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ABSTRACT

This final report includes the results of a research program (EWA 6801) for the investigation and understanding of pulsating hydraulics. The research investigation consisted of three phases:

a. An understanding of the theory and the present state-of-the-art of pulsating hydraulic systems.

b. Development of the basic techniques for analysis of pulsating hydraulic systems and simulation of a single-line pulsating hydraulic system on the analog computer.

c. Application feasibility of pulsating hydraulics to the B-52 hydraulic systems.

A literature review and letter survey was accomplished to obtain an understanding of the theory and the present state-of-the-art of pulsating hydraulic systems. The results of the literature review and letter survey revealed that the literature available on pulsating hydraulics was extremely limited and little research has been done in this area. The basic analytical techniques were developed and used in simulating a single-line pulsating hydraulic system on the analog computer. During the research program, 47 different system configurations were evaluated. These configurations were evaluated for the change in system efficiency with respect to changes in the basic parameters of line lengths, line sizes, and pulsation frequencies. The pulsating hydraulic systems efficiencies were lower than those of continuous flow hydraulic systems. The B-52 application study concluded that the B-52, as designed, is not readily adaptable to pulsating hydraulics application. The present systems are functioning adequately and no real improvement in reliability or weight can be forecast.

As a result of the research investigation, it was concluded that for certain applications, pulsating hydraulic systems offers a definite advantage over continuous flow hydraulic systems. It is in these areas of application that further studies are recommended to define the efficient use of pulsating hydraulics.

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SECTION	1.0

SECTION TITLE: INTRODUCTION

1.0 INTRODUCTION

Fluid power has long been used by man. Sailboats were in use approximately 5000 years ago, and there is evidence that the first water wheels were built between 200 and 100 B.C. The conventional hydraulic systems for transmission of power as we know them today are relatively new. The outbreak of World War II forced an enormous acceleration of research and development effort, aimed at the rapid evolution of armament and strategy. Mechanisms with very high speeds of response were needed to permit the rapid increase in the speeds of operation of military aircraft and automatic fire-control systems. In both of these fields, hydraulic systems proved to be particularly useful because of their very fast response and their great stiffness as seen by the load.

Transmission of power through a fluid medium by means of waves is an old hope -- a basic treatise on the subject was written some 30 years ago in England. Little research, however, has been done in this area. Apparently the possible advantages of "pulsating hydraulics" did not outweigh the limitations, and it was bypassed by more conventional fluid-power systems. However, the many advantages promised by the successful application of "pulsating hydraulics" may partly explain why the subject refuses to fade from consideration.

This document presents the results of a research program (EWA 6801) for the investigation and understanding of pulsating hydraulics. The basic approach of the above program was as follows:

a. Develop an understanding of the theory and the present state-of-the-art of pulsating hydraulic systems.

b. Develop the basic analytical techniques for pulsating hydraulic systems and simulate a basic system on the analog computer.

c. Determine the application feasibility of pulsating hydraulics to the B-52 hydraulic system.

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NUMBER	D3-6576

SECTION	2.0

SECTION TITLE: SUMMARY

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2.0 SUMMARY

This section contains the results of the research program (EWA 6801) for the investigation and understanding of pulsating hydraulics. The results of the program are discussed in the following three parts:

a. Literature review and letter survey.

b. Development of the basic analytical techniques for pulsating hydraulic systems and simulation of a single-line pulsating hydraulic system on the analog computer.

c. Application feasibility of pulsating hydraulics to the B-52 hydraulic system.

The literature available on hydraulic power through the use of pulsating pressure or pulsating flow is extremely limited. The literature review concluded that only two companies (Republic Aviation Corporation and Vickers Incorporated) have accomplished significant research work in this field. The work by Republic, References 1 and 3, concluded that pulsating flow means were suitable for power transmission, whereas the pulsating pressure means might be adapted for signal transmission.

A letter survey was initiated encompassing universities, industry, and government agencies to obtain further information concerning pulsating hydraulics. A total of 55 inquiries were made and resulted in a response of 58 percent. The results of the inquiries showed that relatively little research has been accomplished on pulsating hydraulics.

The basic analytical techniques were developed and used in simulating a single-line pulsating hydraulic system on the Pace 231-R analog computer. During the research program, 47 different system configurations were evaluated. These configurations were evaluated for the change in system efficiency with respect to changes in the basic parameters of line lengths, line sizes, and pulsation frequencies.

The system efficiencies, calculated upstream of the alternator valve, varied from 20 percent to 60 percent. This method includes the losses through the alternator valve and the losses due to the high pressure drop between the pulsating flow and the reservoir. These values of system efficiency are quite low when compared with the efficiency of continuous flow hydraulic systems. The system efficiencies, calculated downstream of the alternator valve, varied from 55 percent to 98 percent. This method does not include the losses through the alternator valve and the losses due to the high pressure drop between the pulsating flow and the reservoir. It appears that the losses through the alternator valve and overboard losses due to the reservoir are extremely large. For the single-line pulsating hydraulic system, a pure pulsating system (direct conversion from mechanical energy to pulsating hydraulic energy) is recommended to overcome these losses and make them comparable with continuous flow hydraulic systems.

The application feasibility study concluded that the B-52, as designed, is not readily adaptable to pulsating hydraulics. The present systems are functioning adequately and no real improvement in reliability or weight can be forecast.

The investigations of pulsating hydraulic systems that have been performed to date, particularly the system simulation on the analog computer, indicate certain areas where further effort is desirable. The simulation techniques should be improved (especially the transport delay circuit) by better programming and by utilization of a hybrid computer. Using the improved simulation techniques, design criteria should be developed by investigating the following areas:

a. Transmission line phenomena (resonance, time delay, efficiency, etc.).

b. Pulse generation methods to increase system efficiency.

c. Pulse generation control (feedback).

d. Multi-fluid systems.

e. Component selection.

There are many areas of application where pulsating hydraulic systems seem advantageous and should be investigated. It is further recommended that, dependent on the results of the above investigations, a single-line pulsating hydraulic system be designed, constructed, and tested to verify the computer and analytical results.

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SECTION 3.0

SECTION TITLE: LITERATURE REVIEW AND THEORETICAL BACKGROUND

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3.0 LITERATURE REVIEW AND THEORETICAL BACKGROUND

3.1 Introduction

This section contains a review of the literature and theoretical background concerning the transmission and use of hydraulic power through pulsating pressure or pulsating flow concepts. Pulsating flow is defined as a flow in which the volumetric flow rate is a periodic function of time. The time average of the volumetric flow rate may be zero or non-zero. Thus, there are two limiting cases of pulsating flow. One limit is steady flow for which the amplitude of the pulsations is zero. The other limiting case is purely oscillating flow for which the time average of the volumetric flow rate is zero. All intermediate types of pulsating flows may be obtained by the superposition of these two limiting cases. The pulsating flow hydraulic systems investigated in this study uses purely oscillating flow for which the time average of the volumetric flow rate is zero.

The literature available on pulsating flow concerns primarily the following topics:

- Pulsating flow measurements.
- Heat transfer studies of pulsating flow.
- Investigation of pulsating flow in rigid and flexible tubes with respect to stability and critical Reynolds' numbers.
- Pulsating flow of blood in the arteries of man and other animals.

The literature on hydraulic power through the use of pulsating pressure or pulsating flow is limited since this is a relatively new field of research.

3.2 Literature Review - Results

An extensive literature review conducted at the beginning of the research program yielded only two references concerning the use of pulsating flow for power transmission. Since these articles are considered as the starting point of this investigation, a thorough summary of each is included. Five references obtained after the initial study are briefly summarized in 3.2.3 through 3.2.7.

3.2.1 Research Investigation of Hydraulic Pulsation Concepts (Reference 1)

This report includes a program in which Republic made a research investigation of the technical feasibility of the transmission and use of hydraulic power through pulsating pressure or pulsating flow concepts (USAF Contract AF33(657)-10522). The pulsating flow system was selected as being more desirable than a pulsating pressure system for the transmission of hydraulic power. Wave systems are believed to be more suitable for control signal transmission than for large power transfer. In power transfer, changes of input impedance of the output devices or of fluid properties resulting from a large amount of power variation will not only change the amplitude and wave length of the standing pressure wave continuously but may destroy this wave completely. The research investigation was carried on by:

- The design effort, and
- The analytical method.

The basic approach to system design was evaluated as follows:

a. Pure pulsating system in which the mechanical energy is converted directly to hydraulic energy in a pulsating form (Figure 1). For a system other than a small actuator or motor system, the weight and design penalties that must be accepted proves that such a system is impractical.

b. Combined system in which the mechanical energy is converted to hydraulic energy in a continuous flow form and then altered to pulsating flow (Figure 2).

The transmission of the pulsating flow energy through the plumbing lines of a pulsating hydraulic system may vary from a single-line to a multi-line system. Figures 1 and 2 illustrate a single-line pulsating system and a three-line pulsating system.

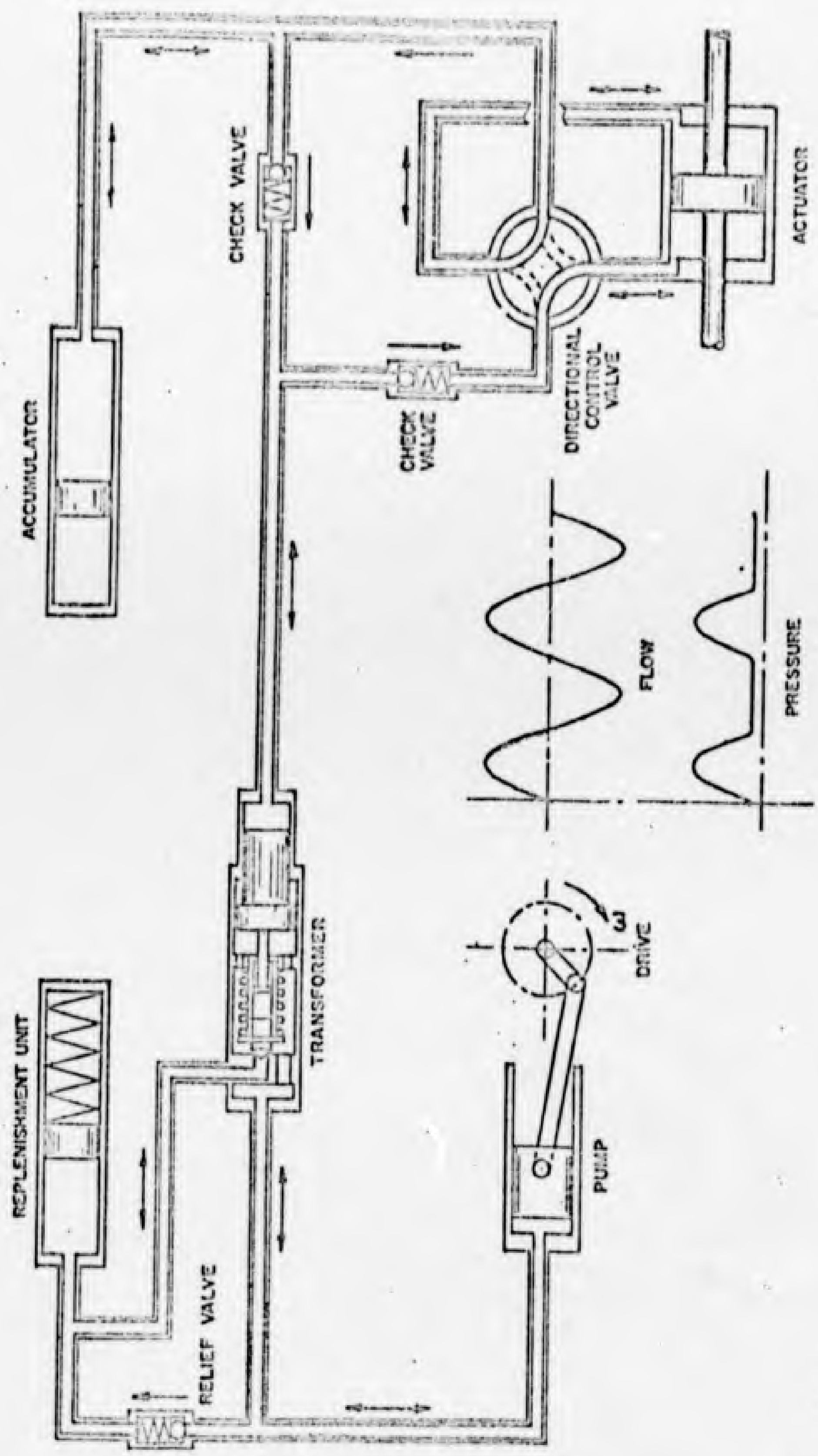
Another comparison was made between pulsating hydraulics systems with constant pressure sources and constant flow sources. For analytical representation, a system with a constant pressure source is defined as one in which the pressure at the input tends to remain constant and the flow varies on demand. This system is exemplified by one in which a pressure-compensated variable delivery pump is used and the pulsations are caused by the introduction of an alternator valve.

A constant flow source is one where pulsating energy is generated by a reciprocating cylinder or similar means. In this system, the flow is a function of the driving frequency and stroke of the pumping cylinder.

A pulsating hydraulic system with a constant pressure source has more advantages than a pulsating hydraulic system with a constant flow source. The following is a comparison of the two systems; the advantages of one system are the disadvantages of the other:

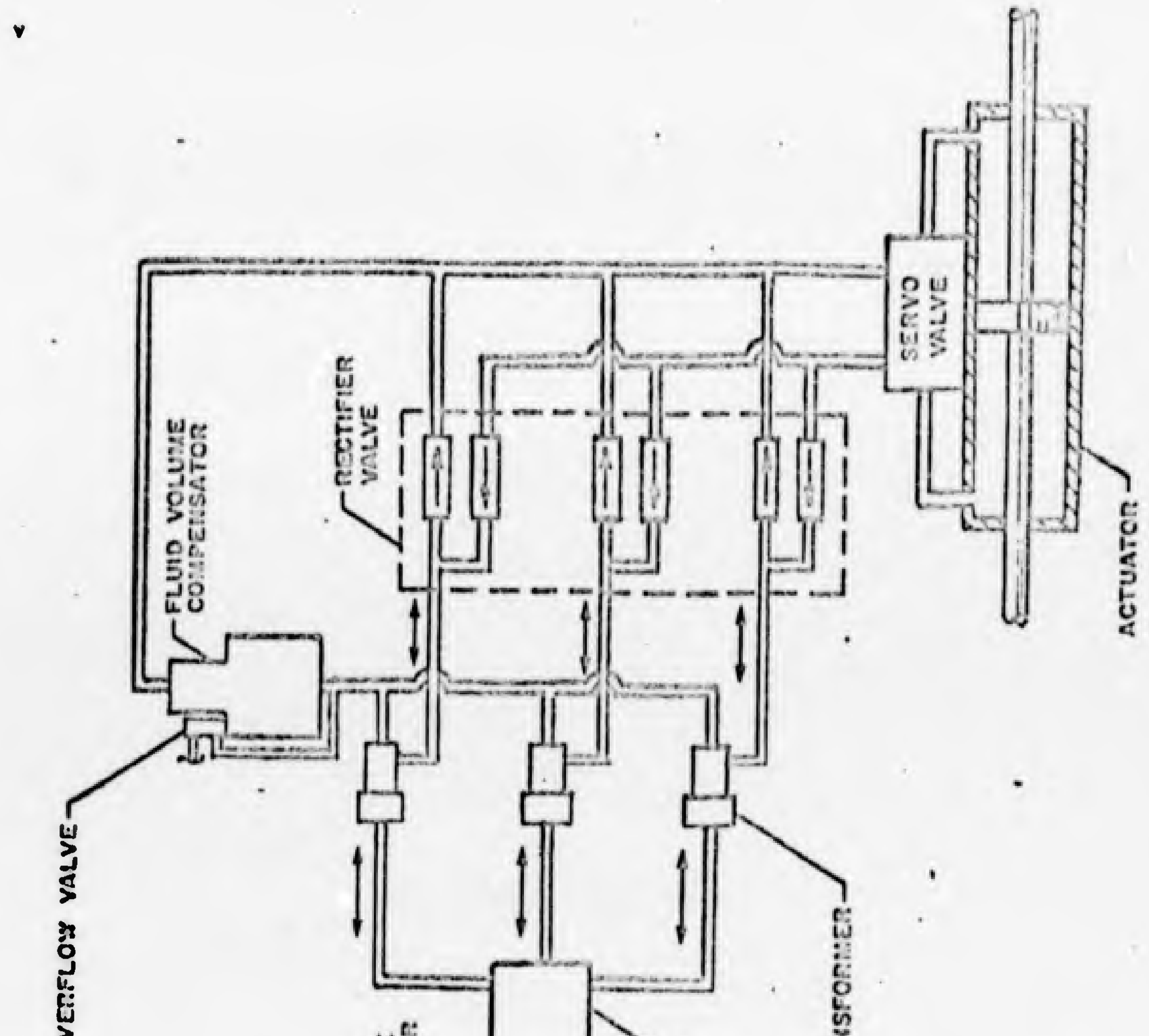
a. Advantages of Systems with a Constant Pressure Source:

(1) Small power is required to drive the pulsating valve; hence, the pulsating frequency is more easily controlled and maintained.



SINGLE-LINE PULSATING HYDRAULIC SYSTEM

FIGURE 1



THREE-LINE PULSATING HYDRAULIC SYSTEM

FIGURE 2

(2) A commercial pressure compensated variable delivery pump can be used in combination with a pulsating valve as the hydraulic power unit of the system. No new pump development is necessary.

(3) Pressure pulses traveling back and forth in the fluid line are nearly square-wave shaped. For the same power transmitted through the line, the maximum pressure level in the line is thus less than that of the fluid line of a system with a flow source.

(4) Depending on the characteristics of the pulsating valve and pump, the flow pulses traveling back and forth in the fluid line can also be made close to square-wave shape, thus reducing the peak flow Reynolds' number. This is important when the Reynolds' number of the flow is near or larger than the critical Reynolds' number.

b. Advantages of Systems with a Constant Flow Source:

(1) No pulsating valve and associated driving mechanisms are needed. Therefore, the system has fewer components.

The investigation indicated that the pulsating flow means were suitable for power transmission. For certain applications, pulsating hydraulics systems have a definite advantage over continuous flow hydraulic systems. Pulsating hydraulics have an advantage over continuous flow hydraulics in the areas of fluid separation or isolation, pressure variation efficiency, and motion or speed synchronization. The following results were obtained during the investigation:

a. Fluid Separation or Isolation

For a continuous hydraulic system, fluid separation or isolation requires additional elements. This results in a component with great functional complexity and reduced reliability. In a pulsating system, the transformer is a simple barrier unit with a diaphragm or piston separator that transmits the energy of the system. The applications or usage for fluid separation are as follows:

(1) Isolation of nuclear radiation.

(2) Wide range of ambient temperatures (especially high temperatures). In this system, the transformer acts as as a separation point between the high temperature fluid of the downstream portion of the system and the lower temperature upstream fluid. Permits use of fluids which are more compatible with the thermal range encountered which results in:

(a) More reliability of components.

(b) Increase efficiency by elimination of special considerations for maintaining a specific temperature range (as bypasses in valves and actuators, air conditioning units, etc.).

(c) Weight saving -- the transformer will be located in a relatively low temperature area and the heat from the high temperature portion of the system will not be transmitted throughout the entire system. As a result, most of the system components can be fabricated of lighter weight and easily machined materials.

b. Pressure Variation Efficiency

Pulsating hydraulics systems has an advantage over continuous flow systems by changing pressure in the system with a transformer. In continuous hydraulics systems, a valve is used to reduce pressure. This valve forms heat and decreases the system efficiency. Pressure increase in a continuous flow hydraulic system requires a complex component and is, therefore, seldom used in aerospace system design.

c. Motion or Speed Synchronization

The alternator valve designed for pulsating hydraulics systems provides an accurate flow division and may save weight over mechanical synchronization devices.

d. Weight

No conclusions can be made on which type of hydraulic system (continuous or pulsating) has the weight advantage. Each hydraulic system must be analyzed and in some cases, a continuous flow hydraulic system may be superior to a pulsating system, while for other cases the opposite is true.

e. Efficiency

The same general remarks previously made also apply here. The system efficiency is sensitive to the system load impedance. For the same load impedance, the system efficiency is rather insensitive to the change of pulsating frequency and flow amplitude. To determine the best system pulsating frequency and flow amplitude, parameters that include pulsating flow line input impedance, resonance conditions, number and size of pulsating flow line, etc., rather than system efficiency, become the deciding factors.

f. Strength

In the detail design of pulsating components, the fatigue effects of cyclic stress loading due to pulsation must be considered. However, the strength requirements are reduced since many components of a pulsating hydraulic system will not be subjected to the high temperature portion of the system.

g. Reliability

The pulsating hydraulics system will result in an increase in the system reliability and a decrease in maintenance requirements through system cleanliness and freedom from contamination. Since the pulsating portion of the system does not flow through the transmission lines, there will be no transfer of contaminants from one portion of the system to another. Therefore, each pulsating component must be tolerant only of self-generated contaminants. The pump is the main contaminant-generating component in a hydraulic system. Since the pump is isolated from the pulsating system, its contaminants will not reach any critical component. The small and compact continuous flow loop located downstream of the rectifier valve can be brought to the desired cleanliness level during assembly and filling. The installation of filters should be unnecessary.

3.2.2 A-F Hydraulics - Russ Henke (Reference 2)

This article contains a brief description and terminology of alternating-flow (A-F) hydraulics. Momentum can be carried through a conductor filled with fluid by two general means:

- By translation of the medium in the conductor (gross motion).
- By compression waves.

Direct-flow (D-F) systems transfer momentum and energy by gross motion of the fluid. There are two general modes of operation for A-F hydraulic systems:

- Vibratory, where gross, but rapid, motion takes place.
- Wave mode, where a standing wave is generated in a single closed conductor.

The following advantages promised by the successful application of A-F hydraulics explains why the subject refuses to fade from consideration:

- No return lines are necessary.
- Reservoirs and their store of fluid could be eliminated.
- Possible reduction in size of pumps and motors.
- Control valves would not have to handle the large volumes of oil currently demanded by some D-F hydraulic systems.
- Control signals could be superimposed on the basic power signals and eliminate electrical controls entirely.

A-F systems may be important in the future. Development waits for necessity as has occurred in so many other areas of fluid power technology -- for example, servo systems, high-pressure systems, etc.

3.2.3 Investigation of Pulsating Flow Hydraulic Concepts (Reference 3)

This report summarizes the work performed by Republic from May 1964 to July 1964. They investigated concepts involved in the conversion of continuous flow hydraulic power into pulsating flow, and its transmission, transformation, and reconversion into continuous flow. Components peculiar to pulsating flow hydraulic systems were designed, fabricated, tested, and combined into a pulsating flow hydraulic system. Also, comparative analyses were made of typical pulsating flow applications for vehicle systems to define the operational and weight advantages of pulsating flow systems.

The results of the pulsating flow system operating tests definitely indicated the feasibility of the transmission of hydraulic power by pulsating means. For the system tested (50 feet long with outside diameters of 1/2 inch and 5/8 inch, respectively), there was little variation in overall system efficiency at pulsation frequencies ranging from 3 cps to 20 cps without a transformer and from 3 cps to 15 cps with a piston type transformer in the system. Tube line vibration is the most serious problem encountered during the test. The vibration was of relatively low amplitude at 3 cps pulsation frequency and became proportionally severe with increased pulsation frequency.

In the comparative system analyses, pulsating flow applications were studies for several specific vehicle installations. These studies indicated that pulsating hydraulics offered a definite vehicle weight advantage when applied to properly selected services.

3.2.4 Study of the Distributed Parameter Hydraulic Line (Reference 4)

This report is the result of a study conducted on a distributed parameter hydraulic line for simulating such a line on the analog computer. The report presents a method of developing and representing the distributed viscous drag, inertial, and compressibility effects of hydraulic fluid flowing through a straight section of tubing. The hydraulic transmission line was made analogous to an electrical transmission line, and the methods used in analyzing electrical lines were used for analyzing hydraulic lines.

3.2.5 Tests of a Capped Hydraulic Line for Distributed Effects (Reference 5)

Tests were conducted in the laboratory to obtain a correlation of the actual hydraulic line test data to the distributed parameter theory developed in Reference 4. The test data were obtained from a section of Number 8 steel hydraulic tubing, capped at one end with the other end coupled to a sinusoidal varying pressure source. This particular hydraulic tubing connection is analogous to an open circuit electrical transmission line.

The results of the correlation provided information for the sinusoidal steady state theory and information for use in the future simulation of hydraulic lines on the analog computer.

3.2.6 Preliminary Study of Non-Steady Flow Hydraulic (Reference 6)

This report is concerned with the description of a preliminary study of non-steady hydraulic flow concepts and applications. This report is concerned with sinusoidal or semi-sinusoidal flow or pressure variations. The objectives of this study were:

- Define the general area of interest.
- Establish concepts of flow or pressure generators, motors, transmission lines, and systems.
- Determine advantages available from concepts developed in this study.
- Provide some applications of components or systems which can be evaluated in detail.
- Determine the problem area.

No calculations or tests were made to determine the feasibility of these concepts.

3.2.7 Pulsating Hydraulics Study (Reference 7)

This document is a proposed program for research and development of pulsating hydraulics concepts. A basic research program is proposed to accomplish the following:

a. Determine the resistance factors for pulsating flow in transmission lines by analytic and experimental means over a broad range of frequencies and amplitudes. This evaluation will include the laminar and turbulent regimes under alternating flow conditions.

b. Investigate synchronous and asynchronous modes of transmission of power, and transformers for pulsating flow hydraulics. Breadboard experimental units will be built and evaluated.

3.3 Letter Survey

A program was instigated to encompass universities, industry, and government agencies to obtain further information concerning pulsating hydraulics. This program consisted of a letter and questionnaire sent to those organizations which are engaged in hydraulic research programs. The results obtained from the questionnaires are tabulated in Appendix A.

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SECTION TITLE: SYSTEM DESIGN

4.0 SYSTEM DESIGN

4.1 Introduction

Transmission of power through a fluid medium by means of waves is an old hope -- a basic treatise on the subject was written some 30 years ago in England. Several different methods or techniques are available for transmission of power using pulsating hydraulic concepts; however, little research in pulsating hydraulic systems has been made.

The basic concepts, types of systems, advantages and disadvantages offered by a pulsating hydraulic system has been summarized in Section 3.0, Literature Review and Theoretical Background. For a more detailed discussion of pulsating hydraulic concepts, References 1, 2, and 3 are recommended.

4.2 Pulsating Hydraulic System for Analog Computer Simulation

Momentum can be carried through a conductor filled with fluid by two general means:

a. By translation of the medium in the conductor (direct flow or pulsating flow system).

b. By compression waves (pulsating pressure system).

The pulsating flow system was selected for our simulation studies because it is more desirable than a pulsating pressure system for the transmission of large amounts of power.

Two basic approaches to system design are available using the pulsating flow concepts:

a. Pure pulsating system -- direct conversion from mechanical energy to pulsating hydraulic energy.

b. Combined system -- the energy is converted to hydraulic energy in a continuous flow form and then altered to pulsating flow.

The combined system was selected for our simulation studies using an alternator valve to convert continuous flow to pulsating flow. A commercial pressure compensated variable delivery pump was used in combination with an alternating valve as the power unit of the system. This type of system eliminates the need for development of a new pump. A discussion of system efficiency for the above two basic systems (pure pulsating and combined) is presented in Section 6.0, Test Results.

This research program was directed towards B-52 application which contains transmission of fluids at high rates and long distances. A single-line pulsating system was selected for the transmission of the pulsating flow energy because it appears more advantageous for B-52 application. The single-line system appears to offer better reliability (less leak points) and a weight saving (no return line) for the long transmission distances. Therefore, the pulsating hydraulic system used for the analog computer simulation was a single-line combined pulsating flow system. A schematic drawing of the pulsating hydraulic system is shown in Figure 3 and a block diagram is shown in Figure 4. The pressures and flows throughout the system are identified by the subscripts on the block diagram.

The single-line pulsating hydraulic system was simulated on the analog computer from the alternator valve downstream. The direct flow hydraulic system upstream of the alternator valve was not simulated during this research program.

4.2.1 System Components

a. Alternator Valve -- A component which converts continuous flow to pulsating flow.

b. Lumped Parameter Transmission Line -- Transmission line parameters are assumed to be concentrated at single locations (lumped) for analysis and their interactions are assumed negligible.

c. Transformer -- A component which transforms pulsating pressure and flow from one amplitude to another. It also acts as a barrier or separator between fluids.

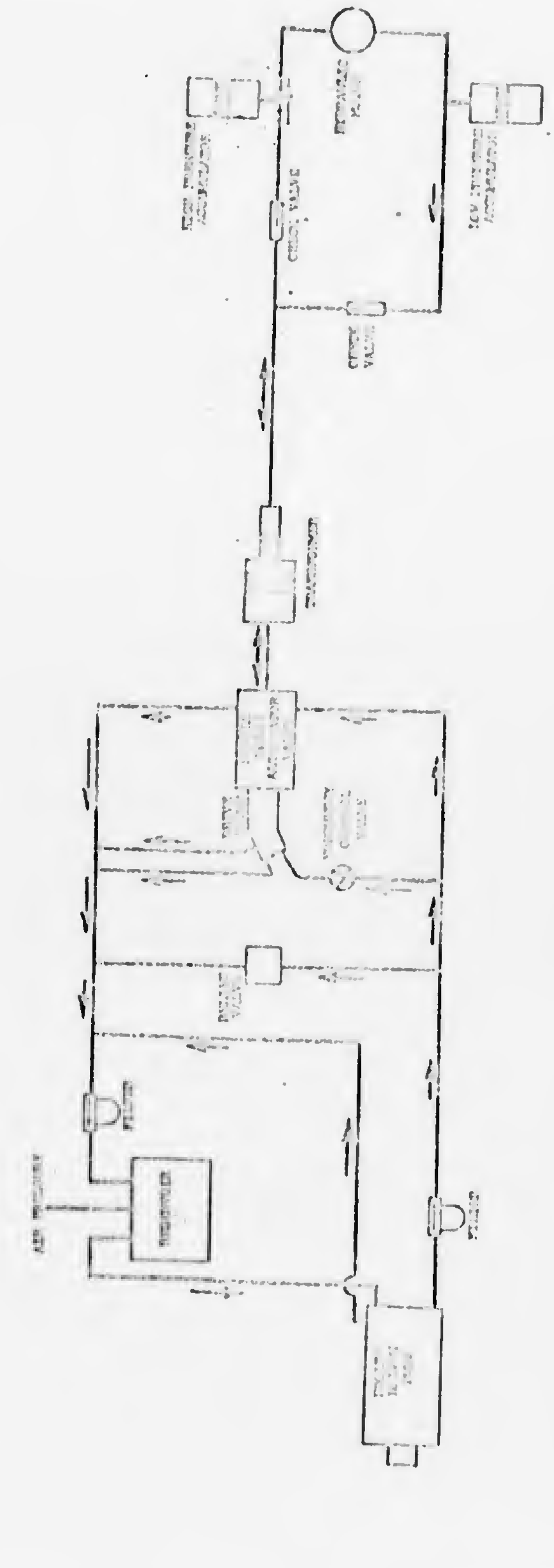
d. Distributed Parameter Transmission Line -- Transmission line parameters (inertance and capacitance) are analyzed as distributed parameters and the frictional effects are lumped at both ends.

e. Hydraulic Rectifiers -- A component which converts pulsating flow to continuous flow.

f. Accumulators -- Used as a ripple filter, fluid-volume control, and to ensure positive pressure on transformer during negative flow.

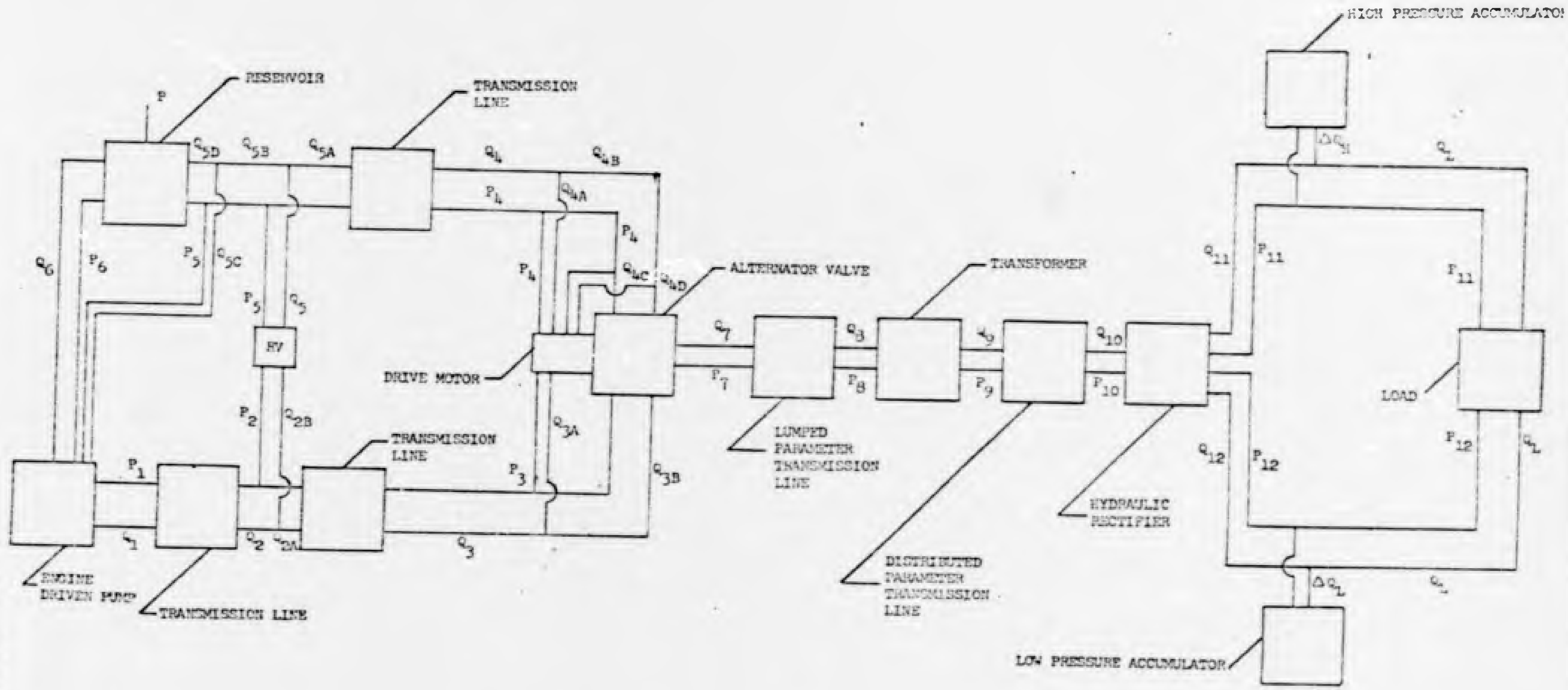
4.2.2 System Design Parameters

In simulating the single-line pulsating hydraulic system, many decisions and assumptions were made pertaining to the system design parameters. Appendix E contains a detailed list of all the design parameters used throughout the research program.



SCHEMATIC OF PULSATING HYDRAULIC SYSTEM

FIGURE 3



BLOCK DIAGRAM OF PULSATING HYDRAULIC SYSTEM

FIGURE 4

CODE IDENT. NO. 81205

NUMBER	D3-6576

SECTION	5.0

SECTION TITLE: ANALYTICAL TECHNIQUES FOR COMPUTER SIMULATION

5.0 ANALYTICAL TECHNIQUES FOR COMPUTER SIMULATION

5.1 Introduction

Evaluation of the proposed systems can be accomplished one of two ways: by an experimental program or by an analytical evaluation of the system. The experimental approach is usually characterized by a minimum of analysis, the construction of a prototype of the system, and considerable "trial-and-error" experimental work. The objectives are to evaluate the experimental data and suggest appropriate modifications which will result eventually in an optimum or nearly optimum design. The cost and time required for this experimental approach are normally much greater than those incurred in an analytical evaluation. In the analytical approach, the task is to derive a set of equations (a mathematical model) whose solution will describe the behavior of the system in terms of its geometry, time, and other physical parameters. These solutions then can be used to obtain operating conditions and parameters which will result in optimum system performance.

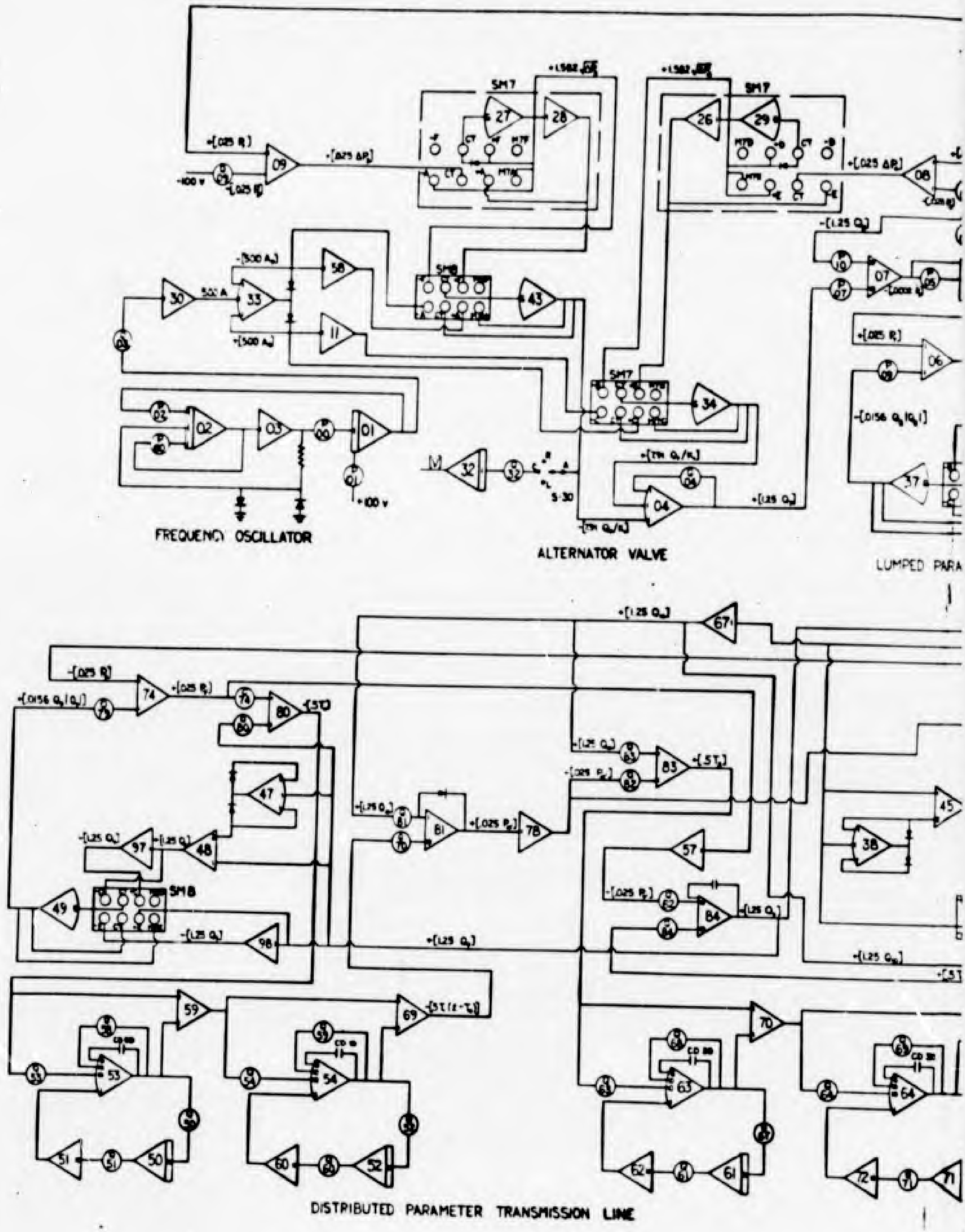
Since the derivation of mathematical models frequently requires approximations, and the results obtained are often based on limited input data, prototype experimentation usually is required. However, prototypes designed on the basis of extensive analytical investigations frequently are near optimum and require little or no modification. The only experimental results required are those which validate the mathematical model. Once the model is validated, additional experimentation can be performed analytically, which results in a considerable cost reduction compared to the experimental approach.

5.2 Analog Computer Simulation

The analytical approach was selected to evaluate the single-line pulsating hydraulic system (Figures 3 and 4). A set of equations (a mathematical model) were derived to describe the system in terms of its geometry, time, and other physical parameters. These equations were scaled and programmed to evaluate the change in system efficiency due to changes in the basic parameters (line length, line size, and pulsation frequency) on the PACE 231-R analog computer. The complete analog computer circuit for the solution of the single-line pulsating hydraulic system is given in Figure 5. The computer scaled variables, computer equations, and potentiometer settings are presented in Appendices B through D.

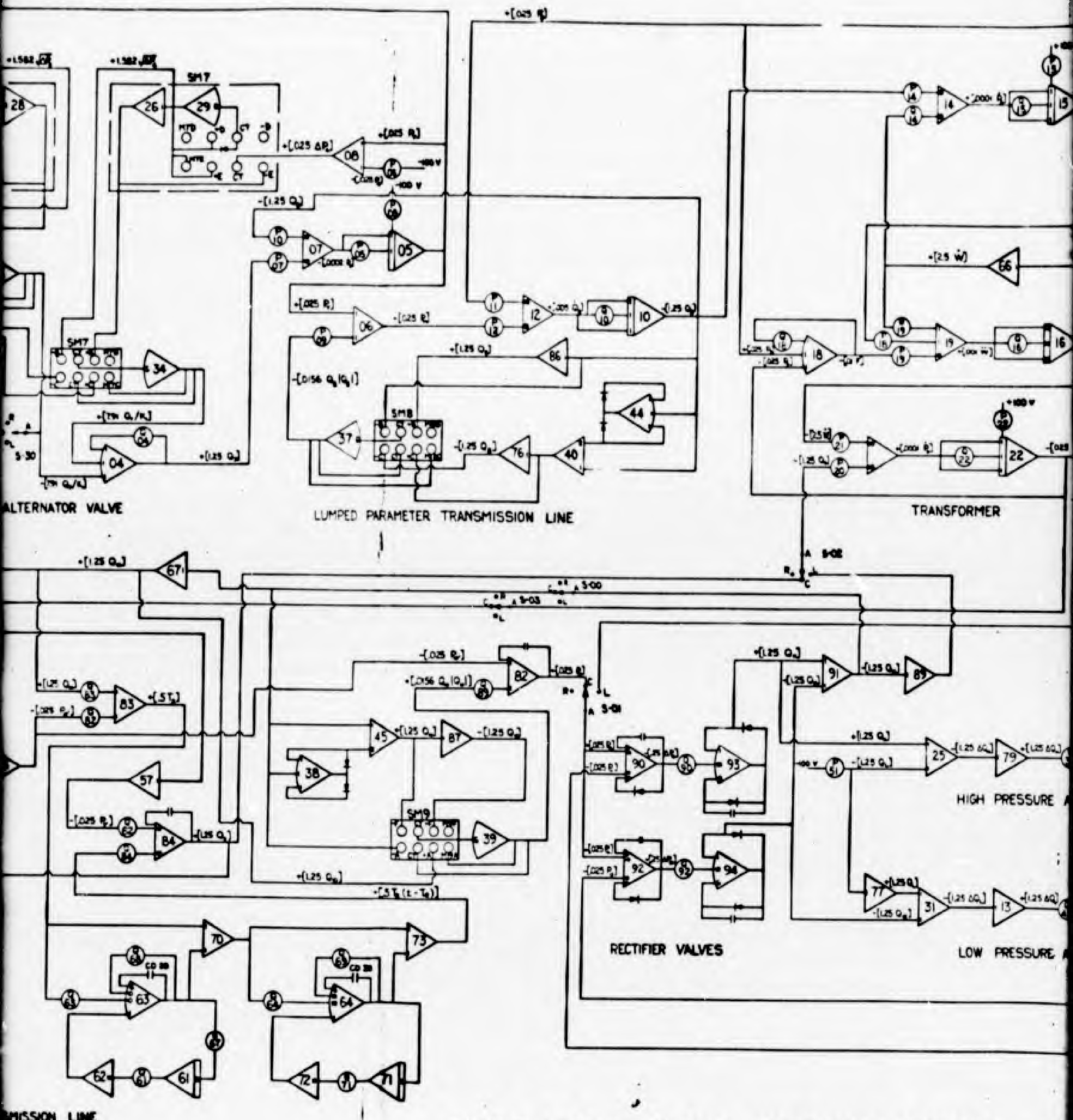
5.3 System Components Equations

To simulate the single-line pulsating hydraulic system on the analog computer, an equation or set of equations had to be derived for each component. The derivation, including assumptions and associated parameters, of the equations for each component is developed below in detail.



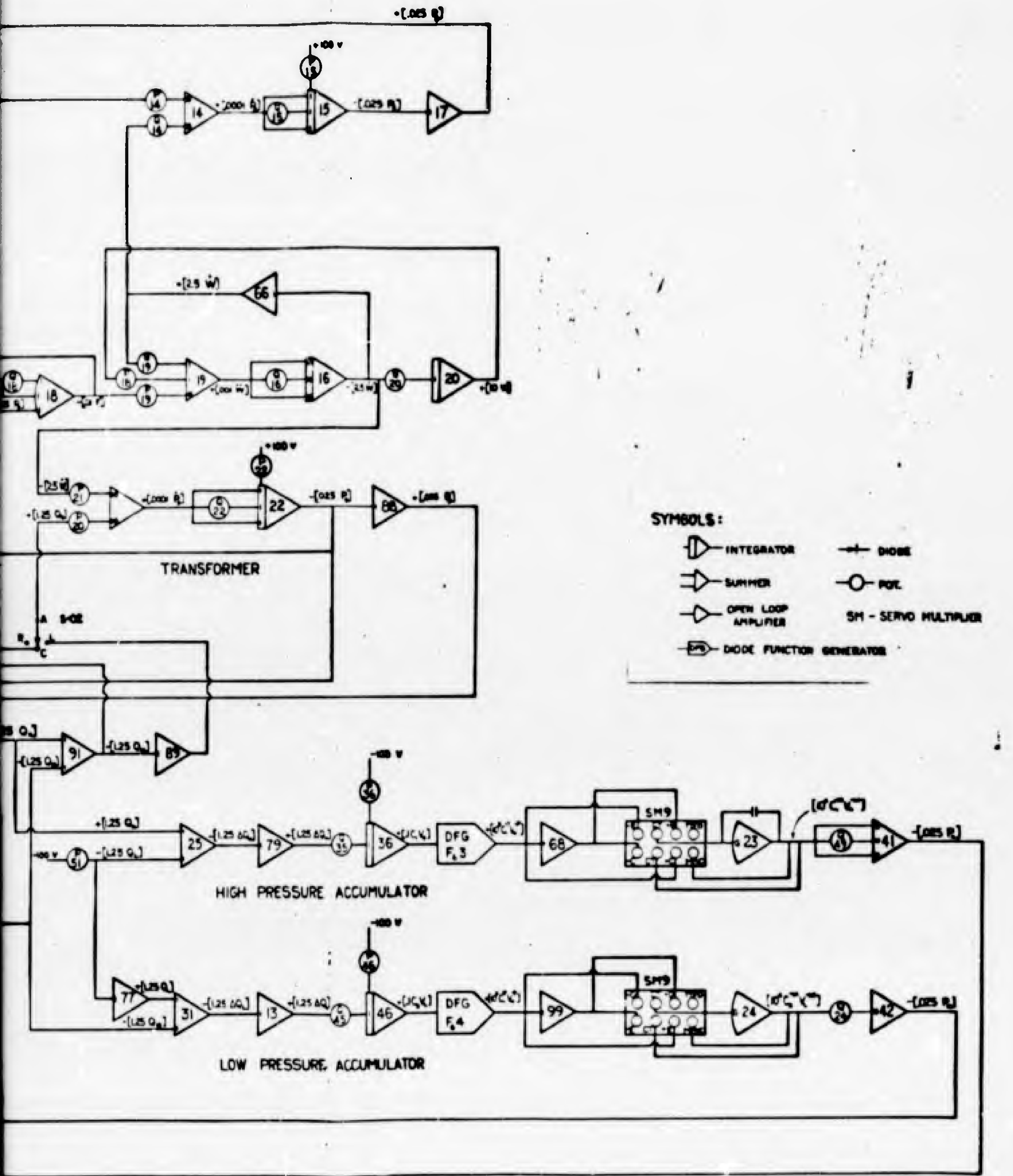
SM7
SM8
FREQUENCY OSCILLATOR
ALTERNATOR VALVE
LUMPED PARA
+100 v
-100 v
S-30
DISTRIBUTED PARAMETER TRANSMISSION LINE

A



SINGLE LINE PULSATING HYDRAULIC SYSTEM - P
FIGURE - 5

B /



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SECT 5.0 PAGE 31

HYDRAULIC SYSTEM - PACE 231-R
GURE - 5

5.3.1 Alternator Valve

The rate of flow through the alternator valve was assumed to obey the orifice equation: (Reference 8)

$$Q = C_D A \sqrt{\frac{2\Delta P}{\rho}} \tag{1}$$

where C_D (discharge coefficient) was assumed a constant value of 0.80. The area of the valve orifice was assumed to vary as a sine function:

$$A = A_o \sin \omega t \tag{2}$$

where $A_o = 0.050$ in.^{2} The desired value for A_o was 0.20 in.^{2}. This was calculated by assuming a pressure drop of 50 psi across the alternator valve. This value was reduced to 0.05 in.^{2} because the transmission line simulation was not adequate to transmit the sudden reversal of pressure to the rectifier valves. This resulted in negative pressures at the rectifier valves which would not occur in the actual system (unless cavitation occurred). The constants of the orifice equation were lumped together to form a new constant K_1:

$$Q = K_1 A \sqrt{\Delta P} \tag{3}$$

where

$$K_1 = C_D \sqrt{\frac{2}{\rho}} \tag{4}$$

5.3.2 Lumped Parameter Transmission Line

Whether a transmission line should be treated as lumped or distributed parameters, depends on the relative magnitude between line length and the wave length of the fluid in the line. A general rule is that when the line length is less than one-eighth of the fluid wave length, the line can be treated as lumped parameters. When this relationship occurs, the spatial variations in flow variables are negligible with respect to temporal changes.

The velocity of sound is given by: (Reference 8)

$$C = \sqrt{\frac{\beta_e}{\rho}} \tag{5}$$

and for the system parameters, $C = 4350$ ft/sec.

The wave length of the fluid is given by: (Reference 8)

$$\lambda = \frac{\text{Velocity of Wave}}{\text{Frequency}} \tag{6}$$

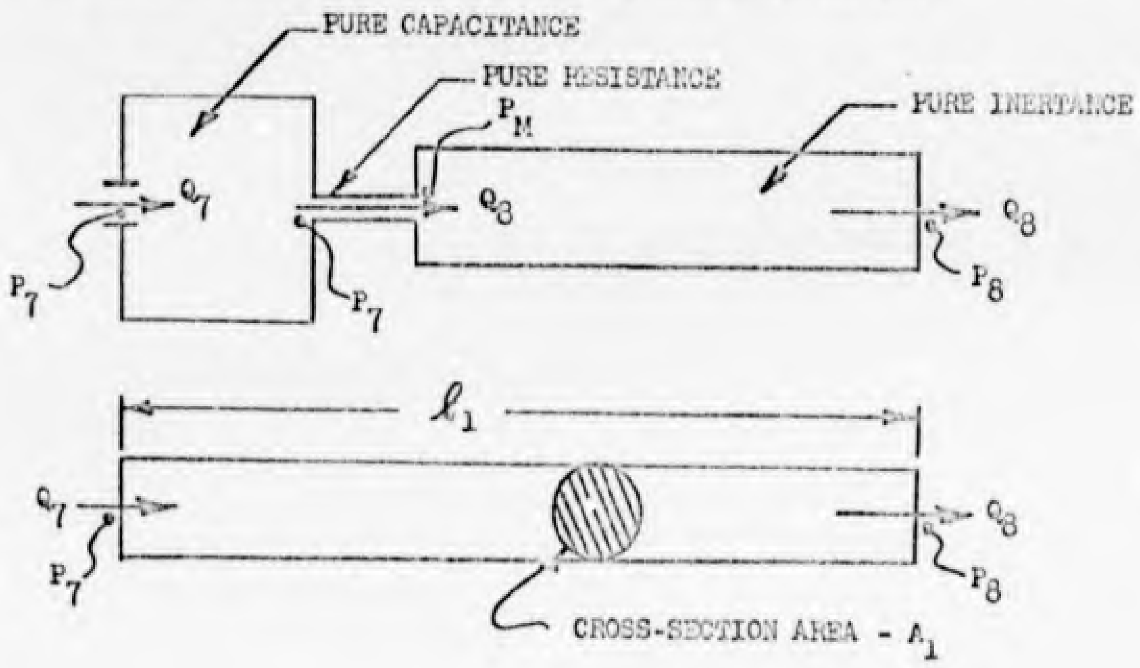
and for the range of frequencies of this research program (4 through 12 cps),

$$\lambda = \frac{4350 \text{ ft/sec}}{12 \text{ cps}} \simeq 363 \text{ ft}$$

and

$$\frac{1}{8} \text{ of } 363 \text{ ft} \simeq 45 \text{ ft}$$

therefore, the transmission line (10 ft) between the alternator valve and the transformer will be analyzed using lumped parameters.



LUMPED PARAMETER TRANSMISSION LINE

FIGURE 6

The governing equations are: (Reference 8)

$$Q_7 - Q_8 = \frac{A_1 \ell_1}{\beta_e} \quad \frac{dP_7}{dt}$$ fluid capacitance

$$P_M - P_8 = \frac{\rho \ell_1}{A_1} \quad \frac{dQ_8}{dt}$$ fluid inertance

$$P_7 - P_M = R_H \, Q_8$$ fluid friction

where

R_H = constant for laminar flow

$R_H = K_2 Q_8$ for turbulent flow

K_2 = constant

The Reynolds number based on the average flow rate of 13.0 gpm (twice the load flow) indicates turbulence flow for the line size of 0.666 inch diameter,

$$R_N = \frac{\rho v D}{\mu} = \frac{vD}{\nu} = \frac{(11.93)(12)(0.666)}{(.018)} = 5300$$

for turbulent flow,

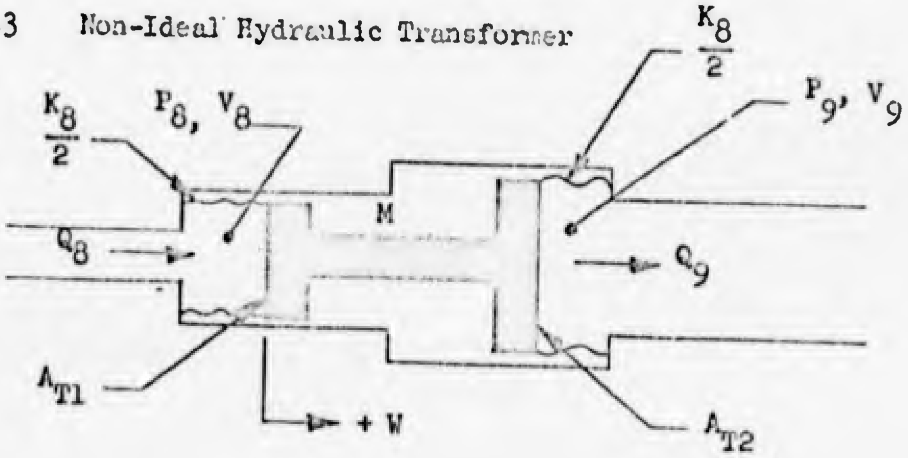
$$K_2 = \frac{8 f_p \ell_1 \rho}{\pi^2 D_1^5}$$

Little information is currently available on the friction characteristics of a fluid in a pulsating system. For fully developed turbulent flow in smooth tubes, the friction factor is given by the empirical Blasius equation;

$$f = \frac{0.316}{R_N^{0.25}} \tag{12}$$

The friction factor for pulsating flow was assumed to be 1.2 f as calculated from Blasius equation.

5.3.3 Non-Ideal Hydraulic Transformer



NON-IDEAL HYDRAULIC TRANSFORMER

FIGURE 7

The governing equations are:

$$F = P_8 A_{T_1} - P_9 \quad A_{T_2} = M \frac{d^2 W}{dt^2} + b \frac{dW}{dt} + K_8 W \quad \text{force equation} \tag{13}$$

$$A_{T_1} \frac{dW}{dt} = Q_8 - \frac{V_8}{\rho} \frac{dP_8}{dt} \tag{14}$$

continuity equations

$$A_{T_2} \frac{dW}{dt} = Q_9 + \frac{V_9}{\rho} \frac{dP_9}{dt} \tag{15}$$

For simplicity, the areas of the input and output ends of the transformer were kept equal for the entire computer program. The transformer parameters were assumed and are given in Appendix E.

5.3.4 Distributed Parameter Transmission Line

The transmission line between the transformer and the rectifier valves was varied in length (50 ft, 125 ft, and 200 ft) during the program. Since the above line lengths are all greater than one-eighth of the fluid wave length, the line was treated as distributed parameters in the analysis. The distributed-parameter equations for a line having distributed inertance and capacitance is given by:

$$P'_{10} + Z_S Q_{10} = P'_9 (t - T_e) + Z_S Q_9 (t - T_e) \qquad (16)$$

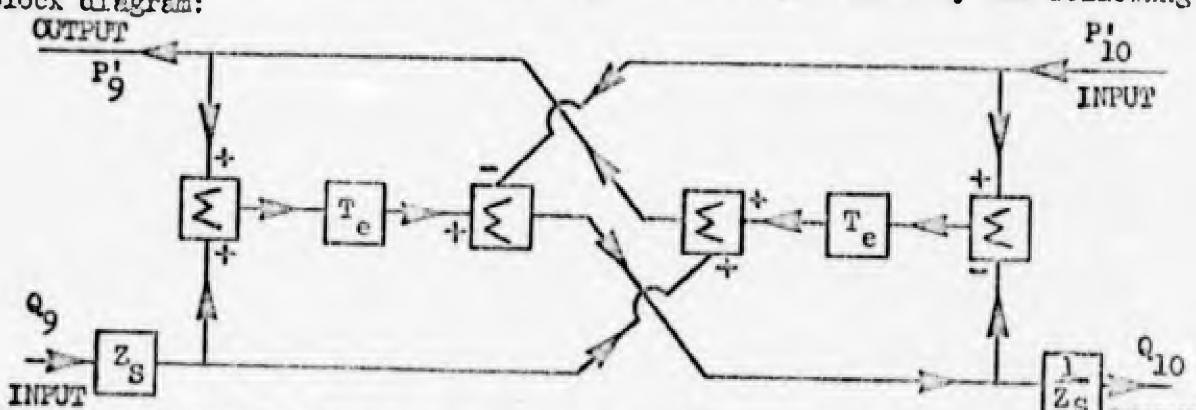
$$P'_9 - Z_S Q_9 = P'_{10} (t - T_e) - Z_S Q_{10} (t - T_e) \qquad (17)$$

where

$$Z_S = \sqrt{\frac{\rho \beta_e}{A_2^{\,2}}} \quad \text{characteristic impedance without friction} \qquad (18)$$

$$T_e = \ell_2 \sqrt{\frac{\rho}{\beta_e}} \quad \text{time for pressure disturbance to travel length of tube} \qquad (19)$$

These two equations (16 and 17) can be represented by the following block diagram:



DISTRIBUTED PARAMETER TRANSMISSION LINE BLOCK DIAGRAM

FIGURE 8

The general equations (16 and 17) can be arranged or transformed into a more useful form for programming on the analog computer,

Equation 16

$$P'_{10} = P'_9 \, (t - T_e) + Z_S \, Q_9 \, (t - T_e) - Z_S \, Q_{10}$$

$$\text{Let } T_1 = \frac{1}{Z_S} \, P'_9 + Q_9 \qquad (20)$$

$$P'_{10} = T_1 \, Z_S \, (t - T_e) - Z_S \, Q_{10} \qquad (21)$$

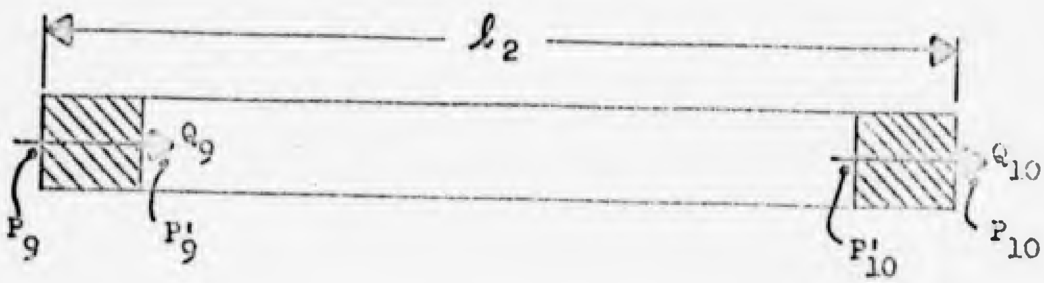
Equation 17

$$Q_9 = \frac{P'_9}{Z_S} + Q_{10} \, (t - T_e) - \frac{P'_{10}}{Z_S} \, (t - T_e)$$

$$\text{Let } T_2 = \frac{1}{Z_S} \, P'_{10} - Q_{10} \qquad (22)$$

$$Q_9 = \frac{P'_9}{Z_S} - T_2 \, (t - T_e) \qquad (23)$$

As yet, no exact simple block diagram representation of a line having distributed inertance, capacitance, and resistance is available. The line frictional effects were lumped at both ends of the transmission line for the analysis.



DISTRIBUTED PARAMETER TRANSMISSION LINE FRICTION EFFECTS

FIGURE 9

$$P_9 - P'_9 = \frac{R_H}{2} Q_9 \qquad (24)$$

$$P'_{10} - P_{10} = \frac{R_H}{2} Q_{10} \qquad (25)$$

where

R_H = KQ for turbulent flow

$$P_9 - P'_9 = K_5 Q_9 |Q_9| \qquad (26)$$

$$P'_{10} - P_{10} = K_6 Q_{10} |Q_{10}| \qquad (27)$$

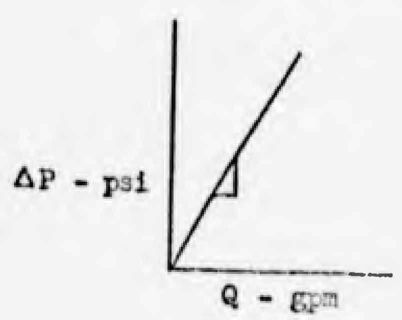
$$K_5 = K_6 = \frac{8 f_p \ell_2 \rho}{\pi^2 D_2^5} \qquad (28)$$

The friction factor for pulsating flow was assumed to be 1.2 f as calculated from Blasius equation.

5.3.5 Rectifier Valves

The resistance or pressure drop through the rectifier valves were assumed to be a straight-line function of flow. Fluid compressibility of the rectifier is neglected.

Assumed relationship



$$\text{Slope} = R_R = \frac{\Delta P}{Q} = 1.847 \ \frac{\text{lbs-min}}{\text{in}^2 - \text{gal}}$$

RECTIFIER VALVE RELATIONSHIP

FIGURE 10

Rectifiers equations,

$$\Delta P_{11} = P_{10} - P_{11} \tag{29}$$

$$Q_{11} = \frac{1}{R_R} \Delta P_{11} \qquad \Delta P_{11} > 0 \tag{30}$$

$$\Delta P_{12} = P_{12} - P_{10} \tag{31}$$

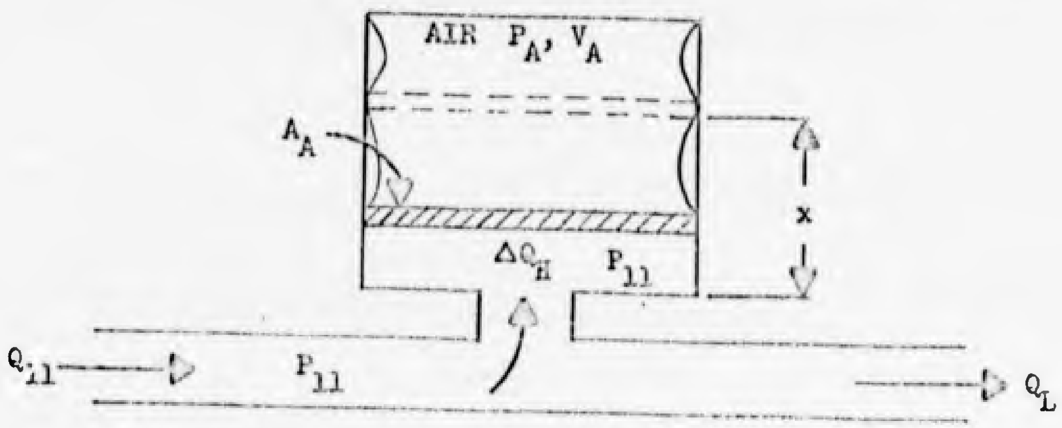
$$Q_{12} = \frac{1}{R_R} \Delta P_{12} \qquad \Delta P_{12} > 0 \tag{32}$$

$$Q_{10} = Q_{11} - Q_{12} \tag{33}$$

5.3.6 Accumulators

Although the construction of the general types of accumulators differ, their basic functions are identical, namely to store pressure energy. Air loaded piston accumulators were used in the pulsating hydraulic system.

In the analysis of the air loaded accumulator, it was assumed that no resistence to flow into the accumulator occurred. Also, the dynamic parameters such as mass and damping of the accumulator piston was neglected.



AIR LOADED PISTON ACCUMULATOR

FIGURE 11

The development of the accumulator equations (References 1 and 8) will be for the high pressure accumulator, but the same equations describe the low pressure accumulator except for different sub-scripts.

Force equilibrium equation,

$$A_A P_{11} = A_A P_A \tag{34}$$

Assuming the compressed air is a perfect gas and the compression and expansion process is adiabatic and reversible,

$$P_A V_A^k = P_i V_i^k \tag{35}$$

$$\Delta Q_H = A_A \frac{dx}{dt} \tag{36}$$

where P_i and V_i are initial air pressure and air volume in the accumulator.

Differentiating Equation 35,

$$P_A k V_A^{k-1} \frac{dV_A}{dt} = - V_A^k \frac{dP_A}{dt} \tag{37}$$

$$\Delta Q_H = - \frac{1}{k} \left(\frac{V_A}{P_A}\right) \frac{dP_A}{dt} \tag{38}$$

From continuity equation of flow,

$$\Delta Q_H = Q_{11} - Q_L \tag{39}$$

$$Q_L = Q_{11} - \frac{1}{k} \left(\frac{V_A}{P_A}\right) \frac{dP_A}{dt} \tag{40}$$

Although Equations 35 and 40 are the basic accumulator equations, another equation is derived from them for programming on the analog computer.

$$V_A = \left(\frac{P_i V_i^{\,k}}{P_A}\right)^{\frac{1}{k}} = P_i^{\frac{1}{k}} \; V_i \Big/ P_A^{\frac{1}{k}} \qquad (41)$$

substituting in Equation 40,

$$Q_L = Q_{11} - \frac{P_i^{\frac{1}{k}} V_i}{k} \; P_A^{-\left(\frac{k+1}{k}\right)} \; \frac{dP_A}{dt} \qquad (42)$$

but $P_{11} = P_A$

$$\Delta Q_H = \frac{P_i^{\frac{1}{k}} V_i}{k} \; P_{11}^{-\left(\frac{k+1}{k}\right)} \; \frac{dP_{11}}{dt} \qquad (43)$$

$$\Delta Q_H = K_3 \, P_{11}^{-K_4} \; \frac{dP_{11}}{dt} \qquad (44)$$

where

$$K_3 = \frac{P_i^{\frac{1}{k}} V_i}{k} \qquad (45)$$

$$K_4 = \left(\frac{k+1}{k}\right) \qquad (46)$$

5.3.7 Load

A load demanding a constant flow (Q_L = 6.5 gpm) was assumed for the pulsating hydraulic system simulation.

5.4 Computer Techniques

The programming of the pulsating hydraulic system on the PACE 231-R analog computer needs some explanation. The analog computer circuit (Figure 5) and the potentiometer settings (Appendix D) are in agreement for run B-B-8.

While the potentiometer settings (Appendix D) are correct for all the runs, in some cases the gains on the amplifiers of Figure 5 were changed to obtain the desired results. A brief discussion of the programming methods used for some of the computer components is included below.

5.4.1 Frequency Oscillator

The time to complete one run was approximately 15 minutes because of the large number of parameters being recorded. During this time, the magnitude of the original sine oscillator for the alternator valve was not stable and gave erroneous results. To remedy this situation, a stable wide frequency oscillator (Reference 9) was mechanized by adding a diode clipping circuit to the usual three amplifier oscillator. It utilizes negative damping plus non-linear positive damping in such a way that the energy balance is maintained at only a determined amplitude, and the solution is a limit cycle.

5.4.2 Transport Delay Simulation

The distributed parameter transmission line contains a phenomenon called transport delay that must be simulated. Mathematically, transport delay is a method that accepts a time-varying input and produces an output which is equal to the input, but displaced in time. This property can be expressed by the equation,

$$y(t) = x(t - T_e) \tag{47}$$

The equations (21 and 23) for the distributed parameter transmission line contain this transport delay, $(t - T_e)$. There are many methods and approximations available for achieving this time delay on the analog computer. The accuracy of the methods is a function of the frequency of the signal to be delayed and the delay time. The accuracy deteriorates for high frequencies and long delay times. The error is a function of the product of the input frequency and the delay time.

The transport delay was simulated by a method based on a theorem of Bode (Reference 10), using a four root-four pole network. The accuracy of the time delay simulation is related to a figure of merit calculated for each network. The figure of merit for the four root-four pole network was $\omega T_e = 8.2$. To achieve better results for the higher frequencies and longer time delays, the delay circuits can be cascaded together. A discussion of the accuracy of the time delay simulation is given in Section 6.0, Test Results.

5.4.3 Accumulators

The accumulator circuit (Figure 5) is rather unique, and a detailed discussion of its development is included below. The size and pre-charge pressure of the accumulators were assumed for the program and the optimum design for the accumulators was not determined. The pressure of the accumulators had a large range of values and a loss of accuracy would be incurred by using the general computer methods in conjunction with a function generator. The method used by Republic Aircraft (Reference 1) was tried but the non-optimized accumulators created scaling and noise problems. The computer programming would have been routine for an optimum designed accumulator (size and pre-charge pressure).

Equation development (high pressure accumulator),

$$\Delta Q_H = K_3 \, P_{11}^{-K_4} \, \frac{dP_{11}}{dt} \tag{47}$$

Integrating both sides of the equation,

$$P_{11}^{-K_4+1} = \left(\frac{1-K_4}{K_3}\right) V_H \tag{48}$$

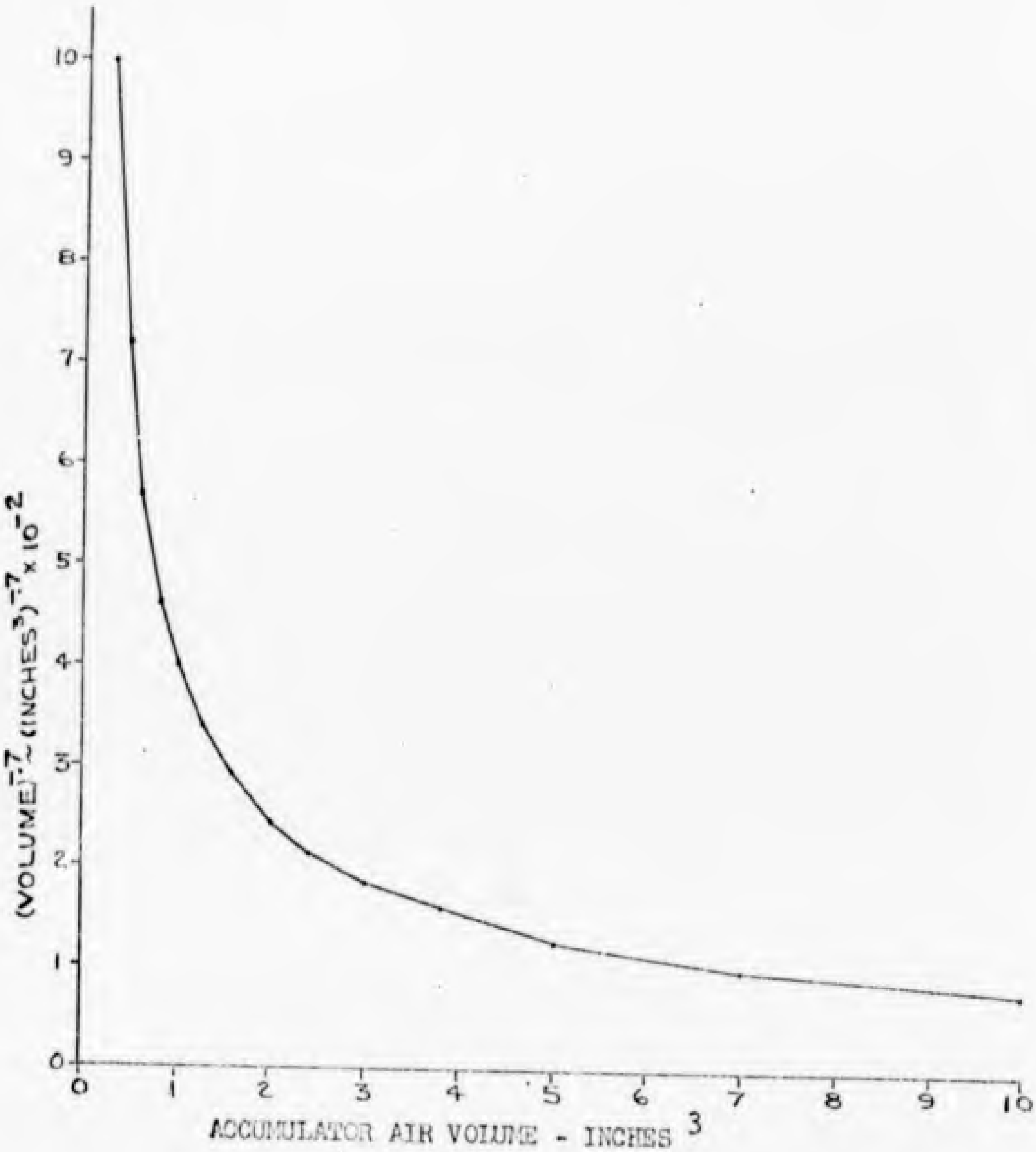
$$K_3^{\left(\frac{1}{1-K_4}\right)} \, P_{11} = \left|1 - K_4\right|^{\left(\frac{1}{1-K_4}\right)} \; V_H^{\left(\frac{1}{1-K_4}\right)} \tag{49}$$

$$K_4 = \left(\frac{k+1}{k}\right) = 1.714 \tag{46}$$

Substituting Equation 46 in Equation 49,

$$K_3^{\left(\frac{1}{1-K_4}\right)} \, P_{11} = \frac{1.601}{V_H^{\,1.4}} \tag{50}$$

Integrating ΔQ_H with respect to time, with the initial condition representing the initial air volume in the accumulator, results in the volume $\left[.1 \, C_1 \, V_H\right]$ of air in the accumulator at any desired time. A diode function generator was used to obtain $\left[10^3 \, C_1^{-.7} \, V_H^{-.7}\right]$ as a function of $\left[.1 \, C_1 \, V_H\right]$. The variable C_1 is a scaling factor used to adapt the diode function generator curve (Figure 12) to the proper abscissa scale for greater accuracy. The normal



DIODE FUNCTION
GENERATOR CURVE

FIGURE 12

range of the abscissa scale is 0 - 1000 in^3, but by letting $C_1 = 5$, the abscissa range is reduced to 0 - 200 in^3.

The value $\left[10^3 C_1^{-.7} V_H^{-.7}\right]$ obtained from the diode function generator was squared to obtain $10^4 \left[C_1^{-1.4} V_H^{-1.4}\right]$. Using this value for $V_H^{-1.4}$, the pressure of the accumulator (P_{11}) was calculated from Equation 50. Although the above computer simulation for the accumulators is involved, the results obtained were in agreement with the accumulator theory.

CODE IDENT. NO. 81205

NUMBER	D3-6576

SECTION	6.0

SECTION TITLE: TEST RESULTS OF COMPUTER SIMULATION

REV LTR:

E 3030 R3

6.0 TEST RESULTS OF COMPUTER SIMULATION

6.1 Introduction

A single-line pulsating hydraulic system was simulated on the PACE 231-R analog computer. The dynamic properties of the system were recorded on a multi-channel recorder, which employs a moving strip of paper, drawn at constant speed past a set of pens. The pens are deflected proportionally to input voltages, and the result is a set of graphs of voltages as function of time. These voltages are transferred to the proper values of the variables they represent by their scale factors. Additional pens, called "event" pens, one at each edge of the paper, produce timing marks at one-second intervals.

It is important to assure that the rate of change of the computer variables is consistent with the dynamic properties of the computer, and that the solution takes place in a reasonable amount of time. To assure that the computer simulation takes place in a reasonable length of time, a method, called time scaling, is used for speeding up or slowing down the solution. In the pulsating hydraulic system simulation, the solution on the computer was slowed down by a factor of 100 ($\beta = 100$) compared to the original problem time.

6.2 Computer Data

During the research program, 47 different system configurations were evaluated. These configurations were evaluated for the change in system efficiency due to changes in the basic parameters of line lengths, line sizes, and pulsation frequencies. Table I correlates the run number to the values for each of the above parameters. The line length between the alternator valve and the transformer was kept constant throughout the program. The line length given in Table I refers to the line between the transformer and the rectifier valves.

A complete set of recording graphs (Figures 13 through 18) for Run No. B-B-8 illustrates the typical dynamic data obtained during each run for the computer circuit. The voltages of the recording graphs were transferred to the proper values of the variables they represent by their scale factors. An explanation is needed for some of the data obtained on the recording graphs. The positive values for the pressure drop of ΔP_{11} and ΔP_{12} (Figure 15), represents the only useful data. The negative values do not have any meaning for this program, for they were amplified by the diode feedback (Figure 5) installed to prevent amplifier overload. This does not effect the program, for when ΔP is negative there is no flow through the rectifier valves. The volume (positive) through the alternator valve for any desired time can be obtained from the efficiency volume graph (Figure 16). The pressure of the high pressure accumulator is shown on the right-hand side of P_{11} (Figure 16). The left-hand side of P_{11} (Figure 16), represents the variation of the accumulator pressure which would be difficult to obtain from the right-hand graph.

TABLE I

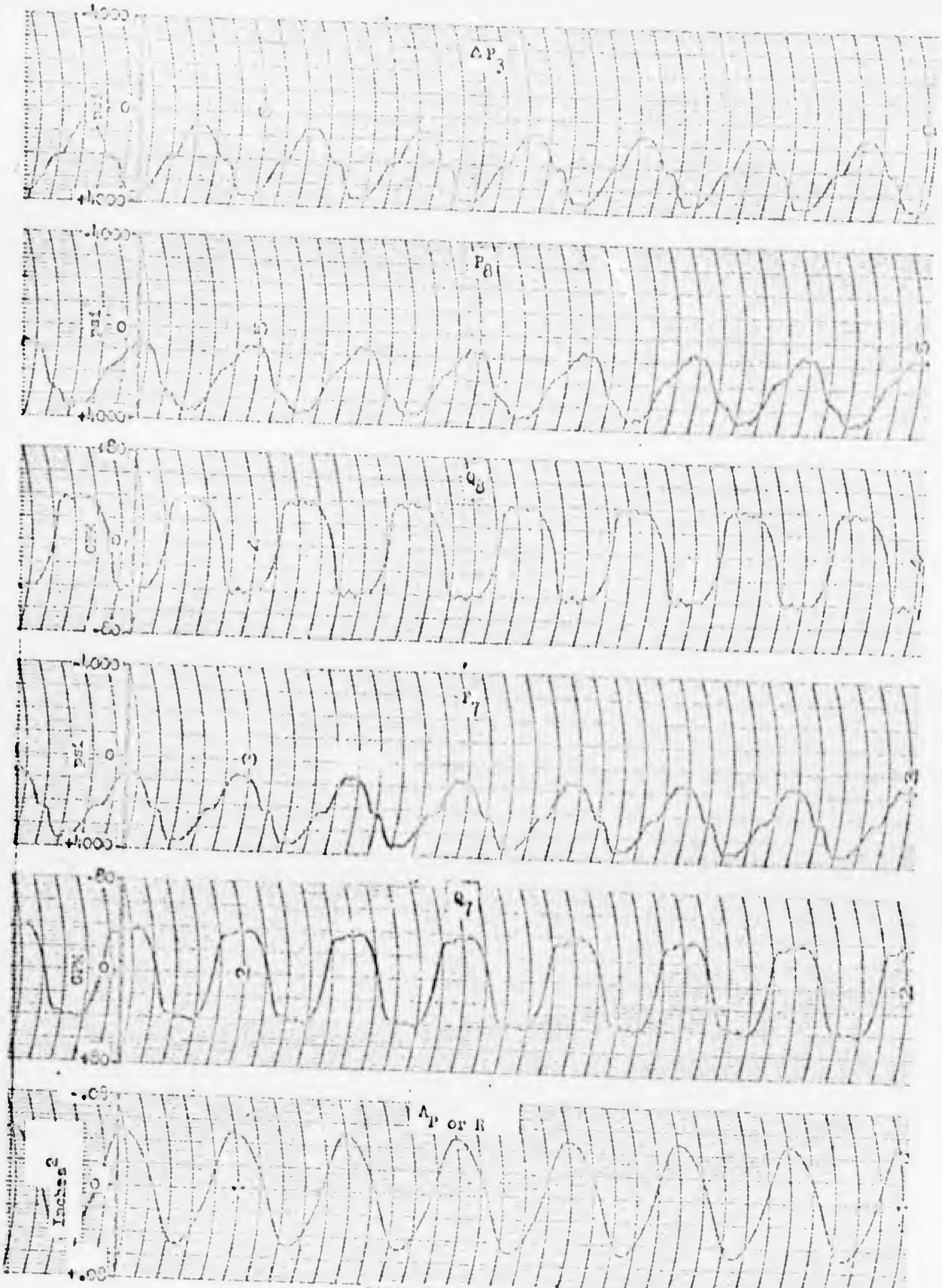
COMPUTER RUN NUMBERS AND PARAMETERS

RUN NUMBER	LINE LENGTH	LINE DIAMETER - I.D.	PULSATION FREQUENCY
A-A-4	50 Ft.	0.444 In.	4 cps
A-A-6	50 Ft.	0.444 In.	6 cps
A-A-8	50 Ft.	0.444 In.	8 cps
A-A-10	50 Ft.	0.444 In.	10 cps
A-A-12	50 Ft.	0.444 In.	12 cps
A-B-4	50 Ft.	0.666 In.	4 cps
A-B-6	50 Ft.	0.666 In.	6 cps
A-B-8	50 Ft.	0.666 In.	8 cps
A-B-8-L*	50 Ft.	0.666 In.	8 cps
A-B-10	50 Ft.	0.666 In.	10 cps
A-B-12	50 Ft.	0.666 In.	12 cps
A-C-4	50 Ft.	0.902 In.	4 cps
A-C-6	50 Ft.	0.902 In.	6 cps
A-C-8	50 Ft.	0.902 In.	8 cps
A-C-10	50 Ft.	0.902 In.	10 cps
A-C-12	50 Ft.	0.902 In.	12 cps
B-A-4	125 Ft.	0.444 In.	4 cps
B-A-6	125 Ft.	0.444 In.	6 cps
B-A-8	125 Ft.	0.444 In.	8 cps
B-A-10	125 Ft.	0.444 In.	10 cps
B-A-12	125 Ft.	0.444 In.	12 cps
B-B-4	125 Ft.	0.666 In.	4 cps
B-B-6	125 Ft.	0.666 In.	6 cps
B-B-8	125 Ft.	0.666 In.	8 cps
B-B-8-L*	125 Ft.	0.666 In.	8 cps
B-B-10	125 Ft.	0.666 In.	10 cps
B-B-12	125 Ft.	0.666 In.	12 cps
B-C-4	125 Ft.	0.902 In.	4 cps
B-C-6	125 Ft.	0.902 In.	6 cps
B-C-8	125 Ft.	0.902 In.	8 cps
B-C-10	125 Ft.	0.902 In.	10 cps
B-C-12	125 Ft.	0.902 In.	12 cps
C-A-4	200 Ft.	0.444 In.	4 cps
C-A-6	200 Ft.	0.444 In.	6 cps
C-A-8	200 Ft.	0.444 In.	8 cps
C-A-10	200 Ft.	0.444 In.	10 cps
C-A-12	200 Ft.	0.444 In.	12 cps

*Transmission line analyzed by lumped parameter method.

TABLE I (CONT'D)

RUN NUMBER	LINE LENGTH	LINE DIAMETER - I.D.	PULSATION FREQUENCY
C-B-4	200 Ft.	0.666 In.	4 cps
C-B-6	200 Ft.	0.666 In.	6 cps
C-B-8	200 Ft.	0.666 In.	8 cps
C-B-10	200 Ft.	0.666 In.	10 cps
C-B-12	200 Ft.	0.666 In.	12 cps
C-C-4	200 Ft.	0.902 In.	4 cps
C-C-6	200 Ft.	0.902 In.	6 cps
C-C-8	200 Ft.	0.902 In.	8 cps
C-C-10	200 Ft.	0.902 In.	10 cps
C-C-12	200 Ft.	0.902 In.	12 cps

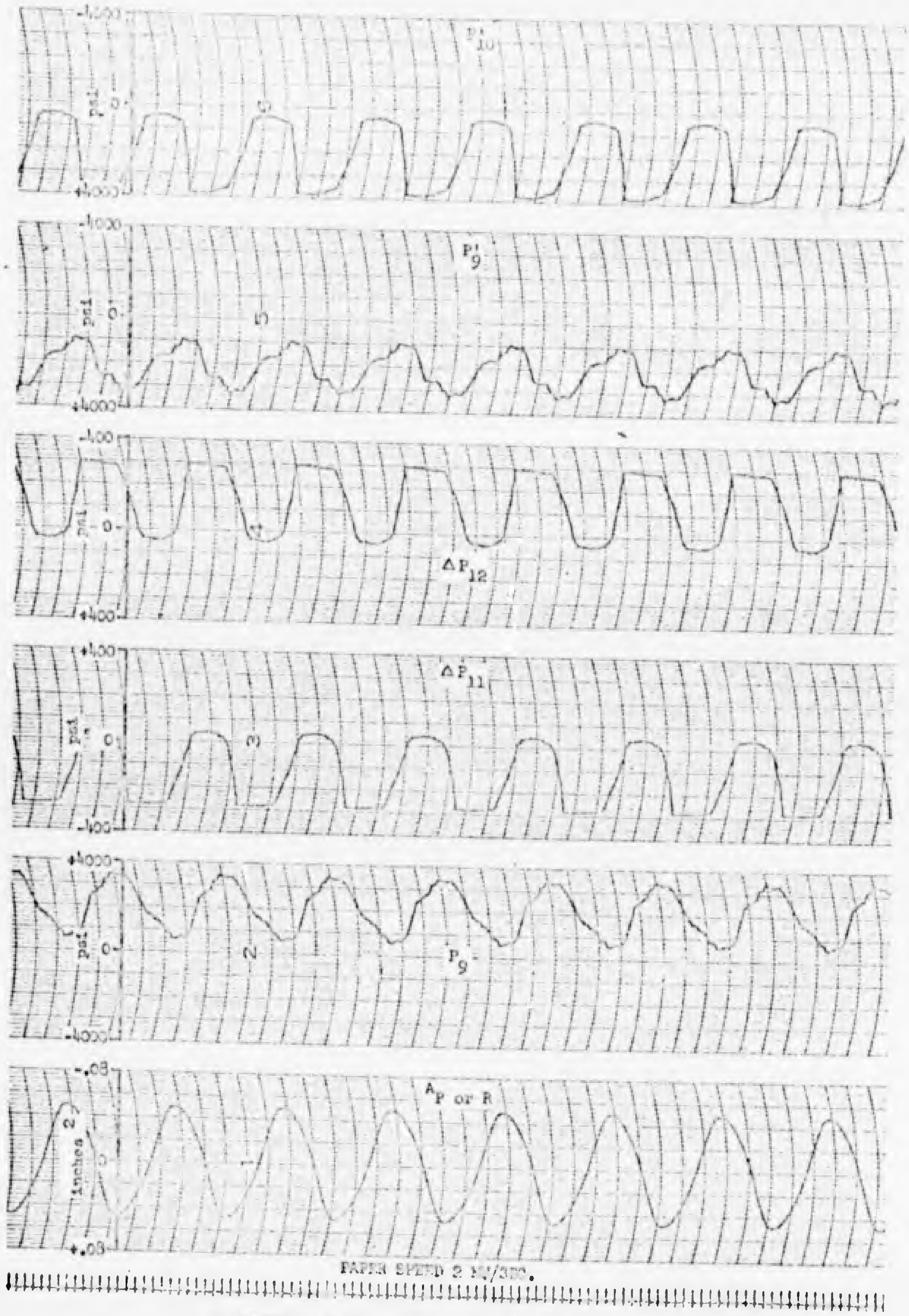


COMPUTER DATA - RUN NO. B-B-8

FIGURE 13

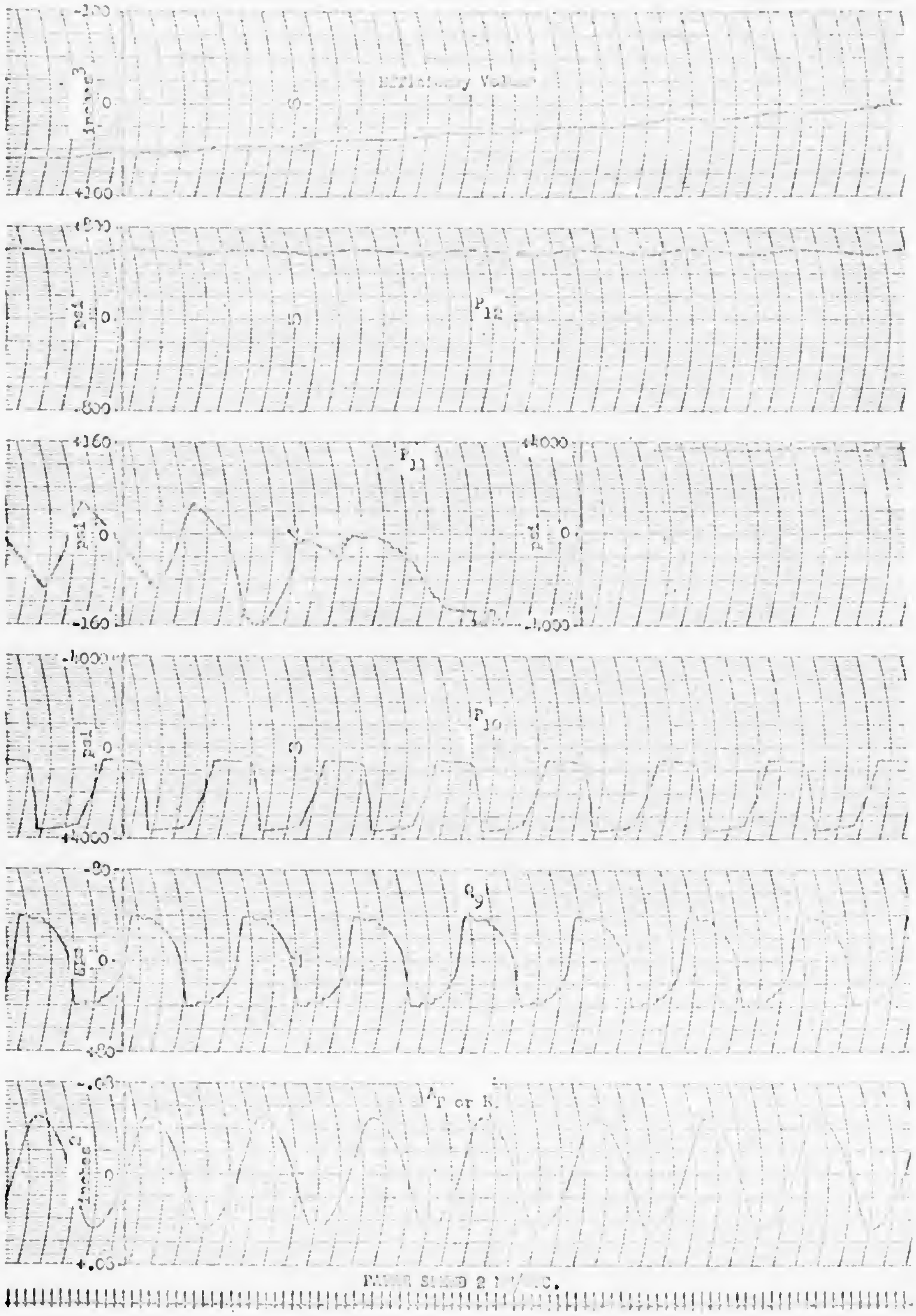
COMPUTER DATA - RUN NO. B-R-8

FIGURE 14



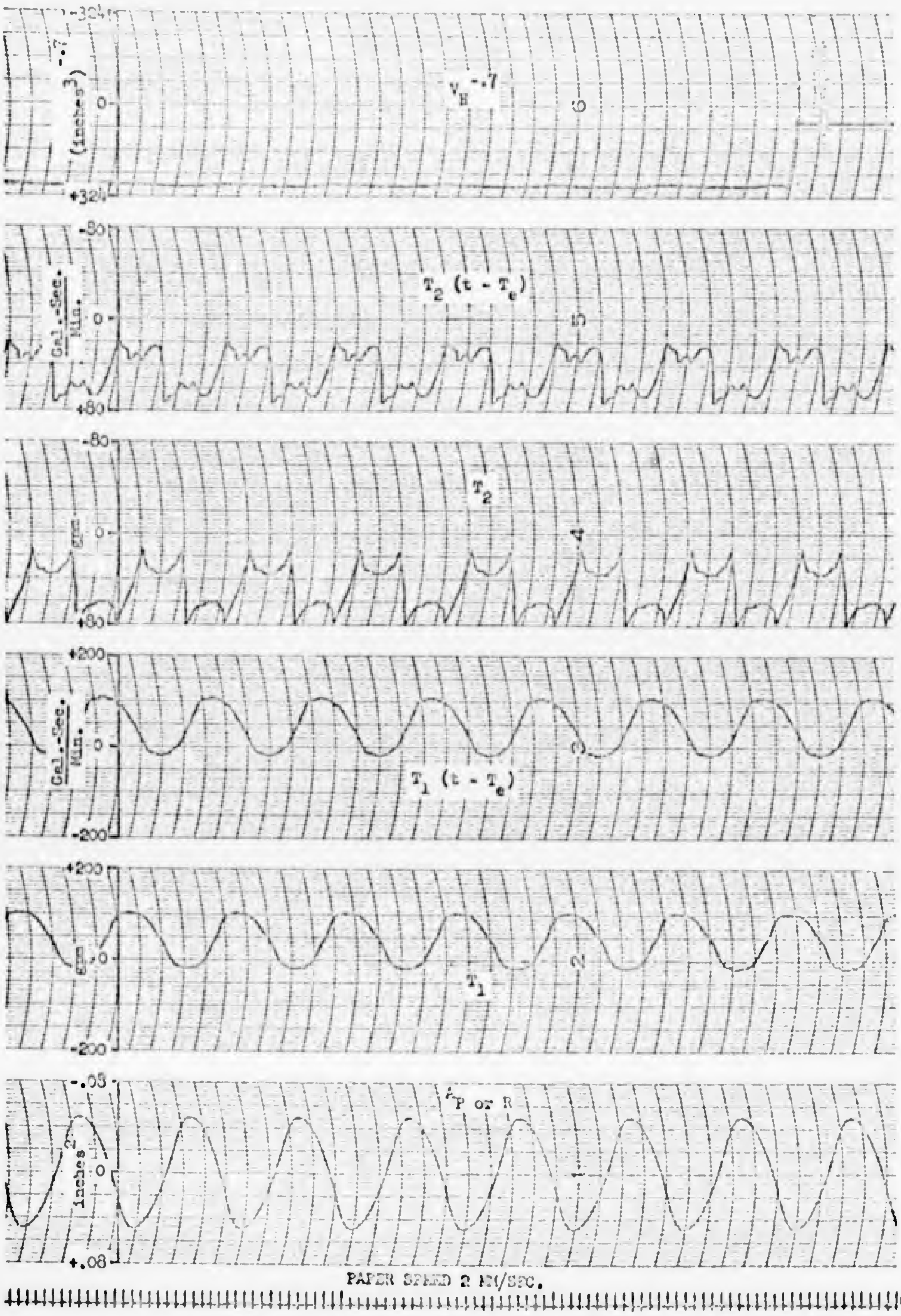
COMPUTER DATA - RUN NO. B-B-8

FIGURE 15



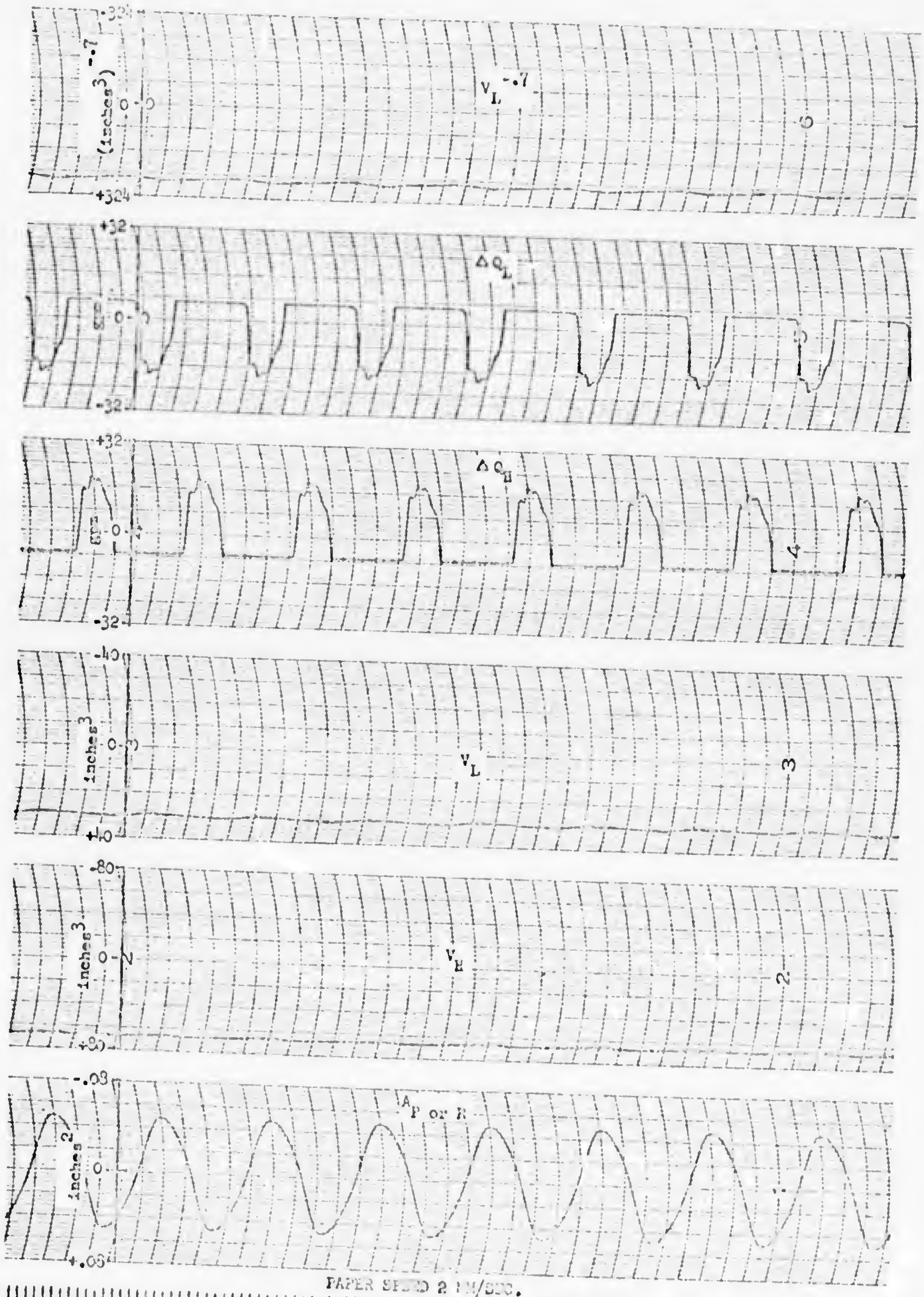
COMPUTER DATA - RUN NO. B-B-8

FIGURE 16



COMPUTER DATA - RUN NO. B-B-8

FIGURE 17



COMPUTER DATA - RUN NO. B-B-8

FIGURE 18

6.3 System Efficiency

One of the objectives of this research program was to determine or evaluate the feasibility of pulsating hydraulic systems. The system efficiency of the pulsating hydraulic systems must be examined to determine whether it is competitive with the system efficiency of continuous flow hydraulic systems. The simulation of the single-line pulsating hydraulic system on the analog computer enabled the calculation of the system efficiency due to changes in the basic parameters of line length, line sizes, and pulsation frequencies. The system efficiency was determined using two different bases:

a. System efficiency calculated upstream of the alternator valve (includes losses to alternator valve and reservoir).

b. System efficiency calculated downstream of the alternator valve.

6.3.1 System Efficiency Calculated Upstream of the Alternator Valve

The system efficiency for the 47 different configurations was calculated upstream of the alternator valve. This method includes the losses through the alternator valve and the overboard losses due to the reservoir. The following method was used to calculate the system efficiency:

$$\text{System Efficiency} = \frac{\text{Work Out}}{\text{Work In}} \qquad (1)$$

Where,

$$\text{Work Out} = (\text{Flow Through Load})\ (\text{Pressure Drop Across Load})$$

$$\text{Work Out} = \left[(6.5\ \text{gpm})(3.85\ \frac{\text{Min-In}^3}{\text{Sec-Gal}})\right] \left[P_{11_{ave.}} - P_{12_{ave.}}\right]$$

$$\text{Work In} = \left[\text{Volume (Positive) through Alternator Valve in one Second}\right] \times \left[\text{Pressure Drop Across Alternator Valve}\right]$$

$$\text{Work In} = \left[\text{Volume } (\text{in}^3/\text{sec})\right] \left[P_3 - P_4\right]$$

The volume through the alternator valve (positive flow) was obtained on the analog computer by integrating Q_7 (positive value obtained from amplifier 43) with respect to time.

$$\text{System Efficiency} = \frac{\left[25\ \text{in}^3/\text{sec}\right]\left[P_{11_{ave.}} - P_{12_{ave.}}\right]}{\left[\text{Volume (in}^3/\text{sec)}\right]\left[3950\ \text{psi}\right]} \qquad (2)$$

The system efficiency is plotted as a function of line length, line size, and pulsation frequency as shown in Figures 19 through 21.

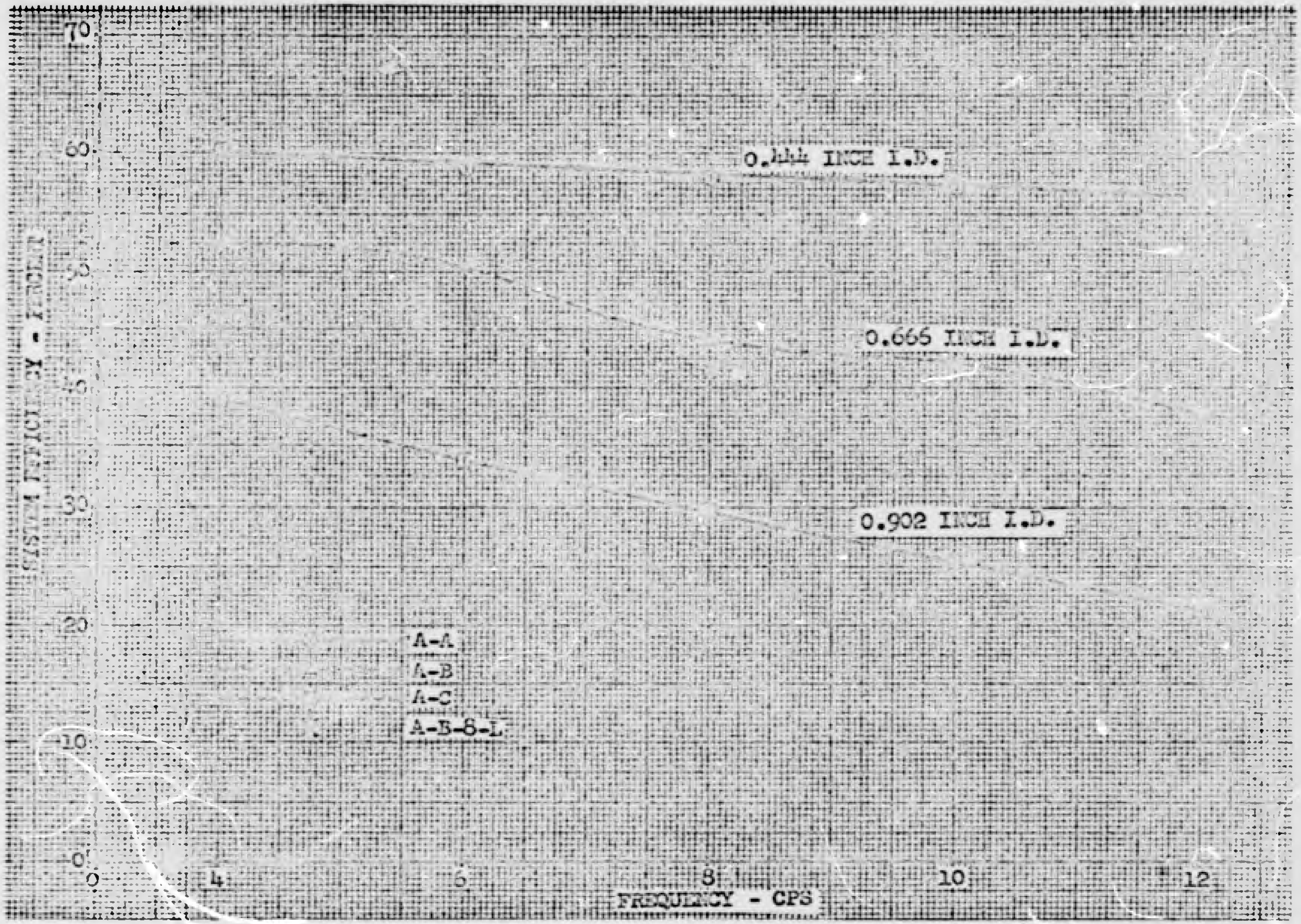
For the 50-foot transmission line (Figure 19), the efficiency varied from 21 percent to 60 percent depending on the line size and pulsation frequency. These values of system efficiency are quite low when compared with the efficiency of continuous flow hydraulic systems (90 percent to 95 percent). The system efficiency for the small line size (I.D. = 0.444 In.) was nearly constant throughout the frequency range and appears not to be a function of pulsation frequency. For the larger line sizes (I.D. = 0.666 in. and 0.902 in.), the system efficiency decreased with increasing values of pulsation frequency. All three curves, representing different line sizes, approximates a straight line relationship with pulsation frequency.

The system efficiency was higher for the smaller line sizes which indicates that the fluid capacitance losses over-shadowed the fluid friction and inertia losses for the 50-foot transmission line. Run No. A-B-8-L had the same configuration as Run No. A-B-8, but the 50-foot transmission line was analyzed using the lumped parameter method instead of the distributed parameter method. The results of the two runs are nearly identical (system efficiency varied three percent) which indicates the lumped parameter method could have been used to analyze the 50-foot transmission line with good results.

For the 125-foot transmission line (Figure 20), the efficiency varied from 20 percent to 45 percent depending on the line size and pulsation frequency. These values of system efficiency are quite low when compared with the efficiency of continuous flow hydraulic systems. The system efficiency was lower for the 125-foot transmission line than the 50-foot transmission line at the low pulsation frequencies but the results were reversed for the large line sizes at the higher pulsation frequencies. The system efficiency curves indicate that for a given line size, there exists certain pulsation frequencies that are more desirable than others.

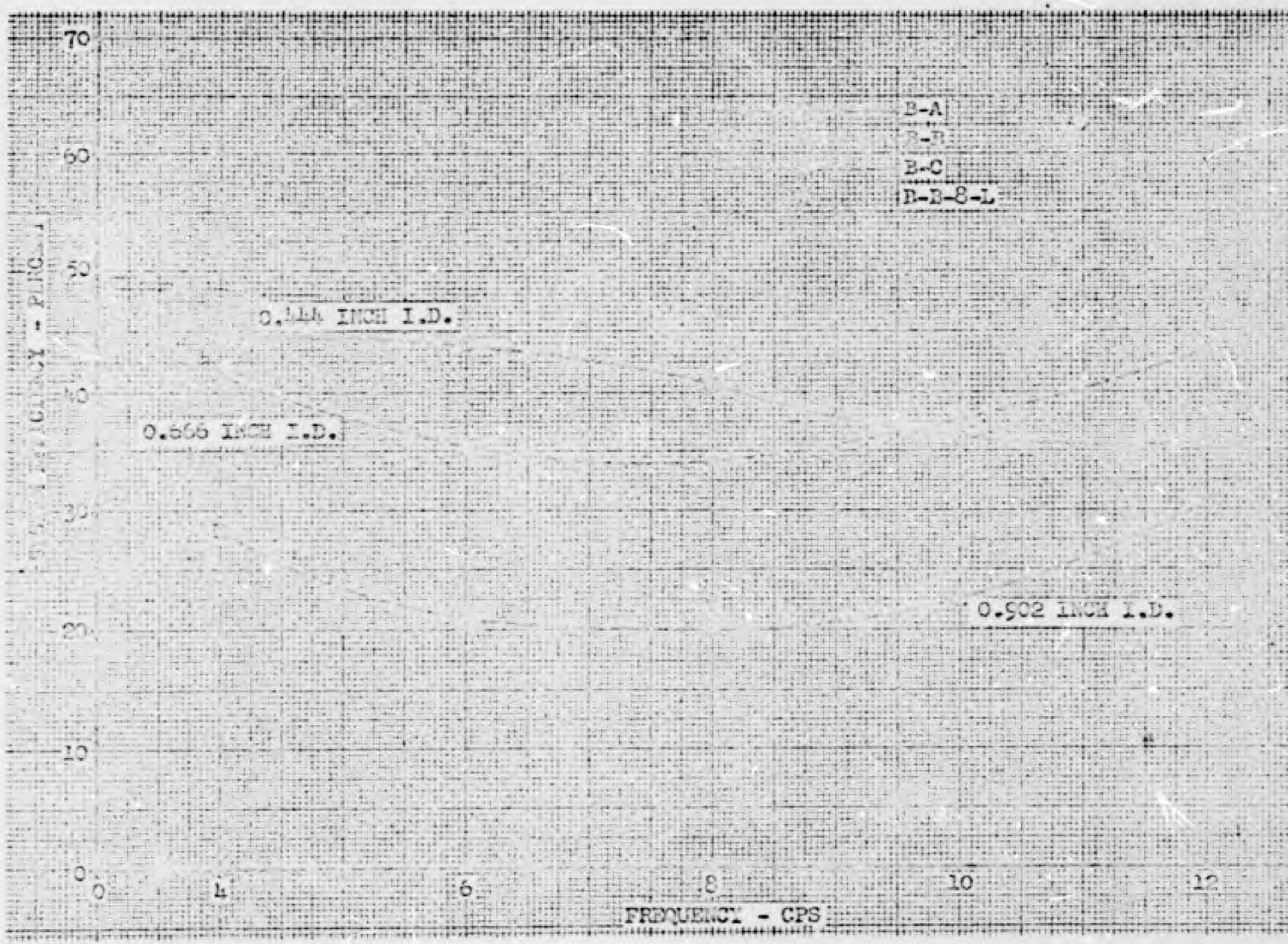
Run No. B-B-8-L had the same configuration as Run No. B-B-8, but the 125-foot transmission line was analyzed using the lumped parameter method instead of the distributed parameter method. The results of the two runs varied considerably (system efficiency varied 12 percent) which indicates the lumped parameter method would give invalid results if used to analyze the 125-foot transmission line.

The recording graphs of the time delay computer circuit were examined for the 50-foot and 125-foot transmission lines and indicated that the simulation of the time delay in the circuit was functioning properly.



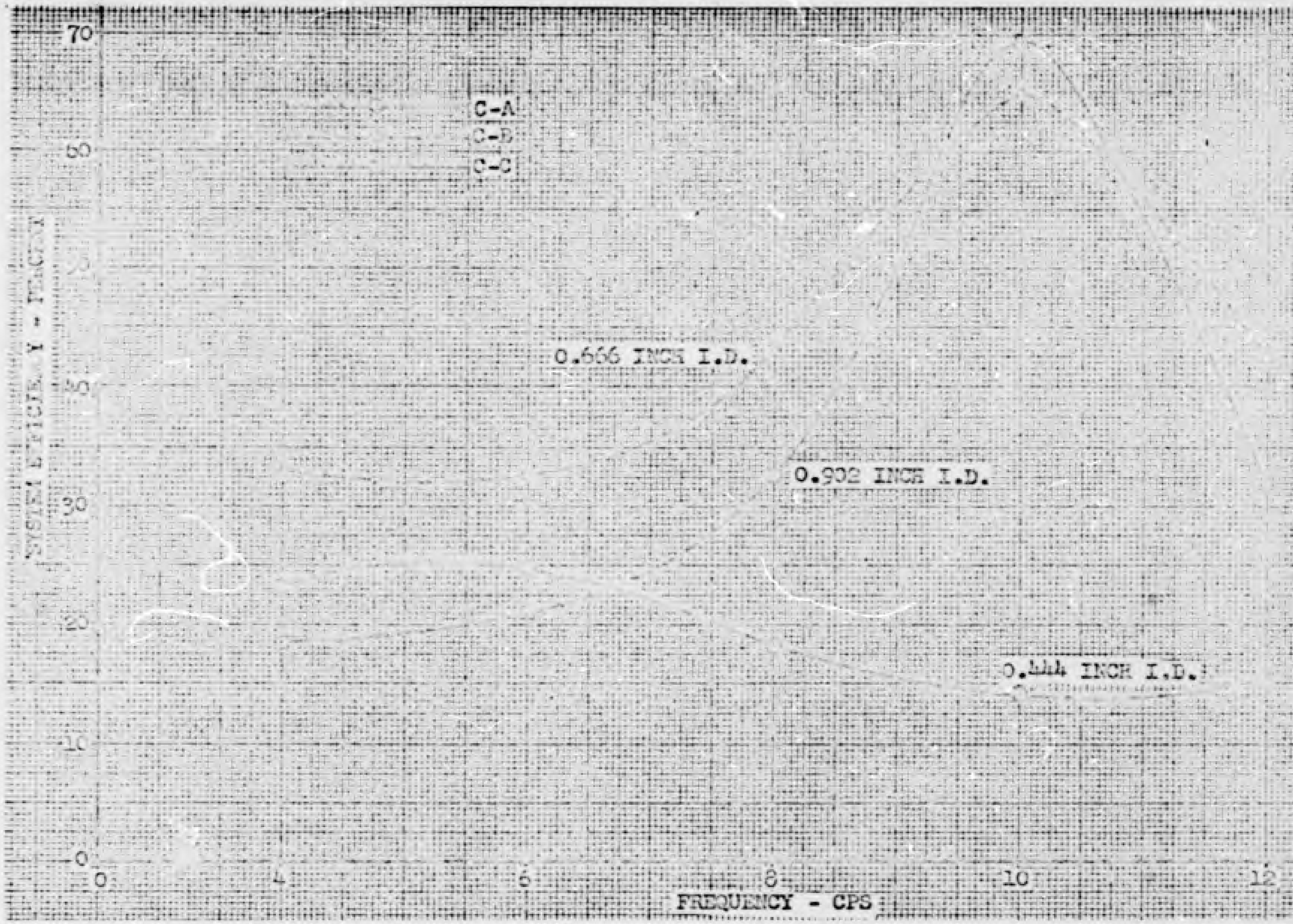
SYSTEM EFFICIENCY (UPSTREAM OF ALTERNATOR VALVE) FOR 50-FOOT TRANSMISSION LINE

FIGURE 19



SYSTEM EFFICIENCY (UPSTREAM OF ALTERNATOR VALVE) FOR 125-FOOT TRANSMISSION LINE

FIGURE 20



SYSTEM EFFICIENCY (UPSTREAM OF ALTERNATOR VALVE) FOR 200-FOOT TRANSMISSION LINE

FIGURE 21

For the 200-foot transmission line (Figure 21), the efficiency varied from 14 percent to 70 percent depending on the line size and pulsation frequency. These high values of system efficiency indicated that the data was invalid, especially at the higher pulsation frequencies. The recording graphs of the time delay computer circuit were examined which revealed that the output voltage did not follow the input voltage exactly which introduced an error in the results. This error is a function of the product of the input frequency and the delay time. The recording graphs did indicate that the time delay circuit was displaced properly.

Therefore, for the 200-foot transmission line at the higher pulsation frequencies, the simulation of the time delay in the circuit was was inadequate and the results are invalid. To achieve better results, a similar delay circuit can be cascaded to the existing circuit to extend the figure of merit value (ωT_e) to meet our program requirements.

6.3.2 System Efficiency Calculated Downstream of the Alternator Valve

The system efficiencies calculated in Section 6.3.1 were quite low when compared with the efficiency of continuous flow hydraulic systems. To determine the extent the losses through the alternator valve and the overboard losses due to the reservoir effected by the system efficiency, the system efficiency was calculated downstream of the alternator valve. Computing the system efficiency by this method, its value will not be penalized by a poor alternator valve design or overboard losses but will indicate the system efficiency due to the pulsating hydraulic circuit. Republic Aviation Corporation (Reference 3) built an alternator valve and determined that the pressure drop through the alternator valve was small.

The system efficiency for the 50-foot and 125-foot transmission lines (30 different configurations) was calculated downstream of the alternator valve. The following method was used to calculate the system efficiency:

$$\text{System Efficiency} = \frac{\text{Work Out}}{\text{Work In}} \tag{1}$$

Where,

Work Out = (Flow through Load)(Pressure Drop Across Load)

$$\text{Work Out} = \left[6.5 \text{ gpm}\right]\left[P_{11_{ave.}} - P_{12_{ave.}}\right]$$

$$\text{Work In} = \int_0^T Q_7 P_7 \, dt \tag{2}$$

The product of flow (Q_7) and pressure (P_7) was integrated with respect to time for one second of problem time. This integration was performed manually from the recording graphs.

$$\text{System Efficiency} = \frac{\left[6.5 \text{ gpm}\right] \left[P_{11_{ave.}} - P_{12_{ave.}}\right]}{\int_0^T Q_7 P_7 \, dt} \tag{3}$$

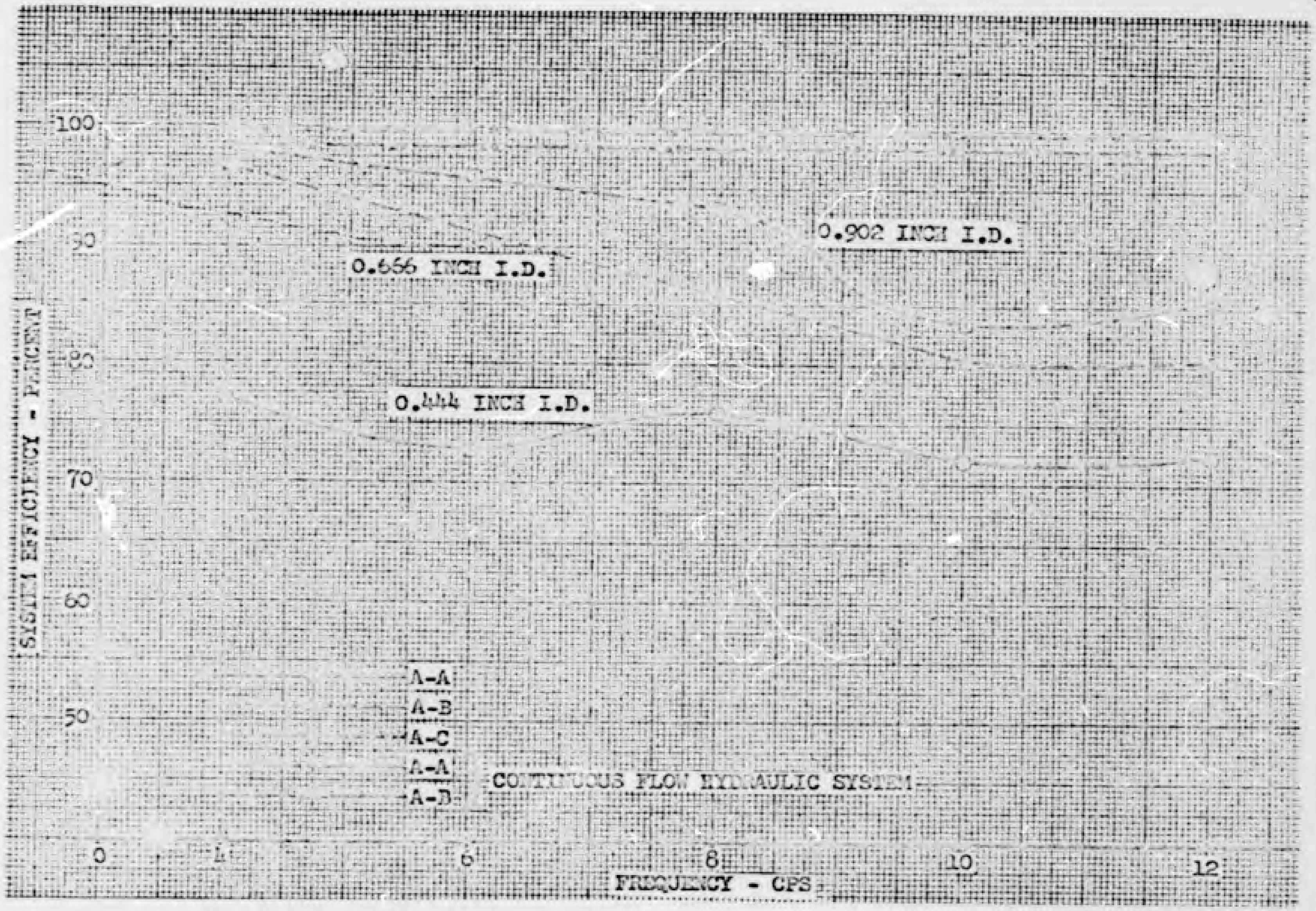
The system efficiency is plotted as a function of line length, line size, and pulsation frequency as shown in Figures 22 and 23.

For the 50-foot transmission line (Figure 22), the efficiency varied from 72 percent to 98 percent depending on the line size and pulsation frequency. These values of system efficiency are higher than the values obtained in Section 6.3.1 (Figure 19), and are nearly competitive with the system efficiency of continuous flow hydraulic systems. The system efficiency for a continuous flow hydraulic system was calculated by assuming only fluid friction losses through the lines and plotted on Figure 22. The system efficiency for the small line size (I.D. = 0.444 In.) was nearly constant throughout the frequency range and appears not to be a function of pulsation frequency. For the larger line sizes (I.D. = 0.666 In. and 0.902 In.), the system efficiency decreased with increasing values of pulsation frequency until reaching 10 cps and then leveled off.

The system efficiency was higher for the larger line sizes (opposite of the results in Section 6.3.1) which indicates that the fluid friction and inertia losses over-shadowed the fluid capacitance losses for the 50-foot transmission line. Comparing the system efficiencies of Figures 19 and 22, it appears that the losses through the alternator valve and overboard losses due to the reservoir are extremely large.

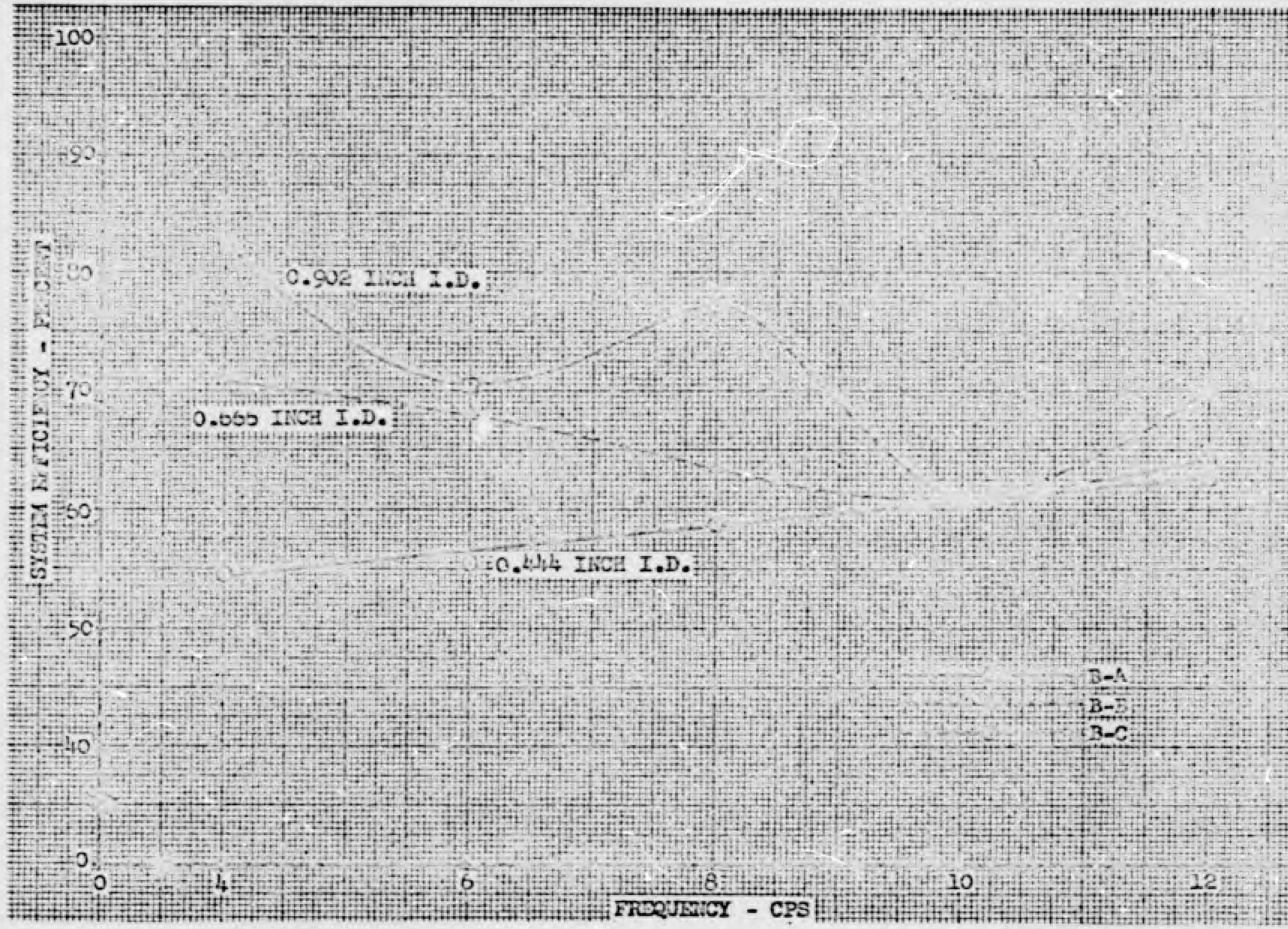
For the 125-foot transmission line (Figure 23), the efficiency varied from 55 percent to 82 percent depending on the line size and pulsation frequency. These values of system efficiency are much higher than the values obtained in Section 6.3.1 (Figure 20), but are still below the system efficiency of continuous flow hydraulic systems. One interesting result obtained from Figure 23 indicated that there was a pulsation frequency (10 cps) where the system efficiency was not a function of line size.

Since the data obtained for the 200-foot transmission line was invalid because of the inadequate time delay simulation, the system efficiency was not calculated downstream of the alternator valve.



SYSTEM EFFICIENCY (DOWNSTREAM OF ALTERNATOR VALVE) FOR 50-FOOT TRANSMISSION LINE

FIGURE 22



SYSTEM EFFICIENCY (DOWNSTREAM OF ALTERNATOR VALVE) FOR 125-FOOT TRANSMISSION LINE

FIGURE 23

THE BOEING COMPANY
MILITARY AIRPLANE DIVISION - WICHITA BRANCH

CODE IDENT. NO. 81205

NUMBER	D3-6576

SECTION	7.0

SECTION TITLE: B-52 APPLICATION STUDY

7.0 B-52 APPLICATION STUDY

7.1 Introduction

The investigation indicated that the pulsating hydraulic systems were suitable for power transmission and for certain applications, pulsating hydraulic systems have a definite advantage over continuous flow hydraulic systems. The development of useful pulsating hydraulic systems has been slow because the necessity for this type of power transmission has not occurred in many areas of fluid power.

7.2 B-52 Application

This section analyzes the advantages and dis-advantages offered by pulsating hydraulic systems pertaining to B-52 application.

7.2.1 Reliability

The pulsating hydraulic system offers a cleaner system with less contamination than a continuous flow hydraulic system. The pump is the main contaminant generating component in a hydraulic system and is isolated in a pulsating system by use of a transformer. Therefore, the contaminants of the pump will not reach any critical component in the hydraulic system.

The reliability of the B-52 hydraulic system has not created any major problems. It is doubtful that a new untried hydraulic system would be beneficial at the present time. If a new hydraulic system was installed that contains critical components, as an electro-hydraulic servo valve, the reliability could become a problem because of contamination.

7.2.2 Weight

It appears that some weight advantage could be obtained by using a single-line pulsating hydraulic system for systems containing long lines (eliminates return line). No general conclusions can be formulated as to the length of line or the amount of weight saving. Each hydraulic system must be analyzed to determine the amount of weight saving, and in some cases, the continuous flow hydraulic system may be superior.

A comparative weight analysis between a pulsating hydraulic system and a continuous flow hydraulic system for the B-52 stabilizer trim hydraulic system is presented in Appendix F. An alternator valve was installed at Station 1157 of the B-52 left body hydraulic system which converted the continuous flow to pulsating flow. This location was selected since the flow divided at this point to the brakes, crosswind trim, and landing gear systems, and to the stabilizer trim system. The pulsating flow energy was transmitted in a single-line to a rectifier valve which converted the pulsating flow to continuous flow for the stabilizer trim hydraulic system.

The results of the comparative weight analysis indicated that the pulsating hydraulic system would weigh approximately 17.0 pounds more than the existing continuous flow system. The results could be reversed if the alternator valve could have been moved forward, thus increasing the pulsating flow line length.

7.2.3 Efficiency

The system efficiency of a pulsating hydraulic system is lower than for a continuous flow hydraulic system. Development of a pure pulsating system (direct conversion from mechanical energy to pulsating hydraulic energy) would eliminate the losses through the alternator valve and overboard losses due to the reservoir. The efficiency of the above system would be comparable with the efficiency of a continuous flow hydraulic system. A dis-advantage of a system containing long lines (Section 7.2.2) is a corresponding decrease in the system efficiency (Figures 16 through 20).

7.2.4 Pressure Variation

Pulsating hydraulic systems have an advantage over continuous flow systems from its ability to change pressure in the system by the use of a transformer. The application of pressure variation is not required for the present B-52 hydraulic system.

7.2.5 High Temperatures and Nuclear Radiation

In a pulsating hydraulic system, the transformer acts as a separation point between high-temperature fluid of the pulsating portion of the system and the fluid of the lower temperature continuous flow portion. The application or usage for fluid separation are as follows:

a. Isolation of nuclear radiation.

b. Wide range of ambient temperatures (especially high temperatures).

At present, the B-52 hydraulic system is not affected with the above problem areas.

7.2.6 Motion or Speed Synchronization

The alternator valve for pulsating hydraulic systems provides an accurate flow division and may save weight over mechanical synchronization devices. The system efficiency containing an alternator valve was extremely low (Section 6.0).

7.2.7 Strength and Vibration

In the detail design of pulsating components, the fatigue effects of cyclic stress loading due to pulsation must be considered. Another problem area to be investigated is the vibration effects on the hydraulic system.

7.2.8 Maintenance

A pulsating hydraulic system requires a very tight and gas-free system, otherwise it would be spongy and response and efficiency would suffer. Care must be exercised in the filling procedure (vacuum filling) of the hydraulic system to produce a tight and gas-free system.

7.3 Conclusions

The B-52, as designed, is not readily adaptable to pulsating hydraulics. The possible advantages (weight, reliability, and synchronization) offered by a pulsating hydraulic system would be minor in comparison to the lower efficiency, strength and vibration problems, and the development of new components for the B-52.

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CODE IDENT. NO. 81205

NUMBER D3-6576

SECTION 8.0

SECTION TITLE: CONCLUSIONS AND RECOMMENDATIONS

8.0 CONCLUSIONS AND RECOMMENDATIONS

8.1 Conclusions

As a result of this research investigation, it was concluded that pulsating hydraulic systems were suitable for power transmission and for certain applications, pulsating hydraulic systems have a definite advantage over continuous flow hydraulic systems. The application feasibility study concluded that the B-52, as designed, is not readily adaptable to pulsating hydraulics. The present systems are functioning adequately and no real improvement in reliability or weight can be forecast. Those areas where it appears that pulsating hydraulic systems will have an advantage over continuous flow hydraulic systems are as follows:

a. High temperature systems

b. Synchronization

c. Cleanliness and freedom of contamination for systems that have critical components

d. Isolation of nuclear radiation

e. Pressure flexibility

Further investigations and development of the above areas must be made before any definite decisions can be made on the future of pulsating hydraulics.

8.2 Recommendations

The investigations of pulsating hydraulic systems that have been performed to date, particularly the system simulation on the analog computer, indicate certain areas where further effort is desirable. The simulation techniques should be improved (especially the transport delay circuit) by better programming and by utilization of a hybrid computer. Using the improved simulation techniques, design criteria should be developed by investigating the following areas:

a. Transmission line phenomena (resonance, time delay, efficiency, etc.)

b. Pulse generation methods to increase system efficiency

c. Pulse generation control (feedback)

d. Multi-fluid systems

e. Component selection

The following areas of application where pulsating hydraulic systems seem advantageous should be investigated:

a. Servo systems

b. Digital valving

c. High temperature systems

d. Control systems

It is further recommended that, dependent on the results of the above investigations, a single-line pulsating hydraulic system be designed, constructed, and tested to verify the computer and analytical results.

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CODE IDENT. NO. 81205

NUMBER	D3-6576

SECTION	9.0

SECTION TITLE: NOMENCLATURE

REV SYM:

E-3030 R1

9.0 NOMENCLATURE

		UNITS
A	Area of Orifice	$In.^2$
A_A	Area of Accumulator Piston	$In.^2$
A_o	Maximum Area of Orifice ($A = A_o \sin \omega t$)	$In.^2$
A_1	Area of Lumped Parameter Transmission Line	$In.^2$
A_2	Area of Distributed Parameter Transmission Line	$In.^2$
A_{T1}	Area of Transformer Piston (Upstream)	$In.^2$
A_{T2}	Area of Transformer Piston (Downstream)	$In.^2$
C	Velocity of Sound	Ft./Sec.
C_D	Discharge Coefficient	--
C_1	Scaling Factor for Diode Function Generator Curve (High Pressure Accumulator)	--
C_2	Scaling Factor for Diode Function Generator Curve (Low Pressure Accumulator)	--
D	Diameter (I.D.)	Inches
D_o	Diameter (O.D.)	Inches
D_1	Diameter of Lumped Parameter Transmission Line (I.D.)	Inches
D_2	Diameter of Distributed Parameter Transmission Line (I.D.)	Inches
E	Young's Modulus of Elasticity	psi
F	Force	Pounds
K_1	Lumped Constant for Orifice	$\frac{\text{Gal.}}{\sqrt{\text{Lb.}}\text{-In.Min.}}$

		UNITS
K_2	Constant for Lumped Parameter Transmission Line Friction Loss	$\frac{Lb.-Min.^2}{In.^2-Gal.^2}$
K_3	Constant for High Pressure Accumulator	$(psi)^{.714}-In.^3$
K_4	Constant for Accumulator	--
K_5	Constant for Distributed Parameter Transmission Line Friction Loss	$\frac{Lb.-Min.^2}{In.^2-Gal.^2}$
K_6	Constant for Distributed Parameter Transmission Line Friction Loss	$\frac{Lb.-Min.^2}{In.^2-Gal.^2}$
K_7	Constant for Low Pressure Accumulator	$(psi)^{.714}-In.^3$
K_8	Spring Constant for Transformer	Lb./Min.
M	Mass of Transformer	$\frac{Lb.-Sec.^2}{Ft.}$
Q	Flow (Subscripts refer to Figure 1)	gpm
ΔQ_H	Flow to High Pressure Accumulator	gpm
ΔQ_L	Flow to Low Pressure Accumulator	gpm
P	Pressure (Subscripts refer to Figures 1, 3, and 6)	psi
P_A	Air Pressure in Accumulator	psi
P_1	Initial Air Pressure in Accumulator	psi
ΔP	Finite Differential Pressure	psi
ΔP_{11}	Finite Differential Pressure $(P_{10} - P_{11})$	psi
ΔP_{12}	Finite Differential Pressure $(P_{12} - P_{10})$	psi
R_H	Hydraulic Resistance Variable	psi/gpm
R_N	Reynolds Number	--

		UNITS
R_R	Rectifier Valve Resistance	psi/gpm
S	Stress	psi
S.F.	Scale Factor	--
T	Wall Thickness	Inches
T_1	Arbitrary Assigned Symbol ($T_1 = \frac{1}{Z_S} P'_9 + Q_9$)	gpm
T_2	Arbitrary Assigned Symbol ($T_2 = \frac{1}{Z_S} P'_{10} - Q_{10}$)	gpm
T_e	Time for Pressure Disturbance to Travel Length of Tube	Seconds
$\mathcal{V}$	Velocity	Ft./Sec.
V	Volume	$In.^3$
V_A	Air Volume of Accumulator	$In.^3$
V_i	Initial Air Volume of Accumulator	$In.^3$
V_H	Air Volume of High Pressure Accumulator	$In.^3$
V_L	Air Volume of Low Pressure Accumulator	$In.^3$
V_8	Volume of Transformer (Upstream)	$In.^3$
V_9	Volume of Transformer (Downstream)	$In.^3$
W	Displacement of Transformer	Inches
Z_S	Characteristic Line Impedance (Without Friction)	$\frac{Lb.-Sec.}{In.^5}$
b	Transformer Damping Coefficient	$\frac{Lb.-Sec.}{In.}$
f	Friction Factor	--
f_p	Friction Factor for Pulsating Flow	--

		UNITS
k	Ratio of Specific Heats of a Perfect Gas	--
ℓ	Length	Feet
ℓ_1	Length of Lumped Parameter Transmission Line	Feet
ℓ_2	Length of Distributed Parameter Transmission Line	Feet
t	Time	Seconds
w	Weight of Transformer	Pounds
x	Displacement of Accumulator Piston	Inches
$\frac{d}{dt}$	Total Derivative with Respect to Time	--
β	Bulk Modulus of Elasticity	psi
β_e	Equivalent Bulk Modulus of Elasticity	psi
λ	Wave Length of the Fluid	Feet
μ	Absolute Viscosity	$\frac{\text{Lb.-Sec.}}{\text{In.}^2}$
ν	Kinematic Viscosity	$\text{In}^2/\text{Sec.}$
ρ	Density	$\frac{\text{Lb.-Sec.}^2}{\text{In.}^4}$
ϕ	Time Scaling Coefficient	--
ω	Frequency	Radians/Sec.

NUMBER	D3-6576

SECTION	10.0

SECTION TITLE: REFERENCES

10.0 REFERENCES

1. Research Investigation of Hydraulic Pulsation Concepts, Technical Documentary Report No. APL TDR 64-109, Wright-Patterson Air Force Base, Ohio, October 1964.

2. Henke, R., "Wave-carried Power Churns Through Motionless Fluids in A-F Hydraulics", Machine Design, February 28, 1963, 115-116.

3. Investigation of Pulsating Flow Hydraulic Concepts, Technical Documentary Report AFAPL-TR-65-83, Wright-Patterson Air Force Base, Ohio, November 1965.

4. Study of the Distributed Parameter Hydraulic Line, Technical Memorandum TM-R5-13, Vickers Incorporated, Troy, Michigan, September 1958.

5. Tests of a Capped Hydraulic Line for Distributed Effects, Technical Memorandum TM-R5-18, Vickers Incorporated, Troy, Michigan, December 1958.

6. Preliminary Study of Non-Steady Flow Hydraulics, Technical Memorandum TM-R5-61, Vickers Incorporated, Troy, Michigan, February 1961.

7. Pulsating Hydraulics Study, Proposal No. TP-R1-90, Vickers Incorporated, Troy, Michigan, May 1962.

8. Blackburn, J. F., Reethof, G., and Shearer, J. L., Fluid Power Control, Cambridge, Technology Press of Massachusetts Institute of Technology, 1960.

9. Korte, E. L., "Stable Wide-Frequency Oscillator", Simulation, January 1964, Page 31.

10. Fifer, S., Analogue Computation, Volume IV, New York, McGraw-Hill Book Company, Inc., 1961, 1244-1254.

11. Investigation and Study of B-52H Hydraulic System Overheat Problems, ECP 1039 (Study), Document T3-1292, The Boeing Company, Wichita, Kansas, July 1962.

12. Caplan, F., "Select Tube Wall Thickness with these Nomograms", Hydraulics and Pneumatics, June 1965, Page 76.

APPENDIX - A

LETTER SURVEY RESULTS

APPENDIX - A

LETTER SURVEY RESULTS

Fifty-eight percent of the questionnaires sent to universities, industries, and government agencies were returned. The questionnaire consisted of the following:

QUESTIONNAIRE

	Yes	No
1. Have you ever been engaged in a research program related to pulsating hydraulics or pulsating flow concepts?	___	___

If yes, answer below.

a. Name of program________________________
b. Supported by___________________________
c. Was a report or thesis written?__________
d. Is it available?________ Source_______________
e. Brief results of the above program

2. Do you know of any companies, universities, or government agencies other than your own who have conducted research on pulsating hydraulics?

3. List any other information regarding the theoretical background, theory and hardware applications for pulsating hydraulics (tests, references, etc.).

The organizations listed below replied to the above numbered questions as follows:

UNIVERSITIES

Air Force Institute of Technology
Mr. H. C. Larsen, School of Engineering
Wright-Patterson Air Force Base, Ohio

No reply

University of Arkansas
Dr. D. A. Gilbrech, Engineering Mechanics
Fayetteville, Arkansas

1. Yes
   a. Critical Reynolds Numbers in Pulsating Flow
   b. National Science Foundation & National Institute of Health
   c. Yes
   d. Yes - University of Arkansas Engineering Experimental Station
   e. Critical Reynolds Numbers for rigid smooth tubes with pulsating flow; some work on flexible tubes.

2. Streeter at University of Michigan
   Sarpkaya at University of Nebraska
   Ostrach at Case Institute of Technology
   O'Brien at John Hopkins University
   Vidya Division of ITEK (Blood in flexible tubes)

California Institute of Technology
Mr. Lagerstrom, Aeronautical Engineering
Pasadena, California

No reply

University of California
College of Engineering, Institute of Engineering Research
Mr. M. Holt, Mechanical Engineering
Berkeley, California

1. No

University of California
College of Engineering, Institute of Engineering Research
Mr. P. Lieber, Applied Mechanics
Berkeley, California

No reply

University of California
College of Engineering, Institute of Industrial Cooperation
Mr. J. W. Miles
Los Angeles 24, California

1. No

Carnegie Institute of Technology
Mechanical Engineering Department, Mr. W. T. Rouleau
Schenley Park, Pittsburg 13, Pennsylvania

1. No
2. Republic Aviation Corporation
3. "Pulsating Flow Hydraulic Systems," Republic Aviation Corporation, RAC-1932-2, NASA N65-14098

Carnegie Institute of Technology
Civil Engineering Department, Mr. T. E. Stelson
Schenley Park, Pittsburg 13, Pennsylvania

No reply

University of Michigan, College of Engineering
Mr. Victor L. Streeter, Civil Engineering
Ann Arbor, Michigan 48104

1. Yes.
   a. Hydraulic Transient Research
   b. University of Michigan, NSF, and Union Pump Co. of Battle Creek, Michigan
   c. Yes
   d. Limited bases - Brief report to be published by Institute of Mechanical Engineering in London, November 1965.
   e. Theory and experiment agree closely in study of transients in reciprocating pumps. Impedance and characteristics methods of analysis used with digital computer.

2. Oklahoma State University - Pat. No. 101, 102, 107, Experiment Station, June 58 - August 1959.

3. Wylie, E.B., "Resonance in Pressurized Piping Systems," ASME Paper No. 65-FE 6. Wylie, E. B., and V. L. Streeter, "Resonance in No. 2 Piping System," ASME Paper No. 65-FE 10.

   Fashbough, R.H., and V.L. Streeter, "Resonance in Liquid Rocket Engine Systems," ASME Paper No. 65-FE 23.

Illinois Institute of Technology
Research Institute, Hydraulics
Mr. T. P. Torda, 10 West 35th Street
Chicago 16, Illinois

Questionnaire forwarded to Dr. G. H. Strohmeier, Vice-President, IIT Research Institute for action.

No reply

Iowa Institute of Hydraulic Research
Hydraulics Laboratory, Mr. H. Rouse
University of Iowa
Iowa City, Iowa

1. No

Engineering Administration Offices,
104 Marston Hall, Dept. of Engineering Mechanics
Mr. D. F. Young, Iowa State University
Ames, Iowa

1. Yes
   a. Unsteady state fluid mechanics
   b. Iowa Engineering Experiment Station, Iowa State University

c. Yes
d. Yes
e. "Electronic Analog Computer for Laminar Flow Problems," Proceedings of the American Society for Civil Engineers, Engr. Mech., Div., 87, 1961. Our effort in pulsating flow has been on biomedical fluid mechanics.

2. University of Minnesota

3. McDonald, D. A., "Blood Flow in Arteries," Williams and Wilkins, Baltimore, Maryland, 1960

John Hopkins University, School of Engineering Science
Mr. Stanley Corrsin, Department of Mechanics
34th & Charles Streets
Baltimore 18, Maryland

1. No
2. Dr. Vivian O'Brien, Applied Physics Laboratory
   P.O. Box 244, R.F.D.-1
   Laurel, Howard County, Maryland
3. Text Book: Richardson, E.G., "Fluid Dynamics"

University of Michigan
Mr. R. B. Keller, Mechanical Engineering Department
Ann Arbor, Michigan

1. No

University of Minnesota
St. Anthony Falls Hydraulic Laboratory, Mr. L. G. Straub
Minneapolis 14, Minnesota

No reply

University of Nebraska
Professor N. H. Barnard, Mechanical Engineering
Lincoln 8, Nebraska

No reply

University of Nebraska
Dr. Turgut Sarpkaya, Engineering Mechanics
Lincoln 8, Nebraska

1. Yes
   a. Mechanism of Turbulence Generation in Pulsating Flow

b. U.S. Army Research Office, Durham, North Carolina
c. Yes
d. Yes - U.S. Army Research Office, Durham, North Carolina
e. Periodic pulsations superimposed upon steady Poiseuille flow rendered the flow more stable and that neutral stability curves are determined uniquely by the average Reynolds number of the ambient flow, a frequency parameter, and a velocity ratio.

2. Additional information may be furnished only at cost.

Oklahoma State University of Agriculture & Applied Science
Mr. J. D. Parker, College of Engineering
Stillwater, Oklahoma

No reply

Pennsylvania State University
Mr. J. L. Shearer, Mechanical Engineering
University Park, Pennsylvania

1. No

University of Pittsburgh
Mr. G. E. Geiger, Mechanical Engineering Department
405 Engineering Hall
Pittsburgh 13, Pennsylvania

1. No
2. No
3. University of Pittsburgh - Knowledge Availability System

Purdue University, School of Engineering
Mr. R. W. Fox, Mechanical Engineering
Lafayette, Indiana

No reply

Rensselaer Polytechnic Institute
Dr. E. A. Saibel, Mechanics Department
Troy, New York

1. No
2. Indian Institute of Technology in Bombay and the Indian Institute of Technology in Kharagpur, India.

South Dakota State College of Agriculture & Mechanics Arts
Engineering Experiment Station
Mr. J. F. Sandfort, State College Station
Brookings, South Dakota

No reply

Stanford University
Mr. J. B. Franzini, Civil Engineering Department
Stanford, California

1. No

Stevens Institute of Technology
Mechanical Engineering Department
Mr. Kurt H. Weil, Castle Point Station
Hoboken, New Jersey

1. No

University of Tennessee
Mechanical Engineering Department
Mr. J. F. Bailey, Perkins Hall
Knoxville, Tennessee

1. No

University of Texas
Mechanical Engineering Department
Dr. W. J. Carter, P.O. Box 7977
Austin 12, Texas

1. Yes
    a. Periodic Flow of Viscous Fluids
    b. Boeing Aircraft Co.
    c. Yes
    d. Yes - BAC D-17188
    e. Frequency response analysis of fluid circuits (circular conduits and non-parabolic velocity profiles).

2. No

Tufts University
Mechanical Engineering Department
Prof. K. N. Astill, Anderson Hall
Medford 55, Massachusetts

1. No

2. Hill P., "Pulsating Flow Experiments," Ph. D. Thesis (M.I.T.-1959)
   Streeter, V.L., - University of Michigan
   Herzl, "An Analysis of a Self Exited Oscillator," ASME Paper 64, FE 24.

Washington State Institute of Technology
Mr. E. R. Tinney, Hydraulics
Pullman, Washington

No reply

University of Wisconsin
Mr. H. L. Harrision, Mechanical Engineering Building
1513 University Avenue
Madison 6, Wisconsin

1. No

Worcester Polytechnic Institute
Alden Hydraulic Laboratory, Prof. Leslie J. Hooper
Worcester 9, Massachusetts

1. No

University of Wyoming
Mr. Eric J. Lindahl, Mechanical Engineering
Laramie, Wyoming

No reply

University of Alabama, Department of Engineering Mechanics
Dr. Harold R. Henry, P.O. Box 6204
University, Alabama 35486

1. No

Case Institute of Technology
Dr. Charles K. Taft, University Circle
Cleveland 6, Ohio

1. No

Catholic University of America
Dr. Paul Chang, Mechanical Engineering Department
620 Michigan Avenue
N.E., Washington, D. C.

No reply

Cornell Aeronautical Laboratory, Inc.
Mr. O. F. Giombini, Cornell University
P.O. Box 235
Buffalo, New York 14221

No reply

University of Illinois, Professor J. M. Robertson
Department of Theoretical and Applied Mechanics
212 Talbot Laboratory
Urbana, Illinois

No reply

Maryland University, Professor John Weske
Institute for Fluid Dynamics and Applied Mathematics
College Park, Maryland

No reply

Rutgers, The State University
Department of Mechanical Engineering
Mr. R. H. Page, University Heights Campus
New Brunswick, New Jersey 08903

No reply

INDUSTRIES AND GOVERNMENT AGENCIES

American Brake Shoe Co.
Mr. Raymond H. Schaefer, Research Director
530 5th Ave., New York 36, New York

Received two replies

First - Aerospace Division, Mr. Edwin L. Shaw,
Vice President Engineering and Research
Oxnard, California

Like most companies engaged in the hydraulic field, we have studied in a cursory nature both the AC type power transmission and the fluid amplifiers. There are some comments which we would like to offer. The transmission of power by AC hydraulic system is an old idea. Synchronous operation of input and output devices have been made for low power level equipment. Unfortunately, when large amounts of power are transmitted reversing flows show a serious problem in line losses, due primarily to the compliance of the fluid and the problem of reversing the fluid direction. Transmitting power by alternating pressure also proves to be extremely difficult, in that a large amount of loss is encountered due to the heating of the alternate compression expansion of the fluid. The studies we have made show no particular weight or space or reliability savings by going to alternating flow or pressure systems.

<u>Second</u> - Hydrodynamics Research Center
Mr. P. B. Burnham, Chief Engineer, System Research
Columbus, Ohio

We have done some brain-storming on "AC Hydraulics" in the past years and came to the conclusions that they are not suitable for systems involving any degree of power transfer. We came to this conclusion since the working fluid does have considerable mass and compressibility associated with it, thereby introducing a considerable amount of dynamics (gain and phase lag characteristics) which could very easily make a system have poor response at best and possibly be inoperable. Also since a transmission line cannot be made with a very low resistance, problems of localized heating at pressure modes in "AC Hydraulics" can be considerable.

Bendix Corporation Research Laboratories Division
Mr. J. B. Kilmer, Manager, Development Planning
Southfield, Michigan

Bendix has conducted in-house studies on pulsating hydraulics. Our tentative conclusions question the advantages of pulsating systems over conventional systems for normal environments up to 700-800° F. However for higher temperatures, 1200° F and up, a pulsating system may have advantage in that a dual fluid system is possible. Such a system would allow the use of Na K or the equivalent in the high temperature environment, while conventional hydraulic fluid could be used for the pressure generation and control in a cooler environment (use of transformer). Our current efforts in pulsating hydraulics are concerned with the development of a proprietary rotary actuator and transmission.

Cook Electric Company
Mr. Maurice G. Hughett, Vice President and General Manager
2700 N. Southport Street
Chicago 14, Illinois

No reply

Electrol, Inc.
Mr. H. M. Billington, Ass't General Manager
85 Grand Street
Kingston, New York

1. No

Gar Precision Parts, Inc.
Mr. Jeremy C. Rice, Technical Director
190 Henry Street
Stamford, Connecticut

1. No

International Business Machines Corporation
Mr. Howard K. Janis, Manager, Technical Information
590 Madison Avenue
New York 22, New York

1. No

Parker Aircraft Company
Mr. Charles E. Cleminshaw, Vice-President
5827 W. Century Blvd.
Los Angeles 43, California

1. No

Westinghouse Electric Corporation
Mr. F. K. Fischer, Development Engineering Laboratory
Essington, Pennsylvania

No reply

National Water Lift Company
Mr. J. J. Jerger, Director of Engineering
2220 Parker Avenue
Kalamazoo, Michigan

No reply

John S. Barnes Corporation
Mr. E. J. Svenson, Research Director
315 South Madison Street
Rockford, Illinois

No reply

Stewart-Warner Corporation
Mr. J. W. Mahanay, South Wind Division
1514 Drover Street
Indianapolis, Indiana

1. No

Hydro-Aire Company
Mr. D. B. Nickerson, Chief Engineer
3000 Winona Avenue
Burbank, California

No reply

Harry Diamond Laboratories
Mr. R. N. Gottron
Washington 25, D. C.

No reply

Honeywell Regulator Co.
Aeronautical Division, Mr. R. W. Mueller
2600 Ridgway Road
Minneapolis 40, Minnesota

1. No

Mr. R. L. Murphy, ATSC-AC/HEAD
Contractor Reports Unit
Scientific & Technical Information Division
National Aeronautics and Space Administration
Washington, D. C. 20546

No reply

Vickers, Inc.
Dr. W. W. Chao, Director of Research and Development
Administrative and Engineering Center
Detroit 32, Michigan

Vickers has been conducting research in the areas of pulsating hydraulics and fluid amplifiers; however, the investigations are conducted with company funds and are a proprietary nature.

APPENDIX - B

COMPUTER SCALED VARIABLES

APPENDIX - B

COMPUTER SCALED VARIABLES

PROBLEM VARIABLE	ESTIMATED MAXIMUM	COMPUTER SCALED VARIABLE
$A_{P \text{ or } R}$	0.200 in^2	$[500\ A_{P \text{ or } R}]$
F	10,000 lbs	$[.01\ F]$
P_7	4,000 psi	$[.025\ P_7]$
P_8	4,000 psi	$[.025\ P_8]$
P_9	4,000 psi	$[.025\ P_9]$
P'_9	4,000 psi	$[.025\ P'_9]$
P_{10}	4,000 psi	$[.025\ P_{10}]$
P'_{10}	4,000 psi	$[.025\ P'_{10}]$
P_{11}	4,000 psi	$[.025\ P_{11}]$
P_{12}	4,000 psi	$[.025\ P_{12}]$
P_M	4,000 psi	$[.025\ P_M]$
$\dot{P}_7$	500,000 psi/sec	$[.0002\ \dot{P}_7]$
$\dot{P}_8$	1,000,000 psi/sec	$[.0001\ \dot{P}_8]$
$\dot{P}_9$	1,000,000 psi/sec	$[.0001\ \dot{P}_9]$
$\Delta P_{3 \text{ or } 4}$	4,000 psi	$[.025\ \Delta P_{3 \text{ or } 4}]$
ΔP_{11}	400 psi	$[.25\ \Delta P_{11}]$
ΔP_{12}	400 psi	$[.25\ \Delta P_{12}]$
$\sqrt{\Delta P_{3 \text{ or } 4}}$	$63_{(psi)}{}^{1/2}$	$[1.582\ \sqrt{\Delta P_{3 \text{ or } 4}}]$

PROBLEM VARIABLE	ESTIMATED MAXIMUM	COMPUTER SCALED VARIABLE
Q_7	80 gpm	$\left[1.25\ Q_7\right]$
Q_7/K_1	12.6 in.-$\sqrt{\text{lb.}}$	$\left[7.91\ Q_7/K_1\right]$
Q_8	80 gpm	$\left[1.25\ Q_8\right]$
$Q_8\|Q_8\|$	6400 $(\text{gpm})^2$	$\left[.0156\ Q_8\|Q_8\|\right]$
Q_9	80 gpm	$\left[1.25\ Q_9\right]$
$Q_9\|Q_9\|$	6400 $(\text{gpm})^2$	$\left[.0156\ Q_9\|Q_9\|\right]$
Q_{10}	80 gpm	$\left[1.25\ Q_{10}\right]$
$Q_{10}\|Q_{10}\|$	6400 $(\text{gpm})^2$	$\left[.0156\ Q_{10}\|Q_{10}\|\right]$
Q_{11}	80 gpm	$\left[1.25\ Q_{11}\right]$
Q_{12}	80 gpm	$\left[1.25\ Q_{12}\right]$
Q_L	80 gpm	$\left[1.25\ Q_L\right]$
$\dot{Q}_8$	20,000 gpm/sec	$\left[.005\ \dot{Q}_8\right]$
ΔQ_H	80 gpm	$\left[1.25\ \Delta Q_H\right]$
ΔQ_L	80 gpm	$\left[1.25\ \Delta Q_L\right]$
W	10 inches	$\left[10\ W\right]$
$\dot{W}$	40 in/sec	$\left[2.5\ \dot{W}\right]$
$\ddot{W}$	100,000 in/sec^2	$\left[.001\ \ddot{W}\right]$
T_1	200 gpm	$\left[.5\ T_1\right]$
T_2	200 gpm	$\left[.5\ T_2\right]$
V_H	1000 in^3	$\left[.1\ V_H\right]$

PROBLEM VARIABLE	ESTIMATED MAXIMUM	COMPUTER SCALED VARIABLE
V_L	1000 in.^{3}	$\left[.1\ V_L\right]$
V	100 in.^{3}	$\left[V\right]$

APPENDIX - C

COMPUTER EQUATIONS

APPENDIX - C

COMPUTER EQUATIONS

1. $\left[500\ A_{P\ or\ R}\right] = 500\ A_o \sin \omega t$

2. $\left[.025\ \Delta P_3\right] = \left[.025\ P_3\right] - \left[.025\ P_7\right]$

3. $\left[.025\ \Delta P_4\right] = \left[.025\ P_4\right] - \left[.025\ P_7\right]$

4. $\left[1.582 \sqrt{\Delta P_{3\ or\ 4}}\right] = 10 \sqrt{.025\ \Delta P_{3\ or\ 4}}$

5. $\left[7.91\ Q_7/K_1\right] = \frac{1}{100} \left[500\ A_{P\ or\ R}\right] \left[1.582 \sqrt{\Delta P_{3\ or\ 4}}\right]$

6. $\left[1.25\ Q_7\right] = 5.22 \left[7.91\ Q_7/K_1\right]$

7. $\left[.0002\ \dot{P}_7\right] = 3.16 \left[1.25\ Q_7\right] - 3.16 \left[1.25\ Q_8\right]$

8. $\left[.025\ P_7\right] = 125/\phi \int \left[.0002\ \dot{P}_7\right] dt$

9. $\left[.0156\ Q_8 |Q_8|\right] = \frac{1}{100} (1.25\ Q_8 |1.25\ Q_8|)$

10. $\left[.025\ P_M\right] = \left[.025\ P_7\right] - .0616 \left[.0156\ Q_8 |Q_8|\right]$

11. $\left[.005\ \dot{Q}_8\right] = 1.91 \left[.025\ P_M\right] - 1.91 \left[.025\ P_8\right]$

12. $\left[1.25\ Q_8\right] = 250/\phi \int \left[.005\ Q_8\right] dt$

13. $\left[.01\ F\right] = 6.57 \left[.025\ P_8\right] - 6.57 \left[.025\ P_9\right]$

14. $\left[.001\ \ddot{W}\right] = -.0114 \left[2.5\ \dot{W}\right] - .142 \left[10\ W\right] + 2.84 \left[.01\ F\right]$

15. $\left[2.5\ \dot{W}\right] = 2500/\phi \int \left[.001\ \ddot{W}\right] dt$

16. $\left[10\ W\right] = 4/\phi \int \left[2.5\ \dot{W}\right] dt$

17. $\left[.0001\ \dot{P}_8\right] = 2.88 \left[1.25\ Q_8\right] - 6.16 \left[2.5\ \dot{W}\right]$

18. $\left[.025\ P_8\right] = 250/\phi \int \left[.0001\ \dot{P}_8\right] dt$

19. $\left[.0001\ \dot{P}_9\right] = 6.16 \left[2.5\ \dot{W}\right] - 2.88 \left[1.25\ Q_9\right]$

20. $\left[.025\ P_9\right] = 250/\phi \int \left[.0001\ \dot{P}_9\right] dt$

21. $\left[.5\ T_1\right] = .441 \left[.025\ P'_9\right] + .40 \left[1.25\ Q_9\right]$

22. $\left[.025\ P'_{10}\right] = 2.28 \left[.5\ T_1\ (t - T_e)\right] - .910 \left[1.25\ Q_{10}\right]$

23. $\left[.5\ T_2\right] = .441 \left[.025\ P'_{10}\right] - .40 \left[1.25\ Q_{10}\right]$

24. $\left[1.25\ Q_9\right] = 1.100 \left[.025\ P'_9\right] - 2.5 \left[.5\ T_2\ (t - T_e)\right]$

25. $\left[.25\ \Delta P_{11}\right] = 10 \left[.025\ P_{10}\right] - 10 \left[.025\ P_{11}\right]$

26. $\left[1.25\ Q_{11}\right] = 2.71 \left[.25\Delta P_{11}\right]$

27. $\left[.25\Delta P_{12}\right] = 10 \left[.025\ P_{12}\right] - 10 \left[.025\ P_{10}\right]$

28. $\left[1.25\ Q_{12}\right] = 2.71 \left[.25\Delta P_{12}\right]$

29. $\left[1.25\ Q_{10}\right] = \left[1.25\ Q_{11}\right] - \left[1.25\ Q_{12}\right]$

30. $\left[1.25\Delta Q_H\right] = \left[1.25\ Q_{11}\right] - \left[1.25\ Q_L\right]$

31. $\left[1.25\Delta Q_L\right] = \left[1.25\ Q_L\right] - \left[1.25\ Q_{12}\right]$

32. $\left[.025\ P_{10}\right] = \left[.025\ P'_{10}\right] - .384 \left[.0156\ Q_{10}|Q_{10}|\right]$

33. $\left[.025\ P'_9\right] = \left[.025\ P_9\right] - .384 \left[.0156\ Q_9|Q_9|\right]$

34. $C_1 \left[.1\ V_H\right] = .308\ C_1/\phi \int \left[1.25\Delta Q_H\right] dt$

35. $(10^3\ C_1{}^{-.7})^2/10^2 = 10^4\ C_1{}^{-1.4}\ V_H{}^{-1.4}$

36. $\left[.025\ P_{11}\right] = 30 \left[\dfrac{10^4\ C_1{}^{-1.4}\ K_3{}^{-1.4}}{1.601}\ P_{11}\right]$

37. $C_2 \left[.1\ V_L\right] = .308\ C_2/\phi \int \left[1.25\Delta Q_L\right] dt$

38. $(10^3\ C_2{}^{-.7}\ V_L{}^{-.7})^2/10^2 = 10^4\ C_2{}^{-1.4}\ V_L{}^{-1.4}$

39. $\left[.025\ P_{12}\right] = 1.5 \left[\dfrac{10^4\ C_2{}^{-1.4}\ K_7{}^{-1.4}}{1.601}\ P_{12}\right]$

40. $\left[V\right] = 16.10/\phi \int \left[7.91\ Q_7/K_1\right] dt$

APPENDIX - D

POTENTIOMETER SETTINGS

APPENDIX - D

POTENTIOMETER SETTINGS

POT. NO.	FORMULA	EXPLANATION OR PURPOSE
P00	ω	Alternator Valve Frequency
P01		Initial Condition on Oscillation - Integrator 01
P02	ω	Alternator Valve Frequency
P05		Obtained from Magnitude and Time Scaling
P06	P_4	Reservoir Pressure (50 psi) - Magnitude Scaling
P07	$(\frac{\beta_e}{A_1 \ell_1})$(S.F.)	Lumped Transmission Line Parameters Times Magnitude Scaling
P08		Initial Condition on P_7 - Integrator 05
P09	K_2 (S.F.)	Friction Loss Parameter of Lumped Transmission Line Times Magnitude Scaling
P10	$(\frac{\beta_e}{A_1 \ell_1})$(S.F.)	Lumped Transmission Line Parameters Times Magnitude Scaling
P11	$(\frac{A_1}{\rho \ell_1})$(S.F.)	Lumped Transmission Line Parameters Times Magnitude Scaling
P12	$(\frac{A_1}{\rho \ell_1})$(S.F.)	Lumped Transmission Line Parameters Times Magnitude Scaling
P14	$(\frac{\beta}{V_8})$(S.F.)	Transformer Compressibility Parameters Times Magnitude Scaling
P15		Initial Condition on P_8 - Integrator 15
P18	$(\frac{K_8}{M})$(S.F.)	Transformer Parameters Times Magnitude Scaling
P19	$(\frac{1}{M})$(S.F.)	Transformer Parameters Times Magnitude Scaling
P20	$(\frac{\beta}{V_9})$(S.F.)	Transformer Compressibility Parameters Times Magnitude Scaling
P21	$(\frac{A_1 \beta}{V_9})$(S.F.)	Transformer Compressibility Parameters Times Magnitude Scaling

POT NO.	FORMULA	EXPLANATION OR PURPOSE
P22		Initial Condition on P_9 - Integrator 22.
P36		Initial Condition on High Pressure Accumulator (Volume of Air) - Integrator 36
P46		Initial Condition on Low Pressure Accumulator (Volume of Air) - Integrator 46
P51	Q_L	Flow Demand (6.5 gpm) - Magnitude Scaling
P60		Magnitude Control of Frequency Oscillator
Q03		Alternator Valve Area Control
Q04	$\frac{1}{K_1}$ (S.F.)	Alternator Valve Parameters Times Magnitude Scaling
Q09	P_3	Source Pressure (4,000 psi) - Magnitude Scaling
Q10		Obtained from Magnitude and Time Scaling
Q14	$(\frac{A_1\beta}{V_8})$(S.F.)	Transformer Compressibility Parameters Times Magnitude Scaling
Q15		Obtained from Magnitude and Time Scaling
Q16		Obtained from Magnitude and Time Scaling
Q18	$\frac{1}{A_1}$(S.F.), $\frac{1}{A_2}$(S.F.)	Gain Control for Transformer Force Equation
Q19	$(\frac{b}{M})$(S.F.)	Transformer Parameters Times Magnitude Scaling
Q20		Obtained from Magnitude and Time Scaling
Q22		Obtained from Magnitude and Time Scaling
Q24	(S.F.)$C_2^{1.4}K_7^{1.4}$(1.601)	Low Pressure Accumulator Parameters Times Magnitude Scaling

POT. NO.	FORMULA	EXPLANATION OR PURPOSE
Q32	K_1 (S.F.)	Efficiency Calculation - Obtained from Magnitude and Time Scaling
Q35	C_1 (S.F.)	High Pressure Accumulator Air Volume Calculation - Obtained from Magnitude and Time Scaling
Q45	C_2 (S.F.)	Low Pressure Accumulator Air Volume Calculation - Obtained from Magnitude and Time Scaling
Q49	(S.F.)$C_1^{1.4}K_3^{1.4}$(1.601)	High Pressure Accumulator Parameters Times Magnitude Scaling
Q50		Time Delay Circuit - Magnitude Scaling
Q51		Obtained from Transport Delay Circuit Parameters and Time Scaling [1]
Q52		Time Delay Circuit - Magnitude Scaling
Q53		Obtained from Transport Delay Circuit Parameters and Time Scaling [1]
Q54		Obtained from Transport Delay Circuit Parameters and Time Scaling [1]
Q58		Obtained from Transport Delay Circuit Parameters and Time Scaling [1]
Q59		Obtained from Transport Delay Circuit Parameters and Time Scaling [1]
Q60		Obtained from Transport Delay Circuit Parameters and Time Scaling [1]
Q61		Obtained from Transport Delay Circuit Parameters and Time Scaling [1]
Q62	(S.F.) $\frac{1}{Z_S}$	Reciprocal of Line Characteristic Impedance Times Magnitude Scaling
Q63		Obtained from Transport Delay Circuit Parameters and Time Scaling [1]

POT. NO.	FORMULA	EXPLANATION OR PURPOSE
Q64		Obtained from Transport Delay Circuit Parameters and Time Scaling [1]
Q67		Time Delay Circuit - Magnitude Scaling
Q68		Obtained from Transport Delay Circuit Parameters and Time Scaling [1]
Q69		Obtained from Transport Delay Circuit Parameters and Time Scaling [1]
Q70	(S.F.) Z_S	Line Characteristic Impedance Times Magnitude Scaling
Q71		Obtained from Transport Delay Circuit Parameters and Time Scaling [1]
Q74	(S.F.) $\frac{1}{Z_S}$	Reciprocal of Line Characteristic Impedance Times Magnitude Scaling
Q79	K_5 (S.F.)	Friction Loss Parameter of Distributed Transmission Line Times Magnitude Scaling
Q80		Obtained from Magnitude Scaling
Q81	(S.F.) Z_S	Line Characteristic Impedance Times Magnitude Scaling
Q82	(S.F.) $\frac{1}{Z_S}$	Reciprocal of Line Characteristic Impedance Times Magnitude Scaling
Q83		Obtained from Magnitude Scaling
Q84		Obtained from Magnitude Scaling
Q89	K_5 (S.F.)	Friction Loss Parameter of Distributed Transmission Line Times Magnitude Scaling
Q90	$\frac{1}{R_R}$ (S.F.)	Reciprocal of Rectifier Valve Parameter Times Magnitude Scaling

POT. NO.	FORMULA	EXPLANATION OR PURPOSE
Q92	$\frac{1}{R_R}$ (S.F.)	Reciprocal of Rectifier Valve Parameter Times Magnitude Scaling

$\triangleright$ 1 Method for determining pots for four root - four pole transport delay simulation on Page 104.

Method for determining pots for four root - four pole transport delay simulation.

T_e = Time for pressure disturbance to travel length of tube (times 100 for $\frac{1}{100}$ time scale).

Calculate,

$a = \frac{5.28}{T_e}$, 2a and 4a

$b = 0.34\ a$

$c = 0.71a$, 2c and 4c

$d = 1.5c$

$a^2 + b^2$

$c^2 + d^2$

POT NO.	VALUE
Q51 and Q61	$a^2 + b^2$
Q53 and Q63	4a
Q54 and Q64	4c
Q58 and Q68	2a
Q59 and Q69	2c
Q60 and Q71	$c^2 + d^2$

APPENDIX - D POTENTIOMETER SETTINGS

POT. NO.	RUN A-A	RUN A-B	RUN A-C	RUN B-A	RUN B-B	RUN B-C	RUN C-A	RUN C-B	RUN C-C
	Settings (Nominal)								
P00					1				
P01	1.0000	1.0000	1.0000	1.0000	1.0000	1.0000	1.0000	1.0000	1.0000
P02					1				
P05	.2500	.2500	.2500	.2500	.2500	.2500	.2500	.2500	.2500
P06	.0125	.0125	.0125	.0125	.0125	.0125	.0125	.0125	.0125
P07	.7150	.3160	.1730	.7150	.3160	.1730	.7150	.3160	.1730
P08	.0250	.0250	.0250	.0250	.0250	.0250	.0250	.0250	.0250
P09	.4670	.0616	.0135	.4670	.0616	.0135	.4670	.0616	.0135
P10	.7150	.3160	.1730	.7150	.3160	.1730	.7150	.3160	.1730
P11	.0848	.1910	.3496	.0848	.1910	.3496	.0848	.1910	.3496
P12	.0848	.1910	.3496	.0848	.1910	.3496	.0848	.1910	.3496
P14	.2880	.2880	.2880	.2880	.2880	.2880	.2880	.2880	.2880
P15	.0250	.0250	.0250	.0250	.0250	.0250	.0250	.0250	.0250
P18	.1420	.1420	.1420	.1420	.1420	.1420	.1420	.1420	.1420
P19	.2840	.2840	.2840	.2840	.2840	.2840	.2840	.2840	.2840
P20	.2880	.2880	.2880	.2880	.2880	.2880	.2880	.2880	.2880
P21	.6160	.6160	.6160	.6160	.6160	.6160	.6160	.6160	.6160
P22	.0250	.0250	.0250	.0250	.0250	.0250	.0250	.0250	.0250
P36	.5000	.5000	.5000	.5000	.5000	.5000	.5000	.5000	.5000
P46	.5000	.5000	.5000	.5000	.5000	.5000	.5000	.5000	.5000
P51	.0812	.0812	.0812	.0812	.0812	.0812	.0812	.0812	.0812
P60	.0068	.0068	.0068	.0068	.0068	.0068	.0068	.0068	.0068

1 The potentiometer settings for P00 and P02 were a function of frequency.

FREQUENCY	P00 AND P02
4 cps	.2512
6 cps	.3768
8 cps	.5024
10 cps	.6282
12 cps	.7536

APPENDIX - D POTENTIOMETER SETTINGS

POT. NO.	Settings (Nominal) RUN A-A	RUN A-B	RUN A-C	RUN B-A	RUN B-B	RUN B-C	RUN C-A	RUN C-B	RUN C-C
Q03	.2500	.2500	.2500	.2500	.2500	.2500	.2500	.2500	.2500
Q04	.1915	.1915	.1915	.1915	.1915	.1915	.1915	.1915	.1915
Q09	1.0000	1.0000	1.0000	1.0000	1.0000	1.0000	1.0000	1.0000	1.0000
Q10	.5000	.5000	.5000	.5000	.5000	.5000	.5000	.5000	.5000
Q14	.6160	.6160	.6160	.6160	.6160	.6160	.6160	.6160	.6160
Q15	.5000	.5000	.5000	.5000	.5000	.5000	.5000	.5000	.5000
Q16	.5000	.5000	.5000	.5000	.5000	.5000	.5000	.5000	.5000
Q18	.1520	.1520	.1520	.1520	.1520	.1520	.1520	.1520	.1520
Q19	.0114	.0114	.0114	.0114	.0114	.0114	.0114	.0114	.0114
Q20	.0400	.0400	.0400	.0400	.0400	.0400	.0400	.0400	.0400
Q22	.5000	.5000	.5000	.5000	.5000	.5000	.5000	.5000	.5000
Q24	.1500	.1500	.1500	.1500	.1500	.1500	.1500	.1500	.1500
Q32	.1610	.1610	.1610	.1610	.1610	.1610	.1610	.1610	.1610
Q35	.0154	.0154	.0154	.0154	.0154	.0154	.0154	.0154	.0154
Q45	.0154	.0154	.0154	.0154	.0154	.0154	.0154	.0154	.0154
Q49	1.0000	1.0000	1.0000	1.0000	1.0000	1.0000	1.0000	1.0000	1.0000
Q50	1.0000	1.0000	1.0000	1.0000	1.0000	1.0000	1.0000	1.0000	.2500
Q51	.2350*	.2350*	.2350*	.3816	.3816	.3816	.1473	.1473	.5892
Q52	1.0000	1.0000	1.0000	1.0000	1.0000	1.0000	1.0000	1.0000	.5000
Q53	.9180	.9180	.9180	.7400	.7400	.7400	.4600	.4600	.4600
Q54	.6510	.6510	.6510	.5252	.5252	.5252	.3260	.3260	.3260
Q58	.9180	.9180	.9180	.3700	.3700	.3700	.2300	.2300	.2300
Q59	.6510	.6510	.6510	.2626	.2626	.2626	.1630	.1630	.1630
Q60	.3444*	.3444*	.3444*	.5613	.5613	.5613	.2164	.2164	.4328
Q61	.2350*	.2350*	.2350*	.3816	.3816	.3816	.1473	.1473	.2946
Q62	.0486	.1100	.2010	.0486	.1100	.2010	.0486	.1100	.2010
Q63	.9180	.9180	.9180	.7400	.7400	.7400	.4600	.4600	.4600
Q64	.6510	.6510	.6510	.5252	.5252	.5252	.3260	.3260	.3260
Q67	1.0000	1.0000	1.0000	1.0000	1.0000	1.0000	1.0000	1.0000	.5000
Q68	.9180	.9180	.9180	.3700	.3700	.3700	.2300	.2300	.2300

*Replug for gain of 10 with respect to Run B-B.

APPENDIX - D POTENTIOMETER SETTINGS

	Settings (Nominal)								
POT. NO.	RUN A-A	RUN A-B	RUN A-C	RUN B-A	RUN B-B	RUN B-C	RUN C-A	RUN C-B	RUN C-C
Q69	.6510	.6510	.6510	.2626	.2626	.2626	.1630	.1630	.1630
Q70	.5145	.2280	.1241	.5145	.2280	.1241	.5145	.2280	.1241
Q71	.3444*	.3444*	.3444*	.5613	.5613	.5613	.2164	.2164	.2164
Q74	.1946	.4410	.8050	.1946	.4410	.8050	.1946	.4410	.8050
Q79	.1170*	.1540	.0338	.2920*	.3840	.0805	.4570*	.6140	.1352
Q80	.4000	.4000	.4000	.4000	.4000	.4000	.4000	.4000	.4000
Q81	.2060*	.9100	.4960	.2060*	.9100	.4960	.2060*	.9100	.4960
Q82	.1946	.4410	.8050	.1946	.4410	.8050	.1946	.4410	.8050
Q83	.4000	.4000	.4000	.4000	.4000	.4000	.4000	.4000	.4000
Q84	.2500	.2500	.2500	.2500	.2500	.2500	.2500	.2500	.2500
Q89	.1170*	.1540	.0338	.2920*	.3840	.0805	.4670*	.6140	.1352
Q90	.2710	.2710	.2710	.2710	.2710	.2710	.2710	.2710	.2710
Q92	.2710	.2710	.2710	.2710	.2710	.2710	.2710	.2710	.2710

*Replug for gain of 10 with respect to run B-B.

APPENDIX - E

SYSTEM DESIGN PARAMETERS

APPENDIX - E SYSTEM DESIGN PARAMETERS

Accumulators

	P_1	V_1
High Pressure Accumulator	2000 psi	100 in.^{3}
Low Pressure Accumulator	100 psi	100 in.^{3}

Alternator Valve

$A_o = 0.050$ in.^{2}

Frequency - 4, 6, 8, 10, and 12 cps

Fluid Properties

MIL-H-5606A (Reference 11)
(for 150° F and 3000 psi)

$\beta = 230{,}000$ psi

$\nu = 0.018$ in.^{2}/sec. = 11.62 centistokes

$$\rho = 0.000079 \frac{\text{lb.-sec.}^2}{\text{in.}^4}$$

Lumped Parameter Transmission Line

MIL-T-6845 Stainless Steel Tubing

Line Length: 10 feet

Line Size: O.D. 0.750 inch
I.D. 0.666 inch

Distributed Parameter Transmission Line

MIL-T-6845 Stainless Steel Tubing

Line Length: 50, 125, and 200 feet

Line Size : O.D. 0.500, 0.750, and 1.000 inch
I.D. 0.444, 0.666, and 0.902 inch

Transformer

$A_{T1}, A_{T2} = 16.44 \text{ in}^2$

$K_8 = 50$ lb./min. (Spring Constant)

$M = 0.423 \dfrac{\text{lb-sec}^2}{\text{ft}}$

$V_8, V_9 = 24.65 \text{ in}^3$

$b = 1.0 \dfrac{\text{lb-sec}}{\text{in}}$ (Damping Coefficient)

$w = 13.62$ pounds

Typical Values for Run No. B-B-8

$$C = \sqrt{\frac{\rho_e}{\rho}} = 4{,}350 \text{ ft/sec}$$

$C_D = 0.80$ (Assumed)

$C_1 = 5$ (Assumed)

$C_2 = 5$ (Assumed)

$E = 27{,}600{,}000$ psi

$$K_1 = C_D \sqrt{\frac{2}{\rho}} = 33.1 \quad \frac{\text{gal}}{\sqrt{\text{lb-in-min}}}$$

$$K_2 = \frac{8 f_p \ell_1 \rho}{\pi^2 D_1^5} = 0.0384 \quad \frac{\text{in-min}^2}{\text{in}^2\text{-gal}^2}$$

$$K_3 = \frac{P_i^{\frac{1}{k}} V_i}{k} = 16{,}300 \ (\text{psi})^{.714} - \text{in}^3$$

$$K_4 = \frac{k+1}{k} = 1.714$$

$$K_5 = K_6 = \frac{8 f_p \ell_2 \rho}{\pi^2 D_2^5} = 0.240 \ \frac{\text{lb-min}^2}{\text{in}^2\text{-gal}^2}$$

$$K_7 = \frac{P_1^{\frac{1}{k}} V_1}{k} = 1{,}920 \ (\text{psi})^{.714} - \text{in}^3$$

$$Q_L = 6.5 \text{ gpm} \qquad (\text{Assumed})$$

$$P_3 = 4000 \text{ psi} \qquad (\text{Assumed})$$

$$P_4 = 50 \text{ psi} \qquad (\text{Assumed})$$

$$R_N = \frac{\mathrm{V}D}{\nu} = 5300$$

$$R_R = 1.847 \ \frac{\text{lb-min}}{\text{in}^2\text{-gal}}$$

$$T_e = \ell_2 \sqrt{\frac{\rho}{\beta_e}} = 0.02875 \text{ Seconds}$$

$$Z_S = \sqrt{\frac{\rho \beta_e}{A_2^2}} = 11.82 \ \frac{\text{lb-sec}}{\text{in}^5}$$

$$\beta_e = 215{,}000 \text{ psi*}$$

$$f = \frac{0.316}{R_N^{0.25}} = 0.0370$$

$$f_p = 1.2f = 0.0444$$

$$k = 1.4 \qquad (\text{Assumed})$$

$$* \ \frac{1}{\beta_e} = \frac{1}{\beta} + \frac{1}{E\left(\frac{D_o}{D} - 1\right)}$$

APPENDIX - F

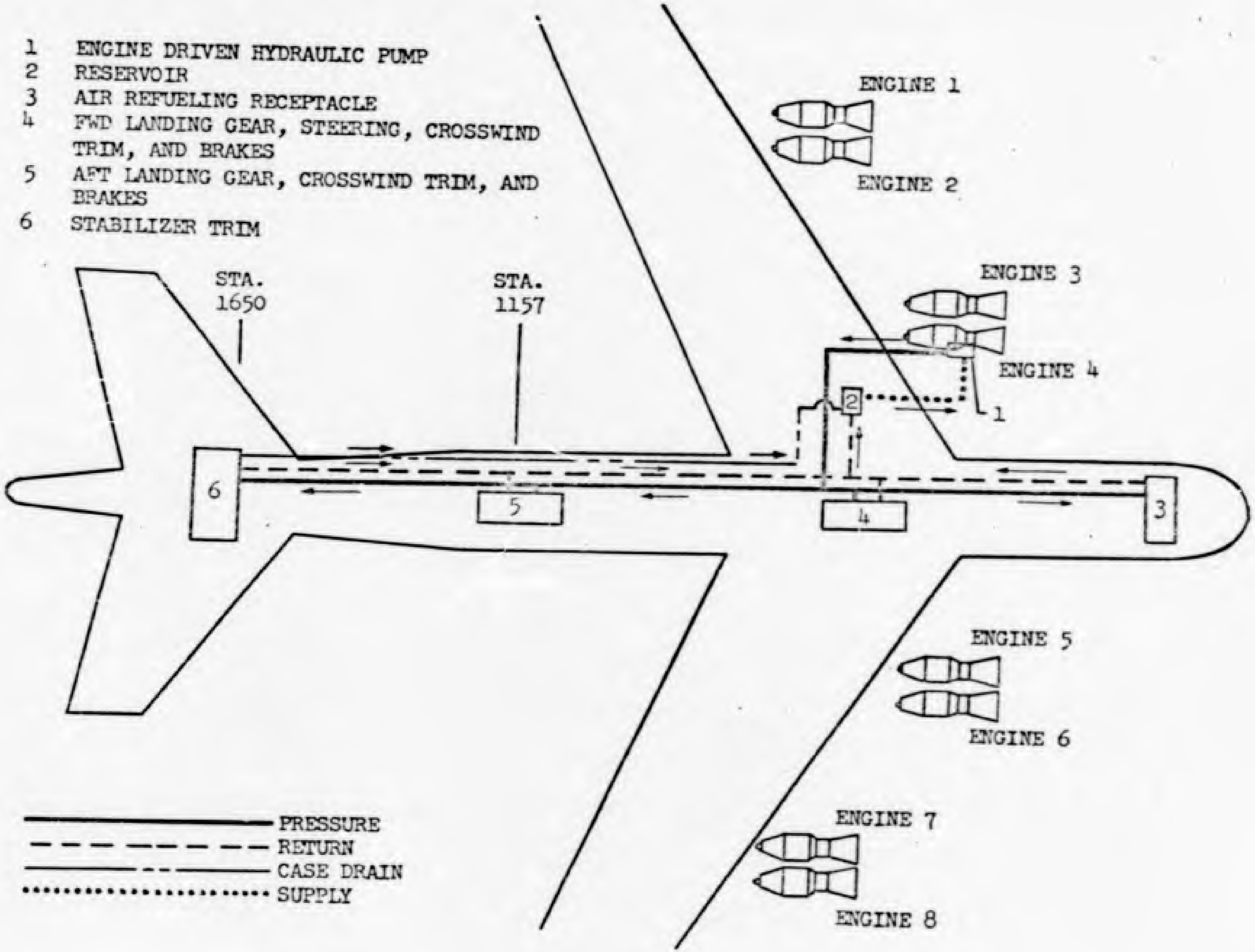
COMPARATIVE WEIGHT ANALYSIS

APPENDIX - F COMPARATIVE WEIGHT ANALYSIS

A comparative weight analysis between a pulsating hydraulic system and a continuous flow hydraulic system for the B-52 stabilizer trim hydraulic system is presented in this appendix. An alternator valve was installed at Station 1157 of the B-52 left body hydraulic system (Figure 24) which converted the continuous flow to pulsating flow. This location was selected since the flow divided at this point to the brakes, crosswind trim, and landing gear systems, and to the stabilizer trim system. The pulsating flow energy was transmitted in a single-line (Figure 25) to a rectifier valve which converted the pulsating flow to continuous flow for the stabilizer trim hydraulic system.

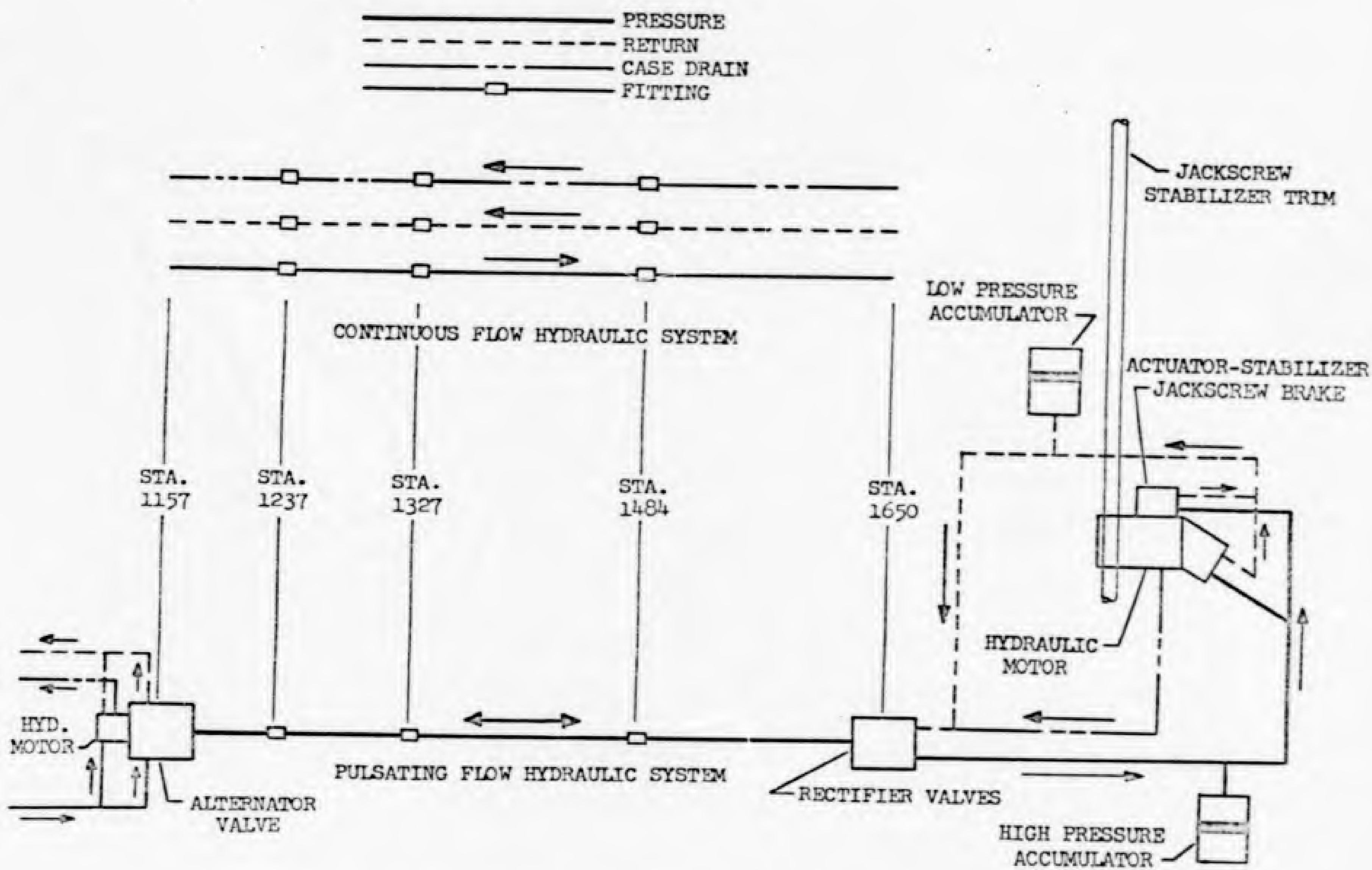
The weight analysis compares the two types of hydraulic systems from and including the alternator valve to and including the hydraulic motor for the stabilizer trim. The continuous flow hydraulic system contains three lines (pressure, return, and case drain) between Stations 1157 and 1650 while the pulsating hydraulic system contains one line. The plumbing required aft of Station 1650 was assumed to be the same for both systems. A schematic showing the relative location of the components of the two systems is shown in Figure 25.

The results of the comparative weight analysis indicated that the pulsating hydraulic system would weigh approximately 17 pounds more than the existing continuous flow system. A summary of the weights of the components in the two systems is presented in Table II. The results could be reversed if the alternator valve could have been moved forward, thus increasing the pulsating flow line length.



B-52 BODY HYDRAULIC SYSTEM (LEFT)

FIGURE 24



COMPARATIVE HYDRAULIC SYSTEM SCHEMATIC (STABILIZER TRIM)

FIGURE 25

TABLE II

B-52 STABILIZER TRIM HYDRAULIC SYSTEM WEIGHTS
(Continuous Flow vs. Pulsating Flow)

ITEM	WEIGHT - LB.	
	Continuous Flow	Pulsating Flow
Lines	23.80	19.85
Fittings	2.07	1.66
Fluid	13.48	6.12
Alternator Valve	——	10.00
Alternator Drive Motor	——	3.00
Check Valves (2)	——	1.00
Accumulators (2)	——	13.00
Hydraulic Motor	——	1.58*
TOTAL	39.35	56.21

*Representative increase in motor weight due to increase in case pressure requirements

Detailed Analysis:

A. Continuous Flow System

Assumed the line length from the alternator valve to the rectifier valve was 500 inches.

1. Pressure Line

Outside Diameter 5/8 inch
Wall Thickness .049 inch
Material - Cold drawn corrosion-resistant steel-AISI 304

$$\text{Weight} = (500 \text{ inches})(0.0252 \frac{\text{lb.}}{\text{inch}}) = 12.60 \text{ lb.}$$

2. Return Line

Outside Diameter 1 inch
Wall Thickness .065 inch
Material - Aluminum alloy-5052-C

$$\text{Weight} = (500 \text{ inches})(0.0187 \frac{\text{lb.}}{\text{inch}}) = 9.35 \text{ lb.}$$

3. Case Drain Line

Outside Diameter 3/8 inch
Wall Thickness .035 inch
Material - Aluminum alloy-5052-C

$$\text{Weight} = (500 \text{ inches})(0.0037 \frac{\text{lb.}}{\text{inch}}) = 1.85 \text{ lb.}$$

Total Line Weight = 23.80 lb.

4. Fittings

No.	Fitting	Wt./Per	Total
3	AN832-10S	0.304 lb.	0.912 lb.
3	AN924-10S	0.059 lb.	0.177 lb.
3	AN832-16D	0.236 lb.	0.708 lb.
3	AN924-16D	0.040 lb.	0.120 lb.
3	AN832- 6D	0.041 lb.	0.123 lb.
3	AN924- 6D	0.009 lb.	0.027 lb.
	Total Fittings Weight		2.067 lb.

5. Fluid

(Mil-H-5606, 52.6 lb/ft.^{3} @ 150°F)

Pressure line

$(0.0631\ ft.^3)(52.6\ lb/ft.^3) = 3.32\ lb.$

Return line

$(0.1718\ ft.^3)(52.6\ lb/ft.^3) = 9.05\ lb.$

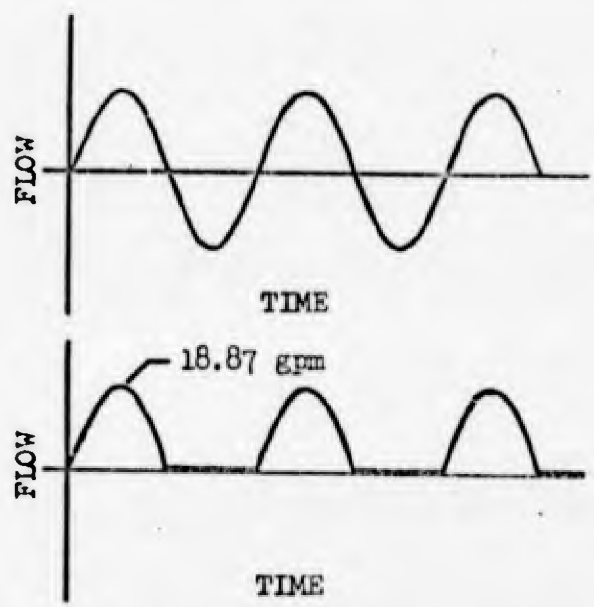
Case drain line

$(0.0211\ ft.^3)(52.6\ lb/ft.^3) = 1.11\ lb.$

Total Fluid Weight = 13.48 lb.

B. Pulsating Flow System

For a single-line pulsating system to produce a rectified flow with the power of a continuous flow of 6 gpm at 3000 psi, the flows and pressures are as follows:



Assumed the flow out of the alternator valve was sinusoidal. Rectification of single-line pulsating flow will produce an interrupted half-wave.

To produce a rectified flow of 6 gpm,

$$\text{Peak Flow} = \frac{6}{0.318} = 18.87\ \text{gpm}$$

Average Pulse Flow = (0.637)(18.87) = 12.0 gpm
(Average of sine wave)

Determining the pulsating flow line size for an average flow rate of 12.0 gpm and by limiting the fluid velocity to 15 ft/sec.

$$A = \frac{Q}{V} = \frac{(12)(231)}{(60)(15)(12)} = 0.257 \text{ in.}^2$$

$$D = 1.128 \sqrt{0.257} = (1.128)(0.507) = 0.571 \text{ in.}$$

To produce an average pressure during each pulse of 3000 psi,

$$\text{Peak Pressure} = \frac{3000}{0.637} = 4710 \text{ psi}$$

Assuming,

Outside Diameter 3/4 inch
Material-AISI 304 Stainless steel tubing (cold drawn)
(Ultimate tensile strength 105,000 psi)
Safety Factor 4

Required wall thickness from Barlow's formula (Reference 12) for thin-wall tubing,

$$T = \frac{P\,D}{2\,S} \tag{1}$$

where,

T = wall thickness, inches
P = internal pressure, psi
D = diameter, inches
S = maximum allowable stress, psi

$$T = \frac{(4710)(0.75)(4)}{(2)(105{,}000)} = 0.0674 \text{ in.}$$

the nearest standard wall thickness for a 3/4 inch O.D. tube is 0.065 inch.

1. Pulsating Flow Line

Outside Diameter 3/4 inch
Wall Thickness .065 inch
Material-Cold drawn corrosion-resistant steel-AISI 304

Weight = (500 inches)(0.0397 lb/inch) = 19.85 lb.

2. Fittings

No.	Fitting	Wt./Per	Total
3	AN832-12S	0.462 lb.	1.386 lb.
3	AN924-12S	0.093 lb.	0.279 lb.
	Total Fittings Weight		1.665 lb.

3. Fluid
(Mil-H-5606, 52.6 lb/ft.^{3} @ 150°F)

Pulsating flow line
$(0.0874 \text{ ft.}^3)(52.6 \text{ lb/ft.}^3) = 4.60$ lb.

Accumulator (assumed 50 in.^{3} for both)
$(0.0290 \text{ ft.}^3)(52.6 \text{ lb/ft.}^3) = 1.52$ lb.

Total Fluid Weight= 6.12 lb.

4. Alternator Valve

Assumed weight 10.0 lb.

5. Alternator Drive Motor

Assumed weight 3.0 lb.

6. Check Valves (2)

Assumed weight 1.0 lb.

7. Accumulators (2)

Assumed 50.0 in.^{3} accumulators, weight 13.0 lb.

8. Stabilizer Trim Hydraulic Motor

The hydraulic motor will have to be re-designed for the pulsating hydraulic system to withstand higher back pressures (case drain is plumbed into return line). Assume new motor will be 10 percent heavier than existing motor.

Hydraulic motor (BAC Spec 10-1173-2)

$(15.75 \text{ lb.})(.10) = 1.575$ lb.

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13. ABSTRACT

This final report includes the results of a research program (EWA 6801) for the investigation and understanding of pulsating hydraulics. The research investigation consisted of three phases:

a. An understanding of the theory and the present state-of-the-art of pulsating hydraulic systems.

b. Development of the basic techniques for analysis of pulsating hydraulic systems and simulation of a single-line pulsating hydraulic system on the analog computer.

c. Application feasibility of pulsating hydraulics to the B-52 hydraulic systems.

A literature review and letter survey was accomplished to obtain an understanding of the theory and the present state-of-the-art of pulsating hydraulic systems. The results of the literature review and letter survey revealed that the literature available on pulsating hydraulics was extremely limited and little research has been done in this area. The basic analytical techniques were developed and used in simulating a single-line pulsating hydraulic system on the analog computer. During the research program, 47 different system configurations were evaluated. These configurations were evaluated for the change in system efficiency with respect to changes in the basic parameters of line lengths, line sizes, and pulsation frequencies. The pulsating hydraulic systems efficiencies were lower than

(See attachment)

13. ABSTRACT for D3-6576 (continued)

those of continuous flow hydraulic systems. The B-52 application study concluded that the B-52, as designed, is not readily adaptable to pulsating hydraulic application. The present systems are functioning adequately and no real improvement in reliability or weight can be forecast.

As a result of the research investigation, it was concluded that for certain applications, pulsating hydraulic system offers a definite advantage over continuous flow hydraulic systems. It is in these areas of application that further studies are recommended to define the efficient use of pulsating hydraulics.