DEVELOPMENT TEST REPORT
FOR
SATURN V HYDRODYNAMIC SUPPORT

November 1965

ER 14037

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FOREWORD

This report has been prepared by the Baltimore Division of the Martin Company, a Division of the Martin Marietta Corporation and is intended to satisfy the requirements of Paragraph 4.1 of the Saturn V Hydrodynamic Support Equipment Design Criteria (88A4100401), except for those development tests which are part of the Head to Head Test Report.

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CONTENTS (continued)

LIST OF FIGURES

I. AIR-TO-OIL INTERFACE TEST

A. SUMMARY

Nuto 146 hydraulic oil was tested to determine its foaming characteristics resulting from sudden air degassing of the oil. The results of the test show that the amount of air which will go into solution with Nuto 146 under high pressure is small. Foaming as a result of sudden decrease in pressure was found to be small and will not adversely effect the operation of the Saturn V Hydrodynamic Support.

The non-gas absorbing, nonfoaming qualities of this oil are attributed to the addition of foam depressing silicones. [The silicones reduce the air absorption, speed the degassing process, and reduce foaming by reducing the surface tension of the oil.] missed machine terms

B. TEST PROCEDURE AND RESULTS

A transparent cylinder was partially filled with Nuto 146 hydraulic oil. The volume of the cylinder above the level of the oil was charged to 2600 psig using air. The cylinder was then sealed off and allowed to stand for 90 hr. under room ambient conditions. There was no measurable change in air pressure throughout the 90 hr. period. At the end of the 90 hr. period, the cylinder was back-lighted and photographed (Fig. 1). The gas pressure was then suddently released by opening a shutoff valve on the gas side of the cylinder. The back-lighted cylinder was then photographed at 15 sec, 2 min. and 7 min. after the pressure was reduced (Figs. 2, 3 and 4, respectively).

Fifteen seconds after sudden release of air pressure, a thin layer of foam formed on the free surface of the oil. After 2 min at atmospheric pressure the layer of foam had dissipated, leaving only scattered, random size surface bubbles. The number of surface bubbles was less after 7 min than at 2 min.

Examination of the number and size of the bubbles immersed in the oil indicates that the rate of degassing of the oil is significantly less after 7 min. This observation is taken to mean that the oil degasses at a reasonably high rate.

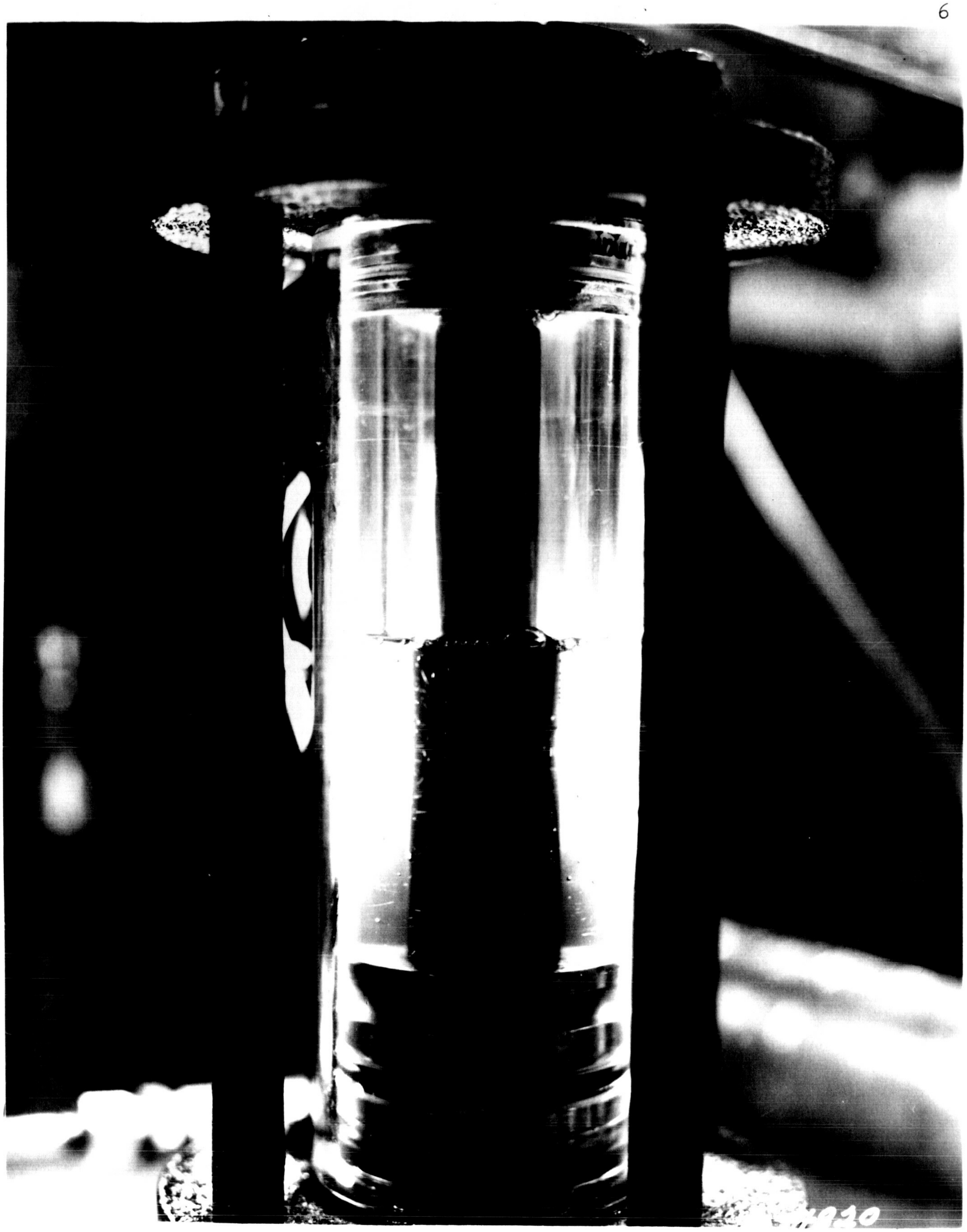
Oil Under Pressure Fig. 1.

15 Seconds After Pressure Release

Fig. 2.

2 Minutes After Pressure Release

Fig. 3.



7 Minutes After Pressure Release   Fig. 4.

II. FLOW CONTROL VALVE DEVELOPMENT TESTS AND MODIFICATIONS

A. SUMMARY

1. Variable Flow Control Valves FCV-1 and FCV-2 (Martin Co. No. 88A4100849-001, Denison No. FC12-23-104)

Static constant flow characteristics of Variable Flow Control Valves FCV-1 and FCV-2 were measured at several flow settings and fluid temperatures. Flow was found to decrease with increasing pressure drop across the valve. Minimum differential pressure drop across the valve for acceptable flow control was found to be 100 psi.

The flow control valve was tested for constant flow characteristics when subjected to sinusoidally varying differential pressures across the valve. The frequency of the differential pressure change was varied from .1 to 20 cps. Flow was found to be frequency dependent and to increase in deviation from set point with increasing frequency.

Three modifications were performed on the valve which resulted in improving the dynamic flow control characteristics to an acceptable level.

2. Flow Control Valves FCV-25 and FCV-26 (Martin Co. No. 88A4100829-003, Manitrol No. 3/8-PCAF-600-2.5)

Flow Control Valves FCV-25 and FCV-26 were tested to determine the minimum pressure drop across the valve for constant flow. The result of the test show that 150 psi is the minimum pressure drop across the valve for acceptable flow control when flow is set at 2.5 gpm. Pressure differentials from 200 to 500 psi were used in setting the flow at 2.5 gpm.

B. TEST PROCEDURE AND RESULTS

1. Variable Flow Control Valves FCV-1 and FCV-2

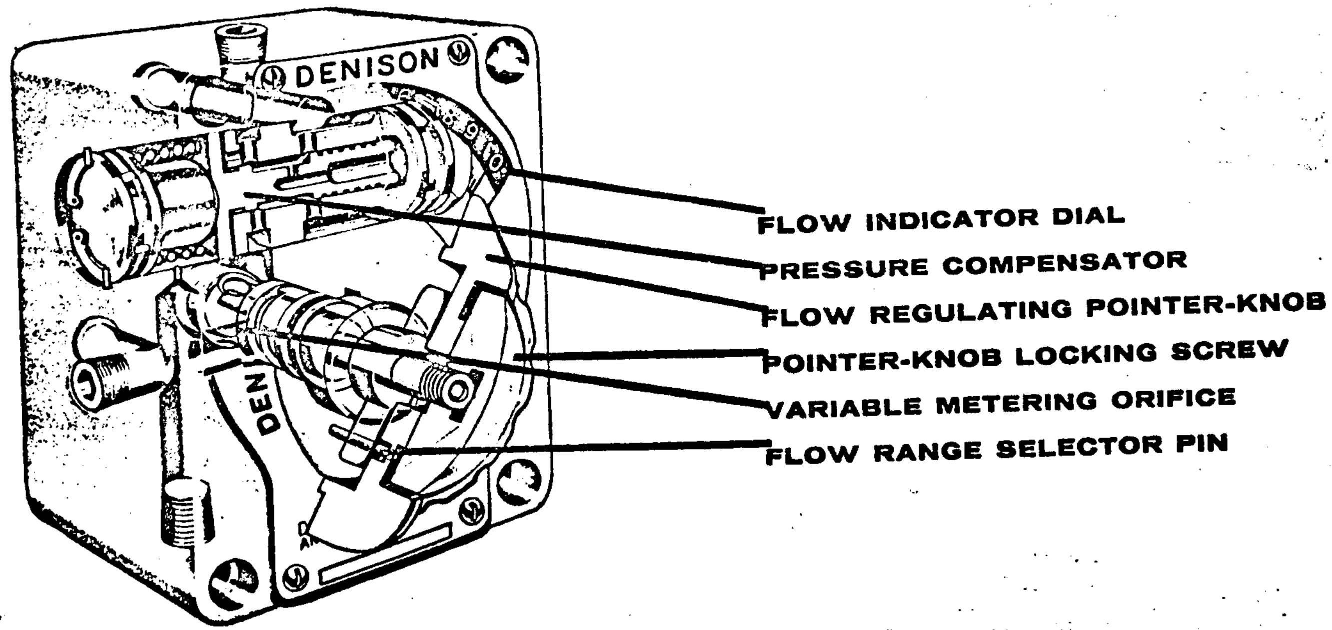
Static Constant Flow Characteristics - The valve was installed for testing in the hydraulic system shown in Fig. 6, which supplied Nuto 146 hydraulic oil at constant pressure and controlled temperature. Oil was supplied at 2900 psi and several constant temperatures. The downstream pressure was varied in steps between 150 psi and 2800 psi. The flow transducer oscillograph trace deflection and the differential pressure were recorded for each step. The results of the test performed at various valve settings, fluid temperatures, and differential pressures is shown in Fig. 7. All data shown was taken after valve modification No. 1. The results indicate that the static flow decreases with increasing pressure differential and that the change in flow is less pronounced at higher fluid temperature (and lower viscosity) than at low temperature. The valve characteristics were not affected by changing the set point from 12 to 6 at range 3. The minimum pressure differential for acceptable flow control was found to be 100 psi.

Dynamic Flow Control Characteristics - The valve was installed for testing in the hydraulic system shown in Fig. 8, which supplied Nuto 146 hydraulic oil at constant pressure and controlled temperatures. The supply pressure to the valve was 2900 psi and downstream static pressure was set at 2400 psi. The sinusoidally varying pressure differential was created by connecting a servo valve in parallel with the back pressure control valve. The servo valve was then driven with a sine wave signal generator. This arrangement made it possible to impose a dynamic pressure fluctuation up to ±120 psi centered around the 2400 psi downstream pressure. Upstream or supply pressure, downstream pressure, and flow were simultaneously and continuously recorded at set dynamic pressure frequencies which ranged from .1 to 20 cps. The ratio of the change in flow through the flow control valve to the change in differential pressure and the phase angle between the change in flow and the change in pressure differential are shown in Figs. 9 and 10 for the stock configuration of the valve and after modifications Nos. 2 and 3. Modifications No. 1, No. 2 and No. 3 are shown in Fig. 11.

The results of the tests show a significant improvement in the response of the valve for maintaining a constant flow under dynamic pressure conditioner. As a result of modification No. 3, a relatively low and constant value of $\Delta Q_3/\Delta P_3$ is maintained up through 1 cps.

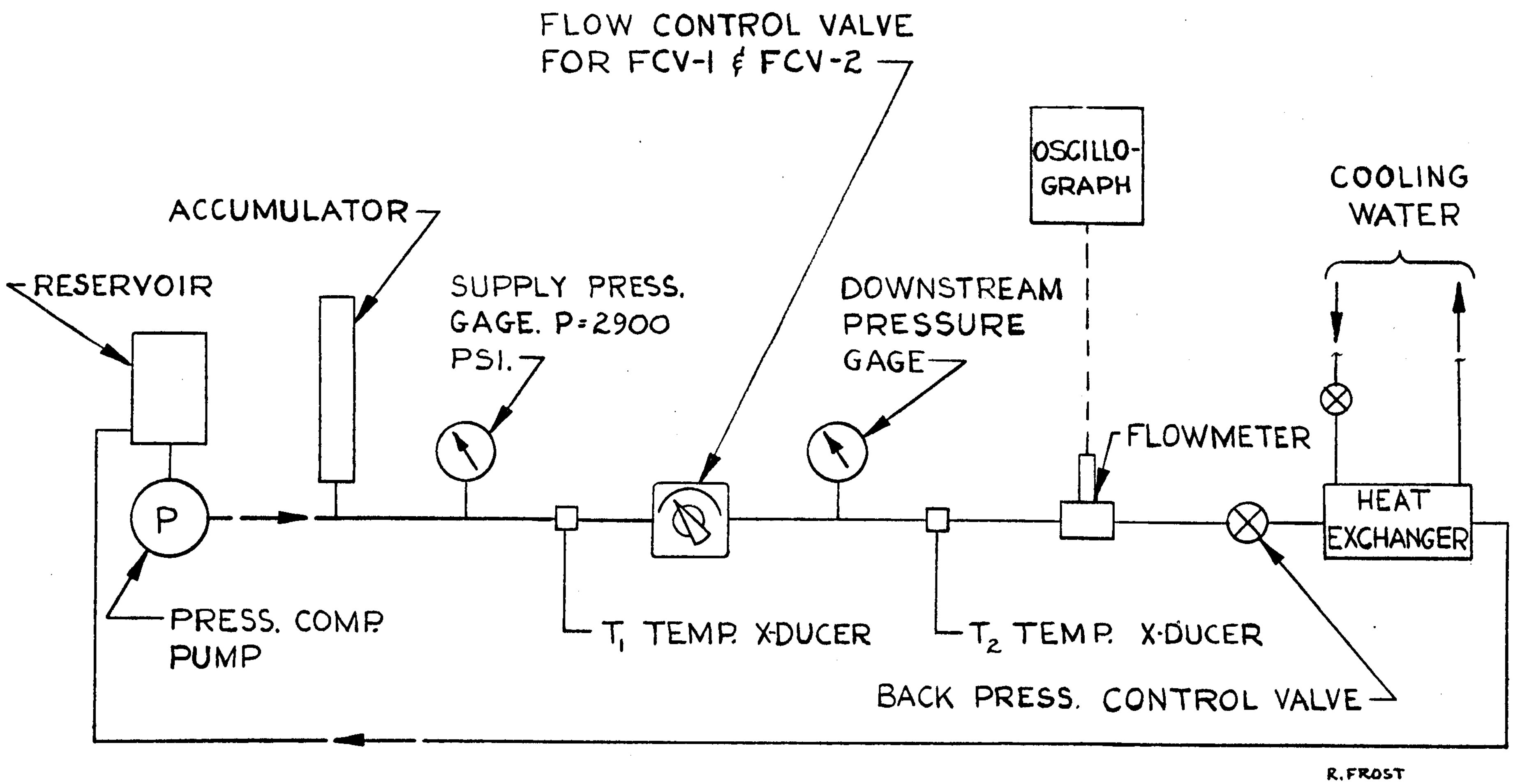
2. Flow Control Valves FCV-25 and FCV-26

Minimum Pressure Drop Test - The valve was installed for testing in a hydraulic system which supplied Nuto 146 hydraulic oil under controlled temperature and pressures. The pressure downstream of the valve was maintained at 2500 $^{+0}_{-25}$ psi. Flow was set at 2.5 gpm with upstream pressures of 3000, 2900, 2800 and 2700 psi for four tests, respectively. For each test, the upstream pressure was reduced in 50 psi increments from the set pressure to 2600 psi and the flow through the valve was measured at each pressure. The results of the test are shown in Fig. 13. The results show that flow control is largely lost at pressure differentials less than 150 psi across the valve.



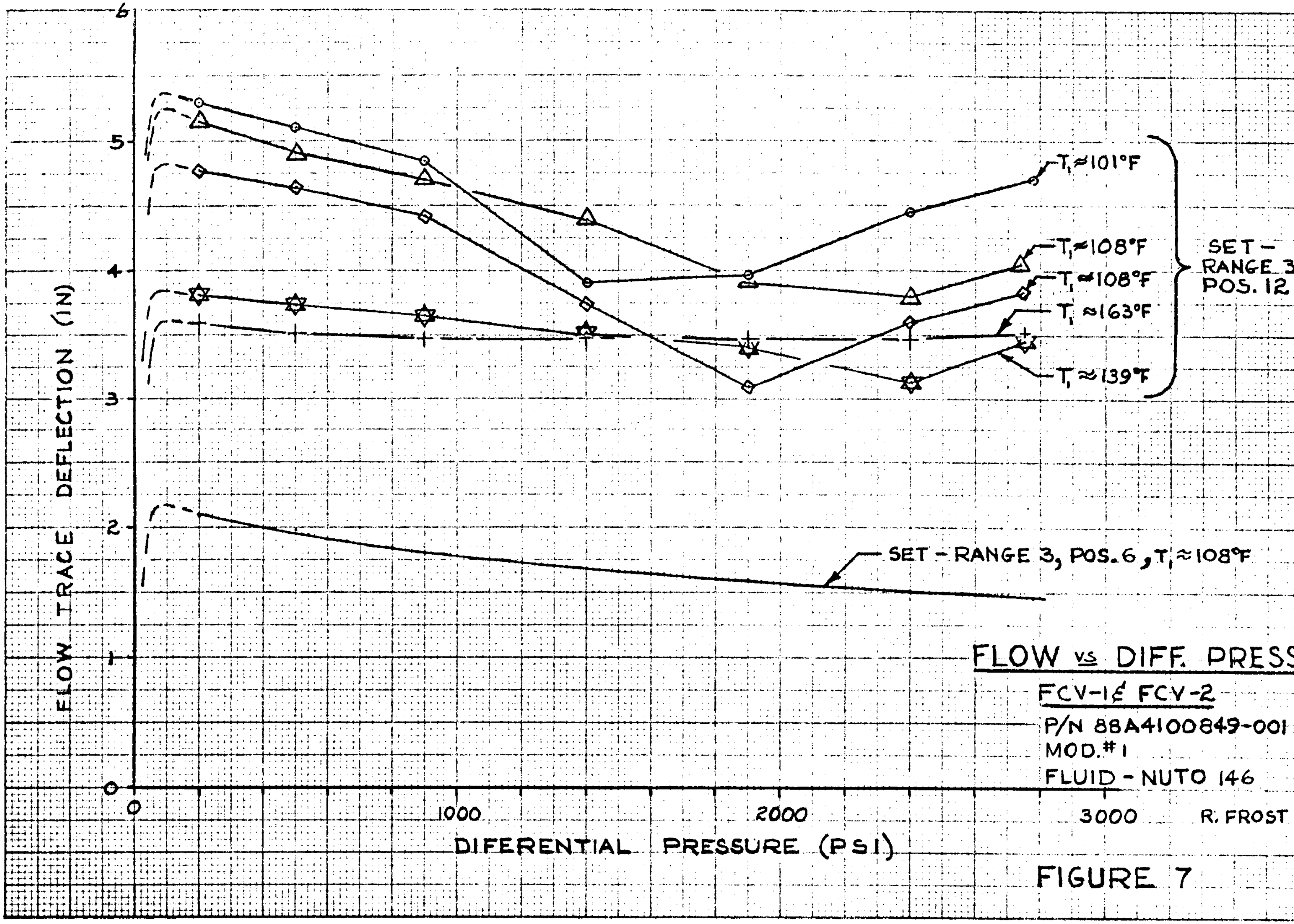
FLOW CONTROL VALVE
FCV-1 & FCV-2
88A4100449-001

FIGURE 5



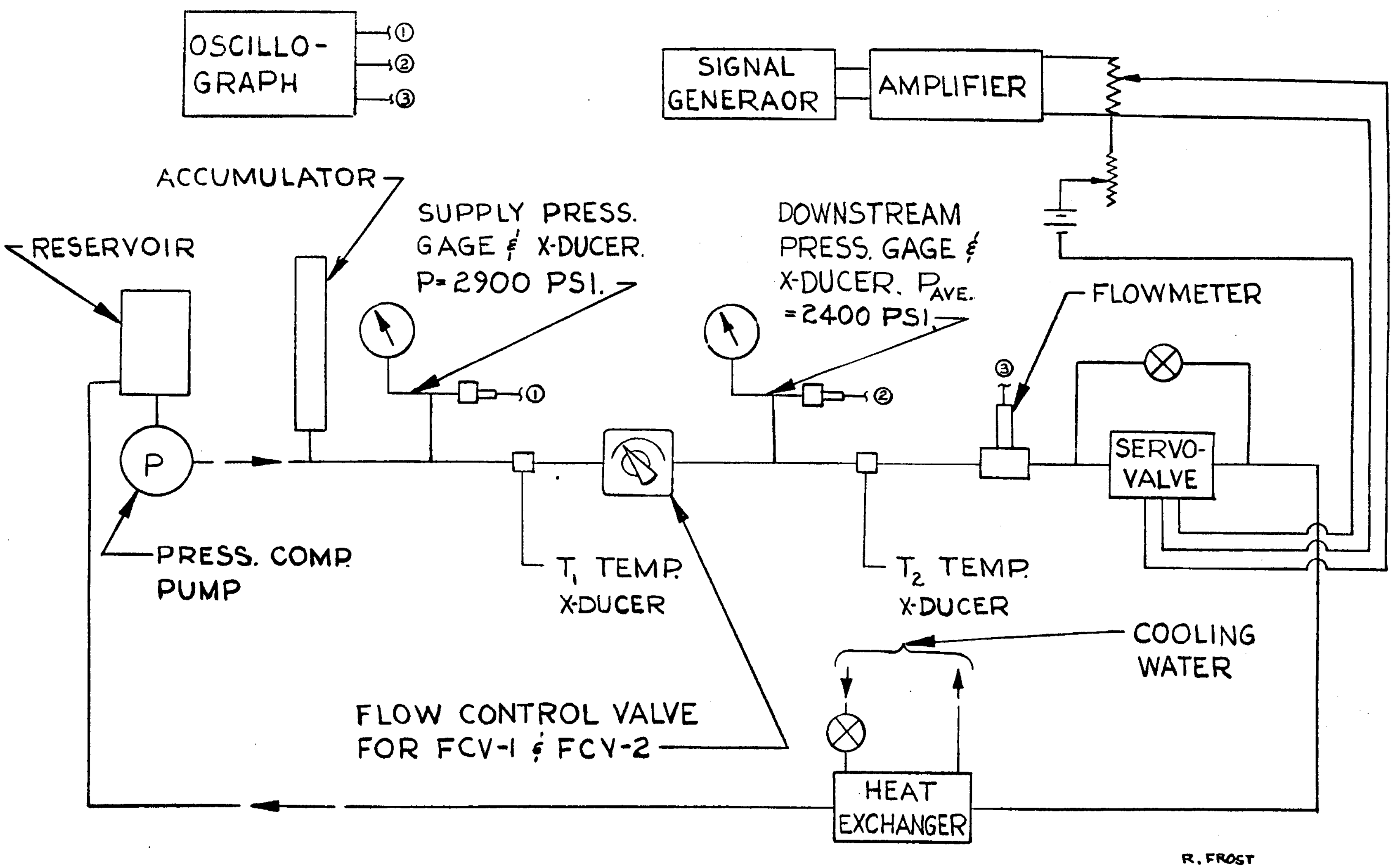
FCV-1 & FCV-2 STATIC CONSTANT FLOW CONTROL TEST SCHEMATIC

FIGURE 6



FLOW TRACE DEFLECTION (IN)
DIFERENTIAL PRESSURE (PSI)
0
1000
2000
3000
1
2
3
4
5
6
$T_1 \approx 101°F$
$T_1 \approx 108°F$
$T_1 \approx 108°F$
$T_1 \approx 163°F$
$T_1 \approx 139°F$
SET - RANGE 3 POS. 12
SET - RANGE 3, POS. 6, $T_1 \approx 108°F$
FLOW vs DIFF. PRESS.
FCV-1 & FCV-2
P/N 88A4100849-001
MOD. #1
FLUID - NUTO 146
R. FROST

FIGURE 7



FCV-1 & FCV-2 DYNAMIC FLOW CONTROL TEST SCHEMATIC

FIGURE 8

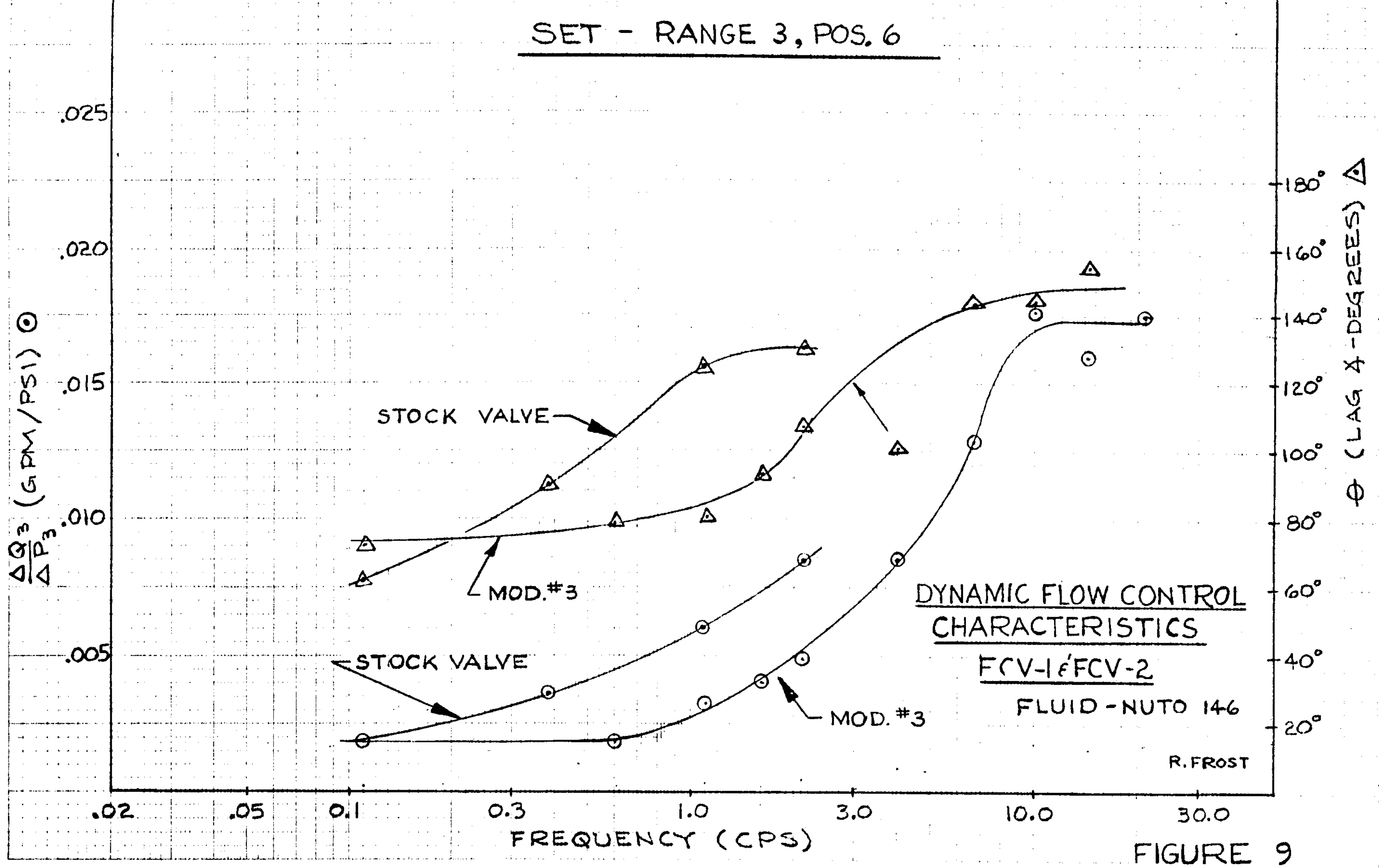

SET - RANGE 3, POS. 6
$\frac{\Delta Q_3}{\Delta P_3}$ (GPM/PSI) ⊙
.025
.020
.015
.010
.005
Φ (LAG ∡-DEGREES) △
180°
160°
140°
120°
100°
80°
60°
40°
20°
STOCK VALVE
MOD. #3
STOCK VALVE
MOD. #3
DYNAMIC FLOW CONTROL
CHARACTERISTICS
FCV-1 & FCV-2
FLUID - NUTO 146
R. FROST
.02
.05
0.1
0.3
1.0
3.0
10.0
30.0
FREQUENCY (CPS)

FIGURE 9

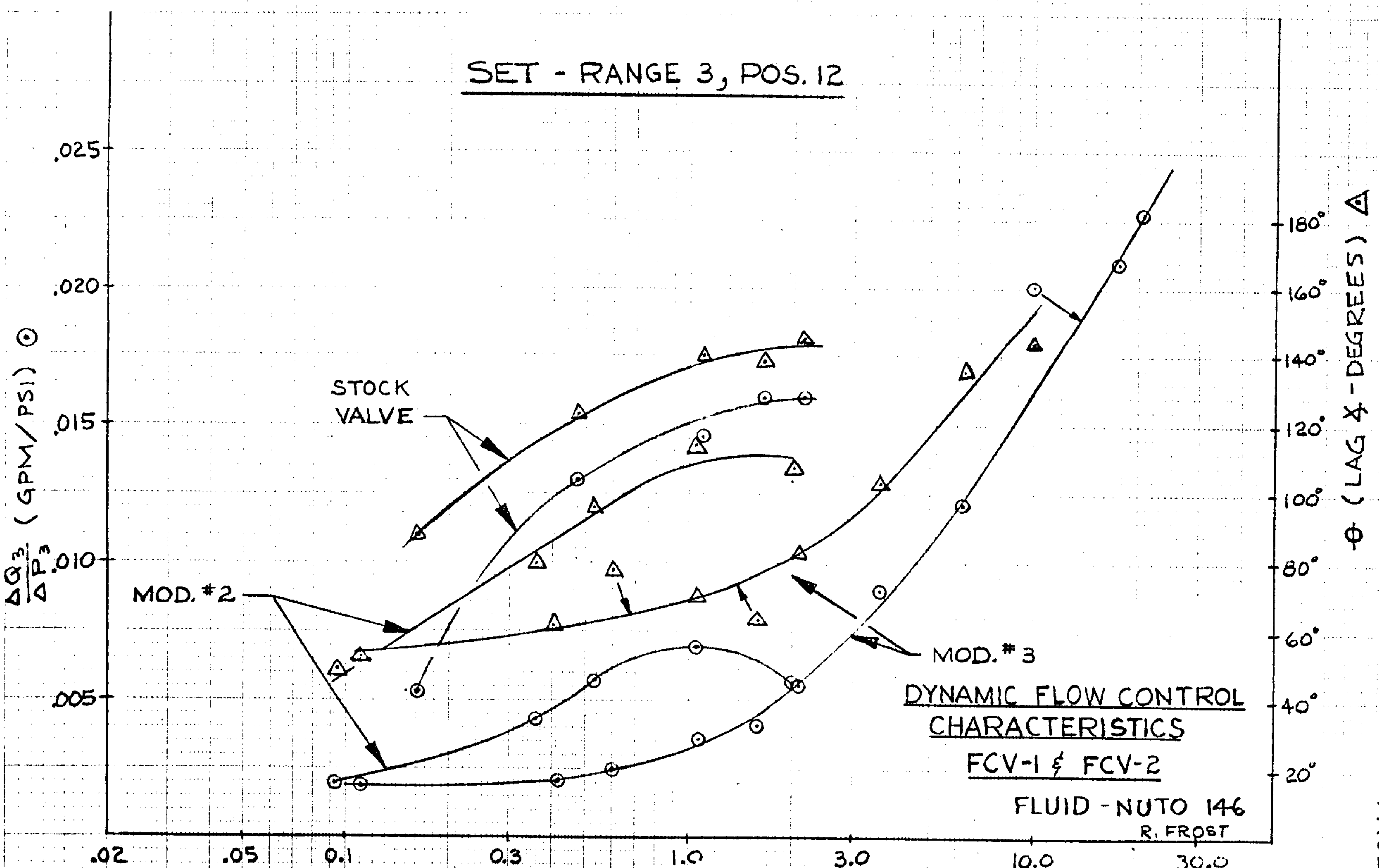
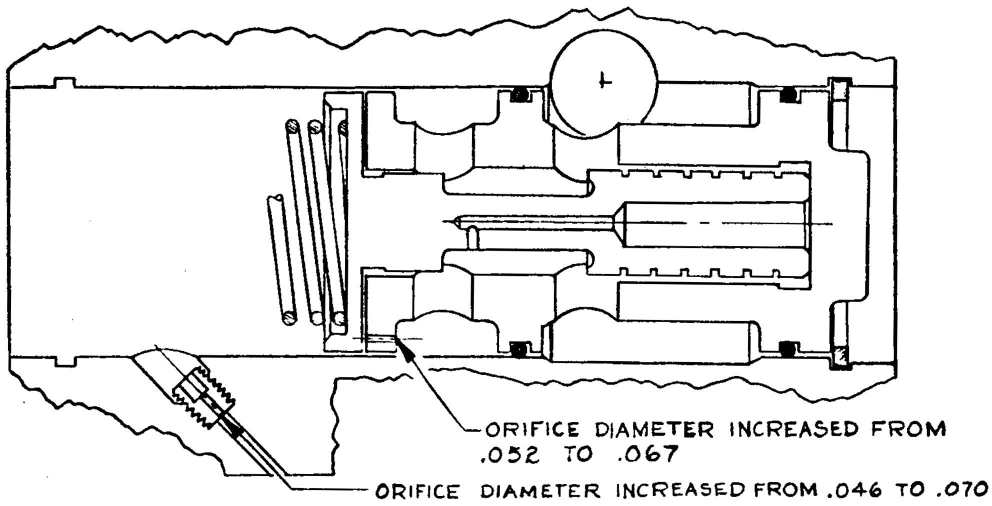
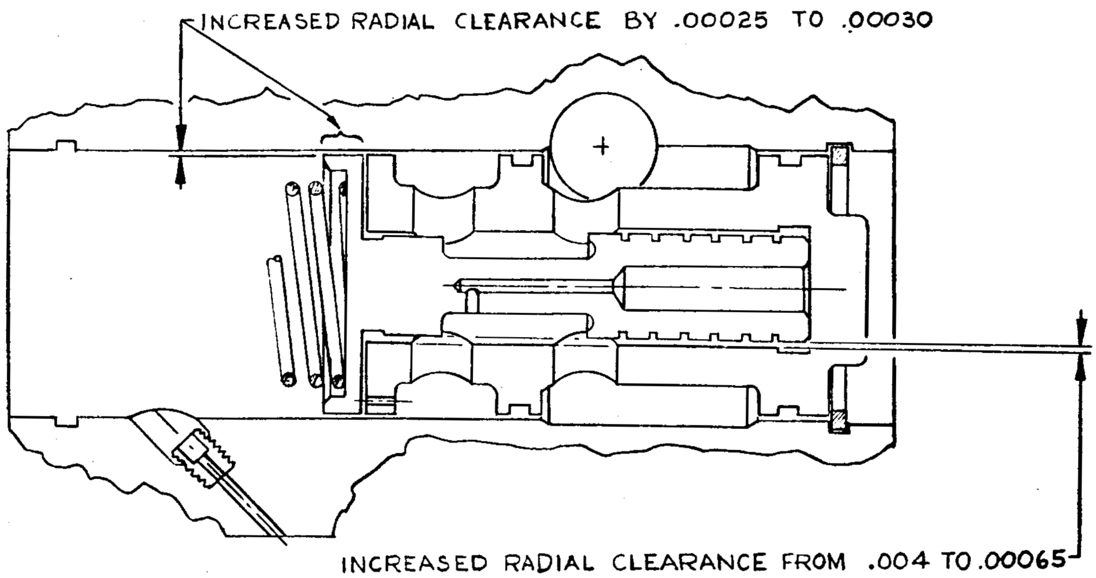

SET - RANGE 3, POS. 12
.025
.020
.015
.010
.005
$\frac{\Delta Q_3}{\Delta P_3}$ (GPM/PSI) ⊙
180°
160°
140°
120°
100°
80°
60°
40°
20°
Φ (LAG ∡ - DEGREES) △
STOCK VALVE
MOD. #2
MOD. #3
.02
.05
0.1
0.3
1.0
3.0
10.0
30.0
DYNAMIC FLOW CONTROL CHARACTERISTICS
FCV-1 & FCV-2
FLUID - NUTO 146
R. FROST

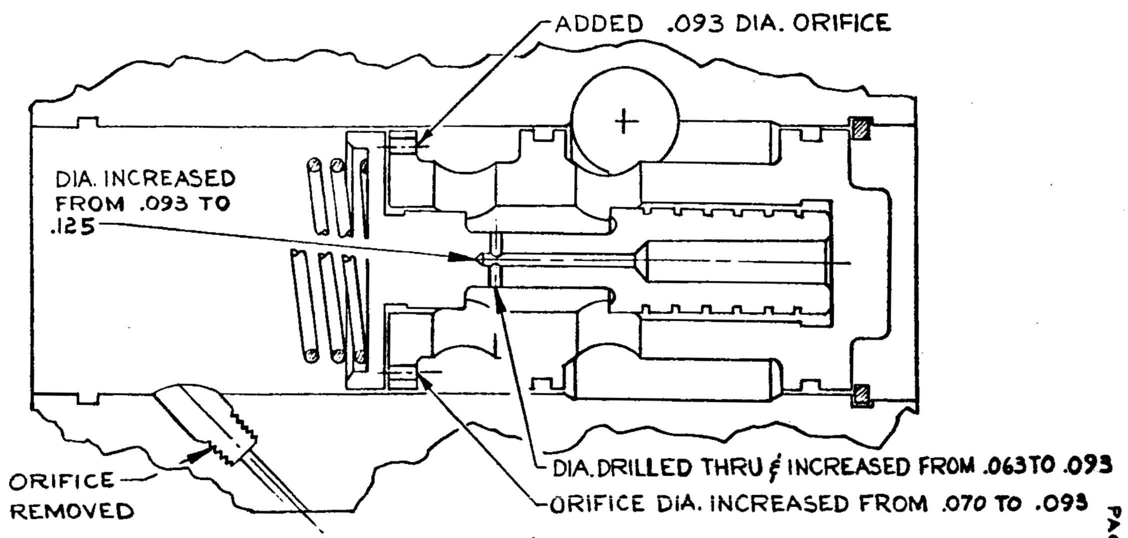
FIGURE 10



MODIFICATION #1



MODIFICATION #2



MODIFICATION #3

FIGURE NO. 11

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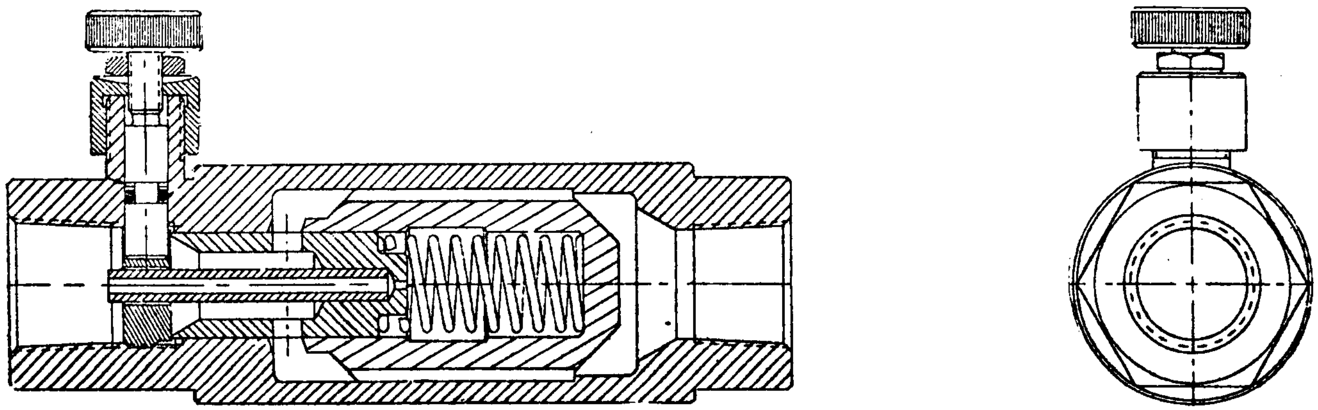

FLOW CONTROL VALVE FCV -25 & -26

88A4100829-003

FIGURE 12

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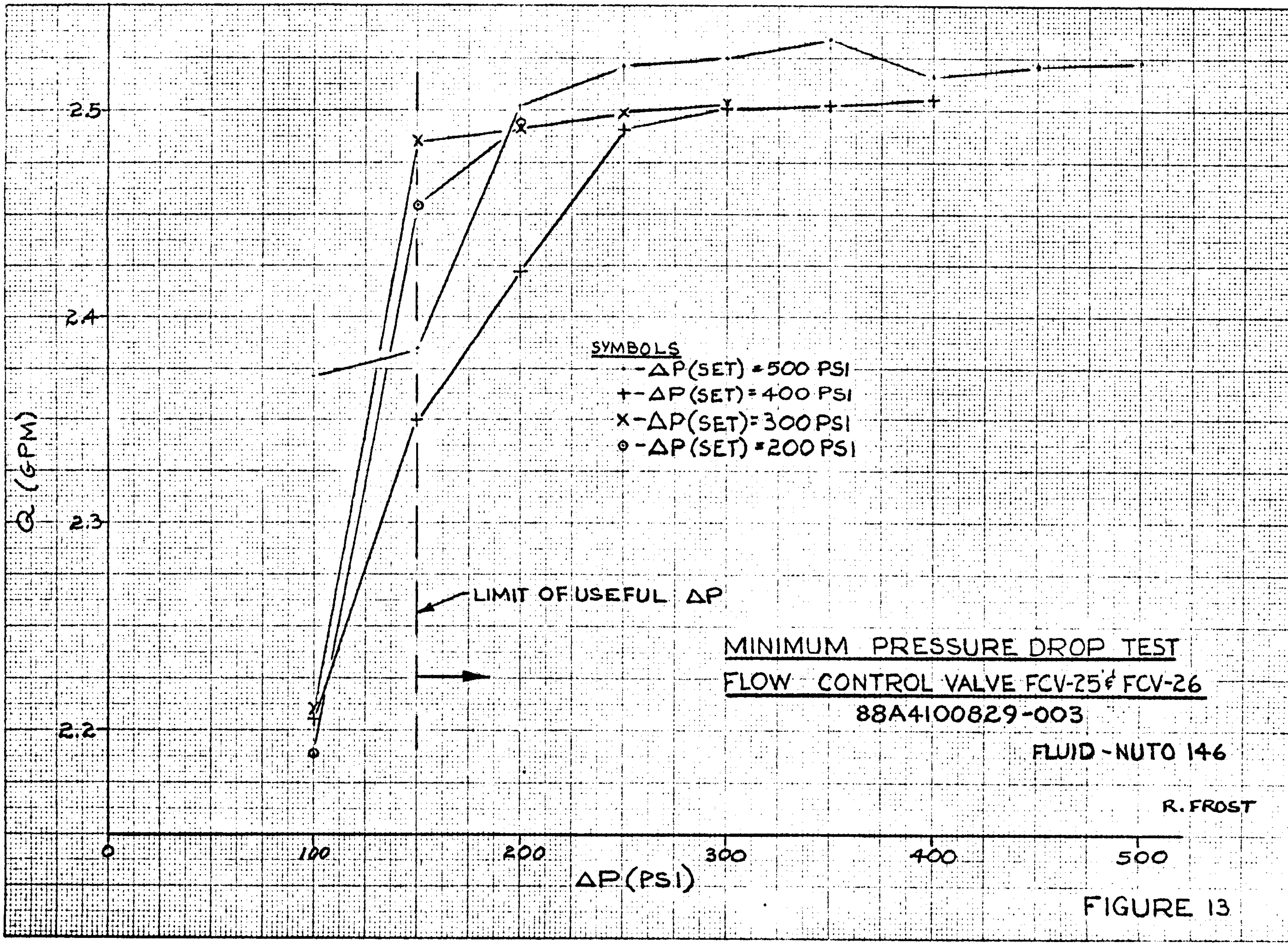

Q (GPM)
2.5
2.4
2.3
2.2
0
100
200
300
400
500
ΔP (PSI)
SYMBOLS
· - ΔP(SET) = 500 PSI
+ - ΔP(SET) = 400 PSI
x - ΔP(SET) = 300 PSI
⊙ - ΔP(SET) = 200 PSI
LIMIT OF USEFUL ΔP
MINIMUM PRESSURE DROP TEST
FLOW CONTROL VALVE FCV-25 & FCV-26
88A4100829-003
FLUID - NUTO 146
R. FROST

FIGURE 13

III. HYDRAULIC PRESSURE REGULATOR TESTS AND MODIFICATION (Martin Co. No. 88A4100499-001, Groove No. WH408-K4)

A. SUMMARY

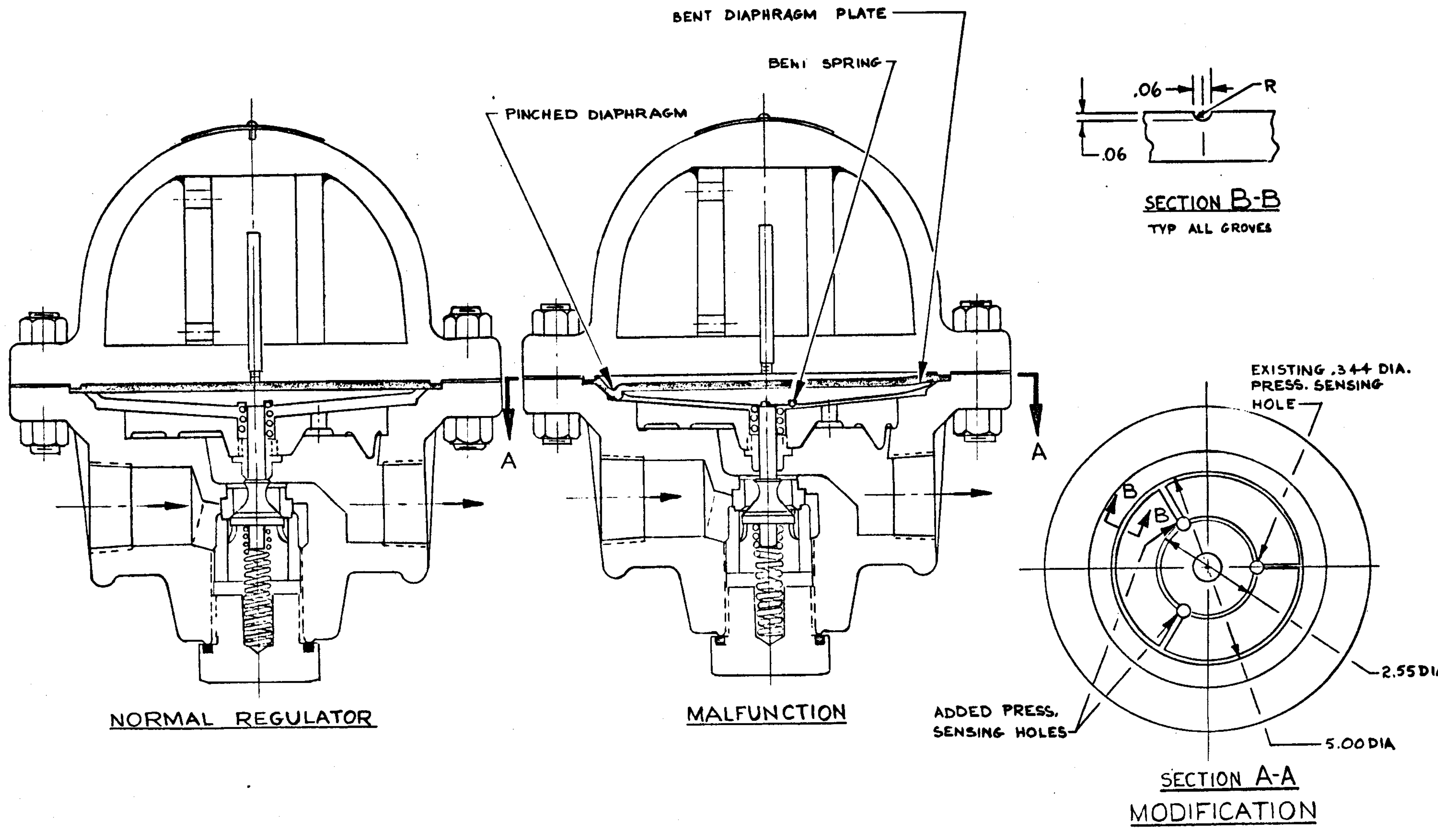
Hydraulic pressure regulators PRV-1, -2, -3 and -4 were used to reduce hydraulic pressure from 2900 psi to 2700 psi and to hold the 2700 psi downstream pressure constant under sinusoidally varying flow demand. The flow demand frequency ranged between .05 and 20 cps. During testing the regulator ceased to regulate and remained in an open position regardless of the supply, downstream or dome pressure. Upon disassembly, the diaphragm, diaphragm plate, and diaphragm spring were found to be damaged. (See Fig. 14).

The malfunction was corrected by adding two more pressure sensing holes and two concentric and connecting grooves to the diaphragm plate seat. (See Fig. 14)

B. TEST PROCEDURE AND RESULTS

The pressure regulator was installed in a hydraulic system supplying Nuto 146 hydraulic oil and used to provide a constant pressure source for system component operation. The supply pressure to the regulator was 2900 psi. The regulator dome was loaded to provide 2700 psi regulated discharge pressure. Average flow demands up to 18 gpm were used. Superimposed upon the average flow were sinusoidally varying changes in flow demand up to approximately 10% of the average flow. Frequencies used were from .05 to 20 cps.

During testing, the regulator stuck in an open positition and could not be made to operate under any combination of system and dome pressures. Upon disassembly and inspection of the regulator the diaphragm spring was found to be deformed laterally, the diaphragm plate was bent upward along the edge on one side, and the rubber diaphragm was pinched in such a way as to indicate that it had been extruded downward between the diaphragm plate and the diaphgram plate seat. (See Fig. 14) Examination of the location of the pressure sensing hole in the diaphragm plate seat and the geometry of the diaphragm plate showed that a pressure unbalance could occur as the diaphragm plate moved up and down such as to create a lateral thrust upon the plate. This thrust would cause the diaphgram spring to bend laterally as the plate is displaced in its seat. When the plate is eccentric to its seat, one edge overlaps the seat shoulder such that the dome pressure can bend the edge upward when the hydraulic pressure is decreased. At the same time the rubber diaphgram is extruded through the gap between the plate and the seal shoulder on the other side. This problem was corrected by drilling two more pressure sensing holes on the same diameter as the original sensing hole and spaced such that all three holes are symmetrically arranged around the center of the seat. A circular groove was machined into the seat such that it connected all the sensing holes. Radial grooves were machined outward from each of the sensing holes to intersect with another concentric circular groove which was machined into the seat. The regulator was then assembled and reinstalled in the hydraulic system. The pressure regulator was used extensively following the modification and no further problems were experienced.



HYDRAULIC PRESSURE REGULATOR PRV-1, -2, -3, & -4

FIGURE 14

R. FROST

IV. CAPILLARY VALVE DEVELOPMENT TESTS (Martin Co. No. 88A4100483-009)

A. SUMMARY

Capillary Valve CV-1, -2, -3, and -4 was installed in a hydraulic system and tested for flow versus opening at constant pressure differential and for flow versus pressure drop at a fixed setting. One laboratory test model valve and two production valves were used in the flow versus opening test. Only the two production valves were used in the flow versus pressure drop test. The data from these tests was compared with calculations for best theoretical performance and found to agree very closely to theory within the desired range. (See Figs. 16 and 17) See Fig. 15 for valve configuration.

B. TEST PROCEDURE AND RESULTS

The capillary valve was installed in a hydraulic system supplying Nuto 146 hydraulic oil at constant pressure and controlled temperature. (See schematic in Fig. 16). The following values and/or tolerances were used for all tests:

P_1 = 800 $\pm$50 psig

$P_1 - P_2$ = Desired value $\pm$10 psi

T_1 = 95° $\pm 2^{\circ}$ F

Flow Setting = Desired value $\pm$ .5 gpm

Flow versus valve opening was measured by opening the valve from the closed position in one or one-half turn increments, as the case may be, and measuring flow at each increment. The pressure drop across the valve was maintained at 300 $\pm$10 psi using the downstream shutoff valve.

Flow versus pressure drop was measured by adjusting the valve opening such that a differential pressure of 300 psi creates a flow of 5 gpm through the valve. The differential pressure was then reduced to 100 psi and the flow was measured. The differential pressure was then increased in 100 psi increments up through 500 psi and the flow was measured at each increment. The results of the test show that the capillary valve conforms to the standard laminar flow equation within the range of use of the valve. The equation for laminar flow through a circular slot is:

$$Q = \frac{\Delta P \pi D h^3}{12 \mu L}$$

where

Q = flow ($in.^3$/sec)

P = pressure differential (psi)

D = average slot diameter (in.)

h = thickness (gap) of the slot (in.)

μ = fluid viscosity (lb-sec/in.^{2})

L = length of flow through the slot (in.)

This equation states that flow is a function of the cube of the flow gap (h) and linear with pressure drop. Examination of Fig. 16 shows that actual flow corresponds well with theoretical flow versus valve opening in the desired range between 1 and 10 gpm. Flows below 1 gpm are somewhat higher than calculated because the calculation does not include the following:

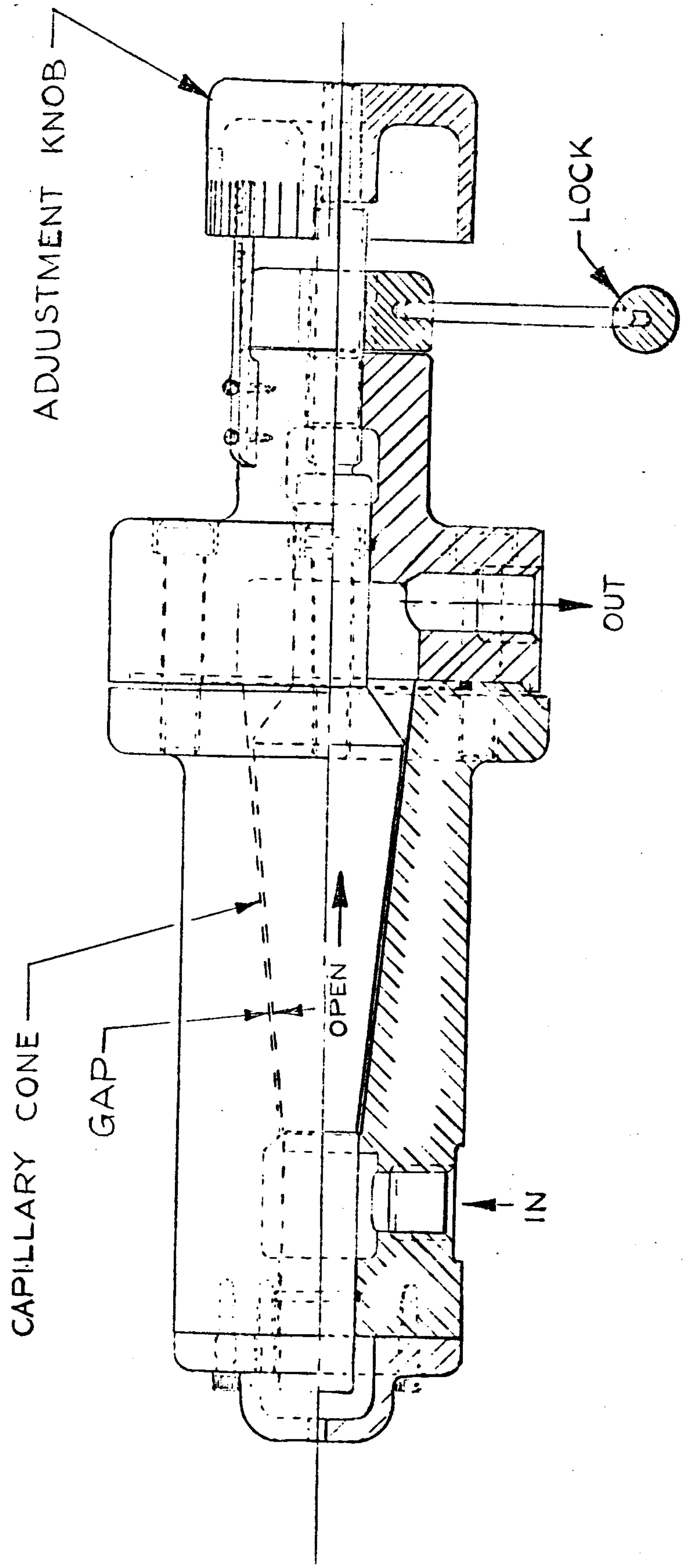
a. Taper tolerance differences between the shaft and the sleeve.

b. Eccentricity tolerance between the shaft and the sleeve.

c. Slight shortening of the flow length (L) through the valve as the valve is closed the last three turns.

All of these effects combined result in increasingly higher than theoretical flows as the valve is closed. This is because the flow gap errors become a relatively large percentage of the flow gap which has a cubic relationship to flow. Flows above 10 gpm become increasingly less than theoretical values with increasing flow. This is due to pressure drops in the capillary valve (other than the capillary passage) becoming a significant part of the total pressure drop across the valve. Since both departures from theory are either above or below the range of usage of the valve, these characteristics are of no significance in the use of the unit. The results of the flow versus pressure drop test show good agreement with the theory that flow is a linear function of pressure differential (see Fig. 17). The relatively large deviation of flow at 100 psi pressure differential probably comes from the error involved in reading small pressure differentials with high range gages.

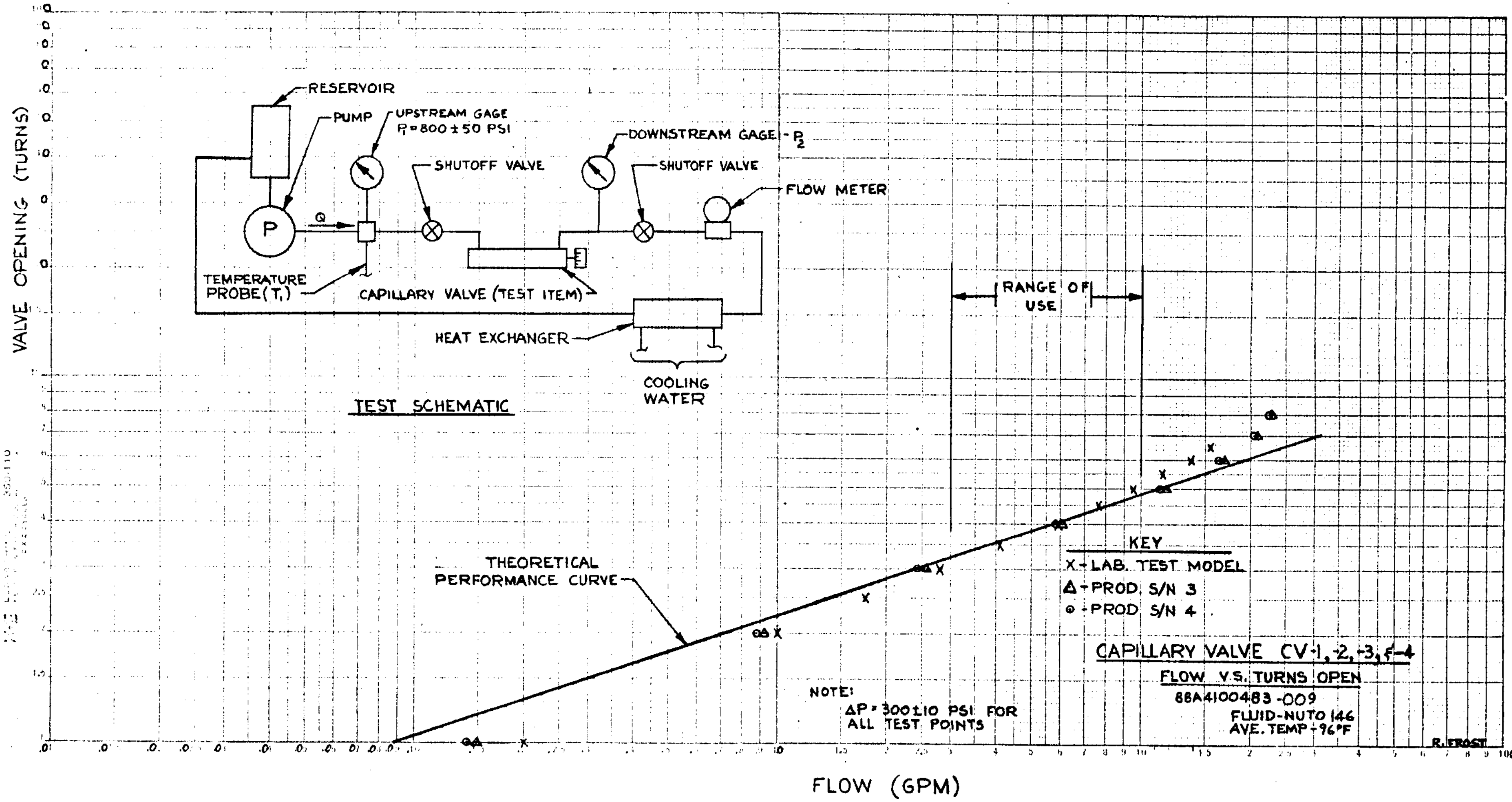
R. FROST



CAPILLARY VALVE CV-1, -2, -3, & -4

88A4100483-009

FIGURE 15



RESERVOIR
PUMP
UPSTREAM GAGE
$P_1 = 800 \pm 50$ PSI
SHUTOFF VALVE
DOWNSTREAM GAGE - P_2
SHUTOFF VALVE
FLOW METER
Q
P
TEMPERATURE PROBE (T_1)
CAPILLARY VALVE (TEST ITEM)
HEAT EXCHANGER
COOLING WATER
TEST SCHEMATIC
RANGE OF USE
THEORETICAL PERFORMANCE CURVE
KEY
X - LAB. TEST MODEL
Δ - PROD. S/N 3
⊙ - PROD. S/N 4
CAPILLARY VALVE CV-1, -2, -3, & -4
FLOW V.S. TURNS OPEN
88A4100483-009
FLUID-NUTO 146
AVE. TEMP - 96°F
NOTE:
$\Delta P = 300 \pm 10$ PSI FOR ALL TEST POINTS
R. FROST
VALVE OPENING (TURNS)
FLOW (GPM)

FIGURE 16

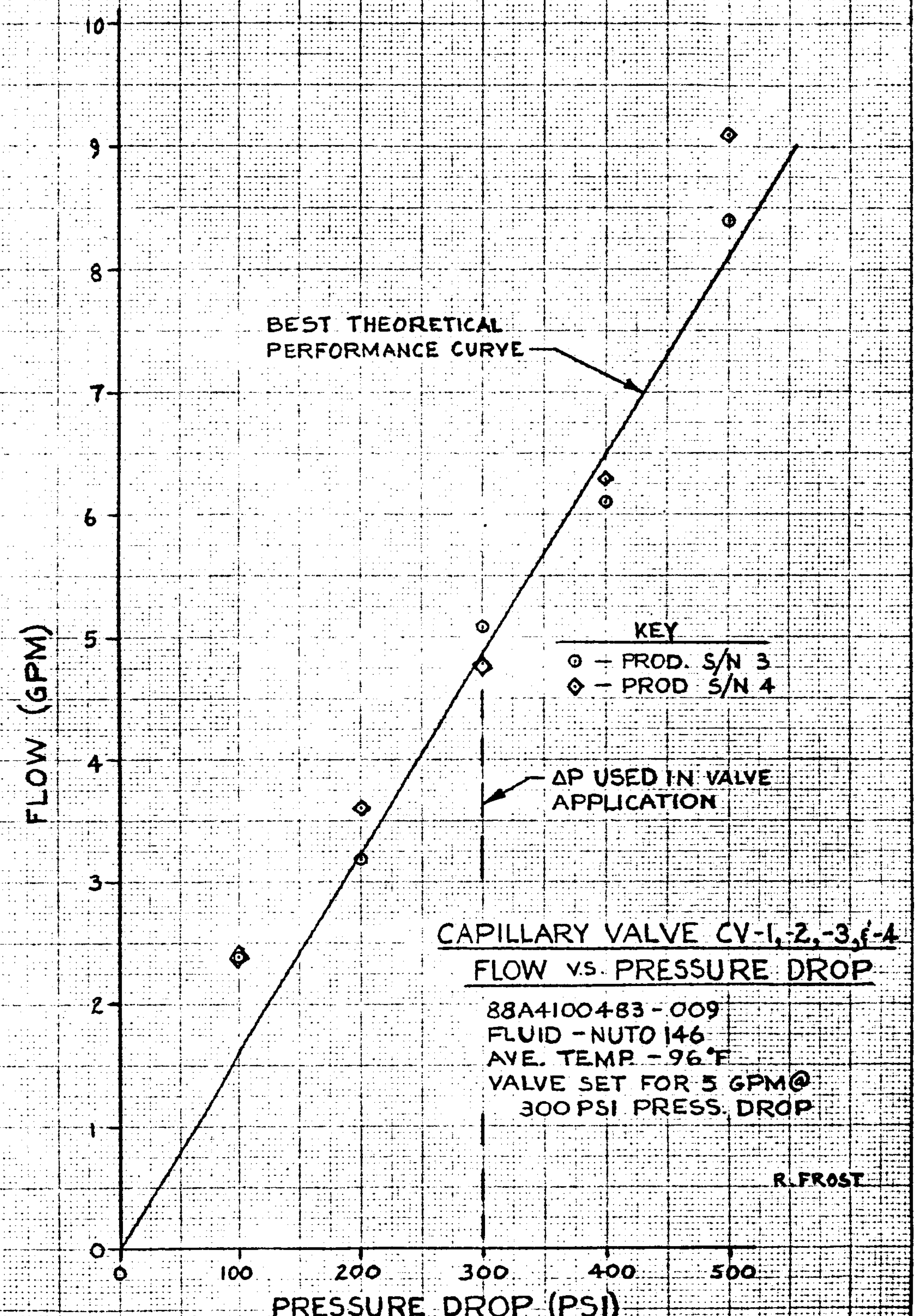

FLOW (GPM)
0
1
2
3
4
5
6
7
8
9
10
0
100
200
300
400
500
PRESSURE DROP (PSI)
BEST THEORETICAL
PERFORMANCE CURVE
KEY
⊙ – PROD. S/N 3
◇ – PROD S/N 4
ΔP USED IN VALVE
APPLICATION
CAPILLARY VALVE CV-1,-2,-3,&-4
FLOW VS. PRESSURE DROP
88A4100483-009
FLUID – NUTO 146
AVE. TEMP. – 96°F
VALVE SET FOR 5 GPM@
300 PSI PRESS. DROP
R. FROST

FIGURE 17

V. FLOAT HEIGHT INDICATING SYSTEM DEVELOPMENT TESTS

A. SUMMARY

During Head to Head testing two problems were experienced with the hydraulic portion of the Float Height Indicating System. It was found that due to the highly viscous nature of the oil, the oil level sight glass (LLI-1) had a very slow response to changes in oil level in the air spring. It was also found that in decreasing the oil pressure upon shutting down, care had to be taken to close CV-17 (see Fig. 19), before the gas in the air spring was lost through the sight glass to the oil cavity under the piston. Air spring gas could also be lost at any time during operation when the gas pressure exceeded the oil pressure (float seated at lower end of piston).

The former problem was considerably improved by increasing the line size between the cylinder and the sight glass from 3/8-in. to 5/8-in. diameter. The latter problem was solved by converting a AN6280-8 check valve to a velocity fuse by moving the poppet spring from the back end to the front end of the poppet (see Fig. 18). This arrangement was found to shut off flow in the reverse direction between 1.4 and 1.8 gpm using Nuto 146 hydraulic oil. The velocity fuse was then installed in the hydraulic line between the sight glass and CV-17 such that rapid decrease of the oil level in the sight glass would cause the valve to close. The valve remains closed until no reverse differential pressure exists and automatically resets itself under that condition.

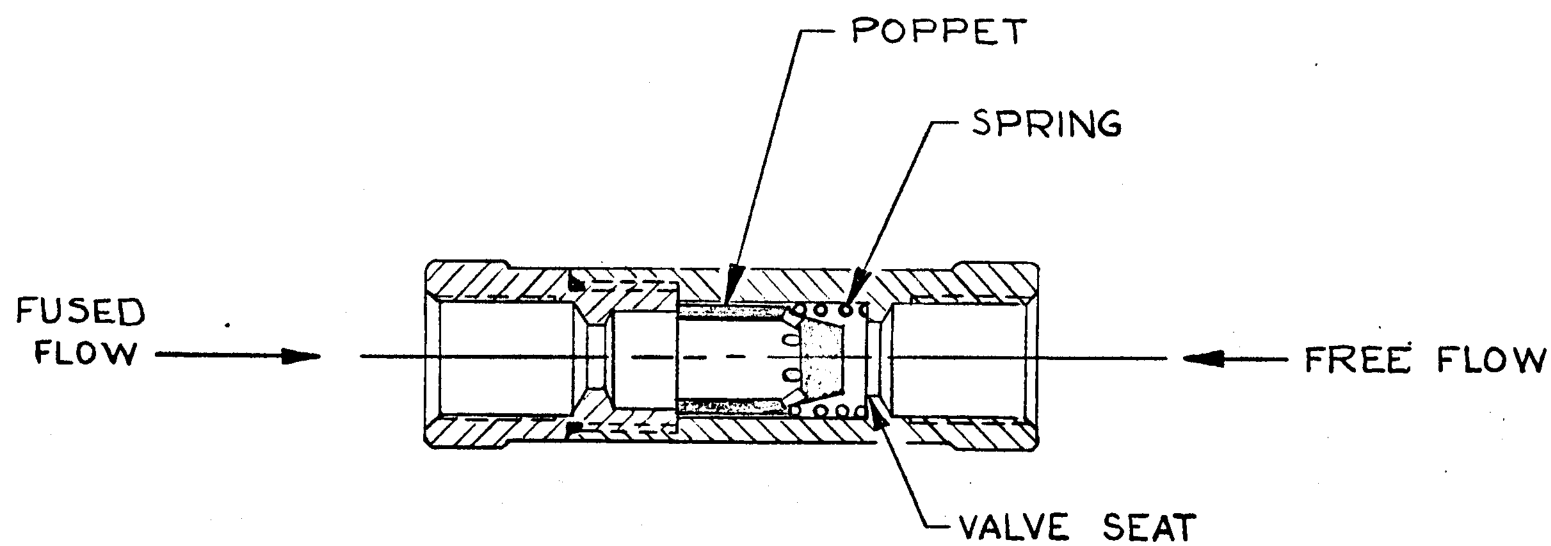
B. TEST PROCEDURE AND RESULTS

1. Sight Glass Lag

The 3/8-in. hydraulic tubing which connected the sight glass to the support cylinder was removed and replaced by 5/8-in. hydraulic tubing of the same length and configuration. None of the components in the line were changed other than fittings which were changed to fit the larger tube size. The result of this change was to make it possible to get a reliable static float height reading from the sight glass and to greatly reduce the lag between the float height and the sight glass reading under changing conditions.

2. Velocity Fuse Development

A AN6280-8 3000 psi hydraulic check valve was converted to a velocity fuse by removing the spring and the poppet from the valve and reinstalling them such that their relative positions were reversed in the valve body (see Fig. 18). This arrangement allows reverse flow through the valve until the pressure drop across the poppet is sufficient to overcome the force of the spring, closing the valve. The velocity fuse was installed in a hydraulic system supplying Nuto 146 hydraulic oil at constant pressure and 95° F (see Fig. 20). The velocity fuse was tested by closing the downstream shutoff valve and applying 1500 to 2000 psi to the system. The shutoff valve was then opened in small increments and the flow was measured at each increment. The flow was incrementally increased until P_2 suddenly decreased to zero pressure, indicating that the fuse had closed. All fuses closed between 1.4 and 1.8 gpm. The velocity fuse was then installed in the line between CV-17 and the sight glass such that free flow was toward the sight glass. When the missile is lowered to the pedestal, the air spring will expand until the float bottoms at the lower end of the piston. At this point, there will be a sudden decrease in hydraulic pressure, causing the oil to be forced out of the sight glass at a high rate, forcing the velocity fuse closed.



HYDRAULIC FUSE

88A4100853-001

FIGURE 18

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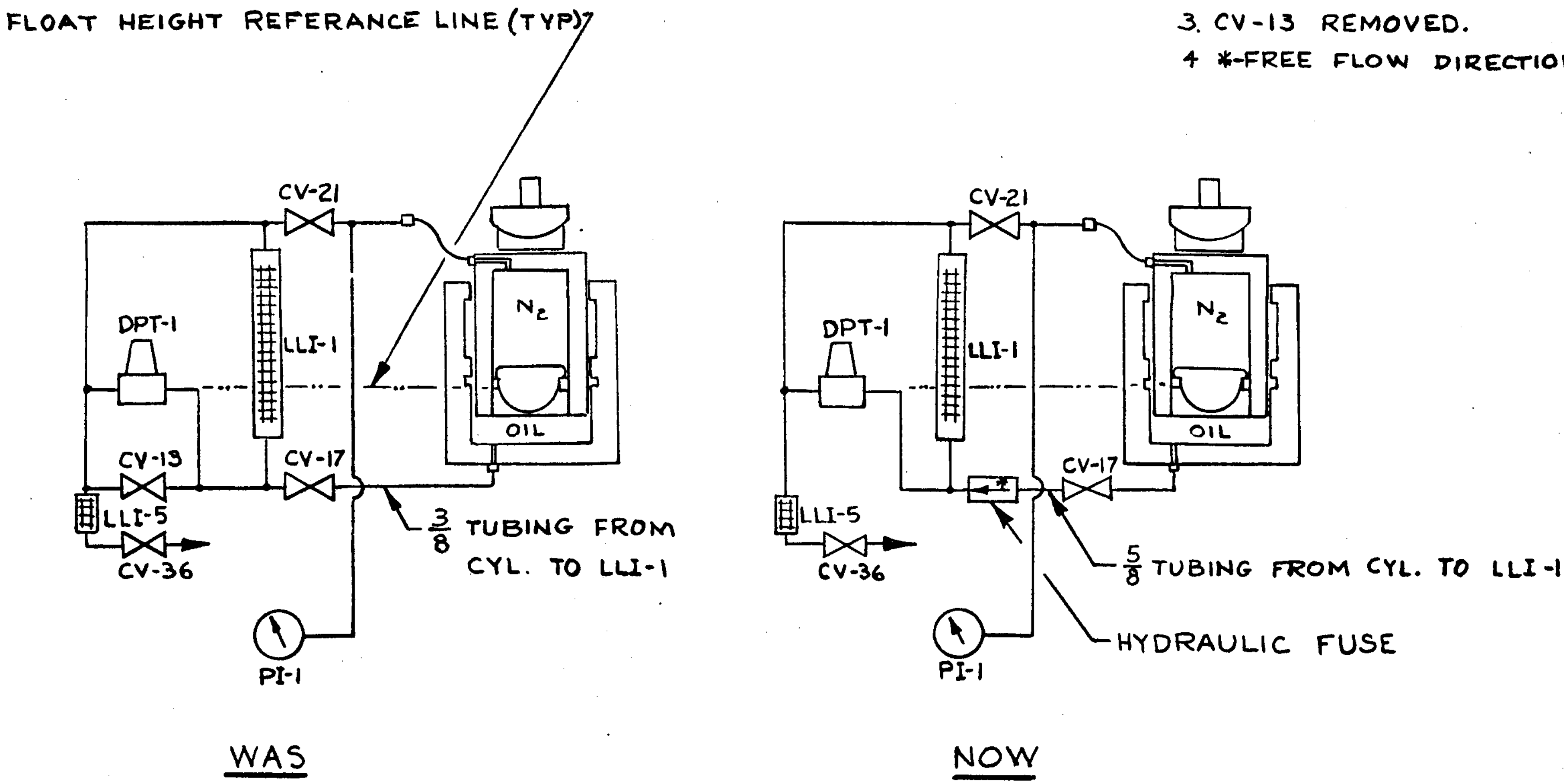
NOTES:

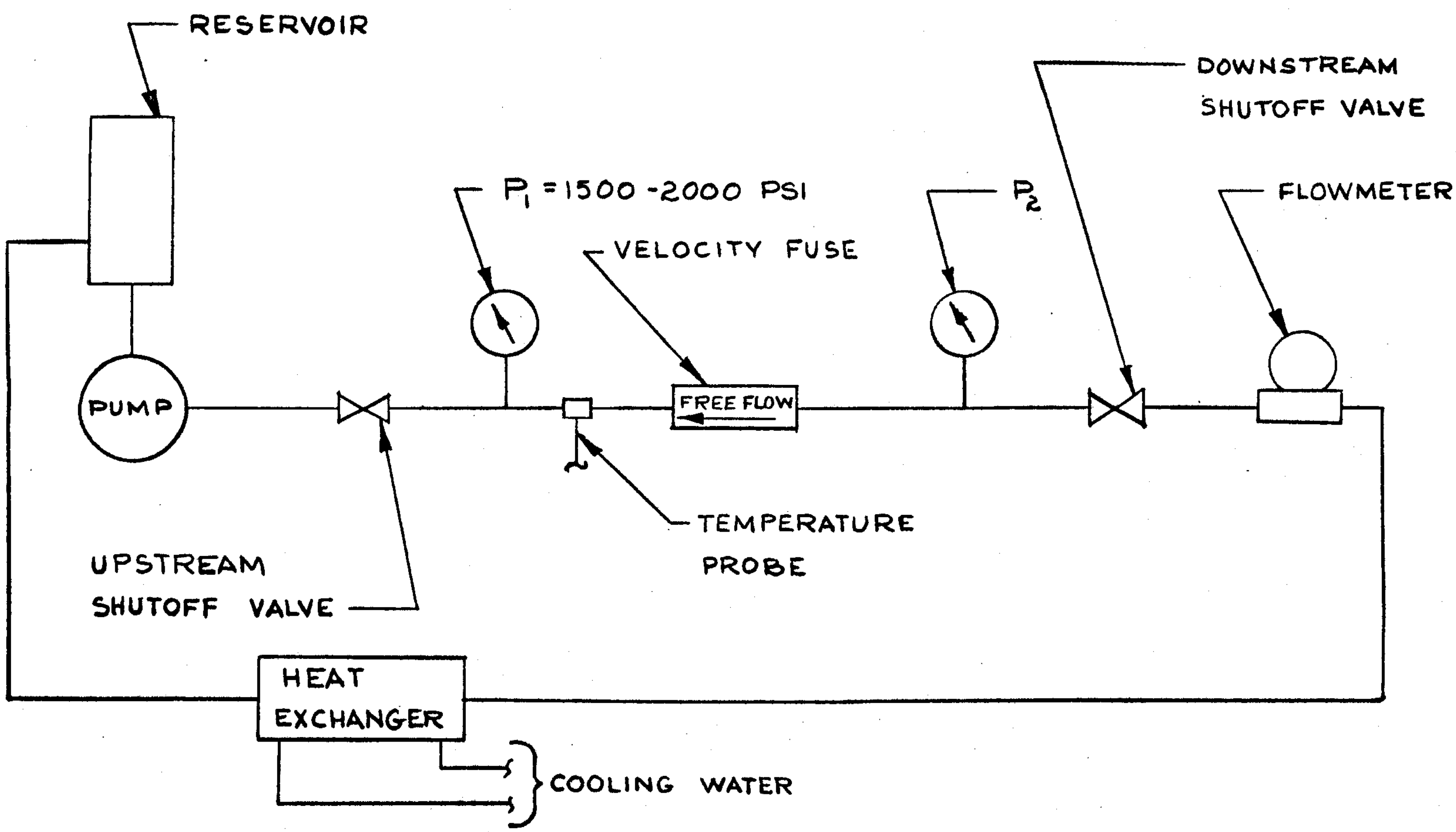
1. SCHEMATIC DETAILS DELETED FOR CLARITY.
2. SUPPORT NO. 1 SHOWN, OTHERS SIMILAR.
3. CV-13 REMOVED.
4. *-FREE FLOW DIRECTION.



FLOAT HEIGHT INDICATING SYSTEM PIPING SCHEMATIC

REF. 88A4100411 R. FROST

FIGURE 19



VELOCITY FUSE TEST SCHEMATIC

FIGURE 20

R. FROST

VI. BEARING CONTACT WARNING SYSTEM DEVELOPMENT

A. SUMMARY

During Head to Head testing of the Hydrodynamic Support, the original Bearing Contact Warning System design was found to be not suitable for the purpose. It was originally assumed that the resistance between the hydrostatic bearing surfaces would be quite high as long as separation existed. Tests showed, however, that suitable operation of the bearing system with resistances as low as .1 ohms was possible. As a result of this, the electrical circuit used in the contact warning system had to be redesigned such that its range of resistance measurement was adjustable and compatible with the values to be measured. (See Fig 21)

B. TEST PROCEDURE AND RESULTS

During Head to Head testing of the Saturn V Hydrodynamic Test Stand, the bearing contact warning lights gave almost constant indication that there was metal-to-metal contact between the hydrostatic bearings. Measurements of the gaps between all the hydrostatic bearings showed that there was sufficient clearance between the mating surfaces. To resolve this problem, the resistance between all four pistons and cylinders was measured with the piston-cylinder assemblies removed from the test stand. Each assembly sat upright on the floor such that the piston was free of all external connections and floating in the cylinder. The ring bearings were pressurized to about 600 psi and a supporting oil flow was supplied to the piston capillary. The pistons were then bounced on their air springs such that piston-cylinder friction measurements could be made and the electrical resistance between the piston and cylinder was continuously measured. All resistance measurements were made using a DC bridge. The results of this test showed that about .1 ohm bearing resistance was a desirable minimum resistance for suitable bearing performance. It was also found that a band of resistance "noise" was superimposed upon the average values of bearing resistance and that this "noise" was a natural characteristic of the bearing. The contact warning system was then developed to fit the new criteria for bearing contact.

C. DEVELOPMENT

The novel development which made the contact warning system operative at low values of contact resistance is a special transformer which effectively multiplies impedance.

For development of the contact warning system, a bread-board system was constructed which simulated the actual system. Referring to Fig 21, simulation was achieved by using the equivalent resistance of the maximum length and exact wire size which run from the console to the bearing transformers located on each pedestal and between the transformers and the bearing halves. The shortest wiring run from the console to the transformers used 210 feet of wire. The longest run uses 330 feet. The wire size for this is number 16 gage. The wiring from the transformers to the bearing halves uses ten feet of number 10 gage wire.

A variable potentiometer was used to simulate the hydrostatic bearing resistance. Before each test, the full scale adjustment potentiometer was adjusted such that one milliampere of current passed through the meter-relay with zero resistance across the bearing surfaces. The simulated bearing resistance was then incrementally increased and the current flow through the measuring circuit was read from the meter-relay at each increment. Three turns ratios for the bearing transformers were tested and the results are shown in Figures 22, 23, and 24. A turns ratio of 53 to 4, primary to secondary (Fig 24), was selected due to the system's sharp cut-off characteristics in the desired range of bearing resistance and the lack of sensitivity to changes in wiring length. A capacitor was added to the D.C. output of the rectifier to act as a filter to eliminate the noise superimposed on the actual bearing resistance.

Any resistance within the range of the resistance measuring system can be selected as a contact warning point by adjusting the high limit contact on the meter-relay. This is achieved by setting the contacts to "make" at some value less than 1 milliampere which corresponds to a known bearing resistance as measured during the bread-board test. When the average current reaches the preset level, contact is made and the events previously described take place.

If, in the future, it is desirable to increase the range of bearing resistance measured by this system, it can be accomplished by increasing the value of the limiting resistor. This allows a proportionally larger voltage drop across the limiting resistor--thus increasing the current flow in the measuring circuit. The meter-relay can then be set for 1 ma maximum and the curve shown in Fig 24 will be displaced to the right at the lower end.

D. SYSTEM OPERATION

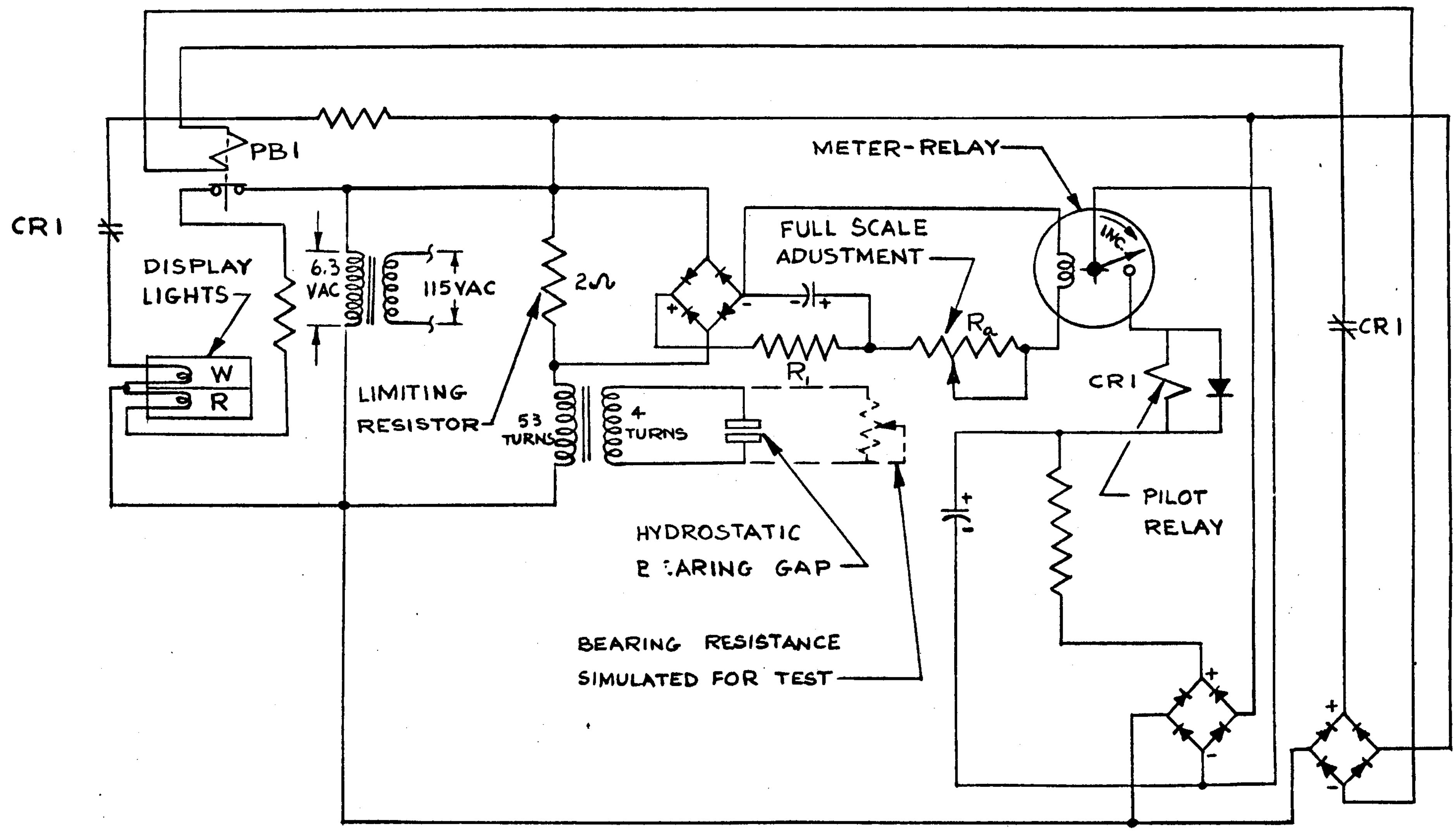
The developed system is composed of one subsystem for each of the 12 bearings, each subsystem detecting the resistance of a pair of mating hydrostatic bearing surfaces. The subsystems are identical but each may be adjusted individually to react any point within a range of contact resistance. This range extends from about .12 ohm to about

10 milliohms. A typical subsystem is shown in Fig 21. Each subsystem actuates a respective light display on the control panel. Prior to contact, a white light is displayed, identifying by legend the bearing indicated as "clear." At the instant of contact, the white light will be extinguished and a red warning light will appear and remain lit regardless of subsequent contact resistance level. Thereafter when the contact resistance again resumes a higher level than the set critical value, the white light will come on again. At this time, both the red and white lights will show. The operator may then reset the warning light by punching the red light which extinguishes it. Resetting cannot be achieved if the contact resistance is still equal to or less than the set critical value. The three possible indications are as follows: First, there is white, i.e., all clear. Second, there is red, i.e., contact exists now. Third, there is white and red, showing that there was contact in the past, but the bearing is clear now.

This operation is achieved by the following (see Fig 21): There is a "contact power on and off" control common to all subsystems. This supplies 115 VAC to each subsystem input transformer. The secondary of the transformer applies 6.3 VAC (open circuit) normally through a limiting resistance across a 53 turn transformer primary. The four turn secondary of this transformer applies its voltage through short leads across the two halves of the bearings, The current flow through the primary gives rise to a voltage drop across the limiting resistance. This voltage is applied through a rheostat to a meter type relay. The high limit contact on the meter-relay operates a pilot relay whose normally closed contacts are in series with the solenoid of the indicator.

The circuitry is so arranged that when bearing resistance is high, the reflected impedance to the primary is very high and minimal current flows permitting the white light to remain on. When the bearing resistance becomes critically low, the apparent primary impedance is also low which increases the current flow in the bearing circuit. This, in turn, causes a high voltage across the limiting resistor, which causes the high limit contacts in the meter-relay to "make." This, through the pilot relay, causes the solenoid PB1 in the indicator to be de-energized, causing the switch to light the "contact" red light. When the bearing resistance becomes high again, and then by reverse action of the prior chain of events, the solenoid is energized again, the red light will not be extinguished. The solenoid is not designed to open the associated switch when re-energized. The operator must actuate the pushbutton to effect a magnetic locking.

Note that when power is off, the indicator solenoid is de-energized, hence dropping the magnetic latch. Therefore, on starting up the hydrodynamic support, all bearing indicators show red initially. As the bearings are separated the white light will appear. The red light may then be extinguished and the indicator relay latched up.



BEARING CONTACT WARNING SYSTEM

FIGURE 21

R. FROST

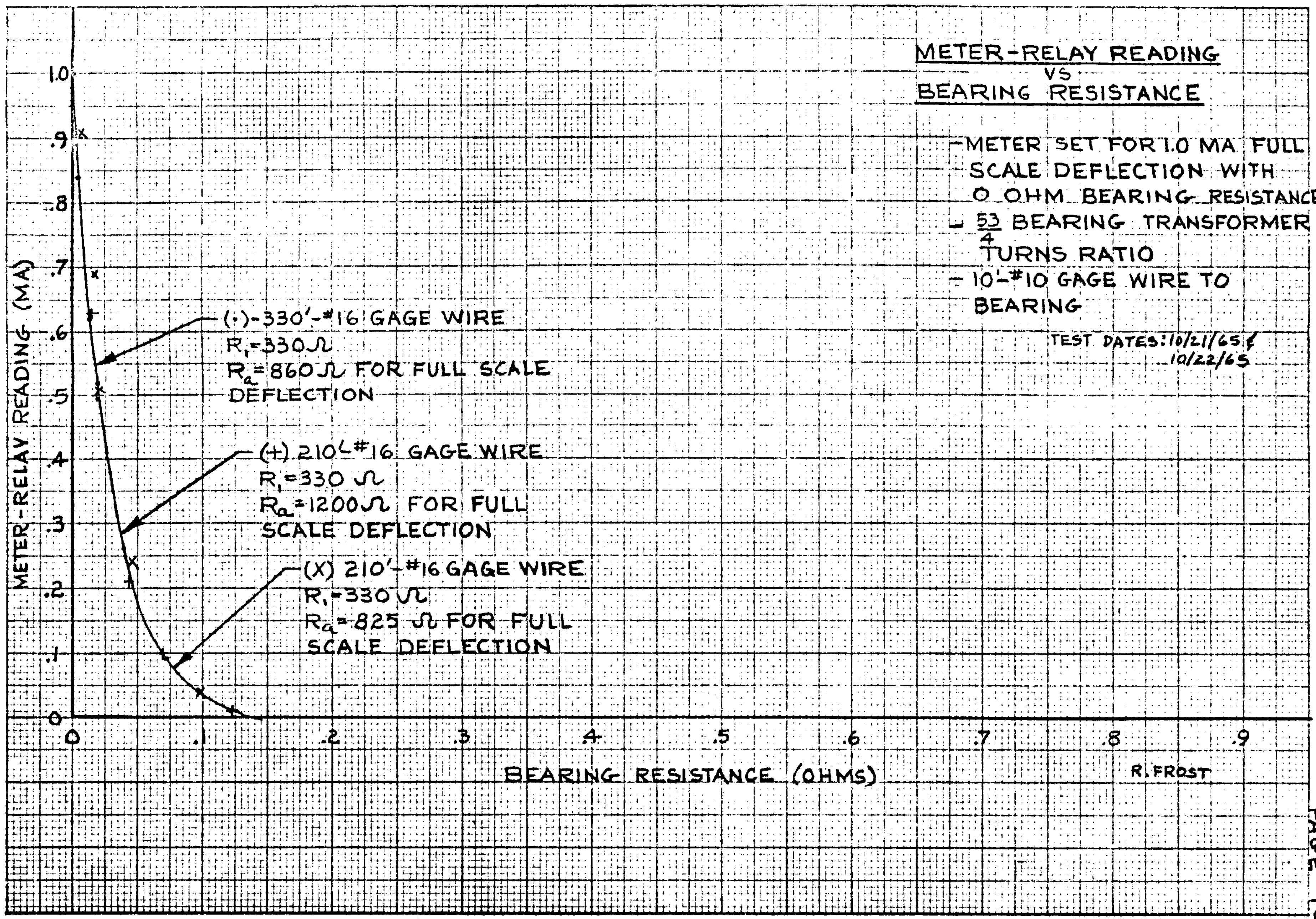

METER-RELAY READING
VS
BEARING RESISTANCE
-METER SET FOR 1.0 MA FULL SCALE DEFLECTION WITH 0 OHM BEARING RESISTANCE
- 53/4 BEARING TRANSFORMER TURNS RATIO
- 10'-#10 GAGE WIRE TO BEARING
TEST DATES: 10/21/65 & 10/22/65
(·)-330'-#16 GAGE WIRE
R1=330Ω
Ra=860Ω FOR FULL SCALE DEFLECTION
(+) 210'-#16 GAGE WIRE
R1=330 Ω
Ra=1200Ω FOR FULL SCALE DEFLECTION
(X) 210'-#16 GAGE WIRE
R1=330Ω
Ra=825 Ω FOR FULL SCALE DEFLECTION
METER-RELAY READING (MA)
BEARING RESISTANCE (OHMS)
R. FROST
0
.1
.2
.3
.4
.5
.6
.7
.8
.9
1.0

FIG. 22

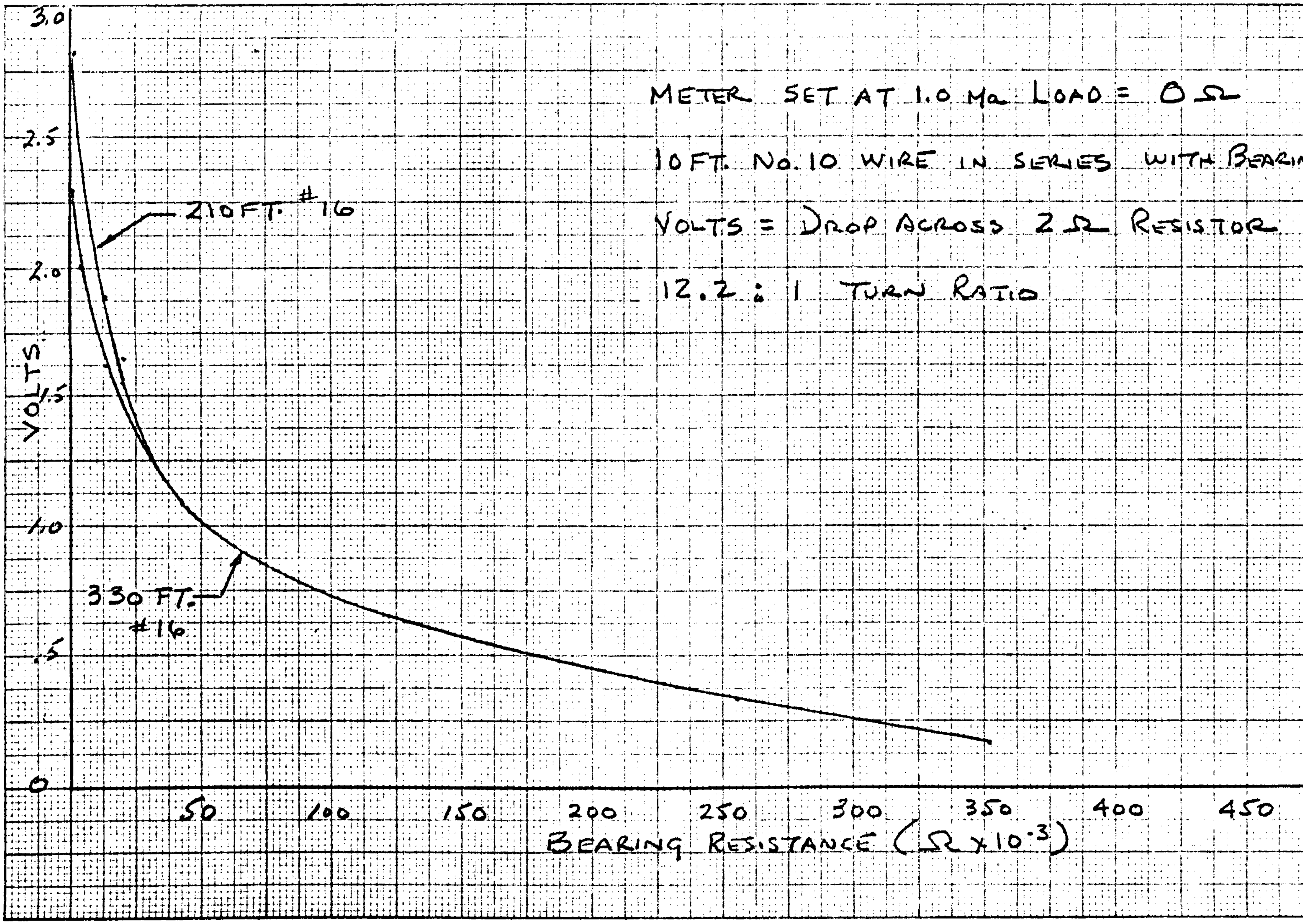

METER SET AT 1.0 Ma LOAD = 0 Ω
10 FT. No. 10 WIRE IN SERIES WITH BEARING
VOLTS = DROP ACROSS 2 Ω RESISTOR
12.2 : 1 TURN RATIO
210 FT. #16
330 FT. #16
VOLTS
3.0
2.5
2.0
1.5
1.0
.5
0
50
100
150
200
250
300
350
400
450
BEARING RESISTANCE (Ω x 10⁻³)

FIG. 23

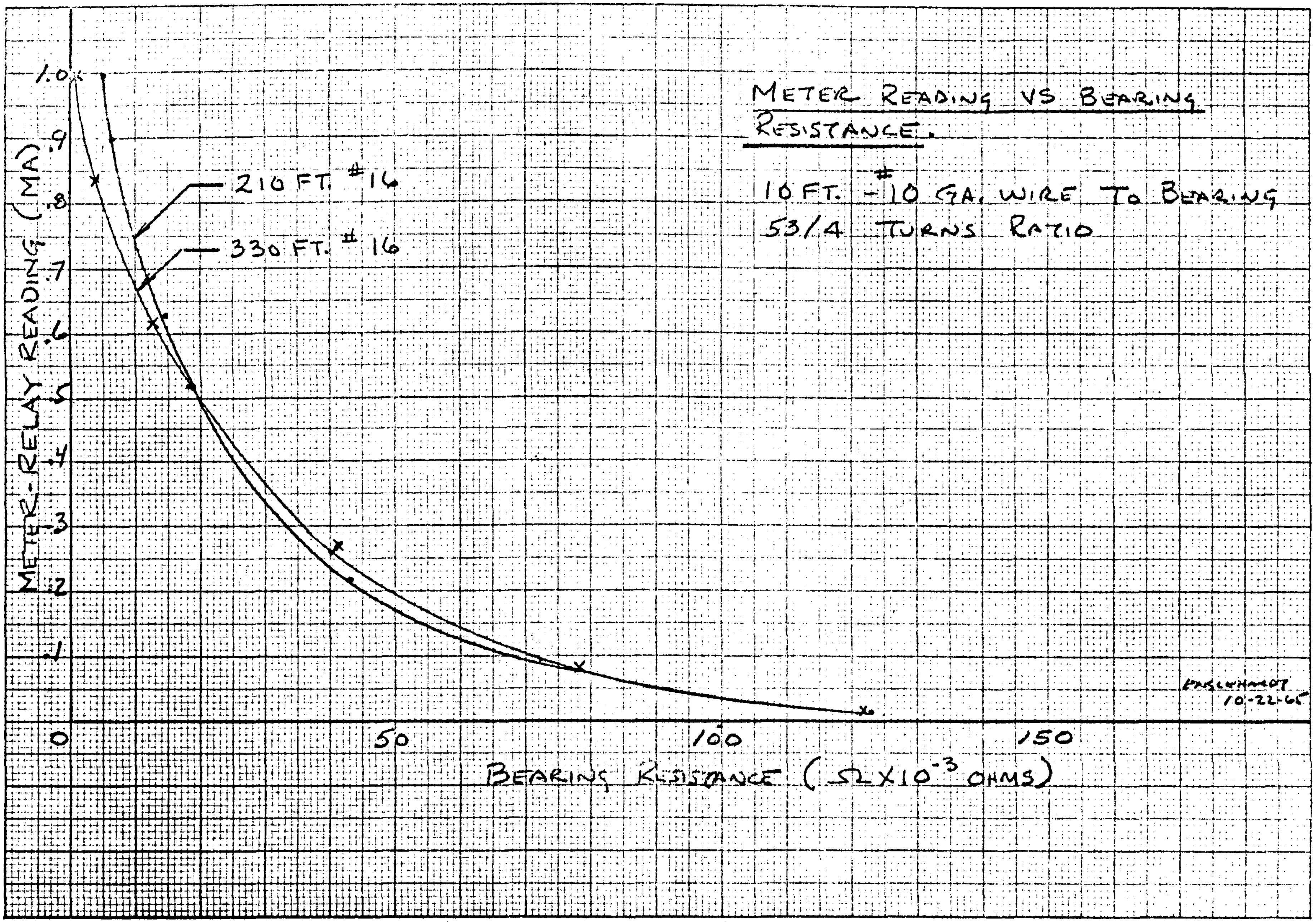

METER READING VS BEARING RESISTANCE.
10 FT. #10 GA. WIRE TO BEARING
53/4 TURNS RATIO
210 FT. #16
330 FT. #16
METER-RELAY READING (MA)
1.0
.9
.8
.7
.6
.5
.4
.3
.2
.1
0
50
100
150
BEARING RESISTANCE (Ω X 10^{-3} OHMS)
10-22-65

FIG. 24

VII. FLOAT SUNK WARNING SYSTEM

A. SUMMARY

During the design phase it was found necessary to provide means to determine whether or not the float contained oil above a critical level. The environmental conditions dictated the use of solid state devices. The classical approach to level indication by heated thermistors was found unsatisfactory due to the overlapping cooling characteristics of oil and gas at the pressure, temperature and oil viscosity encountered in this application. However, the hitherto unappreciated variable thermal lag of the thermistor as a function of immersion or non-immersion was utilized to produce a workable system.

B. DEVELOPMENT

The interface between oil and gas in the piston is defined by a bowl which floats on the oil. It is necessary to determine whether or not the bowl contains oil above a critical level. The dynamic conditions of the environment rule out the use of mechanical devices, while the difficulty of replacement demands devices of high reliability. This led to the selection of a thermistor level sensing system with a proven, high reliability index. The manufacturer took no exception to the pressure, temperature and viscosity conditions.

The device consists of four identical solid state amplifiers. Each furnishes a constant heating current through its thermistor. The resultant voltage drop is then essentially a function of both the current and the resistance of the thermistor. If the thermistor is in oil, the thermistor remains relatively cool, its resistance remains relatively low and the voltage drop across it remains well below a built-in comparison value. When this comparison value is not exceeded the initially displayed "SUNK" indication remains lighted. Conversely, when the thermistor is in air it quickly reaches a temperature and, hence, a resistance level sufficiently high that the voltage drop across it exceeds the comparison value and a switchover takes place, changing the "SUNK" to a "FLOAT" display.

The commercial unit worked well under standard atmospheric conditions with a voltage drop at switchover of about 1.69 v. When operated in 3000 psi nitrogen gas the voltage drop rose to a mean of 3.44 volts, typically. This value was considerably above the comparison value and caused an erroneous SUNK" indication. New voltage reference zeners and new resistances were installed to provide a new reference voltage of 3.7 oh. This level is more than 3 sigmas from the distribution about the 3.44 volt mean. On the instant of submersion the thermistor voltage rose to 4.4 volts mean. The distribution about this mean was more than 3 sigmas from 3.7 volt comparison level. At this point it was discovered that while the initial submersion of the thermistor produced a "SUNK" signal, this signal could be maintained only if the thermistor were in agitated oil. When both the thermistor and the oil were held still, the thermistor soon surrounded itself with a heated shell of oil. The high viscosity of the oil prevented convection currents and the thermistor assumed temperatures actually higher than those occurring in the unsubmerged condition. Hence, an erroneous "FLOAT" signal was produced.

At this juncture the manufacturer provided no solution. However, Martin proceeded with the development of a system based on the observation that the time to critical temperature for a non-immersed thermistor is substantially shorter than for one that is immersed. It was realized at the same time that with the proper heating current values at very low temperatures, say below 60° F, a submerged thermistor never reaches the critical temperature and therefore operates as designed. Obversely, at temperatures at about 130° F the time to critical temperature for submerged and non-submerged thermistors is very close, on the order of 0.5 sec.

C. TEST PROCEDURE

A rather large number of development tests were made in the normal temperature and pressure range of operation. The following is typical:

A sensor was fixed near the bottom of a pressure vessel which was half filled with oil. Power was applied to the sensor simultaneous with the start of a timer. The time to reach critical voltage was indicated by the equipment now in the Hydrodynamic Support. Measurements were made at 3000 psi and at atmospheric pressure. For each pressure one group was made in oil and one group in air. Each group consisted of 10 measurements. Immersion and non-immersion were made alternately by tumbling the container, prior to power application. A 1 minute pause was made between tests.

The following data were obtained:

Time to Critical Temperature (second)

	Atmospheric Pressure		3000 psig Pressure	
	Mean	Range	Mean	Range
Sensor to air	1.74	1.78	1.86	1.95
		1.69		1.80
Sensor in oil	5.67	5.71	6.67	8.10
		5.57		6.20

A similar test at 120° F produced the following data:

Time to Critical Temperature (second)

	Atmospheric Pressure		3000 psig Pressure	
	Mean	Range	Mean	Range
Sensor in air	1.93	+0.08	2.01	+0.09 (a)
		-0.07		-0.1
Sensor in oil	3.01	+0.07	3.09	+0.08
		-0.06 (b)		-0.07

From the above an operating parameter was developed so that there is less than a 0.001 chance of error.

OPERATING PARAMETER:

Maximum time (second) sensor in air: 2.15 sec (from condition (a) above.

Minimum time (second) sensor in oil: 2.92 sec (from condition (b) above.

The timers which act as the discriminant elements are of the ±1.5% variety. No significant deterioration of the circuit's ability to discriminate between the two conditions need be expected as a function of the balance of the circuitry.

VERIFICATION TEST:

After numerous informal tests which yielded favorable results, the as-delivered configuration was tested using the interrogation mode. Correctness of the answers, "FLOAT" or "SUNK" was judged by the performance of the hydraulic parameters. The answers given by the Float Sunk System were correct in all cases.

Number of Groups	Interrogations Per Group	Result	Interval Between Interrogations Seconds	System Pressure (psig)	System Temperature (^{o}F)
9	3	Sunk	60	2300	110
1	3	Sunk	45	1800	110
1	3	Sunk	45	500	108
1	3	Sunk	45	400	108
4	3	Sunk	45	360	106
1	3	Sunk	45	360	104
3	3	Sunk	30	360	103
1	20	Float	60	2300	103

It was noted that during severe agitation, such as evacuation of the float by the purge system, the cooling of the thermistor by high velocity nitrogen will cause erroneous reading of "SUNK."