaced by the circuit illustrated in Fig. 12. It is believed that this cam-controlled, positive-displacement valve circuit would provide a dependable positioning mechanism for a hydraulic slave cylinder.

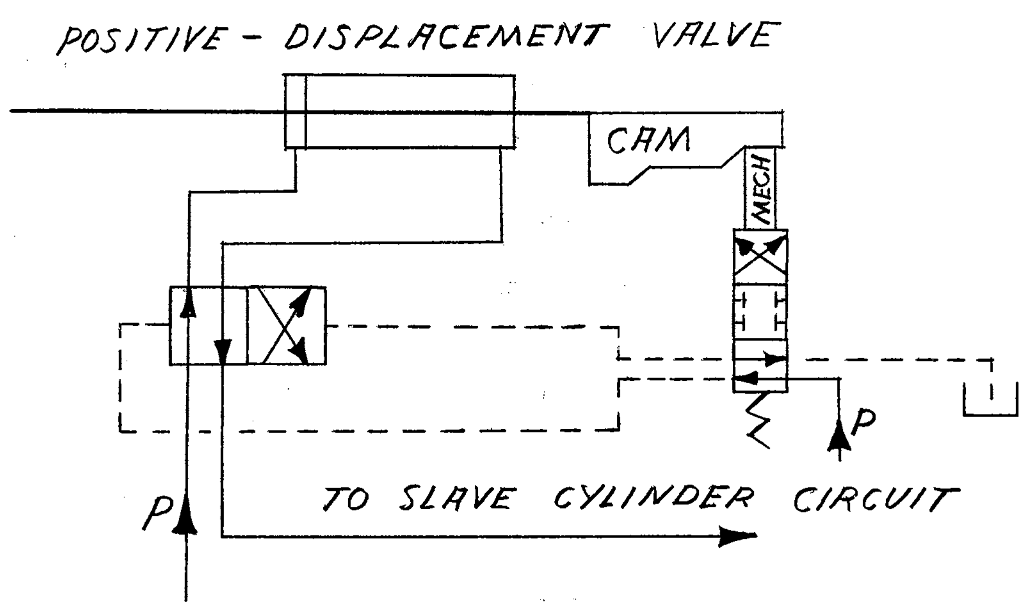


Fig. 12. Proposed Positive-Displacement Valve Circuit

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APPENDIX A

DEFINITION OF COMPONENTS

The following is presented in an effort to provide the reader with a description of each of the standard hydraulic components referred to in this thesis. Each component is identified with respect to its operating characteristics, hydraulic function, and symbolic or graphical representation as approved by the American Standards Association.

Check Valve

A check valve is used to block fluid flow in one direction while permitting free flow in the opposite direction. This function is performed by the use of a closing device such as a swinging disc, spring-seated disc, or spring-loaded ball. (See Fig. 13.)

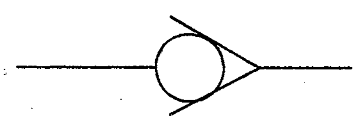

Fig. 13. Check Valve.

Counterbalance Valve.

A counterbalance valve is a pressure-control valve which permits fluid flow at a desired minimum-pressure level in one direction and free flow in the opposite direction. Fluid flow through the valve in one direction is blocked until the minimum-pressure level setting is attained, at which time the valve opens to permit fluid passage. Flow continues in this direction until the pressure of the system falls below the minimum-pressure setting, at which time the valve closes and flow is once again blocked. Fluid flow through the counterbalance valve in the opposite direction is unrestricted. The minimum-pressure or back-pressure level can be varied over a wide range by simply adjusting the spring setting of the valve. This valve was used in this investigation to simulate a given load on the slave cylinder. (See Fig. 14.)

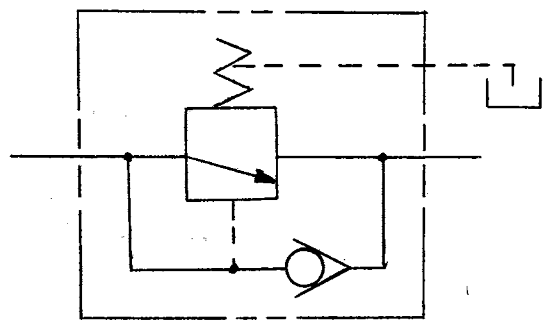

Fig. 14. Counterbalance Valve.

Cylinder

The hydraulic cylinder is a fluid motor usually employed to convert fluid power into reciprocating mechanical power. The symbolic representation for a double-acting, double-rod cylinder is illustrated in Fig. 15.

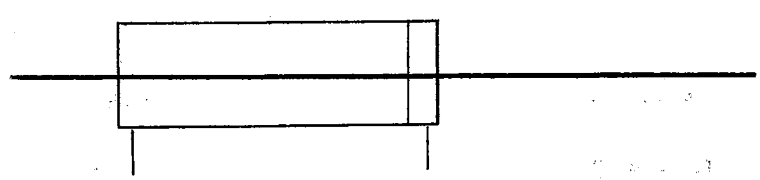

Fig. 15. Cylinder

Directional Control Valve

A directional control valve is used to direct the flow of fluid to other portions of the hydraulic system where fluid application is desired. In a spool-type directional control valve, the movement of the valve spool which determines the direction of fluid flow may be accomplished by the application of electrical force, manual force, hydraulic force, mechanical force, or a combination of these forces. These valves may be either two-way, three-way, four-way, or even more. The number indicates the number of main-line connections in the valve. Usually these valves are either two-position or three-position valves. When a four-way, three-position 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. 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) All four ports were blocked in the neutral position of one combination; and (2) the pressure port was blocked and the rest of the ports were interconnected in the neutral position of the other combinations.

When the symbolic representation for a directional control valve is used, the valve is shown in the neutral position. Figure 16 shows the graphical symbol for a pilot-operated, four-way, two-position, directional valve. Figure 17 is the symbolic representation for a pilot-operated, spring-centered, four-way, three-position valve which has all four ports blocked in its neutral position.

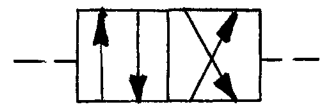

Fig. 16. Directional Control Valve.

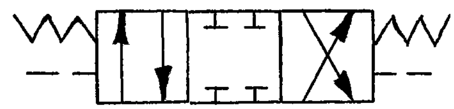

Fig. 17. Directional Control Valve.

The symbolic representation for a manually-operated, four-way, three-position directional control valve which has all four ports blocked when in the neutral position is illustrated in Fig. 18. Figure 19 is the symbolic representation for a solenoid-operated, spring-centered, four-way, three-position, directional control valve which has the pressure port blocked and the other ports connected when the valve is in its neutral position.

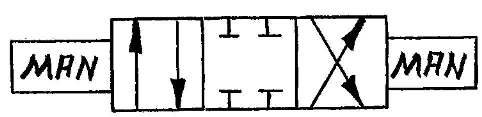

Fig. 18. Directional Control Valve.

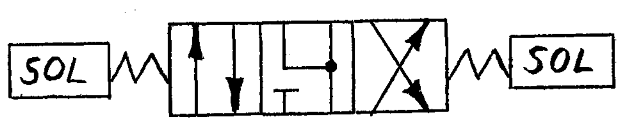

Fig. 19. Directional Control Valve.

A pilot-operated, spring-offset, three-way, two-position, directional control valve or shuttle valve is usually employed when it is desirable to have an auxiliary fluid power source. The main fluid power source is favored by a spring and thus fluid from the main source is allowed to flow through the valve to service the system connected to the outlet port while the fluid from the auxiliary source is blocked. Flow continues in this manner until such time as the main fluid power source pressure falls below that of the auxiliary source, at which time the valve spool shifts and fluid from the auxiliary source services the system and fluid from the main source is blocked. The symbolic representation for shuttle valve is illustrated in Fig. 20.

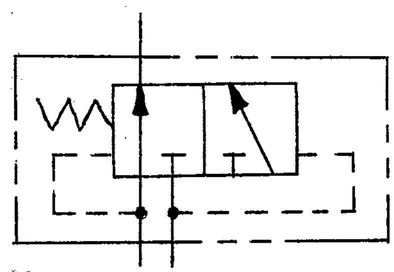

Fig. 20. Shuttle Valve.

Gate Valve

A gate valve is an on- or off-type or shut-off valve. This valve is manually operated and its symbolic representation is shown in Fig. 21.

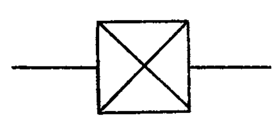

Fig. 21. Gate Valve.

Heat Exchanger

A heat exchanger is used in hydraulic systems to transfer heat from the hydraulic fluid to a cooling fluid. 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. A two-pass, shell-and-tube type heat exchanger with city water passing through the tubes and oil passing through the shell portion was used in this investigation. Figure 22 illustrates the symbolic representation for the heat exchanger.

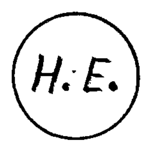

Fig. 22. Heat Exchanger.

Needle Valve.

A needle valve is a manually-operated, non-pressure compensated, volume-regulating type of valve. It employs the use of a long tapered point that permits a very gradual adjustment of the annular flow passage which is formed by the valve seat and the tapered point. Figure 23 is the symbolic representation for a needle valve.

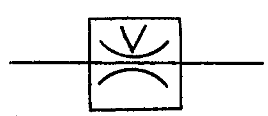

Fig. 23. Needle Valve.

Oil Reservoir

The symbolic representation for an oil reservoir which is vented to the atmosphere is illustrated in Fig. 24.

Fig. 24. Oil Reservoir.

Pilot Check Valve

A pilot check valve is essentially a check valve with additional provisions for allowing reverse flow when desired. It is a free-flowing valve in one direction while the flow is blocked in the opposite direction until such time as pilot pressure is applied which unseats the ball and allows free flow in the reversed direction. (See Fig. 25.)

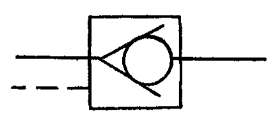

Fig. 25. Pilot Check Valve

Pump Unit

Figure 26 illustrates the symbolic representation for a pump unit which is driven by an electric motor through a variable speed reducer. Although the pump is a fixed-displacement gear type, it becomes a variable volume pump by varying the speed of the pump-input shaft.

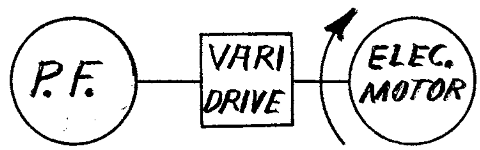

Fig. 26. Pump Unit.

In Fig. 27 a variable displacement pump directly coupled to an electric motor is illustrated.

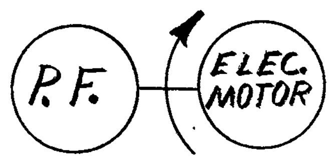

Fig. 27. Pump Unit.

Relief Valve

A relief valve is a safety valve which serves to establish the maximum pressure in a hydraulic system. In operation, the relief valve opens when the system reaches the maximum pressure-level setting and allows the excess volume of fluid to bypass the circuit and return directly to the reservoir. This valve is adjustable over a wide range of pressures. The primary function of the relief valve is to prohibit the pump from being subjected to excessively high pressures. (See Fig. 28.)

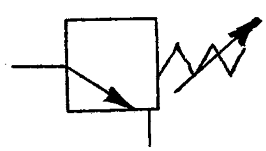

Fig. 28. Relief Valve.

Sequence Valve

A sequence valve is a pressure-control valve. It is used to control the order of fluid flow to the various parts of the hydraulic system by requiring the pressure of the inlet port to reach the desired pressure-level setting before the sequence valve opens, 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 pressures. Figure 29 is the symbolic representation for a sequence valve.

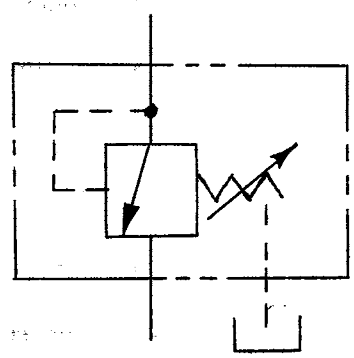

Fig. 29. Sequence Valve

Strainer and Filter

A filter consists of a filter element supported and enclosed within a housing which provides connecting ports for fluid lines. When a filter element can be supported by itself and is used, without the housing, in an installation such as a hydraulic reservoir, it is usually called a strainer. Both the filter and the strainer serve to remove foreign particles which may be present in the hydraulic fluid. Strainers are usually located in the reservoir at both the return and section lines. Figures 30 and 31 are symbolic representations for a strainer and a filter.

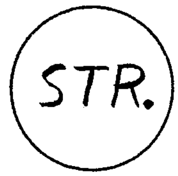

Fig. 30. Strainer.

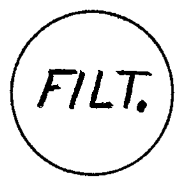

Fig. 31. Filter.

APPENDIX B

FLUID POWER TEST STAND FACILITIES

A fluid power test stand was designed and fabricated during the course of this investigation in order to provide the investigator with the following facilities: (1) A main fluid power source; (2) a support for "bread-boarding" the hydraulic circuit under investigation; (3) a fluid manifold system for connecting the hydraulic circuit under investigation; (4) a drip pan for receiving leakage fluid; (5) a hydraulic-fluid-temperature-control system; and (6) a filtering system to reclaim drip-pan fluid.

A discarded Magna-flux table was rebuilt and modified to serve as the frame for the test stand. Steel grating similar to that used in fire escape ramps was employed to serve as the support for the hydraulic system in this investigation. A drip pan was constructed of sheet steel and a nine-port manifold system was provided for the pressure, the return, and the pilot-return-drain connections. Figure 32 is a schematic diagram of the fluid power test stand circuit and its related facilities.

The test stand circuit is composed of the following three parts: (1) The main pump unit; (2) the recirculating unit; and (3) the fluid reclaiming unit. Fluid in the main pump unit passed from the reservoir through one of the three strainers before entering the suction side of the pump which was powered by a variable-speed drive unit. After passing through the pump the high pressure fluid was available for

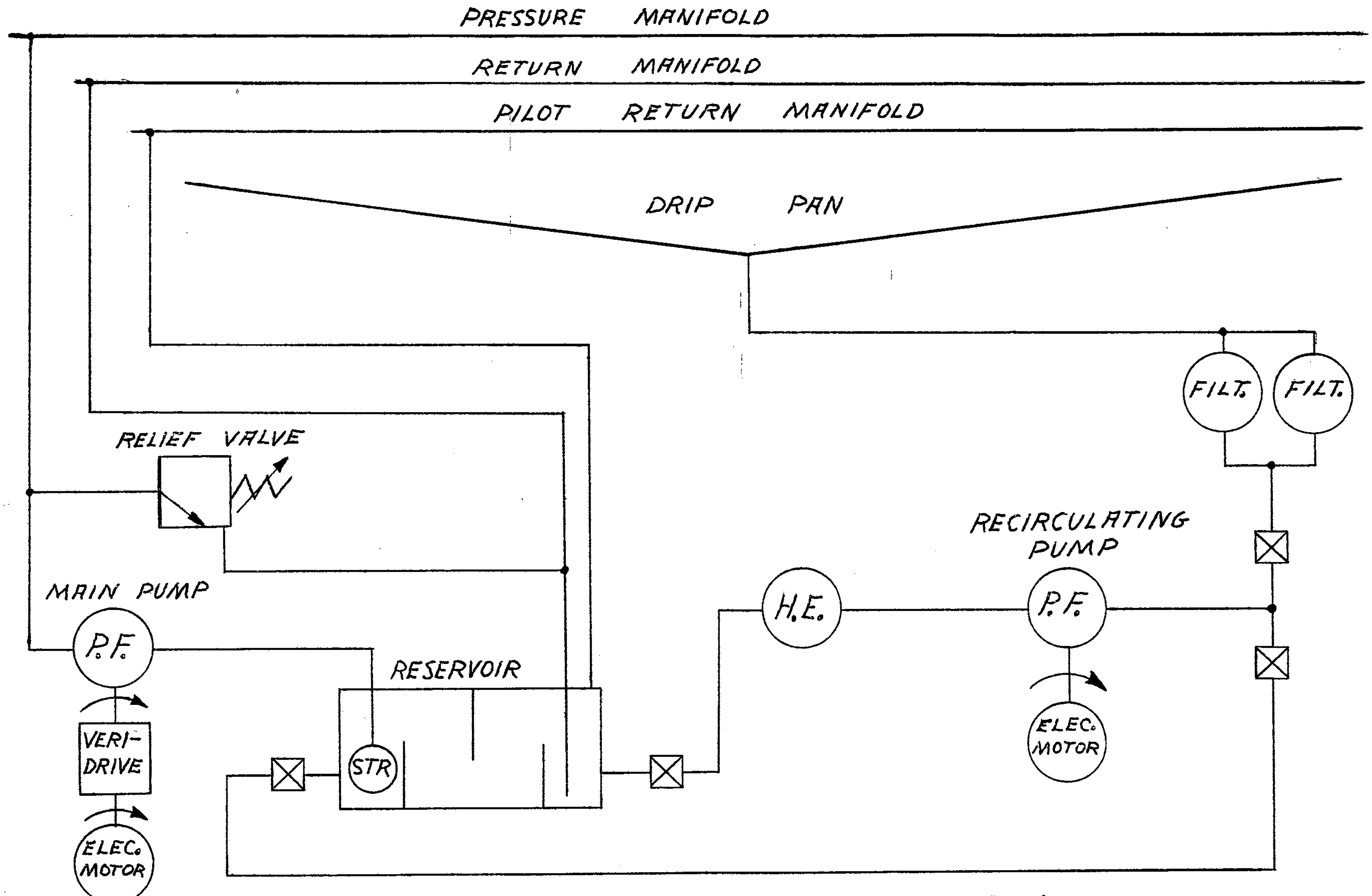

Fig. 32. Graphical Circuit of the Fluid Power Test Stand.

circuit application at the pressure manifold. Any excess volume of fluid was bled off the pressure line by the relief valve and was directed into the return line which was also equipped with a manifold. The combination pilot-return and drain manifold collected fluid at near atmospheric pressure and returned it to the reservoir above fluid level. The recirculating unit was used to maintain the temperature of the hydraulic fluid at some constant level and consisted of a recirculating pump unit and a heat exchanger. Fluid was drawn out of the reservoir through a valve arrangement and pumped through the heat exchanger back into the reservoir. Flow of city water which was used as the cooling medium was regulated through the heat exchanger by a globe valve. The reclaiming unit provided a means of restoring the waste or leakage hydraulic fluid collected in the drip pan to a condition where it was again usable in the hydraulic circuit. In the operation of the reclaiming unit, leakage fluid, which was collected in the drip pan, was drawn through a parallel bank of two filters and a valve arrangement into the recirculating pump where the fluid was then returned to the reservoir by the route of the recirculating system.

Figure 33 is a picture of the completed fluid power test stand ready for operation and Fig. 34 is the calibration curve for the gear pump used in the main pump unit.

Test Stand Apparatus and Equipment

1. Louis Allen Electric Motor: rated, 1150 rpm; 3/4 horsepower; 220-440 volts; 3 phase, 60 cycles; serial no. 1812413.
2. Yale and Towne Variable Displacement, Sliding Block Type Pump: type A-6; serial no. 18467; 1 1/4 inch ports.

Fig. 33. Fluid Power Test Stand

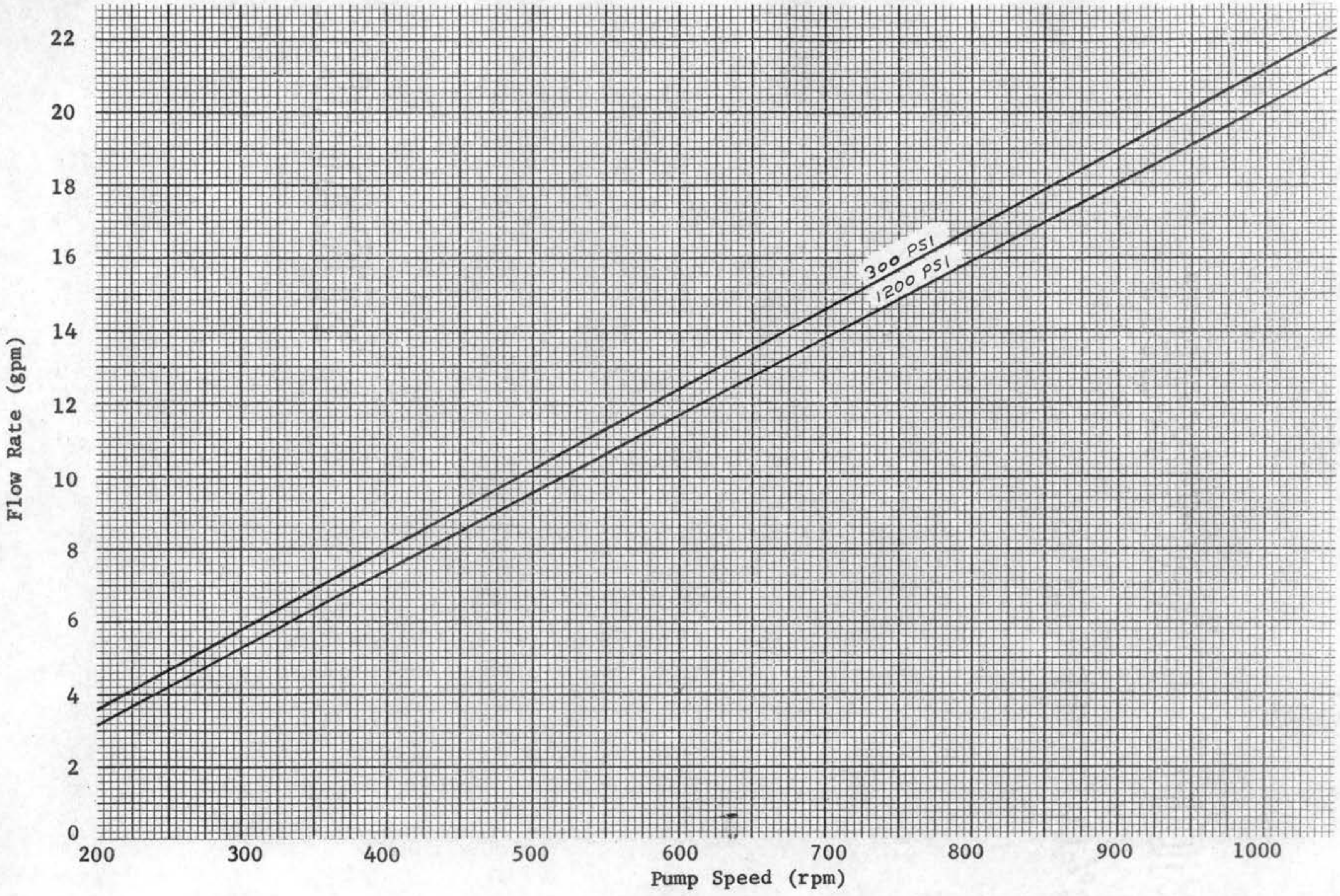

Fig. 34. Performance Curves for Commercial Gear Pump.

3. Young Heat Exchanger: maximum working pressure, 75 psi; model no. 67219; serial no. L11187; number of passes, two; water inlet and outlet through the tubes, 1 inch; oil inlet and outlet to the shell side, 1 inch.

4. Greer Olaer Micronic Filter: two required; model no. and serial no., unknown; rating approximately 10 gallons per minute at 50 psi.

5. Reeve Vari-Speed Moto Drive Unit: size, 6281-C-12; gear ratio, 1.54 to 1; maximum output speed, 1,500 rpm; minimum output speed, 250 rpm; electric motor, Robbins and Myers: rated, 1,150 rpm; 15 horsepower; voltage, 220-440 volts, 3 phase, 60 cycles.

6. Commercial Gear Pump: model no. PD 322, BEEL 206; maximum pressure, 1,500 psi; rotation, clockwise or counterclockwise; gear size, 2 inches; maximum capacity, 50 gallons per minute at 1,000 psi; pump suction, 1 inch; pump discharge, 3/4 inch.

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

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

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

10. Fluid Controls Pilot Type Relief Valve: part no. 1500-6-6; port size, 3/4 inch; pressure range, 50 to 2,000 psi.

11. 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.

APPENDIX C

APPARATUS AND EQUIPMENT

Positive Displacement Valve Circuit

1. Directional Control Valve, (*D-1): Manufacturer, Republic Manufacturing Company; 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-2): Manufacturer, Double A Products Company; model no. DA-180-C; type: pilot-operated, four way, two-position; port size, 1 inch; pilot connections, 1/4 inch.
3. Directional Control Valve, (*D-4): Manufacturer, Double A Products Company, model no. DA-180-C; type: pilot-operated, four-way, two-position; port size, 1 inch; pilot connections, 1/4 inch.
4. Directional Control Valve, (*D-3): Manufacturer, Electrol, Inc.; model no. 185-AN; type: manually-operated, four-way, three-position, all ports blocked in the neutral position; port size, 3/8 inch.
5. Signal Valve, (*S-1 and S-2): Designed by this investigator and built by the Research Apparatus Development Laboratory, College of Engineering, Oklahoma State University; port size, 1/4 inch.
6. Needle Valve (*N-1 and N-2): Manufacturer, P and C Company; model no. 1040A; port size, 1/4 inch

*Indicates the number assigned to this valve in Fig. 7

7. Positive-Displacement Valve: Manufacturer, Boeing Airplane Company, assembly no. 9-14666-501; size 4.84 inch stroke and 2.172 square inch bore area; maximum operating pressure, 3,000 psi; port size, 1/4 inch.
8. Gate Valve (*G-1, G-2, G-3, and G-4): Manufacturer, Henry Voyt Machine Company; model no. 5-9535; port size, 1 inch.

Slave Cylinder Circuit

1. Directional Control Valve, (*D-5): Manufacturer, Denison Engineering Company; 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.
2. Directional Control Valve, (*D-6): Manufacturer, Double A Products Company, model no. DA-180-C; type: pilot-operated, four-way, two-position; port size, 1 inch; pilot connections, 1/4 inch.
3. Shuttle Valve (*D-7 and D-8): Manufacturer, Adel Precision Products Corp.; port no. 20348; port size, 1/4 inch.
4. Signal Valve, (*S-3 and S-4): Designed by this investigator and built by the Research Apparatus Development Laboratory, College of Engineering, Oklahoma State University; port size, 1/4 inch.
5. Needle Valve, (*N-3 and N-4): Manufacturer, P and C Company; model no. 1040A; port size, 1/4 inch.
6. Slave Cylinder: Manufacturer, Carter Controls, Inc.; model CBH, size, 3 inch bore and 24 inch stroke; type, double-acting, double rod, cushioned; maximum operating pressure, 1,500 psi; port size, 3/8 inch.

*Indicates the number assigned to this valve in Fig. 7.

7. Check Valve, (*D-9): Manufacturer, Republic Manufacturing Company; model no. 433-3/4S; port size, 3/4 inch.
8. Counter Balance Valve, (*C-1): Manufacturer, Double A Products Company; model no. SA-180-B; type, internal pilot and external drain; pressure range, 250-500 psi; port size, 1 inch.

Instruments

1. Chrono-Tachometer: Manufacturer, Standard Electric Time Company, model CM.
2. Pressure Pickup: Manufacturer, Electro Products Laboratories; model no. 3700-A Electro Pressuregraph; type, capacitance; maximum pressure, 2,000 psi with 0.062 inch disc.
3. Oscillograph: Manufacturer, Allen B. Dumont Laboratories; model, single-beam cathode-ray; type, 304H; serial no. 2274.
4. A.C. Voltage Regulator: Manufacturer, Sorenson and Company, 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. 2-508.

*Indicates the number assigned to this valve in Fig. 7.

VITA

Joseph Milton Case

Candidate for the Degree of

Master of Science

Thesis: THE POSITIONING OF A HYDRAULIC SLAVE CYLINDER WITHOUT FEEDBACK

Major Field: Mechanical Engineering

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Honors and Awards: Received a one-year senior scholarship from the Schlumberger Well Surveying Corporation; received Oklahoma State University Honors Fellowships in the Department of Mechanical Engineering for the academic years 1957-58 and 1958-59.

Professional Organizations: Member of the Society of Automotive Engineers.