Preliminary Design of a Hydraulic Vibration Machine with Variable Amplitude and Frequency, Using Multistage Amplification and Feedback Control

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PRELIMINARY DESIGN OF A HYDRAULIC VIBRATION MACHINE
WITH VARIABLE AMPLITUDE AND FREQUENCY,
USING MULTISTAGE AMPLIFICATION
AND FEEDBACK CONTROL

A Thesis
Presented to the
Department of Mechanical Engineering
Brigham Young University

In Partial Fulfillment
of the Requirements for the Degree
Master of Science

by
Melvin Joseph Merrell
21 December 1965
This thesis by Melvin Joseph Merrell is accepted in its present form by the Department of Mechanical Engineering of Brigham Young University as satisfying the thesis requirements for the degree of Master of Science.

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Typed by: Norma Scarlett
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NOMENCLATURE

A  Amplitude

A_L  Constant gain of lead circuit

A_{max}  Maximum orifice area, in²

A_o  Orifice area

A_p  Effective area, in²

A_v  Area of power valve pressure surface, in²

b  Damping for power valve, \( \frac{1b \, sec}{in} \)

C  Capacitor

C  Damping for ram, \( \frac{1b \, sec}{in} \)

C_1  Amplification for displacement monitor

C_2  Amplification for velocity monitor

C_3  Amplification for acceleration monitor

C_d  Coefficient of discharge

D  Derivative with respect to time

E  \( 2C_d W \cos \theta \)

E_{in}  Voltage in, volts

E_o  Voltage out, volts

F  Force, lb
\( F_c \) Constant force, lb

\( F_r \) Ram force, lb

\( I \) Current, milliamps

\( J \) \( \sqrt{-1} \)

\( K \) Spring constant, lb/in

\( KG \) Forward loop transfer function

\( KGH \) Open loop transfer function

\( K_v \) Spring constant for power valve, lb/in

\( L \) Effective damping length, in

\( M \) Mass of load, \( \frac{lb \, sec^2}{in} \)

\( M_v \) Mass of power spool, \( \frac{lb \, sec^2}{in} \)

\( o \) As a subscript is point of linearization

\( P \) Pressure difference, psi

\( P_{mv} \) Pressure across Moog valve, psi

\( P_s \) Supply pressure, psi

\( P_v \) Pressure across power valve, psi

\( P_{va} \) Pressure across power valve pressure surface, psi

\( Q \) Flow, \( \frac{in^3}{sec} \)

\( q \) Flow, \( \frac{in^3}{sec} \)

\( Q_m \) Flow from Moog valve, \( \frac{in^3}{sec} \)

\( Q_{pv} \) Flow from power valve, \( \frac{in^3}{sec} \)

\( q_r \) Flow to ram, \( \frac{in^3}{sec} \)
$R_1$ Resistor

$R_2$ Resistor

t Time

$V_t$ Total entrapped volume of fluid in fluid spring

W Circumference of valve, in

X Displacement, in

Y Displacement of power spool, in

Z Displacement of Moog valve spool, in

$\beta$ Bulk modulus of fluid spring oil, \( \frac{\text{lb}}{\text{in}^2} \)

$\zeta$ Damping ratio

$\theta$ Angle of flow leaving orifice

$\rho$ Oil density, \( \frac{\text{lb}}{\text{sec}^2} \)

$\tau$ Inverse frequency

$\omega$ Frequency in \( \frac{\text{radians}}{\text{sec}} \)

$\omega_n$ Natural frequency in \( \frac{\text{radians}}{\text{sec}} \)
CHAPTER I
INTRODUCTION

Essentially there are three types of vibration machines: mechanical, electrodynamic, and electrohydraulic (hydraulic). Each of these machines has characteristics which make it suitable for particular areas of use.

Mechanical. The mechanical vibration machines include direct-drive machines, rotating unbalance device machines, and others. The prominent features of these machines are large displacements and low operating frequencies. Because the forces on machine members increase as frequency increases, the members deform and the amplitude varies accordingly. Each type of vibration machine is affected similarly, but varies with type of construction. Mechanical vibration machines are rarely used as constant displacement machines above 25 cps.

Electrodynamic. This type of machine is noted for its ability to vibrate relatively small masses at high frequencies. Vibration is caused by a force produced by the interaction between a current flow in a driver coil and the magnetic field which cuts the coil. The force to weight-of-machine ratio is quite small compared with the mechanical or hydraulic machines. Because the vibration depends on the frequency of the current
flow, a wide frequency range of operation can be obtained, but to obtain a 25,000 pounds force vector, a machine the height of a man would be required.

**Hydraulic.** The hydraulic vibration machine is a device which derives its name from the method of transmitting power. Transmission is by high pressure fluid flow from a pump to an actuator cylinder. Valves direct flow alternately to each side of the double acting piston. Initial flow is usually controlled by an electrohydraulic servovalve. The prominent features of the hydraulic machine are that large forces and large displacements can be achieved using small components. Currently hydraulic machines are not capable of operating above 300 to 350 cps maintaining any significant displacement. This is referred to later as state of the art frequency.

General information on the subject of vibration machines can be found in the following sources: SHOCK AND VIBRATION HANDBOOK, Vol.2, by Harris and Crede(1), CONTROL SYSTEM COMPONENTS by Gibson and Tuteur(2), and FLUID POWER CONTROL by Blackburn, Reethof and Shearer(3).

The hydraulic vibration machine under preliminary design in this thesis was chosen to fill a structural testing need in the Mechanical Engineering Department and to provide laboratory demonstration capability of a hydraulic control system. It was also planned to design into it a characteristic that the hydraulic machine does not normally have--that

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1Numbers in parentheses refer to the List of References at the end of the thesis.
is higher frequencies of operation. To facilitate this multiple stages of amplification were utilized and a variable spring was designed so that resonance could be maintained throughout much of the operating range of frequency.

It should be said here that this thesis does not include the total design of the vibration machine. A complete design would require investigation of many details that cannot be covered in this thesis. These details will be worked out when the actual machine is built. Some components of the vibration machine were partially designed by two fifth-year mechanical engineering students, Calvin Merrell and Clair Shields.

This thesis essentially accomplishes three things: (1) power matching of hydraulic components capable of performing to the specifications selected. This was done by utilizing a digital computer to study the large number of variables involved; (2) an extension of the state of the art frequency response at the power level chosen using a fluid spring to vary the natural frequency of the load and a variable fluid volume cylinder, both of which are physically realizable; (3) an analysis of the stability of the total dynamic system using root locus technique.
CHAPTER II

SYNOPSIS

Before the details of the preliminary design are given, an understanding of the system would be of value.

The Problem

The problem is essentially one of amplification—that is, transforming an electrical signal input into a relatively large displacement output. This is done in three stages.

The first stage (Fig. 1) is a servovalve consisting of three main parts: a torque motor, a flapper valve (flapper...
and nozzle), and a spool valve. The servovalve first converts an electrical signal to a displacement of the flapper by magnetic forces in the torque motor. The displacement of the flapper controls the flow in the two nozzles which control flow alternately to the ends of the spool. This flow causes the spool to move, and the spool movement allows high pressure flow through one control port while allowing return flow from the load which is the second stage.

The second stage is another spool valve which operates similarly to the one in the first stage. The small flow through the nozzles of the flapper valve on the first stage regulates the movement of the small spool valve. This movement in turn regulates a much larger flow which now can move the larger second stage spool.

The last, or third stage, is a double-acting ram or piston. This has a relatively large area which, when acted upon by a high pressure flow from the second stage, or power valve, produces a large force. The load or object to be vibrated is attached directly to the piston rod. Monitoring the displacement of the ram can be used to control the system.

The variable spring is also directly attached to the ram to facilitate change in natural frequency of the load. This variable spring (Fig. 2)\(^2\) is essentially another double-acting piston in a cylinder, but has no inlet and outlet as does the ram. This cylinder is filled with a fluid that is

---

\(^2\)These are simplified drawings. Detail drawings are available in the Brigham Young University Mechanical Engineering Department.
more compressible than most fluids. By regulating the volume of this fluid, the spring constant can be varied over a wide range.

Fig. 2.—Variable Spring System

The problem is not just one of a single fixed load at a particular frequency and vibration at a fixed amplitude. The vibration machine must be capable of many different loads at different frequencies and amplitudes.

Design Criteria

Specifications. Before the project was started, a tentative set of specifications were agreed upon by those
concerned in the Mechanical Engineering Department. These are as follows:

I. Force and pressure
   A. A force vector of 25,000 pounds maximum
   B. Maximum operating pressure of 3000 psi

II. Amplitude
   A. Two-inch maximum at frequencies up to 10 cps
   B. One-half-inch maximum at frequencies higher than 10 cps

III. Loading
    1000 pounds maximum

IV. Electrical input by function generator

V. Feedback control

VI. Response frequencies obtainable beyond state of the art

Method of Approach

In designing and matching the hydraulic components, a chart called a load locus was used. This chart graphically compared the load power with that of the drive power. The load locus is a graph of force versus velocity or pressure versus flow. If the vibration machine were designed for only one frequency and amplitude, there would only be one curve for the load and one curve for the drive; but since this was not the case, the curve representing the maximum power for the driving component would need to completely enclose all the pressure versus flow curves for the load. This is shown in Figure 3. Using this method, coupled with other pertinent
Fig. 3.—Load Locus

information, matching of all the hydraulic components was made. Because of the many variables involved, much of the work was done by a digital computer (Appendix A, p. 41).

Analysis

1. Oscillator
2. Subtractor
3. Amplifier
4. Moog Servovalve
5. Power Spool Valve
6. Ram
7. Feedback

Fig. 4.—Tentative System

Because of the complexity of the nonlinear system, the equations were linearized. This made it possible to work with the Laplace transformed equations.
With linear equations it was then possible to find the characteristic equation of the input-output relationship and plot a root locus diagram of this equation.

This not only indicated the stability of the system, but allowed comparison of compensation methods.

It was found that a number of electrical lead circuits (differentiating circuits) were necessary ahead of the hydraulic system to accomplish stability.

Linearizing the equations, which represent a very non-linear system, was only justified because the system was least stable at maximum gain. It was at this point of maximum gain that the equations were linearized.
CHAPTER III

DESIGN OF COMPONENTS

Ram Cylinder. Specifications for a ram cylinder with movable heads to reduce fluid volume and which would operate at 3000 psi and produce 25,000 pounds force were given to Calvin Merrell. He calculated strength requirements for the cylinder and components and produced detailed drawings of these components. These drawings are available in the Brigham Young University Mechanical Engineering Department.

The piston rod was designed to carry a maximum axial and bending load of 25,000 pounds and 25,000 inch-pounds respectively.

The ram cylinder heads were designed so they could be adjusted in and out. A slotted plug is attached to the heads and moves with them. The moving heads and slotted plugs were designed to reduce the volume of fluid on both sides of the piston. Because of the compressibility of the fluid, the stiffness, and hence the frequency, is a function of the volume. A simplified diagram of the slotted plug, piston and cylinder are shown in Figure 5.

Power Valve. Almost all the requirements for the power valve were specified. The pressure is 3000 psi, and flow is determined by the capability of the pump since most of the flow from the pump goes to the power valve. It was
necessary then to determine from pump catalogs and other literature what realistic amount of flow could be generated, or what is the largest pump feasible. A pump which would produce 65 to 75 gallons per minute and would require up to 130 horsepower as input was available. This Hp was a feasible requirement for a diesel engine to be acquired by the B.Y.U. The velocity of the ram is set by the maximum flow from the power valve. The ram velocity is found by taking the flow to the ram divided by the piston area. A maximum velocity of 30 in/sec, which corresponds to 72.3 gal/min (264 in$^3$/sec), was assumed for the power valve.

Now that the maximum flow (264 in$^3$/sec) and maximum pressure drop (operating pressure) required from the power valve are known, the question then is, what mathematical relationship is obeyed between these two extremes (what are the other points in the flow-pressure curve)? The flow-pressure relationships can be derived analytically. The analysis involves the following assumptions(4):
1. Ideal fluid
2. Ideal fluid source
3. Ideal geometry of valve
4. Steady state conditions

The assumption of an ideal nonviscous and incompressible fluid is nearly correct under most conditions. At the peak flow rates and with relatively viscous fluids, the effective supply pressure at the valve intake may fall off slightly because of pressure drop in the supply lines; but this should not be more than 5 to 10 per cent in well-designed systems. The incompressibility assumption is also justified as far as phenomena inside the valve are concerned. At normal operating pressures, the finite compressibility of real fluids has only a negligible effect upon the flow through the orifices.

An "ideal" constant-pressure fluid source is one in which the source supplies fluid to the intake of the valve at a constant pressure independent of the flow rate. A constant-flow source supplies fluid at a constant flow rate independent of the change in pressure. It is possible to build sources for which the assumption of ideality is fairly accurate, even for rapid changes in loads; but even though the source may not be absolutely constant, the effect on the system is not excessive in most cases.

"Ideal geometry" means that the edges of the orifices are perfectly sharp and the clearances are zero so that the geometry of the orifice is not a function of valve-stem position. This assumption is usually acceptable except for displacements below one or two micro-inches.
The assumption of steady-state condition is valid for the valve alone.

If the system is such that these four assumptions are valid, the flow-pressure relationship follows the orifice equation

\[ q = C_d A_o \sqrt{\frac{2P}{\rho}} \]  \[1\]

where \( q \) = Flow

\( C_d \) = Coefficient of discharge \( P \) = Pressure difference

\( A_o \) = Orifice area \( \rho \) = Oil density.

\( C_d \) is dependent somewhat upon Reynold's number, but it has been found that in well-constructed valves, a value between .6 and .65 can be used.(5).

The orifice equation is that of a parabola, and the flow-pressure curve between maximums is now defined. The curve then represents the limiting condition for the ram; (i.e., maximum available power).

The previous coverage of flow-pressure curves makes it possible to examine the load locus. A load locus curve is also a flow-versus-pressure curve since flow is equal to the velocity multiplied by the area and pressure is the force divided by the area. If the force of a load of some system is plotted versus the velocity of that system, this is a load locus. The loci vary according to the type of loading.

The load of the ram, which is a load on the power valve, was analyzed as a spring-mass-dashpot system, and the dynamics can be described mathematically in the following way:

\[ F = MD^2X + CDX + KX. \]  \[2\]
The spring with spring constant $K$ is the variable spring mentioned; the mass $M$ is the mass of the load; and leakage and flow resistance act as a dashpot with a coefficient of damping $C$. $D$ is the derivative with respect to time. If the equation is rearranged and a solution of $X = A \sin \omega t$ is assumed, the solution is

$$F = MA\omega^2 \left( \frac{\omega_n^2}{\omega^2} - 1 \right) \sin \omega t + 2\xi \frac{\omega_n}{\omega} \cos \omega t \right). \quad [3]$$

- $F = \text{Force}$
- $M = \text{Mass}$
- $A = \text{Amplitude}$
- $\omega_n = \text{Natural frequency}$
- $\omega = \text{Operating frequency}$
- $\xi = \text{Damping ratio}$

Fig. 6.--
Amplitude Versus Frequency for Ram

Frequency in Cycles Per Second

Amplitude in Inches
If the load locus is plotted for the equation, it turns out to be an ellipse, and its relative shape depends on the variables amplitude, frequency, and natural frequency. A value for damping was obtained for the ram by comparing values of other hydraulic rams and scaling to fit the particular ram (see Assumed Values, p. 53). A computer program (Appendix A, p. 47) was written which varied each of these variables and eliminated all load locus curves except those under the maximum output curve of the power valve. Each locus can be compared to the maximum output curve of the power valve since we know the equation which describes that output (Equation 1). The computer results were the maximum amplitude for each operating frequency. The calculations were made assuming the natural frequency was equal to the operating frequency (Fig. 6).

Now that the flow-pressure requirements for the power valve are known, size, shape and other details for the valve can be determined. A general four-way valve (Fig. 7) along with its equivalent electrical circuit (Fig. 8) is illustrated to facilitate the following derivation. The circuit is that of a loaded Wheatstone bridge with square law arms. These are square zigzag instead of saw tooth to show that hydraulic conductance is not ohmic, but is parabolic. The electrical analogue of an orifice is not a resistor, but a nonlinear varistor. The control valve does not correspond to an ohmic rheostat, but corresponds to a triode.
Fig. 7.-- Spool Valve and Load Schematic

Fig. 8.--Equivalent Circuit

The application of Kirchhoff's first law to the system results in the following equations:
\[ P_1 + P_2 = P_s \quad [4] \quad P_3 + P_4 = P_s \quad [5] \]
\[ P_1 - P_4 = P_m \quad [6] \quad P_3 - P_2 = P_m \quad [7] \]

The \( P \)'s represent pressure drops across the circuit elements.

The subscripts \( s \) and \( m \) refer to the supply and motor (or hydraulic load) respectively. Kirchhoff's second law requires that the following equations hold:

\[ q_1 + q_4 = q_s \quad [8] \quad q_2 + q_3 = q_s \quad [9] \]
\[ q_2 - q_1 = q_m \quad [10] \quad q_4 - q_3 = q_m \quad [11] \]

The \( q \)'s represent rates of flow in the various branches.

The pressure is a scaler quantity in the first law, and Kirchhoff's second law is the electrical analogue of the law of conservation of mass in the hydraulic case.

Finally, using the orifice law yields four more equations:

\[ q_1 = g_1\sqrt{P_1} \quad [12] \quad q_2 = g_2\sqrt{P_2} \quad [13] \]
\[ q_3 = g_3\sqrt{P_3} \quad [14] \quad q_4 = g_4\sqrt{P_4} \quad [15] \]

The \( g \)'s are hydraulic conductances of the orifices \( A_0C_d\sqrt{2/\rho} \).

By assuming a constant pressure system and, hence, ignoring equations 8 and 9, the equations 4 through 15 are tractable. Assuming a zero lap valve, \( q_1 \) and \( q_3 \) are zero if the spool is to the left; and \( q_2 = q_4 = q_s \), so the subscripts on \( q \) are unnecessary. Since

\[ P_m = P_1 - P_4 \quad [16] \quad P_4 = q^2/g^2 \quad [17] \]
\[ P_m = P_1 - q^2/g^2 \quad [18] \quad P_1 = P_s - P_2 \quad [19] \]

hence, \( P_2 = q^2/g^2 \quad [20] \) and \( P_m = P_s - 2q^2/g^2 \quad [21] \)
The power into the load is the product of load pressure and
flow or
\[ H_m = qP_m = qP_s - \frac{2q^3}{g^2}. \] \[
^2
\]
Maximum power will exist when
\[ \frac{DH_m}{Dq} = 0 \quad \text{or} \quad 3P_m = 2P_s \quad P_m = \frac{2}{3} P_s \]
\[ 23 \]
or at maximum power the load pressure \( P_m \) is \( 2/3 \) that of supply
pressure.

Since \( P_m \) is zero at maximum flow, by applying equation
\[ q_{\text{max}} = q_{\text{max}} \sqrt{P_s/2} \] \[
24
\]
and with most hydraulic fluids this is approximately
\[ q_{\text{max}} = 70 A_{\text{max}} \sqrt{P_s}. \] \[
25
\]

\( A_{\text{max}} \) occurs when valve stem displacement is maximum.

From the above equation the maximum orifice area is
\( A_0 = .0689 \text{ in}^2 \). This area can be obtained by multiplying
the peripheral length of the orifice by the displacement (one
sharp edge from the mating sharp edge). It can be seen that
many such combinations could result in an area of \( .0689 \text{ in}^2 \).

To find such a combination, further criteria for determination
must be found. Factors involved are force-displacement
relationships, weight-size relationships, and choice of an
electrohydraulic servovalve that will fit the requirements
of the power valve. Finding the load locus curves for the
power valve, and at the same time choosing a servovalve with
an output that matches the power valve requirements, will
give some of these factors.
The differential equation that describes the forces of the valve stem is (6)

\[ F = M_v D^2 \dot{Y} + bD \dot{Y} + (E P_{pv} + K_v)Y + \rho L D q_r. \]  

\[ F = \text{Force} \quad P_{pv} = \text{Pressure drop across power valve} \]

\[ M_v = \text{Mass of valve spool} \quad K_v = \text{Power valve spring constant} \]

\[ b = \text{Coefficient of damping} \quad L = \text{Damping length} \]

\[ E = 2 C_d W \cos \theta \quad q_r = \text{Flow to ram} \]

\[ \theta = \text{Angle of flow through orifice} \]

As was done previously, the flow-pressure curve of the servo-valve must enclose the load locus curves of the power valve. A value for damping was assumed for the valve in similar fashion as was done for the ram.

A computer program was written which took the points calculated by the previous computer program and related them to the power valve. By assuming a first stage flow, the power valve load locus was compared to the flow-pressure curve of the assumed first stage. This program had quite a few variables and some constants that had to be changed to find the combination desired, such as frequency, amplitude, spring constants, mass and flow.

The program did several things. It determined the maximum displacement of a certain valve mass from which a size could be calculated, the mass of the valve being a function of the valve length and diameter, both of which had to be feasible. The program also determined what flow-rate capability was needed for the first stage valve. This,
too, had to be feasible from the standpoint of availability of such servovalves.

The results of the computer program are as follows: the spool at the orifice must be one inch in diameter with 360 degree port; the weight of the spool can be no more than one pound; the frequency can be as high as 300 cps before the amplitude of displacement of the spool decreases; the pressure surface or spool-end diameter must be one-half inch; and flow rate to the spool end must be at least 15.4 in$^3$/sec; spring constant of the spool valve can be 30 to 100 pounds per inch. Damping length used in the program was .2 inches. This, like the spring constant, is not critical, except that it cannot be negative for stability(7).

Ducting should be at least four times as large as the maximum area of the orifices in order to avoid saturation in the valve(8).

Upon investigating servovalves, it was found that Moog Servocontrols, Inc. could modify their 31-010A series valve so that gain was essentially zero to about 700 cps on the frequency versus decibles graph. This valve would give the flow necessary and operate at 3000 psi. (For gain diagrams see Fig. 9.)

**Fluid Spring.** Specifications for a variable spring were given to Clair Shields who considered different designs for a variable spring and concluded that a fluid spring would be most easily obtained. He then designed and produced drawings of the fluid spring.
Fig. 9.—Amplitude and Phase Response Diagram
The equation of motion for the load, $MD^2X + CDX + KX$, solved for amplitude is

$$A = \frac{F_c/K}{\sqrt{(1 - (\omega/\omega_n)^2)^2 + (25\omega_n/\omega)^2}}$$ \[27\]

Where $F_c$ is a constant force applied to the mass in the system. If $\omega_n$ is much greater than $\omega$ it makes $(25\omega_n/\omega)^2$ the only significant term under the radical sign; thus

$$A = \frac{F_c/K}{25\omega_n/\omega} = 25\omega_n/\omega = \frac{F_c}{25\omega_n^3}$$ \[28\]

Equation 28 says that amplitude is proportional to operating frequency when $\omega$ is much less than $\omega_n$, but because $\omega$ is small in comparison, amplitude is small.

If $\omega$ is much greater than $\omega_n$ then the significant factor is $(\omega/\omega_n)^2$ and

$$A \approx \frac{F_c}{M\omega^2}.$$ \[29\]

This says that amplitude is inversely proportional to the square of frequency. On the other hand, if $\omega_n = \omega$ then

$$A \approx \frac{F_c}{C\omega}.$$ \[30\]

Now amplitude is inversely proportional to the frequency to the first power only. This contrast can be seen readily in Figure 10.

Amplitude decreases with frequency, but the rate can be changed substantially if the natural frequency can be changed to correspond to the operating frequency. This then shows the need for a variable spring.
Both mechanical and fluid springs were considered for this variable spring. The mechanical type considered was a double-wedge cantilever. This system was feasible since spring constant depends on length. However, this design was bulky, and many problems were involved in attachment and positioning.

The fluid spring also presented problems; however, they could be overcome more easily. If the fluid used were incompressible, there would be no spring involved. This not being the case, it can be shown\(^9\) that the spring constant relationship is

\[ K = \frac{4A_p^2 \beta}{V_t} \]  \[31\]

where \(A_p\) is the effective area of the piston.
Thus the spring constant \( (K) \) is inversely proportional to the volume \( (V_t) \) of fluid entrapped on both sides of the spring piston. This says that if the volume is controlled, the spring constant can be controlled. Using a silicone oil which has a bulk density \( (\beta) \) of about 140,000 pounds per square inch, a size and configuration for the fluid-spring piston and cylinder was calculated. Another computer program\(^3\) was written to facilitate varying the area and also calculate volume, frequency, and spring constant for each area chosen. From this a five-square-inch effective piston area was chosen, and a maximum volume of 9.4 gallons, and an operating range from 30 cps to 450 cps were obtained. Since \( \sqrt{K/M} \) equals the natural frequency, then by controlling the volume, and thus controlling the value of \( K \), the natural frequency is also controlled.

\(^3\)Available in the Brigham Young University Mechanical Engineering Department.
CHAPTER IV

ANALYSIS OF SYSTEM

The system is shown in block diagram below.

Fig. 11.--System

The electrical circuits were not designed; however, the magnitudes of amplification and the frequencies necessary were evaluated and considered possible.

First considered in the analysis were the relationships between input and output for the Moog Servovalve. This valve was chosen because it was the best available considering flow rate and frequency response. From a frequency response curve\(^4\) it was obvious that the transfer function was not a simple second order equation with .5 to .7 damping as Moog indicated for the regular valve. To approximate the curve,
Bode plots(10) of a number of first and second order transfer functions were drawn on tracing paper and added. When the right combination of transfer functions were found, their addition compared favorably with the curve for the modified servovalve. The combination that fit best was one with two first order transfer functions with corner frequencies of 540 cps, plus a second order function with a natural frequency of 540 cps and damping equal to .2 (Fig. 9, p. 21).

The transfer function equations in Laplace transform notation are

\[ G_1(s) = \frac{Z}{I} = \frac{1}{(s \tau + 1)\left(s^2 + 1357s + 3393\right)}, \quad [32] \]

where \( 1/\tau = 540 \) cps or 3393 radians/sec.

\[ Z = \text{Displacement of the ram} \]

\[ I = \text{Current} \]

This approximate transfer function deviates from the curve more as frequency is increased above 400 cps; even so, the deviation up to 600 cps is not more than one-half decibel.

The flow relationship or orifice equation is:

\[ Q = 2C_d W \sqrt{2P_{mv}/\rho} = 2C_d W \sqrt{2(P_s - P_{va}/\rho)} \quad [33] \]

\( Q_m = \text{Flow from Moog valve} \)

\( P_{mv} = \text{Pressure drop over Moog valve orifice} \)

\( C_d = \text{Coefficient of discharge} \)

\( W = \text{Peripheral length of valve at orifice} \)

\( \rho = \text{Oil density} \)

\( P_{va} = \text{Pressure drop across power valve end or pressure surface} \)

o as a subscript is point of linearization
Linearizing these equations to make them tractable results in

\[ Q_m - Q_{mo} = K_1(Z - Z_o) + K_2(P_{va} - P_{vao}). \] \[34\]

Further reduction results in:

\[ Q_m = K_1Z + K_2P_{va} + K_3 \] \[35\]

where

\[ K_3 = -(K_1Z_o + K_2P_{vao} + Q_{mo}) \] \[36\]

\[ K_1 = \frac{\partial Q_m}{\partial Z} \] \[37\]

\[ K_2 = \frac{\partial Q_m}{\partial P_{va}}. \] \[38\]

\[ K_1 \] and \[ K_2 \] were taken from the flow-pressure-current curves supplied by the Moog servovalve company. They can also be evaluated directly from the partial derivatives of the orifice equation.

The flow from the Moog valve can be related to the velocity of the power valve since the flow from the Moog valve \( Q_m \) is equal to the area \( A_v \) of the surface it is acting upon in the power valve, multiplied by the velocity of that surface, or

\[ Q_m = A_vD_y. \] \[39\]

The following is an analysis of the power valve equations.

The force balance equation for the power valve is

\[ P_{va} A_v = M_v D^2Y + bD_y + (E_{pv} + K_v)Y + \lambda LDq_r. \] \[40\]

\( A_v \) = Area of power valve

\( Y \) = Displacement of power valve spool

\( M_v \) = Mass of spool

\( b \) = Coefficient of damping

\( P_{pv} \) = Pressure drop across power valve
\( K_v \) = Power valve spring constant
\( L \) = Damping length
\( q_r \) = Flow to ram
\( E = 2C_dW \cos \theta \)
\( \theta \) = Angle of flow from orifice

The term \( \rho L D q_r \) was neglected since it can be made very small by making the effective damping length small (11).

After transforming into \( S \) notation and combining equations 32, 35, 39, and 40,

\[
(K_1 G_1(S) I = (Y/A_v)(-K_2 M_v S^2 + (A_v^2 - K_2 b) S - K_2 T_1) - K_3) \quad [41]
\]

\[
T_1 = EP_{pv} + K_v \quad [42]
\]

Let \((1/A_v)(-K_2 M_v S^2 + (A_v^2 - K_2 b) S - K_2 T_1) = G_2(S) \). \[43\]

Rearranging results in

\[
Y = K_1 G_1(S) I/G_2(S) + K_3/G_2(S) \quad [44]
\]

Now it is known that for the ram, \( q_r = A_r D X \), \[45\]

and \( Q_{pv} = K Y + K_5 P + K_6 q_r \). \[46\]

\( Q_{pv} \) = Flow from power valve
\( q_r \) = Flow to the ram
\( A_r \) = Area of the ram

\[
K_4 = \partial q_r / \partial Y \quad [47]
\]

\[
K_5 = \partial q_r / \partial P_r \quad [48]
\]

\[
K_6 = -(K_4 Y + K_5 P + Q_{ro}) \quad [49]
\]

The force balance equation for the load is

\[
F_r = P_r A_r = M D^2 X + CDX + KX \). \quad [50]
\]
\[ P_r = \text{Pressure drop across the ram} \]
\[ X = \text{Displacement of the ram} \]
\[ F_r = \text{Ram force} \]
\[ M = \text{Mass of load} \]
\[ C = \text{Coefficient of damping} \]
\[ K = \text{Ram spring constant} \]

Combining equations 45, 46, and 47 gives
\[ Y = \frac{X}{K_4 A_r} \left( - K_5 M S^2 + (A_2^2 - K_5 C) S - K_5 K \right) - \frac{K_6}{K_4}. \]  

Combining equations 44 and 51 results in
\[ X = \frac{K_1 G_1(S)}{G_2(S) G_3(S)} I + \frac{K_3}{G_2(S) G_3(S)} - \frac{K_7}{G_3(S)} \]
\[ G_3(S) = \frac{1}{K_4 A_r} \left( - K_5 M S^2 + (A_2^2 - K_5 C) S - K_5 K \right). \]
\[ K_7 = -\frac{K_6}{K_4}. \]

Equation 52 is now an equation in terms of displacement of the ram and input current to the Moog servovalve. The equation is not a direct relationship between \( X \) and \( I \) as would be most desirable. With a little investigation, however, equation 52 can be analyzed as if it were a direct relationship.

Figure 12 shows the block diagram of the open loop transfer functions. The two extra terms (inside the dashed lines) can be analyzed as external signals into the system

![Block Diagram](image)

**Fig. 12.**— KG Plus External Inputs

and need not be considered in the stability analysis. If the system is stable, it will remain so with these external signals added (12).
Now having the linear relationship

\[ \frac{X}{I} = K_1 G_1(S) / G_2(S) G_3(S), \]

this relationship can be investigated for stability by means of root locus. If \( G_1(S), G_2(S) \) and \( G_3(S) \) are substituted into equation 55, the results are

\[
\frac{X}{I} = K_G = \frac{K_1 K_1 A_r A_v / K_5 M_5 V K_2}{(S^2 + (A_r^2 - K_5 C) S + K)(S^2 + (A_v^2 - K_2 b) S + T_1)} \times \frac{1}{(S T + 1)^2(S^2 + 1357 S + 3393)}.
\]

Substituting values for the constants and factoring gives

\[
K_G = \frac{K_1 K_1 A_r A_v / K_5 K_2 M_5 V M^2}{(S + 25^4 + 12513)(S + 25^4 - 12513)(S + 400000)} \times \frac{1}{(S + 3393)^2(S + 2.51)(S + 1354)}.
\]

It can be seen that there are very low frequency roots in the system that need to be compensated by zeros to provide high frequency response. This can be partially accomplished with the type of feedback used. If acceleration, velocity and position feedback are used, then \( C_3 S^2 + C_2 S + C_1 \) can be introduced into the equations. Acceleration could be provided by an accelerometer; velocity, by integrating the acceleration, and position, possibly by an induction potentiometer. By selecting the right ratio between the constants (amplification), two of the low frequency poles can be canceled; that is, if

\[ C_2 / C_3 = 2.51 \]

\[ C_1 / C_3 = .01 \]
then essentially
\[ C_3 s^2 + C_2 s + C_1 = C_3 (s(s + 2.51)) \]
so that
\[
KH = \frac{K_1 K_2 A_r A_t \sqrt{C_3 (s+2.51)}}{(s+254+j2513)(s+254-j2513)(s)(s+400000)} \]
\[ \times \frac{1}{(s+3393)^2 (s+2.51)(s+1354)} \]  \[ [60] \]
To eliminate the other low frequency poles, there must be more compensating circuits in the feedback loop or in front of the Moog valve, one being the equivalent of the other. The root locus plot without more compensation is shown in Figure 13.

![Root Locus Plot](image)

**Fig. 13.--Root Locus, Uncompensated**

It is possible to introduce zeros into the system to eliminate unwanted poles in two ways. The first is usually
bypassed because the network requires an expensive amplifier. The second is a simple RC circuit (see Fig. 14) which places a zero in the region of the critical frequency. This RC circuit is characterized by a pole and a zero with the pole located at a higher frequency (at least two octaves) and, therefore, does not oppose the effects of the zero. The lead circuit will need to be isolated by an amplifier in order that the loads of each circuit may be isolated.

![RC Circuit Diagram](image)

\[ R_1 = \text{Resistor} \quad E_{\text{in}} = \text{Voltage in} \]
\[ R_2 = \text{Resistor} \quad E_o = \text{Voltage out} \]
\[ C = \text{Capacitor} \]

Through trial and error it was found that by canceling the system pole at 1354 with a network zero and by canceling the pole at 3393 by two zeros, the system becomes stable at all values of gain with the damping increasing as the gain increases. Since the pole locations depend on many variables, exact cancellation would be difficult. Because of this there will be some variations in the system transfer function which was desired. This variation will add the extra poles and zeros not completely canceled to the overall system function; but because of the closeness of the zero to the pole, the
resulting small residue at the pole, and the small variation in residues at the others, make such a system still adequate (13).

The system parameters are inherently unstable. These parameters are: compressibility, which is a function of temperature; time constants and viscosity, also temperature sensitive; conduit (i.e., fluid line) resonance and others. In using a compensation method such as this, it is most likely that there will be modifications necessary as problems arise. These problems cannot be predetermined and may not be easily diagnosed and solved. Some of these will be discussed in the recommendations.

The method of critical frequency cancellation will not create the needed force to drive the loads (valves, ram, etc.) at a higher frequency. It can only make higher frequencies possible if the machine is capable of operating at those frequencies. The hydraulic system which included the Moog valve was designed for the high frequencies, which are not possible without the electrical lead network. For low frequencies and low amplitudes the hydraulic system will not be at peak power.

The final closed loop equation is  

$$KGH = \left( \frac{1}{T_2} \right)^4 \left( \frac{1}{T_3} \right)^2 \left( \frac{1}{T_5} \right)$$  

$$= \frac{1}{(S+254+J2513)(S+400000)(S+254+J2513)}$$  

\[61\]

1/ \(T_2 = 3393\) \hspace{1cm} 1/ \(T_5 = 1354\) \hspace{1cm} 1/ \(T_3 = 5000\)
At a value of damping of 0.7, which corresponds to an angle of 45 degrees on the locus (Fig. 15), there exists a value of gain of about $10^7$ assuming $A_L$ equal to 1. By increasing or decreasing $A_L$, the gain would decrease or increase accordingly. This means that by making $A_L$ large ($10^3$ to $10^7$), the damping will always be large enough (0.3 to 0.7) to effect reasonably good response. A sine wave input was envisioned as the usual input; however, other wave forms could be used.

![Diagram](image)

**Fig. 15.** -- Root Locus Compensated System (Plotted near origin only)

The term $(S^2 + (S(A_L^2 - K_C)/(-K_M)) + K/M)$ in Equation 56 is worthy of further examination. The natural frequency $\omega_n$ which is equal to $\sqrt{K/M}$ in the system can be changed because of the variable spring. The mass on the
ram, M, can also be changed which will change the term 
\((A_r^2 - K_5C)/ - K_5M\). Since \(\omega_n\) is decreased when input fre-
quency is decreased, the response is relatively unchanged.
As the mass is decreased, the term \((A_r^2 - K_5C)/ - K_5M\) is
increased and results in the complex pole being further away
from the imaginary axis. This makes it possible to operate
at higher frequencies.
CHAPTER V

CONCLUSIONS AND RECOMMENDATIONS

It has been stated previously that because of the complexity and size of such an undertaking, the scope of this thesis does not include the total design of a hydraulic vibration machine. Because fluid power research is quite young, essentially beginning with the demand of World War II, much is still unknown. The problem of control which is solved in this thesis may be inadequate after the components are actually fabricated. This situation would not be surprising since many of the constants used in the calculations had to be assumed. When these constants, such as damping in the valve and in the ram, are found empirically from the components themselves, then, depending on the deviations from the assumed values, some problems may need resolving.

Many researchers, particularly those who have had experience in this field, indicate that in many fluid control designs some of the control work must be done after a preliminary design of the system has been made and a machine built. An example of the problems arising and not accounted for in this study is that of resonant lines between valve and load. If this problem is encountered when the vibration machine is made, it is recommended that composite lines of different
lengths and/or diameter be tried, the small diameter being placed next to the valve(14). If practical, a conical line is recommended(15).

It is recommended that an oil cooling system be incorporated into the low pressure side of the oil system. It is essential to keep the oil temperature down when full flow is not utilized and to reduce temperature sensitivity effects as much as possible. The fluid spring was also designed to be cooled.

Since the natural frequency of the load is a function of many variables, the operating frequency will not be precisely matched without some manipulation. It is possible that until the system reaches a steady temperature state, a frequency match will be quite difficult. When a steady state condition is achieved, a frequency match should be obtainable by varying the volume of fluid in the fluid spring. If a specific frequency is not too critical, the operating frequency which is controlled by the oscillator can be changed to match the load natural frequency after they are close to each other. A separate signal from the output displayed on an oscilloscope might well be used to check output amplitude against the desired amplitude.

It is likely that there will be frequency drifting due to the noted influences. If this drift is not excessive, readjustment from time to time will be satisfactory. If the drift is excessive, then further control of the environment will be necessary. If the requirements are not too severe,
a condition of near frequency match may be desired. This near match would allow the amplitude to be adjusted either up or down as the drift requires.

In the very high frequency range it will not be possible to correct drift by changing the fluid spring as it is now designed since the adjustable heads on the cylinder must be moved. This will require frequency matching by the input or changing the cylinder design.

Noise problems are possible, but not probable. Flow instabilities sometimes cause high frequency noise, but rarely affects a system's performance\(^{16}\). Electrical noise can be a 60 cps nuisance sometimes, but can be prevented by using a 400 cps electrical system or it can be eliminated after it has become a problem. Amplifier noise is usually much too high a frequency to be a problem in this system.

It is recommended that the valve be made by a valve-manufacturing company which has had experience in this field and can fabricate to tolerances specified on the drawings. If this is not possible, the engineer responsible for this project should follow the construction very closely and be familiar with the requirements and problems connected with such a fabrication. Sections 9.5 and 9.54 of FLUID POWER CONTROL are useful references.

The total design, especially the high pressure items, should be checked by a registered professional engineer before use.
LIST OF REFERENCES


5. Ibid., p. 181.

6. Ibid., p. 381.

7. Ibid., p. 373.

8. Ibid., p. 253.

9. Ibid., p. 166.


16. Ibid., p. 363.

17. Ibid., p. 652.
APPENDIX A

COMPUTER PROGRAMS
POWER VALVE PROGRAM

1 WRITE(6,2)
2 FORMAT( 4OH,MEL MERRELL LOAD LOCUS FOR POWER VALVE)
   WRITE(6,5)
5 FORMAT(36H-INPUT INFORMATION SEE PROGRAM ABOVE)
   DIMENSION B(21),FOG(21),VOG(21),FF(21),V(21),PFORCE(21),
      IPV(21),FPVP(21),VELP(21)
   SUPPLY=3000.0
   ARAM=8.5
   RATCNG = .02
   WTSTOP=.24
   FLOWCH= 3.85
   FLOWSP=25.0
   RATSTR=.3
   FLOW=15.4
   PI=3.1415927
   RATIO=.5
   FWANT=150.
   CPVVIS=.04
   READ(5,2036)
2036 FORMAT(119HO
   1
     WRITE(6,2036)
     ENDIAM=.5
     PORT=360.0
     DIAM=.9
     K= 5.0
2015 PVWATE=DIAM*DIAM*RATIO
     DL = .2
     KCHNG=5.0
     KSTOP=30.0
     BMPV=PVWATE/386.0
     DIAMST=.9
     DIASTP=1.1
     DIAMCH=.10
     ENDSTP=.7
     ENDCHN=.10
     PORTCH=20.0
     PORTSP=360.0
     ENDSTR = .5
     KSTART = 5.
     FREQ=150.0
CVIS=15.0
BULK=100000.
DIVID=20.0
WATE=1000.
FSTOP=500.0
ASTRT=0.0
ACHNG1=0.02
APT=3.0
FCHNG=20.0
FNCHNG=50.0
FNSTOP=500.0
DIVID1=50.0
DIVIDA=DIVID1
ACHNG=ACHNG1
A=ASTRT
BM=WATE/386.0
F=FREQ
WN=F*6.2832853
FN=F
WNP=SQR(K/BMPV)
ZETA=.1
SQIGL=.1
FPN=WNP/6.2831853
R=F*6.2831853
IF(FN)40,40»3
VOL=(4.0*APT*APT*BULK)/(BM*WN*WN)
DO 45 I=2,20
B1=ATAN((-2.0*ZETA*R*WN)/((WN*WN)-(R*R)))
BZ=ARCSIN(0.0)
BINC=-(SQRT((B1-BZ)*(B1-BZ))/DIVID)
B(I)=B(I-1)+BINC
45 CONTINUE
TRANS=0.0
DO 55 I=1,20
FOG(I)=((SIN(B(I)))*(((WN*WN)/(R*R))-1.0)+((2.0*ZETA*WN
1)/R)*(COS(B(I))))
IF(FOG(I)>GE.0) GO TO 552
FOG(I)=-FOG(I)
552 VOG(I)=COS(B(I))
FF(I)=FOG(I)*A*R*R*BM
V(I)=VOG(I)*A*R
PV(I)=SUPPLY-FF(I)/ARAM
FPOG(I)=SIN(B(I))*(((WN*WN)/(R*R)))1.0+3.442*PV(I)/({
1BMPV*R*R})+COS(B(I))*(((2.0*SQIGL*WNP)/R)+DL*(SQRT(PV))
2/(BMPV*R*R))
55 CONTINUE
DO 65 I=1,20
FAVAL=27.7777777*(900.0-(V(I)*V(I)))
IF(V(I)>GE.30.0 OR FF(I)>GE.FAVAL)GO TO 80
IF (TRANS.GT.0. AND (I).GE.20) GO TO 91
65 CONTINUE
A=A+ACHNG
GO TO 550
80 A=A-ACHNG/DIVIDA
TRANS=1.0
GO TO 550
91 PRES=(BULK*APT*A)/VOL
VELMAX=AMAX1(V(1),V(2),V(3),V(4),V(5),V(6),V(7),V(8),V(9)
1),V(10),V(11),V(12),V(13),V(14),V(15),V(16),V(17),V(18)
2,V(19),V(20))
APV=(VELMAX*8.4*360.0)/(3833.9*3.141593*DIAM*PORT)
DO 260 I=1,2
PFORCE(I)=FPOG(I)*APV*R*R*BMPV
VELP(I)=VOG(I)*APV*R
260 CONTINUE
XDOTMX=FLOW/((PI*ENDIAM*ENDIAM)/4.)
XDOTSQ=XDOTMX*XDOTMX
AREA=PI*ENDIAM*ENDIAM/4.
FACTOR=3000.0*AREA/XDOTSQ
DO 280 I=1,2
PVFAVA=FACTOR*-(VELP(I)*VELP(I))
IF (SWITCH.EQ.0.) GO TO 275
IF (VELP(I).LE.XDOTMX.OR.PFORCE(I).LE.PVFAVA) GO TO 286
IF (VELP(I).GE.XDOTMX.OR.PFORCE(I).GE.PVFAVA) GO TO 2240
280 CONTINUE
GO TO 125
IF (F.LT.FWANT) GO TO 190
FORMAT(53HO DIAMETER AT ORIFICE-WEIGHT-PORT-END DIAMET
1ER FLOW,5F 15.5)
WRITE(6,2021)DIAM,PVWATE,PORT,NDIAM,FLOW
WRITE(6,295)
295 FORMAT(102HOFREQUENCY VALVE PRESS VALVEAMPLI ANGLE
1 VALVE FORCE VALVE VELOCITY SQIGL VALVE NAT FREQ
20 )
WRITE(6,3000)F,PV(1),APV,B(1),PFORCE(I),VELP(I),SQIGL,FNP
WRITE(6,110)
110 FORMAT(100HOFREQUENCY AMPLITUDE ANGLE F/AWWM V/A
1W VELOCITY FORCE VOLUME NAT-FREQ ZETA PRES)
WRITE(6,120)F,A,B(1),FOG(20),VOG(1),VELMAX,FF(1),VOL,FN
1,ZETA,PRES
11,F10.5,F10.1)
SWITCH=0.0
GO TO 190
125 A=ASTART
IF (F-FCHNG)145,145,145
F=F+(FCHNG/5.0)
IF (F-FSTOP) \leq 10,10,155

145 F=F+FCHNG
U=F/(F-FCHNG)
IF (F \leq FCHNG) GO TO 149
ACHNG=ACHNG/1.1

149 ACHNG=ACHNG/(U*U)
IF (F-FSTOP) \leq 10,10,155

155 A=ASTRT
DIVIDA=DIVID1
ACHNG=ACHNG1
F=FREQ
IF (FN-FNCHNG) \leq 170,180,180

170 FN=FN+FNCHNG/5.0
IF (FN-FNSTOP) \leq 1,10,200

180 FN=FN+FNCHNG
IF (FN-FNSTOP) \leq 1,10,200

190 K=K+KCHNG
IF (K \leq KSTOP) GO TO 2015
K=KSTART
DIAM=DIAM+DIAMCH
IF (DIAM \leq DIAMSTOP) GO TO 215
DIAM=DIAMST

2215 ENDIAM=ENDIAM ENDCHN
IF (ENDIAM \leq ENDSTP) GO TO 2015
K=KSTART
DIAM=DIAMST
ENDIAM=ENDSTP
RATIO=RATIO-RATCNG

IF (FLOW \leq FLOWSP) GO TO 2015
K=KSTART
DIAM=DIAMST
ENDIAM=ENDSTP
RATIO=RATSTR
FLOW=FLOW+FLOWCH
IF (FLOW \leq FLOWSP) GO TO 2015

GO TO 200
SWITCH=1.0
F=F-FCHNG
A=ASTART
GO TO 10

200 STOP
END
ENTRY LOCI
LIST OF MNEMONIC CHARACTERS FOR POWER VALVE PROGRAM

ARAM AREA OF RAM
BMPV MASS OF POWER VALVE SPOOL
BULK BULK MODULUS OF OIL
CPVVIS COEFFICIENT OF DAMPING ON POWER SPOOL
DIAM DIAMETER OF POWER SPOOL
DIAMCH DIAMETER INCREMENT
DIAMST INITIAL DIAMETER OF POWER SPOOL
DIASTP MAXIMUM DIAMETER OF POWER SPOOL
DIVID ANGLE DIVIDER
DIVIDI INITIAL DECREASING AMPLITUDE DIVIDER
DL DAMPING LENGTH
DIVIDA DECREASING AMPLITUDE DIVIDER
ENDCHN SPOOL END DIAMETER ENCREMENT
ENDIAM SPOOL END DIAMETER
ENDSTR INITIAL SPOOL END DIAMETER
ENDSTP MAXIMUM SPOOL END DIAMETER
FNP POWER VALVE NATURAL FREQUENCY
FPOG F/AWWM FOR POWER VALVE
FWANT INITIAL FREQUENCY
K SPRING CONSTANT FOR POWER VALVE
KCHNG K INCREMENT
KSTART INITIAL SPRING CONSTANT
KSTOP MAXIMUM K
PORTCH PORT INCREMENT
PORTSP MAXIMUM PORT
PV PRESSURE DROP ACROSS POWER VALVE ORIFICE
PVWATE WEIGHT OF POWER SPOOL
SOIGL DAMPING OF POWER VALVE
SUPPLY SUPPLY PRESSURE
XDOTMX MAXIMUM VELOCITY OF POWER SPOOL

FOR OTHERS SEE LIST OF MNEMONIC CHARACTERS FOR RAM PROGRAM
RAM LOAD PROGRAM

WRITE(6,2)
2 FORMAT(29HOMEL MERRELL LOAD LOCUS THREE)
READ(5,4)FREQ,CVIS,BULK,DIVID,VOLCH,WATE,FSTOP,VLSTP,AST1RT,ACHNG,VOLST,APT,FCHNG
4 FORMAT(2F3.0,F2.0,F6.0,F3.0,F3.0,F3.0,F3.0,F3.0,F3.0,F3.0)
1 F5.4,F5.2,F2.1,F3.0)
WRITE(6,506)
506 FORMAT(18H+INPUT INFORMATION)
WRITE(6,6)
6 FORMAT(11HOFREQUENCY VISCOUS-K BULK MOD DELTA A D
1ELTA VOL WEIGHT FREQ-STOP VOL STOP INITIAL A POT AR
2-F)
WRITE(6,8)FREQ,CVIS,BULK,ACHNG,VOLCH,WATE,FSTOP,VLSTP,AS
TRT,APT,FCHNG
8 FORMAT(2F8.0,F13.0,F10.4,F13.0,2F9.0,F12.0,F8.3,F11.2,F7
1.)
FNCHNG=50.0
FNSTOP=500.0
A=ASTRT
BM=WATE/386.0
F=FREQ
FN=0.0
14 R=F*6.283185
WN=FN*6.283185
ZETA=CVIS/(2.*BM*WN)
WN=FN*6.283185/(SQRT(1.0-(ZETA*ZETA)))
B1=ATAN((-2.0*ZETA*R*WN)/(WM*WN)-(R*R))
RN=WN/6.283185
B2=ARCOS(0.0)
30 B=B1
BINc=-(SQRT((B1-B2)*(B1-B2)))/DIVID
IF(FN)35,35,32
VOL=(4.0*APT*A/PT*BULK)/(BM*WN*WN)
35 FOG=((SIN(B)) *((WM*WN)/(R*R)-1.0)+((2.0*ZETA*WN)/R)*(
1COS(B))
IF(FOG)356,40,40
FOG=-FOG
VOG=COS(B)
V=VOG*A*R
PRES=(BULK*A/PT*A)/VOL
FF=FOG*A*R*R*BM

47
FAVAL = 27.777777*(900.0 - (V*V))
IF (FAVAL) 200, 65, 65
65 IF (FF - FAVAL) 7, 95, 200
200 A = A + ACHNG/20.
WRITE (6, 909)
909 FORMAT (2H+5)
B = B1
201 FOG = ((SIN(B)) * (((WN*WN)/(R*R)) - 1.0)) + ((2.0*ZETA*WN)/R)*COS(B))
IF (FOG) 2012, 2, 2, 202
202 VOG = COS(B)
V = VOG*A*R
FF = FOG*A*R*R*BM
PRES = (BULK*APT*A)/VOL
FAVAL = 27.777777*(900.0 - (V*V))
IF (FF .LE. 0.0 AND. V .LE. 29.) GO TO 208
WRITE (6, 901)
901 FORMAT (2H+1)
IF (FAVAL) 208, 2, 7, 207
WRITE (6, 903)
FORMAT (2H+2)
Z = SQRT (B*B)
IF (FOG.LE. FAVAL .AND. Z .GE. B2) GO TO 95
WRITE (6, 905)
905 FORMAT (2H+3)
208 B = B + BINC
IF ((SQRT (B*B)) B2) 201, 201, 200
B = B + BINC
IF ((SQRT (B*B)) B2) 35, 35, 8
A = A + ACHNG
907 FORMAT (2H+4)
WRITE (6, 907)
GO TO 30
95 WRITE (6, 100)
100 FORMAT (100HOFREQUENCY AMPLITUDE ANGLE F/AWWM V/A
1W VELOCITY FORCE VOLUME NAT-FREQ ZETA PRES)
105 WRITE (6, 110) F, A, B, FOG, VOG, FF, VOL, RN, ZETA, PRES
11, F10.5, F10.1)
A = ASTART
IF (F = FCHNG) 117, 120, 120
117 F = F + (FCHNG/5.0)
IF (F = FSTOP) 14, 14, 130
120 F = F + FCHNG
IF (F = FSTOP) 14, 14, 130
130 A = ASTRT
F = FREQ
IF (FN = FNCHNG) 1356, 136, 136
1356 FN = FN + FNCHNG / 5.0
1357 IF (FN = FNSTOP) 14, 14, 150
136  FN = FN + FNCHNG
     IF (FN = FNSTOP) 14, 14, 150
150 STOP
     END
$ENTRY  LOCUS
LIST OF MNEMONIC CHARACTERS FOR RAM PROGRAM

A  AMPLITUDE
ASTRT  INITIAL AMPLITUDE
B  ANGLE
B1  STARTING POINT IN 1ST QUADRANT
B2  ENDING POINT IN FIRST QUADRANT
BINC  ANGLE INCREMENT
BM  MASS OF LOAD
CVIS  COEFFICIENT OF DAMPING FOR RAM
F  OPERATING FREQUENCY CPS
FAVAL  AVAILABLE FORCE CALCULATED FROM CURVE
FF  FORCE
FN  NATURAL FREQUENCY CPS
FNSTOP  MAXIMUM ON NATURAL FREQUENCY
FOG  F/AWWM FOR RAM
PRES  PRESSURE
RN  NATURAL FREQUENCY RADIANS PER SECOND
VOG  V/AW
VOL  VOLUME OF FLUID
Z  ABSOLUTE B
ZETA  DAMPING RATIO FOR RAM
SAMPLE CALCULATIONS

\( A_{\text{max}} \) for Power Valve

\[
q_{\text{max}} = 70 A_{\text{max}} \sqrt{P_s}
\]
\[
A_{\text{max}} = q_{\text{max}}/(70 \cdot \sqrt{P_s})
\]
\[
A_{\text{max}} = 264/70 \cdot 54.77 = .0689
\]

Linearization Constants for Power Valve

\[
K_4 = \frac{\partial q_r}{\partial Y} = C_d W \sqrt{2(P_s - P_r)}/\rho
\]
\[
= .6 \cdot 3.1415 \cdot \sqrt{2(3000)/7.48 \cdot 10^{-5}}
\]
\[
= 16,869
\]

\[
K_5 = \frac{\partial q_r}{\partial P_r} = \frac{(-Y_o C_d W \sqrt{2/\rho})}{(2 \sqrt{P_s - P_r})}
\]
\[
= (-.02 \cdot .6 \cdot 3.1415 \sqrt{2/7.48 \cdot 10^{-5}})/2 \sqrt{3000}
\]
\[
= -.056
\]

Volume of Fluid in Fluid Spring for a Specified Frequency and Mass

\[
K = \frac{4A_p^2 \beta}{\nu_t} = \omega^2_{nM}
\]
\[
V_t = \frac{4A_p^2 \beta}{\omega^2_{nM}}
\]
\[
= 4.25 \cdot 100,000/(628.3)^2 \cdot 2.59
\]
\[
= 9.8 \text{ in}^3
\]
ASSUMED VALUES

The coefficient of damping for the ram was estimated to be 15 lb sec/in on the basis of a similar design (17). The area was different; hence, a scaling process was necessary.

The coefficient of damping for the power valve was assumed to be 0.03 which would place the damping near 0.1 for a natural frequency of 100 cps.

The actual bulk modulus of the commonly used silicon oils is approximately 140,000 to 160,000; however, because of bubbles which are created in the oil, a value of 100,000 was assumed.
PRELIMINARY DESIGN OF A HYDRAULIC VIBRATION MACHINE
WITH VARIABLE AMPLITUDE AND FREQUENCY,
USING MULTISTAGE AMPLIFICATION
AND FEEDBACK CONTROL

An Abstract of A Thesis
Presented to the
Department of Mechanical Engineering
Brigham Young University

In Partial Fulfillment
of the Requirements for the Degree
Master of Science

by
Melvin Joseph Merrell
21 December 1965
ABSTRACT

This thesis covers the preliminary design of a hydraulic vibration machine and was undertaken to fill the need for such a machine in the dynamics area of the Mechanical Engineering Department of the Brigham Young University.

Power matching of several stages of amplification, both electrohydraulic and hydraulic, was accomplished as was the designing of the mechanical components involved and the detailed drawings of these. An analysis necessary to examine stability was also accomplished. The thesis does not cover the fabrication of the vibration machine, and many details will arise when this is undertaken.

The main specifications are: 25,000 pounds maximum force vector, maximum operating pressure of 3000 psi, two inch maximum displacement, and 1000 pounds maximum load.

The system includes an electrical input by a function generator or an oscillator, a compensating lead circuit, a Moog servovalve, a power valve, and a ram actuator. In connection with the ram is a fluid spring which enables operation at resonance over much of the frequency range. The output, acceleration, velocity and displacement are monitored and are used as a feedback control.
A linear approximation was made to study stability.

APPROVED: