NASA CONTRACTOR REPORT

NASA CR - 61005

NASA CR - 61005

FACILITY FORM 602

N65-28847
(ACCESSION NUMBER)

(THRU)

393
(PAGES)

1
(CODE)

(NASA CR OR TMX OR AD NUMBER)

31
(CATEGORY)

A STUDY OF DESIGN GUIDELINES FOR STERILIZATION OF SPACECRAFT STRUCTURES

FINAL REPORT

GPO PRICE $ _____

CFSTI PRICE(S) $ _____

Hard copy (HC) 7.00

Microfiche (MF) 2.00

ff 653 July 65

Prepared under Contract No. NAS 8-11107

GENERAL ELECTRIC

Spacecraft Department
Valley Forge Space Technology Center
Philadelphia 1, Penna.

For

NASA GEORGE C. MARSHALL SPACE FLIGHT CENTER
Huntsville, Alabama

August 1964

Sgt 29760

PREFACE

This is the Final Report of an investigation performed by the Spacecraft Department of the General Electric Company, Valley Forge, Pennsylvania, for the George C. Marshall Space Flight Center of the National Aeronautics and Space Administration, Huntsville, Alabama, under Contract No. NAS 8-11107. Mr. R. Crawford, Structures Research Section, Propulsion and Vehicle Engineering Laboratory, was the project engineer. The contract was initiated on 20 September 1963 with a six-months period of performance.

A STUDY OF DESIGN GUIDELINES FOR STERILIZATION OF SPACECRAFT STRUCTURES

FINAL REPORT

Prepared under Contract No. NAS8-11107 by

GENERAL ELECTRIC

Spacecraft Department

Valley Forge Space Technology Center

Philadelphia 1, Pennsylvania

This report is reproduced photographically
from copy supplied by the contractor.

NASA-GEORGE C. MARSHALL SPACE FLIGHT CENTER

TABLE OF CONTENTS

Section		Page
1	INTRODUCTION	1-1
2	SUMMARY	2-1
3	CONCLUSIONS	3-1
4	RECOMMENDATIONS	4-1
5	TYPICAL PLANETARY LANDER IN STERILIZING ENVIRONMENT	5-1
	5.1 Model Description	5-1
	5.2 Thermal Analysis	5-2
	5.2.1 Summary	5-2
	5.2.2 Analysis	5-6
	5.2.3 Results	5-9
	5.2.4 Conclusions	5-11
	5.3 Structural Analysis	5-26
	5.3.1 Entry Shield	5-27
	5.3.2 Beam Elements	5-30
	5.3.3 Compression Members	5-33
	5.3.4 Ring-Stiffened Shells	5-36
	5.3.5 Tanks and Pressure Vessels	5-37
	5.3.6 Alignment Sensitive Component Mounting	5-45
	5.3.7 Plates and Panels	5-46
	5.3.8 Explosively Actuated Fasteners	5-47
	5.3.9 Relaxation of Highly Stressed Components	5-50
	5.3.10 Thermal Distortion	5-53
	5.4 Design Considerations - Containing Canister	5-55
	5.4.1 General	5-55
	5.4.2 Storage and Handling	5-55
	5.4.3 Container Weight Penalties	5-59
	5.4.4 Heat Transfer Relationships	5-61
	5.4.5 Inflight Separation Problems	5-61
	5.4.6 Techniques for Maintaining Positive Overpressure	5-61
	5.4.7 Flexible Containers	5-62
	5.5 References	5-63

TABLE OF CONTENTS (Continued)

Section		Page
6	JOINT STUDY	6- 1
	6.1 Introduction	6- 1
	6.2 Discussion	6- 1
	6.2.1 General	6- 2
	6.2.2 Manufacturing Characteristics - Structural Joints	6- 2
	6.2.3 Biophysical Characteristics - Structural Joints	6- 5
	6.2.4 Reliability Characteristics - Structural Joints	6- 6
	6.2.5 Thermal Characteristics - Structural Joints	6-10
	6.2.6 Weight Characteristics - Structural Joints	6-11
	6.3 Summary	6-14
	6.4 Bolted Joints	6-14
	6.4.1 Design Characteristics	6-14
	6.4.2 Strength Characteristics	6-17
	6.5 Riveted Joints	6-23
	6.5.1 Design Characteristics	6-23
	6.5.2 Strength Characteristics	6-28
	6.6 Welded Joints	6-29
	6.6.1 Design Characteristics	6-29
	6.6.2 Strength Characteristics	6-31
	6.7 Brazed Joints	6-33
	6.7.1 Design Characteristics	6-33
	6.7.2 Strength Characteristics	6-35
	6.8 Bonded Joints	6-36
	6.8.1 Design Characteristics	6-36
	6.8.2 Strength Characteristics	6-41
	6.9 Miscellaneous Joints	6-46
	6.9.1 Design Characteristics	6-46
	6.9.2 Strength Characteristics	6-46
	6.10 Technique for Improving Joint Conductance	6-47
	6.10.1 General	6-47
	6.10.2 Joints in Vacuum	6-47
	6.10.3 Joints in Air and Other Gases	6-50
	6.10.4 Joints with Metallic Fillers	6-51
	6.10.5 Joints with Non-Metallic Fillers	6-51
	6.10.6 Summary	6-54
	6.11 References	6-54
7	MATERIALS	7- 1
	7.1 Introduction	7- 1
	7.2 Summary	7- 1

TABLE OF CONTENTS (Continued)

Section		Page
	7.2.1 General Data	7- 1
	7.2.2 Metals	7- 2
	7.2.3 Structural Adhesives	7- 2
	7.2.4 Structural Composites	7- 3
	7.2.5 Ablating Materials	7- 3
	7.3 Conclusions	7- 3
8	MANUFACTURING CONSIDERATIONS	8- 1
	8.1 General	8- 1
	8.2 Conclusions	8- 2
9	GENERAL DESIGN GUIDELINES	9- 1
	9.1 Factors Effecting Ease of Sterilization	9- 1
	9.1.1 General	9- 1
	9.1.2 Geometric Complexity	9- 1
	9.1.3 Surface Finish	9- 1
	9.2 Design Guidelines	9- 9
	9.2.1 General	9- 9
	9.2.2 Design and Analysis	9-10
	9.2.3 Joining Techniques	9-14
	9.2.4 Structural Tooling Jigs and Fixtures	9-17
	9.2.5 AGE Equipment	9-18
	9.2.6 Approved Materials List	9-18
	9.3 References	9-19
10	APPENDIX	
	A. Recommendations for Future Work	A- 1
	B. Entry Shield Analysis	B- 1
	C. Beam Analysis	C- 1
	D. Compression Members	D- 1
	E. Ring-Stiffened Shells	E- 1
	F. Plates and Panels	F- 1
	G. Design Considerations for a Biological Barrier	G- 1
	H. Canister Analysis	H- 1
	J. Temperature Distribution in Connectors and Bolted Joints	J- 1
	K. Interface Phenomena for Metals in Contact Under Normal Stress	K- 1
	L. Clean Rooms and Clean Work Stations	L- 1
	M. Functional Flow Diagram of Voyager Sterilization Process	M- 1

TABLE OF CONTENTS (Continued)

Section		Page
N.	Radiation Sterilization of Structures and Components	N- 1
O.	Ultrasonic Sterilization	O- 1
P.	Typical Test Program, Effect of Surface Finish on Ease of Sterilization	P- 1
Q.	Structural Materials	Q- 1

LIST OF ILLUSTRATIONS

Figure		Page
5.1-1	Typical Planetary Lander	5- 3
5.2-1	Maximum Temperature Differential versus Time - Aluminum Vehicle	5- 7
5.2-2	Maximum Temperature Differential versus Titanium Vehicle	5- 8
5.2-3	Temperature Data - Typical Nodes - Run 3, Aluminum Vehicle	5-10
5.2-4	Temperature Data - Typical Nodes - Run 4, Aluminum Vehicle	5-13
5.2-5	Temperature Data - Typical Nodes - Run 6, Aluminum Vehicle	5-14
5.2-6	Temperature Data - Typical Nodes - Run 10, Aluminum Vehicle	5-15
5.2-7	Temperature Data - Typical Nodes - Run 12, Aluminum Vehicle	5-16
5.2-8	Temperature Data - Typical Nodes - Run 13, Titanium Vehicle	5-17
5.2-9	Temperature Data - Typical Nodes - Run 14, Titanium Vehicle	5-18
5.2-10	Temperature Data - Typical Nodes - Run 15, Titanium Vehicle	5-19
5.2-11	Temperature Data - Typical Nodes - Run 16, Titanium Vehicle	5-20
5.2-12	Temperature Data - Typical Nodes - Run 17, Titanium Vehicle	5-21
5.2-13	Temperature Data - Typical Nodes - Run 18, Titanium Vehicle	5-22
5.2-14	Contact Profile Between a Typical Component and Its Mounting Plate	5-24
5.2-15	Effect of Gas Pressure on Thermal Conductivity of Super Insulation	5-26
5.3-1	Typical Lander Cone Structures from General Electric Voyager Study (NAS-W696)	5-28
5.3-2	Stresses in Beams Due to Depthwise Gradients	5-31
5.3-3	Typical Interaction Curve for I-Beam	5-33
5.3-4	Thermally Induced Discontinuity Stresses for Ring-Stiffened Shells of Various Radii	5-37

LIST OF ILLUSTRATIONS (Continued)

Figure		Page
5.3-5	Effects of Temperature on Tensile Yield Strength of Typical Materials	5-40
5.3-6	Spherical Tank Subjected to Internal Pressure - Required Wall Thickness vs. Pressure and Radius Parameters	5-43
5.3-7	Spherical Tank Subjected to Internal Pressure - Radial Displacement vs. Pressure, Radius and Thickness Parameters	5-43
5.3-8	Unrestrained Spherical Tank Subjected to Temperature Rise of 230°F - Radial Displacement vs. Tank Radius	5-44
5.3-9	Spherical Tank Walls Subjected to Radial Temperature Gradient - Maximum Bending Stress vs. Linear Thermal Gradient	5-44
5.3-10	Simplified Typical Explosive Nut Installation	5-48
5.3-11	Typical Shaped Charge Installation	5-49
5.3-12	Schematic of Test Fixture	5-51
5.3-13	Thickness Change vs. Loading Cycle	5-52
5.4-1	Comparison of Typical Advanced Mariner and Voyager Configurations Showing Relationship Between Lander, Orbiter and Canister	5-56
5.4-2	Typical Containing Canister for Planetary-Lander Idealized for Preliminary Analysis	5-57
6.2-1	Summary Sheet, Structural Joints	6- 3
6.4-1	Axial Bolt Load due to Temperature Changes	6-19
6.4-2	Axial Bolt Load vs. Temperature Changes	6-19
6.4-3	Axial Bolt Load vs. Temperature Changes (1/4-in. Steel bolt)	6-20
6.4-4	Axial Bolt Load vs. Temperature Changes (5/16-in. Steel bolt)	6-20
6.4-5	Bolt Load Factor vs. Clamped Material Thickness	6-21
6.4-6	Bolt SHEAR Loads as a Fraction R_i of Total Load P	6-21
6.5-1	Typical Riveted Panel in Manufacturing Jig	6-24
6.5-2	Typical Riveted Panel in Manufacturing Fixture	6-25
6.7-1	Proposed Gemini Adapter	6-35
6.8-1	Time-Temperature Relationship of Typical Adhesives	6-37
6.8-2	Typical Process Sheet - Honeycomb Sandwich Panel	6-39
6.8-3	Typical Biological Load Reduction During Fabrication	6-40
6.8-4	Comparison of Adhesive Specifications	6-42
6.8-5	Qualified Adhesives (Per QPL)	6-43
6.8-6	Effects of Service Temperature and Time on Typical Epoxy	6-44

LIST OF ILLUSTRATIONS (Continued)

Figure		Page
6.10-1	Contact Conductance vs. Contact Pressure for Bare Metal Joints	6-47
6.10-2	Contact Conductance vs. Contact Pressure for Gold, Silver, and Copper Joints in Vacuum	6-47
6.10-3	Thermal Conductance vs. Interface Contact Pressure (Aluminum)	6-48
6.10-4	Thermal Conductance vs. Interface Contact Pressure (Magnesium)	6-48
6.10-5	Idealized Representation of a Bolted Joint with an Annuler Load	6-49
6.10-6	Long Straight Beam Ends Two Equal Oppositely Directed Normal Forces at Corresponsing Points on Opposite Surfaces	6-49
6.10-7	Thermal Conductance of Single Bolt Joint	6-49
6.10-8	Contact Conductance vs. Contact Pressure for Aluminum Joint for Difference Roughness Values	6-50
6.10-9	Predicted and Experimental Conductance vs. Pressure for Stainless-Steel Contact	6-51
6.10-10	Thermal Contact Conductance vs. Contact Pressure	6-52
6.10-11	Thermal Resistance of Metallic Interface with Various Filler Materials	6-53
6.10-12	Thermal Contact Conductance vs. Contact Pressure	6-53
6.10-13	Influence of a Conducting Fluid on Thermal Conductance vs. Contact Pressure	6-54
9.1-1	Survivor Curves for Micro-Organisms, Population versus Exposure Time	9- 3
9.1-2	Typical Curve Showing Resistance of Spores to Destruction by Dry Heat	9- 3
9.1-3	Variation in F_{257^oF} Values for Different Surfaces	9- 5
9.1-4	Variation in D Values for Different Surfaces	9- 6
9.1-5	Surface Finishes by Common Production Methods	9- 7
9.1-6	Possible Variations in Surfaces Having Equal Surface Finish Values	9- 7
9.1-7	Relative Size of Several Organisms	9- 8
9.1-8	Relationships Between a Typical Surface and Typical Organism	9- 8
9.2-1	Riveted Connection — Typical Manufacturing Sequence	9-15

LIST OF TABLES

Table		Page
5.2-1	Sterilization Time	5- 5
5.2-2	Summary of Thermal Analog Run Conditions	5-12
5.2-3	Reduction in Thermal Conductivity Caused by Evacuating the Air from Various Materials* ($100^{o}F$ Mean Temperature)	5-25
5.3-1		5-32
5.3-2	Critical Temperature Gradients Between Columns and Restraining Structure	5-35
5.3-3	Possible Pressurizing Systems and Instruments for Planetary Landers	5-38
5.3-4		5-53
5.4-1	Comparison of Handling Procedures	5-59
6.2-1	Ratings of Joining Techniques, Structural Joint Study	6- 5
6.2-2	Joint Reliability Evaluation - Structural	6- 6
6.5-1	Values of L/t, F_{cy}, E and α Critical Thermal Gradients	6-29
6.6-1	Tensile Strength Comparison (Reference 6.11)	6-32
6.8-1	Suggested Sandwich Construction for Elevated Temperatures	6-45
6.10-1	Technique for Improving Thermal Conductance Across Structural Joints	6-55

1. INTRODUCTION

Terminal heat sterilization has been recommended for the first generation of unmanned space vehicles designed to impact on planetary surfaces. The reasons for this recommendation are numerous. The possible existence of extra-terrestrial life is widely accepted. The introduction onto other planets of organisms which could upset a balanced ecology is felt by many scientists to be undesirable. In addition to moral and philosophical objections to contamination, practical scientific objections have been raised. Introduction of organisms may interfere with subsequent biological observations. Observations which might shed light on the problem of the origin of life are jeopardized if contaminated space vehicles are impacted on planets where they might find favorable environments.

The recent literature on the general problem of spacecraft sterilization is fairly extensive. Sufficient work has been performed to enable many sterilizing techniques to be considered and rejected. Techniques such as radiation, extreme cold, and ultrasonics have been considered. Chemical sterilants have also been extensively studied. Their use on the Ranger program is well documented. For planetary landers however, the use of the most reliable technique, heat sterilization, has been prescribed in the Sterilization Guidelines established by the NASA Office of Space Sciences.

Because of uncertainties regarding actual requirements, a conservative sterilization environment was specified for this study. The environment consisted of a minimum of four cycles in which an equilibrium temperature of 150°C is maintained for 60 hours of each cycle. This study was concerned with the response of vehicle structures to the sterilizing environment, and with design principles and procedures required to make the vehicle withstand this environment. It is a part of a more general study on the behavior of complete vehicles, including electrical equipment and electronic components, subjected to terminal thermal sterilization.

2. SUMMARY

The purpose of this study program was to investigate the effects of terminal sterilization on spacecraft and planetary lander structures and to establish guidelines for the structural design of these vehicles when subjected to thermal sterilization.

The environment resulting from terminal sterilization creates two major problems: transient thermal gradients and material degradation due to elevated temperatures. A simplified model of a typical planetary lander was selected to represent a range of vehicles. This model was used to calculate relative magnitudes of thermal gradients, material exposure times, and typical structural elements. The typical lander was subjected to a spectrum of thermal environments. Although sterilization in a vacuum had been originally specified for this study, thermal analog analyses were also performed for several cases in which a conducting atmosphere was introduced into the containing canister. This permitted the evaluation of vacuum sterilization in comparison to sterilization in a conducting medium.

Using the thermal gradients obtained from the typical lander configuration, it was possible to evaluate the structural response of a number of typical structural elements. The elements selected are representative of the major problem areas on any lander. Analyses were performed in sufficient depth to establish the magnitude of the stress and deflection problem and to formulate general guidelines for future designs. To evaluate the relationship between a vehicle and its containing canister a simplified model of a containing canister was employed. Design problems related to this canister were evaluated and guidelines were prepared.

The problems arising from the sterilization of structural joints were evaluated. Since joints represent potential traps for contaminants containing biological life, a survey of joint types and joining techniques was performed. Typical configurations for Mariner and Voyager vehicles were studied, and a summary tabulation of typical joints was prepared. Joints were grouped by types; i.e., riveted, bolted, bonded, etc., and qualitative ratings of each joining technique were made.

Manufacturing problems associated with structural fabrication and assembly were evaluated. The total exposure time resulting from four sterilization cycles is sufficiently long to cause many materials to degrade. Using the results of the thermal analysis, a survey of materials problems directly attributable to sterilization was made. Tentative general guidelines covering design, analysis and assembly of planetary landers, and their containing canisters were formulated.

3. CONCLUSIONS

The general design guidelines which incorporate the results of this study are given in 9.2.

1. The thermal environment specified for sterilization is a significant design condition for material selection and for the analysis of many structural elements.

2. For any specific vehicle both thermal analysis and thermal testing are desirable in establishing thermal gradients and exposure times.

3. The introduction of a sterile atmosphere into the containing canister is preferable to sterilization in a vacuum. The advantages gained include:

   - Shorter thermal rise times,
   - The elimination of a 14.7 psi overpressure design requirement on the containing canister,
   - The elimination of an atmospheric contamination hazard caused by leaks at the interface between mating sections.

4. Thermal rise times can be reduced by altering joint conductance values. This will be beneficial for some joints in a space vehicle.

5. The design of the containing canister is of major importance in maintaining sterility of the planetary lander. A rigid canister is preferable to a flexible canister.

6. No joints or joining techniques can be rejected solely on the basis of incompatibility with terminal sterilization. Every type of joining technique can be made acceptable from a sterility standpoint by careful planning and controls.

7. Terminal sterilization alone is not sufficient to reduce biological levels to desired values without the use of intermediate cleaning or sterilizing procedures. Periodic biological load reduction is required during fabrication and assembly. This implies separate cleaning operations during the entire assembly sequence. The entire assembly sequence of any configuration must therefore be carefully examined for compatibility with this requirement.

8. Although terminal sterilization is a limiting design condition for some structural elements, existing analytical techniques are

adequate to evaluate this environment, and no gaps in analytical capability are foreseen. Specific configurations require extensive analysis to establish performance in the sterilizing environment.

9. Design for terminal sterilization imposes a weight penalty for some structural elements. These include tanks and pressure vessels, explosively actuated devices, and adhesive bond thickness in multi-layered structures.

10. Many structural materials experience significant irreversible degradation during terminal sterilization. In general, however, the selection of materials will not be a significant obstacle to sterilization. The environment, though severe, is not considered limiting from a materials standpoint. For some materials, additional testing is required.

4. RECOMMENDATIONS

1. For spacecraft structures subjected to terminal heat sterilization, further study of thermal stresses and deflections is recommended in these areas:

   - Multi-layered shells
   - Ring-stiffened shells
   - Space frames
   - Tanks and pressure vessels
   - Non-structural elements such as explosive devices, harnesses and cables including clamps, insulation blankets, waveguides, and pneumatic or fluid plumbing lines.

2. Guidelines for the design of these structural elements should be collected in a design handbook.

3. Where specific vehicle elements or configurations are known, testing for structural response to the thermal environment should be performed.

4. Additional design work should be performed on containing canisters and problems related to their use (e.g., interface with lander, heating techniques).

5. Additional work on alternate sterilization techniques for structural elements is necessary. These techniques would be used for biological load reduction in cases where heat sterilization is not acceptable. Work on the biological aspects of sterility should be collected in handbook form for use by structural designers.

6. Additional studies on material properties should be conducted for some metals and for many non-metals.

   Specific reasons for these recommendations are given in Appendix A, "Recommendations for Future Work."

5. TYPICAL PLANETARY LANDER

5.1 MODEL

5.1.1 BASIC CONSIDERATIONS

In order to evaluate the effect of terminal sterilization it is necessary to describe the vehicle in detail. However, in order to keep the study general and make its results applicable to a range of vehicles rather than a single specific vehicle, an idealized model of a typical planetary lander was established to represent a range of vehicle configurations. The model was sufficiently idealized to prevent the thermal analysis from becoming overly cumbersome, yet sufficiently detailed to obtain meaningful thermal and structural results.

To arrive at a realistic configuration, a review of work on other programs was conducted. The Voyager Design Study prepared by the General Electric Missile and Space Division for the NASA Office of Space Sciences under Contract NAS W-696 was a major source of information. General Electric studies for the Mariner-B vehicle and JPL concepts for the Mariner-C were also useful. The general validity of the configuration was checked by comparison with numerous conceptual configuration drawings by NASA and various aerospace contractors available in the open literature. As a result of this review the model shown in Figure 5.1-1 was prepared. The general characteristics of this model are listed on the figure. The model includes elements of every major structural type (e.g., beams, columns, shells, etc.) which are likely to occur on a typical lander. In this way an analysis of the model would be representative of the range of problems to be encountered on any vehicle.

5.1.2 STRUCTURAL MODEL

The model in Figure 5.1-1 is sufficiently detailed to include: multi-layer shells, beam elements, column elements and panel elements. Joints and details in this figure are shown in sufficient detail to permit thermal idealization. No detail design work was performed for this model. Where specifically designed joints occurred on the Mariner or Voyager vehicles these joints were altered to the simplified details shown on Figure 5.1-1.

From a structural standpoint the model establishes general dimensions and general requirements only.

5.1.3 THERMAL MODEL

The thermal model is shown on Figure 5.1-1. This model was used to perform transient thermal analyses on an analog computer.

Twenty four nodes were used to describe the lander. Node 25 represents the containing canister. This node was subjected to programmed temperature patterns. Node 26 represents an inert gaseous environment.

An axisymmetric arrangement of the nodal points was employed. Nodes 23, 1, 2 and 3 represent the thermal shield and the energy absorbing (crush-up) material. Nodes 24, 4, 5 and 6 represent the structural shell. Nodes 7, 8, 9 and 10 represent an internal structural panel and its components. A sandwich panel was assumed so that nodes 7 and 9 represent one face sheet and the components on one side while nodes 8 and 10 represent the opposite face sheet and the components on the opposite side. The dichotomy occurs in the core midway between the parallel face sheets. It was assumed that the component weight was distributed uniformly over the surface of the mounting panels. Nodes 11 through 16 represent a larger structural panel with its components and is divided in a manner similar to nodes 7 through 10.

Nodes 17, 18 and 19 represent a system of beams and their components. A radial distribution of the structural members was made to facilitate both the thermal and the structural analyses. Node 21 represents the aft cover, node 22, the structural supports for the rocket motor, and node 20, the rocket motor.

The thermal resistance network used in the analog computer showing the relationship between adjacent nodes is presented in Figure 5.1-1. Using this network, it was possible to perform thermal analyses by subjecting node 25, the containing canister, to various programmed heat inputs.

5.2 THERMAL ANALYSIS

5.2.1 SUMMARY

The thermal analysis performed for this study was concerned primarily with:

a. Temperature-time relationships during the sterilization process

b. Techniques and procedures for reducing the time required to reach the sterilization temperature and

c. Techniques for reducing undesirable temperature gradients with the vehicle structure.

REQUIREMENTS FOR IDEALIZED STRUCTURE
FOR UNMANNED PLANETARY SPACECRAFT

1. The structure shall represent a simple lander configuration. A complex vehicle or a complex structural concept is not required for this approach.

2. No specific vehicle shall be represented. The structure will not represent either Mariner, Voyager or any proposed planetary lander, but will be general enough to represent a range of early vehicles.

3. Structure or details of a special type shall be represented only once. As an example, many planetary landers currently in progress feature more than one pressure vessel, more than one boom mounted experiment package. For this model only one of each major type will be considered.

4. The structure shall include portions of each of the following types of structure:

   A. Sheet metal fabrication.
   B. Sandwich
   C. Tubular Space frame.

5. The total weight of the vehicle shall be assumed to be in the general range of other planetary landers (viz Mariner and Voyager).

6. Size, location and type of non structural components including electronics, pneumatics, etc. shall be grossly represented to simplify analysis. Thermal characteristics of these masses shall be representative of black boxes for a typical planetary lander.

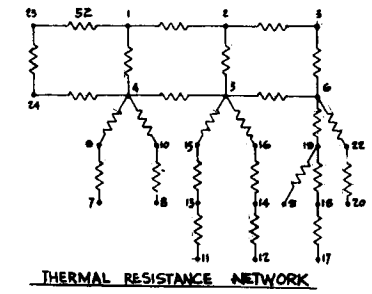
7. Materials shall be selected based upon their successful performance on other space vehicles. The use of new or developmental materials will not contribute to this analysis.

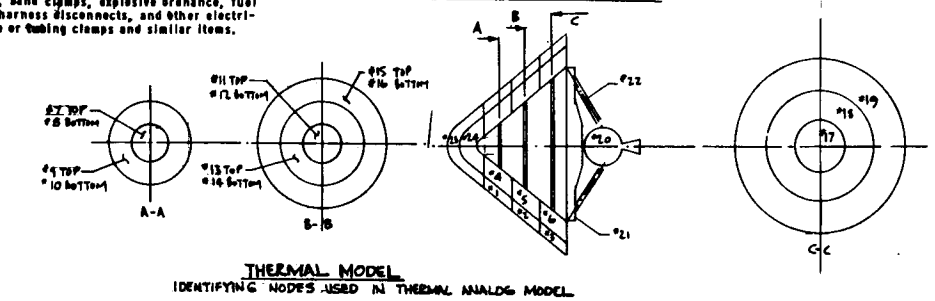
8. Surface finishes, coatings and treatments shall be specified without detailed analysis but based upon judgement and experience on other programs.

9. Material gauges and member cross sections shall be specified without detailed analysis. Sufficient simplifications shall be made to permit a static determination of the member size or skin gage.

10. Critical structural joints, connections, rings, stiffeners, etc. shall be shown in sufficient detail to permit easy thermal idealization.

11. Equipment items which are not primarily structural members shall be excluded from this analysis. These items include hardware often lumped with the structure in weight or system studies and include hardware which is known to be troublesome from a sterilization standpoint. In this category can be included bearings, band clamps, explosive ordnance, fuel valves or fill ports, harness disconnects, and other electrical connectors, cable or tubing clamps and similar items.





WEIGHT DISTRIBUTION FOR TYPICAL LANDER (1400#)

1. FOR 1400# LANDER APPROX 15-20% MAY REQUIRE REMOVAL WHEN HEAT STERILIZATION OCCURS ∴ WT. IN STERILIZATION CHAMBER MAY BE APPROX 1200 LB.

2. TYPICAL WEIGHT DISTRIBUTION FOR 1200 LB VEHICLE
   - 90 LB-HEAT SHIELD
   - 250 LB-STRUCTURE
   - 150 LB-RETARDATION
   - 150 LB- COMMUNICATIONS
   - 150 LB- ADAPTOR
   - 210 LB- SCIENTIFIC EXPERIMENTS
   - 110 LB.- ELEC. & ELECTRONICS
   - 90 LB.- THERMAL CONTROL

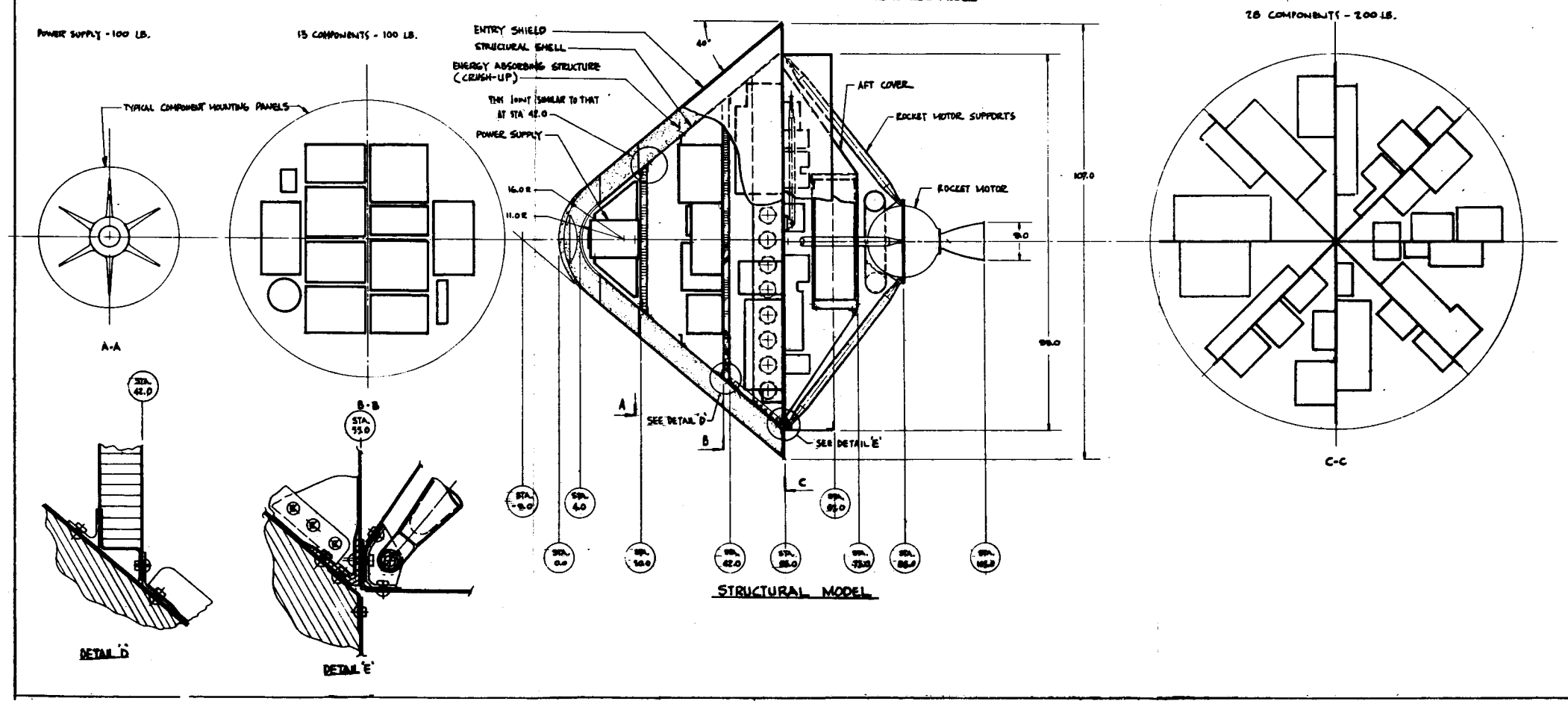


Figure 5.1-1. Typical Planetary Lander

These items are important because (1) many structural materials experience a loss in strength when subjected to elevated temperatures for extended periods and (2) large temperature gradients may result in significant thermal stresses. The importance of item c. is discussed in Section 6.10 which describes techniques for improving joint conductance.

The major thermal analysis effort was concentrated on developing an electronic analog model of a typical planetary lander. This model was used to establish estimates of the most pessimistic and the most optimistic transient characteristics of the lander. The variables considered were: (a) thermal diffusivity of the lander structural material, (b) the presence or absence of a conducting gas within the lander sterile enclosure (canister), (c) the canister temperature-time program. The lander vehicle in Figure 5.1-1 contains examples of a variety of structural joints which were used to obtain conductance values.

Table 5.2-1 gives a summary of the sterilization times involved for various spacecraft configurations and methods of sterilization. Once the entire lander temperature was raised to 300°F, the vehicle was allowed to dwell at that temperature for 60 hours, then cooled down to 70°F. This cycle would be repeated four times in accordance with the contract work statement.

The cooldown time was assumed equal to the heatup time. The total sterilization time shown in Table 5.2-1 includes transient and steady state times.

TABLE 5.2-1. STERILIZATION TIME

Vehicle	Emittance	Condition	Heatup Time * (hours)	Heatup Time ** (% of total sterilization time)	Total Sterilization Time - 4 cycles (hours)
Aluminum	0.9	Vacuum	2	10.5	304
			6	13.4	328
			2	5.2	268
Aluminum	0.9	Gas	6	9.5	296
			2	24.1	464
Titanium	0.9	Vacuum	6	23.9	468
			2	20.0	400
Titanium	0.5	Gas	6	21.1	416
			2	27.3	528
Titanium	0.5	Vacuum	6	28.0	544

* For boundary, Node 25
** Time for coldest node to reach 150°C

Although the titanium vehicles represent an extreme case, it is possible to construct thermally resistant vehicles which would require longer exposure times. As an example, on vehicles like the Voyager-Venus probes, heavy multi-layer insulation is required for internal equipment. The presence of this insulation can add 30 to 40 hours to each sterilization cycle. In order to shorten this time, heaters would have to be located below the insulation.

Two different coating property values are reported in Table 5.2-1. An emittance of 0.9 represents a practical maximum value. The value of 0.5 represents an expected average value.

Figures 5.2-1 and 5.2-2 provide a graphical summary of the range of temperature gradients occurring on the hypothetical spacecraft. The temperature differences indicated are the differences observed between the coldest (inner) and the hottest (outer) points of the lander. It should be noted that these nodes, (1 and 7) are not adjacent nodes; hence these gradients are not likely to occur on any structural element.

These figures indicate that it is possible to have temperature differentials on the order of 200°F even with a 6-hour heat-up time for the canister.

5.2.2 ANALYSIS

A lander vehicle design was selected to illustrate a variety of different mechanical joints. From this design an electronic analog was developed. Figure 5.1-1 shows the thermal resistance network. Internode radiation is not shown, although a set of internode configuration factors was assumed and factored into the model.

The basic equation for a typical node is given below:

$$W_i C_{p_i} \frac{dT_i}{dt} = \sum_j K_{i_j} (T_j - T_i) + \sum_j F_{i_j} (T_j^4 - T_i^4) - \epsilon_i A_i \sigma T_i^4$$

where: A = area

C_p = specific heat

F = radiation interchange factor

K = conductance

W = weight

ϵ = emittance

σ = Boltzmann's constant

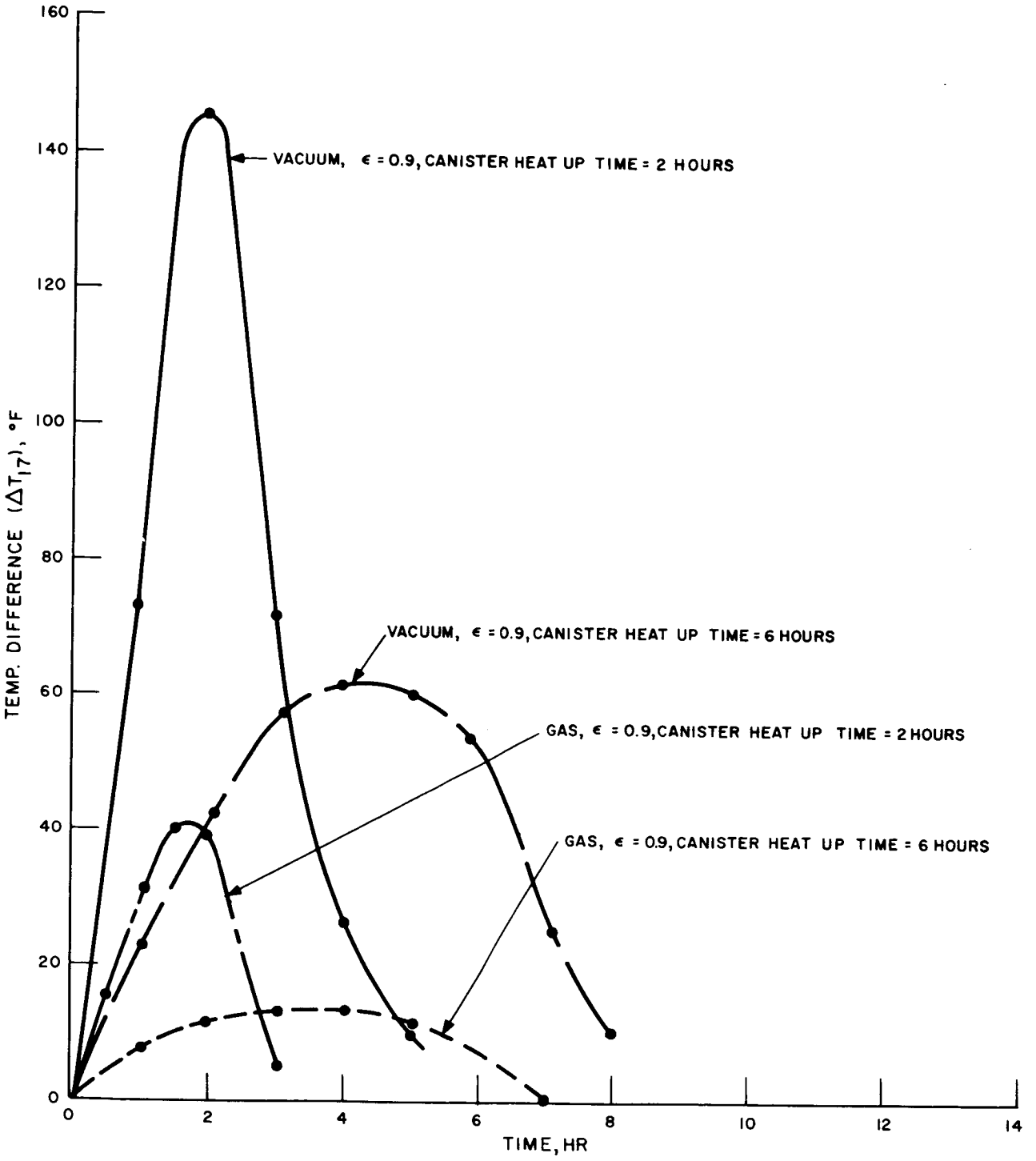


Figure 5.2-1. Maximum Temperature Differential verses Time - Aluminum Vehicle

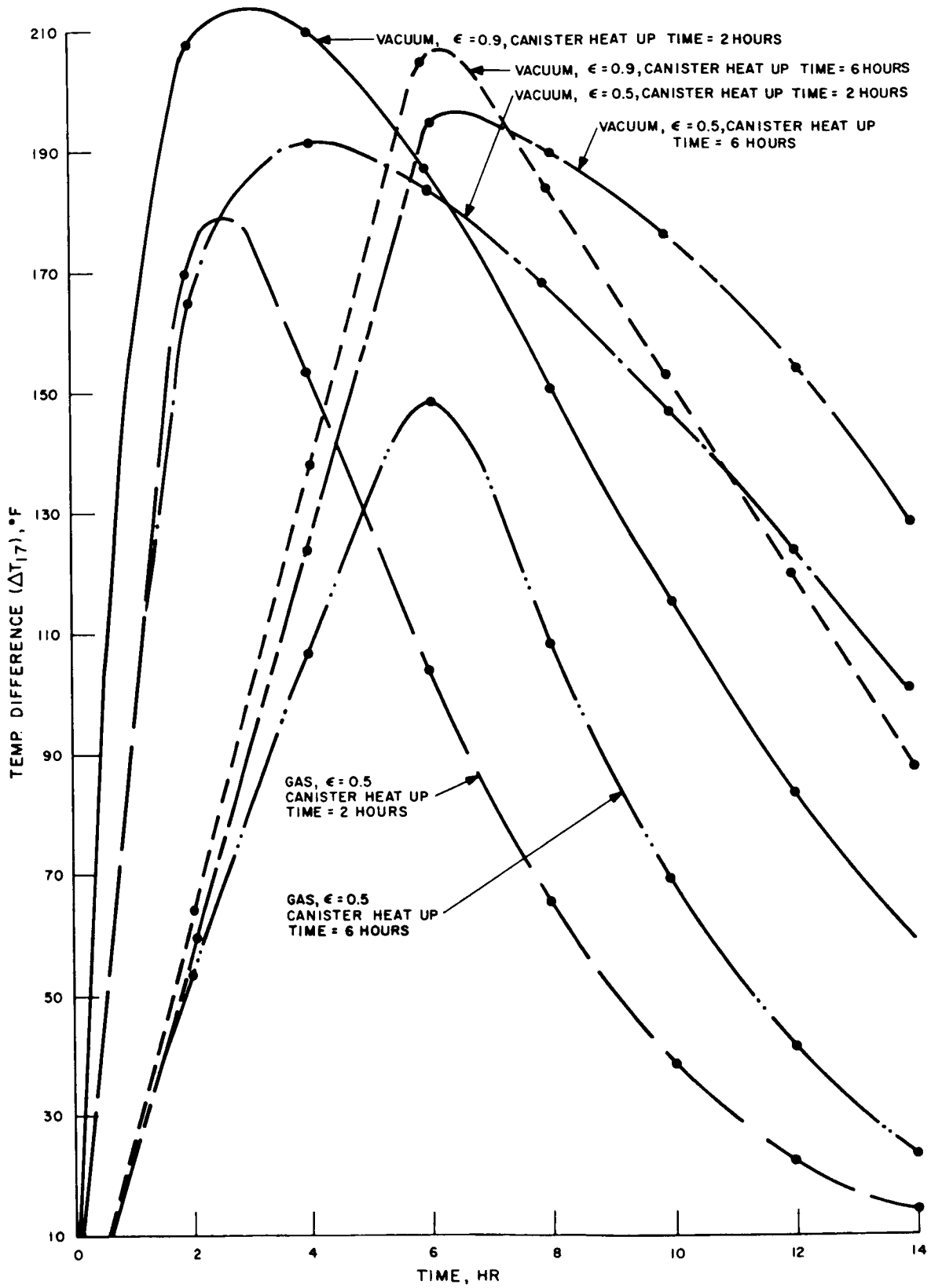


Figure 5.2-2. Maximum Temperature Differential versus Time - Titanium Vehicle

Twenty-four nodes were used to describe the lander. Node 25 was used to represent the canister with a programmed temperature pattern. A 26th node was used to represent the presence of an inert gaseous environment.

The analog model of the simplified lander was set up on an Electronic Associates, Inc. Model 231R special purpose heat transfer computer. Plots of the results were prepared on an EA1 Model 1100E x-y plotter.

No specific gas was considered in those cases where gaseous sterilization was investigated. In order to represent the gas, a representative natural convection film coefficient of 0.5 Btu/(hr)(ft^2)(^{o}F) was assumed. This value falls within the region of expected values. For a relatively complex geometry the only accurate method for determining the actual value is by physical testing. For a practical design it would be desirable to use a high conductivity gas such as helium. Nitrogen could also be used.

5.2.3 RESULTS

The significant computer results are presented in Figures 5.2-3 through 5.2-13. These figures show the nodes across which the most critical gradients were experienced. Table 5.2-2 summarizes all cases considered during this study. The runs are tabulated in the order in which they were performed. Several cases were run in which the canister temperature was taken above 300°F with the thought that the spacecraft would heat more slowly than it did and that the process could be accelerated without causing any portion of the structure to exceed 300°F. As shown on Figure 5.2-3 this procedure resulted in an overshoot by some nodes, and for this reason the heating procedure was altered.

From the data it can be seen that the heating time is sufficiently short, compared to the 60-hour soak time, to suggest that heating the canister above 300°F is undesirable even for a short period.

The most salient characteristic of Figures 5.2-3 through 5.2-7 is the short time required for the spacecraft to reach near equilibrium conditions. There are several reasons for this behavior: (1) an aluminum structure was assumed; (2) the entire interior and exterior of the vehicle was assumed coated with a thermodynamically black coating and (3) brazed honeycomb was assumed for the sandwich structure.

In the examples studied, gaseous sterilization reduced the time to reach sterilization temperature almost 50% from the vacuum sterilization case. This results in a 10% reduction in the total soak time at elevated temperatures. The other advantage of gaseous sterilization is the fact that it reduced the most extreme temperature gradients by as much as 60%. Gaseous sterilization has the added

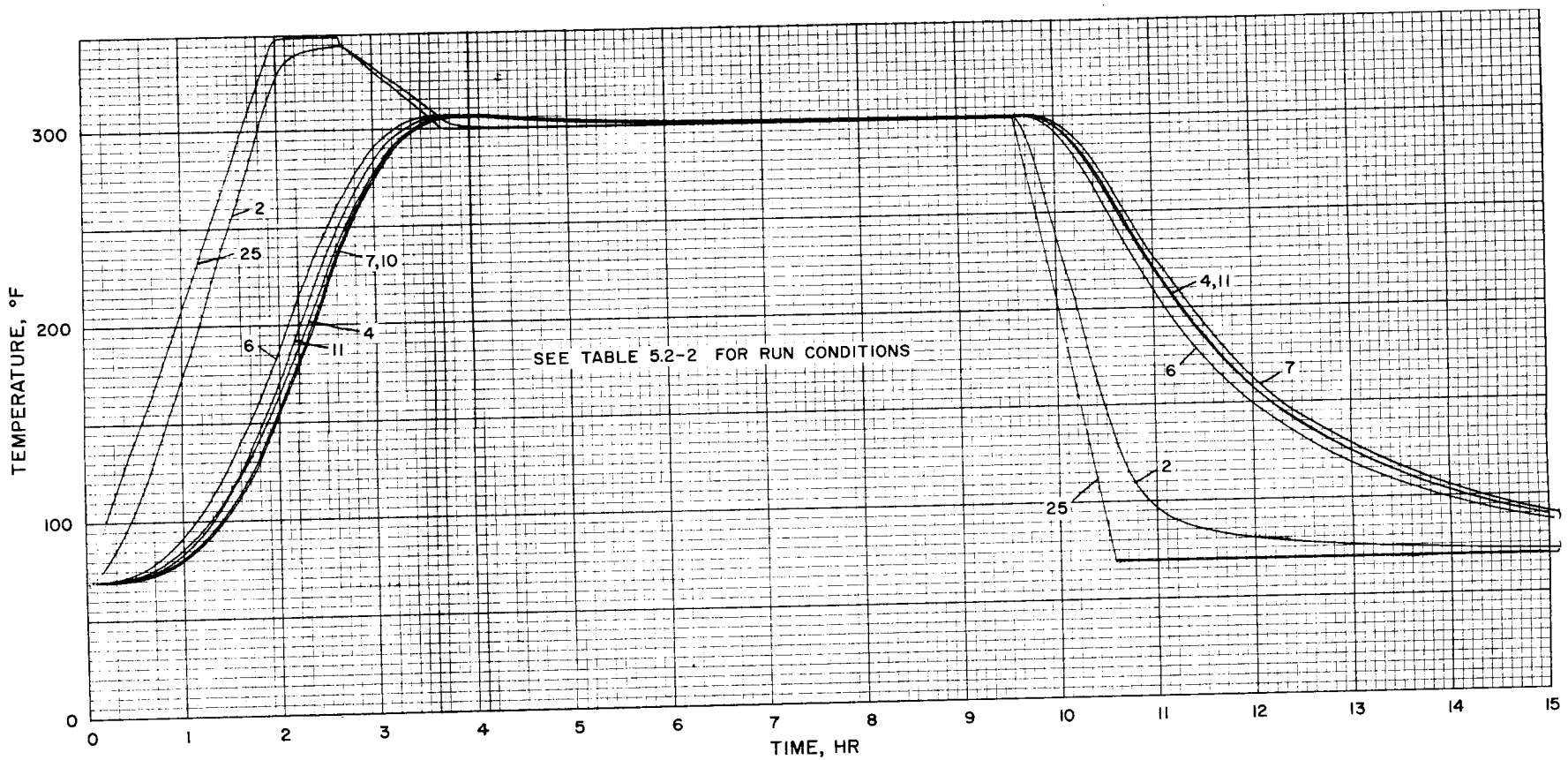


Figure 5.2-3 Temperature Data - Typical Nodes - Run 3, Aluminum Vehicle

advantage of increasing joint conductances and increasing the apparent thermal conductivity of multi-layer insulations. Increasing the time to heat up the canister from 2 hours to 6 hours only increases the total sterilization time by about 8 hours, its major effect being in the reduction of thermal gradients. The most significant temperature gradients appear to exist between the outer and inner portions of the structure rather than within an individual structural member. An interesting feature of Figures 5.2-3 through 5.2-7 is the uniformity with which it is possible to heat the vehicle to sterilization temperatures independent of the heat-up time. These figures represent a best set of conditions in terms of total time required for heat up: i.e., lowest heating time.

The worst set of conditions is shown in Figures 5.2-8 through 5.2-13 which represent a titanium vehicle. The lower thermal conductivity of titanium results in thermal lag of cold node and increased exposure time for the hot nodes.

5.2.4 CONCLUSIONS

Thermal sterilization in a gas is far more efficient than thermal sterilization in a vacuum, both in terms of minimizing total sterilization time and in reducing thermal gradients.

A. High thermal conductivity materials and joining techniques should be used as much as possible, consistent with good design practices. Examples of these techniques include brazed core honeycomb sandwich in preference to bonded sandwich, and welded joints in preference to bolted, riveted, or bonded joints. This conclusion is borne out by comparing the behavior of an all aluminum vehicle with an all titanium vehicle.

B. The capability of providing both programmed heating and cooling is desirable in order to minimize thermal gradients in the vehicle structure.

C. The use of vacuum-type, multi-layer insulation in conjunction with vacuum sterilization is incompatible with short sterilization cycles. The use of multi-layer insulation could add as much as 30 to 40 hours to each sterilization cycle.

D. Techniques for heating and cooling the containing canister warrant further study.

TABLE 5.2-2. SUMMARY OF THERMAL ANALOG RUN CONDITIONS

Figure	Run No.	Lander Material	Heating Time (Hr)	Condition	Emittance	Peak Canister Temp. (°F)	Notes
	1	Alum. structure, brazed honeycomb,* alum. core	1	vacuum	0.9	350	Canister was raised to 350° in one hour. When the coldest point reached 300°F the canister was cooled to 300°F in one hour. The canister was cooled to 70°F in one hour following 60-hour dwell.
	2	Alum. structure, brazed honeycomb, alum. core	1	vacuum	0.9	350	Canister was raised to 350°F in one hour. When the coldest point reached 240°F the canister was cooled to 300°F in one hour. The canister was cooled to 70°F in one hour following 60-hour dwell.
5.2-3	3	Alum. structure, brazed honeycomb, alum. core	2	vacuum	0.9	350	Canister was raised to 350° in two hours. When the coldest point reached 240°F the canister was cooled to 300°F in one hour. The canister was cooled to 70°F in one hour following 60-hour dwell.
5.2-4	4	Alum. structure, brazed honeycomb, alum. core	2	vacuum	0.9	300	Canister was raised to 300°F in two hours. Canister was cooled to 70°F in 1 hour. Dwell time was neglected.
	5	Alum. structure, brazed honeycomb, alum. core	4	vacuum	0.9	300	Cooling cycle was not considered.
5.2-5	6	Alum. structure, brazed honeycomb, alum. core	6	vacuum	0.9	300	Cooling cycle was not considered.
	7	Alum. structure, brazed honeycomb, alum. core	2	vacuum	0.9	300	Canister cooled from 300°F to 70°F in 2 hours. Heating was not considered.
	8	Alum. structure, brazed honeycomb, alum. core	4	vacuum	0.9	300	Canister cooled from 300°F to 70°F in 4 hours.
	9	Alum. structure, brazed honeycomb, alum. core	6	vacuum	0.9	300	Canister cooled from 300°F to 70°F in 6 hours.
5.2-6	10	Alum. structure, brazed honeycomb, alum. core	2	gas	0.9	300	Heating cycle only.
	11	Alum. structure, brazed honeycomb, alum. core	4	gas	0.9	300	Heating cycle only.
5.2-7	12	Alum. structure, brazed honeycomb, alum. core	6	gas	0.9	300	Heating cycle only.
5.2-8	13	Titanium structure fiberglass core sandwich	2	vacuum	0.5	300	Heating cycle only.
5.2-9	14	Titanium structure fiberglass core sandwich	6	vacuum	0.5	300	Heating cycle only.
5.2-10	15	Titanium structure fiberglass core sandwich	2	gas	0.5	300	Heating cycle only.
5.2-11	16	Titanium structure fiberglass core sandwich	6	gas	0.5	300	Heating cycle only.
5.2-12	17	Titanium structure fiberglass core sandwich	2	vacuum	0.9	300	Heating cycle only.
5.2-13	18	Titanium structure fiberglass core sandwich	6	vacuum	0.9	300	Heating cycle only.

* For calculating conductance values brazed joints were assumed. This assumption tends to make rise times more rapid. Bonded joints would probably be used in practice but for this study it was convenient to assume brazed joints.

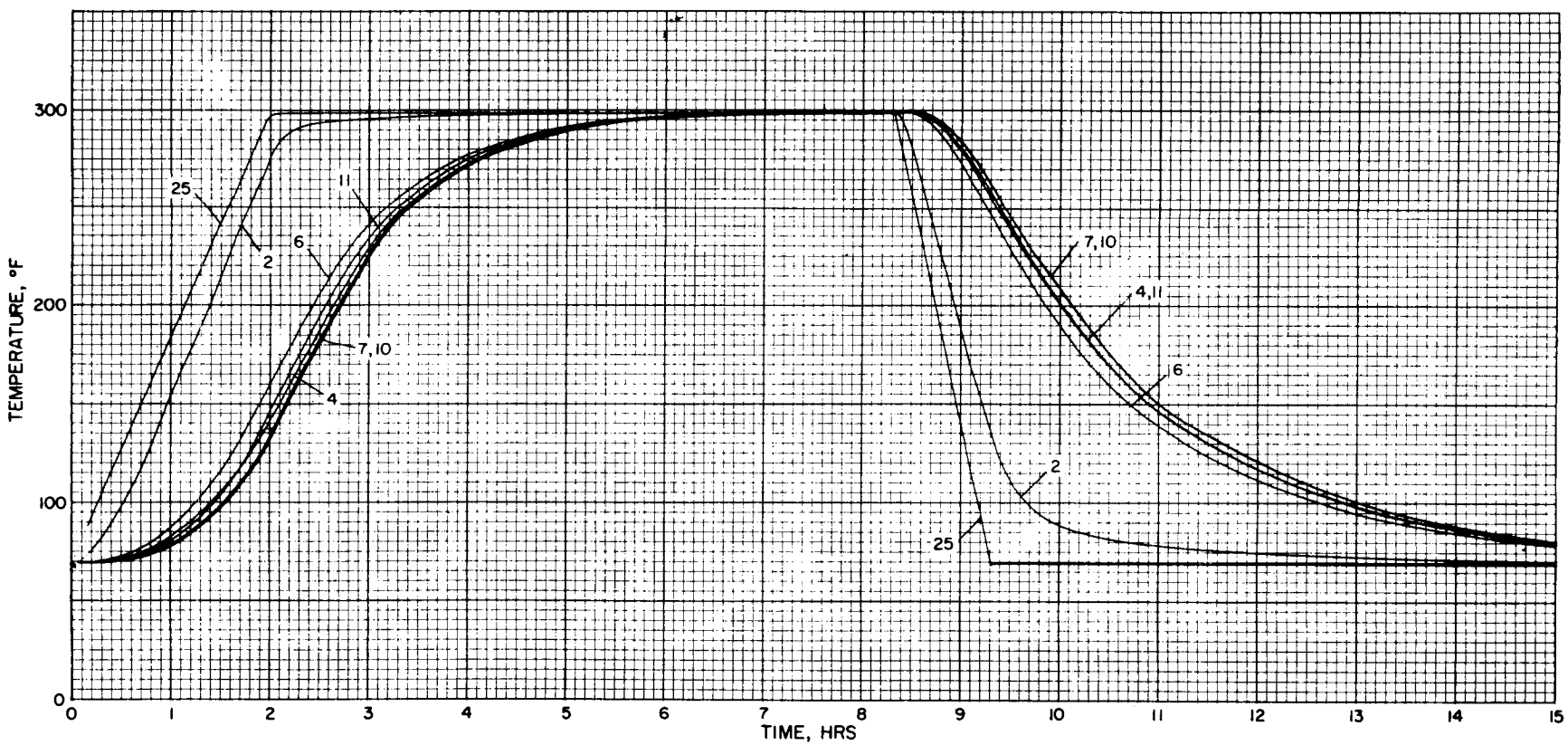


Figure 5.2-4 Temperature Data - Typical Nodes - Run 4, Aluminum Vehicle

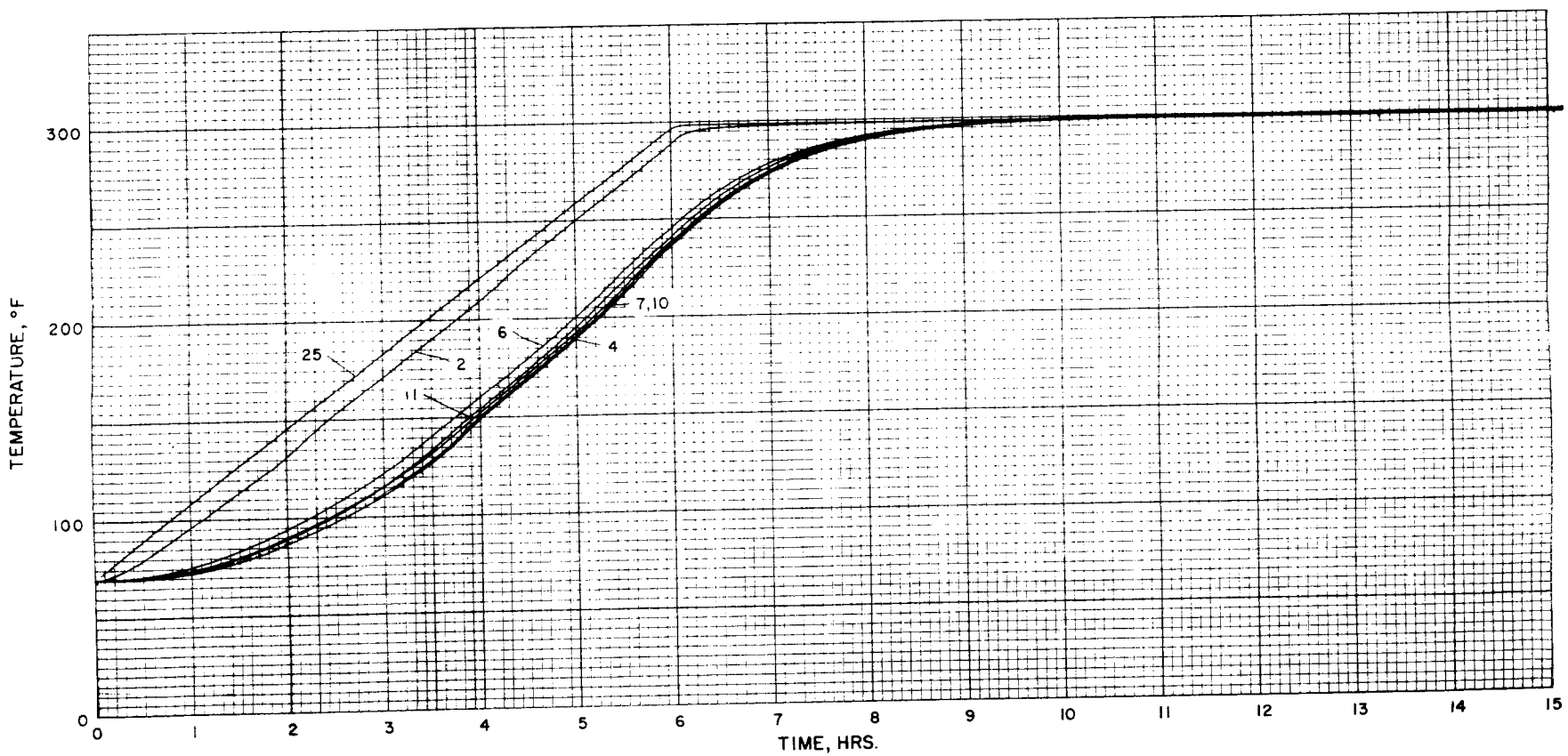


Figure 5.2-5. Temperature Data - Typical Nodes - Run 6, Aluminum Vehicle

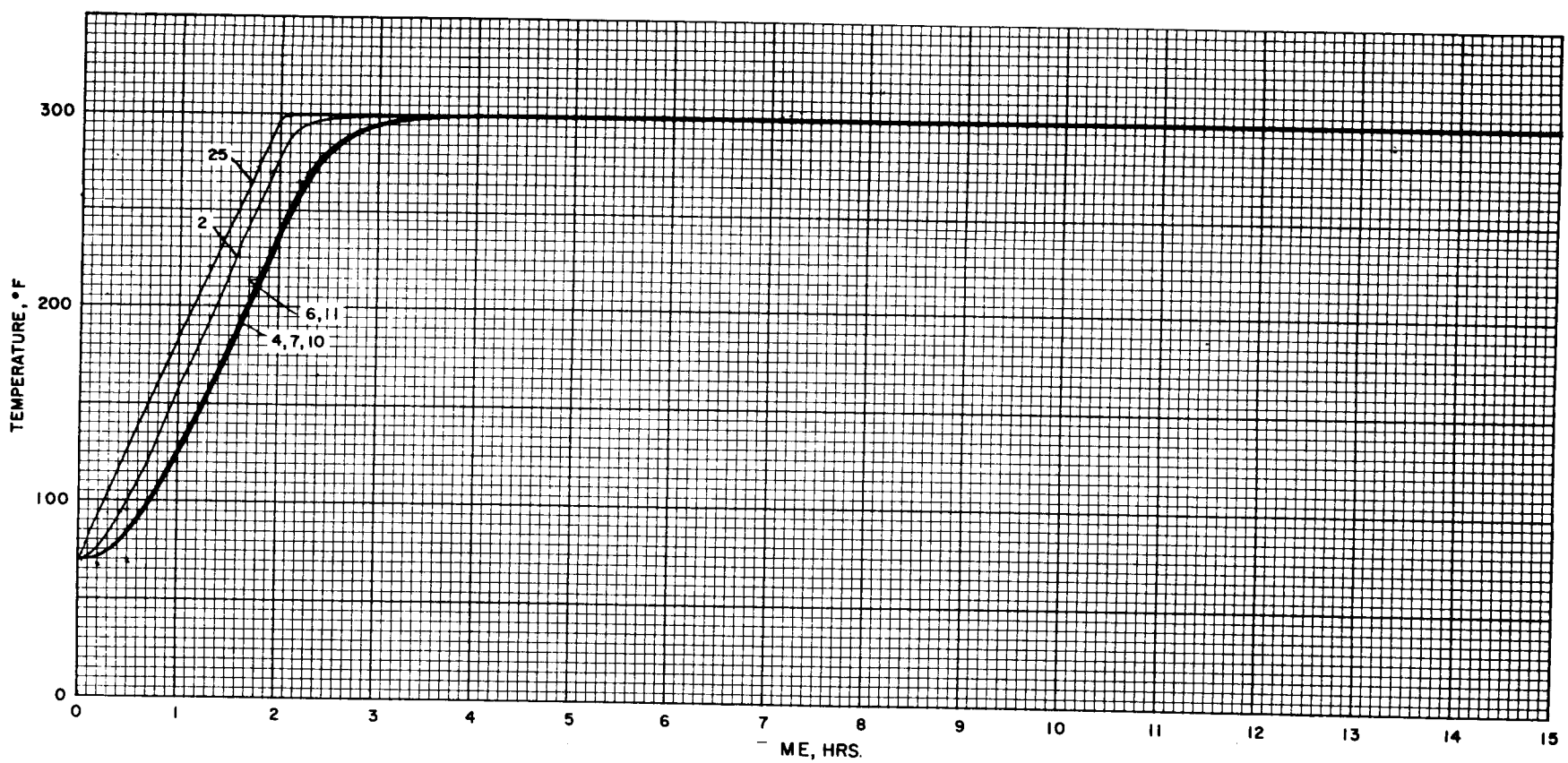


Figure 5.2-6. Temperature Data - Typical Nodes - Run 10, Aluminum Vehicle

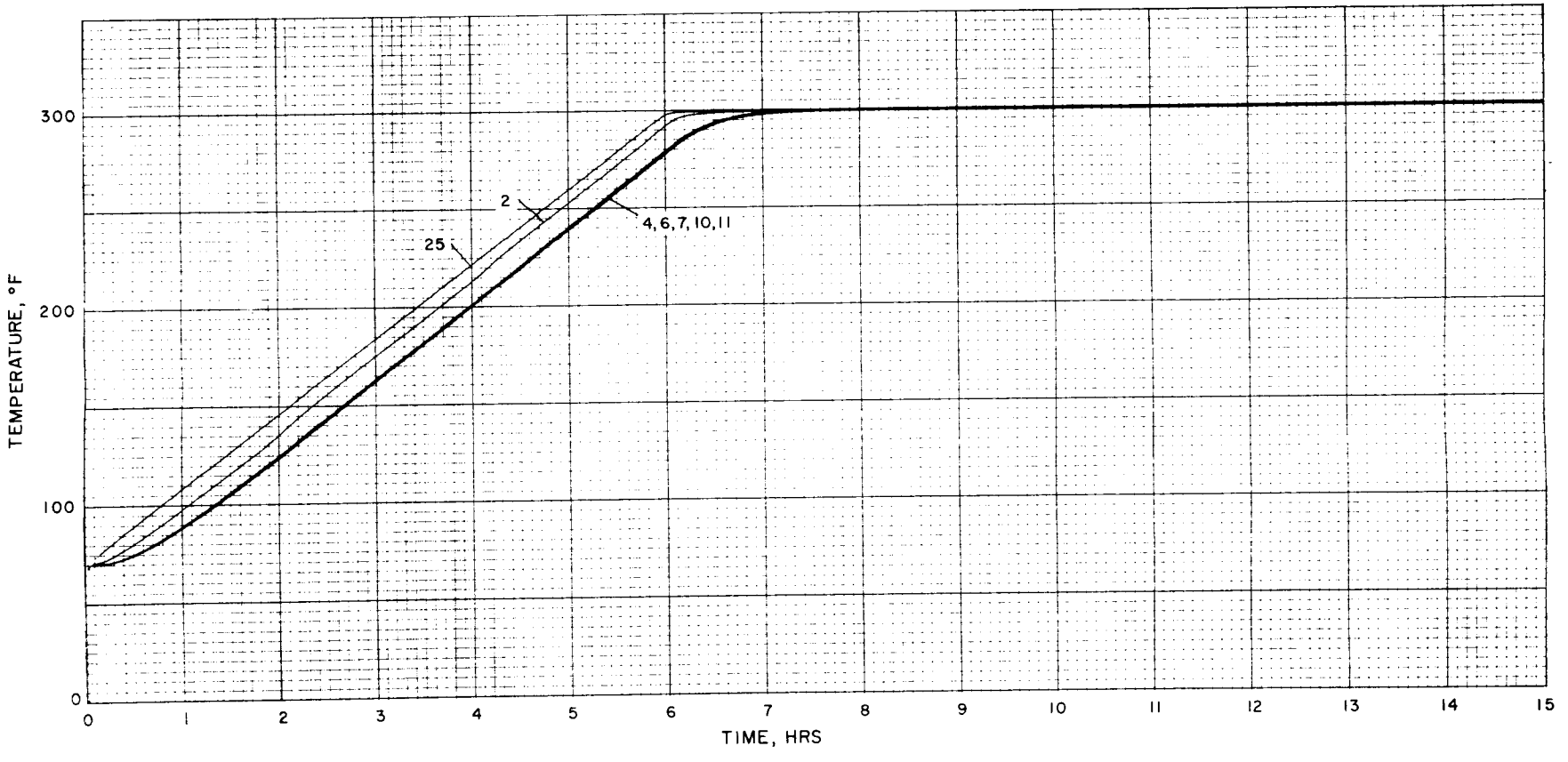


Figure 5.2-7. Temperature Data - Typical Nodes - Run 12, Aluminum Vehicle

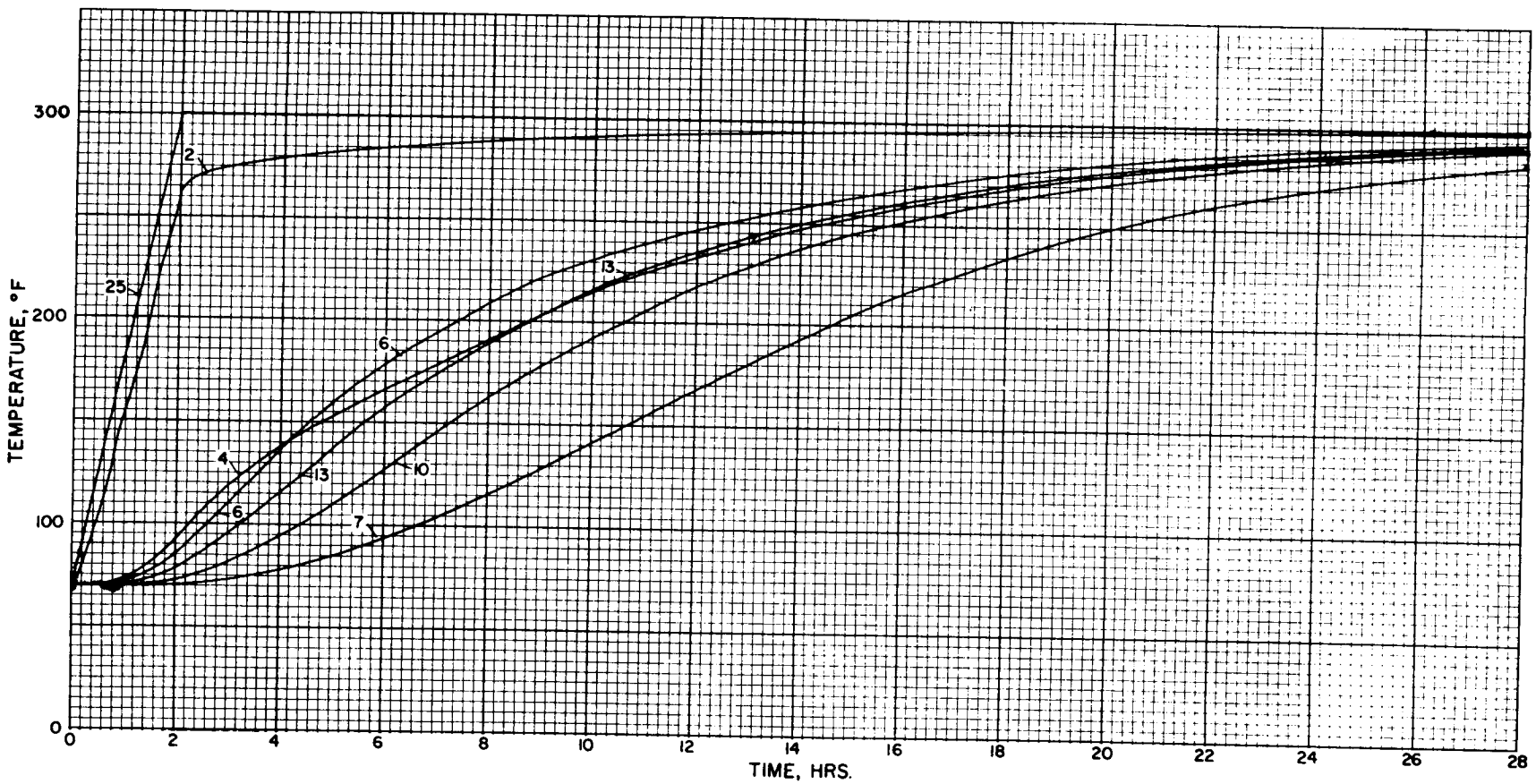


Figure 5.2-8. Temperature Data - Typical Nodes - Run 13, Titanium Vehicle

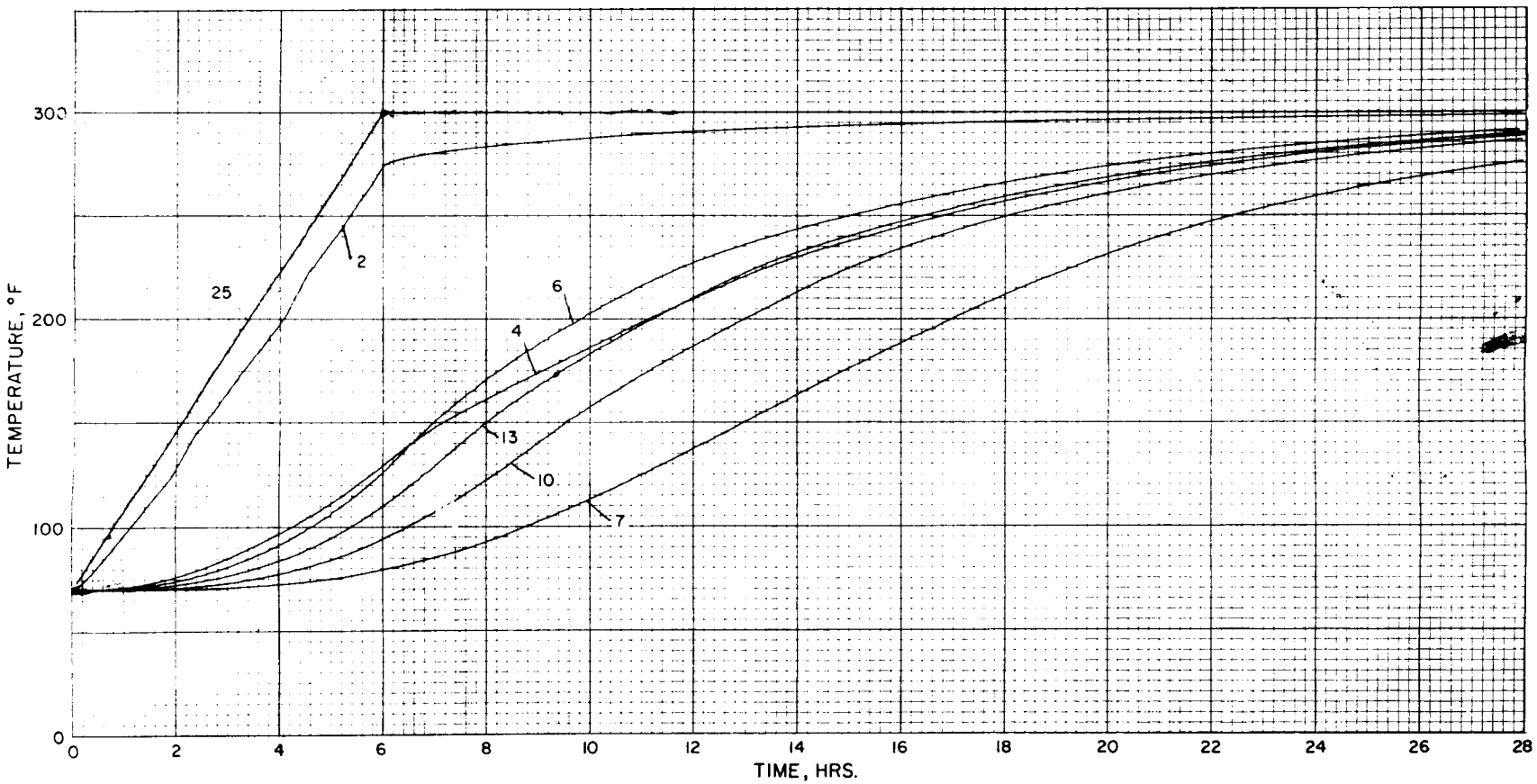


Figure 5.2-9. Temperature Data - Typical Nodes - Run 14, Titanium Vehicle

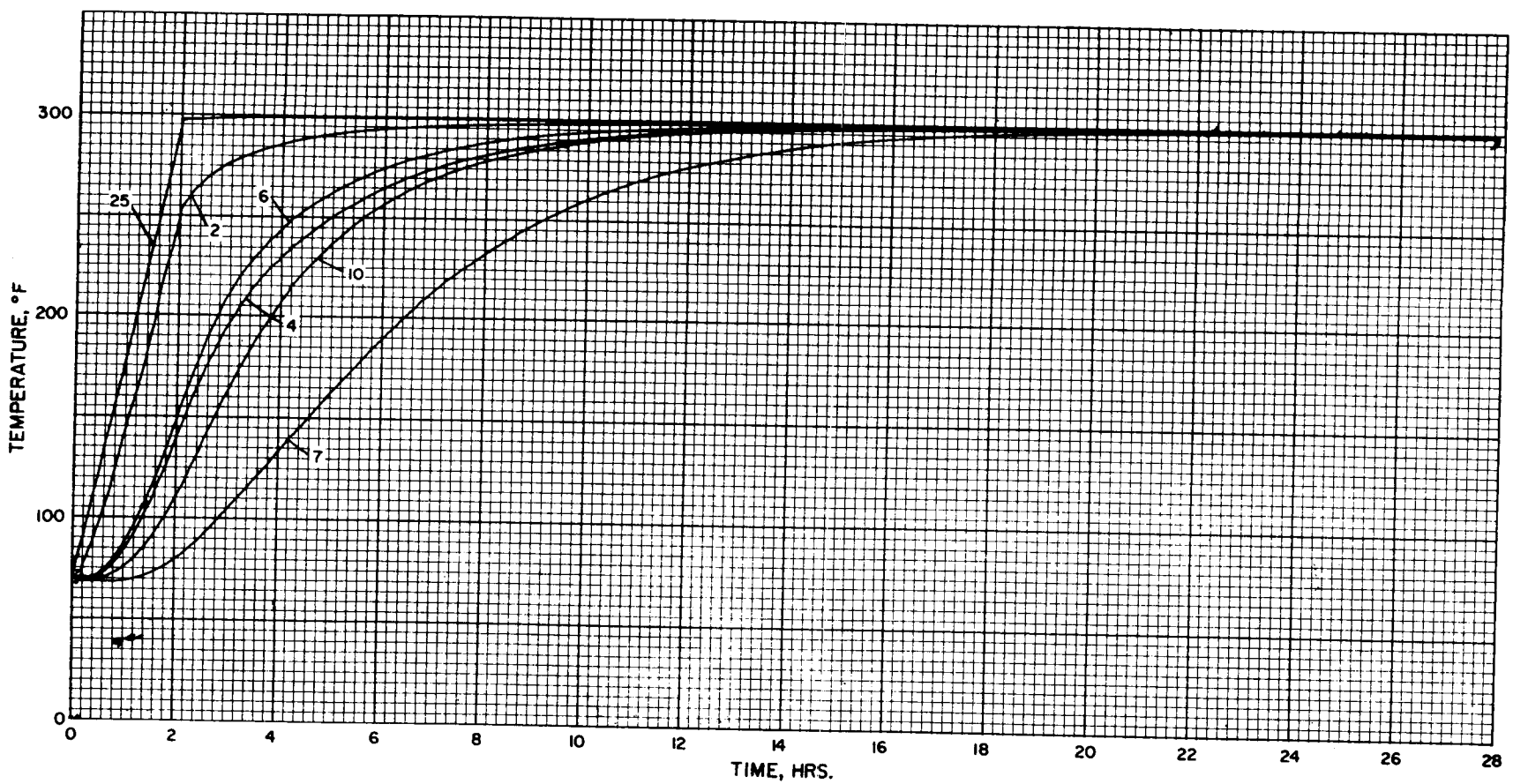


Figure 5.2-10. Temperature Data - Typical Nodes - Run 15, Titanium Vehicle

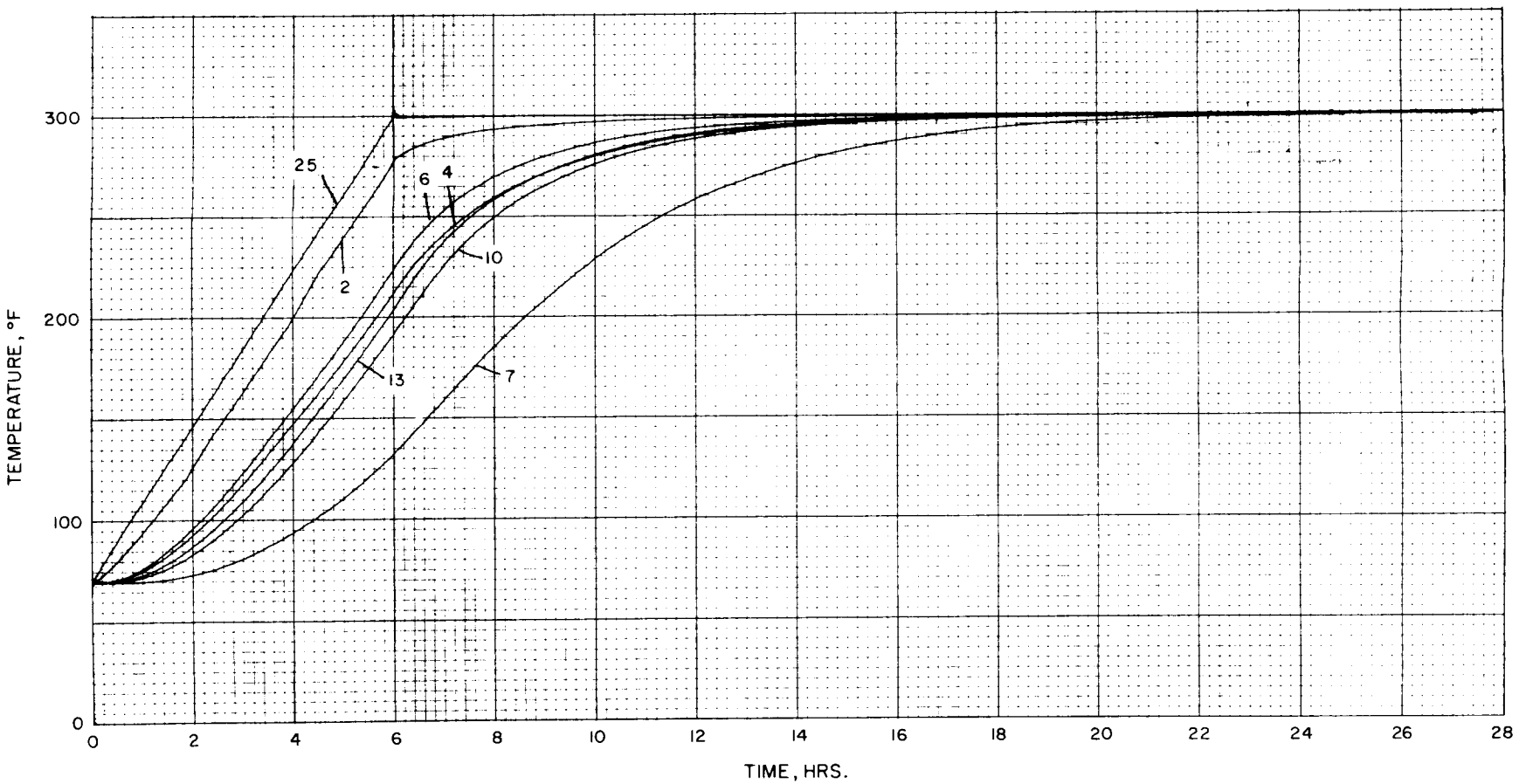


Figure 5.2-11. Temperature Data - Typical Nodes - Run 16, Titanium Vehicle

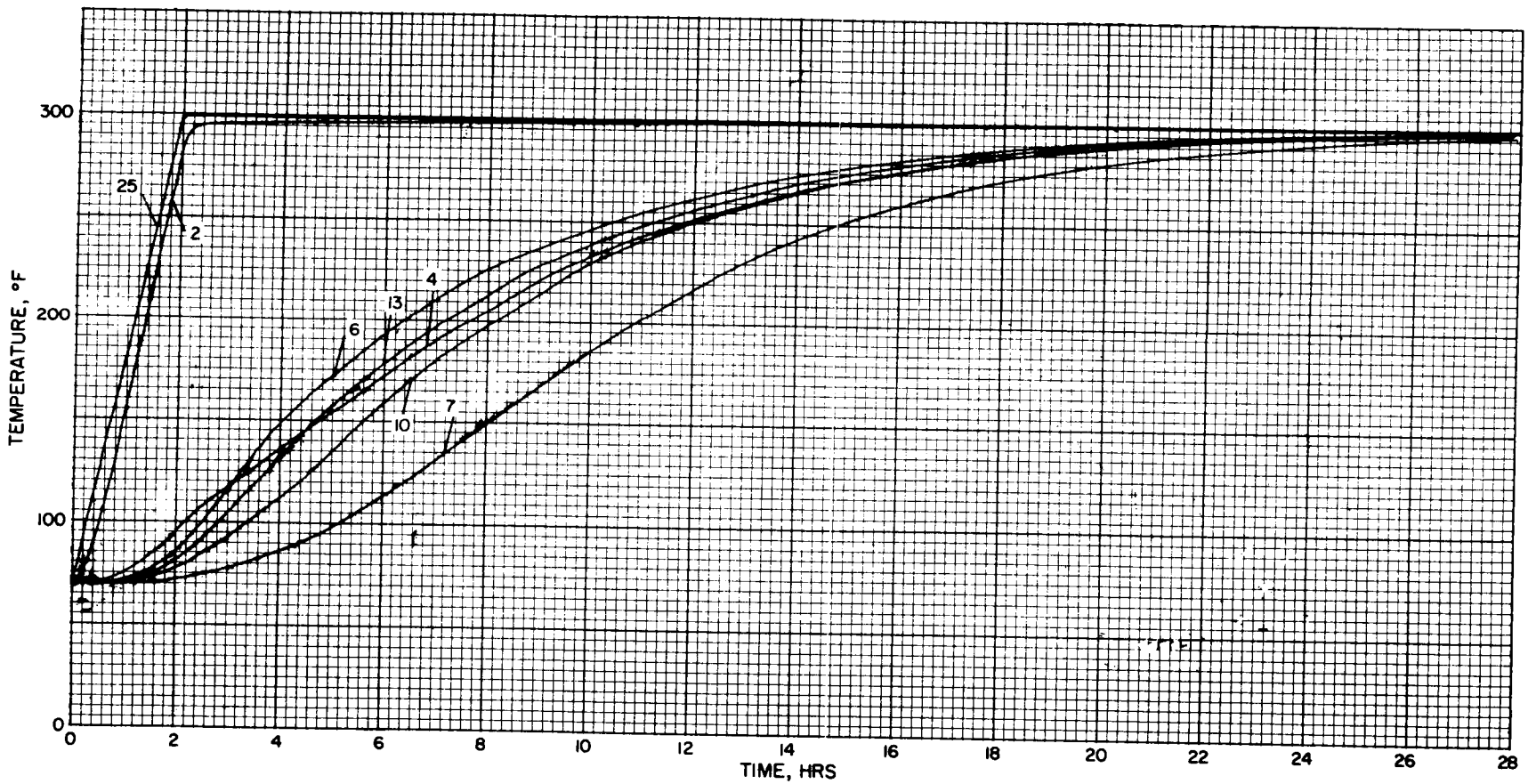


Figure 5.2-12. Temperature Data - Typical Nodes - Run 17, Titanium Vehicle

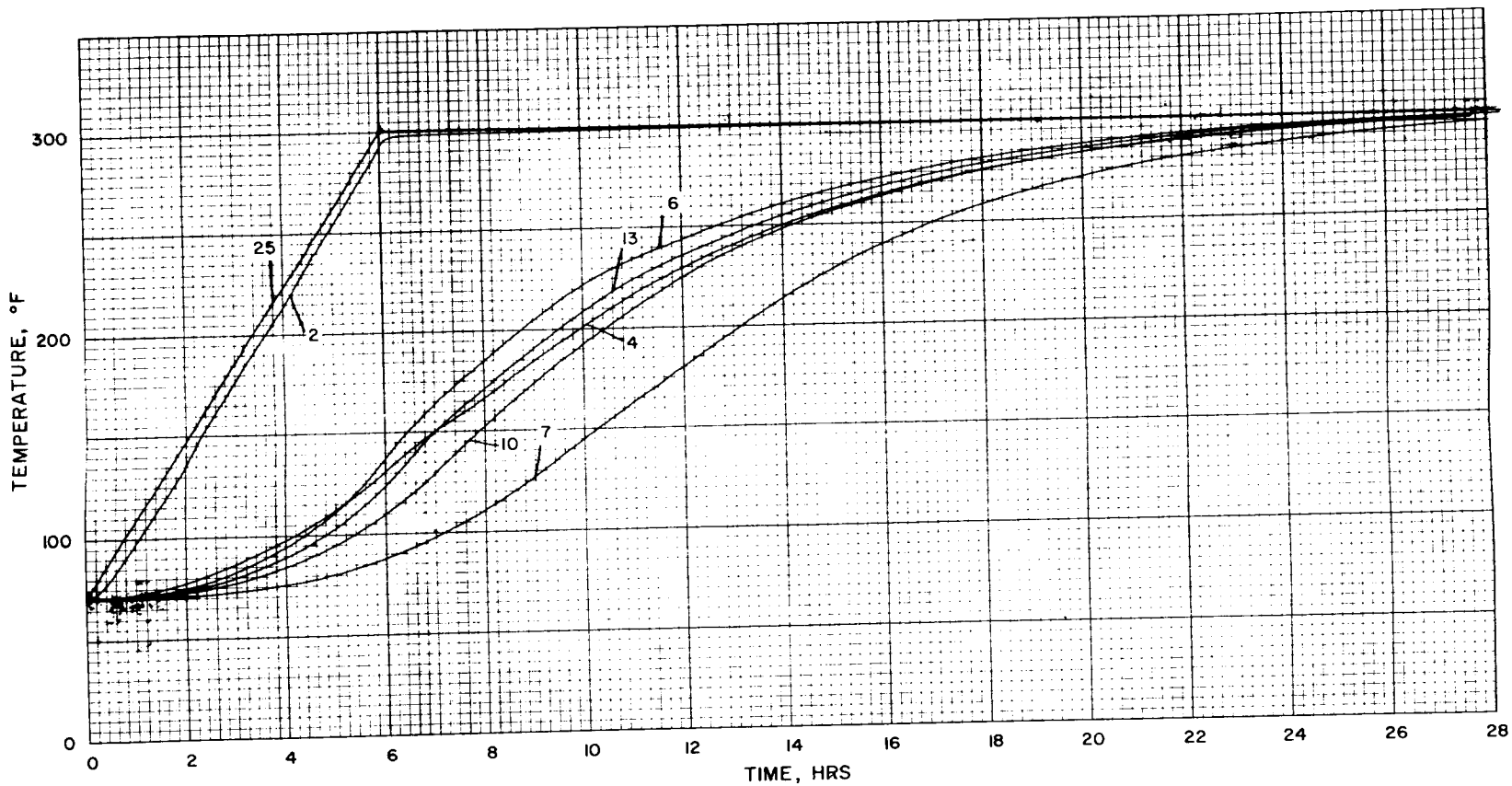


Figure 5.2-13. Temperature Data - Typical Nodes - Run 18, Titanium Vehicle

5.2.5 SPECIAL PROBLEMS

A. Joint Conductance

This analysis purposely omitted considering variations in the contact conductance between the various joints shown in the model. The reason for this is because of the fact that the model chosen is not a good example to illustrate the importance of this variation. The major resistance to heat flow in the idealized model is in conduction through the various structural elements. This condition cannot, however, be regarded as general. One of the major gaps in our ability to predict accurately the thermal behavior of spacecaaft in many cases is the inability to know accurately the value of the contact conductance to be used between two mating surfaces. Considerable experimental data exist on contact conductance between two surfaces under idealized conditions. Comprehensive data on the characteristics of actual design joints are very sparse. Much of the idealized data considers two surfaces in intimate contact with one another with a uniform pressure applied over the two surfaces. For an actual joint this is very far from the case. A report by Aron and Colombo (Reference 5.5.1) shows that, for two plates of equal thickness bolted together, at a distance of 1.3 - 1.7 times the plate thickness from the bolt the pressure is negative, i.e., the compressive load near the bolt actually becomes a tensile load causing the plates to physically separate. This phenomenon has been corroborated in some limited experimental work on the Advent program performed by the General Electric Spacecraft Department.

Figure 5.2-14 shows a profile of the gap existing across the mating surface of a component with a .09 in. thick base plate bolted to a 1/2-in. aluminum core honeycomb with a .060-in. face sheet facing the component base plate and an 0.012-in. face sheet on the opposite surface. It is quite evident from this profile, shown as a function of bolting torque, that increasing the bolt torque does not, at least in this example, provide more complete contact between the two mating surfaces. A series of systematic experiments is required to establish the basic contact conductance data for real joints and to establish basic standards for both spacing and torque. This is highly pertinent to spacecraft sterilization since the resistance through joints can be a controlling factor in the rate at which portions of a spacecraft can be heated up. This subject is discussed in detail in Section 6.10.

B. Insulation Characteristics

In order to establish the general problems associated with sterilizing an insulated system it is appropriate to examine some of the thermal characteristics of super-insulation and other common insulating materials. With super-insulation the spacing between each layer of insulation is smaller than the mean free path of the gas molecules, the heat conduction through the residual gas being proportional to the gas pressure. Therefore, at pressures less than approximately 0.1 μ Hg.

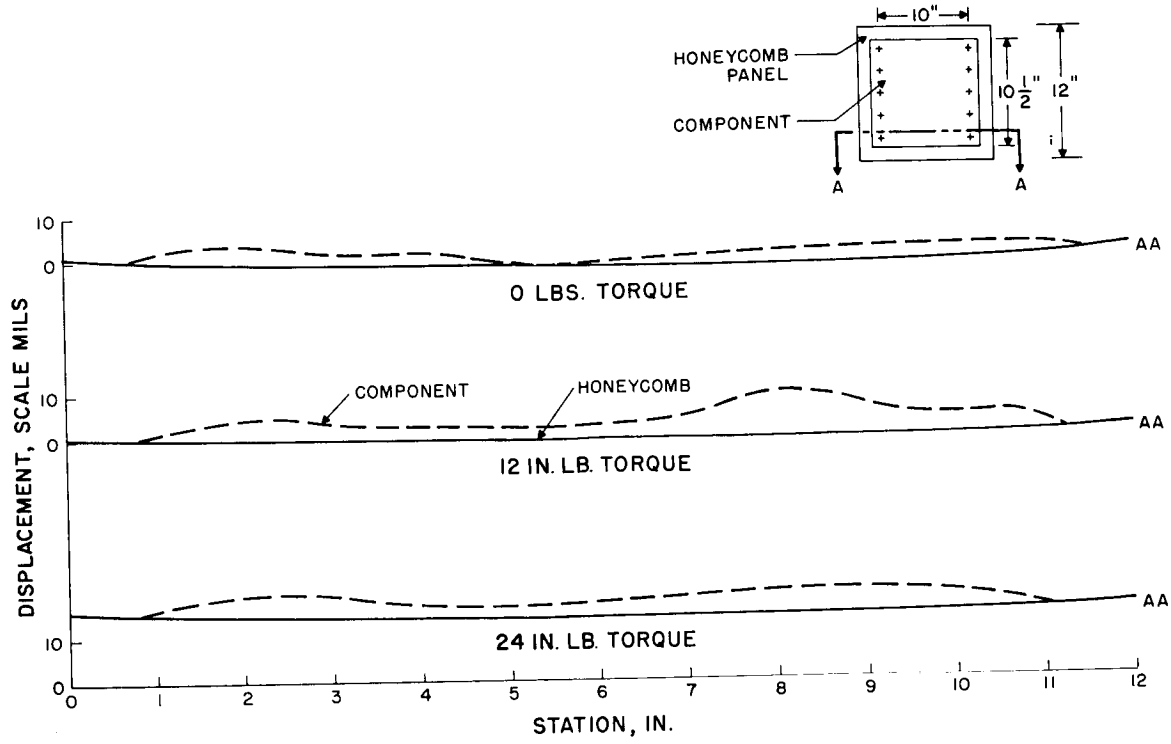


Figure 5.2-14. Contact Profile Between a Typical Component and Its Mounting Plate

gaseous conduction through super insulation is almost negligible. Figure 5.2-15 shows a plot of apparent thermal conductivity as a function of gas pressure for two typical super-insulating materials. Under vacuum conditions this type of insulation can add 30 to 40 hours to the sterilization time of each sterilization cycle in order to raise the insulated item to sterilization temperatures, unless a heater were located under the insulation to short circuit the insulation. In general, insulation defeats the purpose of sterilization. Table 5.2-3 shows the thermal conductivity as a function of pressure for some other common insulating materials.

C. Thermal Sterilization Procedures

From the discussion in Section 5.1 it is apparent that a technique for programmed heating and cooling of the lander enclosure (canister) is called for. Several techniques are possible, including:

1. Heater blankets attached to the heater canister.
2. Utilization of the canister as a radiator similar to the Gemini adapter design, i.e., have a heated fluid (gas) pass through tubes mounted on the interior surface of the canister.
3. Enclosing the entire canister and lander with an oven.

5.3.1 ENTRY SHIELD, BOND AND STRUCTURE

A. General

The prime requirement of any entry shield-bond-structure system of a planetary lander is to perform a given mission surviving all of the ground and flight environments imposed upon it with minimum weight. In order to meet the minimum weight criteria, thermal shields must often be chosen primarily on the basis of thermal and ablation performance during entry. Unfortunately, the optimum shield materials usually have a high coefficient of thermal expansion and low strengths, while the substructure has high strength and lower coefficients of thermal expansion. The bond system, mating the thermal shield to the substructure must be selected to compensate for this thermal incompatibility.

B. Discussion

The bond requirements in all systems are two-fold. The bond itself must not fail due to tensile and shear stresses, while allowing relative deflections between shield and structure to preclude failure of either of these members due to thermally induced loads.

The 300°F sterilization environment becomes a significant design condition for a shield-bond-structure combination because usually, for an orbiting vehicle or for a vehicle in transit to one of the near planets, the most severe cyclic temperature ranges between +250°F and -250°F.

The cold soak condition induces a compression loading of the substructure, a tensile loading of the shield (which is critical for most shielding materials) and a compression and shear loading on the bond. The hot soak condition results in a compression loading on the shield, a tensile loading of the substructure, and a tensile and shear loading of the bond. This latter condition, while not critical for the shield or substructure, is usually critical for the bond and may become the limiting design condition for both the choice of the bond type and its thickness.

As a representative configuration, the shield-bond-structure combinations recommended in the recent Voyager study are considered (see Figure 5.3-1). This study was performed during 1963 by the General Electric Spacecraft Department under contract No. NAS-W696 for the NASA Office of Space Sciences, Washington, D.C. The study considered both Mars and Venus missions, and different types of shield-bond-structure combinations were recommended for each mission.

Recommended for the low heat flux, long term Mars entry condition was an elastomeric shield material (GE-ESM-1000) of 40-50 lb/cu ft density. This material consists of a discontinuous reinforced plastic honeycomb filled with a foamed silicone elastomer. Since each cell of the reinforcing honeycomb is slit and the filler

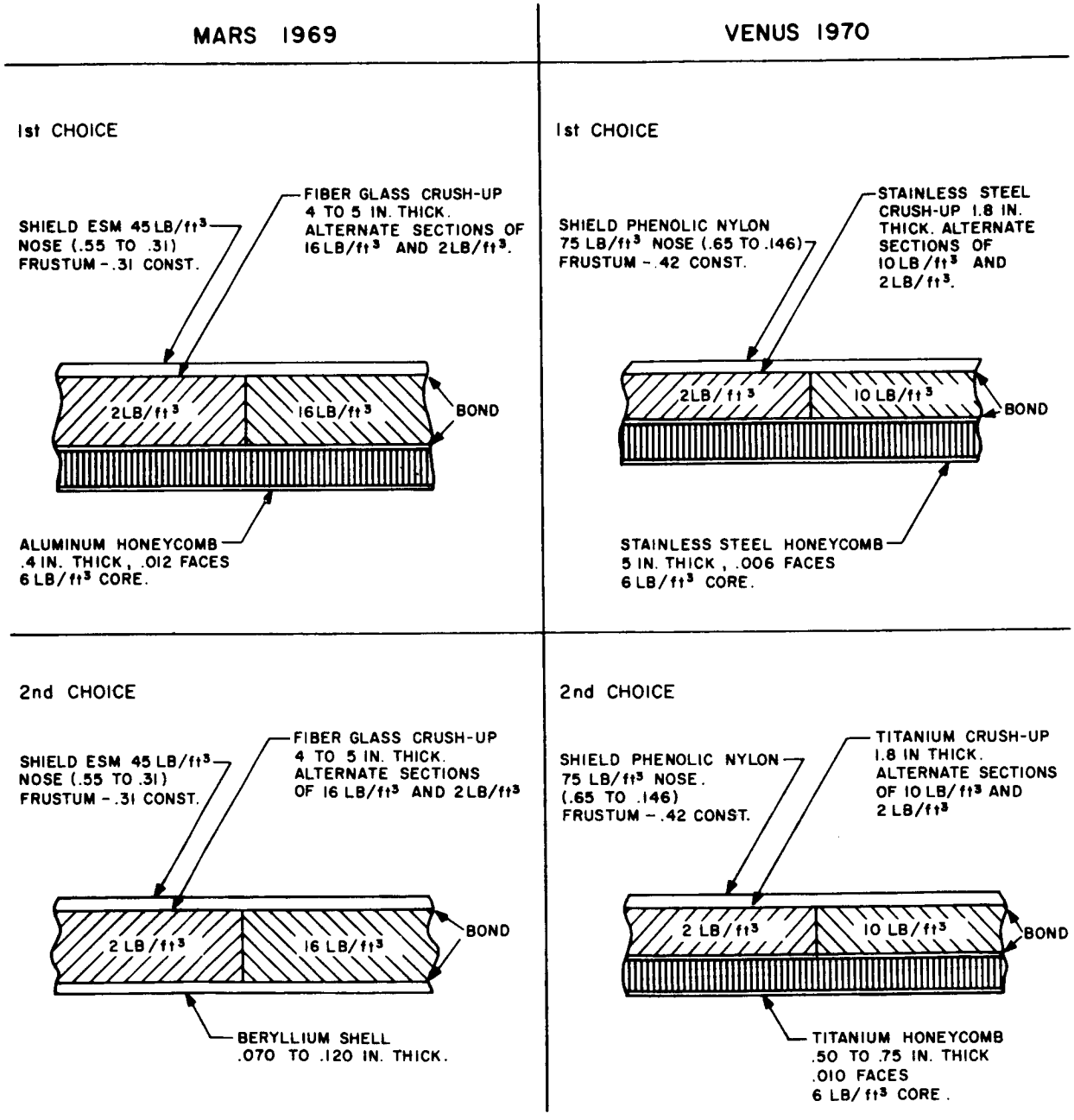


Figure 5.3-1. Typical Lander Cone Structures from General Electric Voyager Study (NAS-W696)

TABLE 5.2-3. REDUCTION IN THERMAL CONDUCTIVITY CAUSED BY EVACUATING THE AIR FROM VARIOUS MATERIALS* (100°F MEAN TEMPERATURE)

Material	Density (lb/cu ft)	Thickness (inch)	Thermal Conductivity at Atmospheric Pressure [Btu/(hr) (ft) (°F)]	Pressure (Microns Hg)	Thermal Conductivity of Reduced Pressure [Btu/(hr) (ft) (°F)]	Reduction of Thermal Conductivity (%)
Air Space between Cu surfaces		1	0.572	0.07	0.187	67.3
Corkboard	6.67	1	0.283	0.06	0.106	62.5
Corkboard	12.4	1	0.314	0.13	0.120	61.8
Cotton	0.78	1	0.288	0.07	0.072	75.0
Cotton	1.53	1	0.257	0.12	0.052	79.8
Cotton	2.62	1	0.248	0.14	0.033	86.7
Coarse glass wool	2.8	1	0.267	0.04	0.055	79.4
Fine glass wool "A"	0.58	1	0.2505	0.08	0.050	80.0
Fine glass wool "A"	1.94	1	0.2145	0.05	0.0195	90.9
Fine glass wool "A"	3.8	1	0.211	0.11	0.012	94.3
Fine glass wool "B"	0.78	1	0.302	0.04	0.665	78.0
Hairfelt	11.1	3/4	0.210	0.10	0.031	88.1
Kapok	0.26	1	0.366	0.10	0.091	75.1
Kapok	0.99	1	0.2445	0.10	0.0375	84.7
Kapok	3.96	1	0.244	0.12	0.0175	92.8
Mineral wool	7.7	1	0.258	0.05	0.037	85.7
Silica aerogel	8.6	1	0.1715	0.07	0.0255	85.1
Special specimen	3.1	1	0.231	0.00	0.0145	43.7
Wood fiber board	15.5	3/4	0.353	0.05	0.075	78.8
Wood fiber	3.5	1	0.2715	0.08	0.052	80.8
Wood fiber	6.9	1	0.277	0.09	0.033	88.1

*From: Rowley, F.B., et al, "Gas is an Important Factor in the Thermal Conductivity of Most Insulating Materials" Heating, Piping and Air Conditioning, Vol. 23 n/2, December 1951, p. 103-106.

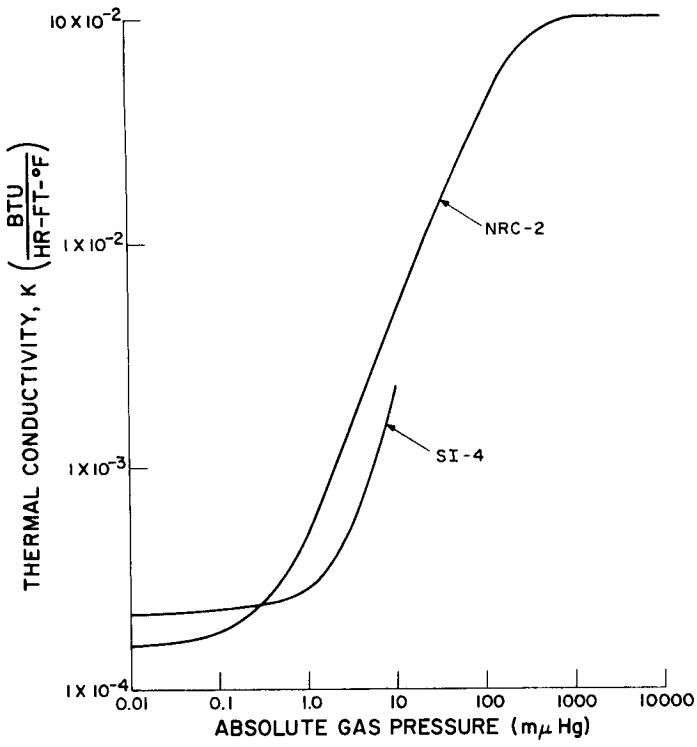


Figure 5.2-15. Effect of Gas Pressure on Thermal Conductivity of Super Insulation

The third alternative represents in all probability the lowest system weight solution although it makes control of the rate of heatup and cool down the most difficult. It may be noted that when thermal insulation is required for mission success then heating techniques or procedures compatible with this insulation must be developed.

5.3 STRUCTURAL ANALYSIS

For the model described in Section 5.1 several structural elements have been selected for analysis in this section. Using the general dimension of the model and relating these dimensions to expected loads and boundary conditions, these elements may be studied in a general way to obtain approximate stresses and deflections resulting from the sterilizing environment. No attempt was made to analyze the specific model shown. The model was used only to indicate typical problem areas and general boundary conditions.

is foamed, this particular shielding material is quite soft, especially at the higher temperatures. At the sterilization soak temperature, the material is strained. However, the modulus of elasticity of GE-ESM-1000 is so low at this elevated temperature that negligible stresses are induced in both the shield-bond combination and the substructure. This material is currently being subjected to extensive thermal cycling tests encompassing the sterilization range as part of its qualification testing. To date, no thermal problems have arisen with this material and, if the sterilization procedure weighs heavily in a tradeoff, this material would be a likely candidate.

For the Venus entry, however, the ablation and thermal requirements dictate the use of a more rigid plastic with an independent bond system. The system recommended utilizes a phenolic nylon shield material and a neoprene base bond. Here, the plastic is much stiffer than for the Mars mission, and the stress induced in the bond becomes significant.

Computer programs are generally used to solve this thermal stress problem. These programs are capable of calculating stress levels in the shield, bond (both rigid and soft) and substructure for a variety of sphere-cone-cylinder combinations with and without stiffener rings. If thermal loads vary longitudinally, circumferentially and radially, different programs are required but the results are easily superimposed. Also available are other thermal stress and distortion computer programs for space frameworks and shells which can be used in combination with the above heat shield programs to solve for compatibility stresses in the secondary structure.

A typical analysis shown in Appendix B indicates some of the general equations which form a basis for the rigorous solution. Of the configurations shown in Figure 5.3-1, the stainless steel construction is the most critical; for, while the coefficient of thermal expansion is higher than for titanium, resulting in a slightly smaller incompatibility between shield and structure, the modulus of elasticity is much higher, resulting in larger required bond loading to force compatibility.

The analysis includes several simplifying assumptions as follows:

a. Bond expansion is neglected.

b. Core shear deformation in both the crushup and the basic honeycomb is neglected.

c. The local effects of bending such as at rings and discontinuities due to radial gradients are neglected. The first two assumptions are conservative, but not overly so, and are adhered to in general design practice. The third assumption, while not conservative, is not particularly critical for general design, and can be omitted for simplicity. As previously stated, computer program routines available make provisions for these effects.

C. Conclusions

For a shield configuration associated with a Venus lander the bond thicknesses are compared for an in-flight soak at 250°F and a sterilization soak at 300°F. An additional bond thickness of .087-inch results from the sterilization requirement. (See Appendix B.) The weight penalty associated with this is 54.4 lb.

The analysis of Appendix B is meant to show a basic trend only. The actual weight penalty incurred by the sterilization would of course be minimized by a better choice of bond material. It should also be pointed out that the lander chosen does have a thick enough shield so that the cold temperature environment was not critical. In many cases the bond will be designed for the cold temperature environment and be of sufficient thickness so that the stresses imposed on the bond during sterilization are not excessive.

The analysis of Appendix B shows that the sterilization thermal environment becomes a governing design condition for lander entry shield-bond-substructure combinations. The sterilization environment may become the limiting design parameter in the choice of

(a) The bond type

(b) The bond thickness

(c) The shield material as affected by bond limitations.

5.3.2 BEAM ELEMENTS

A. General

The bending stresses, prior to thermal sterilization, of members primarily stressed in bending during powered flight, are small. If, for example, the design condition for flight is 20g, the bending stresses at the start of sterilization will be 1/20 of the flight stresses. In addition, it is common practice to design beams on the basis of maximum deflection or natural frequency considerations, rather than stress. This practice will tend to make the stresses at the start of sterilization even lower.

For a beam element (I section) whose dimensions are compatible with the typical lander model the effect of depthwise and spanwise thermal gradients have been considered separately.

B. Depthwise Gradients

1. Discussion

For this study, an I-beam was sized to withstand a uniform load which would be amplified by a factor of 20 in a boost environment. The beam was sized for simply supported and fully fixed end conditions. A simple I shape was used although it is recognized that in practice a fabricated shape composed of separate webs and flanges would probably be used. Another assumption made was that a natural frequency of not less than 100 cps would be required to be compatible with the 20g amplification. For space vehicles, this is representative of load criteria because of the vibrations encountered during the boost phase.

A thermal gradient of 100°F was assumed with the hotter side reaching the sterilization temperature of 300°F. The gradient assumed is not strictly linear. A ΔT across the flange and web junction represents a more realistic picture. The effect of this nonlinearity, however, is small for this type of beam. Thermal stresses were superimposed on a 1g static stress to obtain a gross effect during sterilization. Since the thermal stresses were equal, (dependent upon $\alpha E \Delta T$), and the static stress was a small percentage of design load, the resulting stresses were approximately equal for simply supported and fully fixed conditions. Curves of these results are shown in Figure 5.3-2. The analysis for this case is shown in Appendix C.

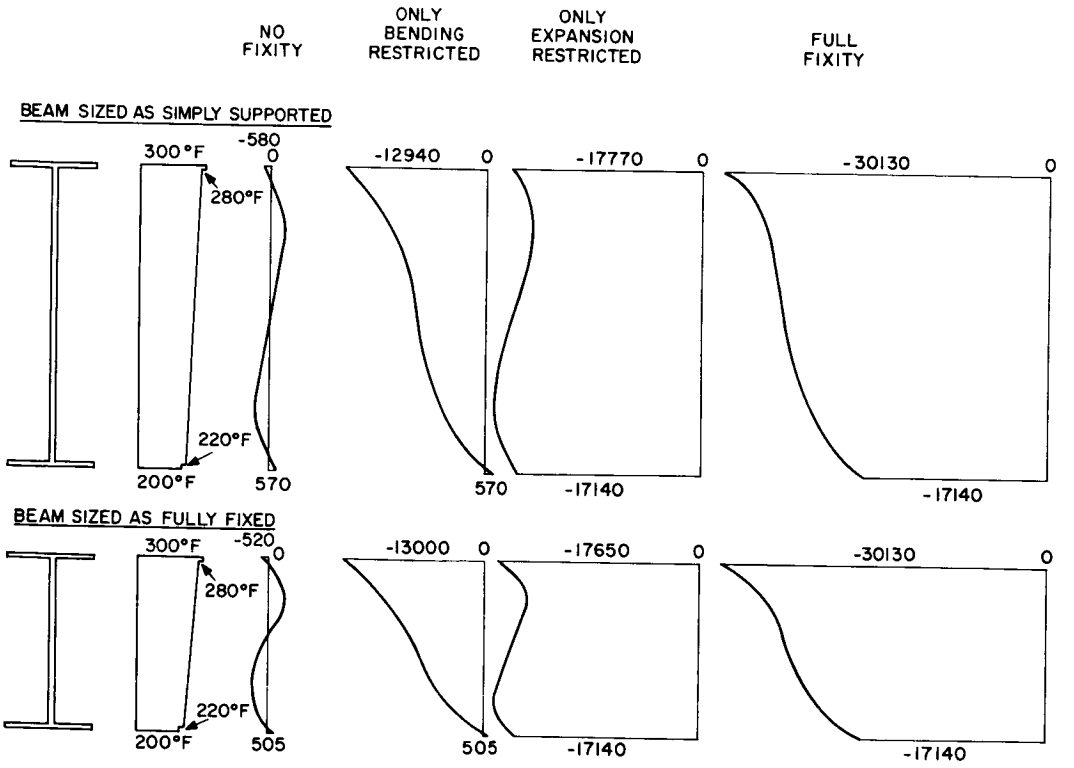


Figure 5.3-2. Stresses in Beams Due to Depthwise Gradients

2. Conclusions

For this case, it is readily apparent that superimposing 1g static loads on the thermal loads will not add an appreciable effect because they add only 1/20 of the original design load. Sensitivity to a thermal environment will, therefore, be a function of the percentage of design load that 1.0g represents.

The results are extremely sensitive to end fixity. With total fixity, the stresses are equal to (α E ΔT). As fixity is decreased, the stresses are in effect relieved and, as is shown in the example for simply supported beams, become rather low. The low stresses are the result of the non-linear gradient only. Assuming full fixity for 2024-T3 aluminum alloy construction, the stress level is α E ΔT or (13.1×10^{-6} in/in/^{o}F) x (230°F) x (10.0×10^{6} psi) = 30130 psi. Adding 1g stresses of 1750 psi brings the total to 31880 psi. For 1000 hours at 300°F, F_{cy}* equals 92% of room temperature F_{cy} which is .92 x 40000 = 36800 psi. For both local instability (such as flange buckling) and overall column instability this level is too high for efficient aluminum design.

A material such as titanium will build up a maximum thermal stress of α E ΔT = 4.6 x 230 x 15.5 = 16,400 psi. This material permits greater fixity and a higher percentage of 1g loading both of which may be required. Table 5.3-1 illustrates this point.

TABLE 5.3-1.

Material	Compression Yield Strength at Room Temperature, (psi)	Compression Yield Strength for 1000 hours at 300°F (psi)	Maximum Thermal Stress (psi)	Maximum Thermal Stress Compressive Yield at 300°F stress
2024-T3 Al	42,000	38,600	30,130	$\frac{30130}{38600} = 78\%$
6AL-4V Ti	126,000	100,800	16,400	$\frac{16400}{100800} = 16\%$

B. Spanwise Gradients

1. Discussion

As heat is applied the beam may expand against the resistance offered by the shell or shell rings. The resultant axial load will be superimposed on the bending loads. Strictly speaking the element will respond as a beam-column. Since the bending stresses may be small, it is likely that most of the response will be as a column.

* F_{cy} = compression yield strength

It may be useful to consider an interaction curve for a typical beam element, shown in Figure 5.3-3.

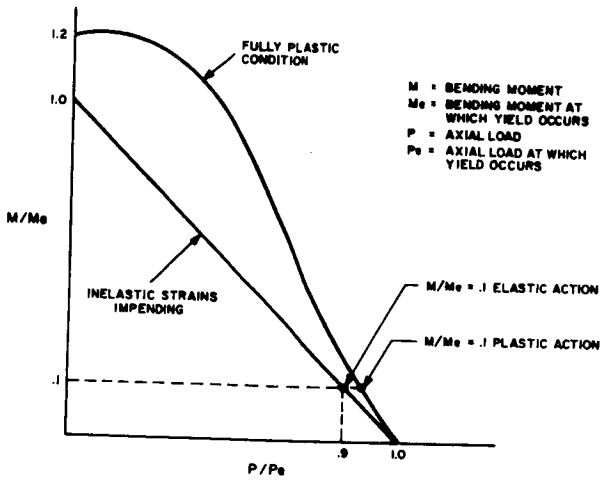


Figure 5.3-3. Typical Interaction Curve for I-Beam

For a 10g design condition M/Me = .1, hence P/Pe = .9. This indicates what is perhaps obvious, that the beam can tolerate a substantial additional axial load, provided of course that it has been stabilized against buckling or local instability. The ability to accept additional load could be increased by permitting inelastic strains to penetrate the beam. The gain is slight however as shown in Figure 5.3-3. The use of this type of curve may be useful in establishing upper limits for permissible axial loads.

2. Conclusions

Beam elements will respond primarily as columns to spanwise thermal loads during sterilization.

The use of interaction curves will be useful in determining maximum permissible loads.

5.3.3 COMPRESSION MEMBERS

A. General

The purpose of this study is to determine the effects of a sterilization environment on column design. Typical columns were selected which are representative of the structure which may be expected on a lander. The columns were optimumly designed for a given length and flight load environment. The critical temperature

gradient (ΔT) between the column and support structure, which would fail the column, was then determined. The analysis is shown in Appendix D.

B. Discussion

Column lengths of 20, 40 and 60 inches were assumed. For typical flight conditions, 1000, 2000 and 3000 pound loads were selected as representative. Only round tubes of 2024-T4 aluminum alloy were considered. Taking into account the degradation of material at 300°F for 1000 hours, the maximum permissible stress level for each of the resulting configurations was determined. From these stress levels, an allowable axial load was determined. Finally, the allowable axial load was expressed as an allowable temperature rise.

The restraining structure was assumed to be rigid and to have the same expansion coefficient (α) as the column. Therefore, ΔT is the temperature difference between column and restraining structure which will cause buckling.

A summation of the major assumptions is:

1. The columns are pin-ended.
2. Longitudinal restraint is total. This means that, as the column builds up load due to temperature, the restraining structure does not deflect due to that load. Actually, deflection of the restraining structure will minimize the load build-up in the column.
3. Material properties used to calculate the column strength at temperature were for 300°F for 1000 hours. At this point it is felt that 1000 hours is a conservative estimate.
4. Creep is not a factor. A check was made to determine the critical stress level assuming the maximum allowable creep to be a conservative 0.1%. The parameter used was T = temperature = 460 + 300°F = 760°R, C = constant dependent upon time and temperature at a constant stress level = 20 and t = time = 1000 hrs. The Larson-Miller parameter = P = T (c + Log t) = 760 x (20 + 3) = 17500. Critical rupture stress = 41,500 psi. Critical stress for assumed creep = 20,000 psi. The worst case ΔT for columns checked was 122°F and the resulting stress was 16,000 psi.
5. No eccentricities exist. This assumption is valid in that any existing eccentricities would be accounted for in the original design and the column sized accordingly. The difference between beam-column effects at room temperature and at 300°F is what is actually neglected.

C. Conclusion

The results of the analysis indicate that long columns designed for light loads at room temperature are most sensitive to a thermal sterilization environment. This may be explained in that longer columns are designed for lower stress levels and the critical ΔT varies directly with the allowable stress level. This may be shown in equation form as:

$$F_{ct} = \alpha E (\Delta T)$$

$$\Delta T = \frac{F_{ct}}{\alpha E}$$

where: F_{ct} = critical compressive stress at temperature

α = coefficient of linear thermal expansion

E = modulus of elasticity

ΔT = temperature change.

Table 5.3-2 shows a comparison of critical temperature differentials between restraining structure and columns as a function of column length and applied load. It is noted that the most critical ΔT obtained was 122°F. A comparison of this ΔT with values obtained in the thermal analysis of Section 5.2 indicates that 122°F is well above the maximum gradient which would be expected for a column installation. It may be concluded that for a range of columns likely to be used on planetary landers, the sterilization environment does not appear to constitute a limiting design condition.

It is of interest to note that for a material such as titanium the αE value is approximately 78, compared to aluminum with 131. By using titanium instead of aluminum columns the critical ΔT values would increase by 131/78 or 1.68. Titanium would offer the further advantage of an increased critical compressive strength at 300°F.

TABLE 5.3-2. CRITICAL TEMPERATURE GRADIENTS BETWEEN COLUMNS AND RESTRAINING STRUCTURE

Load (lb)	Column Length (in)	ΔT cr (°F)
1000	20	214
2000	20	240
3000	20	257
1000	40	154
2000	40	183
3000	40	202
1000	60	122
2000	60	149
3000	60	166

5.3.4 RING STIFFENED SHELLS

A. General

This investigation was concerned with discontinuity stresses in a ring-shell joint when subjected to a thermal gradient within a conservative range of sterilization temperatures. The analysis was limited to cylindrical shells and "Zee" type rings and to cases where shell and ring tend to separate under load. The analysis is shown in Appendix E.

B. Discussion

A cylindrical shell reinforced with a typical Zee stiffener was subjected to a thermal gradient of 150°F and analyzed for the resulting hoop and meridional stresses. A shell thickness of .10 inch and a typical Zee of I_x = .0148 in.^{4} was assumed. The results of this particular case were:

hoop stress = 4630 psi

meridional stress = 12,720 psi

To compare this single case with a similar case, a different stiffener (I_x = .0208 in.^{4}) was attached to a range of cylindrical shells having varying radii and thicknesses. The radii varied between 25 and 75 in. and the shell thicknesses between .010 and .15 in. A ΔT of 100°F was assumed. A parametric type solution is given for various shell radii and thickness while holding the properties of the stiffening ring constant. Actually, it is not expected that the ring would remain constant. However, within the limits of this study, it is felt that the effects of shell radius and thickness are of major importance and a parametric approach is applicable. Using the parametric solution (Appendix E) it was possible to obtain a factor for hoop and meridional stresses directly from a set of curves (see Figure 5.3-4).

To compare this general solution with the specific case discussed earlier, the curves were used to obtain stresses in a .10-inch shell with a 50-inch radius. The curves indicate that:

hoop stress $= k\,\Delta T = 52 \times 100 = 5200$ psi

meridional stress $= k\,\Delta T = 145 \times 100 = 14500$ psi

For the cases where the ring stiffness and thermal gradients varied, the stress levels are easily tolerated. Specific configurations will require careful attention but an approach similar to the one used for this analysis may be useful. The analysis and resulting curves are given in Appendix E.

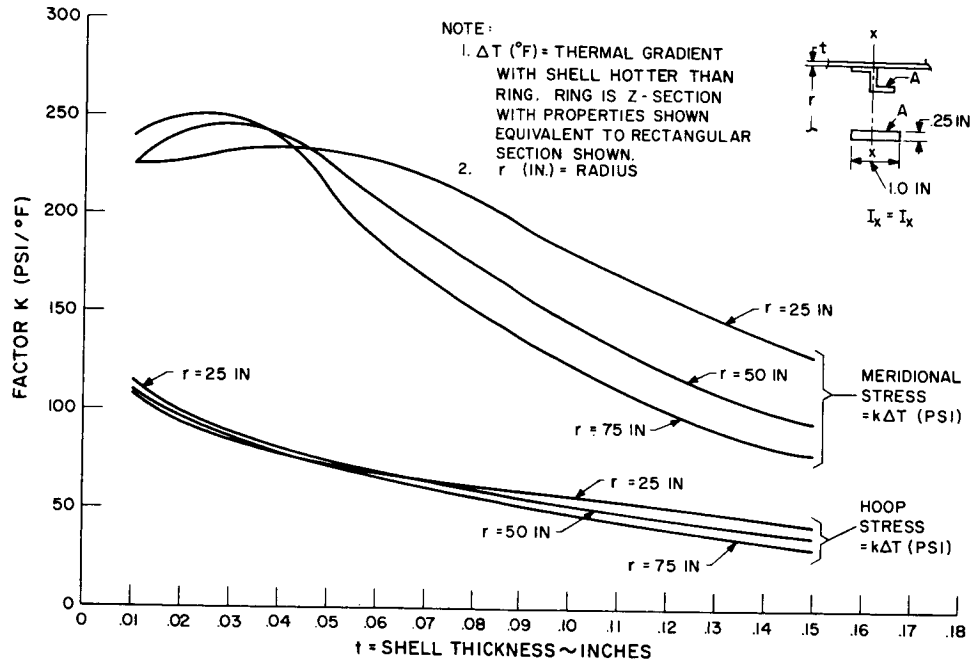


Figure 5.3-4. Thermally Induced Discontinuity Stresses for Ring-Stiffened Shells of Various Radii

C. Conclusion

The stress levels induced by the anticipated thermal environment are not in themselves large. The important consideration is that other stresses may be superimposed on these thermal stresses.

Although flight loading would not be applicable during sterilization, there is a possibility that pressure variation may have a significant effect. This is particularly true in the case of a containing canister if the thermal cycling is done in a vacuum.

Because of the wide range of variables which can affect stress levels, each specific design must be carefully investigated.

5.3.5 TANKS AND PRESSURE VESSELS

A. General

Pressure vessel design becomes an important consideration in structural design for sterilization. The problem arises of whether to design pressure vessels and pressurized systems to withstand a terminal dry heat sterilization at 150°C, or to sterilize the gas separately and charge the system after terminal heat sterilization of the vehicle and tank.

The aim of this specific study was to determine order-of-magnitude weight penalties imposed on pressure vessels when they are designed to withstand terminal heat sterilization, and to investigate alternate sterilization procedures which will eliminate the weight problem.

The study was based on spherical tanks containing gases which closely follow the perfect gas laws in the pressure-temperature range considered. Spherical tanks were selected as the most efficient and most frequently utilized containers for high pressure storage. No attempt was made to analyze imperfect gases and mixtures, or liquids. Items such as liquid propellants and coolants require individual consideration and were considered to be beyond the scope of this investigation.

B. Discussion

1. Terminal Heat Sterilization

Table 5.3-3 contains a list of possible pressurized systems and scientific instruments which may be used on future planetary landers. The type of gas and the volume used may vary somewhat for different vehicles; however, the values shown in the Table may be considered typical. Tankage weights are compared considering the system designed with or without a terminal heat sterilization requirement.

TABLE 5.3-3. POSSIBLE PRESSURIZING SYSTEMS AND INSTRUMENTS FOR PLANETARY LANDERS

System	Gas	Working Pressure (psi)	Pressure at 150°C (psi)	Tank Volume (in^3)	Tank weight without terminal sterilization (lb)	Tank weight for terminal sterilization (lb)	Weight Penalty (lb)
Spin Sterilization	N_2	3000	4300	600	6.8	12.2	5.4
Propulsion Pressurization	He	3000	4300	2000	23.2	41.6	18.4
Gulliver Life Detector (5.2)	CO_2	300	430	120	0.13	0.23	.10
Photoautotroph Life Detector	CO_2	300	430	225	0.26	0.46	.20
Gas Chromatograph (5.3)	He	2000	2870	90	0.67	1.20	.53
Gas Damped Linear Accelerometer	N_2	15	22	1.0	Negligible	Negligible	Neg.
Total					31.06	55.69	24.63

The weight comparison in Table 5.3-3 was based on the use of Titanium 6A1-4V for all tanks; however, approximately the same relationship will apply for steel or aluminum with an appropriate correction for material property reduction. The weight comparison was derived from the pressure increase and a 20% reduction of material properties which take place at a temperature of 150°C. No consideration was given to tank support structure, flight load requirements, thermal coatings, or insulation, any of which could contribute to an additional weight differential. These conditions would require individual consideration.

For pressurized tanks, the reduced properties at elevated temperatures govern while, for other components, the reduced room temperature properties after elevated temperature exposure are of prime importance. Figure 5.3-5 shows the reduction in tensile yield strength at temperature for three representative materials: stainless steel AISI 301, titanium 6A1-4V, and aluminum 7075-T6. MIL-HDBK-5 gives 100-hr data for titanium and aluminum but only 1/2-hr data for steel; however a material data search indicated little change in mechanical properties of the austinitic stainless steels subjected to longer exposure times. It is interesting to note that the reduction of tensile yield strength ranges between 11% for AISI 301 steel and 40% for 7075-T6 aluminum at 302°F (150°C), indicating that material selection is an important criterion for tank design.

Of the 24.63 lb or 80% weight increase, shown in Table 5.3-3, it is apparent that the major portion can be attributed to the large volume, high pressure tanks. Other factors such as tank support and ground handling limit the minimum gauge for the small low pressure tank walls which are used for the experiments, so that the weight increase required due to pressure rise is probably insignificant.

2. Alternate Tank Sterilization

Since a relatively high weight penalty is imposed on a planetary lander when the large volume, high pressure vessels are designed to withstand terminal heat sterilization with the tanks charged, it becomes advantageous to study other sterilization methods which may minimize or eliminate this weight penalty. An alternate method, which minimizes the weight penalty, is to heat-sterilize the vehicle with the tanks installed but not charged, then charge them with a sterilized gas.

This method imposes a sterile interface problem at the charge line coupling. If the biological barrier cannot be removed at the time the tanks are charged, a coupling would be required on the external surface of the barrier with feed lines connecting the barrier coupling to the pressure vessel. The feed lines could be disconnected prior to flight and stowed either on the lander or, preferably, on the inside surface of the barrier. The lines could be disconnected and stowed either remotely or, in the case of a flexible barrier, manually.

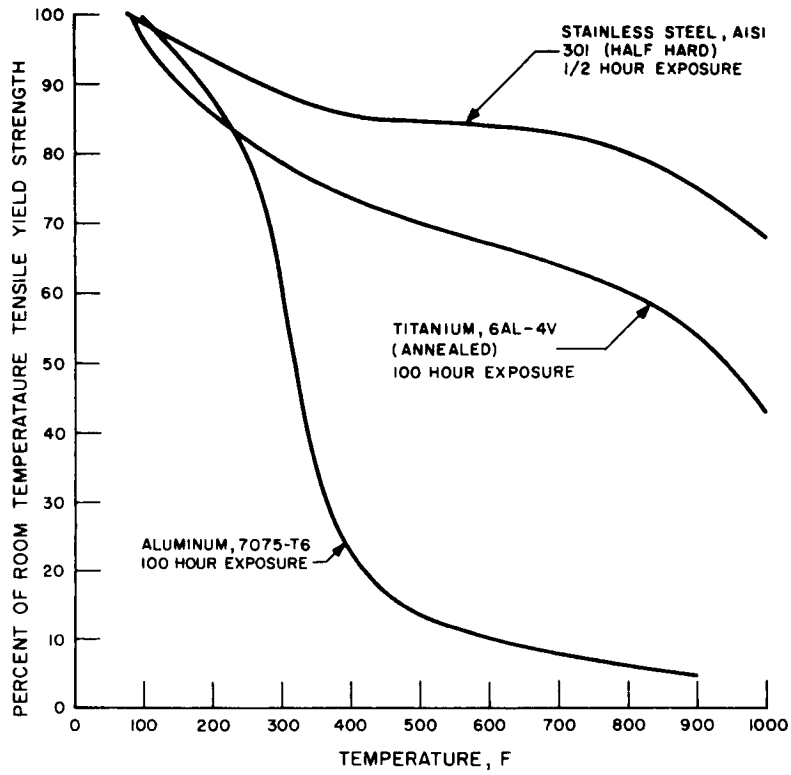


Figure 5.3-5. Effects of Temperature on Tensile Yield Strength of Typical Materials

If it is required that the barrier be removed in a sterile area after terminal heat sterilization, the tanks could be charged at this time. It is highly probable that this requirement will exist to facilitate installation of other equipment which is nc compatible with heat sterilization. A detailed study of the specific problems associated with tanks or pressure vessels would require a specific configuration.

3. Bus or Orbiter Sterilization

For the purpose of this discussion it is assumed that the sterile planetary lander will be enclosed and isolated from a delivery vehicle (bus or orbiter) by a biological barrier. To permit ejection of the lander, and to prevent the barrier from contaminating the planet, a portion of the barrier must be removed at some point between the boost phase and final lander velocity correction. If the biological barrier of the lander is removed prior to separation of the lander from the bus, consideration must be given to the bus attitude control gas. If attitude correction is required after the barrier has been separated, a high probability of contaminating the lander exists.

Figure 5.3-6 shows radial displacement versus pressure, thickness and radius parameters for various values of allowable tensile stress. Figure 5.3-7 shows radial displacement versus pressure, thickness and radius parameter for various values of modulus of elasticity in tension. Figure 5.3-8 shows radial displacement versus tank radius for a given temperature change of 230°F for various values of thermal expansion. The gradient ($\Delta T = 230°F$) is a never-exceed, worst case condition. The curves indicate the radial growth that occurs on an unpressurized tank. Figure 5.3-9 shows maximum bending stress versus a linear thermal gradient through the wall thickness for various values of the product of thermal expansion and modules of elasticity in tension. This indicates in a gross way the stress condition from gradients which must be added to the stress condition resulting from internal pressure. It is of interest that these stresses are independent of the tank wall thickness.

These figures are useful in preliminary design for fast evaluation of the stresses or deformations in spherical pressure vessels. For specific lander configurations where cylindrical or other shaped tanks are used, similar curves can be prepared.

In general, the radial growth of thin-walled, large diameter tanks should be analyzed to prevent damage to the tank. A thick-walled tank should be checked to avoid damaging the support structure. When designing pressure tanks and pressurized systems for a sterile environment, the following guidelines should be adhered to:

1. Decide during the preliminary design phase whether the tanks are required to withstand terminal heat sterilization in a charged condition. This decision should be based on the types and pressures of the liquids and gases to be stored. Personnel safety and weight penalty should be of prime concern.

2. If terminal heat sterilization is selected as the sterilizing method, consideration must be given to safety, tank materials, and material degradation at temperature, insulation, and thermal coatings. Tank supports must be designed to allow for increased growth at temperature. Considerable work has been done on tank support methods. Reference 5.5.4 contains a comparison of various support techniques.

3. If it is decided to charge the tanks with sterile gas or liquid following terminal heat sterilization of the vehicle, problem areas to be considered include material property degradation after extended 150°C temperature exposure, charge line interface design, and design and control of the sterilized gas or liquid charging equipment. The charging process should be added to the vehicle flow chart early in the design stage to allow for facility integration.

To heat-sterilize a gas attitude control system on the bus with the system charged would impose an approximate 80% penalty on the pressure vessels. Since it is not practical to sterilize the entire bus, the attitude control system would have to be heat sterilized, sealed, and installed in the bus as a unit. This method presents a severe safety hazard, and a difficult assembly problem due to the complex tubing runs.

Other methods of sterilizing the attitude control system might include the following. They could be used in entirety or in combination for particular applications.

1. Heat sterilize the system uncharged; seal; install in vehicle, and charge with a sterile gas. This procedure eliminates the weight penalty and the safety hazard. However, the system would have to be charged in a sterile lock to prevent contamination at the charge line interface connection.

2. Install the system uncharged and unsterilized in the vehicle, seal, then charge with a gas containing a sterilizing agent such as ethylene oxide. This method may require considerable research to develop a gas mixture which would fulfill both attitude control and sterilization requirement.

The contract work statement implies that terminal sterilization of the complete vehicle is an absolute requirement in this regard, it is of interest to note that the Propulsion Specification for the Mariner-C Vehicle included these requirements:

- The propulsion system shall be designed to be removable as a unit from the spacecraft.
- The propulsion system when loaded with propellant and pressurized shall be safe to work near.
- The propulsion system shall not require the services of liquid or gaseous umbilicals.

These requirements are of interest because of the apparent conflicts with sterilization requirements. These conflicts point out the need for further study in this area.

C. Design Guidelines

To assess the gross effects of the thermal environment on tanks or pressure vessels several general design curves were prepared. Their purpose is to provide analytical data to show the effect of the sterilization environment on pressurized versus non-pressurized tanks. Approximate weight comparisons can be made rather easily. It was assumed that the tanks were radially unrestrained and that they were spherical.

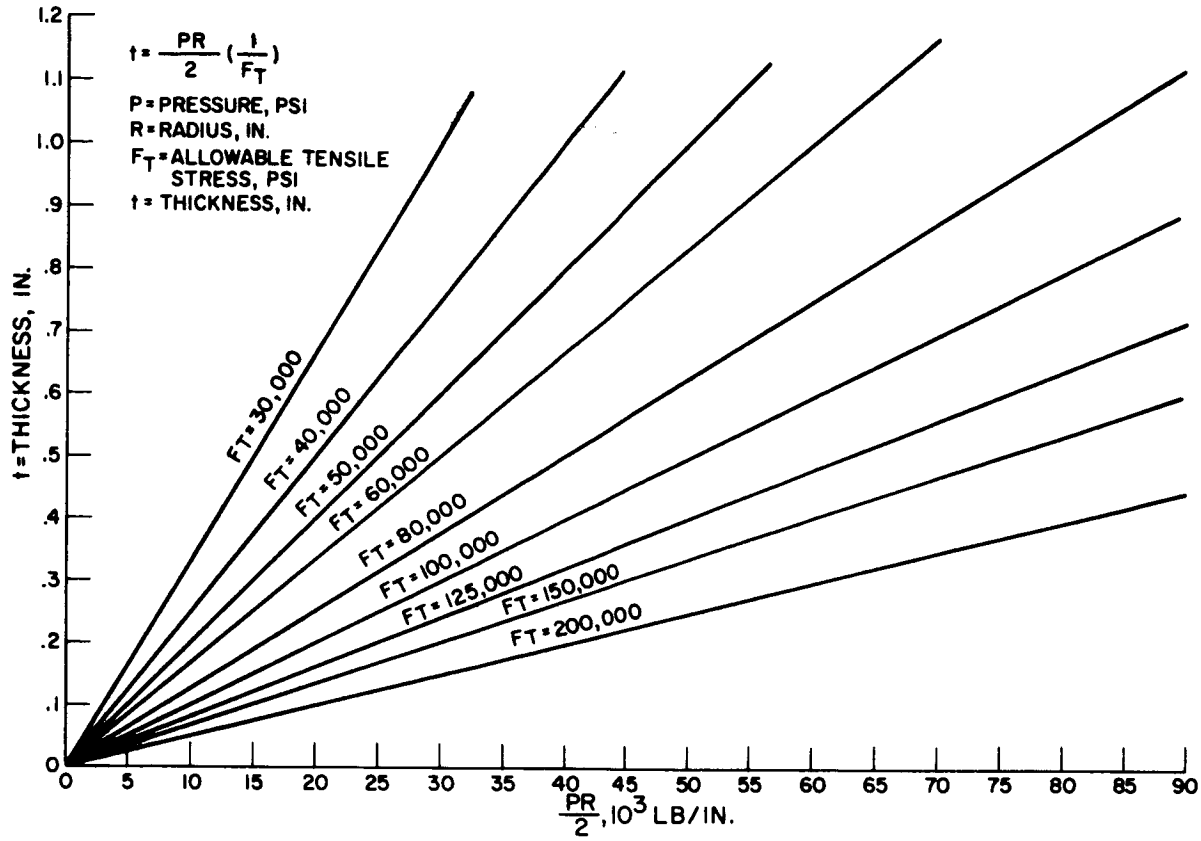


Figure 5.3-6. Spherical Tank Subjected to Internal Pressure - Required Wall Thickness Vs Pressure and Radius Parameter

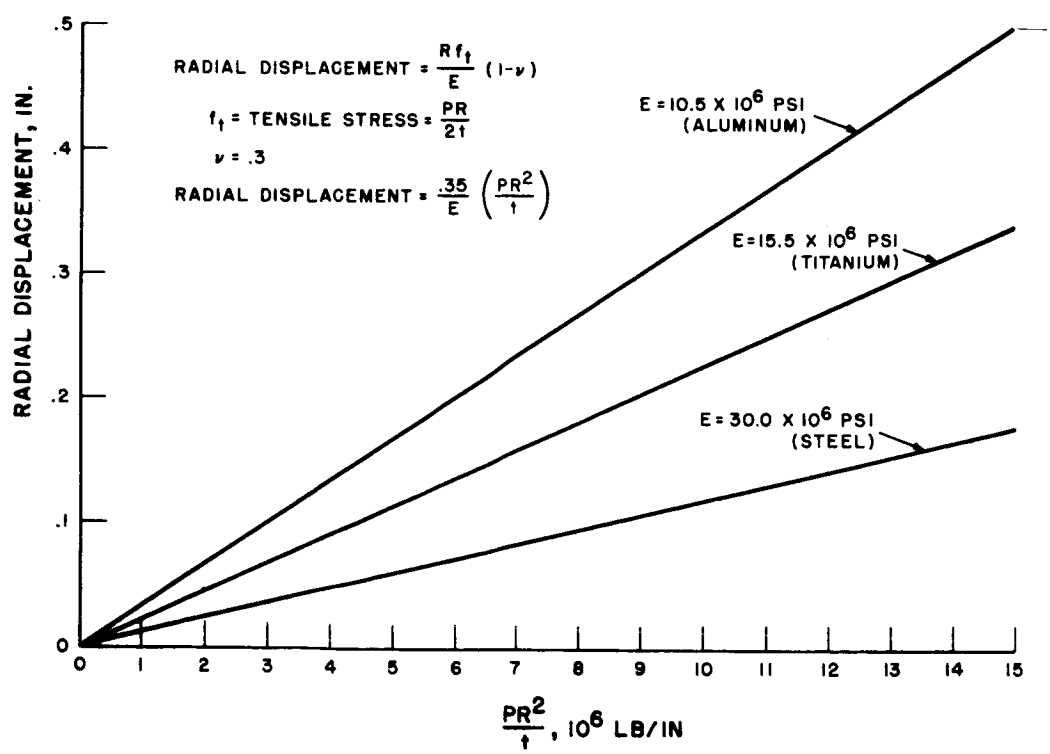


Figure 5.3-7. Spherical Tank Subjected to Internal Pressure - Radial Displacement Vs Pressure, Radius and Thickness Parameters

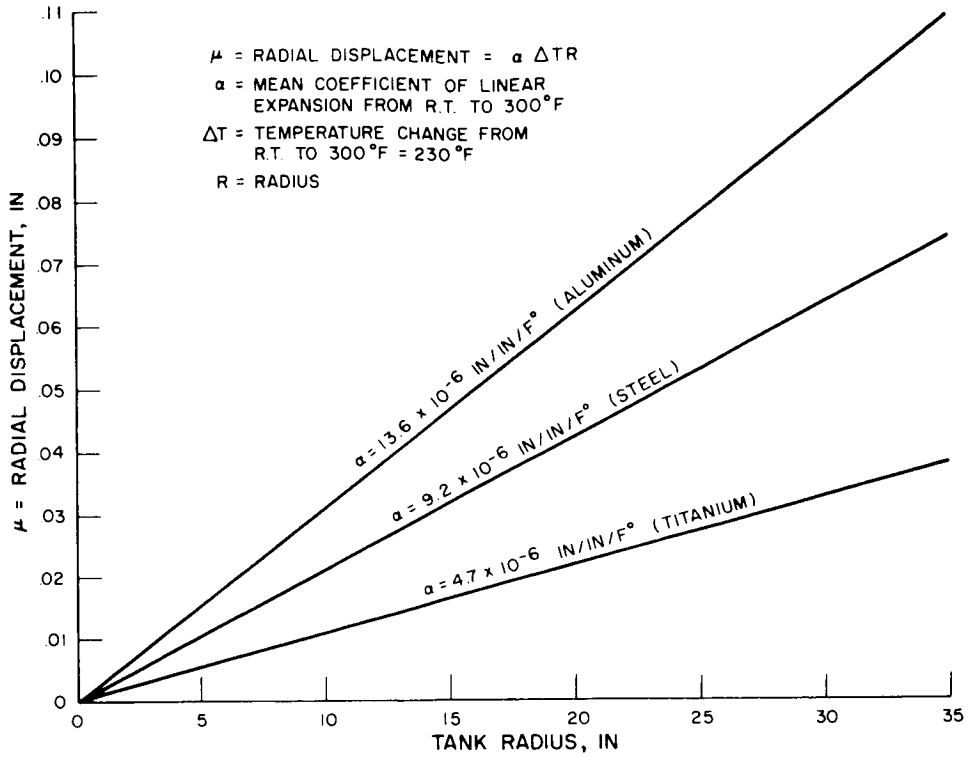


Figure 5.3-8. Unrestrained Spherical Tank Subjected to Temperature Rise of 230°F - Radial Displacement Vs Tank Radius

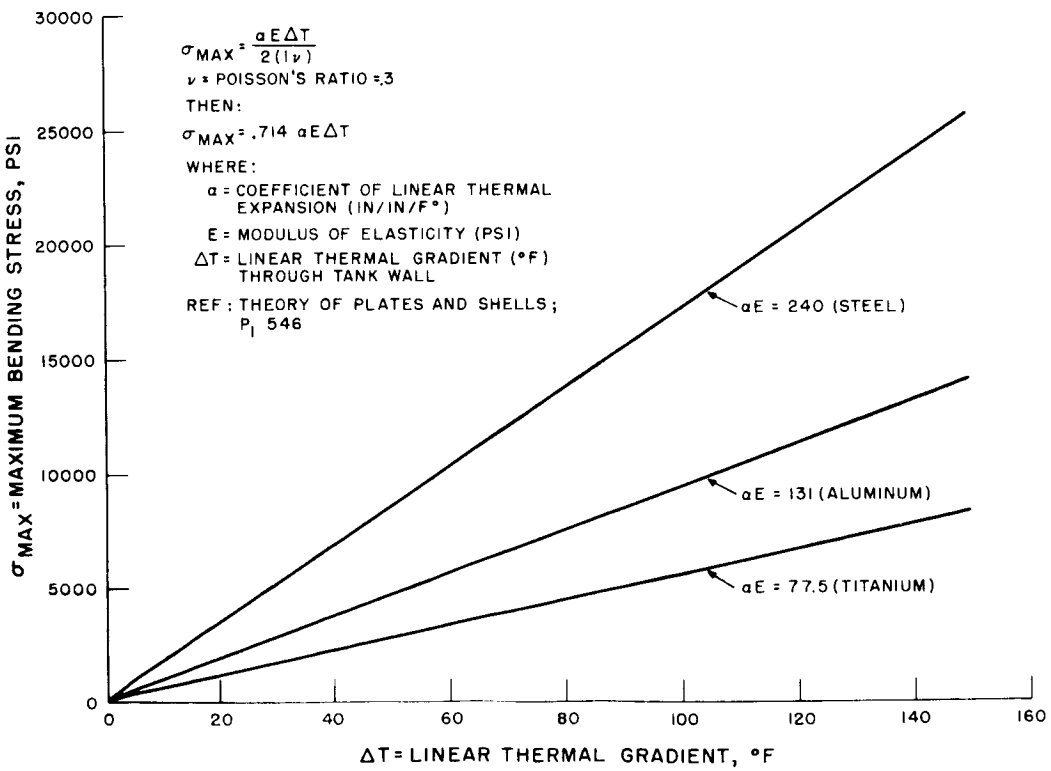


Figure 5.3-9. Spherical Tank Walls Subjected to Radial Temperature Gradient - Maximum Bending Stress Vs Linear Thermal Gradient

5.3.6 ALIGNMENT-SENSITIVE COMPONENT MOUNTING

A. General

Thermal sterilization causes thermal distortions in the vehicle. Current techniques for design and analysis in both elastic and plastic ranges are, in general, adequate to evaluate the stress conditions imposed by thermal sterilization. In the case of accurately aligned components, however, it becomes important to evaluate the effect of thermal sterilization on alignment accuracies or tolerances.

B. Discussion

In discussing design procedures at elevated temperatures, Gatewood (Ref. 5.5.8) states that the apparent way to avoid thermal stresses is to "allow enough deflection to absorb the thermal expansion." Expansion joints are suggested as a possible solution. For alignment-sensitive components on a typical lander, this solution is not practical. Another solution which is offered is to "let the inelastic portion of the stress strain curve provide the deflection to absorb the thermal expansion". From the standpoint of structural design or analysis this approach may be perfectly acceptable. However, such an approach will not be popular with system engineers.

As a purely hypothetical example assume that a sensor requires alignment with a vehicle axis to within $\pm 2.0^{\circ}$. Shop practices may permit alignment to $\pm .5^{\circ}$. Assume that inelastic strains resulting from terminal sterilization could cause a misalignment of $.25^{\circ}$ for each successive cycle. At the end of 4 cycles a misalignment of $\pm 1.5^{\circ}$ could occur. This misalignment is still within acceptable limits as specified. Since the actual extent of misalignment would never be known without inspection, it is unlikely that system engineers would be receptive to this approach. The alternative procedure is to pay the weight penalty associated with design in the elastic range.

For components where alignment is critical, no deflections beyond the elastic range of structural supporting members should be permitted. This implies structural inefficiency. In actual fact many members will be sized by deflection or natural frequency considerations, and the penalties associated with elastic design may be negligible.

A more difficult problem is the anticipation of deflections which result from such factors as:

1. Creep
2. Residual stresses in materials resulting from forming of detail parts.

3. Stresses caused by assembly (e.g., riveting jigs, welding fixtures). Each vehicle will require careful attention to ensure that deformations from all causes are compatible with alignment requirements.

To ensure that critical components such as rocket engines and sensors are correctly aligned following sterilization, it may be necessary to perform alignment checks following sterilization. Certain types of checking could be accomplished using windows or ports in the containing canister. Removal of the lander from the container for alignment checking will defeat the purpose of terminal sterilization unless such removal is performed in a sterile hood or enclosure.

C. Conclusions

Alignment-sensitive components should not be located on structural elements subject to plastic deformations. Deformations resulting from all causes should be considered. Techniques and procedures for verifying alignment of critical components require additional study. The use of ports or windows in the canister may be a practicable solution.

5.3.7 PLATES AND PANELS

A. General

A typical lander will include plates or panels of sheet or sandwich material. The wide possible range of variables such as material, overall dimensions, and thickness, prohibits anything other than a general approach.

The representative material selected for this analysis is 2024-T3 aluminum. The following ranges of panel dimensions were considered.

- aspect ratio (a/b): - 1.0, 1.5 and 2.0 in.
- shortest edge (b): - 3.0, 6.0, 9.0 and 12.0 in.
- thickness (t): - 0.02 to 0.14 in.

B. Analysis

For this range of panel dimensions and thicknesses, two cases were considered: (1) all sides simply supported and (2) all edges clamped. For these conditions the thermal gradients (ΔT_{cr}) required for buckling were calculated. This analysis is shown in Appendix F. Although component mounting boxes are not normally considered as structure, the effects of thermal gradients on these boxes are an

important supplement to the panel analysis. The structural response to the thermal environment for a typical component was evaluated in Appendix F using a matrix approach.

C. Conclusions

The analysis indicates that, for all panels considered, gradients as low as 3° to $5^{o}F$ will cause thermal buckling in a range of realistic panel sizes. Although buckling may occur it does not necessarily follow that the structure then fails. On cool-down, many of the thermally induced buckles and wrinkles may vanish. As a general rule, however, flat plates or panels should not be used for mounting alignment-sensitive components.

The analysis shows that the stresses attributable to thermal sterilization of component boxes will be low, and no problems are foreseen in this area.

5.3.8 EXPLOSIVELY ACTUATED FASTENERS

A. General

On any planetary lander there will be a requirement for remotely controlled structural fasteners. Several of the areas which may require this type fastener will include: bus/lander separation, biological barrier separation, rocket ejection, parachute deployment, and scientific instrument deployment. In all cases the joints must be designed for two prime functions: to withstand powered flight loads, and to perform mechanical separation. Fasteners have been designed which are actuated electrically, hydraulically or pneumatically; however, on space vehicles where requirements of minimum weight and power and maximum reliability predominate, explosively actuated fasteners have proven to be superior.

A typical explosive nut installation used to support and eject a boom-mounted instrument is shown in Figure 5.3-10. During powered flight, mid-course correction, and entry, the bolted joints support the boom. After impact an electrical signal activates the explosive nut, thus freeing the boom from a rigid support and allowing the instrument to extend to its working position.

B. Terminal Heat Sterilization

Terminal heat sterilization imposes a new requirement on the design of explosive devices. A literature search, including brochures from explosives manufacturers, indicates that very little testing has been done on explosives at elevated temperatures for extended time periods. Most present devices are designed for storage and operation at a maximum temperature of $165^{o}F$.

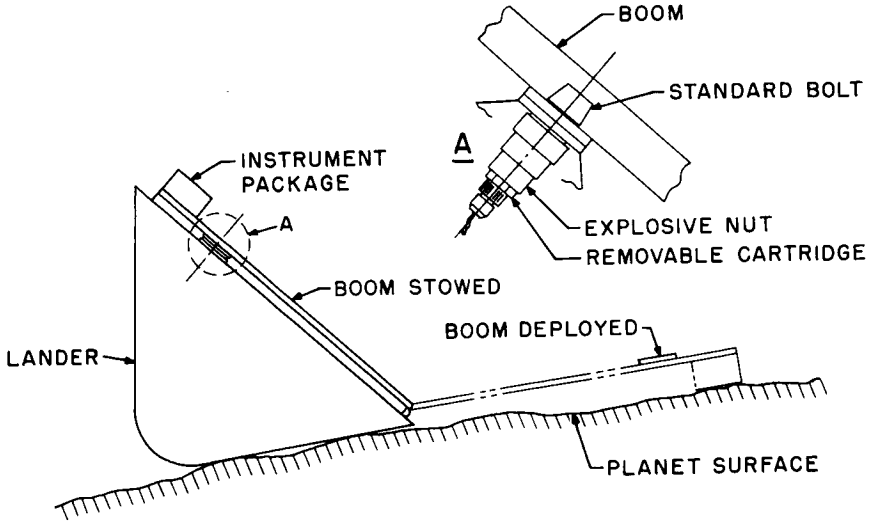


Figure 5.3-10. Simplified Typical Explosive Nut Installation

Time-temperature testing which has been performed may be exemplified by a series of tests by DuPont (Ref. 5.5.5) on (FLSC) Flexible Linear Shaped Charge. Soak temperatures ranged from 230°F to 360°F. The maximum time at temperature was 24 hours. Results of the tests indicate a definite loss in cutting efficiency for the longer exposure times.

With a lack of back-up time-temperature test data, the opinions of various explosive manufacturers and explosive experts vary considerably as to the ability of present state-of-the-art explosive devices to withstand a 150°C sterilization soak for 60 hours or more. A test effort should be initiated to determine the feasibility of heat sterilizing explosives.

Test results may show that the explosive charge must be over-designed to allow for degradation due to the sterilization temperature soak. This could be a critical factor in the reliability of the fastener. The extent of overdesign may be directly dependent upon the soak time, such that any deviation from the soak time to which the fastener is qualified may substantially reduce the reliability.

C. Alternate Sterile Installation

The limiting item on heat sterilization of explosive fasteners is the explosive powder. The mechanical and electrical portions in most existing explosive devices are satisfactory for a heat sterilization environment. If the explosive powder is not capable of heat sterilization, it may be possible to sterilize the powder chemically and install the charge after terminal sterilization of the vehicle. To evaluate the installation problems, explosive fasteners are divided into three types.

1. Fasteners with Integral Explosive Cartridge: This type includes bolts, nuts, pistons, pin-pullers, wire-cutters, etc. The cartridges are welded, soldered or machined integrally with the fastener and cannot be physically removed.

   Installation of this type fastener after terminal sterilization may involve considerable effort. If the fastener is used as a load transfer point during assembly and handling, a substitute fastener or fixture would be required prior to and during vehicle sterilization. After sterilization the substitute fastener would be replaced by the prime explosive fastener. Depending upon the number, accessibility and method of attachment, a major reassembly of the sterile vehicle may be required. This is undesirable since only a minimum amount, and preferably no work at all should be required on a vehicle after it has undergone sterilization.

2. Fasteners with Removable Explosive Cartridges: Fasteners of this type are identical to type 1 except that the cartridges are threaded into the fastener housing. The obvious advantage of this type is that the cartridges may be installed after vehicle sterilization without disturbing the fastener; however, access is still required to the sterile vehicle.

3. F. L. S. C. (Flexible Linear Shaped Charge): F. L. S. C. is normally used for major separations such as stage or bus/lander separation, where its ability to cut load carrying skin can be utilized. The advantage is that it may be used to sever an integral structure through an evenly distributed load path without the interruption which is required by a multiple fastener pattern. For a typical shaped charge installation see Figure 5.3-11.

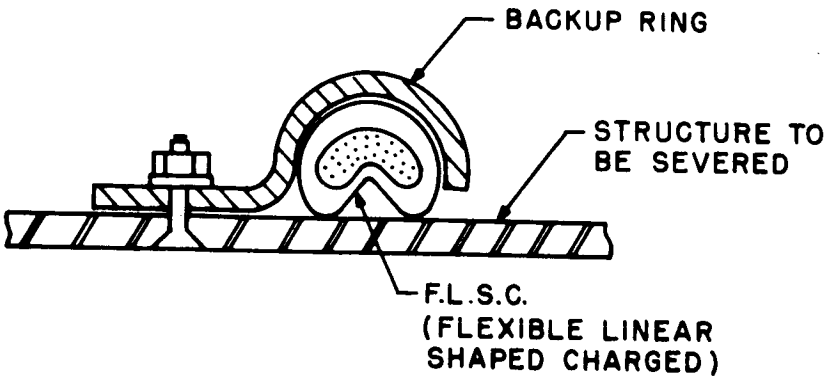


Figure 5.3-11. Typical Shaped Charge Installation

Due to the size and area covered by a shaped charge installation, the magnitude of the installation problem after vehicle sterilization is greater than with the other two types of explosive fasteners. The F. L. S. C., a back-up ring, and in some instances a damping material, must be fastened to the structure by mechanical fasteners. This installation requires maximum vehicle access and a considerable amount of mechanical assembly on a sterile vehicle.

D. Design Guidelines

Explosive fasteners are an efficient, proven method of remote joint separation. Their use on planetary landers should not be eliminated unless a test program proves that explosive charges cannot withstand heat or chemical sterilization.

The most reliable approach appears to be chemical sterilization of the explosive charge and installation after heat sterilization of the vehicle. If this method is used, the two requirements which should be considered during design of the vehicle are access and installation procedure. The order of preference for fasteners to minimize these requirements are:

1. Fasteners with removable explosive cartridges
2. Fasteners with integral explosive cartridges
3. Shaped charges

5.3.9 RELAXATION OF HIGHLY STRESSED COMPONENTS*

A. General

Relaxation defined as the release of stress in structural elements that slowly deform by creep or other phenomena is a problem studied in turbine flanges in the thirties. Bolts of a turbine flange are drawn tightly in assembly by heating and tightening so their contraction on cooling will give added pressure. The turbine subjects the bolt to a combination of stress and high temperature that causes slippage of the bolt. Stresses in the bolts are slowly relieved and distributed, gradually loosening or relaxing the bolts, which can cause leakage between the flanges. Relaxation as it occurred in turbine bolts usually is significant at elevated temperatures. However, relaxation in highly stressed structural members has been observed at room temperature. A mechanical strain measuring

* The analysis and experiments described in this section are based on work performed by the Advanced Technology Laboratories of the General Electric Company under Contracts NAS 8-4012 and 8-11523.

instrument attached to a tensile specimen will cause marked indentations in a specimen strain-cycled above the yield stress. This observation led to the investigation which studied the effect of lateral cyclic motion imparted to a gasket stressed above the yield stress. The objectives of the investigation was to determine the per cent change in gasket thickness as a function of lateral cycles.

Although not specifically applicable to the lander model, the investigation is nevertheless of interest. Any lander will contain highly stressed elements in the form of gaskets, shims or spacers, and this approach to testing may be useful.

B. Test Results

The gasket was simulated by a copper strip 1/16 x 1/8 x 1-1/2 inches long with the load applied normal to the 1/8 x 1-1/2 inch face as shown in Figure 5.3-12. The copper strip was placed between two anvils being pressed together by the normal load. Five tests were conducted with compressive loads producing 12,250; 15,650; 20,000; 25,600; and 40,000 psi stress in the copper strip. For each test a new copper strip with a 0.2% offset yield stress of 7,770 psi was used. When the normal load was applied, one anvil was moved 0.0005 inches laterally with respect to the other. A complete forward and backward movement of the anvil constituted a cycle. After each 1/4 cycle, the change in gap between the anvils was recorded and the load which decreased or relaxed was returned to its initial value. The reduction in load was greatest during the first few cycles, being as high as 3% for each 1/4 cycle. The results of these tests are shown in Figure 5.3-13.

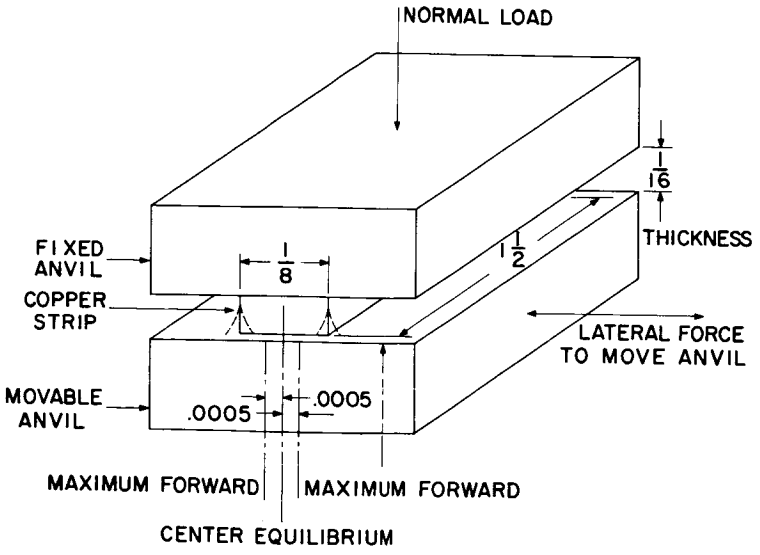


Figure 5.3-12. Schematic of Test Fixture

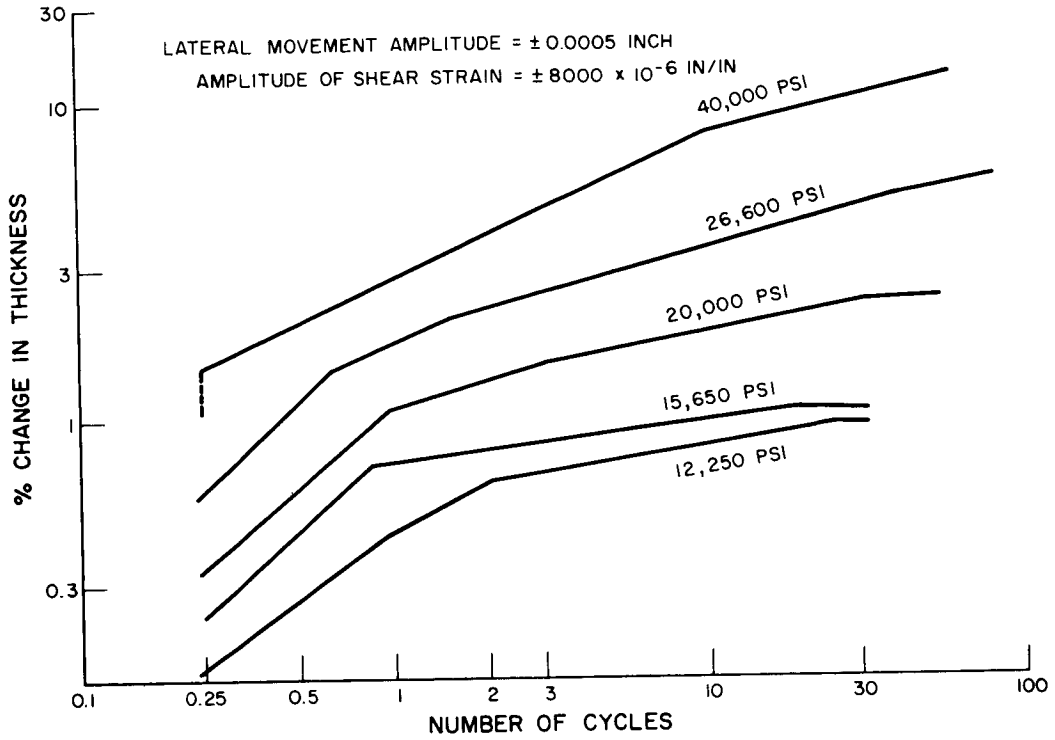


Figure 5.3-13. Thickness Change Vs. Loading Cycle

Figure 5.3-13 is a log-log plot of the per cent change in thickness of a rectangular copper strip as a function of lateral displacement cycles. The per cent change in thickness was obtained by dividing the change in thickness after the load was applied by the original thickness of the copper strip before a compressive load was applied. It can be noted that changes in slopes of the curves occur in the vicinity of the first cycle and in the vicinity of 20 and 30 cycles for compressive loads of 12,250; 15,650; and 20,000 psi. Three distinct regions having different rates of deformation are seen. First, the largest change in thickness occurs during the first cycle. In the second region the rate of thickness change has decreased but still is changing, and in the third region the change of thickness ceases to occur. G. J. Moyar and G. M. Sinclair also observed three distinct regions when they subjected a hollow tube under high axial stress to a cyclic torsional stress. They called the regions: (1) Initial deformation which is analyzed using theories of plasticity, (2) transient strain, and (3) steady state.

The lateral forces required to move the anvil in each of the tests is given in Table 5.3-4. For each test there were four copper strips between the anvil. Values given are the maximum and minimum loads required to displace the center anvil 0.0005 inch. Two trends were evident during the tests. First,

as the number of cycles increased, the forces required to move the anvil increased leveling off in the steady state region. Secondly, a larger force was required to move the anvil away from the center equilibrium position than to move the anvil from an extreme position toward the center equilibrium position. The difference was as much as 200 to 300 pounds.

TABLE 5.3-4.

Normal Stress (psi)	Minimum Load (lb)	Maximum Load (lb)
12,250	1100	1650
15,650	1300	1800
20,000	1725	2400
26,000	2100	3150
40,000	2700	5100

The results presented in Figure 5.3-13 were based on tests which did not exactly match the environment of a gasket in a pipe joint. A gasket is usually circular instead of straight. The circular shape is more resistant to radial or lateral movement than a straight strip. Also, in a gasketed joint the load is not maintained constant when the gasket deforms. Loadings on the gasket will relax. An expected reduction in thickness would be less under the actual case than under the tested condition. However, any effect of load relaxation on gasket thickness is not expected to be seen in the first cycle. Therefore, it can be concluded that, if the forces on the gasket cause a lateral movement of 0.0005 in., a change in thickness of 1/2 per cent can be expected based on the results of these tests if the normal compressive load on the gasket is twice the yield stress.

5.3.10 THERMAL DISTORTION*

Changes in temperature of a structure can result in stress and distortion when the coefficient of expansion is not the same for all members of the structure or when the temperature change is not the same in all parts of the structure. In order to prevent excessive stresses in pipe lines or boosters, it is customary

* The discussion in this section is based on work performed by the Advanced Technology Laboratories of the General Electric Company under Contracts NAS 8-4012 and 8-11523. The TIS (Technical Information Series) Numbers are document numbers for work on these contracts.

to include expansion joints. Similar design considerations may be included elsewhere in a vehicle structure to prevent excessive stresses and possible permanent structural damage.

An idea of the stresses and deformations in local elements of a structure can be obtained by considering results in Section 48 of General Electric Report TIS 63GL44 for fluid connectors used in boosters.

There, consideration was given to the stresses and growth in radius and thickness of a flange having a radial variation in temperature. It was found that, for given temperatures at the inner and outer surface of the flange, the stresses at these surfaces and the radial growth were relatively insensitive to the exact shape of the temperature distribution curve. When the radial distribution of temperature can be described by the function $Ar^n + B$, where A, B, and n are given by Section 61.2 of General Electric Report No. 63GL46 and r is the radius, the method in Section 48 of 63GL48 gives analytical expressions for stress and distortion.

In an example (page 48-5 of 63GL44) of a very rapidly heated flange with an inner radius of 1 inch, an outer radius of 2 inches, and a thickness of 1/2 inch it is found that a temperature change of $500^{o}F$ at the inner wall and $100^{o}F$ at the outer wall causes a stress of -29, 100 psi at the inner wall and a stress of 10,800 psi at the outer wall; a radial growth of .00207 inch at the inner wall and .00414 inch at the outer wall; and a change in thickness of .00292 inch at the inner wall and .00034 inch at the outer wall. Because of the much lower heating rate in the case of sterilization, it must be expected that the stresses would then be only a fraction of those given above. Nevertheless, even these stresses are well within the capabilities of most structural materials.

5.4 DESIGN CONSIDERATIONS — CONTAINING CANISTER

5.4.1 GENERAL

The concept of a containing canister which can serve as a biological barrier has been established through studies by industry and the NASA. This barrier must satisfy numerous design requirements and must be compatible with the sequence of events prior to planetary landing which the vehicle experiences. As an example of the design considerations which might be required for a typical lander, a list of design considerations associated with the biological barrier from an earlier General Electric study is attached as Appendix G.

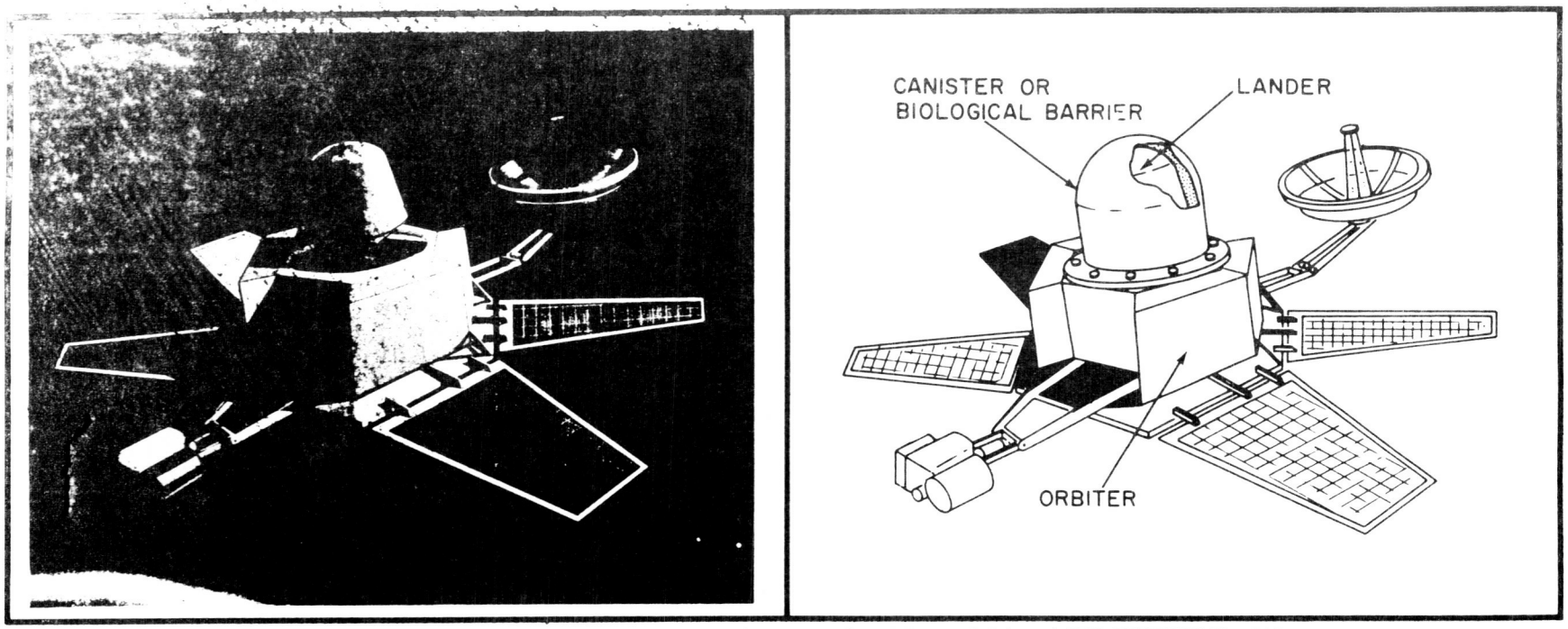
Figure 5.4-1 compares the configuration of an Advanced Mariner Vehicle and a Voyager Vehicle. The relationship between vehicle lander and containing canister is made apparent in this figure. To demonstrate the relationship between a containing canister and the lander vehicle used for this study, see Figure 5.4-2. The comments which are made subsequently are based upon this general model of a canister.

With this information as general background material, a number of important and potentially troublesome, problem areas can be identified. These problem areas fall within several categories which will be discussed separately.

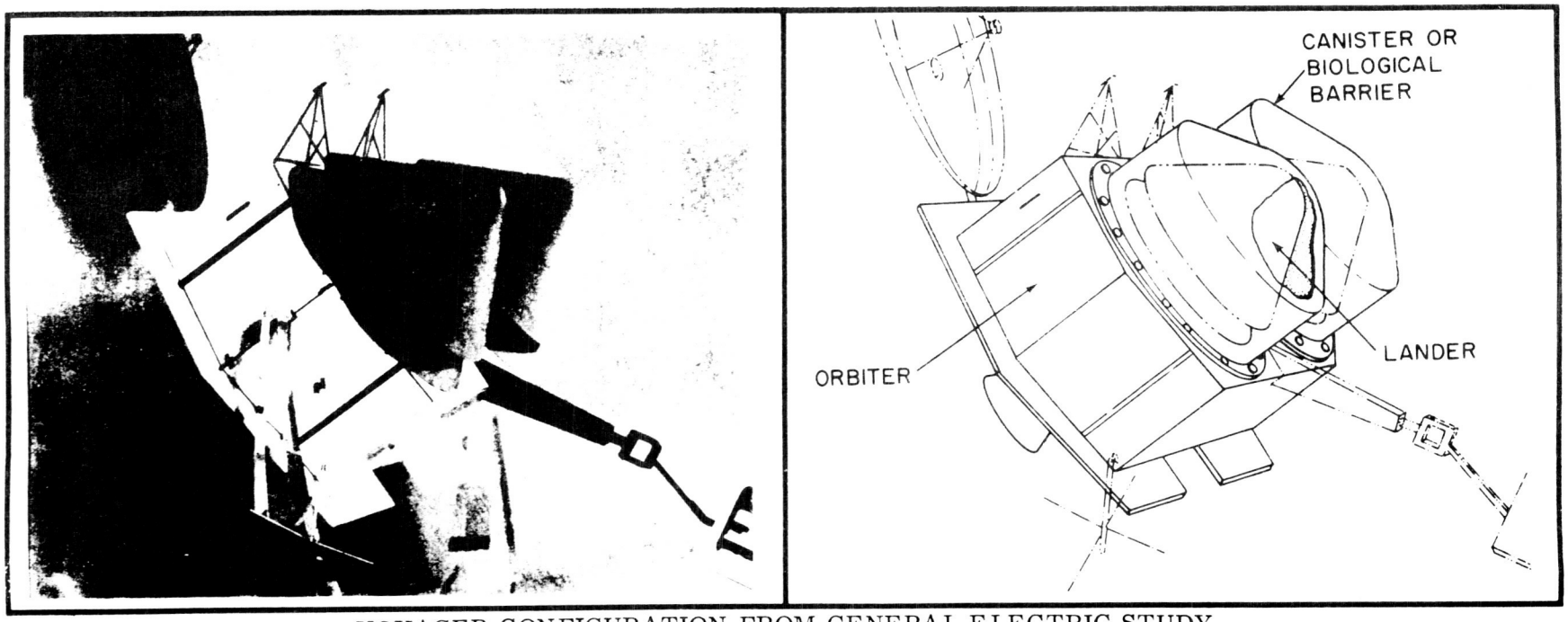
1. Storage and handling of vehicle in protective container.
2. Container weight penalties associated with terminal sterilization.
3. Heat transfer relationships between container and lander.
4. In-flight separation problems resulting from biological barrier.
5. Technique for maintaining positive pressure within sealed container.
6. Problems associated with a flexible container.

5.4.2 STORAGE AND HANDLING

If a vehicle is required to undergo extensive handling, shipping or storage, special provisions must be factored into the structural design. Thin gage materials must have sufficient strength to prevent handling damage. Minimum weight structural design may be impractical from the standpoint of the handling requirement. As an example of the variations in requirements which might be imposed on two identical planetary landers, consider Table 5.4-1 for a hypothetical situation.



ADVANCED MARINER CONFIGURATION (AFTER NASA FACT SHEET D-62)



VOYAGER CONFIGURATION FROM GENERAL ELECTRIC STUDY

Figure 5.4-1. Comparison of Typical Advanced Mariner and Voyager Configurations Showing Relationships Between Lander, Orbiter and Canister

CANISTER WEIGHT COMPARISON

(Ref. Section 5.4.3)

CONDITION	DESIGN CRITERIA	MATERIAL	SKIN GAGE (in.)	(1) GROSS WT. ESTIMATE (LBS.)
I	STERILIZATION	STAINLESS STEEL	.0053	103
I	HANDLING	6061-T6 ALUMINUM	.030	172
II	STERILIZATION	6061-T6 ALUMINUM	.0039	48
II	HANDLING	6061-T6 ALUMINUM	.030	172
III	STERILIZATION	6061-T6 ALUMINUM	.156	892
III	HANDLING	6061-T6 ALUMINUM	.030	172
IV	STERILIZATION	6061-T6 ALUMINUM	.0039	48
IV	HANDLING	6061-T6 ALUMINUM	.030	172

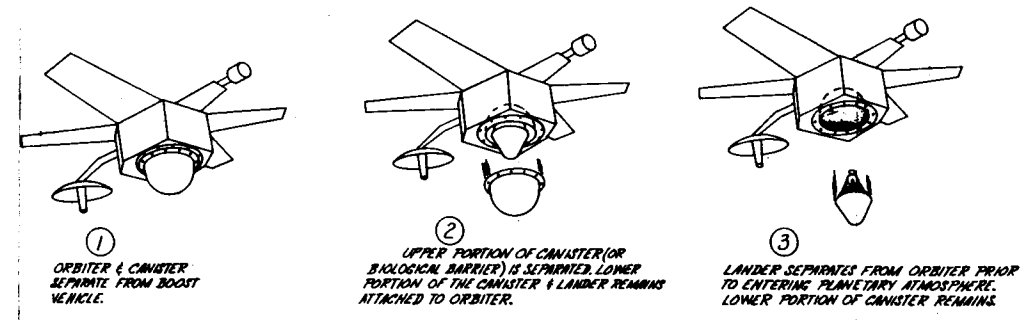
(1) Gross weight includes canister weight derived from structural analysis (Ref. 5.7 appendix I) plus 30 lbs. estimated for attachment rings and fittings. Weight for attachment rings and fittings is a rough estimate and is not based on analysis.

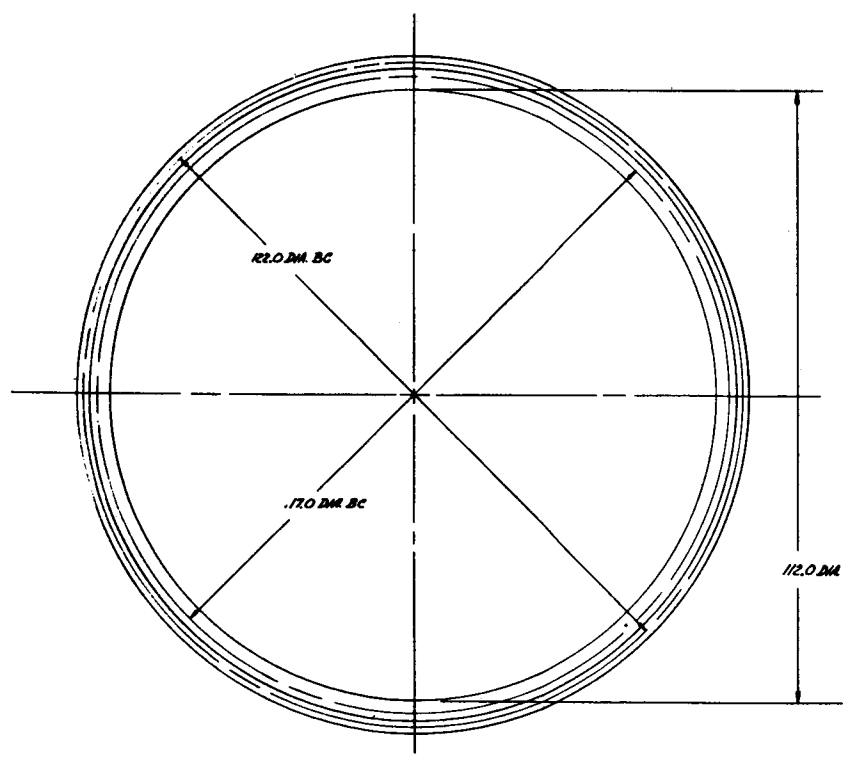
(2) Surface area of canister is taken as 48,000 in.^{2}

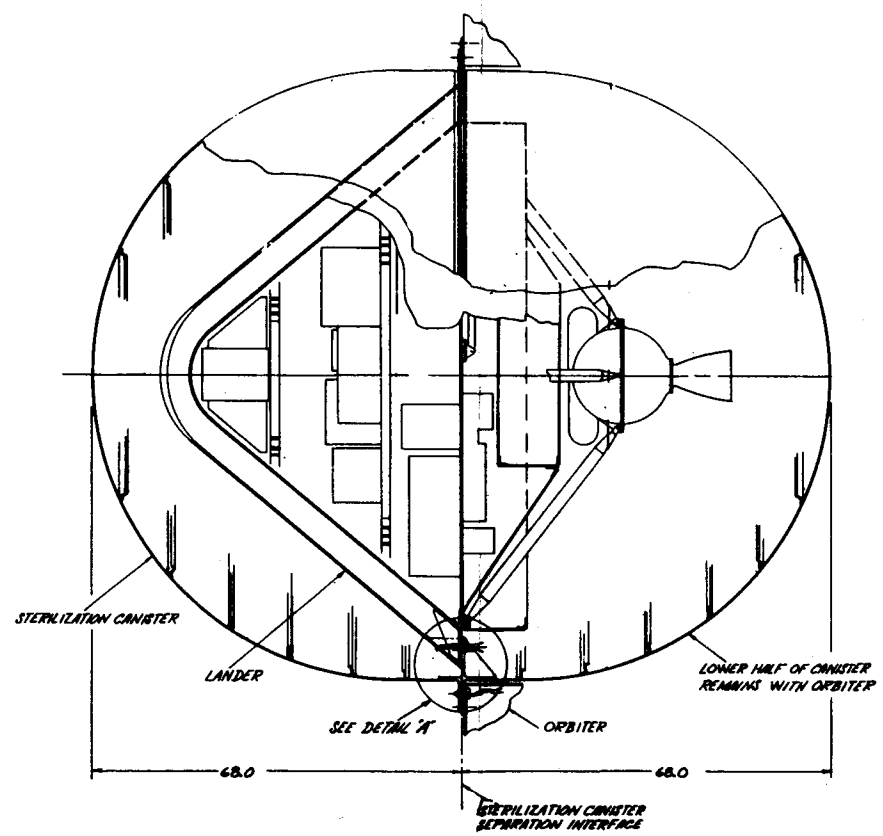
REQUIREMENTS FOR TYPICAL CONTAINING CANISTER FOR IDEALIZED UNMANNED PLANETARY LANDER

1. The containing canister shall represent a sterile barrier into which the typical lander is placed prior to terminal sterilization and in which it remains until some point in orbit beyond the contaminating effects of the Earth atmosphere.
2. The containing canister shall be shown in sufficient detail to indicate general size and functional complexity.
3. No specific canister for a specific vehicle or program shall be represented. This canister shall be general enough to represent a range of problems.
4. The structural details of the canister shall not be considered design solutions. Configuration, thicknesses, ring spacing, etc., shall be considered as typical design solutions rather than specific design solutions. The method of construction shown shall be typical rather than exact.
5. To simplify any weight comparison, which may be desired to assess the penalties associated with vacuum sterilization, spherical ends shall be shown on this canister. This choice is completely arbitrary and does not reflect the results of a design study.
6. The detailed interfaces between lander and canister, canister sections and between canister and orbiter shall be shown in sufficient detail to represent the functional requirements. The details shown do not represent the ideal design solution.
7. To clarify the relationship which will exist between the canister, lander and orbiter, a typical sequence of events is shown.
8. The canister shown does not reflect the method of heating which may ultimately be selected. It is assumed that the canister wall temperatures can be varied as desired.
9. The canister shown does not reflect the design requirements which will result from maintaining an internal positive gage pressure with a sterile gas. The canister does not reflect the design requirements which would result from a requirement to vent the canister during heating, or from a requirement to recharge with a sterile gas during cooling.

TYPICAL SEQUENCE OF EVENTS SHOWING RELATIONSHIP BETWEEN ORBITER, LANDER & STERILIZATION CANISTER







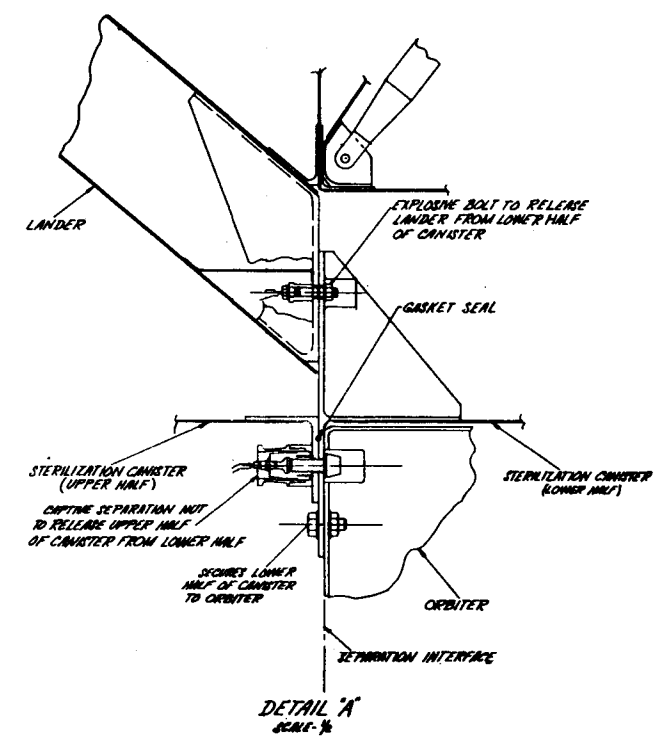


Figure 5.4-2. Typical Containing Canister for Planetary Lander Idealized for Preliminary Analysis

TABLE 5.4-1. COMPARISON OF HANDLING PROCEDURES

Procedure A	Procedure B
1. Contractor fabricates lander	1. Contractor fabricates lander
2. Contractor fabricates orbiter	2. Contractor fabricates orbiter
3. Contractor fabricates canister	3. Contractor fabricates canister
4. Place lander in canister and sterilize	4. Ship lander, orbiter and canister to launch site
5. Ship canister and orbiter to launch site	5. Place lander in orbiter and sterilize
6. Mate canister with orbiter	6. Mate canister with orbiter

For Procedure B, the total length of time during which the lander is enclosed in the canister prior to launch might be measured in hours while, for Procedure A, the lander might remain in the canister for weeks. For long storage times, design requirements for the canister would undoubtedly be more severe.

It has been proposed that a small positive pressure be held within the canister after sterilization to assure no leakage of contaminated air into the vehicle. If welding is not feasible for sealing the canister, mechanical joints may be required at the separation plane or at umbilical interfaces. Difficulty in designing mechanical joints for zero leakage may necessitate the use of a portable pressure source of sterile gas to retain the positive pressure during storage and handling and possibly on the launch pad prior to launch.

5.4.3 CONTAINER WEIGHT PENALTIES ASSOCIATED WITH TERMINAL STERILIZATION

Recommended procedures (Ref. 5.5.7) call for sterilization of a canister in an atmosphere of dry nitrogen at 135°C for 24 hours. In general, it is recommended that this procedure be a terminal requirement and that the sterile container remain sealed until flight. However, as a result of previous studies, it is generally conceded that this requirement will have to be compromised to accommodate certain components which are not capable of withstanding heat sterilization.

If the vehicle is sterilized in vacuo as specified by the Work Statement covering the present study, the enclosing canister must be designed to withstand a positive overpressure of 14.7 psi. This load would be imposed at temperature meaning that reduced allowable values at temperature would apply. This technique appears undesirable since, in addition to the possible weight penalty, maximum reliance must be placed on gaskets or seals to prevent airborne contaminants from entering the canister. The problem of monitoring the canister to determine that no airborne contaminants enter during a long storage period must also be considered.

If a sterile atmosphere is used in the enclosing canister to prevent air from leaking into the can (and as an aid to conductance) then this pressure will increase as heat is applied. Unless this atmosphere is vented, the resulting overpressure could constitute a design condition for the canister. If the pressure is relieved, the resulting cool-down could result in a vacuum. This vacuum would tend to draw airborne contaminants into the canister or to collapse the walls. A possible design solution would be the release of a sterile gas at a pressure in excess of atmospheric back into the can during cool-down. This approach will also add weight in the form of fittings, valves and parts although, from a weight standpoint, it appears desirable.

A gross weight comparison has been made of containing canisters designed to undergo various sterilization procedures. The purpose of this comparison is to emphasize the importance of sterilization techniques on the structural design of the canister. For the purpose of this study, the canister shown in Figure 5.4-2 has been selected as an idealized typical model. The structural analysis from which the weight comparison was made is presented in Appendix H. The comparative weight table is shown in Figure 5.4-2. The four sterilization techniques which were selected for weight comparison are as follows:

Condition I

The canister is pressurized to 1 psig prior to sterilization.

The canister is not vented during sterilization and the pressure rises to 7.8 psig at 150°C.

Upon return to ambient temperature, the pressure within the canister returns to 1 psig, and is retained at this level for monitoring sterilization.

During launch, the canister is vented to maintain a maximum pressure of 7.8 psig.

Condition II

Prior to sterilization, the canister is at ambient pressure.

During heat-up, the canister is vented, maintaining 1.0 psig pressure.

During cool-down, the canister is charged with a sterile gas, maintaining 1.0 psig pressure.

During launch, the canister is vented to maintain 1.0 psig pressure.

Condition III

The canister is evacuated prior to sterilization.

Vacuum is maintained during sterilization.

Vacuum is maintained throughout handling and launch.

Condition IV

The canister is placed inside a vacuum chamber.

The chamber and canister are simultaneously evacuated.

The canister is heated to 150°C for sterilization.

During the return of the chamber to ambient pressure, a positive pressure of 1.0 psi is maintained in the canister to monitor sterilization.

During launch, the canister is vented to maintain a 1.0 psig pressure.

5.4.4 HEAT TRANSFER RELATIONSHIPS BETWEEN CANISTER AND LANDER

A typical relationship between vehicle and canister is shown in Figure 5.4-2. During heat-up, a major conduction path will occur at the interface between the canister and the vehicle. Although this interface will obviously reduce the rise times and shorten the cool-down, it may also introduce local hot spots and severe thermal gradients in the adjacent structure. This area should be examined more closely. To be meaningful, however, some thought must be given to the design of the interface in order to make sure that it is functionally realistic.

5.4.5 IN-FLIGHT SEPARATION PROBLEMS RESULTING FROM BIOLOGICAL BARRIER

At some time between launch and lander separation, a portion of the canister must be removed. Although this is not primarily a structural problem, the structure is affected by the separation techniques which are employed. Springs, rockets, explosive ordnance and cans have been successfully employed in the past; however, the sterilization requirement may impose some new problems in this area. Some of the previously utilized devices such as explosives and rockets may not be capable of the 150°C sterilization environment such that their use may be restricted. If mechanical joints are required for separation, sealing of the joints, particularly over large areas such as the periphery of the lander may result in a considerable problem.

The most highly reliable leak-proof canister would have all joints welded or brazed. However, this limits the type of separation of the canister to a Flexible Shaped Charge. As previously mentioned the use of explosives may be limited; however it may be possible to install the Flexible Shaped Charge on the outside surface of the canister after sterilization.

5.4.6 TECHNIQUES FOR MAINTAINING POSITIVE OVERPRESSURE WITHIN THE SEALED CONTAINER

The design of gas supply equipment, pressure lines, valves, etc. must be considered as a structural problem. Seals, gaskets and O-rings are less clearly structural, but

must also be considered. The leakage requirements for a sterile vehicle, particularly if long storage times are necessary after sterilization, may be more stringent than normal, such that present sealing techniques may have to be altered.

5.4.7 PROBLEMS ASSOCIATED WITH A FLEXIBLE CONTAINER

Flexible canisters made from some material such as plastic have been considered for use as biological barriers for interplanetary landers. Some high temperature plastics are presently available which can withstand the 150°C sterilization temperature with a reasonable strength.

The advantages of a flexible container over a rigid container include: lighter weight, less expensive fabrication and the possibility of making minor adjustments on the vehicle after sterilization, by flexing the container locally at the adjustment area.

The same problems which exist with rigid containers also exist with flexible containers; however, there are additional problems with flexible containers, which on previous studies have prohibited their use. These problems may be summarized in three categories:

1. Structural design
2. Storage and handling
3. Lander separation

A. Structural Design

A flexible container is limited in its ability to withstand a pressure differential. It is more susceptible to creep than a rigid container such as metal or fiberglass, particularly during the sterilization cycle and over long storage periods.

Joint design is a problem between flexible sections and at interfaces where the flexible material is joined to rigid structure. This problem would exist at umbilical interfaces and at the bus or orbiter mechanical interface.

B. Storage and Handling

The storage and handling problems are much greater with a flexible container than with a rigid container. The flexible container is more susceptible to puncture and tear, and a small penetration could propagate rapidly into a catastrophic rupture. This would require a complete resterilization of the vehicle.

Hard points for handling could not be located directly on the container, but would have to be carried through to the hard vehicle structure.

Any sandwiching of the container which would be required at these points could scar the flexible material and lead ultimately to rupture.

C. Lander Separation

Prior to lander separation a portion of the container must be remotely removed to allow free travel of the lander during separation. The problem of removing a flexible container is greater than removing a rigid container. Methods of shearing the container and either imparting a motion, or stowing the sheared segments on the orbiter to prevent fouling of the system are inherently more complicated and less reliable than for a rigid container.

5.5 REFERENCES

5.5.1 W. Aron, G. Colombo, "Controlling Factors of Thermal Conductance Across Bolted Joints in a Vacuum Environment, Douglas Aircraft Co. Engineering Paper #1663.

5.5.2 "Radioisotopic Metabolic Detection of Possible Martian Life-Forms," A.H. Hein, Proceedings of the Lunar and Planetary Exploration Colloquium, Vol. III No. 2, May 5, 1963.

5.5.3 V.I. Oijama, "Mars Biological Analysis by Gas Chromatography," Proceedings of the Lunar and Planetary Exploration Colloquium, Vol. III No. 2 May 5, 1963.

5.5.4 R.I. Johnson, G.R. Arthur, "A Space Vehicle Design for Unmanned Rendezvous," AAS 61-97, August 1-3, 1961.

5.5.5 Technical Information on DuPont Explosive Specialties, Bulletin No. ES-61-1A.

5.5.6 Omitted

5.5.7 H. Coons, NASA MEMO R-RP-R-656 December 31, 1963.

5.5.8 B.E., Gatewood, "Thermal Stresses" McGraw-Hill Book Company, Inc., New York 1957.

6.2 DISCUSSION

6.2.1 GENERAL

Since joints of all types, involving a range of manufacturing processes and procedures, were of interest, it was necessary to group joints into loose categories. The most reasonable classification seems to be grouping by type as shown in Figure 6.2-1, Summary Sheet, Structural Joints. To obtain this summary, a study was made of the proposed configurations for Mariner and Voyager planetary landers. The study was restricted to the lander portion. To evaluate current joining techniques for space vehicles, drawings from other programs were also evaluated.

A joint rating shown in Table 6.2-1 was prepared to compare various joining techniques in order of preference in the following areas:

a. Design

b. Stress analysis

c. Thermal analysis

d. Manufacturing

e. Reliability

f. Biophysics

g. Weight

6.2.2 MANUFACTURING CHARACTERISTICS - STRUCTURAL JOINTS

In many quarters the techniques for achieving sterility are regarded as absolutes. In line with this "black" or "white" view, preference would always be given to a continuously welded joint since it offers no hidden haven for viable organisms. But the achievement of ultimate sterilization occurs within a framework of more complex, collateral considerations, such as:

a. Deployment of equipment

b. Techniques for terminal sterilization

c. Hardware configuration

d. Overall environmental control

e. Surveillance

When these factors are considered and balanced there appears good reason for not insisting on combining the operations of joining and sterilization. Since terminal heating should result in the desired 10^{-4} probability of a single, living organism, the advantages of simple tooling, an unemcumbered working environment, and vibrational damping, for example, should be given serious considerations.

6. JOINT STUDY

6.1 INTRODUCTION

One of the goals of this sterilization study was the evaluation of structural joints which occur in a typical spacecraft. A specific task was to establish methods for avoiding traps which could contain living organisms.

In attempting to classify joints some are easily recognized as "structure" while others are less easily categorized. For the former types there is extensive literature for thermostructural behavior. (References 6-1 and 6-2.) For the latter, which are by far the most troublesome from an analytical standpoint, the literature is much less extensive. Since the intent of this study was to determine the behavior of structural elements, no attempt has been made to identify all of the problems which will occur in the "grey" areas. But during the course of the study it has become apparent that many of the most serious problems connected with spacecraft sterilization will concern these grey areas. Accordingly, to indicate an awareness of the total problem, several joints have been grouped in a miscellaneous category and some of the salient problems have been outlined.

In evaluating the different types of joining techniques, it has been convenient to "start from scratch." That is, facilities, work areas, process controls, etc., have been hypothesized, as required, to make a particular point. In actuality this approach is not too practical. Based upon the announced NASA timetables for planetary landings, it is possible to work backwards and observe that the planning and design work required to make these dates meaningful must now be in progress. This means, that in the case of large clean-work areas or large sterile work areas, planning, scheduling and procurement should already have been initiated. Recommendations, to be completely valid, should be geared to the existing or proposed facilities. This concept was not rigidly adhered to for the joint ratings.

To put the sterilization problem in perspective, a separate study of clean rooms and sterile work areas has been made. In this latter study full consideration was made for state-of-the-art manufacturing capabilities and limitations. For the joint study it has been more fruitful to evaluate joints as if the fabrication of a vehicle which must experience terminal sterilization has begun today. The fabrication techniques described represent what a biologist might desire rather than what is practically attainable; when it is shown what is desired, then it can be determined if it is practical.

An attempt has been made to summarize the techniques and methods by which the thermal conductance across any of these structural joints can be improved.

SUMMARY SHEET - STRUCTURAL JOINTS
SPACECRAFT STERILIZATION STUDY

GENERAL NOTES:

1. THE STRUCTURAL JOINTS ON THIS CHART HAVE BEEN RATED AS A PART OF "A STUDY OF DESIGN GUIDELINES FOR STERILIZATION OF SPACECRAFT", CONTRACT NO. NAS-11107.
2. THE JOINTS SHOWN ARE TYPICAL JOINTS WHICH MAY OCCUR ON A TYPICAL PLANETARY SPACECRAFT FABRICATED IN THE NEAR FUTURE.
3. THE JOINTS SHOWN HAVE PURPOSELY BEEN KEPT GENERAL RATHER THAN USING SPECIFIC JOINTS FROM SPECIFIC SPACE VEHICLES.
4. IN PREPARING THIS CHART USE WAS MADE OF DOCUMENTS AND DRAWINGS ORIGINATED BY GENERAL ELECTRIC, NASA, JPL AND OTHER ORGANIZATIONS. PARTICULAR USE WAS MADE OF MARINER, VOYAGER AND RANGER DRAWINGS AND PHOTOGRAPHS.
5. MANY JOINTS OR DETAILS WHICH OCCUR ON A TYPICAL SPACECRAFT ARE NOT CONSIDERED PRIME STRUCTURE. THESE DETAILS INCLUDE BEARINGS, SEALS, FILL PORTS, TUBE FITTINGS, HARNESS CLAMPS, SOLAR CELL BONDS, THERMAL INSULATION TIEDOWNS ETC. NO ATTEMPT HAS BEEN MADE TO EVALUATE ALL DETAILS IN THIS CATEGORY. A FEW TYPICAL JOINTS HAVE BEEN INCLUDED TO SIGNIFY AN AWARENESS OF THIS PROBLEM AREA.

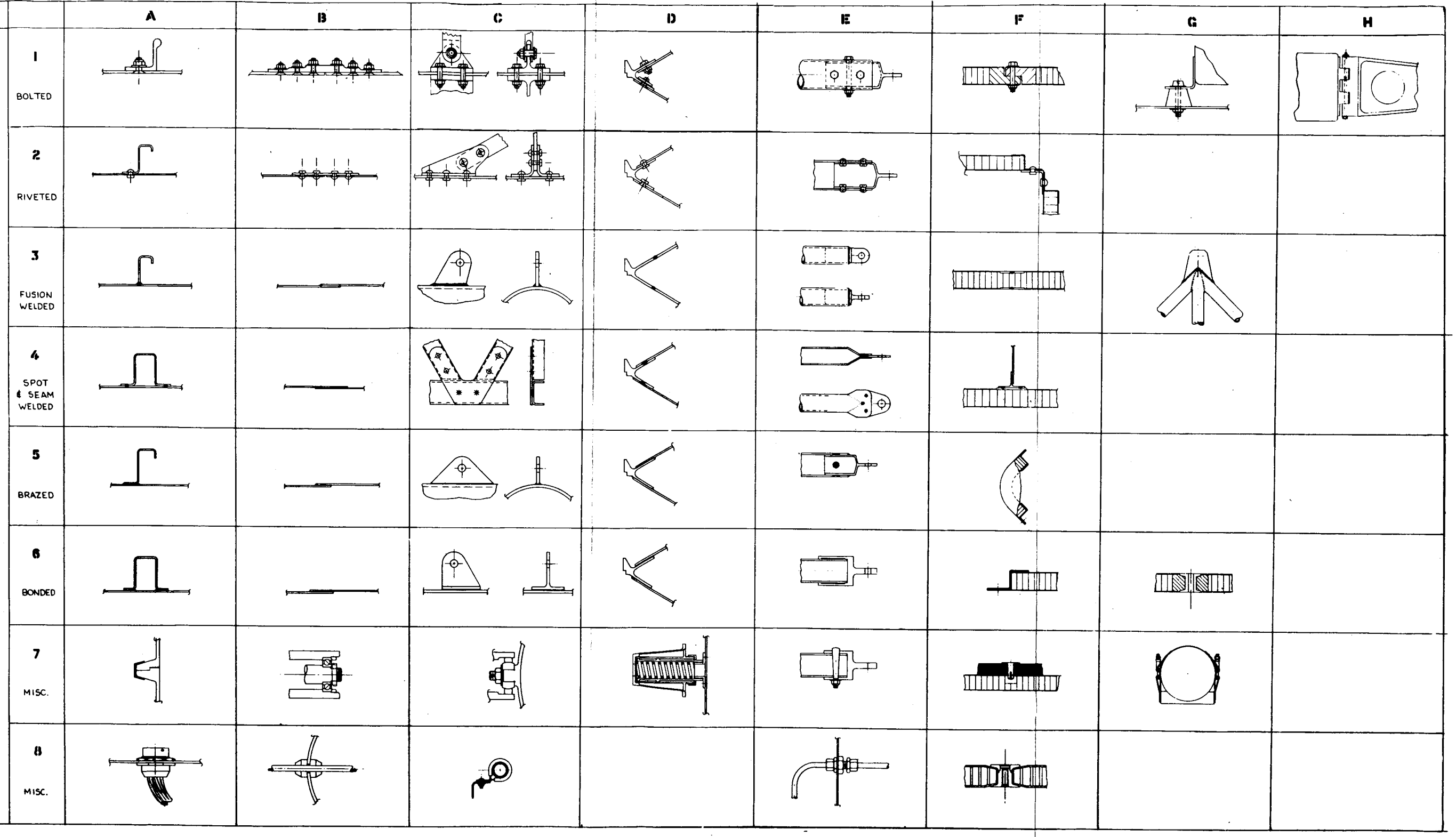


Figure 6.2-1. Summary Sheet, Structural Joints

TABLE 6.2-1. RATINGS OF JOINING TECHNIQUES, STRUCTURAL JOINT STUDY

Category Rating	Design	Stress Analysis	Thermal Analysis	Manufacturing	Reliability	Bio-Physics	Weight
1	Riveted	Bolted	Fusion Welded	Riveted	Bolted	Fusion Welded	Fusion Welded
2	Bolted	Riveted	Brazed	Bolted	Riveted	Seam Welded	Seam Welded
3	Bonded	Fusion Welded	Bonded	Bonded	Bonded	Spot Welded	Spot Welded
4	Fusion Welded	Bonded	Bolted	Fusion Welded	Fusion Welded	Riveted	Riveted
5	Spot Welded	Brazed	Seam Welded	Resistance Welded	Spot Welded	Bolted	Bolted
6	Brazed	Spot Welded	Spot Welded	Brazed	Brazed	Brazed	Brazed
7	—	—	Riveted	—	—	Bonded	Bonded

6.2.3 BIOPHYSICAL CHARACTERISTICS - STRUCTURAL JOINTS

Welding techniques are considered the best due to the heat involved, which tends to sterilize the working area during the manufacturing process. One objectionable feature of this technique is the rough finished surface of the weld bead or fillet which may create places for soil contamination. Riveting and bolting techniques are satisfactory. The loss of temperature-sterilizing action, during manufacture, is offset by the ease of control of process, the lack of fumes and contaminants, and the probability of achieving low surface contamination.

Brazing and bonding offer more chance for contamination during manufacture, less control of the process, and the opportunity to seal in spores. The processes are acceptable, however, and should not be discarded where design considerations favor the use of these techniques. From a sterilization standpoint, no single joining technique can be considered so satisfactory as to demand preference.

6.2.4 RELIABILITY CHARACTERISTICS - STRUCTURAL JOINTS

A. General

The factors considered in the joint evaluation are:

1. Manufacture
2. Marginal Design Practice
3. Inherent Restraints
4. Strength Scatter
5. Embrittlement

The ratings for these individual areas are summarized in Table 6.2-2 and the composite ratings are shown in Table 6.2-1.

B. Manufacture

An undetected, poorly fabricated joint could contribute to a castastrophic part failure due to the thermal cycling sterilization process since it would be the "weak link" in the structure which could lead to a failure of an externally restrained joint when subjected to a rise in temperature.

Consider, for example, a fusion welded lap joint as illustrated by 3B (see Figure 6.2-1) undergoing a rise in temperature, and assume that the joint is externally restrained by adjacent stiff structure. If the welds were poorly manufactured and ineffective over a significant length of weld then the remaining effective portion of the weld could develop high shear stresses leading to failure of the joint.

TABLE 6.2-2. JOINT RELIABILITY EVALUATION - STRUCTURAL

Type	Manufacture	Marginal Design Practice	Restraints	Strength Scatter	Embrittlement	Avg. Reliability Index	Relative Standing (See Table 6.2-1)	Remarks
Bolted	1	-1	0	1	1	.4	1	
Riveted	0	0	-1	0	1	0	2	
Fusion Welded	0	0	0	-1	-1	-.4	4	
Spot Welded	0	0	-1	-1	-1	-.6	5	
Brazed	-1			-1	-1	-1.0	6	Brazing is not applicable to aluminum
Bonded	-1	1	1	-1	-1	-.2	3	Tension and Shock Loads should be minimized.

Reliability Code - Good +1
Fair 0
Poor -1

The factors considered in evaluating the likelihood of an undetected poorly fabricated joint were:

1. Skill required by the fabricating personnel
2. Number of steps involved in fabricating the joint
3. Number of parts required
4. Environment under which the joint is fabricated including lighting, heat, chemical fumes and noise level usually associated with the fabrication process
5. Ease of control of the fabrication process
6. Ease with which flaws may be detected by non-destructive inspection techniques.

Bolted joints were rated as good from the viewpoint of manufacture since: little skill is required by the operator, few steps are required in the assembly process, a favorable environment could easily be maintained, excessive bolt preload could be prevented by use of a torque wrench, flaws may be readily detected by a simple torque test. Under preload, permanent set or thread stripping will exhibit itself by a small effort required to turn the nut or the bolt head. Note however, that sterilization may require the prohibition of lubricant application before torquing.

Riveted joints were rated as fair from the viewpoint of manufacture because somewhat greater skill is required by the operator and because visual inspection is the only non-destructive technique for determining flaws.

Fusion and spot welded joints were rated as fair from the viewpoint of manufacture since: either highly skilled operators or elaborate machinery are required for fabrication, heat and fumes are inherent in the process, careful inspection methods involving magnetic particle, penetrant or radiographic inspection and other techniques are required.

Brazing involves precleaning, fluxing, proper alignment during heating and cooling of the joint, heating to a temperature of 800°F or above, and postcleaning if a corrosive flux is used. The process demands either a skilled operator or elaborate machinery, a large number of steps in the fabrication process, heat and fumes, and careful control of the fabrication process and elaborate inspection techniques involving vacuum, penetrant, radiography or ultrasonic methods. For these reasons, brazing was rated as poor from the viewpoint of manufacture.

Bonded joints involving structural adhesives were rated as poor from the viewpoint of manufacture for much the same reasons that were given for the rating of brazed joints. The process involves preparation of the adhesive, which depends mainly on a chemical phenomenon or its ability to join the adhesive and adherent surfaces. Even slight contamination of the adhesive or the adherent surface can partly or completely destroy the bonding power of the adhesive. The environment inherent in the process involves noxious fumes, chemical irritants and heat requiring elaborate safety precautions.

Many steps are required including preparation of the surfaces, the mixing of the adhesive and the curing of the assembly, each step involving close control. Inspection is primarily of the destructive type, requiring testing of the samples and coupons. Quality assurance relies heavily on control of the processes and sampling of the materials before assembly.

C. Marginal Design Practice

A survey of a number of stress reports shows a tendency to design certain types of structure more closely than others. For joints, the tendency would be to design those joints whose behavior is reasonably well understood with closer margins than those joints about which less is known from experience and analysis. Since there may be considerable variation in the environmental loads, marginally designed joints may fail either during sterilization or in subsequent loading. Reliability concerns itself with the probability of failure and although margins of safety in themselves are no absolute indication of the probability of failure, they nevertheless provide an index to the relative reliability of one joint design with another.

In view of the foregoing discussion, a bolted joint has been rated poor because the designer and stress analyst design this type of joint with confidence based on analysis and past experience so that the tendency would be to have a marginal joint design.

Bonded joints on the other hand are a relatively new fabrication technique and the designer would approach the problem with caution because of a limited experience history. From the viewpoint of marginal design practice, then, bonded joints have been rated as good.

Riveted, welded and brazed joints fall between these two extremes and have been rated as fair.

D. Inherent Restraints

Supposing the joint were not restrained by adjacent stiff structure but nevertheless were restrained from deforming under a thermal gradient without stresses by the characteristics of the joint itself. For example, consider a bolted lap joint in which the bolt had a lower thermal coefficient of expansion than the plates which it joined. Under a temperature rise the bolt and the plate would be stressed due to the unequal rates of growth. A joint such as this would be characterized as having "inherent restraints." If the inherent restraints were high, then there is a danger that under a thermal-sterilization cycle permanent set could take place and either loosen the joint after cooling or introduce distortion, which would increase the likelihood of a stability (column or crippling) failure when subject to future loading. Note that the joints characterized as having high inherent restraints would also tend to develop thermal stresses due to external restraints of adjacent stiff structure. The factors considered in evaluating the restraints inherent to the joint were the stiffness characteristics of the joint in the transverse and lateral directions as well as thermal conductivity across the joint.

Bolted joints tend to be stiff along the axis of the bolt and not stiff in the lateral direction due to bolt hole clearance. Thermal conductivity is low thereby increasing the slope of thermal gradients; bolted joints were rated fair with respect to inherent restraints.

Riveted joints on the other hand are stiff in both directions and were rated as poor.

Fusion welded joints may or may not be stiff depending on the type of weld. A welded lap joint, for example, would be stiff but a butt joint would be stiffer than the parts joined; fusion welded joints were rated as fair.

Spot and Seam welded joints always involve a lapping of structure and would tend to be stiff in the transverse and lateral directions and have low conductivity; these joints were rated as poor.

Brazed joints involve a lapping of structure with filler metal between the metal parts to be joined. Since the filler metal covers a large area (compared to a spot weld or fusion weld), the stiffness in the lateral direction would not be high. In the transverse direction, however, the large area of filler metal would increase the stiffness. Brazed joints are rated as fair from the viewpoint of inherent restraints.

Bonded joints have been rated as good because of the wide area application of the adhesive and because of the low shear modulus of elasticity (two orders below that of most metals). Thermal conductivity may be kept high by the use of a thin adhesive layer of the Eccobond or epoxy type bonds.

E. Strength Scatter

From the viewpoint of reliability, it is desirable to use joints whose minimum design strength values (e.g., three standard deviations below the mean) are close to the mean strength value. A sharply peaked distribution of strength values indicates that fabrication process is able to be closely controlled, resulting in a small scatter in strength properties, and gives confidence in the probability of the actual strength exceeding the design strength.

Bolted joints have been rated as good because the simplicity of the fabrication process and the close control capability, while riveted joints have been rated as fair because the greater complexity of the fabrication process would introduce greater strength scatter.

Welded, brazed and bonded joints all require a number of steps in the fabrication processes and all involve the application of heat, thereby introducing a number of independent variables tending to give wide scatter in strength. For these reasons they were rated as poor with regard to strength scatter.

F. Embrittlement

Embrittlement implies a reduction in ductility (i.e., a reduced capability of deforming plastically before ruptures) due to a physical or chemical change. Embrittlement reduces reliability because it increases the chances of catastrophic part failure.

Bolted and riveted joints have been rated as good from the viewpoint of embrittlement because there is no attendant reduction in ductility.

Welded joints have a marked tendency towards embrittlement unless the preheating and cooling periods are closely controlled and uniformly applied. For this reason they have been rated as poor. The 300°F sterilization temperature is below the usual stress-relieving temperatures and is not expected to affect the embrittlement characteristics of welded joints.

Chemical interaction between the filler metal and the base metal in brazed joints tend to produce brittle inclusions and for this reason brazed joints have been rated as poor.

Little is mentioned in the literature on embrittlement characteristics of a bonded joint, although it may be inferred that the poor shock resistance of bonded joints are due to embrittlement caused by chemical interaction between the adhesive and the adherent. To be on the conservative side, bonded joints have been rated as poor.

6.2.5 THERMAL CHARACTERISTICS - STRUCTURAL JOINTS

A. General

The following thermal characteristics have been considered in preparing the comparative ratings shown in Table 6.2-1. The major assumption made is that the joints represent major heat flow paths.

B. Bolted and Riveted Joints

Joints A, B, C, D, and F have structural stiffness and thus can be considered satisfactory thermal paths (provided the surfaces are finished to 60 microinches (rms) or better and are flat and free of burrs). Data in subsequent references support this.

Joints E, G, and H are less desirable. Joint E, consisting of two curved surfaces cannot provide assurance of mating of interfaces. Tightening bolts may actually reduce interface mating.

Joints G and H are not reliable due to improper or tortuous paths, i.e., too many resistances in series.

Bolted and riveted joints may be improved by using an interface filler such as lead foil, indium foil, or silicone grease to promote heat transfer. Test data on the improvement to be expected can be found in the literature (References 6.3, 6.4 and 6.5). Current practice at JPL on Mariner vehicles is to use indium foil in critical thermal joints.

Assume that the ratio of doubler cross sectional area to the ideal cross sectional area is given by the factor k_D. The equivalent length of the doubler material is represented by L_D. The increase in volume over the length L_η is therefore given by $k_j = \frac{L_D}{L_\eta} k_D$.

Obviously, many other things affect the choice of joint which will give minimum weight under a given set of requirements. Such factors as reliability, accessibility, producibility, material compatibility, connecting member size and configuration, nature of loading, environment, seal requirements, etc., are involved.

This means that every joint must be analyzed separately from an overall system standpoint and therefore one cannot say that a particular joint will always be best for certain loads or for any other particular characteristic. Careful consideration of many factors is necessary to decide the best joint to use.

B. Fusion Welded

Included in this category is the electron-beam welding technique. Fusion welding may be used for primary load-carrying joints and generally requires less additional material than the other types of joints. Consideration must be given, however, to material compatibility (use of weldable alloys, heat treatment required, etc.) and stress relieving requirements. Joint efficiencies of 90-100 percent are not uncommon and reliability is high. Flash welding, where applicable, produces highly efficient, reliable, and lightweight joints.

C. Seam and Spot Welded

These types of connections rate high solely on the fact that material overlap requirements are a minimum. However, consideration must be given to joint efficiencies which can be as low as 65-70 percent in the thicker materials, low reliability under vibration and acoustic loadings, and thickness limitations. Such considerations, particularly the low fatigue strength of this type of joint, will often increase joint weight and dictate its rejection.

D. Riveted

This joint can be designed for efficiencies in the order of 80 percent and are highly reliable. Weight is affected by material overlap requirements, rivet type, and material thickness required to develop the rivet strength (addition of doublers, etc.).

E. Bolted

This joint is similar in construction and characteristics to the riveted joint; however, the attachments are generally heavier. This method finds advantage in joints between heavily loaded members and where disassembly is a requirement. Bolting produces a very reliable and versatile joint.

F. Brazed and Bonded

The size (and weight) of these joints is dictated by the shear strength of the bonding or brazing material. Brazing is limited to materials capable of withstanding high brazing temperatures (steel, titanium, etc.) and a shear strength of 15,000 psi. Considerably lower shear strengths are developed by bonding. Due to these shear strength limitations, the use of these techniques is restricted to thin materials and special applications (honeycomb construction, etc.).

6.3 SUMMARY

Each type of joining technique presently used on space vehicles has been evaluated for the effects of terminal heat sterilization. The joints considered have specific structural advantages which have been proven through years of testing and successful use. Terminal heat sterilization has introduced a new requirement on joint design which requires a re-evaluation of present joining techniques.

This investigation has shown that it is not possible to summarily reject any joining techniques on the grounds of incompatibility with the sterilizing environment. For the specified sterilizing environment, there are joining methods in each of the areas studied which may be selected and successfully employed. The wide range of variables for any configuration make it necessary that joining techniques be evaluated in light of their specific use or a specific configuration.

Future work involving analysis of spacecraft joint performance in a thermal environment is necessary and should be performed on the specific vehicles which will experience thermal sterilization. Variables such as size, shape, and mission performance may have considerable influence on specific joint selection.

6.4 BOLTED JOINTS

6.4.1 DESIGN CHARACTERISTICS

A. General

Bolted joints for structural connections exist in a very wide range of types and sizes. Threaded fasteners may vary in:

1. length
2. diameter
3. head type
4. thread form
5. locking features
6. surface finish or treatment

This wide selection of available fastener types is nothing new to designers, but since the problem of removing organisms is of concern, it should be noted.

C. Fusion Welded Joints

Fusion welds in general are desirable provided there is no reduction in cross sectional areas of the heat carrying members. A fusion weld is not a joint in the thermal sense of the word.

D. Spot-Welded Joints

Spot welds, by their nature, requiring fairly thin section material, are inherently unreliable if used as heat flow paths, since the interfaces surrounding the spot weld are not in contact. A continuous spot weld (seam weld) is slightly better but still has the problem of poor contact immediately adjacent to the weld.

E. Brazed Joints

Provided a reliable braze is obtained, the same conditions apply as are described for Fusion Welds.

F. Bonded Joints

Bonded joints can be considered adequate heat flow paths, subject to bond limitations. These are thermal conductivity of material, quality of adhesion, inclusion of voids, thermal degradation, differential thermal expansion effects, and thickness of bond material. Due to the limited availability of data, no satisfactory conclusions can be made.

G. Miscellaneous Joints

Adequate thermal paths which may be considered are 7A, G, 8A, and C.

6.2.6 WEIGHT CHARACTERISTICS - STRUCTURAL JOINTS

A. General

This study investigates various types of joints and rates them in order of preference from a weight standpoint. The ratings, while considering the weight added by the joint, cannot be based on weight alone and must be somewhat of a compromise between the practical joint and a theoretical joint.

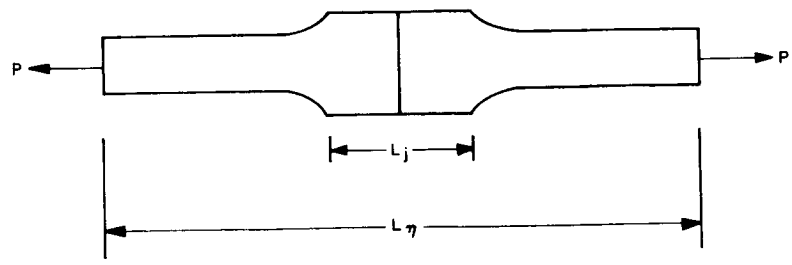
The method used to evaluate the weight efficiency of a joint, as stated in F. R. Shanley's "Weight-Strength Analysis of Aircraft Structure," (Reference 6.45) is as follows:

For weight analysis purposes the optimum weight is multiplied by a non-optimum factor for joints. Such a factor is defined as: $k_j = \frac{\text{Volume of Additional Material With Joint}}{\text{Volume Without Joint}}$.

This factor may be applied to the whole structure or to any portion of it.

The simple tension member could be used as the basis for analyzing the joint efficiency. Consider an ideal case where the volume of material is determined by the ultimate load to be carried, the ultimate tensile stress and the length. The volume of material over the length L_η would be $(Vol)_o = \frac{P}{f_a} L_\eta$, where, L_η is the length between joints and, $(Vol)_o$ is the ideal volume.

A good example of the fusion type weld joint would be:



The ratio of the increased cross sectional area to the ideal volume will be given by $1/\eta$, where η is the usual joint inefficiency factor. The increase in area will extend over some length which is represented by L_j, the average value of the joint. The increase in volume will be given by the ratio

$$k_j = \frac{\Delta \text{ Volume}}{\text{Ideal Volume}} = \frac{L_j}{L_\eta} \left(\frac{1}{\eta} - 1\right) \qquad \text{(a)}$$

Since the length L_η is regarded as the length between joints, it can be seen that the increase in volume over a total length, L, will be proportional to the number of joints. ($L_\eta = L/N$ where N is the number of sections, assuming that joints are equally spaced.) Equation (a) could be written in the alternative form

$$k_j = \frac{L_j}{L} N \left(\frac{1}{\eta} - 1\right) \qquad \text{(b)}$$

The joint length, L_j, will of course vary with the size of the structure and the type of design. In the case of the flat sheet with a riveted joint, the inefficiency caused by the joint extends over the entire sheet length. The L_j/L_η then becomes equal to unity, and $k_j = (1/\eta) - 1$. This is the case usually analyzed in design. The typical values of η in aluminum alloy is from 0.85 to 0.70. The corresponding values of k_j, for the riveted connection, would be 0.18 and 0.43, indicating a joint weight of 20 to 40 percent over the ideal minimum.

In the case of members for which compression or instability conditions are most critical, the stated type of inefficiency is not usually significant. The "doubler" effect is more important and should be briefly outlined.

In the past when discussing sterilization some investigators rejected bolted connections in favor of welding. In some cases it has been suggested that welded structural frames, which could be sterilized internally and externally, be used to support components which could not tolerate the same sterilizing environment; but implicit in this concept is the idea that the components will be attached to the structural frame. The obvious fastening means, bolts, have merely been pushed aside. Once it is admitted that bolts must be used to attach these components then the absolute requirement for a welded frame seems less obvious.

The question may arise, "Why not develop new fasteners which do not require threads; fasteners which could be more easily sterilized?"

It is true that new structural fasteners could be developed which do not require threads. Special high temperature fasteners have been developed which do not use threads. But it is not necessarily true that these new fasteners will be more easily sterilized - either before or after they are installed. The configuration of a new fastener will very likely have other geometric complexities which are equally undesirable.

The objections to threaded fasteners seem to be related to the technique employed for sterilization. If gaseous sterilization is used, (in spite of apparent disadvantages) a tightly sealed joint is a barrier to penetration. Conversely, if heat sterilization is used a tightly sealed joint insures good heat transfer at the joint and quicker rise times to sterilizing temperatures, but in either case, after assembly has occurred, flushing is equally difficult.

In surveying drawings and photographs of other space vehicles, it is obvious that threaded fasteners are widely used. To eliminate them as a fastening technique would be to impose undue hardship on the designer. Furthermore, at present, it is not possible to see what would be gained in terms of ease of reliability of sterilization. An acceptable alternative would be to limit the number, type and range of sizes of fasteners which could be used. This approach has been successfully used on other hardware programs by preparing an approved fasteners list.

Bolts or nuts which feature nylon or plastic pellets for locking should be evaluated in terms of expected temperatures and performance after exposure. In general it is safe to say that no major fastener selection problems will result from the temperatures encountered during sterilization. Any handbook or catalog will list numerous threaded fasteners for applications well beyond 300°F over extended periods.

B. Joint Summary, Case 1A (Stringers Stiffeners, or Component Mountings Flanges to Sheet)

This type of connection is very common on existing space vehicles and it is safe to assume that it will not be ruled out as a result of the sterilizing environment. To reduce the biological load, faying surfaces could be flamed or disinfected at assembly or installation and sterile hardware could be used. If it is not desirable for economic reasons to sterilize these fasteners, they could merely be cleaned (by scrubbing, ultrasonics, etc.) to remove gross contamination.

In many ways this case represents a simple version of a worst-case condition. Several bolts will of course be more objectionable (see Case 1B), but other cases (riveting, welding, etc.) will all tend to be simpler in terms of geometry.

C. Joint Summary, Case 1B (Sheet-to-Sheet Connections)

These connections are basically similar to Case 1A; however, for sheet connections a number of widely used specialty fasteners exist which may have unique problems regarding sterilization.

Consider the following examples:

1. Blind Lockbolt
2. Torque Controlled Locking Bolt
3. Blind Bolt

For these three well known fasteners, the internal surfaces must experience some form of biological load reduction before installation. Ultrasonics followed by separate thermal treatment appears as a suitable solution. The opportunities for entrapping gross contaminants in fasteners of these types are greater than for a simple bolt.

For faying surfaces between large sheets, gaseous sterilization or local flaming (perhaps with portable devices) could be accomplished.

D. Joint Summary, Case 1C (Sheets or Panels to Machined Fittings)

If required, fittings or machined details, because of their relatively small size, could be sterilized separately, bagged, stored, etc. until they are needed. Faying surfaces could be sterilized using techniques mentioned for 1A and 1B.

E. Joint Summary, Case 1D (Curved Sheets or Panels to Machined Rings)

Since numerous fasteners are implicit in a long joint, it is likely that some of the threaded specialty fasteners described in Case 1B would be used. Many fasteners of these types are driven with special guns. Care is required to prevent guns or tools from spraying or spreading gross contaminants (i.e., grease, oils, unfiltered air or moisture, onto exposed surfaces. It may be desirable to require cleaning by scrubbing or wiping between any operations in which power tools (not subject to special sterilizing procedures) are used.

F. Joint Summary, Case 1E (Tubes to Fittings)

Where all parts of structural assembly are small and where a high quality structural connection can be made using hand tools, the assembly can easily be made under controlled conditions such as:

1. glove boxes
2. clean benches
3. clean rooms - sterile hoods

Such assemblies could be fabricated, bagged and opened for use as required.

G. Joint Summary, Case 1F (Sandwich Panel Connections)

Connections between flat or curved sandwich panels should be carefully analyzed for response to thermal environment. Since the panel itself is a composite it may be highly anisotropic and each application should be examined separately.

The thermal stresses from differential expansion across a bolt should be examined since: 1) the bearing capacity of many panels is low and since 2) many core materials fail at a uniform rate, i.e., they do not behave elastically under local out of plane loads.

H. Joint Summary, Case 1G (Metal Flanges or Fittings to Non-Metals)

Insulating standoffs or spacers may be required for some components. These connections require special attention. A good insulator will block heat flow and increase rise time for terminal sterilization. Poor insulators may not be compatible with component life requirements in operation.

All non-metallic spacers must be able to withstand the terminal sterilization environment with no significant loss of load carrying ability. Load reduction with ethylene oxide appears practical.

I. Joint Summary, Case 1H (Hinges or Swing Fittings)

Bolted connections of this type may be troublesome during terminal sterilization due to the relatively long bolt. But for load reduction during manufacture, the problems should be analogous to Case 1C.

J. Conclusions

In summary, while more care must be utilized in the selection of materials, and the actual assembly process must be more closely monitored than the normal space vehicle requires, the use of fasteners of the types discussed are not ruled out. The additional procedures required do not constitute sufficient burden to force even minor undesirable tradeoffs into the vehicle design.

6.4.2 STRENGTH CHARACTERISTICS

A. General

The indiscriminate use of many otherwise acceptable fasteners is not recommended when designing for the sterilization environment. A significant number of fasteners, including all of the aluminum and titanium rivet and bolt types, have restrictions placed on them for use in thermal environments above 300°F. This is primarily because of the loss of strength which these materials experience. When combined with the added stresses of complex thermal expansions, joints of significantly reduced margins of safety at elevated temperatures result.

Along with the restrictions on different bolt types, restrictions must be placed on some types of nuts. These include some of the common steel nuts (such as AN364 and AN365). While a rating of individual attachments is not undertaken here, it can be pointed out as a design guideline that the suitability of each attachment for a specific application must be re-evaluated after subjection to the sterilization environment.

During the thermal sterilization cycle, in most cases, the inertia loads on the joints are negligible, and the significant item of interest may become the irreversible loss of joint strength.

B. Preload Buildup

A discussion of bolt preload buildup due to temperature has been included in this section. Referring to Figures 6.4-1 through 6.4-5, it can be seen that for a simple lap joint, the induced bolt tension is a function of the material thickness and material incompatibility. An obvious design guide may appear to be the use of fasteners of the same material as the mating parts; however, when the shear and bending load buildup in an attachment is also considered (as would the case in a lap joint where the mating elements have restrained ends) the use of steel bolts or rivets is indicated. Steel bolts have the capacity to take the buildup in axial load in addition to not exhibiting a significant loss of strength. The permanent set that may occur with the softer aluminum attachments may result in looseness of the joint after cooling.

C. Load Redistribution

Insofar as load redistribution is concerned, it is possible that during a prolonged exposure at the sterilizing temperature (150°C) such a redistribution in butt splices or lap joints may occur. This problem has been analyzed (Reference 6.2) for a bolted double shear butt splice having from 2 to 9 bolts per side. This analysis was made using the techniques of Reference 6.6 for joints at room temperature and at 500°F. Figure 6.4-6 shows bolt shear loads as a fraction of total load P for joints with varying numbers of bolts. Details of the configuration are given on the figure.

In general it appears that elevated temperatures cause an increase in shear loads on some bolts and a reduction on others, depending on the number of bolts and their relative locations. In a 9-bolt joint, for example, the load on the middle bolt decreases by 5.56 percent at 500°F. From this analysis it may be concluded that load redistribution in bolted shear joints is not a major problem in spacecraft sterilization. Test data presented in Reference 6.1 also indicates that thermal gradients inducing bowing have no appreciable effect on the load in a bolted joint.

Referring to Figure 6.4-6, it can be seen that all of the bolted and riveted joints will be subjected to an axial load variation when the attachment and member materials differ. An analysis for determination of this load increase is presented in Section 6.4.2(D). Steel bolts joining aluminum material have been taken as representative. Bolt diameters of 3/16, 1/4 and 5/16 inches have been considered with mated material thickness ranging from .20 to 1.50 inches. A .06-inch washer was assumed under the head and nut and a suitable shear nut was assumed.

NOMENCLATURE: (ALL DIMENSIONS IN INCHES & IN^2)

t_M = THICKNESS OF MATERIAL

t_S = LENGTH OF SHANK OF BOLT (ASSUMED EQUAL TO t_M PLUS THE .06 WASHER SHOWN)

t_R = THICKNESS OF WASHER UNDER THE NUT (.06"). BOLT IS ASSUMED THREADED FOR THIS THICKNESS HENCE THE ROOT AREA OF BOLT IS USED.

t_G = NUT THICKNESS

D_B = BOLT DIAMETER

$D_M = 2 = D_B$ DIAMETER OF AREA ASSUMED EXERTING LOAD ON BOLT

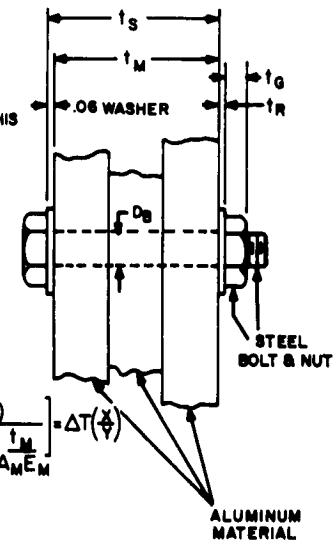
A_M = AREA OF $D_M = .75 D_B^2$

A_S = SHANK AREA OF BOLT

A_R = ROOT AREA OF BOLT THREAD

$$L_B = t_S + t_R + \frac{t_G}{2}$$

$$P_T = \text{AXIAL BOLT LOAD} = \Delta T \left[\frac{(\alpha_M t_M)(\alpha_B L_B)}{\frac{1}{E_B}\left(\frac{t_S}{A_S} + \frac{t_R}{A_R} + \frac{t_G}{2A_R}\right) + \frac{t_M}{A_M E_M}} \right] = \Delta T \left(\frac{X}{Y}\right)$$



ASSUMPTIONS:

1. $\alpha_B = 6.0 \times 10^{-6}$ IN/IN/F°
2. $E_B = 30.0 \times 10^6$ PSI
3. $\alpha_M = 12.5 \times 10^{-6}$ IN/IN/F°
4. $E_M = 10.5 \times 10^6$ PSI

Figure 6.4-1. Axial Bolt Load due to Temperature Changes

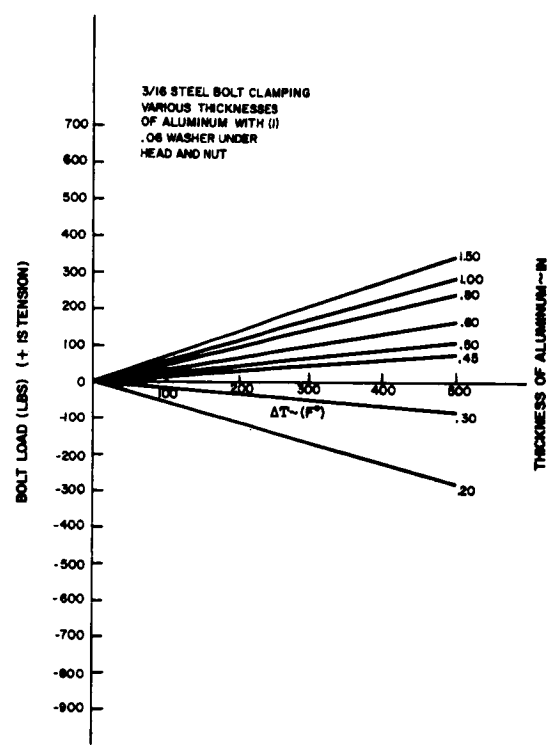


Figure 6.4-2. Axial Bolt Load vs. Temperature Changes

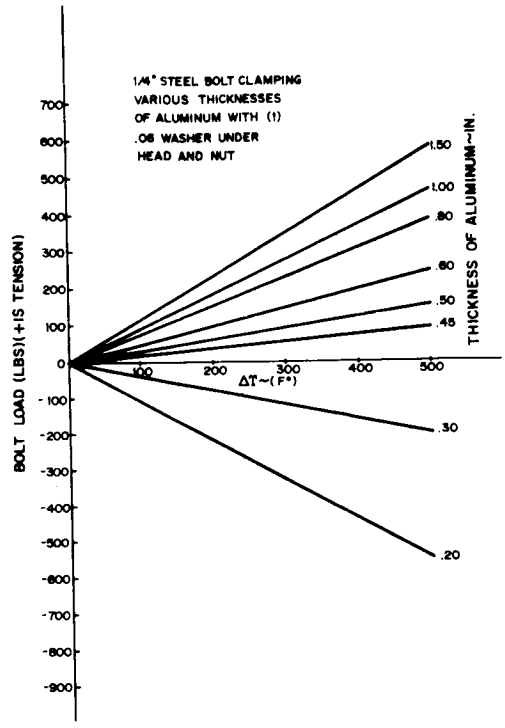


Figure 6.4-3. Axial Bolt Load vs. Temperature Changes (1/4-in. Steel bolt)

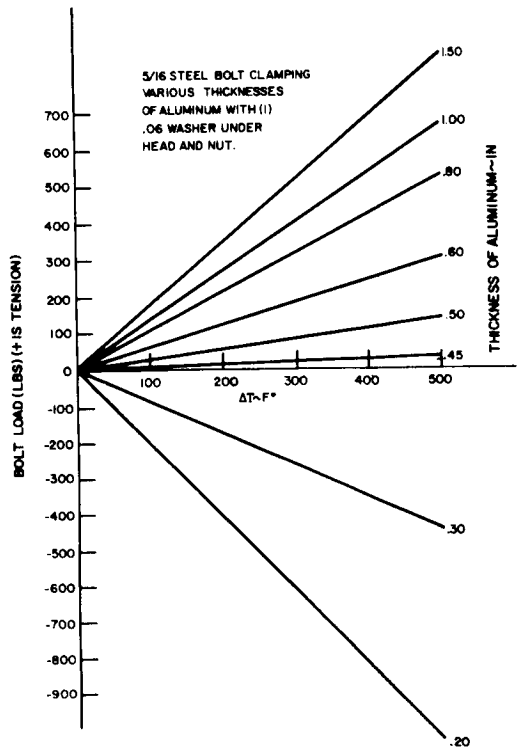


Figure 6.4-4. Axial Bolt Load vs. Temperature Changes (5/16-in. Steel bolt)

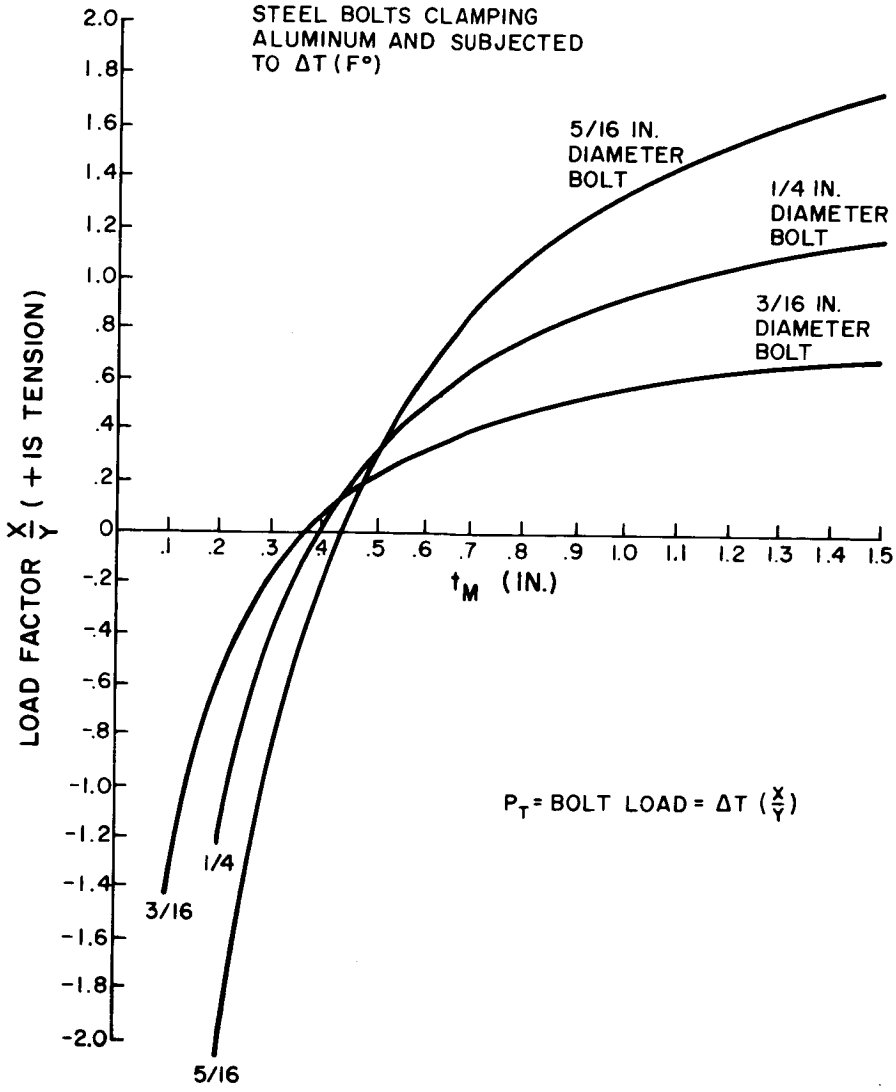


Figure 6.4-5. Bolt Load Factor vs. Clamped Material Thickness

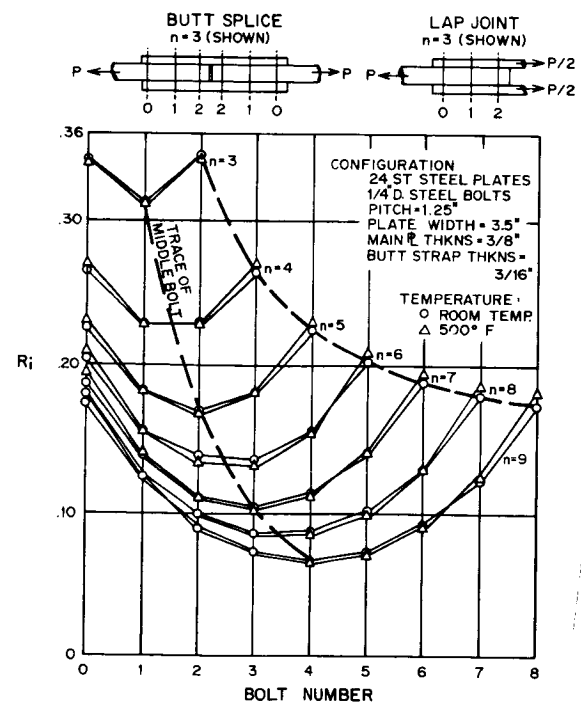


Figure 6.4-6. Bolt SHEAR Loads as a Fraction R_i of Total Load P

D. Shear Buildup

Joints 1B, D, E, F, and G of the joint summary are typical of joints susceptible to shear load buildup in the attachments. This shear load will be a function of the restraint offered the connecting members.

This load buildup (which will usually occur during the heat-up and cool-down) can induce appreciable shear loading as shown in the example below.

For joint 1-F shown, assume L_L is the length of the left hand panel, T_L the temperature at the left end and T_o is the temperature of the attachment denoting the right hand parameters byR.

$$\Delta L_L = \int_o^{L_L} \alpha_L \Delta T(x)\, dx$$

$$\Delta L_R = \int_o^{L_R} \alpha_R \Delta T(x)\, dx$$

For a linear gradient and $\alpha_L = \alpha_R$, $L_L = L_R$, $T_L = T_R$

$$\Delta L_L = \Delta L_R = \int_o^{L} \alpha (c\, x)\, dx = \frac{\alpha\, c\, L^2}{2}$$

where $(T_L - T_o) = (T_R - T_o) = c\, L$.

let: $L = 30$ in. (in this case the panel width);

A (concentrated in face sheets) $= .040\ in^2/in.$;

$\alpha = 12.5\ (10)^{-6}$ in./in./^{o}F (Aluminum);

ΔT, $(T_L - T_o) = 175^o$ (Maximum from thermal model);

$E = 10^7$;

$$\Delta L = 12.5\ (10)^{-6}\ (175)\ \frac{(30)}{2} = .0328 \text{ in.};$$

$$P/in. = \frac{SAE}{L} = \frac{.0328\ (.040)\ (10)^7}{30} = 438 \text{ lbs./in.};$$

The transverse shear capability of the panels used for this example is:

$$q_{max} = 0.040\ F_s = 0.040\ (40,000) = 1600 \text{ lbs./in.}$$

for a fully stabilized panel. It can be assumed then for this example that the sterilization environment will not be limiting for the fasteners, but the generation of a loading condition for which the member was not originally designed is evidenced. In this case, a uniform edge compression load is induced on mating shear panels under a temperature differential of the type prevalent during the cool-down period.

E. Conclusions

In general it can be concluded that if high temperature bolts and nuts are used, no problems will be encountered in the fasteners. However, a more detailed study of joint loads may be indicated.

The analytical tools available for determination of joint loads are fairly basic. The application of the redundant force method to bolted and riveted joints (Reference 6.1) allows for the solution of the forces and discrepancies in a joint by a computer analysis.

An extension of this study could be a computer analysis which would, for different bolt and rivet combinations, show the deformations and load redistributions which would occur and determine from these strains the permanent distortions and resulting loss in joint strength.

6.5 RIVETED JOINTS

6.5.1 DESIGN CHARACTERISTICS

A. General

For this report, riveted joints are discussed from the standpoint that all requirements which are desirable from a biological consideration will be given precedence over other considerations. This general discussion will be more meaningful if Volume V,"Sterilization of the Voyager Design Study,"has been read first. The requirements of design, manufacturing and sterilization were considered simultaneously.

Riveted connections have been widely used on space vehicles in the past. The techniques for design and manufacture are well known and carry over from the airframe industry. Compared to other joining techniques, riveting is relatively simple and inexpensive.

Riveting provides a degree of structural damping not generally associated with other joining techniques. One problem which usually arises in the design of a space vehicle involves successfully "detuning" vibration frequencies. Since little is known about vehicle frequencies as the design is in progress, any technique which provides structural damping is welcomed by designers. The damping effects of riveted structure were noted on early satellites such as Relay and Tiros (Reference 6.7).

Some investigators have compared riveted and welded structures which simulate actual satellite payloads. Their results have indicated the superiority of riveted damping (Reference 6.8).

The Relay satellite featured a riveted structural frame. Riveting was chosen over welding based on considerations of schedule, cost, and structural damping (Reference 6.9).

Mariner 2 featured an all riveted structure and Surveyor will use a blind riveted tubular design with cherry lockbolts (Reference 6.10).

B. Case 2A of Joint Summary (Sheet to Stiffeners or Stringers)

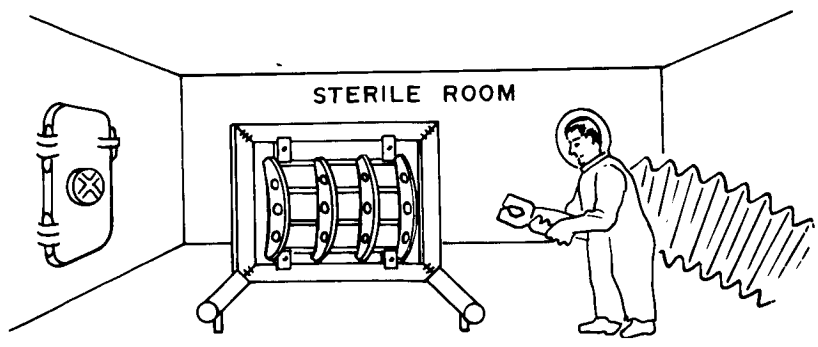
An objection that has been raised concerning sterilization of riveted joints is that at the faying surfaces at the joints, it is impossible to remove 100 percent of the dead organisms. If we assume that a technique existed, that was 100 percent efficient, then it might possibly be used in the case of small riveted stringers or stiffeners.

1. Assembly Within a Sterile Room

In order to consider this situation we must first hypothesize a large sterile room, then a sterile jig (with sterile tooling points). Other items which are implicit in this approach are:

- sterile work suits
- sterile rivets, cleco, cleco guns
- sterile rivet guns
- sterile lubricants for the guns
- sterile air or gas (for pneumatic tools)
- control techniques to verify sterility for a large working area

Assuming a technique for flushing organisms exists, then the ribs and stiffeners shown in Figure 6.5-1 could be sterilized and flushed as piece parts and then placed in the jig. Since some parts of the assembly must be in contact with the fixture, it is thus implied that the fixture will have been sterilized and flushed. The total sterilization and cleaning of larger panels presents an obviously much greater problem. Keeping the panel sterile (and clean) would be extremely difficult.



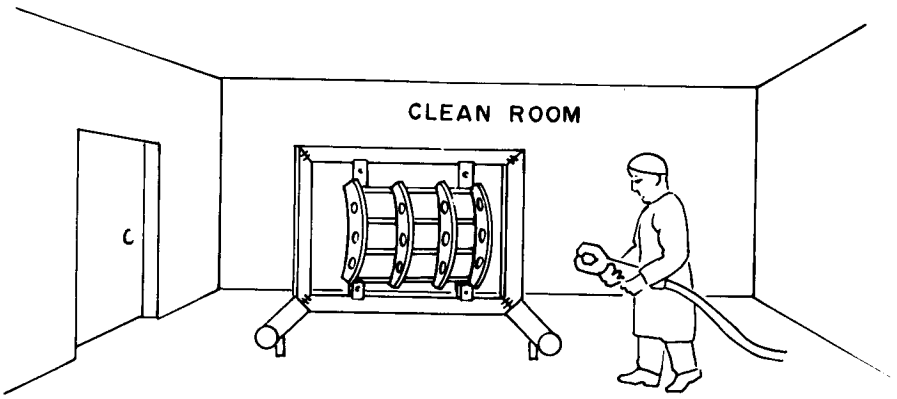
6.5-1. Typical Riveted Panel in Manufacturing Jig

In attaching the stringers and ribs to the panel, certain operations are implied. They include drilling, reaming and deburring. All of these operations imply chips, dust and airborne contaminants. It is true that Whitfield flow clean rooms can handle airborne particulates; but just physically cleaning these residues adds additional steps

and problems that raise the cost and potentially lower the reliability. An example would be the assays required. This further implies that the controls for a sterile assembly room will be vastly different and far more complex than the controls for a clean room.

For this quick look, it is possible to see that the requirement to remove all organisms and particulate from between faying surfaces under sterile conditions imposes many serious problems that may be beyond the state of the technology.

As an alternative, consider Figure 6.5-2.



6.5-2. Typical Riveted Panel in Manufacturing Fixture

2. Assembly in a Clean Room

In Figure 6.5-2 it will be assumed that fabrication takes place in a laminar flow clean room of 1000 particulate/ft^3 or less. It will also be assumed that:

- Individual piece parts may or may not be sterilized but certainly will have been exposed to exhaustive pre-cleaning.
- Jigs and fixtures will be clean but not sterilized.
- Rivets, cleco, rivet guns, deburring tools and other small hand tools may or may not be externally sterilized but would be clean.
- Cleanliness of work area and workmen is subject to controls as required, e.g., clean uniforms, filtered air, positive pressure in work area.
- Faying surfaces will be cleaned or sterilized by a specified process before final assembly (biological load reduction).

Based on assumptions such as these, it is very likely that the completed subassembly would have a high degree of initial cleanliness. Throughout the assembly process, it would have undergone "biological load reduction" by thermal, radiative, ultrasonic and chemical techniques; and consequently, following terminal sterilization would have a much better probability of being truly sterile.

The reason for this lengthy digression is to point out that while flushing may even become possible, it would only be practical for small parts like 2A and 2C rather than for large parts such as sheets and panels. Since flushing is not practical for large surfaces, then faying surfaces between large and small parts can be flame-treated or subjected to spot irradiation. Wiping is not considered a useful method of removing organisms.

C. Case 2B of Joint Summary (Sheet-to-Sheet Connections)

This type of joint implies multiple rows of fasteners. Sheet materials, either flat or curved, are difficult to handle and often require fixtures or tools to be moved or handled. It may be generalized that, as piece parts, sheet details are more difficult to sterilize or disinfect than stringers or stiffeners. This has been pointed out in the previous discussion.

Techniques involving dipping or spraying of sheets exist in the airframe industry and could be modified to perform partial sterilization (as a part of planned "biological load reduction").

A more practical approach to joining large sheets or panels would seem to be:

- Use of gaseous chemical sterilants and/or flaming at faying surfaces.
- Fabricate joint in clean room.
- Protect long joints and seams from contamination in subsequent handling operations by means of sterile protective coverings, such as plastic sheeting.
- Subject completed assembly to terminal sterilization.

D. Case 2C of Joint Summary

As mentioned, sheets, panels or other large details are amenable to currently available sterilizing techniques but due to large bulk, present handling and recontamination problems. Fittings and machined details (depending on their size) are easily sterilized using current techniques and devices. Small details could be sterilized and cleaned by combinations of heat, chemicals, radiation and ultrasonics, all using chambers, tanks or other facilities which are available off-the-shelf.

Fittings and details should be sterilized, bagged and then transported to either a clean room or a sterile room for assembly. Prior to assembly, major structural faying surfaces should be treated by flaming or chemicals. Individual rivets should be chemically pre-sterilized and individual containers opened only just prior to pulling or driving.

E. Case 2D of Joint Summary (Curved Sheets or Panels or Machined Rings)

For sheets and rings associated with typical spacecraft, useful techniques currently exist for sterilizing, bagging and transporting; however, such techniques have not been extended to large items. No insurmountable difficulty is foreseen to extending present techniques to encompass larger structural items.

A most reasonable approach would be:

- Assemble large structural components in a clean room (the degree of cleanliness to be specified by a sterility control group).
- Sterilize faying surfaces at layup (flame, irradiation, gas).
- After drilling and deburring and sterilizing as above, drive or pull pre-sterilized rivets.
- Bag and seal edges with thermal bonding.
- Subject completed assembly to terminal sterilization.

Obviously, the assembly prior to terminal sterilization could not completely satisfy sterility requirements. Equally obvious, however, is the fact that the biological load, particularly between faying surfaces, has been greatly reduced. This aspect of the total sterility problem is so important that iteration is required throughout the process.

F. Case 2E of Joint Summary (Tubes to Fittings)

For combinations such as this, mating piece parts should be separately sterilized. For tube assemblies with end fittings, the following sequence is visualized:

- Sterilize details as previously described and add coverings.
- Enter end fittings and tube into sterile glove box (gaseous type).
- Assemble, using sterile techniques (viz., drilling, reaming, deburring and riveting).
- Bag and seal.
- Exit sterile glove box directly to sterile assembly area.
- Unbag and assemble on vehicle.
- Undergo terminal sterilization.

Such a sequence is lengthy and expensive and as an alternate assembly sequence we could:

- Clean fittings via ultrasonics.
- Sterilize details, thermally, and package till use.
- Assemble in clean room using clean techniques for drilling, reaming, etc.
- Treat faying surfaces before final riveting, as previously described.
- Package.
- Sterilize terminally.

This approach which is certainly simpler and less expensive has also satisfied the two requirements of load reduction during fabrication, and terminal sterilization.

G. Case 2F of Joint Summary (Sheet or Extrusions to Thin Sandwich Sheet)

This case is basically similar to cases 2A and 2B. The sandwich sheets or panels will be more difficult to subject to piece-part sterilization unless special edges and inserts are incorporated into the design. Without these precautions it is possible that moisture or sterilants could collect inside the cells and become troublesome.

Sandwich sheets and panels are generally jigged or tooled in a different manner than sheets. Depending on the facing thickness, panels can be difficult to handle. Thin facings require special handling and transporting techniques.

6.5.2 STRENGTH CHARACTERISTICS

A. General

Most of the discussion which has been summarized for bolted joints is also applicable to riveted joints. Inter-rivet buckling is also applicable to both riveted and bolted joints; however, it is most commonly associated with multiple rivet patterns.

B. Inter-Rivet Buckling

An aspect of bolted and riveted joints which must be considered is the inter-rivet thermal buckling phenomena which will exist if there are thermal gradients through mating thin sheets. Aluminum, 2024-T3, 301 Stainless Steel 1/2 hard, and 6-4 Titanium were selected as typical structural materials to be compared.

Assuming that only a thermal load is acting, the critical thermal strain for elastic buckling between rivets is:

$$\left[\alpha\,(T-T_o)\right]_{CR} = \frac{\pi^2 t^2}{3 L^2} \tag{1}$$

where: t = skin thickness;
L = length of skin between rivet centerlines.

The approximate critical thermal strain when the bending stress induced by the thermal load equals the yield stress of the material at the given temperature is given by:

$$\left[\alpha\,(T-T_o)\right]_{CR} = \frac{\pi^2 t^2}{3 L^2} + \frac{L^2}{4\pi^2 E^2 t^2}\left[F_{cy} - \frac{\pi^2 E}{3}\left(\frac{t}{L}\right)^2\right]^2 \tag{2}$$

Table 6.5-1 shows the values of L/t, F_{cy}, E and α used for each material, and the resultant critical thermal gradients. Values of L/t used were 50 and 25 for each material. Fifty was selected as a conservative maximum which is rarely exceeded in design practice. Twenty-five is a more realistic value used for standard rivet spacing.

be imposed on weldments; whereas a more rigid precleaning would be required for mechanical joints to reduce the biological load.

Welding has been used on spacecraft in the past. In general it has been limited to specific types of joints and has not been accepted as a replacement for mechanical fasteners, particularly on major structural assemblies. Welding is often used for liquid and gas lines, and pressure vessel joints, where its sealing capability has been found superior to mechanical joints. Welding is also used to build-up complex fittings when integral machining is impractical. It is particularly valuable in joining subassemblies which are used for mounting equipment requiring critical alignment. In many instances the small amount of permanent displacement between members at a mechanical joint during handling or boost loading is sufficient to exceed alignment tolerances.

B. Fabrication and Quality Control

In general, fabrication and inspection of welded structure is more difficult and more costly than any other joining method. The contributing factors to this difficulty and higher cost include:

1. Jigs and Fixtures

More elaborate tooling is required for welded joints than for mechanical joints, particularly for large structural assemblies.

2. Equipment and Accessibility

Bulkiness of the welding equipment and the detrimental effect of the high temperature on surrounding structure must be considered in planning the assembly procedure.

3. Repairs and Parts Replacement

Repairs and parts replacement may be necessitated by poor welds, warpage of thin sections, part damage during fabrication, or change in design. Acceptable repairs are difficult to make on welded structure and in some instances the large assemblies must be scrapped.

4. Inspection

Weld quality is dependent upon surface cleanliness, quality of the welding equipment, and ability of the welding personnel. Visual inspection is sufficient for most mechanical joints, whereas structural weldments require more costly radiographic inspection for high reliability joints. Defects which are most commonly found in welds include cracks, porosity, incomplete fusion, inadequate penetration, and inclusions. Any of these defects may cause a catastrophic failure in a highly loaded structural application.

TABLE 6.5.1. VALUES OF L/t, F_{cy}, E AND α CRITICAL THERMAL GRADIENTS

Material	L/t	F_{cy} psi(10^3) At 300°F	E psi (10^6) At 300°F	α (In/In/°F)(10^{-6})	($T-T_{o\,CR}$) (°F)
2024-T3	50	33.6	10.1	12.6	124
2024-T3	25	33.6	10.1	12.6	420
301	50	54.6	23.4	8.9	156
301	25	54.6	23.4	8.9	606
6A1-4V	50	104.6	14.0	4.6	805
6A1-4V	25	104.6	14.0	4.6	1162

It may be noted from the table that the most critical temperature gradient of 124°F occurs in 2024-T3 aluminum with an L/t of 50. A more realistic L/t of 25 shows a sharp increase in the critical gradient to 420°F.

A study of the results of the thermal analog analysis of the model indicates that the maximum temperature gradient which could be expected across a riveted joint would be somewhat less than the minimum of 124° which was found in this analysis. Inter-rivet thermal buckling appears to be of minor concern in design for terminal heat sterilization as long as present design practices are adhered to.

C. Conclusions

As previously discussed in the section on bolted joints, the permanent loss of strength of aluminum rivets and their doubtful load carrying capacity at temperature make them less desirable than monel or other high-strength rivets.

6.6 WELDED JOINTS

6.6.1 DESIGN CHARACTERISTICS

A. General

Upon first consideration of joint design for terminal heat sterilization, welded joints appear to best fulfill the requirements. The advantages of welded joints for this environment include excellent thermal conductivity of the joint, the inherent sterilization of the joint due to the high temperature welding process, and the capability of sealing off previously sterilized cavities or faying surfaces with a continuous bead.

Surface cleanliness is not normally as rigid a requirement on mechanical joints as on welded, brazed, or bonded joints. By virtue of this existing requirement for cleanliness on welded joints, it is possible that no additional precleaning requirements need

C. Case 3A, 3C, 3D and 3F - Joint Summary

These joints are desirable as good thermal conduction paths. They have no faying surfaces which would be difficult to clean during sterilization. Consideration must be given to material and relative material gages to prevent warping or cracking at the weld. Fillet sizes must be controlled.

D. Case 3B and 3E - Joint Summary

This lap joint and rod end are examples of how a weld may be used to seal off faying surfaces to prevent contamination after welding.

E. Case 3G - Joint Summary

This type joint is commonly used on spacecraft to join tubular sections. It offers a good thermal path and light weight design.

F. Case 4A Through 4F - Joint Summary

Resistance welding (spot and seam welding) may be used to replace multiple fastener patterns. Many of the high strength alloys such as 2024 and 7075 aluminum, which cannot be fusion welded, may be resistance welded. Resistance welding shows little advantage over riveting in design for a sterilization environment. The heat application is local and does not suffice for surface sterilization. Resistance welds should be designed for shear only.

6.6.2 STRENGTH CHARACTERISTICS

A. General

In the previous discussion on bolted and riveted joints it was pointed out that the joints will lose some strength after sterilization. The same is true for welded joints. However, there are some additional, but much more subtle, side effects which a welded joint may be subjected to. Residual stresses which may remain from an incomplete stress relieving process after welding can induce slippage in the atoms of the material which in turn can lead to distortion and non-uniform strength characteristics at temperature and load application. These side effects can be very difficult to pin-point and are usually compensated for in design by higher minimum-margin requirements.

One advantage of welding is the fact that usually no material incompatibilities will exist in a welded joint by virtue of its basic construction. This eliminates some of the thermal stress resultants which are inherent in dissimilar bolt and mating material combinations.

B. Material Selection

Most of the high strength titanium and steel alloys commonly used for spacecraft structure are weldable, and materials may readily be selected which exhibit good properties after welding. Some consideration must be given in the local area of the weld to assure a high strength joint, but this may be compensated for by a local material buildup or an increased weld area.

Material selection is most critical in the design of aluminum welded structure. In many cases the high strength materials used for spacecraft construction are non-weldable by the fusion welding process. This is particularly true for the commonly used high strength aluminums, 2024 and 7075. Some success has been achieved in fusion welding 2024 but like 7075 it is still considered non-weldable.

Table 6.6-1 shows a comparison of ultimate tensile strengths of the two commonly used high strength aluminums - 2024 and 7075, and the two commonly used weldable aluminums - 5052 and 6061. The strengths are compared at room temperature both before and after a 1000-hour, 300°F temperature soak, and at 300°F after a 1000-hour soak. The strength of butt welds for 5052 and 6061 are also compared at room temperature and at 300°F.

TABLE 6.6-1. TENSILE STRENGTH COMPARISON (Reference 6.11)

Aluminum Alloy	Room Temp. Properties of Base Metal			Properties at 300°F after 1000 hr. exposure		F_{tu}, psi of arc welded Butt Joint at Room Temp. and 300°F	
	F_{tu} (psi)	E 10^6 (psi)	F_{tu}, (psi,) after 1000 hr. exposure at 300°F	F_{tu} (psi)	E 10^6 (psi)	Room	300°F
Non-Weldable							
2024-T3	64,000	10.5	57,500	48,600	10.0	---	---
7075-T6	76,000	10.3	49,500	32,000	9.4	---	---
Weldable							
5052-H38	39,000	10.1	38,200	31,200	9.5	25,000	21,000
*6061-T6	42,000	9.9	40,400	33,600	9.5	42,000	31,500

*Heat treated and aged after welding.

It is apparent from a review of Table 6.6-1 that the best structural aluminum for use to withstand a sterilizing environment is 2024, which has an ultimate tensile strength of 57,500 psi, after a 1000-hour, 300°F soak. The best weldable aluminum is 6061 with 40,400 psi ultimate tensile strength after a similar soak. A comparison of

these two materials shows the strength penalty which may be imposed on the structure, if welding is selected as the only acceptable joint for aluminum structure.

C. Fusion Welds - Joint Summary

For all the welded joints shown on the joint summary sheet, it can be seen that the stresses induced in the weld depend solely on relative displacements of the mating members. The one exception would be joint 3G which is typical of construction techniques to be avoided. A joint of this type is subjected to the forces which are induced in the members in order to maintain compatible member end displacements. If the members are of equal length and similar material, and external restraint is ignored, these forces will be in the form of end moments. Axial loads will also be induced if the member lengths are different.

It can be concluded that the sterilization requirement does not particularly enhance the use of welded joints, but at the same time will not limit their use except in instances where the joining process provides undesired rotational restraint to the members.

D. Spot Welds - Joint Summary

Spot welding techniques have been proven to be totally undesirable from a fatigue strength point of view. If spot welding is to be used in areas of negligible loading, the sterilization thermal environment may cause failure if tensile loads are induced. Joint 4-F is an example of a shear joint which is susceptible to tension failure under thermal loadings. Thermal test data in Reference 6.2 indicates a loss of as high as 45 percent of room temperature strength of a spot welded lap joint after thermal cycling. This loss was further aggravated by curvature of the joint when gradients were present. Spot welds generally should be avoided.

6.7 BRAZED JOINTS

6.7.1 DESIGN CHARACTERISTICS

A. General

Brazed joints are often used to:

- Join dissimilar metals
- Provide leak tightness
- Provide good electrical or thermal conductivity
- Withstand thermal shocks or large temperature gradients
- Provide corrosion resistance
- Join very thin materials when other techniques are not appropriate.

Brazed joints are used in the design of space vehicles but in general they are not used for prime structure or major load carrying members. Brazing is often used in temperature control systems where all of the characteristics listed above can be used.

Honeycomb sandwich is often brazed although, in general, the cores and facings have been of stainless steel for high temperature applications. Brazed panels can easily be fabricated to operate in the temperature range 500° F to 1000° F where adhesives could not be used. Other sandwich configurations such as truss core, truss grid, or corrugated cores can be used.

Unlike bonded fabrications, brazing is not susceptible to degradation from high vacuum or radiation. Like bonded fabrication, brazing requires that the parts to be joined have a high degree of cleanliness. For a general description of the cleaning and handling requirements of a typical brazed panel see Chapter 25, "Honeycomb Structure, of the American Welding Society's Brazing Manual (Reference 6.12). In particular, the insistence on the use of clean white gloves and air conditioned work room is of interest.

Brazed joints are often used in fluid or gas lines for aerospace applications. A considerable amount of testing and development work has been done to optimize techniques and processes for making these joints.

If active thermal control systems are required for planetary landers then it is likely that an attempt may be made to combine the function of structural and fluid carrying subsystems. Such attempts have been started on the Gemini Program (Reference 6.13). The proposed Gemini Adapter is shown in Figure 6.7-1. This configuration not only combines functions of structural and fluid elements but also provides a degree of protection from penetration by meteorites.

Brazing would unquestionably be a prime contender for joining the open ends of the stringers as shown in Figure 6.7-1.

B. Case 5A of Joint Summary (Sheet to Stiffeners or Stringers)

This type of joint has seldom been used for spacecraft structure. Proposed joints for structural applications have been discussed previously. For stainless steel or titanium brazed panels the brazing temperatures would be quite high. The completed panel would presumably have been cleaned prior to the brazing.

C. Case 5B Through 5D of Joint Summary

These joints have not commonly occurred on space vehicles in the past. Their use could be desirable to improve thermal conductivity. Each case will require individual attention from a sterility standpoint but no problems can be foreseen.

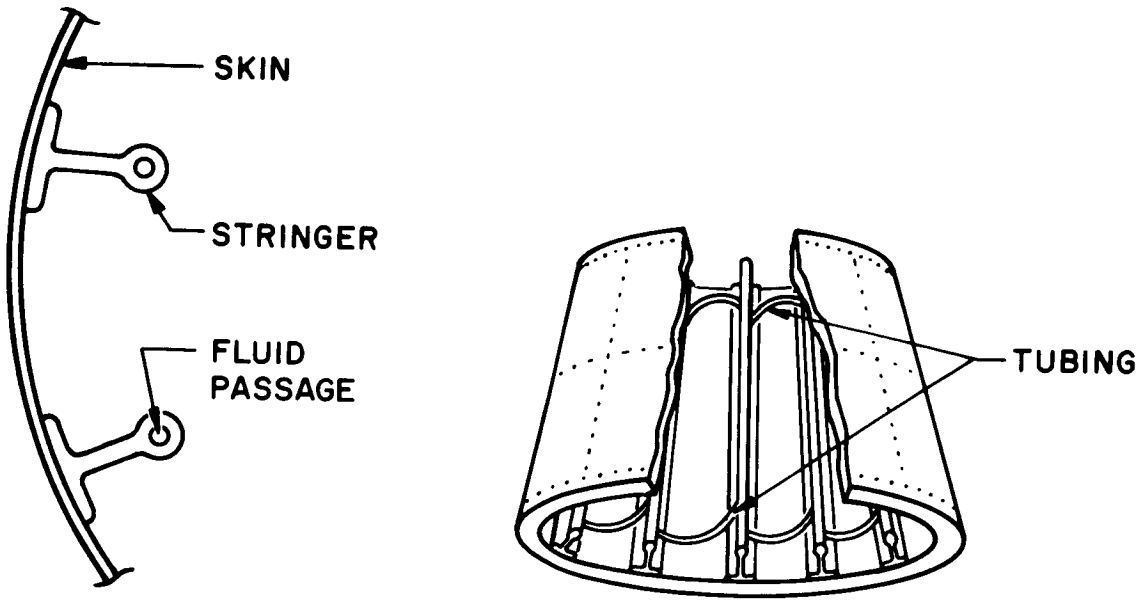


Figure 6.7-1. Proposed Gemini Adapter

D. Case 5E of Joint Summary (Fittings or Connections Between Tubular Members)

Brazed fittings have not commonly been used for purely structural applications. They are widely used for pneumatic and fluid systems. The techniques for these applications are well known. If it is desirable to use brazed connectors of this type to improve conductivity much of the technology from fluid systems will be applicable.

E. Case 5F of Joint Summary (Brazed Sandwich)

The use of brazed structural sandwich panels for planetary landers appear likely.

6.7.2 STRENGTH CHARACTERISTICS

A. General

Brazing is seldom used for joining prime structure; however, its use in components and plumbing connections as structural joints warrants an investigation. As an example, it is often used to make joints in rigid or flexible high frequency waveguide, or in relatively long tubing runs where the dynamic characteristics during powered flight must be considered. Where stiff sections are joined to flexible supports, the response of the combination must be evaluated.

B. Temperature Considerations

The range of temperatures which may be encountered on a launch vehicle is -420°F to 2500°F. Although brazing cannot be used (with aluminum) at the upper end of this range it is useful at lower temperatures. Brazing alloys generally have large differences in expansion from the base metals. At lower temperatures they exhibit good ductility which helps overcome the stress problem (Reference 6.14). The filleting action of the braze alloy aids performance under vibrations or repeated stresses. Many braze alloys yield smoothly filleted joints with a uniform appearance. Some alloys however are both brittle and aggressive, in some cases attacking and alloying with the base material. Where this occurs, the new joint is a new non-uniform, uncharacterized alloy formed by the base material and the braze alloy.

For this case no amount of data or properties of the base material will be sufficient to predict the characteristics of the new joint. Although this problem is not uniquely associated with the sterilization environment, it is mentioned as an objection to brazing for major structural applications.

6.8 BONDED JOINTS

6.8.1 DESIGN CHARACTERISTICS

A. General

Bonded joints are discussed from the standpoint that sterilization requirements will be given prime consideration. This discussion simultaneously considers the requirements of design, manufacturing and sterilization.

A brief survey of existing and predicted space vehicles indicates that adhesive bonding is widely used. Furthermore, recent state-of-the-art surveys (References 6.15 and 6.16) predict that interest in sandwich and composite constructions, using bonding, will increase.

The success of adhesive bonding depends to a large extent on the cleanliness of surfaces. The process specifications of most manufacturers give detailed and explicit procedures to be followed in preparation for bonding. During these cleaning steps it would appear that dirt and gross contamination can be removed in a way that is compatible with the procedures required for biological load reduction. Of interest here is a caution that unless carefully chosen, the cleaning solution may support or even encourage the growth of some highly selective organisms. Some investigators, (Reference 6.17), have reported the presence of viable organisms in chemically pure concentrated hydrochloric acid. Conversely, proper choice of cleaning solution may reduce the organism population several orders of magnitude.

By examining a specific cleaning procedure in detail, it is probable that it may be made to satisfy the requirements of a sterility control program. If the procedures will not positively satisfy sterility requirements, it may be possible that chemical

additives, increased temperatures, prolonged exposure times, or various combinations of the above can be used to modify the cleaning procedures for compatibility with sterility requirements.

One question which recurs when discussing bonding concerns the presence of organisms in the adhesive layer. It must be assumed that organisms are embedded in the adhesive layer (Reference 6.18). Following terminal sterilization, non-viable organism residues will be permanently fixed in an adhesive matrix. Under these conditions it is impossible that the spacecraft supported sensing devices could detect organism residues in the adhesives. It is true that portions of the spacecraft could fragment and fall in the vicinity of the biological sensors. Nevertheless, it is still felt that the possibility of such an occurrence is quite small. If organic adhesives were utilized, recognition of the residue separately from the organic would not be possible.

In the past, some of the objections to the use of adhesives have stemmed from the more general objection to all organics. This objection was based on the idea that some surfaces (specifically the moon) might contain high concentrations of free radicals which would react explosively on coming into contact with organic substances from the earth. This view has not been supported by CETEX (Reference 6.19). For this discussion, therefore, the use of organic adhesives will be permitted.

B. Case 6A of Joint Summary (Sheet to Stiffeners or Stringers)

Joints of this type have found sufficient use on space vehicles to be of interest. Bonded hat sections provide an excellent "fix" for plates or panels with undesirable vibrational frequencies. Bonds of this nature might be made in an oven or autoclave with tooling or fixtures provide required clamping pressures. The adhesives might be in the form of supported or unsupported films or liquids.

Usually, the mating parts will have been thoroughly cleaned and water break tests performed before layup. Temperature ranges can vary from room temperature to several hundred degrees. In general, the times involved are short; in the neighborhood of several hours. As an example of the times involved, Figure 6.8-1 shows temperature-time relationships for two common adhesives.

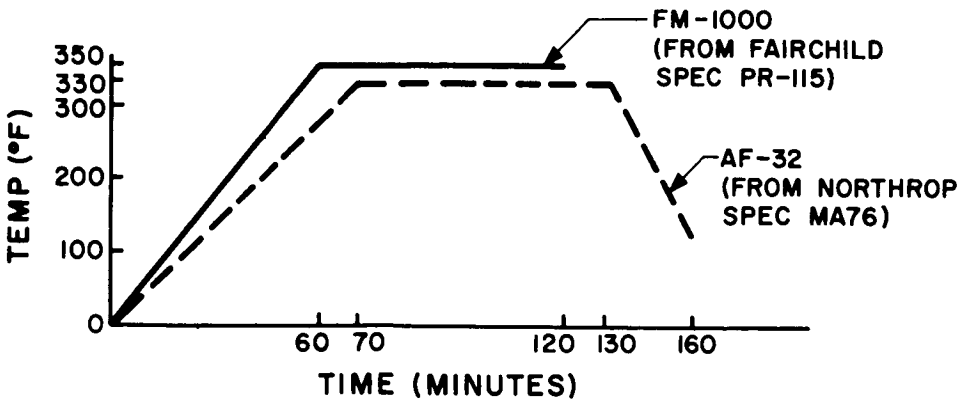
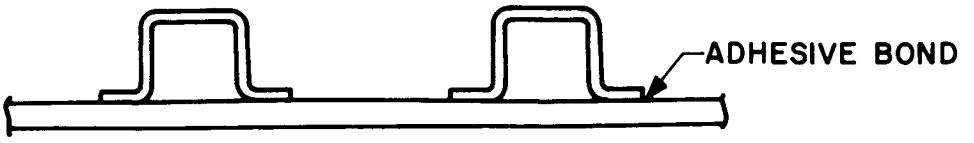


Figure 6.8-1. Time-Temperature Relationship of Typical Adhesives

Consider a simple panel as shown:



In a normal fabrication sequence this panel might experience these operations:

- Detail parts cleaned and degreased
- Bonded at 350°F for 60 minutes
- Terminal sterilization (when finally assembled to the spacecraft).

If it is assumed that following bonding, the panel is bagged, subjected to careful handling, then assembled in a "clean" or relatively clean room (1000 particles/ft^3) then it should begin terminal sterilization with a relatively low biological load. For such a panel, provisions for careful bagging and handling following completion of the bonding process must be provided.

In many cases such a panel would not be complete following bonding. Instead it would be ready for other operations such as trimming to final size, drilling, reaming, milling, etc. When this is the case, panels could be sterilized by an approved procedure then bagged and stored until final assembly. In any event, the point at which the panel is bagged is closely related to the manufacturing procedure. This implies that sterilization control will have to extend into the area of manufacturing planning or that planners will be educated for the specific requirements of sterilization.* As an example of the planning which occurs on a typical panel see Figure 6.8-2. This figure shows a typical process sheet or traveler for a very simple flat sandwich panel (used on the Advent Satellite). If sterilization and bagging is included, the procedures would be only slightly modified. Clean room procedures are assumed throughout. To summarize, these joints should create no special problems.

C. Case 6B of Joint Summary (Sheet-to-Sheet Connections)

For long joints or joints covering large areas, flaming or other sterilizing techniques could be used on adjacent surfaces. This assumes that some techniques are available for producing sterile adhesives or else nothing really is gained by extensive surface preparation. Unquestionably, load reduction would occur, but since the extent or degree cannot be determined it could be omitted from the procedure. No special problems can be associated with this joint.

D. Case 6C of Joint Summary (Sheets or Panels to Machined Fittings)

Bonded joints of this type are less frequently encountered in spacecraft design. If their use should be specified, they would be no more difficult than Case 6A.

*Duties, responsibilities and requirements for a sterilization control group have been extensively documented in other reports. See "Voyager Study," Vol. 5.

1. Prefit Parts

2. Process Details Number of Cleaning Cycles_____

3. Vapor Degrease Core Dry - 15 min. @150°-200°F_____

4. Apply Tape P.O. #_____ FM #1000 Batch #_____

   .0045" Roll #_____ Date of Mfg._____

5. Core Splice FM-47 Batch #_____
   Tape P.O. #_____ Type 0 Roll_____

6. Assemble Parts and Test Panels.
   Corfil as required #615 B#_____

7. Daily Chemical Process**
   Test Acceptance Date_____

8. Sequence Number G.E._____

9. Identify Assembly and Test Panels

10. Platen Lay-up

11. Cure Cycle Number**_____

12. Cure Cycle_____min. Pressure_____HF
    Cure Period_____min. Temp._____

13. Mill - Route

14. Drill - Corefill #615 as required Batch #_____

15. Clean-Up

16. Re-identify as required

17. Inspection - Test Panel

18. Rejection Report #_____ Cause_____

19. Engineering Change #_____

20. Rework - See squawk for method, etc. Authorized by Inspection Supv.
    _____

21. Final Inspection and Stamps Applied

22. Light Spray Coat of Kaylon Lacquer Over Identification

NOTE: **Recording Media of Temperature Control & Test Data on File

Figure 6.8-2. Typical Process Sheet - Honeycomb Sandwich Panel

E. Case 6D of Joint Summary (Curved Sheets or Panels to Machined Rings)

These joints are relatively uncommon on spacecraft with one notable exception. The bond between the elastomeric shield (ablation material) and the structure will fall in this category. It is also conceivable that crush-up structure would be attached to prime structure using adhesive bonding.

Completed subassemblies would in all likelihood be large and some thought will be required for the procedures and techniques by which cleaning and handling will be accomplished. For large subassemblies it has been suggested that heat sterilization (in a chamber) could be performed as a step in the manufacturing sequence. Because of the time and expense involved this should be recommended only when the specific advantages are known.

This truly is a decision that would be required of the sterilization control group of a specific project.

In general the handling, transportation, etc. of large bonded subassemblies can be expected to present problems from the standpoint of recontamination, sheer bulk of sterilizing equipment and associated GSE.

F. Case 6E of Joint Summary (Tubes to Fittings)

Joints of this type are not commonly encountered, but since detail parts are usually small (hence easily cleaned) no specific problems can be foreseen.

G. Case 6F of Joint Summary (Panel Edge Closeouts or Doublers)

These bonded joints are very common although many spacecraft panels use edge potting where no strength is required. This joint would very likely be formed when the core to facing bond is made (i.e., in one operation). For some panels, however, the edge closeout members are added as a subsequent operation. Where the panel is made in one operation, no particular problems are anticipated. When edge closeouts are added as a subsequent operation, an intermediate sterilizing operation, if specified, could be troublesome. Figure 6.8-3 shows a sequence which could occur.

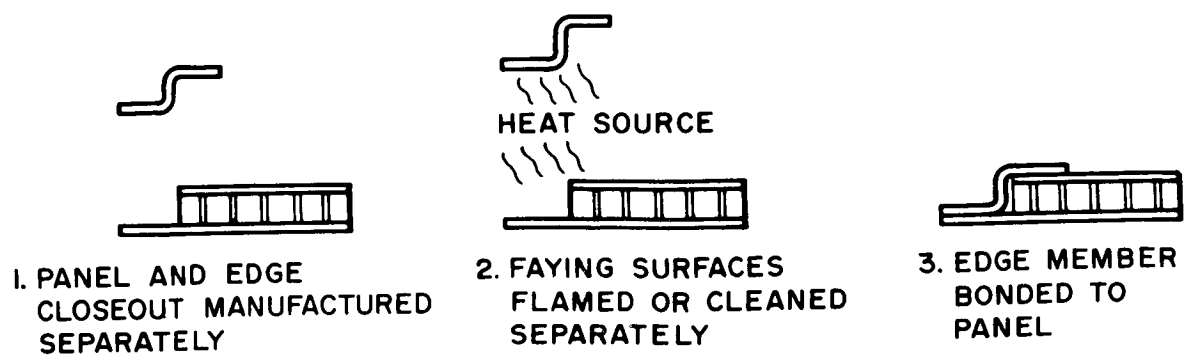


Figure 6.8-3. Typical Biological Load Reduction During Fabrication

H. Case 6G of Joint Summary (Bonded Inserts or Plugs)

Details of this type are commonly encountered. They may be used to attach panels to frame members or to attach components to the panel. Blind or through fasteners for bonding are available in a wide variety for aerospace applications. Specialty fasteners can easily be made. The inserts may be located and cured with the initial core-to-facing bond. One fairly common procedure is to drill, undercut and pot the insert after the panel is completed. Some questions which may arise for a joint of this type are:

- How sterile are the potting compounds or how sterile can they be made?
- How does the potting compound degrade as a result of thermal environment?
- To what degree is loss of strength at high temperature (for typical joints of this class) irreversible?

These questions cannot be answered within the limited scope of this study. However, as a general guideline, inserts of this type should be avoided for the support of heavy components, arms or devices which are preloaded or pretensioned prior to terminal sterilization or any application in which loads are high during high temperature exposure.

6.8.2 STRENGTH CHARACTERISTICS

A. General

Two major qualities of bonded fabrication make it desirable for use on space vehicles. These qualities include:

1. Structural Efficiency

Because composite structures or fabrications permit efficient placement of materials they often perform better in terms of structural efficiency, load per pound of structure, etc., than a homogeneous isotropic material.

2. Damping Characteristics

Since the bond in many sandwich fabrications can absorb energy, sandwich in general has performed well in vibration environments. For this reason it is often used for flaps and control surfaces on high speed aircraft where conventional riveting or spot welding is not acceptable (Reference 6.20).

B. Strength-Temperature Relationships

To be acceptable for terminal sterilization, an adhesive must be able to withstand the specified sterilizing temperatures (in this case 150°C) for a considerable period of time. An investigation of which adhesives can withstand this environment can start

with the Mil-Specs and their QPL's. For structural adhesives two often used specifications are:

MIL-A-25463 Adhesive, Metallic Structural Sandwich Construction
MIL-A-5090 Adhesive, Airframe Structural, Metal-to-Metal

These specifications differ in that MIL-A-5090 uses a lap shear test while MIL-A-25463 uses a standardized peel test. Both of these specifications divide adhesives into Types and Classes. These divisions are summarized on Figure 6.8-4.

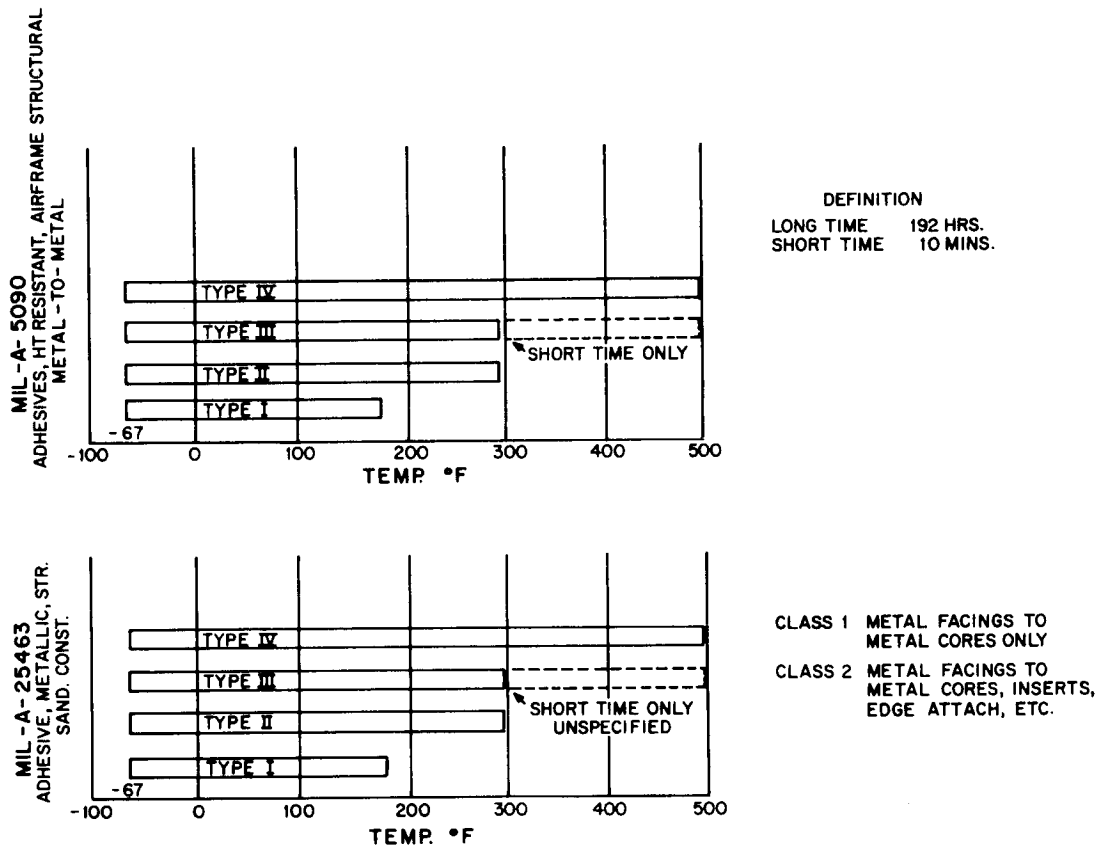


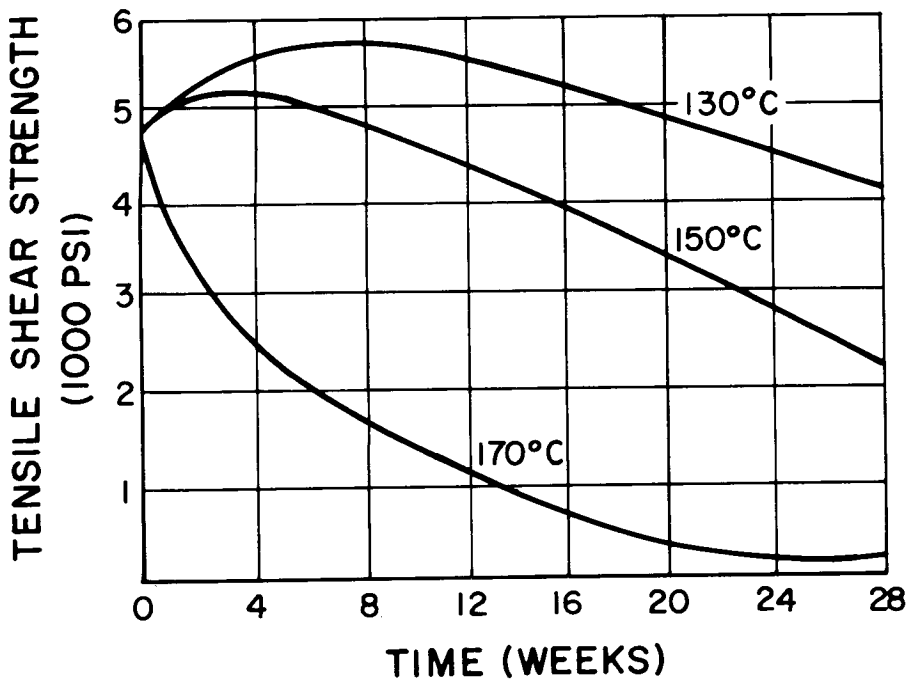
Figure 6.8-4. Comparison of Adhesive Specifications

By checking the latest available QPL's it can be seen that not many adhesives are available which meet the specification requirements for elevated temperature. Figure 6.8-5 shows a comparison of qualified adhesives for these QPL's. Figure 6.8-6 illustrates the effects of time and temperature on a typical epoxy adhesive (Reference 6.21). Adhesives technology is advancing rapidly and should be subjected to continual monitoring.

QPL 25463-7 5 OCT 63		
TYPE	ADHESIVE	MFGR
I CL 2	METAL BOND 408	NARMCO
I CL 1	AF 56205 EC 1956 AF 205 AF 60	3M
II CL 2	HT 424	BLOOMINGDALE
II CL 3	HT 424	"

QPL 5090-17 6 AUG 62		
TYPE	ADHESIVE	MFGR
III	PLYMASTER ACG 1031	PPG
	EPON 422	SHELL
	AEROBOND 422 430	HILLER
	HT 424	BLOOMINGDALE
III & IV	AF T431	3M

Figure 6.8-5. Qualified Adhesives (Per QPL)



NOTES -

1. ADHESIVE CURED AT 170° C
2. STRENGTH MEASURED AT ROOM TEMPERATURE

Figure 6.8-6. Effects of Service Temperature and Time on Typical Epoxy

C. Bonded Sandwich

Bonded sandwich has been used extensively for light-weight spacecraft design. It has been utilized in both the Mariner and Voyager studies made by General Electric.

Core manufacturers offer very little data on their adhesives. This is understandable, due to the relative complexity of testing which is necessary to obtain these values.

It is reasonable to expect that the node bond in core materials will degrade and that some core properties (particularly core shear modulus) will degrade with exposure. Manufacturers are optimistic about the ability of sandwich to withstand high temperature. One manufacturer (Reference 6.22) offers the information shown in Table 6.8-1. Unfortunately the manufacturers do not indicate how much of the total strength loss is due to the loss of strength in the core.

This area may not be considered as an immediate critical problem, but it should certainly be investigated further for any particular design application.

TABLE 6.8-1. SUGGESTED SANDWICH CONSTRUCTION FOR ELEVATED TEMPERATURES

Temp. °F	Core	Facings	Skin Bond	Comments
up to 350	Al	Al	Resin	A suitable bonded aluminum honeycomb sandwich may be expected to retain three-fourths of its design strength at 300°F and about two-thirds of its design strength at 350°F.
up to 500	Special Glass-Plastic	Laminates Resin, Steel or Titanium	Resin	A glass-fabric honeycomb made with heat resistant resins shows good strength at 800°F. Some resins begin to deteriorate after a few hours at this temperature.

D. Ceramic Adhesives

One class of adhesives which may be very useful are the ceramic adhesives. These materials were originally developed for high-temperature, high-strength aircraft applications. Work in this area has produced some adhesives which retain 1000 psi or more after exposure at 1000°F. Laforte (Reference 6.23) has stated that ceramic adhesives can be developed which, when applied under controlled conditions will yield shear strengths in excess of 200 psi at temperatures through 1000°F. At 800°F and 1000°F, shear values may be expected to approach 4000 psi.

6.9 MISCELLANEOUS JOINTS

6.9.1 DESIGN CHARACTERISTICS

A. General

Aside from the previously discussed joining techniques usually associated with structure, there are numerous specialty fasteners used throughout a vehicle which do not warrant individual classification in a general investigation. These fasteners include such items as tube and harness clamps, tube fittings, electrical connectors, spacers, seals, insulation fasteners, etc. Although these items are usually considered secondary by most structural designers, the sterilizing environment may invoke penalties which could lead to structural failure in equipment mounting and connections. Before any specific fastener or joining method is selected for use in a sterilizing environment, a detailed analysis should be performed.

B. Cases 7 and 8 - Joint Summary

All the joints shown in Cases 7 and 8 represent typical joints commonly used on spacecraft. An observance of these joints immediately reflects the many built-in biological "traps" which are inherent in their design. The use of any of these joints precludes the possibility of flushing dead organisms if the joint is assembled prior to sterilization. Precleaning and handling similar to that required for bolted or riveted joints would be sufficient to permit utilization of these specialty joints.

6.9.2 STRENGTH CHARACTERISTICS

A. General

Because of the many types of joints which may be grouped in this category, it is difficult to predict definite problems for any particular joint type. Unlike other fastening methods where variables such as size and materials may be predicted within a rather narrow range, the special joints in this category may have a considerable spread in these same variables.

In general the sterilization thermal environment may introduce a significant load condition and in some cases may require redesign of the joints.

B. Cases 7 and 8 - Joint Summary

Joints A-7 and D-7 are two examples which may be used to point out potential problem areas. Joint A-7 is a marman-clamp device at a separation plane. A steel strap in tension is usually used to provide the forces necessary to prevent separation. If an aluminum ring is used, at temperature forces which will be additive to the initial forces, will be induced in the ring and strap. Conversely, under flight loading, the induced forces are of a relieving nature, so that with a proper choice of initial strap tension, the tie-down load is never exceeded.

Besides resulting in the critical design condition, the thermal environment will also accelerate the creep mechanism, which in this case, will be of a relaxing nature. Since a load readjustment after sterilization may be impractical, the flight design load must be increased by an amount which will ensure the desired minimum value after creep.

The spring shown in D-7 is also susceptible to load build-up and creep, and the thermal environment is a potential trouble maker when fine accuracy is required on the spring constant.

6.10 TECHNIQUES FOR IMPROVING JOINT CONDUCTANCE

6.10.1 GENERAL

Numerous potential solutions have been proposed to improve the thermal conductance across "real" spacecraft joints. This summary will discuss the methods, approaches, and expected usefulness of such data.

6.10.2 JOINTS IN VACUUM

The first condition to be discussed will be for the case of a joint in a vacuum, since this is considered to be the worst case. Bare metal-to-metal conductance for typical spacecraft materials have been reported by Fried (References 6.24, 6.25 and 6.26), Jacobs and Starr (Reference 6.27), Clausing (Reference 6.28) and Aron and Colombo (Reference 6.29). All data except (Reference 6.24) and (Reference 6.29) are for cylindrical sample metal surfaces under load. Reference 6.24 deals with flat, thin plates and Reference 6.29 with a bolted joint made of flat plates. Figures 6.10-1 and 6.10-2 show results of References 6.24 and 6.25 data. Figures 6.10-3 and 6.10-4 show data from Reference 6.25.

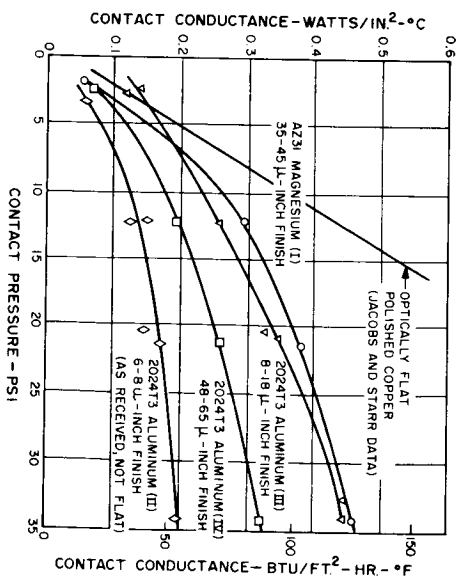


Figure 6.10-1. Contact Conductance vs. Contact Pressure for Bare Metal Joints

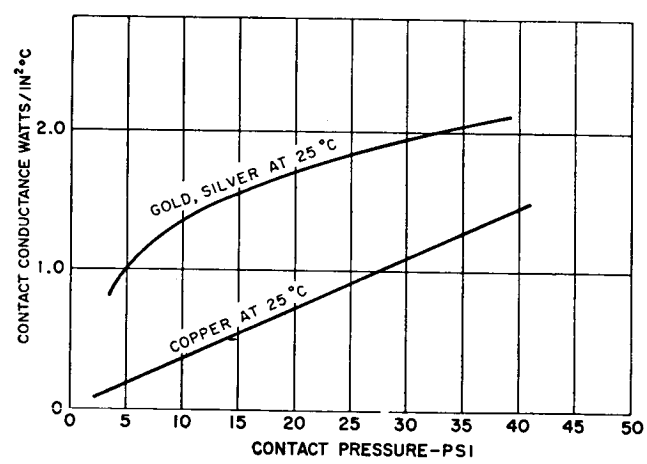


Figure 6.10-2. Contact Conductance vs. Contact Pressure for Gold, Silver, and Copper Joints in Vacuum

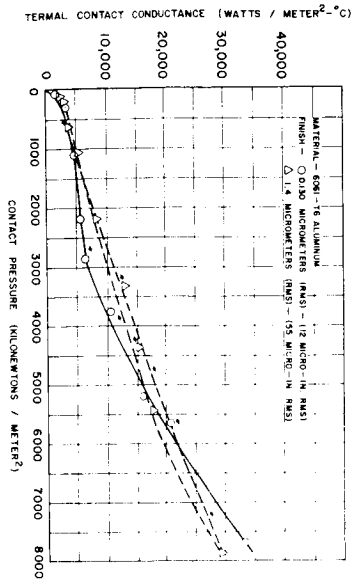


Figure 6.10-3. Thermal Conductance vs. Interface Contact Pressure (Aluminum)

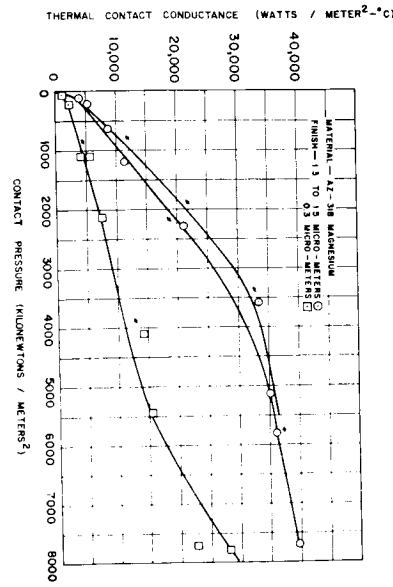


Figure 6.10-4. Thermal Conductance vs. Interface Contact Pressure (Magnesium)

Ideally, the thermal conductance of a real joint should be predictable, provided the contact pressure and contact area are known since sufficient gross thermal conductance data exists for a number of materials and surface finishes. In reality there is no practical way to predict the true contact pressure and true contact area since insertion of a pressure sensor would affect the pressure field in the joint.

The closest approach to prediction of the contact pressure and pressure distribution was performed by Fernlund (Reference 6.30) and refined by Lardner (Reference 6.31). Lardner considers an annular pressure applied normal to the plate representative of the joint, and symmetrical about the hole (or bolt) centerline. This configuration, shown in Figure 6.10-5 shows the idealized representation of a bolted joint with an annular load. The midplane stress is compressive under this loading up to about a radius equal to twice the loading radius, when it becomes tensile.

Coker and Filon (Reference 6.32) as cited in Reference 6.29 reported a case similar to the above and to Reference 6.30. Applying elastic theory to an infinitely long straight beam acted upon by two equal but oppositely directed normal forces at opposite points on the beam (Figure 6.10-6) they were able to show that along the midplane, the pressure reduces to zero at 1.3 times the half thickness of the beam and, further out is replaced by tension. This would indicate that if the single beam were replaced by an equivalent thickness joint (2 beams), these beams would separate at 1.3 t measured along the interface from a point under the load.

Aron and Colombo (Reference 6.29) constructed a two-dimensional photo-elastic model to check this phenomenon and were able to prove it. They also constructed a photo-elastic model of a single-bolt joint (Figure 6.10-7) to obtain a 3-dimensional case. For an estimated 800 lb tension on the bolt they were able to show that a zero pressure point was observed at 1.7 t. The peak pressure was calculated to be 5000 psi. One could thus find the average pressure on the annular contact and use existing thermal conductance data to find the heat flow through the joint, assuming that the interface is otherwise representative of the data to be used.

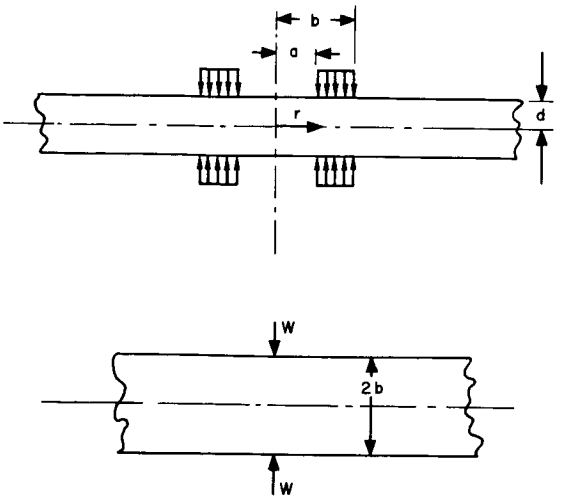


Figure 6.10-5. Idealized Representation of a Bolted Joint with an Anular Load (Data from Reference 6.31)

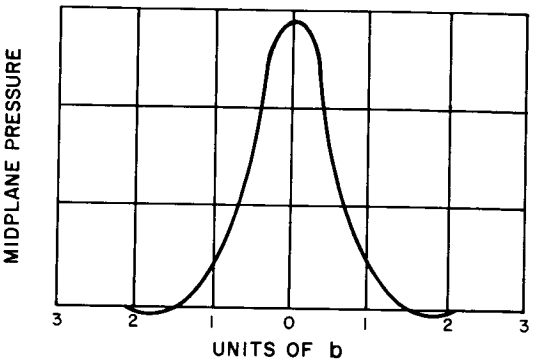
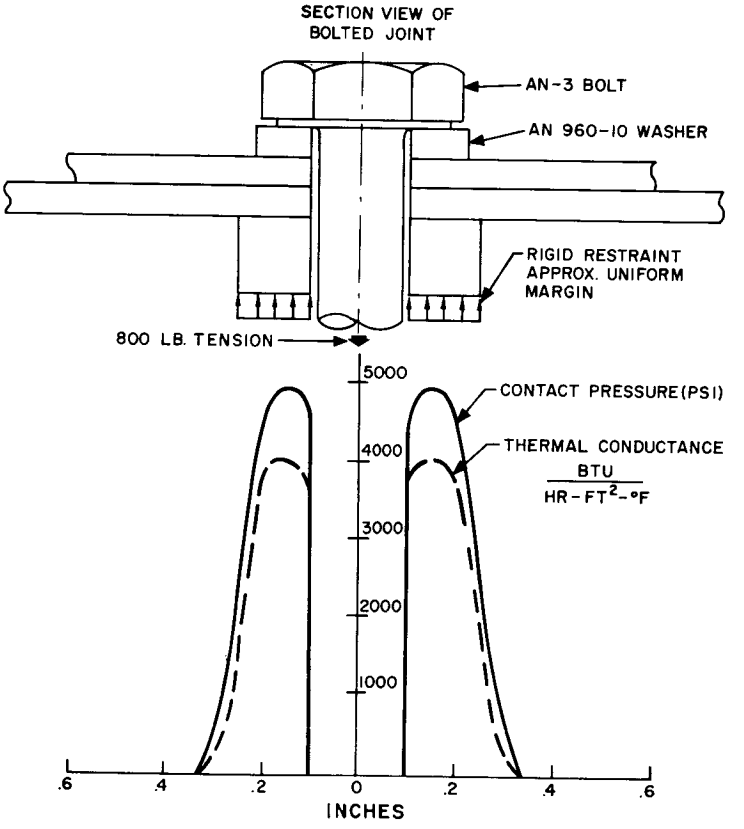


Figure 6.10-6 Pressure under Two Equal, Opossitely Directed Normal Forces at Corresponding Points on Opposite Surfaces of a Long Straight Beam (Data from Ref. 6.31)

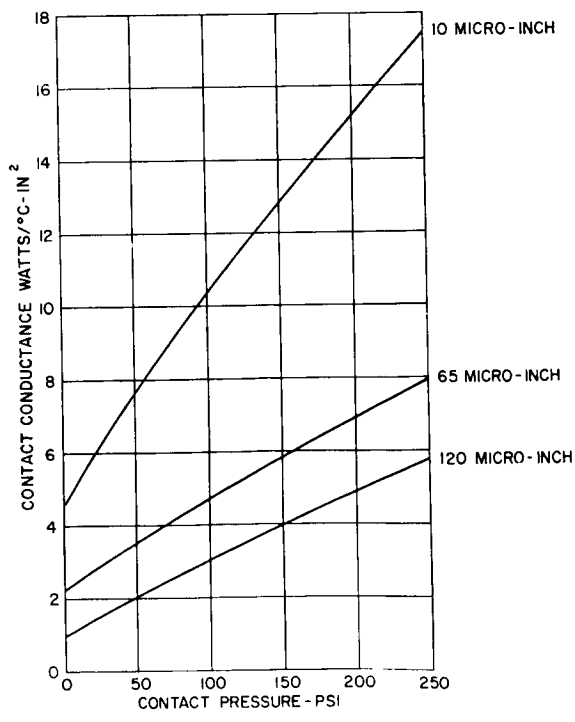


6.10-7. Thermal Conductance of Single Bolt Joint (Data from Reference 6.29)

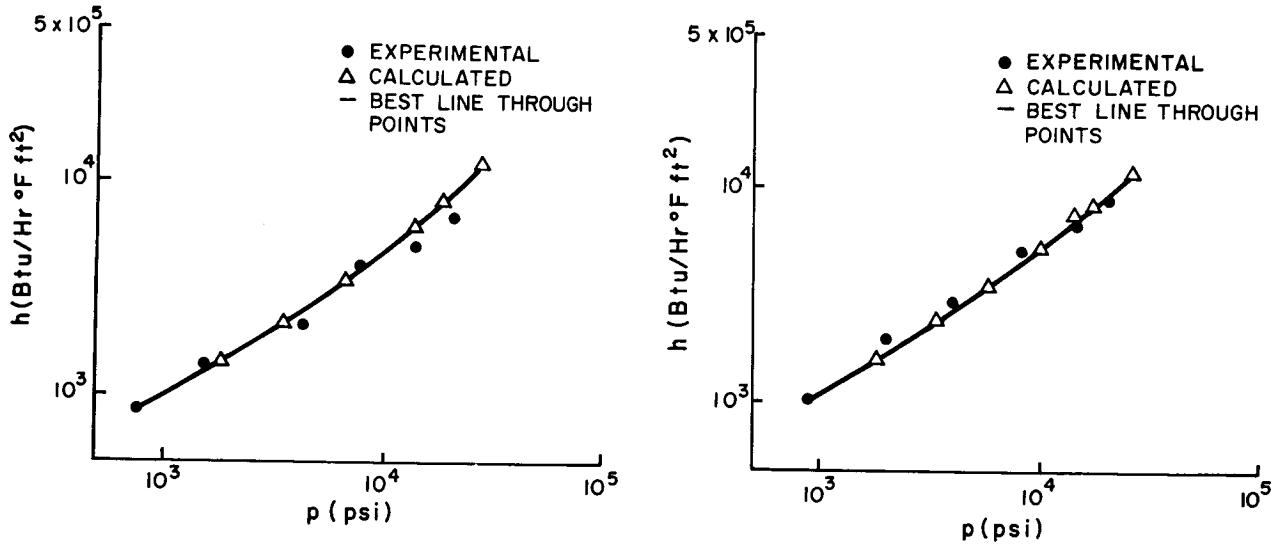
There is no vacuum contact conductance data available in the open literature for riveted joints although some work is currently in progress.

6.10.3 JOINTS IN AIR AND OTHER GASES

The presence of a conducting medium in such a joint tends to overcome some of the limitations inherent in joints in a vacuum. There exists a great deal of thermal contact data in air for simple metallic contacts, riveted contacts, contacts with fillers, etc.; however, no such data for bolted joints is available. The most useful general data for spacecraft-type materials is that reported by Barzelay et al (References 6.33, 6.34, 6.35 and 6.36), Boeshoten and Van Der Held (Reference 6.37), Fenech and Rohsenow (Reference 6.38), Laming (Reference 6.39), Weills and Ryder (Reference 6.40) and Henry and Fenech (Reference 6.44). Figure 6.10-8 shows representative results from Reference 6.34 and Figure 6.10-9 results from Reference 6.44. There exists, however, only a general consistency of qualitative data, since the experimental conditions and surfaces measurement techniques were quite inconsistent. The presence of a conducting gas tends to reduce the over-riding effect of non-flatness somewhat, however, the major heat flow path at all but low contact pressures is by solid conduction. There exists no satisfactory method to determine the surface roughness mating characteristics, other than profilometer traces, which represent the mating for one particular line. This is one shortcoming of the method proposed in References 6.38 and 6.44. The most promising approach would be the use of statistical correlations of representative joint data for which all significant parameters are known. The reader is referred to the cited literature for further details.



6.10-8. Contact Conductance vs Contact Pressure for Aluminum Joint for Different Roughness Values



6.10-9. Predicted and Experimental Conductance vs. Pressure for Stainless Steel Contact (Data from Reference 6.44)

6.10.4 JOINTS WITH METALLIC FILLERS

One approach to minimize the effects of surface finishes and to assure a more reproducible joint from a heat transfer viewpoint, is the inclusion of a deformable metallic filler sheet in the joint.

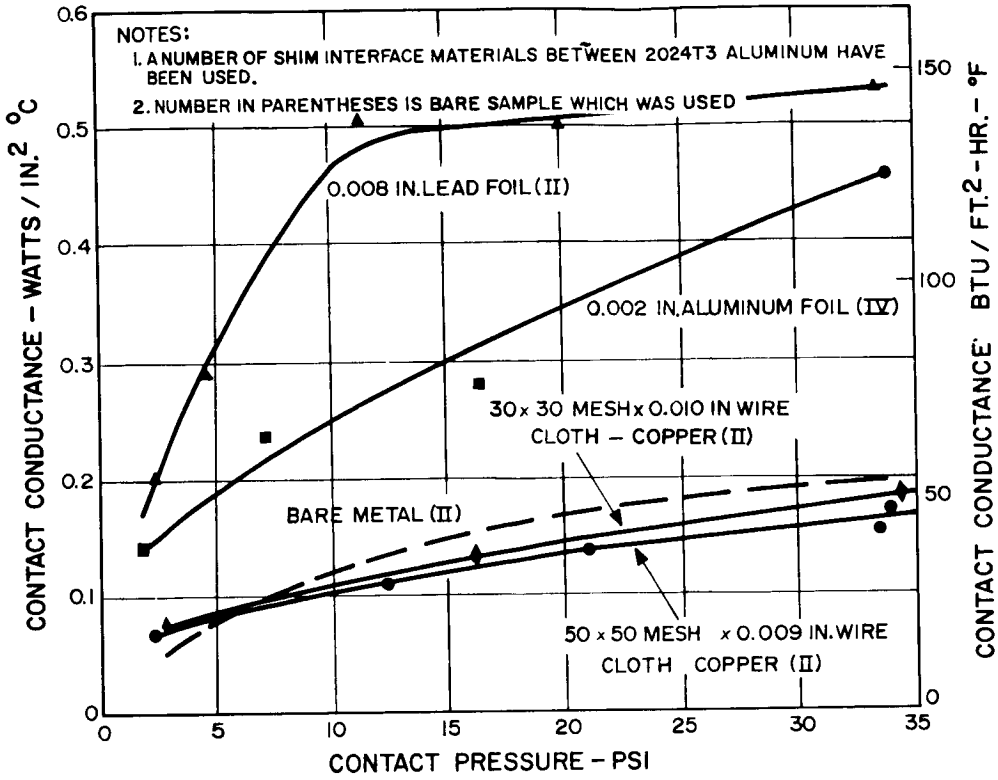
The most prominent material at present is indium foil, a material presently in use or proposed by JPL. From a sterilization standpoint, indium is not desirable because of its low melting point of 312°F.

Its main advantage over lead is its lower density. However, for the intended purpose of thermal sterilization lead or in many cases aluminum foil should be satisfactory.

Data on indium or lead foil in a joint in vacuum is given by Jansson (Reference 6.41), Fried (Reference 6.24) and by Stubstad (Reference 6.42). For a joint in air data is given by Brunot and Buckland (Reference 6.43) and Barzelay (Reference 6.33). Figure 6.10-10 shows data of Reference 6.24, and 6.10-11 data of Reference 6.41. A number of other metallic filler approaches are given by Stubstad (Reference 6.42) who considered Apollo electronics heat sinks in a vacuum.

6.10.5 JOINTS WITH NON-METALLIC FILLERS

In order to provide a heat transfer promoting filler which will flow freely into the surface discontinuities or to provide an electrically non-conductive filler, materials like epoxies, elastomers and greases were used.



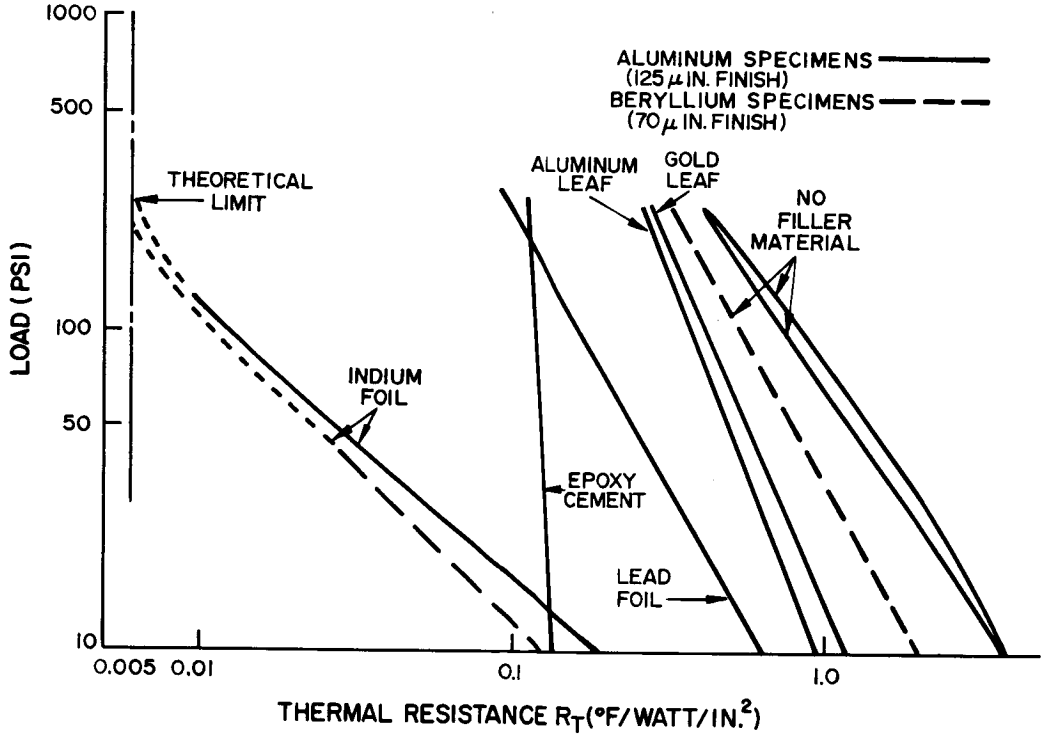
6.10-10. Thermal Contact Conductance vs. Contact Pressure

The filler providing the most remarkable improvement in vacuum heat transfer, high vacuum silicone grease, may not be suitable for sterilization at 150°C. This may not be due to breakdown, but rather due to viscosity change and migration and creep. Data on thermal performance with grease is shown by Fried (Reference 6.26), Clausing (Reference 6.28), and Stubstad (Reference 6.42) for vacuum conditions. Figures 6.10-12 and 6.10-13 show such data.

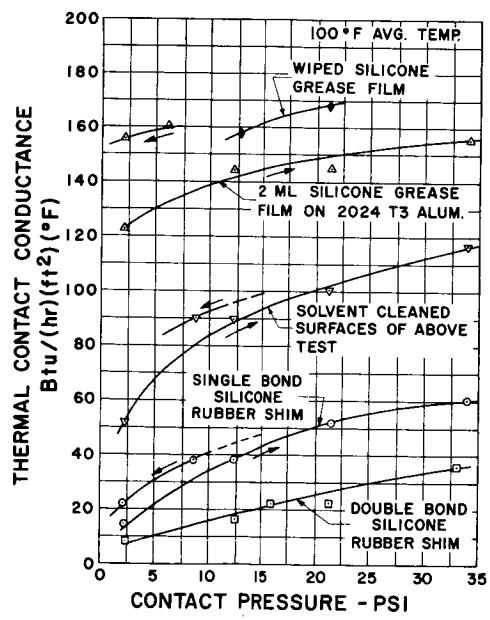
Data for epoxies are shown by Jansson (Reference 6.41). Epoxies may be able to withstand sterilization temperatures, depending upon composition.

Elastomers, primarily room temperature vulcanizing and other silicone rubbers, appear to hold some promise. Fried (Reference 6.26), and Stubstad (Reference 6.42) report data which shows no improvement in heat transfer, but may remain constant with surface waviness effects. See Figure 6.10-12.

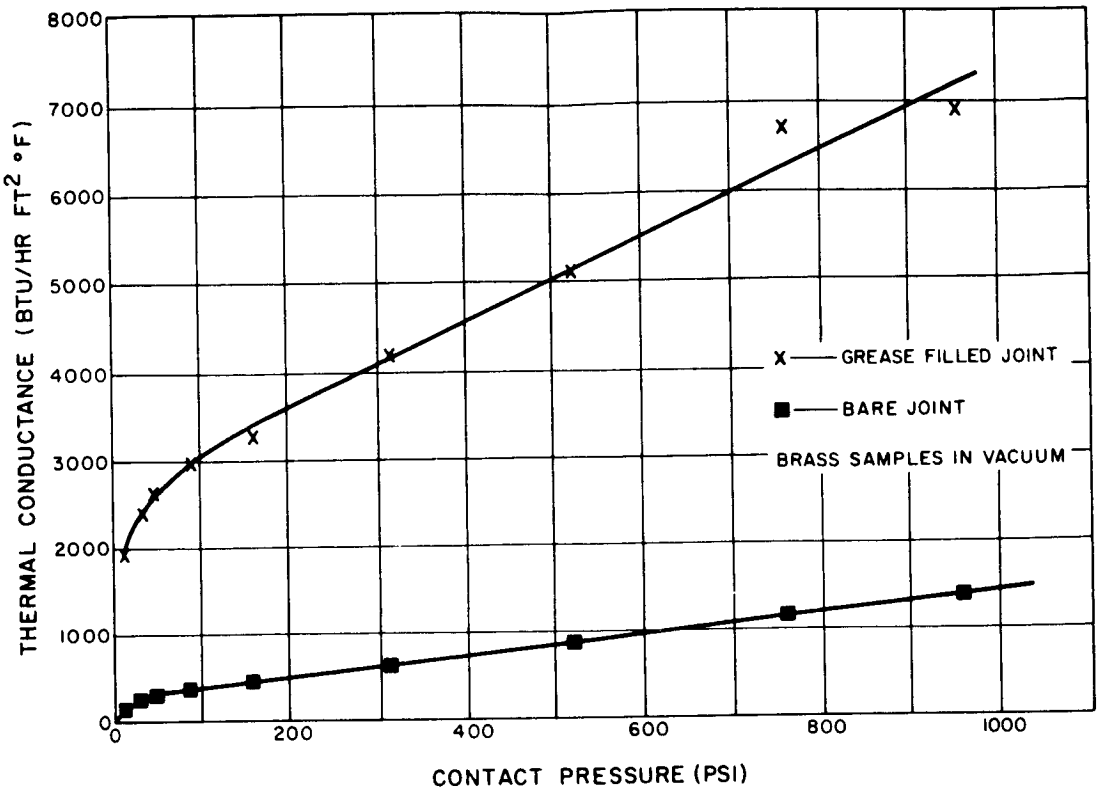
Jansson (Reference 6.41) shows the highest conductance for indium foil, followed by epoxy cement, lead foil, aluminum foil, gold foil in that order. See Figure 6.10-13.



6.10-11. Thermal Resistance of metallic interface with various filler materials (Data from Reference 6.41)



6.10.12. Thermal Contact Conductance vs. Contact Pressure



6.10-13. Influence of a Conducting Fluid on Thermal Conductance vs. Contact Pressure

6.10.6 SUMMARY

This information has been summarized in Table 6.10-1. Specific discussions of "Temperature Distribution in Connectors" and "Temperature Distribution in Bolted Joints" are presented in Appendix J. For a study of "Interface Phenomena for Metals in Contact Under Normal Pressure" see Appendix K.

6.11 REFERENCES

6.1 "Thermo-Structural Analysis Manual," Technical Report No. WADD-TR-60-517, Volume I, August 1962.

6.2 "Thermo-Mechanical Analysis of Structural Joint Study," Technical Report No. WADD-TR-61-151, January 1962.

6.3 M.E. Barzelay and G. F. Holloway, "Effect of an Interface on Transient Temperature Distribution in Composite Aircraft Joints," NASA TN 3824, 1957.

6.4 E. Fried and F.A. Costello, "Interface Thermal Contact Resistance Problem in Space Vehicle," ARS Jour. Vol. 32 No. 2, 1962.

TABLE 6.10-1. TECHNIQUE FOR IMPROVING THERMAL CONDUCTANCE ACROSS STRUCTURAL JOINTS

Joint Type	References	General Comments	Previous Uses on Space Programs
1. Metal-to-metal (no conducting atmosphere)	6.24, 6.28, 6.29, 6.25, 6.27 and 6.26	Reference 6.24 deals with flat plates, Reference 6.29 with straps, remainder with cylinders	Reference 6.24 used by STL, GSFC, JPL, and GE
A. Bolted	6.29		
B. Riveted	None		
C. Brazed	None	No problems expected	
2. Metal-to-metal (conducting atmosphere)	6.39, 6.38, 6.29, 6.36 and 6.44	A considerable amount of data exists, but not all are for spacecraft materials.	None
A. Bolted		Use data of References 6.39, 6.38, 6.33, 6.34 and 6.37	None
B. Riveted	6.35 and 6.36	Aircraft stringer skin data	Aircraft use only
C. Brazed		No problems expected	
3. Filler sheets - metallics	6.24, 6.41, 6.42 and 6.43	Tests were performed in air and vacuum	No space use, but work in References 6.26, 6.41 and 6.42 intended for space use
A. Lead	6.24, 6.41		
B. Indium	6.41, 6.42		JPL uses indium foil
C. Aluminum	6.24, 6.33, 6.43		
D. Etc.	6.41		
4. Fillers - non metallics		Many of these materials are not suitable for thermal sterilization because of temperature limitations	
A. Epoxies	6.41		
B. Greases	6.26, 6.28 and 6.42		Grease has been used by STL
C. Greases w/metallic			Used by Gemini program by McDonnell
D. Elastomers	6.26 and 6.42		In use by STL
E. Other - (thermal fuzz, etc.)	6.42		Reference 6.42 reports a large number of low temperature heat transfer promoting fillers for Apollo program.

6.5 A. Clausing and B. T. Chao, "Thermal Contact Resistance in a Vacuum Environment," U. of Ill. Report ME-TN-242-1, 1963.

6.6 S. J. Rosenfeld, "Analytical and Experimental Investigation of Bolted Joints," NASA TN 1458, October 1947.

6.7 Robert W. Northrop, "Structural Design of the Tiros Meteorological Satellite," paper read at Structural Design of Space Vehicles Conference, Santa Barbara Biltmore, Santa Barbara, California, April 1960.

6.8 Moses Tawil, "Structural Damping Improved by Riveting," Space/Aeronautics, November 1962.

6.9 "Project Relay," Quarterly Technical Report No. 1, Radio Corporation of America, Princeton, N. J., June-August 1961.

6.10 "Research and Development Handbook," Space/Aeronautics, June 1963.

6.11 "Welding Handbook," Fourth Edition, American Welding Society.

6.12 "Brazing Manual," American Welding Society, Inc., Reinhold Publishing Company, 1963.

6.13 J.A. Chamberlin and A.J. Meyer, Jr., "Project Gemini Design Philosophy," Astronautics and Aerospace Engineering, February 1963.

6.14 "Design Criteria for Zero-Leakage Connectors for Launch Vehicles," General Electric General Engineering Laboratories Quarterly Progress Report No. 3 Contract NAS-84012, January 11, 1963.

6.15 R.R. Heldenfels, "Structures and Materials," Astronautics and Aerospace Engineering, November 1963.

6.16 "Dynamics and Structures," Space/Aeronautics, July 1963.

6.17 E.S. Wynne, Comment During Conference on Spacecraft Sterilization, July 9, 1962, NASA, Washington.

6.18 J.T. Cordaro, "Studies on the Prevention of Contamination of Extra Terrestrial Bodies: Bacteriologic Examination of Hermetically Sealed Electronic Components,' SAM Report 62-18, November 1961.

6.19 "Summary Recommendations," Second Meeting of the Adhoc Committee on Contamination of Extraterrestrial Exploration, Planet, Space Science, Pergamon Press, 1959.

6.20 Thorburn R. Rieben, "Sonic Fatigue Testing and Development of Aircraft Panels," Structural Adhesives and Sandwich Constructions WADC-University of Dayton Joint Symposium, November 1957.

6.21 H.A. Burgman, "The Trend in Structural Adhesives," Machine Design, November 1963.

6.22 Honeycomb Sandwich Design Brochure E, Hexcel Products, Inc.

6.23 H.G. Lefort, "Ceramic Adhesives Score in High Temperature Uses," Ceramic Industry, April 1959.

6.24 E. Fried and F.A. Costello, "Interface Thermal Contact Resistance Problem in Space Vehicles," ARS Journal, Vol. 32, No. 2, p. 237, 1962.

6.25 E. Fried, "Thermal Contact Conductance Study," Final Report Contract NAS 8-5207, March 1964.

6.26 E. Fried, "Thermal Joint Conductance in a Vacuum," ASME Paper 63-AHGT-18, presented at ASME-Aviation and Space Conference, Los Angeles, California, March 1963.

6.27 R.B. Jacobs and C. Starr, "Thermal Conductance of Metallic Contacts," Rev. Scientific Instrument, Vol. 10, p. 140, 1939.

6.28 A.M. Clausing and B.T. Chao, "Thermal Contact Resistance in a Vacuum Environment," University of Illinois, Exp. Station Report, ME-TN-242-1, August 1963.

6.29 W.K. Aron and G. Colombo, "Controlling Factors of Thermal Conductance Across Bolted Joints in a Vacuum," ASME Paper 63WA196.

6.30 I. Fernlund, "A Method to Calculate the Pressure Between Bolted or Riveted Plates," Report No. 245, Trans, Chalmers University, Gothenburg, Sweden, 1961 (In English).

6.31 T.J. Lardner, "Thermal Joint Conductance: Midplane Stress Distributions," JPL-SPS 37-19, Vol. IV, 1963.

6.32 E.H. Coker and L.N.G. Filon, "A Treatise on Photo-Elasticity," Cambridge U. Press, 2nd Ed., p. 436.

6.33 M.E. Barzelay, Kin Nee Tong and G.F. Holloway, "Thermal Conductance of Contacts in Aircraft Joints," NACA TN 3167, 1954.

6.34 M.E. Barzelay, Kin Nee Tong and G.F. Holloway, "Effect of Pressure on Thermal Conductance of Contact Joints," NACA TN 3295, 1955.

6.35 M.E. Barzelay, "Range of Interface Thermal Conductance of Aircraft Joints," Syracuse University, NASA TN D-426, May 1960.

6.36 M.E. Barzelay and G. F. Holloway, "Effect of an Interface on Transient Temperature Distribution in Composite Aircraft Joints," NACA TN 3824, April 1957.

6.37 F. Boeschoten and E. F. M. Van Der Held, "The Thermal Conductance of Contacts Between Aluminum and Other Metals," Physics 23, 1957.

6.38 H. Fenech and W. M. Rohsenow, "Prediction of Thermal Conductance of Metallic Surfaces in Contact," ASME Journal of Heat Transfer, February 1963.

6.39 L. C. Laming, "Thermal Conductance of Machined Metal Contacts," Proceedings 1961 International Heat Transfer Conference, Paper No. 8, American Society of Mechanical Engineers, 1963.

6.40 N.D. Weills and E.A. Ryder, "Thermal Resistance Measurements of Joints Formed Between Stationary Metal Surfaces," Trans. ASME, Vol. 71, No. 3, p. 259, 1949.

6.41 R.M. Jansson, "The Heat Transfer Properties of Structural Elements for Space Instruments," MIT-Instrumentation Lab Report E-1173, June 1962.

6.42 W.R. Stubstad, "Measurement of Thermal Contact Conductance in Vacuum," ASME Paper 63-WA-150, November 1963.

6.43 A.W. Brunot and F. F. Buckland (Mrs.), "Thermal Contact Resistance of Laminated and Machined Joints," Trans. ASME, Vol. 71, No. 3, 1949.

6.44 J.J. Henry and H. Fenech, "The Use of Analog Computers for Determining Surface Parameters Required for Prediction of Thermal Contact Conductance," ASME Paper 63-WA-104, November 1963.

6.45 F.R. Shanley, "Weight-Strength Analysis of Aircraft Structures."

7. MATERIALS

7.1 INTRODUCTION

The requirement for terminal sterilization creates a high temperature environment whose effect on materials requires evaluation. A temperature of 150°C for a soaking period of 60 hours is a typical sterilization cycle for ensuring a low probability of organism survival. A maximum of four such sterilization cycles would possibly be required in case of accidental contamination or the necessity of replacing a portion of the vehicle. The thermal analysis of hypothetical lander configurations presented in Section 5.2 of this report indicates that the total time at elevated temperature could run as high as 400-500 hours. A temperature of 150°C for such periods is definitely a hostile environment for many of the commonly used structural materials.

Although this temperature is not considered exceptionally high, it is a new design requirement whose effect on material properties, both at this temperature and at normal operating temperatures subsequent to long term exposure, must be carefully considered. It is theoretically possible to construct a spacecraft structure to withstand thermal sterilization since there are sufficient materials available which can perform satisfactorily in this environment without undergoing property degradation. The design and construction of interplanetary spacecraft will be simplified, however, if high utilization of current materials can be obtained.

7.2 SUMMARY

7.2.1 GENERAL DATA

An evaluation of spacecraft constructional materials and their properties was prepared to determine their compatibility with a thermal sterilization process. This is presented in Appendix P, Structural Materials. For the purpose of this study, it was assumed that a spacecraft would undergo a maximum of four cycles at 150°C with each consisting of a 60-hour soak at this temperature. Results of the thermal analysis of Section 5.2 indicate that this 240-hour period could easily be extended to 400 to 500 hours for certain parts of the vehicle because thermal lag dictates lengthy heating and cooling periods.

The materials evaluated are divided into several categories such as metallic alloys, adhesives, composites, and thermal shield materials. Physical properties, both at room temperature and at elevated temperatures, are presented and discussed as they pertain to thermal sterilization. In particular, thermal conductivity and differences in thermal expansion characteristics of the materials are significant. Coefficients of thermal expansion among the various materials vary from 5×10^{-6} to 100×10^{-6} in/in/°F. Some of the materials, such as reinforced plastic honeycomb or the thermal shield materials,

are actually used for their insulating properties and will lengthen the spacecraft heating and cooling times.

Various mechanical properties at elevated temperatures are presented. In many cases data are also plotted for temperatures above 150°C to show the location of any major change in properties with temperature. The effects of aging at 150°C upon subsequent room temperature properties are also given. For metallic materials, the primary aging effects are thermally induced phase changes which usually result in a loss of strength. Aging of organic materials is more complicated with oxidation playing a major role. The dry, inert gas atmosphere proposed for use during sterilization will decidedly reduce degradation of organic materials.

7.2.2 METALS

Of the metallic materials, magnesium alloys, as a class, are the most questionable for use as structural materials in spacecraft which require thermal sterilization. Currently favored alloys, such as AZ31B-H24 and ZK60A-T5, as well as the new lightweight alloy, LA141, suffer an excessive loss in strength at 150°C. Creep resistance, in particular, is extremely low. The newer thorium-alloys, such as HM21-T8 and ZH62A-T5, do appear suitable for these applications. ZH62A, a casting alloy, is being used on a current satellite, but for its high room-temperature strength rather than its elevated temperature properties.

Several aluminum alloys in common use in current space vehicles are able to withstand thermal sterilization e.g., 2014-T6, 2024-T4, and 6061-T6 are satisfactory for extended exposure to 300°F. The higher strength alloys, 7075-T6, 7079-T6 and 7178-T6, incur loss in room-temperature strength properties as a result of prolonged exposure to 300°F. It is quite likely that they will end up having lower strengths than 2014-T6 and 2024-T4 if multiple sterilization cycles are required.

Titanium and steel alloys were evaluated since they have been proposed for structural materials for Venus landers. The titanium alloys and the stainless steels have excellent strength retention and high creep strength at temperatures as high as 1000°F. In addition, aging at the lower temperatures proposed for sterilization has no noticeable effect on these properties. The low alloy steels are also satisfactory, particularly the AISI-H11 types.

7.2.3 STRUCTURAL ADHESIVES

Organic adhesives are available which can withstand 300°F. These adhesives are polymers and copolymers based upon thermosetting type resins, such as phenolics, or upon the silicone resins, Some of the more commonly used spacecraft adhesives, like the vinyl-phenolics and the polyamide-epoxies, are not acceptable. Polybenzimidazale adhesives, a recently developed type which can be used at temperatures as high as 700°F, require curing at 600°F with a 750°F post cure. Because of these high temperatures their use as typical space vehicle materials will be limited.

Data are presented which show the effects of elevated temperature aging upon organic resins and adhesives. Oxidation as a result of exposure in air results in a significant loss of room temperature properties such as elongation, impact strength, tensile strength, and flexure strength. Aging of similar specimens in dry nitrogen causes much less loss in subsequent properties. In this case the effect is primarily that of heat energy upon the polymer bonds. Scission of the molecular chains or increased cross-linking are two reactions which can occur. Dry, inert gaseous atmospheres should be used when testing organic materials for their resistance to thermal sterilization.

7.2.4 STRUCTURAL COMPOSITES

Honeycomb and laminated composites are suitable for spacecraft subjected to sterilization. A few of the high temperature structural adhesives are suitable for bonding of honeycomb, and special adhesives have been developed. Phenolic-glass appears best of the current reinforced plastic honeycomb materials. For laminated plastics, phenolic-glass or silicone-glass combinations are preferred.

7.2.5 ABLATING MATERIALS

The area where the least amount of data and experience is available is undoubtedly that of the thermal ablation shield and its structural bond. Phenolic-nylon composites have been extensively used and are proposed by General Electric for a Venus lander. This material may be marginal and additional testing will be required to establish its suitability.

A new class of foamed elastomeric thermal shield materials based upon silicone rubber formulations has been developed. These materials are particularly suited to a long time, low heat flux entry such as would be experienced by a Mars lander. Test results to date are encouraging, and no reason is apparent why these materials would not successfully withstand thermal sterilization.

7.3 CONCLUSIONS

In general, structural materials will not be a limiting item in the design of space vehicles to withstand thermal sterilization. Special materials, however, must be substituted for some currently used materials because of their inability to withstand extended periods of time at 300°F.

Even for those materials considered acceptable, additional testing should be conducted to verify properties at temperature and effects of aging. In certain areas, such a thermal shield materials, testing is definitely required.

8. MANUFACTURING CONSIDERATIONS

8.1 GENERAL

Many of the manufacturing and fabrication requirements associated with terminal sterilization were considered by the Joint Study, Section 6 where the biological load reduction concept was introduced. Load reduction was also recommended in the "Final Guidelines for Spacecraft Sterilization" issued by the NASA Office of Space Sciences, Iowa City Summer Study. Implicit in this concept is the existence of work areas in which decontamination can occur or in which contamination levels can be controlled.

The technical literature on sterilization advocates the use of ultrasonics, clean room assembly, monitoring of manufacturing processes, sealed packaging, training and indoctrination, plus dry heat and/or ethylene oxide treatment. With the exception of the application of heat and gas, this program consists of techniques already employed to establish and maintain cleanliness*. For this reason government complaints of shoddy housekeeping in the space industry are both relevant and disturbing. They warrant asking several basic questions, such as:

a. What is lacking in industry's approach to contamination control?

b. Is the capability of the clean room understood and properly exploited?

c. How realistic are the goals of sterilization programs?

d. Is it sufficient to employ cleanliness techniques with greater diligence in order to achieve sterility?

Although the present study was not primarily concerned with manufacturing problems, the answers to these questions are of sufficient interest to justify their inclusion in this study. Furthermore, for many space vehicles, the distinction between "structure" and "vehicle" simply does not exist on the shop floor. The problems and requirements for clean rooms and clean work stations as they relate to the sterilization problem are discussed in Appendix L. Although work is in progress on contamination control for both medical and industrial purposes, the concept of "sterile rooms" today exists only in theory.

*Cleanliness and sterility are akin in the sense that cleanliness promotes sterility, but does not guarantee it. However, the two concepts are not synonymous. For example, sterile lint would be incompatible with the cleanliness required during the assembly of pneumatic hardware.

To illustrate the need for controls and procedures for handling Appendix M, "Functional Flow Diagram For Voyager Vehicle," is included in this Report. Although many of the steps shown on this diagram in Appendix M occur after manufacturing is completed, the diagram illustrates the complexity involved in maintaining a sterile vehicle and indicates the need for careful control.

8.2 CONCLUSIONS

Several general conclusions can be drawn from Appendix L.

a. Biological load reduction should be required on any vehicle designed for terminal sterilization.

b. Because of the size of most landers it is doubtful if glove boxes can be used exclusively. It is likely that clean rooms or clean work stations must be used to assist in load reduction.

c. Previous experience has shown that clean rooms and clean room techniques have been misapplied through a lack of real knowledge of their capabilities and limitations. Care should be exercised to ensure proper use of these expensive facilities. Clean room facilities should not be used solely for psychological considerations.

d. To minimize manufacturing problems, detailed flow plans should be maintained to permit planning and control over all hardware. Since a single break in control procedures could offset the work of a complete sterility program, the need for controls and procedures cannot be overemphasized.

9. GENERAL DESIGN GUIDELINES

9.1 FACTORS AFFECTING STERILIZATION

9.1.1 GENERAL

Although sterilization by heat has been considered during this study, and is in fact the simplest and most efficient technique, other methods of sterilization may be useful and feasible for biological load reduction. The literature on sterilization is extensive (Reference 9.1); engineering problems have also been previously discussed (Reference 9.2); gaseous sterilization techniques are well documented (References 9.3 and 9.4). In particular the techniques for sterilizing the Ranger vehicle are well known (References 9.5 and 9.6). The use of radiation or ultrasonics for load reduction on space vehicles is not as well documented. As a general introduction to these techniques, Appendix N, "Radiation Sterilization of Structures and Components" and Appendix O, "Ultrasonic Sterilization," have been added to this report.

9.1.2 GEOMETRIC COMPLEXITY

It has been suggested that the geometric complexity, traps, cracks and crevasses created by structural joints may form pockets from which biological life cannot easily be removed. It has been stated (Reference 9.7) that structural frames could be made of welded tubing to avoid these joints. Interior surfaces could be sterilized by introducing gaseous sterilants through small holes in the frame.

Although it is true that geometric complexity (e.g., thread forms) does entrap gross contaminants, it is also true that gross contaminants can be removed by conventional cleaning techniques.

For members with equal cross sectional areas a butt weld is less complex geometrically than a bolted connection; the bolted connection entails additional parts and cracks which might entrap large particles of contamination. If these large particles are removed by conventional cleaning techniques, the surface of the material becomes an important factor in the case of sterilization.

9.1.3 SURFACE FINISH

A. Basic Definitions

Few design engineers are familiar with the theory of "biological load reduction." Inasmuch as the spacecraft sterilization program uses this as a major premise, it may be useful to briefly define this concept.

The general theory of bacterial order of death concludes that the population decreases logarithmically when exposed to a lethal influence. With heat above a basic lethal temperature, for example, the slope of the curve would be a function of the time-temperature relationship involved (Reference 9.9). The number of organisms (population) decreases in a predictable, orderly manner until nearly all the population is extinct. Unfortunately, the bottom and top of the lethality curves do not obey the logarithmic theory. For example, a few organisms in a given population may continue to survive and, if conditions again become favorable, reproduce long after the theory has predicted complete extermination. It is obvious that factors are involved that are, as yet, unknown. The fact is that resistance varies from species to species, within a species, and even within a given pure spore population. For this reason alone, after theoretical sterilization time and temperatures have been reached, an additional time and temperature margin must be allowed.

Few microbiologists would be willing to estimate or state precisely what this margin should be. In particular, for an unknown population, extreme (by physical science standards) safety margins should be employed.

At this point, the question may properly be asked, why bother? Why not simply heat for a long enough time to convince even the most stubborn die-hard that the equipment is sterile. The answer to this from a biologist's view is "fine"; this is precisely what is done in the laboratory. Unfortunately, financial considerations, effect of temperatures on materials, and the lack of data as to the functional reliability of treated parts make such a course impractical for flight hardware. It is important, therefore, to find the point where the required degree of assurance of sterility may be equated to a practical time-temperature treatment. Figure 9.1-1 and Figure 9.1-2 are typical curves illustrating the previous discussion.

Introduction of some new terms is convenient at this point. Let D equal the time in hours to reduce a given population of micro-organisms to 90%. (It is apparent that the population would never be certified sterile, no matter how long one treated it.) Then, using an equation given by Stumbo (Reference 9.10),

$$D = \frac{t}{\log A - \log B},$$

where A is the total number of samples heated, multiplied by the number of spores per sample, B is calculated by assuming one surviving spore per container when less than the total number of containers show some organism survival; and t is the exposure time at the given lethal temperature. This equation was developed for moist heat (much more lethal than dry heat) but may be utilized for initial dry-heat calculations.

To the D value concept, a second concept is added, that of an F value where F is the time required for sterilization to the level of less than one spore. According to Schmidt (Reference 9.11), this may be expressed as follows:

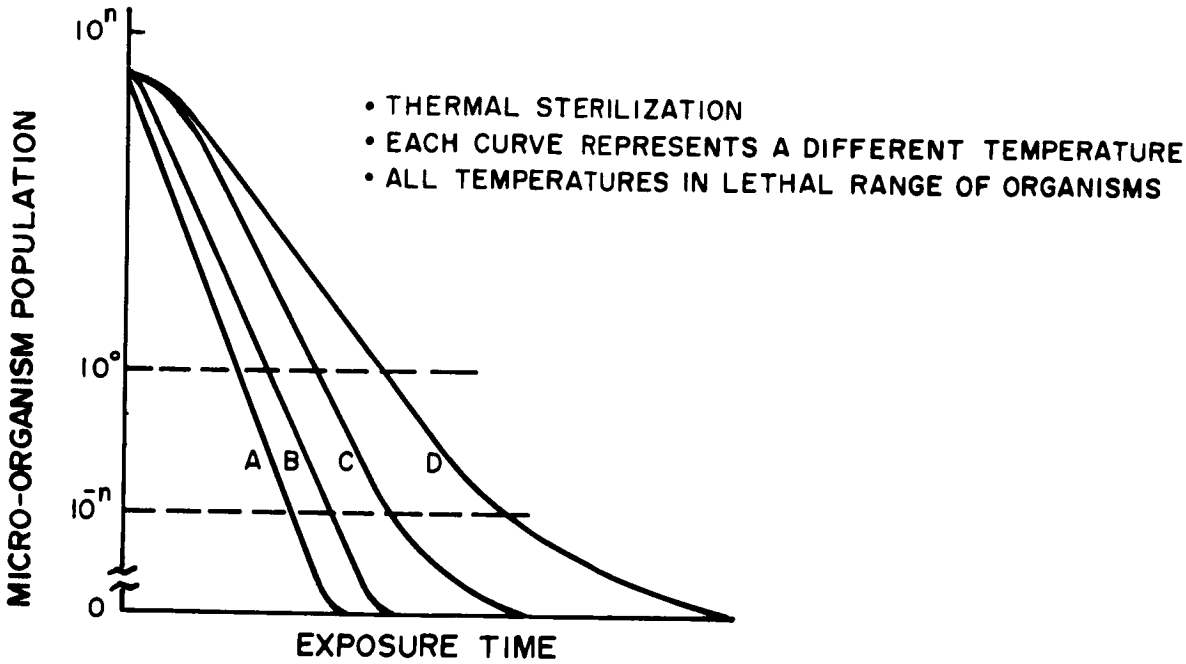


Figure 9.1-1. Survivor Curves for Micro-Organisms, Population versus Exposure Time

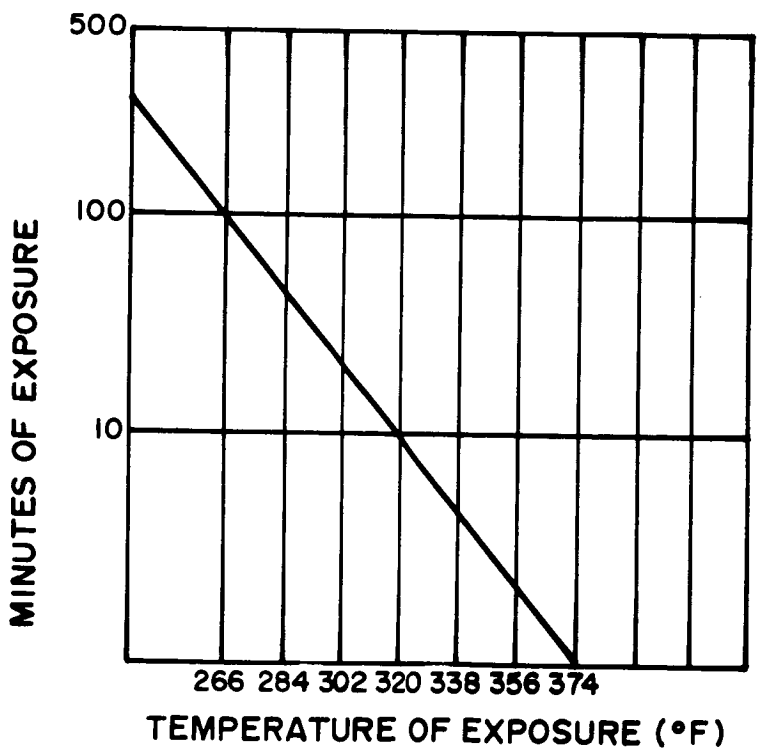


Figure 9.1-2. Typical Curve Showing Resistance of Spores to Destruction by Dry Heat

$$F = D (\log M + 1),$$

when M is equal to the number of spores per replicate, times the number of replicates.

It should be noted that logarithmic reduction is not an established fact but only a theory which generally seems to be borne out by observation.

With these definitions established, it is possible to examine some of the difficulties involved in establishing a numerical indication of the sterility of a space vehicle. First, the total initial population can only be estimated. Secondly, although scientists can establish D values for specific organisms, there is no specific D value for all organisms, and it is necessary to use conservatism in establishing this value. Finally, it is not physically practical to examine a completed space vehicle to determine the level of contamination following a specified sterilizing treatment. These difficulties probably account for the relative severity of the specified thermal treatment.

The usefulness of biological load reduction becomes apparent when Figure 9.1-1 is considered. By starting with a small initial population, it is possible in a given time period to achieve much lower values for sterility. The NASA Office of Space Sciences has established the target for sterilization procedures as 10^{-4}.

If individual details or major subassembly are subjected to sterilization or extensive cleaning, or if care is taken during manufacture to reduce contamination levels, the likelihood of achieving 10^{-4} is increased. The concept of reducing the level of biological life prior to terminal sterilization is known as "biological load reduction."

B. Surface Finish

The work statement for the present contract reads in part:

> "It is highly important to establish methods to avoid traps that could contain living organisms and/or such that could not be cleaned of traces of life. A bolt-and-nut combination cannot be cleaned. Likewise, traces of life can be retained between rivet heads and the part riveted together."

It is important to consider the factors which make a bolt and nut combination difficult to clean. One objective to bolting may be the geometric complexity of a bolted connection when compared to a butt-welded connection. If geometric complexity is considered to be a factor in ease of sterilization, then the question of scale becomes very important. A threaded fastener which to the human eye, appears to have a relatively complex geometric surface, may in fact appear quite different to the organism.

"Where the bacillus lives, gravitation is forgotten, and the viscosity of the liquid, the resistance defined by Stokes Law, the molecular shocks of the Brownian movement, and the electric charges of the ionized medium make up the physical environment and have their potent and immediate effect on the organism. The predominant factors are no

longer those of our scale; we have come to the edge of a world of which we have no experience and where all our preconceptions must be recast." (Reference 9.12).

Surface characteristics may play a part in ease of sterilization. Figures 9.1-3 and 9.1-4 show the variations in F and D values for some organisms on different surfaces (Reference 9.13). From these plots one can infer that rough metallic surfaces are more difficult to clean than smooth surfaces. The range of surfaces which might be encountered on a space vehicle is approximately from 125 micro-inches to 4 micro-inches. Figure 9.1-5 shows some of the processes that might be used to produce these finishes. The methods for measuring these finishes are described in MIL-STD-10A, "Surface Roughness, Waviness and Lay." The general characteristics of surface finishes are discussed in Appendix K.

Surfaces which have the same numerical value for surface finish may vary from one another in relative roughness. Figure 9.1-6 shows how a typical 64-micro-inch finish might vary for different machining techniques. The relationship between organism size and surface finishes is suggested in Figure 9.1-7 and 9.1-8. The organisms shown on these figures were selected to demonstrate typical sizes only. It can be noted that the influenza micro-organism shown in these Figures is below the limits of resolution of the light microscope.

When considering Figure 9.1-8, the removal of dead micro-organisms by wiping appears somewhat less than satisfactory. It becomes even less feasible when it is considered that some bacteria "may secrete mucilaginous substances and become glued to solid

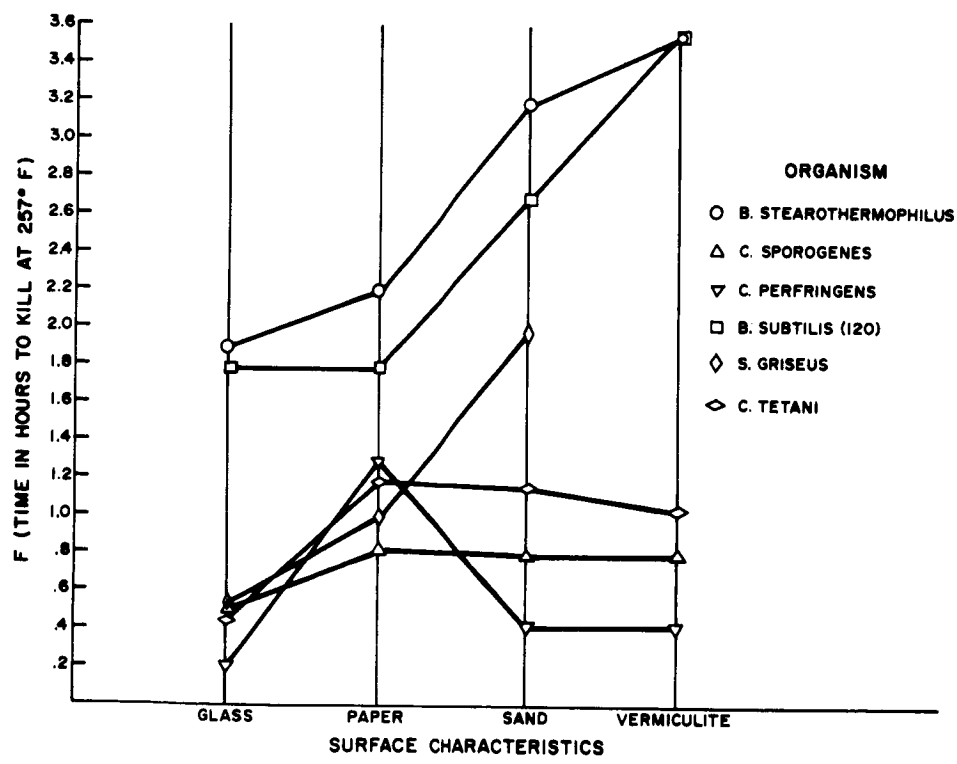


Figure 9.1-3. Variation in F_{257^oF} Values for Different Surfaces

surfaces or they may adhere by means of a morphologically differentiated structure, a holdfast. Solid surfaces serve such organisms as resting places for their growth." (Reference 9.8). It is true that ultrasonic techniques could be used to shake these organisms loose and flush them away, but such techniques would probably never result in complete removal of all viable and nonviable organisms. Fixed-frequency ultrasonic cleaning can leave layers of dirt of contamination, however, sweep-frequency cleaners are available. The load reduction which would result from the use of these or similar techniques is considered very important.

It may therefore be concluded that geometrical complexity is relative not absolute; hence that bolts, nuts or other threaded fasteners may be no more difficult to sterilize (and flush if necessary) than a spherical ball of equal volume. When variations in surface occur as a result of finishes, treatments, coatings or processes, the resistance of organisms (as measured by F or D values) can be expected to vary. As shown in Figures 9.1-3 and 9.1-4, these values do not appear to vary by orders of magnitude. Based on Hobby's estimate of 10^9 organisms on the surface and in the interior of a good "clean" spacecraft, (Reference 9.14) a worst case of 10^{10} organisms can be hypothesized. The ability of a terminal sterilization cycle to lower this load to acceptable levels will depend to some extent on the surfaces and coatings of the vehicle.

The major factors which affect sterilization to a specified probability are: temperature, time of exposure, total initial population, and death rate for a specific organism. The concept of beginning terminal sterilization with a small initial population is therefore a requirement, and previous biological load reduction a necessity. To achieve

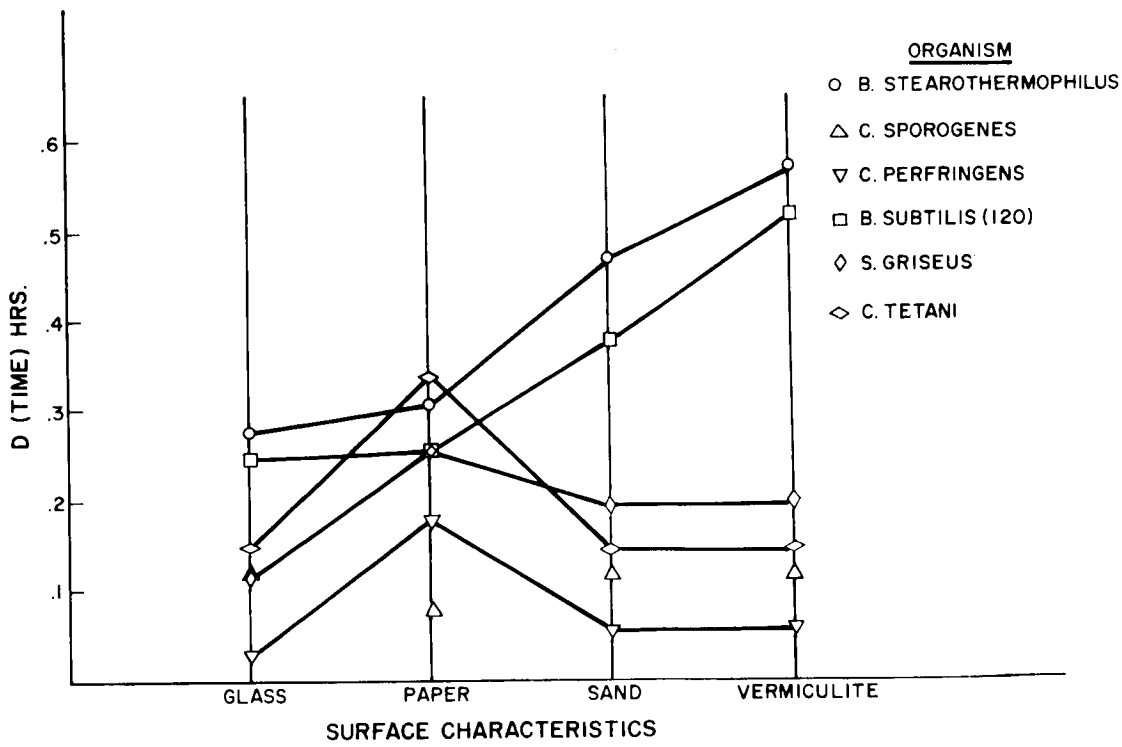


Figure 9.1-4. Variation in D Values for Different Surfaces

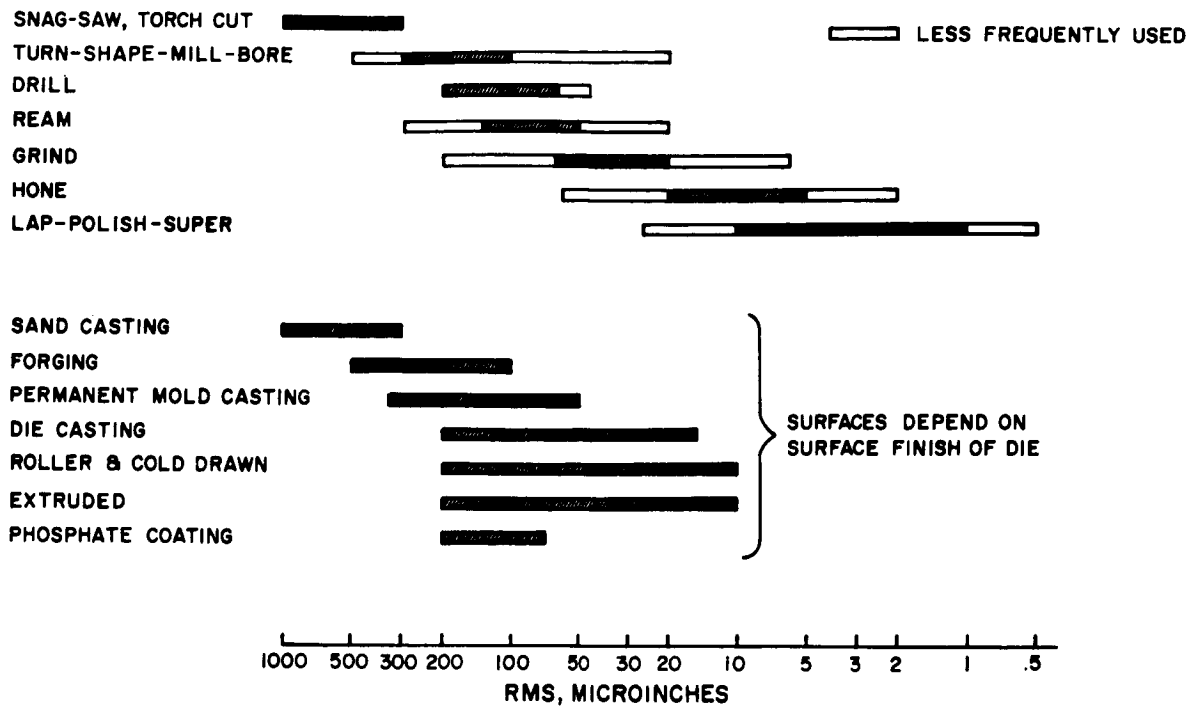


Figure 9.1-5. Surface Finishes by Common Production Methods

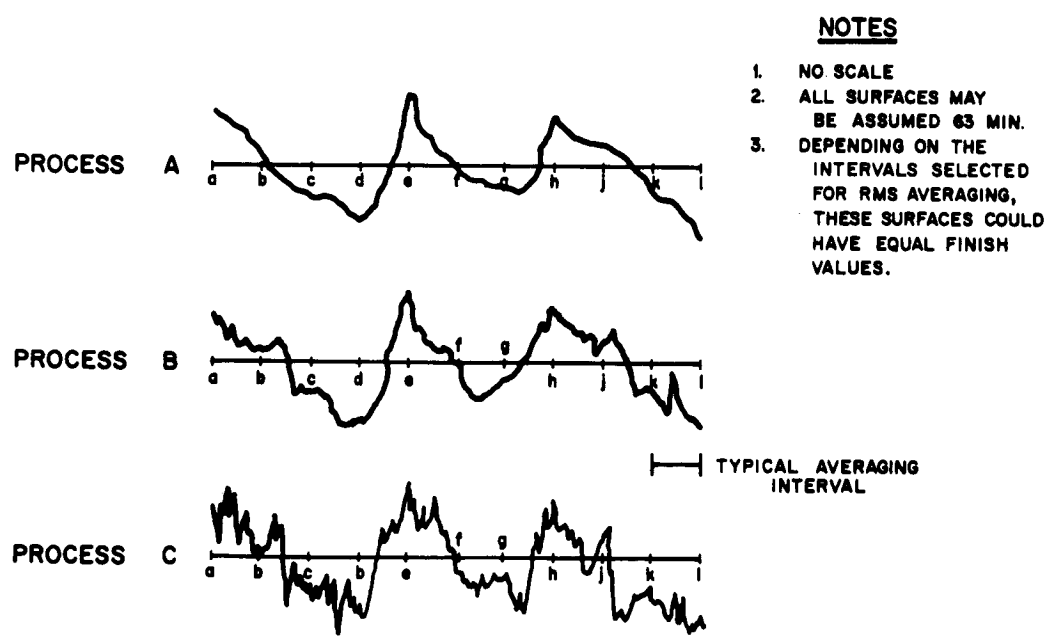


Figure 9.1-6. Possible Variations in Surfaces Having Equal Surface Finish Values

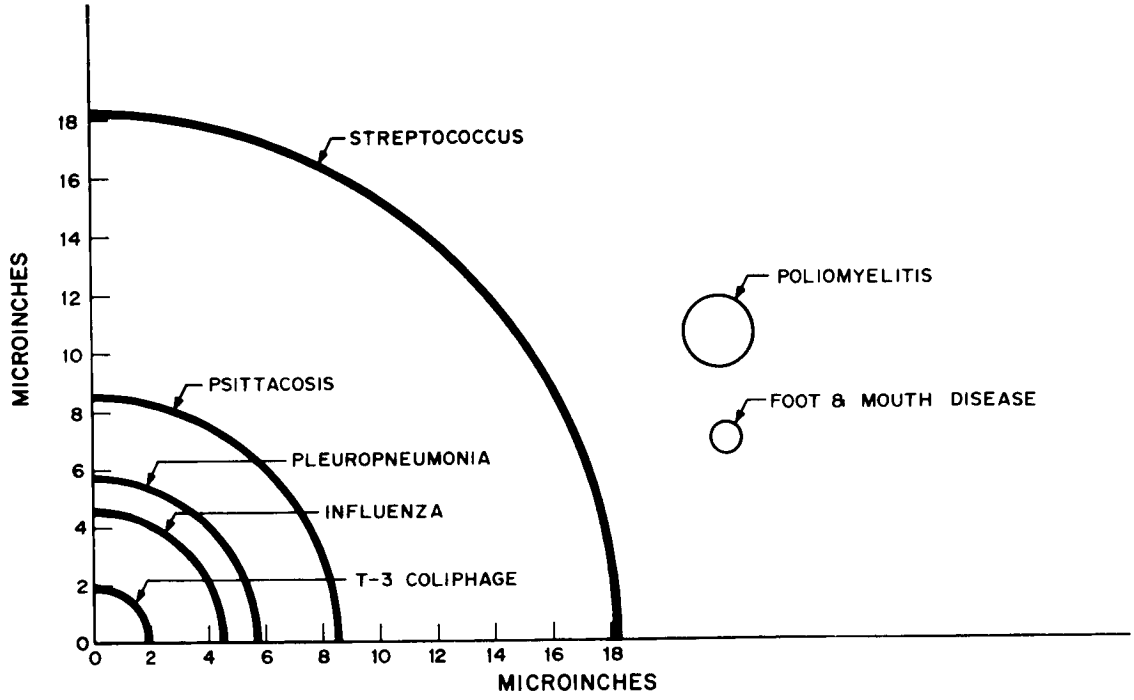


Figure 9.1-7. Relative Size of Several Organisms

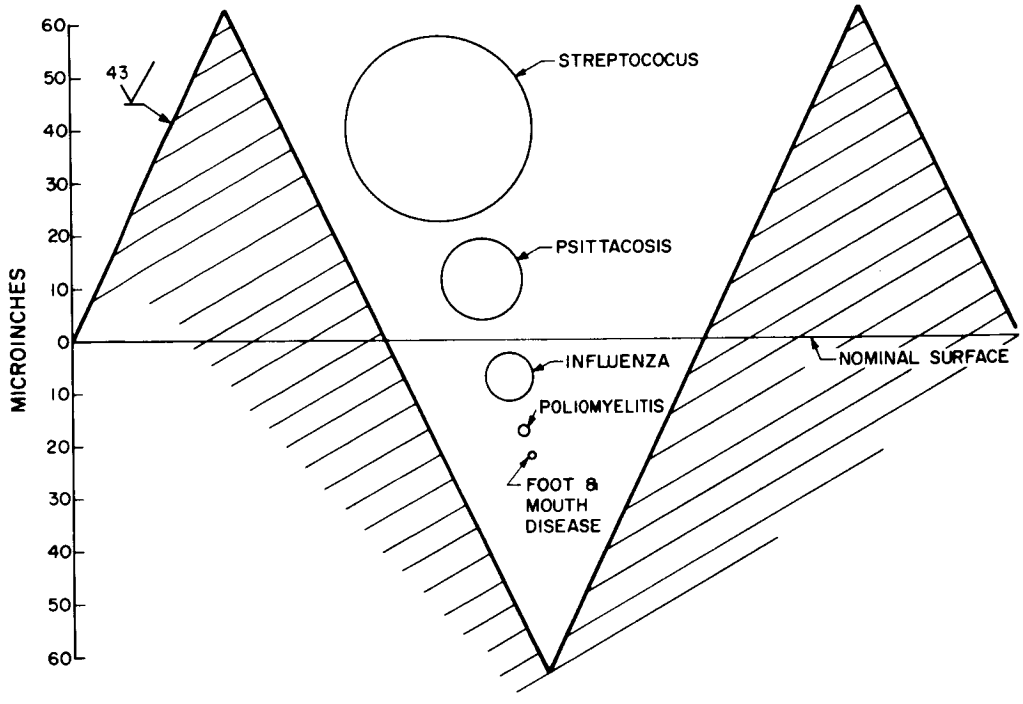


Figure 9.1-8. Relationships Between a Typical Surface and Typical Organism

desired reliability values, it is desirable to verify that D and F values do not experience large changes as a function of surface condition. To establish the relationship between various surfaces and ease of sterilization, certain test programs may be required. A simplified approach to such testing is given in Appendix P.

This discussion has dealt primarily with the effect of surface finish or ease of sterilization. Similar discussions could be made concerning surface coatings, paints and other applied finishes will have unique characteristics and will warrant further investigation.

This discussion has dealt primarily with the effect of surface finish or ease of sterilization. Similar discussions could be made concerning surface coatings, or treatments. Coatings, paints, and other applied finishes will have unique characteristics and will warrant further investigation.

9.2 DESIGN GUIDELINES

The major conclusions of this study have been summarized in the following guidelines for the design of vehicle structues which will experience terminal sterilization. Because of the magnitude of the task and the limitations of the study, no claim to completeness is made. The intent of these guidelines is to alert the designer to a wide range of potential problem areas which may not be apparent at first sight. As more experience is gained in designing structures for terminal sterilization, these guidelines should be revised and refined.

For a typical lander to successfully withstand terminal sterilization without loss of functional reliability, provisions for sterilization must be made early in the design cycle. As an example of the extent to which sterilization requirements have been considered in the design of previous space vehicles, see Section 1.1 and 1.2 of Reference 9.8.

9.2.1 GENERAL

If biological load reduction is used to supplement terminal sterilization, the number of structural elements, members, and fasteners should be kept small. Maximum use should be made of repetitive shapes and members. The fewer the number of parts, the fewer the number of controlled sterilizing procedures which would be required.

A small number of standard parts is preferable to a large number of non-standard parts. The point has been made that detail parts cannot be individually checked for sterility (Reference 9.15). In hospitals, individual scalpels are not checked. Instead certain standard operating procedures of known effectiveness are set up and administrative controls established to see that the procedures are followed. This approach to sterility is also followed in the pharmaceutical and food processing industries. In these cases, however, it is possible to monitor the effectiveness of the sterilization procedures by statistical sampling. Satisfactory results of the sampling are interpreted as indication of success of the procedure.

9.2.2 DESIGN AND ANALYSIS

A. Preliminary Design of Lander

During the preliminary design of the planetary lander (including any component parts which may impact on a planet) all components should be identified according to whether they can or cannot tolerate terminal sterilization by heat.

For the components and elements which cannot tolerate terminal heat sterilization, alternate sterilization procedures should be selected early in the design phase.

This recommendation stems directly from a consideration of structural problems. Consider for example a typical high pressure storage vessel which might be included on a planetary lander. This vessel could contain helium at 3000 psi. The function of the pressure vessel is clearly structural, and it is questionable whether a 60-hour soak at 150°C could be tolerated. For this reason, alternate sterilizing means may be required. Gaseous sterilization might be employed for this component. This technique has been used in the past for highly complex components. Bruch (Reference 9.4) reports the use of gaseous sterilization on a complex heart-lung machine. Regardless of the technique which is finally selected, its selection may depend on structural response to the environment.

B. Entry Shield

The sterilization environment becomes a significant design condition for an entry shield-bond-structure combination. Since typical shielding materials have much higher coefficients of thermal expansion than the common structural metals, the differential expansions and contractions of the shield and structure are significant.

Traditionally, the "hot soak" and "cold soak" environments of the shipping, handling and storage conditions that a re-entry vehicle must survive design the bond thickness required. Additionally, while the minimum shielding thickness is chosen as a function of the many thermal parameters of entry, the cold-soak condition is often limiting on the actual design thickness. The hot-soak condition of storage (usually +160°F) is not usually critical, but under a sterilization temperature of 300°F, this may become the limiting design condition for both the type and thickness of bond.

C. Beam Elements

For depthwise gradients, the sensitivity to a thermal environment will be a function of the percentage of design load that a 1g loading represents.

For spanwise gradients the beam may react primarily as a column, and it should be designed accordingly.

D. Compression Members

For the range of columns likely to occur on a planetary lander, the thermal environment does not appear to constitute a limiting design condition.

The use of titanium offers the advantages of reduced thermal expansion and high allowable stresses.

E. Ring-Stiffened Shells

Discontinuities may occur in the lander shell at reinforcing rings or at bulkhead or panel attachment points. Although the stress levels at these discontinuities may not be high, each condition should be carefully analyzed.

The containing canister may have reinforcing rings. If sterilization is performed in a vacuum, the canister design may be greatly influenced by these discontinuity stresses.

F. Plates and Panels

The thermal gradients required to cause buckling will in general be small. Plates and panels should therefore generally be avoided in mounting alignment-sensitive components.

The thermal stresses in component packages (black boxes) induced by terminal sterilization will, in general, be low and non-limiting.

G. Tanks and Pressure Vessels

During the preliminary design phase it should be decided whether the tanks are required to withstand terminal heat sterilization in a charged condition. This decision should be based on the types and pressures of the liquids and gases to be stored. Personnel safety and weight penalty should be of prime concern.

If terminal heat sterilization is selected as the sterilizing method, consideration must be given to safety, tank materials, material degradation at temperature, insulation, and thermal coatings. Tank supports should be designed to allow for growth at elevated temperatures.

If it is decided to charge the tanks with sterile gas or liquid following terminal heat sterilization of the vehicle, problem areas to be considered include material property degradation after extended 150°C temperature exposure, charge line interface design, and design and control of the sterile gas or liquid charging equipment. The charging process should be added to the vehicle flow chart early in the design stage to allow for facility integration.

H. Explosively Actuated Fasteners

The need for explosively actuated devices on planetary vehicles is sufficiently great to prevent their exclusion because of the terminal-sterilization requirement.

Few property or performance data are available for these devices following exposure to terminal sterilization. Allowance should be made for degradation.

It appears that some devices may have to be added after terminal sterilization. Where this occurs, the order of preference to minimize assembly problems is:

1. Fasteners with removable cartridges or squibs
2. Fasteners with integral explosive cartridges
3. Shaped charges.

I. Alignment-Sensitive Components

During preliminary design the components and equipment items for which alignment requirements are most severe should be determined. The structural elements or members should be chosen which will be used to support these members; analysis should be performed to determine if permanent deflections or set occurs. A determination should be made to see if some permanent set could be permitted.

The preceding comments are perhaps obvious, but it should be emphasized that it is becoming common practice to design for buckling and instability failures in the inelastic range. Gatewood (Reference 9.17) suggests: 1) the use of expansion joints or 2) "Let the inelastic portion of the stress strain curve provide the deflection to absorb the thermal expansion." Either approach may be acceptable for alignment-sensitive components based on theoretical analysis or tests to determine the degree of misalignment caused by the sterilizing environment.

Two alternative solutions which may cost considerable weight would be to 1) design completely within the elastic range, or 2) build into the component or vehicle a means for checking component alignment after sterilization.

J. Fittings

The techniques for minimum weight structural design are fairly well established. No unique requirements for the design or fabrication of structural fittings are recommended as a result of terminal sterilization.

K. Non-Structural Elements

Many non-structural elements including cable clamps, harness bundles, thermal insulation, flexible wave guide and spacers or fillers may deflect or deform and create structural problems. In the design or installation of these elements serious consideration should be given to response during terminal sterilization.

L. Design Release Procedures

The requirements for detailed controls and procedures have been discussed by Jaffee (Reference 9.2) and others. For any particular program the establishment of a central sterility group has been recommended. As a result of this study it is felt that the importance of controls and procedures cannot be overemphasized. Prior to shop release of structural details or assemblies, approval by this group should be indicated on pertinent drawings and specifications.

M. Containing Canister

The storage and handling requirements on the container after sterilization may constitute a critical design condition. Specific requirements for the canister should be established early in the design cycle.

Sterilization in a vacuum creates numerous problems which may include design for a 14.7 psi static external overpressure and for seals to prevent atmospheric contamination. The introduction of a sterile atmosphere at a slight positive gage pressure is recommended over sterilization in vacuum.

Heat transfer relationships between the lander and the canister will require additional analysis.

Maintenance of a positive internal pressure is a design problem, but no obstacles to its solution are anticipated.

Flexible containers or biological barriers create many problems in design, handling, and separation. Their use as a major barrier to contamination is not recommended.

Techniques for heating the canister require additional study.

N. Design Data

For the vehicle which experiences terminal sterilization, a handbook of material properties for design at elevated temperature and following exposure to elevated temperatures should be prepared. Particular emphasis should be placed on non-metallic materials.

A primer or guidelines for different sterilization procedures should be prepared to enable designers and engineers to select alternate sterilization techniques which are compatible with the peculiarities of special components. Selected techniques should be subject to the approval of a central sterility group. Alternate techniques should be selected early during the design phase of the vehicle. This will enable flow plans and procedures to be established early in the life of the program. In this way limiting time items can be detected. In addition to thermal sterilization this primer should include:

1. Ultrasonic sterilization
2. Radiation sterilization
3. Gaseous and chemical sterilization

O. Joint Selection

No joint types or joining techniques can be rejected solely on the basis of incompatibility with the sterilization environment.

P. Thermal Analysis

Thermal analysis by means of an analog model, similar to the analyses performed on this study, should be performed for the specific vehicles which will be terminally sterilized.

Q. Thermal Testing

Thermal testing of a representative configuration should accompany thermal analysis. This testing would verify the analysis and give confidence in its ability to predict behavior.

9.2.3 JOINING TECHNIQUES

A. Types of Joining Techniques

The number of different joining techniques used on a space vehicle should be kept low. Joining techniques should be limited to a number for which the precesses and techniques are well known.

This requirement is not intended to limit the designer to one or two joining techniques. Instead, it would allow him to use whatever techniques the design warrants, but would control the variety of total techniques used. There exists such a variety of joining techniques and the processes vary so widely that it will be difficult for every process to be evaluated with respect to sterility.

1. Riveting

The number of different types of rivets and riveting processes should be limited. Riveted joints should be used only in applications where cleaning procedures, as specified by a central sterility group, can be applied subsequent to the riveting operation. The procedures which would be specified might vary depending upon the type of riveting process that is used.

For rivets driven or pulled by pneumatic tools it is possible that oil or air impingement on the metal surface could be a source of contamination. If riveting is performed in a "clean" shop, it may be necessary to heat or filter the air supply and to use sterile lubricants for the gun.

Explosive rivets would require special attention because of the combustion products, possibly contaminated, which might be expelled. Their use on space vehicles has been negligible in the past. For rivets which are pulled using an electric gun, the resulting contamination would probably be less than for a pneumatic gun.

Speciality fasteners normally grouped with rivets (e.g., rivnuts, torque-loks, etc.) will require evaluation depending on where in the overall fabrication process their use is specified.

Hole preparation should take place in a specified location, using tools and techniques subject to the control and approval of a central sterility group. Shop drills may be pneumatic or electric. With pneumatic drills an air jet impinging on metallic surfaces possibly causes a buildup of microbes. The use of an electric drill may be preferred.

The type of surface finish inside the hole could conceivably be subject to approval by a central sterility group. They might for instance, recommend a procedure similar to the one shown in Figure 9.2-1.

Cleco fasteners, C-clamps, Knuvise clamps and many other tools which normally supplement riveting operations may require special evaluation and controls.

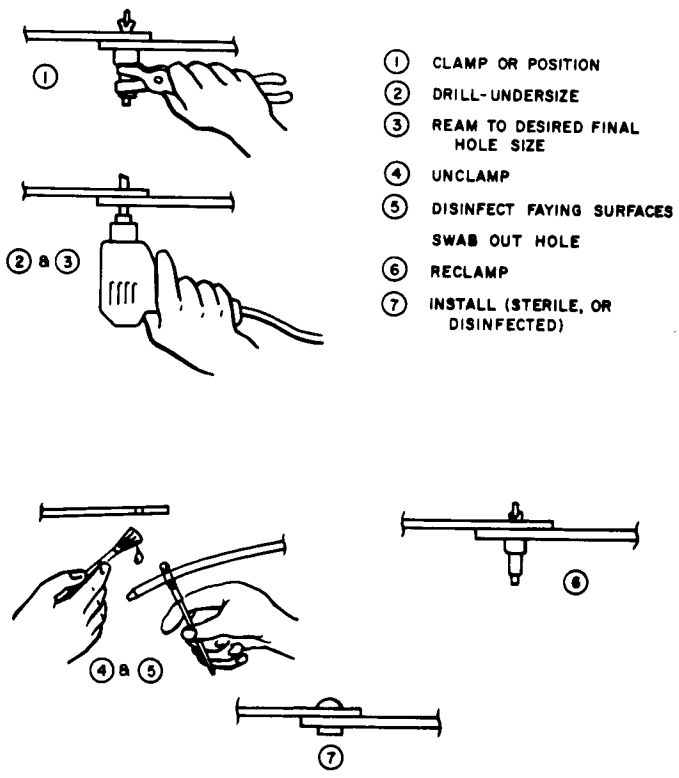


Figure 9.2-1. Riveted Connection — Typical Manufacturing Sequence

2. Bolting

Bolted joints should be used only in applications where the cleaning procedure specified by a central sterility group can be applied before the bolting operation. The number and type of bolts, nuts or specialty fasteners should be limited. An approved materials and fasteners list should also consider limiting thread forms, head types, nut types, and locking features.

Bolts and nuts may be fastened by hand or with power tools. If bolting is performed in a clean room, it might be desirable to exclude power tools and rely on simple hand tools which are easily cleaned. Torque wrenches might offer some advantages in a clean room.

The locking feature used on nuts or bolts is a matter of interest to a sterility group. Obviously, the fewer piece parts, the less chance of a human error. Cotter pins and lock wires or lock washers might be avoided for this reason. Similarly the use of washers should be examined carefully. Self-locking nuts may offer some advantages. The effect of temperature or chemical sterilants on nylock pellets or nylon or plastic looking features in either bolts or nuts will require special evaluation.

Hole preparation for bolted connectors should be subject to the same controls as for riveted joints.

3. Brazing

Brazed joints are permissible when approved by a central sterility group. The brazing process is essentially self sterilizing. For large items which are heated uniformly (viz., sandwich panels) sterilization may occur during fabrication. For smaller parts (viz., brazed joints in tubes or pipes) only some portions of the assembly may be sterilized during manufacture.

In general, since the brazing temperatures will probably be more severe than the sterilizing environment, it is safe to assume that brazed parts could be subjected to the sterilizing environment without harm.

If a brazed sandwich panel fabricated in a retort is considered, it is reasonable to assume it could be taken to a clean-shop floor and removed from its retort or container. Then in a clean room, holes for attachments could be drilled and reamed. These holes could be disinfected and the panel moved to a clean room where additional sterilization could, if desired, be performed. The completely assembled spacecraft will, of course, be subjected to terminal sterilization but the "biological load" on the brazed components would have been considerably reduced.

4. Welding

Welding processes should be approved by a sterility control group. Welding jigs and fixtures should be cleaned by procedures to be approved by the sterility control group. Closed shapes (e.g., tubes) should be provided with holes or parts to facilitate entrance of cleaning solution if cleaning or flushing after sterilization is prescribed. Individual piece parts should be sterilized in a prescribed chemical sterilant or by other techniques as specified by a sterility control group.

The use of welded structure has received wide attention as a good means for sterilizing spacecraft. Welding was mentioned by Phillips and Hoffman (Reference 9.7) as a good means of fabricating structure for space vehicles. Furthermore, the use of welding on space vehicles is expected to increase in the next decade. A recent state of the art (Reference 9.18) survey indicates that in the next decade welded joints will increase from 30 to 60 percent of all joints. The case for welding seems to involve two factors:

a. Weld temperatures will provide local sterilization.
b. Welded configurations in general are "cleaner" and provide fewer traps or pockets for contamination.

Although weld temperatures may provide local sterilization, little is gained. Even assuming that half of all organisms present are killed, the completed weldment still requires sterilization. And, although welded structures may in general be slightly cleaner, it does not necessarily follow that fewer pockets result. In the case of spotwelded skins, for example, the total area not exposed for flushing is not much different than for the same riveted connection.

In general, welded structures are more rigid than riveted or bonded counter-parts. Previous vehicles such as Mariner 2 and Surveyor have used rivets and bolts. The use of riveted structure improves damping characteristics (Reference 9.19).

5. Bonding

Bonding processes, processing specifications and adhesive systems should be approved by a sterility control group. The number of adhesives, cleaning agents and surface treatments used on any particular space vehicle should be kept small.

9.2.4 STRUCTURAL TOOLING, JIGS AND FIXTURES

Tooling, jigs and fixtures used to hold flight hardware at any stage in the fabrication process should be free from loose dirt, grease, oil or other contaminants. The care taken to clean these items depends on their relative position in the overall assembly process.

For tools which are used early in fabrication, a relatively simple process such as scrubbing with detergent, steam cleaning, wiping with disinfectant solution or some combination of these three might be sufficient. For fixtures which are used late in the fabrication process, more elaborate precautions for biological load reduction may be required.

9.2.5 SUPPORT EQUIPMENT

Support equipment of a static nature such as benches, work stands, etc., can be used in clean rooms with few problems since they are in general easy to clean and to keep clean. Moving equipment, however, presents more problems since it implies tires (pneumatic, semipneumatic) brakes, brake fluids, lubricants and lubricant fittings. Any of these items could be of concern in a clean room facility. All AGE items which interface with flight structure hardware should be subjected to cleanliness procedures as specified by a central sterility group prior to mating with flight hardware.

9.2.6 APPROVED MATERIALS LIST

Materials selected for use on a spacecraft structure which is subjected to the sterilizing environment should be selected from an Approved Materials List which is subject to approval of a sterility control group. This materials list should include all structural materials, metals and non-metals. Deviations from an approved list should be subject to approval by a sterility control group.

The use of approved materials lists is common practice on many space vehicle programs. In general, these lists are not welcomed by designers or design engineers who would prefer to select the material based on the specific application. But, in general, some restraints on materials tends to simplify the designers job by limiting the number of materials he must evaluate for an application. By keeping the total number of materials relatively small, several benefits are derived:

a. The total volume of information on stress-strain-time relationships for materials is reduced.

b. The number of materials to be evaluated for a specific application is reduced.

In Reference 9.17, Gatewood has tabulated the physical constants and the allowable stresses which are required for thermo-structural analysis. Although this reference deals primarily with aircraft structures, the techniques and materials also apply to planetary landers since many of the properties vary with temperature. These tabulations are shown in Tables 9.2-1 and 9.2-2 as a general indication of the materials problem.

TABLE 9.2-1. PHYSICAL CONSTANTS NEEDED IN AIRCRAFT STRUCTURES (REFERENCE 9.17)

1. ρ, density
2. C_p, specific heat, Btu/(Lb)(^oF)
3. α, coefficient of linear thermal expansion in/in (^oF)
4. K, thermal conductivity, Btu/(hr)(ft)(^oF)
5. σ, radiation constant, Btu/(hr)(ft^2)($^oF^4$) .1748 (10^{-8}) for black bodies
6. ϵ, emissivity coefficient
7. h, convective heat transfer coefficient, Btu/(hr) (ft^2) (^oF)

TABLE 9.2-2. ALLOWABLE STRESSES NEEDED IN AIRCRAFT STRUCTURES (REFERENCE 9.17)

1. Stress-strain curves; Tension compression, shear
2. Tangent modulus curves; tension compression, shear
3. Secant modulus curves; tension compression, shear
4. Stress-time-temperature-strain data; tension compression, shear
5. E and G, tension, compression, shear.
6. ν, Poisson's ratio; ϵ, percent elongation
7. F_u, ultimate stress, tension compression shear bearing bending
8. Fy, yield stress, tension, compression, shear bending
9. F_{cr}, compression buckling, plates, instable sections, stiffened plates
10. F_{cc}, compression crippling, plates, instable sections, stiffened plates
11. F_c, columm stable sections, instable sections, stiffened plates.
12. F_s, shear modulus, torsion
13. F_{scr}, shear buckling, instable panels
14. F_{sw}, shear ultimate, diagonal tension
15. F_{bcr}, bending buckling, instable sections
16. F_{bcc}, bending crippling, instable sections
17. F_m, mean stress for repeated stress
18. R, combined stress factor

9.3 REFERENCES

9.1 J. J. Perkins, "Principles and Methods of Sterilization," Charles C. Thomas, Publisher - Springfield, Ill., 1960.

9.2 L. D. Jaffee, "Sterilization of Unmanned Planetary and Lunar Space Vehicles - An Engineering Examination," JPL-NASA TN 32-325 (Rev.), March 25, 1963.

9.3 R. S. Lloyd, and J. B. Krahe, "Gaseous Sterilization with Ethylene Oxide," (a supplement to Chapter XXI "Principles and Methods of Sterilization" by J. J. Perkins) American Sterilizer Company, Erie, Pa., 1961.

9.4 C. W. Bruch, "Gaseous Sterilization," Annual Review of Microbiology, Volume 15, 1961.

9.5 "How Ranger Payload is Sterilized," Missiles and Rockets, p. 27, March 20, 1961.

9.6 A. M. Nowitzsky, "Spacecraft Sterilization," Parts I and II, Missiles and Space, August and September 1962.

9.7 C. R. Phillips, and R. K. Hoffman, "Sterilization of Interplanetary Vehicles," Science, Vol. 132, No. 3433 pp. 991-995, 1960.

9.8 "Mariner C Design Characteristics and Restraints, Design Restraints" Jet Propulsion Laboratory Specification Number MC-3-120, December 8, 1962.

9.9 I. J. Pflug, "Thermal Resistance of Microorganisms to Dry Heat," Food Technol. 14, 483-487 October 1960.

9.10 C. R. Stumbo, "A Technique for Studying Resistance of Bacterial Spores to Temperatures in the Higher Ranges." Food Technol. 2, 228-240, 1948.

9.11 C. F. Schmidt, "Thermal Resistance of Microorganisms," Reddish (ed) Antiseptics, Disinfectants, Fungicides, and Chemical and Physical Sterilization, Lea and Lebager, Philadelphia, Pa. p. 831-834.

9.12 D. W. Thompson, "On Growth and Form" Cambridge University Press, Cambridge, 1961.

9.13 M. G. Koesterer, "Final Report, Sterilization of Space Probe Components" Contract NAS-F-31, 1962.

9.14 L. D. Jaffee, "Sterilizing Unmanned Spacecraft," Astronautics, August 1963.

9.15 "Proceedings of Meeting on Problems and Techniques Associated with the Decontamination and Sterilization of Spacecraft," NASA TN D-771, June 1960.

9.16 V. Oyama and E. Wilson, "Application of Gas Chromatography to the Analyses of Organics, Water and Absorbed Gases in the Lunar Crust," American Rocket Society Journal, March 1962.

9.17 B. E. Gatewood, "Thermal Stresses," McGraw-Hill Book Company, Inc., New York, 1957.

9.18 "Research and Development Technical Handbook," Space/Aeronautics, Vol. 40 No. 2, July 1963.

9.19 M. Tawil, "Structural Damping Improved by Riveting," Space/Aeronautics, November 1962.

APPENDICES

APPENDIX A

RECOMMENDATIONS FOR FUTURE WORK

APPENDIX A

RECOMMENDATIONS FOR FUTURE WORK

A.1 ANALYTICAL WORK

This study has indicated structural problems in several areas. In many of these areas, additional work is required. Further study of thermostructural stresses and deflections is recommended in these areas.

A.1.1 MULTI-LAYERED SHELLS

Thermal sterilization of multi-layered shells, formed by the heat shield, bond, crush-up structure, and landing capsule structure proper, is a critical design condition. Stresses have been evaluated for a single, typical lander configuration. Because of the extreme importance of this condition which affects the heat shield-structural integrity, the stresses in the structure, heat shield and bond should be evaluated in terms of suitable parameters, such as the relative rigidities of the shells, allowable strains, thermal expansivities, and cone angle.

A.1.2 RING-STIFFENED SHELLS

Wherever beams or bulkheads are attached to conical shells, heavy rings will be used to transfer the loads in lieu of shell reinforcement. The combination of thin skins and relatively massive reinforcing members causes thermostructural incompatibility problems under thermal sterilization which should be examined.

A.1.3 SPACE FRAMES

Most landing vehicles will contain rocket engines for midcourse correction and/or terminal guidance. These engines can only be located at the rear face of the lander where they must be supported either by heavily reinforced flat diaphragms or by cone-shaped space frames. The second method is generally more efficient. Thermal buckling of the frame members during sterilization heat soak should be investigated with particular emphasis on realistic end supports.

A.1.4 TANKS AND PRESSURE VESSELS

It was shown in this study that spherical pressure tanks may suffer a weight penalty as high as 80% due to thermal sterilization. Because of this unexpectedly serious effect, other commonly used tank configurations, such as cylinders with semi-ellipsoidal ends, should be investigated. The effect of different mounting systems should be included. The question of whether or not solid propellant rockets can be terminally sterilized should be answered.

A.1.5 NON-STRUCTURAL ITEMS AFFECTING STRUCTURAL INTEGRITY

A number of items which, although not structural by themselves, may adversely affect the structural integrity of the vehicle during thermal sterilization. Some of these are: explosive fasteners, harnessing and cabling including clamps, insulation blankets, wave guides, pneumatic and fluid plumbing lines. Their behavior under the sterilization environment should be investigated.

A.1.6 Data on these topics should be collected and presented in handbook form for use by structural designers. During this study, basic guidelines have been established which indicate general directions for desirable or undesirable designs. However, they are not sufficient in themselves to be used as design handbooks. The preparation of a handbook is recommended which would be more specific than the present guidelines. This handbook would include design charts, graphs, and tables from which a designer could determine with very little effort the effect of sterilization on his proposed design.

A.2 TESTING

Where details of a specific lander configuration are known, testing is recommended. Test work is specifically recommended in the following areas:

A.2.1 DISTORTION TESTS

Changes in geometry due to thermal sterilization of typical fabricated elements should be investigated. Foremost among these are ring-stiffened and longeron-stiffened conical shells, possibly with one spherical and one open end.

A.2.2 ALIGNMENT SENSITIVE COMPONENTS

The effect of terminal sterilization on sensors, attitude control nozzles, and scientific experiment packages on a typical vehicle should be investigated. Emphasis should be placed on realistic joints and attachments.

A.2.3 THERMAL PERFORMANCE OF SPACECRAFT JOINTS

The thermal conductance of real spacecraft joints is an area of concern when such joints represent major heat flow paths. An experimental program to investigate such joints to determine or derive performance parameters is needed. Such a program would be similar to studies carried out in the late 1950's on aircraft joints by Barzelay.

A.2.4 LANDER MODEL

A large amount of work of the present study was predicated on the thermostructural analysis of a typical planetary lander. Both thermal and structural analyses should be verified by performing a test on a simplified sub-scale or full scale model of a lander.

A.3 CANISTER DESIGN

The concept of sterilizing a landing vehicle in a canister is well established. This canister is a flight item and must be designed for handling, shipping, and flight loads and for in-flight separation from the lander. Pressure loads due to heating of a sealed canister should be traded off against plumbing and piping penalties incurred by letting the sterilizing gas escape during heating and pumping it back during cooling. The structural interface between the lander and the containing canister will represent a major conduction path during terminal sterilization. The effect of local hot spots should be evaluated.

A.4 STERILIZATION TECHNIQUES

A primer for spacecraft sterilization is required. This primer should include biological information on sterilization techniques and practices including instructions and procedures for cleaning and biological load reduction. This primer should include:

A.4.1 DESIGN FOR BIOLOGICAL LOAD REDUCTION

Thermal sterilization cannot be effective within reasonable time limits unless "biological load reduction" has previously taken place. This implies numerous cleaning and sterilizing steps in the manufacturing process. The design of the vehicle should allow for them. Weak links in the sterilization chain, such as seals and gaskets, should be investigated.

A.4.2 EFFECT OF SURFACE FINISH

A typical test program to evaluate the effect of surface finish on ease of sterilization is described in Appendix P. This type of testing should be performed. The effect of surface coatings, paints, treatments, and porosity on the ease of sterilization should also be evaluated.

A.4.3 EFFECT OF ULTRASONIC CLEANING

The use of ultrasonic cleaning techniques will reduce biological load levels. Few or no data exist to indicate the degree to which these levels are reduced by the techniques and procedures which would be employed on structural elements. Simple tests are proposed to determine the effectiveness of cleaning by ultrasonic techniques.

A.5 MATERIALS TESTING

In the materials area the sterilization requirement will necessitate additional test work. Further study is recommended in these areas:

A.5.1 DETERMINATION OF REQUIRED TESTS

Using sterilization cycles of 60 hours each plus transient times, a total time of 300 hours at 300°F is a reasonably safe assumption. Of all the steel alloys likely to be used in space vehicles, only those tempered in the 400-600°F range (AISI 4130, 4340, etc.) are likely to be affected by that condition. Sufficient data are probably available for aluminum alloys. It appears that few test data on magnesium and titanium alloys at 300°F for 300 hours are available, but a thorough search should be conducted before tests are perfomed.

A.5.2 300-HOUR SOAK DATA FOR ALLOYS

For the alloys previously mentioned, testing is required to provide values at temperature and at room temperature following exposure. Typical properties of interest are modulus of elasticity, yield and ultimate strengths, and thermal expansion.

A.5.3 ORGANIC MATERIALS

Tests should be performed on selected organic materials to obtain properties at elevated temperature and following exposure to elevated temperature. Outstanding among these are structural adhesives.

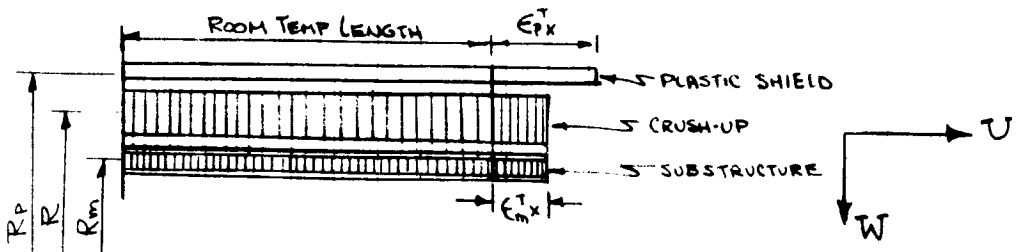
APPENDIX B

ENTRY SHIELD ANALYSIS

HEAT SHIELD ANALYSIS - VENUS MISSION

THE MECHANICS OF THERMAL EXPANSIONS AND CORRESPONDING ELASTIC STRAINS FOR A SEGMENT OF THE VEHICLE ARE SHOWN BELOW.

THERMAL STRAIN



$$\epsilon_{mx}^T = \alpha_m \Delta T$$

$$\epsilon_{m\theta}^T = \alpha_m \Delta T$$

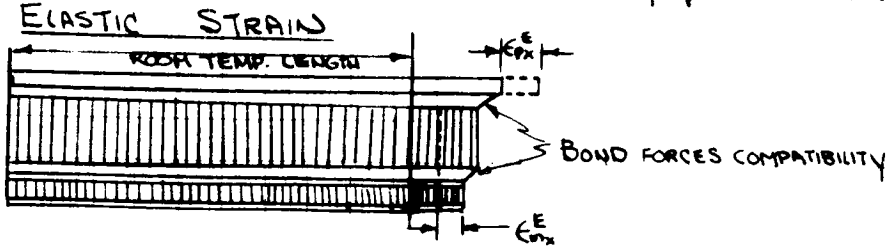
$$\epsilon_{px}^T = \alpha_{px} \Delta T$$

$$\epsilon_{p\theta}^T = \alpha_{p\theta} \Delta T$$

SYMBOLS

ϵ = STRAIN - IN/IN
m = METAL SUBSTRUCTURE
p = PLASTIC SHIELD
x = MERIDIONAL DIRECTION
θ = HOOP DIRECTION
ΔT = CHANGE IN TEMPERATURE
α_m = METAL COEFFICIENT OF EXPANSION
α_p = PLASTIC COEFFICIENT OF EXPANSION
E_m, E_p = MODULUS OF ELASTICITY
σ_m, σ_p = STRESS
ν_m, ν_p = POISSONS RATIO

ELASTIC STRAIN



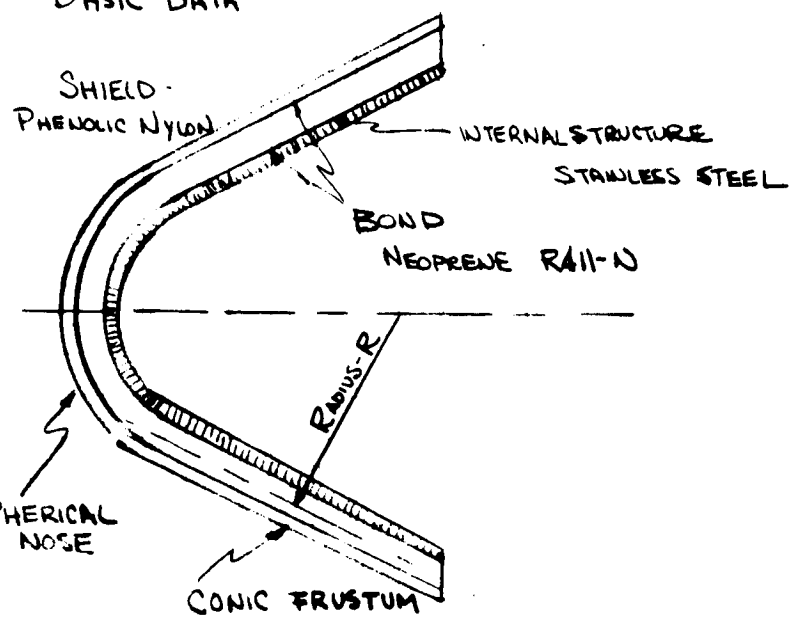
$$\epsilon_{mx}^E = \frac{\sigma_{mx}}{E_m} - \nu_m \frac{\sigma_{m\theta}}{E_m} \qquad \epsilon_{px}^E = \frac{\sigma_{px}}{E_{px}} - \frac{\nu_p \sigma_{p\theta}}{E_{p\theta}}$$

$$\epsilon_{m\theta}^E = \frac{\sigma_{m\theta}}{E_m} - \nu_m \frac{\sigma_{mx}}{E_m} \qquad \epsilon_{p\theta}^E = \frac{\sigma_{p\theta}}{E_{p\theta}} - \nu_p \frac{\sigma_{px}}{E_{px}}$$

BY Y CK. DATE REV.	GENERAL ELECTRIC	PAGE 1 MODEL REPORT

HEAT SHIELD ANALYSIS - VENUS MISSION

BASIC DATA



MATERIAL PROPERTIES

	STAINLESS STEEL	PHENOLIC NYLON	NEOPRENE R-411-N
FTU - PSI	190,000	5000 PSI	20 psi (250°-300°F)
FTY - PSI	170,000	—	—
FCY - PSI	185,000	—	—
FSU - PSI	120,000	—	15 PSI - HOT T. 106 PSI - COLD T.
E - PSI	$30(10)^6$ to $28(10)^6$	$.4(10)^6$ PSI	30 psi
G - PSI	$11.7(10)^6$	—	12 psi
α - IN/IN/°F	$6.5(10)^{-6}$ (70°-600°F) $5.5(10)^{-6}$ (70°-200°F)	$45(10)^{-6}$ (300°F) $25(10)^{-6}$ (-250°F)	—
ν - POISSONS RATIO	.3	.4	—

FORM 1-8136 B (9-58)

HEAT SHIELD ANALYSIS – VENUS MISSION

TOTAL STRAIN (ELASTIC AND THERMAL)

$$\epsilon_{mx} = \epsilon^E_{mx} + \epsilon^T_{mx} = \frac{dU_m}{dx}$$ (U = TOTAL ELONGATION - LONGITUDINAL)

$$\epsilon_{m\theta} = \epsilon^E_{m\theta} + \epsilon^T_{m\theta} = -\frac{W_m}{R_m}$$ (W = TOTAL ELONGATION - RADIAL)

$$\epsilon_{px} = \epsilon^E_{px} + \epsilon^T_{p\theta} = \frac{dU_p}{dx}$$

$$\epsilon_{p\theta} = \epsilon^E_{p\theta} + \epsilon^T_{p\theta} = -\frac{W_p}{R_p}$$

ALSO

$$\epsilon_B = \frac{W_p - W_m}{t_B} = \frac{\sigma_B}{E_B}$$

WHERE σ_B = BOND TENSILE STRESS
$W_p - W_m$ = RELATIVE DEFLECTION OF THE TWO SHELLS IN THE RADIAL DIRECTION
ϵ_B = TENSILE STRAIN OF BOND

$$\epsilon_{xy} = \frac{U_p - U_m}{t_B} = \frac{\tau_B}{G_B}$$

WHERE τ_B = BOND SHEAR STRESS
$U_p - U_m$ = RELATIVE DEFLECTION OF THE TWO SHELLS IN THE LONGITUDINAL DIRECTION
ϵ_{xy} = SHEARING STRAIN OF BOND

COMPATIBILITY EQUATIONS

$$\sigma_{px} t_p + \sigma_{mx} t_{mx} = 0$$

$$\sigma_{p\theta} t_p + \sigma_{m\theta} t_{m\theta} = 0$$

$$t_p \sigma_{p\theta} = -R\sigma_B$$

$$t_m \sigma_{m\theta} = R\sigma_B$$

$$t_p \sigma_{px} = -\int_x^L \tau\, dx$$

$$t_m \sigma_{mx} = \int_x^L \tau\, dx$$

HEAT SHIELD

COMBINING THE ABOVE EQUATIONS, EXPRESSIONS FOR SHIELD, SUBSTRUCTURE AND BOND CAN BE OBTAINED. IT IS REMINDED THAT THE MATHEMATICAL MANIPULATIONS ARE QUITE LENGTHY. AS CAN BE SEEN, AN IMPORTANT PARAMETER FOR AN ACCURATE NUMERICAL SOLUTION IS THE SHEAR STRESS DISTRIBUTION, $\int \tau dx$, IN THE BOND. THIS WILL VARY WITH THE TEMPERATURE SINCE E_B AND G_B ARE FUNCTIONS OF TEMPERATURE. THIS ANALYSIS WILL CONSIDER AN APPROXIMATE SHEAR STRESS DISTRIBUTION INDEPENDENT OF TEMP. FOR THIS EXAMPLE, THE BOND STRESSES AT THE HIGH TEMPERATURE ARE THE MAIN INTEREST. IN ORDER TO CHOOSE A REPRESENTATIVE BOND THICKNESS ALONG THE MERIDIONAL LENGTH, THE SHIELD STRESS ALLOWABLE AT THE COLD SOAK ENVIRONMENT IS USUALLY USED AS THE LIMITING DESIGN PARAMETER.

THE APPROXIMATE EXPRESSIONS FOR SHIELD AND BOND STRESSES ARE:

$$\sigma_p = -\frac{(\alpha_p - \alpha_m)\Delta T}{\frac{1-\nu_p}{E_p} + \frac{t_p(1-\nu_m)}{t_m E_m} + \frac{t_B t_p}{E_B R^2}}$$

$$\sigma_B = -\frac{\sigma_p t_p}{R}$$

$\tau_{B\,(max)}$ - WHICH OCCURS AT THE END OF THE COMPOSITE SHELL

$$= (\alpha_p - \alpha_m)\Delta T \sqrt{\frac{\sigma_B\left(1 + \frac{\nu_p + \nu_m}{2}\right)}{t_B\left(1 - \frac{\nu_p + \nu_m}{2}\right)\left(\frac{1}{t_p E_p} + \frac{1}{t_m E_m}\right)}}$$

HEAT SHIELD ANALYSIS

REQUIRED BOND THICKNESS VIA COLD TEMPERATURE ANALYSIS

$\Delta T = 70^\circ (\text{R.T.}) - (-100^\circ (\text{GLASS TRANSITION})) = -170^\circ F$

α_p (COLD TEMP) $= 25\,(10)^{-6}$ IN/IN/°F

α_m " $= 5.5\,(10)^{-6}$ IN/IN/°F

t_p (FRUSTRUM) = .42 IN. t_m (2 FACES) = .012 IN.

$$\sigma_p = -\frac{(\alpha_p - \alpha_m)\Delta T}{\frac{1-\nu_p}{E_p} + \frac{t_p(1-\nu_m)}{t_m E_m} + \frac{t_B t_p}{E_B R^2}}$$

AT THE COLD SOAK TEMPERATURE OF -250°F, THE BOND PASSES ITS GLASS TRANSITION STAGE AND THE BOND BECOMES RIGID. IT CAN BE SEEN THAT THE LAST TERM IN THE DENOMINATOR APPROACHES ZERO AND THE BOND HAS NO EFFECT ON THE SHIELD STRESSES. THE DESIGN APPROACH IS TO CHOOSE A BOND THICKNESS SUCH THAT THE SHIELD STRESSES AT THE GLASS TRANSITION STAGE ARE LOW ENOUGH SO THAT THE ADDITION OF THE STRESSES FROM THE SUCCEEDING CONTRACTION, WHERE THE BOND COMPRESSIBILITY IS NO LONGER AVAILABLE FOR ALLEVIATING STRESSES, DOES NOT FAIL THE SHIELD.

ASSUME A GLASS TRANSITION POINT OF -100°F

$\Delta T = -250 + 100 = -150^\circ F$

$$\sigma_p = -\frac{(25-5.5)(-150)(10)^{-6}}{\frac{1-.4}{.4(10)^6} + \frac{.42(1-.3)}{.012(30)(10)^6}} = 1260 \text{ psi}$$

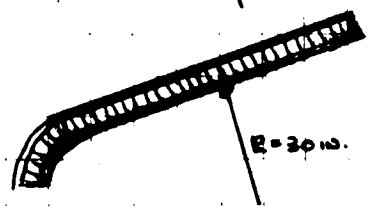
HEAT SHIELD ANALYSIS.

BOND THICKNESS IS NOW CHOSEN SO THAT

$$\sigma_{P\,(70^\circ \text{ TO } -100^\circ)} + \sigma_{P\,(-100^\circ \text{ TO } -250^\circ)} \leq 5000 \text{ psi (TENSILE ALLOWABLE)}$$

$$\sigma_{P\,(-100^\circ \text{ TO } -250^\circ)} \leq 5000 - 1260 = 3840 \text{ psi}$$

IT IS OBVIOUS THAT IN THIS PARTICULAR CASE, THE SHIELD THICKNESS IS SUFFICIENT SUCH THAT THE BOND THICKNESS IS NOT LIMITING IN THE DESIGN AT THE COLD TEMPERATURE. THEREFORE, THE BOND THICKNESS WILL BE DESIGNED FOR ADEQUATE SHEAR AND TENSILE STRENGTH ON THE HIGH TEMPERATURE CYCLE. SINCE WITHOUT THE STERILIZATION REQUIREMENT THE MAXIMUM DESIGN TEMPERATURE WOULD BE +250°F, THE DIFFERENCE IN BOND THICKNESS REQUIRED FOR +250°F AND +300°F IS THE CHANGE BROUGHT ABOUT BY THE STERILIZATION PROCESS.



THE ANALYSIS IS PERFORMED FOR A RADIUS NEAR THE CENTER OF THE FRUSTRUM SINCE THE APPROXIMATE EQUATIONS ARE ONLY VALID FOR R/t IN THIS RANGE.

CHOOSING A NOMINAL SHELL RADIUS OF 30 INCHES, THE BOND THICKNESS WILL FIRST BE DETERMINED FOR ADEQUATE TENSILE STRENGTH AND THEN CHECKED FOR SHEAR STRENGTH. COMBINING THE EQUATIONS FOR σ_B AND σ_P AND SOLVING FOR t_B IN TERMS OF σ_B GIVES:

$$t_B = \frac{E_B R}{\sigma_B}\left[\left(\alpha_P - \alpha_m\right]\Delta T\right) - \frac{\sigma_B R}{t_P}\left[\frac{1-\nu_P}{E_P} + \frac{t_P\left[1-\nu_m\right]}{t_m E_m}\right]$$

HEAT SHIELD ANALYSIS

CONDITION ① HOT SOAK AT 250°F

$E_B = 30$ psi
$F_{TUB} = 20$ psi
$\Delta T = 250 - 70 = 180$

$$t_B = \frac{30(30)}{20}\left[\left\{(45.0 - 6.5)(+180^\circ)(10)^{-6}\right\} - \frac{20(30)}{.42}\left(\frac{1-.4}{4(10)^6} + \frac{.42(1-.3)}{.012(28)(10)^6}\right)\right]$$

$$= .159 \text{ IN}$$

CONDITION ② HOT SOAK AT 300°F

$\Delta T = 300 - 70 = 230$

$$t_B = \frac{30(30)}{20}\left[\left\{(45.0 - 6.5)(230)(10)^{-6}\right\} - \frac{20(30)}{42}\left(\frac{1-.4}{4(10)^6} + \frac{.42(1-.3)}{.012(28)(10)^6}\right)\right]$$

$$= .246 \text{ IN.}$$

THIS INDICATES A Δ BOND THICKNESS OF .087 IN AS REQUIRED BY STERILIZATION.

HEAT SHIELD ANALYSIS

CHECK SHEAR STRESS $G_B = 12$ psi

$$\tau_B = (\alpha_p - \alpha_m)\Delta T \sqrt{\frac{G_B\left(1+\frac{\nu_p+\nu_m}{2}\right)}{t_B\left(1-\frac{\nu_p+\nu_m}{2}\right)\left(\frac{1}{t_p E_p}+\frac{1}{t_m E_m}\right)}}$$

CONDITION ① $\Delta T = 180°F$

$$\tau_B = (45-6.5)(180)(10)^{-6}\sqrt{\frac{12\left(1+\frac{.4+.3}{2}\right)}{.139\left(1-\frac{.4+.3}{2}\right)\left(\frac{1}{.42(.4)(10)^6}+\frac{1}{(.012)(.3)(10)^6}\right)}}$$

$\tau_B = 9.21$ psi

$$M.S. = \frac{15}{9.21} - 1 = \underline{+.62}$$

CONDITION ② $\Delta T = 230°F$

$$\tau_B = (45-6.5)(230)(10)^{-6}\sqrt{\frac{12\left(1+\frac{.35}{2}\right)}{.246\left(1-\frac{.35}{2}\right)\left(\frac{1}{.420(.4)(10)^6}+\frac{1}{(.012)(.3)(10)^6}\right)}}$$

$= 10.65$ psi

$$M.S. = \frac{15}{10.65} - 1 = \underline{+.41}$$

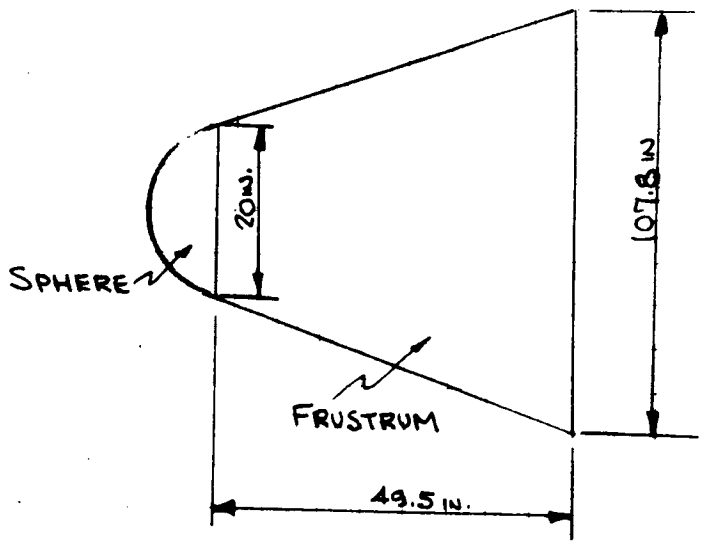
HEAT SHIELD ANALYSIS.

Δ SHIELD STRUCTURE WEIGHT

γ - NEOPRENE 411-N BOND = .046 #/IN^3

Δt = .246 - .159 = .087 IN.

SURFACE AREA - TYPICAL LANDER



SPHERICAL NOSE CAP

ASSUME $R_{AVE} = \frac{10+6}{2} = 8$ IN. $A = 2\pi r^2 = 2\pi(64) = 402$ IN^2

FRUSTUM.

$$S = \pi(R_1+R_2)\sqrt{L^2+(R_2-R_1)^2}$$

$$= \pi(10.0+53.9)\left[(49.5)^2+(53.9-10.0)^2\right]^{\frac{1}{2}} = 13{,}200 \text{ IN}^2$$

Δ WEIGHT = (.087)(.046)(13602) = 54.4 lbs.

HEAT SHIELD ANALYSIS

CONCLUSIONS

THE PREVIOUS ANALYSIS SHOWING A WEIGHT PENALTY OF 54.4 lbs IS MEANT TO SHOW A BASIC TREND ONLY. THE ACTUAL WEIGHT PENALTY INCURRED BY THE STERILIZATION WOULD OF COURSE BE MINIMIZED BY A BETTER CHOICE OF BOND MATERIAL. IT SHOULD ALSO BE POINTED OUT THAT THE LANDER CHOSEN DOES HAVE A THICK ENOUGH SHIELD SO THAT THE COLD TEMPERATURE ENVIRONMENT WAS NOT CRITICAL. IN MANY CASES IT COULD BE ASSUMED THAT THE BOND WILL BE DESIGNED FOR THE COLD TEMPERATE ENVIRONMENT AND BE OF SUFFICIENT THICKNESS SUCH THAT THE STRESSES IMPOSED ON THE BOND DURING STERILIZATION ARE NOT EXCESSIVE.

IN SUMMATION, THE ANALYSIS SHOWS THAT THE STERILIZATION THERMAL ENVIRONMENT BECOMES A BASIC DESIGN CONDITION FOR LANDER ENTRY SHIELD-BOND-SUBSTRUCTURE COMBINATIONS. THE STERILIZATION ENVIRONMENT MAY BECOME THE LIMITING DESIGN PARAMETER IN THE CHOICE OF

(a) THE BOND TYPE
(b) THE BOND THICKNESS
(c) THE SHIELD MATERIAL AS AFFECTED BY BOND LIMITATIONS.

APPENDIX C

BEAM ANALYSIS

BY P. Kuzmik CK. DATE 2-14-64 REV.	GENERAL ELECTRIC	PAGE 1 MODEL REPORT STERILIZATION

BEAMS

ASSUMED CONDITIONS

MATL 2024-T3

LENGTH = 90 IN

W = TOTAL LOAD = 200 LB (1.0 G)

w' = UNIFORM LOAD = $\frac{200}{90}$ = 2.22 LB/IN

ASSUME BEAM WEIGHTS .28 LB/IN (25.0 LB)

THEN:

$$\left.\begin{array}{l} W = 225 \text{ LB} \\ w = 2.50 \text{ LB/IN} \end{array}\right\} 1.0\ G$$

CALCULATE I REQUIRED FOR NATURAL FREQUENCY = 100

$$f = \frac{(nL)^2}{L^2}\sqrt{\frac{gEI}{w}} \qquad \text{(REF THOMSON PG 217)}$$

SIMPLY SUPPORTED -

$$nL^2 = 9.87$$

$$\therefore 100 = \frac{9.87}{8649}\sqrt{\frac{384 \times 10.5 \times 10^6}{2.5} I} = .00114\sqrt{1612.8 \times 10^6 I}$$

$$= .00114 \times 40200\sqrt{I}$$

$$\sqrt{I} = \frac{100}{45.8} = 2.18 \ ; \ I = \overline{2.18}^2 = \underline{\underline{4.75}} \text{ IN}^4$$

ENDS FIXED -

$$nL^2 = 22.4$$

$$\therefore 100 = \frac{22.4}{8649} \times 40200\sqrt{I} = .00259 \times 40200\sqrt{I}$$

$$= 104\sqrt{I}$$

$$\therefore \sqrt{I} = \frac{100}{104} = .96 \ ; \ I = .96^2 = \underline{\underline{.92 \text{ IN}^4}}$$

FORM 1-8136 B (9-58)

BEAMS

CALCULATE MAXIMUM BENDING MOMENTS & DEFLECTIONS
(10 G)

SIMPLY SUPPORTED

$$\text{MAX MOMENT (CENTER)} = \frac{WL}{8} = \frac{225 \times 93}{8} = \underline{2620} \text{ IN LB}$$

$$\text{MAX DEFLECTION (CENTER)} = \frac{5WL^3}{384EI} = \frac{5 \times 225 \times 804357}{384 \times 10.5 \times 10^6 I}$$

$$= \frac{900}{4040 I} = \boxed{\frac{.223}{I}}$$

FIXED ENDS

$$\text{MAX MOMENT (ENDS)} = \frac{WL}{12} = \underline{1750} \text{ IN LB}$$

$$\text{MOMENT AT CENTER} = \frac{WL}{24} = \underline{875} \text{ IN LB}$$

$$\text{MAX DEFLECTION (CENTER)} = \frac{WL^3}{384EI} = \boxed{\frac{.045}{I}}$$

BEAMS

CALCULATE DEPTH OF BEAMS

ASSUME $F_b = 35000$ PSI
ASSUME DESIGN LOAD = 20.0 G
USE I VALUES CALCULATED FOR NATURAL FREQUENCY = 100 GIVEN ON PG 1.

SIMPLY SUPPORTED

MAX MOMENT FOR 20.0 G = 20.0 × 2620 = 52400 IN LB

$F_b = \frac{Mc}{I}$ ∴ C = HALF DEPTH = $\frac{F_b I}{M} = \frac{35000 \times 4.75}{52400} = 3.17$ IN

DEPTH = 2 × C = 2 × 3.17 = **6.34 IN**

ENDS FIXED

AT THE ENDS:

MOMENT = 20.0 × 1750 = 35000 IN LB

$C = \frac{35000\, I}{35000} = I = .92$ IN

DEPTH = 2 × C = 2 × .92 = **1.84 IN**

IN CENTER:

MOMENT = 20.0 × 875 = 17500 IN LB

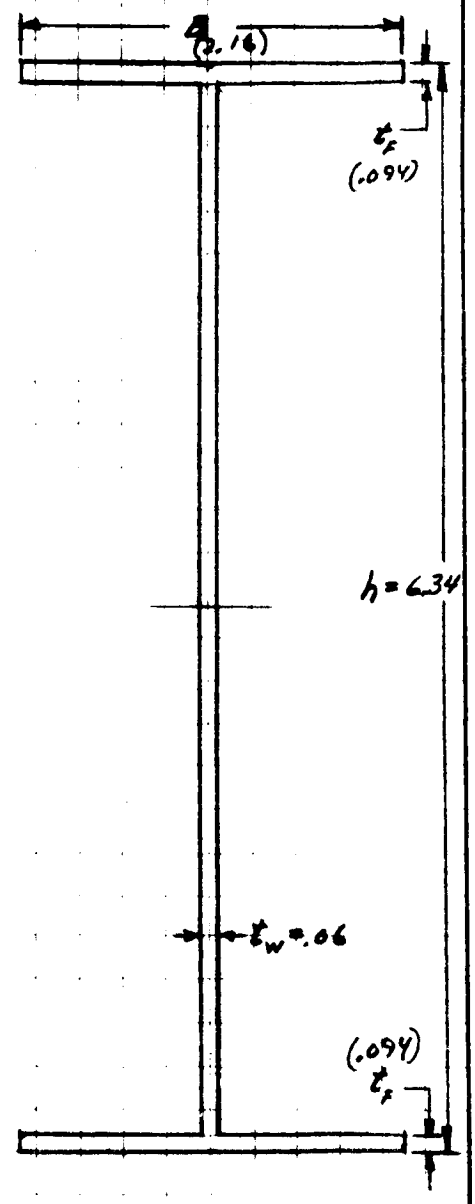
$C = \frac{35000\, I}{17500} = 2 \times I = 2 \times .92 = 1.84$

DEPTH = 2 × C = 2 × 1.84 = **3.68 IN**

∴ USE DEPTH = **3.68 IN**

~~BEAMS~~
BEAM SIZING

SIMPLY SUPPORTED



GIVEN:

$h = 6.34$ IN

$I = 4.75$ IN^4

$f_b = 35000$ PSI

ASSUME $t_W = .06$ IN

$I = I_W + I_{CMS}$

USE 6.0 FOR DEPTH OF WEB

$$I = \frac{.06 \times 6^3}{12} + \left(2 \times B t_F \times \left(3.0 + \frac{t_F}{2}\right)^2\right)$$

TAKE OUT

$$4.75 = 1.08 + \left[2 B t_F \left(3.0 + \frac{t_F}{2}\right)^2\right]$$

$$2 B t_F \times 9 = 3.67$$

$$B t_F = \frac{3.67}{18} = .204$$

ALSO $\frac{.58}{t_F} = 115$; $t_F = \frac{.58}{115} = .0435 B$

$B(.0435 B) = .204$; $.0435 B^2 = .204$

$B^2 = \frac{.204}{.0435} = 4.7$; $B = 2.16$ IN

$\therefore$ $t_F = .0435 B = .0435 \times 2.16 = .094$ IN

$$A = (.094 \times 2.16 \times 2) + (6.14 \times .06)$$
$$= .408 + .370 = \underline{.778} \text{ IN}^2$$

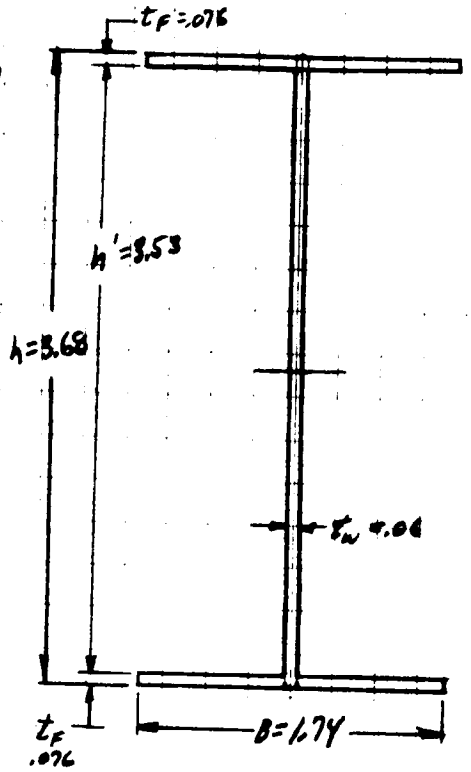
AND 10136 -2004

FORM 1-6136 B (9-56)

BY P. Kuzmik CK. DATE 2-14-64 REV.	GENERAL ELECTRIC	PAGE 5 MODEL REPORT STERILIZATION

BEAMS

BEAM SIZING (CONT'D)



FIXED ENDS

GIVEN:

$h = 3.68$ IN.

$I = .92$ IN^4

$f_b = 35000$ PSI

ASSUME $t_W = .06$ IN

USE h' = WEB DEPTH = 3.50 IN

$A = (2 \times 1.74 \times .076) + (3.53 \times .06)$

$= .265 + .212$

$= \underline{.477}$ IN^2

$I = I_W + I_{CAPS}$

$$.92 = \frac{.06 \times \overline{3.50}^2}{12} + (2Bt_F \times \overline{1.625}^2)$$

$$Bt_F = \frac{.92 - .215}{2 \times 2.65} = \frac{.705}{5.30} = .133$$

$t_F = .0435\,B$ $\left(\text{ie } \frac{.5B}{t} = 11.5\right)$

$$B^2 = \frac{.133}{.0435} = 3.05 \; ; \; B = 1.74 \text{ IN}$$

$t_F = .0435\,B = .0435 \times 1.74 = .076$ IN.

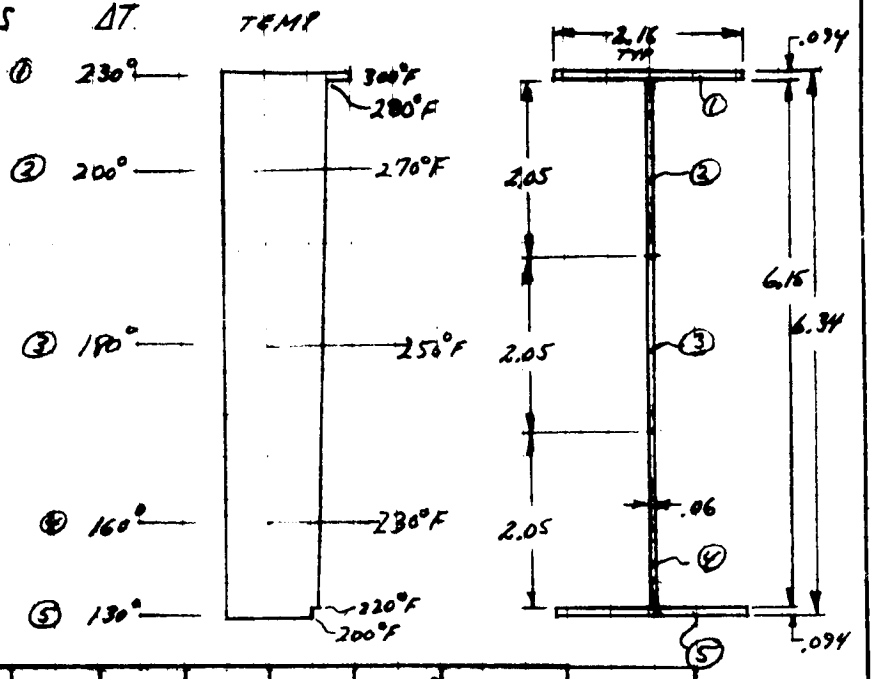
AND 10136-1501

FORM 1-8136 B (9-58)

BEAMS

THERMAL ANALYSIS

SIMPLY SUPPORTED



ELE	AREA IN²	ΔT F°	E ×10⁶ PSI	α ×10⁻⁶ IN/IN-°F	EA ×10⁶	Y IN	EAY ×10⁶	EAY² ×10⁶	EAαΔT	EAαΔT
①	.203	230	10.0	13.1	2.03	6.244	12.68	79.17	38805	6116
②	.123	200	10.1	13.0	1.24	5.172	6.41	33.15	16666	3224
③	.123	180	10.2	13.0	1.25	3.122	3.90	12.18	9126	2925
④	.123	160	10.3	12.9	1.27	1.072	1.36	1.46	2807	2621
⑤	.203	130	10.3	12.8	2.09	0	0	0	0	3478
Σ	.775				7.88		24.35	125.96	66804	18364

$$\varepsilon_0 \Sigma(EA) + \beta \Sigma(EAY) = \Sigma(EA\alpha\Delta T)$$

$$\varepsilon_0 \Sigma(EAY) + \beta \Sigma(EAY^2) = \Sigma(EA\alpha\Delta T Y)$$

$$7.88\varepsilon_0 + 24.35\beta = .018364 \qquad \varepsilon_0 = \frac{.018364 - 24.35\beta}{7.88}$$

$$24.35\varepsilon_0 + 125.96\beta = .066804$$

FORM 1-8136 B (9-58)

BEAMS

THERMAL ANALYSIS

SIMPLY SUPPORTED (CONT'D)

$$\left[(24.35)\left(\frac{.018364-24.35\beta}{7.88}\right)\right]+125.96\,\beta=.066804$$

$$3.09(.018364-24.35\beta)+125.96\beta=.066804$$

$$.056745-75.24\beta+125.96\beta=.066804$$

$$50.72\beta=.010059$$

$$\underline{\underline{\beta=.000198}}$$

$$\varepsilon_o=\frac{.018364-24.35\beta}{7.88}=\frac{.018364-.004821}{7.88}$$

$$=\frac{.013543}{7.88}=\underline{\underline{.001719}}$$

	ε_o	Y	β	E $\times 10^6$	βY	ΔT	α $\times 10^{-6}$	$\alpha\Delta T$	TOTAL STRAIN $\varepsilon=\varepsilon_o+\beta Y-\alpha\Delta T$
1	.001719	6.244	.000198	10.0	.001236	230	13.1	.003013	-.000058
2		5.172		10.1	.001024	200	13.0	.002600	.000143
3		3.122		10.2	.000618	180	13.0	.002340	-.000003
4		1.072		10.3	.000212	160	12.9	.002064	-.000133
5	.001719	0	.000198	10.3	0	130	12.8	.001664	.000055

	THERMAL STRESS $\sigma=E\varepsilon$ (psi)	C	1.0 G BENDING STRESS $\sigma=\frac{MC}{I}=\frac{2620\times C}{4.75}$
1	-580	±3.17	±1750
2	1440	±3.07	1680
3	-30		
4	-1380	±3.07	1680
5	570	±3.17	±1750

NO PROBLEM FOR SIMPLY SUPPORTED BUT

WORST CASE: ASSUME END FIXITY - THEN $\varepsilon=-\alpha\Delta T$ AND $\sigma_T=30130$ PSI

$$\sigma=\sigma_B+\sigma_T=30130+1750=\underline{\underline{31880\text{ PSI}}}$$

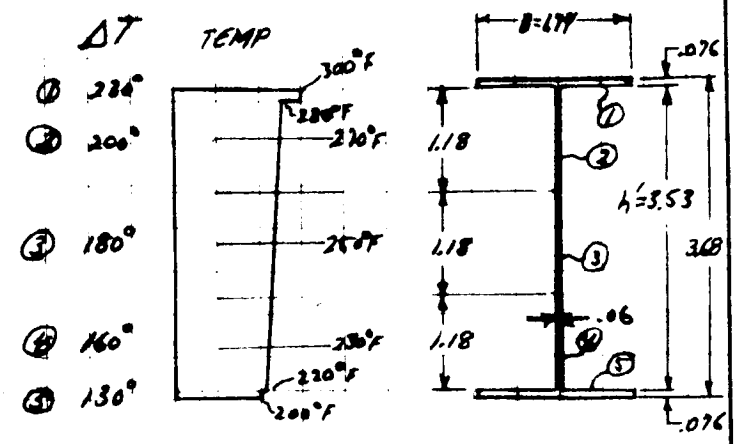
F_B (TEMP) =

FORM 1-8136 B (9-58)

BEAMS

THERMAL ANALYSIS

FIXED ENDS



ELE.	AREA IN²	ΔT F°	E ×10⁶ PSI	α ×10⁻⁶	EA ×10⁶	Y IN	EAY ×10⁶	EAȳ² ×10⁶	EAYαΔT	EAαΔT
①	.132	230	10.0	13.1	1.320	3.616	4.773	17.259	14381	3977
②	.071	200	10.1	13.0	.717	2.988	2.142	6.400	5569	1864
③	.071	180	10.2	13.0	.724	1.808	1.309	2.367	3063	1694
④	.071	160	10.3	12.9	.731	.628	.459	.288	947	1509
⑤	.132	130	10.3	12.8	1.360	0	0	0	0	2263
Σ	.477				4.852		8.683	26.314	23960	11307

$$\varepsilon_0 \Sigma(EA) + \beta \Sigma(EAY) = \Sigma(EA\alpha\Delta T)$$

$$\varepsilon_0 \Sigma(EAY) + \beta \Sigma(EA\bar{Y}^2) = \Sigma(EA\alpha\Delta TY)$$

$$4.852\,\varepsilon_0 + 8.683\,\beta = .011307 \quad ; \quad \varepsilon_0 = \frac{.011307 - 8.683\beta}{4.852}$$

$$8.683\,\varepsilon_0 + 26.314\,\beta = .023960$$

BEAMS

THERMAL ANALYSIS

FIXED ENDS (CONT'D)

$$\left[(8.683)\left(\frac{.011307-8.683\beta}{4.852}\right)\right]+26.314\beta=.023960$$

$$.02024-15.537\beta+26.314\beta=.023960$$

$$10.777\beta=.003720$$

$$\beta=\underline{\underline{.000345}}$$

$$\varepsilon_o=\frac{.011307-8.683\beta}{4.852}=\frac{.011307-(8.683\times.000345)}{4.852}$$

$$=\frac{.011307-.002996}{4.852}=\frac{.008311}{4.852}=\underline{\underline{.001713}}$$

	ε_o	Y	β	E $\times10^6$	βY	ΔT	α $\times10^{-6}$	$\alpha\Delta T$	SIMPLY SUPPORTED $\varepsilon=\varepsilon_o+\beta Y-\alpha\Delta T$	FIXED $\varepsilon=-\alpha\Delta T$	1.0 G BENDING STRESS* $=\frac{Mc}{I}$ c	
①	.001713	3.616	.000345	10.0	.001248	230	13.1	.003013	-.000052	-.003013	1.84	± 3500
②		2.988		10.1	.001031	200	13.0	.002600	.000144	-.002600	1.77	± 3367
③		1.808		10.2	.000624	180	13.0	.002340	.000003	-.002340	.6	± 1141
④		.628		10.3	.000217	160	12.9	.002064	-.000134	-.002064	1.77	± 3367
⑤	.001713	0	.000345	10.3	0	130	12.8	.001664	.000049	-.001664	1.84	± 3500

	THERMAL STRESSES - EE		(1.0 G) + (THERMAL) MAX	
	SIMPLY SUPPORTED	FIXED	SIMPLY SUPPORTED	FIXED
①	-520	-30130	-4020	-33630
②	1440	-26000	4807	-29367
③	-30	-23400	-1171	-24541
④	-1340	-20640	-4707	-24807
⑤	490	-16640	3990	-20140

* $\frac{M}{I}=\frac{1250}{.92}$ ±

FORM 1-8136 B (8-58)

BY P. KURMIK
CK.
DATE 2-27-64 REV.

GENERAL ELECTRIC

PAGE 10
MODEL
REPORT STERILIZATION

BEAMS

SUMMARY OF STRESSES DUE TO THERMAL GRADIENTS

BEAM SIZED AS SIMPLY SUPPORTED

ELEMENT	①	②	③	④	NO FIXITY ⑤	NO FIXITY ⑥	BENDING RESTRICTED ⑦	BENDING RESTRICTED ⑧	EXPANSION RESTRICTED ⑨	EXPANSION RESTRICTED ⑩	FULLY FIXED ⑪
	ε_0	βY	$\alpha \Delta T$	E $\times 10^6$	$\varepsilon_1 = \varepsilon_0 + \beta Y - \alpha \Delta T$	$\sigma = \varepsilon_1 E$ (PSI)	$\varepsilon_2 = \varepsilon_0 - \alpha \Delta T$	$\sigma = \varepsilon_2 E$ (PSI)	$\varepsilon_3 = \beta Y - \alpha \Delta T$	$\sigma = \varepsilon_3 E$ (PSI)	$\sigma = -\alpha \Delta T E$ (PSI)
1	.001719	.001236	.003013	10.0	-.000058	-580	-.001294	-12940	-.001777	-17770	-30130
2	⋮	.001024	.002600	10.1	.000143	1444	-.000881	-8900	-.001576	-15920	-26260
3	⋮	.000618	.002340	10.2	-.000003	-30	-.000621	-6330	-.001722	-17560	-23870
4	⋮	.000212	.002064	10.3	-.000133	-1370	-.000345	-3555	-.001852	-19080	-21260
5	.001719	0	.001664	10.3	.000055	570	.000055	570	-.001664	-17140	-17140

BEAM SIZED AS FULLY FIXED

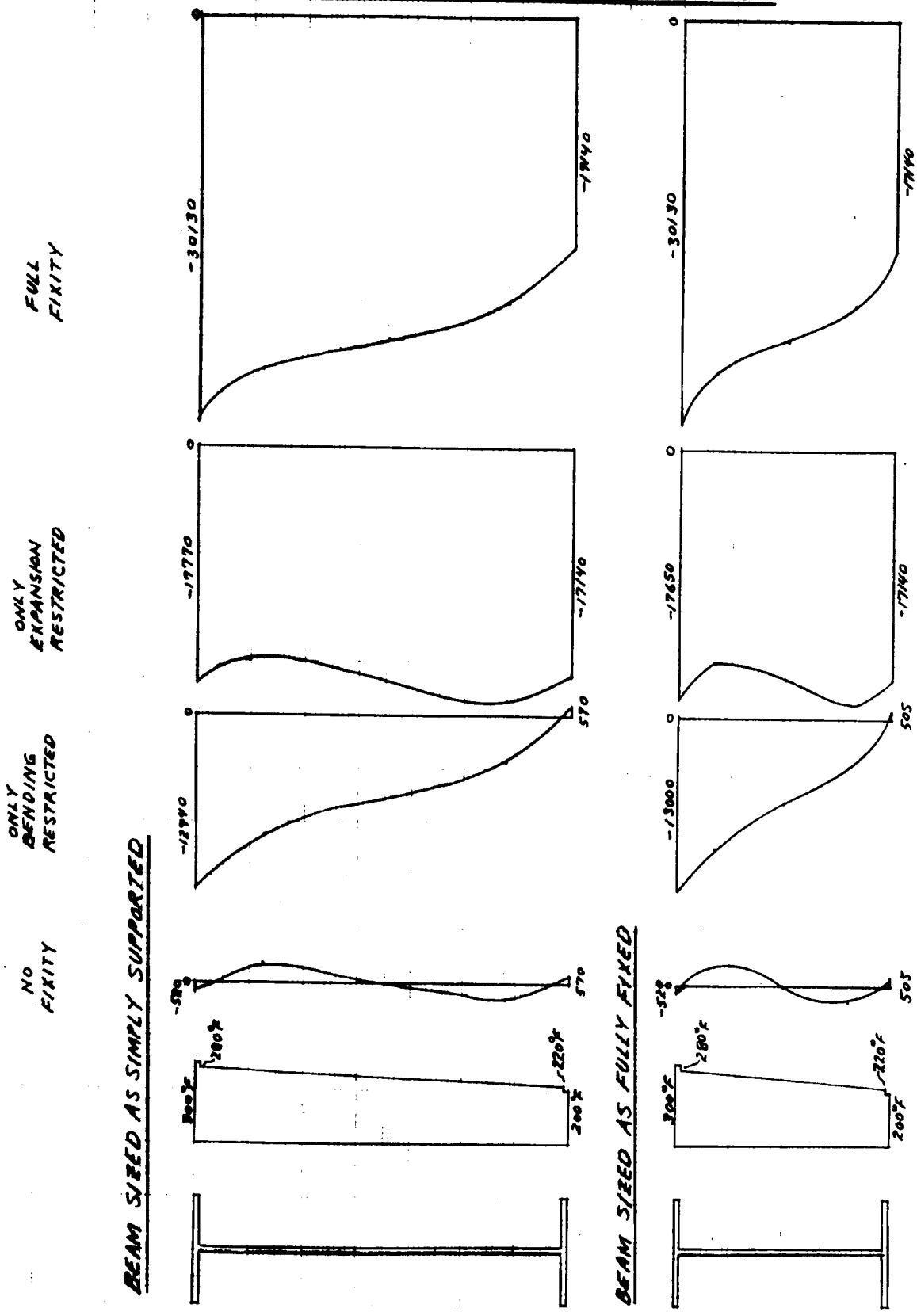
	①	②	③	④	⑤	⑥	⑦	⑧	⑨	⑩	⑪
1	.001713	.001248	.003013	10.0	-.000052	-520	-.001300	-13000	-.001765	-17650	-30130
2	⋮	.001031	.002600	10.1	.000144	1454	-.000887	-8960	-.001569	-15850	-26260
3	⋮	.000624	.002340	10.2	-.000003	-30	-.000627	-6400	-.001716	-17500	-23870
4	⋮	.000217	.002064	10.3	-.000134	-1380	-.000351	-3615	-.001847	-19025	-21260
5	.001713	0	.001664	10.3	.000049	505	.000049	505	-.001664	-17140	-17140

MATERIAL IS 2024-T3 ALUMINUM ALLOY

FORM 1-8136 B (9-58)

STRESSES IN BEAMS DUE TO DEPTHWISE GRADIENTS

P. Kuzmik
2-27-64



APPENDIX D

COMPRESSION MEMBERS

BY P. Kuzmik CK. DATE 2-5-64 REV.	GENERAL ELECTRIC	PAGE 1 MODEL REPORT STERILIZATION

COLUMN DESIGN FOR ROOM TEMPERATURE

GIVEN: $L = 40$ IN ; $P_c = 2000$ LB ; $E = 10.5 \times 10^6$ PSI ; $F_{TY} = 42{,}000$ PSI
(ALUMINUM ALLOY TUBING 2024-T3)

FIND: MAXIMUM F_{cr} WHICH WILL DICTATE A MINIMUM OR OPTIMUM AREA.

APPROACH:

1. USING MIL-HDBK-5 (PG 3.4.2.1) CALCULATE THE TRANSITIONAL L/ρ.
2. CALCULATE F_c ASSUMING VALUES OF L/ρ WHERE:
   FOR SHORT COLUMNS (IE L/ρ < TRANSITIONAL)

$$F_c = F_{co}\left[1 - \frac{0.385\,(L/\rho)}{\pi\sqrt{E/F_{co}}}\right]$$

   FOR LONG COLUMNS (IE L/ρ > TRANSITIONAL)

$$F_c = \frac{C\pi^2 E}{(L/\rho)^2}$$

3. SOLVE FOR D'/t WHICH SATISFIES THE EQUATION

$$F_{cr} = \frac{.4E}{D'/t} \quad \text{(NOTE: } D' \text{ IS AVERAGE DIAMETER)}$$

   THIS IS THE MAXIMUM VALUE FOR THE ASSUMED STRESS LEVEL F_{cr}.

4. USING ρ FROM STEP 2 SOLVE FOR D' REQUIRED FOR GIVEN STRESS LEVEL WHERE

$$\rho = .355\,D' \quad \text{(FOR ROUND TUBES)}$$

5. SOLVE FOR t USING AREA COMPATIBLE WITH STEP 2.

$$A = P/F_c$$

$$\& \quad A = \pi D' t$$

6. CALCULATE ACTUAL D'/t FROM STEPS 4 & 5.

FORM 1-8136 B (9-58)

COLUMN DESIGN FOR ROOM TEMPERATURE (CONT'D)

7. PLOT VALUES OF D'/t FOR ASSUMED STRESS LEVELS AND USE THE HIGHEST STRESS LEVEL PERMISSIBLE AS DICTATED BY STEP 3.

SOLUTION OF EXAMPLE:

1. TRANSITIONAL $L/\rho = 1.732\pi\sqrt{E/F_c} = 78.3$

2. ASSUME $L/\rho = 70$ (SHORT COLUMN) $\therefore F_c = 20400$ PSI

3. MAX $\frac{D'}{t} = \frac{.4E}{F_c} = 206$

4. $L/\rho = 70 = \frac{40}{\rho}$; $\rho = \frac{40}{70} = .572$; $D' = 2.82\rho = 1.613$ IN

5. $A = \frac{P}{F_c} = \frac{2000}{20400} = .098$ IN^2

$A = \pi D' t \therefore t = \frac{A}{3.14D'} = \frac{.098}{3.14 \times 1.613} = .0194$ IN

6. $D'/t = \frac{206}{.0194} = 83$

SINCE MAX D'/t PERMISSIBLE IS 206 (STEP 3) A HIGHER STRESS LEVEL IS POSSIBLE. BY PLOTTING VARIOUS SOLUTIONS AS GIVEN ABOVE THE HIGHEST STRESS LEVEL FOR A GIVEN COLUMN LENGTH AND LOAD MAY BE DETERMINED. THE CURVES ALSO PERMIT LIMITED INTERPOLATION FOR OTHER CONDITIONS OF COLUMN LENGTH AND LOAD.

COLUMN EQUATIONS: ROOM TEMPERATURE

REF MIL-HDBK 5 PG 3.4.2.1

$$F_{co} = F_{cy}\left(1 + \frac{\sqrt{F_{cy}}}{1000}\right)$$

FOR 2024-T3 TUBES $F_{cy} = 42000$; $E = 10.5 \times 10^6$ PSI (ROOM TEMP)

$$\therefore F_{co} = 42000\left(1 + \frac{204.9}{1000}\right) = 42000 \times 1.2049 = \underline{50600} \text{ PSI}$$

TRANSITIONAL L/ρ

$$= 1.732\pi\sqrt{E/F_{co}} = 5.44\sqrt{207.5}$$

$$= 5.44 \times 14.4 = \underline{78.3}$$

PARABOLA STRAIGHT LINE

FOR SHORT COLUMNS ($L/\rho < 78.3$) 1.385

$$F_c = F_{co}\left[1 - 0.385\left(L/\rho\right)/\pi\sqrt{E/F_{co}}\right]$$

$$= 50600\left(1 - \frac{.385\, L/\rho}{3.14 \times 14.4}\right) = 50600\left(1 - \frac{.385\, L/\rho}{45.2}\right)$$

FOR LONG COLUMNS ($L/\rho > 78.3$) 1.381

EULER

$$F_c = \frac{C\pi^2 E}{(L/\rho)^2} = \frac{2.86 \times 10.5 \times 10^6}{(L/\rho)^2} = \frac{103.53 \times 10^6}{(L/\rho)^2}$$

FORM 1-8136 B (9-58)

COLUMN DATA: ROOM TEMPERATURE FOR 2024-T3 TUBES

SHORT COLUMNS

①	②	③	④	⑤
L/ρ	$.385\,L/\rho$	$\frac{.385\,L/\rho}{45.2}$	1.0 − ③	$F_c = 50600 \times$ ④
15	5.8	.128	.872	44120
30	11.6	.257	.743	37600
40	15.4	.341	.659	33350
45	17.3	.383	.617	31220
50	19.2	.425	.575	29100
55	21.2	.469	.531	26870
60	23.1	.511	.489	24740
65	25.0	.553	.447	22620
70	27.0	.597	.403	20390
75	28.9	.639	.361	18270
78.3	30.1	.666	.334	16900

$F_{cy} = 42000$ PSI
$E = 10.5 \times 10^6$ PSI

LONG COLUMNS

①	②	③
L/ρ	$(L/\rho)^2$	$F_c = \frac{103.53 \times 10^6}{(L/\rho)^2}$
78.3	6131	16900
80	6400	16180
90	8100	12780
100	10000	10350
110	12100	8560
120	14400	7190
130	16900	6130
140	19600	5280

EQUATIONS

SHORT COLUMN

$$F_c = F_{co}\left[1 - \frac{0.385\,L/\rho}{\pi\sqrt{E/F_{co}}}\right]$$

$$= 50600\left(1 - \frac{.385\,L/\rho}{45.2}\right)$$

LONG COLUMN

$$F_c = \frac{C\pi^2 E}{(L/\rho)^2} = \frac{103.53 \times 10^6}{(L/\rho)^2}$$

$(C = 1.0)$

BY P. KUZMIK
CK.
DATE 2-1-64 REV.

GENERAL ELECTRIC

PAGE 5
MODEL
REPORT STERILIZATION

COLUMN DESIGN - ROOM TEMPERATURE

①	②	③	④	⑤	⑥	⑦	⑧	⑨	⑩	⑪	⑫	⑬	⑭	⑮
L/ρ	F_c	MAX D'/t	ρ FOR GIVEN L = L/①			D' FOR COLS ④,⑤,⑥ D' = 2.82 ρ			AREAS FOR ⊛ GIVEN ρ = LOADS			AREA/3.14 FOR P =		
*	PSI		L=20	L=40	L=60	L=20	L=40	L=60	1000	2000	3000	1000	2000	3000
30	37600	112	.667	1.333	2.000	1.881	3.759	5.640	.027	.054	.081	.0086	.0172	.0258
40	33350	126	.500	1.000	1.500	1.410	2.820	4.230	.030	.060	.090	.0095	.0191	.0286
50	29100	144	.400	.800	1.200	1.128	2.256	3.384	.034	.068	.102	.0108	.0216	.0324
60	24750	169	.333	.667	1.000	.939	1.881	2.820	.040	.080	.120	.0127	.0254	.0382
70	20400	206	.286	.572	.858	.807	1.613	2.420	.049	.098	.147	.0156	.0312	.0467
78.3	16900	248	.255	.510	.765	.719	1.438	2.157	.059	.118	.177	.0188	.0375	.0563
80.0	16177	260	.250	.500	.750	.705	1.410	2.115	.062	.124	.186	.0197	.0394	.0591
90	12780	329	.222	.444	.667	.626	1.252	1.881	.078	.156	.234	.0248	.0496	.0744
100	10350	406	.200	.400	.600	.564	1.128	1.692	.097	.194	.291	.0308	.0617	.0925
110	8560	490	.182	.364	.546	.513	1.026	1.540	.117	.234	.351	.0372	.0744	.1116
120	7190	584	.167	.334	.500	.471	.942	1.410	.139	.278	.417	.0442	.0884	.1326
130	6130	685	.154	.308	.462	.434	.868	1.303	.163	.326	.489	.0518	.1037	.1555
140	5280	795	.143	.286	.429	.403	.806	1.210	.189	.378	.567	.0601	.1202	.1803

⊛ $A = P/②$

** $F_c = \frac{k_2\sqrt{EE_T}}{D'/t}$; $D'/t = \frac{4.2 \times 10^6}{F_c}$

* TRANSITIONAL $L/\rho = 78.3$ (REF PG ____)

FORM 1-8136 B (9-58)

BY P KUZMIK
CK.
DATE 2-11-64 REV.

GENERAL ELECTRIC

PAGE 6
MODEL
REPORT STERILIZATION

COLUMN DESIGN – ROOM TEMPERATURE (CONT'D)

①	②	③	⑯	⑰	⑱	⑲	⑳	㉑	㉒	㉓	㉔
			(*)(*) RESULTING THICKNESS FOR D'								
			L=20"			L=40"			L=60"		
L/ρ	F_c PSI	MAX D'/t	1000#	2000#	3000#	1000#	2000#	3000#	1000#	2000#	3000#
30	37600	112	.0046	.0091	.0138	.0023	.0046	.0069	.0015	.0030	.0045
40	33350	126	.0067	.0135	.0201	.0034	.0068	.0102	.0022	.0044	.0066
50	29100	144	.0096	.0191	.0288	.0048	.0096	.0144	.0032	.0064	.0096
60	24750	169	.0135	.0270	.0405	.0068	.0136	.0204	.0045	.0090	.0135
70	20400	206	.0193	.0386	.0579	.0097	.0194	.0291	.0064	.0128	.0192
78.3	16900	248	.0261	.0522	.0783	.0131	.0262	.0393	.0087	.0174	.0261
80	16177	260	.0279	.0558	.0837	.0140	.0280	.0420	.0093	.0186	.0279
90	12780	329	.0396	.0792	.1188	.0198	.0396	.0594	.0132	.0264	.0396
100	10350	406	.0546	.1092	.1638	.0273	.0546	.0819	.0182	.0364	.0546
110	8560	490	.0725	.1450	.2175	.0362	.0724	.1086	.0242	.0484	.0726
120	7190	584	.0938	.1876	.2814	.0469	.0938	.1407	.0313	.0626	.0939
130	6130	685	.1194	.2388	.3582	.0597	.1194	.1791	.0398	.0796	.1194
140	5280	795	.1491	.2982	.4473	.0746	.1492	.2238	.0497	.0994	.1491

(*)(*) $A = 3.14 D't$; $t = \left(\frac{A}{3.14}\right) \times \left(\frac{1}{D'}\right)$

FORM 1-8136 B (9-58)

BY P. Kuzmik
CK.
DATE 2-11-64 REV.

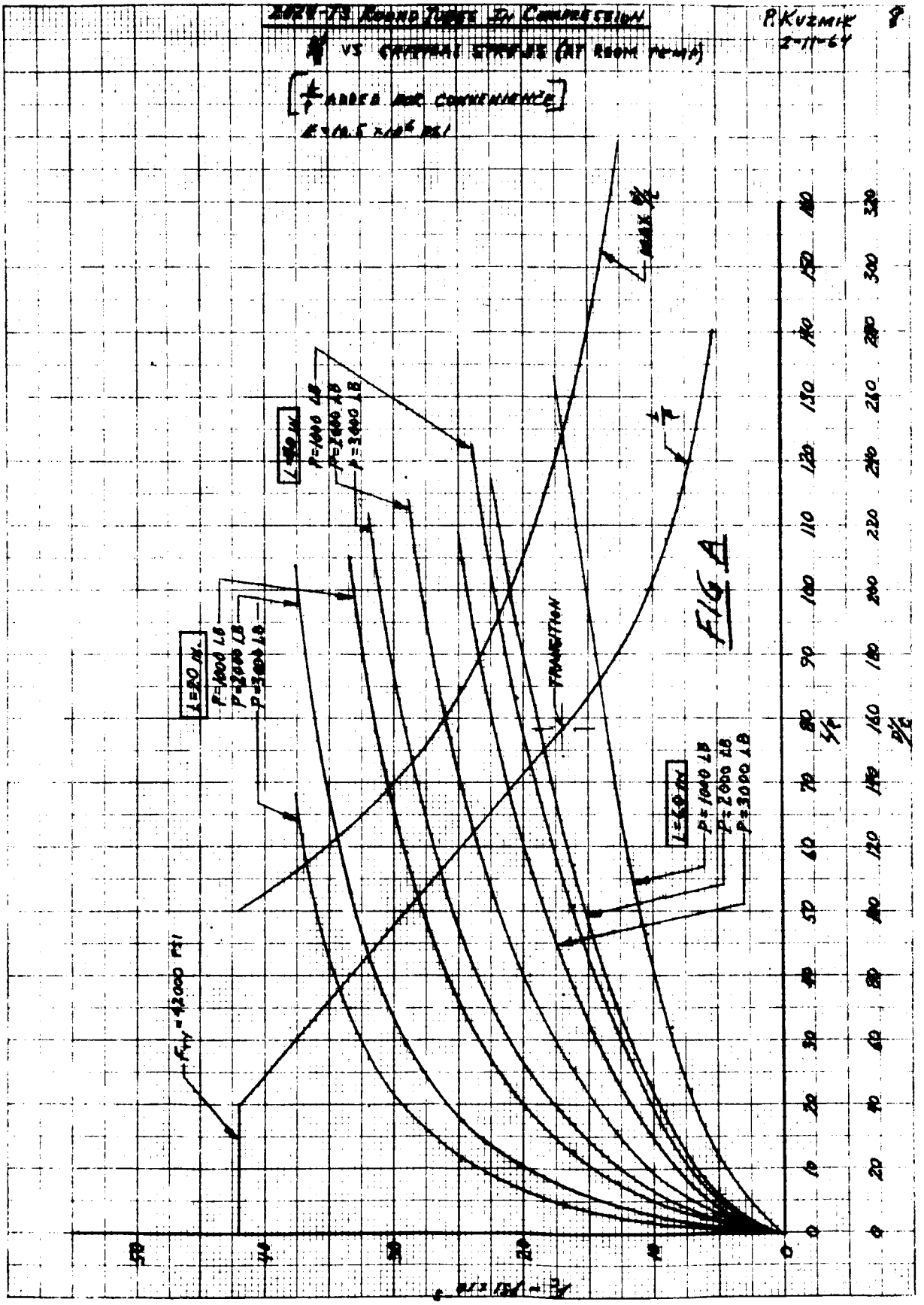
GENERAL ELECTRIC

PAGE 7
MODEL
REPORT STERILIZATION

COLUMN DESIGN - ROOM TEMPERATURE (CONT'D)

(1)	(2)	(3)	(25)	(26)	(27)	(28)	(29)	(30)	(31)	(32)	(33)
			ACTUAL D/t								
			L = 20"			L = 40"			L = 60"		
L/r	F_c PSI	MAX D/t	1000#	2000#	3000#	1000#	2000#	3000#	1000#	2000#	3000#
30	37400	112	409	207	136	1634	817	544	3760	1880	1252
40	33350	126	210	105	70	829	415	276	1923	962	640
50	29100	144	118	59	39	470	235	157	1058	529	352
60	24950	169	70	35	23	277	139	92	627	314	209
70	20900	206	42	21	14	166	83	55	378	189	126
78.3	16900	248	28	14	9	108	55	36	248	124	83
80	16177	260	25	13	8	101	51	34	227	114	76
90	12780	329	16	8	5	63	32	21	142	71	47
100	10350	406	10	5	3	41	21	14	93	47	31
110	8560	490	7	4	2	28	14	9	64	32	21
120	7190	584	5	3	2	20	10	7	45	23	15
130	6130	685	4	2	1	15	8	5	33	17	11
140	5280	795	3	2	1	9	5	3	24	12	8

FORM 1-6188 B (9-58)


2024-T3 ROUND TUBES IN COMPRESSION
VS CRITICAL STRESS (AT ROOM TEMP)
ADDED FOR CONVENIENCE
P. KUZMIK
2-11-64
L=20 IN.
P=1000 LB
P=2000 LB
P=3000 LB
L=60 IN.
P=1000 LB
P=2000 LB
P=3000 LB
TRANSITION
MAX
FIG A

COLUMN CHECK AT TEMPERATURE (300°F 1000 HRS)

THE OBJECTIVE AT THIS POINT IS TO DETERMINE WHAT CHANGE (RISE) IN TEMPERATURE IS CRITICAL FOR A GIVEN COLUMN DESIGNED FOR ROOM TEMPERATURE. USING THE AREAS OBTAINED FOR COLUMNS IN THE PREVIOUS SECTION THE INTERNAL LOAD BUILDUP FOR A TEMPERATURE CHANGE IS OBTAINED BY THE EQUATION

$$P = AE\alpha\Delta T$$

FOR PURPOSES OF THIS STUDY THE FOLLOWING ASSUMPTIONS ARE MADE:

1. COLUMNS ARE FULLY RESTRAINED AXIALLY. THIS SAYS THAT THE RESTRAINING STRUCTURE DOES NOT EXPAND WITH TEMPERATURE. BY USING A GRADIENT VALUE AND ASSUMING EQUAL RATES OF EXPANSION A REASONABLE RESULTING LOAD MAY BE OBTAINED. HOWEVER, FULL RESTRAINT ALSO MEANS THAT THE RESTRAINING STRUCTURE IS INFINITELY STIFF. THIS RESULTS IN CONSERVATISM AND NO ATTEMPT IS MADE HERE TO ACCOUNT FOR IT. ALSO CREEP EFFECTS ARE NOT ANALYZED WHICH RESULTS IN UNCONSERVATISM.

2. VALUE FOR E ARE TAKEN AT 300°F FOR ALL TEMPERATURES CONSIDERED. THE VALUE USED IS 10.0×10^6 PSI.

3. THE VALUE USED FOR α IS MEAN FOR CHANGE FROM ROOM TO 300°F. NUMBER USED IS 13.1×10^{-6} IN/IN/F°. STEPS 2 & 3 ARE CONSERVATIVE FOR TEMPERATURES BELOW 300°F.

FOR GIVEN VALUES OF b/t & L/ρ A CRITICAL STRESS LEVEL IS OBTAINED FOR COLUMNS SUBJECT TO THERMAL ENVIRONMENT. THUS AN ALLOWABLE LOAD OR ΔT IS OBTAINED.

COLUMN CHECK AT TEMPERATURE (CONT'D)

EXAMPLE USING CURVES IN FIG. A

GIVEN: COLUMN WITH 60 IN LENGTH AND AXIAL COMPRESSION LOAD = 2000 LBS.

FIND: WHAT TEMPERATURE RISE IS CRITICAL ASSUMING FULL AXIAL RESTRAINT.

PROCEDURE:

1. FROM FIG A DETERMINE THE MAX D'/t.
   FOR GIVEN CONDITION MAX $D'/t = 199$ ($\psi_p = 67$)
2. READ F_{CR} WHICH IS THE OPTIMUM STRESS LEVEL.
   $F_{CR} = 21{,}200$ PSI.

3. CALCULATE AREA.

$$\frac{P}{A} = F_{CR} \; ; \; A = \frac{2000}{21200} = .0943 \text{ IN}^2$$

4. CALCULATE F_{CR} FOR $D'/t = 199$ (AT 300°F; $E = 10.0 \times 10^6$ PSI)
   $\alpha = 13.1 \times 10^{-6}$ IN/IN/F°

$$F_{CT} = \frac{.4E}{D'/t} = \frac{.4 \times 10.0 \times 10^6}{199} = 20000 \text{ PSI}$$

ALSO FOR $\psi_p = 67$

$$F_{CT} = 19500 \text{ PSI} \quad (\text{USE MINIMUM } F_C = 19500 \text{ PSI})$$

5. CALCULATE ΔT_{CR}

$$\Delta T_{CR} = \frac{P_{TEMP}}{A E_T \alpha_T} = \frac{F_{CR_T}}{E_T \alpha_T} = \frac{19500}{10.0 \times 13.1} = \frac{19500}{131} = 149°F$$

ALSO:
$P_T = 149 \times 131 \times .0943 = 1853$ LB

NOTE: CURVES FOR STEP 4 ARE PROVIDED IN FIG B.
MAXIMUM STRESS LEVEL $= F_{TY} = 38{,}600$ PSI
FOR 300°F EXPOSED 1000 HRS.

FORM 1-8136 B (9-58)

COLUMN EQUATIONS: 300°F FOR 1000 HOURS

FOR 2024-T3 TUBES

$F_{cy} = 38600$ PSI

$E = 10.0 \times 10^6$ PSI

$$F_{co} = F_{cy}\left(1 + \frac{\sqrt{F_{cy}}}{1000}\right) = 38600\left(1 + \frac{196}{1000}\right)$$

$$= 38600 \times 1.196 = \underline{46200} \text{ PSI}$$

TRANSITIONAL L/ρ

$$= 1.732\,\pi\sqrt{E/F_{co}} = 5.44\sqrt{\frac{10.0 \times 10^6}{46200}}$$

$$= 5.44 \times \sqrt{216} = 5.44 \times 14.7 = \underline{80.0}$$

PARABOLA
STRAIGHT LINE

SHORT COLUMNS ($L/\rho < 80.0$) EQ 1.385

$$F_c = F_{co}\left[1 - 0.385\left(L/\rho\right)/\pi\sqrt{E/F_{co}}\right]$$

$$= 46200\left(1 - \frac{.385\, L/\rho}{3.14 \times 14.7}\right) = 46200\left(1 - \frac{.385\, L/\rho}{46}\right)$$

EULER

LONG COLUMNS ($L/\rho > 80.0$) EQ 1.381

$$F_c = \frac{C\pi^2 E}{(L/\rho)^2} = \frac{9.86 \times 10.0 \times 10^6}{(L/\rho)^2} = \frac{98.6 \times 10^6}{(L/\rho)^2}$$

FORM 1-6136 B (9-58)

COLUMN DATA: 300°F FOR 1000 HRS (AT TEMPERATURE)
2024-T3 TUBES

SHORT COLUMNS

①	②	③	④	⑤
L/ρ	$.385\,L/\rho$	$\frac{.385\,L/\rho}{46.0}$	$1.0 - ③$	$F_c = 46200$ ④
15	5.8	.126	.874	40380
30	11.6	.252	.748	34560
40	15.4	.335	.665	30720
50	19.2	.417	.583	26930
60	23.1	.502	.498	23000
70	27.0	.587	.413	19080
80	30.8	.670	.330	15250

$F_{cy} = 38600$ PSI
$E = 10.0 \times 10^6$ PSI

LONG COLUMNS

①	②	③
L/ρ	$(L/\rho)^2$	$F_c = \frac{9.86 \times 10^6}{(L/\rho)^2}$
80	6400	15400
90	8100	12170
100	10000	9860
110	12100	8150
120	14400	6850
130	16900	5830
140	19600	5030
150	22500	4380

EQUATIONS

SHORT COLUMNS:

$$F_c = F_{co}\left[1 - \frac{.385\,L/\rho}{\pi\sqrt{E/F_{co}}}\right]$$

$$= 46200\left(1 - \frac{.385\,L/\rho}{46.0}\right)$$

LONG COLUMNS:

$$F_c = \frac{c\pi^2 E}{(L/\rho)^2} = \frac{9.86 \times 10^6}{(L/\rho)^2}$$

$(c = 1.0)$

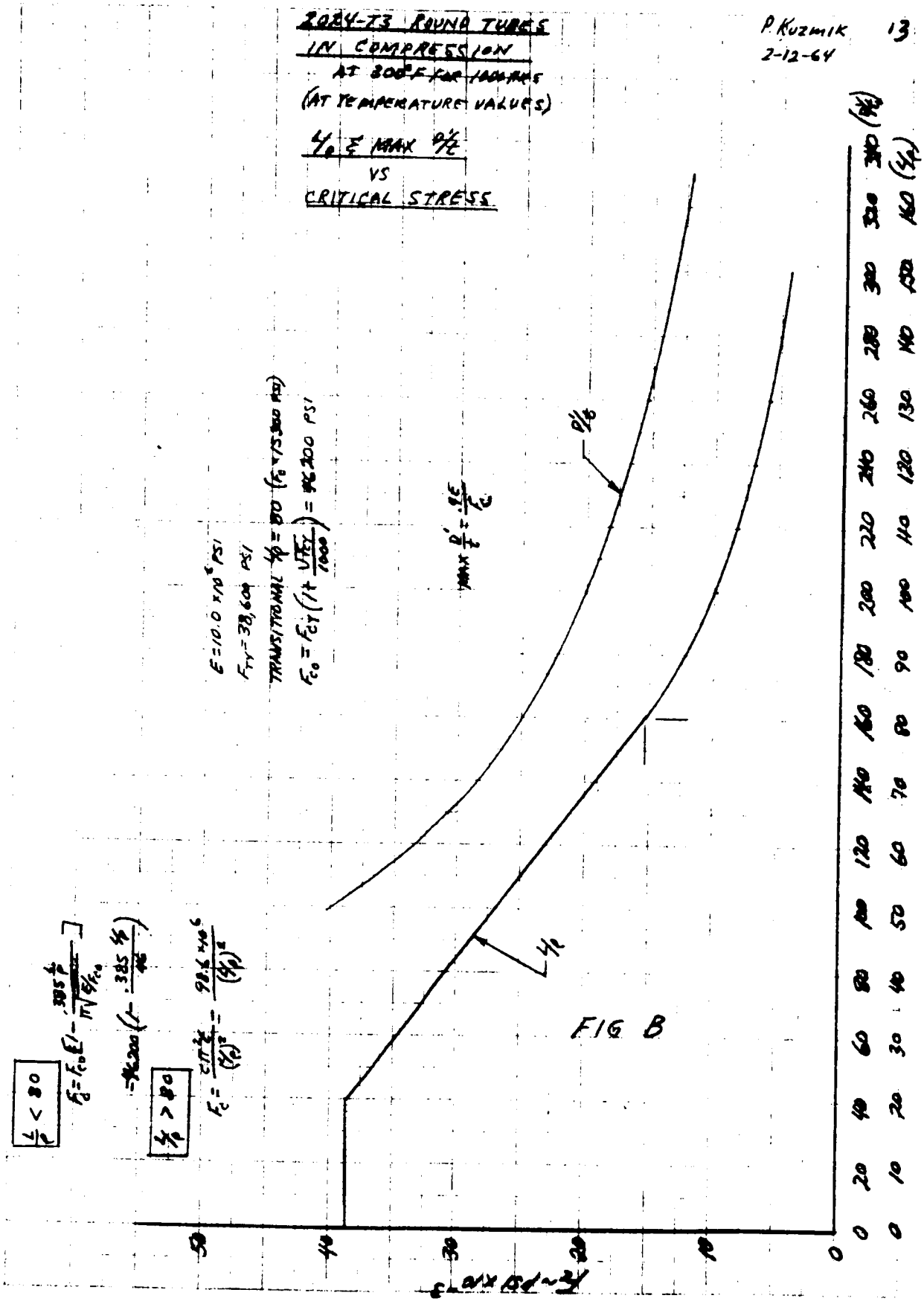

2024-T3 ROUND TUBES
IN COMPRESSION
AT 300°F FOR 1000 HRS
(AT TEMPERATURE VALUES)
L/ρ & MAX D'/t
VS
CRITICAL STRESS
P. Kuzmik
2-12-64
13
E = 10.0 x 10^6 PSI
F_TY = 38,600 PSI
TRANSITIONAL L/ρ = 80 (F_c = 15300 PSI)
F_co = F_cY (1 + √F_cY / 1000) = 46200 PSI
MAX D'/t = .9E / F_c
D'/t
L/ρ
L/ρ < 80
F_c = F_co [1 − .385 L/ρ / (π √(E/F_co))]
= 46200 (1 − .385 L/ρ / 46)
L/ρ > 80
F_c = cπ²E / (L/ρ)² = 98.6 x 10^6 / (L/ρ)²
F_c ~ PSI x 10^-3
0 10 20 30 40 50
0 20 40 60 80 100 120 140 160 180 200 220 240 260 280 300 320 340 (D'/t)
0 10 20 30 40 50 60 70 80 90 100 110 120 130 140 150 160 (L/ρ)

FIG B

COLUMN CHECK AT TEMPERATURE (CONT'D)

TABLE OF CRITICAL ΔT FOR CONDITIONS GIVEN

LOAD P (LB)	LENGTH (IN)	AT ROOM TEMPERATURE MAX* D/t	F_c* (PSI)	AREA $= \frac{P}{F_c}$	L/ρ	AT 300°F (1000 HRS) ** F_c (PSI) FOR D/t	FOR L/ρ	*** MAX P (LB)	⊛ ΔT_{CR} (F°)
1000	20	138	30400	.033	47	28200	28000	924	214
2000	20	123	34400	.058	38	32500	31500	1827	240
3000	20	115	36800	.082	32	34700	33700	2763	257
1000	40	192	22000	.045	67	21000	20200	909	154
2000	40	160	26000	.077	57	25000	24000	1848	183
3000	40	146	28500	.105	51	27200	26400	2772	202
1000	60	248	17000	.059	78	16100	16000	994	122
2000	60	200	21000	.095	69	20000	19500	1853	149
3000	60	179	23400	.128	63	22300	21700	2778	166

⊛ $\Delta T_{CR} = \frac{F_c \text{ (SMALLER AT TEMP)}}{E\alpha} = \frac{F_c}{131}$

*** MAX P = (SMALLER VALUE OF F_c AT TEMP) * (AREA)

** FIG B

* FROM FIG A

FORM 1-6136 B (9-58)

APPENDIX E

RING-STIFFENED SHELLS

BY P. Kurnik CK. DATE 3-9-64 REV.	GENERAL ELECTRIC	PAGE 1 MODEL REPORT STERILIZATION

RING STIFFENED SHELL

OBJECT: DETERMINE THE MAGNITUDE OF DISCONTINUITY STRESSES THAT WOULD BE THE DIRECT RESULT OF A THERMAL GRADIENT ACROSS A SHELL AND RING JUNCTURE.

PROCEDURE: A TYPICAL EXAMPLE OF STRUCTURE IS ASSUMED BUT NO EFFORT IS MADE TO SIZE IT FOR OTHER CONDITIONS. A THERMAL GRADIENT OF 150°F IS ASSUMED. FOR THE SOLUTION AS PRESENTED THE SHELL IS 150°F HOTTER THAN THE STIFFENER CAUSING A TENSION LOAD TO EXIST IN THE FASTENERS.

A REDUNDANT ANALYSIS IS PERFORMED TO CALCULATE THE SHELL MOMENTS AND THE RADIAL LOAD. FROM THESE THE MAXIMUM HOOP STRESSES AND MERIDIONAL BENDING STRESSES ARE DETERMINED.

ALSO INCLUDED IN THIS REPORT IS A SOLUTION SHOWING THE EFFECTS OF SHELL RADIUS AND THICKNESS.

GIVEN:

r = RADIUS = 50 IN.

t = SHELL THICKNESS = .10 IN.

A = RING AREA = .218 IN^2 } AND 10138-2001

I_x = .0148 IN^4

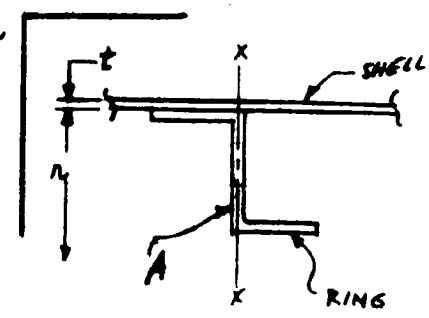
ΔT = THERMAL GRADIENT = 150°F

$$\lambda = \sqrt[4]{\frac{3(1-\nu^2)}{r^2 t^2}} \; ; \; \nu = .3 \quad \therefore \lambda = \frac{1.28}{\sqrt{rt}} = .573 \; ; \; \lambda^3 = .188$$

MATL: 2024-T3 ALUMINUM ALLOY

$E = 10.0 \times 10^6$ PSI } AT 300°F

$\alpha = 13.0 \times 10^{-6}$ IN/IN/°F



FORM 1-6136 B (9-58)

RING STIFFENED SHELL

CALCULATE AN EQUIVALENT RING WITH A RECTANGULAR CROSS SECTION AS SHOWN. EQUIVALENT IN THIS CASE MEANS EQUAL AREA AND EQUAL I_x. EQUAL AREAS WILL YIELD EQUAL RADIAL DISPLACEMENT FOR RADIAL LOAD X_1. THE I_x VALUES MUST BE EQUAL TO YIELD EQUAL STIFFNESS WHEN APPLYING END LOADS X_1 OR X_2 (I.E. TWIST) ON THE RING. (REF TIMOSHENKO, "STRENGTH OF MATERIALS" PART II PAGE 178.)

$$\therefore A = .218 = t_1 a \quad ; \quad t_1 = \frac{.218}{a}$$

AND $I_x = .0148 = \frac{t_1 a^3}{12}$

THEN $t_1\left(\frac{a^3}{12}\right) = \frac{.218}{a} \times \frac{a^3}{12} = \frac{.218}{12} a^2 = .0148$

$$a^2 = \frac{12 \times .0148}{.218} = \frac{.178}{.218} = .815 \text{ IN}^2$$

$$a = \underline{\underline{.91}} \text{ IN}$$

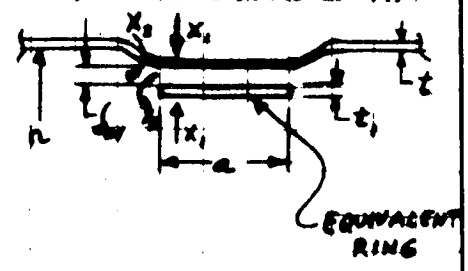
$$t = \frac{.218}{.91} = \underline{\underline{.24}} \text{ IN}$$

ASSUME THE SHELL AND RING ARE SEPARATED. CALCULATE THE RADIAL DISPLACEMENT OF THE SHELL RELATIVE TO THE RING.

$$\delta_{01} = \alpha r \Delta T = 13.0 \times 10^{-6} \times 50 \times 150 = \underline{\underline{.0923}} \text{ IN}$$

THIS IS THE GAP THAT MUST BE CLOSED BY THE REDUNDANT FORCES. THE SHELL AND RING ARE ASSUMED TO HAVE THE SAME SLOPE AT THE JOINT. THIS ASSUMPTION SETS THE SLOPE REQUIREMENT AND PROVIDES THE SECOND EQUATION NECESSARY FOR SOLUTION OF THE TWO UNKNOWNS X_1 & X_2.

$$\therefore \underline{\underline{\delta_{02} = 0}}$$



RING STIFFENED SHELL

MATRIX

$$\begin{bmatrix} \delta_{01} \\ \delta_{02} \end{bmatrix} = \begin{bmatrix} \delta_{11} & \delta_{12} \\ \delta_{21} & \delta_{22} \end{bmatrix} \begin{bmatrix} X_1 \\ X_2 \end{bmatrix} = \begin{bmatrix} .0973 \\ 0 \end{bmatrix}$$

EQUATIONS

$$\delta_{11} X_1 + \delta_{12} X_2 = \delta_{01} = .0973 \text{ IN}$$

$$\delta_{21} X_1 + \delta_{22} X_2 = \delta_{02} = 0$$

SOLVE FOR FLEXIBILITY COEFFICIENTS (δ_{11}; δ_{12}; δ_{21} & δ_{22})

δ_{11} = RADIAL DISPLACEMENT OF SHELL AND RING DUE TO UNIT RADIAL LOAD (IE 1.0 LB/IN). BOTH ARE PLUS BECAUSE BOTH TEND TO CLOSE THE GAP.

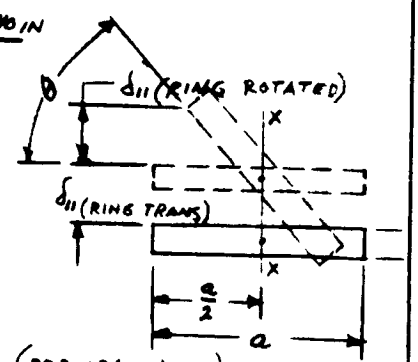
$$\delta_{11\,(\text{SHELL})} = \frac{r^2 \lambda}{2Et} = \frac{2500 \times .573}{2 \times 10^7 \times .10} = \underline{.000718 \text{ IN}}$$

$$\delta_{11\,(\text{RING TRANSLATED})} = \frac{r^2}{Et_1 a} = \frac{2500}{10^7 \times .218} = \underline{.001140 \text{ IN}}$$

$$\delta_{11\,(\text{RING ROTATED})} = \frac{a}{2}\theta = \frac{\frac{a}{2} M_1 r^2}{EI_x} = \frac{\frac{a}{2} \times \frac{a}{2} \times r^2}{E \times \frac{t_1 a^3}{12}}$$

$$= \frac{3r^2}{Et_1 a} = \frac{3 \times 2500}{2180000}$$

$$= \underline{.003420 \text{ IN}}$$



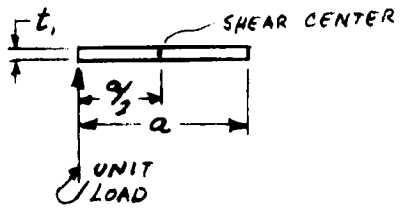
$$\delta_{11} = .000718 + .001140 + .003420 = \underline{\underline{.005278 \text{ IN}}}$$ (PER LB/IN LOAD)

RING STIFFENED SHELL

SOLVE FOR FLEXIBILITY COEFFICIENTS (CONT'D)

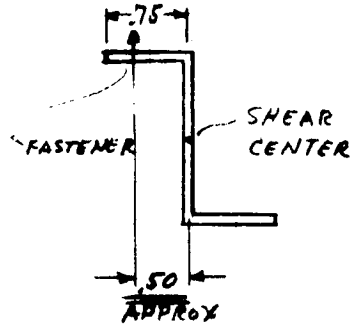
δ_{11} SOLUTION CONT'D: IT SHOULD BE NOTED HERE THAT THE UNIT RADIAL LOAD IS ASSUMED ACTING ON THE EDGE OF THE EQUIVALENT RING. THUS THE COEFFICIENT OF FLEXIBILITY IS A FUNCTION OF $a/2$ AS SHOWN. FOR THE ACTUAL CONDITION WHERE THE FASTENER WOULD BE IN TENSION THE MOMENT ARM WILL VARY WITH THE LOCATION OF THE FASTENER. THE SKETCHES ILLUSTRATE.

IDEALIZED EQUIVALENT RING



$a = .91$ IN
$a/2 = .455$ IN

ACTUAL CONDITION



IN THIS CASE THE COEFFICIENT WOULD BE SLIGHTLY CONSERVATIVE IN THAT THE ACTUAL CASE, WITH A LARGER MOMENT ARM, WOULD BE MORE FLEXIBLE.

AS THIS DISCUSSION INDICATES THE ANALYSIS HEREIN IS APPLICABLE ONLY FOR Z-SECTIONS AND ONLY FOR CASES WHERE THE SHELL AND RING TEND TO PULL APART UNDER LOAD. FOR HAT SECTIONS OR OTHER SYMMETRIC CROSS-SECTIONS WHERE THE SHEAR CENTER IS SYMMETRIC TO THE LOADS NO TWIST OR FLEXIBILITY OF RING DUE TO END LOAD WOULD EXIST. THUS A STIFFER STRUCTURE EXISTS AND LOADS WILL BE HIGHER.

RING STIFFENED SHELL

SOLVE FOR FLEXIBILITY COEFFICIENTS (CONT'D)

δ_{22} = SLOPE OF SHELL AND RING DUE TO UNIT MOMENT LOAD (I.E. IN LB/IN)

$$\delta_{22(SHELL)} = \frac{M_0 \lambda^3}{k}\left[\psi(\lambda x)\right]$$ (REF TIMOSHENKO II PG 8)

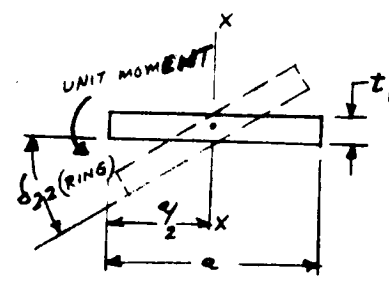
FOR UNIT LOAD AND δ AT LOAD

$$\delta_{22(SHELL)} = \frac{\lambda^3}{k} \quad ; \; k = \frac{Et}{r^2}$$ (REF TIMOSHENKO II PG 166)

$$\therefore \delta_{22(SHELL)} = \frac{\lambda^3}{1} \times \frac{r^2}{Et} = \frac{.188 \times 2500}{10^7 \times .1} = \underline{.000470}$$

$$\delta_{22(RING)} = \frac{r^2}{EI_x} = \frac{r^2}{E} \times \frac{12}{t_1 a^3}$$ (REF TIMOSHENKO II PG 178)

$$= \frac{12 \times 2500}{10^7 \times .24 \times .91^3} = \underline{.016000}$$



$$\delta_{22} = .000470 + .016000 = \underline{\underline{.017070}} \text{ (PER IN LB/IN)}$$

RING STIFFENED SHELL

SOLVE FOR FLEXIBILITY COEFFICIENTS (CONT'D)

δ_{12} = RADIAL DISPLACEMENT OF RING DUE TO UNIT MOMENT (i.e. IN-LB/IN). LOAD AS SHOWN OPENS GAP AND IS TAKEN AS NEGATIVE. NOTE THAT NO RADIAL DISPLACEMENT OF SHELL DUE TO MOMENT OCCURS SINCE ONLY A RADIAL LOAD IS APPLIED.

$$\delta_{12} = -\left(\theta \times \frac{a}{2}\right) = -\left(\frac{h^2}{EI_x} \times \frac{a}{2}\right) = -\left(\frac{h^2}{E} \times \frac{12}{t_1 a^3} \times \frac{a}{2}\right)$$ (REF TIMOSHENKO II PG 178)

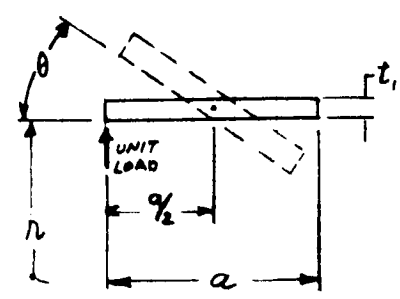
$$= -\frac{6h^2}{Et_1 a^2} = \frac{-6 \times 2500}{10^7 \times .24 \times .91^2} = \underline{\underline{-.007500 \text{ IN}}}$$ (PER IN-LB/IN)

δ_{21} = SLOPE OF RING DUE TO UNIT RADIAL LOAD (i.e. LB/IN). LOAD AS SHOWN PRODUCES SLOPE OPPOSITE TO THAT PRODUCED BY UNIT MOMENTS AND IS TAKEN AS NEGATIVE. NOTE THAT NO SLOPE OF SHELL DUE TO RADIAL LOAD OCCURS.

$$\delta_{21} = -\theta = \frac{-M_h \times h}{EI_x} = \frac{-\frac{a}{2}h^2}{E\frac{t_1 a^3}{12}} = \frac{-6h^2}{Et_1 a^2}$$ REF TIMOSHENKO II PG 178

(WHERE M = UNIT LOAD $\times \frac{a}{2}$)

$$\delta_{21} = \frac{-6h^2}{Et_1 a^2} = \underline{\underline{-.007500}}$$ (PER LB/IN)



RING STIFFENED SHELL

SOLVE FOR RADIAL LOAD X_1 AND MOMENT X_2
(REF PGS 2 & 3).

DEFL.
$$\left(\delta_{11} X_1\right) + \left(\delta_{12} X_2\right) = \delta_{01} = .0973 \text{ IN}$$
$$.53\, X_1 - .75\, X_2 = 9.73$$
$$X_1 = \frac{9.73 + .75 X_2}{.53} = 18.40 + 1.41\, X_2$$

SLOPE
$$\left(\delta_{21} X_1\right) + \left(\delta_{22} X_2\right) = \delta_{02} = 0$$
$$-.75 X_1 + 1.71\, X_2 = 0$$

$$-.75\left(18.40 + 1.41\, X_2\right) + 1.71\, X_2 = 0$$

$$-13.80 - 1.06 X_2 + 1.71\, X_2 = 0$$

$$.65\, X_2 = 13.80$$
$$X_2 = \underline{\underline{21.20}} \text{ IN LB/IN}$$

$$X_1 = 18.40 + \left(1.41 \times 21.20\right)$$

$$= 18.40 + 30.00 = \underline{\underline{48.40}} \text{ LB/IN}$$

RING STIFFENED SHELL

REDUCE THE EQUATIONS FOR DETERMINING RADIAL LOAD (X_1) AND MOMENT (X_2) TO A FUNCTION OF THE THERMAL GRADIENT (ΔT) : (REF PG 7)

$$X_1 = \left(\frac{\alpha \Omega \Delta T \times 10^2}{.53}\right) + \left(\frac{.75 X_2}{.53}\right) = \frac{.065 \Delta T}{.53} + 1.41 X_2$$

$$= (.123 \Delta T) + (1.41 X_2)$$

FROM THE SLOPE EQUATION

$$-.75 X_1 + 1.71 X_2 = 0$$

$$-.75(.123 \Delta T + 1.41 X_2) + 1.71 X_2 = 0$$

$$-.092 \Delta T - 1.06 X_2 + 1.71 X_2 = 0$$

$$.65 X_2 = .092 \Delta T$$

$$X_2 = \frac{.092}{.650} \Delta T = \underline{\underline{.142 \Delta T}} \text{ IN LB/IN}$$

$$X_1 = .123 \Delta T + (1.41 \times .142 \Delta T)$$

$$= .123 \Delta T + .200 \Delta T = \underline{\underline{.323 \Delta T}} \text{ LB/IN}$$

RING STIFFENED SHELL

SOLVE FOR STRESSES IN TERMS OF ΔT

MERIDIONAL STRESSES:

DUE TO MOMENT (X_2)

$$M = \frac{X_2}{2} = \frac{.142\,\Delta T}{2} = \underline{.071\,\Delta T} \text{ IN LB/IN}$$ (REF TIMOSHENKO II PG 8)

DUE TO RADIAL LOAD (X_1)

$$M = \frac{X_1}{4\lambda} = \frac{.323\,\Delta T}{4 \times .573} = .141\,\Delta T \text{ IN LB/IN}$$ (REF ROARK PG 270 CASE 8)

TOTAL $M = .071\Delta T + .141\,\Delta T$
$= .212\,\Delta T$ IN LB/IN

$$S_1' = \text{BENDING STRESS} = \frac{6M}{t^2} = \frac{6}{.01} \times .212\,\Delta T$$

$$= 600 \times .212\,\Delta T = \underline{\underline{127.2\,\Delta T}} \text{ PSI}$$

HOOP STRESSES:

$$S_2 = \frac{X_1 r \lambda}{2t} = \frac{.323\,\Delta T \times 50 \times .573}{2 \times .1}$$

$$= \underline{\underline{46.3\,\Delta T}} \text{ PSI}$$

ΔT (°F)	S_1' PSI	S_2 PSI
50	6360	2315
100	12720	4630
150	19080	6945

RING STIFFENED SHELL - SOLUTION FOR VARIOUS SHELL RADII AND SHELL THICKNESSES FOR GIVEN RING

EQ 1 $$\left[(X_1)\left(\frac{r^2\lambda}{2Et}+\frac{4r^2}{Et_1a}\right)\right]-\left[(X_2)\left(\frac{6r^2}{Et_1a^2}\right)\right]=\alpha r\Delta T$$

EQ 2 $$-\left[(X_1)\left(\frac{6r^2}{Et_1a^2}\right)\right]+\left[(X_2)\left(\frac{r^2\lambda^3}{Et}+\frac{12r^2}{Et_1a^3}\right)\right]=0$$

ASSUME: $E=10.0\times10^6$ PSI ; $\alpha=13.0\times10^{-6}$ IN/IN/°F (ALUMINUM ALLOY)

$\left.\begin{matrix} t_1=.25 \text{ IN} \\ a=1.00 \text{ IN}\end{matrix}\right\}$ EXAMPLE PROBLEM $t_1=.24$; $a=.91$

$\nu=.3 \therefore \lambda=\frac{1.28}{\sqrt{rt}}$

THEN:

EQ 1 $$\left[(X_1)\left(\frac{r^2\lambda}{20t}+1.60r^2\right)\right]-\left[(X_2)(2.40r^2)\right]=13r\Delta T$$

EQ 2 $$-\left[(X_1)(2.40r^2)\right]+\left[(X_2)\left(\frac{r^2\lambda^3}{10t}+4.80r^2\right)\right]=0$$

r	t	r/t	rt	$\sqrt{rt}$	$\lambda=\frac{1.28}{\sqrt{rt}}$	λ/t	λ^3	λ^3/t
25	.01	2500	.25	.500	2.560	256.00	16.777	1677.70
|	.02	1250	.50	.707	1.810	90.50	5.930	296.50
|	.03	833	.75	.866	1.478	49.27	3.228	107.60
|	.05	500	1.25	1.118	1.145	22.90	1.501	30.02
|	.10	250	2.50	1.581	.810	8.10	.531	5.31
25	.15	167	3.75	1.936	.661	4.41	.289	1.93
50	.01	5000	.50	.707	1.810	181.00	5.930	593.00
|	.02	2500	1.00	1.000	1.280	64.00	2.097	104.85
|	.03	1667	1.50	1.225	1.045	34.83	1.141	38.03
|	.05	1000	2.50	1.581	.810	16.20	.531	10.62
|	.10	500	5.00	2.236	.572	5.72	.187	1.87
50	.15	333	7.50	2.739	.467	3.11	.102	.68
75	.01	7500	.75	.866	1.478	147.80	3.228	322.80
|	.02	3750	1.50	1.225	1.045	52.25	1.141	57.05
|	.03	2500	2.25	1.500	.853	28.43	.621	20.70
|	.05	1500	3.75	1.936	.661	13.22	.289	5.78
|	.10	750	7.50	2.739	.467	4.67	.102	1.02
75	.15	500	11.25	3.354	.382	2.55	.056	.37

FORM 1-8136 B (9-58)

RING STIFFENED SHELL

$L = 25$ IN

$t = .01$ IN

EQ 1 $\left[(X_1)\left(\frac{625 \times 2.56}{.2} + 1000\right)\right] - 1500 X_2 = 325 \Delta T$

$9000 X_1 - 1500 X_2 = 325 \Delta T$; $X_1 = \frac{325 \Delta T + 1500 X_2}{9000} = .036 \Delta T + .167 X_2$

EQ 2 $-1500 X_1 + \left[(X_2)\left(\frac{625 \times 16.8}{.1} + 3000\right)\right] = 0$

$-1500 X_1 + 108000 X_2 = 0$

$-1500(.036 \Delta T + .167 X_2) + 108000 X_2 = 0$

$107736 X_2 = 54 \Delta T$

$$\boxed{\begin{aligned} X_2 &= .0005 \Delta T \\ X_1 &= .0360 \Delta T \end{aligned}}$$

$t = .02$ IN

EQ 1 $\left[(X_1)\left(\frac{625 \times 1.81}{.4} + 1000\right)\right] - 1500 X_2 = 325 \Delta T$

$3830 X_1 - 1500 X_2 = 325 \Delta T$; $X_1 = \frac{325 \Delta T + 1500 X_2}{3830} = .085 \Delta T + .392 X_2$

EQ2 $-1500 X_1 + \left[(X_2)\left(\frac{625 \times 5.93}{.2} + 3000\right)\right] = 0$

$-1500 X_1 + 21500 X_2 = 0$

$-1500(.085 \Delta T + .392 X_2) + 21500 X_2 = 0$

$20910 X_2 = 127 \Delta T$

$$\boxed{\begin{aligned} X_2 &= .0061 \Delta T \\ X_1 &= .0874 \Delta T \end{aligned}}$$

RING STIFFENED SHELL

$R = 25$ IN

$\boxed{t = .03 \text{ IN}}$

EQ 1 $$(X_1)\left[\left(\frac{625 \times 1.478}{.6}\right) + (1.60 \times 625)\right] - \left[(X_2)(2.4 \times 625)\right] = 13 \times 25 \times \Delta T$$

$$2540\, X_1 - 1500\, X_2 = 325\,\Delta T \;;\quad X_1 = \frac{325\,\Delta T + 1500\, X_2}{2540} = .128\,\Delta T + .590\, X_2$$

EQ 2 $$-1500\, X_1 + \left[(X_2)\left(\frac{625 \times 3.228}{.3} + 4.80 \times 625\right)\right] = 0$$

$$-1500\, X_1 + 9720\, X_2 = 0$$

$$-1500\,(.128\,\Delta T + .590\, X_2) + 9720\, X_2 = 0$$

$$8835\, X_2 = 192\,\Delta T$$

$$\boxed{\begin{aligned} X_2 &= .022\,\Delta T \\ X_1 &= .141\,\Delta T \end{aligned}}$$

$\boxed{t = .05 \text{ IN}}$

EQ 1 $$(X_1)\left[\frac{625 \times 1.145}{1.0} + 1000\right] - \left[(X_2)(1500)\right] = 325\,\Delta T$$

$$1715\, X_1 - 1500\, X_2 = 325\,\Delta T \;;\quad X_1 = \frac{325\,\Delta T + 1500\, X_2}{1715} = .190\,\Delta T + .875\, X_2$$

EQ 2 $$-1500\, X_1 + \left[(X_2)\left(\frac{625 \times 1.501}{.5} + 3000\right)\right] = 0$$

$$-1500\, X_1 + 4880\, X_2 = 0$$

$$-1500\,(.190\,\Delta T + .875\, X_2) + 4880\, X_2 = 0$$

$$3570\, X_2 = 285\,\Delta T$$

$$\boxed{\begin{aligned} X_2 &= .080\,\Delta T \\ X_1 &= .260\,\Delta T \end{aligned}}$$

RING STIFFENED SHELL

$R = 25$ IN (CONT'D)

$t = .10$ IN

EQ 1 $$(X_1)\left[\frac{625 \times .810}{2.0} + 1000\right] - 1500 X_2 = 325 \Delta T$$

$$1253 X_1 - 1500 X_2 = 325 \Delta T \; ; \; X_1 = \frac{325 \Delta T + 1500 X_2}{1253} = .260 \Delta T + 1.200 X_2$$

EQ 2 $$-1500 X_1 + \left[(X_2)\left(\frac{625 \times .531}{1.0} + 3000\right)\right] = 0$$

$$-1500 X_1 + 3332 X_2 = 0$$

$$-1500(.260 \Delta T + 1.200 X_2) + 3332 X_2 = 0$$

$$1532 X_2 = 390 \Delta T$$

$$\boxed{\begin{aligned} X_2 &= .255 \Delta T \\ X_1 &= .566 \Delta T \end{aligned}}$$

$t = .15$ IN

EQ 1 $$(X_1)\left[\frac{625 \times .661}{3.0} + 1000\right] - 1500 X_2 = 325 \Delta T$$

$$1138 X_1 - 1500 X_2 = 325 \Delta T \; ; \; X_1 = \frac{325 \Delta T + 1500 X_2}{1138} = .286 \Delta T + 1.320 X_2$$

EQ 2 $$-1500 X_1 + \left[(X_2)\left(\frac{625 \times .289}{1.5} + 3000\right)\right] = 0$$

$$-1500 X_1 + 3120 X_2 = 0$$

$$-1500(.286 \Delta T + 1.320 X_2) + 3120 X_2 = 0$$

$$1140 X_2 = 430 \Delta T$$

$$\boxed{\begin{aligned} X_2 &= .377 \Delta T \\ X_1 &= .782 \Delta T \end{aligned}}$$

RING STIFFENED SHELL

$R = 50$ IN

$t = .01$ IN

EQ 1 $\left[(X_1)\left(\frac{2500 \times 1.81}{.2} + 4000\right)\right] - 6000\, X_2 = 650\, \Delta T$

$$26600\, X_1 - 6000 X_2 = 650 \Delta T \; ; \; X_1 = \frac{650\, \Delta T + 6000\, X_2}{26600} = .024 \Delta T + .225\, X_2$$

EQ 2 $-6000\, X_1 + \left[(X_2)\left(\frac{2500 \times 5.93}{.1} + 12000\right)\right] = 0$

$$-6000\, X_1 + 160000\, X_2 = 0$$

$$-6000(.024\, \Delta T + .225 X_2) + 160\,000\, X_2 = 0$$

$$159000\, X_2 = 144\, \Delta T$$

$$\boxed{\begin{aligned} X_2 &= .0009\, \Delta T \\ X_1 &= .0242\, \Delta T \end{aligned}}$$

$t = .02$ IN

EQ 1 $\left[(X_1)\left(\frac{2500 \times 1.28}{.4} + 4000\right)\right] - 6000\, X_2 = 650\, \Delta T$

$$12000\, X_1 - 6000 X_2 = 650 \Delta T \; ; \; X_1 = \frac{650 \Delta T + 6000 X_2}{12000} = .054 \Delta T + .500\, X_2$$

EQ 2 $-6000\, X_1 + \left[(X_2)\left(\frac{2500 \times 2.097}{.2} + 12000\right)\right] = 0$

$$-6000\, X_1 + 38200\, X_2 = 0$$

$$-6000(.054\, \Delta T + .500 X_2) + 38200\, X_2 = 0$$

$$35200\, X_2 = 324\, \Delta T$$

$$\boxed{\begin{aligned} X_2 &= .0092\, \Delta T \\ X_1 &= .0586\, \Delta T \end{aligned}}$$

RING STIFFENED SHELL

$L = 50$ IN

$t = .03$ IN

EQ 1 $$\left(X_1\right)\left[\left(\frac{2500 \times 1.045}{20 \times .03}\right) + \left(1.60 \times 2500\right)\right] - \left(X_2\right) \times \left(2.4 \times 2500\right) = 650\,\Delta T$$

$$8350\,X_1 - 6000\,X_2 = 650\,\Delta T \quad ; \quad X_1 = \frac{650\,\Delta T + 6000\,X_2}{8350} = .078\,\Delta T + .718\,X_2$$

EQ 2 $$-6000\,X_1 + \left[\left(X_2\right)\left(\frac{2500 \times 1.141}{.3} + 12000\right)\right] = 0$$

$$-6000\,X_1 + 21530\,X_2 = 0$$

$$-6000\left(.078\,\Delta T + .718\,X_2\right) + 21530\,X_2 = 0$$

$$21530\,X_2 - 4308\,X_2 = 468\,\Delta T$$

$$\boxed{\begin{aligned} X_2 &= .0271\,\Delta T \\ X_1 &= .0975\,\Delta T \end{aligned}}$$

$t = .05$ IN

EQ 1 $$\left(X_1\right)\left[\left(\frac{2500 \times .810}{20 \times .05}\right) + \left(1.60 \times 2500\right)\right] - \left(X_2\right)\left(2.4 \times 2500\right) = 650\,\Delta T$$

$$6030\,X_1 - 6000\,X_2 = 650\,\Delta T \quad ; \quad X_1 = \frac{650\,\Delta T + 6000\,X_2}{6030} = .108\,\Delta T + 1.000\,X_2$$

EQ 2 $$-6000\,X_1 + \left[\left(X_2\right)\left(\frac{2500 \times .531}{.5} + 12000\right)\right] = 0$$

$$-6000\,X_1 + 14660\,X_2 = 0$$

$$-6000\left(.108\,\Delta T + 1.00\,X_2\right) + 14660\,X_2 = 0$$

$$8660\,X_2 = 648\,\Delta T$$

$$\boxed{\begin{aligned} X_2 &= .075\,\Delta T \\ X_1 &= .183\,\Delta T \end{aligned}}$$

RING STIFFENED SHELL

$R = 50$ IN (CONT'D)

$t = .10$ IN

EQ 1 $\quad (X_1)\left[\left(\frac{2500 \times .572}{2}\right) + (4000)\right] - 6000 X_2 = 650 \Delta T$

$$4717 X_1 - 6000 X_2 = 650 \Delta T \quad ; \quad X_1 = \frac{650 \Delta T + 6000 X_2}{4717} = .138 \Delta T + 1.275 X_2$$

EQ 2 $\quad -6000 X_1 + \left[(X_2)\left(\frac{2500 \times .187}{1} + 12000\right)\right] = 0$

$$-6000 X_1 + 12467 X_2 = 0$$

$$-6000(.138 \Delta T + 1.275 X_2) + 12467 X_2 = 0$$

$$4817 X_2 = 828 \Delta T$$

$$\boxed{\begin{aligned} X_2 &= .172 \Delta T \\ X_1 &= .357 \Delta T \end{aligned}}$$

$t = .15$ IN

EQ 1 $\quad (X_1)\left[\left(\frac{2500 \times .467}{3}\right) + (4000)\right] - 6000 X_2 = 650 \Delta T$

$$4390 X_1 - 6000 X_2 = 650 \Delta T \; ; \; X_1 = \frac{650 \Delta T + 6000 X_2}{4390} = .148 \Delta T + 1.370 X_2$$

EQ 2 $\quad -6000 X_1 + \left[(X_2)\left(\frac{2500 \times .102}{1.5} + 12000\right)\right] = 0$

$$-6000 X_1 + 12170 X_2 = 0$$

$$-6000(.148 \Delta T + 1.370 X_2) + 12170 X_2 = 0$$

$$3950 X_2 = 890 \Delta T$$

$$\boxed{\begin{aligned} X_2 &= .225 \Delta T \\ X_1 &= .457 \Delta T \end{aligned}}$$

RING STIFFENED SHELL

$r = 75$ IN

$t = .01$ IN

EQ 1 $\left[(X_1)\left(\frac{5625 \times 1.48}{.2} + 9000\right)\right] - 13500\, X_2 = 975\, \Delta T$

$50700\, X_1 - 13500\, X_2 = 975\, \Delta T$; $X_1 = \frac{975 \Delta T + 13500\, X_2}{50700} = .0192\, \Delta T + .265\, X_2$

EQ 2 $-13500\, X_1 + \left[(X_2)\left(\frac{5625 \times 3.228}{.1} + 27000\right)\right] = 0$

$-13500\, X_1 + 209000\, X_2 = 0$

$-13500\,(.0192\, \Delta T + .265\, X_2) + 209000\, X_2 = 0$

$205400\, X_2 = 260\, \Delta T$

$$\boxed{\begin{aligned} X_2 &= .00127\, \Delta T \\ X_1 &= .0195\, \Delta T \end{aligned}}$$

$t = .02$ IN

EQ 1 $\left[(X_1)\left(\frac{5625 \times 1.045}{.4} + 9000\right)\right] - 13500\, X_2 = 975\, \Delta T$

$23700\, X_1 - 13500\, X_2 = 975\, \Delta T$; $X_1 = \frac{975\, \Delta T + 13500\, X_2}{23700} = .0412\, \Delta T + .57\, X_2$

EQ 2 $-13500\, X_1 + \left[(X_2)\left(\frac{5625 \times 1.141}{.2} + 27000\right)\right] = 0$

$-13500\, X_1 + 59100\, X_2 = 0$

$-13500\,(.0412\, \Delta T + .57\, X_2) + 59100\, X_2 = 0$

$51400\, X_2 = 555\, \Delta T$

$$\boxed{\begin{aligned} X_2 &= .0108\, \Delta T \\ X_1 &= .0473\, \Delta T \end{aligned}}$$

RING STIFFENED SHELL

$r = 75$ IN

$t = .03$ IN

EQ 1 $\quad X_1\left[\left(\frac{5625 \times .853}{.60} + 9000\right)\right] - \left[(X_2)(13500)\right] = 975\,\Delta T$

$$17000\,X_1 - 13500\,X_2 = 975\,\Delta T; \quad X_1 = \frac{975\,\Delta T + 13500\,X_2}{17000} = .057\,\Delta T + .795\,X_2$$

EQ 2 $\quad -13500\,X_1 + \left[(X_2)\left(\frac{5625 \times .621}{.3} + 27000\right)\right] = 0$

$$-13500\,X_1 + 38600\,X_2 = 0$$

$$-13500(.057\,\Delta T + .795\,X_2) + 38600\,X_2 = 0$$

$$27900\,X_2 = 770\,\Delta T$$

$$\boxed{\begin{aligned} X_2 &= .028\,\Delta T \\ X_1 &= .079\,\Delta T \end{aligned}}$$

$t = .05$ IN

EQ 1 $\quad X_1\left[\left(\frac{5625 \times .661}{1.0} + 9000\right)\right] - \left[13500\,X_2\right] = 975\,\Delta T$

$$12720\,X_1 - 13500\,X_2 = 975\,\Delta T; \quad X_1 = \frac{975\,\Delta T + 13500\,X_2}{12720} = .077\,\Delta T + 1.060\,X_2$$

EQ 2 $\quad -13500\,X_1 + \left[(X_2)\left(\frac{5625 \times .289}{.5} + 27000\right)\right] = 0$

$$-13500\,X_1 + 30250\,X_2 = 0$$

$$-13500(.077\,\Delta T + 1.060\,X_2) + 30250\,X_2 = 0$$

$$15950\,X_2 = 1040\,\Delta T$$

$$\boxed{\begin{aligned} X_2 &= .065\,\Delta T \\ X_1 &= .146\,\Delta T \end{aligned}}$$

RING STIFFENED SHELL

$\Delta = 75$ IN

$t = .10$ IN

EQ 1 $\quad X_1\left[\left(\frac{5625 \times .467}{2}\right) + 9000\right] - 13500 X_2 = 975 \Delta T$

$$10315 X_1 - 13500 X_2 = 975 \Delta T \; ; \; X_1 = \frac{975 \Delta T + 13500 X_2}{10315} = .095 \Delta T + 1.310 X_2$$

EQ 2 $\quad -13500 X_1 + \left[(X_2)\left(\frac{5625 \times .182}{1} + 27000\right)\right] = 0$

$$-13500 X_1 + 27570 X_2 = 0$$
$$-13500(.095 \Delta T + 1.310 X_2) + 27570 X_2 = 0$$
$$9870 X_2 = 1260 \Delta T$$

$$\boxed{\begin{array}{l} X_2 = .128 \Delta T \\ X_1 = .263 \Delta T \end{array}}$$

$t = .15$ IN

EQ 1 $\quad X_1\left[\left(\frac{5625 \times .382}{3.0} + 9000\right)\right] - 13500 X_2 = 975 \Delta T$

$$9715 X_1 - 13500 X_2 = 975 \Delta T \; ; \; X_1 = \frac{975 \Delta T + 13500 X_2}{9715} = .100 \Delta T + 1.390 X_2$$

EQ 2 $\quad -13500 X_1 + \left[(X_2)\left(\frac{5625 \times .056}{1.5} + 27000\right)\right] = 0$

$$-13500 X_1 + 27210 X_2 = 0$$

$$-13500(.100 \Delta T + 1.390 X_2) + 27210 X_2 = 0$$
$$8410 X_2 = 1350 \Delta T$$

$$\boxed{\begin{array}{l} X_2 = .160 \Delta T \\ X_1 = .322 \Delta T \end{array}}$$

RING STIFFENED SHELL

$S_1' =$ MERIDIONAL BENDING STRESS $= \frac{6}{t^2}\left(\frac{X_2}{2} + \frac{X_1}{4\lambda}\right)$

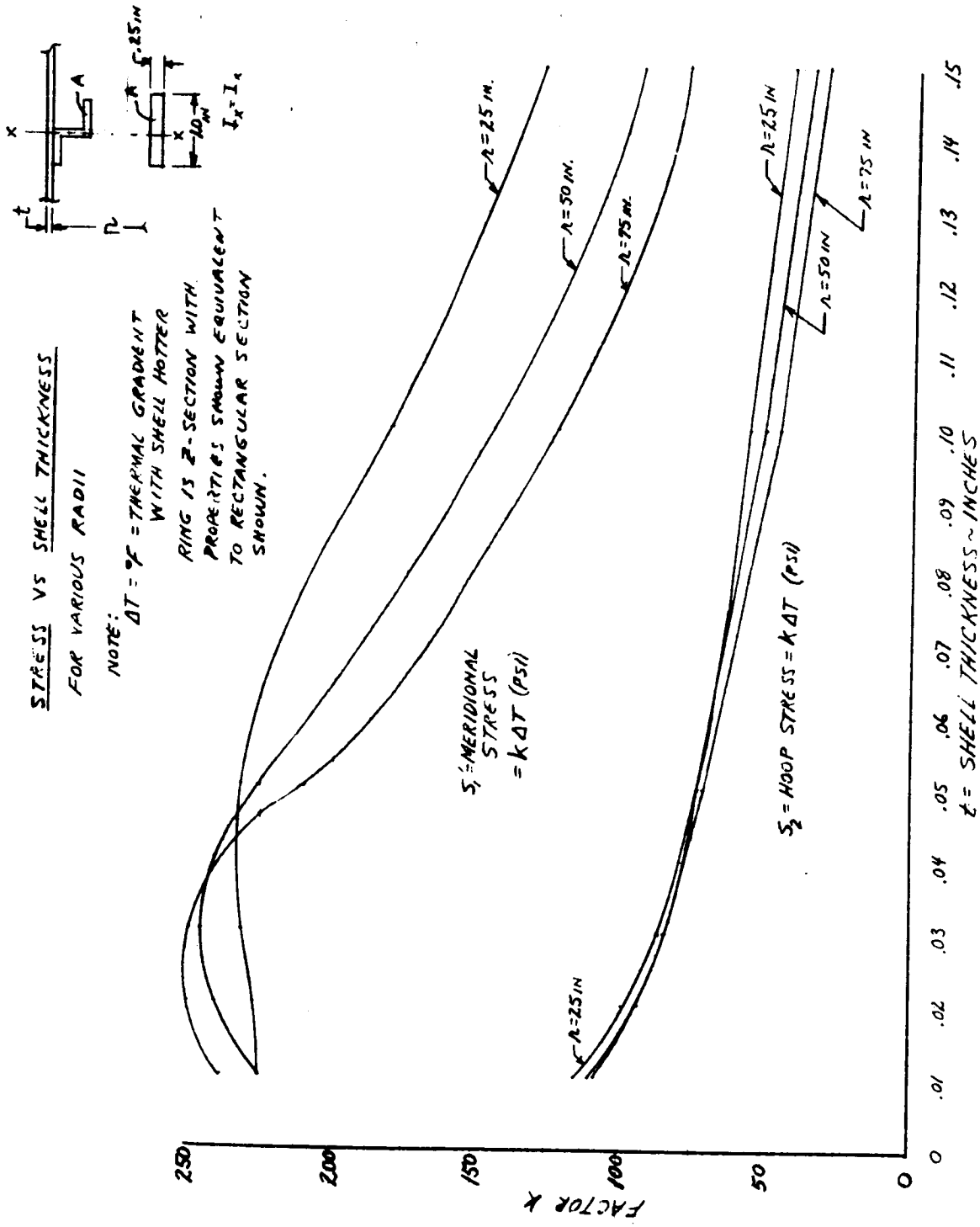
$S_2 =$ HOOP STRESS $= \frac{r\lambda}{2t}(X_1)$

①	②	③	④	⑤	⑥	⑦	⑧	⑨	⑩	⑪	⑫
t	t^2	X_1	X_2	λ	$r\lambda$	4λ	$6/t^2$	$X_2/2$	$X_1/4\lambda$	$S_1' = ⑧[⑨+⑩]$	$S_2 = ③ \times \frac{⑥}{2\times①}$
$r = 25$ IN											
.01	.0001	.0360	.0005	2.560	64.00	10.24	60000	.00025	.0035	225 ΔT	115 ΔT
.02	.0004	.0874	.0061	1.810	45.25	7.24	15000	.00305	.0121	227 ΔT	99 ΔT
.03	.0009	.141	.022	1.478	36.95	5.912	6667	.0110	.0238	232 ΔT	87 ΔT
.05	.0025	.260	.080	1.145	28.62	4.580	2400	.0400	.0568	232 ΔT	74 ΔT
.10	.0100	.566	.255	.810	20.25	3.240	600	.1275	.1747	181 ΔT	57 ΔT
.15	.0225	.782	.377	.661	16.52	2.644	267	.1885	.2958	129 ΔT	43 ΔT
$r = 50$ IN											
.01	.0001	.0242	.0009	1.810	90.50	7.24	60000	.00045	.0033	225 ΔT	110 ΔT
.02	.0004	.0586	.0092	1.280	64.00	5.12	15000	.00460	.0114	240 ΔT	94 ΔT
.03	.0009	.0975	.0271	1.045	52.25	4.180	6667	.0136	.0233	246 ΔT	85 ΔT
.05	.0025	.1830	.0750	.810	40.50	3.240	2400	.0375	.0565	226 ΔT	74 ΔT
.10	.0100	.3570	.1720	.572	28.60	2.288	600	.0860	.1560	145 ΔT	51 ΔT
.15	.0225	.4570	.2250	.467	23.35	1.868	267	.1125	.2446	95 ΔT	36 ΔT
$r = 75$ IN											
.01	.0001	.0195	.0013	1.478	110.85	5.912	60000	.00065	.0033	237 ΔT	108 ΔT
.02	.0004	.0473	.0108	1.045	78.38	4.180	15000	.00540	.0113	250 ΔT	93 ΔT
.03	.0009	.079	.028	.853	63.98	3.412	6667	.0140	.0232	248 ΔT	84 ΔT
.05	.0025	.146	.065	.661	49.58	2.644	2400	.0325	.0552	210 ΔT	72 ΔT
.10	.0100	.263	.128	.467	35.02	1.868	600	.0640	.1408	123 ΔT	46 ΔT
.15	.0225	.322	.160	.382	28.65	1.528	267	.0800	.2107	78 ΔT	31 ΔT

FORM 1-8136 B (9-58)

THERMALLY INDUCED DISCONTINUITY STRESSES FOR RING STIFFENED SHELL

P. KUZMIK
3-13-64



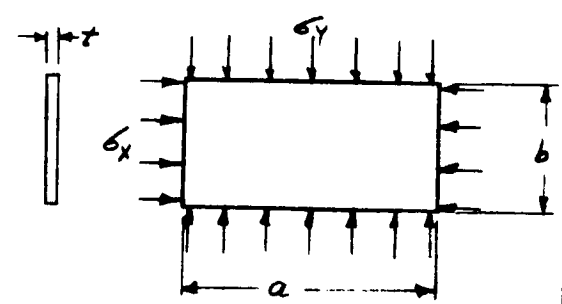
APPENDIX F

PLATES AND PANELS

BY P. KUZMIK CK. DATE 3-19-64 REV.	GENERAL ELECTRIC	PAGE 1 MODEL REPORT STERILIZATION

FLAT PLATE BUCKLING IN A THERMAL ENVIRONMENT

THIS STUDY IS CONCERNED WITH THE EFFECTS OF A THERMAL ENVIRONMENT ON A FLAT PLATE WHICH IS ASSUMED TO BE RESTRICTED FROM EXPANDING IN TERMS OF ΔT. THIS ASSUMES THAT THE RESTRAINING STRUCTURE IS INFINITELY RIGID IN THE PLANE OF THE PLATE AND EXPANDS, DUE TO TEMPERATURE, AT THE SAME RATE AS THE PLATE DOES. THUS FORCES ON THE PLATE ARE A FUNCTION OF THE TEMPERATURE DIFFERENCE BETWEEN PLATE AND STRUCTURE (ie ΔT). THE RESULTS HEREIN (ie ΔT_{CR}) INDICATE ONLY WHEN INITIAL BUCKLING WILL OCCUR. THE LIMITING ΔT WILL VARY AS THE EDGE CONDITIONS VARY. DATA IS GIVEN FOR SIMPLY SUPPORTED AND FIXED EDGES.



$\sigma_X = \sigma_Y$
= APPLIED STRESS $= \frac{E\alpha}{1-\nu}\Delta T$

THE MATERIAL ANALYZED IN THIS REPORT IS 2024-T3 ALUM. PROPERTIES ARE TAKEN AT THE TOP OF THE STERILIZATION RANGE OR 300°F.

$E = 10^7$ PSI
$\alpha = 1.30 \times 10^{-6}$ IN/IN/°F
$\nu = .3$

$$\therefore \sigma_X = \sigma_Y = \frac{10^7 \times 1.3 \times 10^{-7}}{.7}\Delta T = \underline{\underline{186\,\Delta T}}$$

THE ANALYSIS THAT FOLLOWS SOLVES FOR THE LOADS IN ONE DIRECTION AT A TIME AND THEN COMBINES BOTH THROUGH THE USE OF INTERACTION CURVES.

FORM 1-8136 B (9-58)

FLAT PLATE BUCKLING IN A THERMAL ENVIRONMENT

THE EQUATION FOR CRITICAL BUCKLING STRESS IS

$$\sigma_{CR} = k_c\left(\frac{\pi^2 E \eta}{12(1-\nu^2)}\right)\left(\frac{t}{b}\right)^2$$ (REF CONVAIR FORT WORTH SECT 6)

WHERE:

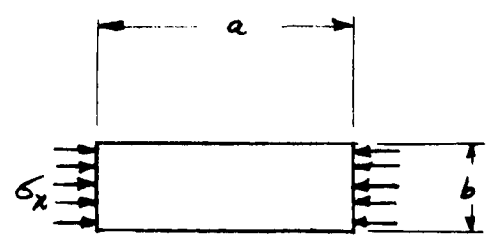
k_c = NONDIMENSIONAL BUCKLING COEFFICIENT
b = SHORTER EDGE (IN)
ν = POISSONS RATIO = .3
t = PLATE THICKNESS (IN)
E = MODULUS OF ELASTICITY (COMP) = 10^7 PSI
η = PLASTICITY COEFFICIENT = 1.0 (FOR ELASTIC RANGE)

$$\therefore \sigma_{CR} = k_c\left(\frac{t}{b}\right)^2(9.04 \times 10^6)$$

SIMPLY SUPPORTED EDGES

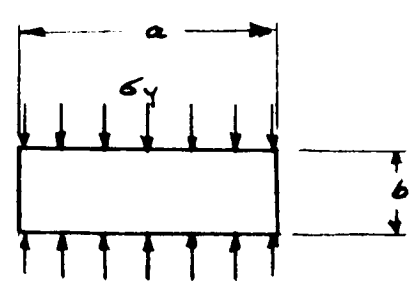
FOR LOAD IN "X" DIRECTION (i.e. σ_x)

a/b	k_c	σ_{CR-X}
1.0	4.00	$36.16 \times 10^6 (t/b)^2$
1.25	4.20	37.97
1.50	4.30	38.87
1.75	4.05	36.61
2.00	4.00	36.16
2.45	4.15	$37.52 \times 10 (t/b)^2$



FOR LOAD IN "Y" DIRECTION (i.e. σ_Y)

a/b	b/a	k_c	σ_{CR-Y}
1.00	1.00	4.00	$36.16 \times 10^6 (t/b)^2$
1.25	.80	2.65	23.96
1.50	.67	2.10	18.98
1.75	.57	1.75	15.82
2.00	.50	1.50	13.56
2.45	.41	1.35	$12.20 \times 10^6 (t/b)^2$



FLAT PLATE BUCKLING IN A THERMAL ENVIRONMENT

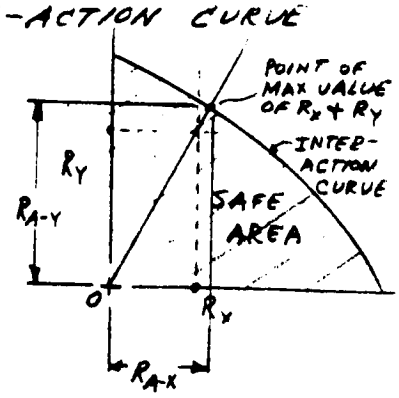
SIMPLY SUPPORTED EDGES

SOLVE FOR THE APPLIED STRESS RATIOS

$$R = \frac{\sigma}{\sigma_{CR}} = \frac{186\,\Delta T_{CR}}{\sigma_{CR}}$$

ASPECT RATIO a/b	σ_{CR-X}	R_X $\left(\times 10^6 (t/b)^2\right)$	σ_{CR-Y}	R_Y $\left[\times 10^6 (t/b)^2\right]$	R_X/R_Y *
1.00	$36.16 \times 10^6 (t/b)^2$	$5.14\ \Delta T$	$36.16 \times 10^6 (t/b)^2$	$5.14\ \Delta T$	1.00
1.25	37.97	$4.90\ \Delta T$	23.96	$7.76\ \Delta T$	.63
1.50	38.87	$4.78\ \Delta T$	18.98	$9.80\ \Delta T$	.49
1.75	36.61	$5.08\ \Delta T$	15.82	$11.76\ \Delta T$	.43
2.00	36.16	$5.14\ \Delta T$	13.56	$13.72\ \Delta T$	.37
2.45	$37.52 \times 10^6 (t/b)^2$	$4.96\ \Delta T$	$12.20 \times 10^6 (t/b)^2$	$15.24\ \Delta T$	.32

* $\frac{R_X}{R_Y}$ IS THE SLOPE OF THE LINE ON THE INTER-ACTION CURVE PLOT AND ORIGINATES AT $R_X = R_Y = 0$. WHERE THIS LINE INTERSECTS THE INTER-ACTION CURVE IS THE POINT THAT REPRESENTS THE LARGEST R_X & R_Y VALUE AND WILL IN TURN REFLECT THE LARGEST ΔT.



$$\frac{R_{A-X}}{R_X} - 1 = \text{MARGIN OF SAFETY} = 0$$

$$\therefore R_X = R_{A-X} \quad (\text{ALSO } R_Y = R_{A-Y})$$

FLAT PLATE BUCKLING IN A THERMAL ENVIRONMENT

SIMPLY SUPPORTED EDGES
SOLVE FOR ΔT_{CR} IN TERMS OF "t" & "b"

ASPECT RATIO a/b	R_X/R_Y	R_{A-X}	R_X $\left(\times 10^6 (t/b)^2\right)$	$R_{A-X} = R_X^*$ $\therefore \Delta T_{CR} =$
1.00	1.00	.500	$5.14\,\Delta T$	$.0973 \times 10^6 (t/b)^2$
1.25	.63	.405	$4.90\,\Delta T$	.0826
1.50	.49	.360	$4.78\,\Delta T$	.0753
1.75	.43	.330	$5.08\,\Delta T$	.0650
2.00	.37	.305	$5.14\,\Delta T$	.0593
2.45	.32	.280	$4.96\,\Delta T$	$.0564 \times 10^6 (t/b)^2$

*FROM PRECEDING PAGE

$$R_X = R_{A-X} \quad (\text{ALSO } R_Y = R_{A-Y})$$

FOR $a/b = 1.00$

$$\frac{5.14\,\Delta T_{CR}}{10^6 (t/b)^2} = .500 \quad ; \quad \Delta T_{CR} = \frac{.500}{5.14} \times 10^6 (t/b)^2$$

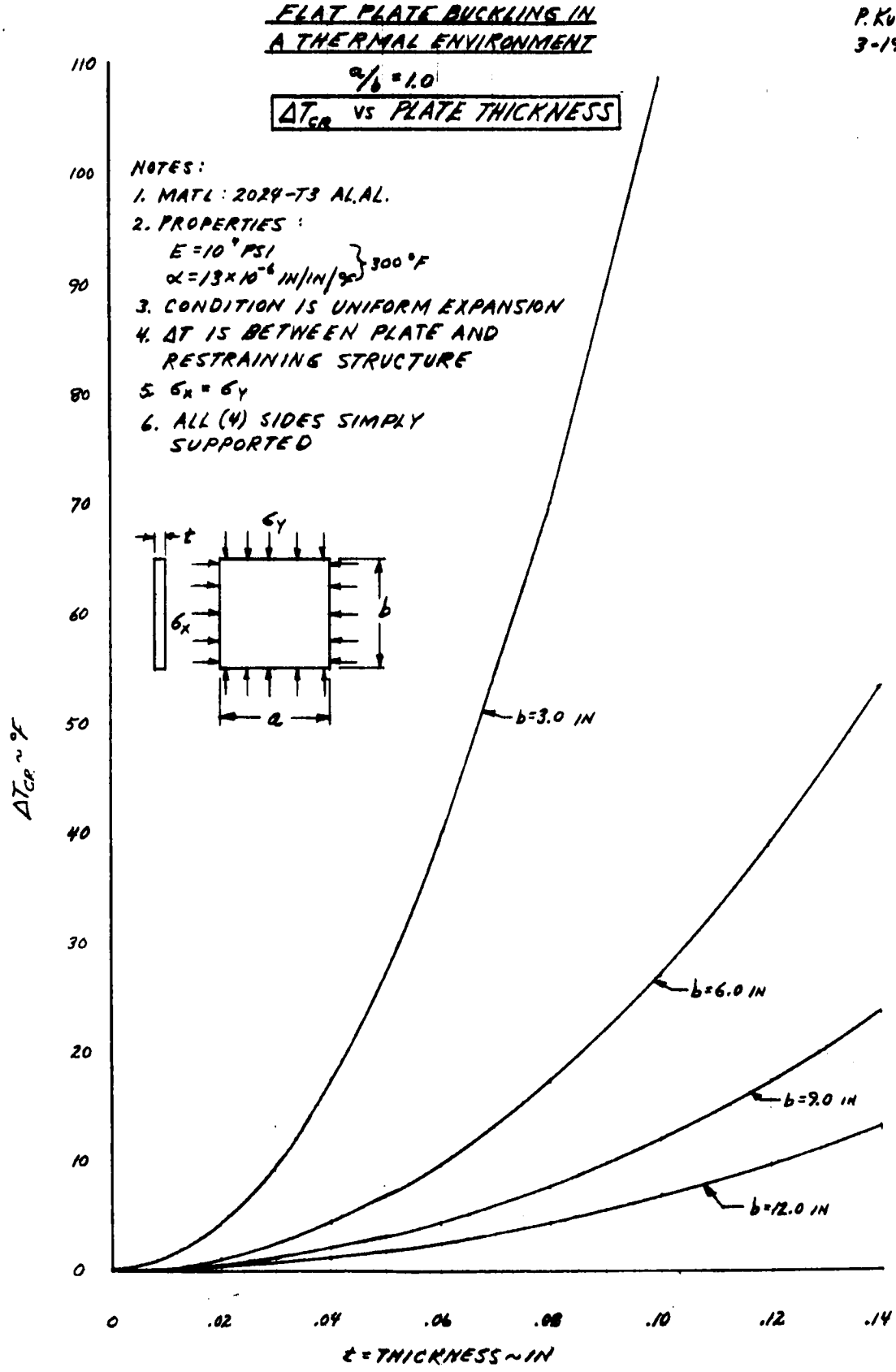
$$\therefore \Delta T_{CR} = .0973 \times 10^6 (t/b)^2$$

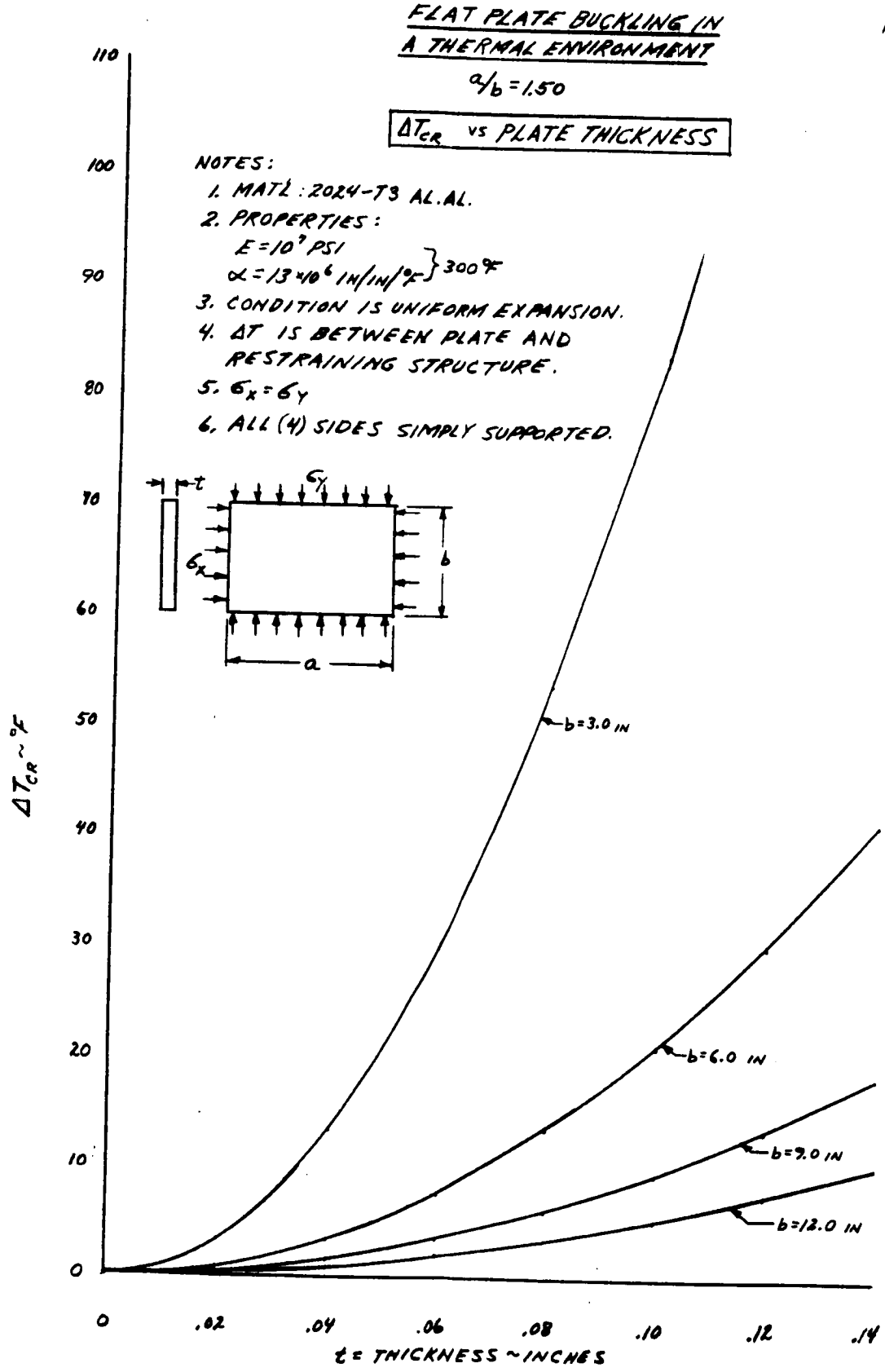
FLAT PLATE BUCKLING IN A THERMAL ENVIRONMENT

SOLVE FOR ΔT_{CR} (SIMPLY SUPPORTED EDGES)

t IN	b IN	$(t/b)^2 \times 10^6$	$^*\Delta T_{CR}$ (°F) a/b = 1.00 W = .0973	a/b = 1.50 W = .0753	a/b = 2.00 W = .0593
.02	3	44.45	4.3	3.3	2.6
.04	↑	177.77	17.3	13.4	10.5
.06		400.00	38.9	30.1	23.7
.08		711.13	69.2	53.5	42.2
.10		1111.09	108.1	83.7	65.9
.12	↓	1600.00	155.7	120.5	94.9
.14	3	2177.81	211.9	164.0	129.1
.02	6	11.11	1.1	.8	.7
.04	↑	44.45	4.3	3.3	2.6
.06		100.00	9.7	7.5	5.9
.08		177.77	17.3	13.4	10.5
.10		277.79	27.0	20.9	16.5
.12	↓	400.00	38.9	30.1	23.7
.14	6	544.43	53.0	41.0	32.3
.02	9	4.94	.5	.4	.3
.04	↑	19.75	1.9	1.5	1.2
.06		44.45	4.3	3.3	2.6
.08		79.01	7.7	5.9	4.7
.10		123.45	12.0	9.3	7.3
.12	↓	177.77	17.3	13.4	10.5
.14	9	241.99	23.5	18.2	14.4
.02	12	2.78	.3	.2	.2
.04	↑	11.11	1.1	.8	.7
.06		25.00	2.4	1.9	1.5
.08		44.45	4.3	3.3	2.6
.10		69.44	6.8	5.2	4.1
.12	↓	100.00	9.7	7.5	5.9
.14	12	136.12	13.2	10.2	8.1

$^*\Delta T_{CR} = W \times 10^6 (t/b)^2$


FLAT PLATE BUCKLING IN
A THERMAL ENVIRONMENT
a/b = 1.0
ΔT_CR VS PLATE THICKNESS
P. KUZMIK
3-19-64
NOTES:
1. MATL: 2024-T3 AL.AL.
2. PROPERTIES:
E = 10^9 PSI
α = 13×10^-6 IN/IN/°F
300 °F
3. CONDITION IS UNIFORM EXPANSION
4. ΔT IS BETWEEN PLATE AND RESTRAINING STRUCTURE
5. σx = σY
6. ALL (4) SIDES SIMPLY SUPPORTED
σY
t
σx
b
a
b=3.0 IN
b=6.0 IN
b=9.0 IN
b=12.0 IN
ΔT_CR ~ °F
t = THICKNESS ~ IN
0
10
20
30
40
50
60
70
80
90
100
110
.02
.04
.06
.08
.10
.12
.14


FLAT PLATE BUCKLING IN
A THERMAL ENVIRONMENT
a/b = 1.50
ΔT_CR vs PLATE THICKNESS
P. KUZMIK
3-19-64
NOTES:
1. MAT'L: 2024-T3 AL.AL.
2. PROPERTIES:
E = 10^7 PSI
α = 13×10^6 IN/IN/°F } 300°F
3. CONDITION IS UNIFORM EXPANSION.
4. ΔT IS BETWEEN PLATE AND RESTRAINING STRUCTURE.
5. σx = σy
6. ALL (4) SIDES SIMPLY SUPPORTED.
t
σy
σx
b
a
b=3.0 IN
b=6.0 IN
b=9.0 IN
b=12.0 IN
ΔT_CR ~ °F
t = THICKNESS ~ INCHES
0
10
20
30
40
50
60
70
80
90
100
110
.02
.04
.06
.08
.10
.12
.14

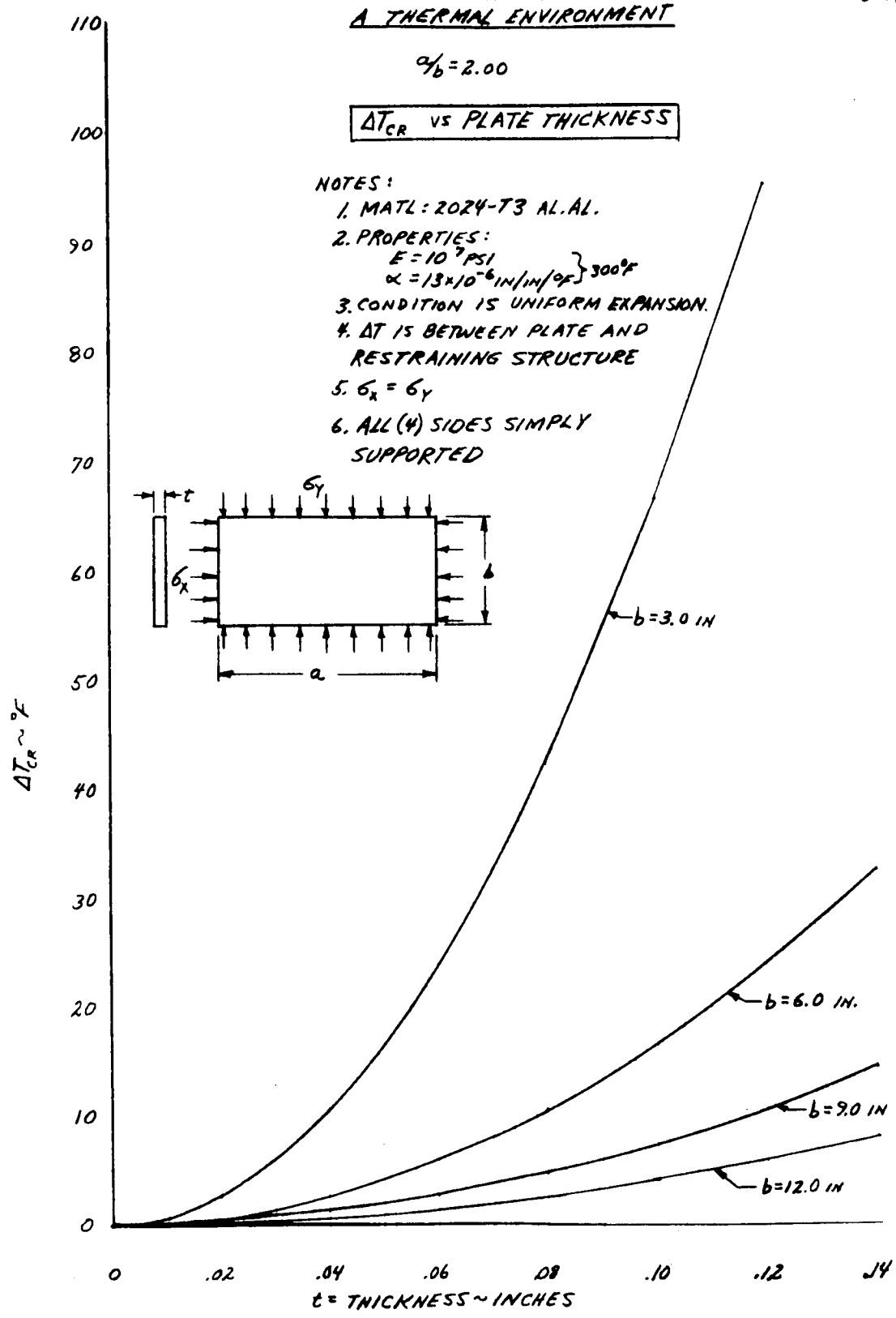
P. KUZMIK
3-19-64

FLAT PLATE BUCKLING IN A THERMAL ENVIRONMENT

$a/b = 2.00$

ΔT_{CR} VS PLATE THICKNESS

NOTES:
1. MATL: 2024-T3 AL. AL.
2. PROPERTIES:
   $E = 10^7$ PSI
   $\alpha = 13 \times 10^{-6}$ IN/IN/°F } 300°F
3. CONDITION IS UNIFORM EXPANSION.
4. ΔT IS BETWEEN PLATE AND RESTRAINING STRUCTURE
5. $\sigma_x = \sigma_y$
6. ALL (4) SIDES SIMPLY SUPPORTED



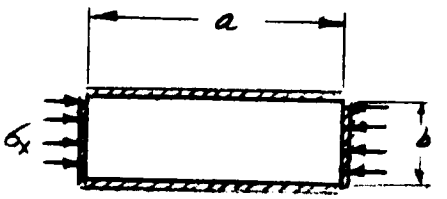
FLAT PLATE BUCKLING IN A THERMAL ENVIRONMENT

EDGES FIXED

$$\sigma_{CR} = k_c \left(\frac{t}{b}\right)^2 (9.04 \times 10^6)$$

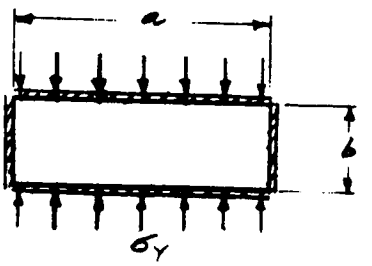
FOR LOAD IN "X" DIRECTION

a/b	k_c	σ_{CR-X}
1.00	10.6	$95.82 \times 10^6 (t/b)^2$
1.25	9.5	85.88
1.50	8.4	75.94
1.75	8.3	75.03
2.00	8.0	72.32
2.45	7.8	$70.51 \times 10^6 (t/b)^2$



FOR LOAD IN "Y" DIRECTION

a/b	b/a	k_c	σ_{CR-Y}
1.00	1.00	10.6	$95.82 \times 10^6 (t/b)^2$
1.25	.80	7.3	65.99
1.50	.67	6.0	54.24
1.75	.57	5.2	47.01
2.00	.50	4.8	43.39
2.45	.41	4.4	$39.78 \times 10^6 (t/b)^2$



FLAT PLATE BUCKLING IN A THERMAL ENVIRONMENT

EDGES FIXED

$$R = \frac{\sigma}{\sigma_{CR}} = \frac{186\,\Delta T_{CR}}{\sigma_{CR}}$$

ASPECT RATIO	σ_{CR-X}	R_X $\times[10^6(t/b)^2]$	σ_{CR-Y}	R_Y $\times[10^6(t/b)^2]$	R_X/R_Y
1.00	$95.82 \times 10^6(t/b)^2$	$1.94\ \Delta T$	$95.82 \times 10^6(t/b)^2$	$1.94\ \Delta T$	1.00
1.25	85.88	$2.17\ \Delta T$	65.99	$2.82\ \Delta T$	.77
1.50	75.94	$2.45\ \Delta T$	54.24	$3.43\ \Delta T$	.71
1.75	75.03	$2.48\ \Delta T$	47.01	$3.96\ \Delta T$	.63
2.00	72.32	$2.57\ \Delta T$	43.39	$4.29\ \Delta T$	.60
2.45	$70.51 \times 10^6(t/b)^2$	$2.64\ \Delta T$	$39.78 \times 10^6(t/b)^2$	$4.68\ \Delta T$	.56

SOLVE FOR ΔT_{CR} IN TERMS OF "t" & "b"

ASPECT RATIO	R_X/R_Y	R_{A-X}	R_X $[\times 10^6(t/b)^2]$	$^*\Delta T_{CR}$
1.00	1.00	.500	$1.94\ \Delta T$	$.2577 \times 10^6(t/b)^2$
1.25	.77	.465	$2.17\ \Delta T$	.2143
1.50	.71	.460	$2.45\ \Delta T$	.1878
1.75	.63	.440	$2.48\ \Delta T$	.1774
2.00	.60	.430	$2.57\ \Delta T$	.1673
2.45	.56	.440	$2.64\ \Delta T$	$.1667 \times 10^6(t/b)^2$

$^*\ R_X = R_{A-X}$

FOR $a/b = 1.0$

$$\frac{1.94\,\Delta T_{CR}}{10^6(t/b)^2} = .50 \quad ; \quad \Delta T_{CR} = \frac{.50}{1.94} \times 10^6(t/b)^2$$

$$= .2577 \times 10^6(t/b)^2$$

FLAT PLATE BUCKLING IN A THERMAL ENVIRONMENT

SOLVE FOR ΔT_{CR} (ALL EDGES FIXED)

t IN	b IN	$(t/b)^2 \times 10^6$	* ΔT_{CR} (°F) $a/b = 1.00$, $W = .2577$	$a/b = 1.50$, $W = .1878$	$a/b = 2.00$, $W = .1673$
.02	3	44.45	11.5	8.3	7.4
.04	↕	177.77	45.8	33.4	29.7
.06		400.00	103.1	75.1	66.9
.08		711.13	183.3	133.6	119.0
.10		1111.09	286.3	208.7	185.9
.12		1600.00	412.3	300.5	267.7
.14	3	2177.81	561.2	409.0	364.3
.02	6	11.11	2.9	2.1	1.9
.04	↕	44.45	11.5	8.3	7.4
.06		100.00	25.8	18.8	16.7
.08		177.77	45.8	33.4	29.7
.10		277.79	71.6	52.2	46.5
.12		400.00	103.1	75.1	66.9
.14	6	544.43	140.3	102.2	91.1
.02	9	4.94	1.3	.9	.8
.04	↕	19.75	5.1	3.7	3.3
.06		44.45	11.5	8.3	7.4
.08		79.01	20.4	14.8	13.2
.10		123.45	31.8	23.2	20.7
.12		177.77	45.8	33.4	29.7
.14	9	241.99	62.4	45.4	40.5
.02	12	2.78	.7	.5	.5
.04	↕	11.11	2.9	2.1	1.9
.06		25.00	6.4	4.7	4.2
.08		44.45	11.5	8.3	7.4
.10		69.44	17.9	13.0	11.6
.12		100.00	25.8	18.8	16.7
.14	12	136.12	35.1	25.6	22.6

$$^{*}\Delta T_{CR} = W \times 10^6 \left(t/b\right)^2$$

P. KUZMIK
3-20-64

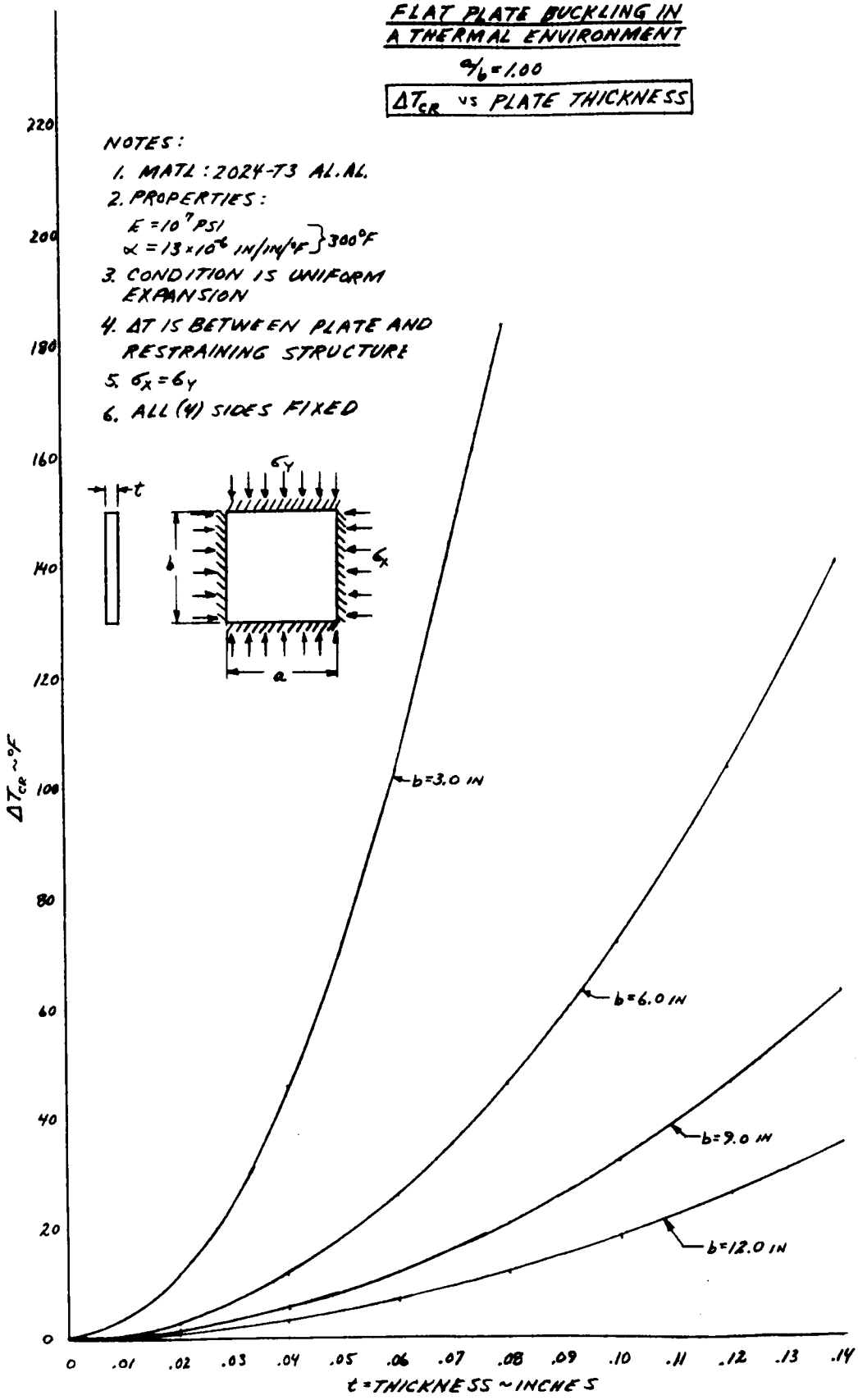
FLAT PLATE BUCKLING IN A THERMAL ENVIRONMENT

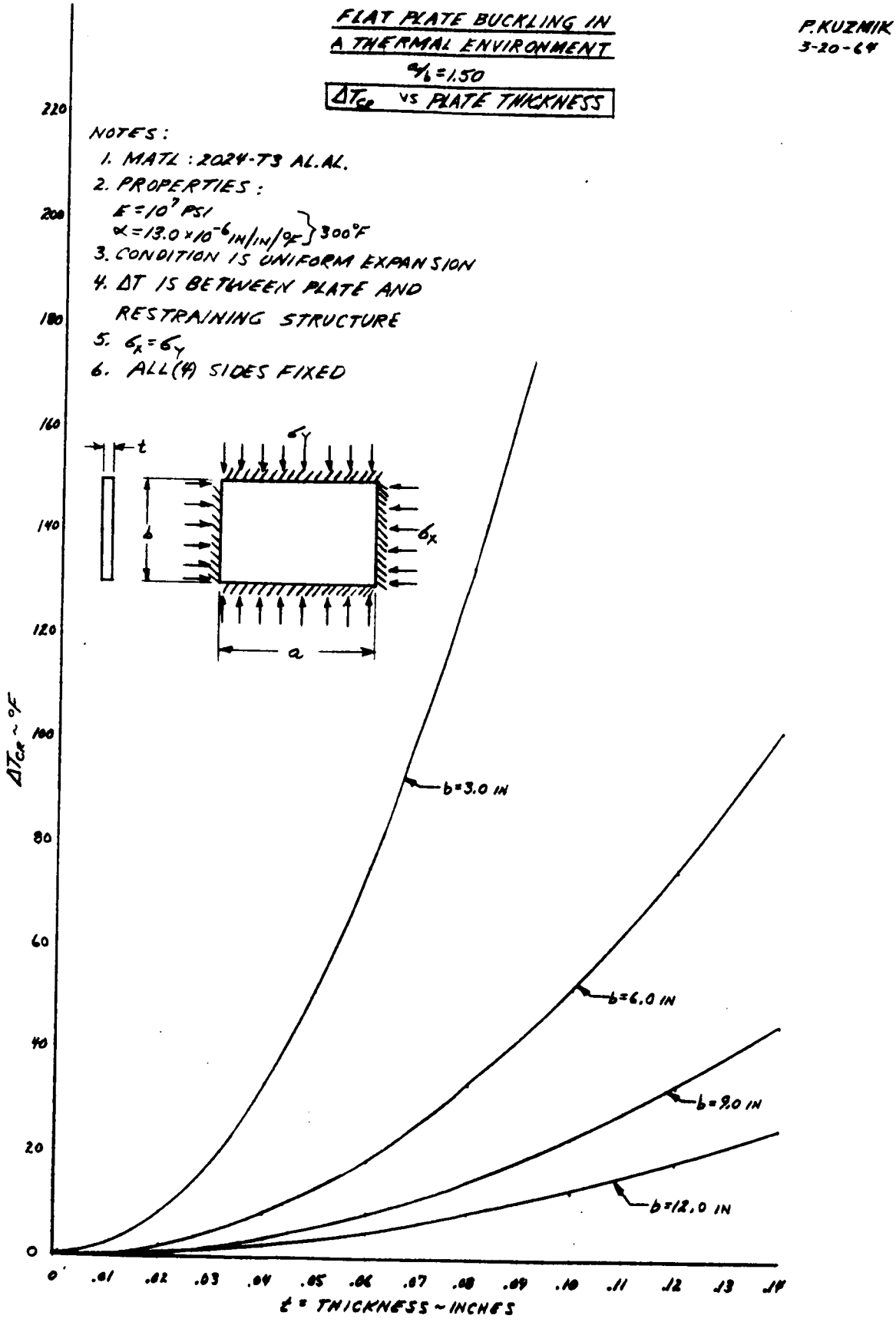
$a/b = 1.00$

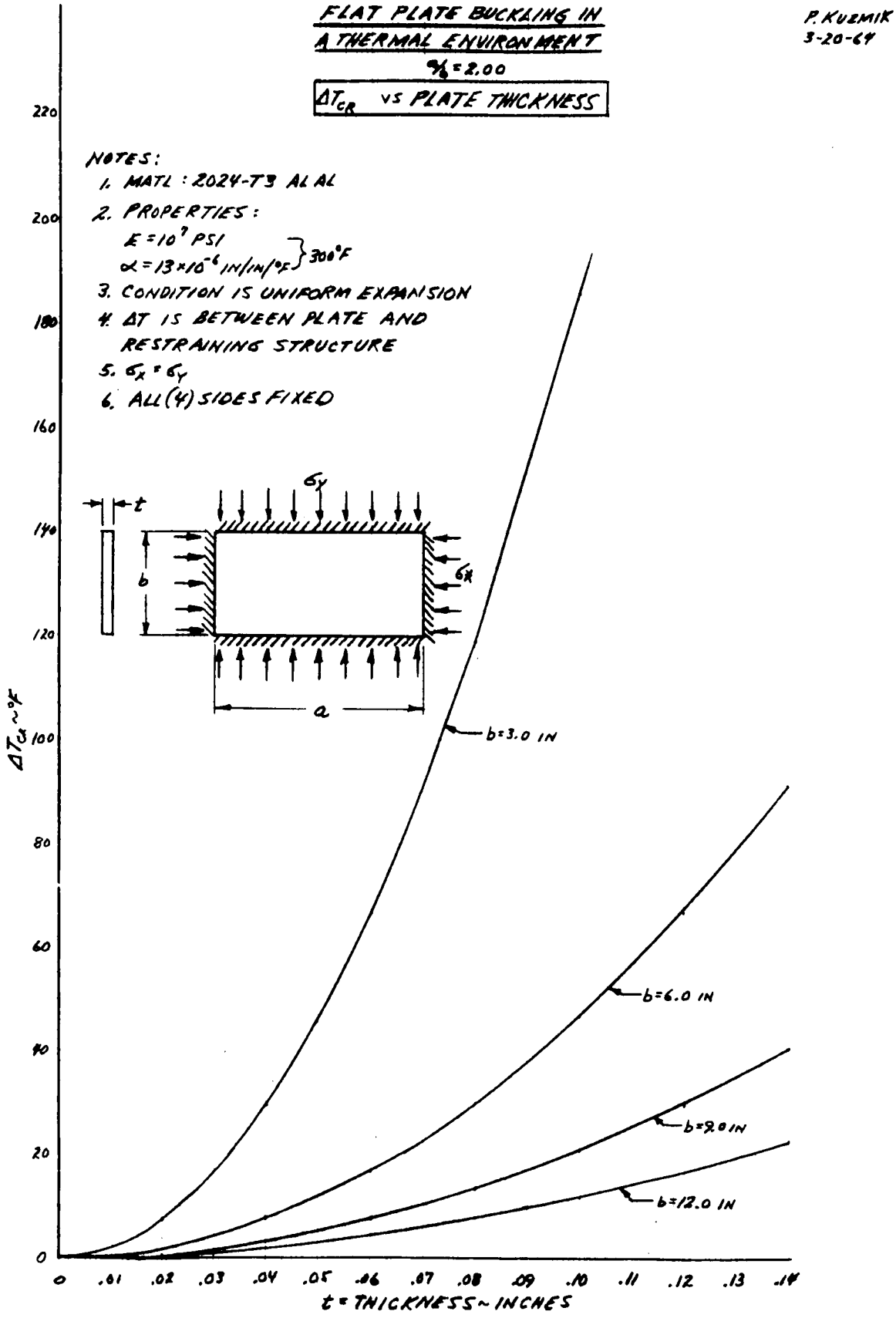
ΔT_{CR} VS PLATE THICKNESS

NOTES:

1. MATL: 2024-T3 AL. AL.
2. PROPERTIES:
   $E = 10^7$ PSI
   $\alpha = 13 \times 10^{-6}$ IN/IN/°F } 300°F
3. CONDITION IS UNIFORM EXPANSION
4. ΔT IS BETWEEN PLATE AND RESTRAINING STRUCTURE
5. $\sigma_X = \sigma_Y$
6. ALL (4) SIDES FIXED



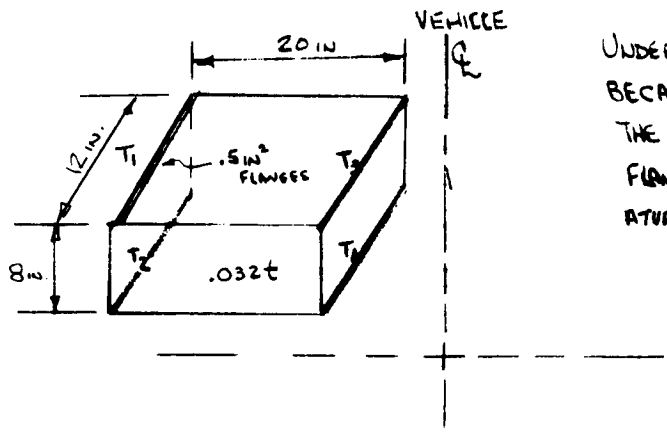

FLAT PLATE BUCKLING IN
A THERMAL ENVIRONMENT
a/b = 1.50
ΔT_CR VS PLATE THICKNESS
P. KUZMIK
3-20-64
NOTES:
1. MATL: 2024-T3 AL. AL.
2. PROPERTIES:
E = 10^7 PSI
α = 13.0 × 10^-6 IN/IN/°F } 300°F
3. CONDITION IS UNIFORM EXPANSION
4. ΔT IS BETWEEN PLATE AND
RESTRAINING STRUCTURE
5. σx = σy
6. ALL (4) SIDES FIXED
t
σy
σx
b
a
b = 3.0 IN
b = 6.0 IN
b = 9.0 IN
b = 12.0 IN
ΔT_CR ~ °F
0
20
40
60
80
100
120
140
160
180
200
220
0
.01
.02
.03
.04
.05
.06
.07
.08
.09
.10
.11
.12
.13
.14
t = THICKNESS ~ INCHES


FLAT PLATE BUCKLING IN
A THERMAL ENVIRONMENT
a/b = 2.00
ΔT_CR VS PLATE THICKNESS
P. KUZMIK
3-20-64
NOTES:
1. MATL: 2024-T3 AL AL
2. PROPERTIES:
E = 10^7 PSI
α = 13×10^-6 IN/IN/°F } 300°F
3. CONDITION IS UNIFORM EXPANSION
4. ΔT IS BETWEEN PLATE AND RESTRAINING STRUCTURE
5. σx = σy
6. ALL (4) SIDES FIXED
σy
σx
t
b
a
b=3.0 IN
b=6.0 IN
b=9.0 IN
b=12.0 IN
ΔT_CR ~ °F
t = THICKNESS ~ INCHES
0
20
40
60
80
100
120
140
160
180
200
220
.01
.02
.03
.04
.05
.06
.07
.08
.09
.10
.11
.12
.13
.14

EFFECT OF THERMAL GRADIENTS ON TYPICAL BOX OR COMPONENT HOUSING STRUCTURE

THIS ANALYSIS IS PRESENTED AS BEING TYPICAL OF THE SHEAR STRESS BUILDUP ON BOX STRUCTURE SUBJECTED TO THERMAL GRADIENTS. COMPONENT HOUSINGS ARE CONSIDERED SINCE THEY WOULD TYPICALLY BE MOUNTED FOR INERTIAL BALANCE AND MOST SUSCEPTIBLE TO GRADIENTS DURING A CONTROLLED STERILIZATION PROCEDURE.

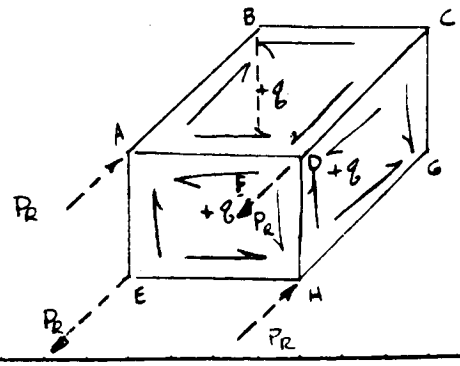
CONSIDER A COMPONENT MOUNTED AS SHOWN



UNDER A VACUUM STERILIZATION, BECAUSE OF ITS LOCATION FROM THE HEAT SOURCE THE FOUR FLANGES CAN HAVE VARYING TEMPERATURES AS SHOWN

LET $T_1 = 250°F$
$T_2 = 200°F$
$T_3 = 180°F$
$T_4 = 100°F$

ASSUME AN ALUMINUM BOX STRUCTURE OF .5 IN² FLANGES AND .032" SHEET - DIMENSIONS AS SHOWN. THE FLANGE AREA INCLUDES AN EFFECTIVE SHEET THICKNESS. TO MAINTAIN COMPATIBILITY BETWEEN THE FLANGE EXPANSIONS, A REDUNDANT SHEAR FORCE IS GENERATED IN THE SHEET.



EXPRESS q IN TERMS OF A SET OF COUPLES P_R

$$2q = \frac{P_R}{12} \qquad q = \frac{P_R}{24} = .417\,P_r$$

USING THE MATRIX FORCE METHOD OF ANALYSIS

$$F = [T \;\; V]\begin{bmatrix} P \\ R \end{bmatrix}$$

THE METHOD OF ANALYSIS THAT BEST SUITS THIS THIS DISTORTION PROBLEM IS TO DEAL DIRECTLY WITH THE INITIAL DISTORTIONS

THEN $F = -VG_{22}^{-1}V^T\delta^*$

WHERE V IS A GEOMETRY MATRIX FOR REDUNDANT P_F

G_{22}^{-1} = INVERSE OF $V^T L V$

L = FLEXIBILITY MATRIX

δ^* = INITIAL DISTORTIONS

$= \alpha L \Delta T$ AT A, E D & H

$= 0$ ELSEWHERE

ORDER OF F

$$\underbrace{\begin{Bmatrix} F_A \\ F_B \\ F_D \\ F_C \\ F_E \\ F_F \\ F_H \\ F_G \\ F_{ABCD} \\ F_{EFGH} \\ F_{ABFE} \\ F_{DCGH} \\ F_{BCGF} \\ F_{ADHE} \end{Bmatrix}}_{F} \quad \underbrace{\begin{Bmatrix} 1 \\ 0 \\ -1 \\ 0 \\ -1 \\ 0 \\ 1 \\ 0 \\ +.417 \\ -.417 \\ -.417 \\ +.417 \\ -.417 \\ +.417 \end{Bmatrix}}_{V}$$

$\delta^*_A = \alpha L \Delta T = 12.5(12)(250-70)(10)^{-6} = .0270$

$\delta^*_E = \alpha L \Delta T = 12.5(12)(200-70)(10)^{-6} = .0195$

$\delta^*_D = \alpha L \Delta T = 12.5(12)(180-70)(10)^{-6} = .0165$

$\delta^*_H = \alpha L \Delta T = 12.5(12)(100-70)(10)^{-6} = .0045$

FLEXIBILITY MATRIX h

$$f_{SHEET} = \frac{ab}{Gt}$$

$$f_{FLANGE} = \frac{L}{6AE}\begin{pmatrix} 2 & 1 \\ 1 & 2 \end{pmatrix}$$ FOR UNIFORM SHEAR LOADING

MEMBER AB
DC
EF
HG

$$f = \frac{12}{6(.5)(10)^7}\begin{pmatrix} 2 & 1 \\ 1 & 2 \end{pmatrix} = 4(10)^{-7}\begin{pmatrix} 2 & 1 \\ 1 & 2 \end{pmatrix}$$

ABCD
EFGH

$$\frac{12 \times 20}{(3.8)(10)^6 .032} = 4(10)^{-7}\ (4934)$$

ABFE
DCGH

$$\frac{12 \times 8}{3.8(10)^6(.032)} = 4(10)^{-7}\ (1974)$$

BCGF
ADHE

$$\frac{20 \times 8}{3.8(10)^6(.032)} = 4(10)^{-7}\ (3289)$$

$$h = 4(10)^{-7}\begin{bmatrix} 2 & 1 & & & & & & & & & & & & & & \\ 1 & 2 & & & & & & & & & & & & & & \\ & & 2 & 1 & & & & & & & & & & & & \\ & & 1 & 2 & & & & & & & & & & & & \\ & & & & 2 & 1 & & & & & & & & & & \\ & & & & 1 & 2 & & & & & & & & & & \\ & & & & & & 2 & 1 & & & & & & & & \\ & & & & & & 1 & 2 & & & & & & & & \\ & & & & & & & & 4934 & & & & & & & \\ & & & & & & & & & 4934 & & & & & & \\ & & & & & & & & & & 1974 & & & & & \\ & & & & & & & & & & & 1974 & & & & \\ & & & & & & & & & & & & 3289 & & & \\ & & & & & & & & & & & & & 3289 & & \end{bmatrix}$$

$$G_{22} = V^T L V$$

$$= 4(10)^{-7}\,[3554.168]$$

$$G_{22}^{-1} = .7034\,(10)^3$$

$$F = [-VG_{22}^{-1}V^T][\delta^*]$$

PARTIONING THE MATRIX $-VG_{22}^{-1}V^T$

$$F = -VG_{22}^{-1}V^T$$

$$\begin{bmatrix} F_A \\ F_B \\ F_D \\ F_C \\ F_E \\ F_F \\ F_H \\ F_G \\ F_{ABCD} \\ F_{EFGH} \\ F_{ABFE} \\ F_{DCGH} \\ F_{BCGF} \\ F_{ADHE} \end{bmatrix} = 703.4 \begin{bmatrix} 1 & 0 & -1 & 0 & -1 & 0 & 1 \\ 0 & 0 & 0 & 0 & 0 & 0 & 0 \\ -1 & 0 & +1 & 0 & +1 & 0 & -1 \\ 0 & 0 & 0 & 0 & 0 & 0 & 0 \\ -1 & 0 & +1 & 0 & +1 & 0 & -1 \\ 0 & 0 & 0 & 0 & 0 & 0 & 0 \\ 1 & 0 & -1 & 0 & -1 & 0 & 1 \\ 0 & 0 & 0 & 0 & 0 & 0 & 0 \\ +.417 & 0 & -.417 & 0 & -.417 & 0 & +.417 \\ -.417 & 0 & +.417 & 0 & +.417 & 0 & -.417 \\ -.417 & 0 & +.417 & 0 & +.417 & 0 & -.417 \\ +.417 & 0 & -.417 & 0 & -.417 & 0 & +.417 \\ -.417 & 0 & +.417 & 0 & +.417 & 0 & -.417 \\ +.417 & 0 & -.417 & 0 & -.417 & 0 & +.417 \end{bmatrix}$$

NOT REQUIRED

NOT REQUIRED

$$\begin{bmatrix} \delta^2 \end{bmatrix} = \begin{bmatrix} F \end{bmatrix}$$

δ^2	F
.0270	-.0045
0	0
.0165	+.0045
0	0
.0195	+.0045
0	0
.0045	-.0045
0	0
0	-.00188
0	+.00188
0	+.00188
0	-.00188
0	+.00188
0	-.00188

MEMBER FORCES

A-B	3.17 lbs	COMPRESSION	AT A END
C-D	3.17 lbs	TENSION	AT D END
E-F	3.17 lbs	TENSION	AT E END
G-H	3.17 lbs	COMPRESSION	AT H END

SKIN SHEARS = 1.32 lbs/in

Conclusions

It can be seen that the effects of sterilization on small compact structures of the black box housing variety are small in that these structures are free to expand.

To expand the results of the analysis to large box structures, and shear panels in general, it can be seen that the redundant shears are dependent on the flexibility of the structure and the differential thermal strains. It follows that combinations of stiff or large area flanges and very thin sheets will produce higher skin shears. Also, as the panel dimensions increase and the skin thickness decreases the possibility of panel instability or diagonal tension wrinkles becomes more prominent.

For most panels proportioned for flight inertia loads however, the induced panel shears should be small enough so that the sterilization requirement is not limiting.

APPENDIX G

DESIGN CONSIDERATIONS FOR A BIOLOGICAL BARRIER

The information contained in this Appendix was prepared by the Spacecraft Department of the General Electric Missile and Space Division for the Voyager Design Study under Contract NAS W-696 with the NASA Office of Space Sciences.

APPENDIX G

DESIGN CONSIDERATIONS FOR A BIOLOGICAL BARRIER

The preliminary design requirements for a biological barrier are as follows:

1. It shall have electrical connections for check-out and test. This will involve an estimated 5 connections.
2. It shall have "hose" fittings for liquids and gases. It is estimated that two fittings for liquid and one fitting for gas will be required.
3. It shall be big enough to contain lander, radio isotope-thermoelectric generator, and protective container, and remote handling gear.
4. It shall have straps, handles, or fasteners for attachment to the shroud.
5. It shall have an opening large enough to accept items without tearing.
6. It shall be as light as possible since the barrier accompanies the spacecraft into space.
7. It shall not be affected by hot (up to 100°C) or cold ethylene oxide.
8. It shall not be affected by heat up to 300°F (150°C).
9. It shall have a method for removing a section after the shroud is ejected.
10. It requires a built-in electrical junction box between lander and orbiter.
11. The purpose of the barrier is to keep out bacteria, spores, and other micro-organisms.
12. To avoid carrying the junction box into space, it shall be possible to detach this before flight.

From: Voyager Design Study
Volume V, Sterilization
Document No. 63SD801
15 October 1963
Contract NAS W-696

APPENDIX H

CANISTER ANALYSIS

BY CK. DATE REV.	GENERAL ELECTRIC	PAGE 1 MODEL REPORT

CANISTER STRUCTURAL DESIGN

Structural Analysis

Introduction

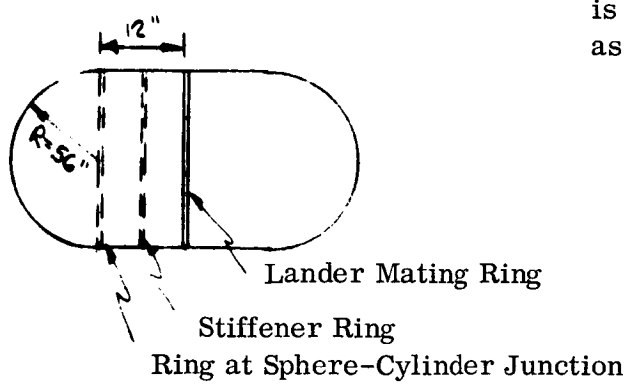
The analyses presented below substantiate the structural weight figures given in the comparative study of the canister requirements for the different sterilization procedures.

The material allowables used are at temperature and a factor of 2.0 is applied to the limit loads in keeping with current pressure vessel design practices.

To simplify the analyses, the discontinuity forces at the lander attachment ring are not included. This is in keeping with the comparative nature of the study. Also, it is assumed that the canister is heated in a manner such that ring-sheet gradients do not exist.

Geometrical Parameters

A sphere-cylinder combination is used. Stiffeners are located as required.



CONDITION I

UNVENTED CANISTER CONTAINING INERT GAS

PRESSURE @ R.T. - 1psig

PRESSURE @ 300°F CONSTANT VOLUME EXPANSION $P_2 = \frac{P_1 T_2}{T_1}$

$$P_2 = 15.7\left(\frac{760}{530}\right) = 22.5 - 14.7 = 7.8 \text{ psig}$$

ULT. DESIGN PRESSURE = 2(7.8) = 15.6 psi

MATERIAL : STAINLESS STEEL F_{ty} @ 300° = .97(170,000) = 165,000 psi

CYLINDER THICKNESS :

$$t = \frac{pr}{F_{ty}} = \frac{15.6(56)}{165,000} = .0053''$$

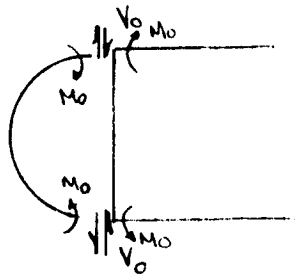
THIS THICKNESS IS TOO SMALL TO BE PRACTICAL FOR HANDLING LOADS. USE ALUMINUM 6061-T6 FOR WELDED CONSTRUCTION

F_{ty} = .83(35,000) = 29,000 psi

$$t = \frac{15.6(56)}{29,000} = .030 \text{ IN.}$$

USE .030" AL. FOR SPHERE ALSO. FOR ACTUAL HARDWARE LOCAL STIFFENING WOULD PROBABLY BE REQUIRED IN AREAS OF HANDLING FITTINGS, GAS PORTS ETC. THESE ARE ESTIMATED HERE, AND THE ONLY STIFFENING RING SIZED IS A RING AT THE SPHERE-CYLINDER JUNCTION.

CONDITION I (CONT.)



FOR UNIFORM SHELL THICKNESS $M_0 = 0$

$$V_0 = \frac{P}{8\sqrt[4]{\frac{3(1-\nu^2)}{r^2t^2}}} = \frac{15.6}{8\sqrt[4]{\frac{3(.91)}{(56)^2(.030)^2}}} = 1.97\,\#/\text{IN}$$

THIS IS A LOCAL SHEAR WHICH CAN EASILY BE TAKEN BY A 1" DOUBLER

$$t = \frac{1.97(56)}{29{,}000} = .0038 \text{ IN}$$

USE .012 IN THICK STRAP

CONDITION II

IN THIS CASE, THE CANISTER IS EITHER CHARGED OR VENTED TO MAINTAIN A ONE PSI PRESSURE DIFFERENTIAL.

$$t_{CYL} = \frac{2(56)}{29{,}000} = .0039''$$

HANDLING CONDITIONS WILL DESIGN THIS CANISTER. ASSUME .030 IN t.

CONDITION III

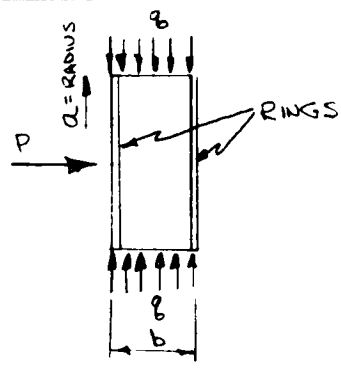
COMPLETE VACUUM. NO CHARGING OR VENTING

$P_{ULT} = 2(14.7) = 29.4$ PSI EXTERNAL

INSTABILITY WILL BE CRITICAL FAILURE MODE. STAINLESS STEEL HAS A HIGHER E/ρ AT TEMPERATURE THAN ALUMINUM, AND HENCE THIS IS A BETTER CHOICE FOR THE SHEET MATERIAL. HOWEVER, IT MUST ALSO BE CONSIDERED THAT THE STIFFENING RINGS WILL BE THE MOST EFFICIENT WHEN A HIGH I/ρ RATIO IS OBTAINED. SINCE I IS A GEOMETRICAL PARAMETER ONLY ALUMINUM STIFFENING RINGS WILL BE MORE EFFICIENT THAN STEEL. WISHING TO AVOID MATERIAL INCOMPATIBILITIES AN ALL ALUMINUM CONSTRUCTION IS CHOSEN.

THE VALUE OF E_c FOR WELDABLE 6061 AT 300°F IS $9.6(10)^6$. FOR AN ACTUAL DESIGN, THE CANISTER AND VEHICLE MATERIALS SHOULD BE THE SAME.

CYLINDRICAL SECTION



LET $b = 9$ IN AS A FIRST CUT RING SPACING.

$q = 29.4$ PSI

$P = 29.4(\pi a^2)$ = TOTAL AXIAL LOAD

$a = 56.0$ INCHES

THE METHOD OF ANALYSIS CONSIDERES THE INTERACTION BETWEEN q_{cr} AND P_{cr}. CURVES OF THE INTERACTION EXPRESSION ARE PRESENTED IN THE MSVD-GE STRUCTURES MANUAL AND THE ANALYSIS FOLLOWS THE FORMAT THEREIN.

β_1 (LATERAL LOAD PARAMETER)

$$= \frac{q(1-\nu^2)}{E} \times 10^8 = \frac{29.4(.91)(10)^2}{9.6} = 279$$

CONDITION III (CONTD)

B_2 (AXIAL LOAD PARAMETER)

$$= \frac{P(1-\nu^2)(10)^8}{2\pi a^2 E} = \frac{29.4\,(.91)(10)^8}{2\,(9.6)(10)^6} = 139$$

K_1 (GEOMETRY PARAMETER) $= \frac{a}{b} = \frac{56}{9} = 6.2$

USE CURVE FOR $K_1 = 6$ $b = 9.33$ IN.

READ for $B_1 = 279$

$B_2 = 139$ $\frac{h}{a}$ REQ'D $= .0035$

$h = .0035(56) = .196$ IN

STANDARD SKIN GAGES IN THIS RANGE ARE .156, .188 & .250
LIGHTER SKIN GAGES WITH CLOSER RING SPACING IS MORE EFFICIENT SINCE AS THE SHELL THICKNESS DECREASES THE REQUIREMENT FOR FLEXURAL RIGIDITY IS TRANSFERRED TO THE STIFFENERS, WHICH CAN SUPPLY A HIGHER FLEXURAL RIGIDITY PER UNIT WEIGHT THAN THE SKIN.

CHECK RING SPACING FOR .156 IN. SHEET

$\frac{h}{a} = \frac{.156}{56} = .0028$

$K_1 = 8$ (EST) $b = \frac{56}{8} = 7$ IN.

USE .156 IN SHEET AND 7 INCH RING SPACING.

IN THE HEMISPHERICAL PORTION THE RING SPACING CAN GRADUALLY BE INCREASED SINCE THE AXIAL LOAD IS REDUCED AS THE PROJECTED AREA DECREASES.

CONDITION III (cont'd)

REQUIRED STIFFENER INERTIA

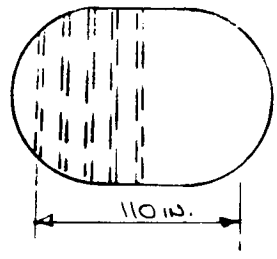
LATERAL LOAD PARAMETER $\Phi_1 = \frac{q a (1-\nu^2)}{E h} = \frac{29.4(56)(.91)}{9.6(10)^6(.156)} = .100(10)^{-2}$

AXIAL LOAD PARAMETER

$$\Phi_2 = \frac{P(1-\nu^2)}{2\pi a E h} = \frac{29.4(56)(.91)}{2(9.6)(10)^6(.156)} = .050(10)^{-2}$$

GEOMETRY PARAMETER $K = \frac{a}{L}$ WHERE L = TOTAL LENGTH OF CYLINDER

L IS CHOSEN HERE TO INCLUDE ALL BUT THE EXTREME SPHERICAL CAPS AS SHOWN.



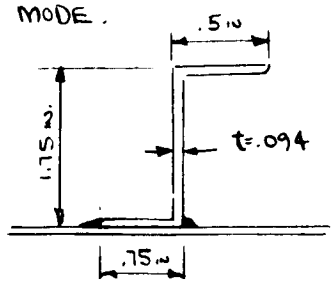
$K = \frac{56}{110} = .508$

READ α_1 FROM CURVE KNOWING Φ_1, Φ_2 & K

$\alpha_1 = 3(10)^{-5}$

$$I_e = \frac{\alpha_1 b h a^2}{(1-\nu^2)} = \frac{3(10)^{-5}(7)(.156)(56)^2}{.91} = .113 \text{ IN}^4$$

A ZEE STIFFENER IS CHOSEN WITH A THICKNESS SUCH THAT LOCAL INSTABILITY OF THE STIFFENER WEB IS NOT A FAILURE MODE.



$I_{xx} = .1104 \text{ IN}^4$

$A = .267 \text{ IN}^2$

THIS IS A PERMISSIBLE SECTION SINCE I_e ABOVE WOULD INCLUDE THE EFFECTIVE SHEET INERTIA

$\frac{b}{t} = \frac{.5}{.094} = 5.3$ UNSUPPORTED EDGE

$= \frac{1.75}{.094} = 18.5$ SUPPORTED EDGES

} ADEQUATE TO PRECLUDE CRIPPLING

FORM 1-8136 B (9-58)

Condition III (Cont.)

It can be seen that the proportion of ring structure to sheet structure is not as efficient as could be realized with closer ring spacing. The optimum design must consider other factors, such as space and clearance requirements. Of course a honeycomb construction would be even more efficient, however this introduces other problems such as decreased thermal conductivity, attachment and cutout design problems and greatly increased cost. Nevertheless, the rough gages given could be taken as representative for an extreme case.

Condition IV

The maximum pressure differential on the canister, providing the chamber and canister are evacuated simultaneously is one psi. This is the same as Condition II and the same canister would suffice.

APPENDIX J

TEMPERATURE DISTRIBUTION IN CONNECTORS AND BOLTED JOINTS

The analyses and experiments described in this Appendix were performed by the Advanced Technology Laboratories of the General Electric Company under Contract No. NAS 8-4012 for the NASA-Marshall Space Flight Center. The TIS Documents referenced in this Appendix were issued under Contract No. NAS 8-4012.

APPENDIX J

TEMPERATURE DISTRIBUTION IN CONNECTORS AND BOLTED JOINTS

J.1 TEMPERATURE DISTRIBUTION IN CONNECTORS

Tube and duct connectors have a relatively compact construction and are ordinarily made of metals having fair conductivity. For this reason it is reasonable to expect that, unless the heat transfer to their surfaces is extremely high, the temperature distribution in them will be nearly uniform.

An estimate of the temperature differences in a connector or similar part during sterilization can be obtained from work done on NASA Contract NAS 8-4012 and reported in General Electric Reports TIS 63GL41 to 46. This work was performed in order to determine the thermal distortion of fluid connectors. Figure J-1 is a copy of Figure 61.10 of that report. On it one can see how the maximum temperature difference across the wall of a flange or pipe varies with thermal constants and dimensions. One can also see that wall curvature has very little effect.

T_o = Initial Temperature
T_f = Final Temperature
K = Thermal Conductivity of Flange Material
h = Heat Transfer Coefficient
s = Thickness
Max ΔT = Maximum Temperature Difference Across Flange While Heating

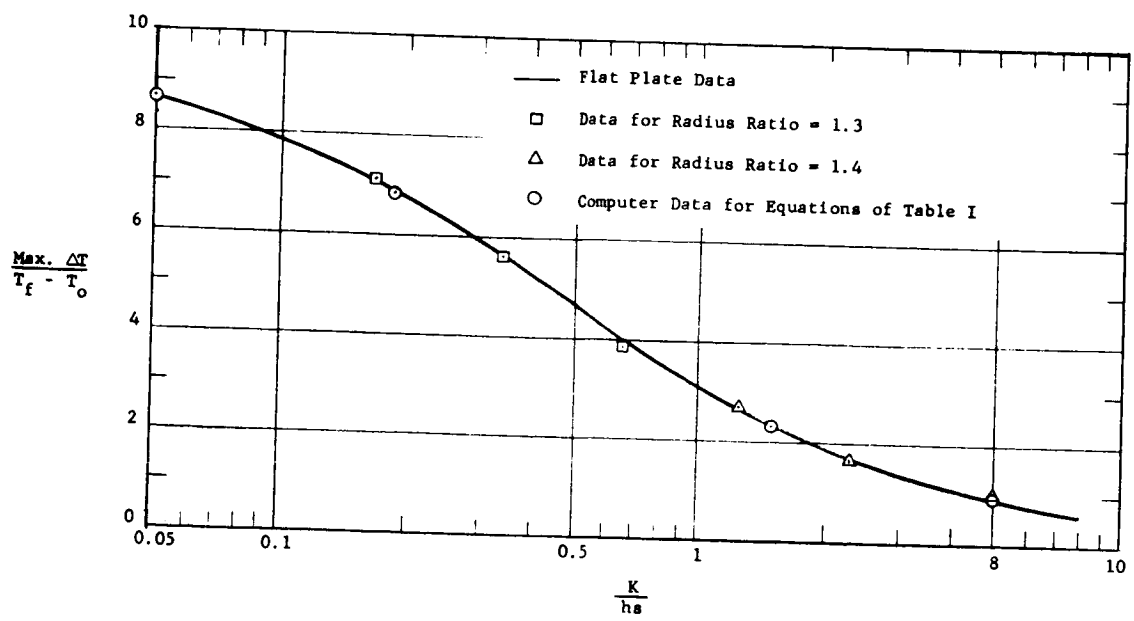


Figure J-1. Maximum Temperature Difference Through Flange Thickness (From Figure 61.10 of TIS 63GL46)

Figure J-1 is based on the use of an effective heat transfer coefficient. In the sterilization process some heat will be transferred by radiation while the remainder reaches the wall by conduction through a surface gas film. (In the case of heating in a vacuum the radiation term will represent nearly all the heat transfer.) Although radiant heat transfer takes place as the fourth power of the absolute temperature, over any particular temperature range it can be linearized. This gives its contribution to the effective heat transfer coefficient, h, in Figure J-1.

Unless the gases surrounding the structure being sterilized are both dense and have a high velocity at the surface of the structure, their contribution to the heat transfer coefficient, h, will be so small that one would enter Figure J-1 at large values of K/hs where the expected maximum temperature difference across the walls of the connector would be less than a few percent of the maximum temperature change.

It is found in Section 61.1 of TIS 63GL46 that for pipes heated by internally flowing gas the contribution to h of the gas flow is given by:

$$h_{gas} = (Nu)\, K_f/D \ (\text{BTU/hr ft}^2\ {}^{\circ}\text{F}) \tag{1}$$

where

K_f = thermal conductivity of fluid (BTU/hr ft^2 $^{\circ}$F)

D = diameter of pipe (ft)

and

$$(Nu) = 0.023\,(Re)^{0.8}\,(Pr)^{0.3}, \text{ Nusselt's number} \tag{2}$$

where

$$(Re) = DG/\mu, \quad \text{Reynold's number} \tag{3}$$
$$(Pr) = C\mu/K_f, \quad \text{Prandtl's number} \tag{4}$$

μ = fluid viscosity (lb/hr ft)

C = specific heat of flange material (BTU/lb $^{\circ}$F)

G = mass flow density (lb/hr ft^2)

The contribution to h due to radiation will depend on the geometric configuration of the radiant heaters as well as on the radiation efficiency of the various surfaces. The maximum radiant heat transfer to the structure will occur if it is completely surrounded and if both the heater and structure surfaces have perfect radiation efficiency. Its value is

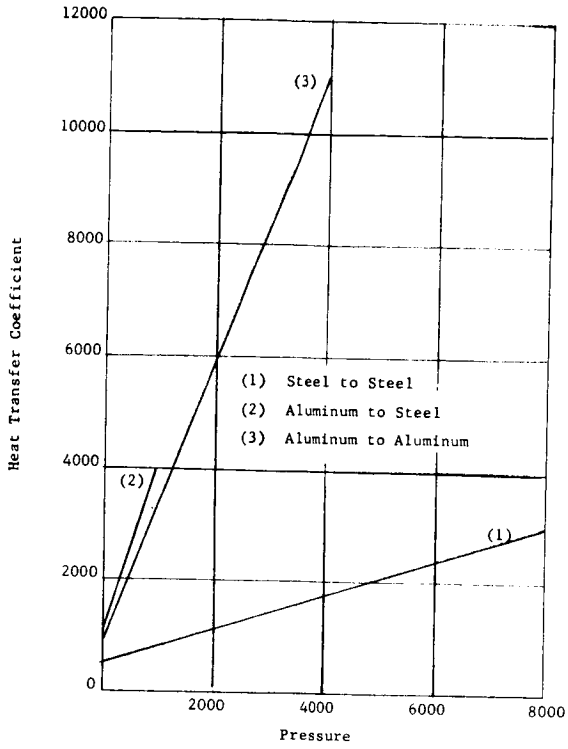


Figure J-2. Heat Transfer Coefficient at Joint Between Two Surfaces

that the response is nearly as though the joint were absent. In an example on page 61-3.4, 35 of TIS 63GL46, it is found that the center of a stainless-steel bolt comes up to half the flange temperature in 100 seconds while for an aluminum bolt this takes only 14 seconds. Since sterilization times are expected to be considerably longer than this, it is reasonable to expect the bolts and flanges to reach a nearly uniform temperature.

In Section 61.4 of TIS 63GL46, a computer method is given for transient analysis of temperatures in flange assemblies. An example given there of a connector consisting of a stainless steel flange joined to an aluminum alloy flange by aluminum bolts shows that the bolt temperature lags behind the wall temperature by only about 20 seconds. Again one is led to conclude that all parts of a connector will be thoroughly heated long before the sterilization process is completed.

$$h_{rad} = B\,(T_H^3 + T_S\,T_H^2 + T_S^2\,T_H + T_S^3) \quad (5)$$

$$(\text{BTU/hr ft}^2\ {}^{o}\text{F})$$

where

T_H = absolute temperature of heater surface (^{o}Rankine);

T_S = absolute temperature of structure (^{o}Rankine);

B = a constant dependent on geometry and radiation coefficients (BTU/hr ft $^{o}F^4$).

For configurations appropriate to sterilization the value of h including both radiant and convective heat flow will be so low that temperature differences through the solid wall of pipes and flanges will be small in comparison with the total temperature rise.

For example, values of h for combinations of convection and radiation coefficients for oxidized bare steel pipe are given in Table 11, page 377, of Mark's Handbook (1951 Edition). These values range from about 1 to 15 BTU per hr sq ft per deg F difference. Corresponding values for flat plates are in the same range. Since the thermal conductivity K for metals falls in the range of 10 (stainless steel) to 80 (aluminum alloy) BTU per hr ft per deg F difference and the wall thickness is likely to be less than 1 inch, it is evident from Figure J-1 that even during the transient heating time the maximum temperature difference from front to back of the wall will be only a small fraction of the total temperature change.

J.2 TEMPERATURE DISTRIBUTION IN BOLTED JOINTS

A knowledge of probable temperature transients in bolts can be obtained from Section 61.3 of TIS 63GL46. The bolts considered there are used to hold the flanges of a fluid connector together. Thermal contact resistance at the bolt head and at the nut is quite dependent on surface finish as well as on material and the pressure between mating parts. Figure J-2 (corresponding to Figure 61.9 of TIS 63GL46) shows some curves for the heat transfer coefficient at joints where the joined metals are steel or aluminum with surface roughnesses of about 50-100 micro inches. This roughness is equivalent to a fairly ordinary machining job that might be done on a flange or bolt. A point to note, especially at lower conductances, is that at the nut end of the bolt there are two resistances in series.

The higher the heat transfer constant at a junction, the more nearly the parts behave as though the joint were not present. In conventional joints, where contact pressures of 10,000 psi or higher are to be expected, the conductivity is so high

APPENDIX K

INTERFACE PHENOMENA FOR METALS IN CONTACT UNDER NORMAL STRESS

The analyses and experiments described in this Appendix were performed by the Advanced Technology Laboratories of the General Electric Company under Contract Nos. NAS 8-4012 and NAS 8-11523 for the NASA-Marshall Space Flight Center.

APPENDIX K

INTERFACE PHENOMENA FOR METALS IN CONTACT UNDER NORMAL STRESS

K.1 Introduction

In virtually all assembled components, some joints are effected by the mechanical fastening together of two metal pieces. Riveted joints, bolted joints and threaded joints all possess junctions of metal in contact, as opposed to an integral metallurgical bond between the pieces. Many configurations utilize only the two components in direct contact; others utilize intermediate pieces, such as gaskets in fluid connectors. In both cases, however, the problem arises as to what the true area of contact is between the parts joined. Since some external force is applied to hold the pieces together, a normal stress exists across the interface between the components. In some geometries, shear stresses also will occur. The result of these contact stresses is that surface deformations occur on one or both components, resulting further in some degree of mating. Were all assembled components produced with perfectly flat mating and perfectly smooth surfaces, then the bodies would be in complete contact under very small normal contact stresses. However, this is never the case. On a microscopic scale, at least, surface asperities will exist, their size depending on the machining or grinding technique used to produce the part. When the components are initially assembled, the parts will contact only on the peaks of the asperities. As the normal load between the parts increases, the asperities (actually now under extremely high stresses) will yield, allowing contact to exist between other asperities. The question arises as to what the distribution of "islands" or "ridges" of contact is when the pieces are assembled to make a satisfactory structural joint.

The importance of this problem in the sterilization process cannot be under-estimated. Three separate problems due to the incompleteness of contact between mated parts can be seen;

a) A small area of contact between mated parts will result in a decrease in thermal conductivity of the assembly across the joint. Roughly, the conductivity is proportional to the area of contact. Should few asperities actually contact, then the heat transfer must be accomplished by convection through the gas existing in the interface voids. Should the system be in a vacuum, then radiation may be the only means of transferring heat across the surface, except for those local areas of contact. In any case, the resulting conductivity will depend on the area of contact and the amount and type of gas filling the voids. Since the sterilization process involves the heating of the entire system to be sterilized, the lack of conductivity due to incomplete mating should be considered.

b) Sterilization has as its goal the removal of all bacteria from the system. When small voids exist within the system, such as exist across material contacts, the possibility of the entrapment of bacteria is present.

c) The most important problem associated with incompleted contacts is that of the possibility, and often the probability, that the voids will be closed. In other words, caverns of air will exist in the interface region which have no access to the outside. The frequency and size of such caverns depend on the stress between the components, the geometry of the mating surfaces, and particularly, the surface finish on the parts. Should these exist, then complete evacuation of the total system is impossible. That gas which is initially in the void will remain during the sterilization process. Any bacteria present will also remain, albeit they may be killed by the heat. The complication due to this phenomenon is that when materials are highly stressed, there is a slow degradation of that stress, and ultimately the voids may open, at a time when the system is in a region requiring sterilization.

With the above problems present, an accurate understanding of the degree of mating possible between structural components in contact is of great interest. During NAS Contracts 8-4012 and 8-11523, both dealing with the fundamentals of the design of leakproof fluid connectors, many experiments were accomplished in which data concerning true area of contact as a function of normal stresses were made available. While the primary purpose of the experiments was to discern how much gas could flow between mated surfaces, the techniques employed yielded information on the resultant degree of mating for several materials and for several surface finishes. The report which follows illustrates the severity of the problems, presents many of the results pictorially, and lists some of the conclusions concerning the material deformation during surface compression.

K.2 The Five Regimes of Material Deformation During the Mating Process

When two surface of metals are brought together and compressed under a normal stress, five separate regimes of material flow can be envisioned. The materials may have different strength properties (as many structural materials do) and they may have different surface finishes. Should they have different yield points, and should the geometry of the parts be such that one piece is smaller than the other, (as in a gasket, or a washer in a bolted joint) then the material flow of the softer smaller parts may be approximated by plane strain deformation. Should the parts be of the same geometry and of the same strength, then the deformation mode is slightly different, but the resultant degree of mating does not vary appreciably (based on experimental evidence).

Consideration of the general case where a component is softer than the other, the following five regimes are present, each as a function of increasing normal stress.

K.2.1 Regime I

Since each surface in contact does have some distribution of asperities, and since the distribution and size of the asperities is a function of the manufacturing process, many of the asperities will be of some representative size. However, some asperities will exist which are higher than the mean peak values, and several will exist which are lower. In the case of machined surfaces, not only are asperities caused by the intentional movement of the cutting tool, but the tool itself has asperities, and its motion is not purely smooth. Thus, minor asperities superimposed on the major asperities exist, along with the accidental deviations of a regular pattern of the major asperities.

As the first asperities come in contact under the lightest of loads, only the highest touch. The contact stresses between these asperities will be at equilibrium at a level exceeding the yield stress. Deformation of the asperities will cease at a contact stress between two and three times the yield stress of the softer material, depending on the geometry of the asperities. (Section K.3). During the initial contact between the bodies, the real area of contact is negligible. The mean contact stress between the bodies (based on the apparent area of contact) is quite low.

K.2.2 Regime II

As the normal stress is increased, the peaks of the asperities of average height come into contact. Again, these asperities deform plastically under increasing load. Since most of the asperities have made at least initial contact, the area of contact increases with normal stress, generally linearly. Regime II is considered to exist up to the point where the material deformation of the individual asperities begins to interfere with that of adjacent asperities. (The deformation of a typical asperities during Regime II is shown in Section K.3). Again, the actual contact stress between asperities is between two and three times the yield strength, depending on the geometry of the asperity. However, the contact stress between the components (based on the apparent area of contact) is most probably less than or at most equal to the yield strength of the weaker material.

K.2.3 Regime III

As the normal stress increases such that the material "pushed off the top" of one asperity interfers with the pile of material from the next asperity, the deformation mode becomes more complex. The process is further complicated by the strain hardening of the softer material. The increase in area of contact with increasing stress begins to diminish. Bulk flow of the material begins. The amount of work required to produce greater mating rises precipitously.

K.2.4 Regime IV

Further increase in normal stress causes the weaker material to flow in bulk. Physical dimensions of the piece change. Should the physical dimensions change such that the apparent area of contact becomes larger, then the true area of contact will increase. However, bulk flow of the weaker material does not appreciably alter the true area of contact. After a certain amount of surface deformation, the asperity stress and strain distribution tends to remain constant. Particularly, if the harder material has asperities, it is virtually impossible to force the softer material into contact with the harder material at the base of these asperities. Infinite stress would be required to accomplish that.

K.2.5 Regime V

When the softer material has deformed in bulk to a certain degree, then the strain hardening and the geometrical contraints will cause that body to appear nearly rigid, and the harder material may flow. However, the deformation of the harder material will be bulk flow, since the surface asperities are already "protected" by their partial contact with the softer material. Hence, should the normal stresses be such that this latter regime exists, the effort may largely be wasted if the intent is to increase true area of contact.

K.3 The Elastic-Plastic Compression Phenomenon

In cases where a small piece of metal with a low yield strength and little potential strain hardening is compressed between two larger pieces of metal with high yield strengths, it is possible to assume that the larger pieces are rigid. As the loads become very high or when strain hardening is appreciable, the assumption becomes less valid.

The above assumption can be used in qualitatively explaining the phenomena which occur during mating of metals. As an approximation, a quantitative explanation can also be given.

For the purposes of discussion and understanding, plane strain is assumed; thus the directions of possible metal flow are reduced to two only. Because of the inability of the metal to flow in both directions laterally under a normal load, the normal stress at which it yields will be higher than in a conventional compression test where no lateral constraints are used.

It can be shown * that:

$$S = Y/\sqrt{1 - \nu + \nu^2} \qquad (1)$$

where S is the normal stress to produce incipient yield, Y is the yield strength as gained by conventional testing, and ν is Poisson's ratio. The above is based on the von Mises yield criterion. For metals ($\nu = 0.3$),

$$S = 1.127Y \qquad (2)$$

A Tresca yield condition will give

$$S = Y \qquad (3)$$

Hence, since the experimental data for ductile metals lies between the two yield criteria, but closer to the von Mises, we should expect yielding to occur in the specimen at about

$$S \sim 1.1Y \qquad (3)$$

Besides the yielding of the softer body in bulk which should begin at a load described above, it must be recognized that at the interface between the surface of the rigid body and the surface of the softer body, asperities exist on both, and the area of contact between is initially small and grows with increased normal load. For a phenomenological explanation of the different phases of deformation which the softer material experiences, one can consider the softer body to have a certain surface geometry and to be a plastic-rigid body; i.e., the body is rigid until sufficient stress is produced at a point to cause plasticity. For that stress level and above, the material at that point is purely plastic. If the surface of the purely rigid body is devoid of irregularities (approximately the diamond burnished case) then the following model can be surmised to explain the softer body deformation.

*R. Hill, The Mathematical Theory of Plasticity, Oxford University Press, 1956.

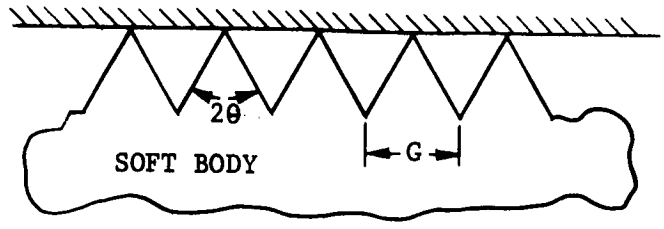


FIGURE K-1. Model of Surface Asperities

The wedge shape, while not being exact, is a close approximation for the machined surface. If one such wedge is considered, it can be shown that the wedge deforms as shown below:

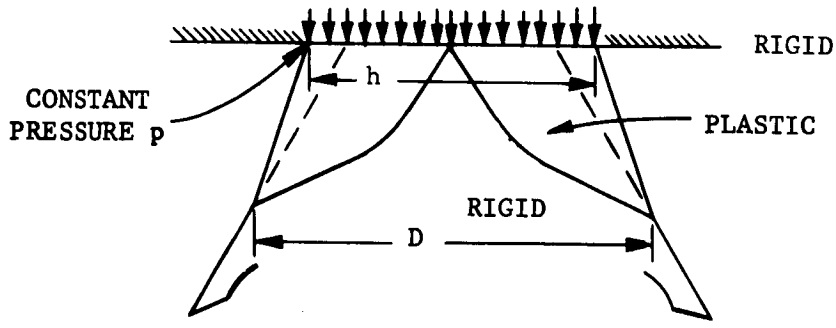


FIGURE K-2. Plastic Wedge Deformation

The load bearing capacity of a deformation is known to be as shown in Figure K-3.

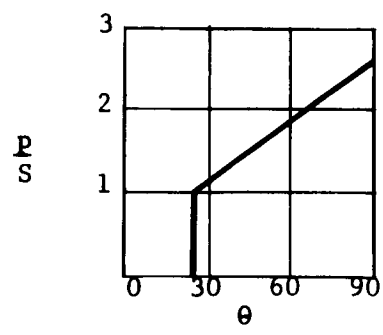


FIGURE K-3. Variation of Wedge Stress With Wedge Angle *

Hence for large angles θ (which are conventional on finely ground surfaces),

$$p \sim 2.63S \qquad (4)$$

*R. Hill, The Mathematical Theory of Plasticity, Oxford University Press, 1956.

As θ becomes close to 180°, the ratio of area over which the pressure acts to the total area of plastic deformation, h/D, becomes *

$$h/D \sim 0.5 \tag{5}$$

Thus, it would appear that up to the load where D approaches the surface pitch G, (and interference between plastic regions from adjacent wedges occurs), we may predict the ratio h/G, the ratio of contact area to apparent area. The average normal stress acting the bulk of the soft body will be $(h/G)\cdot p$. Since p is a constant for a given angle θ, then h will be linearly increasing with the total load F per unit thickness,

$$F \sim hp, \tag{6}$$

and the increase in area of contact (h/G) will be linear with the mean normal stress Q.

$$Q = F/G = hp/G \tag{7}$$

At the point where h/D tends to h/G, then the average normal stress Q becomes

$$Q = 1.315S \tag{8}$$

for large angles θ. Since this average stress would produce incipient bulk yielding, the phenomenon can be explained by this model only to the point where, from *

$$h/G \sim .4 \tag{9}$$

However, from the above, the earlier stages of mating are shown to produce a linearly increasing area of contact with increasing load. For the real case, this explanation is affected by the asperity shape being other than purely wedge shaped, the rigid surface having asperities, and the plastic material having an amount of strain hardening. We can, however, isolate a range of nominal normal stress in which,

a) only surface plastic deformation is produced, not bulk plastic deformation, and

b) the increase in area with increasing stress is roughly linear.

---

*R. Hill, The Mathematical Theory of Plasticity, Oxford University Press, 1956.

K.4 Experiments Performed

Two separate types of experiments have been performed. In the first, annular flat metal rings have been compressed between two flat thick metal bodies of greater strength than the metal rings. Two different metals were utilized for the anvils between which the rings were compressed — 347 stainless steel (0.2% yield strength = 39,175 psi) and 2024 (T4) aluminum (0.2% yield strength = 51,750 psi). Four materials of varying yield strength were used for the rings. The materials and their strength properties are listed below:

Material	0.2% Yield Strength	Tensile Strength
Lead, 99.90% pure, ASTM B29.55	1425 psi	2000 psi
Aluminum, 99.6% pure 1060-0 Temper, ASTM 209-61	4400 psi	10000 psi
Copper, 00.9% pure, Fed. Spec. QQ-C-576	7770 psi	29100 psi
Nickel, 99.4% pure, ASTM B160-58T	13230 psi	45300 psi

Each ring was 0.060 inch thick, had an outside diameter of 1.187 inches, and had an inside diameter of 0.937 inch, thus leaving a ring width of 0.125 inch and a width to thickness ratio of approximately 2:1. Hence, the compression of the rings between the anvils is approximately a plane strain problem; the plastic bulk flow of the rings radially approximates the Prandtl plasticity problem.

Prior to the ring compression, the surface finish of both rings and anvil was recorded, both by Talysurf profilometer and by interference photography. The ring was then compressed normally to a final nominal stress level. Upon removal from the anvils, the ring was photographed, and in many cases, "Talysurfed" again. The anvils were also photographed and a profile taken. Hence, the alteration in surface profiles of both ring and anvils was found. The degree of mating between them was found by inspection of photographs of both surfaces at a given location on the surfaces. The photographs shown herein are results from these experiments. The scratches apparent on the photos are scribe marks made initially on the anvil surfaces in order to locate common points on both surfaces after removal of the ring from the anvil. Tests such as those described above are referred herein as the A Type.

Also performed were a series of experiments in which no ring was used between the anvils. Rather, the anvils were machined such that high relief rings remained on both anvils. The inside and outside radii are the same as those on the rings. The height of each high relief ring was 0.030 inch, thus allowing a geometry similar to the ring experiments to be effected. The anvils were pressed together normally, and the surfaces recorded both before and after the test as in the previous case. Tests of this type are called B Type. The material flow in this case differs from the previous case in that no shear stresses exist at the interface due to the complete symmetry.

In both tests, the compression was monitored by linear differential transformers. The force to compress the system was applied from a tensile testing machine. Hence, the final load, causing the resultant degree of mating was known exactly.

K.5 B Type Tests, Direct Metal to Metal Surface Compression

The results of five such experiments are cited herein. The first case is that where the anvil material was 347 stainless steel. The surfaces were prepared by grinding, generally in a radial direction. The resultant finish was a 40 micro-inch rms surface with a directionality slightly off radial with a small swirl effect. With such a finish, the asperities on the opposing surfaces would not lie parallel, but rather cross at about a 20 degree angle. The original surface profile, as recorded in a tangential direction, is shown in Figure K-4. The normal stress applied to the system was 95,600 psi, or 2.44 times the yield strength of the steel. The alteration of the surface profile was significant, and is shown in Figure K-5. Figure K-6 shows a photomicrograph of one of the surfaces after the compression. The lines running lower left to upper right are the original surface finish lines. The lines running from upper left to lower right are those imposed upon the surface from the compression with the other surface. It can be noted that the density of the latter lines is much less than that of the former, thus evidencing that the mating was not complete. Figure K-7shows a magnified portion of the profile, with interference lines (which show relative depth). It can be noted that the only lines present in the area shown are those from the original grinding. Thus, in this area, no contact was made, even under a nominal stress equal to 2.44 times the yield stress.

The next three tests shown involve 2024 (T4) aluminum anvils circumferentially machined with a radial pitch of 0.001 inch with a wedge shaped tool. The resultant surface finish was approximately 100 micro-inches rms. The surfaces were compressed three times successively. The original surface profile (in a radial direction) is shown in Figure K-8. Figure K-9 shows the profile after a compression test (with a similar profile) under a normal stress of 67,300 psi, equal to 1.3 times the aluminum yield strength. It will be noted that virtually no damage was done to the surface, except for a slight rounding off of asperity peaks. It can be concluded from this that very little true area of contact was effected. Figure K-10 shows the same surface after the second compression, this time with a normal stress of 74,000 psi, or 1.43 times the yield strength. Again, very little surface damage (and, hence, true area of contact) was attained. The final compression, between the same surfaces, is shown in Figure K-11. The normal stress was 86,500 psi, equal to 1.67 times the yield strength. In this case, the anvil high relief ring underwent some plastic deformation in bulk flow. However, the surface profile is essentially the same. Hence, even at 1.67 times the yield strength with this type of surface finish, the true area of contact is small. It can be calculated from the Talysurf profiles that the asperity included angle is approximately 174 degrees. Exact comparison of asperities in the Talysurf traces is impossible since the profiles were not necessarily made at the same point about the periphery of the high relief ring.

The final metal to metal direct compression test reported herein is that of the 2024 (T4) aluminum surfaces diamond burnished circumferentially to a surface finish of approximately 2 or 3 micro-inches, rms. The resultant original finishes were essentially flat with periodic circumferential narrow low relief grooves. The surfaces were about 95% flat surfaces and the grooves representing less than 5% of the total area. Figures K-12 and K-13 show high

magnification photographs of the opposing surfaces after compression together under a normal stress of 43,000 psi (or 0.84 times the yield stress). At the point photographed, a flat piece of foreign matter had been present in the system, and left a low relief mark on each surface. The important information to be noted is that on this type of surface, where the roughness is entirely due to low relief cut, there is really no way for complete area of contact to be effected. There is no material available from either surface to fill up the grooves in the other. Should the stress be increased to well above the yield stress, the material would flow in bulk, while the surface roughness would be preserved.

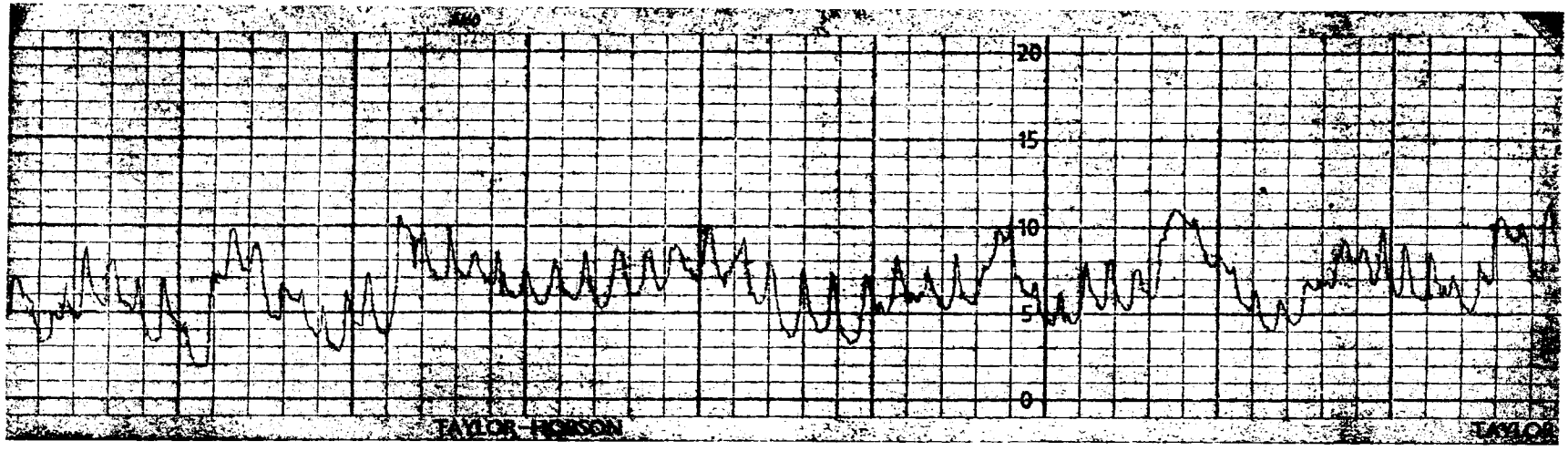


Figure K-4

Original Ground Finish On 347 Stainless Steel Surface; Prior To Mating With Similar Surface Under Normal Load (Scale: height-each small division, 20 micro-inches; lateral each small division, 0.002 inches)

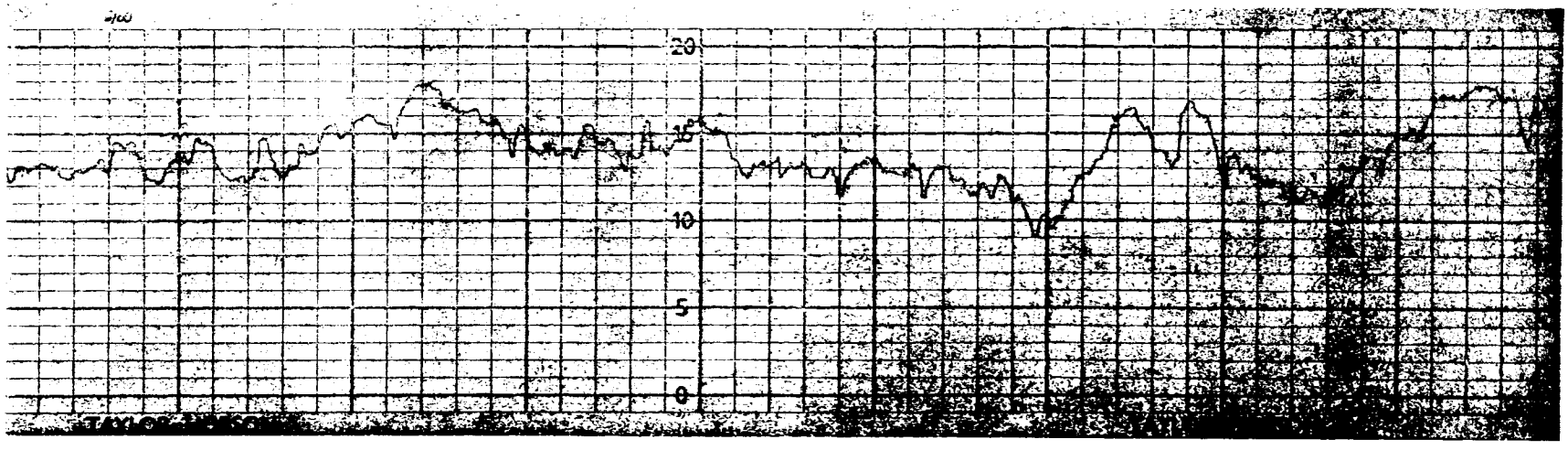


Figure K-5

Surface Profile Of 347 Stainless Steel Surface After Mating With Another Surface Of Same Type As In Figure K-4 (Scale: height-each small division, 20 micro-inches; lateral each small division, 0.002 inches)

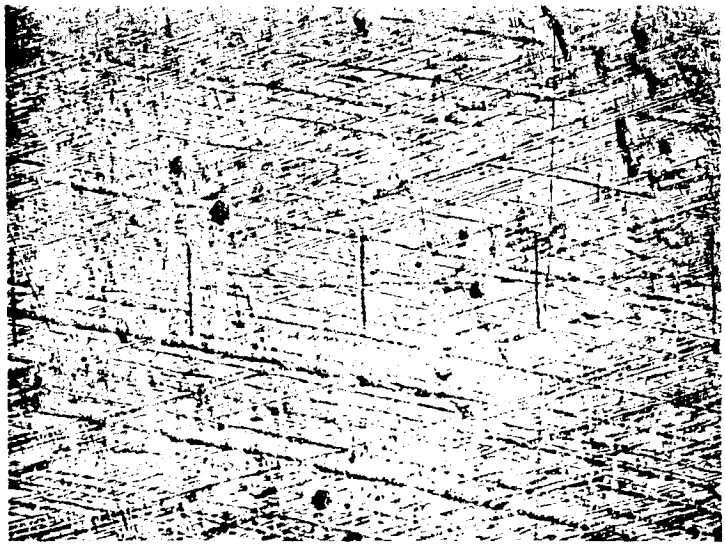


Figure K-6

Photo Micrograph of 347 Stainless Steel Surface After Mating With Similarly Ground Stainless Steel Surface Under A Stress Equal To 2.44 σ_y. (Scale: 0.0125 inch between Scale Marks)

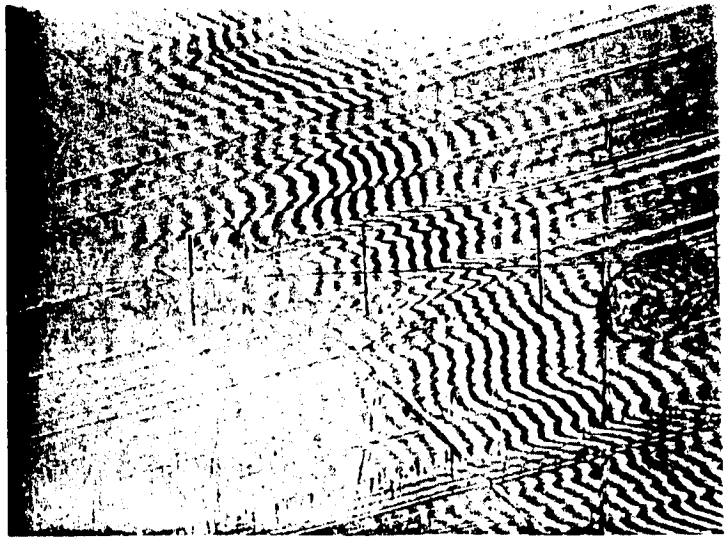


Figure K-7

Magnification of Same Surface As In Figure K-6, Interference Lines Added (Scale: 0.00194 inch between scale marks)

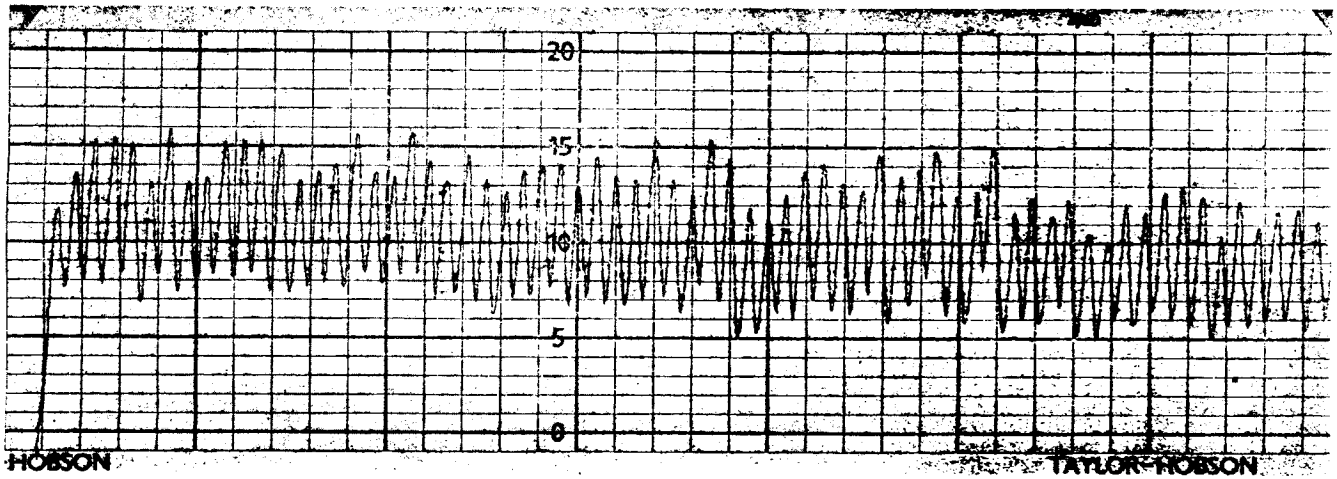


Figure K-8

Original Surface Profile Of Aluminum Anvil, Prior To Mating With Similar Surface (Scale: height-each small division, 50 micro-inches; lateral each small division, 0.002 inches)

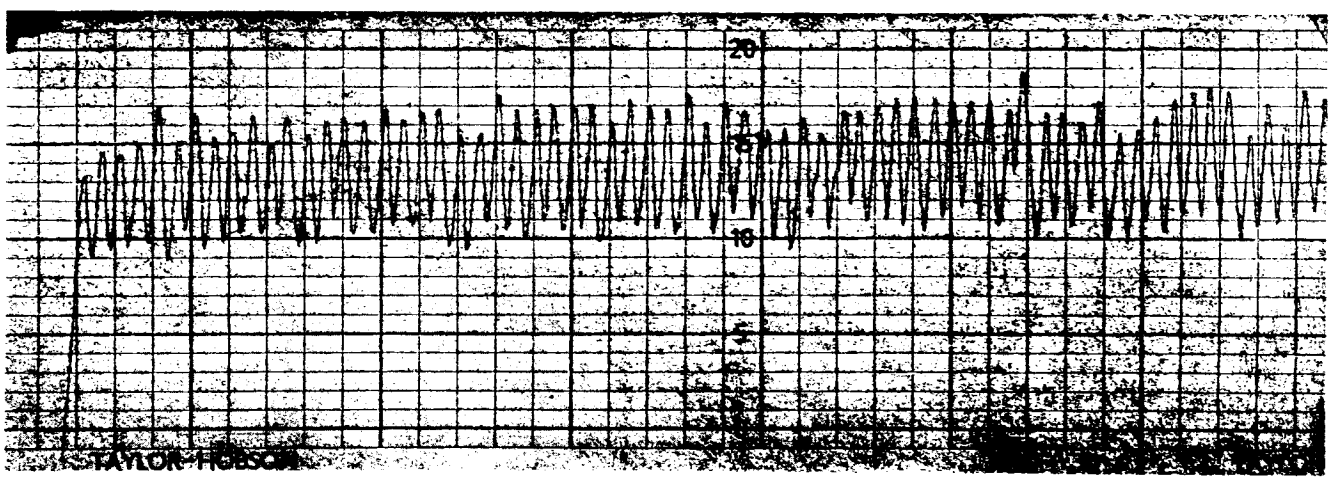


Figure K-9

Surface Finish Of Aluminum Shown In Figure K-8 After First Compression Test (Scale: same as Figure K-8)

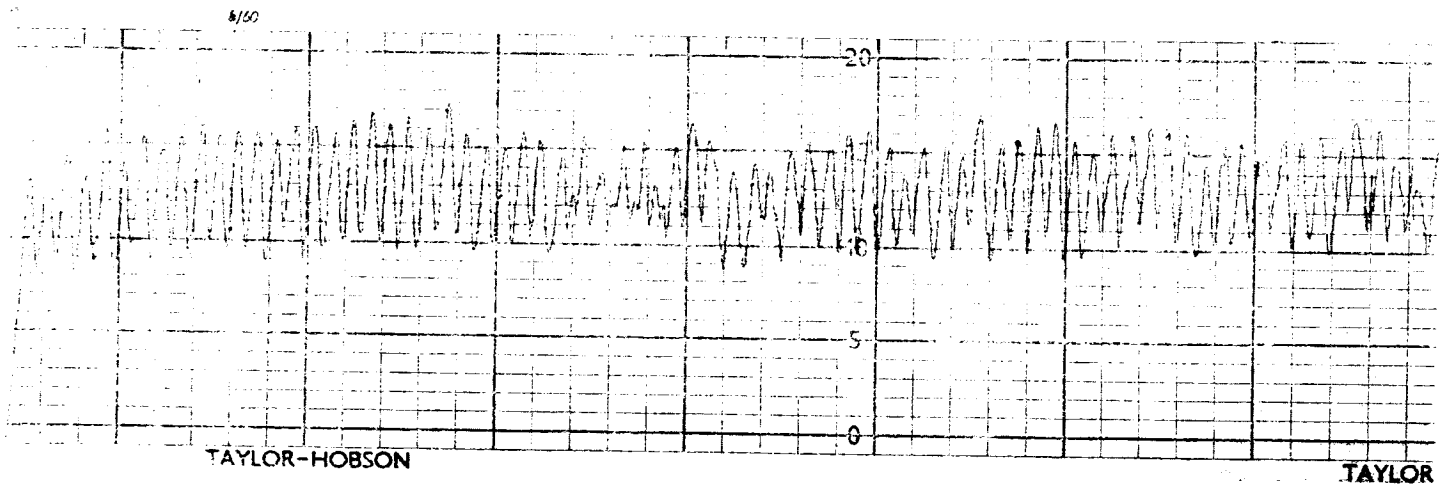


Figure K-10

Surface Profile Of Aluminum After Second Compression Test
(Scale: same as Figure K-9)

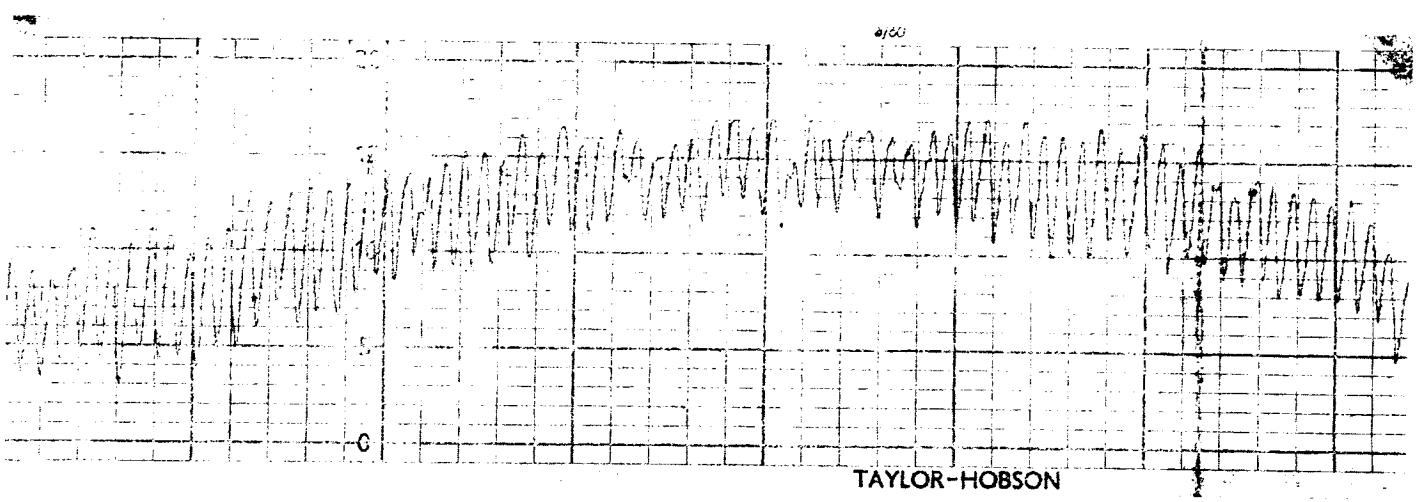


Figure K-11

Surface Profile Of Aluminum After Third Compression Test
(Scale: same as Figure K-10)

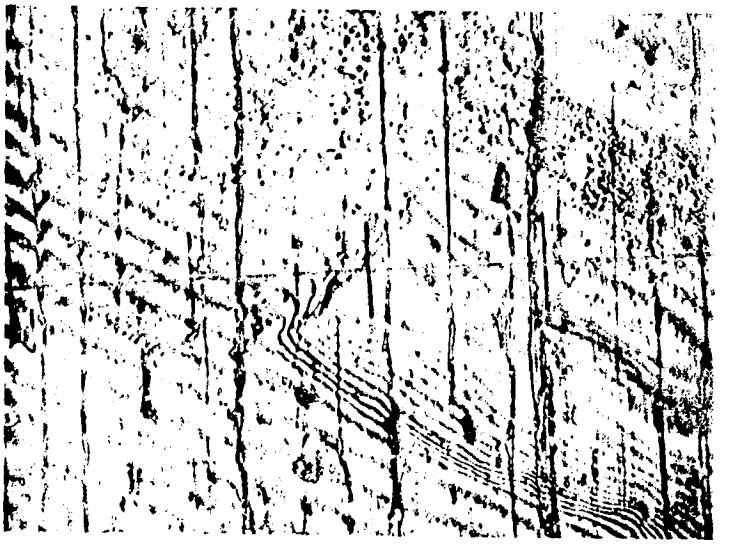


Figure K-12

Diamond Burnished Surface of 2024 (T4) Aluminum After Compression with Similarly Finished Specimen; Note How Relief Grooves Due To The Burnishing Process Running Vertically On The Photo.
(Scale: 0.00194 inches between scale marks)

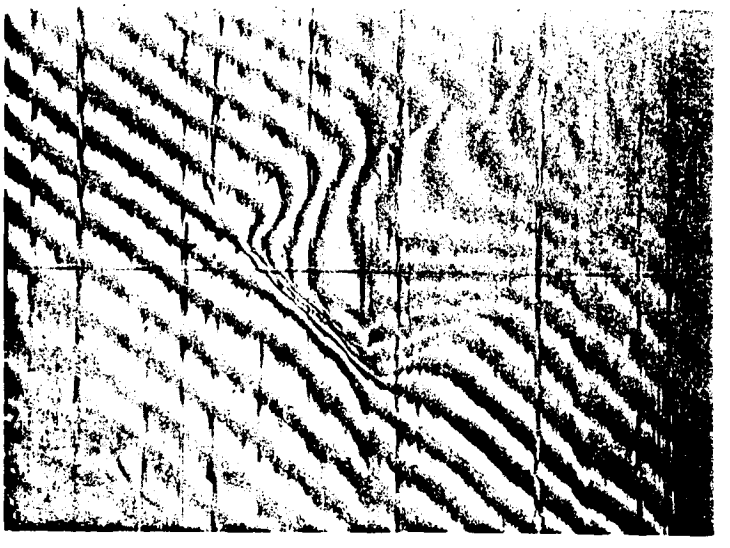


Figure K-13

Aluminum Surface Mated With That Shown In Figure K-12

K.6 A Type Tests, Soft Flat Rings Compressed Between Anvils

The results of seven separate tests involving soft metal flat annular rings compressed between rather rigid anvils are presented; each displays the difficulty found in attaining a large degree of area of contact between mated surfaces. One test shows that the degree of mating in a Prandtl plasticity type arrangement is not uniform across the width of the softer material.

The first four tests involved the use of anvils with extremely smooth surfaces; the surfaces were diamond burnished such that the roughness was approximately 4 micro-inches, rms. As before, the diamond burnished surface is evidenced by a generally flat quality with periodic low relief grooves which are very narrow. Four different ring materials were used, lead, aluminum, copper, and nickel.

The first test is that of 347 stainless steel anvil and a lead ring. Figures K-14 and K-15 show common points on both the stainless steel surface and the lead surface after the mating experiment. The stainless steel surface, of course, is identical in condition to that prior to the mating experiment. The lead was initially quite rough, being cut out of flat stock. The stress applied to the system was 2815 psi, which is equivilent to 1.973 times the yield strength of the lead. It is to be noted that in order to acquire complete sealing, the lead would have to be pushed into the low relief grooves in the stainless steel. As can be noted by comparison of the two figures, no ridges exist on the lead, even though the low relief grooves are quite apparent at the same magnification on the stainless steel surface. Thus, even for lead, which is extremely soft, perfect mating is not possible for such combinations of surface finish. However, in spite of the low relief grooves on the stainless steel, the degree of mating in this particular case is probably above 95%.

The second test utilized a similar diamond burnished aluminum anvil along with a soft aluminum ring. The aluminum ring was circumferentially machined with a radial pitch of 0.001 inch with a wedge shaped tool. The resultant original surface finish on the aluminum surfaces was about 200 micro-inches, peak to valley. The surfaces were pressed together with a stress of 12520 psi, about 2.85 times the yield stress of the soft aluminum (well below the yield stress of the 2024 aluminum). Figures K-16 and K-17 show the resultant degree of mating. The dark lines on the photo of the soft aluminum designate the deep grooves between asperities. The light lines are those tops of asperities which have been flattened and were in complete contact with the harder aluminum. An estimate of the area of contact attained in this system is approximately 50%.

Figures K-18 and K-19 show the results of diamond burnished 2024 aluminum anvils mated with a circumferentially machined copper ring. The copper ring was machined with a radial pitch of 0.001 inch with a wedge shaped tool. The surfaces were forced together with a normal stress of 27875 psi, which is about 3.59 times the yield stress of the copper (still well below the yield stress of the 2024 aluminum). As can be noted, the tops of the asperities of the copper have been flattened are in intimate contact with the aluminum. However, the grooves remain, and only about 50% contact was effected, even with the high stress level used.

The final test done using diamond burnished anvils was that shown in Figures K-20 and K-21. Stainless steel anvils and a nickel ring, circumferentially machined with a resultant finish of about 40 micro-inches, rms, were forced together under a normal stress of 39150 psi, which is approximately 2.96 times the yield stress of the nickel. (This stress level is approximately equal to the yield stress of the stainless steel.) The result, as can be determined from Figure K-21 is that only about 30% area of contact has been attained. Here, where a strain hardening softer material has been used, the mating between surfaces is even more difficult to attain.

The final three tests employed anvils which had been radially ground and soft rings which had been circumferentially machined, thus producing a system with generally orthogonal asperity directions.

The first, shown in Figures K-22 and K-23 utilized a copper ring machined with a radial pitch of 0.001 inch with a wedge shaped tool. The resultant finish was approximately 120 micro-inches peak to valley. The radially ground stainless steel anvil finishes were about 20 micro-inches, rms. The stress applied to the system was 20,970 psi, which is about 2.7 times the yield strength of the copper. The photos show that the degree of mating is about 70%. It can be seen from this that when the original asperities were orthogonal, the stress concentrations are greater and more mating results from a given stress (as compared with previous tests with similar stresses).

Figures K-24 and K-25 show the results of a system subjected to 79,880 psi, or 6.03 times the yield stress of the soft ring, which was nickel. The original nickel circumferentially machined surface was about 150 micro-inches peak to valley. The stainless steel anvil ground finish was about 40 micro-inches, rms. As can be seen, the degree of mating is extremely high, about 80% locally. However, the stress level used was well above the yield stress of the stainless steel, and the fifth regime of material deformation (Section K-2.5) has commenced. Although the surface of the stainless steel was well preserved in this case, the steel underwent definite plastic deformation, and the entire surface under the ring had sunk down appreciable. Even under such an extreme stress level, complete area of contact was impossible. It is also to be noted that nickel, which does strain harden, was the ring material used.

The last A Type test employed a radially ground stainless steel anvil with a 40 micro-inch rms finish. The ring used was aluminum, with a circumferentially machined finish of about 150 micro-inches, peak to valley. The normally applied stress was 15,080 psi, about 3.59 times the yield strength of the aluminum. In this case, a phenomenon which was evident to some degree in several other tests was made extremely obvious. In that the material flow of the ring between the anvils was one directional, i.e., radially inward and outward, the problem was very much like the Prandtl plasticity problem. Shear stresses were present at the interfaces, and the ring underwent shear deformation. The normal stress distribution in such a problem has its highest point at the center of the ring width. The stress drops off near the edge. However, the center part of the ring is well restrained by the remainder of itself. The portion of the ring at the inside and outside edges flows radially against the opposing surfaces. The photos show that while the normal stress is lowest in this region, the combined stresses are such that the mating is extremely good locally. Near the center, the mating has not been effected nearly as well. Figures K-26

through K-29 show the center of the ring; Figures K-30 through K-33 show the mating near the outside edge.

It is to be noted that in such a situation, the voids remaining in the system are entrapped and the gas remaining inside would be extremely difficult to evacuate. Bacteria which exist in those voids, if not killed by the heat treatment during sterilization would be present and perhaps escape when the joint loosened under relaxation of stresses when the system was operational.

Figure K-14

Stainless Steel Diamond Burnished Surface
(Scale: 0.00482 inch between scale marks)

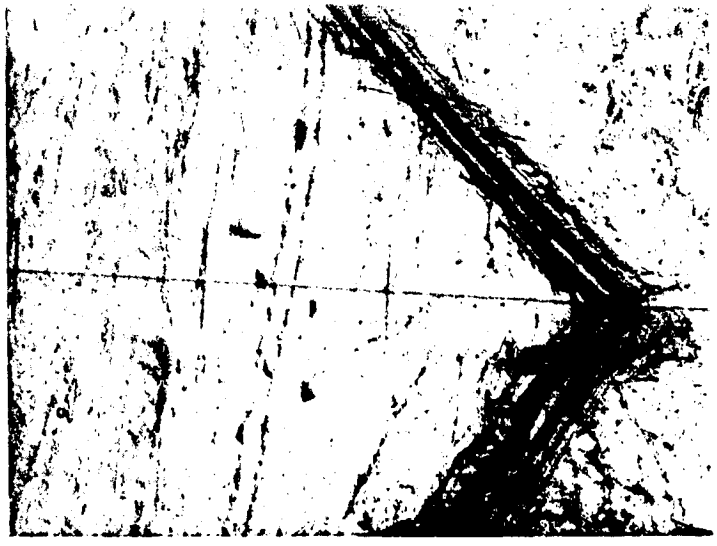


Figure K-15

Lead Surface After Being Pressed Against The Stainless Steel Surface Shown in Figure K-14. (Scale: 0.00482 inch between scale marks)

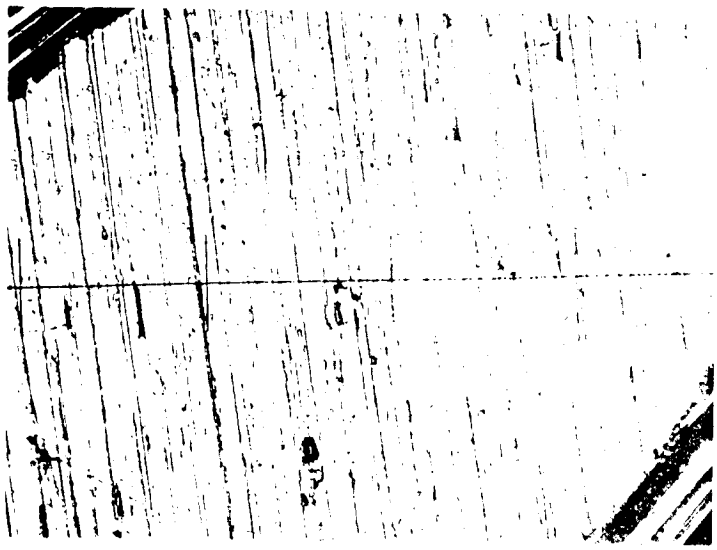


Figure K-16

2024 Aluminum Surface Diamond Burnished (Scale: 0.00482 inch between scale marks)

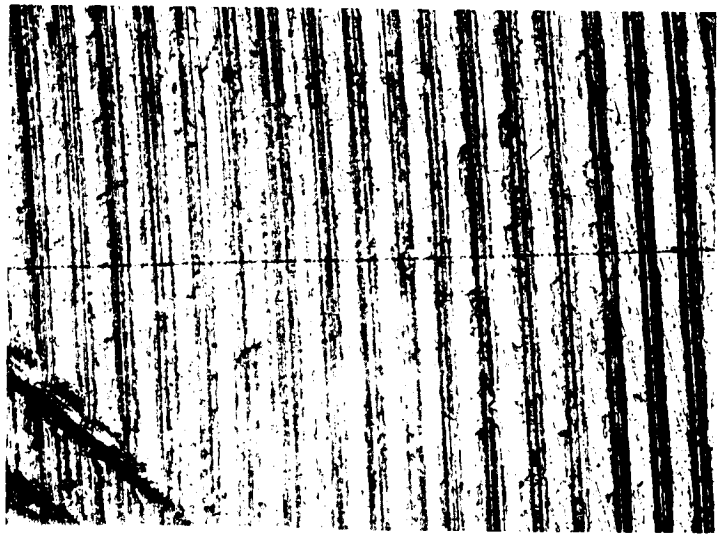


Figure K-17

Soft Aluminum Surface After Mating With Aluminum Surface Above (Scale: 0.00482 inches between scale marks)

Figure K-18

2024 Aluminum Surface Diamond Burnished, Interference Lines Shown (Scale: 0.00194 inches between scale marks)

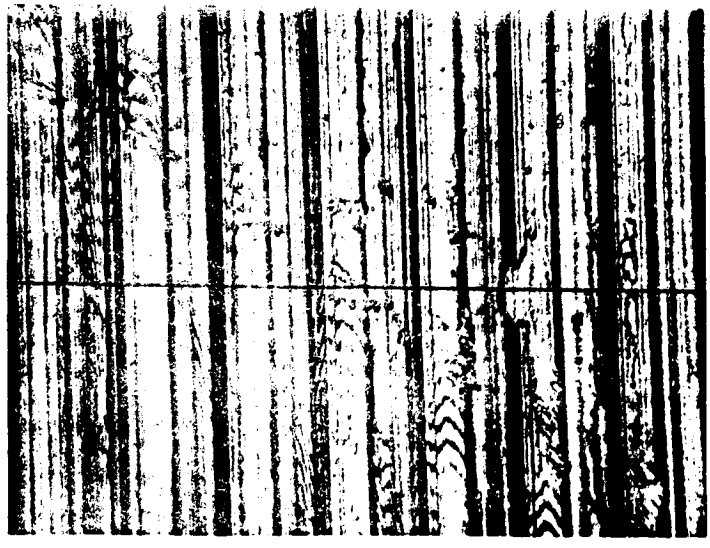


Figure K-19

Circumferentially Machined Copper Surface After Mating With Aluminum Surface of Figure K-19 (Scale: 0.00194 inch between scale marks)

Figure K-20

Diamond Burnished Stainless Steel Surface (Scale: 0.00482 inch between scale marks)

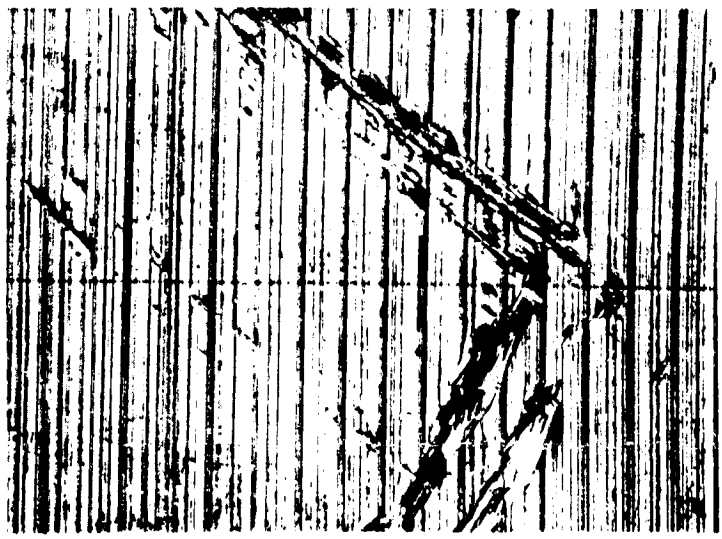


Figure K-21

Nickel Surface, Circumferentially Machined After Mating With The Stainless Steel Surface of Figure K-20. (Scale: 0.00482 inch between scale marks)

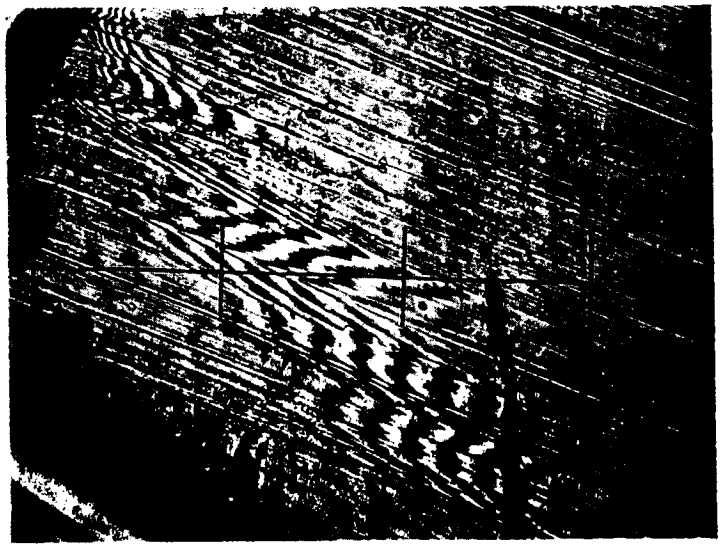


Figure K-22

Stainless Steel Surface, Ground Finish
(Scale: 0.00194 inch between scale marks)

Figure K-23

Copper Surface, Machined After Mating With Stainless Steel Surface of Figure K-22 (Scale: 0.00194 inch between scale marks)

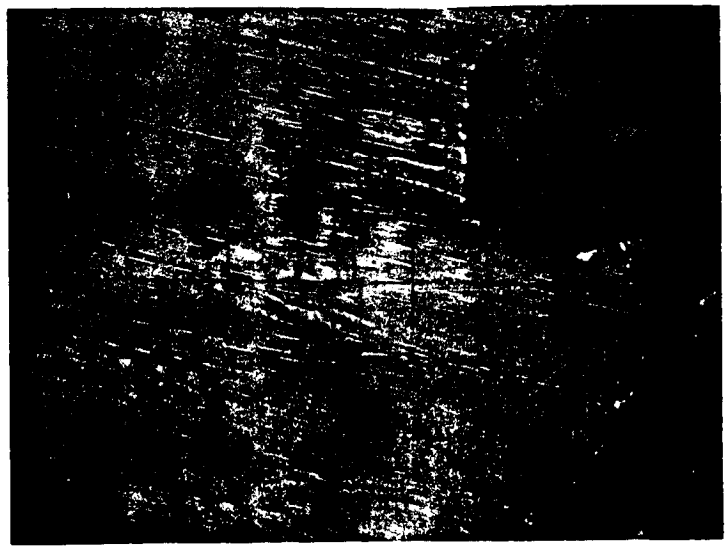


Figure K-24

Ground Stainless Steel Surface Finish
(Scale: 0.00194 inch between scale marks)

Figure K-25

Circumferentially Machined Nickel Surface After Mating With Steel Surface of Figure K-24 (Scale: 0.00194 inch between scale marks)

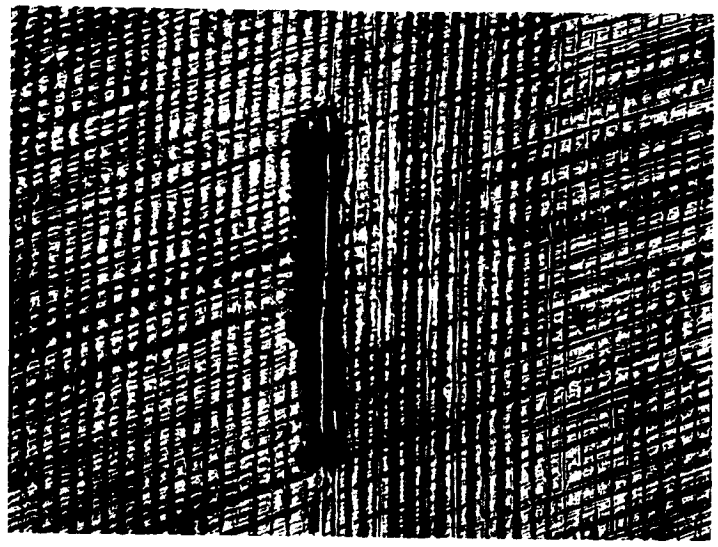


Figure K-26

Aluminum Flat Ring Circumferentially Machined (vertical lines) After Mating With Stainless Steel Surface Radially Ground (near horizontal lines) (Scale: 0.0125 inch between scale marks)
Note: Photo taken at center of ring width.

Figure K-27

Same As Figure K-26 (Scale: 0.00482 inch between scale marks)

Figure K-28

Same As Figure K-26 (Scale: 0.00194 inch between scale marks)

Figure K-29

Same As Figure K-26, With Interference Lines

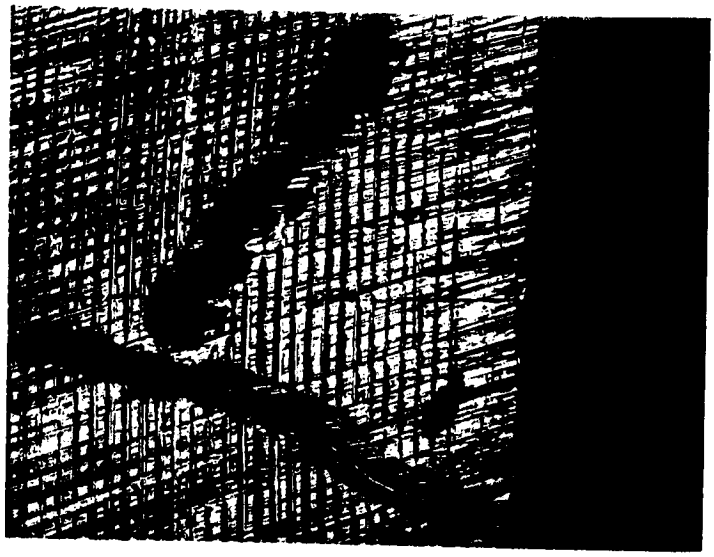


Figure K-30

Aluminum Flat Ring Circumferentially Machined (vertical lines) After Mating With Stainless Steel Surface, Radially Ground (near horizontal lines) (Scale: 0.0125 inch between scale marks)
Note: Photo taken at outside edge of ring.

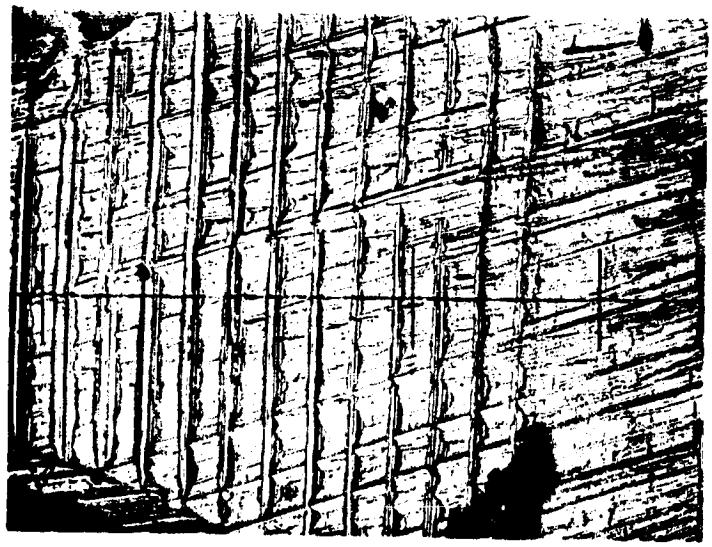


Figure K-31

Same As Figure K-30(Scale: 0.00482 inch between scale marks)

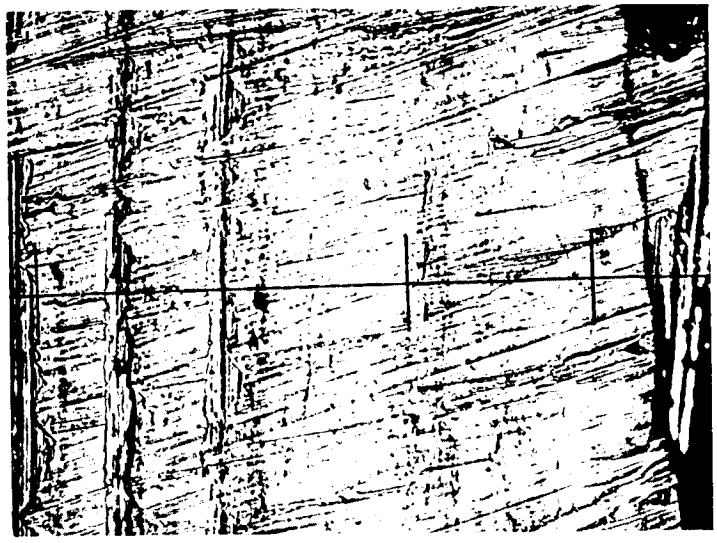


Figure K-32

Same As Figure K-31 (Scale: 0.00194 inch between scale marks)

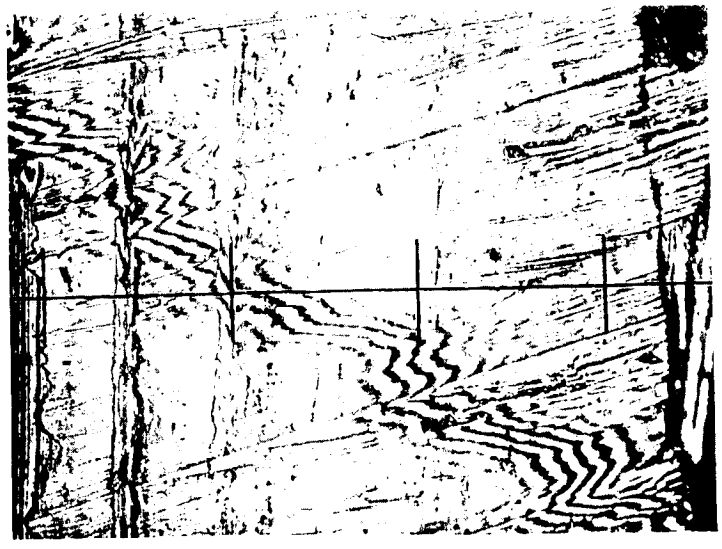


Figure K-33

Same As Figure K-32 With Interference Lines

K.7 Conclusions

1. Complete mating, i.e., 100% real area of contact, is virtually impossible between flat surfaces mated together under normal stress as long as surface asperities exist on either or both surfaces.

2. A strain hardening material is more difficult to mate with another surface under normal stress than a non-strain hardening material.

3. Where shear stresses exist between the surface, the degree of mating is best where the shear stress between the surfaces is greatest, and not necessarily in the area of highest normal stresses.

4. It is virtually impossible to force even soft metals into low relief voids on harder surfaces to the point where intimate contact is made.

5. Once a certain amount of contact is made between surfaces, the local surface stress distribution tends to preclude further mating and the path of least resistance to further deformation is deformation in bulk of the softer body.

K.8 Evaluation of Surface Finishes

A Taylor-Hobson "Talysurf" stylus-type profilograph has been used to obtain profiles of the various finishes used. Such an instrument will yield single-line profiles of surfaces which have as little as two-micro-inch asperities. Not only are traces of the profiles in a given direction gained, but an electronic integrator circuit produces, for any surface of a certain minimum length, a center line average (CLA) roughness. While the "Talysurf" is basically simple to utilize, a thorough understanding of the interpretation of the traces is required.

When inspecting "Talysurf" profilometer traces, it is necessary to insure that:

(1) the direction in which the trace was made is known, and

(2) the different scales present on the trace in the vertical and horizontal directions are understood.

Since the "Talysurf" trace is merely the record of a stylus motion following the contours of the surface in a straight-line travel, the information attained about the surface is limited. Any asperities on either side of the trace are not recorded; nor is there any surety that a trace made parallel to the trace at hand would resemble the recording at hand. Hence, something must be known about the surface prior to producing the trace. The stylus must be moved in the lay direction - the direction across which the asperities run. If a surface has a random asperity distribution, traces in two directions can be made. There is no guarantee that a single trace will yield information indicative of the true character of the surface. For machined parts, and some ground parts, the lay direction is perpendicular to the direction of the tool motion.

The utility of the profilometer tracing lies in the magnification of the record of the stylus displacement in the vertical direction. Since an equal magnification in the travel (horizontal) direction would result in extremely lengthy traces, the horizontal scale is fixed; on all traces the scale is 20:1. Each small division on the paper (0.2 inch) equals 0.01 inch of stylus travel. The vertical magnification can be changed to suit the surface at hand; six magnifications are possible.

Number	Magnification	Each Small Division Equals
1	1,000X	100 microinches
2	2,000X	50 "
3	5,000X	20 "
4	10,000X	10 "
5	20,000X	5 "
6	50,000X	2 "

Hence, for all cases, the vertical magnification is different from the horizontal magnification. The scales utilized are noted for each trace presented herein.

Because of the scale variances, the asperities are much less severe than are shown in the traces. It must be recognized that the enclosed angle for a "peak" will generally be greater than 170°. Hence, while the traces yield a great deal of useful information, they do not represent an actual visual picture of the surface.

With regard to the CLA average, such a characterizing quantity is defined by:

$$CLA = \frac{1}{\ell_o} \int_{x=o}^{x=\ell_o} /y/dx \qquad (1)$$

The CLA system, now an American standard, is sometimes supplemented by the more common rms system. While the CLA value of a surface is automatically computed by the Talysurf, the rms value is not.

The rms value, defined by

$$rms = \frac{1}{\ell_o} \left[\int_{x=o}^{x=\ell_o} y^2 dx \right]^{\frac{1}{2}} \qquad (2)$$

must be gained from a knowledge of the details of the surface profile or be estimated from the CLA value. In both cases, y is the height of the surface from the mean line describing the surface.

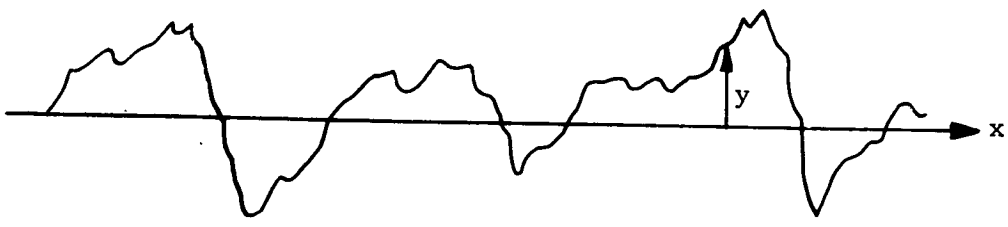
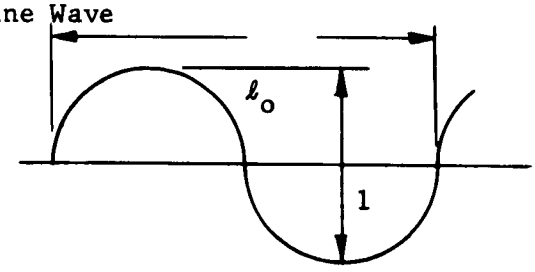
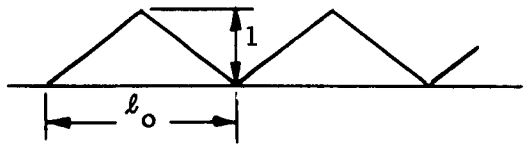
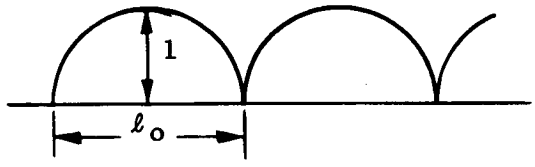
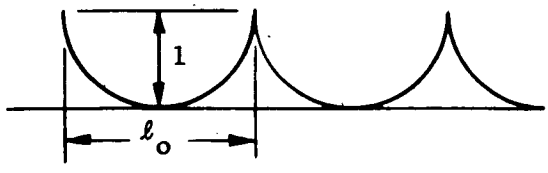
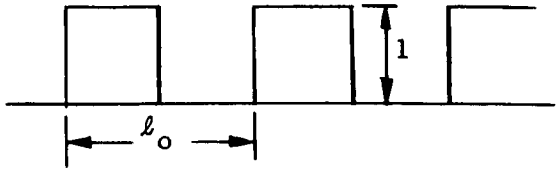


FIGURE K-34. Typical One Dimensional Surface Profile

Following are listed in Table K-I the CLA and rms values from several mathematically described surface geometries. The variations between rms and CLA are given as aids in estimating rms values from CLA values.

TABLE K-I. Mathematical Surface Profiles

Profile	CLA	rms	rms CLA
(1) Sine Wave 	0.318	0.354	11.2%
(2) Saw Tooth 	0.25	0.289	15.6%
(3) Parabola 	0.256	0.298	16.3%
(4) Parabola 	0.256	0.298	16.3%
(5) Step Function 	0.5	0.5	0%
(6) Gaussian Distribution of Surface Asperities	1.0σ*	0.796σ	25.8%

*σ = Standard deviation

K.9 Interpretation of Interference Lines

The apparatus used for interference photography is the Zeiss interference microscope with a polaroid camera attachment. Both ordinary magnified photographs and magnified interference photographs are possible.

In order to precisely locate matching points on the two mated surfaces, local scribe marks were made in the surface profile with a stylus. Such a mark becomes noticeable on the softer material as a result of the plastic flow; photos in the region of this mark are then compared.

In order to interpret the interference photographs, it is necessary to understand the phenomenon which causes the interference lines. Considering light to propagate as a wave, we can speak of the frequency of the light and of the wave length of the light. If two light sources of the same frequency and magnitude are directed along the same path, then the magnitude of the resultant wave produced by the addition of the two waves will depend on the phase between the original two waves. The addition of two light waves to produce a new wave of different magnitude characteristics is called interference. If, by optical devices, the light from a single source is split, then the possibility of interference exists. By reflecting one wave from the object to be seen, and the other from a mirror at a known distance and angle to the object, then the two reflected waves will interfere in a manner so as to produce the interference profile of the object surface.

The resultant lines can be envisioned as contour lines on the object surface as seen if the surface were slightly off normal to the line of vision. Such is shown for a flat surface in Figure K-35. The bands formed can be viewed as contour lines of the surface seen from a direction perpendicular to the parallel planes shown. The vertical distance between planes (and hence bands) is one-half the wave length of the light used. As the angle between line of sight and surface changes, the distance between bands varies; however, the vertical distance between the bands taken as contour lines remains J/2 (one-half the wave length of the light).

Were a groove to exist in the surface, then the bands would show up as in Figure K-36. Again, the direction of light is perpendicular to the planes shown, and the bands are J/2 apart in a direction perpendicular to the line of sight.

Figure K-36 also illustrates the interpretation of the interference bands. Measuring from the center of a dark band to the center of the same band in a location where the band has changed direction, we can note what portion of a half wave length of light the band has displaced. In Figure K-36, the displacement is (0.3 x J/2). Thus, the depth of the groove is 0.3 x J/2).

If the surface had been tipped in a direction 90° to its present direction, a different band pattern would emerge.

Both surface topograph shape and depth can be ascertained by this method.

For the results shown in this report, white light has been used; a value of 11.8 microinches can be taken as a half wave length.

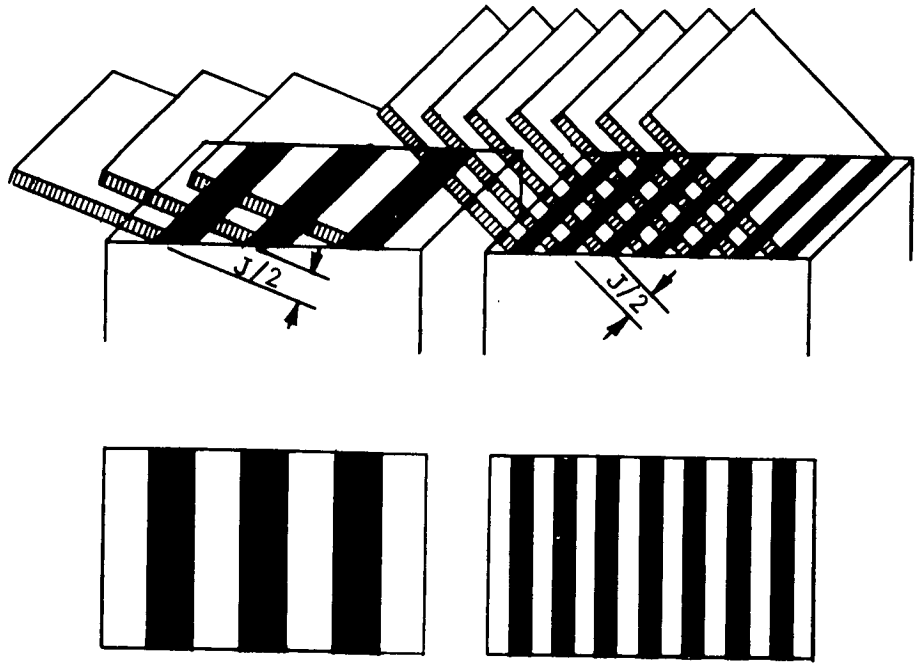


FIGURE K-35 Interference Pattern for Flat Surface

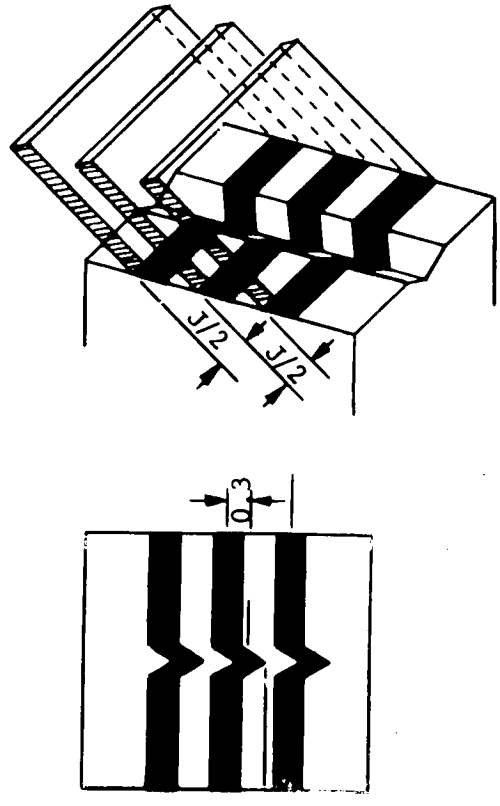


FIGURE K-36 Interference Patterns for Flat Surface with Groove

APPENDIX L

CLEAN ROOMS AND CLEAN WORK STATIONS

The work summarized in this Appendix was performed by the Spacecraft Department of the General Electric Missile and Space Division on a number of Studies and Contracts including Contract NAS W-696 (Voyager Design Study) for the NASA Office of Space Sciences.

L.1.4 CONVENTIONAL CLEAN ROOMS

A conventional clean room represents an attempt to maintain a high degree of cleanliness within an entire enclosure. Since air flow inside such rooms is not always predictable a great deal of emphasis is placed on the avoidance of particle generation. This attempt to minimize the sources of contamination increases enormously the costs of construction and operation. There was a time when designers went so far as to utilize stainless steel in the fabrication of wall panels. A price of $60, $100, and more per square foot of room was not uncommon. Besides material costs, operational expenses mount due to the elaborate procedures and special clothing required to prevent the transfer of dirt from the outside environment to the interior. For example, each man spends from 30 to 45 minutes of each 8-hour work period in entering and leaving the facility.

His clean room garments cost approximately $3.00 weekly for laundering and rental.* In addition, power consumption is costly. The floor plan shown in Figure L.1-1 requires 22,500 cubic feet of air per minute and roughly 23 tons of refrigeration to meet Air Force specifications. These environmental conditions must be maintained twenty-four (24) hours daily for 365 days each year. The air handling equipment is never shut down except in an emergency. Janitorial service is also an expensive item. Two mean are employed full-time with the responsibility of maintaining clean conditions.

*The cost of clean room clothing and entry time has been estimated by NAMES, Olmsted AFB, Pa., to be approximately $1600 a year per worker.

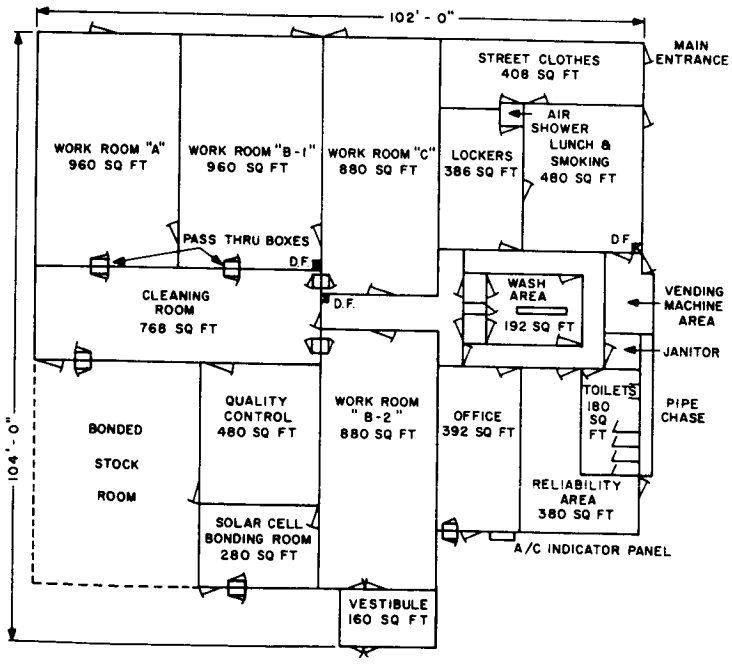


Figure L.1-1. Clean Room Floor Plan (Ref. 4)

Notice that the Air Force specifies plus or minus temperature deviation and emphasizes the rate of temperature change from an established level. This parameter is considered more significant than the range of temperature variation established by the Federal Standard.

Similarly, T.O. 00-25-203 defines the Air Force Standard Clean Work Station under two sets of conditions, operational and "at rest." Table L.1-2 outlines these conditions in comparison with the Federal Standard:

TABLE L.1-2. CLEANLINESS LEVELS IN CLEAN WORK STATIONS

	Federal Standard Class 100	AF Standard Clean Work Station (Operational)	AF Standard Clean Work Station (At Rest)
Particles/Cubic Foot equal to or greater than 0.5 microns	100	1000	100

The clean work station as defined here is equivalent to the Class IV described in the previous T.O. of February 7, 1962.

Nothing in the new Air Force T.O. corresponds to Federal Standard Class 10000.

L.1.3 CLEAN ROOM DESIGN

Clean rooms, themselves, vary greatly in basic design. The conventional type, or the non-laminar flow clean room, was the first to find application in industry. More recently, laminar flow designs have come into being. These laminar flow units take one of two general forms, depending on how the air is introduced. In the case of the down-flow room the incoming air filter bank covers the entire ceiling; the cross-flow room, on the other hand, has one entire wall that serves as the air inlet. Clean rooms also differ in materials of construction. Some have rigid walls; others are flexible, plastic enclosures. The former are apt to be fixed, permanent installations; the latter, likewise, may remain in one location, but, more often, they are intended as portable units.

Similarly, clean work stations employ non-laminar and laminar air flow. The Baker Sterilshield is a good example of the non-laminar flow enclosures. The Whitfield type of bench utilizes the laminar flow principle.

APPENDIX L

CLEAN ROOMS AND CLEAN WORK STATIONS

L.1 CLEAN ROOMS

L.1.1 DEFINITION

A space in which airborne contamination, and if needed, temperature and humidity, is controlled to a far higher degree than conventional air conditioned areas is known either as a clean room or a clean work station.

L.1.2 CLEAN ROOM CLASSES

The Federal Standard No. 209 (issued December 16, 1963) defines three classes of clean rooms. These classifications are based on particle count with a maximum number of particles permissible of 0.5 microns or 5.0 microns and larger, and are established as operational levels. (Operational levels are dust counts taken in a facility during any activity period.)

The three classes of clean rooms are designated as Class 100, Class 10000, and Class 100000.

> Class 100 limits the particle count to a total of 100 particles per cubic foot 0.5 microns and larger.
>
> Class 10000 requires that the particle count not exceed 10000 particles per cubic foot 0.5 microns and larger, or 65 particles per cubic foot 5.0 microns and larger.
>
> Class 100000 has no more than a total of 100000 particles per cubic foot 0.5 microns and larger, or 700 particles per cubic foot 5.0 microns and larger.

These federal standards differ in some respects from the most recent Air Force specification, entitled "Standards and Guidelines for the Design and Operation of Clean Rooms and Clean Work Stations." This latest Technical Order (T.O. 00-25-203, issued July 1, 1963) imposes standards on Air Force activities and contractors performing maintenance and overhaul contractual work for the USAF, and, in addition, provides guidelines for achieving these operating standards. On the whole, the Federal Standard is more stringent in its provisions for the operation of clean work stations.

The new T.O. 00-25-203 defines the Air Force Standard Clean Room and the Air Force Standard Clean Work Station.

The Air Force Standard Clean Room is equivalent to the Federal Standard Class 100000 as far as dust count is concerned and replaces the Class II and Class III categories established by the previous T.O. of February 7, 1962. But the Air Force document goes one step further and defines the Air Force Standard Clean Room under two sets of conditions, operational and at rest. The design standard is the "at rest" standard. These further details are shown in Table L.1-1 below:

TABLE L.1-1. COMPARISON OF THE ENVIRONMENTAL STANDARDS FOR THE FEDERAL STANDARD CLASS 100000, THE AIR FORCE STANDARD CLEAN ROOM (OPERATIONAL), AND THE AIR FORCE STANDARD CLEAN ROOM (DESIGN).

	Federal Standard Class 100000	AF Standard Clean Room (Operational)	AF standard Clean Room (Design)
Particles/Cubic Foot			
Equal to or greater than			
0.5 microns	100000	100000	20000
1.0 microns	20000	20000	4000
5.0 microns	700	700	
Temperature			
Range (any specified level)	67°F to 77°F	67°F to 77°F	67°F to 77°F
Deviation (from specified level)			
Critical application	± 0.5°F	-	-
Less critical application	± 5.0°F	-	-
Rate of change		4.0°F/Hr.	2.5°F/Hr.
Humidity			
Maximum		45	40
Deviation (from specified level)			
General application		-	-
Humidity sensitive application		-	-

Small wonder that government and industry cooperated in holding a Conference on Clean Room Specifications in April 1963 to see what could be done to limit spending on the achievement of cleanliness. According to a Department of Defense spokesman, facility construction at the time was costing $1,000,000 weekly. One feature of the conference proceedings was emphasis on the shortcomings of the conventional approach. These are explained in the following paragraph.

Generally, the handling of the air and its movement in and out of the room are accomplished in accordance with the standards and practices employed in the field of air-conditioning. The only novelties are the use of elaborate and expensive materials of construction and the installation of super-interception filters in the air supply ducts. This approach, in so far as it relies on the building of walls, floors, and ceilings that shed a minimum of contamination, makes sense. But the idea that absolute filters superimposed on an ordinary ventilating system will maintain room cleanliness is fallacious. The dirt that degrades a clean room environment originates with people, processes, and hardware.

The daily variation in contamination level for a typical clean room is shown in Figure L. 1-2. The real problem is to remove this contamination as soon as it is generated. But, unfortunately, the conventional clean room does not do this very effectively. It has, indeed, a very poor self-cleaning capability. Much of the particulate matter remains airborne, settles, mixes again with the air as people disturb the atmosphere, and, in some measure, becomes entrapped in the dead air pockets that exist in the sections of the room that are not swept by the air in passing

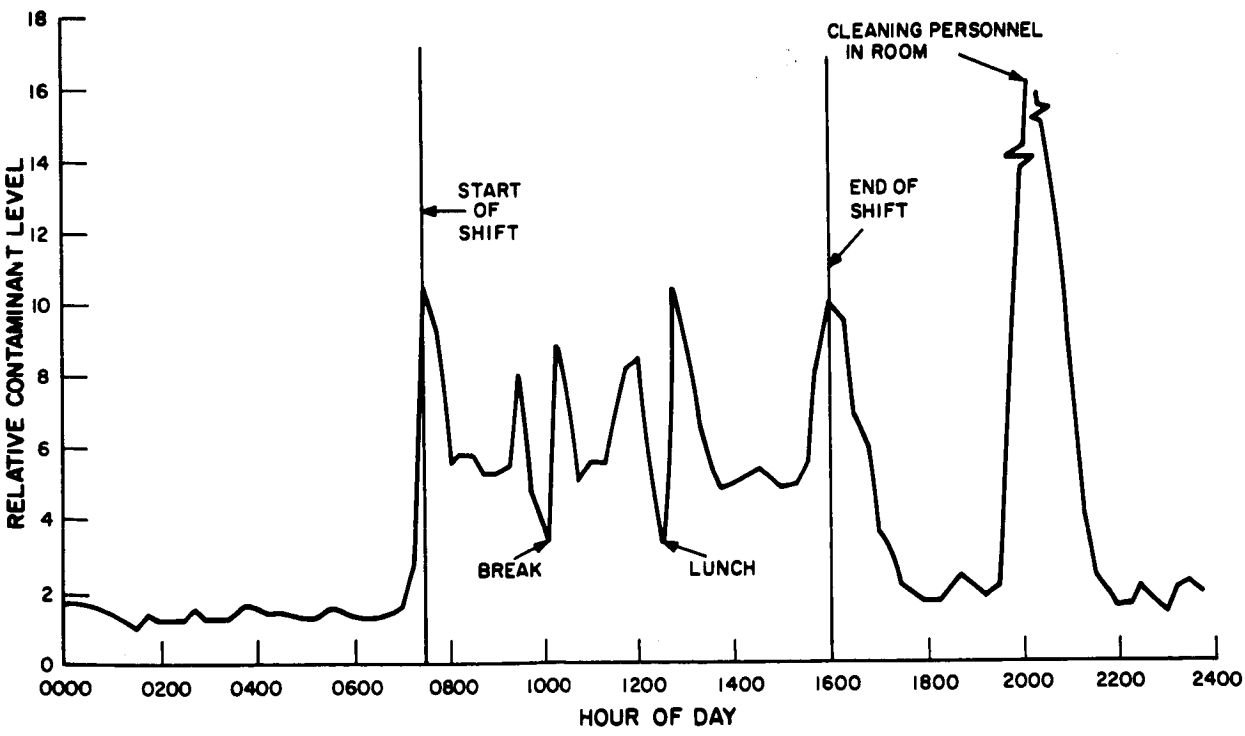


Figure L. 1-2. Typical Clean Room Contaminant Levels

from the inlet to the exhaust ports. In the Figure L. 1-1 installation, the air enters each room from a central air handling unit through two ceiling inlets and exhausts into a single duct through a wall opening located approximately one foot above the floor. In the case of Work Room "C", for example, two, four-foot square louvers located at equal intervals on the center line of the length of the room introduce the filtered air. A five and a quarter square foot grille on the center line of the outside wall serves as the outlet. Obviously, air flow under these circumstances is bound to be highly irregular. And any dust that settles on the floor has to be elevated and reentrained before it can exit through an opening located above floor level. The real cleaning that takes place in a conventional clean room is accomplished by the janitor with the aid of a vacuum system, mops, and wipers. True, some of the dirt does leave through the exhaust ducts. But, by and large, it remains within the confines of the room until the janitor performs some cleaning action. Thus, any given particle of contamination can make innumerable contacts with critical hardware as it floats, settles, and rises aloft again. Under these circumstances there is no way of assuring that a cleaning operation will not be followed by recontamination as long as the hardware remains exposed to the clean room environment. For this reason it would be difficult to implement a sterilization program with the maximum effectiveness and at a minimum cost through the use of the conventional type of clean room. There was a time when owners of such installations claimed they met Class IV standards. With few exceptions (Machine and Tool Blue Book, volume 59, Jan. 1964, article entitled "IBM's Antiseptic Environment" which states that design criteria exceed U.S. Air Force Class IV specifications) appraisals of capability have become more realistic. Generally speaking conventional clean rooms qualify as Class 100000 installations. The one shown in Figure L. 1-3 probably does this more readily than the facility shown in Figure L. 1-1. The long continuous ceiling inlet and the exhaust openings running the entire length of the room at floor level on opposite walls provide in a crude and elemental way some of the features of the down-flow laminar concept.

In summary, the conventional clean room is costly to operate for a number of reasons. Among them are the following:

1. The air-handling equipment must be run continuously.
2. The janitorial or clean-up program must be in effect daily.
3. Special clean room garments must be furnished and maintained.
4. Access procedures result in lost time.
5. Restrictions on movements and operations limit output.
6. Channels of communication become complicated.
7. Control of parts flow necessitates elaborate paper work and procedures.

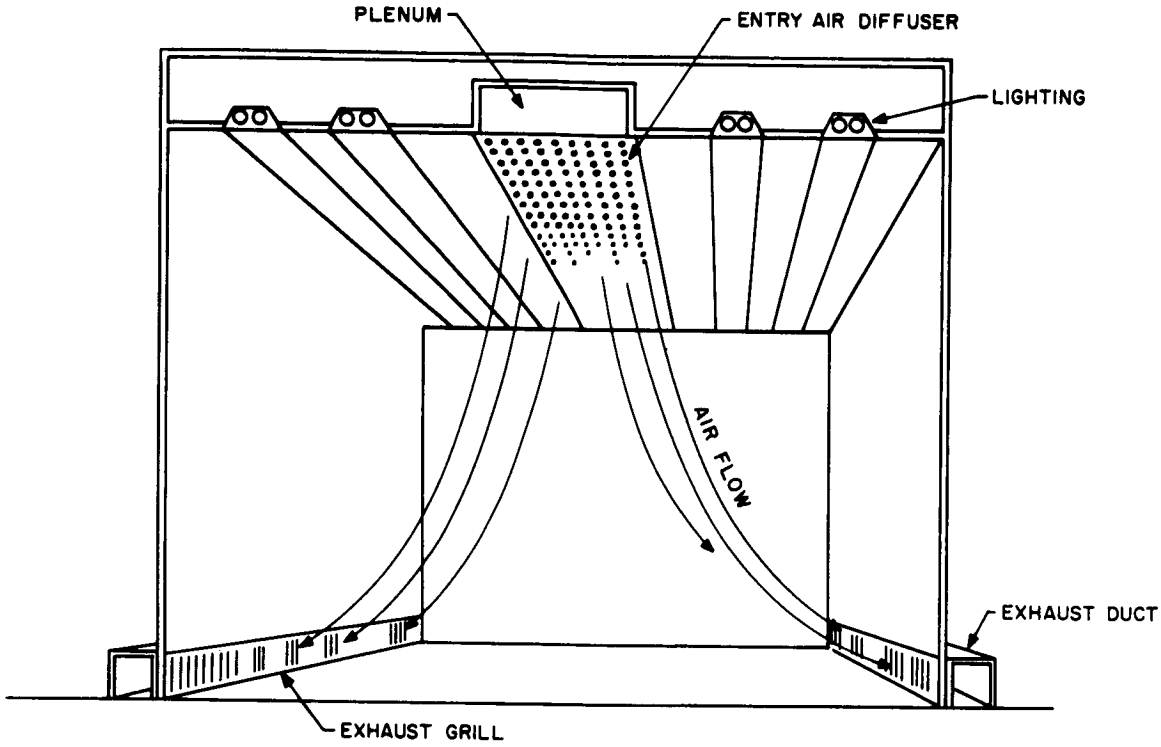


Figure L. 1-3. Conventional Clean Room

8. Many processes closely related to clean room operations must be separately located outside the room because of contamination problems.

9. Dust monitoring must be performed daily.

L. 1.5 LAMINAR FLOW

The laminar flow rooms shown in Figures L. 1-4 and L. 1-5 were developed by Willis Whitfield (Ref. 1) of the Sandia Corporation. His objectives are to maintain cleanliness in the immediate vicinity of the work piece and to prevent contamination once released from having more than a single opportunity to impinge on critical hardware. Therefore, in a laminar flow installation contamination is carried in a direct line from its source to a point of exit from the room. All of this is accomplished by locating the work area directly in the path of air flowing from a large surface made up of absolute filters. With this arrangement the air in a room changes completely 120 times more or less every hour. In a conventional room, on the other hand, the air circulates once every 3 minutes, which is the rate of change employed in ordinary air conditioning installations. For purposes of contamination control, however, 20 changes of air per hour are entirely inadequate.

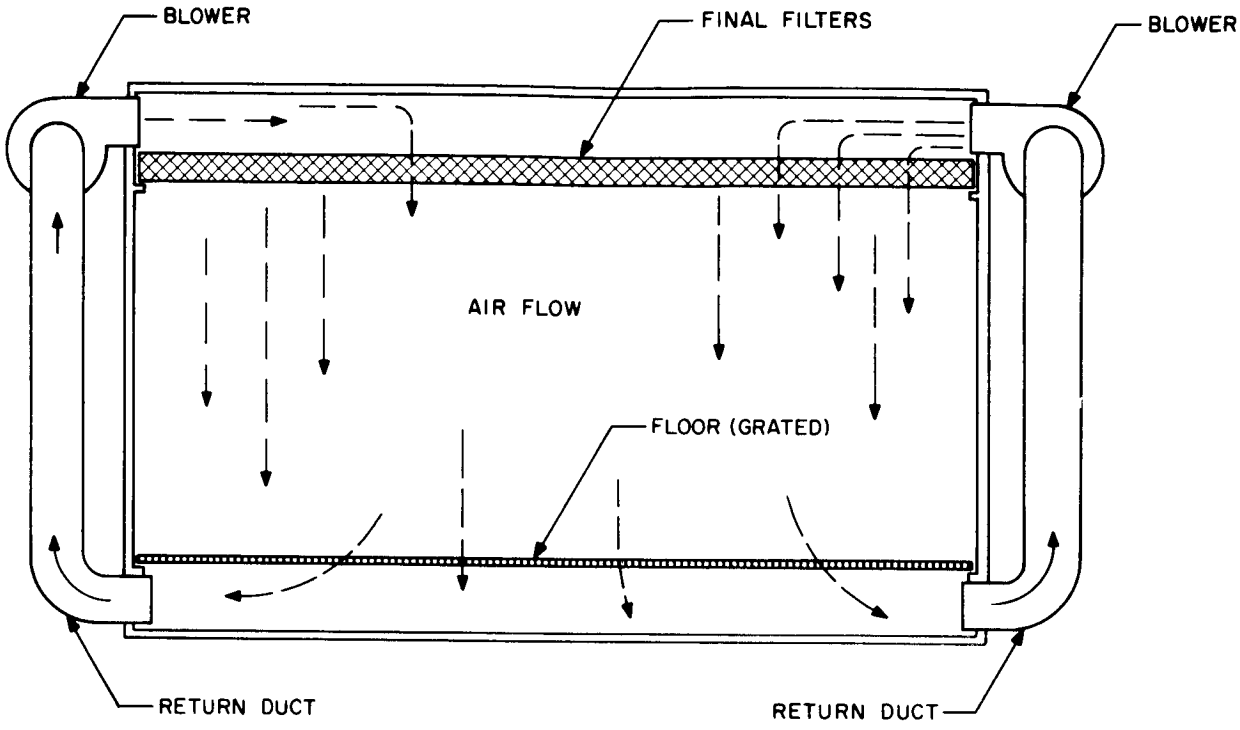


Figure L. 1-4. Ceiling to Floor Laminar Flow Room

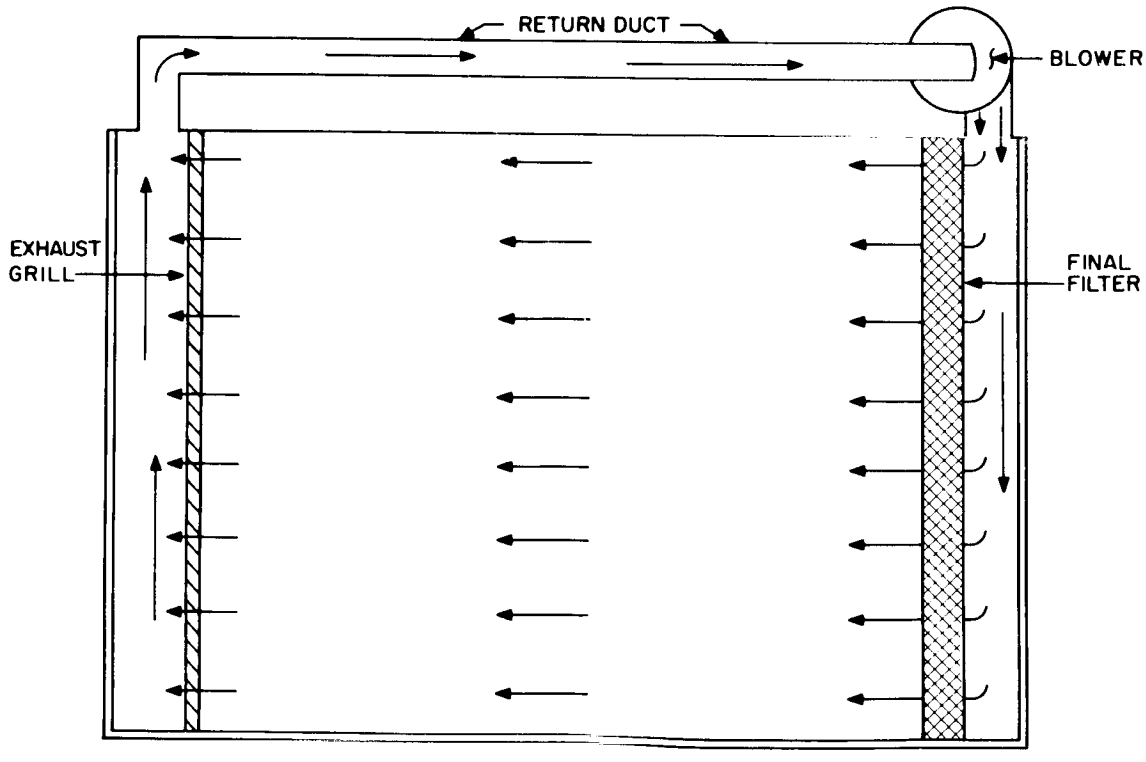


Figure L. 1-5. Cross Flow Room

Willis Whitfield explains an important difference between his and the conventional approach by saying that the cleanliness level in a non-laminar flow room is dependent on the operations performed within it, whereas the cleanliness level in a laminar flow facility is almost independent of the activities going on inside. This attempt to differentiate the two is not totally valid. In the case of the down-flow design Whitfield's statement is substantially correct. Contamination is quickly removed from the area where it is generated and exits through the floor. Parallel streamlines prevent the lateral migration of particles so that each work station in the room can be considered as totally isolated from every other work station. And any particle released from a piece of hardware never returns to the vicinity of its origin as it may in a conventional clean room. In view of these conditions the level of cleanliness is practically independent of the work being accomplished. This is not the case, however, in a cross-flow room. As the air flows from the filter wall to the opposite wall, the contamination level at a particular work station is dependent on the operations performed at other work stations upstream. This situation does, indeed, contradict Whitfield's assertion that the activity level has little or no bearing on the cleanliness level, but it in no way detracts from the benefits and economies provided by the cross-flow principle. Among these are the opportunity to carry on within the same four walls both the cleanest and the dirtiest operations without interference. Class 100 conditions prevail right at the filter wall, while at the opposite end of the room various kinds of dirt producing operations could be tolerated.

Critics of the laminar flow concept argue that laminar flow will not exist as soon as people and equipment enter the air stream. Indeed, this is true, But the breakdown of laminar flow does not degrade the cleanliness of the room. On the contrary, near laminar flow re-establishes itself rather quickly down stream from the object that initiated the turbulence. Furthermore, if the obstruction yields contamination, the particles are carried away within the disturbed streamline. This ability of the room to handle local contamination without degrading the entire work space is due to stratification of the air flow. If some of the streamlines are broken up by hardware, equipment and personnel, other streamlines that are still intact, contain the contamination as it is carried out of the room.

Whitfield's data, based on work done in a rigid, all-weather portable unit, are irrefutable proof of the remarkable self-cleaning capability of controlled air flow patterns. Originally, this unit consisted of a special entry and the main 8 x 10 foot clean room. With experience came the recognition that the entry way contributed nothing, so that in some of the experimental work the doorway remained open. On one of these occasions the wind was blowing 30 to 35 miles per hour, stirring up the desert sand. Yet no increase in dust level was recorded by the Royco counter. At another time, under somewhat the same weather conditions, 15 entries were made by several personnel during a three hour period. This was done without special clothing and without entry cleanup procedures. No effects were recorded. A

room such as this will maintain a cleanliness level below 1000 particles per cubic foot, 0.5 micron size and larger and it will reach this condition in less than 20 seconds. A general bibliography of articles and papers treating laminar flow clean rooms is contained in Reference 2.

L.1.6 COMPARISON OF CONVENTIONAL AND LAMINAR FLOW CLEAN ROOMS

The Air Force Logistics Command summed up its feelings on the relative merits of the two major types of clean rooms by stating flatly that whatever may be the deficiencies of the laminar flow concept "we are generally better off in a laminar flow room than in a conventional room". The reasons for this judgment are:

1. The laminar flow of air provides a self-clean-down capability equal to the task of removing contamination rapidly.
2. The air flow patterns carry air borne contamination away from the work area so that there is no possibility of recontamination.
3. Personnel restrictions are minimized.
4. Costly and unusual materials of construction are not required.
5. Laminar flow makes it possible to use floor space more effectively because less support equipment and area are required. For example, in a 55-ft x 30-ft conventional room, 15-ft x 30-ft might be required for air showers and locker rooms, leaving 1200 ft.^{2} for usable work space. By contrast, a 55-ft x 30-ft laminar flow room would provide 1500-ft.^{2} of work area.
6. Little or no special clothing is required.
7. No time is lost in entry procedures.
8. Dirty operations associated with clean room operations can be conducted in the same area.
9. The air handling equipment operates only 40 hours a week in contrast to the 168 hours involved in the ownership of a conventional room. If a work year consists of 50 weeks, then the laminar flow air conditioner runs 50 x 40, or 2000 hours, whereas the conventional room requires 168 x 50 = 8400 hours.
10. Little janitorial service is needed to maintain cleanliness. At a West Coast General Dynamics location a cross-flow room was activated in late October 1962. As of April 1963 there had been no janitorial service; yet the room met Air Force cleanliness requirements.
11. Dust monitoring is performed only occasionally.
12. Communication problems are eliminated.

Lt. P. Austin of the Olmsted Air Force Base, who is responsible for the publication of T.O. 00-25-203, entered the General Dynamics clean room in street clothing, went through a series of violent physical activities, and then monitored the dust level. The particles generated by this experiment could be seen as they were carried in parallel air streams to the exhaust. Recovery of the room occurred in a matter of minutes. Lt. Austin stated later that had he done the same thing in a conventional room a period of one to two weeks would have been required to restore ultra-clean conditions.

Figure L.1-6 provides a comparison among various classes of clean rooms and clean work stations.

L.1.7 AIRBORNE BACTERIA COUNT IN CLEAN ROOMS

The maintenance of highly sterile conditions during the assembly of large structures poses serious problems. The Voyager Design Study performed by General Electric during 1963, proposed the use of flexible plastic tunnels in which a technician could maneuver and perform his tasks. Separated by a sterile, plastic barrier from direct contact with the hardware, the technician would find the pace of his movements retarded. The time and motion specialists in Manufacturing feel that planned time for operations such as riveting, drilling, torquing, assembly, etc., would increase threefold, at the very least.

In view of the complexities and penalties involved in attempting to extend the principle of glove-box assembly from the bench to an entire room, a careful study should be made of the feasibility of relying solely on laminar flow and absolute filters. By maintaining the work place at all times between the filter bank and the technician, while the latter wears sterile gloves and a mask, it might be possible to prevent bacterial contamination. The basis for such an expectation is the research done at the Sandia Corporation and a report on the bacteria count in industrial clean rooms published by Air Engineering, September 1963, Vol. 5, #9.

At Sandia a half-hour colon resection was performed on a dog located within the filtered air stream of a Whitfield bench. During the entire operation, twenty-eight blood agar plates were exposed at various locations in the vicinity of the dog. Eight of them were placed between the filter and the dog, and of these, six developed a simple bacteria colony, while two remained clear. Two plates were held directly over the operation site directly under the faces of the two surgeons who were both talking through their masks. At the end of the half hour, both of these plates were negative. The plate placed between the hind legs of the dog, in close proximity to the incision, exhibited a single colony. A plate within six inches of the dog's head remained clear. Between the dog and the surgeons were four plates. The colony count on these was 0, 1, 15 and 7 respectively. Even the plates on the floor at the feet of the operators remained relatively sterile. On these, the counts were 8, 6, 13, 11, and 8.

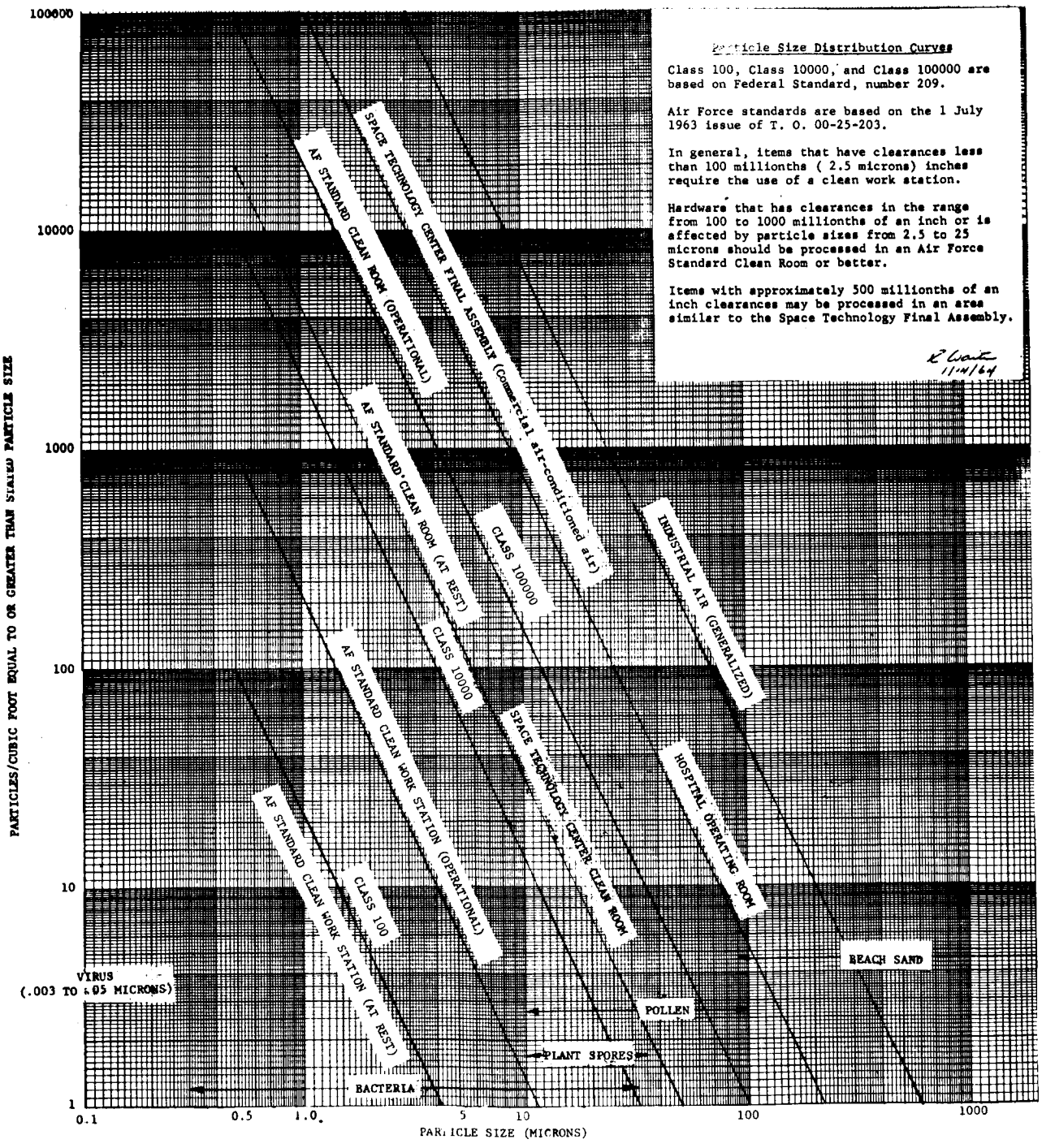


Figure L. 1-6. Comparison Among Various Classes of Clean Rooms and Clean Work Stations

Researchers of the University Health Service at the University of Minnesota have published similarly encouraging findings. They found such a vast difference between the dust concentrations and the bacterial concentrations in industrial clean rooms that they concluded that most of the airborne material is sterile. They reported that, on the grand average, the microbiological contamination level in industrial clean rooms is about 1/3 that found in hospital operating rooms. The overall results are tabulated in Table L. 1-3.

TABLE L. 1-3. COMPARISON OF HOSPITAL OPERATING ROOMS AND INDUSTRIAL WHITE ROOMS

	Number of Samples	Mean Microbial Col./Cu. Ft.	Range Microbial Col./Cu. Ft.
Hospital Operating Rooms	795	10.5	1-81
Industrial White Rooms	76	3.32	1-10

In rooms with dress and conduct conforming to Air Force Standard Clean Room regulations the investigators found mean microbial colonies per cubic foot of air as low as 0.38. And this occurred in rooms where the rate of air changes per hour varied from 6 to 10. Generally speaking, this number of changes is less than good practice prescribes. The Valley Forge room has 20 changes per hour. The newer laminar flow rooms utilize air moving at 100 feet per minute. Thus, the air in the vicinity of any localized work station will change many, many times an hour, so that there is little opportunity for the build-up of microbial contamination. Conceivably, with proper filtering, face masks, sterile gloves, and the assurance that the technician will always be on the downstream side of the hardware, a high degree of sterility can be maintained - one that would not eliminate the need for terminal sterilization, but one that could be regarded as a very satisfactory "biological load reduction".

L. 1-8. CLEAN ROOM PROBLEM AREAS

As a contractor for the Atomic Energy Commission, the Sandia Corporation is obligated to maintain extremely high standards of cleanliness for many components (e.g., fuze switches). C. F. Beld, a Sandia spokesman made the following comments at the Conference on Clean Room Specifications in April 1963. They typify the practical problems which are experienced by many clean room operators.

1. Many people are hard to convince that contamination is a serious consideration. Therefore, they resist the disciplines required to establish and maintain cleanliness. Under pressure, they give precedence to schedules.

2. Non-airborne particulate and film contamination overshadow the air-borne varieties as sources of degradation. Insofar as clean rooms limit and channel the activities of people they contribute to the control of all kinds of contaminants. Otherwise, their effectiveness is greatest in minimizing air-borne particulate matter.

3. Assemblies containing large numbers of parts usually include many crevices and capillary traps. As an example, he cited one piece of hardware with fifty-four (54) parts that requires a total of 153 operations to put together. Of these 153 assembly operations, 41 or approximately 28%, are cleaning operations. These statistics are an indication of the complexities that would attend a program designed to prevent sterility breaks.

4. Most of the contamination problems that plague Sandia stem from inadequate manufacturing plans and instructions, unanticipated sources of pollution, and insufficient awareness of the problems. The important point in this assertion is the admission that unsolved difficulties exist in a field where industry's experience covers a number of years. This means that sterilization programs must go beyond generalties and come to grips with details. In the words of C. F. Beld, contamination control is a full-time job.

L.1.9 HARDWARE REQUIREMENTS

Figure L.1-7 indicates, in general, the level of cleanliness recommended by the Air Force for a number of hardware categories. Items affected by 25 to 50 micron particles can be assembled and repaired in the Air Force Standard Clean Room. These include instruments, gyros, electronic components, precision measurement equipment, oxygen systems, engine pumps and actuators, hydraulic systems, and pneumatic equipment. Such a listing is not all inclusive. Many pieces of hardware that fall in the above-mentioned classifications can be handled in atmospheres with as many as 100 particles per cubic foot in the size range above 100 microns. Typically, such an environment might have one particle over 600 microns, nine particles over 225 microns but less than 600, and 90 particles over 100 microns but less than 225. This condition can be achieved with very little or no filtering and is to be found in many open shop areas. The point is that a lot of space hardware does not call for unusual cleanliness measures. Ordinary housekeeping would suffice.

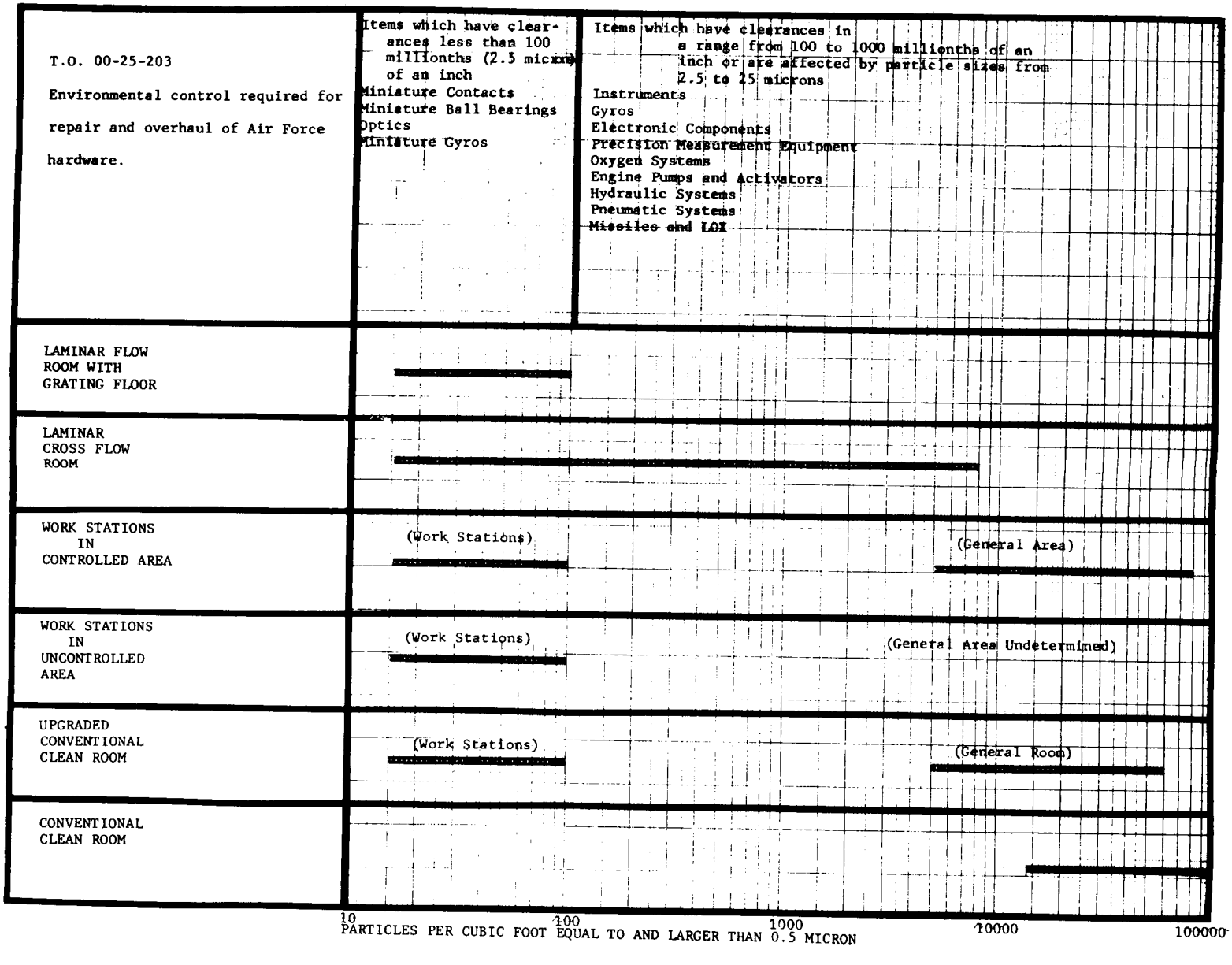


Figure L.1-7. Level of Cleanliness Recommended by Air Force for Various Hardware Categories

Not so long ago industry was obsessed with the idea that the goal of clean room cleanliness was the Class IV condition. Ironically, during this same period the government issued its strongest indictment of industrial housekeeping. Perhaps the lesson here is that preoccupation with unrealistic goals results in failure to do what is prudent. Insistence on unattainable ends discouraged many people from maintaining any interest whatsoever in cleanliness.

On the left side of Figure L. 1-7 are listed those items that can best be handled in clean work stations. These are the miniature pieces of hardware such as contacts, ball bearings, and gyros. Optics also may be benefited by such treatment.

Items on the right hand side of Figure L. 1-7 may be processed in

1. Most well-designed nonlaminar flow clean rooms.
2. Poor quality nonlaminar flow clean rooms in which laminar flow work stations are installed as a means of cleaning up the room's atmosphere.
3. Laminar flow work stations located in uncontrolled areas.
4. Controlled areas such as the Space Technology Center provided that laminar flow work stations are located in the area.
5. Laminar air cross-flow type rooms.
6. Laminar air grating flow type rooms.

Items on the left hand side of Figure L. 1-7 may be processed in:

1. Laminar air grating flow type rooms.
2. Laminar flow work stations.
3. The work locations adjacent to the absolute filters of a cross-flow room.

The non-laminar flow type of room is not suitable for handling the items on the left hand side of the chart.

L. 2 CLEAN WORK STATIONS

L. 2. 1 DESCRIPTION

The clean work station based on the Whitfield principle (Ref. 3) has made it possible to maintain Class 100 conditions with minimum expense and equipment. Until its appearance on the market within the last two years enclosures such as the Baker Steril shield were used to establish localized cleanliness. These units, that were

so popular several years ago, are mentioned here to provide a base for evaluating the performance of the Whitfield bench.

The Armour Research Foundation of the Illinois Institute of Technology reported the following "recovery time" for the Baker Steril shield:

Time, seconds	Particles/ft.^{3}
0	Too high to count
20	6853440
40	95722
90	41630
120	11724
300	3824

It took five minutes to reduce the count to 3824 particles/ft.^{3}. The long clean-up period was attributed to dead air spaces in the corners of the cabinet. These exist because of the way in which the air enters and leaves the enclosure. The inlet is a relatively small opening compared with the overall dimensions of the enclosed workspace. And the exhaust is the rectangular opening which serves primarily as the means of access to the work station located within the cabinet. Another reason for the slow recovery is the leakage of air around the filter. Whitfield demonstrated conclusively that the slightest by-passing of a filter plays absolute havoc with the dust count. And it is only since his work became known that proper care has been taken in the design and installation of gaskets. In fact, it is no exaggeration to say that a clean work station is no better than the gaskets. Naturally, the filter plays a prime role. But filters are fairly standard and are available to all manufacturers of clean benches. How the filter is installed and sealed in position determines the effectiveness of the bench in producing and maintaining a dust free atmosphere.

In contrast to the Steril shield with its 3824 particles/ft.^{3} at the end of five minutes, the Whitfield bench complies with the Class 100 standard in less than a minute. Twenty seconds is the figure most often cited in the literature.

Basically, the bench is a very simple structure of which the most important element is the super interception filter. Figure L.2-1 shows the usual profile. In fact it is so deceptively simple that many manufacturers have rushed into the market place with units. Most of them are remarkably similar in appearance. But some of them are unacceptable because of the producers' efforts to be different. To achieve sales appeal manufacturers tamper with the design fundamentals. From the purchaser's standpoint the following considerations are of the utmost importance:

1. Is the unit vibration-free? This, of course, is a main consideration when a bench is to be used for microscopy. Furthermore, vibration tends to shorten the life of the seals.

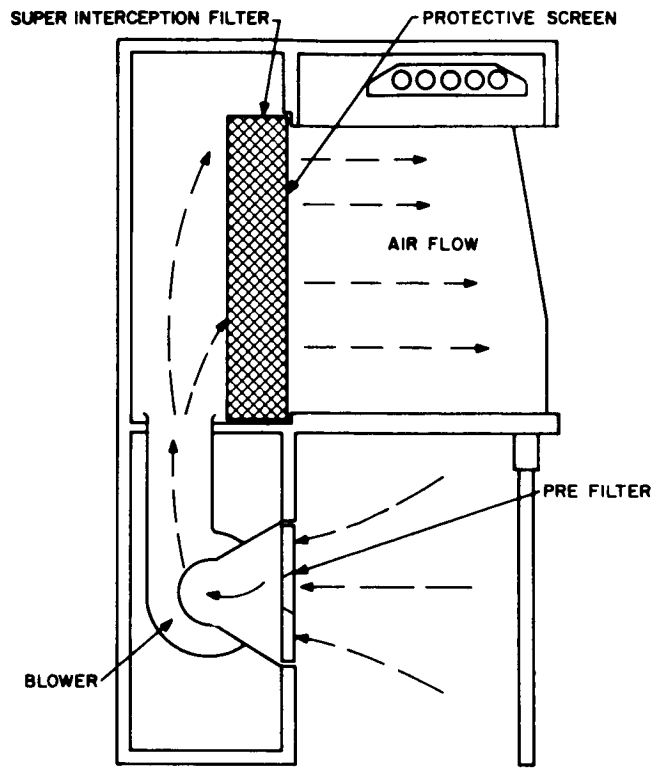


Figure L.2-1. Laminar Flow Clean Bench

2. Is the filter mounting rigid and sufficiently strong to maintain pressure on the gasket? Any leaks whatsoever nullify the effectiveness of the 99.97% filter.

3. Is the air flow at 100 ft./min. uniform within reasonable limits across the face of the filter?

4. Is the work area enclosure rectangular, perpendicular to the face of the filter, and equal to the filter in cross-sectional area? It is important that the work "tunnel" be no larger in area than the filter. Otherwise, a flow of air into the work bench through the open front could occur.

L.2.2 ADVANTAGES

The advantages of the clean work station are numerous:

1. The initial cost is relatively low and is becoming lower. Roughly speaking, an $1800 bench today was priced in excess of $4000 two years ago. With competition mounting, further reductions may be

expected. The engineers at Sandia who developed the prototype have been critical of the price structure all along. They feel that today's levels are excessive. Typical prices for stainless steel construction are:

25 inch length	-	$1000
31 inch length	-	1100
35 inch length	-	1200
49 inch length	-	1350

2. The volume of the controlled environment is commensurate with the size of the work piece. No effort and money need be expended on maintaining operating personnel within the clean atmosphere.

3. The bench can be brought to the work. Localized centers of environmental control can be established wherever the need is greatest. At Sandia, a clean air work station was established in a high bay area in the midst of a group of dirt operations. The floor was concrete rather than vinyl. Sand blasting, paint spraying, and grinding operations were carried on within twenty feet of the open front of the bench. The female operator wore powder, lipstick, a shaggy sweater, no hat, and no smock. She did wear gloves to avoid film contamination.

4. No time is lost in entry procedures.

5. The blowers operate only when the bench is in use.

6. The movements of the operator are unrestricted for all practical purposes. He is not confined by special clothing. The work is rotated so that there is no need for the worker to walk around. Communications are uninhibited.

7. Benches can be set up side by side to facilitate the flow of material from work station to work station without exposure to contamination and without the need for interim packaging.

8. Large rooms can be "cleaned-up" by installing benches so that they cover roughly 20% of the floor area.

9. Special clothing costs are reduced to a minimum and can be entirely eliminated.

10. Unusual janitorial activities and schedules are not needed.

11. The need for dust monitoring is almost non-existent. Checks are made primarily to assure that there are no leaks in the filter and the seals.

12. Maintenance costs are low since the normal operating time is only 24% of the operating time of conventional clean rooms.

The Sandia Corporation maintains an assembly line of benches in the Bulova Watch Company, New York City Plant. To maintain maximum freedom from contamination, Bulova provides two operators per work station who function like a surgeon-nurse team in an operating room. A table in the vicinity of the bench holds the work tools for which one of the operators is responsible. He serves as an assistant, handing the tools to the watchmaker who performs the assembly operations within the bench itself.

L.2.3 GLOVE BOXES

Glove boxes are complete closures that impose a barrier between the work piece and the uncontrolled environment of which the worker himself is a part. This means, ordinarily, that the hands and arms of the operator are encased in an impervious film made of rubber or plastic. This basic approach has undergone expansion to the point where people completely covered are introduced into the "glove-box". These are concepts that require careful study. Many restrictions would accompany such an operation, raising costs. The time required to perform specific tasks would be multiplied on the average by a factor of three or four according to the judgment of our Advanced Planning, Time Standards and Methods engineers.

L.3 REFERENCES

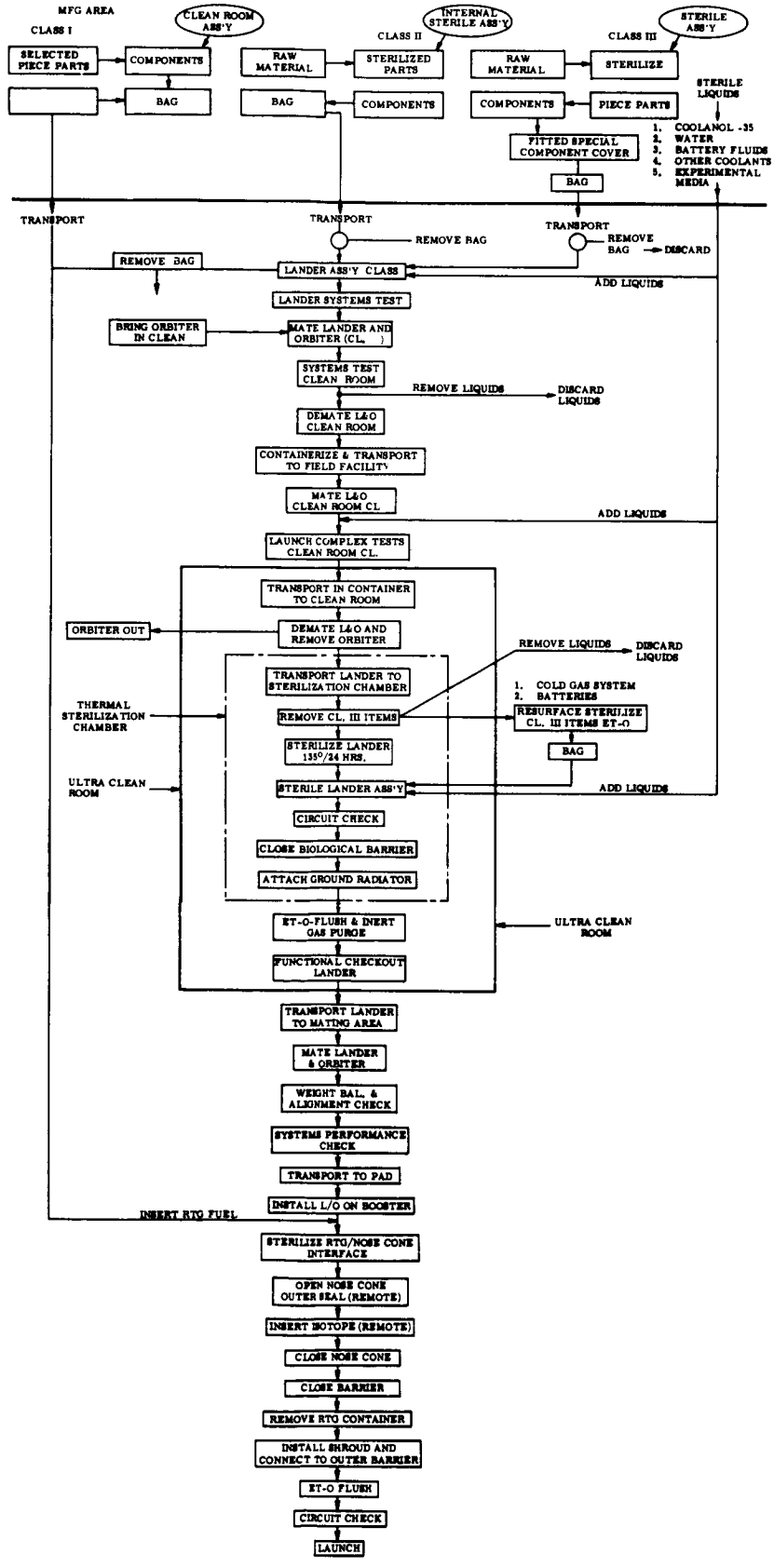
1. Whitfield, W.J.: A New Approach to Cleanroom Design, Sandia Corporation Report SC-4673 (RR), March 1962.
2. Agnew, Boyd: Laminar/Flow Clean Room Handbook. Agnew-Higgins Incorporated, 1963.
3. Mashburn, J.C., W.E. Neitzel and W.J. Whitfield: A Portable Clean Work Station, Sandia Corporation Report SC-4690 (RR), June 1962.
4. Russell, J.A., Waite, R.E., and Kindig, E.V: The Controlled Environment Facility at the Valley Forge Space Technology Center, Spacecraft Department Missile and Space Division, General Electric Company, August 1962.

APPENDIX M

FUNCTIONAL FLOW DIAGRAM OF VOYAGER STERILIZATION PROCESS

The information contained in this Appendix was prepared by the Spacecraft Department of the General Electric Missile and Space Division under contract NAS W-696 (Voyager Design Study) for the NASA office of Space Sciences.

APPENDIX M. FUNCTIONAL FLOW DIAGRAM OF VOYAGER STERILIZATION PROCESS



APPENDIX N

RADIATION STERILIZATION OF STRUCTURES AND COMPONENTS

APPENDIX N

RADIATION STERILIZATION OF STRUCTURES AND COMPONENTS

Irradiation by nuclear particles has been studied as a means of terminal sterilization of the spacecraft. This technique is not considered feasible for the reasons discussed below. The Radiation Effects Information Center at Battelle Institute has been consulted for effects on electronic components.

Exposure energies in ergs can be directly related to specific heats and to weights of particular materials. Thus, a temperature-equivalent can be derived. The integrated neutron flux (nvt) can be roughly translated into energy as follows: 10^8 neutron/cm^2 = (0.1 Mev) 25 ergs/gm. This relationship is the carbon relationship, but the results obtained should be sufficiently close to give an understanding of the problem. For example, an electronic device that will withstand 10^{15} neutrons (steady state) can be expected to withstand 2.5×10^6 rads.

$$10^{15} \text{ nvt } \frac{25}{10^8} = 25 \times 10^7 \text{ ergs/gm}$$

$$\text{Approximately 100 ergs} = \text{one rad.}$$

$$\text{Then } 2.5 \times 10^8 \text{ ergs} = 2.5 \times 10^6 \text{ rads.}$$

A reasonable figure for sterilization purposes would be a requirement of 10^7 rads to insure a lethal dose to micro-organisms. In the case in question then, sterilization by means of radiation would be inadequate. Each case must be individually treated. Trade-offs, however, involving combinations of techniques are possible. An additional consideration is that the components will be subjected to additional radiation during the voyage. This must be estimated and added to the total sterilization dosage when calculating component reliability for the entire trip.

Possible trade-offs include such considerations as the radiation resistance of germanium devices as compared to silicon devices, the effect of shielding on voyage dose, etc. In view of the difficulties of dry heat or chemical sterilization, each potential problem component should be examined to determine radiation tolerance level. It would seem possible that relatively large sections may be efficiently sterilized by radiation.

The effects of combined radiation and temperature are generally unknown. Active investigation is proceeding within this area. As this information becomes available, it should be examined for applicability.

It is most probable that the use of irradiation should be limited to sterilization of items that cannot take thermal or ethylene oxide treatment. Table N-1 below indicates the state-of-the-art and potential development.

TABLE N-1. STATUS OF MATERIALS IN RELATION TO NUCLEAR ENVIRONMENT

Section A. Exposure, ergs gm^{-1}(C)

	Material(1)	Material(2)
Elastomers	5×10^9	5×10^{10}
Plastics	7×10^9	2×10^{12}
Fuels	10^9	5×10^{12}
Lubricants and Hydraulic Fluids	5×10^{10}	5×10^{11}

Section B. Integrated Flux, N/cm^2

	Material(1)	Material(2)
Structural Metals	10^{20}	10^{20}
Ceramics	10^{19}	10^{20}
Electronic Components	10^{15}	10^{18}
Semi-conductors	10^{13}	10^{15}

(1) Current materials are satisfactory to this dosage.
(2) Materials under development may give satisfactory service to this dosage.

APPENDIX O

ULTRASONIC STERILIZATION

APPENDIX O

ULTRASONIC STERILIZATION

The use of ultrasonics has been recommended for cleaning surfaces and small fixtures, nuts, bolts, and rivets, prior to assembly and terminal sterilization.

The use of ultrasonics for degreasing, coagulation and cutting is standard industrial practice. However, for spacecraft sterilization, ultrasonic cleaning methods have been studied for other reasons. The effect of cavitation on surface dirt literally blasts the contaminants from surfaces, blind holes and tiny devices. The time required is usually only in the second to minute range and in as much as actual erosion of the surface occurs, the process can, with proper choice of frequencies and immersion fluids, insure a new and relatively uncontaminated surface. The erosion is so minute (in the time range mentioned) that no harm is done to the material surface from a structural standpoint. Ultrasonic cleaning is most effective on hard, relatively clean surfaces of the type most likely to be used in spacecraft construction.

In the planning of spacecraft sterilization systems, the effect of the ultrasonic cleaning process on biological contamination is of prime importance. Most simple organisms such as unicellular infusoria are killed instantly in a cavitating liquid. The cell walls are ruptured immediately upon exposure (Figure O-1). The effect however is not universal with regard to organism lethality. For example, tuberculosis bacilli when exposed to ultrasonics suffer extensive population losses, yet some survival is evident upon reculture experiments. None the less, the treatment does have the bonus effect of reducing the biological load by lethal action as well as by physical removal. This is of considerable value in avoiding recontamination. The use of bactericidal solutions as the cavitating fluid affords additional protection. The recirculation and microfiltration of the fluid affords an enormous amount of organism and debris removal. For this reason, a strong recommendation is made that all parts and structures, that can be physically exposed to an ultrasonic cleaning, be so treated during manufacture. Where a designer would have a choice, the ultrasonically treatable part should be chosen. Obviously, the treated item must be protected after treatment to insure retention of cleanliness and sterilization. Placement of the cleaned item within a protective bag, sterilization by thermal means, and storage or transport only after completion of this procedure should be mandatory. Plastic containers, for example, the DuPont polyimide series plastics, resembling polyethylene in appearance and handling qualities, withstand heat far beyond the requirements of the problems. Figure O-2 shows a typical flow diagram for ultrasonic treatment.

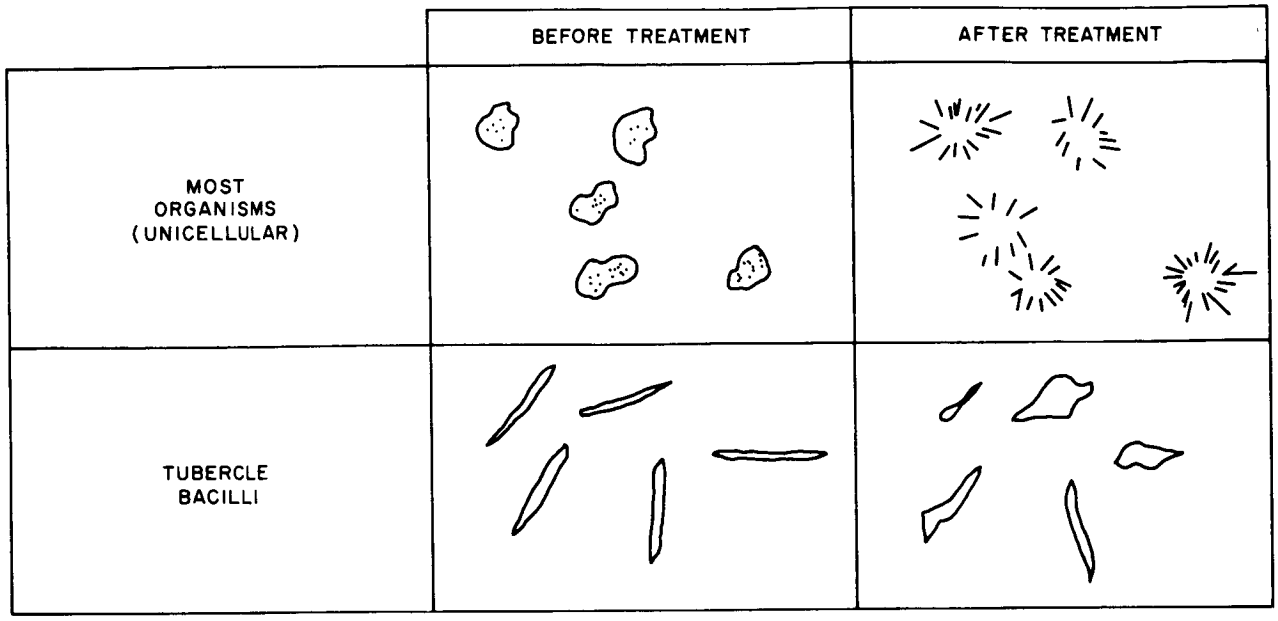


Figure O-1. Effect of Ultra Sound on Micro-organism (Adapted from Glickstein)

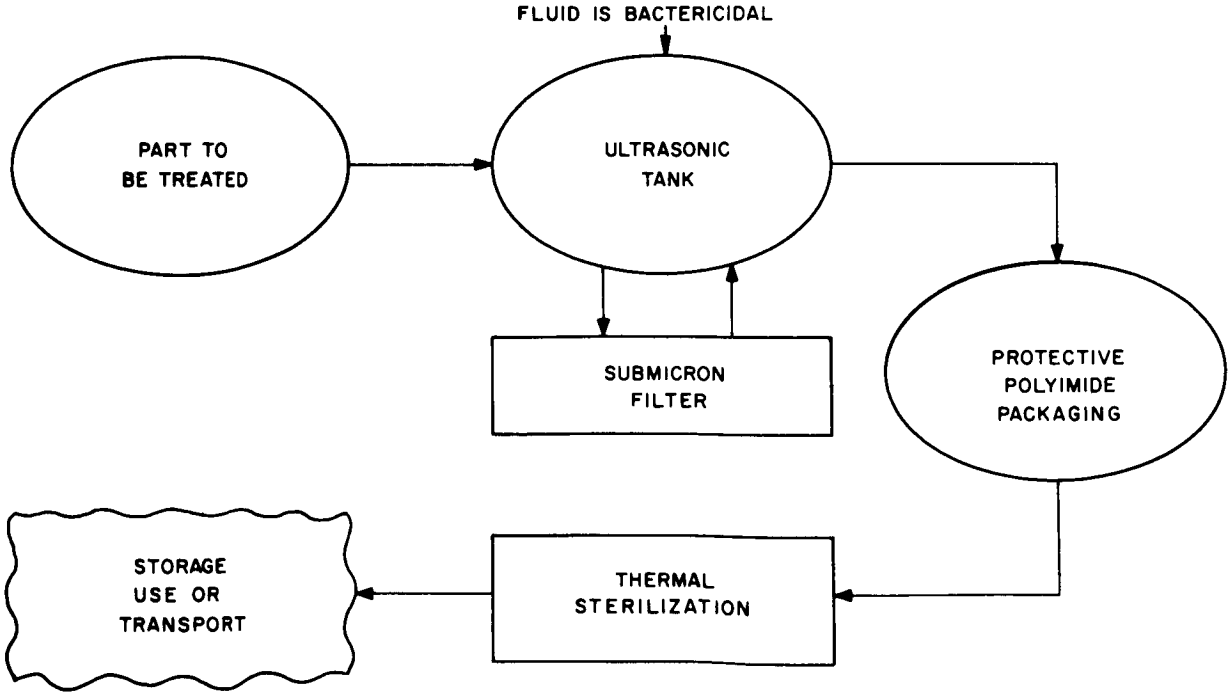


Figure O-2. Flow Diagram of Ultrasonic Treatment

APPENDIX P

TYPICAL TEST PROGRAM, EFFECT OF SURFACE FINISH ON EASE OF STERILIZATION

APPENDIX P

TYPICAL TEST PROGRAM, EFFECT OF SURFACE FINISH ON EASE OF STERILIZATION

Typical Test Program

To indicate an approach to testing which might be useful, a typical approach is outlined. Such a program has not been carried out in this study. This outline is intended to point out the problems and requirements which would accompany testing.

The objective of testing is to determine what effect the finished surface of a material will have upon the ease and efficiency of thermal sterilization.

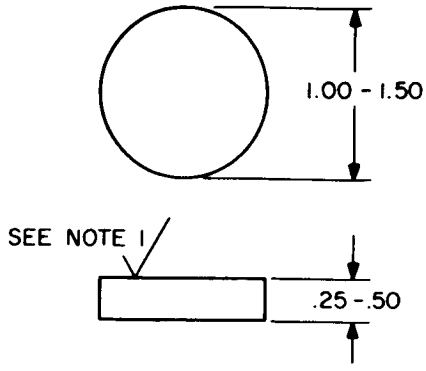
1. Materials needed: Autoclave, culture media, incubator, petri dishes, test organism, atomizer, sample plates (at least 4 sets of 3 finishes each set), tubing, pumps, filters, microscope and slides, camera, timer, Whitfield Flow Clean bench and sundry other pieces of equipment found in a biological laboratory.

2. Utilizing the matrix plan, clean all samples (Figure P-1), flame surfaces and sterilize with heat in autoclave. Sterilization to be accomplished approximately 250°C for 12 hours. Samples would be placed in petri dishes or sealed cans and sterilized inside their container. A set of handling tongs should be sterilized at the same time. Samples must never be handled except with tongs. It is recommended that this experiment should be performed on a Whitfield flow bench.

3. Contamination of samples is accomplished by spraying the surface of the sample with an organism solution, using an atomizer. The control sample, of course, is not contaminated. All samples would be incubated for approximately 6 hours at 33-37°C (or temperature best suited to specific organism growth). At least four (4) different (spore forming) organisms should be used.

4. Samples F (see Figure P-2 for Typical Test Matrix) and samples A would be cultured by placing in the sterile culture media. The sample would be read at 12, 24, 36, 48 and 72 hours. Readings would continue until growth appears equal on all contaminated plates.

5. Sample set E would be cleaned, sterilized, contaminated and given the full thermal treatment series (or at least until sterility is demonstrated).



MAKE FROM 2024-7075 OR
OTHER TYP. SPACECRAFT ALLOYS

NOTES:

1. SURFACE FINISHES TO BE 125, 32 AND 4 μ IN. OVER.
2. ALL SPECIMENS SHOULD BE INSPECTED WITH SURFINDICATOR OR PROFILOMETER AND INSPECTION RECORDS RETAINED.

Figure P-1. Typical Specimen for Surface Finish Experiment

SERIES / SPECIMEN FINISH		A 1 2 3	B 1 2 3	C	D	E	F	G	H
CLEAN		✓	✓	✓	✓	✓	✓	✓	✓
HEAT STERILIZE		✓	✓	✓	✓	✓	✓	✓	✓
CONTAMINATE			✓	✓	✓	✓	✓	✓	✓
STERILIZE @ 135 °C	1 HR			✓					
	2 HR				✓				
	4 HR					✓			
	8 HR						✓		
	16 HR							✓	
	24 HR								✓
CULTURE		✓	✓	✓	✓	✓	✓	✓	✓

NOTES:

1. SERIES A AND B ARE CONTROLS.
2. MATRIX SHOULD BE REPEATED 4 TIMES FOR EACH OF 4 ORGANISMS.
3. MATRIX SHOULD BE REPEATED IF NECESSARY AT 125°C AND 150°C.

Figure P-2. Surface Finish Experimental Matrix

6. One method of culturing which should be done is to use a stiff wire brush to brush the sample surfaces. The brush would be worked in sterile water until the spores and organisms are suspended in the water, aliquot sample of the water planted on the nutrient broth. (About 0.1 ml specimens taken from carefully shaken suspensions would be streaked.)

7. Among the recommended organisms include: Escherichia coli, Bacillus subtilis, Aspergillus terreus, Penicillium citrinum, Aspergillus niger. These are tentatively suggested inasmuch as a considerable knowledge has been gained about them. It is quite possible that others may be of more value.

8. Culture media under consideration include: Thioglycolate broth, wort agar, bone nutrient agar. No specific media have been selected at this time.

9. Adequate means must be taken to insure that the temperature rise on the sample plates is fast enough to break through the thermal adaptation barrier.

The anticipated result of this testing would be a matrix or chart of sterilization efficiency for various surface finishes with respect to temperature, usable by a designer trying to select a finish for structural components or elements.

It is obvious that some parts can be separately and absolutely sterilized. It may be possible to use sterile assembly techniques to fit these parts together. Where this is the case, a relatively rough surface finish could be tolerated. Where major dependence must be placed on terminal sterilization, a very fine finish (with high attendent cost), could be specified. The effect of this choice upon manufacturing and assembly costs and processes may be significant.

APPENDIX Q

STRUCTURAL MATERIALS

This Appendix describes and summarizes materials work by numerous investigators. The data which are presented were taken from a variety of sources which are listed in the References. No materials testing was performed under this contract.

APPENDIX Q

STRUCTURAL MATERIALS

Q.1 INTRODUCTION

The purpose of this investigation was to determine the ability of typical spacecraft structural materials to withstand a terminal heat sterilization environment.

Except for metallic materials, the potential spacecraft materials are grouped in broad functional areas. They include adhesives, composite structural materials, re-entry materials and a miscellaneous category including coatings, insulation, and special materials. Two broad areas were considered for each material category, i.e., properties while at the sterilization temperature, and effects of sustained exposure at this temperature upon subsequent properties at room temperature. The advantages and disadvantages of specific materials are discussed and, where possible, specific recommendations are made.

Q.2 METALLIC STRUCTURAL MATERIALS

Q.2.1 GENERAL

Of all the constructional materials used for spacecraft, metallic materials have been the most thoroughly evaluated, and considerable data are available which are directly applicable to the thermal sterilization problem. Alloys of aluminum, magnesium, titanium, and steels, both low alloy and stainless types, are all used to varying degrees. Variation of physical and mechanical properties with temperature are reasonably well documented for the more common alloys of these metals. The mechanism of creep and time-dependent plastic strain at elevated temperatures of metallic materials is understood and can be compensated for. In addition, metallurgical phenomena, such as diffusion, phase changes and precipitation, which occur as a result of the application of heat have been well studied.

Selection of metallic materials should pose no serious problem as a result of the thermal sterilization environment. A temperature of 300°F is not considered exceptionally high except for the lower melting point metals. Based upon strength, the maximum operating temperatures of metallic materials are roughly proportional to their melting points.

Element	Melting Point (^{o}F)
Magnesium	1202
Aluminum	1220
Iron	2802
Titanium	3300

Based upon this rough rule, it is obvious that magnesium and aluminum alloys will be the most marginal. In fact, certain aluminum and magnesium alloys do have insufficient strength or suffer significant degradation of properties at 300°F. This will be documented and discussed below in the separate sections for each class of alloy. Thermal sterilization will pose no problem for titanium alloys, stainless steels, and certain of the alloy steels since they retain useful properties at temperatures as high as 1000°F.

A potentially serious problem that must be considered is distortion of structures as a result of differences in thermal expansion coefficients for the various alloys. This is graphically illustrated in Figure Q.2-1. As can be seen, the total expansion for aluminum and magnesium between room temperature and 300°F is about twice that for steel and titanium. For magnesium alloys, this difference in strain is equivalent to approximately 9,000 psi. A stress of this magnitude is not excessively high when compared to the elevated tensile properties of magnesium alloys but becomes significant when creep is considered.

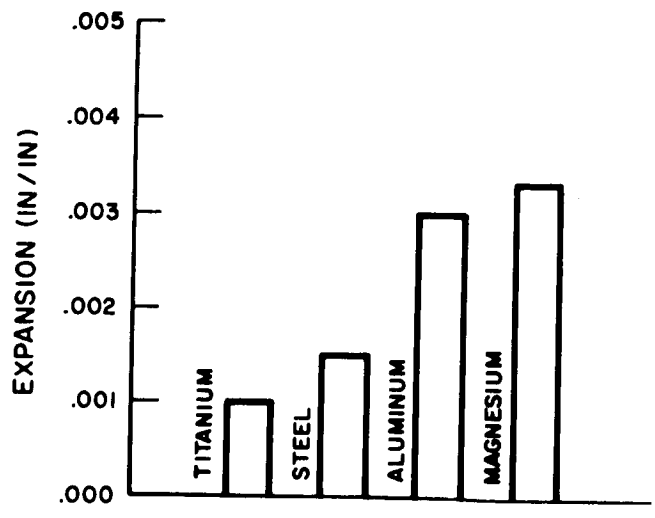


Figure Q.2-1. Total Expansion of Metallic Alloys Between Room Temperature and 300°F

Additional characteristics of metallic alloys must be evaluated to determine their suitability for use in a spacecraft structure to withstand the thermal sterilization cycle. First, strength at the 300°F temperature must be known so that design loads and thermally induced stresses will not cause failure. In particular, the strength properties should be those obtained after prolonged exposures of up to 500-600 hours at temperature. Creep is the second area to be considered since significant distortion may occur during lengthy exposure to 300°F in certain of the metallic alloys. Lastly, the occurrence of irreversible changes during the sterilization cycle must be evaluated.

The decrease in strength with increasing temperature is a reversible process for certain alloys; i.e., when the temperature is decreased, strength increases and returns to its original room temperature value. However, irreversible temperature changes, such as recrystallization, precipitation of second phases, and overaging, will result in changes of subsequent room temperature properties for many of the common structural alloys. Since the spacecraft must undergo the stresses of launch and perform its mission in space, these changes in properties must be well documented.

Q.2.2 MAGNESIUM ALLOYS

A. General Characteristics

Magnesium alloys are extensively used in spacecraft because of their low density of 0.065 lb/in.^{3}, lowest of all constructional alloys. Their strength is quite low, but based upon stiffness-to-density ratio they rank higher than all alloys except beryllium. Chemical compositions for representative magnesium alloys are presented in Table Q.2-1.

AZ31B receives extensive use as sheet, plate, and extrusions. Since this alloy is not heat treatable, strain hardening by cold working is used to increase the strength. In this strain hardened condition, the alloy is identified as AZ31B-H24. Normally, this alloy is not considered suitable for extended service at temperatures above 250°F.

HM21A is a thorium containing alloy also available as sheet and plate. Although somewhat lower in strength at room temperature than AZ31B-H24, it has two distinct advantages. First, it has a higher thermal conductivity and second, improved elevated temperature properties. In particular, thorium as an alloying element increases the creep resistance of magnesium alloys.

ZK60A, is among the highest strength wrought alloys, and is particularly suitable for extrusions and forgings. It is used in the heat treated T5 condition obtained by aging at 300°F for 24 hours directly following the high-temperature forging or extrusion operation.

TABLE Q.2-1. CHEMICAL COMPOSITION AND HEAT TREATMENT FOR MAGNESIUM ALLOYS

Chemical Composition

Percentage

Element	AZ31B	HM21A	ZK60A	ZH62A	LA141
Aluminum	2.5 -3.5	—	—	—	0.75-1.25
Zinc	0.6-1.4	—	4.8-6.2	5.2-6.2	—
Manganese	0.20 Min	0.45-1.1	—	—	0.05 Max
Thorium	—	1.5 -2.5	—	1.4-2.2	
Zirconium	—	—	0.45 Min	0.5-1.1	
Silicon	0.10 Max	—	—	—	
Copper	0.05 Max	—	—	0.10 Max	
Nickel	0.005 Max	—	—	0.01 Max	
Iron	0.005 Max	—	—	—	0.005 Max
Calcium	0.04 Max	—	—	—	
Lithium	—	—	—	—	13.00-15.00
Other Impurities	0.30 Max	0.30 Max	0.30 Max	0.30 Max	

Heat Treatment

Alloy	Solution Anneal	Aging	Condition
AZ31B	Not heat treatable, used in strain hardened cond.		H24
HM21A	None	450°F for 16 hr.	T5
ZK60A	None	300°F for 24 hr.	T5
ZH62A	None	480°F for 12 hr.	T5

For magnesium castings, ZH62A appears to offer the most potential. This alloy, heat treated to the T5 condition by aging at 480°F for 12 hours, has one of the highest yield strengths of all the casting alloys. The high thorium content also imparts improved and creep resistance properties at elevated temperatures.

A relatively new alloy, LA141, has a high potential for application to spacecraft in the form of sheet or extrusions. This alloy contains 14% lithium which makes it about 25% lighter than the other magnesium alloys. It cannot be age hardened to a stable high strength level and so is normally overaged at 350°F. Its low strength at room temperature falls off rapidly with increasing temperature.

B. Physical Properties

Selected physical properties of the magnesium alloys discussed above are listed in Table Q.2-2. Also shown is the variation of these properties with temperatures where such data are available. These properties are of interest in sterilization

TABLE Q.2-2. PHYSICAL PROPERTIES OF SELECTED MAGNESIUM ALLOYS

Temp (^{o}F)	AZ31B	HM21A	AK60A	ZH62A	LA141
		Specific Heat (Btu/lb/^{o}F)			
RT	0.235	.245	—	0.23	0.346
100	0.237	—	—		0.346
200	0.248	—	—		0.347
300	0.260	—	.255		0.348
400	0.270	.265	.265		0.350
		Thermal Conductivity (Btu/hr./ft^2/ft/^{o}F)			
RT	45	80	70	60	25
100	47	—	—	—	25
200	51	—	—	—	24
300	55	—	—	—	24
400	58	—	—	—	24
		Avg. Coefficient of Thermal Expansion (in/in/^{o}F x 10^6)			
RT-200	13.0		14.8	—	21.8
RT-300	14.3	—	14.4	—	—
RT-400	15.1	—	14.9	15.1	22.2

since they contribute to the problem of rapid achievement of thermal equilibrium within the structure so that the total time that portions of the spacecraft will have to be at a high temperature will be decreased. HM21A is the best of the alloys from this standpoint since it has 25% greater thermal conductivity than the typical magnesium alloys. LA141 is the least desirable because of its low thermal conductivity, specific heat, and large coefficient of thermal expansion. Its thermal conductivity is 50% lower, its specific heat is 40% higher, and its external expansion is 45% greater than the more common alloys.

C. Mechanical Properties of Magnesium Alloys

The effects of elevated temperature upon tensile ultimate and tensile yield strengths of the magnesium alloys are shown graphically in Figures Q.2-2 through Q.2-6. These are typical data to indicate trends and are not suitable for design. Data at temperatures greater than 300°F are included to show the relationship between the sterilization temperature and temperatures where properties decrease significantly. This is an indication of the margin of safety in case of accidental overshooting of temperature. For AZ31B-H24 and ZK60A-T5 the strengths are those obtained following 1000-hours exposure at the test temperature. Results for the LA141, HM21A-T8 and ZH62A-T5 are short time properties based upon 0.5 hour time at temperature prior to testing.

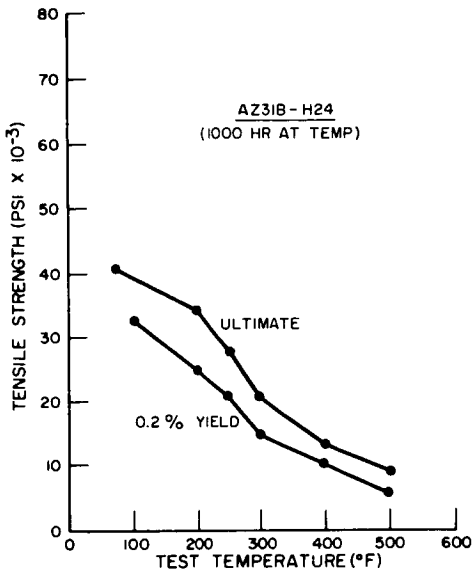


Figure Q.2-2. Elevated Temperature Tensile Strength of AZ31B-H2T Following 1000 Hours Exposure at Test Temperature

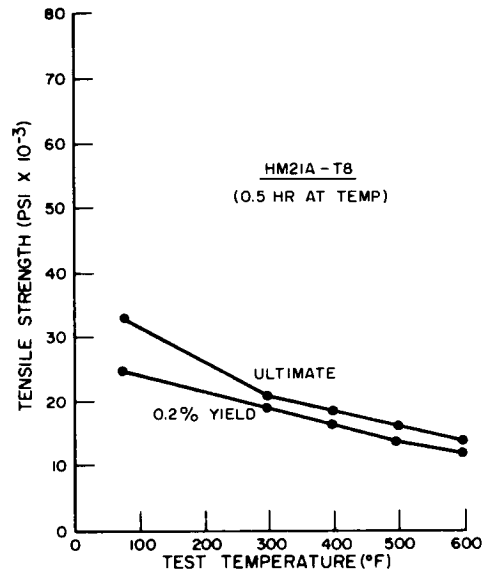


Figure Q.2-3. Elevated Temperature Tensile Strength of HM21A-T8 Following 0.5 Hours Exposure at Test Temperature

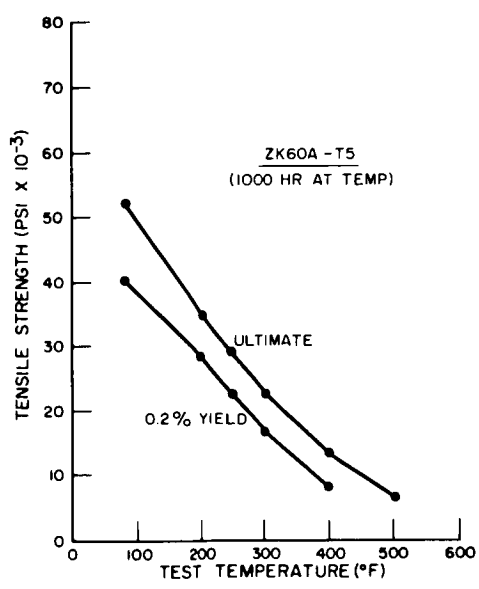


Figure Q.2-4. Elevated Temperature Tensile Strength of ZK60A-T5 Following 1000 Hours Exposure at Test Temperature

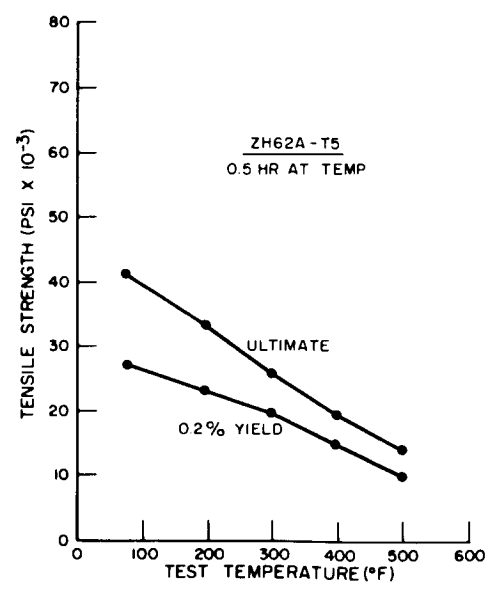


Figure Q.2-5. Elevated Temperature Tensile Strength of ZH62A-T5 Following 0.5 Hours Exposure at Test Temperature

With the exception of LA141, all of the alloys have adequate strength properties at 300°F. Tensile ultimate at this temperature varies between 20,000 and 25,000 psi while tensile yield ranges from 15,000 to 20,000 psi. Although the data for HM21A-T8 and ZH62A-T5 represent a holding period of only 0.5 hours, the thorium content of these alloys should permit them to retain their strength for much longer periods of time. LA141 has extremely poor tensile properties at 250°F, the maximum temperature for which data are available. This alloy appears suitable only for non-load bearing structures. The variation of tensile modulus of elasticity with temperature, shown in Figure Q.2-7 is insignificant for the thorium containing alloys, HM21A and ZH62A, while AZ31B drops about 20% at 300°F. This should not pose a serious problem.

Creep data are presented in Figures Q.2-8 through Q.2-11 in the form of curves showing the stress to produce given percentages of total strain, elastic plus plastic, versus test temperatures for 500 hours or 1000 hours. No data are available for LA141, but as noted previously, this alloy has extremely poor elevated temperature tensile strength and would be expected to have practically no resistance to creep. These creep curves dramatically illustrate the advantages of the thorium bearing alloys. At 300°F AZ31A and ZK60A have negligible creep strength while ZH62A can sustain a stress of 13,000 psi for 1000 hours. The only data available for HM21A was for 100 hours, but there is no doubt that this is one of the most creep resistant alloys.

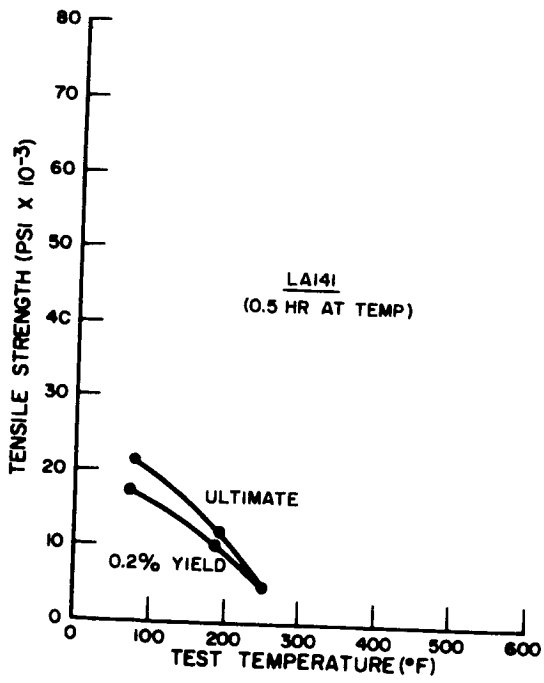


Figure Q.2-6. Elevated Temperature Tensile Strength of LA141 Following 0.5 Hours Exposure Test Temperature

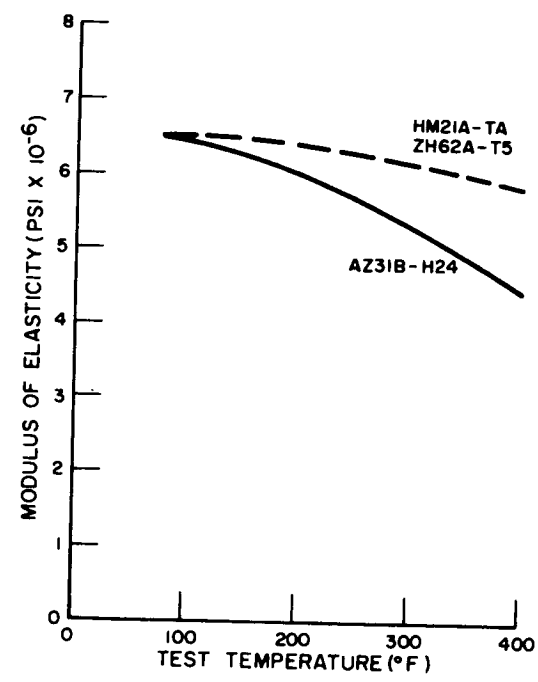


Figure Q.2-7. Effect of Temperature Upon Tensile Modulus of Elasticity

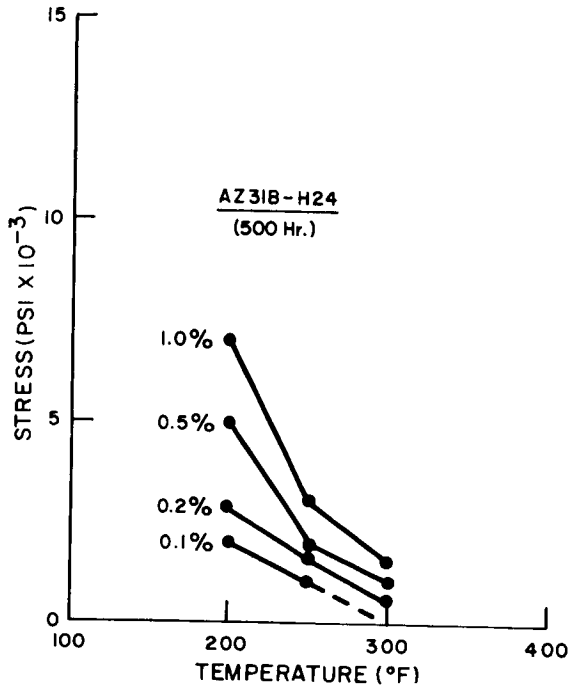


Figure Q.2-8. Stress to Produce Given Total Deformation for AZ31B-H24 in 500 Hours

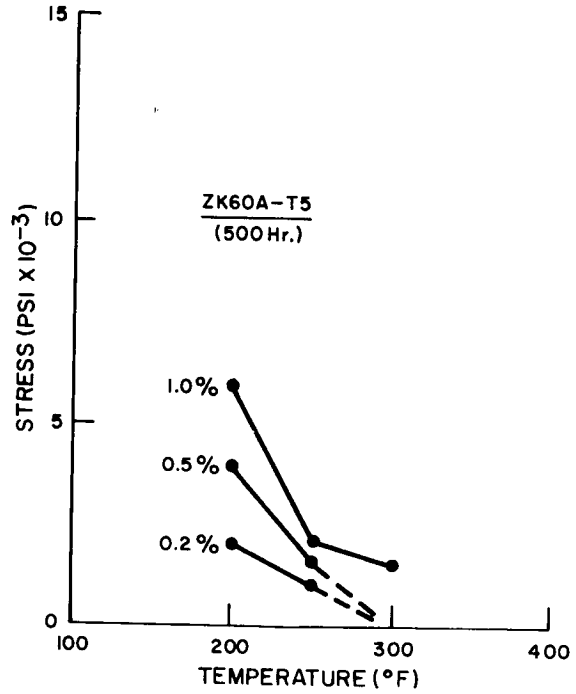


Figure Q.2-9. Stress to Produce Given Total Deformation for ZK60A-T5 in 500 Hours

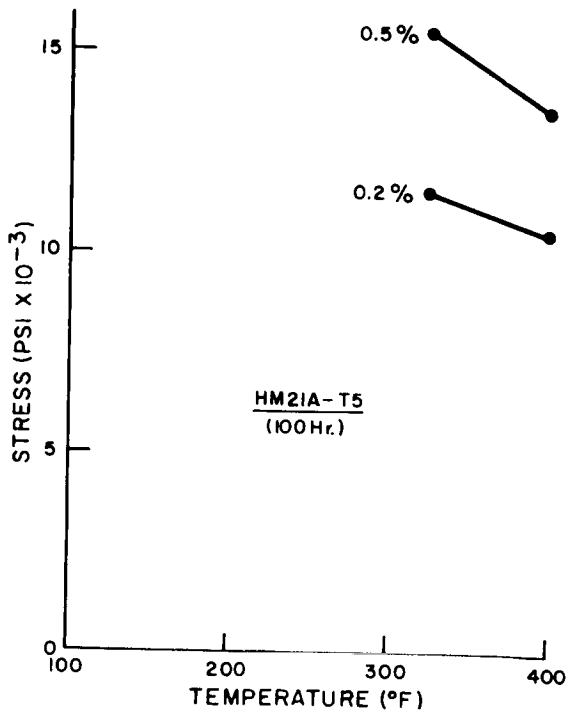


Figure Q.2-10. Stress to Produce Given Total Deformation for HM21A in 1000 Hours

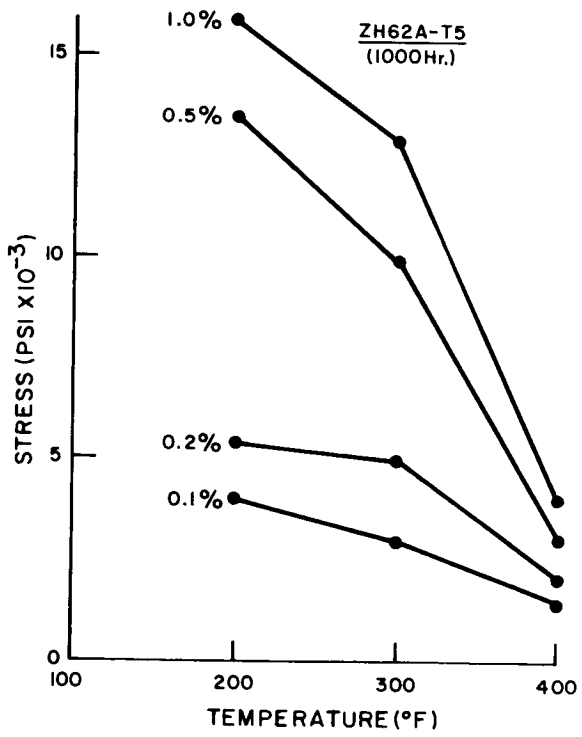


Figure Q.2-11. Stress to Produce Given Total Deformation for ZH62A in 1000 Hours

Figures Q.2-12 and Q.2-13 show the effects of prolonged exposure at elevated temperatures upon subsequent room temperature tensile properties, both for AZ31B-H24 and ZK60A-T5. Both alloys show loss in strength as a result of 1000-hours exposure at temperatures above 200°F. The decrease in yield strength after exposure at 300°F is of a sufficient magnitude that the design-allowable properties would have to be lowered for these alloys. Since ZK60A-T5 is aged at 300°F, it is not surprising that a loss in strength occurs as a result of overaging. The recrystallization temperature for AZ31B-H24 is between 350 and 400°F so a loss in strength in this alloy is also understandable.

The data on HM21A-T8 was incomplete and consisted of short time exposures of approximately one hour between 700 and 800°F. With no noticeable affect on subsequent room temperature tensile strength. No data was available for ZH62A-T5, but it can be assumed that it likewise would resist overaging.

D. Suitability of Magnesium Alloys

Magnesium alloys, attractive for space vehicle structures because of their low density, must be used with care when thermal sterilization is a requirement. This can be seen in Figure Q.2-14 and Q.2-15 which compare behavior at 300°F. In particular, AZ31B-H24 sheet and ZK60A-T5 forgings may not be useable. Both of these alloys are prone to creep at 300°F under extremely low stresses and it is doubtful that they can support anything other than their own weight. In addition,

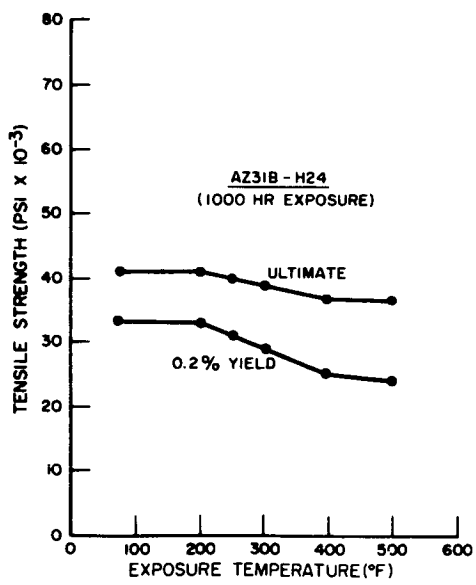


Figure Q.2-12. Effects of 1000 Hours Exposure at Elevated Temperature upon Subsequent Room Temperature Tensile Property of AZ31B-H24

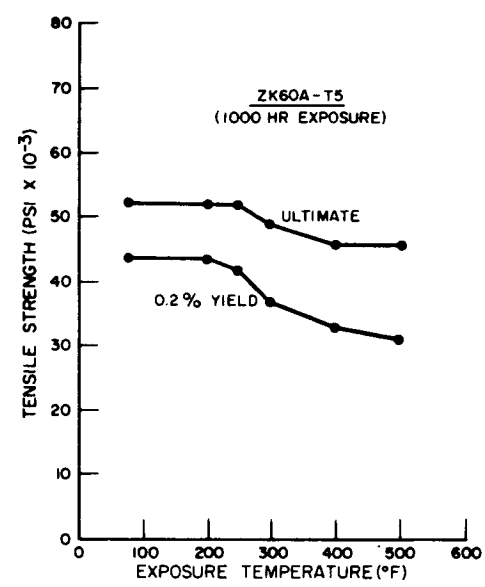


Figure Q.2-13. Effects of 1000 Hours Exposure at Elevated Temperature Upon Subsequent Room Temperature Tensile Property of ZK60A-T5

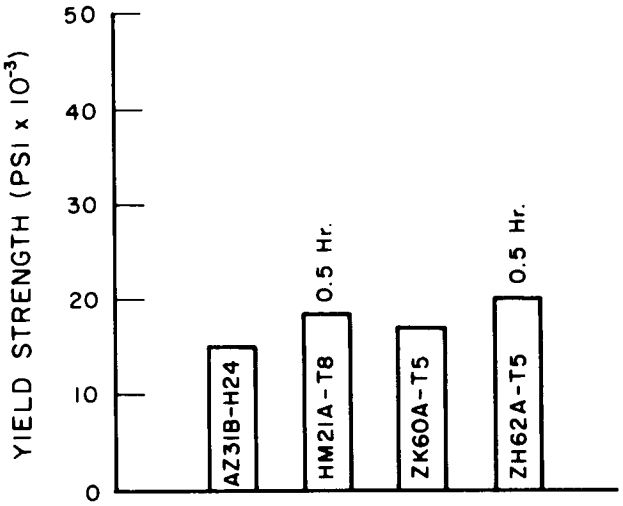


Figure Q. 2-14. Yield Strength of Magnesium Alloys at 3000°F After 1000 Hours at Temperature

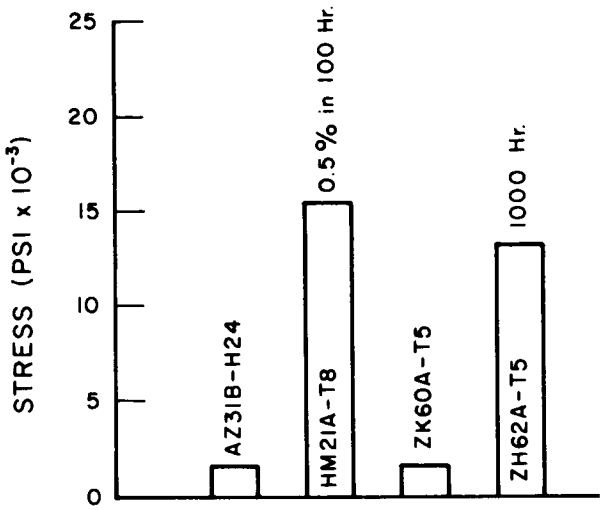


Figure Q. 2-15. Stress to Produce 1% Total Deformation in 500 Hours at 300°F

they both lose subsequent room temperature properties as a result of long-term exposure to the 300°F sterilization environment. The newest and lightest weight magnesium alloy, LA141, will not be suitable for use.

The HM21A-T8 and ZH62A-T5 thorium containing alloys have adequate properties at 300°F and also show no appreciable property changes as a result of long exposures of this temperature. ZH62A is a relatively new alloy, but it is being used currently for structural castings in space vehicles. HM21A has not been used to any degree in space vehicles, but it appears to be an attractive replacement for AZ31B.

HM31A appears attractive as a potential replacement for ZK60A for extrusions. What little data are available indicate good high temperature strength, stability at temperatures up to 600°F, and excellent creep resistance. Typical data for HM31A are shown below:

Temp (°F)	Yield Strength (psi)	Ultimate Strength (psi)	Stress for 0.5% Creep in 100 hr (psi)
RT	33,000	42,000	—
300	26,000	28,000	—
400	23,000	24,000	17,000
500	20,000	21,000	17,000

Q.2.3 ALUMINUM ALLOYS

A. General Characteristics

Because of their low density, high strength, formability and ready availability, aluminum alloys are utilized for a large percentage of the structure of various spacecraft. Some of the higher strength aluminum alloys, e.g., 7075-T6, rate among the strongest metallic materials based upon yield strength-to-density ratio. In general, most aluminum alloys will be suitable for use in spacecraft designed for thermal sterilization. Their high thermal conductivity will help simplify the thermal problems involved in designing for sterilization. Additionally, most of the commonly used alloys retain useful strength at 300°F and their properties are not seriously affected by long exposures at this temperature. Table Q.2-3 presents the chemical composition and heat treatment procedures for the alloys to be discussed in this section.

2014 is a relatively high strength alloy available as sheet, plate, extrusions and forgings. It has been used on space vehicles in the form of machined ring forgings. It is solution heat treated at 925°F followed by age hardening at 340°F for 12 hours; after this treatment it is designated as 2014-T6.

2024 is probably the most extensively used aluminum alloy in spacecraft structures. It is usually used as sheet or extrusions. This alloy is normally heat treated to either the T3 or the T4 condition with some material aging occurring at room temperature. T3 designates appreciable cold work and T4 indicates no cold work following a solution annealing operation at 920°F. Additional age hardening does take place at an elevated temperature of 375°F.

The 2219 alloy is a moderately high strength, heat treatable alloy noted for its weldability. Because of this and its good notch toughness it is being considered for the fabrication of cryogenic propellant storage vessels. This alloy is solution annealed at 1000°F and its T6 aging treatment of 18 hours at 375°F should make it resistant to overaging during thermal sterilization.

TABLE Q.2-3. CHEMICAL COMPOSITION AND HEAT TREATMENT OF SELECTED ALUMINUM ALLOYS

Chemical Composition

Element	2014	2024	2219	6061	7075	7079	7178	356
Copper	3.9 -5.0	3.8 -4.9	5.8 -6.8	0.15-0.40	1.2 -2.0	0.40-0.8	1.6 -2.4	0.020M
Iron	1.0 Max	0.50 Max	0.3 Max	0.7 Max	0.7 Max	0.40 Max	0.7 Max	0.20M
Silicon	0.5 -1.2	0.50 Max	0.20 Max	0.40-0.8	0.50 Max	0.30 Max	0.50 Max	6.5 -7.5
Manganese	0.4 -1.2	0.3 -0.9	0.20-0.40	0.15 Max	0.30 Max	0.10-0.30	0.30 Max	0.10M
Magnesium	0.2 -0.8	1.2 -1.8	0.02 Max	0.8 -1.2	2.1 -2.9	2.9 -3.7	2.4 -3.1	0.20-0.40
Chromium	0.10 Max	0.10 Max	—	0.15-0.35	0.18-0.40	0.10-0.25	0.18-0.40	—
Zinc	0.25 Max	0.25 Max	0.10 Max	0.25 Max	5.1 -6.1	3.4 -4.8	6.3 -7.3	0.10M
Titanium	0.15 Max	—	0.02-0.10	0.15 Max	0.20 Max	0.10 Max	0.20 Max	0.20M
Other, ea.	0.05 Max	0.05 Max	0.05 Max	0.05 Max	0.05 Max	0.05 Max	0.05 Max	0.05M
Other, Total	0.15 Max	0.15 Max	0.15 Max	0.15 Max	0.15 Max	0.15 Max	0.15 Max	0.15M
Zirconium	—	—	0.10-0.25	—	—	—	—	—
Vanadium	—	—	0.05-0.15	—	—	—	—	—

Heat Treatment

Alloy	Solution Anneal	Aging	Condition
2014	950°F	340°F for 12 hr.	T6
2024	920°F	375°F for 12 hr.	T6
2219	1000°F	375°F for 18 hr.	T6
6061	980°F	350°F for 10 hr.	T6
7075	900°F	250°F for 24 hr.	T6
7079	830°F	240°F for 50 hr.	T6
7178	900°F	280°F for 24 hr.	T6
356	None	440°F for 8 hr.	T51

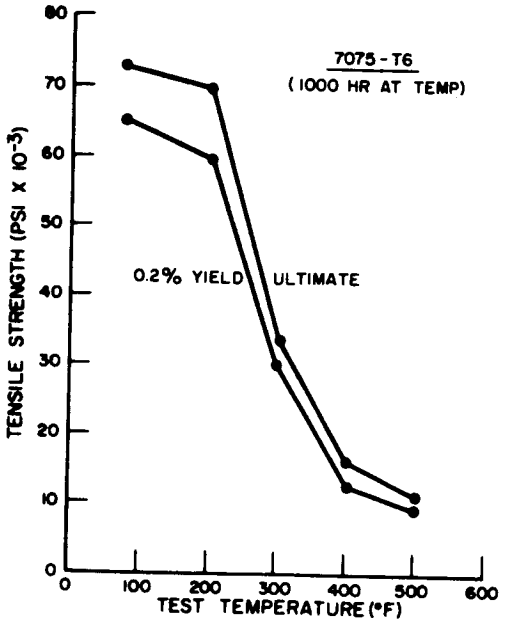


Figure Q.2-20. Elevated Temperature Tensile Strength 7075-T6 Following 1000 Hrs. Exposure at Test Temperature

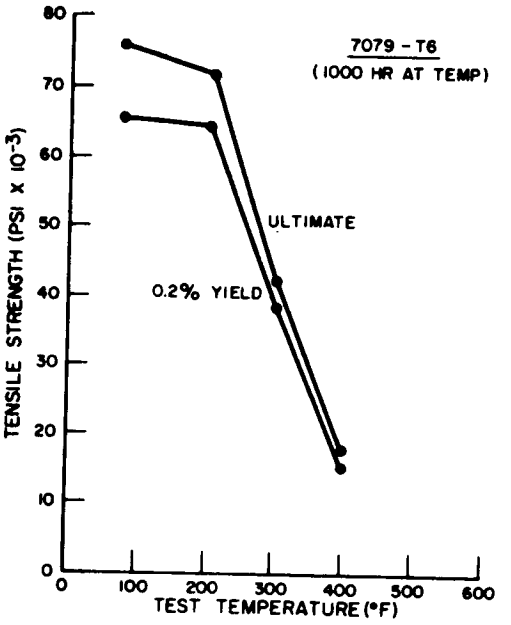


Figure Q.2-21. Elevated Temperature Tensile Strength 7079-T6 Following 1000 Hrs. Exposure at Test Temperature

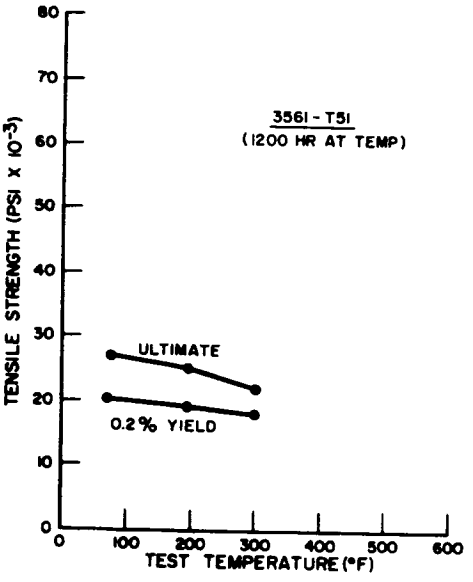


Figure Q.2-22. Elevated Temperature Tensile Strength 356-T51 Following 1200 Hrs. Exposure at Test Temperature

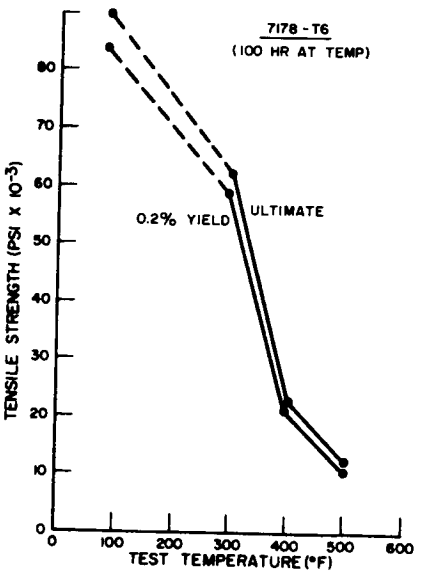


Figure Q.2-23. Elevated Temperature Tensile Strength 7178-T6 Following 100 Hrs. Exposure at Test Temperature

TABLE Q.2-4. PHYSICAL PROPERTIES OF SELECTED ALUMINUM ALLOYS

Temp (°F)	2014	2024	2219	6061	7075	7079	7178	356
				Specific Heat (Btu/lb/°F)				
RT		0.205			0.195			
100		0.210			0.195			
200	0.23	0.220		0.23	0.192		0.23	0.23
300		0.228			0.187			
400		0.232			0.180			
				Thermal Conductivity (Btu/hr/ft^2/ft/°F)				
RT	90	71		90	72	70	70	95
100	-	72		-	74	-	-	-
200	100(250°F)	78		-	79	-	-	-
300	-	85		-	91	-	-	-
400	-	95		-	99	-	-	-
				Avg. Coefficient of Thermal Expansion (in/in/°F x 10^6)				
RT-200	11.6	12.3		13.0	12.9	13.0	13.0	11.5
RT-300	10.9	12.2		-	13.1	-	-	11.7
RT-400	11.2	13.3		13.5	13.5	13.7	-	12.1

To show how mechanical properties other than tensile yield and ultimate are affected by temperature, data are plotted for 2014-T6 in Figure Q.2-24. All of the properties are similarily affected on the basis of percentage of initial room temperature properties although there is some scattering at the higher temperatures.

Creep data for the various aluminum alloys are shown in Figures Q.2-25 through Q.2-31, where stress to produce given percentages of deformation in 1000 hours are plotted versus test temperature. Stress to produce 1.0% creep deformation in 1000 hours at 300°F ranges between 20,000 and 35,000 psi for all alloys. No creep data were available for 7178-T6.

Next to 2024, the 6061 alloy is probably the most used in spacecraft construction. It is not a high-strength alloy, but it has the advantage of being weldable and brazeable. It is readily available in most any desired form and can be easily formed. It is solution annealed at 980°F and aged at 350°F to the T6 condition.

7075, 7079, and 7178 are all of interest because they can be heat treated to the highest strength of the common aluminum alloys. Following solution annealing, they are aged at approximately 250°F. This temperature is lower than that proposed for sterilization. These alloys, therefore, are probably the most susceptible to irreversible structural and property changes during sterilization.

Alloy 356 has been included because of its high usage in structural castings. The heat treatment for this alloy consists of aging at 440°F directly after casting to the T51 condition and a solution anneal at 1000°F followed by aging at 310°F to the slightly higher strength T6 condition.

B. Physical Properties

Various physical properties of interest for thermal sterilization are presented in Table Q.2-4. The available data are somewhat incomplete for 2014, 2219, 6061, 7079, 7178, and 356.

C. Mechanical Properties

The tensile yield and ultimate strengths at elevated temperatures for all of the alloys are plotted in Figures Q.2-16 through Q.2-23. Exposure at temperature was 1000 hours prior to testing for all alloys except 356-T51 and 7178-T6 where holding times were 1200 and 100 hours, respectively. Except for the casting alloy, 356-T51, all alloys have substantial yield strengths at 300°F with the values ranging from 30,000 to 45,000 psi. 356-T51 retains about 90% of its initially low yield strength of 20,000 psi.

All alloys, excluding 2219-T6 and 356-T51 which display a gradual decrease in properties with temperature, show a rather sharp break in tensile strength properties as the temperature is increased. 7075-T6 and 7079-T6 both exhibit this phenomenon between 200 and 300°F, while 2014-T6, 2024-T4, 6061-T6 and 7178-T6 show a marked loss in strength between 300 and 400°F. As would be expected, the temperature range in which this occurs for each alloy correlates with its precipitation hardening temperature. An increase in yield strength is observed for 2024-T4 at 300°F. This is a result of additional age hardening with maximum strengthening obtained after about 500 hours at this temperature.

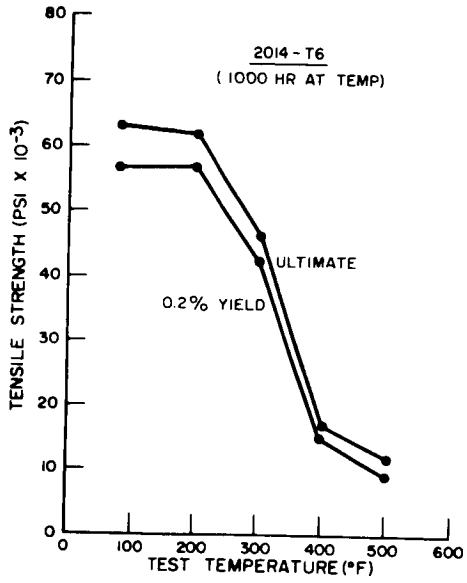


Figure Q.2-16. Elevated Temperature Tensile Strength 2014-T6 Following 1000 Hrs Exposure at Test Temperature

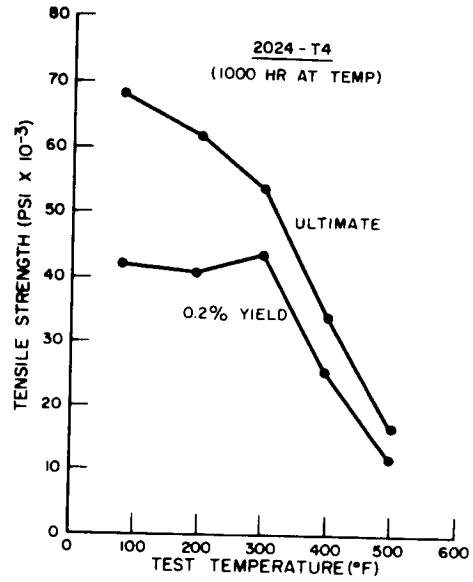


Figure Q.2-17. Elevated Temperature Tensile Strength 2024-T4 Following 1000 Hrs Exposure at Test Temperature

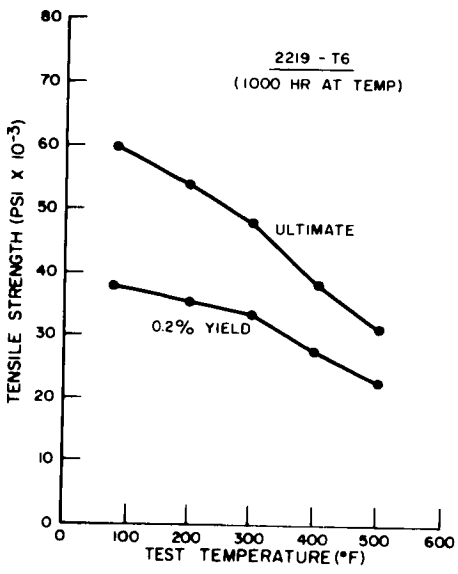


Figure Q.2-18. Elevated Temperature Tensile Strength 2219-T6 Following 1000 Hrs Exposure at Test Temperature

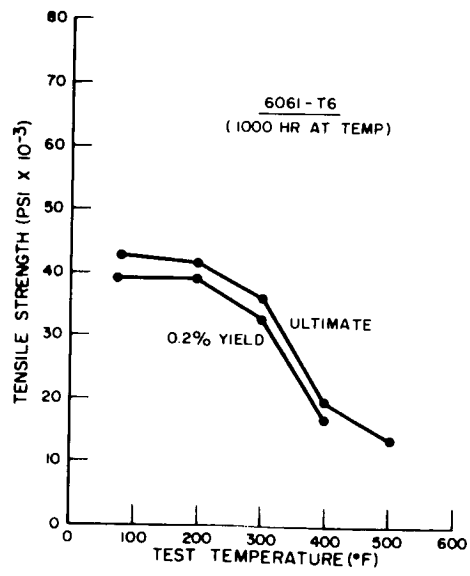


Figure Q.2-19. Elevated Temperature Tensile Strength 6061-T6 Following 1000 Hrs Exposure at Test Temperature

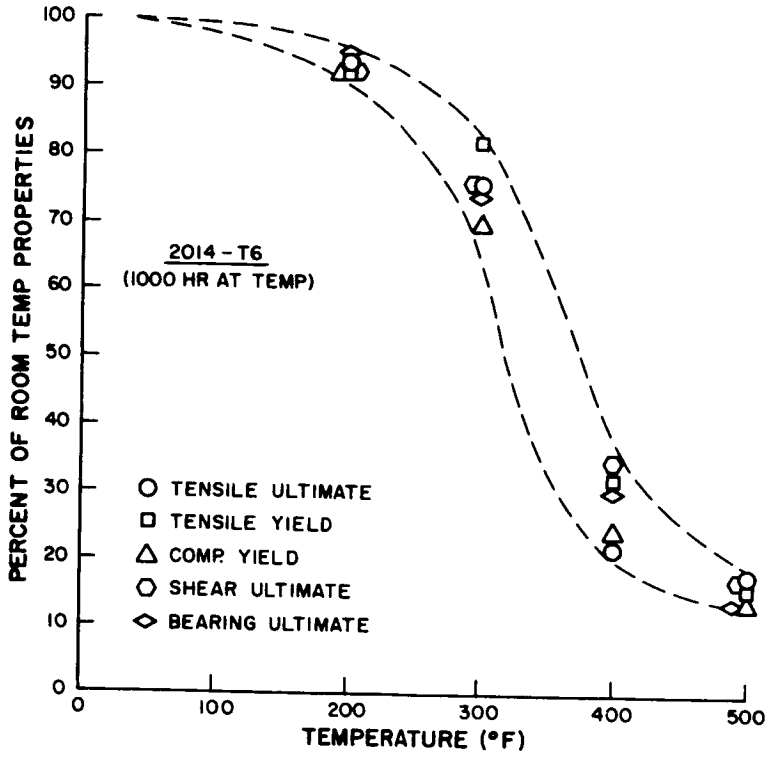


Figure Q. 2-24. Change in Mechanical Design Properties of 2014-T6 with Temperature

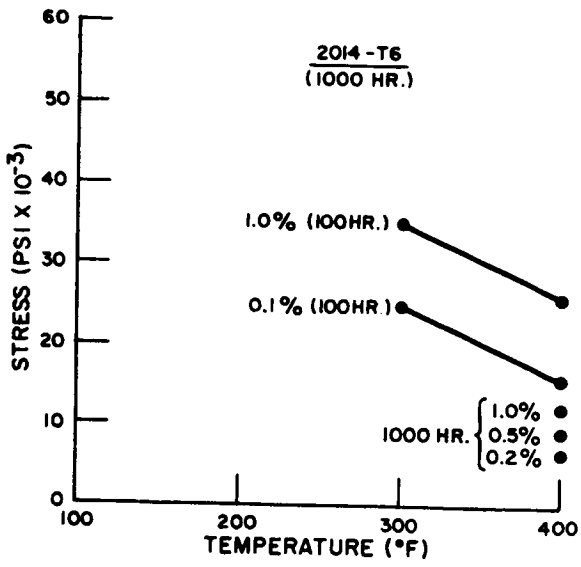


Figure Q. 2-25. Stress to Produce given Creep Deformations in 1000 hours at various Temperatures for 2014-T6

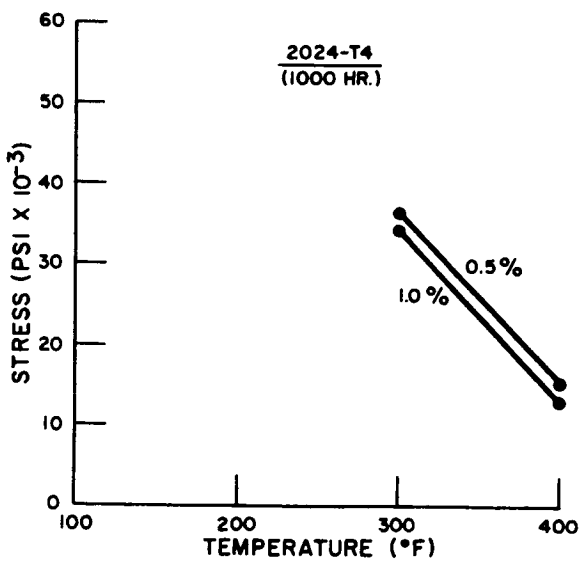


Figure Q. 2-26. Stress to produce given Creep Deformations in 1000 hours at various Temperatures for 2024-T4

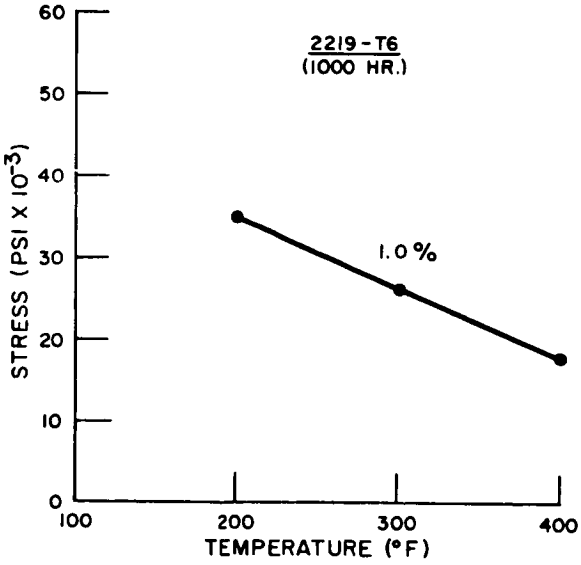


Figure Q. 2-27. Stress to Produce given Creep Deformations in 1000 hours at Various Temperatures for 2019-T6

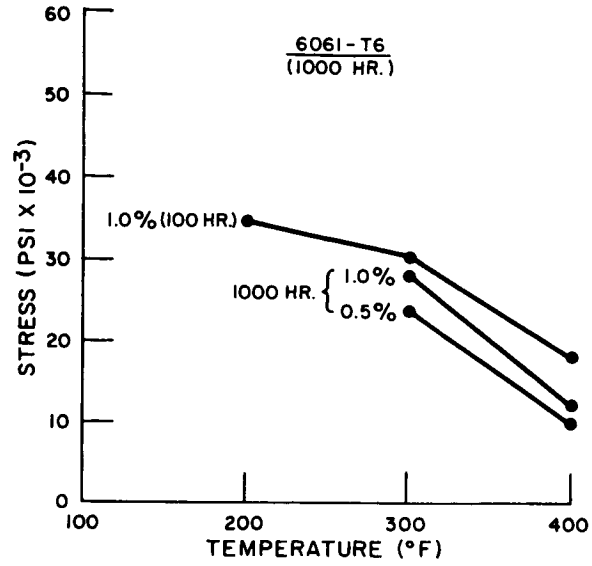


Figure Q. 2-28. Stress to Produce given Total Deformations in 1000 hours at Various Temperatures for 6061-T6

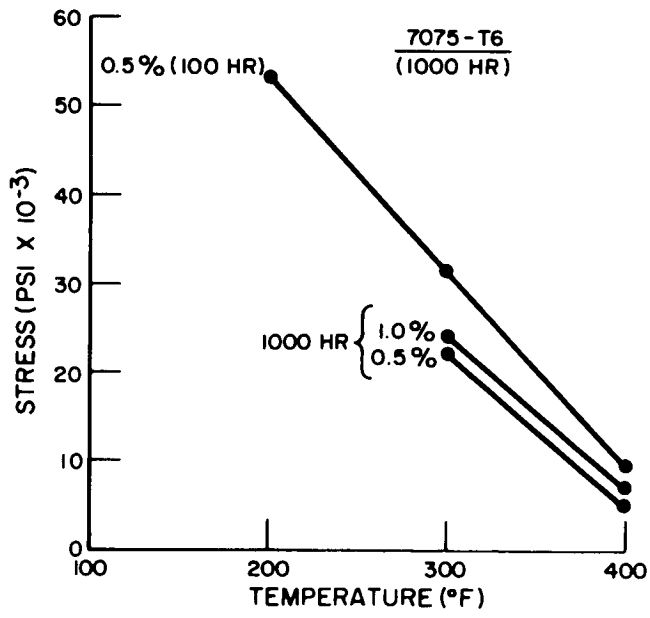


Figure Q. 2-29. Stress to Produce given Creep Deformations in 1000 hour at Various Temperatures for 7075-T-6.

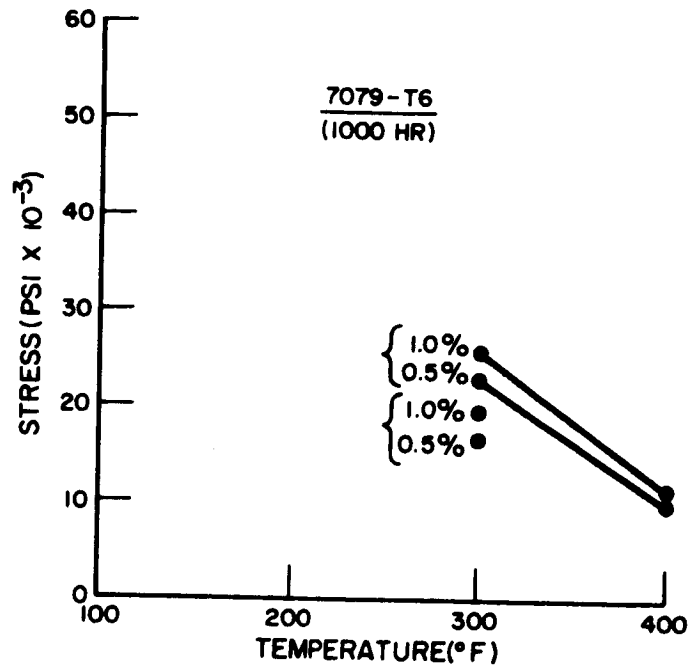


Figure Q. 2-30. Stress to Produce given Creep Deformations in 1000 Hour at Various Temperatures for 7079-T6

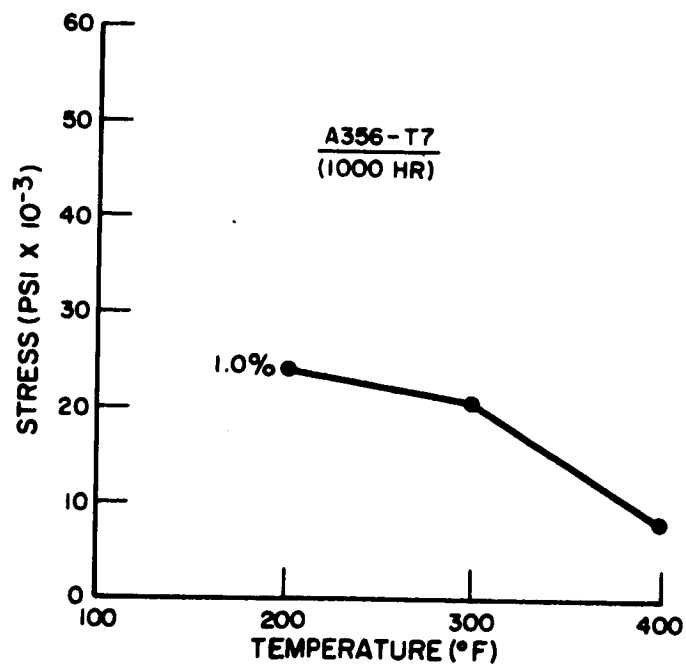


Figure Q. 2-31. Stress to Produce given Total Deformations in 1000 Hour at Various Temperatures for A356-T7

The changes in room temperature tensile properties produced by long term (1000 hours) exposures at various elevated temperatures are shown in Figures Q. 2-32 to Q. 2-39. 2014-T6, 2024-T4, 2219-T6 and 6061-T6 all show essentially no change as a result of exposures up to 300°F. Again an additional age hardening reaction takes place in 2024-T4 at 300°F resulting in a slight increase in subsequent room-temperature yield strength. The high strength alloys, 7075-T6, 7079-T6 and 7178-T6, all exhibit a significant loss in room-temperature properties at exposure temperatures much above 200°F. Again these temperatures at which losses in strength are produced correlate with age hardening temperature ranges. Over-aging causes the decrease in room temperature properties.

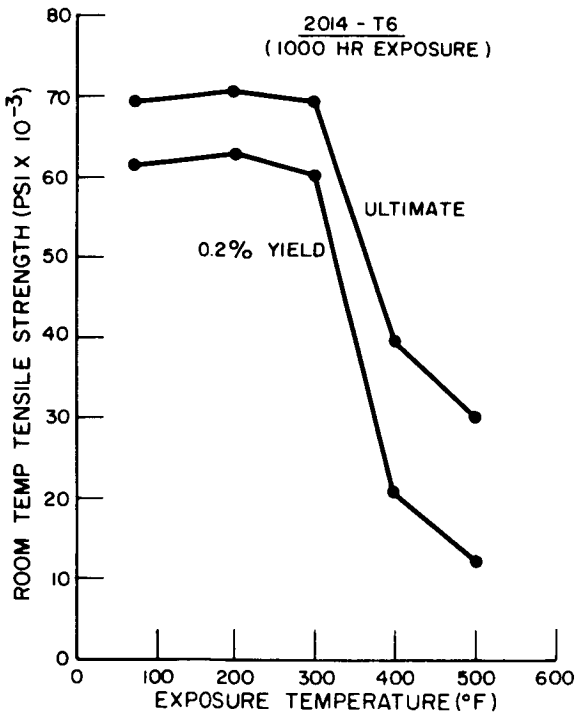


Figure Q. 2-32. Effects of 1000 Hour Exposure at Elevated Temperature upon Subsequent Room Temperature Tensile Properties of 2014-T6

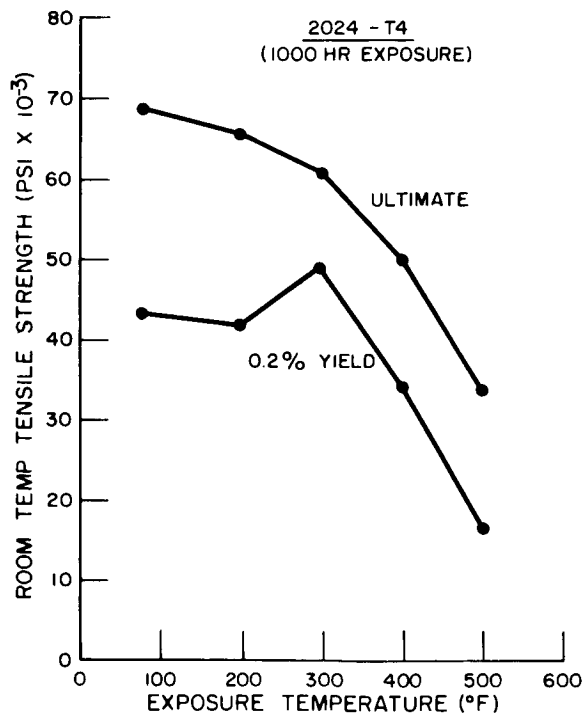


Figure Q. 2-33. Effects of 1000 Hour Exposure at Elevated Temperature upon Subsequent Room Temperature Tensile Properties of 2024-T4

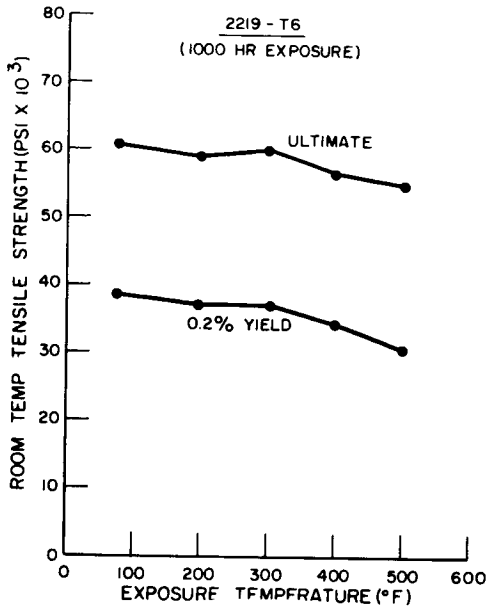


Figure Q. 2-34. Effects of 1000 Hour Exposure at Elevated Temperature upon Subsequent Room Temperature Tensile Properties of 2214-T6

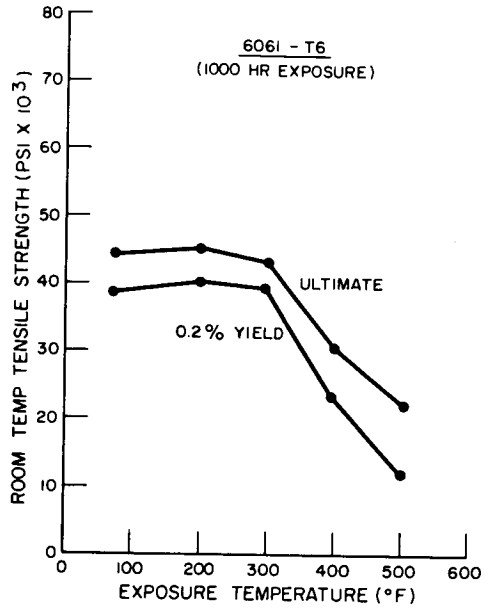


Figure Q. 2-35. Effects of 1000 Hour Exposure at Elevated Temperature upon Subsequent Room Temperature Tensile Properties of 6061-T6

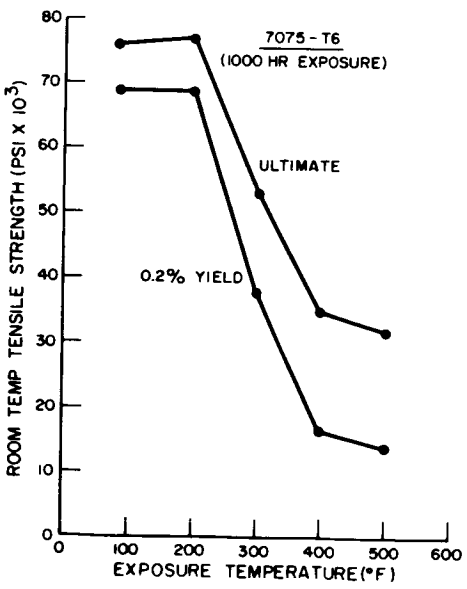


Figure Q. 2-36. Effects of 1000 Hour Exposure at Elevated Temperature upon Subsequent Room Temperature Tensile Properties of 7075-T6

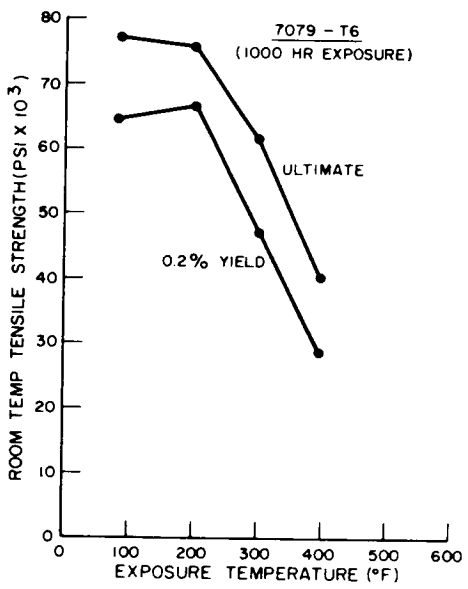


Figure Q. 2-37. Effects of 1000 Hour Exposure at Elevated Temperature upon Subsequent Room Temperature Tensile Properties of 7079-T6

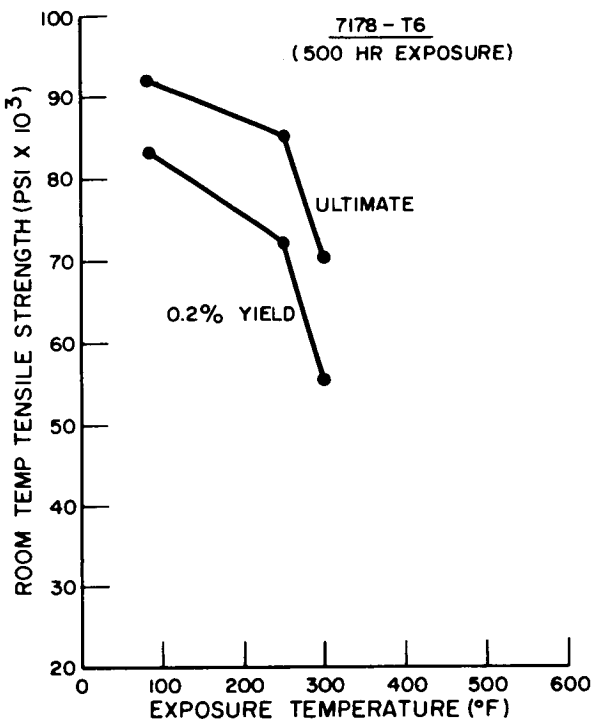


Figure Q. 2-38. Effects of 500 Hour Exposure at Elevated Temperature upon Subsequent Room Temperature Tensile Properties of 7178-T6

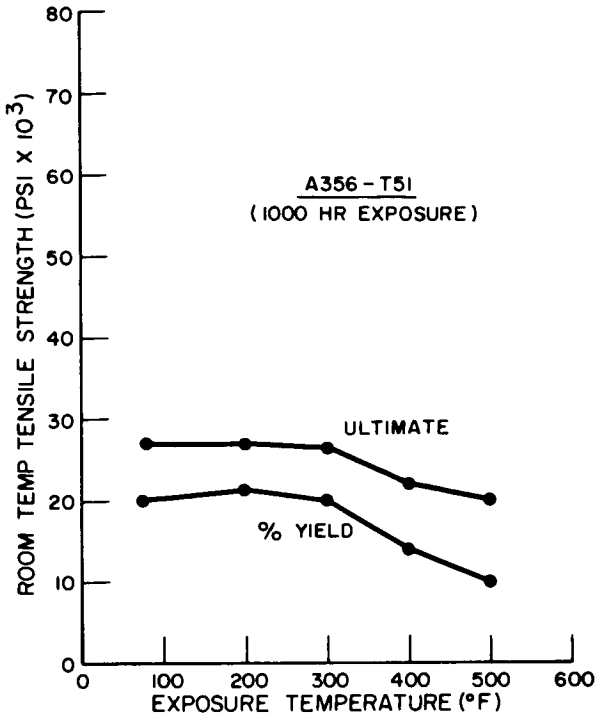


Figure Q. 2-39. Effects of 1000 Hour Exposure at Elevated Temperature upon Subsequent Room Temperature Tensile Properties of A356-T51

D. Suitability of Aluminum Alloys

Aluminum alloys are suitable for spacecraft structures which must undergo thermal sterilization. Their high-thermal conductivity makes them attractive from a heat transfer standpoint but their high coefficient of expansion can pose problems wherever they are in contact with dissimilar metals.

A summary of the effects of the 300°F sterilization cycle upon mechanical properties of the various aluminum alloys is shown in Figures Q. 2-40, Q. 2-41 and Q. 2-42. 2014-T6 and 2024-T4 appear superior to all of the other alloys with the possible exception of 7178-T6. Additional data are required for this alloy before it can be considered. 2014-T6 and 2024-T4 have the highest yield strength and creep strength at 300°F and retain the highest room temperature yield strength following exposure. The high strength alloys, 7075-T6, 7079-T6 and 7178-T6 lose such a large percentage of their strength as a result of overaging at 300°F that they actually end up having lower strength than 2014-T6 and 2024-T4; 6061-T6 withstands the sterilization cycle and although it has lower strength, it should be selected for welded applications.

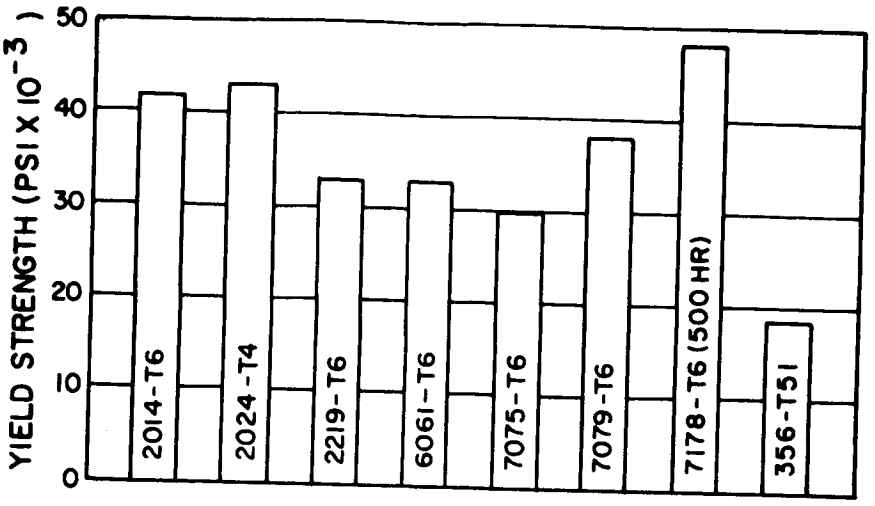


Figure Q. 2-40. Yield Strength at 300°F after 1000 Hours at Temperature

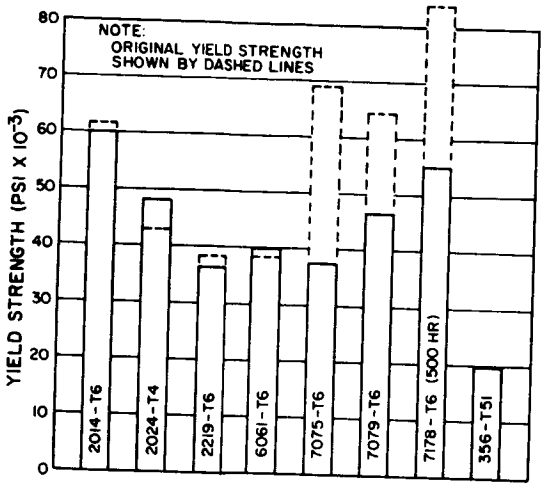


Figure Q. 2-41. Yield Strength at Room Temperature after Exposure for 1000 Hour at 300°F

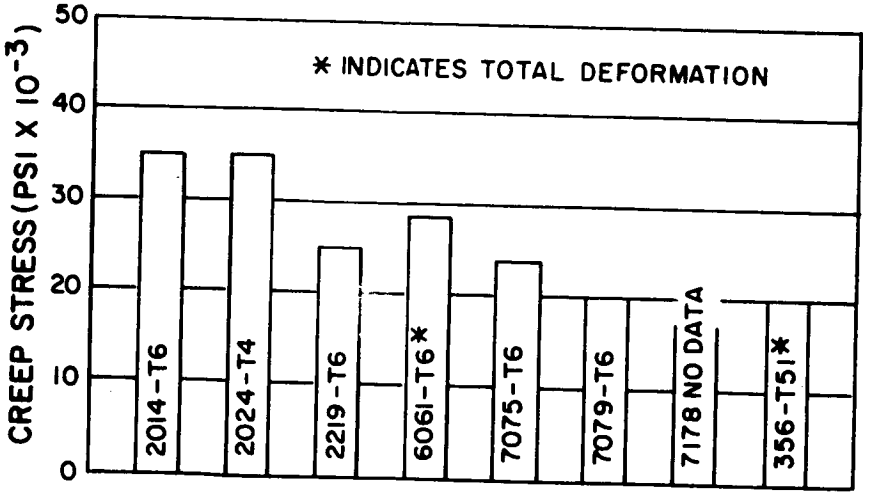


Figure Q. 2-42. Stress to Produce 1.0% Creep Deformation in 1000 Hours at 300°F

Q.2.4. TITANIUM ALLOYS

A. General Characteristics

Titanium alloys are assuming increasing importance as spacecraft materials because of their high strength-to-density ratio. Unalloyed titanium and the 5Al-2.5Sn alloy were used extensively on the Mercury space capsule as structural members while the 6Al-4V alloy has become established as the standard material for pressure vessels on space vehicles. Other than attachment hardware and pressure vessels, the use of titanium alloys has been limited on the current generation of satellites and space probes. They are being considered for structural materials on the Voyager - Venus lander and solar probe vehicles because of their excellent strength to density ratio at temperatures up to 1000°F.

The 300°F thermal sterilization temperature is considered mild for titanium alloys and will not restrict their use. Their low thermal conductivity will intensify the heat transfer problem. If used in combination with aluminum and magnesium alloys, the relatively low thermal expansion of titanium could also result in distortion during heating.

Typical chemical compositions and heat treatment temperatures for the most frequently used titanium alloys are shown in Table Q.2-5. Those titanium alloys which are heat treatable are age hardened at temperatures ranging from 900 to 1050°F.

This is considerably higher than the aging temperatures used for aluminum and magnesium alloys and explains why titanium alloys retain their strength to much higher temperatures.

5Al-2.5Sn, one of the most extensively used alloys, is a non-heat treatable, single-phase alpha alloy with useable strength at 800-1000°F. This alloy is available in all common forms, viz., sheets, plates, bars and forgings. Because it is weldable and retains good notch toughness at extremely low temperatures, it is being used for cryogenic propellant tanks.

TABLE Q.2-5. CHEMICAL COMPOSITION AND HEAT TREATMENT FOR SELECTED TITANIUM ALLOYS

Chemical Composition

Element	5Al-2.5Sn	6Al-4V	6Al-6V-2Sn	4Al-3Mo-1V	13V-11Cr-3Al
Aluminum	5.0	6.0	5.5	4.0	3.0
Vanadium	—	4.0	5.5	1.0	13.0
Chromium	—	—	—	—	11.0
Tin	2.5	—	2.0	—	—
Molybdenum	—	—	—	3.0	—
Iron	—	—	0.7	—	—
Copper	—	—	0.7	—	—
Titanium	Bal.	Bal.	Bal.	Bal.	Bal.

Heat Treatment

Alloy	Solution Anneal	Aging
5Al-2.5Sn	Cannot be age hardened	
6Al-4V	1700°F	1000°F for 4 hrs.
6Al-6V-2Sn	1650°F	1050°F for 4 hrs.
4Al-3Mo-1V	1630°F	925°F for 12 hrs.
13V-11Cr-3Al	1450°F	900°F for 72 hrs.

Mixed alpha-beta, two phase alloys include 6Al-4V, 6Al-6V-2Sn and 4Al-3Mo-IV. These alloys can be heat treated to considerably higher strength than the alpha alloys. The 6Al-4V alloy is probably the most widely used of all the titanium alloys. It has moderately high strength, and is the most weldable of the heat treatable alloys. Gas storage pressure vessels for many space vehicles are fabricated from this alloy. The 6Al-4V alloy is thermally stable for extended periods of time for temperatures up to 950°F.

The 13V-11Cr-3Al alloy is the only all-beta alloy produced. It is the most formable of the titanium alloys and is capable of heat treatment to high strength levels. Its rather poor weldability, low notch toughness and thermal instability above 600°F have restricted its use.

B. Physical Properties

Physical properties which are of importance for thermal sterilization are presented on Table Q. 2-6 for the titanium alloys. No significant differences exist among the properties of the various alloys. As noted previously, the thermal conductivity of titanium alloys is much lower than that of aluminum or magnesium and would affect the heating and cooling rates during sterilization. The thermal expansion of titanium alloys is also much less than aluminum or magnesium alloys and could lead to distortion problems.

C. Mechanical Properties

Room and elevated temperature tensile properties for the various alloys are plotted in Figures Q. 2-43 through Q. 2-47. Except for the 5A1-2.5Sn alloy, all of the data represent properties after 0.5 hours at temperature. All of the alloys retain high strength at the 300°F sterilization temperature. As can be seen in Figure Q. 2-48, the elastic moduli of titanium alloys do not decrease significantly from their value at room temperature.

Creep data at the lower temperatures is limited. Stress to produce 1% creep deformation at 400 and 600°F is given in Table Q. 2-7 for 6A1-4V, 4A1-3Mo-1V and 13V-11Cr-3A1. Although the data are somewhat meager, it is obvious that creep would not be a problem at 300°F. Stability data are shown in Table Q. 2-8. Again, all the test data are for temperatures greater than 300°F the lowest being 500°F. Since the room temperature properties are not significantly affected by the extended periods of time at these high temperatures, it is reasonable to assume that they will withstand 300°F.

Metallurgical stability of welds is of greater concern than that of the parent material shown in Table Q. 2-8 since fusion welds and spot welds in the 13V-11Cr-3A1 alloy tend to be unstable at relatively low temperatures. Resistance welds in the 4A1-3Mo-IV alloy may also degrade; additional data are needed in this area.

D. Suitability of Titanium Alloys

Titanium alloys have the highest strength-to-density ratio at temperatures up to 400°F of any of the commonly used alloys. From this standpoint, they are well suited for structural applications in spacecraft requiring sterilization. It is doubtful whether this advantage of the titanium alloys will result in their replacing aluminum and magnesium alloys to a large extent except for Solar-Probe vehicles and Venus landers where temperatures much higher than those used for sterilization will be encountered.

TABLE Q.2-6. PHYSICAL PROPERTIES OF SELECTED TITANIUM ALLOYS

Temp. (°F)	5Al-2.5Sn	6Al-4V	6Al-6V-2Sn	4Al-3Mo-1V	13V-11Cr-3Al
			Specific Heat (Btu/16/°F)		
RT	.126	.135	.155	.132	.130
200	.131	.135	.157	.132	.128
300	—	—	.158	—	—
400	.137	.137	.160	.134	.137
500	—	—	.161	—	—
600	.145	.144	.162	.140	.160
			Thermal Conductivity (Btu/hr/ft²/ft/°F)		
RT	4.4	3.8	No data	3.9	3.7
200	4.8	4.3		4.4	4.3
300	—	—		—	—
400	5.6	5.1		5.1	5.4
500	—	—		—	—
600	6.3	5.9		6.0	6.7
			Avg. Coefficient of Thermal Expansion (in/in/°F x 10^6)		
RT to 200	5.2	4.8	5.0	5.0	5.2
RT to 400	5.2	5.0	5.0	5.1	5.3
RT to 600	5.3	5.1	5.2	5.3	5.4

TABLE Q.2-7. CREEP STRENGTH OF SELECTED TITANIUM ALLOYS

Temp. (°F)	6Al-4V	4Al-3Mo-1V	13V-11Cr-3Al
	Stress for 1% Creep Deformation in 1000 hrs.		
600	120,000	145,000	—
	Stress for 1% Creep Deformation in 1000 hrs.		
RT	—	—	120,000
400	—	—	100,000
600	80,000	—	—

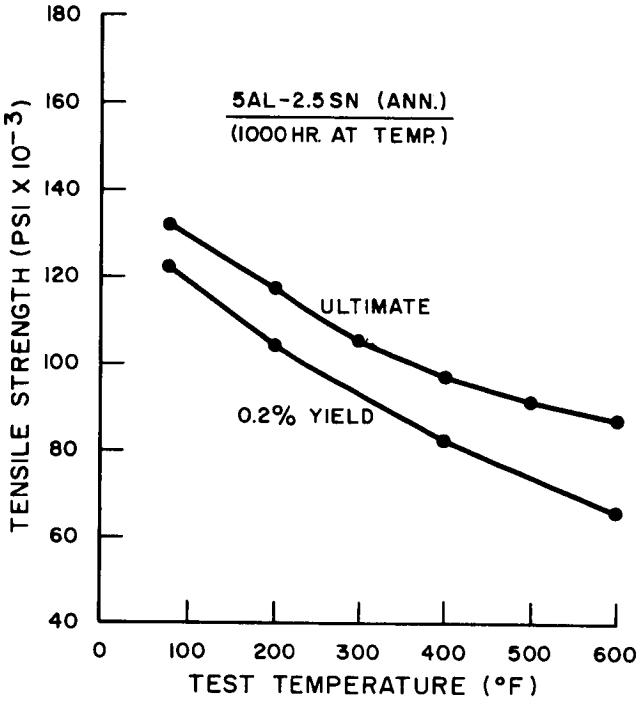


Figure Q. 2-43. Effects of Elevated Temperature on the Tensile Properties of 5Al-2.5Sn (aged) Titanium Alloy

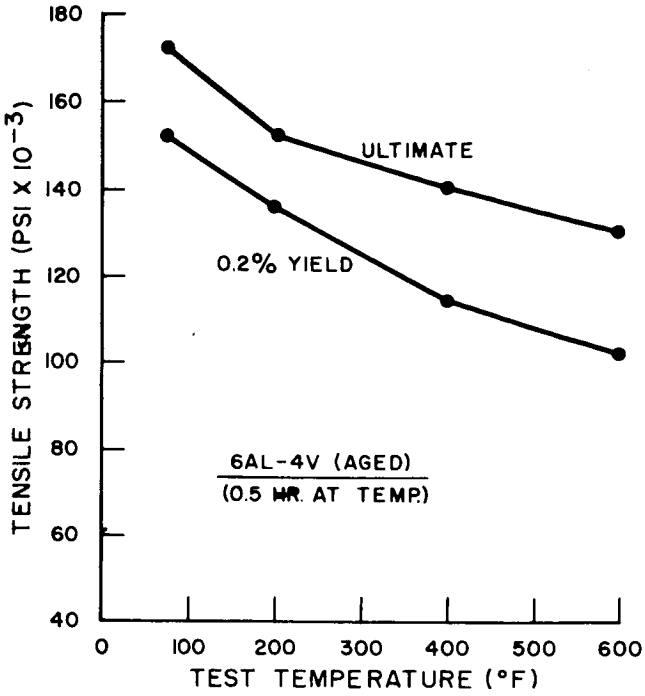


Figure Q. 2-44. Effects of Elevated Temperature on the Tensile Properties of 6Al-2.5Sn (aged) Titanium Alloy

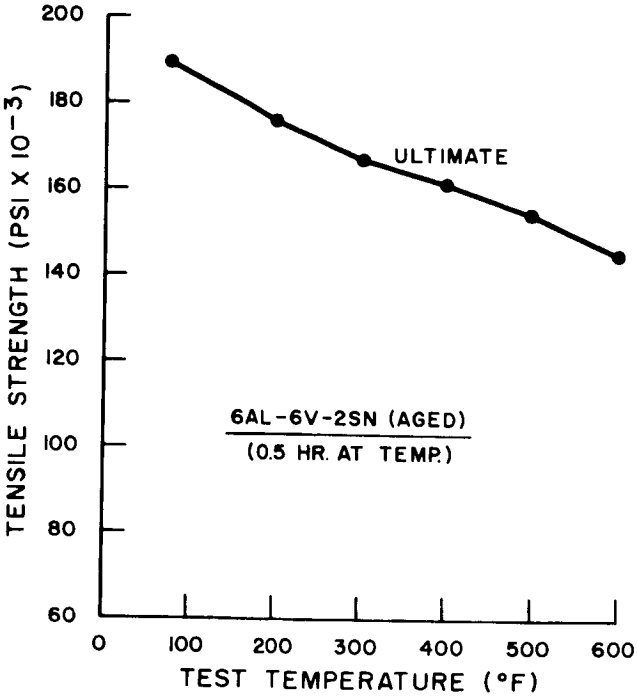


Figure Q. 2-45. Effects of Elevated Temperature on the Tensile Properties of 6Al-6V-2Sn (aged) Titanium Alloy

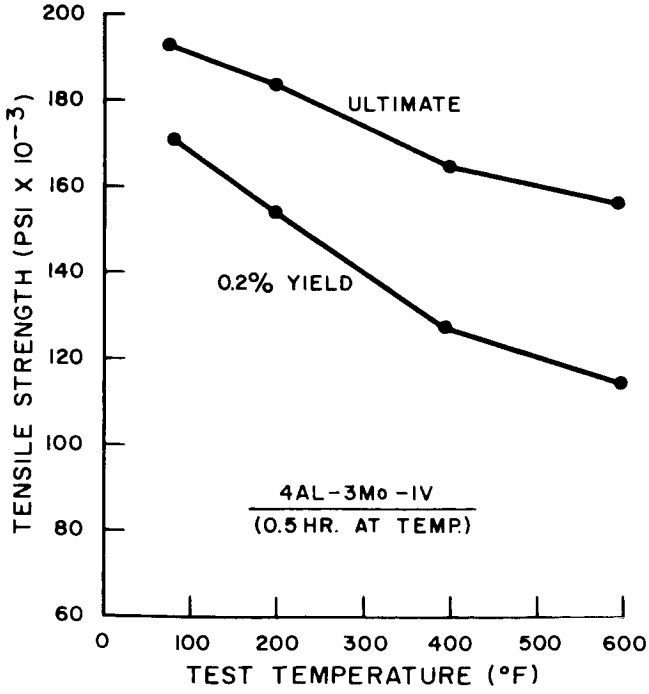


Figure Q. 2-46. Effects of Elevated Temperature on the Tensile Properties of 4Al-3Mo-1V Titanium Alloy

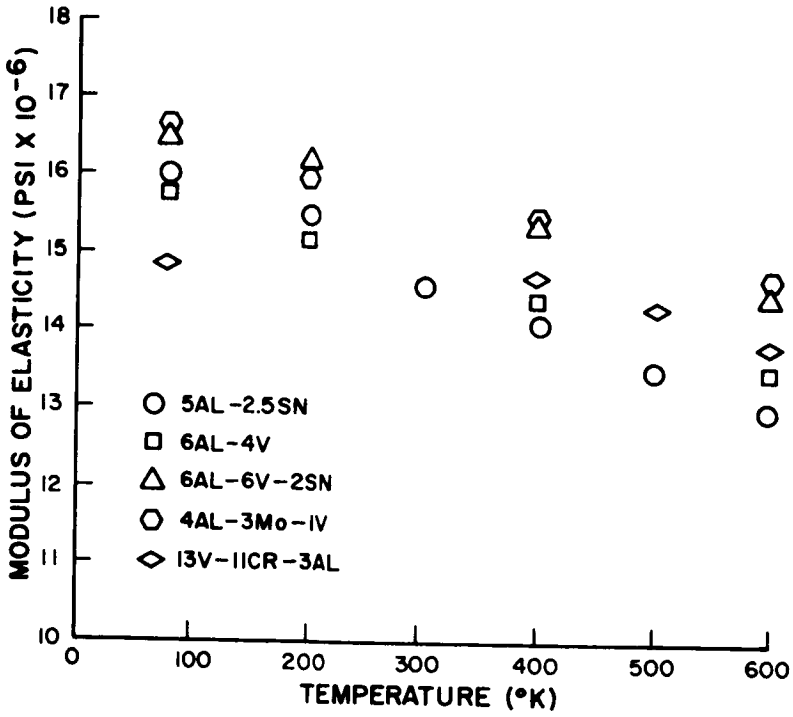


Figure Q. 2-48. Modulus of Elasticity of Titanium Alloys at Elevated Temperatures

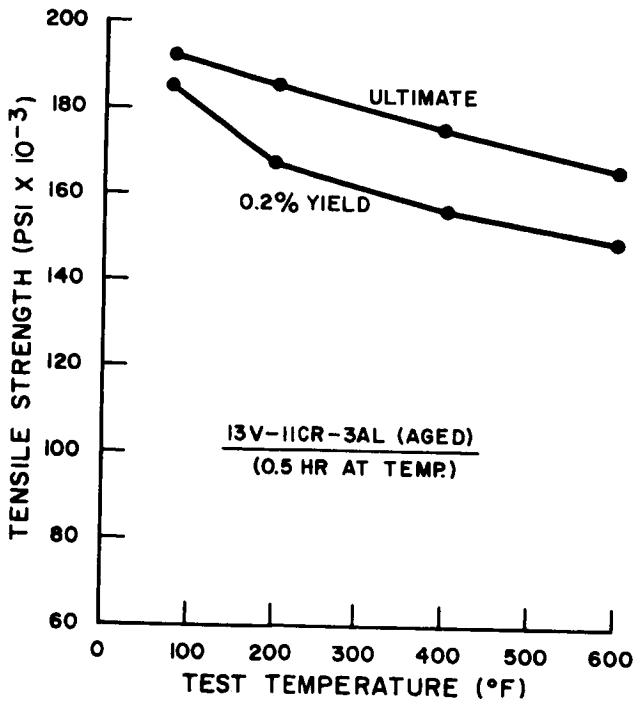


Figure Q. 2-47. Effects of Elevated Temperature on the Tensile Properties of 13V-11Cr-3Al (aged)

TABLE Q.2-8. EFFECTS OF EXPOSURE AT ELEVATED TEMPERATURES UPON SUBSEQUENT ROOM TEMPERATURE PROPERTIES OF TITANUM ALLOYS

Alloy	Exposure Conditions		Resultant Room Temp. Properties				
	Temp. (°F)	Time (Hr)	Stress (psi)	0.2% Y.S. (psi)	Ultimate (psi)	Elong. (%)	R.A. (%)
6Al-4V	—	—	—	124,000	134,000	20	—
(Annealed)	650	300	50,000	133,000	144,000	18	—
6Al-6V-25V	—	—	—	169,000	180,000	11	24
(Annealed)	700	1,000	30,000	168,000	182,000	11	23
4Al-3Mo-1V	—	—	—	167,000	190,000	7	—
(Annealed)	500	1,000	100,000	170,000	190,000	7	—
13V-11Cr-3Al	—	—	—	174,000	202,000	6	—
(Annealed)	500	500	140,000	185,000	201,000	6	—

Q.2.5 STEEL ALLOYS

A. General Characteristics

Steel alloys, like titanium alloys, are not extensively used in space vehicle structures because buckling is usually the limiting criteria. The precipitation hardening stainless steels, and the modified hot work die steels rank with the titanium alloys on the basis of strength-to-density ratios at elevated temperatures. For this reason, these alloys are also being considered for use in the Venus lander and Solar Probe vehicles.

Steel alloys are the most complex of all metallic materials. They vary markedly in properties and behavior depending upon their composition and heat treatment. For this reason, one of each type used most frequently in aerospace vehicles is included in this evaluation of behavior during thermal sterilization. Chemical compositions and heat treatment procedures for the steel alloys discussed are shown in Table Q.2-9.

AISI type 301 austenitic stainless steel cannot be strengthened by heat treatment. Additional strength, if desired, can be obtained through strain hardening by means

TABLE Q.2-9. CHEMICAL COMPOSITIONS AND HEAT TREATMENT PROCEDURES FOR STEEL ALLOYS

Chemical Composition

Element	AISI 301	PH 15-7 Mo	AISI H-11	AISI 4340	18% N, 9%Co, 5%Mo
Carbon	0.15 max	0.09 max	0.38-0.43	0.38-0.43	0.03 max
Manganese	2.00 max	1.00 max	0.20-0.40	0.60-0.80	0.10 max
Silicon	1.00 max	1.00 max	0.80-1.20	0.20-0.35	—
Chromium	16.00-18.00	14.00-16.00	4.75-5.50	0.70-0.90	—
Nickel	6.00- 8.00	6.50- 7.75	—	1.65-2.00	18.00-19.00
Cobalt	—	—	—	—	8.50- 9.50
Molybdenum	—	2.00- 3.00	1.25-1.75	0.20-0.30	4.60- 5.20
Vanadium	—	—	0.30-0.50	—	—
Aluminum	—	0.75- 1.5	—	—	0.05- 0.15
Titanium	—	—	—	—	0.50- 0.80

Heat Treatment

Alloy	Heat Treatment Procedure
AISI 301	Not heat treatable, strengthened by cold working
PH 15-7Mo	Cond TH1050 — Anneal at 1950°F, transform at 1400°F, age at 1050°F.
PH 15-7Mo	Cond RH950 — Anneal at 1750°F, cool to -100°F, age at 950°F.
AISI H-11	Heat to 1850°F, air cool, temperature at 1050°F.
AISI 4340	Heat to 1600°F, oil quench, temperature at 800°F.
18 Ni-9Co-5Mo	Anneal at 1500°F, age at 900°F.

of cold reduction. The fully austenitic steels have been widely used for welded cryogenic propellant tanks since they retain ductility at liquid hydrogen temperatures (-423°F). Their application on current space vehicles is limited to specialized components or attachments.

The semi-austenitic, precipitation hardening class of stainless steels to which PH15-7Mo belongs have an excellent combination of strength, temperature resistance and corrosion resistance. They are being used for brazed honeycomb and structural members for supersonic aircraft where skin temperatures exceed the capability of aluminum alloys. Two types of heat treatments can be used to strengthen the alloy; they differ in the manner in which the austenitic structure

is transformed to martensite. The TH treatment makes use of phase changes at 1400°F while the RH treatment requires refrigeration. Age hardening is then used to strengthen the martensitic structure obtained from both treatments. These steels are being considered for the construction of Venus landers.

The AISI-H11 steels were originally developed for hot working dies. Because of their high strength at temperatures up to 1000°F, they were slightly modified and adopted for use in aerospace applications. These steels are unique among the low alloy class because their martensitic matrix is further strengthened by a precipitation of carbides at temperatures of 900 to 1100°F. This "secondary hardening" provides high strength at elevated temperatures. They are not extensively used in current space vehicles.

AISI4340 is a standard, commonly used low alloy steel strengthened by oil quenches and tempering at 800-900°F. It retains significant strength at temperatures 100°F below that used for tempering. Its use on previous space vehicles has been limited.

The 18% Ni-9% Co-5% Mo, maraging steel, is a newly developed ultra-high strength steel having the highest yield strength-to-density ratio of any commercially available alloy. It requires only a simple aging treatment at 900°F for strengthening. Most important, however, is its retention of notch toughness at extremely high strength levels. This alloy is currently undergoing testing and utilization for developmental hardware, particularly pressure vessels.

B. Physical Properties

Pertinent physical properties are tabulated in Table Q.2-10. Steel alloys, like titanium, have low thermal conductivity and low thermal expansion when compared to magnesium and aluminum alloys. The low alloy steels, H-11 and 4340, have about twice the thermal conductivity of the stainless types (however, it is still low). Care must be taken if steels are used in conjunction with aluminum and magnesium alloys.

C. Mechanical Properties

The effects of elevated temperatures upon the tensile properties of the selected steel alloys are shown in Figures Q.2-49 through Q.2-53. The data for AISI 301 and PH 15-7Mo represent strength at temperature following a 1000 hour prior soak while those for the remaining alloys represent short time (0.5 hrs) at temperature. Strength at 300°F is obviously no problem for any of the alloys. AISI 4340 is the only one of the alloys not normally considered as a material for elevated temperature service. When tempered at 800°F, though, it is perfectly suitable for indefinite use at 300°F.

TABLE Q.2-10. PHYSICAL PROPERTIES OF STEEL ALLOYS

Temp. (°F)	AISI 301	PH15-7Mo	AISI H-11	AISI 4340	18Ni-9Co-5Mo
		Specific Heat (Btu/lb/°F)			
RT	0.110	Similar to 301	Similar to 4340	—	No data available
200	0.118			0.118	
300	0.123			—	
400	0.127			0.125	
600	0.133			0.137	
		Thermal Conductivity (Btu/hr/ft^2/ft/°F)			
RT	8.5	—	—	16-19	No data available
200	9.0	—	—	16-10	
300	10.0	9.5	—	16-19	
400	10.5	10.5	16.5	16-19	
600	11.0	—	16.5	16-19	
		Avg. Coefficient of Thermal Expansion (in/in/°F 10^6)			
RT-200	9.4	6.1	6.1	6.7	—
RT-400	—	6.1	6.5	6.8	—
RT-600	9.5	6.1	—	7.1	5.6

Change in modulus of elasticity with test temperature is shown in Figure Q.2-54. As can be seen, at 300°F all alloys retain better than 90% of their room temperature modulus.

Low temperature creep strength data for the alloys are meager but what are available are shown in Table Q.2-11 and indicate that creep will pose no problem.

Prolonged exposure at 300°F will not affect the subsequent room temperature properties of any of the alloys. Aging data at 500° to 600°F for 500 hours are

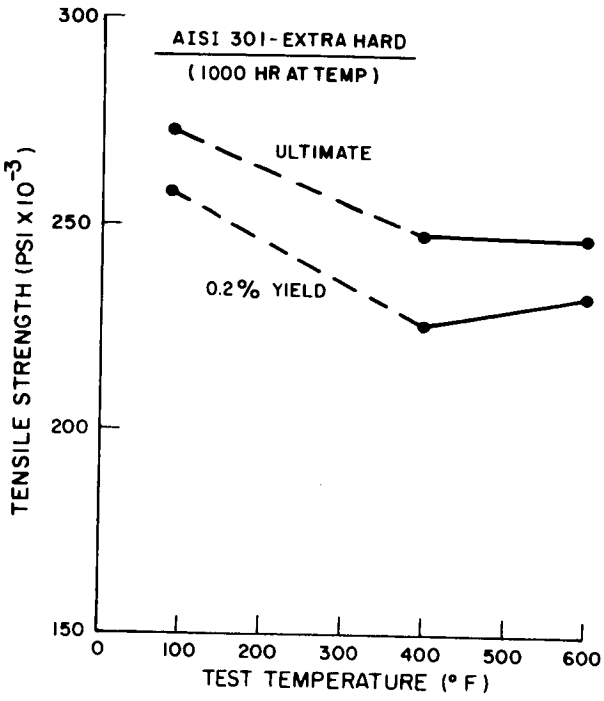


Figure Q.2-49. Tensile Strength of Cold Worked AISI 301 Steel at Elevated Temperatures

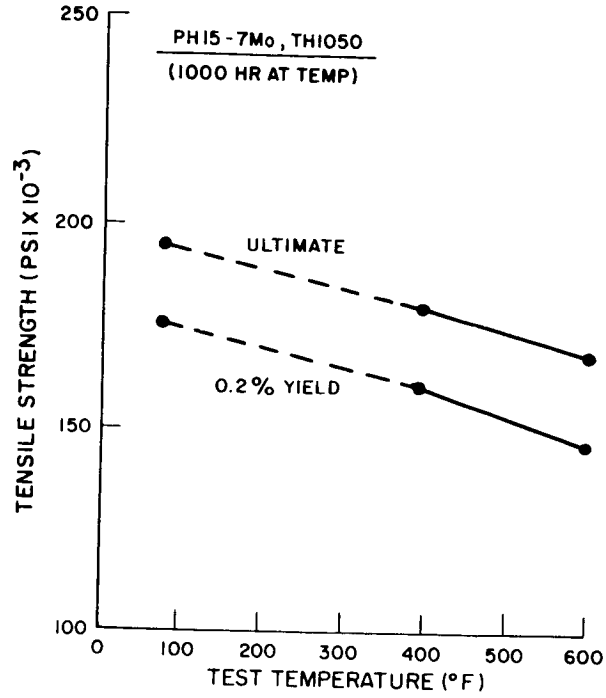


Figure Q.2-50. Tensile Strength of Cold Worked PH15-7Mo, TH1050 Steel at Elevated Temperatures

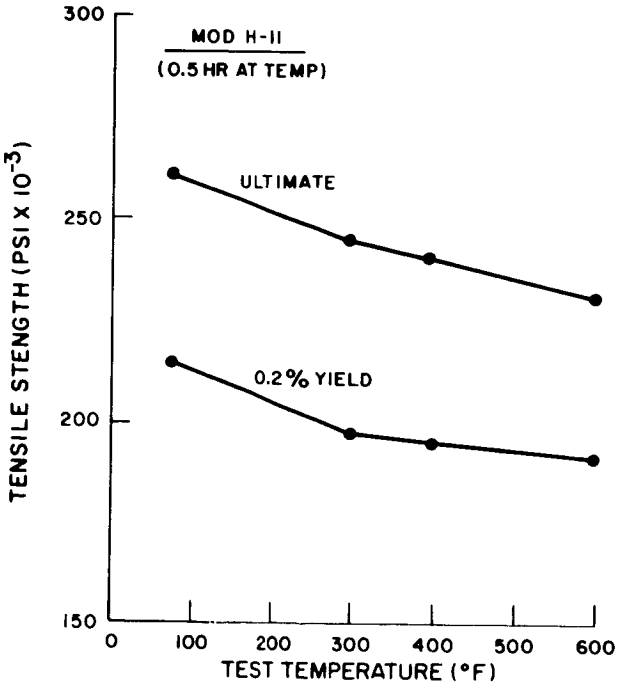


Figure Q.2-51. Tensile Strength of Cold Worked Mod. H-11 Steel at Elevated Temperatures

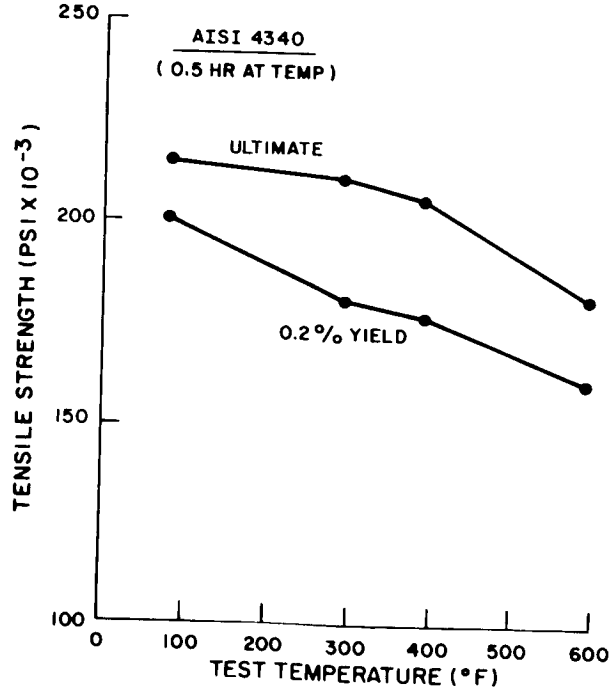


Figure Q.2-52. Tensile Strength of Cold Worked AISI 4340 Steel at Elevated Temperatures

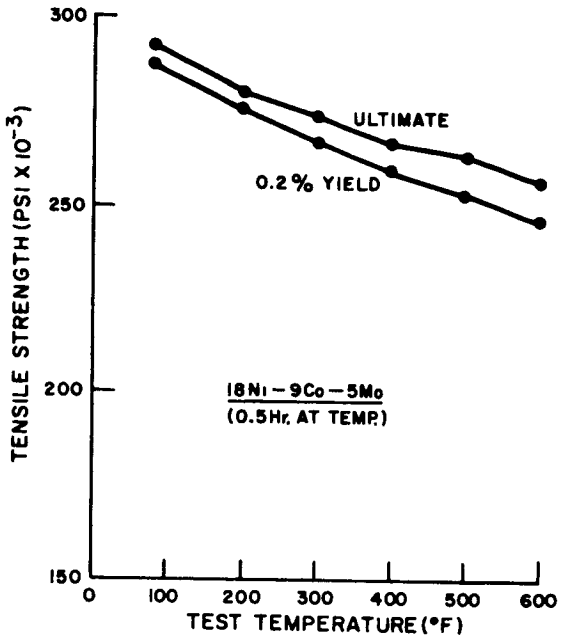


Figure Q.2-53. Tensile Strength of Cold Worked 18Ni-9Co-550 Steel at Elevated Temperatures

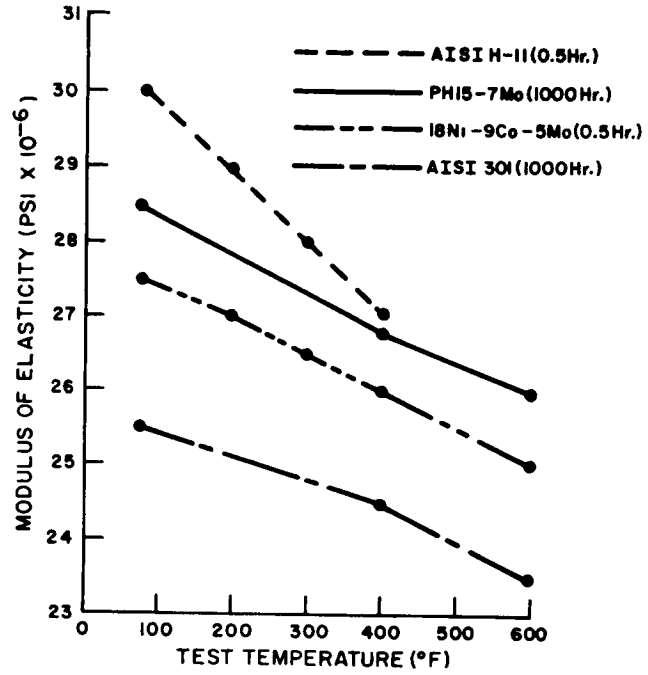


Figure Q.2-54. Modulus of Elasticity of Steel Alloys at Elevated Temperature

presented for PH15-7Mo in Table Q.2-12. No changes are observed following exposure at 500°F while strengthening occurred at 600° and 800°F. Metallurgical phase changes will be very slow at 300°F for PH15-7Mo.

The effect of temperature upon reaction rates can be appreciated by using the empirical tempering parameter devised by Holloman & Jaffe as shown below.

$$\text{Parameter} = \left[\text{Temp } (^{\circ}\text{F}) + 460\right]\left[20 + \log \text{time (hr)}\right]$$

Example:

Material = H-11 steel, tempered at 1050°F for 10 hr

Parameters: $\left[1050 + 460\right]\left[20 + 1\right] = 31,700$

Time at 300°F
(To produce equivalent : tempered properties)

$$\left[300 + 460\right]\left[20 + \log t\right] = 31,700$$

$$\log t = \frac{31,700}{760} - 20$$

$$\log t = 21.8$$

$$t = 10^{22} \text{ hours}$$

TABLE Q.2-11. CREEP STRENGTH OF STEEL ALLOYS

Temp (°F)	Time (Hr)	Creep Deformation (%)	Stress to Produce Given Deformation	
			PH15-7Mo	AISI H-11
600	1000	0.1	132,000	—
600	1000	0.2	150,000	—
600	1000	0.5	—	—
600	1000	1.0	—	210,000

TABLE Q.2-12. EFFECTS OF EXPOSURE AT ELEVATED TEMPERATURES UPON SUBSEQUENT ROOM TEMPERATURE PROPERTIES OF PH15-7Mo, TH1050

Exposure Conditions		Subsequent Room Temp. Properties	
Temp (°F)	Time (Hr)	0.2% Yield (psi)	Ultimate (psi)
RT	—	190,000	205,000
500	500	190,000	205,000
600	500	200,000	210,000
800	500	225,000	240,000

D. Suitability of Alloy Steels

All of the alloy steels evaluated are suitable for use in structures which must undergo thermal sterilization. The low alloy steels, AISI H-11 and AISI 4340, have an advantage over the stainless steels because of their higher thermal conductivity. AISI 4340, or slight modifications of this composition, have been used at higher tensile strength levels than those shown previously. This higher strength is obtained by tempering of the quenched alloy at a temperature of approximately 450°F. It is possible that an alloy so tempered would lose room temperature properties upon prolonged exposure at 300°F. Silicon modified 4340, which is tempered at 600°F, would be a potential candidate to replace 4340 at this higher strength level. The 18Ni-9Co-5Mo alloys appear promising for future use but additional testing will be required.

Q.3 ADHESIVES

Q.3.1 GENERAL

Adhesives will be required for the construction of interplanetary space vehicles. Some typical applications are:

a. Bonding of re-entry shield to supporting substructures.

b. Fabrication of honeycomb panels

c. Bonding of stiffening members

d. Bonding of solar cells

e. Bonding of miscellaneous attachments or instruments to the structure

Adhesives for planetary landers must be capable of withstanding the thermal sterilization treatment, i.e., they must maintain strength at 300°F for periods up to 500 to 600 hours and show no significant loss in subsequent room temperature properties. Like the metallic alloys discussed previously, the organic adhesives currently used for most space vehicles were not selected to meet these criteria, however, adhesives are available which can meet all the requirements posed by thermal sterilization. In addition to the high temperature organic adhesives, ceramic adhesives might prove advantageous in certain applications.

Q.3.2 ORGANIC ADHESIVES

A. General Characteristics

The organic resins used in adhesive formulation are usually divided into two groups - thermoplastic and thermosetting - based upon their behavior under the application of heat. Thermoplastic materials undergo reversible softening and melting with increasing temperatures. Thermosetting resins do not soften and meet but undergo irreversible changes as temperature rises. This decomposition takes place with the evolution of carbon and gaseous phases. The thermoplastic resin, consisting of long linear chains with no cross linking, normally are more flexible and have lower strength than the thermosetting resins which contain cross linking between chains.

Because of their softening and melting, the thermoplastic resins are limited in use to approximately 200°F. Some thermosetting resins, on the other hand, retain useful properties as high as 400°F. A listing of the commonly used resins of each category are shown in Table Q.3-1 with their approximate heat distortion temperatures. Fillers are often added to the basic resin to improve properties and resistance to temperature. Also, resins are blended to improve properties. Thermoplastics are added to thermosetting resins to reduce brittleness while thermosetting resins are added to thermoplastic resins to improve elevated temperature properties. Elastomers may be added to the thermosetting resins to increase flexibility.

TABLE Q.3-1. COMMONLY USED ORGANIC RESINS

Thermoplastic Resins		Thermosetting Resins	
Class	Heat Distortion Temp (°F)	Class	Heat Distortion Temp (°F)
Vinyl	200	Epoxy	450
Polyamide (Nylon)	300	Polyester	400
		Phenolic	300
		Silicone	400

Silicones differ from the normal organic resins in that their molecular chain structure consists of alternating silicon and oxygen atoms. This gives a more stable basic chain than that of a hydrocarbon polymer, so that silicones possess superior stability at elevated temperatures.

Thermal degradation of resins is caused by 1) scission of the molecular chains, 2) oxidation, and 3) hydrolysis. Figure Q.3-1, shows the results of twist tests on various resins and illustrates the significant difference in their ability to resist thermal degradation. The test results are not directly applicable to thermal sterilization, but demonstrate the wide range in elevated temperature behavior among the resin types.

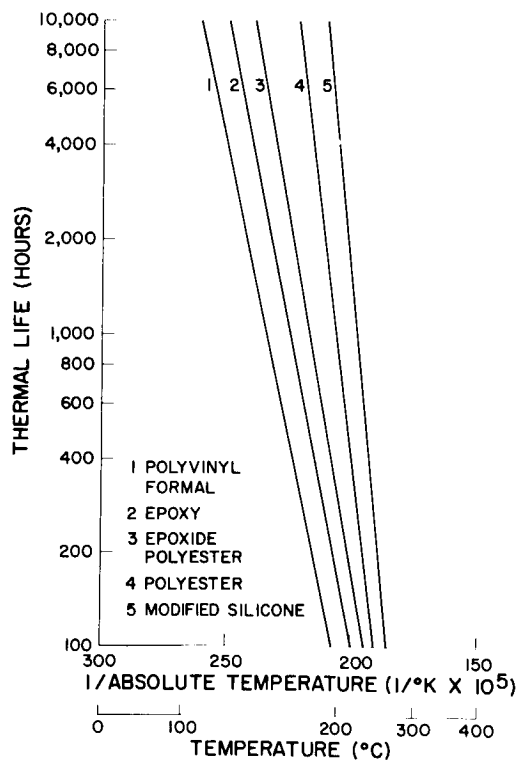


Figure Q.3-1. Comparative Life of Elevated Temperatures for Organic Resin Based Upon Twist Test

Sterilization performed in a chamber filled with a dry, inert gas will reduce thermal degradation caused by oxidation and hydrolysis. The magnitude of the damage contributed by these factors can be appreciated from the curves in Figures Q.3-2 and Q.3-3. Figure Q.3-2 shows the effects on subsequent room temperature elongation of polyamide resin (nylon) as a result of long term aging at 300°F in air compared to aging in nitrogen. Figure Q.3-3 compares room temperature shear strength of adhesive bonds aged at 600°F in air to that of bonds aged in dry nitrogen. Obviously, the inert gas used during thermal sterilization will reduce detrimental effects upon adhesives.

Structural adhesives are classified in military specification MIL-A-5090D in accordance with their maximum useful service temperature as shown below:

Type I - 192 hr. exposures to 180°F

Type II - 192 hr. exposures to 300°F

Type III - 192 hr. exposures to 300°F and 10 min. exposures to 500°F

Type IV - 192 hr. exposures to 500°F

Typical minimum required properties for each type are presented in Table Q.3-2. As the temperature requirements increase, the number of commercial adhesives qualified per QPL-5090-17 decreases rapidly. Only one adhesive has been qualified as meeting the Type IV requirements.

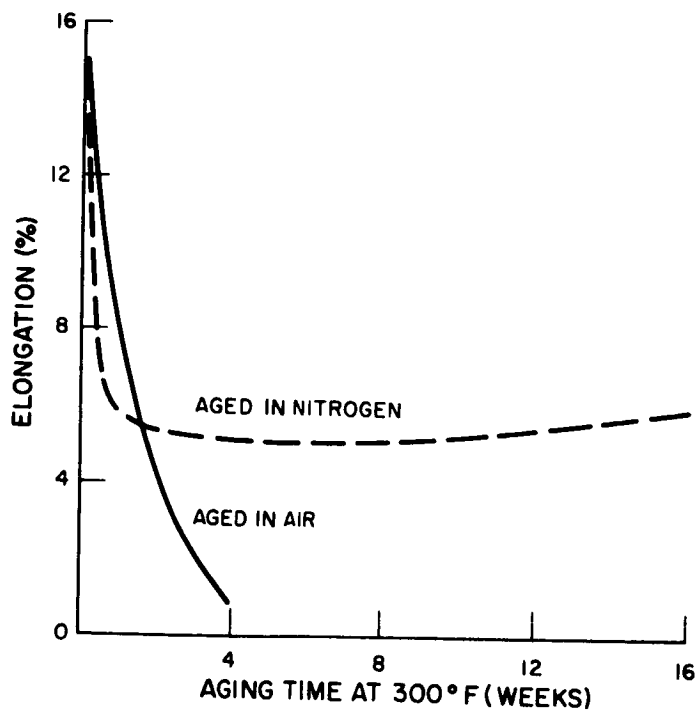


Figure Q.3-2. The Effect of Aging Atmosphere on the Subsequent Room Temperature Elongation of Polyamide Resin

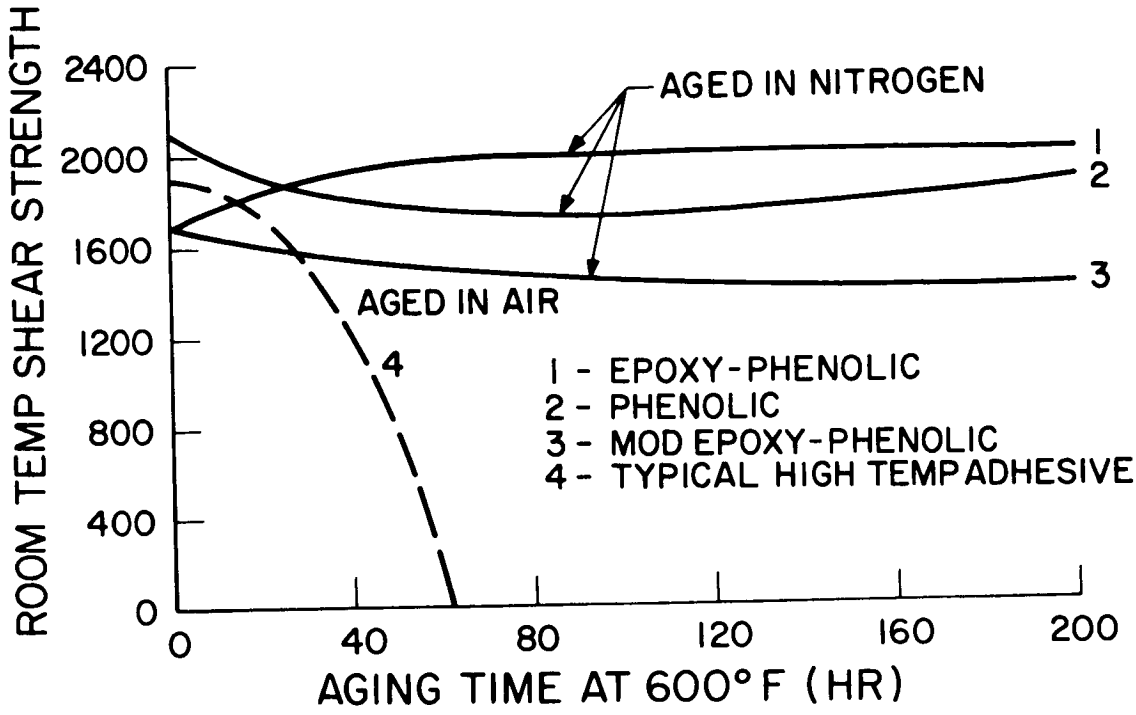


Figure Q.3-3. Aging Time at 600°F Versus Room Temperature, Shear Strength

TABLE Q.3-2. MECHANICAL PROPERTY REQUIREMENTS OF ADHESIVES PER MIL-A-5090D

a) Elevated Temperature Tensile Shear

Test Conditions		Minimum Avg. Strength Requirement (psi)			
Temp (°F)	Time (Hr)	Type I	Type II	Type III	Type IV
RT	-	2500	2500	2500	2500
180	192	1250	-	-	-
300	192	-	2000	2000	2000
500	10 min.	-	-	1850	1850
500	192	-	-	-	1000

b) Elevated Temperature Creep Deformation

Test Conditions			Maximum Avg. Deformation (Inches)			
Temp (°F)	Stress (psi)	Time (Hr.)	Type I	Type II	Type III	Type IV
180	800	192	.015	-	-	-
300	800	192	-	.015	.015	-
500	800	192	-	-	-	.015

The adhesives used on current spacecraft are predominantly of Type I, such as epoxy or epoxy modified with polysulfide or polyamide to improve flexibility. The polysulfide-epoxies would degrade seriously in 48-72 hours of exposure at 300°F. The polyamide-epoxies which are cured at 300°F, would be questionable for structural bonds requiring thermal sterilization.

Several Type III adhesives and the single Type IV appear more than adequate. These include phenolic-epoxies, nitrile-phenolics, epoxy-silicones, and polyamides. The properties of typical adhesives of these types are discussed in detail below.

B. Physical Properties

Thermal-physical properties of interest from the thermal sterilization standpoint are presented for several resins in Table Q.3-3. Such property data on specific adhesive formulations is not readily available. As can be seen, there is a wide range of values for a given property of a resin. In general, the thermosetting and thermoplastic types have similar specific heat and thermal conductivity. Thermal expansion of the thermosetting resins, however, is significantly less than that of the thermoplastic resins and approaches that of aluminum alloys. Thermal expansion is of importance for sterilization. The various combination of materials must be carefully matched to prevent bond failure during heating to 300°F. Also, the very low thermal conductivity of all the resins must be considered.

TABLE Q.3-3. THERMAL PROPERTIES OF COMMON RESINS

Resin	Type	Thermal Conductivity (Btu/hr/ft^2/°F/ft)	Coefficient of Thermal Exp. (in/in/°F x 10^6)	Specific Heat (Btu/lb/°F)
Polyvinyl Chloride	Thermoplastic	0.07-0.10	28-33	0.3 -0.5
Polyamide	Thermoplastic	0.10-0.15	45-70	0.4
Phenolic	Thermosetting	0.1 -0.4	8-25	0.3 -0.4
Polyester	Thermosetting	0.1	20	0.25
Epoxy	Thermosetting	0.1 -0.8	12-30	-
Silicone	Thermosetting	0.1	4-32	-
Typical Metal				
Aluminum Alloy	-	90	14	0.23

C. Mechanical Properties

Tensile lap shear strength for several typical Type I, low temperature adhesives are shown in Figure Q.3-4. Even in these short time tests, 10 minutes at temperature, it is evident that these adhesives are inadequate for service at temperatures of 300°F and above.

Figure Q.3-5 shows curves of tensile shear versus test temperature for several types of commercially available adhesives. These test data were obtained after a 192 hour holding period at test temperature. All on these adhesives retain significant strength to 500°F. The strongest of the adhesives, the nitrile-phenolic, was preconditioned by a 20-hour post cure at 500°F to improve its performance. The strengthening effect of this post cure is shown in Figure Q.3-6.

Limited creep resistance data are listed in Table Q.3-4 for three of the high temperature adhesives - the epoxy-phenolic, aluminum filled epoxy-phenolic, and nitrile-phenolic. All show less deformation at 300°F than the 0.015 in. maximum for MIL-A-5090D, Type III. The nitrile-phenolic and the epoxy-phenolic were both tested at 500°F and, again, deformation was significantly less than the 0.015 in. maximum for Type IV adhesives.

Resistance to aging at 300°F is shown in Figure Q.3-7 for the nitrile-phenolic, Type IV, adhesive. Tensile shear strength of cured (350°F) specimens are compared to those which were aged for 222 hours at 300°F following the cure. No change was affected in room temperature strength but an increase was exhibited in strength at elevated temperatures. This adhesive is normally post-cured at 500°F for optimum elevated temperature performance. Such a treatment should stabilize it against further change at 300°F within reasonable periods of time. In addition, the use of a dry, inert atmosphere during sterilization should limit thermal degradation as discussed previously.

D. Suitability of Adhesives

Adhesives are available which can tolerate the thermal sterilization process. The Type IV, nitrile-phenolic adhesive has been extensively used on Mach 2 aircraft operating at temperatures approaching 300°F. The use of adhesives must be undertaken with care and additional substantiating property data should be obtained.

A limiting factor in the structural application of adhesives is the elevated temperature post cure, or in the case of some newly developed adhesives, the initial curing temperature. For example, the high strength polybenzimidazole adhesive requires a pressure of 200 psi and a 600°F temperature for curing.

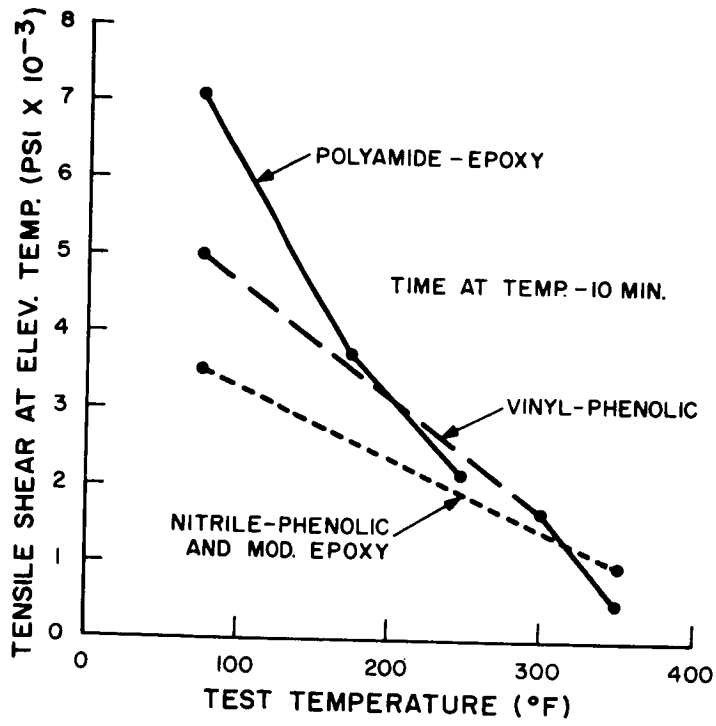


Figure Q.3-4. Effect of Temperature Upon Tensile Shear Strength of Typical Low Temperature, MIL-A-5090D, Type I Adhesives Exposed for 10 Minutes

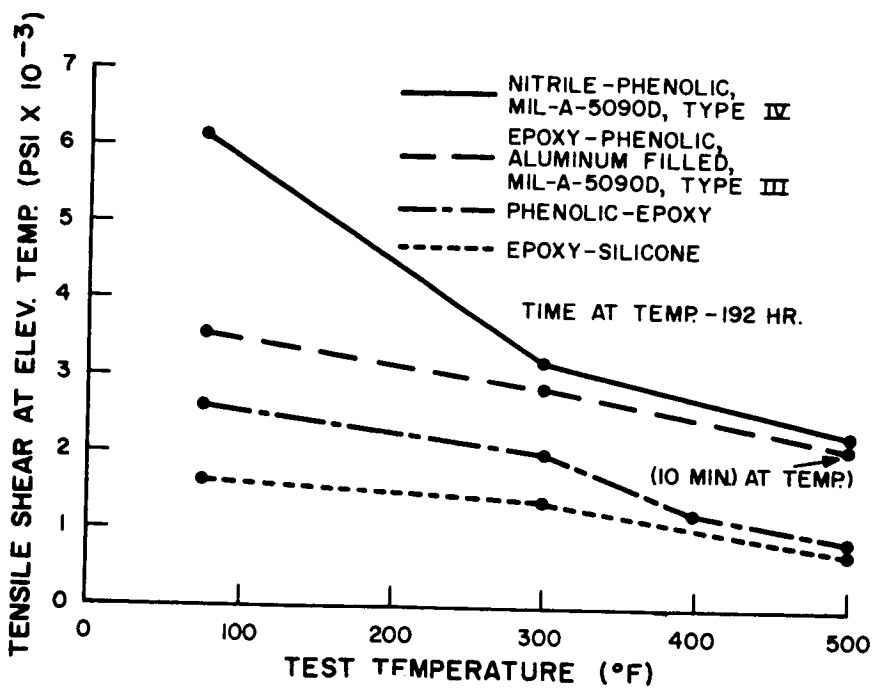


Figure Q.3-5. Effect of Test Temperature Upon Tensile Shear Strength Adhesives for Elevated Temperatures Except Where Noted All Tests Conducted Following 192 Hours at Temperature

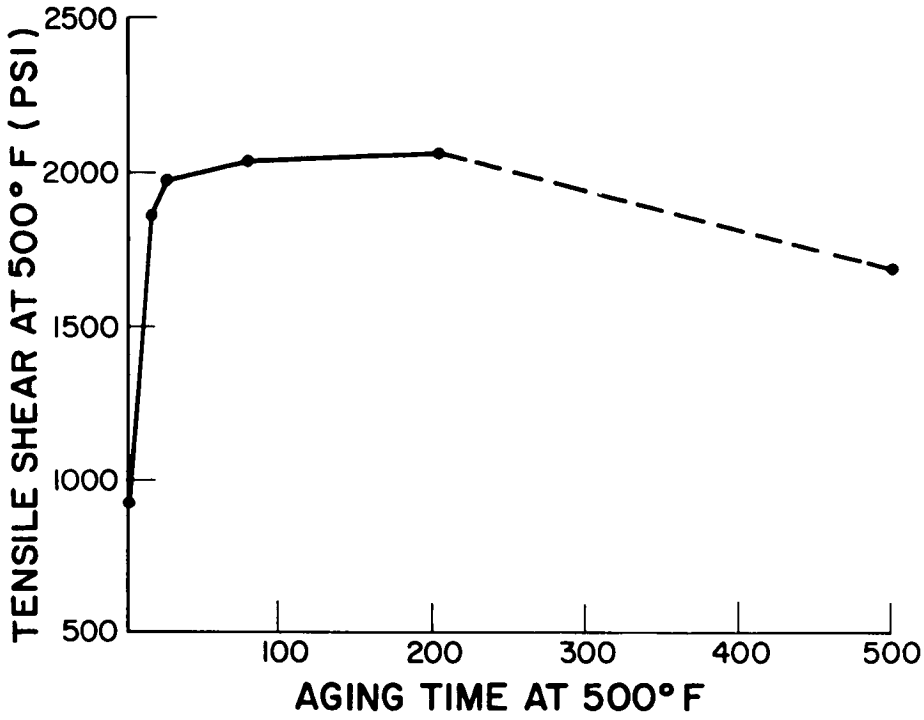


Figure Q.3-6. Effect of 500°F Post Cure Upon 500°F Short Time Tensile Shear of a Nitrile-Phenolic Adhesive, MIL-A-5090D, Type IV

TABLE Q.3-4. CREEP RESISTANCE OF ELEVATED TEMPERATURE ADHESIVES

Temp (°F)	Time (Hr)	Stress (psi)	Total Deformation (Inch)		
			Al Filled Epoxy-Phenolic	Nitrile-Phenolic	Epoxy-Phenolic
300	192	800	0.010	0.0000	0.0000
500	192	800	-	0.0017	0.0018

The nitrile-phenolic adhesive, while cured at a reasonable temperature of 350°F, requires a 500°F post cure for optimum properties. Such temperatures are excessive for the aluminum and magnesium alloys but will cause no problem with titanium and the stainless steels.

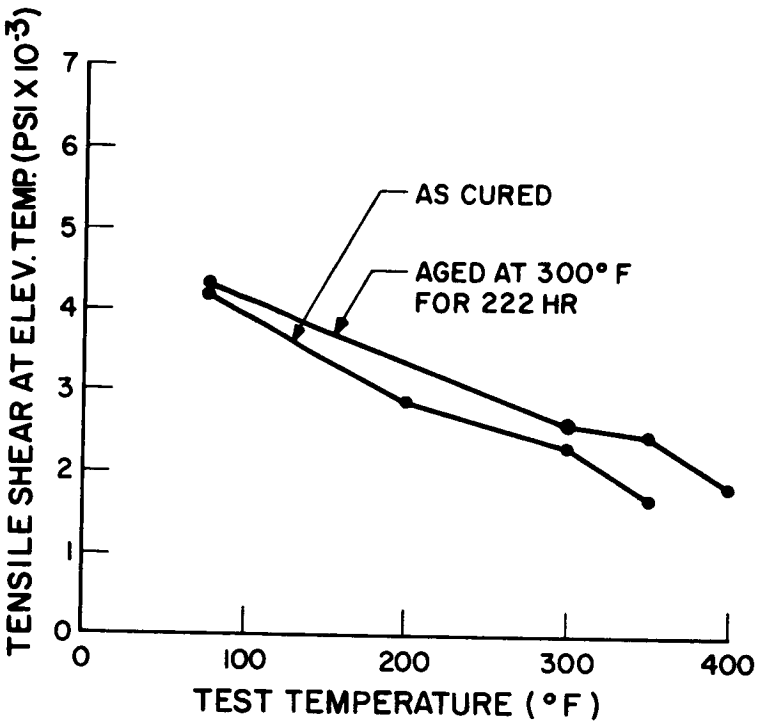


Figure Q.3-7. Effects of Aging at 300°F for 222 hours on the Short Time Elevation Temperature Properties of a Nitrile-Phenolic Adhesive, MIL-5090D, Type IV

Q.3.3 CERAMIC CEMENTS

Ceramic cements are being actively evaluated for application in aerospace structures for service at temperatures above the capabilities of present commercial or developmental organic resin based adhesives. A tentative goal of 1000 hours life under stress at 1000°F has been established. It does not appear that these cements will be required in the fabrication of space vehicles to withstand thermal sterilization since adequate organic adhesives are available. They may be used in specific spacecraft applications, however, where temperature is a problem. For this reason, a brief discussion of their properties and characteristics is included.

These ceramic cements can be divided into two general and rather broad categories based upon whether they are air curing at room temperature or whether they have to be fired at elevated temperatures. The air curing cements consist of mixtures of finely powdered ceramic oxide bonded with silicates of phosphates. The silicate bonded cements are hardened by loss of water of hydration. The phosphate bond is

produced by a chemical reaction between phosphoric acid and the oxides. The cements which require firing to harden are made up of pulverized glass (SiO_2) in a slurry with various oxides added to adjust properties and firing temperatures. Depending upon composition, they are fired between 1000 and 3000°F.

Typical physical properties are listed in Table Q.3-5. Limited tensile shear strength data at room and elevated temperatures are plotted in Figure Q.3-8.

The ceramic adhesives have the following distinct advantages over organic adhesives:

a. Higher strength at elevated temperature.

b. Greater resistance to degradation caused by aging in air.

c. Less susceptibility to radiation damage.

d. Higher thermal conductivity (still low compared to metals).

e. Lower coefficient of thermal expansion.

The major drawback of ceramic cements is their brittleness. Additional testing under flexure and vibration will be required before they can be used as structural materials.

TABLE Q.3-5. PROPERTIES OF TYPICAL CERAMIC CEMENT

Physical Properties	
Density	0.15 lb/in^3
Specific Heat	0.2 Btu/lb/°F
Thermal Conductivity	0.4 - 0.7 Btu/hr/ft^2/°F/ft
Coefficient of Thermal Expansion	2 - 5 x 10^{-6} in/in/°F
Mechanical Properties	
Tensile Ultimate	10,000 psi
Compressive Ultimate	100,000 psi
Modulus of Rupture	23,000 psi

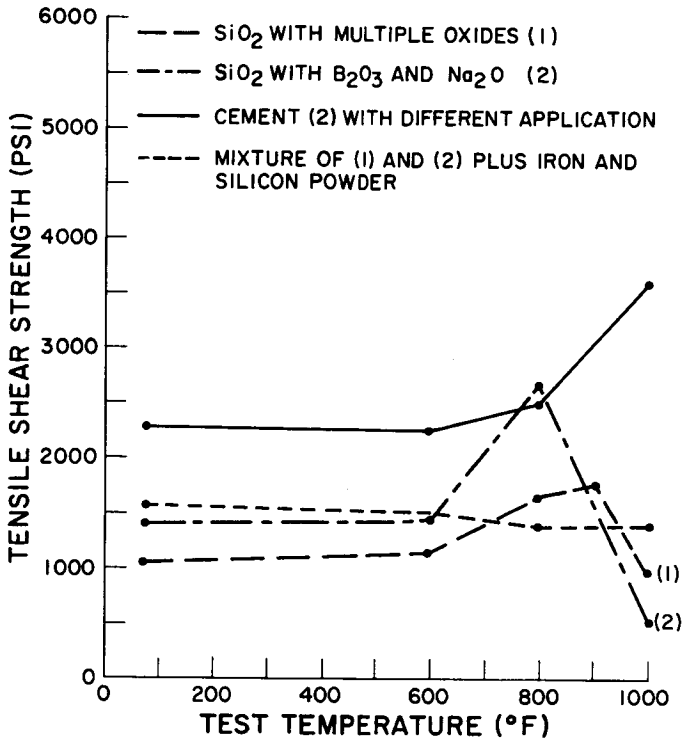


Figure Q.3-8. Short Time Elevated Temperature Tensile Shear Properties of Fixed Ceramic

Development work is in progress to improve ductility of ceramic type adhesives. One technique investigated is to form a sintered cermet (ceramic dispersed in a metal matrix) between the metal surfaces to be joined.

Sintering takes place below the melting point of the metal and produces a continuous metallic matrix with dispersed oxide particles. The resulting joints exhibit improved ductility. Another promising approach is the blending of ceramic powders with brazing alloys.

In general, it does not appear that ceramic cements are serious contenders to replace organic adhesives in space vehicles except for those such as the Solar Probe or Venus Lander, where temperatures approaching 1000°F may be expected. It appears that they should be able to withstand the 300°F thermal sterilization, but much more additional testing is required before they will replace brazing as a joining method for high temperature materials.

A multitude of fabrication problems exist in the production of honeycomb from titanium and stainless steel. One of the principal problems is brazing; so that maximum advantage can be made of the high temperature capabilities of the alloys. The use of stainless steel and titanium alloys has also added impetus to the development of high temperature adhesives such as were discussed previously.

In addition to brazing and adhesive bonding, a new process called roll welding has been developed for making titanium sandwich panels. This process consists of hot rolling an evacuated and sealed pack containing titanium corrugation with iron inserts filling the voids. After rolling, the iron is leached out leaving the diffusion welded titanium sandwich panel.

It is evident that the operational environment for titanium and steel sandwich materials is much more severe than that of thermal sterilization. Honeycomb which is fabricated from these alloys and is acceptable for the higher temperature service application will pose no problems for sterilization. If adhesive bonded panels do not prove satisfactory, brazed panels can be substituted. Thus, from the standpoint of sterilization, it is the aluminum honeycomb and reinforced plastic materials which require the most attention.

Q.4 COMPOSITE STRUCTURAL MATERIALS

Q.4.1 GENERAL

Since extensive use is made of composite materials in space vehicle structures, their ability to withstand the sterilization environment requires evaluation. Composites represent a broad spectrum of dissimilar materials intimately combined so as to take maximum advantage of the most desirable properties of each. For this discussion, however, specific types of composite materials have been selected which are most representative of those made for space-vehicle construction. These include honeycomb sandwich, with both metallic and non-metallic core, and reinforced plastics.

The effects of elevated temperatures upon organic resins are of prime importance in the behavior of these composites. This problem area was reviewed in the previous section on structural adhesives and, in general, the same information applies here. Particular emphasis in this section on composite materials will be on special problems inherent in the adhesive bonding of thin cores and face sheets.

Both metallic honeycomb sandwich and reinforced plastic honeycomb sandwich materials will be required for proposed planetary lander vehicles. The metallic honeycomb will be for structural purposes while the reinforced plastic honeycomb will serve as a crush-up to absorb the energy of impact. For the Mars lander, aluminum and glass-reinforced phenolic honeycomb have been suggested while the higher temperatures on Venus dictates the use of titanium or stainless-steel honeycomb.

Q.4.2 ALUMINUM HONEYCOMB SANDWICH

Aluminum honeycomb sandwich is composed of three materials — the core alloy, face sheet alloy, and organic adhesives. The face sheets may be any aluminum alloy desired, but are most ofter 2024-T3. This alloy is considered one of the best to withstand thermal sterilization based upon data presented in the section on metallic materials. The core material is usually 5052-H83 foil.

Tensile data showing the behavior of this alloy are presented in Figures Q.4-1 and Q.4-2. It apears that the maximum service temperature for this alloy is slightly over 300°F. Significant loss in room temperature strength as a result of long-term elevated temperature exposure does not occur until 400°F. This alloy is strain hardened by cold working and softens by recrystallization above this temperature.

Adhesives for bonding of metallic sandwiches are classified into four types in MIL-A-25463 according to the same maximum service temperature as in the specification on structural adhesives (MIL-A-5090D) discussed previously. Most honeycomb for space vehicles is bonded with Type I adhesives which are limited to 180°F. This adhesive normally would not be satisfactory for thermal sterilization.

High temperature adhesives are available, however, which should provide satisfactory performance. Examples of the type of properties obtained and comparisons with the specification requirements are presented in Table Q.4-1. No definite conclusions can be drawn from this sketchy data.

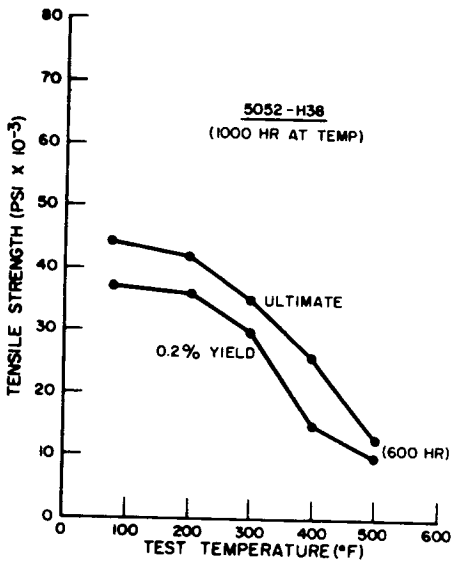


Figure Q.4-1. Elevated Temperature Tensile Properties of 5052-H38 after 1000 hour Exposure at Test Temperature

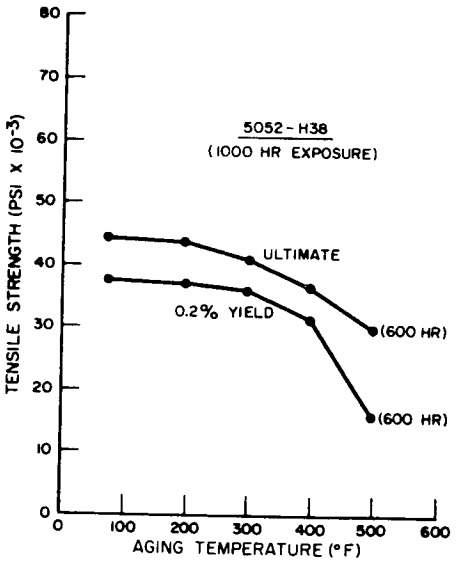


Figure Q.4-2. Effect of 1000 hour Exposure at Elevated Temperature upon Room Temperature Tensile Properties of 5052-H38

TABLE Q.4-1. MECHANICAL PROPERTIES OF TYPICAL ELEVATED TEMPERATURE SANDWICH ADHESIVES, MIL-A-25463, TYPES* II, III, AND IV

Type Adhesive	Temp. (°F)	Time	Property Values	
			Avg. of Test	Spec. Requirement*
Sandwich Peel Strength (in-lb/in width)				
(1) Nitrile-phenolic and mod. epoxy composite	RT	-	30	8.5 minimum
	180	10 min.	15	5.0 minimum
(2) Phenolic-epoxy	RT	-	7	8.5 minimum
	180	10 min.	-	5.0 minimum
(3) Al filled phenolic-epoxy	RT	-	9-11	8.5 minimum
	180	10 min.	-	5.0 minimum
(4) Mod. phenolic	RT	-	4	8.5 minimum
	180	10 min.	-	5.0 minimum
Flatwise Tensile Strength (psi)				
(1) See above	RT	-	1000	450 minimum
	300	10 min.	-	350 minimum
	500	10 min.	-	270 minimum
(3) See above	RT	-	617	450 minimum
	300	10 min.	-	350 minimum
	500	10 min.	290	270 minimum
(4) See above	RT	-	330	450 minimum
	300	10 min.	-	350 minimum
	500	10 min.	320	270 minimum
Flexure Strength (total load in lb)				
(3) Al filled phenolic-epoxy	RT	-	2180	1750
	500	10 min.	1380	1200
Creep Deflection (inches)				
(3) See above	RT	192 hr.	0.000	0.025 max (1000 lb load)
	500	192 hr.	0.008	0.050 max (500 lb load)

*Notes Type I - Long term exposure to 180°F
Type II - Long term exposure to 300°F
Type III - Long term exposure to 300°F + short term to 500°F
Type IV - Long term exposure to 500°F

The aluminum filled phenolic-epoxy looks promising and has been certified to MIL-A-25463 Type II requirements.

Limited data on the effects of aging for 300 hours at 300°F upon subsequent room temperature peel strength are shown in Table Q.4-2. These data indicate a significant loss in this property for both of the adhesives. Aging in nitrogen would probably result in less degradation.

More high temperature data are required on adhesive bonded honeycomb to arrive at a definite conclusion as to its suitability for thermal environment. Although high temperature structural adhesives are available which would be satisfactory, they may not necessarily be suitable for aluminum honeycomb because of their flow and filleting characteristics. The epoxy-phenolic system, developed especially for elevated temperature service, appears to be the most promising.

Q.4.3 REINFORCED PLASTIC STRUCTURAL MATERIALS

It is expected that reinforced plastic will be used on space vehicles that require sterilization. The resin impregnated glass fabric honeycomb crush-up material proposed for the Mars lander was citedpreviously. Similar honeycomb is also being used as a supporting matrix for elastomeric ablation shield materials now undergoing development for planetary re-entry protection. This application will be described in more detail in a later section. In addition, reinforced, laminated plastics can, and probably will, be utilized in many other specific structural applications. Their insulating qualities, however, may prove to be a serious limitation.

TABLE Q.4-2. EFFECTS OF AGING AT 300°F UPON ROOM TEMPERATURE PEEL STRENGTH

Exposure Conditions			
Adhesive	Temp. (°F)	Time (hr)	Peel Strength (in.-lb/in.)
Nitrile-phenolic and mod. epoxy composite	RT 300	- 300	23 - 33 18
Al filled phenolic-epoxy	RT 300	- 300	9 - 11 7

For elevated temperature applications, honeycomb sandwiches fabricated from glass cloth impregnated with phenolic or silicone resins are usually selected. Facing sheets may be laminated reinforced resins or metal. The advantages of these specific types of resins were previously pointed out in the section on adhesives. Both types have service temperature ceilings above 300°F and should be capable of withstanding the thermal sterilization treatment.

As an illustration of the suitability of reinforced plastic honeycomb core, the effect of elevated temperature upon the plate shear strength of phenolic-glass fabric honeycomb core is compared to that on 5052-H38 aluminum alloy honeycomb core in Figure Q.4-3. These test data based upon 30 minute soak at temperature, show that the reinforced plastic core retains a higher percentage of its room temperature shear strength at 300°F than does the aluminum alloy. The effect of prolonged exposure at 300°F and 400°F upon compression strength of phenolic-glass honeycomb is shown in Figure Q.4-4. At 300°F, no change in compression strength occurs up to the maximum soaking period of 192 hours.

It appears that honeycomb fabricated from phenolic impregnated glass fabric will be satisfactory for space vehicles requiring sterilization. Although no data are presented for silicone resin-glass fabric honeycomb panels, they too should be acceptable since the silicone resins rank among the best at elevated temperatures.

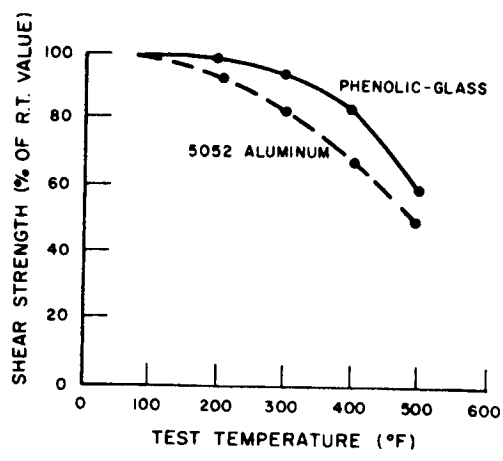


Figure Q.4-3. Short Time Elevated Temperature Shear Strength of Phenolic Glass Fabric Honeycomb Core Versus 5052-H38 Honeycomb Core (30 Min. at Temperature)

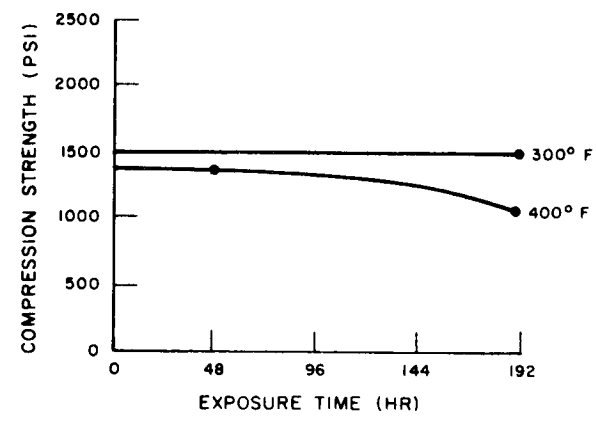


Figure Q.4-4. Effects of Exposure' Time at Elevated Temperature upon Compressive Strength of Phenolic Glass Fabric Honeycomb

One important factor that will affect heat transfer is the insulating qualities of these resin core structural materials. Typical thermal conductivities of 0.07-0.08 Btu/hr/ft^2/^{o}F/ft are obtained for phenolic-glass honeycomb panels. This is considerably lower than the 0.1-0.3 values for the resins themselves and the 5 - 100 values for structural metallic alloys and would complicate the heating and cooling problems.

In addition to reinforced plastic honeycomb, structural components may be made up of laminations or layers of glass fiber and resins. The glass fiber may be woven into cloth or wound as thread around a mandrel to make complex shapes. The latter way is used to fabricate reinforced plastic water cases for solid propellant rockets.

For moderate temperature service, epoxy polyester resins are commonly used. For elevated temperature service, specific heat resistant resins such as phenolics, epoxy-phenolics, or silicones have been developed. Typical physical properties for phenolic-glass laminates are silicon-glass laminates presented in Table Q.4-3.

Data showing the elevated temperature performance of several types of resin impregnated glass laminates are shown in Figures Q.4-5 and Q.4-6. In Figure Q.4-5, ultimate tensile strength after a 1000-hour soak is plotted against test temperature. Figure Q.4-5 illustrates the effect of test temperature upon flexure strength of similar laminates after 1000 hours exposure.

TABLE Q.4-3. PHYSICAL PROPERTIES OF GLASS REINFORCED RESIN LAMINATES

Type	Coefficient of Thermal Expansion (in/in/^{o}F)	Thermal Conductivity (Btu/hr/ft^2/^{o}F/ft)	Sp. Heat (Btu/lb/^{o}F)
Phenolic-glass	18×10^{-6}	0.15	0.3
Silicone-glass	5×10^{-5}	0.06	0.25

Of the commonly used types, the phenolic-glass laminates appear best at 300°F. In fact, significant loss of strength does not occur until over 400°F. The glass cloth reinforced phenyl-silicone actually has greater strength and higher temperature capability than the phenolic-glass laminate, but this is a developmental-type product.

The epoxy-phenolic resin blend appears to be marginal for use at 300°F since flexure strength drops off catastrophically between 300 and 400°F. The reinforced silicones have somewhat low strength but retain it to over 500°F. They should be suitable candidates for thermal sterilization.

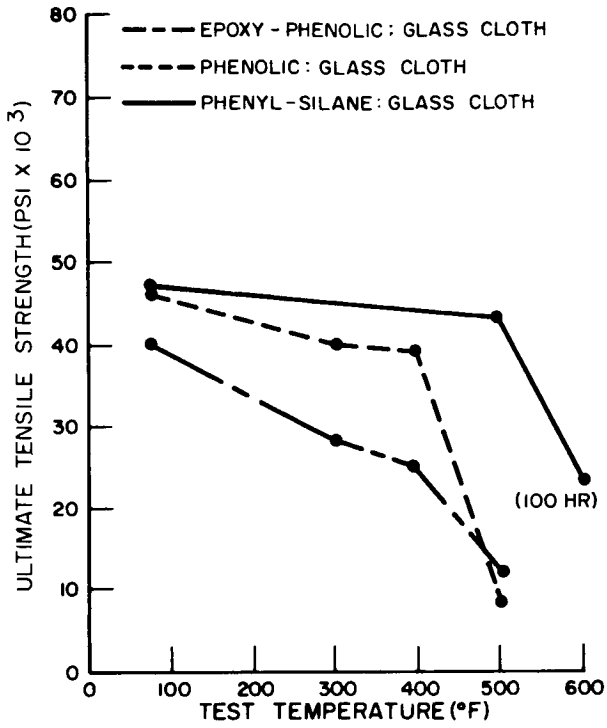


Figure Q.4-5. Elevated Temperature Tensile Ultimate Strength of Resin-Glass Fabric Laminates After 1000 Hours at Test Temperature

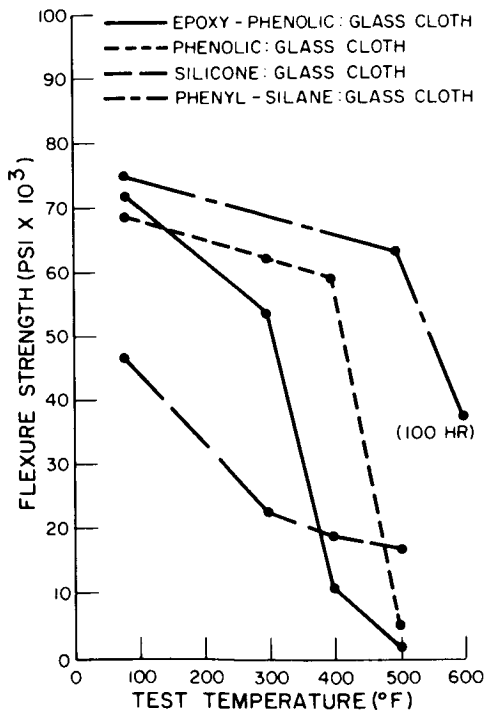


Figure Q.4-6. Elevated Temperature Flexure Strength of Resin-Glass Fabric Laminates after 1000 Hours at Test Temperature

Another newly developed resin system is the polyamide-polyimide blend. Flexure strength data from glass cloth laminates made with this resin are compared at 500°F with silicones and phenolics in Figure Q.4-7. It is evident that there are several potential reinforced plastic materials for use at elevated temperatures.

Data showing the effects of long term aging at elevated temperature upon subsequent room temperature properties of laminates are meager. The information presented in the previous section on organic adhesives would be applicable. In addition, the elimination of oxidation by using the dry, inert atmosphere during sterilization would aid in retarding degradation.

In conclusion, reinforced plastics will be acceptable for vehicle structures requiring thermal sterilization. High temperature resins must be used, however, and additional evaluation may be required on effects of heat aging upon subsequent room temperature properties.

Q.5 THERMAL SHIELD MATERIALS

Q.5.1 GENERAL

An ablative thermal shielding material will undoubtedly be required for planetary landers to protect them from overheating during entry through the planet atmosphere. Thermal shield materials and the adhesives for bonding the shield to the nose cone substructure must be critically examined for their ability to withstand the 300°F thermal sterilization treatment since they will of necessity be an integral part of the structure.

Although designed to withstand the heat generated during re-entry into the earth's atmosphere, it is by no means certain that the materials currently used will undergo exposures of 500-600 hours total time at 300°F without serious degradation. During a typical re-entry, the time at temperature is comparatively short and the insulation properties of the shielding material maintain the interface at 150 to 200°F.

Thermal shield materials have been selected for Mars and Venus landers as part of the recently completed Voyager design study program conducted by the General Electric Company. Phenolic-nylon was specified for the Venus lander while a newly developed elastomeric material, designated ESM, was selected for the Mars lander. These materials are discussed in the following section. Property data applicable to the thermal sterilization problem is limited for both materials. What data are available are presented along with a discussion of the problem areas.

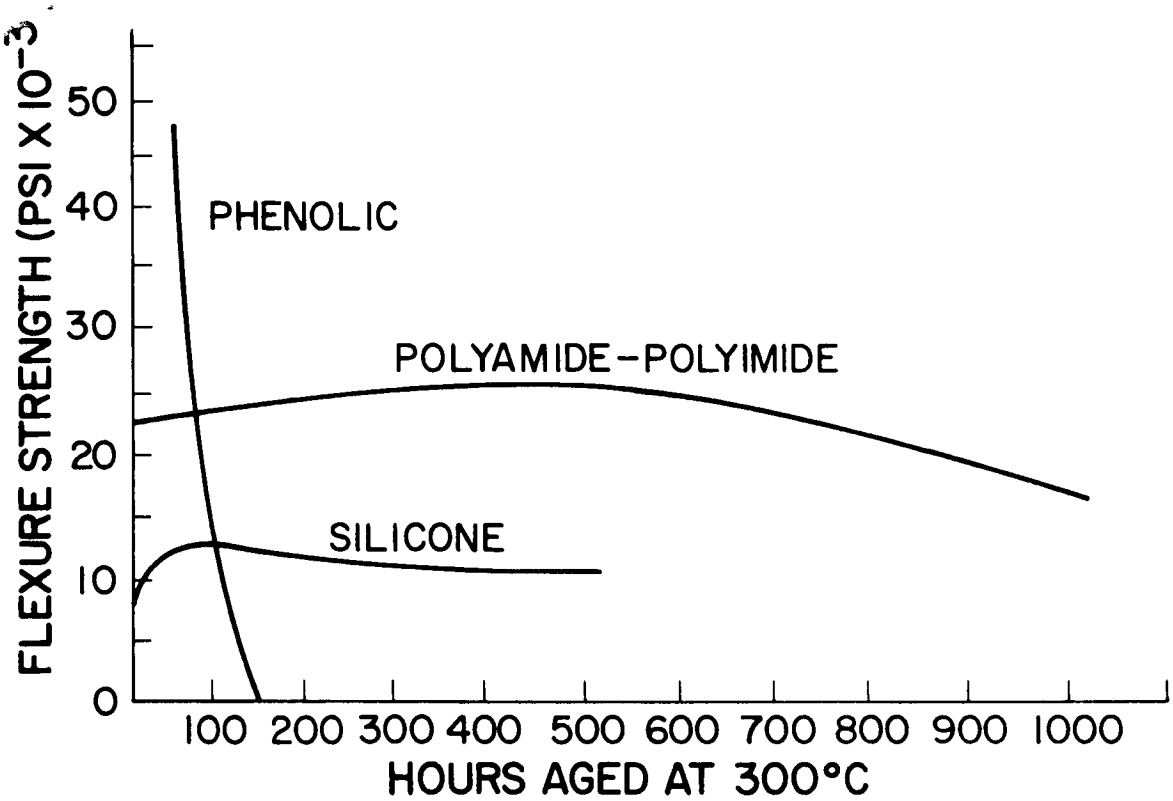


Figure Q.4-7. Effect of Exposure Time Upon Flexure Strength of Resin-Glass Laminates at 575°F

Q.5.2 PHENOLIC NYLON

Phenolic impregnated nylon has been extensively used for thermal shields for ballistic missiles and recoverable earth orbiting vehicles. The forward nose cap is normally press molded from small squares of preimpregnated nylon fabric while the aft portion of the shield is vacuum-bag molded from wound impregnated nylon tape. The curing cycle consists of 15 hours at 250°F.

Typical room temperature physical and mechanical properties for phenolic-nylon are presented in Table Q.5-1. Of significance for sterilization is the rather high coefficient of thermal expansion and the low thermal conductivity.

The thermal expansion for this material is approximately 5 to 10 times that of the steel or titanium substructure proposed for the Venus lander. Phenolic nylon is also being considered as a back-up to the ESM material for the Mars lander. The match in thermal expansion is better for the aluminum and magnesium alloy structure for this vehicle, but still the expansion of the phenolic-nylon will be from 2 to 3 times as great as that of the metallic alloys.

TABLE Q.5-1. ROOM TEMPERATURE PROPERTIES OF PHENOLIC IMPREGNATED NYLON

a) Physical Properties

Density (lb/ft^3)	73
Specific Heat ($Btu/lb/^oF$)	0.4
Thermal Conductivity ($Btu/hr/ft^2/^oF/ft$)	0.14
Coefficient of Thermal Expansion ($in/in/^oF$)	$30 \text{ to } 50 \times 10^{-6}$

b) Mechanical Properties

Tensile Ultimate (psi)	5000
Tensile Elongation (%)	1.0
Modulus of Elasticity (psi)	0.6×10^6

In either case, the shield material will be in compression during the thermal sterilization cycle. Past experience has shown that even with an aluminum structure, a flexible bond must be used to compensate for the differences in expansion. A hard bond, such as an epoxy, results in failure of the phenolic-nylon.

Silicone adhesives have proved suitable for bonding phenolic-nylon thermal shields to aluminum. Neoprene rubber adhesives have also been satisfactory. It is questionable whether the neoprene adhesives will withstand the 300°F exposure since this is considered to be their maximum temperature capability. Silicone rubber adhesives, with maximum use temperatures of approximately 600°F should be suitable for thermal sterilization.

It is uncertain whether or not either adhesive will be suitable for bonding phenolic-nylon to titanium or steel alloys.

Mechanical test data showing tensile ultimate versus short time at temperature for phenolic-nylon are plotted in Figure Q.5-1. At 300°F, tensile strength drops to 3000 psi, about one half of the room temperature value. Strength falls off significantly between 300 and 400°F. Compression strength data would be more pertinent for thermal sterilization, but none was available.

Data showing changes in room temperature properties of phenolic-nylon as a result of long term aging at 300°F are unavailable. Loss of room temperature tensile strength of nylon, itself, as a result of exposure at 285°F is shown in Figure Q.5-2. These aging tests were performed in air so oxidation caused a considerable

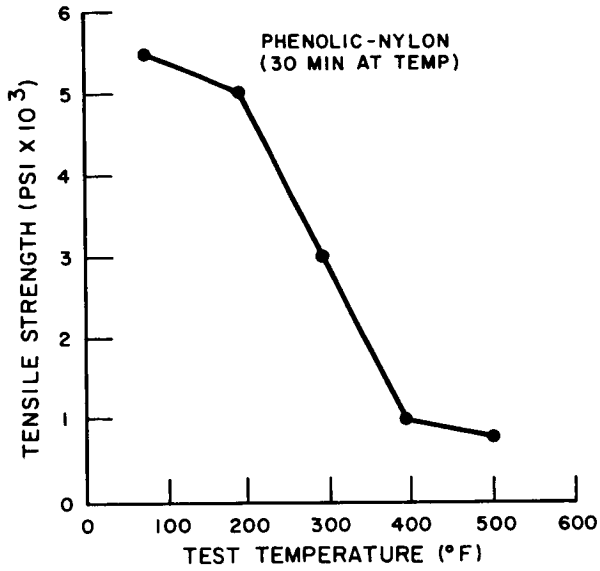


Figure Q.5-1. Effect of Elevated Temperature Upon Ultimate Tensile Strength of Phenolic-nylon, 30 Minutes at Test Temperature

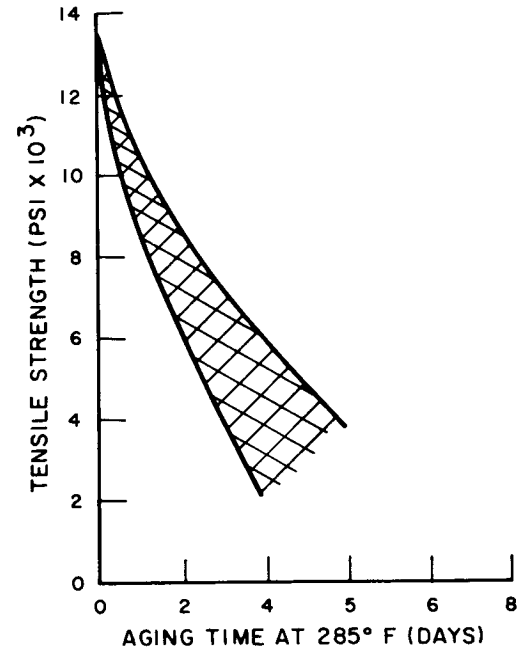


Figure Q.5-2. Effect of Aging at 285°F in Air Upon Subsequent Room Temperature Ultimate Tensile Strength of Nylon Resin

portion of the loss. This can be appreciated by referring to Figure Q.3-2 in the section on adhesives. This curve compares the effects on room temperature ductility of aging in air versus aging in dry nitrogen. The loss in ductility is considerably less for aging in nitrogen and levels off to a constant value. It can be assumed that aging phenolic-nylon laminates in nitrogen will also result in less severe degradation.

From the data available, it cannot be definitely concluded that phenolic-nylon will be satisfactory as a thermal shield on a space vehicle which requires thermal sterilization. In particular, additional data are required on the shield to structure band and long term elevated temperature mechanical property data and aging effects on the phenolic-nylon.

Q.5.3 ELASTOMERIC SHIELD MATERIAL (ESM)

For low heat flux, long time entry conditions, such as for the Mars lander, a special elastomeric material is being developed by the General Electric Company. This material consists of a silicone rubber foamed at mildly elevated temperatures to control density between 20 and 80 pounds per cubic foot. Special additives and fillers have been added to increase the strength of the char layer. The formulation is foamed in place in split phenolic-glass honeycomb whose purpose is also for support of the char layer.

Typical room temperature physical and mechanical properties are shown in Table Q.5-2. Thermal conductivity is significantly lower than for phenolic-nylon and thermal expansion is much greater. The ESM, itself, is being evaluated as a flexible bonding material to absorb the stresses because of differential thermal expansion between the shield and the structure.

Compressive strengths versus short time exposure at elevated temperatures are plotted in Figure Q.5-3. Similar test specimens aged in vacuum at 250°F for 14 days had the same room temperature compressive strength as unaged samples.

TABLE Q.5-2. TYPICAL ROOM TEMPERATURE PROPERTIES OF ESM 1001 P-S, ELASTOMERIC THERMAL SHIELD MATERIAL

a) Physical Properties

Density (lb/ft^3)	40
Specific Heat ($Btu/lb/^oF$)	0.30
Thermal Conductivity ($Btu/hr/ft^2/^oF/ft$)	0.09
Coefficient of Thermal Expansion ($in/in/^oF$)	100×10^{-6}

b) Mechanical Properties

Tensile Ultimate (psi)	58
Tensile Elongation (%)	60
Modulus of Elasticity (psi)	—

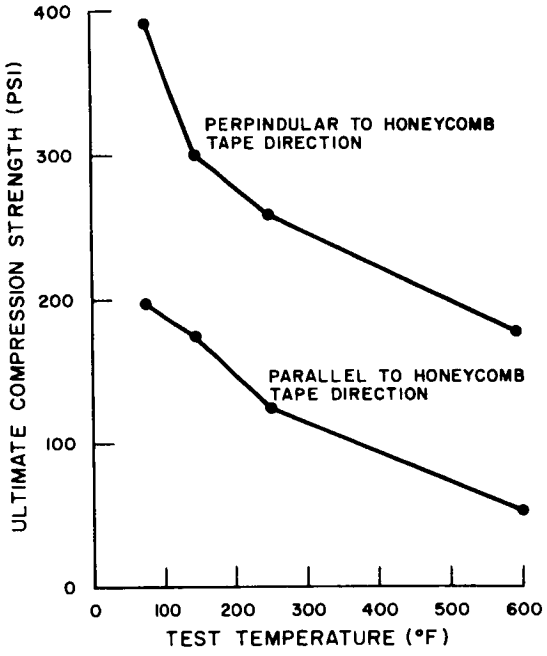


Figure Q.5-3. Short Time Elevated Temperature Ultimate Compressive Strength of ESM 1001; 30 Minutes at Temperature

The elastomeric shield materials show promise but will require additional evaluation. Because they are based upon the silicone resins which show excellent resistance to elevated temperature exposure, it seems a good assumption that they will be able to withstand the thermal sterilization treatment. Limited short time tests of the ESM materials bonded to aluminum substructures have demonstrated their ability to withstand the stresses induced by heating to 300°F.

Q.6 MISCELLANEOUS MATERIALS

Q.6.1 GENERAL

There are numerous materials which are not structural materials in the strictest sense of the word, but are so intimately tied in with the structure that it was felt they should be included in this report. Such items as corrosion protective finishes, thermal control coatings; thermal insulation; and conductive joint cements fall into this category. The potential effects of thermal sterilization of spacecraft upon materials normally used for these functions are presented in limited detail below.

Q.6.2 THERMAL INSULATION

Maintenance of a reasonably uniform interior temperature with space vehicles is a complex problem. Heat input is non-uniform because of on-off operation of electronic components and uneven heating of the vehicle by absorptance of solar infrared radiation. Normally, space vehicles must be well insulated to maintain control over the temperature.

Several types of insulation may be used, depending upon the specific application. In the vacuum of space, multi-barrier radiation insulation is extremely efficient because a large percentage of heat transfer is by radiation. Closed cell organic foams, on the other hand, are more suitable for insulating specific items

from the structure where heat transfer is by conduction. The ability of the organic foams to withstand thermal sterilization depends upon the resin used. As discussed in previous sections, there are many suitable organic resins. The use of such insulation will naturally create sizeable heat transfer problems during sterilization.

Multi-barrier, radiation insulation, on the other hand, may create a different problem. A typical insulation blanket consists of several layers of embossed, aluminized Mylar film. This material has a very low emittance. Thermal conductivity through such a blanket in vacuum is 0.0001 Btu/hr/ft^2/^{o}F/ft. In a gaseous atmosphere where conduction can occur its insulating qualities are not good. This can be seen in Figure Q.6-1 which compares heating rates of the interior and the exterior of a 30 layer, stitched blanket which was placed into an air furnace preset at 300°F. This material should not present serious heat transfer problems during thermal sterilization.

These multilayer blankets have been found to shrink significantly when heated to 300°F. The shrinkage is of a sufficient magnitude to warp space vehicle structures. The problem can be alleviated by preshrinkage of the insulation as shown by data in Table Q.6-1. Subsequent exposure in this case was 270°F instead of 300°F but the principle is demonstrated.

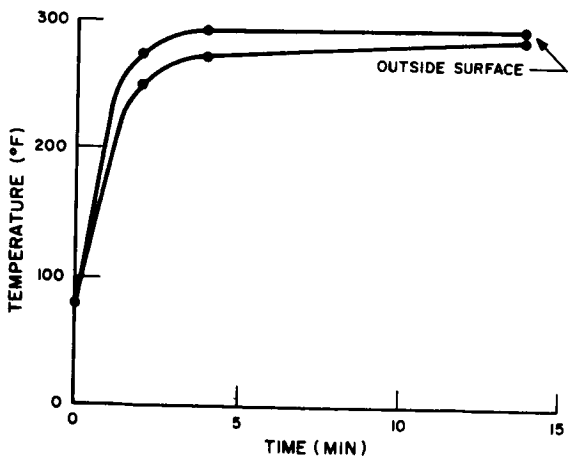


Figure Q.6-1. Rate of Temperature Rise Within a Multi-Layer Embossed, Aluminized Mylar Insulation Blanket

Preshrinkage of actual multi-layer blanket assemblies was performed in dry, purified nitrogen to prevent oxidation of the aluminized layer with a subsequent increase in emittance. Data are presented in Table Q.6-2 which show the advantage of dry nitrogen atmosphere.

This table illustrates a technique potentially useful for other materials requiring thermal sterilization. Many organic materials will undergo permanent dimensional changes as a result of elevated temperature and such a pretreatment for dimensional stability would prove necessary.

Q.6.3 COATINGS

Coatings are used on space vehicles for two purposes — corrosion protection and control of thermal radiative properties. For corrosion protection of aluminum and magnesium alloys, chemical conversion coatings or

TABLE Q.6-1. EFFECT OF PRESHRINKAGE UPON DIMENSIONAL STABILITY OF 1/4 MIL EMBOSSED, ALUMINIZED MYLAR

Sample	Preshrinkage Time @ 300°F (hr)	Dimensional Change After Preshrinking (%)		Dimensional Change After Additional 80 hr @ 270°F (%)	
		Long.	Trans.	Long.	Trans.
1A	1	-5.8	+2.2	-0.4	0.0
1B	1	-6.3	+1.8	-0.3	-0.1
2A	2	-5.9	+2.0	-0.3	-0.2
2B	2	-6.3	+2.2	+0.1	0.0
3A	3	-6.3	+2.4	-0.1	0.0
3B	3	-5.9	+2.0	-0.4	-0.5
4A	4	-6.0	+1.8	-0.1	0.0
4B	4	-6.2	+2.2	0.0	0.0

TABLE Q.6-2. EFFECT OF ATMOSPHERE UPON THERMAL RADIATIVE PROPERTIES OF PRESHRUNK EMBOSSED, ALUMINIZED MYLAR INSULATION BLANKETS

Shrinkage Atmosphere*	Emittance (ϵ_{TN})	Solar Absorptance (α_s)
As rec'd	0.14	0.17
Air	0.23	0.25
Dry, purified N_2	0.08	0.18
Dry, purified N_2	0.09	0.19

*Preshrinkage for 4 hours @ 300°F

anodically applied coatings are commonly used. It is felt that these types of coatings will not be a problem because of thermal sterilization requirements. Zinc chromate primer which is extensively used will also withstand 300°F.

Typical thermal control coatings used are listed in Table Q.6-3. It is felt that these coatings will withstand long term exposure to 300°F without subsequent changes in room temperature emittance and solar absorptance, but no substantiating data are available. Tests on these coatings are advisable.

TABLE Q.6-3. TYPICAL SPACECRAFT THERMAL CONTROL COATING

Coating		Characteristics
Pigment	Resin	
Aluminum	Silicone-allyd	low emittance-low absorptance, cured at 300°F.
TiO_2	Silicone	high emittance-low absorptance, cured at 250°F.
TiO_2	Mod. Silicone	high emittance-low absorptance, recommend cure at 480°F.
Carbon	Acrylic	high absorptance-high emittance, cured at room temperature.

Q.6.4 CONDUCTIVE JOINT FILLERS

Thermal conductivity across joints and the use of metallic foils is discussed in the thermal analysis section of the report. Highly conductive metal foils which retain a minimal amount of strength at 300°F will be suitable. Lead aluminum and indium failures were discussed. Since the 313°F melting point of indium is very close to the internal sterilization temperature its usefulness may be limited.

Silver filled silicone greases, currently being used, will not be suitable. Cements based upon thermosetting resins filled with metal powder do appear useful. A silver filled epoxy cement is available which can withstand continued exposure to approximately 400°F without deterioration.

Q.6.5 TEMPERATURE INDICATORS

Assurance of sterilization may depend heavily upon indirect methods of indication that all portions of the spacecraft reached 300°F. Certain materials, paints and papers, which undergo irreversible color changes at specific temperatures may aid in establishing this assurance particularly during preliminary testing.

Papers are available which turn from white to black at specified temperatures. Papers are available which indicate 10°F steps up to 400°F. Temperature indicating paints have also proved useful for establishing maximum temperature attained during a process or operation. Visual examination following sterilization will pose problems, particularly in internal areas of the vehicle.

Q.6.6 REFERENCES

Many references were consulted during this study. Since the primary purpose was to present general, or typical data, and a discussion of the problems inherent in thermal sterilization specific references were not inserted with the data. The principal sources have been listed below according to the section in which they were used.

A. Metallic Material

1. "Metals Handbook", 8th Ed., Amer. Soc. of Metals, 1961.
2. A. Goldsmith et al, "Handbook of Thermophysical Properties of Solid Materials", MacMillan Company, New York, 1961.
3. A.A. Watts, "Space Materials Reference Book", MSD, General Electric Company, 1963.
4. "Strength of Metal Aircraft Elements", U.S. Dept. of Defense, MIL-HDBK 5, 1961.
5. "Report on Elevated Temperature Properties of Aluminum and Magnesium Alloys", Tech. Publ. 291, ASTM, 1960.
6. "Determination of Design Data for Heat Treated Titanium-Alloy Sheet", ASD-TDR-62-335, Vol. 3, December, 1962.
7. W.P. Achbach and R.J. Favor, "Design Information on Titanium Alloys for Aircraft and Missiles", DMIC Report 145, January 1961.
8. "Properties of Ti-4Al-3Mo-IV", Engineering Bulletin No. 8, Titanium Metals Corp. of America.
9. "Properties of Ti-13V-11Cr-3Al", Engineering Bulletin No. 9, Titanium Metals Corp. of America.
10. "Properties of Ti-6Al-6V-2Su", Engineering Bulletin No. 10, Titanium Metals Corp. of America.
11. M.M. Lemcoe and A. Trevino, "Determination of the Effects of Elevated Temperature on Material Properties of Several High Temperature Alloys", ASD-TDR-61-529, June 1962.
12. D.A. Roberts, D.B. Roach, A.M. Hall, "Physical and Mechanical Properties of Nine Commercial Precipitation Hardenable Stainless Steels", DMIC Report 112, May 1959.

B. Adhesives

1. F.M. Clark, "Insulating Materials for Design and Engineering", John Wiley and Sons, 1962.

2. M.W. Riley, "Adhesive Bonding: State of the Art", "Materials in Design Engineering", December 1961.

3. "Qualification of Scotch-weld Brand Structural Adhesive System, AF-7431, in accordance with MIL-A-5090D", Minnesota Mining and Mfg. Co. Report, October 1961.

4. "Handbook of Adhesives", Bloomingdale Rubber Co., 1961.

5. Normco Materials, Normco Materials Division, Telecomputing Corp.

6. H.G. Lefort, "Ceramic Adhesives Score in High Temp Uses", Ceramic Industry, April 1959.

7. K.W. Cornely, "Refractory Cements Good Up to 5300°F", Mat'ls in Des. Eng., November 1961.

C. Composite Materials

1. "Honeycomb Sandwich, Design Data and Test Methods", Hexcel Products Inc., 1959.

2. "Mechanical Properties of Hexcel Honeycomb Materials", Hexcel Products Inc., 1964.

3. K.R. Hoffman and C.E. Elliot, "High Temperature Resistance of Silicone Laminates", Proc. of 18th Conf., Reinforced Plastics, Soc. of Plastics Ind., February 1963.

4. J.H. Freeman, et al, "Glass Reinforced Organic Resin Laminates for Service to 300°C", 18th Tech. Conf., Soc. of Plastics Eng., January 1962.

5. "Composite Construction for Flight Vehicles", U.S. Dept. of Defense, MIL-HDBK-23, 1959.

D. Thermal Shield Materials

1. R.L. Mehan, "Mechanical Properties of Phenolic-Nylon at Temperatures Between 75 and 665°F", G.E. Report PIR 2243-432, March 1962.

2. G. Bainton and L.S. Lazar, "Behavior of Phenolic-Nylon from -140 to +250°F", G.E. Report PIR 244-105, August 1951.

3. J. Axelson and C. Dolan, "

4. Voyager Design Study, Vol. IV.

DISTRIBUTION

INTERNAL

R-ME-MM
James Dickinson
Fred J. Beyerle
Earl A. Hasemeyer

R-P&VE-AL
Dr. Pearl Fisher

R-P&VE-MM
Felix P. LaIacona

R-P&VE-SAA
Neil W. Flory

R-P&VE-PT
Carlton Garner

R-RP-S
Jerry Johnson

R-SA
Dr. Preston Farish

R-P&VE-SS
Paul Frederick

R-RP-R
Harry Coons

R-RP-DIR
Dr. Ernst Stuhlinger

R-P&VE-SA
Josef F. Blumrich

R-QUAL-J
Eugene Buhmann

R-ASTR-A
Fred Digesu

R-P&VE-SA
Erich Engler

R-P&VE-MEE
Eugene C. McKannan

R-P&VE-N
W. A. Brooksbank

R-P&VE-S
G. A. Kroll

R-P&VE-SAA
Ron G. Crawford (20)

MS-IP

MS-IPL (8)

HME-P

CC-P

EXTERNAL

NASA Headquarters, OIA
Mr. David S. Liberman, Code KR

NASA Headquarters, OSSA
James R. Miles, Sr., Code SL (3)
Donald P. Hearth, Code SL
Oran Nicks, Code SL
Paul Tarver, Code SL
Donald P. Easter, Code SL
William O'Bryant, Code SL
Lawrence B. Hall, Code SB (3)
Dr. O. E. Reynolds, Code SB
Dr. Carl W. Bruch, Code SB
Dr. Freeman H. Quimby, Code SB

DISTRIBUTION (CONTINUED)

NASA Headquarters, OART
Dr. James B. Edson, Code R-1

NASA
Langley Research Center
Langley Field, Virginia
Hampton, Virginia
William D. Mace, Instruments Research
R. S. Van Fleet, Clean Room Technology
W. J. Watson, Mail Stop 159

NASA
Jet Propulsion Laboratory
California Institute of Technology
4800 Oak Grove Drive
Pasadena 3, California
Victor J. Magistrale, 315 (T-201)
Don Kohorst, Div. 35 (3)
Dwain Spenser, Div. 35
Paul L. McGiven, 315 (T-201)
Leo Lunine, Technical Information

NASA
Ames Research Center
Moffett Field, California
Mountain View, California
Dr. Lawrence H. Frommhagen, Exobiology
Dr. Alvin Seiff, Aero-Theromodynamics

NASA
Goddard Space Flight Center
Greenbelt, Maryland
Norman H. MacLeod, Aeronomy and Meteorology

National Aeronautics and Space Administration
Washington, D. C. 20546
Library

DISTRIBUTION (CONTINUED)

National Aeronautics and Space Administration
Ames Research Center
Moffett Field, California 94035
Library

John F. Kennedy Space Center, NASA
Cocoa Beach, Florida 32931
Library

National Aeronautics and Space Administration
MSC -- Florida Operations
Box MS
Cocoa Beach, Florida 32931
Library

National Aeronautics and Space Administration
Flight Research Center
P.O. Box 273
Edwards, California 93523
Library

National Aeronautics and Space Administration
Goddard Space Flight Center
Greenbelt, Maryland 20771
Library

National Aeronautics and Space Administration
Goddard Institute for Space Studies
475 Riverside Drive
New York, New York 10027
Library

Jet Propulsion Laboratory
4800 Oak Grove Drive
Pasadena, California 91103
Library

National Aeronautics and Space Administration
Langley Research Center
Langley Station
Hampton, Virginia 23365
Library

DISTRIBUTION (CONTINUED)

National Aeronautics and Space Administration
Lewis Research Center
21000 Brookpark Road
Cleveland, Ohio 44135
Library

National Aeronautics and Space Administration
Manned Spacecraft Center
Houston, Texas 77058
Library

National Aeronautics and Space Administration
Michoud Operations
P.O. Box 29300
New Orleans, Louisiana 70129
Library

National Aeronautics and Space Administration
Mississippi Test Operations
Bay St. Louis, Mississippi 39520
Library

National Aeronautics and Space Administration
North Eastern Office
30 Memorial Drive
Cambridge, Massachusetts 02142
Library

National Aeronautics and Space Administration
Pacific Launch Operations Office
P.O. Box 425
Lompoc, California 93438
Library

National Aeronautics and Space Administration
Plum Brook Station
Sandusky, Ohio 44871
Library

DISTRIBUTION (CONTINUED)

AEC-NASA Space Nuclear Propulsion Office
Division of Reactor Development
U. S. Atomic Energy Commission
Washington, D. C. 20545
Library

National Aeronautics and Space Administration
Space Nuclear Propulsion Office -- Albuquerque
P. O. Box 5400
Albuquerque, N. Mexico 87115
Library

National Aeronautics and Space Administration
Space Nuclear Propulsion Office -- Cleveland
Lewis Research Center
21000 Brookpark Road
Cleveland, Ohio 44135
Library

National Aeronautics and Space Administration
Wallops Station
Wallops Island, Virginia 23337
Library

National Aeronautics and Space Administration
Western Operations Office
150 Pico Boulevard
Santa Monica, California 90406
Library

NASA Senior Scientific Representative to Australia
c/o Secretary, Department of Supply
339 Swanson Street
Melbourne, C. 1, Victoria, Australia
Library

Instituto Geofisico deHuancayo
NASA Satellite Tracking Station
Apartado 3747
Lima, Peru
Library

DISTRIBUTION (CONTINUED)

National Aeronautics and Space Administration
Space Nuclear Propulsion Office - Nevada
Nuclear Rocket Development Station
P.O. Box 1
Jackass Flats, Nevada 89023
Library

NASA Daytona Beach Office
General Electric Company
P.O. Box 2500
Daytona Beach, Florida 32015
Library

National Aeronautics and Space Administration
Office - Downey
North American Aviation, Inc.
Downey, California 90241
Library

National Aeronautics and Space Administration
Space Nuclear Propulsion Office
Germantown, Maryland 20545
Library