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PERFORMANCE ANALYSIS OF FLARED CONNECTORS

GEORGE N. GRAVES, CAPT, USAF
ALBERT B. SPENCER, JR., 1ST LT, USAF

TECHNICAL REPORT AFRPL-TR-71-19

MARCH 1971

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FOREWORD

This report summarizes research conducted from May 1966 to February 1967 under Phase II of the AFRPL in-house Project Number 305802ERB, "Tube Connector Development." The research was performed by the Subsystems Branch of the Liquid Rocket Division, Air Force Rocket Propulsion Laboratory, Edwards, California.

Capt George N. Graves and Lt Albert B. Spencer, Jr., were the Project Engineers and Mr. D. L. Lank was the Engineering Technician.

This technical report has been reviewed and is approved.

JERRY N. MASON, Capt, USAF
Chief, Subsystems Branch
Liquid Rocket Division

ABSTRACT

Current practices in the selection of separable connectors for rocket system plumbing applications require that an analytical investigation of many connector candidates be accomplished, followed by the selection and the conduct of extensive verification testing. This method is both expensive and time consuming. The Air Force Rocket Propulsion Laboratory has sponsored an investigation to demonstrate an analytical leakage-prediction correlation technique for comparison with actual test results received in the evaluation of the standard AN flared tube connector. Success of this approach would allow the use of this prediction correlation technique on other connector concepts. A secondary objective was to evaluate the performance of the AN flared tube connector. The approach consisted of deriving an analytical leakage-prediction technique using a flow conductance parameter in conjunction with the Meyer hardness index (Reference 1) and comparing these data with the test results received from the evaluation of the flared tube connector. The program did not achieve satisfactory prediction correlation between predicted and test results. However, comparison of predicted results with the test results did show the predicted leakage performance values to be extremely conservative. The evaluation of the flared connectors did demonstrate a 10^{-6} atm cc/sec helium leakage capability of the flared connectors when tested under specific operating conditions.

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SECTION I

INTRODUCTION

Go/no go acceptance tests have been used exclusively by many organizations to determine what connectors to use in aerospace systems. This method is expensive and time consuming. In many instances, it represents a duplication of effort without advancing the state of the art.

As part of the Air Force Rocket Propulsion Laboratory's Tube Connector Development Program, Contract AF04(611)-9578 was initiated with Battelle Memorial Institute (BMI) to formulate a procedure for predicting the performance of separable tube connectors. It was anticipated that the results of this study would provide the basis for more efficient and reasonable procedures for selecting "shelf items" for aerospace systems and would permit rapid screening of many connector designs.

The AFRPL Tube Connector Development Program was established with the general objective of providing an independent evaluation of the fitting concepts developed under contracted programs and of evaluating field service problems and skill levels required for connector fabrication and installation. The specific objectives of this phase of the program were:

1. To develop an in-house connector evaluation capability

2. To evaluate the flared connector for comparison with the AFRPL connector concept

3. To generate data on flared connector performance for comparison with the values predicted by the BMI analysis technique.

SECTION II

FLARED TUBE SEAL CONNECTOR CONCEPT

The flared tube, seal-type connector is probably the most common threaded connector for fluid line connections. This connector utilizes the tubing flare to accomplish both the connector-to-connector seal and the tube-to-connector seal. This design has numerous variations, most of which employ the same basic geometry but differ in the manufacturing and quality control requirements.

The AN flared connector can be procured to government or industry specifications. AN flared connectors consist of a sleeve, a connecting nut, and a union (Figure 1). This connector is manufactured in accordance with military specification MIL-F-5509B, titled, "Fitting, Flared Tube, Fluid Connection." In making a tube connection with this type of connector, the following sequence of operations is generally followed:

1. The nut and the sleeve are placed on the tubing.

2. The tubing is flared by a hand tool or by a flaring machine.

3. The connection is secured by placing the flared tubing in contact with the nose of the union, sliding the sleeve and nut against the flared tubing end, and threading the nut into place on the union. The sleeve is forced against the back of the tube flare by the unit as it is threaded onto the union. As the nut is tightened, the sleeve compresses the back of the tube flare against the nose of the union, thus forming the sealing interface. Actual sealing is accomplished by deformation of the flared tube surface against the mating surface of the union.

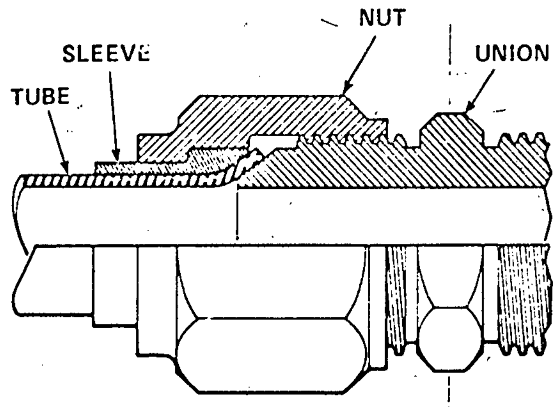

Figure 1. AN Flared Connector

Three distinct geometrical sealing modes are recognized in the connector. These have been categorized as Types I, II, and III and are shown in Figure 2.

In Type I, the mating between the flared cone and the nose of the AN union creates a sealing surface of 0.010 to 0.020 inch wide at the inner edge of the AN union. In Type II, the sealing contact surface ranges from 0.020 to 0.030 inch. The sealing action in Type II is caused by contact of the flared cone with the top edge of the AN union. The Type III seal contact surface is 0.070 to 0.125 inch in width in the middle of the AN union.

A major advantage of the AN flared connector is its simplicity; however, several disadvantages have given rise to modifications and even completely new connector design concepts. Some of the notable disadvantages include:

1. The torque relaxation with time characteristics

2. The inability to obtain sufficient sealing stresses in 1/2-inch and larger connector sizes

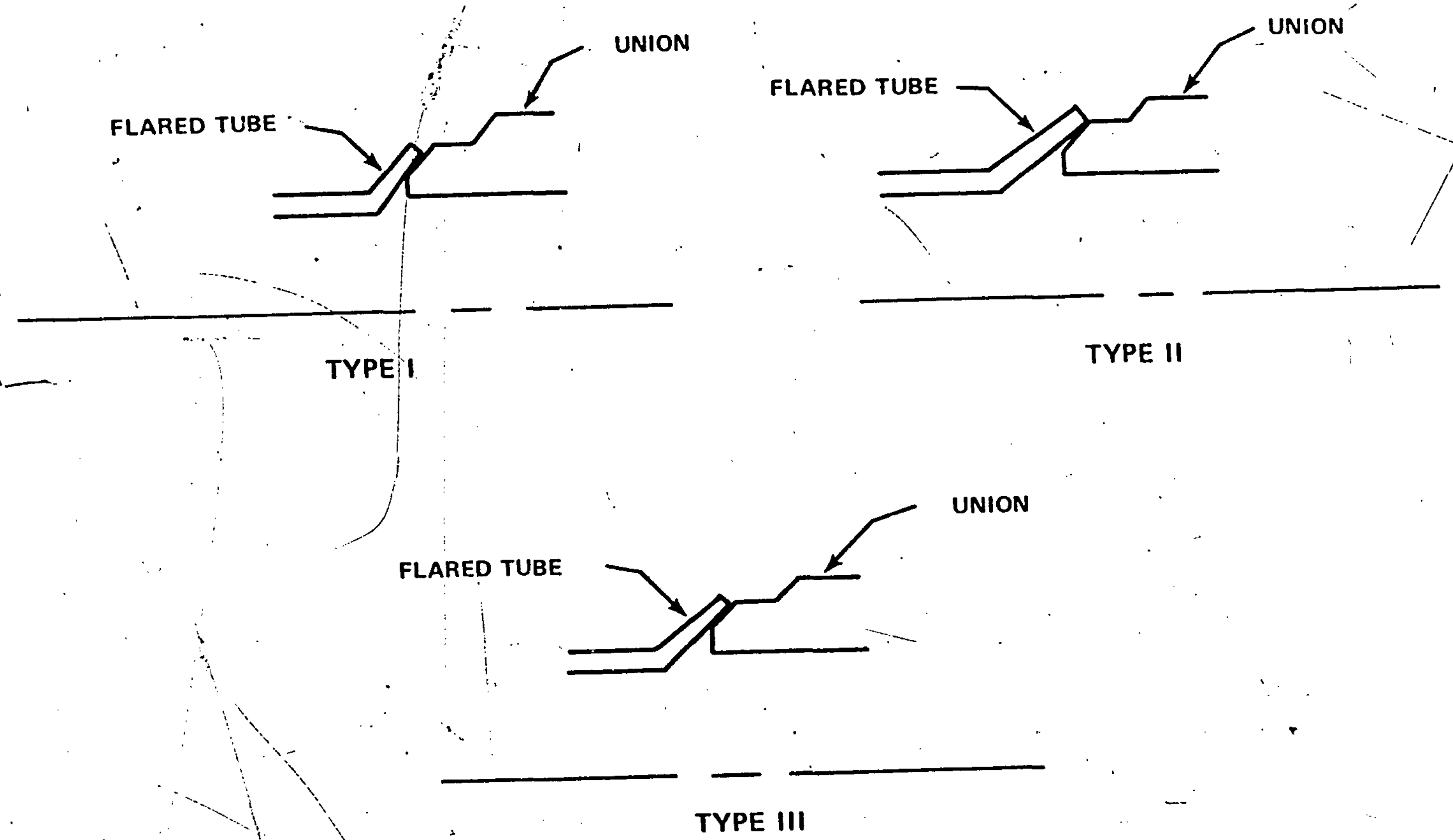

Figure 2. Sealing Mode Geometry

3. Inadequate quality control requirements

4. The tendency of tubing to crack when subjected to flaring operations

Attempts to eliminate the disadvantages just noted have resulted in an AN connector modification that contains a separate seal. This seal is a truncated cone and fits over the end of the 37-degree union nose. The seal is made of a soft material, generally copper or aluminum, offering lower required seating stresses. While presenting a solution to the required seating stress problem, the separate seal has the disadvantage of adding another part to the connector, which could be omitted if proper care is not taken during assembly. Additionally, it does not eliminate the disadvantages previously identified in steps 1, 3 or 4 of the previous paragraph.

SECTION III

EXPERIMENTAL AND ANALYTICAL INVESTIGATIONS

Exact axial loads resulting from the assembly torques are difficult to determine theoretically because the friction factors are generally unknown and can vary depending on the lubricant and the connector geometric changes. Consequently, the minimum and maximum initial axial loads were determined experimentally. This was accomplished with the test configuration shown in Figure 3.

The tubes were welded to adapter plates which were bolted to the tensile machine platform. When the parts were lubricated properly and the recommended torques were applied to the connector, the minimum and maximum axial loads were read directly on the tensile machine.

Table I shows the relationships of torques to maximum axial loads in the flared connector.

Table II shows the axial loads transferred to the AN union and flare for the minimum and maximum torques required of each connector size. The seal contact loads calculated for the three different seal geometries are depicted in Table III.

The seal of a flared connector is a series-type seal. The entire axial load applied by applying torque to the nut is transferred through the connector as seal contact stress, so the compressive load contributes entirely to the seating of the seal.

The axial loads obtained on the initial assembly were used for analysis purposes although additional assemblies were evaluated to determine repeatability of torque values. Frequently, the friction factor will decrease as the connector is repeatedly assembled, and as a result,

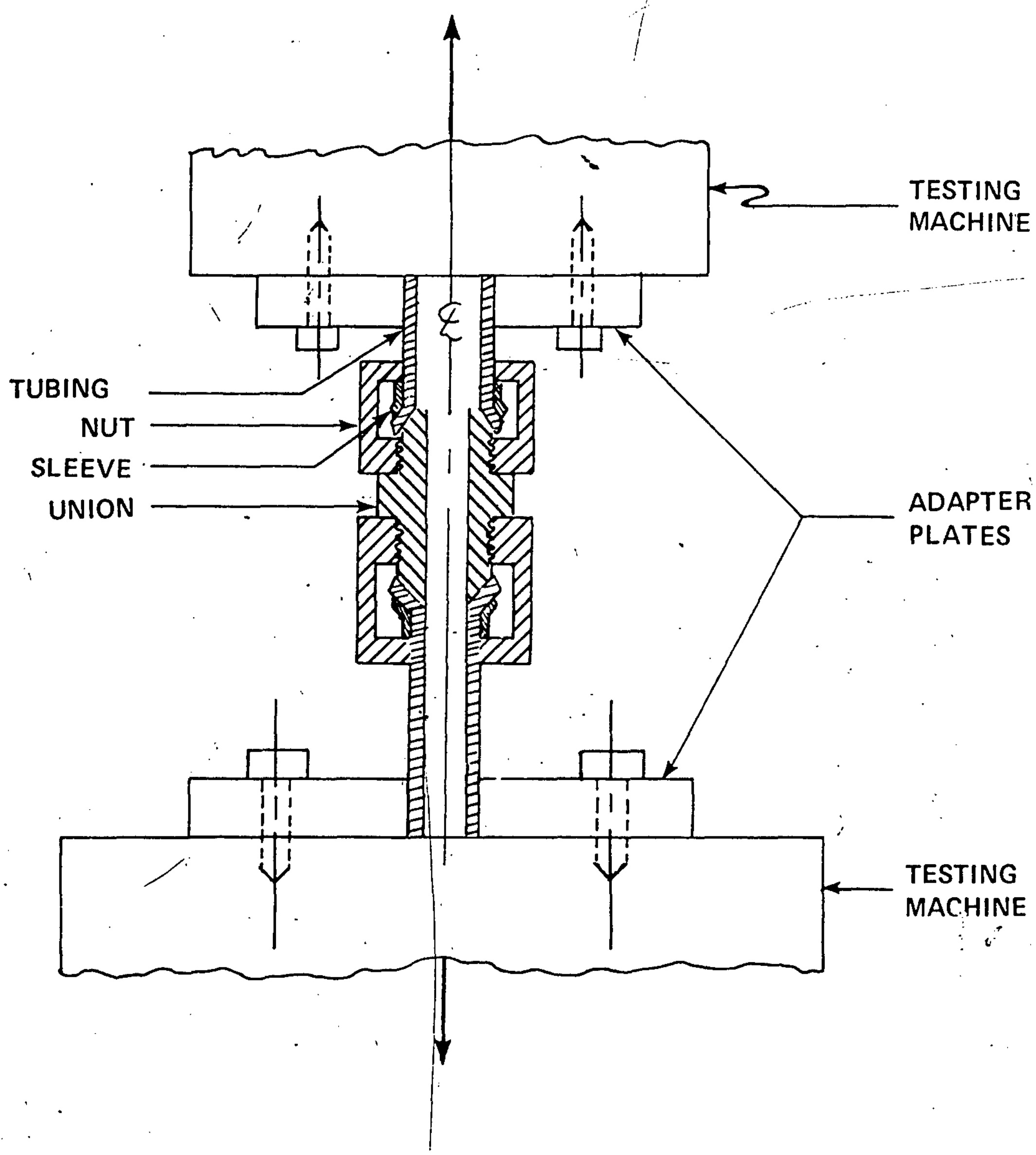

Figure 3. Axial Load Versus Torque Test Setup

TABLE I. TORQUE VERSUS AXIAL LOAD

Tube Size (inches)	Torque (in. /lb)	Axial Load (pounds)
3/8	150	1230
	270*	2200
	300**	2380
	375	3100
	450	3900
1/2	250	2075
	450*	3300
	500**	3650
	625	5230
3/4	375	3000
	650*	5700
	700**	6200
	875	7700
	1050	8650

* Recommended minimum torque
** Recommended maximum torque

the nut will be overstressed if compensation for this friction effect is not considered in applying the assembly torque.

The deflection rates of the compression members in the connector were not determined experimentally. However, the calculated values for the 3/8-, 1/2- and 3/4-inch connectors were 0.104×10^{-6}, 0.174×10^{-6} and 0.134×10^{-6} in. /lb, respectively.

These compressive deflection rates correspond well with the value of 0.222×10^{-6} in. /lb, experimentally determined for the Wiggins DL flared connector by BMI.

TABLE II. AXIAL SEALING LOADS*

	Tube Size (inches)					
	3/8		1/2		3/4	
	Min.	Max.	Min.	Max.	Min.	Max.
Initial Assembly	2200	2380	3300	3650	5700	6200
Type I						
Working Pressure	1420	1600	1870	2200	1050	1550
Proof Pressure	1250	1450	1670	2020	760	1260
Type II						
Working Pressure	1070	1280	1310	1660	1040	1540
Proof Pressure	800	1000	800	1150	-20	480
Type III						
Working Pressure	1240	1430	1630	1980	1550	2050
Proof Pressure	1050	1240	1290	1640	730	1230

* Values are in pounds.

TABLE III. CALCULATED SEALING FORCES*

	Tube Size (inches)					
	3/8		1/2		3/4	
	Min.	Max.	Min.	Max.	Min.	Max.
Initial Assembly	1320	1420	2000	2200	3440	3750
Type I						
Working Pressure	860	965	1130	1340	635	940
Proof Pressure	750	875	1050	1230	1460	760
Type II						
Working Pressure	645	760	790	1020	630	930
Proof Pressure	480	605	485	695	-	290
Type III						
Working Pressure	745	860	990	1200	930	1240
Proof Pressure	635	750	780	990	440	745

* Values are in pounds.

The tensile deflection rates were determined experimentally. The deflection rate of the nut was determined using the arrangement shown in Figure 4. The deformations and deflections occurring in the test fixture and in the adapter flange were subtracted from the total measured deflection rate. The bending deflection rates of the flanged tubes must be added to the deflection rate of the nut to determine the deflection rate of the tensile members. The average values for increasing load conditions were 0.653×10^{-6} in./lb for the 3/8-inch connector, 0.572×10^{-6} in./lb for the 1/2-inch connector, and 0.470×10^{-6} in./lb for the 3/4-inch connector. Large hysteresis loops were found during the loading and unloading of the smaller-size connectors.

The structural loads which a connector must withstand consist of the pressure end load and the tube bending moment. Pressure end load is based on the maximum seal diameter and the operating and proof pressures. The bending moment is based on tubing diameter, wall thickness, and material properties.

Table IV gives the mean seal diameters that were used in calculating the pressure end loads for AN flared connectors.

TABLE IV. EFFECTIVE SEAL DIAMETERS

Tube Size (inches)	AN Seal Diameters (inches)		
	Type I*	Type II*	Type III*
3/8	0.318	0.476	0.397
1/2	0.426	0.654	0.540
3/4	0.664	0.934	0.799

* AN seal geometry type

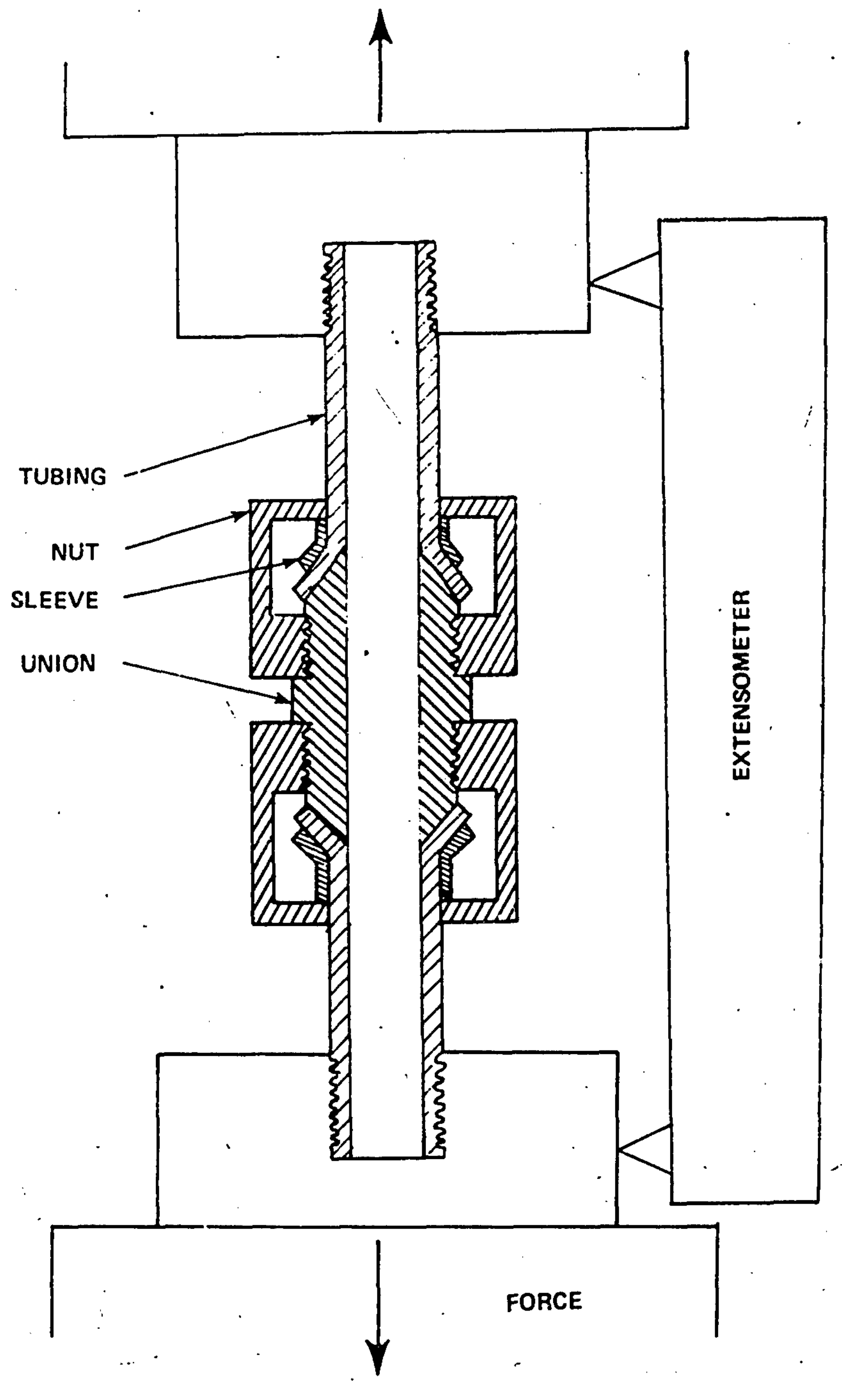

Figure 4. Experimental Arrangement for Determining Nut Deflection

The pressure end load acting on the structure is equal to:

$$F_E = \frac{\pi}{4} G^2 \cdot P$$

where

F_E = pressure end load, pounds

G = effective seal diameter, inches

P = internal pressure, psi

Resulting pressure end loads are presented in Table V.

TABLE V. PRESSURE END LOAD SUMMARY (POUNDS)

Connector Size (inches)	Type I*		Type II*		Type III*	
	Working Pressure	Proof Pressure	Working Pressure	Proof Pressure	Working Pressure	Proof Pressure
3/8	320	478	714	1070	497	745
1/2	572	856	1350	2030	908	1380
3/4	1380	2080	2750	4150	2020	3050

*AN seal geometry type

A bending moment, M, producing bending loads may be present because of tubing misalignment, thermal expansion, or contraction of the tubing system, vibrations, displacements of anchors, or acceleration forces. Bending moments imposed on a connector in a tubing system cannot be determined in advance, since these moments depend upon the specific tubing system, its operating condition, and the location of the connector in the system. The maximum bending moment that can be applied to a connector through the attached tubing is given by the equation:

$$M = SZ$$

where

S = limiting stress of tube material, psi
Z = section modulus of tube cross section, $in.^3$

For thin-walled tubing, the section modulus can be closely approximated by:

$$Z = \frac{\pi D^2 t}{4}$$

where

$$t = \frac{PD}{2S_h} = \frac{PD}{200,000} \geq 0.005 \text{ inch}$$

and

t = tube wall thickness, inches
P = internal pressure, psi
D = tube diameter, inches
S_h = design shear stress, taken as two-thirds of the yield strength at 70^oF

The required bending moment and the equivalent axial load for each size of AN flared connector is shown in Table VI.

TABLE VI. AXIAL LOAD EQUIVALENTS
OF BENDING MOMENTS

AN Connector Size (inches)	Required Bending Moment (in. -lb)	Equivalent Axial Load (pounds)
3/8	50	778
1/2	125	1340
3/4	425	2920

An axial load, equivalent to the bending moment, can be stated in terms of equivalent internal pressure. The maximum bending stress is given as:

$$S_B = \frac{M}{Z}$$

where

S_B = maximum bending stress, psi

M = design bending load, in. -lb

Z = section modulus, in.^{3}

The tensile stress exists at only one point on the tube circumference; it is assumed that the connector may be designed as if S_B existed uniformly all around the tube circumference. The bending load can be expressed as an equivalent internal pressure, P_B given by:

$$P_B = \frac{4 S_B t}{D}$$

The equivalent axial load, F_B (Table VI), is expressed as:

$$F_B = P_B \frac{\pi}{4} G^2$$

and the total equivalent structural end load is:

$$F_T = F_E + F_B$$

which is shown in Table VII.

TABLE VII. TOTAL EQUIVALENT END LOAD

	Connector Size (inches)		
Connector Type	3/8	1/2	3/4
<u>Type I</u>			
Working Pressure	1098	1912	4300
Proof Pressure	1256	2196	5000
<u>Type II</u>			
Working Pressure	1492	2690	5670
Proof Pressure	1848	3370	7070
<u>Type III</u>			
Working Pressure	1275	2248	4940
Proof Pressure	1523	2720	5970

An approximation of leakage was calculated. This is very useful in determining if a connector has the desired sealing potential.

The greatest difficulty in determining sealing criteria is the generation of quantitative relationships among performance parameters that can be used in the prediction of leakage. Leak paths are formed by the interconnection of void spaces in an interface. The void spaces are formed by fabrication marks, damage and/or contaminants. The size and number of leakage paths will depend primarily upon the control established during the fabrication process where widely divergent surface characteristics can be generated.

The relationship between load and leakage is important because they provide correlation among the parameters of interface sealing width, sealing stress, load and leakage. This aspect is most important when analyzing a connector such as the AN flared fitting which depends solely upon axial load for sealing.

To calculate leakage, the conductance parameter and the modified stress ratio were used as two correlation factors. When two surfaces are placed together with some amount of force to form a sealing interface, this condition provides resistance to the flow of a fluid. This resistance to leakage can be represented by a conductance parameter factor (h) and is related to the leakage flowrate (Q) and pressure difference (ΔP) at the interface by $Q = h \Delta P$. The conductance parameter described above is given as:

$$h^3 = \frac{12 \mu (r_o - r_1) P_o Q_o}{\pi (r_o + r_1)(P_2^2 - P_1^2)} - \frac{12.76 \, \epsilon P_o \lambda_o \quad (h^2)}{(P_2 - P_1)}$$

where

h^3	= conductance parameter
P_2, P_1	= inlet and outlet fluid pressures
λ_o	= molecular mean free path of the gas, at standard temperature and pressure conditions
μ	= viscosity of the gas
r_1, r_o	= dimensions of circular interface
P_o	= standard pressure
Q_o	= leakage, atm cc/sec

In determining leakage, a second correlation is the modified stress ratio (MSR). The MSR relates to the apparent contact area, contact load and material hardness and is calculated as follows:

$$MSR = \frac{P^{(2/n')}}{A_A \sigma_M}$$

where MSR = modified stress ratio

P = applied load, pounds

n' = Meyer index

A_A = interface area, in.^{2}

σM = Meyer hardness, psi

The Meyer hardness was determined from sample measurements made on several specimens and the average value was 257,000 psi. The Meyer index was found to be 2.35.

Seal interface areas were found. The results of these calculations are illustrated in Table VIII. The modified stress ratio was calculated and is shown in Table IX for each seal geometry and associated tube size.

TABLE VIII. APPARENT SEAL INTERFACE AREAS

	Seal Geometry (in.^{2})		
Tube Size (inches)	Type I	Type II	Type III
3/8	0.020	0.023	0.044
1/2	0.027	0.039	0.061
3/4	0.042	0.069	0.088

TABLE IX. CALCULATED VALUES FOR MODIFIED STRESS RATIO

Sealing Geometry		Modified Stress Ratio ($\times 10^{-2}$)		
Tube Size (inches)	Torque (in.-lb)	Type I	Type II	Type III
3/8	270	6.04	4.17	2.47
	300	6.70	4.79	2.76
1/2	450	5.79	2.97	2.36
	500	6.51	3.64	2.68
3/4	650	2.50	1.35	1.48
	700	3.10	1.89	1.88

Figure 5 illustrates the correlation between the modified stress ratio and the conductance parameters as a function of the surface roughness. The surface finish was measured for several specimens. Surface roughness ranged from 10 to 50 micro-inches peak-to-valley (PTV) in all specimens. The greatest ~~number~~ of samples had a surface roughness range of 30 to 50 micro-inches ±1 percent. Table X shows the conductance parameter for the appropriate seal geometries and tube sizes.

The contact length must be calculated on the basis of a contact stress equal to the Meyer hardness and the axial and radial loads. The total interface length is defined as the sum of the axial and radial interface lengths because each produces a resistance to flow:

$$L = \frac{F_A}{D_M \sigma_M} + \frac{F_R}{D \sigma_M}$$

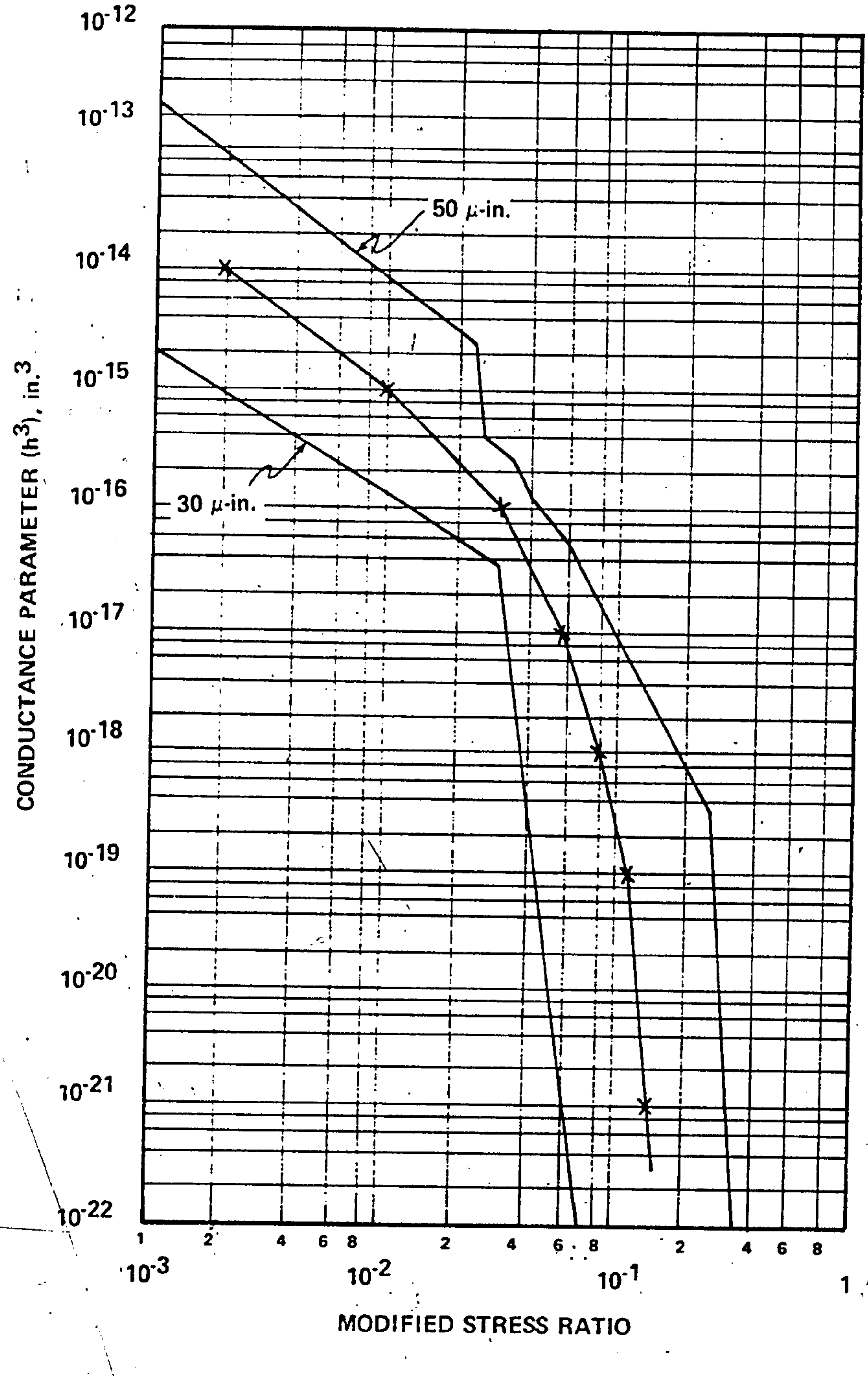

Figure 5. Modified Stress Ratio for Sealing Surfaces Having 30 to 50 μ-in. PTV Roughness

where F_A = axial load, pounds

F_R = radial load = 0

D_M = mean diameter of axial interface

D = diameter of radial interface

σ_M = Meyer hardness = 257,000 psi

TABLE X. CONDUCTANCE PARAMETERS FOR AN FLARED FITTING

(Sealing Surfaces of 30 to 50 μ-in. PTV Roughness)

Sealing Geometry		Conductance Parameter, h^3 (in.^{3})		
Tube Size (inches)	Torque (in.-lb)	Type I	Type II	Type III
3/8	270	9.0×10^{-18}	3.4×10^{-15}	1.8×10^{-16}
	300	5.0×10^{-18}	15.0×10^{-15}	1.5×10^{-16}
1/2	450	12.0×10^{-18}	6.0×10^{-15}	2.0×10^{-16}
	500	5.5×10^{-18}	4.5×10^{-15}	1.1×10^{-16}
3/4	650	150.0×10^{-18}	17.0×10^{-15}	5.0×10^{-16}
	700	100.0×10^{-18}	10.0×10^{-15}	3.0×10^{-16}

Table XI shows the contact length for the various torque loads applied to the seal geometries.

TABLE XI. APPARENT CONTACT LENGTH OF LEAK PATH

Sealing Geometry		Leak Path Length (inches)		
Tube Size (inches)	Torque (in. - lb)	Type I	Type II	Type III
3/8	270	0.0055	0.0028	0.0039
	300	0.0062	0.0033	0.0045
1/2	450	0.0055	0.0025	0.0037
	500	0.0065	0.0032	0.0045
3/4	650	0.0019	0.0014	0.0024
	700	0.0006	0.0020	0.0032

The apparent circumferential contact interface length is found by:

$$W = \pi D_M$$

where D_M = mean diameter of seal contact area

Contact interface length for the appropriate tube sizes is shown in Table XII.

TABLE XII. APPARENT CIRCUMFERENTIAL CONTACT INTERFACE LENGTH

	Sealing Geometries (inches)		
Tube Size (inches)	Type I	Type II	Type III
3/8	1.0	1.495	1.25
1/2	1.34	2.05	1.695
3/4	2.09	2.93	2.51

By knowing the above data, the leakage could be predicted from:

$$Q = \frac{(P_2^{\,2} - P_1^{\,2})}{24 \mu P_1} \quad \frac{W}{l} h^3$$

where
- Q = leakage rate, atm cc/sec
- P_2, P_1 = inlet and exit fluid pressures, psi
- μ = viscosity of fluid (0.411×10^{-6} lb sec/ft)
- W = width of leak path, inches
- l = length of leak path in the flow direction, inches
- h^3 = conductance parameter

The calculated predicted leakage values are shown in Table XIII.

TABLE XIII. CALCULATED LEAKAGE RATES

Sealing Geometry		Leakage Rate (atm cc/sec)		
Tube Size (inches)	Torque (in. - lb)	Type I	Type II	Type III
3/8	270	1.76×10^{-4}	1.96×10^{-1}	6.25×10^{-3}
	300	1.05×10^{-4}	7.35×10^{-1}	4.50×10^{-3}
1/2	450	3.16×10^{-4}	5.30×10^{-1}	9.91×10^{-3}
	500	1.23×10^{-4}	3.1×10^{-1}	4.47×10^{-3}
3/4	650	1.79×10^{-2}	3.85×10	5.65×10^{-2}
	700	3.77×10^{-2}	1.59×10	2.54×10^{-2}

SECTION IV

TEST RESULTS

The following gives the test results from the evaluation conducted on the flared connector under stress-reversal bending, vibration, pressure, impulse, and proof pressure test conditions. The detailed procedures and apparatus used are described in the Appendix.

Stress-reversal bending tests were conducted on the flared connectors. Half of the connectors were assembled at low-torque and the remaining at high-torque conditions. The connectors were tested for 300,000 cycles, and the test stress level was based upon a total combined equivalent stress of 29,000 psi. Results from the stress-reversal bending tests are recorded in Table XIV.

Vibrational and stress-reversal bending are very similar in that they both create the same stress distribution around the tube. The principal difference is the rate of cycling: 25 Hz for stress-reversal bending as compared to 150 Hz for vibration. Additionally, the stress-reversal bending test has the tube mounted as a cantilever beam while the vibration test has the tube mounted as a simply supported beam. The vibration test was conducted in accordance with AFRPL-TM-66-8; results are reported in Table XV.

Pressure impulse testing has two objectives. The first is to consider the water hammer effect upon the sealing surfaces. This may cause the separation of these surfaces, thereby forming a leak path. The second objective is to evaluate the structure's fatigue limits through cyclical, dynamic structural tests. Since pressure impulsing is not generally considered in the design function, a minimum value of 175,000 psi/sec rise and a maximum value of 600,000 psi/sec have been chosen as the test standards in accordance with Society of Automotive Engineers Specification ARP 603.

TABLE XIV. STRESS-REVERSAL BENDING TESTS SUMMARY

A. Condition	Connector Type and Tube Size (inches)* 1/2 F8V17	1/2 F8V16	1/2 F8V18	3/8 F6V30	3/8 F6V31	3/8 F6V32	3/8 F6V33	[illegible] F6V15	[illegible]
Torque (in.-lb)	500	500	370	375	365	300	300	[illegible]	[illegible]
Bending Moment (in.-lb)	126	126	26	51	51	51	50.[illegible]	unknown	unknown
Proof Pressure (psi)	6000	6000	5700	5600	6000	6000	6000	6000	[illegible]

B. Condition	Leakage Measurements for the Respective Condition (atm. cc/sec) 1/2 F8V17	1/2 F8V16	1/2 F8V18	3/8 F6V30	3/8 F6V31	3/8 F6V32	3/8 F6V33	3/8 F6V15	3/8 F6V14
Proof Pressure	8×10^{-10}	6×10^{-7}	1.6×10^{-5}	8.5×10^{-8}	1.4×10^{-7}	1.4×10^{-7}	1.6×10^{-5}	6.2×10^{-8}	5.5×10^{-7}
Leak at Test Start	8×10^{-10}	5.3×10^{-7}	1.2×10^{-5}	1.1×10^{-7}	1.2×10^{-7}	8.3×10^{-8}	1.5×10^{-5}		
Maximum Leak During Test	8×10^{-10}	5.3×10^{-7}	1.3×10^{-5}	1.4×10^{-7}	1.2×10^{-7}	1.2×10^{-7}	2.8×10^{-5}	- -	- -
Leak at End of Test	8×10^{-10}	1.2×10^{-7}	9.0×10^{-6}	4.2×10^{-8}	1.1×10^{-7}	1.0×10^{-7}	2.8×10^{-5}	- -	- -

* Tubing failed at 12,000 cycles due to excessive stress caused by error in test setup.

** Tubing failed at 10,000 cycles due to excessive stress caused by error in test setup.

TABLE XV. VIBRATION TEST SUMMARY

Condition	Connector Type and Tube Size (inches)			
	3/8 F6V19	3/8 F6V18	3/8 F6V17	3/8 F6V16
Stress Level (psi)	5200	5200	5200	5200
Proof Pressure (psi)	4900	4975	4900	- -
Leakage (atm cc/sec) Pressure at Proof	1.3×10^{-8}	3.1×10^{-9}	5.0×10^{-8}	3.0×10^{-5}
Maximum During Test	2.2×10^{-8}	1.8×10^{-8}	4.7×10^{-8}	9.5×10^{-5}
At Finish	1.4×10^{-8}	7.9×10^{-9}	1.1×10^{-8}	9.3×10^{-5}
g Level	2.5	2.5	2.5	2.5

Pressure impulses were induced by rapidly opening or closing valves, thus allowing sudden fluid impulse differentials to be formed in the test item. The test consisted of 20,000 cycles. The average pressure rise rate varied from 200,000 psi/sec (at initial valve opening) to 400,000 psi/sec (at valve fully opened). This pressure rise variation occurred on each cycle.

The proof pressure test is an inspection test to ensure that no gross structural defects exist in the connector before starting structural load tests. Proof pressure, as defined for this series of tests, was 1.5 times the working pressure. The leakage objectives during this series of testing were that the connectors should not exceed 1×10^{-4} atm cc/sec as measured by a helium mass spectrometer.

The test results for pressure impulse and proof pressure are recorded in Table XVI.

TABLE XVI. PRESSURE IMPULSE TEST SUMMARY, AN CONNECTOR

Fitting*	Torque (in.-lb)	Peak Pressure (psi)	Leak Rate at Proof Pressure	Max Leak During Test	Leak Rate at Test End
F6V5	375	5500	4.0×10^{-8}	2.8×10^{-4}	2.6×10^{-4}
F6V6	375	5500	1.7×10^{-9}	8.5×10^{-8}	7.5×10^{-8}
F6V7	375	5500	5.6×10^{-4}	1.4×10^{-3}	1.4×10^{-3}
F6V8	375	4900	2.0×10^{-6}	3.3×10^{-6}	5.1×10^{-7}
F6V9	400	4875	1.3×10^{-3}	**	- -
F6V25	300	5000	1.1×10^{-2}	1.6×10^{-2}	6.5×10^{-3}
F6V26	300	5000	4.0×10^{-7}	2.3×10^{-3}	2.3×10^{-2}
F6V27	270	5000	4.3×10^{-8}	9.7×10^{-4}	8.3×10^{-5}
F6V28	270	5000	2.6×10^{-8}	2.4×10^{-6}	1.5×10^{-7}
F8V9	575	5000	3.4×10^{-4}	6.5×10^{-4}	6.5×10^{-4}
F8V31	625	5500	1	**	- -
F8V32	565	5500	3.1×10^{-1}	**	- -
F8V33	565	5500	5.0×10^{-1}	**	- -
F8V34	565	5500	2.0×10^{-1}	**	- -
F8V35	625	5500	1.7×10^{-1}	**	- -
F8V36	500	5300	2.3×10^{-2}	**	- -
F8V37	625	5100	3.7×10^{-2}	**	- -
F8V38	565	5780	3.7×10^{-2}	**	- -
F8V39	625	4900	3.0×10^{-7}	2.5×10^{-6}	2.5×10^{-6}
F8V40	625	5700	1.8×10^{-2}	**	- -
F8V41	625	5200	1.7×10^{-1}	**	- -

*F6V = 3/8-inch fittings
F8V = 1/2-inch fittings
**Test not performed

SECTION V

DISCUSSION OF RESULTS

Results received from the stress-reversal bending tests conducted on the 3/8- and 1/2-inch flared connectors are given in Table XIV. The connectors were shown to perform remarkably well under the stress-reversal bending condition. No significant degradation in leakage was detected throughout the tests. Comparison of the leakage rates showed that the test items performed significantly better than was predicted. Testing of the 3/8- and 1/2-inch connectors, however, did not confirm a noted deterioration in leakage performance because the larger-size flared connectors were used.

Six 3/8-inch and three 1/2-inch flared tube connectors were tested under stress-reversal bending. Two 3/8-inch connector assemblies prematurely failed due to excessive stress loading that was erroneously applied. Examination of these assemblies showed that circumferential cracking had occurred at the junction of the tubing. The exact applied stress levels were unknown; however, by failing at 10,000 and 12,000 cycles, it was estimated that a total combined equivalent stress level of 35,000 to 40,000 psi was applied.

Four 3/8-inch connectors were tested under vibration conditions with the results tabulated in Table XV. Each connector was subjected to 300,000 cycles at a cyclical rate of 140 and 150 Hz. The vibration frequency was near a resonant condition of the test system which made it difficult to maintain a specific frequency. As shown in the Vibration Test Summary (Table XV), the connectors exhibited significantly lower leakages than predicted, prior to, during and after completion of the vibration cycling.

Twenty-one specimens were prepared for pressure impulse tests, as shown in Table XVI. Nine test specimens of 3/8-inch diameter were subjected to 20,000 pressure impulse cycles. One of the 3/8-inch test specimens was found to leak excessively during proof pressure tests. Twelve, 1/2-inch connections were prepared and tested. Only two of the connections were sufficiently leakproof at proof pressure.

Many of the connectors showed a considerable loss in performance during the conduct of the tests. These performance losses were assumed to be caused by a microfitting problem caused by the input loading of the seal surfaces. Microscopic examination did not confirm this assumption, however.

The 3/8-inch connectors performed two or three orders of magnitude better than the prediction estimated for stress-reversal bending and vibration tests. In the pressure impulse tests, the flared connector's performance was not as high; however, the prediction was within the error range of the test method. The initial sealing performance of the pressure impulse specimens was generally two or three orders of magnitude higher than the analytical prediction. However, the performance degraded during testing and fell to within the predicted ranges.

The suspected primary reason for not accurately correlating the analytical leakage predictions with the test results was the slight deviation in usage of the Meyer hardness parameter. Whereas the Meyer hardness parameter relates an applied load to the projected area of an impression generated on a surface, the use of this parameter was applied to a flared connector design consisting of a truncated cone sealing interface. Other factors that could have caused the disparity in data could have been the three types of sealing interfaces encountered (Figure 2) and the machining variations involved with the manufacture of the connector assemblies.

SECTION VI

CONCLUSIONS AND RECOMMENDATIONS

The following conclusions and recommendations have been made based on the performance prediction and experimental evaluation.

1. Flared connector performance cannot be accurately predicted by the method used. Correlation between the predicted and test results has shown the predicted values to be extremely conservative.

2. Using careful leak-detection inspection methods and under specific operating conditions, the flared connector will perform in the 10^{-6} atm cc/sec leakage range without excess nut torque.

3. It is concluded that the smaller-sized flared connectors show a tendency to seal better than the large sizes.

4. It is recommended that another prediction correlation method be investigated to arrive at satisfactory results.

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2. AF/BSD-TR-61-34, "High Pressure Handbook for Missile Support Systems," December 1961.

3. B. Goobich, J. W. Adams, J. V. Baum and T. M. Trainer, "Performance Analysis of Separable Connectors," Battelle Memorial Institute, Columbus, Ohio, AFRPL-TR-65-190, November 1965.

4. P. Goobich, J. Adams, J. Baum and T. M. Trainer, "Development of AFRPL Threaded Fittings for Rocket Fluid Systems," Battelle Memorial Institute, Columbus, Ohio, AFRPL-TR-65-102, November 1965.

AUTHORS' BIOGRAPHIES

GEORGE N. GRAVES, CAPT, USAF

BSME, Bradley University, June 1962

Capt Graves has served as Project Engineer within the Subsytems Branch of the Air Force Rocket Propulsion Laboratory. His responsibilities have covered advanced liquid rocket tube connector fitting development and test programs.

ALBERT B. SPENCER, JR., 1ST LT, USAF

BSME, University of Missouri, June 1967
MBA, Golden State College, 1969

As Project Engineer within the Subsytems Branch of the Air Force Rocket Propulsion Laboratory, Lt Spencer has been responsible for advanced liquid rocket tube connector fitting development and test program

He has been the author of several reports including:

- AFRPL-TR-69-98, "Tube Connector Development"
- AFRPL-TR-69-101, "Preseated Seal Concept for AFRPL Connector Seal Retention"
- AFRPL-TR-69-159, "Stress Corrosion Investigation of Candidate Braze Alloys"
- AFRPL-TR-69-148, "Automatic Tube Welding and Machining Equipment System Evaluation"

APPENDIX

FITTINGS EVALUATION
TEST REQUIREMENTS AND FACILITIES DESIGN

AFRPL-TM-66-8

FITTINGS EVALUATION

TEST REQUIREMENTS AND FACILITIES DESIGN

LT GEORGE GRAVES, USAF

Author

APRIL 1966

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This Technical Memorandum has been reviewed and approved

AIR FORCE ROCKET PROPULSION LABORATORY
RESEARCH AND TECHNOLOGY DIVISION
AIR FORCE SYSTEMS COMMAND
EDWARDS, CALIFORNIA

1. INTRODUCTION:

The In-House Fittings Evaluation Project was initiated in September 1964 as an integral part of the Air Force Rocket Propulsion Laboratory's Tube Connector Development Program. This project will provide adequate test facilities and a background in test techniques to evaluate tube connector state of the art. The testing experience gained will then be used to evaluate the performance of advanced mechanical tube connectors developed by Battelle Memorial Institute on Contracts AF04(611)-8176 and AF04(611)-9578, "Development of Families of Mechanical Fittings," and an evaluation of the equipment and concepts for welded tube connectors developed by North American Aviation, Los Angeles Division (NAA/LAD) on Contracts AF04(611)-8177 and AF04(611)-9892, "Exploratory Development of Families of Welded Fittings for Rocket Fluid Systems."

This report details the progress made to date in test system design and the operational status of the test facilities.

2. PROJECT PROGRESS:

Considerable progress has been made toward reaching project objectives. An industry survey was conducted to provide guidance in planning tests, establishing test requirements, procedures and instrumentation techniques. Based on the results of the survey, six test systems have been defined. These test systems are burst pressure, pressure impulse, vibration, stress reversal bending, temperature shock and field simulation.

The burst pressure and pressure impulse system have been used to test plain tube sections from 3/8 to 1 inch in diameter. The component parts of temperature shock and vibration systems have been tested. The detailed design drawings are being completed for the stress reversal system. The design criteria are presently being formulated for the field simulation test.

2.1 Survey:

A tubing connector industry survey was initiated to provide general background information, test system requirements, procedures and techniques. Approximately 30 firms, government agencies and professional activities were contacted. A list of the organizations and individuals within the organizations contacted is included as Appendix I. The industry survey provided two types of information. The first type, connector design requirements, as applicable to rocket propulsion, are listed below. The other category, test requirements and instrumentation techniques, will be discussed in paragraph 2.3, Test Requirements.

Pressure ranges	0 - 1,000 psi
	0 - 4,000 psi
	0 - 10,000 psi
Temperature ranges	-425 to 600°F
	-425 to 200°F
Size range	1/8 - 16 inches

Flexure life	300,000 cycles, bending moments of $M = ZS_B$ but not more than $M = 60 (D + 3)$ where M = moment applied to fitting Z = section modulus of tube with fitting S_B = lower of (1) 2/3 yield strength (2) 4/5 stress to rupture (2 yrs) (3) 1/2 200,000 cycle fatigue strength D = outside tube diameter
Vibration	25 to 2,000 cpm at "g" levels to produce stresses equal to 2/3 yield
Compatibility	Rocket propellants (N_2O_4, CTF, LF_2)
Leakage	10^{-7} atm cc/sec of Helium
Temperature shock	Between designed temperature extremes
Repeated assembly	25 times

2.2 Test Requirements:

The requirements placed on each test system are the most important step in planning the evaluation of the performance of a tubing connector. The following paragraphs will describe the requirements for the six test systems. Leakage measurements and proof pressure tests are of such importance throughout the tests that they will be discussed first.

2.2.1 Leakage Measurements:

Leakage may be measured in several different manners depending on the rate and leaking media. The measurement of liquid leakage is very difficult if the leakage rate is less than one cubic centimeter (cc) per minute. Nuclear, ultrasonic and chemical techniques are being developed but are handicapped by high cost or low sensitivity. Therefore, most leakage measurements are made with a gas as the working fluid.

Water displacement is a technique that has been used successfully. The leaking gas is captured in a container under water. The system is generally made so that the captured gas may be maintained at a constant pressure. The volume of water displaced equals the volume leakage of the gas.

This system will measure leaks from high rates down to about 10^{-3} atmospheric cubic centimeters per second (atm cc/sec). Careful work and clever technique will allow measuring leaks of 10^{-4} atm cc/sec; however, the high probability of error inherent in the displacement technique and the considerable time required for measurement limits this technique's usefulness for low leakage rate measurement.

The leakage measurement limits of the water displacement method are 3 to 6 orders of magnitude greater than that expected of the AFRPL connector. For these leakage measurements a helium mass spectrometer leak detector will be used. This technique uses helium as a pressurizing gas and a vacuum chamber to surround the fitting and collect all leaking helium gas. Conventional mass spectrometer techniques are used to sense the helium. The leakage rate can be calibrated with the help of a known leak. This technique will measure leaks in the range of 10^{-5} atm cc/sec to 10^{-9} atm cc/sec. A variation of this technique uses a small variable orifice in a probe which opens the vacuum to the atmosphere. This probe will locate leaks by drawing pressurized helium into the mass spectrometer. The probe technique is not a quantitative technique. Careful operation will locate leaks in the 10^{-6} atm cc/sec range.

Leakage rates will be the prime method of determining failure of connectors, welded connectors and braze fittings. Leakage should not exceed 10^{-7} atm cc/sec. The maximum acceptable leakage for flared fittings will be 10^{-5} atm cc/sec. If leakage degrades to this level any test will be stopped and the flared fitting nut retorqued.

2.2.2 Proof Test:

The proof pressure test is an integral part of all the other tests except the burst pressure test. Proof pressure is defined as 1.5 times the working pressure. This test is an inspection type test to insure that no gross defects exist in the connector seal or mechanical structure. Leakage during this test, for the welded and brazed joints and the AFRPL connector, should be less than 10^{-7} atm cc/sec as measured by a helium mass spectrometer leak detector. Other tubing connectors will have higher allowable leakage values based on their designed capability.

2.2.3 Burst Pressure:

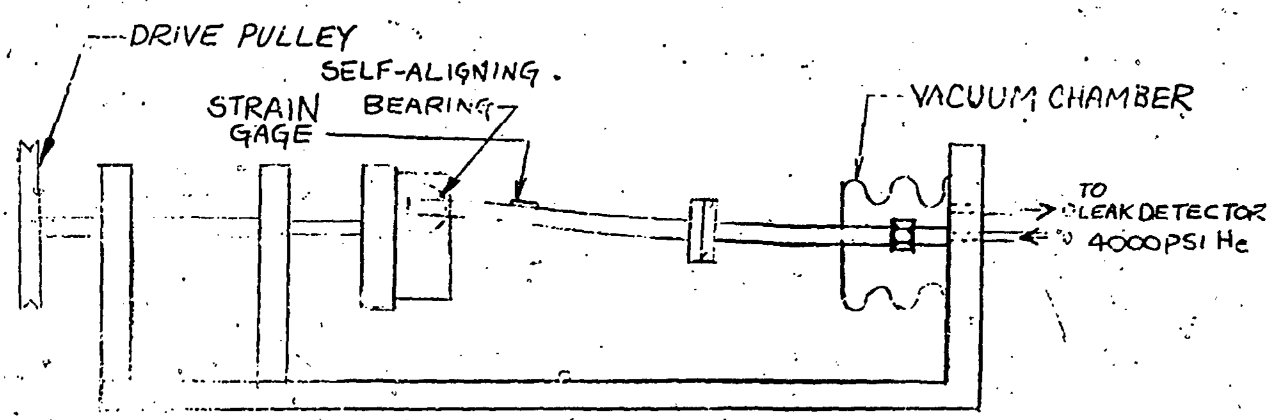
The burst pressure test is a structural test based on the ultimate strength of the structural members of the connector. Burst pressure is defined as two times working pressure without catastrophic failure. The burst pressure should yield structural members of the connector and cause large leaks. These leaks are not measured because they have no relation to designed performance. A burst pressure test not only identifies weak connector structural members; it also demonstrates the actual factor of safety of the connector.

2.2.4 Stress Reversal Bending:

It is highly likely that in rocket propulsion applications tubing lines containing connectors will be subjected to dynamic loading.

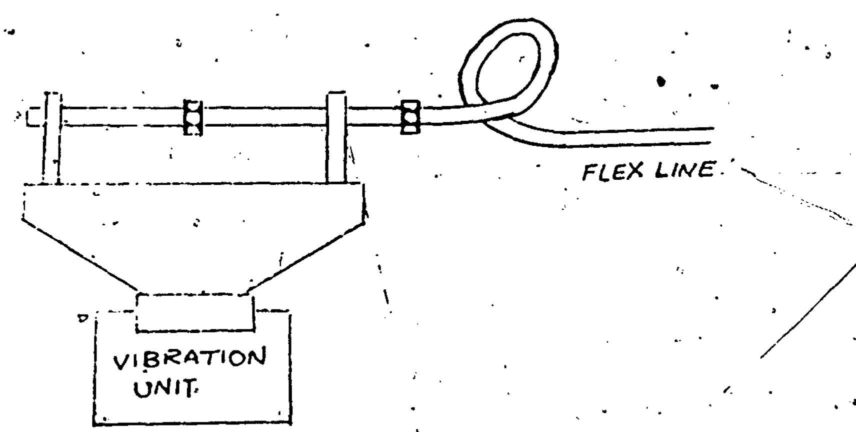
These loads may be caused by vehicle deflection, installation forces, acceleration, etc. The dynamic aspect of these forces generally occurs during rocket engine operation. Thus, the total duration is generally less than five minutes. For dynamic loading, only low total cycle life is required and high stress levels may be maintained.

The test system incorporates a standard stress reversal bending machine design. The connector is mounted near the fixed end and a small vacuum chamber is built around the fitting to collect leakage which is measured with the leak detector. A bellows assembly is used for the vacuum chamber so that it will not affect the stress transmitted to the connector. The stress will be measured by a pair of strain gauges mounted 180 degrees apart on the non-rotating shaft in the plane of maximum stress (see diagram). The bending moment applied to the fitting will be calculated from this and checked by placing strain gauges on plain tube test sections which will be used for checkout of the system. Frequency of flexing will be 25 cycles per second. Half the mechanical tube connectors will be assembled at maximum torque and the remaining half assembled with minimum torque. The connector will be tested for 300,000 cycles or until maximum leakage (as defined in paragraph 2.2.1) is exceeded. The test stress level will be based on the criteria listed in paragraph 2.1.



2.2.5 Vibration:

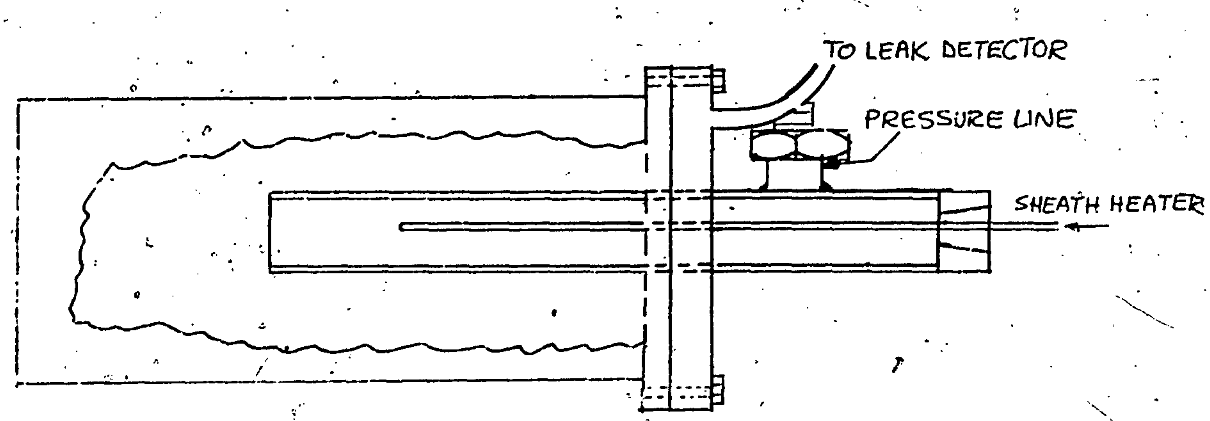
Vibration is one of the most common structural loads and has the potential of relaxing stress which may loosen the nut and unload the seal. The vibration test will load the tube near the connector to 2/3 yield stress (reference paragraph 2.1) of the tube material (20,000 psi for AISI 347). The tube containing the connector will be mounted as an indeterminant beam. The stress will be measured with a pair of strain gauges located on the tube near the connector in the plane of maximum axial stress. Vibration tests will be conducted at the lowest resonant frequency of the test fixture and fitting assembly. The system will be designed for resonance under 500 cps. Duration of the test will be 300,000 cycles or until maximum allowable leakage (as defined in paragraph 2.1) is exceeded.



It should be noted that vibration and stress reversal bending are very similar in that both create the same stress distribution around the tube. The principal differences are the rate of cycling, 25 cps for stress reversal bending as compared to 250 cps or higher for vibration. The stress reversal bending test has the tube mounted on a cantilever beam while the vibration test has the tube mounted as on indeterminate beam.

2.2.6 Temperature Shock:

Between temperature of exhaust products and cryogenic propellants, large temperature gradients may be caused in tubing line connectors. The gradients can have two deleterious effects. High temperatures on the tensioning member (nut) cause relaxation of the sealing load. Thermal gradients in the negative direction raise the stress level in the nut with the possibility of causing yielding or failure. If the nut yields, the sealing load will be reduced upon return to assembly temperature. The system that is designed (see sketch below) is similar to systems which have created transient gradients of 600 to -320°F in less than five minutes.



TEMPERATURE SHOCK SYSTEM

The connector will be pressurized at proof pressure for five minutes. If leakage does not exceed the allowable (10^{-7} atm cc/sec for AFRPL) the pressure will be reduced to working pressure and the connector subjected to six thermal shocks.

Sequence of Thermal Shocks

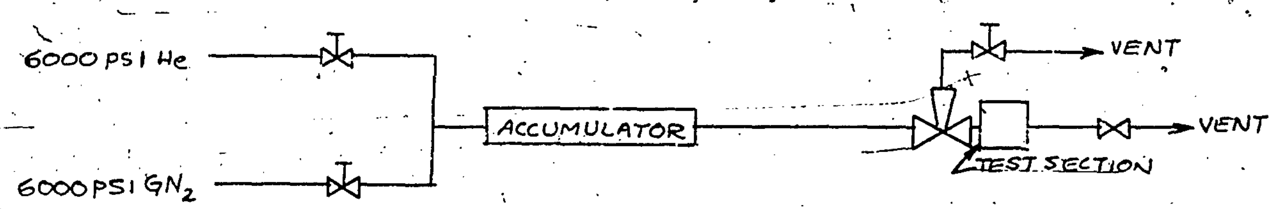
	Cycle Number					
	1	2	3	4	5	6
Initial temperature (Room Temp.)	RT	RT	RT	RT	RT	RT
Intermediate temperature	-320	-320	-320	(1)	(1)	(1)
Transition time	5-8	5-8	5-8	5-12	5-12	5-12
Final temperature (Room Temp.)	RT	RT	RT	RT	RT	RT
Transition time	5-8	5-8	5-8	5-12	5-12	

(1) 600 for AFRPL and other advanced connectors, 300 for flared.

Failure will be leakages greater than those described in paragraph 2.3.1.

2.2.7 Pressure Impulse:

A pressure impulse test was planned to perform two functions. The water hammer effect on sealing surfaces may separate the surfaces forming a leak path. Also, the pressure impulse test is a cyclic dynamic structural test. As the pressure impulse is not generally considered in design, a minimum value of 175,000 psi/sec rise and a maximum value of 600,000 psi/sec has been chosen as a test standard in accordance with the Society of Automotive Engineers ARP 603. Pressure impulses are generally caused by rapidly opening or closing valves. These valves typically have system use cycle lives of 10,000 to 20,000 actuations. The test thus consists of 20,000 cycles.



The system operates with GN_2 for continuous testing due to the high flow ratio (about 20,000 scf per day) and the high cost of He. For leak testing the GN_2 is shut off with a hand valve and helium is cycled through the system. This is accomplished after every 1,750 cycles. The accumulator is used because of the small supply lines to the test area. An electronic switching device is used to synchronize the actuation of the valves. At the start of a cycle both the 3-way and 2-way solenoid valves are in the normally closed position. The 3-way valve is actuated and there is adiabatic charging of the test section volume. After the test section has been charged for half a second the test section is vented through the hand valve, which acts as an adjustable orifice, by returning the 3-way valve to the normally closed position. After another .15 second, the solenoid valve is actuated, which completely vents the system. After the system is vented the solenoid valve is closed and the cycle repeated. Duration of a cycle is approximately 1.1 seconds.

The test section is mounted as closely coupled to the 3-way and 2-way solenoid valves as possible to lower the volume which will be pressurized. Two strain gauge type pressure transducers are mounted one on each end of the test section. The oscillograph trace of a 4,850 psi pressure impulse is shown as figure 1. The average pressure raise rate is 200,000 psi/sec and when the valve poppet is fully open achieves a maximum rate of 400,000 psi/sec. Note the slight decline of maximum pressure due to valve leakage.

2.2.8 Field Simulation:

The field simulation test is one of the most important tests because the installation in real systems may introduce more performance degradation than the design parameter test discussed in the previous paragraphs. The design criteria are presently being defined. The areas of investigation will include:

Repeated assembly limits
Misalignment (axial, angular)
Assembly in confined areas
Contamination limits
Operator technique

The test failure criteria are also being defined; however, generally excessive leakage as defined in paragraph 2.2.1 will be the determining factor.

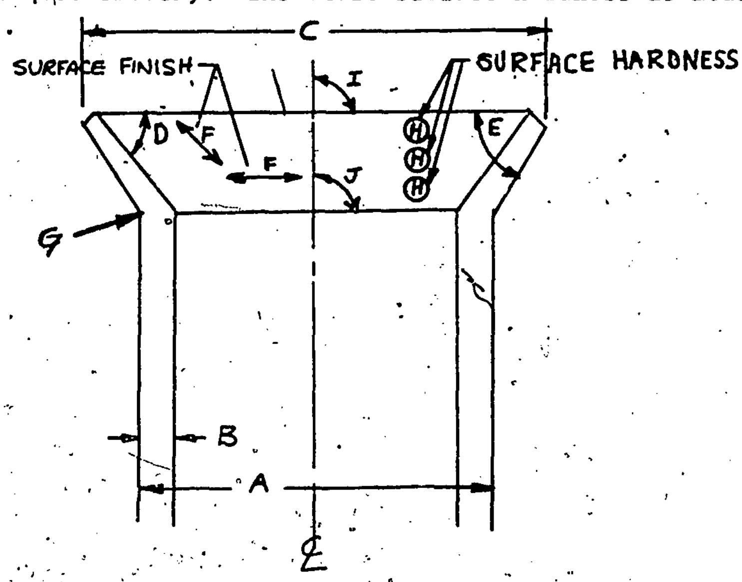
3. INSPECTION PROCEDURES:

3.1 Producing satisfactory flares for MS flared tubing connectors was found to be a difficult task. While there are a series of Military Specifications and Standards covering the subject, the fact remains that in general flares used will not meet these requirements. Furthermore, there is considerable day-to-day difference in flare quality as a result of flaring machines adjustment, operators, tubing materials, techniques, etc.

In order to be able to describe the test items in a manner suitable for analysis of the test data, an inspection procedure was developed. The areas that are inspected are listed below.

Item	Precision	Units
A. Tube O.D.	Nearest .001	Inch
B. Tube Wall Thickness	Nearest .001	Inch
C. Flare O.D.	Nearest .001	Inch
D. Inside Flare Angle	.1°	Angular Degrees
E. Outside Flare Angle	.1°	Angular Degrees
F. Flare Surface Finish	2 Microinches	Peak to Valley
G. Radius	.001	Inch
H. Flare Surface Hardness	2 Units	Brinell Hardness Numbers
I. Flare Squareness	.1°	Angular Degrees
J. Concentricity (flare/tube)	.0001	Inch

The flare surface finish is measured in two directions and at two locations on the flare (see sketch). The flare surface hardness is measured at three locations.



The tube outside diameter, wall thickness and flare outside diameter are measured while the tube is mounted in a precision mill dividing head as there is considerable variation in the above parameters around the circumference of a tube. The flare angles are also measured using the dividing head to hold the flare and a sine bar to measure the angle. A Rockwell superficial hardness test is used to measure the surface hardness of the flare. Flare surface finish is measured with a micrometeric surface finish meter. Similar inspection techniques on AFRPL connectors are being prepared to insure test connectors meet the requirements of proposed Military Standards MS 27850 through MS 27868.

4. FACILITY STATUS:

The facilities used for the test described in the previous section are in various stages of design, fabrication, checkout and use. The status of each test system is discussed below.

4.1 Burst Pressure:

Two burst pressure systems have been used. A hydrostat has been used for burst pressures under 15,000 psi. For burst pressures above 15,000 psi a helium intensifier is used. Because of the inherent danger involved with the release of large amounts of energy at rupture, these tests are conducted behind a concrete wall and vault door in the high vacuum laboratory (Building 8620). The plain tube burst tests have been completed with this facility.

4.2 Stress Reversal Bending:

A stress reversal bending system has been designed and detail part drawings are being completed. Fabrication will be completed by the end of April 1966 and checked out for all sizes by mid-May 1966.

4.3 Vibration:

Vibration tests will be run on the MB C-25 electronic vibrator located in the centrifuge area of Building 8620. A fixture has been fabricated to a design by North American Aviation, Los Angeles Division. This fixture has been mounted on the vibrator and the fixture checked for resonant points. These were identified as 280, 310, 700 and 1550 cps. Pressure lines have been installed into the area, clamps fabricated and flex lines procured. Testing will be initiated upon receipt of strain gages (estimated as 29 April 1966).

4.4 Temperature Shock:

The design of the temperature shock system has been completed and detail drawings finished by the drafting section. Fabrication of the temperature shock system for 3/8 inch tubing has been completed and the component parts checked. Testing of flared connectors will begin 31 March 1966. The test systems for all sizes will be completed by early May 1966.

4.5 Pressure Impulse:

The pressure impulse test system has been fabricated and used to test plain tube sections between 3/8 and 1 inch O.D. Testing with 3/8 inch flared connectors has been initiated.

4.6 Field Simulation:

The field simulation test system is in the design phase. It is estimated that the system will be checked out with flared connectors by mid-May 1966.

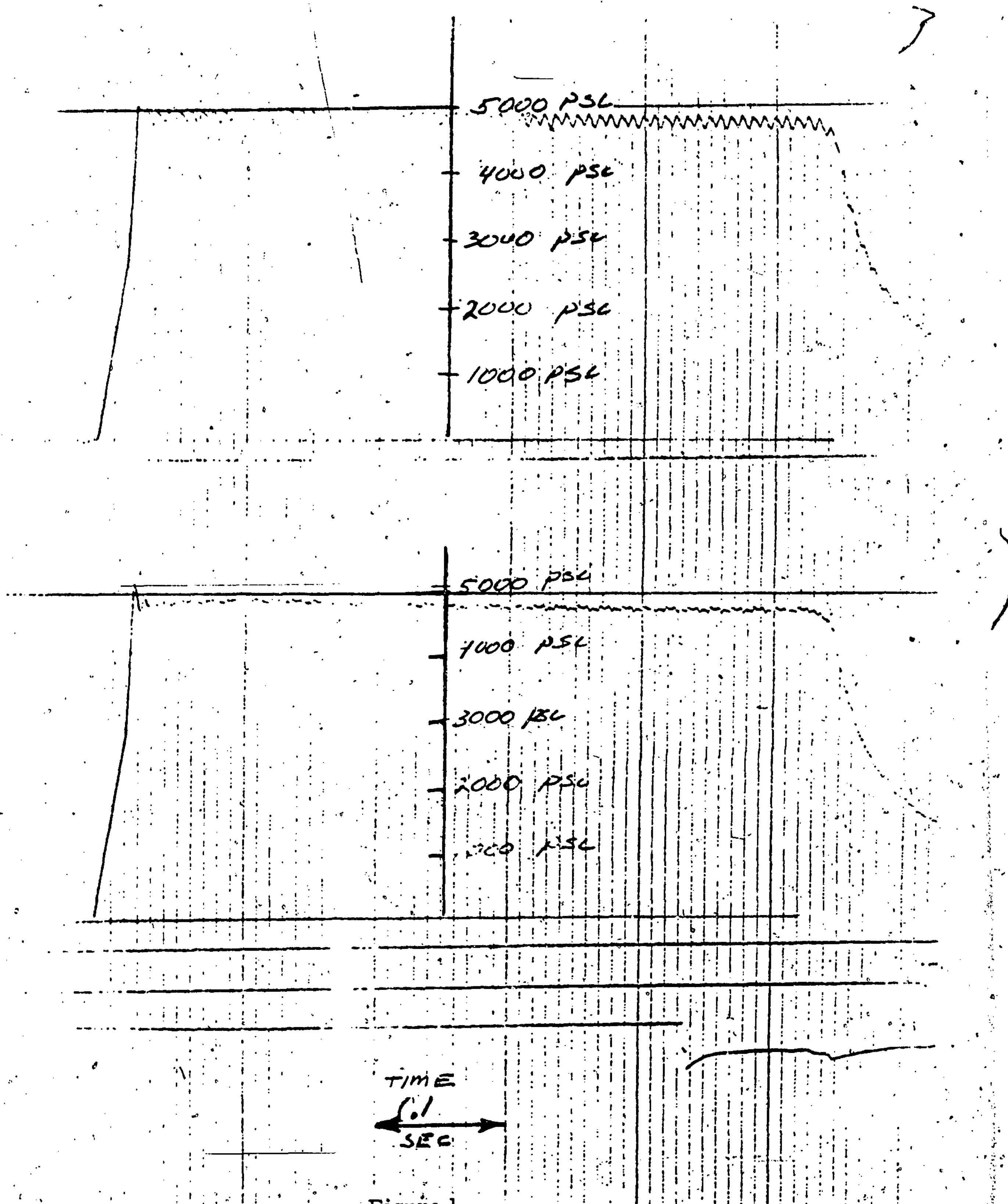

5000 PSI
4000 PSI
3000 PSI
2000 PSI
1000 PSI
5000 PSI
4000 PSI
3000 PSI
2000 PSI
TIME
.1
SEC

Figure 1

APPENDIX I

INDUSTRY SURVEY CONTRACTORS

Government:

Mr. C. Maggio
System Engineering Group (SETPF)
Wright-Patterson AFB, Ohio

Mr. W. P. Britt
R-P&VE-PMC
NASA
Huntsville, Alabama

Mr. P. C. Hass
R-P&VE-PMD
NASA
Huntsville, Alabama

Mr. R. J. Smith
Hq ASD (APL)
Wright-Patterson AFB, Ohio

Mr. A. Bold
AML, Building 76-6
Naval Air Engineering Center
Philadelphia, Pennsylvania

Mr. N. J. Cicere, Jr.
Naval Air Engineering Center
Bureau of Naval Weapons
Philadelphia, Pennsylvania

Mr. D. Rouse
Bureau of Naval Weapons
Department of the Navy
Washington 25, D. C.

Mr. W. Koch
ASL/SENXP
Wright-Patterson AFB, Ohio

Prime and Research Contractors:

Mr. A. C. Farajian
Aerospace Corporation
Applied Mechanics Division
El Segundo, California

Mr. W. D. Padian
North American Aviation, Inc.
Los Angeles Division
International Airport
Los Angeles, California

Mr. A. Pardoe
Space and System Information Division
North American Aviation, Inc.
Downey, California

Mr. M. H. Binstock
Atomics International
8900 De Soto Avenue
Canoga Park, California

Mr. B. Goobich
Battelle Memorial Institute
505 King Avenue
Columbus, Ohio 43201

Mr. Hans Van der Velden
The Boeing Company
Air Plane Division
Renton, Washington

Mr. E. P. Aguilera
Convair, Dept. 528-30
Lindberg Field
San Diego, California

Mr. F. Rathbun
General Electric Company
Bldg. 37-669
Schenectady, New York

Mr. S. Goodman
Electric Boat Division
General Dynamics
Groton, Connecticut

Mr. R. H. Laurell
Dept. 62-22, Bldg. 152
Lockheed-California Company
Burbank, California

Mr. J. F. Slough, Jr.
Dept. 280-053
Los Angeles Division
North American Aviation, Inc.
International Airport
Los Angeles, California

Suppliers and Vendors:

Mr. J. L. Trevino
Mil-Flow
1900 W. Dorothy Lane
Dayton 39, Ohio

Mr. C. R. Keeler
Whittaker Corporation
16217 Lindbergh Street
Van Nuys, California 91409

Mr. C. Kazlauska
11334 Burbank Boulevard
North Hollywood, California

Mr. F. R. Allin
The Weatherhead Company
1736 Standard Avenue
Glendale, California

Mr. W. J. Scott
Voi-Shan Manufacturing Company
Culver City, California

Mr. E. Bentley
Aeroquip Corporation
Burbank, California

Mr. W. Freundlick
Resistoflex Corporation
Roseland, New Jersey

Mr. I. E. Peterson
Parker Aircraft Company
5827 West Century Boulevard
Los Angeles 45, California

Mr. K. Graham
Wiggins Connectors
3424 East Olympia Boulevard
Los Angeles 23, California

Mr. R. H. Williams
Harrison Manufacturing Company
3020 Empire Avenue
Burbank, California 91504

Mr. R. F. PeterJohn
Afco Lunair
701 North Federal Highway
Dania, Florida 33004

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George N. Graves Capt, USAF
Albert B. Spencer, Jr., 1st Lt, USAF

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	Air Force Rocket Propulsion Laboratory Air Force Systems Command, USAF Edwards, California 93523

13. ABSTRACT

Current practices in the selection of separable connectors for rocket system plumbing applications require that an analytical investigation of many connector candidates be accomplished, followed by the selection and the conduct of extensive verification testing. This method is both expensive and time consuming. The Air Force Rocket Propulsion Laboratory has sponsored an investigation to demonstrate an analytical leakage-prediction correlation technique for comparison with actual test results received in the evaluation of the standard AN flared tube connector. Success of this approach would allow the use of this prediction correlation technique on other connector concepts. A secondary objective was to evaluate the performance of the AN flared tube connector. The approach consisted of deriving an analytical leakage-prediction technique using a flow conductance parameter in conjunction with the Meyer hardness index (Reference 1) and comparing these data with the test results received from the evaluation of the flared tube connector. The program did not achieve satisfactory prediction correlation between predicted and test results. However, comparison of predicted results with the test results did show the predicted leakage performance values to be extremely conservative. The evaluation of the flared connectors did demonstrate a 10^{-6} atm cc/sec helium leakage capability of the flared connectors when tested under specific operating conditions.

KEY WORDS	LINK A		LINK B		LINK C	
	ROLE	WT	ROLE	WT	ROLE	WT
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