TIME-DELAY OF VARIOUS TYPES OF HYDRAULIC REVERSING SYSTEMS

By

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PREFACE

Mr. A. G. Comer was one of the first to investigate some of the problems associated with applied hydraulics at Oklahoma State University. (1). Mr. Comer's work concerned the control of a hydraulic cylinder by means of pressure sequencing, which was proven to be a reliable control method. The above mentioned thesis (1) also predicted the operation time of a reversing circuit within six percent. As most of this six percent error was due to inaccuracies in the calculation of the actual shift time[1]; it was desirable to analyze the shift time more closely.

The object of this study was to determine the time-delay associated with mechanical, hydraulic and electro-hydraulic control systems. In the field of applied hydraulics these systems are the most commonly used, and a comparative analysis of their individual characteristics is important.

Many people have assisted the author in setting up the test equipment, performing the necessary tests, and preparing this thesis. Special credit is due to Professor E. C. Fitch for his help throughout the preparation of this report. Much of the value this thesis may have is due to his criticism and suggestions. The author's gratitude is also extended to Professors B. S. Davenport and L. J. Fila and to Messers

[1]The shift time or time-delay is considered to be the period of time starting as a hydraulic piston completes a stroke, and a signal to reverse the direction of travel of the piston is generated; the period ends when the signal shifts a directional control valve and the piston reverses its direction of motion.

George Cooper and John McCandless. It is hoped that the many others who have helped in so many ways towards the completion of this work will read their own names into these words of thanks.

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LIST OF SYMBOLS AND ABBREVIATIONS

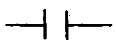
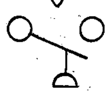
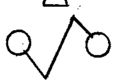
A	Acceleration (in./sec^2)
F or G	Force (lbs)
c	Coefficient of viscous damping (lb-sec/in.)
k	Spring constant (lbs/in.)
d	Characteristic length (in.)
Q	Pump output ($in.^3$/sec)
t	Time (sec)
τ	Torque (in.-lbs)
I	Mass Moment of Inertia (in.-lb-sec^2)
α	Angular acceleration (radians/sec^2)
M	Mass (lb-sec^2/in.)
x or y	Variable distance (in.)
C_1 or C_2	Arbitrary constants
D_v, D_c, or D_r	Diameter of valve spool, cylinder or piston rod respectively (in.)
$\dot{x}$ or $\dot{y}$	First derivative of distance with respect to time
$\ddot{x}$ or $\ddot{y}$	Second derivative of distance with respect to time
L_v or L_c	Length of valve spool travel or cushioned action respectively (in.)
p	Operating pressure (lb/$in.^2$)
psi	Pressure (lb/$in.^2$)
g	Gravitational constant (in./sec^2)
U	Velocity (ft/sec)
V	Volume ($in.^3$)

D Operator notation (d/dt)

gpm Volumetric output (gals/min)

Subscripts are used to distinguish between various lengths on some figures.

The following Electrical Symbols Are Used

Symbol	Abbreviation	Identification
	CR	Contact Relay, normally closed
	CR	Contact Relay, normally open
	LS	Limit Switch, normally open
	PS	Pressure Switch, normally open
	Sol	Solenoid

CHAPTER I

INTRODUCTION

The knowledge and use of applied hydraulics evolved from a crude pump and cylinder arrangement to more complicated equipment such as servomechanisms and involved brain circuitry. Various functions and actions can be performed by the use of hydraulic power controlled by means of fluid signals. These fluid signals can be very important to the successful use of brain circuitry to control machine processes. The dynamic characteristics of the creation and application of mechanical and fluid signals to a directional control valve are studied in this report. The length of the time-delay inherent in a control system may have a decisive influence on which control system is applied to a particular machine.

Four common methods of signal generation and/or application used in applied hydraulics are studied in this investigation. These methods of control are:

1. mechanical
   a. mechanical linkage
   b. cam valve (fluid pilot)

2. electro-hydraulic
   a. limit switches
   b. pressure switches

Earlier studies of Comer (1) and Matthews (2) have been directed towards showing that a pressure rise in a hydraulic circuit is instrumental in producing a reliable signal. Now it is desirable to analyze the application of a signal, and to compare a pressure signal method to other methods of signal generation and application.

The investigation of the dynamic characteristics of the four control systems was carried out on an experimental and analytical basis. The action of each control system mentioned above was timed in a test circuit. The experimental values obtained with the test circuits were then compared to calculated values found from equations derived to express the characteristics of each control system. The development of these equations formed an integral part of this investigation. Application of these equations to similar systems will enable the machine designer to predict the operation time of a particular control system, and help him to select the most appropriate controls.

CHAPTER II

PREVIOUS INVESTIGATIONS

An extensive search throughout the Engineering Index, Industrial Arts Index and many of the leading technical periodicals (3) revealed only one article concerned with the timing of mechanical and solenoid operated valves. (L. Dodge, 1956). Dodge's article (4) was quite valuable, but it is believed that some of the material given was somewhat misleading.[1] Furthermore if the equations given by Dodge were substantiated by experimental tests, no record of the tests was included in his report. Comer (1) gave equations that predicted the time-lag of pressure rise in a closed hydraulic system due to compressibility of the fluid. He applied the results of these equations to the overall time-lag involved in controlling a cylinder by means of pressure sequencing, but no experimental verification of the shifting time-delay was given. No information was available in the periodicals searched on the time-delay that could be expected in a hydraulic reversing circuit containing a solenoid valve as a pilot valve.

[1]cf. Chapter 5.

CHAPTER III

TEST CIRCUITS

Each test circuit consisted essentially of a hydraulic cylinder that would reciprocate automatically, with a counterbalance valve to simulate a load on the cylinder and to provide a means of controlling system pressure. The reciprocation of the cylinder was controlled by the spool position of a directional control valve. Suitable timing devices were applied to measure the time-delay[1] of the circuit.

An attempt was made to time the mechanical control system by using a high speed movie camera to record the motion of a valve handle and the time. A high speed camera could be an extremely accurate timing method, but the available equipment gave a maximum speed of only one frame every 15 milliseconds (64 frames per second) which was too slow for the time periods involved. With a faster camera this technique would give very accurate measurements.

The timing system finally selected for all test circuits consisted of an adjustable frequency square-wave generator with an electronic counter to register the number of pulses. The signal from the square-wave generator was fed through two micro-switches to the electronic

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[1]The time-delay is considered to be the period of time starting as a hydraulic piston completes a stroke, and a signal to reverse the direction of travel is generated; the period ends when the signal shifts a directional control valve and the piston reverses its direction of motion.

counter. The switches were so wired that the first (called the "start" switch) passed the signal to the counter when the shift signal occured, and the second (called the "stop" switch) stopped the signal to the counter when the shift was completed. With the frequency of the square-wave generator adjusted to a thousand cycles per second, the electronic counter registered the time interval in milliseconds.

The mechanical shift controls are shown symbolically in Fig. 3-1. The timing apparatus with the microswitches and the directional control valve for the mechanical system are shown in Fig. 3-2. The hydraulic symbols used in these figures and throughout this thesis are in accordance with American Standards Association practice, except as noted. All main hydraulic lines are shown by solid lines, pilot lines by dashes (ASA standard practice), and the electrical lines are shown by a dash-dot (- ∘ - ∘) arrangement. A symbolic representation of each hydraulic component, with a description, is included in Appendix A.

With the micro-switch arrangement shown in Fig. 3-2, the timing signal was fed from the generator to the counter only while the valve spool contacted neither the "start" nor "stop" switches. Contact between the valve spool and either micro-switch interrupted the timing signal. The basic timing system shown in Fig. 3-2 was employed for all circuits tested, but the arrangement of the "start" and "stop" switches was modified for each system. The switch arrangement is included in the figure of each test circuit (Figs. 3-3, 3-4, and 3-5).

The cam valve reversing circuit and switch arrangement are shown in Fig. 3-3. The "start" switch was a normally open micro-switch so adjusted that it passed the timing signal at the same time the piston rod operated the cam valve. The "start" switch was energized within

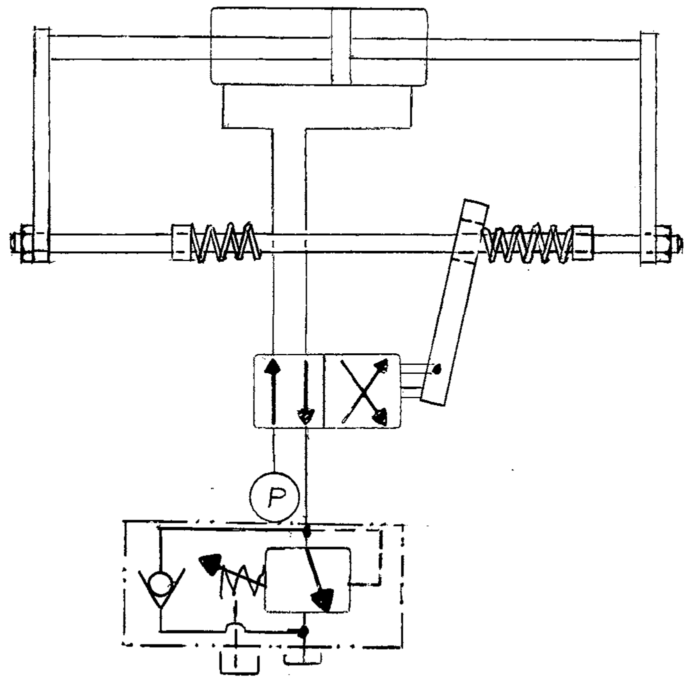

Fig. 3-1. Mechanical test circuit.

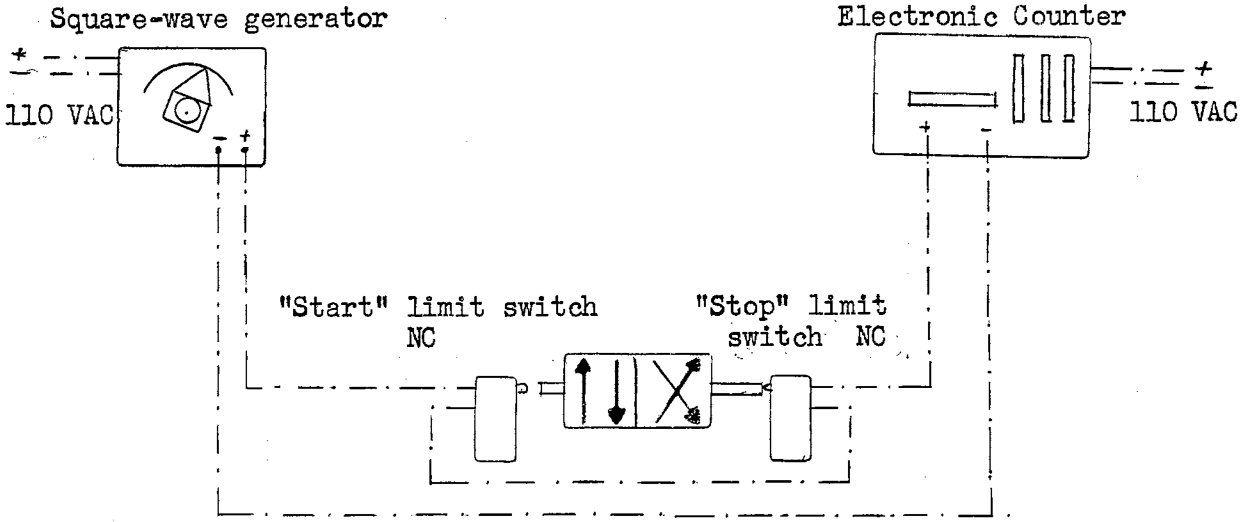

Fig. 3-2. Timing instrumentation, shown on mechanical directional control valve.

a millisecond of the opening of the cam valve. The "stop" switch was a normally closed pressure switch that interrupted the timing signal when sufficient pressure was applied to shift the directional control valve.

The reversing circuit and timing mechanism for the limit switch controls are shown in Fig. 3-4. The timing "start" switch consisted of a normally open limit switch which was energized within one millisecond of the operation of the solenoid valve limit switch (limit switch one in Fig. 3-4). The synchronization of the two switches was checked by use of the square-wave generator and electronic counter. The "stop" switch was a normally closed pressure switch that interrupted the timing signal when sufficient pressure was applied to shift the directional control valve.

A latching relay was needed to operate the electro-hydraulic circuits because the limit and pressure switches gave a momentary impulse which could not insure positive operation with an open center, spring centered, solenoid valve. If some type of detent control were applied to the directional control valve to hold it in position until another reversing signal was received, the latching relay would be unnecessary.

The pressure switch control system is shown in Fig. 3-5. This system is identical to the limit switch controls except for the method of signal pick-up for reversing the cylinder, and the "start" switch used to determine time-delay. The "start" switch was a normally open pressure switch adjusted to pass the timing signal whenever the pressure rose at the end of the stroke of the piston. This pressure occured at the end of the stroke due to the piston "bottoming" in the cylinder.

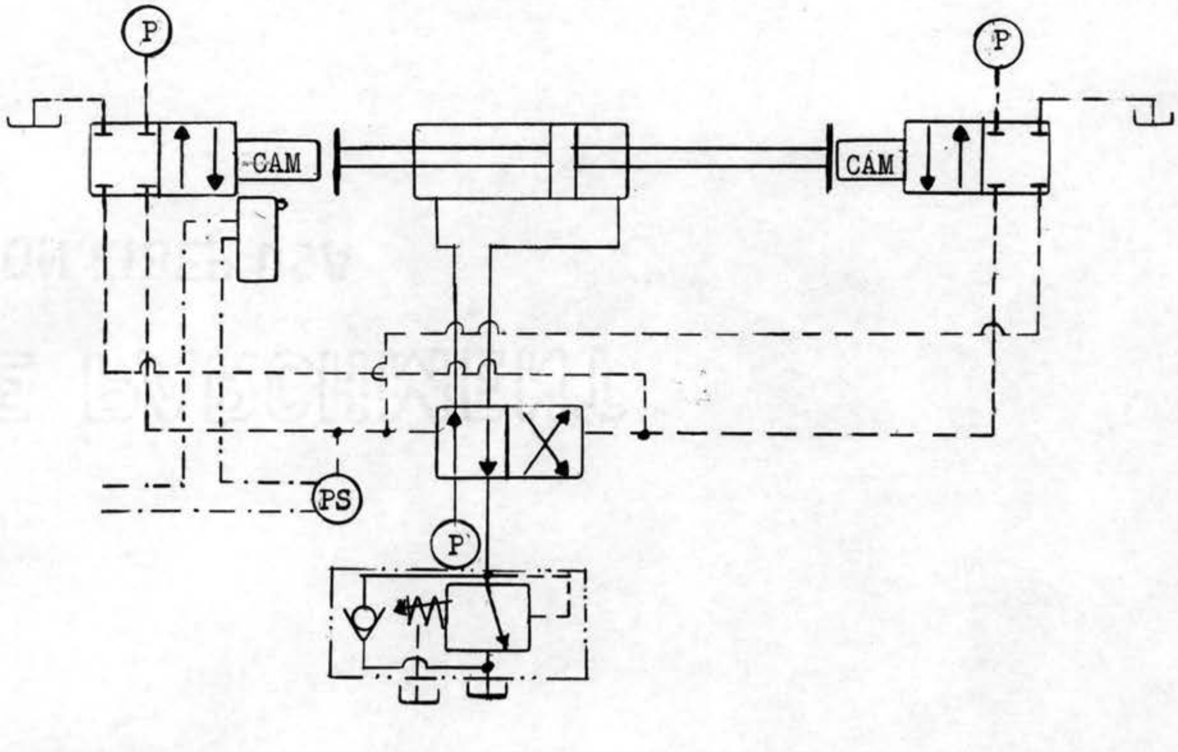

Fig. 3-3. Cam valve control system with "start" and "stop" timing switches.

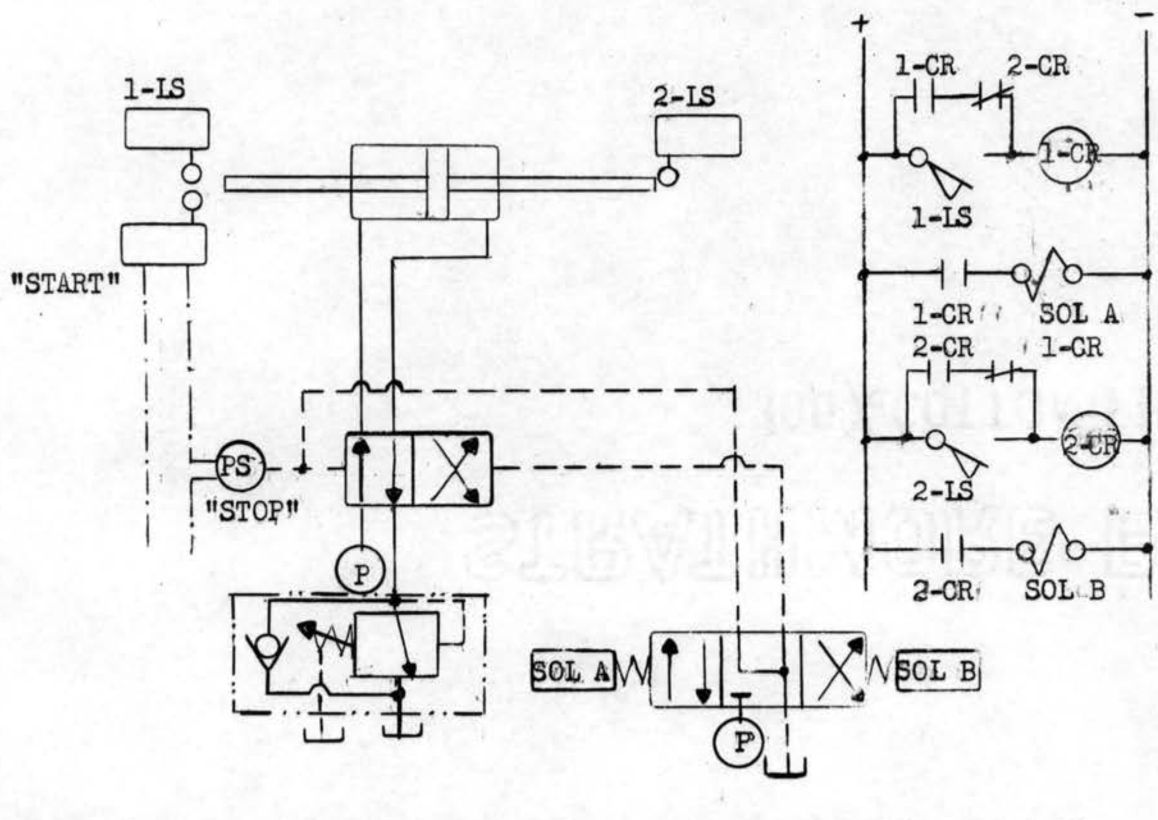

Fig. 3-4. Limit Switch control system with "start" and "stop" timing switches.

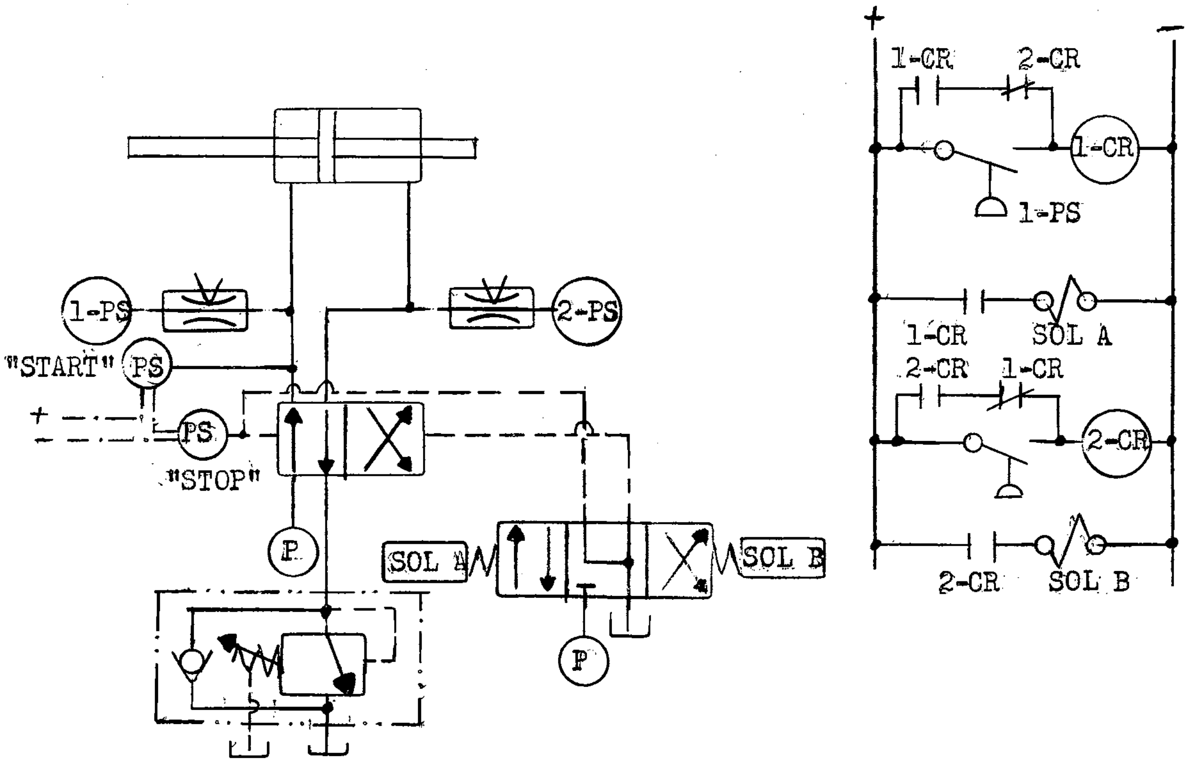

Fig. 3-5. Pressure switch control system with "start" and "stop" timing switches.

CHAPTER IV

TEST PROCEDURES

The fluid power supply used in this investigation was identical to that used by Matthews (2) in an earlier investigation at Oklahoma State University. Essentially it consisted of a power source, pump, reservoir unit and heat exchanger as described in Appendix C. The system could deliver fluid at volumetric output and pressures up to 20 gpm and 1500 psi respectively.

In order to evaluate the effect of pump output and system pressure on time-delay of a hydraulic circuit, tests were run at four pump outputs and two system pressures. A larger range of pressures would be desirable; but because of the high pressure surges, the difficulty of protecting pressure gauges from these surges, and the lack of adequate control of system pressure, only two operating pressures were studied. After the circuit was set up and operated for a time in order to establish equilibrium conditions, tests were run at pump speeds of 500, 650, 800, and 950 rpm and a system pressure of approximately 350 psi. The time-delay for several reversals was measured and recorded for each speed. The test was then repeated at a system pressure of approximately 700 psi.

Whenever pressure switches were used to determine a reversal signal or a timing signal, the pressure setting was checked by the use of a dead weight tester after each setting. This procedure was found to be necessary because of the inaccuracies of the pressure switch scale when used to read pressure directly. The operating pressure for the switch was indicated by a continuity light across the terminals. The continuity light flashed on when the pressure rose to the set value. The pressure on the dead weight tester was then noted and recorded.

CHAPTER V

ANALYSIS OF CIRCUITS AND PRESENTATION OF RESULTS

Introduction

The purpose of this investigation was to determine the time-delay of four of the more common control methods used in applied hydraulics: cam valve-pilot control, mechanical linkage controls, pressure, and limit switch controls. The investigation consisted of two parts; analytical and experimental. The experimental study of each control method was conducted to check the accuracy of the equations developed in the analytical part.

The operating cycle of a hydraulic cylinder can be divided into two separate phases, namely piston travel time, and valve shift time. The shift time itself can be subdivided into: 1. signal generation time, and 2. signal transmission and application time. The piston travel time starts when a piston begins moving away from an extreme position, and ends when the moving parts contact a signal element. The start of the signal generation time is the end of the piston travel time; the signal generation time is completed when the signal element actually emits a signal to shift the directional control valve. The signal transmission and application starts when a signal is formed and ends when a directional valve is shifted to reverse the motion of the hydraulic cylinder. The complete shift time is also called the time-delay of the circuit.

To clarify the concept of signal transmission and application time, the electro-hydraulic operation will be used as an example. The signal transmission and application time for this test (Figs. 3-4, 3-5) started as current was applied to a latching relay through the limit switch or pressure switch. The relay in turn energized a solenoid controlled pilot valve. The signal application time ended when the pilot valve had passed sufficient fluid to shift the directional control valve, which reversed the direction of fluid flow to the cylinder.

Mechanical Control System

Because of the complexities introduced by the viscous friction, and O-ring drag, of a seemingly simple mechanical control system, an exact solution of the time-delay for this system was not attempted nor deemed valuable. Another mechanical control system might have a different physical configuration, yet the procedure followed in the derivation below can be applied to nearly all mechanical linkage controls. Only signal application time was measured and calculated because the many variations possible in a mechanical linkage would affect signal generation time, and the solution is usually relatively easy.

Upper and lower limits for the time-delay can be found by using Newton's basic laws of motion. The symbolic picture of the valve and operating mechanism, shown in Fig. 5-1, depicts the applied spring and damping forces and the masses on which they operate.

For the fastest possible speed, the viscous damping of the mechanical system will be assumed equal to zero. At the instant the detent releases, the torque applied to the valve handle can be expressed by the following equation:

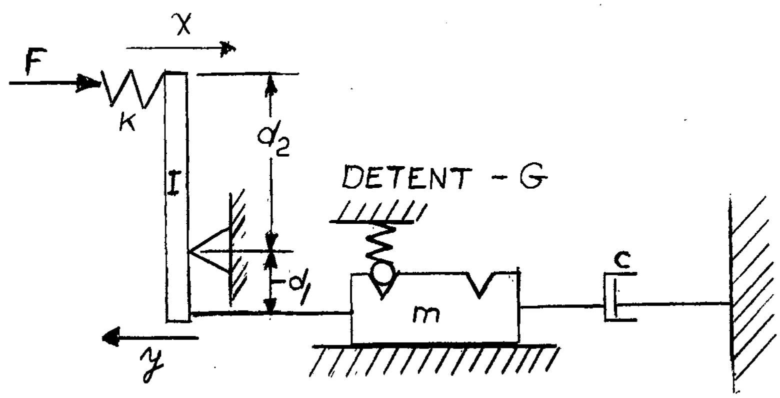

Fig. 5-1. Mechanical Control System.

$$\mathcal{T} = Fd_2 - mad_1 \quad \text{(valid for } t < 0\text{)} \qquad (5\text{-}1)$$

in which $\underline{a}$ is acceleration (in./sec^2), $\mathcal{T}$ is torque (in.-lb) and clockwise torques are considered positive. Because of the detent action on the valve spool, the operating spring would be compressed for some distance, until the force $\underline{F}$ reached a value that would overcome the mechancial detent force $\underline{G}$. Before the detent releases force $\underline{F}$ equals $\underline{G}$. When the detent released, the spring would force the valve handle to the other detent position which would reverse the direction of fluid flow to the cylinder. Calling the distance the valve handle and the spring travels $\underline{x}$, the force $\underline{F}$ decreases by the amount $\underline{kx}$ from the maximum detent force as the valve handle moves ($\underline{k}$ is the spring constant, lbs/in.), or the force of the spring on the valve handle equals $F - kx$. Substituting this value in Eq. (5-1) gives:

$$\mathcal{T} = (F - kx)d_2 - mad_1 \qquad \text{(Valid for } t \geqq 0\text{)} \qquad (5\text{-}2)$$

But since $\mathcal{T} = I\alpha$ these expressions for $\mathcal{T}$ can be equated to yield:

$$(F - kx)d_2 - mad_1 = I\alpha \qquad (5\text{-}3)$$

in which $\underline{I}$ is the mass moment of inertia (lb-in.-sec^2), and $\underline{\alpha}$ is angular acceleration (radians/sec^2).

The acceleration of the valve spool $\underline{a}$ equals $\ddot{\underline{y}}$ equals $\ddot{x}d_1/d_2$ and $\underline{\alpha}$ equals $\ddot{x}/d_2$ with $\ddot{\underline{x}}$ and $\ddot{\underline{y}}$ representing the second derivatives of distance with respect to time. Substitution of these values in Eq. (5-3) gives:

$$(F - kx)d_2 - \frac{md_1^2}{d_2}\ddot{x} = \frac{I\ddot{x}}{d_2} \tag{5-4}$$

Equation (5-4) can be simplified to the form

$$(I + md_1^2)\ddot{x} + kxd_2^2 = Fd_2^2 \tag{5-5}$$

To further simplify Eq. (5-5) let $R = I + md_1^2$ and $U = kd_2^2$.

Equation (5-5) then becomes:

$$R\ddot{x} + Ux = Fd_2^2 \tag{5-6}$$

Dividing Eq. (5-6) by $\underline{R}$ results in:

$$\ddot{x} + \frac{U}{R}x = \frac{Fd_2^2}{R} \tag{5-7}$$

Equation (5-7) can be solved by use of the La Place transform with zero initial conditions to give:

$$(P^2 + \frac{U}{R})\ L\ (x) = \frac{Fd_2^2}{P\,R} \tag{5-8}$$

Solving for the La Placian operator $\underline{L(x)}$ yields:

$$L\ (x) = \frac{Fd_2^2}{PR(P^2 + \sqrt{U/R}^2)} \tag{5-9}$$

From a table of the La Place transform (6), Eq. (5-9) can be solved for x to give:

$$x = \frac{Fd_2^{\,2}}{U} \left(1 - \cos \sqrt{U/R}\;\, t\right) \qquad (5\text{-}10)$$

The physical values from measurement and calculation are:

$M = 0.00412$ lb-sec^2/in. $\qquad d_1 = 1.312$ in.

$k = 2.20$ lb/in. $\qquad d_2 = 4.375$ in.

$F = 12$ lb $\qquad I = 0.01493$ in.-lb-sec^2

Since the time t at $x = 2.084$ is desired, the above constants can be substituted in the equations for U and R and in Eq. (5-10) to yield:

$$2.084 = \frac{12.0(4.375)^2}{42.1} \left(1 - \cos \sqrt{42.1/0.02198}\;\, t\right) \ .$$

This equation can be solved to yield $t = 0.0141$ seconds or $t = 14.1$ milliseconds.

As the measured values for the operating time was considerably above the calculated value, as shown by Table I, and Fig. 5-2, evidently the viscous friction and O-ring drag has a pronounced effect on the motion of the valve spool. This value of time (14.1 milliseconds) will be used as the upper limit for the operating speed of the mechanical system because friction must increase the operating time.

A more realistic consideration of the friction in the mechanical system will be made with the viscous damping c on the valve spool assumed, somewhat arbitrarily, to be 4.0 lb sec/in.. The the equation of motion for the mechanical system Eq. (5-5) becomes:

TABLE I

MECHANICAL SHIFT

Pump Speed (rpm)	Cycles per minute	Average shift time (seconds)	Stroke length (inches)	Theoretical cycles per minute	Pump output (gpm)	System Pressure (psi)
500	36.4	0.0698	12.06	36.6	10.30	300
642	45.8	0.0615	12.25	46.65	13.35	355
810	58.2	0.0560	12.30	59.50	17.10	300
950	67.1	0.0710	12.30	69.00	19.85	355

$$(I + md_1^2)\ddot{x} + cd_1^2\dot{x} + kd_2^2x = Fd_2^2 \qquad (5\text{-}11)$$

Symbolizing cd_1^2 by $\underline{H}$ and introducing the operator $D = dx/dt$, with other symbols as before gives:

$$\left(D^2 + \frac{H}{R}D + \frac{U}{R}\right)x = \frac{Fd_2^2}{R} \qquad (5\text{-}12)$$

The complete solution of Eq. (5-12) is the sum of the particular and general solutions with:

$$D = \frac{-H/R \pm \sqrt{(H/R)^2 - 4U/R}}{2} \qquad (5\text{-}13)$$

Substituting the value for $\underline{H}$ = 68.8 lb-sec-in., and the values of $\underline{U}$ and $\underline{R}$ in Eq. (5-13), it can be solved for $\underline{D}$ to give the two values − 6.3, and − 306.9. The particular solution of Eq. (5-12) is Fd_2^2/U = 5.45 in., and the complete solution is:

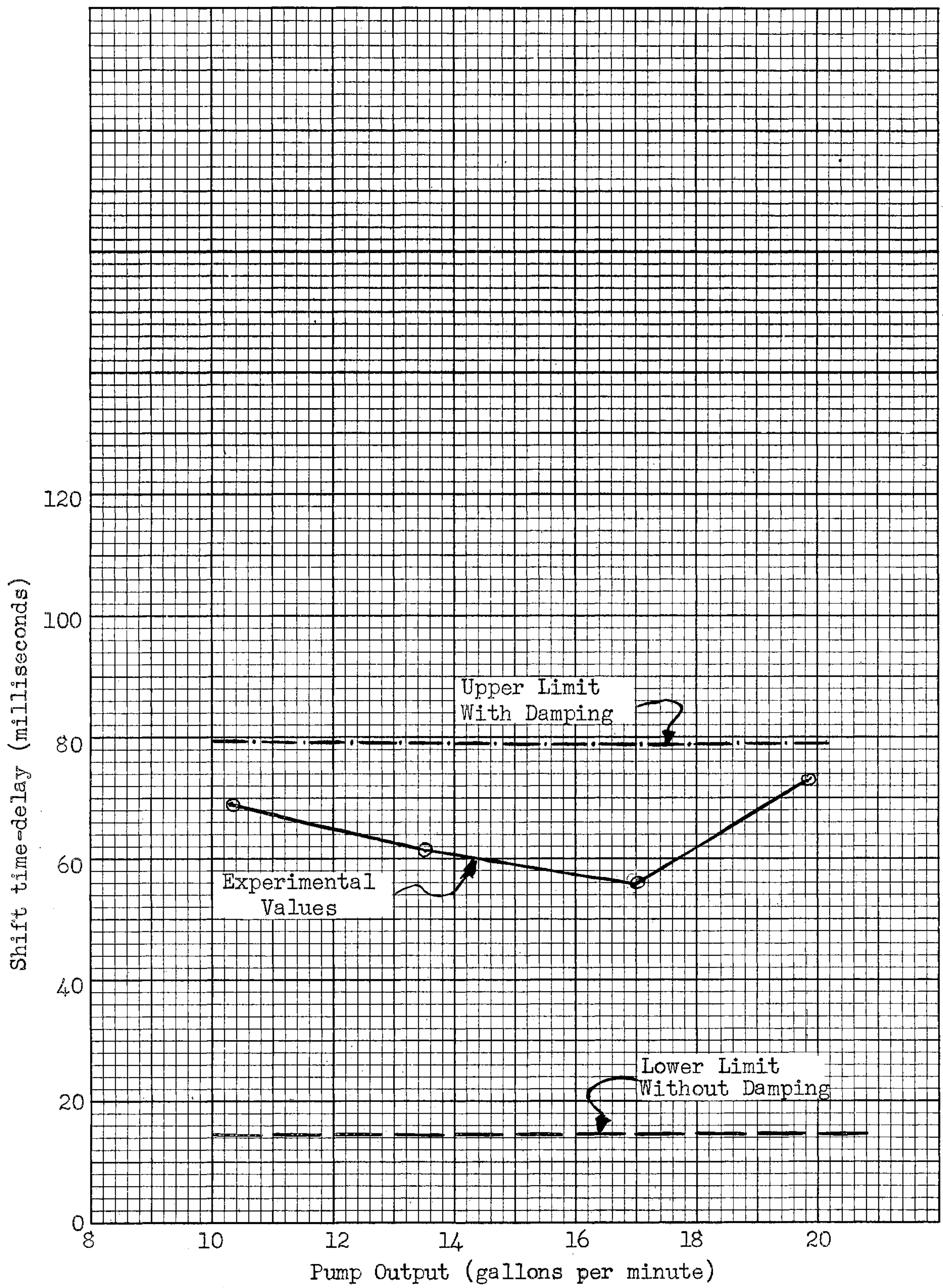

Fig. 5-2. Experimental and analytical curves of mechanical linkage control system.

$$x = C_1 e^{-306.9t} + C_2 e^{-6.3t} + 5.45 \qquad (5\text{-}14)$$

From the initial conditions of $\underline{x} = 0$ and $\dot{\underline{x}} = 0$ at $\underline{t} = 0$, $\underline{C_1} = 0.1142$ and $\underline{C_2} = -5.564$, giving:

$$x = 0.1142\, e^{-306.9t} - 5.564\, e^{-6.3t} + 5.45 \qquad (5\text{-}15)$$

Equation (5-15) was solved by trial and error with the result $\underline{t} = 0.0795$ seconds. This value of $\underline{t}$ was considered to be the largest probable operating time for the mechanical system.

The analysis of the mechanical system was made on the basis of the spring being held motionless which was not the actual case. The momentum of the spring would cause an impact type load on the valve handle, which makes the true solution of Eq. (5-11) a second order partial differential equation (7) that cannot be solved readily. However it is believed that the results of the static analysis already presented can be applied to valves with detent action with only a slight error in certain cases. Whenever the spring and detent characteristics are such that the detent is released before the spring compresses to its solid length, the velocity of the spring has little effect on the operation time of the valve. If however, the spring is compressed solid before the detent releases, then the impact velocity would have a pronounced effect on the valve spool motion. In the case of a spring whose length is small in relation to its diameter, then the effect of the spring velocity is less than with a long, small diameter, spring.

The tests of the mechanical control system indicated that an increase in pump output, or piston speed decreased the time-delay of the valve, except at the highest piston speed tested. This unexpected increase in time-delay at the high piston speed can possibly be explained by the deformation of the O-ring seals used on the valve spool. The higher system pressure caused by the flow characteristics of the counterbalance valve experienced at high pump output, may have deformed the rubber O-rings sufficiently to cause increased drag on the valve spool, which would increase the time-delay of the control system.

Cam Valve Control System

The time-delay of a cam valve control system is dependent on: the pump volume output, the operating pressure, the flow characteristics of the cam valve, the mass of the directional valve spool, the area of the spool on which the pressure acts, the coefficient of viscous damping between the spool and the valve body, and the distance the spool travels to shift the direction of fluid flow. Because of the variations possible in the free travel of the push rod of a cam valve, only the signal transmission and application time was measured and computed. The signal generation time (from first contact between the piston and the cam valve to opening of the valve port) is easily found from volume relationships, and is not detailed in this report. As some of the above factors in the time-delay are negligible, a simple relation can be derived to give the length of signal application time by assuming:

1. The cam valve is fully opened at start of signal to shift the directional control valve.
2. Full pump volume is directed to shift the directional control valve.

3. Cylinder retraction (or effective completion of the shift) begins when the directional valve spool is 1/8 in. past the center position, which opens the valve port.

4. Pressure build-up time in the pilot chamber of the directional control valve is negligible.

Using the above assumptions, the signal application time can be found by dividing the volume of fluid displaced by the value spool by the pump output. This gives:

$$t = \frac{\pi/4 \, D_v^2 \, L_v}{Q} \tag{5-16}$$

in which $\underline{D_v}$ is the diameter of the valve spool (in.), $\underline{L_v}$ is distance of spool travel to open the valve port (in.), $\underline{Q}$ is pump output (in.^{3}/sec), and $\underline{t}$ is time (sec). This relationship neglects the effect of the operating pressure, and gives results that are accurate within 2.4 milliseconds for the system tested at the operating pressures of 300 to 450 psi. The results of Eq. (5-16) are plotted in Fig. 5-3. An empirical relationship that gives results within 2.6 milliseconds for the pressure range of 700 psi in which p_1 is the high operating pressure and p_2 is the low operating pressure, is:

$$t = \frac{\pi/4 \, D_v^2 \, L_v}{Q \sqrt{P_1/P_2}} \tag{5-17}$$

The ratio of the square root of the pressures is applied somewhat empirically, but is logical because in laminar flow the velocity of a fluid in a closed conduit is proportional to the square root of the head loss.

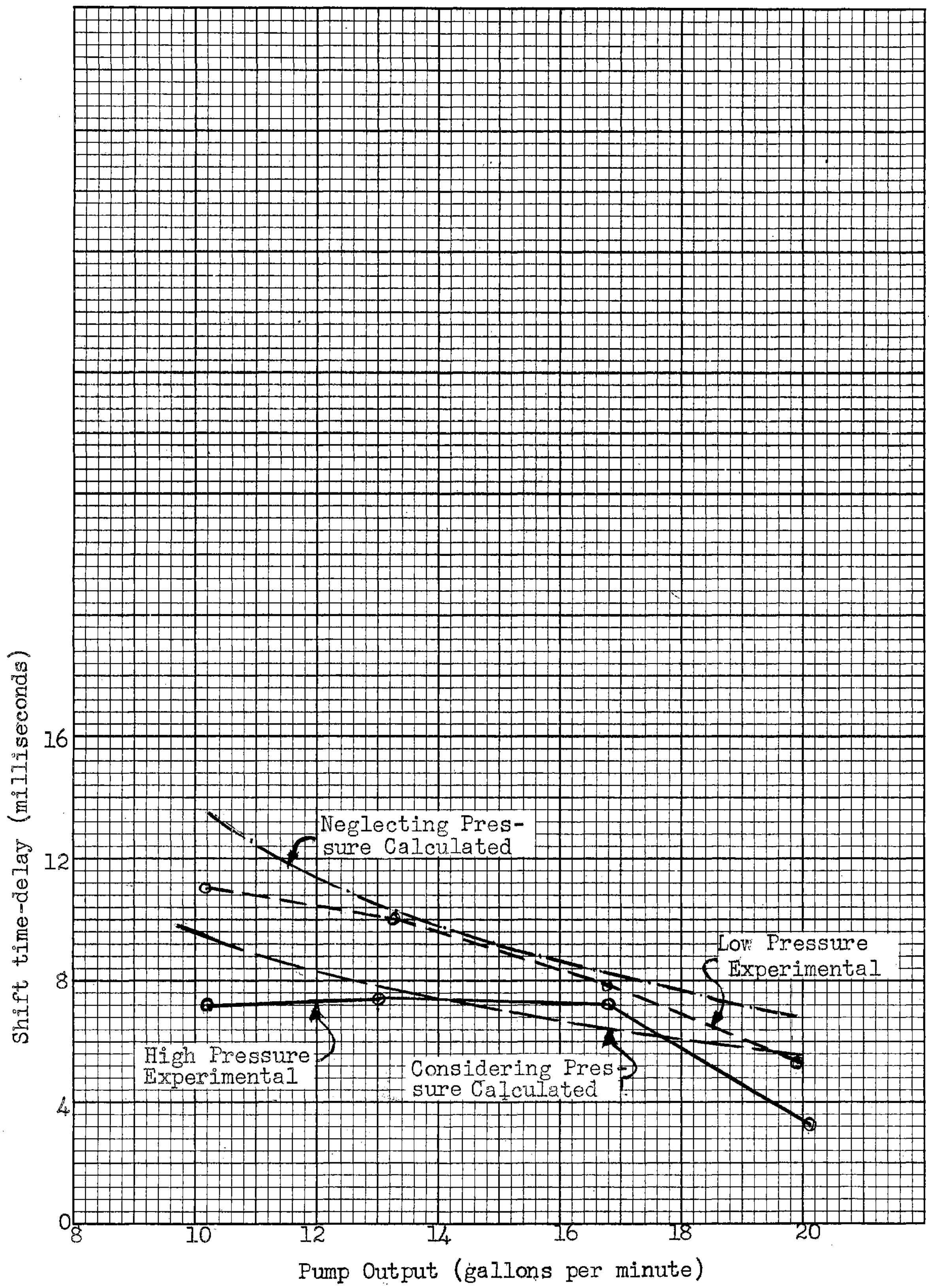

Fig. 5-3. Experimental and analytical curves of cam valve control system.

TABLE II

CAM VALVE CONTROL

Pump Speed (rpm)	Cycles per minute	Average shift time (seconds)	Theoretical cycles per minute	Pump Output (gpm)	"Stop" Signal Pressure (psi)	System Pressure (psi)
489	30.8	0.0111	33.0	10.2	275	335
643	40.6	0.0100	42.8	13.3	275	335
803	50.4	0.0078	54.4	16.8	325	435*
950	55.4	0.0053	64.4	19.9	300	445*
497	29.1	0.0072	33.0	10.2	350	690
631	37.7	0.0074	42.1	13.0	360	690
803	46.8	0.0072	54.4	16.8	350	690
951	51.6	0.0032	65.0	20.1	300	690

Stroke length was constant at 13-1/4 inches.

*Minimum setting available with back pressure valve at these pump outputs.

The cam valve control system is extremely fast in operation, and ranks with the limit switch controls in speed and surge-free operation, as will be discussed later. Because of the simplicity of the cam valve circuit, the time-delay can be easily predicted by calculations with greater accuracy than that needed for most practical purposes.

Limit Switch Control System

The time-delay of a limit switch control system, which utilizes a holding or latching relay and a solenoid pilot valve to control the directional valve, is primarily dependent on the operating speed of the electrical controls. The time-delay of alternating current operated relays and solenoids has not been evaluated analytically, to the knowledge of the author. Many values of operation time are possible, ranging from a few microseconds for extremely fast relays to several hundred milliseconds for some solenoids. For 60 cycle AC operated solenoids, the portion of the electrical cycle the voltage happens to be when a solenoid (or relay) is energized may affect the operation time of the solenoid by as much as three milliseconds. In this investigation the manufacturer's values of operation time will be used in all calculations concerning relays or solenoids. Because of the variations possible in the free travel of a limit switch, the signal generation time is not considered.

The total time-delay of a solenoid, pilot controlled, hydraulic system can be estimated by adding the time-delay of each component in the control system. The time-delay of a relay or solenoid can usually be obtained from the manufacturer. The time-lag that occurs from the opening of the solenoid valve to completion of the shift of the directional control valve can be predicted in the same way as for the cam valve control system. As this value of time-lag is in the range of ten milliseconds, it will frequently be overshadowed by the time-lag of the relay (from two to fifty milliseconds) and the time-lag of the solenoid (from three to a hundred milliseconds)[1]. When considering the delay of the solenoid controlled valve, it should be realized that the solenoid valve port is partially opened before the solenoid

[1]These are representative values taken from several manufacturer's of electrical equipment.

plunger travels its full stroke, and this partial opening may pass sufficient fluid to shift the directional control valve. The time-delay for the limit switch system should be between the values calculated for the cam valve system in Eq. (5-16) as the lower limit (assuming no delay in relays and solenoids) and the sum of Eq. (5-16) and the full operation time of the relays and solenoids as the upper limit. The total operation time of the relay and solenoid used in this investigation is approximately ten milliseconds. The results of the calculations based on the above analysis are plotted in Fig. 5-4. Table III shows the experimental data of tests on the limit switch control system.

TABLE III

LIMIT SWITCH CONTROL

Pump Speed (rpm)	Cycles per minute	Average shift time (seconds)	Theoretical cycles per minute	Pump Output (gpm)	Stroke length (inches)	System pressure (psi)
498	34.6	0.0095	35.8	10.25	12.25	300
670	45.8	0.0122	48.3	13.95	12.38	300
795	53.6	0.0106	56.6	16.70	12.62	355*
960	62.6	0.0072	68.5	20.20	12.62	420*
492	34.7	0.0052	35.6	10.05	12.10	635
657	45.7	0.0076	48.1	13.60	12.10	635
782	53.2	0.0098	58.0	16.40	12.10	635
945	62.0	0.0072	69.0	19.95	12.38	635

* Minimum setting available with back pressure valve.

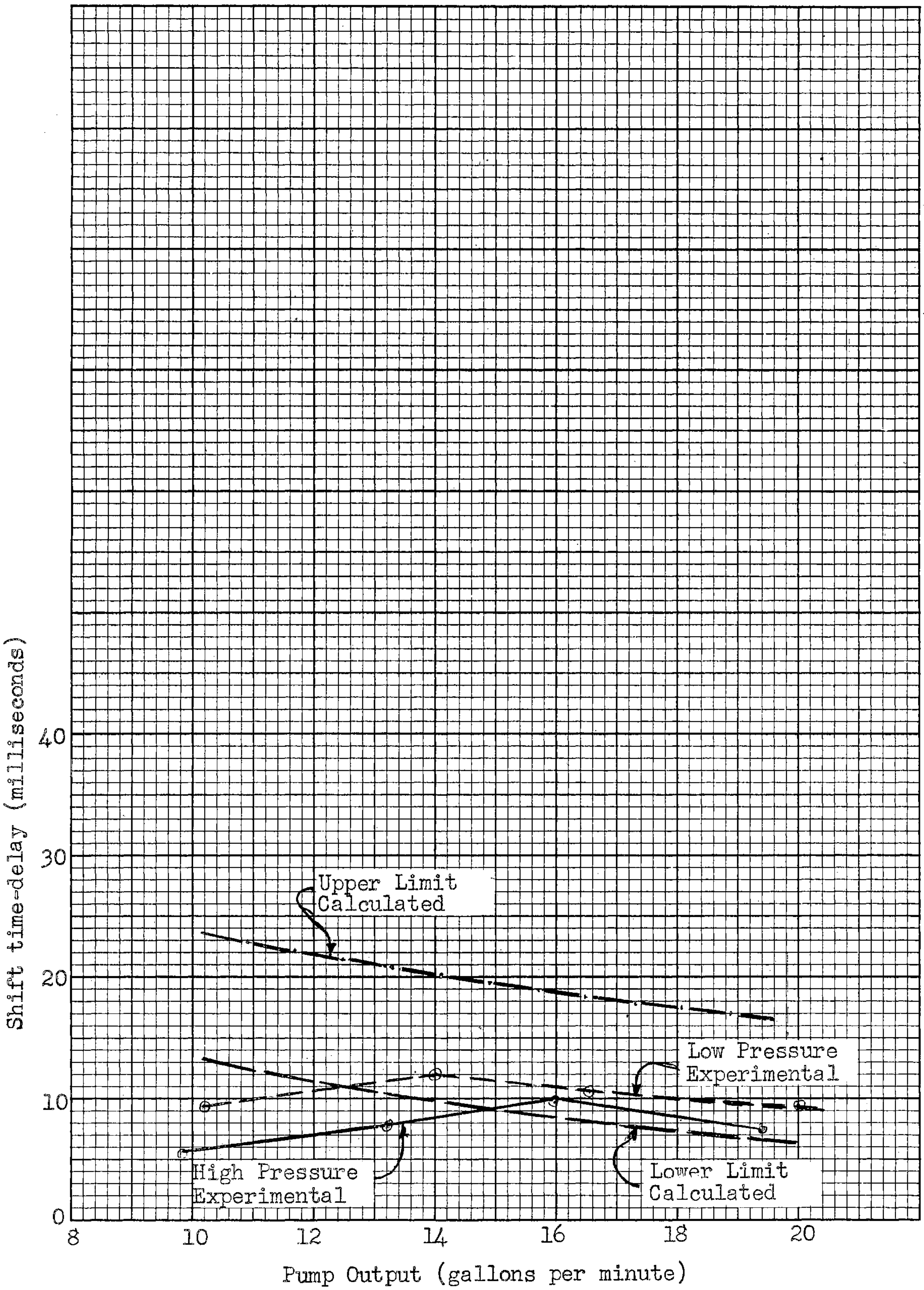

Fig. 5-4. Experimental and analytical curves of limit switch control system.

The "stop" signal used in timing the limit switch control system was a pressure switch connected to the pilot chamber of the directional control valve (Fig. 3-4). It was noticed that at low pump outputs (below 12 gpm) the time measured depended upon the "stop" pressure switch setting, and an increase of 50 psi on the pressure switch would shown a corresponding increase in the measured shift time. As no other timing method was available, the pressure switch was somewhat arbitrarily set at 250 psi for this one reading. At higher pump outputs a change in the pressure setting had little or no effect on the time measurements.

Pressure Switch Controls

The time-delay of the pressure sensing control system is unlike that of the other control systems in that a considerable portion of the time-delay is due to the signal generation time, and a comparatively small part of the delay is due to signal transmission and application. Theoretically the circuit controlling pressure switches could be set for a pressure slightly above the operating pressure, which would decrease the signal generation time appreciably, but due to the high pressure surges occuring in this type control system, the reversing pressure must be set considerably above the operating pressure. In most cases needle valves must also be used to throttle the fluid flow before it reaches the pressure switch. The presence of a needle valve in a circuit complicates the flow pattern.

The time-delay of the pressure switch controls can be analyzed as follows:

1. The signal generation time, which starts as the piston reaches the end of its stroke, and ends when the pressure switch is activated.
2. The signal transmission time which includes the closing of the latching relay (Fig. 3-5), and energizing of the solenoid valve.
3. The signal application time, which ends when the directional valve spool is shifted past midcenter, and the piston starts to retract.

If needle valves are used in the circuit, the signal generation time is necessarily lengthened because the valves permit only a very small flow rate to the pressure switch, and a certain volume of fluid is needed to activate the pressure switch.

The lower limit of time-delay for the pressure switch control system was found by neglecting the delaying influence of the needle valves. The signal generation time includes the travel of the piston through the cushioned portion of the stroke, and the time of pressure build-up in the system due to compressibility of the fluid. The time for the piston to travel through the cushion action of the cylinder was found by dividing the area of the piston times the length of cushioning by the volume output of the pump. The time for pressure build-up in a closed hydraulic system is according to Ernst (5):

$$t = \frac{V \Delta P}{235{,}000\ Q} \tag{5-18}$$

where $\underline{V}$ is the volume (in.^{3}), $\underline{\Delta P}$ is the pressure rise above operating pressure (psi), Q is pump output (in.^{3}/sec), and t is time (sec). The number 235,000 is a constant applicable to most hydraulic fluids.

The signal transmission and application time was the same as for the limit switch circuit, once the contacts of the pressure switch had been closed, and current was applied to the latching relay. Therefore the time-delay of the electrical controls was found from the manufacturer's catalog of the relays or solenoids used in the circuit. The total time-delay is the sum of Eqs. (5-16, 5-18), the cushion travel time, and the electrical delay-time:

$$t = \frac{V \Delta P}{235{,}000\ Q} + \frac{\pi/4\ D_v^{\ 2}\ L_v}{Q} + \frac{\pi/4(D_c^{\ 2} - D_r^{\ 2})\ L_c}{Q} + \text{electrical delay} \qquad (5\text{-}19)$$

where D_c and D_r are diameter of cylinder and piston rod respectively (in.), and L_c is length of cushion action on the cylinder.

The upper limit of time-delay was calculated by adding to the signal generation time, the time that it takes a certain volume of fluid to flow past the needle valve to operate the pressure switch. At the average pressure between system and sequence pressure the flow rate past the needle valves used was approximately 0.1 gpm. As it takes approximately 0.05 in.^{3} of fluid to operate the pressure switches used in this test, it would take 0.130 seconds to operate the pressure switches. This figure is somewhat on the conservative side as to speed of operation, because this flow would start as soon as pressure started to rise above the normal operating pressure. The addition of this total time to Eq. (5-17) is the upper limit of time-delay for the pressure control system. Table IV shows the data taken from experimental tests on this system; this data is compared to a plot of the analytical analysis in Fig. 5-5.

The pressure switch controls were first tested for time-delay by using a limit switch to reverse the cylinder at one end of its stroke, and pressure switch reversing at the other end. This procedure was necessary because there were insufficient pressure switches available to control both ends of the cylinder stroke and to time the circuit. Needle valves were not needed in this circuit because the use of the one limit switch reduced the pressure surges, but as needle valves were necessary when both reversals were controlled with pressure switches, the time-delay was measured again using needle valves.

Mr. J. Case (8) has developed a special valve that reduced the surges associated with pressure sequencing, but this author and others have had little success with commercial valves and parts when used in pressure controlled circuits. The presence of needle valves in the circuit increases the surges, which increases the possibility of component failure. The loud "banging" noises caused by the pressure surges makes pressure controls undesirable for many applications. Unless special equipment is used in pressure reversing controls, it is believed that their application is definitely limited.

Because of the variation in the length of piston stroke in the various systems tested, it would be an unfair comparison to plot the cycles per minute of one control system and compare it directly to the cycles per minute of another system with a shorter stroke. For this reason the measured value of cycles per minute will be multiplied

TABLE IV

PRESSURE SWITCH CONTROL

Pump speed (rpm)	Cycles per minute	Shift pressure (psi) Left	Shift pressure (psi) Right	Average shift time (sec)	Theorectical cycles per minute	Pump output (gpm)	Timing signal Pressure (psi) "Start"	Timing signal Pressure (psi) "Stop"	System Pressure (psi)
498	26.2	1275	1375	0.213 (0.138)[2]	32.1	10.2	290	430	300
649	32.4	1275	1375	0.195 (0.122)[2]	42.4	13.5	290	430	300
809	39.4	1275	1375	0.172 (0.103)[2]	53.4	17.0	375	430	400[1]
959	43.5	1275	1375	0.149 (0.085)[2]	63.5	20.2	350	430	400[1]
484	25.2	1275	1375	0.137 (0.103)[2]	30.3	9.8	700	430	685
650	32.0	1275	1375	0.138 (0.094)[2]	41.5	13.4	675	430	685
802	35.7	1275	1375	0.125 (0.085)[2]	51.6	16.7	650	430	690
935	40.3	1275	1375	0.123	60.6	19.6	675	430	690

[1] Minimum setting available with back pressure valve.

[2] Time in parentheses was taken without needle valves in test circuit.

Cycles per minute were measured with shift pressure set at 790 psi.

Stroke length was constant at 13 7/8 inches.

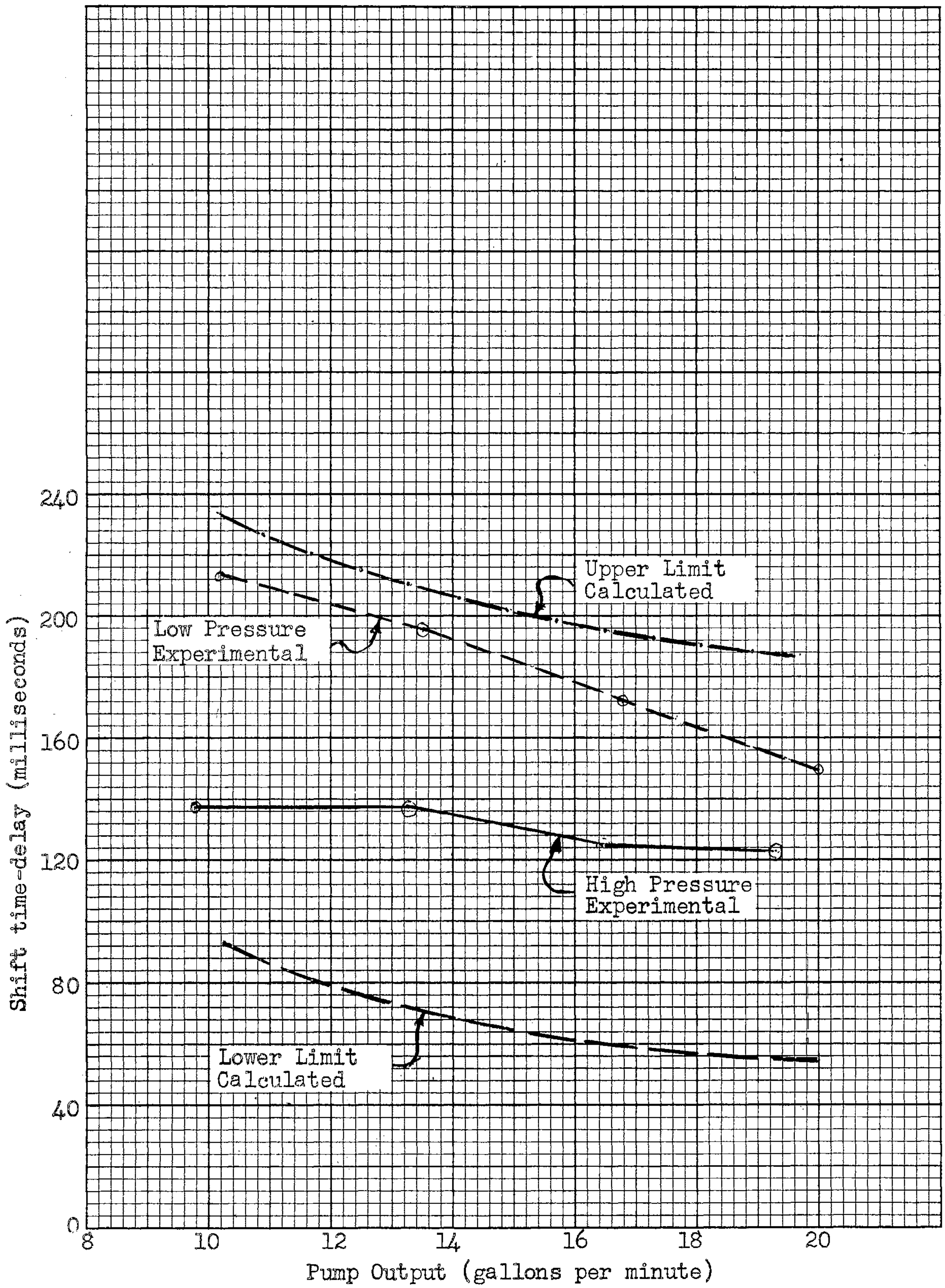

Fig. 5-5. Experimental and analytical curves of pressure switch control system.

by the ratio of measured stroke length to maximum stroke length. The results of this analysis are plotted in Fig. 5-6 and compared to the optimum cycles per minute considering zero shift time. It may be noted that the time-lag of the mechanical system was longer than either that of the cam valve or the limit switch control systems, yet the cycles per minute of the mechanical system is larger than the cycles per minute of the limit switch controls. This seeming contradiction is explained by the fact that the entire travel of the valve spool was timed in the mechanical system, while only a portion of the valve spool travel was timed in the cam valve and limit switch systems. It was assumed in the derivation of operating time for the cam valve system that the cylinder started reversal as soon as a portion of the valve port was opened to the cylinder.

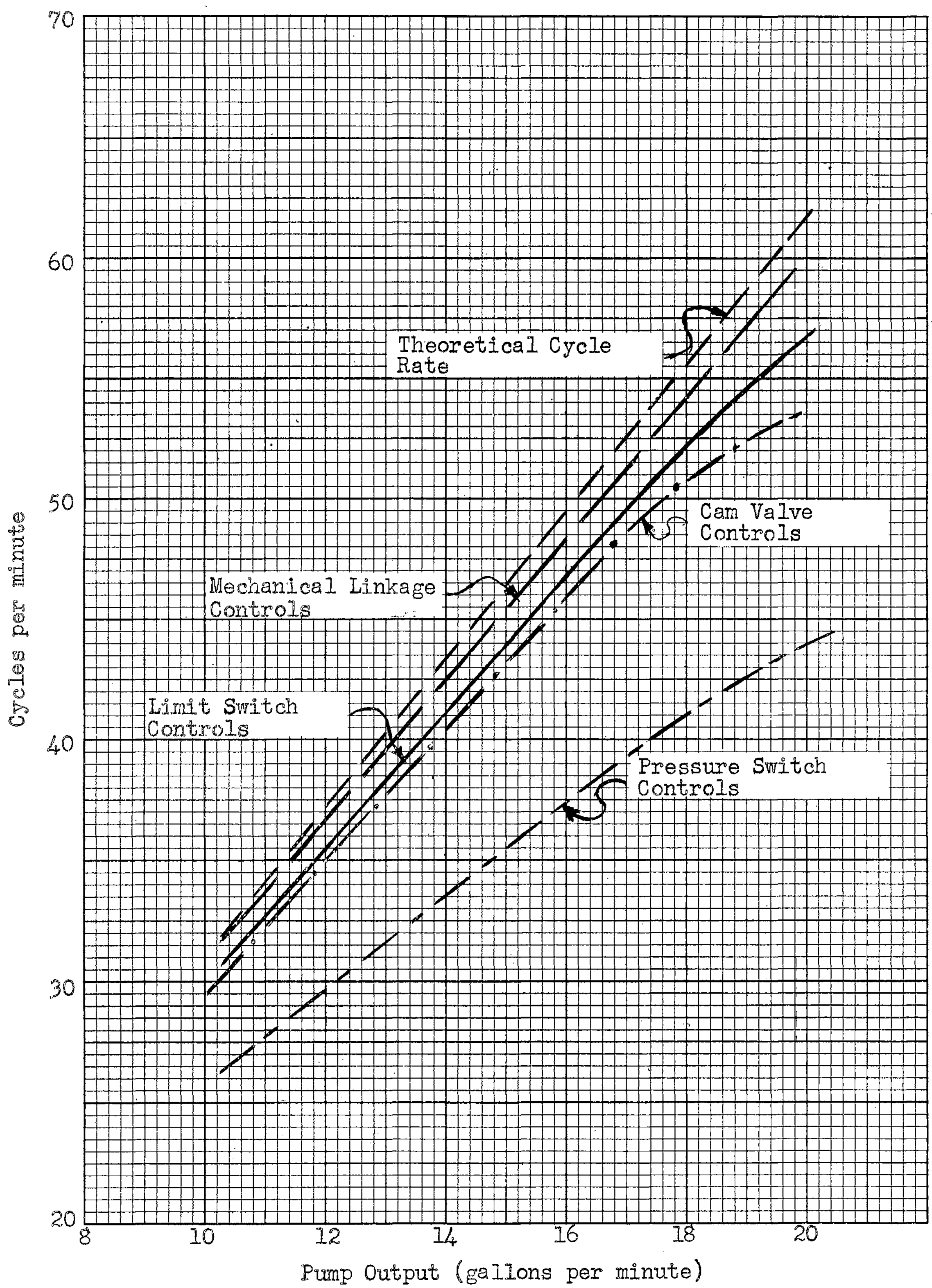

Fig. 5-6. Comparison of mechanical linkage, cam valve, limit switch, and pressure switch controls to theoretical cycle rate.

CHAPTER VI

SUMMARY

This investigation purported to determine the shift time of several types of automatic reversing systems, and to develop equations that would predict this reversal time. It was the first known effort to time the reversal itself, so the measured shift time of a few milliseconds was noteworthy.

Equations were developed that would predict the operation time of each control system within certain limits. It is believed that these equations are applicable to similar systems, with certain restrictions. They will at least enable the designer to make an intelligent estimate of shift time in a hydraulic circuit. The limit switch, cam valve, and mechanical control systems are approximately equal in speed of operation, and reasonably free of pressure surges. The cam valve controls are simple in construction and design, and give the most trouble free operation.

CHAPTER VII

RECOMMENDATIONS FOR FUTURE RESEARCH

Although this investigator could not always predict the shift time as accurately as hoped, and though there is some doubt as to the accuracy of a few points (this doubt arises mainly from the expected shape of the curves) he believes that the results are within the accuracy needed at present. It is only in servomechanisms that a more accurate accounting of operation time is necessary. Moreover it is believed that tests of reversal time can be applied easily to other systems, but because of necessarily small variations in O-ring drag, clearance space, and valve spool design, complete reproducibility of results should not be expected.

However, if it is desired to extend these tests in the future, much could be accomplished by refined testing techniques. Other more accurate timing devices possible are high speed movie cameras (in the range of a thousand frames per second), photo-sensitive cells, and strain gages. A strain gage could be placed on the cam valve push rod, on the limit switch cam follower, or on the pressure switch piping to act as a "start" signal for the timing mechanism. The timing mechanism itself could be an electronic "flip-flop" circuit, an electronic counter, or an oscilloscope with a timing wave imposed on the scope. The "stop" signal could be another strain gage placed on either the end cap of the valve or on the piping leading to the cylinder. The first position would pick up a signal from the valve spool hitting the end cap, the second would pick up the pressure to the other end of the cylinder.

It might also be desirable in the future to more fully investigate the effect of pressure on shift time. This investigation would be quite difficult with a slow shifting control because of pressure surges, but could be applied to limit switch or cam valve controls.

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APPENDICES

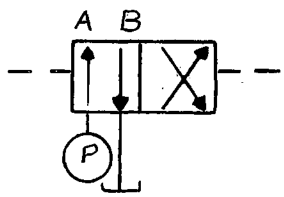
APPENDIX A

DEFINITION OF COMPONENTS

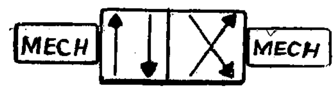
As the test stand with its various components was adequately detailed in a previous thesis (2) it will be described only as a fluid source capable of delivering oil at pressures up to 1500 psi, at flow rates as high as 24 gpm, with suitable manifolding and return lines; and as a test board to which various valves and cylinders could be attached.

The valves and cylinders which were actually used in controlling each circuit are shown in symbolic representation. A description of each part is included with symbols used in accordance with the proposals of the American Standard Association.

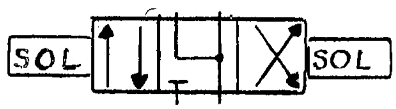
A directional control valve is used to direct the flow of hydraulic fluid to various parts of the system. In operation, this valve requires some type of motive force to move the spool which in turn changes the direction of flow. The motive force can be mechanical, manual, electrical or hydraulic in orgin. These valves may have several main line connections, and usually have two or three positions. A graphical representation of the directional control valves used in this investigation is shown in Fig. A-1, with the ports connected to the normal unactivated position. When the spool is shifted to the extreme left, port A is open to pressure, port B, to tank. In the extreme right position port B is open to pressure, port A, to tank. Only the solenoid valve shown in Fig. A-1c, used a neutral position, and flow through the valve is shown by the connection of the lines in the center square.



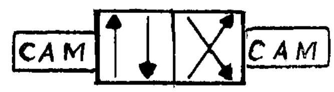
a. Directional control valve, fluid control.



b. Mechanical control.



c. Solenoid control.



d. Cam control.

Fig. A-1. Directional control valves, and their operating forces.

The method of applying the controlling force to the valve is shown by dashed lines in the case of hydraulic control, and by appropriate abbreviations in a control rectangle in the case of mechanical or solenoid control.

The symbol for a variable restriction such as a needle valve is shown in Fig. A-2. The "V" within the enclosure shows that the restriction is variable. The symbol for a double-acting, double rod cylinder is shown in Fig. A-3. A connection to a tank or to the oil reservoir is shown in Fig. A-4a. If the inlet line touches the tank, as in Fig. A-4b, the fluid is released beneath the surface of the oil; Fig. A-4c shows the fluid being released above the surface of the oil, as for a pilot line.

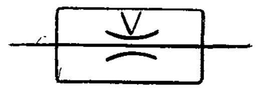


Fig. A-2. Variable restriction.

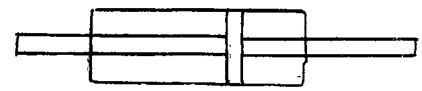
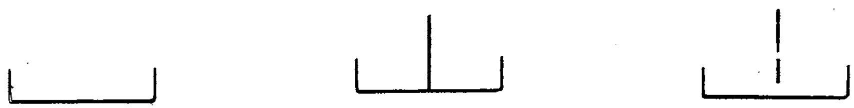


Fig. A-3. Double-acting, double rod, cylinder.



a. Tank symbol. b. Inlet below surface. c. Inlet above surface.

Fig. A-4. Tank symbol, with various types of connections.

The symbol for a pressure switch is shown in Fig. A-5. The dashed line indicates a fluid line, the dash-dot lines indicate electrical connections.

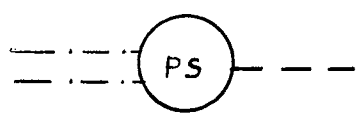


Fig. A-5. Pressure switch.

A counterbalance valve is a pressure-control valve which permits flow at a desired minimum-pressure level in one direction and free flow in the other direction. Fluid flow in one direction is blocked until the desired level of back pressure is attained, at which time the valve opens to permit fluid passage. Flow continues through this valve until

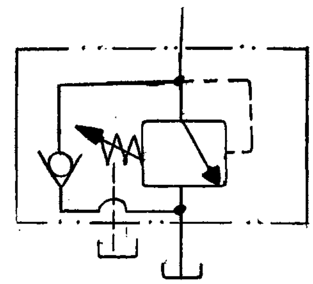


Fig. A-6. Counterbalance valve, shown in component enclosure.

the back pressure of the system drops below the set minimum valve at which time the valve closes. Flow through the counterbalance valve in the opposite direction is unrestricted. This valve is capable of maintaining the back pressure at any of a large range of values. Its primary function in this investigation was to simulate a load on the cylinder in order that the effect of back pressure on shift time-delay could be determined. A counterbalance valve is shown in Fig. A-6.

APPENDIX B

SAMPLE CALCULATIONS

A time study of the cam valve control system for a complete cycle will demonstrate the use of the equations given in the body of the thesis.

At the pump speed of 643 rpm the volume output was 13.3 gpm. The measured time for one cycle was 60 seconds divided by 40.6 cycles per minute equal to 1.478 seconds per cycle.

The calculated time is found from twice the cylinder volume divided by the pump output, plus two times the shift time-lag. (Both figures are multiplied by two because the piston travels the complete length of the cylinder in both directions, and the directional control valve is shifted twice in one cycle). So,

$$t = \frac{2 \cdot \pi/4(D_c^2 - D_r^2)L_c \cdot 60}{13.3(231)} + \frac{2 \cdot \pi/4\, D_v^2\, L_v \cdot 60}{13.3(231)}$$

$$t = \frac{\pi/4(2^2 - 0.75^2)13.25 \cdot 120}{13.3(231)} + \frac{\pi/4(1.187)^2(0.437)(120)}{13.3(231)}$$

$$t = 1.395 + 0.0189 = 1.414 \text{ seconds}$$

The error is $\dfrac{(1.478 - 1.414)\,100}{1.478} = 4.3\,\%$

APPENDIX C

APPARATUS AND EQUIPMENT

Pump Circuit

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

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

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

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

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

Cylinder Control Circuits

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

2. Directional Control Valve: Manufacturer, Double A Products Co.; model no., R6-180-C; type: manually operated, four-way, three-position, detent, all ports blocked in the neutral position; port size, 1 inch; (neutral position detent was blocked by filling groove in valve spool).

3. Counterbalance Valve: Manufacturer, Double A Products Co.; model no., SA2-180-B; type, internal pilot and external drain; pressure range 250 to 500 psi; port size, 1 inch.

4. Directional Control Valve: 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.

5. Latching Relay: Manufacturer, Potter and Brumfield; serial no., EL 5AH; operation, solenoid, with mechanical hold; 110-125 volts, 60 cycle AC.

6. Limit Switch: Manufacturer, Microswitch Corp; catalog no., WZE 7RQ9 TN; capacity, 10 amperes at 125 volts.

7. Pressure Activated Switch: Manufacturer, Barksdale Valve Co; range, 250-3000 psi; serial no., 9612-3-H; port size, 1/4 inch; switch capacity, 10 amperes at 125 volts.

8. Pressure Activated Switch: Manufacturer, Meletron Corporation; model no., 1510-50; range, 2650-3050 psi (adapted for lower pressure); port size, 1/4 inch; switch capacity, 10 amperes at 125 volts.

9. Pressure Activated Switch: Manufacturer, MechaniArts Associates; catalog no., 9L-K45-B1; range, 250-1500 psi; port size, 1/4 inch; switch capacity, 10 amperes at 125 volts.

10. Directional Control Valve: 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, Ashcroft; pressure range, 0 to 5000 psi.

3. Pressure Gauge: Manufacturer, U. S. Gauge Co; pressure range, 0 to 1500 psi.

4. Electronic Counter: Nanufacturer, Berkley Division of Beckman Instruments Inc; model no., 410.

5. Square-wave Generator: Manufacturer, Du Mont; type, 185-A.

6. Dead Weight Tester: Manufacturer, Ashcroft Gauge Division of Manning, Maxwell and Moore, Inc; type no., 1313A.

VITA

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Candidate for the Degree of

Master of Science

Thesis: TIME-DELAY OF VARIOUS TYPES OF HYDRAULIC REVERSING SYSTEMS

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