Final Report

STUDY FOR MAINTENANCE OF HYDRAULIC MOTORS

(27 June 1967 - 27 October 1968)

Contract No. NAS5-10559

Prepared by

Mechanical Engineering Laboratory
Research and Development Center
General Electric Company
Schenectady, New York

for

NATIONAL AERONAUTICS AND SPACE ADMINISTRATION
GODDARD SPACE FLIGHT CENTER
GREENBELT, MARYLAND

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Goddard Space Flight Center

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ABSTRACT

This report describes development of methodology and instrumentation for detection and identification of incipient malfunctions in hydraulic components of antenna drives. Means for acquisition, processing, and interpretation of diagnostic data are presented in order to transfer competence for condition monitoring from the General Electric Company to the Goddard Space Flight Center.

This work has been based on the use of nonintrusive vibration measurements as the means for extracting diagnostic data. Analytical studies have predicted the character of normal vibration signatures and the changes to be expected as the result of various defects. The predictions were tested and confirmed in laboratory and field tests, in which it was shown that many defects can be detected without system disassembly before performance has been degraded, thereby securing the advantages of planned maintenance and eliminating unexpected failures.

The data reduction techniques developed by General Electric have been duplicated at Goddard, using equipment now owned by the National Aeronautics and Space Administration. As a result, exploitation of the technology developed can begin immediately. It is recommended that a program be instituted for acquiring raw data from the field on magnetic tape, and that data be analyzed at Goddard on a regular basis. The analysis and interpretation of data would be aided and speeded by integrating the present instrumentation with a digital computer. This should be done, and an evaluation made of benefits which would be derived from a concept whereby automated diagnoses would be accomplished locally at each antenna site.

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Section 1

INTRODUCTION

SCOPE AND PURPOSE OF THE REPORT

The work described in this report has two goals. The first is development of all the background necessary to permit diagnoses of the condition of hydraulic components of a satellite tracking antenna, on the basis of nonintrusive vibration measurments. This includes development and testing of theory through carefully planned tests and data analysis. This phase of the project establishes the validity of the approach and demonstrates that malfunctions can be detected at an early stage of development without disassembly of the hydraulic system or its components.

The second goal of the work is to describe how such a diagnostic system can be implemented by Goddard Space Flight Center. Detailed information is included on data acquisition, with instructions for use of Goddard's data analysis instrumentation, and recommendations are made for criteria to be applied to the analyses for recognition of defects. This provides Goddard's present capability of performing diagnoses in-house. Means are described whereby the capability could be expanded by the use of a computer to perform many of the functions now carried out by a skilled technical person.

PROJECT BACKGROUND

Because of the critical application of satellite tracking antennas, particularly those used in manned flight monitoring, very high reliability is required to ensure proper operation during critical periods. In this study diagnostic techniques have been developed to contribute to more effective maintenance as a means of achieving that goal. All measurements are made external to the hydraulic system, without penetrations which could themselves decrease reliability. By recognition of incipient defects without teardown inspections, maintenance can be directed only to the items needing repair, and the work can be planned in such a way as to minimize down-time, repair costs, and unplanned outages.

HYDRAULIC SYSTEM, USB ANTENNA

The hydraulic systems of the Unified S-band (USB) antenna are described in detail in its technical manual (Ref. 1). A briefer summary may be found in Proposal No. RFP 525-25164-333 of February 1, 1967, which led to the present contract. That part of the Request for Proposal dealing with the characteristics of the hydraulic system is included as Appendix I of this report.

HYDRAULIC COMPONENTS STUDIED

In choosing those hydraulic components of the USB antenna which would be selected for intensive study, Goddard Space Flight Center applied the following criteria:

1. Each component should be important to proper system operation.
2. Each should differ considerably from the others in function and construction.
3. Each should be typical of a large and important class of hydraulic components.

Application of these criteria not only ensured an immediate usefulness of the results in monitoring the USB antenna drive, but made it possible to extend them to similar devices in other systems. The components chosen were as follows:

1. Hydraulic servo pump, part No. PV07 XS-020-51N-Y2-OYW, Denison Engineering Division, American Brake Shoe Company.
2. Dual-vane pump, part No. V2235-5-2-1AA20-S214-MOD, Vickers, Incorporated
3. Solenoid valve, part No. DID04-33-103-OW-OU-1008, Denison Engineering Division, American Brake Shoe Company.
4. Servo control valve, part No. 410-1158, Aerospace Division, American Brake Shoe Company.
5. Pressure relief valve, part No. CG-03-C-10-S85-MOD, Vickers, Incorporated

The relationship of each of these components to the total system is described in both the system manual referenced earlier and in Appendix I to this report. Briefly, their use is as follows:

The dual-vane pump has two sections fed by a common inlet. The first supplies replenishing oil at five gallons per minute (gpm) and 60 pounds per square inch (psi), while the second delivers flow at 2 gallons per minute. The pressure of this discharge is regulated at 800 pounds per square inch by the pressure relief valve described. This oil is delivered to the servo control valve, which responds to electrical signals in such a way as to control the volume and direction of the output of the hydraulic servo pump. When the antenna is stowed this output is vented through the above referenced solenoid valve; before moving the antenna the solenoid must be actuated so as to close off this bypass and direct the pump output to the hydraulic motors.

Section 2

ANALYTICAL INVESTIGATIONS

PREDICTED MODES OF FAILURE

In order to predict the effect of a malfunction on the vibration signature of a device it is necessary to understand its dynamics and to know the ways in which it is likely to fail. Well-designed hydraulic components of the types studied are inherently long-lived and relatively trouble-free if the usual precautions of maintaining clean hydraulic fluid are followed. The USB antenna uses quality components in a conservative way, and failures of these installations are rare. As a consequence, there is no statistically valid information which can be used to assess the probability of any particular failure. The analyses of probable modes of failure must therefore be based mainly on engineering judgment. Several sources were consulted. These include operational personnel within Goddard Space Flight Center, Collins Radio Company engineers (who designed the system), the component manufacturers, engineers of the General Electric Hydraulic Transmission Project, and engineers of the Fluid Controls Unit of the General Electric Research and Development Center.

Servo Pump Failure Modes

The Denison servo pump design parameters have been de-rated in the application under study so that the pump's life is expected to be 25,000 hours. Some specific problems have been encountered in practice:

1. An instance has occurred when the coupling between the servo pump and the a-c motor loosened. The pump failed to move fluid and the antenna could not be moved.
2. On one occasion a piston apparently seized in a cylinder of a hydraulic motor. Although this is not one of the components under study, it is similar in design principles to the servo pump, and is listed here as a possible servo pump failure mode. This condition was followed by a complete break of the piston rod.

The following failure modes, although not known to have actually happened on USB antenna systems, are held to be probable failure mechanisms for the servo pump:

1. Port erosion -- this condition will change the pressure pulse experienced within each cylinder.
2. An increase in clearance between the pistons and cylinder walls due to gradual wear. Internal leakage and loss in pressure and flow result.

3. Galling at the slipper-creep plate surface. This can occur from lubricant failure during cold starts and the high unit stresses which exist on these surfaces.

4. Fatigue failure at the surfaces between the piston balls and the ball sockets. These are areas of high unit stresses.

5. Worn and leaking shaft seals.

6. Worn ball and roller bearings.

7. Wear and gradual removal of material from the edges of the cylinder block due to large moments existing on the piston in the fully displaced position at high load conditions.

8. Scoring or galling of the surface between the port plate and the cylinder block.

9. Cavitation in the suction passages of the pump resulting in cavitation erosion in the port plate passages.

Vane Pump Failure Modes

Most generally, a vane pump does not fail catastrophically, but just wears out. The result of wear is an increase in clearances between the moving parts and a decrease in pump performance. Possible failure mechanisms are as follows:

1. Vane collapse -- a vane does not move out to the ring surface and therefore does not make its contribution to the total flow and pressure. This is considered a rare occurrence.

2. Bearing failure.

3. Cavitation -- since little or no supercharge pressure exists at the pump inlet, cavitation can take place more readily in the vane pump than in the servo pump.

4. Worn or scored vanes.

5. Scoring of the ring by the vanes, caused by constant rubbing, blocked oil passages at the base of the vanes, and material inhomogeneities.

6. Vane block splitting, caused by excessive moments on the vanes.

Servo Control Valve Failure Modes

Servo control valves can malfunction in the following ways:

1. Sluggish operation due to a sticking spool. Most often, this occurs because of a deposit of silt in the 25×10^{-6} inch clearance space between the spool and the cylinder. If the silt is abrasive, the spool or block may be scored as well, aggravating the tendency to stick.

2. Excessive Leakage.

3. Null shift due to physical unbalance. This can be caused by an increase in the leakage rate on one side of the spool versus the other side.

Solenoid Valve Failure Modes

The most common faults associated with this device are:

1. Internal leakage caused by scoring.

2. Increased friction due to contaminants in the oil.

Relief Valve Failure Modes

This is generally a trouble-free component if held to its design parameters. Malfunctions possible to this item are:

1. Internal leakage due to damaged seats.

2. Failure to hold pressure due to relaxed or broken spring.

3. Fluctuating pressure or chatter due to damaged piston or valves.

SIGNATURE PREDICTIONS

In this phase of the program techniques were developed for predicting the character of vibrations expected to be generated by each of the five hydraulic components studied. This was done in such a way as to make evident the changes to be expected due to the presence of selected critical malfunctions. The emphasis was more on understanding of the phenomenon than on mathematical rigor. Such understanding was essential to permit design of experimental procedures which could be used to detect defects and malfunctions from analyses of the outputs of vibration sensors.

The analytical problem was approached by mathematically modeling each component on a digital computer. The response of the component to a particular stimulus was calculated as will be described. The conversational language in which the programs are written is BASIC (Refs. 2, 3), and they were run on the General Electric 265 time-sharing computer. All computer programs described in this report are included in Appendix II.

Servo Pump

The servo pump is a piston-type pump having seven cylinders in an axial arrangement. Shaft power is applied to cause the piston and cylinder assembly to rotate as sketched in Figure 1. The angle β between the rotating and static elements of the pump may be varied to adjust the rate and direction of the flow. This angle is controlled by a servo valve via a yoke and hydraulic actuators.

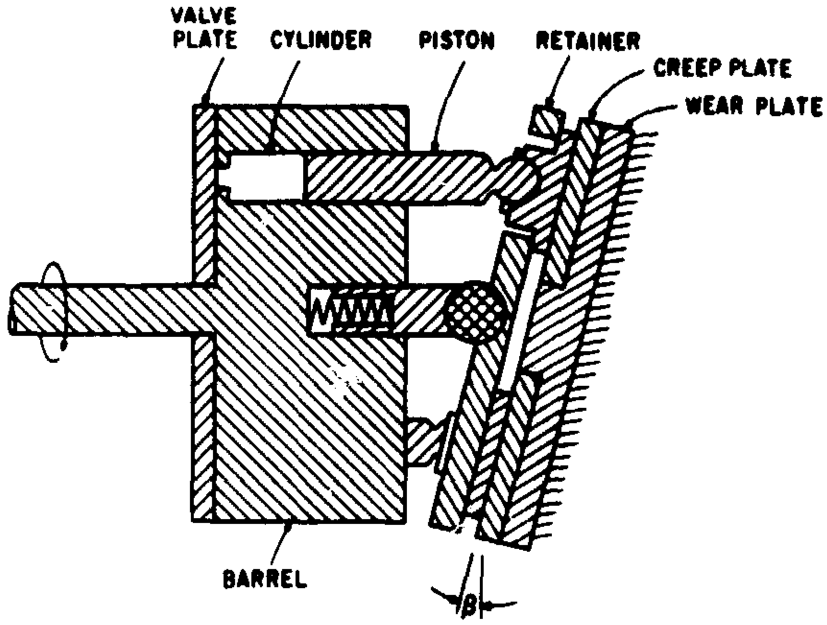


Figure 1. Piston Pump Rotating Assembly

The discharge flow from each cylinder approximates a half-sine pulse. A valving arrangement, including discharge ports in the barrel and suitable slots in the port plate (Figure 2), manifolds the output of the several cylinders to produce a resultant flow having relatively small ripple.

A computer program (GSFC23) has been written which models the operation of the piston pump by computing the flow and discharge pressure (using an orifice for a load) by taking into account both the instantaneous displacement and the leakage of each piston. The axial position of a piston in a cylinder varies with rotation angle θ in a manner which is approximately sinusoidal. As can be seen in the sketch of Figure 3, a point on each piston describes an ellipse as the barrel rotates, since this is the geometric figure formed by the intersection of a plane and a cylinder. Letting the radial distance from the shaft center-line to the cylinders be R, and the position of a point on the piston be X:

$$R^2 + X^2 = Y^2 \tag{1}$$

From the known equation of an ellipse, the distance, Y, from the center to a point on periphery at the angle θ can be expressed in terms of the lengths of the major and minor axes, 2a and 2b.

$$Y^2 = \frac{a^2 b^2}{a^2 \sin^2 \theta + b^2 \cos^2 \theta} \tag{2}$$

but $\quad b = R$

$$a = \frac{R}{\cos \beta}$$

where β is the angle between the rotating and fixed axes, (Figure 1).

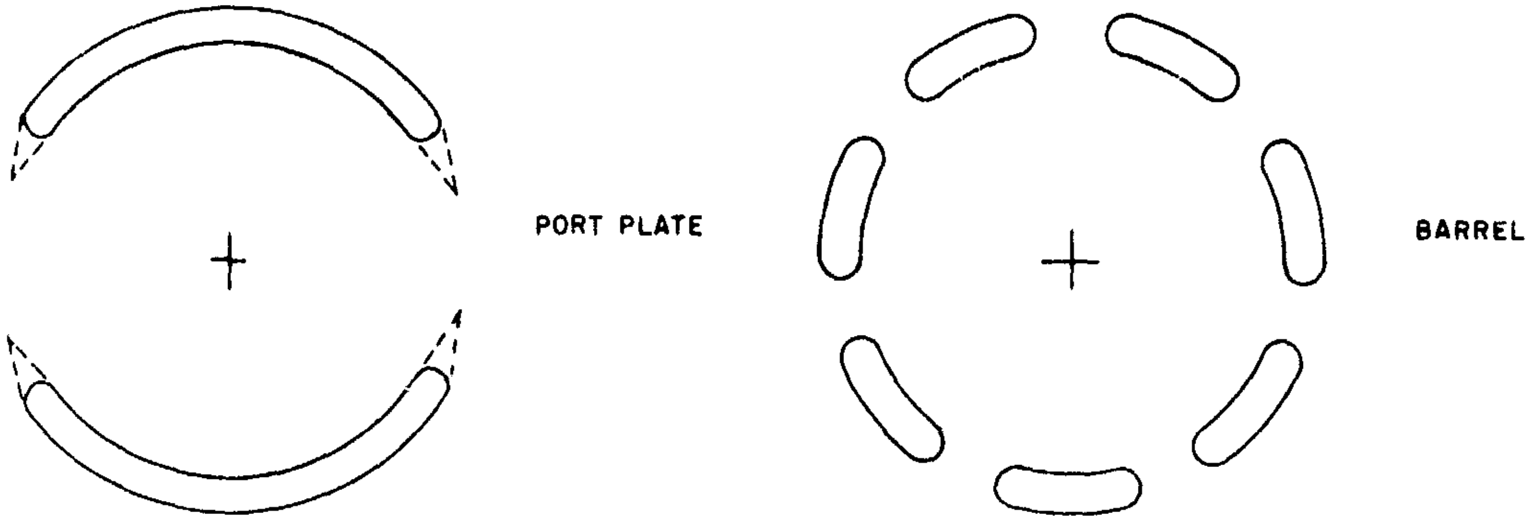


Figure 2. Servo Pump Valving

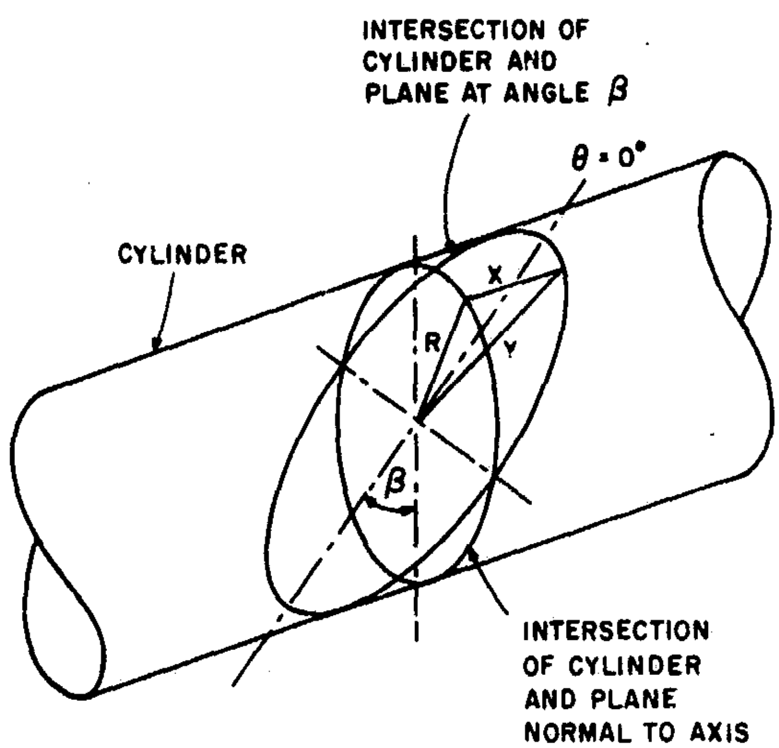


Figure 3. Intersections of Planes With a Cylinder

Substituting and solving for X,

$$X = \frac{R \cos \theta \sin \beta}{1 - \cos^2 \theta \sin^2 \beta} \tag{3}$$

The rate of change of piston position with angle of rotation θ can be found by differentiating:

$$\frac{dX}{d\theta} = -R \sin \theta \sin \beta (1 - \cos^2 \theta \sin^2 \beta)^{-3/2} \tag{4}$$

By means of Equation 4 it is possible to calculate the velocity of each piston as a function of angle, and therefore the pump delivery which could be expected neglecting leakage. The leakage in the annulus between each piston and its cylinder can be calculated by the use of an expression from a standard text (Ref. 4):

$$Q = \frac{3.132 \times 10^6 \; P \; B^3}{\nu L} \pm 360 \; SB) \; \pi D \tag{5}$$

where Q is leakage, cubic inches per minute

P is differential pressure, pounds per square inch

B is clearance between piston and cylinder, inches

ν is kinematic viscosity, centistokes

L is path length, inches

S is piston velocity, feet per second

D is cylinder diameter, inches

(The minus sign is used when pumping, since the piston motion opposes the leakage flow)

If there is a difference in leakage from one cylinder to another this can be accounted for by appropriately adjusting clearance. The instantaneous velocity of each piston is found by the use of Equation 4 and the rotational speed. In this way the instantaneous flow rate can be calculated as a function of rotational angle θ. The pressure developed across the orifice, including both steady and ripple components, may then be found.

The computer program asks for the percentage of full stroke (as could be read from the yoke position indicator on an actual pump) and a nominal value of pressure. From the rated flow at full stroke (20 gpm) the flow rate at this new stroke is computed, and the orifice area determined which gives this nominal pressure; the length of the stroke in inches and the yoke angle are determined. The shaft is rotated in small increments, during each of which the delivered flow (displacement has leakage) is calculated. The true pressure at the orifice is then computed; it is somewhat less than the nominal value originally entered because of leakage. Figure 4 shows an example of how the pressure fluctuation is predicted to vary during one revolution of the pump at half-stroke. The jump in pressure at midrevolution shows the effect of a leaky cylinder. A Fourier analysis of the pressure waveform thus generated determines the magnitude of its spectral components.

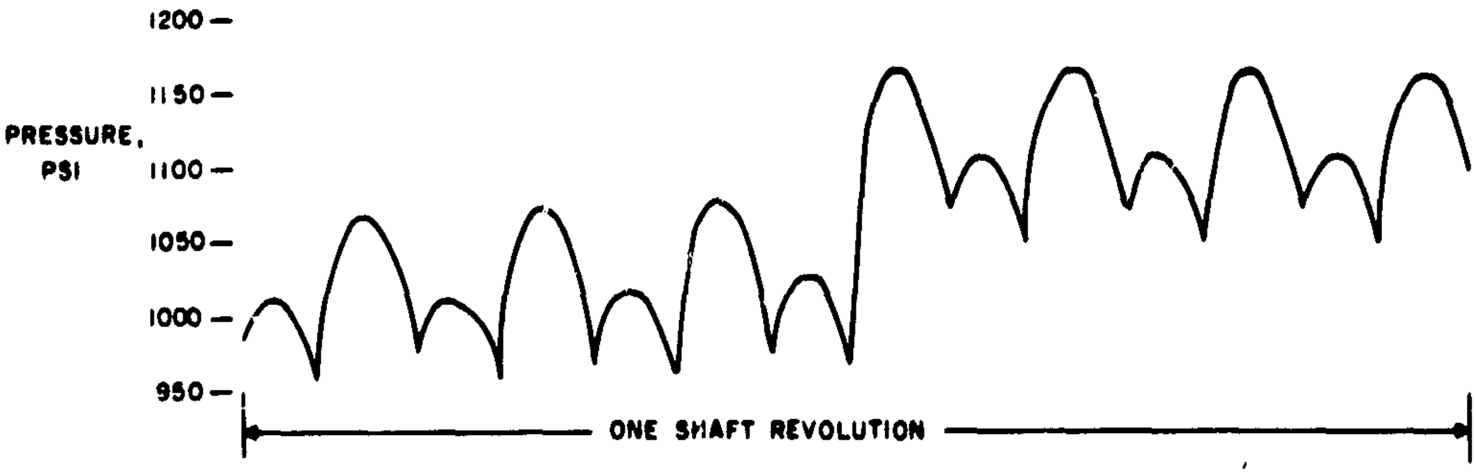


Figure 4. Predicted Pressure Waveform of Servo Pump

As would be expected, the effect of leakage is greatest at higher differential pressures. Table 1 shows the effect of increased clearance in one cylinder at half-stroke at 1300 psi and 450 psi. The nominal clearance is 0.0010 inch; the clearance in the worn cylinder is 0.0014 inch.

When there is no differential leakage the first component of any size in the ripple is the seventh harmonic of running speed. Also present are multiples of seven per revolution, with similar amplitudes. When there is dissimilar leakage in one or more cylinders due to wear or improper manufacture, the most striking effect is a large new component at the one-per-revolution frequency, which for this pump is at 20 hertz. If the pump is normal this component is absent, as described above.

The vibration sensed on the outside of the pump will not be the same as the pressure pulsations described above. This difference is not quantitatively predictable. The pump structure and the piping must act as a high-pass filter, for example, since the d-c flow component must be contained. Also, the structure will have various resonances and antiresonances which will enhance some spectral components while attenuating others. However, the structure cannot generate new spectral components, but can only respond to forcing functions. Therefore, the presence or absence of certain spectral components in the vibration show whether or not these same components are present in the pressure signal.

Various malfunctions of the servo pump (scoring, galling, ball-socket fatigue) will result in scraping or impact-type noise occurring at a rate which is synchronized with the shaft rotation. While such noises will cause

Table 1

PREDICTED SERVO PUMP PRESSURE SPECTRA WITH AND WITHOUT DIFFERENTIAL WEAR OF ONE CYLINDER

Harmonic	Frequency	dB re 1 psi Normal	dB re 1 psi Worn
Nominal Pressure = 1300 psi			
1	20	-88	35
2	40	-96	8
3	60	-94	26
4	80	-96	0
5	100	-96	21
6	120	-96	-2
7	140	30	32
8	160	-101	-5
9	180	-98	16
10	200	-96	-8
11	220	-98	14
12	240	-95	-10
13	260	-100	13
14	280	27	27
Nominal Pressure = 450 psi			
1	20	-97	17
2	40	-104	-9
3	60	-102	8
4	80	-105	-17
5	100	-105	3
6	120	-105	-21
7	140	12	14
8	160	-110	-24
9	180	-107	-2
10	200	-104	-26
11	220	-107	-4
12	240	-103	-28
13	[illegible]	-108	-5
14	280	19	-19

an increase in the overall broad-band signal generated by the pump, they may be expected to be more easily detected and classified by study of the pump waveform than by spectrum analysis. Because such noises occur at the same relative shaft angle in each revolution, they should be separable from continuous processes. Summation analysis, as described in Section 5 of this report, would be useful for developing displays of such transients.

Cavitation in a pump results in a very large increase in broad-band noise, particularly at high frequencies. It occurs whenever the instantaneous pressure drops below the vapor pressure of the hydraulic fluid. The level at which the onset of cavitation is detected will depend on the relative amplitude of the cavitation noise and the noise due to the normal turbulence, which has a similar character.

Vane Pump

The vane pump is a positive displacement pump in which the circular rotor has a series of 12 movable vanes in radial slots around the periphery. These are kept in contact with the contour of the housing by means of a combination of centrifugal force and hydraulic pressure introduced at the bottom of the slots. Two such pumps are mounted on a common shaft.

The outer contour roughly approximates an ellipse, allowing for two inlet regions and two discharge regions (Figure 5). As the rotor spins, the chamber between the rotor and housing expands at the intakes and compresses at the discharges. In between the inlet and discharge ports the outer contour is circular; hence, no pumping takes place. Assume that the circular arcs exceed the vane spacing by 2D degrees, insuring inlet and discharge separation by at least one vane.

The exact contour of the outer housing between the circular arcs has been estimated. At one end of each port area the contour must match the rotor radius, while at the other it matches a larger radius. This contour is approximated by taking an ellipse with these major and minor axes, and dividing it into four equal parts at these axes. Each sector subtends an arc of 90 degrees. Each port on the housing subtends an arc of less than 90 degrees; therefore, the sector of the ellipse may be scaled or "compressed" to fit the available arc. In this way a contour is formed which blends smoothly with the circular arcs. Both the radius and the slope match where the circular arc and the "compressed" elliptical sector join.

The polar equation of an ellipse is

$$r = \frac{a^2b^2}{a^2\sin^2(\theta) + b^2\cos^2(\theta)} \qquad (6)$$

When $\theta = 0$, let $r = R_2$ (one-half the major axis of the ellipse).

When $\theta = \pi/2$, let $r = R_1$ (one-half the minor axis).

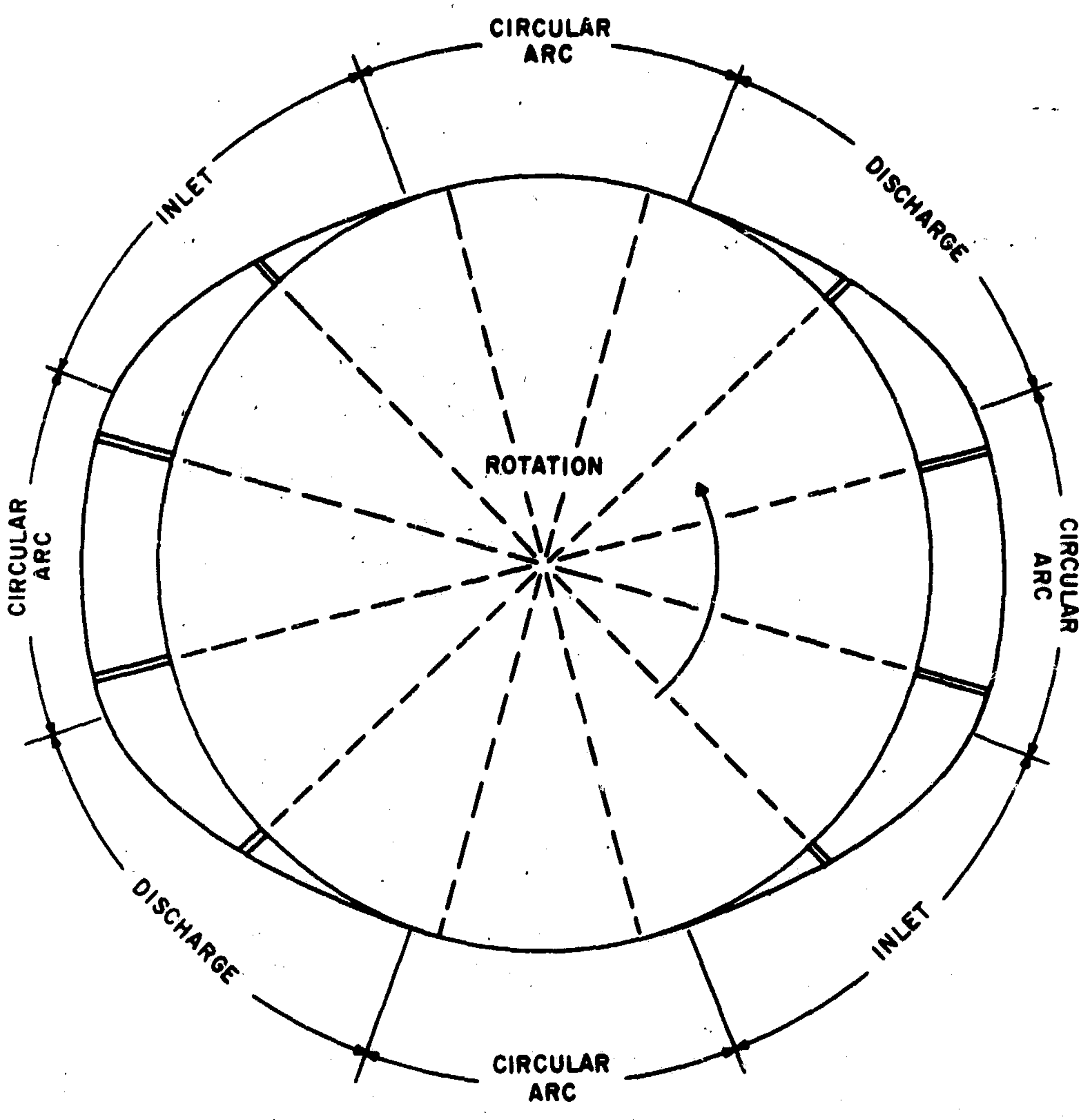


Figure 5. Vane Pump

Therefore:

$$a = \sqrt{R_2} \quad (7)$$

$$b = \sqrt{R_1} \quad (8)$$

Suppose that the sector of the ellipse between $\theta = 0$ and $\pi/2$ is to be scaled or compressed to fit in the arc between $\theta = (\pi/12 + D)$ and $(\pi/2 - (\pi/12 + D))$, or an arc of $(\pi/3 - 2D)$ radians. To do this, let $r(\theta) = r(\varphi)$, where

$$\varphi = [\theta - (\pi/12 + D)] \cdot [(\pi/2) / (\pi/3 - 2D)] \quad (9)$$

In summary, the radius to the outer housing in the first quadrant is as follows:

$$0 < \theta < (\pi/12 + D): \; R_3 = R_2 \quad (10)$$

$$(\pi/12 + D) < \theta < (5\pi/12 - D): \; R_3 = \frac{R_1 \cdot R_2}{R_2 \sin^2(\varphi) + R_1 \cos^2(\theta)} \quad (11)$$

$$(5\pi/12 - D) < \theta < \pi/2: \; R_3 = R_1 \quad (12)$$

If the vanes had zero thickness, and therefore zero volume, the pump discharge would be free of flow perturbations. Perturbations are introduced by the fact that the vanes do have finite volume and therefore occupy space which would otherwise be filled with fluid. Computer program GSFC17 was written to analyze this pump's signature. The position of each vane is continuously noted. When the vane is within the region of a discharge port the vane length is computed, and its volume subtracted from the flow volume which would exist with zero-thickness vanes. In this way the flow volume as a function of shaft angle is calculated.

The length of a vane is L:

$$L = R_3 - R_1 \quad (13)$$

Assuming a vane length (normal to the radius) of unity, the delivery of the pump (neglecting vane thickness) is as follows:

$$Q = \pi(R_2^2 - R_1^2) \cdot \frac{\pi/3 - 2D}{2\pi} \text{ in}^3 \times \frac{1}{\pi/3 - 2D} \cdot \frac{1}{\text{rad}} \times 2\pi \frac{\text{rad}}{\text{rev}}$$

where n is the speed in RPS, or

$$Q = \pi(R_2^2 - R_1^2) \frac{\text{in}^3}{\text{rev}} \text{ from one port or}$$

$$Q = 2\pi(R_2^2 - R_1^2) \frac{\text{in}^3}{\text{rev}} \text{ from both ports} \quad (14)$$

From this must be subtracted the volume of the vanes. The instantaneous volume of a vane is

$$Q_1 = L \times H$$

where H is vane thickness.

The instantaneous sum of all vane volumes in discharge regions is Q_O.

Supposing one vane leaks, one must determine if it is in a position to allow inlet and discharge regions to communicate. Call the leakage flow Q_2.

$$Q_2 = Q \times L1$$

where L1 is the leakage fraction the total net flow is Q_3:

$$Q_3 = Q - Q_O - Q_2 \qquad (15)$$

If a vane is leaky, i.e., makes a poor seal with the outer housing, the discharge port may communicate with the inlet at certain shaft angles. Such communication takes place if the leaky vane but no other vane occupies one of the circular arcs in the outer housing. In that event, a fraction of the discharge flow is allowed to leak back to the inlet, thereby reducing the available flow. The program makes provision for this situation.

In the program the flow is calculated at each of many increments of shaft rotation. This is squared to determine a quantity proportional to the instantaneous discharge pressure, and the result stored. A Fourier analysis is made of the resulting pressure waveform to determine the relative amplitude of the various spectral components.

Figure 6 shows the pressure waveform predicted by this calculation for normal operation and for operation with a single leaking vane. The ripple is very small relative to the static pressure. Table 2 shows the effect of vane width on a pump with no leakage. As expected, if vane thickness is zero there is no ripple. Doubling of vane thickness increases pressure disturbances at the vane-rate (number of vanes times running speed) and its harmonics by 6 decibels.

Table 3 shows the predicted effect of leakage. A single leaking vane gives a new component (plus its harmonics) at four per revolution, since the leak decreases the available discharge flow whenever the leaky vane is located in one of the circular segments of the pump outer housing.

The pressure perturbations generated by the vane pump in its normal condition will be very small, and will be difficult to detect in the presence of background noise arising from flow processes (turbulence) and from other, more powerful, sources such as the piston pump. As with the piston pump, the housing will tend to attenuate low-frequency signals more than those at high frequencies. The waveform sketched in Figure 6 shows that there is a very abrupt drop in pressure 12 times per shaft revolution. When there is leakage present, pressure drops at 12 per revolution will again be present, but their amplitudes are expected to be dissimilar because of the greater

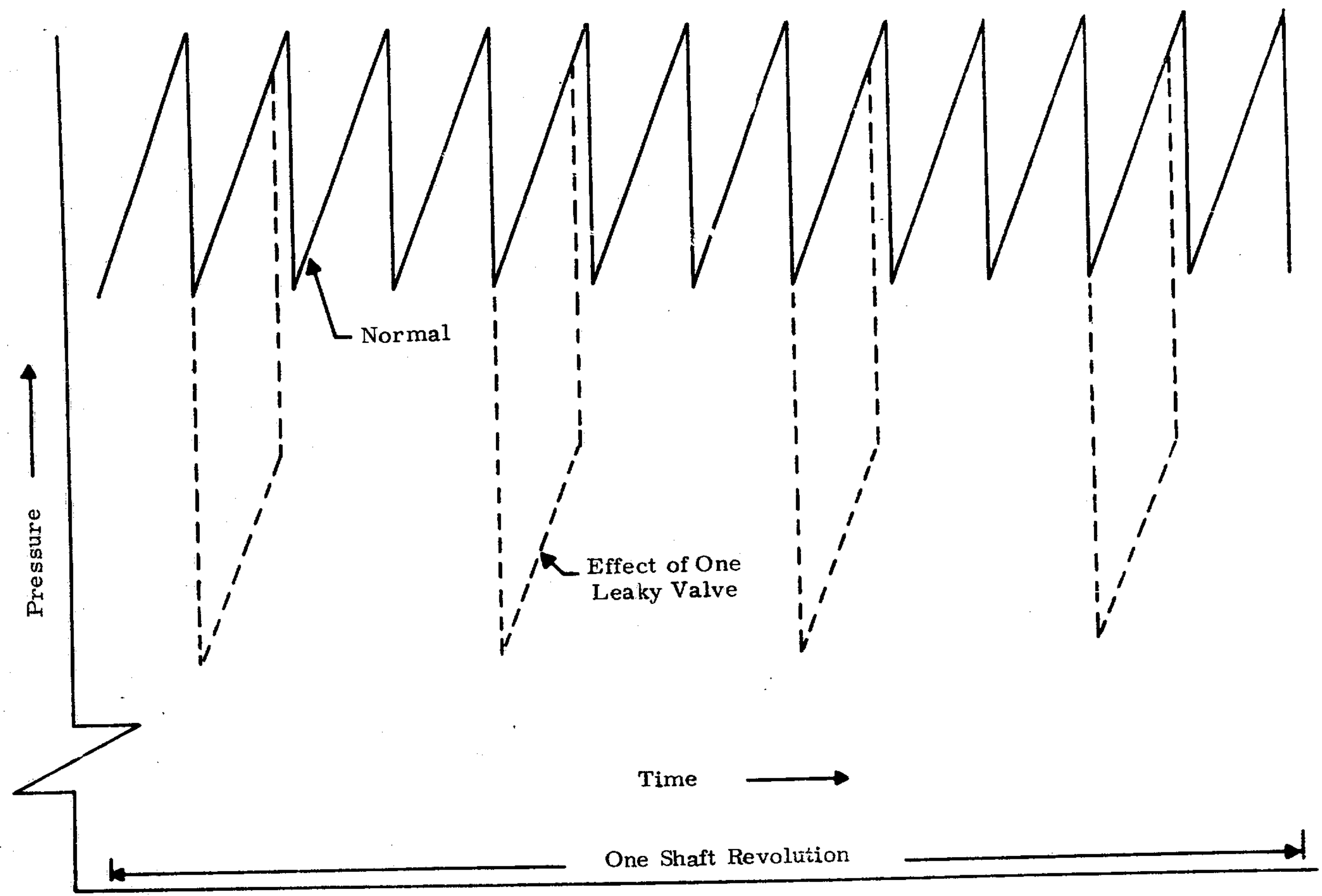


Figure 6. Vane Pump Pressure Waveform

Table 2

PREDICTED SPECTRAL COMPONENTS
OF NORMAL VANE PUMP PRESSURE
(Levels in decibels relative to an arbitrary reference.)

Special Component, Multiple of Pump Running Speed	Vane Width		
	0 inches	0.05 inch	0.1 inch
1	---	---	---
2	---	---	---
3	---	---	---
4	---	---	---
5	---	---	---
6	---	---	---
7	---	---	---
8	---	---	---
9	---	---	---
10	---	---	---
11	---	---	---
12	---	113	119
13	---	---	---
14	---	---	---
15	---	---	---
16	---	---	---
17	---	---	---
18	---	---	---
19	---	---	---
20	---	---	---
21	---	---	---
22	---	---	---
23	---	---	---
24	---	106	112

Table 3

PREDICTED SPECTRAL COMPONENTS
OF VANE PUMP PRESSURE, WITH ONE LEAKY VANE
(Levels in decibels relative to arbitrary reference)

Spectral Component, Multiple of Pump Running Speed	Leakage, % 0	Leakage, % 1
1	---	---
2	---	---
3	---	---
4	---	126
5	---	---
6	---	---
7	---	---
8	---	121
9	---	---
10	---	---
11	---	---
12	119	120
13	---	---
14	---	---
15	---	---
16	---	111
17	---	---
18	---	---
19	---	---
20	---	113
21	---	---
22	---	---
23	---	---
24	112	115

pressure jump experienced when leakage occurs, as seen in Figure 6. As Table 3 shows, the amplitude at three per revolution, with leakage, is expected to be larger than the normal 12 per revolution component.

Vane collapse will cause a large out-of-balance force at one per revolution, which should be easily detected. A one per revolution signal can also arise from other sources, such as leakage in the servo pump, and it may not be possible to differentiate between these. However, a rise in amplitude of this signal component is definitely indicative of system deterioration and would call for shutdown and remedial action.

Figure 6 shows that pulsations are expected in the discharge of the vane pump. For the same reason that there are discharge pulsations (finite vane-width) there will be pulsations in the inlet pressure. The space taken up by a vane cannot simultaneously be filled with hydraulic fluid. Since the two halves of the vane pump both draw their fluid directly from the tank, with no supercharge, they will be more prone to cavitation than the servo pump. Whether or not such cavitation actually takes place depends on the pressure drop in the inlet lines and the inertia of the fluid in them.

It will be shown in later sections of the report that inlet cavitation is normally present and takes the form of 12 bursts of noise per revolution, corresponding to the 12 pulsations in inlet pressure. Leakage would diminish the amplitude of these pulsations, because the leak would effectively supercharge the inlet.

Scoring of the ring or the vanes is expected to cause an increase in high-frequency broad band noise. If it is relatively uniform throughout a shaft rotation its detection will depend on whether the signal is large enough to be detected in the presence of the normal flow noise. If ring wear is non-uniform the vanes will be forced to reciprocate when passing over the worn region. In addition, tangential forces will be set up between the vanes and the ring which in an extreme case would cause splitting of the vane block. These forces will cause impact-type signals whenever a vane passes over the roughened area. The rate would be the vane-rate, which is the number of vanes times the shaft speed. The character of such signals would approximate the normal pump signature, except that in the case of severe wear the amplitude would be much greater.

Splitting of the vane block is a catastrophic failure which will increase unbalance, eliminate normal flow noise and vane-rate pulse generation, and vastly increase high-frequency noise due to the rattling of loose parts. It would be easily detected. It is expected that the malfunctions which would lead to such failure would be detected early enough that this problem could be avoided.

Solenoid Valve

This is a quarter-inch, four-way spool-type valve, actuated by an a-c solenoid. When it is deenergized, an internal spring holds the valve open, permitting the two inlet ports to communicate directly with the tank. When the solenoid is energized, both inlet ports are blocked.

Normal flow through the open valve is limited to two gpm. The instantaneous flow-rate at the moment the valve is opened will be higher if there is air dissolved in the hydraulic fluid.

The motion of the valve has only been predicted for the valve opening, which is accomplished by deenergizing the solenoid coil. The motion upon activation of the solenoid was not attempted, since it was found experimentally that the solenoid force was so high that the closure time would be dependent on the relative point in the a-c cycle at which the switch was closed. If the switch is closed as the a-c voltage swings through zero, there is at first no force, and actuation will not take place for some three or four milliseconds, after which time the instantaneous force will have built up sufficiently to permit valve closing. On the other hand, if the switch is closed near the peak of the a-c wave the force will build up more quickly and operation will be more rapid.

Calculation of the motion of the solenoid valve has been accomplished in two parts. First, consider its motion under normal low-flow conditions. The valve closure time is calculated by predicting net force on the moving system as a function of its position, and performing a numerical integration to determine position as a function of time. When the position is such that the moving system hits its stop, the time from the start of the sequence is noted.

Spring force was determined by experimentally relating force and deflection:

$$F_1 = 30X + 4.75 \tag{16}$$

where F_1 is in pounds

X is position relative to position when deenergized.

Friction forces come about from the O-ring seals. The magnitude has been estimated using data from the Parker O-ring Handbook. There are two components to this force: compression friction and pressure friction.

The running friction due to seal compression in one ring is given by

$$F_a = \frac{(-20.3 + 0.4725\ S - 0.00225\ S^2)}{.25}\ C\pi D \tag{17}$$

where S = Shore hardness of ring

C = Percent seal compression

D = Diameter of rubbing member

For pressures up to 1500 pounds per square inch the running friction due to pressure is given by

$$F_3 = (0.0253P + 10)\,A \tag{18}$$

where P = pressure

A = projected area of seal

A is tabulated by the Parker Company in their handbook; it may be roughly estimated for rod-groove applications as

$$A \sim \pi D\,T \tag{19}$$

where T = O-ring thickness in inches

Combining Equations 17, 18, and 19:

$$F_4 = \left[\left(\frac{-20.3 + 0.4725\,S - 0.00225\,S^2}{25}\right)C + (0.0253P + 10)\,T\right]\pi D \tag{20}$$

where F_4 = friction force per O-ring

It should be noted that $D = 5/32$ in this application, and that P is essentially zero since the O-rings seal only the flow from the valve to the tank; they are shielded from full pressure by the spool valve. There are two O-rings. Assuming their hardness to be 80° Shore, and compression to be ten percent, and taking T to be 0.07 inch (which is appropriate for this diameter), Equation 19 becomes

$$F_4 = 1.025 \text{ pounds per O-ring} = 2.05 \text{ pounds total} \tag{21}$$

Bernoulli forces are generated by the flow of hydraulic fluid through the valve (Figure 7). The magnitude is given by (Ref. 5):

$$F_5 = Q\,U\,\rho \cos 69^\circ \tag{22}$$

where Q is flow, cubic inches per second

U is velocity, inches per second

ρ is mass density, pound $\div$ second^2 per inch^4

The angle θ in Figure 7 is 69° whenever the valve opening is large compared to the clearance between the spool and the stationary member. Since the clearance is so small, the angle may be taken to be 69° with little error.

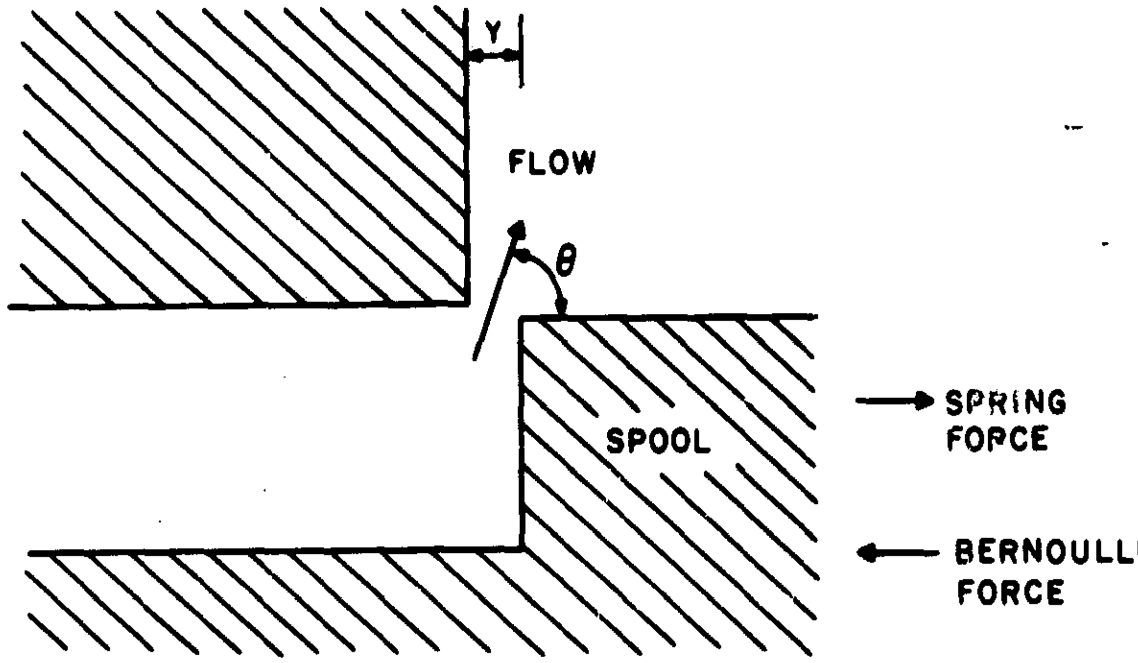


Figure 7. Generation of Bernoulli Forces in Solenoid Valve

The flow velocity U through an orifice is given by

$$U = C_D \sqrt{\frac{2P}{\rho}} \tag{23}$$

where C_D is the discharge coefficient

P is the pressure differential, pounds per square inch

Combining Equations 22 and 23:

$$F_5 = (\sqrt{2\rho}\, C_D \cos 69°)\, Q\sqrt{P} \tag{24}$$

If C_D = 0.6, and using the density appropriate for hydraulic oil, Equation 24 becomes

$$F_5 = 2.92 \times 10^{-3}\, Q\sqrt{P} \tag{25}$$

The normal flow through the solenoid valve is 2 gpm; i.e., 1 gpm through each half of the valve. The pressure differential is 800 pounds per square inch. Therefore, the total Bernoulli force at normal flow is

$$F_5 = 0.64 \text{ pounds} \tag{26}$$

Drag forces arise as the spool translates. They may be estimated by

$$F_6 = C'_D A \left(\frac{1}{2} \rho V_o^2\right) \tag{27}$$

where C'_D is the drag coefficient

A is the spool frontal area, square inch

V_o is the spool velocity, inches per second

For a flat plate, $C'_D \sim 1$. The area (both halves of the spool) is about 1.23 square inches. Thus, Equation 27 becomes

$$F_6 = 0.019\ V_o^2 \qquad (28)$$

Inertial force is calculated from the moving mass and its acceleration. The weight of the moving parts in the solenoid valve is about 0.325 pounds. Hence,

$$F_7 = \frac{0.325}{386} A \qquad (29)$$

where A is acceleration in inches per square second

The motion of the valve may be computed by balancing forces:

$$F_1 = F_4 + F_5 + F_6 + F_7 \qquad (30)$$

or

$$A = \frac{386(F_1 - F_4 - F_5 - F_6)}{0.325} \qquad (31)$$

It will be recalled that F_6 is velocity-dependent. However, $V = 0$ at $T = 0$. In program GSFCO5 an initial acceleration is calculated, then time is allowed to increment. A value for V is established by a numerical integration. This in turn permits calculation of a new acceleration; this iteration is continued until the spool position (found by integrating the velocity) shows the valve to be fully open. Using appropriate constants, this time has been calculated to be seven milliseconds.

The actual operating time will be somewhat in excess of this calculated value because of particular characteristics of the solenoid. When a solenoid is operated on alternating current, as this one is, it would chatter at the times that the applied voltage sevings through zero were it not for an additional feature, called a shading coil, which causes a force component which is out of phase with that of the main coil. When the main coil is deenergized its magnetic field collapses. As it does so, current is induced in the shading coil (which is short-circuited on itself); this results in the closure being sustained a few milliseconds longer than would otherwise be the case. As a consequence, when a normal solenoid valve is deenergized there should be a period somewhat in excess of seven milliseconds between the opening of the switch and the mechanical transient which is generated by the moving mass hitting its stop.

The closure time calculated is that expected from a valve in good condition. Fouling by contaminants in the hydraulic oil will slow the valve.

Spool wear and scoring will increase leakage. This probably will not be detectible acoustically, since the leakage velocities will be very low. If the wear and scoring increase the roughness of the spool, the increased friction will tend to increase closure times. This may be nonlinear; i.e., the friction may be velocity-dependent.

Where there is energy stored in the system because of air in the lines the problem is different. This situation has been studied and tested by assuming that the air is stored in an accumulator of given volume. Initially, the full accumulator pressure appears across the valve, allowing high flow rates. The high flow velocities through the valve generate large Bernoulli forces, in such a direction as to oppose the spring force as in Figure 7. As a result, the valve cracks open until Bernoulli and spring forces balance. Drag forces are negligible since the valve velocity is very low. As the accumulator discharges, the Bernoulli force decreases and the valve opens wider. Finally, the accumulator completely discharges and the spring drives the valve spool to its full open position.

Define Y as the valve opening, as in Figure 7.

$$Q = UA \tag{32}$$

where $A = \pi D_1 Y$

D_1 = spool outer diameter

Combining Equations 23 and 32:

$$Q = \left(\pi D_1 C_D \sqrt{\frac{2}{\rho}}\right) \sqrt{P}\, Y \tag{33}$$

Calling the term in parenthesis C_2:

$$Q = C_2 \sqrt{P}\, Y \tag{34}$$

Combining Equations 33 and 23 in Equation 22:

$$F_5 = 2\pi D_1 Y C_D^2 P \cos 69^\circ \tag{35}$$

The Bernoulli force must be equal to the difference between spring and friction forces:

$$F_5 = F_1 - F_4$$

or

$$F_5 = (30\hat{X} + 4.75) - 2.05 \tag{36}$$

Now $X = X_0 - Y$. X_0 is the spool position at the time the valve starts to open. Therefore, considering Bernoulli forces from both halves of the valve,

$$2\,(2\pi D C_D^2 \cos 69^\circ)\, YP = 30\,(X_0 - Y) + 2.70$$

Calling the quantity in parenthesis C_1,

$$Y = \frac{30X_o + 2.70}{2PC_1 + 30} \tag{37}$$

Initially, P is known. Y is calculated by Equation 37. From this, by Equation 33, the flow through each half of the valve can be calculated. After an increment of time the volume of fluid in the accumulator has decreased by $Q\,\Delta t$. P, the pressure applied to the valve by the accumulator, is now recalculated.

Before being installed in the hydraulic loop the accumulator has a volume V_T completely filled with gas at pressure P_1. After application of a pressure P at the liquid connection ($P > P_1$), the volume of gas becomes

$$V_G = \frac{P_1 V_T}{P} \tag{38}$$

The fluid volume is

$$V_F = V_T - V_G = V_T \left(1 - \frac{P_1}{P}\right) \tag{39}$$

The flow through the valve decreases the fluid volume to a new value, V'_F.

$$V'_F = V_T \left(1 - \frac{P_1}{P}\right) - Q\,\Delta t \tag{40}$$

This leads to a new gas volume, V'_G.

$$V'_G = V_T - V'_F \tag{41}$$

The new pressure applied by the accumulator is

$$P' = \frac{P_1 V_T}{V'_G}$$

$$= \frac{P_1\ V_T}{V_T - \left[V_T\,(1 - P_1/P)\right] - Q\,\Delta t}$$

$$P' = \frac{P_1\ V_T}{\frac{P_1}{P}\ V_T - Q\,\Delta t} \tag{42}$$

One can now go back to Equations 33 and 37, recalculating Y and Q, and a new $Q\,\Delta t$. The iteration is continued until the accumulator is fully discharged. The parameters of the solenoid valve are such that the valve opening during accumulator discharge is very small, a few thousandths of an inch. Consequently, to the time for accumulator discharge must be added a time delay in closing due to inertial effects as previously calculated. These calculations have been programmed in GSFC19. If the accumulator volume is assumed to be one quart, with a precharge of 1200 psi, the anticipated opening time totals approximately 240 milliseconds.

Servo Valve

The servo valve is a four-way electrohydraulic device used as an element of a closed-loop servo. A torque motor operates a jet-pipe hydraulic preamplifier which actuates the second-stage spool. The spool is controlled by a null-balance force-feedback servo.

The servo valve has the response characteristics of a low-pass filter. As such, it may be analyzed by the techniques used to calculate the response of electrical filters. Such a filter has a time delay and a rise-time when presented with a unit step-function. The response to such a transient can be calculated from the known amplitude and phase characteristics of the filter. These are shown in Figure 8 as given by the manufacturer's literature and as measured by Goddard on a particular valve. This analysis follows the treatment of the same problem by Guillemin (Ref. 6).

Consider a linear low-pass filter with input terminals 1 and output terminals 2. The output may be expressed as:

$$e_2\,(t) = \int_{-\infty}^{\infty} g\,(\omega)\,h\,(\omega)\,e^{j\omega t}\,d\omega \tag{43}$$

where $g(\omega)$ is the Fourier Transform of the input time-function

$h(\omega)$ is the input-output transfer function of the device

The input time-function $f(t)$, which is a unit step-function, can be expressed as:

$$f(t) = 0 \quad \text{for } t < 0$$

$$f(t) = 1 \quad \text{for } t > 0$$

$$g(\omega) = \frac{1}{2\pi}\int_{-\infty}^{\infty} e^{j\omega t}\,dt$$

$$= \frac{1}{j2\pi\omega} \tag{44}$$

The transfer function can be written in polar form:

$$h(\omega) = H(\omega)\,e^{-j\theta(\omega)} \tag{45}$$

$H(\omega)$, the amplitude response, and $\theta(\omega)$ the phase response, are known quantities, either supplied by the manufacturer or measured on a test stand.

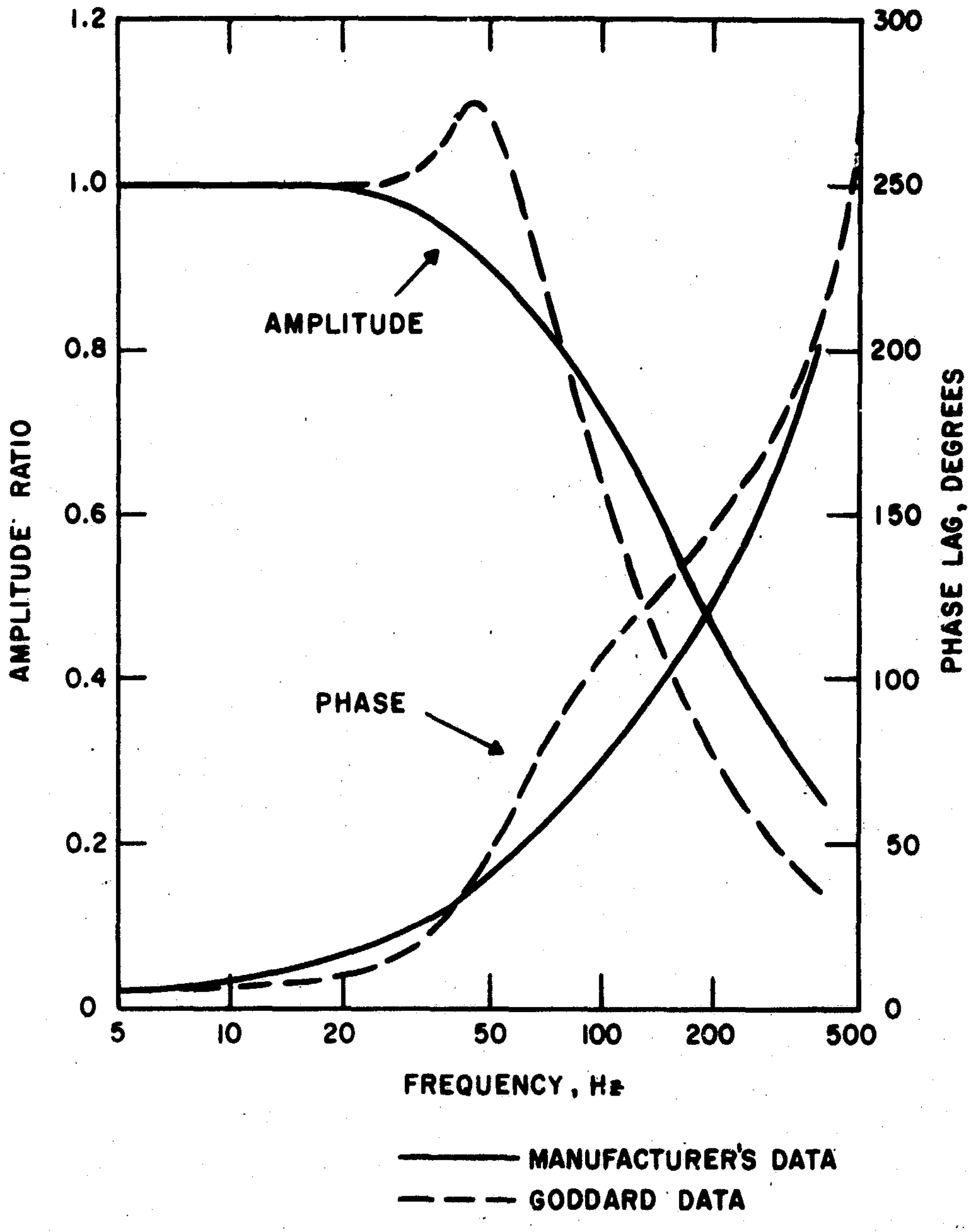


Figure 8. Amplitude and Phase Responses of a Servo Valve

Then:

$$e_2(t) = \frac{1}{2\pi j} \int_{-\omega_1}^{\omega_1} H(\omega) \frac{e^{j(\omega t - \theta(\omega))}}{\omega} \, d\omega \tag{46}$$

where ω_1 is a very high frequency. Separating Equation 4 into real and imaginary components:

$$e_2(t) = \frac{1}{2\pi j} \left[\int_{-\infty}^{\infty} \frac{H(\omega) \cos\left[\omega t - \theta(\omega)\right]}{\omega} \, d\omega \; t{+}j \int_{\omega_1}^{\omega_1} \frac{H(\omega) \sin\left[\omega t - \theta(\omega)\right]}{\omega} \, d\omega \right. \tag{47}$$

Consider the first intergral. $H(\omega)$ is an even function and $\theta(\omega)$ is an odd function, so the integrand must be odd. The integrand (with symmetrical limits) would therefore disappear except for the pole at the origin. In the vicinity of the origin $\theta(\omega)$ and ω approach zero, and $H(\omega)$ approaches 1, so the first integral can be rewritten as:

$$\frac{1}{2\pi j} \int_{-\rho}^{\rho} \frac{d\omega}{\omega} \quad \text{where } \rho \to 0 \tag{48}$$

This has a singularity at the origin. Calculating the residue:

$$\text{Res}(0) = \lim_{\omega \to 0} \frac{d\omega}{\omega} = 1$$

In Reference 7, Hildebrand's Theorem IV (p. 530):

$$\lim_{\rho \to 0} \int f(Z) \, dZ = j\alpha \left[\text{Res}(a)\right]$$

where: a is the value of Z having a singularity and α is the intercepted angle at $Z = a$. α is positive for integration in the ccw direction.

Therefore:

$$\int_{-\rho}^{\rho} \frac{d\omega}{\omega} = j\pi$$

and Equation 48 becomes equal to one-half.

$$e_2(t) = \frac{1}{2} + \frac{1}{2\pi} \int_{-\omega_1}^{\omega_1} H(\omega) \frac{\sin\left[\omega t - \theta(\omega)\right]}{\omega} \, d\omega$$

$$= \frac{1}{2} + \frac{1}{\pi} \int_{0}^{\omega_1} H(\omega) \frac{\sin\left[\omega t - \theta(\omega)\right]}{\omega} \, d\omega \tag{49}$$

The response of the servo valve to a step-function input has been computed by program GSFC18, using both the manufacturer's data and that measured by Goddard (Figure 9). While the results disagree in detail, they both show that some five to sever milliseconds is required for the valve to fully respond. The fact that a finite response is predicted at zero time is due to slight errors in the data.

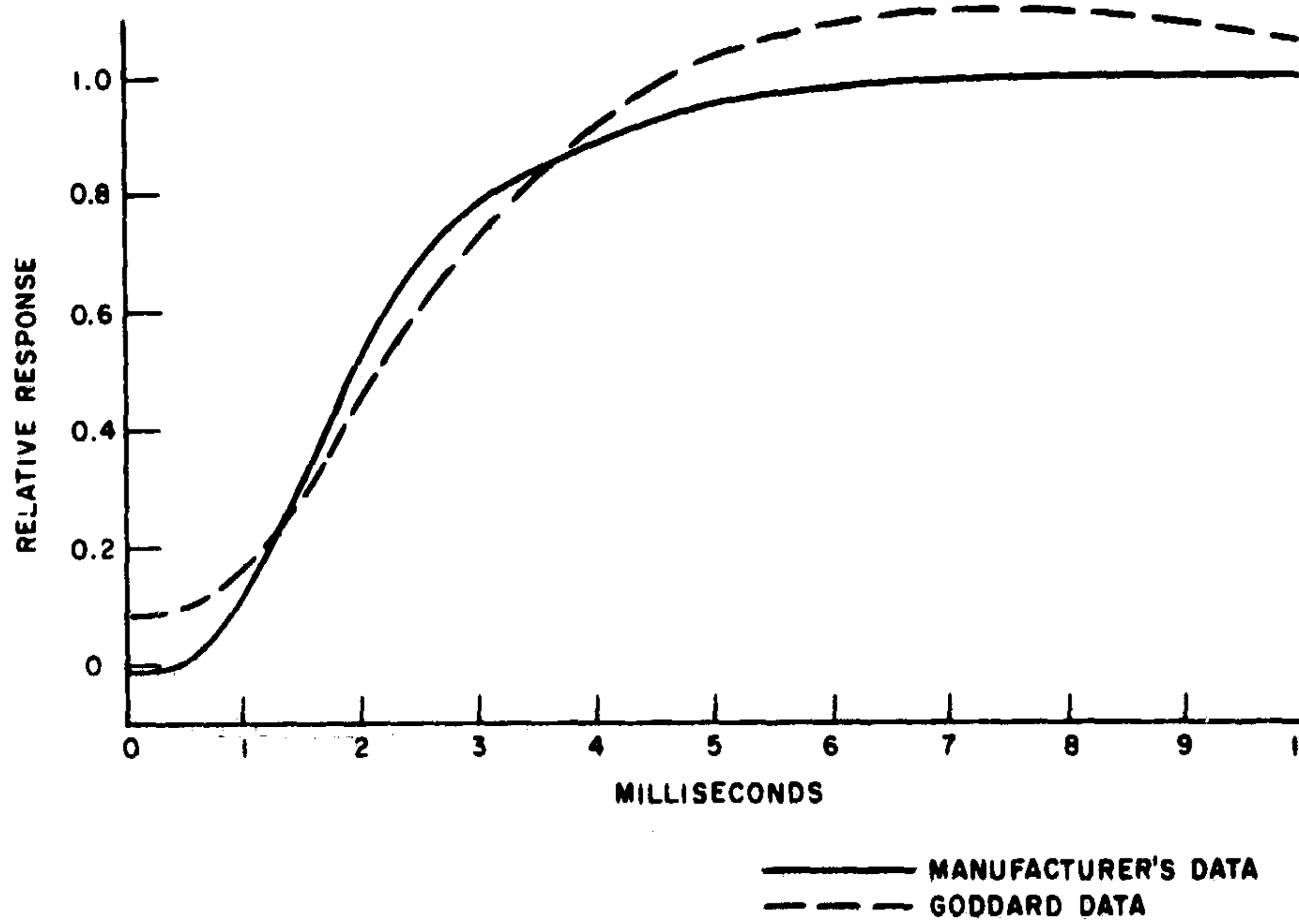


Figure 9. Response of a Servo Valve to a Step-function Input Signal

Flow noise is the only signal generated by the servo valve. The turbulent flow excites the case, where it may be sensed by an accelerometer. When the valve is closed the only flow is due to leakage, plus a very small flow in the hydraulic preamplifier, but as the valve opens the amplitude of the flow noise will increase. When the noise amplitude reaches a steady value the spool may be assumed to be in the opened position. If a step-function is applied electrically with the spool in the neutral position, the variation in flow-noise with time indicates the time required to actuate the valve.

Contamination of the valve will lead to stickiness. This will tend to slow the valve response and to introduce stick-slip friction. The latter will cause the valve to move in jerks rather than a smooth motion. In such cases, the flow-noise is expected to increase in amplitude stepwise. Wear and scoring of the spool are expected to decrease the valve frequency-response, which in turn leads to slower response to a transient. Scoring, wear, or excessive tolerances will increase internal leakage.

Pressure Relief Valve

The pressure relief valve is shown in cross-section in Figure 10. An orifice is assumed to be the load. Details of the construction are shown in Figures 11, 12 and 13. When the line pressure exceeds a preset level, the pilot or ball valve begins to open.* Part of the line pressure appears across the pilot valve, and part across the piston because of head loss in the capillary. This drop is in such a direction as to force the piston upward, thereby opening the bypass valve. The flow bypass decreases the line pressure. The process continues until equilibrium is reached.

The pressure at which the relief valve regulates is set by a balance of forces. The various parameters are identified on the appropriate figures.

Balance of forces on the piston:

$$H_1 R \pi (R_5^2 - R_6^2) = K_5 (X_6 + S) \tag{50}$$

Balance of forces on the ball valve:

$$(H_0 - H_1) R \pi R_2^2 = K(X_0 - X_2) \tag{51}$$

In the usual situation $X_6 >> S$, $X_0 >> X_2$. With this assumption, these equations may be combined:

$$\left(H_0 - \frac{K_5 X_6}{R \pi (R_5^2 - R_6^2)} \right) R \pi R_2^2 = K X_0 \tag{52}$$

or

$$H_0 = \frac{K X_0}{R \pi R_2^2} + \frac{K_5 X_6}{R \pi (R_5^2 - R_6^2)} \tag{53}$$

The line pressure for which the regulator is set is:

$$P_0 = H_0 R = \frac{K X_0}{\pi R_2^2} + \frac{K_5 X_6}{\pi (R_5^2 - R_6^2)} \tag{54}$$

In order to calculate the dynamic response of the relief valve, as in program GSFC14, it is useful first to develop expressions relating to several key components of the valve.

Capillary Tube (Figure 12). If it is assumed that the flow through the tube, Q, is known, then the velocity is:

$$V_1 = \frac{Q}{\pi R_3^2} \tag{55}$$

*This valve actually uses a cone-type rather than a ball-type pilot valve. Although the analysis was started before this was known, this should not have any effect on the conclusions reached.

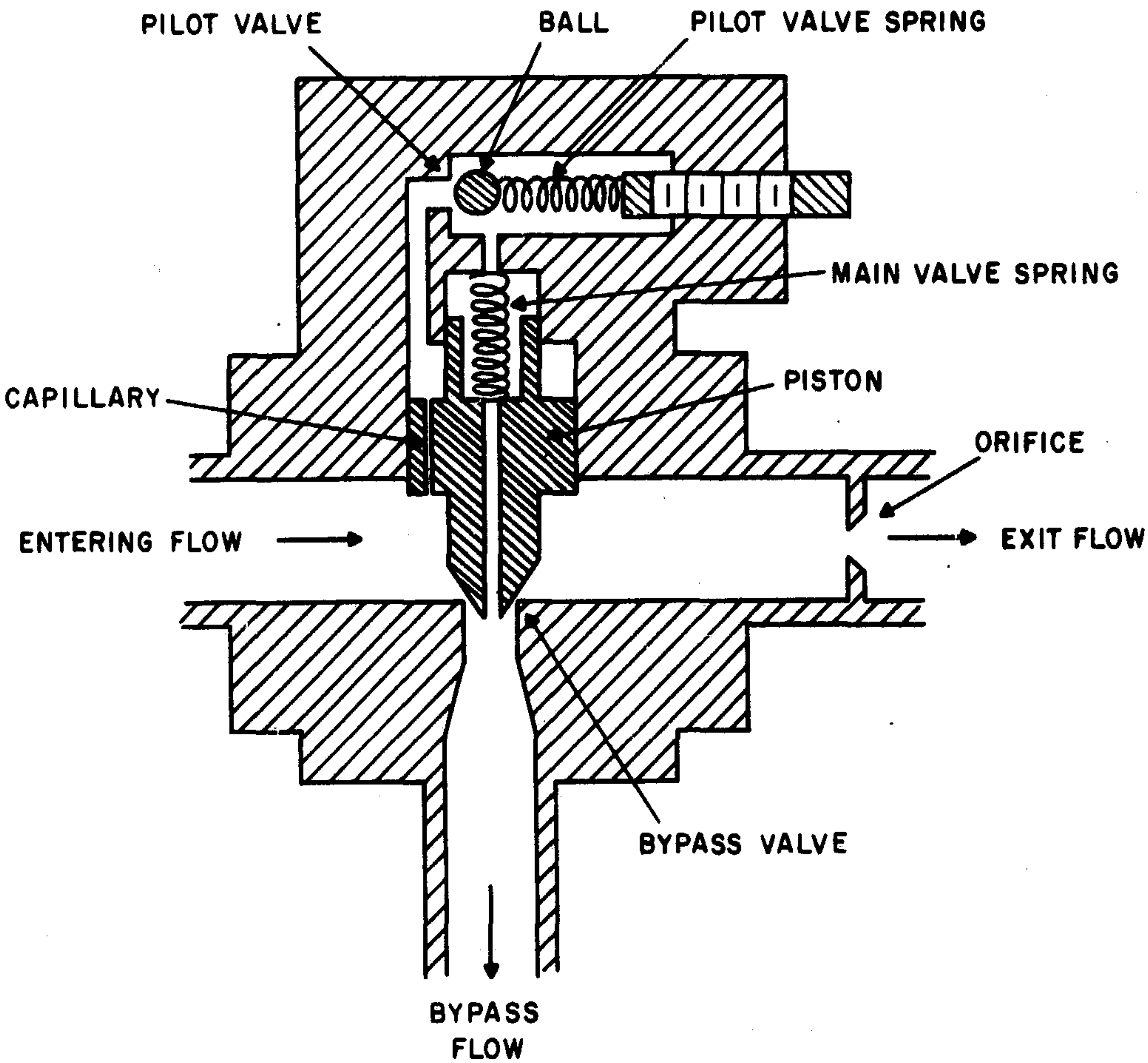


Figure 10. Cross-section of a Pressure Relief Valve

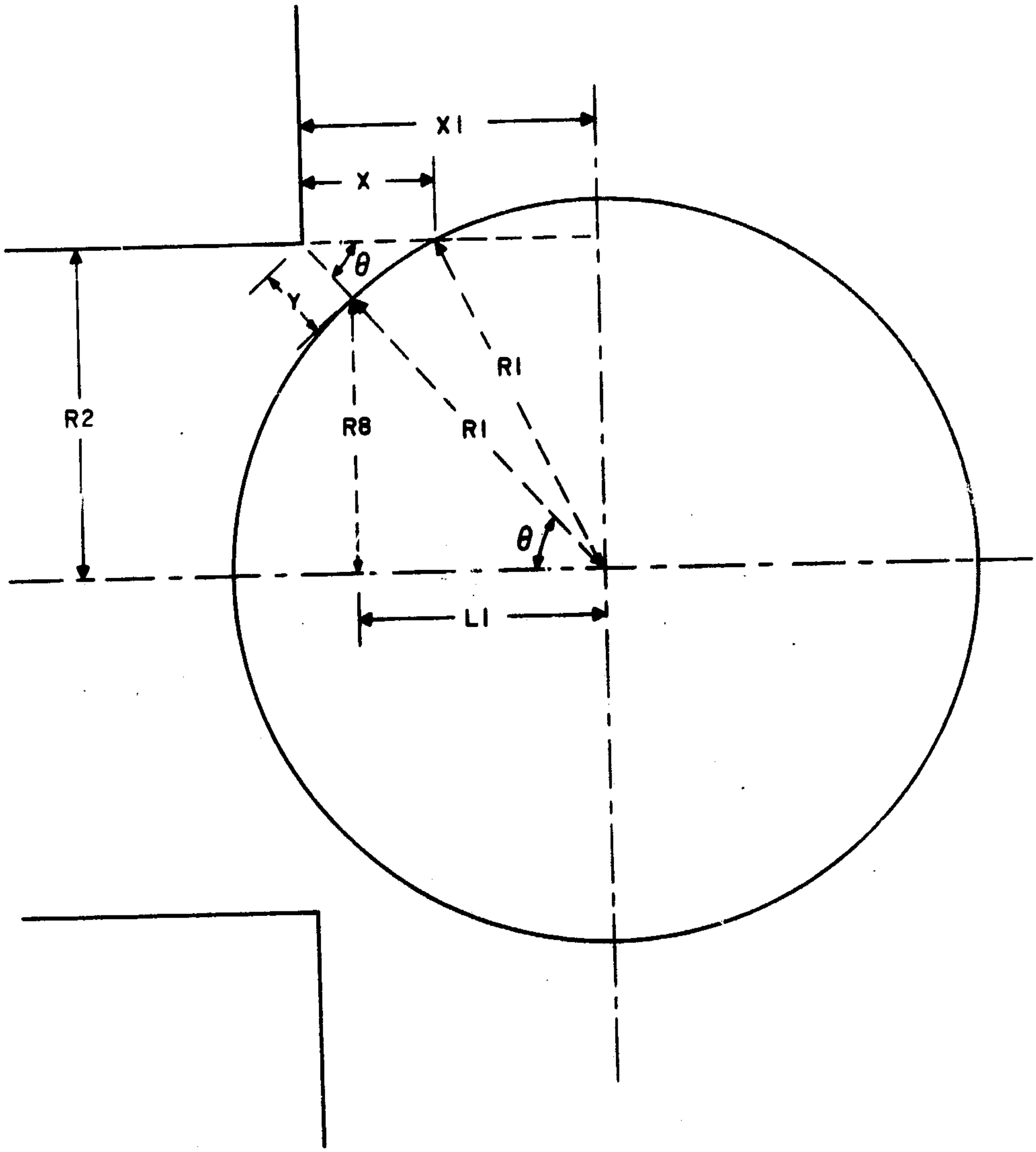


Figure 11. Details of a Pilot Valve

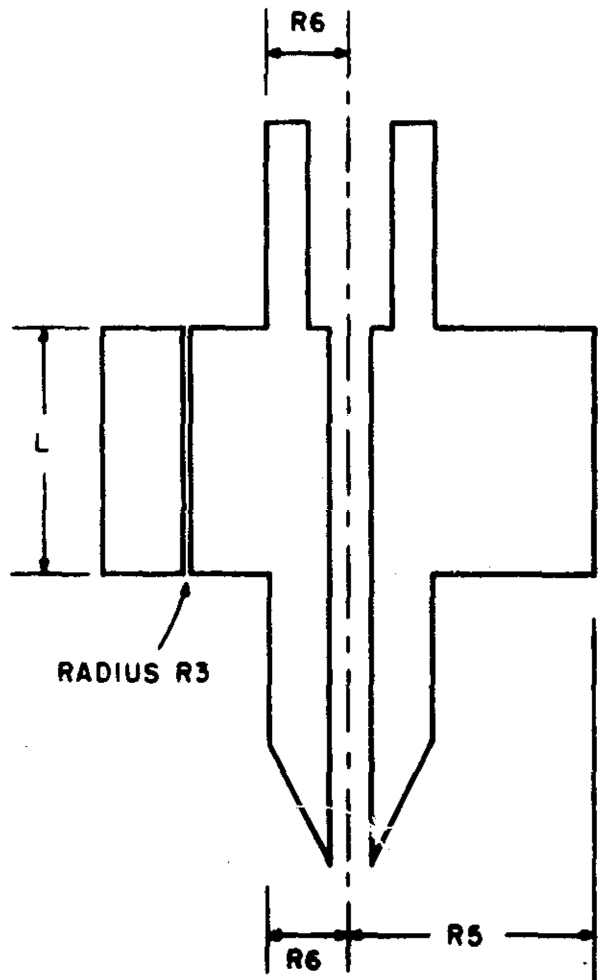


Figure 12. Details of Piston

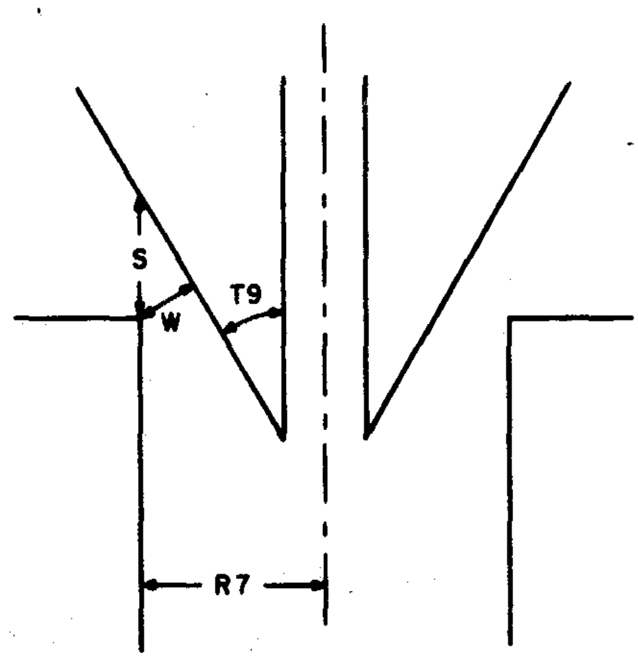


Figure 13. Details of Bypass Valve

Computing the losses due to this flow gives the entering loss:

$$H' = \frac{0.5\ V_1^2}{2 \cdot 32.2}$$

The Reynolds Number, N, is determined from V_1. From N are calculated the the friction factor, F, and then the friction loss.

$$H'' = \frac{F\ L}{2\ R_3} \cdot \frac{V_1^2}{2 \cdot 32.2}$$

The exit loss is:

$$H''' = \frac{V_1^2}{2 \cdot 32.2}$$

Three components are added to get the total head loss in the capillary:

$$H_1 = \left(1.5 + \frac{FL}{2R_3}\right) \frac{V_1^2}{2 \cdot 32.2} \qquad (56)$$

Ball Valve (Figure 12.) The drop across the ball valve is the difference between line pressure and the capillary drop:

$$H_2 = H_0 - H_1 \qquad (57)$$

The flow through the orifice of area A_2 can then be calculated:

$$Q_2 = 0.65\ A_2 \sqrt{2 \cdot 32.2\ H_2} \qquad (58)$$

Therefore,

$$A_2 = \frac{Q_2}{0.65\sqrt{2 \cdot 32.2\ (H_0 - H_1)}} \qquad (59)$$

Referring to Figure 12,

$$\frac{R_3}{R_1} = \frac{R_2}{R_1 + Y}$$

$$Y = \frac{R_1 R_2}{R_3} - R_1 \qquad (60)$$

Now the area of a sector of a circle of radius, r, and arc length, b, is

$$\frac{b \cdot r}{2}$$

Therefore,

$$A_2 = 2\pi \cdot R_2 \left(\frac{R_1 + Y}{2}\right) - 2\pi \cdot R_3 \left(\frac{R_1}{2}\right)$$

or

$$\pi R_1 R_8^2 + R_8 A_2 - \pi R_1 R_2^2 = 0$$

$$R_8 = \frac{-A_2 + \sqrt{A_2^2 + 4\pi^2 R_1^2 R_2^2}}{2 \pi R_1} \tag{61}$$

Knowing R_8,

$$L_1 = \sqrt{R_1^2 - R_8^2} \tag{62}$$

Now
$$\theta = \tan^{-1} \frac{R_8}{L_1} \tag{63}$$

and

$$(X_1 - X) = \sqrt{R_1^2 - R_2^2} \tag{64}$$

$$X_1 = \frac{R_2}{\tan \theta} = \frac{R_2 L_1}{R_8} \tag{65}$$

Therefore,

$$X = \frac{R_2 L_1}{R_8} - \sqrt{R_1^2 - R_2^2} \tag{66}$$

This value of X is based on the valve area required to pass a given flow. by Equation 51,

$$X_2 = \frac{(H_o - H_1) R \pi R_2^2}{K} - X_0 \tag{67}$$

This value of X_2 is based on pressure drop across valve. Note that X and X_2 must be equal, since they are both measures of the same quantity.

Bypass Valve (Figure 13).

$$\frac{Y}{S} = \cos (T_9)$$

$$Y = S \cos (T_9) \tag{68}$$

For small values of S, the valve open area is approximated by

$$A = 2\pi R_7 Y$$

$$= 2\pi R_7 S \cos (T_9)$$

The flow that is bypassed can be calculated from this area and the line pressure H_o.

$$Q_7 = 0.7 \ A\sqrt{2 \cdot 32.2 \ H_0} \qquad (69)$$

or

$$Q_7 = 0.7 \cdot 2\pi \ R_7 \ S \cos (T_9)\sqrt{2 \cdot 32.2 \ H_0} \qquad (70)$$

Orifice (Figure 10). The flow passed through the orifice is calculated from its radius R_4 and the pressure head H_0 of the line.

$$Q_0 = 0.65 \ \pi \ R_4^2\sqrt{2 \cdot 32.2 \ H_0} \qquad (71)$$

or

$$H_0 = \left(\frac{Q_0}{0.65 \ \pi \ R_4^2}\right) \cdot \frac{1}{2 \cdot 32.2} \qquad (72)$$

These expressions are combined in the computer program in the following sequence, in order to calculate the response of the valve to a transient. The bypass is initially assumed closed; line pressure (as a function of time) is determined from the moment of release.

1. At t = 0, calculate a very rough approximation to the pilot flow by assuming that the ball valve is wide open and all the line pressure must therefore appear across the capillary. Taking into account only entrance and exit losses, a value for Q is calculated. This must be larger than the true value of Q because several losses were neglected.

2. The basis of this calculation of Q, new values of Q are chosen which increment upward toward this value. The assumed Q establishes a flow velocity in the capillary so that friction in the capillary can be considered. Calculate the drop in the capillary, H_1, thereby defining H_2, the drop across the ball valve. H_1 establishes the pressure differential, and hence the fluid pressure on the piston as well.

3. Balance the forces on the piston: fluid pressure, spring force, drag, and acceleration. Calculate piston acceleration and displacement. The latter defines the bypass valve opening. This and the pilot flow give a new flow to the load, and therefore a new value for line pressure. Displacement also defines Q_8, the flow displaced by the piston, which must dump through the ball valve.

4. The flow through the ball valve is the sum of the capillary flow and the flow displaced by the piston. Use these to calculate X, the ball valve opening.

5. The pressure across the ball valve is the line pressure less the capillary drop. Use this to calculate X_2, the ball valve opening.

6. A solution is taken to exist if $X \sim X_2$. If this is not the case, another and slightly larger value of Q is chosen in step 2, and the process repeated.

7. After a solution is found at t = 0, time is incremented and the entire sequency repeated.

In order to run the program a variety of valve dimensions are required. Since many of which were not available, they were estimated. The resulting predictions of line pressure are not correct in absolute terms, but it is believed that they are a reasonable qualitative indication of the way the valve responds. In particular, they show that the valve responds to the assumed transient by regulating to its preset pressure without overshoot (Figure14).

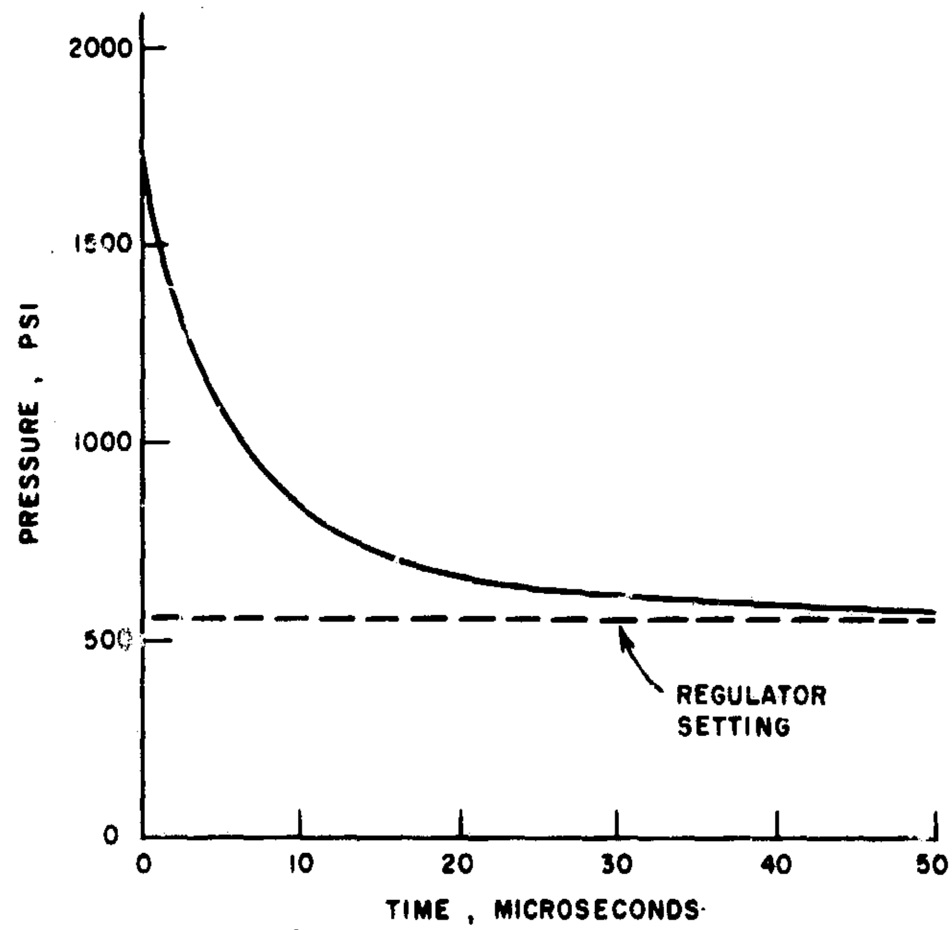


Figure 14. Predicted Response of Pressure Relief Valve

The pressure is approached asymptotically. Consequently it may be concluded (in agreement with the manufacturer) that the relief valve normally operates without fluctuating pressure or chatter. The only signal from a normal valve will be relatively steady broad-band noise, due to flow. It follows that fluctuating flow noise, chatter, and impacting in the valve are manifestations of instability and are indicative of valve malfunction. The manufacturer's literature says chatter is due to damage of the piston or valve seat, or dirt in the poppet valve.

The above calculations assume that:

1. All the inertia is effectively in the main valve assembly.
2. Propagation time of pressure disturbances is negligible.

The first assumption is a good one, as is demonstrated below. The force on the ball is about as follows:

$$F \cong \Delta P \cdot \pi \cdot 0.1^2$$

If $\Delta P \sim 800$, then $F \sim 24$ pounds

Ball volume $= \frac{4\pi}{3} r^3 \sim 4 \times 10^{-3}\ in^3$

Its weight is:

$$4 \times 10^{-3} \cdot \frac{1}{1728} \cdot 500\ \frac{lb}{ft^3} \sim 10^{-3}\ \text{pounds}$$

$$\frac{F}{W} \sim \frac{24}{10^{-3}} \sim 24 \cdot 10^3\ \frac{\text{pound force}}{\text{pound weight}}$$

For the piston --

if $H_1R = P_1 = 100$ psi, the pressure differential

then $100\ \pi(0.5^2 - 0.2^2) = 60$ pounds, the force on the piston

$$\frac{F}{W} = \frac{60}{0.3} = 200\ \frac{\text{pound force}}{\text{pound weight}}$$

The assumption of negligible propagation time is more debatable. The velocity of sound, $c \sim 5000$ feet per second

~ 0.6 inch per microsecond

The calculations show appreciable valve response in one microsecond, which obviously cannot be true since valve dimensions are on the order of an inch or two. On the other hand, the calculated valve motion is relatively low after 20 to 30 microseconds have elapsed. In all probability, the program overestimates the orignial transient correction but is adequate for the latter portions of the regulating process.

Rolling-element Bearings

Stresses are generated in both the races and rolling elements of ball and roller bearings during their normal operation. Within the service life of the bearing these are accommodated with negligible outward effect, but a point is finally reached when these subsurface stresses result in a spall, in which a small amount of surface material "pops out," leaving a rough depression. Once spalling starts, the bearing deteriorates rapidly (Figure 15).

Development of spalls in a properly designed and used bearing would be expected to take a long time, but generation of such local defects may be accelerated by a number of factors, as indicated in Table 4.

Table 4

FAILURE MODES FOR ROLLING ELEMENT BEARINGS

1.	Normal Fatigue	Starts as small flaking at local spots on the rolling surfaces. Develops into erosion of one or several large areas. This is the normal failure mode for a properly used bearing.
2.	External Vibration	Bearings may be damaged both when operating and stationary. Damage usually takes the form of periodic indentation of races.
3.	Overload	Generally accelerates the fatigue failure process.
4.	Electric Current Passages	Can happen in electric machinery. Sudden high currents cause local craters in balls and races. Alternating currents normally produces regular repeating impressions.
5.	Corrosion	Usually starts as local areas with rough surface structure. Develops into roughness and pits distributed over all rolling surfaces.
6.	Lack of Lubrication	Increases temperature and wear of most stressed areas.
7.	Contamination of Lubricant	Increases wear, may cause corrosion, depending on type of contaminant.

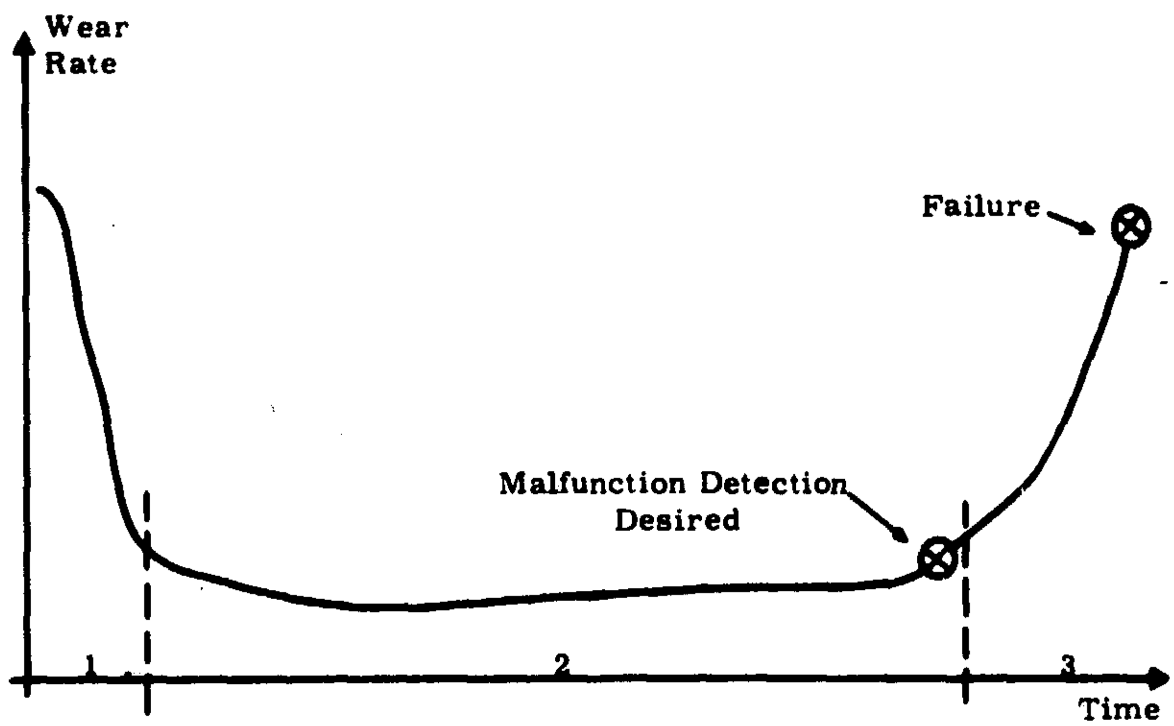


Figure 15. Typical Wear Rate Function for Rolling Element Bearing

If any of the six conditions in Table 4 are present except fatigue, the long low-wear period is usually shortened, but the wear curve is otherwise similar. The essential fact is that once increased wear has started for any reason, the bearing is statistically close to the end of its life.

A malfunction detection system should react when the wear has just started to increase. In most cases the increased wear is physically caused by one or several local defects on either a ball or a race. The defect may be a fatigue crack, a corroded spot, an electric arc spot, etc. Obviously an impact of some sort can be expected every time this defect spot is in contact with another surface in the bearing. The problem is to detect this set of impacts while the defect is still small and local and does not yet affect the general bearing performance. Three fundamental malfunctions should be considered:

- Local defect in outer race
- Local defect in inner race
- Local defect in a rolling element

A local defect is then understood to be a single dent or pit which is so small that bearing performance in terms of drag, alignment, etc., is not affected. Any bearing defect can then be considered a superposition of several fundamental malfunctions.

A defect in the inner or outer race of a bearing generates an impact every time a ball rolls over it. Likewise, if the defect is in a ball surface, an impact is generated every time it hits the inner or outer races. If the bearing rotates with constant speed, each defect generates a regular set of impacts with an "impact repetition frequency" that depends upon bearing speed and location of the defect. Assuming that no sliding occurs in the bearing, those fundamental frequencies can be computed. The result is:

Outer race malfunction $$f_e = \frac{n}{2} f_r \left(1 - \frac{BD}{PD} \cos \beta \right) \tag{73}$$

Inner race malfunction $$f_i = \frac{n}{2} f_r \left(1 + \frac{BD}{PD} \cos \beta \right) \tag{74}$$

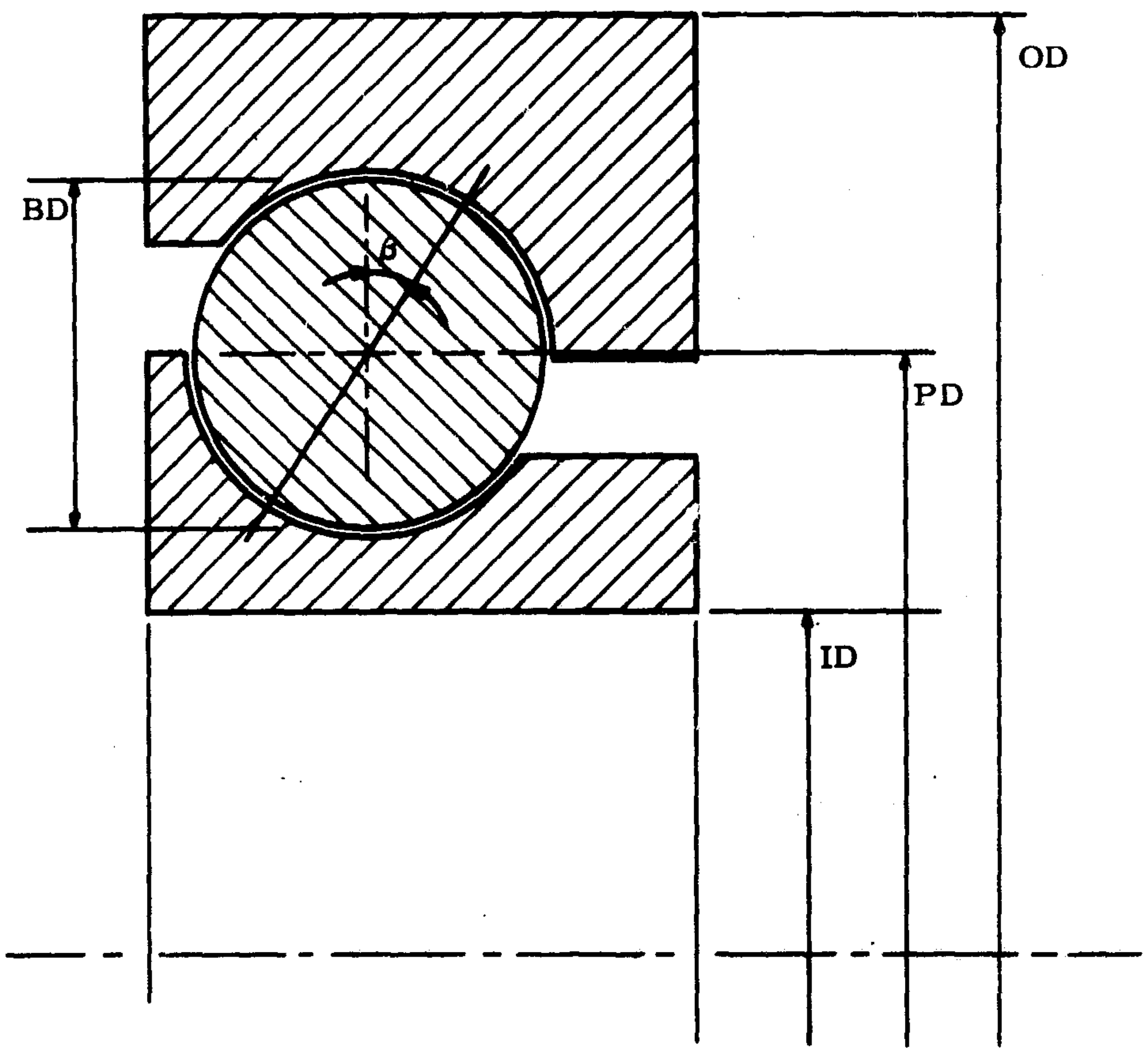
Ball or roller malfunction $$f_b = \frac{PD}{BD} f_r \left(1 - \left(\frac{BD}{PD}\right)^2 \cos^2 \beta \right) \qquad (74)$$

where f_r is the relative speed between inner and outer race (revolutions, per second), and BD, PD, β, and n as defined in Figure 16.

The mechanical signatures which generally can be expected from the three fundamental malfunctions are shown in Figure 17. It is assumed in the figure that the bearing has a stationary outer race and a rotating inner race, and a radial load which is fixed relative to the outer race. It should be noted that all impacts are not of the same height. The impact size must depend upon the load where the impact is generated and how both inner race and the balls rotate relative to the load direction. Each impact is followed by a "ringing". i. e., a structural response which depends more upon the support structure than the bearing itself.

The amplitude of the defect signals will generally be low relative to back ground noise. Summation analysis for detection of low-level signals of this character will be discussed in Section 5 of this report.

The expressions predicting the periodicity of disturbances arising from local defects of the inner race, the outer race, or the rolling elements has been programmed for the computer (program BRGMFN). It has been run with the use of appropriate constants for the servo pump roller bearing, the servo pump ball bearings, and the vane-pump needle bearing. The print-outs are shown as Table 5.



OD = Outer Diameter
ID = Inner Diameter
PD = Pitch Diameter
BD = Ball Diameter
β = Contact Angle
n = Number of Balls

Figure 16. Basic Ball Bearing Dimensions

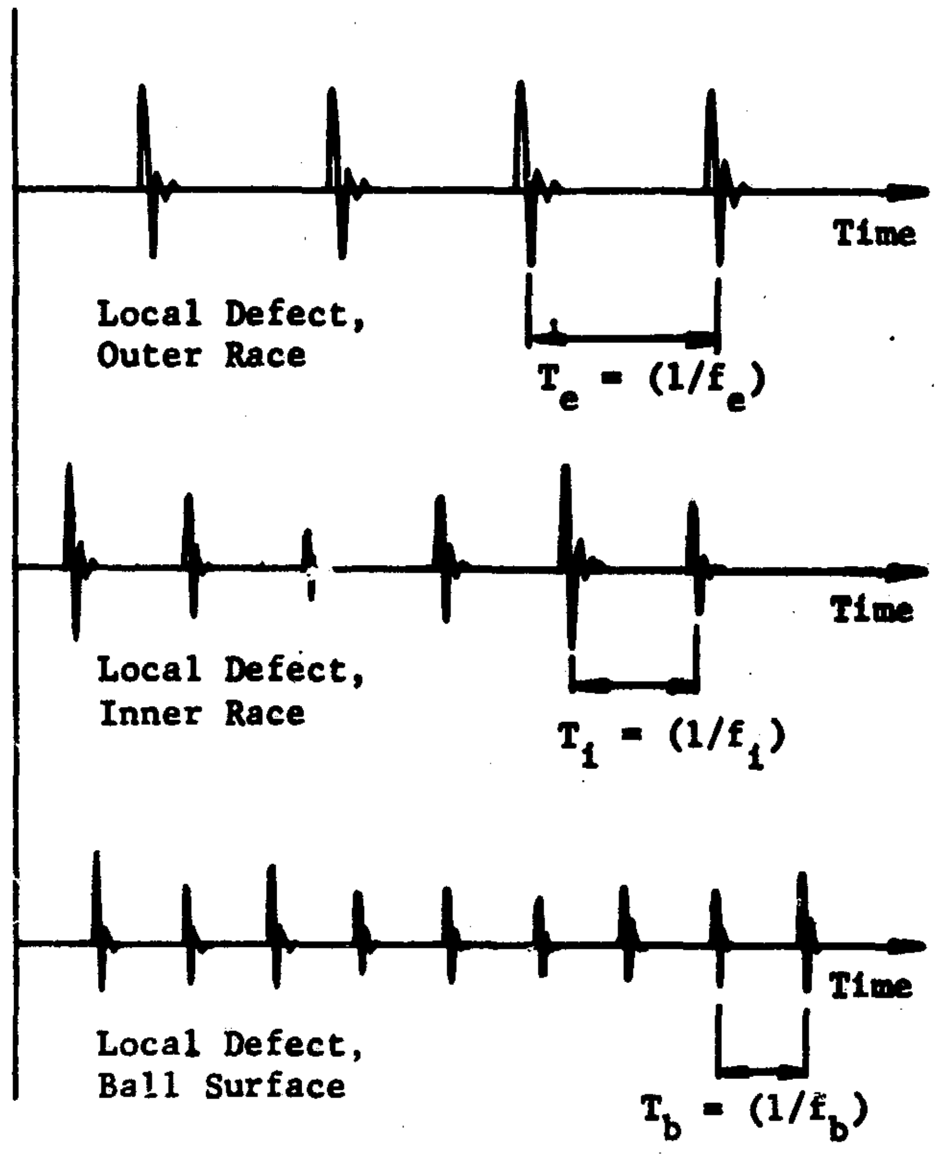


Figure 17. Bearing Defect Signatures

Table 5

PREDICTED PERIODICITIES OF BEARING DEFECT SIGNATURES

```
10 REM   SERVO PUMP ROLLER BEARING
RUN


BRGMFN     10:50    AUGUST 30,1968

WHAT ARE INNER RACE RPM, OUTER RACE RPM, NO. OF ROLLING
ELEMENTS, ROLLING ELEMENT DIA., PITCH DIA., AND CONTACT
ANGLE?1175,0,27,.5,5.9312,0

INNER RACE RPS= 19.58
PERIOD, MSEC= 51.06
CAGE RPS= 8.97
PERIOD, MSEC= 111.53
ROLLING ELEMENT RPS= 115.33
PERIOD, MSEC= 8.67
                              DEFECT LOCATION
               INNER RACE     OUTER RACE     ROLLING ELEMENTS
FREQ.,HZ        286.66         242.09         230.65
PERIOD, MSEC.   3.49           4.13           4.34


TIME:    2 SECS.
```

```
10 REM   SERVO PUMP THRUST BEARINGS
RUN


BRGMFN     10:52    AUGUST 30,1968

WHAT ARE INNER RACE RPM, OUTER RACE RPM, NO. OF ROLLING
ELEMENTS, ROLLING ELEMENT DIA., PITCH DIA., AND CONTACT
ANGLE?1175,0,10,.5,2.936,15

INNER RACE RPS= 19.58
PERIOD, MSEC= 51.06
CAGE RPS= 8.18
PERIOD, MSEC= 122.23
ROLLING ELEMENT RPS= 55.94
PERIOD, MSEC= 17.88
                              DEFECT LOCATION
               INNER RACE     OUTER RACE     ROLLING ELEMENTS
FREQ.,HZ        114.02         81.81          111.88
PERIOD, MSEC.   8.77           12.22          8.94


TIME:    2 SECS.
```

Table 5 (Cont'd)

PREDICTED PERIODICITIES OF BEARING DEFECT SIGNATURES

```
1 REM  VANE PUMP NEEDLE BEARING
RUN



BRGMEN      15:34    ST WED 10/09/68

WHAT ARE INNER RACE RPM, ØUTER RACE RPM, NØ. ØF RØLLING
ELEMENTS,RØLLING ELEMENT DIA., PITCH DIA., AND CØNTACT
ANGLE? 1175,0,33,.0625,.6875,0

INNER RACE RPS= 19.58
PERIØD, MSEC= 51.06
CAGE RPS= 8.9
PERIØD, MSEC= 112.34
RØLLING ELEMENT RPS= 106.82
PERIØD, MSEC= 9.36
                                     DEFECT LØCATIØN
                  INNER RACE         ØUTER RACE       RØLLING ELEMENTS
FREQ.,HZ           352.5              293.75           213.64
PERIØD, MSEC.      2.84               3.4              4.68



RAN       3.59 SEC
```

Section 3

EXPERIMENTAL INVESTIGATIONS

TEST PLAN

An experimental program was designed to confirm the accuracy of the analytical work on signature prediction and to devise test procedures for detection of incipient malfunctions in the identified components of an actual installation. The program had four parts.

First, preliminary vibration measurements were made on the USB antenna at Goddard Space Flight Center, as a means of establishing the approximate vibration levels on the various components, the general character of the signals, and the degree of cross-talk which could be expected from one component to another. This information was used to guide the development of the rest of the test plan. One conclusion reached was that vibration levels were moderate (on the order of 1 g); hence exceptional system sensitivity was not required. The predominant vibration signal was that of the servo pump, consisting mainly of multiples of seven times the running speed. While some traces of this vibration were noted on components other than the servo pump, these could be eliminated by operating the pump in the yoke-centered condition which eliminates pumping.

It was also found that servo pump noise was mainly below 2000 hertz, so that its effects could also be minimized by high-pass filtering. Since valve noise was found to be mainly at high frequencies, above 2000 hertz, such filtering could be put to practical use. Vane pump vibration appeared to be independent of the servo pump yoke position, so that its signature could be measured with the servo pump set for quiet operation.

Test procedures and techniques could not be developed conveniently on the USB antenna at Goddard because of its heavy commitments for other purposes. Tests on a special facility offered more freedom and flexibility. Consequently, such a facility was designed and built at the General Electric Research and Development Center, using hydraulic components supplied by Goddard Space Flight Center. The facility was designed in such a way that each component was operated under conditions of pressure and flow which closely approximated those on an actual installation. It was arranged so that, to a large degree, the various components could be exercised independently.

This facility was first operated using new hydraulic components in an as-delivered condition. Test procedures were developed and the analytical predictions of vibration characteristics tested. When it was felt that a sufficient understanding had been reached, the hydraulic components were returned to Goddard, where arrangements were made for their disassembly and for introduction of defects. These defects were so designed that they would not compromise the performance of the component to the point where it would fail to normally perform its hydraulic function. On the other hand, their nature and extent were to be such that if they were noted during a teardown inspection, remedial action would be called for.

After the defects were introduced, the components were returned to the Research and Development Center for re-test. These tests were designed to establish test techniques suitable for defect detection, and to show that the defects could be detected.

Finally, the components with introduced defects were sent back to Goddard, where they were installed on the USB antenna of the Network Test and Training Facility. Analyses of the vibration signals were then made using electronic instrumentation owned by Goddard. These tests established that tests on the laboratory facility at the Research and Development Center and on an actual installation were equivalent. It further demonstrated the suitability of the Goddard instrumentation for such analyses, and showed that almost all of the introduced defects could be detected by the developed techniques and tools.

DEFECTS INTRODUCED

Several specific defects were intentionally introduced into the hydraulic components by Goddard personnel (except as noted). In addition, a defective vane pump sent in from the field was made available for test. A description of these defects follows.

Servo Pump

The nominal clearance between each piston and its cylinder is normally 0.0010 inch. This was increased to 0.0014 inch in one of the seven by turning down the piston. The actual clearance in the other six was not established.

While not a mechanical defect, cavitation was induced in the servo pump as an example of an unsuitable condition. This was accomplished by reducing its supercharge.

Vane Pump

The vane pump was intentionally damaged by scoring the edges of four consecutive vanes in the control portion only. This destroyed the quality of the seal between these vanes and the ring.

A damaged ring from a control pump was also tested. Damage was detected by Goddard through oil analysis and confirmed by a teardown inspection. Figure 18 is a photograph of the damage, which consisted of a

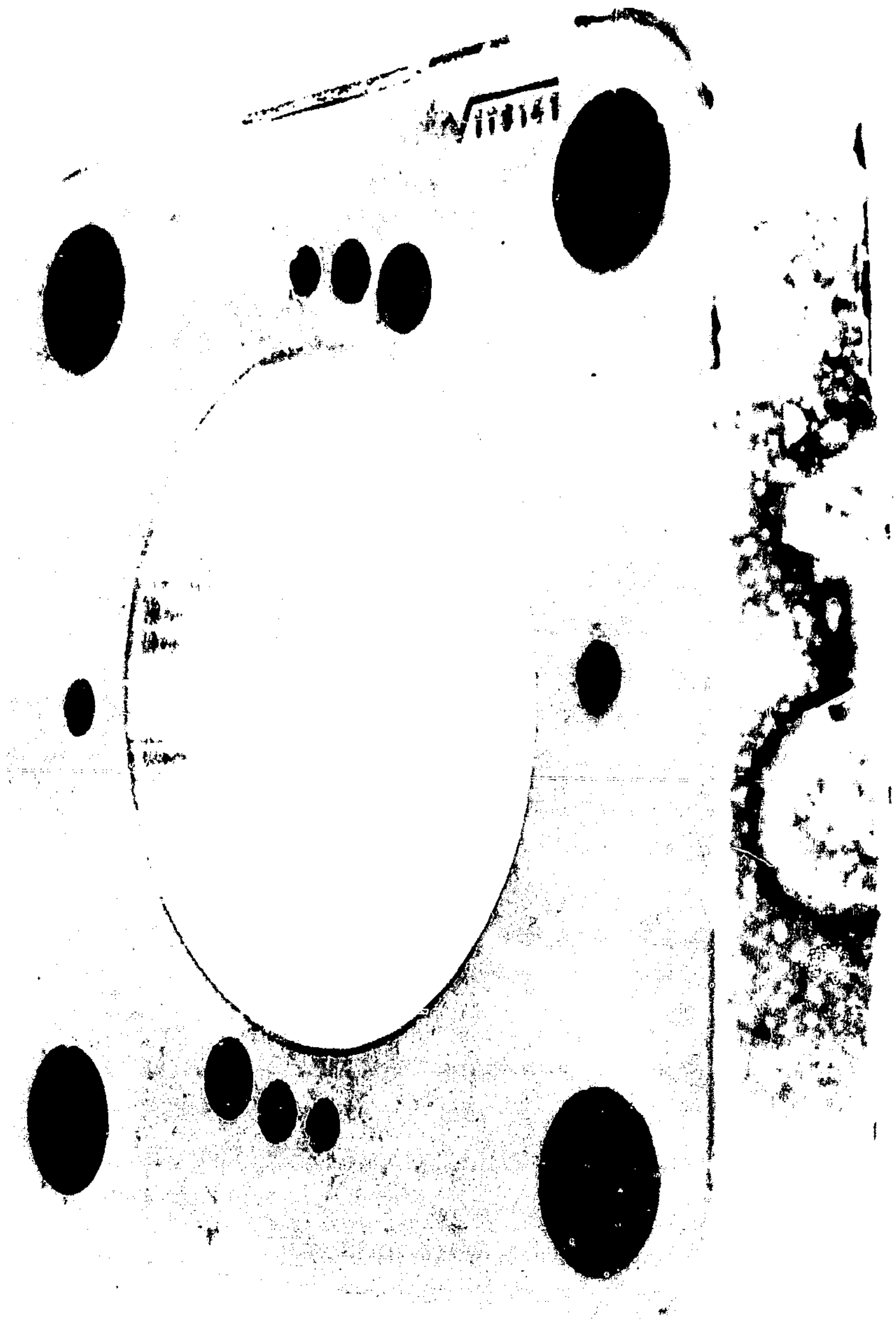


Figure 18. Defective Ring, Control Portion of Dual-vane Pump

series of ridges on the inner surface of the ring. A second damaged area, not visible in the picture, was diametrically opposite that which can be seen. A possible explanation for the damage is partial blockage of the oil passages at the base of the vanes. This would restrict their freedom of motion and increase wear on the ring.

Solenoid Valve

The spool of the solenoid valve was scored. This was expected to increase leakage and add stiction.

Relief Val.e

The seat of the poppet, or pilot, valve was scored.

Servo Valve

The servo valve was sent to the manufacturer for its modifications, which included increase of its internal leakage. Figures 19 and 20 are V-curves run by Goddard on this valve before and after the defect was introduced. Prior to modification leakage was low, but there was some offset. The manufacturer corrected the offset and increased the leakage by a factor of about three.

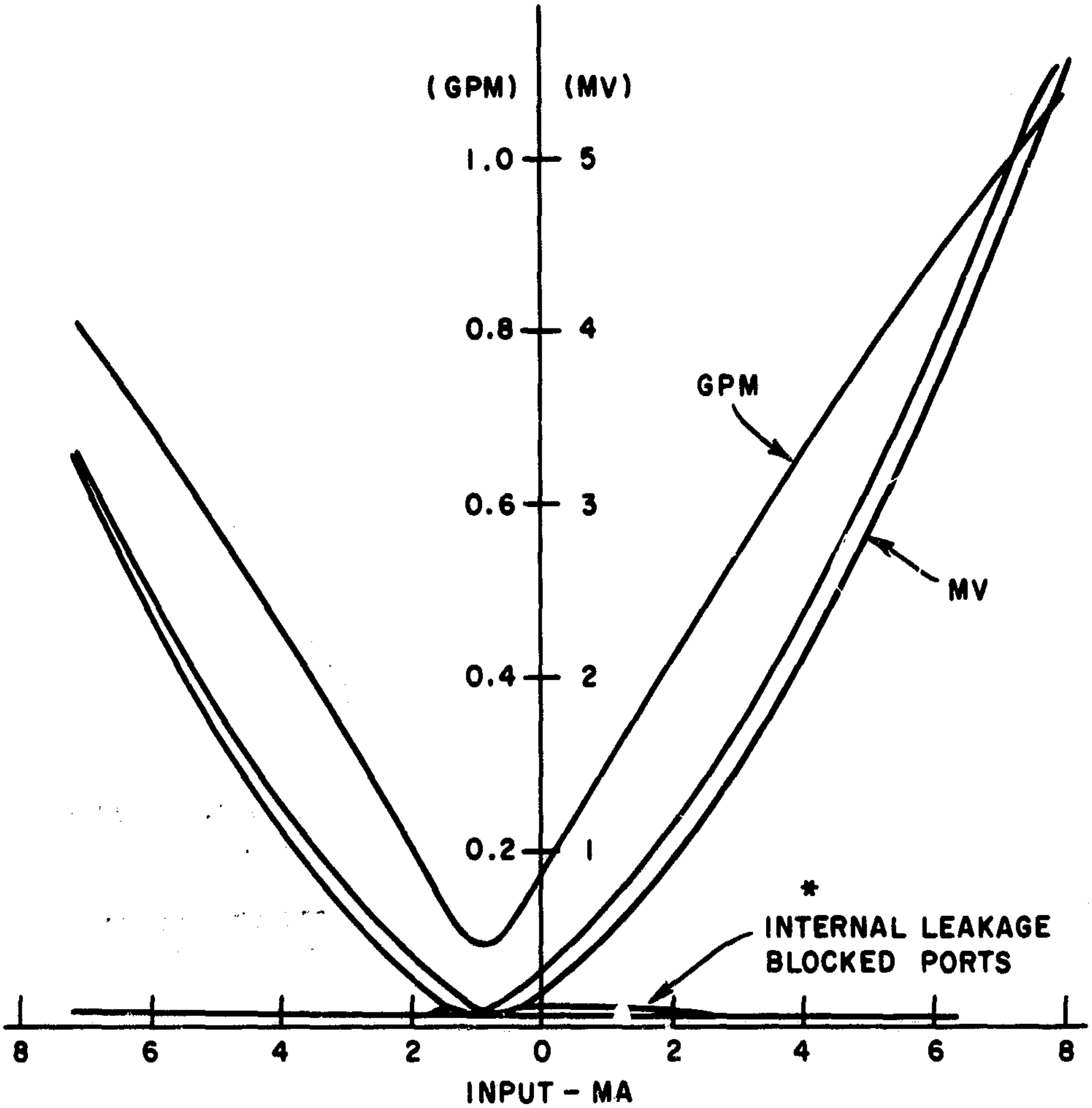
Bearings

Local defects simulating spalled races were introduced by the Research and Development Center. A high-powered pulsed laser was used to erode the material, as follows.

Servo Pump Roller Bearing. Three defects, graded in size, were added 0.25 inch apart, as sketched in Figure 21. They were so oriented that, in normal rotation, first the smallest spall would be excited by a roller, then the next largest, and finally the largest. Since the circumferential spacing between rollers is 0.75 inch, this arrangement results in a continuous sequence of impacts, all equally spaced in time. The largest defect was about 0.5 inch across, an estimated 40 mils wide, and 10 mils deep. The second was 0.5 inch across, about 15 mils wide, and 2 mils deep. The smallest was like the second but only about 0.25 inch across.

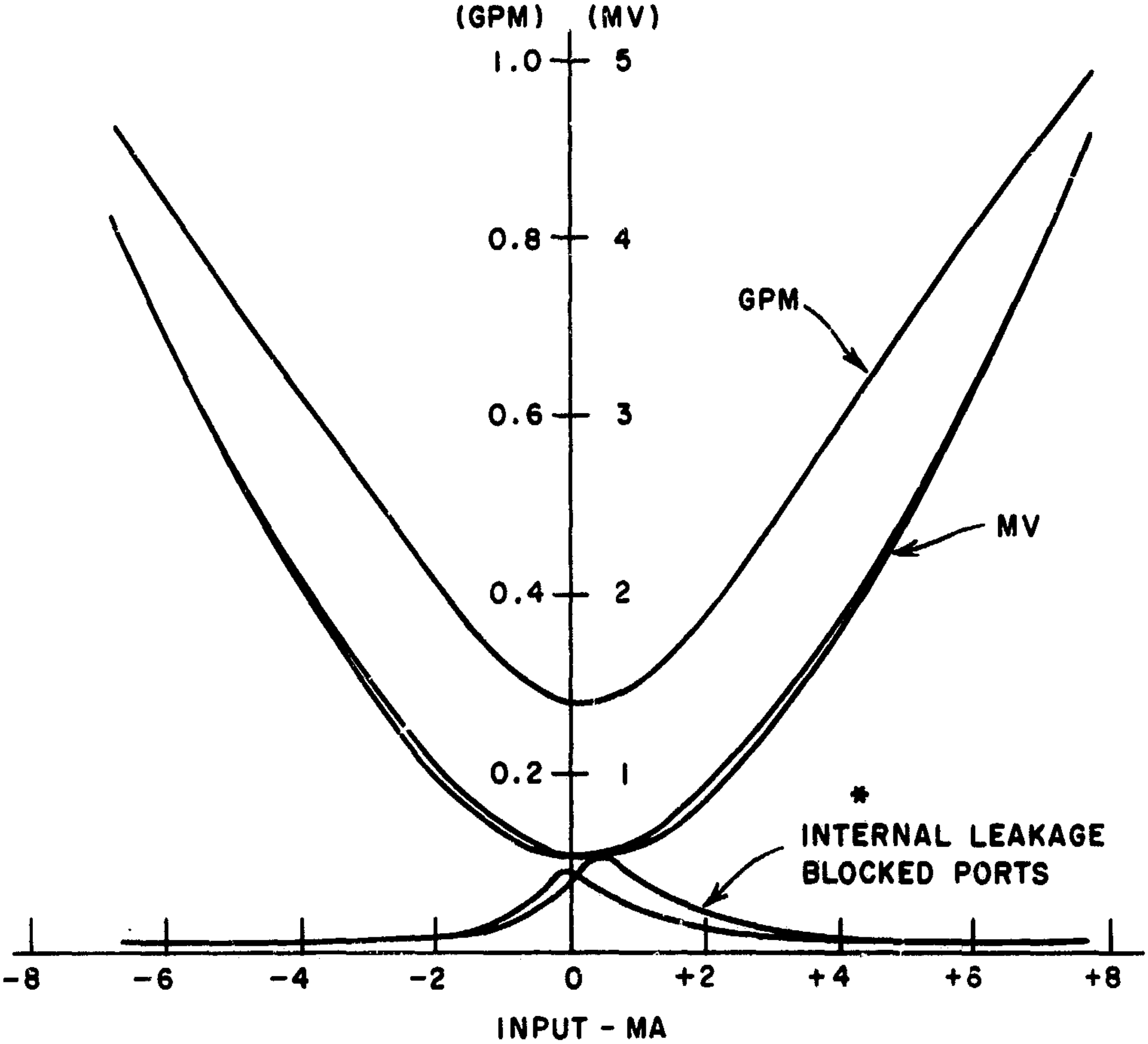
Servo Pump Ball Bearing. The servo pump also includes a duplex pair of ball bearings. The outer race of one bearing had a defect introduced, shown in Figure 22 at a magnification of 47.5. It measured about 60 mils across, 30 mils wide in the direction of ball travel, and 40 mils deep. It was centered over the normal ball track, which could be seen as a very light burnishing of the race.

Vane Pump Needle Bearing. The shaft of the vane pump serves as the inner race of the needle bearing. A defect was introduced which was the



* MAX. LESS THAN 0.1 GPM

Figure 19. V-curve of Servo Valve as Initially Received (Supply Pressure 1000 psi)



* MAX. AT NULL = 0.28 GPM
OFF NULL = LESS THAN 0.1 GPM

Figure 20. V-curve of Servo Valve After Modification (Supply Pressure 1000 psi)

same length as one of the needles, 10 mils wide, and 20 mils deep (Figure 23). The intent was to orient the defect parallel to the axis of the shaft; this is the expected orientation of a natural spall. The actual defect was skewed as sketched in the figure.

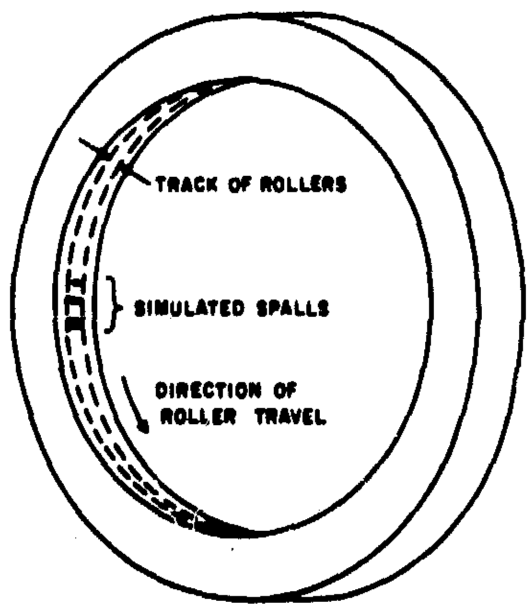


Figure 21. Orientation of Roller Bearing Simulated Spalls

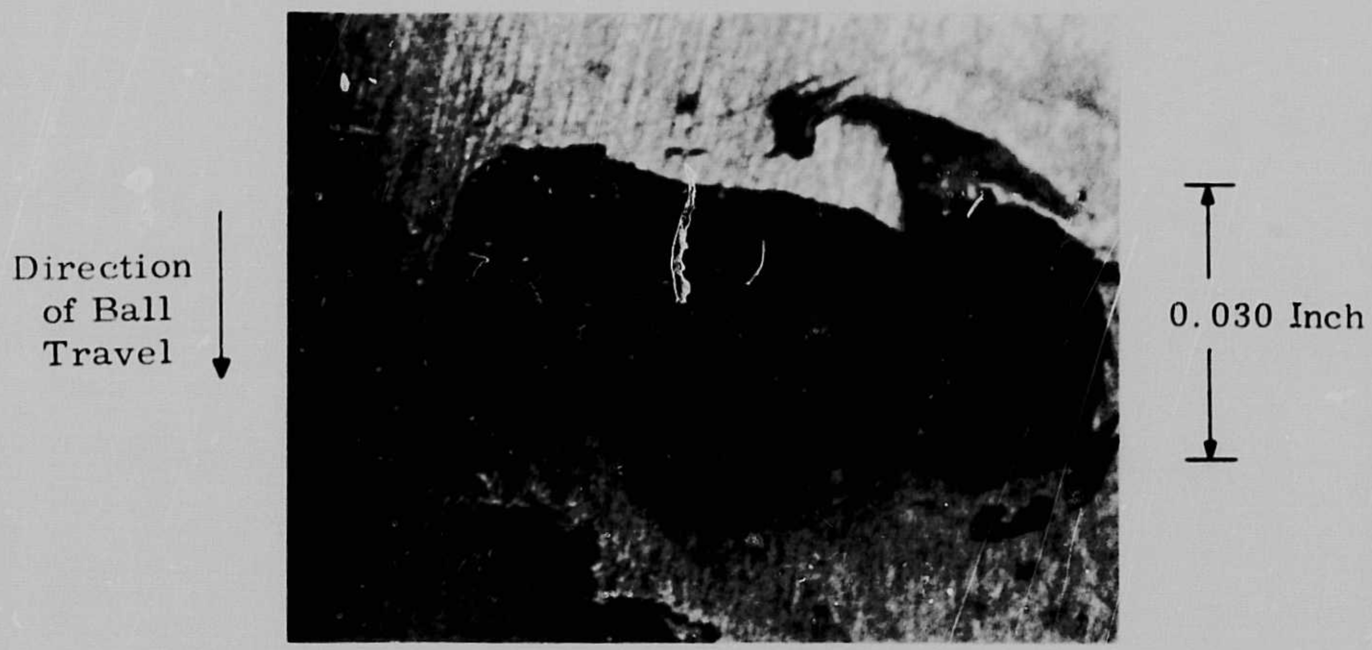
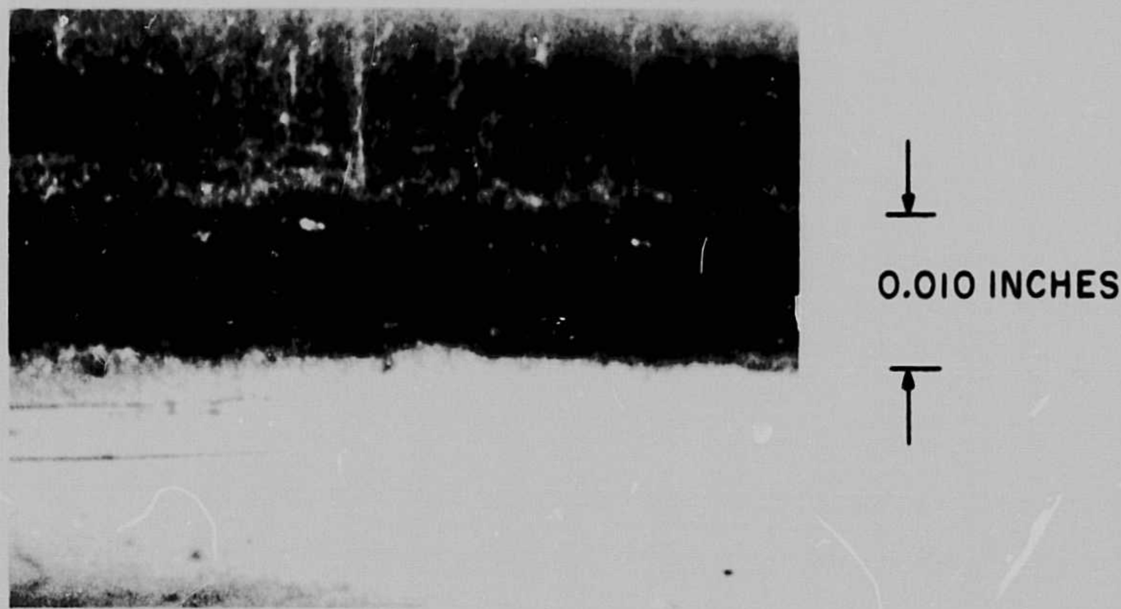
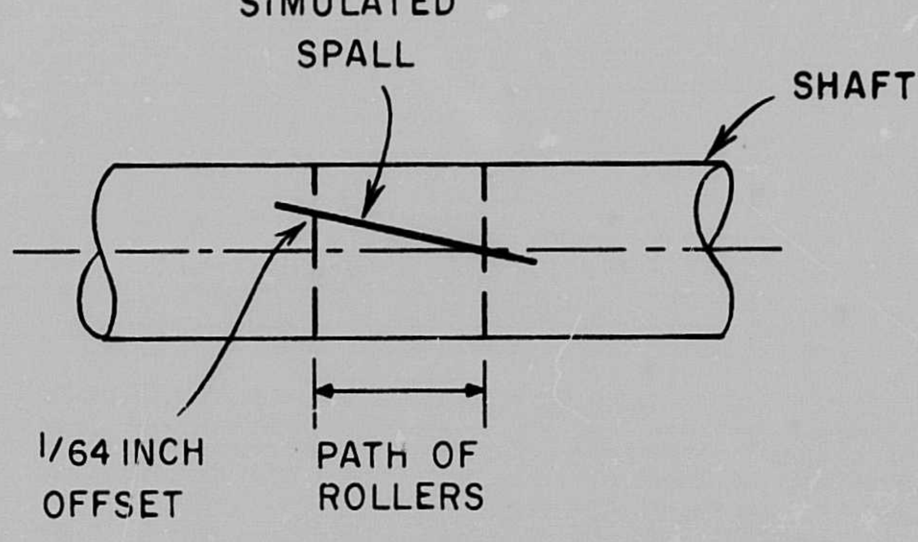


Figure 22. Appearance of Simulated Spall in Servo Pump Ball Bearing



APPEARANCE



ORIENTATION

Figure 23. Vane Pump Needle Bearing Simulated Spall

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Section 4

DATA ACQUISITION

TRANSDUCERS

The characteristics of the signals to be analyzed are such that the transducers chosen must respond linearly over a wide frequency range. Since they must operate on antennas out-of-doors they must be able to withstand rain and moderately wide range of temperatures. Practical considerations of the data acquisition process require that they be able to drive moderate lengths of cable so that the recording or analyzing equipment can be remote from the antenna.

The transducers purcha ed for this project meet all of these requirements. The equipment furnished is the Endevco Model 28450 Underwater Accelerometer System. Being an accelerometer, the transducer has good sensitivity, wide frequency range, and high linearity. The quality which makes this particular system unusual is the fact that it can operate while submerged in salt water. While the present application is less severe, this capability ensures that performance will not be affected by rain, salt spray, or hydraulic fluid. Each system, of which six were supplied, comprises the following:

1. Piezoelectric accelerometer with integral amplifier
2. Cable 25 feet in length (which could be extended to 300 feet).
3. Electronic module for directing + 28 volts direct current to the sensor and relaying the acceleration signal to subsequent instrumentation.

Important system characteristics, excerpted from the manufacturer's literature, are as follows:

Sensor weight: 150 grams

Sensor material: 316 stainless steel

Nominal sensitivity: 100 millivolts per g

Frequency response: 5-10000 hertz ±5 percent

Maximum acceleration: ±100 g peak

Output impedance: 50 ohms in series with 36 mf

Operating temperature range: $+20^{\circ}$ to $+125^{\circ}F$

Nonoperating temperature range: -65° to $+185^{\circ}F$

An additional sensor was added to the system to generate a once-per-revolution trigger pulse. The transducer chosen was the Electro Products Laboratories Model 3015-A, a variable reluctance type, which consists chiefly of a coil wound around a permanent magnet. When the magnetic field is disturbed, as by passing close to a piece of steel, a voltage is generated in the coil. This pickup was installed through an access cover on the electric motor which serves as prime mover. It is placed in such a way that a grease fitting on the motor shaft passes nearby, generating a well-defined pulse each time the shaft revolves.

Transducer-operating Console

An operating console has been built and delivered to make the use of the transducers more convenient. It serves the following purposes:

1. Supplies + 28 volts direct current for energizing the transducer amplifiers
2. Provides a calibrating signal
3. Monitors current drawn by each system
4. Provides necessary connectors to accept sensors and distribute signals

Figure 24 is a photograph of the console. Its circuit diagram is shown in Figure 25. In normal operation the built-in power supply energizes all systems simultaneously. A voltmeter measures the output of this supply, while a milliameter can be used to measure the current (nominally 10 ma) to each system.

The current-measuring switches have spring returns which automatically disconnect the meter when the switch is not purposely depressed. When the operate-calibrate switch is on OPERATE the signal from each transducer is directed to a connector on the rear apron. In the CALIBRATE position the signals are disconnected, and a 60 hertz sinusoid derived from the power line is connected simultaneously to all outputs and to a monitor terminal on the front panel. Since the actual sensitivities of the transducers are all very close to the nominal sensitivity of 100 millivolts per g, the calibration signal may conveniently be set equivalent to some given acceleration level, such as one g RMS.

Transducer Attachment Fixtures

The transducer of the Endevco 2850 system, as purchased, has a stud, 1/2 inch long with 1/4-28 thread, for attachment. This stud has been shortened to just under 1/4 inch in the transducers supplied so as to reduce the depth of thread required for mounting.

Long experience on the part of the General Electric Company has shown that proper mounting of transducer is absolutely essential if the expected

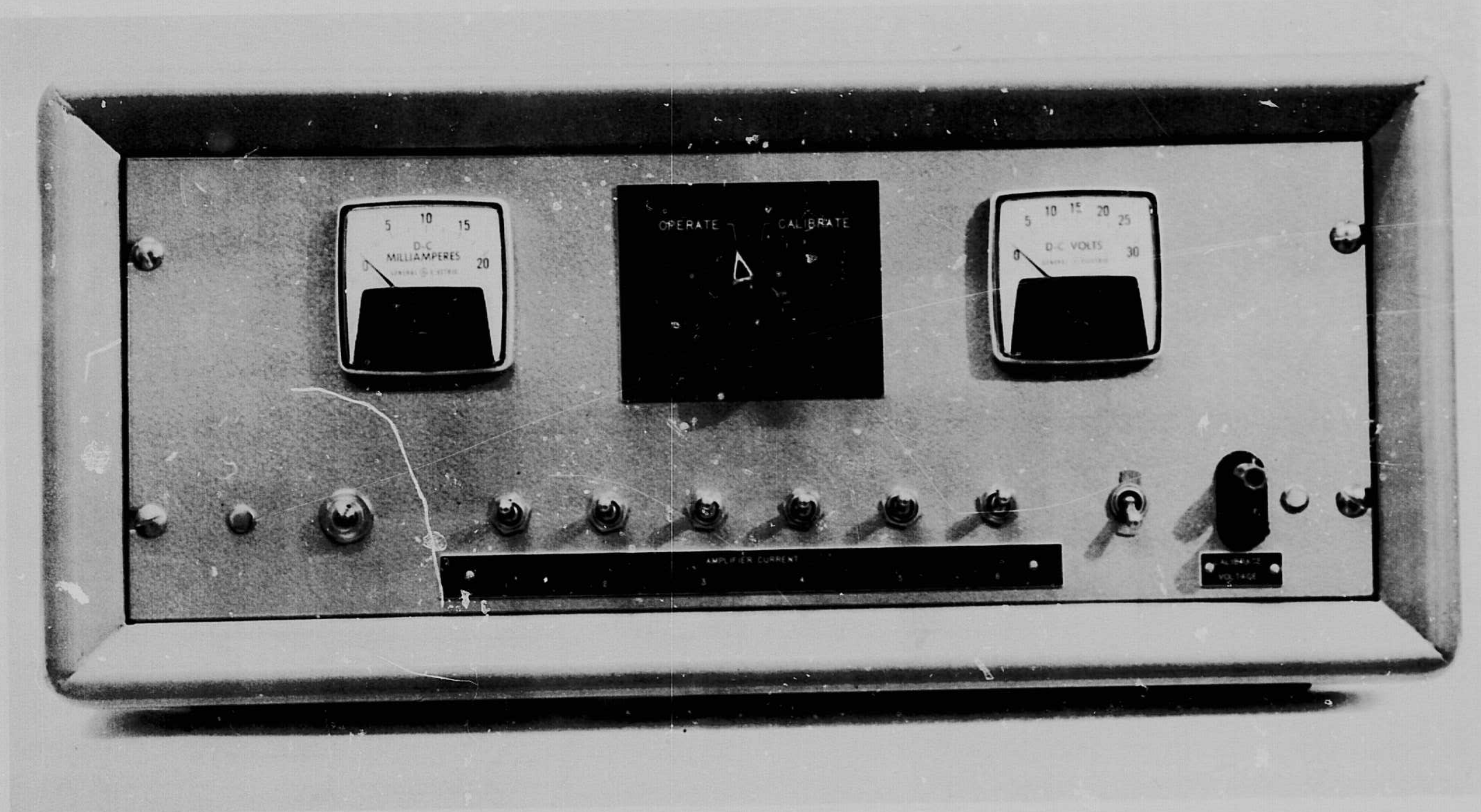


Figure 24. Accelerometer Console

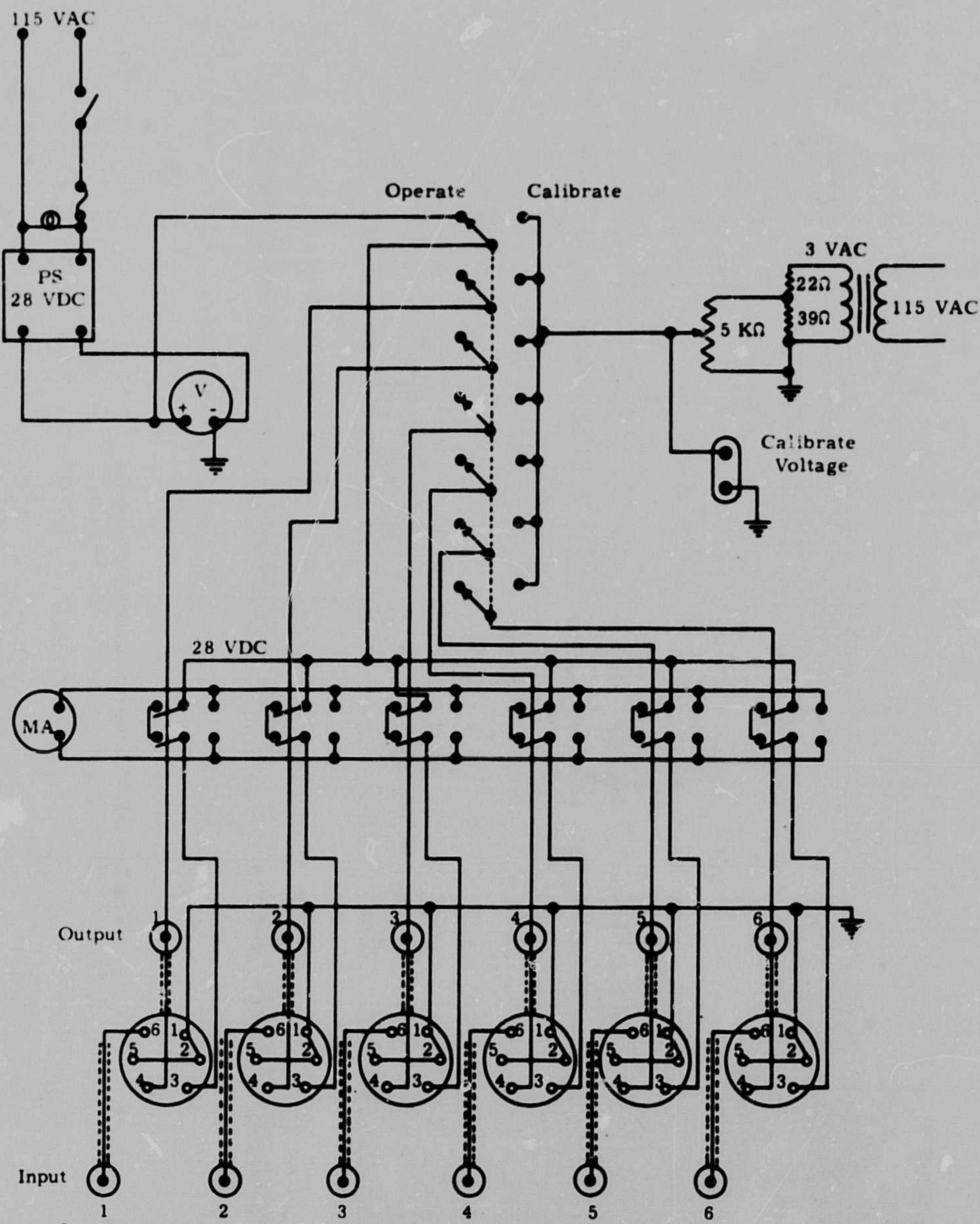


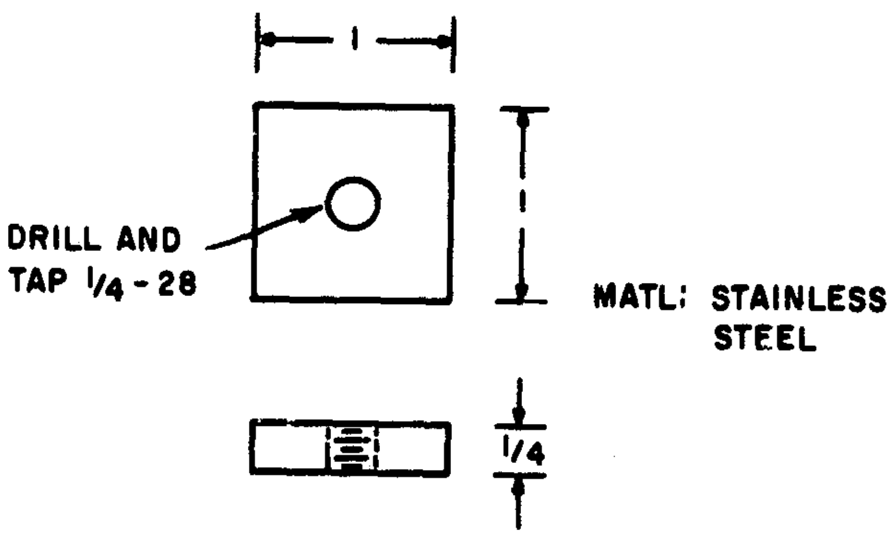
Figure 25. Transducer Console, Circuit Diagram

performance is to be realized. A proper mounting surface has two important qualities: 1) it must be very smooth, flat, and clean; 2) the tapped hole which accepts the transducer stud must be perpendicular to the surface with very small error.

When the transducer is screwed to such a surface (which should be lightly lubricated) the entire flat bottom surface of the transducer mates with the vibrating surface. As the two surfaces strike they immediately "lock"; increasing torque does not result in further rotation of the transducer. This results in minimum strain of the transducer base, which could lead to erroneous signals, and ensures that the vibration surface and the transducer move in unison at all frequencies within the system capabilities.

It is virtually impossible to prepare a suitable surface as described above on a piece of equipment in the field. It is very difficult to achieve even in a factory. As a consequence, it is usually convenient to prepare a mounting block having a suitable surface, and then attach the block to the vibrating surface by means of a cement, as will be described.

The mounting blocks prepared, as shown in Figure 26, measure one inch by one inch by one-fourth inch, and are made of stainless steel to resist corrosion.



NOTE: One surface of block to be specially prepared. Face off, drill, and top in lathe without unchucking, then grind to No. 16 finish.

Figure 26. Accelerometer Mounting Block

The mating surface is turned flat in a lathe, drilled and tapped without being unchucked, then ground flat. Very good results can be obtained when cementing the mounting block to relatively uneven surfaces. Response of the system, when a proper cement is used, is not affected by the cement to frequencies at least as high as ten kilohertz.

Epoxy cements have suitable mechanical characteristics and are perfectly satisfactory, but require four hours or more to harden and cure before measurements can be made. Equally good from a mechanical standpoint but much faster curing (about 15 minutes) is dental cement.* A powder and fluid solvent are mixed to a putty-like consistency and spread on the clean mating surfaces, which are then pressed together. The resulting bond is very hard but fairly brittle, so that the mounting block can be broken free with a sharp blow if necessary.

Transducer Attachment Locations

Transducer attachment locations which proved useful in these tests were as follows.

Servo Pump. There is one pickup on the roller bearing outer race, as shown in Figure 27. This location can be used to monitor vibrations of this bearing and general vibration due to pulsations in delivered flow. A second location is on the port block. This also can be used to monitor vibration due to pulsations, and is a good position for sensing vibrations of the duplex-pair thrust bearings.

Dual-vane Pump. One pickup is located on the cover and one on the outer body, as seen in Figure 28 and identified in Figure 5-22 of instruction manual ME-1528. These locations allow one to monitor the vibration of each portion of the dual pump. The bearing vibration is best sensed by a transducer directly over the bearing on the inner body.

Solenoid Valve. One transducer is placed on the end of the valve opposite the coil, with the transducer sensitive axis parallel to the spool.

Servo Valve. One transducer is located on an end of the valve, its sensitive axis being parallel to the spool.

Pressure Relief Valve. One pickup is mounted on the top of the body of the valve, its sensitive axis being normal to the mounting plate.

TEST FACILITIES

A test facility was designed and built by the Research and Development Center as a means of studying the signature of the selected hydraulic

*A suitable cement is F88 adhesive, made by Tridox Products Incorporated, P.O. Box 91, Gladwyne, Pennsylvania 19035.

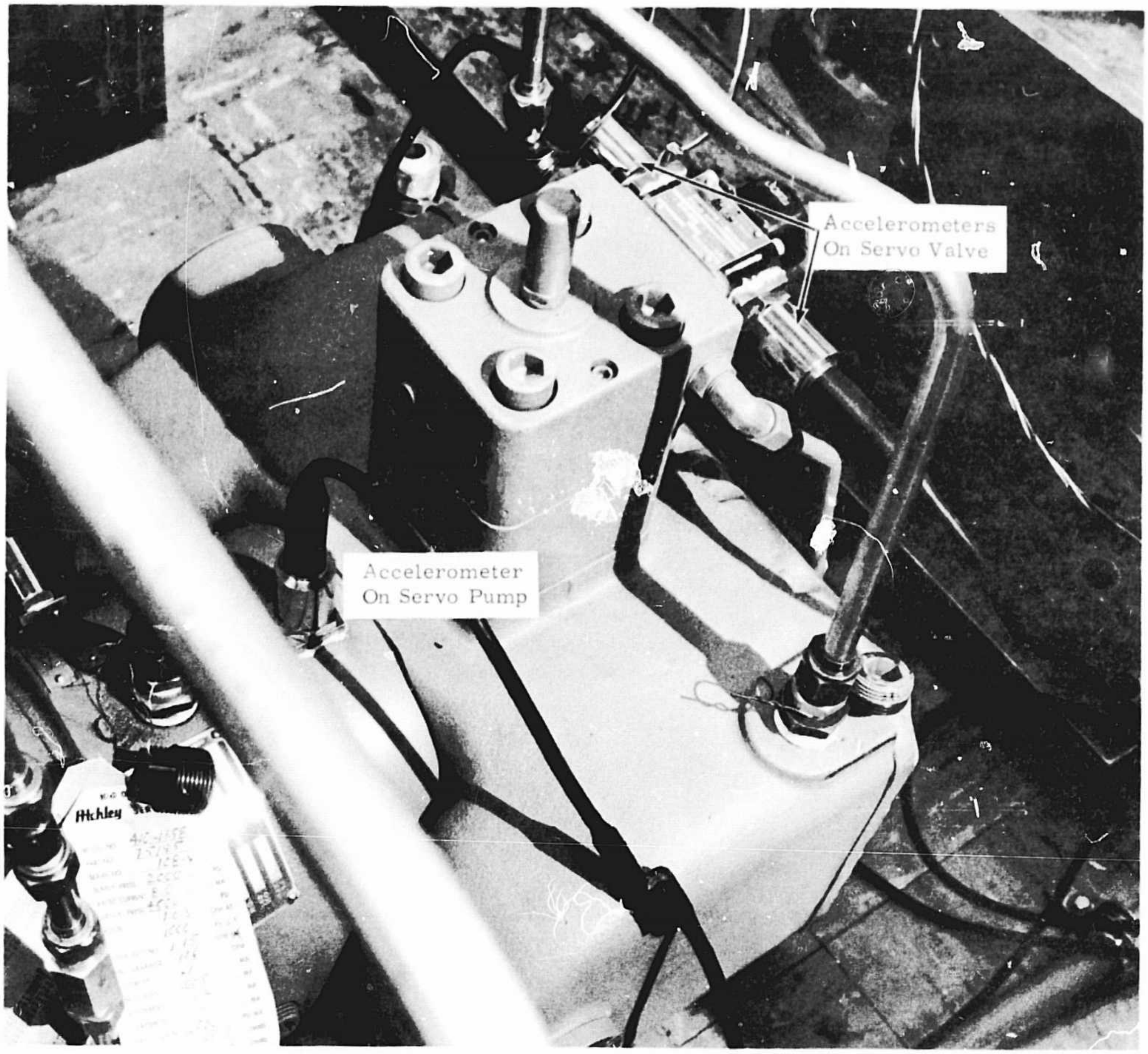


Figure 27. Accelerometers on Outer Race of Servo Pump Roller Bearing and on Servo Valve

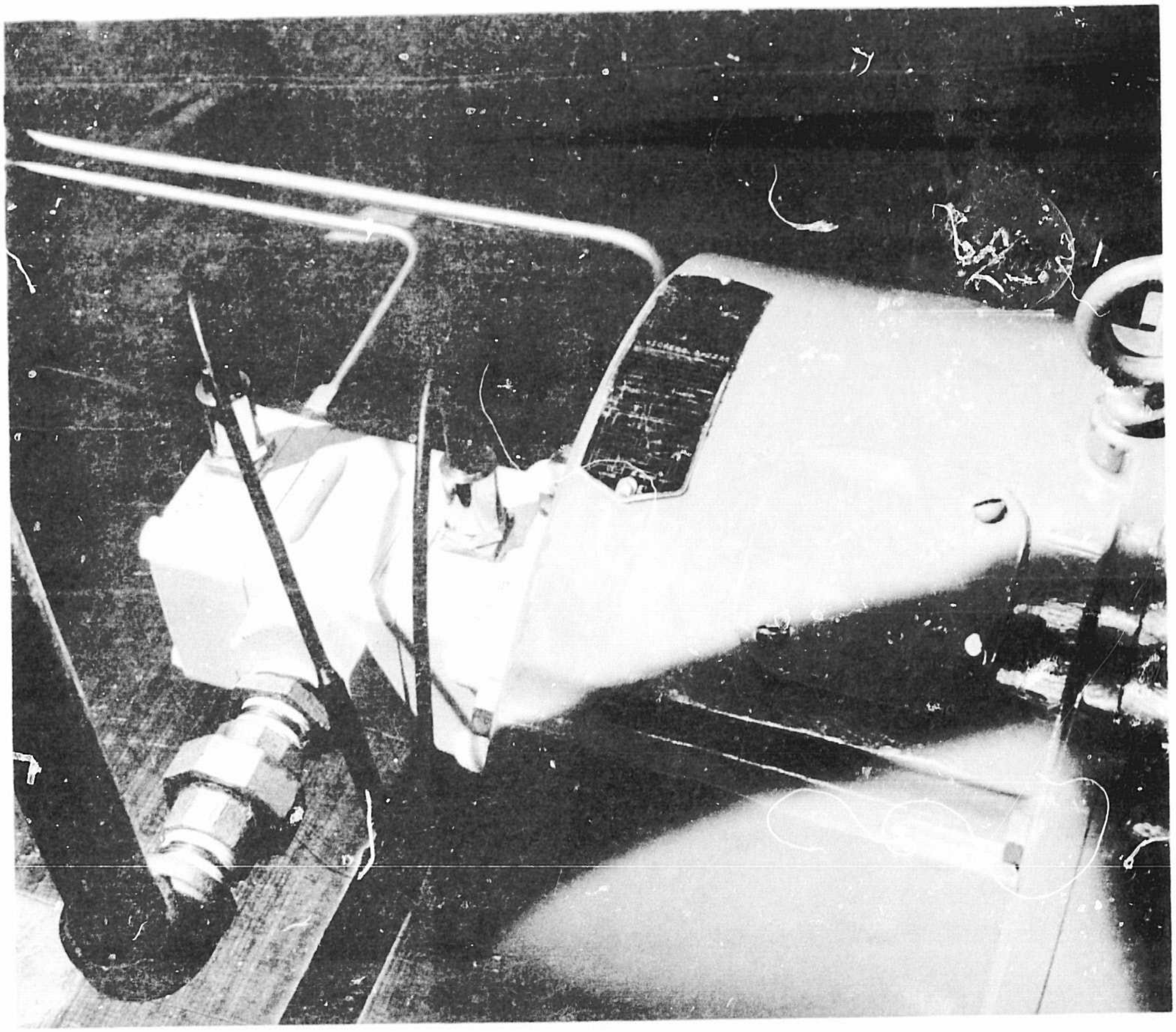


Figure 28. Accelerometers on Vane Pumps

components. While closely simulating the actual installation, it gave considerably more freedom in exercising the components and developing test procedures than would be possible on an antenna. The facility as built had two independent parts. The first, powered by a pump owned by the Center, was designed for the testing of the solenoid valve, pressure relief valve, and servo valve. It is shown in Figure 29; the schematic is Figure 30.

The pressure regulator, servo valve, and solenoid valve are mounted on their respective manifolds. Flow from the supply pump, at a pressure of approximately 1500 psi, is set to 2 gpm by passing all the oil directly to the return line through a flow meter. To do this, valves 2, 3 and 10 are closed and valves 1 and 8 are open. After the supply pump is set to move the desired quantity of oil, valves 1 and 8 are closed. Opening valves 2 and 10 allows the Vickers regulator to be tested. For the regulator test, valves 11 and 12 remain closed. Opening valve 10 slightly simulates system leakage in the control pressure lines in an actual hydraulic power unit. Pressure and flow variations through the regulator can be accomplished by setting the regulator adjusting screw and valve 10 to the desired positions.

To operate the servo valve, the regulator is set to 800 psi and this pressure is allowed to act on the servo valve when valve 10 is closed and valves 11 and 12 are opened. When the servo-valve torque motor is activated, the motion of the spool admits this control pressure to one of the two servo-valve accumulators. The brief flow of fluid into the accumulator, as it charges up to line pressure, simulates the flow through the servo valve when the valve is used to move the actuator pistons in a servo pump.

The solenoid valve may be operated by admitting pump flow and pressure directly from the pump through valves 3, 5 and 6. (Valve 2 is closed so that the regulator and the servo control valve are not in the loop.) The solenoid valve is normally open. When it is energized, flow through the valve stops, and accumulator 3 charges up to full line pressure. When the valve is deenergized, the line supply is dumped into the return line through the solenoid valve. The resulting supply-line pressure drop occurs over a finite time period as a result of the action of the accumulator in the line. It is possible in this way to approximate the operating conditions of the valve when it functions in an actual hydraulic power unit.

The second part of the test facility (Figure 31) was designed for testing the servo and vane pumps, and also the servo valve. Figure 32 is a schematic diagram of this facility.

A 55-gallon drum serves as the storage tank. The dual-vane pump and the servo pump are mounted conventionally on the drive motor. The control portion of the vane pump supplies filtered flow, regulated at 800 psi, to the servo control valve. Oil pumped from the replenishing side of the vane pump is passed through the case of the servo pump for cooling purposes.

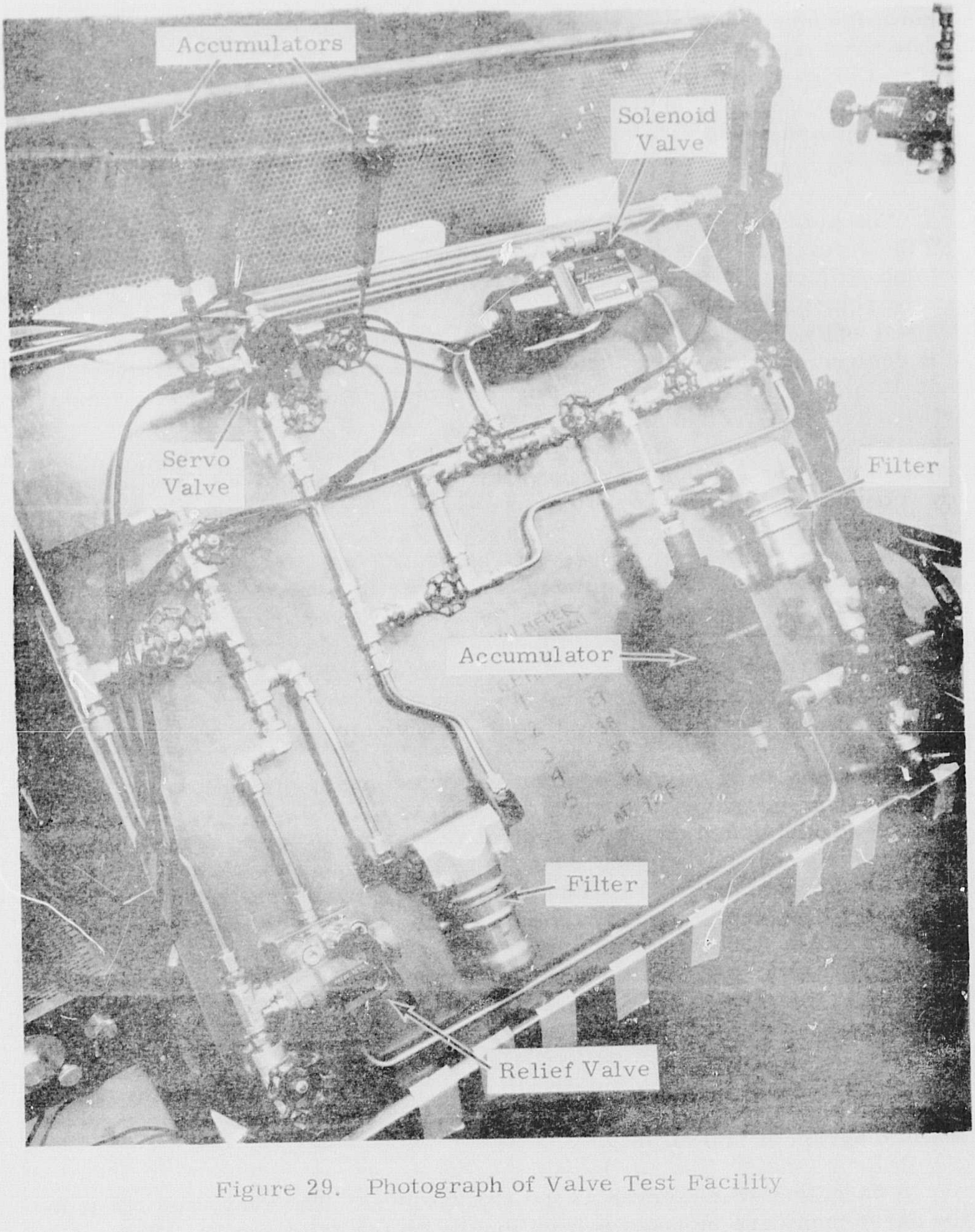


Figure 29. Photograph of Valve Test Facility

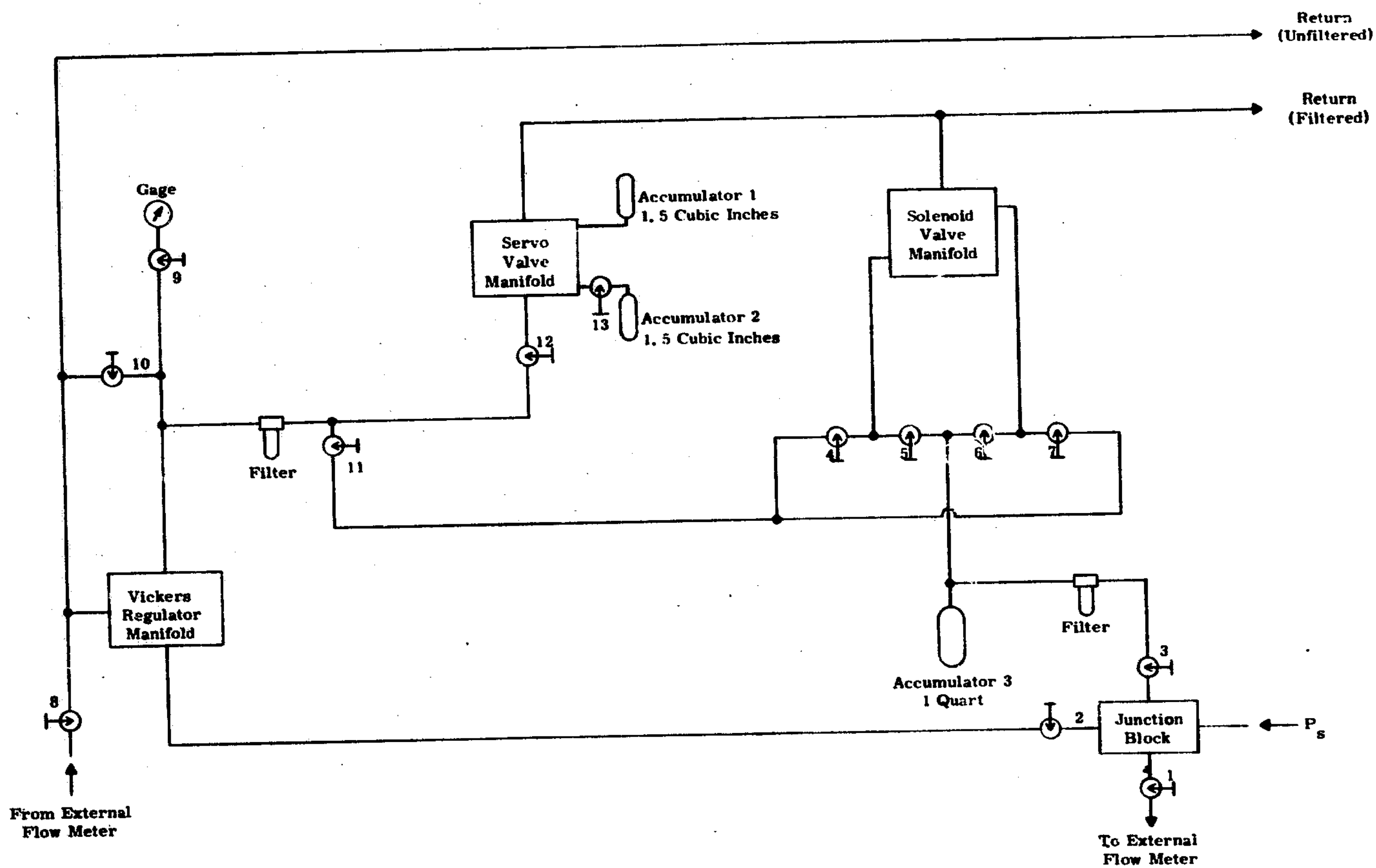


Figure 30. Hydraulic Schematic Diagram for Valve Tests

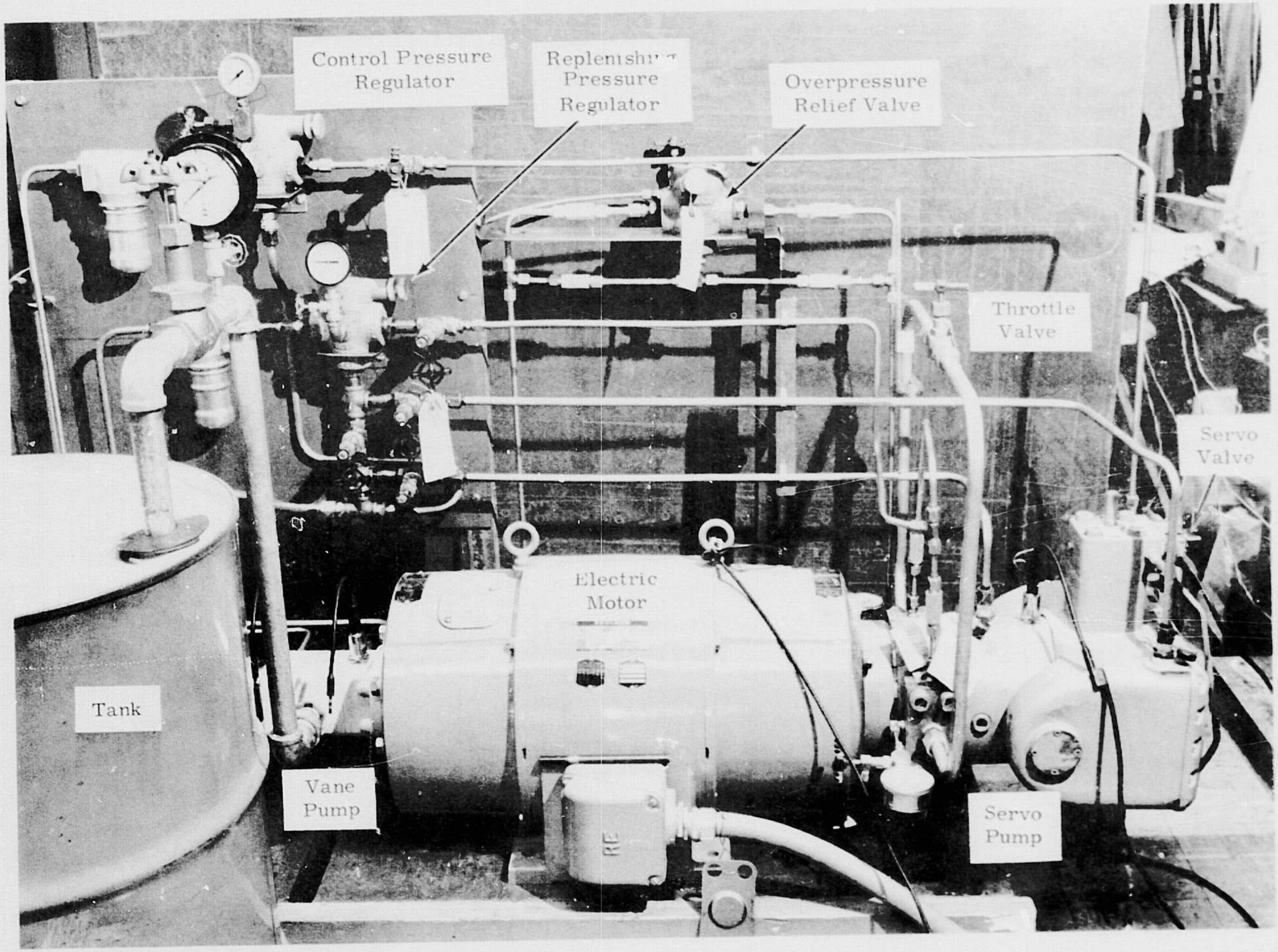


Figure 31. Photograph of Pump Test Facility

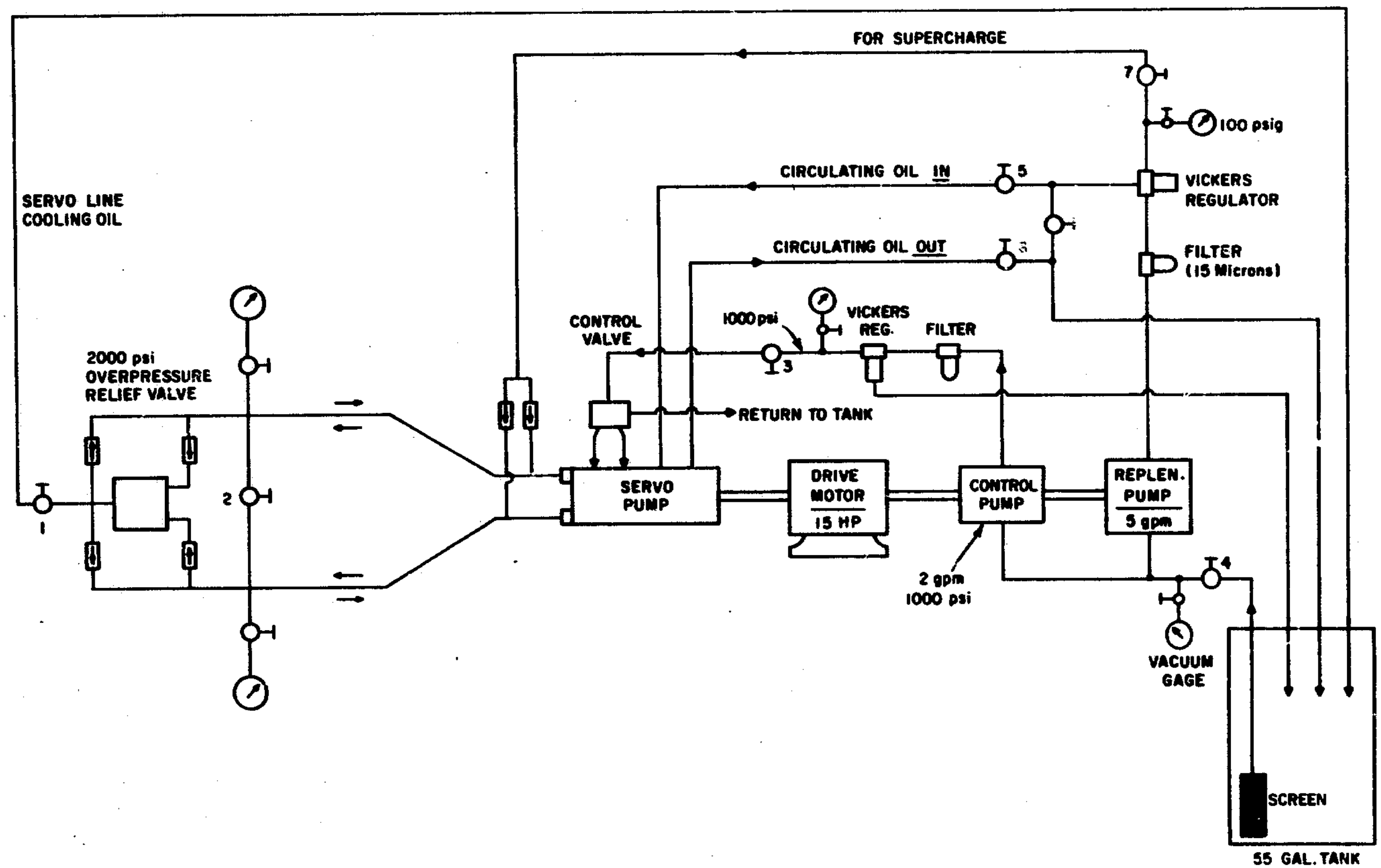


Figure 32. Hydraulic Schematic Diagram for Pump Tests

The pressure-controlled side of the regulator in the replenishing line provides the supercharge pressure for the servo pump.

A throttle valve is used as the load for the servo pump. Its closure and the pump stroke determine output pressure. A series of check valves and an overpressure relief valve (set at about 2000 psi) provide protection incase the throttle is ever accidentally fully closed. A valve in the tank return from the high pressure lines is used to bleed off a portion of the servo pump discharge flow. This tends to prevent overheating by causing a small amount of oil to be continuously replaced. Since the facility does not include a heat exchanger the permissible operating time is determined, first, by the rise in servo pump discharge temperature, and, second, by the temperature of the oil in the tank.

A self-generating magnetic transducer placed in proximity to a grease fitting on the motor shaft provides a once-per-revolution reference signal which is useful in data analysis.

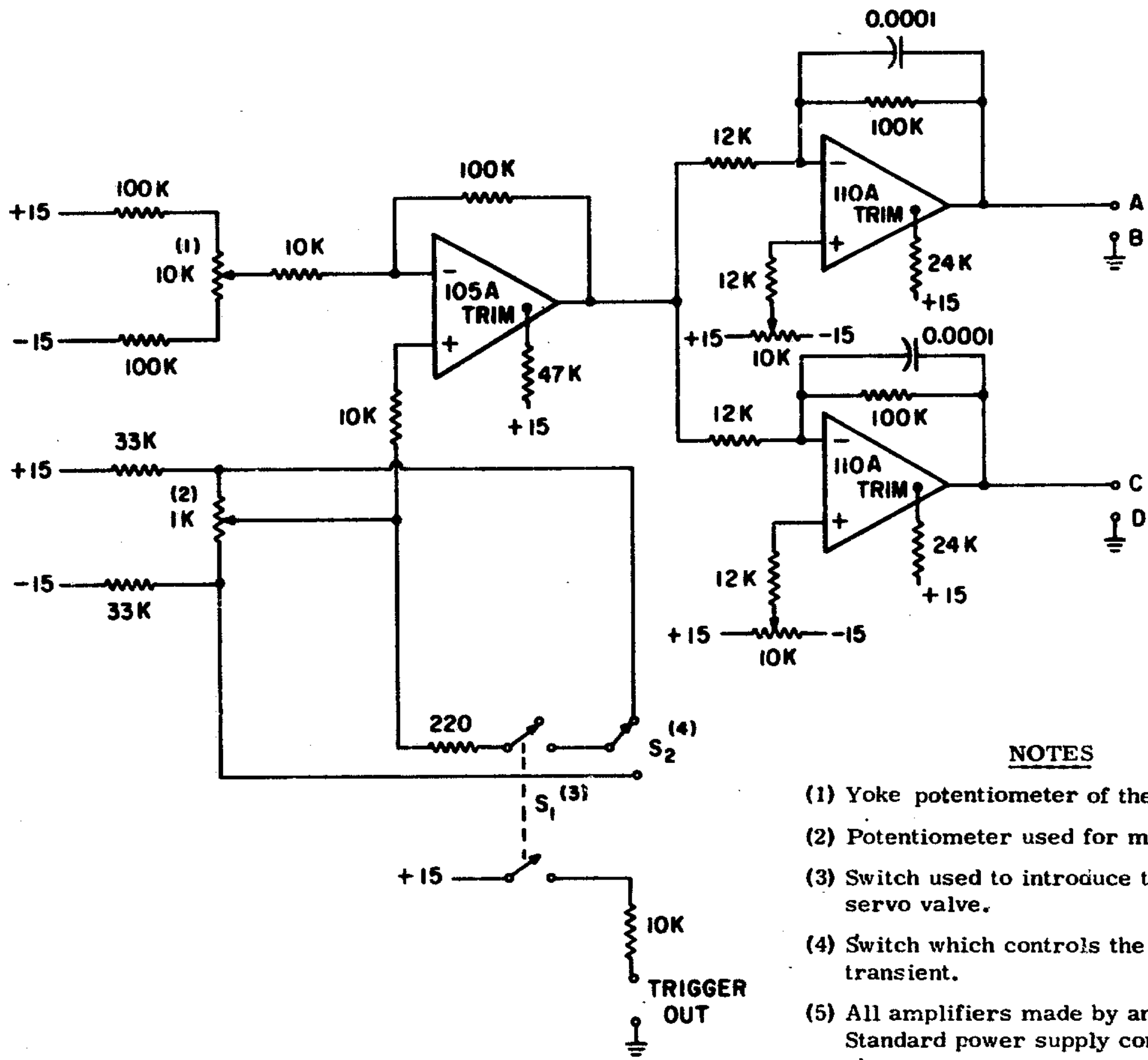
A servo amplifier (Figure 33) was built as a means of controlling the stroke of the servo pump through operation of the servo valve. This amplifier has two inputs: a d-c error signal from an adjustable potentiometer, and a signal from the yoke-position potentiometer built into the servo pump.

The two output amplifiers each drive one of the coils of the servo valve. An initial adjustment on each amplifier makes its output zero when the error signal is zero and the yoke is centered. This amplifier can also be used to introduce transients into the servo valve. A switch supplies a step-function error signal to the amplifier and simultaneously generates a trigger signal which indicates the time at which the transient was initiated.

Tests were also performed on the USB antenna of the Network Test and Training Facility at Goddard. This facility was modified for testing as follows:

1. Transducers were mounted on the hydraulic components as previously described.
2. For solenoid valve tests the solenoid coil was unwired from its normal connection and rewired as in Figure 34. This component was tested with the antenna in the stowed position.
3. The servo valve was tested by rewiring it and the yoke potentiometer as shown in Figure 33. The valve was then controlled using the servo amplifier shown in the same figure.
4. A pickup was added to the motor to generate a once-per-revolution trigger signal.

No system modifications are required for test of the servo pump, vane pump, or pressure relief valve.



NOTES

(1) Yoke potentiometer of the servo pump.

(2) Potentiometer used for manual control.

(3) Switch used to introduce transients into servo valve.

(4) Switch which controls the sense of the transient.

(5) All amplifiers made by analog devices. Standard power supply connections not shown.

Figure 33. Servo Amplifier Circuit Diagram

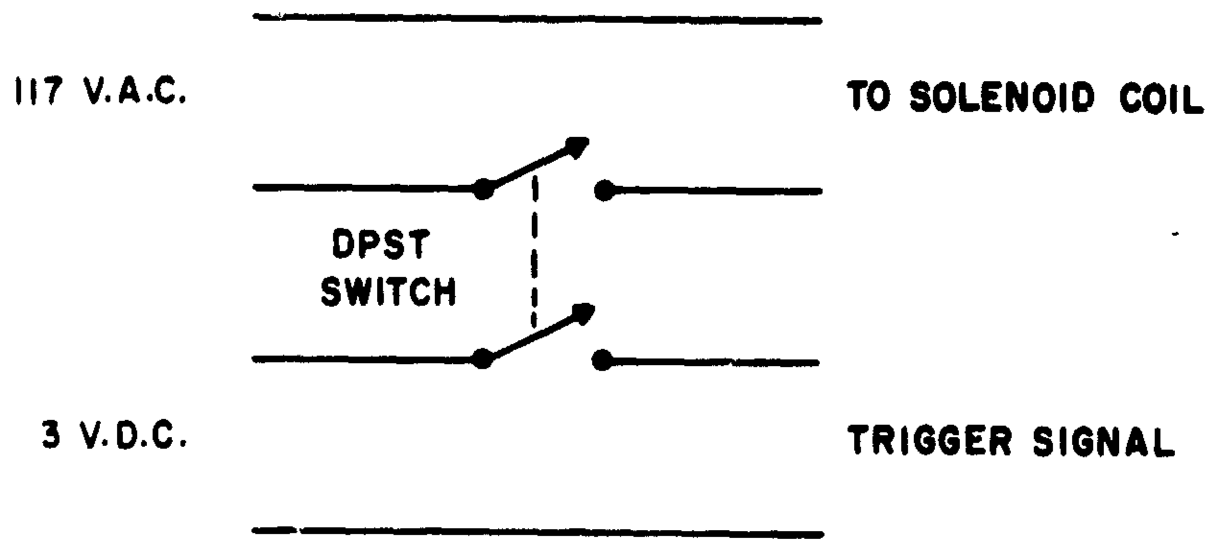


Figure 34. Wiring for Solenoid Valve Test

TAPE RECORDINGS

The tests may be performed and analyzed "live," i.e., without tape recording, or the signals may be taped for subsequent analysis. When the signals are tape-recorded, a bandwidth of ten hertz to 20 kilohertz will be satisfactory except as noted below. Recordings in the FM mode are suggested as being generally more precise and accurate than recordings in the direct mode.

As will be discussed, response to frequencies considerably higher than 20 kilohertz is necessary for study of the vibration of the servo valve, even though these frequencies are well above the upper range of linear response of the accelerometers. Response to about 40 kilohertz is adequate. Since absolute information is not required from these signals, the direct record mode is acceptable.

There are three important factors which contribute to making satisfactory tape recordings. These include proper documentation, high-fidelity recording, and suitable calibration.

A tape recording has value only when the information recorded can be meaningfully retrieved. This requires complete documentation of all the pertinent conditions under which the recording is made. Figures 35 and 36 show examples of such records. The data at the top of the first sheet (Figure 35) give the reel identification number, the make and model of the recorder used, the tape speed, the date of the recording, and a description of the subject of the recording. (A block diagram of the instrumentation preceding the tape recorder is often helpful.) The data in the middle of the sheet tells whether each track was recorded FM or direct, and, if FM, the center frequency for each track. The table at the bottom of the first page (Figure 35) gives the parameter recorded on each track.

REEL NUMBER: GSFC 6A
TAPE RECORDER: CEC PR 3300
TAPE SPEED, IPS: 60
DATE: 8-21-68
TEST DESCRIPTION: Test of components with introduced defects at NTTF.

x-axis hydraulic — Endevco 28450 system — Arvee 851B decade ampl. (optional) — calibrated attenuator — Tape recorder

IRIG CH.	1	2	3	4	5	6	7
FM FREQ.	54KHz		64		54		54
DIR. REC.							

IRIG CH.	8	9	10	11	12	13	14
FM FREQ.		64					
DIR. REC.				X		X	

IRIG CHANNEL	PARAMETER
1	Servo pump roller bearing vibration
2	—
3	Servo pump discharge port vibration
4	—
5	Replenishing pump vibration
6	—
7	Control pump vibration
8	—
9	Voice commentary
10	—
11	Pressure relief valve vibration
12	—
13	1/rev trigger signal
14	—

Figure 35. Sample Tape Recorder Parameter Log

REEL NUMBER: GSFC 6A

RECORDING	COUNTER (REVOLUTIONS)	TEST DESCRIPTION
1	0-206	Calibration 100MV RMS at 60Hz, ch. 1, 3, 5, 7 Calibration 0.5V p-p sq. wave ch. 11 Net attenuation ea. channel (in sequence) 0, -10, 0, 0, 0, -10, -10 DB
2	206-690	Run with servo pump yoke centered Net attenuations 0, -10, 0, 0, 0, -10, -10 DB
3	702-1178	Run east at 1 deg./sec. Net attenuations 0, -10, 0, 0, 0, -10, -10 DB
4	1178-1648	Run east at 2 deg./sec. Net attenuations 0, -10, 0, 0, 0, -10, -10 DB
5	1648-1980	Run east at 3 deg./sec Net attenuations 0, -4, 0, 0, 0, -10, -10 DB
6	1980-2245	Run east at 4 deg./sec. Net attenuations 0, 2, 0, 0, 0, -10, -10 DB
7	2245-2665	Run west at 3 deg./sec. Net attenuations 0, -4, 0, 0, 0, -10, -10 DB
8		
9		
10		

Figure 36. Sample Tape Recorder Test Log

It is good practice to include a voice commentary on each tape. The redundancy is valuable in case the data sheets are ever lost, and the commentary is often an aid in identifying important events during tape playback.

The second sheet (Figure 36) describes each of the recordings or tests which are arranged serially on the tape. The tabulation includes the reel number for cross-referencing to the previous sheet, and the recording number, the footage counter range for that recording, and the test condition. The latter should include attenuator and/or gain settings of the instruments.

One of the tests should be a calibration. With this it will always be possible to relate the retrieved signals to absolute vibration levels. The value of being able to measure absolute levels may not always be immediately apparent; in fact, it may not always be necessary. On the other hand, it is not always possible to foresee all the future uses of recorded data. A calibration can always be ignored, but it cannot be added with confidence at a later date.

Making a recording which is a faithful reproduction of the original signal has two aspects. First, the recorder and its mode of operation must have sufficient frequency response to encompass all significant signal components. Second, the signal must be recorded so as to make best use of the recorder's dynamic range. If the signal is too strong it will overload the recorder and cause distortion, while if it is too weak it will be obscured by the recorder's inherent self-noise. It is common practice for magnetic tape recorders to be set up to accept a one-volt RMS sine wave with some small (on the order of 3%) distortion. The maximum excursion of such a signal is $\pm\sqrt{2}$ volts. If nonsinusoidal signals are recorded on this machine their amplitude should be adjusted so that their maximum excursion approximate this range. This ensures that both distortion and noise will have minimal effects.

TEST PROCEDURES

Recommended tests for the various hydraulic components are described below. As previously discussed, the information can be taped for later analysis or analyzed live.

Servo Pump

Record the pump vibration with yoke centered, i.e., at zero stroke. Place the antenna so that it looks at one horizon. Take data while driving at constant velocity to the opposite horizon.

Vane Pump

Record vibration while running with the yoke of the servo pump centered.

Solenoid Valve

Stow the antenna in any convenient position Wire the solenoid coil as shown in Figure 34. With hydraulic power supply operating (servo pump yoke centered) to supply system pressure, actuate the solenoid coil. Analysis is made as the coil is deactivated. Analysis requires the vibration signal and a trigger signal generated by the switch shown. Repeat with the hydraulic power supply shut down.

Servo Valve

Replace the normal servo amplifier with the one shown in Figure 33. Make the necessary connections to the yoke potentiometer, but not the servo valve. Energize the amplifier. Adjust the manual control potentiometer to its centered position. (The STEP FUNCTION switch should be OFF.)

The voltage at the output of each output amplifier should be approximately zero. If it is in error, adjust the bias potentiometer to correct to zero. Connect the servo valve to the amplifier and release the brakes. A slight correction of the manual control potentiometer will make the antenna velocity very close to zero.

To take data, set the DIRECTION switch to A, and turn the STEP FUNCTION switch to ON. This will cause the antenna to turn at constant velocity. The step-function switch may be returned to OFF in about one second. The process is repeated with the direction switch set to B. The operation of the STEP FUNCTION switch also generates a trigger signal which is required in the data analysis.

Pressure Relief Valve

Measure the vibration with hydraulic power supply operating, the antenna being stowed.

Section 5

DATA REDUCTION TECHNIQUES

Data reduction techniques serve two purposes. They extract wanted information from a background which may contain much that is extraneous, and they present the wanted information in a meaningful way. This could mean that it is displayed so as to show the important details of the signal, or it may mean that the information is formatted in a way which makes it suitable for transmission to other equipment (such as a digital computer) for further processing.

SPECTRUM ANALYSIS

A generalized signal consists of some combination of discrete-frequency spectral components and random noise. The discrete-frequency components can be separated out by means of frequency filtering, provided that the energy of the desired signal exceeds the sum of all other signals, as measured within the passband of the filter. By decreasing the filter bandwidth the noise is increasingly discriminated against, while acceptance of sinusoidal signals within the passband is unchanged. While one might at first conclude that this process could be continued indefinitely so that weaker and weaker sinusoids could be detected, there are practical limitations in building such an analyzer.

As the filter becomes more selective its tuning becomes more critical, and finally a point is reached when it is often found that the supposedly stable sinusoid actually is wandering somewhat in frequency. Its bandwidth is not really zero, but finite, and an excessively narrow filter will actually increase measurement errors. There are other practical problems having to do with the permissible tuning rate of the filter (which is an important consideration when a tunable filter is to be scanned over a range of frequencies), filter stability, and the shape of the filter passband. Despite these problems, the technique is well suited to the separation or extraction of particular components of complex signals within the limitations described.

TRANSIENT AVERAGING OR SUMMATION

There are occasions when as much or more information is contained in the waveform of a signal than in the amplitude of its spectral components. This is particularly true if there is a significant repetitive transient event. In many cases the desired signal will be obscured by unwanted signals. If the unwanted signals differ considerably in frequency from the desired signal, they can be separated out by filtering. However, if they are similar in frequency, filtering is of little value and recourse must be made to other techniques.

If a signal repeats itself in a cyclical manner, a powerful tool called signal summation or transient averaging can be used to extract it from other signals which are nonperiodic or of an unrelated periodicity. Since the periodicity of the wanted signal is known, it is summed on itself as shown diagramatically in Figure 37 to develop the function $F(\tau)$:

$$F(\tau) = \sum_{n=1}^{N} f\left[\tau + (n-1)T_S\right] \qquad 0 \leq \tau \leq T \qquad (76)$$

where n is the number of summations

T_S is the period of the repetitive function

T is the period of scan of the device which perform the summation

τ relates to the rate at which the summed signal is read out; this requires τ seconds for a signal representing T seconds in real time.

If there is a signal with periodicity T_S in the input it will sum directly as n, the number of summations, since there is coherence from one sum to the next. Signals which are incoherent add not as n, but as its square root. This is analogous to adding together two random signals of equal amplitude; their sum is $\sqrt{2}$ times the amplitude of either component rather than twice the size.

As a result of the difference between the rate at which coherent and incoherent signals are summed, the ratio of the wanted to unwanted signals in the sum is $\sqrt{n}$ times the same ratio at the input to the summation device. For most signals, on the order of 100 summations (giving a ten-to-one signal-to-noise ratio improvement) are adequate to extract the wanted signal from the background. Since improvement goes as the square-root of the number of summations, many more than 100 would be required to get an additional significant improvement. Very large numbers of summations may not give as much improvement as expected because of slight instabilities in the signal, such as small speed changes.

The general organization of a device suitable for forming the function $F(\tau)$ is shown in Figure 38. Two inputs are required: the signal itself, which may be preprocessed as will be described, and a trigger signal having the same periodicity as the signal which is to be extracted. If the desired signal is the waveform developed by each revolution of a rotating device, such as a pump or motor, the trigger would be a once-per-revolution pulse generated by a transducer on the shaft. If one is searching for bearing defects, which are generated at a rate which is not an integer multiple of running speed, the trigger would come from an externally programmed source whose periodicity was adjusted to the proper value.

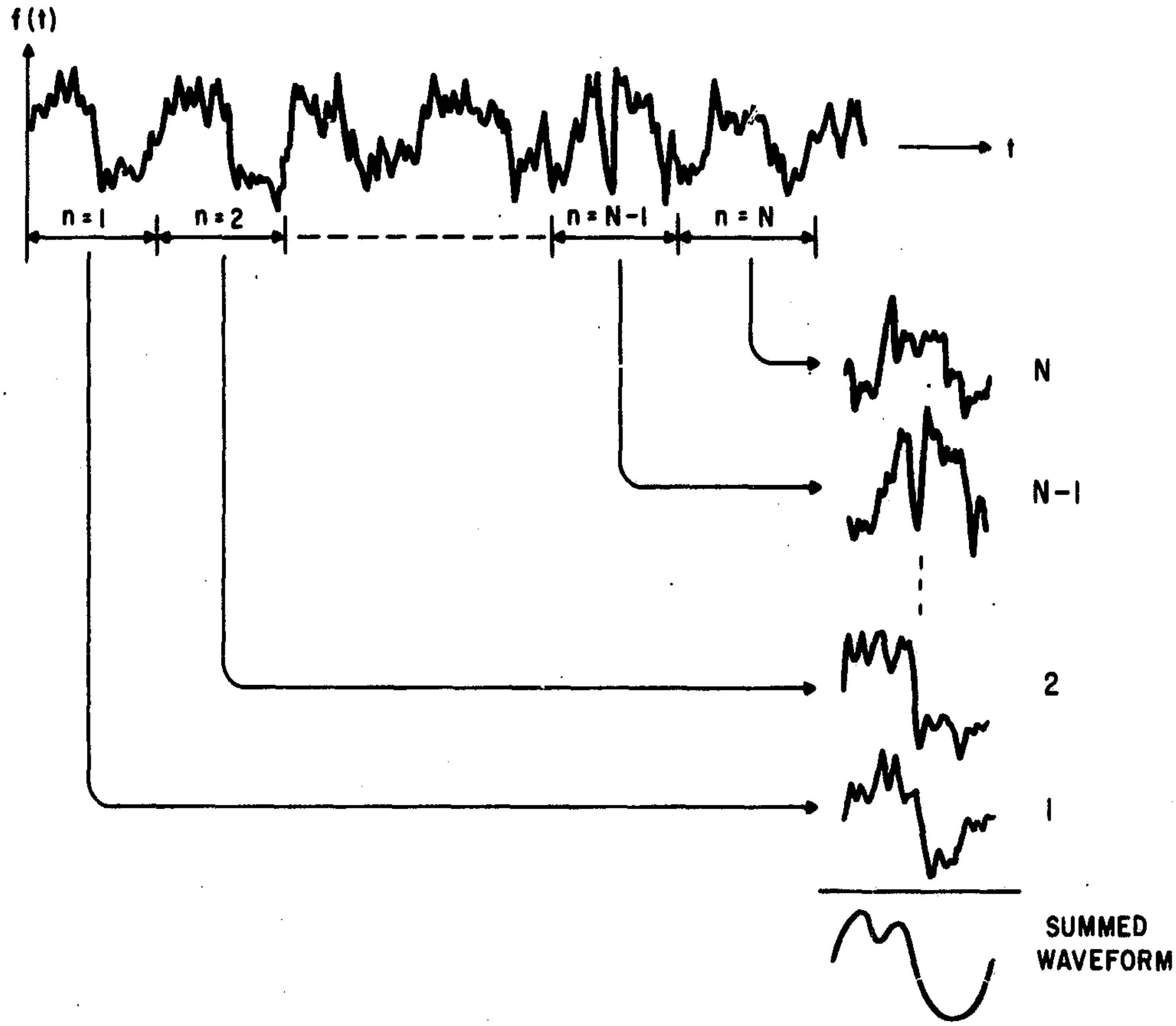


Figure 37. Example of Summation or Transient Averaging

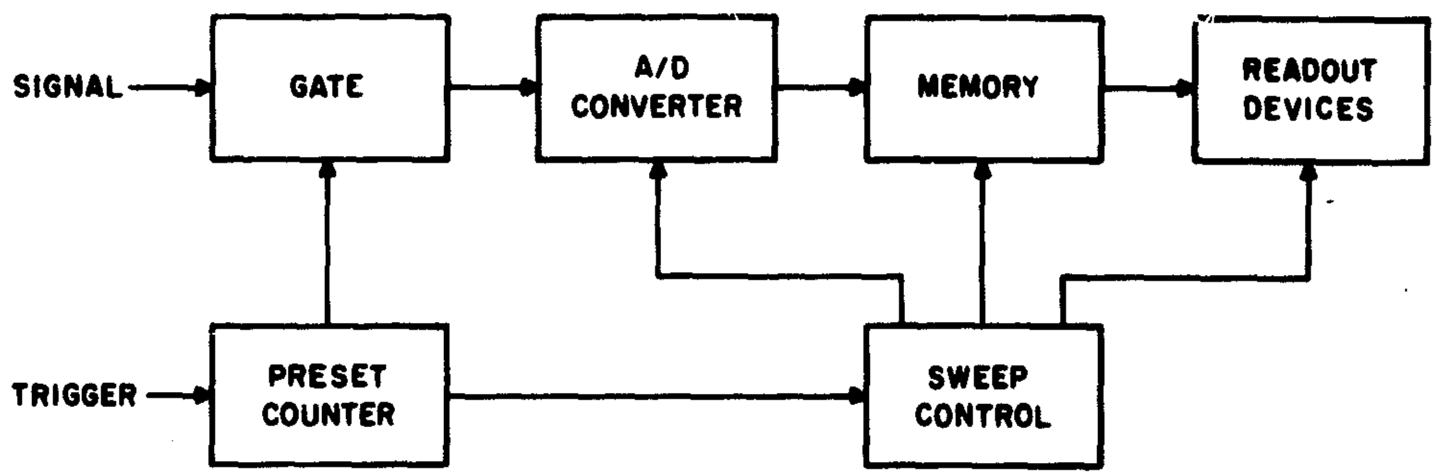


Figure 38. Organization of a Summation Analyzer

The trigger signals are fed to a preset counter which controls a gate. If the number of trigger signals counted is less than the preset number, the signal to be analyzed is passed to the succeeding stages; when the preset number is reached the gate opens and the summation process ceases.

The summation device itself has three major components:

- Analog-to-digital converter
- Memory bank
- Scanning control

The heart of the analyzer is the memory, usually a bank of magnetic cores. Under control of the scanner the analog-to-digital converter looks at the incoming signal briefly and digitizes it; the result is stored in the first address of the memory. The scanner then advances one step and the process repeats, with the digitized signal being stored in the second address. The process continues until all addresses have been contacted, thus storing in the memory a digitized representation of the selected portion of the incoming signal.

Where another trigger pulse comes along the entire process is repeated, each digitized signal being added to that already stored in the appropriate address of the memory. This continues until the preset counter stops the process after a predetermined number of summations.

If the desired signal is known to be within a certain frequency range and extraneous signals (such as noise) are known to be mainly in a different part of the spectrum, the signal may be prefiltered before summation. When this is done the signal-to-noise ratio is improved even before summation, so that the desired signal can be extracted with fewer summations than would otherwise be required.

Some transient signals which recur periodically, such as the impacts characteristic of defective bearings, are not necessarily phase-coherent from one occurrence to another. Each impact is like a burst of noise; they occur at regular intervals but the waveshapes differ. Not being coherent, such signals do not sum in such a way as to be enhanced relative to the background. However, the summation process can aid in their detection if the incoming signal is rectified before being summed. When this is done it is the envelope of the periodic signal which is recovered, rather than the signal itself.

The transient signal, even though weak, adds energy at the same addresses on each summation, while other nonsynchronized events add randomly, first in one address, then another etc., so that their contribution is spread out rather than concentrated. As a result, those signals with the proper periodicity are again made evident.

SPECTRAL ANALYSIS AFTER AVERAGING

The signal stored in the core memory of the summation device contains only signals whose frequencies are integer multiples of the trigger signal. If the trigger is derived from the shaft of a 1200 revolution per minute machine, for instance, it occurs every 50 milliseconds, or at a 20 pulse-per-second rate. The summed signal contains mainly multiples of 20 hertz, other signals being discriminated against by the summation process. A Fourier analysis of the summed signal can be used to determine the amplitude of each of these components.

The advantage of this (over a Fourier analysis of the raw signal) is that the analysis is made of a signal which is effectively prefiltered to eliminate most of the unwanted components. As a consequence, the work of the analyzer is made easier and the requirements placed on it are less stringent. The technique is particularly useful for determining the level of very weak components, which would otherwise be difficult to measure.

The Fourier series can be calculated digitally by reading out the memory into a computer, which performs the necessary calculations. The signal in the memory will equal or exceed one period of the signal, as described in Equation 76; the calculation requires only the data in j sequential address, where

$$j = \frac{T_S}{T} \cdot N \tag{77}$$

N is the number of addresses in the memory

Alternately, the Fourier analysis could be performed by a conventional analog spectrum analyzer. To do this, the output of j addresses is read out sequentially at a uniform rate in a continuously circulating manner. That is, after the signal stored in the j'th address is read out the sequence is restarted beginning with the first address. When this is done the output is a representation of the original signal with components lacking the proper periodicity largely removed. Depending on the rate at which the memory elements are read out, the frequencies present in this reconstructed signal may or may not be the same as their equivalents in the original signal which was summed.

$$\frac{f_{out}}{f_{in}} = \frac{\Delta t}{\Delta t'} \tag{78}$$

where Δt is the period represented by each address of the memory during summation

$\Delta t'$ is the time interval between successive addresses during readout

TIMING OF TRANSIENTS

Among the tests required for evalution of the condition of the servo and solenoid valves are measurements of their operating time in response to a transient. As previously described, the test procedure includes generation of a trigger signal which acts as the time reference. The vibration signal measured on the valve is used to indicate its response. In the case of the solenoid valve, the desired signal is that generated by the moving parts as they hit their stop, while the motion of the servo valve is deduced from the change in flow noise.

In any measurement problem, it is necessary to take data in such a way as to discriminate against unwanted signals while retaining significant information. The unwanted signals are largely vibrations due to the rotating machinery, the motor, and the pumps. It was found experimentally that these could be largely removed by high-pass filtering.

The solenoid valve impact noise is a relatively powerful source, and a two kilohertz cutoff was adequate. The signal from the servo valve, mainly flow noise, was obscured unless the cutoff was set to at least 20 kilohertz. It will be recalled from the earlier discussion of transducer characteristics that this is above the linear range of frequency response, and in fact includes the resonant frequency, which is on the order of 35 kilohertz. As a result, the amplitude information in the signal is of little value. However, the absolute value of the amplitude is not required, and the relative amplitude (as a function of time) is adequate. The timing of events in the signal is easily determined by displaying the signal on an oscilloscope. The sweep, which is calibrated, is triggered at the onset of the transient.

Section 6

DATA REDUCTION SYSTEM

SYSTEM DESCRIPTION

The data reduction equipment used by the Research and Development Center had two major components: the Radiometer FR2a frequency analyzer, and the Enhancetron 1024 summation device.

The Radiometer is a heterodyne-type constant-bandwidth tunable filter covering the frequency range from 20 to 15000 hertz. The bandwidth is selectable: 2, 8, or 25 hertz. The spectrum analyses shown in the report were made by mechanically sweeping the filter, using two hertz bandwidth. The filter output was plotted as a function of frequency using the Bruel and Kjaer 2304 Level Recorder. This is, in effect, a recording voltmeter having logarithmic characteristics.

The Enhancetron is a summation or transient averaging device with a core memory having 1024 addresses. Its trigger came either from the one-per-revolution sync pulse generated by the shaft position sensor or, for bearing defect analysis, from a laboratory-designed and -built pulse generator, called a Time Mark Generator, with easily selectable periods (by digital switches).

The system, as used for diagnostic purposes at Goddard, will now be described in detail. The following is a list of NASA-Goddard equipment used for summation:

1. Tape recorder - Sangamo 4700
2. Summation computer - TMC-CAT-400
3. Analog computer - EAI RACE TR-48
4. Digital computer - EAI DES-30
5. Oscilloscope
6. Band-pass filter - Krohn-Hite
7. X-Y plotter - EAI 1110
8. Counter-timer - Computer Measurements 226B
9. Function generator - H. P. 203A

Also used is a GE-provided instrument that contains variable-gain amplifiers and an option to full-wave rectify the accelerometer signals before summation.

Figure 39 is a block diagram of the summation systems. The signal from the tape recorder is processed by a band-pass filter. After filtering, it is amplified, and then rectified if desirable. This is now the input that is summed by the CAT 400, which has 400 memory addresses. To identify the start of each data cycle the once-per-revolution signal from a shaft pickoff or an artifically generated signal with a period correct to 10 microseconds is used to trigger the CAT sweep after passing through an AND gate. The other input into the AND gate is the output of the preset counter. As long as the residual count in the present counter is greater than zero, there is an output from the counter. At the end of each sweep of the CAT, a pulse is generated. These are then counted by the preset counter, decreasing until zero is reached. At this time there is no output from the counter; the AND circuit is disabled, preventing any further pulses from triggering the CAT sweep. The summed data can now be read into an X-Y plotter or into a spectral analyzer.

A block diagram of the spectral analysis of the CAT output is shown in Figure 40. The equipment used is:

1. Summation computer - TMC-CAT-400
2. Analog computer - EAI-RACE TR-48
3. Digital computer - EAI-DES-30
4. Counter-timer - Computer Measurements 226B
5. Special pulse generator - Intercontinental Instruments PG-1
6. Spectral analyzer - Singer LF-26
7. Power spectral density analyzer - Singer PDA-1
8. Oscilloscope

Normally, the summation time is set slightly longer than the once-per-cycle period. If the summed data are to be analyzed spectrally, only that part of the memory covering exactly one cycle or an integer multiple must be used. An error will produce an apparent discontinuity in the signal. Not only will a shift in the frequency distribution occur, but additional data will be generated that did not exist in the presummed signal. The correct number of CAT addresses to be used is given by:

$$n = 400 \cdot \frac{T_{sig}}{T_{cat}} \tag{79}$$

T_{sig} = period of signal

T_{cat} = period of one complete CAT summation

n = number of addresses used during period T_{sig}

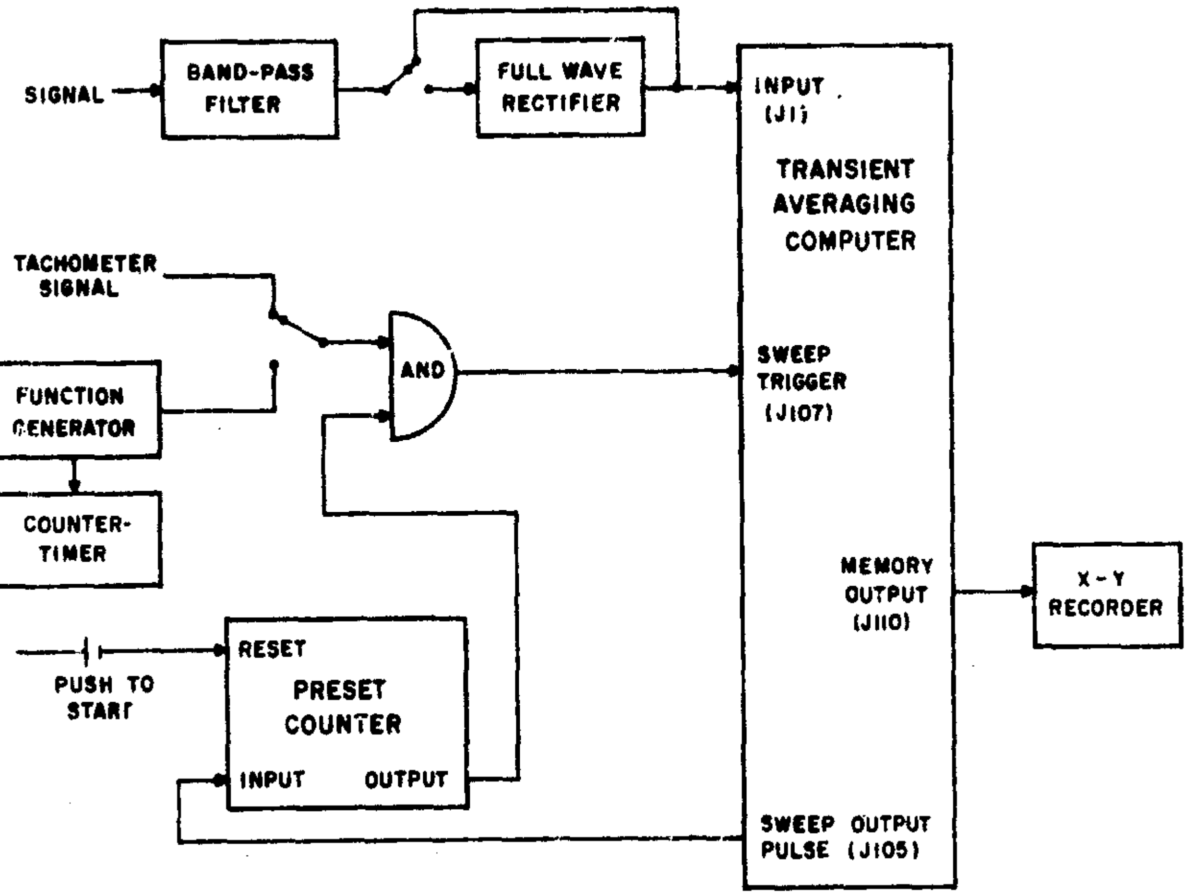


Figure 39. Block Diagram Showing Use of the Computer of Average Transients (CAT) as a Transient Averager.

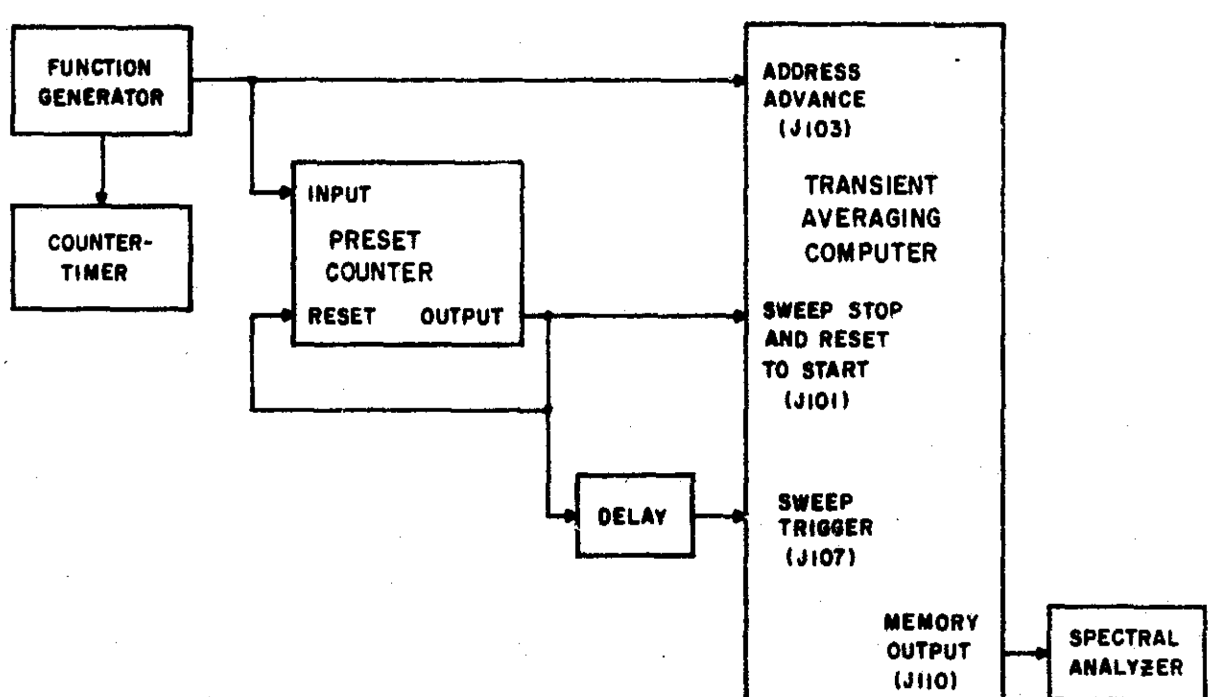


Figure 40. Block Diagram Showing Use of CAT in Conjunction With Spectrum Analyzer

This number is set into the preset counter of Figure 40. A pulse generator operating at a frequency less than the limiting speed of CAT address advance (20 kHz) causes the memory to be read out bit by bit to the spectral analyzer. To avoid the possibility of missing one bit, the advance rate is normal operated at less than 10 kilohertz. The pulse-generator output operates a preset counter. When the count reaches the correct value a pulse stops the CAT memory sweep and resets it to start. It also resets the preset counter. Since the CAT sweep start must be delayed at least 50 microseconds to allow for fly-back time, a short delay of approximately 75 microseconds is inserted between the counter output and the sweep start. The delay should not exceed the period between address advance pulses (100 microseconds or longer when the address advance rate is 10 kilohertz or less), or the first address count will be lost. The memory is thus continously cycled so that it can be spectrally analyzed. If the memory output is displayed on an oscilloscope, it will appear as a continuous wave without discontinuities.

A connection diagram of a summation and spectral analysis system using NASA-Goddard equipment appears in Figure 41. In the spectral analysis mode the impulse generator pulses are processed in the following manner. The output of differentiator 7 (point A) is a sharp spike, as shown in Figure 42A. When the present counter reaches zero its output (Figure 41, point B) as shown in Figure 42B is a sharp spike. This is expanded to 50 microseconds (Figure 42C) by a monostable counter (Figure 41, point C) in order to properly operate AND gate 8E. Monostable counter MT8 expands the pulse to 80 microseconds (Figure 42D). The end of the 80-microsecond pulse is used to generate a delayed pulse, using AND gate 7A and differentiator DIF 9 (point E Figure 41, and Figure 42E). The pulse is then expanded to 50 microseconds (Figure 42F) to operate the digital logic that controls the external triggering of the CAT 400 sweep.

OPERATING INSTRUCTIONS

To implement the summation and subsequent spectral analysis, the specific equipment owned by Goddard is interconnected according to Figure 41. For summation, the following procedure should be followed:

1. Connect the equipment as shown in Figure 41. The input signal is connected to input J1 on the front of the panel. (The output from the CAT memory to the X-Y recorder and the spectral analyzer is in the rear.)

2. Turn on the equipment previously listed and allow to warm up for 15 minutes. The CAT 400ON switch is at the rear of its case.

3. Set the number of CAT summations on counters DC-1 and DC-0 (of EAI-DES-30) to the desired number.

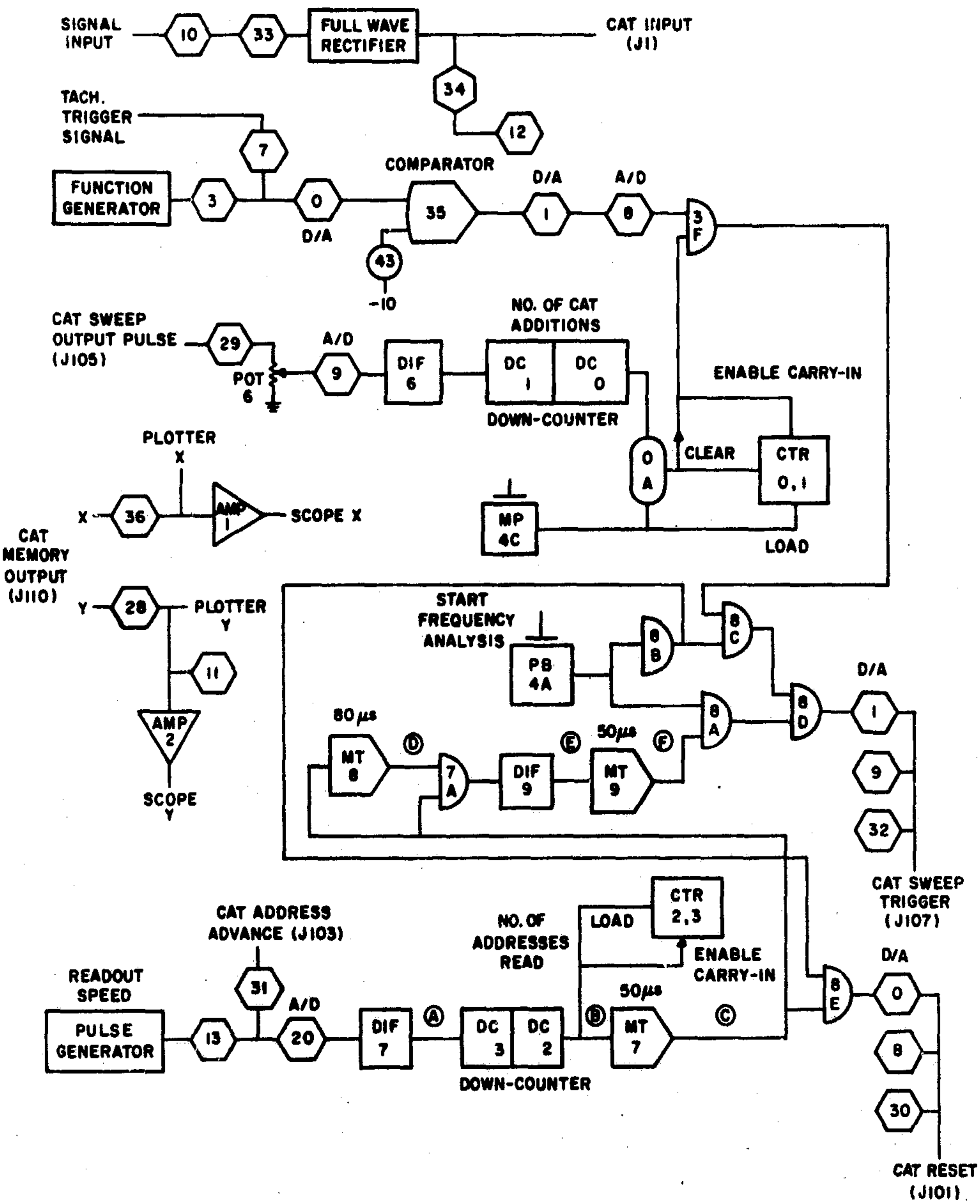


Figure 41. Connection Diagram for the Use of CAT

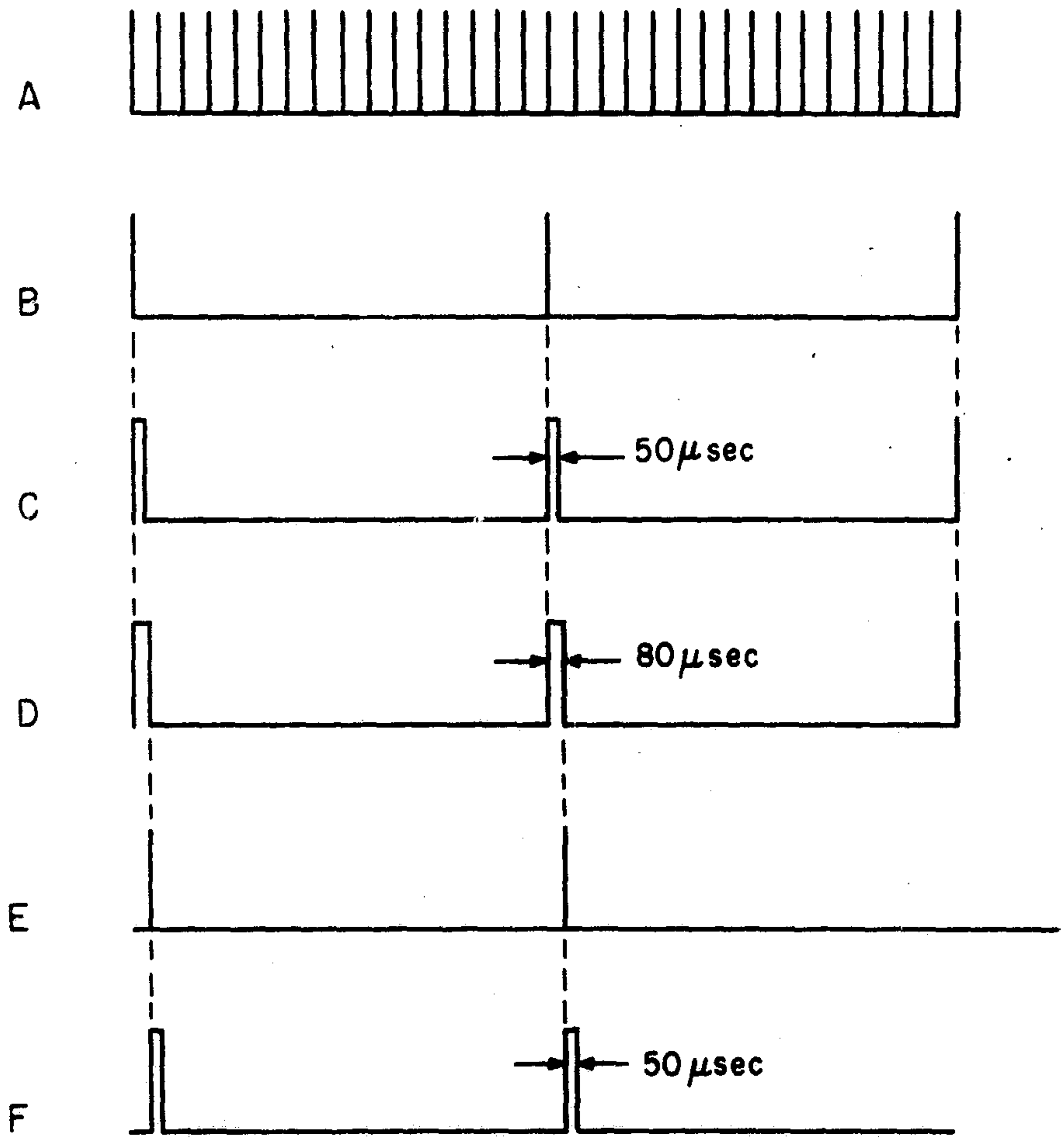


Figure 42. Waveforms of Analysis Control Circuits

4. Set the CAT 400 adjustments as follows:

Name	Position
PROGRAM switch (in rear)	C
Potentiometer (in rear)	1/4 turn from full CW
ACC. -NORM. -REV. switch	ACC.
DATA TRANSFER switch	off
INPUTS IN USE switch	1
SWEEP DELAY	0
STIM. -PROMPT switch	any position
TRIGGER	EXT.
SWEEP TIME	slightly larger than signal period
VERT. RANGE	10^4 (May be set in other positions if different ranges are desired as determined by sample run.)
Inputs	switch 1 only turned on
EXT. -ADD-SUB.	SUB.
USE-TEST	USE
PLOT-PRINT	PRINT

5. Push the display button. The information previously stored in the memory will appear on the oscilloscope.

6. Press the reset button to erase the memory. Memory erase will work only when the CAT is in the display mode.

7. Set the desired number of CAT additions on digital counters 0 and 1 (DC-0, DC-1, Figure 41)

8. Play data tapes on the magnetic recorder, and adjust signal peak amplitudes to be slightly less than ± 3 volts as measured on an oscilloscope.

9. If a tachometer signal is also recorded, it should be adjusted to approximately two volts, zero to peak.

10. If a tachometer signal is not available, set the H. P. 203A function generator at the desired frequency and, using the square wave output, set the frequency to give the correct period as measured by the counter-timer, Computer Measurements 226B.

11. Set the rectifier switch to the desired position.

12. Set the upper and lower frequency limits of the filter within 100 cycles of the anticipated passband.

13. Return the tape recorder to that part of the tape that contains the data to be summed, and start the playback.

14. Push the CAT start button.

15. Push momentary switch MP4C on the EAI DES 30 panel. The first triggering pulse will start the CAT summation. It will continue until the preset counter has counted down to zero. During this period the signal can be monitored on the CAT oscilloscope. To be sure that the optimum sweep trigger setting has been used, the process should be repeated, increasing and decreasing the summing period in approximately 10-microsecond steps.

16. To make a permanent recording of the summed data, insert paper in the X-Y recorder and turn on the vacuum switch on the left side.

17. To bring the CAT memory readout to the first address, do the following in the order shown:

    Push the stop switch.

    Push the readout switch.

18. Place the right-hand switch of the X-Y recorder in the OPERATE-PEN UP position. The pen will now move to the start position and can be adjusted by a zeroing pot. The ordinate can be adjusted by the Y zeroing pot.

19. The position of the X-Y pen in the last memory address can be checked by depressing the F. S. (X) button (S102) in the lower left corner of the rear CAT panel. The pen position may be set by adjusting the X-Y gain while the FS(X) button is depressed. Since this may affect the zero position, it should be rechecked and readjusted if necessary. Two or three adjusting cycles may be needed for proper positioning of the X-Y pen.

20. Place the PLOT-PRINT switch in the PLOT position. When all of the memory has been read out, return this switch to the PRINT position.

If a spectral analysis of the summed data is desired, the following additional steps should be followed:

21. Place the EXT.-ADD-SUBTRACT switch on EXTERNAL.

22. Turn the sweep-rate rotary switch to EXT.

23. Turn the input switch off.

24. PLOT-PRINT switch should be on PRINT.

25. Push the stop switch.

26. Push the readout switch. This places the readout in the first memory address.

27. Push the stop switch.

28. Push the start switch.

29. See the Singer Metrics Manual for calibrating and operating instructions of the spectral analysis equipment.

30. Unless the address advance frequency is adjusted for different sweep rates during summation and playback, the frequency spectrum from the analyzer will be shifted from its true value, as in Equation 78.

    To make playback frequencies the same as those originally tape-recorded, set the address advance frequency as follows:

$$f_{\text{address}} = \frac{400}{T \cdot R} \qquad (80)$$

    where T = CAT sweep period during summation

    R = ratio of tape recorder speed during playback to recorder speed during recording.

    Calculate the frequency using Equation 80, and adjust the pulse generator (Intercontinental Instruments PG-1) to correspond to this value.

31. Using Equation 79, set the correct number of CAT addresses in DC2 and 3 on the EAI-DES-30 panel.

32. Push latch switch PB4A on the EAI-DES-30. The correct amount of memory is now extracted from the CAT memory and continuously circulated.

33. Start the Singer spectral analyzer.

34. At the end of the spectral analysis, again push latch switch PB4A on the EAI-DES-30. This stops recirculation of the CAT memory.

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Section 7

RESULTS OF ANALYSES

In this section of the report actual analyses of signals will be presented and discussed as recorded on the test facility at the General Electric Company's Research and Development Center (R&DC) and the Network Test and Training Facility (NTTF) of Goddard Space Flight Center (GSFC). These may be compared with the signature predictions described earlier. It must be emphasized that all of the components tested with defects were fully operable, and that their malfunctions would not have been detected during their usual operation.

SERVO PUMP

The signature of the normal servo pump has been measured by the summation, or transient averaging technique for a wide variety of operating conditions. Since the pump is driven by a large induction motor its speed is relatively invariant, but, depending on the demands of the system, it can vary in flow and discharge pressure. The direction and quantity of the flow is set by the yoke angle (β in Figure 1); this is in turn set by hydraulic actuators via the servo valve. Figure 43 shows a measurement of percentage of full stroke, as a function of antenna velocity. The determination was made by reading the yoke position indicator on the antenna at NTTF as the antenna was slewed (temperature about 95° F, wind calm).

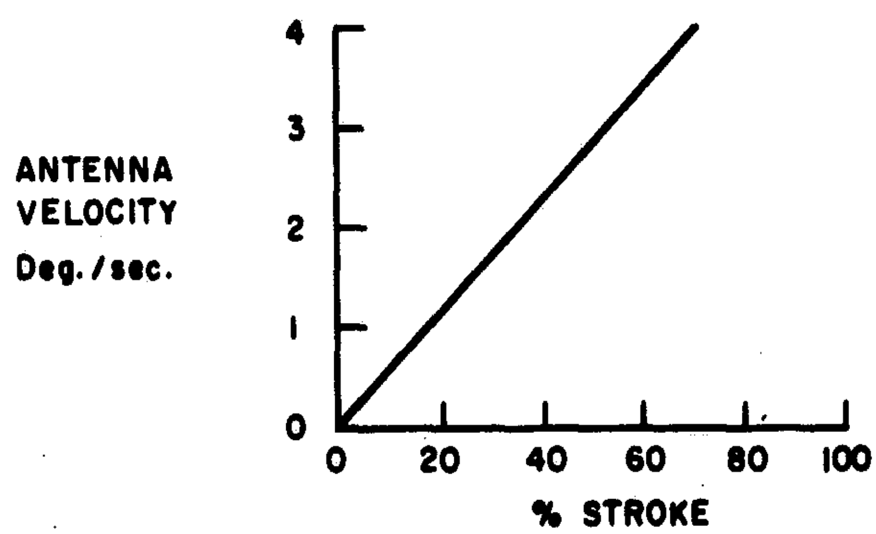


Figure 43. Servo Pump Stroke as a Function of Antenna Velocity

The pressure developed by the pump depends on the load imposed by the hydraulic motors. While there are no pressure gauges on the actual antenna drive, the instruction manual says that it normally runs about 500 psi. High winds, unbalance of the antenna structure, and friction in the gears are cited as factors influencing the pressure.

While the flow and pressure were not both individually adjustable on the antenna, they could be controlled at will in laboratory tests. The percentage of full stroke could be varied with the manual control on the servo amplifier (Figure 33), and the discharge pressure was varied by means of the throttle valve shown in Figure 32. Measurements were made at nominal values of zero stroke, half stroke, and full stroke, but the greatest emphasis was placed on the half-stroke condition, since this is fairly typical of the stroke achieved in practice at moderate and high antenna velocities (Figure 43).

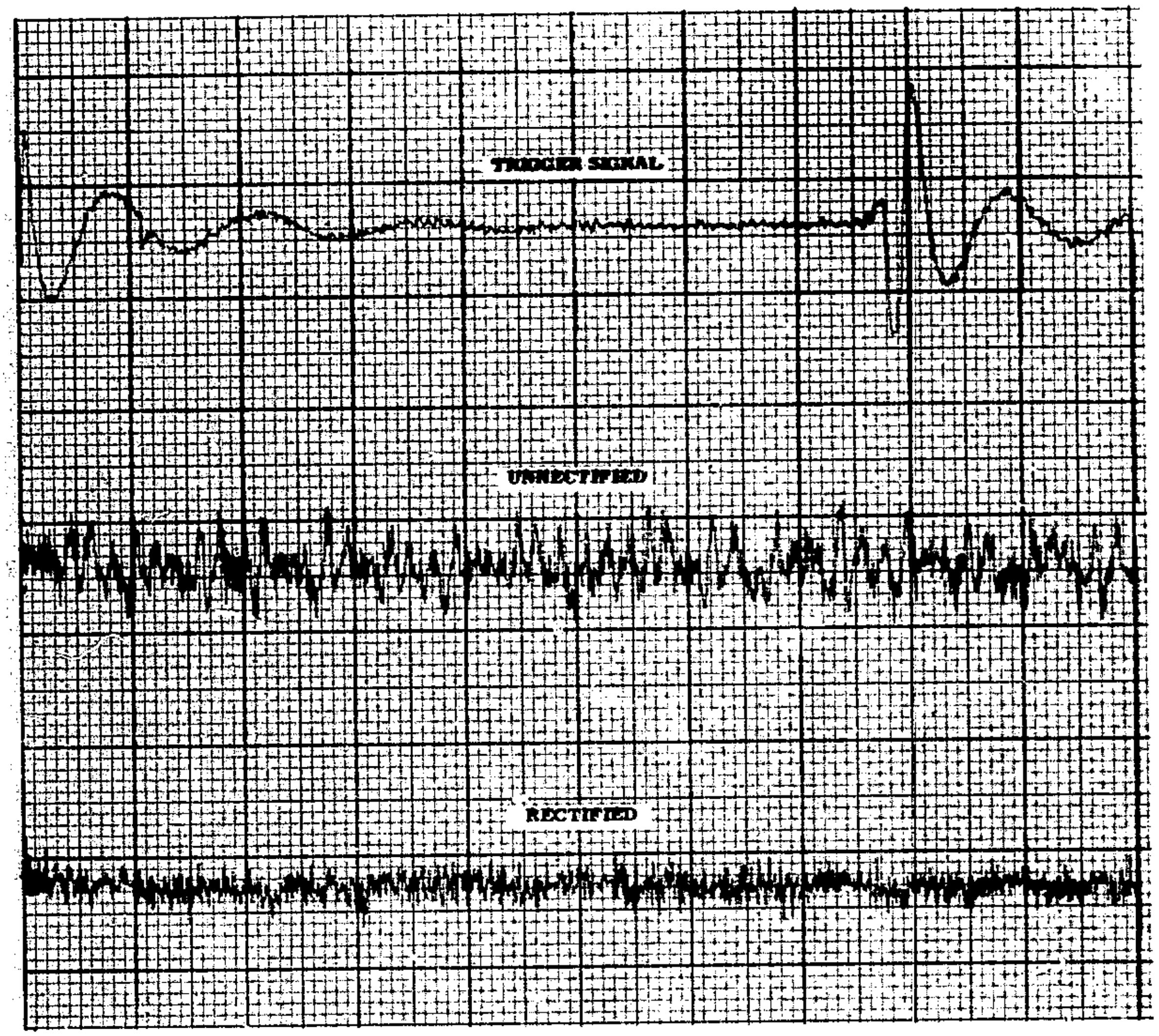
When the yoke is driven to produce full stroke there is a mechanical stop to limit the angle β to a tolerable maximum value; when the stop is reached, the pump noise changes abruptly. Since this effect has limited operational significance it was not studied; it does mean, however, that the analyses at full stroke are of limited usefulness, since the signature could vary greatly with a negligible change in pump operation.

Figure 44 shows summation analyses at zero stroke and zero pressure, with both the raw signal and the rectified signal being summed. (In these analyses the transducer location was over the servo pump roller bearing, except as noted.) In common with subsequent analyses, a representation of the once-per-revolution trigger signal is also included. If the stroke is exactly zero, the servo pump has no displacement and no flow, so the vibration signature should have no character, as was the case. When stroke is increased, even with the throttle valve open so that pressure is very low, the situation changes and the summed signal (particularly that analyzed without rectification) shows a distinctive repetitive character (Figure 45).

Certain sequences within the waveform repeat seven times per revolution, corresponding to the seven pistons of the pump. As pressure increases to 400, then 1100 psi (Figures 46 and 47), the vibration increases in amplitude and the relative amplitude of the harmonics changes, but the general character of the signal is about the same. Figures 48 and 49 are analyses at full stroke, at 0 and 1100 psi.

If the signal is passed through a 1000-hertz low-pass filter to remove noise due to flow and friction, the waveform captured is relatively "cleaner" for a given number of summations than if the raw signal is used. Figures 50 and 51 are examples, corresponding to Figures 46 and 47. The same taped data were analyzed twice in Figure 50 to demonstrate repeatability. This prefiltering may be useful at times, but should be supplemented by an analysis of the raw signal as well, to ensure that the filtering has not thrown away significant information.

Corresponding analyses of the pump in the laboratory after leakage was introduced are shown in Figures 52 to 55 (zero stroke; 50 percent stroke at 0, 400, and 1100 psi, no filtering before summation) and Figures



TEST LOCATION

Laboratory Test Facility	X
Network Test and Training Facility	

SERVO PUMP OPERATING PARAMETERS

Stroke, % : 0
Pressure, PSI: 0

SIGNAL PRECONDITIONING

No Filter	X
1 kHz LP Filter	
20 kHz HP Filter	
Unrectified	X
Full Wave Rectified	X

ANALYSIS INSTRUMENTATION

GE - R&DC	X
NASA - GSFC	

TRIGGER SIGNAL

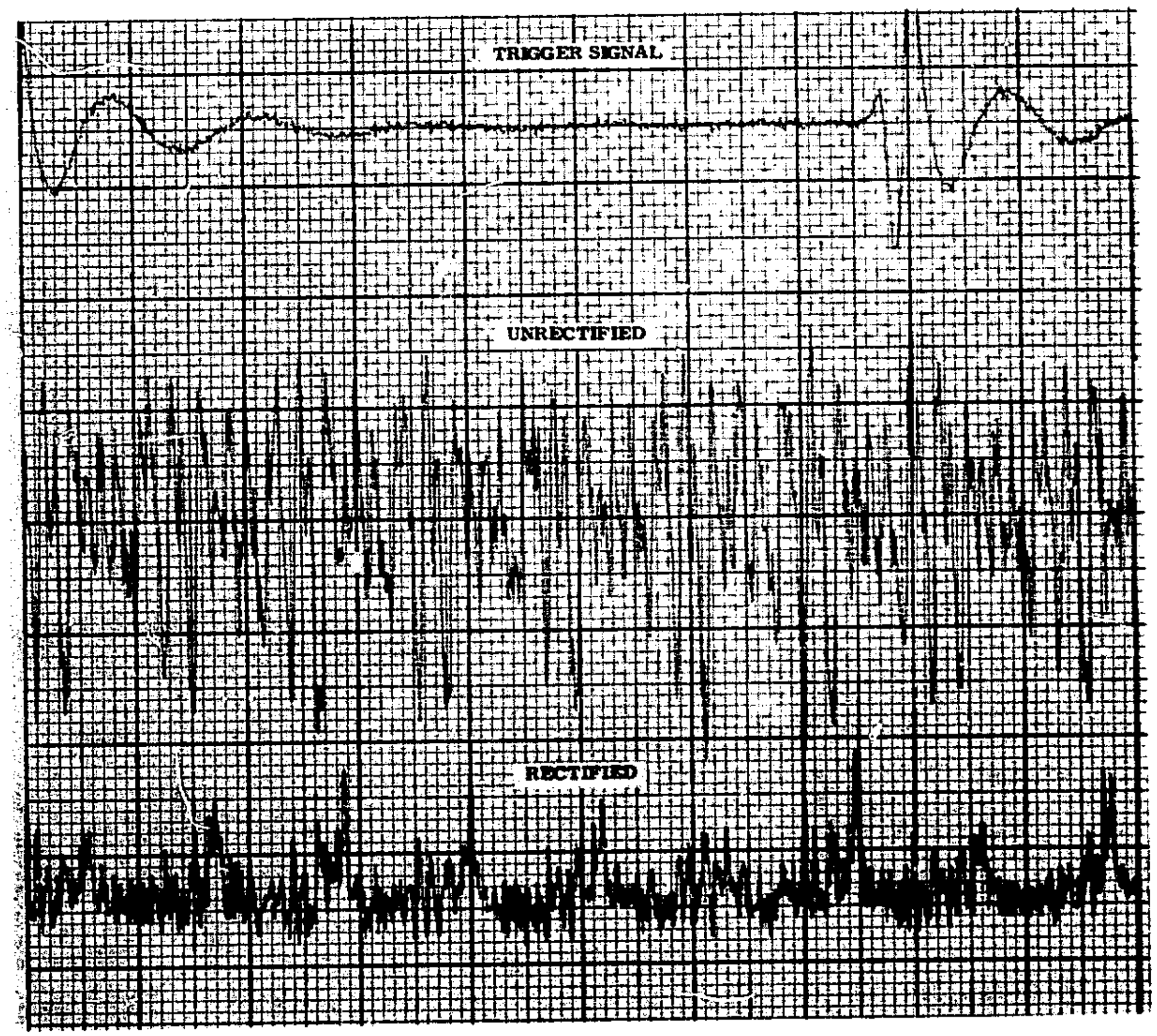
Shaft Position Sensor	X
Pulse Generator	
Switch	

Period, msec Real Time:

SUMMATION PARAMETERS

Number of Summations: 200
Sweep Period, msec Real Time: 62.5

Figure 44. Summation Analyses, Normal Servo Pump



TEST LOCATION

Laboratory Test Facility	X
Network Test and Training Facility	

SERVO PUMP OPERATING PARAMETERS

Stroke, % : 50
Pressure, PSI : 0

SIGNAL PRECONDITIONING

No Filter	X
1 kHz LP Filter	
20 kHz HP Filter	
Unrectified	X
Full Wave Rectified	X

ANALYSIS INSTRUMENTATION

GE - R&DC	X
NASA - GSFC	

TRIGGER SIGNAL

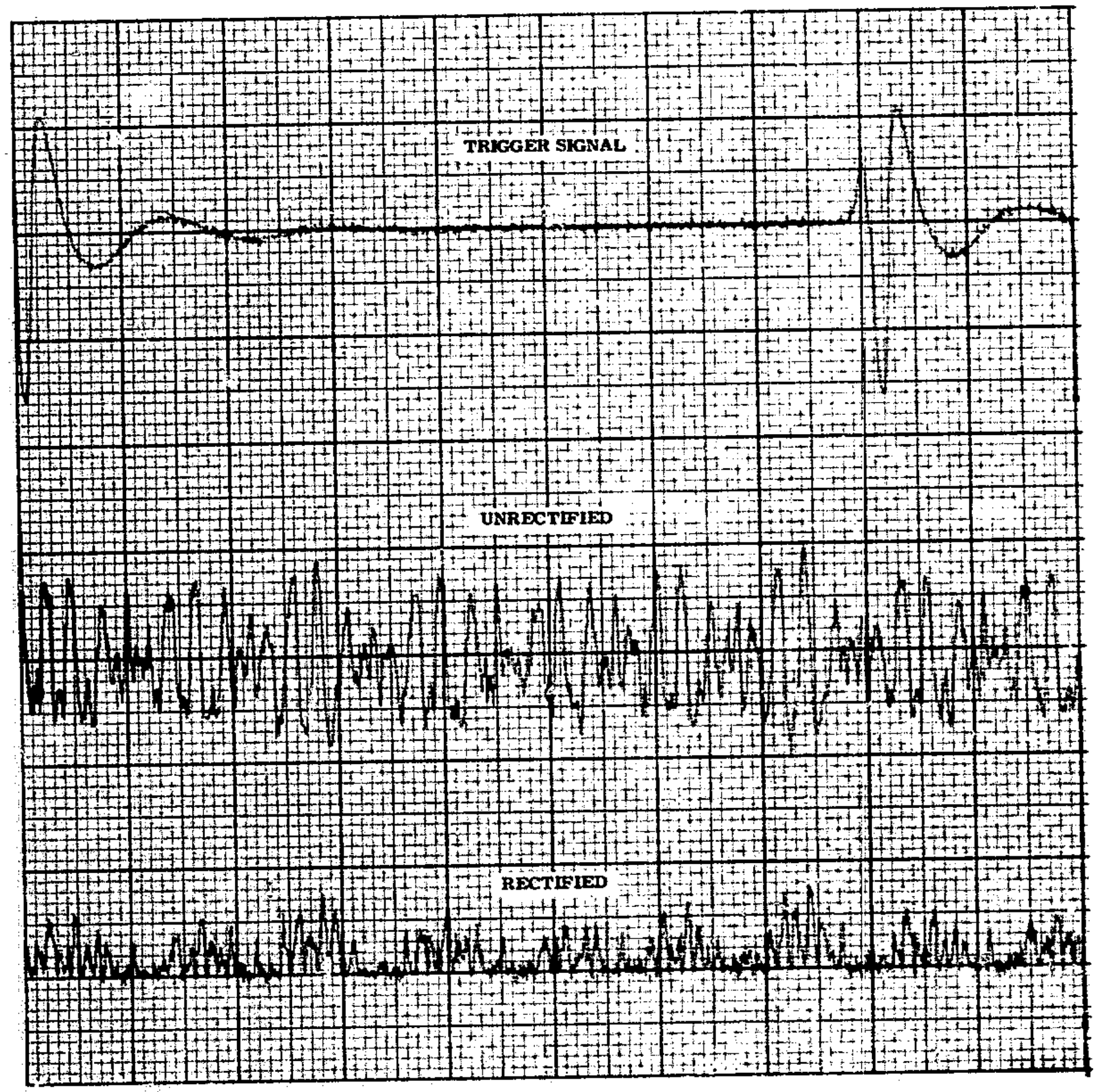
Shaft Position Sensor	X
Pulse Generator	
Switch	

Period, msec Real Time:

SUMMATION PARAMETERS

Number of Summations: 200
Sweep Period, msec Real Time: 62.5

Figure 45. Summation Analyses, Normal Servo Pump



TEST LOCATION

Laboratory Test Facility	X
Network Test and Training Facility	

SERVO PUMP OPERATING PARAMETERS

Stroke, % : 50
Pressure, PSI : 400

SIGNAL PRECONDITIONING

No Filter	X
1 kHz LP Filter	
20 kHz HP Filter	
Unrectified	X
Full Wave Rectified	X

ANALYSIS INSTRUMENTATION

GE - R&DC	X
NASA - GSFC	

TRIGGER SIGNAL

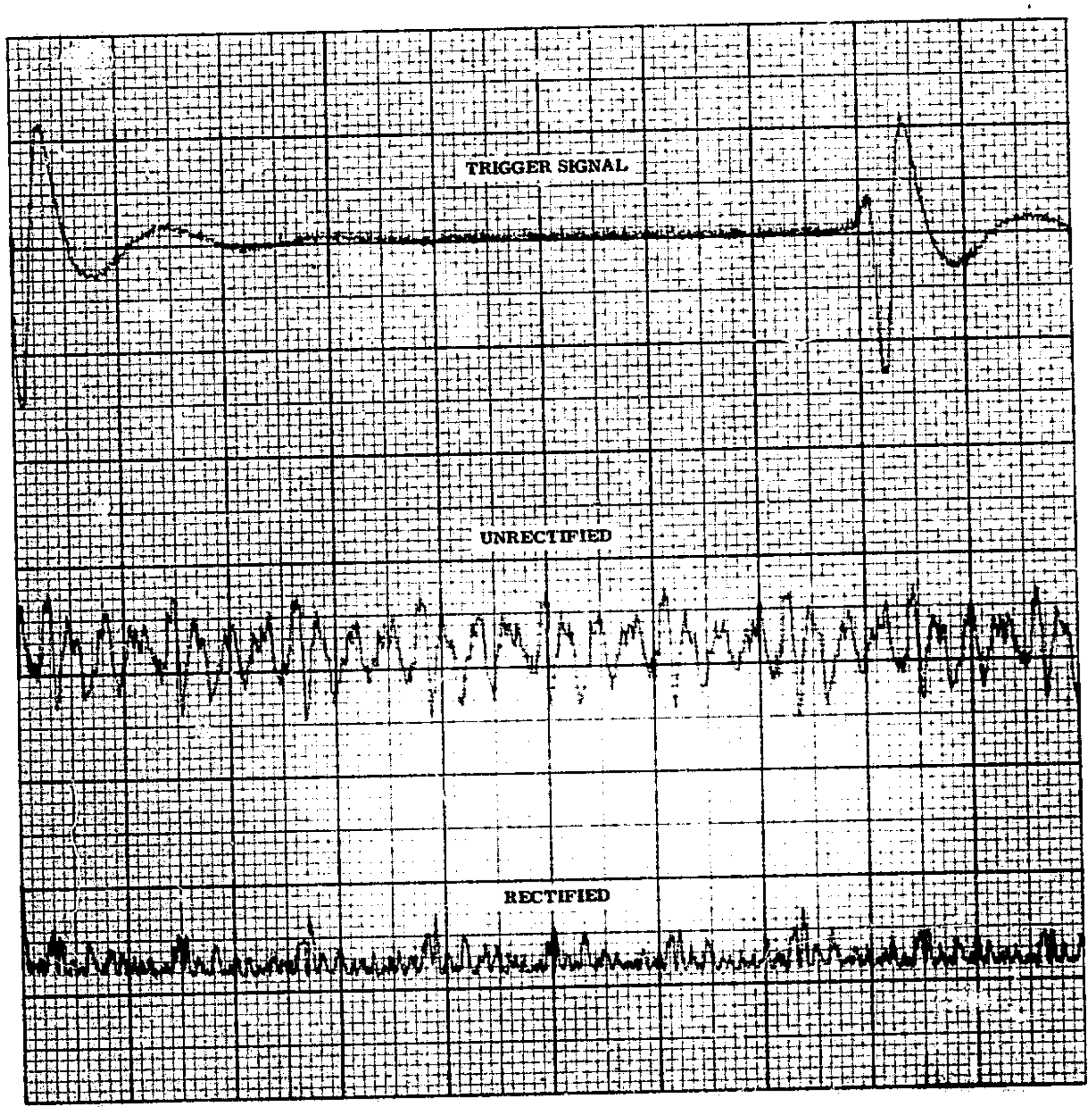
Shaft Position Sensor	X
Pulse Generator	
Switch	

Period, msec Real Time:

SUMMATION PARAMETERS

Number of Summations: 100
Sweep Period, msec Real Time: 62.5

Figure 46. Summation Analyses, Normal Servo Pump



TEST LOCATION

Laboratory Test Facility [X]
Network Test and Training Facility [ ]

SERVO PUMP OPERATING PARAMETERS

Stroke, %: 50
Pressure, PSI: 1100

SIGNAL PRECONDITIONING

No Filter [X]
1 kHz LP Filter [ ]
20 kHz HP Filter [ ]
Unrectified [X]
Full Wave Rectified [X]

ANALYSIS INSTRUMENTATION

GE - R&DC [X]
NASA - GSFC [ ]

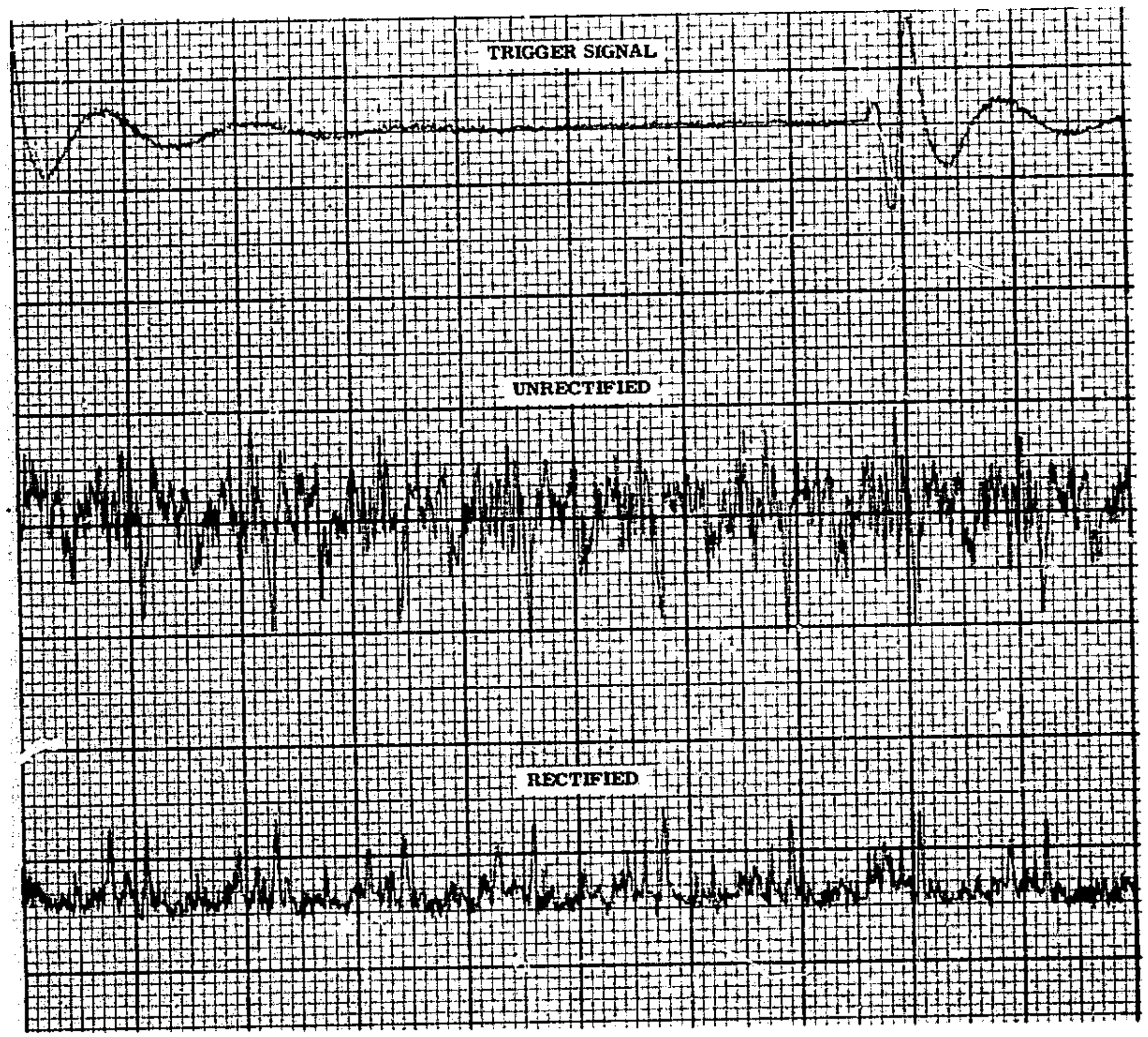
TRIGGER SIGNAL

Shaft Position Sensor [X]
Pulse Generator [ ]
Switch [ ]
Period, msec Real Time:

SUMMATION PARAMETERS

Number of Summations: 100
Sweep Period, msec Real Time: 62.5

Figure 47. Summation Analyses, Normal Servo Pump



TEST LOCATION

Laboratory Test Facility [X]
Network Test and Training Facility [ ]

SERVO PUMP OPERATING PARAMETERS

Stroke, %: 100
Pressure, PSI: 0

SIGNAL PRECONDITIONING

No Filter [X]
1 kHz LP Filter [ ]
20 kHz HP Filter [ ]
Unrectified [X]
Full Wave Rectified [X]

ANALYSIS INSTRUMENTATION

GE - R&DC [X]
NASA - GSFC [ ]

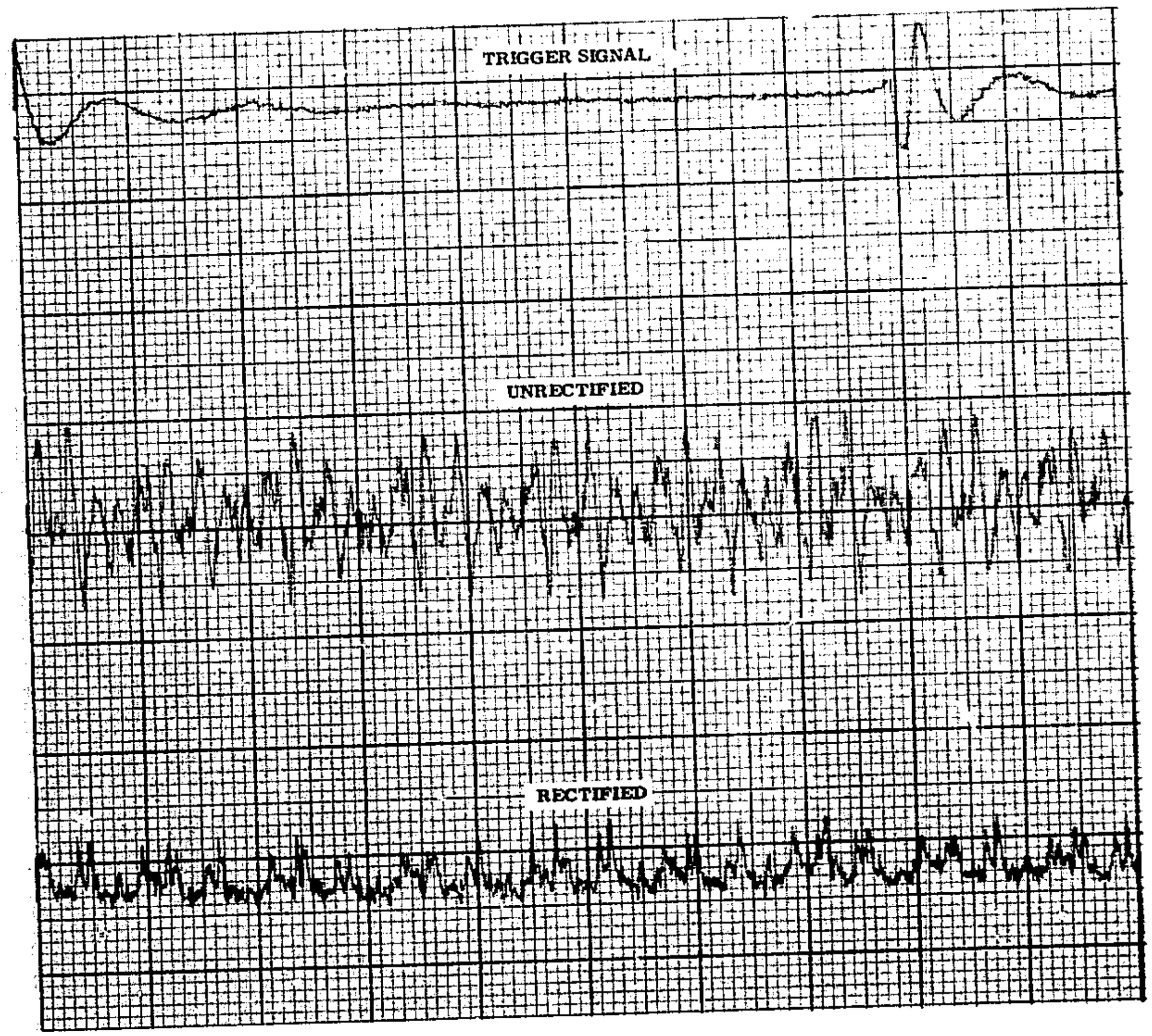
TRIGGER SIGNAL

Shaft Position Sensor [X]
Pulse Generator [ ]
Switch [ ]
Period, msec Real Time:

SUMMATION PARAMETERS

Number of Summations: 200
Sweep Period, msec Real Time: 62.5

Figure 48. Summation Analyses, Normal Servo Pump



TEST LOCATION

Laboratory Test Facility	X
Network Test and Training Facility	

SERVO PUMP OPERATING PARAMETERS

Stroke, %: 100
Pressure, PSI: 1100

SIGNAL PRECONDITIONING

No Filter	X
1 kHz LP Filter	
20 kHz HP Filter	
Unrectified	X
Full Wave Rectified	X

ANALYSIS INSTRUMENTATION

GE - R&DC	X
NASA - GSFC	

TRIGGER SIGNAL

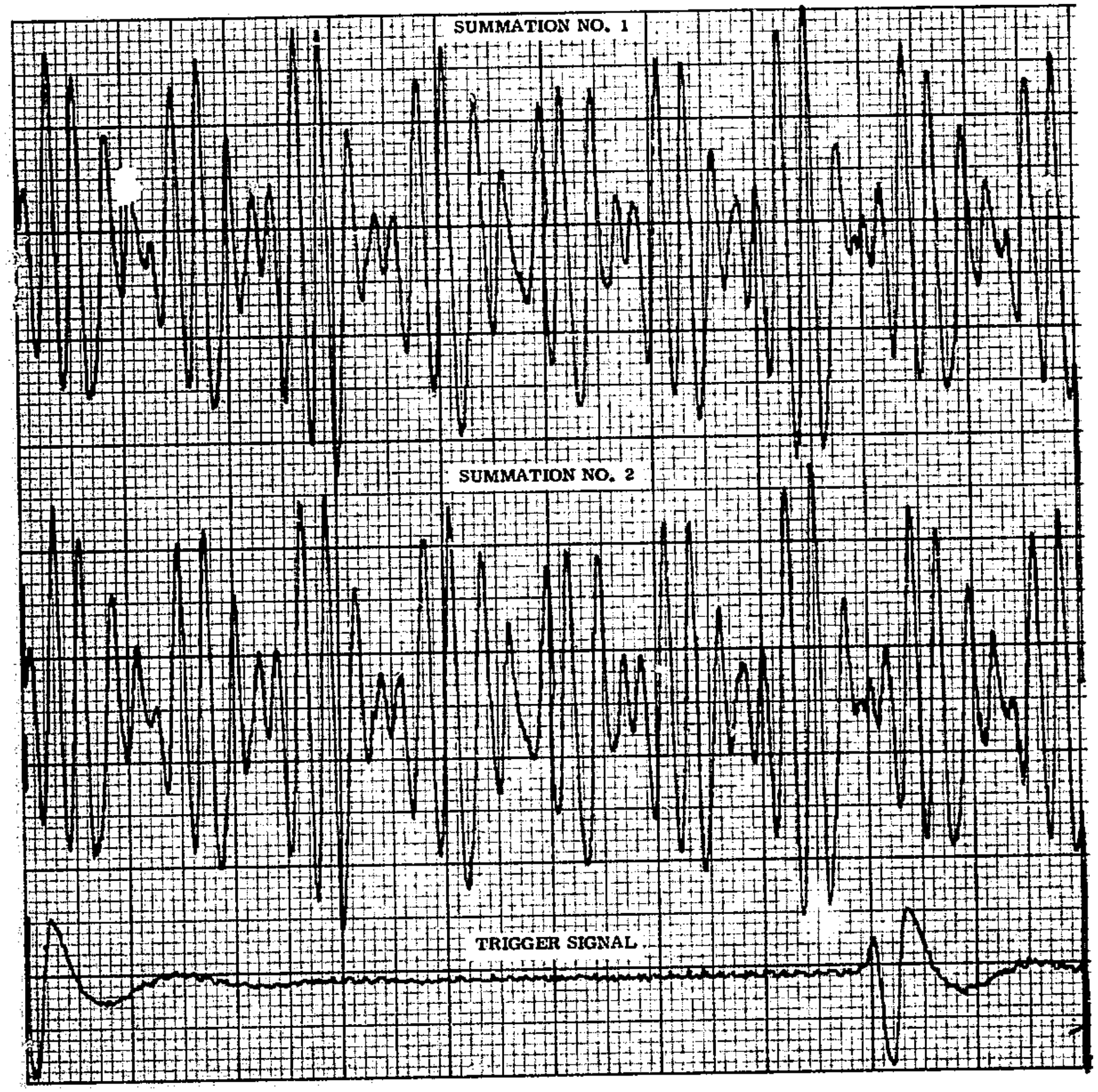
Shaft Position Sensor	X
Pulse Generator	
Switch	

Period, msec Real Time:

SUMMATION PARAMETERS

Number of Summations: 200
Sweep Period, msec Real Time: 62.5

Figure 49. Summation Analyses, Normal Servo Pump



TEST LOCATION

Laboratory Test Facility [X]
Network Test and Training Facility [ ]

SERVO PUMP OPERATING PARAMETERS

Stroke, % : 50
Pressure, PSI: 400

SIGNAL PRECONDITIONING

No Filter [ ]
1 kHz LP Filter [X]
20 kHz HP Filter [ ]
Unrectified [X]
Full Wave Rectified [ ]

ANALYSIS INSTRUMENTATION

GE - R&DC [X]
NASA - GSFC [ ]

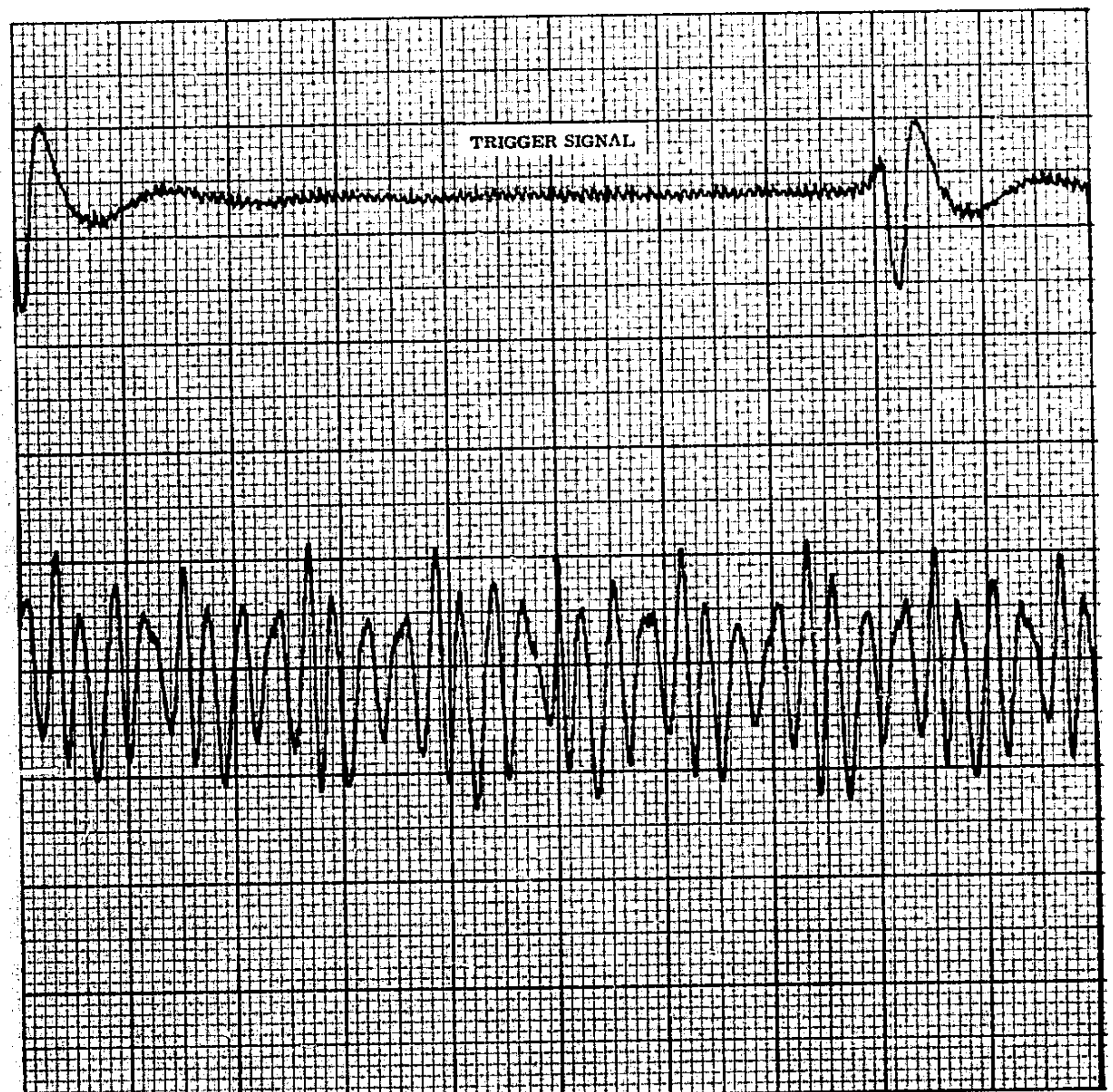
TRIGGER SIGNAL

Shaft Position Sensor [X]
Pulse Generator [ ]
Switch [ ]
Period, msec Real Time:

SUMMATION PARAMETERS

Number of Summations: 200
Sweep Period, msec Real Time: 62.5

Figure 50. Summation Analysis, Normal Servo Pump



TEST LOCATION

Laboratory Test Facility	X
Network Test and Training Facility	

SERVO PUMP OPERATING PARAMETERS

Stroke, %: 50
Pressure, PSI: 1100

SIGNAL PRECONDITIONING

No Filter	
1 kHz LP Filter	X
20 kHz HP Filter	
Unrectified	X
Full Wave Rectified	

ANALYSIS INSTRUMENTATION

GE - R&DC	X
NASA - GSFC	

TRIGGER SIGNAL

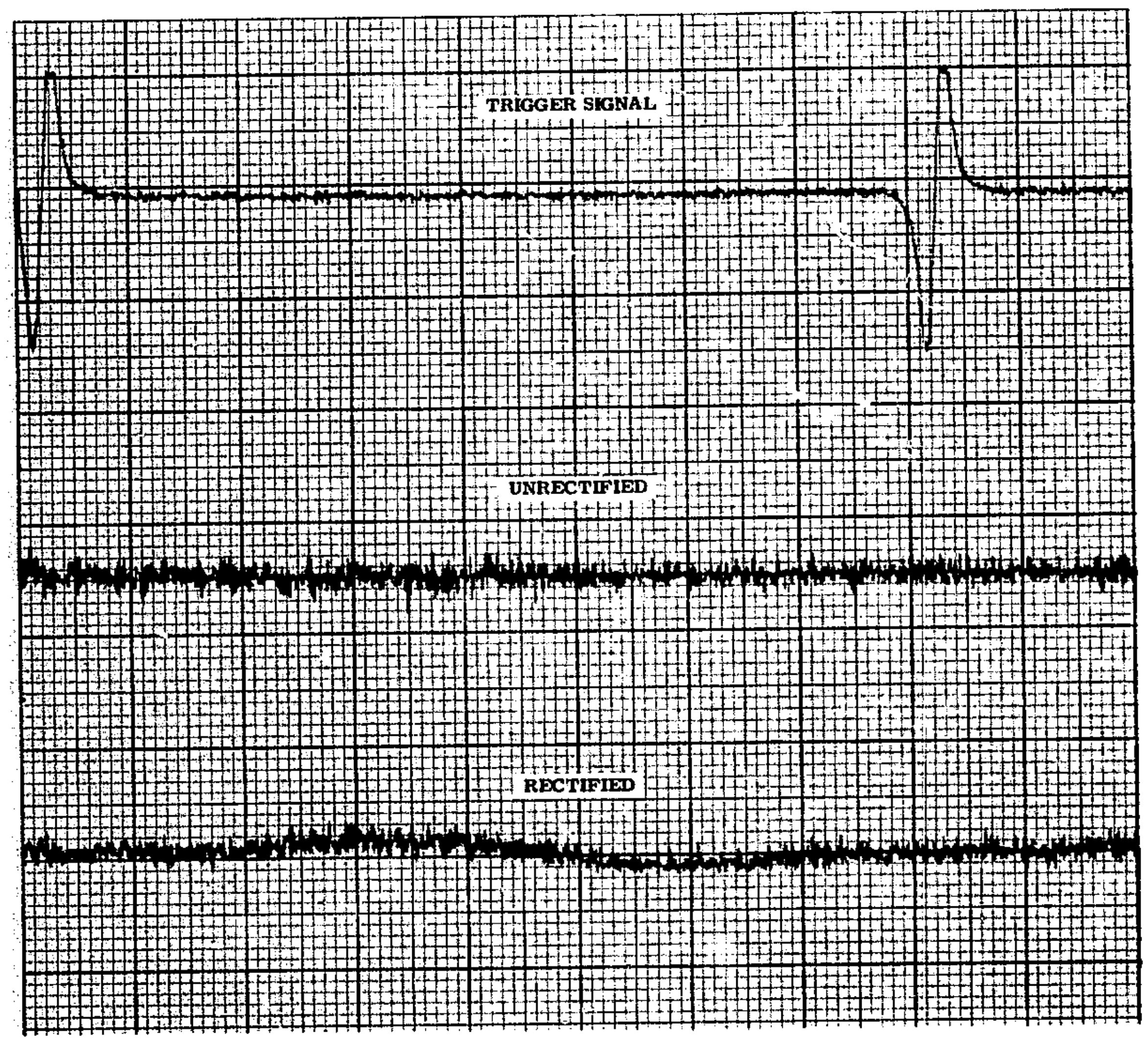
Shaft Position Sensor	X
Pulse Generator	
Switch	

Period, msec Real Time:

SUMMATION PARAMETERS

Number of Summations: 200
Sweep Period, msec Real Time: 62.5

Figure 51. Summation Analysis, Normal Servo Pump



TEST LOCATION

Laboratory Test Facility	X
Network Test and Training Facility	

SERVO PUMP OPERATING PARAMETERS

Stroke, % : 0
Pressure, PSI: 0

SIGNAL PRECONDITIONING

No Filter	X
1 kHz LP Filter	
20 kHz HP Filter	
Unrectified	X
Full Wave Rectified	X

ANALYSIS INSTRUMENTATION

GE - R&DC	X
NASA - GSFC	

TRIGGER SIGNAL

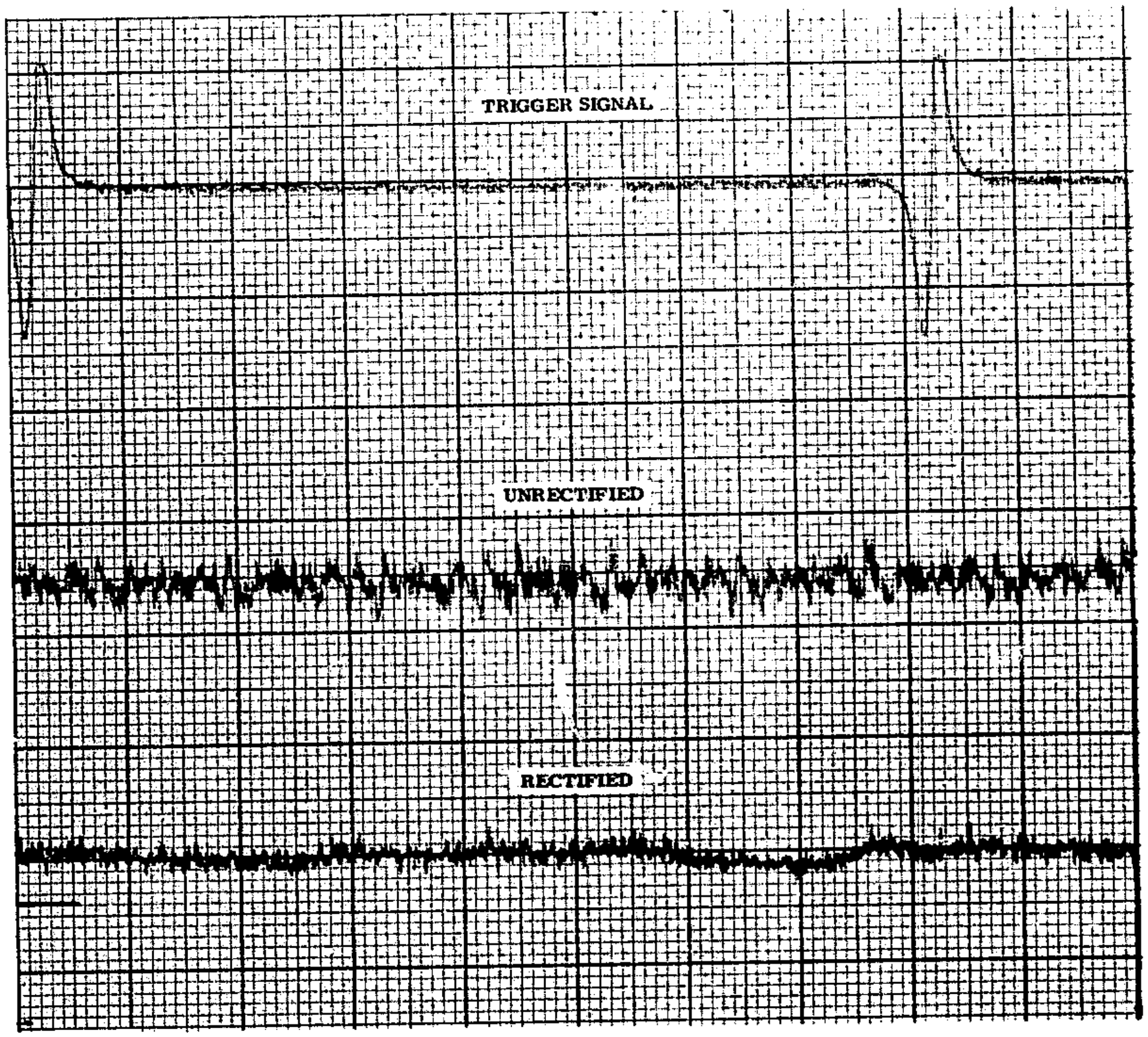
Shaft Position Sensor	X
Pulse Generator	
Switch	

Period, msec Real Time:

SUMMATION PARAMETERS

Number of Summations: 200
Sweep Period, msec Real Time: 62.5

Figure 52. Summation Analyses, Servo Pump With Malfunction



TEST LOCATION

Laboratory Test Facility	X
Network Test and Training Facility	

SERVO PUMP OPERATING PARAMETERS

Stroke, %: 50
Pressure, PSI: 0

SIGNAL PRECONDITIONING

No Filter	X
1 kHz LP Filter	
20 kHz HP Filter	
Unrectified	X
Full Wave Rectified	X

ANALYSIS INSTRUMENTATION

GE - R&DC	X
NASA - GSFC	

TRIGGER SIGNAL

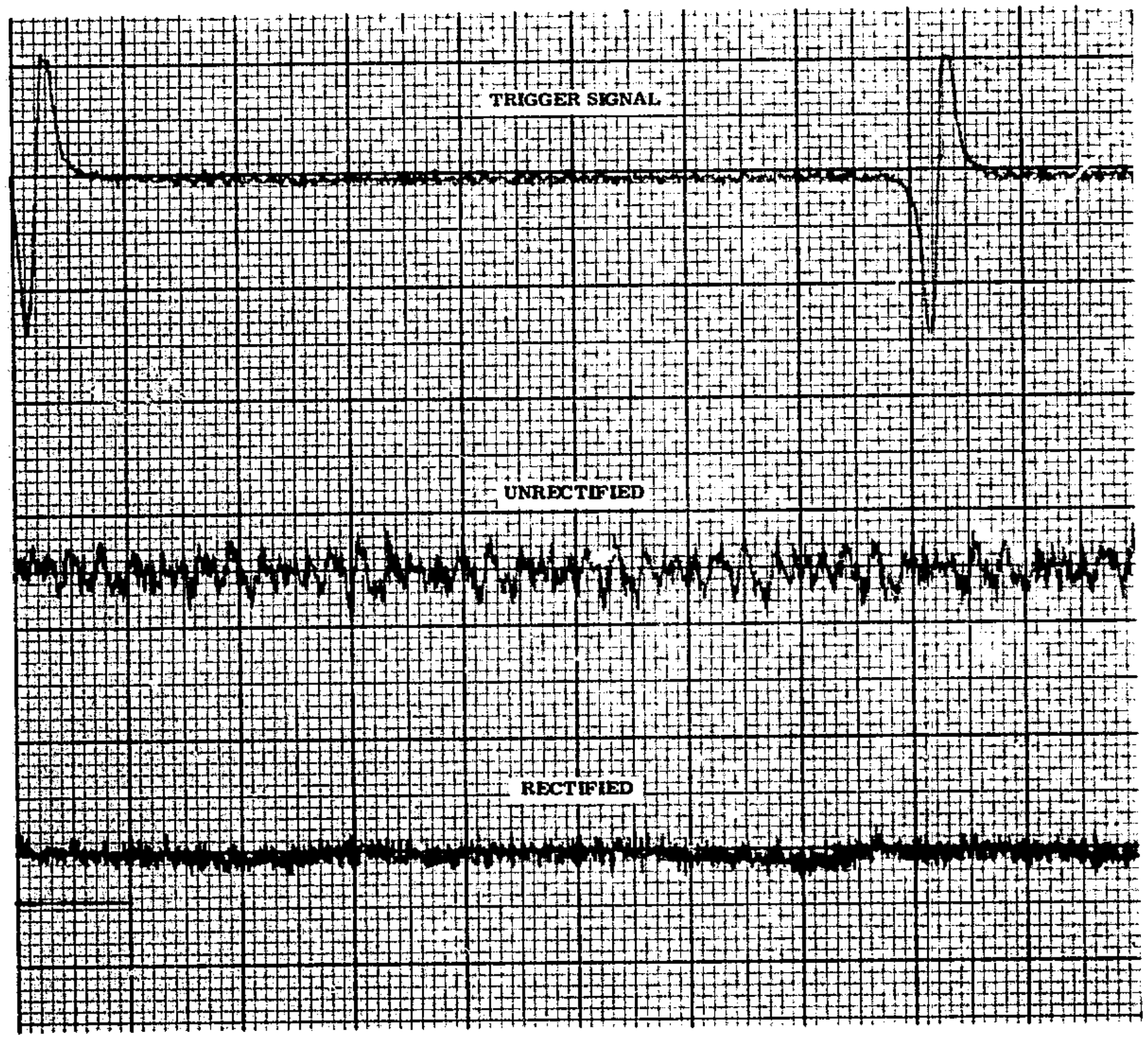
Shaft Position Sensor	X
Pulse Generator	
Switch	

Period, msec Real Time:

SUMMATION PARAMETERS

Number of Summations: 200
Sweep Period, msec Real Time: 62.5

Figure 53. Summation Analyses, Servo Pump With Malfunction



TEST LOCATION

Laboratory Test Facility	X
Network Test and Training Facility	

SERVO PUMP OPERATING PARAMETERS

Stroke, % : **50**
Pressure, PSI: **400**

SIGNAL PRECONDITIONING

No Filter	X
1 kHz LP Filter	
20 kHz HP Filter	
Unrectified	X
Full Wave Rectified	X

ANALYSIS INSTRUMENTATION

GE - R&DC	X
NASA - GSFC	

TRIGGER SIGNAL

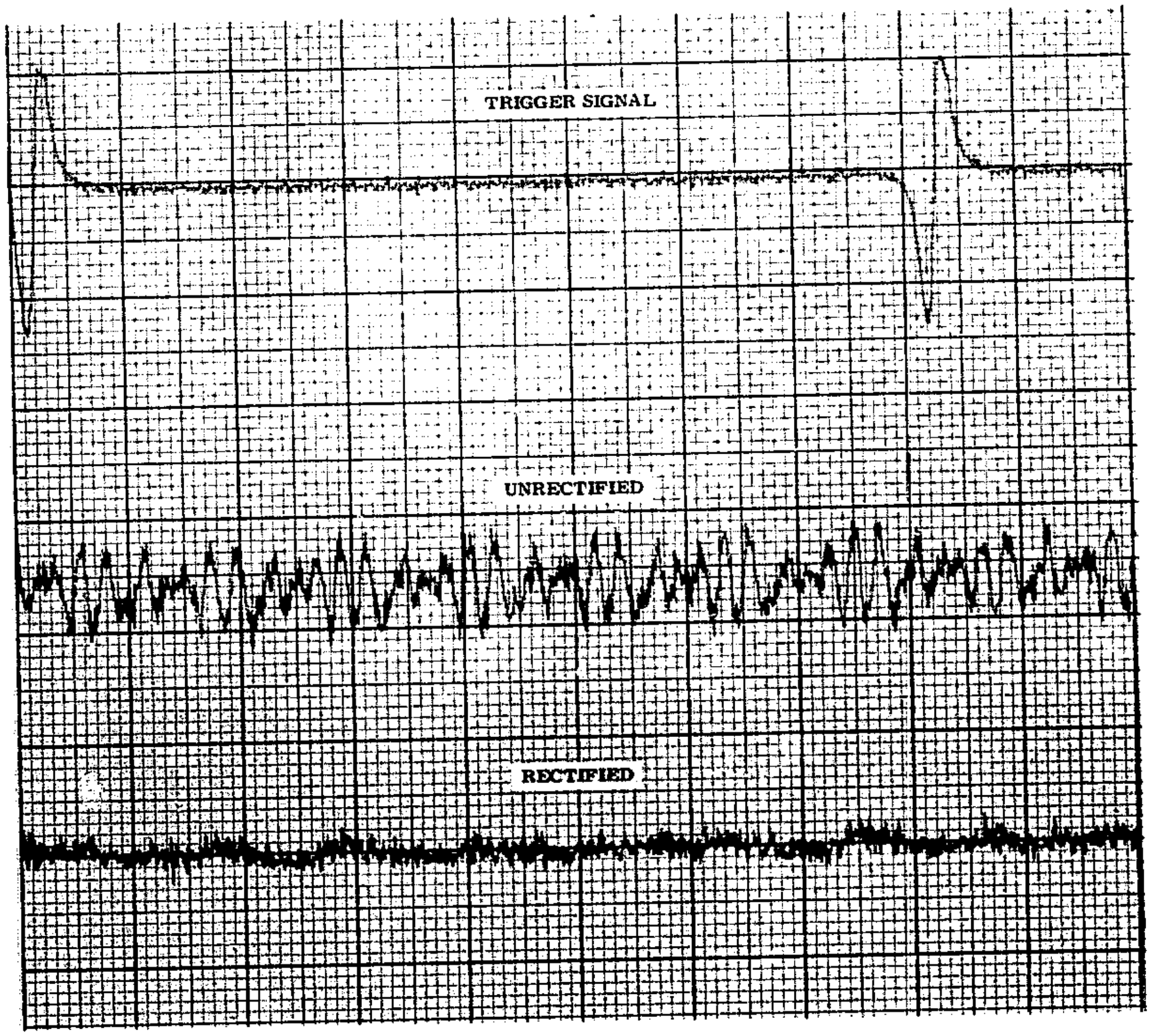
Shaft Position Sensor	X
Pulse Generator	
Switch	

Period, msec Real Time:

SUMMATION PARAMETERS

Number of Summations: **200**
Sweep Period, msec Real Time: **62.5**

Figure 54. Summation Analyses, Servo Pump With Malfunction



TEST LOCATION

Laboratory Test Facility	X
Network Test and Training Facility	

SERVO PUMP OPERATING PARAMETERS

Stroke, % : 50
Pressure, PSI: 1100

SIGNAL PRECONDITIONING

No Filter	X
1 kHz LP Filter	
20 kHz HP Filter	
Unrectified	X
Full Wave Rectified	X

ANALYSIS INSTRUMENTATION

GE - R&DC	X
NASA - GSFC	

TRIGGER SIGNAL

Shaft Position Sensor	X
Pulse Generator	
Switch	

Period, msec Real Time:

SUMMATION PARAMETERS

Number of Summations: 200
Sweep Period, msec Real Time: 62.5

Figure 55. Summation Analyses, Servo Pump With Malfunction

56 and 57 (50 percent stroke at 400 and 1100 psi, 1000 hertz low-pass filter before summation).

Figure 58 shows a series of similar analyses taped and analyzed at NTTF. Antenna velocity was the variable. Pressure is not known, but can be assumed to be about 500 psi; stroke can be estimated from Figure 43. The expected increase in vibration with increasing stroke is very clear.

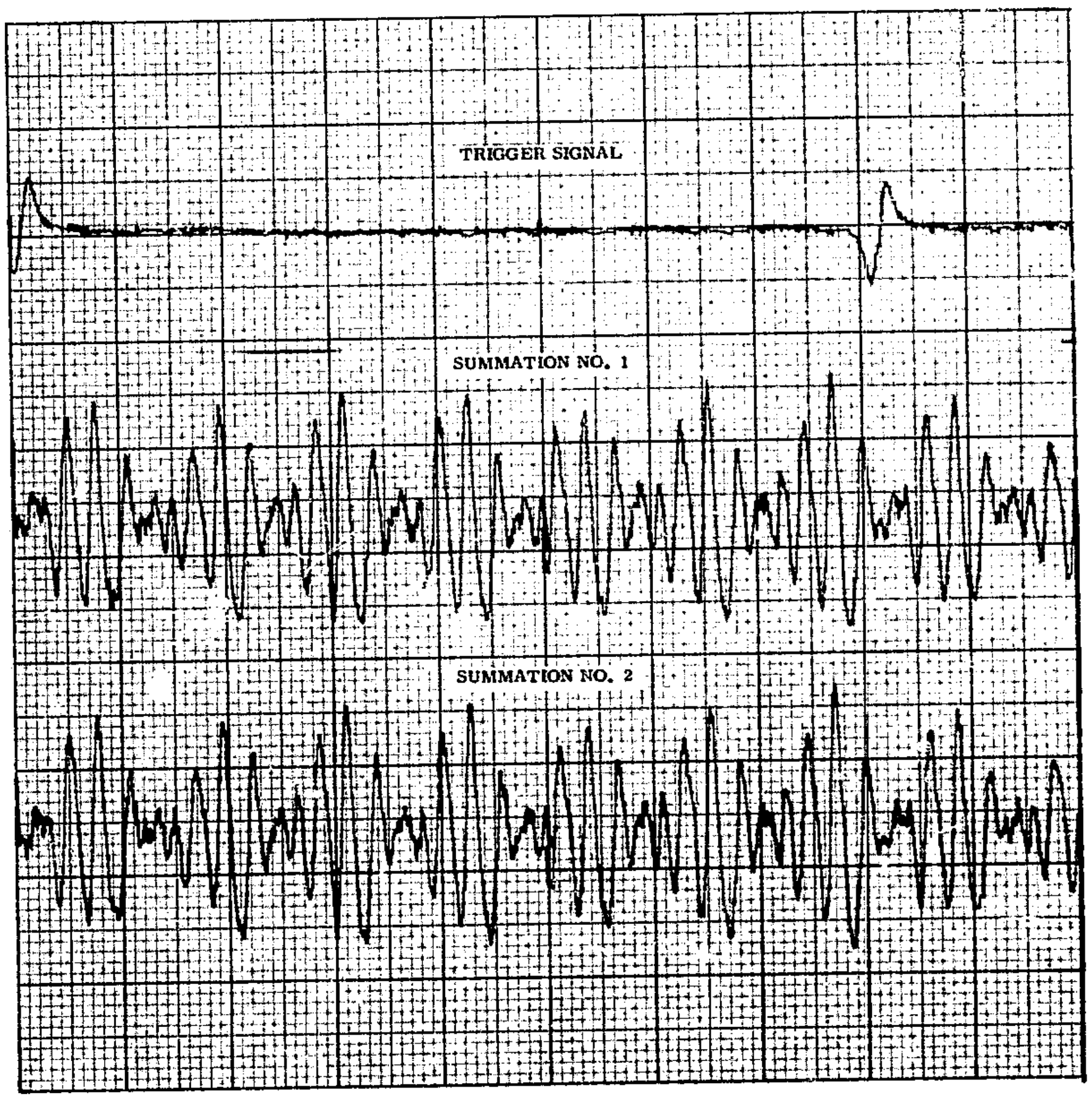
It is interesting to compare traces 12 and 14, which are comparable except for direction of rotation of the antenna. In terms of pump operation, this means that the sign of the angle β in Figure 1 was reversed. The two traces match quite well if one of the two is reversed in phase (i.e., down-going portions of the curve are made up-going) and in time (i.e., time is assumed to run from right to left).

The reversal of the waveform in time comes simply from the change in β. With direction of rotation held constant, the sequence of events leading to the ripple reverses when β reverses. This can be shown by means of computer program GSFC23 (Appendix II). However, the program does not predict a change in phase or sign of the disturbances. This tells much about the way that the pressure fluctuations are coupled to the outside of the pump. If the vibrations were due to hoop stresses such as are typical of a pressure vessel, they would not have changed sense or sign as the sign of β was reversed. However, if the vibrations were due to moments generated between rotating and stationary portions of the pump, the sense of the fluctuations would change with the sign of β. Since the latter condition is consistent with the experimental evidence, it is safe to assume that the motion measured on the exterior of the servo pump is due chiefly to vibratory moments which are driven by the pressure fluctuations.

Since the responsiveness of the mechanical structure will differ with frequency, one should not expect a 1:1 relationship between the predicted pressure waveform and the measured vibration. At low frequencies the fluctuating forces act, in effect, against a very stiff spring. At some higher frequency the structure begins to "break up" because of resonances, and as a consequence the vibration amplitude for a given driving force is greater.

The validity of this prediction is demonstrated quantitatively in Table 6, which compares predicted pressure fluctuations and measured vibrations as a function of frequency for a normal pump, operating at 50 percent stroke and a pressure of 1400 psi. The last column is the vibration (in decibels relative to one g) per psi of pressure, as a function of frequency. The trend of this column shows that the responsiveness of the structure is about the same at 140 and 280 hertz, but that increasingly larger vibration amplitudes are realized for a given force at 420 and 560 hertz.

Unfortunately, the stiffness of the structure at low frequencies discriminates against the responses, particularly at once per revolution or 20 hertz,



TEST LOCATION

Laboratory Test Facility	X
Network Test and Training Facility	

SERVO PUMP OPERATING PARAMETERS

Stroke, % : 50
Pressure, PSI: 400

SIGNAL PRECONDITIONING

No Filter	
1 kHz LP Filter	X
20 kHz HP Filter	
Unrectified	X
Full Wave Rectified	

ANALYSIS INSTRUMENTATION

GE - R&DC	X
NASA - GSFC	

TRIGGER SIGNAL

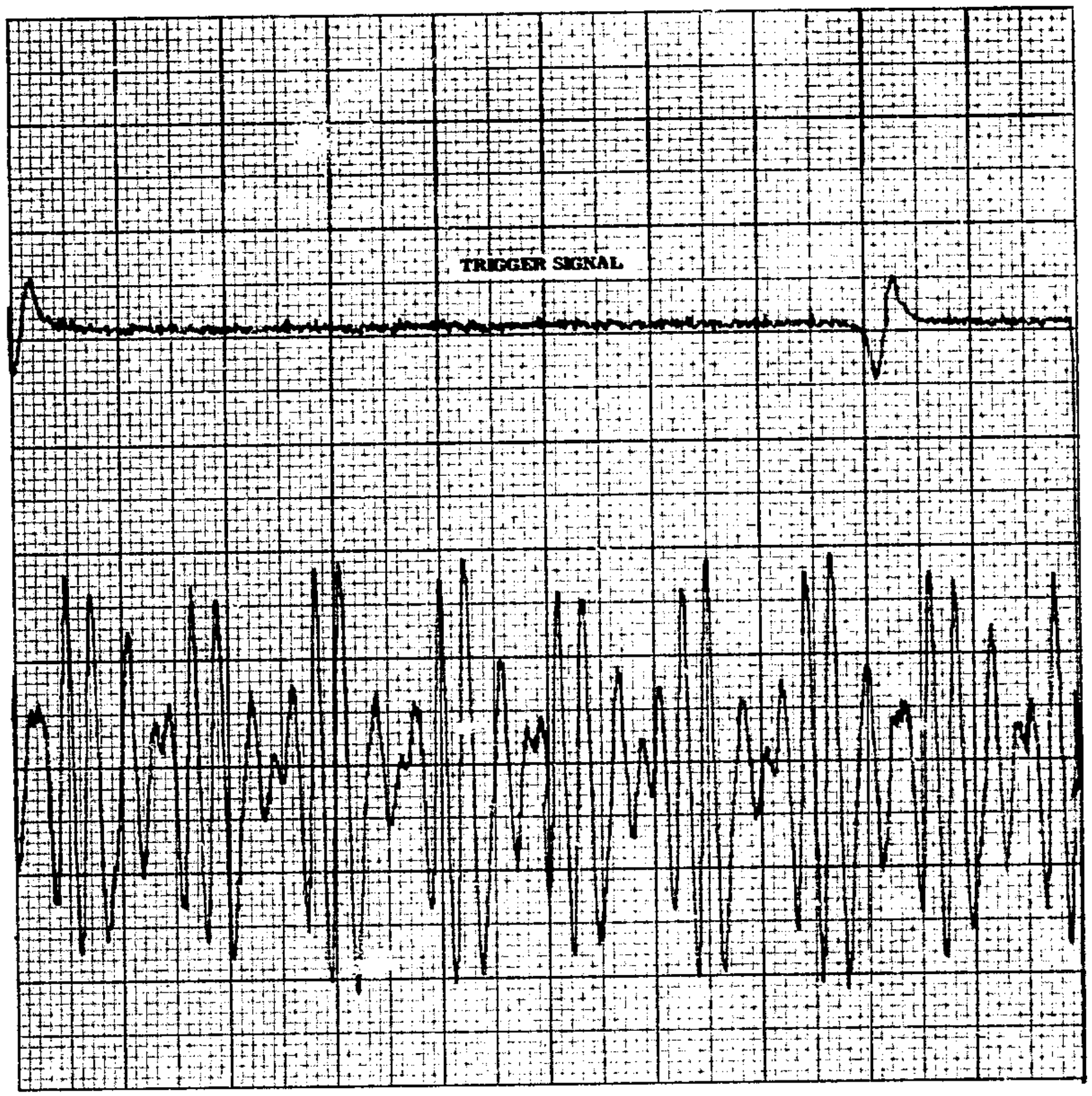
Shaft Position Sensor	X
Pulse Generator	
Switch	

Period, msec Real Time:

SUMMATION PARAMETERS

Number of Summations: 200
Sweep Period, msec Real Time: 62.5

Figure 56. Summation Analysis, Servo Pump With Malfunction



TEST LOCATION

Laboratory Test Facility	X
Network Test and Training Facility	

SERVO PUMP OPERATING PARAMETERS

Stroke, % : **50**
Pressure, PSI: **1100**

SIGNAL PRECONDITIONING

No Filter	
1 kHz LP Filter	X
20 kHz HP Filter	
Unrectified	X
Full Wave Rectified	

ANALYSIS INSTRUMENTATION

GE - R&DC	X
NASA - GSFC	

TRIGGER SIGNAL

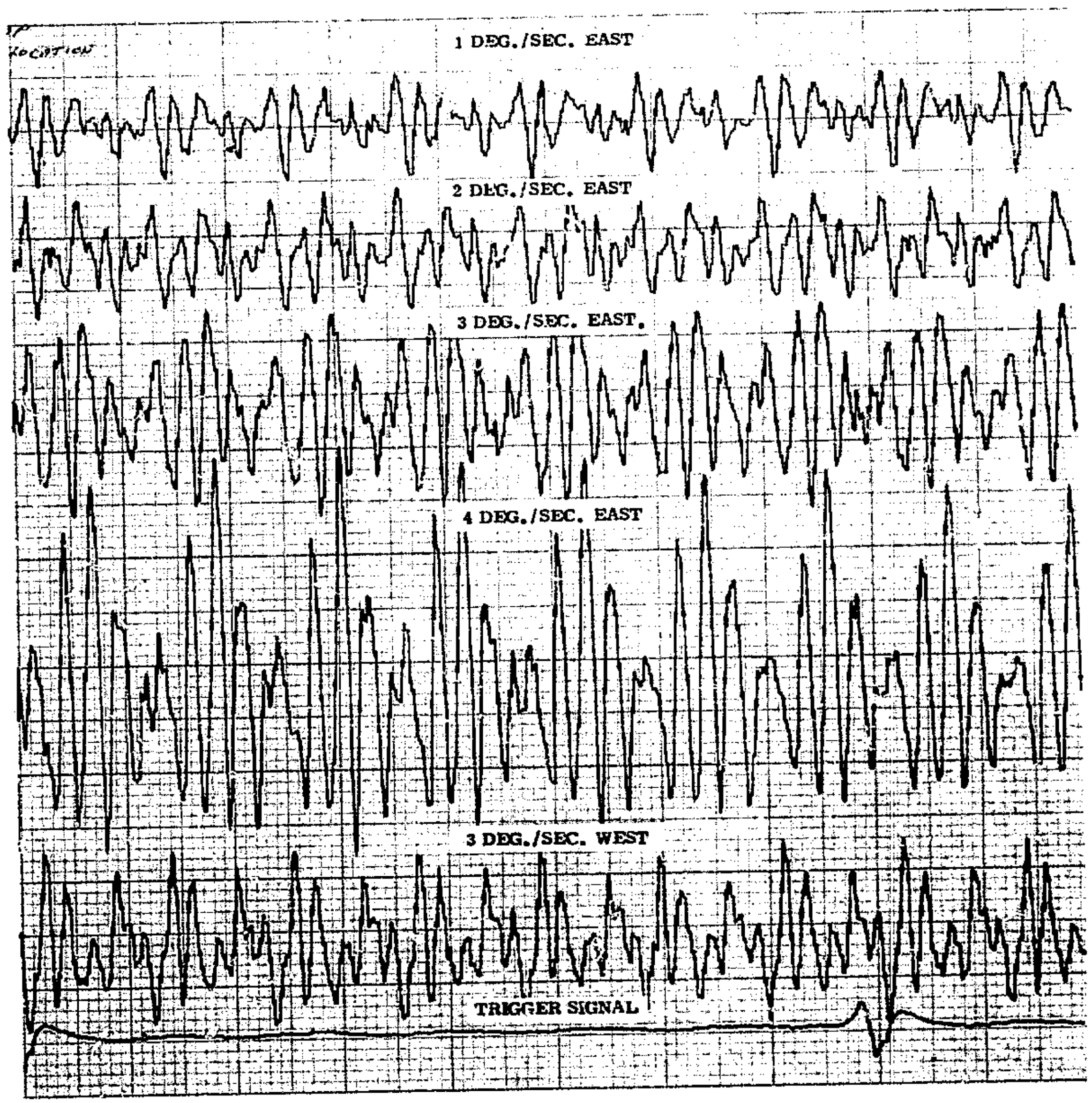
Shaft Position Sensor	X
Pulse Generator	
Switch	

Period, msec Real Time:

SUMMATION PARAMETERS

Number of Summations: **200**
Sweep Period, msec Real Time: 62.5

Figure 57. Summation Analysis, Servo Pump With Malfunction



TEST LOCATION

Laboratory Test Facility	[ ]
Network Test and Training Facility	[X]

SERVO PUMP OPERATING PARAMETERS

Stroke, % : } See Text
Pressure, PSI : }

SIGNAL PRECONDITIONING

No Filter	[X]
1 kHz LP Filter	[ ]
20 kHz HP Filter	[ ]
Unrectified	[X]
Full Wave Rectified	[ ]

ANALYSIS INSTRUMENTATION

GE - R&DC	[ ]
NASA - GSFC	[X]

TRIGGER SIGNAL

Shaft Position Sensor	[X]
Pulse Generator	[ ]
Switch	[ ]
Period, msec Real Time:	

SUMMATION PARAMETERS

Number of Summations: **200**
Sweep Period, msec Real Time: **62.5**

Figure 58. Summation Analyses, Servo Pump With Malfunction

Table 6

COMPARISON OF PREDICTED PRESSURE SPECTRA
AND MEASURED VIBRATION SPECTRA OF SERVO PUMP

Frequency (Hz)	Predicted Pressure (dB re 1 psi)	Measured Vibration (dB re 1 g)	Vibration, dB re 1 g per psi of Pressure
140	32	-17	-49
280	27	-23	-50
420	21	-13	-34
560	13	- 4	-17

which are indicative of a leaky cylinder. The waveforms shown do not have an obvious once-per-revolution component which can be identified with the inserted malfunction. They have been supplemented with spectrum analyses, made with the 2 hertz bandwidth Radiometer analyzer (Figures 59 to 62). While there is an identifiable once-per-revolution component in the analysis of the malfunctioning pump which is not present in the analysis of the normal pump, the increase in vibration amplitude is very small and is not considered reliable as an indicator. The fact that the amplitude of this component does not increase with pressure indicates that it probably arises from some other source than the leaky cylinder.

It will be noticed that there are strong spectral components in many of the runs near 40 and 80 hertz. By displaying these components and the once-per-revolution trigger simultaneously on an oscilloscope it has been shown that they are not synchronous with the shaft rotation, and therefore cannot be associated with the pumping action. Since they are asynchronous, they must arise in the bearings. The strongest spectral components are those which are due to the pumping action, at multiples of seven per revolution.

In view of the difficulties encountered in detecting the leaky cylinder, it is worthwhile considering the size of the leak in question. The computer program (GSFC23) was modified slightly to permit calculation of the volume of fluid leaked per shaft revolution; this was related to the nominal displacement to give the percentage of fluid leaked. Under the usual conditions of 50 percent stroke and 500 psi pressure, the leakage for all cylinders (through the annulus between piston and cylinder) totals 2.7 percent. If the clearance of one cylinder is increased from 1.0 to 1.4 mils, the leakage increases only to 3.4 percent, or a change of 0.7 percent.

The total internal leakage in a normal pump runs about 10 percent.* This comes from the leakage through the annuluses just described, from

*Meeting with Messrs. Hightower, Peavler, et al, July 19, 1967.

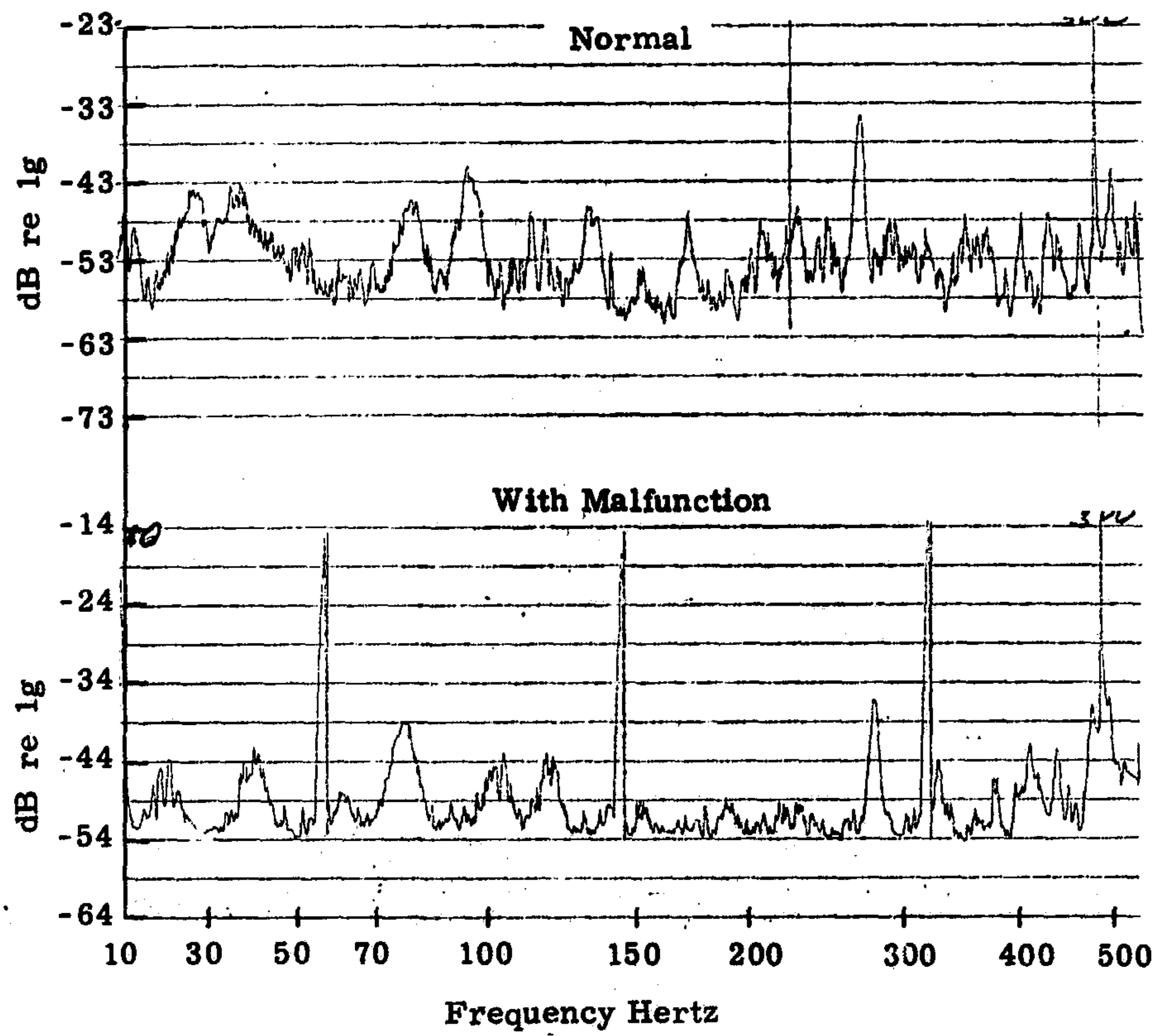


Figure 59. Servo Pump Spectra (0 Stroke, 0 psi)

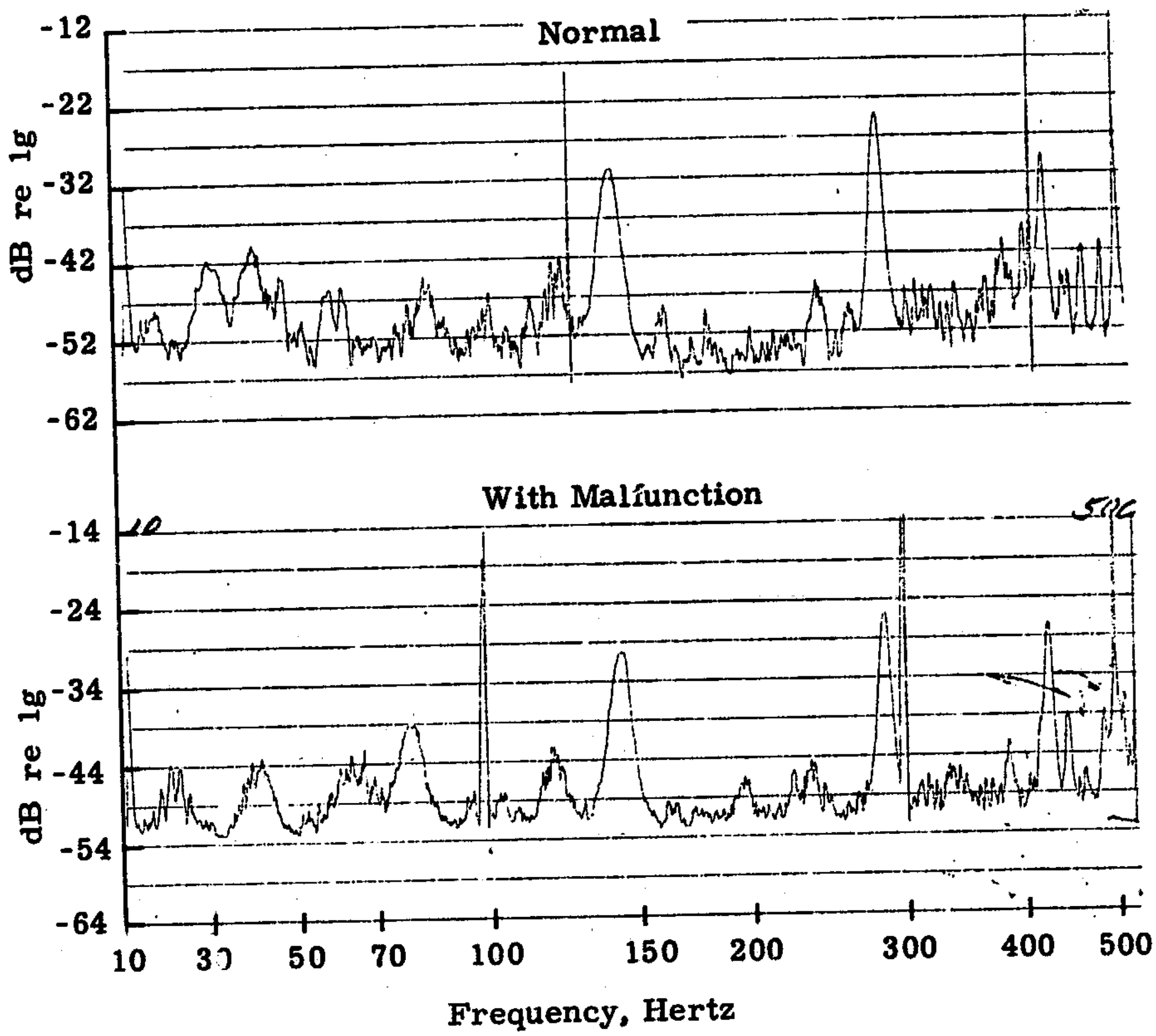


Figure 60. Servo Pump Spectra (50% Stroke, 400 psi)

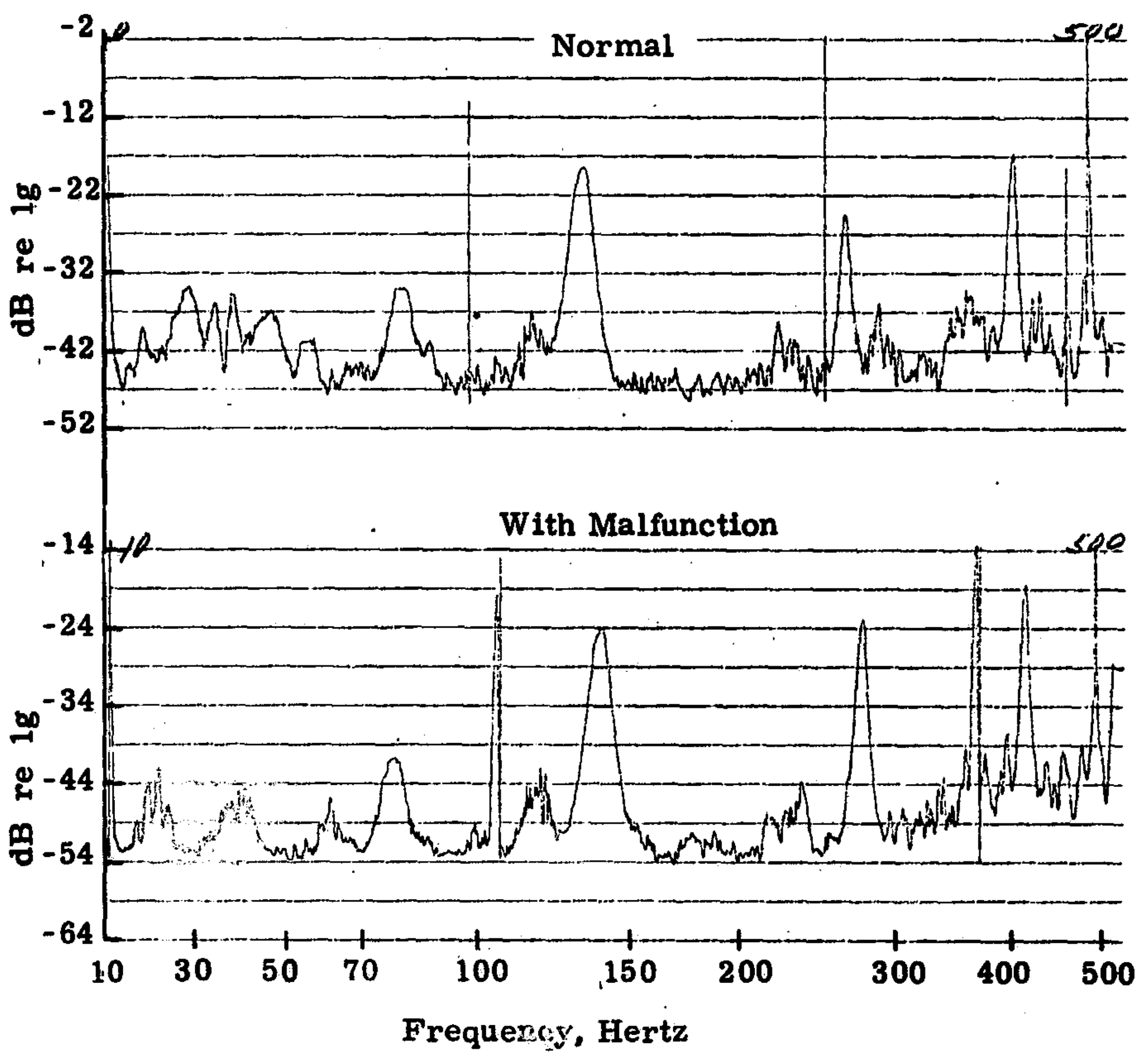


Figure 61. Servo Pump Spectra (50% Stroke, 1100 psi)

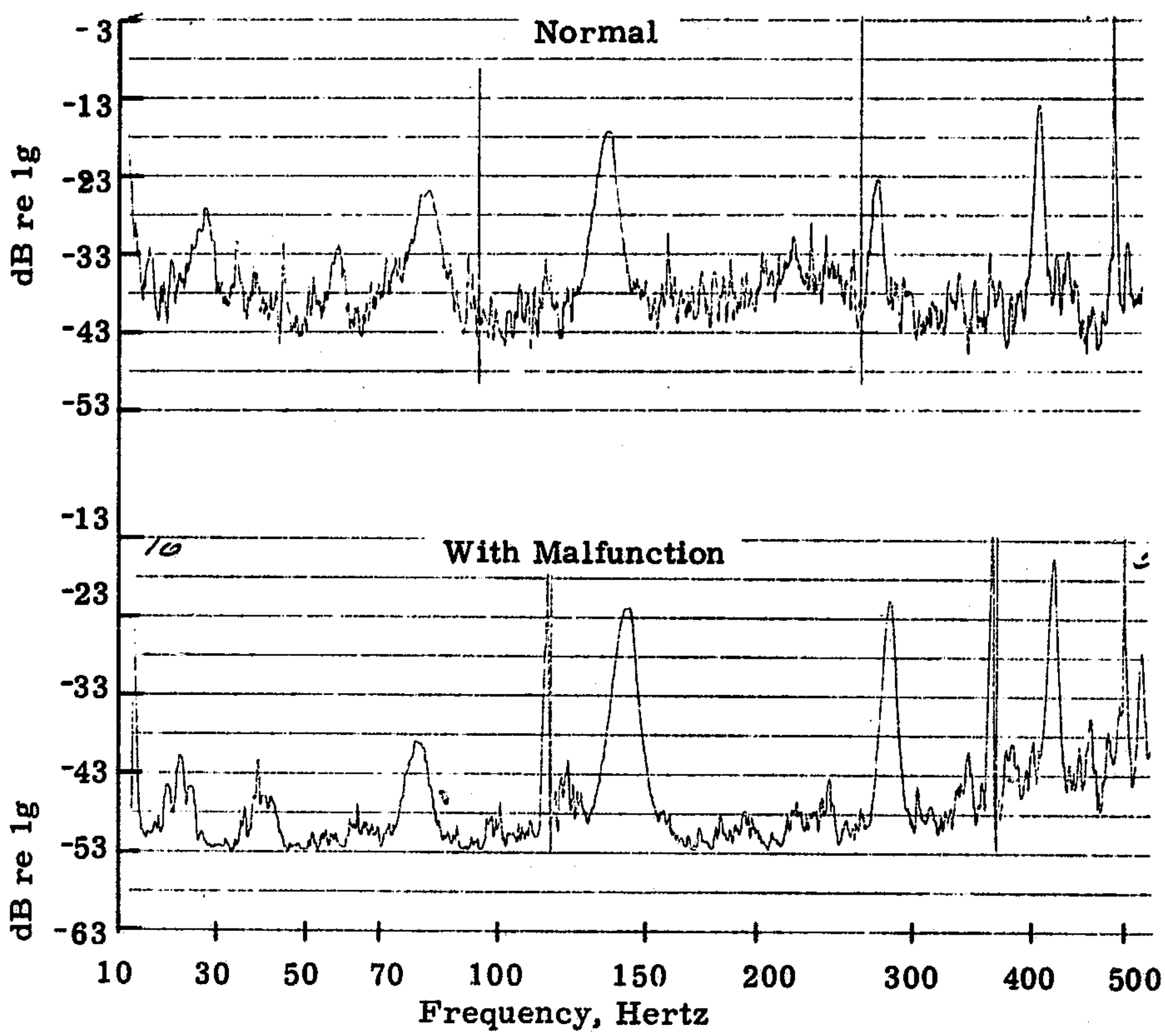


Figure 62. Servo Pump Spectra (50% Stroke, 1400 psi)

capillaries in each piston which feed oil to the ball-and-socket joints shown in Figure 1, and from leakage between the rotating assembly and the port plate. The leakage introduced represented a significant percentage change in clearance, but its effect on the total performance was quite negligible.

A test was run in which the servo pump was starved for replenishing oil in order to induce cavitation. The pump was first run normally at 50 percent stroke and 500 psi pressure. The pressure in the replenishing line was then dropped as far as possible (about 30 psi) by means of the pressure regulator shown in Figure 32. This was not sufficient to induce cavitation; therefore, the valve in the replenishing line was partially closed to cause cavitation along with a corresponding drop in discharge pressure. Two runs were made, with discharge pressures of 400 and 450 psi.

Figure 63 shows photographs of the raw waveforms measured at these two pressures. Peak accelerations were +10 and +16 decibels relative to one g at 450 and 400 psi discharge pressures, respectively. Results of summation analyses using rectified signals are shown in Figure 64 for the accelerometer location over the thrust bearings. Such an analysis captures the repetitive portions of the waveform envelope.

Cavitation occurs when the pressure at the pump inlet falls below the vapor pressure. If cavitation is fully developed this condition will be present throughout the cycle; when cavitation is less than fully developed it occurs only at those instants during the cycle when the proper conditions are met. As would be expected, cavitation noise was first experienced at the instant that each of the seven pistons achieved its maximum velocity while ported to the inlet, causing seven bursts of noise per revolution. The signature was clearest at the transducer location closest to ports. When the discharge pressure dropped to 400 psi the duration of the bursts of cavitation was longer than those at 450 psi, and the amplitude was greater. In either case, the envelope of the waveform shows an order-of-magnitude increase relative to that experienced under normal conditions.

Figure 65 shows waveforms photographed from tape recordings made on the X-axis system and the Y-axis system at NTTF on July 21, 1967, when preliminary measurements were made in order to find the approximate character of signals from the system. The recordings were made on the servo pump at antenna velocities of four degrees per second, east-to-west, west-to-east, and south-to-north. The peak accelerations were, respectively +24 decibels, +18 decibels, and +20 decibels relative to one g. Each photograph includes a time period of 50 milliseconds, corresponding to one full rotation of the pump. The similarity between these waveforms and those in Figure 63 is striking. It appears very likely that the pumps were cavitating at the time that the recordings were made. The energy bursts seen in the photographs were not evident at antenna velocities of one, two, or three degrees per second.

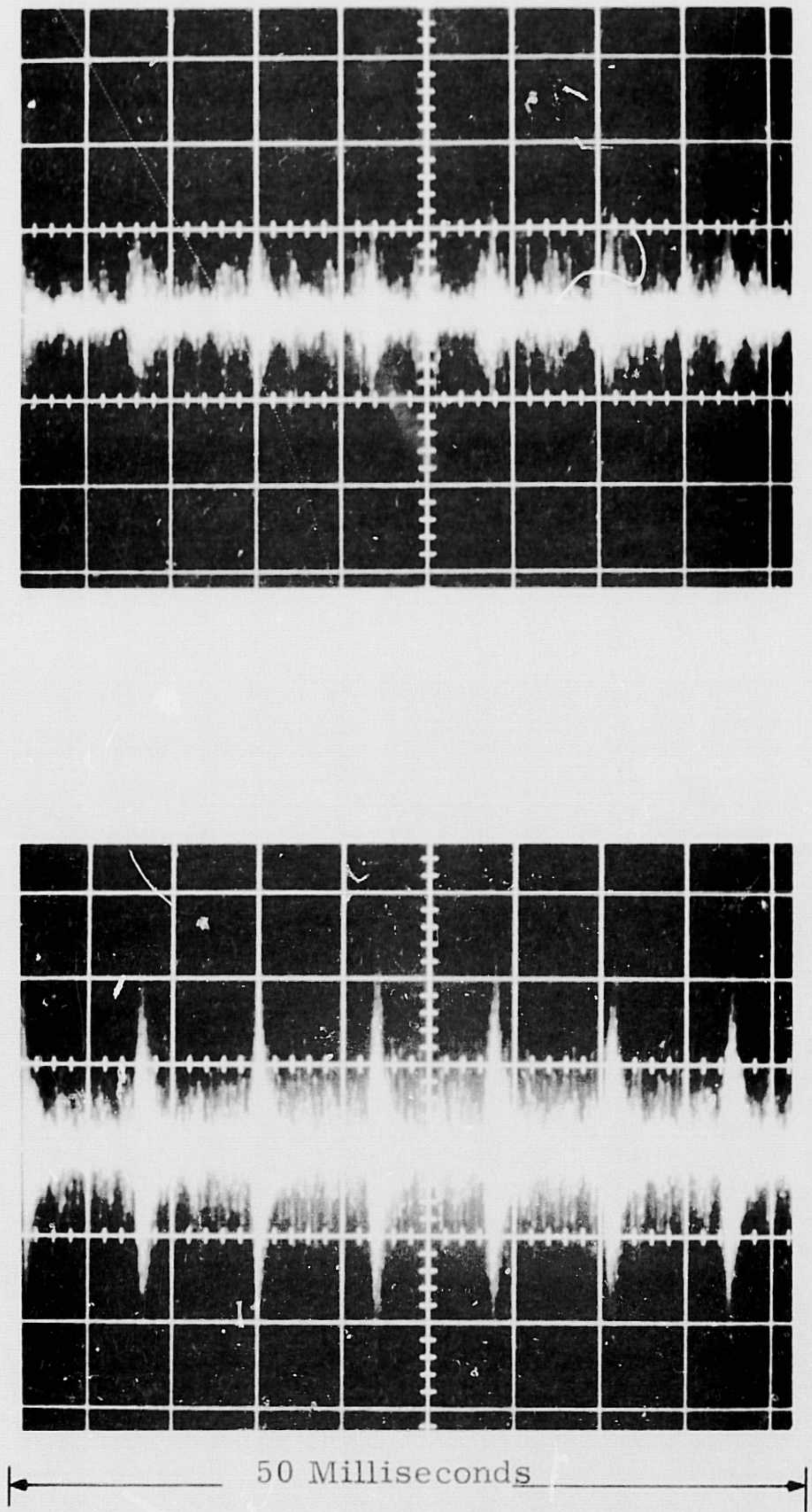
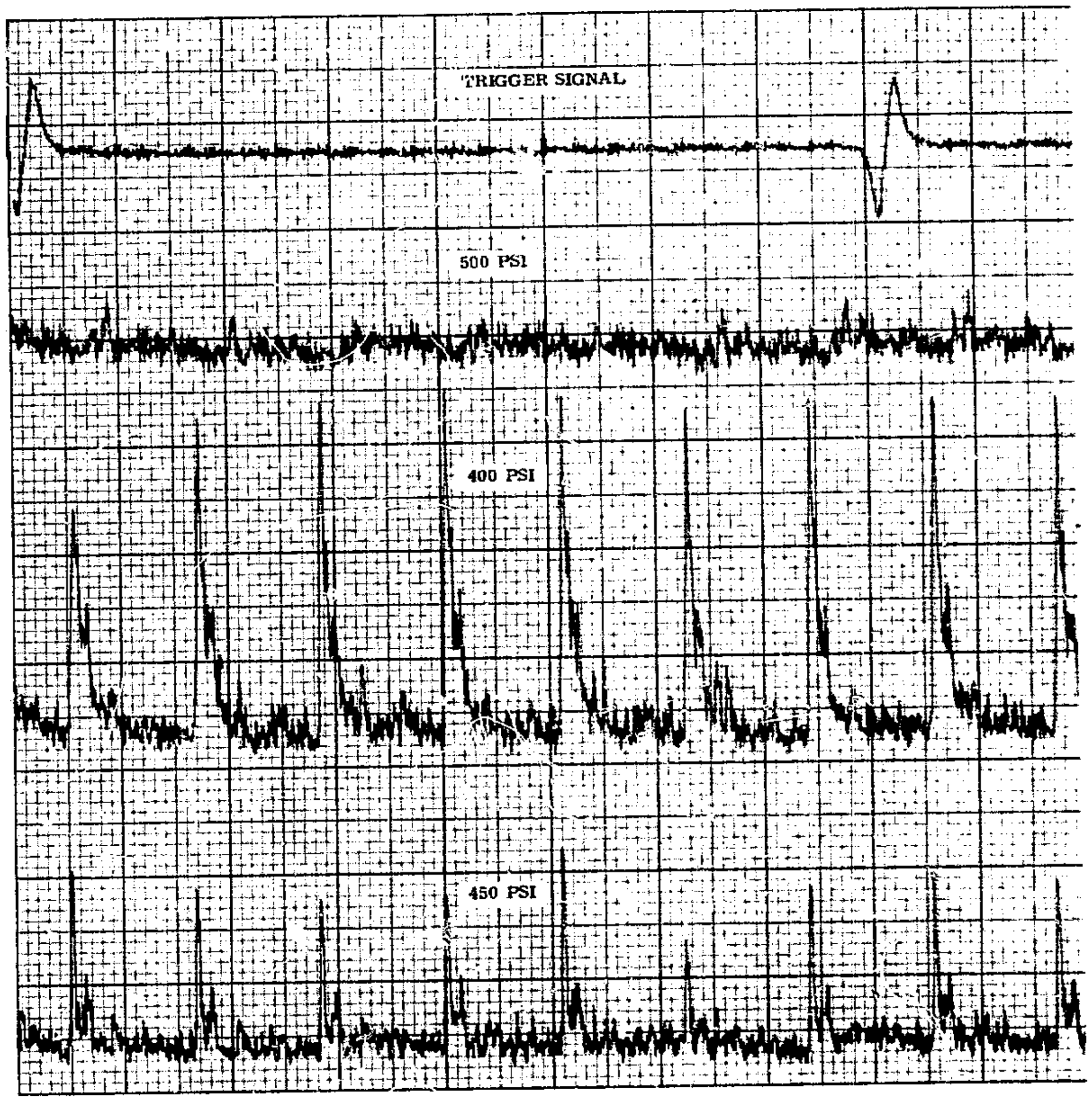


Figure 63. Cavitation Waveforms Measured in Laboratory Tests



TEST LOCATION

Laboratory Test Facility	X
Network Test and Training Facility	

SERVO PUMP OPERATING PARAMETERS

Stroke, % : 50
Pressure, PSI: See Text

SIGNAL PRECONDITIONING

No Filter	X
1 kHz LP Filter	
20 kHz HP Filter	
Unrectified	
Full Wave Rectified	X

ANALYSIS INSTRUMENTATION

GE - R&DC	X
NASA - GSFC	

TRIGGER SIGNAL

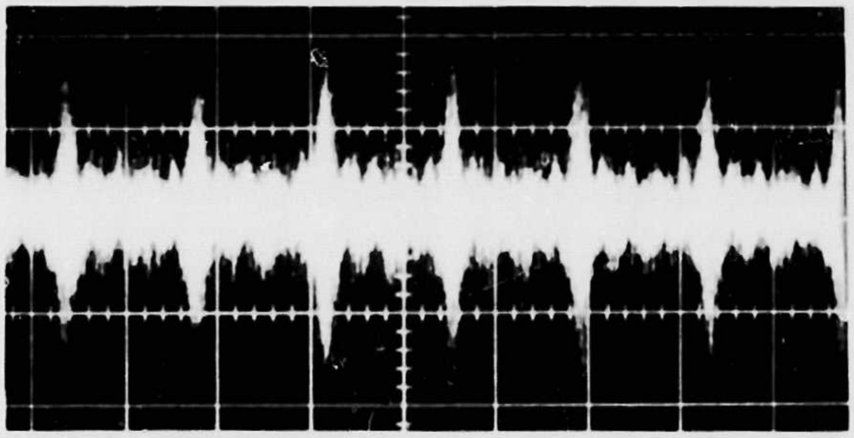
Shaft Position Sensor	X
Pulse Generator	
Switch	

Period, msec Real Time:

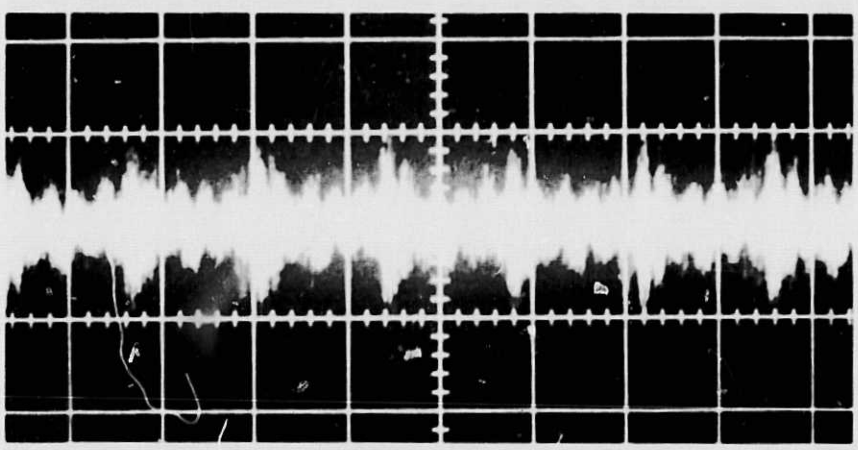
SUMMATION PARAMETERS

Number of Summations: 250
Sweep Period, msec Real Time: 62.5

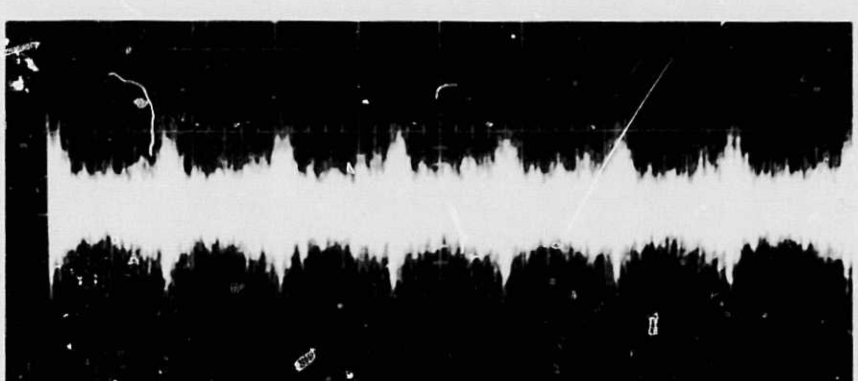
Figure 64. Summation Analyses, Normal Servo Pump With Cavitation



X Axis
4 Degrees per Second
West



X Axis
4 Degrees per Second
East



Y Axis
4 Degrees per Second
North

50 Miliseconds

Figure 65. Servo Pump Waveforms, at the Network Test and Training Facility, Goddard Space Flight Center

Figure 66 shows the result of summing rectified signals from the servo pump vibration, using signals recorded on August 21, 1968. This pump was not either of those represented in Figure 65; it was the pump with introduced malfunctions which was tested in the laboratory. The X-axis system was the one in use. The trace recorded in an antenna velocity of three degrees per second shows nothing unusual. The other, at four degrees per second, again shows the seven noise bursts per revolution characteristic of cavitation. It may be compared to Figure 64, which was recorded with cavitation in the laboratory.

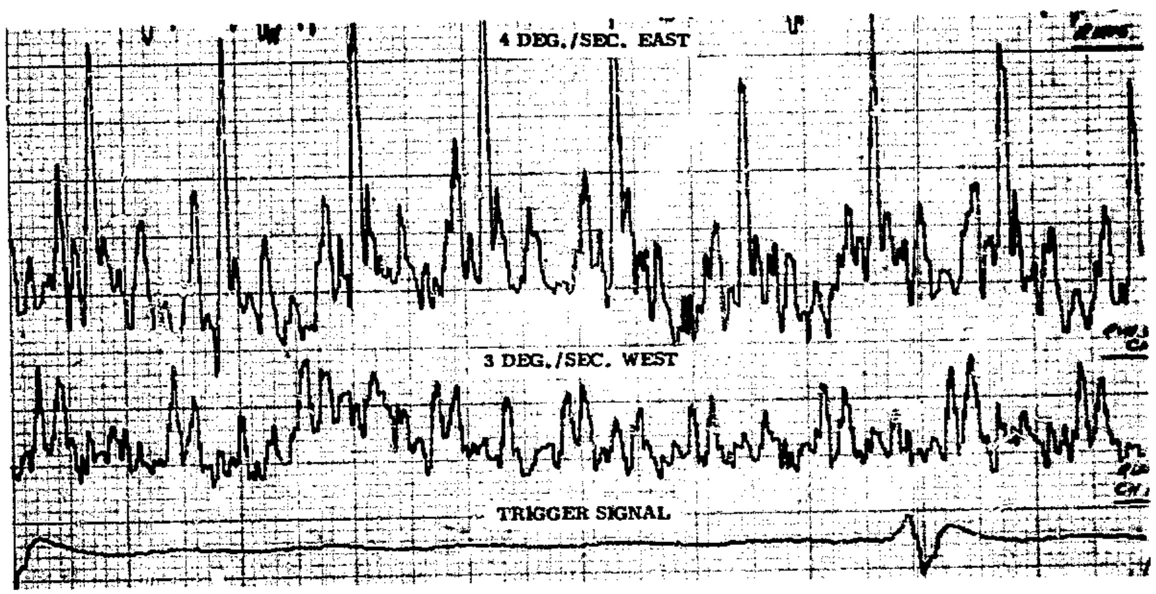
The results given include tests with a total of three different pumps on two different systems. All the measurements indicate that the servo pump on the antenna will cavitate when the antenna is run at a velocity of four degrees per second, but that operation at lesser velocities does not cause cavitation. The consistency of these results suggests a characteristic inherent in the system design, and not a function of a particular set of hardware. This effect was not found in laboratory tests. Since cavitation can cause erosion of pump parts, it would be a conservative measure to limit antenna operation to velocities of three degress per second or less unless absolutely necessary.

DUAL-VANE PUMP

The dual-vane pump has two sections: control and replenishing. Under normal conditions the signals present on both are virtually identical in character and amplitude.

Figure 67 shows one of these signals as recorded directly from the oscilloscope without any processing. The shaft-position trigger signal is also shown. The normal waveform is seen to be a succession of evenly spaced bursts of noise. There are twelve of these for each shaft revolution, corresponding to the twelve vanes in each rotor. All bursts are similar in amplitude, about 10 g's peak. Their high amplitude and random character indicate that they are due to localized cavitation at the pump inlet, as predicted earlier.

This signal is relatively "clean", i. e., free from interference from other sources. This is particularly true when the servo pump is operated under conditions of zero or very low stroke. When the servo pump is delivering considerable power its vibration, as measured on the vane pump, will not necessarily be negligible. While summation or transient averaging is not essential to clarify the waveform of a clean signal, it serves a useful function in that it can store the waveform for later analysis. This could include permanent storage by means of an X-Y plot of the information in the memory, or the waveform could be fed to a digital computer which was programmed to recognize certain patterns.



TEST LOCATION

Laboratory Test Facility	☐
Network Test and Training Facility	☒

SERVO PUMP OPERATING PARAMETERS

Stroke, %: } See Text
Pressure, PSI: }

SIGNAL PRECONDITIONING

No Filter	☒
1 kHz LP Filter	☐
20 kHz HP Filter	☐
Unrectified	☐
Full Wave Rectified	☒

ANALYSIS INSTRUMENTATION

GE - R&DC	☐
NASA - GSFC	☒

TRIGGER SIGNAL

Shaft Position Sensor	☒
Pulse Generator	☐
Switch	☐
Period, msec Real Time:	

SUMMATION PARAMETERS

Number of Summations: 100
Sweep Period, msec Real Time: 62.5

Figure 66. Summation Analyses, Servo Pump With Malfunction

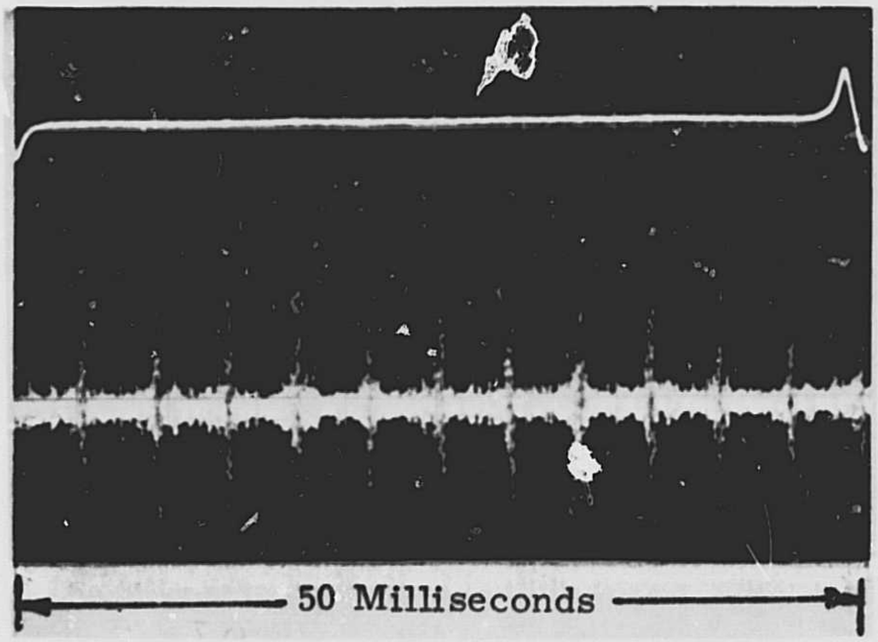
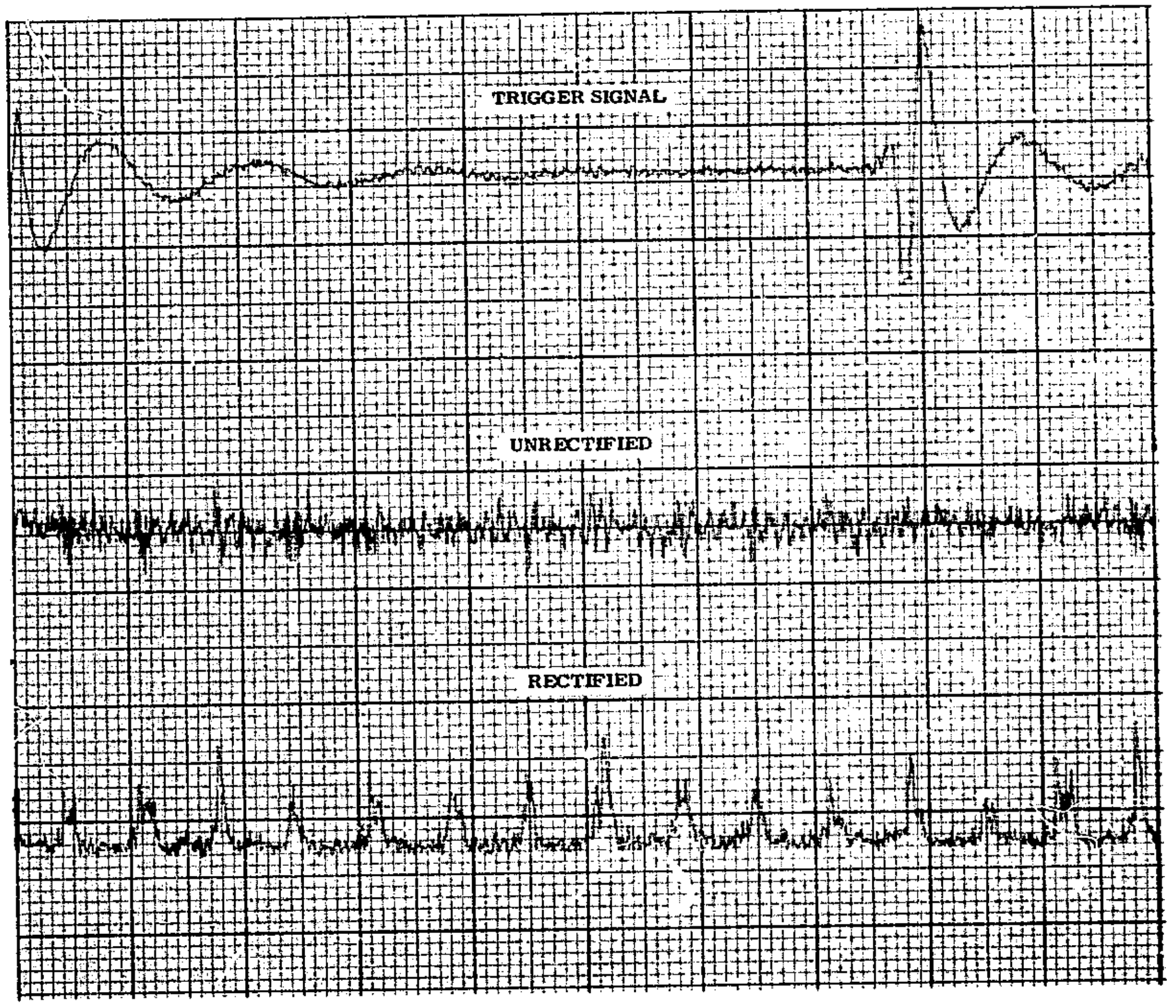


Figure 67. Vane Pump Vibration

Figure 68 is an X-Y plot of the memory of the Enhancetron 1024 after 200 summations of the waveform of a normal pump. The servo pump yoke was centered. The summation of the unrectified signal shows nothing; there were no periodic components of significant size in the vibration. The summed rectified signal captures the envelope of the waveform, clearly showing the twelve bursts of noise. While the bursts are not identical, the amplitudes and characters are similar. The effect of "cross-talk" from the servo pump is shown in Figure 69. It was delivering 1200 psi at half stroke. The summed unrectified signal now shows some periodic components, but close inspection reveals that they are multiples of seven per revolution, and are therefore associated with the piston pump and not the vane pump on which the vibration was measured.

The effect of the introduced defect, damage to four successive vanes of the control pump, can be seen in Figures 70 and 71. The vibrations were measured while running at the Research and Development Center's test facility. All parameters in these two figures were identical except the transducer location and the gain preceding the summation device: the gain used during analysis of the defective pump was twice that used during signal summation of the normal pump. In comparing the two signatures it is not only the amplitude of the waveform which is significant, but also its character. Comparing the lower traces (rectified signals), the normal pump exhibits the usual twelve clean bursts per revolution, while the defective pump has a signature which is very different. The twelve bursts are much reduced in amplitude. Instead of relative quiet between bursts, there is almost continuous noise. The leakage suppressed cavitation by effectively supercharging the pump, while frictional noise increased due to the roughened vanes.



TEST LOCATION

Laboratory Test Facility	X
Network Test and Training Facility	

SERVO PUMP OPERATING PARAMETERS

Stroke, % : 0
Pressure, PSI : 0

SIGNAL PRECONDITIONING

No Filter	X
1 kHz LP Filter	
20 kHz HP Filter	
Unrectified	X
Full Wave Rectified	

ANALYSIS INSTRUMENTATION

GE - R&DC	X
NASA - GSFC	

TRIGGER SIGNAL

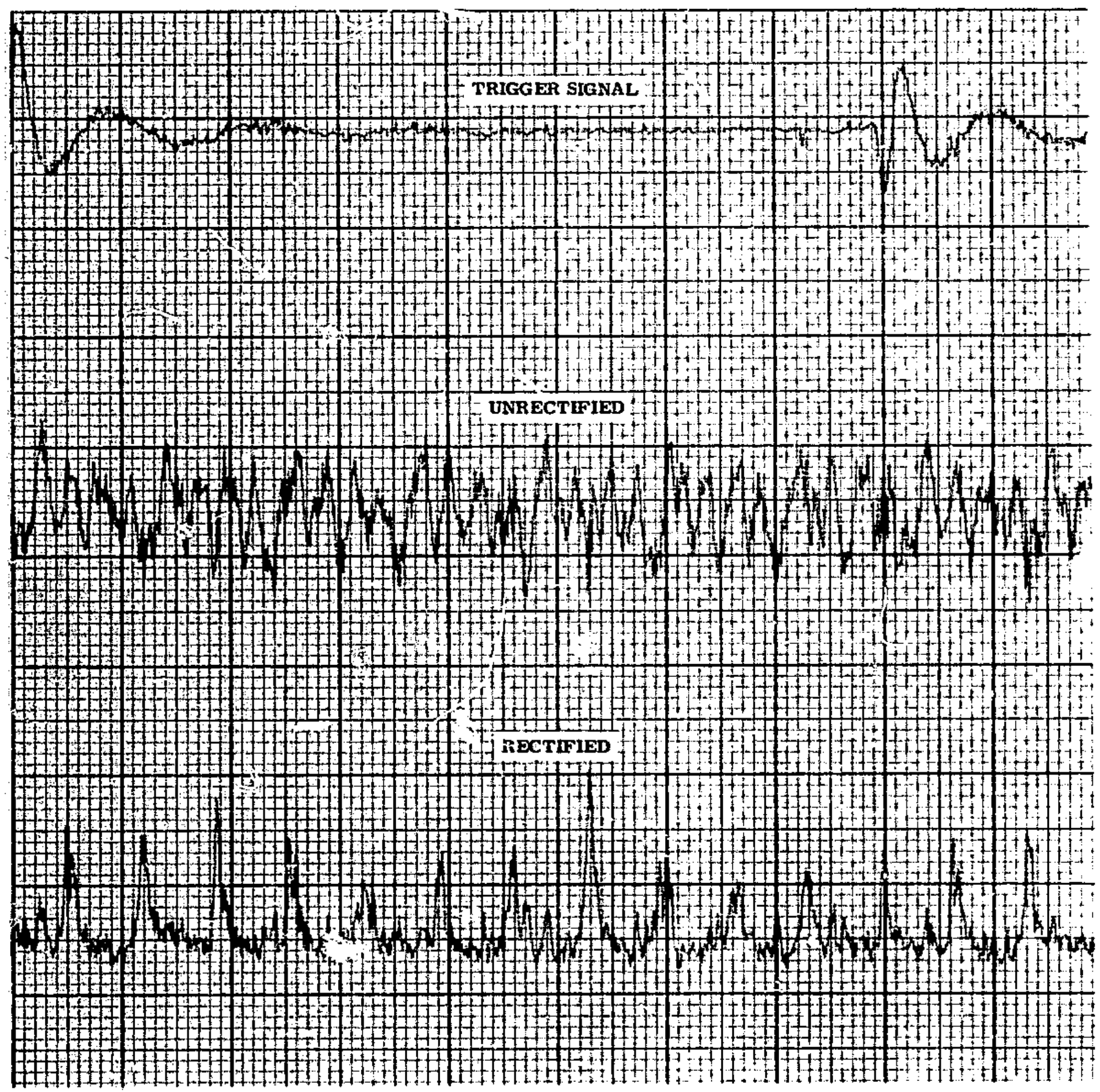
Shaft Position Sensor	X
Pulse Generator	
Switch	

Period, msec Real Time:

SUMMATION PARAMETERS

Number of Summations: 200
Sweep Period, msec Real Time: 62.5

Figure 68. Summation Analyses, Normal Vane Pump



TEST LOCATION

Laboratory Test Facility	X
Network Test and Training Facility	

SERVO PUMP OPERATING PARAMETERS

Stroke, % : 50
Pressure, PSI: 1200

SIGNAL PRECONDITIONING

No Filter	X
1 kHz LP Filter	
20 kHz HP Filter	
Unrectified	X
Full Wave Rectified	X

ANALYSIS INSTRUMENTATION

GE - R&DC	X
NASA - GSFC	

TRIGGER SIGNAL

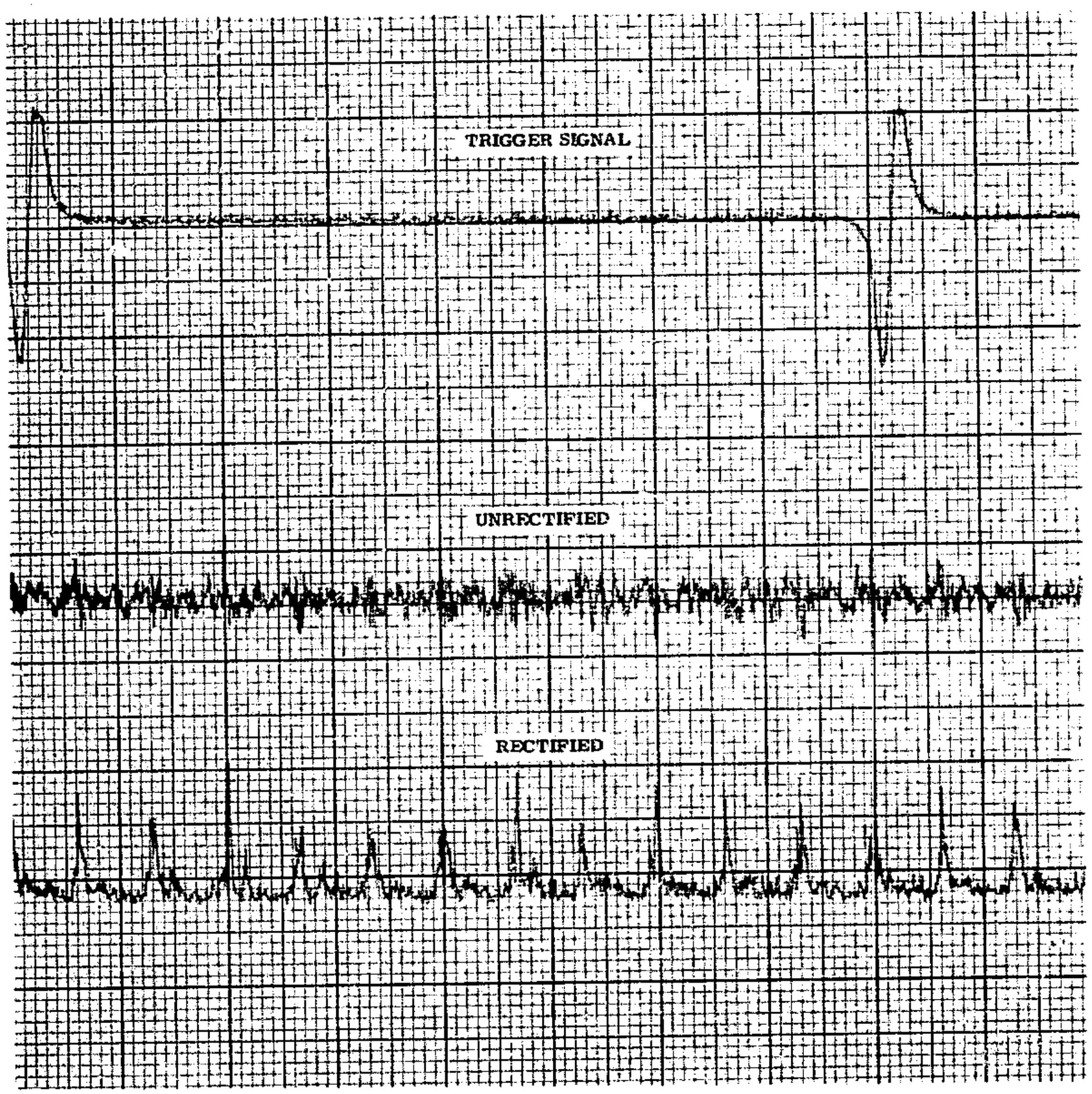
Shaft Position Sensor	X
Pulse Generator	
Switch	

Period, msec Real Time:

SUMMATION PARAMETERS

Number of Summations: 200
Sweep Period, msec Real Time: 62.5

Figure 69. Summation Analyses, Normal Vane Pump



TEST LOCATION

Laboratory Test Facility	X
Network Test and Training Facility	

SERVO PUMP OPERATING PARAMETERS

Stroke, %: 0
Pressure, PSI: 0

SIGNAL PRECONDITIONING

No Filter	X
1 kHz LP Filter	
20 kHz HP Filter	
Unrectified	X
Full Wave Rectified	X

ANALYSIS INSTRUMENTATION

GE - R&DC	X
NASA - GSFC	

TRIGGER SIGNAL

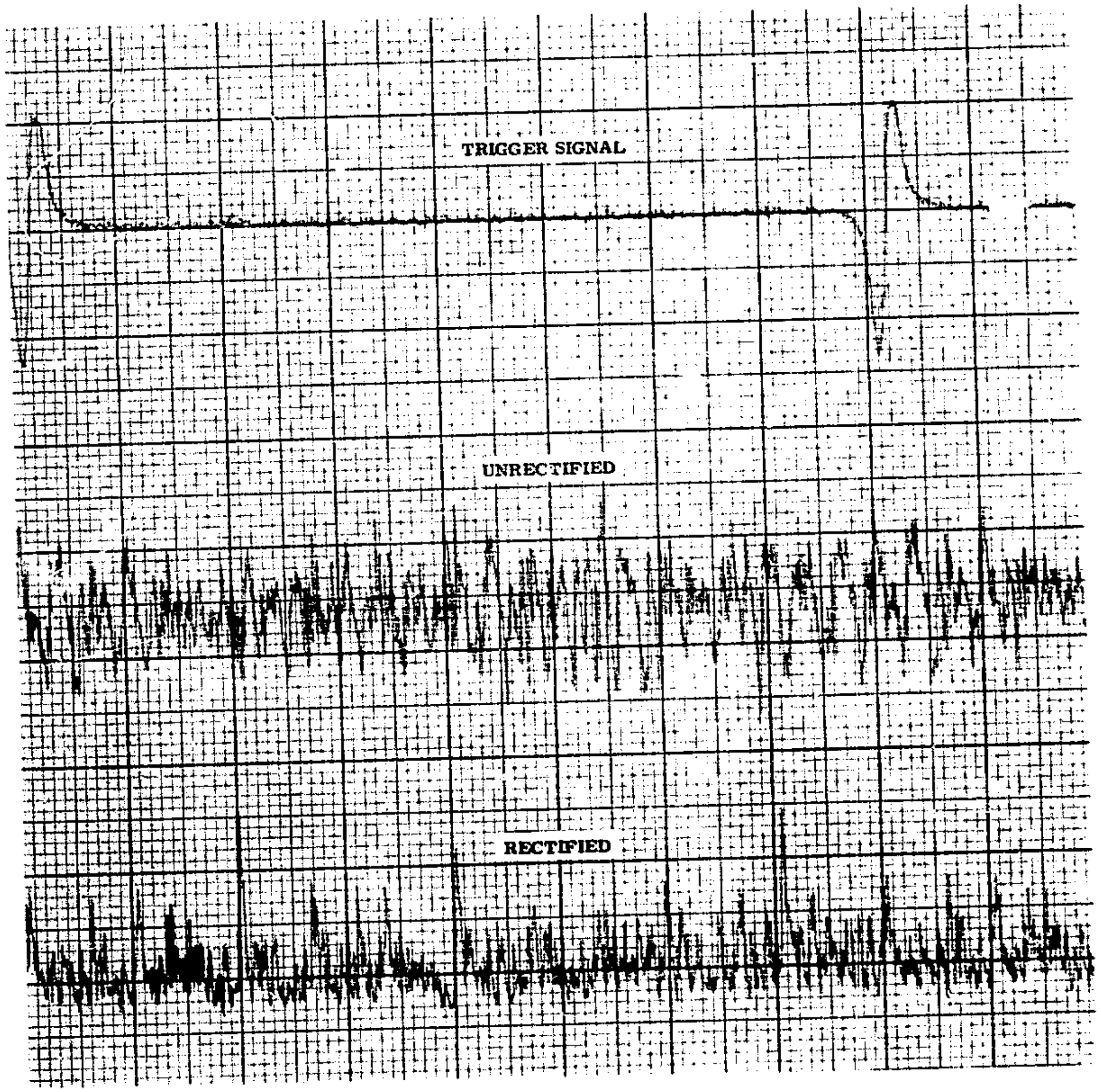
Shaft Position Sensor	X
Pulse Generator	
Switch	

Period, msec Real Time:

SUMMATION PARAMETERS

Number of Summations: 200
Sweep Period, msec Real Time: 62.5

Figure 70. Summation Analyses, Normal Replenishing Pump (Control pump had leaky vanes.)



TEST LOCATION

Laboratory Test Facility	X
Network Test and Training Facility	

SERVO PUMP OPERATING PARAMETERS

Stroke, % : 0
Pressure, PSI: 0

SIGNAL PRECONDITIONING

No Filter	X
1 kHz LP Filter	
20 kHz HP Filter	
Unrectified	X
Full Wave Rectified	X

ANALYSIS INSTRUMENTATION

GE - R&DC	X
NASA - GSFC	

TRIGGER SIGNAL

Shaft Position Sensor	X
Pulse Generator	
Switch	

Period, msec Real Time:

SUMMATION PARAMETERS

Number of Summations: 200
Sweep Period, msec Real Time: 62.5

Figure 71. Summation Analyses, Control Pump With Leaky Vanes

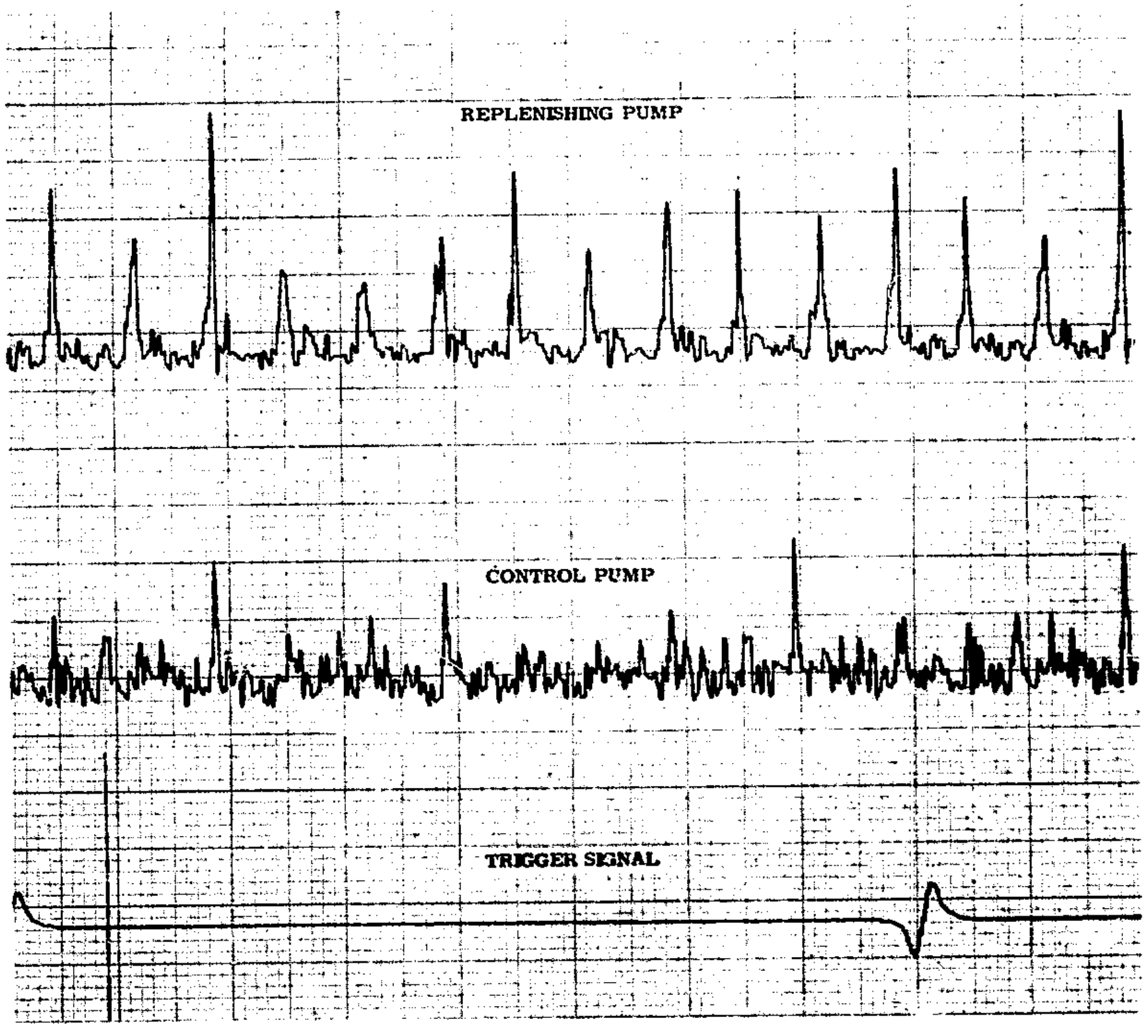
The identical magnetic tape which was used to generate Figures 70 and 71 was reanalyzed at Goddard using the Goddard analysis instrumentation previously described. There were again 200 summations, but this time the gain was the same for each analysis (Figure 72). The waveforms produced almost exactly duplicate the earlier results using the R&DC analysis system.

The same signals from the vane pump were re-recorded and reanalyzed using Goddard equipment after the pump was installed on the NTTF antenna (Figure 73). There are differences in detail between these analyses and those made from the earlier tape, but the character of the signals is not changed and the identity of the defective pump is very clear.

Spectrum analyses were made of the signals from the normal replenishing pump and the defective control pump (Figure 74) using the Radiometer analyzer with two hertz bandwidth. The replenishing pump vibration is mainly broad-band noise having a spectrum level (energy in a one-hertz band) of about -48 decibels relative to one g. In the defective pump the level of the broad-band noise is much lower; this is in keeping with the envelopes of the waveforms presented earlier. Various spectral components are also evident, at or near 40, 60, 80, 100, 120, 140, and 280 hertz. The components at 140 and 280 hertz are obviously related to the servo pump. The 80-hertz signal is actually not synchronous with shaft rotation as described in the discussion of servo pump spectra. It probably arises from a bearing, since bearings are capable of generating such nonsynchronous signals. The vibration at 40 hertz seems to have two components spaced very close in frequency. The larger is nonsynchronous, while the smaller is tied to running speed.

The computer analysis of a pump with one defective vane indicated that new vibrations would be generated at 80 hertz and its harmonics, since high and low pressure regions were allowed to communicate via the leak four times per revolution. With four successive leaky vanes other patterns are established, and 40 hertz and its harmonics may also be expected. This is in fair agreement with the measurements, but detection of a bad pump from waveform analysis is easier, as previously discussed.

Another example of the ability of waveform analysis to detect malfunctions is found in the following example. Goddard personnel, through routine analysis of hydraulic oil samples drawn from units in the field, detected metal particles suggesting deterioration of a vane pump. When the pump was inspected at Goddard's request, severe "washboarding" of the ring of the control portion of the pump was found, as seen in Figure 18. The roughness appeared at the point visible in the photograph and also in the area diametrically opposed, which is hidden. As a result, the vanes had to reciprocate in an abnormal way when passing these areas. Moments were also applied to the rotor or vane block in such a way as to cause it to eventually split. The damaged ring, the rotor, and the twelve vanes of the



TEST LOCATION

Laboratory Test Facility	X
Network Test and Training Facility	

SERVO PUMP OPERATING PARAMETERS

Stroke, % : 0
Pressure, PSI: 0

SIGNAL PRECONDITIONING

No Filter	X
1 kHz LP Filter	
20 kHz HP Filter	
Unrectified	
Full Wave Rectified	X

ANALYSIS INSTRUMENTATION

GE - R&DC	
NASA - GSFC	X

TRIGGER SIGNAL

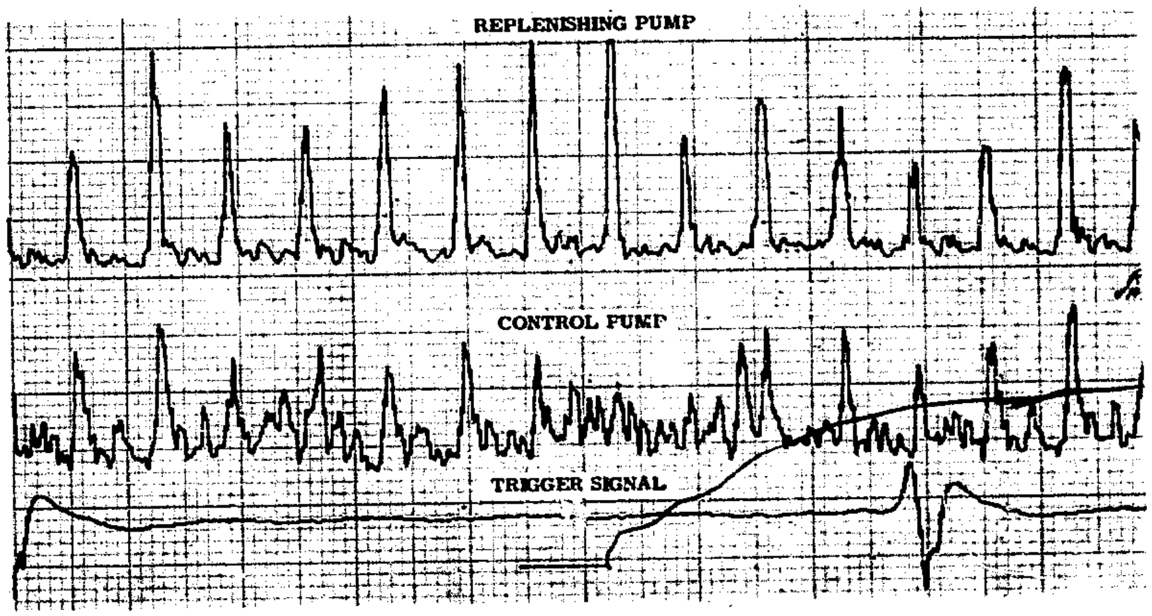
Shaft Position Sensor	X
Pulse Generator	
Switch	

Period, msec Real Time:

SUMMATION PARAMETERS

Number of Summations: 200
Sweep Period, msec Real Time: 62.5

Figure 72. Summation Analyses, Control Pump (Leaky Vanes) and Replenishing Pump (Normal)



TEST LOCATION

Laboratory Test Facility	☐
Network Test and Training Facility	☒

SERVO PUMP OPERATING PARAMETERS

Stroke, %: 0
Pressure, PSI: 0

SIGNAL PRECONDITIONING

No Filter	☒
1 kHz LP Filter	☐
20 kHz HP Filter	☐
Unrectified	☐
Full Wave Rectified	☒

ANALYSIS INSTRUMENTATION

GE - R&DC	☐
NASA - GSFC	☒

TRIGGER SIGNAL

Shaft Position Sensor	☒
Pulse Generator	☐
Switch	☐

Period, msec Real Time:

SUMMATION PARAMETERS

Number of Summations: 200
Sweep Period, msec Real Time: 62.5

Figure 73. Summation Analyses, Control Pump (Leaky Vanes) and Replenishing Pump (Normal)

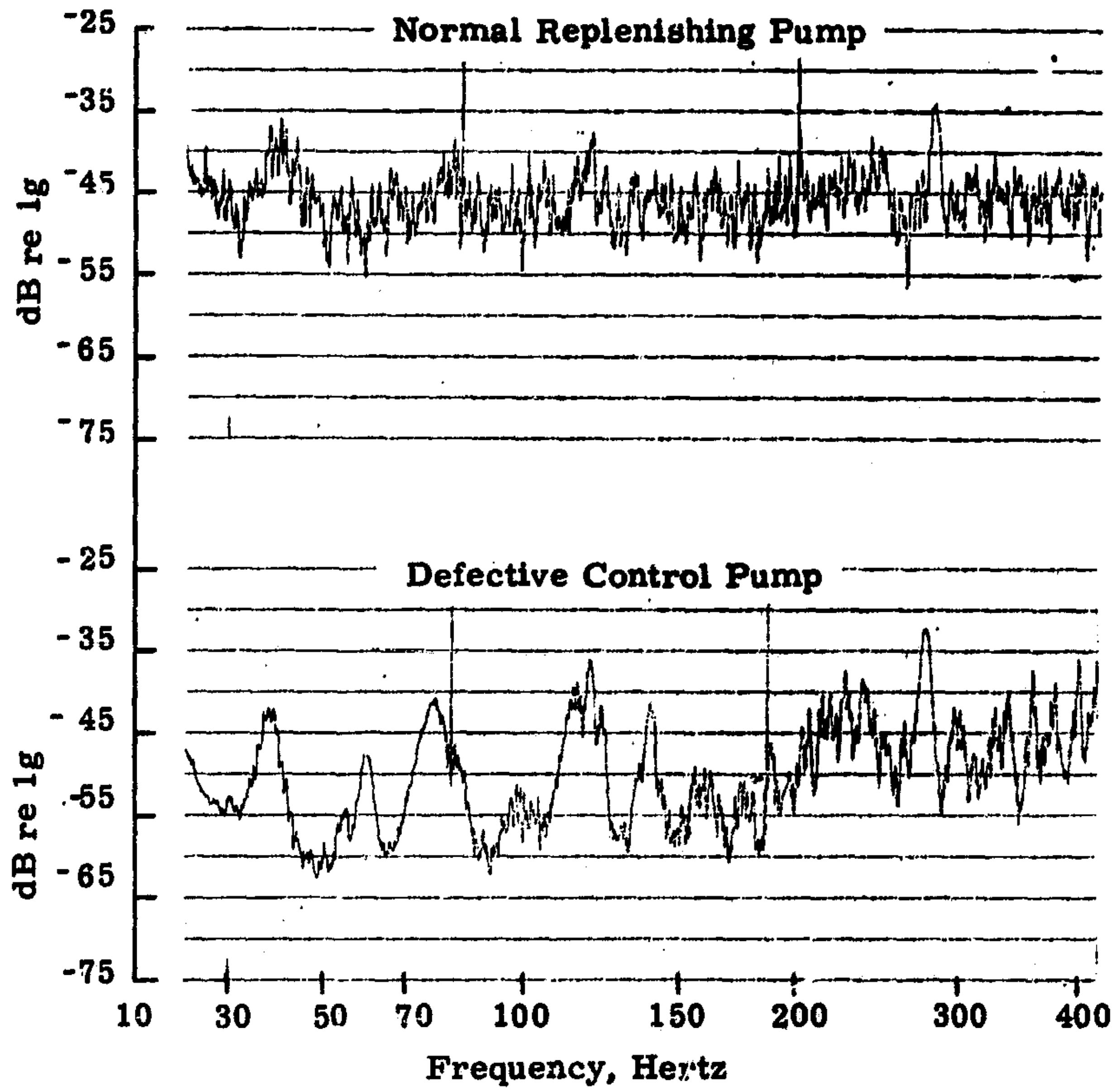


Figure 74. Vane Pump Spectra

control pump were returned to Goddard, and then sent to the Research and Development Center for test.

The components from the defective pump were installed in place of the normal control pump components then being tested. In order to protect the servo pump from possible damage due to released metallic particles this pump was physically removed from the prime mover and all oil connections to it were closed off. All the control pump flow was returned to the tank via its regulator. Valve 1, in Figure 32, was opened so that the replenishing oil would also have a return path to the tank from its pressure regulator. There was no indication from usual operation of the pump that its condition had been degraded.

The vibration signals from the two portions of the pump were rectified before being summed in order to capture the repetitive features of the waveform envelope. The summation process was triggered by the signal from the shaft position sensor, and 200 sums were made. The net gain in the replenishing pump analysis was five times as great as the gain used in analyzing the damage control pump. The results are shown in Figures 75 and 76.

The signal from the replenishing pump appears to be normal, as it should be. The curve is perhaps a little "rougher" than usual, some of the control pump vibrations having been picked up by the sensor on the replenishing pump. The vibration signal measured on the control pump shows many abnormalities. The bursts of noise are about ten times larger than normal, and they differ widely in amplitude. The period between major bursts, which would normally be quiet, shows considerable noise. As a result, the abnormal section of the pump is easily identifiable. Its signature departs so much from that which is normal that application of signature analysis techniques could quite certainly have been successful in detecting the malfunction at a much earlier date than it was achieved by oil analysis.

SOLENOID VALVE

The opening time of the solenoid valve was first measured under normal conditions of pressure and flow without the use of an accumulator to simulate air in the lines. The test was run many times; Figure 77 shows representative traces from the oscilloscope. (A two-kilohertz high-pass filter was used.) The transient at the left has proved to be electrical in origin. It is a switching transient which generally comes about two milliseconds after the sweep is initiated, but in some tests is absent. It has been theorized that it is due to arcing at the switch contacts as the magnetic field of the solenoid coil collapses. Its occasional absence may be caused by the switch being opened (by chance) at a propitious time in the a-c line cycle.

The second transient, some 15 milliseconds after start of the sweep, is mechanical in origin and is due to impacting of the moving system on its stop. The timing is controlled by the various forces in the valve -- spring,

TEST LOCATION

Laboratory Test Facility	X
Network Test and Training Facility	

SERVO PUMP OPERATING PARAMETERS

Stroke, % : NA
Pressure, PSI : NA

SIGNAL PRECONDITIONING

No Filter	X
1 kHz LP Filter	
20 kHz HP Filter	
Unrectified	
Full Wave Rectified	X

ANALYSIS INSTRUMENTATION

GE - R&DC	X
NASA - GSFC	

TRIGGER SIGNAL

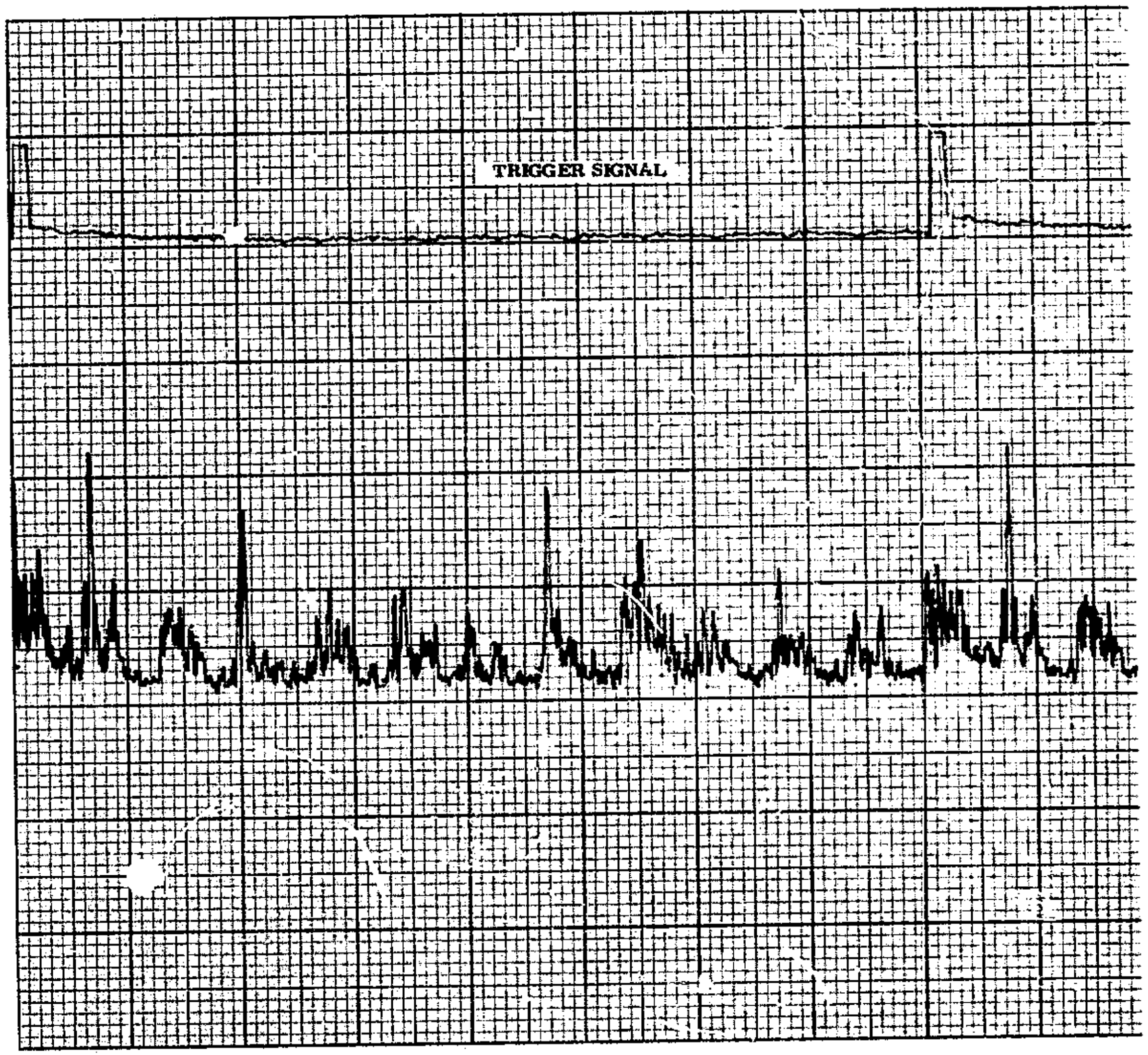
Shaft Position Sensor	X
Pulse Generator	
Switch	

Period, msec Real Time:

SUMMATION PARAMETERS

Number of Summations: 200
Sweep Period, msec Real Time: 62.5

Figure 75. Summation Analysis, Normal Replenishing Pump
(Control pump had defective ring.)



TEST LOCATION

Laboratory Test Facility	X
Network Test and Training Facility	

SERVO PUMP OPERATING PARAMETERS

Stroke, % : NA
Pressure, PSI: NA

SIGNAL PRECONDITIONING

No Filter	X
1 kHz LP Filter	
20 kHz HP Filter	
Unrectified	
Full Wave Rectified	X

ANALYSIS INSTRUMENTATION

GE - R&DC	X
NASA - GSFC	

TRIGGER SIGNAL

Shaft Position Sensor	X
Pulse Generator	
Switch	

Period, msec Real Time:

SUMMATION PARAMETERS

Number of Summations: 200
Sweep Period, msec Real Time: 62.5

Figure 76. Summation Analysis, Control Pump With Defective Ring

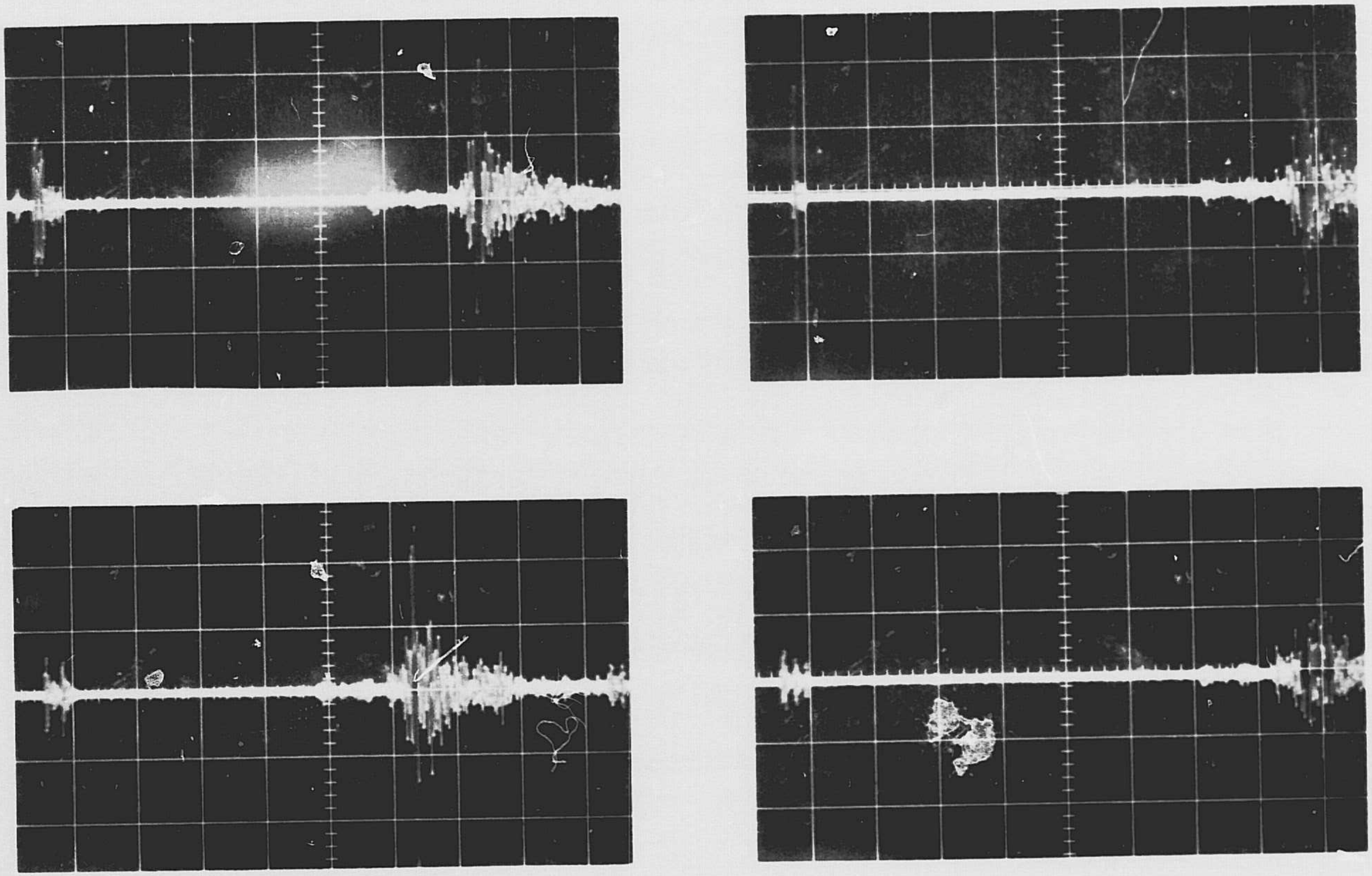


Figure 77. Tests of Normal Solenoid Valve (20 Milliseconds Sweep)

drag, friction, etc. -- and is predicted to be about ten milliseconds. Actually, the time tends to be greater than predicted, and is somewhat variable from run to run. These variations are probably due to changes in friction at the O-ring seals.

After scoring of the spool the valve was retested (Figure 78). Instead of running slower, as might be expected as a result of increased stiction, the speed actually increased so that the valve opened sooner. As before, there was some spread in the data. Figure 79 shows the opening time for

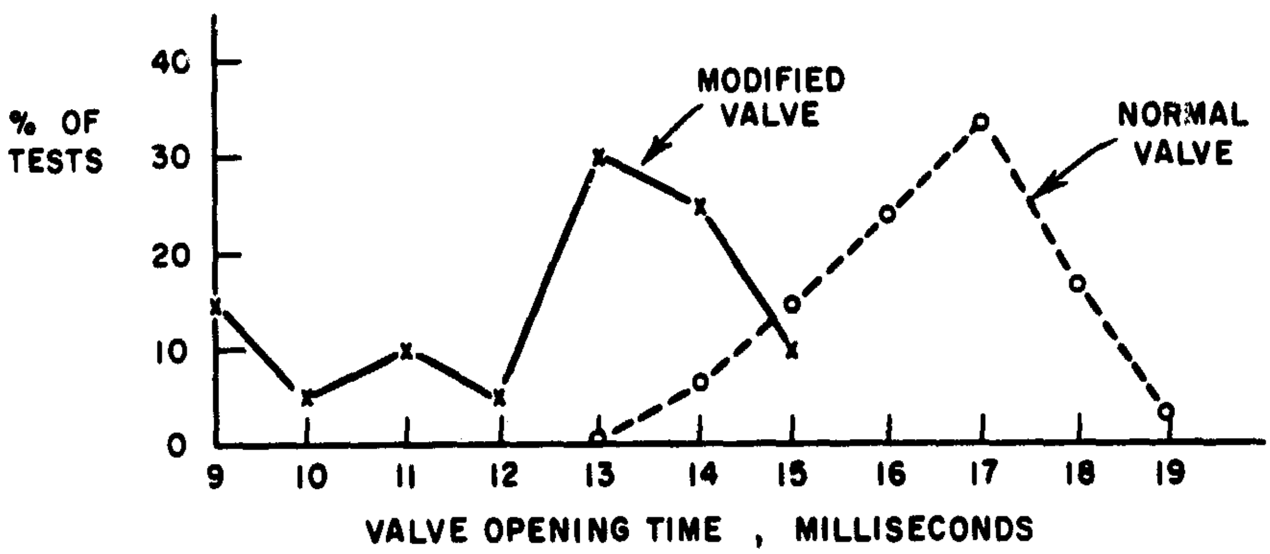


Figure 79. Results of Repeated Tests of Solenoid Valves

some 60 tests, half with a normal valve and half after scoring. The spread of operating times of the scored valve is greater than the spread for the normal valve. It is probable that the O-ring compression was not exactly the same after the valve was reassembled following its modification. This could have the noted effect.

Figure 80 shows two tests of the scored valve run at the NTTF facility. The top trace is the trigger signal; one test in particular shows contact bounce in the switch. The lower trace is the vibration signal. Results are comparable to similar measurements in the laboratory.

It may be concluded that the solenoid valve operates about as expected, but that valve scoring of the degree introduced was not detected, since the effect of such scoring was not as great as that of other variables, particularly friction.

The effect of air in the lines was simulated in the laboratory by adding an accumulator as previously discussed. The calculated operating time under the test conditions was approximately 235 milliseconds. Figure 81 shows the measured response of the normal valve in the laboratory, after

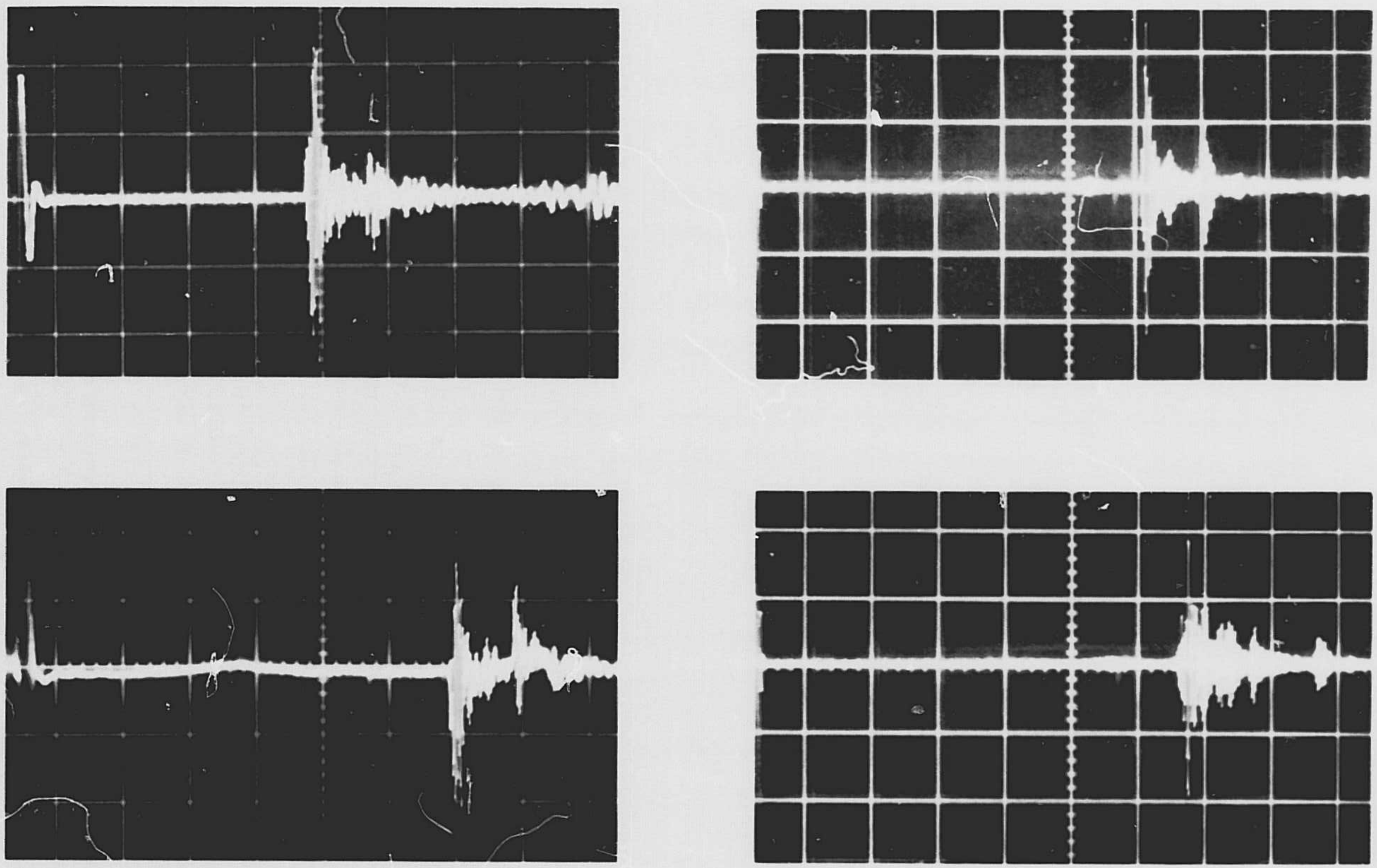
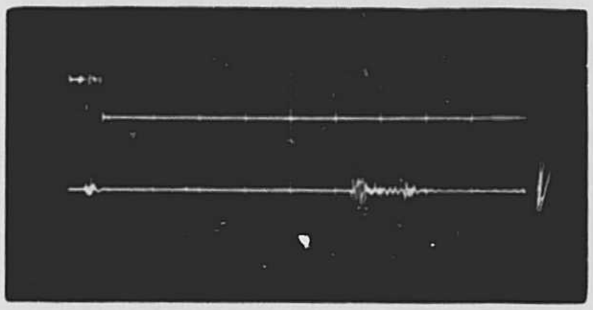
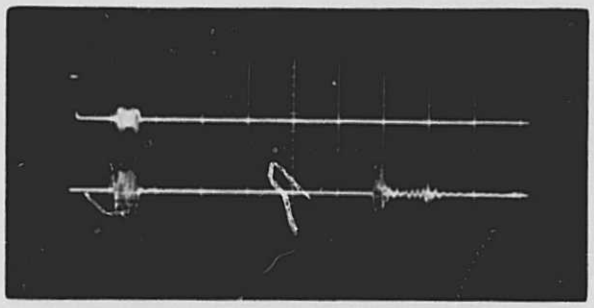


Figure 78. Tests of Solenoid Valve With Defect (20 Milliseconds Sweep)



Top Trace: Electrical Trigger

Bottom Trace: Vibration Signal



|←20 Milliseconds →|

Figure 80. Tests of Solenoid Valve with Defect, at the Network Test and Training Facility, Goddard Space Flight Center

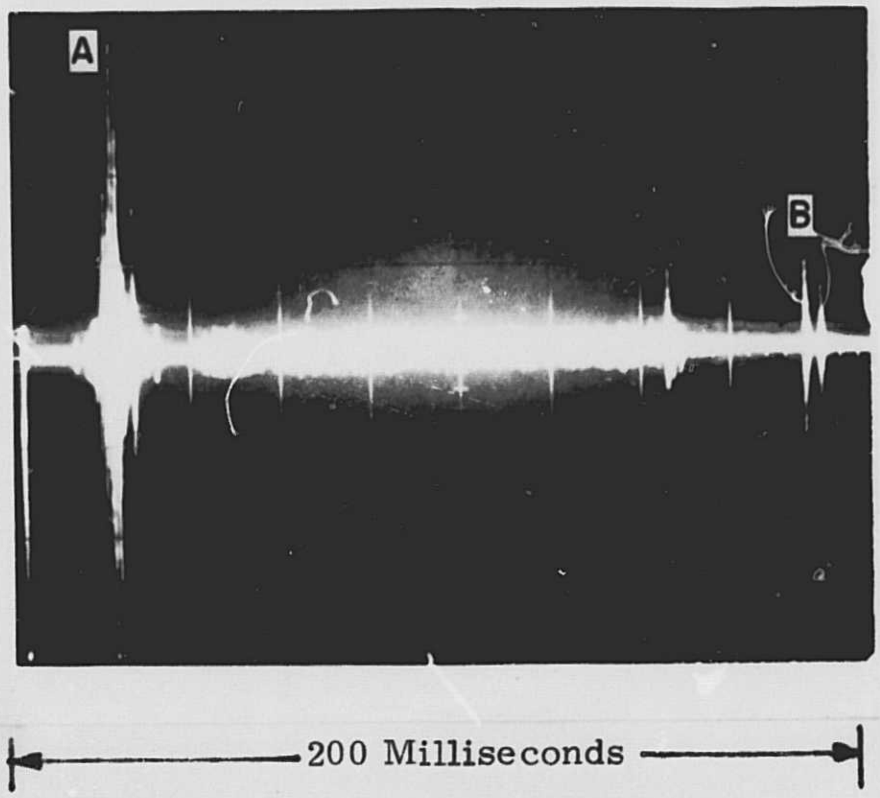


Figure 81. Opening of Normal Solenoid Valve with Accumulator

the signal is passed through a 20-kilohertz high-pass filter. The following sequence of events is represented.

At t = 0 the switch opens. Soon thereafter the magnetic forces drop to zero and the spool begins to move toward the open position, pushing the solenoid armature with it. When the spool is about half-way in its total travel it begins to crack open, permitting flow. At that time large Bernoulli forces are generated by discharge of the energy stored in the accumulator. These forces brake the spool, but the armature (which is not solidly fastened to the spool) continues to move until it impacts on its stop, producing the signal seen at A. The spool momentarily rests at an equilibrium position which depends on the balance of forces. As the accumulator discharges, the flow noise is relatively stable in amplitude. When the accumulator is fully discharged, the spool opens under spring force. Its stop produces transient β after an elapsed time of 175 milliseconds.

Figure 82 shows the equivalent signal from the malfunctioning valve. The filter was set for two kilohertz high-pass. Operating time increased to 220 milliseconds. This is the opposite effect from that noted without the accumulator, but is not unexpected. With the accumulator in the hydraulic circuit the operating time is set mainly by the time required to discharge the accumulator. This is relatively insensitive to friction effects, since the spool is almost stationary during this process. On the other hand, the discharge time is influenced by the discharge coefficient of the valve. Scoring or roughening of the spool could cause a decrease of this coefficient, which would increase the operating time as already noted.

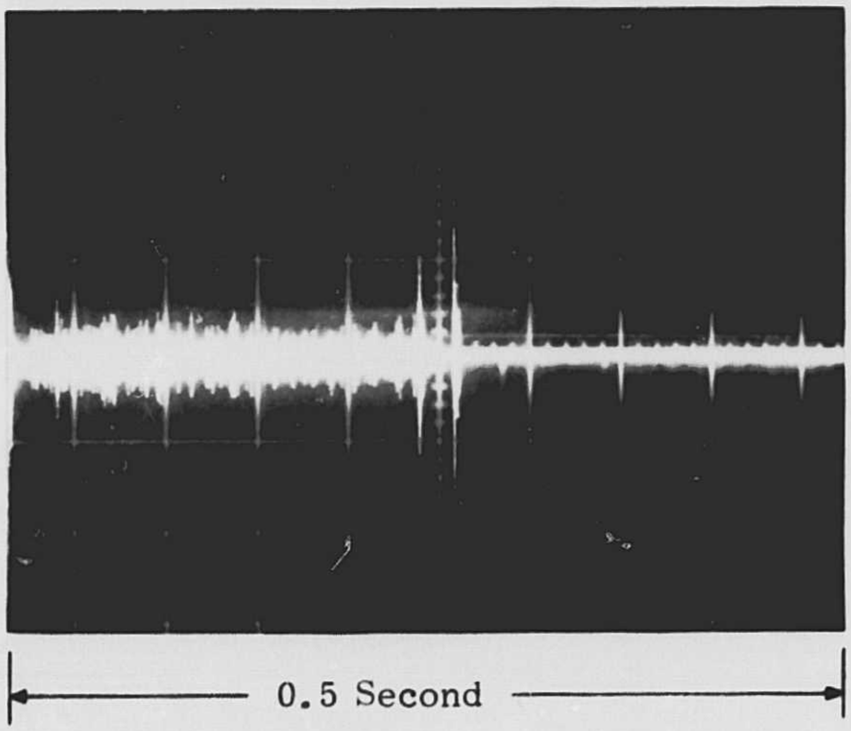


Figure 82. Opening of Defective Solenoid Valve, with Accumulator

Comparison of Figures 81 and 82 with Figure 80 (measured at NTTF), shows that there was negligible air in the lines during the latter measurement. The signal measured was characteristic of that expected without stored energy. The good correlation between the prediction and measured results in the laboratory indicates that the presence of air could be detected if it is present in significant amounts.

SERVO VALVE

The operation of the servo valve is monitored by measuring the variation in flow-noise through the valve as it responds to an electrical input. When a zero net error signal is applied to the coils, the hydraulic preamplifier and a feedback spring on the spool allow the spool to assume a neutral position. An error signal causes the spool to deflect to one side or the other, depending on its sign. This allows the valve to pass flow from its inlet to the appropriate discharge port. As the servo nulls out, the net error is again reduced to zero and the spool recenters, shutting off flow.

Figure 83 shows idealized V-curves for: a) normal valves, and b) a valve with offset or unbalance. In each case, there is some value of error for which flow is minimized (leakage only). In the normal situation this occurs when the error is zero. In a qualitative sense, flow noise will vary with the error signal in much the same way, being of minimum amplitude when flow is minimum and greatest at the high flows, corresponding to large errors.

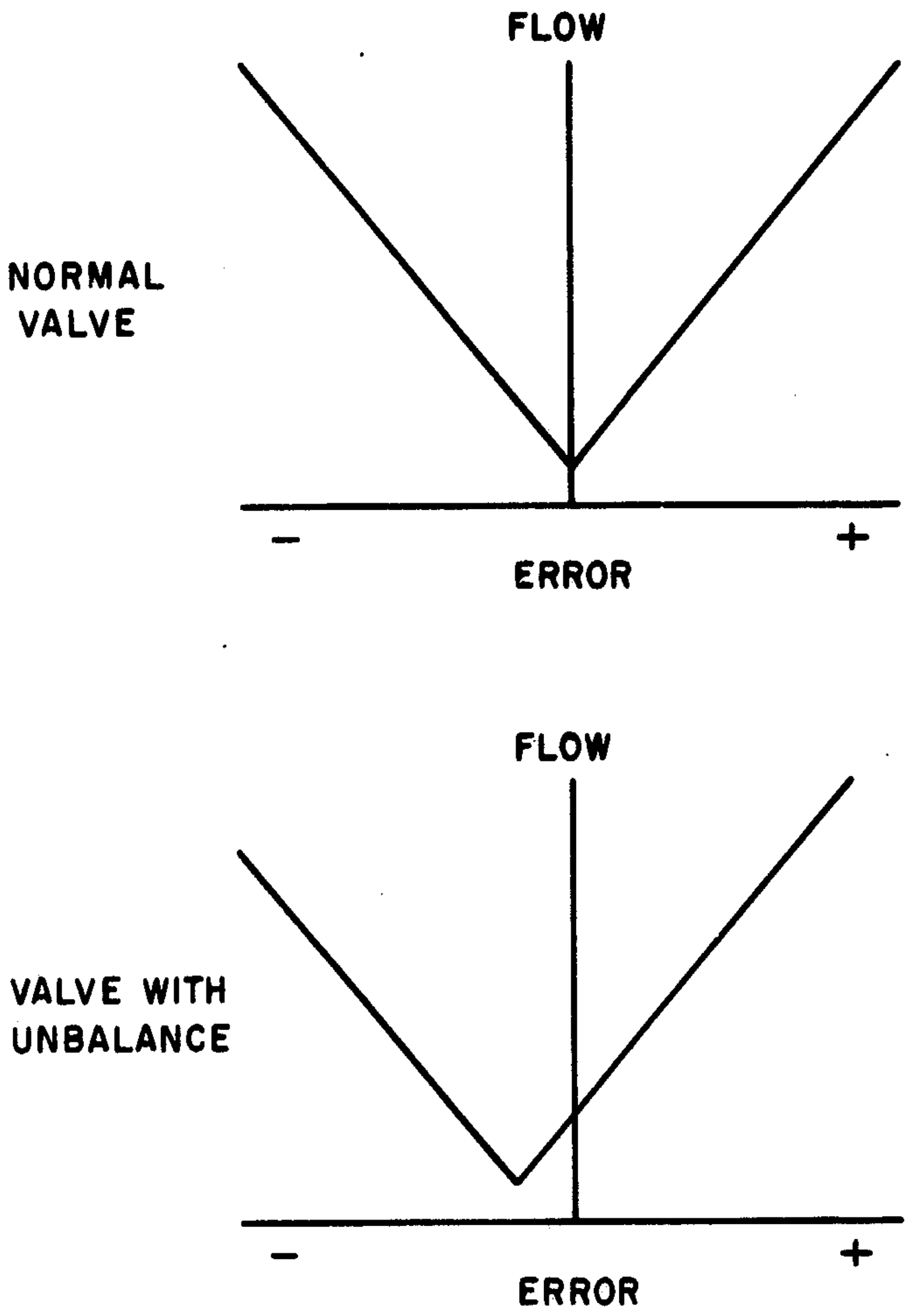


Figure 83. Idealized V-curves of Normal and Unbalanced Servo Valves

As discussed in the mathematical analysis leading to the servo valve response, the servo valve takes a few milliseconds to respond after being excited by a step-function transient. As a consequence, the flow noise through a normal valve is expected to vary with time after a step-function input as shown in Figure 84(a), assuming that the valve is centered before the input occurs. If the valve is unbalanced, the flow noise will not be the same in the two directions corresponding to positive or negative errors. This is sketched in Figures 84 (b and c).

Figure 19 shows actual V-curve measurements on the servo valve which was originally supplied for test. In its original condition it showed unbalance similar to that sketched in Figure 84(b), and was tested on the valve test facility (Figures 29 and 30). For those tests the valve was electrically excited by a battery; there was no servo amplifier and no feedback. As a consequence, an error signal would cause the spool to translate to one side and stay there. Since the flow through the valve was directed to a small accumulator, it would only persist until the accumulator was charged. Once the charging was complete there would be no flow noise except the noise of the jet in the hydraulic preamplifier. Flow noise versus time through the unbalanced valve would be expected to vary on this facility as sketched in Figure 85.

Figure 86 shows the results of measurements on the unbalanced valve whose V-curve is shown in Figure 19. The signals were passed through a 20-kilohertz high-pass filter. The top trace in each photograph is the electrical trigger signal used to initiate the sweep. whose total duration was 50 milliseconds. Correspondence between the predicted and actual signatures of the unbalanced valve is very good, showing that signature analysis techniques can detect this condition. The test outlined is applicable to a test stand but not to testing on the antenna unless the brakes are locked, since it does not permit adequate control of antenna velocity or position.

The characteristic response time of the servo valves was tested in more detail by means of the servo amplifier of Figure 33 and the test facility of Figures 31 and 32. A transient, manually inserted, causes the valve to pass flow in the appropriate direction. As the servo pump yoke responds, it gradually reduces the valve flow back to its minimum value through feedback action via the yoke position potentiometer. Unbalance cannot be detected acoustically since the feedback causes the valve to seek the null point of the V-curve, regardless of whether or not this corresponds to zero net-error voltage. Unbalance could be detected at null from coil voltage measurement.

The sequence can be seen in Figure 87, which shows the variation in noise following a step-function transient electrical input to a normal valve (20 kilohertz high-pass). The two traces represent the vibration on opposite sides of the valve. The flow noise is at first low; then, when the

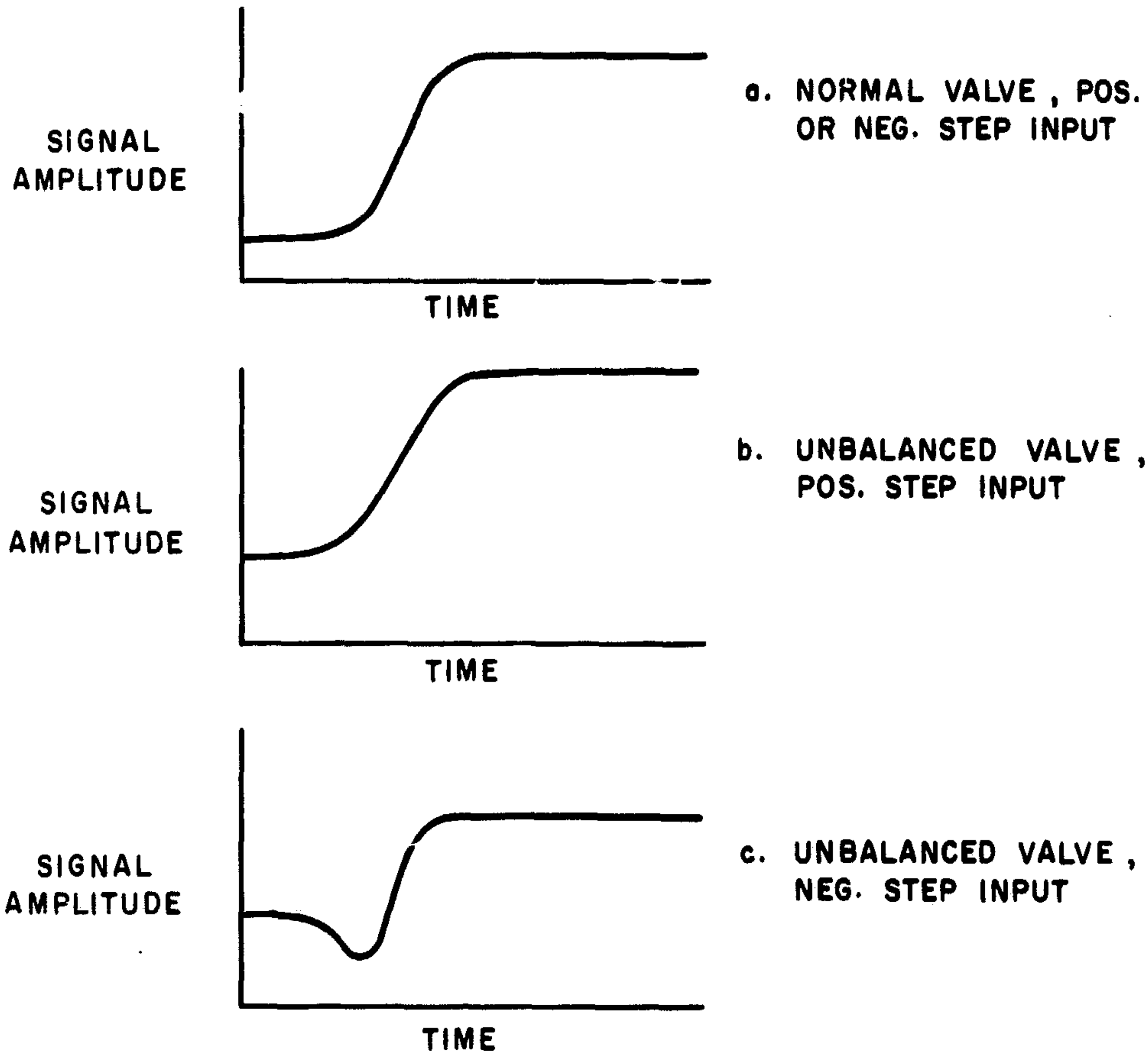


Figure 84. Predicted Variation of Flow Noise from Normal and Unbalanced Servo Valves with Step Input

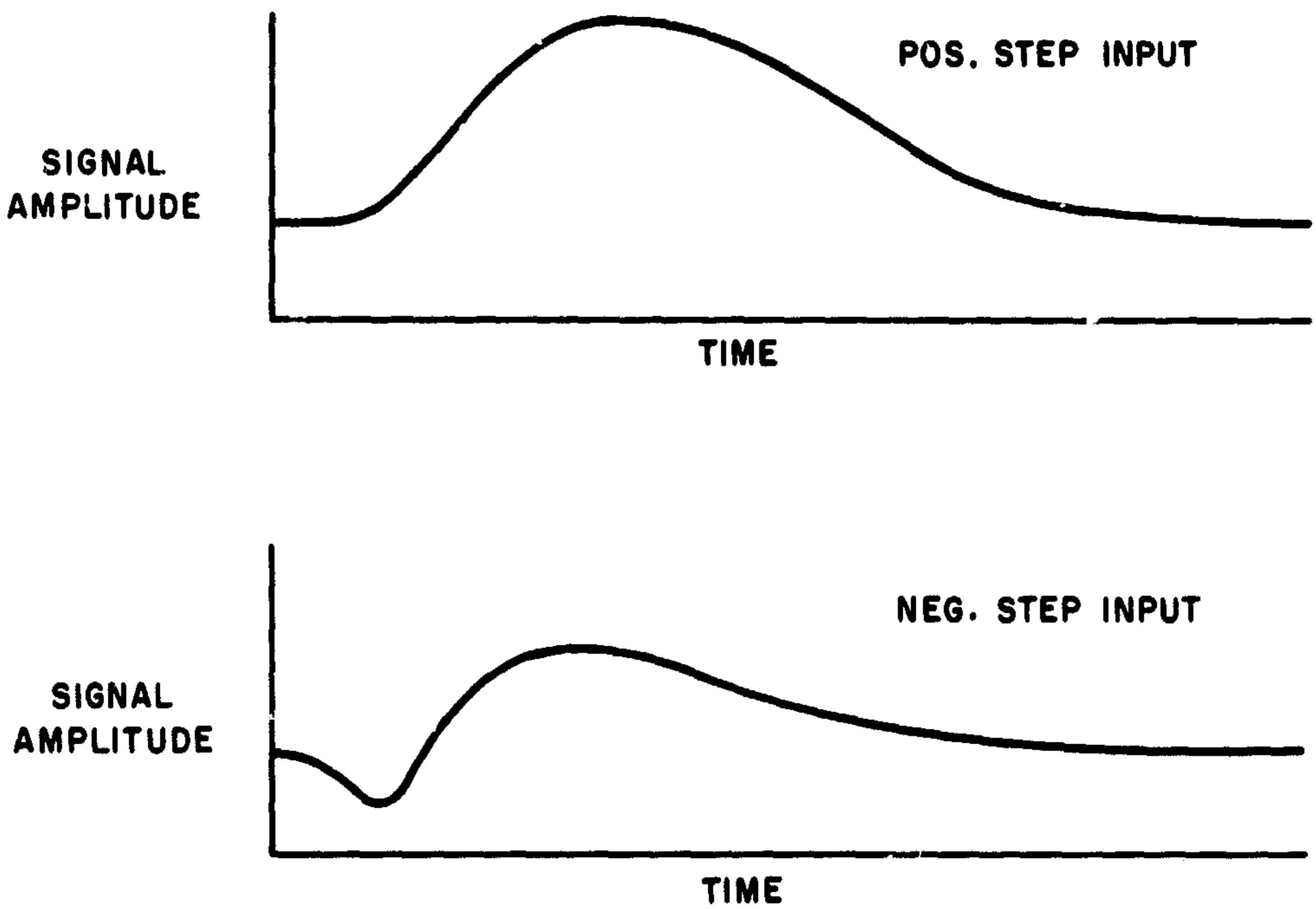
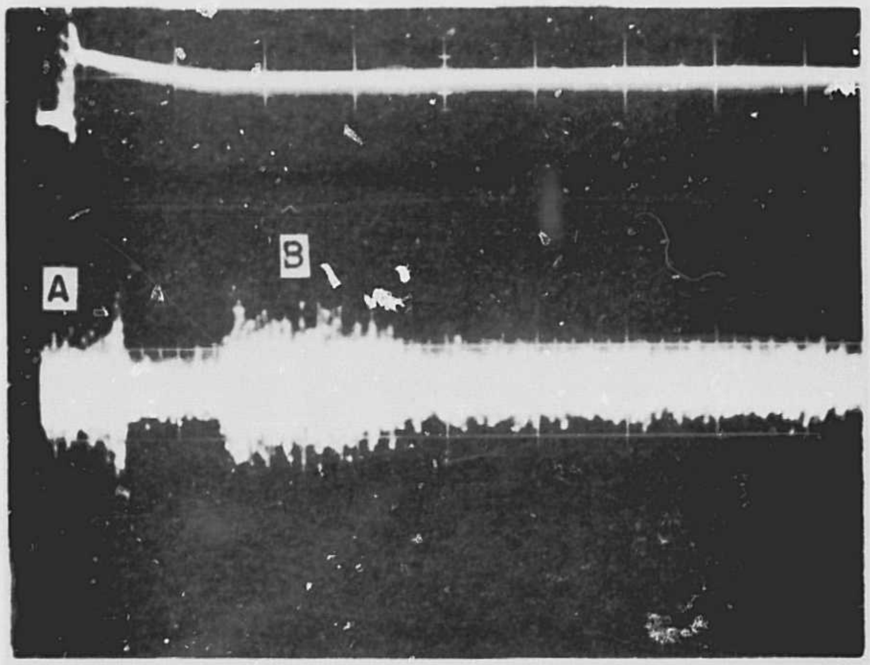
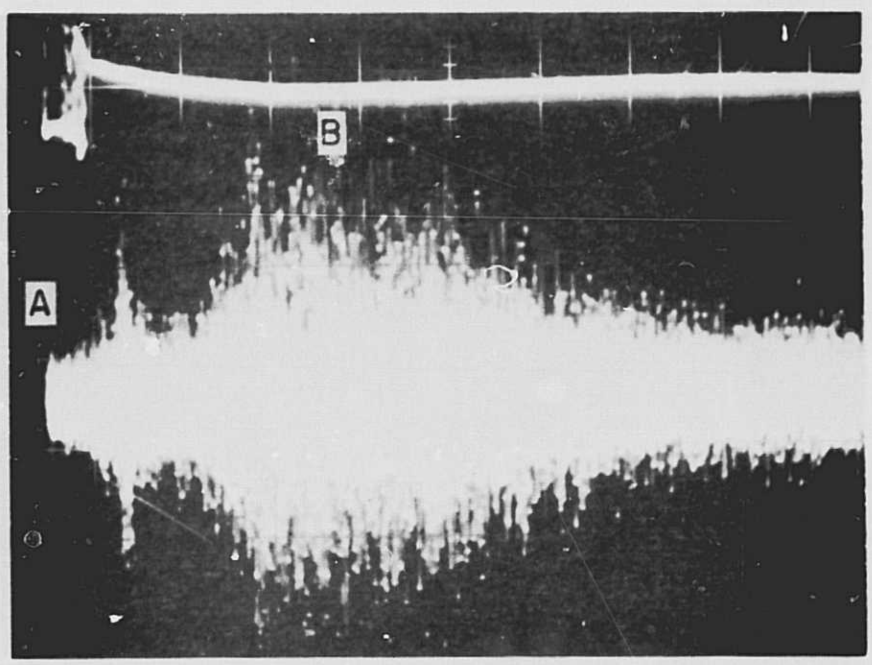


Figure 85. Predicted Variation of Flow Noise of Unbalanced Servo Valve and Test Facility



50 Milliseconds

Direction 1



Direction 2

Figure 86. Flow Noise of Unbalanced Servo Valve After Step-function Input

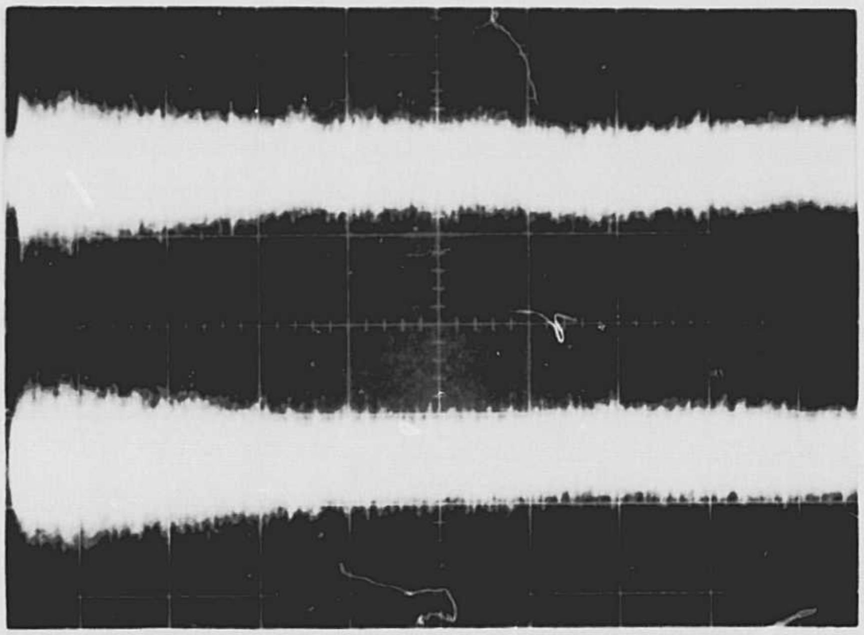


Figure 87. Response of Normal Servo Valve

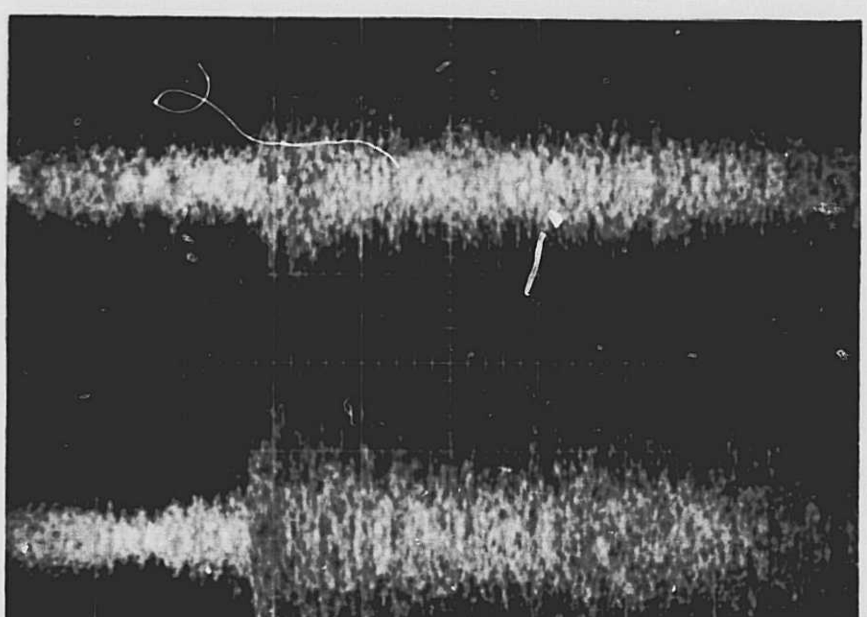


Figure 88. Response of Normal Servo Valve

valve responds, it increases abruptly. After about a second, the noise returns to the original condition.

Figure 88 is an expansion of the first part of the above transient. The full response takes 15 milliseconds, which is twice as long as was predicted. There is no significant difference in the noise when the sense or direction of the transient is reversed. The discrepancy between predicted and measured operating times could be explained if the frequency response of the valve were poorer than that used in the prediction. (The amplitude and phase characteristics of the actual valve tested were not measured). Figures 89 and 90 correspond to the two previous laboratory measurements, except that they were measured on the antenna at NTTF.

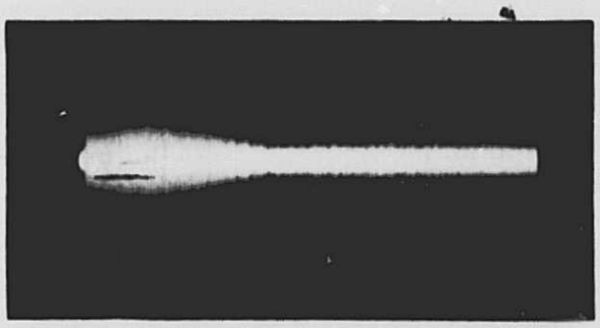
Figure 20 shows the modifications in the servo valve which were made by the manufacturer. In the rebuilding, the unbalance was removed from the valve and its leakage greatly increased. The responses of the leaky valve, measured in the laboratory on the servo pump and at NTTF, are shown in Figure 91. The flow at null was so great that no variation in flow noise could be measured, even though the valve appeared to function in its usual manner. The signals shown in Figure 91 were passed through a 20-kilohertz high-pass filter. Other passbands were tried with similar results. It is concluded that signature analysis techniques can be used to detect abnormally high leakage. If leakage is normal, the response time of the valve can be measured; this in turn relates to the frequency response.

PRESSURE RELIEF VALVE

The pressure relief valve normally operates continuously to maintain flow at 800 psi to the servo valve and the antenna braking system. Excess flow is returned to the drain manifold. Flow noise is generated by the flow through the servo valve and by that which is bypassed. The total of these two flows is 2 gpm, the rating of the control portion of the vane pump.

Figure 92 shows the flow noise generated by a normal valve. The data were recorded at NTTF during the preliminary investigation, in July 1967. There was no filtering of the signal, and a single sweep of the oscilloscope is shown at five milliseconds per centimeter. The signal amplitude is 3 g's RMS. Although there were several noise measurements of pressure relief valves in the laboratory, they are not presented because a question exists as to whether they were run at the rated 2 gpm flow.

Figure 93 corresponds to Figure 92, but was made a year later (also at NTTF), using the valve with scoring of the pilot valve seat. The RMS amplitude is 2.5 g's. There is no apparent difference between the two traces, in amplitude or character, which would disclose the malfunction. It must be assumed that the introduced defect was not sufficiently serious to cause chatter, hunting, or other instabilities in the pressure relief valve.

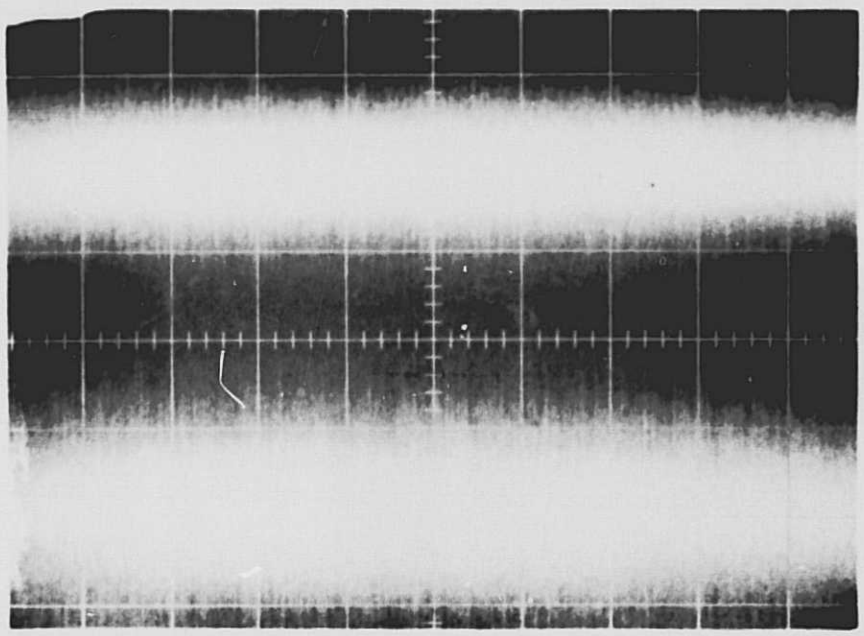


|← 2 Seconds →|

Figure 89. Normal Servo Valve Test at the Network Test and Training Facility, Goddard Space Flight Center

|← 50 Milliseconds →|

Figure 90. Normal Servo Valve Test at the Network Test and Training Facility, Goddard Space Flight Center



Vibration Signals Measured on Opposite Sides of Valve

2 Seconds

Laboratory Test

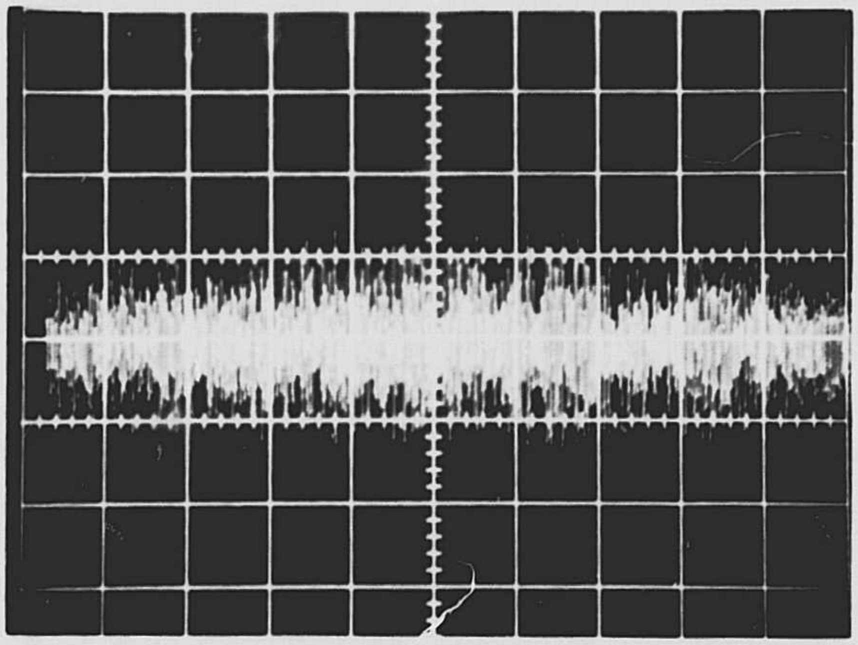
Trigger Signal

Vibration Signal

2 Seconds

Test at the Network Test and Training Facility

Figure 91. Tests of Servo Valve with High Leakage



|←——50 Milliseconds——→|

Figure 92. Flow Noise of Normal Pressure Relief Valve, Tested at the Network Test and Training Facility, Goddard Space Flight Center

|←—50 Milliseconds—→|

Figure 93. Flow Noise of Pressure Relief Valve with Defect, Tested at the Network Test and Training Facility, Goddard Space Flight Center

BEARINGS

Rolling-element bearings produce a "tick" or "click" whenever a local defect contacts one of the bearing surfaces. The periodicity of this transient may be predicted, as previously described, from the running speed and geometry of the bearing. In practice, the periodicity may differ slightly from the prediction for one or more of the following reasons.

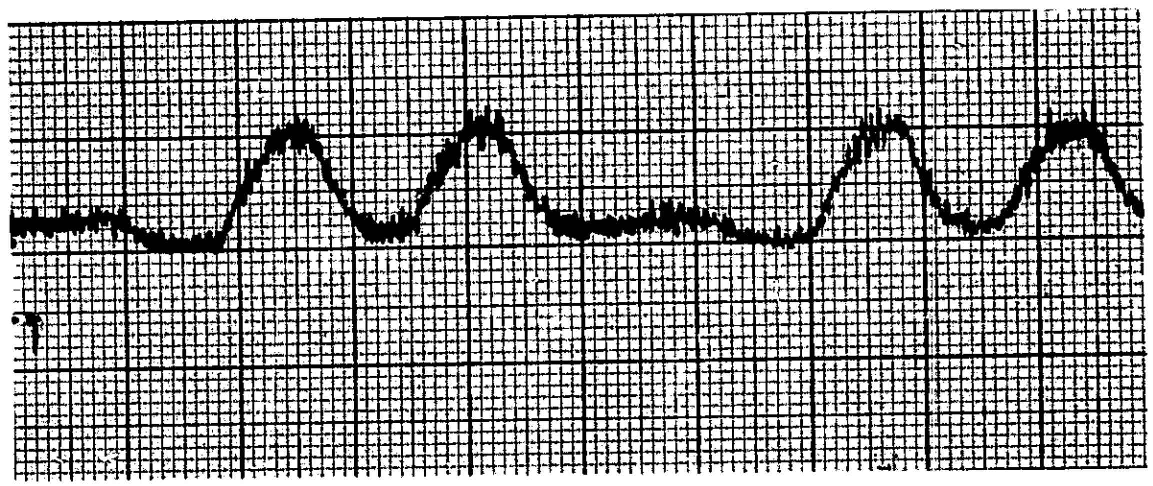
- Bearing dimensions may not be exactly known
- There may be some slippage between elements rotating at different velocities.
- In ball bearings the angle β (Figure 16) is usually not accurately known, since it depends somewhat on the thrust load.

Since the noise is not synchronous with shaft speed, the summation analysis which searches for the signal is triggered by an external source rather than by the shaft-position signal.

Since the periodicity is not exactly known, it is usually necessary to vary the period of the trigger in steps on either side of the calculated value to ensure proper coverage. In the figures that follow this has been done; it will be noted that the calculated and measured periodicities differ slightly. In all of these analyses the signal was rectified before being summed. In some of the summations the tape-recorded signal was played back more slowly than it was recorded, in effect stretching time by the recording speed/playback speed ratio. This was done for two reasons:

- The analog-to-digital converter in the Enhancetron 1024 summation device performs better with the slower signals.
- When the tape speed is halved, for example, the effective accuracy of the Time Mark Generator, used to generate trigger signals, is effectively doubled. In real time this device may be set (by means of digital switches) to a precision of ten microseconds. Such a setting is usually adequate, but in some instances finer control is desirable.

Figure 94 is a plot showing detection of the defects in the roller bearing. A total of 300 summations were made. The periodicity was measured as 4.06 milliseconds, as opposed to the prediction of 4.13 milliseconds for an outer race defect. The tape from which the analysis was made was recorded on July 17, 1968. While all three defects are visible, the second and third appear to have the same amplitude, connoting approximately the same defect size. It will be recalled that the third defect was actually the largest (Figure 21).



TEST LOCATION

Laboratory Test Facility	X
Network Test and Training Facility	

SERVO PUMP OPERATING PARAMETERS

Stroke, % : 0
Pressure, PSI : 0

SIGNAL PRECONDITIONING

No Filter	X
1 kHz LP Filter	
20 kHz HP Filter	
Unrectified	
Full Wave Rectified	X

ANALYSIS INSTRUMENTATION

GE - R&DC	X
NASA - GSFC	

TRIGGER SIGNAL

Shaft Position Sensor	
Pulse Generator	X
Switch	

Period, msec Real Time: 4.06

SUMMATION PARAMETERS

Number of Summations: 300
Sweep Period, msec Real Time: 7.81

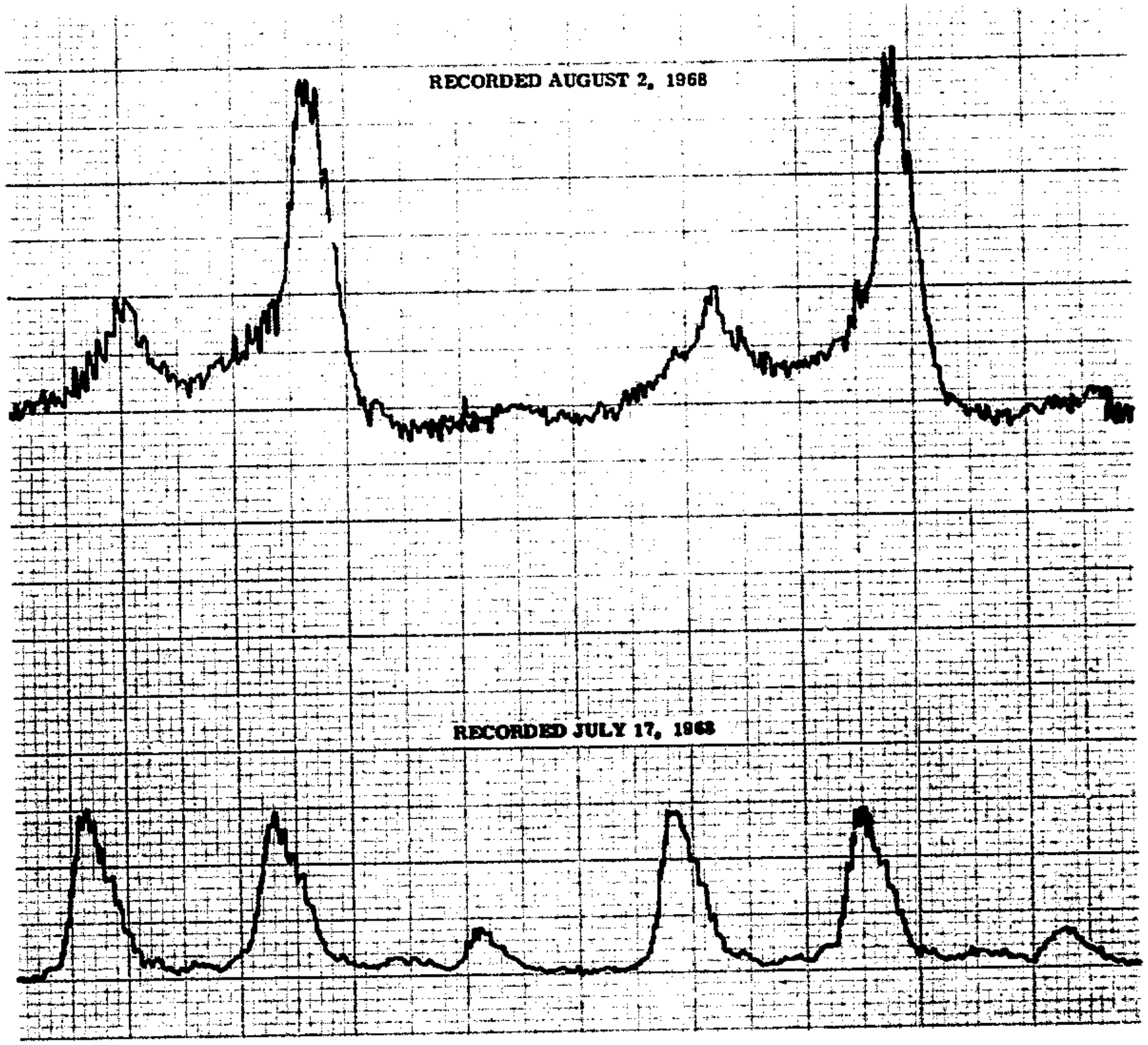
Figure 94. Summation Analysis, Servo Pump Roller Bearing With Simulated Spall

Figure 95 shows a second analysis of this tape at Goddard, and an analysis of another tape, made in the laboratory on August 2, 1968. Here the sequence of transients, from weak to moderate to strong, is as expected. The earlier results occurred when the simulated spalls were "fresh", and it is probable that there was a roughness at the edges of the two larger defects which masked their differences in size. After running in for several hours, the edges wore down and results were as expected.

Figure 96 is a plot of an analysis made of the signal recorded from a transducer over the servo pump duplex-pair of thrust bearings. (There were 350 summations made.) A periodicity of 12.22 milliseconds was predicted for an outer-race defect, assuming a 15 degree contact angle, while the actual periodicity was 12.17 milliseconds. The recurring burst of noise is clearly seen.

A defect was also introduced on the vane pump shaft, which is the inner race of a needle bearing. The predicted periodicity is 2.84 milliseconds. Despite a detailed search, the signature of this defect was not found. The explanation for the failure lies in the orientation of the defect, sketched in Figure 23. The simulated spall is not oriented along the shaft axis, as a normal spall would develop, but is skewed by about 1/64 inch. As a result of the skew, a roller of the bearing traverses the defect in a sweeping motion, instead of all points on the roller meeting it simultaneously. Because the line of contact of each roller bridges the defect, the noise characteristic of a locally defective bearing is not generated.

Summation analyses corresponding to Figures 94 to 96 were made from tapes recorded before the defects were introduced, as a means of checking the technique. As expected, nothing was found.



TEST LOCATION

Laboratory Test Facility	X
Network Test and Training Facility	

SERVO PUMP OPERATING PARAMETERS

Stroke, % : 0
Pressure, PSI : 0

SIGNAL PRECONDITIONING

No Filter	X
1 kHz LP Filter	
20 kHz HP Filter	
Unrectified	
Full Wave Rectified	X

ANALYSIS INSTRUMENTATION

GE - R&DC	
NASA - GSFC	X

TRIGGER SIGNAL

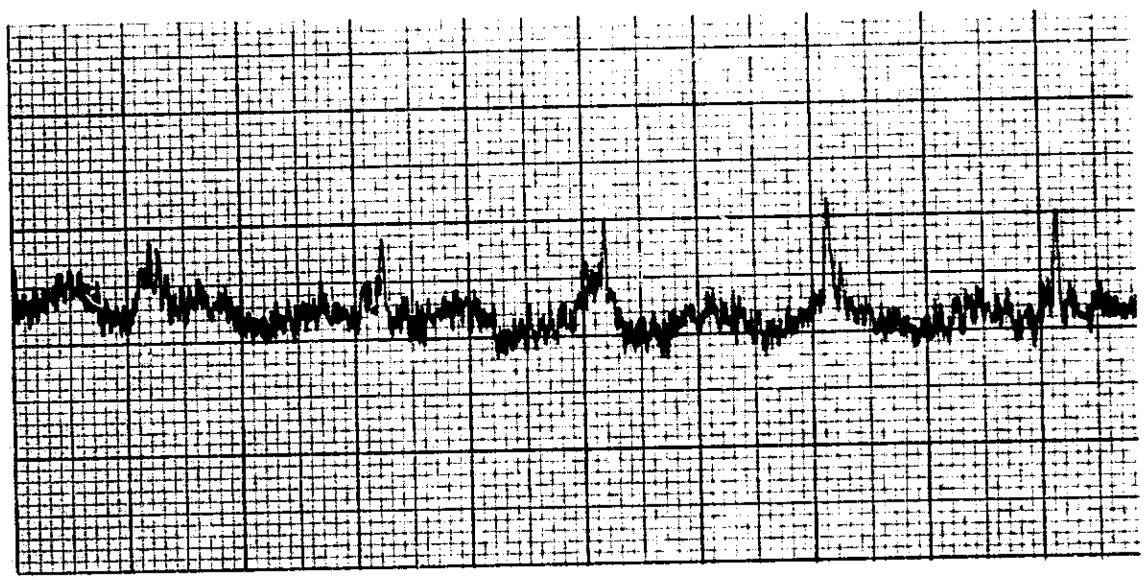
Shaft Position Sensor	
Pulse Generator	X
Switch	

Period, msec Real Time: 4.07

SUMMATION PARAMETERS

Number of Summations: 300
Sweep Period, msec Real Time: 7.81

Figure 95. Summation Analyses, Servo Pump Roller Bearing With Simulated Spall



TEST LOCATION

Laboratory Test Facility	X
Network Test and Training Facility	

SERVO PUMP OPERATING PARAMETERS

Stroke, % : 0
Pressure, PSI : 0

SIGNAL PRECONDITIONING

No Filter	X
1 kHz LP Filter	
20 kHz HP Filter	
Unrectified	
Full Wave Rectified	X

ANALYSIS INSTRUMENTATION

GE - R&DC	X
NASA - GSFC	

TRIGGER SIGNAL

Shaft Position Sensor	
Pulse Generator	X
Switch	

Period, msec Real Time: 12.17

SUMMATION PARAMETERS

Number of Summations: 350
Sweep Period, msec Real Time: 62.5

Figure 96. Summation Analysis, Servo Pump Ball Bearing With Simulated Spall

Section 8

CRITERIA FOR SIGNATURE EVALUATION

The purpose of diagnostics is to detect and identify various malfunctions of the hydraulic components, without disassembly, before the defect becomes so great as to preclude normal operation of the antenna drive system. This requires not only proper data acquisition and analysis instrumentation, but also establishment of criteria for evaluating the results of the analyses so that a decision can be made to continue normal operation or to schedule a repair.

Criteria, by their very nature, may be expected to grow in sophistication as experience is gained. Correlation of test results with field service data will cause some criteria to be modified and others to be added. However, the work included in this report represents a considerable body of experience in itself. The correlation of measurements and predictions, and the good agreement between analyses of data taken in the laboratory and on the antenna give credence to these results. As a consequence, criteria for evaluation of diagnostic signatures can be stated with confidence that their application will be useful for detection of malfunctions.

In effect, recommended criteria have been scattered throughout the previous discussions on signature prediction and results of analysis. The codification given below draws this information together in one place. Transducer locations and test procedures are assumed to be as previously discussed.

SERVO PUMP

The following tests apply to the servo pump.

1. If the vibration level at a frequency of one per revolution (approximately 20 Hz) increases in amplitude when running at high antenna velocities, relative to the level measured with yoke centered, differential wear of one or more cylinders is present to an abnormal degree.

2. If the vibration amplitude of a given pump at multiples of seven per revolution increases over a period of time (all tests at the same pumping condition), wear within the cylinders is increasing.

3. When the raw signal from a normal pump is summed using the shaft position transducer as a trigger, a waveform is recovered within which there is a characteristic function repeating seven times per revolution, as in Figures 45 through 47. If this is not found, there is a suspicion of malfunction.

4. When the rectified signal is summed as in Test 3, above, there may be a seven per revolution periodicity in the recovered waveform envelope. If the character of the summed signal is as in Figure 64 and the signal appears as in Figures 63 or 65, with a peak amplitude greater than 5 g's, the pump is cavitating. If the antenna velocity at the time of measurement is four degrees per second this is usual, although not desirable. If it occurs at lower velocities it indicates low supercharge pressure due to obstructed lines, dirty filters, or a defective replenishing pump.

5. If the summed rectified signal shows a periodicity of one per revolution there is probably scoring or galling at the valve plate of the wear plate.

6. If there are two relatively short bursts of noise per revolution in the summed rectified signal, spaced one-half revolution apart, there is probably wear at a ball-socket joint.

DUAL-VANE PUMP

A test of the dual-vane pump would include the following points.

1. The normal vane pump signature consists mainly of 12 equally spaced bursts of noise per revolution, with an amplitude of about 10 g's peak. The bursts are normally of about equal amplitude (range of 2:1). Deviations from the normal should be considered suspect.

2. Unusually low amplitude in any of the bursts described above in No. 1 is indicative of leaky vanes.

3. Measurable vibrations at exact multiples of three per revolution are indicative of a leaky vane.

4. When the raw, rectified signal is summed, the ratio of peak energy (in the twelve bursts of noise) to energy between bursts should be about 3:1. If it is less, the ring is roughened, there is scoring of the vanes, or both.

5. If the twelve bursts of noise are present more than one third of the time, i. e., if the duration of each burst is more than about 2 milliseconds, some blockage of the vane pump inlet is to be suspected.

6. If the twelve bursts of noise exceed 10 g's peak by a significant margin, there is washboarding of the ring, which may cause splitting of the vane block.

7. If one or more of the usual 12 bursts per revolution are missing, one or more vanes have collapsed.

SOLENOID VALVE

1. The time required for the valve to open after being deenergized should be between 10 and 20 milliseconds when tested at no-flow with the hydraulic pumps shut down. Shorter times indicate worn and leaky O-ring seals. Longer times indicate excessively tight seals or stiction due to spool scoring.

2. The time required for the valve to open when the system is pressurized by the pump should be about the same as Test 1, above. If there is flow noise as in Figure 81, there is air in the system.

SERVO VALVE

1. Noise above 20 kilohertz should vary in amplitude as shown in Figures 87 and 88 after a transient input. If the noise level does not increase by about 3:1 from minimum to maximum noise, there is excessive leakage.

2. If noise level does vary properly as in No. 1, above, the time delay in the valve after a step input should not exceed 15 milliseconds. Longer response times indicate deteriorated frequency response.

3. When tested as in No. 2, above, the noise level should change smoothly from minimum to maximum and back to minimum. If the noise level varies step-wise there is striction in the spool.

PRESSURE RELIEF VALVE

1. The overall vibration level should be about 3g RMS, as in Figure 92. Fluctuations in amplitude with time should not exceed 2:1. Greater fluctuations indicate hunting due to damage, probably in pilot valve assembly.

2. Short bursts of noise of high amplitude indicate chatter because of damage to the pilot valve.

BEARINGS

1. When the raw rectified signal is summed using a trigger period calculated with Equation 73, 74, or 75, there should not be any disturbance measured with this periodicity. Some searching in the vicinity of the calculated period may be necessary. If a disturbance is found it identifies a spall in the appropriate part of the bearing.

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Section 9

IMPROVED DIAGNOSTIC INSTRUMENTATION

The instrumentation now owned by Goddard Space Flight Center has been proven capable of detecting a wide variety of malfunctions in hydraulic components. The rationale and methodology which have been applied to the servo and vane pumps and the solenoid, servo, and pressure relief valves can probably be extended to other components of the antenna with similar success.

At present the instrumentation for data acquisition and analysis is best operated by a skilled individual who is conversant with the requirements and capabilities of the system. As use of diagnostic procedures increases, it becomes more and more desirable to make the system simpler and quicker to operate. This may first come about through improvements in the present analog instrumentation, but it is likely that the final realization of the system will involve pattern recognition using a digital computer.

At present, it is probably most economical to feed data from a variety of sources to a central analysis facility. In the future, the benefits of diagnostics may be better realized through a decentralization which puts an analysis system at each antenna site. Several steps will be described in this section which would lead toward improved diagnostic capability.

DATA ACQUISITION INSTRUMENTATION

For the present, as experience is gained and the capabilities of diagnostics are explored, data will be recorded at the various antenna sites on magnetic tape and sent to Goddard for analysis. A team could travel from site to site to make recordings, carrying with them the required instrumentation, or the recordings could be made by station personnel at each site. If the former alternative is chosen the team can be trained to take the proper data using general purpose equipment, but if station personnel are to take the data it is highly desirable that special instrumentation be designed and built to make the procedure as simple as possible. If this is not done there will be questions as to calibrations, correctness of connections, overloading of data channels, and the like; questions which if not resolved could make the analysis meaningless.

If special purpose instrumentation of a particular, standard design is supplied, it will also be possible to setup a standardized step-by-step instruction for its use. This would ensure consistency of the data-taking procedure. Differences in results of analyses could then be confidently ascribed to differences in the hydraulic components and not to errors in data acquisition.

Such instrumentation should include everything necessary for data-taking except the tape recorder, since this is already available at each site. The following equipment should be included:

1. Transducers
2. Attachment fixtures and cement
3. Calibration means
4. Built-in amplifiers of known, fixed gain
5. Means for exercising the servo valve and the solenoid valve
6. Wires and cables
7. Instruction manual

The instrumentation should be designed in such a way as to minimize the possibilities for errors in its use. It should include no more controls than necessary and should be preconnected with appropriate gains or attenuations built in. It should be self-contained, compact, rugged, and light. Construction of this system is felt to be of first importance since diagnostics can be of value only if the original data are beyond suspicion.

ANALOG DATA REDUCTION INSTRUMENTATION

The present data reduction instrumentation, previously described, can successfully detect many malfunctions of the hydraulic components. The only present lack is a 20-kilohertz high-pass filter, which is required for servo valve analyses. The Research and Development Center has used for this purpose a variable filter with slope of 24 decibels per octave and found it satisfactory. A filter with these characteristics should be either purchased or synthesized with operational amplifiers.

The Hewlett-Packard 203A Function Generator has been used to provide the trigger signal in summation analyses for detecting bearing defects. While it can perform the function, it is difficult to use for three reasons.

1. It is calibrated in terms of frequency, while the important characteristic of its output is really its period.

2. Its calibration is too coarse to be read accurately; hence a counter must be used to read out the period of the signal which it is generating.

3. It is extremely difficult to adjust its output period in the small increments which are needed for bearing analysis.

For these reasons a more suitable replacement should be considered. One solution would be to duplicate the Time Mark Generator which was used by the Research and Development Center. Its characteristics are such that it meets the objections to the H. P. 203A listed above. In addition, it has an automatic search feature whereby it can deliver a measured pulse train at a given periodicity; then, automatically step itself through a sequence of pulse trains with increasingly longer periodicities. It automatically clears the memory of the summation device whenever the period steps.

COUPLING TO A DIGITAL COMPUTER

At present it is necessary for a skilled technical person to apply criteria to diagnostic signatures in order to make decisions as to the proper course of action. This is an inexpensive way to get started, since equipment costs are minimized. It is a way of getting experience quickly, but it has certain long-term disadvantages. First, the human operator is flexible, but he may make mistakes. He is relatively slow, and cannot process large quantities of data. Also, it would probably not be practical to put a trained man at each antenna site if the decision is made to perform the diagnosis on-site rather than at a central facility. These objections are overcome (at the expense of more sophisticated instrumentation) if a computer takes over the analysis function.

Data containing diagnostic information comes from the transducers as analog voltages. This must be converted to digital form before it can be accepted by a computer. The most obvious way to do this is to perform a high-speed analog-to-digital (A/D) conversion of the raw signal and store it in the computer memory for further processing. While this seems straightforward, it is not practical at the present state of the art.

According to sampling theory, the sampling rate of the converter must be at least twice the highest frequency of interest. If the highest frequency is ten kilohertz or so, which is reasonable for diagnostic purposes, it is seen that even a short data sample yields an enormous amount of information to be stored. The computer time required for processing is correspondingly large. This approach is technically feasible, but not economical. (This situation may change in years ahead as advances are made in computer technology.)

In order to limit the amount of information which must be transmitted to the computer, some form of data preprocessing is desirable. This involves putting some kind of a buffer between the analog data and the computer. A transient-averaging or summation device, such as the Goddard-owned Computer of Average Transients (CAT) can perform this function. As previously described, it performs the required A/D conversion and stores the averaged response to a stimulus; this is important for diagnostic purposes. Because of its summation capability it reduces a very large amount of information (the raw data) to a relatively small amount of information (that which is stored in its memory), effectively discarding extraneous information while retaining that which is significant.

Digitizing of Component Signals

Before discussing the coupling of the CAT to the computer it is worthwhile to reconsider the ways that the CAT can be used to digitize the signals from hydraulic components. There are two requirements: a stimulus (used to trigger the sweep of the CAT), and a signal which contains a response to the stimulus.

Servo Pump. The stimulus to pump vibration is revolution of the shaft, and the shaft-position variable-reluctance transducer provides the electrical impulse used for triggering. The signal to be analyzed may or may not be rectified before summing, depending on what a particular test is trying to accomplish. Several different summations will be required, one corresponding to each test, and the result of each is transmitted to the computer. Many examples of such summations are included in this report.

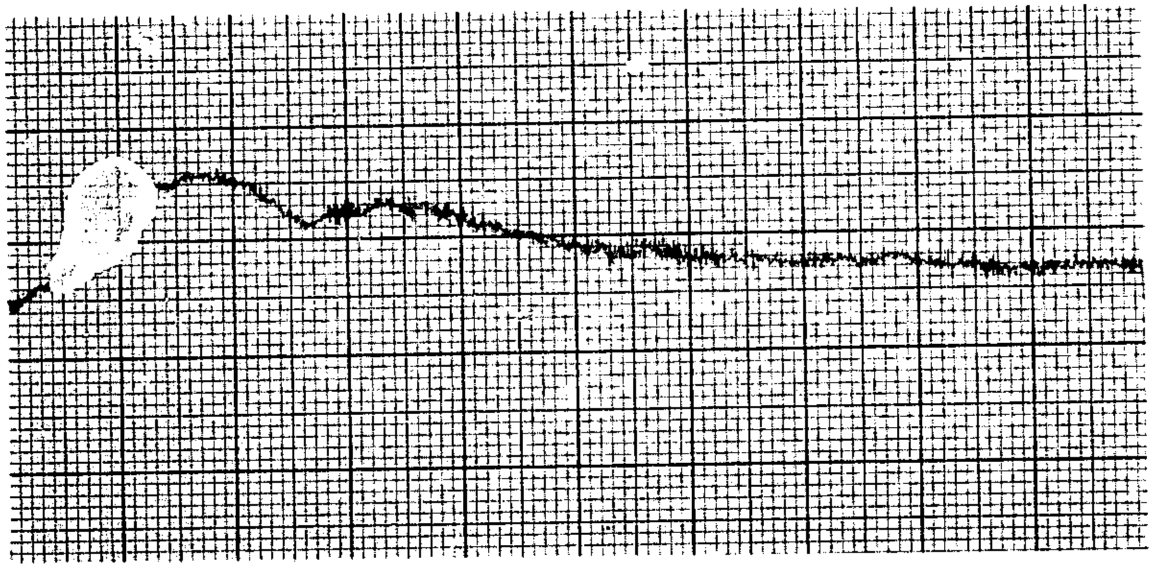
Vane Pump. Requirements for vane-pump analyses are the same as for the servo pump. Fewer tests will be required to characterize a given pump since, unlike the servo pump, the operating condition is fixed. The CAT has been used for such analyses.

Solenoid Valve. While a transient averaging device has not been used to study the solenoid valve, this could readily be done. The stimulus would be the opening of the switch which energizes the solenoid coil. It would be best to rectify the vibration signal before summing, since the analysis requires only the shape of the envelope.

Servo Valve. The servo valve signature could be captured by the CAT by the use of the stop-function input as the trigger. The vibration signal would be rectified before summing. That technique has not been used routinely in this study, but was included on an experimental basis to show its practicality. Figure 97 is the summed signal; it corresponds to the lower trace in the oscilloscope photograph shown in Figure 86.

Pressure Relief Valve. The pressure relief valve does not respond to an externally applied stimulus, as do the components previously described. Using an arbitrary repetitive trigger signal it would be possible to digitize and measure the average amplitude of the signal (by rectification before summation), but it would not be possible to capture details of the waveform envelope. Tests may indicate, however, that it is possible to get an adequate A/D conversion in the CAT on a single sweep, in which case a single arbitrary trigger could be applied and the waveform envelope captured in a single sweep. If this move proves to be practical, a single measurement would give both the average signal amplitude and a measure of the fluctuation of the noise, supplying the full information needed for a diagnosis.

Bearings. This report shows several examples of summations of bearing signals for defect detection. The trigger is supplied by an external source of the proper periodicity. This analysis would be the most difficult of all to computerize since the proper trigger periodicity is not exactly known. As a consequence, a search must be made using periodicities in a range about the expected value. While the search procedure could be automated, the logic required to recognize that the proper periodicity had been found would be complicated.



TEST LOCATION

Laboratory Test Facility	X
Network Test and Training Facility	

SERVO PUMP OPERATING PARAMETERS

Stroke, % : NA
Pressure, PSI: NA

SIGNAL PRECONDITIONING

No Filter	
1 kHz LP Filter	
20 kHz HP Filter	X
Unrectified	
Full Wave Rectified	X

ANALYSIS INSTRUMENTATION

GE - R&DC	X
NASA - GSFC	

TRIGGER SIGNAL

Shaft Position Sensor	
Pulse Generator	
Switch	X

Period, msec Real Time:

SUMMATION PARAMETERS

Number of Summations: 20
Sweep Period, msec Real Time: 125

Figure 97. Summation Analysis, Unbalanced Servo Valve

In summary, the CAT could easily be used for complete automated analyses of servo and vane pumps, solenoid valves, and servo valves. At least a partial analysis, and perhaps a full analysis, of pressure relief valves could be made. Bearing analyses could be made, but more effort would be required to implement the process. The stored waveforms, after transmission to the computer, could be analyzed by the shapes of their envelopes, by spectral content (from Fourier Analyses), or by the timing of events.

Information resulting from summations is stored by the CAT in its core memory. This can be read out in two ways. For the first output, each memory element is read in sequence and reconverted from digital to analog form. This output, coupled with another signal proportional to the number of the memory address, can be used to make an X-Y plot on an oscilloscope face, or can be fed to an X-Y recorder for permanent storage. This output was used in generating the plots included in this report. A second output takes the digital data from the core memory and presents it in digital form, using the BCD code. Address data are also available in BCD form.

The General Electric Company Research and Development Center has already built a facility for its own use which couples the Enhancetron 1024 summation device to the General Electric Time-sharing Computer System. The Enhancetron lacks a digital output such as is offered by the CAT, but does have an analog output. Consequently, the system devised would be compatible with the analog output of the CAT.

Figure 98 is a block diagram of this equipment. The memory of the summation device is read out address-by-address using a Hewlett-Packard 2401C Integrating Digital Voltmeter (DVM). The digital output of the DVM

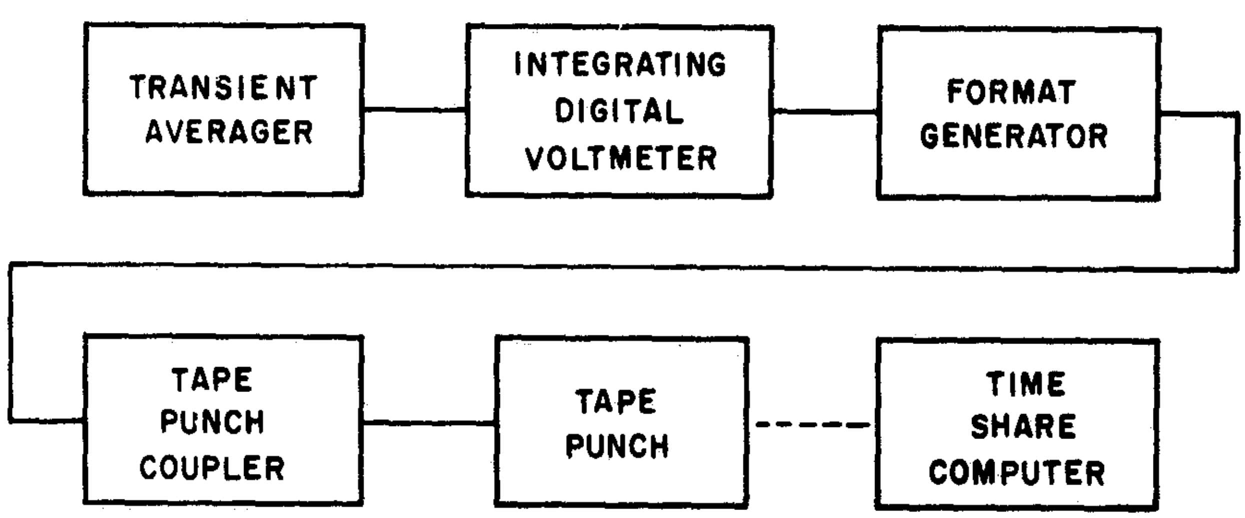


Figure 98. Coupling of Transient Averager to the Time-sharing Computer

is not in a format acceptable to the computer without editing. The editing functions are automatically performed by the Format Generator, which inserts line numbers, adds symbols to separate items of data, and other functions. The Format Generator output drives an H.P. 2545A Tape Punch Coupler and H.P. 2545C Paper Tape Handler.

The end result is a punched paper tape which is accepted by the conventional teletype machine of a Time-sharing Computer terminal. This tape may be formatted (at the operator's option) for either FORTRAN or BASIC languages, either of which may be used in the time-sharing system. The tape does not require any editing other than that automatically performed by the Format Generator.

As an example of how this system can be used, a computer program was written to analyze data from vane pumps. The signal from the vane pumps was assumed to be rectified and summed, as represented by Figures 68 to 73. In the program called VANE 1 (a listing is included in Appendix II) lines 200 to 233 are data from the paper tape. Every other address of the Enhancetron was read out to approximate the 400-address memory of the CAT. Line 1000 is the relative gain used; line 1010 is the test number. Line 1120 is the period corresponding to one shaft revolution; it is used in line 1130 to compute the number of addresses between the normal 12 noise bursts.

In lines 1140 to 1270 the actual waveform is cross-correlated against an idealized waveform. The idealized waveform is shifted in time relative to the actual waveform, and when the correlation function is a maximum the positions of the 12 noise bursts are considered located. The program then looks between the 12 peaks to determine signal amplitude in what is normally a quiet period, measures the amplitude of each of the 12 bursts, finds the largest of them, determines the average amplitude of the twelve bursts, and then takes various ratios between these measured values.

Starting at line 1702, the various measurements and ratios are compared to previously stored limits. If any parameter is out of limits, note is made and the number of out-of-limit parameters counted. Depending on the number counted, the program recommends various actions, starting at line 2400.

The results of three runs of the program are given in Tables 7 to 9: Test Number 1 (Table 7) used data from a normal pump; Test Number 102 (Table 8) was that of the control pump with the damaged ring shown in Figure 18; Test Number 101 used data from the replenishing pump which was run simultaneously with the defective control pump.

Not by chance, the print-out characterizes the pump of the first test as normal. In fact, its characteristics were used in setting allowable limits for the measured parameters.

The pump with the defective ring is readily recognized as in need of repair since it is out-of-limits in seven of the nine categories which the

Table 7

COMPUTERIZED DIAGNOSIS OF NORMAL CONTROL PUMP

VANE 15:38 MARCH 14,1968

TEST NUMBER 1

SIGNAL ANALYSIS

RMS VALUE ØF SIGNAL BETWEEN PEAKS= 46.6181

MAX. PEAK AMPLITUDE= 207

MEAN PEAK AMPLITUDE= 145

RMS VALUE= 65.0622

MAX. PEAK TØ MEAN PEAK RATIØ= 1.42759

PEAK TØ RMS RATIØ= 3.18157

MEAN PEAK TØ RMS RATIØ= 2.22864

RATIØ ØF SIGNAL BTW. PEAKS TØ RMS AMPL.= .716517

DISCRIMINANT ANALYSIS

NØ DISCRIMINANTS ØUT ØF LIMITS.

RECØMMENDED ACTIØN

NØRMAL MAINTENANCE ØNLY.

TIME: 29 SECS.

Table 8

COMPUTERIZED DIAGNOSIS OF CONTROL PUMP
WITH DEFECTIVE RING

VANE 15:41 MARCH 14,1968

TEST NUMBER 102

SIGNAL ANALYSIS

RMS VALUE ØF SIGNAL BETWEEN PEAKS= 640.976

MAX. PEAK AMPLITUDE= 2595

MEAN PEAK AMPLITUDE= 1424.58

RMS VALUE= 749.901

MAX. PEAK TØ MEAN PEAK RATIØ= 1.82159

PEAK TØ RMS RATIØ= 3.46046

MEAN PEAK TØ RMS RATIØ= 1.89969

RATIØ ØF SIGNAL BTW. PEAKS TØ RMS AMPL.= .854748

DISCRIMINANT ANALYSIS

EXCESSIVE SIGNAL AMPLITUDE BETWEEN PEAKS

EXCESSIVE PEAK AMPLITUDE

EXCESSIVE MEAN PEAK LEVEL

EXCESSIVE RMS AMPLITUDE

EXCESSIVE MAX. PEAK TØ MEAN PEAK RATIØ

DEFICIENT MEAN PEAK TØ RMS RATIØ

RATIØ ØF SIGNAL BTW. PEAKS TØ RMS AMPL. EXCESSIVE

RECØMMENDED ACTIØN

7 DISCRIMINANTS ØUT ØF LIMITS. IMMEDIATE ØVERHAUL REQUIRED.

TIME: 29 SECS.

Table 9

COMPUTERIZED DIAGNOSIS OF NORMAL REPLENISHING PUMP ON SAME SHAFT AS DEFECTIVE CONTROL PUMP

VANE 15:45 MARCH 14,1968

TEST NUMBER 101

SIGNAL ANALYSIS

RMS VALUE ØF SIGNAL BETWEEN PEAKS= 109.574

MAX. PEAK AMPLITUDE= 320.

MEAN PEAK AMPLITUDE= 217.986

RMS VALUE= 121.528

MAX. PEAK TØ MEAN PEAK RATIØ= 1.46798

PEAK TØ RMS RATIØ= 2.63314

MEAN PEAK TØ RMS RATIØ= 1.79371

RATIØ ØF SIGNAL BTW. PEAKS TØ RMS AMPL.= .901633

DISCRIMINANT ANALYSIS

EXCESSIVE SIGNAL AMPLITUDE BETWEEN PEAKS

DEFICIENT MEAN PEAK TØ RMS RATIØ

RATIØ ØF SIGNAL BTW. PEAKS TØ RMS AMPL. EXCESSIVE

RECØMMENDED ACTIØN

3 DISCRIMINANTS ØUT ØF LIMITS. SCHEDULE INSPECTIØN WITHIN ØNE MØNTH.

TIME: 29 SECS.

program considers. The third test, of the control pump, indicates that something is wrong but the defect is not yet critical. Actually, this pump was normal. The confusion arises because of cross-talk between the normal replenishing pump and the defective vane pump. Vibrations generated at the bad ring propagated through the structure and were sensed by the accelerometer on the replenishing pump.

This program is a good example of what the computer can do, since it includes the three essentials of any diagnostic process: signal analysis, comparison of analysis with discriminants, and a recommendation for action.

The approach outlined above using the Time-sharing Computer is an attractive one for the development of programs and techniques for automated analysis, and for the processing of modest amounts of test data. Its great advantage is in breaking down the barrier between the engineer and the computer by virtue of its easy access, the simplicity of modifying programs, and virtually instant turn-around time from data entry to printing of results.

If large quantities of data are to be handled, batch processing is usually more economical than time-sharing. The CAT may still be the core around which the system is built. The Technical Measurement Corporation, manufacturer of the CAT, produces a compatible device, called the Model 535EL Teletype Page Printer, which can act as the interface between the CAT and a computer (Figure 99). The 535EL is a modified ASR33 Teletype. It may be purchased from the Technical Measurement Corporation,

Figure 99. Coupling of CAT to the Computer for Batch Processing

or they will modify an existing ASR 33. It accepts the information stored in the CAT (which is in BCD form), performs a parallel-to-serial conversion, and produces a punched tape in ASCII code.

The 535EL prints 10 data words on a line, separated by spaces. Data words have either five or six digits, depending on the particular model of CAT 400 which is used. Carriage returns and line feeds are supplied automatically, as is a stop code when memory readout is complete. The operator may choose odd or even parity. Other information which may be desired, such as an identification code for each run, can be entered on the tape from the teletype keyboard. The punched paper tape produced in this way can be handled directly by many computer centers.

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NEW TECHNOLOGY

In the course of this investigation a background of theory was built up which permitted prediction of the externally measured vibration of certain important hydraulic components, under normal operating conditions, and with certain defects which were held to be those most likely to occur in service. The defects or malfunctions studied were to be of an incipient nature; i. e., of such character as to permit apparently normal operation but of a nature which would become progressively more serious with time, leading to ultimate failure unless remedial action is taken.

Tests of normal components and of components with defects were made and analyzed using suitable data reduction techniques. It was demonstrated that vibration signatures of both normal and malfunctioning components were as predicted, and that defects can be detected from such analyses without disassembly of either the system of the hydraulic components. It is believed that the latter capability constitutes a "new technology."

Application of this technology by the National Aeronautics and Space Administration requires three capabilities:

- Data acquisition
- Data reduction
- Ability to evaluate the reduced data

The techniques, equipment, and procedures which constitute the new technology are discussed in detail in three sections of This Final Report:

Section 4: Data Acquisition

Section 6: Data Reduction System

Section 8: Criteria for Signature Evaluation

Taken together, they constitute a handbook which NASA personnel may use to perform diagnostic investigations of hydraulic components of the Unified S-Band antenna. Other sections of the Final Report give the foundation upon which the technology is built, and discuss ways in which it can grow in capability.

Section 11

RECOMMENDATIONS

As a result of the program described in this report, the Goddard Space Flight Center has the present capability of performing diagnoses on hydraulic components. In order to implement this it will be necessary to provide means whereby data from the field can be sent to Goddard for analysis and interpretation. Since the quality of the diagnosis is dependent on the quality of the input signals, steps must be taken to ensure that good data are supplied. It is unrealistic to believe that uniformly reliable information will come from the field unless a special effort is made. This could take the form of one of the following three alternatives:

1. Assemble a set of data acquisition equipment as described in Section 4 of this report and train a test crew in its proper maintenance and use. This crew would visit the remote sites and tape-record data for transmittal to Goddard.

2. Design and build a rugged, portable, integrated unit which would include everything necessary to acquire diagnostic data except the tape recorder. In its design the data-taking procedure should be simplified as much as possible in order to minimize errors in its use. Prepare a simple instruction manual for use by station personnel. Ship the system from site to site, having station personnel take the data on the tape recorder which is part of the normal equipment complement at each antenna location.

3. The third alternative would be similar to the second except that the equipment would not need to be portable; a separate device would be made part of the permanent equipment at each site.

The overall value of diagnostics should be increased by extending it to applications other than those studied thus far. This could include other hydraulic components of the USB antenna, other antenna drives of similar design, nonhydraulic machinery, and nonmechanical applications such as electronic circuits. A considerable body of knowledge already exists in the General Electric Company pertaining to these related areas.

Automation of the centralized diagnostic facility through coupling to a digital computer was described in Section 9. It is recommended that this be done through a three-step process.

1. Use the present capabilities and equipment of General Electric to develop diagnostic computer programs using the time-sharing computer. The time-share approach is ideal for this purpose because of the ease with which programs can be modified and tested and the near-instantaneous response which eliminates the problem of turn-around time.

2. Acquire the equipment from Technical Measurement Corporation which serves as the interface between the CAT 400 and a general purpose computer.

3. Integrate the programs developed in (1) and the equipment in (2), to yield the desired computerized system. This would be able to handle relatively large amounts of data at low cost by making use of existing computer facilities at Goddard.

Following this, consideration should be given to performing diagnoses on-site at each antenna. This would allow frequent diagnostic tests and fast response to diagnostic information. The diagnostic process would be amenable to complete automation. The concept for such an automated system is shown in Figure 100. A programmer, using a medium such as punched tape as its memory, would control the process. It would send commands to the electronic subsystem to put the antenna through a series of diagnostic exercises. Transducers on the hydraulic components would transmit signals to the CAT 400 as previously described. The CAT would read in raw data and read out its memory under control of the programmer. The digitized data would be formated for the particular computer in use by the format control. The computer, as directed by the programmer, would process the data, perform discriminant analyses, identify incipient defects, and recommend remedial action. The latter might include a proposed schedule for repairs and replacement part numbers.

On-site analysis should be considered a tentative, long-range goal, whose need would be established through experience. If centralized data reduction gives an early enough warning of incipient problems, it may never be needed. On the other hand, there is a clear advantage to computerizing the centralized data analysis facility, and it is recommended that this process be undertaken now.

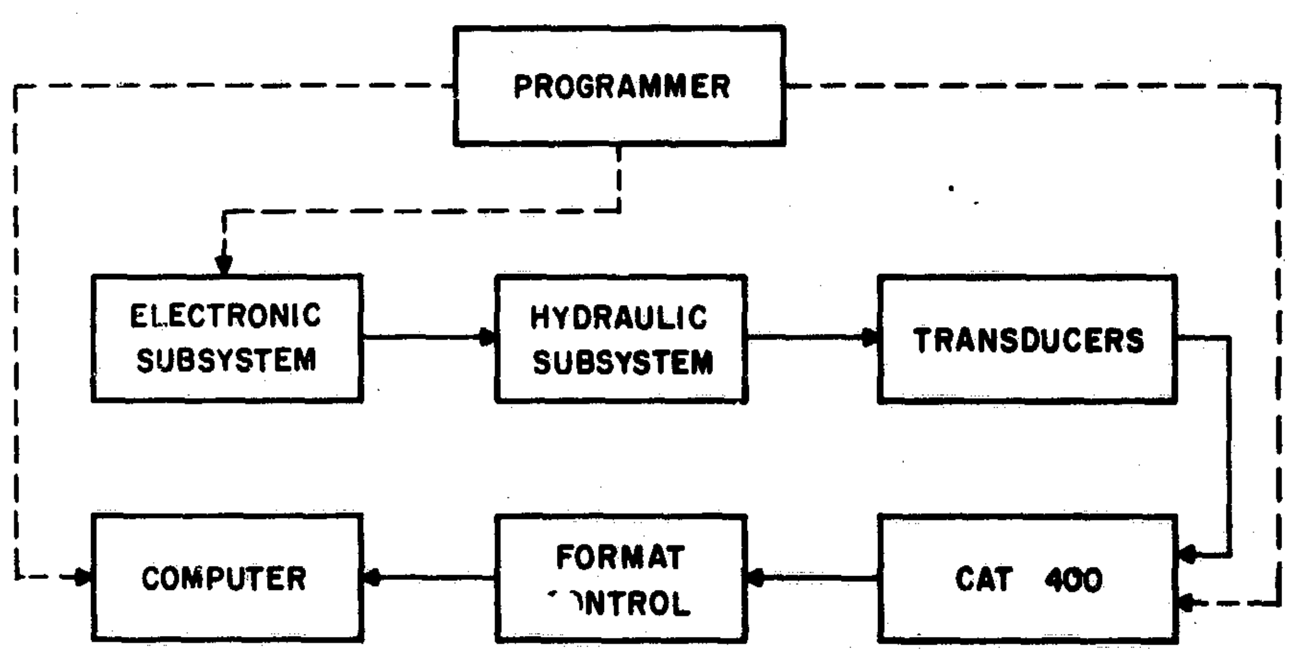

Figure 100. Automatic On-site Diagnostic System

Section 12

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Appendix I

DRIVE SUBSYSTEM DESCRIPTION

In order to clarify the requirement, a description is given here of a typical drive subsystem. This is one of the more complex of the drives of interest; more simple systems exist, as well. These consist of constant pressure pumps with relatively large servo valves controlling flow to fixed displacement drive motors or rotary actuators. The drive described illustrates most of the salient features of all types. (See Figure I-1.)

The servo control and drive system consists of transistorized electronic circuits which control a hydraulic drive system. The hydraulic drive system, in turn, rotates a satellite tracking antenna on an X and Y axis in a manner that permits satellite tracking from horizon to horizon. The control signals are amplified by the servo amplifier and applied to a servo control valve on the appropriate hydraulic drive unit. The X and Y axis hydraulic drive units are functionally identical. Each of the hydraulic drive units is controlled by separate control signals from a servo amplifier. The servo control valve positions the yoke of a variable-displacement type of hydraulic pump. The position of the pump yoke determines the direction of flow and the flow rate of the hydraulic oil pumped by the variable displacement pump. The direction of hydraulic oil discharge from the main hydraulic pump determines which of the two hydraulic motors drives the antenna. Since each hydraulic motor rotates the antenna in only one direction, the direction of hydraulic oil flow is directly associated with the direction of antenna rotation.

A linear potentiometer is also connected to the main hydraulic pump yoke. The potentiometer indicates the pump yoke position and gives an indication of the direction and quantity of hydraulic oil flow in the system. The electrical signal from the potentiometer is fed back to the servo amplifier through the servo box. The hydraulic motors are the fixed-displacement piston type, which provide mechanical rotary motion when driven by hydraulic oil delivered by the main pump. Each hydraulic motor shaft is connected to a reduction gear train which drives the antenna. The gear train and motor connections also provide an antibacklash system to allow change of direction of the antenna without any time delay. Each gear assembly is equipped with a brake assembly to hold the antenna stationary. These brakes are not hydraulically applied; however, hydraulic pressure is used in releasing the brakes.

The main servo pump has a flow rate of zero to 20 gallons per minute and a maximum pressure rate of 5000 pounds per square inch. The maximum pump pressure is limited to 2000 pounds per square inch by an adjustable relief valve. While limited to 2000 pounds per square inch by the relief valves, the pressure developed by the servo pump to drive the

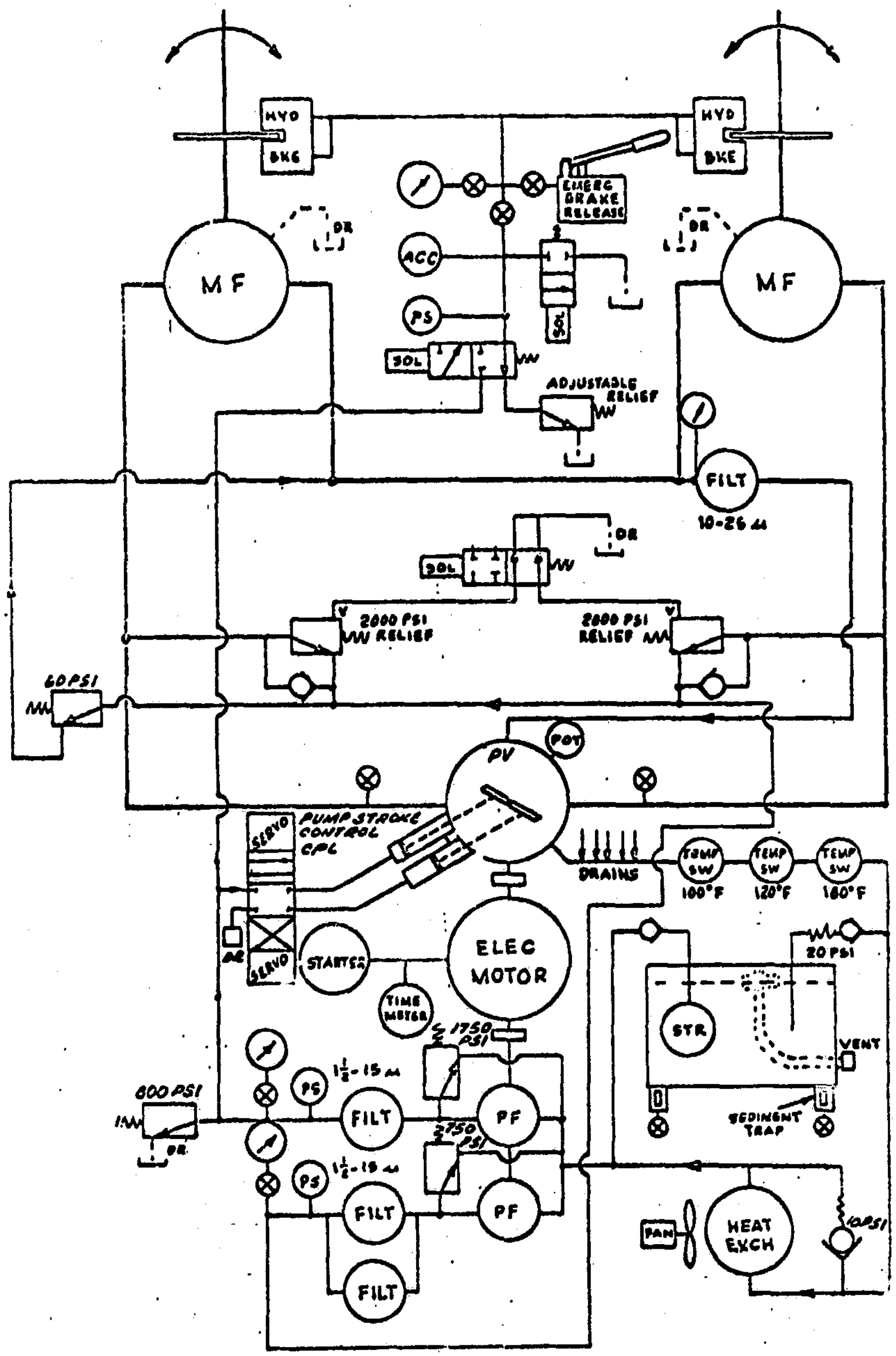

Figure I-1. Hydraulic Subsystem, Block Diagram antenna is dependent upon antenna wind loading, unbalance, rate of acceleration, and frictional forces within reduction gears and the hydraulic motors. The main servo pump is controlled by a servo valve which positions the yoke of the servo pump to vary the displacement and also to change the direction of hydraulic oil flow through the system. As the yoke is moved away from the center position, the flow begins. The direction of hydraulic oil flow depends upon which side of center the yoke is moved. The hydraulic oil flow increases to a maximum as the pump yoke moves away from the center position.

A constant-displacement, dual-vane hydraulic pump is mechanically coupled to an electric prime mover. This dual-vane pump has a hydraulic oil flow rating of five gallons per minute in the forward vane and two gallons per minute in the rear vane. The forward vane has a maximum pressure rating of 1500 pounds per square inch, and the rear vane is rated at 2000 pounds per square inch maximum. Discharge pressure from the dual-vane pump is controlled by relief valves. In the rear vane, a two gallon per minute flow of hydraulic oil at a pressure of 800 pounds per square inch supplied hydraulic oil to the servo valve that control the servo pump. The rear vane also supplies the necessary hydraulic pressure to release the antenna brakes. In the forward vane, a five gallon per minute supply of hydraulic oil, at a pressure of 60 pounds per square inch, furnishes replenishing oil for the servo pump and back pressure for the antibacklash function. Adjustable pressure switches in the front and rear vanes monitor the dual-vane pump output. If the pressure drops below a safe reading these pressure switches will open, and shut down the hydraulic drive power unit.

The hydraulic motors are a fixed displacement type. The shaft, pistons, piston rods, and cylinder block assemblies revolve as an integral unit which is referred to as the rotating group. The valve plate has two curved ports, each extending nearly 180° and separated by solid areas. When hydraulic pressure is applied to one of the valve plate ports, it is felt by the pistons joined to that port through the cylinder block and the pistons are forced to move away from the valve plate, causing the entire rotating group to rotate. The hydraulic oil in the cylinder bores is discharged into the low-pressure port of the valve plate. If the application of the high and low pressures is reversed at the ports, the rotating group rotates in the opposite direction. Since these hydraulic motors are set at a fixed 30° angle, the speed of the motor shaft is determined by the flow rate of the hydraulic oil supplied to the valve plate.

An antibacklash torque is built into the hydraulic drive system by the series method of connecting the hydraulic motors. When one motor is the driving motor, using the main servo pump discharge pressure, the other motor is kept in a condition of trying to drive in the opposite direction. The second hydraulic motor tries to drive in the opposite direction because of the difference in the pressure in the line between the two hydraulic motors and the pressure supplied by the replenishment vane of the dual-vane pump. When one motor is driving in one direction, the second motor is actually driven backward by the bull gear on the respective axis. The purpose of the antibacklash system is to remove backlash from the gear train of the hydraulic motor that is driven by the bull gear. When the servo control system calls for a reversal of direction, the nondriving or driven hydraulic motor is immediately in position to drive the antenna.

Major Equipment

COMPONENT NAME	MANUFACTURER	MANUFACTURER'S PART NUMBER	RELATED TECHNICAL PUBLICATION
Hydraulic Reservoir	Collins Radio Co.	529-0623-005	NA
Dual Vane Hydraulic Pump	Vickers, Inc. Detroit, Mich.	V2235-5-2-1AA-20-S214-MOD	Vickers Vane Type Double Pump I-1886S
Fluid Pressure Filter (with switch)	Aircraft Porous Media, Inc. Glen Cove, N. Y.	AC-3255-12ZABS	NA
Pressure Switch S3	Static "O" Ring Pressure Switch Co. Kansas City, Mo.	9N-F5	Static "O" Ring Pressure Switch General Instruction
Relief Valve	Vickers, Inc. Detroit, Mich.	CG-03-C-10 MOD	Vickers High Pressure Relief Valve, I-1585S
Fluid Pressure Filter (without switch)	Aircraft Porous Media, Inc. Glen Cove, N. Y.	AC-3255-12ZABN	NA
Pressure Switch S2	Static "O" Ring Pressure Switch Co. Kansas City, Mo.	5N-F3	Static "O" Ring Pressure Switch General Instruction
Relief Valve	Vickers, Inc. Detroit, Mich.	CG-03-B-10 MOD	Vickers High Pressure Relief Valves, I-1585S
Relief Valve	Denison Eng. Div., American Brake Shoe Co. Columbus, Ohio	R2V123Ys	Denison R2V Relief Valves, Bulletin SUR4
Yoke Potentiometer and position indicator	Denison Eng. Div. American Brake Shoe Co. Columbus, Ohio	035-28978	Denison Axial Piston Pump, 700 Series

Major Equipment (Cont'd)

COMPONENT NAME	MANUFACTURER	MANUFACTURER'S PART NUMBER	RELATED TECHNICAL PUBLICATION
Hydraulic Servo Pump	Denison Eng. Div. Am. Brake Shoe Co. Columbus, Ohio	PV07XS-020-51N-YZO-OYW	Denison Axial Piston Pump 700 Series
Servo Control Valve	Aerospace Div. Am. Brake Shoe Co. Oxnard, Calif.	410-1158	Model 410 Servo-Valve Bulletin
Solenoid Valve	Denison Eng. Div. Am. Brake Shoe Co. Columbus, Ohio	DID04-33-103-oW 04-1008	Denison 4-Way Valves, Bulletin SD-4178
AC Motor	Reuland Electric Co. Alhambra, Calif.	324U	Reuland Hydraulic Pump Motor
Brake Actuator Unit	Collins Radio Co.	529-0781-001	NA
Hydraulic Motors	Vickers, Inc. Detroit, Mich.	L3-MFS-2003-30-15-21	Vickers Drawing E830970
Heat Exchanger Assembly	Collins Radio Co.	529-0709-001	NA

Technical Data

ITEM	CHARACTERISTIC	DESCRIPTION
Electric Motors		
Prime Mover	Power Requirements	220/440 volts, 3 phase, 60 cycles, 15 hp at 1200 rpm
	Duty Cycle	Continuous
Heat Exchanger Fan	Power Requirements	220/440 volts, 3 phase, 60 cycle, 1/4 hp 1725 rpm

Technical Data (Cont'd)

ITEM	CHARACTERISTIC	DESCRIPTION
Servo Pump	Volume Capacity	20 gpm at 1200 rpm
	Output Pressure	5000 psi maximum
	Volume Control	Feedback potentiometer, servo valve, and stroking pistons
Dual Vane Pump	Rotation	Clockwise as viewed from driveshaft end
	Pump Speed	1200 rpm, 600 rpm minimum drive speed
	Maximum Pressure	
	Control Vane	2000 psi
	Replenishment	1500 psi
	Flow Rating	
	Control Vane	2 gpm
	Replenishment Vane	5 gpm
Servo Valve	Rated Flow	1 gpm with 1000 psi drop across the valve
	Operating Pressure	3000 psi maximum
	Coil Resistance	1000 ohms
Hydraulic Motors	Speed	3600 rpm maximum, 50 rpm minimum
	Operating Pressure	3000 psi maximum

Technical Data (Cont'd)

ITEM	CHARACTERISTIC	DESCRIPTION
Filters	Rated Pressure	3000 psi
	Proof Pressure	4500 psi
	Burst Pressure	7500 psi
	Flow Capacity	16 gpm
	Maximum Particle Passed	15 microns diameter
	Operating Temperature Range	-65° to $+275^\circ$F
Hydraulic Drive System	Operating Temperature	40°F to 120°F ambient
	Relative Humidity	0 to 100%
	Sand and Dust	Desert dust storm conditions
	Rain	Up to 4" per hour
	Wind	Up to 140 mph
	Salt Atmosphere	As encountered at sea
	Operating Altitude	0 to 7000 feet

Appendix II

COMPUTER PROGRAMS

GSFC23

```
10 REM  THIS PROGRAM COMPUTES THE INSTANTANEOUS PRESSURE DELIVERED
11 REM  BY A 7 PISTON PUMP TO AN ORIFICE LOAD AS THE INPUT SHAFT
12 REM  ROTATES.  THE PRESSURES MAY BE PRINTED OUT OR OMITTED
13 REM  AT THE OPERATOR'S OPTION.  A FOURIER ANALYSIS IS MADE
14 REM  OF THE RESULTING PRESSURE WAVE TO DETERMINE THE AMPLITUDE
15 REM  OF ITS SPECTRAL COMPONENTS.
50 PRINT "TYPE 1 IF PRESSURES ARE REQUIRED, OTHERWISE 2";
55 INPUT D
90 LET T1=0
100 REM  R2 IS RADIUS FROM CL TO CYLINDER CENTER
110 LET R2=1.5
120 LET P=3.14159265
130 PRINT "ENTER PERCENT STROKE, NOMINAL PRESSURE";
140 INPUT S1,P1
150 LET S1=S1/100
154 PRINT
155 IF D=2 THEN 160
156 PRINT " M       ";"PRESS."
157 PRINT
160 REM  FLOW IN CU. IN./SEC.
170 LET Q0=20*231*S1/60
180 REM  CALC. A0, LOADING ORIFICE AREA IN SQ. IN.
190 REM  CALC. HEAD IN IN.
200 LET P1=P1*1728/(62.4*.85)
220 LET A0=Q0/(.65*SQR(2*386*P1))
230 REM  CALCULATE STROKE AT MAX FLOW
240 LET S2=20*231/(7*.785*1200)
250 REM  CALC. ACTUAL STROKE
260 LET S2=S2*S1
270 REM  CALC. YOKE ANGLE
280 LET B2=ATN(S2/(2*R2))
290 LET Q3=0
300 LET I=14*7
310 DIM S(98)
320 FOR M=1 TO I
330 REM  ANGULAR INCREMENT
340 LET T=2*P/I
350 LET A=T*M
360 LET Q2=0
370 FOR K=1 TO 7
380 LET B=A+2*(K-1)*P/7
390 IF B>P THEN 410
400 GOTO 470
410 IF B<2*P THEN 680
420 IF B>3*P THEN 440
430 GOTO 470
440 IF B<4*P THEN 680
450 REM  B IS THE ANGLE OF THE K'TH CYLINDER
460 REM  V IS RATE OF CHANGE OF PISTON POSN. WITH ANGLE
470 LET V=R2*SIN(B)*SIN(B2)*(1-(COS(B))↑2*(SIN(B2))↑2)↑(-1.5)
```

GSFC23 CONTINUED

```
480 REM  L IS LENGTH OF PISTON IN CYL.
490 LET L=R2*COS(B)*SIN(B2)/SQR(1-(COS(B))↑2*(SIN(B2))↑2)
500 LET L=2.5-L
510 REM  CALC. LEAKAGE
520 REM  B9 IS CLEARANCE IN INCHES
530 IF K=1 THEN 560
540 LET B9=1E-3
550 GOTO 580
560 LET B9=1.4E-3
570 REM  L1 IS LEAKAGE IN CU. IN./MIN.
580 LFT L8=3.132E6*P1*B9↑3*12/(40*L)
590 LET L9=360*V*2*P*1200/60*1/12*B9
600 LET L1=P*1*(L8-L9)
610 REM  CALC. LEAKAGE PER INTERVAL T
620 LET L1=L1*T/(2*P*1200)
630 REM  Q1 IS FLOW OF GIVEN PISTON AND CYL.
640 LET Q1=.785*V*T
650 LET Q1=Q1-L1
660 REM  Q2 IS TOTAL FLOW IN ANG. INCR., CU. IN.
670 LET Q2=Q2+Q1
680 NEXT K
690 REM  CALC. FLOW RATE, CU. IN./SEC.
700 LET V2=Q2*2*P/T [illegible]0
710 REM  CALC. PRESS. HEAD, IN.
720 LET S=1/(2*386)*(V2/(.65*A0))↑2
730 LET S(M)=S*62.4*.85/1728
736 LET T1=T1+S(M)
737 IF D=2 THEN 740
739 PRINT M;S(M)
740 NEXT M
741 LET S(0)=S(98)
742 PRINT "AVG. PRESSURE=";T1/I;"PSI"
750 REM  CALC. FOURIER SERIES
755 PRINT
756 PRINT "HARMONIC","FREQUENCY","DB RE 1 PSI"
760 LET M=I
770 FOR N=1 TO 14
780 LET A1=0
790 LET B1=0
800 FOR K=1 TO I
810 LET Y=(S(K)+S(K-1))/2
820 LET R=(K-.5)*2*P*N/M
830 LET A=Y*SIN(R)
835 LET A1=A1+A
840 LET B=Y*COS(R)
845 LET B1=B1+B
850 NEXT K
860 LET A1=A1*2/M
870 LET B1=B1*2/M
880 LET C=SQR(A1↑2+B1↑2)
890 LET C1=20/LOG(10)*LOG(C)
895 LET C2=INT(C1+.5)
900 PRINT N,20*N,C2
910 NEXT N
920 END
```

```
50 REM  THIS PRØGRAM CØMPUTES THE WAVEFØRM ØF THE ØUTPUT ØF A
51 REM  VANE PUMP ØF BALANCED DESIGN.  THE PRØGRAM ASSUMES 12 VANES.
52 REM  A FØURIER ANALYSIS ØF THE WAVEFØRM IS MADE TØ DETERMINE ITS
53 REM  SPECTRAL CØMPØNENTS.  LEAKAGE ØF ØNE ØF THE VANES MAY BE SIM-
54 REM  ULATED.  WHEN THE LEAKY VANE IS IN SUCH A PØSITIØN AS TØ
55 REM  ALLØW A DISCHARGE PØRT TØ CØMMUNICATE WITH AN INLET PØRT
56 REM  THE AVAILABLE FLØW IS DECREASED BY AN ARBITRARY AMØUNT WHICH
57 REM  MAY BE CHØSEN.
90 PRINT "HARMØNIC","RMS VALUE","DB RE 1E-6"
96 REM  THE PRØGRAM ASSUMES A VANE LENGTH(NØRMAL TØ THE RADIUS) ØF 1.
98 REM  H IS VANE THICKNESS
100 LET H=.1
105 REM  N IS THE NUMBER ØF INCREMENTS INTØ WHICH ØNE SHAFT RØTATIØN
106 REM  IS TØ BE DIVIDED.
110 LET N=120
115 REM  SAVE STØRAGE SPACES IN THE CØMPUTER MEMØRY FØR THE N VALUES
116 REM  ØF PUMP ØUTPUT TØ BE CALCULATED.
120 DIM Q(120)
125 REM  THE ØUTER CASING ØF THE PUMP APPRØXIMATES AN ELLIPSE.
126 REM  2*R1 AND 2*R2 ARE THE MINØR AND MAJØR AXES.  R1 IS ALSØ TAKEN
127 REM  TØ BE THE RØTØR RADIUS.
130 LET R1=2
140 LET R2=2.5
150 LET P=3.1415927
158 REM  Q IS THE VØL. DELIVERED AT ØNE EXIT PØRT PER REV, NEGLECTING
159 REM  VANE THICKNESS.
160 LET Q=P*(R2↑2-R1↑2)
163 REM  THERE ARE TWØ DISCHARGE PØRTS.
165 LET Q=2*Q
166 REM  CERTAIN PØRTIØNS ØF THE ØUTER CASING ØF THE PUMP ARE CIRCULAR
167 REM  ARCS.  THE LENGTH ØF THESE ARCS IN RADIANS EXCEEDS THE SPACING
168 REM  BETWEEN VANES BY 2*D.
170 LET D=P/72
180 FØR M=1 TØ N
190 LET Q0=0
200 LET Q2=0
205 REM  CØMPUTE THE FLØW RATE AT EACH ANGULAR INCREMENT.
208 REM  T IS THE ANGLE IN RAD.
210 LET T=(M-1)*2*P/N
215 REM  K IS THE NUMBER ASSIGNED TØ A GIVEN VANE.
220 FØR K=1 TØ 12
228 REM  T1 IS THE ANGLE ØF THE K TH VANE
230 LET T1=T+(K-1)*2*P/12
240 IF T1<(P/12+D) THEN 390
245 REM  THE AVAILABLE FLØW IS DECREASED IF A VANE IS IN THE ARC
246 REM  SUBTENDED BY A DISCHARGE PØRT.
248 REM  IN 250 TØ 280 DETERMINE IF A VANE IS IN THE FIRST DISCHARGE
249 REM  REGIØN.
250 IF T1>(5*P/12-D) THEN 270
260 GØTØ 310
```

GSFC17 CØNTINUED

```
270 IF (T1-2*P)<(P/12+D) THEN 390
280 IF (T1-2*P)>(5*P/12-D) THEN 390
288 REM  T2 IS THE ANGLE FRØM THE CØMPRESSED ELLIPSE USED TØ
289 REM  CALCULATE THE RADIAL WIDTH ØF A VANE IN THE DIS. REGIØN.
290 LET T2=(T1-2*P-P/12-D)*P/(2*(P/3-2*D))
300 GØTØ 320
310 LET T2=(T1-(P/12+D))*P/(2*(P/3-2*D))
320 LET S=SIN(T2)
330 LET C=CØS(T2)
338 REM  R3 IS THE RADIUS.
340 LET R3=R1*R2/(R2*S↑2+R1*C↑2)
345 REM  L IS THE VANE WIDTH.
350 LET L=R3-R1
358 REM  Q1 IS THE VØLUME ØF ØNE VANE.
360 LET Q1=L*H
362 REM  THERE ARE TWØ DISCHARGE PØRTS.
363 LET Q1=2*Q1
370 LET Q1=Q0+Q1
378 REM Q0 IS THE VØLUME ØF ALL VANES IN THE DISCHARGE REGIØNS.
380 LET Q0=Q1
388 REM  N1 IS THE LEAKY VANE.
390 LET N1=6
400 IF K<N1 THEN 480
405 IF K>N1 THEN 480
407 REM  IN 411 TØ 456 DETERMINE IF THE LEAKY VANE IS IN A PØSITIØN
408 REM  TØ ALLØW HIGH AND LØW PRESSURE REGIØNS TØ CØMMUNICATE.
411 LET V=1
412 IF T1<(P/12-D) THEN 426
414 IF T1<(7*P/12-D) THEN 430
416 IF T1<(13*P/12-D) THEN 434
418 IF T1<(19*P/12-D) THEN 438
420 IF T1<2*P THEN 442
425 GØTØ 450
426 IF T1>0 THEN 460
428 GØTØ 450
430 IF T1>(5*P/12+D) THEN 460
432 GØTØ 450
434 IF T1>(11*P/12+D) THEN 460
436 GØTØ 450
438 IF T1>(17*P/12+D) THEN 460
440 GØTØ 450
442 IF T1>(23*P/12+D) THEN 460
450 IF V>1 THEN 480
452 LET T1=T1-2*P
454 LET V=2
456 GØTØ 412
458 REM  L1 IS THE LEAKAGE.
460 LET L1=.01
470 LET Q2=Q*L1
480 NEXT K
```

GSFC17 CONTINUED

```
488 REM  Q(M) IS THE NET FLØW.
490 LET Q(M)=Q-Q0-Q2
510 NEXT M
512 REM  PERFØRM FØURIER ANALYSIS.  FLØW IS SQUARED IN 560 TØ YIELD
513 REM  A QUANTITY PRØPØRTINAL TØ PRESSURE.
515 LET Q(0)=Q(N)
520 LET M=N
522 REM  24 CØEFFICIENTS ARE CØMPUTED.
525 FØR N=1 TØ 24
530 LET A1=0
540 LET B1=0
550 FØR K=1 TØ M
560 LET Y=((Q(K)+Q(K-1))/2)↑2
570 LET F=(K-.5)*2*P*N/M
580 LET A=Y*SIN(F)
590 LET A1=A1+A
600 LET B=Y*CØS(F)
610 LET B1=B1+B
620 NEXT K
630 LET A1=A1*2/M
640 LET B1=B1*2/M
650 LET C=SQR(A1↑2+B1↑2)
655 REM  EXPRESS ANSWER IN DB RELATIVE TØ AN ARBITRARY REFERENCE.
660 LET C1=20/LØG(10)*LØG(C/1E-6)
670 PRINT N,C,INT(C1+.5)
680 NEXT N
690 END
```

GSFC05

```
10 REM  THIS PROGRAM CALCULATES THE TIME REQUIRED FOR THE SOLENOID
11 REM  VALVE TO ACTUATE AFTER BEING DE-ENERGIZED.  THE PRESSURE IS
12 REM  ASSUMED TO BE 800 PSI.
90 PRINT "SECS.","LB. FORCE","POSN., IN."
100 LET X=.15
110 LET R=.85*62.4/(1728*386)
120 LET V0=0
140 LET A0=0
150 LET Q=.0005
160 FOR T=Q TO .1 STEP Q
360 REM  SPRING FORCE
370 LET F=-30*X-4.75
372 REM  FRICTION FORCE
373 LET F=F+2.05
374 REM  BERNOULLI FORCE
375 LET F=F+2*2.92E-3*231/60*SQR(800)
376 REM  DRAG FORCE
377 LET F=F+1*1.23*.5*R*V0↑2
380 LET A=F*386/.325
400 LET V=V0+(A0+A)/2*Q
420 LET X=X+(V0+V)/2*Q
430 LET V0=V
435 LET A0=A
440 PRINT T,F,X
450 IF X<0 THEN 500
480 NEXT T
500 END
```

GSFC19

```
10 REM  THIS PROGRAM CALCULATES THE TIME REQUIRED FOR A 1 QT.
11 REM  ACCUMULATOR TO DISCHARGE THROUGH THE SOLENOID VALVE.
12 REM  XO IS THE POSITION OF THE SPOOL AT WHICH THE VALVE STARTS
13 REM  TO OPEN.  IN THE PROGRAM THIS IS ALLOWED TO ASSUME A RANGE
14 REM  OF VALUES, BUT IN THE ACTUAL VALVE IT IS ABOUT .08 INCHES.
90 PRINT "XO","MILLISECS."
100 FOR XO=0 TO .15 STEP .01
110 LET P=1500
114 REM  V9 IS THE ACCUMULATOR VOLUME.
115 LET V9=231/4
117 REM  P1 IS THE ACCUMULATOR INITIAL CHARGE.
118 LET P1=1200
119 REM  VO IS THE INITIAL VOLUME OF FLUID IN THE ACCUMULATOR.
120 LET VO=V9-V9*P1/P
129 REM  V IS THE INSTANTANEOUS FLUID VOLUME IN THE ACCUMULATOR.
130 LET V=VO
140 LET J=.001
145 LET D=3/8
150 LET C3=.6
154 LET R=.85*62.4/(1728*386)
155 LET C1=6.28*D*C3↑2*COS(69*3.14/180)
156 LET C2=3.14*D*C3*SQR(2/R)
159 REM  T IS TIME IN SECONDS.
160 FOR T=0 TO 1 STEP J
170 LET Y=(30*XO+2.70)/(2*P*C1+30)
175 LET Q=C2*Y*SQR(P)
179 REM  Q IS THE FLOW THROUGH EACH HALF OF THE VALVE.
205 IF T=0 THEN 220
210 LET V=V-J*Q*2
212 IF V<0 THEN 300
219 REM  P IS NOW THE PRESSURE APPLIED TO THE FLUID SYSTEM BY THE ACC.
220 LET P=P1*V9/(V9-V)
250 NEXT T
300 PRINT XO,T*1000
320 NEXT XO
400 END
```

```
GSFC18     16:53    NØV 22,1967

10 REM  THIS PRØGRAM CØMPUTES THE RESPØNSE ØF THE SERVØ VALVE TØ A
11 REM  UNIT STEP FUNCTIØN INPUT AT T=0 BASED ØN THE PUBLISHED
12 REM  AMPLITUDE AND PHASE CHARACTERISTICS.
13 REM  W IS ANGULAR FREQUENCY IN RAD./SEC.
90 PRINT "SECØNDS","AMPLITUDE"
150 FØR T=0 TØ .01 STEP .00025
152 LET P=3.1416
155 LET G1=0
157 REM  WO IS THE HIGHEST FREQUENCY FØR WHICH THE VALVE CHARACTERISTICS
158 REM  ARE GIVEN.
159 LET W0=2*P*500
160 LET W1=W0/100
180 FØR W=0 TØ WO STEP W1
188 REM  W2 IS THE AVERAGE FREQUENCY IN EACH STEP ØF THE INTEGRATIØN
189 REM  WHICH FØLLØWS.
190 LET W2=W+W1/2
198 REM  THE FØLLØWING SUBRØUTINE CALCULATES THE AMPLITUDE RESPØNSE
199 REM  AT W2 FRØM A STRAIGHT LINE SEGMENT APPRØXIMATIØN.
200 GØSUB 1000
218 REM  THE FØLLØWING SUBRØUTINE CØMPUTES THE PHASE AT W2 FRØM A
219 REM  STRAIGHT LINE SEGMENT APPRØXIMATIØN.
220 GØSUB 1500
225 REM  INTEGRATIØN STARTS.
230 LET X=W2*T-A1
300 LET G=G1
320 LET G1=Y*SIN(X)/W2*W1
340 LET G1=G1+G
400 NEXT W
450 REM  END ØF INTEGRATIØN.
500 LET S=.5+G1/3.1416
600 PRINT T,S
650 NEXT T
660 GØTØ 2000
1000 LET F=W2/(2*P)
1001 LET Y=.602771-9.55414E-5*F+.0025*ABS(F-30)-7.23684E-3*ABS(F-50)
1002 LET Y=Y+3.25725E-3*ABS(F-107)+1.00858E-3*ABS(F-205)
1003 LET Y=Y+3.75473E-4*ABS(F-343)
1010 RETURN
1500 LET A1=.575*F+.95*ABS(F-50)-.934211*ABS(F-70)
1501 LET A1=A1-.145577*ABS(F-165)+.104787*ABS(F-400)
1505 LET A1=A1*P/180
1510 RETURN
2000 END
```

GSFC14

```
50 REM  THIS PROGRAM COMPUTES THE PRESSURE DELIVERED TO AN ORIFICE
51 REM  LOAD BY A PRESSURE RELIEF VALVE AS A FUNCTION OF TIME.  AN
52 REM  INITIAL POSITION FOR THE BYPASS VALVE IS ASSUMED. THE FLOW
53 REM  INTO THE VALVE IS ASSUMED CONSTANT.
110 REM  H1 IS THE HEAD LOSS IN THE CAPILLARY.
120 LET H1=0
125 REM  R IS THE FLUID DENSITY.
130 LET R=.85*62.4
140 REM  Q4 IS THE RATED FLOW INTO THE VALVE AND LOAD.
150 LET Q4=.007
160 REM  Q7 IS THE FLOW BYPASSED FROM THE LOAD BY THE REGULATOR.
170 LET Q7=0
180 REM  Q0 IS THE FLOW TO THE LOAD.
190 LET Q0=Q4-Q7
200 REM  R4 IS THE RADIUS OF THE ORIFICE LOAD.
210 LET R4=.03
220 LET R4=R4/12
230 REM  K5 IS THE SPRING CONSTANT OF THE MAIN VALVE ASSY.
240 LET K5=100
250 LET K5=K5*12
260 REM  X8 IS THE INITIAL COMPRESSION IN K5
270 LET X8=1
280 LET X8=X8/12
290 REM  R7 IS THE SEAT RADIUS IN THE MAIN VALVE ASSY.
300 LET R7=.5
310 LET R7=R7/12
320 REM  M IS THE PISTON WEIGHT.
330 LET M=.3
340 LET M=M/32.2
350 REM  R5 AND R6 ARE THE PISTON OUTER AND INNER RADII.
360 LET R5=.5
370 LET R6=.2
380 LET P=3.1416
390 LET A5=P*(R5↑2-R6↑2)/144
400 REM  T9 IS THE PLUNGER HALF-ANGLE IN THE MAIN VALVE ASSY.
410 LET T9=P/6
420 REM  R1 IS THE BALL RADIUS IN THE PILOT VALVE ASSY.
430 LET R1=.12
440 LET R1=R1/12
450 REM  R2 IS THE SEAT RADIUS OF THE BALL VALVE.
460 LET R2=.1
470 LET R2=R2/12
480 REM  R3 IS THE CAPILLARY RADIUS IN THE PILOT VALVE ASSY.
490 LET R3=.005
500 LET R3=R3/12
510 REM  L IS THE CAPILLARY LENGTH.
520 LET L=.5
530 LET L=L/12
540 REM  K IS THE SPRING CONSTANT OF THE SPRING IN THE BALL VALVE ASSY.
550 LET K=300
```

```
560 LET K=K*12
570 REM  XO IS THE INITIAL COMPRESSION IN K
580 LET X0=.04
590 LET X0=X0/12
600 REM  S IS POSITION OF THE PLUNGER IN THE MAIN VALVE ASSY.
610 LET S=0
620 REM  SO IS THE INITIAL VALUE OF S.
630 LET S0=0
640 REM  V IS THE PLUNGER VELOCITY.
650 LET V=0
660 REM  VO IS THE INITIAL VALUE OF V.
670 LET V0=0
680 REM  AO IS INITIAL ACCELERATION.
690 LET A0=0
695 LET P0=X0*K/(P*R2↑2*144)+K5*X8/(A5*144)
696 PRINT "REGULATOR SET FOR" PO "PSIG."
697 PRINT
698 PRINT "MICROSECS.","LINE PRESSURE, PSIG."
700 LET T2=1E-6
710 LET T3=50*1E-6
720 FOR T1=0 TO T3 STEP T2
725 REM  HO IS HEAD LOSS ACROSS LOAD.
730 LET H0=(Q0/(.65*P*R4↑2))↑2/(2*32.2)
740 REM  COMPUTE A VERY ROUGH APPROXIMATION TO THE PILOT FLOW
750 REM  NEGLECTING FRICTION AND BALL VALVE LOSSES.
755 REM  H2 IS HEAD LOSS ACROSS BALL VALVE.
760 LET H2=ABS(H0-H1)
765 REM  Q IS FLOW THROUGH THE CAPILLARY.
770 LET Q=P*R3↑2*SQR(2*32.2*H2/1.5)
810 REM  IN 800 TO 1130 COMPUTE X2, THE BALL DISPL. BASED ON PRESSURE
820 REM  APPLIED TO THE BALL, AND COMPUTE X, THE BALL DISPL. BASED
830 REM  ON ASSUMED PILOT FLOW.  SOLUTION EXISTS WHEN THEY ARE EQUAL.
840 LET Z6=100
850 LET Q1=Q/Z6
860 LET Q3=2*Q
870 FOR Q=Q1 TO Q3 STEP Q1
875 REM  V1 IS VELOCITY IN THE CAPILLARY.
880 LET V1=Q/(P*R3↑2)
890 LET V1=ABS(V1)
900 REM  COMPUTE FRICTION FACTOR IN CAPILLARY.
901 LET N=V1*R3*2*R/(32.2*1.7E-6*144)
902 IF Q=0 THEN 940
903 IF N<1E3 THEN 907
904 IF N>1E5 THEN 909
905 LET F=.3164/N↑.25
906 GOTO 940
907 LET F=64/N
908 GOTO 940
909 LET F=.0032+.221/N↑.237
920 REM  COMPUTE PLUNGER POSITION BY BALANCING FORCES.  POSITION THEN
```

GSFC14 CONTINUED

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930 REM  DETERMINES NEW LINE PRESSURE AND SOLUTION IS REPEATED.
940 LET H1=(1.5+F*L/(2*R3))*V1↑2/(2*32.2)
990 REM F5 IS FORCE ON PISTON DUE TO DROP ACROSS CAPILLARY.
1000 LET F5=A5*H1*R
1005 REM  F6 IS SPRING FORCE ON PISTON.
1010 LET F6=K5*(S+X8)
1015 REM  F7 IS DRAG FORCE ON PISTON.  DRAG COEFF. ASSUMED 1.
1020 LET F7=1*A5*.5*R*V↑2
1030 IF S<S0 THEN 1050
1040 GOTO 1060
1050 LET F7=-F7
1055 REM A6 IS PISTON ACCELERATION.
1060 LET A6=(F5-F6-F7)/M
1065 REM  V IS VELOCITY.
1070 LET V=V0+(A6+A0)/2*T2
1075 REM  S IS DISPLACEMENT.
1080 LET S=S0+(V+V0)/2*T2
1090 IF S<0 THEN 1110
1100 GOTO 1120
1110 LET S=0
1120 LET Q7=.7*2*P*R7*S*COS(T9)*SQR(2*32.2*H0)
1122 REM  Q8 IS FLOW CAUSED BY PISTON UPWARD MOTION.
1125 LET Q8=V*A5
1130 LET Q0=Q4-Q7-Q-Q8
1145 REM  Q2 IS FLOW THROUGH BALL VALVE.
1150 LET Q2=Q+Q8
1155 REM  A2 IS BALL VALVE AREA.
1160 LET A2=Q2/(.65*SQR(2*32.2*(H0-H1)))
1170 LET R8=-A2+SQR(A2↑2+4*P↑2*R1↑2*R2↑2)
1180 LET R8=R8/(2*P*R1)
1190 LET L1=SQR(R1↑2-R8↑2)
1200 LET X1=R2*L1/R8
1205 REM  X IS BALL VALVE OPENING BASED ON FLOW VOLUME.
1210 LET X=X1-SQR(R1↑2-R2↑2)
1215 REM  X2 IS BALL VALVE OPENING BASED ON PRESSURE DROP ACROSS VALVE.
1220 LET X2=(H0-H1)*R*P*R2↑2/K-X0
1230 IF X2<X THEN 1250
1240 NEXT Q
1250 PRINT T1*1E6,H0*R/144
1260 LET V0=V
1270 LET A0=A6
1280 NEXT T1
1400 END
```

```
200DATA144,137,137,244,310,401,489,360,247,147,144,152,148,163,173,
201DATA217,151,160,182,182,140,140,140,137,129,140,133,140,134,133,
202DATA125,137,138,140,137,134,125,183,240,201,190,256,186,155,130,
203DATA139,144,145,152,139,163,170,179,179,125,152,126,133,122,126,
204DATA126,130,129,126,123,122,122,123,126,130,125,168,187,168,240,
205DATA352,205,198,159,144,148,163,148,137,134,138,118,130,123,126,
206DATA129,130,122,122,122,122,118,118,117,117,117,117,122,117,117,
207DATA137,166,152,255,355,198,155,175,147,170,147,117,137,133,125,
208DATA125,117,125,121,117,133,117,125,129,125,113,122,121,121,117,
209DATA122,121,118,117,1[illegible],147,143,151,151,124,140,124,190,182,121,
210DATA121,121,117,125,139,198,147,121,125,121,113,114,113,122,117,
211DATA118,133,118,114,113,114,118,117,137,122,129,199,217,193,139,
212DATA214,163,160,144,117,133,121,126,117,117,113,121,113,117,132,
213DATA148,133,117,125,117,116,117,125,117,112,113,113,118,128,125,
214DATA170,163,129,132,117,129,118,130,117,122,114,198,182,152,140,
215DATA122,118,130,113,122,122,118,114,118,118,115,118,118,114,122,
216DATA114,122,163,171,156,318,222,199,187,144,163,156,133,118,125,
217DATA148,118,118,118,114,114,130,122,118,118,122,115,118,122,122,
218DATA114,118,122,123,114,122,123,160,183,194,276,187,153,153,156,
219DATA139,126,172,153,140,145,141,141,134,126,119,119,122,114,119,
220DATA114,123,126,118,119,114,115,114,119,118,119,160,187,160,214,
221DATA192,149,134,123,134,130,131,133,126,122,115,114,127,118,126,
222DATA119,118,115,119,118,119,115,115,118,123,114,119,115,123,118,
223DATA156,210,161,237,152,138,126,126,134,168,168,123,130,118,134,
224DATA119,118,123,122,113,114,119,115,118,119,114,119,114,119,114,
225DATA119,115,119,119,156,164,160,279,226,175,126,133,134,118,118,
226DATA118,122,118,122,122,122,122,115,114,125,113,114,114,114,118,
227DATA118,114,115,118,122,117,114,114,148,271,191,519,268,245,171,
228DATA123,144,140,138,140,202,202,130,140,167,175,134,138,126,130,
229DATA115,118,118,130,130,114,126,126,130,129,123,119,140,202,256,
230DATA206,241,245,156,133,130,145,140,140,134,134,152,179,164,164,
231DATA141,134,127,125,126,118,123,114,122,118,115,115,114,114,115,
232DATA114,115,176,171,148,318,268,206,171,152,155,148,152,140,126,
233DATA122,122,115,118,118,114,118,118,115,115,119,119,119,114,114,
1000 LET T=5
1010 LET V=102
1040 PRINT
1041 PRINT
1042 PRINT "TEST NUMBER"V
1043 PRINT
1044 PRINT
1050 PRINT " "," ","SIGNAL ANALYSIS"
1051 PRINT
1052 PRINT
1100 DIM A(509),B(12)
1110 MAT READ A
1115 MAT A=(T)*A
1120 LET T=50.6
1125 LET D=0
1130 LET P1=INT(510*T/(62.5*12)+.5)
```

JANE1 CØNTINUED

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1140 FØR K=0 TØ P1
1145 LET C1=0
1150 FØR M=0 TØ 3*P1+5
1170 FØR N=0 TØ 2
1180 LET B=0
1200 IF M=N*P1+K THEN 1220
1210 GØTØ 1230
1220 LET B=1
1230 LET C=A(M)*B+C1
1240 LET C1=C
1245 NEXT N
1250 NEXT M
1260 IF C1<D THEN 1280
1265 LET D=C1
1270 LET M1=K
1280 NEXT K
1295 LET C1=0
1297 LET D=0
1298 LET Y=INT(M1-P1/4)
1299 LET X=INT(Y-P1/2)
1300 FØR N=1 TØ 12
1306 LET X1=X+N*P1
1307 LET Y1=Y+N*P1
1310 FØR M=X1 TØ Y1
1320 LET B=A(M)↑2+C1
1321 LET C1=B
1325 LET D=D+1
1330 NEXT M
1340 NEXT N
1350 LET B=SQR(B/D)
1360 PRINT "RMS VALUE ØF SIGNAL BETWEEN PEAKS="B
1361 PRINT
1367 LET D=0
1368 LET X=INT(M1-P1/2)
1369 LET Y=INT(X+P1)
1370 FØR N=1 TØ 12
1372 LET G=0
1378 LET X1=X+N*P1
1379 LET Y1=Y+N*P1
1380 FØR M=X1 TØ Y1
1390 IF A(M)<G THEN 1410
1400 LET G=A(M)
1410 NEXT M
1415 LET B(N)=G
1420 NEXT N
1430 FØR N=1 TØ 12
1440 IF B(N)<D THEN 1460
1450 LET D=B(N)
1460 NEXT N
1470 PRINT "MAX. PEAK AMPLITUDE="D
```

JANE1 CONTINUED

```
1471 PRINT
1475 LET H=0
1480 FØR N=1 TØ 12
1490 LET H=H+B(N)
1500 NEXT N
1510 LET H=H/12
1520 PRINT "MEAN PEAK AMPLITUDE="H
1521 PRINT
1525 LET S=0
1530 FØR M=0 TØ 509
1540 LET S=S+A(M)↑2
1550 NEXT M
1560 LET S=SQR(S/510)
1570 PRINT "RMS VALUE="S
1571 PRINT
1572 LET U=D/H
1573 PRINT "MAX. PEAK TØ MEAN PEAK RATIØ="U
1574 PRINT
1580 LET Q1=D/S
1590 PRINT "PEAK TØ RMS RATIØ="Q1
1591 PRINT
1600 LET Q2=H/S
1610 PRINT "MEAN PEAK TØ RMS RATIØ="Q2
1611 PRINT
1620 LET Q3=B/S
1630 PRINT "RATIØ ØF SIGNAL BTW. PEAKS TØ RMS AMPL.="Q3
1700 PRINT
1701 PRINT
1702 PRINT " "," ","DISCRIMINANT ANALYSIS"
1710 PRINT
1720 PRINT
1750 LET Z=0
1800 IF B>90 THEN 1820
1810 GØTØ 1830
1820 PRINT "EXCESSIVE SIGNAL AMPLITUDE BETWEEN PEAKS"
1821 PRINT
1825 LET Z=Z+1
1830 IF D>400 THEN 1840
1835 GØTØ 1850
1840 PRINT "EXCESSIVE PEAK AMPLITUDE"
1841 PRINT
1845 LET Z=Z+1
1850 IF H>290 THEN 1870
1860 GØTØ 1880
1870 PRINT "EXCESSIVE MEAN PEAK LEVEL"
1871 PRINT
1875 LET Z=Z+1
1880 IF S>130 THEN 2000
1890 GØTØ 2020
2000 PRINT "EXCESSIVE RMS AMPLITUDE"
```

JANE1 CONTINUED

```
2001 PRINT
2010 LET Z=Z+1
2020 IF U>1.5 THEN 2022
2021 GØTØ 2029
2022 PRINT "EXCESSIVE MAX. PEAK TØ MEAN PEAK RATIØ"
2023 PRINT
2024 LET Z=Z+1
2029 IF Q1>3.5 THEN 2040
2030 GØTØ 2050
2040 PRINT "EXCESSIVE PEAK TØ RMS RATIØ"
2041 PRINT
2045 LET Z=Z+1
2050 IF Q1<2.5 THEN 2070
2060 GØTØ 2080
2070 PRINT "DEFICIENT PEAK TØ RMS RATIØ"
2071 PRINT
2075 LET Z=Z+1
2080 IF Q2<2 THEN 2100
2090 GØTØ 2110
2100 PRINT "DEFICIENT MEAN PEAK TØ RMS RATIØ"
2101 PRINT
2105 LET Z=Z+1
2110 IF Q3>.8 THEN 2130
2120 GØTØ 2400
2130 PRINT "RATIØ ØF SIGNAL BTW. PEAKS TØ RMS AMPL. EXCESSIVE"
2140 LET Z=Z+1
2400 IF Z>0 THEN 2500
2410 PRINT "NØ DISCRIMINANTS ØUT ØF LIMITS."
2500 PRINT
2501 PRINT
2502 PRINT " "," ","RECØMMENDED ACTIØN"
2510 PRINT
2511 PRINT
2520 IF Z=0 THEN 2600
2530 IF Z>3 THEN 2650
2550 GØTØ 2620
2600 PRINT "NØRMAL MAINTENANCE ØNLY."
2601 PRINT
2605 GØTØ 9999
2620 PRINT Z"DISCRIMINANTS ØUT ØF LIMITS.  SCHEDULE INSPECTIØN WITHIN"
2621 PRINT "ØNE MØNTH."
2622 PRINT
2630 GØTØ 9999
2650 PRINT Z"DISCRIMINANTS ØUT ØF LIMITS.  IMMEDIATE ØVERHAUL REQUIRED."
9999 END
```