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### AFRPL-TR-69-97

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## DEVELOPMENT OF AFRPL FLANGED CONNECTORS FOR ROCKET FLUID SYSTEMS

T. M. Trainer, J. V. Baum, J. R. Thompson, and N. D. Ghadiali

> Battelle Memorial Institute Columbus Laboratories

#### TECHNICAL DOCUMENTARY REPORT NO. AFRPL-TR-69-97

### July 1969

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by

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### FORE WORD

This report summarizes research conducted under Phases II and III of USAF Contract AF 04(611)-11204 from January, 1966 to March, 1969. The research was performed by the Columbus Laboratories of Battelle Memorial Institute under the auspices of the Air Force Rocket Propulsion Laboratory, Edwards Air Force Base, with Capt. John L. Feldman and Capt. F. M. Cassidy serving as project monitors. The principal investigators were N. D. Ghadiali, D. E. Netter, J. G. St. Clair, J. R. Thompson, Research Engineers; J. V. Baum and B. Goobich, Associate Fellows; E. C. Rodabaugh, Senior Engineer; and T. M. Trainer, Project Manager.

This technical documentary report has been reviewed and is approved.

John L. Feldman Captain, USAF Project Engineer

### ABSTRACT

A 3-year program was conducted to develop a family of flight-weight flanged tube connectors which utilize the "Bobbin" seal concept for both 6061-T6 aluminum and Type 347 stainless steel tubing systems. In addition, engineering support was furnished during the evaluation of AFRPL stainless steel threaded connectors by selected organizations. The activities in relation to the first objective consisted of: (1) the selection of bolted, flanged connectors as the best means of joining tubing from 1 to 16 inches in diameter. (2) the development of stainless steel and aluminum Bobbin seals for typical flanged connector sizes, (3) the preparation of a detailed computer program for the optimum design of flanged connectors incorporating Bobbin seals, (4) the design, fabrication, and qualification testing of representative flanged connectors, and (5) the preparation of designs for a family of stainless steel and aluminum flanged connectors. The activities directed toward the second objective consisted of: (1) the investigation of seal-removal tools, (2) the design and evaluation of connector modifications to achieve seal retention and misalignment limitation, (3) the investigation of increased radial seal loading techniques, and (4) the investigation of stress relaxation in threaded connectors. In addition to a summary of the technical activities, the report contains the computer design program for flanged connectors, and designs for aluminum and stainless steel flanged connectors for pressures up to 1500 psi and for tube sizes through 16 inches. It is recommended that MS standards and specifications be prepared for cryogenic stainless steel and aluminum flanged connectors for tubing through 3 inches in diameter. Additional developmental work is recommended in regard to larger flanged connectors, and in regard to certain aspects of the stainless steel threaded connectors.

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#### ABBREVIATIONS AND SYMBOLS APPEARING IN REPORT BODY\*

- **F**<sub>S</sub> Radial sealing force per inch of seal disk circumference, lb/in.
- F<sub>T</sub> Radial force per inch of seal tang outside circumference at the root of each seal disk, lb/in.
- S<sub>v</sub> Material yield stress, psi
- D<sub>0</sub> Seal disk outside diameter, in.
- De Seal tang outside diameter, in.
- D<sub>i</sub> Seal tang inside diameter, in.
- L Seal tang length, in.
- t Seal tang thickness, in.
- **P**<sub>T</sub> Pressure on outside seal tang circumference, psi
- K Dimensionless factor for relating seal tang length to seal tang inside diameter and seal tang thickness
- Sa Allowable stress at most severe operating condition, psi
- P Operating pressure, proof pressure, or burst pressure, psi
- D Tube outside diameter, in.
- TAL Connector design axial load, lb
- PL Pressure end load, lb
- TBM Tube bending moment, lb-in.
- Z Tube section modulus, in.<sup>3</sup>
- S<sub>A</sub> Minimum stress for calculating tube bending moment, psi
- AL<sub>B</sub> Equivalent axial load, 1b
- S<sub>B</sub> Bending stress, psi
- T Tube-wall thickness, in.
- MSL Minimum residual seal load, lb
- BL Maximum allowable bolt load, lb

\*Individual listings of abbreviations and symbols have been prepared for each Appendix, where applicable.

FS	Factor of safety
As	Bolt tensile stress area, in. <sup>2</sup>
Db	Basic bolt diameter, in.
S	Axial stress in bolt, in.
n	Number of threads per inch
To	Torque, lb-in.
R	Factor relating bolt torque, stress, and diameter
LT	Seal disk thickness, in.
DSA	Axial deflection rate of seal tang, in./lb
SEE	Modulus of elasticity of seal material, psi
σ <sub>θ</sub>	Circumferential stress, psi
a	Inner seal tang radius, in.
Ъ	Outer seal tang radius, in.
$\mathbf{P_i}$	Internal pressure, psi
Po	External pressure, psi
r	Location of stress, in.
∆F <sub>S</sub>	Radial seal load increment contributed by seal disk, lb/in.

k

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#### DEVELOPMENT OF AFRPL FLANGED CONNECTORS FOR ROCKET FLUID SYSTEMS

by

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I

#### INTRODUCTION

In April, 1962, the Air Force Rocket Propulsion Laboratory initiated a program at Battelle's Columbus Laboratories to develop a family of separable fluid connectors capable of meeting the stringent requirements of present and future rocket propulsion systems. The primary objective of this program, Contract AF 04(611)-8176, was to design and evaluate lightweight mechanical connectors that could successfully seal helium to  $2 \times 10^{-7}$  atm cc/sec in missile-system environments. As a result of this program, connector design criteria were established, a unique Bobbin seal was conceived, and a threaded connector incorporating the Bobbin seal, for tubing sizes up to l inch, was designed and its feasibility demonstrated. This work is described in Technical Documentary Report No. RTD-TDR-63-1115, dated December 1963 (AD 426290).

A subsequent program, Contract AF 04(611)-9578, was initiated in December 1963, to design families of threaded-type tube connectors and to demonstrate their ability to meet the extreme requirements of missile fluid systems. Elbows, tees, crosses, and unions made from Type 347 stainless steel, René 41, and 6061-T6 aluminum were designed for standard tubing sizes ranging from 1/8 to 1 inch, and limited qualification tests were performed with selected connector configurations for 3/8- and 3/4-inch tubing systems. Preliminary MS standards and specifications were prepared for selected sizes of 4000-psi stainless steel connectors and 1000- and 750-psi aluminum connectors. This effort is described in Technical Documentary Report No. AFRPL-TR-65-162, dated November 1965 (AD 474789).

Two concurrent efforts were then initiated. In one effort, the Air Force Rocket Propulsion Laboratory reviewed the preliminary standards and specifications, purchased a number of 4000-psi stainless steel connectors, conducted qualification tests with many of the connectors, provided connectors to various facilities for evaluation, published MS standards and specifications,  $(1)^*$  and conducted tests directed toward improving the sealing reliability of 3/4- and 1-inch stainless steel connectors. Concurrently, a three-phase program was conducted by Battelle-Columbus under Contract AF 04(611)-11204. Under Phase I, a number of procedures were investigated for achieving a softer sealing surface on 6061-T6 aluminum seals, and an overaging process was selected as being ideal for aluminum seals of all sizes. The Battelle work is described in Technical Documentary Report No. AFRPL-TR-67-191, dated July 1967 (AD 817843).

\*MS Standards and Specifications and references are given on page 194.

Phase II, the major effort of Contract AF 04(611)-11204, was directed toward the development of flight-weight stainless steel and aluminum flanged-type separable connectors utilizing the Bobbin seal for tubing diameters from 1 through 16 inches. The activities included the investigation of promising types of flanged connectors; the design, fabrication, and qualification testing of selected connectors; and the preparation of specifications for families of stainless steel and aluminum connectors. Phase III included support for the work at the Rocket Propulsion Laboratory on threaded connectors, and the development of a modified threaded-connector configuration to provide seal retention and a limitation to the degree of misalignment that can be imparted to the connector during assembly. This report describes the work on Phases II and III under the following major headings:

FLANGED-CONNECTOR DESIGN FLANGED-CONNECTOR EVALUATION FLANGED-CONNECTOR SPECIFICATIONS THREADED-CONNECTOR SUPPORT

#### FLANGED-CONNECTOR DESIGN

The flanged-connector-design work consisted of four major efforts: (1) the development of Bobbin seals for tubing sizes from 1 through 16 inches, (2) the investigation of flanged-connector-design criteria, (3) the development of a flanged-connector-design procedure, and (4) the design of representative connectors.

#### Design Requirements

As defined by the work statement, flanged connectors were to be designed for the following service ranges:

Size, inches	Pressure, psi	Temperature, F
1 to 3	0 to 6000	-423 to 200
1 to 3	0 to 4000	<b>-42</b> 3 to 600
1 to 16	0 to 1500	-423 to 200
1 to 16	0 to 1500	0 to 1200

The flanged connectors were to be designed for a 5-year storage requirement and for service with the following fluids:

Oxidizers	Fuels	Gases
N204	N2H4	Helium
H <sub>2</sub> O <sub>2</sub>	UDMH	Nitrogen
C1F3	UDMH/NH (50/50)	Hot gas from bi-
F <sub>2</sub>	MAF	propellant-, mono-
RFNA	MHF	propellant-, and
Compound "A"	MMH	solid-propelli.nt
	H <sub>2</sub>	gas generator

On the basis of the work with threaded connectors and the consideration of the tubing systems generally used in rocket propulsion systems, it was mutually agreed and specified in the contract work statement that the tubing materials of primary interest to the program were 6061-T6 aluminum and Type 347 stainless steel.

#### Development of Bobbin Seals for Large-Diameter Tubing

At the beginning of the program, a study was made to determine the best plating for stainless steel seals. Tentative dimensions for stainless steel and aluminum Bobbin seals were then established for typical large tubing sizes, on the basis of the design criteria developed previously for threaded connector seals. Seals of these dimensions were fabricated and tested, and seal-design formulas were developed for inclusion in the flange-connector-design computer program. The formulas were subsequently modified as a consequence of the qualification testing.

#### Sealing Principles of the Bubbin Seal

Figure 1 shows the configuration of a typical Bobbin seal and the four major stages of seal installation. First, the seal is positioned in the flange seal cavity. As the flange members are moved closer together, contact between the seal and the flanges is established at the edges of the seal disks. An increase in the axial force on the flanges causes the Belleville-type disks to rotate at the disk root radii and to take up the diametral clearance. Further deflection of the disks causes an interference fit at the sealing surfaces on the circumference of the seal disks, and as the deflection increases, the contact force on the sealing surfaces exceeds the compressive yield strength of the seal material and the ID of the seal becomes smaller. When the seal is fully assembled, the flanges bear against the seal tang, and continued application of the axial force needed for preloading is transmitted through the short cylindrical tang.



FIGURE 1. FOUR STAGES OF BOBBIN SEAL ASSEMBLY

Low helium leakage can be achieved reliably in a metal-to-metal seal only if plastic deformation is attained at the sealing interface. The disks of the Bobbin seal provide mechanical-force magnification to create a sufficient sealing stress to achieve plastic deformation at the interface while keeping the axial seal seating force to a minimum. Consequently a minimum-weight connector or housing can be obtained. The seal tang serves four important functions: (1) it resists the radial sealing force until the proper contact stress is established at the radial sealing interface; (2) it yields owing to the radial sealing force after the proper contact stress is reached, and, by yielding, maintains a relatively constant sealing contact stress as the flanges are brought into contact with the tang; (3) it facilitates the independent deflection and yielding of each seal disk, which may not occur simultaneously owing to variations in diametral clearance; and (4) it provides a loading path so that the connector is preloaded independently of the radial sealing force, and the force on the sealing surfaces is unaffected by changes in the external loads on the connector.

### Corrosion Compatibility of Seal Plating Materials

The yield strength of the surface of the circumference of the seal disks determines the radial sealing force that must be developed. If possible, the basic seal material should be softer than the flange material, thus circumventing the cost and quality control problems associated with plating. This was accomplished for 6061-T6 aluminum connectors by the use of overaged 6061-T6 aluminum for the Bobbin seals. However, when both the seal and the flange materials are soft, as is the case with Type 347 stainless steel, or when a high-strength seal material is required, plating should be used on the seal.

Although soft nickel has been used almost exclusively as the plating material for stainless steel during the development of the AFRPL threaded and flanged connectors, nickel is not compatible with all missile fluids. Therefore, as a part of this program, a study was made to determine the optimum plating materials for stainless steel Bobbin seals for all typical missile fluid systems, and to consider the corrosion problems of the two structural materials of interest - 6061-T6 aluminum, and Type 347 stainless steel.

Table 1 is a list of candidate plating materials. In the first group are common materials for which considerable propellant-compatibility data exist. The second group comprises the precious metals for which some compatibility data exist. The third group consists of exotic metals for which little or no compatibility data exist.

Metal	Hardness	Melting Point, F
	Group 1	
Nickel, ann.	64(a)	2635
Copper, ann.	40(a)	1980
Lead	4(a)	62.0
Tin, ann.	7(a)	<b>4</b> 50
	Group 2	
Gold, ann.	25(a)	1945
Silver, ann.	25-35(a)	1760
Platinum, ann.	38-52(a)	3225
Tantalum, ann.	150(b)	5425
	Group 3	
Palladium, ann,	40(b)	2825
Rhodium, ann.	122(b)	3560
Cerium	31(b)	1480
Lanthanum	51(b)	1690
Praseodymium	45(b)	1685

### TABLE 1. CANDIDATE METALS FOR PLATING STAINLESS STEEL BOBBIN SEALS

(a) Brinell Hardness Number.

(b) Vickers Hardness Number.

Preliminary Plating Selection. The following conclusions were reached during the initial stages of plating evaluation and selection:

- (1) Tin appeared to be the only metal that might exhibit reasonable compatibility with all candidate propellants. However, its resistance to  $N_2O_4$ ,  $H_2O_2$ , and  $F_2$  is questionable, and it has a low melting point.
- (2) Tantalum appeared to have excellent compatibility with all fuels and oxidizers except fluorine-containing oxidizers, but plating experience with tantalum is very limited.
- (3) Copper, nickel, lead, and silver show poor compatibility with some of the candidate propellants. Only copper and nickel have outstanding compatibility in some propellants, and nickel is preferred over copper.
- (4) Gold and platinum have good resistance to several fluids.
- (5) The plating materials selected on the basis of the preliminary studies were: gold, platinum, and nickel.

<u>Corrosion Resistance</u> <u>Candidate Materials to Fluids</u>. The corrosion ratings for the tentatively selected plati materials and for the materials of construction are given in Table 2. The fluids are separated in groups: oxidizers, propellants, and products of combustion. Specific compatibility data were not available for all metals exposed to all of the media in each temperature range. In some instances, the ratings in Table 2 are based on the performance of similar alloys, or the performance of the alloy in similar media. A combination of metal and environment is listed as questionable when insufficient data are available from which to make a good estimate or when there are discrepancies in the available data.

Nickel is suitable for chlorine and fluorine-containing media in the medium and upper temperature ranges, and for two of the hydrazine propellants. Nickel is not compatible with  $H_2O_2$  or RFNA, and it is not compatible with the nitric acid formed in water-contaminated N<sub>2</sub>O<sub>4</sub>. Gold is compatible with RFNA and N<sub>2</sub>O<sub>4</sub>. However, no candidate plating material is compatible with H<sub>2</sub>O<sub>2</sub>, and all plating materials are questionable for N<sub>2</sub>H<sub>4</sub>, MMH, MAF, and MHF.

Aluminum (6061-T6) is compatible with all the fluids except HC1 and possibly HF which are products of combustion. Use of aluminum is limited to 200 F systems. The compatibility of Type 347 stainless steel with most of the environments being considered is excellent. In the temperature range up to 200 F there exists a susceptibility to stresscorrosion cracking in environments containing chlorides. In the temperature range up to 600 F, stainless steel can handle the fluorine-containing fuels, but at higher temperatures these fluids are expected to cause rapid attack. However, since the connectors will be subjected to the medium and higher temperatures for only limited periods, the use of stainless steel should be satisfactory even at 1200 F.

Galvanic Attack. The problem of galvanic attack is particularly important because of the 5-year storage requirement. Various combinations of coatings and materials of construction were considered. In general, the use of aluminum seals in a stainless steel connector or of nonaluminum seals in an aluminum connector should be avoided because the galvanic couple will accelerate the corrosion of the aluminum. Of course, this is true only with those fluids that will serve as an electrolyte.

The galvanic couple created by Type 347 stainless steel in contact with nickel is not considered to be a significant factor affecting the 5-year-storage requirement. There is some doubt about the gold-stainless steel combination, although this combination should also be acceptable with fluids that do not act as electrolytes.

<u>Conclusions</u>. On the basis of the work conducted, it was concluded that nickel should be used if possible. Gold should be used when nickel is questionable and gold is acceptable. For those applications where neither plating is satisfactory, it will be necessary to use unplated stainless steel seals; this will result in some reduction in sealing reliability.

#### Preliminary Design of Large-Diameter Seals

The design of large-diameter Bobbin seals began with the consideration of Bobbin seal parameters based on work with threaded-connector seals. These parameters

TABLE 2. COMPATIBILITY OF TENTATIVELY SELECTED PLATING MATERIALS AND MATERIALS OF CONSTRUCTION WITH VARIOUS FLUID MEDIA

Temperature									£	the film of the second s					Gaset	and Pro	ducts o	3	- <u>8</u> 1	mbusto
Range and Allows	N2O.	H <sub>2</sub> O <sub>2</sub>	RFNA Oxidia	El 2	CLF3	CLF5	N2H4	HIMOU	50/50	HMM	MAF	E.	H <sup>2</sup>	H <sub>2</sub> O	So So	8	N N	\$1	≖	He
-23 to 200 F												I	,	(	c	ţ	,	Ċ		
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Platimm	000	N C	υυ	0.0	00	υ U	0 U	<del>0</del> 0	Qυ	0 U	<u>o</u> 0	ວບ	ງ ບ	י ט נ	່ວບ	, o o		οc		00
6061-T6 aluminum 347 stainless stoel	50	ט כ	ט נ	<b>v</b> v	00	o o	U	U	U	υ	υ	U	o	υ	υ	د	ر	נ		>
-423 to 600 F													(	C	ţ	ر	Ĺ	C		с С
Nickel	( <b>1</b> )	ei (	4	υÇ	υC	υο			* *				υυ	י ט כ	יטכ	000	ουί	, o c		000
Gold Platinum	<b>0</b> 0	• •	. 4		00	00	at 1	•		<b>1</b> 5 <b>1</b> 1	-c -s	at et	υυ	υυ	υυ	ას	ງບ	ຸບ		, U
347 stainless steel	0	4	4	υ	J	ر	6	•	•	l										
600 to 1200 F													ر	Ľ	C	U	U	υ		υ
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Gold Platinum	20	. 4	•	N N	¥	Ň	٩	•	•	•	<b>A</b> A	n, 1	υυ	ט נ	ט כ	່ວບ	2 2	0		ົບ
347 stainless steel	¥	-	4	N N	ž	U N	**	4	•	•	L	i							- 11	

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C = compatible
NC = not compatible
Q = questionable
Q = media decomposes owing to temperature.
(a) The nitric acid formed by water contamination of N2O<sub>4</sub> is contosive to nickel.

included (see Figure 1) the disk root radii, the sealing interface, the disk proportions, the initial and final angles of the seal disks, and the seal tang dimensions.

Disk Root Radii. During the previous two contracts, two types of problems were encountered at the roots of the seal disks. For certain combinations of disk thickness and tang strength, the compressive load on the seal disks was sufficient to cause a slight bulging of material at the root of the seal disks as the disks were rotated during seal assembly. This bulging, which was noticed primarily with aluminum seals, prevented proper mating of the flanges with the seal tang for preloading. As summarized in Technical Documentary Report No. AFRPL-TR-67-191, this problem was solved for smalldiameter aluminum seals by offsetting the seal disks slightly inward from the edges of the tang. This is illustrated in Figure 2. It was decided that this feature would be used in large aluminum and stainless steel seals. As shown in Figure 2, this offset required a socond radius at the root of each seal disk.



#### FIGURE 2. SELECTED MODIFICATIONS FOR SEAL DISK ROOTS

Another problem noted in the earlier work was the cracking of material at the inside corners of the seal disk roots. This seemed to be associated with the sharpness of the inside root radii, and the problem was overcome in small-diameter seals by making these radii larger, as shown in Figure 2. This approach was also selected for inclusion in the large-diameter seals.

Scaling Interface. As indicated in Figure 3, the initial contact at the seal interface occurs at the edge of the seal disk along a relatively narrow surface. Moreover, because of the rotation of the sealing disk, Angle B diverges, thus tending to reduce the contact surface even urther. In the small-diameter seals, the potential problem of a narrow sealing surface is overcome because the interacting stresses and plasticdeformation mechanism in these seal sizes causes the edge of the seal disk to deform in such a way that the seal surface width is increased. Figure 4 shows a typical seal interface for small-diameter seals. The deformation mechanism is the same for plated and unplated seals.



FIGURE 3. SEAL INTERFACE AT MOMENT OF CONTACT



FIGURE 4. TYPICAL SEAL INTERFACE FOR SMALL-DIAMETER SEALS

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Because it was expected that the disks of large-diameter seals would be designed to provide the same radial force per inch of seal circumference as the small seals, the cross sections of the disks were expected to be similar, and it was believed that the large seal disks would deform in a manner similar to that of the disks of the smalldiameter seals. However, during the preparation of the proposal for this contract, it was found that the disks of tentative 6-inch seals did not deform at the edges, and a very narrow sealing surface resulted. This action was confirmed during Phase II of this program by the fabrication and assembly of tentative 3-inch stainless steel seals. Although the reason for this change in seal disk deformation was not known, it was assumed to be characteristic of large-diameter seals and was believed to be associated with the radius of curvature of the seal disks.

Two means were considered for achieving an increase in seal width for largediameter seals. The first, an increase in radial force per inch of seal circumference, produced no significant change in seal disk deformation. The second is illustrated in Figure 5. Geometric analysis showed that the seal disk rotated approximately 6 degrees while the diametral clearance was being overcome. It was hypothesized that if the eventual sealing surface on each seal disk were machined with a reverse angle of 8 to 10 degrees as shown in Figure 5a, the flange and seal disk sealing surfaces at the moment of initial contact would be similar to that shown in Figure 5b. As assembly of the seal was continued, it was believed that the angle between the sealing surfaces would decrease to approximately 0 degrees, and that the high radial sealing force during the final stages of assembly would be sufficient to maintain seal contact across the entire disk sealing surface.



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#### FIGURE 5. MODIFICATION OF DISK SEALING SURFACE

(See Figure 3 for initial configuration.)

Stainless steel seals were fabricated according to the preliminary 3-inch design shown in Figure 6. Examinations of cross sections of assembled seals (Figure 7) showed that the seal interface was approximately 0.017 inch and that no cracking



FIGURE 6. PRELIMINAR DESIGN OF 3-INCH SEAL





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occurred at the inner disk root radii. The results were sufficiently conclusive to warrant incorporation of the reverse-angle feature for large-diameter seals.

As described in Technical Documentary Report No. AFRPL-TR-67-191, the reverse-angle feature was investigated as a means of possible improvement of the sealing action of small-diameter aluminum seals. Since no change in seal interface or sealing action could be observed, it was decided to omit this extra machining operation for seals for threaded connectors. However, the reverse-angle feature proved to be effective in helping to achieve satisfactory assembly of misaligned flanged connectors. Thus, if the misalignment-limitation feature developed during Phase III of this program and described in this report is not sufficient to prevent misalignment problems, the reverseangle feature might still be desirable for seals for threaded connectors.

Disk Proportions and Initial and Final Disk Angles. The selection of the proportions and of the initial and final angles for the seal disks is determined by a number of factors; the most important are the available axial force, the manufacturing tolerances to be accommodated, and the deformation characteristics of the metal. For the larger threaded connector seals, it was found that desirable values for the disk dimensions were: (1) initial disk angle, 30 degrees, (2) final disk angle, 10 degrees, (3) seal disk thickness, 0.027 inch, and (4) seal disk length, 0.090 inch. Although no change in radial force per inch of seal circumference was anticipated for the large-diameter seals, it was realized that the larger machining tolerances associated with the larger diameters required additional consideration of the disk proportions and angles.

Consideration of the initial disk angle led to the conclusion that an increase from 30 to 35 degrees would be desirable for the large-diameter seals. The 35 degrees was not believed to be too steep to prevent proper toggling action of the seal disks during assembly, and a 20-degree rotation from 35 degrees would provide a 33 percent greater diametral change than would a 20-degree rotation from 30 degrees. While this change in initial seal disk angle would increase the axial force required to seat the seal, the seal seating force needed in large-diameter connectors was estimated to be much less than the preload force needed to resist pressure and structural loading. (In contrast, for the smaller threaded connectors, the seal seating force was a significant percent of the required preload.) Satisfactory assembly was achieved with the 35-degree initial angle on the 3-inch seal design shown in Figure 6.

Although the best length of seal disk was difficult to estimate, it was apparent that the 0.027-inch thickness used on the small-diameter seals would be difficult to machine in large diameters. Consequently, the disk thickness was increased to 0.040 inch. On the basis of general disk proportions and machining tolerances, the following approximate disk lengths were selected: (1) 0.165 inch for tubing through 9 inches in diameter, (2) 0.230 inch for tubing through 12 inches in diameter, and (3) 0.305 inch for tubing through 16 inches in diameter. As described later, these dimensions were subsequently revised as a result of the qualification tests.

Seal Tang Dimensions. As described in Technical Documentary Report No. AFRPL-TR-65-162, the deformation mechanism of the Bobbin seal is a complex combination of elastic and plastic strains in the axial as well as the radial direction. Because the development of an accurate theoretical strain analysis would have been very costly, the design of threaded connector seals was based on a combination of theoretical and experiment.' information. It was decided that the same method would be used to establish tang dimensions for the large-diameter seals. The design approach used for the threaded- and flanged-connector seal tangs began with the following steps:

- (1) The radial sealing force per inch of seal disk circumference,  $F_S$ , was selected as 600 lb/in. (a discussion of this value is given on page 165).
- (2) The radial force per inch of outside seal tang circumference,  $F_T$ , at the root of each seal disk due to  $F_S$  varies as a ratio of seal disk outside diameter,  $D_O$ , to seal tang outside diameter,  $D_e$ , viz.,

$$F_{T} = F_{S} D_{o} / D_{e} .$$
 (1)

(3) The radial force per inch of seal tang outside circumference from the two seal disks,  $2F_T$ , can be equated to a pressure,  $P_T$ , on the outside circumference of the seal tang (this assumes that the effect of the two disk forces is distributed evenly across the short seal tang length)

$$\mathbf{P}_{\mathbf{T}} = 2\mathbf{F}_{\mathbf{T}} / \mathbf{L} , \qquad (2)$$

where L = seal tang length.

(4) The pressure needed to produce a totally plastic state in the seal tang<sup>(2)</sup> is

$$P_{T} = \frac{2}{\sqrt{3}} S_{y} C \ln (D_{e}/D_{i}) = 2F_{T}/L , \qquad (3)$$

where

 $S_v = seal material yield stress, psi$ 

C = 0.72 [from calculations by Beliaev and Sinitskii<sup>(3)</sup>]

 $D_i = scal tang inside diameter, in.$ 

 $D_e =$  seal tang outside diameter, in.

Thus, for a given inside or outside tang diameter and a given seal material yield strength, Equation (3) can be used to establish a number of seal tang dimensions that will produce the desired radial force per inch on the outside circumference of the seal tang.

The tang of the 3-inch seal shown in Figure 6 was established by using Equation (3) and by selecting the tang inside diameter to be approximately the same as the inside diameter for heavy wall tubing. Examination of the microsections, shown in Figure 7, showed that the nickel plating was less deformed than expected. However, it was decided that this basis of calculation should be used until more experimental data could be obtained.

Because it was not clear whether the optimum seal width of 0.020 inch (see page 165) could be achieved in the large-diameter seals, tentative tang dimensions were calculated for two widths of sealing surfaces, 0.015 and 0.020 inch. Also, in anticipation that a higher strength material might be required to keep the large-dismeter seals sufficiently small in cross section, tang dimensions were calculated for three typical yield-stress levels of candidate seal materials. The results of these calculations for three typical seal sizes are shown in Table 3.

Yield Strength of Seal Material, Sy, psi	Width of Sealing Sur- face, in.	Tang Inside Diameter, D <sub>i</sub> , in.	Tang Outside Diameter, D <sub>e</sub> , in.	Seal Disk Outside Diameter, D <sub>o</sub> , in.	Tang Length, L, in.
30,000	0.020	1,800	2.230	2.500	0,120
		3.585	4.430	4.700	0.410
		7.450	8.730	9.000	0.690
	0,015	1,900	2,230	2,500	0.165
		3,800	4,430	4.700	0.305
		7.745	8.730	9.000	0.525
60,000	0.020	2.000	2,230	2,500	0.110
		4.000	4.430	4.700	0.215
		8.070	8.730	9.000	0.330
	0.015	2,060	2,230	2,500	0.085
		4.100	4.430	4,700	0,165
		8.230	8,730	9.000	0,255
90,000	0.020	2,065	2,230	2,500	0.083
		4.140	4.430	4,700	0,145
		8,275	8,730	9.000	0.228
	0.015	2.115	2,230	2.500	0,058
		4.200	4.430	4.700	0,115
		8.395	8,730	9.000	0.168

TABLE 3. TENTATIVE TANG DIMENSIONS FOR THREE TYPICAL SEAL SIZES

In addition, the graphs shown in Figures 8 and 9 were prepared using the above mathematical analysis. From Figure 8 it can be seen that for a small tang length, the use of a material with a yield strength of 60,000 to 90,000 psi will greatly reduce the size and weight of the larger seals and the size and weight of the larger connectors. It can be seen from Figure 9 that an increase in seal tang length results in a reduction in the outside diameter of the seal, thus reducing the diameter and weight of the connector. However, the weight of the seal remains high if a material yield strength of 30,000 psi is used. Further, if the L/t ratio for the tang is too great, the assumption that the radial disk forces can be equated to a uniform external pressure on the tang is no longer valid. A judgmental compromise between a reduction in tang thickness and a limitation on tang length was approximated by the expression:

$$L = K (tD_i)^{1/2}$$
, (4)

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FIGURE 8. TANG THICKNESS VERSUS TANG INSIDE DIAMETER FOR A SMALL TANG LENGTH AND THREE SEAL-MATERIAL YIELD STRENGTHS



FIGURE 9. TANG THICKNESS VERSUS TANG INSIDE DIAMETER FOR THREE TANG LENGTHS

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L = tang length, in.

K = 0.250

t = tang thickness, in.

 $D_i$  = inside tang diameter, in.

Equations (3) and (4) form a set of simultaneous equations which can be solved through iteration on a digital computer. The tang calculations for typical yield-strength values are shown in Table 4.

As the program developed, it became increasingly obvious that the seal tang would often have to be as long as possible to achieve the lightest connector. The reduction in tang thickness as a function of an increase in tang length was computed by increasing K in Equation 4. These data are shown in Table 5. To investigate the practical limits of seal tang length, seals were fabricated from overaged aluminum and from Type 304 stainless steel for the 2- and 4-inch sizes with various L/t ratios. Assembly data for these seals are given in Table 6.

Examination of the microsections of the stainless steel seals showed that bowing of the tang was negligible at an L/t ratio of less than 2.5 for 2-inch seals and 3.5 for 4-inch seals. Typical cross sections are shown in Figures 10, 11, and 12. The bowing in the 2-inch seal even at an L/t ratio of 3.80 was not too great. Although the aluminum seals did not deform uniformly because of faulty material, it was apparent that an L/t ratio of 3.50 would be satisfactory.

On the basis of the L/t tests, it was concluded that the tang could be designed with a maximum L/t ratio of 3.50 for seal sizes at least as large as 4 inches, and probably larger. However, it was decided that a tang thickness equal to the tube wall thickness would be specified, except for thin-wall tubing (low pressure systems) and for L/t ratios greater than 3.50.

#### Evaluation of Preliminary Large-Diameter Seals

Preliminary, representative large-diameter seals were evaluated by deformation and leakage tests using techniques formulated during the development of threadedconnector seals.

Deformation Evaluation of 2- and 4-Inch-Diameter Seals. During work with seals for threaded connectors, it was found that the sealing capabilities of the seals could be reasonably estimated by means of the maximum axial loads needed to assemble the seals, and by examinations of cross sections of the seal interface (for example, see Figure 7). Similar studies were made with 2- and 4-inch nickel-plated austenitic stainless steel seals with dimensions approximately those shown for a material yield stress of 30,000 psi in Table 3. The actual tang dimensions, the peak axial loads required to assemble each seal, and the calculated minimum force per inch of seal circumference are shown in Table 7. TABLE 4. COMFUTED DIMENSIONS OF SEAL TANGS FOR K = 0.25 FOR DIFFERENT SEAL-MATERIAL YIELD STRENGTHS

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D <sub>i</sub> , in.	S <sub>y</sub> , pei	, n.	L, in.	D <sub>i</sub> , in.	s <sub>y</sub> , pei	D, in.	L, in.	D <sub>i</sub> , in.	S <sub>y</sub> , psi	De, in.	L, in.
1000-1	3000,0000	<u>1.4457</u>	0,1305	1000°7	3600.0000	K.7785	6.3943	11.0000	3000 0000	2000	
1.0000	4000°0000	].3646	0.1141	4.0100	4000° 0000	64n5	0.3557	1	4000.00004	7277.11	5 50 50 5 50 50 5 50 50
1.0000	0000 00009	1.3122	0.1067	· • • • • • • • • • • • • • • • • • • •	5000.0000 	6.5507	0.32FK	1:.000	50100.0000	11.6709	0.4875
0000 *	70006.0000	1.2475	~ E 60 ° 0		7000 0000	1.4200 1.4200		00000-11	50000 00000 70000 0000	11.5934	0.4577
1.0000	80000.0000	1.2258	0.0886	6.000	80000.0000	1104.9	0.2788	11.0000	8000-0000	0.554.0	0.4340
1.0000	9000.0000	1.2042	0.0848	4.0000	9000.00006	6.37n5	0.2676	11.0000	9000°00006	104401	0.4]46
1.0000	100000.0000	1.1937	0.08]5	4.000	10000.0000	6.3451	0.25Rn	11.0000	100000.0000	114-11	
2.0000	3000°0000	2.5505	0.1978	7.0000	30000-0000	7.8180	0.4352	12.0000	30000.0000	AF79.51	0.6164
		1.40°.	51110	1.0000	4000,0000 5055, 5006	7.6732	0.3979	0000.01	4000.0000	12.8021	0.5575
2-0000			1501.0	7 0000		1.5740	0.3537	12 2000	5000,0000 50000,00000	12.69.21	0.5160
2.0000	70000.0000	020M .		7.000	7000.0000	7-6614	0.9220		70000 0000	12.61.66	0.4945
2.0000	96000.0000	1185.4	0.1371	7.0000	8000.00008	7.4218	0.3083	12.000	80000°0000	12.55.55	0.4595 0.4305
2-0100	9000.0000	2.2595	4161.0	7.0000	0000.00006	7.38e6	0.2961	12.000	9000.0000	12+4451	
2.0000	100000.0000	2.2416	0.1265	7.000	10000.0000	7-3630	0.2854	12.0000	10000.0000	12.4334	0.4057
		7479.5		F.000C	30000.00005	8.8540	0.4742	0000 11	30000-0000	13.9905	0.6493
3.0000	50000.0000			9.000	4000.0000	8.7070	0.4283		+0000 0000	13, 9233	0.5874
0000 E	6000.0000	FORE F	0.1971	R.0000	50000.0000	A.6147	0.3961	0000.51		13.70A6	0.543A
5. a0 00	7000.0000	3.3506	0.1865	H.0000	6000.0000 70000.0000	R.5347	7176.0	15.0000	70000-0000	13.6268	0.5107
3.7700	80000.0000	3.3212	0.1778			R.4870	F525.0	13.0000	8000,0000		
0000 E	90000.0000	3,2957	0.1704		9000°00008		4099°D	13.0000	9000.00009	13.4775	
0000*5	10000.0000	3.2754	P.1643	0000.4	10000.0000	8.3792		13.0000	10000.0000	13.4449	0.4288
4.0000	3000.0000	4.6841	0.3047	0000	3000.0000	9.8871	0.5117	14.0000	30000.0000	15.0239	0.6814
		5792.4	0.2743	9.000	4000.0004	9.73n4	0.4623	14.000	4000.0004	14.A435	0.6166
0000 •			1562.0	9.000.9	50000.0000	9.62A3	0.4276	1.0000	0000-00005	14.7260	G.57AA
0000 *	7000.0000	1010101 4 - 3847	542200	00000	60000-0000	9.5557	0.4014	14 0000	5000.0000 70000 0000	14.6473	0.5361
4.0000	B000.0000	4.3515	0.2142	0000.6	1000.0000	9.5010	0.3805	14.0000	80000-0000	[0/C.*]	0.5085
4-0000	9000.00009	4.3246	0.2055		0000°0000	9.4579	0.3634	14.0000	90000.0000	(+-)/97	0.4103 4444
4.0000	10000.0000	4.3073	0.1980	0000-6	10000-0000	0,304.0	0.440.0	14 0000	10000 0000		
0000 .	1000 . NOUNE	5.7344	0.3510	10.000	3000,0000	10.9179	0.5477	15.000	30000.0000		0.400
5,0000 5,0000		5.6040 5.6040	0.3163	10.0000	40000-0000	10.7558	0.4951	15.0000	40000.0000	5.8478	0.6450
*******	6000°0000	5.4589	0.2730	10.000	50000,0000	10.6503	0.4581	15.0000	5000.0000	15.7476	0.5973
5.0000	70000.0000	5.4135	0.2594		60000.0000 70000 0000	10.5752	0.4300	15.0000	60000.0000 70000 5000	15.6570	0.5610
5.000	Bn000,0000	5.3779	0.2475								0.5322
5.0000	9000.0000	5.3491	0.2376	10.000	9000-0000	10.4380	0.3740	15.0000	9000,0000	14000	4×12.0
nhau*c	10000*00001	5,3251	0.2290	10.0000	100000.0000	10.4081	0.3607	15.0000	10000.0000	15.4664	5174-0
								16.0000	3000.0000	17.0695	0.7434
								16.0100	4000.00004	16.8812	0.6729
								14.0000	50000.0000	16.7585	0.6231
								16.0000	60000.0000	16.6711	0.5853
								16.0000	8000,0000	16.6052	0.5553
								16.0000	9000.0000	16.5113	0.5097
								14.0000	100000.0000	16.4765	0.4917

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D <sub>i</sub> , in.	D <sub>e</sub> , in.	t, in.	к
1	1.4460	0.1306	0.2500
3	2.5509	0.1979	0,2500
3	3. 6251	0.2544	0.2500
4	4.6846	0.3048	0.2500
5	5.7349	0.3511	0.2500
6	6.7791	0.3944	0.2500
	7.8186	0.4354	0.2500
5	8.4544	0.4744	0.2500
1	1 3837	0 14H2	0 3100
	2 6 /50	0 2260	0.3100
	3 5 306	V_LLU	
	A 6912	0 3403	0 3100
	8 6 740	0 40 20	0 31 0 0
-	5,0347	0.4020	0.3100
	7 70736		
	1,7010	0,5000	0.3100
	-0-(JOH_	0_0451	
,	1.3331	0,10/5	0.3404
. <u> </u>	2 0 1 30	0.2303	0.3000
-1	3,4695	0,3311	0.3400
	4.5197-	0,3978	0_3800
2	5,5730	0,4590	0.3400
	6.5865	0_5162	0_3000
	7,6166	0,5704	0,3400
<u> </u>	8.6439	0.6220	0.3800
1	1.3009	0,1830	0.4400
<b>č</b>	_ <b>2,3/</b> 35_	0_2812	
3	3,4249	0,3635	0,4400
	4.4660_	0_4370	0.4400_
5	5,5007	0,5045	0,4400
	6.5312	0.5676	0.4400
7	7,5584	0.62/3	0.4400
<u> </u>	R_5833	0.6842	0.4407_
1	1,2754	0_1979	0,5000
_2	2.3423_	0.3048	0.5000
3	3,3895	0.3944	0,5000
4	4.4273	0.4744	0.5000
5	5,4593	0.5479	0,5000
. 6	6.4872	0.6167	0.5000
- 7	7.5123	0.6817	0.5000
8	8.5351	0.7436	0.5000

TABLE 5.	COMPUTED DIMENSIONS OF SEAL TANGS FOR D 1 THROUGH
	HINCHES USING DIFFERENT VALUES OF KISy - 10,000 PSI)

D., in.	2., in,	L, in.	k k
		·	
1	1.2547	0,2122	0.5600
2	2.316A	0.3274	0.5600
3	3,3607	0,4241	0,5600
4	4.3957	0.5:04	0.5600_
5	5,4254	5,5897	0.5600
6	6.4514	0.6638	0.5600
	7.4746	0.7339	0.5000
8	8.4958	0.0007	9.5600
1	1.2375	0.2260	0.6200
2	2.2954	0.3493	0.6200
3	3.3366	0.4528	0.6200
4	4.3694	0.5451	0.6200
5	5.3972	0.6299	0.6200
6	6-4214	0.7093	0.6200
7	7.4432	0.7843	0.6200
<u>H</u>	A.4630	0.8558	0.6200
1	1.5509	0,2415	0.4900
2	2,2/48	0,3739	0.6900
٤	3,3131	0,4850	0.6900
4	4,3436	0.5842	0.6900
5	5,3695	0,6753	0,6900
6	6.3921	0,7605	0.6900
7	7.4124	0.8411	0.6900
8	8,430A	0,9179	0.6900
1	1.2084	0,2544	0.7500
2	2,2597	0,3944	0.7500
3	3.2959	0.5118	0.7500
4	4,3248	0,6167	0.7500
5	5,3193	0,7130	0.7500
6	6.3707	0.8031	0.7500
	7.3899	0.8883	0.7500

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TABLE 6. ASSEMBLY DATA FOR STAINLESS STEEL AND ALUMINUM SEALS WITH DIFFERENT L/t RATIOS

Seal Size		Tang	Tang	т /с	Arrial I	Some lb	Axial Seating Force (Peak 1)
Material	Specimen	t, in,	Length, L, in.	Ratio	Peak 1	Peak 2	Circumference, 1b/in,
2-in,	1	0,229	0,215	0.939	5,676	6,000	708
aluminum	Z	0,199	0,239	1,201	5,496	5,880	688
	3	0,179	0.270	1.508	5,760	5,784	720
	4	0,154	0,296	1,922	5,460	5,520	682
	5	0,130	0. 322	2, 477	5,124	5,488	642
	6	0.102	0.352	3,451	4,584	4,872	572
4-in,	7	0.306	0,332	1.084	9,960	10,200	702
aluminum	12	0,123	0,538	4,374	7,560	8,400	532
2-in,	13	0,235	0,213	0.906	7,560	-	932
stainless	14	0.200	0.246	1,230	7,120	7.200	880
	15	0.180	0.269	1.494	7,475	7,800	922
	16	0,153	0.294	1,922	7,080	7,200	872
	17	0.128	0,318	2.484	6,730	6,820	830
	18	0,098	0.378	3,857	6,240	6,480	770
4-in,	19	0,268	0,330	1,231	11,000	11,500	775
stainless	20	0.230	0,380	1.652	11,400	11,750	803
	21	0,205	0.414	2,020	10,750	10,650	758
	22	0,173	0.454	2,624	10,250	11,600	723
	23	0.138	0,494	3,580	10,000	10,450	704
	24	0, 101	0,541	5,350	9,000	9,200	633



9A681

FIGURE 10. 2-INCH STAINLESS STEEL SEAL L/t = 3.857

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FIGURE 11. 4-INCH STAINLESS STEEL SEAL

L/t = 3, 580



9**A**685

9**A** 684

FIGURE 12. 4-INCH STAINLESS STEEL SEAL



Specimen	Seal Out- side, Diam., D <sub>o</sub> , in.	Tang In- side Diam., D <sub>i</sub> , in.	Tang Thick- ness, t, in.	Axi11 F Peak 1	Peak 2	Axial Seating Force (Peak 1) per Inch of Seal Cir- cumference, 1b/in.
1	2,505	1,920	0.155	5,225	5,775	734
2.	2,508	1.820	0.205	6,975	7,725	980
3	4,709	4.020	0.205	8,250	10,350	700
4	4,709	3,820	0.305	7,950	10,000	676
5	4,700	3.622	0.405	10,800	13,250	897

TABLE 7. DIMENSIONS AND MAXIMUM AXIAL ASSEMBLY FORCESFOR NICKEL-PLATED STAINLESS STEEL SEALS

The maximum axial forces per inch of seal circumference corresponded well with the forces required on threaded-connector seals to seal helium to  $10^{-7}$  atm cc/sec. Examination of the seal interfaces for the different specimens led to the following conclusions:

- Except for Specimen 1, all seals had a seal interface width close to 0.020 inch (all specimens had a sealing surface machined on a 10-degree reverse angle on each seal disk).
- (2) A higher force per inch of seal circumference resulted in Specimen 2 having a better seal interface than Specimen 1.
- (3) A higher force per inch of seal circumference resulted in Specimen 5 having a better seal interface than Specimens 3 and 4.

Leakage Evaluation of 2-, 4-, 8-, 12-, and 16-Inch-Diameter Seals. Because of the promising results of the deformation tests with the 2- and 4-inch-diameter seals, the decision was made to conduct leakage tests with representative seals. Overaged aluminum was selected as the seal material for major use because it is easily machined, it requires no plating, and its sealing action is believed to be similar to that of nickelplated stainless steel. The dimensions for 4-, 8-, 12-, and 16-inch-diameter seals and the seal seating forces are given in Table 8.

The peak seal seating forces per inch of seal circum/erence were almost 50 percent greater than the average 450 lb/in. of seal circum/erence achieved for aluminum seals for threaded connectors. Unfortunately, the construction of the leakage test fixture did not provide for adequate restraint of the deflection of the fixture because of pressure end loads. However, on the basis of the adequate sealing of the seals up to the different pressures (2000 psi maximum) at which deflection caused leakage, it was concluded that the seals were satisfactory. It was also decided that the higher axial force per inch of seal circumference (750 lb/in.) would be retained for the aluminum seals for flanged connectors.



#### TABLE 8. DIMENSIONS OF ALUMINUM SEALS FOR LEAKAGE TESTS

Note: All dimensions in inches,

Because soft nickel plating has approximately one-third the yield strength of aluminum, and because nickel-plated stainless steel seals were apparently performing satisfactorily in threaded connectors with a design axial seal seating force of 600 lb/in. of seal circumference, a design value of 750 lb/in, of seal circumference was also selected for stainless steel seals for flanged connectors. A 2-inch nickel-plated stainless steel seal with a measured peak axial seal seating force of 768 lb/in, of seal circumference gave a leak rate of 6.6 x  $10^{-9}$  atm cc/sec with a helium pressure of 2000 psi.

Determination of Minimum Seal Load. The design of the Bobbin seal substantially separates the axial preload from the radial sealing load. However, some residual axial load is required and tests were conducted with aluminum seals to determine the reduction in axial seal load that could be tolerated without excessive seal leakage. Representative overaged 6061 aluminum seals were assembled and pressurized with helium at 2000 psi. In initial preload equal to the pressure end load plus one-half the seal seating load was applied. Once a leakage rate (Column 4 in Table 9) was established, the applied external load was decreased in increments until the leakage rate exceeded the allowable limit. The difference between the threshold load (Column 5) and the pressure end load was the minimum axial seal load.

For the 2-inch seal, the minimum axial seal load was 31 percent of the seal seating load, whereas for the 6- and 8-inch seals, the minimum loads were 45 and 41 percent, respectively. In the case of the 4-inch seal, the external load was actually less than the pressure end load before an increase in leakage rate could be detected. As a general rule for designing the connector, it was decided that the residual axial load should not be less than 50 percent of the axial seal seating load.

Nomina) Seal Size, in.	Axial Seal Seating Load, lb.	Total Initial Axial Load, lb	Initial Leakage Rate at 2000 Psi, atm cc/sec/in. of seal circumference	Load at Allowable Leakage Rate, lb	Pressure End Load, lb.	Minimum Axial Seal Load, lb
2	5,500	11,650	9.3 x 10 <sup>-10</sup>	10,600	8,900	1700
4	9,500	36,250	$4.6 \times 10^{-10}$	28, 500	31,200	-2700(a)
6	13,500	73,000	$3.2 \times 10^{-8}$	70,000	64,000	6000
8	16,000	134,000	$2.2 \times 10^{-10}$	119,000	112,400	6600

TABLE 9. MINIMUM AXIAL SEAL SEATING LOADS FOR ALUMINUM SEALS

(a) This value indicates that the final external force was less than the pressure end force.

#### Development of Computerized Seal-Design Procedure

Seal-Design Procedure for Qualification Tests. On the basis of the results from the seal-development work, the following step-by-step procedure was derived to design the seals for the connectors used in the qualification tests. The tube inside diameter was calculated in the tube-design section of the computerized connector-design program. In the seal-design section of the program, the seal inside diameter was made equal to the tube inside diameter plus 0.04 inch (to allow for seal contraction) for tube sizes through 8 inches in diameter, or to the tube inside diameter plus 0.06 inch for tube sizes greater than 8 inches. As shown by Curve 1 of Figure 13, the seal disk height was calculated as a step function of tube size. The seal disk thickness was made 0.04 inch for all tube sizes.

The seal tang thickness was first made equal to the tube wall thickness, and Equation (3) was used with the appropriate material yield stress and the required radial force to calculate a tang length. The ratio of seal tang length to seal tang height was checked to determine that it was not greater than: (1) 3.5 for tube sizes up to and including 3 inches, (2) 2.5 for tube sizes up to and including 8 inches, and (3) 2.0 for tube sizes greater than 8 inches. If the calculated ratio was greater than allowable, the tang thickness was increased by steps of 0.005 inch and the calculation of seal tang length was repeated until a satisfactory ratio was obtained. The previously established seal disk dimensions were then used with the selected tang to determine the final seal design.

Revised Seal-Design Procedure. As discussed on pages 144 through 146, the qualification tests showed that the seal design procedure had to be revised for three reasons. First, the selected disk thickness of 0.040 inch had to be increased because of the pressure-impulse requirements of the high-pressure aluminum connectors. Second, the disk height of the larger, high-pressure stainless steel seals had to be increased to provide more elastic recovery for thermal-gradient requirements. Third, the radial force per inch of seal circumference for all stainless steel seals, and in particular for the high-pressure seals, had to be increased to provide better sealing



FIGURE 13. SEAL DISK HEIGHT VERSUS TUBE SIZE

contact to overcome imperfections and manufacturing variations in the nickel surface of the seals. During consideration of these requirements, it also became apparent that shorter disk heights were needed for the seals for tubing less than 3 inches in diameter.

The design procedure was revised to begin with the calculation of seal disk height as a linear function of tube size. As shown by Curve 2 in Figure 13, the linear variation was made to approximate the disk height of the 3-inch qualification-test seals because of the satisfactory thermal-gradient performance of the 3-inch stainless steel connectors. The seal disk height for the 1-inch size was based on engineering judgment and on seals used under Contract AF 04(611)-8176 for threaded connectors. For the larger seals, the seal disk height was made as great as possible without increasing the basic connector structure as determined by the operating conditions and the material properties. Following the seal-disk-height calculation, the inside diameter of the seal was made equal to the tube inside diameter plus an allowance for seal contraction proportional to the seal disk height.

The seal tang thickness was initially made equal to 0.9 times the tube wall thickness, and a tang length was calculated using Equation (5a):

$$L = 2F_{T} / [S_{v} \ln (D_{e}/D_{i})], \qquad (5a)$$

where

L = tang length, in.

- $F_T$  = radial force per inch of outside seal tang circumference at root of each seal disk, lb/in. [FT was made equal to 750 lb/in. for all aluminum seals, 1200 lb/in. for low-pressure stainless steel seals, and 2500 lb/in. for high-pressure stainless steel seals.]
- $S_v$  = seal material compressive yield stress, psi
- $D_e$  = tang outside diameter, in.
- $D_i$  = tang inside diameter, in.

On the basis of additional consideration of allowable bowing in the seal tang, the maximum allowable ratio of seal tang length to seal tang thickness was set at 4 for all tube sizes, and an iterative check (see page 23) was performed on the calculated ratio until a satisfactory ratio was obtained. Equations (5b) and (5c) below show the calculations derived for seal disk thickness. The maximum of the two values obtained was the designed disk thickness. These equations were based on the successful performance of a seal disk thickness of 0.060 inch in the pressure-impulse test of the 8-inch aluminum connector, and on the judgmental application of beam-strength considerations to other disk sizes.

$$LT = F_T D_O / D_S S_V , \qquad (5b)$$

and

$$LT = 0.04 + 0.001 D \sqrt{P/12}$$
, (5c)

where

- LT = seal disk thickness, in.
- $D_0$  = seal disk outside diameter, in.
- D = tube outside diameter in.
- P = operating pressure, psi,

#### Investigation of Connector-Design Criteria

Several criteria must be considered in the design of separable connectors for large tubing systems. An investigation was made of these criteria prior to the development of a computerized, connector-design program. In addition to the seal development work described previously, this work included: (1) design approximations for conventional flanged connectors; (2) investigation of nonconventional connector configurations; (3) comparison of conventional and nonconventional connectors; (4) investigation of connector thermal gradients; (5) investigation of bolt parameters; and (6) investigation of stress relaxation considerations.

#### Design Approximations for Conventional Flanged Connectors

Conventional connectors include either integral or loose-ring flanges fastened together by bolts. These configurations are illustrated in Figure 14. The important factors that determine the physical dimensions and weight of conventional connectors



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include the (1) design conditions, (2) seal dimensions, (3) material properties, (4) tubewall thickness, (5) connector loads, (6) bolt strength, and (7) wrench clearance. Each of these factors is discussed briefly, and approximate comparative weights are given for conventional integral and loose-ring connectors.

Design Conditions. On the basis of the requirements of missile systems as defined by the work statement, five combinations of temperature, pressure, material, and tube size were selected for design approximations. These are listed in Table 10. Some design penalties were imposed for the low-pressure system because the computed tube-wall thickness for pressure retention (for smaller tube sizes especially) was not great enough for normal handling or fabrication of the tubing system, and heavier tube-wall thicknesses required a heavier connector to withstand the higher bending moment loads.

Tube Diameter, in.	System Pressure, psi	Maximum Temp, F	Material
1 to 16	100	200	Type 347 SS
1 to 16	1500	200	Type 347 SS
1 to 16	1500	200	6061-T6 Al
1 to 3	4000	600	Type 347 SS
1 to 3	6000	200	Type 347 SS

TABLE 10. CONDITIONS FOR DESIGN APPROXIMATIONS

For the design approximations, no consideration of thermal gradients was included in the load and/or stress analyses. However, on the basis of concurrent laboratory measurements of thermal gradients (see page 56), it was decided not to use steel bolts with aluminum flanges.

Seal Dimensions. As shown in Figure 15a, if the seal OD is relatively large, the bolt-circle diameter will be larger than that required to accommodate wrench clearance at the bolt head. If the seal OD is small and the seal length is excessive (Figure 15b), then the length of the bolt may be longer than necessary. It is desirable, therefore, to select an "optimum" seal configuration (Figure 15c) which will not penalize the flange design. For comparison purposes, the seal dimensions selected were based on a seal tang thickness equal to the tube-wall thickness. The other dimensions of the seal were then selected to satisfy seal-performance parameters as discussed previously.

Material Properties. To provide a common basis for stress and weight calculations, values were selected for the minimum strength properties of the connector materials. Some of these values are listed in Table 11. For 6061-T6 aluminum connectors, the bolt material selected was 2024-T3 aluminum, while for stainless steel connectors, the bolt material selected was heat-treated, high-strength A286. Also, for analysis purposes, factors of safety of 1.5 for aluminum materials and 1.25 for steel materials were selected.







b. Tang too Long





# FIGURE 15. EFFECT OF SEAL TANG ON CONNECTOR DESIGN

	6061-T6 Aluminum	2024-T3 Aluminum	Type 347 SS	High-Strength A286
Modulus of Elasticity, 10 <sup>6</sup> psi	9.9	9.9	28.0	29.0
Density, 1b/in. <sup>3</sup>	0.098	0.098	0.288	0.286
Room Temperature (70 F)				
Tensile Yield Strength, psi Ultimate Tensile Strength, psi	35,000 42,000	50,000 62,000	35,000 90,000	131,000 200,000
200 F				
Tensile Yield Strength, psi Ultimate Tensile Strength, psi	32,200 38,100	47,000 59,000	30,000 76,500	128,000 196,000
600 F				
Tensile Yield Strength, psi Ultimate Tensile Strength, psi	-	-	25,000 68,000	120,000 180,000

# TABLE 11. MATERIAL MINIMUM-STRENGTH PROPERTIES SELECTEDFOR CONNECTOR COMPARISONS(a)

(a) Some of these properties were revised in later designs.

<u>Tube-Wall Thickness</u>. The tube-wall thicknesses selected for this study were calculated by the following formula from Section I of the ASA Code for Pressure Piping:

Tube-Wall Thickness = 
$$\frac{1.1 \text{ PD}}{2S_a + 0.8P}$$
, (6)

where

- S<sub>a</sub> = allowable stress of tube material at most severe operation condition, psi
- P = operating pressure, proof pressure, and/or burst pressure, psi
- D = tube outside diameter, in.
- 1.1 = factor applied for fabrication tolerance.

The most severe operating conditions that determined minimum tube-wall thickness were (1) proof pressure (one-and-one-half times maximum working pressure) for Type 347 stainless steel, and (2) burst pressure (two times maximum working pressure) for 6061-T6 aluminum. For the 100-psi conditions, a minimum wall thickness was selected on the basis of weldability and ease of handling. Table 12 shows the calculated tube-wall thicknesses.

		Tube-Wall 7	hickness, inc	hes	
Tube	6061-T6 Aluminum,	·····	<u>Type 347 S</u>	tainless Steel	
Diameter,	1500 Psi,	100 Psi,	1500 Psi,	4000 Psi,	6000 Psi,
D, in.	200 F	200 F	200 F	600 F	200 F
1	0.042	0.022	0.040	0.120	0.147
2	0.084	0.028	0.080	0.240	0.294
3	0, 126	0.035	0.120	0.360	0.441
4	0, 168	0.035	0.160		
5	0.210	0.050	0.200		
6	0.252	0,050	0.240		
7	0,294	0,050	0.280		
8	0.336	0.050	0.320		
9	0.378	0.050	0.360		
10	0.420	0.050	0.400		
11	0.460	0.050	0.440		
12	0,504	0.063	0.480		
13	0.546	0.063	0.520		
14	0,588	0.063	0.560		
15	0.630	0.063	0.600		
16	0.672	0,063	0.640		

## TABLE 12. TUBE-WALL THICKNESS CALCULATED FOR CONNECTOR COMPARISONS

<u>Connector Loads</u>. Three types of loads (pressure end load, tube bending load, and minimum residual seal load) combine in different ways to determine the operating conditions. The connector design axial load (TAL) was determined on the worst-case basis of:

- (1) Axial load due to proof pressure plus minimum seal load
- (2) Axial load due to operating pressure plus equivalent bending load plus minimum seal load.

The design axial loads (TAL) for each tube OD and combination of temperature, pressure, and material are shown in Table 13.

	Design	n Axial Load	(TAL), 1b	
Tube	6061-T6 Aluminum,	Туре	347 Stainles	s Steel
Diameter, D,	1500 Psi,	1500 Psi,	4000 Psi,	6000 Psi,
in.	200 F	200 F	600 F	200 F
1	3,951	3, 472	8,296	12,082
2	13,414	12,754	32, 942	45,083
3	30, 592	30, 958	59, 502	83, 480
4	47,868	46, 367		
5	64, 348	62, 525		
6	84,770	81,686		
7	107, 356	103, 814		
8	135, 187	128, 758		
9	165, 122	159,032		
10	204,083	194, 326		
11	248,840	241,760		
12	292, 546	284,865		
13	334, 785	331, 507		
14	397, 426	381,681		
15	452,201	434,853		
16	510,487	492,636		

TABLE 13. DESIGN AXIAL LOADS CALCULATED FOR CONNECTOR COMPARISONS

The pressure end loads at operating and proof pressure conditions were calculated:

$$PL = \frac{\pi P(D_o)^2}{4}, \qquad (7)$$

where

ŧ

PL = pressure end load, lb

P = operating pressure or proof pressure, psi

 $D_0$  = seal outside diameter, in.

The tube bending moment was calculated on the basis of Equations (8) and (9) and the lower value was selected. The static bending moment due to misalignment at assembly was defined as half of the tube bending moment.

$$TBM1 = Z S_A , (8)$$

$$TBM2 = 60 (D + 3)^3 , (9)$$

where

TBM1 and TMB2 = tube bending moment, 1b-in.

Z = tube section modulus, in.<sup>3</sup>

S<sub>A</sub> = minimum of 2/3 yield strength, 0.8 stress to rupture in 2 years, or 0.5 fatigue strength, psi

D = tube outside diameter, in.

The equivalent axial load due to tube bending was calculated:

$$AL_{B} = S_{B} \pi T (D-T) , \qquad (10)$$

where

ALB = equivalent axial load, lb

S<sub>B</sub> = bending stress, psi

T = tube-wall thickness, in.

D = tube outside diameter, in.

The minimum residual seal load, MSL, was calculated to be 10 percent of the assembly-seal sealing load, i.e., about 75 lb/in. of seal circumference:

$$MSL = 75 \pi (D_0)$$
, (11)

where

MSL = minimum seal load, lb

 $D_0$  = seal outside diameter, in.

Bolt Strength. For design approximation, high-strength A286 bolts and 2024-T3 aluminum bolts were selected for the stainless steel and aluminum connectors, respectively. The room-temperature strength properties of these bolt materials are shown in Figures 16 and 17. The maximum allowable bolt loads at maximum operating temperature for various bolt sizes are shown in Table 14. The allowable bolt loads were computed as follows:

$$BL = \frac{S}{FS} A_{a}, \qquad (12)$$

where

BL = maximum allowable bolt load, lb

Sy = tensile yield strength of bolt material at maximum operating temperature, psi

FS = factor of safety; for 2024-T3, FS = 1.5, for A286, FS = 1.25

$$A_s = \text{tensile stress area} = \frac{\pi}{4} \left( D - \frac{0.9743}{n} \right)^2$$
  
and  $D = \text{basic major bolt diameter, in.,}$   
 $n = \text{number of threads per inch.}$ 







FIGURE 17, STRENGTH OF A286 BOLTS BASED ON JOHNSON'S APPROXIMATION

Bolt Size,	2024-T3 Aluminum,	Až	286
in. – TPI	200 F	200 F	600 F
10-32	<u>مينا ڪري ٿي. جي جي محمد مين محمد مين محمد مين محمد مين محمد م</u> هنه هن	2,050	1,925
1/4-28	1,139	3, 734	3,505
5/16-24	1, 818	5,960	5, 595
3/8-24	2,751	9,015	8,463
7/16-20	3, 719	12, 186	11,440
1/2-20	5,011	16, 418	15,413
9/16 18	6,360	20, 835	19,560
5 <b>/8-</b> 18	8,010	26,273	24,665
3/4-16	11,686	38, 284	35,940
7/8-14	15,963	52, 297	49,095
1-12	20,774	58,061	63.894

TABLE 14. MAXIMUM ALLOWABLE BOLT LOADS, POUNDS

Seench Clearance. One of the bolt characteristics that influences the dimensions of the connector is the bolt-head size and its corresponding wrench-clearance requirement. One minimum limit on the bolt-circle diameter is established by the tube outside diameter plus the wrench-clearance diameter. A limit on the maximum number of bolts that can be used in a flange is the space between the bolts required for wrench clearance. Table 15 lists the across-flats dimensions and the wrench-clearance diameters that were used in this comparison study.

Bolt Size, in.	Hexagon Head Size Across Flats, in.	Wrench- Clearance Diameter, in			
1/4	7/16	0.88			
5/16	1/2	0.94			
3/8	9/16	1,00			
7/16	11/16	1.08			
1/2	3/4	1.34			
9/16	7/8	1.51			
5/8	15/16	1.59			
3/4	1-1/16	1.73			
7/8	1-1/4	1.94			
1	1-7/16	2,16			

#### TABLE 15. BOLT-WRENCH CLEARANCE

Integral-Flange Connector. The total-axial-load data in Table 13 were used to determine the number and size of bolts required for integral flange connectors for each representative tube size and pressure condition. A computer program was established for designing integral flanges in accordance with the formulas and methods set forth in the ASME Code for Unfired Pressure Vessels - Section VIII. Bolt-circle inputs were determined from the minimum of two conditions: (1) connector hub  $OD + 2 \times (nut radius + wrench clearance between nut and hub) or (2) seal-cavity-clearance diameter + 2 x bolt radius. Flange OD inputs were found by adding two times the bolt diameter to the calculated bolt circle.$ 

Loose-Ring Flange Connector. The number and size of bolts required for the locering configuration were identical to those required for the integral-flange connectors. Loose-ring-thickness calculations were made using the bolt-circle and ring OD dimensions established for the integral-type flanges. The stub flanges were designed by a computer program set up for the loose-ring design. The OD of the stub flanges was the minimum that would contain the seal cavity while providing adequate bearing area between the flange face and the loose ring.

Weight Comparison. Comparative weights for the integral and the loose-ring connectors were computed and are shown in graphical form in Figures 18 through 21.

#### Investigation of Nonconventional Connector Configurations

The investigation of nonconventional connectors was initiated by a search for basic configurations. Three major sources were searched: (1) patents, (2) commercially available connectors, and (3) connector concepts available from the literature and from past Battelle projects. Approximately 60 patents were obtained, studied, and classified according to the basic design approaches. Letters were sent to approximately 316 companies requesting design information on available or promising connector configurations. A search was made of the extensive literature on connectors collected under Contract AF 04(611)-8176, and of the connector concepts conceived during that project and during other Battelle projects on connectors. From this work, five connector configurations were selected as containing design features worthy of detailed study.

To permit a comparison of nonconventional and conventional configurations, a preliminary design of each nonconventional connector was developed for the design conditions given in Table 10. Type 347 stainless steel was chosen as the material for all nonconventional connector designs. This selection permitted a more general comparison of the different configurations and simplified the comparison procedure. The tube-wall thickness and the total axial load were common to all conventional and nonconventional connector configurations. The residual seal-seating-load requirement was also common to all connectors.

<u>X-Connector</u>. The X-connector designed by the Parker Aircraft Company is shown in Figure 22. The outer portion or fastener of this connector serves the same function as the bolts in a conventional connector. The fastener is fabricated with end diameters large enough to pass over the stub flanges during assembly. A special tool is used to swage or deform the end portions of the fastener inwardly and around the stub flanges. The fastener is contained and deformed in such a way that it seats the seal and provides joint preload.

A 1/2-inch X-connector and a 3/4-inch X-connector have been developed through funding from General Electric and NASA, and tests have been conducted satisfactorily with helium at 4000 psi. Development of the X-connector with the Bobbin seal in the larger tube sizes would be somewhat questionable because of the large axial seal-seating motion and high joint preloads which are required. Furthermore, the connector is a



FIGURE 18. WEIGHT OF TYPE 347 STAINLESS STEEL LOOSE-RING AND INTEGRAL FLANGE CONNECTORS FOR 100-PSI, 200 F SYSTEMS



FIGURE 19. WEIGHT OF 6061-T6 ALUMINUM LOOSE-RING AND INTEGRAL-FLANGE CONNECTORS FOR 1500-PSI, 200 F SYSTEMS



FIGURE 20. WEIGHT OF TYPE 347 STAINLESS STEEL LOOSE-RING AND INTEGRAL-FLANGE CONNECTORS FOR 1500-PSI, 200 F SYSTEMS



FIGURE 21. WEIGHT OF TYPE 347 STAINLESS STEEL LOOSE-RING AND INTEGRAL-FLANGE CONNECTORS FOR 6000-PSI, 200 F SYSTEMS

semipermanent rather than a separable connector. However, not only does the connector have unique features, but it appears to have been investigated more thoroughly than any other nonconventional connector in the past few years.

A design similar to the X-connector, shown in Figure 23, was established for the estimate of connector weight. The weight of the connector fastener was calculated using the computer program established for the nut of the threaded-flange connector. The weights of the stub flanges were made identical to those of the stub flanges of the conventional loose-ring connector.

<u>V-Band Connector</u>. The V-band connector is an acceptable tube fastener for many industrial and aerospace applications. Several design variations are possible in the V-band structure, but limitations must be imposed owing to tube size, system pressure, and permissible leakage. A three-segment band interconnected by the stud and nut arrangement shown in Figure 24 was chosen as the best configuration for liquid rocket propulsion systems. This design is more suitable for large tube diameters and for the high axial seal-seating motion required by the Bobbin seal. Also, a more accurate preload can be realized with this configuration than with a single-fastener, single-strap configuration, since less relative motion exists between the band and flange pressure faces during assembly.

The flange-face pressure angle and the thickness of the flange beyond the tube OD were chosen to provide the axial motion required for seal-seating and joint preload. The included angle on the V-band was not allowed to exceed 35 degrees. The distance between the pressure faces of the V-band was determined from the summation of dimensions of the contained flanges and seal. These parameters, together with the total-axial-load data and an assumed friction angle, provided the inputs to a computer program established to calculate the weight of the V-band. The flange weights were calculated by the stub-flange computer program.

<u>Compression-Collar Connector</u>. The load-carrying member of this connector is a two-segment collar similar to a two-segment V-band with parallel faces on the inwardly projecting flanges. The design as presented by Allied Research Associated is shown in Figure 25. One of the inwardly projecting flanges of this connector is tapped for sockethead screws which bear against a hardened steel ring on one of the flange faces. These screws, when tightened against the hardened steel ring, produce axial motion which forces the flanges against the seal to provide joint preload.

The fastener for the redesigned compression collar is in the form of a continuous thin-walled nut as shown in Figure 26. The two end rings of the nut member are designed to contain the bending loads, and this permits the thin-wall construction of the interconnecting cylinder. This design was chosen because it represents a lighter weight than can be realized with the thick-walled two-segment design, where bending loads must be carried through the entire length of the connector. Computations of the weight for the tut, the nut ring, and the flanges were accomplished by computer programs. The weights of these elements plus the socket-head screws and hardened steel ring make up the total connector weight.

<u>Breech-Type Connector</u>. The breech-type connector uses a nut member to carry the total axial load over and around the seal area. The design as it is manufactured by Thornhill-Craver Company, Inc., is shown in Figure 27. One of the flanges, called the lug hub member, contains outwardly projecting tapered shelves. The nut member contains inwardly projecting tapered shelves which contact the tapered surfaces of the



FIGURE 22. X-CONNECTOR



FIGURE 23. X-CONNECTOR APPROXIMATION USED FOR WEIGHT ANALYSIS



FIGURE 24. THREE-SEGMENT V-BAND CONNECTOR



### FIGURE 25. COMPRESSION-COLLAR CONNECTOR DESIGNED BY ALLIED RESEARCH ASSOCIATES



FIGURE 26. REDESIGNED COMPRESSION-COLLAR CONNECTOR

hub-member shelves. Relative rotating motion between the lug hub and nut member produces an axial motion and compressive force on the flanges and seal. Only a few degrees of nut member rotation are required for connector preload. The torque is applied by the arm and bolt device shown in Figure 27.

The redesigned breech-type connector is shown in Figure 28. To save weight, the arm and bolt arrangement for applying axial preload was removed. Splines were designed on the nut and hub to provide the contact surfaces through which a special tool can transmit joint assembly torques. Cammed surfaces on the joint loading tabs were designed with a 9-degree lead angle, and tab-face areas were sized to provide a bearing stress less than the allowable for Type 347 stainless steel. The nut-wall thickness was calculated to withstand the total axial load in tension, together with the bending loads imposed by the tabs. These data were used in the nut and flange computer programs to complete the required calculations. The total connector weight consists of the stub- and tab-flange weights and the tab, spline, and nut weights.

<u>Threaded-Flange Connector</u>. A gear-powered union designed by Resistoflex Corporation contains a worm and worm-gear arrangement to reduce the wrench torque required to assemble the connector (see Figure 29). This type of torquing device increases connector weight for two reasons: (1) the gear reduction assembly is an integral part of the connector and (2) the diameter of the load path is considerably greater than the flange OD and this produces increased bending moments, requiring thicker nut-wall construction. A redesign of this connector produced the more familiar threaded-flanged connector, shown in Figure 30. Weight data for the threaded flange and nut were provided by separate computer programs set up for the threaded-flange connector, while the plain-hub data came from the stub-flange computer program. The high nut torques required for preloading this design must be provided by a special tool attached to the connector through the hub and nut splines.

One design of a special tool which could be used with this connector is shown in Figure 31. A chain similar to a timing chain would be wrapped around the hub spline of the connector and tightened to anchor the tool and to sustain the countertorque imposed by the tool. A second chain would be wrapped around the tool drive gear and nutspline area of the connector, and the ends would be connected to form an endless chain drive. This drive chain would be tightened by an idler wheel to complete the attachment of the special tool. Torque could be applied to the connector nut through either of two wrench positions on the special tool. The low-torque position would be used to attain greater speed in seating the seal, and the high-torque position would be used to apply the preload.

#### Comparison of Conventional and Nonconventional Connectors

Weight is believed to be the most important parameter in the comparison of conventional and nonconventional connectors. The information pertaining to weight, and to comparisons of a more general nature are discussed in the following sections.

Weight Comparison. The connector weights for the representative systems are summarized in parametric-curve form in Figures 32 through 35. The weights shown represent the flange and fastener portion of the connector external to the tube outside diameters. The weight of the seal is not included. The V-band connector is the heaviest for the four conditions investigated. The high weight exhibited by this configuration









FIGURE 29. RESISTOFLEX GEAR-POWERED UNION



FIGURE 30. RESISTOFLEX GEAR-POWERED UNION REDESIGNED FOR MINIMUM WEIGHT





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FIGURE 33. CONNECTOR WEIGHTS FOR A 1500-PSI, 200 F SYSTEM







FIGURE 35. CONNECTOR WEIGHTS FOR A 6000-PSI, 200 F SYSTEM

is due primarily to the noncontinuous fastener element. The fastener does not have the structural-strength advantage that can be realized with a composite ring-and-cylindrical-fastener arrangement, and the imposed bending moments must be sustained in a structure similar to a simple beam. The fastener must also carry the tensile load due to total axial load. The fastener thickness required to carry the combined tensile and connector-imposed bending loads is in excess of four times the thickness required for the pure tensile load.

According to the curves representing the compression-collar and breech-type connectors for each of the four conditions. there is no weight advantage of these connectors as compared with the conventional integral-flange connector. The curves denoting the loose-ring flange, the threaded flange, and the X-connector show that these connectors are the lightest in weight of those investigated during the analysis. However, the feasibility of the X-connector is low when examined in relation to the program requirements for zero leakage, reassembly, and use with the Bobbin seal.

The threaded-flange connector represents the lowest weight design of all the connectors investigated. A more detailed weight comparison for the loose-ring and threaded-flange connectors is shown in Table 16. Weight savings in excess of 25 percent are possible in several instances, as shown by this comparison. The low weight of the threaded-flange connector is due to the fact that the axial load path is contained within a minimum envelope around the seal cavity.

System	Tube	Stainless Ste	el Connector We	ights, lb
Pressure	Diameter,	Loose-Ring	Threaded	Percent
nd Temperature	in.	Flange	Flange	Lighter
	1	0.40	U. <b>26</b>	35.0
	3	2.00	1.66	17.0
100 psi	5	5.13	4. 20	18.1
200 F	8	10.76	9, 8(	8.4
	12	24.77	24.15	
	16	45.06	45.26	
	1	0.56	0.49	12.5
	3	3.82	3.21	16.0
1500 psi	5	9.68	7.31	24,5
200 F	8	20.51	16, 37	20.2
	12	60.87	43.60	28.4
	16	105.73	89. 54	15.3
4000 ps.	1	0.95	0.71	25.2
600 F	2	3.77	2.71	28.1
	3	7.26	5.44	25.1
6000 psi	1	1.06	0.82	22.6
200 F	2	4.09	3, 09	24.4
-	3	7.95	6. 47	18.6

 
 TABLE 16.
 WEIGHT COMPARISON FOR LOOSE-RING AND THREADED-FLANGE CONNECTORS
 Miscellaneous Comparisons. A second area of connector comparison was concerned with some general features of each design. The features evaluated are shown in Table 17. The conventional integral-flange and loose-ring-flange connectors are combined under the single heading "Bolted Flange" in Table 17 since the designs are identical for the features being considered. The best overall rating for these general design features was shown by the bolted-flange and compression-collar connectors.

Condition	Bolted	Compression	Threaded	Breech	Segmented	X-
	Flange	Collar	Flange	Type	V-Band	Connector
Special tools	Good	Good	Undesirable	Undestrable	Good	Undesirable
Required clearance - beyond	Acceptable	Acceptable	Good	Good	Acceptable	Undesirable
connector envelope Preload accuracy	Good	Good	Acceptable	Acceptable	Acceptable	Acceptable
Required machining tolerance Flange load distribution with axial misalignment	Good Good	Good Good	Acceptable Undesirable	Undesirable Undesirable	Acceptable Acceptable	Acceptable Acceptable

TABLE 17, COMPARISON OF GENERAL DESIGN FEATURES

A third area of comparison was the estimated assembly time required for each connector type. The calculations were based on "Assembly Time Units". One (1) Assembly Time Unit (ATU) was defined as the time required to insert one bolt (for the smaller connectors) through the holes of a conventional connector, to start the nut, and to run the nut to a hand-tight condition (IATU = 15 to 30 seconds). The ATU values for basic assembly functions are shown in Table 18.

Connector Assembly	Tube D	lameter
Function	l Through 4 In.	5 Through 16 In.
Hand Assembly	1	2 :0 3
Seal seating	1/2 per nut turn	3/4 per nut turn
Preload torque	1	1 to 2

TABLE 18. SINGLE-BOLT ASSEMBLY-TIME-UNIT REQUIREMENTS

Table 19 gives the breakdown and summarization of ATU's for the individual assembly functions of bolted-flange connectors. Table 18 was prepared for conventional connectors in particular, but slight alterations made the data applicable to compressioncollar and V-band connectors. Tables 20 and 21 present the estimates for these two configurations. Although time estimates for breech-type and threaded-flange connectors were less accurate, since their assembly functions could not be directly associated with the assembly functions of a simple bolt, Tables 22 and 23 present the estimated assembly times for these two connectors. The final column in each of the five assembly-time tables represents the range of estimated real time. The summary of assembly-time estimates, as listed in Table 24, represents the average of this real-time range.

	Tube			Turns to	Single-Bolt ATU					Time
Pressure. psi	Diameter, in.	Bo No.	olts Size	Seat Seal	Hand Assembly	Seal Seating	Preload Torque	Total	Total A TU	Range, min
	1	6	# 10	3-3/4	1	2	1	4	24	6 to 12
100	н	20	1/4	3-1/2	2	3	1	6	120	30 to 60
	16	36	1/4	6	3	4	I	в	288	72 to 144
	1	6	# 10	3-3/4	1	2	L	4	24	6 to 12
1500	н	16	3/8	2-3/4	2	3	2	7	112	28 to 56
	16	30	1/2	4-1/4	3	4-1/2	2	9-1/2	285	71 to 143
6000	1	6	# 10	3-3/4	i	2	1	4	24	6 to 12
	3	10	3/8	2-3/4	1	1-1/2	2	4-1/2	45	11 to 22

TABLE 19. BOLTED-FLANGE-CONNECTOR ASSEMBLY-TIME ESTIMATES

TABLE 20, COMPRESSION-COLLAR-CONNECTOR ASSEMBLY-TIME ESTIMATES

	Tube			Turns to	Joint	Single - Bolt A TU					Time
pressure, Diameter, par in. No	$\frac{B}{No.}$	olts Size	Seat Scal	Haod Assembly	lnøert Seal Bolt Seating		Preload Torque	Total	Total ATU	Range, min	
	1	6	# 10	5-5/4	\$	1	2	1	4	27	7 to 14
100	8	20	1/4	3-1/2	5	1	3	1	5	105	26 10 52
	16	54	1/4	6	8	ì	4	1	6	212	53 to 100
	1	6	# 10	3-3/4	3	1	2	1	4	27	7 to 14
1500	B	20	5/16	3	5	1	2-1/2	1-1/2	5	105	26 to 52
	16	34	7/16	4-1/4	8	1	3-1/2	2	6-1/2	229	57 to 114
6000	1	6	# 10	3-3/4	3	1	2	1	4	27	7 to 14
	3	12	5/16	3	4	ĩ	1-1/2	1-1/2	4	52	13 to 26

	Tube			Nut Turns to		Single	Nut ATU			Time
Pressure, psi	Diameter, in.	No.	Nuts Size	Seat Seal	Hand Assembly	Seal Seating	Preload Torque	Total	Total ATU	Range, min
	1	6	# 10	7	1/2	5	1/2	6	36	9 to 18
100	8	6	5/16	5	1	4	1	6	36	9 to 18
	16	<b>, 6</b>	7/16	8	1-1/2	6	1-1/2	9	54	14 to 28
	1	6	# 10	7	1/2	5	1/2	6	36	9 to 18
1500	8	6	3/4	3-1/2	1	6	1	8	48	12 to 24
	16	6	1-1/4	5	1-1/2	9	1-1/2	12	72	18 to 36
6000	1	6	1/4	6-1/2	1/2	5	1/2	6	36	9 to 18
	3	6	5/8	4	1	5	1-1/2	7-1/2	45	12 to 24

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TABLE 21. V-BAND-CONNECTOR ASSEMBLY-TIME ESTIMATES

TABLE 22. BREECH-TYPE-CONNECTOR ASSEMBLY-TIME ESTIMATES

Pressure, psi	Tube Diameter, in.	Joint Hand Assembly	Attach Special Tool	Seat Seal	Preload Joint	Lock Joint	Remove Tool	Total A TU	Firne Range, min
	ı	2	Not read	2	2	2		8	2 to 4
100 8 16		~ ~	16	14	2	6	6	46	12 to 24
	4	24	40	4	10	10	92	23 to 46	
	1	Z	Not read.	2	2	2		8	2 to 4
1500	Ř	2	16	14	8	6	6	52	13 10 26
1300	16	4	24	40	16	10	10	104	26 to 52
6000	,	,	8	2	2	2	4	20	5 to 10
	3	3	8	4	2	3	4	24	6 to 12

Pressure, psi	Tube Diameter, in.	Joint Hand Assembly	Attach Special Tool	Seat Seal	Preload Joint	Remove Tool	Total ATU	Time Range, min
100	1	2	No. reqd,	2	2	-	6	1-1/2 to 3
	8	4	16	14	2	6	42	11 to 22
	16	8	24	40	4	10	86	21 to 42
1500	1	2	Not regd.	2	2		6	1-1/2 to 3
	8	4	16	14	8	6	48	12 to 24
	16	8	24	40	16	10	98	25 to 50
6000	1	2	8	2	2	4	18	4-1/2 to 9
	3	3	8	4	2	4	21	5 to 10

#### TABLE 23. THREADED-FLANGE-CONNECTOR ASSEMBLY-TIME ESTIMATES

TABLE 24. SUMMARY OF ASSEMBLY-TIME ESTIMATES

Pressure, psi	Tube Diameter, in.	Bolted Flange	Compression Collar	Threaded Flange	Breech Type	Segmented V-Band
100	1	9 min	10 min	2 min	3 min	l4 min
	8	45 min	39 min	16 min	18 min	14 min
	16	1 hr 45 min	1 hr 20 min	31 min	35 min	21 min
1500	1	9 min	10 min	2 min	3 min	14 min
	8	42 min	39 min	18 min	19 min	18 min
	16	1 hr 45 min	1 hr 25 min	38 min	39 min	27 min
6000	1	9 min	10 min	7 min	8 min	14 min
	3	18 min	19 min	8 min	9 min	18 min

The threaded-flange connector shows the lowest time requirement for the largest range of system parameters. The V-band connector requires slightly lower assembly times in tube diameters from 8 to 16 inches at system pressures of 1500 psi and below.

Conclusions and Recommendations. On the basis of the investigations described above, the following conclusions were reached:

- (1) Optimized conventional integral- and loose-ring-flange connectors will be lighter in weight than most flanged connectors that might be developed from nonconventional configurations.
- (2) Nonconventional threaded-flange connectors offer the promise of a weight savings of up to 25 percent as compared with conventional flanged connectors.
- (3) The successful development of large, threaded-flange connectors depends on the development of easily used assembly tools.

The following recommendations were made concerning the development of flanged connectors during the remainder of the program:

(1) Conventional integral- and loose-ring-flange connectors should be developed for all sizes and service conditions.
(2) With funds in addition to those allocated, a parallel program should be undertaken to develop large threaded-flange connectors and appropriate assembly tooling. This development could be undertaken in the following major steps: (a) the design of representative connectors and associated assembly tools and (b) the fabrication and field evaluation of selected connectors and assembly tools.

#### Investigation of Connector Thermal Gradients

As described in Technical Documentary Report No. AFRPL-TR-65-162, a separable connector can achieve very low helium leakage rates over a wide temperature range only if the connector design can accommodate the effects of thermal gradients. This degree of design sophistication is not usually required for connectors containing liquids, and as a result, most separable-connector designs do not include a detailed thermalgradient analysis. The development of such an analysis technique for the AFRPL threaded connectors is believed to be a major reason for the success of the connector in maintaining low gas leakage over large temperature ranges. Consequently, the decision was made to incorporate a thermal-gradient analysis in the design procedure for flanged connectors.

The effects of thermal gradients on a connector are strongly dependent on the configuration and operating principles of the connector. As discussed previously, the Bobbin seal specifically incorporates a feature to minimize the effect of changes in axial load on the effectiveness of the seal. Because of the complex configuration of flanged connectors, a theoretical analysis of heat transfer in a connector is not only very costly, but is also subject to significant inaccuracies. Thus, the decision was made to estimate the connector thermal gradients by making temperature measurements on connector parts similar to those expected for the final connectors.

Experiments With Flange Segments. The initial thermal-gradient tests were conducted with bolted flange segments. The use of segments that were sized to simulate a typical portion of a connector flange greatly reduced not only the cost of the parts but alio the cost of the tests. It was believed that the results of these tests would facilitate the selection of optimum connector proportions. The details of the segments are shown in Figures 36 and 37. Two segments of each size were bolted together, and thermocouples were attached to the bolt and to the segment as shown in Figure 38. The sides of the segments were insulated, and the surface simulating the inside surface of the connector and the tube wall was subjected to hot and cold fluid temperatures according to the required connector temperature limits.

Figure 39 shows the thermocouple readings when the inner surface of a simulated high-pressure, stainless steel connector was placed in contact with liquid nitrogen. The sudden drop in temperature near TC-6 after 11 minutes was attributed to increased heat transfer between the nitrogen and the connector as local boiling of the nitrogen subsided. The decision was made to mount the simulated connector in a vertical position to permit the bubbles to escape more readily and more closely simulate a flowing liquid.

Figures 40, 41, and 42 show typical results for vertically mounted specimens. The average flange temperatures were obtained from the arithmetic average of four thermocouples in the flange segments. One thermocouple located in the center of the bolt was



Tube Diam In	Pressure. psi	•	в.	с	D	E	F	G	н	t	к	No of Sim Bolts	Deg. of Sim. Flange
1	6000	0, 38	27-1/2	0.68	0.49	0. <b>149</b>	3, 87	0.63	0.12	0.201	0.19	6	60
3	6000	0.67	17-1/2	1.32	0, 94	0 44	2.95	1.55	0,15	0.397	0.34	10	36
1	4000	0.40	27-1/2	0.65	0.46	0.1196	4.00	0.50	0. 12	Ŭ. 201	0 20	6	60
3	4000	0 72	17-1/2	1.15	0. 83	0.375	3. 22	1.28	0 15	0 332	0.36	10	36
1	1500	0.48	27 - 1/2	0.57	U. 38	0.0418	4.17	0.33	0.12	0 201	0 24	6	60
3	1500	1.08	21-1/2	0, 81	0. 56	0.1196	3.54	0.96	0, 15	0. 266	0 54	8	45
5	1500	1.45	19-1/2	0. <b>88</b>	0. 67	0.20	3. 11	1.39	0, 15	0. 332	0 73	10	36
8	1500	1,45	11	1.20	0. 82	0.320	2.61	1.89	0, 15	0. 397	0.73	16	22. 5
12	1 500	1.45	8	1.65	1.15	0.48	1.78	2.72	0, 19	0.531	0.73	34	15
16	1500	1.45	5-1/2	1. 81	1, 31	0.64	1.00	3.50	0.19	0. 531	0.73	32	11.25

Note All dimensions in inches unless otherwise noted.

FIGURE 36. DETAILS OF SEGMENTS FOR THERMAL-GRADIENT TESTS FOR SIMULATED TYPE 347 STAINLESS STEEL BOLTED FLANGES



Tube Diam. in.	٨	В	c	ם	E	F	6	н	1	ĸ	No of Sim. Balts	Deg. of Sim. Flange
1	0,48	27-1/2	0.7 <b>3</b>	0 48	0.042	4 23	0.27	0.15	0. 266	0.24	6	60
3	1.08	21-1/2	1.06	0.69	0.126	3.75	0 75	0.15	0. 397	U 54	8	45
3	1.20	14-1/2	1 27	U. 84	0 210	3. 27	1.23	0.15	0. 531	0.60	12	30
0	1 28	10	1.71	1 09	0. <b>336</b>	2.82	1.68	0 15	υ <b>656</b>	0 64	18	20
12	1 44	7-1/2	2.38	1 38	0. <b>504</b>	2.09	2. 41	V. 15	0 813	0 72	24	15
16	1.53	6	2.55	1 67	0 672	1.50	3 00	0 15	0 938	0 76	32	11. 25

FIGURE 37. DETAILS OF SEGMENTS FOR THERMAL-GRADIENT TESTS FOR SIMULATED 1500-PSI 6061 ALUMINUM BOLTED FLANGES

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FIGURE 42. RESULTS OF THERMAL-GRADIENT TESTS WITH FLANGE SEGMENTS SIMULATING A 5-INCH, 1500-PSI ALUMINUM CONNECTOR WITH ST AINLESS STEEL NUTS AND BOLTS



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FIGURE 41. RESULTS OF THERMAL-GRADIENT TESTS WITH FLANGE SEGMENTS SIMULATING A 5-INCH, 1500-PSI ALUMINUM CONNECTOR

WITH ALUMINUM BOLTS AND NUTS



used to measure the bolt temperature. The results showed that a significantly greater thormal-gradient problem exists with stainless steel connectors than with aluminum connectors. The tests also showed that it would be undesirable to use steel bolts with aluminum connectors despite their higher strength and greater resistance to damage due to handling. Figure 43 shows the maximum temperature difference measured for different sizes of aluminum and stainless steel connector flange segments when in contact with liquid nitrogen and boiling water.

Experiments With Connectors. Thermal-gradient measurements were made with 2- and 3-inch connector assemblies made of Type 347 stainless steel and 6061-T6 aluminum in both integral and loose-ring configurations. For the integral-flange assembly, measurements were made of the temperatures at the Bobbin seal tang, the integral flange at the bolt circle, and the bolt shank between the flanges. For the loose-ring assembly, temperatures were measured at the Bobbin seal tang, the outside diameter of the stub flange, the loose ring at the bolt circle, and the bolt circle, and the bolt shank midway between the flanges. Table 25 shows the results of the tests. A comparison of the integral-flange-to-bolt temperature differences for the 3-inch aluminum and stainless steel connectors in Table 25 (35 F and 55 F, respectively) shows good correlation with the values measured for the 3-inch-connector flange segments shown in Figure 43b and d (25 F and 62 F, respectively).

			Temperature Difference, F						
Tube Diameter, in.	Flange Type	Material	Seal to Integral Flange	Integral Flange to Bolt	Seal to Stub Flange	Stub Flange to Loose Ring	Loose Ring to Bolt		
3	Integral	Type 347 SS	70	55					
2	Integral	Type 347 SS	30	49					
3	Integ ral	6061-T6 A1	60	35					
2	Integral	6061-T6 A1	30	50					
3	Loose Ring	Type 347 SS			65	65	75		
2	Loose Ring	Type 347 SS			35	60	60		
3	Loose Ring	6061-T6 Al			30	30	48		
2	Loose Ring	6061-T6 Al			40	32	34		

TABLE 25.RESULTS OF THERMAL-GRADIENT MEASUREMENTSFOR CONNECTOR ASSEMBLIES AT ROOM TEMPER-<br/>ATURE EXPOSED TO LIQUID NITROGEN

Deflection Measurements. Measurements were made of the deflections in the flanges of the 2- and 3-inch aluminum and stainless steel connectors subjected to liquid nitrogen temperatures. The connectors were assembled and the bolts were stressed to selected preload levels. When the connectors were filled with liquid nitrogen, the deflections of the flange elements were measured to determine the effects of the thermal gradients. The results of these measurements are shown in Figures 44 through 47.

As shown in Figure 44. when the 3-inch, Type 347 stainless steel, integral flanged connector was cooled, the thermal gradients caused the flange elements to rotate



FIGURE 43. 212 F AND -323 F THERMAL-GRADIENT TEST RESULTS FOR 1500-PSI ALUMINUM AND STAINLESS STEEL CONNECTOR FLANGE SEGMENTS



FIGURE 44. FLANGE DEFLECTION FOR 3-INCH, 1500-PSI, STAINLESS STEEL INTEGRAL-FLANGE CONNECTOR



FIGURE 45. FLANGE DEFLECTION FOR 3-INCH, 1500-PSI STAINLESS STEEL LOOSE-RING-FLANGE CONNECTOR



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FIGURE 46. FLANGE DEFLECTION FOR 3-INCH, 1500-PSI ALUMINUM INTEGRAL-FLANGE CONNECTOR



FIGURE 47. FLANGE DEFLECTION FOR 3-INCH, 1500-PSI ALUMINUM LOOSE-RING-FLANGE CONNECTOR



FIGURE 48. PRESSURE PLUS BENDING LOADS (AXIAL) FOR TYPE 347 STAINLESS STEEL TUBE CONNECTORS

conversely to the direction of rotation during bolt-up. This resulted from axial shrinkage of the seal and the flange plus radial shrinkage of the tube adjacent to the flange. Thus, in the flange-design program, it appeared necessary to account for the effects of thermal gradients in the radial direction as well as in the axial direction.

#### Investigation of Bolt Parameters

A survey was made of the technical literature available at Battelle-Columbus on bolted fasteners. Approximately 120 references were selected and cataloged. Also, about 25 reports on material properties were surveyed and cataloged. Catalogs and technical literature on other types of fasteners were included in the information survey. Although several interesting types of fasteners were identified, it was decided that no fastener configuration seriously challenged nut and bolt fasteners for the conventional integral- and loose-ring-flange connectors that were selected for development.

<u>Preliminary Investigation of Fasteners for Stainless Steel Connectors</u>. Preliminary calculations for estimating fastener loads per inch of connector circumference were made on the basis of the system requirements given in Table 10. Tentative values of the axial loads due to pressure and bending that must be sustained by the fasteners for Type 347 stainless steel tubing systems are given in Figure 48.

The strengths of bolt-nut fasteners were obtained from catalog data for commerially available socket-head-type bolts with an ultimate tensile strength of the order of 190,000 psi. Figure 49 shows the strength/weight ratios for these fasteners using three different bolt spans or grips. It can be seen that the smaller bolt sizes give a somewhat better strength/weight ratio. Figure 50 shows the tensile load that can be sustained by bolt-nut fasteners made from A286, on the basis of data published in NASA CR-357<sup>(4)</sup>. Preliminary selections of A286 bolt sizes for the three high-pressure service ranges are listed in Table 26.

Preliminary Investigation of Fasteners for Aluminum Connectors. A search was made for suppliers of high-strength aircraft-quality aluminum bolts, and for information on bolt yield strength at 200 F and -400 F. Fastener supply companies and fastener manufacturers were contacted. Several companies could supply 6061-T6 aluminum bolts, and a few could supply 2024-T4 aluminum bolts from stock, but none could supply higher strength aluminum bolts from stock. None of the companies contacted could supply information on bolt yield strength for any of the alloys at the desired temperatures or at room temperature.

A preliminary estimate was made of the axial loads that must be sustained by the fasteners for the 6061-T6 aluminum connectors. An estimate was made of the tensileload capability of 2024-T4 aluminum bolts in sizes from 1/4 to 7/8 inch, as illustrated in Figure 51. Table 27 shows the preliminary selection made of the bolt size and number required for bolted flanged connectors for 1500-pri aluminum connectors.

Two quotations were obtained for 7075 aluminum bolts, as shown in Table 28. It appeared possible to obtain bolts of the diameter and length required for the connectors in AN 300 series 7075-T73 aluminum with a NAS 624-type 12-point head. Commercialquality-aluminum bolts in alloy 7075-T6 manufactured in compliance with American Standard B18.2-1955 also appeared to be available as required. However, it was concluded that the use of 7075 aluminum bolts for the connectors would impose a substantial cost disadvantage.



FIGURE 49. STRENGTH/WEIGHT RATIOS FOR UNF SOCKET-HEAD CAP SCREWS WITH LIGHT HEX FULL-HEIGHT LOCK NUT AND WASHER



FIGURE 50. FASTENER LOAD CAPACITY (A286 BOLT AND NUT)



FIGURE 51. FASTENER LOAD CAPACITY (2024-T4 ALUMINUM)

Tube	6000-Psi, 200 F System		4000 600 F :	-Psi, System	1500-Psi, 200 F System	
Diameter, in,	Bolt Size	No. of Bolts	Bolt Size	No. of Bolts	Bolt Size	No. of Bolts
1	No. 10	6	No. 10	6	No. 10	6
2	5/16"	8	1/4"	8	No. 10	6
3	3/8"	10	5/16"	10	1/4"	8
4					5/16"	8
5					5/16"	10
6					3/8"	12
7					3/8"	14
8					3/8"	16
9					3/8"	20
10					3/8"	24
)1					1/2"	20
12					1/2"	22
13					1/2"	24
14					1/2"	26
15					1/2"	30
16					1/2"	32

14.

# TABLE 26.PRELIMINARY SELECTION OF A286 BOLT SIZESFOR STAINLESS STEEL CONNECTORS

# TABLE 27.PRELIMINARY SELECTION OF 2024-T4 ALUMINUM<br/>BOLT SIZES FOR 1500-PSI, 200 F, 6061-T6<br/>ALUMINUM CONNECTORS

Tube Diameter, in.	No. of Bolts	Size of Bolts
1	6	1/4-28
2	6	5/16-24
ĉ	8	3/8-24
4	10	1/2-20
5	12	1/2-20
6	14	9/16-18
7	16	9/16-18
8	18	5/8-18
9	20	5/8-18
10	22	5/8-18
11	20	3/4-16
12	24	3/4-16
13	26	3/4-16
14	24	7/8-14
15	26	7/8-14
16	30	7/8-14

7075-T73 AN	3DD Series	7075-T6 American Standard B18,2-1955		
Size	Cost per 100 Lb, \$	Size	Cost per 100 Lb, \$	
BM9138-4-7	439.00	$1/4-28 \times 3/4$	49.70	
BM9138-6-20	524,40	$3/8-24 \ge 2$	46.00	
BM9138-8-30	593.00	$1/2 - 20 \times 3$	53,35	
BM9138-10-40	844.20	$5/8 - 18 \times 4$	83,50	
BM9138-12-54	1067.10	$3/4 - 16 \times 6$	140.00	
BM9138-14-67	1863.00	$7/8 - 14 \times 7$	168.00	

#### TABLE 28. QUOTED COST OF 7075 ALUMINUM BOLTS

Preliminary Consideration of Threaded-Fastener Parameters. A study was made of published information on such factors as preload, fatigue failure, vibration resistance, corrosion resistance, surface finish, hardness, lubrication, fastener stiffness, and the effect of successive tightenings.

Three methods of predicting bolt preload or tension are commonly used: (1) measuring the applied torque, (2) measuring the fastener's extension, and (3) estimating the amount of nut rotation. Patented configurations have begun to appear recently, which are based on the yielding of a part of the bolt-nut assembly. Measurement of the applied torque is the method most often used, although some of the new designs show considerable promise. Price and  $Trask^{(5)}$  investigated the following factors and their effect on bolt tension:

- (1) Structure material
- (2) Bolt grip length
- (3) Lubricant
- (4) End of fastener turned
- (5) Number of successive tightenings.

Variations in the structural materials showed little difference in the bolt torque for lubricated parts. However, steel bolts mated with a titanium structure showed higher torques than steel bolts mated with an aluminum or steel structure for nonlubricated, asreceived parts. The maximum bolt load on aluminum structures and other low compressive yield materials is limited by local deformations of the structure at the area of contact with the bolt head and nut. The grip length and the end of the fastener to which tightening torque is applied had little effect on the torque-preload relationship.

Lubrication of the mating parts, however, had a significant effect on the torque required to produce a specified bolt load. The torque required to stress a nonlubricated (specimens as-received) 3/4-inch bolt to 100,000 psi was 5140 lb-in. This torque was 2590 lb-in, when the specimen was lubricated. Price and Trask noted similar results at other stress levels for 3/4-inch and 1/4-inch fasteners.

A computational method for relating bolt torque, stress, and diameter was suggested by these investigations. The ratio R as determined by Equation (13) was used:

$$R = \frac{1000 T_0}{D_b S},$$
 (13)

where

 $T_o = torque, lb-in.$ 

 $D_b = bolt diameter, in.$ 

S = axial stress in bolt, psi.

A factor of 1000 was used for convenience in locating the decimal point. Values of R were calculated by Price and Trask for 1/4-inch and 3/4-inch steel bolts for lubricated and as-received conditions for three stress levels. These values are shown in Table 29.

Bolt Stress,	R (Lub	ricated)	R (As R	eceived)
psi	1/4 In.	3/4 In.	1/4 In.	3/4 In.
100,000	3.52	3.48	8,24	6.00
120,000	3.57	3.59	8,29	5,98
140,000	3.60	3,66	8,51	5.92

TABLE 29. RATIO R FOR 1/4-INCH AND 3/4-INCH STEEL BOLTS

For lubricated fasteners the value of R was nearly the same for 1/4-inch bolts as it was for 3/4-inch bolts when the bolts were stressed to the same axial stress level. The change in R with changes in stress level was also small. When the formula was applied to bolts in the nonlubricated or as-received condition, however, the ratio was not constant for different bolt sizes.

Repeated installation of threaded fasteners was found to have a significant effect on the torque-tension ratio. Repeated installation for as-received fasteners, in general, required higher torque for a given tension than was required on the first installation. The installation torque for lubricated fasteners was reported to be essentially constant after the second assembly. This effect is shown in Table 30. All torques are for an axial stress of 100,000 psi in the bolt. The lock-nut torque shown is the torque required to turn the lock nut with no axial load on the bolt.

From the report by Price and Trask, it can be concluded that the use of lubricated fasteners permits relatively accurate prediction of bolt tension and reliable reuse of bolt and nut.

Other investigations, however, have shown that lubrication tends to reduce the vibration resistance of threaded fasteners to a marked degree. One comparison<sup>(6)</sup> of the effect of vibration for lubricated versus nonlubricated fasteners shows that lubricated fasteners lost all preload after 1000 to 2000 cycles, while nonlubricated fasteners held for the duration of the test extending to 125,000 cycles. The nuts used were plain nuts. In other tests, various lock nuts were evaluated for vibration resistance. These included nonmetallic insert, beam type, distorted thread, castellated, and plain. Of these, the nonmetallic-insert type performed best, with the beam type performing next best. Castellated types and those with a distorted thread were less effective in resisting the effects of

First I	nstallation	Second I	nstallation	Third Ir	stallation	Fourth Installation	
Total Torque	Lock-Nut Torque	Total Torque	Lock-Nut Torque	Total Torque	Lock-Nut Torque	Total Torque	Lock-Nut Torque
		1,	4-Inch Fast	ener Lubri	icated		
88	24	68	6	71	6	63	6
95	12	65	5	62	4	65	3
88	8	60	3	54	-	57	_
82	19	68	5	64	5	66	3
89	11	62	4	68	4	60	4
183	18	286	24	357	24	-	-
244	28	36 0	27	-	-	-	-
253	35	378	35	-	-	-	-
237	27	350	24	-	-	-	-
179	6	295	-	377	-	<b>4</b> 08	-
		3,	4-Inch Fast	ener Lubri	cated		
2590	-	2550	-	2450	-	2400	_
2530	-	2220	-	2030	-	2040	-
2590	-	2320	-	2370	-	2240	-
2800	-	2350	-	2240	-	2180	-
2550	-	2240	-	2140	-	1990	-
		3/	4-Inch Faste	ner As Re	ceived		
5140	180	5780	130	6090	100	6250	80
4960	180	6450	100	6720	80	6940	50
4500	230	5960	130	7020	100	6610	80
3700	180	3830	100	4220	100	4500	80
4200	190	4010	80	4270		4550	

# TABLE 30.EFFECT OF REPEATED ASSEMBLY ON TIGHTENING<br/>TORQUE FOR STEEL BOLTS(a)

(a) From National Bureau of Standards Report 7308, "The Relation Between Torque and Tension for High Strength Threaded Fasteners", J. I. Price and D. K. Trask.

vibration. In general, high installation torque for a given bolt stress level will give the highest resistance to vibration. The results of another investigation<sup>(7)</sup> indicated that if a fastener system meets room-temperature vibration requirements, it should present no problems at cryogenic temperatures.

The fatigue of bolted fasteners has received much attention (8-12) in recent years. The most important factor is the preload of the bolt. Increasing the bolt preload has the effect of reducing the minimum-to-maximum stress ratio. Although increasing the bolt preload increases the mean stress, the fatigue strength of the bolt is improved.

One study of the fatigue life of threaded fasteners as affected by the ratio of minimum to maximum working load showed that the fatigue life exceeded 2 million cycles for a ratio of 0.5, dropped to 200,000 cycles at a ratio of 0.4, and dropped further to 100,000 cycles for ratios below 0.3. The angularity of the bearing surface also had a marked effect on fatigue life. An angular deviation of 1 degree reduced the fatigue life by 38 percent and an angular deviation of 2 degrees caused a 91 percent reduction in fatigue life.

Unengaged threads on the bolt and reuse of the nut each have been shown to cause a marked effect on the fastener fatigue life. If two or more threads are unengaged on the bolt, the fastener fatigue life is about three times as great as that in an installation where the nut is close to thread run-out. Reuse of stressed nuts has been shown to decrease the fastener fatigue life about 32 percent on the first reuse. The life then decreases to about 50 percent on the second reuse and remains at this level for repeated reuse.

The relative stiffness of the bolt and bolted assembly were major factors affecting the fatigue life of bolted assemblies. Fatigue life was improved for bolts with relatively low stiffness by designing the surrounding structure assembly, including the washer, with a relative high stiffness. Increasing the thread root radius generally improved the fatigue life. Special nuts, designed to distribute the preload along the nut, are apparently of questionable value from the standpoint of fatigue life. Data on fatigue life for these nuts show much scatter, but generally the fatigue life is somewhat improved by the use of special nuts.

Nuts having a nonmetallic collar insert for vibration resistance show reduced fatigue life. This is probably due to a reduced number of threads to accommodate the collar insert. Higher nut height generally improves fatigue life. The use of a lubricant also improves fatigue life but reduces vibration resistance.

The effect of higher temperatures on fastener fatigue life is both good and bad. One effect of higher temperature is to increase creep rate; thus the higher stressed portions of the nut relax, giving a more even distribution of stress along the nut and, thereby, increased fatigue life. The increased creep rate at higher temperatures, however, has the effect of reducing bolt preload, decreasing the minimum-to-maximumstress cycle ratio, and reducing the fatigue life. Additional information on the fatigue life of bolts at higher temperatures is needed. There is no dependable way to evaluate the fatigue properties at higher temperatures from room-temperature data.

Chemical methods of locking threaded fasteners appear to be of questionable value. Properly applied chemical locking methods may outperform prevailing torque lock nuts. However, for chemical locking compounds to perform reliably, the compound must be applied to a grease and oil-free surface. Since manufacturing, shipping, and assembly techniques usually require the use of oils, it becomes necessary to clean the bolts and nuts before using chemical locking compounds.

The following design guidelines are suggested for reliable bolted joints:

- (1) Use bolts and nuts of as high a strength as practical
- (2) Beam-type prevailing-torque lock nuts are preferred
- (3) Use a predetermined lubricant and lubricating procedure when installing the fastener
- (4) Install the fastener according to predetermined torque-preload data

- (5) Install the fastener with the highest preload consistent with the material's strength and application
- (6) Use Class 3 threads
- (7) Use bolts and nuts with generous thread-root radius
- (8) Use hardened washers or hardened faces on the structure to eliminate the use of a washer
- (9) Bolts may be reused but do not reuse a nut
- (10) Maintain at least two full threads on the bolt under the nut
- (11) Periodically retighten all fasteners in a joint
- (12) Use a bolt with a coefficient of expansion similar to the joint in which the bolt is used.

#### Investigation of Stress-Relaxation Considerations

Two major types of system conditions appeared likely to bring about the leakage of a connector because of stress relaxation. One was the system with a maximum operating temperature of 1200 F. Austenitic stainless steel exhibits a noticeable creep rate at this temperature and it was obvious that connectors designed to operate at this temperature would have to sustain considerable creep without failure. Furthermore, the operating time of each connector at temperature would have to be carefully recorded. A method for predicting the operational life of such connectors was developed by E. C. Rodabaugh and M. Cassidy (see Appendix A) on a NASA program.

Another system requirement that could cause stress-relaxation problems was a storage life of 5 years without a significant increase in leakage. This condition anticipates the standby requirement of missiles with occasional evaluation of the various missile systems. Although the normal maximum temperature conditions of this requirement are only 200 F (with occasional excursions to 600 F for the periodical evaluation of stainless steel systems), the compact nature of flanged connectors does not provide for large amounts of preload energy storage. Thus, a small amount of creep or stress relaxation in the connector can cause large changes in the preload conditions. In essence, the decision was made to design the connectors with sufficiently low stresses that creep and relaxation would be negligible.

Only a small amount of long-term creep and stress-relaxation data were available for the materials and temperatures of interest. Therefore, an experimental program was established to determine allowable design stress levels. In the following paragraphs the nature of creep and relaxation is reviewed briefly, the experimental program and its results are described, and the resulting design guidelines are defined for a 5-year-life requirement.

<u>Creep and Relaxation</u>. Although closely related, creep and relaxation are distinctly different effects. Creep is the tendency for material to exhibit time-dependent strains at a constant stress level; i.e., with a constant force on a bolt, creep will be evidenced by a gradual increase in bolt length. Relaxation is the reduction of stress in time under a constant strain; i.e., for a bolt tightened between two rigid flanges, stress relaxation will result in a gradual reduction in bolt load without a change in the assembled bolt length. Both of these characteristics are typical of metals at elevated temperatures. The four primary parameters in each phenomenon are time, temperature, stress, and rate of creep or relaxation.

Current creep and relaxation theories are of little use in design problems except as they may be used to guide the organization and use of test data. With current theories, data can be interpolated with a fair degree of confidence. Some data can be extrapolated to longer time periods with confidence. However, the extrapolation of data must usually be done with caution, and tests simulating the actual operating conditions should be conducted if possible.

Most creep and relaxation tests have been conducted at elevated temperatures. The materials and temperatures of interest for the connectors are not usually thought of as conditions in which creep is a problem. Recently, for some requirements, such as the long flying time of supersonic transports (36,000 hours), the need for long-term, low-level creep data has been identified, and special machines are being designed and fabricated to obtain such data. However, the measurement of small amounts of creep and relaxation with most present equipment is difficult and the results are subject to a degree of inaccuracy. A significant assist in this problem is that most materials are relatively stable at low temperatures and low stress levels, and a straight-line relationship can be extrapolated with considerable confidence. At elevated temperatures, in the range of stresses customarily used in design, it has been found that plotting the stress versus the second-stage or minimum creep rate produced gives an approximate straight line on a log-log plot.

The physical mechanisms causing relaxation in a material are believed to be very nearly the same as, if not identical to, those causing creep. For this reason it might seem that creep data could be used to predict relaxation data accurately. Unfortunately, stress level and creep rate are significant factors affecting creep behavior, and since stress and relaxation rate are constantly changing in a component experiencing relaxation, a large amount of creep data are needed to make accurate relaxation estimates. On the other hand, it is much more difficult to conduct a relaxation test than a creep test because of the relaxation loading and measuring requirements. For this reason methods have been developed for estimating relaxation from creep data.

Summary of Available Creep and Relaxation Data. Searches were made for creep and relaxation data: (1) on 6061-T6 aluminum at room temperature and at 200 F, (2) on Type 347 stainless steel at 200 F and 600 F, and (3) on A286 at 200 F and 600 F. Table 31 lists data supplied by one manufacturer for 6061-T6 at 75 F and 212 F. These data are compatible with data from two other sources. No information was obtained on the relaxation of 6061-T6 aluminum. Although considerable creep data are available for Type 347 stainless steel and A286 at temperatures between 1000 F and 1400 F, no creep data were obtained at room temperature and 600 F. Likewise, no relaxation data were obtained for these materials at those temperatures.

On the basis of the available theories and the nature of Type 347 stainless steel, the higher temperature creep information cannot be used to predict the creep behavior at room temperature and 600 F. The creep data on 6061-T6 can be used to extrapolate the effects at the given stress levels for the required design life of 40,000 hours. However, the extrapolation of this information to lower stress levels was not believed to be possible. Since the maximum design stress level in the connector was expected to be below the yield strength, it was not possible to predict the creep behavior of the aluminum at most of the anticipated design-stress levels.

Temperature,	Time Under	Stres	s, ksi, for R	upture and Ci	eep in Time	Indicated
F	Stress, hr	Rupture	1.0% Creep	0.5% Creep	0.2% Creep	0.1% Creep
75	0.1	45	45	44	43	42
	1	45	45	43	42	42
	10	45	44	43	42	42
	100	45	44	42	42	41
	1000	45	43	42	41	41
212	0.1	41	41	40	<b>4</b> 0	40
	1	40	40	<b>4</b> 0	39	38
	10	39	39	39	37	36
	100	38	37	37	35	34
	1000	37	36	35	33	32

# TABLE 31. STRESS RUPTURE AND CREEP PROPERTIESFOR 6061-T6 ALUMINUM

Experimental Determination of Creep Data. From a consideration of the probable modes of missile operation, it was decided that the connectors would remain at room temperature except for those brief periods when the missile was statically operated. On this basis, 10 hours was selected as the longest time that a connector would experience a maximum temperature. Because it was difficult to determine when, during the 5-year storage period, a connector would experience the maximum temperature, the most stringent operational requirement was selected as consisting of a period of 10 hours at maximum temperature, followed by a room-temperature environment for 5 years.

Since the expected period of high-temperature operation was well within normal creep-testing periods, it was decided that specimen tests should be conducted for 100 hours. The additional test time would make the estimate of behavior during the first 10 hours more reliable. For the aluminum, six stress levels were selected: 35, 33, 30, 25, 20, and 15 ksi. These stress levels not only bracketed the expected design stress levels, but also correlated with some of the stress levels of the available data. For the stainless steel, eight stress levels were selected: 50, 45, 40, 35, 30, 25, 20, and 15 ksi. These were chosen to bracket the expected design levels and to insure measurable creep at the higher values. To provide some indication of material stability, three heats were tested. Although the maximum test time possible was 10,000 hours, it was believed that data obtained over this period could be used to extrapolate behavior to 40,000 hours.

Tables 32 and 33 show the data for the specimens prepared for the roomtemperature tests. Foil strain gages were attached to the room-temperature specimens, reference gage-length measurements were made for each specimen, and the specimens were stressed in a multiple-lever-arm system. The desired stress on each specimen was obtained by varying the cross sectional area of the specimens. Thus, different stresses could be obtained, although one lever arm was used to load several specimens in series. The elevated-temperature tests were conducted in standard creep-test machines.

Material	Stress, ksi	Specimen Diameter, in.	String	Number of Specimens
347 55	50	0,2500	1	6
347 SS	45	0.2635	1	6
347 SS	40	0.2798	1	6
347 SS	35	0,2500	2	6
6061-T6 Al	35	0.2500	2	6
6061-T6 A1	33	0,2580	2	6
3 <b>47 SS</b>	30	0,2500	3	6
347 SS	25	0.2740	3	6
6061-T6 A1	30	0.2500	3	6
347 SS	20	0.2500	4	6
347 SS	15	0.2890	4	6
6061-T6 A1	15	0.2890	4	6
6061-T6 A1	25	0,2500	5	6
6061-T6 A1	20	0.2798	5	6

## TABLE 32. 10,000-HOUR CREEP TESTS, THREE HEATS FOR EACH MATERIAL, TWO SPECIMENS FOR EACH HEAT

TABLE 33.100-HOUR CREEP TESTS, THREE HEATS FOR EACH<br/>MATERIAL, ONE SPECIMEN FOR EACH HEAT,<br/>0.2500-DIAMETER CAGE SECTION

Material	Test Temperature, F	Strees, ksi	Number of Specimens
6061-T6	200	15	3
6061- <b>T</b> 6	200	20	3
6 06 1 <i>~'</i> T6	200	25	3
6 06 <b>1 - T</b> 6	200	30	3
6061-T6	200	33	3
6061- <b>T</b> 6	200	35	3
347 SS	600	15	3
347 SS	600	20	3
347 SS	600	25	3
347 SS	600	30	3
347 SS	600	35	3
347 SS	600	40	3
347 SS	600	45	3
347 SS	600	50	3

100-Hour Creep Data for 6061-T6 Aluminum at 200 F. Figures 52 through 58 show the results of the 100-hour creep tests for 6061-T6 aluminum at 200 F. Total strain includes the elastic strain, the initial plastic strain, and the creep strain. The initial point on each curve is the first strain reading taken after application of the load, and represents the total elastic and initial plastic strain. Continued increase in strain with increasing time represents creep strain.

It was concluded from these data that operational stress levels should be kept below 30 ksi. The variation between the three materials was not unusual, and the data were averaged to obtain a design basis.

100-Hour Creep Data for Type 347 Stainless Steel at 600 F. Figures 59 through 65 show the results of the 100-hour creep tests for Type 347 stainless steel at 600 F. The data are reported in the same manner as for 6061-T6 aluminum. As indicated, no creep strain occurred at a stress level of 35 ksi for any of the specimens. Since the design stress levels for the connector were not expected to exceed 35 ksi, it was concluded that creep and relaxation would not be a problem for the expected 10-hour elevatedtemperature conditions for Type 347 stainless steel connectors.

10,000-Hour Creep Data for 6061-T6 Aluminum at 70 F. Table 34 shows the creep strain estimated for the 6061-T6 aluminum specimens. Although every attempt was made to set up and conduct the tests carefully, the long duration of the tests and the small amounts of creep involved prevented accurate measurements with the available equipment. However, by making the assumption that no creep occurred at the lowest stress level, it was possible to develop the approximations shown in Table 34. The usefulness of these data is substantiated in part by another Battelle program that showed a creep strain of 10 microinches in 1400 hours for 6061-T6 at a stress of 24,600 psi. These data extrapolated to 10,000 hours would indicate a strain of 35 microinches. These tests showed that a maximum design stress for aluminum of 30 ksi would be satisfactory.

Tensile Stress,		Creep St	rain, $\mu$ in.	in.
ksi	A	K	R	Average
35	- 5	240	210	148
33	- 35	55	180	67
30	0	75	110	62
25	-10	110	100	67
20	÷5	35	35	27
15	Assumed to be	zero		

TABLE 34.	10,000-HOUR CREEP STRAIN FOR 6061-T6
	ALUMINUM AT 70 F

Note: The negative strain values were believed to be caused by temperature effects on the equipment used to measure the strain.

10,000-Hour Creep Data for Type 347 Stainless Steel at 70 F. The same problems encountered with the aluminum specimens were also encountered with the stainless steel creep specimens. By making the same assumption (zero creep at 15 ksi stress) the values shown in Table 35 were developed. As with the aluminum values, it is assumed that these strains are useful approximations for the stress levels of interest to the program. It was apparent from the tests that a maximum design stress of 35 ksi would be satisfactory for Type 347 stainless steel.



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FIGURE 55. CREEP CURVES FOR 6061-T6 ALUMINUM BAR AT 200 F AND 15,000 AND 20,000 PSI

FIGURE 56. CREEP CURVES FOR 6061-T6 ALUMINUM BAR AT 200 F AND 25,000 AND 30,000 PSI



FIGURE 57. CREEP CURVES FOR 6061-T6 ALUMINUM BAR AT 500 F AND 33, 000 PSI



FIGURE 58. CREEP CURVES FOR 6061-T6 ALUMINUM BAR AT 200 F AND 35,000 PSI



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FIGURE 59. CREEP CURVES FOR TYPE 347 STAINLESS STEEL BAR (C-MATERIAL) AT 600 F



FIGURE 60. CREEP CURVES FOR TYPE 347 STAINLESS STEEL BAR (I-MATERIAL) AT 600 F



FIGURE 61. CREEP CURVES FOR TYPE 347 STAINLESS STEEL BAR (W-MATERIAL) AT 600 F

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FIGURE 62. CREEP CURVES FOR TYPE 347 STAINLESS STEEL BAR AT 600 F AND 30,000 AND 35,000 PSI



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FIGURE 64. CREEP CURVES FOR TYPE 347 STAINLESS STEEL BAR AT 600 F AND 45,000 PSI



FIGURE 65. CREEP CURVES FOR TYPE 347 STAINLESS STEEL BAR AT 600 F AND 50,000 PSI

Tensile Stress,	, Creep Strain, $\mu$ in./in.			
ksi	c	I	U	Average
50	103	204	102	1 36
45	64	91	73	76
<b>4</b> 0	37	25	34	32
35	38	54	23	38
30	25	<b>4</b> 6	44	38
25	1	17	18	12
20	19	12	0	10
15	Assumed to b	e zero		

#### TABLE 35. 10,000-HOUR CREEP STRAIN FOR TYPE 347 STAINLESS STEEL AT 70 F

<u>10,000-Hour Creep Data for 2024-T351 and A286 Bolts</u>. Long-term, roomtemperature creep tests were conducted with  $^{2}024$ -T351 aluminum and A286 steel bolts. The program involved conducting duplicate tests on two bolt sizes at two stress levels for each alloy. The aluminum bolts were 1/4 and 1/2 inch in diameter and 3 inches long. Each bolt size was stressed at 33,000 and 42,000 psi. The steel bolts were 1/4 and 3/8 inch in diameter and were also 3 inches long. The steel bolts were stressed at 80,000 and 100,000 psi. The 3/8-inch rather than 1/2-inch steel bolts were used because of the load limitations of Battelle's creep machines.

Two foil strain gages were attached near the center and on opposite sides of each bolt. These two strain gages, along with two dummy gages, formed a 4-arm bridge. The output of each set of two gages was read on a Baldwin SR-4 strain indicator. This output was averaged and the deformation in microinches calculated for each specimen. The grips holding the specimens were designed so that a change in the length of the bolts could be measured with micrometers.

The test results summarized in Tables 36 and 37 include size, stress, total strain achieved in each bolt after 10,000 hours, creep strain, and total residual strain after removal of stress.

It is evident from Table 36 that very little creep occurred in the 2024-T351 alloy in 10,000 hours of exposure. The creep strain recorded at the stress of 33,000 psi was  $6.4 \mu in./in.$  and at 42,000 psi it was 15.5  $\mu in./in.$  In the case of the A286 steel, a relatively large amount of deformation occurred in 10,000 hours in the specimens stressed at 100,000 psi. The average residual strain was about 1518  $\mu in./in.$  or about 0.15 percent. Only 474 (average) microinches or about 30 percent of this total was actual creep strain. The balance was initial plastic strain obtained on loading because the stress exceeded the proportional limit of the material. The lower stress of 80,000 psi produced an average total strain of 141  $\mu in./in.$  Of this total, 24 microinches represented creep strain, and the balance represented the instantaneous plastic strain obtained on loading.

Specimen	Bolt Diameter, inch	Stress, ksi	Total Strain at 10,000 Hours, $\mu$ in. /in.	Creep Strain, µin./in.	Total Residual Strain, µin./in
		Strain	-Gage Measurement		<u></u>
3	1/4	33	$3.07 \times 10^3$	20,5	20.5
5	1/4	33	$3.08 \times 10^{3}$	5,5	2.5
7	1/2	33	$3.14 \times 10^3$	- 3	5
8	1/2	33	$3.14 \times 10^3$	2,5	1
				6.4 avg	7.2 avg
2	1/4	42	$3.91 \times 10^3$	13.5	11
4	1/4	42	$3.96 \times 10^3$	24.5	14,5
9	1/2	42	4.01 x $10^3$	4.5	-6
10	1/2	42	4.03 x $10^3$	<u>19.5</u> 15.5 avg	$\frac{+3.5}{5.8}$ avg

# TABLE 36.SUMMARY OF CREEP DATA FOR 2024-T351 ALUMINUM<br/>BOLTS AT ROOM TEMPERATURE (80 F)

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# TABLE 37.SUMMARY OF CREEP DATA FOR A286 STEEL BOLTS<br/>AT ROOM TEMPERATURE (80 F)

Specimen	Bolt Diameter, inch	Stress, ksi	<b>Total Strain</b> at 10,000 Hours, μin./in.	Creep Strain, uin, /in,	Total Residual Strain, µin./in.
		Strain	-Gage Measurement		
2	1/4	80	$3.05 \times 10^3$	e	172
5	1/4	80	$3.04 \times 10^3$	19	177
8	3/8	80	$2,89 \times 10^3$	47	125
10	3/8	80	2.85 x $10^3$	20	91
				24 avg	141 avg
4	1/4	100	5.16 x $10^3$	480	1491
6	1/4	100	5, $12 \times 10^3$	422	1497
7	3/8	100	5.09 x $10^3$	465	1409
11	3/8	100	5, 35 x $10^3$	530	1675
				474 avg	1518 avg

### Development of a Computerized Flange-Connector Design Procedure

This effort was started with an extensive survey of the literature. Of approximately 1000 references that were identified, over 250 were obtained and studied, including several different flange design procedures. The following general steps were common to these design procedures:

- Assembly of information on design parameters such as pressure, temperature, imposed loads, material properties, and material compatibility
- (2) Generation of a preliminary design
- (3) Analysis of the preliminary design on the basis of stress, deflection, etc.
- (4) Adjustment and reanalysis of the preliminary design
- (5) Selection and definition of a final design.

The flange design procedure that was developed for the requirements of this program was incorporated into two digital computer programs entitled IRFDP (Integral Flange Design Procedure) and LRFDP (Loose-Ring Flange Design Procedure). A listing of both programs, together with a brief discussion of the use of the programs, is presented in Appendix B. The major features of the design procedure are discussed below. In evaluating the operation and output of the procedure, it should be remembered that flange connectors incorporating the Bobbin seal have been developed specifically to achieve very low gas leakage for a wide range of operating temperatures and pressures. In other flange design procedures, the capability of designing for specific thermalgradient temperatures and weight optimization is not usually included.

#### Procedure for Designing Flanged Connectors

The procedure starts with a trial design and, by stepwise modification, ends with a design that satisfies selected performance requirements and predetermined connector parameters. Then, by selected modification of some connector dimensions, alternative designs are obtained which satisfy the performance requirements. From these alternative designs, the minimum-weight connector design is selected. A schematic diagram of the overall approach is shown in Figure 66.

<u>Predetermined Parameters</u>. The predetermined parameters are derived from the operational and functional objectives for the connectors. For example, the general design configurations (see Figure 67) are significant predetermined parameters. The materials of construction and their properties are other predetermined parameters. Others more directly relatable to the application are the minimum radial and axial seal loads per nch of seal circumference, the length-to-thickness ratio for the seal tang, the dimensions of the seal disks, the bolt-wrench clearance dimensions, the minimum and maximum bolt spacing, the minimum dimensions for the distance from the bolt circle to the flange outside diameter, and the bolt-hole clearance.







FIGURE 67. CONNECTOR CONFIGURATIONS

Type 347 stainless allel (annealed condition) and 6061-T6 aluminum are the two basic connector materials. For stainless steel connectors, the bolt material selected is A286, and the seal material is an austenitic stainless steel with soft nickel plating. For the aluminum connectors, the bolt material is 2024-T356, and the seal material is overaged 6061 aluminum alloy.

The predetermined seal loads and dimensional parameters were the result of the seal-design studies and of the qualification tests. The bolt-wrench clearances were established from the dimensions of commercially available wrenches. Bolt-hole spacing requirements were established on the basis of design experience and the ASME Unfired Pressure Vessel Code, (13)

<u>Performance Requirements</u>. An important part in the development of a design procedure is the establishment of performance requirements in sufficient detail that the requirements become integrated into the procedure. Seven computer sections were used to define the design performance requirements:

- (1) Bolt-up
- (2) Operating
- (3) Proof pressure
- (4) Burst pressure
- (5) Pressure impulse
- (6) Thermal shock
- (7) Misalignment.

The design procedure section for bolt-up imposes the conditions of initial assembly of the connector with no operating loads applied. This section is used as a base-line condition in the computer program. One of the conditions of initial assembly is that the axial bolt loading during bolt-up must be sufficient to seat the seal.

The section for operating conditions includes the requirements that the connector sustain operating pressure at maximum temperature, a specified value of stressreversal-bending load, and a minimum axial seal load which will assure that the leakage rate of the connector does not exceed the specified minimum.

The proof-pressure section requires satisfactory connector operation with an applied pressure of 1.5 times the maximum operating pressure at maximum temperature, while the burst-pressure section specifies an applied pressure of 2.0 times the maximum operating pressure at ambient temperature. The pressure-impulse section includes a fatigue stress selected to permit 200,000 pressure cycles from 0 psi to approximately 1.5 times the maximum operating pressure at ambient temperature.

The thermal-shock section includes the requirement that the connector perform satisfactorily when temperature differentials exist between the connector components. Numerical values for the temperature differentials are supplied as inputs to the design program. The design program for this condition includes load calculations relating the flexibility of the connector components to dimensional changes due to temperature differentials under normal connector operating-load conditions.

The misalignment section specifies a maximum bending load as a function of tubing size and tubing-material yield strength.

<u>Trial Design</u>. The selection of a trial design is an important part of an iterativetype design program. Trial dimensions are selected which describe, in full, the connector to be designed. Also, if the trial dimensions are close to the dimensions of a satisfactory design, the number of iterations is reduced. In the connector design procedure, minimum dimensions are selected for the trial design. For example, the trial bolt size is the minimum size, No. 10 (0. 190-in. diameter', p. ovided in the bolt-size input data. The trial flange thickness is three times the tube wall thickness. This is a minimum value chosen arbitrarily on the basis of experience. The trial bolt-circle diameter is calculated on the basis of the wrench clearance meeded for the trial bolt size. This provides a minimum bolt-circle diameter for the trial design. Other dimensions include the reinforcement at the flange and tube junction.

<u>Design Analysis</u>. For most mechanical parts, there are at least two analyses that should be carried out: (1) load analysis and (2) stress analysis. Many other analyses may also be required, such as thermal analysis, fluid-flow analysis, and vibration analysis. For the tube connectors, only load and stress analyses are conducted. Consideration of thermal characteristics and vibration characteristics is included as part of the load analysis. The consideration of fluid leakage past the seals is explicitly included in the limits established for the minimum seal loads.

As shown in Figure 66, the design analysis provides feedback to the trial design. Changes are made in the trial design to compensate for any unsatisfactory load or stress conditions discovered in the analysis. For example, if the axial seal load is too low, the trial-design bolt load is increased and the analysis is repeated. Also, if the calculated bolt stress is too high with the maximum possible number of bolts, the trial design is changed to include the next larger size bolt. Similarly, if one of the flange stresses is too high, the trial flange thickness is increased and the analysis is repeated.

Design Dimensions. After changes are made in the trial design and the analysis results satisfy all the load and stress requirements, the trial design becomes a satisfactory design.

Parameter Modifications and Optimum Design. Selected parameters can then be systematically varied and the results examined to determine an optimum design. The parameter variations must not obviate the design objectives, however.

For the integral connector, the parameters which are varied are the maximum height of the reinforcing hub at the juncture of the flange and tube, the bolt-circle diameter, and the flange outside diameter. For the loose-ring configuration, these parameters are varied, as is the amount of contact area between the loose ring and the flange. The optimum design is selected on the basis of (1) minimum weight, (2) dimensional compatibility of integral and loose-ring-flange designs, and (3) manufacturability.

### Computer Program for Designing Flanged Connectors

The computer program is the mechanism for accomplishing design synthesis and optimization. The form of the computer program is important only because the designer uses the computer program as a tool in the design procedure. Secondary attention must be given to modifying the computer program for efficiency in computer utilization.

In the flange design program, a "building block" or sectional approach is taken to provide flexibility during preparation of the program. This approach also allows relatively easy program modifications to adapt to any future changes in program objectives. Some aspects of the computer program for tube connectors are discussed to further illustrate the design synthesis and optimization procedure. A simplified schematic of the tube connector program is shown in Figure 68. Inputs not only include such things as material properties and system requirements, but provision is also made for the variation of dimensional parameters that influence design optimization.



FIGURE 68 SCHEMATIC OF FLANGED-CONNECTOR COMPUTER PROGRAM

Critical Design Parameters. Critical design parameters are selected on the basis of their importance to connector performance. Some of these critical parameters are briefly discussed.

### 1. Seal Loads

The maintenance of intimate physical contact between the seal and the mating flanges under all required loading conditions is one of the primary design aspects of the tube-connector program. Parameters which describe this aspect are the seal loads in the radial and axial directions. Seal loads, in turn, are dependent on the pressure and structural loads applied to the connector, the dimensional stability of the seal and flanges, and the relative flexure of the seal and flanges under loads.

#### 2. Bolt Load

During bolt-up, the bolt load is applied directly to the seal through the flanges. Then, as other loads are applied, the bolt load must be adequate to provide a margin for maintaining minimum loads on the seal. A somewhat less evident function of the bolt load is to minimize the effects of cyclic loading on the connector. The possibility of fatigue failure is reduced in some cases by the maintennance of unidirectional loads. Also, the chance of loosening of threaded joints is reduced by the maintenance of steady-state loads.

#### 3. Tube Bending Moment

The tube bending moment is considered as an external load applied to the connector; however, the magnitude of the tube bending moment is dependent on the tube properties and on deflections in the tube system. In the connector design program, tube bending moments are calculated on the basis of tube material properties and cyclic loading of the tube. The influence of the tube bending moment applied to the connector design is usually substantial.

#### 4. Internal Pressure

For most pressure vessels, the hoop stress resulting from internal pressure is the most important design parameter. However, for bolted flanged connectors, the structure for transferring axial loads provides increased wall thickness and therefore the hoop stress is not a critical parameter. However, the internal pressure is an important design parameter, particularly because the pressure loading in the axial direction tends to reduce the seal loads. The proof pressure is 1.5 times the maximum operating pressure, while the burst pressure is 2.0 times the maximum operating pressure.

#### 5. Thermal Gradients

Calculations are made for the effects of thermal gradients on connector flange and seal loads. The program does not include an analysis procedure to determine values for thermal gradients. However, it does include calculations to determine the relative changes in loads and deflections of connector elements in response to thermal gradients. Worst-case emperature differences between bolts, thanges, and seals are provided as inputs to the program. The thermal model used in the program is illustrated in Figure 69. Average worst-case temperature differences are taken to exist between the bolt and the flange and between the flange and the tube. The seal-element temperature is the same as the tube temperature. Relative changes in physical dimensions occur from the temperature differences. For example, when a connector is suddenly exposed to the flow of cryogenic fluid, the tube shrinks radially, the seal shrinks axially and radially, and the bolt shrinks axially. The relative significance of these dimensional changes depends on the flexibility of the connector elements and the load changes that follow.



#### FIGURE 69. FLANGE-CONNECTOR THERMAL MODEL

#### 6. Impulse Pressure

The peak value of the pressure cycle is specified as 1.57 times the operating pressure and the minimum value is atmospheric pressure. For this type of pressure cycling, fluctuating stress levels are experienced at the flange-to-tube joint. The resistance of the flange-to-tube joint to fatigue loads is improved by a reinforcing hub.

An investigation was made of the first design approach used in determination of hub dimensions as a result of the failure of the 3-inch, 1500-psi aluminum connector during the vibration test (see page 111). A stress-concentration factor of 2.0 had been selected to account for stress amplification at the juncture of the hub and the flange. Also, the calculations of maximum and minimum stress levels at the critical regions for assessment of fatigue resistance were based on combined stress levels. On review of this approach, it was found that the stress-concentration factor could, in some cases be greater than 2.0, depending on the size of fillet radius selected and the relative dimensions of the hub, flange, and tube. Also, the combined stress-level formulation did not adequately account for changes in direction of the maximum and minimum values of the calculated combined stress.

The computer program was extended to select stress-concentration factors based on connector dimensions (flange OD, hub OD, and fillet radius) and published values of calculated stress-concentration factors. (14) Also, initial choices for the fillet radii were made a part of the design procedure according to the tube size. The initial radii are 1/8, 3/16, 1/4, and 5/16 for tube sizes through 4, 8, 12, and 16 inches, respectively. The program includes the option of fixing the fillet radius according to a specified input value.

Fatigue-stress calculations were modified to consider the maximum and minimum stresses at critical sections in the axial direction. Then the assessment of resistance to fatigue failure is made according to the idealized stress-versus-strain-history procedure as outlined in Section III of the ASME Boiler and Pressure Vessel Code for Nuclear Vessels. <sup>(15)</sup> The computer program includes a fatigue safety factor of 2.0 which is consistent with the method of establishing the value of the applied bending moment based on 0.5 times the fatigue strength of the tube material, and with the appropriate stress-concentration factor applied to the maximum and minimum calculated axial stress values. Calculations for maximum and minimum values of the combined stresses, based on the octahedral shear stress theory, are also made and compared with the allowable yield strength of the connector material.

In addition to the above described improvements in the method of calculation of fatigue stress resistance, an arbitrary minimum value of hub-reinforcement thickness at the hub-flange juncture was established at 1.5 times tube-wall thickness for aluminum connectors and 1.25 times tube-wall thickness for stainless steel connectors. These minimum values are intended to compensate for observed relative sensitivities of the materials to fatigue damage and for possible inaccuracies in machining.

Load Calculations. After a trial connector design is generated, the designanalysis procedure starts with the determination of loads on the connector components. For convenience, preliminary calculations are made to establish the deflection rates of the connector components. For example, the axial deflection rate of the seal is given by:

$$DSA = \frac{L}{2 \times SEE \times 3.14 \times D_e \times t}$$
(14)

where

DSA = Axial deflection rate of seal tang, in. /lb

- L = Seal tang length, in.
- SEE = Modulus of elasticity of seal material, psi
  - $D_{e}$  = Seal tang outside diameter, in.
    - t = Seal tang thickness, in.

Figure 70 illustrates in somewhat more detail the computer-program section used in the design analysis which includes load and stress calculations. These calculations are made in sequence for each of seven design conditions. The component loads for each condition are dependent upon initial bolt-up loads, pressure-bending moments, and thermal gradients. Some sample calculations are presented for bolt-up loads and operating-pressure loads. Dimensional notations are shown in Figures 71 and 72.





### FIGURE 70. SCHEMATIC OF DESIGN ANALYSIS







### Sample Calculations<sup>†</sup>

For bolt-up loads the horizontal forces are related by:

SL1 = BL1.

(15)

+ In the calculations the asterisk symbol indicates multiplication. See Appendix B for further discussion of computer computation.
The flange moment is represented by:

$$M\phi l = RSL l * XT/2 + BL l * H2 + SL l * H3.$$
 (16)

Similar equations are prepared for the operating-pressure loads:

$$BL2 = FPL2 + SL2 + TL2$$
, (17)

$$M\phi^{2} = -RSL2 * XT/2 + BL2 * H^{2} + SL2 * H^{3} + FPL2 * H^{4} + TL^{2}$$
  
\* H^{5} + MP2 ± MC2 - (P2 \* \pi \* B \* SX) \* XT/2 . (18)

Then the changes from the bolt-up condition to the operating-pressure condition are obtained:

$$(\mathbf{BL1} - \mathbf{BL2}) + \Delta \mathbf{BL} = \Delta \mathbf{SL} - \mathbf{TL2} - \mathbf{FPL2}; \qquad (19)$$

$$\Delta M \varphi = -\Delta RSL * XT/2 + \Delta BL * H2 + \Delta SL * H3 - FPL2 * H4 - TL2 * H5 + (P2 * \pi * B * SX) * XT/2 - MP2 - MC2 .$$
(20)

Two more equations are then obtained from consideration of the relative changes in axial and radial dimensions:

$$\Delta BL * DBA = \Delta SL * DSA + \left(\frac{\Delta M\phi}{H^2 + H^3}\right) * DFA + BCR * (XT + SX)$$
(21)  
+ FCR \* XT ;

$$\Delta RSL * DSR - (P2 * B * SMD)/(4 * TA * SE) = -\Delta M\varphi + DFR * XT/2$$
(22)  
- (P2 \* B \*  $\left(\frac{A + B}{2}\right)/(4 * \left(\frac{A - B}{2}\right) * FE)$ .

Equations (19) through (22) can now be solved simultaneously to obtain values for  $\triangle BL$ ,  $\triangle SL$ ,  $\triangle RSL$ , and  $\triangle M\varphi$ . Then, starting with known bolt-up loads, the connector loads can be calculated for the operating pressure.

A similar method is used for calculating the connector loads for each design condition. For thermal gradients, for example, when the effects of relative thermal expansion are included in the analysis, the calculations include the simultaneous solution of six equations.

Load Checks. After the loads are calculated for each design condition, a check is made to determine whether the loads are satisfactory. If the axial seal load for the operating pressure, say, is calculated to be less than the required minimum, the bolt-up load is increased and the analysis is repeated. This iteration loop is illustrated in the partial computer-logic diagram shown in Figure 73.





<u>Stress Calculations</u>. With emphasis on minimum weight and strength-to-weight ratios in aerospace applications, it becomes necessary that stress calculations be performed for all suspected possible failure modes. For flanged connectors, the criterion of failure generally involves deformation rather than rupture; thus, the material yield strength is an important stress limit. The modulus of elasticity and the coefficient of thermal expansion are important material parameters in connector designs where thermal gradients are significant.

To determine the dimensions of connector designs which are judged to be satisfactory, it is necessary to establish allowable stresses for the materials of construction. Allowable stresses, in relationship to material properties, involve a complex interaction between the accuracy of the stress calculation methods, the accuracy with which loads are known, and the objective functions of the components. For example, the allowable stress for bolts is set relatively lower than the allowable stress for the flanges for several reasons: (1) bolts are loaded in tension, and in case of overload, there is no

ability to transfer loads to any other part of the structure; (2) the only bolt stress calculated is due to axial loading, while in reality the bolts have bending stresses due to flange rotation; (3) the tightening of bolts with a wrench introduces shear stresses and a reduction in axial yield strength; and (4) the threads on the bolts form notches with accompanying stress concentrations.

When a bending-moment load is imposed on the connector through the tubing, a maximum tensile stress will exist at one point on the circumference of the tube section. A diametrically opposed maximum compressive stress will arise simultaneously. However, compressive failure is not likely since the pressure end load reduces the compressive stress. While the maximum tensile stress exists at only one point on the circumference, it can be conservatively assumed that the connector may be designed as if the maximum bending stress existed all around the tube circumference.

Consideration should also be given to fatigue damage due to cyclic bending of the tube-connector assembly. In addition to bending stresses, the connector internal pressure creates axial, radial, and circumferential stresses. A simplified stress-time relationship of the combined stresses in the axial direction is represented graphically in Figure 74.



FIGURE 74. STRESS-TIME REPRESENTATION

The allowable fatigue stress when variable and steady stresses are present can be determined with the aid of a modified Goodman diagram. A more complex fatigue problem is created by impulse-pressure variation from 0 to 1.57 times operating pressure. This results in fluctuating stresses in the axial, radial, and circumferential directions. The problem of impulse-pressure fatigue is simplified if fluctuating stresses in the axial direction only are considered. This is the approach taken in the computer program.

The general methods used for stress analysis in the connector design program are based on the same stress theories used in the formulation of the ASME Unfired Pressure Vessel Code. (13) The computer-program logic diagrams applying to parts of the stress calculations for operating pressure are shown in Figures 75 and 76. Also shown on these diagrams are some of the iteration loops corresponding to stress checks. For example, when the calculated bolt stress exceeds the design allowable stress for the bolts, the trial design is changed by increasing the bolt size. Then the design analysis is repeated (load and stress analysis) for the "new" dimensions of the trial design with the "new" bolt size.







FIGURE 76. PARTIAL LOGIC DIAGRAM FOR STRESS CHECKS ON JUNCTION OF HUB AND FLANGE, CONDITION 2

### **Design** Optimization

An optimum design is one selected from among alternative designs which exhibits maximum (or minimum) values for selected characteristics. For tube connectors for aerospace applications, the optimum design is selected to minimize weight. In the connector-design program, some other characteristics are also considered in the selection of optimum designs. These include matching the bolt-circle diameters for integralflanged and loose-ring connectors. Also, for ease of manufacturing, the bolt-circle diameters are selected in increments of 1/16 inch.

<u>Parameter Modifications</u>. Alternative designs from which the least-weight designs were selected for the designs given in Appendix C were obtained by systematic variations of input parameters of the computer design program. The input parameters that were varied were: (1) the relative height of the hub reinforcement, (2) the bolt-circle diameter, (3) the bolt size, (4) the flange outside diameter, and (5) the relative amount of overlap of the loose ring and its corresponding integral flange.

Design Selection. If only one design characteristic is selected to be minimized or maximized, the selection of the optimum design is simplified. When two or more characteristics are used, they must be ordered and weighted with respect to importance. In the selection of optimum connector designs, the criteria used were: (1) minimum weight, (2) dimensional compatibility of integral- and loose-ring-flange designs, and (3) selected dimensional limits for some items to improve manufacturability and handling of the connectors.

### Design of High-Temperature Flange Connectors

In 1964, under Contract NAS-8-11523, Battelle developed an analysis procedure for estimating the relaxation time to leakage for separable tube connectors. The procedure was based upon the steady-state creep law, revised to account for the effects of primary creep.

At the beginning of the program described by this report, the possibility of including this procedure in the flange-design computer program was considered. However, as the work developed, it was mutually agreed that the needs of the Air Force were not sufficient to warrant the expense of this effort. For those instances where a hightemperature flange design is required, it is believed that the optimum design for a flange with a Bobbin seal can be most easily determined by use of the computer design program in conjunction with the "hand" calculation defined by the relaxation procedure. Accordingly, the entire final report on Contract NAS-8-11523, Subcontract No. 63-30, has been included in this report as Appendix A. The information contains not only information on the relaxation analysis procedure and its use, but also summary information on high-temperature materials.

### Design of Representative Connectors

It was originally intended that, prior to the experimental program, the computerized flange-design procedure would be used to design a number of representative connectors for different tube diameters and system requirements. However, several factors resulted in a modification of this objective. First, because of the existence of three basic connector configurations (integral/integral, loose-ring/loose-ring, and integral/loose-ring), a large number of designs would be required. Second, results from the experimental program might necessitate significant dimensional changes in the designs. Third, very few state-of-the-art large connector configurations were available for comparison with the representative Bobbin seal connectors. Fourth, if a specific flange design were needed at any time in the program, it could be obtained quickly. Finally, since considerable effort was expended in developing an experimental program that would determine the capability of the design procedure to produce a satisfactory flange-connector design for the specified system requirements, the decision was made to consider the test connectors as representative of the requirements. Illustrative aspects of these designs are discussed, and dimensions are given for all of the connectors considered for the experimental program. Subsequently, some of these connectors were eliminated from the test program.

## Connectors for Type 6061-T6 Aluminum Systems

Tables 38 through 41 summarize the major system input conditions and present the dimensions developed for integral/integral and loose-ring/loose-ring flange connectors for 3-, 8-, and 16-inch tubing for 100 and 1500-psi aluminum systems. The tables also give the dimensions for compatible integral and loose-ring flanges which are needed to form an integral/loose-ring assembly. A comparison of the connector part weights gives an indication of the weight penalty that is required for this dimensional compatibility. For these designs, the seals are overaged 6061 aluminum, the flanges are 6061-T6 aluminum, and the bolts and loose rings are 2024-T351 aluminum.

### Connectors for Type 347 Stainless Steel Systems

Tables 42 through 45 summarize the major system input conditions and present the dimensions developed for integral/integral and loose-ring/loose-ring flange connectors for 3-, 8-, and 16-inch tubing for 100 and 1500-psi stainless steel systems. The tables also give the dimensions for compatible integral and loose-ring flanges which are needed to form an integral/loose-ring assembly. The weight penalty for using compatible flanges can be seen in the tables.

Tables 46 and 47 give three system input conditions and the associated flange dimensions for 4000-psi connectors for stainless steel systems. The original intention was to fabricate and test a connector for the full temperature range from -423 to 600 F. However, the weight penalty caused by the thermal gradients was very high. When a reexamination of the possible future conditions showed that a system would be either for hot gas or cryogenic fluid, the decision was made to test two connectors. Further design analysis showed that connector fabrication costs could be saved if the hot-gas connector was fabricated and tested first, and the cryogenic connector was fabricated by remachining the hot-gas connector. This approach necessitated a small weight sacrifice between the optimum cryogenic connector and the reduced hot-gas connector because the bolt circle and size could not be changed, but the resulting connector was believed to be an acceptable test of the design procedure.

#### Conclusions From Design Optimization

As a result of the design optimization of the experimental connector configurations, it was concluded that the bolt-circle diameter is the most important variable in achieving a lightweight connector. Every effort should be made to keep this to a practical minimum. As illustrated by the 4000-psi connector, it is also important to select an accurate thermal gradient because of the small amount of deflection in the smallest connector parts. It was also concluded that little weight penalty is paid for making the integral and loosering flanges compatible.

#### TABLE 38. INPUT DATA, DIMENSIONS, AND ESTIMATED WEIGHTS FOR EXPERIMENTIAL, ALUMINUM 100-PSI INTEGRAL-PLANGE CONNECTORS



TABLE 19 - INPUT DATA, DIMENSIONS, AND ESTIMATED WEIGHTS FOR EXPERIMENTAL, ALUMINUM 100-PSI DAGE-RING CONNECTORS





loput Data

hes falses		*1				10	H)		Fian	ec-to-4 a	a bot then	nat gradic	nt, F		•
Sen I al inter	to use hatu	n failes	•			-10170	-0		Flar	ec ultima	te strengt!	1 114			42.660
ing na san sing. Na shi mini san sa	ne i nitel the	rinal oral	iund I						Fiar	we vield i	the set is a	6.)			32, 200
	ing the state the st	anal chada	ant F				1.		Rum	ultimate	uncaeth				62 000
	and the state	the next of	rate of t						Etter	wardet stre	unth mu	1			47 600
	ange trong t		ndurat F						Man 11	ultamate	streamth .	(ML)			62 600
	ange enveren	de ana dia							illustra Illustra	statist state	units max				47 000
nange der	-4.31 + ++10 1	und under K	raqueore e				0		CH I I	yie to stre	utter ler				47,000
							Dim	ICESIONS	m.						
MA 10.	TIN	TW	RL HD	RUD	RT	FOD	FID	<u>FT</u>	BX.	SUD	\$D	<u>SL</u>	Holt Size	No. of Huits	Weight, Ib
						NiM	Comparable	ic with to	tegrai Slan	ers.					
	1, 1991	0.028	1 100	1 165	0.296	1 1.97	2 944	0 279	1, 398	3, 54	2.984	0.490	1/16	14	0.745
×	M. 194919	6,046	9 742	8 287	0 505	8 147	7 930	0 450	9.212	8,79	1 970	0 672	1/4	24	3,94
16	11-1440	0,043	14 240	16 550	0 <b>85</b> 0	17.444	15.870	0 840	17,709	17,29	15 930	0 855	174	76	18.19
						6	mpatible	with lates	ral Flange						
1	1, 100	0.028	4, 100	1 165	0.296	1, 697	2 944	0 279	1.898	3.54	2 984	0.490	3716	14	0.745
*	м, (ни)	0.036	1 \$16	8 287	0 520	8 947	7 910	0 450	9, 275	8,79	7.970	0 672	1/4	24	4.1
10	11 1100	065	18 42.0	16 380	0 925	17,444	15 870	0 840	17 892	17.29	18 930	0.855	1/4	78	19.89

#### TABLE 40. INPUT DATA, DIMENSIONS, AND ESTIMATED WEIGHTS FOR EXPERIMENTAL, ALUMINUM 1800-PSI INTEGRAL-FLANGE CONNECTORS



TABLE 41 INPUT DATA, DIMENSIONS, AND ESTIMATED WEIGHTS FOR EXPERIMENTAL<sub>3</sub>. ALUMINUM 1500-PSI LOOSE-RING CONNECTORS



#### Isiput Dat :

Oncrating	DECEMBER 1	<b>N</b> 1				1.50	a .		Fla	nee-to-sea	I bot then	naterada	ent. F		16
Operating	leioperatur	14020	t.			-424/200	1		Fla	age ultima	te stiengtl	1. (%)			42,000
B H-to-rin	ic cold the	rmal grad	wut F				1		Flar	nge yaeld s	trength, p	61			32, 200
16-11-10-110	g hot the ri	nal gradu	ent f			10	1		Rug	ultimate	strength.	061			62,000
Ring-to-fla	inge cold i	bernal g	radionit, F			30	ł		Ring	z vield stre	ngth, psi	•			47,000
Ring-to-fl.	me hor th	ermal gra	idient F			14	•		boli	ultimate	strength.	psi .			62.000
Flange-to-	ncal cold i	hermal g	radiucite. F			4	4		Bott	yield stre	ngth, pai	•			47 (HN
							Dim	ាខាមសារន	ш.						
Size 10.	TOD	1.W	ROD	RL()	RT	FL D	FID	FT	BC	SOD	SED	SL	Bolt Size	No. of Bolts	Weight, Ib
						Not	Compatible	e With Int	egral Flag	ges					
.1	4,000	0, 114	4.546	3 202	0.544	3.562	2,771	0,659	3, 889	3, 350	2 811	0.473	5/16	16	1,86
×	8,000	0.306	10 673	8.424	1, 120	9.080	7.389	1 640	9.611	\$.310	7.429	0.567	1/2	28	17.48
16	16,000	0,611	24., 895	16.860	2, 150	18 084	14 780	1 450	19, 021	16 438	14,838	0,627	<b>٦/8</b>	28	121.70
						Ce	mpatible \	with Integ	ral Flange	\$					
3	3,000	0, 114	4.845	3 193	0 649	3.547	2.771	0.614	4, 192	3.360	2.811	0,473	5/16	16	2.27
8	8,000	0.306	10.673	8.404	1 120	9 080	7 389	1.640	9. <del>6</del> 11	8.310	7.429	0,567	1/2	28	17.48
16	16,000	0,611	20, 895	16.800	2 150	18.084	14,780	3.450	19.021	16.438	14.838	0,627	7/8	28	121.70

95

ø



TABLE 43. INPUT DATA DIMERSIONS, AND ESTIMATED WEIGHTS FOR EXPERIMENTAL. STAINLESS STEEL 100-PSI LOOSE-RING CONNECTORS



Input Data

Operating pressure inst	100	"lange-to-scal hot thermal gradient. F	G
Operating temperature range, F	-423/200	Flange ultimate strength at RT pai	90, 000
Bolt-to-ring cold thermal gradient, 1	23	Flange yield strength at 200 F, psi	30, 300
Bolt-to-ring lot thermal gradient, 1	7	Ring ultimate strength at 200 RT pst	130,000
Ring-to-Bange cold thermal gradient, F	18	Ring yield strength at 200 F, rsi	85,000
Brosto-flange bot the mal gradient.	ti -	Boit ultimate strength at RT, psi	130,000
Flange-to-scal cold thermal gradient. F	15	Bolt yield strength at 200 F. psi	85, 00

							Dime	<b>ns</b> ton: 10	1						
5125 10	top	TW	ROD	RID	RT	FOD	FID	FT	B(	SOD	SID	SL.	Bolt Sise	No of BL!*	Weight, 1b
		L				Not Con	spatible W	ith Integr	al Flanges						
4	1 000	0.028	4 121-	.1 061	0 236	3 713	2 944	0.324	3 914	1 550	2 984	0 509	3716	12	1 98
	N 000	0 0:15	1 768	8 177	0 400	8 973	7.010	0.495	8 238	8.810	7 970	C 697	1/4	20	13 40
1ú	16 000	0 0455	18 285	16 352	0 685	17 490	15 870	0.460	17 754	17 327	15 930	0 878	1/4	48	<b>4</b> 9-76
						Comp	arihic Will	integral	Hanges						
	1.000	0.028	4 11-	1.061	0.236	3 713	2 944	0.324	3 914	3 5.0	2 984	0 509	3/16	12	1 976
	= 000	0 035	9,825	# 177	0 400	8 971	7 930	0 495	+ 295	8 810	7 970	0 697	1/4	20	10 690
lu	11. 000	0.965	18 400	16 <b>14</b> 0	0 7 10	17 490	15 870	դ յլք.	17 932	17 327	15 910	0 878	1/4	56	51 800

TABLE 44. INPUT DATA, DIMENSIONS, AND ESTIMATED WEIGHTS FOR EXPERIMENTAL, STAINLESS STEEL 1500-PSI INTEGRAL-FLANGE CONNECTORS



TABLE 45 INPUT DATA, DIMENSIONS, AND ESTIMATED WEIGHTS FOR EXPERIMENTAL, STAINLISS STEEL 1500-PS1 LOOSE-RING CONNECTORS



								Input D	at z						
		Operating	e pressure	DSI.			1,50%		Fian	ge-to-sea	l hot the	mat grad.	cnr. F	14	
		Operating	e temperat	lune range	1 <sup>°</sup>		-423-202		Flan	e uitinai	te strengt	6 00		20.000	
		liolt-to-t	ing cold th	licitual gra	dicht, F		70		Flan	vi vield v	trouth	54 1955		35 000	
		Golt to - r	ing hot the	mal grad	ient, r		23		Rine	ultunate	411 n.m.):	ns.		130,000	
		Ring-to-f	lange cole	I thermally	gradient,	F	55		Rine	stald ste	nath an	1		N5 000	
		Rung-to+f	lange hot	thermal gr	radicat.	1	i4		fk-1r	ultor at a	trenoth			100 000	
		Dauge-te	s-stal colu	I themain	eradient		44		No. In	and the stand	and gen,	pai		13 2,000	
					<b>_</b>		0.1		2011	yit to site	igni, tai			85,000	
							D	or environ	s, 10						
Sizi _ 10	100	L.M.	100	RUD	RT	FOD	TID	FT	80	500	SID	\$1.	Boit Size	No of Beits	Weight th
						No	( Compan	He With	Integral F	langes					
4	1 (100	0, 1, 0	4, 410	1 157	0 455	3, 527	2.760	0.765	, 788	1 160	2.799	0 488	1.14	16	A (M)
	5 000	0 110	9,369	8 148	0 881	5.068	7 359	1 606	9.065	8 316	7 399	0.551	3.5	32	36.05
16	16 000	0.041	18 787	16 299	1.612	17.3.4	14 718	4,960	17.725	16 432	14 778	0 640	1.2	52	161,00
						í	ompatable	Wath In	itegial Fia	nges					
	1 000	0.450	4 40%	3 1 3 9	0 4.10	E 50 c	2 760	0,750	5 KJH	3 360	2 799	0 488	1.4	16	4.34
ч	5 000 S	0 120	9 892	8 744	0.896	8 653	7 359	1 575	4 098	F. 190	7 399	0.579	0.4	10	• 35 0× 40
16	16 000	0 641	16,757	16 200	1 612	17 194	14 718	4,000	17 725	16 432	14.778	0.649	1/1	5%	23,60 161 00

# TABLE 46. INPUT DATA, DIMENSIONS, AND ESTIMATED WEIGHTS FOR EXPERIMENTAL, STAINLESS STEEL 4000-PSI INTEGRAL-FLANGE CONNECTORS

 $\mathbf{c}^*$ 



				nput Data			23/600 F 3	ystem	70/8 10 F Sy	ste m	-423/200	System		
		c	perating pro	saure, psi			4,000		4,00	o		4000		
		0	perating ter	mperature ra	nge, F		-423/600		70/60	0	- 423	1.200		
		B	olt-to-flang	e cold there	nal gradient,	, F	55			1		55		
		B	olt -to -flang	e hot therm	al gradient,	F	55		5	5		18		
		F	larige -to -se	al cold ther	mal gradient	, F	70			1		70		
		F	lange -to -se	al hot therm	al gradient,	F	70		7	0		23		
		F	lange ultim	ate strength	at RT, psi		90,000		90,00	0	90	.000		
		F	lange yield	strength at (	500 F, psi		25,000		25,00	0	30	. 000		
		B	olt ultimate	strength at	RT, psi		130,000		130,00	0	130	.000		
		B	oit yield str	ength at 600	F. ps)		75,000		75,00	0	85	600		
					Dim	ensions, ii	٦.							
Size, in.	TOD	TW	FOD	FID	FT	_ <b>B</b> C	н	HL	SOD	SID	SL	Bolt Size	No. of Bolu	Weight, 1b
						-42	3/600 F Sys	tem						
ڌ	3,00	0.361	5.419	2.280	2.437	4.625	0, 185	ī. 360	3.166	2.317	0.216	3/8	18	24.53
						70	/600 F Syst	em						
3	3,00	0, <b>361</b>	4.982	2.277	1.853	4.1875	0.185	1, 361	3.086	2.317	0.247	3/8	15	15. 27
						-42	3/200 F Sy	stem						
3	3.00	0, <b>30</b> 6	4.982	2.389	1.738	4, 1875	0, 185	1. 282	3.197	2.429	0.258	3/8	16	14 42

TABLE 47. INPUT DATA, DIMENSIONS, AND ESTIMATED WEIGHTS FOR EXPERIMENTAL, STAINLESS STEEL 4000-PSI LOOSE-RING CONNECTORS

ROD		( ( (			FOD	₽  + sī	<u>↓</u> <u>7</u> <del>√</del>		FID	& SID	Bo	lt Size	
		Input Da	ta		-4237	ouu F Sys	te m	70/600 F	System	-423/	200 F Syster	n	
	Operatine D	essure. Dat				4.000		4.0	000		<ul> <li>φ. ωυθ</li> </ul>	-	
	Operating to	morature	ange, F			423/600		70/0	500		423/200		
	Bolt to tring	cord therm	al gradien	nt, ł		70			1		70		
	Bolt to ging	hot therma	i gradient	F		70			70		23		
	Ring-to flat	ige cold the	rmal grad	ent, F		55			1		55		
	Ring-to-flat	ge hot then	mal gradu	ent, F		55			55		18		
	Flange -to -s	al cold the	rmal grad	icai, E		55			1		55		
	Flange-to #	al hot then	mal gradu	ent, F		55			55		18		
	Hangs ultin	ate strengt	1 #1 RT. 1/	51		90,000		ao, 0	000		90,000		
	Plange yield	strength at	100 F p	а		25,000		25,0	000		30,000		
	Ring ultima	e strength a	e RT, psi			130,000		130,0	909		130.000		
	Ring yield s	rength at 60	30 C. PSC			15,000		75,0	906		85,000		
	Bult ultimat	e strength a	tRI, psi			000,000		130,0	900		130,000		
	Bolt yield H	rength at co	N) Е, рът			75,090		75,0	000		85,000		
		_		Dinstan	ions, in.								
Size, in. TOD	TW RO	) RID	R1	FOD	FID	FT	BC	SOD	510	51	Boli Size	No. of Bolts	Weight, In
ن زن از	∾.361 5.41	છે. તે તેમરે		<b>4</b> , 128	-423/60 1. 150	2 377	4. 625	<b>3</b> . Ibú	2.317	0.216	3/8	18	21, 52
(بىز ئ	0.361 5-23	2 3.382	0,723	4. 041	<u>7076</u> 2.277	0 5ystem 1 837	4. <b>43</b> 8	1.006	2.317	0.247	3/#	16	15.32
J ] ()	0. <b>30</b> 6 5.23	2 9 480	0 656	4. 041	-423.72 2.389	00 F SYH: 1.678	<u>1.11</u> \$ 9375	3, 197	₹. <b>42</b> 9	0.25H	3/N	18	14 12

### FLANGED-CONNECTOR EVALUATION

The laboratory evaluation of representative flanged connectors is described in seven parts: (1) qualification test objectives, (2) qualification testing of small aluminum connectors, (3) qualification testing of small stainless steel connectors, (4) qualification testing of small, high-pressure stainless steel connectors, (5) qualification testing of large aluminum connectors, (6) qualification testing of large, stainless steel connectors, and (7) summary and conclusions.

#### Qualification Test Objectives

A test plan was prepared and approved prior to the qualification testing. The plan was formulated to demonstrate through tests of representative connectors under worst-case, performance conditions, that the design procedure is applicable to the following:

- (1) Flanged connectors made of aluminum or stainless steel
- (2) Aluminum flanged connectors for operation from -423 to 200 F
- (3) Stainless steel flanged connectors for operation over three temperature ranges: (a) -423 to 200 F, (b) -423 to 600 F, and (c) ambient to 1200 F
- (4) Aluminum flanged connectors for operation from 0 to 1500 psi, and stainless steel flanged connectors for operation from 0 to 6000 psi.

### Selected Tests

The following general tests were selected as representative of worst-case performance conditions. The procedures followed for each connector are summarized as a part of the discussion of the test results.

1. Proof Pressure. The connector was to be subjected to one-and-one-half times the maximum operating pressure at the maximum design temperature. The maximum acceptable leakage was to be  $2 \times 10^{-7}$  atm cc/sec of helium per inch of seal circumference, and the connector was to show no permanent distortion or damage.

2. Thermal Gradient. The leakage was to be measured at the minimum and maximum design temperature and during the thermal transients. Maximum acceptable leakage with maximum operating pressure was to be  $2 \times 10^{-7}$  atm cc/sec of helium per inch of seal circumference.

3. Stress-Reversal Bending. The connector was to be cantilever mounted and subjected to 200,000 cycles, or the life of the tubing, of stress-reversal bendin , with maximum operating pressure at the maximum design temperature. Maximum acceptable

leakage was to be  $2 \times 10^{-7}$  atm cc/sec of helium per inch of seal circumference. The rate of application was not to exceed 1800 cpm, and the moment applied to the fitting during these tests was to be M = ZS<sub>b</sub>, but not more than 60 (D + 3)<sup>3</sup>, where

M = moment applied to the connector, in-lb

Z = section modulus of the tubing used with the connector, in.<sup>3</sup>

Sb = lower of (a) 0.667 x short time yield strength (b) 0.800 x stress to rupture in 2 years (c) 0.5 x 200,000 cycle fatigue strength

a, b, and c are evaluated at maximum connector design temperature

D = tubing diameter, in.

4. Pressure Impulse. The connector was to be given a proof-pressure test prior to the pressure-impulse test. All impulse testing was to be conducted at ambient temperature at the rate of  $35 \pm 5$  cycles per minute. Each impulse cycle was to constitute a rise from approximately 0 psi to a peak surge pressure and drop to approxiniately 0 psi. The peak surge pressure was to be within 1.43 to 1.57 times the working pressure, as shown by an oscillograph or other electronic measuring device. The connector was to complete 200,000 impulse cycles, after which it was again to be given a proof-pressure test.

5. Vibration. The connector was to be vibrated for 200,000 cycles at an approximate resonant frequency of 160 cps, at the maximum design temperature with the maximum operating pressure. The moment at the fitting was to be the same as for the stress-reversal-bending test.

6. Repeated Assembly. A limited number of connectors were to be assembled and disassembled 25 times and subjected to operating pressure. No leakage greater than  $2 \times 10^{-7}$  atm cc/sec of helium per inch of seal circumference was to be acceptable.

7. Misalignment. The connector was to be assembled with angular misalignment of such magnitude that the bending load would equal one-half the bending load used in the stress-reversal-bending test.

8. Tightening Allowance. The connector was to be assembled to the design fastening load plus 10 percent. If no damage occurred, the fastening load was to be increased in increments of 10 percent until damage was detected.

#### **Connectors** Selected for Test

The representative connectors selected for test are shown in Table 48. The selection of these connectors is discussed below.

Tube OD,	Alum 0 to 15 -423 to	inum, 00 Psi, 0 200 F	Stainle 0 to 15 -423 to	ss Steel, 00 Psi, 0 200 F	Stainless Steel, 0 to 4000 Psi, -423 to 600 F		Stain Ambie	less Steel, ent to 1200 F Max Pressure to be
in.	100 Psi	1500 Psi	100 Psi	1500 Psi	100 Psi	4000 Psi	100 Psi	Determined
3	x	x		x	x	x	х	x
8		x	х	x				
16	x	x	x	x				

TABLE 48.SELECTED REPRESENTATIVE CONNECTORS

The materials that can be selected for the four service ranges listed on page 3 of this report are stainless steel for all ranges, and aluminum for the low-pressure range from -423 to 200 F. The representative connectors were selected from four of the five possible service range – material combinations. The fifth combination, stainless steel for 0-6000 psi at -423 to 200 F, was not selected because the applicability of the design procedure to such connectors was expected to be demonstrated by tests with the other selected connectors, particularly the 4000-psi connectors.

The design steps required for low- and high-pressure flange connectors are not necessarily the same. Consequently, high- and low-pressure connectors were included in each service range. Two low-pressure connectors were omitted because the other connectors were expected to demonstrate the applicability of the computerized procedure to these connectors.

The high pressure for the 1200 F service range was to be established on the following basis. The connector used for the test was to be dimensionally the same as the connector designed for the high-pressure test of the 0 to 4000-psi service range. A period of time (approximately 3 hours) at 1200 F was to be selected so as to be representative of high-temperature systems and also to be compatible with the test operations. The pressure was then to be selected as the maximum pressure at which the connector could operate satisfactorily for the time period selected. (As the testing effort encountered unexpected difficulties, tests with 1200 F connectors were deleted from the program.)

If there were no limitations in the facility test capabilities, it would have been necessary only to test connectors of the smallest and largest sizes to demonstrate the applicability of the design procedure over the complete size range. However, an 8-inch connector was the largest connector that could be subjected to the stressreversal bending test at Battelle. Consequently, 8- and 16-inch connectors were selected to represent the largest connectors, depending on the tests required.

Three-inch connectors were selected to represent the smallest connectors because: (a) the design procedure for smaller sizes included allowances for fabrication limitations, (b) the 3-inch size represented the largest size in two service ranges, and

(c) use of a 3-inch size in all service ranges permitted considerable standardization of test equipment and procedures and consequent cost savings in the test program.

# Tests Selected for Each Connector

It was not possible to subject each connector assembly to every test. One reason was that three of the tests (stress-reversal bending, vibration, and pressure impulse) were fatigue tests. Another reason was that the assembly required for a thermalgradient test could not readily be used for a vibration test. It was decided that two connector assemblies of each design would be used for a complete series of tests. It was also decided that it was not necessary to conduct a complete series of tests for every connector design if the tests for one connector design demonstrated to an acceptable degree the applicability of the design procedure to other connector designs.

As shown in Table 49, a complete series of tests was planned for two connector designs: (1) the 3-inch, 1500-psi aluminum connector, and (2) the 3-inch, 1500-psi stainless steel connector. (The test numbers coincide with those given previously.) These tests were expected to demonstrate the applicability of the design procedure to small connectors for all performance aspects except: (1) low pressure, (2) pressure in excess of 1500 psi, and (3) operation at 600 F.

The low-pressure operation of small connectors was to be demonstrated by Tests 1-3 and 8 on an aluminum connector, and by Tests 1, 2, and 4 on a stainless steel connector at 600 F. Test 5 was not planned for low pressure because the stressreversal bending at low pressure, combined with the vibration test at 1500 psi, was expected to demonstrate the ability of the connector to sustain vibration at low pressure. Tests 6 and 7 were not planned for low pressure because the connector features related to repeated assembly and misalignment were not significantly affected by the lowpressure design considerations.

The applicability of the design procedure for pressures higher than 1500 psi was to be demonstrated by Tests 1-4 and 8 with a 4000-psi stainless steel connector assembly at 600 F. The overall performance capability of connectors at 600 F was to be demonstrated by the same tests and by Tests 1, 2, and 4 at 600 F on a lowpressure stainless steel connector. Tests 5-7 were not to be conducted for the same reasons described above in relation to the low-pressure demonstrations.

As shown in Table 49, tests with the 8-inch connectors were to demonstrate the performance of large connectors under all conditions except pressure impulse (Test 4), vibration (Test 5), and tightening allowance (Test 8). Tests 6 and 7 were planned for only one assembly each because the design considerations for repeated assembly and misalignment were not expected to be significantly affected by either pressure or material.

The performance of large connectors during Tests 4 and 8 was to be demonstrated with 16-inch connectors. Because of the importance of Tests 1 and 2, Test 1 was to be conducted with every 16-inch connector, and Test 2 was to be conducted with two of the 16-inch connectors. Test 5, the vibration test, was not planned for any large connectors because the test was difficult to conduct with large sizes, and because the resistance to vibration was to be demonstrated by vibration tests with the 3-inch connectors, by stress-reversal-bending tests with the 8-inch connectors, and by pressureimpulse tests with the 16-inch connectors.

Tube O	D,	Aluminum,	-423 to 200 F	Stainless Steel,	-423 to 200 F
3	Tests:	1-3,8	1-4, 5-8		1-4, 5-8
	Assembly:(a)	L-L	I-L, I-I		L-L, L-L
8	Tests:		1-3,6	1-3,7	1-3
	Assembly:		I-L	I-L	I- L
16	Tests:	1, 2, 4, 8	1,4,8	1,2,4,8	1,4,8
	Assembly:	I-L	I-L	I-L	I-L
	S	tainless Steel	, -423 to 600 F		
		100 Psi	4000 Psi		
3	Tests:	1,2,4	I-4, 8		

I-I

## TABLE 49. QUALIFICATION TESTS SELECTED FOR REPRESENTATIVE CONNECTOR ASSEMBLIES

Assembly:

(a) 1-I = integral/integral flange assembly
 I-L = integral/loose-ring flange assembly
 L-L = loose-ring/loose-ring flange assembly.

I-L

### Qualification Testing of Small Aluminum Flanged Connectors

The test schedule shown in Table 49 was followed, although the type of assembly was altered for some of the tests. Details of the test procedures and the test results are discussed for each of the eight types of tests.

### Test 1. Proof Pressure

<u>heater</u> was located on the integral flange.

Next, an assembled connector and the seal pressurizing line were attached to the base-plate assembly. A Bobbin seal was placed between the transition flange and the base plate. Finally, the vacuum chamber was placed around the connector and the pressurizing, electrical, and vacuum leads were connected.

Each test was started by pressurizing the volume enclosed by the primary connector seal and the secondary seal with helium and by pressurizing the remaining internal connector volume with nitrogen. Both volumes were pressurized simultaneously to the proof-pressure level (150 psi for the 100-psi connector, and 2250 psi for the 1500-psi connector). After 5 minutes, the pressure in both volumes was reduced to zero and the connector temperature was increased to 200 F, at which time the internal pressure in both volumes was again raised to the proof-pressure level. After 15 minutes at maximum temperature and pressure, the connector was cooled to room temperature while the proof pressure was maintained in the secondary seal volume and the connector volume. After 5 minutes at room temperature the test was concluded. The helium mass spectrometer was used to monitor leakage during the entire test.

Test Results. The maximum leakage rate measured during the test of the 100-psi connector was  $1.3 \times 10^{-9}$  atm cc/sec per inch of seal circumference, while the maximum leakage rate measured for the 1500-psi connector was  $3.1 \times 10^{-9}$  atm cc/sec per inch of seal circumference.



FIGURE 77. THERMAL-GRADIENT ASSEMBLY

### Test 2. Thermal Gradient

Test Procedures. The thermal-gradient test was conducted with the 100-psi and 1500-psi assemblies used for the proof-pressure tests. Figure 78 is a schematic drawing of the major components of the thermal-gradient test system that was tried initially. (The connector assembly was rotated  $180^{\circ}$  from the position shown in Figure 77.) At the start of the test, the LN<sub>2</sub> tank was filled with LN<sub>2</sub> and pressurized to 50 psi. The cryogenic valve was used to valve the LN<sub>2</sub> into the connector. When the boiling of the LN<sub>2</sub> had raised the pressure in the entire system to 1500 psi and the vent valve was venting gaseous nitrogen, the pressure in the LN<sub>2</sub> tank was increased until an equilibrium condition had been reached for the in-flow of LN<sub>2</sub> and the outflow of gaseous nitrogen.

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#### FIGURE 78. THERMAL-GRADIENT-TEST SYSTEM

Operation of this system was unsatisfactory because the large heat sink of the base plate prevented rapid cooling of the connector. It was found that sufficiently rapid cooling could be obtained if the base plate was precooled with dry ice and if the stub flange extension of the thermal-gradient assembly was cooled independently during the cooling cycle. A second type of vacuum chamber was made which completely enclosed the stub flange extension, thus eliminating the elastic seal at that point which would not seal in the presence of liquid nitrogen. Metal-to-metal contact was maintained between the stub flange extension and the vacuum chamber so the extension could be cooled by immersion of the lower part of the vacuum chamber in LN<sub>2</sub>.

The temperature of the heaters was controlled by a Variac. Three thermocouples were used to determine the thermal gradients. They were located on the stub flange, on the middle of one of the studs, and on the loose ring midway between two bolt holes. The thermocouples were also found to provide a good indication of the presence of liquid nitrogen in the connector.

At the start of the thermal-gradient test, the steel base plate was precooled with dry ice, and the extension of the stub flange was precooled with liquid nitrogen. The connector was then heated to 200 F. (The leakage of each connector at this temperature had been checked during the proof-pressure test.) The heaters were then turned off and the volume enclosed by the primary connector seal and the secondary seal was pressurized with helium to the maximum working pressure (100 psi and 1500 psi), while the remaining connector volume and the LN<sub>2</sub> standpipe were pressurized simultaneously, and LN<sub>2</sub> was forced into the connector. When thermal equilibrium had been reached, the in-flow of LN<sub>2</sub> was stopped, and the heaters on the connector were used to raise the connector temperature to 200 F. The test was designed to cool the connector as rapidly as possible to simulate the sudden flow of LN<sub>2</sub> through a connector that had been exposed to the hot sun. The heating cycle was conducted relatively slowly to simulate heating from atmospheric conditions.

Test Results. The thermal gradients, equilibrium temperatures, and maximum leakage rates for the 100-psi and 1500-psi connectors are shown in Table 50. The difference between the average thermal gradients obtained during the qualification tests and those obtained for the simulated assemblies (see Table 26) were caused by differences in the location of the thermocouples. It was believed that the test procedure had produced representative maximum thermal-gradient conditions. It can be seen that all of the measured leak rates were well below the required value of  $2 \times 10^{-7}$  atm cc/sec per inch of seal circumference.

Maximum Operating		Thermal Gra	adient, F		Maximum Leakage
Pressure, psi	Cycle	Stub Flange to Loose Ring	Loose Ring to Stud	Equilibrium Temp, F	Rate <sup>(a)</sup> , atm cc/sec
100	1	60	12	-292	1.8 x 10 <sup>-9</sup>
	2	56	13	-242	$0.8 \times 10^{-9}$
	3	50	12	-293	$0.8 \times 10^{-9}$
	4	48	16	-297	$0.8 \times 10^{-9}$
	5	67	9	-297	$1.2 \times 10^{-9}$
	Avera	age 65	12	-294	1.1 x 10 <sup>-9</sup>
1500	1	34	15	-225	$1.2 \times 10^{-8}$
	2	37	12	-217	$0.9 \times 10^{-8}$
	3	67	18	-228	$0.4 \times 10^{-8}$
	4	77	(b)	-210	$0.6 \times 10^{-8}$
	5	54	(b)	-292	$0.6 \times 10^{-8}$
	6	64	(Ъ)	-212	$0.6 \times 10^{-8}$
	Avera	ige 58	15	-231	$0.7 \times 10^{-8}$

TABLE 50. THERMAL GRADIENTS, EQUILIBRIUM TEMPERATURES, AND MAXIMUM LEAKAGE RATES FOR 3-INCH, ALUMINUM FLANGED CONNECTORS

(a) Helium leakage rate per inch of seal circumference.

(b) Thermocouple failed after three cycles,

## Test 3. Stress-Reversal Bending

<u>Test Procedures</u>. This test was conducted with the thermal-gradient assemblies that were used for Tests 1 and 2 for the 100-psi and 1500-psi connectors. Each assembly was mounted as the base of a rotating cantilever beam in the test equipment shown in Figure 79. The cantilever beam was designed to provide mechanical resonance at the operating speed of the machine (1800 rpm). The cantilever beam was clamped to the stub flange extension provided on the thermal-gradient assemblies, and a fine-tuning weight was used to provide adjustment for mechanical resonance of the different assemblies. Strain gages were attached to the cantilever beam to determine the moment being applied to the connector.



FIGURE 79. 4-INCH-DIAMETER PIPE SPECIMEN BEING INSTALLED IN STRESS-REVERSAL-BENDING MACHINE

The first 1500-psi aluminum connector was fabricated with flanges that were too thick. This connector was used to calibrate the equipment for the proof-pressure test, the thermal-gradient test, and the stress-reversal-bending test. As the speed of the stress-reversal-bending calibrations for this connector was increased to 1800 rpm, the strain gages or the cantilever beam recorded a reduction in strain. This could not be changed by any adjustment of the tuning weight. An examination of the equipment showed that the cantilever beam was not stiff enough to maintain the configuration of a simple cantilever at speeds above 500 rpm, and a reverse bend was being introduced dynamically. Not only did this make the strain-gage readings worthless, but an excessive moment was being applied to the connector. This problem was corrected by selecting a low test speed. However, it was decided that strain gages would be applied to each connector as a check on the strain gages on the cantilever beam. Each thermal-gradient assembly was mounted in the test equipment and heated to 200 F. The volume between the primary-secondary seals was pressurized with helium at the maximum working pressure (100 psi and 1500 psi), and the connector volume was simultaneously pressurized with nitrogen to the same pressure. A bending moment of 2521 in-lb was applied to the 100-psi connector, and a bending moment of 9500 in-lb was applied to the 1500-psi connector. A speed of 392 cpm was selected for each test.

<u>Test Results</u>. The 100-psi connector was subjected to 218,250 cycles of stressreversal bending before failure occurred. The failure was a group of three or four very small holes located 3 inches from the bottom of the mounting flange on the tubular portion of the stub flange. It was believed that the failure was caused by localized stresses resulting from an indentation in the 0.028-inch tube wall approximately 1/4 inch from the point of failure. The maximum helium leakage rate measured during the test was  $1.5 \times 10^{-9}$  atm cc/sec/inch of seal circumference.

The 1500-psi connector was subjected to 241,500 cycles of stress-reversal bending before failure occurred. The failure was a fine circumferential crack, approximately 1/2 inch long, located 1-1/2 inches from the O-ring groove, on the tubular portion of the stub flange. The maximum helium leakage rate measured during the test was 0.35 x  $10^{-9}$  atm cc/sec per inch of seal circumference.

Because neither failure in the aluminum connector assemblies occurred in the connector material, and because each test achieved the required 200,000 cycles of stress-reversal bending, the test results were judged to be satisfactory.

### Test 4. Pressure impulse

Test Procedures. This test was conducted with a 3-inch, 1500-psi "vibration assembly" instead of a "thermal gradient" assembly. The vibration assembly consisted of a connector without a secondary seal, with tube extensions, and with a pressurizing port, similar to the connector shown in Figure 80.



FIGURE 80. VIBRATION ASSEMBLY

Figure 81 shows schematically the arrangement of the pressure-impulse equipment. Hydraulic pressure was adjusted for 2250 psi at the output of the power unit, and a solenoid-actuated directional-control valve was used to control the flow to the test fitting. A Bell-O-Fram type diaphragm was used to separate the hydraulic oil from the alcohol in the test fitting. A switch and motor-driven gear-cam arrangement was designed to operate the solenoid valve to produce the desired cycling rate. Needle valves were installed in the inlet and return lines of the directional-control valve to provide adjustable restriction on the flow of oil and thereby control the rate of pressure rise.



#### FIGURE 81. SCHEMATIC DIAGRAM OF PRESSURE-IMPULSE EQUIPMENT

The connector was first assembled for a helium leak check. Because of a previous failure in the vibration test (see page 111) a modified 3-inch, 1500-psi aluminum connector was used which contained a stronger hub and sixteen 5/16-inch studs. The nuts were torqued to 110 in-lb, using a 50/50 mixture of Lubriseal and MoS<sub>2</sub> to lubricate the stub threads and nut collars. A leakage check at room temperature showed no detectable leakage at 2250 psi of helium. Pressure traces recorded at various times throughout the test showed that sharp impulses were being obtained at a rate of 40 cycles per minute, and that dwoll times of approximately 3/5 second existed at both the maximum and minimum pressure levels. A typical trace is shown in Figure 82.



### FIGURE 82. PRESSURE CYCLES MEASURED FOR 3-INCH, 1500-PSI ALUMINUM CONNECTOR

Test Results. The testing was terminated after 200,000 impulse cycles with no evidence of liquid leakage. The connector was drained and heat was applied to assist in removal of the alcohol. A leak check showed no detoctable leakage at a helium pressure of 2250 psi.

Subsequently, when the seal in the 8-inch, 1500-psi aluminum connector failed after approximately 14,700 cycles of pressure-impulse testing, the seal from the 3-inch connector was given a Zyglow examination. Fatigue cracks were found at the base of both seal disks, which indicated possible failure at about 300,000 cycles. This showed a need for an increase in the thickness of the seal disks for the 3-inch, 1500-psi connector.

### Test 5. Vibration

Test Procedures. The vibration test was conducted with a 1500-psi vibration assembly similar to that shown in Figure 80. The assembly was mounted as a simply supported beam. The connector was supported at two nodes, and a yoke-shaped device was fabricated to transmit the input from the Calidyne shaker to the connector at the nodes. A dummy connector was used to check out the test equipment and procedures.

Despite the tests with the dummy connector, considerable difficulty was encountered with the first assembly. The instrumentation used to monitor the test consisted of a vibration meter for monitoring acceleration, velocity, and displacement; an Ellis meter for strain readout for statically calibrated strain gages placed on a tubing extension; and an oscilloscope for strain-level monitoring. Many attempts were made to adjust the tuning weights and connector support locations, but it was impossible to maintain the required frequency of vibration, and the strain never rose above about 50 percent of the required strain. When the connector failed prematurely, it was decided that the connector design had to be modified for a second test and that an improved test procedure had to be developed.

The next vibration test was conducted with a vibration assembly which contained strengthened hubs. The connector was supported at two locations by small steel cables and the center of the connector assembly was fastened horizontally to the Calidyne machine. This arrangement was found to provide good frequency and amplitude control for the assembly. The flexible cables did not adversely affect the vibration of the connector. The connector was pressurized with water at 1500 psi and an enclosure with

radiant heaters was used to heat the assembly to 200 F. The instrumentation of the second test was the same as for the first test, with the addition of a pressure gage and a thermocouple. A test with a third vibration assembly was conducted identically to the test with the second assembly.

<u>Test Results</u>. A pretest room-temperature leak check of the first 1500-psi vibration assembly showed a leakage rate of  $6.5 \times 10^{-8}$  atm cc/sec per inch of seal circumference. During the vibration test, the first assembly broke at the base of the integral flange after only about 50,000 cycles at an average stress of 7000 psi (compared with a design stress of 14,000 psi with a bending moment of 9500 in-lb). This premature failure occurred despite the fact that the connector was neither pressurized nor heated. Examination of the stub flange showed an extensive crack at the flange-to-hub location, indicating incipient failure of that flange also.

A second vibration assembly was fabricated similar to the first but with strengthened hub sections (see page 86). The connector assembly was pressurized to 1500 psi with water, heated to 200 F, and vibrated at the design stress level at a frequency of 160 cps. Failure occurred after 28,200 cycles at the bulkhead in the simulated tubing, about 6 inches from the connector. This premature failure was a surprise because the calculated and measured strain at this location was approximately one-half the strain at the transition joint, the point of expected failure. Although a flaw in the bar stock could have caused the failure, the bulkhead was believed to have caused an unexpected stress raiser.

In the third vibration assembly, the bulkheads were moved to the ends of the assembly, about 18 inches from the connector. With a frequency of approximately 179 cps and the design bending moment, the connector failed at the tube-to-hub transition section after 53,000 cycles. This was the expected location of failure because of the change in section, and the test was considered to be an acceptable test of the connector because the connector was shown to be stronger than the tubing or the tube-to-hub transition. However, the failure of the tube-to-hub transition at such a low stress level was believed to represent a critical problem for the system designer, because substantial evidence was believed to have been developed that an unknown stress raiser was being encountered in the 3-inch tubing system in the vibration mode. (The further investiga-tion of this problem is included in a follow-on effort, Contract No. F04611-69-C-0028).

### Test 6. Repeated Assembly

<u>Test Procedures</u>. The repeated assembly test for a 3-inch, 1500-psi aluminum connector was conducted with flanges from the first, failed vibration assembly. The test utilized the seal cavities and the undamaged inside diameter of the flanges. O-ringsealed supporting base plates were made so an axial load could be applied to the flanges in a Universal testing machine. Pressurized helium was introduced to the inside of the seal area through one base plate, while a cylindrical extension of the other base plate met with an O-ring seal on the outside diameter of the first base plate to form a vacuum chamber for the helium mass spectrometer. All of the assembled seals were checked with 1500-psi helium pressure. Test Results. Table 51 shows the maximum leakage rate measured for each assembled seal. After the test, an examination of the connector sealing cavities with a 20-power lens showed no sign of wear or deformation at the surfaces where sealing occurred.

Seal	Leakage Rate <sup>(a)</sup> , atm cc/sec	Seal	Leakage Rate <sup>(a)</sup> , atm cc/sec
1	<b>x</b> (b)	14	X
2	x	15	0.14 x $10^{-9}$
3	x	16	x
4	x	17	x
5	x	18	x
6	x	19	$0.27 \times 10^{-9}$
7	x	20	x
8	x	21	x
9	x	22	x
10	x	23	x
11	x	24	x
12	$0.81 \times 10^{-9}$	25	x
13	$0.97 \times 10^{-9}$		

TABLE 51. REPEATED-ASSEMBLY-TEST LEAKAGE DATA FOR 3-INCH 1500-PSI ALUMINUM CONNECTOR

(a) Helium leakage rate per inch of seal circumference.

(b) Denotes no detectable leakage.

#### Test 7. Misalignment

<u>Test Procedures.</u> The misalignment test was conducted with the 3-inch, 1500-psi "vibration assembly" that had been used for the pressure impulse test. The cavities of this connector were machined to a depth of 0.080 inch instead of the standard 0.020 inch to produce a greater length of seal contact area for conditions of extreme misalignment.

When a tubing system is assembled in practice, it is expected that axial misalignment will exist between component parts of the system. Physical displacement of interconnecting tube ends is required to compensate for axial misalignment, and tube-end displacement results in angular misalignment at the connectors. The assembly of angularly misaligned connector halves produces a bending moment which is distributed among the system components and tubing. The bending moment will increase during connector bolt-up until the angular misalignment between connector halves has been reduced to zero. This action produces more axial motion on one side of the connector than the other, with a resultant relative motion between the sealing disks and connector cavities. The seal must be able to withstand this sliding action while maintaining its ultimate requirement for sealing.

A test fixture was fabricated to simulate a misaligned tubing condition. One half of the connector was clamped to a rigid base with its tube axis at an angle of 2 degrees to the base. The other connector half was positioned but not clamped by a support close to the connector. Shims were used under this support to provide angular changes in the connector axis which would add to or subtract from the 2-degree angle established by the clamped connector half. Initial test misalignments of 1, 2, and 3 degrees were established by this method.

A tube extension providing a moment arm was added to the unclamped connector half. A bending moment equal to 1/2 the moment established for stress-reversal bending was required for this test, and the moment was provided by placing a weight on the tube extension. During the test it was found that the moment-producing weight caused a high transverse load on the seal due to leverage about the connector inner support block which acted as a fulcrum. A force of this magnitude would not be encountered in the assembly of an actual tubing system, and its effect was therefore removed from the test setup through the use of an outer tube support.

The seal was assembled into the connector at the desired angle of misalignment and the nuts were snugged down fingertight on the connector bolts. The weight was placed on the connector tube extension which was supported by the inner and outer tube supports. The connector bolts were then tightened with progressively increasing torque in a boltto-bolt sequence. As the bolt load was increased, the unrestrained connector half raised from the supports and a moment was applied to the entire connector. The bolt torque was increased in a continued bolt-to-bolt sequence until the required bolt preload was attained on all bolts. This assembly procedure satisfied the test requirement of initial angular misalignment of the connector halves while providing the bending moment and transverse loads that are expected during the process of bolt-up. In fact, the application of the full moment during practically the entire bolt-up procedure was significantly more stringent than will be encountered in practice, when the moment will normally increase with bolt tightening.

Test Results. Leakage checks were made with 1500-psi helium on each of the assemblies resulting from initial misalignment angles of 1, 2, and 3 degrees. No measurable leakage was found on any of the assemblies. Because some hand guidance was required during the early portion of the bolt-up of the 3 degrees of misalignment, it was apparent that a greater seal cavity depth will be required if a misalignment capability of 3 degrees is to be provided in the 3-inch connector.

### Test 8. Tightening Allowance

<u>Test Procedures.</u> Each of the fourteen No. 10-32 studs of the 100-psi connector was initially tightened with a torque of 24 in-lb, using a 50/50 mixture of Lubriseal and MoS<sub>2</sub> applied to the nut faces and stud threads. Each of the fourteen 3/8-in. studs of the 1500-psi connector was initially tightened with a torque of 175 in-lb with the same lubrication procedure used for the 100-psi connector. Measurements were then made of the dimensions believed to be most useful in determining the occurrence of yielding in the connector (see Figure 83). Each connector was then tightened with additional torque increments of 10 percent, with measurements being made after each torque increment.



FIGURE 83. TIGHTENING -ALLOWANCE TEST MEASUREMENTS ON 3-INCH ALUMINUM CONNECTORS Measurements were made at four equally spaced locations.

<u>Test Results</u>. The measurements showed that the greatest portion of the total change in the selected dimensions occurred during the first 10 percent increment of stud overtorque. Because a considerable amount of radial taper in the loose rings occurred as the result of the connector preloads, it is believed that the continued increase in radial taper of the rings caused a shift in the center of pressure between the nut face and loose ring. This shift in pressure toward one side of the stud resulted in an increase in the friction moment arm, which required a larger torque to overcome friction. The portion of the applied torque remaining for conversion to axial stud load was lessened, with a resultant decrease in connector dimensional changes. The radial taper of the loose rings essentially stopped when the stud shanks prevented further increase in the taper.

In the 100-psi connector, increased elongation of the stude was noted with the application of 50 percent overtorque. This was judged to be evidence of yielding of the stude. Although stud yielding was not encountered on the 1500-psi connector with 50 percent overtorque, initial yielding was indicated for the next 10 percent increment. In either case, the sealing capability of the connector was unchanged by the application of 150 percent of the recommended torque.

### Qualification Testing of Small, Stainless Steel Connectors

The test schedule shown in Table 51 was followed. Details of the test procedures and test results are discussed for each of the eight types of tests.

### Test 1. Proof Pressure

Test Procedures. The proof-pressure test procedures described previously for the 1500-psi 3-inch aluminum flanged connector were used for the 3-inch stainless steel flanged connector. The connector contained two loose-ring flanges machined as part of a thermal-gradient assembly similar to that shown in Figure 77. A torque of 95 in-lb was applied to the sixteen 1/4-inch A286 studs, using the same lubrication procedures as for the aluminum fasteners. Test Results. The maximum leakage rate measured during the proof pressure test was  $1.52 \times 10^{-8}$  atm cc/sec of helium per inch of seal circumference.

### Test 2. Thermal Gradient

Test Procedures. The test procedures developed for the 3-inch aluminum connectors were used for the 1500-psi stainless steel connector.

Test Results. The thermal gradients, equilibrium temperatures, and maximum leakage rates for the 1500-psi stainless steel connector are shown in Table 52. Some difficulty was encountered in controlling the flow of the liquid nitrogen in the first five cycles and this resulted in variation and high readings for the equilibrium temperatures for these cycles. However, for the sixth cycle, the liquid nitrogen was introduced without pressure in the connector so that a very low equilibrium temperature was reached as fast as possible (boiling gas could be vented immediately). The connector was then pressurized for a leak check. Because the thermal gradients in the sixth test were comparable to those in the other tests, and because the thermal gradients were believed to be the most severe condition of this test, the test was judged to be realistically severe. The measured rate of helium leakage remained at very low values for all the test cycles.

	Thermal Gra	dients, F	Equilibrium	Maximum Leakage		
Cycle	Stub Flange to Loose Ring	Loose Ring to Stud	Temperature, F	Rate, atm cc/sec <sup>(a)</sup>		
1	100	146	-130	3.1 x 10 <sup>-9</sup>		
2	88	152	-150	2.6 x 10-9		
3	86	142	-160	$2.6 \times 10^{-9}$		
4	107	113	-100	$3.7 \times 10^{-9}$		
5	86	142	-165	$1.0 \times 10^{-9}$		
6	87	134	-300	$1.0 \times 10^{-9}$		
Average <sup>(b)</sup>	93	139	-141	$2.6 \times 10^{-9}$		

TABLE 52. THERMAL GRADIENTS, EQUILIBRIUM TEMPERATURES, AND MAXIMUM LEAKAGE RATES FOR 1500-PSI, 3-INCH STAINLESS STEEL CONNECTOR

(a) Helium leakage rate per inch of seal circumference

(b) Average shown for first 5 cycles only.

#### Test 3. Stress-Reversal Bending

Test Procedures. This test was conducted with the 1500-psi thermal-gradient assembly used for the proof pressure and thermal gradient tests. The procedures were identical to those used for the 1500-psi aluminum connector. A bending moment of 11,288 in-lb was applied to the stainless steel connector.

<u>Test Results</u>. The stainless steel connector was subjected to 352,860 cycles of stress reversal-bending without failure. The maximum helium leakage rate measured during the test was  $1.42 \times 10^{-9}$  atm cc/sec per inch of seal circumference.

### Test 4. Pressure Impulse

Test Procedures. A 3-inch, 1500-psi stainless steel "vibration assembly" (similar to that shown in Figure 80) was subjected to a pressure impulse test using the same procedures described for the 3-inch, 1500-psi aluminum connector. The stainless steel connector, which had no secondary seal, consisted of an integral/loose-ring configuration. An assembly torque of 110 in-lb was applied to the sixteen 1/4-inch A286 bolts.

A room-temperature leak check of the assembled connector showed no detectable leakage of helium at a pressure of 2250 psi. Operation of the test equipment produced a temperature-stabilized condition with a maximum pressure of 2180 psi (which was greater than the mimimum specified pressure).

Test Results. The test was terminated after 200,005 pressure-impulse cycles with no indication of liquid leakage. A leak check with 2250-psi helium after removal of the alcohol showed no detectable helium leakage.

### Test 5. Vibration

<u>Test Procedures</u>. The connector used for the 3-inch, 1500-psi stainless steel vibration test was the integral/loose-ring vibration assembly that had been used previously for the pressure impulse test. On the basis of the failures encountered in the vibration tests of the aluminum connectors it was decided that the flange-to-hub radii should be increased. Consideration was given to machining a new assembly with increased hub thicknesses, but it was decided that the stainless steel would probably be less susceptible to the effects of the stress raiser than the aluminum, and the thinner hub would probably be satisfactory.

Strain gages were attached to the assembled connector at the flange-to-hub radius, at the tube-to-hub transition, and on the tube. The strain gages were calibrated statically for the required bending moment, and the leakage rate was monitored for 30 minutes with the connector pressurized with 1500-psi helium. No leakage could be detected.

A room-temperature vibration check with no internal pressure produced a tube strain of 464  $\mu$ in./in. at a resonant frequency of 186 cps. Strains of 458 and 710  $\mu$ in./in. were measured at the tube-to-hub transition and the hub-to-flange radius, respectively.

The assembly was pressurized, heated, and tested in the same manner as was the 1500-psi aluminum connector. Although the power to the Calidyne machine had to be slightly increased periodically, the test was completed without incident.

Test Results. No indication of liquid leakage was detectable after 475,000 cycles. The water was removed from the connector and the connector was heated to remove residual moisture. A leak check with 1500-psi helium showed no detectable leakage.

Although the connector showed no leakage, the diameter of the simulated tubing between the connector and the bulkheads had enlarged 1/16 inch. Because the calculated

and measured stresses in the tubing before and during vibration did not exceed the yield stress of the stainless steel, and in fact were about one-half the yield stress, it was hypothesized that the same, unknown, stress raiser existed in the 3-inch stainless steel connector as in the 3-inch aluminum connector.

## Test 6. Repeated Assembly

Test Procedures. The 3-inch, 1500-psi stainless steel thermal-gradient assembly was adapted for use in the repeated assembly test. Seal loading was accomplished in a Universal testing machine. An axial load was applied to the connector through a tube whose mean diameter was equivalent to the connector bolt-circle diameter. The load was transmitted through the tube and connector loose ring to the stub flange. The load path from this point was through the scal tang to the integral flange, and then through the connector to the base plate with its pressurizing fittings. A vacuum chamber, consisting of a tube closed at one end, was placed over the assembly. The testing machine pressed against the closed end of the vacuum chamber, which rested on the loading tube described above. The open end of the vacuum chamber moved down over an O-ring seal located in the outside circumference of the base plate. This arrangement not only provided an accurate means of measuring the seal seating forces, but also saved the time normally required to tighten the studs.

An axial load of 25,000 pounds was applied as a preload to each of the 25 seals tested, and all of the seals were leak checked with 1500-psi helium pressure.

Test Results. Table 53 shows the maximum leakage rates measured for the 25 stainless steel assemblies. At the time, it appeared that the existence of four seals with excessive leakage rates was caused by poor quality control of the nickel plating. However, growing evidence from the use of stainless steel seals in threaded connectors, as well as from subsequent tests with the 8-inch flanged connectors, indicated that a significant increase was required in the radial sealing load of stainless steels to obtain reliable performance. Examination of the seal cavities of the connector flanges with a 5-power microscope showed no wear or deformation at the surfaces where sealing occurred.

### Test 7. Misalignment

<u>Test Procedures</u>. The misalignment test was conducted with the 3-inch, 1500-psi stainless steel "vibration assembly" that was used for the vibration test. The enlargement of the tubing extension that occurred in the vibration test had no influence on the misalignment test. The test procedures described previously for the 3-inch, 1500-psi aluminum connector were used for the stainless steel connector. The required bending moment of 5640 in-lb was achieved by placing a 141-pound weight 40 inches from the connector on the extension beam.

Test Results. Tests were conducted with initial misalignment angles of 1, 2, and 3 degrees. A sliding action was noted between the seal and its cavity during each test, and greater motion was evident as the misalignment angle was increased. Some plating was scraped from the seal as a result of this motion, but the material came from areas which did not affect the ultimate sealing surfaces, which were on a 10-degree reverse angle. No leakage could be detected with 1500-psi helium on any of the three assemblies resulting from initial misalignments of 1, 2, and 3 degrees.

Seal	Measured Leakage(a), atm cc/sec	Remarks
1	<b>x</b> (b)	
2	x	
3	x	
4	7.2 $\times 10^{-10}$	Leakage within specified limit
5	$1.11 \times 10^{-8}$	Leakage within specified limit
6	×	<b>U</b>
7	×	
8	x	
9	x	
10	x	
11	Greater than 2, 26 x $10^{-6}$	Large flakes in plating on sealing surface
12	×	
13	x	
14	Greater than 2.7 x $10^{-6}$	Pits in plating across sealing surface
15	x	
16	Greater than $4.5 \times 10^{-6}$	One large flake in plating on sealing surface
17	x	
18	x	
19	x	
20	x	
21	x	
22	$1.76 \times 10^{-8}$	Leakage within specified limit
23	x	
24	5.7 x 10-7	Pits in plating across sealing surface
25	x	· · · ·

#### TABLE 53. LEAKAGE DATA FOR REPEATED-ASSEMBLY TEST WITH 3-INCH, 1500-PSI STAINLESS STEEL CONNECTOR

(a) Helium leakage rate per joch of seal circumference.

(b). Denotes no detectable leakage

### Test 8. Tightening Allowance

Test Procedures. The 3-inch, 1500-psi stainless steel thermal-gradient-assembly connector containing two loose rings was used for this test. The connector was assembled by applying the maximum torque of 110 in-1b to each of the sixteen 1/4-inch A286 studs. The same lubrication technique was used as for the aluminum connector, and the same measurement and overtorquing procedure was used. In addition, a leak check was made with 1500-psi helium after the initial assembly and after the application of 50 percent torque.

Test Results. The greatest dimensional change after the application of 150 percent of recommended bolt torque occurred in the tilting of the loose rings. However, this was not considered to be excessive. The incremental changes in stud length were consistent and did not suggest a condition of yielding. Neither the pretest nor the posttest leakage check on the connector with 1500-psi helium produced detectable leakage. Calculations of stud capacity showed that failure of the studs could be expected starting at about 170 percent of recommended torque.

# Qualification Testing of Small, High-Pressure Stainless Steel Connectors

The applicability of the design procedure for designing connectors for pressures higher than 1500 psi was to be demonstrated by Tests 1 through 4 and 8 conducted on a 3-inch, 4000-psi, -423 to 600 F, stainless steel, integral-integral flange connector assembly. The overall performance capability of connectors at 600 F was to be demonstrated by these tests combined with tests on a low-pressure, 3-inch connector at 600 F.

A reexamination of the primary objectives of the laboratory program resulted in the decision to minimize the high-temperature activities. As a result of this decision, the 1200 F tests were eliminated completely, and the low-pressure 600 F tests were eliminated. It was believed that any significant problem at 600 F would be revealed by the test of the 3-inch, 4000-psi connector at 600 F.

When a 4000-psi connector was designed for the full temperature range of -423 to 600 F, the connector was very heavy (see Tables 46 and 47). Since this temperature range was not expected in a single system, 4000-psi connector designs were optimized for a cryogenic system (-423 to 200 F) and a hot-gas system (ambient to 600 F). The dimensions of these connectors showed that by accepting a slightly heavier than optimum cryogenic connector, it was possible to use dimensions for the cryogenic connector that could be obtained by remachining the hot-gas connector. This choice was selected, and the hot-gas connector was fabricated and tested first. The connector was then disassembled, the flanges were modified, and the cryogenic connector was assembled and tested.

Because two 4000-psi connector designs were tested in this revised approach, the test schedule was reexamined. It was decided that Tests 1 through 3 would be conducted on the hot-gas connector, and Tests 1, 2, and 8 would be conducted on the cryogenic connector. When a question of seal disk strength arose later in the program, a pressure impulse test (Test 4) was also conducted with the cryogenic connector.

### Tests for a 3-Inch, 4000-Psi Hot-Gas Connector

Proof-Pressure Test Procedures. The proof-pressure test was conducted with a thermal-gradient assembly similar to that shown in Figure 77. Because of the high temperature, disks were machined on the tubular portions of the connector, and the vacuum chamber was welded to the disks after the connector was assembled. Lubrication was applied to the nut-bearing surfaces and threads of the 3/8-inch, A286 connector bolts, and a preload torque of 296 in-1b was applied to each of the 16 bolts. External heaters were mounted on the outside diameter of each of the integral flanges. Wiring was routed through the vacuum chamber electrical feed-through, and after the connections were checked, the vacuum chamber was welded in place.

Difficulty was encountered with the heating of the connector because the heaters (which had a higher capacity and a higher voltage than the heaters used for the 200 F tests) shorted out in the vacuum required for leakage measurements. A 5500-watt heater had been purchased and modified to fit inside the connector during the

thermal-gradient test. It was decided that this heater would be used for the proofpressure test. Although the heater filled most of the connector cavity, the remainder of the cavity was filled with water to optimize the flow of heat from the heater to the connector. The water did not boil because of the high pressure.

Although an ambient test at  $60^{\circ}0$  psi showed a leakage of only  $1.16 \times 10^{-9}$  atm cc/sec of helium per inch of seal circumference, an increase in connector temperature to 600 F resulted in an excessive leak which flooded the mass spectrometer. An examination of the seal showed an imperfection in the nickel plating at the point of leakage. However, it was estimated that this problem may have been aggravated by load interaction between the sealing disks caused by a very short seal tang. Seals with twice the tang length were then fabricated. Load ring tests showed that the redesigned seals had a radial load of about 2550 pounds per inch of seal circumference as compared with a load of about 1300 pounds per inch for the original seal design. Although the lower value was believed to have been sufficient, it was estimated that the higher value would pose no problems for the high-pressure connector.

<u>Proof-Pressure Test Results.</u> Seven proof-pressure test cycles were conducted to "wring out" the new seal design. To simplify installation, a secondary seal was not used during these tests. Although leakage from the connector was not monitored continuously, a constant 6000-psi helium pressure was maintained inside the connector. Leakage was measured at ambient temperatures (considered the condition most likely to leak) and no leakage was detected after any cycle. This series of tests was considered to be ample for the proof-pressure test requirements.

<u>Thermal-Gradient Test Procedures</u>. The connector was then disassembled and reassembled with an unused primary seal, and with a secondary seal. Six cycles of high-temperature thermal-gradient testing were conducted from ambient to 600 F with heat being supplied by the internal heater. The seal leakage was monitored continuously while the pressure inside the connector was maintained at 4000 psi and the helium pressure between the primary and secondary seals was maintained at 4000 psi. Temperatures were monitored throughout each test cycle with thermocouples on the OD of the seal tang, the OD of a flange, and on the midpoint of a bolt. The maximum temperature differential occurred between the seal tang and the bolt during the heating portion of each cycle.

<u>Thermal-Gradient Test Results</u>. The indicated leakages and maximum temperature differentials are shown in Table 54. The rate of heat input was increased each cycle but no significant increase in seal leakage could be associated with the increasing temperature differentials.

Stress-Reversal-Bending Test Procedures. The stress-reversal-bending test was conducted in a manner similar to that used for the 3-inch aluminum and stainless steel connectors. The base plate of the thermal-gradient assembly was rigidly attached to the test machine, and the required bending moment (12,960 in-lb) was applied to the connector through an extension beam. Temperature-compensated strain gages were attached to the tubular section between the connector and the base plate. The gages were first statically calibrated at room temperature with the assembly in a horizontal position and with weights on the extension beam, and these values were compared with those calculated

for the tubular section using the dimensions and the required moment. This comparison was then used to calculate the amount of strain that should be indicated at 600 F with a reduced modulus of elasticity. Finally, the beam deflection required in the machine to produce the required strain was measured by dial-indicator gages and shown to be in close agreement with the calculated deflection requirement.

Test Cycle	Max Leakage Rate(a), atm cc/sec	Max Temp Differential Between Seal Tang and Bolt, F
1	$4.6 \times 10^{-9}$	132
2	5.4 x $10^{-9}$	198
3	4.1 x $10^{-9}$	195
4	$3.7 \times 10^{-8}$	201
5	4.8 x $10^{-9}$	223
6	6.7 x $10^{-9}$	242

TABLE 54.	THERMAL-GRADIENT-TEST DATA FOR 600 F, 3-1	INCH,
	4000-PSI STAINLESS STEEL CONNECTOR	

(a) Helium leakage rate per inch of seal circumference.

The temperature of the connector was stabilized at 600 F by the Variac-controlled internal heater, the internal pressure of the connector and the helium pressure between the seals were raised to 4000 psi, and the cycling was begun. Pressure, temperature, and helium leakage were monitored continuously throughout the test, while the straingage readings were checked at half-hour intervals. The cycle rate was 378 cycles per minute.

Stress-Reversal-Bending Test Results. The test was terminated after 202,230 cycles of stress-reversal bending, during which time no leakage could be detected from the connector.

Tests for a 3-Inch, 4000-Psi Cryogenic Connector

<u>Proof-Pressure Test Procedures</u>. As described previously, the cryogenic 4000psi connector was very similar to the hot-gas 4000-psi connector. The connector was assembled with a primary and secondary seal, and the 3/8-inch A286 bolts were tightened with a torque of 296 in-lb. Because the heating cycle was less critical for the cryogenic connector than for the hot-gas connector, heating was accomplished by passing steam through a copper tube which was coiled about the welded vacuum chamber and the exposed connector tubular sections.

The proof-pressure test of the 3-inch, 4000-psi cryogenic connector was initiated by pressurizing the connector to 6000 psi at ambient temperature for approximately 1 hour. The pressure was then reduced to 1000 psi and the connector was heated to 200 F. Next, the pressure was raised to 6000 psi for approximately 30 minutes, and the connector was then cooled to ambient temperature with a constant internal pressure of 6000 psi. Proof-Pressure Test Results. No leakage could be detected during two such proof-pressure tests.

<u>Thermal-Gradient Test Procedures</u>. Because of the high pressure of the 4000-psi connector, it was not possible to use the same thermal-gradient-test techniques that were used for the 3-inch, 1500-psi aluminum and stainless steel connectors. The modified procedure consisted of using the liquid-nitrogen test stand to cool the connector as quickly as possible by allowing liquid nitrogen to flow rapidly into the connector at ambient pressure. To hasten the cooling process, the lower part of the connector was immersed in liquid nitrogen during the cooling cycle. The volume between the secondary and primary seals was continuously pressurized to 180 psi, and the mass spectrometer was used to monitor connector leakage continuously. When an equilibrium temperature had been reached, the liquid-nitrogen supply was stopped, and the connector and the secondary-primary seal volume were pressurized to 4000 psi.

<u>Thermal-Gradient Test Results.</u> Four cycles were conducted between temperature extremes of -300 and +195 F. Because the cooling was accomplished in approximately 1 hour, and because the thick flanges of the high-pressure connector represented large masses as compared to the seal, it was believed that the thermal-gradient design parameters were given a severe test. The largest leakage measured during the four cycles was  $3.3 \times 10^{-8}$  atm cc/sec of helium per inch of seal circumference.

<u>Pressure-Impulse Test Procedures</u>. Because a stress-reversal-bending fatigue test had already been conducted on the hot-gas 4000-psi connector, a second fatigue test was not originally scheduled for the cryogenic 4000-psi connector. However, as described later, pressure impulse tests produced seal disk failure in the 8-inch, 1500-psi aluminum connector and extensive seal disk cracking in the 3-inch, 1500-psi aluminum connector. The seal disks in the 3-inch, 4000-psi stainless steel connector had the same thickness as the seal disks in the 3- and 8-inch, 1500-psi aluminum connectors. Although stainless steel was known to be much tougher than the aluminum, it still appeared likely that a pressure-impulse test at 6000 psi might fail the stainless steel disks of the 4000-psi connector. Thus, a pressure-impulse test of the 4000-psi connector was initiated primarily to test the seal disks and secondarily to conduct another fatigue test of the connector structure.

Much of the equipment used to conduct pressure-impulse tests on the 3-inch, 1500-psi connectors was used to test the 4000-psi connector. A modification consisted of substituting an intensifier for the oil/alcohol separation unit so the 2250-psi output pressure of the original equipment could be used to produce 6000-psi pressure impulses. In this arrangement, it was necessary to use a light oil in the connector instead of alcohol, but this was believed to be acceptable since the mode of possible failure was expected to leak oil. The secondary seal was machined away in the assembled connector prior to the test so the pressure impulses could reach the primary seal.

Pressure-Impulse Test Results. For the most part the test was conducted without incident. At one time the intensifier galled and had to be reworked slightly, but by using
a heavy oil at the intensifier seal: it was possible to continue. In all, 1,072,422 pressure-impulse cycles were completed without leakage at the primary seal. Upon disassembly of the connector, a Zyglow examination of the seal showed no cracking of the seal material.

<u>Tightening-Allowance Test Procedures.</u> The 3-inch, 4000-psi cryogenic stainless steel connector was assembled by applying the recommended torque of 296 in-1b to each of the 3/8-inch, A286 bolts. The same lubrication and measurement was used as had been used for the 3-inch, 1500-psi aluminum and stainless steel connectors. The torque was then increased in increments of 10 percent.

<u>Tightening-Allowance Test Results</u>. An overtorque of 90 percent was applied without indication of yielding of the bolts. However, at this torque the nut corners had become badly rounded and application of additional torque was not possible.

# Qualification Testing of Large Aluminum Connectors

The applicability of the computerized procedure for designing large aluminum flanged connectors was to be demonstrated by the fabrication and lest of connectors for 8- and 16-inch tubing systems. As the test program proceeded, the test schedule was reexamined. It was decided that the number of high-pressure, large tubing systems would be low and that the test of a 16-inch, 1500-psi connector would have relatively little applicability. On the other hand, it was also decided that there would be many lowpressure systems, and the absence of an 8-inch, 100-psi connector was undesirable in view of the large size difference between the 3-inch and the 16-inch, 100-psi connectors. Therefore, the decision was made to replace the 16-inch high-pressure connector with an 8-inch low-pressure connector. At the same time, the schedule of tests was revised slightly and the test schedule shown in Table 55 was undertaken.

TABLE 55. REVISED TEST SCHEDULE FOR LARGE ALUMINUM CONNECTORS

Tube OD, in.	Max Operating Pressure, psi	Tests	Assembly Type
8	1500	1,2,5-8	Integral/Loose-Ring
8	100	1, 2, 5, 8	Integral/Loose-Ring
16	100	1, 2, 4, 8	Integral/Loose-Ring

Following the successful completion of over 1,000,000 vibration cycles on the 8-inch, 100-psi connector, the decision was made to substitute a pressure-impulse test for the vibration test of the 8-inch, 1500-psi connector. A seal design problem revealed by this pressure-impulse test demonstrated the wisdom of this decision.

#### Test Assemblies

As described in some detail in previous sections, the tests of the 3-inch connectors were conducted on two types of assemblies: (1) a thermal-gradient assembly or (2) a vibration assembly. Because of the large size of the 8- and 16-inch connectors and because of the more limited test program planned for the large connectors, a different type of test assembly was developed.

Figures 84 and 85 show pictures of the 8-inch, 100- and 1500-psi test assemblies. Figure 86 shows a picture of the 16-inch, 100-psi assembly, while Figure 87 shows a cross section of this assembly. All the large test assemblies had essentially the same configuration. Each assembly contained one loose-ring flange and one integral flange. The internal flange, a short tube section, a tube end closure, and a mounting flange were designed to be machined from one piece of aluminum plate. The mounting flange contained holes for bolt attachment to a rigid mounting pad during the dynamic tests.

The loose-ring stub flange was fabricated from a cylindrical forging approximately 12 inches long. The stub flange design contained a tubular section which extended approximately one tube diameter beyond the tapered hub portion of the flange. Beyond the tube section, the wall thickness was increased to provide material for an O-ring groove and for threaded bolt holes for the attachment of an end closure.

For each assembly, a vacuum chamber was made with a closed end so the entire test assembly could be lowered into the vacuum chamber. An  $\Omega$ -ring seal then was used to seal the opening of the vacuum chamber with the edge of a disk machined on the end closure of the stub flange tube extension.

#### Test l. Proof Pressure

Test Procedures. Proof-pressure tests were conducted with both 8-inch assemblies and with the 16-inch assembly. During assembly, a torque of 60 in-1b was applied to the twenty-four 1/4-inch studs of the 8-inch, 100-psi assembly, and to the seventy-eight 1/4-inch studs of the 16-inch, 100-psi assembly. A torque of 500 in-1b was applied to the twenty-eight 1/2-inch studs of the 8-inch, 1500-psi assembly. A 50/50 mixture of Lubriseal and MoS<sub>2</sub> was used to lubricate the stud threads and the nut collars.

For the 8-inch assemblies, two-piece ring heaters were clamped to the tubular portion of the connectors. The use of electrical heaters would have been difficult for the 16-inch connector because of the number of heaters required for the large area, and because of the large thermal gradients that would have been caused by the rapid loss of heat from the highly conductive aluminum. The procedure used for heating the 16-inch assembly consisted of submerging the entire vacuum-chamber-enclosed-connector assembly in a tank of water kept at 200 F. The assembly was cooled by raising it out of the tank and allowing it to air cool.

In each proof-pressure test, the connector was pressurized with helium initially to 1-1/2 times the maximum operating pressure at ambient temperature for 30 minutes. The pressure was then decreased to about 10 psi and the temperature of the connector was raised to 200 F. The pressure was then raised again to 1-1/2 times the maximum operating pressure and both temperature and pressure were maintained for 30 minutes.



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FIGURE 84. 8-INCH, 15,000-PSI ALUMINUM TEST ASSEMBLY

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FIGURE 85. 8-INCH, 100-PSI ALUMINUM TEST ASSEMBLY

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The pressure was then reduced to approximately 10 psi, the connector was cooled to room temperature, and the pressure was raised again to the proof pressure for 15 minutes. The leakage of the connectors was monitored continuously by a helium mass spectrometer.

<u>Test Results</u>. No helium leakage could be detected during the proof pressure test of either the 8-inch, 100-psi connector, or the 8-inch, 1500-psi connector.

The measurement of an accurate leak rate was not possible for the 16-inch connector because considerable helium entered the vacuum chamber through the 16-inch and 22-inch elastomeric O-rings that were used to seal the tube closure. Approximately 2 hours were required to heat the connector assembly. Although two O-rings were used in series to separate the helium in the connector from the vacuum chamber, the helium permeation rate of O-rings increases rapidly with temperature and time (for the first 4 hours). (For instance, the measured permeation rate of helium through a 15-inch O-ring after 2 hours at 176 F was  $3 \times 10^{-6}$  atm cc/sec.) The second O-ring reduced the flow of helium into the connector considerably, but the second O-ring tended to saturate with helium and act as an accumulator. This provided a flow of helium into the vacuum chamber that was fairly high and out of phase with the heating and cooling cycle.

Consideration was given to welding the closure in place. However, the welded closure would have greatly increased the costs of the subsequent tests. Therefore, the decision was made to determine the proof-pressure leakage characteristics of the connector by two measurements: (1) a room-temperature measurement with 150-psi internal pressure, and (2) an estimate of the amount of helium entering the vacuum chamber in addition to the leakage measured from the O-ring.

No leakage could be detected at room temperature with 150-psi internal pressure. During eight heating and cooling cycles conducted over a 6-day period, no leakage could be detected as resulting from the Bobbin seal. During these heating and cooling cycles, variations in helium pressure and variations in time were used in an attempt to produce a helium reading that could be differentiated from the slowly fluctuating flow of helium from the O-ring. These variations provided a good measure of the response of the O-ring installation so that good confidence was achieved that no measurable helium was being supplied to the vacuum chamber of the Bobbin seal.

# Test 2. Thermal Gradient

Test Procedures. With the 3-inch connectors it was possible to maintain maximum working pressure in the connectors and maximum helium pressure on the primary seal while the connectors were cycled between the temperature extremes. The large size of the 8- and 16-inch connectors prevented this method of operation because the pressurized liquid-nitrogen reservoir could not supply sufficient liquid nitrogen. Therefore, the test procedure was modified for the large connectors.

The procedure for the 8-inch connectors consisted of filling the internal volume of each connector as rapidly as possible with liquid nitrogen at atmospheric pressure. When an equilibrium temperature had been reached, the liquid-nitrogen supply was cut off and the connector was pressurized with helium to the maximum working pressure. Although the gas inside the connector was not pure helium, it was believed that the percentage was sufficiently high that the test results would be acceptable. The pressure was held for 30 minutes and the temperatures were recorded. The temperature of the connector was then raised to 200 F for 30 minutes while the pressure was maintained at the maximum working pressure. The connector was then cooled to room temperature and after 15 minutes the connector pressure was reduced to ambient, and another cooling cycle was started.

The procedure used for the 16-inch connector was similar, but the connector was not heated to 200 F because of the O-ring permeation problem described under Test 1. Instead, the connector was allowed to return to room temperature overnight.

Test Results. Four thermal-gradient cycles were conducted for each 8-inch connector. No helium leakage could be detected during any of the cycles for either connector. Table 56 shows the temperatures measured for the major components of the 8-inch, 1500-psi connector. The tube did not reach a low temperature because the level of liquid nitrogen was raised only enough to cover the connector area. On the 8-inch, 100-psi connector, only the integral flange and tube thermocouples were operating and these read -270 F and -10 F, respectively, for each run. The test of the 8-inch, 100-psi connector was judged to be valid, however, because it was known that the thermal gradients were smaller in the 100-psi connector than in the heavier, 1500-psi connector.

	Temperature, F			
Test	Loose Ring	Integral Flange	Stud	Tube
1	-121	-252	-25	-10
2	-125	-260	-30	-12
3	-112	-252	-13	0
4	-112	-267	-15	-05

TABLE 56. TEMPERATURES OF MAJOR COMPONENTS OF<br/>8-INCH, 1500-PSI ALUMINUM CONNECTOR

Three thermal-gradient cycles were completed on the 16-inch connector. In each cycle the connector was cooled from room temperature to -265 F in approximately 4 hours, and the connector was pressurized with helium for 45 minutes. No helium leakage could be detected during any of the three cycles. This test was judged to be valid because the eight heating cycles conducted under Test 1 were believed to have amply demonstrated the sealing capability of the connector at 200 F.

#### Test 4. Pressure Impulse

Test Procedures. As mentioned previously, the successful vibration test of the 8-inch, 100-psi connector (see page 124) resulted in the decision to subject the 8-inch 1500-psi connector to a pressure-impulse fatigue test instead of a vibration fatigue test. Thus, pressure impulse tests were conducted on the 8-inch, 1500-psi assembly and on the 16-inch, 100-psi assembly.

The test procedures were similar to those used for the 3-inch connectors. Because of the large volume of the connectors it was not possible to use the oil/alcohol separator and oil was pumped directly into the connectors. The equipment shown schematically in Figure 81 was used to create approximately 40 pressure cycles per minute. Figures 88 and 89 show the traces for the 8-inch and the 16-inch connectors. Although it was not possible to obtain pressure dwell in the 16-inch connector, it was believed that the effect of the pressure peaks was satisfactory. During the testing, each connector was placed in a container and a fluid-level switch was arranged to shut off the equipment upon a leakage of approximately 1 pint of oil.



FIGURE 88. PRESSURE CYCLES MEASURED FOR 8-INCH, 1500-PSI CONNECTOR



FIGURE 89. PRESSURE CYCLES MEASURED FOR 16-INCH, 100-PSI CONNECTOR

Test Results. Although the pressure-impulse test of the 8-inch, 1500-psi connector was eventually completed successfully, considerable difficulty was encountered with failure in different parts of the assembly. Each of these is discusse ! briefly.

Following initiation of the pressure-impulse test, the test was automatically terminated after 14,694 cycles when one of the seal disks sheared at the juncture with the tang over a length of about 1/2 inch. This failure came as a surprise because the seal disks for the 3-inch, 1500-psi connector had essentially a same cross section and yet the 3-inch connector did not fail during 200,000 pressure-inpulse cycles. A Zyglow examination of the 8-inch seal showed extensive cracking along much of the base of both seal disks, showing conclusively that the seal was under strength. A Zyglow examination of the 3-inch seal also showed two significant cracks about 3 inches long in the same area, indicating that the 3-inch seal disk was also under strength. On the basis of beam-strength approximations, new seals for the 8-inch, 1500-psi connector were machined with seal disk widths of 0.060 in the sather than the initial 0.040 inch. During the replacement of the Bobbin seal. it was also found that the tongue-ingroove seal housing for the O-ring seal on the end closure had developed fatigue cracks because of the flexing of the flat end closure. The tongue was machined off and the groove was modified to form a simple face seal.

The connector was reassembled with a modified seal and given a successful proofpressure test. The next failure occurred after an additional 16,890 cycles when two of the high-strength socket-head cap screws on the end closure failed and the O-ring extruded through the gap which was created at the face seal. It was apparent that the flexure of the end closure had to be reduced to reduce the bending loads on the endclosure bolts. This was accomplished by placing an O-ring sealed piston on the inside of the connector against the end closure. A raised ring near the OD of the piston was used to prevent high forces from being applied at the center of the end closure.

The next failure occurred after an additional 20,964 cycles (52,548 total cycles) when the base component cracked. The base had been designed to simulate a flanged opening in a component, such as a pump, and the cylindrical wall and the bottom of the base had been made extra strong because the base was expected to be used for other tests. Despite this precaution, however, the high-pressure end load (approximately 120,000 pounds) caused excessive flexure of the flat bottom of the base, and a fatigue crack developed at the juncture of the base bottom and the cylindrical wall. There was no indication that the failure was associated with the part of the base representing the integral flange. A new base was fabricated with increased thickness and radii at the tube-to-base juncture.

The connector was reassembled with the new base component and with a modified seal, and a proof-pressure test was conducted successfully. After 38,704 cycles, 14 end-closure bolts failed in fatigue (59,668 cycles on these bolts). Although minor damage was done to the piston, it was possible to reassemble the end closure with a special grade of high-strength cap screws and the test was continued.

After an additional 40, 400 cycles, the tubular portion of the stub flange developed an axial crack about 6 inches from the stub flange. At this point, a total of 131,652 pressure-impulse cycles had been imposed on the connector. The acceptable conditions stated for this test did not include failure of the tube. However, the stress-reversal bending test does include the concept that the test is successful if the connector is demonstrated to be stronger than the adjacent tubing. Because both of these tests are fatigue tests, it was judged that the pressure-impulse test of the 8-inch, 1500-psi aluminum connector was successful in that the connector was shown to be stronger in fatigue than the adjacent tubing.

For the pressure-impulse test of the 16-inch, 100-psi connector, two precautions were taken to prevent nonconnector failures similar to those described for the 8-inch, 1500-psi assembly. First, the base was clamped at the periphery to a laboratory mounting pad to prevent excessive deflection of the large flat area of the base. Second, two pistons were machined with raised rings near the outside diameters, and the pistons were placed in the connector one behind the other, against the tube end closure to prevent excessive deflection of the flat end closure. The test was stopped after 1,223,500 cycles. No failures had occurred in any of the parts of the connector-simulation assembly, and no evidence of leakage or material yielding could be detected.

The calculated stress in the tubing wall of the 8-inch, 1500-psi connector was 27,500 psi. The calculated stress for the tube wall of the 16-inch, 100-psi connector was 18,000 psi. The difference in the stresses for the two connectors resulted from the need to increase the calculated minimum wall thickness of the 16-inch connector for machining purposes. Since most of the connectors would be designed to the higher stress value, it was obvious that the allowable stress in the tubing would have to be lowered to assure a minimum fatigue life of 200,000 pressure-impulse cycles.

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#### Test 5. Vibration

Because of the premature failure experienced at the tube-to-transition during the vibration testing of the 3-inch aluminum connectors, it was believed that a similar failure would probably be experienced in the 8-inch connectors. In fact, a follow-on program (Contract No. FO4611-69-C-0028) was initiated to include the vibration testing of additional 8-inch connectors. When exceptionally long fatigue life was obtained during the vibration testing of the 8-inch, 100-psi connector, the planned vibration test of the 8-inch 1500-psi connector was replaced with a pressure-impulse test.

<u>Test Procedures</u>. The 8-inch, 100-psi assembly was bolted in an upright position to the laboratory mounting pad used for the Calidyne machine. Strain gages had been mounted on the stub flange. One pair of gages was located at the tube-to-hub transition, while another pair of gages was located 1/2 inch from the transition on the tubular portion of the stub flange. (The strain gages were calibrated before the test with the assembly mounted in a horizontal position. The strain at the transition was approximately twice the strain in the tubular portion.)

Calculations indicated that a pipe extension would have to be fastened to the end closure of the stub flange to reduce the natural frequency of the assembly to 160 cps. A 9-1/2-inch length of 6-inch steel pipe with a 3/16-inch wall was equipped with a flange which was then fastened to the end closure. The extension was then struck a sharp blow and a recording of the strain reaction at the strain gages showed a natural frequency of the assembly of 136 cps. This was judged to be acceptable, and the Calidyne machine was attached to the flange of the pipe extension by a horizontal bolt. The instrumentation used to monitor the test was the same as that used for the 3-inch connectors.

<u>Test Results.</u> It was decided to vibrate the 8-inch connector initially at a low stress level, at room temperature, and unpressurized. A bending moment of 13,400 in-lb was imposed for 603,740 cycles. This produced a measured strain corresponding to a stress of 11,000 psi at the tube-to-hub transition. When no damage at this stress level could be detected, the design bending moment of 18,000 in-lb was imposed for approximately 2.8 x 10<sup>6</sup> cycles. This produced a stress of 14,800 psi at the tube-to-hub transition. Following this test, the connector was pressurized with 100-psi helium for 1 hour and no helium leakage could be detected by the mass spectrometer.

Consideration of the unexpected success of the 8-inch vibration test resulted in the thought that the following test conditions might have contributed to the test results: (1) the 8-inch connector was tested at room temperature, while the 3-inch connector was tested at 200 F, and (2) the 8-inch connector was unpressurized, while the 3-inch

connector was pressurized with water. To determine whether these different test conditions were responsible for the different test results, the 8-inch connector was vibrated again, but the connector was heated to 200 F and pressurized with water to 100 psi.

The second test was conducted at the design bending moment for 3 hours, during which time the connector was subjected to approximately 1 million cycles. When no failure occurred during this test either, despite the previous fatigue cycles, it was concluded that a size effect existed between the 3- and 8-inch connector, and the stress raiser which existed in the 3-inch connector did not exist in the 8-inch connector.

# Test 6. Repeated Assembly

Test Procedures. The repeated-assembly test on the 3-inch connector consisted of 25 assemblies. No appreciable wear or damage could be noted in either flange cavity. Because the radial seal loading was designed to be the same for the 8-inch seals as for the 3-inch seals, it was decided that six assemblies of the 8-inch, 1500-psi connector would be sufficient provided no leakage was encountered, and no damage was caused to the flange cavities. Because of the large size of the seals, and the test time and helium required for each test, this change in procedure was expected to reduce test costs appreciably.

In each test, twenty-eight 1/2-inch studs were torqued to 500 in-lb and the connector was pressurized to 1500-psi with helium for 1 hour, while the leak rate was monitored by the mass spectrometer.

<u>Test Results</u>. No helium leakage could be detected during any of the six repeatedassembly tests. After the tests, no damage could be detected on either flange sealing surface.

# Test 7. Misalignment

Test Procedures. The purpose of this misalignment test was to determine the sealing capability of the connector when it is subjected to a combination of axial and angular misalignment which results in a specific bending moment at the seal. It was decided that the test of the 8-inch, 1500-psi connector would be the most stringent for the large connectors. The bending moment required for this connector was one-half the tube bending moment of 79, 860 in-lb required for the stress-reversal-bending test.

A test fixture was fabricated to allow horizontal mounting of the test assembly to withstand the bending moment. A 15.5-ft-long 6-inch steel pipe with a 3/16-inch wall was then bolted to the stub flange closure. Supports were fabricated to allow angular positioning of the stub flange against the integral flange so a Bobbin seal would just be contained inside the seal cavities of both flanges. The resulting angular misalignment between the faces of both flanges was approximately ! degree. Supports were placed at the stub flange and at the end of the pipe extension. The bending moment at the Bobbin seal supplied by the pipe alone was 20, 200 in-lb. A 102-pound weight was placed 197 inches from the Bobbin seal to produce another 20, 200 in-lb, making a total bending moment of 40, 400 in-lb.

The seal was assembled into the connector and the nuts were snugged down fingertight. The nuts were then tightened with progressively increasing torque in a stud-tostud sequence until each nut had been tightened with a 500 in-lb torque. During the torquing procedure the pipe was raised from its supports due to the flange faces becoming parallel. Thus, for the better part of the torquing procedure, the full moment was applied to the connector, simulating the most stringent misalignment condition. When the full torque was achieved, the flange faces were parallel.

A second seal was assembled at the same angle of misalignment. In this case, however, a 235-lb weight was used at a distance of 11 feet from the seal. The pipe length was modified to give a lower moment, and the total moment applied to the connector was 40,200 in-lb. All other procedural steps were the same as those used during the assembly of the first seal. The second test was conducted to give a higher shear loading at the seal.

<u>Test Results</u>. Each of the misalignment tests was concluded by pressurizing the connector with 1500-psi helium while maintaining the full bending moment on the connector. For each assembly, no leakage rate could be detected by the helium mass spectrometer.

#### Test 8. Tightening Allowance

<u>Test Procedures.</u> The test procedures used for this test were essentially the same as those used for the 3-inch connectors. After the nut faces and stud threads were lubricated with a 50/50 mixture of Lubriseal and MoS2, the connectors were assembled with the recommended torque - 60 in-1b for the 8- and 16-inch, 100-psi connectors (each connector utilized 1/4-inch studs), and 500 in-1b for the 8-inch, 1500-psi connector. The measurements shown in Figure 83 were then made. Each connector was then tightened with the additional torque increments of 10 percent and additional measurements were made until some form of material yielding was detected.

<u>Test Results</u>. As with the 3-inch connectors, the limitation in overtorque was established by failure of the studs. This is believed to be the most desirable form of limitation because the studs can be replaced if excessive torque is applied. The 1/4inch studs of the 16-inch connector began to fail between 30 and 40 percent overtorque, while the 1/4-inch studs of the 8-inch, 100-psi connector began to fail between 60 and 70 percent overtorque. This appeared to be caused by the fact that the loose ring of the 8-inch connector angled more than the loose ring of the 16-inch connector and bound against the studs, delaying stud failure. A somewhat similar action had been encountered in the 3-inch connectors. The 8-inch, 1500-psi connector exhibited bolt failure in the overtorque increment above 20 percent. The degree of overtorque provided by the large aluminum connectors was not as great as had been expected. However, the provision of increased bolt strength would result in a significant increase in connector weight unless a higher strength bolt material were used. It was decided that the specifications would include the allowable torque limits to prevent overtorque of the bolts during connector assembly.

# Extra Test - Burst

A burst test was not scheduled for any connector under Contract AF 04(611)-11204 because the other tests, in particular the pressure-impulse test, were believed to demonstrate the required strength of the connectors. However, with the successful completion of the 16-inch pressure-impulse test, it was decided that a burst test of this connector would be desirable. Aluminum connectors are more likely to fail in burst than stainless steel connectors because the yield strength of aluminum is much closer to its ultimate strength than is the case for stainless steel, and aluminum does not strain harden during yielding as does stainless steel. Further, as mentioned above, the wall thickness of the 16-inch connector was greater in relation to the maximum pressure requirements than is the case for higher pressure aluminum connectors. Thus, a burst test of this connector-simulation assembly was expected to be a stringent test of the connector design.

The 16-inch, 100-psi connector burst with an internal pressure of 490 psi. The rupture occurred in the middle of the machined tubing section (see Figure 86) - the area of highest stress. Despite an internal pressure of about 5 times the maximum operating pressure of the connector, no yielding of the connector parts could be detected, and no leakage of the oil was apparent. This test was believed to be an impressive demonstration of the aluminum-connector-design parameters relating to strength.

# Qualification Testing of Large, Stainless Steel Connectors

The applicability of the computerized procedure for designing large stainless steel flanged connectors was to be demonstrated by the fabrication and test of connectors for 8- and 16-inch tubing systems. At the same time that the schedule of tests was revised for the large aluminum connectors the schedule of tests for the large stainless steel connectors was also revised. However, in addition to substituting an 8-inch, 100-psi connector for a 16-inch, 1500-psi connector, it was also decided that a 16-inch, 100-psi stainless steel connector need not be fabricated and tested. This decision was based on the belief that the 16-inch, 100-psi aluminum connector would demonstrate the suitability of the design procedure for large, low-pressure connectors. The revised test schedule for the large stainless steel connectors is shown in Table 57.

# TABLE 57. REVISED TEST SCHEDULE FOR LARGE STAINLESSSTEEL CONNECTORS

Tube OD, in.	Max Operating Pressure, psi	Tests	Assembly Type
8	1500	1,2,5,7,8	Integral/Loose ring
8	100	1,2,5,8	Integral/Loose ring

It will be noted that neither the original test schedule nor the revised test schedule included a repeated-assembly test. This decision was based on the belief that early, successful tests with the stainless steel seals in the threaded connectors indicated few, if any, problems with stainless steel seals. Thus it was expected that the tests with the 3-inch stainless steel seals would be a sufficient check on the design principles for stainless steel seals for flanged connectors. This was an unfortunate decision. Not only were some problems encountered with the 3-inch seals, as described previously, but the increased use of the stainless steel threaded connectors revealed a sealing reliability problem with 3/4- and 1-inch seals. In addition, extensive problems were encountered with the sealing of the 8-inch, 1500-psi stainless steel flanged connector in the final months of the program. As a result, it was necessary to recommend an overrun effort to investigate the performance of the stainless steel seals in more detail. The uncompleted tests (No. 8 for the 100-psi connector and Nos. 5, 7, and 8 for the 8-inch, 1500-psi connector) would be included in an overrun effort and the test results reported in a supplemental report to this document.

# Test l. Proof Pressure

Test Procedures. The 100- and 1500-psi stainless steel test assemblies had the same configuration as the 8-inch, 1500-psi aluminum test assembly shown in Figure 84. One difference was that the stub-flange tubing extension component consisted of a piece of tubing welded to the steel flange. This design made large stainless steel forgings unnecessary.

The proof-pressure test procedures described previously for the 8-inch aluminum assemblies were used to conduct Test 1 for both 8-inch stainless steel assemblies. A torque of 100 in-1b was applied to the twenty 1/4-in. A286 stude of the 100-psi connector, while a torque of 372 in-1b was applied to the thirty-four 3/8-inch A286 bolts of the 1500-psi connector. The lubrication procedure was the same as described for previous assemblies.

Test Results. No leakage could be detected during the proof-pressure test of the 8-inch, 100-psi connector. The proof-pressure test of the 8-inch, 1500-psi connector proved to be a significant obstacle in the test program. In addition to equipment problems, a design problem was revealed for the Bobbin seal. The primary aspects of these problems are summarized.

Attempts to proof-pressure the 8-inch, 1500-psi connector were unsuccessful with two different seals. An examination of the coined nickel surface of the seals at first seemed to indicate insufficient radial sealing force. However, a hardness check of the nickel plating showed that the nickel plating had not been annealed. This had happened twice during the development of the stainless steel seals for threaded connectors, and its reoccurrence emphasized the need for careful control of the fabrication steps of the stainless steel Bobbin seals. Unfortunately, annealed nickel plating looks the same as unannealed plating.

The proof-pressure test with the first annealed seal showed a minor increase in leakage as the connector was heated to 200 F. When the connector was cooled to ambient, the helium leakage increased to about  $3 \times 10^{-7}$  atm cc/sec per inch of seal circumference.

Because this leakage was beyond the allowable amount, the test was terminated. An examination of the nickel on the seal showed a smaller than normal coined surface, indicating too low a radial sealing force. In addition, a closer inspection of the general condition of the plating indicated a poorer quality of plating than that obtained on the 3-inch stainless steel seals.

To obtain a better idea of the effect of the reduced quality of the nickel plating, three annealed seals that were slightly oversized were given a light machining cut to produce a smooth nickel surface that fitted snugly in the connector flanges. Three proofpressure cycles were conducted with one of these seals. During the first cycle, no leakage could be detected. During the second cycle, the helium leakage at 200 F was approximately  $5 \times 10^{-8}$  atm cc/sec per inch of seal circumference, and at ambient temperature the leakage increased to about  $2 \times 10^{-7}$  atm cc/sec. The test was terminated when the leakage at 200 F during the thermal cycle was  $1.5 \times 10^{-7}$  atm cc/sec of helium per inch of seal circumference. Although the leakage values obtained were still within the allowable amount, the increasing nature of the leakage, despite the special machining, indicated that the seal was not satisfactory.

Because sealing problems resulting from insufficient sealing load had developed with the stainless steel threaded connectors, and because a relatively small surface was coined on the 8-inch seal, it was decided that an increased radial sealing force was required. This was obtained for the 8-inch, 1500-psi connector by doubling the length of the seal tang. Calculations indicated that this would increase the radial sealing face from about 750 lb/in. of seal circumference to about 1300 lb/in. While new seals were being fabricated, a reinforcing ring was made to fit inside the tang of an original seal to increase the radial sealing load of that configuration to about 1300 lb/in. This technique made possible an interim test of the higher sealing load. Tests with this seal were somewhat uncertain because of problems that developed with the O-ring seal on the vacuum chamber. H vever, after several proof-pressure cycles similar to those conducted for the 16-inch aluminum connector, no significant leakage could be detected from the Bobbin seal.

#### Test 2. Thermal Gradient

Test Procedures. Two series of thermal-gradient tests were conducted with the 8-inch, 100-psi connector. The first series was conducted with the same procedures described previously for the 8-inch aluminum connectors. The second series was conducted in a similar manner except that the connector was not placed in a vacuum chamber. Instead, a small probe was used with the helium mass spectrometer to "sniff" for helium leakage along the circumference of each disk of the Bobbin seal. While this procedure cannot be used to measure the rate of a helium leak, it is a good "go-no-go" indication of the presence of a helium leak. The latter procedure was used also for all of the tests with the 8-inch, 1500-psi stainless steel connector.

<u>Test Results.</u> During the first series of tests, three thermal-gradient cycles were conducted with the 8-inch, 100-psi connector. The maximum measured helium leakage during these cycles was  $1.2 \times 10^{-9}$  atm cc/sec per inch of seal circumference. When the thermal-gradient tests with the 8-inch, 1500-psi connector encountered considerable difficulty, the 8-inch, 100-psi connector was reasonabled with another seal,

and six thermal-gradient cycles were conducted in a second series of tests. No evidence of leakage could be detected by the helium probe during these tests. A thermocouple attached to the integral flange of the connector showed that the stabilized cold temperature reached during both series of tests was between -315 and -300 F. A STATE

Because of the problems encountered with the vacuum chamber during the successful proof-pressure test with the reinforced seal for the 8-inch, 1500-psi connector, it was decided that the first thermal-gradient test would be conducted without the vacuum chamber. The helium probe showed that this seal leaked excessively on the first cold cycle. On the assumption that this leakage was caused by improper action of the reinforced seal, the connector was disassembled and reassembled with one of the modified seals that had then been fabricated.

Although no leakage could be detected during the hot portion of the thermalgradient cycle, some leakage was evident during the first cold cycle. This leakage increased noticeably during the second cold cycle, and during the third cold cycle it was obvious that the leakage was excessive. When the connector was disassembled, an examination of the Bobbin seal and the lower connector flange showed that several small pieces of copper had gotten into the flange cavity and prevented proper seating of the seal. The connector was then reassembled with a second modified seal.

No leakage could be detected during the first cold cycle with the second modified seal. However, definite leakage could be detected during the second cycle, and two locations of excessive leakage were identified during the third cold cycle. When the connector was disassembled and the seal was examined, two flaws could be seen at the sources of leakage. One appeared to be caused by material left in the flange cavity, while the other appeared to be a crack in the nickel plating.

At this point a careful inspection was made with a 10-power glass of the sealing surfaces of all of the used and unused seals. It was apparent that several of the unused, modified seals had flaws which made sealing questionable. It was also apparent that the 1300 lb/in. sealing force of the modified seals was still not enough to create a consistently shiny coined sealing surface. The lack of consistency seemed to be associated with the effect of the reverse taper of the seal on the sealing action as well as with variations in the nickel plating. With the decision that the sealing force was not sufficient to overcome the fabrication variations in the large seal, a reinforcing ring was made to fit inside the seal tang and to increase the radial sealing load by approximately 100 percent. A load ring was made for the seal, and an assembly with a reinforced, modified seal showed a radial load of 2800 lb/in. of seal circumference.

The modified seal with the best nickel plating was then equipped with a reinforcing ring and the connector was assembled with the expectation that a satisfactory thermalgradient test would be achieved. Although the first cycle was conducted with no indication of leakage, leakage was again indicated during the second cycle, and by the fourth cycle excessive leakage was indicated over almost 180 degrees of the Bobbin seal.

With the failure of this test, it was apparent that the leakage was being caused by a factor other than an insufficient radial sealing load. During one of the earlier tests, the temperature of the seal tang and the integral fiange was measured to determine whether differential radial expansion was causing the leakage. Calculations based on these measurements indicated that there was sufficient elasticity in the seal to maintain an adequate sealing load for the thermal gradients. However, a comparison of the 8-inch seal with the 3-inch seal and the 1-inch threaded seal showed that the 8-inch seal had less than half the compressive strain of the 3-inch seal, and less than one-fourth the compressive strain of the 1-inch seal. Even though some of the compressive strain in all the seals is in the plastic region, the austenitic stainless steel used in the seals is a strain-hardening material so that an increase in strain indicates some increase in elastic-recovery capability. Because the effective thermal-gradients in the connector structure can only be approximated, it was decided that one of the reinforced modified seals would be machined to produce longer sealing disks and provide more compressive strain in the seal. It was possible to make the disks twice as long in this approach, which made the compressive strain similar to that in the 3-inch seal. all the second se

A load test was first made with one of the modified seals machined with longer disks. Although the same size reinforcing ring was used as had been used on the previous modified seals, the material removed from the tang to produce the longer disks resulted in a measured radial load of 2200 lb/in. of seal circumference. This was judged to be adequate. A modified scal with good nickel plating was then machined with longer disks, a reinforcing ring was added, and the connector was reassembled with this seal. During seven thermal-gradient cycles, no leakage could be detected from the Bobbin seal; this indicated fairly conclusively that differential thermal expansion was the basic cause of leakage in the thermal-gradient test rather than insufficient radial load.

As described subsequently in this report, it also became apparent during this period of time that a radial sealing load of 1500 lb/in. of seal circumference was desirable for the 3/4- and 1-inch threaded-connector stainless steel seals. Because of the increased variations that are normally encountered in fabricating larger parts, and because of significantly increased chances of accidental damage to the sealing surfaces, an increase in radial sealing load for the 8-inch seals was desirable, in any case, even though insufficient radial load was probably not the cause of leakage in the experimental assembly. However, the increased mass required to achieve high radial sealing loads in the larger seals represented a design problem requiring further consideration.

# Test 5. Vibration

<u>Test Procedures</u>. The procedures used to test the 8-inch, 100-psi aluminum connector in its final series were also used to test the 8-inch, 100-psi stainless steel connector. The design bending moment produced a stress at the tube-to-hub transition of 16,800 psi.

Test Results. Failure occurred in 372,000 cycles at the welded juncture of the stub flange and the extension piece of tubing. The welded joint was located approximately 4 inches from the stub flange and since it was in a lower stress zone than the connector, failure of the carefully made weld was not expected. Some thought was given to remaking the weld and continuing the test. However, the vibration tests of the 3-inch connectors showed that the stainless steel parts were much less susceptible to failure in vibration than the aluminum parts, and since the 8-inch, 100-psi aluminum connector had successfully withstood  $1.0 \times 10^6$  vibration cycles, there seemed to be no need to obtain additional vibration cycles for the stainless steel connector.

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## Test 8. Tightening Allowance

Test Procedures. Because of the possible conduct of additional tests with the 8-inch, 100-psi and 8-inch, 1500-psi stainless steel connectors in an overrun effort, it was not possible to conduct a complete tightening-allowance test. The overtorque increments at each connector were stopped at 30 percent to prevent failure of the expensive A286 studs. However, the procedures up to this point were identical to those described for previous connectors. An initial torque of 100 in-1b was applied to the twenty 1/4-inch A286 studs of the 100-psi connector, while an initial torque of 372 in-1b was applied to the twenty-four 3/8-inch A286 bolts of the 1500-psi connector.

Test Results. No indication of yielding was obtained within the 30 percent overtorque limitation. As with previous connectors, failure was expected to occur first in the studs and bolts. Tests with the 3-inch stainless steel connector indicated that the 1/4-inch studs would fail with about 70 percent overtorque. Tests with individual bolts showed that the 3/8-inch bolts would fail with about 40 percent overtorque.

# Test Summary

#### Aluminum Flanged Connectors

Test 1. Proof Pressure. The primary purpose of this test was to check the initial sealing capability and the strength of the connectors. Proof-pressure tests were conducted with the following connectors: 3-inch, 100-psi; 3-inch, 1500-psi; 8-inch, 100-psi; 8-inch, 1500-psi; and 16-inch, 100-psi. In the tests with the 3-inch connectors, the maximum helium leakage rate measured was  $3.1 \times 10^{-9}$  atm cc/sec per inch of seal circumference. No helium leakage could be measured during the tests with the large connectors. No yielding of any connector structure could be detected.

Test 2. Thermal Gradient. This test was conducted to determine the effect of maximum thermal gradients on connector sealing. Several thermal-gradient cycles were conducted with each of the assemblies used for the proof-pressure tests (see above). The maximum helium leakage rates measured for the 100- and 1500-psi, 3-inch connectors were, respectively,  $1.8 \times 10^{-9}$  and  $1.2 \times 10^{-8}$  atm cc/sec per inch of seal circumference. No helium leakage could be measured for any of the large connectors.

Test 3. Stress-Reversal Bending. This test was one of three fatigue tests used during the course of the program. Stress-reversal-bending tests were conducted with the 100- and 1500-psi, 3-inch connectors. Although both assemblies lasted the required 200,000 cycles, failures occurred in the tubing portions of both assemblies before 250,000 cycles had been completed. The maximum helium leakage measured during these tests was  $1.5 \times 10^{-9}$  atm cc/sec per inch of seal circumference. Test 4. Pressure Impulse. Pressure-impulse tests, another type of fatigue test, were conducted with a 3-inch, 1500-psi assembly, an 8-inch, 1500-psi assembly, and a 16-inch, 100-psi assembly. The 3-inch assembly used for this test was not the same assembly as was used for the stress-reversal bending test. Although the 3-inch assembly withstood the required 200,000 cycles satisfactorily, substantial cracking was subsequently found at the base of the seal disks. The 8-inch seal failed at the base of a seal disk after approximately 14,700 cycles. Modified seals were machined with the width of the seal disks increased 50 percent. This type of seal showed no cracking after approximately 80,000 cycles. After several miscellaneous equipment failures, the tubular portion of the 8-inch stub flange component failed after approximately 130,000 cycles. The connector was judged to be satisfactory. No failures occurred in the 16-inch assembly after  $1.2 \times 10^6$  cycles.

Test 5. Vibration. Vibration tests, the third type of fatigue test, were conducted with three 3-inch, 1500-psi assemblies, and an 8-inch, 100-psi assembly. The first 3-inch assembly failed at the base of the integral flange after only about 50,000 cycles with approximately one-half the required bending moment. The second 3-inch assembly, with strengthened connector hubs, failed at a bulkhead in the tubing portion of the assembly after about 28,000 cycles under the required test conditions. The third assembly, with modified tubing sections, failed after 53,000 cycles at the tube-to-transition section under the required test conditions. In the second and third assemblies, failures occurred in a far shorter time than was indicated by the strain measured by carefully calibrated strain gages placed on the connectors. In contrast, the 8-inch connector withstood 1.0 x  $10^6$  vibration cycles under the required test conditions without failure.

Test 6. Repeated Assembly. Because of the high sealing forces needed to seal helium, repeated-assembly tests were conducted to determine whether repeated connector assembly caused damage to the connector flanges. Twenty-five reassemblies were conducted with a 3-inch, 1500-psi connector, and six reassemblies were conducted with an 8-inch, 1500-psi assembly. The maximum helium leakage measured for the 3-inch tests was  $0.97 \times 10^9$  atm cc/sec per inch of seal circumference, while no leakage could be measured during the 8-inch tests. No damage could be detected on any connector flange after the tests.

Test 7. Misalignment. A misalignment test was established to demonstrate that a connector could be assembled satisfactorily under a certain amount of misalignment. The test was conducted with a 3-inch, 1500-psi assembly, and an 8-inch, 1500-psi assembly. The 3-inch connector was assembled satisfactorily at angles of 1, 2, and 3 degrees and no helium leakage could be measured after these assemblies. The 8-inch connector was assembled satisfactorily twice with a 1-degree angle of misalignment, and no helium leakage could be measured after either assembly. It was estimated that these angles represented realistic conditions for the two connector sizes.

Test 8. Tightening Allowance. The final test of each connector assembly consisted of tightening the studs in increments of 10 percent over the maximum torque until yielding was detected. For the 100-psi and 1500-psi, 3-inch connectors, an overtorque of 40 and 50 percent, respectively, was accomplished satisfactorily. The studs then began to yield in the next increment of overtorque. For the 100-psi, 8- and 16-inch connectors and for the 8-inch, 1500-psi connector, the overtorque capability was 60, 30, and 20 percent, respectively. Differences in overtorque capability result from the step function by which the computerized design program selects the connector bolt sizes.

Extra Test - Burst. No burst test was scheduled for the contract. However, the unfailed 16-inch, 100-psi connector provided a good opportunity to evaluate the burst safety margin provided by the computerized design procedure. The tubing portion of the assembly burst at a pressure of 490 psi. No deformation could be detected in the connector after the test.

## Stainless Steel Flanged Connectors

The purpose of each of the eight standard tests is summarized briefly in the above summary of the aluminum connector tests.

Test 1. Proof Pressure. Proof-pressure tests were conducted with a 3-inch, 1500-psi assembly, two 3-inch, 4000-psi assemblies, an 8-inch, 100-psi assembly, and an 8-inch, 1500-psi assembly. The maximum helium leakage measured for the 3-inch, 1500-psi assembly was  $1.5 \times 10^{-8}$  atm cc/sec per inch of seal circumference. The first test with a 3-inch, 4000-psi, 600 F assembly resulted in excessive leakage at 600 F. A new seal was machined with an increase in the radial sealing load and no leakage could be detected during seven proof-pressure cycles. No helium leakage could be detected for the 3-inch, 4000-psi cryogenic assembly or for the 8-inch, 100-psi assembly. However, the proof-pressure test for two assemblies of the 8-inch, 1500-psi connector resulted in an excessive leakage. When a seal was modified to provide an increased sealing load, no leakage could be measured during the proof-pressure test. No yielding could be detected on any of the connectors as a result of the proof-pressure tests.

Test 2. Thermal Gradient. This test was conducted with each of the assemblies used for the proof-pressure test. The maximum helium leakage measured during six thermal-gradient cycles of the 3-inch, 1500-psi connector was  $3.7 \times 10^{-9}$  atm cc/sec per inch of seal circumference. The maximum leakage measured during six cycles of the 3-inch, 4000-psi, 600 F assembly was  $6.7 \times 10^{-9}$  atm cc/sec per inch of seal circumference, while the maximum leakage measured for the 3-inch, 4000-psi cryogenic connector was  $3.3 \times 10^{-8}$  atm cc/sec. A total of 10 thermal-gradient cycles were conducted on the 8-inch, 100-psi assembly with no detectable leakage. Considerable difficulty was encountered in the testing of the 8-inch, 1500-psi assembly. It was not until the length of the seal disks and the radial sealing load were increased that seven thermal-gradient cycles were conducted without an indication of helium leakage.

Test 3. Stress-Reversal Bending. Stress-reversal-bending tests were conducted with a 3-inch, 1500-psi assembly and a 3-inch, 4000-psi, 600 F assembly. The 3-inch, 1500-psi assembly withstood approximately 350,000 cycles without failure, and the 3-inch, 4000-psi assembly withstood approximately 200,000 cycles without failure. The maximum helium leakage measured for the 3-inch, 1500-psi assembly was  $1.4 \times 10^{-9}$  atm cc/sec per inch of seal circumference. No leakage could be measured during the test of the 3-inch, 4000-psi assembly.

Test 4. Pressure Impulse. Pressure-impulse tests were conducted with a 3-inch, 1500-psi assembly (different from the one used for the stress-reversal bending test) and a 3-inch, 4000-psi cryogenic assembly. The 1500-psi assembly withstood 200,000 cycles without failure and with no detectable leakage. The 4000-psi assembly withstood approximately  $1.0 \ge 10^6$  cycles without failure and without detectable leakage.

Test 5. Vibration. Vibration tests were conducted with a 3-inch, 1500-psi assembly, and an 8-inch, 100-psi assembly. The 3-inch assembly (which had been used for the pressure-impulse test) withstood 475,000 cycles without failure and without leakage. The 8-inch assembly failed after 372,000 cycles at the weld joint between the stub flange hub and the tube extension. No leakage could be detected during the test and the connector was judged to be satisfactory.

Test 6. Repeated Assembly. Twenty-five reassembly tests were conducted with a 3-inch, 1500-psi assembly. Four assemblies showed excessive leakage. At the time it was judged that the excessive leakage was caused by poor plating. However, on the basis of tests with other stainless steel seals, it subsequently appeared that these seals indicated insufficient radial sealing load.

Test 7. Misalignment. A misalignment test was conducted with a 3-inch, 1500-psi assembly. Assemblies were made at initial misalignment angles of 1, 2, and 3 degrees. No leakage could be detected after any assembly.

Test 8. Tightening Allowance. Tightening-allowance tests were conducted with the same assemblies as were used for the proof-pressure tests. The 3-inch, 1500-psi connector withstood 50 percent overtorque without yielding, while the 3-inch, 4000-psi assembly withstood 90 percent overtorque without yielding. The 8-inch, 100-psi assembly and the 8-inch, 1500-psi assembly were subjected to only a 30 percent overtorque because the assemblies and the fasterners were needed for work under another contract. Tests with individual fasterners showed that the 100-psi assembly would have withstood about 70 percent overtorque, while the 1500-psi assembly would have begun yielding with 40 percent overtorque.

# Discussion of Critical Design Parameters

For the most part, the qualification tests verified the applicability of the computerized program for designing lightweight flanged connectors for aluminum and stainless steel tubing systems. As would be expected, the many design parameters and test conditions combined to produce several critical areas. Both the satisfactory and unsatisfactory critical areas are discussed briefly in relation to the major components of the aluminum and stainless steel connectors.

#### Aluminum Flanged Connectors

Bobbin Seal. The radial sealing load (750 lb/in. of seal circumference) selected for the aluminum Bobbin seal produced excellent initial sealing. The residual axial load (50 percent of the axial seating load) maintained excellent sealing during the performance tests. The use of overaged 6061-T6 aluminum as the seal material was entirely satisfactory.

Two problems were revealed for the Bobbin seal. First, the seal disks were too thin to pass the pressure-impulse test for 1500-psi systems. Second, calculations showed that the length of the seal disks was marginal for some connector sizes (particularly 8 and 16 inch) for providing sufficient elastic recovery for cryogenic thermalgradients. (This condition was found to exist despite the successful thermal-gradient tests of the 8-inch connectors.)

For the specifications shown in Appendix C, the design of the seal disks was revised to overcome these difficulties. There is little doubt that the thermal-gradient problem has been corrected by a lengthening of the disks of selected seal sizes. However, the calculations for the seal disk thickness are approximations, and tests are needed to verify the pressure-impulse capability of the new length/thickness dimensions of the disks for 1500-psi seals for 8- and 16-inch systems. The revised seal design did not significantly change the low-pressure seals and only marginally changed the highpressure 3-inch seals. Consequently these successful designs need not be retested.

Integral and Loose-Ring Flanges. The design of the flanges incorporates many stress calculations and assumptions. Although the performance of both types of flanges was excellent in most regards, one minor and two major problems were revealed in the test program. The minor problem was caused by the stress calculation for the thickness of the seal cavity lip for the stub flange providing insufficient material for machining tolerances for the large connectors. This has been corrected in the specifications.

A major problem resulted when the initial design program did not provide sufficient strength at the hub-to-flange joints. This problem was corrected early in the laboratory program and the revised design was shown to be satisfactory in the subsequent tests. The revised design has been included in the specifications. In the second major problem, the tubing wall selected for the high-pressure connectors was not sufficiently strong to provide a minimum of 200,000 pressure-impulse cycles. The calculation for the tubing wall thickness was revised, on the basis of considerable design data, and the specifications incorporate the revised calculation. Because of the importance of failure of the tubing, the adequacy of the calculation should be checked by additional pressureimpulse tests.

In addition to the stress-calculation revisions, consideration of the flange configurations during the laboratory program led to the decision to design the flanges with a slight tube extension so the tubing weld would not occur at the change in section provided by the tapered flange hub. This configuration has also been incorporated in the specifications. Loose Ring. The loose ring represents a difficult design problem. While the loose ring tilted more for some test connectors than for others, the studs limited the degree of tilt to an acceptable value. Because of the relatively close fit of the studs in the holes of the loose ring, a considerable increase in the weight of the loose ring would be required to prevent any binding against the studs. The tilting of the loose ring did not adversely affect the assembly of any connectors within the required torque values, and the design basis for the loose ring was believed to be satisfactory.

Bolts, Studs, and Nuts. The performance of the aluminum studs and nuts was excellent in all cases. The adequacy of the design basis was particularly well demonstrated with the 8-inch, 1500-psi as sembly in which high-strength steel socket-head screws and cap screws kept failing during the pressure-impulse test of the connector. There is little doubt that the use of aluminum studs and nuts poses problems. The threads are easily damaged in handling and the bearing surfaces have to be lubricated carefully to prevent galling. Further, the proximity of the yield strength of aluminum to its tensile strength seriously limits the overtorque capability of aluminum fasteners, requiring extra care to achieve the proper torque. However, calculations show that aluminum fasteners must be used if a lightweight aluminum connector is to maintain good sealing for cryogenic thermal gradients. By selecting 2024-T4 material, and by using standard machining practices, it is believed that practical aluminum fasteners have been designed.

<u>General Comments.</u> In reviewing the tests of aluminum connectors, there was evidence that each increase in connector size introduced unexpected problems. The 3-inch connector introduced vibration problems that had not been encountered with aluminum threaded connectors, and the 8-inch, 1500-psi connector introduced fatigue problems that had not been encountered with the 3-inch, 1500-psi connector. These problems seemed to result primarily from the unanticipated effects of high pressure on large surface areas, and on the tendency of aluminum to crack with overstressing rather than to yield and redistribute the stresses as stainless steel does. Because of these problems, it was concluded that the satisfactory performance of large (over 8 inch), 1000- and 1500-psi connectors remains in some question until they are built and tested. The dimensions for such connectors were included in the specifications, and it was believed that the chances are good that they would function satisfactorily. However, the connectors were not viewed with the same confidence as those that were within the limits of the connectors that were built and tested.

#### Stainless Steel Flanged Connectors

At the start of the program, the stainless steel connectors were expected to produce fewer problems than the aluminum connectors. This resulted primarily from the greater toughness of stainless steel than aluminum (as discussed above). Because of this expectation, the aluminum connectors in each size were scheduled to be tested before the stainless steel connectors. Unfortunately, in the latter stages of the laboratory program, major problems developed with the stainless steel seal for the 8-inch, 1500-psi connector. Although considerable progress was made toward the solution of these problems, it was not possible to demonstrate the adequacy of the solution for the large connector sizes. At the end of the program, a letter was forwarded, recommending the investigation of adequate solutions to the large-diameter stainless steel seal problem and the conduct of the necessary tests to demonstrate these solutions.

Bobbin Seal. The dimensions selected for the stainless steel Bobbin seals were satisfactory in several respects. The disk thickness was satisfactory for the radial loading and for the pressure-impulse tests. (For instance, the 0.040-inch thick disk for the 3-inch, 4000-psi connector withstood 1,000,000 pressure-impulse cycles without any sign of fatigue damage.) Also, the residual axial seating loads were satisfactory for the various performance conditions.

The major seal problems that developed related to the radial sealing load or to the inability of the seal to accommodate the radial thermal gradients in the large, highpressure connectors. The radial-sealing-load problem was found to exist in the 3/4- and 1-inch threaded connectors as well as in the flanged connectors. On the basis of work in both types of connectors, the design radial sealing load was increased from 750 lb/in. of seal circumference for all seals to 1200 lb/in. for 100, 200, and 500-psi connectors, and to 2500 lb/in. for pressures of 1000 psi and higher. These revised values were included in the specifications.

In conjunction with the radial sealing load problem, it was also found that the 8-inch seal did not provide sufficient elasticity for the thermal gradients caused by the heavy flanges for the 1500-psi connectors. Although a 16-inch, 1500-psi connector was not tested, the problem was shown by calculation to be even worse for this connector, and other large, high-pressure connectors were believed to have the same limitation. It was shown by the successful test of the 8-inch, 1500-psi connector that this problem could be corrected by increasing the length of the seal disks. Increased seal disk lengths were then selected for all large connectors and the thickness of the disks was increased to sustain the higher radial loading (2500 lb/in.). These revised dimensions were included in the specifications.

On basis of the qualification tests and on selected calculations, it was believed that the specification dimensions could be viewed with confidence for all connectors through 3 inches, and for 100-, 200-, and 500-psi connectors through 8 inches. However, for the 8-inch, 1500-psi connector, and for all connectors larger than 8 inches, selective tests were desirable to substantiate the revised seal design parameters. It was possible that the disk length in some of the larger, high-pressure connectors would have to be increased still further. If this became necessary, it was believed that a modified seal design should be used in which the disks were machined farther down into the seal tang, leaving some material between the disk sides.

Integral and Loose-Ring Flanges. The design basis for these connector components was found to be satisfactory in almost every respect. On the basis of the problems revealed by the aluminum flanges, the hub-to-flange joint was strengthened, the stubflange seal cavity lip was increased for the larger sizes, and a tube extension was added to the tapered hub to remove the weld joint from the change in section at the tapered hub. Loose Ring. The tilting of the loose ring that was encountered in the aluminum loose rings was even more pronounced in some of the stainless steel connectors. However, as with the aluminum connectors, the tilting did not seem to interfere with the normal assembly of the connectors, and the decision was made to leave the design unchanged.

Bolts, Studs, and Nuts. The performance of the A286 studs and nuts was excellent. No problems were encountered in the assembly of the fasteners or in the performance of the connectors, and ample overtorquing capability was demonstrated. The high cost of A286 fasteners and the long lead time often required to get the parts will undoubtedly be a problem for some time. However, the A286 fasteners have so many technical advantages that it is believed that no other fastener will offer serious competition within the foreseeable future.

# FLANGED-CONNECTOR MILITARY STANDARDS AND SPECIFICATIONS

The flanged connector military standards and specifications developed under Contract AF 04(611)-11204 are discussed in three sections: (1) background, (2) general standards and specifications characteristics, and (3) recommendations for flangedconnector military standards and specifications.

# Background

Under Contract No. AF 04(611)-9578, military standards and specifications were prepared in draft form for AFRPL threaded connectors for 4000-psi stainless steel tubing systems and 1000- and 750-psi aluminum tubing systems. Specifically, the items prepared consisted of: (1) military standards containing assembly instructions and part drawings for typical connector configurations for six tubing sizes: 1/8, 1/4, 3/8, 1/2, 3/4, and 1 inch, (2) a military specification for the plating of soft nickel on stainless steel Bobbin seals, and (3) a military specification for the qualification of AFRPL fittings by prospective manufacturers. Although the preparation of these items required considerable effort, ample guidance was obtained from previous military documents concerning the appropriate format and necessary information.

At the beginning of Contract AF 04(611)-11204, it was mutually agreed that there was little precedence for determining the best form for military standards and specifications for flanged connectors. Flanged connectors for the aerospace industry are usually designed specifically for each system to insure minimum weight. On this basis, it appeared desirable to develop a computerized design procedure that could be used by aerospace designers to determine the dimensions of AFRPL flanged connectors for each aerospace system application.

While this general concept was attractive, further mutual consideration led to the conclusion that it was not practical as an Air Force procurement procedure. As the computer program was developed it became clear that a certain amount of engineering judgment would be required for the selection of input data and for the selection of optimum connector dimensions. If flanged connectors were obtained by the Air Force through use of the design procedure, no control could be exercised over these judgment points. Further, mistakes in the use of the computer program might not be revealed until late in the procurement cycle.

Attention was then directed toward the preparation of a large "steam table" of dimensions that would be sufficiently comprehensive that all foreseeable system requirements could be met closely by selection of the proper connector parts, either by the Air Force or by the contractor. While this approach had considerable merit, the large number of connector sizes and types made it very difficult to specify tolerances for dimensions and assembly torques. Also, there seemed to be a distinct possibility that the large number of connectors might include some combination of parts that would not perform properly under certain application requirements.

Late in Contract AF 04(611)-11204 it was decided that the best procurement procedure would be a combination of the two basic approaches. Military standards and specifications were to be prepared for the sizes and pressures mutually believed to be the most commonly used in the foreseeable future. By including the computer program for the design procedure in this report, it would also be possible for connector dimensions to be developed for special applications. Arrangements could be made between the Air Force and each contractor concerning the necessary steps for qualifying the design of each special connector.

# General Characteristics of Flanged-Connector Military Standards and Specifications

It is anticipated that the military standards and specifications eventually prepared by the Air Force for flanged connectors will consist of three documents: (1) an assembly of military standards, (2) a military specification for soft nickel plating, and (3) a military specification for the requirements of fluid connectors. Each of these is discussed briefly.

# Military Standards

Military Standards for AFRPL threaded connectors were issued as part of Procurement Specifications MIL-F-27417 (USAF). Five sheets (MS 27850) were prepared describing the proper assembly of the fitting. One sheet (MS 27851) showed the Military Standard numbers for each stainless steel connector assembly, while MS 27856 showed the numbers for each aluminum connector assembly. One sheet was then prepared for each connector part (MS 27852 through MS 27855, and MS 27857 through MS 27868), showing the necessary manufacturing information for producing each part for each of six tubing sizes. (Figure 90 shows MS 27855, which was prepared for stainless steel seals).

Similar military standards are expected to be issued for AFRPL flanged connectors. One set of assembly instructions will probably be satisfactory for all the flanged connectors. However, one drawing of an assembled connector will probably be required for each material, system pressure, and service condition so the necessary part numbers can be given. For each assembly drawing, one military standard sheet will probably be required for each component, i.e., integral flange, loose-ring flange, loose ring, seal, bolt, stud, and nut. If military standards are prepared on this basis for the system conditions included in Appendix C, approximately 70 military standards will be required. Because of the large number of tubing sizes and connector dimensions to be included on each military standard, it may be desirable to issue the flanged-connector military standards in groups, with the most common system conditions being given first priority.

# Nickel-Plating Specifications

A military specification, MIL-P-27418 (USAF), has been issued for plating soft nickel on stainless steel Bobbin seals. Except for minor changes that might be needed to update the document, the specification is believed to be satisfactory for the Bobbin seals of the flanged connectors.





#### Flanged-Connector Requirements Specification

A military specification, MIL-F-27417 (USAF), has been issued to establish the requirements of AFRPL threaded fittings for use in rocket-engine fluid systems. By defining different manufacturing and performance criteria to be met by AFRPL connectors, a means was expected to be provided by which sample connectors from any manufacturer could be evaluated for quality. By means of this procedure, it was expected that interested manufacturers would be placed on a qualified products list.

It is expected that a similar military specification will be issued for AFRPL flanged connectors. Because of the larger sizes and much greater variety of flangedconnector types, it is believed that the performance evaluation of connectors should be minimized and that greater detail should be included concerning inspection and handling of the manufactured parts.

# Recommendations for Flanged-Connector Military Standards and Specifications

The uncertainty concerning the final form of the flanged-connector military standards and specifications led to the mutual decision to limit the expenditures under Contract AF 04(611)-11204 to the preparation of typical drawings, nominal component dimensions, and recommendations concerning selected types of information. As Air Force decisions were eventually made concerning the military standards and specifications that should be issued, it was expected that the Rocket Propulsion Laboratory, using information in this report as a basis, could prepare the necessary documents. The recommendations prepared by Battelle-Columbus are discussed in three categories: (1) military standards, (2) nickel-plating specifications, and (3) connector performance specifications.

# Military Standards

Appendix C contains the drawings and nominal dimensions prepared by Battelle-Columbus for the flanged-connector military standards. First, suggested assembly instructions have been prepared. Next, an assembly drawing is shown which is believed to be suitable for all of the assembly military standards. Finally, nominal part dimensions have been prepared for Connectors 1 through 10 of the following flanged connectors. Dimensions for Connectors 11 through 27 are to be generated after testing during an overrun effort.

- 1. 100-psi, aluminum, cryogenic, 1 through 16 inches
- 2. 200-psi, aluminum, cryogenic, 1 through 16 inches
- 3. 500-psi, aluminum, cryogenic, 1 through 16 inches
- 4, 1000-psi, aluminum, cryogenic, 1 through 16 inches
- 5. 1500-psi, aluminum, cryogenic, 1 through 16 inches
- 6. 100-psi, stainless steel, cryogenic, 1 through 16 inches
- 7. 200-psi, stainless steel, cryogenic, 1 through 16 inches

8.	500-psi, stainless steel, cryogenic, 1 through 16 inches
9.	1000-pai stainless steel cryogenic 1 through 16 inches
10.	1500-nei stainless steel cryogenic i through 16 inche
•	
11.	2000-psi, stainless steel, cryogenic, 1 through 3 inches
12.	2500-psi, stainless steel, cryogenic, 1 through 3 inches
13.	3000-psi, stainless steel, cryogenic, 1 through 3 inches
14.	4000-psi, stainless steel, cryogenic, 1 through 3 inches
15.	5000-psi, stainless steel, cryogenic, 1 through 3 inches
16.	6000-psi, stainless steel, cryogenic, 1 through 3 inches
17.	100-psi, stainless steel, hot gas, 1 through 3 inches
18.	200-psi, stainless steel, hot gas, 1 through 3 inches
19.	500-psi, stainless steel, hot gas, 1 through 3 inches
20.	1000-psi, stainless steel, hot gas, 1 through 3 inches
21.	1500-psi, stainless steel, hot gas, 1 through 3 inches
22.	2000-psi, stainless steel, hot gas, 1 through 3 inches
23.	2500-psi, stainless steel, hot gas, 1 through 3 inches
24.	3000-psi, stainless steel, hot gas, 1 through 3 inches
25.	4000-psi, stainless steel, hot gas, 1 through 3 inches
26.	5000-psi, stainless steel, hot gas, 1 through 3 inches
27.	6000-psi, stainless steel, hot gas, 1 through 3 inches

It is obvious from an examination of the nominal dimensions of the connector parts that a number of tolerances must be established for the final military standards. It is recommended that normal machining practices be followed for the most part. As an exception, the thickness of the seal disks must be controlled relatively closely, as must the dimensions and the surface finish of the outside diameter of the seal disks, and of the inside diameter of the lips of the flange seal cavities. The outer seal disk edges and the corners of the flange seal cavities must also be closely controlled.

#### Nickel-Plating Specification

The following comments are made in accordance with the respective paragraphs of MIL-P-27418 (USAF). The paragraphs not discussed are believed to be applicable to the plating of seals for flanged connectors.

3.2 Workmanship. On the basis of work with seals for threaded connectors, it is believed that a third paragraph (3.2.3 Sealing Surfaces) should be added to Paragraph 3.2. A suggested wording for this paragraph is: "The examination of the circumference

of each seal disk for conformance with 3, 2, 1 and 3, 2, 2 shall be conducted with a 5-power glass."

<u>4.2 Separate Specimens</u>. It is suggested that the following sentence be added to the paragraph for 4.2: "The examination of the separate specimens shall be conducted with a 5-power glass".

<u>4.3.2.</u>] Visual Inspection. It is suggested that the following sentence be inserted between the two existing sentences: "The examination of the circumference of each seal disk shall be conducted with a 5-power glass".

5. PREPARATION FOR DELIVERY. The handling damage that has occurred with the seals manufactured by Scientific Advances Inc. has amply demonstrated the need for the inclusion of packaging instructions. Such damage will be greatly amplified for the larger, heavier, flanged connector seals. The proper wording for this section of the specification can probably best be devised by the Rocket Propulsion Laboratory, using existing terminology. The major requirements to be included are: (1) that each seal be individually packaged and (2) that the packaging include some means for protecting the circumference of the seal disks until installation of the seal in the connector.

# Flanged-Connector Requirements Specification

The following comments are made in accordance with the respective paragraphs of MIL-F-27417 (USAF). Although there are many similarities between threaded and flanged connectors, the number of differences are believed to be great enough to warrant the preparation of a separate specification for flanged connectors.

1. SCOPE. The existing wording should be adequate with the substitution of "flanged connector" for "fittings".

2. APPLICABLE DOCUMENTS. The Rocket Propulsion Laboratory can best identify the applicable government documents.

3. REQUIREMENTS. This is a large section of the specification and considerable work must be done to make it applicable to flanged connectors. The paragraph on materials (3,2) must be revised to include the material for the aluminum hardware and the stainless steel hardware. The paragraph on design and dimensions (3,3) must be revised to exclude information applicable to threaded fittings (such as 3,3,1) and to include information on flanged connectors such as information on studs, bolts, and nuts. Paragraphs 3,4 and 3,5 appear to be generally applicable.

Paragraph 3.6 on performance needs major revision. In fact, it is believed that serious consideration should be given to the complete elimination of performance testing. The manufacturing variations that are critical to successful performance have been identified, for the most part. Thus, it is believed that pertinent manufacturing capability can best be evaluated by the careful inspection of selected critical components, such as different size seals.

Paragraphs 3, 7 and 3, 8 appear to be generally applicable to flanged connectors.

4. QUALITY ASSURANCE PROVISIONS. Paragraphs 4.1 and 4.2 appear to be generally applicable to flanged connectors. Paragraphs 4.3 through 4.7 are performance requirements which should be reexamined and either reduced or eliminated in accordance with the comments above. Instead, Paragraph 4.8 on examinations should be greatly expanded to require the manufacture and inspection of selected, critical, and typical components.

<sup>D</sup>aragraph 4.9 on test methods should be largely eliminated and replaced with a description of applicable examination methods. Paragraphs concerning inspection, such as 4.9.6 through 4.9.11, should probably be retained.

5. PREPARATION FOR DELIVERY. The Rocket Propulsion Laboratory is best able to evaluate the applicability of this material.

6. NOTES. The material in this paragraph appears to be generally applicable to flanged connectors.

# THREADED-CONNECTOR SUPPORT

Contract AF 04(611)-11204 established Phase III to provide engineering support during the introduction of AFRPL threaded connectors (see Figure 91) to industrial firms and government agencies. Although it was anticipated that support might be required in regard to assembly tools and inspection techniques, the exact nature of the support was to be determined during the course of the contract.



# FIGURE 91. AFRPL THREADED CONNECTOR

Work was conducted on four aspects of AFRPL stainless steel threaded connectors: (1) seal removal, (2) seal retention and misalignment limitation, (3) increased radial seal loading, and (4) stress relaxation.

#### Seal Removal

Although axial loads of up to 3000 pounds are required to assemble Bobbin seals in threaded connectors, the forces required to remove the seated seals are much lower. When the nut of a threaded fitting has been removed, the fitting halves can be separated by hand by a cantilever force applied to one of the connector flanges. A moment of about 150 in-lb is sufficient tor the 1-inch connector, with lower moments being required for the smaller connectors.

When the connector halves are separated, the Bobbin seal remains in one of the connector flanges. The most straightforward way of removing the seal is to place the blade of a medium-sized screwdriver at the root of the exposed seal disk and to strike

the handle of the screwdriver a sharp blow. Two or three such blows are almost always adequate to force the Bobbin seal from the connector flange.

Although a screwdriver can be used effectively, such a tool is not desirable if the skill of the technician is limited. There is a strong tendency to use the blade of the screwdriver to pry the seal from the flange, using the flange lip as a fulcrum. Although prying is successful if done by the proper tool, such action with a screwdriver is undesirable because it tends to nick the edge of the relatively soft stainless steel of the flange. Such a nick can cause material to protrude into the seal cavity, and the protrusion damages seals during subsequent assembly.

A prying action is satisfactory for removing the Bobbin seal from a connector flange if the forces are applied properly. Figure 92 shows a simple tool based on this precept that was designed and tested satisfactorily during the program. By dimensioning the yoke to fit snugly between the end of the flange and the exposed seal disk, it is possible to exert a high axial force without a sharp corner of the tool digging into the lip of the flange. The tool can also be used to separate the flanges, as shown in Step 1. After the seal is broken loose, it can be removed easily from the flange. It is believed that this principle could be incorporated into an inexpensive tool with different size yokes for each connector size.



FIGURE 92. SEAL-REMOVAL TOOL

#### Seal Retention and M.salignment Limitation

During the early part of the program, it became apparent that two improvements would be desirable for the threaded connectors: (1) a method of retaining the seal in a flange cavity during assembly of the connector, and (2) a method of limiting the degree of misalignment at which the connector could be assembled. These problems were first considered separately and their parameters are discussed separately. However, the most desirable solution included features applicable to both improvements, and this configuration is described on pages 161-164.

# Seal Retention

Most of the laboratory assemblies of the threaded connectors had been made with the connectors in a horizontal position, and with one flange of the connector relatively unattached. In this assembly mode it was comparatively easy to secure the locse Bobbin seal in one connector flange with one hand while the unattached connector half was brought in contact with the seal by the other hand. It was then relatively easy to snug the nut fingertight while the connector halves were kept pressed together. Hundreds of such assemblies were made at Battelle-Columbus and at the Rocket Propulsion Laboratory without difficulty.

As the AFRPL threaded connector was described to various representatives of industry and government, an increasing number of comments were received concerning the fact that many field assemblies would be more difficult to complete than the laboratory assemblies. Not only would space around the connector often be limited, but in many cases the connector flanges would be attached to rigid or heavy components that would be difficult to move. In such installations, it would usually not be possible to hold the seal in one flange while the other flange was brought into place. With both hands being used to move or support the flanges, it would be easy to jar the flange containing the scal sufficiently to dislocate the seal from the seal cavity. Not only would this be a nuisance, but, more importantly, the sealing surface would probably be damaged. Thus, practical experience indicated a need for snapping the seal in place in one flange prior to assembly of the connector.

Consideration of the configuration of the AFRPL connector led to the early conclusion that a logical means of retaining the seal was to extend one of the flanges to enclose most of the Bobbin seal. The space between the seal disks could then be used for an appropriate element to bear against the inside diameter of the flange extension. Figure 93 is a simplified drawing of this basic approach.

Available Space. One of the basic parameters was the space available for the retention action. In the existing specifications, the length of the seal disk and the length of the seal tang were the same for the 3/8-, 1/2-, 3/4-, and 1-inch seals. Thus, if a retention configuration could be found to work within these dimensions, the same configuration could be used for four of the six specified seals, and few changes would be required in the specifications.



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#### FIGURE 93. MODIFIED FLANGE AND SEAL WITH SPRING

After working with this envelope, it became apparent that sufficient radial space was available, but that the axial space was very limited. The axial limitation stemmed primarily from the relatively large movement needed to seat the seal. When this distance, with its tolerances, was added to a 0.015-inch clearance needed between the seated flanges, the flange extension was very short, and the end of the flange extension was located approximately at the middle of the unseated seal. This meant that simple circular retention shapes such as that shown in Figure 93 were unacceptable and that any retention device located on the seal had to extend to both sides of the midpoint between the seal disks.

Friction Retention. In one approach, seal spring clips were designed to be pushed down by the flange extension as the seal was pushed into the flange cavity. By exerting a radial force against the ID of the extension, the spring clip would hold the seal in the cavity. In these configurations, the extension was not undercut (see Figure 93).

With the proper force and the proper surface conditions on the lip and the spring clip, the seal could be held satisfactorily by this design. The major limitation to this concept was the possible variation in friction between the flange extension and the clip. A small film of lubricant on the clip reduced the friction to the point where the seal could be easily jarred out. It was believed that control of  $\cdots$  surface conditions of the lip and clip would be difficult. A second problem with this design was the absence of an axial force tending to hold the seal back in the seal cavity. Thus it would be expected that seals would be only partially inserted on a certain percentage of connectors and this could lead to improper assembly of the connectors.

Snap-Action Retention. As improved seal-retention methods were sought, the following features were identified as being desirable:

- (1) The clip should be easily placed on a completed Bobbin seal
- (2) The seal (with clip attached) should be easily inserted in the connector

- (3) The clip should hold the seal snugly in the connector
- (4) The seal and clip should be removable by hand prior to assembly (this may be desired to inspect the seal and the seal cavity)
- (5) The clip should not interfere with proper assembly of the connector
- (6) The seated seal and clip should be easily removed from the connector upon disassembly of the connector.

The snap-action requirements emphasized the short axial dimensions available. One requirement was that the flange extension had to be undercut to provide an indentation for the retention device. Second, the retention device had to have a cam rise on the leading edge to force the protrusions down. Finally, the retention device had to have a reverse cam on the opposite side of the protrusion so the seal could be removed from the seal cavity by hand.

By using all of the distance between the inside of a seal disk and the midpoint of the seal (approximately 0.090 inch), it was possible to accomplish these actions. However, the taper of the seal disk was found to limit the amount of radial deflection available for inward deflection during insertion of the seal. Thus, in the final design, as shown in Figure 94, the flange extension was given a fairly sharp lip, and the protrusions of the retention device were made to extend at an angle from the main support ring, which clamped around the seal tang.

#### Misalignment Limitation

During July 1967, approximately twenty 3/4-inch stainless steel AFRPL unions were installed in a 4000-psi liquid-hydrogen piping system at the Rocket Propulsion Laboratory. Despite careful assembly of the connectors, a pressure test showed that five of the connectors leaked badly. It was noted that the leaking connectors were noticeably misaligned prior to the final tightening. Since misalignment conditions can be expected in most tubing and piping systems, sample parts were sent to Battelle-Columbus to determine whether misalignment was the problem and whether modification of the connector configuration could assure satisfactory assembly despite initial misalignment.

After an extensive examination of the parts, it was concluded that leakage had occurred because of two basic factors: (1) large, initial misalignment angles, and (2) insufficient radial sealing forces in the seals. The radial sealing forces are discussed in more detail in a subsequent section. The large misalignment angles seemed to have caused leakage for one of three reasons: (1) the seal, which was not fully enclosed by both flange cavities, was forced initially against the edge of a connector flange, damaging the seal and preventing sealing when the connector was loosened and assembled properly, (2) the large angles and high moments prevented proper seating of the seal disks, and (3) a sealing surface of the seal was damaged as the seal slid in the flange cavity during straightening of the misaligned flanges.

Although it was mutually agreed with the Project Monitors that some misalignment had to be tolerated in a threaded connector, it was also agreed that the large angles (up to 7 degrees) made possible by the dimensions of the connector were too great for the reliable sealing of high-pressure helium. Therefore, an effort was undertaken to modify the connector to limit the amount of possible misalignment.


a. Spring Clip

b. Retention Flance

1



c. Modified Nut

# FIGURE 94. PARTS TO ACCOMPLISH SEAL RETENTION AND MISALIGNMENT LIMITATION

Consideration of the misalignment problem led to the basic decision that only minor changes in the connector configuration would be acceptable. This decision was based on the conviction that the existing fitting was simple and that this simplicity was strongly conducive to proper use of the fitting. On the other hand, the introduction of more complicated connectors in the past by industry had demonstrated that an increase in the number of connector parts or an increase in the complexity of assembly could override the advantages that the added features were designed to accomplish. Thus it was decided that changes designed to limit misalignment would be limited to changes in the existing parts.

It was apparent that improper assembly due to misalignment was possible in the connector because the nut could be engaged with the threaded flange even though the plain flange was turned at a large angle to the nut. This assembly angle was possible because of two dimensional relationships: (1) the clearance between the ID of the nut flange and the shoulder of the plain flange and (2) the clearance between the OD of the plain flange and the ID of the nut barrel. Thus, the general method selected to limit the misalignment angle of the threaded connector was to reduce the clearances for these dimensional relationships.

Although this concept was effective in reducing the angle of misalignment at which the connector could be assembled, it was still possible for the seal cavities in the two flanges to become sufficiently misaligned that the seal would not always be guided into the opposing seal cavity. Consideration of this problem produced the concept of using the threads on the nut to position the plain flange. In this way, engagement of the nut with the threaded flange would automatically align the plain flange with the threaded flange.

This guidance was believed to be most easily accomplished by locating the flange extension for seal retention on the plain flange. Then, by extending the threads deeper into the nut (to enclose the Bobbin seal in the assembled connector) the nut threads could be used to enclose the OD of the plain flange extension. This configuration is described more fully below.

# Selected Design for Seal Retention and Misalignment Limitation

Figure 94 shows the configuration of the spring clip, the modified plain flange, and the modified nut selected to accomplish seal retention and misalignment limitation. Figure 95 shows the four major stages of assembly of the connector. The modification of the parts, the assembly steps, and the evaluation of the configuration are discussed below.



Step I. Spring Clip Placed on Bobbin Seal



Step 2. Seal-Clip Assembly Snapped into Retention Flange



Step 3. Nut Threaded Past Interrupted Thread



Step 4. Nut Threaded onto Other Flange, Tightened with Proper Torque

FIGURE 95. ASSEMBLY STEPS OF MODIFIED CONNECTOR 161

Spring Clip. As shown in Figure 94, the spring clip consisted of a C-shaped spring made from flat spring stock that was snapped over the seal tang. Two humps were formed, one near each end of the "C". These humps protruded outward beyond the seal disk diameter. The humps were offset from the rest of the material so the major part of the clip was in line with the tang, but the humps were displaced slightly in the axial direction. This allowed the humps to snap under the flange extension lip. The radii of the humps provided the entrance camming action and the exit camming action. By having the clip touch the tang circumference only at the ends of the "C" and at the middle of the "C", the clip could deflect radially, thus providing a radial spring force. By making the width of the material touching the tang the exact width of the tang, the clip provided good axial rigidity for the seal. The major limitation of the design was the fact that the clip would snap in place in the flange extension only if the humps were slanted toward the flange cavity. However, when the seal was reversed there was no snapping action, and the looseness of the seal was believed to be sufficient evidence that the seal was not properly installed.

Considerable time was required to develop a procedure for fabricating the spring clips. The final process consisted of the following steps:

- (1) Layout and cut 1/8-inch strips of spring steel
- (2) Layout dimples
- (3) Form dimples on first die
- (4) Finish-form dimples on second die
- (5) Grind off stack to make offset dimple and reduce ring width on bench grinder
- (6) Form ring around mandrel
- (7) Secure clips to heat-treating mandrel
- (8) Heat-treat and draw
- (9) Finish fit to seal.

Plain Flange. Most of the connector modifications were limited to the plain flange. Because the plain flange was also equipped with threads, the modified flange was entitled the retention flange. The modifications to the plain flange and their purpose are discussed briefly below.

- (1) The plain flange lip was extended, and an undercut was provided on the ID of the extension for the spring clip.
- (2) A hub was added to the tube portion of the plain flange to provide a closer fit with the ID of the nut flange.
- (3) An external thread, identical to the thread on the threaded flange, was added to the OD of the plain flange, excluding the flange extension. By threading the nut over and past this interrupted

thread, the nut provided protection for the exposed disk of the retained scal prior to assembly of the connector.

Nut. The nut remained unchanged except that the threads were extended deeper into the nut to provide guidance for the OD of the plain flange extension.

Connector Assembly. The major assembly steps of the modified connector are as follows:

- (1) Snap the seal into the retention flange
- (2) Thread the nut completely over the interrupted threads on the retention flange
- (3) Engage the nut with the threaded flange, using the fingers to thread the nut as far as possible
- (4) Tighten the nut slowly to the required torque.

Note. When the nut is loosened, it first encounters the interrupted thread on the retention flange. It is necessary to turn the nut slowly while the retention flange is pushed away from the threaded flange. When the nut threads engage the interrupted threads, the nut can then be removed quickly.

Modified Connector Evaluation. Five 3/4-inch retention flanges and modified nuts were fabricated and a number of spring clips were made. Three connectors were forwarded to the Rocket Propulsion Laboratory for evaluation, and two connectors were retained for evaluation at Battelle-Columbus. The results of these evaluations are discussed briefly.

One of the major questions about the configuration was the ability of the nut thread to exert a thrust against the interrupted thread as the fitting was disassembled. Although the primary purpose of threads is to exert high axial loads, the specific function required in the design was believed to be unusual, if not unique. As a preliminary evaluation of this action, a threaded flange was clamped vertically in a vise. A nut was threaded past the interrupted threads on a retention flange and the nut was then threaded on the threaded flange, with the tubular portion of the retention flange extending upward. Weights up to 100 pounds were placed on top of the tube of the retention flange, and the nut was used to raise the weights by creating thrust against the leading threads of the retention flange. The ease with which the weights were raised indicated that the necessary axial force could be supplied to separate the flanges of an assembled fitting to the point that the nut would engage the threads of the retention flange.

The evaluation of the spring clips was not as promising. Although some of the clips worked satisfactorily, it was apparent that the camming action was influenced substantially by small changes in the dimensions of the clips and of the inside disk root radii and tang length. It would be expected that properly made dies would alleviate much of the dimensional variation in the spring clips. However, it appeared that the production costs required to control the related dimensions of the spring clips and of the seal could increase the cost of the seal/spring-clip assembly as much as 50 percent over the cost of the basic seal.

Even more problems were encountered in the evaluation of the misalignment limitation features. The interrupted threads on the modified flange were subject to seizure because of galling, contamination, and peening from the nut. An almost insurmountable problem developed because of the relatively close fit between the outside circumference of the seal disks and the inside surface of the lip of each flange seal cavity. Thus, even with the initial misalignment limited to 1 degree by the tolerance between the nut and the modified flange, the edge of the seal disk extending from the retention flange was not automatically piloted into the seal cavity of the threaded flange by the tightening of the nut. Under these conditions, it would be expected that a number of seal disks would be damaged through contact with the edge of the flange cavity lip, preventing proper sealing action.

<u>Conclusions and Recommendations</u>. It was concluded that neither the spring clip nor the modifications to limit misalignment during assembly were practical. Two courses of future action are recommended. First, work should be done on a tool to preset the seal in either the plain flange or the threaded flange as a means of retaining the seal. Some thought has been given to this approach during the preparation of the report and it appears promising. Second, it is recommended that the assembly procedure for AFRPL connectors require that all parts be sufficiently loose initially that the preseated seal can be placed by hand into the opposing seal cavity and the nut made fingertight. The proper preload would then be applied. The degree of misalignment possible in this procedure would be limited by the depth of the seal cavity.

# Increased Radial Seal Loading

Starting in 1967, a growing amount of evidence was obtained that the radial sealing loads in the 3/4- and l-inch, 4000-psi stainless steel seals and in certain flanged connector seals were not sufficiently high to insure reliable sealing on assembly. The work in this area increased as the extent of the problem became better known, until, in the last months of the program, the increase in radial sealing loads for selected threaded and flanged connectors was defined as a major area requiring additional theoretical and laboratory effort.

Although the answer to this problem cannot be presented in this report, it is believed advisable to record a relatively systematic description of the problem with the threaded connectors, of the work conducted so far, and of the most promising avenues for solution. It is hoped that this record will serve as a helpful reference during any future work. The pertinent information for flanged connector seals has been included in earlier report sections.

The record is presented in the following major sections: (1) theoretical background, (2) developmental background, (3) qualification tests at Battelle-Columbus, (4) qualification tests at the Rocket Propulsion Laboratory, (5) test installation at the Rocket Propulsion Laboratory, (6) seal investigations at the Rocket Propulsion Laboratory, (7) seal investigations at Battelle-Columbus, (8) candidate materials and configurations, and (9) recommended future activities.

# Theoretical Background

The theoretical background to the problem is discussed in four categories: (1) forces required to seal helium, (2) the Bobbin seal configuration, (3) seal material selection, and (4) seal design.

Forces Required to Seal Helium. Extensive work has been done by the IIT Research Institute (16, 17) and by the General Electric Advanced Technology Laboratories (18) on the forces required between two metal surfaces to seal helium to approximately 2 x 10<sup>-7</sup> atm cc/sec per linear inch of seal (the maximum leakage specified for the Air Force separable connectors). Although the relationships developed for the different parameters are somewhat complex, in general it was shown by both facilities that the force between two metals with fairly smooth surfaces ( $\approx 16$  rms) must be from 2 to 3 times the yield strength of the softer material to achieve low helium leakage. The step-by-step application of the IITRI analysis procedure to a separable connector is described on pages 143-148 of a final report on threaded connectors<sup>(19)</sup>.

Although the work by IITRI and General Electric showed that random surface conditions had a significant effect on the actual leakage exhibited by two metal surfaces, neither facility developed design guidelines to overcome these occurrences. On the basis of previous work with many kinds of static seals, and keeping in mind the high forces needed to yield metallic surfaces, project personnel at Battelle-Columbuc selected an 0.020-inch-wide sealing surface as being optimum for the AFRPL threaded and flanged connectors. Extensive experience at many facilities has shown that metal sealing surfaces approximately 0.005 inch wide (a width commonly found in highperformance aerospace valves) encounters many leakage problems because of damage and contamination. On the other hand, the 0.060-inch-wide sealing surfaces found in relatively dependable O-ring installations would require excessive force in a metal-tometal seal. Thus, the 0.020-inch-wide surface (which is found in many installations using small O-rings) was a judgmental selection between two extremes. As an example, for a soft nickel plating with a yield strength of about 10,000 psi, the force on an 0.020inch-wide sealing surface would be 600 lb/in, to obtain a pressure equal to 3 times the yield strength of the nickel.

Bobbin Seal Configuration. The development of the Bobbin seal configuration has been described in detail in two reports (19, 20), and the principles of the sealing action are summarized on pages 4 and 5 of this report. Two features of the seal are of particular interest to this discussion: (1) the toggle action of the seal disks and (2) the yielding of the seal material during assembly.

The toggle action of the seal disks was developed as a means of amplifying the axial force to attain a higher radial sealing force. As noted above, it was realized that high sealing forces would be required. By keeping the axial force to a minimum, the size and weight of the threaded fittings would be kept to a minimum. In practice, the toggle action of the seal disks produced a radial sealing force approximately twice the axial seal seating force. Although this was very desirable, the mechanical advantage of the toggle action, and the sensitivity of toggle angle to initial diametral tolerances between the seal and the seal cavity made it difficult to determine the radial sealing force accurately from a measured axial seating force.

The yielding of the seal structure was included in the seal design as a means of limiting the sealing forces that could be developed by dimensional variations in production parts. While this objective was accomplished to a certain degree, the actual effect

on the radial sealing forces of differences in the seal dimensions and differences in the material yield strength were much greater than had been anticipated.

Seal Material Selection. Although considerable thought was given to the selection of material for the structure of the threaded fittings (19, 20), the selection of austenitic stainless steel for the connector flanges resulted in the rather automatic selection of annealed austenitic stainless steel for the secl. In appreciable amount of effort was spent in selecting the best plating for the seal, and this work was amplified for the flanged connector seals (see pages 5-7). Nickel and gold plating were selected initially, and this selection was verified in the later effort.

When no apparent problems developed with the austenitic stainless steel seals during the development work, the specifications were based on this material. Thus, when the need arose for an increase in radial sealing force, two problems existed: (1) the problem of achieving increased strength within the specification dimensions, and (2) a lack of information concerning candidate seal materials other than annealed austenitic stainless steel. The work on these problems is described in detail subsequently.

<u>Seal Design</u>. At one time an attempt was made to develop an exact procedure for designing Bobbin seals. This was not successful because of the complex combination of elastic and plastic strains that were developed in the seal. An approximate design procedure was then developed, and seal design nomographs were developed for inreaded scals using this procedure and experimental data. <sup>(19)</sup> Essentially, this calculation procedure was based on the concept that the seal tang developed compressive, plastic strains and that the radial sealing force was developed by the force needed to compress the seal tang.

As work was continued on Contract AF 04(611)-11204, it became increasingly apparent that this design procedure was not sufficiently accurate for small seals. It appeared that the basic reason for inaccuracy was that the material in the seal disks made a significant contribution to the radial sealing force in the small seals.

A new design procedure was then developed which included the material in the seal disks as well as the material in the seal tang. Although the calculation was still an approximation, the results of the revised procedure agreed much more closely with the experimental results for threaded seals, and it was judged to be acceptable for large as well as small seals. The revised seal-design procedure is described below.

For simplicity, the assembled seal disk was assumed to be a completely flattened cone with forces and dimensions according to Figure 95.





 $F_S$  = radial seal force per inch of seal disk circumference, lb/in.

 $F_T =$  radial force per inch of seal tang outside diameter at the root of each seal disk, lb/in.

 $D_0$  = seal disk outside diameter, in.

- $D_e = seal tang outside diameter, in.$
- LT = seal disk thickness, in.

The forces and dimensions for the seal tang model are shown in Figure 97.



FIGURE 97. SEAL TANG STRESS MODEL

 $D_i$  = seal tang inside diameter, in.

L = tang length, in.

t = tang thickness, in.

On the basis of the elastic stress/strain theory for thick-walled cylinders<sup>(21)</sup>, circumferential stress is related to pressure as follows:

$$\sigma_{\theta} = -\frac{a^2 b^2 (p_0 - p_i)}{(b^2 - a^2)} \cdot \frac{1}{r^2} + \frac{p_i a^2 - p_0 b^2}{b^2 - a^2} , \qquad (23)$$

where

 $\sigma_{\theta}$  = circumferential stress, psi

a = inner cylinder radius, in.

b = outer cylinder radius, in,

p<sub>i</sub> = internal pressure, psi

 $p_0 = external pressure, psi$ 

r = radial location of stress, in.

Considering the seal disk first, if the circumferential stress at  $D_e$  equals the yield strength of the material, the radial seal force increment contributed by the seal disk is:

$$\Delta F_{S} = S_{y} \frac{(D_{o}^{2} - D_{e}^{2}) \times LT}{2D_{o}^{2}}$$
(24)

where

P<sub>i</sub> = 0

 $r = a = D_{p}/2$ 

 $S_v \approx$  yield strength of seal material, psi

 $\sigma_{\theta} = \mathbf{S_v}$ 

 $\Delta F_S$  = radial seal force increment contributed by the seal disk, lb/in.

$$b = D_0/2$$
.

Similarly, the radial seal force increment contributed to each seal disk by the tang was calculated:

$$\Delta F_{T} = S_{y} \frac{(D_{e}^{2} - D_{i}^{2}) \times \frac{L}{2}}{2D_{e}^{2}}$$
(25)

where

 $\Delta F_T$  = radial seal force increment contributed to each seal disk by the tang, 1b/in.

$$r = a = D_i/2$$
  
 $b = D_e/2$ .

It was assumed that the radial force at the inside diameter of the seal disk was distributed radially to the sealing surface. Thus:

$$\mathbf{F}_{\mathbf{S}} = \Delta \mathbf{F}_{\mathbf{S}} + \Delta \mathbf{F}_{\mathbf{T}} \times \frac{\mathbf{D}_{\mathbf{e}}}{\mathbf{D}_{\mathbf{o}}} \quad . \tag{26}$$

The foregoing calculations were based on limits from idealized elastic seal elements. Examinations of the dimensions of typical seals showed that sufficient radial compression existed to create plastic deformation at the inside diameter of the seal disks and in the seal tang. Using plastic theory<sup>(2)</sup> for the seal tang, the seal tang radial force contribution was calculated;

$$\Delta F_{T}P = \frac{L}{2} \times S_{y} \ln \frac{D_{e}}{D_{i}} , \qquad (27)$$

and then

$$\mathbf{F}_{\mathbf{S}} = \Delta \mathbf{F}_{\mathbf{S}} + \Delta \mathbf{F}_{\mathbf{T}} \mathbf{P} \times \frac{\mathbf{D}_{\mathbf{e}}}{\mathbf{D}_{\mathbf{o}}} \qquad (28)$$

# Developmental Background

The development of low-leakage threaded connectors began in early 1962. The major steps of each contractual effort are summarized on pages 1 and 2 of this report. Of primary importance to this discussion is the fact that little difficulty was experienced with the performance of nickel-plated, stainless steel Bobbin seals during the two major contracts on the development of threaded fittings. Thus, instead of exploring the performance characteristics of the Bobbin seal in detail, the majority of the work was concerned with designing, fabricating, and testing different types of connectors for aluminum as well as stainless tubing systems.

In fact, the performance of aluminum Bobbin seals was in need of significant work in 1966, at the start of Contract No. AF 04(611)-11204, and Phase I of this contract was established to complete that development. An excellent solution was found to this problem by the use of overaged  $6061-T6^{(22)}$ .

The work under Contract No. AF 04(611)-9578(19) to develop specifications for stainless steel Bobbin seals consisted of testing several 3/8- and 3/4-inch seals made to dimensions based on the original calculations (see page 14). In the first series of seals tested, five 3/8-inch and five 3/4-inch seals were made of Type 309 CRES. Although leak tests with these seals were very good, the disks did not deform properly. In a second series of tests, four 3/8-inch seals and four 3/4-inch seals were made with variations in selected dimensions. These tests were used to determine satisfactory tang thicknesses and disk thickness. In a third series of tests, four 3/8-inch seals were made to the selected dimensions and leak tested, Although the widths of the coined sealing surfaces were less than the optimum 0.020 inch, the sealing results were very good, and the radial loads, as estimated from the measured axial loads, were twice as high as required. As described in detail in the final report (19), the design equations then were modified on the basis of the experimental data, and a seal design nomograph was developed incorporating these changes. (A nomograph was also developed for aluminum threaded connector seals.) Seals were then designed for six tubing sizes. Table 58 shows the dimensions of these seals and the maximum axial loads required to seat sample seals made to these dimensions from Type 310 stainless steel.

It should be noted that radial sealing loads had not been measured for some time because the technique used was expensive. Instead, on the basis of early seal load tests, it was assumed that the radial sealing load was twice the axial seal seating load. It can be seen from Table 58 that the final stainless steel seals were thought to provide radial sealing loads from about 1150 lb/in. to about 1350 lb/in.

Tube Size, in.	Inside Tang Radius, <sup>I</sup> i	Tang Thickness, t	Outside Tang Radius, <sup>r</sup> e	Outside Disk Radius, <sup>r</sup> o	Tang Length, L	Seal Disk Thickness, LT	Minimum Axial Seal Seating Load, P <sub>A</sub> , 1b	P <sub>A</sub> per In. of Seal Circumference, 1b/in.
1/8	0.050	0.036	0.086	0.171	0.095	0.027	665	620
1/4	0.095	0.043	0.138	0.222	0.120	0.030	950	680
3/8	0.145	0.049	0.194	0.278	0.150	0.026	1215	683
1/2	0,191	0.053	0.244	0.329	0.150	0.026	1388	662
3/4	0.286	0.069	0.355	0.440	0.1 <b>5</b> 0	0.026	1595	572
1	0. <b>37</b> 2	0.089	0.461	0.546	0.150	0.026	2030	586

# TABLE 58.MEASURED DIMENSIONS, COMPUTED AND EXPERIMENTAL VALUES<br/>FOR AXIAL FORCE FOR TYPE 310 CRES BOBBIN SEALS

For a sealing surface width of 0.020 inch, these forces provided sealing pressures from 57,500 psi to 67,500 psi. By using soft nickel plating with a yield stress of about 10,000 psi, it was concluded that the seals would provide a sealing force equal to approximately six times the yield strength of the softer material. Thus the seals appeared to provide ample sealing force, and the specifications for the connectors were prepared, incorporating these dimensions. Figure 90 shows the final seal specifications that were issued, on the basis of this work.

# Qualification Tests at Battelle-Columbus

Qualification tests of representative connectors were conducted at Battelle-Columbus as a part of Contract AF 04(611)-9578. A total of twenty-eight 4000-psi stainless steel connectors (20 unions, 4 elbows, and 4 tees) successfully withstood the qualification tests with maximum helium leakage never exceeding  $3.9 \times 10^{-7}$  atm cc/sec per inch of seal circumference. This represented a total of 41 seals successfully tested. On two occasions excessive leakage occurred, once owing to human error and once when an insufficient preload was developed.

A statistical summary was prepared using the maximum leakage measured for each seal. When the data were normalized on the basis of the logarithmic equivalent of the leakage rate, it was found that the data approximated a normal distribution. The mean recorded maximum leakage was 2.6 x  $10^{-8}$  atm cc/sec of helium per inch of seal circumference.

# Qualification Tests at the Rocket Propulsion Laboratory

Concurrently with Contract AF 04(611)-11204, the Rocket Propulsion Laboratory prepared final specifications for the 4000-psi stainless steel connectors, and purchased a number of connector components from Scientific Advances, Inc., (SAI) in Columbus, Ohio. Qualification tests similar to those conducted at Battelle-Columbus were conducted at the Rocket Propulsion Laboratory. Although these test data were not completely summarized, the results, in substance, duplicated the results obtained at Battelle-Columbus.

# Test Installation at the Rocket Propulsion Laboratory

As described previously, twenty 3/4-inch, 4000-psi AFRPL stainless steel unions were installed in a liquid-hydrogen system at the Rocket Propulsion Laboratory. Five of these connectors leaked badly. Three of the connectors and two seals were sent to Battelle-Columbus for examination.

The separate seals were examined first. The disks appeared to have been fully deflected, the tang had apparently been satisfactorily yielded, and a seal seating surface approximately 0.010 inch wide was visible on the four sealing disks. Under a medium-power microscope, the sealing surface of one disk was seen to have a pronounced ridge where the seal had apparently been pressed against the edge of the flange cavity during an improper partial assembly prior to final assembly. The ridge was sufficiently large that the seal could not have sealed properly even in an aligned connector. The other seal showed no obvious cause of leakage. The only visible differences between this seal and seals from past Battelle work were (1) tool marks were visible in the nickel plating, indicating that the surface finish of the machined seals was rougher than Battelle's seals had been, and (2) the nickel on the yielded surfaces had a mat finish, while past Battelle seals showed a very shiny surface where the nickel was yielded.

The absence of structural abnormalities in the separate seals indicated that it was necessary to examine the sealing surfaces at the source of leakage in the three connectors. To find the leaks in these connectors, three corners of the hex nuts of each connector were machined away. This exposed the outside of the seals but left the connectors held together by the three remaining nut corners. An O-ring-sealed plug was made to fit inside the connectors, and the inside of each connector was pressurized with helium. Soap solution was used to locate the sources of leakage.

The three connectors were then cut in half so that each connector half was still held together by 1-1/2 corners of the nuts. The connectors were cut so the leak was contained in one half. This half was then separated so the sealing surfaces at the leak could be examined. The other half was encapsulated, polished, and etched so the cross sections of the assembled connectors could be examined and the hardnesses of the nickel plating could be measured.

An examination of the leakage area in one connector showed that the edge of the flange cavity had been nicked by a sharp object, causing a peened, inward protrusion of the metal. This protrusion scratched the nickel plating on the sealing surface during assembly of the seal. It was believed that this seal could not have sealed satisfactorily even if the connector had been aligned.

The other two connectors showed no obvious cause of leakage. The general appearance of the seals was identical to that of the separate seals. The surfaces of the flange cavity did not exhibit abnormalities, although it appeared that the seals had not pressed as tightly against the flange surface in the general area of leakage as they had

against other areas and against the flange which did not show a leak. Although the disk edges in one connector showed a somewhat different type of deformation, in general it was concluded that the seal structures had deformed normally. The coined sealing surfaces were approximately 0,010 inch wide.

Unfortunately, Battelle's seals that were made to the same dimensions as the leaking seals had been discarded or were assembled in connectors on test. However, it was possible to find seals made of stainless steel and René 41 from earlier research. The shiny appearance of the sealing surfaces on the Battelle seals indicated that the nickel had yielded more than was indicated by the mat sealing surface of the leaking seals. It was suspected that the plating of the leaking seals might be too hard. However, hardness readings showed that the hardness of the plating on the leaking seals was within specifications and compared with the hardness readings taken on a crosssectioned René 41 seal.

Consideration was then given to the possibility that the seal was not providing sufficient resistance to yielding. This could have been caused by the material in the seals having a low yield strength, or by interaction between the two sides of the seal. Thus, if one disk grossly yielded the seal structure, this might have significantly reduced the resistance to yielding in the other disk-tang area.

Scientific Advances, Inc. had made tensile specimens from the material used for the seal. The company records showed that the yield strength of the material in the 3/4-inch seals was approximately 35,000 psi. This was within the specifications and compared favorably with the yield strength of the Type 310 stainless steel seals of the same dimensions tested at Battelle-Columbus.

However, the difference in appearance of the nickel between the leaking seals and Battelle's past seals had not been explained. Battelle's Laboratory Record books and reports for all of the work on stainless steel seals were reviewed. Two aspects of the past work seemed to be pertinent.

First, it appeared that higher radial sealing pressures had probably been attained during the early research on separable connectors. It was not possible to compare the values directly because so many different seal dimensions had been used, and most of the radial-seal-seating loads were not measured directly. The axial-seal-seating loads were always measured, but these included the force necessary to rotate the disks, and the mechanical advantage of the rotating disks was difficult to estimate. However, the axial-load peaks of early seals with dimensions similar to those of the leaking seals were generally about 30 percent higher than the axial-load peaks of the leaking seals. In addition, the thickness of the disks at the hinge line of the final seals (0.027 inch) was about 30 percent higher than the thickness (0.020 inch) of most of the disks of the early seals. Thus, a greater amount of the axial load required to seat the leaking seals was needed to rotate the disks (this force does not contribute to the radial seal-seating force) than was the case with the seals with thinner disks. In conclusion, it was estimated that the radial seal-seating pressure on the leaking seals was probably significantly lower than for the early seals.

The second aspect was the difference between the appearance of the axial sealseating-load curve for the early seals and the appearance of that for the seals made to dimensions in the specifications. Most of the early curves showed two distinct humps. (See subsequent Figure 99.) Each hump represented the seating action of a seal disk. Two humps were created because one disk was always stronger than the other disk. Many of the curves for the specification seals had one full hump and a partial hump. It was believed that the absence of a full second hump may have resulted from the first disk causing the tang to yield sufficiently that the second disk was not supported as much as normal by the tang. The possibility of interaction between the two sides of the tang was increased for the larger seals because the tang was shorter in relation to the tang thickness than was the case for the 3/8-inch seals. This interaction might explain the failure of the seals to function satisfactorily in the misaligned connectors despite the fact that all the seals functioned satisfactorily in the aligned connectors.

As a result of the examination of the connector parts and of Battelle's past work, the following conclusions were reached:

- (1) The unretained Bobbin seal was susceptible to damage and improper installation during assembly of the connector.
- (2) The leaking connector parts were made within the specifications. Although not apparently a cause of leakage, the machined surface of the seals was noticeably rougher than were earlier Battelle seals.
- (3) The radial seal-seating pressure in the 3/4-inch connector was satisfactory for an aligned connector, but higher sealing pressures were attained in many of Battelle's earlier seals.
- (4) As the seal tang is made shorter in relation to its thickness, a ratio may be reached for which the disks do not seat sufficiently independently.

The following steps were recommended to overcome the apparent deficiencies of the connector:

- (1) A means should be devised to retain the seal in one of the flange cavities during assembly of the connector.
- (2) Consideration should be given to changing the rms specification on the seals from 32 to 16.
- (3) The radial sealing force of each size of stainless steel seal should be measured, and if necessary, increased.
- (4) The length-to-thickness ratio for the tangs of all sizes of stainless steel seals should be checked to insure independent seating action of the sealing disks.

# Seal Investigations at the Rocket Propulsion Laboratory

The decision was made to conduct work under Steps 3 and 4 at the Rocket Propulsion Laboratory. Because of the installation problems with the 3/4-inch connectors, it was decided that work would be concentrated on this connector size. Hewever, it was believed that the results of this work would probably apply to 1-inch seals also. Strain-gaged rings were fabricated at Battelle-Columbus from high-strength steel to replace the retaining lip of the 3/4-inch connector flanges. Plugs were machined to fit inside the rings so the seal tang could be compressed between the plugs and the seal disks would press outwardly against the load rings. These components were calibrated and then forwarded to the Rocket Propulsion Laboratory.

First, the maximum radial sealing loads were measured for seven SAI seals. The first two seals were seated normally, with the disks both being seated in one loading. The next five seals were loaded to seat only one disk at a loading. The results of the load tests with these seals are shown in Table 59. Also shown are the measured maximum axial loads, the original seal disk diameters, and the percent of sealing surface that was matted, or did not seem to be as shiny as desired. Figure 98 shows a plot of the maximum measured radial load as a function of the maximum measured axial load.



FIGURE 98. COMPARISON OF MAXIMUM RADIAL AND AXIAL SEAL-SEATING LOADS - 3/4-INCH SAI STAINLESS STEEL SEALS

The major conclusion from this work was that the radial scaling load varied much more than was anticipated for scals made by the same manufacturer from one bar of material. Tentative conclusions were: (1) the initial fit of the scal was not a major load-determining factor, (2) dimensional variations within the scal probably were a major load-determining factor, (3) the prevalence of the mat surface indicated the need for increased scal strength, and (4) the rule-of-thumb estimate that the radial scaling force was twice the axial scaling force was an effective approximation.

In a second series of tests it was shown that significant interaction existed between the seal disks. In essence it was shown that the radial load established by the first disk to seat could be reduced as much as 50 percent when the second disk was seated.

In a third series of tests, 3/4-inch seals were machined from Type 304L stainless steel to the dimensions in the specifications. Radial sealing forces from 5 seals varied fairly uniformly from 520 lb/in. to 830 lb/in. of seal circumference. It was thus shown that the introduction of a material-properties variation created a set of

	Seal Disk	Max Axial	Maximum I Şeal Circumi	Percent of Matted Sealing	
Specimen <sup>(a)</sup>	Diameter <sup>(b)</sup> , in.	Load, 1b	Axial	Radial	Surface
1	0.884	1260	452	965	33
	0.887	1740	623	1275	33
2	0.885	1135	407	790	25
	0.887	1480	530	1174	10
3	0,882	1380	495	935	33
	0.887	2280	817	1385	None
4	0.887	1400	502	880	None
	0.887	1485	533	900	33
5	0.885	1330	477	885	50
	0.887	1580	567	1065	25
6	0.886	1410	506	990	25
	0.887	1775	637	1380	33
7	0.885	1400	502	1050	33
	0.881	1480	530	940	75

#### TABLE 59. LOAD TESTS MADE WITH SAI 3/4-INCH STAINLESS STEEL SEALS AT ROCKET PROPULSION LABORATORY

(a) Disks of Specimens 1 and 2 seated during one loading. Disks of Specimens 3 through 7 seated one at a time.

(b) ID of flange lip = 0.888 in.

(c) Based on ID of flange lip.

readings almost completely below those obtained from the SAI seals. It was also shown, however, that the lowest sealing load was still very close to the original design minimum of 600 lb/in.

In a fourth series of tests, 3/4-inch seals were made from Type 304L stainless steel with different tang lengths. These seals were seated to determine the effect of increased tang length on the radial sealing load and on disk interaction. It was found that a 100 percent increase in tang length resulted in a 50 percent increase in radial sealing load. A greater increase in tang length did not increase the radial sealing load significantly because the seal disks were not strong enough to yield a stronger tang.

On the basis of the work conducted at the Rocket Propulsion Laboratory, it was concluded that the radial sealing load of the 3/4- and 1-inch stainless steel seals should be increased and that the increased load should be accomplished by lengthening the seal tang since this would tend to separate the seating action of the seal disks.

#### Seal Investigations at Battelle-Columbus

The work on the seal retention and misalignment problems of threaded connectors (see pages 157-164) precipitated a more intensive investigation of the radial sealing loads of the stainless steel threaded seals. This occurred because it was not possible to select a final seal-retention or misalignment-limitation configuration until the seal dimensions had been selected.

As the parameters associated with an increased seal tang length were examined more closely, it became apparent that this approach had several disadvantages. The most important disadvantages were an increase in the size and weight of the connectors. While the increase required for the 3/4- and 1-inch connectors might have been ac ceptable, the same dimensional increases on the 3/8- and 1/2-inch connectors might have been unacceptable. Related to these problems were secondary problems associated with the change of many of the connector specifications. In addition to the problems associated with an increased tang length, several higher strength materials seemed to be capable of producing the required increase in radial sealing load within the specification seal dimensions. If such a material were used, the higher yield would provide greater springback in the seal and might minimize the problem of disk interaction. Thus, the decision was made to develop an improved seal within the specification dimensions.

The accomplishment of this objective proved to be much more difficult than anticipated. However, considerable work on the problem was accomplished, and it is summarized in the following sections.

Radial Sealing Load Calculations. The improved procedure for calculating radial sealing loads (see pages 166-169) was used to calculate the loads for the 3/4-inch SAI seals. In the original calculation, 30,000 psi had been used as the yield strength of austenitic stainless steel. However, this is a minimum tensile yield strength, and a more extensive search of the literature produced data showing that the compressive yield strength of austenitic stainless steel could be between 40,000 and 50,000 psi for the degree of strain produced in the Bobbin seals. When 50,000 psi was used in the revised calculation procedure, the calculated radial sealing load for the 3/4-inch seals agreed well with the average of the experimental results obtained at the Rocket Propulsion Laboratory for the 3/4-inch SAI seals.

The decision was then made to calculate the radial sealing loads for all the seals in the specification. This calculation was of particular interest because the sealing surfaces of the 3/8- inch SAI seals seemed to be consistently shiny, and it was hoped that the radial sealing load for this seal might be used to estimate the required radial sealing load increases in the 3/4- and 1-inch seals. Table 60 shows the radial sealing loads calculated for the specification seals, using 40,000 and 50,000-psi yield strengths. It can be seen that the calculations for the 3/4-inch seal using a 40,000-psi yield strength agreed well with the average values measured at the Rocket Propulsion Laboratory for 3/4-inch seals made of Type 304L stainless steel.

If the radial sealing load for the 3/8-inch seal was satisfactory, these calculations showed that the 1/8- and 1/4-inch seals were also satisfactory. The calculations also showed that the 1/2-inch seal was slightly understrength, while the 3/4-inch seals were almost 50 percent understrength.

Tubing Size, in.	Radial Scaling Load, 1b/in., Using S <sub>y</sub> = 40,000 Psi	Radial Sealing Load, 1b/in., Using S <sub>y</sub> = 50, 000 Psi		
1/8	948.9	1187.3		
1/4	913.9	1142.3		
3/8	912.2	1139.5		
1/2	805.9	1007.6		
3/4	728.8	901.5		
1	700.0	875.0		

TABLE 60.RADIAL SEALING LOADS CALCULATED FOR SPECIFICATION<br/>STAINLESS STEEL SEALS WITH REVISED CALCULATION<br/>PROCEDURE

With the need for an increase in radial sealing load further substantiated, the question arose concerning the capability of the connector structures to sustain the loads resulting from stronger seals. The axial load required of the nut would be increased, and the radial load on the lip of the connector flanges would be increased. Work was then undertaken to determine the limits of these factors.

<u>Axial Seal-Seating-Load Calculations</u>. The following formula was used to calculate the axial loads expected for the minimum and maximum torques listed in the specifications:

$$T = 0.2 dF_1$$
, (29)

where

T = torque, in-lb

d = thread diameter, in.

 $F_1$  = axial force, lb.

Table 61 shows the minimum torques and resulting axial forces, and the total axial seal-seating loads as defined on two bases: (1) experimental values from Table 58, and (2) calculated values based on Table 60, using a ratio of axial load to radial load of 1 to 2.

 TABLE 61.
 AXIAL LOADS FOR SPECIFICATION TORQUES AND AXIAL SEAL-SEATING

 LOADS FOR STAINLESS STEEL SPECIFICATION SEALS

Tubing Size, in.	Min Torque, in-1b	Min Axial Force, 1b	Exp. Seal Seating Force, 1b	Calc. Seal Seating Force, Sy = 40,000 Psi, Ib	Calc. Seal Seating Force, Sy = 50,000 Psi, 1b
1/8	75	666	665	520.0	650.0
1/4	200	1600	950	650.0	808.0
3/8	380	2533	1215	810.0	1000.0
1/2	490	2800	1388	840.0	1110.0
3/4	1240	5510	1595	1010.0	1260.0
1	2300	8764	2030	1200 0	1510.0

From the information in Table 61 it appeared that the available axial loads would permit significant strength increases in 3/8-inch and larger seals. The 1/8-inch seal appeared to be very marginal, and the 1/4-inch seal did not provide much margin if the possible variations in seal loading were considered.

Flange Strength Limitations. An increased diametral requirement is probably the biggest problem to be overcome in designing a lightweight threaded connector with a radial seal. Thus, the lips on the connector flanges were made as thin as possible. On the other hand, the stress calculations for the flange lips were very approximate, and the effect of simplification in the calculations was difficult to estimate. When the specification connectors were designed, a minimum lip thickness of 0,065 inch was selected. If the OD of the seal for a given connector size was such that extra space was available with the use of a standard thread size on the nut, then the lip thickness was increased. Table 62 shows the nominal flange lip thicknesses for the plain flanges of the specification connectors.

Tubing Size, in.	Plain Flange Lip Thickness, in.
1/8	0.075
1/4	0,065
3 / 8	0.065
1/2	0,070
3/4	0.079
1	0.065

# TABLE 62. PLAIN FLANGE LIP THICKNESS FOR STAINLESS STEEL SPECIFICATION CONNECTORS

From Table 62 it can be seen that the 1/4-, 3/8-, and 1-inch plain flanges all had the minimum allowable thickness. Because of its increased diameter, it was concluded that the 1-inch plain flange was the weakest component. Experiments were conducted with the 1-inch plain flange to obtain an experimental approximation of its radial load capacity.

A 1-inch seal was machined from A286 (tensile  $S_y = 102,000 \text{ psi}$ ), and the seal was partially seated in a 1-inch plain flange. A diametral expansion of 0.0002 inch was measured with an axial load of 3500 pounds. At 4500 pounds, an additional expansion of 0.001 inch was measured. The test was stopped, the seal tang was reduced 50 percent, and the seal was completely seated with an axial load of 5350 pounds. At this load, the lip diameter had increased 0.002 inch and some flange yielding was indicated. It was concluded that the 1-inch plain flange could sustain an axial seal seating load of 4000 pounds without damage, and an occasional load of approximately 5000 pounds. This conclusion was partially verified later when a 1-inch seal made of 19-9 DL was seated with a maximum axial load of 4000 pounds without yielding of the flange lip. The ID of the plain flange lip (the OD of the seated seal) is 1.1 inches. If a ratio of axial-to-radial force of 1 to 2 is selected, the tests indicated that the 1-inch plain flange could withstand a radial sealing force of about 2300 lb/in. With an occasional force of about 2900 lb/in. It then appeared that the plain flanges as dimensioned by the specifications would tolerate a significant increase in the radial sealing load.

Load Testing of 3/8-Inch SAT Seals. Because it appeared that the 3/8-inch seal might be found to be satisfactory, load tests were conducted with three 3/8-inch SAI seals. The disks of the first specimen were seated one at a time, while the disks of the second and third specimens were seated during one assembly. For the second type of assembly, one disk was seated in a load ring, and one disk was seated in a flange. Table 63 shows calculated and experimental loads for the 3/8-inch seals. Figure 99 shows the axial load trace obtained from the tensile machine, and associated strain readings obtained from the load ring. An examination of the sealing surfaces showed uniformly shiny, coined sealing surfaces about 0.015 inch wide.

Specimen(a)	Maximum Axial Load, 1b	Maximum L Seal Circumf Axial(b)	oad per In. of Ference, 1b/in <u>.</u> Radial	Calculated Radial Load, lb/in., Sy = 50, 000 Psi	Radial Load After Seating of Second Disk, lb/in.
1	1235	705	1410	1139.5	
-	1500	856	1460		8°
2	1200	685	1150	1139.5	1150
	1285	735	(c)		
3	1400	300	1440	1139.5	1230
	1730	990	(c)		

# TABLE 63.CALCULATED AND EXPERIMENTAL LOADS FOR 3/8-INCH<br/>STAINLESS STEEL SPECIFICATION SEALS

(a) Disks of Specimen 1 seated one at a time, disks of Specimens 2 and 3 seated during one loading.

(b) Based on ID of flange lip = 0.564 in.

(c) Seal seated in one load ring and one flange.

Three tentative conclusions were reached from these tests:

- The revised calculation was applicable to 3/8-inch seals as well as 3/4-inch seals, and was probably accurate for all sizes of seals for threaded connectors.
- (2) A radial sealing load of 1200-1500 lb/in. of seal circumference appeared to be a good minimum value for threadedconnectors
- (3) The appearance of two pronounced humps on the axial load curve actually indicates more, rather than less, disk interaction as had been thought.



FIGURE 99. AXIAL LOAD TRACE AND RADIAL LOAD INCREMENTS FOR 5/8-INCH STAINLESS STEEL SAI SEALS (SPECIMENS NO. 2 AND 3)

Investigation of Plating Quality. Tests were then conducted to determine the effect of increased radial sealing loads on 3/4 and 1-inch SAI seals. Specimens were forwarded to Battelle-Columbus from the Rocket Propulsion Laboratory. When these appeared to have rougher plating surfaces than expected, additional specimens in most sizes were obtained and examined. In all, approximately 30 SAI seals were examined with a 10-power glass. Typical types of variations included pits, plating nodules, grainy surfaces, edge nicks, scratches, inclusions, and machining marks under the plating.

Although it was believed at first that the quality of the plating should be improved, a discussion with SAI personnel of the production plating of the seals indicated that the plating quality was good as defined by the present state of the art. Although various steps could be taken to improve the plating, these would result in a substantial increase in plating costs. Therefore, it appeared desirable to increase the radial sealing load sufficiently to make the current plating quality acceptable. A similar conclusion was reached at the Rocket Propulsion Laboratory after an examination of the plating. In any case it was clear that the range of radial sealing loads that could occur within the specification for the 3/4- and 1-inch seals would result in a significant percentage of seals with excessive leakage if the nickel plating were of the current quality.

Increased Sealing Loads for Current Plating Quality. The possibility of obtaining reliable sealing for seals with current plating quality was investigated by assembling a number of 3/4- and 1-inch SAI seals with insert rings in the tangs to provide additional radial sealing loads. The disks were able to sustain radial sealing loads of about 1700 lb/in, before their columnar strength was exceeded. Approximately 20 seals were assembled with varying radial sealing loads. On the basis of detailed examinations of the sealing surfaces before and after seating, it was concluded that an average radial sealing load of 1500 lb/in. would give good sealing with the current plating. Starting at a radial sealing load of about 1200 lb/in., a shiny coined sealing surface was formed that was about 0.015 inch wide. At a load of 1500 lb/in., a similar surface was formed which was about 0.020 inch wide. In both cases, many of the plating variations were coined out. When edge nicks or pits occurred, the coined surface was sufficiently wide to create a seal around the discontinuity.

During a detailed review of this work at the Rocket Propulsion Laboratory, it was mutually agreed that attempts would be made to develop seals with radial sealing loads ranging from 1200 lb/in. to 1700 lb/in. Since the 3/8-inch and smaller seals already provided these loads, attention was to be directed to the 1/2-, 3/4-, and 1-inch seals.

## Candidate Materials and Configurations

Over a period of several weeks at the end of Contract AF 04(611)-11204, personnel at the Rocket Propulsion Laboratory and Battelle-Columbus worked closely together in developing and evaluating candidate materials and configurations.

Two basic approaches were visualized for developing the necessary increase in radial seal loading: (1) the use of a stronger material for the entire seal or (2) the use of an insert ring of strong material in the tang of an austenitic stainless steel seal. Called the one-metal and two-metal seal approaches, each had advantages and disadvantages.

The one-metal seal approach offered lower production costs and better production control. However, fewer materials were available for consideration, and the performance of a stronger seal structure against the softer material of the flange: was difficult to envision.

The two-metal seal approach offered higher production costs and poorer production control. On the other hand, several materials appeared to be candidates for the strengthening element, and the performance of the seal disks against the flanges was thought to be more predictable.

Candidate Materials for One-Metal Seals. Candidate material properties for a satisfactory one-metal seal could be closely specified. The compressive yield strength had to be from 50 percent to 100 percent more than for the SAI seals (which was about 50,000 psi). The coefficient of thermal expansion had to be about  $9.5 \times 10^{-6}$  in./in./F to match the austenitic stainless steel flanges. The material had to be compatible with candidate rocket propulsion fluids. It also had to be available in bar stock, cost about \$1.00 per pound, and have a machining index of approximately 50.

All relatively common stainless steels were considered. The martensitic and ferritic stainless steels were not acceptable because of their low coefficient of thermal expansion (approximately  $5.5 \times 10^{-6}$  in./in./F). The age-hardenable stainless steels had a low coefficient of thermal expansion and were generally too strong. The nickel-base superalloys were too strong and were not corrosion resistant to N<sub>2</sub>H<sub>4</sub> and fluorine. A286 was shown to be too strong (see page 178) and was not compatible with N<sub>2</sub>H<sub>4</sub> and fluorine. A cobalt-base superalloy, V-36, appeared to have attractive properties, but it was not generally available and little corrosion data could be found.

The most promising materials for a one-metal seal were: (1) cold-worked Type 304 stainless steel, (2) 19-9DL stainless steel, and (3) a new stainless steel by Armco, called 21-6-9.

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Type 304 stainless steel is capable of being cold worked to very high yield strengths. The yield strength of full-hard tubing was found to range from 115,000 to 150,000 psi. Cold-worked bars were available in limited quantities and by special order. Because the general properties of cold-worked Type 304 appeared to be ideal for use in the austenitic stainless steel flanges, the major problems seemed to be associated with the availability of the material in the required yield strength, and the effect of the residual stresses on the machining of the thin seal disks. The further investigation of this material is discussed below.

The second candidate, 19-9DL, was an iron-base superalloy. It had an ideal yield strength (70,000 to 85,000 psi), was readily available, and cost about \$1.00 per pound. It had a coefficient of thermal expansion of  $10 \times 10^{-6}$  in./in./F, and its machining index was comparable to austenitic stainless steel. Although an examination of the composition of 19-9DL indicated that it should be more corrosion resistant than A286 (another iron-base superalloy), little data existed for exposure to rocket propulsion fluids. The lack of corrosion data appeared to be the biggest problem with 19-9DL.

The third candidate, 21-6-9, was very similar to austenitic stainless steels. The coefficient of thermal expansion, cost, and machining index were favorable. Although its corrosion resistance to rocket propulsion fluids was not known, its composition indicated that it would be similar to Type 316 stainless steel. The biggest problem with 21-6-9 was the fact that the annealed tensile yield strength was not very high (50,000 to 60,000 psi). While this was some improvement over annealed austenitic stainless steel, it was not as much as desired. Another problem was the fact that there was only one supplier, and 21-6-9 was not readily available in the bar sizes of interest. Although 21-6-9 was able to be cold worked to achieve higher yield strengths, such material was much less available than cold-worked Type 304 stainless steel.

Candidate Materials for Two-Metal Seals. The two-metal seal configuration was envisioned as consisting of an austenitic stainless steel Bobbin seal with some percentage of its tang replaced by a relatively high-strength stainless steel. The percentage to be replaced would depend on the strength of the insert ring, although it was apparent that some of the original tang material had to remain to join the seal disks during machining and to insure proper seating of the seal disks.

Most of the materials discussed for the one-metal seal were also considered for the two-metal seal. Particular attention was given to A286 and Inconel 718. However, because of the corrosion problems with these materials, the final candidate material was hard-drawn and tempered Type 304 stainless steel tubing. Although this material was not readily available in the tubing size and wall thickness desired, there appeared to be several companies that would be willing to supply it in limited quantities for about \$1.50 per pound. When used with an annealed Type 304 stainless steel structure, the combination appeared to be ideal. However, as described below, preliminary tests simulating such a configuration produced unanswered questions about the performance of the two-metal seal.

Limited Tests With One-Metal Seals. Limited tests were conducted with onemetal seals to assist in the formulation of recommendations for follow-on effort. Small quantities of each candidate material were obtained and load tests were conducted with seals machined to the specification dimensions.

A 1-inch bar of 21-6-9 was obtained readily as a sample from Armco (it was not possible to obtain a slightly larger size bar for the 1-inch seals during 6 weeks of requests). The sample had a tensile yield strength of 57,500 psi, which was said to be about average. Fur 3/8-inch and three 3/4-inch seals were machined from this bar and load tested. The test results are shown in Table 64, as is a calculated radial seal-ing load based on an assumed compressive yield strength of 65,000 psi. Although the first two 3/4-inch seals were seated before a 3/4-inch load ring was fabricated, the measured axial loads for all seals and the measured radial sealing loads for Specimen 3 were about as anticipated. It was not possible to explain the similarity of the 3/8-inch, 21-6-9 results shown in Table 64 with those obtained with 3/8-inch seals made of annealed Type 310 stainless steel (see Table 63). An examination of the disk surfaces of the 3/8-inch, 21-6-9 seals showed significant areas of crazing, which indicated a greater surface compressive yielding than had been observed on other seals. It was concluded that the material was promising, but that future tests should determine whether 21-6-9 was weaker in compressive yield than expected.

Tubing Size,	, <u>, , , , , , , , , , , , , , , , , , </u>	Maximum Axial	Max Load p Circumfer	er In. of Seal ence, lb/in.	Calculated Radial Load, lb/in.,
in.	Specimen	Load, lb	Axial	Radial	Sy = 65,000 Psi
3/8	1	1440	822	1420	1485
3/8	2	1230	702	1235	1485
3/8	3	1420	810	1320	1485
3/8	4	1525	870	1600	1485
3/4	1	2400	860		1170
3/4	2	2400	860		1170
3/4	3	2100	752	1240	1170

TABLE 64.	LOAD-TEST RESULTS FOR 3/8- AND 3/4-INCH
	SEALS MACHINED FROM 21-6-9

A 1-1/4-inch bar of 19-9DL was readily obtained. An accompanying specification listed a tensile yield strength of 84,000 psi. Several sources listed a minimum tensile yield of 70,000 psi, but the specimen that was obtained was apparently typical. One 3/4inch seal and one 1-inch seal were machined from this bar. Table 65 shows the results obtained from these seals and the calculated radial sealing load based on an assumed compressive yield strength of 95,000 psi. The load results were about as anticipated and the seal deformation appeared to be normal. The 1-inch plain flange did not yield with this assembly, substantiating the earlier tests (see page 178) with an A286 seal. Although the edges of the seal disks did not flatten out but indented slightly into the softer flange material, neither seal was more difficult to remove than the SAI seals. This indicated that flange indentation may not be a problem with a stronger seal. However, it also appeared that a reverse angle may have to be machined on the seal disks if a satisfactorily wide sealing surface is to be obtained. (This procedure worked well for the flanged-connector seals.)

Tubing Size,	Seal	Maximum Axial Load,	Max Load p Circumfer	er In, of Seal ence, lb/in.	Calculated Radial Load, lb/in.,	
in.	Material	1b	Axial	Radial	S <sub>y</sub> = 95,000 Psi	
3/4	19-9DL	2900	1040	2120	1715	
1	19-9DL	4000	1160	1530	1660	
3/4	304	2700	968	2030	1715	

# TABLE 65.LOAD-TEST RESULTS FOR 3/4- AND 1-INCH SEALSMACHINED FROM 19-9DL AND ANNEALED ANDCOLD-DRAWN TYPE 304 STAINLESS STEEL

A 1-inch bar of annealed and cold-drawn Type 304 stainless steel was found and purchased. Subsequently, an additional 1-inch bar and 1-1/4-inch bar were located but not purchased. These bars had specified tensile yield strengths of 90,000, 89,000, and 84,000 psi, respectively. Through discussions with several steel manufacturers, it was determined that, although annealed and cold-drawn Type 304 bars are usually purchased according to a minimum yield strength, it would be practical to specify a minimum and a maximum yield strength, provided the range was sufficiently broad. On the basis of these discussions, values of 70,000 and 90,000 nsi would be acceptable. Ears could be obtained to this specification at any time from a mill if 1000 pounds were ordered. The price would be approximately \$0.90 per pound. If smaller quantities were desired, calls to specialty suppliers would have to be made, similar to those made by Battelle-Columbus. While additional effort would be needed to determine accurately the most practical methods for specifying and ordering cold-worked Type 304, it appeared to be practical to obtain the material with the proper yield strength.

There had been concern that a cold-worked bar would have wide stress variations. Table 66 shows hardness readings taken from the Type 304 bar that was purchased. It was concluded that the uniformity was quite good for the purposes of the Bobbin seal.

Location on 11-Ft Bar	Surface of Bar	Center of Bar	One-Half Radius
One End	106	94	98
Middle	104	91	98
Other End	105	93	98

#### TABLE 66. ROCKWELL B HARDNESS READINGS TAKEN FROM 90,000-PSI-YIELD, COLD-WORKED TYPE 304 STAINLESS STEEL

One 3/4-inch seal was machined from this bar. The machining was judged to be somewhat easier than for annealed Type 304 stainless steel. There was no noticeable effect from residual stresses on the seal dimensions. The results from the load test of this seal are shown in Table 65. Some concern was felt about the ductility of the coldworked material. However, the accompanying specificiation showed 37 percent elongation, and the deformation of the seal appeared to be normal. The same conditions noted for the edge of the 19-9DL seal disks were also noted for the Type 304 seal disks.

Limited Tests With Two-Metal Seals. A number of load tests were made with SAI seals equipped with insert rings as a means of increasing the radial sealing loads on the plating. On the basis of this work, it appeared relatively easy to select a highstrength material which could be used to replace a portion of the seal tang and obtain the same increased loads. Although detailed consideration was given to A286 and Inconel 718, the only material that was finally acceptable was cold-drawn and tempered tubing made of Type 304 stainless steel. The Superior Tube Company listed a yield-strength range of 115,000 to 150,000 psi for this material. Although this was a wide yieldstrength range, tests with the SAI seals had shown that good disk deflection could be obtained even when the tang did not yield. Thus, it was hypothesized that if the weakest tubing did not permit the tang to yield, the behavior of the seal with this material would be the same as it would be with the highest yield tubing. The decision was made to simulate the 115,000-psi-yield tubing by using 99,000-psi-yield A286. Insert rings were designed to provide comparable strength, and one 3/4-inch and one 1-inch SAI seal were modified accordingly.

The load test with the 1-inch seal gave the expected results. Although the tang yielded slightly, a radial load of 1490 lb/in. was measured and the sealing surfaces were comparable with those obtained with previous seals in which the tang did not yield. The 3/4-inch seal, however, was a disappointment. The load curve obtained on the tensile machine did not appear normal, and the radial load was only 1180 lb/in. This occurred despite the fact that the tang yielded less than it did in the 1-inch seal. When a second 3/4-inch seal gave duplicate results, it was concluded that an unknown factor was occurring in the deformation of the 3/4-inch seal.

#### **Recommended Future Activities**

Because of the promise of the one-metal seal approach, and because of the apparent deformation problem with the two-metal seal, it was recommended that future activities be concentrated on developing a satisfactory one-metal seal.

At the conclusion of the program, it appeared that the 21-6-9 should be selected for the 1/2-inch seal and that annealed and cold-drawn Type 304 should be selected for the 3/4- and 1-inch seals. However, there were still several areas about these materials and about 19-9DL that needed further understanding. Because the development of these areas might change the evaluations, it was recommended that a program be initiated in which additional material specimens would be purchased, a number of seals of each material and size would be machined and plated, and a number of load tests and selected performance tests would be conducted. Corrosion tests of the 19-9DL and 21-6-9 should also be included.

On the basis of this preliminary work, the best materials and seal designs should be selected for the 1/2-, 3/4-, and 1-inch seals and a comprehensive series of load

tests and performance tests should be conducted to demonstrate the adequacy of the design. Following this effort, the specifications should be revised accordingly.

### Stress Relaxation

A previous section of the report has discussed theoretical and experimental eftorts to determine the effect of stress relaxation in flanged connectors. The objective of this work was to demonstrate that flanged connectors would perform satisfactorily with a system storage life of 5 years. Similar, though less extensive, work was conducted in relation to threaded connectors.

It was estimated that the high-pressure (4000 psi) stainless steel threaded connectors would be more likely to show the effects of stress relaxation than the lowpressure (1500 psi) aluminum connectors. Further, it was believed that the 3/4-inch connectors would be representative of all the stainless steel threaded connectors. Thus, typical assembly modes were visualized for a number of 3/4-inch connectors, and test conditions were developed to simulate various missile storage applications.

It was expected that stress-relaxation might affect the performance of the threaded connectors in two ways: (1) by reducing the axial force applied by the nut and (2) by reducing the radial force at the sealing surfaces. If the axial force applied by the nut were reduced (by relaxation of the nut or of the connector flanges), combined pressure and structural loads on the connector might result in insufficient residual axial load on the seal, causing excessive leakage. Relaxation in the seal as a result of high compressive loading, or relaxation in the lips of the connector flanges as a result of high tensile and bending stresses, might reduce the radial load at the sealing surfaces sufficiently to cause leakage. Consideration of these failure modes led to the decision to use strain gages to measure nut relaxation, and helium leakage to measure excessive reduction in radial sealing loads.

#### Test Specimens

Table 67 shows the assembly conditions for the 22 test connectors used for the investigation of the effects of stress relaxation. Each connector was fabricated according to the dimensions in the specifications. Because stress relaxation increases rapidly with stress, it was expected that the amount of initial nut tension would significantly influence the amount of stress relaxation occurring in the nut. The torque levels shown in Table 67, and the different assembly methods were selected to produce the levels of initial nut tension expected in practice.

The nut of each connector is designed as a spring to maintain the proper axial load on the connector despite connector thermal gradients caused by hot or cold fluids. Thus the measurement of strain in the nut was an ideal means of determining the effect of stress relaxation on the axial load produced by the nut. The nut consists of three major sections: (1) the threaded portion, (2) the inward projecting nut ring, and (3) the relatively thin nut hub which connects the ring and the threaded portion. While the ring and the threaded portion contain large hoop strains, the nut hub is subjected for the most part to axial strains, and the strain gages were located on the nut hub.

Specimen	Asservibly Condition	Assembly Method	Nut Load, Ib	Helium Leak Rate on Assembly, atm cc/ sec per inch of seal circumference
1 <sup>(a)</sup>	Typical preload	Open-end wrench	5075	$2.2 \times 10^{-8}$
2(a)	Typical preload	Open-end wrench	4830	$0.5 \times 10^{-8}$
3(a)	Typical preload	Open-end wrench	<b>44</b> 00	(c)
4	Min torque, 100 ft-lb	Open-end torque wrench	5100	(c)
5	Min torque, 100 ft-lb	Open-end torque wrench	7450	$3.2 \times 10^{-9}$
6	Min torque, 100 ft-lb	Open-and torque wrench	3230	4.9 x 10 <sup>-7(d)</sup>
7(a)	Typical preload	Open-end wrench	<b>463</b> 0	$1.4 \times 10^{-8}$
8(a)	Typical preload	Open-end wrench	<b>437</b> 0	0.48 x 10 <sup>-9</sup>
9(a)	Typical preload	Open-end wrench	7800	$2.2 \times 10^{-8}$
10	Max torque, 116 ft-lb	Loc Rite torque wrench	7080	0.66 x 10 <sup>-9</sup>
11	Max torque, 116 ft-lb	Loc Rite torque wrench	8830	(c)
12	Max torque, 116 ft-lb	Open-and torque wrench	9050	(c)
13	Max torque, 116 ft-lb	Open-end torque wrench	10380	2.6 x $10^{-9}$
14	Max torque, 116 ft-lb	Open-end torque wrench	7480	(c)
15	Max torque, 116 ft-lb	Open-end torque wrench	7860	(c)
16	Max torque, 116 ft-lb	Open-end torque wrench	5370	$1.2 \times 10^{-8}$
17	Max torque, 116 ft-lb	Open-end torque wrench	6420	$0.9 \times 10^{-9}$
18	Max torque, 116 ft-lb	Open-end torque wrench	7340	0.69 x 10 <sup>-9</sup>
19	Max torque, 116 ft-lb	Open-end torque wrench	7150	0.48 x 10 <sup>-9</sup>
<sub>20</sub> (b)	Max torque, 116 ft-lb	Open-end torque wrench	7370	1.1 x 10 <sup>-8</sup>
21(b)	Max torque, 116 ft-lb	Open-end torque wrench	7740	(c)
22(b)	Max torque, 116 ft-lb	Open-end torque wrench	<b>993</b> 0	4.7 x $10^{-8}$

# TABLE 67. ASSEMBLY CONDITIONS FOR 3/4-INCH, STAINLESS STEEL CONNECTORS FOR STRESS-RELAXATION TESTS

(a) Nut was tightened to achieve approximate desired strains in nut.

(b) Nut threads were electropolished.

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(c) No leakage could be detected,
(d) Connector was left on test even though measured leak rate was excessive on assembly,

One gage was placed on each of three of the six nut flats midway between the ring and the threaded portion. Although it would have been desirable to place the gages on alternating nut flats, this would have prevented proper tightening of the nut with a wrench. Thus two of the gages were placed on adjacent nut flats.

The gages on each nut were calibrated before the connectors were assembled. This was done by loading each nut in a tensile machine and reading each gage at loadlevel increments. This method was used to circumvent the effects on the measured strains of triaxial stresses in the nut and minor variations in strain-gage locations. Table 68 shows the increment of strain for each gage at the major tensile loads.

#### **Test Conditions**

Three types of tests were selected to represent various types of storage systems: (1) static, (2) static with constant, maximum system pressure and end loading, and (3) static with periodic stress-reversal-bending tests. These tests also represented increasing probability of connector leakage.

Each connector for the static relaxation test was equipped with a flange at one end and a welded closure at the other. The flanged end was mounted to a manifold that was connected to a source of 4000-psi helium. When the leakage of the connectors was checked periodically, the manifold was pressurized, and an O-ring sealed vacuum chamber was slipped over each connector. A helium mass spectrometer connected to the vacuum chamber was used to measure the leakage of the connector.

For the static, full-load tests, each connector was mounted to a manifold similar to the static test connectors. In addition, the closed end of each connector was attached to a hydraulic cylinder. For the duration of the test, the manifold and the connector were kept pressurized by 4000-psi helium, and the hydraulic cylinders were kept pressurized to exert a constant, maximum axial structural load on the connectors. For periodic leakage measurement, the hydraulic cylinders were disconnected and a vacuum chamber was slipped over each connector and helium leakage was measured with a mass spectrometer.

Each connector subjected to the dynamic tests was assembled as part of a cantilever beam to fit the stress-reversal-bending equipment used during Contract AF 04(611,-9578. The connectors were maintained in a static condition without load until the periodic checking period. At that time, each connector was subjected to 200,000 cycles of stress-reversal bending and each connector was leak-checked with the helium mass spectrometer.

Leak checks and dynamic tests were conducted at 1, 5, 9, and 24-month intervals. Also at these time intervals, strain-gage readings were made with no load, with internal pressure, and with internal pressure plus axial tension loading.

Nut	Gage	<u>Nut</u>	Tensile 1 2000	oad, 15, 3000	Strain, 4000	Microin,/i 5000	n	Strain for 6000- Lb Tenuile Load
1	а	180	170	170	180	190	200	1090
	b	165	205	205	205	195	210	1185
	с	210	220	220	225	223	200	1300
2	a	220	180	205	230	230	235	1300
	b C	205	175	1/5	190	205	205	1090
rt.	•	205	2.24	336	170	205	202	1200
Ľ.	a b	205	235	235	230	235	240	1380
	c	243	220	215	210	220	235	1345
4		195	230	225	235	220	225	1330
	ъ	240	190	205	215	210	230	1290
	с	260	235	230	240	240	250	1455
5	8	270	235	225	230	225	225	1410
	ь	235	240	225	225	220	220	1365
	c	230	205	205	210	215	220	1285
6	a	260	270	245	245	245	245	1510
	ь	205	240	240	235	245	240	1405
	¢	250	220	220	220	230	220	1360
7	a	160	165	190	210	210	210	1145
	ь	230	215	220	225	230	230	1350
	Ľ	290	123	~~;	220	230	230	1320
8	8 5	275	210	200	190	195	190	1210
	c	240	200	190	200	200	210	1240
	-	205	225	330	220	220	225	1335
,	a b	225	190	205	220	220	230	1290
	c	235	225	230	235	235	245	1405
10		180	170	200	200	205	210	1165
	ъ	300	230	230	220	220	230	1430
	с	250	240	220	215	215	220	1365
11	a	180	190	195	200	205	200	1170
	ь	170	180	185	200	210	215	1160
	c	200	200	205	210	220	235	1270
12	a	245	255	230	235	215	220	1400
	ь	220	220	210	210	210	220	1290
	C	100				205		1155
13	a 5	180	160	165	180	195	190	1070
	ç	225	205	215	215	220	220	1300
14		235	260	240	220	230	240	1435
	b	185	165	180	195	200	205	1130
	c	180	150	180	185	195	210	1100
15	4	150	150	155	185	190	190	1020
	ь	195	195	200	200	210	205	1205
	c	215	220	225	200	215	210	1285
16		195	210	210	210	205	215	1245
	ъ	240	185	200	200	200	195	1220
	c	240	215	220	225	225	230	1355
17		230	230	210	200	215	220	1305
	ь	240	210	200	190	190	190	1220
	c	180	212	193	190	200	200	1170
18		235	200	195	190	195	195	1210
	D C	220	190	200	205	220	220	1223
10	-		220	210	70%	216	210	1255
19	ĥ	205	205	210	205	215	210	1235
	c c	245	220	215	220	220	225	1345
20		205	185	190	205	205	210	1200
	ь	180	215	215	230	230	230	1300
	c	250	225	215	220	215	220	1345
21		215	220	210	210	205	210	1270
	Ь	230	210	215	220	215	225	1315
	c	285	220	220	220	230	230	1405
22	٥	155	195	185	190	195	195	1115
	ъ	195	185	180	205	190	195	1150
	C	195	200	200	220	420	445	1200

TABLE 08. INCREMENT OF STRAIN FOR THE NUT STRAIN GAGES AT THE MAJOR TENSILE LOADS, 3/4-INCH STAINLESS STEEL CONNECTORS

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#### Test Results

The tubing of one connector broke during the last stress-reversal-bending "exercise" and it was not possible to leak-check this connector. However, none of the remaining 21 connectors showed an increase in helium leakage during any test above the values measured on initial assembly. In fact, in most cases it was not possible to detect any leakage.

After the strain-gage readings were taken at the 24-month interval, an examination of the readings showed variations which were not explainable by the action of the fitting structure. A discussion of this problem with a Battelle-Columbus specialist led to the conclusion that changes in the resistance of the lead wires during the 2-year period had introduced excessive reading variations. Since the total change in nut tension was the measurement of primary interest, it was decided that the purposes of the test would still be met if strain-gage readings were taken before and after the disassembly of each connector. These strains could then be compared with the strains developed on assembly to determine the relaxation in each nut.

Table 69 shows the tests conducted with each connector, the strains measured by each gage on assembly and on disassembly, and the estimated initial and final axial loads on each nut. Of the 22 nuts involved, six showed an increase in average strain. This resulted when a strain gage that read low on assembly indicated an increasingly higher value during the 2 years, closer to that read with the other two gages. Of the remaining 16 connectors, the average strains for three connectors decreased less than 10 percent, seven decreased less than 20 percent, and four decreased less than 35 percent. Of the remaining two connectors, the average strain for one decreased about 45 percent, and that for the other decreased about 70 percent. In general it was found that the nuts with the highest initial tensile load showed the largest reductions in strain. If the initial tension was above the required minimum of 5690 pounds, the residual tension was within 20 percent of the required minimum. The one exception was the connector that experienced a 70 percent reduction in tension to give a residual tension of 2580 pounds. These results, combined with the fact that no connectors exhibited increased leakage during the 2-year test period, indicated that stress relaxation is not a source of leakage in threaded connectors during 2 years of storage. Since the rate of stress relaxation reduces on a log-log basis with a reduction in stress, it is believed that the successful 2-year-storage tests indicate that the connectors will be satisfactory for a 5-year-storage period.

Connector	Type of Test	Nut Strain, µin./in., 6000-Lb Load	Nut Strains on				Let.	Nut Strains on				Eo.	Load
				5	<u>c</u>	Av.	Load, 10		b	c .	<u>n./in.</u>	Lond 1b.	1b
1	Static, Loaded	1192	1050	1085	890	1008	5075	1045	1025	890	987	4960	-113
ž	Static, Loaded	1190	530	1160	1185	958	4830	<b>95</b> 0	1090	1085	1042	5250	+420
3	Static	1345	635	1145	1175	985	4400	1040	1145	1110	1098	4900	+500
	Stetic, Loaded	1358	830	1730	900	1153	5100	825	1360	865	1017	4500	-600
ŝ	Static, Loaded	1353	1410	2255	1360	1682	7450	1280	1580	1245	1368	6060	-1390
6	Static	1425	710	1075	515	767	3230	720	815	660	732	3080	-150
,	Static, Loaded	1272	565	1280	1105	983	4630	1005	1215	1050	1090	5140	+510
i i	Static, Loaded	1173	450	1040	1075	855	4370	825	955	1035	938	4790	+420
ů.	Static	1343	1210	2030	2000	1747	7800	1630	1935	1770	1778	7930	+130
10	Static. Loaded	1320	1505	1490	1685	1560	7080	1340	1485	1500	1442	6550	-530
ii	Static Loaded	1200	1565	2025	1705	1765	8830	1280	1830	1475	1528	7640	-1190
12	Static	1282	1485	2690	1620	1932	9050	1275	1680	1025	1323	6190	-2860
	Static Loaded	1205	1590	2640	1830	2020	10380	1210	2310	1625	1715	8820	-1560
16	Static, Londed	1222	1155	2355	1065	1525	7480	1040	1330	955	1108	5440	-2040
15	Static	1170	1055	2330	1210	1532	7860	875	1050	850	925	4740	-3120
14	Dynamic	1270	940	1400	1075	1138	5370	800	1120	945	955	4520	-850
	Denamic	1232	1005	1650	1 100	1318	6420	1165	1893	1115	1392	6780	+360
	Dynamic	1215	1250	2150	1130	1510	7340	745	385	460	\$30	2580	-4760
10	Dechamic	1288	1650	1590	1360	1533	7150	250	1580	1430	1253	5840	-1310
20	Static Loaded	1282	1 140	2085	1 300	1575	7370	960	1845	1050	1285	6020	-1350
20	Static Londed	1330	1200	2640	1110	1717	7740	1085	1640	1145	1290	5820	-1920
22	Static	11/5	1420	2915	1500	1945	9930	1300	1895	1355	1517	7740	-2190

TABLE 69. MEASURED STRAINS AND AXIAL MUT LOADS FOR 3/4-INCH STAINLESS STEEL CONNECTORS BEFORE AND AFTER 24 MONTHS STRESS RELAXATION TESTS

# CONCLUSIONS 'ND RECOMMENDATIONS

Many conclusions and recommendations have been given in different parts of the report. The major items are repeated here in three categories: (1) aluminum flanged connectors, (2) stainless steel flanged connectors, and (3) threaded-connector support.

#### Conclusions

#### Aluminum Flanged Connectors

- (1) An effective, computerized procedure has been developed for designing AFRPL flanged connectors for aluminum tubing systems.
- (2) Satisfactory qualification tests have been conducted for 100-psi, 200-psi, and 500-psi connectors through 16 inches in diameter, and for 1000-psi and 1500-psi connectors through 3 inches in diameter.
- (3) Additional qualification tests are required for selected performance aspects of 1000-psi and 1500-psi connectors through 16 inches in diameter.
- (4) Additional test data are required concerning the maximum stress level that can be permitted for all 3-inch connectors in a vibration mode.

#### Stainless Steel Flanged Connectors

- (1) An effective, computerized procedure has been developed for designing AFRPL flanged connectors for stainless steel tubing systems.
- (2) Satisfactory qualification tests have been conducted for pressures of 6000 psi for sizes through 3 inches in diameter, and for pressures of 100 psi for sizes through 8 inches in diameter.
- (3) Additional qualification tests are required for selected performance aspects for pressures through 1500 psi and for sizes through 16 inches in diameter.

#### Threaded-Connector Support

- (1) Stress relaxation is not expected to cause leakage in AFRPL threaded connectors.
- (2) The seal retention and misalignment limitation concepts developed and evaluated during the program are not practical for the purposes of the AFRPL connector.

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(3) The radial scaling load should be increased for the scals for the 1/2-, 3/4-, and 1-inch stainless steel connectors. Promising approaches have been determined.

#### Recommendations

# Aluminum Flanged Connectors

- It is recommended that specifications be prepared for 100-psi, 200-psi, and 500-psi connectors through 16 inches in diameter, and for 1000-psi and 1500-psi connectors through 3 inches in diameter.
- (2) It is recommended that additional qualification tests be conducted to investigate the following performance features of the 1000-psi and 1500-psi connectors through 16 inches in diameter: (a) deformation characteristics of the seal structure and (b) resistance of the connectors to pressure-impulse cycles.
- (3) It is recommended that additional vibration tests be conducted with 3-inch connectors to determine an appropriate stress level for a minimum life of 200, 000 cycles.

# Stainless Steel Flanged Connectors

- (1) It is recommended that specifications be prepared for pressures through 6000 psi for connector sizes through 3 inches in diameter, and for pressures of 100 psi for sizes through 8 inches in diameter.
- (2) It is recommended that additional qualification tests be conducted to investigate the following performance characteristics for connectors for pressures through 1500 psi, and for sizes through 16 inches: (a) initial sealing reliability, and (b) resistance to thermal gradients.
- (3) It is recommended that dimensions be developed for the remaining connectors listed on page 152.

# Threaded Connector Support

- (1) It is recommended that a seal retention concept be investigated on the basis of the preseating of the seal in one of the connector flanges prior to assembly of the connector.
- (2) It is recommended that the assembly procedures of the AFRPL connector include the requirement that the parts be sufficiently aligned

and free on assembly to permit the seal disks to be placed completely within the flange cavities and the nut made fingertight.

(3) It is recommended that a program be initiated to select the best designs for stronger 1/2-, 3/4-, and 1-inch stainless steel seals, and that the adequacy of these designs be demonstrated by qualification tests.

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# APPENDIX A

# RELAXATION DESIGN OF SEPARABLE TUBE CONNECTORS

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#### RELAXATION DESIGN OF SEPARABLE TUBE CONNECTORS

#### Ьу

L. M. Cassidy, E. C. Rodabaugh, D. B. Roach, and T. M. Traimer

#### INTRODUCTION

The design of separable tube connectors for operating below the temperature which produces significant creep of the metal used in the connector may be based on conventional elastic analysis methods. For example, the ASME Unified Pressure Vessel Code<sup>[1]</sup> gives a widely accepted method for designing bolted-flanged joints, including an approximation of the required bolt load and an elastic analysis of the flanges to insure adequate flange strength for carrying the required bolt load.

Even at temperatures where crewp does not occur, the design of a conventional separable tube connector is not simply a strength problem, since the usual criterion of failure is leakage and leakage may occur without necessarily overstressing any part of the connector. In practice, conventional connectors are initially loaded (tightened bolts in a boiled-flanged connector, tightened nut is a threa'd connector) so that the elastically stored forces in the connector are sufficient to pievent separation of the connector parts due to the subsequently applied service loads arising from internal pressure or loads on the attached pipe or tubing. Accordingly, not only the stresses but also the strains or displacements are significant in connector design.

At operating temperatures where significant creep occurs, the design method for the connector rau<sup>--</sup> take into account the relaxation of the elastically stored forces which occur as a result of the plastic flow of the metal components. In boltad-flanged joints designed in accordance with the ASME Code<sup>(1)</sup>, this relaxation effect is taken into account in an indirect and approximate manner by the use of allowable stresses which are based on creep or stress-to-rupture properties of the material. These allowable stresses, however, do not necessarily reflect the relaxation characteristics of bolted-flanged joints in general, and use of the method may result in excessively conservative design or inadequate performance over the desired service life.

There have been previous discussions and simplified analyss  $\{2,3,4,5,6,7\}$  of relaxation in bolted joints. Unfortunately, a practical solution considering primary creep does not appear to have been developed. The method of analysis presented in this report is intended as a workable approach to the design of separable tube connectors at temperatures where creep or relaxation occurs.

The only known published ds's on the elevated-temperature testing of separable connectors in order to study leakage is a very comprehensive set of tests<sup>(8,9,10)</sup> by the British Pipe Flanges Research Committee on bolted-flanged joints. The results of these tests indicate a definite relation between the time to leakage and the temperature. However, the results are not amenable to a theoretical analysis because of the lack of sufficient specimen creep or relaxation data on the materials used in the test.

Material properties are, of course, interrelated with the design procedure. Accurate information of material properties (such as yield strength, modulus of elasticity, and creep or relaxation rates) are needed to apply the design procedure. Accordingly, this report includes a discussion of material properties and tabulation of properties for various materials with desirable high-temperature properties. The section of the report on material properties leads to a selection of an optimum material for uses in connector design for aerospace application at 1440 F operating temperature.

#### SUMMARY

Materials were evaluated for use at 1440 F, with the result that René 41 is the material choice for the connector design. René 41 is equivalent in strength to the other candidate materials at 1440 F, and has a greater amount of data available. In addition, René 41 is readily available and has seen considerable research and service experience. Nevertheless, the lack of a suitable family of creep curves at 1440 F led to the recommendation that creep or relaxation data be generated for René 41.

With the expectation that relaxation data will not be available for most materials, the design method is directed primarily toward the utilisation of creep data, with a discussion on the use of relaxation data. The widespread scatter in creep data as well so possible secondary effects, such as those discussed in Appendix C, led to the use of a design factor of safety.

The design procedure utilises the steady-state or power law of creep with an assumed sero time reduction in stress to account for primary creep. This approach is conservative for short times, the degree of conservation depending upon the amount of primary creep exhibited by the material at the design temperature. A comparison is shown for predicting relaxation from creep data for various creep theories.

Both bolisd-flanged and threaded connectors are included in the relaxation design, and sample calculations are provided to illustrate the design method for each type of connector. The effect of temperature differentials and external loads on the leakage pressure is considered. Secondary effects such as stress concentrations, creep bending of bolt, dynamic creep, gasket creep, and flange rotations are discussed briefly, but are not included as an integral part of the design procedure.

The design method is suitable for hand calculations, but could be expedited and easily adapted for solution on a high-speed computer. The optimum (minimum weight) design for a given value of leakage pressure can be determined only after calculating a

Aplumbers is raised paragineer designate Referentes on page 19.

\*\*Acatage pressure refers to die value of uniform linternal pressure of or balow which toberable loads go eases easer, and shmidd not be seeduard with the value of gashes pressure required for (1) initially making the gashes or (2) readonal pressure required on the gashes in order to prevent accessive loads age relea. The determination of these gathest pressions is not servered in this report. wide range of geometries. A suitable sumber of calculations enables the static<sup>90</sup> design to be rated in accordance with various combinations of leakage pressure and time. The detailed gasket design is not considered herein, since Reference (11) is quite comprehensive in this respect.

Various methods of analyses are discussed for the creep design of tubes. The data available indicate that a suitable approach for the tube design is to design for the tangential (hoop) stress on the basis of uniaxial tensile creep data.

#### RECOMMENDATIONS

On the basis of the subject design study, the following recommendations are made in support of the over-all design effort for separable tube connectors:

- (1) Greep and/or relaxation data on René 41 at 1440 F, preferably relaxation data, should be generated.
- (2) It is recommended that a design factor of safety of 2.0 be used. However, one or more René 41 connectors should be tested at 1440 F in order to substantiate this factor of safety.
- (3) Avoid yielding of the connector components due to stresses incurred during installation.
- (4) Even though retightening of the connector compensates for the bolt or nut relaxation, care must be taken not to incur excessive deformations in the component pairs which could lead to rupture.
- (5) In order to be conservative, the cycle time should be measured from the start of the heating cycle to the end of the cooling cycle.
- (6) Consideration should be given to various design configurations. For example, loose-type bolted flanges may offer certain advantages over integral-type flanges which would not be realised in a static design.

#### NOMENCLATURE

- a mean coefficient of the rmal expansion, in/in/\*F
- A = area of bolt, gasket, tube, or flange, in.
- c = inner radius of flange or tube, in.
- C1 = coefficient in steady-state creep law
- C2 = coefficient in intercept stress law
- d = outer radius of flange or tube, in.
- o = deflect'on, in.

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- = moment arm for flange bending, in.
- w normal strain, in/in.
- د = etrain rate = 🙅 , in/in/br
- E = modulus of elasticity, psi
- FB = bolt flexibility, in/lb
- $\mathbf{F}_{\mathbf{B}}^{'}$  = bending flexibility of the "bolt" (threaded connector), in/1b
- Fy = flange flexibility at bolt circle, in/lb
- F.S. = design factor of safety
- g radius to centerline of gaskat, in.
- flange thickness, in.
- θ rotation, radians
- KB = creep rate of bolts, in/hr
- Kg == bending creep rate of the "bolt" (threaded connector), in/hr
- Kr = creep rate of flanges, in/hr
- L = influence coefficient from Page 138, Paragraph UA+47, of the ASME Code(1)

LB, Ly = effective longth of bols and flange, respectively, in.

- m = exponent in intercept stress law
- M = flange bending moment + Pe, in-1b
- v Poisson's ratio

Mattic design rofers to a conventional type design such as the Abbil Bouler Cadel 3), which is not time dependent.

<sup>\*\*</sup>The momentature of the A1512 Code<sup>[1]</sup> is used assistively in the sample Lade calculations on pages A. 3 and A. 4. and is use absorbers in the report where necessary for elarity.

- p = uniform internal pressure, pai
- P « total bolt load, ib
- $P_{\frac{1}{2}}$  = design value for residual bult load =  $\frac{P_{1}}{P_{1}R_{1}}$ , lb
- ry = flazibility ratio = Fy/FB
- $\tau \hat{y} = flexibility ratio = F_B^2/F_B$
- TK = Creep ratio = Kg/Kg
- TK = treep ratio = Kg/Kg
- $\mathbf{R} = \text{life factor} = \frac{\mathbf{i} + \mathbf{r}_{\mathbf{F}}}{\mathbf{i} + \mathbf{r}_{\mathbf{K}}} \text{ (bolted-flanged connector) or } \frac{\mathbf{i} + \mathbf{r}_{\mathbf{F}} + \mathbf{r}_{\mathbf{F}}'}{\mathbf{i} + \mathbf{r}_{\mathbf{K}} + \mathbf{r}_{\mathbf{K}}'} \text{ (threaded connector)}$
- J = normal stress, psi
- t = time, hr
- AT . temperature differential, "F
- V = influence coefficient from Figure UA-51, 3, Fage 144, Paragraph UA-51 of the ASME Code<sup>[1]</sup>
- w = width of gast st. in,

#### Subscripts

*	- axial load	r = radial direction in tube
B	= balt	R = room temperature
¢	= creep	t = time
•	• elastic	T = tube or design temperature
5	= flange	• circumferential direction in tube
G	a gasket	y = yield
м	* bending moment	= = axial direction

- o a intercept or sero time
- ept or sero time

#### CREEP AND RELAXATION

Creep and relaxation are closely related, even though they are not necessarily interchangeable in a stress-analysis problem. These phenomena play an important role in the behavior of a separable connector operating st elevated temperatures, and are best illustrated by machanical models.

Creep is the tendency for a material to exhibit time-dependent strains at a constant stress level, typical of metals at elevated temperatures. Figure 1 shows a typical creep curve (neglecting tertiary creep) and the corresponding Maxwell-Kelvin model. Figures 1 is obtained by superposition of the Maxwell and Kelvin models of Figures 2 and 3, representing the stady-state and primary creep, respectively. The enclosed design procedure utilises an equivalent Maxwell model, shown as the dashed curve in Figure 1. The total strain is made up of an instantaneous strain and a time-dependent (strain) et al. The instantaneous or intercept strain and a time-dependent (strain) et al. The instantaneous or intercept strain cudes the elastic strain c<sub>0</sub> plus an additional strain C<sub>200</sub><sup>m</sup> which conservatively accounts for the primary creep period.

Relaxation is the reduction or relaxation of stress in time under a constant strain. Elastic strains are replaced by creep strains, causing a progressive reduction in stress. The models in Figures 1-3 can be considered relaxation models by assuming a constant total strain rather than a constant stress.

There has been considerable literature written on how to predict relaxation from creep date, few agreeing on the best creep theory to use. Although there are very few test data available on the same material to enable a comparison of various theories, Finnis and Heller<sup>17</sup> made a comparison of copper at 165 C with an initial stress of 13,500 pei. Figure 4 shows that the steady-state creep law with the intercept stress reduction used herein is conservative for short times and shows good agreement with the experimental results for longer times. Of course, the time scale would be quite reduced for a superalloy at 1640 F.

The most basic form of a separable connector would be flexible bolting in a pair of rigid flanges, in which case the model of Figure 1 would suffice for a relaxation analysis. The tightness of the joint would be dependent on the ability of the bolt to resist cross deformations. However, the flange assembly can also be represented by a Maxwell model in parallel with the bolt assembly. The flange can increase or decrease the rate of belt relaxation, depending upon the characteristics of the flange geometry. A property designed flange seesably bould be superior to a rigid flange is that the flange elastic recovery as a spring will related the bolt relaxation.

#### MATERIALS

At the initiation of the program, it was thought that the connector might be exposed to service temperatures in the range from 70 F to a maximum of 1700 F. The upper temperature limit was subsequently changed to 1440 F. Consequently, material properties are evaluated at both 1700 F and 1440 F, and a material selection made at 1440 F.

#### Introduction

At the initiation of the materials survey, it was apparent that certain material properties were of major importance in ustabilishing the optimum connector design. Strength over the desired temperature range was the prime criterion. Is addition, because of the continued requirement to design space vehicles with minimum weight, density (or strength-to-density ratio) was a major consideration. Also, because service conditions involve cycling from maximum temperature to room temperature, thermal shock, embrittlement, and oxidation were also considered important criteria.

Of the various strength parameters, creep strength and resistance to relaxation were considered of utmost importance. In the installation of a separable connector, the connector is tightened to a given stress. To prevent leakage during service, this stress should remain sufficiently high during Service. Relaxation or creep during service will tend to loosen the joint and may give rise to leakage. In addition, good rupture strength and ductility are deemed necessary to insure against complete, catastrophic failure of the connector. Also, because the connector will be installed and tightened at room temperature, consideration must be given to thermal expansion of the connector during heating to the maximum temperature. Variations in thermal expansion of different materials comprising the tubing and the connector could lead to tightening of the connector and undue stress, or to loosening of the connector and leakage.

In addition, consideration was given to the availability, fabricability, machinability, and weldability of the various candidats materials. Of particular importance was the general engineering "\* ow-how" in the handling and use of the various alloys. Finally, since the connector should be copable of being loosened or tightened periodically, consideration was given to bonding or self-weiding of the various candidate materials under service conditions.

#### Material Properties

As indicated above, candidate alloys for separable connectors must have good creep, rupture, and relaxation strengths over the anticipated service-temperature range. Good resistance to thermal shock, oxidation, and embrittlement during cyclic service is also required. These requirements are almost identical to those involved in selection of alloys for turbine-buckst application in aircraft gas turbines. Turbinebucket materials, however, are not selected on the basis of relaxation resistance, although rupture strength and creep resistance are major requirements. At the present time, turbine-bucket materials are selected on the basis of rupture strength and thermal-shock resistance. Thus, it was evident that the requirements for separable connectors were sufficiently similar to these of turbine buckets that a summary of the various turbine-bucket materials would likely yield those alloys having the most desired combination of properties for the design of high-temperature separable connectors.

A survey of published property data for various nickel- and cobalt-based superalloys was conducted. Emphasis was placed on creep and rupture strength at 1700 F, and on short-time tensils properties over the service-temperature range. A search of the literature revealed that relaxation data were not available for the bulk of the alloys of interest. On the basis of this survey, the following alloys were selected for more thorough examination:

Wre	sight Alloys
(1)	Astroloy
(2)	Nimonic 105
(3)	Nimonic 115
(4)	René 41
(5)	Udimet 700
(6)	Unitemp 1753
(7)	Waspaloy
c	ast Alloys
<u>c</u> (8)	GMR 235
<u>C</u> (8) (9)	ART Alloys GMR 235 IN-100
<u>C</u> (8) (9) (10)	GMR 235 IN-100 Inco 713C
<u>C</u> (8) (9) (10) (11)	GMR 235 IN-100 Inco 713C Nicrotung
<u>C</u> (8) (9) (10) (11) (12)	GMR 235 IN-100 Inco 713C Nicrotung SM 200
<u>C</u> (8) (9) (10) (11) (12) (13)	GMR 235 IN-100 Inco 713C Nicrotung SM 200 SM 302

These alloys have the highest strength at 1700 F. All have been, or, are being considered for use as turbine buckets in gas-turbine engines. Not all of these alloys are roadily available, and not all are weldable.

In addition, three more common, lower strength superalloys on which extensive experience in fabrication and usage is available were studied for comparison purposes. These were:

(15)	inconel X
(16)	N 155
(17)	5 816

Typical compositions for these 17 alloys are given in Table 1.

#### Alloys for Use at 1700 F

The typical properties of the 17 candidate alloys are given in Table 2. It will be noted in this table that short-time tensils and rupture data were available for the bulk of the alloys. Creep data at 1700 F were quite often lacking. Where creep data were not available at 1700 F, information on the creep strength it lower temperatures was included.

TABLE 1	HOMMAL	COMPOSITIONS	07	SUPERALLOYS
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On the basis of yield strength at 1700 F and creep and rupture strengths at 1700 F, it is evident in Table 2 that the cast alloys have significantly higher strength than the wrought alloys. Of the 17 candidate alloys, SM 200, IN 100, Nicrotung, and Udimet 700 are the highest strength alloys at 1700 F. Of tisses only Udimet 700 is a wrought material. Of the wrought materials, the order of decreasing strength at 1700 F may be listed as Udimet 700, Nimonic 115, Unitemp 1753, Rene 41, and Waspaloy.

In general, it was found that the cohelt-based alloys were of lower strength than the nickel-based alloys.

It will be noted that the coefficience of thermal expansion of these alloys were quice similar in the temperature range from 1600 to 1800 F. Likewise, the density values for the nickel-base alloys were quite similar. Those alloys containing significant amounts of tungsten, of course, had slightly higher density values than those containing little tungsten. Because the density values were so similar, there was little merit in comparing the alloys on a strength-to-density basis.

#### Alloys for Use at 1440 F

It was subsequently decided that the maximum operating temperature for the connector should be reduced from 1700 to 1440 F. Consequently, a survey of the properties of the 17 alloys listed in Table 1 was made to ancertain the most promising alloys at this reduced temperature. The properties of the alloys at 1400 and 1500 F are summarised in Table 3. Where possible, values at 1404 F were interpolated.

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The data given in Table 3 show that the wrought alloys compare more favorably with the cast alloys at this temperature than they did at 1700 F. It will be noted that on the basis of yield strength at 1400 to 1500 F, René 41 and Astroloy (Udimet 730) compare favorably with SM 200 and IN 100. On the basis of strength or rupture in 10 hours at 1400 F, SM 200, Udimet 700, Nimonic 115, and René 41 have very similar strengths. For longer times at 1400 F and at 1500 F, René 41 compares less favorably with these other alloys. It will be noted that at 1500 F and above; Waspaloy has strength properties equal to René 41. In the range of 1400 to 1500 F, René 41 has appreciably better short-time tensile properties and higher short-time (10-hour) rupture and creep strengths than Waspaloy.

#### Material Selection

On the basis of the material properties cited in the preceding section, SM 200, Udimet 700, Nimonic 115, IN 100, and René 4i appear to have the highest strength at 1440 F. SM 200 is a cast alloy which is used principally as the turbine bucket material in the J57 engine. Extensive experience in its use and handling has not been gained as yet. In addition, it is not normally considered a weldable alloy. Udimet 700 has gained appreciable usage, and many shops are familiar with forging and machining the alloy. It is not readily rolled and is not considered weldable. The Nimonics are British alloye not readily available in the United States, and extensive data and experience are not available on Nimonic 115. IN 100 has gained little commercial acceptance, and little is known of the alloy other than the properties cited.

Extensive experience, up the other hand, has been gained in the use of René 41. The alloy is available as forgings, billeta, ber, and sheet from reveral sources. It has excellent strength properties over the temperature range from 70 to 1500 F. In fact, it was for this temperatures range that René 41 was specifically designed. It was not designed to be used at temperatures above 1500 F.

Not these teasing, René 41 was selected as the norst promising material for high-timin rature separable connectors. Table 3 often the short-time-to-sale and the available interperupture properties of René 41. Complete time-to-sale and the available interperupture properties of René 41. Complete time-to-sale and the available interperupture properties of René 41. Complete time-to-sale and the available interperupture properties of René 41. Complete time-to-sale and the available interperupture properties of René 41. Complete interperature use to obtain specific data for densing purposes, a tamby of complete creep surves at a number of stresses at 1442 F is required. Such curves were not available in the literature. In mist creep testing, loade are usually select: ito produce rupture in 1000 hours or less. Such loade can result in 0.2 to 0.3 pes cent deformation on loading. At lower stress levels where smalles per cent deformation on loading occurs, rupture to somplete sceep data are not ruported. Nevertheless, a few complete creep curves for René 41 in the temperature range of interest were unsovered in Referencee (12,13).

Relaxation data on Hené 41 were not available in the literature.

Revé 4) is a precipitation-hardenable nickel-base alloy. The material is usually supplied in the solution-annealed condition. Hardwhing is accomplished by aging the alloy at 1400 or 1650 F. This causes precipitation of a complex nickel-litaniumsluminum phase (called the gamma-prime phase) and produces excellent strength over the temperature range of interest.

Poine 41 is readily lorged and considerable experience has been accumulated by many forge shops in working the alloy. Like most high-strength nickel-base alloys, Rom 41 work hardons rapidly during machining and is considered difficult to machine. (coertheles, it is machined by many shops on a production basis.

Rone 41 is considered a weldable alloy. Nevertheless, as is the case with all high-strength procipitation-hardenable alloys, welding may be somewhat of a problem. Webling in the solution-astroated condition (the low-attength condition) is recommended. After welding, the part should be veannealed and aged to obtain the properties cited. Webling in the signal condition, particularly under conditions of stress, is not recommended. This is true of all precipitation-hardenable inckel-base alloys. In general, it has been found that the greates the aluminum plue titanium content, the higher is the strength and the more difficult are the welding operations. The use of a significantly more welfable inaterial will consequently entail a very appreciable reduction in strength and a si able weight penally.

Two specific heat treatments have been developed for René 41: one to develop maximum short-time strength and one to develop maximum long-time creep and repture strengths. The torene treatment involves solution treating at 1930 F for 4 hours, air ding, sid aging at 1400 F for 16 hours. This treatment yields good short-time whe properties and acceptable 10-hour supfore and treas atrengths. The latter -atment, involving schono treating at 2500 F for 4 hours, instructing, and aging at 1650 F for 4 hours, produces somewhat reduced short-time treasile properties but

wile properties and acceptable 10-boar implicite and crease strengths. The latter - attract, involving schonon treating at 2150 F for 4 hours, are calling, and aging at 1650 F for 4 hours, produces somewhat reduced short-first retensile properties but significantly improved long-time crease and rupture strength. As indicated previously, Reré 41 should be wellard in the solution-treated condition. Recent welding studies have revealed problems in weld metal cracking for material welded after the hightemperature treatment (2150 F). The use of the lower temperature treatment (1950 F) has reportedly eliminated weld-metal cracking problems. For this reason, a solutiontreatment temperature of 1950 F and an aging treatment of 1400 F are economiceded. With such a heat treatment, high short-time tensile properties and good 10-hour rupture and creep proberies will be obtained. A sacrifice in long-time creep and rupture strength will result. Long-time strength is not a prime requirement for the present application.

René 41 has excellent oxidation resistance over the spacified service-temperatire range. It is not known to become embrittled in this temperature range. While originaritive data on instance to thermal shock are not available, it is believed that the all-thas comparable or better resistance to thermal shock than the other candidate materials.

Finally, it is not anticipated that René 41 will show a tendency to bonding or ord welding during service. Alloys which contain appreciable amounts of chromium, aluminum, and titanium are exceedingly difficult to bond. Bonding is accomplished only in atmospheres capable of reducing the stable spinel-type oxide on the surface of the alloy and under conditions of high temperature, high stress, and relative metal flow at mating surfaces. Consequently, self-weiding of connectors of this alloy is not to be expected. If sticking of the connector is encountered, oxidsion of the mating surfaces of the connector slould reduce, if not eliminate, the problem. Heating the connector at about 1800 F in an strongsphere containing a low oxygen content should produce a heig, this oxide layer which is exceedingly difficult to remove. Such an oxide film should prevent self-welding of the connector. If self-bonding presents, the deposition of a very this layer of Stanium or aluminum on the mating surfaces of the connector is recommended. The this film should be diffused into the connector and oxidized by litering to high temperatures alon has solution treating at 1950 F. This should prevent solf-welding.

In summation, Kens 41 has been selected as the material of construction for high-temperature separable (conscions operating from room temperature to 1440 F. Certain other riloys have better creep strength than René 41, however, the combination of properties obtains the in René 41, as well as the availability and experience gained on for alloy, strongly recommendant.

This survey and the recommendations were made with limited knowledge of the billing instartial that will be used with the Rene 41 connector. If the connector is to be userial its the tobing, including of the tubing material is required before the weldability enounce compatibility of the connectors and the tubing can be ascertained. Welding dissiontlar materials is outcome analyze problem. While it is not the propose therein to elect a tubing materials as outcome tubing intervals, analyze as a second tubing materials can be associated with René 41 connectors. While it is not the propose therein to elect a tubing materials, consideration was given to various tubing materials capable of being employed with René 41 connectors. René 41 tubing is not surrently comments as addulie. The high-temperat in tubing materials capable of being employed is the application and surrently sorth in a ly exclude are insoned X. Hastelloy X. In one 178, and Waspalos. In onel X. Hastelloy X, and how and 178 are of significantly different composition then Rine 41, and the heat treatmost employed to string the there allows as not comparable with that recommended for Rene 41, and has similar phoateket is discovered as minder on one wide the sum of Rene 41, and has a similar phoateket is discovered by operations to be a similar for all so compatible with treat of Waspalos, is quite similar is with that recommended for Rene 41, and has a similar phoateket is discovered by the similar with the streatment of Rene 41, and has similar phoateket is discovered by the similar similar with the streatment of Rene 41 is compatible with treat is Waspalow, and will be a Rene 41 connectors to Waspaloy tubing it reactible.

for the store suggested that Weatably be given fail consideration as the tubing ogaters, but us is in the tubing ogaters, but the compared the connector should be

in the solution-treated condition for welding. After welding, the assembly must be completely heat treated. The heat treating sequence recommended is as follows.

- (1) Solution treat at 1975 F for 4 hours, air cool
- (2) Stabilize at 1550 F for 24 hours, air cool
- (3) Age at 1400 F for 12 hours, air cool,

This heat treatment is recon insided for Waspaloy. The René 41 connectors, when subjected to this heat treatment, will have properties comparable to those reported herein.

#### Design Properties of Rene 41

The miniinum or steady-atale creep rates are determined from the master creep curve of Reference (14), shown in Figure 5. The data of Figure 5 are replotted in Figure 6 for 1440 and 1500 F. The straight dashed lines in Figure 6 are a conservative fit to the data at the two temperatures. Thi corresponding constants are:

### At 1440 F, $C_1 = 4,23 \times 10^{-26}$ ,

#### n = 4,97.

At 1500 F.

# C1 = 1.30 x 10-26.

#### •

# n = 4.82.

A rapid initiation determining the constants  $C_j$  and in from creep data, using the creep law  $\xi \neq C_j\sigma^n$  , is as follows:

Determine the strain rates  $i_1$  and  $i_2$  for the stresses  $J_1$  and  $J_2$ , respectively. From the steady-state creep law,  $i_1/l_2 = (J_1/J_2)^n$ , where n is the only unknown, and is easily solved. After calculating n, the constant  $C_1$  can be determined from  $i_1 = C_1J_1^{n}$ .

The constants  $C_2$  and m for the intercept stress were obtained from the test data and plotted curves of Reforence (13). Since the intercept data of Reference (13) are not very consistent, the following constants are considered only approximate in magnitude. This is one reason for the recommendation that creep data be generated for René 41 at 1440 F.

#### At 1440 F.

#### Cz = 1.25 x 10-8,

### m × 1.00.

### At 1500 F.

# $C_2 = 3.93 \times 10^{-11}$ ,

#### m = 1,57,

The design data for René 61 at 1640 F and 1500 F are plotted in Figure 7 for various stress levels.

Other design propertise of René 41 are obtained from Tables 2 and 3.

The yield strength  $v_y = 120,000$  psi at 70 F,

### $\sigma_y = 103,000$ per at 1440 F,

### ວູ = 97,000 ps: ວະ 1500 F.

### The modulus of electricity $E_{\chi\chi}$ = 32, 0 $\times$ 106 pci at 70 F ,

 $E_{\rm T} = 24.4 \times 106$  µei at 1440 F,

#### » 3 н 24.0 x 106 рет at 1500 F.

The mean coefficient of the rmai expansion  $\alpha = 8.3 \times 10^{-6} in/in/*F$  at 1440 F.

a = 8.5 x 10"6 in/in/\*F at 1500 F.

#### DESIGN PROCEDURE

#### Basis for Design Procedure

The design procedure considers only the elastic and creep behavior of the bolts and flangs assembly. Secondary effects such as flangs rotation due to internal pressure, creep of the gashet, creep of the threads, creep benuing of the bolt, and dynamic creep are not included in the design procedure, but are discussed in Appendix C. The design factor of safety is intended to account for these secondary effects in addition to the scatter in immerial properties.

The gashed (weat) design for mechanical fittings is not included in the design procedure, but is discussed in Reference (11), where it was concluded that plastic yielding of seal surfaces is descrable for effective sealing. An initial sealing stress as high as 2.75 times the yield errength of the metal gasket material was considered lice essary to achieve the descred degree of yielding in order to keep the leakage rate of licium below the to arou is decided.

It is assumed that the bolts are flexible compared to the gasket in order that the application of internal pressure results in negligible bolt-load chaiges. It is not necevery that the gasket be the only load path through the assembly.

The design procedure is intended primarily as a method for calculating relaxation of the connector assembly, and does not include a failure analysis of the component parts. It is assumed that the material possesses utility to withstand the progressive deformation incurred if the connector assembly is retightened after each cycle. Various methods of analysis available for predicting the tube strains are discussed in Appendix D. Thuse methods are intended for the tube design at locations sufficiently removed from discontinuities such as the connector assembly.

The detailed design procedure for bolted-flanged and threaded connectors is described step-by-step for the purpose of obtaining a curve of leaks, a pressure versus time for a time to leakage for a specific value of the design pressure). The effects of external loads, temperature differentiels, and retightening, and the use of relaxation data are discussed sejarately from the banic design procedure.

The equations used in the design procedure are derived in Appendix 3.

#### Bolted-Flanged Connectors

The basic design procedure for a bolted-flanged connector consists of nine specific steps in order to arrive at a curve of leakage pressure versus timo. Figure 8 shows a typical flunge geometry,

#### (1) Design Conditions

Establish the design température, design pressure, nominal connector diameter, tube geometry, life requiremente, and temperature differentials.

#### (2) Material Properties

Determine the yield strength  $\sigma_y$  and the modulus of elasticity E at both room temperature and the design tomperature, and the coefficient of thermal expansion u at the design temperature.

. From a family of creep curves at the design temperature, determine the constants  $\zeta_1,\ C_2,\ m,\ snd$   $\pi$  to fit the following equation:

$$s = \epsilon_0 + C_2 \sigma_0^m + C_1 J_0^n \epsilon$$
, (1)

The creep constants should be selected to give the best fit to the experimental data in the range of stress levels expected in the design<sup>®</sup>.

#### (3) ASME Code Design

The design procedure can be applied to any arbitrary design. However, it is desirable to use the ASME Boiler and Pressure Vessel Code<sup>(1)</sup> for the initial static design in order to establish a balanced design. Also, the Code-rated flange can subsequently be rated on a life basis for comparison.

Using the rules of the ASME Code<sup>(1)</sup>, establish either a loose-type or integraltype flange geometry for an appropriate value of internal pressure. This pressure, of course, should be at least as high as the design pressure multiplied by the ratio of the vold-to-hot modulus of elasticity, and multiplied by the design factor of safety. The initial gasket seating load will be dependent on the type seal used and the maximum permissible leakage rate, and need not follow the rules of Roference (1). The allowable Code stress is assumed to equal two-thirds of the room-temperature yield strength in order to avoid yielding of the assembly during the bolt-up operation.

The ASME Code, will establish a static design at room temperature. The initial boli load and bolt stress at room temperature are designated as PBR and JBR, respectively.

#### (4) Flexibility Equations

Determine the bolt (lexibility  $\mathbf{F}_{\mathbf{B}}$  (bolt deflection due to a unit bolt load):

$$\mathbf{F}_{\mathbf{B}} = \frac{\mathbf{L}_{\mathbf{B}}}{\mathbf{A} \cdot \mathbf{E}_{\mathbf{T}}} \quad , \tag{2}$$

where

 $L_{\rm B}$  = effective length of the bolt, usually taken as the total length between the nut bearing surfaces plus one nominal bolt diameter,

AB = total bolt area based on root diameter.

Determine the flange flexibility  $F_{\overline{F}}$  (total deflection of two flanges at the bolt circle due to a unit bolt load).

For a loose-type flange,

$$F_{F} = \frac{3.82 e^{2}}{E_{T}h^{3}\ln d_{F}/c_{F}}$$
(3)

Buther despois on the sizep constants and the intercept stress to bustion are given us pages 6–15, uno 8-5.

For an integral-type flange,

$$F_{F} \sim \frac{1.29 V_{a}^{2}}{L_{V} c_{V} (d_{T} - c_{T})^{5/2} E_{T}}$$
(4)

#### (5) Creep Equations

Calculate the rate of creep deflection of the bolts  $K_{\rm B}$  and the rate of creep deflection of the two flanges at the bolt circle,  $K_{\rm F}$ :

 $K_F = \frac{4\sigma_{F1}^n C_1 e_F}{r}$ ,

$$\frac{C_{I}P_{t}}{A_{B}^{n}},$$
(5)

16:

where

and

$$\sigma_{\mathbf{Ft}} \cdot \left(\frac{\mathbf{h}}{2\mathbf{c}_{\mathbf{F}}}\right)^{\frac{1}{n}} \left(\frac{\mathbf{P}_{\mathbf{t}}\mathbf{0}}{2\pi}\right) = \left[\frac{\left(2^{1+\frac{1}{n}}\right)\left(1,\frac{1}{n}\right)\left(2,\frac{1}{n}\right)}{\left(\mathbf{d}_{\mathbf{F}}^{-1-\frac{1}{n}}-\mathbf{c}_{\mathbf{F}}^{-1-\frac{1}{n}}\right)\mathbf{h}^{2+\frac{1}{n}}}\right]$$

KB °

Equation (6) applies only to a loose-type flange. The creep rate of an integraltype flange is assumed equal to that of the loose-type flange, provided the maximum design stresses according to Reference (1) are equal. The procedure for an integraltype flange is to calculate the maximum design stress by Reference (1). Then determine the thickness of a loose-type flange with the same flange inner and outer radii, moment arm, and maximum design stress as the integral-type flange, this results in an equivalant strength loose-type flange. Using the loose-type flange geometry, calculate the creep rate from Equation (6), assuming this to be the creep rate of the integraltype flange.

#### (6) Fiexibility and Creep Ratios

Calculate the life factor R for the purpose of conveniently evaluating a flange geometry on a life basis:

R

$$=\frac{1+r_{\mathbf{F}}}{1+r_{\mathbf{K}}},$$
 (7)

 $r_F = flexibility ratio = F_F/F_B$ ,

rK = creep ratio = KF/KB .

Equations (5) and (6) should be calculated in terms of the bolt load  $P_t,$  in order that the creep ratio  $r_{\rm K}$  be calculable.

#### (7) Intercept Stress Reduction

Calculate the bolt stress,  $\sigma_{BT}$ . The initial room-temperature bolt stress  $J_{BR}$  is reduced to a new value  $\sigma_{BT}$  at the design temperature due to the change in modulus of elasticity:

$$J_{BT} = J_{BR} \left( \frac{E_T}{E_R} \right) . \tag{8}$$

Calculate a further reduction in bolt stress from  $2_{\rm BT}$  to a value  $2_{\rm 0}$  due to the effects of primary creep:

$$J_0 = J_{BT} = \frac{E_T C_2 J_0^{m}}{R}$$
 (9)

Equation (9) is conveniently solved for  $J_0$  by trial and error. The bolt stress  $J_0$  or corresponding bolt load  $P_0$  represents the actual starting point in the life calculations.

#### (6) Bolt Relaxation

Calculate the time t required for the bolt stress to relax from an initial value  $\sigma_0$ , to a new value  $J_1$ :  $r = \left(\frac{-2}{r}\right)^{n-1}$ 

$$t = \frac{R}{C_{4}E_{T}(n-1)} \left[ \frac{1 - \left(\frac{1}{c_{0}}\right)}{t_{1}} \right]$$
(10)

#### (9) Leakage Pressure

Solve Equation (10) successively in order to obtain a curve of residual bolt stress  $\sigma_i$  or losd  $P_i$  versus time. Convert these results to an allowable leakage pressure versus time curve by the curvessive solution of Equation (11):

$$\mathbf{P}_{\mathbf{t}} = \mathbf{P}_{\mathbf{p}} \times \mathbf{P}_{\mathbf{G}} \quad , \tag{11}$$

 $F_{L} = \frac{P_{L}}{F_{L}}$ 

where

 $P_{p} = bolt load due to pressure = "pg<sup>2</sup> .$ 

P<sub>G</sub> + residual gasket load (or load across fiange face) required to prevent excessive leakage (the determination of this load us not included in this report).

The terms V and L in Equation (4) conform to the non-epitature of Reference (1)

In Equation (11),  $P_t$  is established directly from the  $\mathcal{T}_t$  versus t curve by applying a suitable factor of safety to the load rather than the time scale. The value  $P_G$  is dependent on the type of gaskst used, and may or may not be a function of the internal pressure p.

### Threaded Connectors

The design procedure for threaded connectors commences with Step 1 (design conditions) and Step 2 (material properties) of the bolted-flanged connector design procedure. Sin additional steps are then required for the threaded connector in order to obtain a leakage proseure vorsus time curve.

Reference (15) can be used for the static design of a threaded connector. The nomenclature for the parts of the threaded connector is based on the functional analogy to the parts of the bolted-flanged connector and is not to be taken as a literal description of the parts. A threaded connector model is shown in Figure 9. The nut is considered the "bolt" member of the analysis, and contributes both axial and bending deformations. The flange and union members in Figure 9 are considered the "flange" member of the analysis with axial deformation only.

(1) "Bolt" Flexibility

Calculate the "bolt" (lexibility  $\mathbf{F}_{B}$ , which represents the axial flexibility of the nut and has the same form as Equation (2):

$$\mathbf{F}_{\mathbf{B}} = \frac{\mathbf{L}_{\mathbf{B}}}{\mathbf{A}_{\mathbf{B}}\mathbf{E}_{\mathbf{T}}} , \qquad (12)$$

where LB and AB are the effective nut length and area, respectively.

Galculate the bending flexibility of the "boli"  $F_{\mathbf{p}}$  by using the development on pages 194-197 of Reference (15) for inwardly projecting flanges (see Figure 10):

$$\mathbf{F}_{\mathbf{B}}^{\perp} = \mathbf{e}^{2} \left(\frac{\hat{\mathbf{e}}}{\mathbf{M}}\right) , \qquad (13)$$

where

$$\frac{\theta}{M} = \frac{3(1-\nu^2)}{\pi d_T \beta (d_T - c_T)^3} \mathbf{E}_T \left[ \frac{(1-\nu^2)}{2\beta d_T} \left( \frac{h}{d_T - c_T} \right)^3 t_T \frac{d_T}{c_F} + 1 + \frac{\beta h}{2} \right]$$

and

$$\beta = \sqrt{\frac{3(1-y^2)}{A_T^2 (d_T^{-}c_T)^2}}$$

(2) "Flamge" Flexibility

Galculate the axial flexibility of the "flange", Fr:

$$F_{\overline{F}} = \frac{L_{\overline{F}}}{A_{\overline{F}}E_{\overline{T}}} \qquad (14)$$

The effective length  $L_F$  can be considered equal to  $L_B$  in Equation (12). A\_F is the cross-sectional area of the "flange".

(3) "Bolt" Creep

Calculate th

s axial creep rate of the "bolt", K<sub>B</sub>:  

$$K_{B} = \frac{C_{j} P_{t}^{n} L_{B}}{A_{B}^{n}}.$$
(15)

Calculate the bending creep rate of the "bolt",  $K_B^{\prime}$ , in a manner similar to an integral-type flange<sup>0</sup>, except that the maximum design stress is determined from Reference (15) for an inwardly projecting flange.

The longitudinal hub stress  $S_{\rm H},$  radial ring stress  $S_{\rm R},$  and tangential ring stress  $S_{\rm T},$  respectively, are

$$\mathbf{S}_{\mathbf{H}} = \frac{\frac{3M}{\pi d_T (d_T - c_T)^2} \left[ \frac{(1 - \frac{y^2}{2\beta d_T} \left( \frac{h}{d_T - c_T} \right)^3 + in \frac{d_T}{c_F} + 1 + \frac{\beta h}{2} \right]}{\frac{3(1 + \frac{\beta h}{2})M}{\pi d_T h^2} \left[ \frac{(1 - \frac{y^2}{2\beta d_T} \left( \frac{h}{d_T - c_T} \right)^3 + in \frac{d_T}{c_F} + 1 + \frac{\beta h}{2} \right]},$$
(16)

$$S_{T} = \frac{3(1-\nu) \ln M}{2 \pi d_{T} c_{F} (d_{T} - c_{T})^{3} \beta \left[ \frac{(1-\nu^{2})}{2\beta d_{T}} \left( \frac{h}{d_{T} - c_{T}} \right)^{3} \ln \frac{d_{T}}{c_{F}} + 1 + \frac{\beta h}{2} \right]}$$

where  $\beta = \sqrt[4]{\frac{J(1-y^2)}{d_T^2(d_T-c_T)^2}}$ 

"Bes procedure en page 18 fer integral-type Bangen.

The maximum stresses from Equations (16) should be combined according to the method of Reference (1). For a threaded connector, Equation (6) herein should be used with a constant of two rather than four because there is one rather than two flange members.

#### (4) "Flange" Green

Galculate the axial creep of the "fiange", Ky, which has the same form as Equation (15):

$$\frac{C_1 P_1^{n} L_p}{K_p = \frac{\Lambda_p}{\Lambda_p}},$$
 (17)

(18)

(5) Flexibility and Creep Ratios

Calculate the life factor R: 
$$R = \frac{1+r_F+r_F'}{1+r_R+r_K'} \ , \label{eq:R}$$

where

where

$$r_F + F_F/F_B , r_F + F_B/F_B ,$$

rK\*KF/KB , rK\*KB/KB .

#### (6) Nut Relexation

Determine the relaxation of the nut from Equation (10), but using the life factor R from Equation (18). Use Equation (11) to calculate the leakage pressure versus time.

#### External Loads

Determine the effect of external loads on the leakage pressure by revising Equation (11) to read:

$$\mathbf{P}_{t}^{\prime} = \mathbf{P}_{p} + \mathbf{P}_{G} + \mathbf{P}_{A} + \mathbf{P}_{M} , \qquad (19)$$

where  $P_A$  and  $P_M$  include the effects of external thrusts and moments, respectively:

$$P_{A} = \frac{\pi p K_{A} c_{T}^{2}}{p_{M}} = \frac{\pi p K_{M} c_{T}^{2} (d_{T}^{2} + c_{T}^{2})}{\frac{2d_{T}g}{2}}$$
(20)

Solve Equation (19) successively in order to obtain a leakage pressure versus time curve.

The constants  $K_A$  and  $K_M$  are used to define the magnitude of the stress due to tube thrusts  $(\sigma_A)$  and moments  $(\sigma_M)$ , respectively, in terms of the axial tube stress  $(\sigma_T)$  due to internal pressure:

к

$$\mathbf{A} = \frac{\sigma_{\mathbf{A}}}{\sigma_{\mathbf{T}}} , \quad \mathbf{K}_{\mathbf{M}} = \frac{\sigma_{\mathbf{M}}}{\sigma_{\mathbf{T}}} ,$$
$$\sigma_{\mathbf{T}} = \frac{\mathbf{p}\mathbf{c}_{\mathbf{T}}^{2}}{(\mathbf{d}_{\mathbf{T}}^{2} - \mathbf{c}_{\mathbf{T}}^{2})} .$$

#### Temperature Differentials

The analysis of temperature differentials is limited her vin to the case of components at uniform temperatures, but at a different average temperature from another component of the assembly, but at a different average temperature than the flange and tube assembly, probably the more critical case. In the other case, the flange and bolt assembly is assumed to be at a different average temperature than the teles. The change is indexided that the temperature differentials are of a short-time nature, occurring almost instautaneously for the purpose of analysis. A more refined analysis would be required for temperature differentials varying considerably with time.

#### Bolt-Flange Differential

Letting a positive value of  $\Delta T$  represent the finnge at a higher average temperature than the bolt, calculate the change in bolt stress  $\Delta\sigma_{jk}$ :

$$\Delta \sigma_{\mathbf{B}} = \frac{\alpha (\Delta T) \mathbf{E}_{\mathbf{T}}}{1 + r_{\mathbf{F}}} \qquad (21)$$

For a threaded connector, the change in "bolt" (nut) stress is

$$\Delta \sigma_{\mathbf{B}} = \frac{a(\Delta T) \mathbf{E}_{\mathbf{T}}}{1 + r_{\mathbf{F}} + r_{\mathbf{F}}^{2}} \qquad (22)$$

The change in boit or nut load in Equations (21) and (22) should be evaluated in a different manner, depending upon the nature of the differential. A positive temperature differential at the start of the cycle would lead to an increase in the boil load. This increase should be used to guard against overstressing of the boil (or other connector components), and neglected for short-time differentials in the relaxation analysis. If a negative temperature differential ehould occur near the end of the operating cycle, the reduction in holi coal should be udded to Equation (21) to determine a reduced leakage pressure:

(23)

where  $P_T = \Delta \sigma_B A_B$  (bolt-load change due to temperature differential).

### Flange-Tube Differential

The analysis by Dudley<sup>(16)</sup> or an extension of the equations presented by The analysis by Dudley<sup>1427</sup> or an extension of the equations presented by Rodubuugh in discussion of Reference (17) can be used to evaluate bolt-load changes due to the tube and flange assembly being at different avorage temperatures. Thue raethod of Rodabaugh<sup>(17)</sup> is preferred, since it is consistent with the banic equations of the ASMs Code<sup>(11)</sup>. At the start of the operating cycle the tube will possibly be hotter than the remaining assembly, thus causing a reduction in the bolt load. The opposite effect may occur upon cooling down the assembly. An increase in the bolt load should be used to guard against overstressing the connector assembly, whereas decrease in the bolt load should be used in Equation (23).

### Retightening

If the assembly is not retightened between operating cycles, a continuous operating cycle is assumed, consisting of the total time of the individual cycles. This procedure is consistent with the assumption that creep strains are irreversible in nature.

In the event that retightening is contomplated, each cycle is realisized according to the creep design procedure (not repeating static design) as though it were the first cycle. This is a conservative approach because the flange life will probably improve on subsequent cycles to the estimating the primary creep strains are nonrecurring, as evidenced by the British Flange Tests<sup>(9)</sup>.

#### Relaxation Data

The design procedure is based on the assumption that only creep data are available. Of course, relaxation data are preferred. Relaxation data should be available as a family of curves, starting from various initial stress levels, sufficient to cover the range of initial stress levels expected in the design,

Since the relaxation test is usually run with no elastic follow-up (R \* 1), the time required for a connector assembly bolt or nut stress to relax from an initial stress  $\sigma_0$ to a final value  $\sigma_t$  is obtained directly from the relaxation test data, multiplied by the life factor R determined from Equation (7) or (18). If the relaxation tests are run with elastic follow-up, the relaxation time is determined by multiplying the test data (time) by the ratio of the life factor R of the connector to that of the test assembly. Inter-polation of the test data may be required for epecific values of the initial stress  $\sigma_0$  not available in the test data. The use of a direct ratio for various values of the life factor is based on the assumption that the life factor is a constant in the analysis. Baumann<sup>(2)</sup> justifies this approach, regardless of the creep law observed by the material.

The direct use of relaxation data results in a residual stress versus time curve, similar to that yielded by Equation (10). In generating relaxation data, it would be desirable to run at least one test with sufficient elastic follow-up to simulate the life factor of a connector assembly.

### Redesign Considerations

The designer may find that the calculated connecto-life for a particular design In e designer may time that the calculated connecto life for a particular design pressure is less than the required design life. If the temperature and material have already been fixed, the only changes that can be made in order to obtain an improved life are geometry changes. Ordinarily, the connector inner diameter, boil geometry and gesket geometry will be kept constant during a redesign. Therefore, the logical changes to make if the life factor R in Equation (10) is to be improved are changes in the flange thickness, flange width, and moment arm.

Changes in the moment arm a can have an appreciable effect on the life factor R. Changes in the moment arm e can have an appreciable effect on the life factor R. Of course, an increase in the moment arm will have to be compensated for by changes in the flange width and/or thickness (or hub geometry for an integral-type flange) in order that the flange stresses be maintained at an acceptable level. It is recommended that improvements in the connector life be made by simultaneous changes in the moment arm for flange bending, the flange width, and the flange thickness. Effects of changes in the length of the bolt Lg (perhaps with the use of a rigid bolt collar) should also be investigated. investigated.

If the desired connector life is not attained by the changes described above, the next logical step would be to redesign the bolt geometry and garket geometry in order to allow for greater percentage reductions in the initial bolt load bufore leakage occurs.

### DISCUSSION OF SAMPLE CALCULATIONS

The sample calculations are presented in order to adequately demonstrate the de-eign procedure, and not to ar-live at cptimum design connectors. An optimum design can be determined only after calculating a wide range of connector geometries, thus anabling the selection of a minimum-weight design for a given combination of design (laskage) pressure and life.

The lowse-type and integral-type flange geometries of Figures 11 and 12, respectively, have nearly the same initial stress  $\sigma_{\phi}$ , and the life factor R differs by only 10 per cent. Therefore, the results in Figures 13 and 14 for the loose-type flange very nearly apply for the integral-type flange. The loose-type flange is heavier. This will not always be the case since the loose-type flange. and frequently have a much higher flexibility than the equivalent integral-type flange.

The factor of safety used in Figure 13, together with the intercept stress reduction and the bolt load reduction due to change in elastic modulus, loads to a maximum design pressure of 2980 pei in Figure 14, assuming no axternal loads or temperature differ-satials. It is shown that a temperature difference between the bolt and flange assembly

of -100 F severely affects the leskage pressure and establishes a limiting life of 36 hours. The curves of Figure 14 indicate that, up to approximately 0.5 hour, there is negligible change in the leakage pressure. This would infor that, for accumulated life cycles totaling 0.5 hour or less, retightening after each cycle would not be very beneficial.

René 41 at 1500 F does not exhibit a great deal of primary creep, as shown by the relative insensitivity of the initial stress  $v_0$  to the life factor R in Figure 15. For some materials, the effect of the life factor on the reduced intercept stress would be a streng influence  $t_{\infty}$  chroning an optimum design. The life factor R is a convenient measure of the relative merits of various designs with the same initial bolt stress because it is essentially a geometry-dependent constant in the relaxation equation, all other constants in Relation (10) being a function only of the material and temperature.

It is usually considered desirable to preload the bolt or mit as high as possible without yielding any portions of the connector assembly. The high preload is intended to delay the relaxation of atress or load. Various values of the initial bolt stress were used in the relaxation equation (27). The results in Figure 16 indicate that, for design lives over 1-2 hours, there is very little advantage gained by higher prestressing. In fact, the initial prestress G<sub>0</sub> of 45,000 psi would be more desirable for longer lives since it will provide greater safety against yielding. The initial stress of 90,000 psi at 1500 F for René 41 would be difficult to obtain because of the intercept stress reduction, the elastic modulus change, and the limitation imposed on the room-temperature bolt preload to avoid yielding. preload to avoid yielding.

Results of the threaded-connector geometry of Figure 17 are shown in Figure 18. The geometry of Figure 17 does not represent a well-balanced design. As a result, the axial nut stress  $\sigma_{\rm BR}$  is limited to 40,000 pei (room-temperature prestrees) to avoid overstressing the inwardly projecting flange. The low prestress results in the relatively slow relaxation of the nut load shown in Figure 18. It is quite possible that the optimum static design will also provide a good design for relaxation, depending upon the material and design temperature. However, for some combinations of material and temperature, the sound static design proced may not suffice because of the importance of the interaction of bolt and flange creep and flexibility ratios. For a static design, the integral-type bolted flange is usually lighter than the loose-type flange. This may not be true for relaxation design. In some cases, loose-type flanges offer advantages in the way of increased flexibility and the ability to better withstand certain types of temperature differentials.

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LMC:ECR:DBR:TMT/all

#### APPENDIX A

#### SAMPLE CALCULATIONS

The sample calculations are all performed on René 41 at 1500 F. The following design values apply to René 41 at room temperature and at 1500  $\mathbb{R}^6$ .

The yield strength:

oy = 120,000 pei at 70 F,

$$\sigma_y = 97,000 \text{ psi at } 1500 \text{ F}.$$

The modulus of elasticity:

- ER = 32.0 x 10<sup>6</sup> psi at 70 F,
- ET = 24.0 x 10<sup>6</sup> pei at 1500 F.

The mean coefficient of thermal expansion  $a = 8.5 \times 10^{-6}$  in/in/\*F at 1500 F.

The croop and intercept constants at 1500 F are:

 $C_1 = 1, 30 \times 10^{-26}, C_2 = 3,93 \times 10^{-11},$ 

= 4, 82,

The ASME Code calculations ar-, based on an internal pressure of 10,000 psi. This corresponds to a reduced value of the design pressure equal to 10,000  $(E_T/E_R)$  (1/F.S.) = 3750 psi for a factor of safety equal to two. Therefore, the maximum possible value of the design pressure is 3750 psi, and will be further reduced due to the effecte of primary creep.

m = 1, 57,

#### Loose-Type Bolted Flange

#### ASME Code Design<sup>64</sup>

The allowable stress for use in Reference (1) equals 2/3 (120,000) = 80,000 psi. The geometry is shown in Figure 11.

"Malerial properties for liend 41 are given on pages 18-18 "Use the portent latter of the Allast Could 13. The tube thickness can be determined from other considerations, such as discussed in Appendix D, to equal 0,210 in.

For the purpose of the bolt design, assume that a residual gasket stress of 3P (P = 10,000 psi, the ASME Gode design pressure) is required to maintain the gasket east. Also, assume that the gasket plating material has a yield strength of 10,000 psi and that it requires a gasket stress of three times the yield strength (y = 60,000 psi) to seat the metallic gasket. For a gasket width 2b = 0,125 in. (assume full width is effective),

Hgs = 2b x 3, 14G x 3P = 45,900 lb,

$$H = 0.785 G^2P = 119,000 lb,$$

$$W_{ml} = H + H_p = 164,900 lb,$$

W<sub>m2</sub> = 3, 14bGy = 45, 900 1b.

The required bolt area is

The design bolt load W = 164, 900 lb.

$$A_{m1} = \frac{164,900}{80,000} = 2.06 \text{ im}^2.$$

Use 10-9/16-in.-diameter boits with a total root area equal to 1, 89 in,  $^2$ . The boit stress is

For the flange design,

$$x_G = h_T = 0.663 \text{ in.}, h_D = 0.813 \text{ in.},$$

- HD = 0.785 BZP = 102,000 16,
- H = 0.785 G<sup>2</sup>P = 119,000 lb,
- H<sub>G</sub> = W-H = 45,900 1b,
- H<sub>T</sub> = H-H<sub>D</sub> = 17,000 1b,
- Mp = Hphp = 82,800 1b,
- MT HThT = 11, 300 1b,
- MG = HGhG = 30, 500 1b,
- $M_0 = M_D + M_T + M_G = 124,600$  in-1b,
- $S_{\rm T} = 80,000$  pai,
- K = A/B = 1.90.

From Figure UA-51, 1 of Peference (1), Y = 3, 3.

Then,

$$t = \sqrt{\frac{Y M_o}{B S_T}} = 1,20 \text{ in.}$$

Flexibility and Creep

From the equations on pages 18-20,  $F_B = 1,78/E_T$ ,  $F_F = 1,51/E_T$ ,  $K_B = 0.157C_1P_1^{4,82}$ ,  $o_{Ft} = 0.215P_t$ ,  $K_F = 2.405 \times 10^{-3}C_1P_t^{4,82}$ ,

r<sub>F</sub> = 0,849 , r<sub>K</sub> = 0,0153,

#### Bolt Relaxation

From Equations (8) and (9),

R = 1,82.

$${}^{O}_{BT} = 87,300 \left(\frac{24 \times 10^6}{32 \times 10^6}\right) = 65,400 \text{ psi,}$$

$${}^{O}_{O} = {}^{O}_{BT} = \frac{E_{T}C_{2} o_{O}^{m}}{R} \quad . \tag{24}$$

Solving Equation (24) by trial and error,  $\sigma_0 = 52$ , 100 pei. It is convenient to solve Equation (24) for various values of R in order to construct Figure 15, thus eaving considerable calculating time when optimizing the flange geometry. For a rigid flange (R = 1),  $\sigma_0$  would equal 46,000 pei.

$$18 \begin{bmatrix} \left(\frac{O_1}{G_0}\right)^{n-1} \\ -\frac{O_1}{O_1} \end{bmatrix}$$
 (25)

With 0<sub>0</sub> = 52, 100 psi, the successive solution of Equation (25) results in the residual balt-land surve of Figure 13.

### Lookage Pressure

In order to convert residual bolt load to loakage pressure assume a factor of eafery F, S, = 0, 0 on the residual load, establishing the design surve of Figure 13, For example, if it is assumed that loakage will occur when the gasket pressure reaches a value of 3p, Equation (11) becomes

The solution of Equation (26) for various values of the internal pressure p results in the leakage pressure versus time curve  $(K_A = K_M = 0, \Delta T + 0)$  of Figure 14. For example, at p = 2600,  $P_{\chi}^{-2} = 33,900$  lb. The time t from Figure 13 for 33,000 lb is 4,9 hours.

### Effect of Initial Strees

The relaxition equation for the geometry of Figure 11 is

$$t = 1,55 \times 10^{10} \left[ \frac{1 - \left(\frac{q_{e}}{q_{e}}\right)^{n-1}}{\sigma_{e}^{n-1}} \right]$$
 (27)

The effect of varying the balt stress  $\sigma_0$  in Equation (87) is shown in Figure 16.

#### istegral-Type Bolted Flange

### ASME Code Design®

An integral-type flange was designed in accordance with Reference (1). The gashet geometry, moment arm, outer flange radius, tube geometry, and design boil load are the same as for the loose-type flange of Figure 11. The integral-type flange geometry is shown in Figure 12:

 $h_T = 0,868$  in.,  $h_G = 0.663$  is.,  $h_D = 0.863$  in.

From page A-2,

W = 164,900 lb,
$H = 0.785 G^2 P = 119,000 lb,$
H <sub>D</sub> = 0, 785 <b>B<sup>2</sup>P</b> = 74, 500 3b,
HG = W-H = 45,900 1b,
H <sub>T</sub> = H-H <sub>D</sub> = 44,500 lb,
$M_D = H_{Dh_D} = 64,206$ lb,
$M_{T} = H_{T}h_{T} = 38,600$ lb,
MG = MGhG = 30, 500 lb,
$M_{\phi} = M_{D} + M_{T} + M_{C} = 133,300 \text{ in-1b},$
Using an allowable stress of 80,000 psi and a design moment of 133,300 in-1b, the ness of an integral-type flange is calculated from Reference (1) to be 0,95 in.

#### Floxibility and Groop

From Equations (2) and (5),

PB = 1,35/ET.

Since the finage design stress of 80,000 pei is the same as that of the equivalent loose-type flange, the flange croop rate is assumed to equal that of the loose-type flange:

 $X_T = 2,405 \pm 10^{-3}C_1P_1^{4,83}$ 

FF = 1,43/ET.

The flamibility of the integral-type flange from Equation (4) is

Thes

ry = 1.06, r K = 0.0204, R = 2,02.

#### Bolt Relaxation

From Figure 15, do = 52,900 pei. The bolt relat  $t = 1, 69 \times 10^{14}$   $\left| \frac{1 - \left(\frac{q}{q_0}\right)^2}{\frac{1}{q_0}} \right|$ 

Equation (28) does not yield revalts significantly different from Equation (25) for a losss-type flangs,

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**Threaded Connector** 

Figure 17 shows a threaded-connector geometry, which represents an arbitrary of dimensions rather than an optimum static design. The design procedure is used to calculate the dusign life of the geometry of Figure 17, based on René'41 at 1500 F. Figure 9 is a guide for the proper dimensions to use in the analysis. uto an arbitrary sot

### Bolt Flexibility

$$A_B = \pi (1, 05^2 - 0, 90^2) + 0, 91 in, 2,$$

•

The axial flexibility of the bolt is

$$\mathbf{F}_{\mathbf{B}} = \frac{\mathbf{E}_{\mathbf{B}}}{\mathbf{A}_{\mathbf{B}}\mathbf{E}_{\mathbf{T}}} = 1,26/\mathcal{Z}_{\mathbf{I}}$$

 $= \mathbf{F}'_{\mathbf{B}} = \mathbf{e}^{2} \left(\frac{\theta}{\mathbf{M}}\right) = 1, 14/\mathbf{E}_{\mathbf{T}},$ 

# = 19,8/ET,

The bending flexibility of the bolt is

• •

where

and

$$\frac{1.05+0.90}{2} = \frac{0.86+0.60}{2} = 0.24$$

Flange Flexibility

$$\begin{split} & L_{F} = L_{B} = 1, 15 \text{ in}, ; \\ & A_{F} = \pi (0, 84^{2} - 0, 50^{2}) = 1, 54 \text{ in}, ^{2}, \\ & F_{F} = \frac{L_{F}}{A_{F}E_{T}} = 0, 75/E_{T}. \end{split}$$

Bolt Creep

The axial croop of the bolt is  

$$K_B = \frac{C_1 P_t^{\ B} L_B}{A_B^{\ B}} = 1.62 C_1 P_t^{\ 4.62}.$$

In order to obtain the bending creep rate of the bolt, the stresses for an inwardly projecting flange are calculated from Reference (15). The critical combination of these stresses by the ASME Boller Code $^{(1)}$  is

$$\frac{\mathbf{S}_{\mathbf{H}} + \mathbf{S}_{\mathbf{T}}}{\mathbf{z}} = 9, 18 \text{ M},$$

For a maximum design stress equal to  $2/3 \, c_y = 80,000$  psi, the design moment equals 80,000/9, 18, or M = 8710 in-1b. With a moment arm a = 0, 24 in., this corresponds to an initial mut design load of 36,300 lb. In this case, the initial load of 36,300 lb is not limited by the axial stress in the nut, but by the stress in the inwardly projecting flange.

The thickness of an equivalent loose-type flange is determined from Reference (1) using the basic geometry of the inwardly projecting flange of Figure 17. The equivalent thickness equals 0.575 in. The creep rate of the equivalent loose-type flange, calculated from Equation (6), is

$$K'_{B} = \frac{2\sigma_{Ft}^{B}C_{1}ec_{F}}{h} = 1.26 C_{1}P_{t}^{4.82}$$
,

where

Flange Groep

$$K_{F} = \frac{C_{1}P_{t}^{n}L_{F}}{A_{F}} = 0,145 C_{1}P_{t}^{4,82}.$$

Flexibility and Creep Ratios

$$r_F = 0.595$$
,  $r'_F = 0.905$   
 $r_K = 0.0796$ ,  $r'_K = 0.693$ 

$$L = \frac{1 + r_{\rm F} + r'_{\rm F}}{1 + r_{\rm M} + r'_{\rm H}} = 1,41.$$

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(28)

..... The initial bolt stress is

$${}^{0}_{BR} = P/A_{B} = 34,300/3,91 = 40,000 pei,$$
  
 ${}^{0}_{BT} = 40,000 \left(\frac{E_{T}}{E_{R}}\right) = 30,000 pei.$ 

From Equation (9),

Nut Relaxation

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From Equation (10),

$$t = 1.16 \times 10^{10} \left[ \frac{1 - \left(\frac{\sigma_{e}}{\sigma_{0}}\right)^{n-1}}{\sigma_{e}^{n-1}} \right] .$$
 (29)

The relaxation curve is shown in Figure 18.

 $\mathbf{P}_{t}^{\prime}$ 

#### External Loads

It is assumed that  $K_A = K_M = 0.50$  in this example. The leakage criterion used in Equation (26) is used in Equation (19) to calculate a leakage prossure versus time curve. For example (for the connector shown in Figure 11), at a pressure p of 2000 psi, from Equation (26),

$$P_p + P_G = 33,000$$
 16.

From Equation (20),

### PA = 7440 16, PM = 5920 16.

Then, from Equation (19),

The design curve of Figure 13 gives t = 0, 37 hour. The results are shown in Figure 14 (KA = KM = 6, 50,  $\Delta$  T = 0).

#### Bolt-Flange Temperature Differential

It is assumed that at the beginning of the cooling-down cycle, the bolt assembly is hotter than the flange assembly, or

AT = -100 F.

From Equation (21) and the geometry of Figure 11,

$$\Delta \sigma_{\mathbf{B}} = \frac{a(\Delta T)\mathbf{E}_T}{1 + r_F} = 11,000 \text{ psi.}$$

The reduction in bolt load due to the thermal gradient is

 $\mathbf{P}_{\mathbf{T}} = (11,000)(1.09) = 20,000 1b.$ 

 $\mathbf{P}_{t}^{\prime} = \mathbf{P}_{p} + \mathbf{P}_{G} + \mathbf{P}_{T}.$ 

From Equation (23),

From Equations (26) and (30), at a pressure p = 1500 psi,

 $P'_{2} = \pi(1500)(1,95^{2}) + 3(1300)(1.53) + 20,800 = 45,580 16.$ 

From Figure 13, t = 0.52 hour.

The results of the analysis are shown in Figure 14 (K<sub>A</sub> = K<sub>M</sub> = 0,  $\Delta T$  = -100 F).

#### APPENDIX B

#### DERIVATION OF EQUATIONS

Flange Flexibility

#### Loose Type

Timeshenke (18) gives the flange rotation \$7 due to & flange moment M as

$$\theta_{\rm F} = \frac{1254}{2\pi E_{\rm T}h^3 \ln d_{\rm F}/c_{\rm F}}$$

and the flangs deflection at the bolk circle is

then

$$F_{\rm F} = \frac{10 g}{p^2} = \frac{3.82a^4}{{\rm S_Th^3 \, in \, 4g/a_F}} \,. \tag{33}$$

Integral Type

Wasstrom and Bergh(17) present an equation for the flange rotation:

$$\theta_{\rm F} = \frac{0.645 \text{ VM}}{L \sqrt{c_{\rm F}} (4_{\rm T} - c_{\rm T})^{5/2} E_{\rm T}}.$$
 (34)<sup>6</sup>

Since the total flange moment M = Pe, and the deflection  $\delta_{\Gamma} = \theta_{\Gamma} e_{i}$ 

$$F_{F} = \frac{2\delta_{F}}{P} = \frac{1.29 \text{ Ve}^{2}}{L \sqrt{c_{F}} (d_{T} - c_{T})^{5/2} E_{T}}$$
(35)\*

Boit Creep

From the steady-state crosp law,

$$\epsilon_{\mathbf{B}} = C_1 a_{\mathbf{B}}^n t$$
, (36)

$$\delta_{\mathbf{B}} = \frac{C_1 \mathbf{P}_t^n \mathbf{L}_{\mathbf{B}}}{\mathbf{A}_{\mathbf{B}}^n}, \qquad (37)$$

or, the bolt deflection is

$$K_{\mathbf{B}} = \frac{\delta_{\mathbf{B}}}{t} - \frac{C_{1}P_{t}^{n}L_{\mathbf{B}}}{A_{n}^{n}}$$
(38)

### Flange Greep

 $a_{B} = \frac{P_{1}}{A_{P}}$ 

Marin<sup>(19)</sup> developed equations for the steady-state creep of a circular ring of rectangular cross section subjected to uniform moments. The flange rotation and maximum stress on the inner surface are . . . . . 18 .

$$\Theta_{\mathbf{F}} = \left[ \left( \frac{\mathbf{P}_{\mathbf{1}} \bullet}{2\pi} \right) \frac{\left( z^{1+\frac{1}{m}} \right) \left( 1 - \frac{1}{m} \right) \left( z + \frac{1}{m} \right)}{\left( d_{\mathbf{F}}^{1-\frac{1}{m}} - c_{\mathbf{F}}^{1-\frac{1}{m}} \right) h^{2+\frac{1}{m}}} \right]^{-1} G_{\mathbf{1}} \mathbf{t} , \qquad (39)$$

$$\sigma_{\mathbf{F}\mathbf{t}} = \left( \frac{\mathbf{h}}{2c_{\mathbf{F}}} \right)^{\frac{1}{m}} \left( \frac{\mathbf{P}_{\mathbf{1}} \bullet}{2\pi} \right) \left[ \frac{\left( z^{1+\frac{1}{m}} \right) \left( 1 - \frac{1}{m} \right) \left( z + \frac{1}{m} \right)}{\left( d_{\mathbf{F}}^{1-\frac{1}{m}} - c_{\mathbf{F}}^{1-\frac{1}{m}} \right) h^{2+\frac{1}{m}}} \right] . \qquad (40)$$

Equations (39) and (40) are based on the steady-state creep law  $c = C_0 O^n$  and assume the creep stress distribution in the flange. There exists sufficient test and theoretical data on the creep bending of beams<sup>(20)</sup>, 21, 22, 23, 24) to justify that:

(1) Plane sections remain plane.

- (2) The creep deflections can be predicted on the basis of the steady-state creep law.
- (3) Individual fibers of the beam creep at a rate equivalent to that predicted by a tensile creep test.
- (4) The creep stress distribution is attained shortly after the commencement of the test and does not change for the remainder of the test. This is analytically confirmed by Hult<sup>(24)</sup>. Because of the like nature of the beam and ring deformations, the above assumptions are also assumed to hold true for the creep de-flection of a flange.

From Equations (39) and (40).

$$\theta_{\mathbf{F}} = \frac{2\sigma_{\mathbf{F}_1}^n C_1 c_{\mathbf{F}_1}}{h} . \tag{41}$$

"The terms V and L to Equations (34) and (38) runfarm to the sumancipture of Refer

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(31)

(30)

The

where

Then.

$$K_{F} = \frac{2e\theta_{F}}{t} + \frac{4\sigma_{F_{t}}^{n} C_{1}ee_{F}}{h}.$$
 (42)

### Intercopt Stress Reduction

It is assumed that the initial stress  $\sigma_{\rm BT}$  is reduced in zero time to a new value  $\sigma_0$  which has a total intercept strain  $c_0$  equal to the initial bolt stress  $\sigma_{\rm BT}$  divided by the modulus of elasticity ET. From Figure 19,

$$\sigma_{\mathbf{BT}} = \sigma_{\mathbf{0}} + \mathbf{E}_{\mathbf{T}} C_2 \sigma_{\mathbf{0}}^{\mathbf{m}} . \tag{44}$$

The constants C2 and m are determined by a best fit to the intercept data from a family of creep curves at the design temperature.

For a flexible flange, the interaction of the flange and bolt has to be accounted for. The primary creep of the bolt can be represented by

$$\delta_{\mathbf{B}\mathbf{c}} = C_{\mathbf{z}}\sigma_{\mathbf{0}}^{\mathbf{m}} L_{\mathbf{B}}. \qquad (45)$$

The change in bolt stress is determined by the elastic bolt recovery:

$$\sigma_0 = \sigma_{BT} = \frac{\delta_{B0} E_T}{L_B} . \tag{46}$$

From Figure (20), in order to maintain compatibility of bolt and flange,

$$\delta_{\text{TOTAL}} = \delta_{\text{Fe}} = \delta_{\text{Fe}} = \delta_{\text{Be}} = \delta_{\text{Be}} . \tag{47}$$

By definition,

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$$r_{\mathbf{F}} = \frac{\partial \mathbf{F} \mathbf{a}}{\partial \mathbf{B} \mathbf{a}}, \quad r_{\mathbf{K}} = \frac{\partial \mathbf{F} \mathbf{g}}{\partial \mathbf{B} \mathbf{c}}$$
$$\mathbf{R} = \frac{1 + r_{\mathbf{F}}}{1 + r_{\mathbf{K}}},$$

and Equation (17) becomes

$$\delta_{\mathbf{B}u}(\mathbf{r}_{\mathbf{F}}+1) = \delta_{\mathbf{B}c}(\mathbf{r}_{\mathbf{K}}+1),$$
 (40)

$$\frac{\delta_{B,g}}{\delta_{B,c}} = \frac{1 + r_{B'}}{1 + r_{F'}} = \frac{1}{R}.$$
(49)

Substituting (45) and (46) in (49).

$$c_0 = s_{BT} - \frac{E_T C \rho_0}{R} , \qquad (50)$$

For a rigid flange (R = 1), Equation (50) is identical to Equation (44). The development of Equation (50) for a ilexible flange assumes that the creep ratio  $r_{\rm K}$  is the same for primary creep as it is for steady-state creep.

### Bolt Relaxation

The analysis of bolt relaxation is similar to that of References (2, 6). The enclature is defined as follows:

Ly - original length between nut bearing surfaces

δ' • L<sub>p</sub> - L<sub>p</sub> = initial flange deflection plus bolt extension (or total initial relative movement between nut and bolt)

 $K_{F}t = \delta_{FC} = aroup deflection of flanges$ 

 $K_{Bt} = \delta_{Bc} = creep deflection of bolte$ 

 $\mathbf{F}_{\mathbf{F}}\mathbf{P} = \delta_{\mathbf{F}_{\mathbf{F}}}$  = elastic flange deflection

FBP + oBe = elastic bolt deflection.

# In order that the flange and belts maintain contact at any time,

 $\mathbf{L}_{\mathbf{F}} = \delta_{\mathbf{F}\mathbf{G}} = \delta_{\mathbf{F}\mathbf{G}} + \mathbf{L}_{\mathbf{B}} + \delta_{\mathbf{B}\mathbf{G}} + \delta_{\mathbf{B}\mathbf{G}} \,,$ 

 $\delta' = \delta_{\mathbf{Be}} \left( \mathbf{i} + \mathbf{r_{K}} \right) + \delta_{\mathbf{Be}} \left( \mathbf{i} + \mathbf{r_{F}} \right).$ 

...

$$\delta' = \delta_{\mathbf{B}\mathbf{c}} \left(1 + \mathbf{r}_{\mathbf{K}}\right) + \frac{\delta_{\mathbf{B}} \mathbf{I}_{\mathbf{T}}}{\mathbf{E}_{\mathbf{T}}} \left(1 + \mathbf{r}_{\mathbf{F}}\right),$$

 $\delta_{Be} = \frac{\delta_{B} L_{B}}{E_{T}} ,$ 

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$$\sigma_{20} = \frac{E_T}{L_B} \left[ \frac{\delta' - \delta_{BC} \left(1 + r_K\right)}{1 + r_T} \right].$$
  
Differentiate (53) with respect to time:

$$\frac{d\sigma_{B}}{dt} = \frac{E_{T}}{L_{B}} \left( \frac{1 + r_{H}}{1 + r_{F}} \right) \frac{d\delta_{BC}}{dt}$$

From the steady-state croop law,

$$\dot{\epsilon}_{c} = C_{1}\sigma_{B}^{h} + \frac{d\epsilon_{c}}{dt} + \frac{d}{dt} \left(\frac{\delta_{B}c}{L_{B}}\right),$$

$$\frac{a\sigma Bc}{dt} = L_B C_1 \sigma_B^{n} .$$
(55)

(53)

(54)

Substitute (55) in (34);

$$\frac{d\sigma_{B}}{\sigma_{B}} = \frac{-E_{T}C_{1}dt}{R} .$$
(36)

Integrate (56), letting  $\sigma_{\mathbf{B}} = \sigma_{\mathbf{t}}$ :

$$\frac{\sigma_t^{-n+1}}{1-n} = \frac{-E_T C_1 t}{R} + C,$$
 (57)

where C is a constant of integration. At t = 0,  $\sigma_t = \sigma_p$ . Therefore,  $C = \frac{\sigma_0^{-n+1}}{n}$ 

From (57),  

$$t = \frac{R}{C_1 E_T (n-1)} \left[ \frac{1 - \left(\frac{2t}{\sigma_n}\right)^{n-1}}{\sigma_t^{n-1}} \right].$$
(58)

Equation (58) also applies to the relaxation of the nut in a threaded connector design with R defined by Equation (18).

#### External Loads

#### Anial Load

The axial load  $P_A$  is assumed to be a tensile force which reduces the gasket pressure. The magnitude of the tube stress  $\sigma_A$  due to the axial load  $P_A$  is defined as a constant times the axial tube stress due to internal pressure:

$$\mathbf{P}_{\mathbf{A}} = \sigma_{\mathbf{A}} \mathbf{A}_{\mathbf{T}} = \sigma_{\mathbf{A}} \pi \left( \mathbf{d}_{\mathbf{T}}^2 - c_{\mathbf{T}}^2 \right), \qquad (59)$$

where

$$\sigma_{\mathbf{A}} = \mathbf{K}_{\mathbf{A}} \sigma_{\mathbf{T}} = \frac{\mathbf{p} \mathbf{K}_{\mathbf{A}} \mathbf{c}_{\mathbf{T}}^{2}}{\left(\mathbf{d}_{\mathbf{T}}^{2} - \mathbf{c}_{\mathbf{T}}^{2}\right)} \quad . \tag{60}$$

$$P_{A} = \pi p K_{A} c \frac{2}{T}$$
 (61)

### Bending Moment

The required additional bolt load  $P_M$  accounts for the tendency of the tube bonding moment  $M_T$  to reduce the gasket pressure. The magnitude of the tube stress  $\sigma_M$  due to the bending moment  $M_T$  is defined as a constant  $K_M$  times the axial tube stress due to internal pressure:

$$\sigma_{\mathbf{M}} = \mathbf{K}_{\mathbf{M}} \sigma_{\mathbf{T}} = \frac{\mathbf{p} \mathbf{K}_{\mathbf{M}} c_{\mathbf{T}}^{2}}{\left( \mathbf{d}_{\mathbf{T}}^{2} - c_{\mathbf{T}}^{2} \right)} .$$
 (62)

From beam bending theory,

From (59) and (60),

$$\sigma_{M} = \frac{4MT^{d}T}{\pi(dT^{4} - cT^{4})}$$
 (63)

Combining (62) and (63),

$$M_{T} = \frac{\pi_{P} K_{M} c_{T}^{2} (d_{T}^{2} + c_{T}^{2})}{4d_{T}}$$
 (64)

The bending moment  $M_T$  is considered to reduce the gasket pressure by the amount  $G_0$  on one side of the gasket. If the gasket width is assumed small compared to the gasket radius,

$$a_{\rm G} = \frac{M_{\rm T}}{\pi g^2 w} = \frac{\mu K_{\rm M} c_{\rm T}^{-2} (a_{\rm T}^2 + c_{\rm T}^{-2})}{4 a_{\rm T} g^2 w}$$
 (65)

If the maximum gashet pressure is assumed to act on the complete gashet surface,

$$P_{M} = A_{C} a_{C} = 2\pi g_{W} a_{C} = \frac{m \mu K_{M} c_{T}^{2} (a_{T}^{2} + c_{T}^{2})}{24 \pi g} . \qquad (66)$$

(51)

(52)

### Bolt-Flange Temperature Differential

The thermal bolt strain  $\Delta c_B = \alpha(\Delta T)$  corresponds to a thermal bolt deflection  $\Delta L_{s_B} = \alpha(\Delta T)L_{s_B}$  which is distributed according to the flexibility of the flange and bolt. From Figure 30,

$$a(\Delta T)L_{B} = \delta_{Fe} + \delta_{Be} = \delta_{Be} (r_{F} + \epsilon).$$
 (67)

The change in bolt stress is

From (67) and (68),

$$\Delta \sigma_{\mathbf{B}} * \left( \frac{\delta_{\mathbf{B}} *}{L_{\mathbf{B}}} \right) \mathbf{E}_{\mathbf{T}} \quad . \tag{66}$$

$$\Delta \sigma_{\mathbf{B}} = \frac{\alpha(\Delta T) \Xi_{\mathbf{T}}}{\frac{1}{2} + r_{\mathbf{F}}} . \tag{69}$$

Following a similar procedure for the threaded connector,

$$\Delta \sigma_{\mathbf{B}} = \frac{\alpha(\Delta T) \mathbf{E}_{\mathbf{T}}}{1 + \mathbf{r}_{\mathbf{F}} + \mathbf{r}_{\mathbf{F}}^{T}}$$
(70)

#### APPENDIX C

#### SECONDARY REPECTS

### Stress Concentrations

Because of the accelerated rate of creep at higher stress levels, the presence of Because of the accustorised rate of creep at higher stream iteres is the presence of stress concentrations such as bolt threads might be expected to increase the over-all creep rates of the assembly. However, the results of the British Flange Tests(9) indicate that the creep of the nut assembly was not exceedingly high if the nut material was equivalent to the bolt material. However, when carbon resen nuts were used with alloy steel bolts, the creep of the nut assembly was exceeding.

A comprehentive review of stress concentrations under creep conditions was presented in Reference (23). The strength of a specimen with a notch under creep of ditions is attributed partly to the ability of the material in the notch area to flow ray and redistribute stresses at high temperatures; also because the greater volume of lower stressed material away from the notch restrains the over-all deformations. rapidly

#### **Creep Bending of Bolt**

The tendency of the bolt in a flanged joint assembly to develop bending moments due to flange rotations and shifting of the nut reaction is well known. Under creep con-ditions, however, the stress distribution becomes more favorable beccuss of the re-distribution of the bending stresses. Balley(3) developed a solution, based on the steady-state croep law, for the creep stresses due to combined axial and bending loads on a rectangular croes section: the results are shown in Figure 21. The results for a solid circular section should be comparable because of the relative values of the plastic bending factor for the rectangular and solid circular sections.

#### Dynamic Creep

Superimposed alternating stresses are known to affect the creep rate of metals. However, Lassn<sup>24</sup>) showed that the presence of alternating stresses did not appear to affect greatly the creep rate for most materials. Results for Waspain with 500 F from Reference (27) are shown in Figure 22.

Langenecker(28) has recently discussed a problem of possible significance to connectors operating in a high acoustic radiation field. He found that intense ultra-sound could have a marked effect on the physical properties of solids,

### Gasket Creep\*

The gasket will contribute a certain amount of creep and flexibility to the connector assembly. This contribution is neglected in the design procedure in view of the rela-tively short length of the gasket in most designs. Also, for metallic gaskets, the streases in the gasket are usually lower than those of the bolt assembly. In fact, the gasket area should be cheesen, on the basis of the relative creep strengths of the gasket and bolt material, such that the creep rate of the gasket is negligible. The inclusion of the gasket creep and flexibility properties in the design procedure will unduly complicate the equations, with very little gain in design accuracy for most geometries.

### Flange Rotations

Changes in the flange rotation and corresponding bolt-load changes due to the application of internal pressure are neglected in the design procedure. These effects

The design procedure could be excended to seconds the offects of gashes crosp in an approximate, but quite conservative

re considered secondary from the standpoint of over-all bolt relaxation, but should be unsidered in the static design of the connector if found to be significant.

APPENDIX D TUBE DESIGN

The tube design is considered on page 18 only to the extent that it affects the strength of the connector. The tube, however, is subjected to progressive creep de-formations with each operating cycle. The tube thickness should estify the minimum requirements for a creep design. Some of the appropriate theories avails" is for the tube creep design are reviewed below.

Finnie and Heller<sup>(7)</sup> present the equation for the tangential strain rate in a thin-walled tube with closed ends under internal pressure:

- • •

$$\dot{\epsilon}_{\phi} = C_1 \left(\frac{\mathbf{p} \mathbf{c}_T}{\mathbf{d}_T - \mathbf{c}_T}\right)^n \left(\frac{3}{4}\right)^{\frac{n+1}{2}} = \dot{\epsilon} \left(\frac{3}{4}\right)^{\frac{n+1}{2}} , \qquad (71)$$

where  $\dot{c} = C_1 \sigma^0$  = the strain rate which would be produced by the tangential stress

acting alone in simple tension.

The creep of thin-walled tubes under internal pressure, axial loads, and moments is considered in Reference (7),

Finnie and Heller<sup>(7)</sup> also analysed the creep of thick-walled tubes subjected to internal pressure. Based on the Mises flow law and the steady-state creep law  $\ell = C_1 \sigma^n$ , the radial, tangential, and axial stresses, respectively, are

$$\sigma_{r} = -p \frac{\left(\frac{d_{T}}{T}\right)^{2/n}}{\left(\frac{d_{T}}{c_{T}}\right)^{2/n} - 1},$$

$$\sigma_{\phi} = p \frac{\left(\frac{2-n}{n}\right) \left(\frac{d_{T}}{T}\right)^{2/n} + 1}{\left(\frac{d_{T}}{c_{T}}\right)^{2/n} - 1},$$

$$\sigma_{g} = p \frac{\left(\frac{1-n}{n}\right) \left(\frac{d_{T}}{T}\right)^{2/n} - 1}{\left(\frac{d_{T}}{c_{T}}\right)^{2/n} - 1},$$

(72)

where  $r = variable radius, c_T \leq r \leq d_T.$ 

Equations (72) reduce to the well-known Lamé formulas for n = 1. The tangential strain rate is .

$$\dot{\epsilon}_{\phi} = C_1 \left(\frac{3}{4}\right)^{\frac{n+1}{2}} \left[ \frac{p}{\left(\frac{d_T}{c_T}\right)^{2/n}} - \frac{(2)}{n} \right]^{\frac{n}{2}} \left(\frac{d_T}{r}\right)^2 .$$
(73)

The steady-state creep stresses and strain rates in a thick-walled tube under com-bined internal pressure and axial load are presented by  $\beta$  innis<sup>(29)</sup>. Numerical solutions to the equations are tabulated for various values of the variables. In the case where the additional axial load is either small or large compared to the axial load due to internal pressure, simple approximate solutions are given. Finnis<sup>(30)</sup> also pointed out the lack of complete agreement between the theoretical and experimental studies on multiaxial creep in tubes, attributing the discrepancies to the effect of hydrostatic stress which is usually assumed negligible in creep strain predictions.

A method for predicting the creep failure time for either thin-walled or thick-walled circular cylinders can be found in Reference (31). True stress and true strain are used and the steady-state creep law is assumed valid until failure. The failure time is based on plastic instability, or the time at which the vessel wall is no longer suf-ficient to hole the load. Simplified formulas are obtained for both thin-walled and verythick-walled cylinders.

Creep-rupture tests<sup>(32)</sup> were run on Type 316 stainless steel tubes at high tem-peratures and pressures. The results of the tests compare favorably with those of uni-smal tension tests if effective stress and effective strain rate are used. Rattinger and Padlog<sup>(33)</sup> also present a method of analysis for predicting creep rupture of cylindrical shells using conventional uniaxial creep data.

A method of analysis<sup>(34)</sup> for predicting the stresses and strains in thick-walled cylinders considers primary as well as secondary creep. The snalysis is quite com-plicated and justifies a computer solution,

Equations (71) and (73) result in quite favorable strain rates for higher values of the exponent n. On this basis, the design procedure of the ASME Boiler Code<sup>(1)</sup>, whic compares the maximum design stress to the creep rate or rupture stress of a unianial tension test, provides for a safe design. which





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FIGURE 1. TYPICAL CREEP CURVE

FIGURE 5. MASTER CREEP CURVE FOR RENE 41 BAR





FIGURE 8. TYPICAL FLANGE GROMETRY





A-41978 FIGURE 10. INWARDLY PROJECTING FLANGE









FIGURE 12. INTEGRAL-TYPE FLANGE GEOMETRY (RENÉ 41 + 1500 F)







FIGURE 14. LEAKAGE PRESSURE VERSUS TIME FOR BOLTED-FLANGED CONNECTOR (SHOWN IN FIGURE 11)

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FIGURE 16. EFFECT OF INITIAL BOLT STRESS ON RELAXATION (FLANGE GEOMETRY OF FIGURE 11)



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FIGURE 19. INTERCEPT STRESS REDUCTION





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FIGURE 20. CREEP OF FLANGE AND BOLT

FIGURE 21. CREEP BENDING OF RECTANGULAR SECTION (n = 6)





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# APPENDIX B

### COMPUTER PROGRAM FOR DESIGNING AFRPL FLANGED CONNECTORS

This appendix is intended for those who plan to use the computing programs IRFDP and LRFDP for designing flange connectors of the types shown in Figure 67 of this report. Program IRFDP is used to design connectors containing two integral flanges, and Program LRFDP is used to design connectors containing two loose-ring flanges. By appropriately controlling certain input values of both programs, a design can be obtained for a connector containing one integral flange and one loose-ring flange.

A detailed description is given of the input and output sections of Program IRFDP. Program LRFDP is similar to IRFDP, and the difference is discussed. Also, the integral/loose-ring connector-design approach is described.

Programs IRFDP and LRFDP were written in FORTRAN IV language for the CDC 6400. A list of both the programs is included. It is believed that the programs can easily be converted to run on a different machine with a FORTRAN IV compiler.

## Item-by-Item Description of Input Data for IRFDP

The computer input cards that are necessary to perform the design using IRFDP will be described in this section. The numbers refer to the card numbers as read by the program. The following FORMATS were used for the input:

1020 FORMAT (8F10.0)

1021 FORMAT (8F10.4)

1022 FORMAT (8F10.9)

1023 FORMAT (2110)

# Card 1

Read 1020, TBMI

TBMI = Tube bending moment input, lb-in.

Set TBMI = 0.001 if the value of the bending moment is to be selected internally according to Equations (8) and (9) on page 31.

Card 2

Read 1021, TOD, TWI

TOD = Tube outside diameter, in.

TW1 = Tube wall thickness input, in.

If  $TWI \le 0.001$ , then the tube wall thickness is internally selected by Equation (6) on page 29. In the final program, Equation (6) also includes tube design for impulse-pressure fatigue.

### Card 3

Read 1020, TU, TY, TF, TR, TFRO

- TU = Ultimate tensile strength of tube material at maximum operating temperature, psi
- TY = Tensile yield strength of tube material at maximum operating temperature, psi
- TF = Fatigue strength for R = 1.0 of tube material at maximum operating temperature, psi
- TR = 2 Year-Rupture strength of tube material at maximum operating temperature, psi
- **TFRO** = Fatigue strength for R = 0.0 of tube material at maximum operating temperature, psi.

### Card 4

Read 1021, BCDI, HUBII

BCDI = Bolt circle diameter input, in.

If  $BCDI \le 0.001$ , the bolt circle diameter is calculated, also if  $BCDI \le 0.001$ , II = 0 and BNI = 0.0 (See Card 30 and Card 31)

HUBII = Hub reinforcement height, in.

If HUBII < 0.001, the reinforcement height is calculated by the program.

Card 5

Read 1021, GIOTWI

GlOTWI = Maximum allowable hub reinforcement ratio,

where

$$G10TWI = \frac{(TW + HUBI)}{TW}$$

where

TW = tube wall thickness, in.

HUBI = Selected hub height, in.

Set G10TWI = 3. to 5.

# Card 6

Read 1021, FRI, FSFI

FRI = Radius input for fillet at junction of flange and hub reinforcement, in. If FRI = 0.0, the radius is computed by the program. It is recommended that a value no smaller than that computed be used.

FSFI = Factor of safety used in fatigue evaluation. A factor of 2.0 was chosen for all designs.

# Card 7 - Card 9

Read, 1020, FUR

Read, 1020, FU, FY, FSU, FCY, FE, FG, FYC, FYR

Read, 1022, FRHO, FMU, FTE, FCR

- FUR = Ultimate tensile strength of flange material at room temperature, psi
  - FU = Ultimate tensile strength of flange material at maximum operating temperature, psi
  - FY = Tensile yield strength of flange material at maximum operating temperature, psi
- FSU = Ultimate shear strength of flange material at maximum operating temperature, psi
- FCY = Compressive yield strength of flange material at maximum operating temperature, psi

FE = Young's modulus of elasticity for flange material, psi

- FG = Shear modulus for flange material, psi
- FYC = Tensile yield strength of flange material at minimum operating temperature, psi
- FYR = Tensile yield strength of flange material at room temperature, psi
- **FRHO** = Flange material density, 1b/in.<sup>3</sup>

FMU = Poisson's Ratio for flange material

FTE = Coefficient of thermal expansion for flange material, in. /°F

FCR = Flange material creep rate.

# Card 10 - Card 13

Read 1021, (R[NN], NN = 1,8) Read 1021, (R[NN], NN = 9,11) Read 1020, (FF[NN], NN = 1,8) Read 1020, (FF[NN], NN = 9,11)

R = Stress Ratio - the algebraic ratio of the minimum stress to the maximum stress in the stress cycle. Eleven standard ratios are read in. They range from -1.0 to 1.0 in steps of 0.2.

FF = Corresponding fatigue strength for flange material at maximum operating temperature for a given number of cycles, psi. The values of FF are obtained from a modified Goodman diagram for the flange material.

# Card 14

Read 1021, SODI, SIDI, SELI, TAI, TAOTW, SLEGTI

SODI = Seal outside diameter input, in.

SIDI = Seal inside diameter input, in.

SELI = Seal tang length input, in.

TAI = Seal tang height input, in.

TAOTW = Initial ratio of tang height to tube wall thickness. A value of 0.9 to 1.0 is recommended. SLEGTI = Seal disk thickness, input, in.

Let SODI < 0.001 for the program to calculate seal dimensions.

## Card 15

Read 1020, RSLI, SSLI, MSLI

RSLI = Radial sealing load per inch of seal circumference, lb/in.

SSLI = Axial sealing load per inch of seal circumference, lb/in.

MSLI = Minimum axial sealing load per inch of seal circumference, lb/in.

### Card 16 - Card 17

Read 1020, SCY, SEE, SCYC, SCYR, SYR

Read 1022, STE

SCY = Compressive yield strength of seal material at maximum operating temperature, psi

SEE = Young's modulus of elasticity for seal material, psi

- SCYC = Compressive yield strength of seal material at minimum operating temperature, psi
- SCYR = Compressive yield strength of seal material at room temperature, psi

SYR = Tensile yield strength of seal material at room temperature, psi

STE = Coefficient of thermal expansion for seal material, in. / °F.

# Card 18 - Card 29

Read 1021, (BH[NN], NN = 1, 8)

Read 1021, (BH (NN), NN = 9, 15)

Read 1021, (BWC[NN], NN = 1, 8)

Read 1021, (BWC[NN], NN = 9, 15)

Read 1021, (BS[NN], NN = 1, 8)

Read 1021, (BS[NN], NN = 9, 15)

Read 1021, (BTA[NN], NN = 1, 8)

Read 1021, (BTA[NN], NN = 9, 15)

Read 1021, (NWT[NN], NN = 1, 8)

Read 1021, (NWT[NN], NN = 9, 15)

Read 1021, (DAF[NN], NN = 1, 8)

Read 1021, (DAF[NN], NN = 9, 15)

BH = Hole size for bolts, in.

BWC = Bolt wrench clearance, in.

BS = Nominal bolt size, in.

BTA = Bolt tensile area, in.  $^2$ 

NWT = Weight of a nut, lb.

DAF = Diameter across flats for bolt head, in.

Note: The order of reading the above bolt geometry for 15 different sizes is from the smallest size to the largest size of bolt.

# Card 30 - Card 31

Read 1023, II, IMAX

Read 1020, BNI

II = Index for selection of bolt from bolt table input

 $\mathbf{IMAX} = 16$ 

BNI = Number of bolts, input.

Note: See Card 4. If  $BCDI \leq 0.001$ , then II = 0 and BNI = 0.0

# Card 32 - Card 33

Read 1020, BY, BE, &U, BYC, BYR

Read 1022, BTE, BCR, BRHO

- BY = Tensile yield strength of bolt material at maximum operating temperature, psi
- BE = Young's modulus of elasticity of bolt material, psi
- BU = Ultimate tensile utrength of bolt material at maximum operating temperature, psi

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- BYC = Tensile yield strength of bolt material at minimum operating temperature, psi
- BYR = Tensile yield strength of bolt material at room temperature, psi
- BTE = Coefficient of thermal expansion of bolt material, in. / °F
- BCR = Bolt material creep rate
- BRHO = Bolt material density, 1b/in.<sup>3</sup>

# Card 34

Read 1020, P, PP, PB, PIM, TMAX, TMIN

P = Operating pressure, psi

PP = Proof pressure, psi

PB = Burst pressure, psi

PIM = Impulse pressure, psi

TMAX = Maximum operating temperature, F

TMIN = Minimum operating temperature, F.

# Card 35

Read 1020, DTC1, DTC2, DTH1, DTH2

- DTC1 = Thermal gradient at minimum temperature from seal or tube to flange, F
- DTC2 = Thermal gradient at minimum temperature from flange to bolt, F
- DTH1 = Thermal gradient at maximum temperature from seal or tube to flange, F
- DTH2 = Thermal gradient at maximum temperature from flange to bolt, F.

# Card 36

Read 1021, FS1, FS2, FS3

FS1 = Factor of safety for flange and seal material

FS2 = Factor of safety for bolt material

FS3 is not presently used.

For additional guidance, the actual cards used for designing 3-inch, 1500-psi integral-integral aluminum connectors are reproduced in Figures B-1 and B-2.

### Description of Computer Output for IRFDP

The interpretation of the output of Program IRFDP is given below. The value of each variable is printed directly below its symbolic definition. The output sheets obtained for the 3-inch 1500-psi integral-integral aluminum designs are reproduced on pages B-60 through B-62.

The input to the design is first printed out exactly in the same order as read in. This must be reviewed, as it provides a check to insure that the proper data were key punched. As an example, a value of 0.001 is printed under TBMI (for definition of symbols of input item, see pages B-2 through B-8), hence TBMI = 0.001 lb-in. The design output is then printed and consists of several parts. First the tube bending moment (TBM, lb-in.) and corresponding bending stress (TBS, psi) are printed and then the torque per bolt (TRQ, lb-in.) required to assemble the connector is printed.

The second part of the output consists of the calculated component loads for the seven design conditions as described on page 81 of this report. Also, refer to Figures 71 and 72. The definitions of the loads are as follows:

MO = Equivalent flange moment, lb-in
BL = Axial bolt load, lb
SL = Axial seal load, lb
RSL = Radial seal load, lb
MP = Flange moment due to pressure, lb-in.

MC = Flange moment due to creep, lb-in.

Hence MO1 means the equivalent flange moment under Condition 1, and so on. These definitions apply to all the seven design conditions. However, for Condition 2 and Condition 6, the above definitions are expanded and some new terms are defined as follows:

### **Condition 2**

MO2, BL2, SL2, RSL2 are component loads with creep under Condition 2.

MO2P, BL2P, SL2P, RSL2P are component loads under Condition 2 with no creep.

MO2M, BL2M, S12M, RSL2M are component loads under Condition 2 with no creep and a minus bending moment.

### Condition 6

MCTG6 = Flange moment due to cold thermal gradient, lb-in.

MHTG6 = Flange moment due to hot thermal gradient, lb-in.

MO6C, BL6C, et al., are component loads under Condition 6 at minimum temperature with creep.

MO6CN, BL6CN, et al., are component loads under Condition 6 at minimum temperature without creep.

MO6H, BL6H, et al., are component loads under Condition 6 at maximum temperature with creep.

MO6HN, BL6HN, et al., are component loads under Condition 6 at maximum temperature without creep.

The stress-concentration factor (KTF) used in the fatigue analysis is printed out. The next section of the output consists of stress values for the seven design conditions. In the following description, A means the value at the inside point of the junction of flange and tube and B implies the values are at the outside point. The other stress values are at the flange inside diameter. Also, the definitions for design conditions are as described in the load values printout.

EXA = Equivalent maximum stress at A, psi

ENA = Equivalent minimum stress at A, psi

EXB = Equivalent maximum stress at B, psi

ENB = Equivalent minimum stress at B, psi

SMAX A = Maximum stress in axial direction at A, psi

SMIN A = Minimum stress in axial direction at A, psi

FFC A = Allowable fatigue stress at A, psi

SMAX B = Maximum axial stress at B, psi

SMIN B = Minimum axial stress at B, psi

FFC B = Allowable fatigue stress at B, psi

SH = Flange axial stress due to bolt load or seal load, psi

SRS = Flange axial stress due to radial seal load, psi

SPR = Axial stress in flange due to pressure, psi

SPA = Axial stress in flange due to axial pressure load, psi

SB = Axial stress in flange due to bending load, psi

SR = Radial stress in flange, psi

ST = Tangential stress in flange, psi

BTS = Bolt tensile stress, psi.

The next section consists of the final dimensions of the design. These dimensions are shown in Figure B-3 and include:

H = Axial length of hub reinforcement, in.

HUBI = Hub reinforcement height, in.

G1 = Hub reinforcement height plus tube wall thickness, in.

FR = Fillet radius, in.

TW = Tube wall thickness, in.

TOD = Tube outside diameter, in.

A = Flange outside diameter, in.

B = Flange inside diameter, in.

XT = Flange thickness, in.

BCD = Bolt circle diameter, in.

SOD = Seal outside diameter, in.

SID = Seal inside diameter, in.

SEL = Seal tang length, in.

TA = Seal tang height, in.

SLEGT = Seal leg thickness, in.

BS(I) = Nominal bolt size, in.

BH(I) = Hole size, in.

BTA(I) = Bolt tensile area, in.<sup>2</sup>

BWC(I) = Wrench clearance, in.

BN = Number of bolts

WGT = Approximate weight of connector assembly up to and including hub reinforcement length, lb.

### Input Description for Program LRFDP

In the following description, the order of card input is 1 to 7, 8, 9a, etc.

Cards 1 to 3, Same as in IRFDP

Card 4, Read 1021, AI, HUBII,

where

AI = Stub flange diameter input, in. If AI = 0.00, then the program computes the diameter.

HUBII, same as in IRFDP.

Card 8a, Read 1021, FCYC, FCYR.

FCYC = Compressive yield strength of flange material at minimum temperature, psi.

FCYR = Compressive yield strength of flange material at room temperature, psi.

Card 9 - Card 33, Same as IRFDP

Card 33a, Read 1021, BCDI

BCDI = Bolt circle diameter input, in. If  $BCDI \leq 0.000$  then II = 0.0 and BNI = 0.0 (see Cards 30-31).

Card 33b - 33d

Read 1020, RU, RY, RSU, RCY, RE, RG, RYC, RYR

Read 1020, RCYC, RCYR

Read 1020, RTE, RCR, RRHO

RU = Ultimate tensile strength of ring material at maximum temperature, psi

RY = Tensile yield strength of ring material at maximum temperature, psi

RSU = Ultimate shear strength of ring material at maximum temperature. psi

RCY = Compressive yield strength of ring material at maximum temperature, psi

RE = Young's modulus of elasticity for Ring material, psi

RG = Shear modulus for Ring material, psi

- RYC = Tensile yield strength of ring material at minimum temperature, psi
- RYR = Tensile yield strength of ring material at room temperature, psi
- RCYC = Compressive yield strength of ring material at minimum temperature, psi

RCYR = Compressive yield strength of ring material at room temperature, psi

RTE = Coefficient of thermal expansion for ring material, in. / °F

RCR = Ring material creep rate

**RRHO** = Density of ring material, 1b/in.<sup>3</sup>

Card 34, Same as in IRFDP.

Card 35, Read 1020, DT1C, DT2C, DT3C, DT1H, DT2H, DT3H

DT1C = Thermal gradient at minimum temperature from seal or tube to flange, F
DT2C = Thermal gradient at minimum temperature from flange to ring, F
DT3C = Thermal gradient at minimum temperature from ring to bolt, F
DT1H = Thermal gradient at maximum temperature from seal or tube to flange, F
DT2H = Thermal gradient at maximum temperature from flange to ring, F
DT3H = Thermal gradient at maximum temperature from ring to bolt, F.

Card 36, Same as in IRFDP.

### Description of Output for LRFDP

As in program IRFDP, the input data is first printed out

The design output consists of the same sections as in IRFDP with a few additional print-outs in the "stress for all conditions" section and in the "dimensions" section.

In the stress section after the bolt tensile stress (BTS), the contact stress between the ring and the stub flange (SFB) is printed for all design conditions.

In the dimension section the following interpretations are made:

A = Outside diameter of stub flange, in.

B = Inside diameter of stub flange, in.

XT = Stub flange thickness, in.

ROD = Loose-ring outside diameter, in.

RID = Loose-ring inside diameter, in.

RFR = Ring fillet radius which fits in with the flange fillet radius, in.

The rest of the output interpretation is as described in IRFDP. The dimensions for connectors with loose-ring flanges are shown in Figure B-4.

### Designing an Integral Loose-Ring Connector

- (1) Design a connector using TRFDP
- (2) Design a connector using LRFDP
- (3) Select the maximum bolt circle diameter from Steps 1 and 2
- (4) If the bolt circle diameter is maximum in Step 1, then run Program LRFDP with the values as the input in Card 33a (LRFDP). Also, the required number of bolts and the bolt index are read in (Card 30-31) as obtained in Step 1.
- (5) If bolt circle diameter is maximum in Step 2, then run Program IRFDP with this value as the input in Card 4 (IRFDP). Also, the required number of bolts and the bolt index are read in (Card 30-31) as obtained in Step 2.
- (6) By using dimensions in Steps 1 and 4 or Steps 2 and 5, a matching connector is obtained for an integral flange and a loose-ring flange.

```
PROGRAM IREDP (INPUT, OUTPUT, TAPEGO=INPUT)
       TYPE HEAL LT.MSL.K.LCE.LCD.L.MP2.MC2.MO1.NA1.NA2.NA3
       TYPE REAL NBI.NB2.NB3.MO2.MO2P.MO2M.MP3.MO3.MP5.MO5.MO6.MC6.
      1MCTG6,MHTG6,MOGCN,MOGH,MOGHN,MO7,MH,MP6
       TYPE HEAL MSLI
       TYPE REAL NWT
       TYPE REAL KTP
       DIMENSION R(11) .FF(11) .BH(15) .BWC(15) .BS(15) .BTA(15) .NWT(15)
       DIMENSION DAF (1)
C TUBE LOAD-INPUT
 1020 FURMAT (8F10.0)
 1021 FURMAT (8F10.4)
1022 FORMAT (8F10.9)
 1023 FORMAT (2110)
  999 REAU 1020. TBM1
       IF (EOF+60) 998,997
C TUBE GEOMETRY
  997 READ 1021, TUD, TW1
C TUBE MATERIAL PROPERTIES
       READ 1020+TU+TY+TF+TR+TFR0
C FLANGE GEOMETRY
       READ 1021+BCDI+HUBII
       REAU 1021,GIOTWI
       READ 1021.FRI.FSFI
C FLANGE MATERIAL PROPERTIES
       REAU 1020+FUR
       READ 1020 + FU + FY + FSU + FCY + FE + FG + FYC + FYR
       READ 1022 FRHO. FMU. FTE.FCR
       READ 1021: (R(NN), NN#1.8)
       READ 1021+ (R(NN) + NN=9+11)
      READ 1020: (FF (NN), NN=1.8)
READ 1020: (FF (NN), NN=9.11)
C SEAL GEOMETRY
      REAU 1021+SUDI+SIDI+SELI+TAI+TAOTW+SLEGTI
C SEAL LOADS-INPUT
      READ 1020, RSLI, SSLI, MSLI
C SEAL MATERIAL PROPERTIES
       READ 1020.SCY. SEE, SCYC, SCYR, SYR
      READ 1022.STE
C BOLT GEOMETRY
      READ 1021+ (BH (NN) , NN=1,8)
       REAU 1021 + (6H(NN) + NN=9+15)
       READ 1021, (BWC (NN), NN=1,8)
       REAU 1021, (BWC (NN), NN=9,15)
       REAU 1021+ (BS(NN) + NN=1.08)
      READ 1021+ (BS(NN) + NN=9+15)
      REAU 1021; (BTA (NN), NN=1,8)
      REAU 1021, (BTA (NN), NN=9,15)
      HEAD 1021+ (NWT (NN) + NN=1+8)
      HEAD 1021 + (NWT (NN) + NN=9+15)
      READ 1021+ (DAF (NN) + NN=1+8)
      READ 1021 (DAF (NN) NN=9,15)
C BOLT INPUT
      READ 1023,11, IMAX
      READ 1020.8N1
```

```
C BOLT MATERIAL PROPERTIES
```

```
HEAD 1020+8Y+8E+80+8YC+8YR
HEAD 1022+8TE+8CR+8RHO
C SYSTEM PRESSURE AND TEMPERATURE
      REAU 1020, P.PP, PB, PIM, TMAX, TMIN
      REAU 1020+DTC1+DTC2+DTH1+DTH2
      READ 1021, FS1, FS2, FS3
9020 FORMAT (8F10.0)
9021 FORMAT (8F10.4)
9022 FURMAT (8F10.8)
9023 FORMAT (2110)
      PRINT 726
  726 FORMAT (25X, 19HINPUT TO THE DESIGN///)
      PRINT 2020
2020 FORMAT (16H TUBE LOAD-INPUT)
      PRINT 2021
2021 FORMAT (JA. 4HTBM1)
      PRINT 9021.TBM1
      PRINT 2000
2000 FORMAT (14H TUBE GEOMETRY)
      DELINI SCOT
2001 FORMAT (6X, 3HT00, 7X, 3HTW1)
      PRINT 9021.TOD.TW1
      PRIN1 2002
2002 FORMAT (25H TUBE MATERIAL PROPERTIES)
      PRINT 2003
2003 FORMAT (7X+2HTU+8X+2HTY+8X+2HTF+8X+2HTR+8X+4HTFR0)
      PRINT 9020.1U.TY.TF.TR.TFRO
PRIMI 2004
2004 FORMAT (16H FLANGE GEOMETRY)
      PRINT 2005
2005 FORMAT(5x+4HBCDI+5x+5HHUBII)
      PRINT 9021.BCD1.HUBII
      PRINT 800
 HOD FORMAT (3X, 6HG10TWI)
      PRINT 9021.GIOTWI
      PRINT 2037
2037 FURMAT (6x, 3HFRI, 6x, 4HFSFI)
      PRINT 9021+FRI+FSFI
      PRINT 2006
2006 FORMAT (27H FLANGE MATERIAL PROPERTIES)
      PRINT 801
 801 FURMAT (6X+3HFUR)
      PRINT 9020+FUR
      PRINT 2007
2007 FURMAT (7X+2HFU+8X+2HFY+7X+3HFSU+7X+3HFCY+8X+2HFE+8X+2HFG+7X+
     13HFYC+7X+3HFYR)
      PRINT 9020, FU, FY, FSU, FCY, FE, FG, FYC, FYR
      PRINT 2008
2008 FORMAT (54,4HFRH0,5X,3HFMU,7X,3HFTE,7X,3HFCR)
      PRINT 9022+FRH0+FMU+FTE+FCR
      PRINT 2009
2009 FURMAT (5X+4HR(1)+6X+4HR(2)+6X+4HR(3)+6X+4HR(4)+6X+4HR(5)+6X+
     14HR(0)+6X+4HR(7)+6X+4HR(8))
      PHINT 9021+R(1)+R(2)+R(3)+R(4)+R(5)+R(6)+R(7)+R(8)
      PRINT 2010
```

2010 FORMAT (54+4HR (4) , 5X+5HR (10) +5X+5HR (11))

PRINT 9021+R(9)+R(10)+R(11) PRINT 2011 2011 FORMAT (4X+5HFF(1)+5X+5HFF(2)+5X+5HFF(3)+5X+5HFF(4)+5X+5HFF(5)+5X+ 15HFF (6) + 5X + 5HFF (7) + 5X + 5HFF (8) ) PRINT 9020+FF (1)+FF (2)+FF (3)+FF (4)+FF (5)+FF (6)+FF (7)+FF (8) PRINT 2012 2012 FORMAT (4X+5HFF(9),4X,6HFF(10),4X,6HFF(11)) PRINT 9020, FF (9), FF (10), FF (11) PRINT 2013 2013 FORMAT (14H SEAL GEOMETRY) PRINT 2014 2014 FORMAT (SA.4HSODI,6X.4HSIDI,6X.4HSELI.6X.3HTAX.5X.5HTAOTW.4X.6HSLE 1G11) PRINT 9021, SOUL, SIDI, SELL, TAL, TAOTW, SLEGTI PRINT 2015 2015 FORMAT (18H SEAL LOADS INPUTS) PRINT 2016 2016 FURMAT (5X, 4HRSLI, 4X, 4HSSLI, 4X, 4HMSLI) PRINT 9020, RSLI, SSLI, MSLI PRINT 2017 2017 FORMAT (25H SEAL MATERIAL PROPERTIES) PRINT 2018 2018 FORMAT (7X+3HSCY+7X+3HSEE+6X+4HSCYC+6X++HSCYR+7X+3HSYR) PRINT 9020, SCY, SEE, SCYC, SCYR, SYR PRINT 2019 2019 FURMAT (3X+3HSTE) PRINT 9022.STE PRINT 2022 2022 FORMAT (14H BOLT GEOMETRY) PRINT 2023 2023 FORMAT {4x=5HBH(1),5x,5HBH(2),5x,5HBH(3),5x,5HBH(4),5x,5HBH(5),5x, 15HBH(6),5X,5HBH(7),5X,5HBH(8)) PRINT 9021, BH(1), BH(2), BH(3), BH(4), BH(5), BH(6), BH(7), BH(8) PRINT 2024 2024 FORMAT (4X, 5HBH (9), 4X, 6HBH, 10), 4X, 6HBH (11), 4X, 6HBH (12), 4X, 6HBH (13) 1,4X,0HUH(14),4X,6HBH(15)) PRINT 9021+8H(9)+8H(10)+8H(11)+8H(12)+8H(13)+8H(14)+8H(15) PRINT 2025 2025 FURMAT (4X+6HBWC(1)+4X+6HBWC(2)+4X+6HBWC(3)+4X+6HBWC(4)+4X+6HBWC(5 1),4%,6HBWC(6),4%,6HBWC(7),4%,6HBWC(8)) PRINT 9021+BWC(1),BWC(2),BWC(3),BWC(4),BWC(5),BWC(6),BWC(7),BWC(8) PRINT 2026 2026 FORMAT (4X+6H8WC(9)+3X+7H8WC(10)+3X+7H8WC(11)+3X+7H8WC(12)+3X+ 17H8Wc(13)+3X+7H8Wc(14)+3X+7H8Wc(15)) PRINT 9021.BWC(9).BWC(10).BWC(11).BWC(12).BWC(13).BWC(14).BWC(15) PRINT 2027 2027 FORMAT (5X,5HBS(1),5X,5HBS(2),5X,5HBS(3),5X,5HBS(4),5X,5HBS(5),5X, 15H8S(6),5X,5H8S(7),5X,5H8S(8)) PRINT 9021,85(1),85(2),85(3),85(4),85(5),85(6),85(7),85(8) PRINT 2028 2028 FORMAT (5X+5HBS(9)+4X+6HBS(10)+4X+6HBS(11)+4X+6HBS(12)+4X+6HBS(13) 1+4x+0HH5(14),4x,6H8S(15)) PRINT 9021+85(9)+85(10)+85(11)+85(12)+85(13)+85(14)+85(15) PRINT 2029 2029 FORMAT (4×+6HBYA(1)+4×+6HBTA(2)+4×+6HBTA(3)+4×+6HBTA(4)+4×+6HBTA(5 1),4X,6HBTA(6),4X,6HBTA(7),4X,6HBTA(8))

PRINT 9021, BTA(1), BTA(2), BTA(3), BTA(4), BTA(5), BTA(6), BTA(7), BTA(8) PRINT 2030 2030 FORMAT (4X+6HBTA(9)+3X+7HBTA(10)+3X+7HBTA(11)+3X+7HBTA(12)+3X+7HBT 1A(13), 3X, 7HBTA(14), 3X, 7HBTA(15)) PRINT 9021, BTA (9), BTA (10), BTA (11), BTA (12), BTA (13), BTA (14), BTA (15) PRINT 2053 2053 FORMAT (4X+6HNWT(1)+4X+6HNWT(2)+4X+6HNWT(3)+4X+6HNWT(4)+4X+6HNWT(5 1), 4X, 6HNWT (5), 4X, 6HNWT (7), 4X, 6HNWT (8)) PRINT 9021, NWT (1), NWT (2), NWT (3), NWT (4), NWT (5), NWT (6), NWT (7), NWT (8) PRINT 2054 2054 FORMAT (4X+6HNWT(9)+3X+7HNWT(10)+3X+7HNWT(11)+3X+7HNWT(12)+3X+7HNW 17(13)+3X+7HNWT(14)+3X+7HNWT(15)) PRINT 9021. NWT (9) . NWT (10) . NWT (11) . NWT (12) . NWT (13) . NWT (14) . NWT (15) PRINT 2038 2038 FORMAT (4X+6HDAF (1)+4X+6HDAF (2)+4X+6HDAF (3)+4X+6HDAF (4)+4X+6HDAF (5) 1.4X.6HDAF(6).4X.6HDAF(7).4X.6HDAF(8)) PRINT 9021+DAF(1) .DAF(2) .DAF(3) .DAF(4) .DAF(5) .DAF(6) .DAF(7) .DAF(8) PRINT 2039 2039 FORMAT (4X+6HDAF (9)+3X+7HDAF (10)+3X+7HDAF (11)+3X+7HDAF (12)+3X+7HDAF 1(13), 3X, 7HDAF(14), 3X, 7HDAF(15)) PHINE 9021+DAF (9) +DAF (10) +DAF (11) +DAF (12) +DAF (13) +DAF (14) +DAF (15) PRINT 2050 2050 FORMAT (11H BOLT INPUT) PRINT 2051 2051 FORMAT (8X+2HII+3X+4HIMAX) PRINT 9023, 11, IMAX PRINT 2052 2052 FORMAT (6X. 3HBNI) PRINT 9021, BNI PRINT 2031 2031 FORMAT (25H BOLT MATERIAL PROPERTIES) PHINT 2032 2032 FORMAT (HA+2HBY+BX+2HBE+BX+2HBU+7X+3HBYC+7X+3HBYR) PRINT 9020+BY+BL+BU+BYC+BYH PRINT 2033 2033 FORMAT (6X, 3HBTE, 7X, 3HBCR, 6X, 4HBRHO) PRINT 9022+BTE+BCR+BRHO PRINT 2034 2034 FORMAT (32H SYSTEM PRESSURE AND TEMPERATURE) PRINT 2035 2035 FORMAT (9X,1HP,9X,2HPP,8X,2HPB,7X,3HPIM,6X,4HTMAX,6X,4HTMIN) PHINT 9020+P+PP+PB+PIM+TMAX+TMIN PRINT 2055 2055 FORMAT (5x, 4HDTC1, 6x, 4HDTC2, 6x, 4HDTH1, 5x, 4HDTH2) PRINT 9020+UTC1+UTC2+DTH1+DTH2 PRINT 2036 2036 FURMAT (7X+3HFS1,7X+3HFS2,7X+3HFS3) PRINT 9021+FS1+FS2+FS3 KOUNT=0 TUBE CALCULATIONS TUBE WALL . TUBE BENDING MUMENT. TUBE BENDING STRESS C TW2=1.1+PP+TOD/(2.+TY+.8+PP) TW3=1+1+P8+T00/(2++TU++8+P8) IF (TFR0 .E4. 0.0) GO TO 16 TW4=1.1\*PIM\*TOD/(2.\*TFR0+.8\*PIM) GO TO 17 16 TW4#TW3

B - 17

```
17 CONTINUE
   IF (.001-TW1)1,1,2
 1 TWETW1
   GO TO 3
 2 TW=MAX1F(TW2+TW3+TW4)
 3 W=3.1416+(TOU+++-(TOU+2.+TW)++4)/(32.+TOD)
   IF (TUM1 .EQ. 0.001)5.4
 4 TBM=TBM1
   GO TO 5
 5 TBM2=#+.6667+TY
   TBM3=8*.8*TR
   TEMARNA, SATE
   TBM5+4INIF (TBM2, TBM3, TBM4)
   TBM6#60, # (TOD+3.) ##3
   TBMEMINIF (TBM5, TBM6)
 6 TBS=TBM/W
   P]=3,1416
    IF(T00-3.0)26,26,27
26 BLOm.5+THS+PI+(TOD-TW)+TH+(.5+.5+(TOD/3.))
   GO TO 33
27 BLO#.5+TUS+PI+(TOD-TW)+TW
33 BLI=MAXIF (BLO, SSLI#PI#TOD)
BNP=4.
    IF (TOD-3.0) 60.60.6)
60 I=1
    GO TO 62
61 I=2
62 XT=3.+TW
    GQuTW
    IF (.001-HUBII) 64,63,63
63 G1=1.25*TW
    GO TO 65
64 G]=TW+HUBII
65 8=100-2. TW
    IF(.001-FRI)700,701,701
700 FREFKI
    GO TO 702
701 IF (TOD-4.) 703.703,704
703 FR=0.125
    GO TO 702
704 IF (TOD-8.) 705, 705, 706
705 FR=0.1875
    GO TO 702
706 IF (TOD-12.)707.707.708
707 FR=0.25
    GO TU 702
708 FR=0.3125
702 CONTINUE
    SLEG=0.1+0.012+TOD
    SID= (TOD-2. *TW) +2. *SLEG*0.15
 14 CANTAOTW*TW
 15 SOD#SID+2.*TA+2.*SLEG
    D3#SID+2.*TA
    SLEGT1=RSLI#(SOD/D3)/SCYR
    SLEUT2=0.04+(0.001+P++0.5+TOU)/12.0
    SLEGT=MAAIF (SLEGT1, SLEGT2)
```
```
TARSL=RSLI#(SOD/D3)-SCYR#(SOD#30D=D3#D3)#SLEGT/(SOD#SOD)
      SEL=2. #RSLI/(SCYR#ALOG(D3/SID))
      SX=StL/2.
      LT=SEL/TA
   18 IF (4.0-LT) 19, 19, 25
   19 TA=TA+.005
      GO TO 15
   25 IF (.001-SOUI) 32, 32, 28
   32 SOD#SOD1
      SID#5101
      SEL=SELI
      TAETAL
      SLEGT=SLEGTI
      SX=SEL/2.
      LOAD CALCULATIONS
С
   28 MSL=MSLI+PI+SOD
      PHELIMINARY BOLT CALCULATION
C
  107 IF (I-IMAX) 108+1009+1009
 1009 PRINT 1010
 1010 FORMAT (JA+36HBOLT REQUIRED NOT AVAILABLE IN TABLE)
      GO TO 999
  108 HUBI=G1-TW
      BCD1=SUD+2.*((SLEGT*6.0*RSLI*FS1/FY)**.5)+BH(I)
      BCD2=SUD+2.*(RSLI+FS1/FSU)+BH(7)
      BCD3=TOD+BMC(I)+S*#HOBI
      BCD4=2.* (.060+TOD*.0075)+SOD+BH(I)
      BCD6=TUD+2.*HUBI+2.*FR+DAF(I)
      BCD5=MAX1F (BCD1, BCD2, BCD3, BCD4, BCD6)
      IF (.001-BCDI)110-113-113
  110 IF (BCD5-BCDI) 112,112,1005
 1005 PRINT 1006
 1006 FORMAT (3X+32HBCD INPUT SMALLEP THAN ALLOWABLE)
  115 BCD=BCDI
      UN=UNI
      I = I I
      GO TO 114
  113 BCD=0CU5
      BN=BCD+PI/BWC(I)
      NB=RN15.
      N=2*NB
      BN=N
  114 A=8CD+2,#8H(I)
      H2=(2.*BCU-A-B)/4.
      H3= (A-8-2+TA)/4.
      H4=((A+B)/4+)-(1+/3+)*(SOD**2+SOD*B+B**2)/(SOD+B)
      H5=((A-B)/4.)-(TW+HUBI)/2.
      H6=H5
      FLANGE SIZE CALCULATION
С
  201 CUNTINUE
      H=2 .0+ (8+60) ++ .5
      HG= (BCU-8-G0)/2.
      K=A/B
      T=((K+k)+(1+0+8+55246+ALOG10(K))+1+0)/((1+04720+1+94480+K+K)+(K-1+
     1))
      U = ((K^*K) + (1 + 0 + 8 + 552 + 6 + ALOG10(K)) + 1 + ) / (1 + 36136 + (K^*2 + 1 + ) + (K + 1 + ))
      Y=(1,/(K-1.))*(.66845+5,717*((K*K)*ALOG10(K)/(K**2-1.)))
```

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Z=(K+K+1.)/(K+K-1.)
      Fz, 909-,031*(G1/G0-1.)-,1605*(G1/G0-1.)**.5
      V=.55+.15+(01/G0-1.)~.558+(G1/G0-1.)**.5
      HU= ( (8+G0) +G0) ++.5
      LCE=F/HO
      FCD= (A\A) +HO+20++5
  202 L=(XT+LCE+1+)/T+(XT++3)/LCD
      GE=0.5*(G1+G0)
      HPO# (8+GE) ++.5
      C1=(3.+*.5)*(2.-FMU)/(4.*FE*(1.-FMU**2))
      C2=(3./(1.-FMU++2))++.5
      C3=(12.+(1.-FMU++2))++.25
      C4=((12.*(1.-FMU**2))**.25)/(2.*(1.-FMU**2))
      HETA= (3.*(1.-FMU++2)/((((YOD/2.)+GE-GO)++2)*(GE)++2))++.25
      HOP= (8+GE) ++ .5
      GAMMA=(GL*(FMU+Z)+(1.+C3*XT/HOP))/(1.+(C4*B*GE**3*(FMU+Z)/(HOP*X**
     1+3) * (2+C3+XT/HOP)))
      DT=FE+GE++3/(12,*(1.-FMU++2))
      UF=FE+XT++3/(12.+().+FMU++2))
      XP=((((1.+FMU)/(1.+FMU))+K++2+1.)+B)/((K++2-1.)+(1.+FMU)+2.+OF)
      AT=(PI/4,)*(TOD+#2-8##2)
      ZH=P1*((TOD-2.*G0+2.*G1)**4-(TOD-2.*TW)**4)/(32.*(TOD-2.*G0+2.*G1)
     1)
      AH=(((TOU-2,*G0+2,*G1)**2-(TOD-2,*G0)**2)*P1/4.)
      DEFLECTION RATES
С
      DSA=- (SX) / (SEE+PI+ (8+TA) +TA)
      USR==(H+TA)/(2.*PI*SEL*TA*SEE)
      DBA=(.5/BE)+(((SEL+2.+XT)/2.)+(4./(PI+BS(I)+2))+((SEL+BS(I)
     1+2.*XT)/(2.*BTA(1)))/BN
      DFR=(.91+V)/(L+H0+G0++2+FE)
      DFA=-DFH+(H2+H3)++2=(1,/(2.+PI+FG+XT))+ALOG(BCD/(8+TA))
     1-(XT)/((PI/4.)*(BCD**2-B**2)*FE)
      RATIU=A/TOD
  IF (RATIO .LE. 1.1)711,712
711 KTF=0.179*((T<sup>O</sup>D/FR)**.5)+.914
      GO TO 718
  712 IF (RATIO .LE. 1.2)713,714
  713 KTF=0.202*((TOD/FR)**.5)+.86
      GO TU /1H
  714 IF (RATIO .LE. 1.5)715,716
  715 KTF=0.233*((TOD/FR)*+5)+.79
      GO TO 718
  716 KTF=0.282+((TOD/FR)++.5)+.69
  718 CONTINUE
      LOAD CALCULATIONS CONDITION 1 AND 2
С
      P1=0.
      P2=P
      TBM1=0.
      TBM2=TBM
      MP2=((L+H0+G0++2+FE)/(.91+V))+(C1+B+#2+GAMMA+P2/(XT#(XT++2+C2#GE+G
     1AMMA)))
      FA1×0.
      FA2=P2>(P1/4)+(8++2)
      FPL1=0.
      FPL2=P2+(P1/4+)*(S0D++2-B++2)
      RSL1=RSL1+PI+SOU
```

```
B-20
```

```
111 SL1=8L1
      MU1=-RSL1+XT/2.+BL1+H2+SL1+H3
      MC2#MIN1F((L+G1++2+(8+TW)+FE+FCR)+(MO1+FE+FCR/FYR))
C STRESS CALCULATION CONDITION 1
  310 SH=((8L1*H2+SL1*H3)/(L*G1**2*(8+G0)))
  311 SRS=((RSL1+XT/2,)/(L+G1++2+(D+G0)))
      SR=M01+(1+333*XT+LCE+1+)/(L+XT++2+(8+G0))
      ST=MU1+(Y/(XT++2+B))-Z+SR
      58=0.
      SPR=0.
      SPA#0.
      SCEA=0.
      SCEH=0.
      FSS=UL1/(TOD#PI#XT)
      SSC*SL1/(((B+2.*TA)**2-8**2)*P1/4.)
      BTS=BL1/(BN+BTA(I))
      IF (875-878/FS2) 314,314,414
  414 IF (.001-BNI) 314,314,313
  313 I=I+1
      GU TO 107
  314 IF (SR-FYR/F51) 315.315.302
  315 IF (ST-FYR/FS1)316,316,302
  316 IF (FSS-FSU/FS1) 317,317,302
  317 IF (SSC-SCYR/FS1) 330+330+619
  619 PRINT 620
  620 FORMAT (JX, 18H SSC EXCEEDS LIMIT)
  330 XA1=-SH+SH+SPR+SPA+SRS
      XA2=SCEA+SR
      XA3#ST
      NA1=-SH-SB+SPR+SPA+SRS
      NA2=-SCEA+SR
      NA 3=ST
      EXA=.70711+((XA1-XA2)++2+(XA2-XA3)++2+(XA3-XA1)++2)++3
      ENA#.70711*((NA1-NA2)++2+ (NA2-NA3)++2+ (NA3-NA1)++2)++.5
      EA - AMAX) (ABS(EXA) , ABS(ENA) )
      IF (EA-FYR/FS1) 352, 352, 302
  352 X81#+SH+SH=SPR+SPA-SRS
      X82=-SCE8+SR
      X83*ST
      NB1#+SH-SB-SPR+SPA-SRS
      NH2=-SUF8+SR
      NB3=5T
      £x8=.70711*((x81-x82)*+2+(x82-x83)*+2+(x83-x81)*+2)**.5
      ENB=.70711*((NB1-NH2)**2*(NB2-NB3)**2*(NB3-NB1)**2)**.5
      EB=AMAX1 (ABS (EXB) , ABS (ENB) )
      1F (EB-FYR/FS1) 362, 362, 302
 402 IF(HUBII .LT. 0.001)802.302
802 IF(G101WI .LT. 0.001)804.803
 803 GIOTW=GI/TW
      IF (GIUTW .GT. GIOTWI) 302,804
  804 61=61+0.005
      XT=XT=0.030
      GO TU 108
  302 XT#XT++015
      IF (X1-5"+100) 505'1003'1003
1003 PRINT 1004
```

```
1004 FORMAT (3X.31HFLANGE THICKNESS EXCEEDS 2. * TOD)
      GU TO 999
  850 XT=XT+.015
      KOUNI = KUUNT + 1
      IF (KUUNT .LE. 5)860+202
  860 IF (G1 .GT. 1.75+TW) 851,202
  851 G1=G1-0.005
      GO TO 108
C
      LOAD AND MOMENT CALCULATION FOR CONDITION 2
  362 TL2=FA2+TES#AT
      Q21*-FA2-F 2L2-YBS+AT
      422* (P2+PI+6+SX) +XT/2 -- FPL2+H4-HP2+MC2-TL2+H5
      Q23*+(P2*8*(8+TA))/(4.*TA*SEE)-(P2*8*((A+B)/2.))/(*.*((A-B)/2.)
     1#FE)
      Q24=-Q21+UBA/ (DBA-DSA) +BCR+ (XT+SX) / (DBA-DSA)
      Q25=UFA/((DBA-DSA)+(H2+H3))
      #50=H5+051+055
      427=-DFR+XT/(2.+DSR)
      428*1.+027*(XT/2.)-025*(H2+H3)
      Q29==(U23/DSR) + (XT/2+)+Q26+Q24+(H2+H3)
      MU2=M01-Q29/Q28
      SL2=SL1-024-(029/028)+025
      RSL2=RSL1-(029/028) #027-023/DSR
      HL2=HL1-Q24-(Q29/Q28) +Q25-Q21
If (SL2-HSL) 331, 332, 334
  331 BL1=1.01+8L1
      GO TO 111
  332 421P=-FPL2-FA2-T85+AT
      W22H=(H5+H1+B+5X)+X1/2.-FPL2+H4-MP2-TL2+H5
      Q23P=+(P2*B*(B+TA))/(4.*TA*SEE)-(P2*B*((A+8)/2.))/(4.*((A+8)/2.)
     1*FE)
      Q24P==Q21P+DBA/(DBA=DSA)
      425P=DFA/((DHA-DSA)+(H2+H3))
      426P=H2+021P+022P
      427P=-DFR#XT/(2.#0SR)
      Q28P=1.+Q27P+(XT/2.)=Q25P+(H2+H3)
      Q29P=+ (Q23P/DSR) * (XT/2.) +Q26P+Q24P* (H2+H3)
      MOSb=WO1=ASAb/0686
      SL2P=5L1-424P- (029P/028P) +025P
      RSL2P=RSL1-(Q29P/Q28P)+Q27P=U23P/DSR
      BL2P=BL1-Q24P=(Q29P/Q28P)+Q25P=Q21P
      421M=-FPL2-FA2+TBS#AT
      Q22M=(P2*P1*8*SX)*XT/2,-FPL2*H4-MP2-H5*(FA2-TB$*AT)
      423M=+ (P2*B*(B+TA))/(4+*TA*SEE)=(P2*3*((A+B)/2+))/(4+*((A+B)/2+))/
     1#FE)
      425M#UFA/((UBA-DSA) *(H2+H3))
     M59W#H5+051W+055W
     427M=-UFR+XT/(2.+USR)
     Q28M=1.+U27M*(XT/2.)-Q25M*(H2+H3)
     Q29M=-(Q23M/DSR) + (XT/2+) +Q26M+Q24M+(H2+H3)
     MO2M-M01-42914/428M
     SE2M=SE1-424M- (Q29M/Q28M) +Q25M
     HSL2MEHSL1- (029H/028H) +027H-023H/DSR
     BL2M#BL1-Q24M- (Q29H/Q28M) + Q25H-Q21H
     STRESS CALCULATIONS CONDITION 2
```

C

B 22

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SH2P=(HL2P+H2+FPL2+H4+TL2+H6+SL2P+H3)/(L+G1++2+(B+G0))
SH2M=(HL2M+H2+FPL2+H4+(FA2-THS+AT)+H6+SL2M+H2)/(L+G1++2+(B+G0))
     SRS2P=(MSL2P*XT/2.)/(L*G1+*2*(B+G0))
     SRS2M=(RSL2M*X1/2,)/(L+G1++2+(B+G0))
     SPR2=MP2/(L+G1++2+(8+G0))
     SB2=TBM/ZH
     SPA2=+A2/AH
     SCEB4#0
     SCEA2=22
     SR2P=MU2P+(1.333+XT+LCE+1.)/(L+XT++2+(B+00))
     SR2M=MU2M#(1.333*XT*LCE+1.)/(L*XT+*2*(B+G0))
     ST2P=MU2P+ (Y/ (XT++2+B)) -Z+SR2P
     ST2M=MO2M+(Y/(XT++2+8))-2+5R2M
     FSS2=BL2P/(TOD+PI+XT)
     SSC2=SL2M/(((B+2++TA)++2-B++2)+P1/4+)
     BTS2P=BL2P/(BTA(I) +BN)
     IF (BTS2P-BY/FS2) 344+344+415
415 IF (.001-BNI) 344, 344, 313
344 IF (Sn2P-FY/FS1) 345, 345, 302
 345 IF (ST2P=FY/FS1) 346, 346, 302
 346 IF (FSS2-FSU/FS1) 347+347+302
 347 IF (SSC2-SCY/FS1) 348, 348, 621
 621 PRINT 622
 622 FORMAT (3X+19H SSC2 EXCEEDS LIMIT)
 348 XA1=-SH2P+SB2+SPR2+SPA2+SRS2P
     XA2=-SCEA2+SR2P
     XA3×5T2P
     NA1=-SH2M-SB2+SPR2+SPA2+SRS2M
     NA2=-SCEA2+SH2M
     MST2=EAN
     EXA2=./0711#((XA1-XA2)##2+(XA2-XA3)##2+(XA3-XA1)##2)##.5
     ENA2=. 10/11*((NA1-NA2) **2+(NA2-NA3) **2+(NA3-NA1) **2) **.5
     XA1=XA1-SPA2
     NA1=NA1-SPA2
     IF (AdS (XA1) - AUS (NA1) ) 602+602+601
 602 CMAXZA=NAL
     CMINZA=XA1
     GO TO 503
 601 CMAX2A#XA1
     CMINZA=NA1
 603 EA=AMAX1 (ABS(EXA2), ABS(ENA2))
     IF (CA-FY/F51)1352,1352,302
1352 CALTZA= (CMAX2A-CMIN2A)/2.
     CMNZA= (CMAX2A+CMIN2A)/2.
     FALI2A=CALI2A+FSFI
     SALTZA=FALTZA
     SMNZAECMNZA
     SCHKZA#ABS (SMN2A+SALT2A)
     IF (SCHR2A .GT. FU) 402.805
 805 IF (SCHK2A-FY)806,806,807
 807 IF (AUS (SALT2A) - FY) 808+808+809
 808 SMN2A=FY-SALT2A
     60 10 806
 809 SMN24=0.
```

E06 SMAX2A=SMN2A+LALT2A MIN2A=SMN2A=ALT2A

```
RC1=SM1N2A/SMAX2A
      Macin
 1350 M=M+1
       IF (RC1-R(M))1351,7000,1350
 1351
      FFC2A=((R(M)-RC1)/(R(M)-R(M-1)))+(FF(M)-FF(M-1))+FF(M-1)
      GO TO 7001
 7000 FFC2A=FF(M)
 7001 CONTINUE
       IF (AUS (SMAX2A) -FFC2A) 1363, 1363, 402
 1363 XH1=SH2P+SH2-SPR2+SPA2-SRS2P
       AU2=-SCEU2+SR2P
      XU3=SI2P
      NU1=SH2M-SU2-SPR2+SPA2-SRS2M
      N85=-2CER5+2K5W
      MS15*68N
      EX82=./0/11*((X81=X82)**2+(X82=X83)**2+(X83=X81)**2)**.5
      EN82=. /0711*((N81-N82)**2+(N82-N83)**2+(N83-N81)**2)**.5
      EB=AMAX1 (ABS(EXB2) + ABS(ENB2))
      XB1=XB1-SPA2
      NU1=NU1-SPA2
      IF (LB-FY/FS1) 720,720,302
  720 IF (ABS (X81) - ABS (NB1) ) 605,605,604
  605 CMAX28=NB1
      CHIN28=X81
      GO TU 1362
  604 CMAX28=X81
      CWIN5R=NRJ
 1362 CALT28=(CMAX28-CMIN28)/2.
      CMN2ds (CMAX2B+CHIN2B)/2.
      FALT28=CALT28=FSF1
      SALT28=FALT28*KTF
      SMN20=CMN28+KTF
      SCHK28=ABS(SMN28+SALT28)
      IF (SCHK28.GT. FU) 402.810
  810 1F (SCHK28-FY)811.811.812
  812 IF (ABS (SALT28) -FY) 813,813,814
  813 SMN20=+Y-SALT28
      60 TO H11
  814 SMN20=0.
  B11 SMAX2H=SMN26+SALT2H
      SMIN2B=SMN2B-SALT2B
      RU2=SMIN28/SMAX2
      M=()
 1360 M=M+1
      IF (HC2-R(M))1361,7002,1360
 1361 FFC20=((N(M)+RC2)/(R(M)+R(H-1))+(FF(M)+FF(M+1))+FF(M+1);
      60 10 7003
 7002 FFC20=FF(M)
 7003 CONTINUE
      1F (ABS (SMAX2B) - FFC2B) 380, 380, 402
      LUAU CALCULATIONS CONDITION 3
С
  380 H3=PH
      THM3=0.
      FA3=P3+(P1/4.)+(8++2)
      FPL3=P3+(P1/4+)+(S00++2+8++2)
      TL3=FAJ
```

8-24

```
MP3=((L+HU+60++2+FE)/(.91+V))+(C1+B++2+GAMMA+P3/(XT+(XT++2+C2+GE+G
     IAMMA)))
      Q31=-FPL3-TL3
      032=-FPL3*H4-MP3+(P3*PI*B*SX)*X1/2.-TL3*H5
      W33#+ (Y3+8+ (8+TA))/(4.+TA+SEE)+ (P3+8+(1A+8)/2.)/(4.+(A-8)/2.)+
     1(と)
      Q34=-U314084/(DBA-DSA)
      (035=0FA/((UBA=0SA)+(H2+H3))
      M30=H5+031+035
      437=-0+R*XT/(2.+0SR)
      (438=1++437*XT/2++435*(H2+H3)
      Q39== (U33/D5R) + (XT/2.) + Q36+Q34+ (H2+H3)
      M03=M01-W39/W38
      SL3=SL1=434-(434/438)#435
      HSL3=RSL1-(439/038) +437-433/DSR
      BL3=6L1-034-(039/038)+035-031
      IF (SL3-MSI.) 331, 390, 390
      STRESS CALCULATIONS CONDITION 3
С
  390 5H3=(BL3*H2+SL3*H3+FPL3*H4+TL3*H6)/(L*G1**2*(B+G0))
      SR53=(H5L3+XT/2.)/(L+G1++2+(H+G0))
      SPR3=MP3/(L*G1++2+(8+G0))
      583=0.
      SPA3=FA3/AH
      SCE83=0.
      SCLAJ=P3
      SR3=M03*(1.333*XT*LCE+1.)/(L*XT**2*(8+G0))
      513=M03+(Y/(XT++2+8)) -Z+5R3
      FSSJ=HL3/(TOD+PI+XT)
      SSC3=SL3/(((8+2.*TA)**2-8**2)*PI/4.)
      BISJ=ELJ/(BIA (I) +BN)
      11 (8TS3-8Y/F52)2344,2344,416
 416 IF (.001-HNI)2344,2344,313
2344 IF (SR3-FY/FS1)2315+2315+302
2315 1F (ST3-FY/FS1)2316+2316+302
2316 IF (FSS3-FSU/FS1) 2317,2317,302
2311 IF (55C3-SCY/FS1)2330+2330+623
 623 PRINT 624
 624 FORMAT (3X, 19H SSC3 EXCEEDS LIMIT)
2330 XA1=-SH3+SH3+SPR3+SPA3+SRS3
      XA2=-SLEA3+SR3
      XA3=5T3
      NA1=+SH3+SH3+SPR3+SPA3+SRS3
      NA2=-SCEA3+SR3
      NA3#ST3
      EXA3= . /0/11*((XA1-XA2)**2+(XA2-XA3)**2+(XA3-XA1)**2)**.5
      ENA3=./0711*((N 1-NA2)++2+(NA2-NA3)++2+(NA3-NA1)++2)++5
      EA=AMAX1 (ABS(EXA3) + ABS(ENA3))
      1+ (LA-+Y/+S1) 2363, 302, 302
2363 X81=5H3+583=5PR3+5PA3=5RS3
      XB2=-SCE03+SR3
      883=513
      NE1=5H1-583-5PR3+5PA3-5R53
      NBS==SCEH3+2K3
      NB3=ST3
      EXB3=, /0/11*((XB1-X82)**2*(X82-X83)**2*(X83-X81)**2)**.5
      ENB3= . /0/11*((NB1-NB2) ++2+(NB2-NB3) ++2+(NB3-NB1) ++2) ++.5
```

13-25

```
EU#AMAX1 (AUS (EXU3) .AUS (ENU3) )
If (EN-FY/F31)2373,302.302
      LOAD CALCULATIONS CONDITION 4
Ç
 2373 P4=P8
      FA4*P4* (PI/4.)* (8++2)
      FPL4=P4+(PI/4,)+(S0D++2-B++2)
      TL4=+ A4
      BL4=FPL4+FA4
С
      STRESS CALCULATION FOR CONDITION &
      8754#8L4/(8TA(1)+8N)
      IF (8754-BU/FS2) 4344,4344,417
  417 IF (.001-8NI) 4344.4344.313
С
      LOAU CALCULATION CONDITION 5
 4344 P5=FIM
      FA5=P5* (P1/4.) + (8++2)
      FPL5=P5+(PI/4.)+(SOD++2-B++2)
      115=FA5
      MP5=((L+H0+G0++2+FE)/(_91+V))+(C1+B++2+GAMMA+P5/(XT+(XT++2+C2+GE+G
     1AMMA)))
      051=-5PL5-TL5
      452=-FPL5*H4-MP5+ (P5*PI+6*SX) *XT/2.-TL5*H5
      1FE1
      454=-051=08A/(DBA-05A)
      Q55=UFA/ ((084-054) + (H2+H3))
      050=H5+021+025
      957==DFR#XT/(2.+USR)
      U58=1.+U57+(XT/2.)-U55+(H2+H3)
      459=-(453/DSR) + (XT/2.) + 456+454+ (H2+H3)
      M05=M01-459/458
      SL5=>L1-454-(459/458)+455
      RSL5=RSL1=(059/058)*057-053/DSR
      BL5=0L1-054-(059/058)+055-051
      IF (SL5-MSL) 331,5390,5390
 5390 SH5= (8L5+H2+SL5+H3+FPL5+H4+TL5+H6) / (L+G)++2+ (B+G0))
      SRS5=(RSL5+XT/2.)/(L+G]++2+(B+G0))
      SPR5=MP5/(L+G1++2+(B+G0))
      585=0.
      SPA5=FA5/AH
      SCENS=0.
      SCEA5=P5
      SR5=M05+(1.333+XT+LCE+1.)/(L+X)*+2+(B+G0))
      515=MUS+(Y/(XT++2+B))=Z+5R5
      FSS5=BL5/(TOD+PI+XT)
      SSC5=SL5/(((B+2,*TA)**2~B**2)*PI/4.)
      HTS5+HL5/(HTA(I) +BN)
      IF (8755-844/FS2) 5344, 5344, 418
  418 IF (. U01-8NI) 5344, 5344, 313
 5344 IF (SR5-FYR/F51)5315,8315,302
 5315 IF (>15+FYR/F51)5316+5316+302
 5316 IF (FSS5_FSU/FS1) 5317,5317,302
 531/ IF (5505-SUYR/FS1)5330,5330,625
  625 PHINT 626
 626 FORMAT (3X+19H SSCS EXCEEDS LIMIT)
 5330 SH1=SH
      SRS1=SHS
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SPR1=0.
     SB1×U.
     SPAL=0.
     SCEB1=0.
     SR1=SH
     511#51
     SCEALEO.
     XA1*-SH5+S85+SPR5+SPA5+SRS5
     XA2=-SCEA5+SR5
     XA3=ST5
     NA1=-SH1-SB1+SPR1+SPA1+SRS1
     NA2=-SCEA1+SR1
     NA3#ST1
     EXA5=.70/11+ ((XA1-XA2)++2+ (XA2-XA3)++2+ (XA3-XA1)++2)++2+
     ENA5=,70711*((NA)-NA2)**2+(NA2-NA3)**2+(NA3-NA1)**2)**.5
     IF (ABS(XA1)-ABS(NA1))5602,5602,5601
5602 CMAX5A=NA1
     CMINDA=XA1
     GO TO 5603
5601 CMAX5A=XA1
     CMIN5A=NA1
5603 EA=AMAX1 (ABS (EXA5) + ABS (ENA5))
IF (EA=F YR/FS1)5352,5352,302
5352 CALT5A= (CMAX5A-CMIN5A) /2.
     CMN5A=(CMAX5A+CMIN5A)/2.
     FALTSA=CALTSA
     SALTSA=FALTSA
     SMN5A=CMN5A
     SCHKSA=ABS (SMN5A+SALT5A)
     IF (SCHK5A .GT. FUR) 850,815
 815 IF (SCHK5A-FYR) 816,816,817
817 1F (ABS(SALT5A)-FY)818+818,819
 818 SMNSA=FYR-SALT5A
     GU TO 816
819 SMNSA=U.
816 SMAXSA=SMN5A+SALT5A
     SMIN5A=SMN5A-SALT5A
     RC1=SMIN5A/SMAX5A
     M=0
5350 M=M+1
     IF (RU1-R(M))5351,7004,5350
5351 FFC5A=((R(M)-RC1)/(R(M)-R(M-1)))*(FF(M)-FF(M-1))+FF(M-1)
     60 10 7005
7004 FFC54#FF(M)
7005 CUNTINUE
     IF (AUS (SMAX5A) -FF C5A) 5363, 5363, 850
5363 X81#+SH5+585=SPR5+SPA5=SR55
     XB2=-SCE#5+SR5
     XB3#ST5
     NB1=+SH1-SB1-SPR1+SPA1-SRS1
     NB2=-SCEB1+SR1
     NB3=ST1
     £XH5=.70711+((X6)=XB2)++2+(XB2-XB3)++2+(XB3-XB1)++2)++5
     ENH5=./0/11*((NB1=NB2)**2*(NH2=NB3)**2*(NB3=NB1)**2)**.5
     LB=AMAK1 (AH5(EXB5), ABS(ENB5))
     1F (EH-FAH/ES1) 151 451 305
```

```
721 1F (ABS(X81)-ABS(N81))5605,5605,5604
5605 CMAX58=N81
      CWIN28*X81
     GU TO 5362
5604 CMAX58=X81
      CHINDU=NUL
5362 CAL158= (CMAX58-CMIN58)/2.
      CMNDB=(CMAX58+CMIN58)/2.
     FALISH=CALISH
      SALTOB=FALTOBOKTE
      SMN58=CMN58+KTF
     SCHK58=ABS (SMN58+SALT5H)
      IF (SCHA58 .GT. FUR) 850,820
 H20 IF (SCHN58-FYR) 821,821,822
 822 IF (AUS (SAL 158) - FYR) 823,823,824
 823 SMN56=FYR-SALT58
     GO TO H21
 824 SMN50=0.
 821 SMAX58=SMN58+SALT58
     SMINSH=SMN5H-SALT58
     HC2=SMIN5H/SMAX5B
     M=0
5360 M=M+1
      IF (RC2-R(M)) 5361,7006:5360
5361 FFC50=((K(M)-RC2)/(R(M)-R(M-1)))*(FF(M)-FF(M-1))*FF(Mu1)
     GO TO 7007
7006 ++C50=+F(M)
7007 CONTINUE
      IF (ABS (SMAX58) -FF C58) 5403,5403,850
     LOAD CALCULATIONS CONDITION 6 COLD.
С
5403 Po=P
     FA6=26+ (21/4.) + (8442)
     FPE6=P6+(P1/4.)+(S0D++2-8++2)
     16=FA6+ . 5+ TBS+AT
     MP6=((L+HU+G0++2+FE)/(。91+V))+(C1+U++2+GAMM&+P6/(X1+(X]++2+C2+GE+&
    1AMMA)))
     MC6=MC2
     MCTU9=(P1/3+)+(B++2)+(BETA++2)+((FL+TH++3)/(12++(1++FHU++2)))+FTE+
    MHTG6=(PI/3.)*(B*+2)*(BETA**2)*((FE*TN**3)/(12.*(1.-FMU**2)))*FTE*
    1 (UTH1) + (1 + HETA + XT)
     MHTUG=-MHTGO
     TMINI=THIN
     TMIN2=TMIN+DTC1
     TMIN3=TMIN+DTC2+DTC1
     IMAXI=IMAX
     TMAX2=TMAX-DTH1
     IMAXJ=IMAX-DTH2-DTH1
     WEIC=-IL6-FPL6
     U03(=(P6+8+(H+TA))/(4,*TA*SEE)~(P6+8*((A+B)/2.))/(4,*(A+B)/2.)*FE
    1)-FTE*(TMIN2-70.0)*(B+1A)/2.+5%E*(TMIN1-70.0)*(B+TA)/2.
     G64C=((-G61C+D8A)/(U8A-D5A))+(B6段(XT+SX)+BTE+(TMIN3+70,0)+(SX+-T)
    1-STE+SX#(TMIN1-70.0)-FTE+XT#(TMIN2-70.0))/(D8A-DSA)
     465C=UFA/((H2+H3)*(UBA=DSA))
     066C=H2+061C+062C
```

467C=-UFR#XT/ (2. +DSR) 068C=1.+067C+(XT/2.)+065C+(H2+H3) 469C== (063C/DSR) + (XT/2.) + 066C+064C+ (H2+H3) M06=M01-0690/0680 SL6=SL1-W64C-(069C/068C)+065C RSL6=RSL1= (Q69C/Q68C) +Q67C-Q63C/DSR 846=011-064C-(069C/068C)\*065C-061C IF (SL6-MSL) 331,6390,6390 6390 061CN=-TL6-FPL6 962CH= (P6\*PI\*B\*SX) \*X1/2.-FPL6\*H4-MP6+MCTG6-TL6\*H5 W63CN= (P6\*8\*(8+TA))/(4.\*TA\*SEE) - (P6\*8\*((A+8)/2.))/(4.\*((A-8)/2.)\*F 12)-FIE+(TMIN2-70.0)+(B+TA)/2.+STE+(TMIN1-70.0)+(B+TA)/2. W64CN=((~461CN+DBA)/(DBA-DSA))+(BTE+(TMIN3-70.0)+(SX+XT)+STE+SX+(T 1MIN1-70.0)-FTE\*XT\*(IMIN2-70.0))/(DBA=DSA) 065CN=UFA/((H2+H3)\*(DBA-DSA)) 066CN=H2+061CN+062CN Q67CN=-UFR\*XT/(2.+DSR) 068CN=1.+067CN+(XT/2.)-Q65CN+(H2+H3) W69CN=-(W63CN/DSR) \* (XT/2.) +Q66CN+Q64CN\*(H2+H3) M06CN=M01-069CN/068CN SL6CN=5L1-064CN+ (069CN/068CN) +065CN HSLOCN=H5L1-(069CN/068CN)+067CN-063CN/DSR 846CN=811-084CN- (069CN/068CN) #065CN-061CN C STRESS CALCULATION CONDITION 6 COLD SH6CN=(BL6CN#H2+SL6CN#H3+FPL6#H4+TL6#H6)/(L#Q1##2#(B+GQ)) SRS6CN=(R5L6(N\*XT/2.)/(L\*G1\*\*2\*(B+G0)) SPR6CN=MP6/(L@G1\*#2\*(8+G0)) SCG6LN=MCTG6/(L#G1##2#(B+G0)) 586CN= . 54 TBM/ZH SPACCN=FA6ZAH SCEB8C=0. SCEA60=P6 SR6UN#HO6CN#(]\_333#XY#ECE#]\_}/(|#XT#2.#(B+GO)) 516CH=HU6CN+ (Y/ (X1++2+8)) -Z+SR6CN FSS6CN=BLOCN/(TOD+PI#XT) 55C6UN=5L6CN/(((B+2+\*TA)\*\*2-8\*\*2)\*PI/4.) BTS6CN=BL6CN/(BTA(I)+BN) IF (BTS6CN-BYC/FS3) 6344,6344,419 419 1F (+001-MNI)6344,6344,313 6344 18 (SHOCN-FYC/F51)6315,6315,302 6315 1F (ST6UN-FYC/FS1)6316+6316+302 6316 1F(FSS6CN-FSU/FS1)6317,6317,302 6317 IF (55060N-SCYC/F51)6330.6330.627 627 PRINT 528 628 FORMAL (JA+21H SSC6CN EXCEEDS LIMIT) 6330 XA1=-SHACH+SH6CN+SPR6CN+SPA6CN+SRS6CN-SCG6CN XA2=-SCEA6C+SP6CN XA3#STOCN とXA6レN=+107注2や((XA1-XA2)442+(XA2-XA3)442+(XA3-XA3)4#2)##2)##665 EAREAABCH IF (LA=FYC/FS1)6363.6363.302 6363 XHI#+SHPCN+SHPCN-SPR6CN+SPA6CN-SRS6CN+SCG6CN XU2=-SCENDC+SHOON XU3=515CN £X86~N==1071)•((XB注=X82)+42+(X82=X83)##2+(X83=x83)##2>##2>##65

EH=EABOCN

13 J.)4

```
1F (EH-FYU/FS1)6371,6371,302
      LOAD CALCULATIONS CONDITION 6 HOT
С
 6371 Q01H=-1L6-FPL6
      Q62d=(P6+PI+6+SX)+XT/2.-FPL6+H4-MP6-MC6+MHTG6-TL6+H5
      Q63H=(P6+U+(B+TA))/(4.+TA+SEE)-(P6+B+((A+B)/2.))/(4.+((A+B)/2.)+FE
     1)-FTE*(TMAX2-70.0)*(B+TA)/2.+STE*(TMAX1-70.0)*(B+TA)/2.
      464H=((-461H+DBA)/(DBA-DSA))+(BCR*(XT+SX)+BTE*(7MAX3-70.0)*(5X+XT)
     1-STE#5x#(TMAX1-70.0)=FTE#XT#(TMAX2-70.0))/(DBA-DSA)
      U65HEDFA/((H2+H3)+(UBA-DSA))
      4994=H5+001H+065H
      467M=-UFR*X1/(2.4DSR)
       58n=1.+Q67H+(X1/2.)-Q65H+(H2+H3)
      WB H== (W63H/DSR) # (XT/2.) + 466H+ 664H# (H2+H3)
      MQ6H#M()1-469H/Q68H
        5H=SL1=464H= (469H/468H) +465H
         6H=H5L1-(069H/068H) #067H-063H/USR
      8L6H=8L1-064H- (069H/088H) +065H-061H
       F (SL6H-M5L) 331,8390,8390
 6.90 4 HN=-116-FPL6
      062 IN= (P6+PI+8+SX) +XT/2.-FPL6+H4-MP6+MHTG6-TL6+H5
      963HN=(P6464(B+TA))/(4.4TA4SEE)=(P64646((A+B)/2.))/(4.4((A+B)/2.))
     1E) -FTE* (IMAX2-70.0)*(B+TA)/2.+STE*(TMAX1-70.0)*(B+TA)/2.
      464HN=((-461HN+0BA)/(08A-0SA))+(8TE+(TMAX3-70.0)+(SX+XT)-STE+SX+(T
     1MAX1-70.0)-FTE#XT#(TMAX2-70.0))/(DBA-DSA)
      465HN=0FA/((H2+H3)+(DBA-DSA))
      066HN=H2#061HN+062HN
      067HN=-UFR#XT/(2.+05R)
      468HN=1.+467HN+(XT/2.)-965CN+(H2+H3)
      469HN=-(463HN/DSR)*(XT/2.)+466HN+464HN*(H2+H3)
      M06HN=M01-069HN/068HN
      SL6HN=5L1-464HN- (Q69HN/Q68HN) +Q65HN
      HSL6MN=RSL1-(Q69HN/Q68HN)+Q67HN-Q63HN/DSR
      BL6HN=0L1-Q64HN- (Q69HN/Q68HN) +Q65HN+Q61HN
C STRESS CALULATION CONDITION 6 HOT
      5H6HN= (RL6HN*H2+SL6HN*H3+FPL6*H6+TL6*H6)/(L*G1**2*(B+G0))
      SKS6HN= (K5L6HN+X1/2.)/(L+G1++++(8+60))
      SHG6CN=MHTG6/(L+G1++2+(B+G0))
      SPR6HN=MP6/(L+G1++2+(B+G0))
      SEGHN=.5+TBM/2H
      SPAGMN=FA6/AH
      SCEN6H=0.
      SCEA6H=Po
      SH6HN=MU6HN4 (1.333*XT*LCE+1.)/(L*XT*2.*(B+G0))
      516HN=H06HN*(Y/(XT##2#8))-2#586HN
      FSSOHN=HLOHN/(TOUOPIOXT)
      SSCOMN=SLOHN/(((B+2+*TA)**2-8**2)*PI/4.)
      BISCHN=HLCHN/(BIA(I)+BN)
      IF (UISONN-BY/FS3)8344+8344+420
  420 IF (+001-6H1) 8344+8344+313
8344 11 (SR6HN-FY/F51)8315,8315,302
H315 IF (STONN-FY/FS1) H316, 8316, 302
8316 IF (FSSOMN-FSU/FS1)8317,8317,302
831/ IF (SSCHHN-SCY/FS1)8330,8330,629
  658 AHINI 030
  630 FORMAT (3X+2)H SSC6HN EXCEEDS LIMIT)
B330 XA1X-SHAHN+SBAHN+SPR6HN+SPA6HN+SRS6HN-SHG6CN
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13-30

```
XA2=-SCEAGH+SR6HN
      XA3=STOHN
      LXADHN= . /0/114 ( (XA1-XA2) + /2+ (XA2-XA3) ++2+ (XA3-XA1) ++2) ++.5
      EA=EXANHN
      IF (LA-FY/FS1)8363,302,302
 8363 XB1=+SH6HN+SB6HN-SPR6HN+SPA6HN-SRS6HN+SHG6CN
      XU2=-SCEUGH+SR6HN
      XU3=STOHN
      EX86mN=.70711+((X81-X82)++2+(X82-X83)++2+(X83-X81)++2)++5
      FR=FXR0HN
      IF (EB-FY/FS1)8371+302+302
      LUAU CALCULATION CONDITION 7
С
 8371 H/=0.
      FA7=V.
      FPL1=0
      TL7==5#185#AT
      071 = (0+R/USR) + (XT/2.) + (XT/2.)
      W72= (DFA) / (DBA-DSA)
      U73=TL/+UHA+(H2+H3)/(DBA-DSA)-TL7+H2-TL7+H5
      M07=MU1-U/3/(1.-U71-072)
      RSL7=RSL1-(M01-M07) +DFR+(XT)/2.+DSR
      SL7=SL1-TL7+DBA/(DBA-DSA)- (MO1+MO7)+DFA/((H2+H3)+(DBA-DSA))
      867=861-561+567+T67
      1F (0.001-UN1) 871.870.870
  HTU BISCHK=MAXIF (BIS2P,BIS3,BIS6HN)
      IF (BISCHK .LT. 0.83*84/FS2)873.871
  873 IF (8154 .LT. 0.83*8U/FS2)872+871
  H15 BN1=(BN#BISCHK#F$5)/(0.83#BY)
      BN2=(P1+BCD)/(2.+(BS(I)+XT))+2.
      BN=MAX1F(BN1,BN2)
      NB=BN/2.
      N=2#NH
      BN=14
      BNI=BN
      11=1
      BCD1=BCD
      GO TU 114
  871 CONTINUE
      T1=P1/4.*1004#2
      B1=P1/4.*(TOD+2.*FUBI)**2
      WGT=2.*(11/4.*(A*+2-TGD++2-BN+BH(I)*+2)*XT)*FRHQ+2.*BN+PI/4.*(BS(I
     1) ++2) * (SX+XT+2.*85(I)) *BRH0+2.*8N*NWT(I) +2.*H*FRH0*(1./3.*(B1+T1+(
     101011)00.5)-T1)-2.0PI/4.0(SOD002-TOD002)0.0250FRH0-PI/4.0(SOD002-(
     18+2.+TA)++2)+SLEG+TAN(PI/9.)+FRH0
      THQ=(BL1+85(I))/(BN+5.)
      IF (PI*HCU/HN=2.*(BS(I)+XT))333+333+1000
 1000 BRINI 1001
 1001 FORMAT (JUH MAXIMUM SPACING NOT SATISFIED)
  333 CONTINUE
      641N1 151
  727 FORMAT(LH )
      PRINT 728
  128 FORMAT (25X+14H DESIGN OUTPUT///)
      DHINE 5800
 2000 FURMAT(1x+250BENDING MOMENT AND STRESS)
      BRINE 5801
```

2801 FURMAT (61. 3HTHM. 7X. 3HTBS) PHINT 9020. TBM. TBS PHINI 2803 2803 FURMAT (7X. 3HTRO) PRINT YOZU, TRU PRINT JOUL 3001 FURMAT (25H LOADS FOR ALL CONDITIONS) PRINT 3002 3002 FORMAT (61, 3HM01, 7X, 3HBL1, 7X, 3HSL1, 6X, 4HRSL1) PRINT 9020+MO1+BL1+SL1+RSL1 PRINI 3003 3003 FORMAT (6X, 3HM02, 7X, 3HBL2, 7X, 3HSL2, 6X, 4HRSL2, 7X, 3HMC2, 7X, 3HMP2) PRINT 9020.M02.8L2.SL2.RSL2.MC2.MP2 PRINE 3004 3004 FURMAT (5X+4HMO2P+6X+4HBL2P+6X+4HSL2P+5X+5HRSL2P) PRINT 9020, MO2P, BL2P, SL2P, HSL2P PRINT 3005 3005 FURMAT (5x,4HMO2M,6x,4HBL2M,6x,4HSL2M,6x,5HRSL2M) PHINT Y020, MO2M, BL2M, SL2M, RSL2M PRINE 3006 3006 FURMAT (6X, 3HM03, 7X, 3HBL3, 7X, 3HSL3, 6X, 4HRSL3, 7X, 3HMP3) PRINT 9020+M03+BL3+SL3+RSL3+MP3 PRINI 3007 3007 FORMAT (6X+3HBL4) PRINT 9020,8L4 PRINE 3008 3008 FORMAT (6X, 3HM05, 7X, 3HHL5, 7X, 3HSL5, 6X, 4HRSL5, 7X, 3HMP5) PRINT 9020, M05, BL5, SL5, RSL5, MP5 PRINE 3009 3009 FORMAT (6X+3HM06,7X+3H8L6+7X+3HSL6+6X+4HRSL6+7X+3HMC6+7X+3HMP6) PRINT 9020, M06, BL6, SL6, RSL6, MC6, MP6 PRINT 3010 3010 FURMAT (5X+5HMCTG6+5X+5HMHTG6) PRINT 9020, MCTG6, MHTG6 PRINE 3011 3011 FURMAT (5X, 5HHOGCN, 5X, 5HBLGCN, 5X, 5HSLGCN, 4X, 6HRSLGCN) PRINT 9020, MOSCN, BLOCN, SLSCN, RSLSCN PRIN1 3012 3012 FORMAT (5X,4HMO6H,6X,4HBL6H+6X,4HSL6H,5X,5HRSL6H) PHINT Y020, MO6H, BL6H, SL6H, RSL6H PHIN1 3013 3013 FURMAT (5X+5HMO6HN+5X+5HBL6HN+5X+5HSL6HN+4X+6HRSL6HN) PRINT YOZU, MOGHN, BLOHN, SLOHN, RSLOHN PRINE 3014 3014 FURMAT (6x, 3HM07, 7x, 3HBL7, 7x, 3HSL7, 6x, 4HRSL7) PRINT 020, M07, BL7, SL7, RSL7 PRINI 9104 9104 FURMATIIX+27HSTRESS CONCENTRATION FACTOR) PRINT 9105 9105 FURMAT(6X+3HKTF) PHINT 9021.KTF 541N1 3012 3015 FORMAT (28H STRESSES FOR ALL CONDITIONS) PHINT JUIG 3016 FORMAT (6X+3HEXA+7X+3HENA+7X+3HEXB+7X+3HENB) PRINT YOZU+EXA, ENA+EXB. ENB

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B 32

PRINT JOIT 3017 FURMAT (54.4HEXA2.6X.4HENA2.6X.4HEXB2.6X.4HENB2) PHINT 9020 . EXA2 . ENA2 . EXB2 . ENB2 PRINT 3018 301H FURMAT (4X+6HSMAX2A+4X+6HSMIN2A+5X+5HFFC2A+4X+6HSMAX2B+4X+6HSMIN2B+ 15X+5HFF(2H) PHINE YOZO, SMAXZA, SMINZA, FFCZA, SMAXZU, SMIN2B, FFCZU PRINT 122 722 FURMATISX, 4H5H2P, 5X, 5H5R52P, 6X, 4H5H2M, 5X, 5H5R52M, 6X, 4H5PR2, 7X, 3H5B 12+6X+4HSPAZ) PRINT 9020, SH2P, SRS2P, SH2M, SRS2M, SPH2, SB2, SPA2 FEL INIAH 723 FURMAT (5X+4HSH2P+6X+4HSR2M+6X+4HST2P+6X+4HST2M) PRINT 4020, SR2P, SR2M, ST2P, ST2M PRINE 3019 3019 FURMAT (5x.4HEXA3.6X.4HENA3.6X.4HEXB3.6X.4HENB3) FRINT 4020.EXA3.ENA3.EXB3.BB3 PRINE 3020 3020 FORMAT (5x,4HEXA5,6X,4HENA5,6X,4HEX85,6X,4HEN85) PRINT 9020 EXAS. ENAS. EXHS. ENBS PRINT 3021 3021 FURMAT (4X+6HSMAX5A,4X,6HSMIN5A,5X,5HFFC5A,4X,6HSMAX58,4X,6HSMIN58, 15X+5HF+C58) PHINT 9020, SMAX5A, SMINSA, FFC5A, SMAX58, SMINSB, FFC50 FRINT 124 724 FORMAT (64, 3HSH5, 6X, 4HSRS5, 7X, 3HSB5, 6X, 4HSPA5, 7X, 3HSR5, 7X, 3HST5, 16X,4HSPR5) PRINT 9020, SH5, SRS5, SB5, SPA5, SR5, ST5, SPR5 PRINT 725 725 FORMAT (6X, 3HSH1, 6X, 4HSRS1, 7X, 3HSB1, 6X, 4HSPA1, 7X, 3HSR1, 7X, 3HST1, 16X, 4HSPR1) PRINT 9020, SH1, SRS1, SB1, SPA1, SR1, ST1, SPR1 PRINT 3023 3022 FORMAT (54+6HEXA6CN+5X+6HEX86CN+5X+6HEX86HN+5X+6HEX86HN) PRINT 9020 + EXAGEN, EXBOEN, EXAGHN, EXBOHN PRINT 3029 3029 FURMAT (62+3HBTS+5X+5HBTS2P+6X+4HBTS3+6X+4HBTS4+6X+4HBTS5+4X+ 15HBISOCN, 4X, 6HBIS6HN) PRINE 9020, BTS, BTS2P, BTS3, BTS4, BTS5, BTS6CN, BTS6HN PRINT 3023 3023 FORMAT (11H DIMENSIONS) PRINT 2902 2HO2 FURMAY (94,1HH+64,4HHU81,84,2HG1,8X,2HFR) PRINT Y021+H+HUBI+G1+FR PHINE 3024 3024 FURMAT (6x+2HTW+7x+3HTOD) PRINT YORLOTWOTOD CSOE INING 3025 FORMAT (6x, 1HA, 9x, 1HB, 8x, 2HXT, 7x, 3HBCD) PRINT Y021+A+H+XT, BCU PRINT 3026 3026 FURMAT (64+3HSUD, 7X+3HSID, 7X, 3HSEL . BX, 2HTA. 5X+5HSLEOT) PRINT JO21.SUD.SID.SEL.TA.SLEGT LEDE INTHA 3027 FURMAT (6x+5HBS(I), 5x, 5HBH(I), 4x, 6HBTA(I), 4x, 6HBWC(I), 6x, 2HBN) PHINT 9021,85(1),88(1),874(1),8WC(1),8N

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PRINT 3028

3028 FORMAT (6x, 3HWGT)

PRINT 9021+WGT

PRINT 9103

9103 FURMAT(1H1)

GO TO 999

998 CALL EXIT

END
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PROGRAM LAROP (INPUT.OUTPUT.TAPE60#INPUT)
      TYPE HEAL LI. MSL. K. CE. LCD. L. MP2. MC2. MO1. MA1. NA2. NA3
      TYPE REAL NR1+NR2+NR3+MO2+MO2P+MO2H+MP3+MO3+MP5+M05+M06C+MC6+
     1HCIG6+MHIG6+MO6CN+MO6H+MO6HN+MO7+MH+MP6
      TYPE HEAL KR
      TYPE REAL MPL.MR2.MR2P.MR2M.MR3.MR5.MR6C.MR6CN.MR6H.MR6HN.MR7
      TYPE HEAL NWT
                                                            TYPE HEAL KIF
      TYPE REAL MOLT
      DIMENSION R(11) + FF(11) + BH(15) + BWC(15) + RS(15) + BTA(15) + NWT(15)
      DIMENSION DAF (15)
      NNN=1
C TUBE LOAD-INPUT
  999 DEAD 1020. TRM1
      TF (EOF+60) 998.997
C TURE GEOMETRY
  997 PEAU 1021+TOD+TW1
 1020 FORMAT (3F)0.0)
 1021 FORMAT (HF10.4)
 1022 FORMAT (HE10.9)
 1023 FORMAT (2110)
C TUBE MATERIAL PROPERTIES
      READ 1020 + TU+ TY + TE+ TR+ TERO
C FLANGE GEUMETRY
      ITAURATANISUL GAJA
                                                          - · · ·
      READ 1021+6101WT
      READ 1021 FRI FSFI
C FLANGE MATERIAL PROPERTIES
      READ 1020-FUR
      READ 1020 + FU.FY + FSU + FCY + FE + FG + FYC + FYR
      PEAD 1020 .FCYR. FCYR
                                                          . ... ....
      PEAD 1022 + FRHO + FMU + FTE > FCP
      READ 1021. (R(NN), NN=1+8)
      READ 1021. (R(NN).NN=9.1))
      DEAD 1020. (FF (NN) . NN=1.8)
      READ 10204 (FE(NN) + NN=9+11)
C SEAL GEOMETRY
      READ 1021+SOOT+STUI+SELI+TOJ+TAOTW+SLEGTT
C SFAL LUADS-THPUT
      READ 1020 . RSLT. SSLT. MSLT
C SEAL MATERIAL PROPERTIES
      HEAD 1020+SCY+SEF+SCYC+SCYR+SYR
      READ 1022.STF
C HOLT GLOMETRY
      DEAD 1(21+ (BH(NN)+NN=1+8)
      RFAD 1021 + (RH(NN) + NN=9+15)
      READ 1021 + (RWC (NN) + NN=1+8)
      READ 1021 . (RWC(NN) . NN=9.15)
      REAU 1021 . (AS (NN) .NN=) .8)
      READ 1021 . (AS(NN) .NN=9+15)
      DEAD 1021+ (RTA (NN) +NN#1+B)
      READ 1021+ (RTA(NN)+NN#9+15)
      DEAD 1021 . (NAT (NN) . NN=1,8)
      READ 1021 . (NAT (NN) . NN=9.15)
      READ 102) . (DAF (NN) , NN=1.8)
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DFAU 1021, (DAF (NN), NN=9,15)

```
C ROLT INPUT
      DEAD 1023.TT. THAX
      OFAD INCONT
C HOLT MATERIAL PROPERTIES
      READ LIDO , RY , RE , HU , RYC , RYP
      READ LOPP. HTE , ACR , BRHO
C RING GEDMETRY AND PROPERTIES
      PEAD 1021, PCDT
      PEAD 1020, PH, RY, RSU, RCY, RE, PG, RYC, RYR
      READ 1920-ROYC. ROYR
      READ 1022. HTE .RCH. RRHO
C SYSTEM PRESSURE AND TEMPERATURE
      HEAD LOPE, P. PP. PH. PIM. TMAX. TMIN
      HETO+HITO+012C+013C+011H+UT2H+013H
      READ 1021+FS1+FS2+FS3
      ORINT 126
  126 FORMAT (25% , 19HINPUT TO THE DESIGN///)
      PRINT 2020
 2020 FORMAT (THE LOAD-INPUT)
      PHINT 2021
 2021 FORMAL (3x,4HTHM))
      PRINT 4021 THAT
      DETAL SUUD
 2000 FORMAL (14H TURE GEOMETRY)
      PRINT 2001
 2001 FORMAT (68, 3HTOD.78, 3HTW1)
      PRINT 9021, TOD. TW1
      PRENT 2002
 2002 FURMAT (25H THRE MATERIAL PROPERTIES)
      PHINT 2003
 2003 FURMAT (7X+2HTU+RX+2HTY+8X+2HTF+8X+2HTR+8X+4HTFR0)
      PHINT 9020. TU.TY. TF. TR. TFRO
      PR[N7 2064
 2004 FORMAT (16H FLANGE GEOMETRY)
      PPINT 2005
 POUS FORMAT(7x, 2HAI, 5X, SHHUHII)
      DRINT YOZI . AT . HUBIT
      PRINT HOD
  HOD FORMAT (3X, 64410TWI)
      PRINT JO21.GIOTWT
 2037 FORMAT (6x, 3HERT. 6X, 4HESET)
      PRINT YORI.FRI.FSFI
      PHINT 2006
 2006 FORMAT (27H FLANGE MATERIAL PROPERTIES)
      PRINT HOL
  HOI FORMAL (6X. THELIP)
      PRINT POPALFUR
PRINT POOT
 2007 FURMAT (7x, 24F1, HX, 24FY, 7X, 34FSU, 7X, 34FCY, 8X, 24FE, 8X, 24FG, 7X,
     13HFYC+/X+3HFYR)
      PRINT 4020, FIL, FY . FSU . FCY . FE. FG. FYC. FYR
      PHINT 2040
 2940 FORMAT (FX.4HECYC.6X.4HECYR)
      PPINT GC20.FCYC.FCYR
      PRINT 2008
```

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2008 FORMAT (5x,4HERHO,5X,3HEMU,7X,3HETE,7X,3HECR)
9020 FORMAT (9F10.0)
9021 FORMAT (PF10.4)
9022 FORMAT (RETORS)
9023 FORMAT (2110)
PHINT 4022. FRHO. FMU.FTE.FCH
PD05 TMIRE
2009 FORMAT (5x.4HR(1).6X.4HR(2).6X.4HR(3).6X.4HR(4).6X.4HR(5).6X.
14HH(6)+6X+4HH(7)+6X+4HH(H))
9RINT 9021-R(1) - R(2) - R(3) - R(4) - R(5) - R(6) - R(7) - R(8)
PRINT 2010
2110 FORMAT (5X.4HR(9).5X.5HR(10).5X.5HR(11))
$PRI 41 = 4021 \cdot P(9) \cdot P(10) \cdot P(11)$
PRINT 2011
2011 FOHMAL (AX.SHEE(1).5X.SHEE(2).5X.SHEE(3).5Y.SHEE(A).5X.SHEE(5).5Y.
15HFF (6) +5X-5HFF (7) +5X +5HFF (8) )
DRINT 9020 SELIN SEE 23 SEE 23 SEE 23 SEE 24 SEE 25 SEE 25 SEE 27 SEE 29
2012 EOHMAT (4X, EHEE/0), 4X, 6HEE/10), 4X, 6HEE/11))
$\frac{\partial P(NT)}{\partial P(NT)} = \frac{\partial P(T)}{\partial P(T)} + $
2013 EURANI (144 SEAL GEOMETRY)
2014 FORMAT ISY ANSADIARY ANSIDIARY ANSIDIARY SHIATASSA SHIATASIA AY ANSIF
DRINT 9021-SOUT-SIDT-SELT-TAT-TACTW-SLECTT
DRINE 2015
PRINT 2016
2016 FORMAT (TAH SEAL LUADS INPOLS) PRINT 2016 2016 FORMAT(SA, AHRS) I, AX, AHSSI I, AX, AHMSI I)
PHINT 2016 PHINT 2016 2016 FORMAT(5x+4HPSLT+4X+4HSSLT+4X+4HMSLT) PHINT 9020-PSLT-SSLT+MSLT
PHINT 2016 PRINT 2016 PRINT 9020.RSLI.SSLI.MSLI PRINT 2017
PRINT 2016 PRINT 2016 PRINT 9020.RSLI.SSLI.MSLI PRINT 2017 2017 FURMAT (25H SEAL MATERIAL PROPERTIES)
PRINT 2016 PRINT 2016 PRINT 9020.RSLI.SSLI.MSLI PRINT 2017 2017 FORMAT (25H SFAL MATERIAL PROPERTIES) PRINT 2014
PHINT 2016 2016 FORMAT (TAH SEAL LUADS INPOTS) PHINT 2016 PHINT 9020.RSLI.SSLI.MSLI PHINT 2017 2017 FORMAT (25H SFAL MATERIAL PROPERTIES) PHINT 2018 2018 FORMAT (74.28504.74.28504.645040.644.485040.74.285040.74.285040.
PHINT 2016 2016 FORMAT (54.4HPSLI.4X.4HSSLI.4X.4HMSLI) PHINT 9020.RSLI.SSLI.MSLI PHINT 2017 2017 FORMAT (25H SFAL MATERIAL PROPERTIES) PHINT 2018 2018 FORMAT (7X.3HSCY.7X.3HSEE.6X.4HSCYC.6X.4HSCYR.7X.3HSYR) PHINT 9020.SCY.SEE.SCYC.SCYP.SYR
2015 FORMAT (TAH SEAL LUADS INPOTS) PRINT 2016 2016 FORMAT(5A+4HPSLI+4X+4HSSLI+4X+4HMSLI) PRINT 9020.RSLI+SSLI+MSLI PRINT 2017 2017 FORMAT (25H SFAL MATERIAL PROPERTIES) PRINT 2018 2018 FORMAT (7X+3HSCY+7X+3HSEE+6X+4HSCYC+6X+4HSCYR+7X+3HSYR) PRINT 2019 PRINT 2019
PHINT 2016 2016 FORMAT (TAH SEAL LUADS INPOTS) PHINT 2016 PHINT 9020.RSLI.SSLI.MSLI PHINT 2017 2017 FORMAT (25H SFAL MATERIAL PROPERTIES) PHINT 2018 2018 FORMAT (7X.3HSCY.7X.3HSEE.6X.4HSCYC.6X.4HSCYR.7X.3HSYR) PHINT 9020.SCY.SEE.SCYC.SCYR.SYR PHINT 2019 2019 FORMAT (3X.2HSTE)
<pre>PHINT 2016 PHINT 2016 2016 FORMAT(5A+4HPSLI+4X+4HSSLI+4X+4HMSLI) PHINT 9020.RSLI+SSLI+MSLI PHINT 2017 2017 FORMAT (25H SFAL MATERIAL PROPERTIES) PHINT 2014 2019 FORMAT (7X+3HSCY+7X+3HSEE+6X+4HSCYC+6X+4HSCYR+7X+3HSYR) PHINT 9020+SCY+SEE+SCYC+SCYR+SYR PHINT 9020+SCY+SEE+SCYC+SCYR+SYR PHINT 2019 2019 FORMAT (3X+3HSTE) PHINT 9020 STE</pre>
PHINT 2016 PHINT 2016 2016 FORMAT(5A+4HPSLI+4X+4HSSLI+4X+4HMSLI) PHINT 9020.RSLI+SSLI+MSLI PHINT 2017 2017 FORMAT (25H SFAL MATERIAL PROPERTIES) PHINT 2014 2018 FORMAT (7X+3HSCY+7X+3HSEE+6X+4HSCYC+6X+4HSCYR+7X+3HSYR) PHINT 9020+SCY+SEE+SCYC+SCYR+SYR PHINT 9020+SCY+SEE+SCYC+SCYR+SYR PHINT 2019 2019 FORMAT (3X+3HSTE) PHINT 9022,STF PHINT 2022
<pre>&gt;&gt;15 FORMAT (TAH SEAL LUADS INPOTS)</pre>
<pre>PHINT 2016 PHINT 2016 PHINT 9020.RSLI.4X.4HSSLI.4X.4HMSLI) PHINT 9020.RSLI.SSLI.MSLI PHINT 2017 2017 FURMAT (25H SFAL MATERIAL PROPERTIES) PHINT 2018 2019 FURMAT (7X.3HSCY.7X.3HSEE.6X.4HSCYC.6X.4HSCYR.7X.3HSYR) PHINT 9020.SCY.SEE.SCYC.SCYR.SYR PHINT 2019 2019 FURMAT (3X.3HSTE) PHINT 2022 2022 FURMAT (14H HOLT GEOMETRY) PHINT 2023</pre>
PHINT 2016         PHINT 2016         PHINT 9020.RSL1.SSL1.MSL1         PHINT 2017         POIT FORMAT (25H SFAL MATERIAL PROPERTIES)         PHINT 2014         POIR FORMAT (7X.3HSCY.7X.3HSEE.6X.4HSCYC.6X.4HSCYR.7X.3HSYR)         PHINT 2019         POIR FORMAT (3X.3HSCY.SCY.SCYR.SYR         PHINT 2019         POIR FORMAT (3X.3HSTE)         PHINT 9022.STF         PHINT 2022         POIR FORMAT (14H HOLT GEOMETRY)         PHINT 2023
<pre>PHINT 2014 PHINT 2014 PHINT 2017 PHINT 2017 PHINT 2017 PHINT 2017 PHINT 2017 PHINT 2014 PHINT 2014 PHINT 2014 PHINT 2014 PHINT 4020+SCY+SEE+6X+4HSCYC+6X+4HSCYR+7X+3HSYR) PHINT 4020+SCY+SEE+SCYC+SCYR+SYR PHINT 4020+SCY+SEE+SCYC+SCYR+SYR PHINT 2019 PHINT 2019 PHINT 2019 2019 FORMAI (3X+3HSTE) PHINT 2023 2022 FORMAI (14H HOLT GEOMETRY) PHINT 2023 2022 FORMAI (14H HOLT GEOMETRY) PHINT 2023 2023 FORMAI (4X+SHRH(1)+5X+SHRH(2)+5X+SHRH(3)+5X+SHRH(4)+5X+SHRH(5)+5X+ PHINT(2)+5X+SHRH(2)+5X+SHRH(4)+5X+SHRH(5)+5X+</pre>
<pre>Prive Pormat (IAH SEat L(IADS INPUTS)</pre>
<pre>&gt;&gt;15 FORMAT (THH SEAL LOADS INPOLS)</pre>
<pre>&gt;</pre>
<pre>&gt;&gt;15 FORMAT (IAH SEAL L(MOS INPOLS)</pre>
<pre>Prive Prive Prive</pre>
<pre>Prist Pormal (THH State (TWP) TWP(TY)</pre>
<pre>&gt;&gt;15 FORMAT (TAH SEAL L(MAS INPOIN) pHINT 2014 &gt;&gt;16 FORMAT(5A,4HPS(I,4X,4HSSLT,4X,4HMSLT) pHINT 2017 &gt;&gt;17 FORMAT (25H SFAL MATERIAL PROPERTIES) pHINT 2019 &gt;&gt;18 FORMAT (7X,3HSCY,7X,3HSEE,6X,4HSCYC,6X,4HSCYR,7X,3HSYR) pHINT 4020,5CY,SEE,SCYC,SCYR,SYR pHINT 4020,SCY,SEE,SCYC,SCYR,SYR pHINT 2019 &gt;&gt;19 FORMAT (3X,3HSTE) pHINT 4022,STF pHINT 2023 &gt;&gt;2022 FORMAT (144 HOLT GEOMETRY) pHINT 2023 &gt;&gt;215 FORMAT (4X,5HBH(1),5X,5HBH(2),5X,5HBH(3),5X,5HBH(4),5X,5HBH(5),5X, 15HBH(6),5X,5HBH(7),5X,5HBH(R) pHINT 2024 &gt;====================================</pre>
<pre>&gt;&gt;1(3 FORMAT (14H SEAL LUADS INFOLS)</pre>
<pre>&gt;&gt;15 FORMAT (1AH SEAL LINE) INPOTS)     pFINT 2016 &gt;&gt;16 FORMAT (5% 4HPSLI +4% 4HSSLI +4% 4HMSLI)     pFINT 2017 &gt;&gt;17 FORMAT (25H SFAL MATERIAL PROPERTIES)     pFINT 2014 &gt;&gt;18 FORMAT (7% 3HSCY 7% 3HSEE 6% 4HSCYC 6% 4HSCYR 7% 3HSYR)     pHINT 2014 &gt;&gt;18 FORMAT (17 + 3HSCY 7% 3HSEE 6% 4HSCYC 6% 4HSCYR 7% 3HSYR)     pHINT 2019 &gt;&gt;19 FORMAT (14 + 40] T GEOMETRY)     pHINT 2022 &gt;&gt;2022 FORMAT (14 + 40] T GEOMETRY)     pHINT 2023 &gt;&gt;2022 FORMAT (14 + 40] T GEOMETRY)     pHINT 2023 &gt;&gt;212 FORMAT (14 + 40] T GEOMETRY)     pHINT 2023 &gt;&gt;212 FORMAT (14 + 40] T GEOMETRY)     pHINT 2023 &gt;&gt;212 FORMAT (14 + 40] T GEOMETRY)     pHINT 2023 &gt;&gt;212 FORMAT (14 + 40] T GEOMETRY)     pHINT 2023 &gt;&gt;224 FORMAT (14 + 40] +5% 5HRH(2) +5% 5HRH(3) +5% 5HRH(4) +5% 5HRH(5) +5% +     15HH(6) +5% 5HRH(1) +5% 5HRH(2) +5% 5HRH(3) +5% 5HRH(4) +8H(5) +6HRH(6) +6H(1) +6</pre>
<pre>Prive 2014 Prive 2016 Prive 2016 Prive 2017 Point 2017 Point 2017 Point 2017 Point 2018 Prive 2018 Prive 2018 Prive 2018 Prive 2018 Prive 2018 Prive 2018 Prive 2018 Prive 2018 Prive 2019 Prive 2019 Prive</pre>
<pre>Print 2016 Print 2016 Print 2017 Point 2017 Point 2017 Point 2017 Point 2017 Point 2014 Point 2020 Point 2020 Point</pre>
<pre>print 2016 2016 F0FMAI(5x,4HPSL1,4X;4HSSL1,4X;4HMSL1) pFINT 9020.RSL1.SSL1.MSL1 pFINT 2017 2017 F0FMAI (25H SFAL MATERIAL PROPERTIES) pFINT 2014 2019 F0FMAI (7x,3HSCY,7x,3HSEE.6X;4HSCYC.6X;4HSCYR,7X,3HSYR) pFINT 2019 2019 F0FMAI (7x,3HSCY,7X,3HSEE.6X;4HSCYC.6X;4HSCYR,7X,3HSYR) pFINT 9020.SCY,SEE.SCYC.SCYR,SYR pFINT 9020.SCY,SEE.SCYC.SCYR,SYR pFINT 9020.SCY 0020 F0FMAI (3x,3HSTE) pFINT 9020.SCY 0020 F0FMAI (14H H0LT GEOMETRY) pFINT 9020.SCY 0022 F0FMAI (14H H0LT GEOMETRY) pFINT 2023 2022 F0FMAI (4x,5HRH(1),5X,5HRH(2),5X,5HRH(3),5X,5HBH(4),5X,5HBH(5),5X, 15HDH(6),5X,5HBH(7),5X,5HRH(2),5X,5HRH(3),5X,5HBH(4),5X,5HBH(5),5X, 15HDH(6),5X,5HBH(7),5X,5HRH(2),5X,5HRH(3),5X,5HBH(4),4K,6HBH(5),5X, 15HDH(6),5X,5HBH(7),5X,5HRH(2),5X,5HRH(3),5X,5HBH(4),4K,6HBH(1), pFINT 2023 2024 F0FMAI (4x,5HRH(1),4X,5HBH(1)) pFINT 2024 2024 F0FMAI (4x,6HRH(1),4X,6HBH(1)) pFINT 2024 2024 F0FMAI (4x,6HRH(1),4X,6HBH(1)) pFINT 2025 2025 F0FMAI (4x,6HRH(1),4X,6HBH(1)) pFINT 2025 2025 F0FMAI (4x,6HRWC(1),4X,6HBWC(2),4X,6HBWC(3),4X,6HBWC(4),4X,6HBWC(5) 1),4X,6HHWC(6),4X,6HBWC(7),4X,6HBWC(2),4X,6HBWC(3),4X,6HBWC(4),4X,6HBWC(5) pFINT 2025 2025 F0FMAI (4x,6HRWC(1),4X,6HBWC(2),4X,6HBWC(3),4X,6HBWC(4),4X,6HBWC(5) pFINT 2025 2026 F0FMAI (4x,6HRWC(1),4X,6HBWC(2),4X,6HBWC(3),4X,6HBWC(4),4X,6HBWC(5) 1),4X,6HWC(6),4X,6HBWC(7),4X,6HBWC(2),4X,6HBWC(1),4X,6HBWC(5),4WC(5),4WC(6),8WC(7),4WC(5),4WC(6),8WC(7),4WC(5),4WC(1),3X,7HBWC(12),3X, 17HBYC(11),3X,7HBWC(12),4WC(10),3X,7HBWC(11),4X,7HBWC(12),3X, 17HBYC(12),3X,7HBWC(10),4X,7HBWC(10),4X,7HBWC(11),4X,7HBWC(12),3X, 17HBYC(12),3X,7HBWC(12),4WC(10),4X,7HBWC(11),4X,7HBWC(12),3X, 17HBYC(12),3X,7HBWC(12),4WC(10),4X,7HBWC(11),4X,7HBWC(12),3X, 17HBYC(12),3X,7HBWC(12),4WC(10),4X,7HBWC(12),3X,7HBWC(12),3X, 17HBYC(12),3X,7HBWC(12),4X,7HBWC(12),3X,7HBWC(12),3X, 17HBYC(12),3X,7HBWC(12),4X,7HBWC(12),3X,7HBWC(12),3X,7HBWC(12),3X, 17HBYC(12),3X,7HBWC(12),3X,7HBWC(12),3X,7HBWC(12),3X, 17HBYC(12),3X,7HBYC(12),3X,7HBWC(12),3X,7HBWC(12),3X,7HBWC(12),3X,7HBWC(12),3X, 17HBYC(12),3X,7HBYC(12),3X,7HBWC(12),3X,7HBWC(12),3X,7HBWC(12</pre>

ı,

1

	PHINT 9021.HAC(9).HWC(10).HWC(11).HWC(12).HWC(13).HWC(14).HWC(15)
	PRINT 2027
5051	FORMAT (5X,5H95(1),5X,5H85(2),5X,5H85(3),5X,5H85(4),5X,5H85(5),5X,
	(5HbS(6) + 5X + 5HBS(7) + 5X + 5HBS(R) :
	PHINT 9021, RS(1), HS(2), HS(3), RS(4), HS(5), BS(6), BS(7), BS(8)
	PHINT 2028
SUSH	FURMAT (5x,5HR5(9),4x,6HBS(10),4x,6HBS(11),4x,6HBS(12),4x,6HBS(13)
	$\frac{1}{10} + \frac{1}{10} $
2.120	PRIME COPY COMMET AND COPY
6064	FURMAT (4x, 5HHTA(1), 4x, 5HHTA(2), 4x, 5HHTA(3), 4x, 5HHTA(4), 4x, 5HHTA(5
	//*****//////////////////////////////
	$\frac{1}{2} \frac{1}{2} \frac{1}$
2030	
1	$\Delta(13) \bullet 3X.7 He TA(14) \cdot 3X.7 He TA(15))$
	PRINT 9021-HTA (9) - 9TA (10) - 8TA (11) - 8TA (13) - 8TA (13) - 8TA (14) - 8TA (15)
	PRINT 2053
2053	FORMAT (4X.6HNWT()) .4X.6HNWT(2) .4X.6HNWT(3) .6X.6HNWT(4) .4X.6HNWT(5
•	) 44,6HNWT(6).4X.6HNWT(7).4X.6HNWT(8))
	PRINT 902] NWT (1) NWT (2) NWT (3) NWT (4) NWT (5) NWT (6) NWT (7) NWT (8)
	PRINT 2054
2054	F()HMAT (4x,6HNWT(9),3X,7HNWT(10),3X,7HNWT(11),3X,7HNWT(12),3X,7HNW
1	T(13)+3X+7HNWT(14)+3X+7HNWT(15))
	PRINT 9021. NWT (9) . NWT (10) . NWT (11) . NWT (12) . NWT (13) . NWT (14) . NWT (15)
	PRINT 203A
5028	FORMAT(4x,6HDAF(1),4X+6HDAF(2),4X+6HDAF(3),4X+6HDAF(4)+4X+6HDAF(5)
١	4X + 6HDAF(6) + 4X + 6HDAF(7) + 4X + 6HDAF(8)
	PHINT 9021, DAF (1), DAF (2), DAF (3), DAF (4), DAF (5), DAF (6), DAF (7), DAF (8)
2020	PHINT 2039
20.39	FURMATI4X, BHDAF (9), 3X, THUAF (10), 3X, THUAF (11), 3X, THUAF (12), 3X, THUAF
1	(137+3A+7HBAF(14)+3X+7HUAF(15)) UNLINE MORE DAR (DAR (DAR (DAR (DAR (DAR (DAR (DAR
	DDINT 2000 DDINT 2000
2055	ROHMAT (114 CONT INDUT)
2000	CHINT 2051
2051	FUHMAT (HK. 2HTT. 3X. 4HTMAX)
••••••	PRINT 9023.TI.TMAX
	PRINT 2052
2052	FURMAT (KX, HANT)
	PRINT 4021.HNI
	PRINT 2031
5031	FORMAT (256 BOLT MATERIAL PROPERTIES)
	PRINT 2032
5035	FORMAT (HX+2HRY+AX+2HBE+8X+2HBU+7X+3HBYC+7X+3HBYR)
	PRINT POPO, RY, RE, BU, RYC, BYR
	PPINT 2033
2033	FUHMAI = (6X, 3HBTE, 7X, 3HBCR, 6X, 4HBRH())
	MATHE JULY HIF ON CHONHOU
2055	
4 Y D D	PUTTAL MANICAM HING GEUMEIRT AND PRUPERTIEST
2054	ENERT EVINE ENERT (67.4000)
5433	
	PRINT 2057
2057	FOHMAT (7%, 2HRI) 8X, 2HRY, 7X, 3HRSU, 7X, 3HRCY, 8X, 2HRF, 8X, 2HRG, 7X, 3HRYC
	and the second

	1,7X,3HRYR)			
	PHINT 9020 PH. RY . NSU . RCY . RE . RG . RYC . RYR			-
	PRINT 2058			
2 11 42 54	M FURIANT (SX-AMPCYCANX-AMPCYP)			
	$\mathbf{D}\mathbf{P}$ (NT $\mathbf{Q}$ $\mathbf{D}$ $\mathbf{D}$ $\mathbf{D}$ $\mathbf{D}$ $\mathbf{D}$ $\mathbf{D}$			
2050				
×0.24		ta - t at appatricitation monthaire mo		
	PRINT MUZZARIFARCHARRHU			
	PRINT 2034			
2034	4 FORMAT (32H SYSTEM PRESSURE AND TEMPERATURE	E)		
	PRINT 2035		- ·	
2035	5 FURMAI (9x,1HP,9x,2HPP,8x,2HPP,7x,3HPIM,6x,	4HTMAX 96X .	SHIMIN)	
	PRINT 9020.P.PP.PH.PIM.TMAX.TMIN			
	PRINT 2050			
2050	0 FORMAI (6×,4HDT1C,6×,4HDT2C,6×,4HDT2C,6×,4H	UT]H,6X,4H	)T2H,6X,4	HDT3
1	1 (4 )			
	PHINT 4020, DIIC, DISC, DI3C, DI1H, DISH, DT3H			
	PRINT 2036			
2036	5 FORMAT (74, 3HF51, 7X, 3HF52, 7X, 3HF53)			
•	PRINT 9021.FS1.FS2.FS3			
9110	OCUNTINUE			
	KOUNTEO			
С	TUBE CALCULATIONS TURE WALL STUBE BENDING M	OMENT . TUBE	BENDING	STRESS
-	TW2=1.1*PP+TOD/(2.+TY+.8+PP)		· · · · · ·	•
	TW3=1.1*PR#TOD//2.#TU+.H*PR)			
	$TE(TER) = EC = 0.01 \ GO \ TO \ IA$			
	TW4=1.190TN0T00//2.0TER0+.900TM)			
14	(1) 1) 17 2 TWASTWZ			
17				
. /				
	1F10UV1=1#1119196			
ι				
~	- (50-11) J - テレールAY3デノアレス テレス エルムス			
	/ [WAMAAIF(]W2# W3# W4] 			
.5	5 W=3•[4[0*(10)**4*(10)*****/*)***4//(3/•**(0))			
	TEXTRM1 (CQ ) 001) D(4			
4				
	GU 10 6			
5	5 THM2=WP+6667+TY			
	THMJEWP.HATR			
	TUM4=W#.54TF			
	TBN5=MINIF(TBM2+TBM3+TBM4)			
	TBM6=0(+*(TOD+3+)**3			
	THMEMINIF (THM5.THM6)			
6	5 TUS=TUM/W			
	TMU=T01)-TW			
	PJ=3,1416			
	tF(TOD-3,0)26,26,27			
26	5 ALO=.5+TAS+PI+(TOD-TW)+TW+(.5+.5+(TOD/3.))			
	6C N 07			
27	7 RL0=,54TRS*PI*(TOD-TW)#TW			
33	3 HLI=MAXIF(HLO,SSLI*PI*TOD)			
	ANP=4.			
	TF (TOD-3.0)60.60.61		- · ·	- · ·
60	) T=1			
		· · · · · · · · · · · · · · · · · · ·		

	61	1=2
	62	xT= 1, 4Tw
	•	LT=2,41w
		TF () = EDDT1144.43.43
	63	ション・マン・ションション (アウキマロスタウム) 一方子 モーン 人名かん
	0.4	
	<b>ت ח</b>	
	/00	
	701	[F(1)09-4.)703,703,704
	103	
	794	TE (TOU-4.) 705.705.706
	705	FHE41*1414
	105	IF (100-12,)707,707,708
	707	FH=0,25
		50 TO 702
	708	FR=),3125
	70.	CONTROP
		SLEG=0.1+1.0124TOU
		SIN= (T-)N+2, +T+)+2, +SLEG+0,15
	14	ΤΑΞΤΛΟΤΨ4Τμ
	15	50)=51U+2,+TA+2,+SLEG
		03=ST0+2.4TA
		SLEGT]=RSLT# (SOD/D3)/SCYR
		SLEGT2=0.04+(0.0014H840.54100)/12.0
		SLEGT=MAX1F(SEFGT1.SLEGT2)
-	-	YAKSL #RSL 1 ( (SOD/D3) - 5CYR4 (SOD+SOD-D3+D3) + SLEGT/ (SOD+SOD)
		SEL=2+#HSLI/(SCYR4ALOG(D3/SID))
		SX=SFL/2.1
		I T=5FL/TA
	18	TH (4, 0-LT) 19.19.29
	19	TA = TA + 0.00G
		60 TO 15
	25	12.22.32.32.32.28
	32	$\varsigma(0) = \varsigma(0) I$
	• •	STU=STUT
		SEL=SEL I
		TATTAL
		SLEGIESLEGTI
		SX=SFL/2
С		LOAD CALCHLATIONS
•	28	MSL=45L [49] 4500
С		PRELTHINARY HOLT CALCULATION
•		
	108	HUHI=01-T4
		$\Delta = \Delta \Delta = \Delta \Delta = (1 + 0) + 1 + 1 + 2 + 0 + (1 + 9 + 6) + 9 + 5)$
		$n = (F_{0} - F_{0} + F_{0}) + T_{0} + (A + F_{0})$
		pIu=T(D+P,*(Hupt=0))
		Δ]=((5+4H))/(CT#FCY)+(RTD##2))##-5
		x2=Snu+2_0//SlcGT#6.0##SlT#FS1/FV1##_51
		A SOU+2.0/DSET0FSI/FSU)
		The character of the full fill fills in the second se

```
45=500+2.*(0.050+1004.0075)
      44= 44×1F(41+42+43+45+A)
      TF (.001-AT)110.110,113
  110 4241
      60 10 107
  113 4=44
  107 JF (I-IMAX) 115.1009.1009
 1000 PHINE 1010
 1419 FORMAT (3X, 36HROLT REQUIRED NOT AVAILABLE IN TABLE)
      60 TO 1999
  116 RCU)=A+HH(T)
      AC05=100+5 * MUNUT+9AC(1)
      BCD3=MAX1F(HCD1.BCD2)
      TF(+001-HCDT)115+115+114
  115 HOUSACOT
      RN=HNI
      1=11
      60 10 117
  114 RCU=HCU3
      RU=HCUMPTZRHC(T)
      NH=HN/2.
      N=24NB
      AN=N
  117 SMD=R+1A
      c=(2.**++10)/3.
                                                     · · · ·
      RCD=RCD+2.0RH(1)
      H2=0.54(C-(A+H)/2.)
      H3=0.54((A+B)/2.-5MD)
      H4=((4+H)/4,)-(1,/3,)*(SOD*#2+SOD#8+8##2)/(SOD*8)
      H5=0.5#(RCD-C)
      но=((A-8)/4.)-(Tk+HU9I)/2.
                                           الا المرجم ومنتقد والمراجع المراجع المراجع
                                                                    - - ---
      FLANGE SIZE CALCULATION
С
      H=2. 14 (1461) 44.5
      K=A/4
      T=((K*K)*(],0+R.55246*ALOG10(K))-1.0)/((].04720+1.54480*K*K)*(K-1.
     i))
      IJ=((K*K)*(1,0+8,55246*ALOG10(K))-1,)/(1,36136*(K**2-1,)*(K-1,))
      Y=(1./(K-1.))+(.66845+5.7)7+((K#K)*ALOGIn(K)/(K##2-1.)))
      7= (K#K+1.)/(K#K=].)
      F=. 404-.0314(G1/60-1.)-.1605+(G1/60-1.)++.5
      v=+55++15+(61/60-1+)++558+(61/60-1+)+++5
      HO=((H+G3) 4G0) 44.5
      LCE=F/HO
      I CU≈(U/V) #H0#S0##2
  202 1= (XT#LCE+1.)/T+ (XT##3)/LCD
      cf=0.5*(61+60)
      HHU= (H+GE) ++.5
      C1=(3.**.5)*(2.-FHU)/(4.*FE*(1.-FMU**2))
      02=(3=/(1==FM:14#2))##.5
                                         03=(12++(1++F414+2))+++25
      C4=((12.*(1.-F4U**2))**.25)/(2.*(1.-F4U**2))
      RETA=(3.*(1.-FM/J##2)/((((100/2.)+GE-GO)##2)*(GE)##2))##.25
      H()님=(님부(J트) ㅎㅎ. 특
      GAMMA=(GF#(FMU+Z)#(1.+C3#XT/HOP))/(1.+(C4#B#GE##3#(FMU+Z)/(HOP#XT#
     1+3)+(2+C3+XT/HOP)))
      nT=FF#GE##3/(12.#(1.-FMU##2))
```

```
1)F=FF44T443/(12.4(1.-F41)442))
      *F=(((((1,+F 41))/(1,-FMU))+K++2+1,)+H)/((K++2-1,)+(1++FMU)+2,+OF)
      A1=(い1/4。) & (10)0442-13442)
      7H=P14((T0)-2.+G0+2.+G1)+44-(T0D-2.+TW)++4)/(32.+(T0D-2.+G0+2.+G1)
     11
      AH=((((F())-2,*G()+2,*G1)**2-(T())=2,*G())**2)*PI/4.)
С
      NEFLECTION RATES
      05A=-(5X)/(SFF+PT+(B+TA)+TA)
      NSH=-(H+TA)/(P. +P1+SEL+TA+SEE)
  203 PHA=((SX+XT)/(AF4PT4AS(I)##2/4.))+(RT+A<(I)/2.)/(AE#ATA(I))
      DBA=03AZAN
      1)FR=(-914V)/([0H00600424FE)
      DFA==DFH+(H2+H3)++2=(1./(2.+P1+FG+XT))+aLOG(C/SMD)
     1-(XT)/((P1/4.)+(A++2-B++2)+FF)
      nHH#A./(PT#RF#(RT##3)#ALOG(ROD/RTD))
      DRA==URR#(H5##2)=(1./(2.#PI#RG#RT))#ALOG(BCD/C)
     1-(HT)/((PT/4.)*(HCD**2-RID**2)*RE)
      RING CALCULATIONS
С
      KR=ROU/RTD
      YR=(1+/(KR+1+))+(+66845+5+717+((KR+KR)+ALOG10(KR)/(KR+#2-1+)))
      RATIO=A/TOD
      TF (RATIO .LE. 1.1)711.712
  711 KTF=0.1794((100/FR) #4.5)+.914
      GU TO 11A
  712 TF (HATIO +LE+ 1+2)713+714
  713 KTF=0.202+((TOD/FR)++.5)+.86
      60 TO /18
  714 TF (RATIO .LE. 1.5) 715, 716
  60 IC 714
  716 /TF=0+242+((TOD/FR)4++5)+.69
  119 CONTINUE
С
      LOAD CALCULATIONS CONDITION 1 AND 2
      P1=0.
      9239
      THM1=0.
      THM2=TOM
      MP2=((L+H0+G0++2+FE)/(+91+V))+(C1+B++2+GAMMA+P2/(XT+(XT++2+C2+GE+G
     I AMMA ) ) )
      FA1=1.
      FA2=02+(01/4.)+(4++2)
      FPL1=0.
      FPL2=P2*(PI/4,)*(500**2-8**2)
      ASLI=HSLT+PT+SOD
  111 SL1=HL1
      M01=-H5L1#X1/2.+HL1#H2+SL1#H3
      MC2=41N1F((L+31++2+(B+TW)+FE+FCR),(M01+FE+FCR/FYR))
      4H1=HL14H5
C STRESS CALCULATION CONDITION 1
  310 SH=((UL1+H2+S) 1+H3)/(L+G1++2+(B+G0)))
  31] $RS=((HSL]*XT/2.)/(L*G]**2*(R+GO)))
      SR=4014(].3334XT4LCE+1.)/(L4XT4424(B+G0))
      ST=4014 (Y/(XT++2+8))-245R
      SH=0.
      SPR=n.
```

```
B-42
```

SPA=1.

```
SCLA=0.
     SCEH=0.
     FSS=AL1/(TON+PT*XT)
     SSC=SL1/(((A+2.+TA)++2-B++2)+P1/4.)
     HTS=HL1/(HN#HTA(T))
     SFH=4.4HL1/(PT+(A++2-RID++2))
     SIK= (MK1444) / (QT4424RID)
     RSS=BL1/(A4PT4RT)
     IE (SEH-ECABAS") 151+151+155
 155 IF (+001-01) 151-151-150
 120 A=A+.015
     60 10 116
 121 TF (HTS-HYP/FS2) 314+314+903
 903 77 (0-11) 314.313.313
 313 7=1+1
     60 TO 107
 314 TF (SQ-FYQ/FS1) 315+315+302
 315 TF (ST-FY0/FS1) 316, 316, 302
 316 TF (FSS-FSU/FS1) 317,317,302
 317 [F (SSC-SCYR/FS1) 318+318+619
 619 PRINT 520
 620 FORMAT (3X, 19H SSC EXCEEDS LIMIT)
 314 TF (STH-RYR/FS1) 319+319+303
 319 TF (RSS-RSU/FS1) 330+330+303
 402 TF (HUBIT .LT. 0.001) 402.302
AND TE (GIUTAT .LT. 0.001)804.803
A03 G10TV=61/TW
     TE (G10TW .GT. G10TWI) 302.804
 H04 G1=G1+0.005
     xT=XT=0.030
     GO TO 108
 102 XT=XT++015
     TF (XT-2. +TON) 202-1003+1003
LODS PRIME 1004
1004 FURMAT (3x, 3) HELANGE THICKNESS EXCEEDS LIMIT )
     GO TO 999
 303 HT=RT+.015
     TF (RT-2.+TON) 203.1005.1005
1005 PRINT 1016
1006 FORMAT (31, 29HOING THICKNESS FACEEDS 2. TOD)
     60 TO 999
850 XT=XT+.015
     KOUNT=KOHNT+1
     TF (KOUNT . IF. 5) 860,202
360 TF (G] +GT. 1.754TW)351+202
 851 61=01-0.005
     60 10 109
 130 XA1=-SH+SH+SP4+SPA+SRS
     XAZ=- SCEA+SR
     X A 3=51
     NA1=-SH-SH+SPR+SPA+SRS
     NA2=-50F 1+54
     NA3=ST
     FXA=, / 7110((XA1-XA2)++2+(XA2-XA3)++2+(XA3-XA1)++2)++2+
     FNA= , 70711+ ( (NA1-NA2) ++2+ (NA2-NA3) ++2+ (NA3-NA] ) ++2) ++ 5
     FA =AMOXI (ARS(FXA) . ABS(ENA))
```

```
18 (HA-EYO/851) 352,352,302
  352 441=+50+54-507+504-585
      メンルキーノクトン・イン
      XH HECT
      NH1=+5H=54-5PR+5PA-585
      NHZ=-SCER+SR
      NHJEST
      FXH=。{(!?!l+((XR)=XB2)++2+(XB2=XB3)++2+(XB3=XB1)++2)+4+2)+4+5
      +Nd=,/v711v((441-482)##2+(NB2-NB3)##2+(NB3-NB1)##2)##.5
      FH=AMAX1 (ARS (FXR) +ARS (ENR))
      TE (EH-EYD/ES1) 362, 362, 302
      LOAD AND MOMENT CALCULATION CONDITION 2
C
  362 TI ZEFAR+TASHAT
      421=+4L2=112
      P22=-FP1.2*F4+(P2*PI*H*SX)*(XT/2.)-MP2-MC2-TL2*H6
      123=-(P2+14((4+1)/2.))/(4.+((A-B)/2.)+FF)+(P2+B+SMD)/(4.+TA+SEE)
      P24=(BCRP(CX+XT+RT))/(DBA=DRA)
      H2S=HFA/((H2+H3)+(DHA-HRA-HSA))
      H20= (UHA-DRA) + (R24-R21) / (DHA-DRA-DSA)
      H27=R264 (H2+H3)+R21 #H2+ (R23*XT)/(05R#2.)+R22
      P28=(1.-(R25+(H2+H3))-(DFR+XT++2)/(DSR+4.))
      MO2=N01-H27/H2A
      SL2=SL1-P27#H25/R28-R26
      PSL2=R5L1+(R27+0FR+XT)/(R28+2.+0SR)-R23/NSR
      HLZ=HL1-R21-R26-R27#R25/R28
      42=422+44
      1F (SL2-MSL) 331, 332, 332
  11 HL]=1.0]*HL1
      40 10 111
  332 H7]P=-+PL7-TL7
     P22P==FPI, 2+H4+ (P2+P1+B*SX) + (XT/2+) =MP2=TL2+H6
     R23P==(P2+44((A+R)/2.))/(4.*((A-R)/2.)*FE)+(P2*B*SMD)/(4.*TA*SEE)
     4244=10.
     R25P=UFA/((H2+H3)*(DRA=DRA=DSA))
     R26P=10HA-PRA) + (R24P-R21P) / (DRA-DRA-DSA)
     P27P=H26P+(H2+H3)+R21P+H2+(P23P+XT)/(DSP+2+)+R22P
     P20P=(1.+(P25P+(H2+H3))+()FP+X***2)/(D5P+4.))
     MO2P=401-P27P/P2AP
      SL28=5L1-0278#025878288-8268
     RSL2P=+SL1+(427P+UFR+XT)/(R2AP+2.+USR)-p23P/USR
     HL2P=HL1-R21P-R26P-R27P#R25P/R28P
     VRZP#HL2P#H5
     RETUS PLP-FARAT
     R224=+FP1,2+H4+(P2+P1+H+SX)+(X1/2,)=MP2+H6+(FA2+TBS+A1)
     R23M==(P24R4((A+R)/2,))/(4,+((A+R)/2,)4FF)+(P24B4SMD)/(4,0TA4SEE)
     Q24:1=0.
     R25M=DFA/((H2+H3)+(DRA-DRA-DSA))
     R261= (UHA-DRA) + (R24M-R21M) / (DBA-DRA-DSA)
     R27M=H26M+(H2+H3)+R21M+H2-(R23M+XI)/(DSP+2+)+R22M
     W2HH=(1.-(R25M+(H2+H3))-(DFR*XT**2)/(D5P+4.))
     14(121=141) - 977 / 928M
     SE2M=SE1-827M4025M/828M-826M
     RSL2M=RS(1+(R27M#DFR#XT)/(R28M#2.#DSR)-R23M/DSR
     4L24=HL1-P21M-026M-827M4825M/828M
     MP201=HL2MHU5
     STRESS CALCULATIONS CONDITION 2
```

С

```
SH2P=(HL2P+H2+FPL2+H4+TL2+H6+SL2P+H3)/(|+G1++2+(B+G0))
           SH2M=(HL2M+H2+FPL2+H4+(F42-TRS+AT)+H6+SL2M+H3)/(L+G1++2+(B+G0))
           SRS2P=(HSL2P#XT/2+)/(L#G1##2#(B+G0))
           SR52M=(R5L2M#XT/2.)/(L#G1##2#(8+G0))
           SPR2=MP2/(L+G1++2+(B+G0))
           SPS=LHW/1H
           SPA2=+ 42/4H
                                                                                         • •
                                                                                                         -----
                                                                                                                                SCEH2=0
           SCEA2=P2
           SR2P=M02P#(1.333#XT#LCE+1.)/(L#XT##2*(B+GO))
           SR214=MU2M#(1+333#XT#LCE+1+)/(L#XT##2#(0+G0))
           ST2P=M02P+(Y/(XT++2+B))-Z+SP2P
           ST2M=M02M# (Y/(XT##2#A))-Z#SR2M
                                                                                                    · · · · · · · · · · · · · ·
           ESSPEBL 2PZ (TOD+PT+XT)
           SSC2=SL2M/(((R+2.+TA)++2-R++2)+P1/4.)
           HTS2P=HL2P/(BTA(T) #BN)
           SF820=4.#8L20/(01#(A##2-RID##2))
           SF854=4.48L20/(01+(A++2-RI0++2))
           s[kSb=(wbSbakd)\(kt#eS#BID)
                                                                                                          SIKSH= (WHSMAAB) / (KIAASABID)
           R2255=HF565(VA61481)
           SEBREAMAX1 (SEHOP . SEBRM)
           TF (SFH2-FCY/2.1343+343+123
  123 TF(+001-AT) 343-343-120
   343 IF (HT52P-HY/FS2) 344, 344, 913
                                                                                                   سسي متعادية المستنبة النبية المتنابية بالمتناب المتاري المس
  913 TF (0=[1) 344+313+313
  344 TF (SR2P-FY/FS1) 345, 345, 302
  345 TF (ST2P-FY/FS1) 346, 346, 302
  346 [F (FS52-FSII/FS1) 347, 347, 302
  347 TE (SSC2-SCY2EST) 348+348+621
  621 PRINT 622
                                                                                                         -----
  622 FORMAT (3X. 14H SSC2 EXCEEDS LIMIT)
  348 TF (STH2P-RY/FS1) 349+349+303
  344 TF (STH2M-RY/FS1) 350, 350, 303
   350 TF (RS52-PSU/ES1) 351+351+303
  351 KA1=-SH2P+SH2+SPR2+SPA2+SPS2P
           XAZ=-SCEA2+SR20
                                                                                                     - and a construction of the relation of the second se
           XA3=ST2P
           NA1=-SH2 M-SH2+SPR2+SPA2+SPS2M
          NAZ=-SCEA2+SR2M
          NAJ=SI24
          FX42=• 10711+((XA1-XA2)++2+(XA2-XA3)++2+(XA3-XA1)++2)++.5
          FNA2=+ 10711+ ((NA1-NA2) +*2+ (NA2-NA3) **2+ (NA3-NA1) **2) **.5
          XA1=XA1~SPA2
          NAL-VAL-SPA2
          TF (ARS(XA1) - ARS(NA1)) 602.602.601
  402 CMAX2A=NA1
          CMINPA=XAL
          60 TO 603
  601 CMAX24=XA1
          CML J2A=NA1
 HOR FARAMAX (ARS(EXAP), ABS(ENA2))
          1F (EA-FY/FS1) 1352+1352+302
1352 CALT2H= (CMAX24-CMIN2A)/2.
          CMNSV=(CHAX30+CWINSV)15"
          FALTPA=CALTPAOFSFI
```

```
SALTON=FALTON
     SNNSV=CWNSV
     SCHKPA=ARS (SMNDA+SAL TZA)
     TE (SCHK2A .GT. FU) 402.805
 BOS TE (SCHEPA-EY) ANG. 806.807
 HOT TE (ABS(SALTZA) - FY) ROB, BOB, BOB
 HOR SMN2D=FY-SALT2A
     GO IN NON
 H09 SMN24=0.
 406 SMAKPA=SMNPA+SALTZA
     SMI V24=SMN24-SALT24
     QC = SMIN2A/SHAY2A
     м=0
1350 M=M+1
     TF (RC1-R(M)) 1351.7000.1350
1351 FFC24=((P(M)-PC1)/(P(M)-P(M-1)))*(FF(M)-FF(M-1))*FF(M-1)
     GO TO 7001
7000 FFC24=FF(M)
7001 CONTINUE
     TE (AHS (SHAX2A) - FEC2A) 1363.1363.402
1363 XH1=SH2P+SH2=SPR2+SPA2=SR52P
     XH2=-SCFH2+5P2P
     XH3=512P
     NH1=5H2M-5R2-5PR2+5P42-5R52M
     NH2=-SCEH2+SP2M
     NH3=SIZM
     FXH2=./0/11+((XR1-XH2)++2+(XH2-XH3)++2+(XB3-XB1)++2)++.5
     ENU2=+70711+((NR1-NR2)++2+(NR2-NH3)++2+(NB3-NH1)++2)++.5
     EH=AMAX1 (AHS (EXR2) + AHS (ENR2))
     XB1=XB1-5PA2
     NH1=NH1-SPAZ
     TF (ER-FY/FS1)720+720+302
 120 [F (ARS(XR1)-ARS(NU1))605+605+604
 605 CMAX2d=NP1
     CWINSH=XH1
     60 TO 1362
 604 CMAR2H=XR1
     CWIN5R=NH]
1365 CALIDH= (CMVX58-CMIN58) 15.
     CWNSB= (CHAX2H+CMIN2H) /2.
     FALT2H=CALT2H#FSFI
     SALTOHEFALTORAKTE
     SMNSHECMNSHAKTE
     SCHK2H#AHS (SMN2R+SALT2H)
     IF (SCHK29.GT. FU) 402,810
 HID TF (SCHK28-FY) A11+811+812
 A12 TF (A45 (SALT29) - FY) A13,813,814
 A13 SMN2R=FY-SALT29
     GU TO HI]
 8]4 SMN2440.
 HII SMAX2H=SMV2R+SALT2B
     SMIN2H=SMN2H-SALT2H
     RC2=SHIN2A/SHAX2R
     M = 0
1360 4=4+1
     TE (RC2-R(M)) 1361+7002+1360
```

	1361	FFC2B=((R(M)-RC2)/(R(M)-R(M-)))*(FF(M)-FF(M-1))+FF(M-1))
		GO TO 7003
	7002	FFC2B=FF(M)
	7003	
~		TF (AHS (SMAX2H) - FF (2H) 380, 380, 402
C	30.	LOAD CALCULATIONS CONDITION 3
	3140	*DM3~0
		27 A 1970 20 (D 1 7 A - 1 A - 1 A - 2 A -
		EPE 3=P3#(P1/4)*(5***)
		TI BECAB
		MP3=((1+H0+G0++2+FF)/(-91+V))+(C1+B++2+CAMMA+P3/(XT+/XT+2+C2+GE+G
	1	
		031==FPL 3-7L 3
		H32=-FPL 3+H4+ (P3+P1+9+5X) + (XT/2+) = MP3=TL 3+H6
		R33=-(P3*8*((A+B)/2.))/(4.*((A-B)/2.)*FF)+(P3*8*SMD)/(4.*TA*SEE)
		R34=n.
		H35=DFA/((H2+H3)#(DHA-DRA-DSA))
		P36=(UHA-DRA)+(R34-R31)/(DBA-DRA-DSA)
		R37=R36# (H2+H3) +R31#H2= (R33#XT) / (DSR#2+) +R32
		$R36 = (1 - (R35*(H2+H3)) = (DFR*XT**2)/(DSR*4_))$
		MOJ#MUL-R37/R3H
		R3F3#021+(R3/90FK#41//(R35#20*03R/#R33/03R
		HD3-01 390K HD3-011-431-432-432443344334430
		TE (SI 3-MSI 1331, 300, 300
c		STRESS CALCHEATTONS CONDITION 3
C	396	SH3=(N) 34H2+SI 34H3+FPL 34H4+TL 34H6)/(L #G14#2#(B+G0))
	- • 0	SR53=(HSI 3+XT/2.)/(L+G1++2+(B+G0))
		SPR3=MP3/(L*G1**2*(B+G0))
		5R3=0.
		SPA3=FA3/AH
		5CEB3=0.
		SCEA3=P3
		SR3=M03#(1.333#XT#LCE+1.)/(L#XT##2#(B+Gn))
		\$13=MU3+(Y/(XT++2+R))=2+5R3
		F333=91, 17 (T91)#P1#X1) cs(1)=51 277770, a atalaha2-04000A0174 N
		~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~~
		ATS3=6.981.3/(ATA(T)#8N)
		AIS3=8L3/(ATA(T)*8N) SFH3=4.*HL3/(PT*(A**2-RID**2)) STK3=(MR3*YR)/(PT*(A**2*RID**2))
		ATS3=BL3/(ATA(T)+BN) SFH3=4+HL3/(PT+(A++2-RID++2)) STR3=(MR3+YR)/(RT++2+RID) RS53=BL3/(A+PT+RT)
		ATS3=BL3/(HTA(T)+BN) SFH3=4+HL3/(PT+(A++2-RID++2)) STH3=(MR3+YR)/(RT++2+RID) RS53=BL3/(A+PI+RT) 1F(SFH3-FCY/2,)2314+2314+124
	124	ATS3=HL3/(HTA(T)+HN) SFH3=4.+HL3/(PT+(A++2-RID++2)) STH3=(MR3+YR)/(RT++2+RID) RSS3=HL3/(A+PI+RT) TF(SFH3=FCY/2.)2314+2314+124 TF(.001-AI)2314+2314+120
	124 2314	ATS3=HL3/(HTA(T)+HN) SFH3=4.+HL3/(PT+(A++2-RID++2)) STK3=(MR3+YR)/(RT++2+RID) RS53=HL3/(A+PI+RT) TF(SFH3=FCY/2.)2314.2314.124 TF(.001=AI)2314.2314.120 TF(HT53=HY/FS2)2344.2344.923
	124 2314 923	ATS3=HL3/(HTA(T)+HN) SFH3=4.+HL3/(PT+(A++2-RID++2)) STK3=(MR3+YR)/(RT++2+RID) RSS3=HL3/(A+P[+RT) TF(SFH3-FCY/2.)2314+2314,124 TF(.001-AT)2314+2314,120 TF(HTS3-HY/FS2)2344+2344,923 TF(0-II) 2344+313+313
	124 2314 923 2344	ATS3=HL3/(HTA(T)+HN) SFH3=4.+HL3/(PT+(A++2-RID++2)) STR3=(MR3+YR)/(RT++2+RID) RSS3=HL3/(A+PI+RT) TF(SFH3-FCY/2.)2314+2314,124 TF(.001-AI)2314+2314,120 TF(HTS3-HY/FS2)2344+2344,923 TF(0-II) 2344,313,313 TF(SP3-FY/FS1)2315+2315,302
	124 2314 923 2344 2315	ATS3=HL3/(HTA(T)*HN) SFH3=4.*HL3/(PT*(A**2-RID**?)) STR3=(MR3*YR)/(RT**2*RID) RSS3=HL3/(A*PI*RT) IF (SFH3=FCY/2.)2314.2314.124 IF (.001-AI)2314.2314.120 IF (HTS3=HY/FS2)2344.2344.923 IF (0-II) 2344.313.313 IF (SP3=FY/FS1)2315.2315.302 IF (ST3=FY/FS1)2316.2316.302
	124 2314 923 2344 2315 2316	ATS3=HL3/(HTA(T)+HN) SFH3=4.+HL3/(PT+(A++2-RID++2)) STK3=(MR3+YR)/(RT++2+RID) RSS3=HL3/(A+PI+RT) TF(SFH3=FCY/2.)2314.2314.124 TF(.001-AI)2314.2314.120 TF(HTS3-HY/FS2)2344.2344.923 TF(0-II) 2344.313.313 TF(SP3=FY/FS1)2315.2315.302 TF(ST3=FY/FS1)2315.2315.302 TF(FS3=FSH/FS1)2317.2317.302
	124 2314 923 2344 2315 2316 2317	ATS3=HL3/(HTA(T)+HN) SFH3=4.+HL3/(PT+(A++2-RID++2)) STK3=(MR3+YR)/(RT++2+RID) RSS3=HL3/(A+PI+RT) TF(SFH3-FCY/2.)2314.2314.124 TF(.001-AI)2314.2314.120 TF(HTS3-HY/FS2)2344.2344.923 TF(0-II) 2344.313.313 TF(SD3-FY/FS1)2315.2315.302 TF(ST3-FY/FS1)2315.2315.302 TF(SS3-FSH/FS1)2317.2317.302 TF(SS3-SCY/FS1)230.2330.623
	124 2314 923 2344 2315 2316 2317 623	ATS3=HL3/(HTA(T)+HN) SFH3=4.+HL3/(PT+(A++2-RID++2)) STK3=(MR3+YR)/(RT++2+RID) RS53=HL3/(A+P[+RT) TF(SFH3=FCY/2.)2314.2314.124 TF(.001-AI)2314.2314.120 TF(HT53=RY/FS2)2344.2344.923 TF(0-II) 2344.313.313 TF(SP3=FY/FS1)2315.2315.302 TF(ST3=FY/FS1)2315.2315.302 TF(SS3=FSH/FS1)2317.2317.302 TF(SSC3=SCY/FS1)230.2330.623 PRINT 624 FOPHAL(2Y_10+_C5C3_FXCEFDS_LIMIT)
	124 2314 2344 2315 2315 2317 623 623	ATS3=HL3/(HTA(T)*HN) SFH3=4.*HL3/(PT*(A**2-RID**?)) STK3=(MR3*YR)/(RT**2*RID) RSS3=HL3/(A*P[*RT) TF(SFH3=FCY/2.)2314.2314.124 TF(.001-AT)2314.2314.120 TF(HTS3=RY/FS2)2344.2344.923 TF(0-II) 2344.313.313 TF(SP3=FY/FS1)2315.2315.302 TF(ST3=FY/FS1)2315.2315.302 TF(SSC3=SCY/FS1)2316.2316.302 TF(SSC3=SCY/FS1)2317.2317.302 TF(SSC3=SCY/FS1)2310.2330.623 PRINT 624 FORMAT(3x,19H_SSC3 ExCEEDS (.IMIT) TF(STH3=PY/FS1)231.2331.203
	124 2314 2344 2315 2316 2317 623 623 623 2331	ATS3=BL3/(HTA(T)*BN) SFH3=4.*HL3/(PT*(A**2-RID**2)) STR3=(MR3*YR)/(RT**2*RID) RSS3=HL3/(A*PI*RT) TF(SFH3-FCY/2.)2314.2314.124 TF(.001-AI)2314.2314.120 TF(HTS3-HY/FS2)2344.2344.923 TF(0-II) 2344.313.313 TF(SP3-FY/FS1)2315.2315.302 TF(ST3-FY/FS1)2315.2315.302 TF(SS3-FSH/FS1)2317.2317.302 TF(SSC3-SCY/FS1)2317.2317.302 TF(STR3-HY/FS1)231.2330.2330.623 PRINT 624 FORMAT(3X.19H SSC3 EXCEEDS (IMIT) TF(STR3-HY/FS1)231.2331.303 TF(SS3-SS1/FS1)231.2331.303

	A M ジャット 2 5 0 1 m - 5 0 2 - 5 0 m - 5 0 m - 5 4 5 4 5 4 5 4 5 0 5 3
	NM J= 17.7 FY NJ= - 7.6 7 + H / J J + L - J A - NAA-, / J A Y A - NAA-, / V A Y A - NAA- A - F
	F X A 3 = + ( 1 / 1 / W ( X A 1 = X A / / W C + X A / X A A 3 / W C + (X A 2 = X A / / W C + X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X A 2 - X
	ENAST (U/1) ((NA) -NA2) **2 (NA2-NA3) **2 (NA3-NA1)**2.1**2
	= A - A - A A + (A - A - A - A - A - A - A - A - A - A
	10 10 ATE 1/6 51 12 36 31 3921 192
4 36 3	XH1=SH3+SH3=SPR3+SPR3=SRS3
	XM2=+3L143+5H3
	NH1=2H3+2H3=2H23=2H23
	FXB3=+70711+((XR1-XB2) ==2+(XB2-XB3) ==2+(XB3-XB1) ==2) ==5
	ENH3=+/0711+((NH1=NH2)++2+(NB2=NB3)++2+(NH3=NH1)++21+++5
	FH=AMAAAAAAAABCCCCACABCCCCCCCCCC
_	TF (F9-FY/F51)2373,302,302
С	LOAD CALCULATIONS CONDITION 4
2373	p4=Pq
	FA4=P4#(NT/4.)#(H##2)
	FPL4=P4*(P1/4.)*(S0)**2-R**2)
	τί 43 β Αφ
_	BL4=FPL4+FA4
C	STRESS CALCULATION FOR CONDITION 4
	9T54=HL4/(HTA(T)*BN)
	TF (HTS4-H1/FS2) 4344+4344+933
<u>- 933</u>	TF(0-11)4344,313,313
С	LOAD CALCULATION CONDITION 5
4 3 4 4	PS=PIM
	FA5=P5*(PI/4.)*(R**2)
	FPL5=P5*(PI/4。)*(500**2-8**2)
	MP5=((L#H0#G0##2#FE)/(+91#V))*(C1#B##2#GAMMA#P5/(X1#(X1##2#C2#GE#G.
1	
	857==FV_59#44+(859P[9893]*(X1/2+)=MP5m1[59H6
	R53=+(P5#94((A+B)/2+))/(4+#((A+B)/2+)#FE)+(P5#6#5MP1/(4+#EA*565)
	$P_{3}=0rA/((H_2+H_3)) = (D_1A+U_2A_1)$
	H 30 = (1/3A - 1/2A) # (H 34 - K 3 [ / / (U 55 - U KA - U 3A)) 
	Q3N=(1++(N5)*(Q2+N3))=(UFR*A(**C)/(U34*4+)) N(5+N0)=(F=205+
	うしつ=うし 1~2~う / 20つう/200/200/200/200/200/200/200/200/200/20
	R2C7=K3L1+(K5/RUKWA)/(K58W***V2K/-K33/U3K
	MRDAHLD*HD *E/CI K-NCI 1001 E000.5000
	【F いつしつ = m つし) 551 6 つ 370 6 つ 370 
246	SHOF (RLOWHZ+SLOWHS+FMLOWHS+ILOWHS+ILOWHS/ILWUIWWZ)/
	▶ M 3 7 至 1 m 5 1 / 9 # 1 / 7 # 1 / 1 L * 0 L * 0 K * 2 M * 10 V 1 / 1 / 2 M * 10 × 20 V 1 / 2 M * 10 × 20 V 1 / 2 M * 10 × 20 V 1 / 2 M * 10 × 20 V 1 / 2 M * 10 × 20 V 1 / 2 M * 10 × 20 V 1 / 2 M * 10 × 20 V 1 / 2 M * 10 × 20 V 1 / 2 M * 10 × 20 V 1 / 2 M * 10 × 20 V 1 / 2 M * 10 × 20 V 1 / 2 M * 10 × 20 V 1 / 2 M * 10 × 20 V 1 / 2 M * 10 × 20 V 1 / 2 M * 10 × 20 V 1 / 2 M * 10 × 20 V 1 / 2 M * 10 × 20 V 1 / 2 M * 10 × 20 V 1 / 2 M * 10 × 20 V 1 / 2 M * 10 × 20 V 1 / 2 M * 10 × 20 V 1 / 2 M * 10 × 20 M * 10 × 20 V 1 / 2 M * 10 × 20 V 1 / 2 M * 10 × 20 V 1 / 2 M * 10 × 20 V 1 / 2 M * 10 × 20 V 1 / 2 M * 10 × 20 V 1 / 2 M * 10 × 20 V 1 / 2 M * 10 × 20 V 1 / 2 M * 10 × 20 V 1 / 2 M * 10 × 20 V 1 / 2 M * 10 × 20 V 1 / 2 M * 10 × 20 V 1 / 2 M * 10 × 20 V 1 / 2 M * 10 × 20 V 1 / 2 M * 10 × 20 V 1 / 2 M * 10 × 20 V 1 / 2 M * 10 × 20 V 1 / 2 M * 10 × 20 V 1 / 2 M * 10 × 20 V 1 / 2 M * 10 × 20 V 1 / 2 M * 10 × 20 V 1 / 2 M * 10 × 20 V 1 / 2 M * 10 × 20 V 1 / 2 M * 10 × 20 V 1 / 2 M * 10 × 20 V * 10 × 20

```
SCE45=P5
      585=MU54(1.333#XT#LOF+1.)/(I#XT##2+(8+64))
      ST5=005+(Y/(XT++2+A))-Z+SR5
     FSS5=HL5/(TOD#PT#XT)
     SSC5=5L5/(((H+2.+TA)++2-B++2)+PI/4.)
     R[SS=HLS/(KTA(T)+UN)
      SFH5=4.#8L5/(PT#(A##2-RID##2))
      STH5= (MH5+YR) / (RT++2+RID)
     RSS5=HLS/(PT+A+RT)
      TF (SF85-FCYR/2.)5314+5314+125
 125 TF (+001-AT) 5314+5314+120
5314 TF (HTSS-HYR/FS2)5344.5344.943
 943 JF (0-II) 5344+313+313
5344 TF (SP5-FYR/FS1)5315+5315+302
5315 IF (STS-FYR/FS1) 5316+5316+302
5316 TF (FS55-FSU/FS115317,5317,302
5317 TF (5505-50VP/FS1) 5330, 5330, 625
 625 PRINT 626
 626 FURMAT (38.19H SSC5 EXCEEDS LIMIT)
5330 TF (STH5-RYR/FS1) 5331+5331+303
5331 IF (R555-R5U/F51) 5332, 5332, 303
5332 SH1=5H
     SHS1=585
     SPR1=0.
     SH1=0.
     SPA1=".
     SCEH1=0.
     SR1=SH
     ST1=ST
     SCEA1=0.
     XA1=-SH5+S95+S085+SPA5+SR55
     YA2=-SCEAS+SH5
     XA3=4T5
     NA1=-SH1-SH1+SPR1+SPA1+SR51
     NAZ=-SCEA1+SP1
     NA3=ST1
     EXAS= . 70711+ ( (YA1-XA2) ++2+ (XA2-XA3) ++2+ (XA3-XA1) ++2) ++.5
     FNA5= . 70711 + ( (NA1-NA2) ++2+ (NA2-NA3) ++2+ (NA3-NA1) ++2) ++.5
     TF (ARS(XA1) - ARS(NA1)) 5602, 5602, 5601
SON2 CMAXSA=NAL
     CMINSA=XA1
     60 TO 5603
5601 CMAX54=XA1
     CMINGA=NA1
5603 FA=AMAX1 (AHS(EXA5), ARS(ENA5))
     TF (EA-FYP/FS1) 5352+5352+302
5352 CALTSA= (CMAX5A+CMIN5A)/2.
     CMN5A=(CMAX5A+CMIN5A)/2.
     FALTSA=CALTSA
     SAL TSAEF AL TSA
     SMNSA=CMNSA
     SCHKSAZAHS (SMNSA+SALT5A)
     TF (SCHK54 .GT. FUR) 850+815
 915 TF (SCHK54-FYR) A16, 816, 817
 H17 IF (AHS (SALTSA) - FY) 818+818,819
```

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HTR SMNSA=FYR+SALTSA

```
GO TO 416
 HIG SMNSA= ..
 ALS SMAXSA=SMAISA+SALTSA
     SMINGA=SMNGA-GALTSA
     RC1=SHINSA/SHAX5A
     M = 0
5350 M=M+1
      TF (RC1-R(M)) 5351.7004.5350
5351 FFC54=((R(N)-RC1)/(R(M)-R(M-1)))+(FF(M)-FF(M-1))+FF(M-1)
      GO 10 7005
7004 FFC54=FF(M)
7005 CONTINUE
      TH (ANS (SUAXEA) -FECSA) 5363+5363+850
5363 XH1=+ 3H5+ 585- 5885+ 5845-5855
      XH2=-50FH5+5H5
      XH3=915
      NH1=+SH1-SH1-SPR1+SPA1-SRS1
      NH2=-SCEP1+SP1
      NHJ=ST1
      FX85=+70711+((X31-X32)++2+(X82-X83)++2+(X83-X81)++2)++5
      FN85=.70711+((NR)-NR2) ++2+(NR2-NR3) ++2+(NB3-N81) ++2) ++.5
      FH=AMAX1 (AHS (FXR5), AHS (EN85))
      1E (EH-FAD/E21) 151 + 151 + 305
 721 TF (AHS(XH1) - ARS(Nd1)) 5605.5605.5604
5605 CMAK50=N81
      CMINSH=XR1
      GU TO 5362
5604 CMAX53=XH1
      CMINSH=NH1
5362 CALTEN= (CMAXER-CMINER) /2.
      CMN5R= (CMAX5R+CMIN5R)/2.
      FALTSHECALTSH
      SALTSH=FALTSH#KTF
      SMNSH=CMNSH+KTF
      SCHKSH=AHS (SUNSA+SALTSH)
      TE (SCHK5H .GT. FUR) 850+820
  H20 IF (SCHK5H-FYR) 821+821+822
  822 TE (ARS (SALTSH) - FYR) 823+823+824
  H23 SMNSH=FYP-SALTSH
      60 TO 521
  424 SMN5H=0.
  421 SMAXSH#SMNSH+SALTSH
      SMINSH=SMNSH-SALTSB
      RC2=SHINSH/SHAX58
      M = ()
 5360 M=M+1
      TF (RC2-R(M)) 5361 .7006 .5360
 5361 FFC5H=((R(M)-PC2)/(R(M)-R(M-1)))*(FF(M)-FF(M-1))+FF(M-1)
      GO TO 7007
 7006 FFC5R=FF(M)
 7007 CONTINUE
      TF (AHS (SMAX5H) - FFC5R) 5403.5403.850
      LOAD CALCULATIONS CONDITION & COLD.
C
 5403 P6=P
      FA6=P64(P1/4.)+(4+42)
      FPL5=P6#(PT/4.)#(S0D##2-B##?)
```

•

TLO=FA6+ .S#THS#AT	
MP6=((L#H0#G0##2#FF)/(.91#V))#(C1#	++2+camma+P6/(XT+(XT++2+C2+GE+G
1 AMMA) )	
MChau(2	
DICJEDIJC	a a na an an an an an ann ann an an an a
()TH1#=()T1++	
N1H5==()15H	
NTH3≠=0T3H	
TMINISIMTN	
TMINZETMIN+DTC1	· · · · · · · · · · · · · · · ·
TMIN3=TMIN+DIC2+DIC1	
TM(N A = [M   N A M   O A A M   O A A M   O A A M   O A A M   O A A M   O A A M   O A A M   O A A M   O A A M   O A A M   O A A M   O A A M   O A A M   O A A M   O A A M   O A A M   O A A M   O A A M   O A A M   O A A M   O A A M   O A A M   O A A M   O A A M   O A A M   O A A M   O A A M   O A A M   O A A M   O A A M   O M   O A A M   O A A M   O A A M   O A A M   O A A M   O A M   O A M   O A M   O A M   O A M   O A M   O A M   O A M   O A M   O A M   O M   O M   O M   O M   O M   O M   O M   O M   O M   O M   O M   O M   O M   O M   O M   O M   O M   O M   O M   O M   O M   O M   O M   O M   O M   O M   O M   O M   O M   O M   O M   O M   O M   O M   O M   O M   O M   O M   O M   O M   O M   O M   O M   O M   O M   O M   O M   O M   O M   O M   O M   O M   O M   O M   O M   O M   O M   O M   O M   O M   O M   O M   O   O M   O M   O M   O   O   O   O   O   O   O   O   O   O   O   O   O   O   O   O   O   O   O   O   O   O   O   O   O   O   O   O   O   O   O   O   O   O   O   O   O   O	
TMAX3=1MAX+01H2+01H1	and a constant of an and a constant of the con
TMX4#TMXX+DTH3+DTH2+DTH1	
MCTGA= (PI/3.) + (R++2) + (HETA++2) + (JFE	+TW++3)/(12.+(1FMU++2)))+FTE+
1(D1C1)4(1。+AFTA4XT)	
IF (DTU1 .FQ. 0.0)7008,7009	
7009 MHTGA=MCTG6+DTH1/DTC1	
GO TO 7010	
TODA MATGAEMOTGA	
HOIVETFFLOFILA HOIVETFFLOFILA	MD4 MATCA TI CHUG
	MPh=MC0+MC100=1204H0
Rh3L=(P6+H+SMI))/(4.*TA*SEE)-(P6+H*(	((A+H)/2.))/(4.*((A+H)/2.)*PE)
1-FTE*(IMIN2-70.0)*SMD/2.+STE*(TMIN)	(-70.0) *SMD/2.
₽64C=(HCR#(SX+XT+KT)+BTE#(TMIN4=70.	(1) * ( <x+xt+rt) *<="" =="" ste*(tmtn1="70.0)" th=""></x+xt+rt)>
15X-FTE4XT+([MIN2-70,0)-RTE4RI4(JMIN	13-70.011/(DBA-DRA)
R65C=DFA/((H2+H3)+(DBA=DRA=DSA))	
R66C=(0HA-DRA)+(R64C-R61C)/(DRA-DRA	-054)
R67C=R66C+(H2+H3)+R61C+H2-(R63C+XT)	/(DSR#2.)+R62C
P68C=(1(H65C+(H2+H3))-(DFR+XT++2)	/ (DSp#4.))
M06C=1401-867C/068C	
	SD1 -04 30 /050
	(3K) - H03(703K
	• • • • • • • •
MMD(2010)200	
TF (SL6C-MSL) 331,6390,6390	· · · · · · · · · · · · · · · · · · ·
390 R61CN=-FPLA-TLA	
962CN=-FPL6+H4+(P6+P1+H+SX)+(XT/2.)	-MP6+MCTG6-TL6+H6
R63CN=(P6+8+SMD)/(4.+TA+SEE)-(P6+8+	·((A+R)/?•))/(4•#((A-B)/2•)#FE)
1-FTE+(TM1N2-70.0)*SMD/2.+STF+(TM1N)	-70.01*SM0/2.
R64CN=(BTF+(TMTN4-70.0)+(SX+XT+RT)+	STE# (TMIN1-70.0) #
1SX-FTEPXT+(TMTN2-70.0)-RTF#RT+(TMIN	(3-70.0))/(DBA+DRA)
865CN=UFA/((H2+H3) + (DHA-DRA-DSA))	neer one the grade at a first state of the first of the state of the s
REACHE (URA-DRA) & IREACH-REATINA / INPA-	
	DRA-05AV
	DRA-DSA)
	DRA-DSA) +************************************
MUBUNISMUTSU670N/Q680N	DRA-DSA) +**T)/(DSR+2+)+R62CN 2)/(DSR+4+)}
	DRA-DSA) +XT)/(DSR+2+)+R62CN 2)/(DSR+4+)}
SLACN=SL1-RA7CN#RA5CN/R68CN-R66CN	DRA-DSA) *XT)/(DSR*2+)+R62CN 2)/(DSR*4+)}
SL6CN=SL1-R67CN#R65CN/R68CN-R66CN RSL6CN=RSL1+(R67CN#DFR#XT)/(R68CN#2	DRA-DSA) *XT)/(DSR*2+)+R62CN 2)/(DSR*4+)}
SL6CN=SL1-R67CN#R65CN/R68CN-R66CN RSL6CN=RSL1+(R67CN#DFR#XT)/(R68CN#2 RL6CN=HL1-R61CN-R66CN-R67CN#R65CN/R	DRA-DSA) *XT)/(DSR*2+)+R62CN 2)/(DSR+4+)} **DSR)-R63CN/DSR 68CN

C STRE	SS CALCULATION CONDITION & COLD
	SHOCH= (HI 6CN#H2+SI 6CN#H3+FPI 6#H4+TL6#H6) / (1#G]##2# (8+G0))
	SHS6CN# (USI 6CN+XT/2.)/(L+G1++2+(B+G0))
	SPR6CN=MP67(1+G1++2+(R+G0))
	SCG6CN=MCTG6/(1+G1++2+(B+G0))
	SH6CN#.5#THM/7H
	5ΡΑ6ΩΝ#ΕΛ67ΔΗ
	SCEBAC=0.
	SCEASC#PS
	SP6CN=MU6CN+(1,333+XT+LCE+1.)/(L+XT+2.+(R+G0))
	ST6CN=406CN++(Y/(XT++2++))-2+5P6CN
	FSS6CN=HL6CN/(T00*PI*XT)
	SSC6CN=S1 6CN/(((R+2.*TA)**2-R**2)*P1/4.)
	HTS6CN=BLSCN/(RTA(T) + BN)
	STRACN= (MRACN+YR) / (RT++2+RID)
	RSS6CN=BI 6CN/(A*PI*RT)
	SF86CN=4.+HL6CN/(PI*(A**2-R10**2))
	1F (SF86CN+FCYC/2.)6314.6314.126
126	TF(+001-AT)6314+6314+120
6314	TF (HTSOCU-RYC/FS3) 6344, 6344, 953
951	TF(0-11)6344.313.313
6344	TF (SR6CN-Fyr/FS1)6315+6315+302
6315	TF (ST6CN-FYC/FS1) 6316+6316+302
6316	YF (FSS6CN-FSU/FS1)6317+6317-302
6317	1F (55C6CN-5CYC/F51) 6330 6330 627
627	PRINT 628
628	FORMAT (3X.21H SSC6CN EXCEEDS LIMIT)
6330	TF (STH6CN-RYC/FS1)6331+6331+303
6331	TF (R\$56CN-RSU/F\$1)6332+6332+303
6332	XA1=-SH6CN+SH6CN+SPR6CN+SPA6CN+SRS6CN-SCG6CN
··· — • · ·	XAZ=+SCEA6C+SR6CN
	XA3=ST6CN
	FXA6CN=.70711+((XA1-XA2)++2+(XA2-XA3)++2+(XA3-XA1)++2)++5
	FA=EXA6CN
	1F (EA-FYC/FS1) 6363, 6363, 302
6363	XH1=+SH6CN+SH6CN-SPR6CN+SPA6CN-SRS6CN+SCG6CN
	XAZ=SCERAC+SRACN
	xB3=STOCN
	EX86CN=+70711+((X81-X82)++2+(X82-X83)++2+(X83-X81)++2)++5
	FREEXIGECN
	IF (ER-FYC/FS1) 6371,6371,302
С	LOAD CALCULATIONS CONDITION & HOT
6371	R61H=-FPL6-TL6
	R62H=-FPL6+H4+(P6+P1+H+SX)+(XT/2+)-MP6-MC6+MHTG6-TL6+H6
	R63H=(P6+R+SMI))/(4.+TA+SEE)-(P6+B+((A+B)/2.))/(4.+((A-B)/2.)+FE)
1	1-FTE#(TMAX7-70.0)#SMD/2.+STE#(TMAX1-70.0)#SMD/2.
	R64H=(HCR+(SX+XT+RT)+BTE+(TMAX4-70,0)+(SX+XT+RT)-STE+(TMAX1-70,0)+
1	15X-FTE#XT#(TMAx2-70.0)=RTF#RT#(TMAX3-70.0))/(DBA=DRA)
<b></b>	R65H=DFA/((H2+H3)*(DRA-DRA-DSA))
	R66H=(DBA-DRA)+(R64H-R61H)/(DRA-DRA-DSA)
	R67H=R66H+(H2+H3)+R61H+H2=(R63H+XT)/(DSR+2+)+R62H
	R68H=(l(R65H+(H2+H3))-(DFR+XT++2)/(DSR+4.))
	M06HaM01-R57H/R58H
	SL6H=SL1-R67H#R65H/R68H-R66H
	RSL6H=RSL1+(R67H+DFR+XT)/(R68H+2.+DSR)-R63H/DSR

```
MR6H=HL6H#HS
      TF (SL6H-MSL) 331 . 8390 . 8390
 A390 R61HN#-FPL6-TL6
      R62HN=-FPL6+H4+(P6+PT+B+SX)+(XT/2.)-MP6+MHTG6-TL6+H6
      R63HN≈(P6+B+S4D)/(4.+TA+SEE)-(P6+B+((A+B)/2.))/(4.+((A-B)/2.)+FE)
     1-FTE+([MAX2-70.0)+SM0/2.+STE+(TMAX1-70.0)+SM0/2.
      R64HN=(BTE+(TMAX4-70.0)*(SX+XT+RT)-STE+(TMAX1-70.0)*
     15X-FTE*XT*(TMAX2=70.0)-RTE*RT*(TMAX3-70.0))/(DBA-QRA)
      R65HN#DFA/((H2+H3)+(DBA=DRA=DSA))
      D66HN= (DHA-DPA) + (R64HN-R61HN) / (DBA-DRA-DSA)
      R67HN=R66HN+ (H2+H3)+R61HN+H2- (R63HN+XT)/(DSR+2.)+R62HN
      R68HN=(1.=(R65HN+(H2+H3))=(DFR+XT++2)/(DSR+4.))
      MOGHNEMO1-R67HN/R68HN
      SLOHN=SL1-RG7HN#R65HN/R68HN-R66HN
      RSL6HN=RSL1+(R67HN#DFR#XT)/(R68HN#2.#DSR)-R63HN/DSR
      BL6HN=HL1-R61HN-R66HN-R67HN+R65HN/R68HN
      MROHN=HLAHN+H5
C STRESS CALULATION CONDITION 6 HOT
      SHOHN= (HL6HN+H2+SL6HN+H3+FPL6+H6+TL6+HK)/(L+G1++2+(B+G0))
      SRS6HN= (R5L6HN+XT/2.)/(L+G1++2+(R+G0))
      SHG6CN=MHTG6/(L+G1++2+(B+G0))
      SPR6HN=MP6/(L#G]##2#(H+G0))
      SH6HN=+5+TPM/ZH
      SPA6HN=FA6/AH
      SCEH6H=0.
      SCEA6H=P6
      SR6HN=MO6HN#(1.333#XT#LCE+1.)/(L#XT#2.#(8+60))
      ST6HN=M06HN# (Y/ (XT##2#8)) - Z#SR6HN
     FSS6HN=HL6HN/(TOD*PI*XT)
      55C6HN=SL6HN/(((B+2.*TA)**2-B**2)*P1/4.)
     HTS6HN=BL6HN/(BTA(I) +HN)
      STR6HN# (MR6HN#YR) / (RT##2#RID)
     PSSSHN=HLSHN/(A*PI*RT)
      SF86HN=4.#RL6HM/(PI#(A##2-RID##2))
      TF (SF86HN-FCY/2.)8314.8314.127
 127 TF(.001-AT)8314.8314.120
8314 IF (HTS6HN-RY/FS3) 8344+8344+963
 963 TF (0-11) 8344+313+313
8344 TF (SR6HN-FY/FS1)8315.8315.302
8315 TF (ST6HN-FY/FS1)8316,8316,302
8316 IF (FSS6HN-FSU/FS1)8317+8317+302
8317 IF (SSC6HN-SCY/FS1)8330.8330.629
 629 PRINT 631
 630 FORMAT (3X. PIH SSC6HN EXCEEDS LIMIT)
8330 IF (STR6HN-RY/FS1)8331,8331,303
A331 TF (HSS6HN-RSU/FS1)8332+8332+303
8332 XA1=-SH6HN+SH6HN+SPR6HN+SPA6HN+SR56HN-SHG6CN
      XA2=-SCEA6H+SR6HN
     XA3=ST6HN
     FXA6HN=.70711+((XA1-XA2)++2+(XA2-XA3)++2+(XA3-XA1)++2)++.5
     FA=EXA6HN
     TF (EA-FY/FS1) 8363, 302, 302
A363 XH1=+SH6HN+SB6HN-SPR6HN+SP46HN-SR56HN+SHG6CN
     XH2=-SCER6H+SR6HN
     XH3=516HN
     FXH6HN=.70/11+((XH1-XB2)++2+(XB2-XB3)++2+(XB3-XB1)++2)++.5
```

```
FH=EXHOHN
      TF (ER-FY/FS1) H371+302+302
     LOAD CALCULATION CONDITION 7
¢
A371 P7=0.
      FA7=0.
      FPL7=0
      TL /= SATHSAAT
      $71=-TL7
      P72=-117446
      R73=1.
      R74=∩.
      P75=DFA/((H2+H3)*(DBA+DRA+DSA))
      R76=(UHA-DRA) + (R74-R71) / (DBA-DRA-DSA)
     P77=R76# (H2+H3)+R71#H2= (P73#XT)/(DSR#2.)+R72
      R7H=(1.=(475+(H2+H3))=(DFR+XT++2)/(DSR+4.))
      M07=MU1-H77/H78
      517-511-877#875/878
      RSL7=RSL1+(R77*DFR*XT)/(R78*2.*DSR)-R73/DSR
      HL7=HL1-H71-H76-R774R/5/R78.
      MH7=HL7+H5
      TF(0,001-HNT)871,870.870
  870 RISCHK=MAX1F (HTS2P, BTS3, BTS6HN)
      IF (HTSCHK _LT, 0.83*RY/FS2)873+871
  H73 TF (HT54 .LT. 0.83#80/FS2) 872.871
  B 2 BN1=(BN#BTSCHK#FS2)/(0+834BY)
      RN2=(PI+BCD)/(2.+(BS(I)+XT))+2.
      HN=MAX1F (HN1+HN2)
      NR=RN\S.
      N=2*NB
      AN=N
     BNISHN
                                           -----
      11=1
      RCDT=HCD
      GO TO 117
  971 CONTINUE
      T1=P1/4.4T00##2
     R1=PI/4.*(TOD+2.*HURI)**2
WGT=2.*PI/4.*((A**2-TOD**2)*XT*FPHO+(ROD**2-RID**2-BN*BH(I)**2)*RT
     1+RRHO+BN+RS(I)++2+(SX+XT+RT+2++RS(I))+RRHO)+2++BN+NWT(I)+2++H+FRHO
     1+(1./3.*(R1+T1+(R1*T1)**.5)-T1)-2.*PI/4.*(SOD**2-TOD**2)*.025*FRH0
     1-PI/4.*(SOD+#2-(H+2.*TA)*#2)*SLEG*TAN(PT/9.)*FRH0
      TRQ=(HL1#45(I))/(BN#5.)
 IF (PI*HCD/BN-2.*(BS(I)*XT))333;333:1000
1000 PHINT 1001
 1001 FORMAT (30H MAXIMUM SPACING NOT SATISFIED)
  333 CONTINUE
      IF ((BCU-RID)/2.-BH(I))118,119,119
 11A PRINT 35
  35 FORMAT (3x,41HDIFFERENCE IN BCD AND RID LESS THEN BH(I))
  119 CONTINUE
      PRINT 127
 727 FORMAT(1H )
      PRINT /28
  728 FORMAT(25X+14H DESIGN OUTPUT///)
      PHINT 2800
 2800 FORMAT(1X+25HRENDING MOMENT AND STRESS)
```
3H01	FURMAI(6x, 3 H 11M, 7X, 3 H (95)
	PWINT 4020, TRM. TRS
	PHINT 2803
5803	FORMAT (7X, 3HTHQ)
	PRINT 9020, TRJ
	PRINT 3001
3001	FORMAT (25H LOADS FOR ALL CONDITIONS)
	PRINT 3002
3005	FORMAT (6X, 3HM01, 7X, 3HBL1, 7X, 3HSL1, 6X, 4HRSL1)
	PRINT 9020,401,811,811,8511
	PKINT 3003
3003	FORMAT (6X.3HM02.7X.3HBL2.7X.3HSL2.6X.4()RSL2.7X.3HMC2.7A.3HMP2)
0.00	PRINT 9020-M02-BL2-SL2-RSL2-MC2-MP2
	PRINT 3004
3004	EDRMA ( (5X-4)-402P+6X-4HBI 2P+6X+4HSI 2P-5X-5HBSI 2P)
3004	BRINT 9020-M020-BI 2P-SI 2P-BI 2P
	BRINT ROAD
3005	ENDMAT (EY ALMOOMSEYSAHRIOMSEYSAHSIOMSEYSAHSIOMSEYSAHSIOMSEYSAHSIOMSEYSAHSIOMSEYSAHSIOMSEYSAHSIOMSEYSAHSIOMSEYSAHSIOMSEYSAHSIOMSEYSAHSIOMSEYSAHSIOMSEYSAHSIOMSEYSAHSIOMSEYSAHSIOMSEYSAHSIOMSEYSAHSIOMSEYSAHSIOMSEYSAHSIOMSEYSAHSIOMSEYSAHSIOMSEYSAHSIOMSEYSAHSIOMSEYSAHSIOMSEYSAHSIOMSEYSAHSIOMSEYSAHSIOMSEYSAHSIOMSEYSAHSIOMSEYSAHSIOMSEYSAHSIOMSEYSAHSIOMSEYSAHSIOMSEYSAHSIOMSEYSAHSIOMSEYSAHSIOMSEYSAHSIOMSEYSAHSIOMSEYSAHSIOMSEYSAHSIOMSEYSAHSIOMSEYSAHSIOMSEYSAHSIOMSEYSAHSIOMSEYSAHSIOMSEYSAHSIOMSEYSAHSIOMSEYSAHSIOMSEYSAHSIOMSEYSAHSIOMSEYSAHSIOMSEYSAHSIOMSEYSAHSIOMSEYSAHSIOMSEYSAHSIOMSEYSAHSIOMSEYSAHSIOMSEYSAHSIOMSEYSAHSIOMSEYSAHSIOMSEYSAHSIOMSEYSAHSIOMSEYSAHSIOMSEYSAHSIOMSEYSAHSIOMSEYSAHSIOMSEYSAHSIOMSEYSAHSIOMSEYSAHSIOMSEYSAHSIOMSEYSAHSIOMSEYSAHSIOMSEYSAHSIOMSEYSAHSIOMSEYSAHSIOMSEYSAHSIOMSEYSAHSIOMSEYSAHSIOMSEYSAHSIOMSEYSAHSIOMSEYSAHSIOMSEYSAHSIOMSEYSAHSIOMSEYSAHSIOMSEYSAHSIOMSEYSAHSIOMSEYSAHSIOMSEYSAHSIOMSEYSAHSIOMSEYSAHSIOMSEYSAHSIOMSEYSAHSIOMSEYSAHSIOMSE
3902	PUNHAL CAALAMMA MAA AMAGA TAALAMA
	PRINE BURGHURMERLEHEBLEHEBLEHE
30.04	PRINT 2000 Primat (24 Juna 77, 300 3.77 306) 3.47,4006) 3.77,304031
2010	PUTMAL (0K)3H403((A)3H023((A)3H023(A)3HH323(A)3HH323(A)3HH32)
	PRINT AUCUMMUSHESISLAIRESISHPA
	PRINT 2007
3097	FURMA1 (6X+3H9L4)
	PRINT 9020,8L4
	PRINT 3008
3004	FORMAL (6X, 3HM05, /X, 3HHL5, 7X, 3HSL5, 6X, 4HRSL5, 7X, 3HMP5)
	PRINT 9020, M05+8L5, SL5+8SL5, MP5
	PRINT 3009
3009	FORMAT (5X,4HM06C,6X,4HBL6C,6X,4H5L6C,5X,5H4SL6C,6X,3HMC6,7X,3HMP6
	PRINT 9020,MOAC, PLAC, SLAC, RSLAC, MC6, MPA
	PRINT 3010
3010	FORMAT (5X+5H4CTG6+5X+5HMHTG6)
	PRINT 9020,MCTG6,MHTG6
	PRINT 3011
3011	FORMAT (5X,5HM060N,5X,5HBL60N,5X,5H5L60N,4X,6HRSL60N)
	PRINT 9020, MOSCN, BLECN, SLECN, RSLECN
	PRINT 3012
3012	FORMAT (5X,4HM96H,6X,4HBL6H,6X,4HSL6H,5X,5HRSL6H)
	PRINT 9020.MO6H.BL6H.SL6H.RSL6H
	PRINT 3013
3013	FORMAT (5X.5HMOGHN.5X.5HBLGHN.5X.5HSLGHN.4X.6HRSLGHN)
	PRINT 9020 MOGHNOBI 6HNOSI 6HNORSL6HN
	PRINT 3014
3014	FORMAT (6X.3H407.7X.3HBL7.7X.3HSL7.6X.4HPSL7)
2 · · ). •	PRINT 9020-MC7-BL7-SL7-RSL7
	PRINT 9104
0104	FORMALLIX, 27HSTRESS, CONCENTRATION, FACTORS
7174	
0105	EORMAT(6Y, SHKTC)
7103	
	THE ATEL FOR FORTE
	PPINT 1015
2015	PRINT 3015
3015	PRINT 3015 FURMAT (28H STOFSSES FUR ALL CONDITIONS)

3016	FORMAT (AX, 3HEXA, 7X, 3HENA, 7X, 3HEXB, 7X, 3HENB)
	PHINT YOPO,FXA,ENA,EXB,ENB
	PRINT 3017
3017	FORMAT (5X,4HFXA2,6X,4HENA2,6X,4HEXB2,6X,4HENB2)
	PHINT 9020, EXA2, ENA2, EXA2, ENA2
	PRINT JOIR
3019	FORMAT (4x, 6HSMAX2A, 4X, 6HSMIN2A, 5X, 5HFFC2A, 4X, 6HSMAX2B, 4X, 6HSMIN2B,
1	5X+5HFFC2H)
	PRINT 9020, SMAX2A, SMINZA, FFCZA, SMAX2R, SMIN2B, FFCZB
	PRINT 722
252	F0RMAT(5X+4H5H2P+5X+5HSR\$2P+6X+4HSH2M+5X+5HSR\$2M+6X+4HSPR2+7X+3H\$B
J	2+6×+45PA21
	PFINT 9020, 5420, 58520, SH2M, SRS2M, SPR2, SH2, SPA2
	PRINT 723
723	FORMAI(5x+4H%APP+6X+4HSA20+6X+4HST2P+6X+4HST2M)
	PRINT 9020, SR2P, SR2E, ST2P, ST2M
	PRINT 3019
3019	FORMAT (BX.LPFXA3.6X.4HENA3.6X.AHEXB3.6X.4HENB3)
	PRIMT 9020.EXA3.ENA3.EXB3.ENB3
	PRINT 3020
3030	FORMAT (5x, 4HE (45, 6X, 4HENA5, 6X, 4HEXB5, 6X, 4HENB5)
	PRENT 4020, FXAS, FNAS, EXBS, ENAS
1	PRINT 3021
3051	FORMAT (4X+6HS4AX5A+4X+6HSMIN54+5X+5HFFC5A+4X+6HSMAX5B+4X+6HSMIN5B+
<b>1</b>	5ו5HFFC5B)
	PHINT 9020, SMAX5A, SMINSA, FFC5A, SMAX58, SMIN58, FFC58
14 m .	PRINT 724
174	FURMAI (5X,375H5,6X,445K55,7X,345H5,5X,445PA5,7X,345K5,7X,3H5T5,
1	5,7,4,4,7,4,8,7,7,7,7,7,7,7,7,7,7,7,7,7,7
	PRINE YUZU ( 5P5 ( 5P35 ) 5P3 ) 5P45 ( 5 ( 5 ) 5P95
****	PRINE (25)
123	FURMAI (0X \$ 305M) \$ 0A \$ 4H 3H 3 [ \$ 7 X \$ 3H 3H 1 5 X \$ 4H \$ MAI \$ 7 X \$ 3H 3H 3H 3 [ \$ 7 X \$ 3H 3 [ \$ 7 X \$
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3766 1	FUNNERI - LONG DEFRADUNG DATUELABBURG DATELAGBENG (DASCERADUER) BUTNERI - UDOA - EXAZENI - EXAZUNI - EXAZUNI - EXAZUNI
3030	FOINT JYEN FOINT (LY NUDICLEY EHRICODLEY AUDICDLEY AUDICALES EXTERDIOTSELEY.
1027	KUNNAT ANAN JUDI JIJAN DUDI JEKINAN MUDI JENDAN
	DECTATION AND AND ADDRESS ATCA. ATCA
	DEINT REAL
3037 3	CORMAL LYY, BUCCH, SY CUSEROD, SY SUSEROM, AV ANSERRAAX ANSERSAAY, ANSE
1.1.1.1.1.1.1.1.1.1.1.1.1.1.1.1.1.1.1.	RACNIAXISHSERAHNY
	PRINT 9020. SER. SERZA. SERZA. SERS. SERS. SEDACN. SERAN
	DRINT 3023
3023 1	FORMAT (1)H DIMENSIONS)
1	PRINT 2802
2802 1	FORMAT (9X .) HHAAX . 4HHUB1 . 8X . 2HG1 . 8X . 2HE DY
	PRINT 9071.H.H.HIATIGI.FR
1	PRINT 3024
3024	FOHMAT (6X.2HTw.7X.3HTQD)
	PHINT 9021. TW. TOD
1	PRINT 3025
3025	FORMAT (6X.1HA.9X.1HB.8X.2HXT.7X.3HBCD)
1	PRINT 9021.A.R.XT.HCO

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	PRINT 3030
3030	FORMAT (6X, 3HROD, 7X, 3HRID, 7X, 2HRT)
	PRINT Y021, POD, RIU, RT
	PRINT 3026
3026	FORMAT (6X, 34500, 7X, 34510, 7X, 345EL, 8X, 24TA, 5X, 545LEGT)
	PRINT 9021+SOD+SID+SFL+TA+SLEGT
	PRINT 3027
3027	FORMAT (6X,5HB5(]),5X,5HBH(]),4X,6HBTA(]),4X,6HBWC(])+6X,2HBN)
	PRINT 9021,85(1),8H(1),8TA(1),8WC(1),8N
	PRINT 3028
3028	FORMAT (6%, 3HWG1)
	PRINT 9021,WGT
	PHINT 9103
9103	FORMAT(1H1)
	GO TO 999
99R	CALL EXIT
	END

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# FIGURE B-1. PUNCHED INPUT CARDS 1 TO 18 FOR 3-INCH 1500-PSI ALUMINUM INTEGRAL, INTEGRAL CONNECTOR

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1 2 2 4 2 8 7 8 8 10 10	1⊈[12]16-15 10[12]18-19-24[21-22 633.521-18.4€	23 26[25 28 27 28]26 28 28	rulat 24 at atta at a anier e	13 ef eside an es andaño sa 21 25	an at an antar ar an	ten andere en stander varigi	and the second
*	and the second division of the second divisio						

# FIGURE B-2. PUNCHED INPUT CARDS 19 TO 36 FOR 3-INCH 1500-PSI ALUMINUM INTEGRAL-INTEGRAL CONNECTOR

(Read from Bottom of Page to Top) E-59

## INPUT TO THE DESIGN

10

11145 1 0AD	- 1						
TRAN	TNPU						
1001		_ <del></del>					
• () V L Q • THOE - DE () M	CT.LY						
TUNE GEOMI					······		
	1#1						
3.0000	0.000		· · · · · · · · · · · · · · · · · · ·		<del></del>		
IUNE MAILE	RIAL PROPE	4116.5	<b>F</b> . 3				
		<u>It</u>	22200	IFRO			
42000 El Ange Gui	JACTUV	19000	32200	28400			
PLANGP DE	JHEINT			·····			
HCUI	MUHII MUHII						
7.0000	0.0000						
910TW1							
1.0000			· · · · · · · · · · · · · · · · ·				
FM1	+ 5t 1						
0.0000	2.0000	3-111-00	· · · · · · · · · · · · · · · · · · ·				
FLANGE MA	IERIAL PROP	CHILLS					
FUR							
45000	2.0		يرد هر سن	er	**	EVA	EVD
<u>U1</u>	<u> </u>	<u> </u>		<u></u>	<u> </u>	<u> </u>	<u> </u>
42000	0000	<u>51800</u>	35500	9900000	3800000	40000	35000
FRMU		<u>FIE</u>	PLH				·····
-1000000	.3300000	• 20091300	.00020000	0.45		0(7)	() ( )
<u>R(1)</u>	4(2)	H(3)	H(4)	<u>H(5)</u>	<u> </u>	<u> </u>	R(8)
-1,0000			-+4000	2000	0.0000	. < 0 0 0	• • • • • •
<u> </u>		<u>×(1)</u>		·····			
.6000	-3000	1.0000	<b>66</b> ( ) )				<b>FF</b> (0)
<u> </u>	++ (2)	<u> </u>	FF (4)	<u> </u>	<u>FF(6)</u>	<u> </u>	77 (8)
	21000	22000	24000	50000	28400	30000	32000
<u> </u>	PF(10)	<u> </u>					
33000	0.000	34000					
SEAL GEUME	EIRY			TAOTH	<u></u>		
5001	2101	DELI	141		SLEGIL		
0.0000	0.0000	1.0000	0.0000	• 9000	0.0000		
SEAL LUADS	5 INPUTS	1					
RSLI	<u></u>	<u> </u>	·				
850	510 Stat BOOMER	375					
SEAL MATER	TAL PHUPE	COVO	CAVD	EVP			
364	555 0000000	5070	3618	716			
17200	AA00000	40500	22000	32200			
517. 0000-300							
.00001300							
HULI GEUME	1 PC 1	() +	0444	nu / 5 v	0	04/71	
<u></u>	<u>нн ( с )</u>	<u>×H(3)</u>	<u>1871(4)</u>		RH [0]	<u>BU(1)</u>	6110
.7010	)でのち <u>。</u>	0856.	0166.	•4090	.5310	•3730 04/15:	.0000
<u>HH(A)</u>	<u></u>	<u>nn(11)</u>	041151	<u>nn(13)</u>	DM(14)	(6)/10	·
•7970	0160.	1.0500	101640	1.3150	1.4340	1.0030	0.00101
HWC(1)	HWC(2)	H#C(3)	BWC (4)	HWC (5)	MAC (0)	BWU(/)	DWL(B)
. 5230	- KOBU	• 7230	02180	• 4080	1.0520	1.4000	1.3120
HWC(A)	<u>HWC(10)</u>	HWC(11)			<u>DWC(14)</u>	0WC(12)	
1.6250	えょりりりう	2.2500	2.3/30	2./500	3,0000	きゅくうりり	

RS(1)	H5(2)	HS(3)	BS(4)	85(5)	R\$(6)	BS(7)	<b>BS(8)</b>
.1900	.2500	.3125	.3750	.4375	.5000	.5625	.6250
<u>BS(A)</u>	HS(10)	83(11)	H2(15)	BS(13)	BS(14)	HS(15)	
.7500	. 4720	1.0000	1.1250	1.2500	1.3750	1.5000	
BIA(1)	HIA(C)		HTA (4)	BTA(5)	BTA(6)	BTA(7)	BTA(8)
.0200	.0304	+ 0580	.0878	.1187	.1599	.2030	.2560
HTA(9)	HTA(1V)	HTA(LL)	ЫГА(1Ż)	BTA(13)	BTA(14)	BTA(15)	
.3730	.5070	.6630	.8560	1.0730	1.3150	1.5810	
NWT(1)	NWF(C)	(E) T (N)	NWT(4)	NWT (5)	NWT(6)	NWT (7)	NWT(8)
.0000	. 9032	.0040	.0009	.0100	.0150	. 0200	.0310
NWT (Y)	NWT(10)	NWT (11)	NWT (12)	NWT (13)	NWT(14)	NWT (15)	
. 0440	.10/0	.1011	.1.100	.1600	1900	.2300	
005(1)	UAF (C)	DAF (3)	DAF (4)	DAF (5)	DAF (6)	DAF (7)	DAF (8)
. 3750	.4315	.5000	.5025	.6875	7500	.8750	.9375
DAE(9)	DAE(10)	()AF(11)	(IAE (12)	DAE (13)	DAE (14)	DAF (15)	
1 0625	1.2500	1.43/5	1.6250	1.8125	2 4620	2.2500	
	1.07.000	104313	1.02.00	100152	C . 092V	202300	
THE THE OT	Т:сал у						
11	1944						
U	10						
0.0000	T						
HULI MAIER	TAL PROPER	1123	0.40	010			
		<u> </u>	<u>BTC</u>				
47000	9900000	52000	28100	50000			
RIF	нск	BRHO					
•0001300	.00020000	-10000000					
SYSTEM PRE	SSURE AND	IEMPEHATURE		<b>P</b> . A 14			
e.	44	PB	PIM	IMAX	TMIN		
1200	2200	3000	5320	200	-425		
DIC1	0102	D1H1	DTH2				
60		20	<u> </u>				
F 51	F '52	FSB					
1.1000	1.25.0	1.0000					
A ## 11		NESIGN	OUTPUT				
<u> </u>							
RENALING MO	MENT AND S	IRESS					
TBM	THS						
7360	4(101)						
THO				······			
102							
LOADS FUH	ALL CUNULT	IVINS				· · · · · · · · · · · · · · · · · · ·	
MD1	н 1	St 1	RSI 1				
1.541	26004	26064	8856				
14201		612	PSI 2	MCO	MDO		
NOL			11200	825			
14019	20117	4440 01 10	11102	062	3444		
M() <b>C</b> P			11734	····			
14776	24320	5847	11/34				
MUSW	HLC11	<u> </u>	HOLEM				
14990	25014	23573	11/0/	ND 5			
MUJ	HL 3	51,1	<u>KSLJ</u>	MPJ	the second to the second to the second se		
15034	28400	9031	13149	2100			
RL 4							
25914							

M05	HL.5	SL5	RSLS	MP5		
15054	24573	8273	13391	5396		
M06	HLD	ՏԼԵ	HSLO	MC6	MP6	
13233	24145	5830	9777	825	3444	
MCTG6	MH106					······································
133	-44					
MOBEN	ALOCH	SL6CN	RSL6CN			
13389	25546	7231	9802			
MO6H	BLOH	SL6H	RSL6H			
15129	54715	9857	12360			
мабны	BLORIN	SL 6HN	RSL6HN			
15286	24213	11258	12385			
<u>M07</u>	HL1	SL7	RSL 7			
14528	26995	21637	8856			
STRESS CONCE	NTRATION F	ACTOR				
KTF		-				
2.0715						
STRESSES FOR	ALL CUNUI	TIONS				
EXA	ENA	EXH	ENB			
24053	24053	6984	6984			
EXA2	ENAZ	EXH2	EN82			
14100	19420	6057	4334			
SMAXZA	SMINZA	FFC2A	SMAX28	SMIN2B	FFC2B	
-150]]	-4014	31326	30207	9202	30954	
SH2P	SRS2H	SH2M	SRS2M	SPRZ	SB2	SPA2
18017	5157	18246	51/2	3454	2642	2236
SH2P	SREE	ST2P	ST2M			
6711	6000	11643	11811			
EXAS	ENAS	EXH3	ENB3			
13386	13306	4454	4454			
EXA5	ENAS	EXBS	ENBS			
12968	24053	4419	6984			
SMAX5A	SMINDA	FFC5A	SMAXSB	SMIN58	FFC5B	
-14625	-3115	31872	30295	20961	33270	
SH5	SRSS	SH5	SPAS	SR5	ST5	SPR5
17913	5806	0	3504	6837	11862	5412
SH1	SRSI	SB1	SPAL	SR1	ST1	SPR1
18517	3673	0	0	6622	11489	0
EXAGON	EXBOCIV	EXAGHN	EXBe	HN		
19444	12504	22630	14293			
HT5	HTSZH	BTS3	HTS4	BTS5	BTS6CN	BTS6HN
28091	31810	30675	27925	30790	27529	31868
DIMENSIONS						
H	HŪBI	Gl	FR			
1.2026	.2080	•4002	.1250			
T W	TOD					
-1422	3.0000					
Λ	fi	XT	BCD			
4.9421	2.7351	.4705	4.2861			
<u> </u>	SIU	SEL	TA	SLEGT	······································	
3.3144	2.7705	.5273	•1339	.0497		
μ5(1)	НН(1)	ATA(I)	BWC(I)	BN		
. 1125	. 3260	.0580	.7200	16.0000		
wGT					······································	
2.7732						



FIGURE B-3. DIMENSIONS FOR CONNECTORS WITH INTEGRAL FLANGES



FIGURE B-4. DIMENSIONS FOR CONNECTORS WITH LOOSE-RING FLANGES

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#### APPENDIX C

#### NOMINAL DIMENSIONS AND ASSEMBLY INSTRUCTIONS FOR AFRPL FLANGED CONNECTORS FOR CRYOGENIC SERVICE

This appendix contains nominal dimensions and assembly instructions for aluminum and stainless steel AFRPL flanged connectors for cryogenic service. The maximum operating pressures are 100, 200, 500, 1000, and 1500 psi. The line sizes include pipe and tubing from 1 inch through 16 inches in diameter. Standard wall thicknesses have been selected for the pipe sizes. The tubing wall thicknesses have been based on stress calculations because of the lack of industry standards. The dimensions are apclicable to the assemblies shown in Figure 1: (1) integral/integral flange assembly, (2) loose-ring/loose-ring flange assembly, and (3) integral/loose-ring flange assembly. Stud dimensions are given in addition to bolt dimensions because the stud form facilitates the use of special fastener materials.

As discussed in the report body, it is recommended that military standards be prepared: (1) for all connectors for tubing from 1 inch through 3 inches in diameter, (2) for aluminum connectors for pressures of 100, 200, and 500 psi for tubing through 16 inches in diameter, and (3) for stainless steel connectors for 100 psi for tubing through 8 inches in diameter. While the other connectors are probably satisfactory, further testing is recommended for selected connectors prior to the preparation of military standards. Some of this testing and the preparation of additional connector dimensions will be accomplished as a part of further work under Contract AF 04(611)-11204. A supplementary report will summarize this information.

It was mutually realized that considerable work would be required by the Air Force to complete the preparation of the recommended military standards. However, because of the number of decisions that had to be made by the Air Force concerning the scope and form of the military standards and specifications, it was decided to limit the data presented in this report. General recommendations concerning various aspects of possible standards and specifications are contained in the report body. The nominal dimensions and assembly instructions for cryogenic connectors are contained in Tables C-1 through C-16 of this appendix. Additional information is given below.

#### Materials

Connectors have been designed for use with aluminum and stainless steel tubing materials. The material for the connector flanges has been made the same as the tubing to minimize welding and corrosion problems. Different materials have been selected for other parts of the connectors to achieve improved performance. The materials for each type of connector are discussed briefly.

#### Aluminum Connectors

Aluminum alloy AMS 4117 (6061-T6) was selected for the integral flanges and the loose-ring flanges.

Overaged 6061-T6 aluminum was selected for the seal material on the basis of the work described in Technical Documentary Report No. AFRPL-TR-67-191. In an appendix to that report it was recommended that MIL-F-27417 be revised to include the necessary information for aluminum threaded connectors. Thus the military standards for flanged connectors could bear the same material designation, i.e.: aluminum alloy, AMS 4117, overaged per MIL-F-27417, paragraph 3.2.1.1.2.

The material selected for the aluminum bolts and nuts was aluminum alloy AMS 4119 (2024-T351).

#### Stainless Steel Connectors

Corrosion-resistant steel AMS 5646 was selected for the stainless steel integral flanges and loose-ring flanges.

The material selected for the stainless steel seals was the same as for the threaded connectors, i.e.: corrosion-resistant steel, AMS 5639, AMS 5650, or AMS 5651. The nickel plating required for the stainless steel seals for flanged connectors should be covered by MIL-P-27418.

The material selected for the bolts and nuts was A286. The specification given in MS 27852 for the A286 threaded-connector nuts was AMS 5735. In bolt form this material is cold-worked to a high yield strength and these are the types of bolts that were purchased and used in the test program. It is not known whether a standard specification exists for this condition.

#### Nut Form

For the test program, self-locking, hexagonal, ring-base, corrosion-resistantsteel A286 nuts were used. They were machined as per MS 21043. The 2024-T351 aluminum nuts were plain hexagonal type and machined as per AN 315-UNF-3B.

It is recommended that Specification AN 315 with UNF-3B threads be used for aluminum 2024-T351 nuts. The corrosion-resistant-steel nuts with UNF-3B threads can be made according to MS 21043. MS 21043 will have to be extended to include larger size nuts. As an alternative, the steel nuts can also be made according to AN 315.



a. Integral/Integral Flange Assambly



b. Loose-Ring / Loose-Ring Ficage Assembly



c Integral / Loose - Ring Flangs Assembly

## FIGURE C-1. TYPICAL AFRPL FLANGED-CONNECTOR ASSEMBLIES

Bolts and studs both are shown; seals are shown unseated.



1

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.£	9-6 9-6 9-6	1- N 9-3- 0-9- 9-9-	849°,	1 4 4 1 - C	CE1.		ECE,		1.996	1 . 1 00		1,243	1.367	1.590	.503	503.	561		6E33	199-	. 733	.747	. 69÷	47 A		303°1	1.100	1.150		- <b>0</b>	1.500
14	123		188		<b>5</b> 87,	<b>X</b> 83''	188 188 188 188 188 188 188 188 188 188	999 491 10 12 10 12 10 12	857	5	220 250	944  979 	Elt	1313	.125	-155	441		5710	10 10 10	52 T	.163	• 168	881 ·		- 28c -	-256	-250			616.
¥.	**** ****	2. 194 194	2°666 3ª194	3.05	199.4	5.121	6.247	177-1	9.559	16.652	12.657 12.652	13+852	14.653	16.542	• 12 € • 1 • 12 € • 1	) c <b>6</b> 34	21134	2.676	30264	1.9.5	1.65.4	5.213	6.107	840°7		0.0	10.470	11.767	11-912	14.842	5.97B
r.	6996 • 589 • 589 • 5	2.093	2-545 10-665	9 + 7 6 6 4 - 7 6 6	4.509	9,009 9,009 9,009 9,009 9,009 9,009 9,009 9,009 9,009 9,009 9,009 9,009 9,009 9,009 9,009 9,009 9,009 9,009 9,009 9,009 9,009 9,009 9,009 9,009 9,009 9,009 9,009 9,009 9,009 9,009 9,009 9,009 9,009 9,009 9,009 9,009 9,009 9,009 9,009 9,009 9,009 9,009 9,009 9,009 9,009 9,009 9,009 9,009 9,009 9,009 9,000 9,000 9,000 9,000 9,000 9,000 9,000 9,000 9,000 9,000 9,000 9,000 9,000 9,000 9,000 9,000 9,000 9,000 9,000 9,000 9,000 9,000 9,000 9,000 9,000 9,000 9,000 9,000 9,000 9,000 9,000 9,000 9,000 9,000 9,000 9,000 9,000 9,000 9,000 9,000 9,000 9,000 9,000 9,000 9,000 9,000 9,000 9,000 9,000 9,000 9,000 9,000 9,000 9,000 9,000 9,000 9,000 9,000 9,000 9,000 9,000 9,000 9,000 9,000 9,000 9,000 9,000 9,000 9,000 9,000 9,000 9,000 9,000 9,000 9,000 9,000 9,000 9,000 9,000 9,000 9,000 9,000 9,000 9,000 9,000 9,000 9,000 9,000 9,000 9,000 9,000 9,000 9,000 9,000 9,000 9,000 9,000 9,000 9,000 9,000 9,000 9,000 9,000 9,000 9,000 9,000 9,000 9,000 9,000 9,000 9,000 9,000 9,000 9,000 9,000 9,0000 9,0000 9,0000 9,0000 9,00000 9,0000 9,0000 9,0000 9,0000 9,0000 9,0000 9,0000 9,0000 9,0000 9,00000 9,0000 9,00000000	5.69 <b>6</b>	6.643 8.008	8.625	10.004	12,200	12.730	14.095	16+00	1.250	].500	2,000		3.000	3.500	4.000 6.500	5.090	5.583	6.000		8.622 8.623	10.190	10.752	360400		16.000
с,	2.646 20.090 10.000	14-450 14-450	14. <b>850</b> [4.900	12.606	18.948	180064 280064	20.000	32,030	44.660	996°34	54.000	56.600	62.200	72.300	6.000 10.690	10-030	12.000	14.000	14.003	14.000	14¢000 18.000	14.000	28.500	20.000		46.000	46.000	48.000	000.44	54.000	79.000
r		1000	. 201 . 201	.265 265	265	• • • • • • • • •	532	265.	265	. 265	5.90 5.90 5.90 5.7	.320	-265	\$92"	201	.201	192.		102.	- <b>2</b> 6.5	. 265	.265	e 265	- 265 - 265	202	542.	.265	826		621	•32€•
694	366 366 966	194		2.052 .742	1.264	1.527	116-	1.054	1.955	1.622	6154	2.863	1.0054	20012	. 266	.402	457	523	+15.	1,052	242	154.	1.527	928	1.0.1	1.915	1.547	2,370	1.653	2,165	2.475
<b>1</b> 6-	442° 442°	410 10 10 10 10 10 10 10 10 10 10 10 10 1	, 154 , 169	445°	.624	132	.570	- 19 - 19 - 19	505.	. 745	2002	1.) 70	100.	1.630	264	.309	, 339 650	966.	+14.	.564	0 8 9 ° ° ° ° ° ° ° ° ° ° ° ° ° ° ° ° ° °	.585	.762	649.	320.	196	.872	1.092		1.169	1.296
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				191			.125	521:				160	. 165	52.	9 - 2 - 4 	\$50°	ele.	ii:	110 <b>.</b>	571.		5210	.125	125						.198			052	.256	.250			125	.125	-125	521.	121	.125	.125							.25e	520	220		EIC.
AAA			12.000	10.000	16-000	12.000	18.000	16.950	1000 • 44		32.000	30.000	\$0°000	44°000		000.45	56.000	52.000	000*76			10.000	14-050	14.000				28° 600	24.000	26.005	32,000		32,000	36.000	42,000	44,000	40,000	000	6.000	8.000 .5	10,000	12.000	16.000	16.000	18.000	26.000	22.000	24.000	000-22	28.000 30.000	30.900	28.000	26.900	26.000	26.000
182				1961	201	265	.265	- 265			245	.328	.326	.326	100	.397	146.	69 • • • 0	166.		192.	292	.245	.265		1256	191	328	397	196.	26E"		531	531	162	165,		201	265	-265		841.	.326	-397	195.	197	694	699°	155.	.531 531	593.	4 <u>7</u> 9.	161.	.937	1.962
TAR.		216	592.	176	315	360	. 496	1465	600°	1000 ·	.690	e 788	• 416	1.061		1.361	1.532	1.636	195.1	- 22		.359	.593	644.				967	196.	1.017	1.100	987.1	1040 1040 1040	1.555	1.689	1.72	1,928	892.	.320	-372			654	.837			1.116	1.144	1.36		1.601	165.1	1.652	2.133	2.385
1.894		001	2.121	2.59.6	2.664	3.149	3.743	4.249	4.751		6.306	6.962	465.8	9.006	11.101	8448	12.375	14.508	000.01			2.142	2,565	2.671	10101			5.246	5,639	6.329	6.914 		10.212	10.930	12,233	12,939	14.224	1.067	1.326	1.617		2.682	3,196	3.891	4.246	5.283	5.830	6.251	A29.0	8.261 8.860	10.226	10.11	162.21	13.075 14.292	16.336
1.756		2.210	269.5	395.5	3.359	3.947	6.619	5.048	5.734	7.052	- 1.308-	6.114	9.575	10,409	12.755	14.103	14.790	16.528			2.007	2.928	3.507	304 °C			922.9	6.479	1.199	7.457	216.8	120.01	12.199	12.042	14,198	14.968	18.580	1.797	2.156	2.46]		3.695	4.286	5.140	126.6	6.808	7.484	7.855	0 + C + R	10.405	12.095	12.764	14.419 11	14.971	19.072
121-2		2.691	3.237	3.925	3.761	6.477	5.149	5.578	407.0	1.708	7.836	d. 770	162.01	12.715	1.4.4.0	14.897	15.58+	17.456		26112	2.709	3:450	4:037				7.020	7,135	7,993	8°451	001°A		11.011	13.904	15.260	10.010	19.892	2.199	2.686	2,991		135.4	4.942	466.4	050.7	7.602	8.422	d. 793		11.467	13.261	10.076	E10-01	24% - RT	21.196
1.666	1.260	1.500	2.000	546.5	2.500	3.000	3.500	4.000	4.500	5.563	6.000	6.625	000*6	0.000	10.750	12.000	12.750				1.500	2.000	2.375	2.500			4.500	5.000	5,563	0000			19.000	10.750	12.900	12.750		1.300	1.250	1.500	2.375	2.500	3.000	3-500	000.4	5.000	5.563	0000.9	C70+0	8.625	10.060	10.750		14.000	16.000
201.0	999.0	500.0	500.0	390.0	300.0	0.002	500.0	500.0	202	500.0	505.0	500.0	0.005	0.000	500-0	500.0	500.0				1000-0	1000-0	1000.0			1000.0	1000.0	1000.0	1001.0	1600.0				1000.0	1000.0			1506.0	1500.0		1500.0	1500.0	1500.0	1506.0	1500.0	1500.0	1500.0	1500.0		1500.0	1500.0	1500.0	0.0001	1500.0	1500.0

break all edges .005 max d not applicable to duminum seals;  $\begin{array}{c} \mathbf{v} \in \mathbf{$ TABLE C-4. SEAL, ALUMINUM, CRYOGENIC FLUID CONNECTOR Note: **√**35•±1 VIEW A , REF Ľ, w 1.000 1.000 1.000 1.000 1.000 1.000 1.000 1.000 1.000 1.000 1.000 1.000 1.000 1.000 1.000 1.000 1.000 1.000 1.000 1.000 1.000 1.000 1.000 1.000 1.000 1.000 1.000 1.000 1.000 1.000 1.000 1.000 1.000 1.000 1.000 1.000 1.000 1.000 1.000 1.000 1.000 1.000 1.000 1.000 1.000 1.000 1.000 1.000 1.000 1.000 1.000 1.000 1.000 1.000 1.000 1.000 1.000 1.000 1.000 1.000 1.000 1.000 1.000 1.000 1.000 1.000 1.000 1.000 1.000 1.000 1.000 1.000 1.000 1.000 1.000 1.000 1.000 1.000 1.000 1.000 1.000 1.000 1.000 1.000 1.000 1.000 1.000 1.000 1.000 1.000 1.000 1.000 1.000 1.000 1.000 1.000 1.000 1.000 1.000 1.000 1.000 1.000 1.000 1.000 1.000 1.000 1.000 1.000 1.000 1.000 1.000 1.000 1.000 1.000 1.000 1.000 1.000 1.000 1.000 1.000 1.000 1.000 1.000 1.000 1.000 1.000 1.000 1.000 1.000 1.000 1.000 1.000 1.000 1.000 1.000 1.000 1.000 1.000 1.000 1.000 1.000 1.000 1.000 1.000 1.000 1.000 1.000 1.000 1.000 1.000 1.000 1.000 1.000 1.000 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1500000 1 1500000 1500000 15000000 150000000000			. 550 . 794 . 7944. . 794 . 794 . 794 . 794 . 794 . 794 . 794 . 794 . 794 . 79	882 995 995 197 197 197 197 197 197 197 197 197 197	 563 646 646 750 750 750 750 750 750 815 750 1 256 1 264 1 264 1 264 1 264 1 264 1 264 1 264 1 264 264 1 264 264 264 264 264 264 264 264 264 264	5	6.420 6.427 6.447 6.447 6.447 6.447 6.447 6.447 6.447 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126 1.2.126	

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ALUMINUM, INTEGRAL/INTEGRAL AND LOOSE-RING/ LOOSE-RING CRYOGENIC FLUID CONNECTOR STUD. TABLE C-7.



200-1	1.090	.150	1.476	640.	114.	14.	9.000 P
					554 ·	114	8.000
1005	2.000		1.784		464		12.000
2005	č.375		2.213	640.		.616	10.040
5-0-5	305.2	0,, 7 •	1.444	.043	• 533	553.	16.000
5 u C • č	3.640	しぶく・	2.304	540.	•639	.634	12.000
500.0	3.506	042.	2.600	.043	-712	. 712	18.000
		057.	4.50%			971.	18.000
	5.091		590 - F		028.	62H.	26.000
500-1	5.543	Err.	3.512	640.	.956	454.	22.000
200	000	0-2-	342.5	.063	116.	116.	12.000
500.0	524.1	و ا ٦ .	3. H66	- U63	1-050	1.050	10.000
	2024 2010	• ·		0 4 4 9 4 1 4	1-162	1.102	
- 305	16.600		5.2.2			[] 4 - ]	000
500-1	1. 150	5/1.	5.154	649.	1.442	1.462	* 6.00
500.0	000-21		6,030	.063	1.603	1.603	54,040
500	15-151		6 - 2 - 9	1 4 3 <b>.</b>	1.655	1.655	56.000
1.004	1911-1-1-1-1-1-1-1-1-1-1-1-1-1-1-1-1-1-			690 . 690 .	102-1	104°I	000.52
	0.001						6.000
1000			1.0.1				600°
100-	244	061.	1.812	600.	-5-1		10.050
1-00-1	2.00U	.52.	162.5		<b>c</b> £ <b>9</b> .	574.	10.044
1.00.1	215.2	-55-		.00.	0.1.	171	1.000
				500.	1500	540.	
1.000.1					1-6-		16.000
1000	.0.0	- 1.		101			20.000
1000.		. 3/ -	3+1+1	640.	1.055	1.00	16.000
1 v0u	5.015	E ] 6 .	3.421	E 90 7	1.043	1.960	20.00
1000-	144.6	5.5.	1 - 1 - 1 - 1 - 1 - 1 - 1 - 1 - 1 - 1 -	107.		1.16.	
	1000	1.	5.105	690	144.1	146.1	10.00
1010.	1.6.1	17.7.	5.342	£ 90 °	505.1	1.582	3*.044
1000-0	00 <b>0</b> *01	000.	411.4	• • • •	1.405	1.505	12.003
1000	19.1.1	004.	1. <u>1</u> .	. U63	1.422	1.962	10,001 10,001
10001							000.54
	5		1.3.4	. 96.	245.2		000.04
1000	1.0.0	174.	6/1-41	640	2.044	2.44	
1500.0	1.000	0.50	020-1	.00.	454		8.009 6.000
1500.0	1.500	042.	611.5	. 063	-591	.591	000
1590.0	2.000	.250	996.	.063	. 659	.659	12.000
1500.0	2.375		2.859	690.	64 °	567	10.000
1500.0	3.000	ete.	3.216	E90.	683.	.883	16.000
1500.0	3.500	.375	3.613	.063	.997	497	16.000
1500.0	• • • • •	515°	4.032	690.	1.102	1.102	16.000
1560.0				590	1.197	1.197	26.000
1500.0	5.563	874	1946.4	. 063	1.346	1.346	22.000
1500.0	6.000	8E4.	5.076	.063	1.378	1.378	24.000
1500.0	6-625	.500	5.763	.00.	1.567	100.1	
1500.0	0000	005.	6.51K		1.862	1,862	30.000
1500.0	10.000	563.	7.857	.063	2.105	2,105	30.000
1500.0	054.01	1625	8,635	.063	2,365	2,365	20.000
1501.0	12.000	2051 2051	4°847	500°	1000 V	3.784	28,000 26,000
1500.0	14.000	.875	11.591	.063	3.117	3,117	26.000
1509.0	16.000	1.000	13.485	.063	3.621	3,621	26.000

TABLE C-8. FLANCE, INTEGRAL, CRES, CRYOGENIC FLUID CONNECTOR



¥	500		.563	592,	904.		. 667	.705	(c, '	.767	4001	ffa,	949.	196.	1 c 9 6 A	1.020	1.150	1.233	1.243	1.347	1.500	.500	-517	. 533	.567	2 <b>6</b> 2*	- 690 				167	.804	668,	.975	. 96 7	1.005	1.100	1.150	1.233	1.203	790°1	986.1
1	125	125	.125	.125	-125	.125	.125	.125						<b>9</b>	.250	. 259	. 250	.250	<b>.</b> ][]	616.	<b>.</b>	.125	,125	.125	.125	-125	282	621.	100		.189	108	.188	577"	.188	<b>65</b> 2 °	. 250	.250	.230			r[[.
z	1.034	1.544	2.054	2.517	2.564	3.064	3.691	4.097	4.681	5.087	5.617	6.097	6 <b>68</b> .9	6.107	8.909	10,182	11.037	32.203	13.002	14.192	16.202	1.034	1.294	1.544	2.054	2.50	2.564	9.000	169-2		5.007	5.007	6.697	6-89-9	8.142	606.8	10.147	11.02Y	12.193	13-002	:4°518	16.264
I	1.000	1.500	2.000	2.375	2.500	3.000	3.500	¢ • 0 0 9	<b>4.500</b>	5.000	5.563	6.030	6.625	8.000	8.625	10.000	16.750	12.000	12.750	14.00ô	16.000	1.600	1.253	1.560	2°000	2,375	2.200	000° M	3°560		5.000	1997 - 19 19	6.438	6.625	B.030	979.9	10.000	10.750	12.000	12.750	14.000	16.095
ני	6.000 6.000	9.000	A.000	10.007	10.000	10.000	10.000	10,000	14.000	12.000	20.000	14.000	26.000	1000.01	34.000	30.050	46.000	36.000	60 <b>.</b> 000	44.000	52.000	6.000	8.000	8.000	000-0	12.000	10.000	10.000	10.000		12.000	22.000	14.900	26.000	22.400	38.000	32.000	52.000	44.000	64.000	56.000	74.000
I	102.	-201	.201	.201	102.	.201	, 265	.265	.265	.265	.265	. 265	.265	,265	.265	.265	. 265	.265	.265	.265	。265	.201	.201	.201	.201	.201	.201	.201	592.			265	.265	.265	.265	.265	.265	.265	. 265	.295	.265	. 265
9	.325	204.	.467	.764	.523	.574	1.052	.142	1.200	1E8°	1.527	116.	1.671	1.054	1.915	1.602	2.370	1.757	2.863	1.899	160.5	.325	.366	.402	.467	.764	, 523 	• 21 •	1.052			1.527	116.	1.671	1.178	1.915	1.472	2.376	1.767	2,863	2.361	2,355
la.	,324	446.	.384	.540	+1+.	****	.699	.555	. 789	.615	579.	. 660	1.002	, 765	1.167	1.080	1,362	1.165	1,575	1.320	1.455	+2E,	956.	000°	\$6C°	.555	•2•		• 2 2 •			987	. 705	1.947	967.	1.212	1.109	1.437	1.307	1.650	1.529	1.718
لعنا	446.		1.944	2,245	2.444	2.944	3,334	3.930	455.4	059.4	5.345	5.930	6.407	7.930	e.407	4.870	10.482	11.670	i2.42n	13.870	15.870	**6.7	1.194	1 - 446	1.944	2.245	2.444	2.944	4 A A A A A A A A A A A A A A A A A A A				5.930	6.407	7.912	8.40T	068.5	10.482	11.869	12.420	13.647	15.825
4	1.070	1.6.1	2.144	2.463	2.665	7.197	.1.60 <sup>R</sup>	4.229	\$\$4.4	5.263	5.693	6.307	4.799	R.374	R.866	10.365	10.994	12.422	12.985	16.459	16.517	1.070	1.341	1:612	2 = 2 + 4	2,463	2.665	3.197		A 7 7 * *			6.307	6.799	8.352	6.866	10.397	10.994	12.422	12.985	14.447	16.472
U	1.364	1.918	2.462	2,790	2.995	9.539	3.962	4.595	5.030	5,653	4,097	6.721	7.228	R.636	9.143	10,475	11.522	12,980	13.561	15.075	17.171	1.364	1.641	1.918	2442	2.790	2,995	3,539	3.962			60.6	6.721	7.228	9.014	645.9	10.897	11.522	12.960	13,561	15.053	17,126
æ	1.171	400°2	2,870	9.199	3,405	3.450	A64.4	5.072	5.508	6.132	6,630	7.292	7.12	9.33 <sup>9</sup>	4.456	11.408	12.065	13.543	14,135	15.668	11.794	1.772	2.049	2,327	2,872	3.201	3.467	3,953				6.620	7.207	7.716	9.317	9.656	11.430	12.066	13.543	14,135	15.646	17.749
۲	2.173	9.129	3.272	1.661	3.807	4.352	4.964	5.602	AL0.0	6.662	7.160	7.732	8.242	9.809	10.366	11.938	12.546	14.073	14.605	10.148	18.324	2.174	2.451	2 c 7 2 9	3.274	3.603	3.809	55 <b>6 •</b> •	1/6**		1 - 2 - 0 2 - 5 - 5 - 5	7-150	7.737	9,246	9.847	396.386	11.940	12,596	14.073	14.665	16.176	16.279
PTVEX TUME	000 • 1	062.1	2.000	2.375	2.500	3.000	3.500	4.000	4.500	5.000	5,563	6.000	6.625	H.000	A.625	10.000	10.750	12.000	12.750	14.000	14.000	1.000	1.259	1.500	2.000	2.375	2.00	000.0	3.500			5.563	6.000	6.625	8.000	8.625	10.00	10.750	12.000	12.750	14.000	16.600
PRESSURE	100.0	0.001	0.001	100.0	100.0	200.0	100.0	100.0	100.0	100.0	100.0	100.0	100.0	100.0	100.0	100.0	100.0	100.0	100.0	190.0	100.0	200.0	260.9	200.0	200.0	200.0	200.0	200.0	200-0			200.0	200.0	20010	200.0	200.0	200.0	200.0	200.0	200.0	200.0	200.0

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444.	1.194	1.444	1.946	2.245	2.432	3334	3.891	4.334	4-864 7 315		6.407	7.782	8,329	9.728 1	0.420		3.619	5.564	446.	1.183	1.419	2.245	2,365	2.836	3.260			5.047	5.676	6.065	7.569	7.951 194-7	0.020	1.353	1.936		. 420	1.150	1.845	2.157	2.200	3.063	3.680	4.020	4.6U0	915-5	6-055	955.1	1.623		1 560.1	1.9/6.1	2.8/4	••/14
1.070	146-1	1.612	2.144	2+63	2.654	2.606	4.186	• • 652	5.196	2000	6.799	8.219	8.776	0.221	0.938	2,273 J	4.214	6.206	1.150	1.410	1.007	2.473	2.698	3.207	3.661	4.215		5.553	6.211	6.625	8°.48		0.748	2.130	2.737		1.118	459-1	×،133	<.+14 	150.5	14.6	+.10+	1/***	5.040 111	040.0	C20.0	114.1	d.516	4.418 5.42	1.409 1	2.620 1	3.400 1	1 148.0
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1.026	1.257	1.538	2.055	2.470	2.570	3.092	3.635	4,129	9-0-0 1 1 20	547.5	6-176	6.826	<b>6.230</b>	6.780	10.205	10.925	16.602	14.185	16.192	1.035	1.298	1.569	2.105			904 °C	4.232	4.607	5.185	5.682		9.137	8.774	10.124	10.921	12.151	14-176	16.203	1.042	élt.l	1.088	2 2 2 2 2 2 2 2 2 2 2 2 2 2 2 2 2 2 2	2+659	1.184	3.600			2.002	0.111	دد/ ۵۰	8.151	0.000	10.158	827-01	11.075	14.65	205.01
1.173	2.051	2.329	2.676	3.205	3.400	3.925	744.4	5.038			7.117	7.726	9.185	9.765	11.327	12.073	13.407	15.475	17.546	2.004	2.265	2.520				4.622	5.107	5.455	6.20 <b>8</b>	0%9* <b>9</b>	1.196	556.6	9.798	11.362	11.489	13.480		17.569	1.472	2.233		~	1/0.5	4.079			198.0	6.64 D	101-2	1.616	412.5	975.5	11.203	12-14	004-4	¢¢0.¢1	240.11
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	1.309	1°213		1.922	242 · I	1.973	2.458	941 °2	2.026	3.200	964.6	3.726	4.029	992 • •	5.119	5.008	6.57	1.476	1.603	15/01	2.083	40E 2	2.729	2°874	3.244	3.548	3.788	999°E	4.528	5,257	6.163	6.127	1.561	1.584	1.275	662.5	2.462	2.722	106°.5	3,623	3.678	112.4	4°708	5.066	5,920	121.1	6.743	7.968 8.568	7 - <b>7</b> - <b>7</b>			
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INTEGRAL/INTEGRAL AND LOOSE-RING/ CRES, STUD, TABLE C-14.



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\$.000 <b>6.</b> 000	<b>8.</b> 538	12.400	14,000	1 + . 0 0 0	12.000	18.000	26.000	24.000		44.000	000.04	0.0.0	56,090	66.000	10,000 6.60		10.000	14.966	12.010	14.688	20.000	24.063	25.000	24.000	30.060 30.060	000 · 6 •	42.009 42.009	52.000	52.000	000,02	56.000	8.00U	10-000	10.000	000°47	10.000	20.000	24.04U	¢9.64V	100°07	24.040	94.000		*** 044	36.00V	(110.04	22.000	C	Ż	Ъ	32	Ę	ļ
••75	497	249.	.601	(18.		116.	1.050	1-025	1.227	- 597			1.613	1.857	000-2		505	.657	2120	964	• <b>•</b> •	1.010	1.107	1.258	141.1	1.476	1.522	1,712	1.934	2.123	2.498	145.		197	1 <b>5</b> 8.	) ** • * *	1.030	541.I	642.1		\$12.1	101	246-1	540.2	2-200	2.5566	2.625						
-+51 	104.	543.	109.	500°	4 . H	116.	1.050	1.025	1.277	1.297	1.474		1.613	1-457	0C0*7		505	.657	717.	490	626	501.1	1.107	1.258	1.144	1.470	1.522	1.712	1.934	2.123	2,498	1.5.	140.	- 780			1.030	541-1	1.233	1-363	464.1	1.709	1.4.2	2.045	215.2	4.565	2-8-5						
E90,	6963	.063	. 663	.00.	E 40 .	. 663	.063	690 -	. 06.	.063	. 063	500.	.063	E90.	200.		.063	690 -		690 -			.063	590.	190°		690.		663		.043	590.			E90.		E00.		.063	500.	. 063	690°	[97.	. úč.j	103.		690.						
1.016	861°.	2.377	2.216	3.075	3.022	595.5 574.5	3. 451	3-851 4 247		926.4	5.582	5,301	6.140	7.115	1 . 00 . 1	1.992	2,137	864.5	2.8630 2.868	3.206	3-594	4.162	4.178	4.719	5.053	5.569	5°775 4-384	6.535	7. 359	0-116	9.193	2.000		4.304	5 <b>51.</b> 5	3.5.6	3.870	462.4 494.4	*. d]H	157.5	00/00	5.05d	+55 - 1	4.00/	00/ 90	1000	10.601						
•190	067.	-190	.190	052.	1250	052.	-256	. 250	.250	-250			EIE.		. 1 90 . 1 90	196	.190	99 .	.250	•52•	-250	50 50 50 50	.256			616.	616. 616.		. 375 . 175	54.0.	904.	0 <b>h</b>   • 1	0 Å ] •	500	202	000	107.		[[[.		516.	<u>.</u>	516.	516.		006.							



## TABLE C-15. INSTALLATION OF AFRPL FLANGED CONNECTORS FOR CRYOGENIC SERVICE

	Pipe/Tube	Wrench Tor minum Con Bolt -	que for Alu- nectors-Per Lb-In.	Wrench Torque for Stain- less Steel Connectors- Per Bolt ~ Lb-In.						
Pressure	OD	Minimum	Maximum	Minimum	Maximum					
100	1.0	12.4	13.0	21.5	23.6					
100	1 25	12 4	13.0	20.0	22.0					
	1.65	14.7	15.4	23.8	26.2					
	2 0	16.4	17.2	32.0	35.2					
	2 375	22.9	24.0	46.2	50.6					
	2,5(5	17 7	18.6	32.0	35.2					
	3.0	21 4	22 5	38.8	42.6					
	1.5	60.7	65 0	108.1	118.0					
	4.0	36 4	38 5	78.2	85.6					
	4.0	53 6	56.0	101.3	111.0					
	5.0	41 6	43.7	83.1	91.0					
	5 563	54 3	57.2	105.7	116.0					
	6.00	46.3	48.5	87.2	95.6					
	6,625	53 6	56.5	98.9	108.6					
	8.0	59.6	62.8	106.9	117.5					
	8 625	53 4	56.0	102.0	112.0					
	10.00	54.7	57.2	102.0	112.0					
	10.75	60 5	63.0	101.5	111.5					
	12 0	54 6	57.0	105.0	115.5					
	12.0	104.9	110.0	98.7	118.5					
	14.0	54.0	56.6	103.0	113.0					
	14.0	55.3	58.4	103.9	114 0					
200	1.0	13.0	14.0	22.2	24.4					
200	1 25	13.0	14.0	20. 0	23.9					
	1.5	15.8	17.0	25.2	27.7					
	2 0	17.7	19.0	34.0	37.4					
	2 375	24.3	25.5	40.4	44.5					
	2.5	19.6	21.0	34.6	38.0					
	3.0	23.9	25.0	42.4	46.7					
	3.5	55.8	59.0	113.6	124.0					
	4.0	47.3	50.0	86.4	95.0					
	4 5	58.6	62,0	108.6	118,0					
	5 0	54.9	58.0	93.6	103,0					
	5. 563	55.2	58.6	103.0	113,0					
	6.00	56.0	59.0	101.2	111.5					
	6 625	53.2	56,0	99.3	109.0					
	8.0	55.3	58.6	101.7	111.0					
	8, 625	58, 1	62.0	100.8	110,0					
	10.00	55,8	59.0	102.5	112.0					
	10.75	102.0	108.0	100.0	110.0					
	12 0	57.1	60.0	102.8	112.5					
	12.75	105,6	110.0	104.3	115,0					
	14.0	106,6	111.0	102.5	112,0					
	16.0	105.0	110.0	101.5	111.5					

		Wrench Tos minum Con	que for Alu- nectors-Per	Wrench Torque for Stain less Steel Connectors- Per Bolt - Lb-In							
	Pipe/Tube	Bolt -	Lb-In.	Per Bolt	lt – Lb-in.						
Pressure	OD	Minimum	Maximum	Minimum	Maximum						
500	1.0	14.8	16.0	25.8	28.4						
500	1.25	18.9	20.6	32.2	35.4						
	1.5	18.5	20.0	29.0	31.8						
	2.0	22. 3	23.6	40.6	44.6						
	2. 375	53.0	55.6	45.6	50.0						
	2.5	23.0	25.0	44.9	49.4						
	3.0	53.6	56. 2	42.9	47.2						
	3.5	52.9	55.5	94, 2	104.0						
	4.0	56.4	59.2	108.2	119.0						
	4.5	55.3	58.0	100.0	110.0						
	5.0	53.2	55.7	95.2	104.0						
	5, 563	111.0	117.9	95.8	105.0						
	6.0	63.5	67.0	101.0	111.0						
	6 625	104 4	109 0	99. A	110.0						
	8.0	106.0	111 0	109.2	120.5						
	8 625	115 6	121 0	109.2	121 0						
	10.0	191 0	200 0	197.0	210.0						
	10.0	191.0	200.0	194 8	214 0						
	10.75	170.0	200.0	194.9	214 0						
	12.0	197.7	207.0	204 5	274 0						
	14.75	220,0	238.0	107 4	217 0						
	14.0	310.9	332. U	212 6	232 0						
1000	16.0	104.0	165.U	20.9	47 6						
1000	1.0	23. Y	25.0	39.0	47.4						
	1.25	24.5	20.0	38.5	42.4						
	1.5	25.4	27.0	38.8	42.1						
	2.0	51.8	54.5	40.8	45.U						
	2, 375	54.7	57.5	45.5	50.0						
	2.5	52.0	55,4	91.0	100.0						
	3.0	55.4	58.4	101.3	111.0						
	3.5	105.8	111.0	98.0	108.0						
	4.0	102.2	107.0	102.8	113.0						
	4.5	205.5	215.0	108.2	119.0						
	5.0	107.5	112.5	109.4	120.1						
	5.563	202.5	212.0	182.8	200.0						
	6.0	194.2	204.0	181.2	199.0						
	6.625	194.5	204.0	186.7	205.0						
	8.0	306.7	322.0	194.4	213.0						
	8.625	322.5	338.0	199.7	220.0						
	10.0	484.0	510.0	225.9	247.0						
	10.75	502.0	520.0	228.7	251.0						
	12.0	511.0	530.0	348. 6	383.0						
	12.75	550.0	580.0	360.3	396.0						
	14.0	800.0	840.0	390.9	429.0						
	16.0	1097.0	1140.0	608.4	668.0						
1500	1.0	21.8	23.0	37.2	40.9						
	1.25	53, 5	56, 5	40,0	44.0						
	1.5	52.6	55.0	42.0	46, 2						
	2.0	55.4	58.0	104.5	114.5						
	2.375	111.0	117.0	97.0	106,8						
	2.5	100.5	105.0	91.0	100.0						
	3.0	102.0	107.0	106.2	117.0						
	3.5	193.0	203.0	112.6	123.0						
	4.0	182.0	191.0	186.8	205.0						
	4,5	199.0	210.0	177.0	190.0						
	5.0	188.0	197.0	192.6	210.0						
	5,563	311.0	327.0	203, 0	223.0						
	6.0	318.0	332.0	214.7	236.0						
	6.625	501.0	525.0	333, 4	368.0						
	8.0	510.0	535.0	359.0	395.0						
	8,625	575.0	610.0	378, 8	416.0						
	10.0	830.0	870.0	413, 5	454.0						
	10.75	1180.0	1240, 0	443,8	487.0						
	12.0	1860. 0	1960.0	883, 4	970.0						
	12,75	1997.0	2080.0	706.0	770.0						
	14.0	2962.0	3000.0	959.4	1051.0						
		443.0	4600 0	071 0	1190 0						

TABLE C-15. (Continued)

## TABLE C-16. INSTALLATION OF AFRPL CRYOGENIC FLANGED CONNECTORS





A connector may consist of two integral flanges, two loose-ring flanges, or one flange of each type, as shown. Weld the integral flange directly to the tubing. For a loose-ring flange first place the loose ring on the tubing. Be sure the radius of the loose ring faces the flange. Weld the loose-ring flange to the tubing.





Remove the dust caps. Inspect Surfaces A, B, C, D, and E for dirt. contamination, or surface irregularities. If necessary, rinse the surfaces with methylethylketone and dry parts with a filtered air stream or wipe parts with clean paper tissue moistened with methylethylketone. Replace the dust caps.



Step 3

Coat the threads of the stude and bolts and the bearing surfaces of the nuts and bolt heads with lubricant. Apply the lubricant sparingly. Work the nuts on the male threads by hand. Remove the nuts and inspect the thread surfaces for an even coating of lubricant.



Step 4

Remove the seal from the protective wrapping. Inspect Surfacer A of the seal disks with a 5-power lens. Discard the seal if the surfaces are scratched or irregular. Clean the surfaces with methylethylketone if dirty.





Step 5

Remove the dust caps from the flanges and locate the seal in the flange cavities. Care must be exercized to avoid nicking the edges of the seal. Handling with clean hands is permissible, but the sealing surfaces must not be contaminated with dirt or lubricant.



## Step 6

Tighten the nuts of the studs or bolts fingertight while maintaining an axial force on the flange to keep the seal in place. Tighten the nuts with a calibrated torque wrench to the recommended torque. The nuts should be tightened in sequence with a maximum of 1/2 turn per nut. Several tightening cycles are required to seat the seal and preload the connector.

C - 36

Unclassified				
Security Classification				
	DOCUMENT CONTI	ROL DATA - R	L D	
(Security classification of title, bod	ty of abstract and indexing a	nnotetion must be e	ntered when t	he overall report is classified)
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Battelle Memorial Institute				
505 King Avenue Columbus Obic	49201		au. Under	
REPORT TITLE		·····		
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Trainer, Thomas M.	Thompson, John R.			
Baum, Joseph V.	Ghadiali, Nuruddin	D.		
REPORT DATE		74. TOTAL NO. OI	PAGES	75. NO. OF REFS
July 1969		311		22
. CONTRACT OR GRANT NO.		Se. ORIGINATOR	REPORT NU	IMBER(\$)
AF 04(611)-11204		AFRPL-T	<b>l- 69</b> - 97	
5, PROJECT NO.				
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ABSTRACT				
A 3-year program was conduct "Bobbin" seal concept for both 6061- support was furnished during the evalu- activities in relation to the first object of joining tubing from 1 to 16 inchest typical flanged connector sizes, (3) to connectors incorporating Bobbin seals connectors, and (5) the preparation of activities directed toward the second and evaluation of connector modifical increased radial seal loading technique to a summary of the technical activities designs for aluminum and stainless stel it is recommended that MS standards nectors for tubing through 3 in. in dia connectors, and in regard to certain a	ed to develop a family of T6 aluminum and Type uation of AFRPL stainless ctive consisted of: (1) the in diameter, (2) the develop the preparation of a deta of designs for a family of objective consisted of: ations to achieve seal return ues, and (4) the investig ties, the report contains eel flanged connectors for and specifications be pro- ameter. Additional develop aspects of the stainless st	of flight-weight f 347 stainless stee s steel threaded of he selection of bo velopment of stai iled computer pr ation, and qualifi stainless steel ar (1) the investiga ention and misal ation of stress rel the computer des or pressures up to epared for cryoge elopmental work isel threaded con	langed tube el tubing sys onnectors b olted, flang nless steel a ogram for th cation testin id aluminum tion of seal- ignment lim axation in t sign program 1500 psi an nic stainles is recomme nectors.	connectors which utilize the stems. In addition, engineering y selected organizations. The ed connectors as the best means and aluminum Bobbin seals for ne optimum design of flanged ing of representative flanged in flanged connectors. The removal tools, (2) the design initation, (3) the investigation of hreaded connectors. In addition in for flanged connectors, and d for tube sizes through 16 inche is steel and aluminum flanged co- ended in regard to larger flanged
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Security Classification

## Unclassified Security Classification LINK B LINK C 14. LINK A KEY WORDS ROLE WT ROLE WT ROLE WT Flanged Connectors Threaded Connectors Metal Seals **Bobbin Seals** Thermal Gradients Fatigue Testing AFRPL Connectors Nickel Plating Stress Relaxation Helium Leakage Cryogenic Testing

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