

THE DESIGN OF A HYDRAULIC
ORIFICE FLOW CONTROLLER

by

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Department of Mechanical Engineering, May 18, 1951

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TABLE OF CONTENTS

	Page
ABSTRACT.....	ii
ACKNOWLEDGMENTS.....	iii
CHAPTER I INTRODUCTION TO THE PROBLEM.....	1
CHAPTER II CORNING EXPERIMENTAL HYDRAULIC FEEDER.....	4
A. Description.....	4
B. Dynamic Analysis.....	7
C. Suggested Changes to Increase Stability.....	9
CHAPTER III THE PROPOSED DESIGN.....	11
A. System Specifications.....	11
1. Dynamic.....	11
2. Physical.....	19
3. Economic.....	22
B. System Design.....	22
C. Components.....	25
1. Hydraulic Flow Control Valve.....	25
2. Torque Motor and Error Amplification.....	33
3. Auxiliary Components.....	33
4. The Needle Drive and Supporting Framework.....	38
D. Dynamic Analysis.....	38
E. Economic Considerations.....	60
CHAPTER IV CONCLUSIONS.....	62
BIBLIOGRAPHY.....	64

Superscripts in the text refer to numbered items in the bibliography.

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ABSTRACT

The mass production of glassware includes a feeding operation, in which a machine meters a molten glass stream into discrete gobs of closely-controlled shape and weight, in accordance with a prearranged schedule. A number of adjustments built into the machine permit its adaptation to the gob formation for a variety of finished products and insure continued satisfactory operation despite varying conditions in the glass stream.

This thesis includes a discussion of a hydraulic position-servomechanism gob-feeder developed by the Corning Glass Works, and suggestions for improving its dynamic operation.

The main part of the thesis constitutes a development of the specifications for, and a complete redesign of, the feeder, to provide a more convenient, flexible system with improved dynamic characteristics. The proposed design should be capable of operation over a range of speeds and gob weights considerably broader than now experienced.

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CHAPTER I

INTRODUCTION TO THE PROBLEM

The continuous-process production of glass "pressware", such as kitchen ovenware and television tubes, consists of metering a molten glass stream into tear-drop-shaped gobs of correct shape and weight, locating these gobs in a female die, pressing the glass to final shape with a male die, and subsequent heat treatment and cooling of the molded articles.

The glass is melted and compounded in a large gas or oil-fired furnace from which it flows by gravity into the forehearth, a long (of the order of twenty feet) trough of rectangular cross section. During its passage through the forehearth the glass is "conditioned" and carefully temperature controlled by a series of gas flames directed onto the glass surface. A gate valve at the end of the forehearth directs and controls the flow of molten glass into a ceramic bowl.

In the metering or feeding process, the molten glass is forced through an orifice in the bottom of this bowl by a ceramic needle which is moved vertically in a pattern so as to conform with a program on a circular cam. The gob, formed to the desired (and exacting) weight and shape specifications by the program cam is shorn from the parent melt and falls into a female die on a rotary table. The table is quickly indexed once to a position which brings the glass beneath a mating die where the still-molten glass is pressed into its final shape. The shaped article is heat-treated at subsequent indexing stations and then removed and placed on a continuous belt which passes through an annealing furnace.

This thesis is concerned with the feeding operation just described. In current commercial practice, the programming of the needle is accomplished by a mechanical linkage. The program cam directly activates a lever which carries the needle. The acceleration for the downward stroke of the needle against the viscous resistance of the molten glass is furnished by gravity, and the fidelity of the needle's movement to the cam program is abetted by mounting additional weights at the needle. The limited gravitational acceleration possible in the absence of a positive downward driving force severely restricts the production speed of the entire process, since beyond a certain point fidelity of reproduction of the program cam cannot be insured by increased loading of the needle assembly.

As the ceramic needle alone weighs approximately 200 pounds, the program cams have to be manufactured from steel plate and heat-treated to withstand the wear incurred by accelerating and decelerating the needle assembly. Even so the life of these cams is definitely limited, and since a different camming surface is required for each different article manufactured, cam preparation is an expensive operation.

These production limitations have been largely overcome by the use of an experimental hydraulic feeder developed by the Corning Glass Works. Essentially a position servomechanism with mechanical linkage and using hydraulic power, this feeder represents a great improvement over the previous all-mechanical system - the system being used by all other companies. Certain stability problems, however, have developed in the experimental model. The second part of this paper is

a discussion of the existing experimental feeder, together with a dynamic analysis of it and suggestions for improving the stability of the feeder. The third part of this paper is a complete redesign of the feeder, incorporating some features suggested by the existing model.

CHAPTER II

CORNING EXPERIMENTAL HYDRAULIC FEEDER

A. Description

The experimental feeder is mounted on the end of the fore-hearth, which is approximately 7 feet above the floor level of the glass plant and is reached by a ladder and catwalk. A 1/2 hp a-c induction motor mounted on the catwalk adjacent to the feeder synchronizes the operations of the feeder, press and index table. A friction disc variable-speed drive and worm and worm-wheel speed-reduction unit, also on the catwalk, reduce the motor speed to the operating speed of 5-50 articles a minute. The reduced-speed shaft directly drives a plate cam which operates the glass shear. This same shaft drives the program cam through a mechanical differential which permits adjusting the phase of the needle program with regard to the rest of the cycle. An extension of the reduced-speed shaft drives a flexible shaft which interconnects and synchronizes the feeder with the press and index table by means of a ganged series of plate cams on the press frame; the plate cams actuate air valves connecting the press and index table. This flexible shaft tends to "wind up" under the torsional load of the plate cams, resulting in non-uniform rotary motion.

~~A schematic of the existing experimental feeder only is shown in Figure 1.~~ The program cam is rotated once per feeding cycle; it translates a follower which in turn translates, through a 2:1 or 4:1 lever, the spur gear of the differential, consisting of the gear and two racks. The rack on the right, through a lever of fixed ratio, moves the

piston of a four-way valve against a static spring force. The valve ports oil to the up (or down) chamber of a fixed-piston, moving-cylinder ram, which moves the ceramic needle with respect to the fixed orifice, thus controlling gob formation. The feedback from the ram to the left-hand rack of the differential is all mechanical and includes a link of adjustable length which permits varying the steady-state, lowermost position of the needle with respect to the orifice. The adjustment can be reached from the floor level. Also in the feedback is a lever with a movable pivot which permits varying the magnitude of the feed-back signal, thereby varying the total stroke of the needle with respect to the lift of the program cam. This adjustment can be reached only from the catwalk level.

A constant-pressure source furnishes hydraulic fluid at 340 to 370 psi, through the metering four-way valve, to the activating cylinder; the fluid acts on three square inches during the up stroke and two square inches during the down stroke.

The actual gob formation is influenced by a number of variables, of which some cannot be predicted or controlled to the extent desirable. If the problem were completely understood and the variables properly under control, the program cam together with the speed control of the cam would completely prescribe the proper solution. As it is, temperatures and therefore viscosity vary within certain limits, glass properties and conditioning vary with time even for a single run, and stratification of the glass in the forehearth is unavoidable; these and additional unexplained parameters make necessary the incorporation of a number of adjustments. The stepless variable-speed drive, the phase

adjustment between the process-sequencing drum cam and the program cam, the needle zero position, and the stroke adjustment are four such adjustments. There is also provision for adjusting the needle laterally and longitudinally with respect to the orifice and for raising and lowering the sleeve, a ceramic cylinder surrounding and mounted coaxially with the needle, and rotated at constant speed so as to minimize the stratification of the glass in the bowl.

B. Dynamic Analysis

Figure 2a represents a block diagram of the existing experimental feeder with each block representing a mechanical entity, i.e., a lever, part of the differential, etc.; the lower case letters refer to the lever arms of Figure 1; K_f is the proportionality factor between the ram velocity and the displacement of the pilot-piston valve, while ω_n , the natural frequency of the system and ζ the damping ratio, are combinations of system parameters as derived on page 42; s represents a derivative with respect to time. Figure 2b is a revision of the block diagram as shown in Figure 2a to better illustrate its dynamic characteristics.

Writing the transfer function

$$\frac{Y_1}{y} = \frac{AG(s)}{1 + AG(s)}$$

results in the equation

$$\left[s \left(\frac{s^2}{\omega_n^2} + \frac{2\zeta s}{\omega_n} + 1 \right) + \frac{ceK_f}{df} \right] Y_1 = \left(\frac{ceK_f}{df} \right) y$$

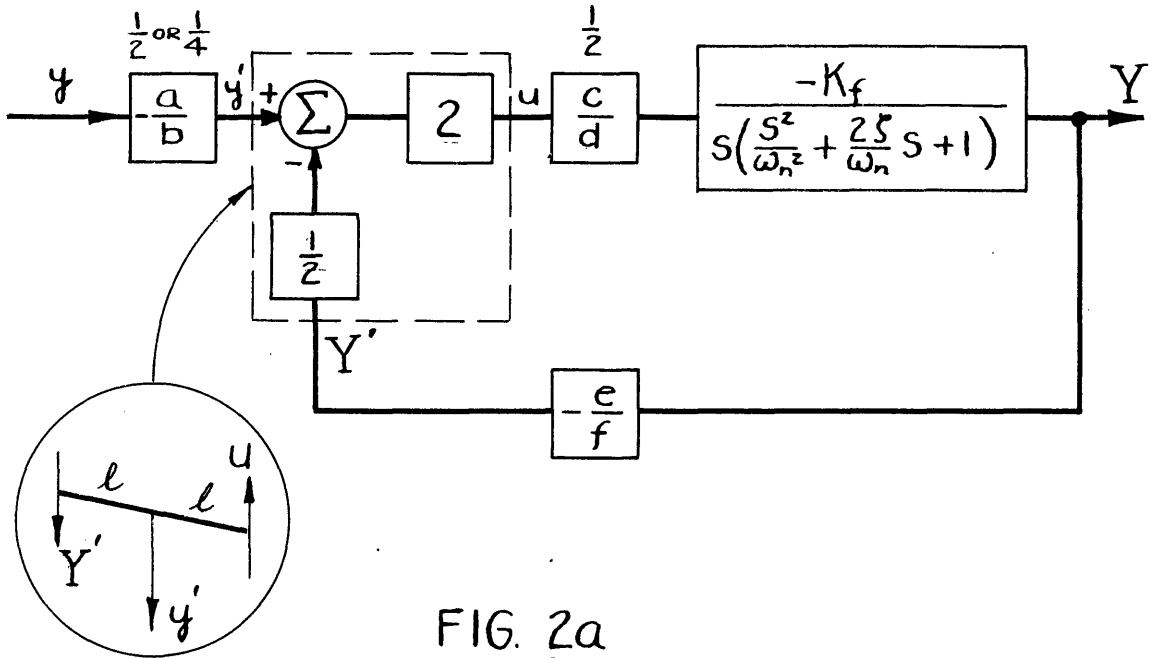


FIG. 2a
BLOCK DIAGRAM OF
FIGURE 1.

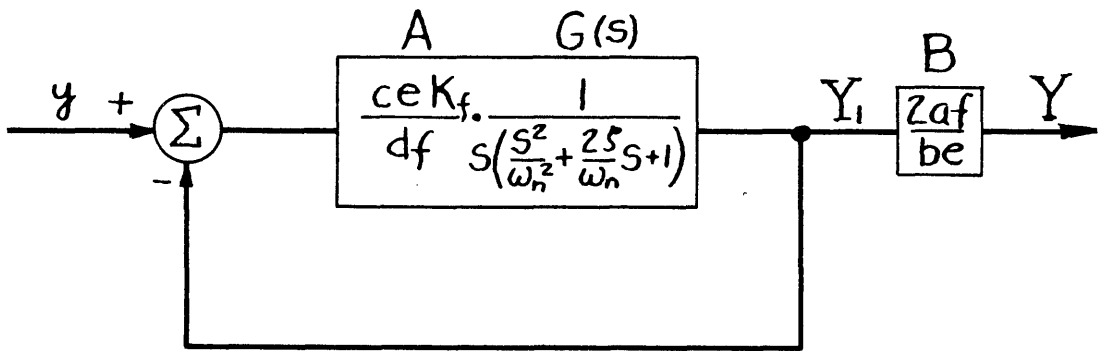


FIG. 2b
REARRANGEMENT OF FIGURE 2a.

AD

which demonstrates that for the steady state, when time derivatives are equal to zero,

$$Y_1 = y.$$

Now
$$Y = \frac{2af}{be} Y_1$$

or
$$Y = \frac{2af}{be} y,$$

which demonstrates that moving the adjustable pivot of lever e-f so as to increase e reduces the stroke relative to the cam lift.

However the ratio e/f appears also in the dynamic part of the solution, being part of the overall gain of the system ceK_f/df .

Since the system is of the third degree, with a single integration, increasing the gain moves the locus of the overall transfer function of the system on a Nyquist plot toward the point of instability. Thus an attempted reduction of the stroke unfortunately increases the gain and leads to instability, manifested by violent hunting of the entire system. This phenomenon has been somewhat alleviated, but not obviated, by increasing the damping ratio of the hydraulic system by fostering increased fluid leakage in the pilot piston by beveling the otherwise-square pilot-piston lands. The resonant condition is also avoided by achieving reduced stroke of the needle by maintaining the stroke-adjust lever fulcrum in a position of stability and by using program cams of lower lift to provide a smaller input.

C. Suggested Changes to Increase Stability

If the stroke adjustment were not included in the closed loop,

stroke adjustment would not affect system stability. It is proposed, therefore, that the stroke adjustment be removed to a point external to the loop, for example, between the program cam and differential spur gear. An adjustable pivot on lever a-b together with the present device of using cams of $7/8$ " , $1\ 1/4$ " or $2\ 1/2$ " rise might provide strokes over the desired range of 0 to 8".

CHAPTER III

THE PROPOSED DESIGN

A. System Specifications

1. Dynamic

The physical phenomena associated with gob formation are only recently receiving adequate attention; accordingly little quantitative information is available with which to establish system specifications. To a considerable extent the specifications outlined by Corning and purported to be the maxima incurred by them in their feeding operations are but a reflection of the capabilities of their present experimental prototype feeder. For example, maximum force levels of 2400 pounds on the up stroke and 1600 pounds on the down stroke were prescribed. These figures simply represent the maximum hydraulic forces obtainable with their system including the Vickers hydraulic pressure source. The maximum source pressure is 800 psi and ram areas are three square inches "up" and two square inches "down". It should be noted however that these forces are obtainable only for zero velocity of the needle; any flow associated with ram velocity would require pressure drops across the orifices of the pilot valve and thereby reduce the available pressure at the needle-ram piston.

Similarly the maximum velocities suggested as being the maximum required (24 inches per second for the up stroke and 36 inches per second for the down stroke) can be derived from the flow capacity of the Vickers unit at the present normal operating pressure of 355 psi. The horsepower

input to the Vickers unit is 5 hp and assuming an overall efficiency of the Vickers unit of 0.75:

$$\eta = \frac{\text{hp}_{\text{out}}}{\text{hp}_{\text{in}}} = \frac{\Delta p Q}{\text{hp}_{\text{in}}}$$

$$Q = \frac{\eta \text{ hp}_{\text{in}}}{\Delta p} = \frac{0.75(5)(33000)(12)}{355 \times 60} = 70 \text{ in}^3/\text{sec}$$

$$v = \frac{Q}{A} = \frac{70}{2} = 35 \text{ in/sec for the down stroke}$$

$$= \frac{70}{3} = 23.3 \text{ in/sec for the up stroke}$$

Again note that these velocity maxima occur at force levels considerably less than the 2400 and 1600 pounds suggested. The source fluid pressure is 355 psi and some pressure differential must be experienced across the pilot-valve ports to provide the flow. The forces actually available at the needle for these maximum velocities are therefore appreciably less than $355 \times 2 = 710$ pounds down and less than $355 \times 3 = 1065$ pounds up; compare these practical values with the 1600 and 2400 pounds specified by Corning.

An analysis of the forces expected in the needle suggests that the major resistance is due to the viscous-friction resistance of the molten glass, which of course varies linearly with the velocity of the needle. Therefore maximum force levels are experienced at maximum velocities; the specification of maximum forces independent of velocity, and maximum velocities independent of the forces (implied by Corning's specifications) is therefore inapplicable to the problem at hand.

A perhaps more rational approach to the question of forces encountered is shown in Figure 3. This represents, approximately to scale, a cross section of the bowl, showing the orifice, sleeve and needle. If we select from available dimensional data the combination of factors which will maximize the forces, the relationships below will obtain, under the following assumptions: 1) that the primary force resisting needle translation is viscous shear, τ , 2) that the velocity gradient is linear between needle and sleeve and between needle and bowl, and 3) that since the downward acceleration of the needle approximates harmonic motion, maximum velocity will occur when the stroke equals $D/2$.

$$F = \sum \tau S \quad \text{where} \quad \tau = \mu \frac{du}{dx}$$

S = surface area of cylinder (in^2)

μ = viscosity of molten glass ($\# \text{sec}/\text{in}^2$)
(4500 poise or $0.065 \# \text{sec}/\text{in}^2$)

$\frac{du}{dx}$ = velocity gradient

A = diameter of orifice ($5''$)

B = diameter of needle ($7''$)

D = stroke of needle ($3 \frac{7}{8}''$)

l_1 = the total head of glass ($16''$) minus
 l_2 ($\approx 13.5''$)

l_2 = the clearance E ($5/8''$) + $1/2 D$ ($\approx 2.5''$)

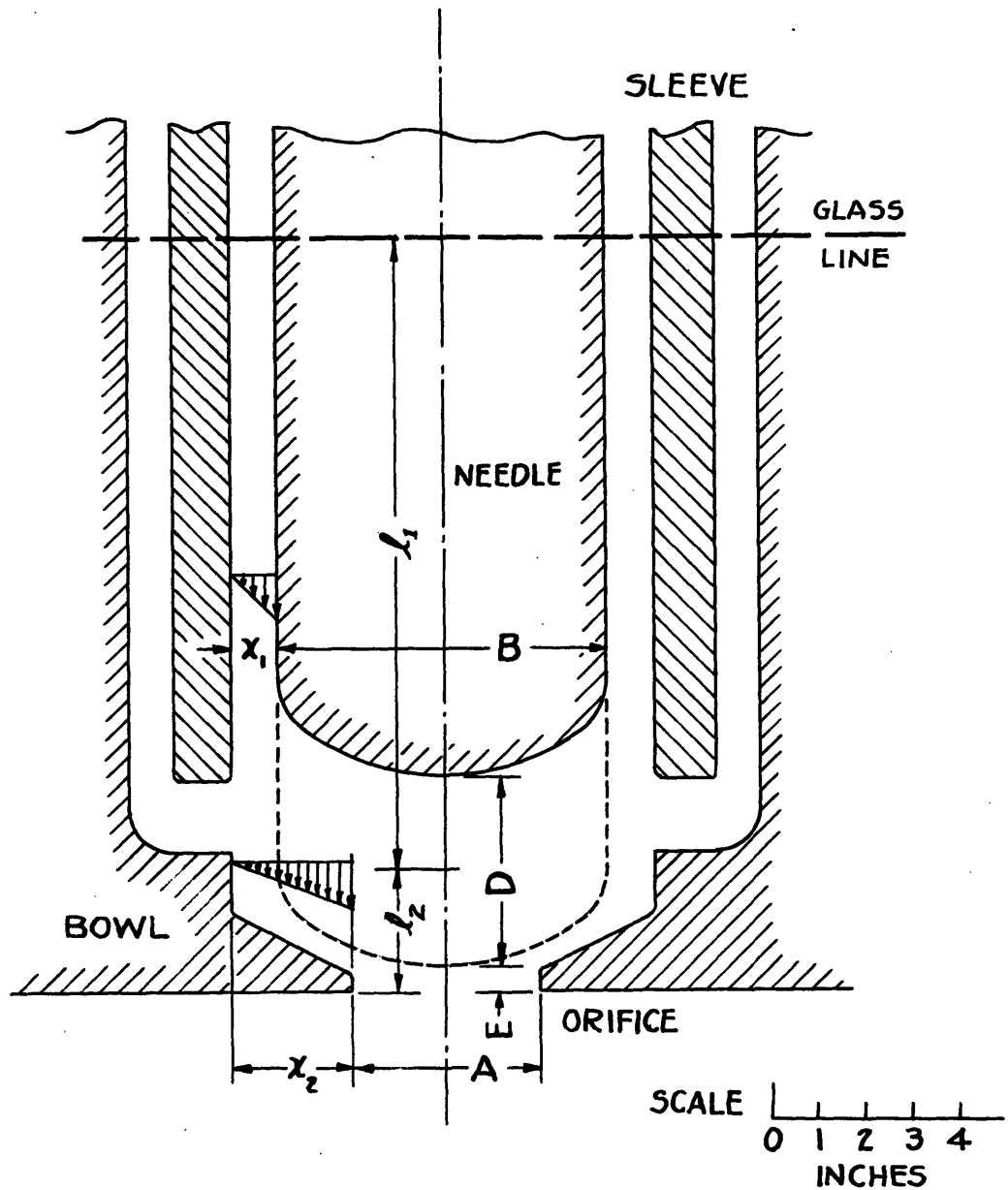
x_1 = clearance between sleeve and needle ($1''$)

x_2 = clearance between needle and bowl ($\approx 2''$)

β = coefficient of viscous friction ($\# \text{sec}/\text{in}$)

Then

$$F = \mu \left(\frac{V_{\max}}{x_1} \right) \pi B l_1 + \mu \left(\frac{V_{\max}}{x_2} \right) \pi A l_2$$



GOB WEIGHT (OZ.)	NEEDLE DIAMETER B (IN.)	ORIFICE DIAMETER A (IN.)	STROKE D (IN.)	CLEARANCE E (IN.)	DROPS PER MINUTE
60	7	3.25	1.25	0.625	6
212	7	5.0	3.875		4
162	7	4.5	3.25	0.75	5

SLEEVE INSIDE DIAMETER = 9.0 IN.

FIG. 3
DIAGRAM AND DATA FOR
VISCOS COEFFICIENT ANALYSIS

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Substituting numerical values in the above equations yields:

$$F = v_{\max} (0.065) \pi \left(\frac{7 \times 13.5}{1} + \frac{5 \times 2.5}{2} \right)$$

$$F = 20.5 v_{\max} = \beta v_{\max}$$

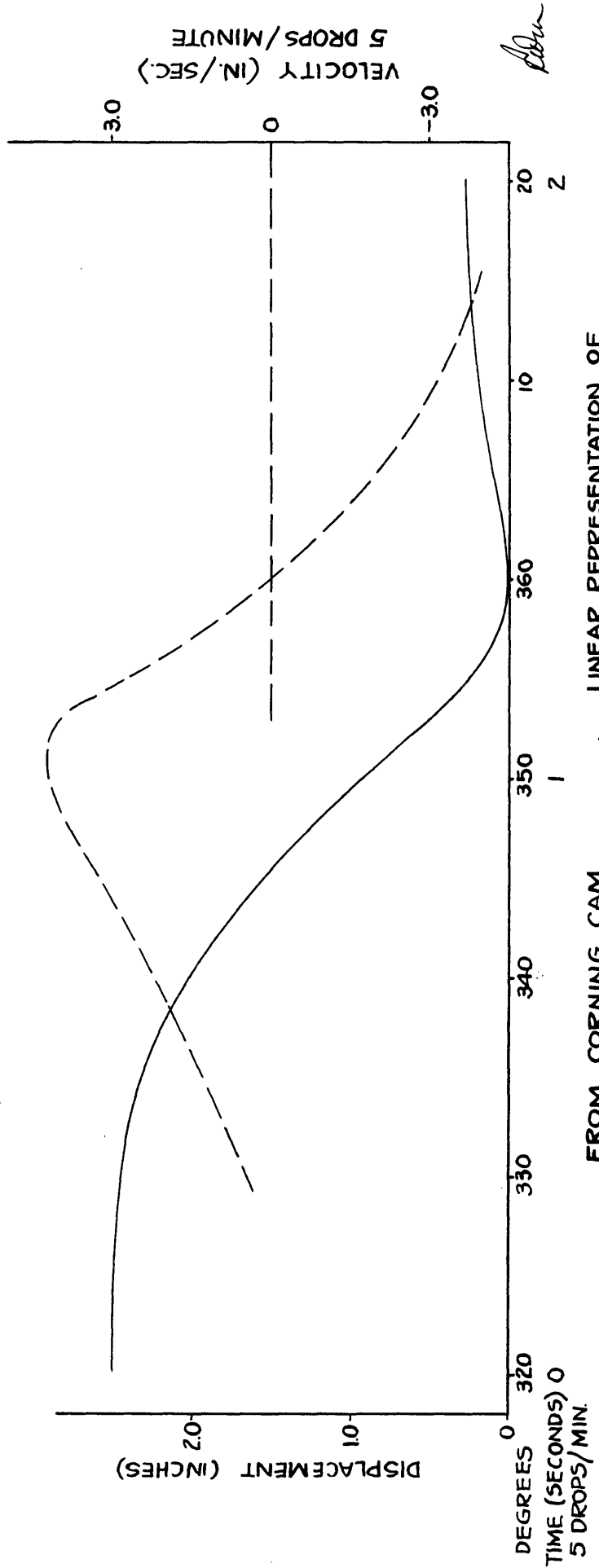
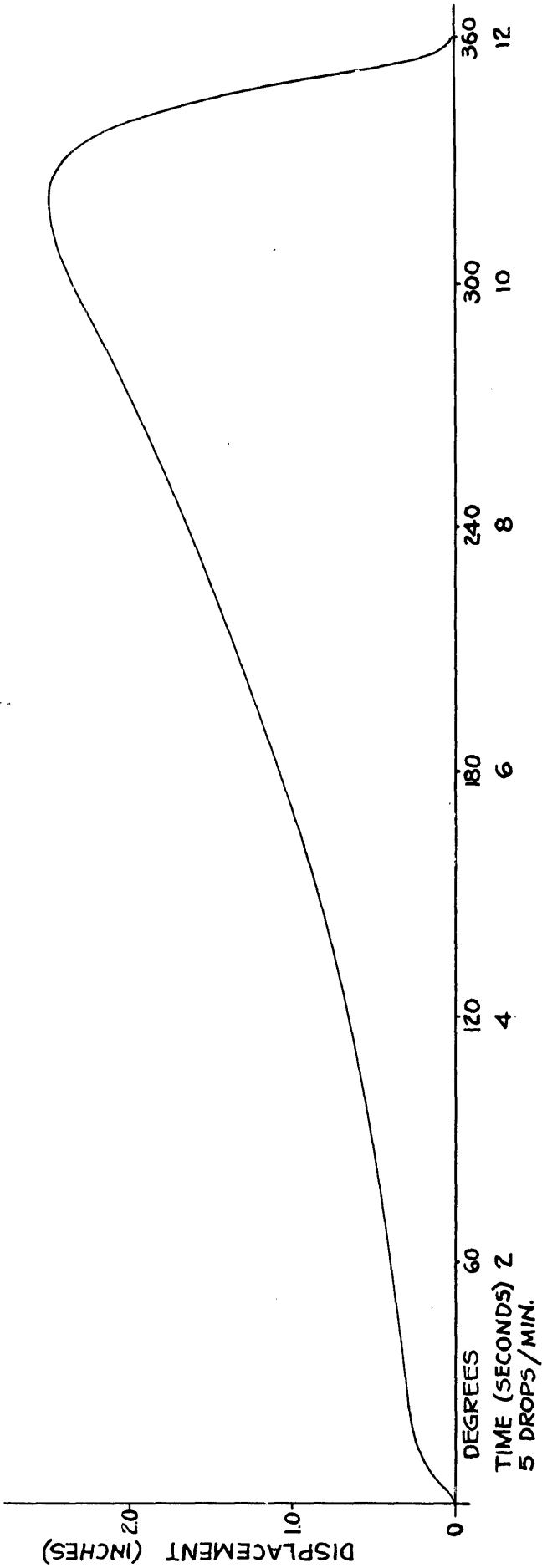
or $\beta = 20.5$

No allowance has been made for turbulence or flow except across the linear velocity gradient in the molten glass, and the actual condition existing at the orifice edge has been ignored in the above simplified analysis. Therefore as an engineering approximation and to insure adequate available force, the coefficient of viscous friction calculated above as being 20.5 is selected as 40 for the proposed design specifications.

To establish the actual velocities and accelerations experienced by the needle, a series of actual program-cam drawings were procured and from them was selected the cam with the most critical profile. The full rotation displacement diagram of the cam together with the displacement and velocity diagram of the section of greatest rate of change (to an extended abscissa scale) are reproduced in Figure 4. At the cyclical speed prescribed for this cam (5 drops/min, i.e., 5 revolutions/min) the maximum velocity and acceleration as measured from the cam drawing and transferred to the needle are 5.65 in/sec and 23.4 in/sec² (a 2.5" rise on the cam causes a 3.25" stroke; therefore, there is an amplification of 1.3 between cam and needle). As far as can be ascertained from available cam drawings, although higher cyclical speeds are used (Corning specified a speed range of 5 to 50 drops/min, but the fastest cam drawing available

was 6 drops/min), these maxima represent approximately the values experienced now, since, if cyclical speed were increased, stroke would decrease in about the same ratio. However since velocity and force and therefore the power requirements of the system are so interrelated, it is thought wise to design for far higher speeds than are now experienced, and for that matter to try to obtain of the order of the 24"/sec velocity originally prescribed by Corning for this design.

As far as permissible errors are concerned, the only requirements available were those set down in a specification applying to a data transmission system which was being considered as a substitute for the flexible shaft in the present system. This specification permitted a 1 degree deviation between generator and receiver. Actually the question of error between input and output does not seem too critical for several reasons. Zero steady-state positional error is easily achieved due to the inherent integration of an hydraulic transmission. As a matter of fact, however, the needle is not at rest at any time during normal operation; it is always moving down at a rapid rate to expel the glass from the orifice and create the gob, then drawing back quickly to taper off the neck of the gob to facilitate the shearing of the gob from the parent melt. Then there is a fairly steady slow rise during which time the pressure distribution around the needle is such that atmospheric pressure outside the orifice and glass viscosity combine to prevent the melt from oozing out of the uncovered orifice while at the same time permitting new glass to flow into the void created by the retracting needle. The full revolution displacement of the typical cam selected and shown in Figure 4 illustrates this program.



FROM CORNING CAM DRAWING NO 15048-21 FIG. 4 LINEAR REPRESENTATION OF TYPICAL PROGRAM CAM.

To return to the error discussion, a single integration while yielding a zero steady-state positional error will result in a finite steady-state velocity error. The second integration, necessary to reduce the velocity error to zero, is not justified in terms of the stability problem which would be introduced by this second integration; if the velocity error proves objectionable, it can be, and is in fact, reduced by redesign of the program cam. The shape of these cams, in the first place, is wholly empirical - a cut and try proposition. Whether the needle is exactly reproducing the program cam (which it never can, or there would not be any feedback) is irrelevant provided the needle is displacing glass properly. Since the cam is altered until the needle does its job properly, and since the needle sees almost the same loading conditions each cycle (there are no fortuitous or random loads), the method of cam design will automatically compensate for any velocity (and acceleration) errors. This is all by way of saying that we do not require a system which will very accurately reproduce an input signal under a very great variety of load conditions. Of course, any possible improvement in the velocity constant, consistent with economic considerations will be exploited.

Production experience indicates that proper glass displacement requires that the needle motion be smooth and free of hunting or oscillation. We must therefore design for a system with a minimum of overshoot and oscillation and be certain that the band width is adequate to insure that such overshoot occurs at frequencies well above those frequencies which are components of the input signal. A Fourier analysis² of the typical cam indicates the following harmonic components:

$$f(t) = a_1 \sin \omega t + a_2 \sin 2\omega t + \dots a_n \sin n\omega t + b_0 + b_1 \cos \omega t + b_2 \cos 2\omega t + \dots b_n \cos n\omega t$$

$a_1 = -0.638$	$b_1 = + 0.035$
$a_2 = -0.290$	$b_2 = - 0.125$
$a_3 = - 0.314$	$b_3 = - 0.194$
$a_4 = + 0.114$	$b_4 = - 0.106$
$a_5 = - 0.117$	
$a_8 = + 0.096$	$b_8 = - 0.062$
$a_{12} = + 0.063$	$b_{12} = - 0.027$
$a_{16} = + 0.029$	$b_{16} = + 0.010$
$a_{20} = + 0.020$	

Thus the amplitude of the 20th harmonic is only 2% of the curve amplitude and may be considered negligible. For this particular curve the first harmonic has the frequency of 5 cpm or 0.083 cps; therefore, a bandwidth of 1.66 cps would be adequate. At the maximum operating frequency of 50 cpm, for a cam of this profile, (and it should be remembered that at higher operating speeds the profiles would be less severe since the gob weights and therefore strokes are less) a bandwidth of 16.6 cps would be adequate. A 20% overshoot of frequency components higher than the 20th harmonic should cause no trouble; therefore, an M_p of 1.3 is selected.

2. Physical

Past experience with feeders indicates that certain adjustments must be readily accessible to the mechanic overseeing production in order

that he may correctively influence gob formation during a production run. These adjustments include cycle-speed adjustment, program-cam phase adjustment, stroke adjustment and zero needle-position adjustment. On the existing feeder these adjustments are located at various places and on different levels of the feeder. As oftentimes an inter-related adjustment of several of these parameters is necessary, and since it is only by actually observing the effect in terms of altered gob shape that the efficacy of the adjustment can be determined, it would be desirable if these four adjustments could be amalgamated into an actual control panel located so that the mechanic could observe the gob formation while actually manipulating the variables.

The other adjustments, mentioned in Chapter II - the orientation of the needle with respect to the orifice and the vertical position of the sleeve - are more infrequently varied; therefore, centralized control of these adjustments is not essential. Similarly, the accessibility of the program cam, changed once per run, is not important.

As implied before, however, the size of the program cam is of considerable importance. Normal machine-shop procedure in the production of cams consists of first accurately reproducing the cam drawing on 0.040" steel stock in the template shop, which is well equipped for accurate work and staffed with precision machinists. This cam then goes to the general machine shop where it is used as a template in a profiling machine for producing the final product in heavy stock. Although the camming forces associated with the existing experimental hydraulic feeder are much reduced from those of the all-mechanical feeder, the inertias, spring forces and lifts are still too great to allow the use of the 0.040" template cam as the program cam. Currently the Corning hydraulic

feeder uses profiled $3/32$ " steel cams. It would therefore be of great advantage to design so as to use the 0.040" template cam, thus economizing on machine-shop use and avoiding the inherent inaccuracies of any additional reproduction process.

Furthermore it would be desirable to reduce the cam sizes from the $7/8$ ", $1\ 1/4$ " and $2\ 1/2$ " lift on a 4" base circle (dimensions now utilized) to dimensions such that the cam could be easily handled in ordinary interplant mailing envelopes and filed in minimum space. Thus the same envelope that contained an order for so many casseroles could also contain the cam to do the programming. The object then is to make the cam small enough to be easily made and handled, yet not so miniature as necessarily to complicate its manufacture or enhance unavoidable manufacturing deviations. Accordingly a cam dimension of 1" lift on a 2" base circle will be specified. Needle strokes in the range 0 to 8" will be provided by means other than by varying cam lift.

Essentially the index table, feeder and press are three separate pieces of equipment, and it is oftentimes desirable to be able to move them relative to each other. There should therefore be a minimum of interconnection between the units and such interconnections as are necessary must be flexible.

Durability, ease of maintenance, and minimization of complexity must be paramount considerations. This is to be a factory-located production device. It must therefore be capable of withstanding the normal abuse accorded machinery by production people and it must continue to function for interminable periods of round-the-clock production with no interruptions. It is of course obvious that any time delays incurred

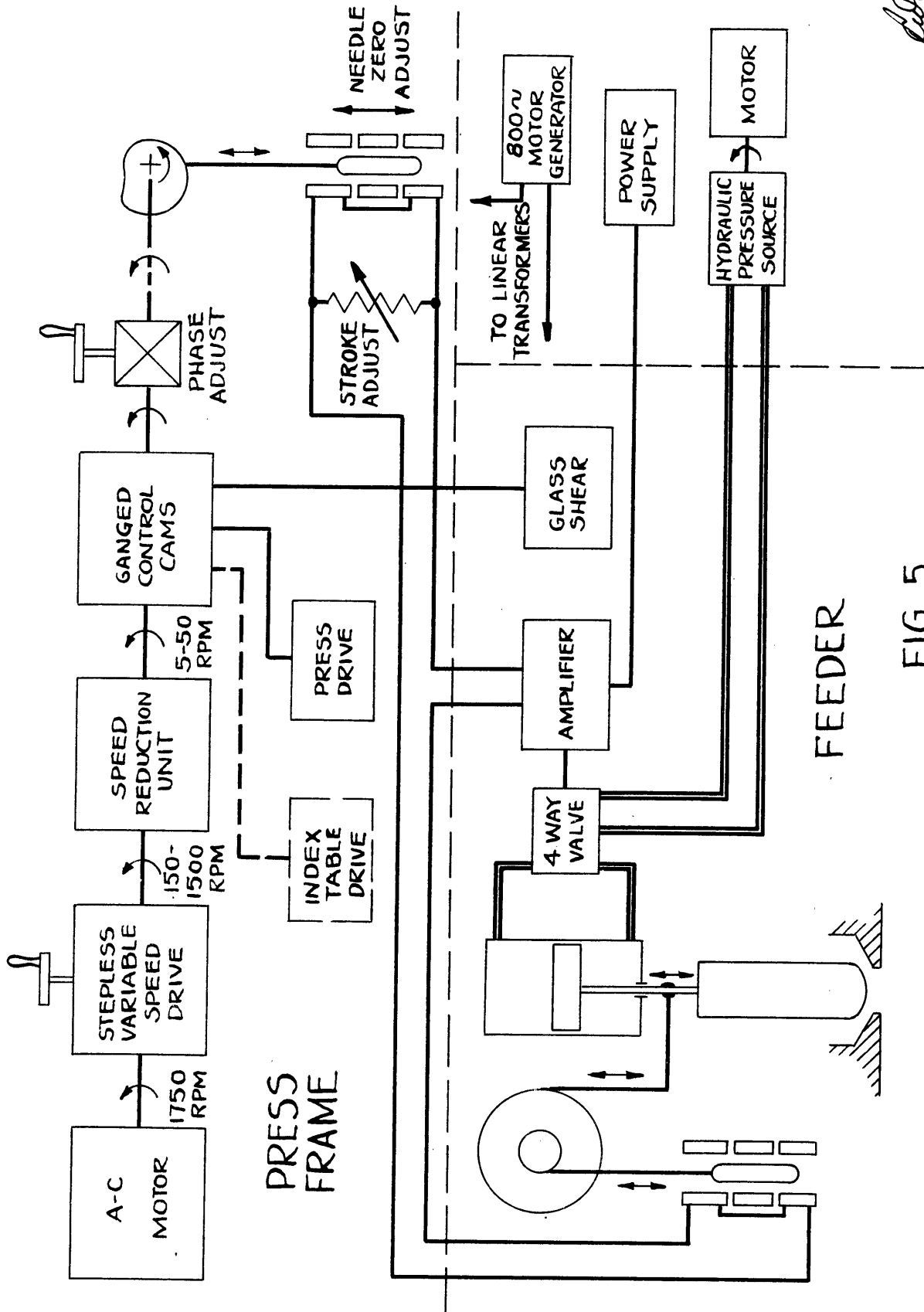
while processing a stream of molten glass can result in complete congestion of the system and lead to even greater complications in clearing equipment and re-establishing flow. Normal maintenance will be provided by regular shop mechanics who have little knowledge of, or experience in working on, intricate electronic, electrical, or hydraulic equipment. Components must therefore be capable of long-time uninterrupted service between general overhauls with but little except cursory inspection and lubrication between times.

3. Economic

Economic considerations dictate that commercially available components be used as much as possible to reduce engineering, manufacturing and testing costs to a minimum. Fortunately, system specifications are not so severe as to necessitate extensive component development work.

B. System Design

An investigation of commercially available power, control and measuring components, and various arrangements thereof, culminated in the system proposed in Figure 5. In this system the drive motor, together with its variable-speed drive and speed-reduction unit is mounted on the rear of the press frame. This location of the motor eliminates the need for the refrigeration required for the motor in the existing feeder because of the motor's proximity to the forehearth. A direct coupling from the speed-reduction unit drives the main synchronizing control cams which in turn operate air-operated valves. As can be seen from the figure for the proposed system, most of these air control lines



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FIG. 5
 PROPOSED FEEDER SYSTEM
 HYDRAULIC ORIFICE FLOW CONTROLLER

terminate on the press itself and so present no mobility problem. Some of the air lines do connect with the index table but the connection and mobility problem between control cams and index table in this design is at least no more complicated than it is in the experimental feeder where the drive motor is mounted on the feeder frame and drives the press and indexing control cams on the press frame by means of a flexible shaft. Ganging the feeder, press and index control cams in one group eliminates this flexible-shaft connection. The only operation of the feeder to be controlled by the ganged control cam is the glass shear which now involves one interconnecting air line. The possibility of operating this shear by a microswitch, electro-pneumatic valve combination could be investigated if the less flexible air line proves too objectionable.

The program cam is also located on the press frame and is driven off the control-cam shaft, with a mechanical phase-shifting device, possibly a differential, interposed. The program cam translates a follower which carries an electro-mechanical transducer which generates a voltage proportional to displacement. Provision is made for varying the length of the follower or the position of the electrical pick-up device so as to conform with the needle zero-adjust specifications. A manual adjustment which electrically activates the output of the transducer provides the stroke adjustment. Thus in one convenient location, from which the view of the feeder is unobstructed, and away from the uncomfortable ambient temperature of the forehearth, the four desired adjustments are grouped.

The electrical output of the program-cam transducer (suitably attenuated), is compared with the output of a similar device moving with

the needle. The deviation between these two signals is amplified and activates an hydraulic valve which meters hydraulic fluid from a suitable pressure source to the needle-ram cylinder.

C. Components

1. Hydraulic Flow Control Valve

A four-way control valve and hydraulic amplifier developed by the Dynamic Analysis and Control Laboratory (DACL) of the Massachusetts Institute of Technology for guided-missile applications and subsequently commercially marketed for industrial applications by Hydraulic Control Company of Roxbury, Massachusetts, seems uniquely adapted to this problem. The need is for a valve that will very accurately meter fluid at pressures adequate for the force levels involved and in quantities to provide the requisite velocities in response to an amplified electrical signal and with a speed of response adequate to provide the bandwidth specified. An analysis of the valve in the contemplated system indicates its acceptability as shown below.

The following data is abstracted from DACL Report #50¹. The model MX has the following characteristics:

Pressure (psi)	Maximum Flow (in ³ /sec)	Maximum Power (hp)
3000	33.8	5
1000	19.2	1

Let p_1 = the source pressure (psi)
 Δp_2 = the pressure drop per port (psi)
 Δp = the pressure drop across the load (psi)

A schematic hydraulic circuit is shown in Figure 6. Tracing the circuit shows that

$$p_1 = \Delta P + 2\Delta p_2. \quad (1)$$

Assume all other head losses including pipe friction and leakage are negligible. For the short pipe lengths involved the pipe frictional head is very small compared to the pressure heads utilized; a subsequent analysis of leakage will prove this also to be negligible.

Now for an orifice, which a metering-valve port essentially is,

$$Q = Ca \sqrt{\frac{\Delta p_2}{\rho}}$$

where Q = flow (in³/sec)

a = port area (in²)

C = product of the flow and velocity constants for the orifice

ρ = mass density of fluid
 $\left(\frac{\text{\#sec}^2}{\text{ft}^4}\right)$

Experimental evidence shows that C is fairly independent of flow and pressure drop; a is held constant at its maximum value. Combining constants

$$Q = K_f \sqrt{\Delta p_2}$$

where K_f is an overall flow coefficient equal to $Ca\sqrt{\rho}$. Q_{\max} for a given pressure drop is given. For Q_{\max} there is no pressure drop across the load; therefore, the source pressure divides equally between the ports. For the 3000 psi supply pressure

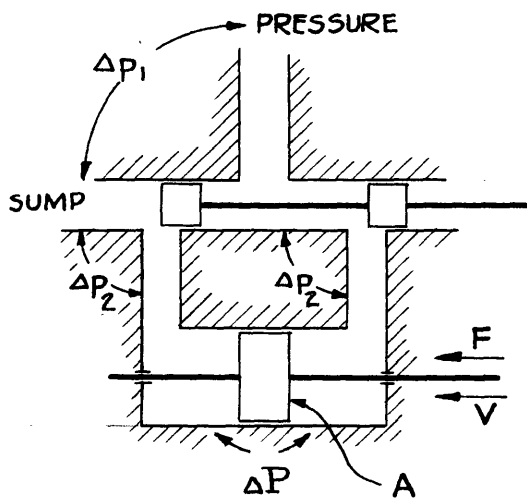


FIG. 6
SCHEMATIC HYDRAULIC CIRCUIT

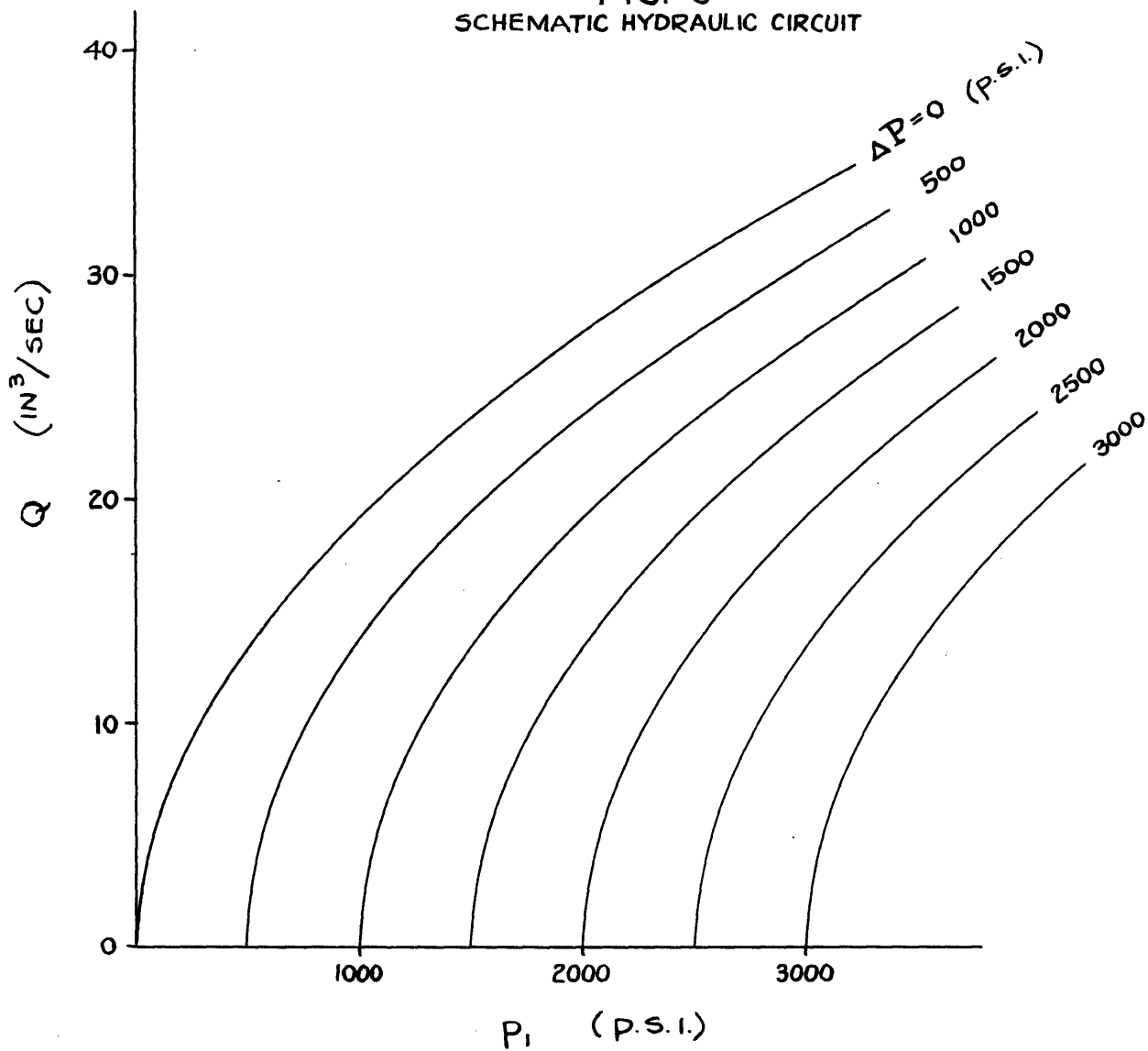


FIG. 7
MX VALVE
FLOW CURVES

Eden

$$(K_f)_{3000 \text{ psi}} = \frac{Q}{\sqrt{\Delta p_2}} = \frac{33.8}{1500} = 0.875$$

For the 1000 psi supply pressure

$$(K_f)_{1000 \text{ psi}} = \frac{Q}{\sqrt{\Delta p_2}} = \frac{19.2}{\sqrt{500}} = 0.858$$

which in fact illustrates that K_f is substantially constant. Take as the value for K_f the average of the values calculated above; then

$$Q = 0.865 \sqrt{\Delta p_2} \quad (2)$$

Using equations (1) and (2), a family of curves (parabolas) can be plotted of Q vs p_1 for different values of ΔP ; see Figure 7. From these curves, for given values of supply pressure (a known variable) the curves of Q and delivered hp vs ΔP (Figure 8) can be plotted where

$$\text{hp} = Q \Delta P / 6600$$

Flow and power for different ΔP and p_1 can be easily read off. It is seen that maximum power occurs where ΔP is equal to $2/3 \times p_1$.

As a check of the reliability of these curves for estimating power output of the valve under various source and loading pressures, it is seen that the calculated horsepower curves for 1000 and 3000 psi supply pressure maximize at 1.1 and 5.6 compared with the published output of 1 and 5 hp at these supply pressures. The discrepancy is probably attributable to the simplifying assumptions of the analysis, and to the fact that the published figures are both conservative, and of single-place accuracy.

Since for this feeder system force is proportional to velocity, and since therefore force and velocity maximize simultaneously requiring maximum power, we shall use as design values the values of Q and ΔP occurring at maximum power to most efficiently apply the valve.

Applying this reasoning and writing the equations which define the force and velocity in terms of system constants:

$$\begin{array}{ll}
 F = A\Delta P & \text{where } F = \text{force on ram (\#)} \\
 Q = VA & A = \text{area of ram (in}^2\text{)} \\
 F = \beta V & V = \text{velocity of ram (in/sec)} \\
 & \beta = \text{coefficient of viscous} \\
 & \quad \text{friction (\#in/sec)}
 \end{array}$$

Solving these equations simultaneously:

$$V^2 = \frac{Q\Delta P}{\beta}$$

Using values from the curves shown in Figure 8 at maximum power:

	$\beta = 20.5$			$\beta = 40$		
	calculated value			conservative engineering assumption		
p_1	1000	1500	2000	1000	1500	2000
V_{\max}	18.9	25.8	32.2	13.6	18.4	23.1
A	0.582	0.527	0.498	0.81	0.74	0.693
F_{\max}	389	527	664	540	740	925

Even with the very conservative over-estimate of β , it is seen that the valve is capable of meeting specifications. At 1500 psi supply pressure the velocity maximum at maximum force (18.4 in/sec) is 3 times

John

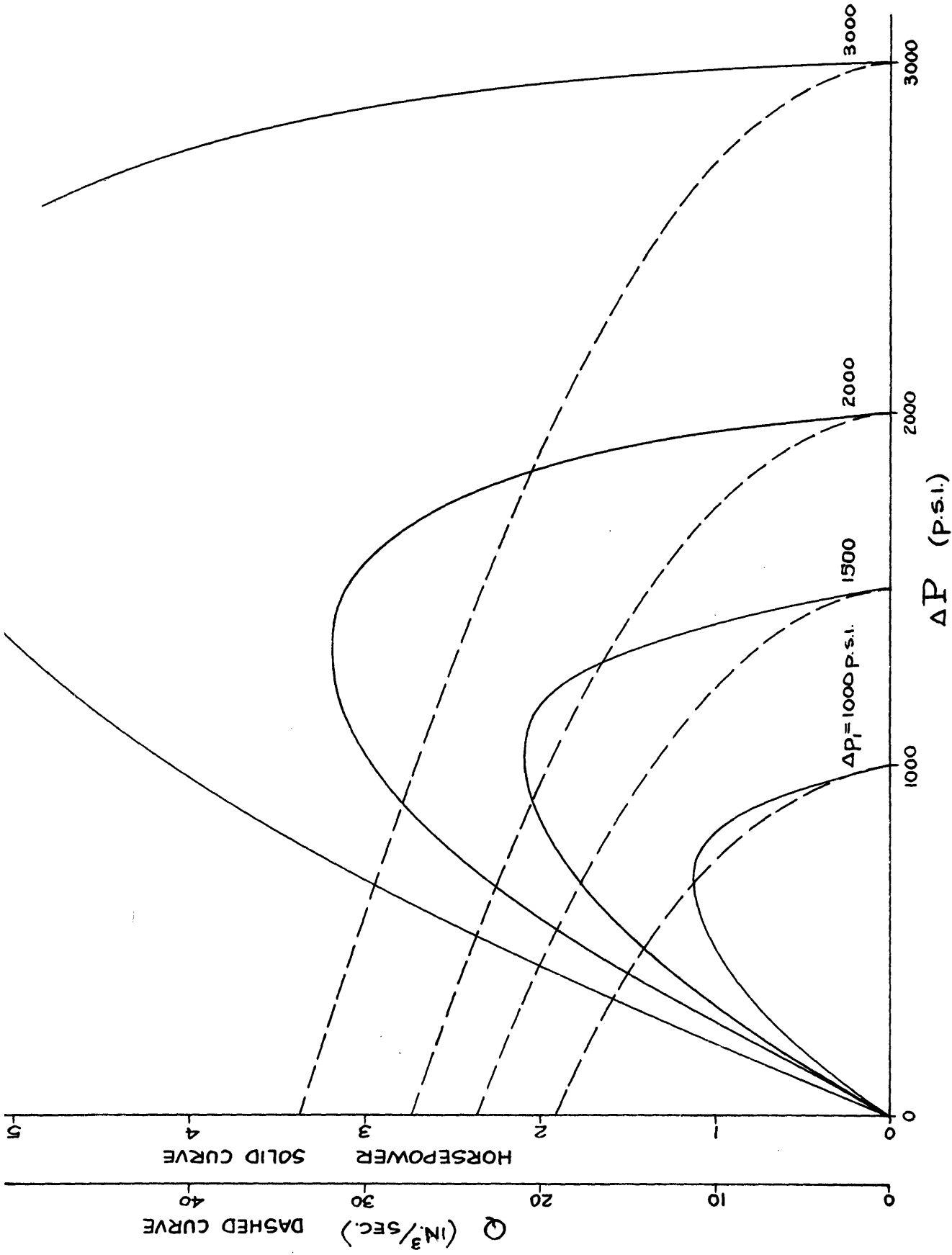


FIG. 8
MX VALVE POWER AND FLOW CURVES

that required by the present system (5.65 in/sec). And by going to 2000 psi supply pressure the 24"/sec velocity figure specified, exaggerated though it is, is almost achieved. The valve and the rest of the system are engineered for up to 3000 psi, so the margin of safety is considered adequate for any conceivable force or velocity demands.

Figure 9 is a photograph of the MX valve and amplifier. In operation the torque motor translates the pilot piston in proportion to the amplified electrical error signal. The movement of the pilot piston from neutral position ports oil to cylinders at either end of the power piston. Thus oil flow metered by the pilot piston locates the power piston, which in turn meters hydraulic fluid from the source to the ram at high pressure and back to the sump at exhaust pressure. The dynamics of the hydraulic-amplifier-valve combination will be more completely described in the section devoted to the dynamic analysis of the loop.

The effect of leakage on the dynamic properties of the system will indicate that considerable leakage flow can be tolerated. This justifies increasing the clearances in the valve, thus lowering manufacturing cost, and even more important, minimizing the deleterious effects of dirt in the system. Refer to the discussion of leakage in the section on dynamic analysis.

A recent analytical investigation, supported by experimental evidence, of valve stroking forces indicates that the force necessary to move the pilot piston can be materially reduced by a redesign of the pilot piston. This information is contained in Project Meteor Report #65⁴ and should be applied so as to ameliorate the specifications required of the torque motor described in the next section.

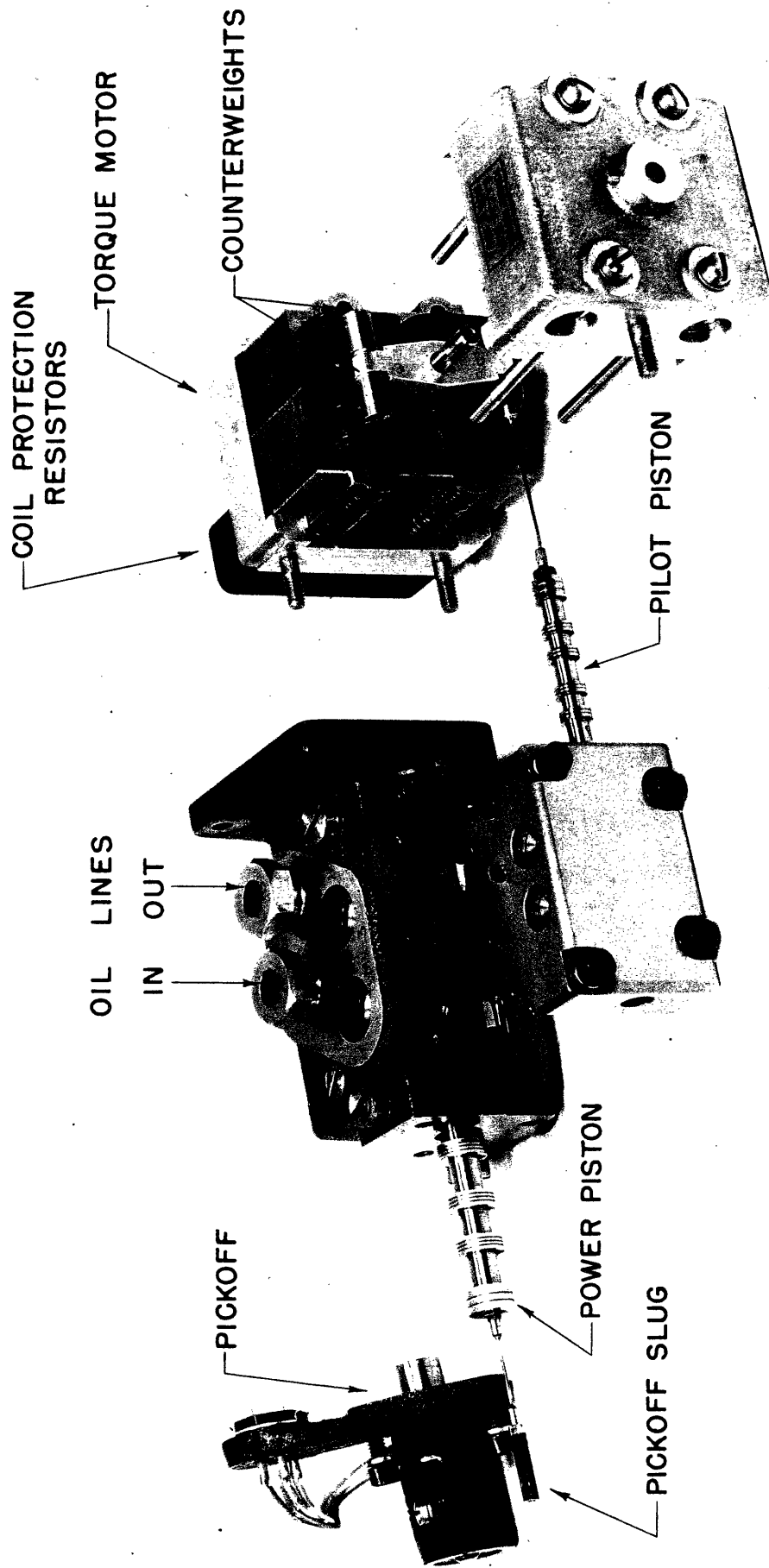


FIG. 9 SERVO VALVE PARTIALLY DISASSEMBLED

2. Torque Motor and Error Amplification

As currently manufactured the MX valve is stroked by a direct-current torque motor, Model No. 35 manufactured by Transonics Corporation of Bedford, Massachusetts. For reasons enumerated later the error signal is the envelope of an amplitude-modulated 800 cps carrier. Current practice with this combination of modulated-carrier error signal and d-c torque motor involves a-c amplifying the error signal, demodulation and filtering, cascading compensating networks, then amplifying the error as a d-c signal to drive the torque motor; this combination is shown in Figure 10a. The promising development of an a-c torque motor by a group at the Dynamic Analysis and Control Laboratory under the direction of Dr. R.H. Frazier offers an alternative arrangement as shown in Figure 10b, which reduces the electronic components appreciably. The most noteworthy improvement in this arrangement is the elimination of the d-c amplifiers which are inclined to drift, thus requiring regular inspection and adjustment by competent technicians.

The a-c torque motor development program has advanced to the point where successful operation is assured. Physically the unit is very similar to the present d-c Model 35 motor shown in Figure 9, the alterations consisting primarily of a laminated armature and different field coils.

3. Auxiliary Components

a. The a-c induction motor, variable-speed drive, speed-reduction unit, synchronizing control cam, and mechanical phase-adjust device as used in the existing feeder are completely applicable to the proposed design.

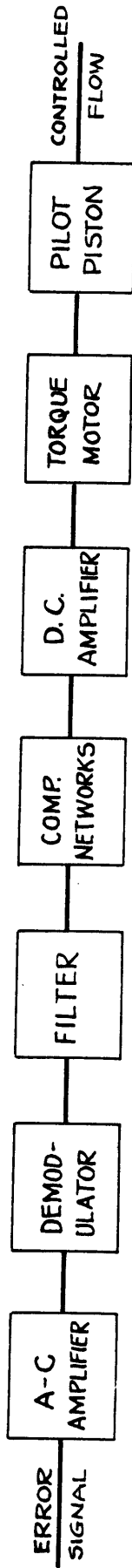


FIG. 10a
SYSTEM WITH
D.C. TORQUE MOTOR

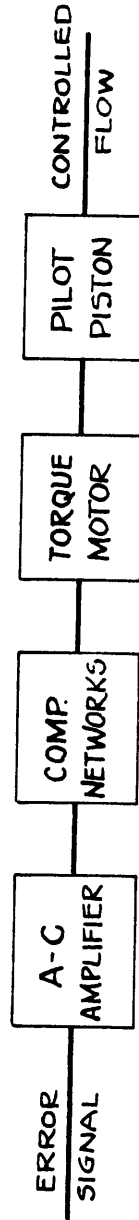
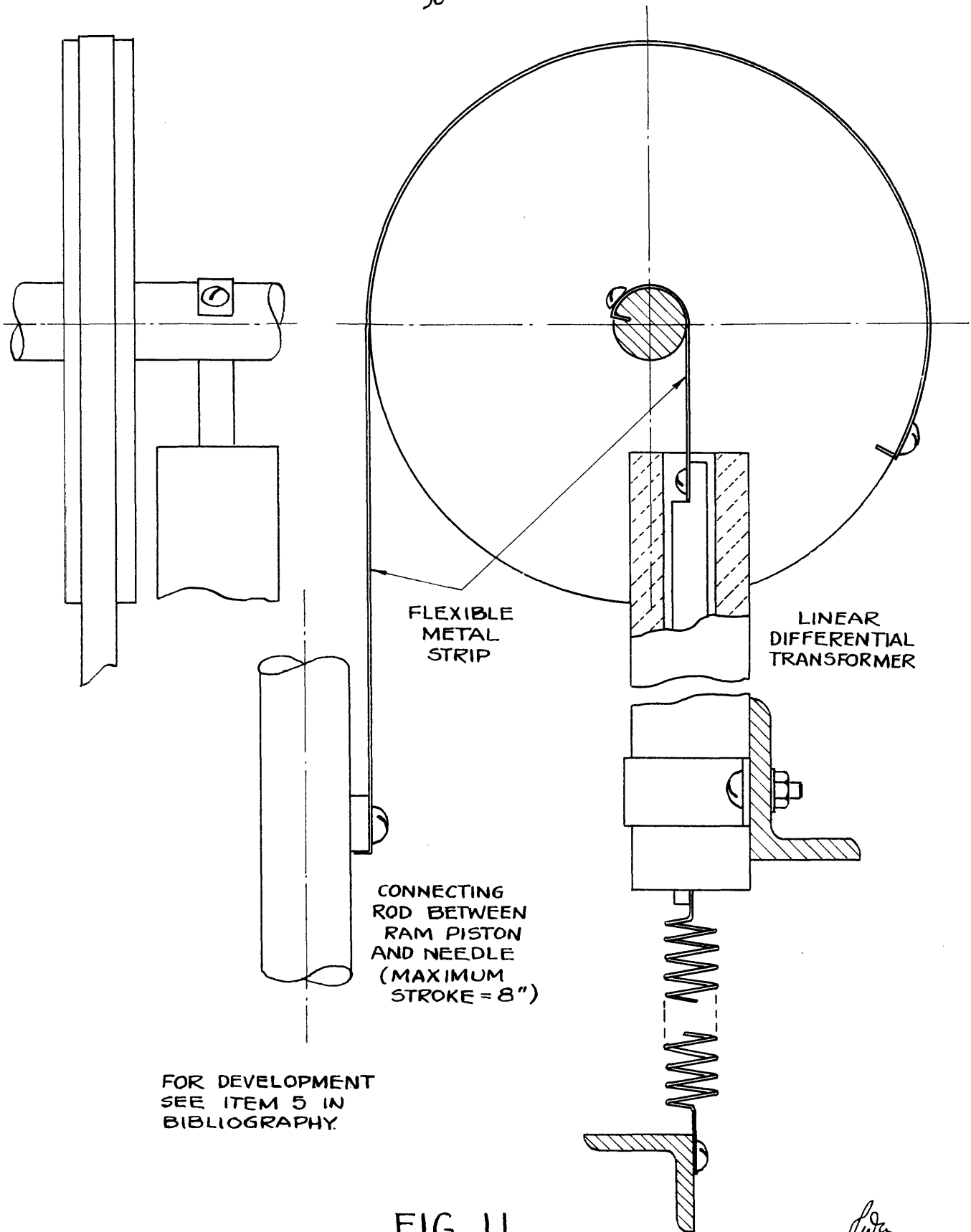


FIG. 10b
SYSTEM WITH
A.C. TORQUE MOTOR

Eden

b. The transducers which convert the position of the follower and needle into proportional electrical signals are so-called linear differential transformers.³ A core of magnetic material connected to the moving unit varies the coupling between three coils arranged coaxially about the core. There is no contact and therefore no friction between the coil and core, and eccentricity or transverse motion between coil and core do not affect accuracy. The center winding is excited with 800 cps current from a motor-generator set; therefore the output of the outside secondary windings, wired in series opposition, is a carrier of 800 cps modulated according to the position of the core. The 800 cycle error signal is a high enough frequency to prevent excessive and objectionable oscillation of the torque-motor armature. However the low-amplitude oscillation of the modulated signal does serve effectively as built-in "dither", a term applied to an enforced and desired high-frequency, low-amplitude oscillation of the hydraulic components of the valve, which reduces sticking and static friction effect. The higher-frequency carrier also favorably increases the sensitivity of the transformers. Excitation of 800 cps is selected since reliable motor-generator sets of this frequency which require an absolute minimum of maintenance attention are commercially available.

The particular model of linear transformer selected is type 10-LC manufactured by Schaevitz Engineering of Camden, New Jersey. This transformer is wound on a ceramic coil-form and can be successfully operated up to 300°C. The standard manufacturing accuracy of linearity (1%) between core motion and voltage output should be adequate. The sensitivity of this model is 0.0008 V/0.001"/ volt input at 60 cps. Operating at 800 cps and with the maximum input voltage of 10 V the sensitivity



FOR DEVELOPMENT
SEE ITEM 5 IN
BIBLIOGRAPHY.

FIG. 11
PROPOSED FEED-BACK LINK

W. A.

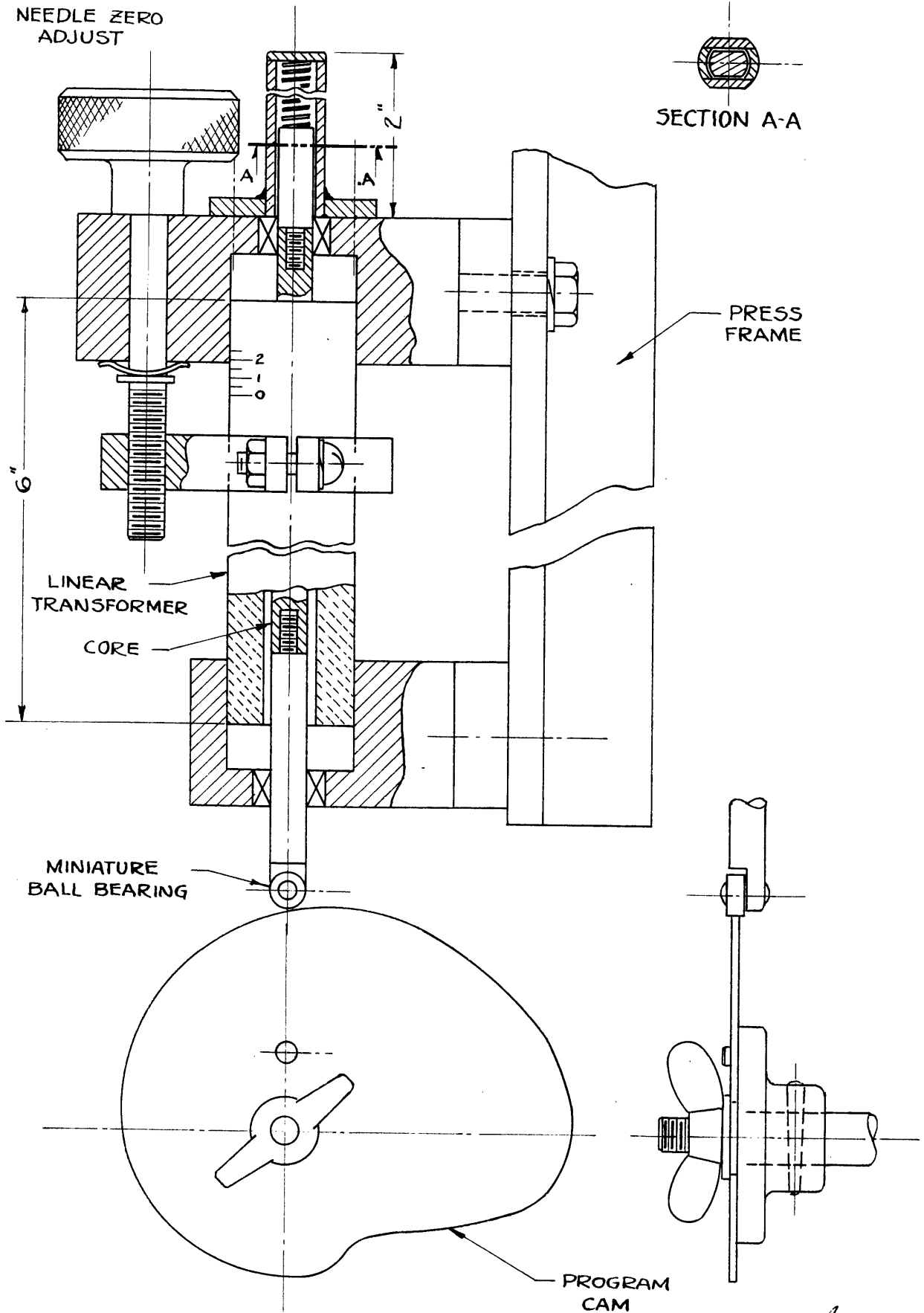


FIG. 12
INPUT AND NEEDLE
ZERO ADJUST

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will be about 10 volts/inch. The 0.500" - 0 - 0.500" or 0 - 1.000" range is based on the desired cam lift of 1.00". Similar transformers will be used for both input-measuring and output-measuring devices. The 0 - 8.00" stroke of the needle will be reduced to the 0 - 1.00" range of the transformer by a device similar to that shown in Figure 11.

A sketch of the arrangement of the program cam, follower and input linear transformer, showing the zero-adjust control is shown in Figure 12.

4. The Needle Drive and Supporting Framework

The design of the ram, and the needle and sleeve support and adjustment, together with the overall supporting framework and the selection of a commercial hydraulic power source, are included in a Bachelor's thesis entitled, "The Design of a Hydraulic Orifice Flow Controller".⁵

D. Dynamic Analysis

Figure 13a is a block diagram representation of the servo-mechanisms components of Figure 5 where:

K_1 = the transfer function for a linear differential transformer

K_2 = the transfer function for an a-c amplifier

K_3 = the transfer function for the a-c torque motor and driving amplifier

$\frac{K_4}{s}$ = the transfer function for the pilot piston-power piston combination

$\frac{K_5}{s \left(\frac{s^2}{\omega_n^2} + \frac{2\zeta s}{\omega_n} + 1 \right)}$ = the transfer function for the ram and load

y = the input from the program cam

Y = the output or the needle displacement relative to the orifice

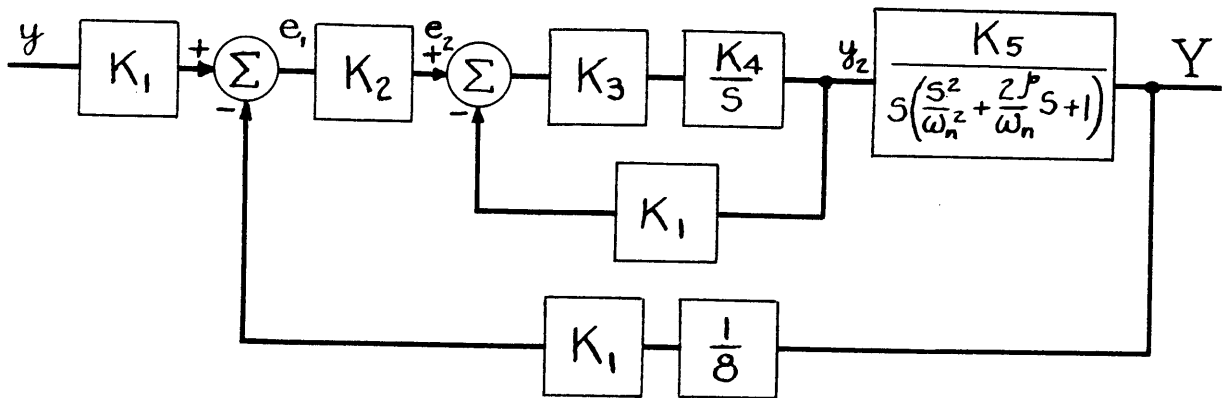


FIG. 13a
PROPOSED SYSTEM
BLOCK DIAGRAM.

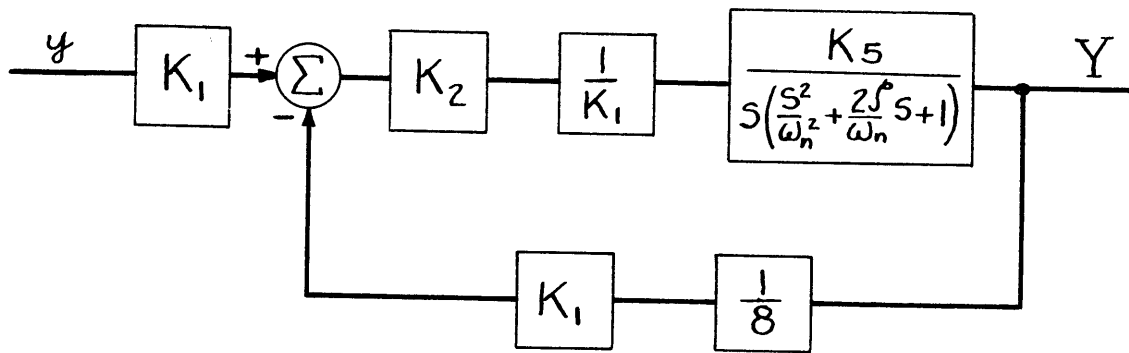


FIG. 13b
SIMPLIFICATION OF
FIG. 13a.

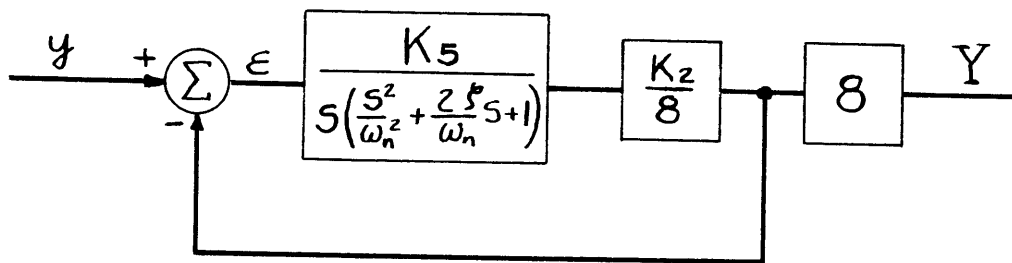


FIG. 13c
REARRANGEMENT OF
FIG. 13b.

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These transfer functions represent simplifications in that 1) the a-c torque motor is not a pure gain, but has some dynamic properties. Its delay of less than 1.7 milliseconds¹, however, is so small compared with the delays of the rest of the system that approximation as a pure gain is valid. Similarly the pilot piston-power piston combination is not a pure integration, but also includes a small delay. Again this delay, which is manifested by a computed natural frequency of over 1000 cps¹, is negligible for the system under consideration.

The integration of the power piston, together with the integration of the ram piston, result in two poles at the origin of the s plane and an inherently unstable system. System stability could be achieved by elaborate compensation while maintaining the double integration. However, the integration of the power piston is more easily eliminated (minimized) because the MX valve is made with a linear differential transformer pickoff connected to the power piston. The output of this pickoff is fed back in an internal loop as shown in Figure 13a. Writing the transfer function for the internal loop:

$$y_2 = \left(\frac{K_3 K_4}{s} \right) e_2 - \left(\frac{K_1 K_3 K_4}{s} \right) y_2$$

rearranging:

$$\frac{y_2}{e_2} = \frac{1}{K_1} \cdot \frac{1}{\left(\frac{s}{K_1 K_3 K_4} + 1 \right)}$$

If the product $K_1 K_3 K_4$ can be made large enough, the delay of the transfer function will be of the same small order as the delays of K_3 and K_4/s , and can be ignored. The transfer function, over the frequency bandwidth of

concern, will then approximate the pure gain constant, $1/K_1$. Figure 13b is Figure 13a redrawn to include this simplification. Figure 13c is a second revision of the system block diagram to a form representing unity negative feedback.

1. Evaluation of Transfer Functions

$$K_1 = 10 \text{ volts/inch}$$

$$K_2 = \text{gain constant (value to be determined)}$$

K_3 The torque motor is currently being developed; therefore this constant is not as yet determined. Its value does not influence the overall system gain, however, since the internal-loop feedback effectively replaces the K_3K_4/s feed-through transfer function with the feedback constant K_1 . The only requirement is that K_3 must not be very small (as shown below).

$K_4/s = 2320/s$ at 1500 psi source pressure and when the ram is delivering maximum power, i.e., when $\Delta P = 1000$ psi, $\Delta p_2 = 250$ psi. The pilot-piston port sensitivity = 0.067 gpm/mil = 0.258 in³/sec/mil. Power-piston diameter = 0.375 in, its area = 0.111 in²; therefore power-piston velocity (Q/A) = 0.258/0.111 = 2.32 in/sec/mil = 2320/sec.

$$\frac{K_5}{s \left(\frac{s^2}{\omega_n^2} + \frac{2\zeta}{\omega_n} s + 1 \right)} = \frac{Y}{Y_2}$$

Combination of the above demonstrates that the product $K_1K_3K_4(=23200K_3)$ can be very large, and the time constant or delay is very small, justifying the simplification of Figure 13b.

The last transfer function will be justified and its parameters defined in the following paragraphs.

a) Literal Evaluation

Let p equal source pressure ($\#/in^2$)

V volume of oil under compression in the system between valve and ram piston (in^3)

B bulk modulus of oil ($\#/in^2$)

M mass of ram, needle and connecting components $\frac{\#sec^2}{in}$

β viscous friction coefficient (of molten glass) $\frac{\#sec}{in}$

A ram area (in^2)

F force on ram piston ($\#$)

L leakage coefficient for the system

k_2 flow sensitivity of power piston (gpm/mil) $(\frac{in^3}{sec} \frac{1}{mil})$

y_2 displacement of power piston (in)

Y displacement of needle (in)

The flow relation for the system is

$$Q_V = Q_R + Q_L + Q_C$$

where

$$Q_V = \text{valve flow} = k_2 y_2$$

$$Q_L = \text{leakage flow} = Lp$$

$$Q_R = \text{ram flow} = \frac{dY}{dt} A$$

$$Q_C = \text{compressibility flow} = \frac{V}{B} \frac{dp}{dt} \quad \text{from } p = \frac{B\Delta V}{V}$$

$$Q_C = \frac{dV}{dt} = \frac{V}{B} \frac{dp}{dt}$$

Substituting into the flow relation:

$$k_2 y_2 = \frac{dY}{dt} A + Lp + \frac{V}{B} \frac{dp}{dt}$$

For the ram at maximum power:

$$F = \frac{2}{3}pA = \beta \frac{dY}{dt} + M \frac{d^2Y}{dt^2} = \beta \dot{Y} + M\ddot{Y}$$

$$p = \frac{3}{2} \frac{1}{A} (\beta \dot{Y} + M\ddot{Y})$$

$$\frac{dp}{dt} = \frac{3}{2} \frac{1}{A} (\beta \ddot{Y} + M\dddot{Y})$$

Again substituting into the flow relation:

$$k_2 y_2 = YA + \frac{3}{2} \frac{L}{A} (\beta \dot{Y} + M\ddot{Y}) + \frac{3}{2} \frac{V}{AB} (\beta \ddot{Y} + M\dddot{Y})$$

Rearranging and in terms of the La Place variable:

$$\frac{Y}{y_2} = \frac{K_5}{s \left[\frac{s^2}{\omega_n^2} + \frac{2\zeta s}{\omega_n} + 1 \right]}$$

where

$$\omega_n = \text{natural frequency} = \sqrt{\frac{2A^2B + 3L\beta B}{3VM}} \quad (\text{rad/sec})$$

$$\zeta = \frac{\text{system damping}}{\text{critical damping}} = \frac{3}{2} \frac{(BLM + V\beta)}{[3VMB(2A^2 + 3L\beta)]^{1/2}}$$

$$K_5 = \frac{2k_2A}{2A^2 + 3L\beta} \quad (1/\text{sec})$$

b) Numerical Evaluation

$$B = 0.27 \times 10^6 \text{ #/in}^2$$

Assume:

$$p = 1500 \text{ #/in}^2 \text{ from the investigation of the MX valve.}$$

At this pressure the velocity maximum of 18.4 in/sec should be adequate.

then

$$A = 0.74 \text{ in}^2$$

$$k_2 = 0.10 \text{ gpm/mil} = 385 \text{ in}^2/\text{sec at 250 psi pressure drop per port, which occurs at source pressure of 1500 psi and maximum power.}$$

Assume:

$$V = 0.74 (8) + \frac{(3/16)^2 (\pi/4) (48)}{\text{ram cylinder}} = 7.25 \text{ in}^3 \text{ high pressure line}$$

$$M = 300 \# \frac{1 \text{ sec}^2}{386 \text{ in}} = 0.78 \# \text{ sec}^2/\text{in}$$

$$\beta = 40 \text{ from analysis of viscous friction coefficient}$$

$$L = Q_L/P$$

$$Q_L = Q_{L1} + 2Q_{L2} \quad \text{where}$$

$$Q_{L1} = \text{leakage past ram piston (in}^3/\text{sec)}$$

$$Q_{L2} = \text{leakage past valve power piston (in}^3/\text{sec)}$$

Refer to Figure 14 which illustrates the land and port construction of the power piston and schematically represents the ram piston. For neutral position of the power piston, (see Figure 14a) the pressure is equalized across the ram piston, and the only flow is leakage flow from the high pressure port to the two exhaust ports. Each leakage path consists of two orifices in series, and assuming geometrically similar orifices, the supply pressure will divide equally across the two orifices ($\Delta p_2 = 750 \text{ psi}$). In actual manufacture the power piston is initially

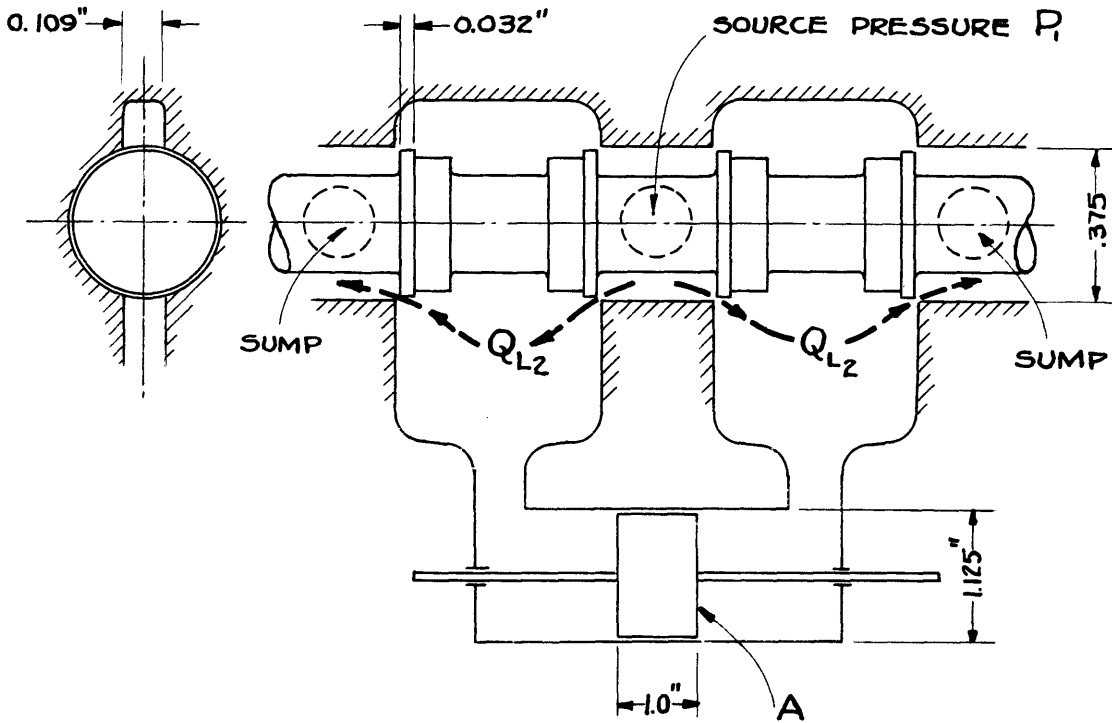


FIG. 14a
POWER PISTON OF MX VALVE
NEUTRAL POSITION

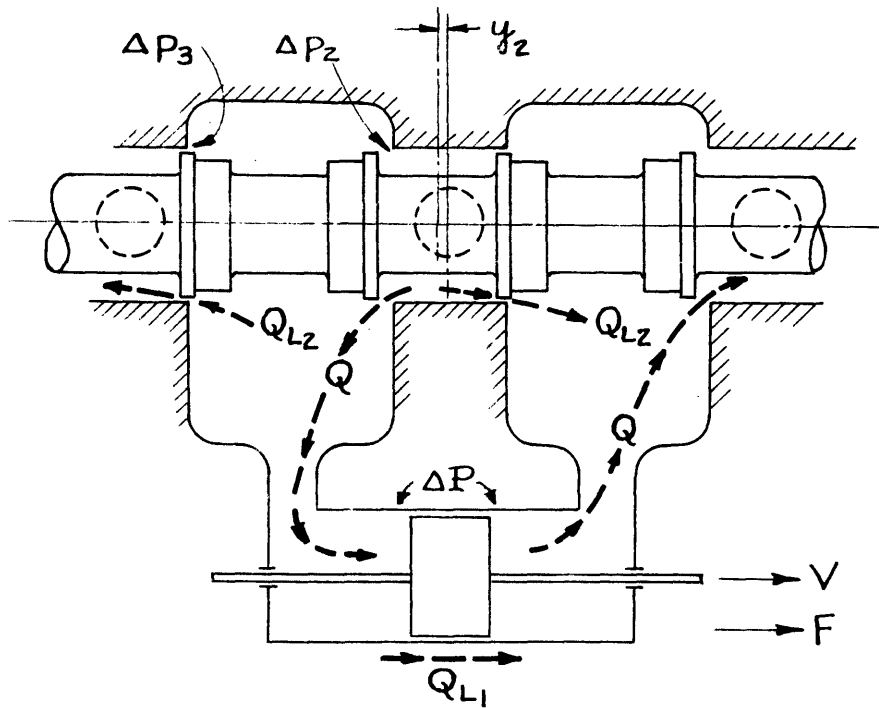


FIG. 14 b
POWER PISTON OF MX VALVE
DISPLACED POSITION

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made with over-lapped lands, which are subsequently ground narrower according to an experimental procedure to provide a linear flow vs displacement curve. This process results in underlapped lands with orifices of 2 to 3 times the piston radial clearance with the piston in neutral position.¹ For a 0.0001" radial clearance and an average underlap of 0.00025" then the effective port orifice length is $10^{-4} \sqrt{1^2 + (2.5)^2} = 0.00027"$. For each orifice the total port width is 0.218", therefore, the leakage area (A) is $0.00027 \times 0.218 = 0.000059 \text{ in}^2$. The flow coefficient (C) for a square-edged orifice is approximately 0.7. Applying the orifice equation, for each path,

$$Q_{L2} = CA \sqrt{\frac{\Delta P_2}{\rho}} = 0.039 \text{ in}^3/\text{sec}$$

(laminar leakage across the piston land is very small)

and since there are two leakage paths in the valve and no leakage across the ram piston, for neutral position of the power piston:

$$Q_L = 0.078 \text{ in}^3/\text{sec}$$

As the power piston is displaced from neutral position to provide flow (Figure 14b) to the ram, the leakage paths are as shown. Two leakage paths exist in the valve, and one exists across the ram piston. The leakage flow across the ram piston is always laminar flow since the radial clearance is very small compared to the piston length. Similarly, for appreciable (over 0.001") displacements of the power piston, the two leakage paths in the valve have laminar flow. The criterion applied is that laminar flow results if the path length is ten or more times the clearance dimension.

For laminar flow:

$$Q_L = w \left[\frac{\Delta p b^3}{12 \mu l} + \frac{V b}{2} \right] \quad \text{where}$$

p is the pressure drop across the length of the path ($\#/in^2$)
 b is the radial clearance (in)
 w is the width of the path (in)
 μ is the oil viscosity ($\#/sec/in^2$)
 l is the length of the path (in)
 V is the velocity of the ram piston relative to the ram cylinder (in/sec) (For the power piston V is effectively zero.)

The positive sign indicates that the pressure difference and velocity are in the same direction, both contributing to leakage.

The oil viscosity is taken as 80 SSU or 2.0×10^{-6} $\#/sec/in^2$, which is the viscosity of Univis J-58 at a temperature of approximately 155 F. This is a conservative estimate of viscosity, and if anything is on the low side (i.e., temperature assumption is high); actual experience with the Corning experimental feeder indicates that, while ambient temperatures in the vicinity of the feeder and forehearth are high, there is sufficient circulation of hydraulic fluid to keep its temperature down and therefore viscosity up. The possible eventual need for oil cooling, however, should not be completely overlooked.

The width of the major laminar leakage path for the power piston (for displacements small compared with the piston land width of 0.032") is approximately the total port width (0.218"). As the displacement increases to 0.035" the width of the leakage path increases, of course, to the full piston circumference. There is some sort of transition in path width for intermediate displacements. However leakage across each port can be calculated as the sum of two separate laminar flows: one flow, the flow across a path of total port width but of short length - a function of displacement; the second flow, the path across the piston circumference minus the total port width for a path length equal to the land width of 0.032".

Assuming:	for the power piston	for the ram piston
	$b = 0.0001"$	$b = 0.001"$
	$l_1 = y_2 - 0.00025"$ where	$l = 1.0"$
	0.00025" is underlap	
	$l_2 = 0.032" =$ piston land	
	width	
	$w_1 = 0.218"$	$w = \pi(1.125) = 3.54"$
	$w_2 = \pi(0.375) - 0.218 =$	
	0.96"	

where

y_2 = the displacement of the power piston from neutral position

Q = the work-producing flow based on power-piston flow sensitivity of 0.10 gpm/mil = $385y_2$ in³/sec

A = the area of the ram = 0.74 in²

V = the velocity of the ram = $Q/A = 385y_2/0.74 = 520y_2$

F = the viscous force exerted by the glass on the ram

$$\text{piston} = \beta V = 40 V = 20800 y_2 \quad (\#)$$

ΔP = the pressure drop across the ram = F/A =

$$20800y_2/0.74 = 28200y_2 \quad (\text{psi})$$

p_1 = the source pressure = 1500 psi

Δp_2 = the pressure drop across a work-producing orifice =

$$(p_1 - \Delta P)/2 \quad (\text{psi})$$

Δp_3 = the pressure drop across each of two leakage paths

$$\text{in the valve} = p_1 - \Delta p_2 = (p_1 + \Delta P)/2 =$$

$$750 + \Delta P/2 \quad (\text{psi})$$

$$Q_{L1} = \frac{P_3 b^3 w}{12\mu l} + \frac{wVb}{2} = 1.48 \times 10^{-4} \Delta P + 1.77 \times 10^{-3} V$$

$$Q_{L2} = \frac{P_3 b^3}{12\mu} \left[\frac{w_1}{l_1} + \frac{w_2}{l_2} \right] = \Delta p_3 \left[\frac{0.91 \times 10^{-8}}{y_2 - 0.00025} + 0.125 \times 10^{-5} \right]$$

$$Q_L = Q_{L1} + 2Q_{L2}.$$

TABLE I

y_2	Q	V	P	P_3	Q_{L1}	Q_{L2}	Q_L
0.0015	0.577	0.78	42	770	0.006	0.0065	0.019
0.003	1.15	1.56	84	790	0.015	0.0037	0.022
0.005	1.93	2.60	140	820	0.026	0.0026	0.031
0.007	2.70	3.64	200	850	0.039	0.0022	0.043
0.010	3.85	5.20	280	890	0.050	0.0019	0.054
0.020	7.70	10.4	560	1030	0.101	0.0018	0.104
0.035	13.6	18.4	1000	1250	0.181	0.0015	0.184

Since we are considering the dynamic system when it is delivering maximum horsepower, $Q_L = 0.184 (\approx 0.2)$ for a ram-piston clearance of 0.001"

$$L = \frac{Q_L}{P} = \frac{0.2}{1500} = 1.33 \times 10^{-3}$$

Substituting those numerical values, at maximum power,

$$\omega_n = 133 \text{ rad/sec}$$

$$\zeta^p = 0.212$$

Of the physical parameters which influence ω_n and ζ^p ,

A is determined by the available power and required ram velocity

B is a physical constant for the fluid

β is determined by the analysis of forces in the glass medium

M depends on the weight of the ceramic needle and the strength requirements of the supporting members

V depends on the displacement of the ram cylinder and on the distance between valve and ram

L depends on the clearances in the valve and ram.

Since the natural frequency varies as $(VM)^{-1/2}$ the moving mass and the volume of oil under compression are kept as low as possible so as to maintain adequate bandwidth.

Numerical substitution indicates that leakage has little effect on natural frequency and considerable effect on the damping ratio.

Q_L (in ³ /sec)	ω_n rad/sec	ξ
0.2	133	0.21
0.6	135	0.245
1.3	138	0.300

This increased leakage could be manifested by increased clearance either at the ram or at the power piston. Reference to the equations pertaining to leakage and to the table showing the magnitude of the separate leakage paths, particularly at higher power levels, suggests increasing the power piston clearance from the 0.0001" now permitted. The increased clearance will not only reduce machining costs of original equipment but will also contribute considerably to continued, uninterrupted service, since the lodging of small dirt particles in small-clearance spaces deleteriously affects valve performance.¹

The table below gives leakage flow at 1500 psi supply and 0.001" ram-piston clearance, for valve power piston clearance of 0.005" and 0.001" as compared with the previous calculations for power piston clearance of 0.0001". At the increased clearances laminar flow across the power-piston landwidth of 0.032" has been added to the orifice flow for neutral position ($y_2 = 0$).

TABLE II

Power Piston Clearance	0.0001"				0.0005"		0.001"		
	y_2	Q	Q_{L1}	Q_{L2}	Q_L	Q_{L2}	Q_L	Q_{L2}	Q_L
	0	0	0		0.078		0.388		1.045
	0.005	1.93	0.026	0.0026	0.031	0.325	0.676		
	0.010	3.85	0.050	0.0019	0.054	0.238	0.526	1.9	3.85
	0.020	7.70	0.101	0.0018	0.104	2.225	0.551	1.8	3.70
	0.035	13.6	0.181	0.0015	0.184	0.187	0.555	1.5	3.18

The curves represented by the calculations in Tables I and II are plotted in Figure 15. For 0.0001" power-piston clearance, the primary leakage path is seen to be across the ram-piston clearance, the flow increasing approximately linearly with displacement of the power piston. Increased displacement means increased flow to the ram and increased ram velocity. The increased velocity directly results in increased velocity leakage and indirectly causes increased pressure drop across the ram piston, which also adds to leakage flow. Power-piston leakage is insignificant except for small displacements.

The actual flow in the transition region from orifice flow in neutral position to laminar flow at some displacement ten or so times the clearance is probably not accurately represented by this analytical attempt. Fortunately, the power demands upon the system are directly related to power-piston displacement. Since we are more concerned with the limiting characteristic of the system at higher power levels, an accurate estimate of leakage flow for small power-piston displacement is not essential.

Increasing power-piston clearance to 0.0005" increases valve leakage by a factor of 125 and makes valve leakage the primary path. Again the transition region demonstrates peculiarities, but over the full range of displacements the leakage can be estimated as approximately $0.6 \text{ in}^3/\text{sec}$.

For power-piston clearance of 0.001" the valve leakages are becoming of the same order as the working fluid flow resulting in gross inefficiencies.

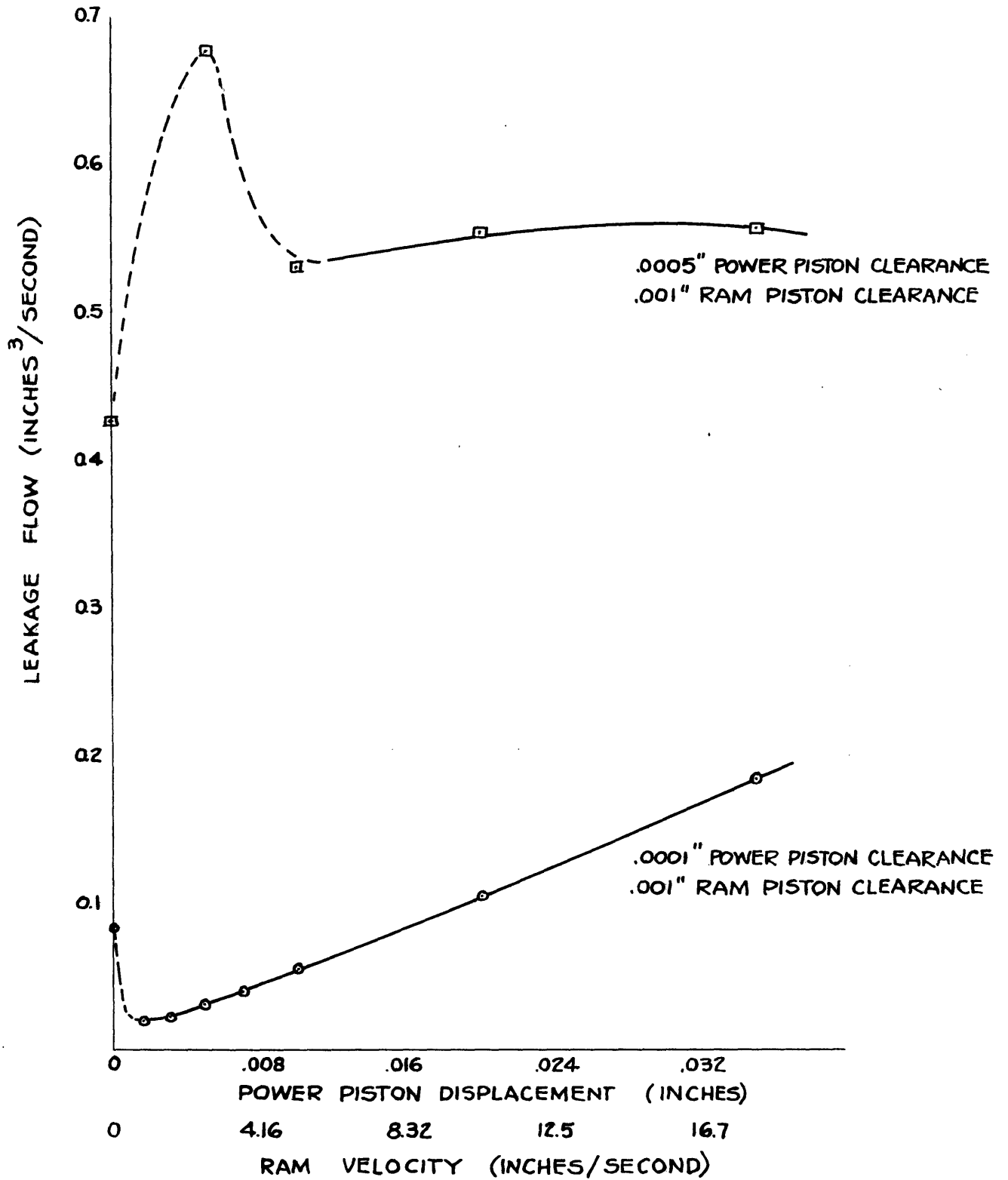


FIG. 15
SYSTEM LEAKAGE

AW

It is interesting to note that at the higher power-piston clearances, the leakage is greater in the displaced condition, where only laminar flow obtains, than in the neutral position where the flow is part orifice and part laminar. Analytically this is due in the first place to higher pressures existing across the valve ports (Δp_3) when the piston is displaced and the ram is loaded; and in the second place to the fact that the laminar flow is a function of the third power of clearance. Note that at the minimum clearance (0.0001") the situation is reversed and higher flow occurs through the orifice. It would be interesting to compare these admittedly sketchy analytical results with experimental evidence.

The combination of ram clearance of 0.001" and valve clearance of 0.0005" will be adopted as the best compromise. The valve clearance is increased by a factor of 5 which should reduce manufacturing cost, and the clearance should be adequate to pass dirt particles not trapped by the system's 5-micron filter. The resulting leakage is not excessive and will be taken as 0.6 in³/sec.

Substituting estimated numerical values into the system transfer function, we arrive at the following overall feed-through transfer function:

$$KG(s) = \frac{Y_1(s)}{(s)} = \frac{500}{s \left[\left(\frac{s}{135} \right)^2 + \frac{(0.245)}{135} s + 1 \right]} \cdot \frac{K_2}{8} = \frac{62.5K_2}{s(5.5 \times 10^{-5} s^2 + 3.63 \times 10^{-3} s + 1)}$$

where K_2 will have to be adjusted to meet the system stability specification of $M_p = 1.3$.

Writing the transfer function as a function of frequency:

$$KG(j\omega) = \frac{62.5 K_2}{j\omega [5.5 \times 10^{-5} (j\omega)^2 + 3.63 \times 10^{-3} j\omega + 1]}$$

Non dimensionalize by taking,

$$u^2 = 5.5 \times 10^{-5} \omega^2$$

$$u = 7.4 \times 10^{-3} \omega$$

Then,

$$KG(ju) = \frac{0.463 K_2}{ju [(ju)^2 + 2(0.245) ju + 1]}$$

Plotting the frequency-variant part of the function on a log-molulus, phase-angle plot⁶ (see Figure 16) for $M_p = 1.3$, the non-dimensional system gain $K_u = -11.6$ db at the non-dimensional resonant frequency of $u_r = 0.95$. Therefore,

$$K_u = 0.256 = 0.463 K_2$$

$$K_2 = 0.553$$

$$\text{System resonant frequency} = \omega_r = \frac{u_r}{7.4 \times 10^{-3}} = 128 \text{ radians/sec.}$$

$$\text{System velocity constant} = K_V = 62.5 K_2 = 34.3 \text{ l/sec}$$

The magnitude and phase vs frequency plots of the system are illustrated in Figure 17.

For the uncompensated system, the magnitude of response is seen to attenuate to a maximum value of 3.5 db over a range of frequencies prior to peaking to 2.3 db near the resonant frequency of approximately 20 cps. The significance of this attenuation has to be evaluated in terms

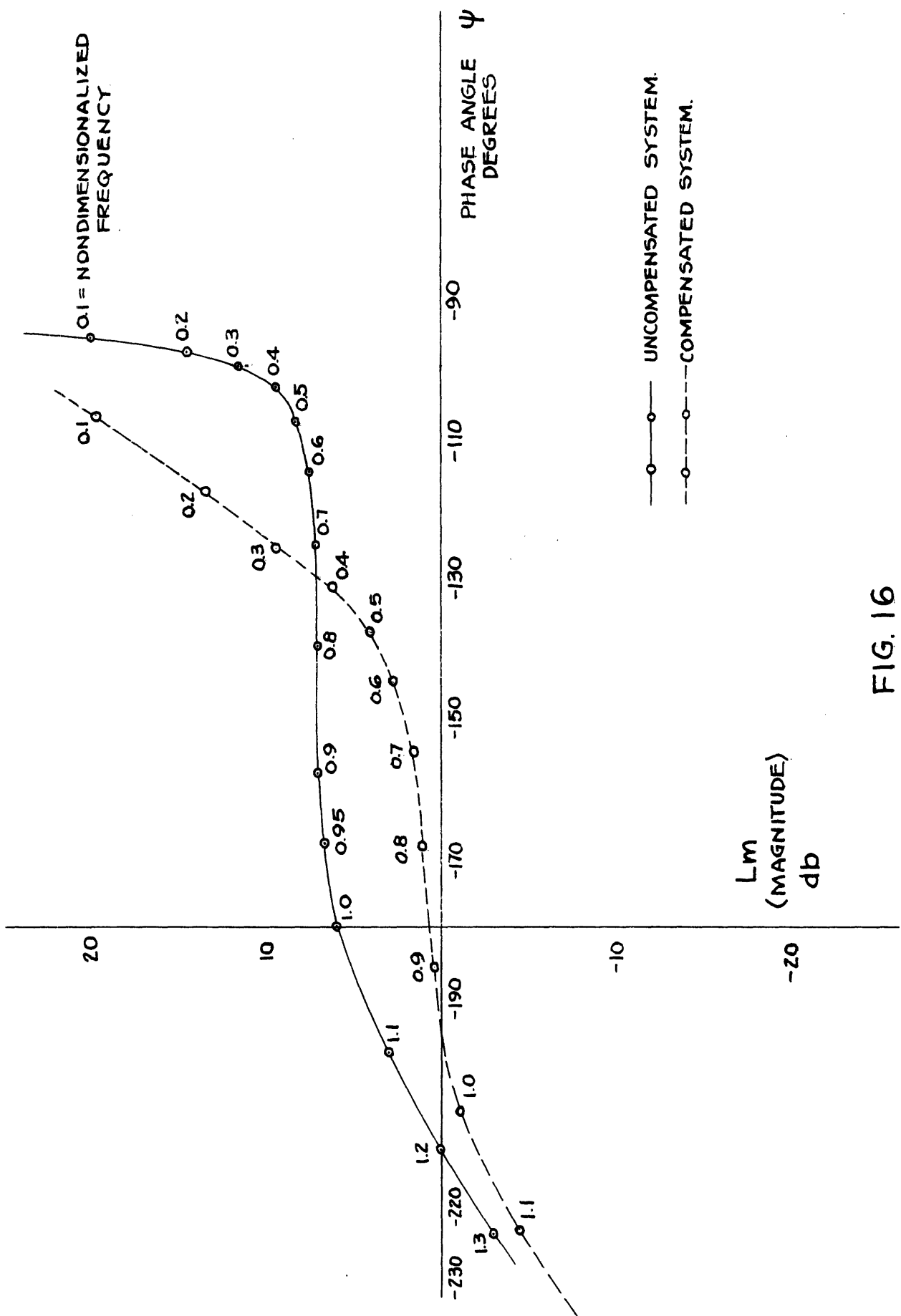


FIG. 16
LOG MODULUS VS. PHASE ANGLE

Bob

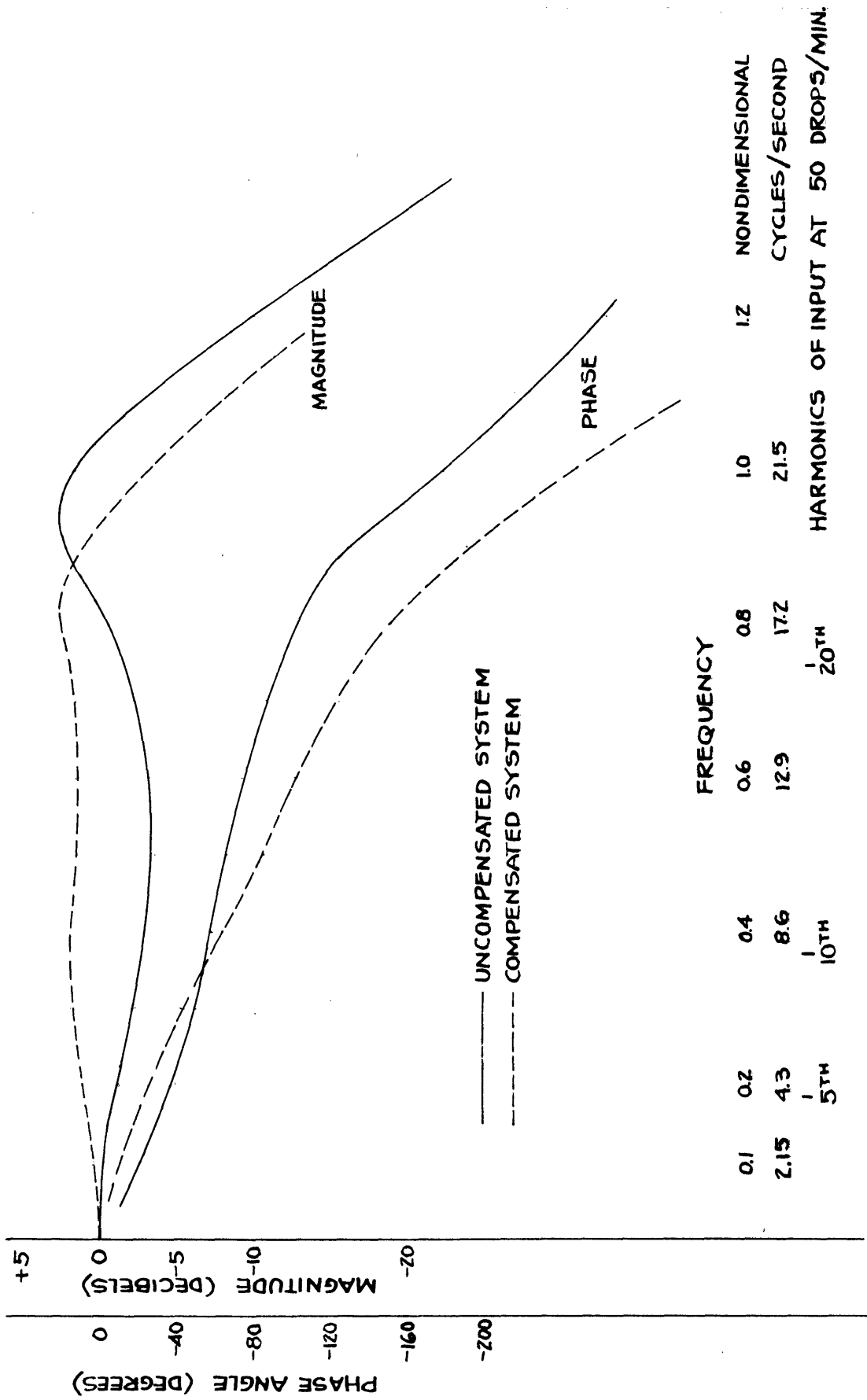


FIG. 17

SYSTEM RESPONSE VS. FREQUENCY

Edna

of the system specifications. Included on the frequency scale are the frequencies at which the various harmonic components of the input occur. Comparing the contributions of these harmonics to the input signal, as evaluated from the Fourier analysis of the typical input cam on page 13, we see that at the maximum operating speed of 50 drops per minute, the frequencies which are somewhat attenuated represent very small contributions to the input signal. It is felt that the attenuation of these components will not seriously affect the fidelity of reproduction. At the operating speeds in current practice - of the order of 5 drops per minute - the twentieth harmonic occurs at 1.66 cps and, therefore, none of the frequencies of interest undergo any attenuation.

Rederiving and evaluating the system transfer function at lower power levels, i.e., when the pressure drop across the ram piston is considerably less than $2/3$ of the source pressure, will indicate that the natural frequency, i.e. bandwidth, is reduced, and the damping ratio is increased. However, since the input curve, for low-power requirements has no abrupt changes, there is no need for bandwidth adequate to provide response to the higher frequency components of the Fourier analysis, since these higher frequencies do not contribute to that portion of the input information. Similarly, the increased damping, although it results in a slower speed of response, will not affect system performance since the changes in input information at this lower power portion of the input curve are very gradual.

To give some idea of the transient speed of response, at maximum power, where the changes in the input curve are most severe, the

bandwidth of 100 radians will result in a build-up time of approximately 0.03 seconds in response to a square-wave input. Comparing this speed of response with the time rate of input information (5-50 drop cycles per minute) indicates that, in comparison with the input information (which never even approaches a square wave), the response time is so short, that the system is operating practically at steady-state conditions at all times.

This loop, like any other, is amenable to compensation. The extent of compensation attempted in this thesis has been restricted, due both to time limitations on the part of the author, and to the impracticability of applying extensive alterations to a system transfer function which, representing the sum total of a number of engineering estimates and assumptions, at best only approximates the real situation.

As an example of what might be done, however, the effect of a lag network with an attenuation factor of 3, and a time constant of 0.0074 second cascaded with the original transfer function is investigated. The intent is to minimize the sag in the magnitude response curve and to increase the velocity constant. Reference to Figure 17 indicates that this network flattens the response over a somewhat narrower, but still adequate, bandwidth, although it results in some amplification of the higher harmonics. The new resonant frequency is 105 rad/sec and the velocity constant is 72 sec.⁻¹

Further compensation will be justified following more accurate experimental evaluation of the system parameters.

E. Economic Considerations

In light of the author's lack of experience in industrial engineering and fabricating costs, this section must be limited to an assessment of the purchase prices of the principle components discussed in the thesis.

The DACL MX valve is manufactured by Hydraulic Control Company of 87 Terrace Street, Roxbury, Massachusetts. The valve currently sells for approximately \$650; this price will be altered by the changes to increase clearances, which should materially reduce the cost, and the changes to reduce stroking force, which may somewhat increase the cost.

For an estimate of the a-c torque motor cost, compare it with the currently-offered type 35 d-c torque motor, sold for approximately \$175 by Transonics Incorporated of Bedford Airport, Bedford, Massachusetts. The laminated armature construction of the a-c motor will probably increase the price somewhat.

The linear differential transformers, series C, type 10 L-C, altered for 800 cps excitation, sell for \$40 each and two are required. The manufacturer is Schaevitz Engineering, Crescent Boulevard at Drexel Avenue, Pennsauke Township, Camden, New Jersey.

The specifications, cost and manufacturer of the hydraulic pressure source are covered in John Dunn's thesis, Item 5 in the bibliography, from which may also be estimated the cost of constructing the needle drive and supporting framework.

The main drive motor, speed adjust and reduction units, together with the synchronizing plate cams and differential are as now used by Corning; they are therefore in a position to appraise the cost of the combination.

Additional units to be developed, the cost of which must be included in this estimate are the assembly at the program cam, as shown in Figure 12, the feedback assembly as suggested in Figure 11 and developed in John Dunn's thesis⁵, and the electronic amplifiers and compensating networks.

A power supply to the amplifier and an 800 cps motor generator set must also be purchased.

CHAPTER IV

CONCLUSIONS

About the best way to evaluate this thesis is to contrast the 300 to 350 hours devoted to it by the author to the six months or a year or longer that the actual physical development of the machine would certainly take. In light of this considerable time difference, the thesis becomes more of a survey of what an improved feeder should and could do than a complete "blueprint" which need only be transformed into physical reality.

The development of the system specifications points up the lack of information vital to the full development of the machine, information which the author estimated by simplified analyses. Before expensive and extensive redesign is considered, these parameters should be established by more elaborate analyses, and what is more important to the engineer, by experimental evidence. These investigations would include a study of the viscous forces between the glass in the bowl and the moving needle as a function of velocity and acceleration of the needle. An analysis of foreseeable drop rates and weights should be instituted to establish values for maximum velocities and accelerations of the needle. Then, if the project to correlate gob shape and weight with the program cam profile continued its present development to the stage where a mathematical statement of the cam profile would insure the desired gob, the development of a really high performance servomechanism,

with high or infinite velocity and acceleration constants would perhaps be justified. At present there appears no economic basis for such a high performance design. An experimental evaluation of leakage and its effect upon the system ought to be correlated with the analytical results.

Armed with this information a dynamic solution could be prepared with greater confidence that it would coincide with reality. Then the dynamic elements of the design, the valve and torque motor, and associated electronic equipment could be coupled to a dummy load whose response duplicated that of the actual needle. The inertia loading of the dummy could be made equal to that of the actual needle, and viscous loading could be simulated using lower viscosity easier-handled fluids with higher velocity gradients. Subjecting this similar system to the velocity and acceleration input expected in the actual feeder would constitute a positive engineering check on the feasibility of going ahead with the complete redesign.

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