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Zero Leakage Design For Ducts And Tube Connections For Deep Space Travel

TUBE CONNECTOR DESIGN PRINCIPLES AND EVALUATION

by

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FOR DEEP SPACE TRAVEL

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Final Report Prepared under Contract No. NAS 8-11523

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FOREWORD

This is Volume V of a six volume final report covering work accomplished by the Research and Development Center of the General Electric Company, Schenectady, New York from 5 July 1963 to 30 June 1967. This program was sponsored by the Missile and Space Division of the General Electric Company, Philadelphia, Pennsylvania, under National Aeronautics and Space Administration Contract NAS 8-11523, "Zero Leakage Design for Ducts and Tube Connectors for Deep Space Travel."

The six volumes contained in this final report are:

Volume I -- "Fundamental Investigations"

Volume II -- "Connector Concept Studies"

Volume III -- "Guide in Selecting Duct, Tubing and Gasketing Materials for Space Vehicles and Missiles"

Volume IV -- "New Connector Designs and Testing"

Volume V -- "Tube Connector Design Principles and Evaluation"

Volume VI -- "X-Connector Feasibility Studies"

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Section 1

INTRODUCTION

This is Volume V of a six volume report covering work accomplished during the period from July 5, 1963 to June 30, 1967, under NASA Contract NAS 8-11523, "Zero Leakage Design for Ducts and Tube Connections for Deep-space Travel."

SCOPE OF WORK

The work reported in this volume is divided into three specific subjects:

- Principles of Connector Testing
- Design Evaluation of Commercial Connectors
- Design Rules for Threaded Connectors

The work performed on this contract has involved many tests of new designs. Thus, the opportunity was available to criticize and examine customary test procedures. The conclusions and recommendations concerning those test procedures are given in Sections 2 and 3 with detailed discussion of the subject given in Sections 4 and 5.

Designers of commercial connectors are faced with decisions in a number of critical areas in order to arrive at a design (how to seal, how to fasten to the tube, how to make separable, etc.). For each critical area of the design, there are options open to the designer. Therefore, an evaluation of a number of different commercial connectors discloses the virtue of each option selected by the designers. Such an evaluation was performed under this contract.

In the course of working on this contract, rules for the design of threaded connectors were evolved. They are presented in Appendix I of this volume.

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Section 2

CONCLUSIONS

PRINCIPLES OF CONNECTOR TESTING

The customary series of testing was found to be satisfactory considering both the cost of testing and the achieved realism. Two exceptions to this were:

1. Vibration and shock tests based on "g" level alone are completely unsatisfactory. This is true whether or not the "g" level is monitored at the input to the tube-connector assembly or at the connector itself. The reason is that the stress in a vibrating or shocked connector-tube assembly is dependent on the configuration of the span as well as the "g" level. If the span is not specified then there is no point in specifying "g" level.
2. Short term leak tests can be misleading in judging the performance of certain types of connectors. If the connector has sealing surfaces in series (whether they occur deliberately or by accident) separated by large volumes, the steady-state value of the leak may not be reached until the connector has been pressurized for a very long time. Thus, a connector designed for use in a deep-space mission may begin to leak at a high rate after several weeks, even if it performed satisfactorily for several hours in the test laboratory.

DESIGN EVALUATION OF COMMERCIAL CONNECTORS

1. When a fitting is designed so that bending loads are transferred gradually (over a long axial distance) from the tube to the fitting, its performance under vibratory conditions should be better than a similar fitting with short transition. Performance will also depend on how the tube is fastened to the fitting. When annealed tubing is used, welding appears to be superior to either swaging to brazing. Hardened tubing was not tested, but welding will obviously be more difficult with hardened tubing because of the softening of tube material at the junction.
2. An alternate load path which transfers bending and tensile loads around the sealing surfaces improves sealing performance. When such an alternate load path is provided of material which is stiffer than the seal, bending and axial loads do not disturb the seal.
3. A number of the evaluated connectors carry extra weight which is necessary only because of assembly techniques. For example, some threaded connectors have wrenching flats on the union which are unstressed in operation but which are necessary in the assembly process. As another example, the swaged permanent fitting requires extra weight because swaging is done from the outside in. This requires a heavy collar which contributes little to strength but is necessary to hold this type of swage together.

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Section 3

RECOMMENDATIONS

PRINCIPLES OF CONNECTOR TESTING

Present test procedures are generally satisfactory, but should be changed at a gradual pace and eventually become more realistic. For example, future vibration tests should be performed at operating temperatures and pressures. They are now customarily performed at room temperature with no pressurization.

Two aspects of tests now performed are misleading and require immediate correction.

1. The vibration and shock performance requirements for fluid connectors should not be specified solely by the number of cycles at a certain "g" level. A much superior method of specification for vibration and shock performance is to state that the connector-tubing assembly should survive a given number of cycles of a certain level of stress, as measured in the tube near the fitting. A rule of thumb for vibration is survival of 10^6 cycles at 1/3 the yield stress of the strongest tubing to be used with the connector.

2. A connector should be tested so as to disclose the leak rate over its intended lifetime. When one connector has two or more seals in series separated by large cavities, this point takes on importance. If the connector must have low leak rate over long periods of time, two test options are available:

   - Test over a sufficient period of time to disclose the level of long-term leak.
   - Test each seal separately by drilling bypass holes at different places on different samples.

An explanation of the mechanism underlying this recommendation, and a method for estimating leak rate time is given in Appendix II.

DESIGN EVALUATION OF COMMERCIAL CONNECTORS

1. When new connector designs are undertaken, the primary engineering effort should be directed toward developing vibration resistance at the tube-fitting interface. The fitting should have a tapered transition section of generous length. Its stiffness should approximate that of the tube at the interface. The stiffness of the fitting should then increase gradually toward the center of the fitting. Welding is recommended at the tube-fitting interface for annealed tubing when it is practical to have

the necessary equipment present. Swaging is recommended at that interface when hard tubing is used, or when welding equipment cannot be made available.

2. When a deformable seal is used, an alternate load path should be provided.

3. It should be recognized in designing new connectors that extra weight incorporated for convenience of assembly serves no purpose in operation. Accordingly, the designer should select the nut-union combination only when it is necessary that the connector be separable. Likewise, the weight of flanges, clamp rings, and bolts should only be accepted when separability is required. Swaged construction offers promise where separability can be sacrificed and a semi-permanent connector can be tolerated. Sleeve-to-sleeve swages should be developed to form a lightweight design which is semi-permanent. Tube-to-sleeve swages offer lightweight design especially when they are performed by expanding the tube out into the sleeve. This type of construction should be further developed, however, to survive the vibration environment.

Section 4

PRINCIPLES OF CONNECTOR TESTING

INTRODUCTION

The proof of any design of fabrication effort is its ability to survive a realistic test program. Where test parameters already exist, the particular design must pass those tests without question. In new technologies and situations, parameters must be critically selected to impose rigid yet truly realistic test criteria.

The section covers some test philosophies resulting from critical assessment and modification of various existing procedures. They represent the kinds of tests imposed on model connectors designed on this contract and described in Volume IV of this report series.

NEW CONNECTOR DESIGNS TESTING

Several new sealing concepts were developed during this contract. They were incorporated into connector configurations and subjected to a rigorous test program.

In the early states of testing, military specifications concerned with test paraneters were consulted and relied on to give useful test criteria. The final test programs use military specifications as a basis, but where necessary, modified to obtain indicative test parameters.

Pressure Tests

The major function of any connector is to seal a fluid under pressure. The program's pressure testing followed a standard design, proof, and burst sequence. The proof and burst requirements are based on the size of connector to be tested. For sizes up to and including 1.5 inches outside diameter, the proof pressure was two times operating pressure with the burst requirement at four times operating pressure. For sizes above 1.5 inches outside diameter, the proof was taken as 1.5 times the operating pressure with burst at 2.5 times the operating pressure. Helium test gas was used in combination with mass spectrometer leak detection techniques which lend themselves to very small leak detection. Where extremely high pressures without leak detection are required (i.e. burst) the gas can be substituted by liquid to preserve safety. An outline of the test philosophy follows.

Initial pressure testing at the design pressure is accompanied by a leak check. The proof test philosophy was to monitor leakage to the design pressure level when practical.

The test criteria dictate that a connector can leak at pressures above the design pressure. However, small gross yielding can take place up to proof pressure and reduction back to design pressure should again result in a leak-tight connector.

It is necessary for burst requirements to be met without catastrophic yielding or parts destruction. The connector was required to remain integral at all times. A return to a non-leaky connector at design pressure was not a requirement.

Leak tests are not usually specified to disclose leak rates over the indicated operating life of the connector. Also, when a connector has seals in series separated by large cavities, it requires a long period of time to achieve any detectable leakage. Leakage builds up slowly because each cavity must be charged by flow through the upstream seal before any gas flows through the down stream seal. A method for calculating the leak rate in such a situation is given in Appendix II. This method can be used to extimate a proper time period for leak tests.

Vibration Tests

All vibration tests were made using a maximum bending stress level at the junction between connector and tube or duct, rather than to fix a certain "g" level of vibration application. If a "g" level is specified, the stress that results at the connector can be of any value depending on the vibration test mount. The more meaningful parameter chosen was the designation of a stress level commensurate with the yield value of the tube or duct material. A stress level of 10,000 psi or 1/3 of the yield was selected as the stress criterion. In all cases, strain gage transducers were used to monitor strain at some point on the tube or duct's surface. This could be correlated to the required stress at the noted junction. As a cross-check, vibration amplitude was monitored using a microscope. These readings transformed into a "g" level provide a cross-check on the stress. A nominal internal helium pressure of 200 psi was imposed during vibration. Leakage monitored throughout the test provided combined pressure and vibration test data.

A bellows vacuum enclosure provided the necessary vacuum for leak detector operation. It did not hinder application of the vibrational conditions nor did it lend any structural support to the connector assembly.

All vibration tests were made with the connector assembly operating at its resonant frequency. This was necessary to obtain the required stress levels on the simply supported vibration fixture. The fixture and the test procedure are similar to those described in Section 5 of this volume. Actual test fixture details are included in Volume IV of this report series.

Shock Tests

A short time shock pulse (approximately 3 milliseconds) was used for shock tests conducted on this program. Like the vibration test, a stress level of 1/3 the tube's yield stress was the test criterion, thereby avoiding the "g" level support program. Prior to all shock tests a dummy connector with the same assembly resonant frequency as the one to be tested was instrumented and used to set the shock level requirements.

Impulse Tests

Impulse tests were not made for any connector developed for this contract. Since equipment to perform impulse tests would have had to be fabricated, this expense was not justifiable for the amount of data expected. However, the effect of an impulse condition should be considered for each design.

Temperature Extreme Tests

Pressure leak tests were made of the connector assemblies at their rated pressures and at their approximate temperature extremes. Where a connector was designed for -423°F it was checked to liquid nitrogen temperature (-321°F). The added expense of testing with liquid hydrogen is avoided whenever the results at -321°F can be extrapolated to -423°F with confidence. The high temperature conditions were obtained using peripheral heaters. For all extreme temperature environments, thermocouples were used to monitor temperature as close to the connector seal area as possible.

Seal Load Removal Tests

All the model connectors employing flanges and bolts were checked under conditions of bolt load reduction. These were tests to determine the effect on the seal's integrity of reduction of applied seal stresses. They were an effort to duplicate an actual service condition whereby seal loading stresses could be reduced by the loosening of bolts due to creep, vibration, etc.

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Section 5

DESIGN EVALUATION OF COMMERCIAL CONNECTORS

INTRODUCTION

A number of commercially available connectors were purchased and tested to better understand sealing principles as directly applied to separable fluid connectors. The connectors purchased were representative of certain illustrative seal aspects. They were all subjected to common parameters of pressure and vibration to establish a base on which to evaluate each principle.

The purpose of this portion of the study was to test various principles in the most economical manner.

It was found to be less expensive, for example, to buy and test commercial connectors that are swaged to the tube than to develop a swaging technique and build special swaged connectors. Another advantage of using commercially available hardware was the fact that they have been proven and are less likely to run into difficulties not related to the point of interest.

It was not the purpose of the tests to in anyway "qualify" particular products or vendors and neither the tests conducted nor the number of items tested are appropriate for that purpose. Each of the seal principles has been separately studied qualitatively with respect to its advantages, shortcomings, and general characteristics. Given in this section are the types of connectors tested with a short evaluation of them.

CONNECTORS TESTED

The connectors tested were chosen on the basis of one or more significant aspects. A test program was carried out and performance was noted under a given set of test conditions for each.

The choice of the connector was determined by the particular areas of interest such as method of sealing, configuration and means of attachment. Commercially available units were purchased rather than to develop a number of connectors employing the aspects to be evaluated. It was definitely not an effort to evaluate the connector as a product nor to assign a figure of merit to rate one over any other. The approach was to indicate salient features of the designs and test those features accordingly as applied to the overall task of sealing fluids.

The following is a list of the connectors chosen.

1. Swaged Gasketed Ferrule (Figure 1)
2. Swaged Permanent Fitting (Figure 2)

3. Two Part Conical Ferrule Fitting (Figure 3)
4. Conical Gasket Threaded Fitting (Figure 4)
5. Conical Gasket Clamped Fitting (Figure 5)
6. Butt Seal Gasket Connector (Figure 6)
7. One Part Conical Ferrule Fitting (Figure 7)
8. Deformable Gasket Joint (Figure 8)
9. "V" Seal Joint (Figure 9)
10. Face Seal Connector (Figure 10)

All connectors were of a one inch tube size to allow representative results to be obtained. The swaged permanent fitting was not available in this size and a half-inch size was used.

Table I gives relative data concerning commercial connector weights and length of tubing required for removal in order to place each specific product in service. The weights were derived from actual measurement of the connector hardware. The lengths assume new work installations and do not consider allowances for installation tools, etc.

CONNECTOR DESCRIPTION

Figures 1 through 10 show a schematic of the connectors tested along with a description of their structure and assembly.

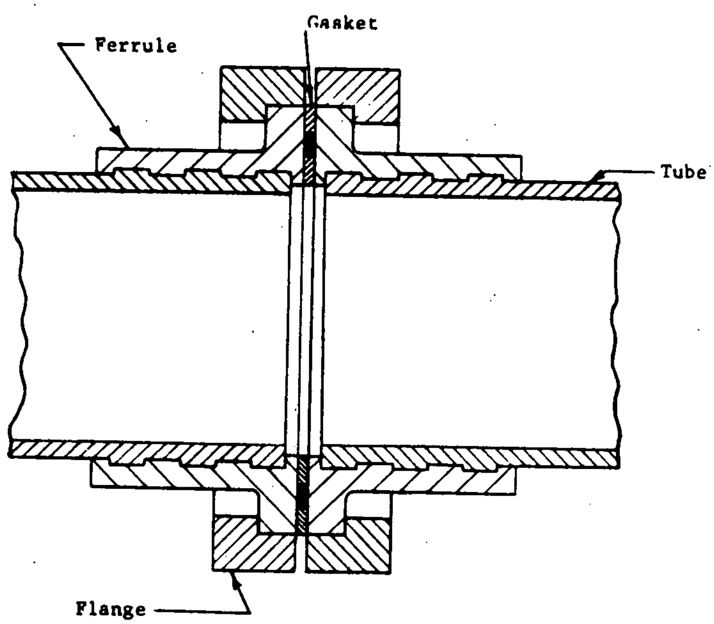


Figure 1. Swaged Gasketed Ferrule

Table I

COMMERCIAL CONNECTOR DATA

	Connector	Tube Length for Removal (inches)	Connector Weight (pounds)
1.	Swaged Gasketed Ferrule	1/8 + gasket thickness	0.3 (estimated)
2.	Swaged Permanent Fitting	1/16*	0.1521*
3.	Two Part Conical Ferrule Fitting	15/32	0.635
4.	Conical Gasket Threaded Fitting	0.72	0.531
5.	Conical Gasket Clamped Fitting	0.575	0.470
6.	Butt Seal Gasket Connector	3.80**	1.294
7.	One Part Conical Ferrule Fitting	1.00	0.818
8.	Deformable Gasket Joint	2 13/16	1.218
9.	"V" Seal Joint	3 1/4	0.836
10.	Face Seal Connector	1 3/32	0.301

* These values relate to the 1/2 inch outside diameter size used.

** This "ℓ" dimension is the total length of tube connector minus the weld preps made for attaching the tubing. The same basic fitting can be obtained with socket weld ends which will decrease this dimension.

The fitting shown in Figure 1 consists of two ferrules, two swivel flanges, a gasket, and four bolts and nuts. The ferrules are swaged to the tube ends with a special tool that fits into the tube end and rolls the tube radially outward and into the circumferential grooves in the ferrule. The ferrules are axially drawn together by the swivel flanges. Four bolts and nuts through the swivel flanges hold the connector together. The seal consists of a gasket clamped between the two mating surfaces of the ferrules.

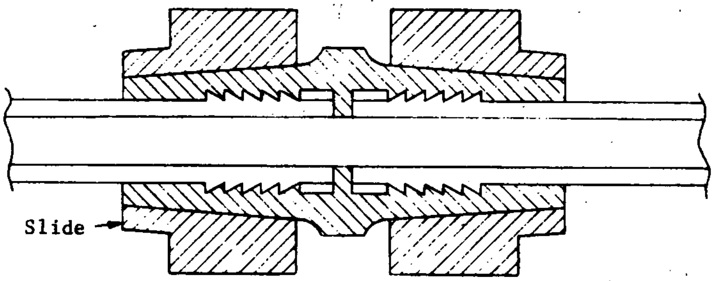


Figure 2. Swaged Permanent Fitting

The swaged permanent fitting shown in Figure 2 is a high pressure tube fitting. It consists of a basic fitting such as a union, elbow, or tee, and slides for each end. Annular serrations are machined in both ends of the inside diameter of the fitting. The outside diameters of the fitting ends are slightly tapered. Matching tapers exist on the inside diameters of two corresponding slides. When the slides are forced up the tapers by a special assembly tool, elastic deformation of the tube takes place causing the tube to be forced into serrations of the fittings. Tube connections made in this manner allow tubes of various materials and hardnessess to be used.

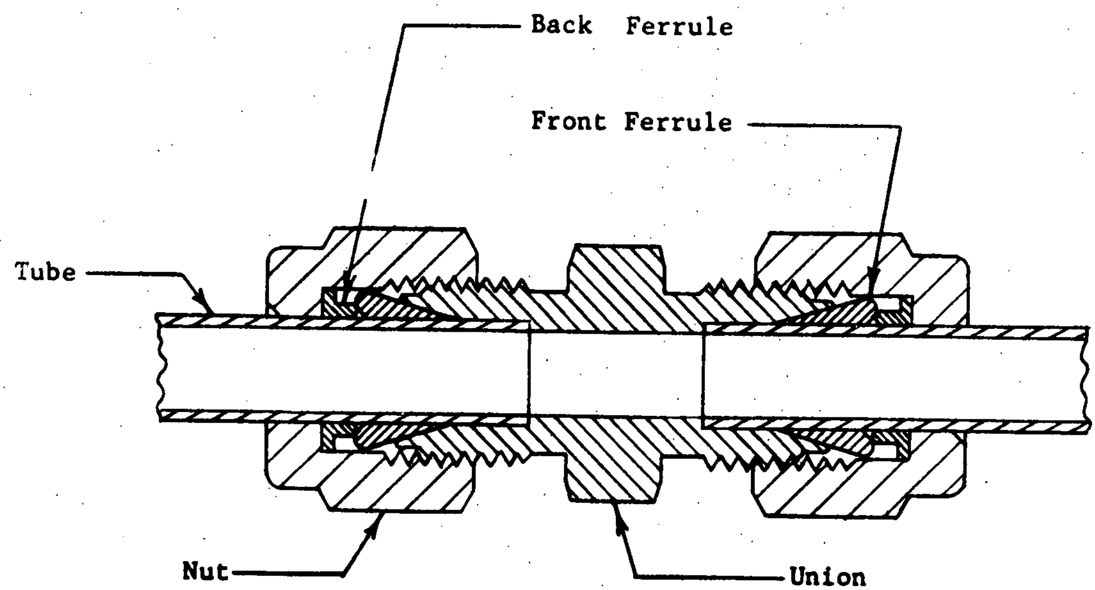


Figure 3. Two Part Conical Ferrule Fitting

The fitting in Figure 3 consists of a double ended union with a nut, front ferrule and back ferrule on each end. All of these parts are usually made of the same material. At the first assembly, the tube is slid through the ferrules into the union until it bottoms in the union. The nut is made finger tight and given another 1 1/4 turns with a wrench. This causes the ferrules to be swaged into the tube. In this way, the mechanical connection and seal are made simultaneously. The ferrules are permanently attached to the tube. The sealing interfaces are between the union and front ferrule and the front ferrule and tube. The front ferrule may be considered a type of gasket.

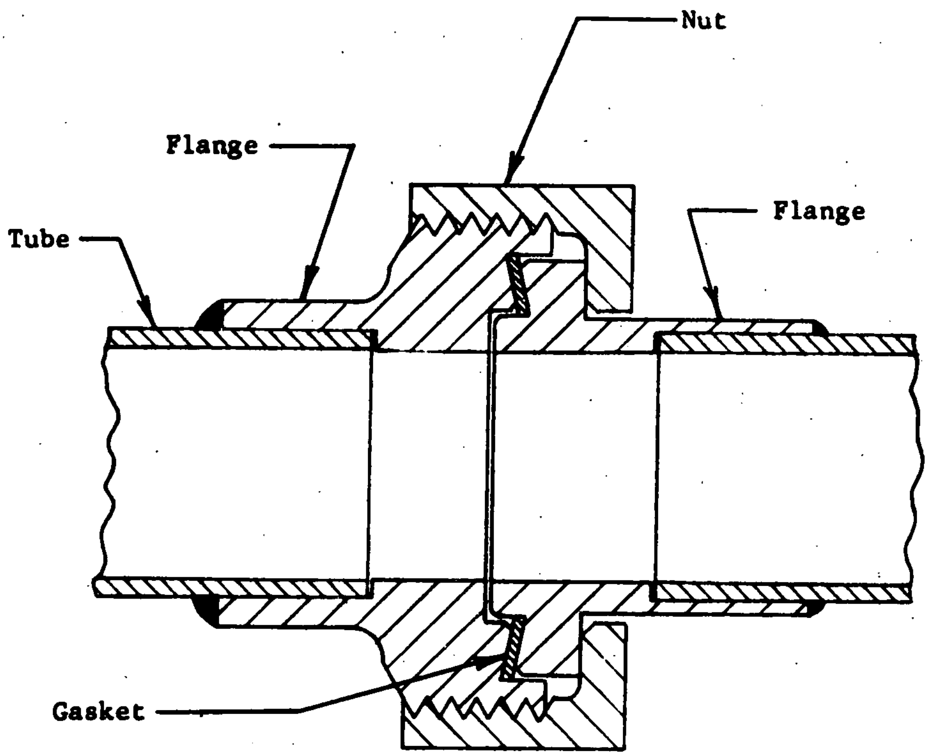


Figure 4. Conical Gasket Threaded Fitting

The fitting shown in Figure 4, for a small diameter tubing, consists of a nut, threaded female flange, gasket and male flange. The flanges are welded or brazed to the tubing. The shoulder of the male flange is engaged by the nut and as the nut is threaded on the female flange, the two flanges compress the gasket between them. The gasket and sealing faces of the flanges are conical. The initial gasket angle is greater than the angles of the mating flange faces and the slant length of the gasket is greater than that of the flange faces. As the flanges are brought together, the gasket is compressed radially and a seal is effected between the outer edge of the gasket and the female flange, and the inner edge of the gasket and the male flange. The nut is torqued until the gasket is firmly clamped between the flange faces.

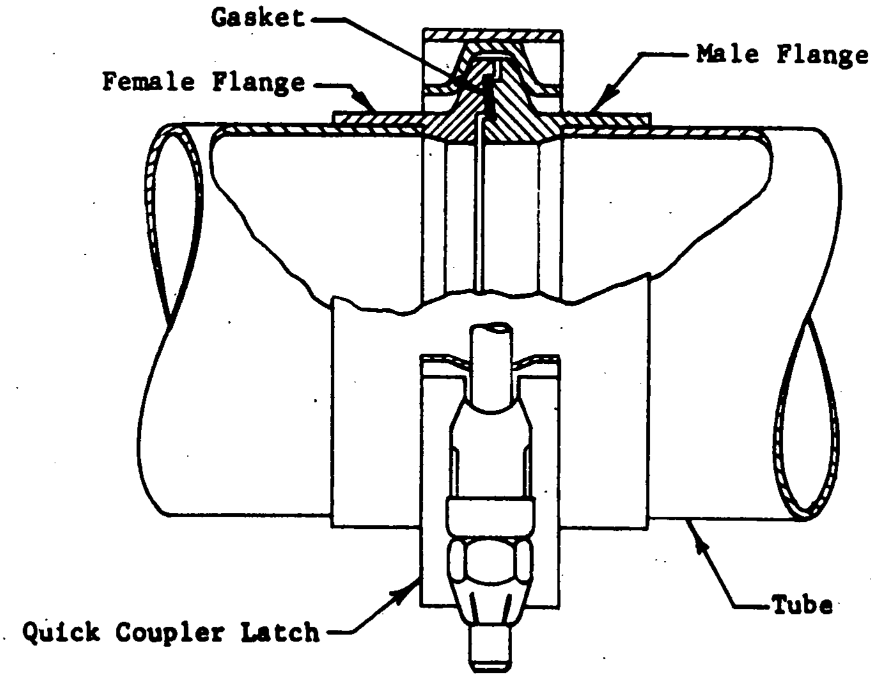


Figure 5. Conical Gasket Clamped Fitting

Figure 5 shows a connector designed for tube sizes of one to twelve inches. It consists of a quick coupler latch, female flange, male flange, and gasket. The flanges are welded or brazed to the tubing. The latch is a segmented ring that fits over the flanges. As a T-bolt on the latch is tightened, the latch length decreases circumferentially and the latch moves in radially. Due to the mating tapers on the latch and flanges, the flanges are drawn together as the latch bolt is tightened. The gasket and sealing faces of the flanges are conical. The initial gasket angle is greater than the angles of the mating flange faces and the slant length of the gasket is greater than that of the flange faces. Therefore, as the flanges are brought together by the latch, the gasket is compressed radially. In this way a seal is effected between the outer edge of the gasket and the female flange, and the inner edge of the gasket and the male flange. The latch bolt is tightened until the gasket is axially clamped between the flanges.

The collar type union shown in Figure 6, consists of a threaded hub, a plain hub, a seal ring, and a collar. The hubs are permanently attached to the tubing by welding. The seal ring is made of steel which is sometimes plated and the ring cross-section is a "T" section with the flange on the inside diameter. The seal ring web is clamped between the hubs when the collar is tightened onto the threaded hub.. The angle of the seal flange lips is slightly less than that of the mating hubs and as the seal ring is clamped between the hub, the seal flange lips are deflected by the hubs. In tnis way a seal is effected between the seal flange lips and the hubs. The seal is pressure energized in that the pressure acting on the inside of the seal increases the normal pressure between the seal flange and hubs. The seal web provides a mechanical connection between the hub and does not perform any sealing function.

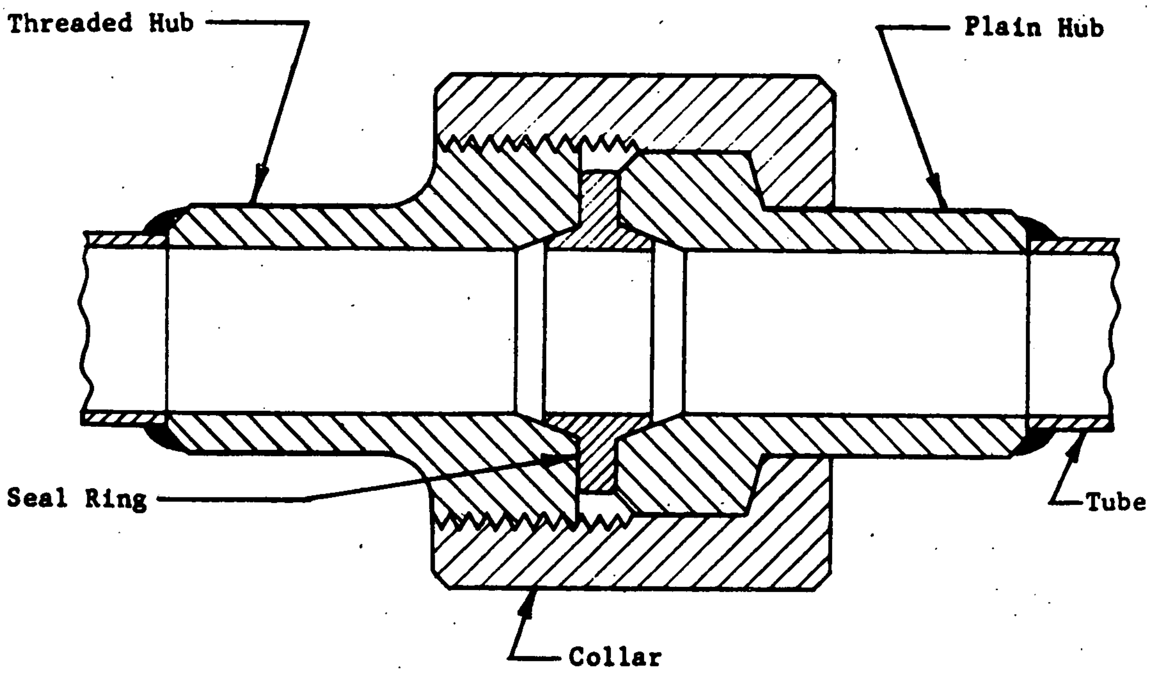


Figure 6. Butt Seal Gasket Connector

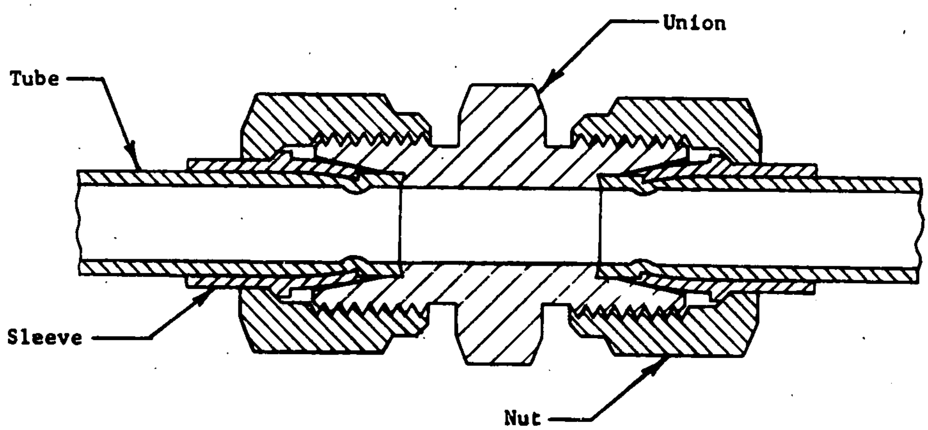


Figure 7. One Part Conical Ferrule Fitting

The flareless tube fitting (Figure 7) consists of a union, two sleeves and two nuts. The sleeves are permanently swaged into the tubing in a pre-setting operation. The connector assembly begins with the insertion of the tube into the union. Then the nut is threaded onto the union and, as the nut is tightened, the sleeve is pressed against the sealing surface of the union. There are two parallel leakage paths; one is through the swaged connection between the sleeve and tube and the other is through the interface between the sleeve and union. An added feature of this connector is that the sleeve is forced radially outward against the nut during assembly to prevent the nut from loosening during vibration.

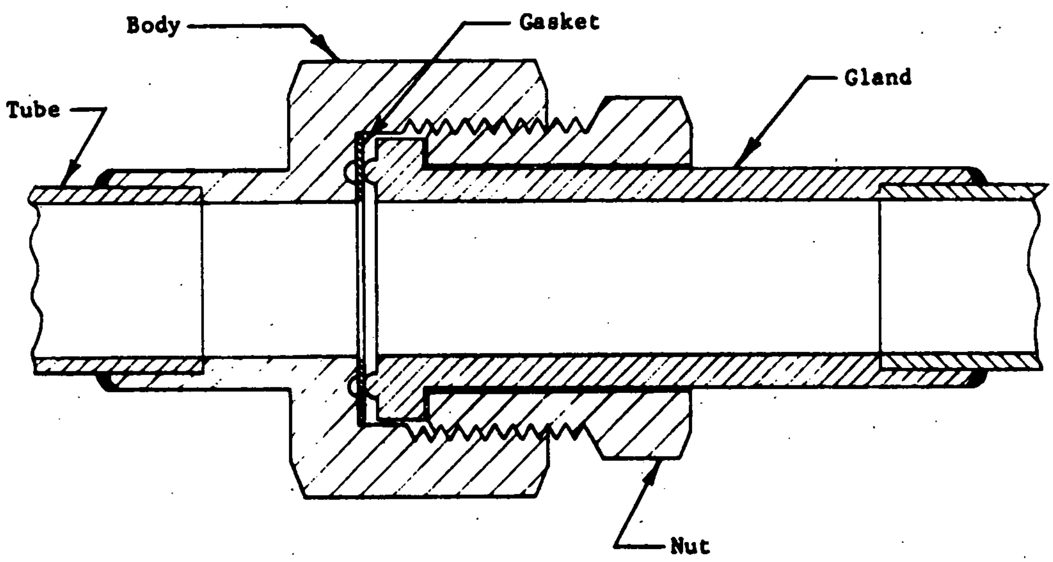


Figure 8. Deformable Gasket Joint

The coupling shown in Figure 8 although designed for ultra-high vacuum service, is suitable for high pressure service. It consists of a body, gland, nut, and gasket. The body and gland have tube socket weld ends for welding the connector to the tubing. The nut has an external thread which mates with the internal thread of the body. The gland has a flange on the end mating with the body. The nut slides over the gland and contacts the backside of the flange. When the nut is tightened into the body, the gasket is axially compressed between the body and gland. The gland has a raised surface that forces the flat gasket into a groove in the body.

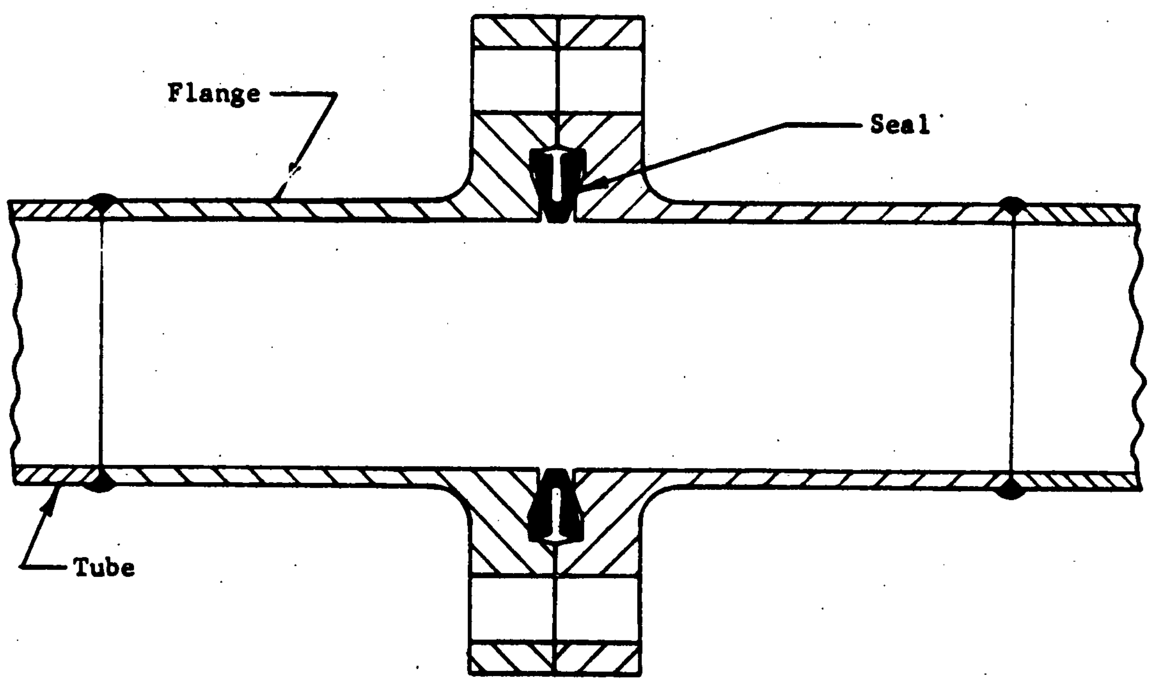


Figure 9. "V"Seal Joint

The connector in Figure 9 consists of two flanges, a seal, and a number of bolts and nuts. The flanges are permanently welded to the tubing. The connector is assembled by placing the gasket in the recess cut in each flange face and torquing the bolts until the flange faces are in direct contact. The seal has a "V" shaped cross-section with the "V" facing radially outward. The seal is made of steel and coated with a soft material. As the flanges are brought together, the edges of the seal are forced into the corners of the flange face recess and a seal is effected.

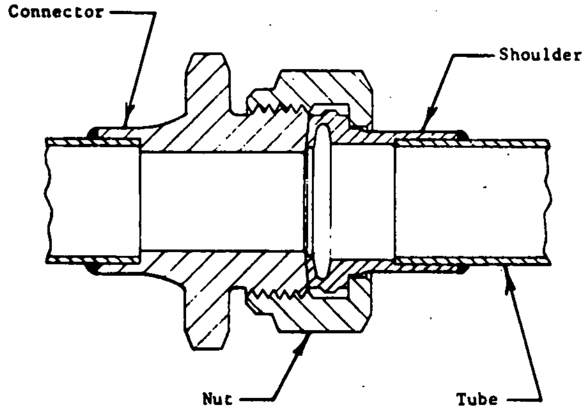


Figure 10. Face Seal Connector

This connector (Figure 10) consists of a shoulder, nut, and connector. The shoulder and connector are welded to the tubing. The bearing surface of the nut engages the shoulder and axially brings it into contact with the connector as the nut is threaded onto the connector. The forward lip of the shoulder is pressed against the connector to effect the seal. The mating surfaces on the shoulder and connector are sometimes plated with a metal which has a lower yield strength than the base metal. A load path separate from the seal is provided in the shoulder.

TEST RESULTS

Table II is a summary of the test induced on the various connector assemblies, but does not include results of temperature extreme tests imposed on the swaged gasketed ferrules, although results are mentioned in the following paragraphs. Appendix III gives a general description of tests and a detailed description of assembly procedures. Each assembly was attached by manufacturer's recommended means to respective lengths of tubing. The tubing served two purposes:

- Provided a means for internally pressurizing the connector
- Served to support it during vibration testing in a beam fashion

In all but one case, the tube used was type 316 stainless steel, 1 inch outside diameter by 0.065 inch wall thickness. The swaged permanent fitting was tested using 1/2 inch outside diameter by 0.065 inch wall tubing.

For pressure tests of all the fittings, the means of attachment to the tube is tested as are the connector sealing characteristics. The pressure test fixture is such that its vacuum seal is out-board of the weld, brazes or mechanical attachments. In this manner of testing, failure of the attachment is judged a connector failure. Leak test of the tube welds or brazes prior to connector testing assures an initially sound joint.

In all cases where vibration failures were noted the failure was observed by a drop in resonant frequency of the assembly. In some cases a gradual decrease was noted, in others, an immediate change was observed. Close surveillance was necessary to preclude accessory equipment damage when the change occurred.

Swaged Gasketed Ferrule

Two separate assemblies were made and checked. One was made with the tube stop machined out to eliminate a possible second seal. Leakage testing of each ferrule is noted because each was checked separately by use of the modified vacuum chamber test fixture. The first unit was vibration tested in two stages. An initial test was made with a bending stress of 8,900 psi at the junction between tube and connector. The remainder of the test carried the total cycles to one million with the bending stress at the 10,000 psi level. The second ferrule unit was tested in the approximate same manner but with 10,000 psi stress for both stages of vibration. Vibration failure was noted by a decrease in assembly resonant frequency.

Leakage was observed for all four ferrule connections after vibration testing as observed by the results in Table II. In addition, two separate assemblies were made and tested at temperature extremes of $+700^{\circ}$ F. with 1000 psi internal pressure. The fitting parts and tubing were all of stainless steel material. Neither assembly leaked during the test or at each equilibrium temperature.

Swaged Permanent Fitting

The one assembly tested was completely successful.

Two Part Conical Ferrule Fitting

It was decided to vibration test both assemblies even though leakage was noted at assembly. In the initial leakage tests, both connectors' rate of leakage appeared to be independent of change in applied pressure, and in general, did not reach an equilibrium leakage condition at each pressure level attained. As a result, a specified amount of time at each pressure increment was allowed before a leakage reading was taken. For the first assembly, whose maximum leakage was approximately 4.3×10^{-3} atm cc/sec before vibration test, the time allowance was three (3) mintues at 50 psi increments, and five (5) minutes at 100 psi increments after vibration test. Leakage of this assembly attained a maximum of approximately 3.5×10^{-3} atm cc/sec at 1000 psig after vibration at the assembly resonant frequency for one million cycles. This leakage was approximately what it was before vibration.

For the second assembly, the time allowed between readings was five minutes at 50 psi increments before vibration. Leakage of this connector at 1000 psig before vibration was about 3.36×10^{-4} atm cc/sec. After a pressure checking of this assembly, residual helium became trapped in its parts and would not be pumped out. The physical characteristics of this fitting are such that it is possible to trap helium between the tube and its component sleeves. A vibration test was carried out for comparative purposes with this condition in existence.

After vibration test at 10^6 cycles on this second assembly, a leak check could not be performed because the necessary vacuum could not be obtained. Residual helium was still observed to be present and an unsuccessful attempt was made to flush it out with nitrogen gas at 30 psig pressure for 1-1/3 hours. Finally, the connector was disassembled. The now loose sleeves and tubing were flushed with air and the connector reassembled. The nuts were torqued to 200 pound-feet, which positioned the nuts about 1/6 turn beyond their position before disassembly. A leak test was again made but a vacuum could not be obtained. The test was suspended.

A third assembly was made using new front and back ferrules and new lengths of tubing but using the nuts and body from assembly number two. The tubing was polished in the area of the sealing surfaces to remove any longitudinal scratches that would hamper sealing. The previous tests did not employ tubing with prepared surfaces. The outside diameter of the tube for this test varied from 1.00 to 0.995 inch. One-third of a turn was required to snug-up on the tube from the initial fingertight position. The nut was turned another 1-1/4 turns to effect the seal. Subsequent pressure test showed a maximum leakage of approximately 5.2×10^{-3} atm cc/sec at 1000 psig. An incremental change in test pressures resulted in a lagging leakage rate change or practically no leakage change in some cases. As a result, a time increment of three minutes between readings was established to allow reaching steady-state condition.

A pressure of 1000 psig was left on the connector for a period of one hour and 47 minutes to check on the time variability of leakage. At the start of the period, leakage was about 4.8×10^{-3} atm cc/sec. At the end of the period, leakage increased to 5.2×10^{-3} atm cc/sec. This is a change of about 10 percent indicating that the time base chosen was acceptable.

Conical Gasket Threaded Fitting

Disassembly of the initial leaky assembly resulted in discovery of a nicked gasket. Reassembly with a new gasket showed less leakage. A vibration test on the second gasketed joint was not made.

Conical Gasket Clamped Fitting

All tests were met and passed successfully.

Butt Seal Gasket Connector

The first assembly was made with the parts as received from the manufacturer. The gaskets and seal surfaces were observed to be nicked and scratched. The second assembly employed polished seal surfaces. However, both assemblies showed leakage as noted in Table II.

One Part Conical Ferrule Fitting

The one assembly successfully survived all imposed tests with no leakage.

Deformable Gasket Joint

Both assemblies tested met with failure during the vibration test at the accumulated cycle levels as shown in Table II. Each failure was observed as a decrease in the assembly's resonant frequency. Post-inspection showed failure was due to excessive gasket deformation.

"V" Seal Joint

The one assembly met all tests successfully.

Face Seal Connector

The lubricant coated seal surface fitting was assembled and vibration tested. After running about 10 seconds, the assembly resonant frequency changed from 283 cps to 273 cps. The test was stopped and visual inspection showed a cracked braze joint at one end of the fitting. With a hand effort, the tubing was pulled out of the fitting. Visual inspection showed the surface that was "wetted" by the braze was only about two-thirds of the total joint area, resulting in a structurally poor joint.

The ends of the tube lengths used with the fitting were machined down to 0.095 inch outside diameter for distance of 3/8 - inch to allow brazing. Normally, the tubing is "sized" to the proper diameter keeping the approximate wall thickness constant. The bending stress level adjacent to the brazed joint was set at 10,000 psi during vibration. In this case a somewhat higher stress level existed there since the tube diameter was smaller within the joint. No further tests were made.

Table II

SUMMARY OF VIBRATION TEST RESULTS

Connector	Pre-vibration Pressure (psi)	Leak Test Leak* (atm-cc/sec)	Post-vibration** Pressure (psi)	Leak Test Leak* (atm-cc/sec)	Comments
1. Swaged Gasketed Ferrule					
Assembly #1					
Ferrule #1	1000	0	1000	1.7×10^{-2}	
Ferrule #2	1000	0	1000	10^{-1} (Est.)	
Assembly #2					
Ferrule #3	1000	0	--	--	Failed at 254,000 cycles.
Ferrule #4	1000	0	--	--	Vacuum not obtainable.
2. Swaged Permanent Fitting					
Assembly #1	2000	0	2000	0	
Assembly #2	2000	0	--	--	Assembly #2 not tested.
3. Two Part Conical Ferrule Fitting					
Assembly #1	1000	4.3×10^{-3}	1000	--	Vacuum not obtainable
Assembly #2	1000	3.4×10^{-3}	1000	3.5×10^{-3}	for Assembly #1.
Assembly #3	1000	5.2×10^{-3}	--	--	Assembly #3 not tested.
4. Conical Gasket Threaded Fitting					
Initial Assembly	650	10^{-3}	700	10^{-3}	No test on second assembly.
Second Assembly, New Gasket	1000	0	--	--	
5. Conical Gasket Clamped Fitting	1000	0	1000	0	
6. Butt Seal Gasket Connector					
Initial Assembly	50	10^{-2}	--	--	
Second Assembly, Polished Surfaces	1000	10^{-2}	1000	10^{-2}	Initial assembly not tested.
7. One Part Conical Ferrule Fitting					
Assembly #1	1000	0	1000	0	Second assembly not
Assembly #2	--	--	--	--	tested.
8. Deformable Gasket Joint					
Assembly #1	1000	0	--	--	First assembly failed at 178,000 cycles, second
Assembly #2	1000	4×10^{-7}	--	--	at 590,000 cycles. Vacuum not obtainable in either case.
9. "V" Seal Joint					
Assembly #1	1000	0	1000	0	Second assembly not
Assembly #2	--	--	--	--	tested.
10. Face Seal Connector					
Dry Lubricant Coated	1000	1.9×10^{-6}	--	--	Coated assembly failed after 2800 cycles.
Polished	--	--	--	--	Vacuum not obtainable thereafter. Polished assembly not tested.

* Zero leak defined as less than 10^{-7} atm-cc/sec

** All tests of 10^6 vibration cycles unless noted

APPENDIX I

Appendix I

DESIGN RULES FOR THREADED CONNECTORS

INTRODUCTION

The design rules presented here are based on analytical and/or experimental studies. A good basis for any design can be made by consideration of each rule and how it applies to a particular situation. All designs cannot possibly possess all points enumerated, and it then becomes a choice between points of interest. It is the aim of this section to provide an aid in making proper design choices and to strengthen sound judgment concerning choices and/or evaluations.

DESIGN RULES

Type of Connector

1. A threaded connector should be used on small tubing (approximately one inch or less) and also in cases where the assembly torque is not excessive (approximately 2000 pound-inches or less.)
2. A bolted flanged connection should be used where a threaded connector is not applicable.

Configuration of Connector

1. Connector disassembly should be accomplished without interference to other parts of the piping system.
2. Separate the seal, structure and attachment to the tube.
3. Use a threaded nut mating with a threaded stub end (union) and a flanged tubular section for the mechanical connection.
4. Place the seal between the union and flanged section.

Seal

1. In most designs considered, plastic flow of one of the materials at the seal interface is necessary for sealing.
2. The plastically deformed material at the seal interface should be a gasket that is used only once.
3. Plastic flow should be induced by a combination of normal and shearing stresses.
4. The seal should be isolated from load variations and any relative motion of the connector elements.

5. Radial seals are less sensitive to load and deflection changes of the structure than axial seals.
6. The sealing surfaces should have a fine machined finish. The direction of tool marks should be normal to the potential leakage paths for ease of establishing a seal.
7. Elastomeric gaskets are suitable for low pressure room temperature applications where they are compatible with the contained fluid.
8. Metal gaskets are suitable for extremes in temperature and high pressures. Materials must be compatible with the system fluid.
9. Pressure energization of the gasket is desirable at high pressures.
10. Radiation effects should be considered.

<u>Connector Structure</u>

1. Yielding of the structure should not be a requisite for obtaining a seal.
2. Structure should support the gasket when internal pressures are large.
3. Keep load path separate from seal.
4. Design for reasonable external forces and moments and specify maximum values to be used in the design of the piping system.
5. Design for thermal transients and steady-state conditions.
6. Design for transverse and torsional vibration.
7. Design for creep and stress relaxation.
8. The structure should protect the seal from variations in load or relative movements of parts.
9. With the seal located in the compression member of the connector, it is advisable to make the tension member (the nut) stiffer than the compression member. Thus, the major part of an axial load is taken by the tension member and the effect on the seal is minimized.
10. Buttress threads should be considered to reduce the radial component of the axial force on the nut and allow a lighter nut design.
11. Materials must be compatible with fluid contained and environment.

<u>Attachment to Tubing</u>

1. Attachment should be a permanent joint.
2. Procedure should be capable of standardization.
3. Strength of joint should be equal to that of tube.

4. For long fatigue life, the simplest and most direct method of attachment is welding or brazing. With annealed tubing welding or brazing is reliable. However, with hardened tubing a swaged connection may be more compatible with rule 3.

Testing of Connector

1. The testing should include all of the most severe operating conditions.
2. Leakage testing should be done with a sensitive and accurate leak detector.
3. The testing should simulate the cyclical operating conditions encountered during the life of the connector, with careful attention given to the compromise between cost and realism.

APPENDIX II

APPENDIX II

LEAK DETECTION INVOLVING VOIDS AND LEAKAGE RESISTANCES IN SERIES

by

L. G. Gitzendanner

INTRODUCTION

It has long been recognized that whenever it is necessary to leak test a device it is undesirable to permit a design which will or may develop voids in series with possible leak paths. For example, if two heavy plates are to be welded together it is best to make either a full penetration weld or a partial penetration weld from only one side. It is undesirable to allow partial penetration welds from opposite sides since a void can be left between the welds. It would then be difficult to check and assure that there is not a leak through both welds since the speed of response of the leaking system may be very slow. Nevertheless, there are some systems where, for other reasons, it is necessary or desirable to use a system that creates voids in series with the leakage path.

It is the object of this report to examine the transient leakage versus time response of a system which has a redundancy of seals with voids between them, all in series along a potential leak path. It is assumed that the leak is to be determined by means of a mass spectrometer helium leak detector. The effect of changing some parameters is also investigated. Specifically, the system considered is a fitting connected to a tube via swaging. It was concluded that a better connection would result if the inside of the fitting were relieved in a manner to leave a series of circular lands which would grip the tube. The space between the lands generates an unavoidable void, even though undesirable from a leak detection viewpoint.

ANALYSIS

The diagram for the fluid flow path and its equivalent electrical circuit are shown in Figure 1 and Figure 2. The electrical circuit can readily be set up with element values proportional to those of the fluid circuit elements and with a scaling factor selected to yield a convenient time scale for recording. The response may then be conveniently recorded. The response of the fluid system will be the same except for the scaling factor on time and a conversion factor between current and fluid flow.

It should be noted that the circuit shown in Figure 2 is equivalent to that in Figure 1 only for the condition of molecular flow. Examination of the leakage rate as a function of helium pressure on a number of test specimens which had leaks, leads to the conclusion that for practical systems (or components of systems) a leak will usually be molecular flow if it is 10^{-6} atm cc/sec, or less. It may be molecular flow at one or two orders of magnitude higher leak rate, but could also be laminar flow or, more likely, in a state of transition between molecular and laminar. If one makes predictions based on assumed molecular flow and in fact is dealing with a flow rate which is in transition or laminar flow, the predictions will be conservative in the sense that the length of time to detect a given leak rate will be shorter than predicted. The electrical analogy is probably sufficiently accurate for leaks up to 10^{-4} atm cc/sec.

One may wonder as to the effect of atmospheric or other gases trapped in the voids on the rate of flow of helium during the leak detection process. It is fortunate that in molecular flow the flow of a given gas is determined by its partial pressure difference across a given restrictor. Thus, the presence or absence of other gases is of no concern and for simplicity in thinking we can assume that the helium used for leak detection is the only gas present.

For the particular system considered, it was desirable to use a design that resulted in the void volumes increasing linearly.

Thus, if there were ten volumes, they would have the proportions 1, 2, 3, 4 . . . . . . 9, 10. It was desired to see how leak detection time might be influenced if the number of voids were reduced in number, but with the linear relationship maintained and the size of the largest void maintained. Thus, on the same scale, void sizes for four voids instead of ten would be 2.5, 5, 7.5 and 10.

In the electrical equivalent circuit the largest capacitor, which corresponds to the largest void, was taken as 10 microfarads.

The sizing of the resistances is considered next. The sum of the flow resistances in series can be determined for any assumed leak rate and helium supply pressure. However, there is no way of knowing ahead of time how the resistance will be distributed. It could be essentially all at one place (across one land) with the remaining restrictions effectively "wide open." It could be across several lands. It could be across all lands. The "worst" distribution is a function of ones definition of "worst." A uniform distribution of resistance among the lands has some physical justification, is approximately a "worst" case, and was assumed for purposes of analysis.

In the electrical equivalent circuit resistances were all made 0.1 megohm. Coupled with the capacitance values noted, this value of resistance results in a convenient time scale for recording.

Figure 3 is a table of the capacitance values used for n = 1, 2, 4, 6, 8, and 10. In all cases all resistances were 0.1 megohm. Figures 4-9 are traces of output voltage versus time for a step input of voltage. The time of the step is shown on the second trace and for convenience in reading the chart has been transferred to the output channel trace. A timing mark also recorded is shown in Figures 4 and 5 only, but the location of points at exactly five second intervals from the point of application of voltage are noted on all figures. Note that each major chart division is approximately one second.

The vertical scales for Figures 4-9 have each been adjusted so steady state deflection is equal to full scale.

The problem of determining the appropriate time scale and leakage scale for the response of a fluid system is considered next. In a typical problem we may have a permissible leakage rate and a sensitivity to detect a somewhat smaller rate. We would like to know how long we must test to assure that if there is a leak at all, we will have detected it or it is less than the permissible rate.

Specifically, we want assurance that we have not simply missed it by an inadequately short test.

To arrive at a time scale factor for the fluid flow problem, we adapt the following units and introduce the concept of a time constant:

Electrical System			Fluid System		
Term	Symbol	Units	Term	Symbol	Units
Current	i	Amperes	Flow Rate	Q	atm cc/sec
Potential	e	Volts	Pressure	P	$\#/in^2$
Capacitance	C	Farads	Void Capacity	C	atm cc/$\#/in^2$
Resistance	R	Ohms	Resistance	R	$(\#/in^2)/(\text{atm cc/sec})$
Time	t	Seconds	Time	t	Seconds

For the electrical system with n = 1 (that is, a circuit having one capacitance and two resistances) it may readily be shown that the time constant is $\tau_E = \frac{R_1 R_2}{R_1+R_2} C_1 = \frac{1}{2} R_1 C_1$ if $R_2 = R_1$. For all R's = .1 megohm and $C_1 = 10$ microfarads, as noted in Figure 3, we find $\tau = 0.5$ seconds, that is, in 0.5 second we should get to $(1 - \frac{1}{\epsilon})$ or 63% of full scale. Indeed this is the case and Figure 4-9 are therefore in real time, as we expect they must be.

For the equivalent fluid system the time constant, τ_F, is $\tau_F = \frac{1}{2} R_1 C_1$. The ratio of τ_F / τ_E is the scaling factor on time we should apply. For example, if a steady state leak is 10^{-6} atm cc/sec with 1000 $\#/in^2$ pressure differential,

then $R_{total} = 2R_1 = 10^9 \ (\#/in^2) \big/ (\text{atm cc/sec})$

and $R_1 = 5 \cdot 10^8 \ (\#/in^2) \big/ (\text{atm cc/sec})$

If we have a void volume of 0.1 cc, then

$$C_1 = \frac{0.1 \text{ cc}}{14.7 \ (\#/in^2) \big/ \text{atm}} = 0.0068 \text{ atm cc} \big/ \#/in^2$$

$$\tau_F = \frac{1}{2} \ 5 \cdot 10^8 \times 0.0068 = 1.7 \times 10^6 \text{ seconds}$$

and $$\frac{\tau_F}{\tau_E} = \frac{1.7 \times 10^6}{0.5} = 3.4 \times 10^6$$

Thus, to reach 63% of steady state in the fluid system would take $3.4 \times 10^6 \times 0.5$ seconds, or 1.7×10^6 seconds. To reach 10% would take $3.4 \times 10^6 \times 0.053$ seconds or 1.8×10^5 seconds.

The configuration of specific interest is for n = 10 rather than for n = 1. Examination of the chart, Figure 9, shows approximately 3.6 seconds to get to 10% of steady state. It must be noted, however, that, for the same 1000 $\#/in^2$ pressure differential, to make steady state leakage equal to 10^{-6} atm cc/sec we must reduce R_1 to

$$R_1 = \frac{1}{n+1} \cdot 10^9 = \frac{1}{11} \cdot 10^9 = 9.09 \cdot 10^7 \; (\#/in^2) / (\text{atm cc/sec})$$

Thus, τ_F is changed to

$$\tau_F = \frac{1}{2} \times 9.09 \times 10^7 \times 0.0068 = 3.1 \times 10^5 \text{ sec}$$

and our scaling factor on time becomes

$$\frac{\tau_F}{\tau_E} = \frac{3.1 \times 10^5}{.5} = 6.2 \times 10^5$$

The time required to reach 10% for a 10^{-6} atm cc/sec leak across 11 barriers and 10 voids, as noted, would thus be

$$t = 3.6 \text{ seconds} \times 6.2 \times 10^5 = 2.23 \cdot 10^6 \text{ seconds.}$$

Figure 10 tabulates the time in seconds to reach 1%, 10% and 63% of steady state for various values of n and an assumed steady state leak of 10^{-6} atm cc/sec in all cases. The value for $t_{1\%}$

n	$t_{1\%}$	$t_{10\%}$	$t_{63\%}$
1	$1.8 \cdot 10^4$	$1.8 \cdot 10^5$	$1.7 \cdot 10^6$
2	---	$6.8 \cdot 10^5$	$2.38 \cdot 10^6$
4	$4.05 \cdot 10^5$	$1.22 \cdot 10^6$	$3.65 \cdot 10^6$
6	$4.85 \cdot 10^5$	$1.36 \cdot 10^6$	$3.97 \cdot 10^6$
8	$9.08 \cdot 10^5$	$1.66 \cdot 10^6$	$5.0 \cdot 10^6$
10	$1.36 \cdot 10^6$	$2.23 \cdot 10^6$	$6.5 \cdot 10^6$

FIGURE 10

and n = 1 is calculated since it represents a time too short to be read from the chart with any reasonable accuracy, all other values are based on the charts of Figures 4-9.

Examination of Figure 10 will show that as n increases the time to reach 63% of steady state changes by an amount which could largely be attributed to the increased void volume. However, the time to reach 10%, or 1% of steady state changes much more rapidly and shows not only the effect of the void volume increasing as n increases, but of the series arrangement of flow resistances and voids.

If we assume a geometry represented by n = 10 and assume a leak detection sensitivity of 10^{-7} atm cc/sec, we can determine the time required to detect leaks of various magnitudes. These are tabulated in Figure 11.

Figure 11

Steady State Leak Rate in atm cc/sec	Time to Detect Presence of Leak, Seconds
10^{-7}	infinite
10^{-6}	2.23×10^{6}
10^{-5}	1.36×10^{5}
10^{-4}	$\sim 1 \times 10^{4}$ (estimated)

It is perhaps useful to estimate the sensitivity required of a leak detector to detect a leak of a given size within a given length of time. To do this we will restrict ourselves to times which are short enough that we can use an approximate analysis, valid for short times only, without too great a loss of accuracy. Specifically, we solve for the situation where time is less than the smallest time constant, R_xC_x, in which case we can neglect back pressure and compute the flow through any restrictor as simply being proportional to the high side pressure divided by flow resistance. Solving for instantaneous flow in terms of steady state flow one finds

$$q_n = \frac{P}{R_n}$$

$$P_n = \int_o^t \frac{1}{C_n} q_n \, d\,t = \frac{P}{R_n C_n} \quad t$$

$$q_{n-1} = \frac{P_n}{R_{n-1}} = \frac{P}{R_n C_n \cdot R_{n-1} C_{n-1}} \quad t$$

$$P_{n-1} = \int_o^t \frac{1}{C_{n-1}} \quad q_{n-1} \, d\,t = \frac{P}{R_n C_n \cdot R_{n-1} C_{n-1}} \cdot \frac{t^2}{2!}$$

etc., or in general

$$P_1 = \frac{P}{R_1 C_1 \cdot R_2 C_2 \; . \; . \; . \; . \; R_n C_n} \quad \frac{t^n}{n!}$$

and $$q_o = \frac{P_1}{R_o} = \frac{P \; t^n}{R_o n! \; R_1 C_1 \cdot R_2 C_2 \; . \; . \; . \; . \; R_n C_n}$$

Q, the steady state value of q_o, is of course given by

$$Q = \frac{P}{R_o + R_1 + \; . \; . \; . \; . \; + R_n}$$

For the case where all R's are equal and the C's vary linearly (i.e. $C_{n-1} = 2C_n$, $C_{n-2} = 3C_n$, etc.) we find that

$$q_o = Q \frac{1}{(n+1)! \;\; n!} \cdot \left(\frac{t}{R_n C_n} \right)^n \qquad \text{or}$$

$$q_o = Q \frac{1}{(n+1)! \;\; n!} \cdot \left(\frac{n \; t}{R_1 C_1} \right)^n$$

Because of our assumptions, we can only use this solution for $t < R_n C_n$

Consider the case where $n = 10$, $C_{10} = 0.00068$ (atm cc)/(#/in^2) and $R_{10} = 9.09 \times 10^5$ (#/in^2)/(atm cc/sec). The steady state leak rate for 1000 #/in^2 pressure difference would then be 10^{-4} atm cc/sec. The length of time over which the short time solution would be valid would be

$$t < R_{10}\, C_{10} = 9.09 \times 10^5 \cdot 0.00068 \text{ seconds}$$

or $t \simeq$ ten minutes

In this length of time the output flow, and hence the sensitivity required to detect a steady state leak of 10^{-4} atm cc/sec, would be:

$$q_o = \frac{10^{-4} \text{ (atm cc/sec)}}{11! \cdot 10!} \cdot \left(\frac{620}{620}\right)^{10} = 7 \cdot 10^{-19}$$

Quite obviously the leak would not be detected, and even if we increased time by a factor of 10 we may judge that the leak may not be detected. The latter conclusion is reached since the short time formula, because it neglects back pressure effects, would give too high a flow; but it would still indicate flow around the limit of flow detectable on the spectrometer leak detector.

If we modified our input assumptions such that C_1 remains 0.0068 (atm cc)/(#/in^2) and the steady state leak remains 10^{-4} atm cc/sec, then for various values of n we find the following sensitivities required to detect a leak in one hour.

n	Sensitivity Required (atm cc/second)
1	2.6×10^{-6}
2	3.8×10^{-7}
3	7.5×10^{-8}
4	1.8×10^{-8}
5	4.6×10^{-9}

Values are not tabulated for n over 5 since one hour would be greater than the shortest time constant and the simplified formula developed for short times would not be valid. The sensitivity required to detect any smaller steady state leak in one hour will be that noted above multiplied by the ratio of leakages raised to the nth power. Thus, to detect a 10^{-5} atm cc/sec leak with n = 3 would require $7.5 \times 10^{-8} \times \left(\frac{10^{-5}}{10^{-4}}\right)^3$ or 7.5×10^{-11} sensitivity. It is not suggested one attempt to estimate times to detect larger leaks since larger leaks would almost certainly not be molecular flow and the calculations would be in error.

CONCLUSIONS

1. The length of time required to detect a leak is very likely to be excessive if the system has restrictions and voids in series along the leak path, unless the sensitivity of the mass spectrometer used to detect it is many orders of magnitude greater than the leak.
2. Increasing the number of lands and void volumes in series increases the time to detect a steady state leak of a given magnitude. The factor of increase in time is approximately proportional to the increase in total void volume if the steady state leak is roughly twice the threshhold of detection. Thus, for small leaks (near the threshhold of detection) the penalty, in terms of lengthened leak test time, introduced by using a larger number of lands and voids is not severe. For larger leaks the penalty is greater, but since the test must be run for a time dictated by the smallest leak we wish to detect, the increased time associated with larger leaks is of no practical concern.

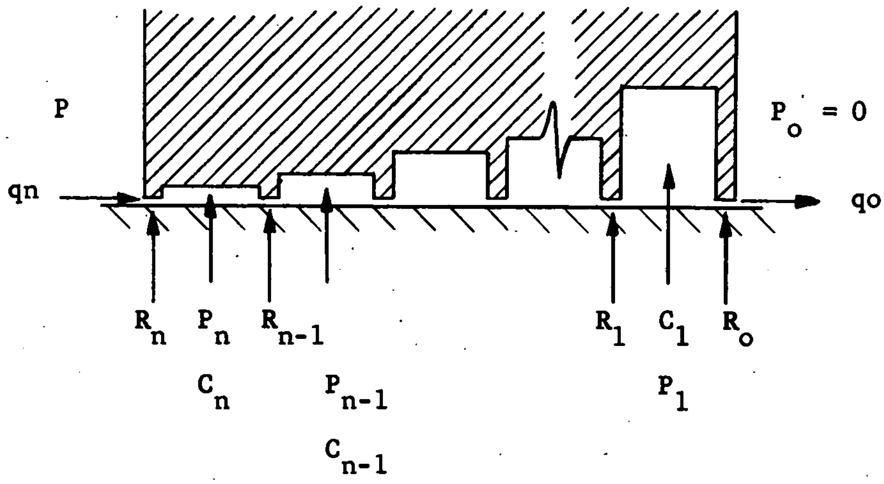


FIGURE 1

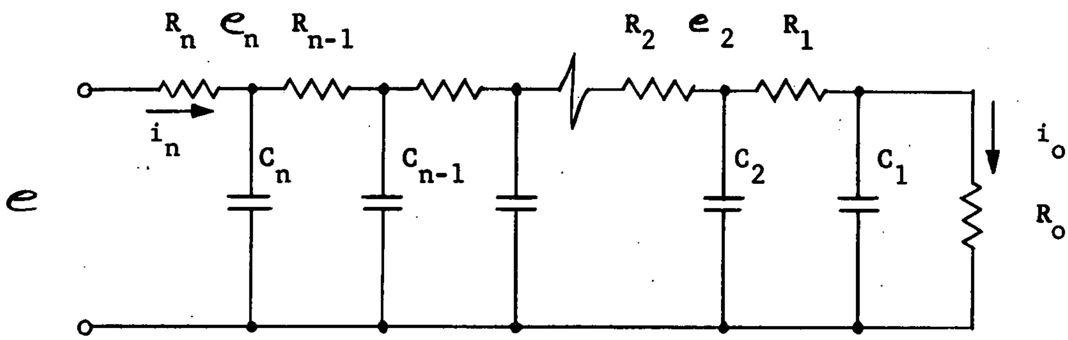
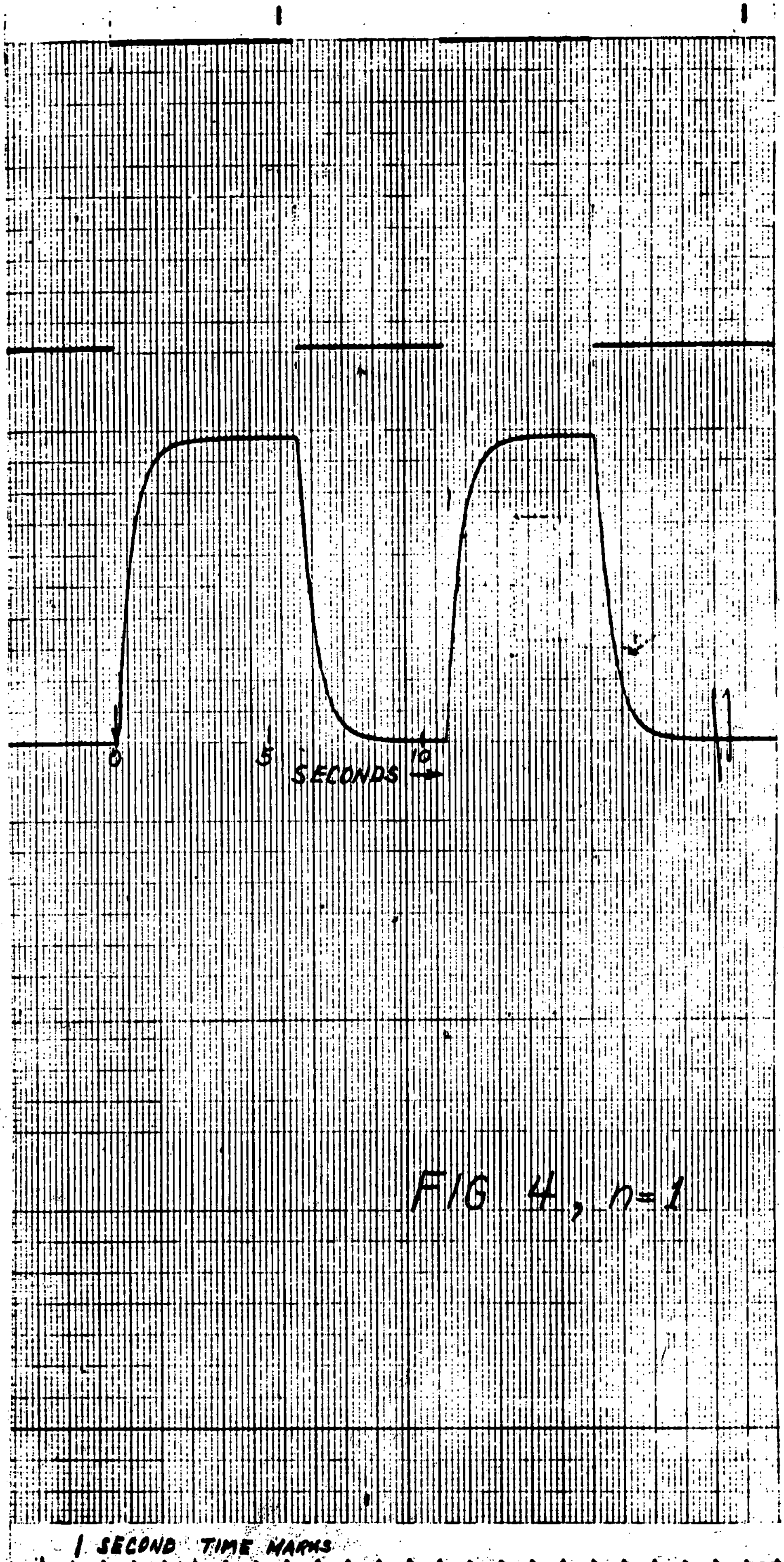


FIGURE 2

Capacitance in Microfarads

n	C_1	C_2	C_3	C_4	C_5	C_6	C_7	C_8	C_n	C_{10}
1	10									
2	10	5								
4	10	7.5	5.0	2.5						
6	10	8.33	6.67	5.0	3.33	1.67				
8	10	8.75	7.5	6.25	5.0	3.75	2.5	1.25		
10	10	9	8	7	6	5	4	3	2	1

FIGURE 3


0
5
SECONDS
10
FIG 4, n=1
1 SECOND TIME MARKS

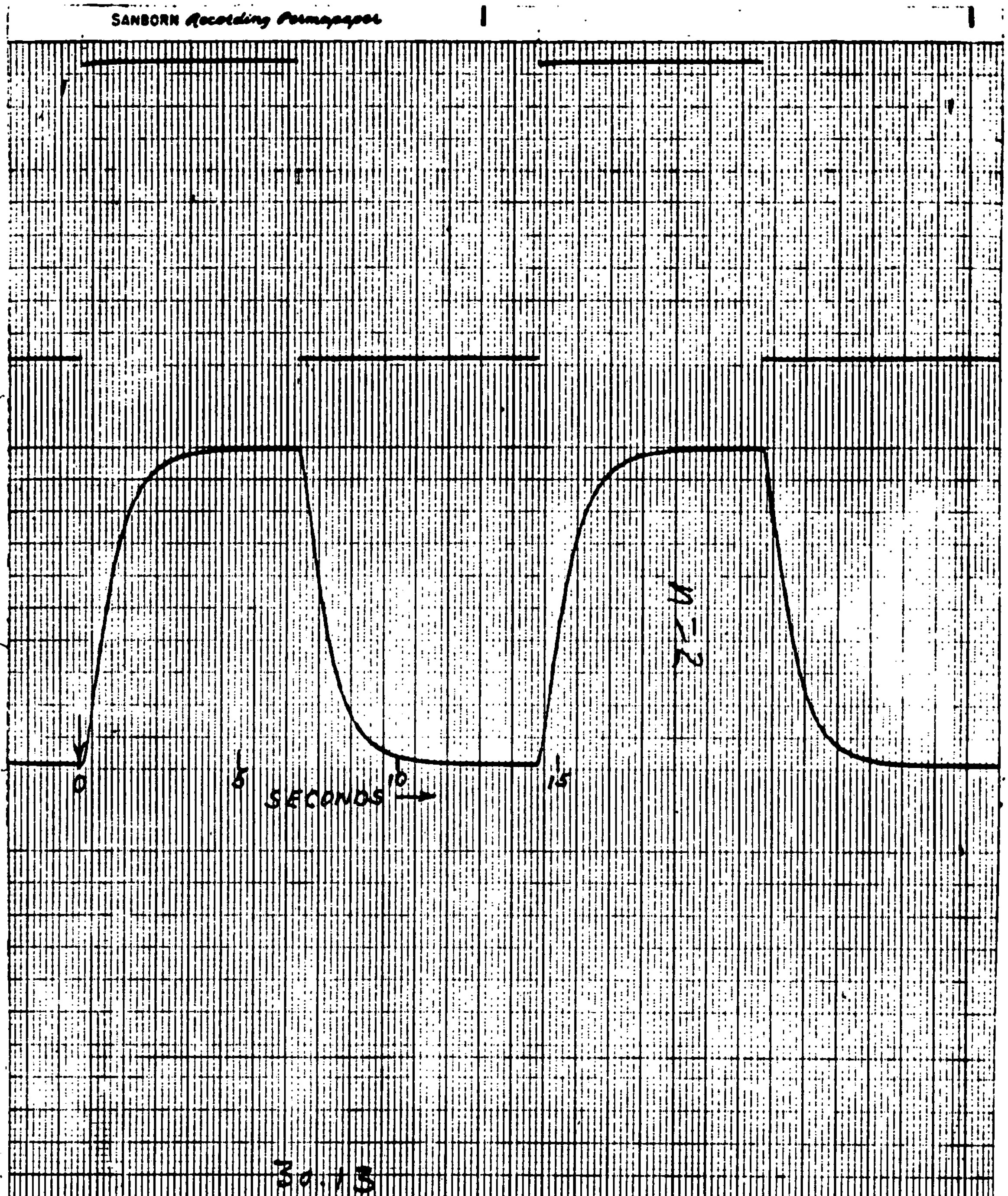
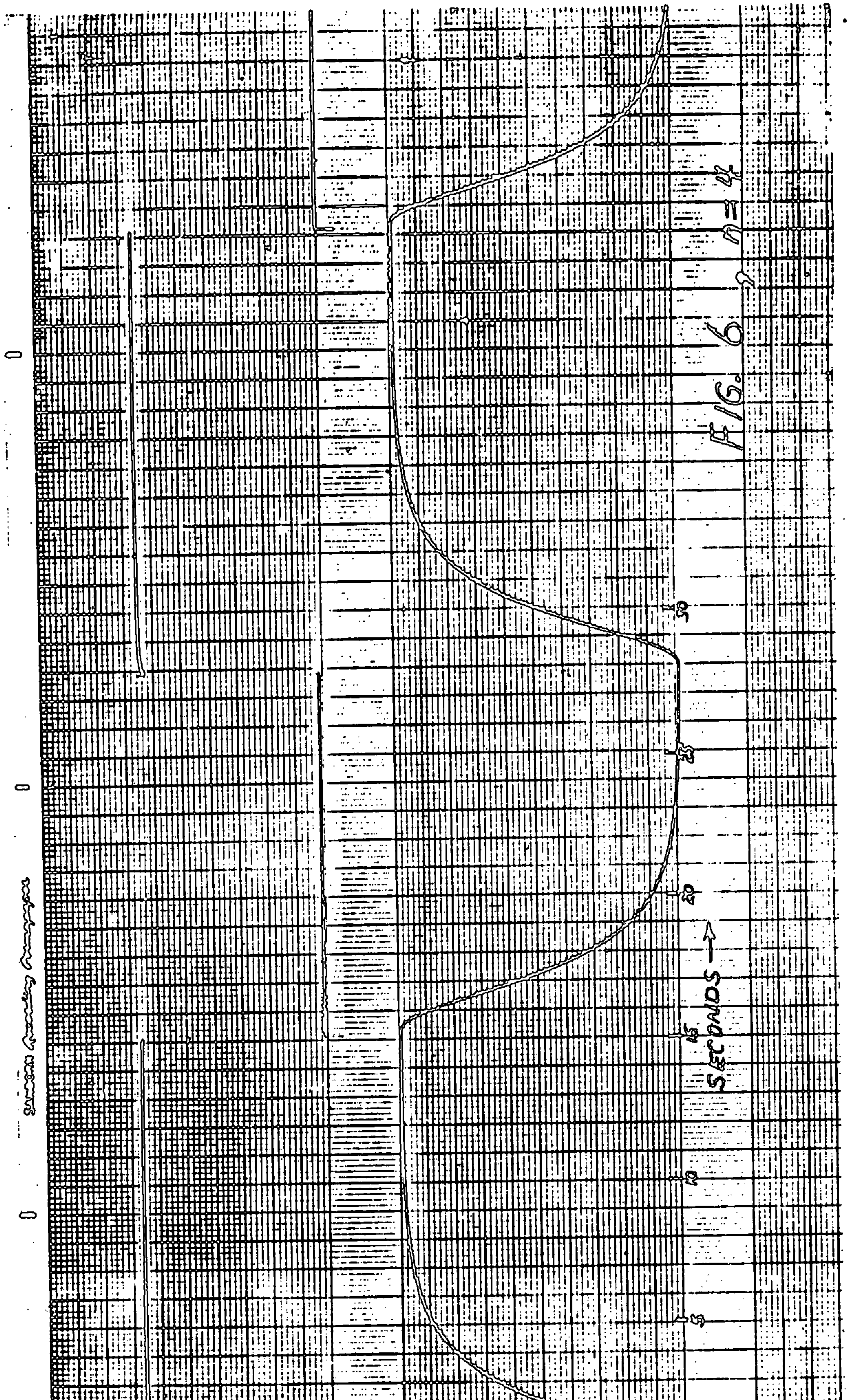


FIG. 5, n=2

1 SECOND TIME MARKS


0
5
10
15
20
25
30
SECONDS →
FIG. 6 , n = 4

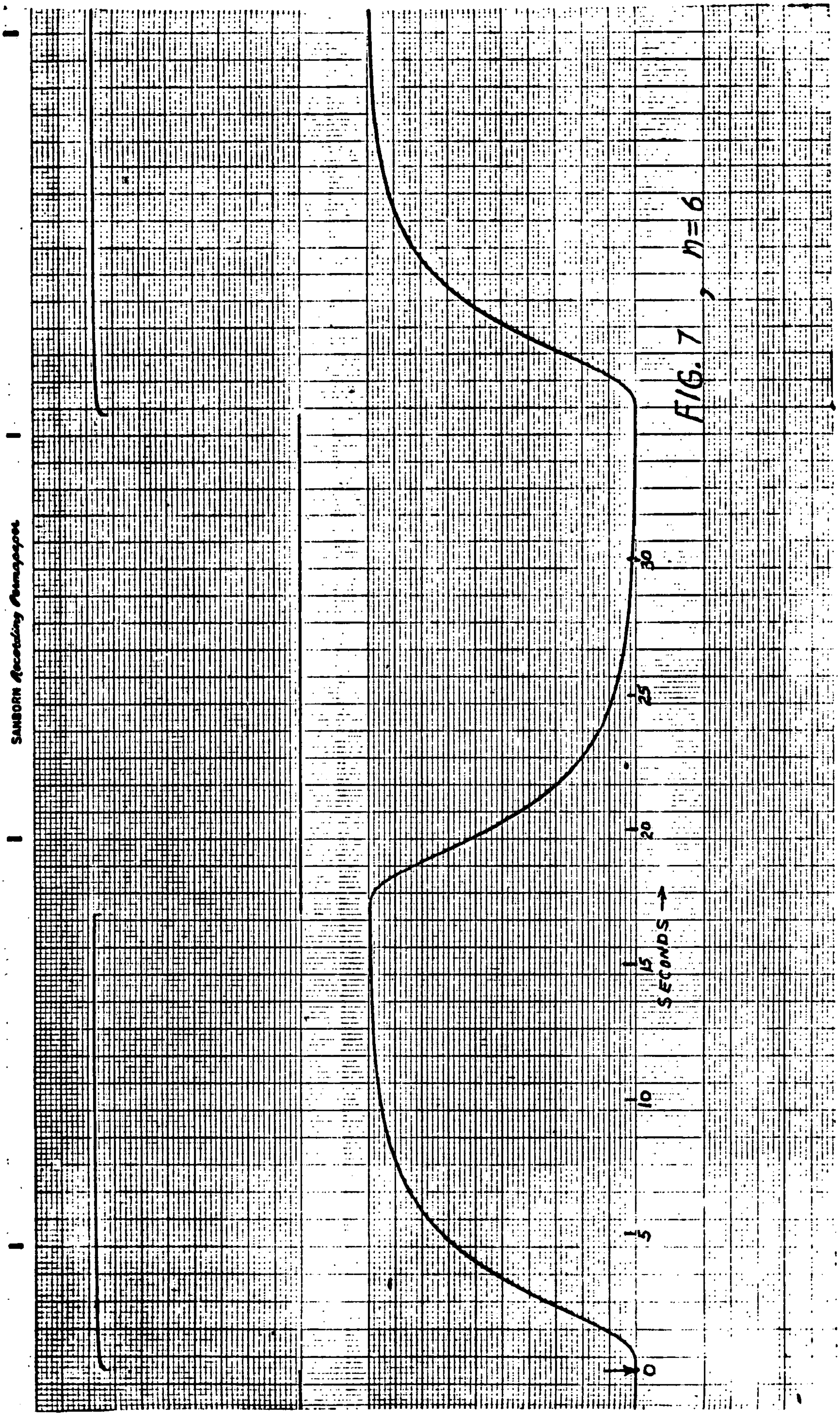


FIG. 7, n=6

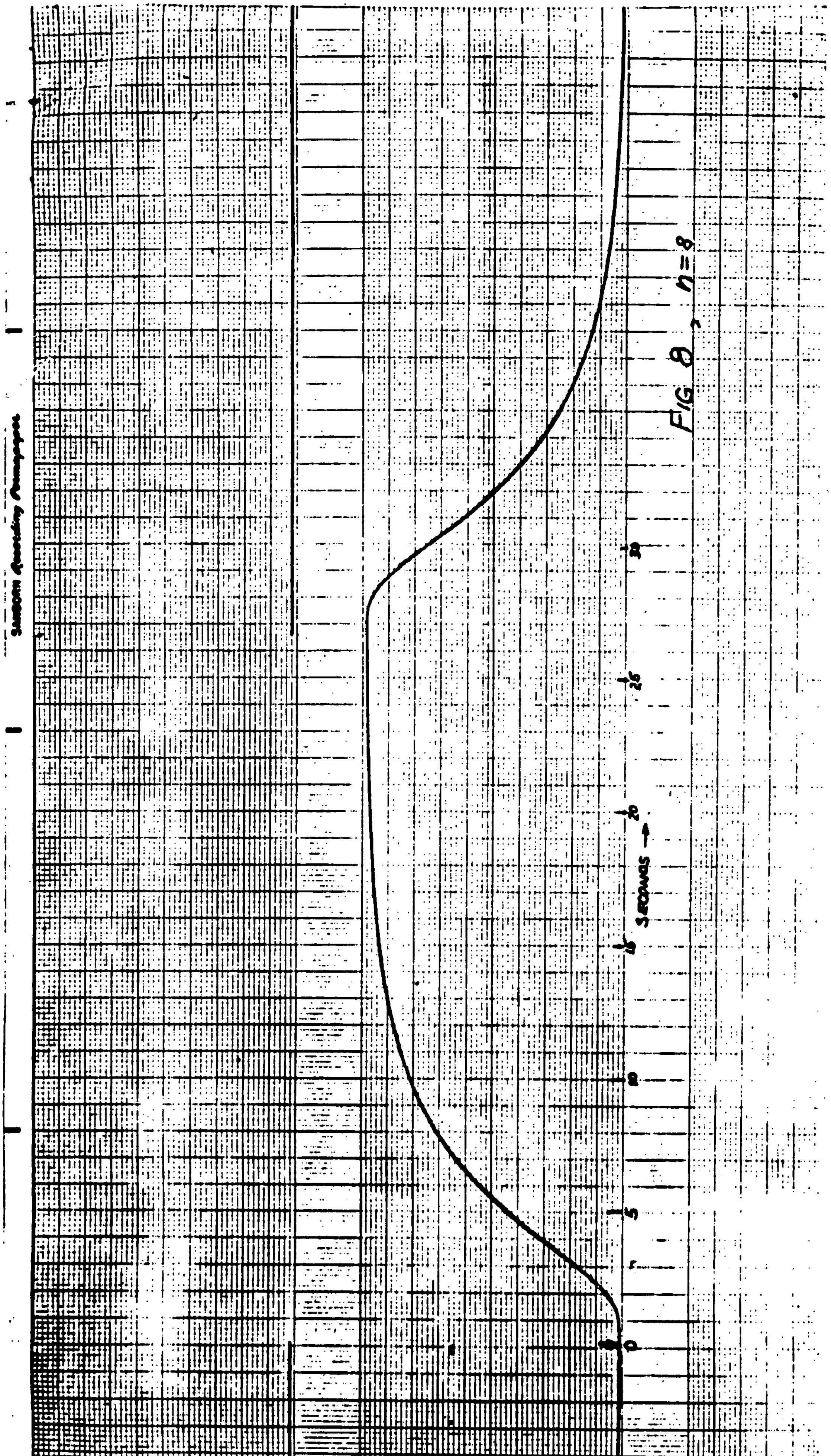
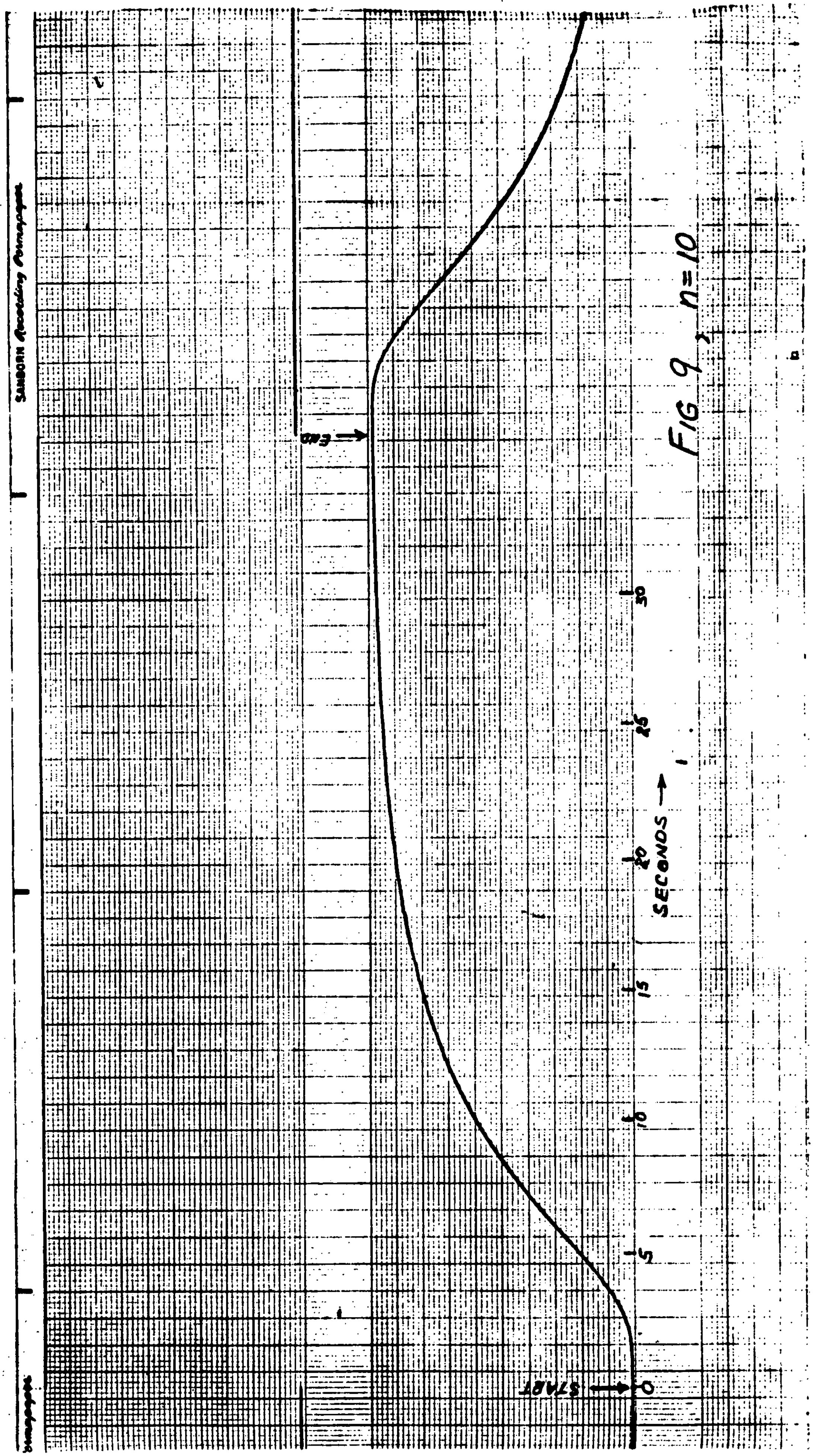


FIG 8, n=8


SANBORN Recording Permapaper
FIG 9, n=10
SECONDS →
0
5
10
15
20
25
30
START
END

APPENDIX III

TEST AND ASSEMBLY DESCRIPTIONS FOR COMMERCIAL CONNECTORS

TEST EQUIPMENT AND PROCEDURES

Three representative tests were made:

- A qualitative leak check with 1000psi internal pressure (the swaged permanent fitting was pressure checked to 2000 psi), using a mass spectrometer helium leak detector with sensitivity to 10^{-7} atm cc/sec.

- A vibration test with no internal pressure and at such amplitude as to allow a 10,000 psi peak bending stress. This is to be applied at the junction between the tube and connector. At the same time, the force level required would be the equivalent to produce 200g's acceleration level at the connector. Imposition of the "g" and stress level criteria followed here is not recommended. The stress criterion only was chosen for the model connector tests described in Section 4 which chronologically followed those reported on here. The tube strains were monitored using strain gages to assist in producing the proper loading levels. The test plan was to subject the connector assembly to one million vibratory cycles at its resonant frequency.

- A post-vibration leak test with 1000 psi internal pressure.

Cryogenic (-321° F.) and high ($+700^{\circ}$ F.) temperatures were imposed on the swaged gasketed ferrule in addition to the above conditions.

Figure 1 is a photograph of the vacuum chamber used for all of the leak tests. It is shown with the swaged gasketed assembly. A connector is usually mounted inside of the chamber with its tube ends protruding out both ends. The entire assembly is vacuum shrouded to observe any leakage. The ferrule assembly is a special case and steps had to be taken to test leakage only through its leak path; the swage between tube and ferrule. Each ferrule was checked separately with the main gasket seal between both ferrules, excluded from the leak test as shown in Figure 2.

The tube end seals are shown in Figure 1. They are essentially "O" ring seal plugs which fit into the tube ends (reamed first for smoothness) and are held in place by hardened steel pins. The removable plugs allow transfer from assembly to assembly so that one pair services all assemblies (the swaged permanent assembly used ends silver brazed in place). One plug is a blank while the other has provision for a threaded high pressure fitting to supply helium to the connector.

The vibratory tests were made at the connector-tube assembly's first mode resonant frequency. A typical assembly is shown in Figure 3 mounted in the

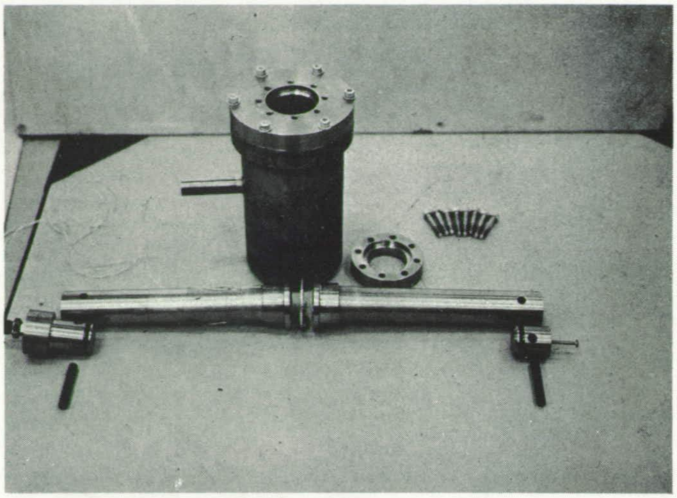


Figure 1. Swaged Gasketed Connector Dis-assembled and Test Devices

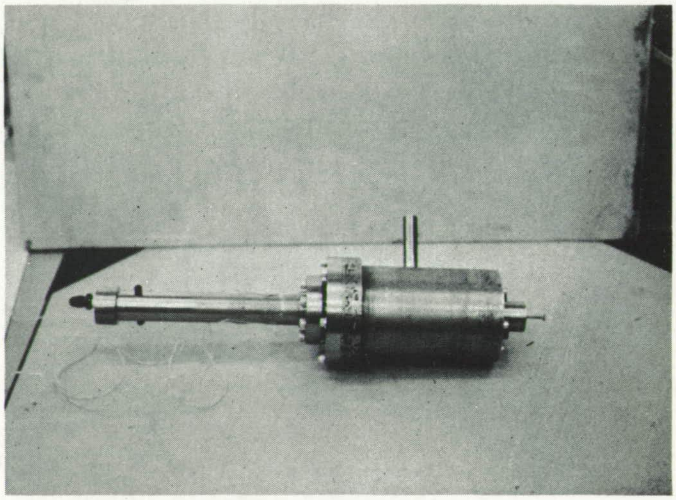


Figure 2. Instrumented and Assembled Swaged Gasketed Ferrule in Pressure Test Fixture

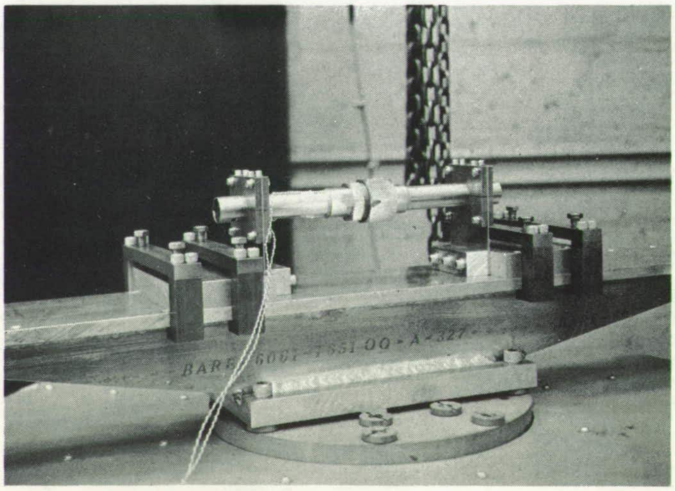


Figure 3. Connector-tubing Vibration Test Assembly

vibration fixture which in turn is mounted to a vibration shaker. The connector tubing assembly is mounted in a simulated hinged support. The test fixture is shown by sketch in Figure 4. The flexures provide a hinge action as the connector is vibrated in the vertical direction. They provide resonant amplification in order to establish 200 g's acceleration at the connector. This limitation, plus the requirement that the tubing at the connector is stressed to 10,000 psi, fixes the length of the span between flexures.

The curve given in Figure 5 shows the relation between the connector weight (W) in pounds, and the effective span length (ℓ_{eff}), in inches, for 1.0 inch outside diameter by 0.065 inch wall type 316 stainless steel tubing. This size tube was used for all tests except the swaged permanent fitting test which used 1/2 inch outside diameter by 0.065 inch wall tubing, type 316 stainless steel. The actual span length in inches is determined as follows:

$$\ell_{actual} = \ell_{eff} - 1.0 \text{ inch}^{*} + (\text{length of connector between tubing connection points})$$

If the span length is correct, then the resonant frequency (fn) will correspond to that indicated in Figure 6.

The curve given in Figure 5 was plotted from the following expression:

$$(\ell_{eff})^2 + 2.47 \frac{W(\ell_{eff})}{\mu} - 9.88 \frac{M}{\mu A} = 0$$

Where: ℓ_{eff} = effective length of span in inches

M = bending moment in tubing at connector in inch-lbs.

A = acceleration of the connector in g's

μ = the weight in pounds per inch length of tubing

W = weight of connector in pounds

The first mode resonant frequency plotted in Figure 6 was calculated from the following expression:

$$fn = \frac{1}{2\pi}\left[\frac{48\ EIg}{(\ell_{eff})^3\ (W + 0.5\ \mu\ell_{eff})}\right]^{1/2}$$

Where:

E = the modulus of elasticity of the tubing material in pounds/(inches)^2

I = the area moment of inertia of the tubing about a diameter in (inches)^4

* The 1.0 inch subtracted from the above was determined from experience and represents the added tubing length due to the flexibility of the tubing connector joint.

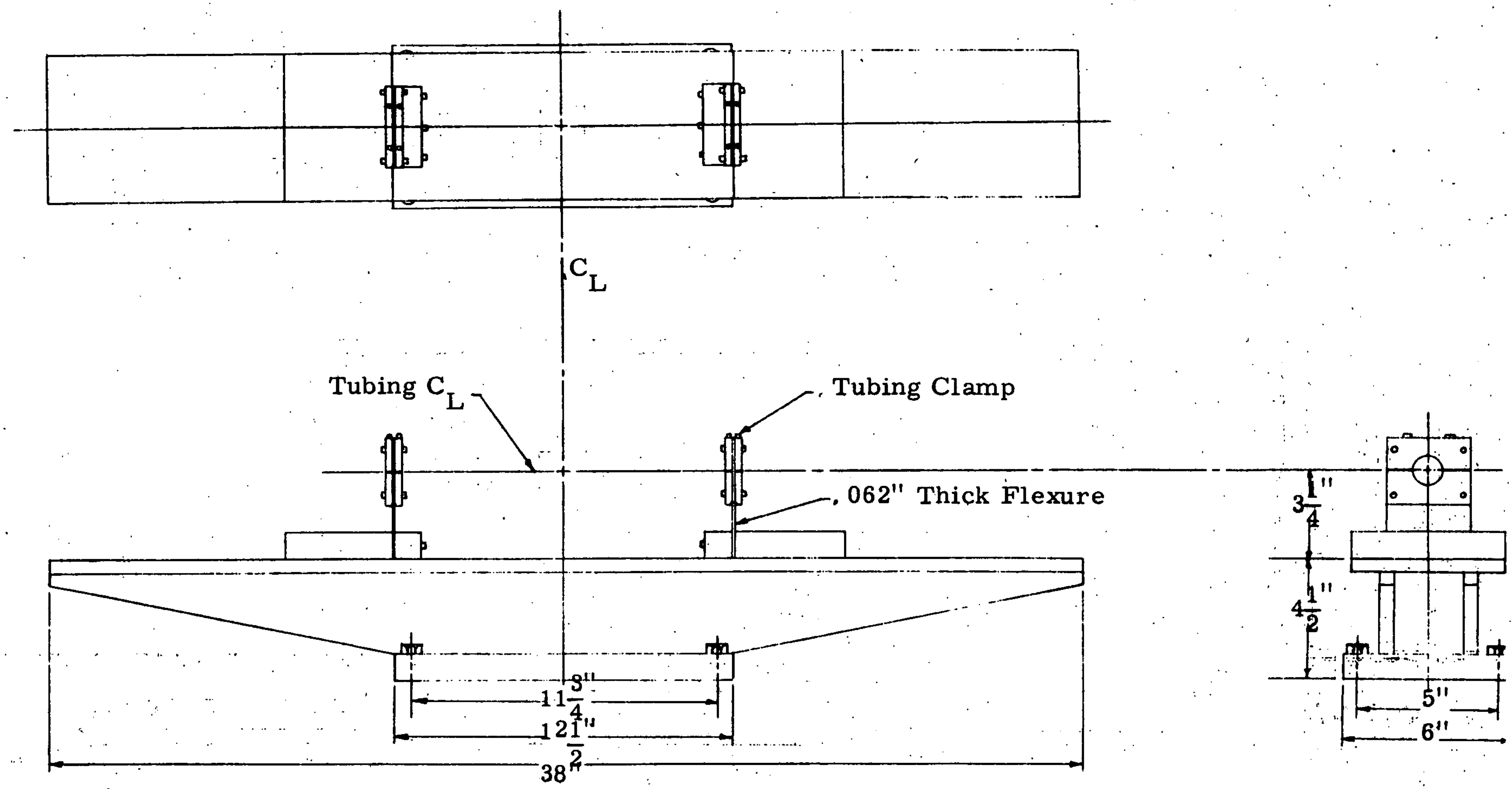


Figure 4. Assembly Drawing of Connector-tubing Vibration Test Fixture

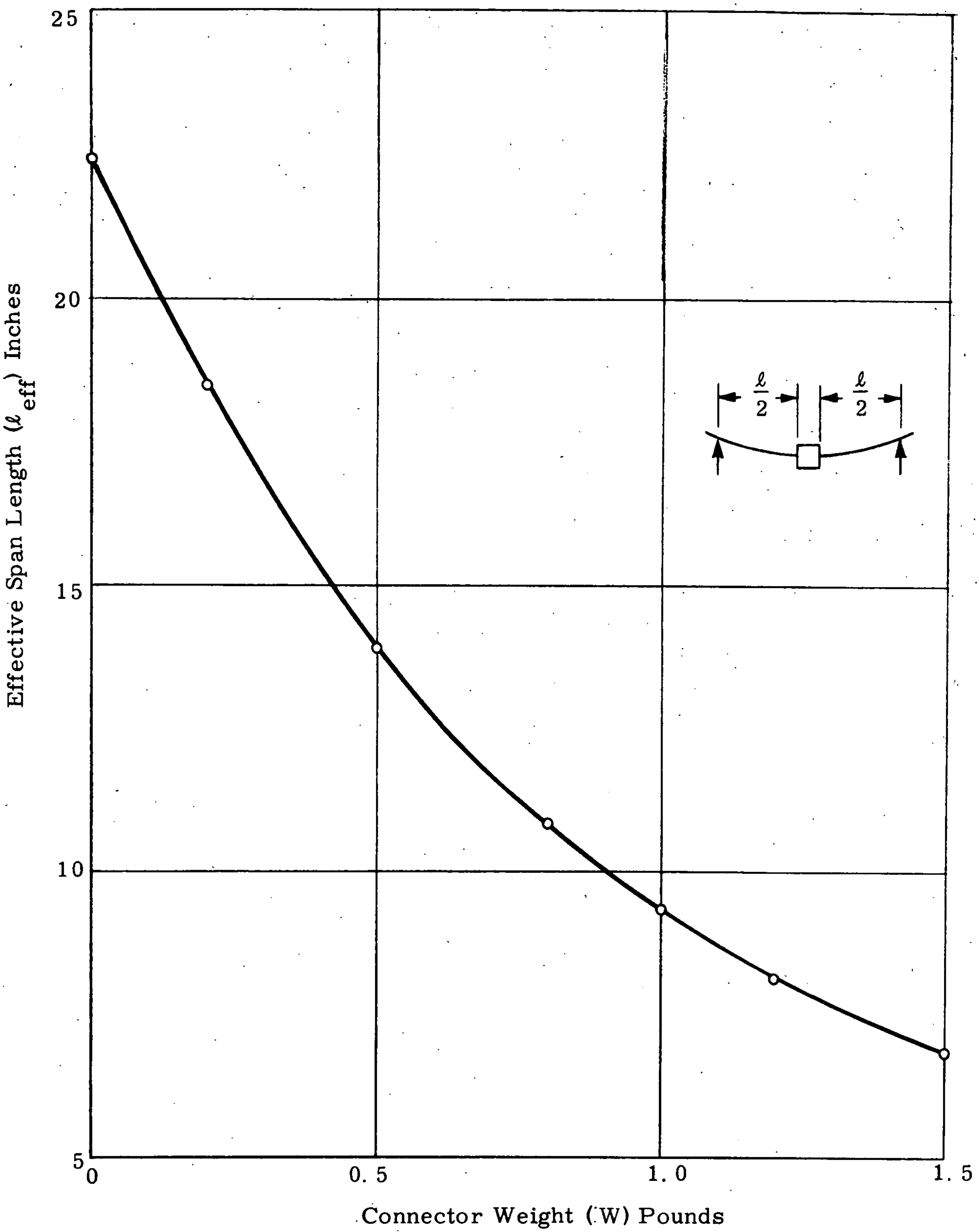


Figure 5. Span Length Versus Connector Weight for 1.0 inch OD by 0.065 inch Wall Stainless Steel Tubing Vibrating at its First Mode Resonant Frequency.

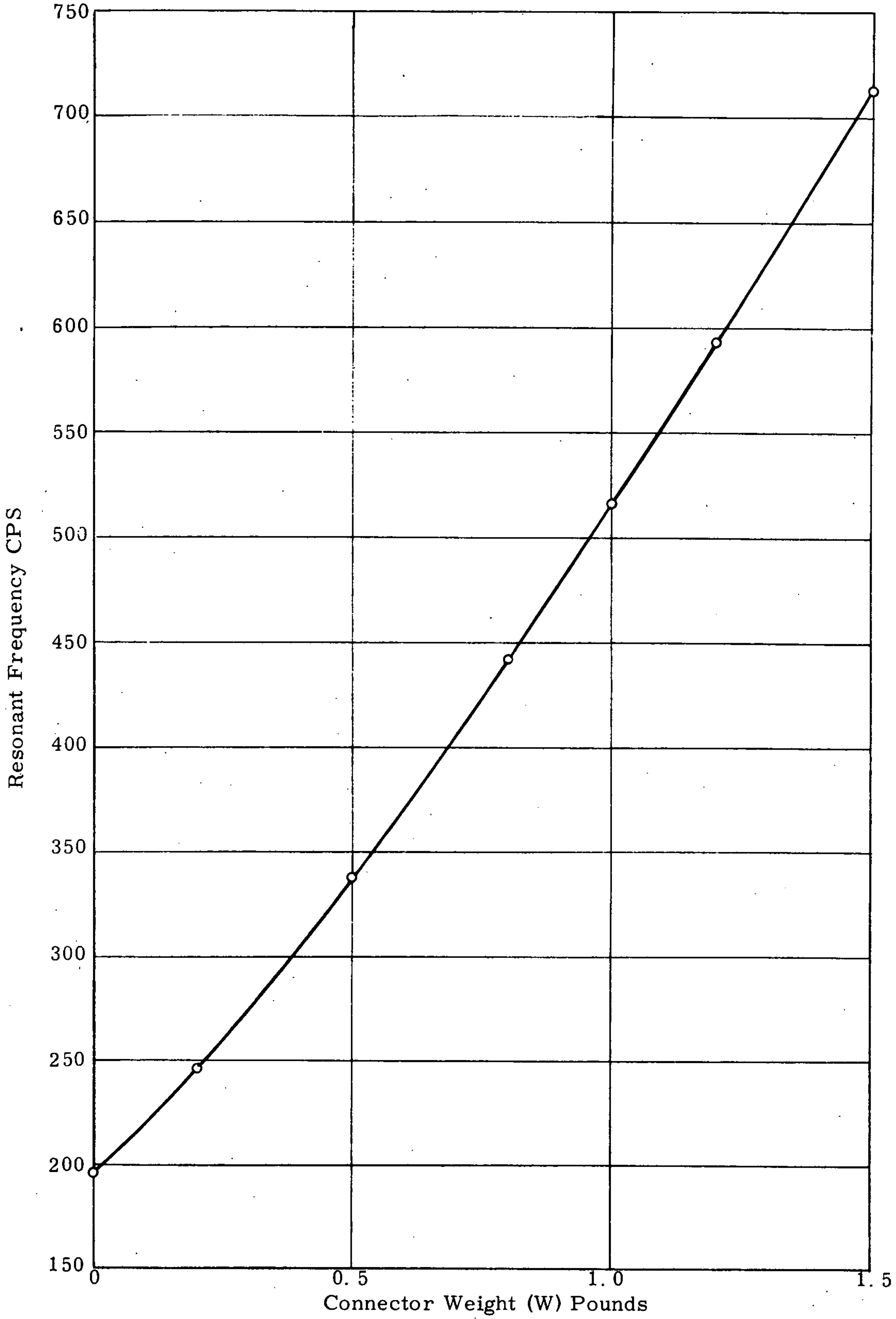


Figure 6. First Mode Resonant Frequency versus Connector Weight for 1.0 inch OD by 0.065 inch Wall Stainless Steel Tubing

Strain gages were mounted on the outer surface of the tube to monitor maximum bending strain while undergoing vibration. With a static calibration (done by hanging a known weight at the center of the fitting), the bending moment as a function of strain could be observed while vibrating.

To determine the "g" level, the displacement of the fitting was monitored using a microscope with a calibrated scale eye piece.

ASSEMBLY PROCEDURES

Each connector was assembled according to the manufacturer's catalog information. The tube ends were prepared commensurate with the requirements specified. Where parts of the connector were to be in rubbing contact, they were lubricated with spray on Moly-kote dry lubricant. The following are descriptive comments about each assembly, how it was made, the special tools required, and assembly procedure in general.

Swaged Gasketed Ferrule

The ferrules can be seen prior to assembly in Figure 7. Figures 8 to 10 show various stages of swage tool assembly. The ferrule is clamped in the fixture and the roller swage tool inserted. The end is rotated until swaging takes place as indicated by a torque wrench reading at the roller assembly. The roller tool is so designed that as the main tapered stem is rotated, the three longitudinal rollers travel up the tapered stem expanding outward and performing the swage. Two assemblies (4 ferrules) were made and tested. The first in the normal way, the second after boring out the tube stop portion of the ferrule. This eliminated the possibility of a secondary seal being formed.

The vacuum chamber required for performing leakage tests at the extreme temperatures is shown schematically in Figure 11. The configuration allows for collecting of leakage solely through the swage and excludes the gasket seal between swaged units. The fabricated hardware is shown in Figure 12. Figure 13 shows the assembled components lying in a flask prior to the cryogenic test. Liquid nitrogen is poured into the flask and the parts allowed to reach temperature equilibrium. The assembly is then pressurized and a leak check made.

A third assembly was made and tested under temperature extremes. Figure 14 shows the apparatus prior to making the high temperature test which utilized a laboratory type oven.

It has been determined that when the two swaged components are made of the same material, the presence of a temperature environment does not affect the leak tightness.

Swaged Permanent Fitting

Two swaged permanent fitting assemblies were made employing a special assembly tool. A fitting is shown disassembled in Figure 15. The special tool required to accomplish the mounting is shown in Figure 16. The tool was

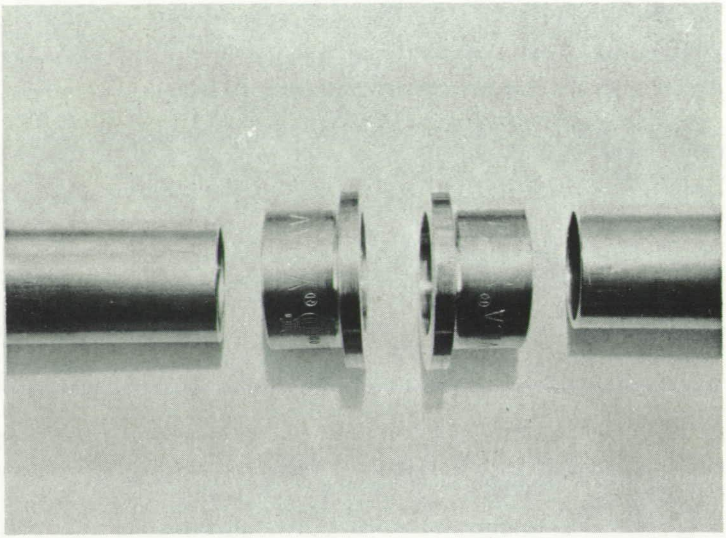


Figure 7. Swaged Gasketed Ferrules, with Tubing

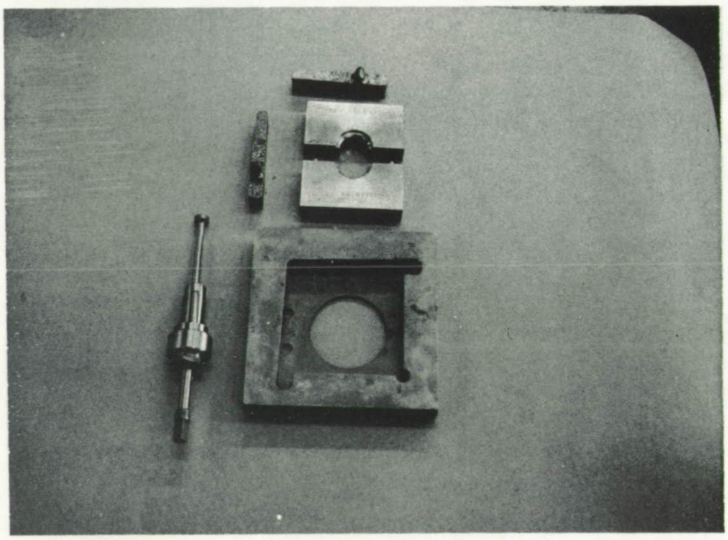


Figure 8. Swage Fitting Assembly Tools

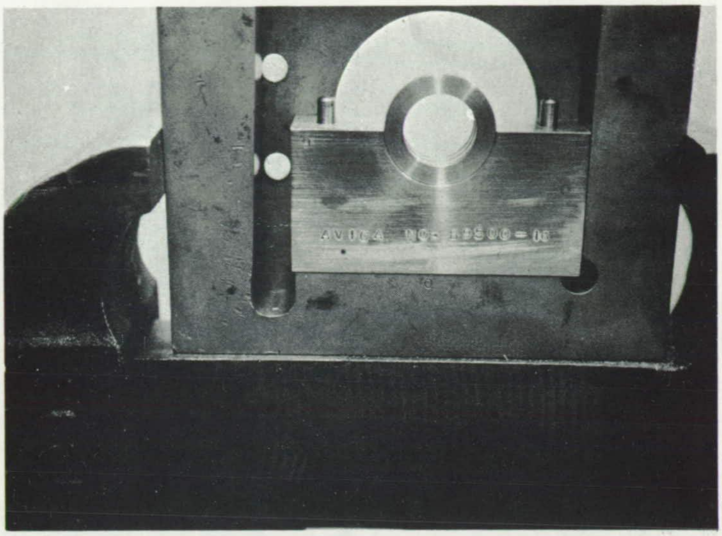


Figure 9. Swaging Tool, Partially Assembled

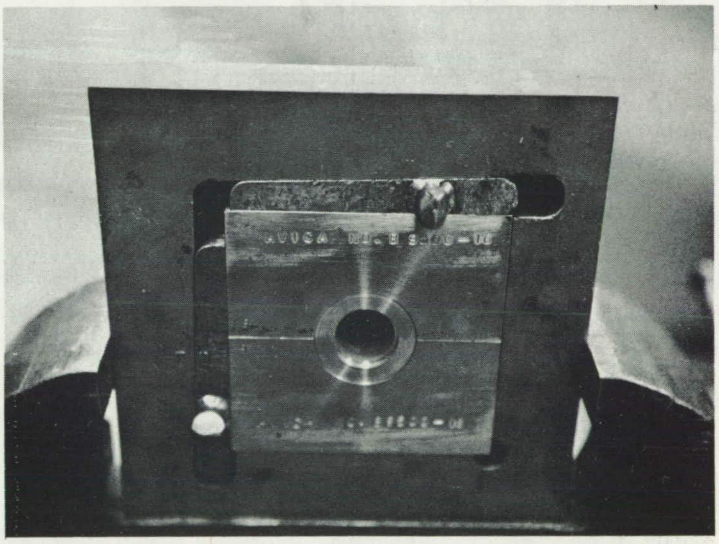


Figure 10. Swaging Tool, Assembled

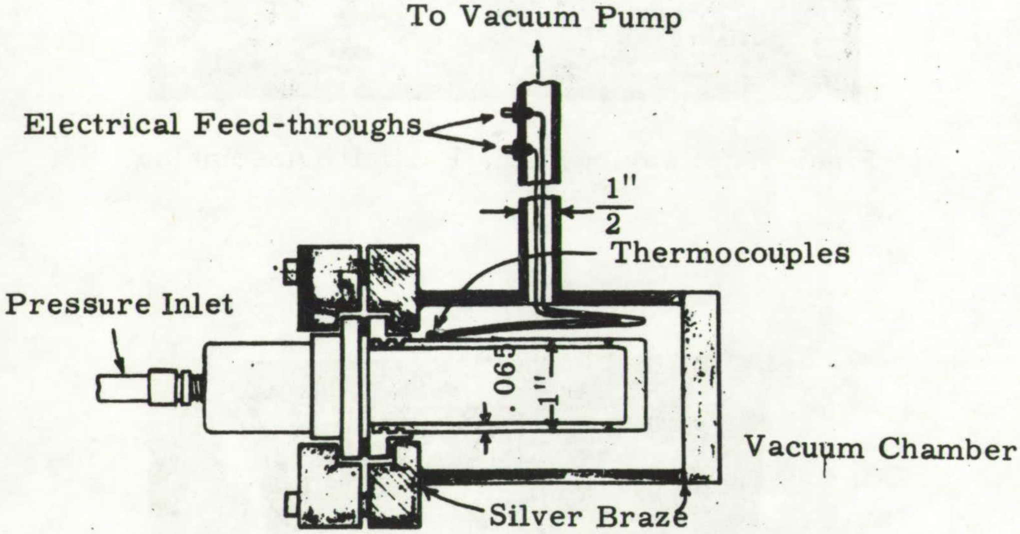


Figure 11. Swaged Connector Vacuum Chamber Test

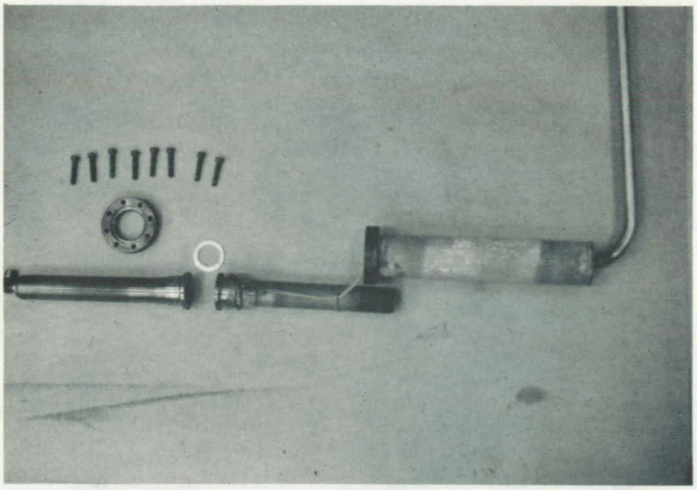


Figure 12. Swaged Connector Hardware

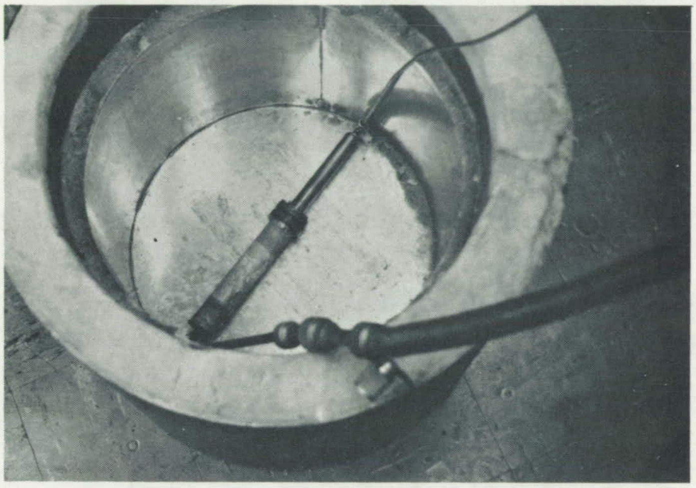


Figure 13. Assembled Components Prior to Cryogenic Test

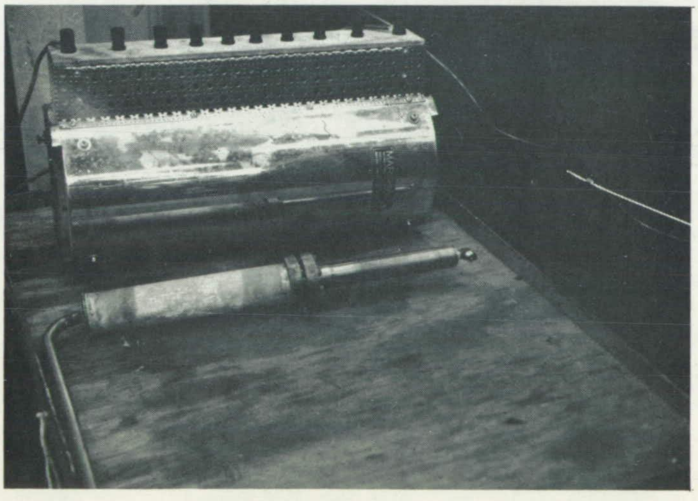


Figure 14. High Temperature Test Apparatus

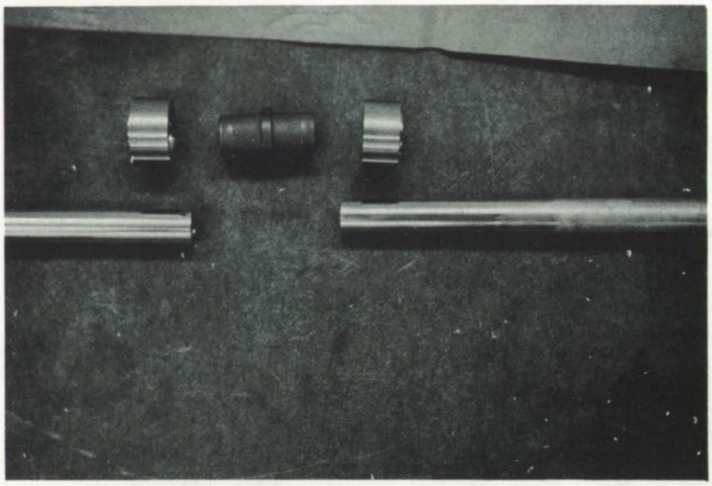


Figure 15. Swaged Permanent Fitting Components

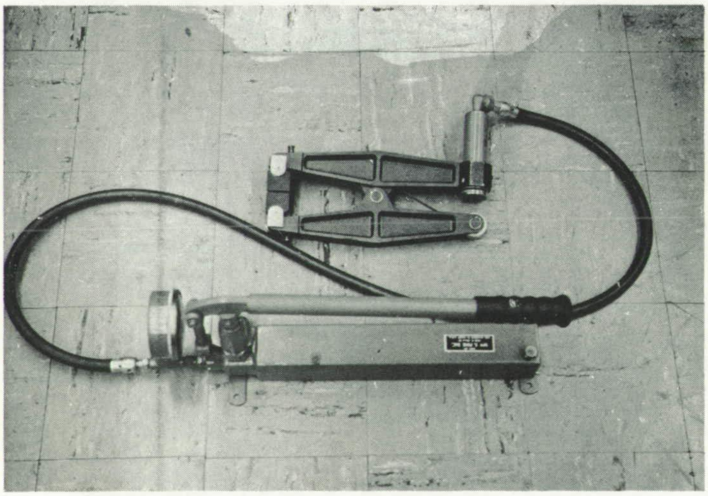


Figure 16. Tool for Assembling Swaged Permanent Fitting

furnished by the manufacturer and the swage operation conducted by their representative on General Electric premises.

The tool is a hand operated hydraulic pump which supplies hydraulic pressure to a piston at one end of the "scissors" beam arrangement. Hydraulic pressure measurement is made by the integral pressure gage. As the hand pump is operated, the "scissors" open up at the piston and close at the fitting end resulting in sufficient driving force to seat the fitting sliders on the tapered union. These parts can be seen by referring to Figure 17 and noting that the sliders are the outermost ring shaped parts at the top. The union is located between them.

Two Part Conical Ferrule Fitting

Two union assemblies were made using procedures exactly as specified in the manufacturer's catalog. No preparation was made on the tube other than squaring off and de-burring the ends. The assembly catalog procedure stated that the fitting should first be made "finger tight". Then, for high pressure systems, the nut should be tightened until the tube is no longer free to rotate in the fitting. This would then be the starting point for applying sufficient sealing force by turning the nut an additional 1-1/4 turns. Because the tube was slightly under-sized, an additional one-half turn was required from the catalog "finger tightness" value to create enough holding force on the tube to prevent it from rotating. A 2400 pound-inch torque wrench was used to measure the required torque to attain the necessary 1-1/4 turns. However, the torque was in excess of the maximum capable of being measured by the wrench.

Conical Gasket Threaded Fitting

Assembly was made by welding the tubes to the respective connector ends. The nut and bolt were lubricated with spray-on Moly-kote prior to applying the clamp to the two connector halves. The nut was torqued to the recommended 80 pound-inches.

Conical Gasket Clamped Fitting

Figure 18 shows the parts in disassembled condition. An oxide was noticed on the sealing surfaces of both ends of the connector after they were welded to the tube ends. They were cleaned by immersing them in an acidic cleaning agent. The cleaning agent was at a temperature between 50^{o} and 60^{o} C. The male flange cleaned up after one-half hour of immersion time, but the female flange retained a grayish colored oxide even after one hour. The original oxide color had been a straw color.

All parts were flushed with water and again chemically cleaned with acetone before assembly. The nut was torqued to the recommended 1375 pound-inch. A sharp rise in torque was felt at about 1000 pound-inch. This indicated seating of the gasket and metal-to-metal contact.

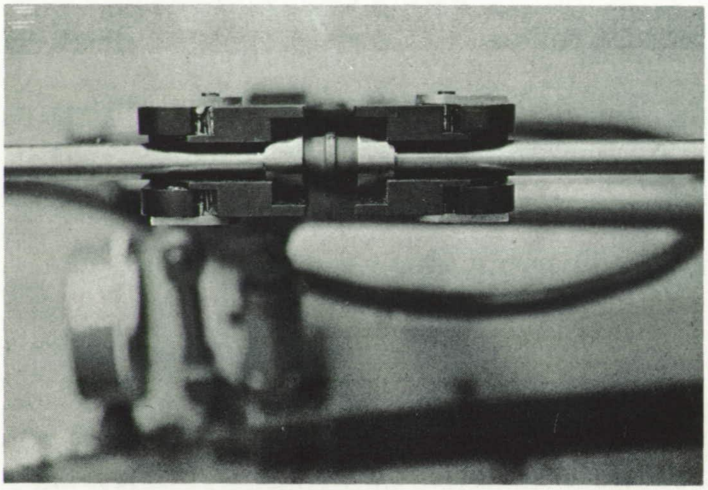


Figure 17. Assembly of Swaged Permanent Fitting

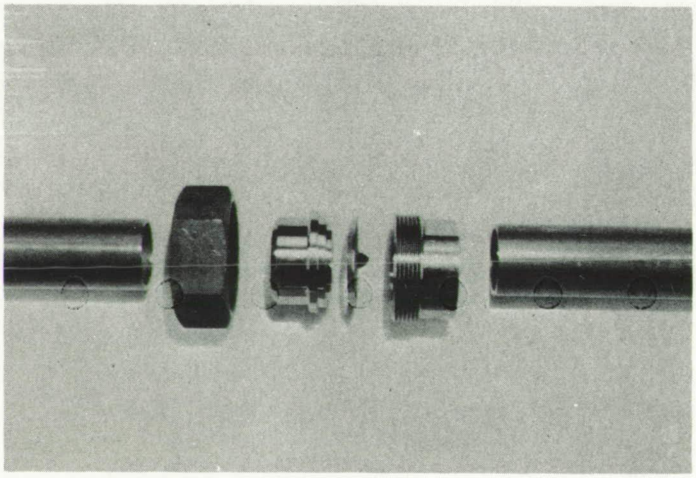


Figure 18. Components of Conical Gasket Clamped Fitting

Butt Seal Gasket Connector

The particular connections used were made for butt welding to a pipe system. Weld preparations were machined into the pipe preparation ends to accomodate tube ends. This type of connector employs a gasket as a seal. The ones used for the tests were silver plated. Recommended torque to make the assembly was 200 pound-feet.

One Part Conical Ferrule Fitting

After preparatory square up, de-burring and longitudinal scratch removing operations on the tube ends, the sleeves were pre-set onto their respective tubes. Scratches were removed from only one set of tubes so that a comparison could be made of the seal achieved under both conditions. Pre-setting, the act of mechanically swaging the cutting edge of the sleeve into the tube to make a seal, was done by means of a pre-set tool. When a sleeve has been "set" on a length of tubing, the connection can be disassembled as necessary. The sleeve remains with the tube and cannot be used again in another connector by virtue of its "biting-in" type of mechanical joint. All parts that were to rub during assembly of the fitting were lubricated with spray-on Moly-kote. When the sleeve was placed on the tube, the fit was somewhat loose. The tube measured from 0. 001 inch to 0. 004 inch under 1 inch, and the sleeve from 0. 008 to 0. 012 inch over 1 inch. The fitting nut was tightened until the tubing could not be turned. With this the start of the pre-setting operation, the nut was turned one turn. The torque required to do this was in excess of 2400 lb-inches.

When the connection was disassembled, metal chips were observed on the sleeve taper, the nut and union threads, and the seat. The tapered ring sealing surface became coated with cadmium plating from the sleeve. The sleeve was able to rotate about one half turn on the tube (this is acceptable).

The reduced inside diameter of the tubing was measured to be 0. 850 inch. This was a result of the sleeve's being swaged into the tube periphery. The minimum acceptable catalog figure is 0. 840 inch.

After clean-up and assembly, the nut was tightened the recommended one-sixth turn beyond the point of torque build-up. Again, torque in excess of 2400 pound-inches was required to accomplish this.

Deformable Gasket Joint

The connector is primarily a vacuum type of connection. However, its features also lend themselves to high pressure sealing. As recommended by the manufacturer, sufficient tightening torque was applied until the soft aluminum gasket was deformed and essentially a total metal to metal contact existed. A sharp torque rise indicated bottoming occurred after about a seven-eights turn of the nut. This required approximately 1500 pound-inches of torque. The torque was then increased to 2400 pound-inches, one turn of the nut.

The parts were from Monel, necessitating the tube joint be made by copper brazing.

"V" Seal Joint

This connector is a pressure energized flanged type. The flanges are held together with six 1/4-28 bolts. It is a flanged connector with a "V" cross-section metallic gasket between them. After welding both connector assemblies to the tubing, the sealing surfaces checked within the specified tolerance for flatness of 0.005 inch total indicator reading. The ovality was also within the specified tolerance of 0.003 inch total indicator reading.

The bolts and nuts were lubricated with spray-on Moly-kote before assembly. The recommended bolt torque of 50 pound-inches on each of the six flange bolts was applied incrementally. The bolts were torqued first to 20 pound-inches then 40 pound-inches, and finally 50 pound-inches. The cross-torque method was used to achieve uniform torque take-up.

Face Seal Connector

The connector parts were silver brazed to their respective tube ends with the assistance of a chill block (Figure 19) loaned by the manufacturer. The aluminum chill block measured 2-1/2 inches square by 4 inches with a number of shallow (3/16-inch deep) blind threaded holes to accept standard size tube ends from 3/16 to 1-1/2 inches outside diameter. A pipe tapped hole to accept water inlet and outlet is at either end. The block is hollowed out and baffled to effect high heat transfer. This method of brazing proved successful in keeping the sealing surfaces clean.

There were two separate assemblies available. The first has both sealing surfaces coated with dry film lubricant, and the second highly polished (8RMS) sealing surfaces. The sealing surfaces are located at the right end of the threaded sleeve just left of center in Figure 20 and the left end of the threaded nut end just to the right of center.

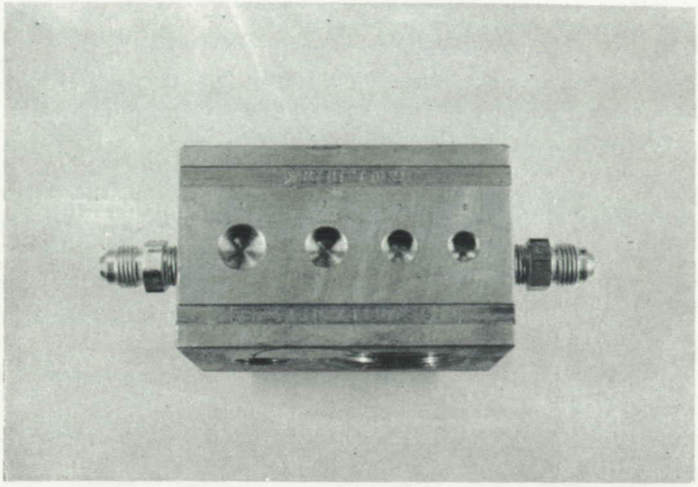


Figure 19. Chill Block Assembly for Brazing Face Seal Connectors to Tubes

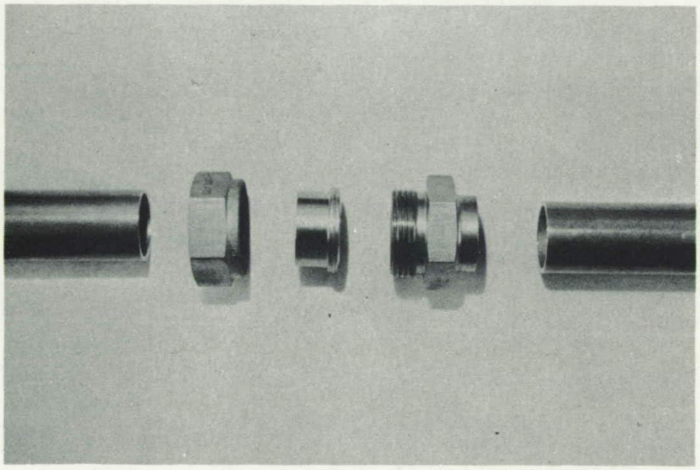


Figure 20. Face Seal Connector Components