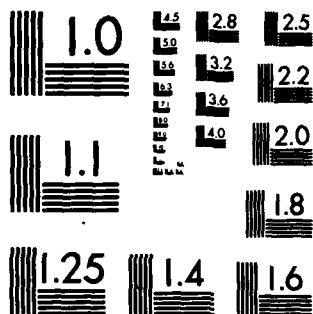




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**LOW TEMPERATURE EVALUATION OF
ADVANCED TECHNOLOGY HYDRAULIC SYSTEM (8,000 psi)**

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P.O. Box 225907
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AUGUST 1983

Technical Report NADC-82132-60
Final Report for Period September 1982 through April 1983

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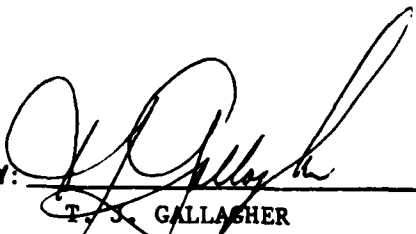
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➤ Frequency response and zero load rate tests at varying temperatures on the two circuits established the minimum temperatures at which performance requirements were met. Time versus temperature data from A-7D Category II Tests was used in heat transfer calculations to provide a comparison of warm up times for aircraft systems using 3000 or 8000 psi.

Results of all tests and analysis indicate that there is no essential difference in warm up time for systems operating at 3000 or 8000 psi.

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PREFACE

This final report was prepared by the Vought Corporation under Naval Air Development Center Contract N62269-82-C-0362.

The sponsoring agency was the Naval Air Systems Command, Washington, D.C. (Mr. Steve Hurst, AIR 310I, Program Manager). The work was administered under the technical direction of Mr. Douglas O. Bagwell, Naval Air Development Center, Warminster, Pennsylvania 18974. Vought's program manager was Mr. W. A. Poindexter and principal investigator was Robert Olsen.

Appreciation is expressed to Mr. G. K. Fling of Vought for support and consultation during the program and for technical review of the draft final report. Also, appreciation is extended to Rockwell International, Columbus Aircraft Division, Mr. W. Bickel, for use of the Lightweight Hydraulic System Aileron actuator for the 8000 psi tests in this program.

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LOW TEMPERATURE EVALUATION OF ADVANCED
TECHNOLOGY HYDRAULIC SYSTEM (8000 PSI)

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1. INTRODUCTION

The objective of this program was to determine if the use of 3/16 inch tubing with MIL-H-83282 fluid in an 8000 psi hydraulic system degrades system performance or results in unacceptable waiting time for system to warm to operating temperature. Dynamic performance can be affected due to increase in viscosity of hydraulic fluids at low temperature. The combinations of high flow demand, small tube diameters, and low temperatures present a unique set of problems which are solvable given adequate information and test data. Some performance characteristics such as line and orifice pressure drop at steady state conditions may be analyzed using fairly simple math models. Dynamic performance of a flight control actuator is much harder to predict. A test of the actuator with proper instrumentation under specified conditions is extremely valuable to verify analytical predictions.

System performance was measured in terms of the fluid temperature at which a servo actuator met minimum frequency response and zero load rate criteria. The acceptability of 8000 psi system performance was based upon an equivalent 3000 psi system. The time required from system start up until oil temperature allows the system to meet minimum frequency response requirements is important because it affects weapon system availability and response time once put on an alert status.

For a hydraulic servo system, the actuator output position at any point in time should correspond to the position commanded. When a uniform cyclic input is fed into a servoactuator, the output follows the input very closely at low frequencies. But at high frequency, the output does not follow the input as accurately. The output signal begins to rise later than the input and reaches a maximum sometime after the input does. This error between input and output is a measure of performance and is expressed by the ratio of the output magnitude to the input magnitude. This ratio is called the amplitude ratio.

Another measure of performance from the same comparison of input signal to output signal is the angle between the input curve and the output curve which is called the phase angle. When the output begins to follow the input at high frequencies, the output is said to lag the input. The frequency at which the output lags the input by 90 degrees is usually specified as a critical performance criterion.

A third measure of performance is zero load surface rate which determines line size and orifice area for a servoactuator. The surface rate in degrees per second can be converted to the equivalent actuator piston speed in inches per second. If zero load velocity is reduced relative to design requirements at low temperatures, the frequency response will also be reduced.

2.0 DISCUSSION OF TEST PROGRAM

2.1 Program Plan

The program plan consisted of the following steps:

- o Select a test circuit from an aircraft which is representative and can be duplicated in a test.
- o Perform frequency response and zero load rate tests on equivalent circuits at 3000 psi and at 8000 psi. Determine the fluid temperature(s) at which minimum frequency response and rate criteria are met for each system.
- o Using published data for low temperature environmental tests of aircraft, estimate the length of time for the 8000 psi system to meet minimum frequency response and zero load rate and compare against the time for a 3000 psi system.

2.2 Test Circuits

The aircraft circuit used in tests was the wing plumbing for the A-7 aircraft aileron actuator which is installed as shown on Figure 1. This circuit was selected because of relative high flow rate, long line lengths, and small diameter tubing. In order to accommodate the system within a small volume which could be temperature controlled, the lengths for pressure and return lines for the two hydraulic systems supplying the actuator were averaged and the total number degrees of bends were tabulated.

A-7 Aircraft Aileron Actuator Plumbing Analysis - 3000 PSI System

System	Function	Length	Tube Dia. -- Inches				Total Deg
			1/4	Total Deg.	3/8	No. Bends	
PC1	Pressure	10.95	26	1273	8.07	8	348
PC2	Pressure	15.32	21	1400	9.01	8	398
PC1	Return	11.19	27	1228	7.96	8	329
PC2	Return	16.31	22	1528	8.99	8	390
		53.77	96	5429	34.03	32	1465

AVG LENGTH 1/4 TUBE = $53.77/4 = 13.4$ feet

AVG DEG 1/4 DIA TUBE = $5429/4 = 1357$ degrees

AVG LENGTH 3/8 DIA TUBE = $34.03/4 = 8.5$ feet

AVG DEGREES 3/8 DIA TUBE = $1465/4 = 366$ degrees

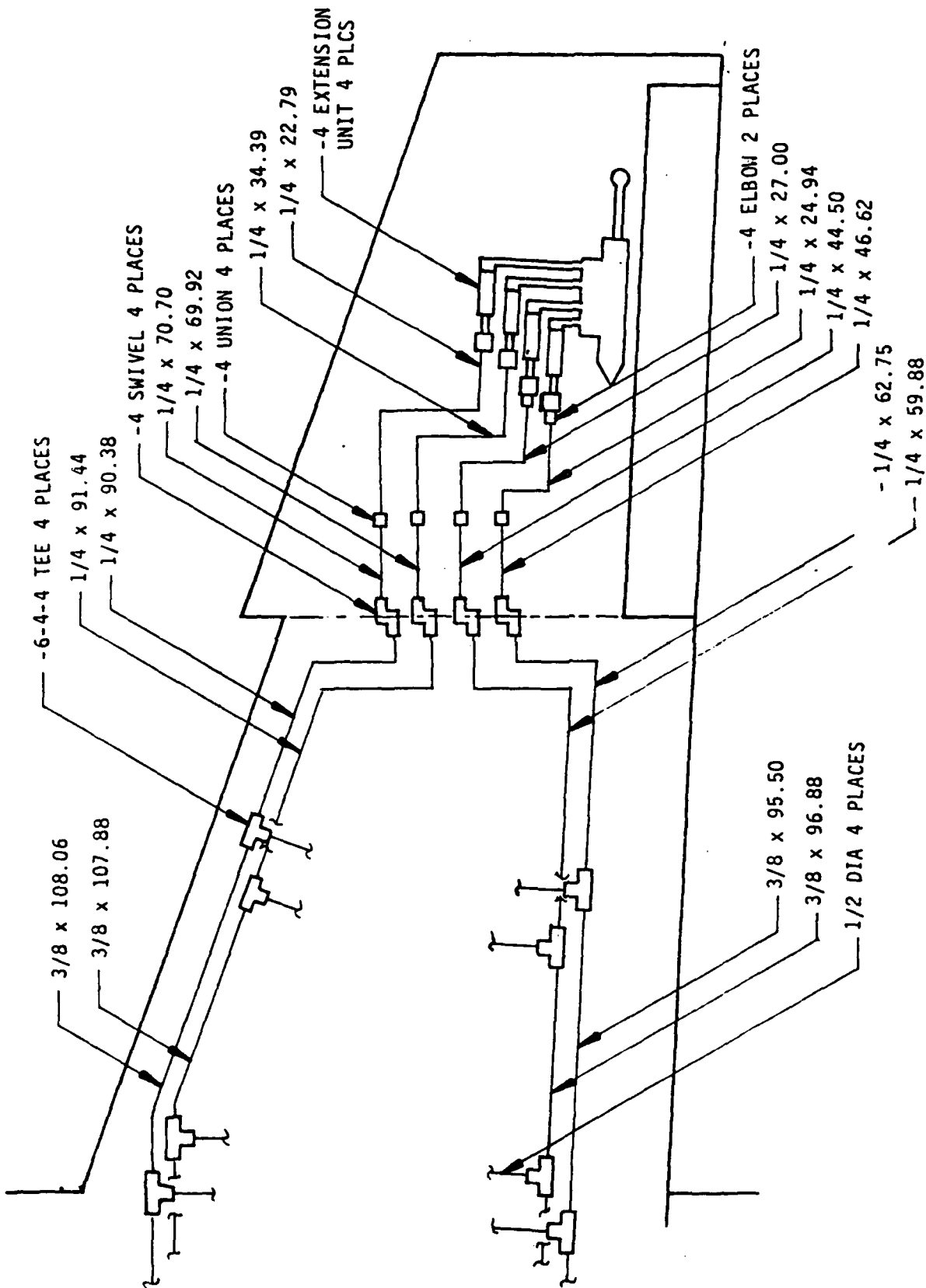


FIGURE 1. A-7 AIRCRAFT AILERON CIRCUIT has long line lengths with high flow demand.

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The test circuit shown on Figure 2 was used to duplicate line lengths and bends as closely as possible. Test circuit tube diameters, line lengths, and total degrees of bends are tabulated below.

Pressure-psig	System	Application	3/16		TUBE DIA 1/4		INCHES 3/8	
			FT LGTH	BEND DEG	FT LGTH	BEND DEG	FT LGTH	BEND DEG
3000	PC1	Pressure	--	--	13.4	1440	8.5	360
8000	PC1	Pressure	13.4	1440	8.5	360	--	--
3000	PC2	Pressure	--	--	13.4	1440	8.5	360
8000	PC2	Pressure	13.4	1440	8.5	360	--	--
3000	PC1	Return	--	--	13.4	1440	8.5	360
8000	PC1	Return	13.4	1440	8.5	360	--	--
3000	PC2	Return	--	--	--	13.4	1440	8.5
360	8000	Return	13.4	1440	8.5	360	--	--
--								

To further insure a fair comparison of 3/16 and 1/4 diameter tubing affect on performance, coil tube assemblies for the 3000 psi and the 8000 psi test circuit were designed in 1/4 inch and 3/16 inch tubing respectively to accept the motion of the actuator housing with respect to the test fixture. The test installation is shown in Figure 3.

2.3 Test Actuators

The actuator used for the 3000 psig circuit was a production aileron actuator (PN 215-82031) from the A-7 aircraft. The actuator was designed and fabricated by Vought Corporation. The actuator used for the 8000 psi circuit was an aileron actuator (PN 83-00221) from the Lightweight Hydraulic Systems Program Ref [1]. The 8000 psi actuator was also designed and fabricated by Vought. The two actuators were designed for the same performance requirements with the exception of pressure. Output thrust was to be the same, except in order to use standard seals, thrust of the 8000 psi actuator was 33% higher in the extend direction. Servovalve spool diameters were the same. Servovalve orifice area for each actuator was designed for a zero load rate of 10 inches/second. The input linkage gain was the same. The installed length and stroke were the same. The pressure and return port locations were the same. MIL-H-83282 fluid was used for both circuits. Some pieces of hardware were common to both actuators. Figures 4 and 5 are cross section assembly drawings of the two actuators used.

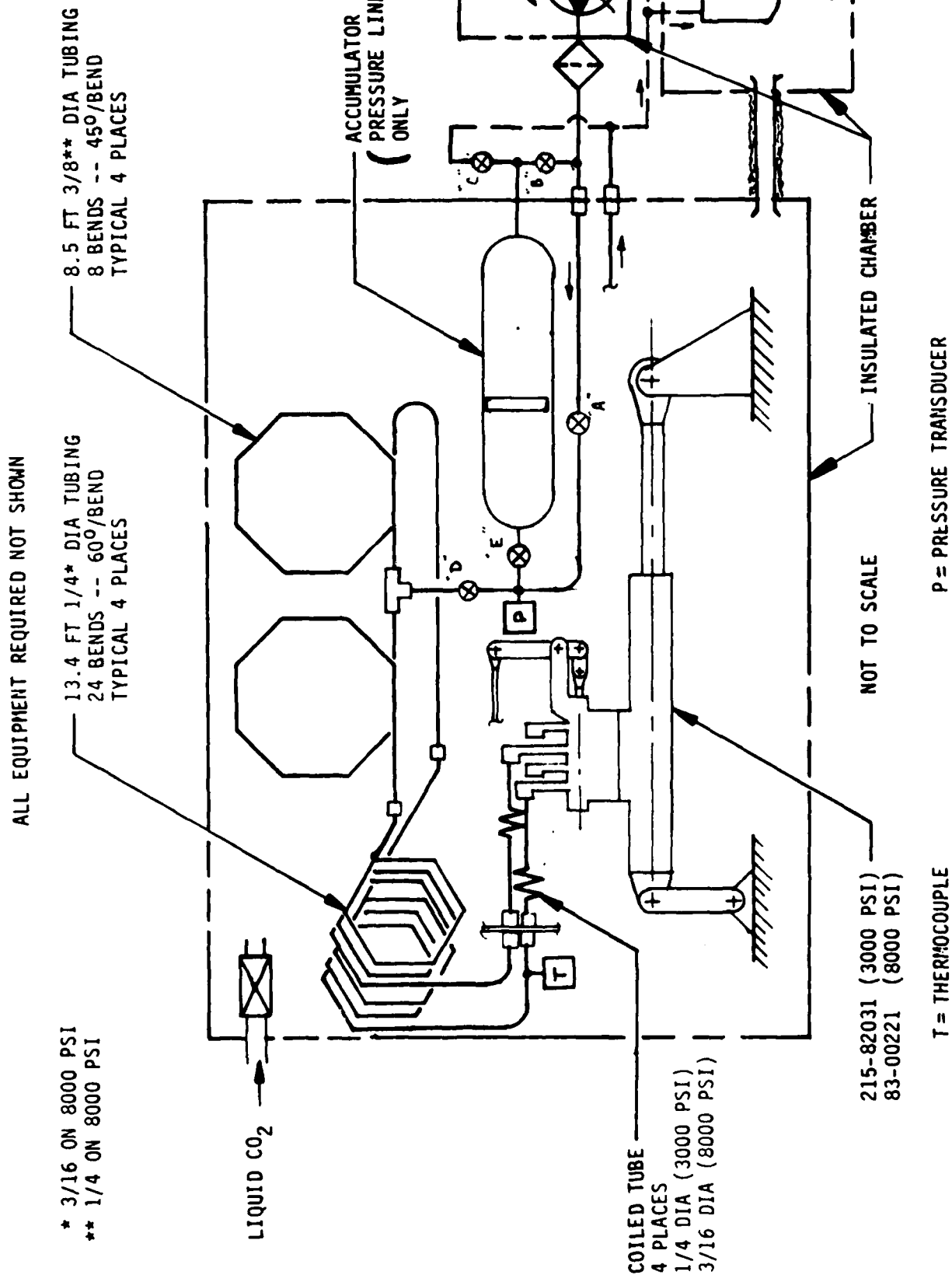


FIGURE 2. LOW TEMPERATURE TEST CIRCUIT duplicates line lengths and bends in a small volume.

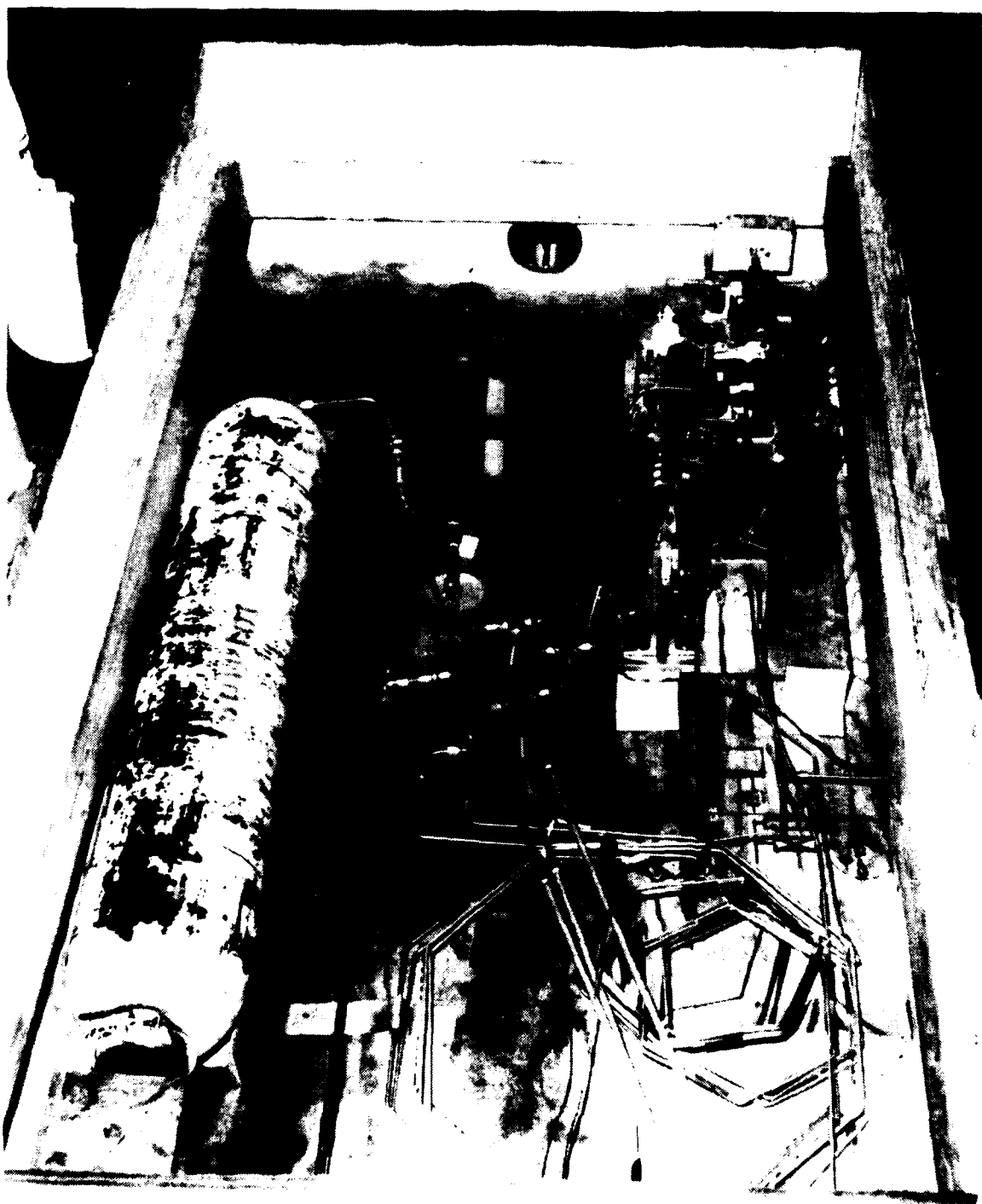


FIGURE 3. PHOTOGRAPH OF TEST INSTALLATION shows test actuator with coiled tubing for flexible plumbing (upper right) 10 gallon accumulator (left), and simulated wing tubing (lower right).

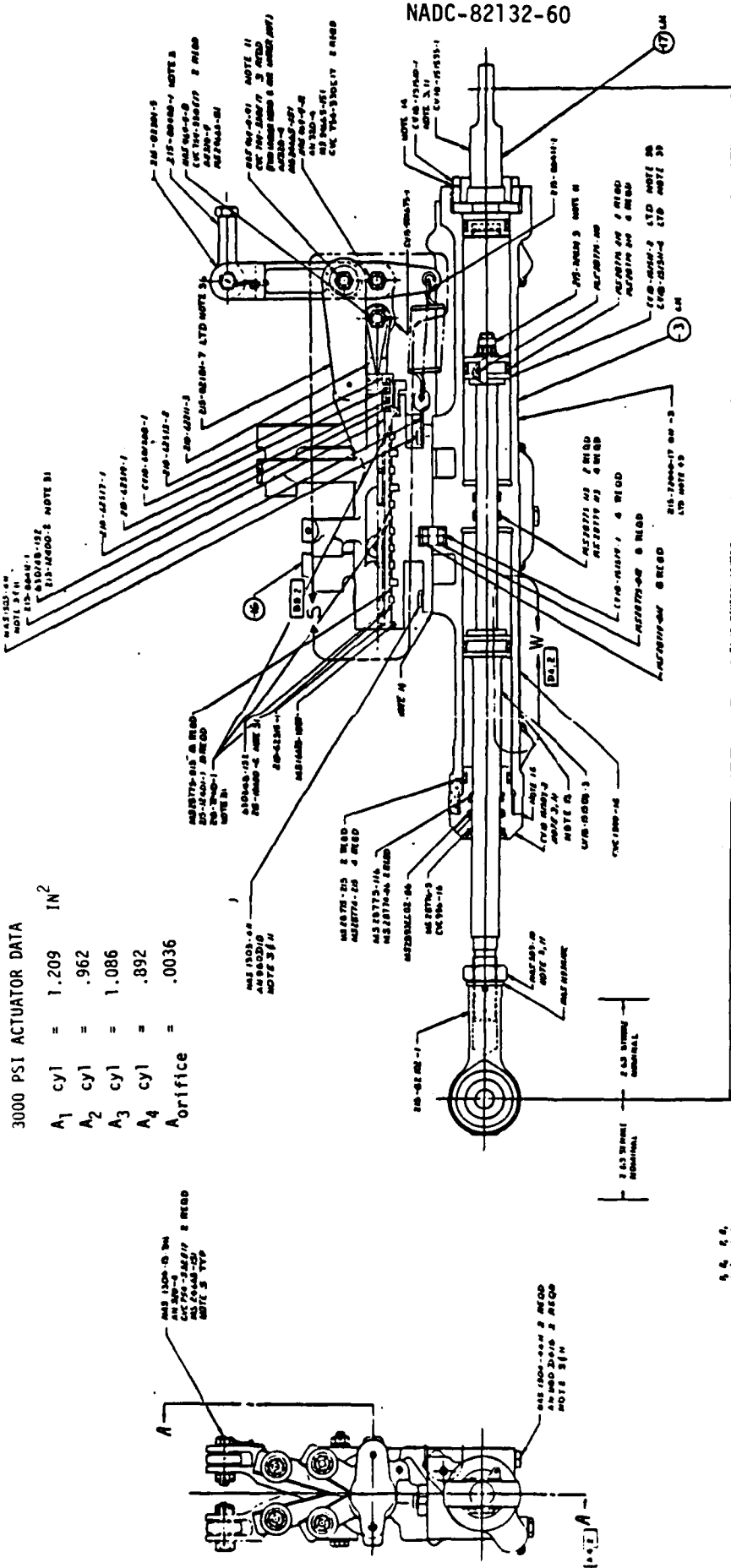


FIGURE 4. A-7 AIRCRAFT 3000 PSI AILERON ACTUATOR has $\pm .06$ inch mechanical input, 5.3 inch total stroke, and 1/4 inch ports.

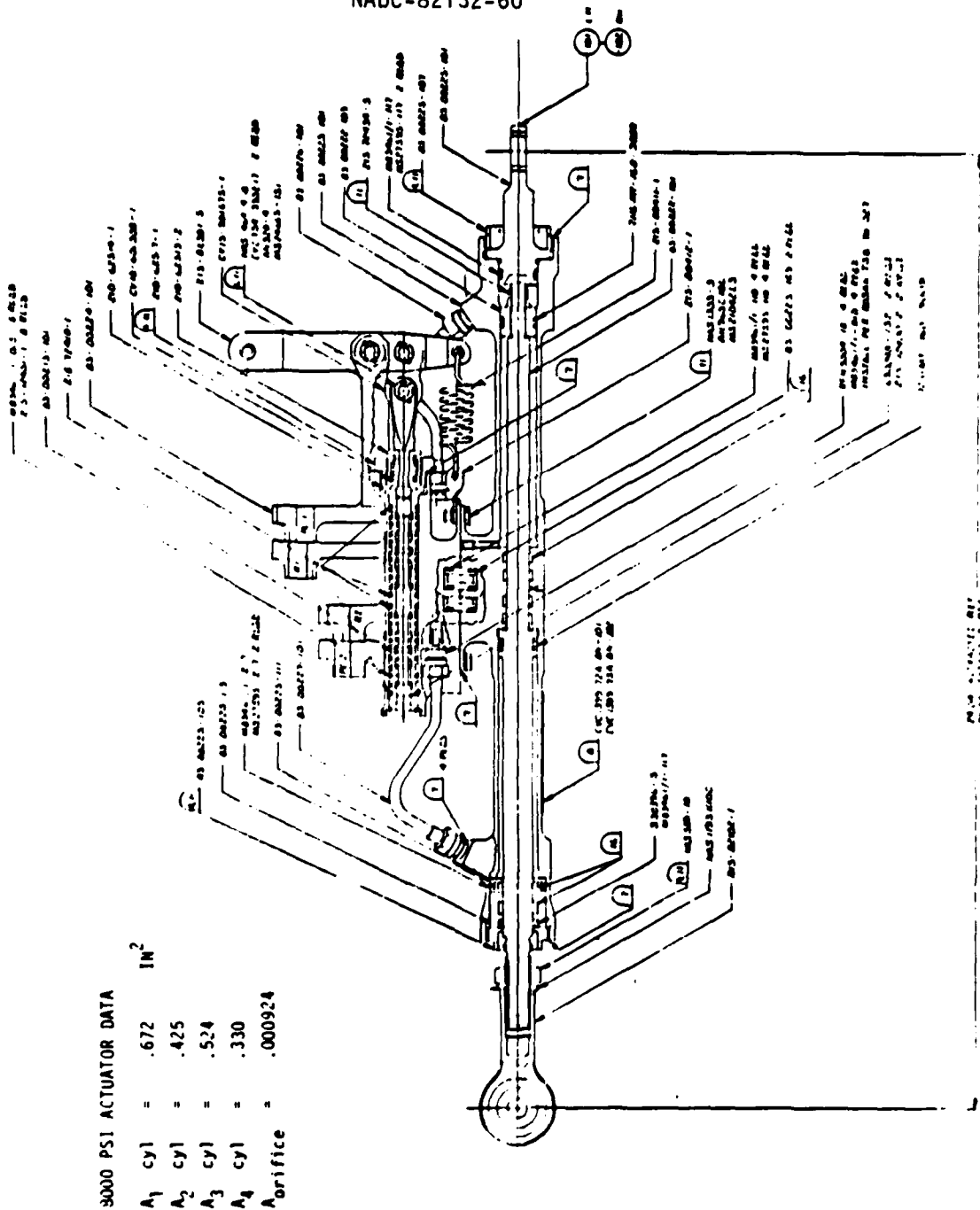


FIGURE 5. A-7 AIRCRAFT 8000 PSI AILERON ACTUATOR has +/- .06 inch mechanical input, 5.3 inch total stroke, and 3/16 inch ports.

2.4 Results of Performance Tests

The frequency response results were measured against a performance criteria of -3 db amplitude ratio (AR) and -45 degree phase angle ($\emptyset$) at 8.4 Hz. The value of 8.4 Hz was selected after calculating the frequency at which each actuator would have -3 db AR and -45 degree $\emptyset$ using known orifice areas, cylinder areas, and idler gain. These calculations assumed full supply pressure to the valve and gave 8.4 Hz for the 8000 psi actuator and 9.94 Hz for the 3000 psi actuator. The value of 8.4 Hz was selected because it would result in the most conservative comparison of the two systems.

2.4.1 Frequency Response Results - The test plan called for frequency response tests to be made on each test circuit at fluid temperatures of -40, -20, 0, +20, +40, +80, and +120 degrees fahrenheit. Due to the uncertainty of cooling a 10 gallon reservoir of oil to the desired test temperature, actual test temperatures varied from the ideal values. Therefore, all test data plotted shows the actual fluid temperature. Amplitude ratio and phase angle were plotted continuously against frequency at each temperature. This data was surveyed and the following data was obtained.

Frequency at Amplitude Ratio of -3db versus Temperature

Frequency at Phase Angle of -45° versus Temperature

Figures 6 and 7 show plots of the data for Amplitude ratio and phase angle respectively. In order to determine the temperature at which each system met the minimum criteria of 8.4 Hz at -3db or at -45 degrees, various curve fits were attempted on each set of points until a reasonably good fit was obtained. The temperatures at which minimum frequency response criteria were met were -18 and +14 degrees F respectively for amplitude ratio of -3db at 8.4 Hz for the 3000 and 8000 psi systems (Figure 6). The temperatures at which -45 degrees phase angle occurred at -8.4 Hz were -20 and +23 degrees F, respectively for the 3000 and 8000 psi systems. The 8000 psi data points above 50 degrees were ignored because they are assumed to be the result of an instrumentation failure.

2.4.2 Zero Load Rate Test Results - As with the frequency response tests, the test plan called for zero load rate tests to be conducted at fluid temperatures of -40, -20, 0, 20, 40, and 120°F. Because of variation in actual test temperature, all data plotted shows the actual fluid temperature for each data point.

In order to determine the temperatures at which each system met a zero load rate of 10 inches/second, the rate versus temperature data for each system was plotted using various curve fits. The best fit was obtained using a fourth degree polynomial curve for each set of data. Figure 8 shows the plotted rate versus temperature data. The 3000 psi circuit gave 10 inches/second at 34°F. The 8000 psi circuit gave 10 inches/second at 45°F. Table 1 summarizes the results of test data.

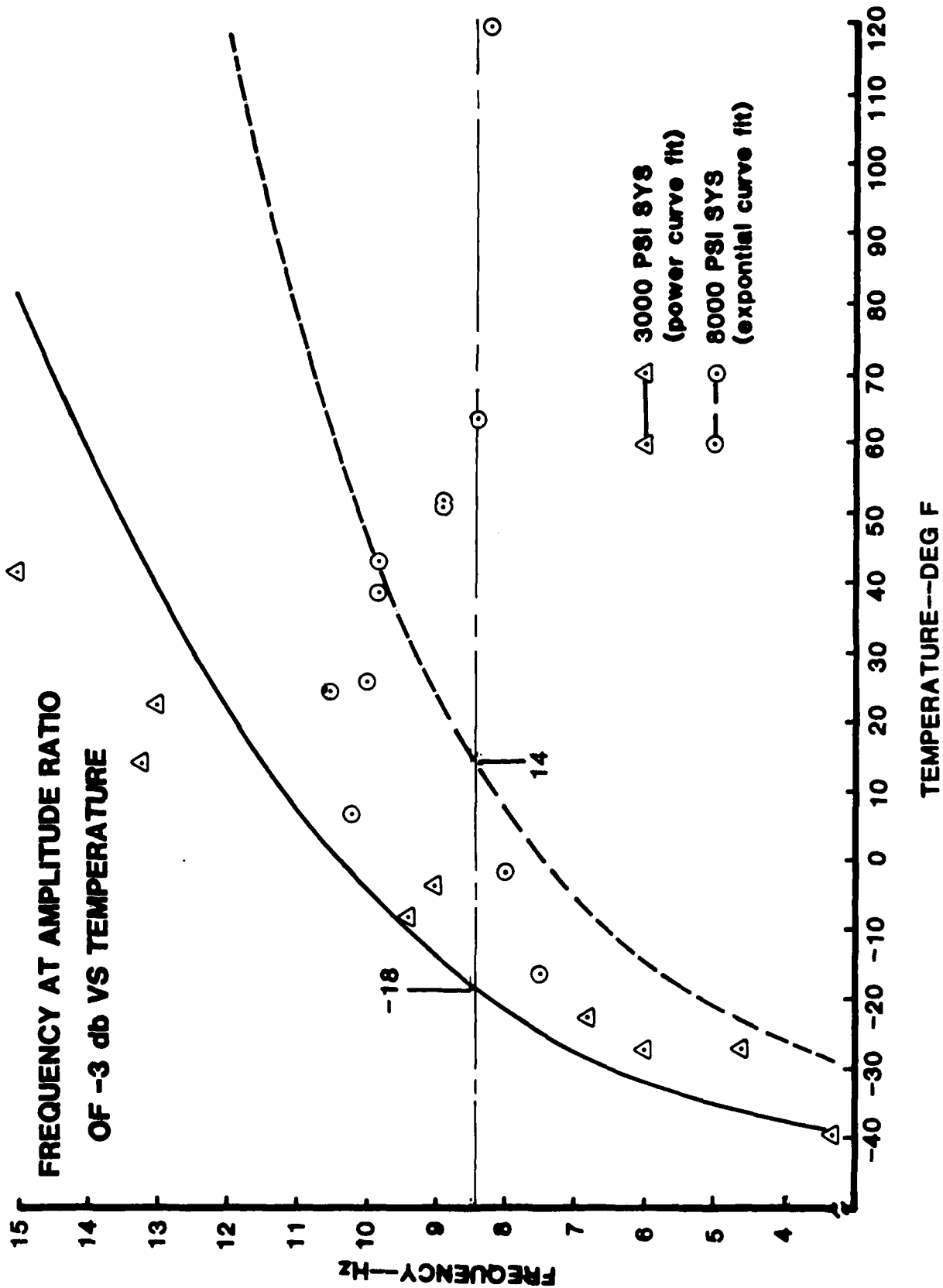


FIGURE 6. FREQUENCY AT AMPLITUDE RATIO OF -3db VERSUS TEMPERATURE establishes temperature at which each system has -3db amplitude ratio at 8.4 Hz.

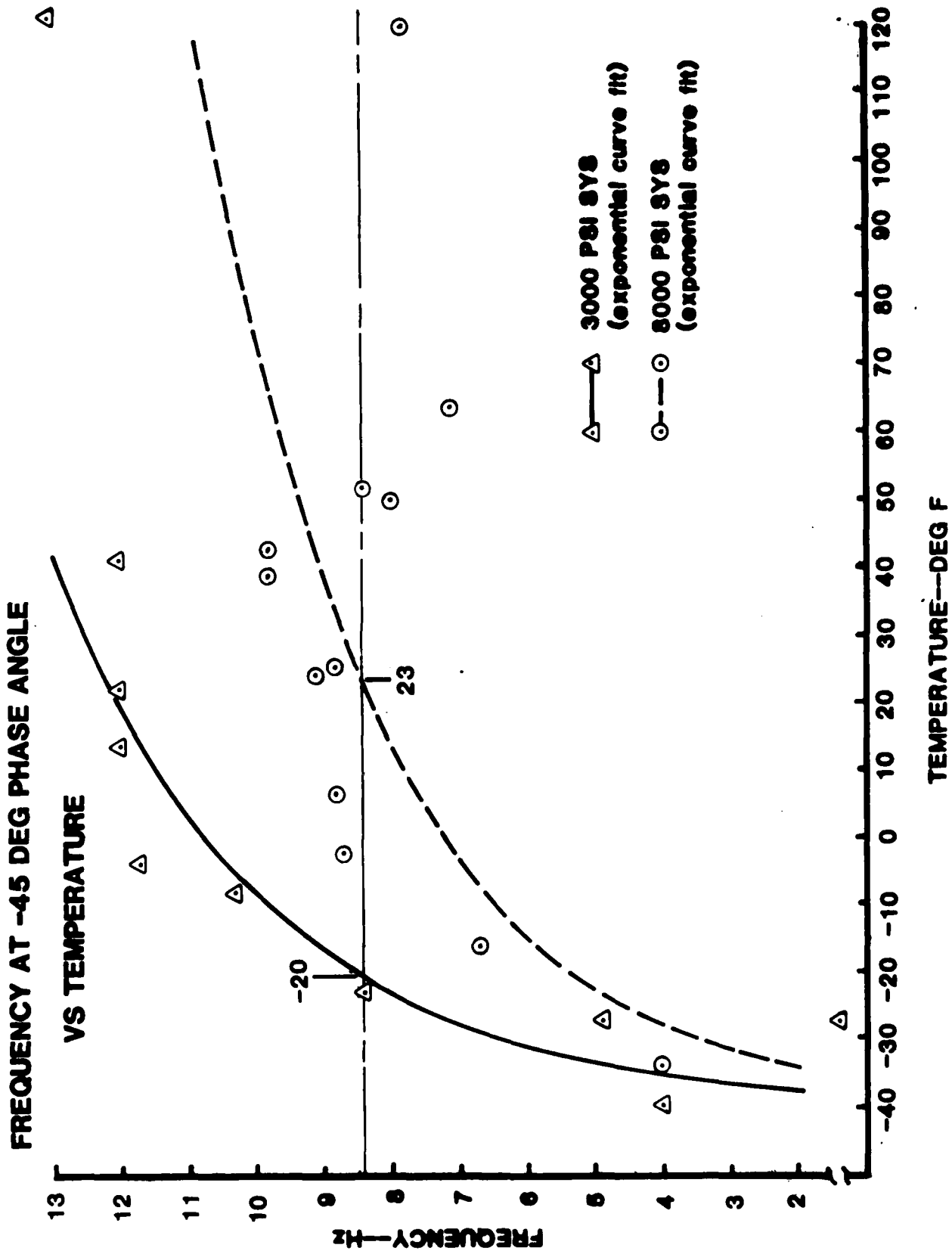


FIGURE 7. FREQUENCY AT PHASE ANGLE OF -45 DEG VERSUS TEMPERATURE establishes temperature at which each system has -45 deg phase angle at 8.4 Hz.

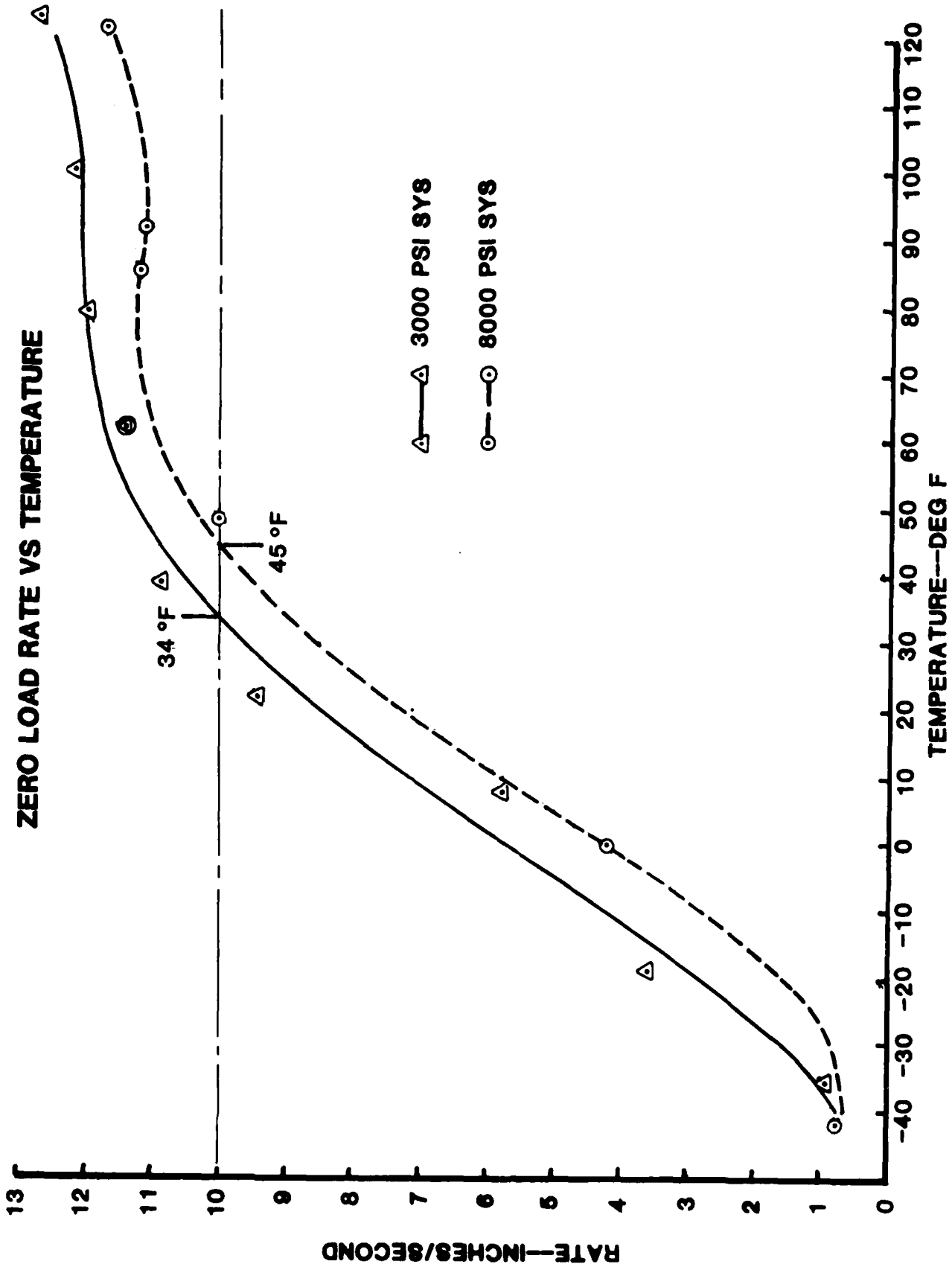


FIGURE 8. ZERO LOAD RATE VERSUS TEMPERATURE establishes temperature at which each system has 10 in/sec rate.

TABLE 1. TEST RESULTS SUMMARY

	TEMPERATURE - °F	
	<u>3000 PSI</u>	<u>8000 PSI</u>
<u>AMPLITUDE RATIO</u>		
-3db at 8.4 Hz	-18	14
<u>PHASE ANGLE</u>		
-45 deg at 8.4 Hz	-20	23
<u>ZERO LOAD RATE</u>		
10 in/sec	34	45

3.0 ESTIMATE OF 8000 PSI AIRCRAFT SYSTEM WARM UP TIME

3.1 Basis for Comparison

All data from tests was on a simulated portion of a total aircraft hydraulic system. The tests established the temperatures at which a typical circuit might reach acceptable performance. The problem remaining was to correlate data from the test circuit with the same circuit in an aircraft which has been cooled to the same low temperature. The aircraft data to be used for comparison is from reference [2] for an A-7D aircraft in environmental tests at Eglin AFB. The wing circuit plumbing in the A-7D is identical to the aileron circuit simulated in performance tests. The specific data used from reference [2] was Run No. 19, -40 degrees F, conducted 18 October 1969. Test conditions in reference [2] were:

- (1) Temperature transducers were installed at critical locations throughout the aircraft hydraulic system. This instrumentation included a temperature transducer to record PC2 pressure oil inlet temperature at the left hand aileron actuator.
- (2) The total aircraft was cold soaked at -45 degrees F for 52 hours.
- (3) The engine was started and time versus temperature data was recorded.
- (4) The flight controls were not cycled. The only flow throughout the hydraulic system during the time period of interest was quiescent case drain and servovalve neutral leakage.
- (5) Hydraulic fluid was MIL-H-5606.

Table 2 reproduces the data from reference [2] in part. The elapsed time from engine start and PC2 aileron inlet oil temperature is shown.

The plan for estimating the difference in warm up time for typical aircraft 3000 psi system and a typical aircraft 8000 psi system was as follows:

- (1) Determine a heat transfer model for the aileron circuit.
- (2) Compute the rate of heat into the 3000 psi aileron circuit.
- (3) Using the same heat transfer model, substitute surface areas, masses, and coefficients of specific heat for an 8000 psi aileron circuit to calculate oil temperature at aileron inlet versus time.
- (4) Rate of heat into the 8000 psi circuit is identical to the rate of heat into the 3000 psi circuit assuming the same horsepower losses in each system.

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TABLE 2. Data From A-7D Category II Tests

Soak Time: 52 Hours

Soak Temperature -45°F

This is a listing of data used with time zeroed to begin with engine start.

<u>Time-sec</u>	<u>Outside Air Temp - °F</u>	<u>PC2 Pump Out Temp - °F</u>	<u>LH Aileron Inlet Temp - °F</u>	<u>PC2 Pump Out Pressure PSI</u>
0	- 38.2	- 38	- 32	25
8	- 38.2	- 40	- 32	49
16	- 38.2	- 35	- 32	77
23	- 40.0	- 23	- 32	56
31	- 40.0	- 17	- 32	46
70	- 38.2	- 19	- 29	2196
133	- 40.0	54	- 31	2981
195	- 38.2	101	- 20	2964
258	- 38.2	127	3	2957
381	- 38.2	160	53	2946
433	- 38.2	170	60	2897

3.2 Heat Transfer Model for Aileron Circuit

In the time period of interest, heat transfer into the aileron circuit is part of the heat generated by the pump. This heat is carried into the aileron circuit by the oil flow in the pressure line due to servovalve neutral leakage. Figure 9 is a plot of the aileron inlet oil temperature versus time from reference [2]. This figure gives the time to reach critical temperatures in the 3000 psi circuit for the total aircraft. These temperatures all lie within the linear portion of the curve of Figure 9, therefore, the heat transfer model only duplicated the linear portion of the curve. The heat transfer calculation procedure given in reference [3] was adapted for use in this problem. Assumptions made were: (1) All heat transfer is by convection through walls at tubing, (2) flow rate of warm oil into circuit is constant, and (3) no heat is generated due to pressure drop in lines. The method is an iterative procedure with the following essential steps:

- (1) Calculate the heat input in BTU in a given constant time increment.
- (2) Calculate an estimate of aileron inlet temperature rise using the known heat input, mass of oil, mass of tubing, and respective specific heat coefficients.
- (3) Calculate the average temperature rise of the circuit based upon the initial temperature rise estimate.
- (4) Calculate the heat dissipated based upon the average temperature rise.
- (5) Calculate the net heat added.
- (6) Calculate a revised estimate of temperature rise at aileron inlet.
- (7) Calculate new circuit temperature.
- (8) Go to step (1).

For the 3000 psi system, a value for rate of heat input was established by trial and error for a linear time versus temperature curve which coincided with the linear portion of the actual time versus temperature data from reference [2]. The rate of heat input established by this procedure for the 3000 psi system was .660 BTU/sec into the aileron circuit. Figure 10 shows the data used for the 3000 psi and equivalent 8000 psi aileron wing circuits used in the heat transfer model. In order to calculate the time versus temperature curve for an equivalent 8000 psi system, it was recognized from the plot of the 3000 psi aircraft data that a finite time period was required for hot oil to flow from the pump to the aileron actuator.

Analysis of the aircraft test data from reference [2] indicates that aileron inlet temperature began to rise when the PC2 pump pressurized the system at 133 seconds after engine start. Extension of the linear portion of the aircraft test data to -40°F gave 170 seconds elapsed time as the start of

AILERON INLET OIL TEMPERATURE VS TIME

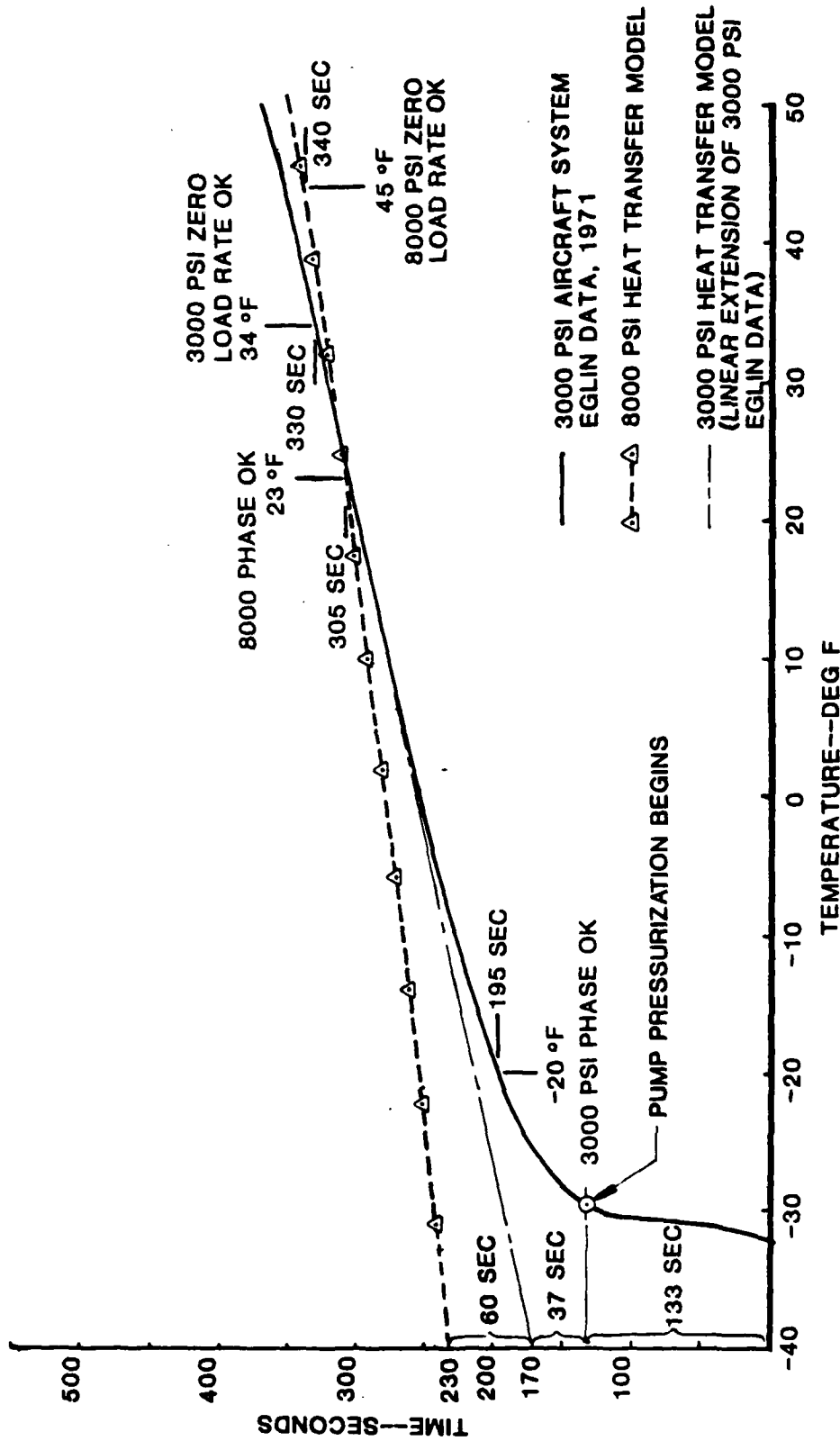
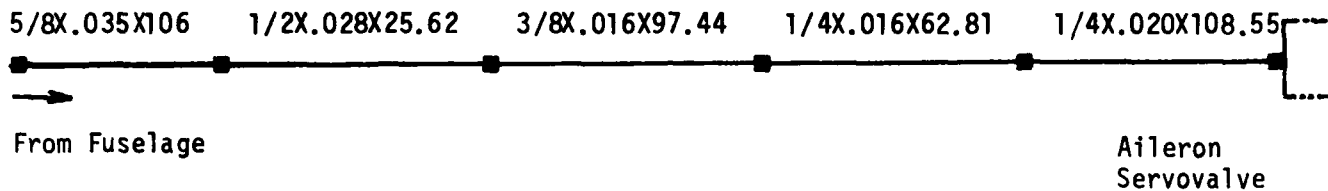


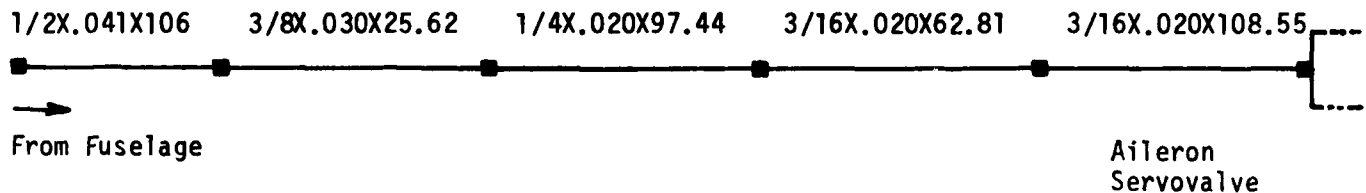
FIGURE 9. AILERON OIL INLET TEMPERATURE VERSUS TIME establishes time to temperature for minimum performance of each system.

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3000 PSI SYSTEM



$$\begin{aligned}
 \text{Tube Surface Area} &= 497.75 \text{ IN}^2 = 3.46 \text{ Ft}^2 \\
 \text{Tube Volume} &= 12.01 \text{ IN}^3 = 3.603 \text{ lb} \quad \text{STL} \\
 \text{Oil Volume} &= 44.72 \text{ IN}^3 = 1.44 \text{ lb} \quad \text{MIL-H-5606} \\
 \text{Cv Steel} &= .109 \text{ BTU/lb-}^\circ\text{F} \\
 \text{Cv MIL-H-5606} &= .48 \text{ BTU/lb-}^\circ\text{F} \\
 \text{Total Surface} &= 3.46 + .35^* = 3.81 \text{ FT}^2 \\
 \text{Total } \Sigma \text{ mCv} &= .109(3.603 + .36^*) + .48(1.44 + .14^*) = 1.19 \text{ BTU/}^\circ\text{F}
 \end{aligned}$$

8000 PSI SYSTEM



$$\begin{aligned}
 \text{Tube Surface area} &= 374.16 \text{ IN}^2 = 2.6 \text{ Ft}^2 \\
 \text{Tube Volume} &= 10.31 \text{ IN}^3 = 3.093 \text{ lb} \quad \text{STL} \\
 \text{Oil volume} &= 22.85 \text{ IN}^3 = .72 \text{ lb} \quad \text{MIL-H-83282} \\
 \text{Cv STEEL} &= .109 \text{ BTU/lb-}^\circ\text{F} \\
 \text{Cv MIL-H-83282} &= .45 \text{ BTU/lb-}^\circ\text{F} \\
 \text{Total Surface} &= 2.6 + .26^* = 2.86 \text{ Ft}^2 \\
 \text{Total } \Sigma \text{ mCv} &= .109(3.093 + .45(.73 + .07^*)) = .731 \text{ BTU/}^\circ\text{F}
 \end{aligned}$$

* 10 Percent added for Fittings

FIGURE 10. CIRCUIT DATA USED IN HEAT TRANSFER STUDY is based upon aileron plumbing in fuselage and wing of A-7 aircraft.

heating for the 3000 psi heat transfer model. In order to calculate the elapsed time to start of heating for the 8000 psi heat transfer model, the sum of three time periods must be calculated.

$$t_{8000} = t_1 + t_2 + t_3$$

where: t_1 is the elapsed time from engine start until pump pressurizes system. $t_1 = 133$ seconds which is equal to t_1 for the 3000 psi heat transfer model.

t_2 is the delta time from system pressurization to the straight line intersection at -40°F . $t_2 = 37$ seconds which is equal to t_2 for the 3000 psi heat transfer model.

t_3 is the delta time due to difference in neutral leakage flow of the 3000 psi and 8000 psi systems.

Therefore, $t_3 = 0$ for the 3000 psi heat transfer model. For the 8000 psi heat transfer model, t_3 is proportional to the total elapsed time for the 3000 psi heat transfer model and inversely proportional to the maximum allowable neutral leakage in equivalent 3000 psi and 8000 psi aircraft hydraulic systems. Assuming maximum neutral leakage for each circuit, the time for a unit volume of fluid to flow from the PC2 pump to the aileron actuator inlet is 30.76 seconds. For an equivalent 8000 psi system, the time for a unit volume of fluid to flow from the PC2 pump to the aileron inlet is 43.1 seconds. Since t_1 and t_2 for the 3000 psi heat transfer model are identical to t_1 and t_2 for the 8000 psi heat transfer models, $t_{8000} = t_{3000} + t_3$.

The equation to solve for t_3 is:

$$\frac{t_{3000} + t_3}{t_{3000}} = \frac{t_{\min} - 8000}{t_{\min} - 3000}$$

Substitution of known values gives:

$$\begin{aligned} \frac{170 \text{ sec} + t_3}{170 \text{ sec}} &= \frac{43.10 \text{ sec}}{30.76 \text{ sec}} \\ t_3 &= 68 \end{aligned}$$

Therefore: $t_{8000} = 133 + 37 + 68 = 238$ seconds

Because the iterative heat transfer model used 10 second increments for each iteration, a time delay of 230 seconds was used for the 8000 psi heat transfer model.

Using a time delay of 230 seconds for heating to begin in the 8000 psi aileron circuit, a rate of heat transfer into the circuit of .660 BTU/sec, and the respective surface areas, masses, and specific heat coefficients, values for 8000 psi aileron inlet oil temperature versus time were calculated. The calculations are shown on Tables 3 and 4. A plot of aileron inlet oil temperature versus time is shown on Figure 9 for the 3000 psi aircraft data and the calculated equivalent 8000 psi aircraft. From Figure 9, the time differentials to attain the same performance standards may be determined. The following times and temperatures were obtained for -45 degree lag in phase angle and 10 in/sec no load rate and shown on Table 5. The maximum time difference is for phase angle and is equal to 1.83 minutes. This represents the additional time for the 8000 psi system to be operational if the aircraft is cold soaked, the engine started, and the system allowed to warm up strictly by means of circulation of warm oil at neutral leakage flow rates. It should be pointed out that if control surfaces were cycled periodically after engine start up, there would be no time difference to attain the same performance for the two systems.

Referring back to Figure 9, the difference in the slope of the linear portion of the time versus temperature curves is due to the reduced surface area and lower mass of the equivalent 8000 psi circuit compared to the 3000 psi circuit.

TABLE 3. HEAT TRANSFER STUDY - 3000 PSI SYSTEM duplicates linear portion of time versus temperature data from A-7D Category II Tests.

CONSTANTS:

.....
 TUBE SURF AREA - SQFT = 3.81
 TUBE MATERIAL MASS-LB = 3.96 (STEEL)
 OIL MASS - LB = 1.58 (MIL-H-5606)
 SPECIFIC HT COEF (STEEL) = 0.11
 SPECIFIC HEAT COEF (OIL) = 0.48
 HEAT INPUT - BTU/SEC = 0.66
 SUM MASS*SPECIFIC HEAT = 1.19

A	B	C	D	E	F	G	H	I
TIME -	HEAT IN -	1ST EST	AVG OIL	DELTA T	HEAT	NET HEAT	2ND EST	EST AIL
SECONDS	BTU	OF TEMP	TEMP RISE		DISSIPAT	ADDED -	OF TEMP	INLET
		RISE			- BTU	BTU	RISE	TEMP - F
		(1)	(2)	(3)	(4)	(5)	(6)	(7)
0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	-40.00
10.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	-40.00
20.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	-40.00
30.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	-40.00
40.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	-40.00
50.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	-40.00
60.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	-40.00
70.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	-40.00
80.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	-40.00
90.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	-40.00
100.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	-40.00
110.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	-40.00
120.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	-40.00
130.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	-40.00
140.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	-40.00
150.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	-40.00
160.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	-40.00
170.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	-40.00
180.00	6.60	5.54	-37.23	2.77	0.07	6.53	5.47	-34.53
190.00	6.60	5.54	-31.76	8.24	0.22	6.38	5.35	-29.17
200.00	6.60	5.54	-26.41	13.59	0.36	6.24	5.23	-23.94
210.00	6.60	5.54	-21.17	18.83	0.50	6.10	5.12	-18.82
220.00	6.60	5.54	-16.05	23.95	0.63	5.97	5.00	-13.82
230.00	6.60	5.54	-11.05	28.95	0.77	5.83	4.89	-8.92
240.00	6.60	5.54	-6.16	33.84	0.89	5.71	4.79	-4.14
250.00	6.60	5.54	-1.37	38.63	1.02	5.58	4.68	0.54
260.00	6.60	5.54	3.31	43.31	1.14	5.46	4.58	5.12
270.00	6.60	5.54	7.88	47.88	1.27	5.33	4.47	9.59
280.00	6.60	5.54	12.36	52.36	1.38	5.22	4.37	13.97
290.00	6.60	5.54	16.73	56.73	1.50	5.10	4.28	18.24
300.00	6.60	5.54	21.01	61.01	1.61	4.99	4.18	22.43
310.00	6.60	5.54	25.19	65.19	1.72	4.88	4.09	26.52
320.00	6.60	5.54	29.28	69.28	1.83	4.77	4.00	30.52
330.00	6.60	5.54	33.28	73.28	1.94	4.66	3.91	34.43
340.00	6.60	5.54	37.19	77.19	2.04	4.56	3.82	38.25
350.00	6.60	5.54	41.02	81.02	2.14	4.46	3.74	41.99
360.00	6.60	5.54	44.76	84.76	2.24	4.36	3.66	45.65
370.00	6.60	5.54	48.41	88.41	2.34	4.26	3.58	49.22
380.00	6.60	5.54	51.99	91.99	2.43	4.17	3.50	52.72
390.00	6.60	5.54	55.49	95.49	2.52	4.08	3.42	56.14
400.00	6.60	5.54	58.91	98.91	2.61	3.99	3.34	59.48
410.00	6.60	5.54	62.25	102.25	2.70	3.90	3.27	62.75

NOTES:

1. (HEAT INPUT)/(SUM MASS OIL, TUBING TIMES SP HT) = COL B/(SUM MASS*CV)
2. TEMPERATURE OF INLET OIL AT START OF TIME PERIOD + 1/2 COL C
3. COL D - AMBIENT TEMPERATURE
4. HEAT DISSIPATED = (.00069444) (10 SEC) (AREA) (COL E)
5. COL B -COL F
6. NET HEAT ADDED/(SUM MASS*CV) = COL G/(SUM MASS*CV)
7. TEMPERATURE OF INLET OIL AT START OF TIME PERIOD + COL H

TABLE 4. HEAT TRANSFER STUDY - 8000 PSI SYSTEM estimates time versus temperature using same method as used for 3000 psi system.

CONSTANTS:

TUBE SURF AREA - SQFT	=	2.86
TUBE MATERIAL MASS-LB	=	3.40 (STEEL)
OIL MASS - LB	=	0.80 (MIL-H-83282)
SPECIFIC HT COEF (STEEL)	=	0.11
SPECIFIC HT COEF (OIL)	=	0.45
HEAT INPUT - BTU/SEC	=	0.66
SUM MASS*SPECIFIC HT	=	0.73

A	B	C	D	E	F	G	H	I
TIME -	HEAT IN -	1ST EST	AVG OIL	DELTA T	HEAT	NET HEAT	2ND EST	EST AIL
SECONDS	BTU	OF TEMP	TEMP RISE		DISSIPAT	ADDED -	OF TEMP	INLET
		RISE			- BTU	BTU	RISE	TEMP - F
		(1)	(2)	(3)	(4)	(5)	(6)	(7)
0.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	-40.00
10.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	-40.00
20.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	-40.00
30.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	-40.00
40.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	-40.00
50.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	-40.00
60.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	-40.00
70.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	-40.00
80.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	-40.00
90.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	-40.00
100.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	-40.00
110.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	-40.00
120.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	-40.00
130.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	-40.00
140.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	-40.00
150.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	-40.00
160.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	-40.00
170.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	-40.00
180.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	-40.00
190.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	-40.00
200.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	-40.00
210.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	-40.00
220.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	-40.00
230.00	0.00	0.00	0.00	0.00	0.00	0.00	0.00	-40.00
240.00	6.60	9.01	-35.49	4.51	0.09	6.51	8.89	-31.11
250.00	6.60	9.01	-26.60	13.40	0.27	6.33	8.65	-22.46
260.00	6.60	9.01	-17.95	22.05	0.44	6.16	8.42	-14.04
270.00	6.60	9.01	-9.54	30.46	0.61	5.99	8.19	-5.85
280.00	6.60	9.01	-1.35	38.65	0.77	5.83	7.97	2.11
290.00	6.60	9.01	6.62	46.62	0.93	5.67	7.75	9.86
300.00	6.60	9.01	14.37	54.37	1.08	5.52	7.54	17.40
310.00	6.60	9.01	21.91	61.91	1.23	5.37	7.33	24.73
320.00	6.60	9.01	29.24	69.24	1.38	5.22	7.14	31.87
330.00	6.60	9.01	36.38	76.38	1.52	5.08	6.94	38.81
340.00	6.60	9.01	43.32	83.32	1.65	4.95	6.75	45.57
350.00	6.60	9.01	50.07	90.07	1.79	4.81	6.57	52.14
360.00	6.60	9.01	56.64	96.64	1.92	4.68	6.39	58.53
370.00	6.60	9.01	63.04	103.04	2.05	4.55	6.22	64.75

NOTES:

1. $(\text{HEAT INPUT}) / (\text{SUM MASS OIL, TUBING TIMES SP HT}) = \text{COL B} / (\text{SUM MASS} \cdot \text{CV})$
2. TEMPERATURE OF INLET OIL AT START OF TIME PERIOD + 1/2 COL C
3. COL D - AMBIENT TEMPERATURE
4. $\text{HEAT DISSIPATED} = (.00069444) (10 \text{ SEC}) (\text{AREA}) (\text{COL E})$
5. COL B - COL F
6. $\text{NET HEAT ADDED} / (\text{SUM MASS} \cdot \text{CV}) = \text{COL G} / (\text{SUM MASS} \cdot \text{CV})$
7. TEMPERATURE OF INLET OIL AT START OF TIME PERIOD + COL H

TABLE 5. ESTIMATED TIME TO WARM UP SUMMARY

TIME TO WARM UP FROM -40 °F

TEMPERATURE -- °F TIME --MINUTES

3000	8000	3000	8000
------	------	------	------

AMPLITUDE RATIO AND PHASE OK -20 23 3.25 5.08

TIME DIFFERENCE 3000 vs 8000 $\Delta = 1.83$

ZERO LOAD RATE OF 10 IN/SEC 34 45 5.50 5.67

TIME DIFFERENCE 3000 vs 8000 $\Delta = 0.17$

4.0 CONCLUSIONS

The adoption of 3/16 inch OD tubing in 8000 psi systems using MIL-H-83282 fluid will not affect weapon system readiness at low temperatures. The test results and analysis of this program show that even though the temperature to achieve the minimum performance criteria was higher for the 8000 psi test circuit in each instance, the time to achieve the temperatures was comparable for the two systems when the total aircraft is considered.

The aircraft data used to establish the time to achieve minimum operating temperature came from low temperature tests on an instrumented aircraft with a 3000 psi hydraulic system. The test procedure for the low temperature tests required starting the engine and reaching the elapsed time and temperatures. No cycling of controls was done during the period of interest. As a result, the 3000 psi aircraft system required 5.5 minutes to reach 34 degrees F at which the zero load rate was met. The only means of heat transfer from warm fluid was by neutral leakage of the system. If the flight controls had been cycled immediately after engine start, the zero load rate temperature would have been reached much more rapidly and would have been reduced to one to two minutes.

The 8000 psi aileron circuit tested was representative of the recommendations of Ref [1] for tube diameters but duplicated 3000 psi system line lengths. The performance of the 8000 psi circuit could have been improved if the circuit tube diameters and line length for each diameter had been designed for reduced line losses and increased actuator rate. This was not done in the interest of reducing the number of variables changed from the 8000 psi circuit to the 3000 psi circuit. Also, duplication of line lengths acted to make the analysis more conservative by not making any circuit changes to reduce the temperatures at which the 8000 psi system met minimum performance criteria. Relatively simple line pressure drop and orifice area math models of the circuit could have been used to determine the changes required to improve performance of the 8000 psi circuit to coincide or more nearly match that of the 3000 psi circuit at each temperature.

The no load rate results showed only a minor difference between the 3000 psi and 8000 psi systems. Calculation of the theoretical zero load rate for each actuator assuming full supply pressure to the valve shows the 3000 psi actuator to be moderately overdesigned with an average rate of 13.15 in/sec. The 8000 psi actuator meets the requirement with an average zero load rate of 10.84 in/sec. The orifice slot width on the 8000 psi actuator would have to be .009 versus .0077 to match the 3000 psi actuator theoretical rate.

5.0 CRITIQUE OF TEST RESULTS

5.1 Test Temperatures

Ideally, the performance tests were to be conducted at temperatures of -40, -20, 0, +20, +40, and +120 degrees inlet oil temperature to the aileron actuator. In order to provide sufficient oil at relatively constant temperature for each test, a 10 gallon bladder type accumulator was plumbed into the circuit. The accumulator was pressurized by oil pressure from the pump so that outlet pressure of the accumulator would be fairly constant. Prior to each test, the accumulator, actuator, and all test circuit plumbing were cold soaked for a number of hours. The cold soak time was calculated using a published formula for a cylindrical object. However, as tests were conducted, the resultant oil temperature was not the "ideal" test temperature. This variance is due to the uncertainty of knowledge of the actual bulk oil temperature in the accumulator as there was no way to insert a submerged thermocouple into the accumulator. Therefore, all temperatures in plotted data of test results are based upon actual temperatures recorded during the tests. Also, additional tests were made to assure a sufficient number of data points.

5.2 Test Actuators

As nearly as possible, the 3000 psi and the 8000 psi actuators used in the tests were designed for the same performance requirements. The 8000 psi actuator was on loan from the 8000 psi "iron bird" tests being conducted at the Columbus Division of Rockwell International. The 8000 psi tests were conducted first, then the actuator was returned and 8000 psi plumbing was removed from the test circuit in preparation for the 3000 psi performance tests. It was discovered after the actuator was returned to Rockwell that one of the piston seals in the 8000 psi actuator had failed and the other piston seal was failing. The failed seal had leakage of 80-100 cc/min. The immediate concern was if the leaking piston seals had any affect on the test results. It was concluded that the leaking pistons would not affect test results for the following reasons:

- (1) The tests were run with no load on the actuator. Large differential pressures were not created across the pistons; therefore, leakage during tests would only be the result of a few hundred psi differential pressure across the seal.
- (2) Only one of the piston seals had failed. The actuators are designed for one half of the required thrust and full rate to be available from each half. Therefore, the capability of the seal which had not failed would have been sufficient for the rate dependent tests being conducted.
- (3) The piston seal leakage at worst would only make the 8000 psi actuator operate with reduced performance and would make any conclusions drawn from the tests more conservative.

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APPENDIX A - TEST DATA

TABLE A-1 3000 PSI Amplitude Ratio and Phase Angle Test Data

<u>RUN NO.</u>	<u>"IDEAL" TEST TEMP</u>	<u>ACTUAL AVERAGE OIL TEMPERATURE</u>	<u>AR Hz@ -3db</u>	<u>PHASE Hz@ -45°F</u>
12	120	121.3	15.7	13.0
13	40	41.3	15.0	12.0
14	20	22.2	13.0	12.0
15	0	14.0	13.2	12.0
16	- 20	- 27.4	6.0	1.4
17	- 40	- 27.2	4.6	4.9
18	0	- 3.8	9.0	11.7
19	- 20	- 8.2	9.4	10.3
20	- 20	- 22.9	6.8	8.4
21	- 40	- 39.7	3.3	4.0

TABLE A-2 3000 PSI Zero Load Rate Test Data

<u>RUN NO.</u>	<u>"IDEAL" TEST TEMP</u>	<u>ACTUAL AVERAGE OIL TEMPERATURE</u>	<u>ZERO LOAD RATE</u>
1	60	62.6	11.4
2	80	79.8	12.0
3	100	100.7	12.2
4	120	123.4	12.7
5	40	39.1	10.9
6	20	22.2	9.4
7	0	8.3	5.8
8	- 20	- 18.5	3.6
9	- 40	- 35.7	.92

APPENDIX A - TEST DATA

TABLE A-3 8000 PSI Amplitude Ratio and Phase Angle Test Data

<u>RUN NO.</u>	<u>"IDEAL" TEST TEMP</u>	<u>ACTUAL AVERAGE OIL TEMPERATURE</u>	<u>AR Hz @ -3db</u>	<u>PHASE Hz -45 Deg</u>
3	120	119.7	8.2	7.8
4	40	43.0	9.8	9.8
5	52	51.9	8.9	8.4
6	20	25.3	10.0	8.8
7	0	6.8	10.2	8.8
8	- 20	- 16.1	7.5	6.7
9	- 40	- 33.7	2.5	4.0
10	- 20	- 1.9	8.0	8.7
11	0	24.4	10.5	9.1
12	60	63.5	9.8	7.1
13	40	50.1	8.9	8.0
14	20	38.9	8.4	9.8

TABLE A-4 8000 PSI Zero Load Rate Test Data

<u>RUN NO.</u>	<u>"IDEAL" TEST TEMP</u>	<u>ACTUAL AVERAGE OIL TEMPERATURE</u>	<u>ZERO LOAD RATE</u>
15	80	85.7	11.2
16	90	91.7	11.1
17	120	121.6	11.7
18	60	62.6	11.4
19	40	48.6	10.0
20	0	- 0.4	4.2
21	- 40	- 42.2	.76

APPENDIX B

REVIEW OF WARM UP TIME ON SELECTED AIRCRAFT

1.0 INTRODUCTION

References [2, 4, 5, 6, 7, and 8] were reviewed for any information on hydraulic system performance at low temperatures which may be applicable to this program. The aircraft tested in low temperature environments and reported upon in the listed references were the A-7D, T-38A, A-10A, F-4E, F-16, and F-105B. Of particular interest were references [5 and 7] on the A-10 and F-16 respectively, because the fluid used was MIL-H-83282. One difficulty encountered in making this review was the lack of uniformity in instrumentation and test conditions. Not all reports had time versus temperature data. Some general conclusions may still be drawn.

2.0 RESULTS OF REVIEW

The review is summarized on Table B-1 for the six hydraulic systems. System hydraulic pumps reached full system pressure as low as 10 to 15 seconds and up to 133 seconds after engine start at temperatures of -20°F and below. The test procedures varied from no cycling of hydraulic powered functions initially to immediate cycling of hydraulic powered functions after engine start. Systems using MIL-H-83282 typically reported marked slow down of flaps and speed brake at temperatures of -20°F and below. No outstanding flight control temperature related performance problems were reported. Most hydraulic system problems were external leakage and leakage across accumulator piston seals.

3.0 CONCLUSION

Because no major flight control warm up problems were reported, it can be concluded that if an 8000 psi system has the same or lower warm up times, system availability and readiness will be acceptable.

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TABLE 8-1. LOW TEMPERATURE PERFORMANCE NOTES ON SELECTED AIRCRAFT

AIRCRAFT	HYD FLUID and SYS PRESSURE	TEST TEMPERATURE - °F	COMMENT ON FLIGHT CONTROLS	COMMENT ON SECONDARY SYSTEM
A-7D Ref [2]	MIL-H-5606 3000 psi	-65°F, -45°F, -25°F, 0°F, 70°F	"Operation of the flight controls (power actuation packages) was satisfactory from the standpoint of available power and function." A 133 second delay occurred from engine start until PC2 pressure of 2981 psi was reached.	"During extremely low temperature operation, some variations may be noted in flap and landing gear cycle times."
T-38A Ref [4]	MIL-H-5606 3000 psi	-65°F, -40°F, -20°F, 0°F, 70°F	No warm up time comments	No warm up time comments
A-10A Ref [5]	MIL-H-83282 3000 psi	-50°F, -40°F, -20°F 0°F, 70°F	"Initial Primary Flight Control System (PFCS) and Secondary Flight Control System (SFCS) response and movements were extremely slow and exhibited significant stiffness below -20 degrees F. However, no objectionable lag, sluggishness, or feel stiffness was noted after approximately 20 minutes of engine run time."	Slow speed brake rates and asymmetric flaps were reported at temperatures of -20°F and below. Fifteen to 20 minutes of engine run time warmed systems sufficiently to return operation to normal.
F-4E Ref [6]	Fluid Not Reported 3000 psi	-65°F, -45°F, -25°F, 0°F, 70°F	No warm up time comments	No warm up time comments
F-16A Ref [7]	MIL-H-83282 3000 psi	-40°F and -20°F	"..... the system warm up characteristics were generally adequate. The hydraulic pumps began pressurizing the hydraulic system during the engine start sequence."	Pressure fluctuations and reduced rate was noted at -20°F for speedbrake. Conclusion was that problems could be encountered at lower temperatures.
F-105B Ref [8]	MIL-H-5606 3000 psi	-65°F, -30°F, 0°F, 70°F	No warm up time comments	No warm up time comment

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