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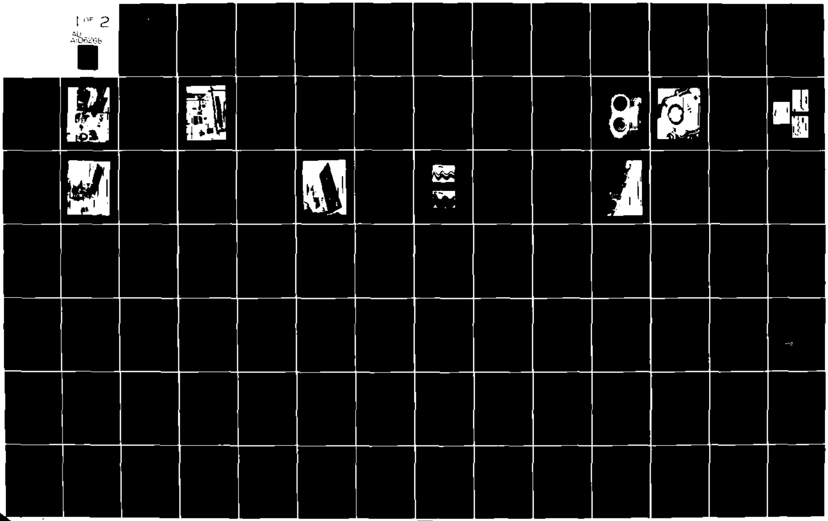

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EVALUATION OF MS-6 FIRE-RESISTANT FLUID FOR 8000 PSI LIGHTWEIGHT HYDRAULIC SYSTEMS

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20. ABSTRACT (Continue on reverse side if necessary and identify by block number)

Evaluation tests and an application study were conducted on a high temperature, fire-resistant silicone base hydraulic fluid designated NADRAUL MS-6. The information developed was used to determine the practicality of using MS-6 in aircraft 8000 psi lightweight hydraulic systems. The test results indicated that MS-6 can be used in 8000 psi systems, but further development effort is needed to improve pump performance and reduce the tendency of

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20. ABSTRACT (Continued)

MS-6 to foam. The application study disclosed that no serious design problems would result from the higher viscosity, higher density, and lower bulk modulus of MS-6; weight and space penalties would be incurred, however.

SUMMARY

Evaluation tests and an application study were conducted on a high temperature, fire-resistant silicone base hydraulic fluid designated NADRAUL MS-6. The tests and study provided information to determine the practicality of using MS-6 in aircraft 8000 psi lightweight hydraulic systems.

Evaluation tests conducted were: pump performance, line pressure drop, pressure surge, restrictor flow, servo actuator frequency response, flow control valve operating characteristics, and solenoid valve internal leakage. The results indicated that MS-6 can be used in 8000 psi hydraulic systems, but further development effort is needed to improve pump performance and reduce the tendency of MS-6 to foam.

The application study examined the impact of MS-6 viscosity, density, and bulk modulus on system and component design, weight and space savings, and actuator stiffness. No serious design problems were anticipated with pumps, actuators, or miscellaneous components. Weight and space penalties of 11.8% and 6.3%, respectively, would be incurred if MS-6 were used to replace MIL-H-83282 fluid in the A-7E 8000 psi lightweight hydraulic system. A stiffness analysis performed on the F-14 horizontal stabilizer actuator disclosed that use of MS-6 at 8000 psi would result in an actuator almost identical in size to the 3000 psi version using MIL-H-5606.

PREFACE

This report documents an investigative program conducted by Rockwell International Corporation, North American Aircraft Division, Columbus, Ohio, under Contract N62269-79-C-0287 with the Naval Air Development Center, Warminster, Pennsylvania. Technical direction was administered by Mr. J. Ohlson, Head, Materials Application Branch, Aircraft and Crew Systems Directorate, Naval Air Development Center (6061).

This report presents the results of laboratory tests and an application study to evaluate a fire-resistant silicone base fluid for use in 8000 psi lightweight hydraulic systems. This work is related to tasks performed at the Naval Air Development Center under AIRTASK No. A3205203/001B/F54-543-203, TASK AREA No. WF41-451-208, and at Rockwell International under Contract N62269-78-C-0363.

Acknowledgement is given to Mr. B. Holland and Mr. A. Jacob for their assistance in the preparation of this report.

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1.0 INTRODUCTION

1.1 BACKGROUND INFORMATION

MIL-H-5606 has been used in military aircraft hydraulic systems for more than 30 years. Performance of this petroleum base fluid has been satisfactory except in two areas: flammability and maximum operating temperature. Flammability is an important safety consideration because of the potential for hydraulic fluid induced fires following accidents, component failures, or combat damage.

The Navy initiated a program in 1972 to develop a high temperature, fire-resistant hydraulic fluid. The candidate developed is a silicone base fluid designated NADRAUL MS-6, Reference 1.

MS-6 has higher viscosity, higher density, and lower bulk modulus than MIL-H-5606. Aircraft systems using MS-6 fluid must be designed to accommodate these differences. The evaluation presented in this report assesses whether such redesign could be accomplished without nullifying penalties.

1.2 PROGRAM OBJECTIVES

Overall objectives of the program were:

- Conduct tests to determine if MS-6 fluid could be utilized in 8000 psi hydraulic systems
- Assess aircraft design changes necessary to accommodate MS-6 fluid
- Make recommendations as to the practicality of utilizing MS-6 to reduce losses due to hydraulic fires

1.3 TECHNICAL APPROACH

The program was conducted in two phases.

Phase I	MS-6 Performance Evaluation
Phase II	MS-6 Application Study

Phase I - A test system was fabricated for determining pump performance, line losses, pressure surges, servo actuator response, and control valve characteristics using MS-6 fluid at 8000 psi. Test results were compared to data previously obtained with MIL-H-27601 and MIL-H-83282 fluids, References 2, 3, and 4. Viscosity and bulk modulus of MS-6 were determined. Program funding did not permit modification of the test equipment for optimum operation with MS-6.

Phase II - Using information given in Reference 5, an evaluation was made of the impact of MS-6 on line sizing, actuator sizing, cooling requirements, and system weight and volume in the A-7E 8000 psi lightweight hydraulic system. Utility functions were additionally included. Employing data given in Reference 3, an analysis was made to determine the effects of using MS-6 at 8000 psi on the F-14 stabilizer actuator stiffness and size.

2.0 MS-6 PERFORMANCE EVALUATION

2.1 EVALUATION TESTS

2.1.1 Test System

The system contained two sections--a power section and a test section. Four different test sections were used: (1) flowmeter calibration, (2) pressure surge, (3) line pressure loss, and (4) servo actuator. The power section is shown pictorially on Figure 1 and schematically on Figure 2. Descriptions of the test sections are given in subsequent paragraphs.

Pressure tubing in the power section was 21-6-9 CRES; return tubing was 6061-T6 aluminum. Standard MS flareless fittings and MS static seals were used throughout the system. Components which previously contained MIL-H-83282 were carefully cleaned to minimize fluid contamination of MS-6. Two 5 micron (absolute) aircraft-type filters were used to maintain fluid cleanliness. The system was filled with 7.5 gallons of MS-6. Flow rate in the pump case drain and system return lines was measured by turbine meters with readout on frequency counters. Fluid temperature was sensed by thermocouples at four locations in the system and monitored on a multi-channel temperature indicator. Temperature stabilization was accomplished by means of a duration-adjusting type controller and oil-to-water heat exchangers.

2.1.2 Flowmeter Calibrations

Flow measurements were required in all MS-6 evaluation tests. Viscosity and density effects of MS-6 on the turbine flowmeters were unknown. The turbine meters were therefore calibrated with MS-6 fluid. Four meters were checked:

Quantity	Flowmeter	Flow Range, gpm
1	Fischer & Porter M/N 10C1507A	0.01 to 1.0
2	Fischer & Porter M/N 10C1510A	0.2 to 10.0
1	Fischer & Porter M/N 10C1510A	0.5 to 15.0

The test setup is shown pictorially on Figure 3 and schematically on Figure 4. Calibrations were based on travel of a piston rod in a double-ended cylinder having a 2-1/2 in. bore, 1 in. rod, and 12 in. stroke. Two optical switches were utilized; one located near each end of piston rod travel. One switch simultaneously started an electronic counter, which totaled turbine meter pulses, and an electronic timer. The second switch stopped the counter and timer simultaneously. Thus,

$$\frac{\text{Total number of pulses, counts}}{\text{Time interval, sec.}} = \text{Average Frequency, Hz}$$

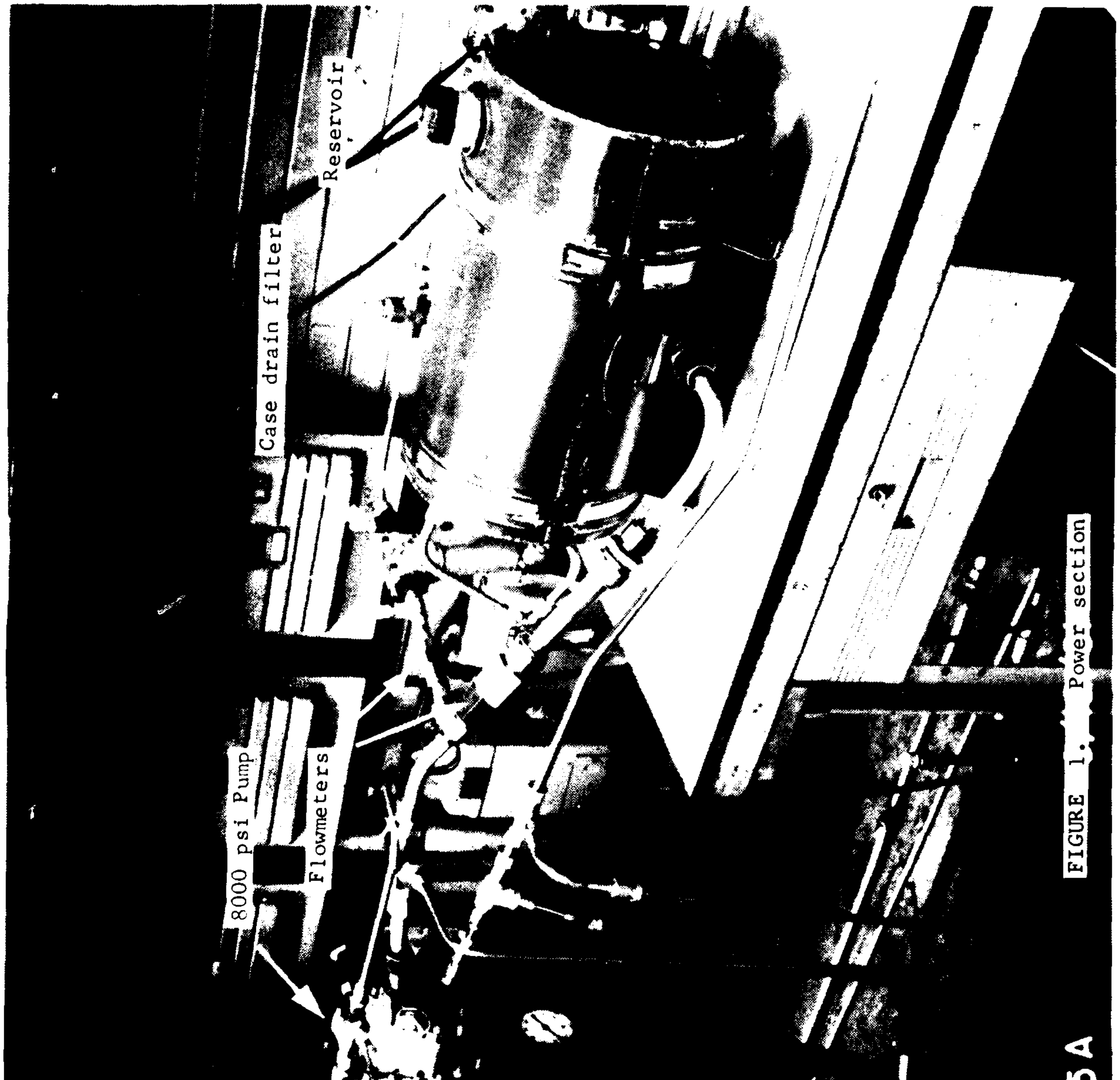


FIGURE 1. Power section

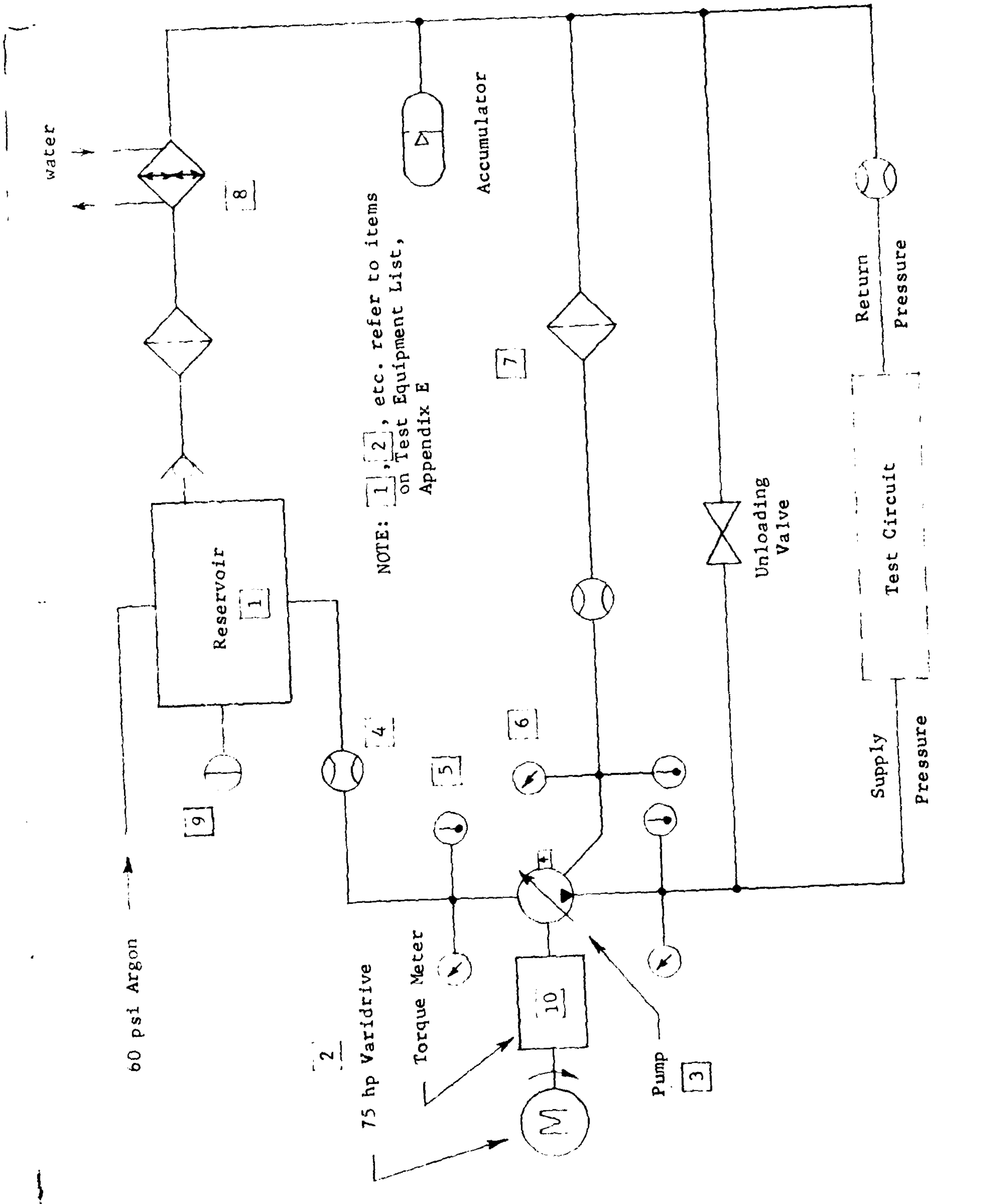


FIGURE 2. Power system schematic

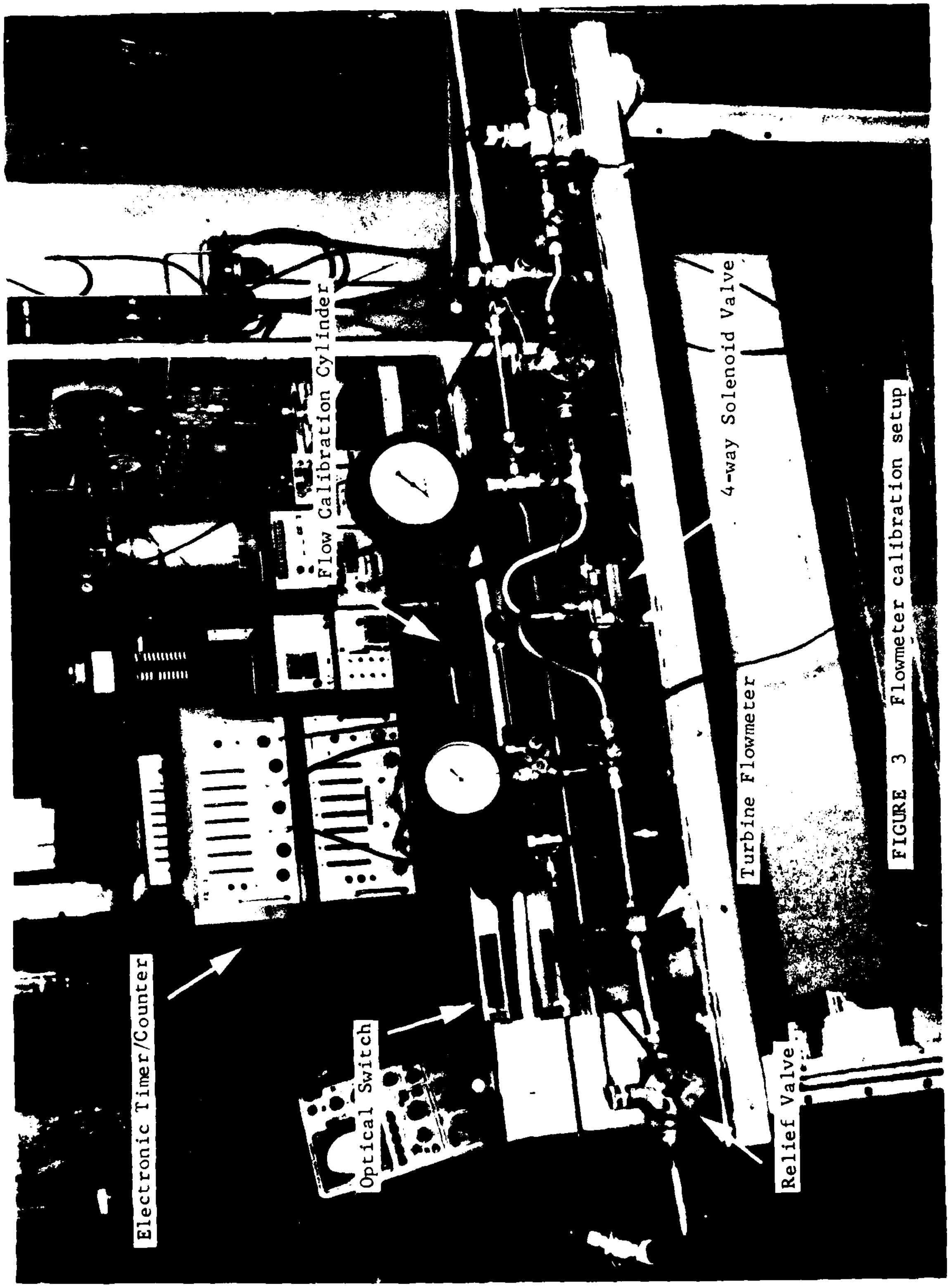


FIGURE 3 Flowmeter calibration setup

NOTE: 5, 6, etc. refer to items on Test Equipment List, Appendix E

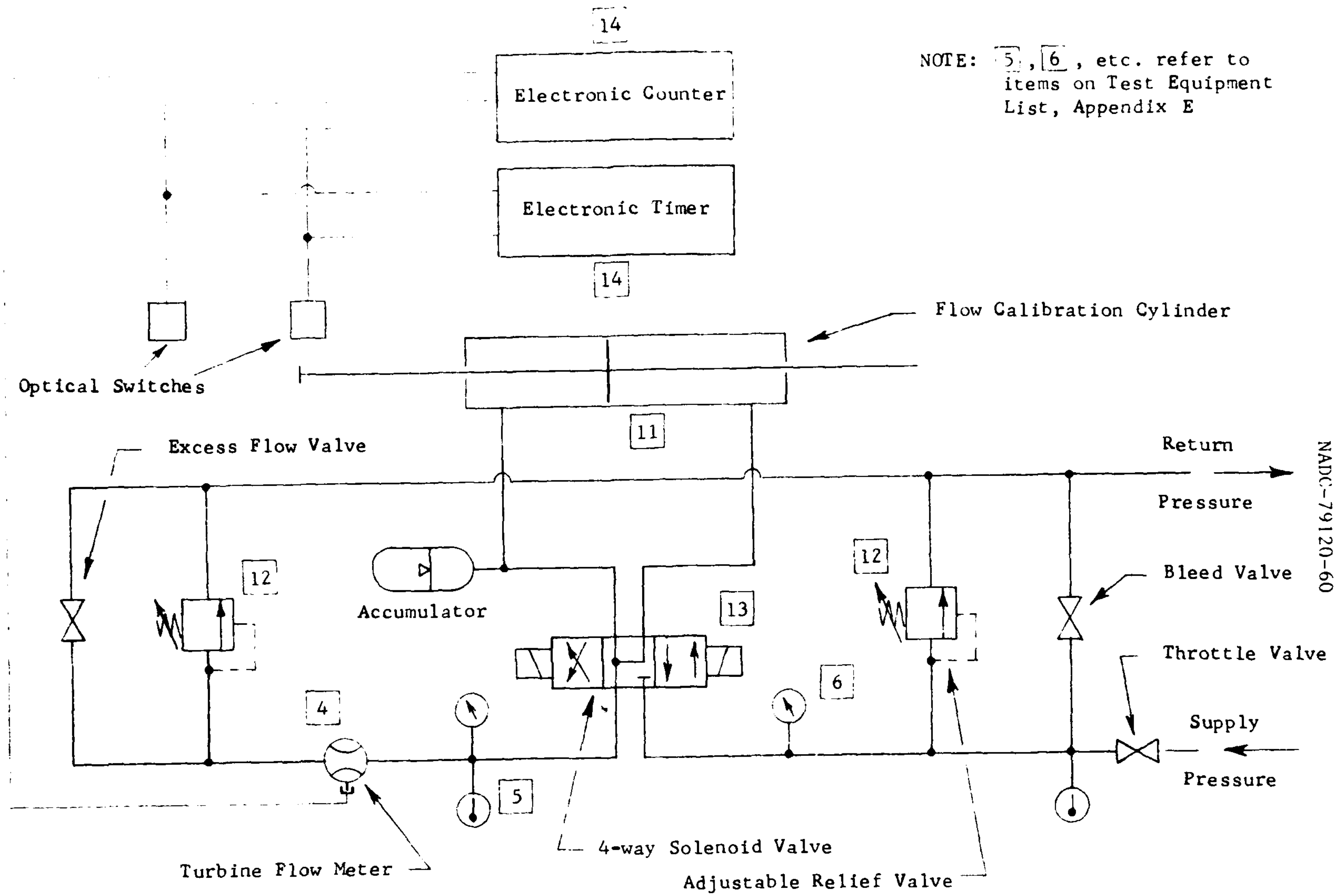


FIGURE 4 Flow meter calibration system schematic

Knowing the net piston area and precise distance between activation of the switches provided fluid volume passed through the flowmeter. Calibrations were conducted with fluid temperatures of +110, +170, and +220°F at pressures from 100 to 200 psi.

2.1.3 Component Tests

This section presents the results of tests conducted on a pump, transmission lines, flow restrictors, servo actuator, control valve, and solenoid valve using MS-6 fluid. Performance comparisons of these components operating with MIL-H-83282 fluid are made in Section 2.3.

2.1.3.1 Pump - The test pump was built by the Aerospace Division of Abex Corporation in Oxnard, California, for the Naval Air Development Center, Reference 2. The unit, identified as M/N AP6V-57, S/N 109422, is a pressure compensated, variable delivery, axial piston pump designed for operation with MIL-H-27601 and MIL-H-83282 fluids. Rated output at 4000 rpm and +240°F inlet fluid temperature is 14 gpm at 7850 psi.

The test plumbing circuit and instrumentation are shown schematically on Figures 5 and 6. Two types of tests were conducted: performance and transient response.

Performance Test Results - Performance test parameters were as follows:

Compensator Setting:	8000 psi
Pump Speed:	4000 rpm
Pressure:	Inlet, 60 psig Discharge, psig (recorded) Case drain, psig (recorded)
Fluid Temperature:	Inlet +110, +200°F Discharge, °F (recorded) Case drain, °F (recorded)
Flow:	Inlet, gpm (recorded) Case drain, gpm (recorded) Discharge, gpm (calculated from return flow)
Input Torque:	lb-in (recorded)

Pertinent areas examined were overall efficiency, heat rejection, and operating temperature characteristics. Performance data are presented in Table 1.

Discharge flow was satisfactory at 4000 psi (only). Case flow decreased to near zero at 6000 psi. Operation at 6000 psi for periods longer than 1 minute would probably damage the pump due to excessive temperature buildup. Overall efficiency peaked at 76.1% near 6000 psi.

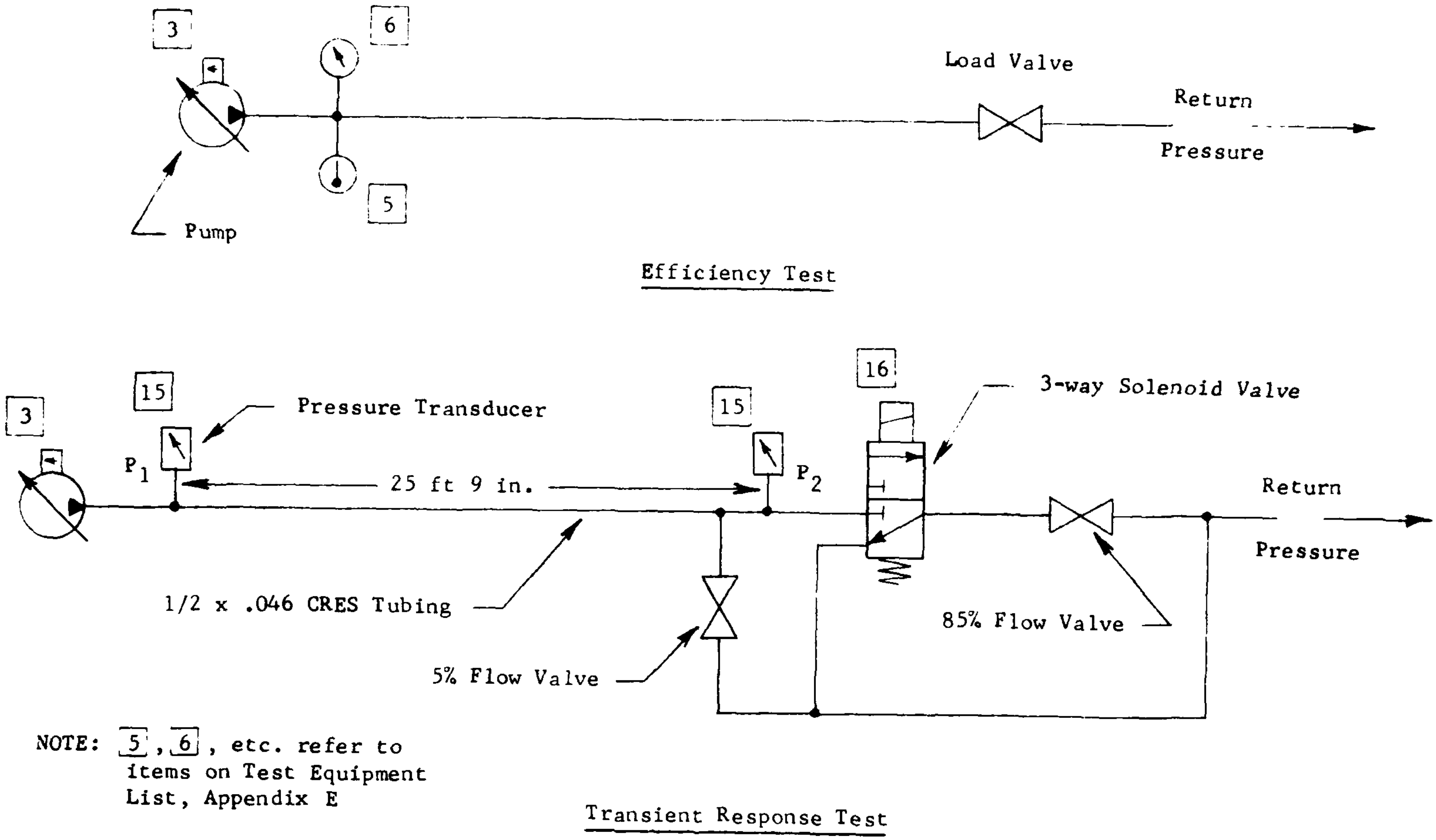


FIGURE 5. Pump test plumbing schematic

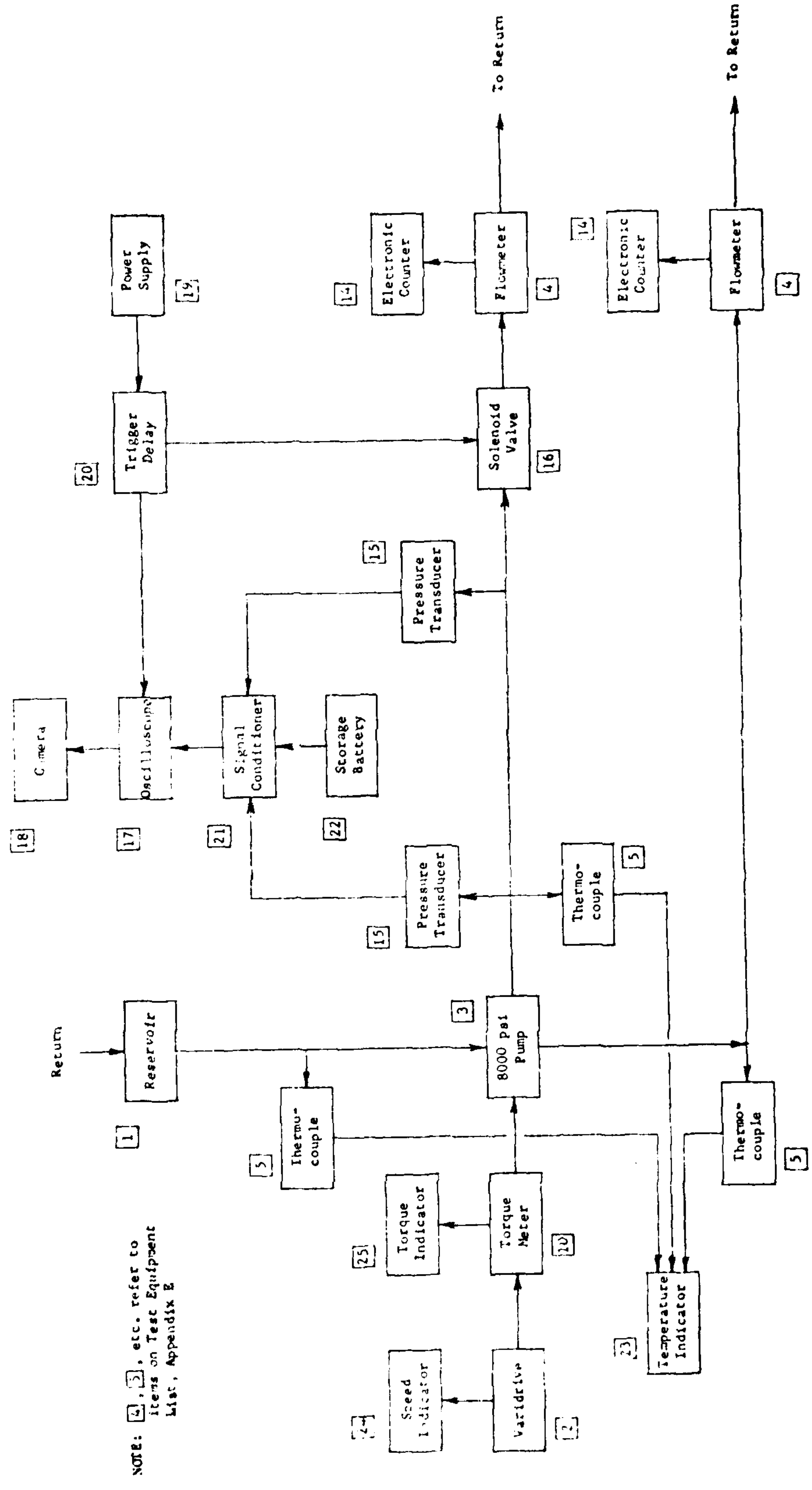


FIGURE 6. Pump test instrumentation schematic

TABLE 1. Pump Performance Data

Pressure, PSIG		Temp. Rise, °F		Flow, GPM		Input Torque, Lb-In	Heat Rejection, BTU/Min.	Overall Efficiency, %
Disch.	Case Dr.	Inlet to Disch.	Inlet to Case Dr.	Disch.	Case Dr.			
4000	240	16	35	14.4	1.16	757	633	68.9
4500	240	18	40	14.3	1.11	838	686	69.6
5000	225	19	43	13.3	.91	855	673	70.8
5500	200	20	45	11.2	.47	761	544	73.4
6000	160	21	87	8.9	.10	640	412	76.1
6400	140	21	95	6.5	.51	535	427	70.4
6500	135	20	93	5.9	.58	507	428	68.6
6600	130	21	93	5.4	.61	484	421	67.7
*6800	80	-	83	0	1.0	135	363	0

PUMP SPEED: 4000 RPM

INLET PRESS: 60 PSIG

INLET TEMP: +200°F

*Compensator set for 8000 psi using MIL-H-83282

A severe instability occurred with inlet fluid temperatures less than approximately +160°F. Operation below +160°F with less than full discharge flow could have damaged the pump. Pressure compensation was affected with +200°F inlet temperatures such that maximum discharge pressure was 6800 psi. The low cut-off pressure was attributed to insufficient fluid bulk modulus which resulted in low precompression pressure in the pump cylinders. The inadequate precompression unbalanced the hanger causing the pump to be de-stroked without compensator control. Pump operating limitations were as follows:

Compensator Setting, psi	Inlet Fluid Temp., °F	Pump Speed, rpm	Discharge Press., psi	Remarks
8000	+110	4000	8000	Pump unstable at full cut-off
8000	+200	4000	6800 (max.)	Pump stable, inadequate precompression pressure
8500	+110	800 to 1800	8000	Pump stable when not at full cut-off
8500	+200	800 to 1700	8000	Stable operation
		1700 to 2400	8000	Unstable operation
		2400 to 2600	8000	Stable operation

An indication of pump wear was obtained by examining debris accumulated in the pump case drain filter. This check was performed following completion of all MS-6 evaluation testing. Total operating time on the pump with MS-6 fluid was 35.4 hours. A patch made of filter debris contained few wear particles, Figure 7. A contamination check of system return fluid produced the following particle count:

Test	Micron Size Range				
Hours	5-15	15-25	25-50	50-100	100+
35.4	20054	750	44	20	0
NAS 1638 (Reference) Class 8)	64,000	11,400	2025	360	64

The pump was partially disassembled at the conclusion of all testing for inspection of the port plate/barrel face wear surfaces. The port plate surface was in excellent condition. The barrel face was in good condition except for a dark purple discoloration on the outer annular area, Figure 8. The barrel was 4130 steel; the face was plated with shoe bronze (95% copper, 5% tin). The discoloration was believed to be copper oxide formed as a result of high local temperatures. The high temperatures were probably caused by marginal lubricity of MS-6 due to foaming.

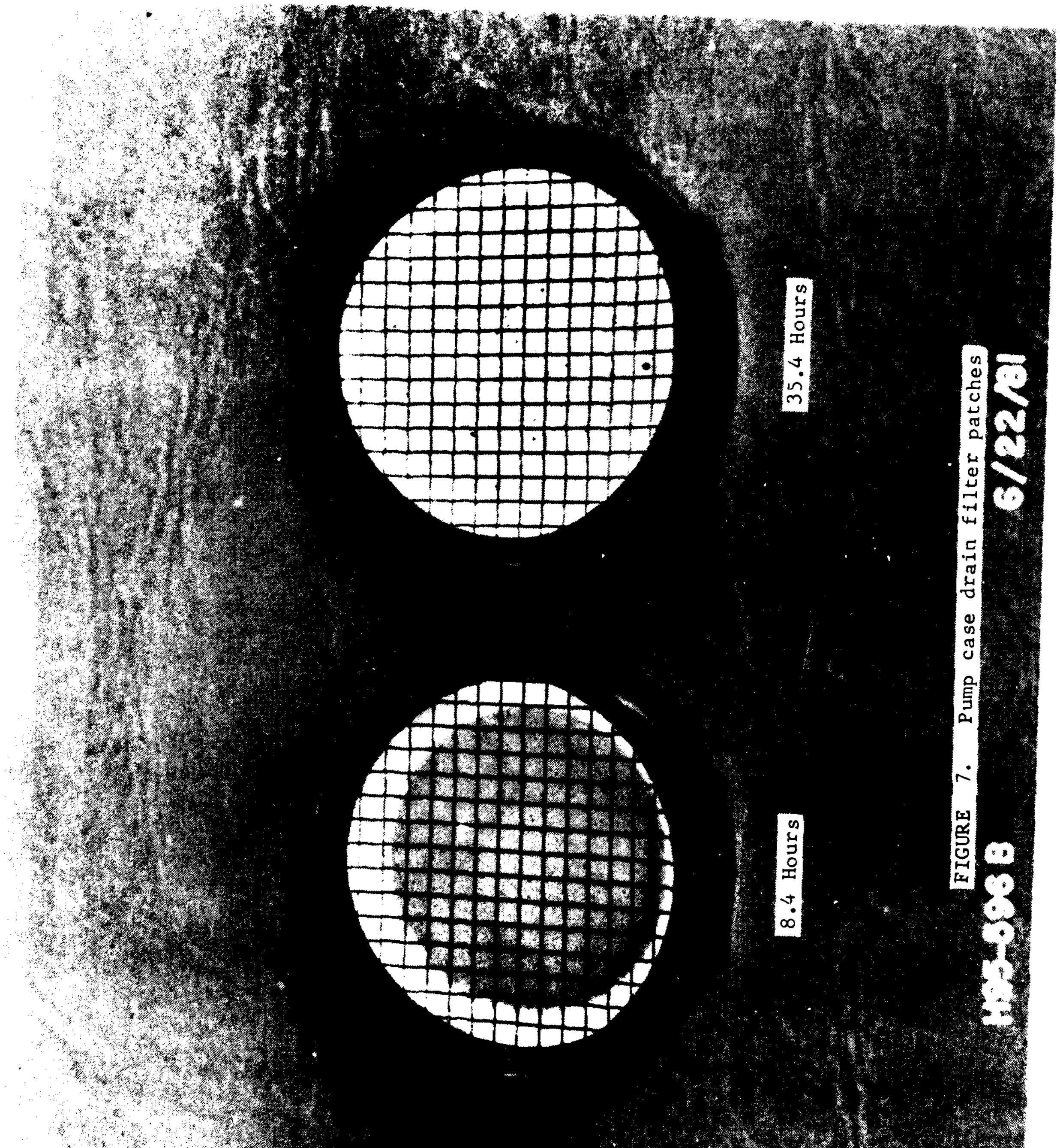


FIGURE 7. Pump case drain filter patches

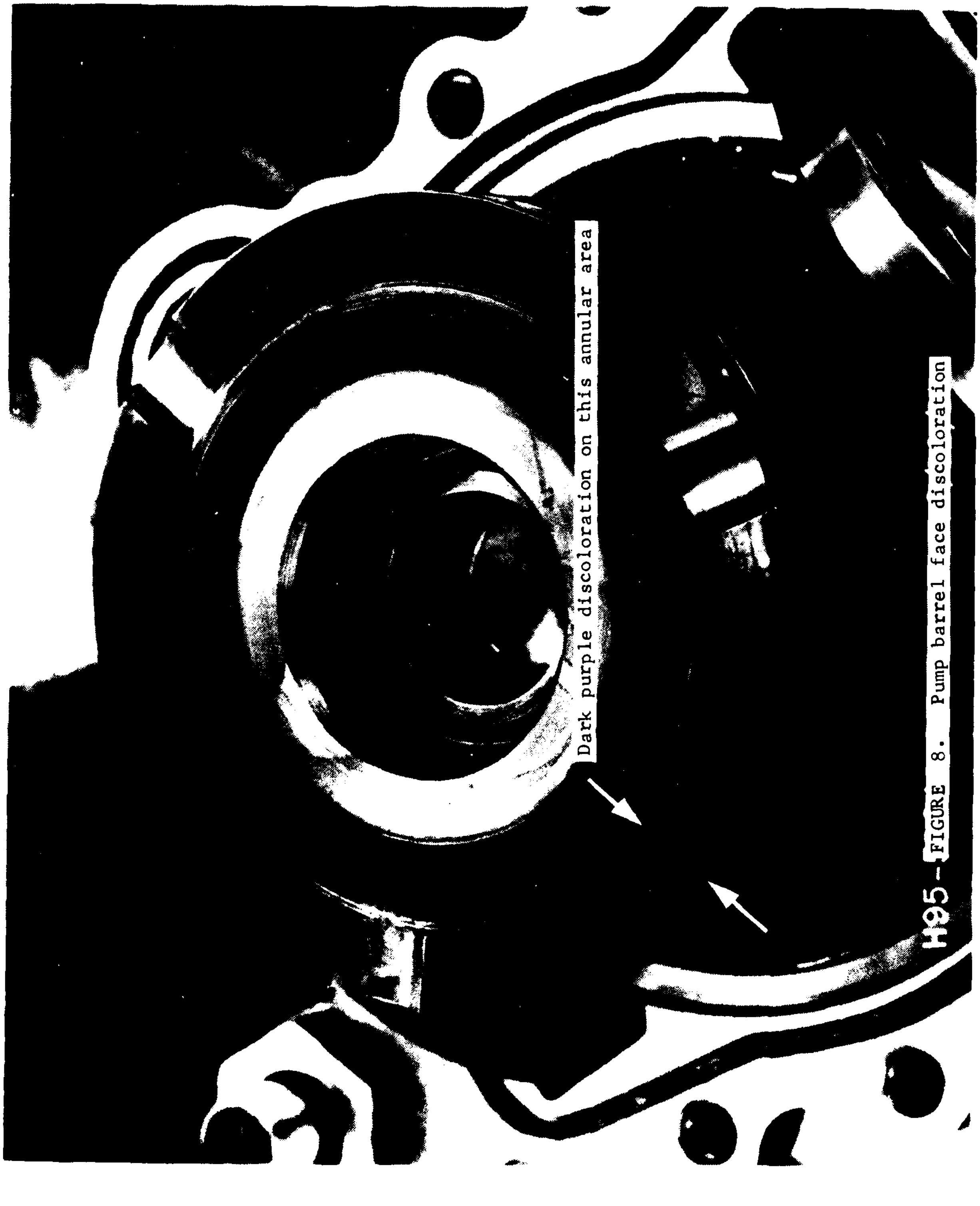


FIGURE 8. Pump barrel face discoloration

A brief investigation was conducted to try to establish the temperature at which the discoloration formed. A copper strip was put in a beaker of MS-6 fluid, covered, and placed in an oven. The results are summarized below:

	Oven Temperature	Time at Temperature	Remarks
*	+350°F	16 hr.	No effects observed
	+400°F	4.5 hr.	No effects observed
	+450°F	21.5 hr.	MS-6 fluid amber color, some discoloration on copper strip
*	+500°F	1.5 hr.	MS-6 fluid black color, severe discoloration on copper strip

*New MS-6 fluid and copper strip used.

The copper oxide thus appears to have formed on the pump barrel face at temperatures between +400 and 500°F.

Transient Response Test Results - Discharge flow was varied from 5% to 90%, then 90% to 5% of rated pump output by means of a fast acting solenoid valve and pre-set load valves. Discharge pressure transients were sensed by a strain gage type transducer and observed on an oscilloscope. The test parameters were:

Compensator Setting:	8000 psi
Pump Speed:	4000 rpm
Pressure:	Inlet, 60 psig Discharge (recorded)
Fluid Temperature:	Inlet, +200°F
Flow Change:	Discharge, 12.6 to 0.7 gpm 0.7 to 12.6 gpm

Photographs of the transient pressure wave form are shown on Figure 9; pump ripple at zero discharge flow is also shown. Transient response results were as follows:

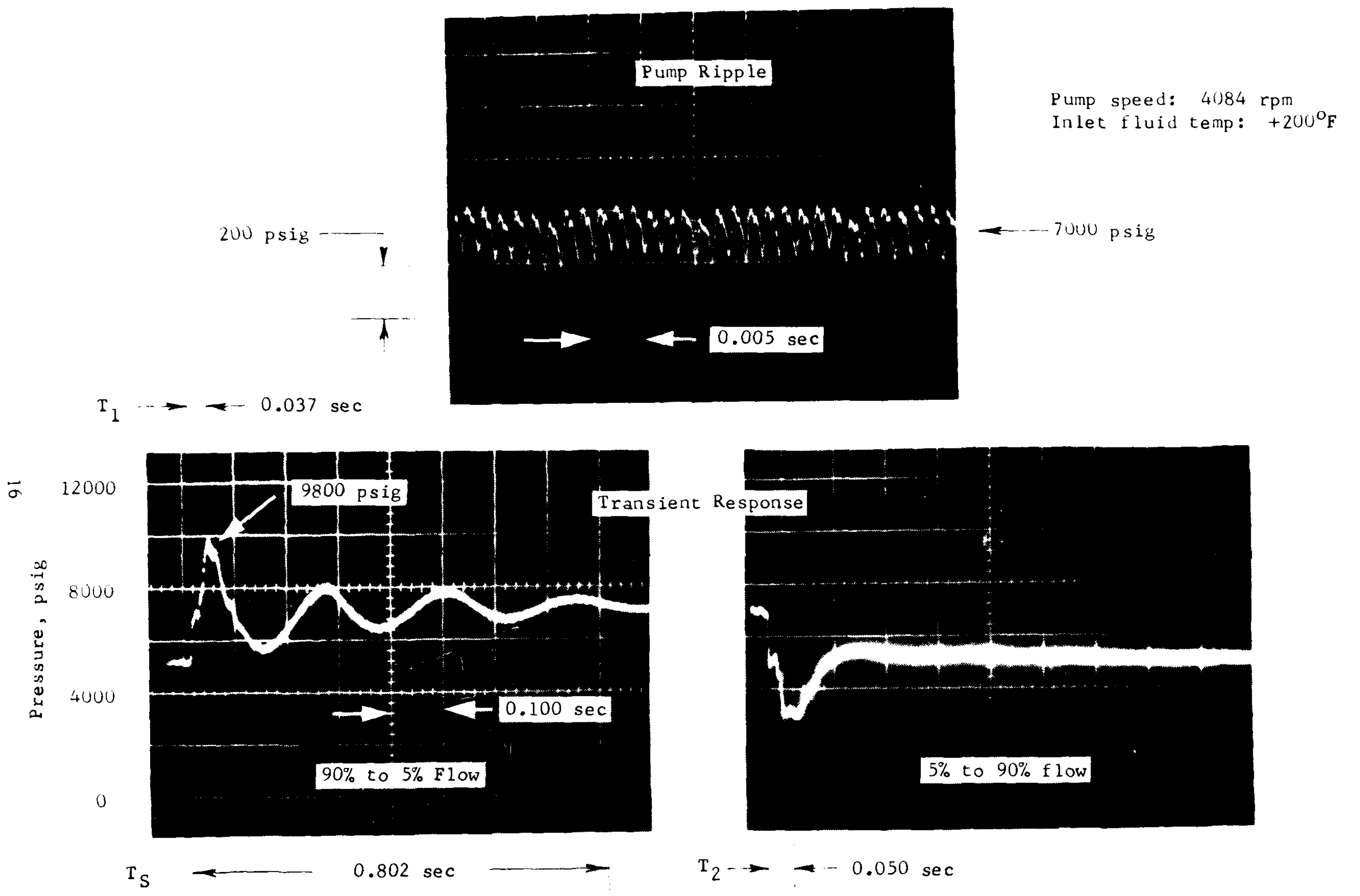


FIGURE 9. Pump ripple and transient response

Condition	Observed Time, sec.		Pressure, psig
90% to 5% flow	T_1	0.037	9800 (overshoot)
5% to 98% flow	T_2	0.050	-
Stability	T_S	0.802	-
Ripple	-		200 (peak-to-peak)

Since pump performance was poor using MS-6 fluid, results of the transient response tests were questionable. If the pump were designed to operate with MS-6, pressure overshoot, stability, and ripple would all be affected.

2.1.3.2 Transmission Line

2.1.3.2.1 Pressure Drop - The test setup is shown pictorially on Figure 10 and schematically on Figure 11. Three tubing sizes were evaluated. All tubing was 21-6-9 CRES and straight.

Tube O.D., in.	Wall Thickness, in.	Test Length, in.
3/16	.020	12
1/4	.023	24
1/2	.046	84

A thermocouple and pressure tap were located at the entrance and exit of each test length. Fiberglass insulation was used to minimize fluid temperature changes. Pressure drop in the test tubing was measured by a differential pressure gage with a 100 psi full scale dial. Flow was sensed by a turbine meter with readout on an electronic frequency counter. Flow was measured at return pressure. MS-6 density data given in Reference 1 was used to calculate flow at operating pressure. The test parameters were:

Operating Pressure:	4000, 6000, 8000 psi
Pressure Drop:	Recorded (100 psid max.)
Fluid Flow:	Recorded
Fluid Temperature:	+110 $\pm 2^{\circ}F$ +200 $\pm 2^{\circ}F$ (Average of entrance and exit temperatures in test section)

Pressure drop data at 8000 psi are given on Figure 12. Data recorded at 4000 and 6000 psi were used for fluid viscosity determinations (see Section 2.2.1). Data on Figure 12 cover a fluid velocity range of 5 to 25 ft/sec. The relatively high viscosity of MS-6 caused all flow to be laminar.

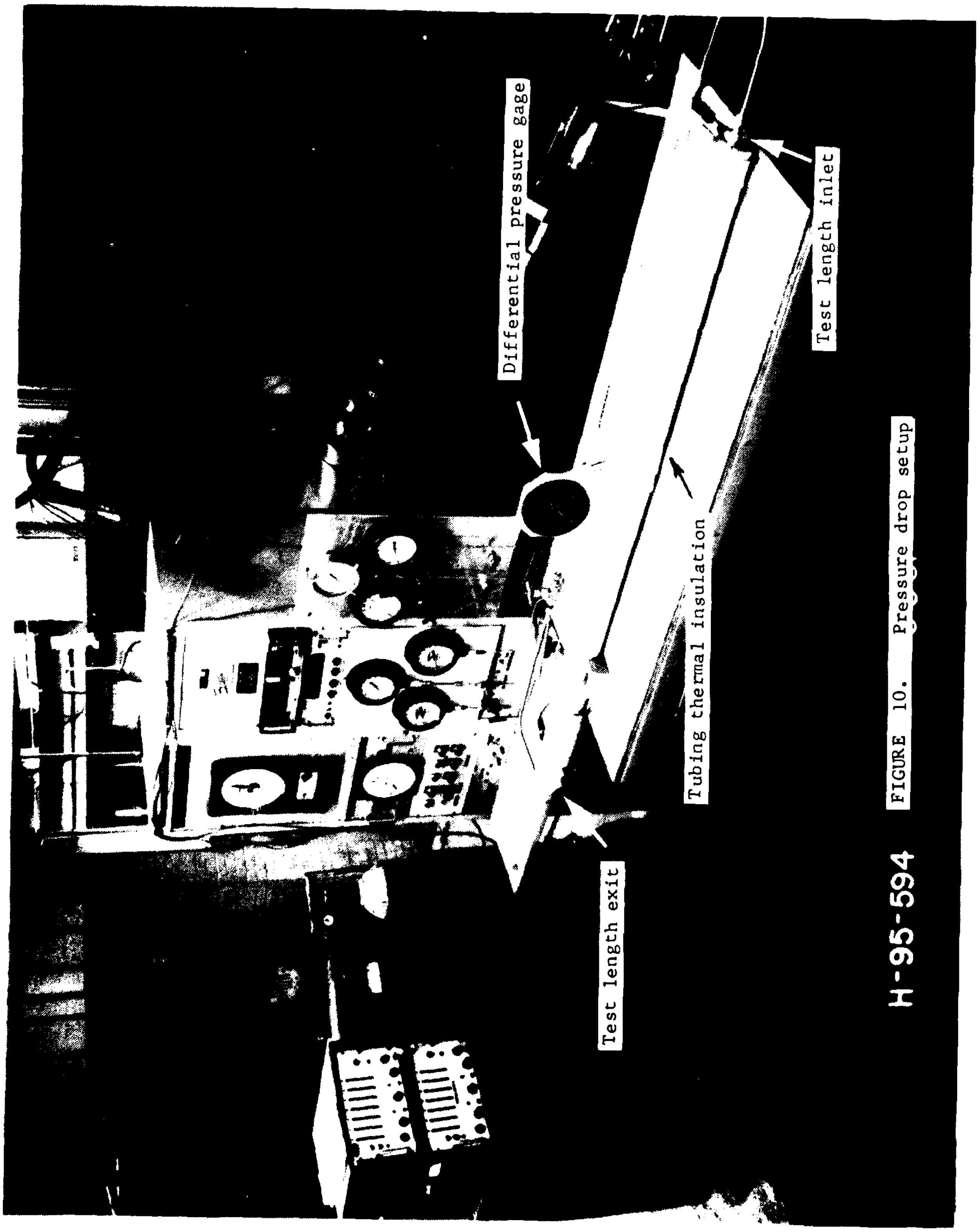


FIGURE 10. Pressure drop setup

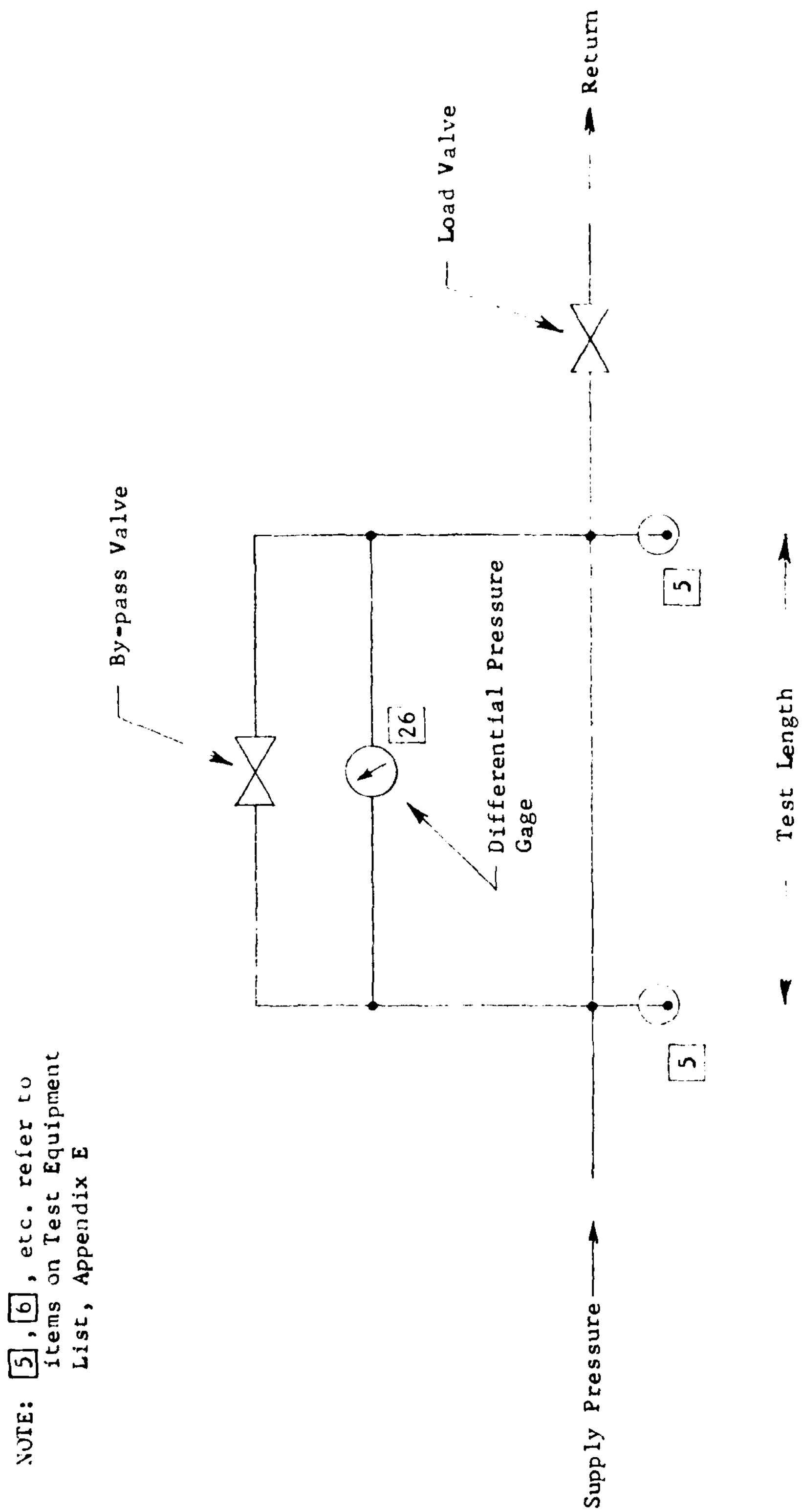


FIGURE 11. Pressure drop setup schematic

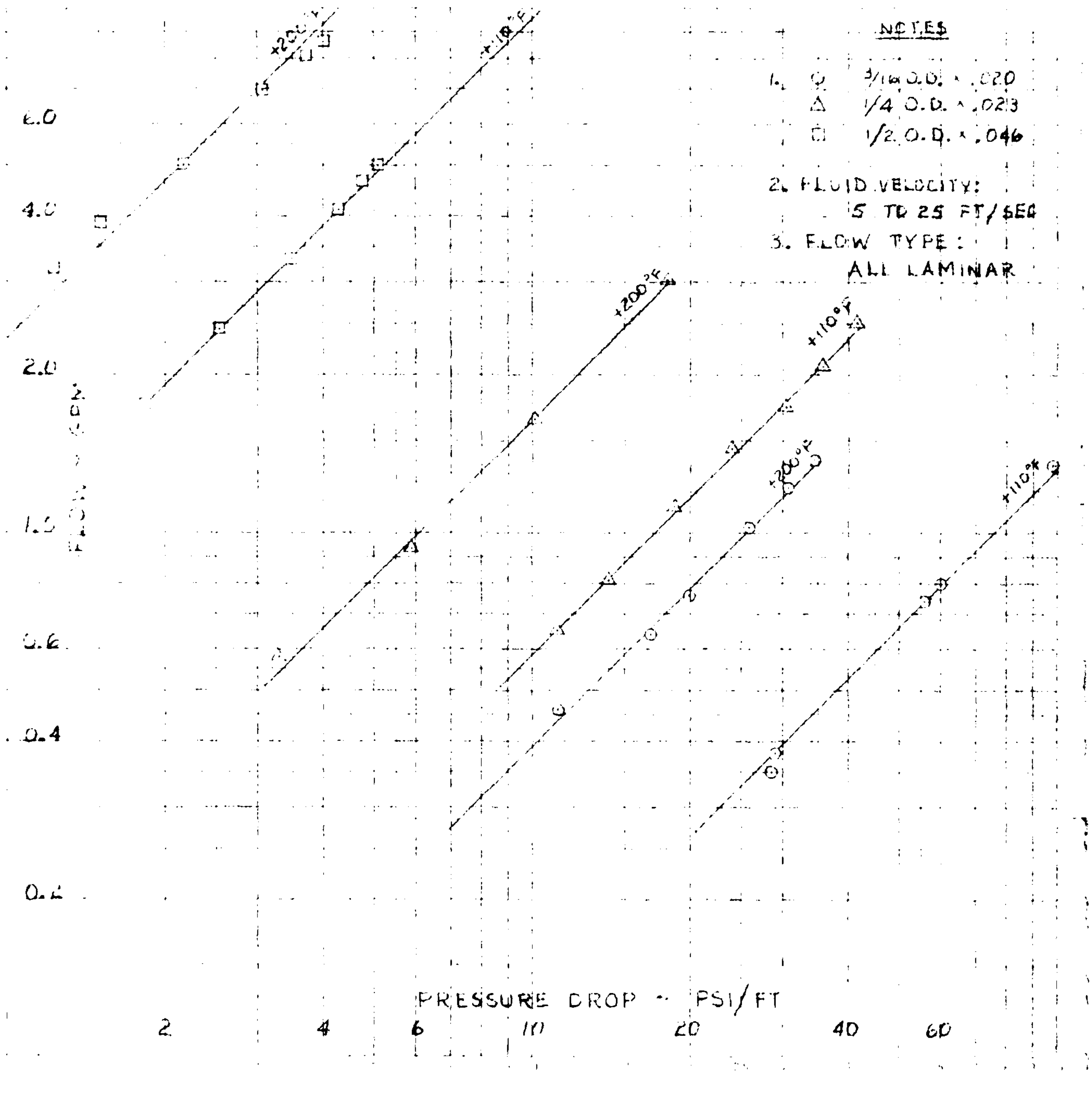


FIGURE 12. Tubing pressure drop at 8000 psi

2.1.3.2.2 Pressure Surge - Pressure surges were created by stopping fluid flow with a fast acting solenoid valve, Sterer P/N 15390-1, Reference 4. The test setup is shown pictorially on Figure 13 and schematically on Figure 5. Two strain gage type pressure transducers were used to measure surges. One transducer (P_1) was located near the discharge port of the pump, the second (P_2) was immediately upstream of the solenoid valve. Readout of the two transducers was simultaneous on a dual beam oscilloscope. Pump speed was varied to obtain desired fluid velocities. The test parameters were:

Operating Pressure:	4000, 8000 psi
Fluid Temperature:	+110, +200°F
Fluid Velocity:	15, 25 ft/sec.
Tubing:	1/2 O.D. x .046 21-6-9 CRES 25 ft. 9 in. between transducers

The largest surge occurred at P_2, Figure 14. Typical surges are shown in Figure 15. The P_2 transient wave form contained two discernible features: (1) surge due to the water hammer effect, and (2) overshoot due to pump dynamics. The water hammer surge was calculated using,

$$\Delta P = V\sqrt{\rho B_e} \qquad \text{(Reference 4)}$$

where,

ΔP = surge magnitude

V = fluid velocity

ρ = fluid mass density

B_e = effective bulk modulus (fluid and mechanical compliance)

The measured water hammer surge agreed well with the calculated surge as shown below.

Operating Pressure, psi	Fluid Temperature, °F	Fluid Velocity, ft/sec	Measured Water Hammer, psi	Calculated Water Hammer, psi
4000	+110	15	767	788
		25	1279	1313
	+200	15	643	693
		25	1072	1156
8000	+110	15	823	889
		25	1371	1481
	200	15	709	799
		25	1182	1331

FIGURE 13. Pressure surge setup

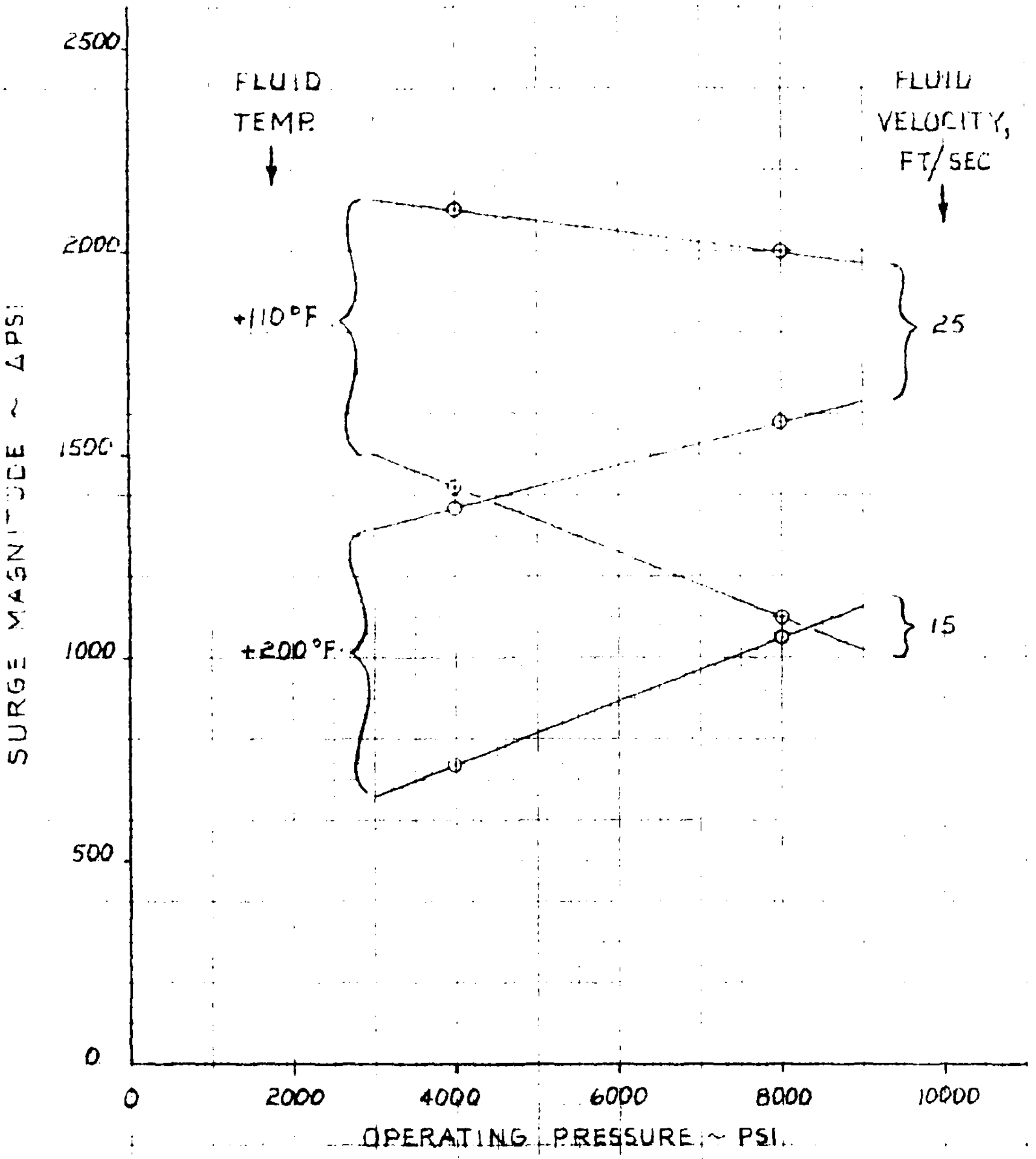
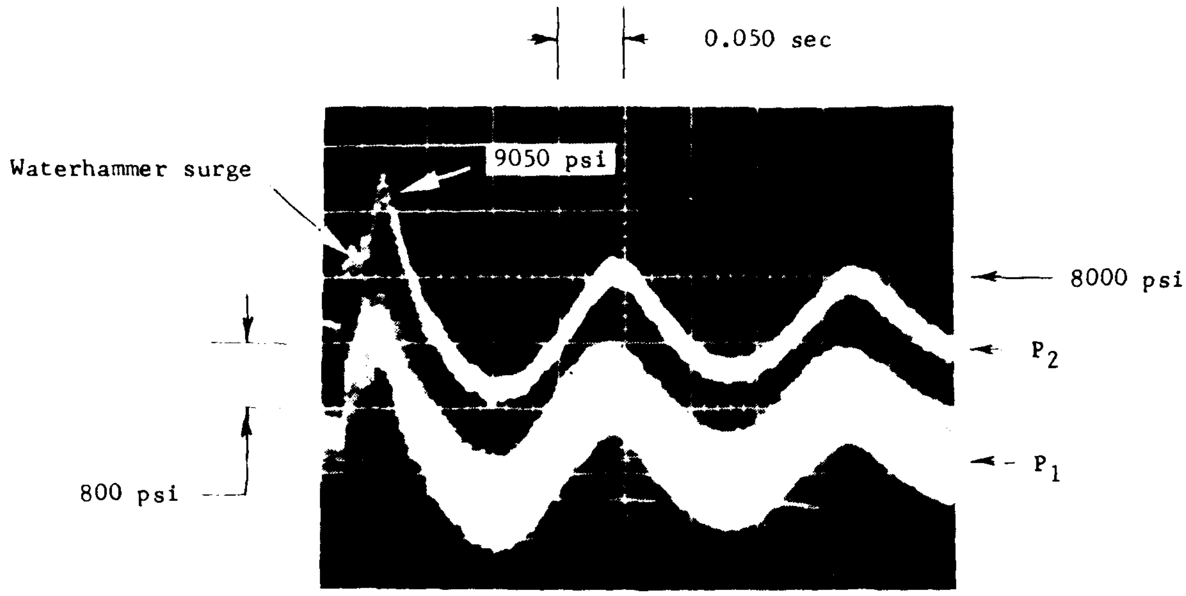
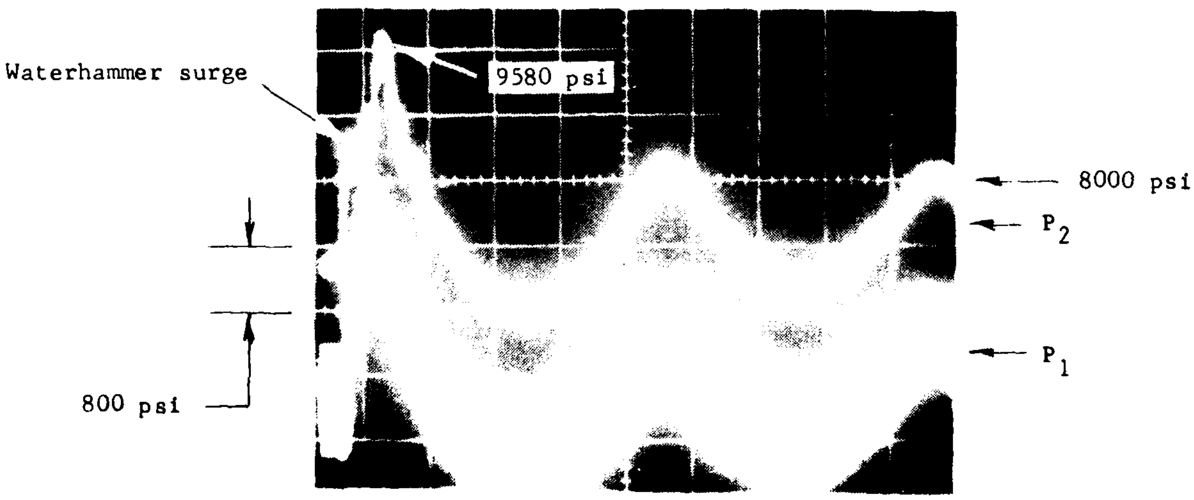


FIGURE 14. PRESSURE SURGE MAGNITUDE AT P_2



Fluid Velocity: 15 ft/sec
Fluid Temperature: +200°F



Fluid Velocity: 25 ft/sec
Fluid Temperature: +200°F

FIGURE 15. Pressure surge wave form

Pressure overshoot caused by pump dynamics was affected by pump performance. Pump operation was sensitive to fluid temperature and speed, and was occasionally erratic. Validity of the overshoot data was therefore questionable.

2.1.3.3 Flow Restrictor - Two flow restrictors designed for MIL-H-83282 fluid at 8000 psi were evaluated:

Restrictor P/N	Rated Flow, gpm	Remarks
Lee P/N REFX0380250AB	2	Previously used in 200 hour endurance test, Ref. 9
Lee P/N JEFX0483000A	4	Previously used in 150 hour compatibility test, Ref. 5

The test parameters were:

Differential Pressure:	7800 psi
Inlet Fluid Temperature:	+110°F
Return Flow:	Recorded

Flow at 8000 psi was calculated from return flow using MS-6 density information given in Reference 1. The results were:

Restrictor Size	Flow Rate Using MS-6
2 gpm	1.54 gpm
4 gpm	3.28 gpm

2.1.3.4 Servo Actuator - The test actuator was designed and fabricated by Rockwell International for the tests reported in Reference 10. The servo actuator, identified as P/N 4212-01, is a dual tandem unit with one section designed to operate at 6000 psi, the other at 9000 psi. The 9000 psi section (only) was used for the MS-6 tests. Basic actuator data are:

Piston travel (full)	8.2 in.
Piston diameter	2.5 in.
Piston net area	2.84 in^2
Output force @ 8000 psi	22,700 lb

The input lever on the servo actuator was controlled by a small driver actuator powered by a 1000 psi supply system containing MIL-H-5606 fluid. The servo actuator was operated open loop; the driver actuator was run closed loop. The servo actuator was installed in a mass load fixture as shown on Figure 16. The load mass weighed 2000 lb. and was supported in a rigid structure designed to minimize jig dynamics, Reference 2.

The test plumbing circuit and instrumentation are shown schematically on Figures 17 and 18. Two types of tests were conducted: frequency response and maximum velocity.

Frequency Response Test Results - Test parameters were:

Supply Pressure:	8000 psi
Fluid Temperature:	Inlet, +110°F Cylinder, +130°F(est.)
Input Lever Motion:	±0.050 in. (sinusoidal)
Input Frequency Range:	1 to 25 Hz
Open Loop Gain:	27.9 in^3/sec/in.

Open loop response is shown on Figure 19. Resonant frequency of the servo actuator/system was 19.5 Hz. Gain margin at resonance was 3 db.

Maximum Velocity Test Results - Hard-over inputs were applied to the servo actuator lever. Stroke time was measured on a two channel strip chart recorder: one channel was actuator piston position, the other was a 10 Hz time signal. Test parameters were:

Supply Pressure:	8000 psi
Fluid Temperature:	+110°F (inlet)
Input Lever Travel:	±0.086 in. (square wave)

Maximum piston velocity was 4.21 in/sec.

2.1.3.5 Flow Control Valve - The test valve was a 4-way proportional flow control design built for the program reported in Reference 2. The unit, identified as P/N 4212-03-11, was a spool/sleeve configuration containing 'V' shaped flow notches on 2 lands. Basic valve data were:

Spool Land Diameter	0.250 in.
Spool Travel (rated)	±0.086 in.
Design Overlap	±0.0025 in.

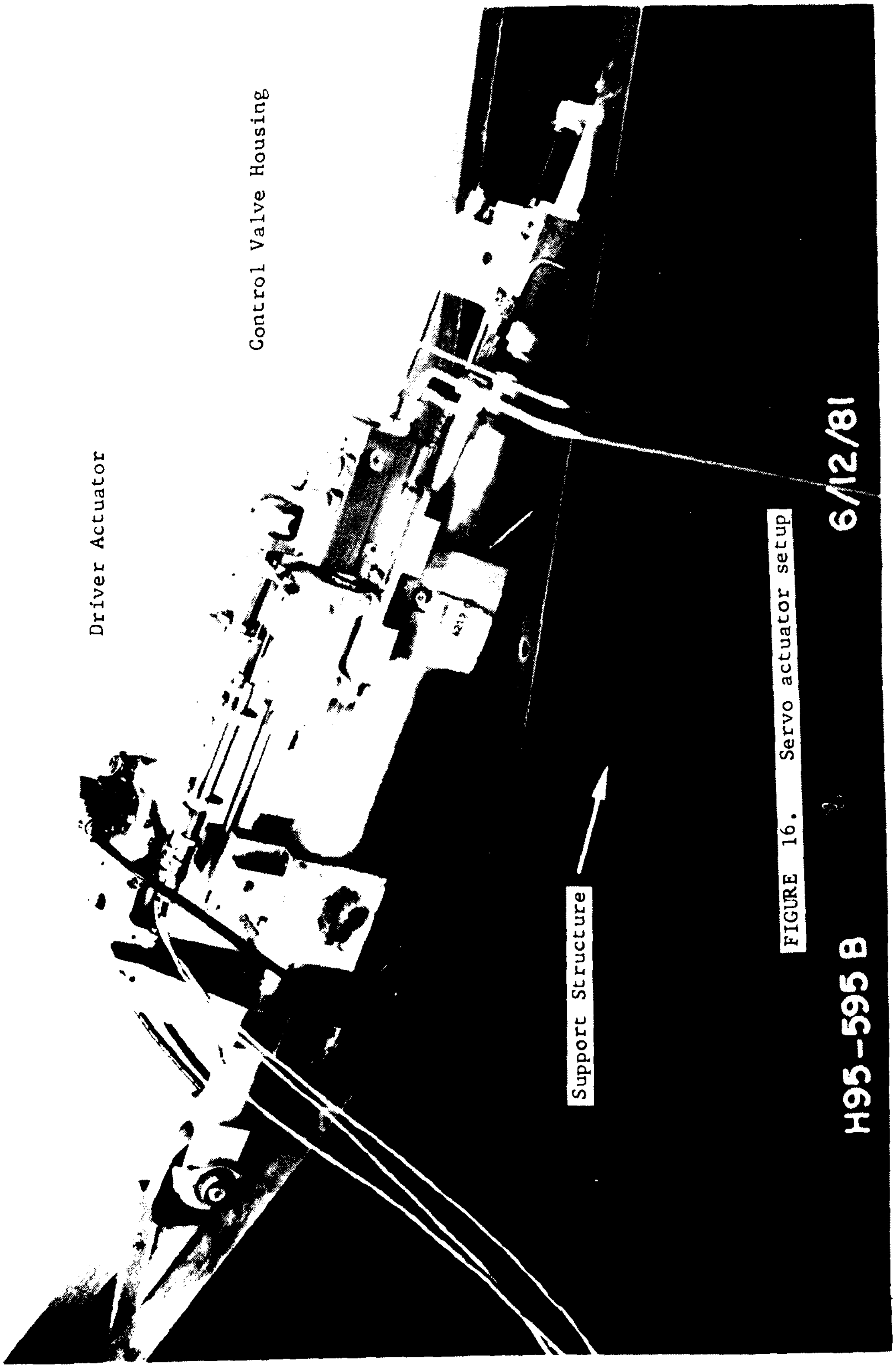


FIGURE 16. Servo actuator setup

NOTE: 27, 28, etc. refer to items on Test Equipment List, Appendix E

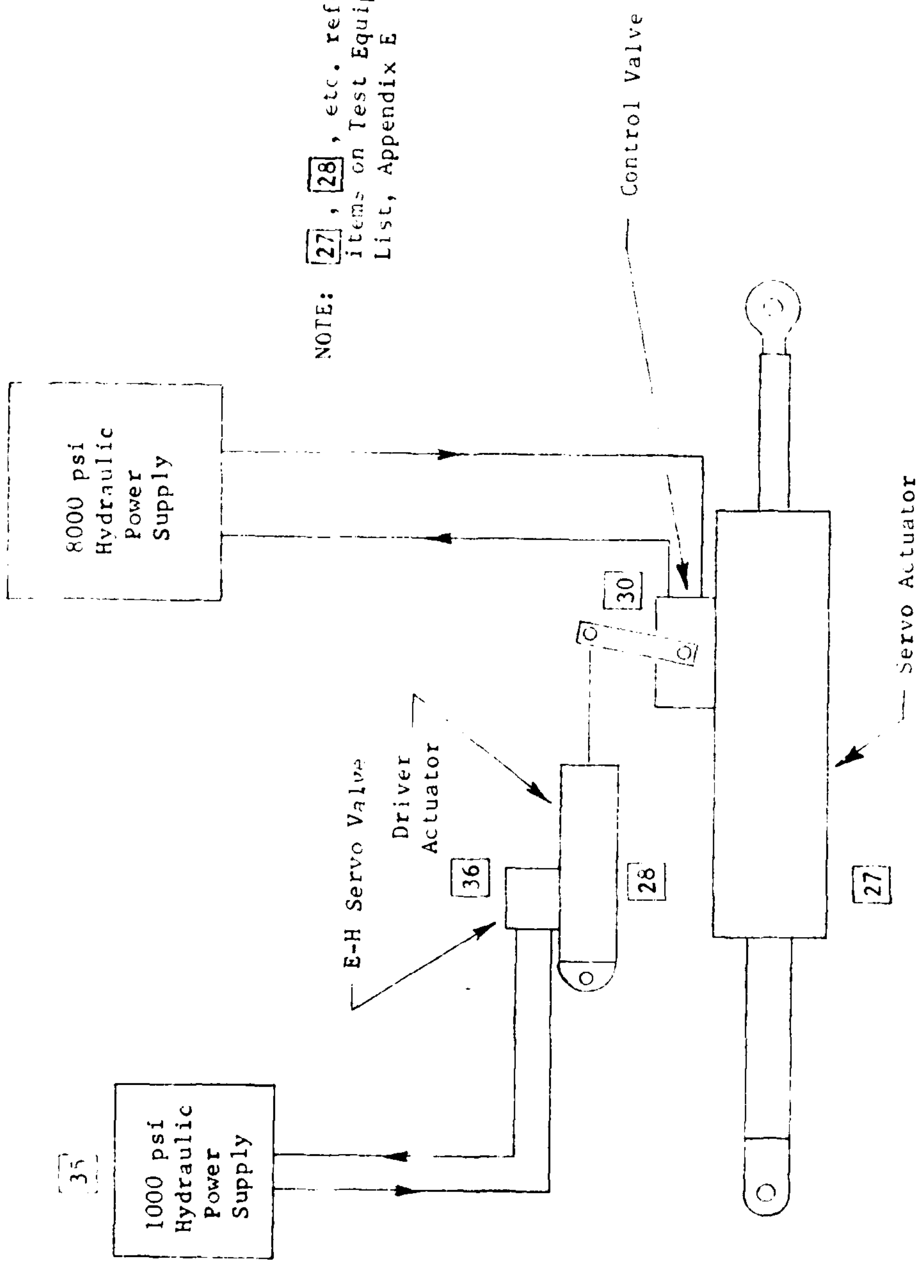
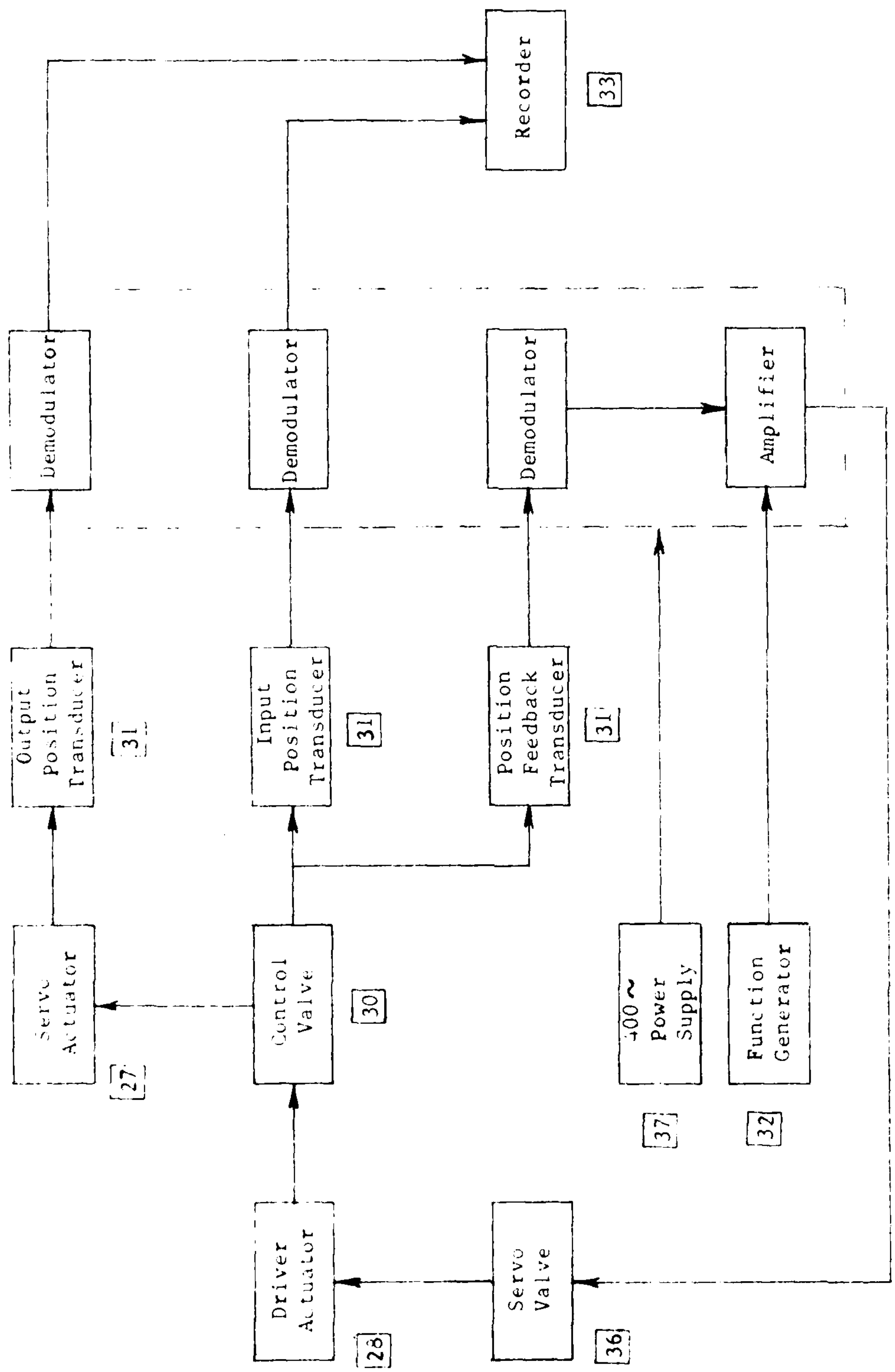


FIGURE 17. Servo actuator plumbing schematic



NOTE: 30 , 31 , etc. refer to items on Test Equipment List, Appendix E

FIGURE 18. Servo actuator instrumentation schematic

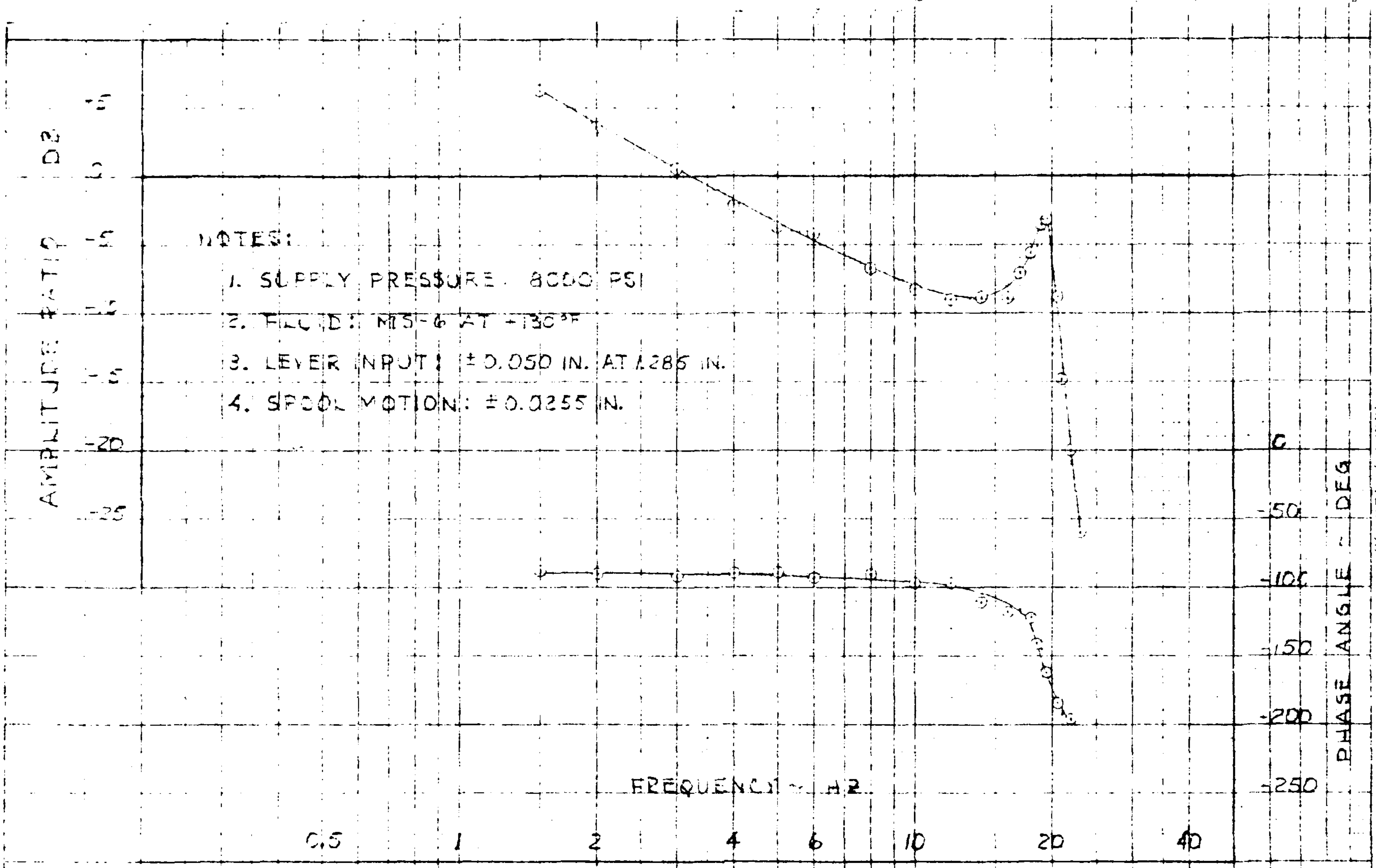


FIGURE 19. SERVO ACTUATOR OPEN LOOP RESPONSE

The valve was installed in a housing mounted on the servo actuator, Figure 16. Flow paths between the housing and actuator were blocked by replacing C1 and C2 porting tubes with undrilled tubes. Access to cylinder ports C1 and C2 was through holes machined in the housing. A turnbuckle and dial indicator were used to control and measure valve input lever position. Valve operating characteristics were determined using the test circuit shown on Figure 20. Flow gain, pressure gain, and internal leakage tests were conducted. Test parameters were:

Supply Pressure:	8000 psi
Inlet Fluid Temperature:	
Flow Gain	+130°F
Pressure Gain	+110°F
Internal Leakage	+110°F
Cylinder Ports C1 and C2:	
Flow Gain	Interconnected
Pressure Gain	Blocked
Internal Leakage	Blocked
Return Flow:	
Flow Gain	Recorded
Internal Leakage	Recorded
Cylinder Port Pressures C1 and C2:	
Pressure Gain	Recorded

Test Results - Valve flow gain and pressure gain are shown on Figures 21 and 22. Both curves have forms typical of those obtained with hydrocarbon base fluids. Flow gain averaged 154 $in^3/sec/in$; pressure gain averaged 2.67×10^6 psi/in.

Internal leakage was so low due to the high viscosity of MS-6 that the planned test data could not be taken. The range of the turbine flowmeter used was 0.01 to 1.0 gpm. Maximum leakage occurred at null and was approximately 0.008 gpm (30 cc/min). Leakage at spool positions of ± 0.086 in. was estimated to be less than 0.003 gpm (10 cc/min).

2.1.3.6 Solenoid Valve - Internal leakage in a 3-way solenoid valve was measured. The valve checked was Sterer P/N 15390-1 (see Section 2.1.3.2.2). The test parameters were:

Supply Pressure:	8000 psi
Fluid Temperature:	+90°F
Electrical Power: (28 VDC)	off, then on
Port Information:	
P	pressurized
C1	closed
R	open

NOTE: [5], [6], etc. refer to items on Test Equipment List, Appendix E

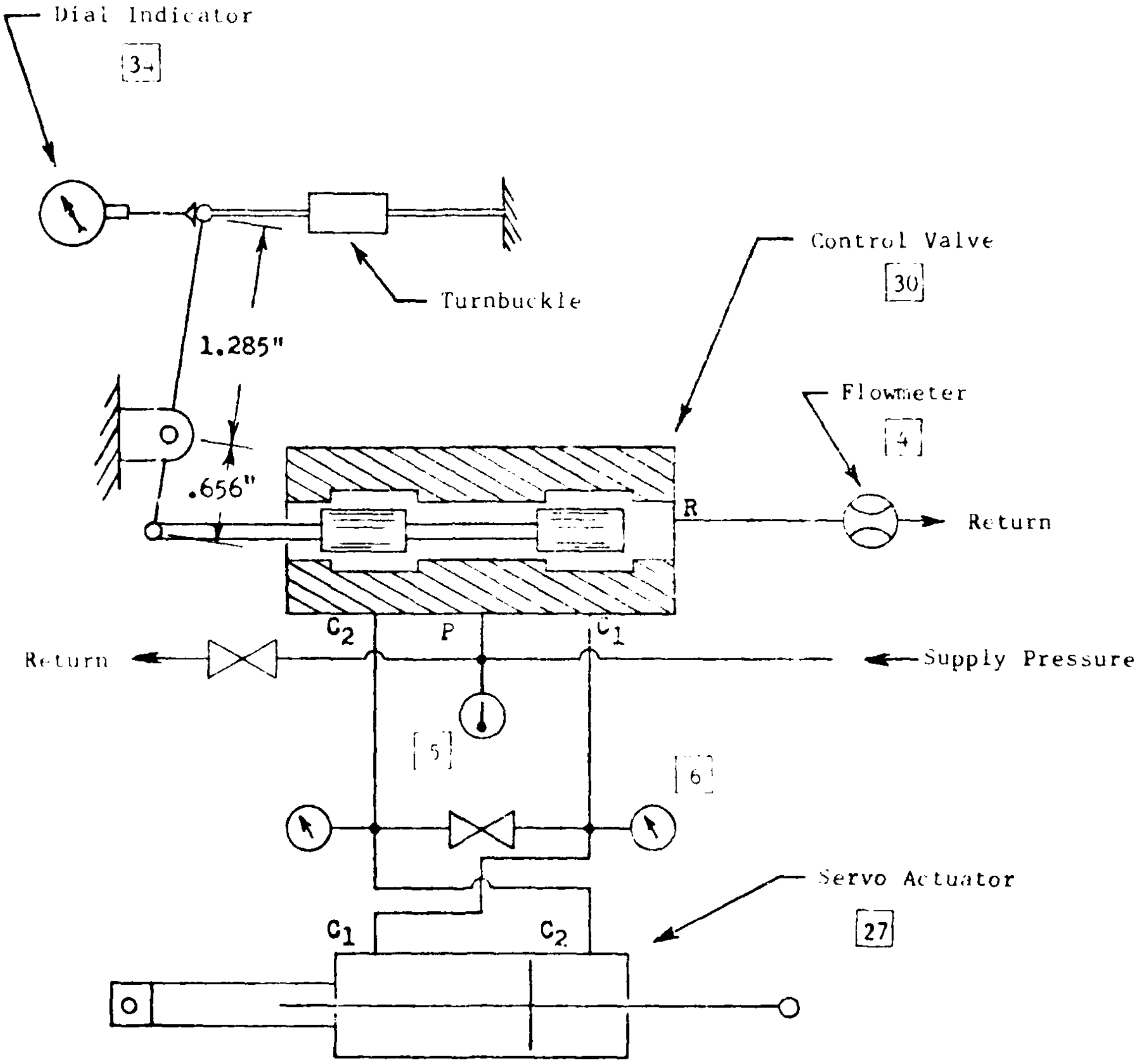


FIGURE 20. Flow control valve setup schematic

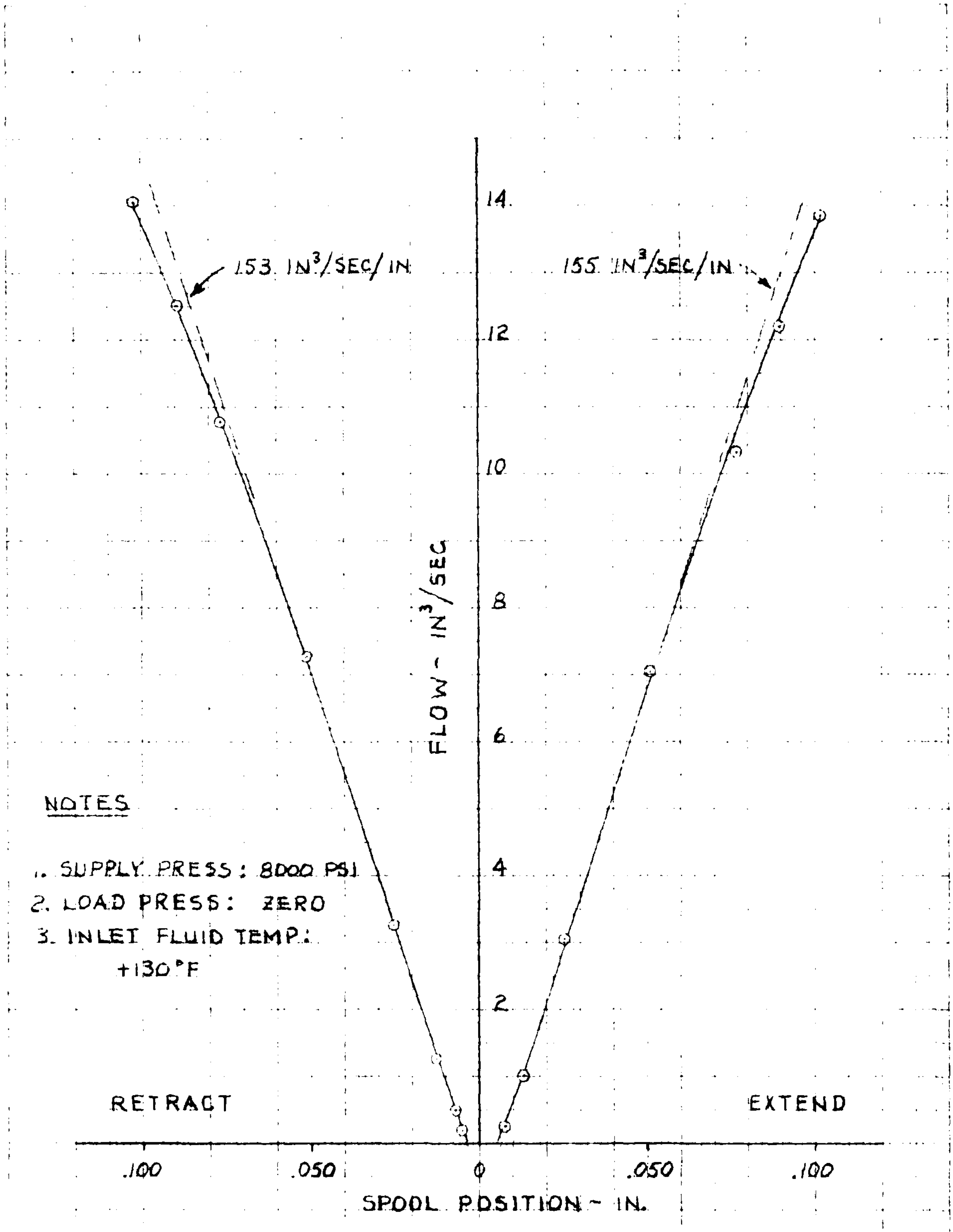


FIGURE 21. FLOW GAIN

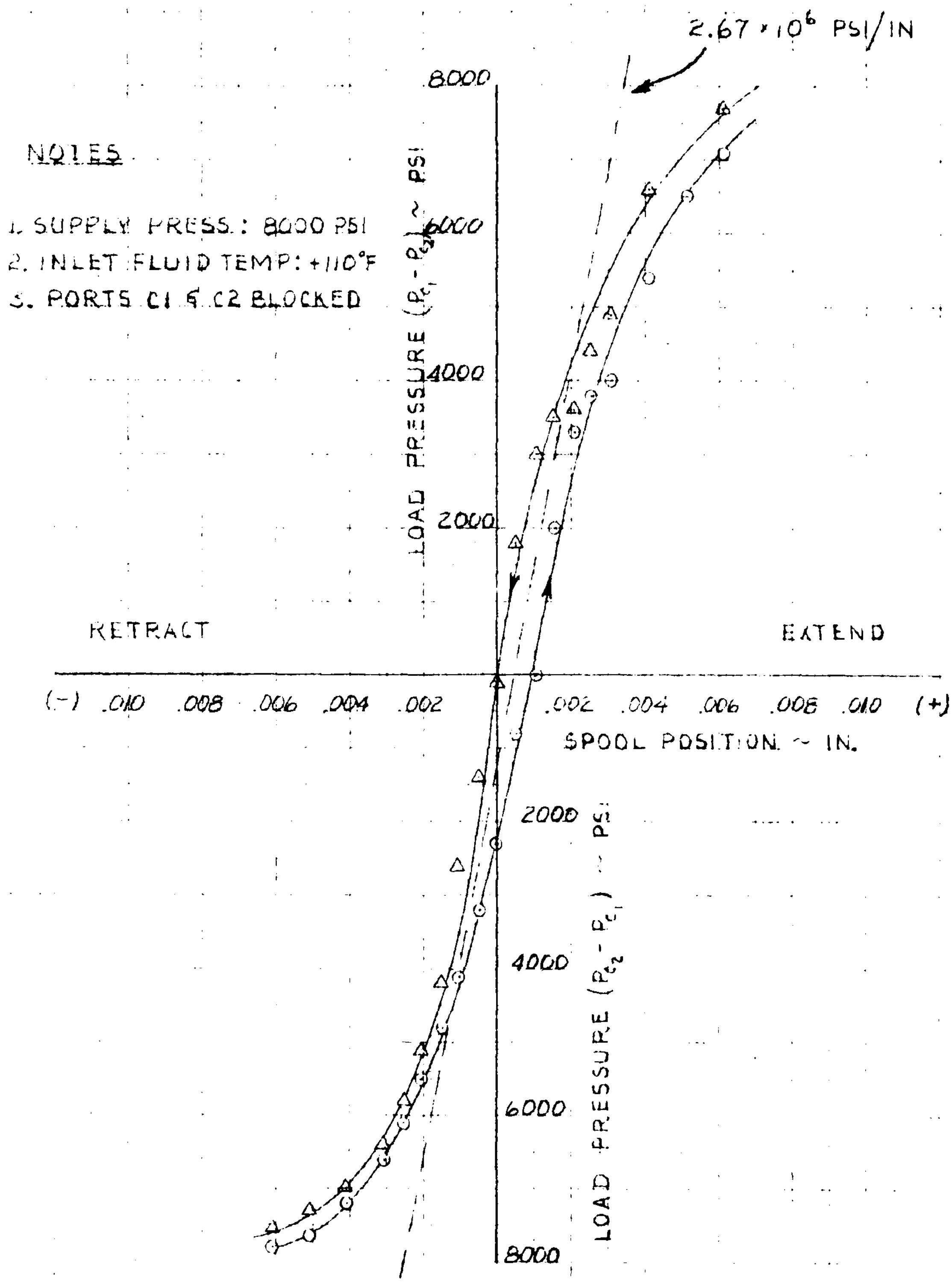


FIGURE 22. PRESSURE GAIN

Test results were as follows:

Valve, off/on	Internal Leakage, drops/min
off	16
on	10

2.1.4 Reservoir Comments

The test system reservoir was an air pressurized design containing a swirl chamber for deaeration and baffles for inverted flight. The reservoir is normally pressurized by engine bleed air. Reservoir capacity is 5.7 gallons.

The reservoir was pressurized with 60 psig of argon for the MS-6 tests. At the conclusion of a day's testing, the system was shutdown, and reservoir pressure was released. At this time, system fluid temperature was usually +160°F or higher, and reservoir fluid level was near the mid-point as indicated by a sight gage. The next morning before system startup, the sight gage always indicated an over-full condition (with room temperature fluid). After system startup, the fluid level returned to normal. The increase in fluid level between shutdown and startup was believed to result from gradual release of dissolved gas in system fluid. Release of the gas produced a slight pressure (perhaps 2 psi) which pushed fluid into the vented reservoir.

2.1.5 Foam Observations

Pump damage was a concern during the evaluation tests. After one period of especially rough operation, the case drain filter bowl was removed to make a visual check for pump wear debris. This was done within 5 minutes after system shutdown. The filter element was covered with white foam and the bowl contained less than 1/4 of its normal fluid volume.

Patches were taken of case drain and return filter debris at the conclusion of MS-6 testing. The bowls were removed in the morning after a day of test runs. No foam was observed on either filter element. Both bowls contained less than 1/2 of their normal quantity of fluid.

System plumbing was changed several times to install various circuits during the course of testing. When tube fittings were loosened for the first time, an abnormally large quantity of fluid was released because of slight pressures within the system. This occurred even when changes were made in the morning before the system was operated.

The foregoing comments are related to the tendency of MS-6 to absorb gases and foam. This characteristic is undesirable in hydraulic systems because of degrading effects on fluid bulk modulus, lubricity, and cavitation.

2.2 FLUID PROPERTY DETERMINATIONS

2.2.1 Viscosity

The increase in viscosity of MS-6 with pressure was determined from data taken during the transmission line tests, see Section 2.1.3.2.1. Under laminar flow conditions, fluid viscosity is related to tubing pressure drop by the equation,

$$\nu = \frac{d^4 \, \Delta P}{2.915 \, \rho \, Q} \qquad \text{(Ref. 3)}$$

Where, ν = kinematic viscosity, centistokes
d = tube inside diameter, in.
ΔP = pressure drop, psi/ft
ρ = fluid mass density, $lb\text{-}sec^2/in^4$
Q = flow rate, gpm

The test data were utilized as follows:

d I.D. of 1/2 x .046 tubing adjusted for expansion due to pressure and temperature. Adjustments not made for other tubing sizes because of negligible effects.

ΔP ΔP gage readout converted to psi/ft of test section length.

ρ Mass density values taken from Reference 1 (see Appendix A).

Q Flow measured at return pressure. Flow at operating pressure calculated from return flow.

MS-6 viscosity determinations based on tubing pressure drop are shown on Figure 23. Tube diameter had some effect on the apparent viscosity.

2.2.2 Bulk Modulus

When fluid flow is stopped suddenly by a fast acting solenoid valve, a pressure surge is developed. The surge travels as a pressure wave from the valve back to the flow source at the speed of sound through the fluid. Wave velocity is related to system compliance and fluid density as follows:

$$\beta_e = \rho \, c^2 \qquad \text{Eq. 1} \qquad \text{(Ref. 7)}$$

Where, β_e = effective bulk modulus
c = velocity of pressure wave
ρ = mass density of fluid

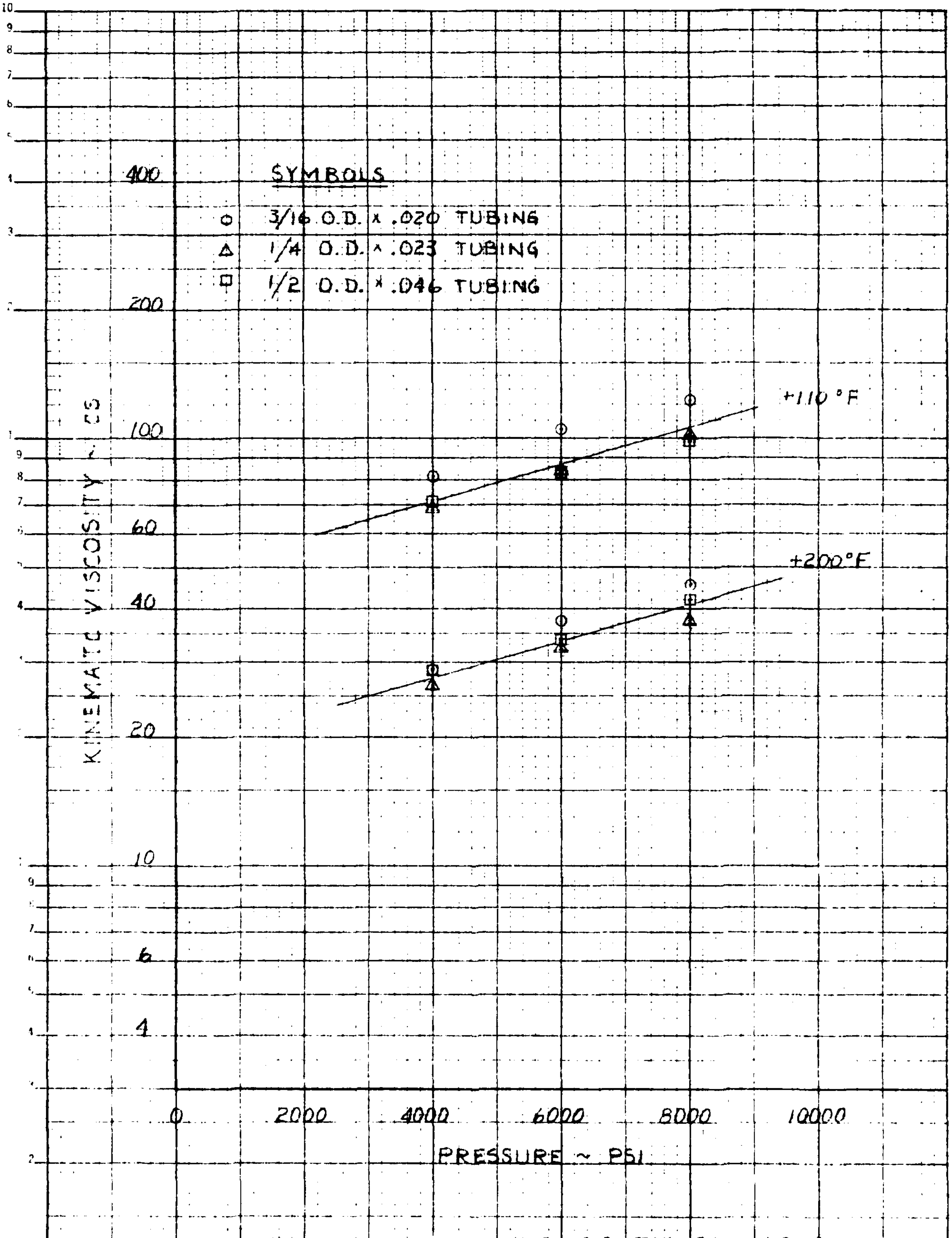


FIGURE 23. VISCOSITY OF MS-6

now, $$\frac{1}{\beta_e} = \frac{1}{\beta_f} + \frac{1}{\beta_t} \qquad \text{Eq. 2}$$

Where, β_f = adiabatic tangent bulk modulus of fluid
β_t = mechanical compliance of tubing

and, $$\frac{1}{\beta_t} = \frac{2}{E}\left[\frac{r_o^2 + r_i^2}{r_o^2 - r_i^2} + \mu\right] \qquad \text{Eq. 3}$$

Where, E = modulus of elasticity of tubing material
r_o = outside radius of tubing
r_i = inside radius of tubing
μ = poissons ratio of tubing material

Equations 1, 2, and 3 were used as the basis for determining adiabatic tangent bulk modulus of MS-6. The pressure surge test, section 2.1.3.2.2, provided pressure wave velocity data. MS-6 mass density information was taken from Reference 1 (see Appendix A). Tubing mechanical compliance was obtained by two methods: actual measurement and calculation. Procedure details are given in the following sections.

Mechanical Compliance of Tubing - The test tubing was 1/2 in. O.D. x 0.046 in. wall 21-6-9 CRES. Actual dimensions were 0.5002 in. O.D. and 0.4068 in. I.D. The modulus of elasticity of the tube material was 28,500,000 psi and poissons ratio was 0.29.

The change in tube outside diameter with applied hydraulic pressure was measured using an electronic micrometer, Sheffield M/N 51 EHB. With 8000 psi pressure applied, stretch in the tube O.D. was 0.00057 in.

The change in tube O.D. was calculated using,

$$\Delta D = \frac{2\, r_i\, P}{E}\left[\frac{r_o^2 + r_i^2}{r_o^2 - r_i^2} + \mu\right] \qquad \text{(Ref. 7)}$$

Where, ΔD = change in outside diameter
P = applied pressure

The calculated change in O.D. with 8000 psi applied was 0.00059 in. The close agreement between the measured and calculated ΔD dimensions provided confidence in the parametric values used to letermine tube compliance. Tubing compliance was calculated, using Equation 3, to be 2,740,000 psi.

Effective Bulk Modulus - Pressure wave velocity in MS-6 was determined at +110 and 200°F at 4000 and 8000 psi. Velocity was based on the time for the wave to travel from pressure transducer P_2 located immediately upstream of a solenoid valve to transducer P_1 located near the discharge port of the pump. The transducers were 25 ft. 9 in. apart. Time was determined from an oscilloscope photograph of the two pressure traces. Time resolution was better than 0.001 sec. The pressure wave velocity (c) was:

MS-6 Temp., °F	Pressure Wave Velocity, in/sec at 4000 psi	at 8000 psi
+110	41,950	43,840
+200	36,950	39,430

The mass density (ρ) of MS-6 is: (see Appendix A)

MS-6 Temp., °F	Mass Density, lb-sec^2/in^4 at 4000 psi	at 8000 psi
+110	9.83×10^{-5}	10.04×10^{-5}
+200	9.45×10^{-5}	9.71×10^{-5}

Using Equation 1, the effective bulk modulus (β_e) was calculated to be:

MS-6 Temp., °F	Effective Bulk Modulus, psi at 4000 psi	at 8000 psi
+110	173,000	193,000
+200	129,000	151,000

Adiabatic Tangent Bulk Modulus of MS-6 - Using Equation 2, the adiabatic tangent bulk modulus of MS-6 was calculated. The results are shown on Figure 24.

It should be noted that fluid in the test system reservoir was in direct contact with argon at 60 psig. Since MS-6 has a strong affinity for gases, system fluid probably contained a significant quantity of dissolved argon, (see Section 2.1.4). This obviously resulted in lower bulk modulus values than if the fluid contained no gas.

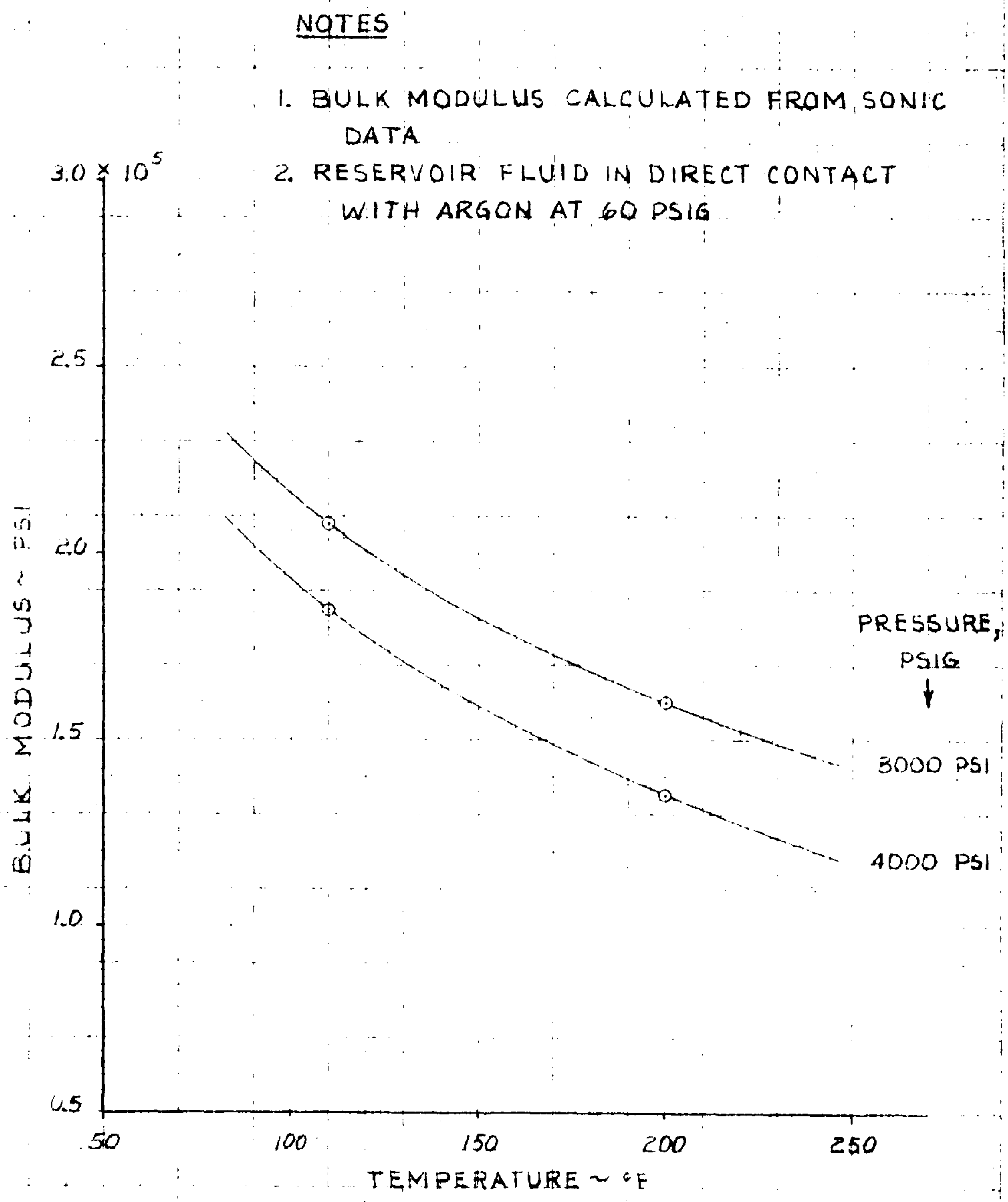


FIGURE 24. ADIABATIC TANGENT BULK MODULUS OF MS-6

2.3 PERFORMANCE COMPARISONS

2.3.1 Pump

Ideally, the pump performance comparison should have been made between two identical models; one designed for use with MS-6 fluid, the other designed for MIL-H-83282 fluid. This would reveal true differences in overall efficiency, heat rejection, and transient response. This type of evaluation was not feasible, however, for economic reasons. Comparisons presented in this section show, instead, the performance degradation that occurs when a pump operates with a fluid it was not designed to use. Pump performance with MS-6 is compared to performance using MIL-H-83282 fluid. Baseline data were generated in the LHS Exploratory Development Program reported in Reference 4. Operating conditions were:

Pump compensator setting:	8000 psi
Pump speed:	4000 rpm
Inlet fluid temperature:	+200°F

Discharge flow, heat rejection, and overall efficiency are compared on Figure 25. Effects of hanger unbalance is evident on the MS-6 curves; discharge flow drops off too rapidly, the heat rejection curve is mis-shaped, and overall efficiency is low.

Operating temperatures were higher using MS-6 as follows:

Fluid Temperature Rise	Using MIL-H-83282	Using MS-6
Inlet-to-discharge	18°F	21°F
Inlet-to-case drain	64°F	83°F

Principal factors causing the higher temperature rise were lower pump efficiency and lower fluid specific heat.

Transient response data are compiled below. Pump transient response characteristics were satisfactory using MS-6.

Condition	Using MIL-H-83282	Using MS-6	*Requirement (Max.)
90% to 5% flow	0.013 sec	0.041 sec	0.050 sec
5% to 90% flow	0.078 sec	0.056 sec	0.050 sec
Stability	0.070 sec	0.805 sec	1.0 sec
Ripple	±150 psi	±100 psi	±240 psi
Pressure overshoot	1600 psi	1800 psi	1600 psi

*See Reference 10

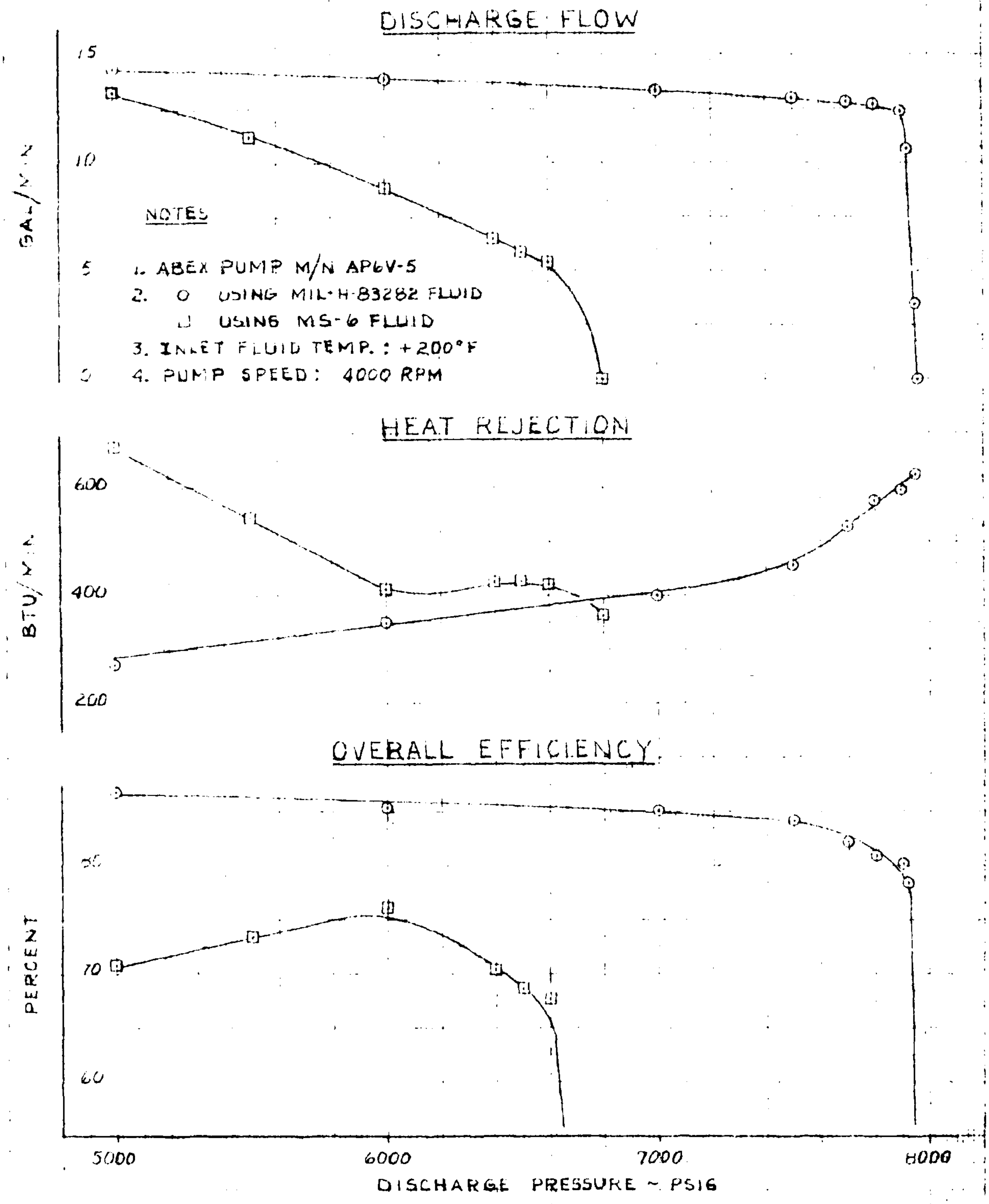


FIGURE 25. PUMP PERFORMANCE COMPARISONS

Although transient response was satisfactory, overall pump performance using MS-6 was poor: hanger unbalance caused incorrect pressure compensation; instability occurred with fluid temperatures lower than +160°F; pump operation was rough, based on audible observations. Performance degradation resulted from the detrimental effects of several MS-6 properties:

MS-6 Property	Principal Effect on Pump Performance
Lower bulk modulus	- Incorrect precompression - Incorrect hanger moment
Higher viscosity	- Lower internal leakage flow (causes higher fluid temperatures)
Higher density	- Lower flow gain in compensator valve (Causes slower response)
Lower specific heat	- Higher fluid temperatures
Tendency to foam	- Additional reduction in bulk modulus - Increased possibility for cavitation - Lowers lubricity (Causes localized high temperatures)

The lower bulk modulus, higher viscosity, and higher density of MS-6 can be accommodated during pump design. The tendency to foam appears to be a natural characteristic of MS-6 and could prove difficult to change. Two corrective measures which should be considered are 1) use of a bootstrap reservoir and 2) use of an anti-foam agent.

2.3.2 Line Pressure Drop

Six plots comparing straight tubing pressure drop characteristics of MS-6 with MIL-H-83282 are presented in Appendix B. A more fundamental comparison based on pressure drop for the same flow and tube diameter is given on Figures 26 and 27. This approach shows the least difference in pressure drop occurs when a comparison is made with turbulent flow present in both fluids; the K factor averages 1.7 for this situation. The largest difference occurs when both fluids have laminar flow; in this case, K is more than 6.0 for temperatures over +200°F (Figure 26). The K factor lies between the two curves if the given conditions produce turbulent flow in MIL-H-83282 and laminar flow in MS-6.

MS-6 pressure losses range from 1.5 to 6 times greater than MIL-H-83282 losses under normal operating temperatures. In order to maintain pressure drops within acceptable limits, line diameters must be increased to accommodate MS-6. The larger tube sizes plus the higher fluid density will produce a weight penalty compared to use of hydrocarbon base fluids. (see Section 3.0.)

2.3.3 Pressure Surge

Pressure transient peaks recorded using MS-6 were compared to peaks observed using MIL-H-83282 (documented in Reference 4). MS-6 surges averaged 29% higher than MIL-H-83282 surges, Figure 28.

NOTES

1. K FACTOR BASED ON SAME FLOW AND TUBE DIAMETER
2. ΔP_{83282} = PRESSURE DROP USING MIL-H-83282
3. $\Delta P_{MS\text{-}6}$ = PRESSURE DROP USING MS-6
4. IF COMPARISON IS BETWEEN TURBULENT AND LAMINAR FLOW, K FACTOR LIES BETWEEN THE TWO CURVES.

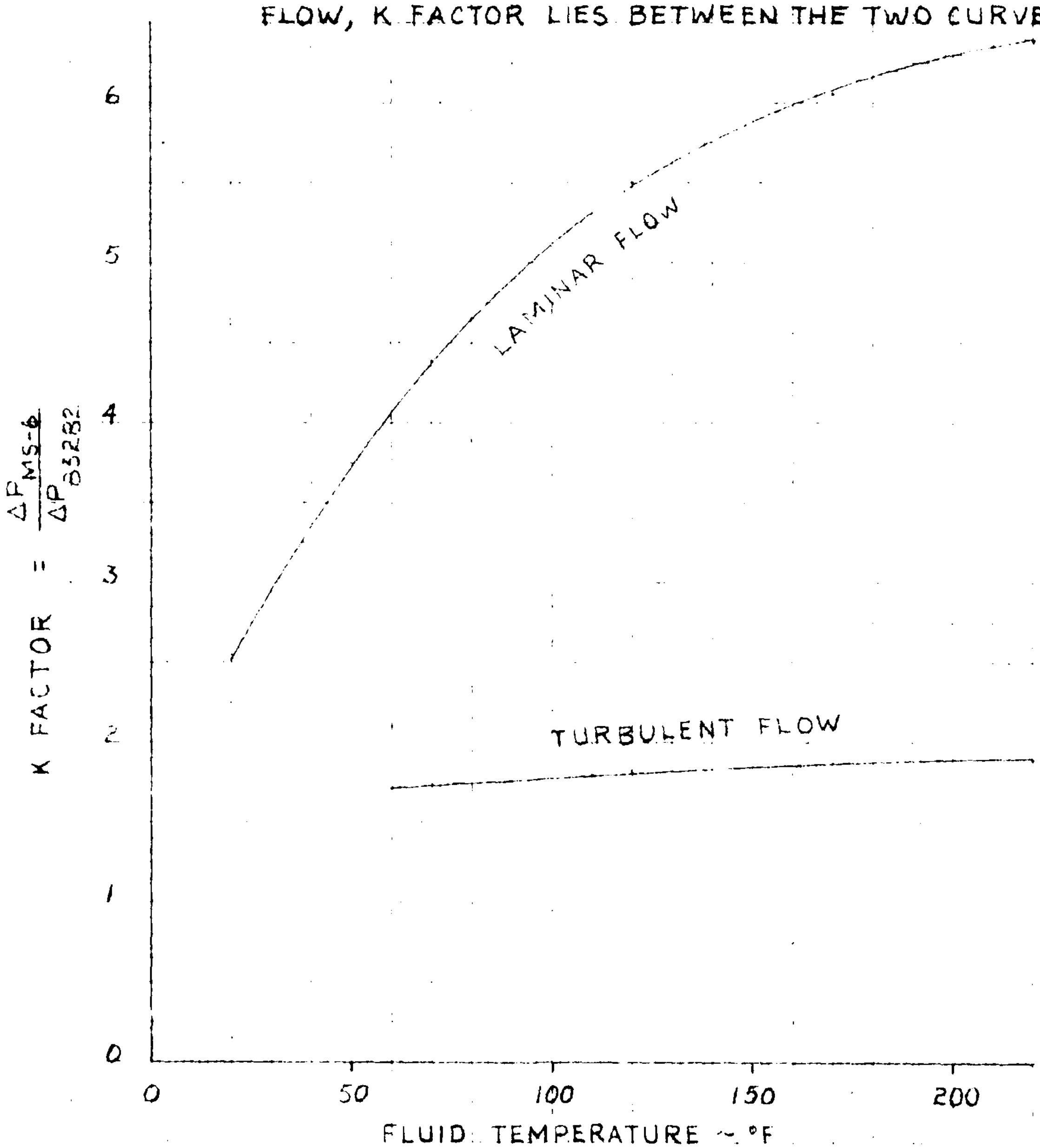


FIGURE 26. INCREASE IN PRESSURE DROP USING MS-6 AT 8000 PSI

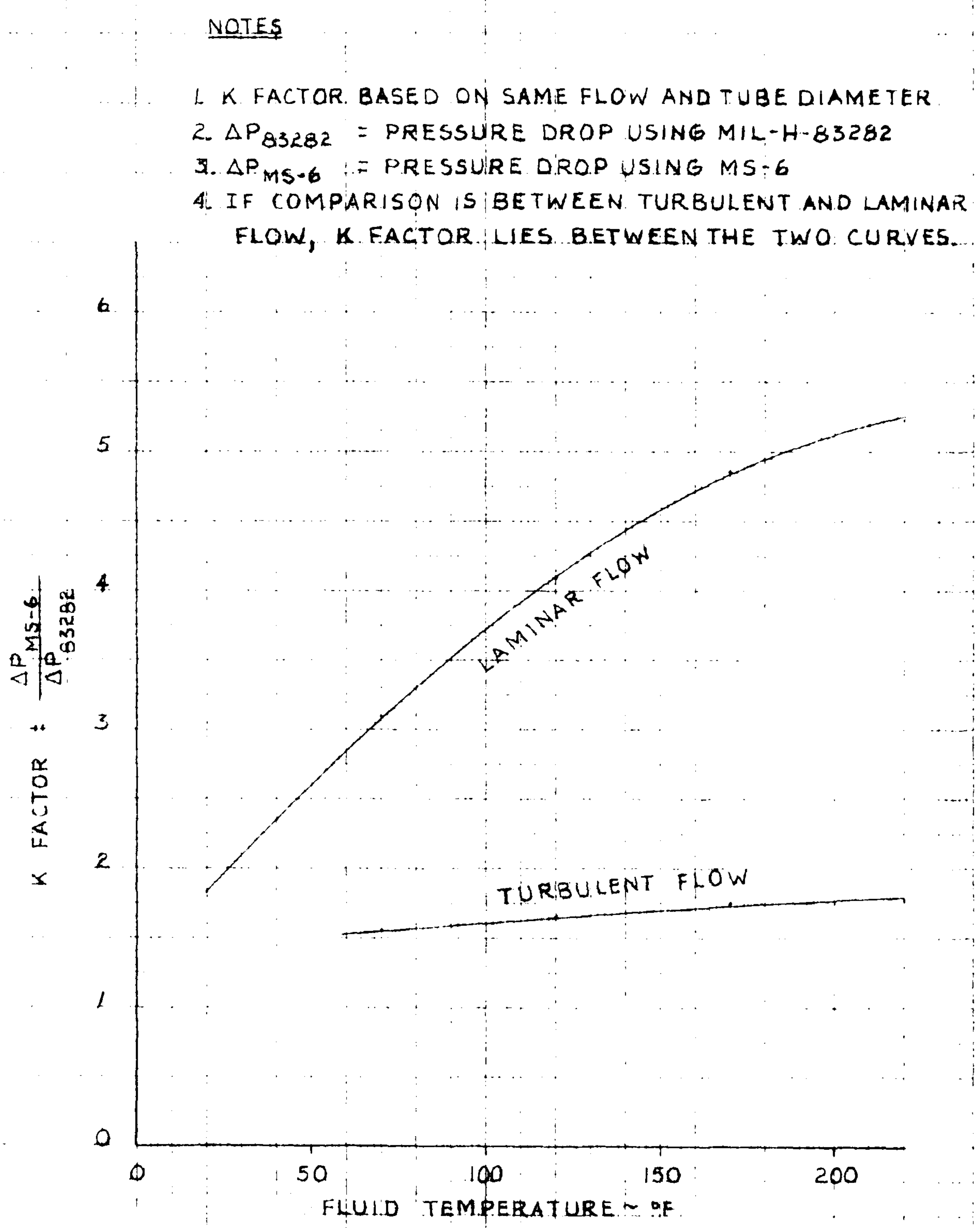


FIGURE 27. INCREASE IN PRESSURE DROP USING MS-6 AT 100 PSI

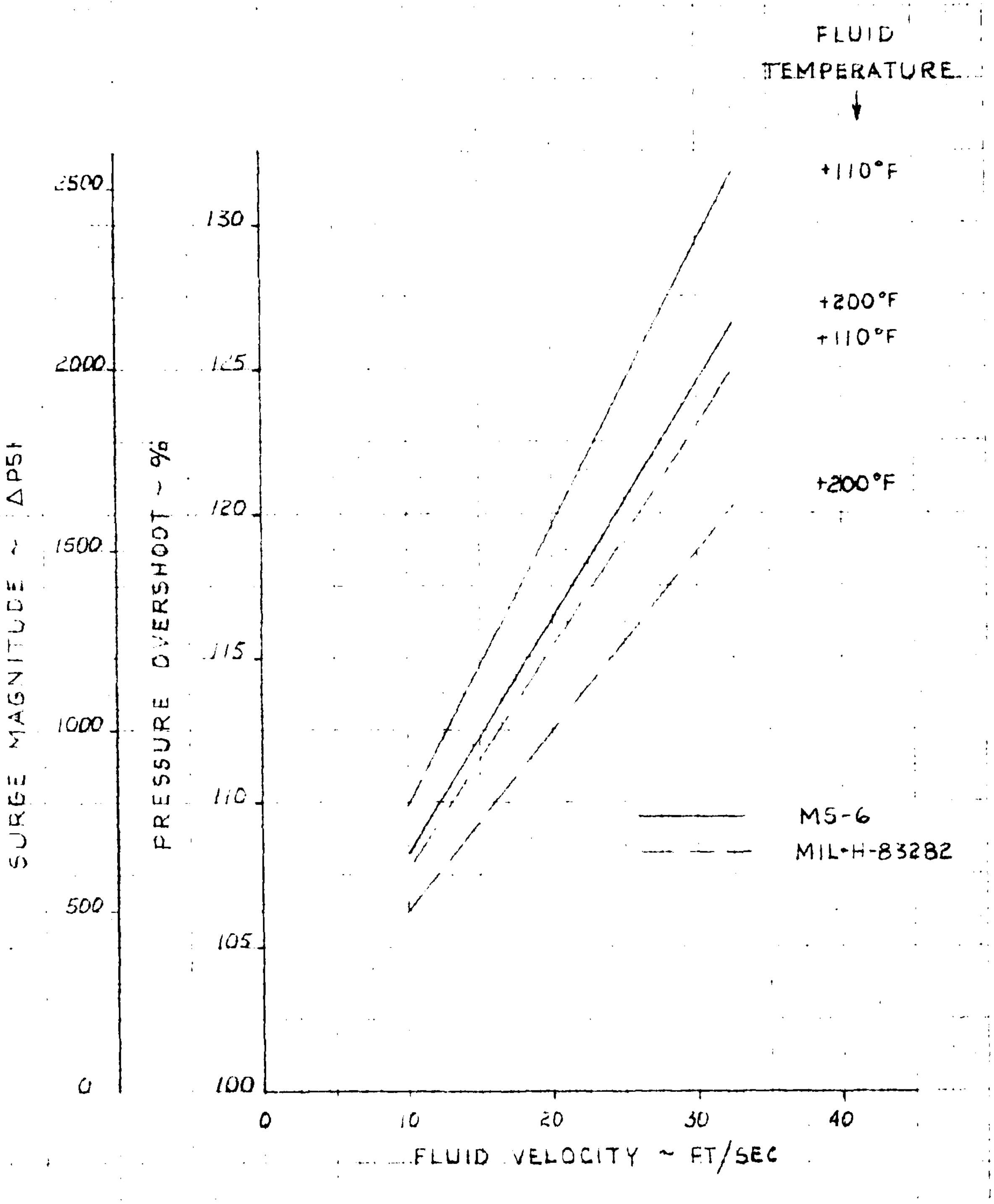


FIGURE 28. PRESSURE SURGE COMPARISONS

Data collected during the MS-6 surge test did not provide a good basis for evaluating surge magnitude because of poor pump performance. Since a principal factor determining transient pressure peaks is pump dynamics, generation of valid surge data will require use of a pump designed for MS-6.

2.3.4 Flow Restrictor

Flow through a sharp edged orifice is found by:

$$Q = C_d A \sqrt{2\left(\frac{P_1}{\rho_1} - \frac{P_2}{\rho_2}\right)}$$

Where,
- Q = flow rate through orifice
- C_d = orifice coefficient of discharge
- A = orifice area
- P_1 = orifice inlet pressure
- P_2 = orifice exit pressure
- ρ_1 = fluid mass density at orifice inlet
- ρ_2 = fluid mass density at orifice exit

Since the density of MS-6 is approximately 25% higher than the density of MIL-H-83282, lower orifice flows will result. Under the test conditions -- 8000 psi operating pressure and +110°F inlet fluid temperature -- orifice flow using MS-6 should be approximately 12% less, theoretically, than with MIL-H-83282 fluid. The test results indicated a larger difference, however.

Restrictor	Flow Rate Using MIL-H-83282	Flow Rate Using MS-6	Percent Difference
2 gpm	1.86 gpm	1.54 gpm	-17.2
4 gpm	3.88 gpm	3.28 gpm	-15.5

The coefficient of discharge for a sharp edged orifice is 0.61, regardless of geometry and fluid viscosity, provided flow is turbulent. If the flow is not fully turbulent, the coefficient can be somewhat higher than 0.61. If flow is laminar, the coefficient will be lower than 0.61. The difference between the calculated reduction in orifice flow (12%) and the actual reduction (≈16%) is probably related to changes in coefficient of discharge resulting from dissimilar Reynold's numbers.

Orifices can easily be re-sized to accommodate MS-6. The test data indicate that use of MS-6 density information and a coefficient of discharge approximately 5% less than that used for hydrocarbon base fluids should provide satisfactory re-sizing.

2.3.5 Servo Actuator

Actuators operating at 8000 psi have a lower stiffness than comparable actuators operating at 3000 psi. The weakest element in the actuator spring system is the fluid column.

$$K_f = \frac{4\beta_f A \eta}{S}$$

Where,
K_f = stiffness of fluid column (actuator at mid-stroke)
β_f = adiabatic tangent bulk modulus of fluid (taken at one-half of system pressure)
A = piston net area
S = piston stroke (total)
η = ratio of fluid volume swept by piston to the total fluid volume contained between the piston and control valve (typically between 0.85 and 0.95 for an integrally mounted valve)

Reduction in stiffness due to a smaller piston diameter is partially offset by an increase in fluid bulk modulus at 8000 psi. The change in actuator/system natural frequency (ω_n) resulting from use of MS-6 instead of MIL-H-83282 is not large, however, because 1) the actuator fluid column is but one factor affecting system spring rate, and 2) ω_n is not related directly to actuator stiffness, but to a square root function.

$$\frac{1}{K_T} = \frac{1}{K_S} + \frac{1}{K_A} + \frac{1}{K_f}$$

Where,
K_T = total spring rate of system
K_S = spring rate of support structure
K_A = spring rate of actuator structure
K_f = spring rate of actuator fluid column

and,

$$\omega_n = \sqrt{\frac{K_T}{M_e}}$$

Where,
ω_n = undamped natural frequency of system
K_T = total spring rate of system
M_e = effective mass of moving components

Comparisons of actuator stiffness and natural frequency for the system tested are as follows:

Fluid	Fluid Bulk Modulus, psi	Actuator Stiffness, lb/in	Actuator/System Natural Frequency, Hz Undamped (Calculated)	Damped (Observed)
MIL-H-83282	[1]246,000	195,000	25.2	22.5
MS-6	[2]171,000	153,000	22.4	19.5

[1]See Figure A-4
[2]See Figure 24

A 30% decrease in fluid bulk modulus produced a 13% reduction in actuator/system damped natural frequency. This magnitude reduction in ω_n is considered acceptable, and should not seriously degrade performance.

2.3.6 Flow Control Valve

Changes in valve operating characteristics resulting from the use of MS-6 are predictable, and can be easily accommodated by design modifications.

Flow Gain - Valve flow gain decreased from 182 in^3/sec/in using MIL-H-83282 to 154 in^3/sec/in using MS-6; a 15.4% reduction. Flow gain is based on the orifice flow equation, and the percentage decrease agrees well with the reduction observed in the orifice flow tests discussed in section 2.3.4. Use of MS-6 density information and a coefficient of discharge approximately 5% less than that used for hydrocarbon base fluids should provide satisfactory orifice re-sizing.

Pressure Gain - Pressure gain increased from 2.4 x 10^6 psi/in using MIL-H-83282 to 2.67 x 10^6 psi/in using MS-6; an 11% improvement. An approximate expression for pressure gain is,

$$K_p \approx \frac{32\,\mu\,C_d}{\pi\, r_c^2}\sqrt{\frac{P_s}{\rho}} \qquad \text{(Ref. 7)}$$

Where,
K_p = pressure gain
μ = absolute viscosity
C_d = coefficient of discharge
P_s = supply pressure
r_c = radial clearance between spool and sleeve
ρ = fluid mass density

Fluid viscosity appears to be the dominant factor tending to increase pressure gain. The improved pressure gain is useful but not a significant advantage.

Null Leakage - Null leakage was 94 cc/min with MIL-H-83282 and 30 cc/min with MS-6; a 68% reduction. The decrease was due to differences in fluid viscosity as shown by the following equation.

$$Q_c = \frac{\pi W r_c^2}{32 \mu} \quad \text{(Ref. 7)}$$

Where, Q_c = null flow (centered spool, blocked cylinder ports)
W = area gradient of valve
r_c = radial clearance between spool and sleeve
μ = absolute viscosity of fluid

The significantly lower leakage rate results in smaller power losses - an obvious advantage for MS-6 over MIL-H-83282.

2.3.7 Solenoid Valve

Internal leakage through components such as solenoid valves is a power loss which should be minimized. A comparison of leakage rates through the test valve using MIL-H-83282 and MS-6 fluids is given below.

Valve Mode, Off/On	Internal Leakage, Drops/Min MIL-H-83282	MS-6	Percent Reduction
Off	18	16	11%
On	20	10	50%

In general, reduction in leakage can be expected to be proportional to the difference in absolute viscosity between the two fluids for given operating conditions.

2.3.8 Fluid Properties

Viscosity - Two types of viscosity are used in flow analyses: absolute and kinematic. They are related as follows:

$$\mu = \rho \nu$$

Where, μ = absolute viscosity
ρ = mass density
ν = kinematic viscosity

Kinematic viscosity is easily determined at atmospheric pressure and is the measure commonly used to describe the consistency of fluids. Absolute viscosity is more suitable, however, for comparing viscosity differences since it is directly related to line pressure drop and to component internal leakage.

The increase in absolute viscosity of MS-6 over MIL-H-83282 is illustrated on Figure 29. At temperatures near -20°F, there is no appreciable difference in viscosity between MS-6 and MIL-H-83282. At +200°F and 8000 psi, MS-6 pressure drops will be approximately 6 times more and internal leakage rates 6 times less than with MIL-H-83282.

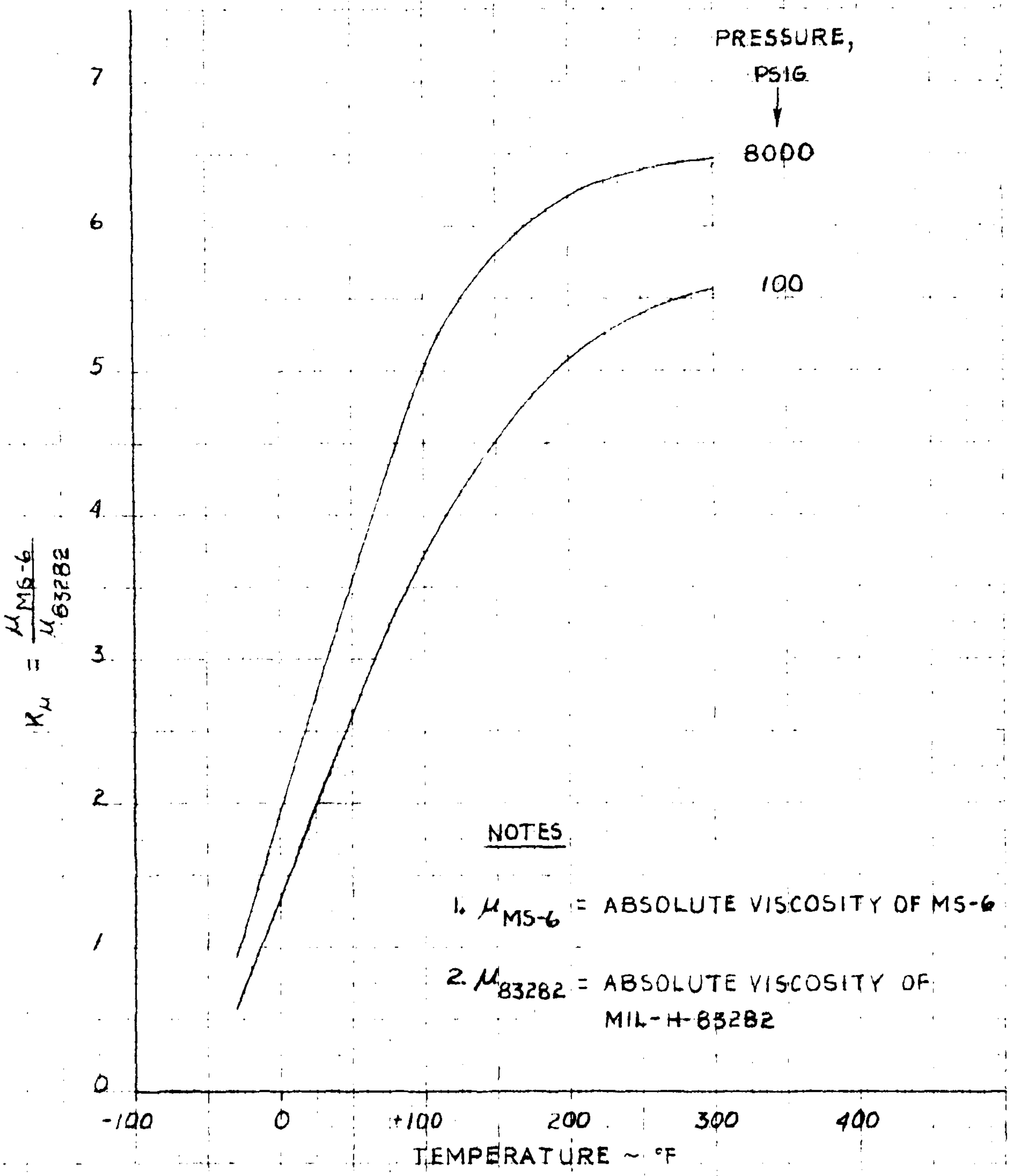


FIGURE 29. INCREASE IN VISCOSITY OF MS-6 OVER MIL-H-83282

Fluid viscosity is a major factor in hydraulic system design and performance. Thorough consideration must be given to the effects of operating conditions on viscosity and effects of viscosity changes on system performance. Viscosity requirements are often conflicting; good lubrication and low internal leakage necessitate moderately high viscosity, while low line losses and fast control response dictate low viscosity. Viscosity-related design factors for MS-6 and MIL-H-83282 are listed on Table 2.

Table 2. Viscosity-Related Design Factors

	MIL-H-83282	MS-6
Max. Operating Temp		
Rated	+400°F	+400°F
2 cs temp	+290°F	over +400°F
Min. Operating Temp		
Rated	-40°F	-65°F
Cold start (2500 cs)	-37°F	-63°F
500 cs temp	- 8°F	-10°F
Line Pressure Drop		
Laminar flow	-	up to 6 x higher
Turbulent flow	-	approx. 1.7 x higher
Power Losses (Internal leakage)	Moderate	Very low
Component Wear (Lubricity)	Moderate	Questionable (due to foaming)
Component Response	Moderate	Slower
Damping	Moderate	Increased

Bulk Modulus - Fluid compressibility is an important consideration in the design of servo systems. Interaction of the fluid spring with the mass of mechanical parts and load produces a natural frequency which often limits dynamic performance. Bulk modulus is a fundamental factor in pump design, affecting precompression volume, hanger balance, and timing.

Two types of fluid bulk moduli are used in design: adiabatic tangent and adiabatic secant. The adiabatic tangent is used for functions that occur rapidly with small pressure excursions, such as oscillating servo actuators. Adiabatic secant is for applications with large pressure changes in small time intervals such as pumps. The adiabatic tangent modulus is usually determined from tests in which the velocity of sound in the fluid is measured.

Bulk modulus values published in the literature are normally for fluids with minimal amounts of dissolved gas. Actual values encountered in working systems will be less because of entrained and dissolved gases. Dissolved gas reduces bulk modulus only slightly; entrained air (free bubbles) can reduce bulk modulus substantially. The amount of gas hydraulic fluid can absorb is a function of pressure, temperature, and time. The rate of absorption decreases as bubble size increases. Under atmospheric conditions, air solubility of petroleum oils is approximately 10% by volume while silicone fluids is approximately 24%, Reference 1.

Saturation pressure is that pressure below which entrained gas is present and above which all gas is forced into solution. The amount of gas entrained has a significant effect on the magnitude of saturation pressure. Ratios of gas-to-oil volumes (at atmospheric pressure) on the order of 15:1 can result in saturation pressures near 4000 psi, Reference 12. A large reduction in bulk modulus occurs if the operating pressure is below the saturation pressure.

Reservoir fluid was in direct contact with argon at 60 psig in the MS-6 tests. At high flow rates, fluid residency time in the reservoir was insufficient for all gas to come out of solution and additional argon was introduced before the fluid re-circulated through the pump. This process continued until the MS-6 could absorb no more gas and equilibrium was reached for the given operating conditions. The volume of entrained and dissolved argon in the 7.5 gallons of system fluid was not known, but the amount present was not believed to be sufficient to raise saturation pressures above operating pressures.

A comparison of the adiabatic tangent bulk modulus of MS-6 with MIL-H-83282 is given in Figure 30. The MS-6 modulus averaged 33% lower than the MIL-H-83282 modulus. The principal effect of this reduction was on pump performance, see section 2.3.1.

The relatively low bulk modulus of MS-6 is considered a disadvantage, but not a serious obstacle. Pumps can be designed to accommodate the lower modulus; actuator servo systems can use MS-6 with a minor degradation in performance. Selection of MS-6 for a specific application will probably be based on trade-offs involving factors other than bulk modulus.

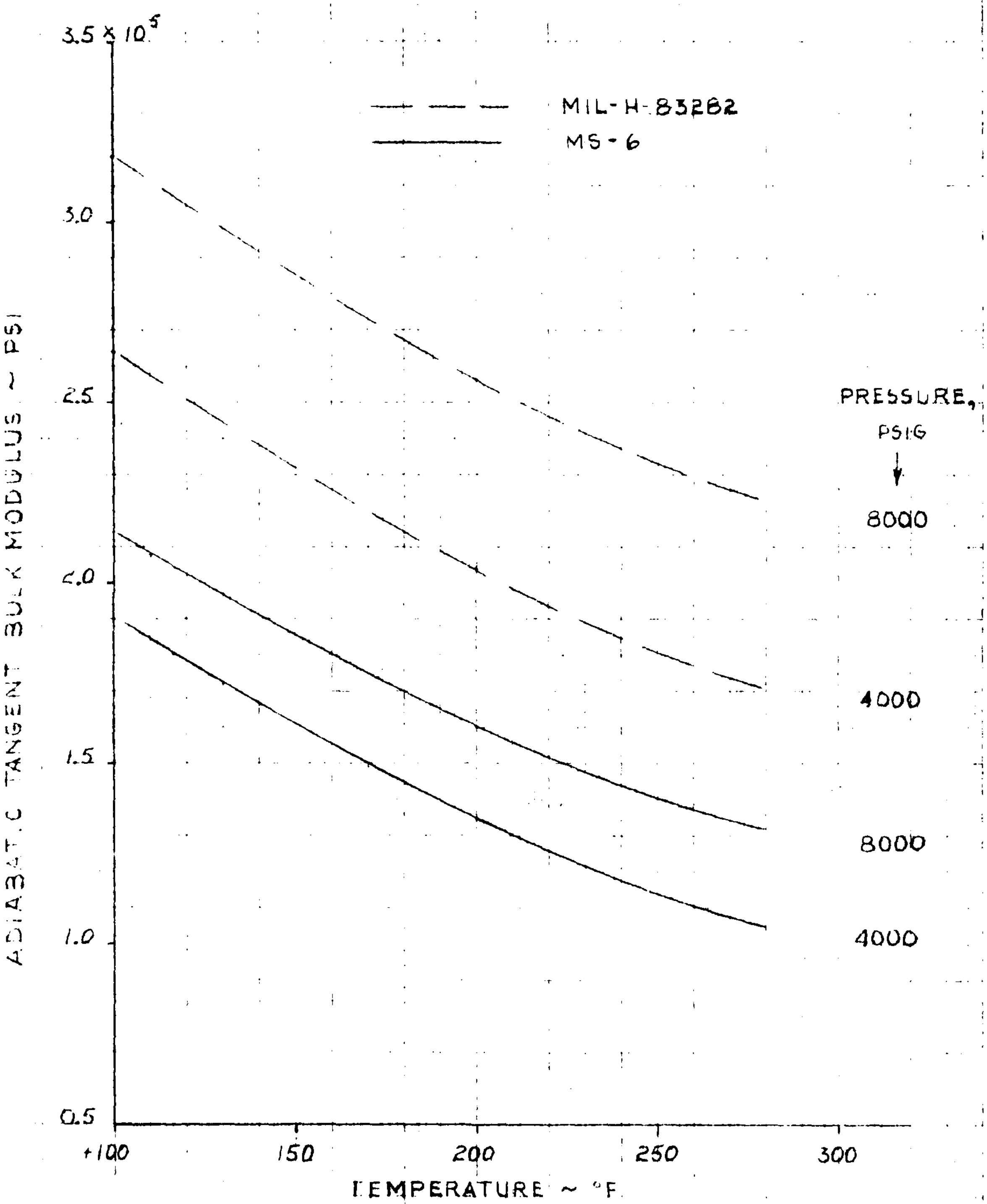


FIGURE 30. BULK MODULUS COMPARISONS

3.0 APPLICATION STUDY

3.1 A-7E LIGHTWEIGHT HYDRAULIC SYSTEM

3.1.1 System Description

The A-7E lightweight hydraulic system is comprised of three independent systems--FC-1, FC-2, and utility--as shown on Figure 31. FC-1 and FC-2 operate at 8000 psi and contain all primary and secondary flight control actuators. The utility system operates at 3000 psi and powers all remaining hydraulic functions. Configuration details are given in Reference 5.

3.1.2 Study Areas

The impact of higher viscosity, lower bulk modulus, and higher density of MS-6 on the design of the 8000 psi flight control system was examined. Areas covered were:

Pump	- Design changes required to accommodate MS-6
Distribution Lines	- Lines re-sized, where necessary, to maintain actuator working pressure within $\pm 2.5\%$ of existing values. Working pressure = 8000 psi minus pressure and return line ΔP's at rated flow and operating temperature.
	- Branch lines, likely to cold soak in flight, sized to provide a pressure drop no greater than the existing line at the cold soak temperature.
	- Pump suction line re-sized for the minimum temperature at which full flow could be delivered.
Reservoir	- Volume increased as required by changes in system volume. Reservoir pressure to be increased if necessary to prevent pump cavitation.
Actuators	- Flight control actuators reviewed to determine if reduced stiffness was acceptable and, if not, actuator re-sized.
Components	- Components and orifices re-sized to account for flow rate changes.
Power Requirements	- Factors affecting system power examined.
System Cooling	- Factors affecting heat generation and temperature buildup studied.

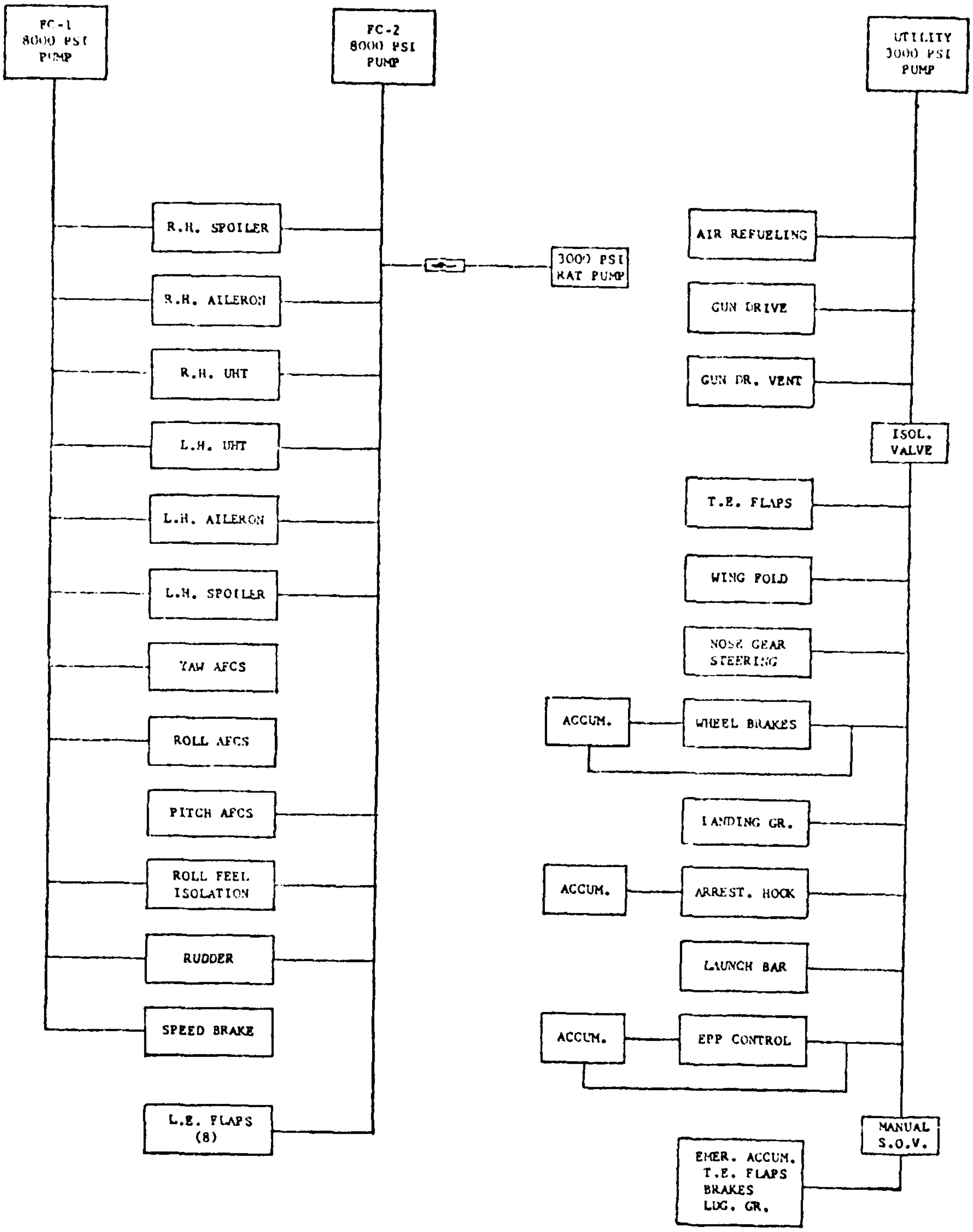


FIGURE 31. A-7E lightweight hydraulic system

3.1.3 Design Modifications

3.1.3.1 Pump - Fluid bulk modulus, viscosity, and density play major roles in pump design. In order to achieve maximum overall efficiency and long life, aircraft pumps must be designed specifically for a given type of hydraulic fluid. Operation with other types could result in degraded performance.

Hanger balance is affected by fluid bulk modulus. A typical hanger configuration is shown on Figure 32. Hanger position is controlled by the compensator actuator and a return spring. Hanger loading results from inertial forces and cylinder pressures--both transmitted to the hanger through the piston shoes. Pump timing, i.e., the point at which high pressure fluid is delivered to the outlet port and the point at which the inlet port is uncovered, affects shoe forces. Precompression--pressure that is built up in the cylinder before fluid is released to the outlet port--also affects shoe forces. Precompression is based on cylinder volume and fluid bulk modulus. Using a fluid with a low bulk modulus reduces precompression and therefore shoe forces. A sufficiently low bulk modulus can unbalance the hanger and de-stroke the pump. Hanger unbalance occurred during the pump evaluation tests (see section 2.1.3.1). Proper pump design would correct this condition.

Internal leakage rates are affected by fluid viscosity. Built-in leakage is necessary to provide lubrication and cooling flow. Principal controlled paths include cross-port leakage, piston/cylinder leakage, and piston shoe/hanger leakage. Reduced leakage, caused by a high viscosity fluid, could cause local temperatures to increase due to inadequate cooling flow. Higher viscosity would, however, provide a thicker oil film and decrease wear if sufficient clearances and lubricity were present. As reported in section 2.1.3.1, the test pump had indications of high local temperatures, but little evidence of wear. Pump design could easily be modified for higher viscosity fluids.

Orifice flow is affected by fluid density. A higher density fluid would decrease flow gain in the compensator valve and slow response. An increase of 12% in the valve area gradient should provide the correct flow for MS-6 fluid.

High gas solubility and a tendency to foam are characteristics of MS-6 which could cause cavitation in the suction porting. The higher density of MS-6 adds to the possibility for cavitation during rapid acceleration of the suction line fluid column. Higher minimum inlet pressure, increased reservoir pressurization, anti-foam agents, or gas removal equipment may be required to resolve this potential problem area.

3.1.3.2 Distribution Lines - Pressure and return line losses were determined in distribution circuits for the primary flight control actuators and leading edge flap subsystem. The analysis compares LHS losses using MIL-H-83282, losses using MS-6, and losses using MS-6 after line re-sizing. MS-6 losses before re-sizing are from 1.7 to 10 times

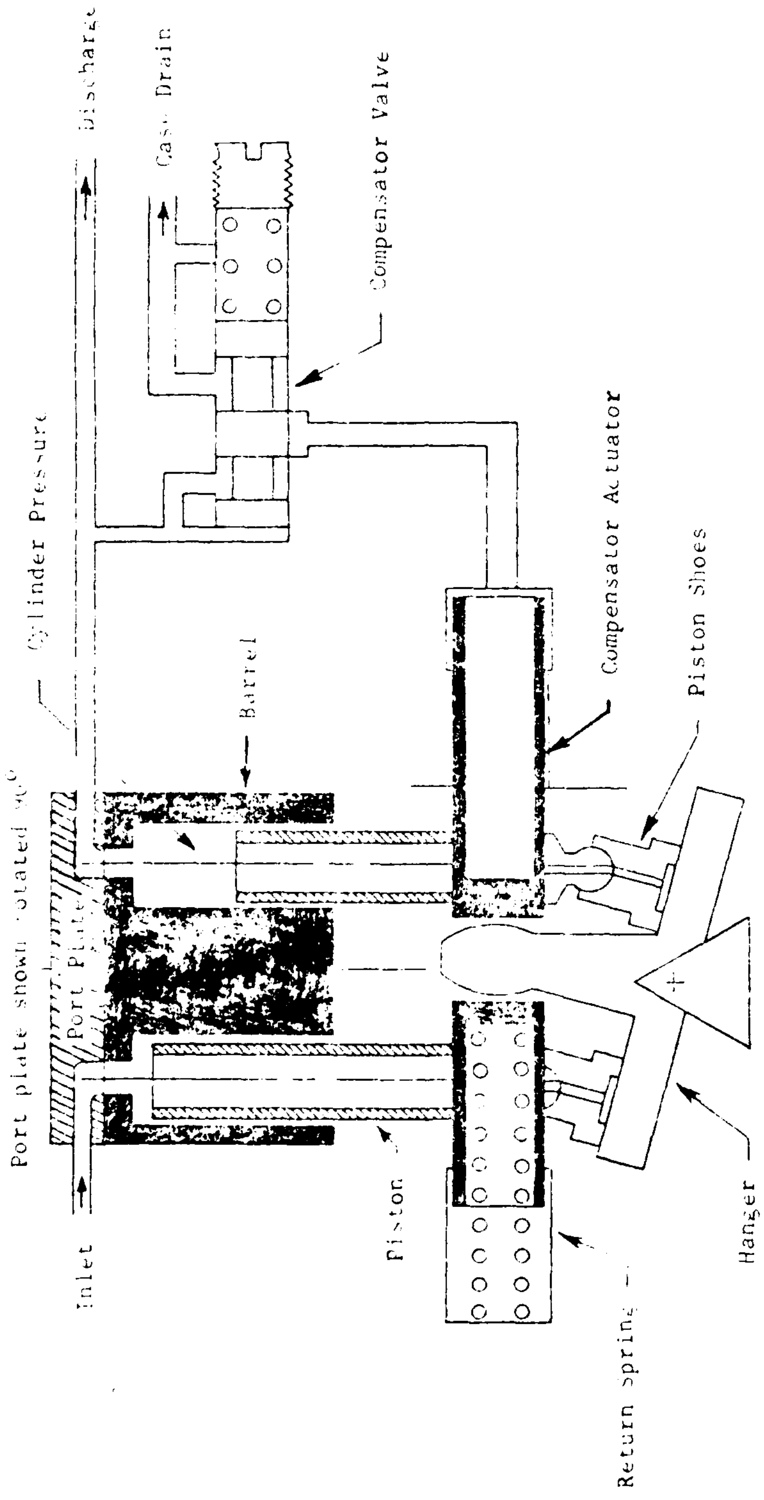
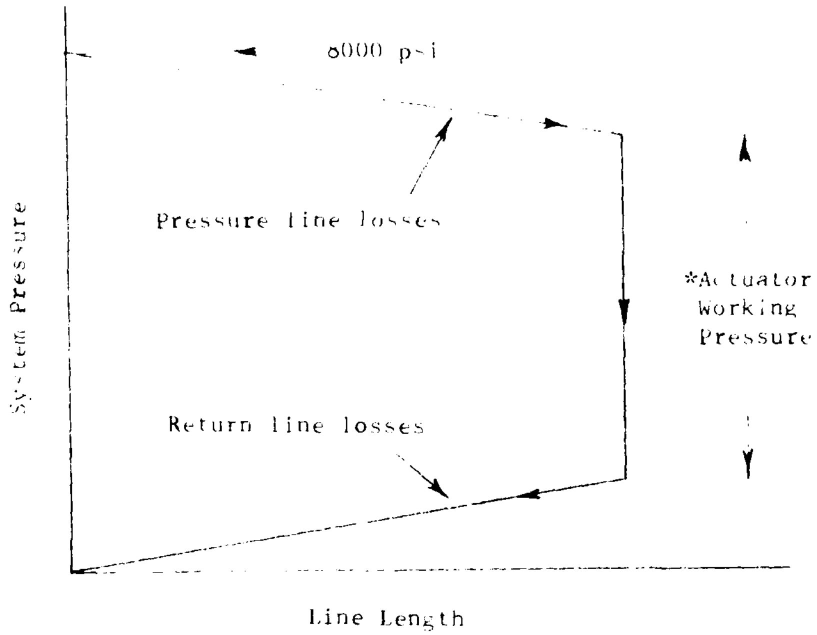


FIGURE 32. Aircraft pump hanger configuration

higher than LHS line losses, Table 3. Lines were re-sized when actuator working pressure was reduced more than 2.5% by line pressure losses, Figure 33.



*Maintain within ±2.5% for line re-sizing

FIGURE 33. Actuator working pressure

MS-6 losses in the LHS aileron, spoiler, and L.E. flap circuits were very high due to the use of -3 size tubing. MS-6 losses in the LHS rudder and UHT circuits reduced working pressures by 4.5 to 6%. A summary of lines re-sized is given in Table 4.

3.1.3.3 Pump Suction Line - Suction line diameter and reservoir pressurization must be designed to provide sufficient pressure at the pump inlet to prevent cavitation. Minimum pressure is a function of pump configuration, size, and speed. The presence of entrained or dissolved gases in the fluid increases the possibility for cavitation. Since the solubility of gas is greater in MS-6 than in MIL-H-83282, the required minimum inlet pressure could be affected. For this analysis, however, the minimum inlet pressure for the MS-6 pump was assumed to be the same as for the LHS A-7 pump (35 psig). Reservoir pressurization was 90 psig.

TABLE 3. Distribution Line Losses

Sub-System	*Line Losses, psi		
	MIL-H-83282 (Existing)	MS-6 Before Re-Sizing	MS-6 After Re-Sizing
FC-1			
R.H. UHT Actuator	530	897	480
L.H. UHT Actuator	475	805	444
Rudder Actuator	316	763	409
Aileron Actuator	716	2990	901
Spoiler Actuator	284	2803	323
FC-2			
R.H. UHT Actuator	447	773	259
L.H. UHT Actuator	566	975	382
Rudder Actuator	246	634	244
Aileron Actuator	803	5356	877
Spoiler Actuator	630	2742	767
L.E. Flap System			
O.B. Panel, O.B. Actuator	1034	5174	942
O.B. Panel, I.B. Actuator	976	4900	906
Center Section, O.B. Actr.	674	3372	612
Center Section, I.B. Actr.	674	3368	612

*Line Losses = pressure line losses + return line losses

Flow Condition = Maximum surface rate

Operating Temperatures: Fuselage = +180°F
Wing = +100°F

TABLE 4. Lines Re-Sized Due to Excessive Pressure Losses

Subsystem	Line Size		Length, ft.
	Existing	Re-Sized	
FC-1	-3	-4	52
FC-2	-3	-4	94
L.E. Flap	-3	-4	139
		TOTAL	285
FC-1	-4	-6	60
FC-2	-4	-6	65
		TOTAL	125
FC-1	-6	-8	11
FC-2	-6	-8	38
		TOTAL	49

The two design conditions where pump inlet pressure must be maintained are:

1. Maximum steady state flow. Flow which occurs at the lowest temperature the pump can deliver full flow.

2. Dynamic response flow. Flow which occurs during the interval between zero flow demand and full flow demand. Minimum suction pressure must be maintained during this interval to prevent cavitation.

Maximum Steady State Flow - Maximum flow (10 gpm) can be achieved with temperatures down to approximately +20°F where total system line losses approach 8000 psi. The LHS pump suction line consists of a 5/8 in. O.D. (-10 size) line 128 in. long with three 90° bend fittings and a 3/4 in. (-12 size) quick disconnect. Steady state pressure drop values for the LHS and MS-6 configurations are compared on Table 5. Based on the assumed conditions, an increase in suction line size from a -10 to -16 and an increase in disconnect size from -12 to -16 is required.

Dynamic Response Flow - Pressure required to accelerate fluid in the suction line is determined by,

$$P = \frac{WLV}{144\ gt}$$

where,

P = pressure, psi

W = peak flow rate, lb/hr

L = line length, ft

V = fluid peak velocity, ft/sec

g = acceleration due to gravity, ft/sec^2

t = pump response time, sec (=0.050)

Fluid temperature selected for the analysis was +100°F--well below expected operating temperatures of +180 to +220°F. Pump inlet pressures using MIL-H-83282 and MS-6 are compared in Table 6. Line and disconnect sizes must be increased to 1 inch (-16 size) to maintain adequate pressure at the pump inlet under dynamic conditions.

Discussion - An increase in line and disconnect size to 1 inch (-16) is necessary to meet both steady state and dynamic flow requirements. The pressure margin using MS-6 in -16 line and disconnect sizes is higher than with MIL-H-83282 in the existing system. The larger margin would allow minimum inlet pressure to be increased to 55 psig (existing minimum is 35 psig) and ease the cavitation problem.

TABLE 5. Pump Inlet Pressure with Steady State Flow at +20°F

Configuration	Suction Line ΔP, psi (ΔP_1)	Disconnect ΔP, psi (ΔP_2)	Press Avail.@ Pump Inlet, psi ($90-\Delta P_1-\Delta P_2=\Delta P_3$)	Margin, psi (ΔP_3-35)
A-7 LHS using MIL-H-83282	24.5	27	38.5	3.5
A-7 LHS using MS-6	133.5	72.5	-116	-151*
-12 suction line, -16 disconnect using MS-6	73.2	6.9	9.2	-25.8*
-16 suction line, -16 disconnect using MS-6	25.1	6.9	58	23

TABLE 6. Pump Inlet Pressure under Dynamic Conditions

Configuration	Suction Line & Disc. ΔP, psi (ΔP_1)	Reqd. Accel. Press, psi (ΔP_2)	Press Avail.@ Pump Inlet, psi ($90-\Delta P_1-\Delta P_2=\Delta P_3$)	Margin, psi (ΔP_3-35)
A-7 LHS using MIL-H-83282	12.0	32.7	45.3	10.3
A-7 LHS using MS-6	44.6	40.6	4.8	-30.2*
-12 suction line, -16 disconnect using MS-6	31.5	28.9	29.6	-5.4*
-16 suction line, -16 disconnect using MS-6	6.9	15.6	67.5	32.5

*Inadequate margin

3.1.3.4 Reservoir - Increased line sizes required to accommodate the higher viscosity of MS-6 result in a larger total system fluid volume. Thermal expansion of this larger volume produces excess fluid which must be handled by the reservoir. Conversely, the reservoir must have sufficient reserve to provide make-up fluid as system fluid cools and contracts. Increase in system volume was as follows:

	System Volume Increase, in^3
Distribution Lines	21.3
Suction Line and Hose	53.8
Aileron Hoses	5.8
Spoiler Extension Units	4.3
Miscellaneous Components	4.8
TOTAL	90.0 in^3

Reservoir re-sizing was based on criteria given in military specification MIL-R-8931, Table 7. Use of MS-6 results in a reservoir volume increase of 31 in^3. No change in reservoir pressurization is required (see section 3.1.3.3).

3.1.3.5 Primary Flight Control Actuator Stiffness - None of the original 8000 psi surface actuators were stiffness critical; gain margins of at least 10 db were achieved. Use of MS-6 would lower actuator stiffness about 20%. Since horizontal stabilizer actuators are considered to be the most sensitive with regard to stiffness, the LHS UHT actuator was studied.

An analysis of the UHT actuator/installation stiffness showed that use of MS-6 would reduce the gain margin by 5 db. This reduction was not considered serious. Therefore, the LHS surface control actuators need not be modified (increase piston net area) to regain lost stiffness.

Servo Control Valve - The higher density of MS-6 directly affects orifice flow as follows,

$$Q_{MS-6} = Q_{83282}\sqrt{\frac{\rho_{83282}}{\rho_{MS-6}}}$$

where,

Q_{MS-6}	=	Orifice flow rate using MS-6
Q_{83282}	=	Orifice flow rate using MIL-H-83282
ρ_{MS-6}	=	Mass density of MS-6
ρ_{83282}	=	Mass density of MIL-H-83282

TABLE 7. Volume Change in FC-1 Reservoir*

	A-7 LHS using MIL-H-83282	Re-sized LHS using MS-6
System volume, in^3 (excluding reservoir)	397	487
Reservoir reserve	68.5	68.5
Volume change, actuators	94.0	94.0
Thermal contraction (+70 to -40°F)	36.3	42.4
5% of total system capacity including reservoir (full-refill)	35.8	41.9
Fluid compression & component expansion (2.5% for MIL-H-83282, 3.0% for MS-6)	17.5	25.2
Thermal expansion (+70 to ±275°F)	67.6	79.0
Required reservoir volume, in^3	320.0	351.0

*FC-1 and FC-2 reservoirs were made identical for economic reasons. Sizing was governed by FC-1 volume requirements. FC-2 reservoir therefore had excess capacity. MS-6 reservoir sizing would also be based on FC-1 requirements.

The density of MS-6 is approximately 25% higher than MIL-H-83282, therefore,

$$Q_{MS-6} = Q_{83282}\sqrt{\frac{\rho_{83282}}{1.25\rho_{83282}}} = .894Q_{83282}$$

To maintain the same flow rate, the MS-6 orifice area must be 1/.894 = 1.118 times the MIL-H-83282 orifice area. If valve stroke is unchanged, the orifice slot area must be increased by 11.8%, theoretically. (see section 2.3.4) The effect on slot width would be as follows:

	Servo Valve Slot Width, in.	
Actuator	LHS design using MIL-H-83282	Re-sized slot using MS-6
Rudder	.005	.0056
UHT	.030	.0335
Aileron	.008	.0089
Spoiler	.010	.0118
RFI	.005	.0056

Null Leakage - The higher viscosity and density of MS-6 reduces null leakage significantly. At temperatures above +100°F, leakage rates using MS-6 will be about 1/6 of the rate using MIL-H-83282.

Actuator	Null Leakage Using MIL-H-83282,cc/min	Null Leakage Using MS-6, cc/min
UHT	16	3.0
Rudder	21.5	4.0
Aileron	4.6	.9

Although the lower null leakage results in smaller power losses and heat generation, too low a rate could raise fluid temperatures in the pump loop by not distributing the hot fluid to extremities of the system for heat dissipation. The effect of leakage rate on heat dissipation for a typical distribution line is shown on Figure 34. For the assumed configuration, maximum heat removal occurs at a leakage rate near 60 cc/min. At rates above 200 cc/min., a net heat gain occurs.

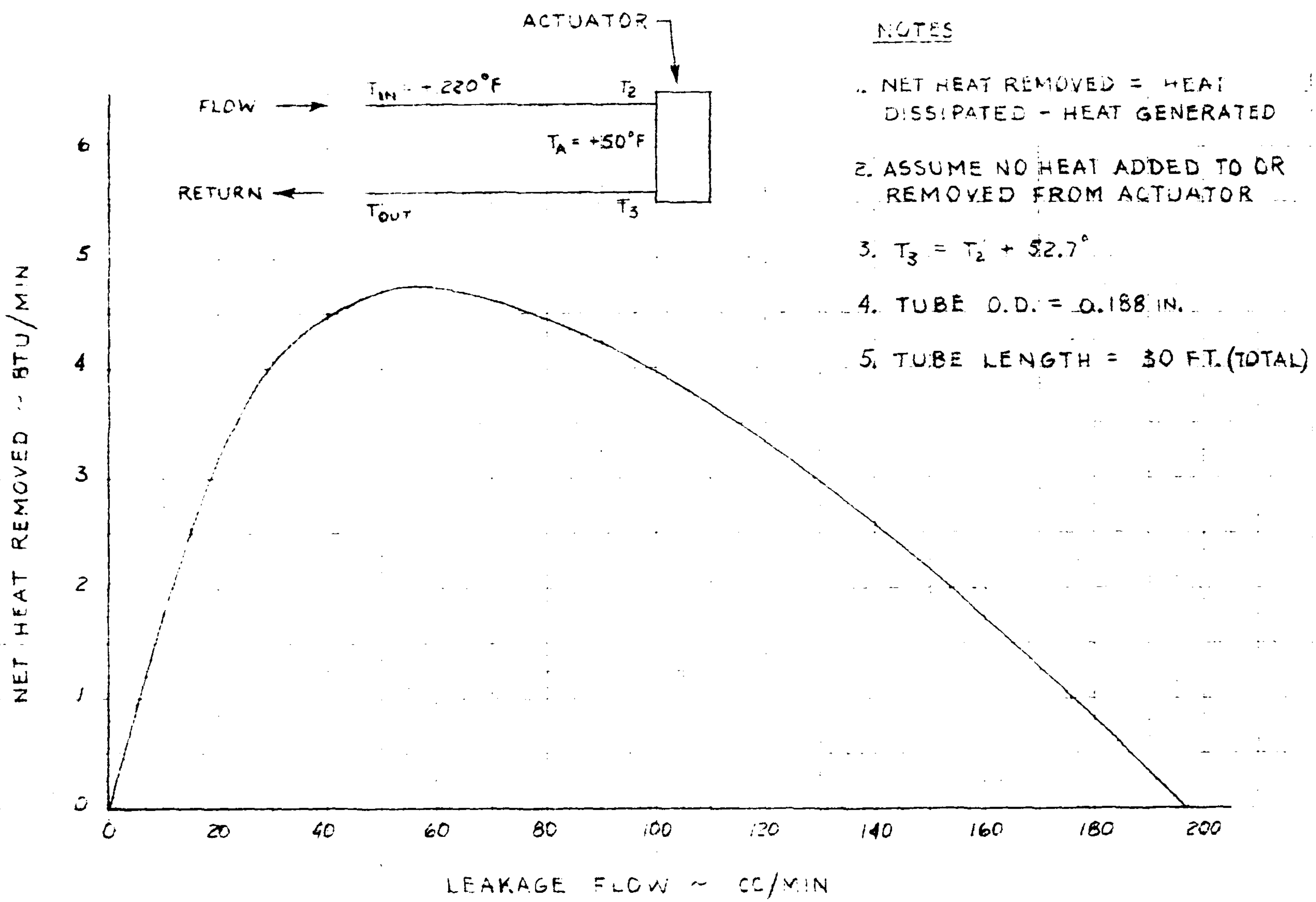


FIGURE 34. HEAT DISSIPATION FROM DISTRIBUTION LINES

If required, the spool/sleeve diametral clearance could be increased to raise leakage rates. Leakage is proportional to the cube of the diametral clearance, thus,

$$\frac{Q_{83282}}{Q_{MS-6}} \quad \left(\frac{C_{d_{83282}}}{C_{d_{MS-6}}}\right)^3$$

where,

Q = null leakage rate

C_d = diametral clearance

As an example, the spool/sleeve diametral clearance in the UHT valve would be increased by 75% as follows:

$$\frac{C_{d_{83282}}}{C_{d_{MS-6}}} \quad \left(\frac{16}{3}\right)^{1/3} = 1.75$$

3.1.3.6 Miscellaneous Components

Seals - Compatibility of elastomeric seals with the fluid is a requirement in the design of all hydraulic systems. Fortunately, MS-6 is compatible with standard nitrile (Buna N) elastomers. Hydraulic fluids normally cause O-rings to swell. Volume swell with MS-6 is low, compared to swell experienced with MIL-H-83282, and could cause sealing problems at low temperatures. Evaluation of the performance Buna N O-rings with MS-6 at extreme temperatures (-65 to +275°F) is recommended. Testing should be conducted using standard gland dimensions in both static and dynamic seal applications.

Filters - The higher viscosity and density of MS-6 will raise the pressure drop across both the housing and element. Housing pressure drop is proportional to fluid density, while element losses are related to viscosity. Housing pressure drop will be 1.25 to 1.5 times higher using MS-6; element losses will be 4 to 5 times higher. A comparison of rated flow pressure drop at +100°F with the A-7 LHS pressure filter is given below.

	Pressure Drop, psi	
	A-7 LHS using MIL-H-83282	A-7 LHS using MS-6
Housing	35	44
Element	10	40
TOTAL	45	84

The higher pressure drop occurring with MS-6 will have little effect on filter performance and will add only slightly to overall system pressure losses and heat generation.

The filter housing contains a ΔP indicator which operates when excessive pressure drop occurs due to a loaded-up element. The setting would have to be increased to retain the same dirt holding capacity.

Return system filters contain a by-pass relief valve. The setting would have to be increased to accommodate MS-6. This would require use of a different rate spring to keep the same full flow pressure drop.

Quick Disconnects - Effects of using MS-6 on disconnect performance are minor. No design changes are required.

Restrictors - Restrictor flow rates will be reduced approximately 15% due to the higher density and viscosity of MS-6. All restrictors must therefore be re-sized. Tube connections on the restrictor must also be changed where transmission line diameters are increased. If the restrictor is used in series with other components, the total circuit must be evaluated before re-sizing the restrictor. For example, if the restrictor was used in series with a shut-off valve and the valve was not re-sized, a larger pressure drop would occur across the valve and a smaller drop across the restrictor would be used.

Check Valves - Pressure losses would be approximately 1.8 times higher using MS-6 fluid in existing valves; this is not considered serious. An insignificant ΔP change would occur if valve size must be increased to accommodate larger line diameters.

Selector Valves - The leading edge flap and speed brake selector valves both have restrictors in their subsystems. Any increase in pressure drop across the valve due to using MS-6 could be compensated when the orifices are re-sized. A small increase in pressure loss across the AFCS actuator shut-off valves due to using MS-6 would be acceptable; no changes would be required in these valves.

Swivels - Each wing fold swivel contains six separate swivels; two in the flap circuit and four in the aileron circuit. Swivel porting must be changed from -3 to -4 to accommodate a larger line size (see section 3.1.3.2). Any increase in pressure drop across the swivel due to using MS-6 would be partially offset by the larger port size.

Hoses - The only hoses affected by the use of MS-6 are at the aileron actuator. Pressure drop increases from 49 to 226 psi if the existing -3 size hoses are used; the ΔP increase drops to 66 psi if -4 hoses are used. Since tubing in the aileron distribution system must be increased to -4 size, the hoses should also be -4. The -4 hoses may not have sufficient flexibility, however, for the aileron installation. (The original 3000 psi installation had special, high flexibility -4 hoses.) If -3 size hose is used, the higher pressure drop could be offset by increasing the diameter of a portion of wing system plumbing.

Coiled Tubing - The lower flow requirements resulting from operation at 8000 psi permitted much of the -4 size tubing in the A-7 lightweight hydraulic system to be reduced to -3. Use of -3 tubing permitted coil tube installations not possible with -4 lines because of space constraints. Coiled tubing were utilized in the leading edge flap, spoiler, and roll-feel-isolation actuator installations.

MS-6 flow rates in the flap coils would produce a total of 430 psi pressure drop per cylinder at operating temperatures (vs. 86 psi using MIL-H-83282). This increase can be compensated for in sizing the flap circuit restrictors and poses no problem.

MS-6 flow rates in the spoiler application produces a pressure loss which seriously reduces actuator working pressure. The total loss (pressure + return) is 1840 psi at operating temperature (vs. 360 psi using MIL-H-83282). Since a larger tubing size cannot be used, the only apparent option would be to use line extension units similar to those employed in the original 3000 psi installation.

Flow requirements of the roll-feel-isolation actuator are low, and the MS-6 pressure loss in the coils would be acceptable.

3.1.3.7 Power Requirements - Flight control actuator power requirements establish system flow demand. Power consumption is proportional to actuator net area. As discussed in section 3.1.4.3, reduced actuator stiffness resulting from use of MS-6 was not serious and no change in actuator size was necessary. Therefore, system hydraulic power requirements would be the same using MIL-H-83282 and MS-6.

Power extraction from the engine is a function of pump size, efficiency, and system flow demand. With flow demand unchanged, pump size would be unchanged. Pump efficiency using MS-6 is not known, but it was assumed that if a pump were designed to use MS-6, overall efficiency levels would be comparable to pumps designed to use MIL-H-83282. If this were true, then power extraction would be the same as for the existing A-7 LHS pumps.

3.1.3.8 System Cooling - Factors which affect heat generation and temperature buildup are:

- Pump size and efficiency
- Distribution line losses
- Power losses across restrictors, valves, filters, etc.
- Power losses at servo actuators
- Air flow and temperature
- System surface area
- System and fluid weight and specific heat

Effects of using MS-6 fluid on each of the above factors will be examined. System cooling requirements will then be discussed.

Pump - More heat is generated by the pump than by any other component in a hydraulic system. Maximum heat rejection of the A-7E 8000 psi pump is currently 380 BTU/min. Heat generation increases with pump size and inefficiency. Since no change in pump size or efficiency would occur as a result of using MS-6 (see section 3.1.3.7), there would be no difference in heat generated by a pump designed to use MS-6.

Distribution Lines - Pressure losses which occur in transmission lines are a source of heat. Tube diameters were increased to accommodate the higher viscosity of MS-6 and maintain pressure losses at acceptable levels (see section 3.1.3.2). Since pressure losses in the re-sized MS-6 system and in the original system using MIL-H-83282 were approximately equal, there would be no difference in heat generation.

Restrictors, Valves, Filters, Etc. - All restrictors must be re-sized for use with MS-6 fluid. The new size can be selected to offset increased pressure losses which occur across unchanged components in series with the restrictor. Since the overall circuit pressure drop is unchanged, there would be no difference in heat generation.

Components where added heat generation would occur were as follows:

Components Not Re-Sized	Pressure Drop Increase at Full Flow @ +200°F, psi	Power Loss, hp	Heat Added, BTU/min
Pressure Filter	22	0.128	5.44
Return Filter	14	0.082	3.46
Pressure Disconnect	15	0.088	3.71
Wing Fold Swivel	48	0.040	1.71
Check Valves	24	0.140	5.94
Aileron Hoses	34	0.029	1.21
	TOTAL	0.510	21.48

The maximum power loss at full flow is approximately 0.5 hp (21.5 BTU/min). At steady state, low flow conditions, power losses would reduce to less than 2 BTU/min or 0.5% of the heat generated by the pump. Therefore, heat added by the above components was considered negligible.

Servo Actuators - Pressure drops across control valve orifices result in significant heat generation. For example, under no load operating conditions, all of system pressure is throttled across the valve. Since flow requirements for the MIL-H-83282 and MS-6 actuators are the same (see section 3.1.3.5), heat generated by control valves will also be the same. Fluid temperature rise will be approximately 11% higher with MS-6, however, than with MIL-H-83282.

$$\frac{\Delta T_{MS-6}}{\Delta T_{83282}} = \frac{(\rho \cdot C)_{83282}}{(\rho \cdot C)_{MS-6}}$$

where,

ΔT = fluid temperature rise

ρ = fluid mass density

C = fluid specific heat

At +200°F,

$$\Delta T_{MS-6} = 1.109 \times \Delta T_{83282}$$

As an example, assume all of system pressure (8000 psi) is dropped across the UHT actuator control valve at rated flow (4 gpm) using MIL-H-83282 fluid. Heat generated would be 786 BTU/min and fluid temperature would increase 52.7°F. The temperature rise using MS-6 would be 52.7 x 1.109 = 58.5°F. The higher temperature rise would cause an increase in the rate of system temperature buildup, but would have no effect on the final stabilization temperature (see Cooling Requirements).

Null leakage using MS-6 is significantly less than with MIL-H-83282. If leakage is too low, however, the heat dissipation potential of the long distribution lines would not be utilized (see section 3.1.3.5). For this study, it was assumed that null leakage was maintained at the existing LHS level by increasing spool/sleeve diametral clearance. No change in heat generation would therefore occur.

Air Flow and Temperature - Cooling requirements are affected by compartment air flow and ambient temperature. The increase in system volume due to larger lines, reservoir, etc., should have no effect on compartment air flow or temperature.

System Surface Area - The larger line and reservoir sizes required by MS-6 increase surface area and improve heat dissipation. The surface area of the re-sized MS-6 system is approximately 8% greater than the original system (38.9 vs. 36 ft^2). The larger area would reduce system stabilization temperature and result in a slightly cooler running system.

System and Fluid Weight and Specific Heat - Weight and specific heat of system fluid and components affect the rate of temperature buildup. The heavier weight of the MS-6 system will tend to decrease the rate of rise. The lower specific heat of MS-6 combined with the larger volume of fluid will tend to increase the rate of temperature rise. A summary of the effect of differences in weight and specific heat is as follows:

	A-7 LHS using MIL-H-83282	Re-sized LHS using MS-6
Specific heat of fluid, C_F, BTU/lb/^oF	0.545	0.391
Specific heat of system materials, C_S, BTU/lb/^oF	0.12	0.12
*Weight of fluid in system, W_F, lb	25.0	33.2
*Weight of system components, W_S, lb	215.0	224.4
$C_F \times W_F$, BTU/^oF	13.63	12.98
$C_S \times W_S$, BTU/^oF	25.80	26.93
$\sum C \times W$, BTU/^oF	39.43	39.91
$\frac{1}{\sum C \times W}$, ^oF/BTU	.0254	.0251

*See section 3.1.4.

The temperature rise per BTU added is nearly identical for both configurations. The calculation excludes heat dissipation and assumes an even temperature distribution.

Cooling Requirements - A summary of the effects of using MS-6 on the parameters which affect cooling requirements is given below:

Parameter	Effect of Using MS-6
Pump	Negligible
Distribution Lines	Negligible
Restrictors, Valves, etc.	Negligible
Servo Actuators	Negligible
Air Flow and Temperature	Negligible
System Surface Area	Reduces cooling requirements
System and Oil Weight and Specific Heat	Negligible

The effect of the increase in surface area on system bulk fluid temperature was analyzed. The study was conducted on FC-1, since this was the smaller of the two systems. Although the computations did not take into account differences in compartment temperatures, air flows, or local hot spots, the results provide an indication of trends and magnitude of changes. The simplified thermal analysis was based on

$$T_{D_t} = \frac{E_L}{\sum K A}\left(1 - e^{-\frac{\sum K A}{\sum C W} t}\right) + T_{D_i}\, e^{-\frac{\sum K A}{\sum C W} t} \qquad \text{(Ref. 12)}$$

and

$$T_F = T_{D_t} + T_A$$

where,

T_F = bulk fluid temperature, $^{\circ}F$

T_{D_i} = fluid temperature minus ambient temperature at time t=0, $^{\circ}F$

T_{D_t} = fluid temperature minus ambient temperature at time t, $^{\circ}F$

T_A = ambient temperature, $^{\circ}F$

E_L = system heat load, BTU/hr.

K = overall heat transfer coefficient, $BTU/hr/ft^2/^{\circ}F$

A = surface area of system lines, actuators, valves, etc., ft^2

C = specific heat, $BTU/lb/^{\circ}F$

W = weight of fuid, components, tubing, etc., lb

t = time, hr.

Values used for the equation parameters were as follows:

	A-7 LHS using MIL-H-83282	Re-sized System using MS-6
System heat load E_L, BTU/hr.	22,800	22,800
Specific heat, C_F, fluid, BTU/lb/°F	0.545	0.391
Specific heat, average, C_S, Ti, Steel BTU/lb/°F	0.12	0.12
*Weight of Fluid, W_F, lb	25.0	33.2
*Weight of Components W_C, lb	215	224.4
Heat Transfer Coefficient, Average for System, BTU/hr/ft²/°F	2.5	2.5
Surface Area, Total System, ft²	36.0	38.9
Ambient Temperature, °F	+20 & +90°F	+20 & +90°F

*See section 3.1.4.

Data derived from the analysis are plotted on Figure 35 As shown, MS-6 system bulk fluid temperatures run slightly lower than MIL-H-83282 system temperatures Although the improvement may be partially offset by minor unaccounted for losses, the analysis does indicate there would be no significant change in cooling requirements when using MS-6 fluid.

3.1.4 Weight and Space Analysis

3.1.4.1 Introduction - A weight and volume study was conducted on the A-7E flight control system and reported in Reference 5. The analysis compared the weight and volume of a system operating at 8000 psi with an equivalent system operating at 3000 psi. The utility system was not evaluated in the Reference 5 study.

3.1.4.2 Approach - The MS-6 weight and space analysis covers both the flight control and utility systems. The study compares component weight and volume in an A-7E 8000 psi hydraulic system using MIL-H-83282 with a re-sized 8000 psi system using MS-6. Data from Reference 5 was used as a starting point for the flight control system analysis; only those components which had a size change were studied. A qualitative procedure was used for the utility system because the cost of a quantitative analysis was prohibitive.

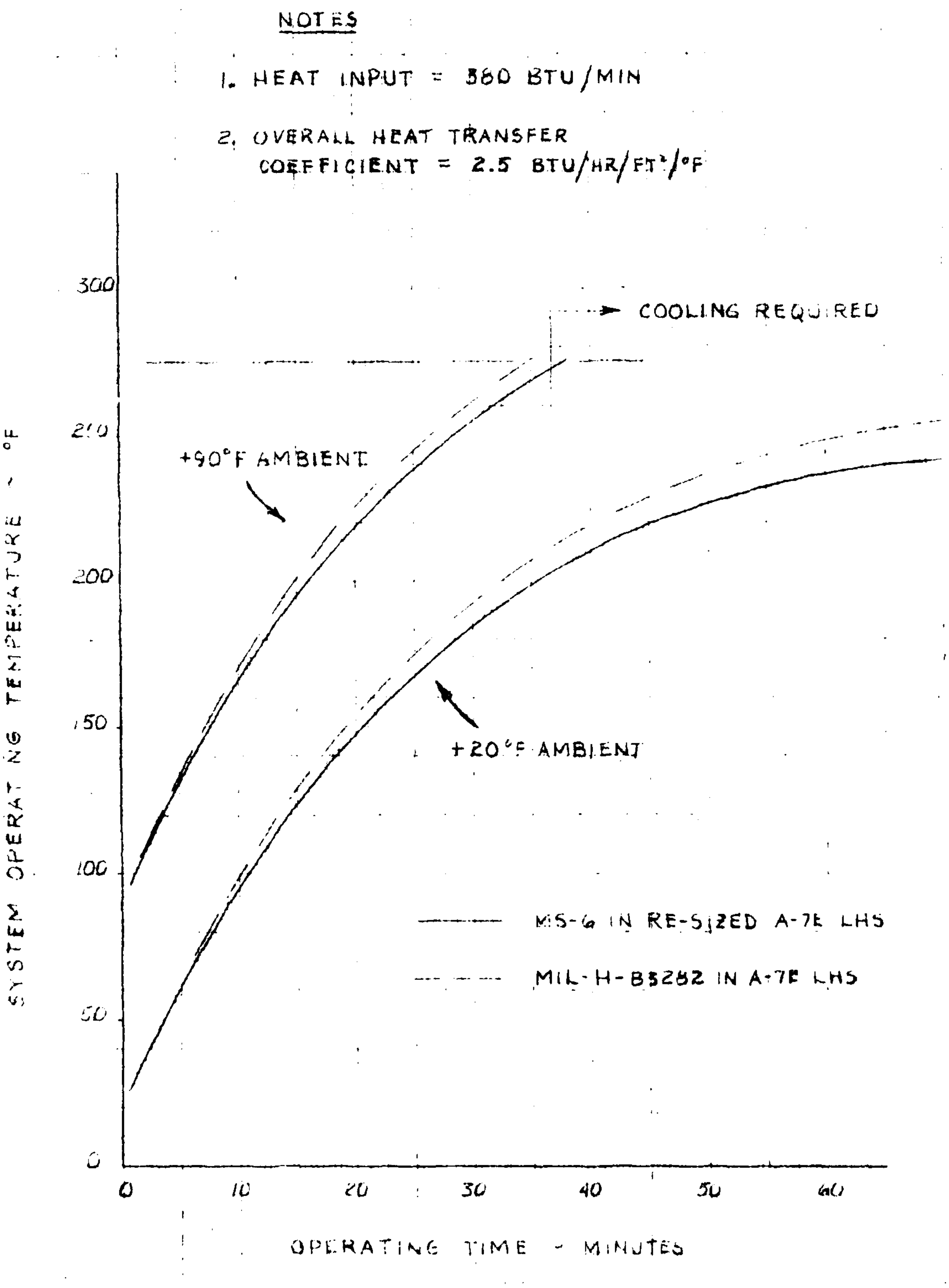


FIGURE 35. COMPARISON OF STABILIZATION TEMPERATURES

3.1.4.3 Flight Control System - A summary of the weight and volume changes which would result from the use of MS-6 in the A-7E 8000 psi flight control system is presented on Table 8. Weight and volume comparisons between the MIL-H-83282 and MS-6 systems are summarized below:

	*A-7 LHS using MIL-H-83282	Re-sized LHS using MS-6	Savings Change
Weight	449.7 lb	502.8 lb	-11.8%
Volume	5207 in^3	5538 in^3	- 6.3%

Weight and volume savings achieved by operating at 8000 psi instead of 3000 psi are compared below:

	Savings Compared to 3000 psi System	
	*8000 psi System using MIL-H-83282	8000 psi System using MS-6
Weight	30.2%	22.0%
Volume	36.3%	32.2%

*Data Source: Reference 5.

The principal reason for operating at 8000 psi is smaller and lighter weight hydraulic components. Weight savings of 30% and space savings of 40% are ultimate goals of the LHS Advanced Development Program, Reference 5. Use of MS-6 in lightweight hydraulic systems would reduce potential weight and space savings approximately 20%.

3.1.4.4 Utility System - This study examines component weight and volume changes which would occur if the fluid in a future 8000 psi utility system were changed from MIL-H-83282 to MS-6. Results are summarized on Table 9; analysis details are given in Appendix C. The change to MS-6 would require a small increase in system weight and volume--probably less than 10%. Principal modifications would be larger tubing sizes in four subsystems and re-sized orifices in eleven subsystems.

TABLE 8. A-7E 8000 PSI Flight Control System, Impact of MS-6 on Weight and Volume

[1]ITEM	DESCRIPTION	MODIFICATION FOR MS-6	DRY WEIGHT, LB	EXTERNAL VOLUME, IN^3
1	Pump (2)	Suction port size increase, -10 to -16	+0.24	-
2,3	Reservoir (2)	Capacity Increase, 320 to 351 in^3	+1.75	+72.0
19	Disconnect, Suction (2)	Size Increase, -10 to -16	+2.79	+19
29-32	Restrictor, L.E. (8)	Port Size Increase, -3 to -4	+1.1	-
37	Swivel, Wing Fold (2)	Port Size Increase, -3 to -4	+0.3	-
44,46, 54	Check Valve (6)	Port Size Increase	+0.3	-
69,70	Hose, Suction (2)	Size Increase -10 to -16	+2.79	+19
77,84	Hose, Aileron Actuator (8)	Size Increase, -3 to -4	+0.51	+ 8
None	[2]Plumbing, 285 Ft.	Size Increase, -3 to -4	+8.75	+73
None	[2]Plumbing, 125 Ft.	Size Increase, -4 to -6	+3.32	+41
None	[2]Plumbing, 49 Ft.	Size Increase, -6 to -8	+3.45	+51
None	[2]Plumbing, 18 Ft.	Size Increase, -10 to -16	+2.56	+103
None	Line Extension Units, Spoiler (8)	Units Replace -3 Coiled Tubing	+5.25	-55
None	Fluid, Hydraulic	Volume Increase, Density Increase	+20.0	-
		TOTALS	+53.11 lb.	+331 in^3

[1]See Reference 5, Drawing 8696-580001

[2]Includes Tubing, Fittings, and Clamps.

TABLE 9. A-7E 8000 psi Utility System,
Impact of MS-6 on Weight and Volume

SUBSYSTEM	IMPACT OF MS-6 ON WEIGHT & VOLUME
Air Refueling	None
Gun Drive	Sizable Increase
Gun Drive Vent	None
Trailing Edge Flaps	None
Wing Fold	None
Nose Gear Steering	Small Increase
Wheel Brakes	None
Landing Gear	Small Increase
Arresting Hook	None
Launch Bar	None
Emergency Power Package Control	None
Emergency Accumulator	None
Utility Power Circuit	Sizable Increase
Total impact of a change to MS-6 on A-7E 8000 psi utility system weight and volume: Small Increase	

3.2 F-14 HORIZONTAL STABILIZER ACTUATOR

3.2.1 Introduction

An application study of very high pressure hydraulic systems to the F-14 aircraft was made by Grumman Aerospace Corporation and documented in Reference 3. The report provides detail information to substantiate weight and volume savings achieved if the operating pressure level were increased from 3000 psi using MIL-H-5606 fluid to 8000 psi using MIL-H-27601 (a shear stable hydrocarbon base fluid).

The F-14 horizontal stabilizer actuator was selected for the current study because stiffness was a critical design factor in the original installation. Since the bulk modulus of MS-6 is lower than that of MIL-H-5606 and the increase in operating pressure does not compensate for the difference, the actuator must be modified to maintain the required stiffness. An analysis was made to estimate weight and volume changes if the 3000 psi actuator using MIL-H-5606 were designed to operate at 8000 psi using MS-6 fluid.

3.2.2 Actuator Design Parameters

Basic design parameters of the F-14 horizontal stabilizer actuator were as follows:

Actuator type:	dual tandem, unbalanced, redundant rods
Hydraulic fluid:	MIL-H-5606
Operating pressure:	3000 psi
Nominal out-stroke load: (2 systems)	123,000 lb.
Length extended:	54.00 in.
Stroke:	8.75 in.
Bore diameter:	5.477 in.
Rod diameter:	2.875 in.
Required spring rate:	550,000 lb/in (min.)
Weight:	93.92 lb

3.2.3 Analysis Results

Analysis details are given in Appendix D. The results disclosed that an actuator operating with MS-6 at 8000 psi can meet stiffness and envelope requirements of the F-14 3000 psi actuator. Basic parameters of the 3000 psi and 8000 psi actuators were as follows:

	*F-14 Actuator using MIL-H-5606 at 3000 psi	Re-sized Actuator using MS-6 at 8000 psi
Actuator spring rate (1 system)	550,000 lb/in. (minimum)	560,000 lb/in.
Maximum force output (1 system)	51,000 lb	115,000 lb
Stroke	8.75 in.	8.75 in.
Bore diameter	5.477 in.	5.375 in.
Rod diameter	2.875 in.	3.25 in.
Net effective area (1 system)	17.1 in^2	14.4 in^2
Weight	93.92 lb	Essentially no change
Volume	1904 in^3	Essentially no change

*Data source: Reference 3.

Output load capability of the 8000 psi actuator was more than twice that of the 3000 psi actuator, and probably would not be acceptable with existing backup mounting structure. Weight and volume savings normally achieved by operating at 8000 psi did not occur.

4.0 CONCLUSIONS

4.1 EVALUATION TESTS

A summary of evaluation test results is presented on Table 10. The tests indicated that MS-6 can be used in 8000 psi systems, but two problem areas were encountered: poor pump performance and hydraulic fluid foaming. Performance of the 8000 psi test pump was seriously degraded by the MS-6 silicone base fluid; the pump was designed to use a hydrocarbon base fluid. MS-6 has a relatively high gas solubility. This affinity for gas and the tendency to foam contributed to poor pump performance by reducing fluid bulk modulus, lowering fluid lubricity, and causing possible cavitation.

4.2 APPLICATION STUDY

A summary of the A-7E application study is given on Table 11. Weight and space savings that would result from using MS-6 at 8000 psi in the A-7E primary flight control system were as follows:

	Compared to Using MIL-H-83282 at 3000 psi, Savings	Compared to Using MIL-H-83282 at 8000 psi, Change in Savings
Weight	22.0%	-11.8%
Volume	32.2%	- 6.3%

Weight and space savings usually achieved by operating at 8000 psi did not occur when MS-6 was used in the F-14 horizontal stabilizer actuator. The output load capability of the re-sized actuator was more than twice the required output and probably would not be acceptable with the existing backup mounting structure.

4.3 UTILIZATION OF MS-6

Results of the evaluation test and application study disclosed MS-6 can be employed in 8000 psi systems to reduce potential losses caused by hydraulic fires. Minor design modifications would be required in pumps, actuators, and miscellaneous components to accommodate MS-6; use of MS-6 would incur weight and space penalties, however. An area that requires development effort is the tendency of MS-6 to foam. A means of reducing this tendency is needed to achieve satisfactory pump performance.

TABLE 10. Summary of Evaluation Test Results

Item	Summary
Pump	- Use of MS-6 seriously degraded performance of an 8000 psi pump designed for MIL-H-83282 fluid
Transmission Line Losses	- MS-6 losses 1.5 to 6 times higher compared to MIL-H-83282
Pressure Surges	- MS-6 surges 29% higher compared to MIL-H-83282 - Test data should be verified due to poor pump performance
Flow Restrictors	- MS-6 flow 16% less compared to MIL-H-83282
Servo Actuator	- Actuator stiffness 22% less using MS-6 than when using MIL-H-83282 - Actuator/system natural frequency 13% lower using MS-6
Flow Control Valve	- MS-6 flow gain 15% less compared to MIL-H-83282 - MS-6 pressure gain 11% higher - MS-6 null leakage 68% lower
Solenoid Valve	- MS-6 internal leakage significantly lower compared to MIL-H-83282
Tendency of MS-6 to Foam	- Reduced fluid bulk modulus - Increased possibility for cavitation - Lowered fluid lubricity

TABLE 11. Summary of A-7E Application Study

I. DESIGN MODIFICATIONS

Pump	- Pump must be designed for MS-6 fluid to obtain satisfactory performance
Distribution Lines	- 285 ft. of -3 size changed to -4 125 ft. of -4 size changed to -6 49 ft. of -6 size changed to -8
Pump Suction Line	- 18 ft. of -10 size changed to -16 - -12 size disconnect changed to -16
Reservoir	- Volume increased 31 in^3
Primary Flight Control Actuator Stiffness	- Reduced stiffness not considered serious. Piston net area not changed.
	- Control valve slot width increased 12%
	- Spool/sleeve diametral clearance increased to obtain satisfactory null leakage
Seals	- Gland dimensions may require modification
Filters	- ΔP indicator and by-pass relief valve settings adjusted
Restrictors	- Orifice areas increased 15%
	- Orifice sizing can be adjusted to compensate for higher pressure losses in other parts of circuit
Check Valves	- No modification required except where line sizes changed
Quick Disconnects	- No modification required except where line sizes changed
Selector Valves	- No modification required
Wing Fold Swivel	- No modification required except port size increased from -3 to -4
Aileron Hose	- Use of -3 size hose requires size increase in wing plumbing
Coiled Tubing	- No modification required at L.E. flap and RFI actuators
	- Use of coiled tubing at spoiler actuator not practical. Line extension units required
Power Requirements	- No modification required
Cooling Requirements	- No modification required

II. WEIGHT AND SPACE ANALYSIS

Primary Flight Control Systems

	Original A-7E 3000 psi system using MIL-H-5606	A-7 LHS using MIL-H-83282	Re-sized LHS using MS-6
Weight, lb	644.4	449.7	502.8
Volume, in^3	8173	5207	5538

5.0 RECOMMENDATIONS

The MS-6 evaluation tests disclosed that further development effort is needed.

Pump

An 8000 psi pump designed to use MS-6 fluid should be obtained. Performance and endurance testing would provide information required to establish the suitability of MS-6 for use in lightweight hydraulic systems.

Tendency of MS-6 to Foam

Two approaches to alleviate foaming should be considered: (1) use of a bootstrap reservoir and (2) use of an anti-foam agent.

1. Bootstrap Reservoir - A bootstrap type reservoir would eliminate contact of MS-6 with pressurized gas and lessen the foam problem. This type reservoir should be employed when the MS-6 8000 psi pump is tested.

2. Anti-Foam Agent - Addition of a suitable foam inhibitor to MS-6 is required. The additive must have long life, be effective at high temperatures, and resist breakdown under severe fluid shearing conditions. It is recommended that the anti-foam agent be evaluated during the MS-6 8000 psi pump tests.

Elastomer Swell

Swell of nitrile O-rings exposed to MS-6 is low and could result in seal leakage. Extreme temperature tests should be performed on both static and dynamic seals to determine if gland dimensions require modification for MS-6 systems.

6.0 REFERENCES

Reference No.

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2 J. N. Demarchi, Dynamic Response Test of Very High Pressure Fluid Power Systems, NR70H-533, North American Rockwell Corporation, Columbus Division, Contract N00156-70-C-1152, 9 December 1970, Unclassified.

3 J. N. Demarchi and R. K. Haning, Application of Very High Pressure Hydraulic Systems to Aircraft, NR72H-20, Columbus Aircraft Division, North American Rockwell Corporation, Contract N62269-71-C-0147, March 1972, Unclassified.

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7 H. E. Merritt, Hydraulic Control Systems. New York: Wiley & Sons, 1967.

8 J. Stauffer, Dynamic Response of Very High Pressure Fluid Power Systems, NR69H-65, North American Rockwell Corporation, Columbus Division, Contract N00019-68-C-0352, 16 April 1969, Unclassified.

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AD-A106 268 ROCKWELL INTERNATIONAL COLUMBUS OH NORTH AMERICAN AI--ETC F/G 11/8
EVALUATION OF MS-6 FIRE-RESISTANT FLUID FOR 8000 PSI LIGHTWEIGH--ETC(U)
OCT 81 J N DEMARCHI, R K HANING N62269-79-C-0287
UNCLASSIFIED NR81H-69 NADC-79120-60 NL

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10 Component Specification, LHS-8810, Pumps, Hydraulic Variable Delivery, 8000 psi, North American Aircraft Division, Rockwell International Corporation, dated 25 August 1980.

11 R. L. Peeler and H. S. Yaplee, Isothermal Bulk Modulus of Hydraulic Fluid - Hydrogen Mixtures, California Research Corporation, Contract AF33(616)-3184, October 1957.

12 G. R. Keller, Hydraulic System Analysis, 3rd Edition, Industrial Publishing Co., Cleveland, Ohio, 1978.

7.0 LIST OF ABBREVIATIONS

AFCS	automatic flight control system
BTU/min	British Thermal Units per minute
C1	cylinder port #1
cc/min	cubic centimeters per minute
CRES	corrosion resistant
cs	centistoke
EPP	emergency power package
^{o}F	degrees Fahrenheit
FC-1	flight control #1
ft	feet
ft/sec	feet per second
gpm	gallons per minute
hp	horsepower
hr	hour
Hz	Hertz (cycles per second)
I.B.	inboard
I.D.	inside diameter
in.	inch
in^2	square inches
in^3	cubic inches
in/sec	inches per second
lb	pound
lb/in	pounds per inch
L.E.	leading edge
LHS	lightweight hydraulic system

max.	maximum
M/N	model number
min	minute (time)
NAAD	North American Aircraft Division
NADC	Naval Air Development Center
NAS	National Aerospace Standard
No.	number
O.B.	outboard
O.D.	outside diameter
ΔP	differential pressure
P/N	part number
psi	pounds per square inch
psig	pounds per square inch gage pressure
rpm	revolutions per minute
sec	second (time)
S/N	serial number
UHT	unit horizontal tail

APPENDIX A

FLUID PROPERTY DATA

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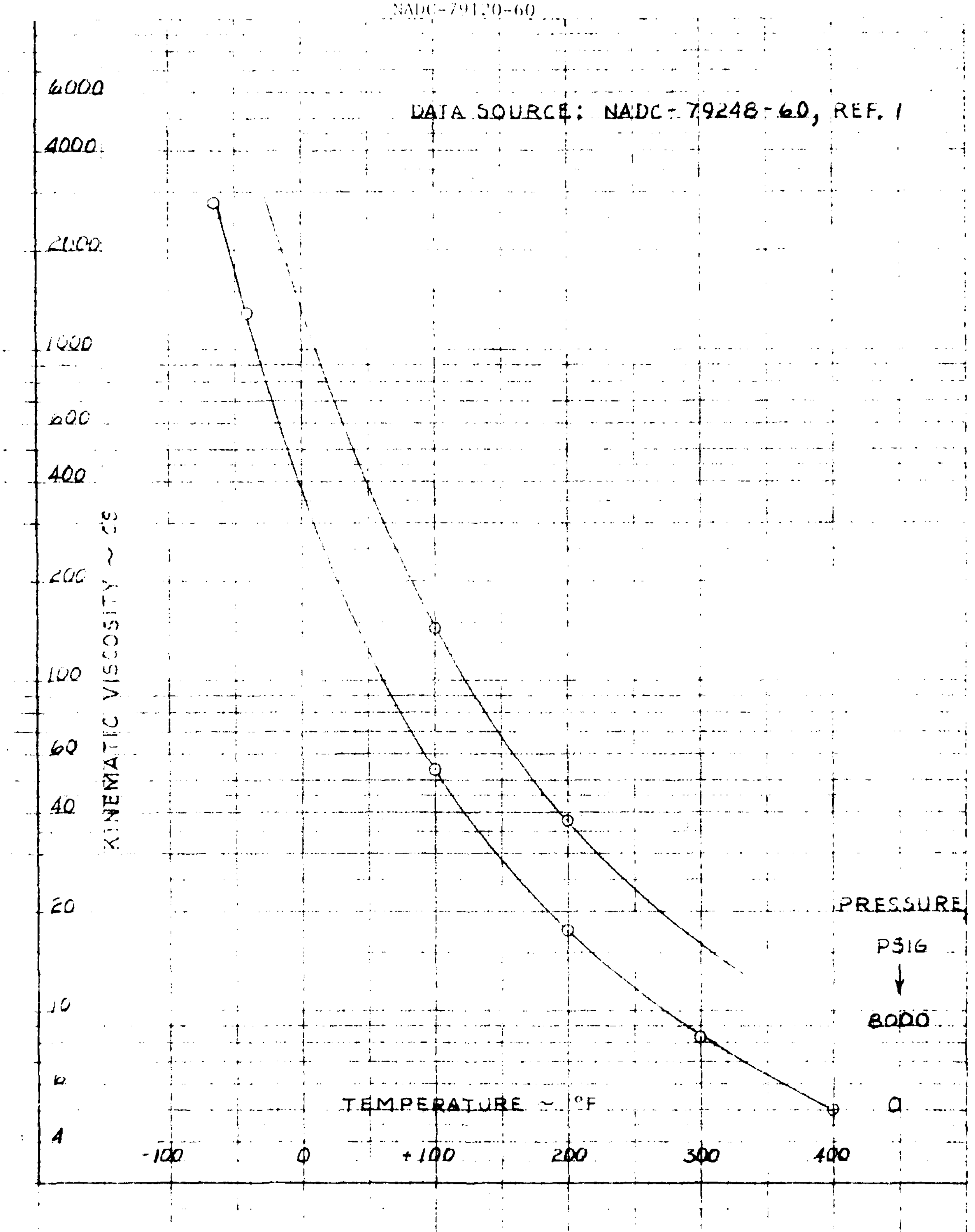

FIGURE A-1. VISCOSITY OF MS-6

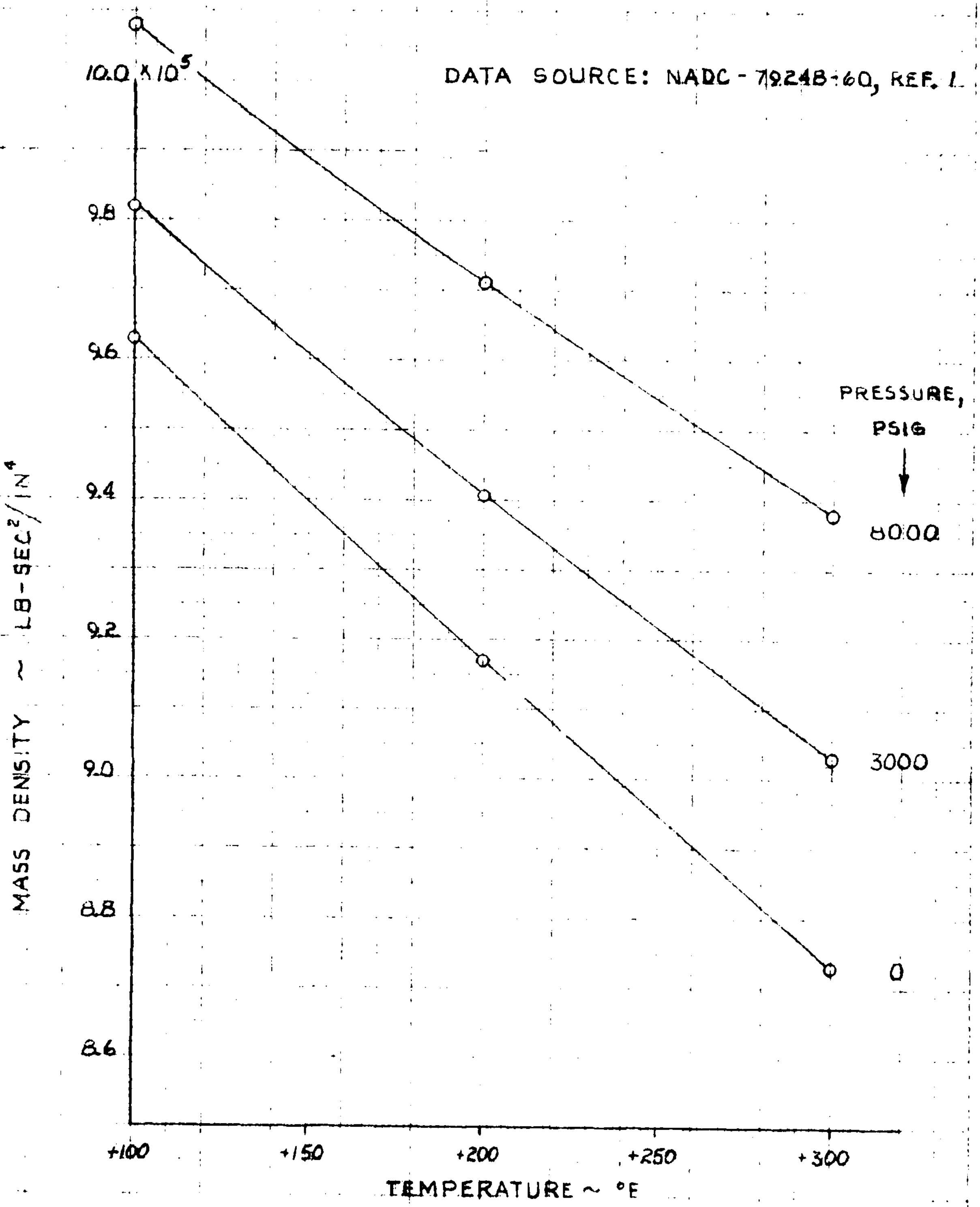

FIGURE A-2. MASS DENSITY OF MS-6

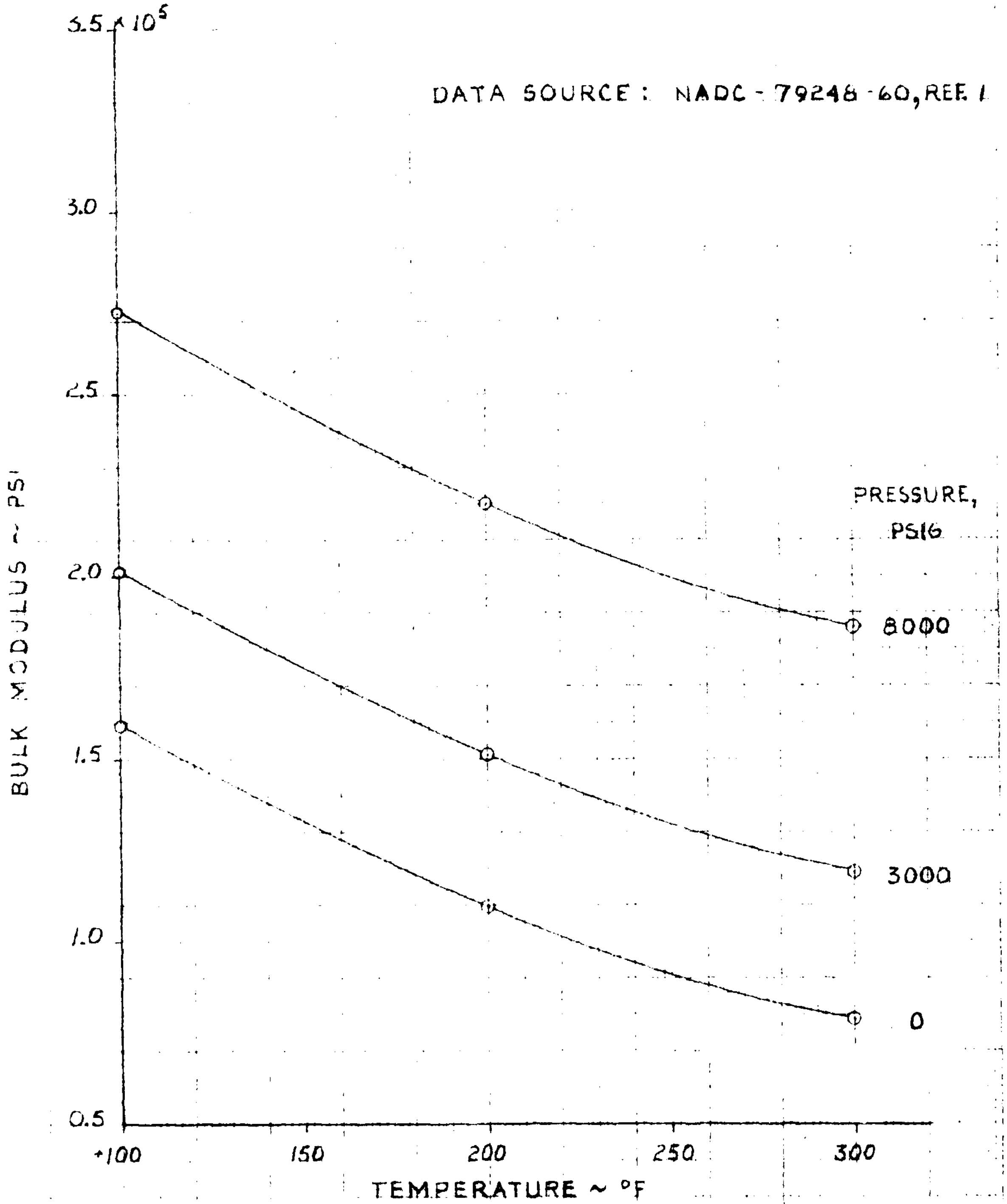

FIGURE A-3. ADIABATIC TANGENT BULK MODULUS OF MS-6

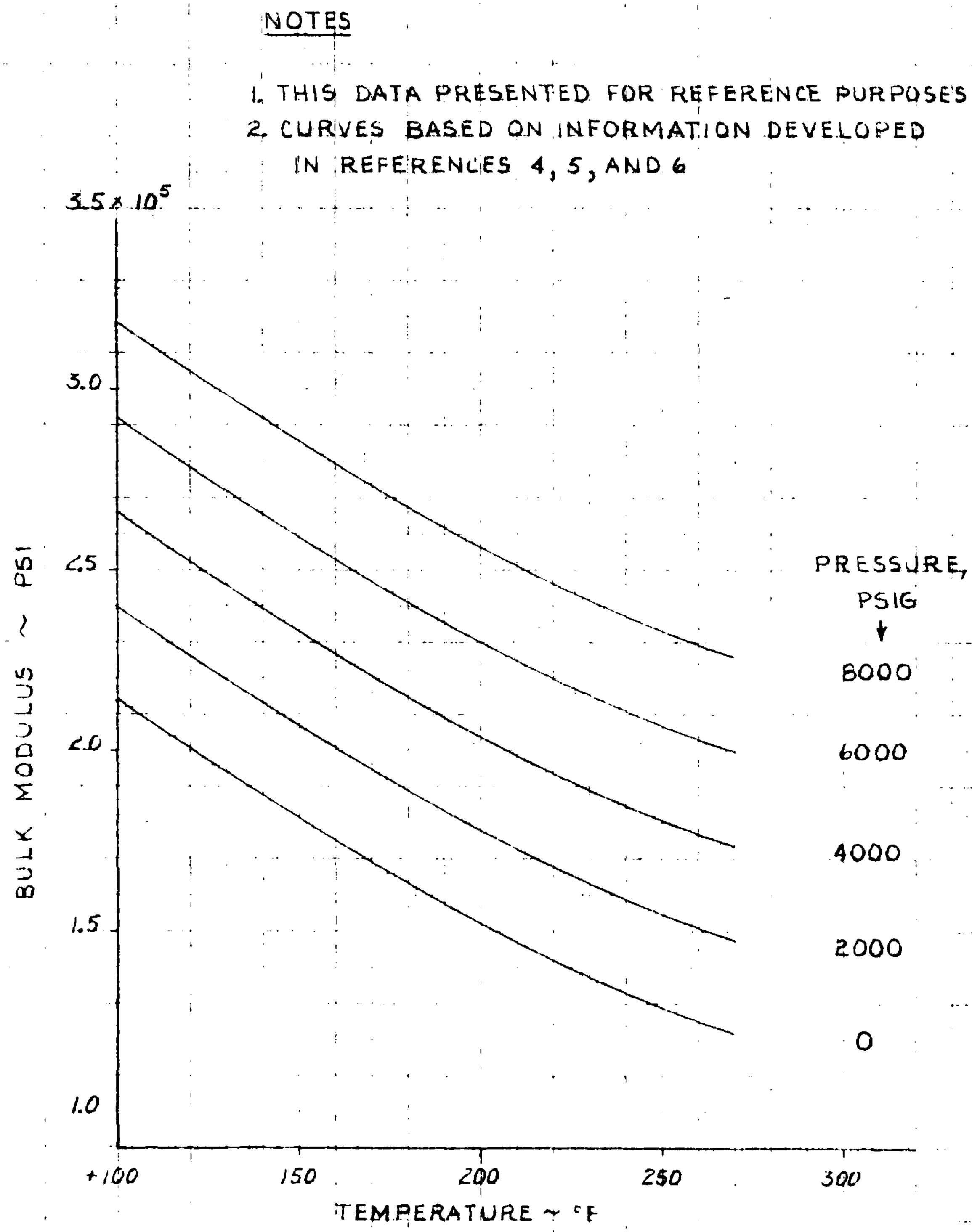

FIGURE A-4. ADIABATIC TANGENT BULK MODULUS OF MIL-H-83282

APPENDIX B

PRESSURE DROP COMPARISONS

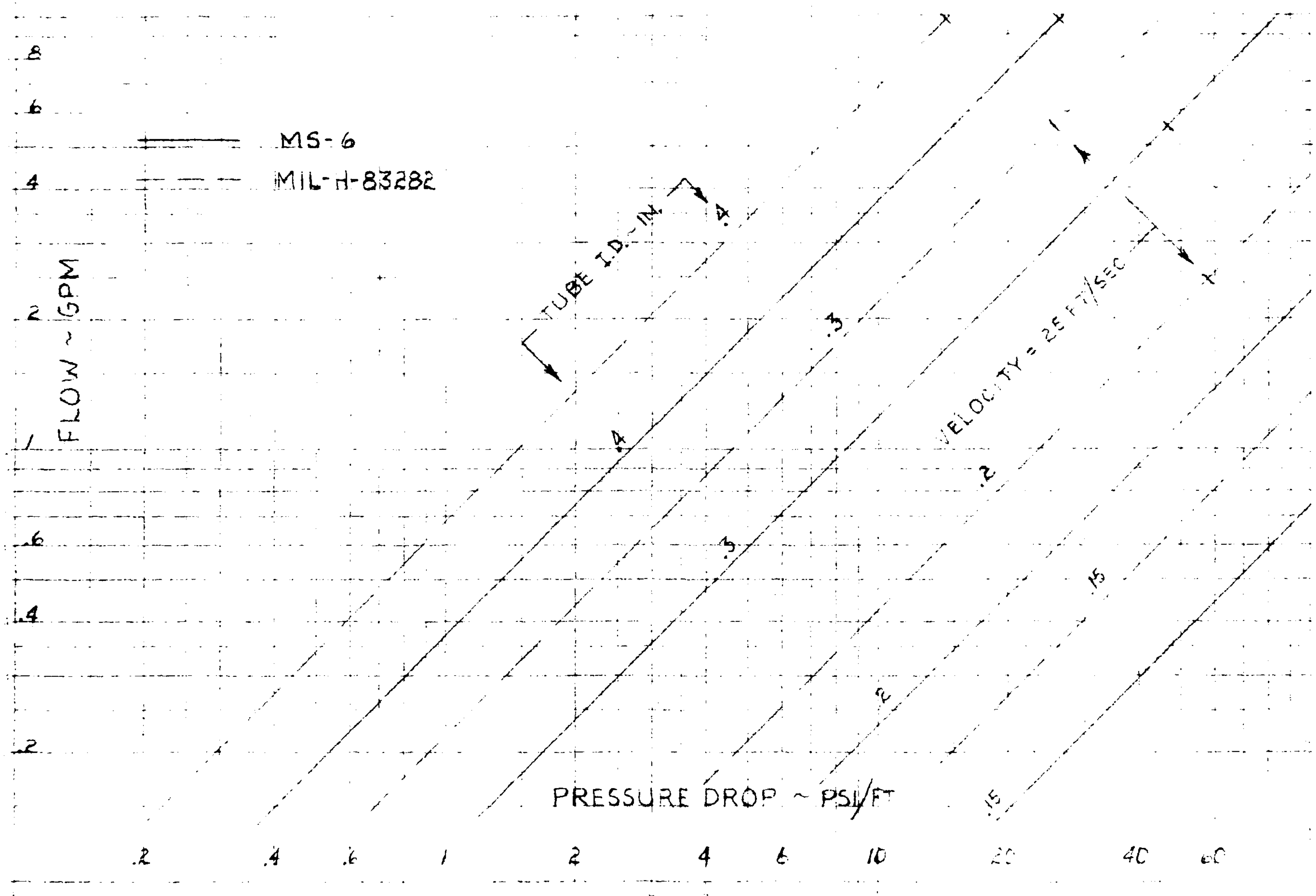

FIGURE B-1. PRESSURE DROP COMPARISONS, 100 PSI, +20°F

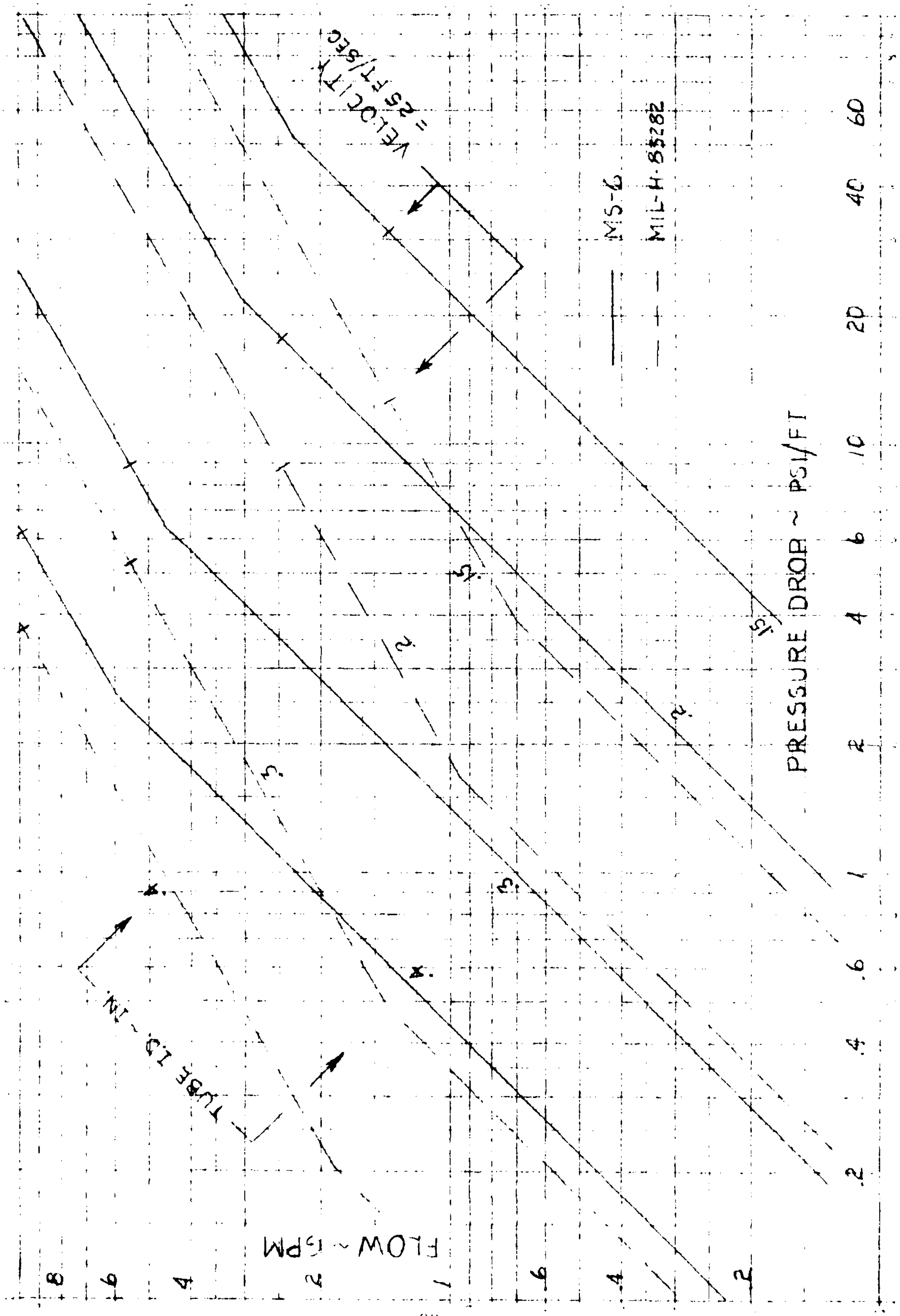

FIGURE B-2. PRESSURE DROP COMPARISONS, 100 PSI, +120°F

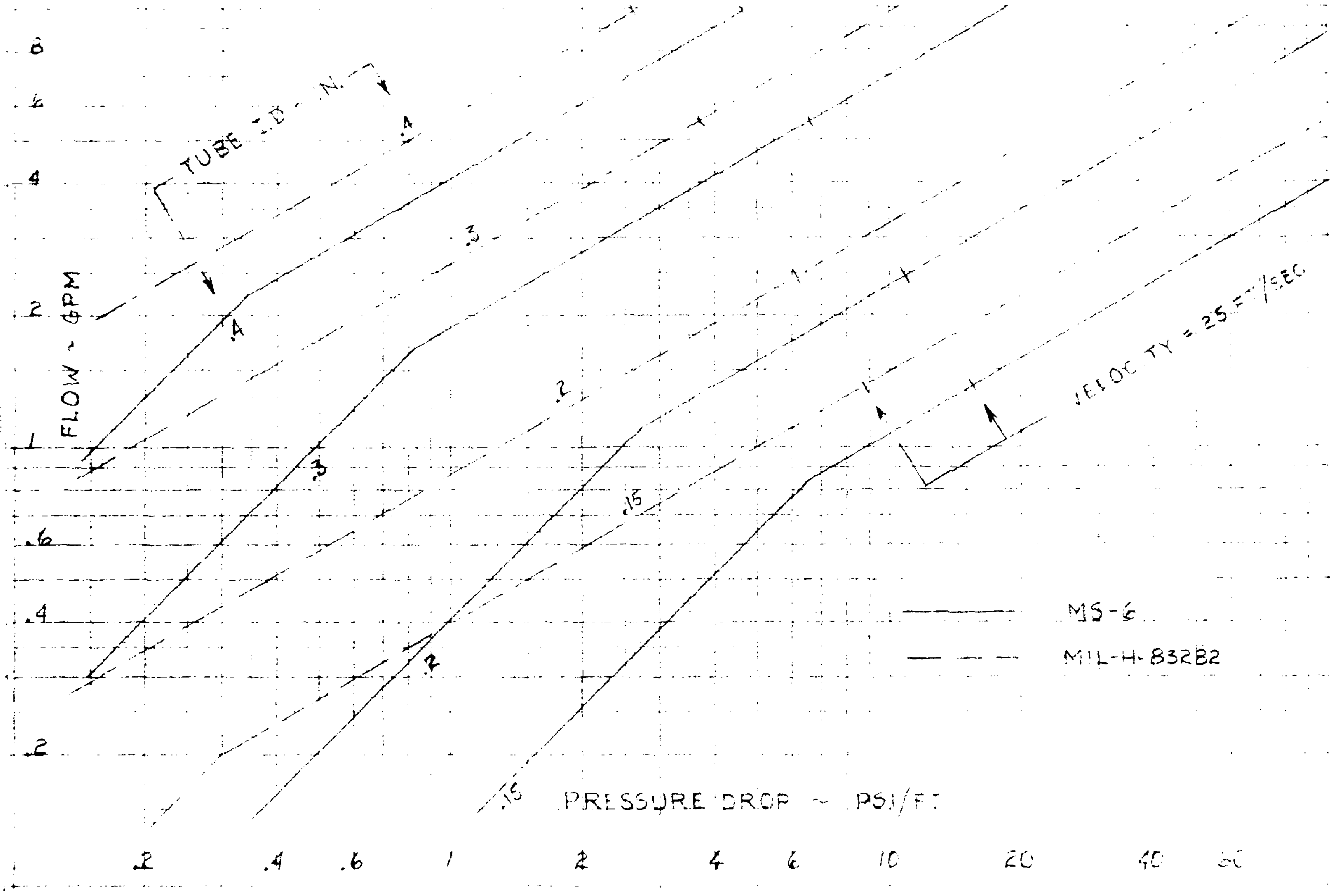

FIGURE B-3. PRESSURE DROP COMPARISONS, ... 220 F

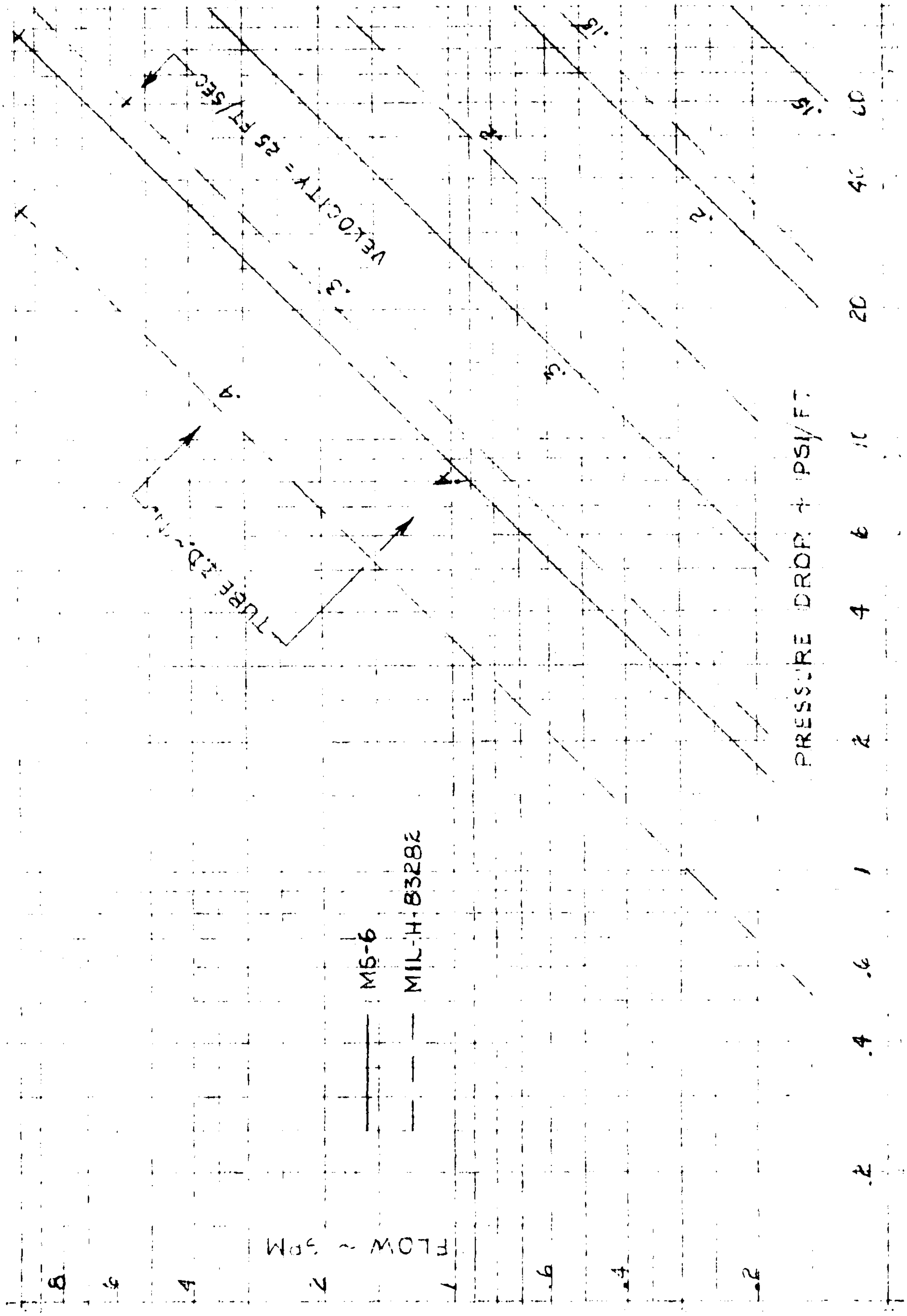

FIGURE B-4. PRESSURE DROP COMPARISONS, 8000 PSI, -20°F

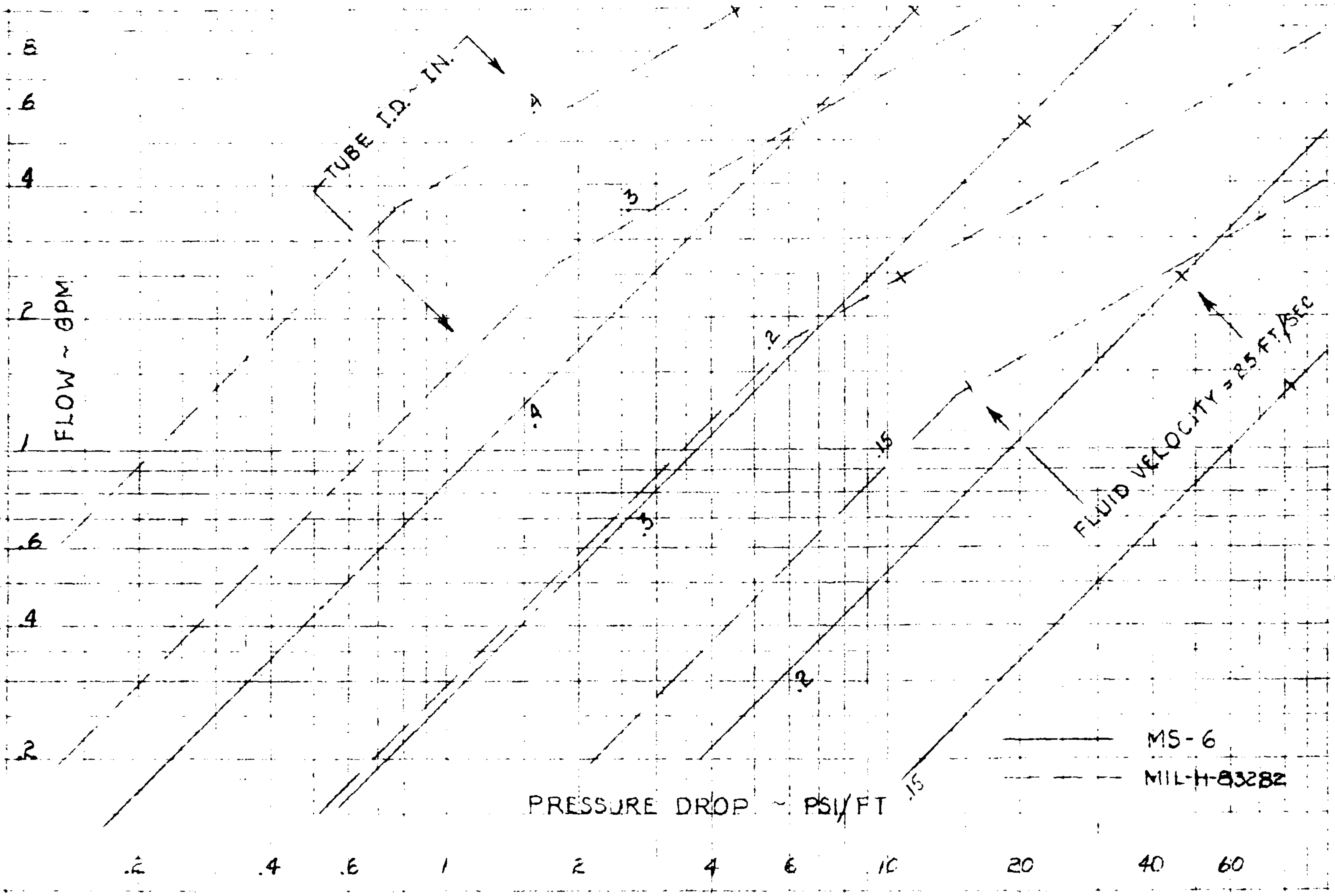

FIGURE B-5. PRESSURE DROP COMPARISONS, 8000 PSI, + 20°F

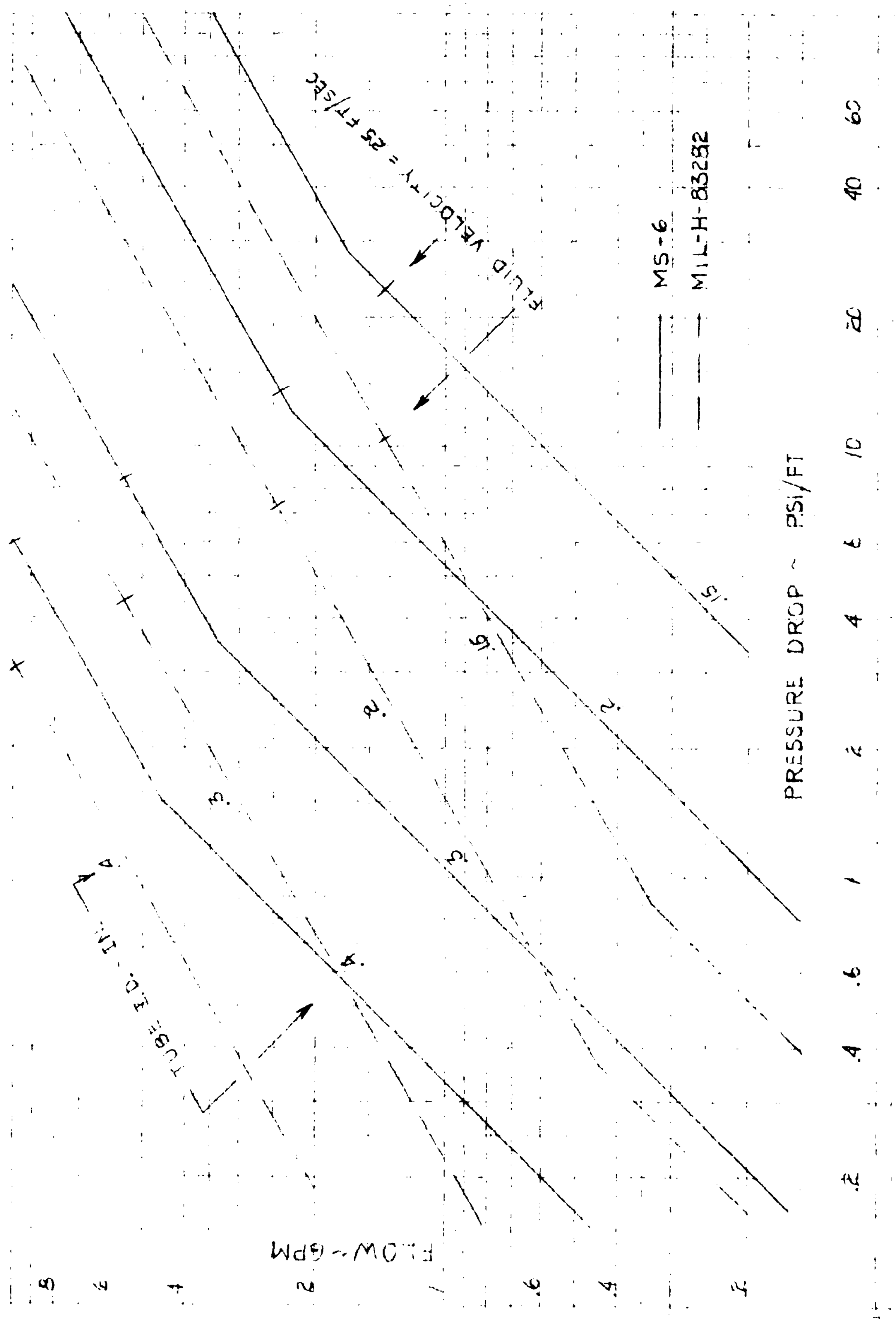

FIGURE B-6. PRESSURE DROP COMPARISONS, 8000 PSI, 220°F

APPENDIX C

UTILITY SYSTEM WEIGHT AND VOLUME STUDY

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TABLE C-1. Air Refueling Probe Subsystem

Component	Modification for MS-6	Effect on Subsystem Wt./Vol.
Actuator	None	None
Selector Valve	None	None
Restrictors	Re-size orifices	None
Check Valve	None	None
Tubing/Fittings	None	None
Impact on utility system weight and volume:		None

TABLE C-2. Gun Drive Subsystem

Component	Modification for MS-6	Effect on Subsystem Wt./Vol.
Motor	Minor internal changes	None
Control Valve	Re-size orifices, increase port size	Negligible increase
Relief Valve	None	None
Loading Valve	None	None
Dual Rate Valve	Re-size orifices, increase port size	Negligible increase
Filter	Minor internal changes	None
Subsystem Tubing/Fittings	Increase tube diameters	Significant increase
Power Circuit Tubing/Fittings	Increase tube diameters	Significant increase
Impact on utility system weight and volume:		Sizable increase

TABLE C-3. Gun Drive Vent Subsystem

Component	Modification for MS-6	Effect on Subsystem Wt./Vol.
Actuator	None	None
Control Valve	None	None
Restrictor	Re-size orifice	None
Hose	None	None
Tubing/Fittings	None	None
Impact on utility system weight and volume:		None

TABLE C-4. Trailing Edge Flap Subsystem

Component	Modification for MS-6	Effect on Subsystem on Wt./Vol.
Flap Actuators	None	None
Control Valve	None	None
Flow Equalizer	Re-size orifices	None
Locking Valve	None	None
Relief Valves	None	None
Bypass Valve	None	None
Shuttle Valve	None	None
Restrictors	Re-size orifices	None
Swivels	None	None
Tubing/Fittings	None	None
Impact on utility system weight and volume:		None

TABLE C-5. Wing Fold Subsystem

Component	Modification for MS-6	Effect on Subsystem Wt./Vol.
Wing Fold Actuators	None	None
Lock Pin Cylinders	None	None
Sequence Valve	None	None
Selector Valve	None	None
Restrictors	Re-size orifices	None
Thermal Relief Valve	None	None
Tubing/Fittings	None	None
Impact on utility system weight and volume:		None

TABLE C-6. Nose Gear Steering Subsystem

Component	Modification for MS-6	Effect on Subsystem Wt./Vol.
Actuator	Possible diameter increase to maintain actuator stiffness	Small increase
Selector Valve	Increase porting size	Negligible increase
Bypass Valve	None	None
Damper Orifice	Re-size orifice	None
Relief Valve	None	None
Servo Valve	Re-size orifices	None
Line Extension Units	None	None
Swivels	None	None
Tubing/Fittings	Increase tube diameters	Significant increase
Impact on utility system weight and volume:		Small increase

TABLE C-7. Wheel Brake Subsystem

Component	Modification for MS-6	Effect on Subsystem Wt./Vol.
Wheel Brakes	None	None
Power Brake Valve	None	None
Anti-Skid Valve	Re-size orifices	None
Shut-off Valve	None	None
Check Valve	None	None
Emergency Brake Valve	None	None
Swivel	None	None
Shuttle Valve	None	None
Tubing/Fittings	None	None
Impact on utility system weight and volume:		None

TABLE C-8. Landing Gear Subsystem

Component	Modification for MS-6	Effect on Subsystem Wt./Vol.
Gear Actuators	None	None
Door Actuators	None	None
Selector Valve	Increase porting size	Small increase
Emergency Selector Valve	Increase porting size	Small increase
Restrictors	Re-size orifices and porting	Negligible increase
Shuttle Valve	Re-size porting	Negligible increase
Swivels	None	None
Tubing/Fittings	Increase tube diameters on approx. 50% of lines	Significant increase
Impact on utility system weight and volume:		Small increase

TABLE C-9. Arresting Hook Subsystem

Component	Modification for MS-6	Effect on Subsystem Wt./Vol.
Actuator	Change damper spring rate, Re-size orifice	None
Overboard Dump Valve	None	None
Selector Valve	None	None
Restrictor	Re-size orifice	None
Check Valves	None	None
Tubing/Fittings	None	None
Impact on utility system weight and volume:		None

TABLE C-10. Launch Bar Subsystem

Component	Modification for MS-6	Effect on Subsystem Wt./Vol.
Actuator	Re-size orifice	None
Control Valve	None	None
Restrictors	Re-size orifices	None
Swivels	Re-size orifices	None
Tubing/Fittings	None	None
Impact on utility system weight and volume:		None

TABLE C-11. EPP Control Subsystem

Component	Modification for MS-6	Effect on Subsystem Wt./Vol.
Actuator	Increase port size	Negligible increase
Restrictor	Re-size orifice, increase port size	Negligible increase
Selector Valve	Increase port size	Negligible increase
Shut-off Valve	Increase port size	Negligible increase
Check Valve	Increase valve size	Negligible increase
Swivels	Increase port size	Negligible increase
Tubing/Fittings	Increase line size	Sizable increase
Impact on utility system weight and volume:		Negligible increase

TABLE C-12. Emergency Accumulator Subsystem

Component	Modification for MS-6	Effect on Subsystem Wt./Vol.
Accumulators*	None	None
Dump Valves*	None	None
Thermal Relief Valves*	None	None
Gas Charging Valves*	None	None
Impact on utility system weight and volume:		None

*one each in the following emergency circuits:
- Wheel Brakes
- Trailing Edge Flaps
- Landing Gear
- Emergency Power Package Control

TABLE C-13. Utility Power Circuit

Component	Modification for MS-6	Effect on Subsystem Wt./Vol.
Pump	Re-time, re-balance hanger, re-size control valve orifice, re-size leakage clearances, increase suction port size	Negligible increase
Reservoir	Increase capacity	Significant increase
Pump Hoses	Increase size of suction and pressure hoses	Small increase
Pump Disconnects	Increase size of suction and pressure disconnects	Small increase
External Disconnects	None	None
Filters	Increase by-pass relief valve and ΔP indicator settings	None
Relief Valve	Possible spring rate change	None
Overboard Relief Valve	Possible spring rate change	None
Check Valves	Increase port size	Negligible increase
Damper Accumulator	None	None
Pressure Transmitter and Switch	None	None
Tubing/Fittings	Increase tube size	Significant increase
Impact on utility system weight and volume:		Sizable increase

APPENDIX D

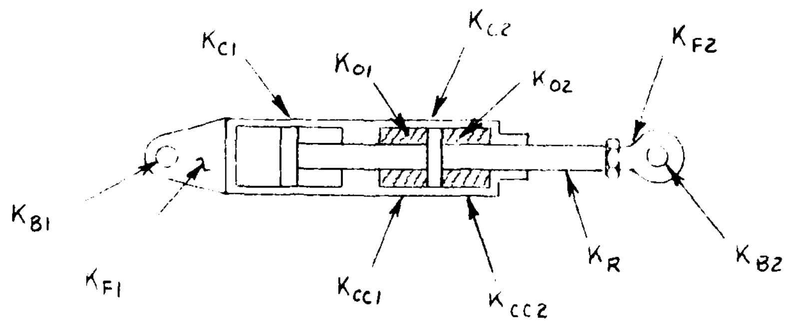
F-14 ACTUATOR STIFFNESS ANALYSIS

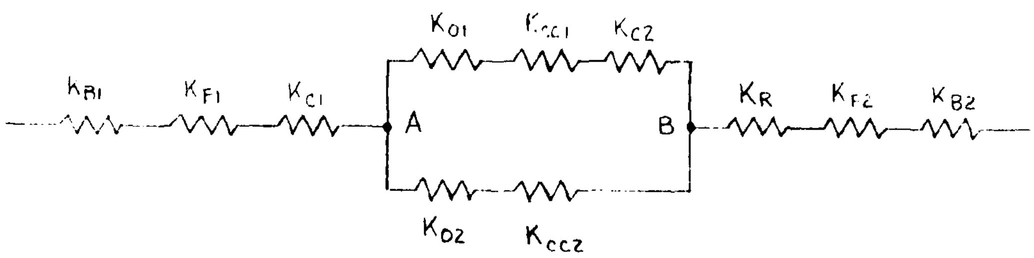
NOTE: See Reference 3 for baseline information.

F-14 HORIZONTAL STABILIZER ACTUATOR STIFFNESS CALCULATIONS

1. CRITICAL CONDITION: ONE SYSTEM OUT, PISTON CENTERED, NO OUTPUT LOAD

2. ACTUATOR SPRINGS:





3. PARAMETERS:

OPERATING PRESSURE	8000 PSI
FLUID	MS-6
FLUID ADIABATIC TANGENT BULK MODULUS @ 4000 PSI & +225°F (β_f)	126,000 PSI
REQUIRED ACTUATOR STIFFNESS	550,000 LB/IN
MODULUS OF ELASTICITY OF STEEL	29×10^6 PSI
POISSONS RATIO FOR STEEL (ν)	0.265

* [illegible]:

[illegible]	[illegible]	[illegible]
[illegible] DIA.	[illegible]	[illegible]
[illegible] NET AREA	$A_{[illegible]}$	[illegible] IN.
[illegible]		[illegible]
[illegible] DIA.		[illegible]
[illegible] AREA	$A_{[illegible]}$	[illegible]
[illegible] AREA	A_P	[illegible]
MAX. [illegible] FORCE [illegible]		[illegible]
STROKE	S	[illegible]
[illegible] LENGTH	L	[illegible]

* [illegible] ITERATION TO [illegible] ACTUA[illegible]

[illegible] ACTUATOR STIFFNESS CHECK:

K_{B1} = 23.5 × 10^6 LB/IN $\qquad K_{B2}$ = 17.5 × 10^6 LB/IN

K_{F1} = 33.8 × 10^6 LB/IN $\qquad K_{F2}$ = 50.0 × 10^6 LB/IN

SOURCE OF ABOVE VALUES: REFERENCE 5

$$K_{C1} = K_{C2} = \frac{A_C E}{L} = \frac{4.473 \times 29 \times 10^6}{9.8} = 13.24 \times 10^6 \text{ LB/IN}$$

$$K_{H1} = K_{H2} = \frac{2 A_P \beta_e}{S} = \frac{2 \times 14.395 \times 126{,}000}{8.75} = [illegible].415 \times 10^6 \text{ LB/IN}$$

$$K_{cc1} = K_{cc2} = \frac{A_P \beta_c}{L} = \frac{A_P E}{L}\left[\frac{T\,(D_o + D_i)}{(1+\nu)D_o^2 + (1-\nu)D_i^2}\right]$$

WHERE, β_c = COMPLIANCE OF CYLINDER

T = CYLINDER WALL THICKNESS

$$K_{cc1,2} = \frac{4.395 \times 29 \times 10^6}{9.8}\left[\frac{.253(5.881 + 5.375)}{(1+.285)\,5.881^2 + (1-.285)\,5.375^2}\right]$$

$$= 1.86 \times 10^6 \text{ LB/IN}$$

$$\frac{1}{K_x} = \frac{1}{K_{01}} + \frac{1}{K_{cc1}} + \frac{1}{K_{c2}} = \left[\frac{1}{.415} + \frac{1}{1.86} + \frac{1}{13.24}\right] \times \frac{1}{10^6}$$

$$K_x = .33 \times 10^6 \text{ LB/IN}$$

$$\frac{1}{K_y} = \frac{1}{K_{02}} + \frac{1}{K_{cc2}} = \left[\frac{1}{.415} + \frac{1}{1.86}\right] \times \frac{1}{10^6}$$

$$K_y = .34 \times 10^6 \text{ LB/IN}$$

$$K_{AB} = K_x + K_y = (.33 + .34) \times 10^6 = .67 \times 10^6 \text{ LB/IN}$$

$$K_R = \frac{A_R E}{L} = \frac{5.89 \times 29 \times 10^6}{11.6} = 14.73 \times 10^6 \text{ LB/IN}$$

$$\frac{1}{K_{ACT}} = \frac{1}{K_{B1}} + \frac{1}{K_{F1}} + \frac{1}{K_{C1}} + \frac{1}{K_{AB}} + \frac{1}{K_{R}} + \frac{1}{K_{F2}} + \frac{1}{K_{B2}}$$

$$\frac{1}{K_{ACT}} = \left[\frac{1}{20.5} + \frac{1}{33.8} + \frac{1}{13.24} + \frac{1}{.67} + \frac{1}{14.73} + \frac{1}{80} + \frac{1}{17.5}\right] \times \frac{1}{10^6}$$

$$K_{ACT} = 560{,}000 \text{ LB/IN}$$

APPENDIX E

TEST EQUIPMENT LIST

NOTE: Item numbers refer to $\boxed{1}$, $\boxed{2}$, etc. coding on Figures 2, 4, 5, 6, 11, 17, 18, and 20.

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TEST EQUIPMENT LIST

Item No.	Test Equipment Description
1	Reservoir, 5.7 gal., Rockwell International Corporation P/N 247-58149
2	Varidrive, 75 hp, U.S. Electric Motors, Type VEU-GSDT
3	Pump, variable delivery, 8000 psi, Abex M/N AP6V-57, S/N 109422
4	Turbine Flowmeter, Fischer & Porter M/N 10C1510A
5	Thermocouple, Rockwell International Corporation P/N - None
6	Pressure gage, dial type
7	Filter, 5μ, Aircraft Porous Media P/N AC-900-121
8	Heat exchanger, 275 psig, Whitlock Type MHIR-4-B-C1
9	Float Switch, Revere P/N F-83000-31
10	Torque Meter, 2000 lb-in, B&F Instruments M/N 2000 CB3
11	Hydraulic Cylinder, Hannifin P/N - unknown
12	Relief Valve, adjustable, Denison M/N RV061 103A
13	Solenoid Valve, 4-way, Bendix 1013695-4
14	Electronic Counter, Beckman/Berkeley M/N 7370
15	Pressure Transducer, Viatran M/N 122E-F76
16	Solenoid Valve, 3-way, Sterer P/N 15390-1
17	Oscilloscope, dual beam, Tektronix Type 502A
18	Oscilloscope Camera, Hewlett Packard M/N 196A

TEST EQUIPMENT LIST (Continued)

Item No.	Test Equipment Description
19	DC Power Supply, Trygon M/N SHR40-1.5
20	Trigger Delay, Rockwell International P/N - None
21	Signal Conditioner, Rockwell International P/N - None
22	Storage Battery, Piqua P/N 24-39
23	Temperature Indicator, Brown Instruments M/N 156x63P12
24	Electronic Counter, Erie Pacific M/N 720
25	Torque Indicator, B&F Instruments M/N 1480-11
26	Differential Pressure Gage, Barton M/N 227
27	Servo Actuator, Rockwell International P/N 4212-01
28	Servo Actuator, Rockwell International P/N 247-58715
29	Demodulator/Amplifier, Rockwell International P/N - None
30	Control Valve, Rockwell International P/N 4212-03-11
31	Induction Potentiometer, Collins P/N CGLO-LMT-11103
32	Function Generator, Hewlett Packard M/N 202A
33	Recorder, Gould/Brush Mark 220
34	Dial Indicator, Federal M/N C81S
35	Hydraulic power supply, 2.5 gpm at 1000 psi using MIL-H-5606 fluid, Rockwell International P/N - None
36	Electro-Hydraulic Servo Valve, Cadillac Gage M/N FC300-2-71
37	AC Power Supply, Darcy/Behlman M/N 161A

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