BLOCK II SOLID ROCKET MOTOR (SRM) CONCEPTUAL DESIGN STUDY CONTRACT NAS 8-37295

FINAL REPORT - VOLUME I

APPENDICES

Prepared for:

GEORGE C. MARSHALL SPACE FLIGHT CENTER
NATIONAL AERONAUTICS AND SPACE ADMINISTRATION
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PROPULSION DIVISION

31 December 1986

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TABLE OF CONTENTS

VOLUME I - FINAL REPORT

APPENDIX A

MID-TERM REPORT
(SUPERSEDED AND REVISED)

BLOCK II SOLID ROCKET MOTOR (SRM) CONCEPTUAL DESIGN STUDY CONTRACT NAS 8 - 37295

MID-TERM REPORT

SUBMITTED TO:

NATIONAL AERONAUTICS AND SPACE ADMINISTRATION
MARSHALL SPACE FLIGHT CENTER, ALABAMA 35812

SUBMITTED BY:

ATLANTIC RESEARCH CORPORATION
PROPULSION DIVISION
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TR-PL 12127

REV. A

FOREWORD

This document is submitted as a Midterm Report satisfying the following requirements of Contract NAS8-37295, Conceptual Design Studies of a Block II Space Shuttle Solid Rocket Motor (SRM):

1) Conceptual Design Package (Preliminary),

2) Preliminary Development and Validation Plan (Preliminary).

TABLE OF CONTENTS

TABLE OF CONTENTS (CONTINUED)

TABLE OF CONTENTS (CONTINUED)

1.0 INTRODUCTION

This Midterm Report describes Atlantic Research Corporation's Block II Space Shuttle Solid Rocket Motor (SRM) Conceptual Design Study Program. The objective of this program is to provide a verifiable SRM concept which eliminates all deficiencies identified with the SRM designs in Space Shuttle Mission 51-L. The Conceptual Design must offer improved flight safety, reliability, and design confidence while maintaining compatibility with Space Shuttle vehicle and launch facilities. Improvements in performance and cost are desirable but secondary.

1.1 ORGANIZATION

The Atlantic Research Corporation (ARC) Block II SRM team is shown in Figure 1.1.1. Major support has been provided by the contractor and consultants shown. The ARC team members identified are also further supported by Propulsion Division Staff on a specific task basis.

1.2 APPROACH

Interaction among the four contractual tasks is shown in Figure 1.2.1, Block II SRM Contract Study Plan. The design studies task implements the primary program objective of developing a Block II SRM design offering improved flight safety and reliability. Sub-tasks shaded in Figure 1.2.1 have been completed. Review of SRM literature and detailed discussion with NASA personnel has identified deficiencies in the Mission 51-L SRM and required improvements such as elimination of asbestos. Study topics and criteria were selected based on the information. These are discussed in Section 2.0.

A summary of the Preliminary Development and Validation (D&V) Plan is shown in Section 4.0. The Capability Assessment Task was initiated early in the program to provide support to the Design Concept Study Task in monolithic versus segmented SRM handling, transportation, and facilities.

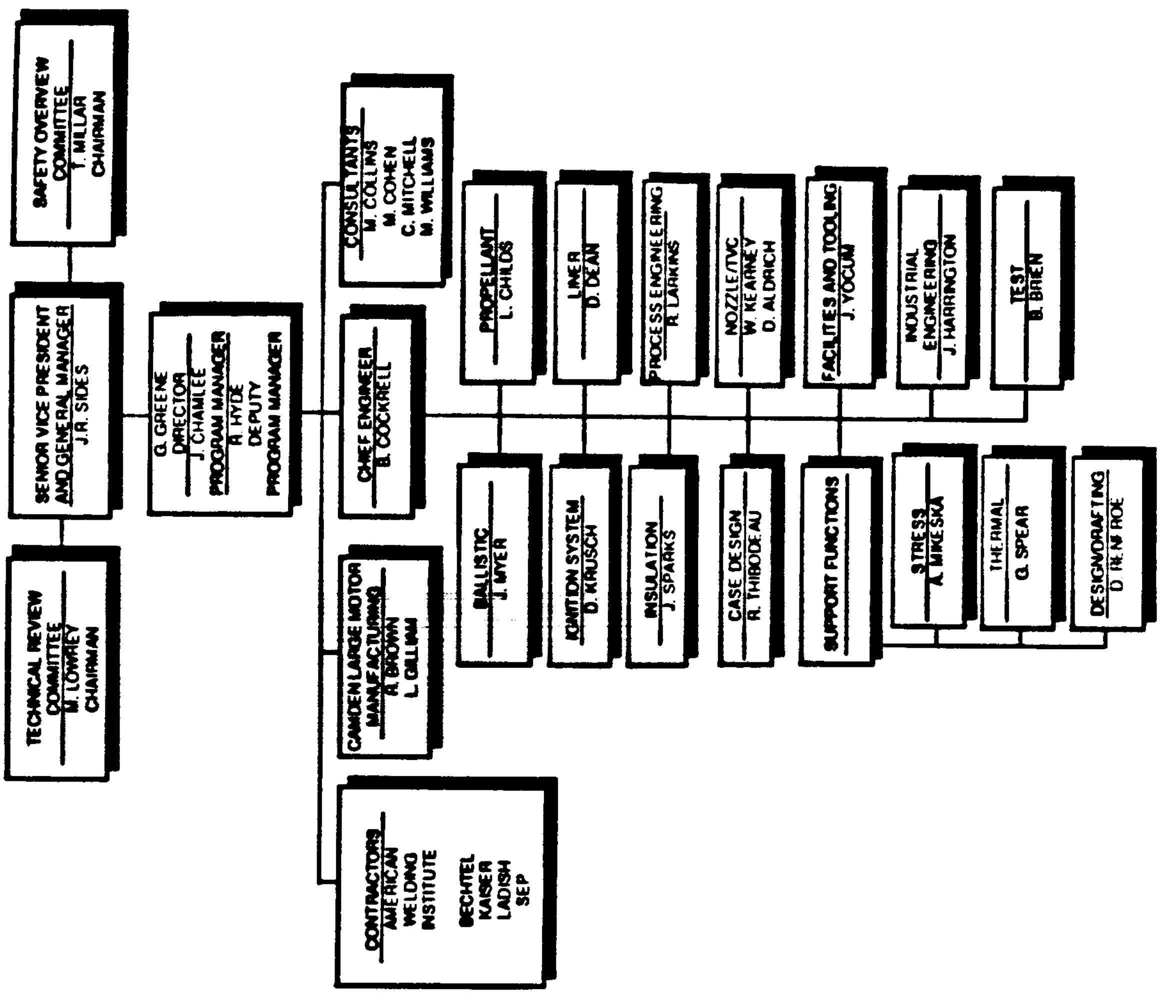

FIGURE 1.1.1. ARC BLOCK II SRM PROGRAM TEAM.

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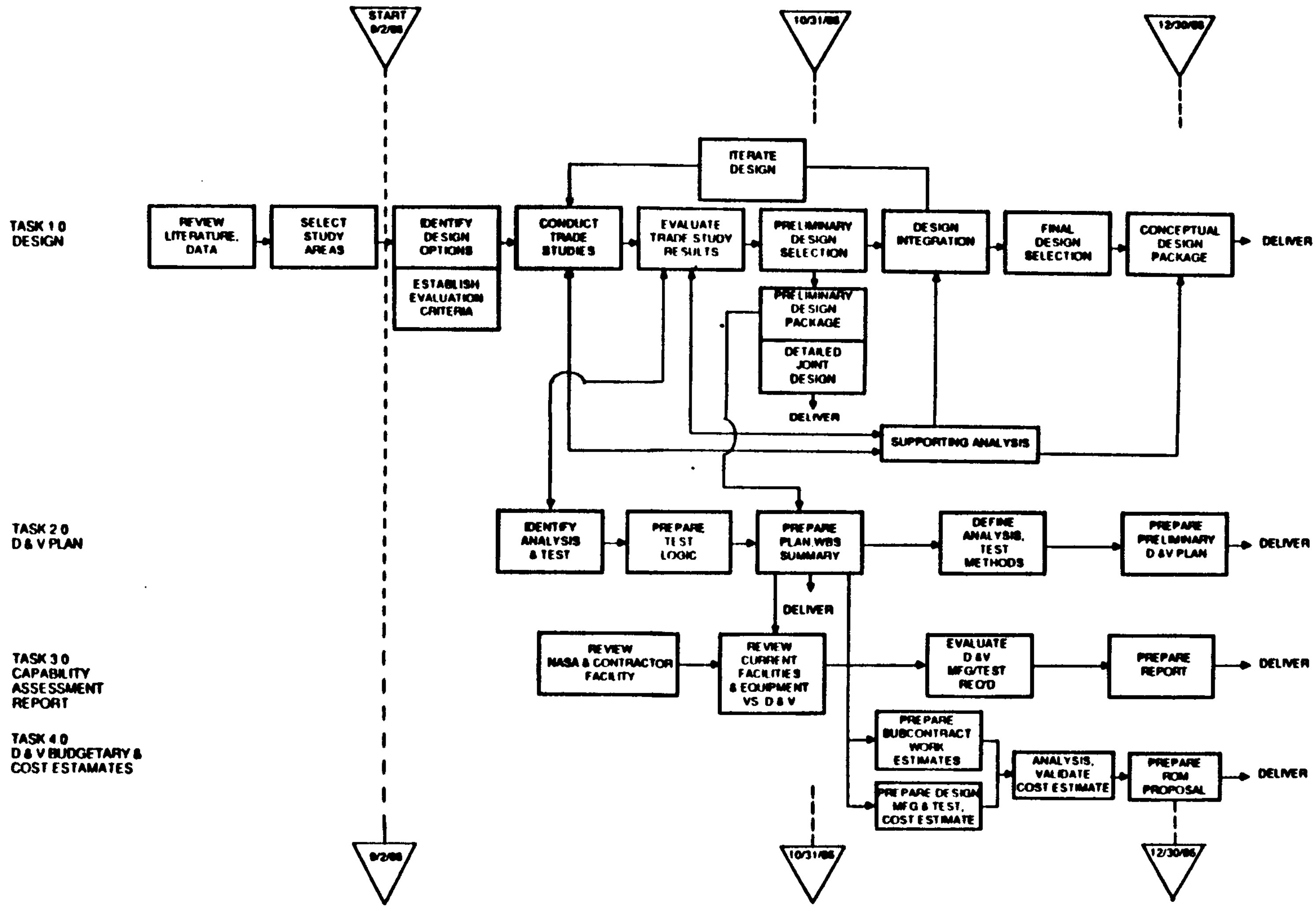


FIGURE 1.2.1. CONTRACT STUDY FLOW DIAGRAM.

1.3 PRELIMINARY DESIGN CONCEPT

The preferred Block II SRM preliminary design concept that has emerged from trade and design studies to date is depicted in Figures 1.3.1 and 1.3.2 and is described as follows.

The SRM is a segmented design having casting segment lengths identical to those of the 51L design. One double length D6AC case segment has been substituted for each prior pair of two adjacent 160" or 120" long segements. This has eliminated four 51L type factory joints. The three field joints connecting the four casting segments are of the inline bolted flange type, each fastened with high strength steel studs with Inconel 718 nuts on each end. Each field joint incorporates redundant non-elastomeric face seals; one "Flexotallic" gasket-type (non-asbestos) seal and one metallic "C"-type seal.

The nozzle-to-case joint also incorporates redundant face-type seals; one metallic "C"-type seal and one elastomeric o-ring seal.

All internal insulation joints are of the unvented type with a labrynth path which precludes direct exposure of the joint seals to hot combustion gases. Mating insulation joints are filled with low strength, high strain room temperature cure sealant. Stress relief features are incorporated in the insulation near the mating joints to permit relative motion of the insulation components without overstressing the insulation joint sealant.

The propellant formulation and grain configuration of each casting segment is identical to the 51L design.

The case insulation design is a hybrid system to optimize weight and performance. A Kevlar/silica/Hypalon material is used next to the case wall because its low thermal diffusivity provides the optimum thermal protection for the reusable case. To provide erosion protection near field joints and in areas which are exposed during propellant burn such as the aft case, the Hypalon insulation will transition to an NBR/phenolic with boric acid filler (USR-3800). The molded inhibitors will also be made from USR-3800. The castable liner will be a CTPB material for compatibility similar to the current liner material with the asbestos fibers replaced with another filler material.

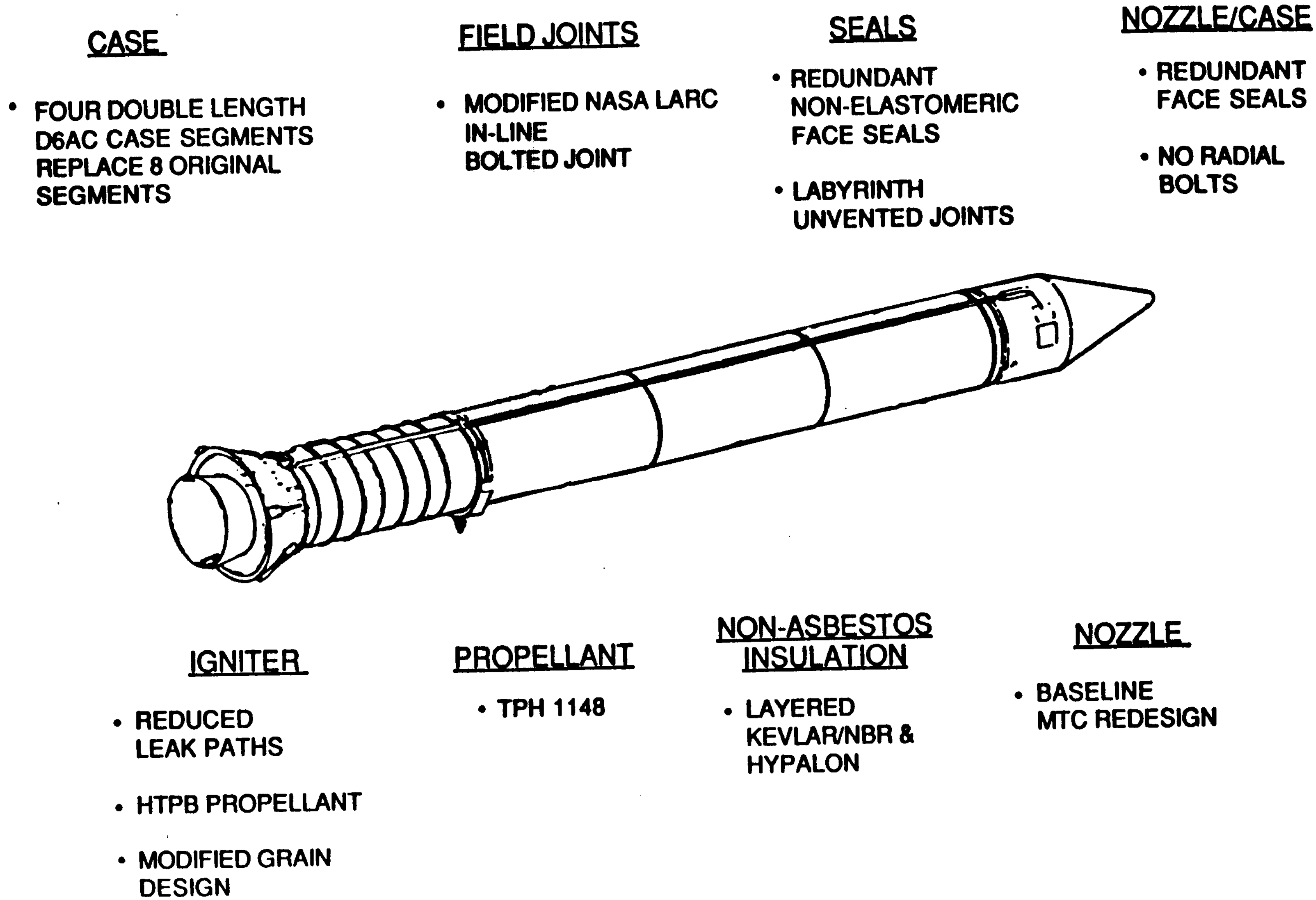


FIGURE 1.3.1. PRELIMINARY BLOCK II SRM DESIGN FEATURES.

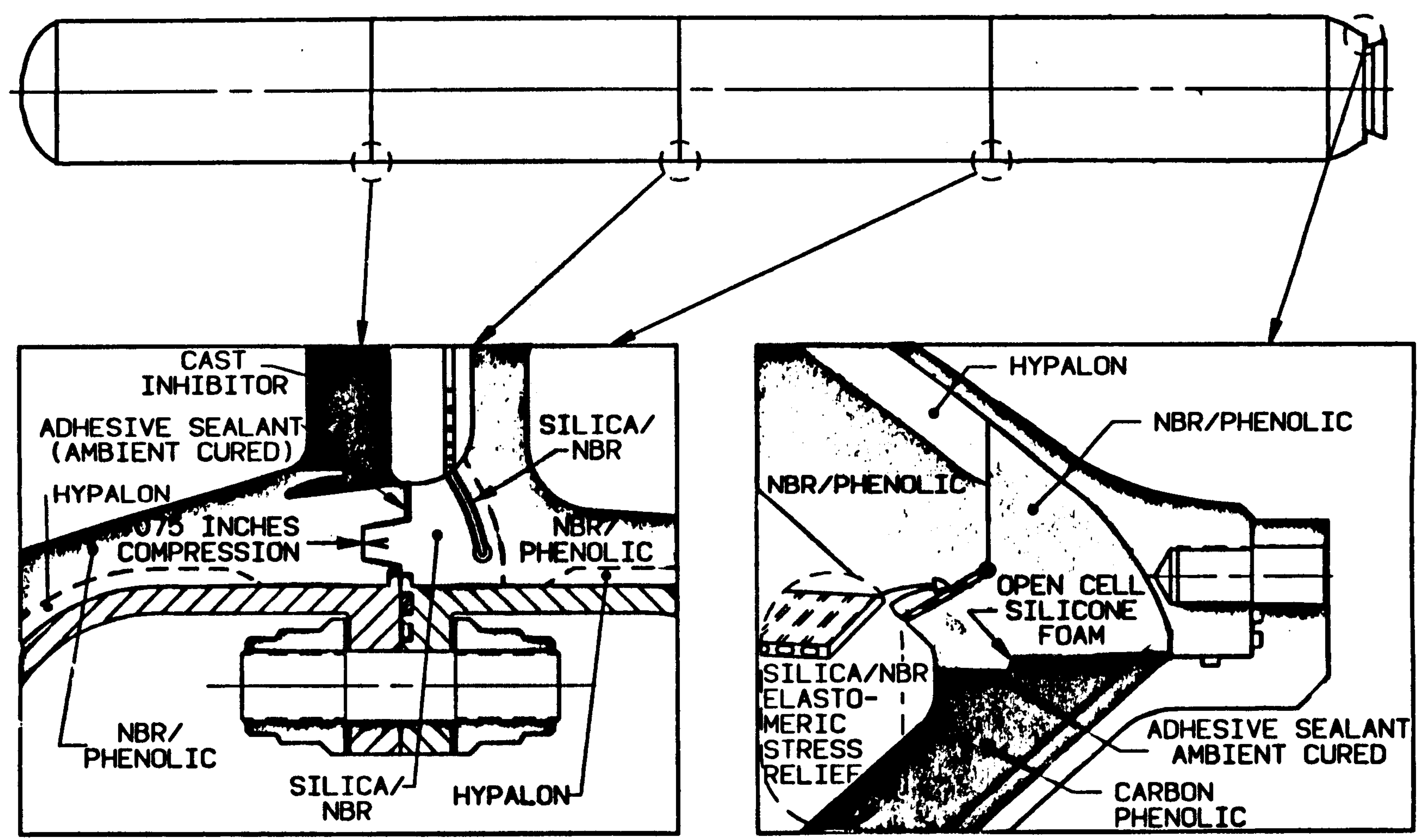


FIGURE 1.3.2. ARC BLOCK II SRM FIELD JOINTS.

The nozzle configuration is basically the same as the 51L configuration except certain materials have been changed to eliminate asbestos and/or to eliminate pocket erosion problems. Also, internal joints have been reconfigured as needed to provide redundant seals.

The preferred igniter design consists of an integral igniter adapter and case with a bolt-on aft closure formed from 200 maraging steel. The igniter assembly is insulated with Kevlar and silica-filled Hypalon and loaded with 18% aluminized HTPB propellant. All joints are sealed using t-ring variants and metal c-rings.

2.0 TRADE STUDIES

Scope

Trade studies were conducted to select preferred Block II SRM design features and materials in the following areas:

- Design Approach (segmented vs. monolithic),
- Motor case,
- Joints and seals,
- Non-asbestos insulation,
- Propellant and liner,
- Igniter,
- Nozzle.

To support these trade studies, additional transporation, handling, and assembly analyses were conducted for both the segmented and monolithic approaches.

Methodology

The following trade study methodology was established and followed to ensure use of consistent and unbiased criteria in the ranking and evaluation of competing design approaches.

Candidate designs were compared on the basis of relative reliability, cost, and payload capability. Definitions of these ranking categories are as follows:

<u>Reliability</u> - the ability of the SRM to successfully function and propel the shuttle through the intended trajectory without threatening the safety of the flight crew.

<u>Cost</u> - Total life cycle cost, both recurring and non-recurring, to design, develop, fabricate, transport, and assemble the SRMs needed to support 15 shuttle flights per year for 10 years.

<u>Payload Capability</u> - Number of pounds of payload that can be injected into low earth orbit by the shuttle using the candidate SRMs.

Of the ranking categories, the most important by far is reliability. Weighting factors were therefore assigned to each ranking category to account for their relative importance. Reliability was assumed to be twice as important as the other two categories combined; hence, reliability was assigned a weighting factor of 0.65 (or 65%). Cost and payload capability were considered to be approximately equal, hence, cost was assigned a weighting factor of 0.20 (20%) and payload capability was assigned a weighting factor of 0.15 (15%).

The next step was to devise criteria to assess the relative merit of competing designs in each ranking category. For this purpose it was decided to use a scoring system from 1 to 10, where 1 is worst and 10 is best. Specific criteria used is shown in Table 2.0.1.

2.1 DESIGN APPROACH TRADE STUDY (updated 12/19/86)

2.1.1 CANDIDATE DESIGN APPROACHES

Two basic design approaches were considered. They were (1) segmented design, having propellant grain segments identical to the Block I design, and (2) monolithic design, having a single full length, one piece propellant grain. Two motor case segment variants were considered for each of these basic approaches. The first variant uses 11 case segments identical in length to the 11 Block I design case segments. The second variant uses longer case segments as follows. The forward, forward center, and aft center casting segments each use a one-piece, 320" long, cylindrical, weld-free case segment in place of the two (160" each) cylindrical case segments used in the Block I design. The aft casting segment uses a one-piece, 326" long, weld-free, cylindrical case segment in place of the three (86", 120", and 120") cylindrical case segments used in the Block I design. The segmented candidates were assumed to use 51-L type factory joints and new, improved field joints. The monolithic candidates were assumed to use 51-L type factory joints throughout. The current PBAN propellant was assumed for all candidates. Rationale for consideration of these candidates is as follows.

Current Grain Segments, Current Case Segments

This candidate enjoys the distinct advantage that all required manufacturing, transportation, handling, and assembly facilities, equipment, and procedures are well defined and proofed (with the possible exception of minor changes associated with (1) improved field joints, and (2) asbestos-free insulation). It should therefore represent the lowest cost approach for second sourcing. However, its joint reliability will be less than the monolithic approach since it still has three field joints (albeit improved).

TABLE 2.0.1. TRADE STUDY EVALUATION CRITERIA SCORING SYSTEM.

CATEGORY	SCORE	CONDITION
RELIABILITY	10	MUCH BETTER THAN 51L (APPROACHING 1.0)
	5	EQUIVALENT TO 51L DESIGN
	1	MUCH WORSE THAN 51L DESIGN
COST	10	9% LESS THAN 51L DESIGN
	7	EQUIVALENT TO 51L DESIGN
	3	12% HIGHER THAN 51L DESIGN
	1	18% HIGHER THAN 51L DESIGN
PAYLOAD CAPABILITY	10	9% MORE THAN 51L DESIGN
	7	EQUIVALENT TO 51L DESIGN
	3	12% LESS THAN 51L DESIGN
	1	18% LESS THAN 51L DESIGN

Current Grain Segments - Longer Case Segments

Use of longer case segments would provide inert weight and assembly cost benefits through elimination of five factory joints. The inert weight saved could then be assigned to the field joints to provide an increase in joint reliability, or conversely could provide an attendant increase in payload weight capacity. The longer case segment facilities/processes must be developed, but all other required facilities/equipment/procedures exist. It should represent the next lowest cost approach to second source, but would still have whatever small residual unreliability is associated with the three improved field joints.

Monolithic Grain - Current Case Segments

The major advantage of the monolithic candidate is that joint reliabilty is maximized since no field joints exist. It is also possible that payload weight advantages could accrue due to reduced case insulation weight and increased propellant weight. Use of current case segments to assemble the entire case prior to casting is justified since no unreliability problems have been identified with factory joints. However, the monolithic approach in general has several formidable disadvantages. These include (1) potential reliability degradations in the propellant grain/case insulation/case bond areas arising from the yet undeveloped fabrication/casting processes required, (2) safety issues associated with transporting such a large, propulsive SRM from the manufacturing facility to the launch facility, (3) high cost to develop and procure the required manufacturing facilities, (4) high cost of the D&V program needed to provide the required large data base, and (5) high cost to develop and procure the equipment and facilities needed to handle, transport, and assemble the monolithic SRM.

Monolithic Grain - Longer Case Segments

Advantages and disadvantages of this candidate are the same as the previous monolithic candidate, except that a payload advantage would accrue due to reduction of inert weight associated with elimination of five factory joints.

Advantages and disadvantages of the candidate design approaches are shown in Table 2.1.1.

TABLE 2.1.1. DESIGN CONCEPT TRADE STUDY.

DESIGN CONCEPT	ADVANTAGES	DISADVANTAGES
CURRENT GRAIN SEGMENTS - CURRENT CASE SEGMENTS	• REQUIRED FACILITIES, EQUIPMENT, PROCEDURES ARE DEFINED AND PROOFED • LOWEST COST TO SECOND SOURCE	• UNRELIABILITY ASSOCIATED WITH 3 FIELD JOINTS • WEIGHT ASSOCIATED WITH 7 FACTORY JOINTS
CURRENT GRAIN SEGMENTS - LONGER CASE SEGMENTS	• EXCEPT FOR LONGER CASE SEGMENTS, REQUIRED FACILITIES, EQUIPMENT, PROCEDURES ARE DEFINED AND PROOFED • NEXT LOWEST COST TO SECOND SOURCE • INCREASED PAYLOAD DUE TO REDUCED WEIGHT OF FACTORY JOINTS, OR: • INCREASED RELIABILITY OF BEEFED-UP FIELD JOINTS FOR SAME TOTAL INERT WEIGHT	• UNRELIABILITY ASSOCIATED WITH 3 FIELD JOINTS • DEVELOPMENT AND FACILITIZATION COSTS OF LONGER CASE SEGMENTS
MONOLITHIC GRAIN - CURRENT CASE SEGMENTS	• NO FIELD JOINTS - MAXIMUM JOINT RELIABILITY • PAYLOAD CAPABILITY INCREASE DUE TO DECREASED INSULATION AND INCREASED PROPELLANT	• DEVELOPMENT AND FACILITIZATION COST TO FABRICATE MONOLITHIC GRAIN • FACILITY/EQUIPMENT COST TO TRANSPORT AND HANDLE MONOLITHIC MOTOR • D&V COSTS TO DEVELOP LARGE DATA BASE REQUIRED • PROPELLANT GRAIN/CASE INSULATION/CASE BOND RELIABILITY UNCERTAINTIES

TABLE 2.1.1. CONTINUED.

DESIGN CONCEPT	ADVANTAGES	DISADVANTAGES
MONOLITHIC GRAIN - CURRENT CASE SEGMENTS (CONTINUED)		• TRANSPORTATION SAFETY CONSIDERATIONS • WEIGHT ASSOCIATED WITH 10 FACTORY JOINTS
MONOLITHIC GRAIN - LONGER CASE SEGMENTS	• NO FIELD JOINTS - MAXIMUM JOINT RELIABILITY • PAYLOAD CAPABILITY INCREASE DUE TO DECREASED INSULATION AND INCREASED PROPELLANT • INCREASED PAYLOAD DUE TO REDUCED WEIGHT OF FACTORY JOINTS	• DEVELOPMENT AND FACILITIZATION COST TO FABRICATE MONOLITHIC GRAIN • FACILITY/EQUIPMENT COST TO TRANSPORT AND HANDLE MONOLITHIC MOTOR • D&V COSTS TO DEVELOP LARGE DATA BASE REQUIRED • PROPELLANT GRAIN/CASE INSULATION/CASE BOND RELIABILITY UNCERTAINTIES • TRANSPORTATION SAFETY CONSIDERATIONS

2.1.2 DISCUSSION OF MONOLITHIC SRM DESIGN ISSUES

Issues bearing on the ranking criteria of flight reliability, payload capacity, and cost were raised in several areas for the monolithic design approach and are addressed in the following paragraphs.

Design Integrity Issues

Ballistic performance and grain design studies were conducted to verify that the thrust vs. time performance delivered by the existing segmented design could be duplicated using a one piece monolithic grain design. These analyses identified two monolithic grain designs capable of duplicating the existing thrust-time trace - one having slots at the head end of the motor and one having slots at the nozzle end of the motor.

Abbreviated structural analyses were conducted to assess the stress-strain state of the monolithic grains identified by the ballistic design studies. These analyses indicated acceptable stress-strain conditions would exist in the grain bore, slot, slot termination, and grain end termination areas over the specified environmental and operational ranges.

Manufacturing Issues

Manufacturing trade studies were performed to determine the most reliable and cost efficient method of production for a one-piece, monolithic SRM grain with current length or longer case segments. Where possible, manufacturing procedures similar to those for the SRM Block I segmented design were selected. However, special procedures, equipment and facilities will be required in many circumstances to produce a monolithic grain to the specified configuration.

Case insulation integrity may be affected. Since the insulation layup takes place once the case segment assembly is complete, the problem of handling a very long steel case is imposed. It is unknown whether or not the rubber can be kept at a constant temperature as it is fed into the length of the motor. Also, an extremely large autoclave is required for the vulcanization process, not to mention the size of the vacuum bag.

Problems encountered with similar applications of large vacuum bags include premature deflation of the bags which tends to ruin the insulation.

A 116-foot casting-segment length causes concern over the lining and propellant casting operations. A sling-lining technique is optimum for large-diameter cases regardless of segment length; however, a significant redesign of the liner applicator will be required in order to line the full length of the case in one continuous operation. Similarly, due to the long propellant drop height for the monolithic grain configuration, an alternate propellant casting method must be developed. The tooling required for casting will be more complex and costly than that currently used to cast a Block I SRM segment. Large, heavy-duty cranes and equipment are needed, as well as a 15-story deep pit and 15-story high building structure.

The new casting method envisioned utilizes a segmented bayonet which is lowered into the case for casting. Bayonet segments are withdrawn from the case as the propellant level rises. Although the propellant drop height problem is alleviated, the casting method still involves working with tooling at great heights above the bottom of the casting pit. This poses serious safety issues.

Major uncertainties exist relative to liner, propellant and bond integrity of the one-piece, monolithic grain configuration. Maximum propellant fill time, consistent with current casting flow rates for the Block I SRM segments, is roughly 5 1/2 days. This casting fill time is greater than the liner and propellant cure times. Therefore, propellant at the bottom or forward section of the SRM grain will be fully cured before casting at the top or aft section is complete. Effects of a cure gradient on the propellant bond and bulk properties are not entirely understood at this time. The effect of hydrostatic pressure gradient might also affect propellant properties.

Mandrel insertion and extraction in a monolithic grain will compromise manufacturing safety and reliability factors, as well as cost. Trade studies indicate that a segmented mandrel is most suitable for a single-piece monolithic grain. The mandrel would consist of a segmented inner core, which is pre-assembled prior to insertion into the case, and segmented fins, which are pre-assembled, inserted and attached to the inner core. Due to the size of the aft case opening, the fins must be lowered through the clearance between the inner core and case, and attached to the inner core within the case. This task is challenging regardless of slot location, aft or forward. Core popping also proves to be somewhat difficult in a monolithic grain for several reasons. Mandrel extraction is stressful to the mandrel tooling itself due to the adhesive forces between the core and propellant. Hydraulic systems both at the top and bottom of the casting pit are required to provide force to initially release the mandrel. An intermediate system near the aft section is also necessary to remove and detach the extracted mandrel sections. Drop height is still a concern as heavy-duty cranes and large equipment are operated 116 feet above the bottom of the casting pit.

Other manufacturing processes which involve special procedures for the one-piece design include breakover, x-ray and grain finishing. As noted in other operations, the size and weight of the monolithic SRM greatly hinder the processing and handling flow. The equipment required to manage such a motor is not easily maneuverable. Although the number of handling steps is greatly reduced for a monolithic SRM, the level of difficulty assigned to each operation is significantly increased. This has direct bearing on manufacturing safety and reliability. Should problems occur during processing the motor or a defect detected which may cause rejection by quality control, an entire 1.1 million pound motor may be lost. This is expected to create tremendous pressure on program personnel to accept or repair a marginal motor in order to avoid schedule slippage or to take a multimillion dollar loss.

Handling Issues

The larger physical dimensions and higher weight of the monolithic motor relative to the segmented motor requires much larger and sturdier construction of handling equipment. This results in higher costs of handling equipment, tooling and facilities. The cost multiple of monolithic vs. segmented motor handling is much more than the respective weight multiple. A monolithic motor will require handling equipment and facilities at KSC that do not currently exist. A multimillion dollar hoisting facility would be required. The VAB currently handles the KSC hoisting requirements for the segmented motor but does not have the capability to handle a monolithic motor.

There are also safety concerns involved in handling a monolithic motor. For example, when the motor is being lifted out of the casting pit it will be suspended 13 to 14 stories above the bottom of the pit in the worst case. An error or accident at this point could be catastrophic.

Transportation Issues

Transportation of a monolothic motor from Camden, Arkansas to KSC presents serious problems due to its size and weight. Rail and barge transport were the two modes considered for the shipment of the motors. Of these two modes, barge transport was the only one deemed suitable for the monolithic motor. Shipping by rail was found to be unsuitable for the following reasons:

- The length of the motors makes curves, trackside obstacles, rail yards and adjacent rail lines difficult to negotitate and hazardous to cargo.
- The weight would require some rails, rail beds and bridges to be fortified. This is a very expensive prospect.
- The many hazards that would be encountered over the route are reason for concern because of the propulsive nature of the monolithic motors.
- There are many regulatory obstacles dealing with size of cargo, its weight, the custom-built railcars and its hazardous properties.
- The cost of the railcars is an estimated $6 million each. At least 20 would be required.

Waterborne transport is the only other candidate. This mode would, however, require a sizable capital expenditure for:

- Camden River dry dock loader
- LC-39 dry dock conversion (KSC)
- Rail connection at Camden
- Rail connection at LC-39 (KSC)
- 1500 ton river barge with railcar capacity
- 1000 ton railcar
- Rail extension to KSC SRM TS

There is still a question of safety regarding the propulsive nature of the monolithic motor. The estimated potential range of the motor is 300 to 500 miles in its shipping configuration, where the igniter and the nozzle have not been installed and both attach ports are fully open.

Monolithic SRM issues are summarized in Table 2.1.2.

2.1.3 RELIABILITY ASSESSMENT

The most important single attribute of the Block II SRM is its reliability. All of the candidate design approach concepts will provide higher reliability than the 51L design through improvement or elimination of field joints. For purposes of comparison, on a scale of 1 to 10 with 10 being best, the 51L configuration was assigned a reliability ranking value of 5.

The segmented designs incorporate field joints consisting of in-line bolted joints with redundant, non-elastomeric, non-pressure actuated face seals. Joint gap opening at the seal locations due to pressurization is practically non-existent. Non-vented labrynth insulation joints have also been incorporated at each field joint to preclude exposure of the seals to hot combustion gases. No other SRM case-liner-insulation-propellant features have been changed from the 51L configuration except for substitution of non-asbestos insulation. Since the demonstrated reliable features of the case/grain assembly have been retained, and the reliability of the field joints has been dramatically improved, the segmented designs were assigned a reliability ranking value of 8.8

The monolithic design candidates have the advantage of having no field joints at all; thus, they represent an ultimate 10 in joint reliability. However, serious uncertainties exist in other areas. Due to the long fill time, propellant near the head end of the motor will be completely cured before casting is complete. This cure gradient, along with the varying hydrostatic pressure caused by the propellant head, could adversely affect propellant physical and bond integrity. Further, the propellant liner near the top of the motor (late in the fill) will be completely cured before the uncured propellant is cast onto the liner. This could adversely affect bond integrity.

Due to the long, confined interior of a monolithic motor, increased difficulties will be experienced in applying insulation and liner to the motor interior. This could manifest itself in reduced reliability of the insulation and liner.

Defects detected by NDT of cast motors can potentially lead to reduced reliability of monolithic motors. This arises from the fact that a rejectable defect could cause loss of an entire monolithic motor but loss of only one casting segment of a

TABLE 2.1.2. MONOLITHIC MOTOR ISSUES.

DESIGN

- BALLISTICS - NO PROBLEMS; CAN DUPLICATE THRUST-TIME TRACE (FINS FWD OR AFT)
- GRAIN STRUCTURAL - NO PROBLEMS; EQUIVALENT STRESSES/STRAINS

MANUFACTURING

- SEGMENTED MANDRELS
- SEGMENTED CASTING BAYONET
- FILL TIME (PROPELLANT/LINER INTEGRITY)
- DEEP CASTING PITS, TALL HEAVY DUTY CRANES
- FLAWS - HIGH COST IF REJECT; LOW RELIABILITY IF ACCEPT
- INSULATION INTEGRITY

HANDLING

- SIZE AND COST OF HANDLING EQUIPMENT
- SAFETY

TRANSPORTATION

- ONLY VIABLE MODE IS BARGE
- LARGE CAPITAL EXPENDITURE REQUIRED
- SAFETY (PROPULSIVE UNIT)

segmented motor. It can therefore be expected that heavier pressure will exist for program personnel to accept or repair marginal conditions in the case of monolithic motors, thereby degrading reliability.

Because of these concerns, even though joint reliability is maximized, overall reliability of a monolithic motor was assigned a reliability ranking value of 7.4

2.1.4 PAYLOAD CAPABILITY ASSESSMENT

The design concept candidates were assessed for differences in payload capability resulting from differing SRM inert weights and propellant weights. The 51L configuration was assumed to be the baseline, capable of carrying a 60,000 lb. payload into low earth orbit. Payload influence coefficients were assumed to be:

$$(1)\ \frac{\Delta PL}{\Delta W_i} = \frac{-1\ \text{lb}}{+5.5\ \text{lb}} = -0.182\ \frac{\text{lb payload}}{\text{lb SRM inerts}}$$

$$(2)\ \frac{\Delta PL}{\Delta W_p} = \frac{+1\ \text{lb}}{+12.0\ \text{lb}} = +0.083\ \frac{\text{lb payload}}{\text{lb SRM propellant}}$$

The results of this assessment, shown in Table 2.1.3, show that payload capability change ranges from -0.8% to +0.3% for the segmented candidates, and ranges from +3.1% to +4.2% for the monolithic candidates. The better payload performance of the monolithic candidates is largely due to the extra propellant assumed to bridge the gaps between original grain segments. However, a significant portion of the added propellant would have to be cut out in the form of longer longitudinal slots in order to tailor to the correct thrust-time trace shape, which is expected to largely negate the assumed propellant weight increase.

The ranking scores shown in Table 2.1.3 were assigned in accordance with the criteria presented in Section 2.0.

TABLE 2.1.3. SRM PAYLOAD ASSESSMENT.

	CHANGES FROM 51L CONFIGURATION			
	SEGMENTED		MONOLITHIC	
	CURRENT CASE SEGMENTS	LONGER CASE SEGMENTS	CURRENT CASE SEGMENTS	LONGER CASE SEGMENTS
CASE WT, LB	+ 2,829	- 621	0	- 3,450
INSULATION WT, LB	- 4,333	- 4,333	- 7,733	- 7,733
INERT WT, SUM LB	- 1,504	- 4,954	- 7,733	-11,183
INERT EQUIV. P/L WT, LB	+ 274	+ 902	+ 1,407	+ 2,035
PROPELLANT WT	+ 1,189	+ 1,839	+12,359	+12,909
PROP. EQUIV. P/L WT, LB	+ 99	+ 153	+ 1,026	+ 1,071
TOTAL P/L WT, LB	+ 373	+ 1,055	+ 2,433	+ 3,106
% P/L CHANGE (60,000 LB BASE)	+ 0.6	+ 1.8	+ 4.1	+ 5.2
RANKING SCORE	7.2	7.6	8.4	8.7

2.1.5 COST ASSESSMENT

Large cost differences were anticipated between segmented and monolithic approaches in the areas of manufacturing, handling, transportation, and assembly. Significant differences in both recurring and non-recurring costs were expected. It was therefore decided that relative cost rankings should be based on total Life Cycle Cost (LCC) for the postulated mission model of 15 flights per year for 10 years.

SRM Life Cycle Cost Model

Considering recurring costs, a total of 300 SRMs is needed for the 15 flights per year, 10-year mission scenario. Based on the requirement for 19 reuses, and considering attrition, it was assumed that 20 new sets of SRM cases, nozzle metal parts, and igniter metal parts would be required. It therefore follows that 280 refurbishments of the SRM cases, nozzle metal parts, and igniter metal parts would be required.

Non-recurring costs include D&V costs, SRM manufacturing facility costs, transportation system costs, and KSC assembly/erection facility system costs.

Total Life Cycle Cost is defined as the sum of recurring and non-recurring costs.

The SRM Life Cycle Cost model is summarized in Table 2.1.4.

Recurring Costs

Normalized recurring cost estimates for each candidate design approach are summarized in Table 2.1.5. Unit costs were estimated for the discrete elements indicated, multiplied by the number of elements required, and summed to obtain the total recurring costs. The segmented candidate consisting of current case segment lengths and current casting segment lengths was chosen as the baseline and its total cost was normalized to 100%. All other costs were then normalized to the baseline cost. Unit costs of the longer case segment designs include amortized costs of added facilities and startup activities required to produce the longer case segments.

TABLE 2.1.4. SRM LIFE CYCLE COST MODEL.

- 300 SRMs (10 YR x 15 FLT/YR x 2 SRM/FLT)
- 20 NEW HARDWARE SETS (CASE, NOZZLE, IGNITER)
- 280 REFURBISHMENTS (CASE, NOZZLE, IGNITER)
- TOTAL LIFE CYCLE COST = RECURRING COSTS + NON-RECURRING COSTS

RECURRING COSTS	NON-RECURRING COSTS
• 20 NEW HARDWARE SETS (CASE*, NOZZLE, IGNITER)	• SRM D&V COST
• 280 REFURBISHMENTS (CASE, NOZZLE, IGNITER)	• SRM MANUFACTURING FACILITY COSTS
• 300 CASE PREPARATIONS	• TRANSPORTATION SYSTEM COSTS
• 300 NOZZLE FABRICATIONS	• ASSEMBLY/ERECTION FACILITY COSTS
• 300 IGNITER FABRICATIONS	
• 300 INSULATION/INHIBITOR FABRICATIONS	
• 300 LINER/PROPELLANT FABRICATIONS	
• 300 MOTOR FINISHING OPERATIONS	
• 300 SHIPMENTS TO KSC	
• 300 KSC ASSEMBLY/ERECTION OPERATIONS	

* INCLUDES AMORTIZED STARTUP AND FACILITIES COSTS RELATED TO LONGER CASE SEGMENTS WHERE APPROPRIATE.

TABLE 2.1.5. SRM RECURRING COSTS.

QTY	ELEMENT	TOTAL RECURRING COSTS, %			
		SEGMENTED		MONOLITHIC	
		CURRENT CASE SEGMENTS	LONGER CASE SEGMENTS	CURRENT CASE SEGMENTS	LONGER CASE SEGMENTS
20	CASE, NOZZLE, IGNITER METAL PARTS	9.1	10.2	9.2	10.2
280	CASE, NOZZLE, IGNITER REFURB.	1.8	1.7	1.8	1.7
300	CASE PREPARATION	1.5	1.3	2.3	2.2
300	NOZZLE FABRICATION	19.7	19.7	19.7	19.7
300	IGNITER FABRICATION	0.7	0.7	0.7	0.7
300	INSUL./INHIB./LINER FABRICATION	4.8	4.8	3.7	3.7
300	PROPELLANT FABRICATION	52.7	52.7	56.4	56.4
300	MOTOR FINISHING	6.7	6.7	7.7	7.7
300	SHIPMENT TO KSC	2.5	2.5	3.2	3.2
300	KSC ASSEMBLY/ERECTION	0.5	0.5	0.3	0.3
		100.0	100.8	105.0	105.8

Non-recurring Costs

Normalized non-recurring costs are summarized in Table 2.1.6 for the candidate design concepts. These costs have also been normalized such that 100% represents the total non-recurring cost of the baseline candidate.

Total Life Cycle Costs

Normalized total Life Cycle Costs are summarized in Table 2.1.7. Again, all costs have been normalized such that 100% represents the total Life Cycle Cost of the baseline candidate.

The cost ranking system discussed in Section 2.0 was used to compute cost rating scores. The segmented design using current case segments was assigned a rating score of seven since its total LCC is representative of the current configuration.

2.1.6 RANKING AND SELECTION OF PREFERRED DESIGN APPROACH

Table 2.1.8 summarizes the results of the Design Approach Trade Study. Rating scores from 1 to 10 (10=best) were derived for each candidate in the ranking criteria categories of reliability, cost, and performance as discussed in the preceding sections. These scores were multiplied by the appropriate weighting factors shown to obtain weighted scores. The weighted scores were then summed to obtain an overall score. As shown, the segmented design having longer case segments and current length casting segments had the highest score of all the candidates, and was therefore chosen as the preferred design concept approach.

2.2 SRM MOTOR CASE TRADE STUDY

The NASA space shuttle solid rocket motor (SRM) case was evaluated for the feasibility of fabricating casting segments from current (51-L type) length case segments or from one piece 320" and 326" long case segments. In this study, materials and processes were identified that could be used in either case configuration and a trade study was performed to define which material and which configuration was best suited for the rocket motor case. The configurations assessed are shown in Figure 2.2.1. The existing factory joint configuration made from two 160" case segments will be referred

TABLE 2.1.6. SRM NON-RECURRING COSTS.

	TOTAL NON-RECURRING COSTS, %			
	SEGMENTED		MONOLITHIC	
ELEMENT	CURRENT CASE SEGMENTS	LONGER CASE SEGMENTS	CURRENT CASE SEGMENTS	LONGER CASE SEGMENTS
SRM D&V PROGRAM	51.6	51.6	67.0	67.0
SRM MANUFACTURING FACILITY	45.7	45.7	69.6	69.6
SRM TRANSPORTATION SYSTEM	2.7	2.7	29.3	29.3
KSC ASSEMBLY/ERECTION FACILITY	0.0	0.0	6.3	6.3
	100.0	100.0	172.2	172.2

REV. A

TABLE 2.1.7. SRM LIFE CYCLE COSTS.

ELEMENT	TOTAL LCC, % SEGMENTED CURRENT CASE SEGMENTS	TOTAL LCC, % SEGMENTED LONGER CASE SEGMENTS	TOTAL LCC, % MONOLITHIC CURRENT CASE SEGMENTS	TOTAL LCC, % MONOLITHIC LONGER CASE SEGMENTS
RECURRING COSTS	78.3	78.9	82.2	82.7
NON-RECURRING COSTS	21.7	21.7	37.3	37.3
TOTAL LCC	100.0	100.6	119.5	120.0
RANKING SCORE*	**7**	**6.8**	**0.5**	**0.3**

[redacted]

TABLE 2.1.8. DESIGN APPROACH TRADE STUDY SUMMARY.

		SEGMENTED				MONOLITHIC			
		CURRENT CASE SEGMENTS		LONGER CASE SEGMENTS		CURRENT CASE SEGMENTS		LONGER CASE SEGMENTS	
CATEGORY	WEIGHT	RAW	WTD	RAW	WTD	RAW	WTD	RAW	WTD
RELIABILITY	0.65	8.8	5.72	8.9	5.79	7.3	4.75	7.4	4.81
COST	0.20	7.0	1.40	6.8	1.36	0.5	0.10	0.3	0.06
PERFORMANCE	0.15	7.2	1.08	7.6	1.14	8.4	1.26	8.7	1.31
TOTAL			8.20		8.29		6.11		6.18

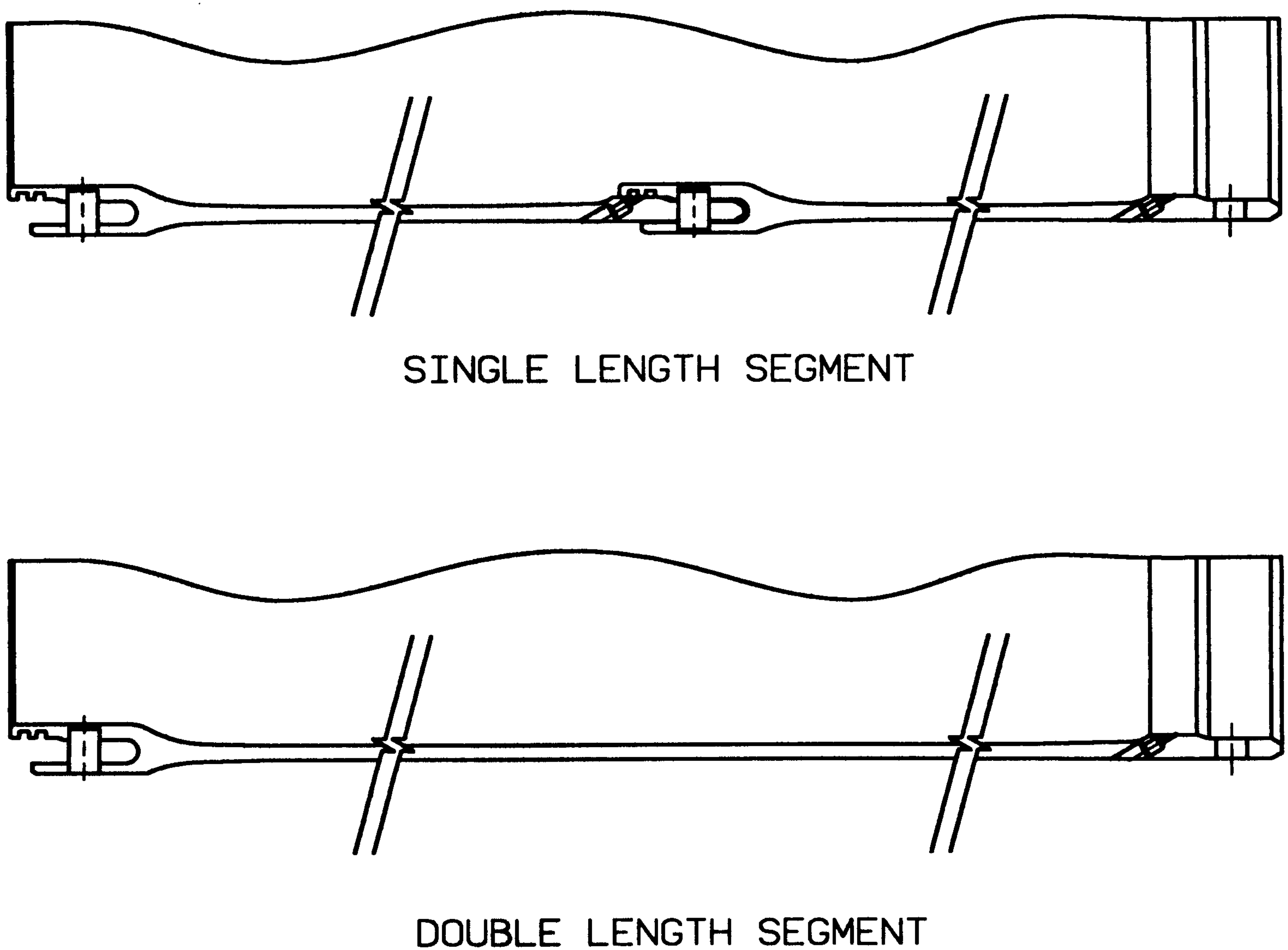


FIGURE 2.2.1. SRM CASE SEGMENT CONFIGURATION.

to as the single length configuration while the one piece 320" long case segment configuration will be named the double length configuration.

The double length case configuration is a viable concept for use in current casting segments. This length case can be fabricated as a monolithic structure or as a welded two piece case. The double length case would require no mechanical insulated joint or pressure seal and would focus the case integrity on a circumferential weld or on parent material properties for a monolithic double length case. The potential weight savings per joint would result in a weight savings of 690 lbs for each joint. This would allow for either increased payload or reallocation of weight to another area of the rocket motor assembly. The case could be proof tested to verify weld integrity and magnetic particle inspected or fluorescent penetrant inspected to examine the weld zone or parent material for cracking. The elimination of a mechanical joint would reduce the chance of saltwater corrosion and the possibility of stress corrosion cracking. The double length segment would eliminate potential rework associated with a single segment, mechanically pinned joint. The various features of the double length case configuration are addressed in Figure 2.2.2.

The current case material, D6AC, was originally selected for its superior strength in a non-welded configuration. The heat treatment level was controlled below the maximum strength level of the material to improve its toughness. At the time of selection, D6AC was a widely used rocket motor case material and possessed a large experience data base. The choice was good. Several other materials have emerged as dependable material candidates since the initial case selection and the trade off between a single length case segment and a double length case segment. These additional variables mandate a new investigation to determine the best material choices when applied to different manufacturing methods. Material properties and behavior combined with manufacturing process considerations will trade off to create the best case material and configuration.

The candidate materials are shown in Figure 2.2.3. Each material shown has production experience and is currently in inventory. The materials vary in ultimate tensile strength from 200,000 psi to 260,000 psi and can be downgraded in tensile strength to improve toughness. The chemical composition of these materials is shown in Figure 2.2.4. Several missile systems that use these steels are shown in Figure 2.2.5.

ADVANTAGES	DISADVANTAGES
WEIGHT SAVINGS OF 649 LBS PER FACTORY JOINT	INCREASED INSPECTION IF WELDED
NO DISASSEMBLY OF FACTORY JOINTS RESULTS IN HIGHER RELIABILITY AND LOWER REFURBISHMENT COST	CASTING LENGTH SEGMENT REQUIRES LARGER FORMING MACHINES AND RAW MATERIAL STOCK
LESS CHANCE OF SEAWATER DAMAGE TO JOINT	

FIGURE 2.2.2. ASSESSMENT OF DOUBLE LENGTH CASE SEGMENT.

D6AC

250 MARAGING

4330V

4340

300M

200 MARAGING

FIGURE 2.2.3. SRM CASE CANDIDATE MATERIALS.

MATERIAL CHEMICAL COMPOSITION*

MATERIAL	ELEMENT (% WT)									
	C	Mn	Si	Cr	Ni	Mo	V	Co	Ti	Al
D6ac	.48 .42	.90 .60	.30 .15	1.20 .90	.70 .40	1.10 .90	.10 .05	-	-	-
200 MARAGING	-	-	-	-	18	3.3	-	8.5	.20	.10
4340	.43 .38	.80 .60	.35 .20	.90 .70	2.00 1.65	.30 .20	-	-	-	-
4330 V	.33 .28	.85 .65	1.45	.90 .70	2.00 1.65	.30 .20	.10	-	-	-
250 MARAGING	-	-	-	-	18	5.0	-	8.5	.40	.10
300 M	.46 .40	.90 .65	1.80 1.45	.95 .70	2.00 1.65	.45 .30	.05	-	-	-

* BALANCE Fe

FIGURE 2.2.4. MATERIAL CHEMICAL COMPOSITION*

D6AC	TITAN III D	20+
	SHUTTLE SRM	2,500
	PATRIOT	1,900
	SRAM A	1,050
	HARM	200
	AMRAAM	700
	PHOENIX	
MARAGING STEELS	STINGER FLIGHT	18,000
	STINGER LAUNCH	18,000
	TOW	421,000
	DRAGON	220,000
	F111 ESCAPE GAS GENERATOR	1,300
4330 V	MK 106	1,180
4340	PEACEKEEPER STM	60
300 M	MK 104	1,000

FIGURE 2.2.5. MATERIAL HISTORY (9/86).

Most of these missile systems have welded construction in the rocket motor cases. Many have thin wall sections unlike the SRM case but the flaw sizes become more critical at the thinner wall sections.

The major criteria for analyzing suitability of a material for the SRM case application are tensile strength, strength to density ratio, overall body stiffness, fracture toughness, susceptability to stress corrosion cracking and the many manufacturing and handling considerations associated with each material. High tensile strength enables designing to thinner wall sections or combining lower strength with increased toughness for a given material. The various room temperature properties are tabulated in Figure 2.2.6 at representative strength levels. The stiffness of the rocket motor case is directly affected by the material's modulus of elasticity, the diameter and wall thickness of the case wall and the number of mechanically jointed case segments. By trading wall thickness, joint quantities and material properties, an optimum case material can be selected. Integral to this discussion, the effects of fracture mechanics must be incorporated to determine the effects of toughness on tensile strength and wall thickness and the capability to manufacture a case in the selected material. Corrosion effects play an important role in material selection because they affect material tensile strength and wall thickness required to satisfy damage tolerant properties. Selected damage tolerant properties of the candidate materials is shown in Figure 2.2.7.

Manufacture of the case segments in the single length and double length configurations are dependent on the material selected and the processes that are available to fabricate the desired form. Maraging steels are suitable for shear spinning over long lengths because they have little impact on the heat treatment facilities and are less likely to distort during heat treatment. Conversely, quench hardenable steels require close control over the heat treatment process and heat treatment of such a large length with low distortion may be impractical. Several large metal structure fabricators were contacted and asked to participate in a manufacturing study to assess the materials and forming processes that could be implemented in a SRM case. These contractors were asked to evaluate their capability to produce a case of the single and double length size and the relative risks associated with each process. These contractors are listed in Figure 2.2.8. Some observations about the various materials were made relative to existing technology and expertise and the following generalizations were made:

ALLOY	ULTIMATE STRENGTH (KSI)	YIELD STRENGTH (KSI)	ELONG-ATION	DENSITY (LB/CU IN)	TENSILE MODULUS (MSI)	ULTIMATE STRENGTH/ DENSITY/RATIO
C-200	210	200	8	.286	26.5	734,000
C-250	255	245	3	.286	26.5	891,600
300M	280	230	5	.283	29	989,400
4340	260	215	10	.283	29	918,400
4330V MOD	220	186	5	.283	29	777,400
D6AC**	195	180	8	.283	29	689,000

*REFERENCE: MIL-HDBK-5D

**NASA SRM CASE MATERIAL SPECIFICATION

FIGURE 2.2.6. ROOM TEMPERATURE MECHANICAL PROPERTIES.*

ALLOY	PRODUCT FORM	TYS (KSI)	PLANE STRAIN THICKNESS (IN)	AVG, L-T K (KSI-$\sqrt{IN}$)	3.5% N CL K (KSI $\sqrt{IN}$)
C 200	PLATE	215	.48	145.0	70.0
C 250	PLATE	249	.48	92.0	40.5
300 M	PLATE	230	.50	62.5	13.6
4340	PLATE	194	.50	72.2	16.8
4300 V	PLATE	196	.48	103.0	25.0
D6AC	PLATE	217	.48	64.0	(~30)

•REFERENCE: DAMAGE TOLERANCE DESIGN HANDBOOK TEST DATA RESULTS

FIGURE 2.2.7. DAMAGE TOLERANCE PROPERTIES.*

AMERICAN WELDING INSTITUTE	KNOXVILLE, TN
BABCOCK & WILCOX	LYNCHBURG, VA
BATH IRON WORKS	BATH, ME
AUTOSPIN	CARSON, CA
LADISH CO.	CUDAHY, WI
LITTON	PASCAGULA, MS
NEWPORT NEWS	NEWPORT NEWS, VA
RANOR	WESTMINSTER, MA
ROHR INDUSTRIES	CHULA VISTA, CA
SANDUSKY CASTING	SANDUSKY, OH
DEFENSE METALS INFORMATION CENTER -BATTELLE	COLUMBUS, OH
WESTINGHOUSE	PENSACOLA, FL

FIGURE 2.2.8. SRM CASE MANUFACTURING SURVEY.

- Maraging steels offered lower distortion due to the heat treatment cycle;

- Quench hardenable steels were slightly easier to machine;

- Maraging steels were simpler to heat treat;

- Welding was easily performed on maraging steel. Welding carbon steels was more difficult.

Materials could be obtained in various forms as shown in Figure 2.2.9. Large forgings, similar to the existing forgings used for the single case segment, could be roll forged into a ring in preparation for further reduction in area at subsequent processing stages. The same forging could be upset and spin forged against a roller die to net the same case. Initial formation of the cylindrical stock could be performed by a cylindrical casting of the low alloy steels. D6AC and the maraging steels would not be suitable for this process since a vacuum arc remelt is required for these alloys and alloy segregation may occur in the maraging steel. Rolled and welded sheet stock, the mainstay of the aerospace rocket motor industry, is the final form. A thicker rolled and welded cylinder could be manufactured and subsequent forming operations could be employed to reduce the wall thickness of the case. This process would also planish the weld area and decrease weld effects at the longitudinal seam. The candidate materials and processes are tabulated in Figure 2.2.10 and show the relation of material to process.

The heat treatments for the material candidates fall into two categories: quench and temper for D6AC, 4340, 4330V and 300M and maraging for 200 and 250 maraging steel. A typical heat treatment cycle is shown in Figure 2.2.11. The major differences between the two types of material heat treat cycles are the number of heat treatment steps and the severity of temperature fluctuation. In the quench hardenable steels, several steps are necessary to obtain final physical properties. During the process, the materials are subjected to severe changes in temperature over short time periods which are necessary to harden the materials but these temperature shocks have a tendency to distort the motor case. The large size of the SRM case is nonconducive to maintaining roundness and straightness during the severe heat treat cycles. With maraging steels, only two steps are required to heat treat the materials to final strength levels. The aging process requires a low temperature and a short time to obtain the final physical properties. An intermediate working process is possible by forming the material

FORGING

- RING ROLL FORGING
- SPIN FORGING

CASTING

- CENTRIFUGAL CASTING

SHEET

- ROLL AND WELD

FIGURE 2.2.9. SRM CASE RAW MATERIAL FORM.

	D6AC	200 MARAGING	4330 V	4340	300M	250 MARAGING
ROLL FORGE	Y	Y	Y	Y	Y	Y
ROLL AND WELD	Y	Y	Y	Y	Y	Y
CENTRIFUGAL CAST	N	N	Y	Y	N	N
SHEAR SPIN	Y	Y	Y	Y	Y	Y

Y = YES; N = NO

FIGURE 2.2.10. MOTORCASE MANUFACTURING TECHNIQUES.

PROCESS STEP	D6AC	250 MARAGING	4330V	4340	300M	200 MARAGING
NORMALIZE	1700° F A.C.	N/A	1650° F A.C.	1650° F A.C.	1700° F A.C.	N/A
SOLUTION ANNEAL	N/A	1500° F	N/A	N/A	N/A	1500° F
AUSTEN-ITIZE	1650° F	N/A	1600° F	1500° F	1600° F	N/A
SALT QUENCH	475° F SALT	N/A	400° F	400° F	400° F	N/A
OIL QUENCH	160° F MAX	N/A	160° F	160° F	160° F	N/A
DOUBLE TEMPER (APPROX-IMATE)	1100° F	N/A	850° F	900° F	900° F	N/A
MARAGING	N/A	900° F	N/A	N/A	N/A	900° F
TOTAL STEPS	5	2	5	5	5	2

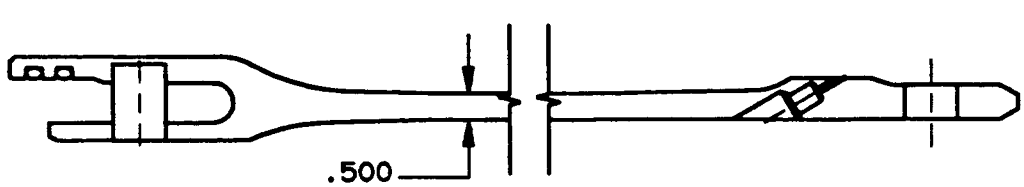


FIGURE 2.2.11. MOTOR CASE PROCESSING TYPICAL HEAT TREATMENT.

after the solution anneal. Excellent dimensional stability is possible with maraging steel. Through hardenability is excellent on the higher carbon quench hardenable steels and with both alloys of maraging steel.

Weldability of the various alloys and the resultant physical properties are significant factors for welded case construction. While all materials are weldable, different processes must be applied to weld the different materials in the defined thickness. Joint configuration varies depending on the welding process used on the material. Welding speeds and heat input must be controllable to ensure a repeatable process. Preheat and postheat temperature along with weld wire selection can tailor the weld joint to specific physical properties. Non-destructive testing methods must give accurate data about the weld joint with high reliability. Depending upon the welding method, fixturing of the motor case segments can define the final product integrity so care must be exercised when designing weld fixturing. Refurbishment of the case becomes a factor with a welded joint so the final joint design must provide for protection from corrosion by controlling the pre-weld fit-up and allowing for material removal after welding to eliminate gaps.

Some welding techniques that were considered are: electron beam (EB), laser beam (LB), tungsten/inert gas (TIG) and metal/inert gas (MIG). All four processes can be automated and tailored to weld the candidate materials. Laser and electron beam welds are suitable for all materials providing the joints can be controlled to minimize fit-up gaps. The TIG and MIG are welds are suitable for the maraging steels. The low alloy steels can be welded prior to heat treatment but distortion cannot be controlled as closely as welding after heat treatment. The low heat treatment levels with maraging steels enables good physical distortion control and the material can be welded prior to solution anneal or prior to aging. The best properties are obtained by welding prior to solution anneal. A typical flow chart of the welding process is shown in Figure 2.2.12. Typical joint configurations are shown in Figure 2.2.13.

Trade studies were conducted that evaluated all of the features previously discussed that pertain to the SRM case configuration, material and forming process. The trade study involved a numerical representation of the tangible aspects of reliability, payload weight and cost impact as well as the perceived values associated with materials and processes. The scoring criteria used for this trade study were as given in Section 2.0. Figures 2.2.14 through 2.2.16 are the tabulations of the numerical values assigned to the candidate materials in several manufacturing processes.

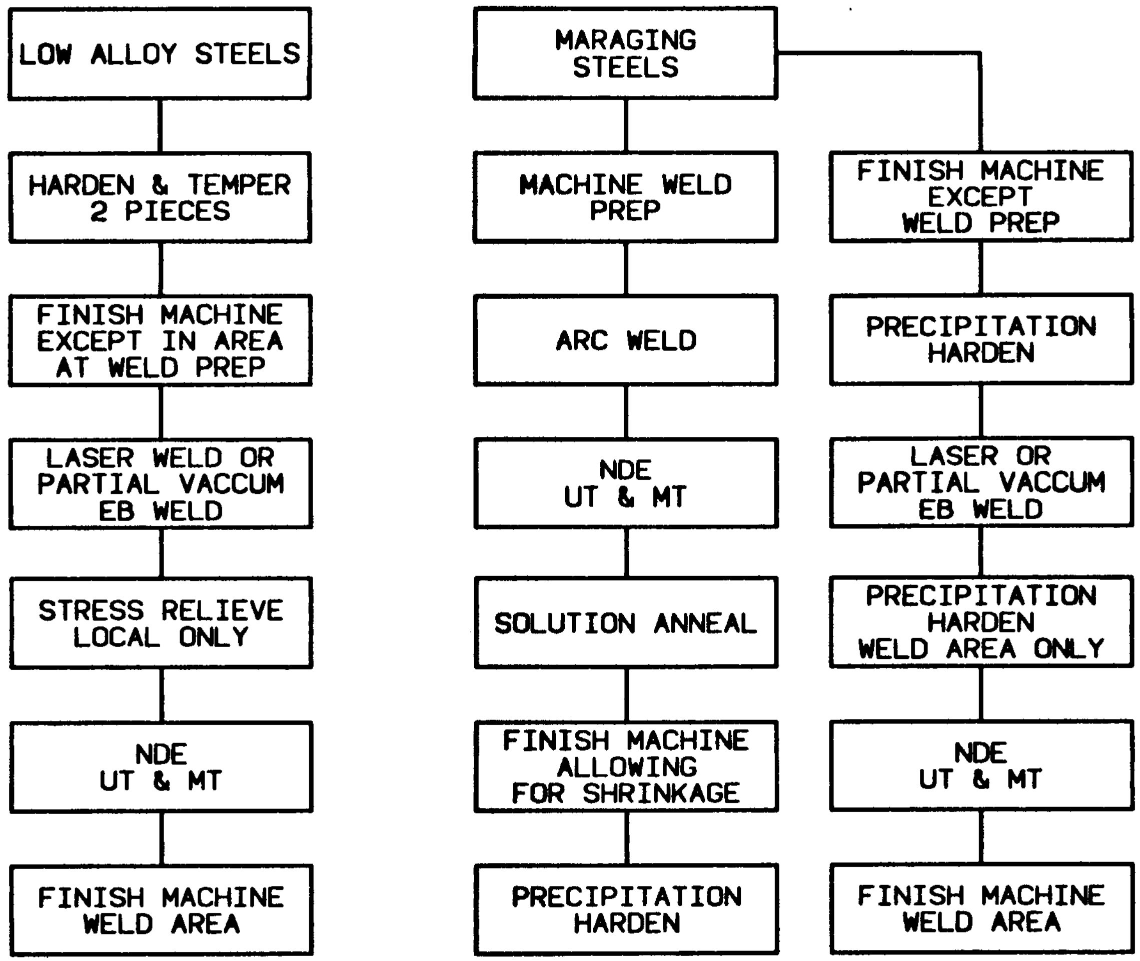
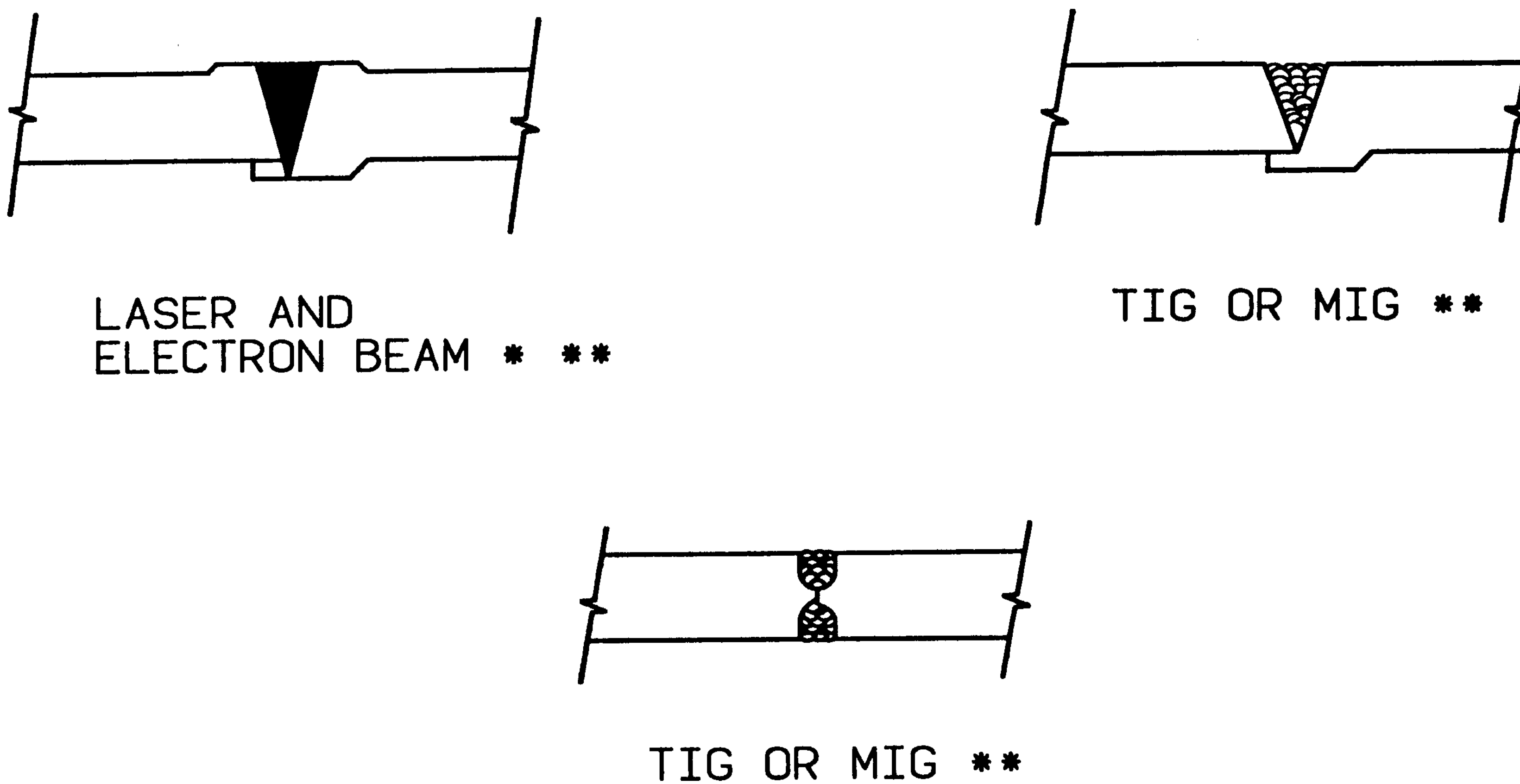


FIGURE 2.2.12. WELDING PROCESS.



* JOINT FIT-UP GAP CONTROLLED TO .000-.007

** JOINT PREHEAT WITH RESISTANCE HEATERS

FIGURE 2.2.13. JOINT PREP.

SINGLE CASE SEGMENT NO MIDCASE CIRCUMFERENTIAL WELD		RELIABILITY					PAYLOAD WEIGHT			COST					
		FLIGHT	DESIGN	MANUFACTURE	COROSION	AVERAGE	INERT	PROPELLANT	AVERAGE	REFURBISHMENT	HARDWARE	SUBSEQUENT MANUFACTURING	TRANSPORTATION	FACILITIES	AVERAGE
ROLL FORGE	D6AC	5	5	5	5	5.00	7	7	7.00	7	7	7	7	7	7.00
	200 MARAGING	7	7	6	5	6.25	7	7	7.00	7	5	7	7	7	6.60
	4330 V	5	6	5	5	5.25	7	7	7.00	7	7	7	7	7	7.00
	4340	5	5	4	4	4.50	8	8	8.00	7	7	7	7	7	7.00
	300 M	6	4	4	4	4.50	9	9	9.00	7	4	7	7	7	6.40
	250 MARAGING	6	7	6	5	6.00	8	8	8.00	7	5	7	7	7	6.60
ROLL AND WELD	D6AC	4	4	6	4	4.50	7	7	7.00	7	8	7	7	7	7.20
	200 MARAGING	4	5	7	4	5.00	7	7	7.00	7	8	7	7	7	7.20
	4330 V	4	4	6	4	4.50	7	7	7.00	7	8	7	7	7	7.20
	4340	3	3	5	3	3.50	8	8	8.00	7	8	7	7	7	7.20
	300 M	2	3	5	3	3.25	9	9	9.00	7	7	7	7	7	7.00
	250 MARAGING	4	5	7	4	5.00	8	8	8.00	7	8	7	7	7	7.20
CENTRIFUGAL CAST	4330 V	2	5	5	4	4.00	7	7	7.00	8	7	7	7	7	7.20
	4340	2	5	5	4	4.00	8	8	8.00	8	7	7	7	7	7.20
SHEAR SPIN	D6AC	5	5	5	5	5.00	7	7	7.00	7	7	7	7	7	7.00
	200 MARAGING	7	7	6	5	6.25	7	7	7.00	7	5	7	7	7	6.60
	4330 V	5	6	5	5	5.25	7	7	7.00	7	7	7	7	7	7.00
	4340	5	5	4	4	4.50	8	8	8.00	7	7	7	7	7	7.00
	300 M	6	4	4	4	4.50	9	9	9.00	7	5	7	7	7	6.60
	250 MARAGING	6	7	6	5	6.00	8	8	8.00	7	5	7	7	7	6.60

FIGURE 2.2.14. SINGLE CASE SEGMENT - NO MIDCASE CIRCUMFERENTIAL WELD.

DOUBLE LENGTH CASE SEGMENT — NO MIDCASE CIRCUMFERENTIAL WELD		RELIABILITY					PAYLOAD WEIGHT			COST					
		FLIGHT	DESIGN	MANUFACTURE	COROSION	AVERAGE	INERT	PROPELLANT	AVERAGE	REFURBISHMENT	HARDWARE	SUBSEQUENT MANUFACTURING	TRANSPORTATION	FACILITIES	AVERAGE
ROLL FORGE	D6AC	9	8	9	6	8.00	8	8	8.00	8	7	8	6	7	7.20
	200 MARAGING	9	8	8	6	7.75	8	8	8.00	8	6	8	6	7	7.00
	4330 V	9	7	8	6	7.50	8	8	8.00	8	7	8	6	7	7.20
	4340	9	7	7	5	7.00	9	9	9.00	8	6	8	6	7	7.00
	300 M	9	6	7	5	6.75	10	10	10.00	8	6	8	6	7	7.00
	250 MARAGING	9	8	8	6	7.75	9	9	9.00	8	6	8	6	7	7.00
ROLL AND WELD	D6AC	4	4	6	5	4.75	8	8	8.00	7	8	8	6	7	7.20
	200 MARAGING	4	5	7	5	5.25	8	8	8.00	7	7	8	6	7	7.00
	4330 V	4	4	7	5	5.00	8	8	8.00	7	8	8	6	7	7.20
	4340	4	3	6	4	4.25	9	9	9.00	7	7	8	6	7	7.00
	300 M	4	3	5	4	4.00	10	10	10.00	7	7	8	6	7	7.00
	250 MARAGING	4	5	7	5	5.25	9	9	9.00	7	7	8	6	7	7.00
CENTRIFUGAL CAST	4330 V	3	4	6	4	4.25	8	8	8.00	8	6	7	6	7	6.80
	4340	3	4	6	4	4.25	9	9	9.00	8	6	7	6	7	6.80
SHEAR SPIN	D6AC	9	8	9	6	8.00	8	8	8.00	8	6	8	6	7	7.00
	200 MARAGING	9	8	8	6	7.75	8	8	8.00	8	4	8	6	7	6.60
	4330 V	9	7	8	6	7.50	8	8	8.00	8	6	8	6	7	7.00
	4340	9	7	7	5	7.00	9	9	9.00	8	6	8	6	7	7.00
	300 M	9	6	7	5	6.75	10	10	10.00	8	5	8	6	7	6.80
	250 MARAGING	9	8	8	6	7.75	9	9	9.00	8	4	8	6	7	6.60

FIGURE 2.2.15. DOUBLE LENGTH CASE SEGMENT - NO MIDCASE CIRCUMFERENTIAL WELD.

DOUBLE LENGTH CASE SEGMENT		RELIABILITY					PAYLOAD WEIGHT			COST					
WITH MIDCASE CIRCUMFERENTIAL WELD		FLIGHT	DESIGN	MANUFACTURE	COROSION	AVERAGE	INERT	PROPELLANT	AVERAGE	REFURBISHMENT	HARDWARE	SUBSEQUENT MANUFACTURING	TRANSPORTATION	FACILITIES	AVERAGE
ROLL FORGE	D6AC	8	8	9	5	7.50	8	8	8.00	8	8	8	6	7	7.40
	200 MARAGING	9	8	8	5	7.50	8	8	8.00	8	8	9	6	7	7.60
	4330 V	8	8	8	5	7.50	8	8	8.00	8	8	8	6	7	7.40
	4340	8	8	7	4	6.75	9	9	9.00	8	8	8	6	7	7.40
	300 M	8	6	7	4	6.25	10	10	10.00	8	7	8	6	7	7.20
	250 MARAGING	8	8	9	5	7.50	9	9	9.00	8	8	9	6	7	7.60
ROLL AND WELD	D6AC	3	4	6	4	4.25	8	8	8.00	7	9	8	6	7	7.40
	200 MARAGING	3	5	7	4	4.75	8	8	8.00	7	9	9	6	7	7.60
	4330 V	3	4	7	4	4.50	8	8	8.00	7	9	8	6	7	7.60
	4340	3	3	6	3	3.75	9	9	9.00	7	9	8	6	7	7.60
	300 M	3	3	5	3	3.50	10	10	10.00	7	7	8	6	7	7.00
	250 MARAGING	3	5	7	4	4.75	9	9	9.00	7	9	9	6	7	7.60
CENTRIFUGAL CAST	4330 V	3	3	5	4	3.75	8	8	8.00	8	6	7	6	7	6.80
	4340	2	3	5	4	3.55	9	9	9.00	8	6	7	6	7	6.80
SHEAR SPIN	D6AC	8	8	9	5	7.50	8	8	8.00	8	7	8	6	6	7.00
	200 MARAGING	9	8	8	5	7.50	8	8	8.00	8	8	9	6	6	7.40
	4330 V	8	8	8	5	7.50	8	8	8.00	8	7	8	6	6	7.00
	4340	8	8	7	4	6.75	9	9	9.00	8	7	8	6	6	7.00
	300 M	8	6	7	4	6.25	10	10	10.00	8	7	8	6	6	7.00
	250 MARAGING	8	8	9	5	7.50	9	9	9.00	8	8	9	6	6	7.40

FIGURE 2.2.16. DOUBLE LENGTH CASE SEGMENT - WITH MIDCASE CIRCUMFERENTIAL WELD.

Reliability was subdivided into four subcategories. Flight reliability refers to the actual usage of the material. Design reliability refers to the ability to analyze and design a reliable motor given the constraints of the processes and materials. Manufacturing reliability is the degree to which a design can be produced in production. Corrosion reliability addresses the long term effects of corrosion on reliability.

Payload weight addresses the inert and propellant weight of the various materials and processes and the possible gains or losses that can be attained with each candidate.

Cost is subdivided into five subcategories to identify the greatest affected areas. Refurbishment costs identify the impact on cost after motor retrieval until new insulation is applied. Hardware cost is the cost of a casting length case and subsequent manufacturing refers to the pinning, o-ring installation and insulation that is applied on the existing segment. Transportation and facility impact compares increased or decreased costs on the current single length case at process points subsequent to insulation.

The weighted composite scores are tabulated in Figures 2.2.17 through 2.2.19. The selected concept based on these trade studies is the double length case fabricated from D6AC steel with no circumferential weld. The next alternative is the double length case segment fabricated from 250 maraging steel with no welds.

2.3 JOINTS AND SEALS TRADE STUDIES

Philosophy

The selection of a joint-seal design was driven by one underlying goal: namely, to improve the SRB field joint reliability of the system so that there is virtually no possibility of failure. Still, the design has to be manufacturable at a reasonable cost, and it must not reduce the payload capacity by an excessive amount. Thus, a ranking formula was devised which weights reliability at 65 percent, payload at 15 percent, and cost at 20 percent, i.e.,

$$R = .65\ F_R + .15\ F_P + .20\ F_C,$$

where R is the ranking fraction and F_R, F_P, and F_C are rating fractions for reliability,

		RELIABILITY	PAYLOAD WEIGHT	COST	TOTAL WEIGHTED SCORE
ROLL FORGE	D6AC	3.250	1.050	1.400	5.700
	200 MARAGING	4.062	1.050	1.320	6.432
	4330 V	3.412	1.050	1.400	5.862
	4340	2.925	1.200	1.400	5.525
	300 M	2.925	1.350	1.280	5.555
	250 MARAGING	3.900	1.200	1.320	6.420
ROLL AND WELD	D6AC	2.925	1.050	1.440	5.415
	200 MARAGING	3.250	1.050	1.440	5.740
	4330 V	2.925	1.050	1.440	5.415
	4340	2.275	1.200	1.440	4.915
	300 M	2.112	1.350	1.400	4.862
	250 MARAGING	3.250	1.200	1.440	5.890
CENTRIFUGAL CAST	4330 V	2.600	1.050	1.440	5.090
	4340	2.600	1.200	1.440	5.240
SHEAR SPIN	D6AC	3.250	1.050	1.400	5.700
	200 MARAGING	4.062	1.050	1.320	6.432
	4330 V	3.412	1.050	1.400	5.862
	4340	2.925	1.200	1.400	5.525
	300 M	2.925	1.350	1.320	5.555
	250 MARAGING	3.900	1.200	1.320	6.420

FIGURE 2.2.17. SINGLE LENGTH CASE SEGMENT - NO MIDCASE CIRCUMFERENTIAL WELD.

		RELIABILITY	PAYLOAD WEIGHT	COST	TOTAL WEIGHTED SCORE
ROLL FORGE	D6AC	5.200	1.200	1.440	7.840
	200 MARAGING	5.037	1.200	1.400	7.637
	4330 V	4.875	1.200	1.440	7.515
	4340	4.550	1.350	1.400	7.300
	300 M	4.387	1.500	1.400	7.275
	250 MARAGING	5.037	1.350	1.400	7.787
ROLL AND WELD	D6AC	3.087	1.200	1.440	5.727
	200 MARAGING	3.412	1.200	1.400	6.012
	4330 V	3.250	1.200	1.440	5.890
	4340	2.762	1.350	1.400	5.512
	300 M	2.600	1.500	1.400	5.500
	250 MARAGING	3.412	1.350	1.400	6.162
CENTRIFUGAL CAST	4330 V	2.762	1.200	1.360	5.322
	4340	2.762	1.350	1.360	5.472
SHEAR SPIN	D6AC	5.200	1.200	1.400	7.800
	200 MARAGING	5.037	1.200	1.320	7.557
	4330 V	4.875	1.200	1.400	7.395
	4340	4.550	1.350	1.400	7.300
	300 M	4.387	1.500	1.360	7.247
	250 MARAGING	5.037	1.200	1.320	7.557

FIGURE 2.2.18. DOUBLE LENGTH CASE SEGMENT - NO MIDCASE CIRCUMFERENTIAL WELD.

		RELIABILITY	PAYLOAD WEIGHT	COST	TOTAL WEIGHTED SCORE
ROLL FORGE	D6AC	4.875	1.200	1.480	7.555
	200 MARAGING	4.875	1.200	1.520	7.595
	4330 V	4.875	1.200	1.480	7.555
	4340	4.387	1.350	1.480	7.217
	300 M	4.062	1.500	1.440	7.002
	250 MARAGING	4.875	1.350	1.520	7.745
ROLL AND WELD	D6AC	2.762	1.200	1.480	5.442
	200 MARAGING	3.087	1.200	1.520	5.807
	4330 V	2.925	1.200	1.520	5.645
	4340	2.437	1.350	1.520	5.307
	300 M	2.275	1.500	1.400	5.175
	250 MARAGING	3.087	1.350	1.520	5.957
CENTRIFUGAL CAST	4330 V	2.437	1.200	1.360	4.997
	4340	2.275	1.350	1.360	4.985
SHEAR SPIN	D6AC	4.875	1.200	1.400	7.475
	200 MARAGING	4.875	1.200	1.480	7.555
	4330 V	4.875	1.200	1.400	7.475
	4340	4.387	1.350	1.400	7.137
	300 M	4.062	1.500	1.400	6.962
	250 MARAGING	4.875	1.350	1.480	7.705

FIGURE 2.2.19. DOUBLE LENGTH CASE SEGMENT - WITH MIDCASE CIRCUMFERENTIAL WELD.

payload change, and cost, respectively, and are all based on a value between 1 and 10 where 10=best. The rating and ranking calculations are illustrated further into the report. The rating criteria are defined in Table 2.3.1.

Joint Selection and Trade Features

Five joint-seal designs were chosen for this trade study. These candidates are listed in Table 2.3.2 along with seal options. Seal characteristics are listed in Table 2.3.3. Sketches are shown in Figures 2.3.1 and 2.3.2. To accomplish these selections, we first must consider the undesirable features of the 51-L original tang and clevis design. The most important of these are:

1. The gap between the o-ring sealing surfaces tended to open during motor pressurization.

2. The primary o-ring may not function (fill the gap) quickly enough if it is exposed to a low temperature environment, and it might be out of position due to back pressure from a leak check.

3. If the insulation is breached and there is a flow path to the o-ring and circumferentially around the o-ring, the thermal resistance of an elastomeric o-ring is poor, since rubber and plastic o-ring materials deteriorate in the neighborhood of 350°F.

   We note here that the insulation design is an extremely important part of the overall field joint system and is discussed under Section 3.4. This trade study is limited to the metallic case joints and seals.

4. The joint is difficult to evaluate by structural analysis, since tolerances involving the pin-pin hole fit, the tang and clevis fit, shim fit, and out of roundness, as well as the affects of pin-pin hole friction, call for a myriad of assumptions. The Langley Research Center did an admirable job analyzing the original design (Reference 1, Section 3.1) but the above objections were apparent. Additionally, this analysis indicated probable yielding of the pins and tang and clevis pin holes, which further complicates the evaluation of the 51-L joint design.

The concept of a bolted, flat seating joint using face seals was ARC's primary consideration to overcome the above mentioned undesirable features:

TABLE 2.3.1. TRADE STUDY CRITERIA.

- RELIABILITY
  - ANALYSIS TRACTABILITY - PREDICTABILITY
    - OVERALL PRESSURE VESSEL PARTS
    - FASTENERS
    - SEALS AND GAPS
  - THERMAL RESISTANCE (IF INSULATION BREACHED)
    - PRELIMINARY AND SECONDARY SEAL RESISTANCE
    - FLANGE GAP AREA (CIRCUMFERENTIAL FLOW)
    - SEAL SEATING GROOVE FLOW AREA
    - EFFECT OF COLD TEMPERATURE
  - JOINT MECHANICS/STRESS STATES IN JOINT SHELL AND FLANGES
    - PEAK STRESSES AROUND FASTENER HOLES
    - FASTNER STRESSES
    - SEAL GAP OPENING
    - EASE OF SEAL INSTALLATION AND SEGMENT ASSEMBLY
    - REUSE DAMAGE AND RISK
- COST
  - MANUFACTURABILITY
  - ASSEMBLY/DISASSEMBLY
- PERFORMANCE
  - JOINT WEIGHT GAINED WITH RESPECT TO 51-L BASELINE
  - PROPELLANT WEIGHT LOST WITH RESPECT TO 51-L BASELINE

TABLE 2.3.2. JOINTS AND SEALS DESIGN SUMMARY.

METAL JOINT CANDIDATES	SEALS
51-L ORIGINAL DESIGN	FACE SEALS
NASA/MTC CAPTURE FEATURE	ELASTOMERIC
CAPTURE FEATURE WITH FACE SEAL	METAL C-RINGS
ARC BOLTED JOINT	METAL GASKET/GRAFOIL FILLER
ANGLE BOLTED JOINT	BORE SEALS
MODIFIED LARC IN-LINE BOLTED JOINT	ELASTOMERIC

TABLE 2.3.3. SEAL CHARACTERISTICS.

- SEALS ARE TRADED WITH RESPECT TO METAL JOINT TYPE

- GENERAL CHARACTERISTICS

	TYPE	MAXIMUM ALLOWABLE TEMPERATURE °F	MAXIMUM ALLOWABLE GAP (IN)	RESILIENCY	DAMAGE TOLERANCE
ELASTOMERIC	FACE/BORE	350	.013	HIGHLY TEMPERATURE DEPENDENT	LOW
METAL C-RING	FACE	3000	.013	UNCHANGED OVER TEMPERATURE RANGE	HIGH
METAL COMPOSITE GASKET	FACE	3000	.012	UNCHANGED OVER TEMPERATURE RANGE	HIGH

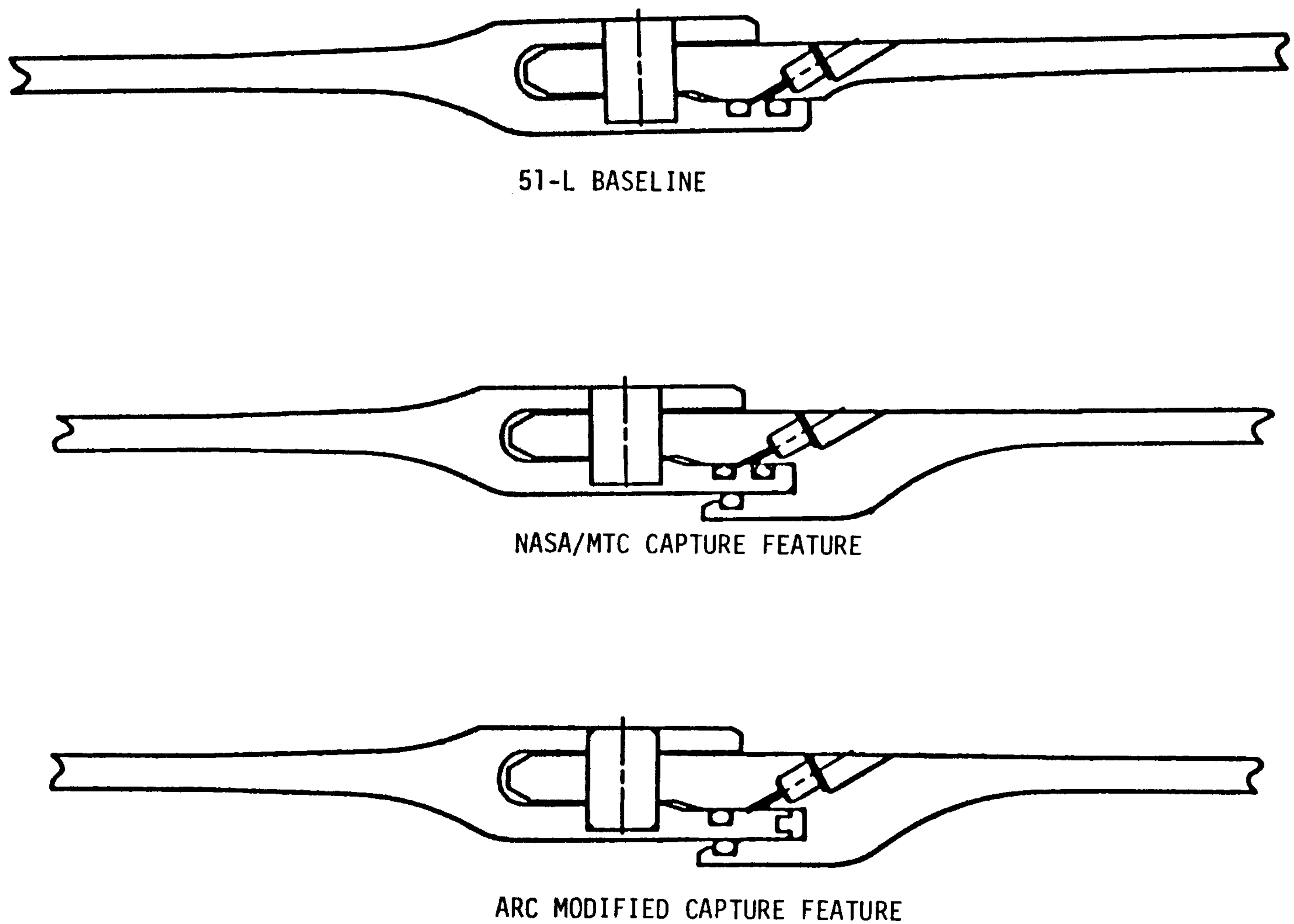


FIGURE 2.3.1. TANG AND CLEVIS JOINT CANDIDATES.

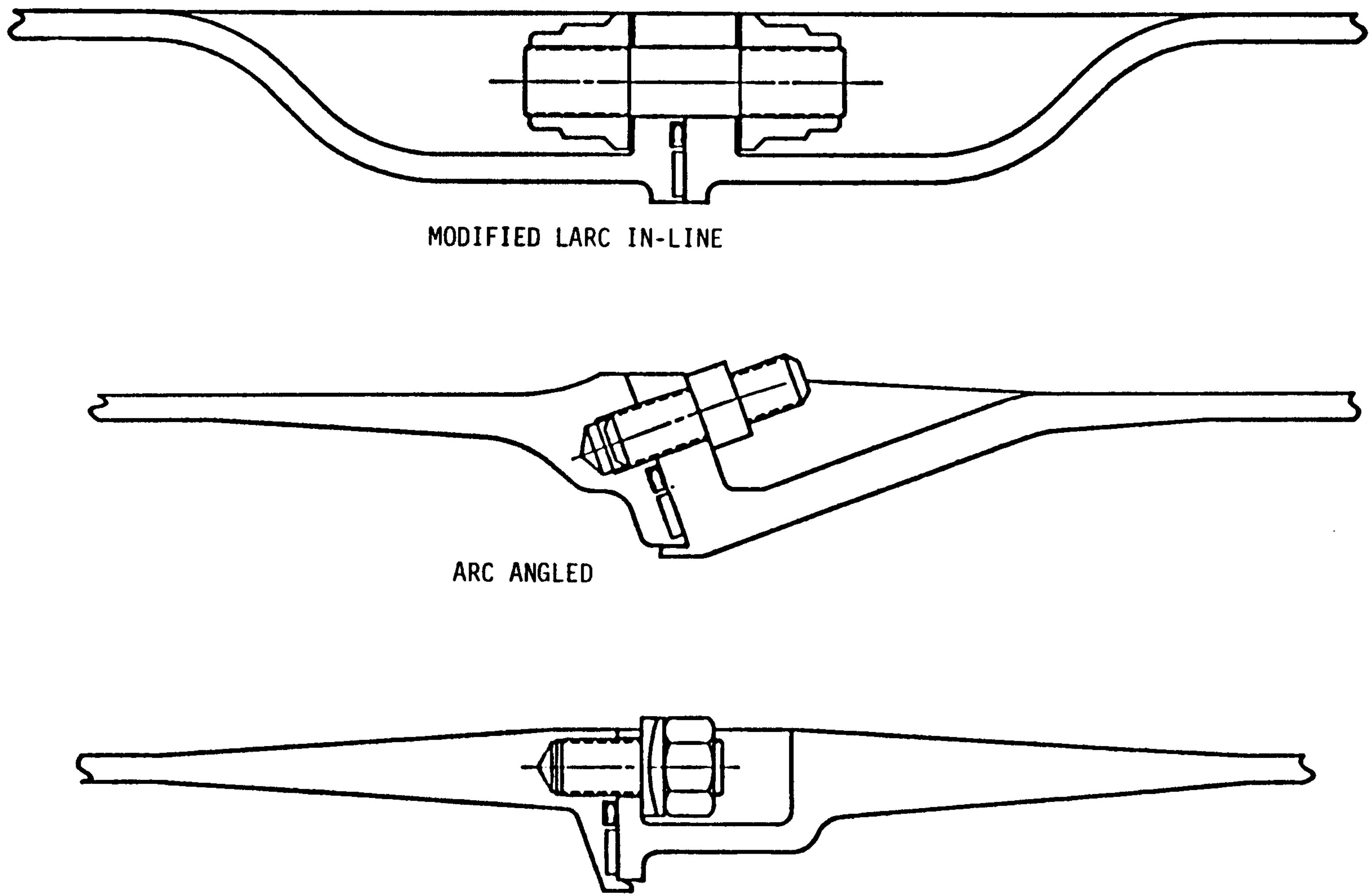


FIGURE 2.3.2. BOLTED JOINT CANDIDATES.

1. The flat seating surfaces make possible the use of metallic seals. A jacketed Inconel material with a graphite foil filler makes a perfect static seal that is not effected by leak check back pressure and is resistant to temperatures of over 3000°F. Metallic (Inconel) c-rings were choosen for the back-up or secondary seals, and will have the same high temperature resistance, while taking up less flange width.

2. The flat seating surfaces and horizontal seal grooves provide easy installation of the seals, and more importantly, easy inspection. Mating the upper segment will not scratch the seals or surfaces and will not lead to metal shavings or other debris.

3. The metallic gaskets are not pressure activated and are not sensitive to low environmental temperatures. Furthermore, both the metal gasket and the c-ring design have been used extensively in the steam generation industry (nuclear and conventional) to contain high pressure and high temperature steam vapor.

4. Finally, the bolted joint design is analytically tractable, and as such, the flange opening and state-of-stress can be kept to acceptable limits by engineering analysis that is not subject to assumptions or other guesswork. The first-cut design analysis is presented in Section 3.1.

In October of 1986, ARC was fortunate to receive a presentation by LARC of their in-line bolted joint. The joint design is very similar to the ARC design and was an obvious choice for one of the prime candidates in the trade study. One advantageous feature is the lack of threaded stud holes, which, in the case of the ARC design, would require inspection for reuse. We note that, for reasons mentioned above, the LARC in-line joint has been rated using the metal gasket and c-ring seal combination of the ARC design. This is presently thought to be a more reliable combination than the original LARC Viton o-ring and metal c-ring combination.

We note that both the ARC axially bolted joint and the LARC in-line bolted joint are receiving design structural analysis iterations and are not, at present, optimal. The LARC design received a slightly higher ranking, presented subsequently.

ARC has also included an angle bolted joint for consideration. This joint is intended to retain the main features of the axially bolted design, but it allows the additional feature of hydraulically pre-tensioned studs. Since pretensioning is essential to the bolted design (to minimize flange opening), it was decided to include this configuration as a backup. This is for the eventuality that stud elongation methods should fall short as a stud or bolt preloading scheme.

The NASA/MTC capture feature was also included in the trade study, since it has (a) been chosen for the 1988 SRB, (b) been extensively scrutinized, and (c) received a good structural analysis study by LARC (Reference 1, Section 3.1). The design is thought to serve as a good baseline for these studies, i.e., it should rank lower than the so-called clean paper designs.

The final subject of this trade study is the capture-feature concept with an elastomeric face seal between the capture feature lip and the inner clevis arm. This addition is thought to add reliability to the o-ring arrangement, since it would have the characteristics of a static seal although not the temperature resistance.

Trade Study Results

The ARC axially bolted joint was structurally analyzed in detail via the NASTRAN finite element method, and the results are presented in Section 3.1 of the Design Studies. Since this analysis covered only the "first cut" configuration, and since the results show a certain amount of overstressing and excess gap opening, the model weight was increased by 20 percent, and the propellant loss by 10 percent for the trade studies.

The NASA/Langley structural analyses (References 1 and 2, Section 3.1) were used heavily for the evaluation of the "capture feature" tang and clevis joint and the LARC "in-line bolted joint." The ARC "angle bolt joint" was evaluated by estimate only, since no detailed structural analysis was performed.

The trade study rating criteria, ratings, and explanatory comments are given for all of the joints in Tables 2.3.4 through 2.3.8. We note that reliability ratings are defined on a scale of 1 to 10. The final reliability factor is defined as the average of the individual ratings, and is in no way related to the probability of failure from classical reliability engineering. The same averaging is performed for the manufacturability and cost criteria. The payload weight factor is normalized to a 60,000 pound payload, and as such it represents the percent of payload lost or gained due to the presence of redesigned joints. The weight factors are summarized in Table 2.3.9. The absolute inert and propellant weight changes are shown in Table 2.3.10.

TABLE 2.3.4. TRADE STUDY RATING - NASA/MTC CAPTURE FEATURE.

RATING CRITERION	RATING	COMMENTS
RELIABILITY		
1. ANALYSIS TRACTABILITY - PREDICTABILITY		
A. OVERALL PRESSURE VESSEL PARTS	10	THOROUGH ANALYSIS BY NASA/LANGLEY AND ARC.
B. FASTENERS	6	ANALYSIS BY NASA/LANGLEY SHOWS SENSITIVITY TO PIN POSITIONING AND POSSIBLE YIELDING OF PIN AND YIELDING AROUND TANG AND CLEVIS PIN HOLES.
C. SEALS AND GAPS	8	NASA/LANGLEY ANALYSIS SHOWS SMALL GAP OPENING WITH ASSUMED INTERFERENCE FIT OF CAPTURE FEATURE.
2. THERMAL RESISTANCE (IF INSULATION BREACHED)		
A. PRIMARY AND SECONDARY SEAL RESISTANCE	4	RUBBER O-RINGS GOOD ONLY TO 350°F.
B. FLANGE GAP AREA (CIRCUMFERENTIAL FLOW)	8	GAP FLOW MINIMIZED BY CAPTURE FEATURE.
C. SEAL SEATING GROOVE FLOW AREA	8	SOME AREA OPEN DUE TO SEATING OF O-RING(S).
D. EFFECT OF COLD TEMPERATURE ON SEALING ABILITY	9	O-RINGS STAY WARM DUE TO STRIP HEATERS.
3. JOINT MECHANICS - STRESS STATES		
A. STRESS STATE IN JOINT SHELL AND FLANGES	10	WITHIN ALLOWABLES VIA NASA/LANGLEY AND ARC ANALYSIS.

TABLE 2.3.4. CONTINUED.

RATING CRITERION	RATING	COMMENTS
B. PEAK STRESSES AROUND FASTENER HOLES	6	LOCAL YIELDING INDICATED BY ANALYSIS.
C. STRESS IN FASTENERS	6	SOME BEARING YIELDING INDICATED BY ANALYSIS.
D. SEAL GAP OPENING	10	GAP OPENING WITHIN ALLOWABLE FOR ALL THREE O-RINGS.
E. EASE OF SEAL INSTALLATION AND SEGMENT ASSEMBLY	6	O-RINGS 12 FOOT DIAMETER MUST ROLL INTO CYLINDRICAL GROOVES. METAL CHIPS OR O-RING DAMAGE POSSIBLE.
F. REUSE DAMAGE AND RISK	8	PIN HOLES MAY SUFFER MINOR YIELDING
COST AND PRACTICALITY		
1. MANUFACTURABILITY	8	REQUIRES CLOSE TOLERANCES TO MAKE CAPTURE LIP WORK (DIAMETER, ROUNDNESS, THICKNESS).
2. ASSEMBLY/DISASSEMBLY	6	REQUIRES PRECISE ROUNDING FOR INTERFERENCE CAPTURE LIP TO ENGAGE INNER CLEVIS.

WEIGHT

1. JOINT WEIGHT GAINED WITH RESPECT TO 51-L BASELINE: 220 LB
2. PROPELLANT WEIGHT LOSS WITH RESPECT TO 51-L BASELINE: -80 LB

TABLE 2.3.5. TRADE STUDY RATING - MODIFIED ARC CAPTURE FEATURE.

RATING CRITERION	RATING	COMMENTS
RELIABILITY		
1. ANALYSIS TRACTABILITY - PREDICTABILITY		
A. OVERALL PRESSURE VESSEL PARTS	10	THOROUGH ANALYSIS BY NASA/LANGLEY AND ARC.
B. FASTENERS	6	ANALYSIS BY NASA/LANGLEY SHOWS SENSITIVITY TO PIN POSITIONING AND POSSIBLE YIELDING OF PIN AND YIELDING AROUND TANG AND CLEVIS PIN HOLES.
C. SEALS AND GAPS	8	NASA/LANGLEY ANALYSIS SHOWS SMALL GAP OPENING WITH ASSUMED INTERFERENCE FIT OF CAPTURE FEATURE.
2. THERMAL RESISTANCE (IF INSULATION BREACHED)		
A. PRIMARY AND SECONDARY SEAL RESISTANCE	4	RUBBER O-RINGS GOOD ONLY TO 350°F.
B. FLANGE GAP AREA (CIRCUMFERENTIAL FLOW)	8	GAP FLOW MINIMIZED BY CAPTURE FEATURE.
C. SEAL SEATING GROOVE FLOW AREA	9	SOME AREA OPEN DUE TO SEATING OF O-RING(S). NO FLOW AREA AROUND FACE SEAL.
D. EFFECT OF COLD TEMPERATURE ON SEALING ABILITY	10	O-RINGS STAY WARM DUE TO STRIP HEATERS. FACE SEAL REMAINS IN COMPRESSION.
3. JOINT MECHANICS - STRESS STATES		
A. STRESS STATE IN JOINT SHELL AND FLANGES	10	WITHIN ALLOWABLES VIA NASA/LANGLEY AND ARC ANALYSIS.
B. PEAK STRESSES AROUND FASTENER HOLES	6	LOCAL YIELDING INDICATED BY ANALYSIS.
C. STRESS IN FASTENERS	6	SOME BEARING YIELDING INDICATED BY ANALYSIS.

TABLE 2.3.5. CONTINUED.

RATING CRITERION	RATING	COMMENTS
D. SEAL GAP OPENING	10	GAP OPENING WITHIN ALLOWABLE FOR O-RINGS AND FACE SEAL.
E. EASE OF SEAL INSTALLATION AND SEGMENT ASSEMBLY	6	O-RINGS 12 FOOT DIAMETER MUST ROLL INTO CYLINDRICAL GROOVES. METAL CHIPS OR O-RING DAMAGE POSSIBLE.
F. REUSE DAMAGE AND RISK	8	PIN HOLES MAY SUFFER MINOR YIELDING
COST AND PRACTICALITY		
1. MANUFACTURABILITY	8	REQUIRES CLOSE TOLERANCES TO MAKE CAPTURE LIP WORK (DIAMETER, ROUNDNESS, THICKNESS).
2. ASSEMBLY/DISASSEMBLY	6	REQUIRES PRECISE ROUNDING FOR INTERFERENCE CAPTURE LIP TO ENGAGE INNER CLEVIS.

WEIGHT

1. JOINT WEIGHT GAINED WITH RESPECT TO 51-L BASELINE: 220 LB
2. PROPELLANT WEIGHT LOSS WITH RESPECT TO 51-L BASELINE: -80 LB

TABLE 2.3.6. TRADE STUDY RATING - ARC ANGLE BOLTED JOINT.

RATING CRITERION	RATING	COMMENTS
RELIABILITY		
1. ANALYSIS TRACTABILITY - PREDICTABILITY		
A. OVERALL PRESSURE VESSEL PARTS	10	STATE OF ART THREE-DIMENSIONAL FINITE ELEMENT MODEL.
B. FASTENERS	6	SUBJECT TO ASSUMPTIONS REGARDING FRICTION UNDER NUTS AND BETWEEN FLANGES.
C. SEALS AND GAPS	8	INFLUENCED BY BOLT ASSUMPTIONS.
2. THERMAL RESISTANCE (IF INSULATION BREACHED)		
A. PRIMARY AND SECONDARY SEAL RESISTANCE	10	FILLED GASKET AND C-RING GOOD TO >2000°F.
B. FLANGE GAP AREA CIRCUMFERENTIAL FLOW)	8	GAP MINIMIZED BY BOLT PRETENSION.
C. SEAL SEATING GROOVE FLOW AREA	10	GASKET FILLS GROOVE.
D. EFFECT OF COLD TEMPERATURE ON SEALING ABILITY	10	NEITHER SEAL SENSITIVE TO COLD.
3. JOINT MECHANICS - STRESS STATES		
A. STRESS STATE IN JOINT SHELL AND FLANGES	10	ENGINEERED TO BELOW ALLOWABLES.
B. PEAK STRESSES AROUND FASTENER HOLES	10	SHOULD BE ACCEPTABLE EVEN IF SOME YIELDING OCCURS.
C. STRESS IN FASTENERS	10	SHANK STRESS BELOW ALLOWABLE. STUDS NOT TO BE REUSED. TAPPED HOLE THREADS DESIGNED TO AVOID GROSS YIELDING AND TO BE INSPECTED.

TABLE 2.3.6. CONTINUED.

RATING CRITERION	RATING	COMMENTS
D. SEAL GAP OPENING	10	ENGINEERED TO ACCEPTABLE VALUES (<.012).
E. EASE OF SEAL INSTALLATION AND SEGMENT ASSEMBLY	6	SEALS MUST BE DEMONSTRATED TO WORK IN ANGLED GROOVES.
F. REUSE DAMAGE AND RISK	8	STUD HOLE THREADS REQUIRE INSPECTION.
COST AND PRACTICALITY		
1. MANUFACTURABILITY	6	CONICAL SURFACES MAY PRESENT A "FLATNESS" PROBLEM, UPPER SEGMENT REQUIRES NEW MILLING PROCESS.
2. ASSEMBLY/DISASSEMBLY	8	ROUNDING NECESSARY. USE OF HYDRAULIC BOLT TENSIONERS WILL BE A BIG HELP IN FIELD ASSEMBLY.

WEIGHT

1. JOINT WEIGHT GAINED WITH RESPECT TO 51-L BASELINE: 932 LB
2. PROPELLANT WEIGHT LOSS WITH RESPECT TO 51-L BASELINE: -375 LB

TABLE 2.3.7. TRADE STUDY RATING - ARC IN-LINE AXIAL BOLTED JOINT.

RATING CRITERION	RATING	COMMENTS
RELIABILITY		
1. ANALYSIS TRACTABILITY - PREDICTABILITY		
A. OVERALL PRESSURE VESSEL PARTS	10	THREE-DIMENSIONAL CONFIGURATION FIT-UP CAN BE PRECISELY MODELED
B. FASTENERS	10	STUDS, NUTS, AND BOLT-PRETENSION CAN BE PRECISELY MODELED.
C. SEALS AND GAPS	10	GAP OPENING NOT SENSITIVE TO UNKNOWNS AND ASSUMED CONDITIONS
2. THERMAL RESISTANCE (IF INSULATION BREACHED)		
A. PRIMARY AND SECONDARY SEAL RESISTANCE	10	GASKET AND C-RING GOOD FOR 2500 AND 3000°F
B. FLANGE GAP AREA (CIRCUMFERENTIAL FLOW)	8	GAP OPENING CONTROLLED TO <.012 BY STUD PRETENSION AND FLANGE GEOMETRY
C. SEAL SEATING GROOVE FLOW AREA	9	GASKET ALLOWS NO FLOW, C-RING DOES NOT CIRCUMFERENTIALLY FILL GROOVE.
D. EFFECT OF COLD TEMPERATURE ON SEALING ABILITY	10	NEITHER GASKET NOR C-RING SENSITIVE TO LOW TEMPERATURE
3. JOINT MECHANICS - STRESS STATES		
A. STRESS STATE IN JOINT SHELL FLANGES	10	ENGINEERED TO BE BELOW ALLOWABLE LEVELS
B. PEAK STRESSES AROUND FASTENER HOLES	10	ENGINEERED TO MAINTAIN ACCEPTABLE AMOUNTS OF YIELDING
C. STRESS IN FASTENERS	10	BULK STRESS BELOW ALLOWABLE. STUDS NOT TO BE REUSED. TAPPED HOLE THREADS DESIGNED TO AVOID GROSS YIELDING. ROOTS TO BE INSPECTED.

TABLE 2.3.7. CONTINUED.

RATING CRITERION	**RATING**	**COMMENTS**
D. SEAL GAP OPENING	10	ENGINEERED TO BE ACCEPTABLE
E. EASE OF SEAL INSTALLATION AND SEGMENT ASSEMBLY	10	SEALS PLACED IN HORIZONTAL GROOVES AND INSPECTED AT THIS LEVEL. MATING SEGMENTS COMPRESS SEALS WITH NO SCRAPING.
F. REUSE DAMAGE AND RISK	8	STUD HOLE THREADS REQUIRE INSPECTION.
COST AND PRACTICALITY		
1. MANUFACTURABILITY	8	MATING SURFACES AND GROOVES EASY TO MACHINE AND INSPECT. NEW MILLING PROCESS NECESSARY TO CUT ALCOVES AND FLUTES.
2. ASSEMBLY/DISASSEMBLY	8	ROUNDING REQUIRED TO ALIGN TOP SEGMENT. TOLERANCES WILL BE REASONABLE.

WEIGHT

1. JOINT WEIGHT GAINED WITH RESPECT TO 51-L BASELINE: 828 LB
2. PROPELLANT WEIGHT LOSS WITH RESPECT TO 51-L BASELINE: -306 LB

TABLE 2.3.8. TRADE STUDY RATING - MODIFIED LARC IN-LINE BOLTED JOINT.

RATING CRITERION	RATING	COMMENTS
RELIABILITY		
1. ANALYSIS TRACTABILITY - PREDICTABILITY		
A. OVERALL PRESSURE VESSEL PARTS	10	THREE-DIMENSIONAL CONFIGURATION AND FIT-UP CAN BE PRECISELY MODELED.
B. FASTENERS	10	BOLTS AND PRETENSION EFFECTS CAN BE PRECISELY MODELED
C. SEALS AND GAPS	10	GAP OPENING NOT SENSITIVE TO UNKNOWNS AND ASSUMPTIONS.
2. THERMAL RESISTANCE (IF INSULATION BREACHED)		
A. PRIMARY AND SECONDARY SEAL RESISTANCE	10*	METAL C-RING GOOD TO 3000°F, BUT SECONDARY O-RING ONLY TO 350°F.
B. FLANGE GAP AREA (CIRCUMFERENTIAL FLOW)	8	FLANGE SEPARATION (GAP) MINIMIZED BY BOLT POSITIONING AND PRE-TENSION.
C. SEAL SEATING GROOVE FLOW AREA	9*	RINGS FILL UP ONLY PART OF THE GROOVES.
D. EFFECT OF COLD TEMPERATURE ON SEALING ABILITY	10*	NO EFFECT ON C-RING, DETERS O-RING ACTIVATION RATE.
3. JOINT MECHANICS - STRESS STATES		
A. STRESS STATE IN JOINT SHELL AND FLANGES	10	ENGINEERED TO BE BELOW ALLOWABLE LEVELS.
B. PEAK STRESSES AROUND FASTENER HOLES	10	SHOULD BE ACCEPTABLE EVEN IF SOME YIELDING OCCURS.
C. STRESS IN FASTENERS	10	SHANK STRESSES BELOW ALLOWABLES. THREAD STRESS CONCENTRATIONS ACCEPTABLE SINCE BOLTS OR STUDS ARE NOT REUSED.

*REPLACE RUBBER O-RING BY METAL JACKETED GASKET.

TABLE 2.3.8. CONTINUED.

RATING CRITERION	RATING	COMMENTS
D. SEAL GAP OPENING	10	ENGINEERED TO ACCEPTABLE VALUES (<.001).
E. EASE OF SEAL INSTALLATION AND SEGMENT ASSEMBLY	10	SEALS PLACED IN HORIZONTAL GROOVES. CAN BE INSPECTED AT THIS LEVEL. MATING SEGMENT SIMPLY COMPRESSES SEALS WITH NO SCRAPING.
F. REUSE DAMAGE AND RISK	10	ALL JOINT PARTS EASILY INSPECTED.
COST AND PRACTICALITY		
1. MANUFACTURABILITY	8	MATING SURFACES AND GROOVES EASY TO MACHINE AND INSPECT. NEW MILLING PROCESS NECESSARY TO CUT ALCOVES.
2. ASSEMBLY/DISASSEMBLY	8	WILL REQUIRE ROUNDING TO ALIGN TWO SEGMENTS. TOLERANCES WILL BE REASONABLE.

WEIGHT

1. JOINT WEIGHT GAINED WITH RESPECT TO 51-L BASELINE: 943 LB
2. PROPELLANT WEIGHT LOSS WITH RESPECT TO 51-L BASELINE: -1000 LB

TABLE 2.3.9. PAYLOAD WEIGHT CHANGES.

	THREE REDESIGNED FIELD JOINTS, FOUR FACTORY JOINTS ELIMINATED	
JOINT DESIGN	ΔPL^{*} (LB)	NORMALIZED PAYLOAD FACTOR**
MODIFIED LARC IN-LINE BOLTED	-225	9.96
ARC IN-LINE AXIAL BOLTED	+11	10.00
ARC ANGLE BOLTED	-63	9.99
MODIFIED ARC CAPTURE FEATURE	+400	10.07
NASA/MTC CAPTURE FEATURE	+400	10.07

*ΔPL = CHANGE IN PAYLOAD

= -.182 (CHANGE IN INERT WEIGHT) + .083 (CHANGE IN PROPELLANT WEIGHT)

**NORMALIZED PAYLOAD FACTOR = $\frac{60000 + \Delta PL}{60000}$ (10),

WHERE BASELINE PAYLOAD = 60000 LB

TABLE 2.3.10. INERT AND PROPELLANT WEIGHT CHANGES PER BOOSTER.

	THREE REDESIGNED FIELD JOINTS, FOUR FACTORY JOINTS ELIMINATED Δ(WT) = 3[REDESIGN - 51-L] - 4[51-L]
STEEL	
MODIFIED LARC IN-LINE BOLTED	3(943) - 4(690) = 69 LB
ARC IN-LINE AXIAL BOLTED	3(828) - 4(690) = -276 LB
ARC ANGLE BOLTED	3(932) - 4(690) = 36 LB
MODIFIED ARC CAPTURE FEATURE	3(220) - 4(690) = -2100 LB
NASA/MTC CAPTURE FEATURE	3(220) - 4(690) = -2100 LB
PROPELLANT	
MODIFIED LARC IN-LINE BOLTED	3(-1000) - 4(-110) = -2560 LB
ARC IN-LINE AXIAL BOLTED	3(-306) - 4(-110) = -478 LB
ARC ANGLE BOLTED	3(-375) - 4(-110) = -685 LB
MODIFIED ARC CAPTURE FEATURE	3(-80) - 4(-110) = 200 LB
NASA CAPTURE FEATURE	3(-80) - 4(-110) = 200 LB

The calculation of the final joint/seal ranking factor is illustrated by sample in Table 2.3.11, and the summary of all rankings is given in Table 2.3.12.

We note that the LARC in-line bolted joint was modified for this evaluation by replacing the primary o-ring seal with a metal jacketed graphite foil-filled type of gasket. This should be a better seal for withstanding the back pressure of the leak check, since the gasket will not shift position. Additionally, the temperature resistance of the metal gasket is far superior to a rubber o-ring and should be capable of withstanding exposure temperatures beyond 2500°F. Also, an aligning shear lip, similar to that on the ARC bolted joint, is presently planned as a modification.

Conclusions and Recommendations

The net results of the trade studies are shown in Table 2.3.13. The recommended joint redesign is the NASA/Langley in-line bolted configuration with the above mentioned modifications, and, perhaps, some slight geometry changes still to be determined.

2.4 INSULATION TRADE STUDIES

The insulation trade study was begun with an identification of the asbestos-containing materials in the SRM which must be replaced. Asbestos is found in the current design in the case internal insulation, the igniter internal and external insulation, the molded and cast inhibitors, the liner, the nozzle, and in certain components in the Safe and Arm (S&A) device. Discussions of the liner, nozzle, igniter, and S&A material replacements appear in those respective sections of this report. This section presents the effort conducted on the case insulation. The insulation selected for the igniter and molded inhibitor resulted from the trade study conducted for the case insulation.

The initial effort for the case insulation trade study was to evaluate potential fabrication processes. This evaluation will be presented in Section 2.4.1. Once a fabrication process was selected, an insulation trade study based on the selected process was conducted. This case insulation trade study is presented in Section 2.4.2.

TABLE 2.3.11. SAMPLE CALCULATION OF RANKING FACTOR FOR MODIFIED LARC IN-LINE BOLTED JOINT.

1. NORMALIZED PAYLOAD FACTOR, K_{PL}:

$$\Delta PL = -.182\ (\Delta W_I) + .083\ (\Delta W_P)$$

$$= -.182\ (69) + .083\ (-2560)$$

$$= -225$$

$$K_{PL} = \frac{60000 + \Delta PL}{60000}\ (10)$$

$$= \frac{60000 - 225}{60000}\ (10)$$

$$= 9.96$$

2. RELIABILITY FACTOR, K_R:

$$K_R = \frac{\text{SUM OF RELIABILITY RATINGS}}{\text{NUMBER OF RATINGS}}$$

$$= \frac{11(10) + 9 + 8}{13}$$

$$= 9.77$$

3. COST FACTOR, K_C:

$$K_C = \frac{\text{SUM OF COST RATINGS}}{\text{NUMBER OF RATINGS}}$$

$$= \frac{2\ (8)}{2}$$

$$= 8.00$$

4. RANKING FACTOR:

$$\text{RANK} = .15\ (K_{PL}) + .65\ (K_R) + .20\ (K_C)$$

$$= .15\ (9.96) + .65\ (9.77) + .20\ (8.00)$$

$$= 9.44$$

TABLE 2.3.12. FINAL JOINT TRADE STUDY RANKING.

JOINT DESIGN	SEALS PRIMARY	SEALS SECONDARY	WEIGHTED SCORE
MODIFIED LARC IN-LINE BOLTED*	JACKETED GASKET	METAL C-RING	9.44
ARC IN-LINE BOLTED	JACKETED GASKET	METAL C-RING	9.35
ARC ANGLE BOLTED	JACKETED GASKET	METAL C-RING	8.69
ARC MODIFIED CAPTURE FEATURE	RUBBER GASKET	RUBBER O-RING	7.96
NASA/MTC CAPTURE FEATURE	RUBBER O-RING	RUBBER O-RING	7.85

RANKING FACTOR = .20 (COST) + .15 (WEIGHT) + .65 (RELIABILITY)

*SHEAR LIP ADDED
GASKET SEAL IN PLACE OF O-RING SEAL

TABLE 2.3.13. JOINT TRADE STUDIES CONCLUSIONS AND RECOMMENDATIONS.

- NASA/LANGLEY IN-LINE BOLTED JOINT RECOMMENDED FOR FIELD JOINT REDESIGN WITH MODIFICATIONS
  - METAL JACKETED GASKET AS A PRIMARY SEAL
  - IN-BOARD SHEAR LIP
- JOINT TO BE FOAM FILLED WITH ALCOVES COVERED
- DETAILED STRUCTURAL ANALYSIS SHOULD BE CONTINUED TOWARD OPTIMIZATION

2.4.1 FABRICATION PROCESS TRADE STUDY

The large-scale of the Shuttle SRM's presents processing problems that are not commonly confronted in smaller scale rocket motors of similar design. These problems cannot only effect cost and schedule, but may also raise questions regarding quality. The manufacture of a reliable insulation on or near schedule will require a process method that is physically feasible for the types of insulations that are currently available to replace the asbestos filled NBR. Because the process plays a major role in the quality and reliability of a SRM insulation, a trade study was conducted to rate a variety of state-of-the-art processes that could be used to manufacture viable insulation candidates.

The processing methods were rated on their 1) potential to reproduce a reliable insulation, 2) the performance of the materials processible by that method, and 3) the cost advantages/drawbacks. The weighting factors for this rating criteria are 75% for reliability, 15% for performance, and 10% for cost. The processes were rated on a scale of 1 through 10, with 10 being the highest rating. The fabrication processes selected for the trade are listed and rated in Table 2.4.1. Included are the characteristics of uncured polymer binders that are processible with each process. Of these processes, the automated (ribbon winding) lay-up process was given the overall highest rating and is proposed as the prinicpal processing method of the SRM insulation.

The automated lay-up process consists of a computer controlled extrusion-winding operation. This operation, graphically displayed in Figure 2.4.1 lays an extruded ribbon of insulation into the case as the case is rotated. A premixed gumstock is fed into a single screw extruder and extruded into a ribbon of a controlled temperature which provides good tack and pliability without scorching the rubber binder. The ribbon is transferred from the extruder along a roller conveyer arrangement and positioned on to the case by a rubber applicator wheel. This applicator applies pressure to the ribbon during lay-up by a pneumatic control to ensure that the ribbon lay-up simulates that of shingles, where one strip partly overlaps the prior strip. the applicator moves axially within the case as the insulation is wrapped. The thickness of the insulation can be controlled by varying the axial speed with respect to the angular speed of the case; e.g., to thin the insulation, the applicator is moved along the axis at a greater rate so there are less ribbon overlaps. The opposite is done to thicken the insulation. To provide tack between the case and ribbon, the case wall will be coated with a film of the insulation

TABLE 2.4.1. FABRICATION PROCESS TRADE STUDY.

FABRICATION PROCESS	INSULATIONS	REPRODUCIBILITY RATING RELIABILITY (WEIGHT FACTOR - 0.75)	PERFORMANCE (WEIGHT FACTOR - 0.15)	COST (WEIGHT FACTOR - 0.10)	TOTAL WEIGHT SCORE
• HAND SEMI-MECHANIZED LAY-UP (CURRENTLY USED)	GUMSTOCKS	7	7	7	7.00
• AUTOMATED LAY-UP (RIBBON WIND)	GUMSTOCKS	9	8	8	8.75
• CAST	CTPB BASE (LIQUIDS)	6	7	3	5.85
• INJECTION MOLD	ANY	3	8	1	3.55
• SLING LINE	CTPB BASE DC93-104	3	8	1	3.55
• BOND IN CURED INSULATION	ANY	6	8	4	6.10

AUTOMATED LAY-UP (RIBBON WIND) SELECTED

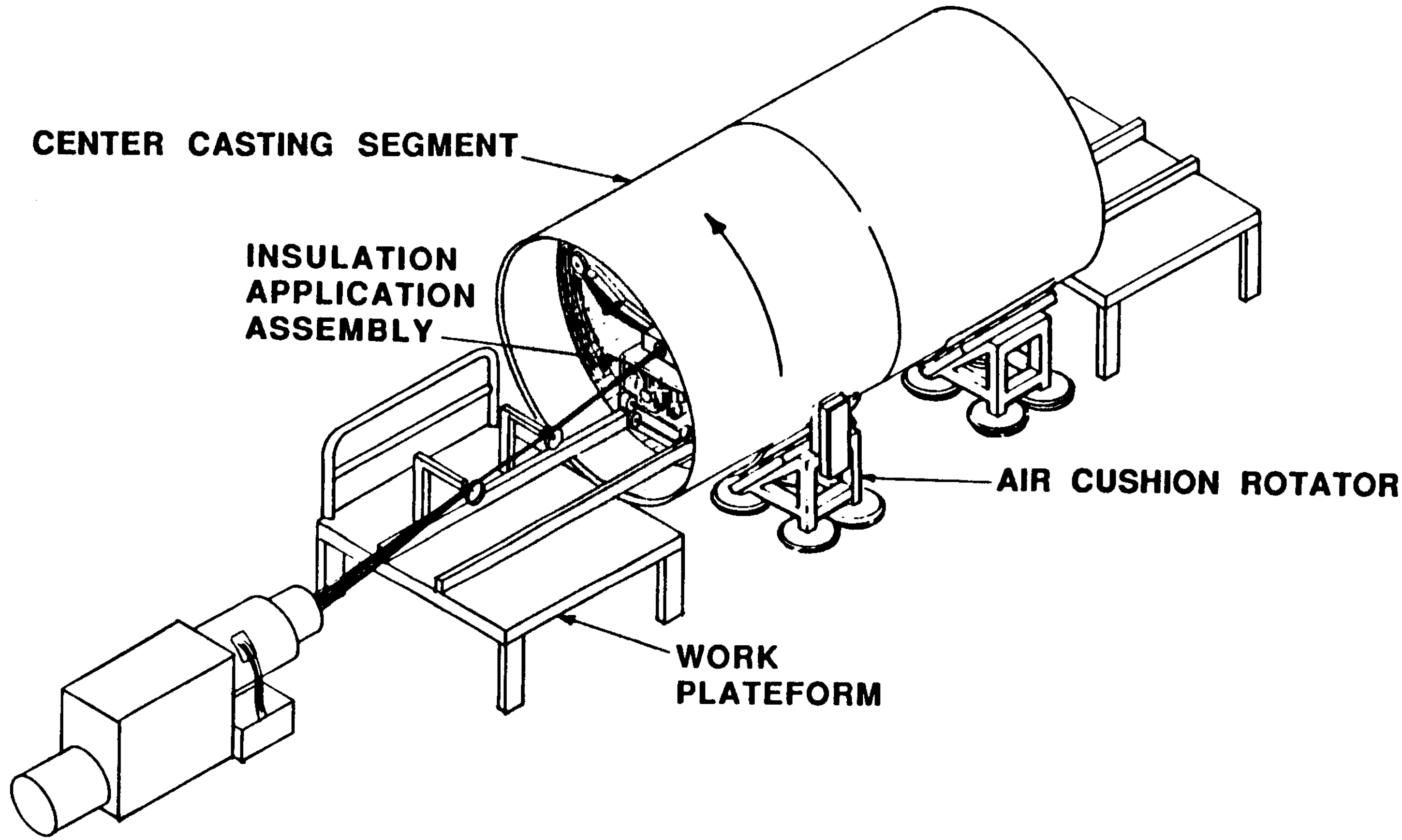


FIGURE 2.4.1. AUTOMATED RIBBON - WIND INSULATION LAY-UP PROCESS.

binder by a spray-on application. This is conducted by spraying on the solvenated binder and allowing the solvent to flash off. As in the current process, the case will first be coated with a primary, co-curing adhesive before the tack coat and insulation are applied. The automated process will also be used to lay-up designated inhibitor flaps. The insulation will be bagged for autoclave cure, as conducted on prior flight motors except that a one-piece butyl rubber bag is recommended, over the nylon bagging film, to minimize vacuum bag leaks during autoclave cure. The final step will be to cure the ribbon wound lay-up in an autoclave to net-shape.

The automated ribbon winding lay-up process was given the highest scores in each category of the rating criteria. This process was given the highest rating in the reproducibility/reliability category. This is mainly due to the computer controlled operation, which greatly increases the redundancy of operation while reducing intensive labor and eliminating the use of the solvents to activate tack during lay-up. The latter two factors are the primary problem with the semi-mechanized lay-up operation used to build SRM insulations to date. Another advantage of the automated lay-up process is that the void content in the lay-up is minimized due to the increased tack of the freshly extruded ribbon and the controlled pressure from the applicator wheel on the ribbon during lay-up. The extrusion operation eliminates the need for calendering stocks which broadens the scope of materials that can be considered. Many of the high performance asbestos-free insulations have inherently poor tack at room temperature and do not calender well; however, they are readily processible by an extruded ribbon winding process. In addition, prepreg fabric materials may also be considered for wind processing. Although the capital costs of the equipment is moderately high, the technology behind the equipment is not only advanced but proven. Extruded ribbon winding operations are currently used to manufacture a variety of rubber products, from tires to rocket motor insulations. Start-up (after equipment procurement) and qualification tests would require several months' effort; however, this process method is extremely cost effective based on its high output rate and its expected low attrition rate.

The Hand Semi-Mechanized lay-up process technique was used to build SRM insulations to date. Of the processes rated this operation is the most labor intensive. Other problems are that it is a slow process, and solvents are often used to activate tack on calendered stock. Aside from these, the material selection would be somewhat more limited than the automated lay-up process because calendered stock is required. Costs would be slightly higher than the automated process mainly due to the additional labor involved in calendering, and hand lay-up. Aside from these problems, the hand semi-mechanized process was rated number 2 overall.

A casting process was also given a high rating, but not the highest due to the higher potential of voids to occur in the insulation. This could be a serious problem because it is very difficult to accurately x-ray insulations on the case. The material selection of castable insulations is limited. A major problem with this process is the tooling costs and the complexity of equipment that would be required to pump, or cast insulations between the case and the casting tooling over the segment length.

Bonding in a cured insulation would use the automated lay-up process except the insulation would be layed-up on a mandrel and cured, machined, and then bonded into the case. This process features some notable advantages, such as full X-ray inspection capability and greater ease in processing; however, the bond integrity would always be in question due to the difficulty in loading the insulation into the case. Injection molding and sling lining, although optimal processes for many end products, would be impractical for an insulation of this size and dimension.

2.4.2 CASE INSULATION TRADE STUDY

ARC conducted an insulation trade study to find a non-asbestos replacement for the NBR/asbestos/silica insulation. The study was divided into two parts. The first step was a review and selection of binders and fibers. The selected binder/fiber combinations were then evaluated in a detailed trade study.

The binders that were considered for insulations are listed in Table 2.4.2. Three areas, specific gravity, thermal conductivity and ARC/industry data base were considered important for the binder selection. Four binders have a specific gravity lower than the baseline NBR and there is an established data base on SBR, EPDM, polyisoprene and polybutadiene. All of the binders have equivalent or lower thermal conductivities than NBR. For the binders being considered, Hypalon has the lowest thermal conductivity. Of the seven binders listed with an established data base, ARC considered NBR, Hypalon, SBR, EPDM, and polyisoprene to be the best for consideration in the detailed trade study.

TABLE 2.4.2. INSULATION BINDER TRADE STUDY.

BINDERS	SPECIFIC GRAVITY	THERMAL CONDUCTIVITY BTU/ft-hr-°F	ARC AND INDUSTRY DATA BASE	SELECTED
NBR	0.98	0.143	X	X
HYPALON	1.20	0.065	X	X
POLYURETHANE	1.14	0.10		
NEOPRENE	1.20	0.112	X	
SBR	0.94	0.143	X	X
EPDM	0.86	0.15	X	X
FLUOROCARBON	1.68	0.13		
SILICONE	1.35	0.13		
POLYISOPRENE	0.93	0.082	X	X
FLUOROSILICONE	1.40	0.13		
POLYBUTADIENE	0.90	0.13	X	

There were many different types of organic and inorganic fibers considered for replacing the asbestos. The fibers are listed in Table 2.4.3. None of the inorganic fibers can be processed in standard rubber mixing equipment, such as a banbury or a mill, without severe fiber damage or breakage. Fiber breakage will reduce the char strength of the insulation. The only inorganic fiber considered for the detail trade study was carbon. A carbon/EPDM insulation which is manufactured using a solvent process is currently being used in the shuttle. Two organic fibers, Kevlar and PBI, have primarily been used in the industry as asbestos replacements. Based on ARC's experience, Kevlar performs better than PBI. Also, ARC has experience with a cellulose fiber that provides good char and acts as a coolant when it decomposes. Therefore, Kevlar was chosen as the primary fiber for replacing the asbestos and two formulations will be considered with the cellulose and carbon fiber.

The detailed trade study was divided into three areas; reliability, weight and cost. The trade study is given in Table 2.4.4. The weighting factors for three areas were 75 percent for reliability, 15 percent for weight and 10 percent for cost. Reliability was sub-divided into four areas: thermal (25%), manufacturing/processing (30%), compatibility (10%) and mechanical properties (10%). The thermal area was divided into material affected rate (MAR) and thermal protection. MAR was given a higher weighting factor because of the requirement of a 2.0 safety factor. Manufacturing/processing was divided into four areas. Viscosity/scorch was rated the highest because of the need to 1) be extrudable, and 2) have uniform cure/properties throughout the thickness of the material. Green tack and ribbon integrity were rated the next highest, respectively, because once the material is extruded it must stick to itself to provide a uniform lay-up, minimized air entrapment and provide a controlled process. Compatibility was divided into vacuum outgassing and aging with the latter given a higher weighting factor. Mechanical properties were divided into bond, strain and modulus. Bond was given the highest weighting factor in order to maintain compatibility with the current liner. Cost was subdivided into recurring and non-recurring. Recurring cost was given a higher weight factor than non-recurring because it is the material cost. Non-recurring is the equipment cost which will be about the same for all of the materials.

The candidate materials (binder/fillers) are also given in Table 2.4.4. Ten candidates plus the control, NBR silica/asbestos were evaluated in the detailed study. The best three insulations in the area of reliability were USR-3800, EPDM Kevlar/cellulose/silica, and Hypalon silica/Kevlar. The primary reason for their high reliability

TABLE 2.4.3. FIBER INGREDIENTS.

TYPE	FIBER
INORGANIC	ALUMINA-BORIA-SILICA
	ALUMINA-SILICATE
	BORON
	ALUMINA-CHRONICA-SILICA
	CARBON
	GLASS
	SILICON CARBIDE
	ZIRCONIA SILICATE
ORGANIC	KYNOL
	POLYESTER
	COTTON
	RAYON
	PBI
	KEVLAR
	NYLON
	PAN
	CELLULOSE

TABLE 2.4.4. CASE INSULATION TRADE STUDY CANDIDATE MATERIAL RATINGS.

RATING PARAMETER	WEIGHTING FACTOR	NBR SILICA/ ASBESTOS	EPDM SILICA/ KEVLAR	EPDM SILICA/ KEVLAR CELLULOSE	HYPALON SILICA/ KEVLAR	USR-3800 NBR/PHENOLIC BORIC ACID	EPDM SILICA	NBR SILICA	NBR SILICA/ MODIFIED	TIR 701 POLYISOPRENE KEVLAR/ SILICA	SBR SILICA/ KEVLAR	EPDM CARBON FIBER
THERMAL												
- MATERIAL AFFECTED RATE	0.18	0.90*	1.44	1.62	1.26	1.80	0.54	0.54	0.90	0.72	0.54	1.62
- THERMAL PROTECTION	0.07	0.35	0.56	0.63	0.70	0.35	0.42	0.21	0.35	0.63	0.35	0.35
MANUFACTURING/PROCESSING												
- RIBBON INTEGRITY-PLIABILITY	0.07	0.56	0.56	0.56	0.56	0.49	0.56	0.56	0.28	0.42	0.56	0.14
- GREEN TACK	0.09	0.63	0.36	0.45	0.63	0.36	0.36	0.72	0.72	0.72	0.72	0.18
- CURE REVERSION RATES	0.04	0.28	0.28	0.28	0.36	0.28	0.20	0.24	0.24	0.16	0.24	0.20
- VISCOSITY - SCORCH	0.10	0.80	0.40	0.40	0.80	0.60	0.20	0.90	0.50	0.40	0.90	0.20
COMPATIBILITY												
- VACUUM OUTGASSING	0.03	0.30	0.30	0.30	0.30	0.30	0.30	0.30	0.30	0.30	0.30	0.30
- AGING	0.07	0.70	0.70	0.70	0.70	0.56	0.56	0.70	0.70	0.70	0.70	0.70
MECHANICAL PROPERTIES												
- BOND	0.06	0.54	0.30	0.42	0.48	0.48	0.48	0.42	0.48	0.42	0.42	0.42
- STRAIN	0.03	0.21	0.12	0.09	0.12	0.24	0.30	0.30	0.06	0.09	0.30	0.18
- MODULUS	0.01	0.07	0.08	0.09	0.08	0.08	0.04	0.04	0.09	0.09	0.04	0.07
WEIGHT	0.15	0.75	1.35	1.20	0.75	1.05	1.50	1.05	0.90	0.45	0.90	1.50
COST												
- RECURRING	0.08	0.64	0.48	0.48	0.48	0.80	0.64	0.64	0.48	0.48	0.48	0.24
- NON-RECURRING	0.02	0.16	0.16	0.16	0.16	0.16	0.16	0.16	0.16	0.16	0.16	0.12
TOTAL RATING =		6.89	7.09	7.38	7.38	7.55	6.26	6.78	6.16	5.74	6.61	6.22

* RATING VALUE = WEIGHT FACTOR x RELATIVE PERFORMANCE

rating is due to thermal performance. The candidate with the least reliability was an EPDM silica. The best three candidates for weight were EPDM carbon fiber, EPDM silica and EPDM silica/Kevlar. All of these have an EPDM binder which has the lowest specific gravity. All the candidates were close on cost except for the EPDM carbon fiber which is the most expensive because of the solvent process used in manufacturing the material. A diagram showing the overall ratings for each of the candidates in the three categories is given in Figure 2.4.2. The best three insulations are Hypalon silica/Kevlar, USR-3800 and the EPDM Kevlar/cellulose/silica. Design studies using these materials are presented in Section 3.4.1.

2.5 PROPELLANT TRADE STUDIES

The objective of this task was to improve SRM performance and reliability via propellant, liner, and igniter propellant formulations. The constraints set for the selection process were existence of a solid data base, retention of the thrust-time trace, unchanged or improved variability, maintained structural margins, a 0.364 in/sec burning rate requirement (625 psi), and asbestos elimination. These issues are summarized in Table 2.5.1.

An examination of available propellants quickly eliminated the higher performance aluminized Class 1.1 propellants because of the increased hazards, and the alternate or "clean" propellants based on ammonium nitrate because of performance degradation and their not yet being state-of-the-art. The trade candidates were therefore limited to a PBAN formulation such as TPH-1148 in the SRM and an improved performance HTPB propellant. The evaluation factors were heavily weighted for reliability (motor integrity, producibility, etc.) which was assigned 60%. Hazards and payload impact were each assigned 15% with cost considerations having the least influence (10/%).

ARCADENE 360B, the HTPB propellant used in these trade studies, is compared to TPH-1148 in Table 2.5.2. ARCADENE 360B must be slightly modified to meet the burning rate requirement by reducing Fe_2O_3 percent which becomes very similar to that used in TPH-1148. ARCADENE 360B has been extensively characterized over a wide range of burning rates. Atlantic Research Corporation (ARC) has produced over 28 million pounds of this propellant at our Camden, Arkansas facility. The current production rates for the Vought MLRS program is over 70,000 lbs/day, making it one of the most produced propellants in the world.

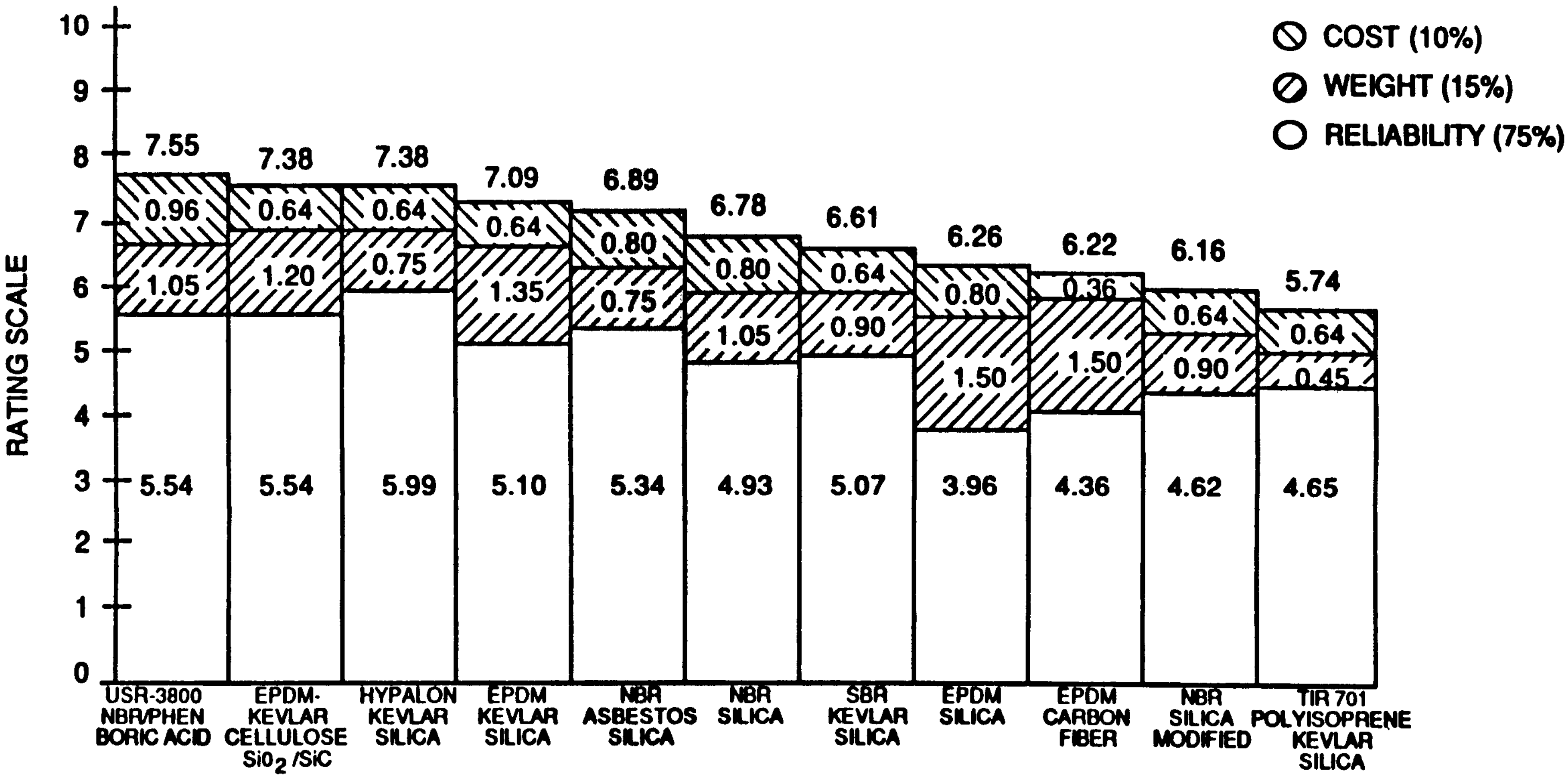


FIGURE 2.4.2. RATINGS OF INSULATION CANDIDATES.

TABLE 2.5.1. PROPELLANT AND LINER TRADE STUDY.

OBJECTIVE:

- SELECT PROPELLANTS (MOTOR AND IGNITER) AND LINER WITHOUT ASBESTOS.

CONSTRAINTS:

- SOLID DATABASE (MUST EXIST OR BE CREATED)
- MAINTAIN THRUST vs TIME
- MAINTAIN OR IMPROVE VARIABILITY
- MAINTAIN STRUCTURAL MARGINS
- BURNING RATE - 0.364 AT 625 PSI
- ELIMINATE ASBESTOS

SCOPE:

- PBAN vs HTPB (HIGH SOLIDS)
- LINER
- IGNITER PROPELLANT (TPH - 1178 vs IMPROVED COMPOSITION)

TABLE 2.5.2. PROPELLANT FORMULATIONS.

	TPH-1148	ARCADENE 360B
BINDER	PBAN/EPOXY	R-45HT/IPDI
Fe_2O_3	0 - 0.3%	0.1 - 0.3%
AL POWDER	16%	18%
TOTAL SOLIDS	86%	88%
AP (COARSE/FINE)	70/30	70/30

C-2

Table 2.5.3 summarizes the trade studies showing slightly higher rating for the current PBAN propellant, primarily because of demonstrated reliability in numerous flights and full scale motor firings in which a problem or malfunction due to propellant has never been identified to our knowledge.

ARCADENE 360B outscores the PBAN propellant in all other categories except for initial non-recurring costs for D&V to qualify the propellant in full scale SRM motors.

However, in the event that higher performance (payload) is required, ARC offers a very credible approach with ARCADENE 360B with a vast database and production proven reliability and producibility.

More details of the trades in each category follow.

Propellants are compared for producibility and reliability in Table 2.5.4.

In assessing the trades, scores of 1 to 10 were assigned with 7 being the norm assumed for TPH-1148 unless a deficiency existed. ARCADENE 360B has not been cast and fired in an SRM motor and therefore lacks the full-scale demonstration. This is the primary reason for the lower score. In terms of propellant produced, 28 million is less than 70 million but the difference is not significant since both numbers are very large. The gap will close rapidly as 360B production for MLRS continues. The 360B mechanical properties are superior to any PBAN propellant and can be tailored for higher strain. Although not required in this motor, 360B mechanical properties are significantly better at low temperature.

An asbestos fire-free liner must be formulated and demonstrated for both propellants. The insulation composition will affect the liner selection. An asbestos-free liner (ARL-151) has been demonstrated with ARCADENE 360B but may require some modification for the SRM application.

Table 2.5.5 compares propellant hazards.

Both propellants exhibit similar hazard characteristics typical of state-of-the-art Class B (military 1.3) composite formulations. ARCADENE 360B is rated slightly better for exhaust products based on its slightly lower CO concentration.

TABLE 2.5.3. PROPELLANT AND LINER TRADE STUDY RESULTS SUMMARY.

		PBAN TPH-1148		HTPB ARCADENE 360B	
CRITERIA	FACTOR	SCORE	WEIGHTED VALUE	SCORE	WEIGHTED VALUE
MOTOR INTEGRITY (PRODUCIBILITY, RELIABILITY, ETC.)	0.60	6.3	3.78	4.6	2.76
HAZARDS	0.15	7.0	1.05	7.3	1.10
IMPACT ON PAYLOAD	0.15	7.0	1.05	10.0	1.50
COST: NONRECURRING RECURRING	 0.05 0.05	 7.0 7.0	 0.35 0.35	 1.0 8.0	 0.05 0.40
TOTAL SCORE			6.58		5.81

TABLE 2.5.4. MOTOR PRODUCIBILITY AND RELIABILITY TRADE (PROPELLANT LINER).

PARAMETER	TPH-1148	SCORE	ARCADENE 360B	SCORE
FULL-SCALE MOTOR DEMONSTRATION	YES	7	NO	1
QUANTITY OF PROPELLANT PROCESSED (LB)	>70 MILLION	7	> 28 MILLION	6
MECHANICAL PROPERTIES s_m (psi) e_m (%) E_o (psi)	 113 37 518	7	 170 ± 10 35 ± 5 800 ± 100	7
BURNING RATE REPRODUCIBILITY	<2% DIFFERENCE FROM TARGET	7	± 1% ON MIXES WITHIN LOT	7
PROPELLANT DATABASE	EXTENSIVE	7	EXTENSIVE	7
ASBESTOS-FREE LINER	NO	4	YES WITH MODIFICATION	4
PROPELLANT/LINER BOND (NO ASBESTOS)	UNKNOWN	4	UNKNOWN	4
D & V	MINIMAL REQUIRED	7	SUBSTANTIAL EFFORT REQUIRED	1
TOTAL SCORE		50		37
AVERAGE SCORE		6.3		4.6

TABLE 2.5.5. PROPELLANT HAZARDS TRADE.

PARAMETER	TPH-1148	SCORE	ARCADENE 360B	SCORE
PROCESSING AND HANDLING HAZARDS	SOA COMPOSITE PROPELLANT	7	SOA COMPOSITE PROPELLANT	7
CURED PROPELLANT AND LOADED MOTOR PROPERTIES	CLASS B (MIL CLASS 1.3)	7	CLASS B (MIL CLASS 1.3)	7
TOXIC PROPELLENT EXHAUST PRODUCTS (MOLES /100g) HCl Al_2O_3 CO	 0.5860 0.2965 (S) 0.8555	7	 0.5837 0.2989(S) 0.7697	8
TOTAL SCORE		21		22
AVERAGE SCORE		7.0		7.3

The ARCADENE 360B performance advantages are clearly shown in Table 2.5.6. Specific impulse and density are both higher for ARCADENE 360B. The 25,175 propellant weight is calculated only from the density difference in the HPM Motor Design. A 2000 lb. extra insulation penalty was assumed for the higher flame temperature of 360B. Using a 1.2 g/cm^3 assumed insulation density, this converts to a 22,340 lb. additional 360B propellant. Using the assigned influence coefficients, a +1490 lb. payload is calculated. This figure does not include any additional thrust contribution from the higher theoretical specific impulse of 360B since some efficiency loss must also be assumed from the extra 2% Al.

The primary requirement for the liner is to reliably bond the propellant to the insulation. Secondary requirements are environmental and process. It must also be asbestos-free. It must have a reasonable pot life for application, it must stay in place on application to vertical insulation, and it must bond to propellant after being held at the cure temperature for a 30-hour casting period. All the requirements are met by the current system except the asbestos-free one.

The trade summarized in Table 2.5.7 evaluates a minimum change (Option I) in which the asbestos is replaced by another fibrous filler against a change (Option II) in which filler, polymeric composition, and other ingredients may be changed. The strongly weighted reliability criterion forces the trade results to Option I because it offers minimum change to the current system. Option II allows a potential payload increase from an estimate of weight savings which result from a decrease in liner thickness. Decreasing the current .057" to .020" corresponds to a decrease in weight of 1,000 pounds per SRM. If the volume lost were filled with propellant, a payload gain of 296 pounds would result.

Thin liners are a strong point in ARC motor technology. These thin liners frequently owe their success to internal barriers.

The existing shuttle rocket motor liner is based on Minuteman technology in which the liner also played a signficant role in insulating the case.

TABLE 2.5.6. PROPELLANT PERFORMANCE TRADE (HPM MOTOR DESIGN).

PARAMETER	TPH-1148	ARCADENE 360B	Δ
I^{0}_{sps} (lbf-sec/lbm)	261.9	263.1	+1.2
DENSITY (lbm/in^3)	0.0635	0.0650	+0.0015
VOLUMETRIC SPECIFIC IMPULSE ($lbf\text{-}sec/in^3$)	16.63	17.09	+0.46
VACUUM SPECIFIC IMPULSE (lbf-sec/lbm)	268.5	269.7	+1.2
PROPELLANT WEIGHT (lb) (NO INSULATION CHANGE)	1,110,136	1,135,311	+25,175
INCREASE IN INSULATION WEIGHT (lb)	———	2,000	+2,000
NOT INCREASE IN PROPELLENT WEIGHT (lb)	———	———	+22,340
Δ PAYLOAD (lb)	———	———	+1490 *
SCORING RANK	7	10	———

* + 0.083 (22,340) - 0.182 (2000) = 1490

TABLE 2.5.7. LINER TRADE.

FORMULATION VARIATION	EXISTING FORMULATION	CHANGE ONLY FIBROUS FILLER		CHANGE MAIN INGREDIENTS AND FIBROUS FILLER; POSSIBLY USE BARRIER	
ADVANTAGES	HIGHLY RELIABLE	BONDING TO V44 & TP-H1148 ESTABLISHED; AGING BEHAVIOR KNOWN		MAY BE REQUIRED TO BOND TO NEW INSULATION; BARRIER ALLOWS WEIGHT SAVINGS PAYLOAD (+296 lbs)	
DISADVANTAGES	NOT ACCEPTABLE DUE TO ASBESTOS; THICK LINER CAUSES INERT WEIGHT PENALTY	NEED TO DEFINE PROCESSIBILITY & CURE CHARACTERISTICS, AND VERIFY BONDLINE PERFORMANCE		EXTENSIVE TESTING REQUIRED TO QUALIFY NEW SYSTEM	
RATING CRITERIA		SCORE	WEIGHTED VALUE	SCORE	WEIGHTED VALUE
RELIABILITY		7	4.20	4	2.40
HAZARDS		--	--	--	--
PAYLOAD IMPACT		7	1.05	8	1.20
COST		7	0.70	4	0.40
TOTAL			5.95		4.00

2.6 NOZZLE TRADE STUDIES

2.6.1 INTRODUCTION

The primary goal of the nozzle portion of the Block II study program is to significantly increase the reliability of the nozzle assembly and attachments by utilizing well developed state-of-the-art technologies. A secondary goal is to increase motor performance so as to offset weight penalties incurred in maximizing the reliability of the overall motor design, when this is consistent with the primary goal. The approach chosen to accomplish this is based on retaining the existing nozzle concept. This means every attempt will be made to minimize the impact on the flex bearing, the nozzle flow surface contour, and the TVC system. The existing metal parts will be used if possible, but where changes are necessary, minimum impact on forging dimensions will be attempted. The nozzle study will, therefore, investigate alternative design approaches where justified according to the following criteria:

- Where there is a demonstrated problem area requiring an improvement in reliability/ flight safety;
- If an existing technology can be incorporated that improves reliability/flight safety or increases performance with no loss in reliability/flight safety;
- If the Block II SRM design dictates a nozzle design change.

Based on this approach, it was decided to concentrate the initial effort on trade studies for the nozzle-to-case attachment joint and the nozzle liner material. The joint study considers both the metal hardware and the insulation with emphasis on the sealing aspects of the assembly. Alternative inlet/throat liner materials are being investigated primarily to further reduce or hopefully eliminate the potential for "pocketing" erosion such as that experienced on STS-8. These studies are discussed and preliminary results are presented in this interim report. The hardware joint study is included in this section, while the insulation joint study is in the insulation section.

Also discussed, in Section 3.3 of this report, is a design study conducted for the five nozzle subassembly joints which currently have simplex seals. Alternative designs were investigated which meet the Block II redundancy and verifiability requirements. All the data necessary to complete these studies have not yet been generated. However, the progress to date is reported and the anticipated choices are indicated for the purpose of providing a preliminary design. In addition, studies are also being conducted in several areas which are not addressed in this interim report. All asbestos-containing materials will be eliminated from the Block II design. Replacement materials will be found for the several adhesives and gap fillers used in the nozzle assembly and the elastomer used in the radiation shield. Since a design change is required, alternate concepts for the radiation shield will be considered. The use of improved liner materials will also be investigated for all areas of the nozzle. The complete results of all these studies will be fully documented in the final report.

2.6.2 NOZZLE-TO-CASE JOINT HARDWARE

This section presents the results completed to date and a status of the trade study on the SRM Nozzle-to-Case Joint hardware (herein called the NTCJ). Three alternate NTCJ concepts were selected as possible candidates. These candidate designs, along with the current redesigned NTCJ, will be rated according to a set of evaluation criteria in order to select an optimum NTCJ design.

A preliminary design has been selected based upon the work completed to date. The final selection will be made once the trade study has been completed.

The objective of the trade study is to improve the reliability of the NTCJ design. An emphasis is being placed on the sealing mechanism of the NTCJ design in considering any improvement in the reliability.

The critical objective to improve the reliability is to obtain a sealing mechanism that would be independent of (or not highly effected by) the response of the NTCJ to the required loading conditions or the manufacturing process. This will be accomplished by both making changes to the hardware configuration and changing the position of the primary and secondary seals.

The three alternate NTCJ designs, along with the current baseline redesign, are shown in Figure 2.6.1. These new design concepts shown in the figure are the 1) dual face seal, 2) shear retention lip, and 3) capture latch.

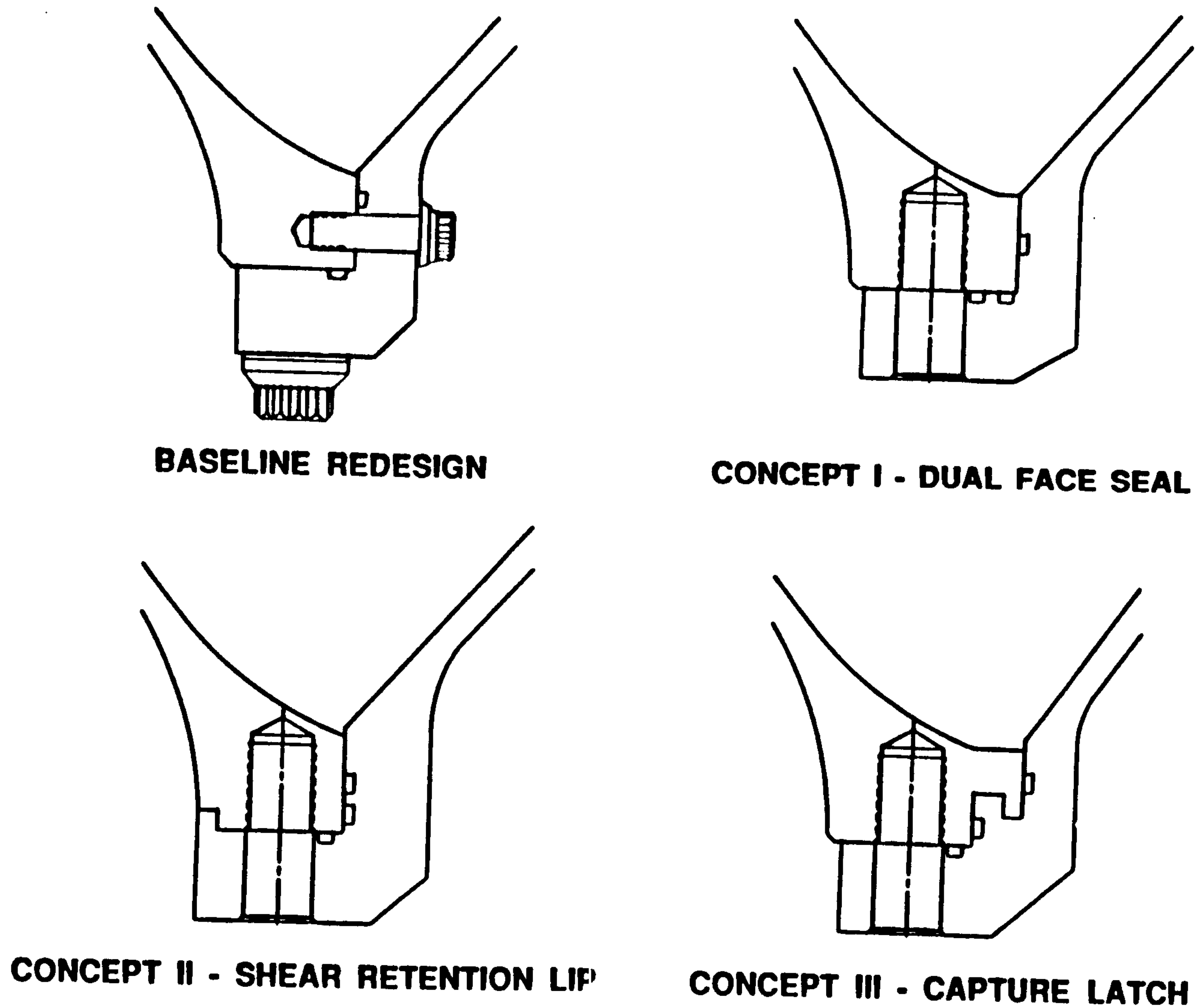


FIGURE 2.6.1. NOZZLE-TO-CASE JOINT CONCEPTS.

The dual face seal concept is similar to the original NTCJ with one major exception, the primary and secondary seals are now both face seals. The width of the flange mating surface was increased to accommodate the second seal. Since no bore seal is involved, extremely close radial tolerances are not required.

The shear retention lip concept is a two-seal type configuration; utilizing both a face and a bore seal. A shear lip has been incorporated on the nozzle component of the NTCJ. This feature minimizes the relative radial motion between the nozzle and case flanges ("rounding"). This rounding is a major contributor to gap opening. In addition, the primary seal has been moved in an attempt to make the seal gap less dependent on joint rotation.

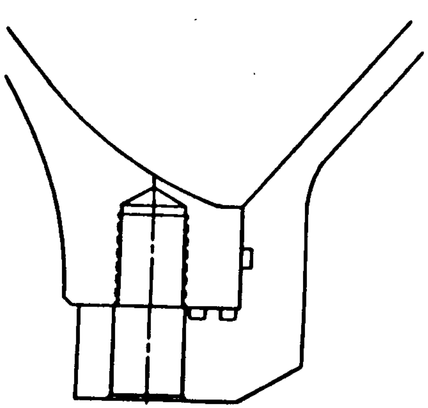
The capture latch concept is also a two-seal type concept, and has interlocking tabs on both the nozzle and case components in order to reduce gap opening due to "rounding." These tabs also serve to reduce gap opening due to joint rotation.

Each of these concepts also have some disadvantages associated with them. A description of the advantages and disadvantages associated with each concept is shown in Figure 2.6.2. All three configurations utilize a third seal, so that the primary seal function can be verified in the direction of operation.

The evaluation criteria to be used are summarized in Table 2.6.1. As shown, it consists of three areas, namely; reliability, performance, and cost. These three areas were selected due to their importance to the NTCJ design. The relative emphasis of the three criteria in the design is shown by their weighting factors. The reliability has obviously been assigned the highest weighting factor. The motor performance rating will be based on inert weight impact. Cost evaluation will include the impact of new forging tooling if this is required.

The reliability of each of the concepts will be estimated by the use of finite element analysis and conventional strength of materials analysis. The performance and cost criteria will then be evaluated once a reliable structural design has been determined.

Concept I has been selected as the preliminary design (Figure 2.6.3). The main advantage of this design over the other concepts is that it is not as complex. In



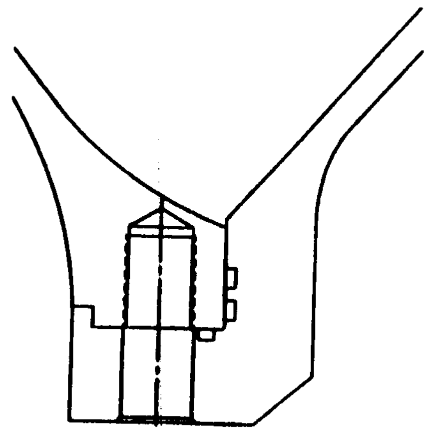
CONCEPT I - DUAL FACE SEAL

ADVANTAGES

- Reduces gap opening due to joint rotation
- No additional bolts or seals required
- Close radial tolerances not required due to face seal feature
- Primary and secondary seal preloaded due to face seal configuration

DISADVANTAGES

- Possible increase in inert weight
- Initial cost impact to change design
- Single seal type less redundant



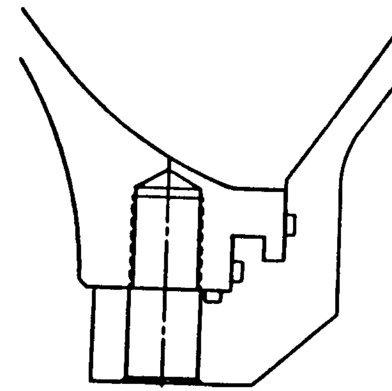
CONCEPT II - SHEAR RETENTION LIP

ADVANTAGES

- Shear lip reduces effect of gap tolerances on gap opening
- Primary seal location to minimize effect of joint rotation on gap opening
- Utilizes two seal types for redundancy
- No additional bolts or seals required

DISADVANTAGES

- Possible increase in inert weight
- Close tolerances required
- Initial cost impact to change design



CONCEPT III - CAPTURE LATCH

ADVANTAGES

- Capture latch reduces effect of gap tolerances on gap opening
- Capture latch reduces effect of joint rotation on gap opening
- Utilizes two seal types for redundancy
- No additional bolts or seals required

DISADVANTAGES

- Complex design
- Close tolerances required
- Possible increase in inert weight
- Initial cost impact to change design

FIGURE 2.6.2. NOZZLE-TO-CASE JOINT CONCEPT EVALUATION.

TABLE 2.6.1. EVALUATION CRITERIA FOR NOZZLE-TO-CASE JOINT STUDY.

- RELIABILITY (65 PERCENT).
  - GAP OPENING DUE TO JOINT ROTATION.
  - GAP OPENING DUE TO "ROUNDING" (TOLERANCES).
  - POTENTIAL FOR SEAL DAMAGE.
- PERFORMANCE (15 PERCENT).
  - INERT WEIGHT IMPACT.
- COST (20 PERCENT).
  - NONRECURRING (NEW FORGINGS).
  - RECURRING.

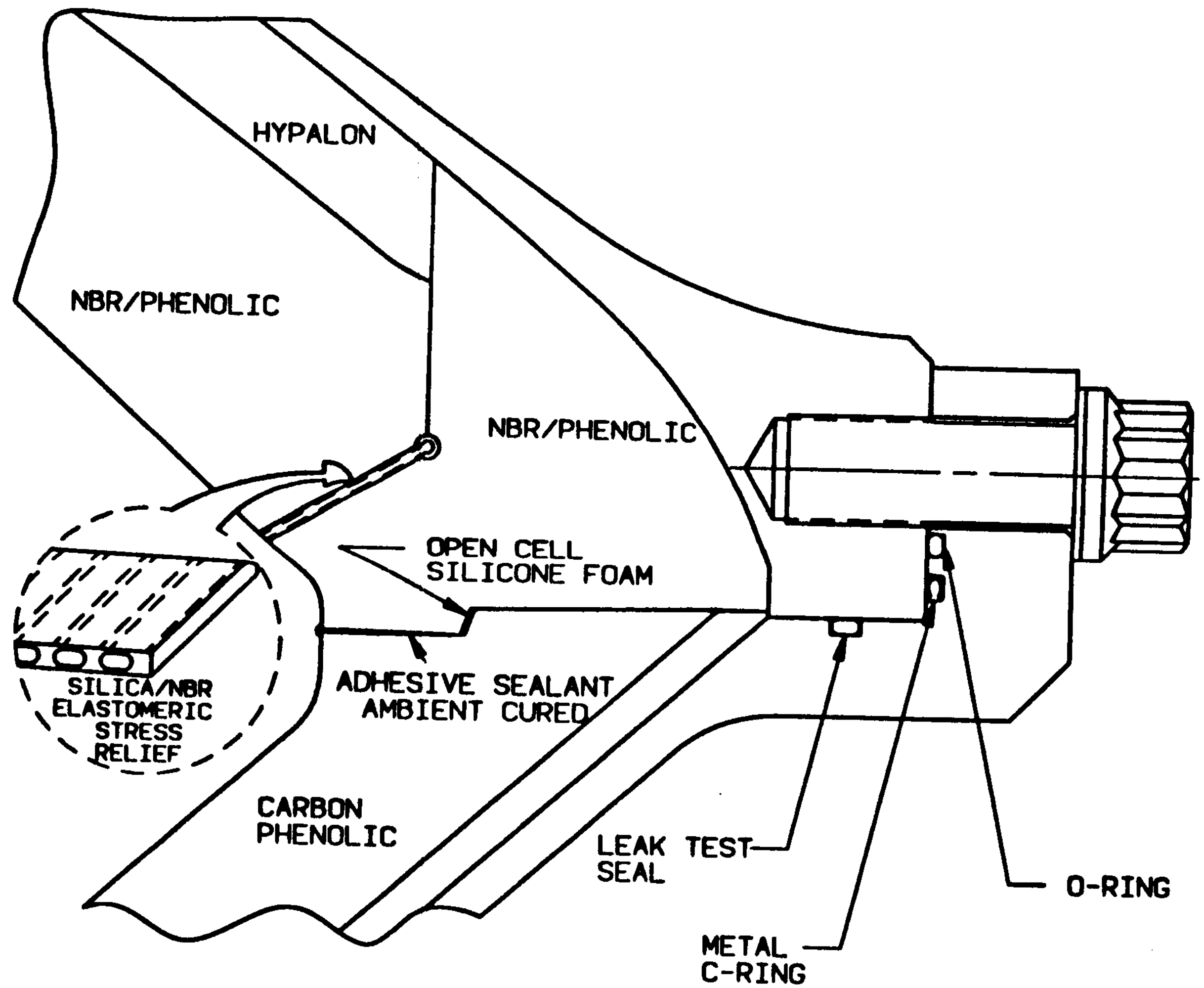


FIGURE 2.6.3. NOZZLE-TO-CASE JOINT PRELIMINARY DESIGN.

addition, the manufacturing and assembly of the NTCJ is simplier (not as close dimensional tolerances required). The final design chosen will probably be similar to Concept I.

The trade study, as far as performance and cost is concerned, still needs to be completed. Once this has been completed and a final design selected, an in-depth structural analysis will need to be done to ensure the structural integrity of the NTCJ. This analysis will have to take into consideration the preload on the bolts, dimensional tolerancing, and required loading conditions in order to properly establish the NTCJ reliability, i.e., seal mechanism.

Final performance and cost numbers will also be generated for the final design.

2.6.3 NOZZLE THROAT LINER MATERIALS

Liners used on the current SRM nozzle (Figure 2.6.4) are fabricated from rayon-based carbon cloth/phenolic tape. The performance of this liner material has been adequate with the notable exception of anomolous gouging and "pocketing" erosion, which was first noted in the STS-8A nozzle forward nose ring and aft inlet ring, as shown in Figure 2.6.4. The nozzle inlet liners on this motor and several subsequent flight motors have failed to meet the 2X requirement on erosion. NASA and the current nozzle fabricator have spent considerable time and effort studying this problem, and as a result of stringent material and processing controls and revised processing techniques, have virtually eliminated the incidence of pocketing erosion in the last six successful flights. However, although a number of items have been identified as potential contributing factors, the problem is not considered solved since the failure mechanisms are not fully understood, and several studies are still underway to evaluate all facets of the rayon-based material's manufacture and response to the operating environment which might relate to the problem.

The Block II study program offers the opportunity to investigate more extensive changes to the inlet/throat liner (Figure 2.6.5) than have previously been considered to solve the pocketing erosion problem. Therefore, a trade study was initiated to evaluate two types of material changes. The first is a direct substitution where liners would be fabricated using the existing tape wrap techniques but with a

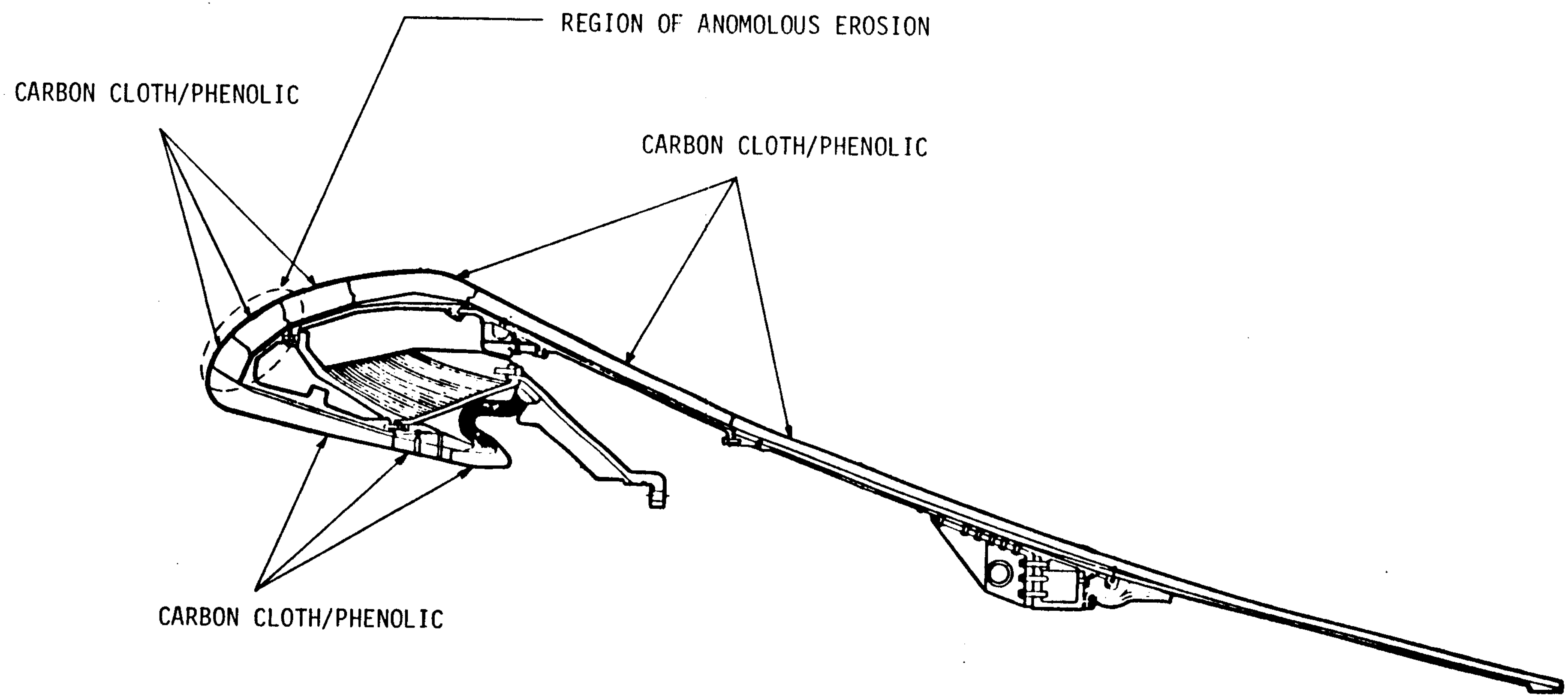


FIGURE 2.6.4. SRM NOZZLE LINER.

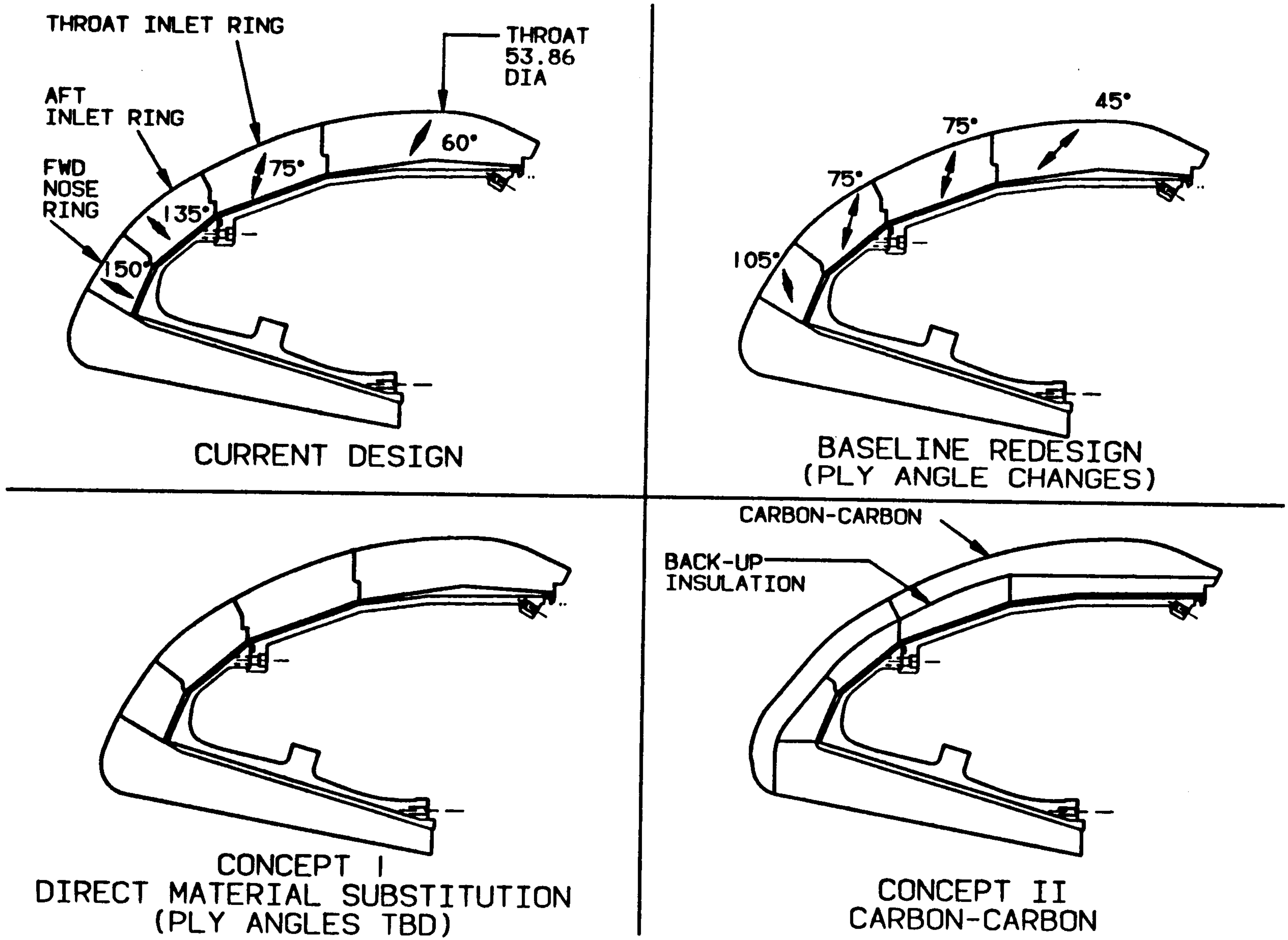


FIGURE 2.6.5. INLET/THROAT LINER CONCEPTS.

different material, while the second is a more extensive material change requiring a different fabrication technique. Several test programs evaluating alternative tape wrapped ablative liner materials were conducted prior to the STS-8 flight. The materials tested there were considered as potential candidates of the first type since measured erosion, char, and thermal performance data existed relating to SRM usage. These ablatives included polyacrylonitrile (PAN) based carbon cloth/phenolic and PAN or pitch based graphite cloth/phenolics. Since tests and analysis conducted as part of the STS-8 investigation indicated that graphite cloth/phenolic or PAN based materials offered increased resistance to pocketing erosion, these were chosen as the ablative materials to be included in this trade study. However, in many of the rocket motor nozzle designs currently being produced or developed, tape wrapped ablative liners have been supplanted by multidirectionally reinforced carbon-carbon materials for the inlet/throat region. While the data base for this relatively new class of materials is not yet as extensive as that for the tape wrapped ablative materials, it is rapidly growing. The inherent three dimensional strength of these materials would virtually eliminate pocketing erosion, and, therefore, carbon-carbon was included as the material of the second type. For the purposes of this initial evaluation, 3-D carbon-carbon was taken as typical of all architectures.

Table 2.6.2 gives a summary of the advantages and disadvantages for each of the three nozzle liner materials. As shown, all three appear to offer increased resistance to pocketing erosion. However, if any of these materials were chosen for a Block II SRM design, an extensive materials test program would have to be conducted to verify these characteristics and their reproducibility. All three also offer increased motor performance due to decreased erosion. Carbon-carbon offers the highest gains in both these areas, but also requires the most development. There are several significant issues associated with utilizing carbon-carbon in this application. Thermal requirements and physical constraints lead to a radius/thickness ratio $(\frac{R}{t})$ between 20 and 40, which is outside the region of current designs. Buckling due to restraint of the high temperature carbon-carbon liner thermal growth imposed by the low temperature steel housing becomes a concern. This could be aggravated by high backside pressure buildup if a degradable (charring) backup material such as carbon/phenolic is used, and gases due to pyrolosis are not adequately vented. Nondegradable backup materials are becoming available; however, their lower insulation properties would lead to higher housing temperatures at soak out unless overall liner/insulation thickness is increased. Evaluation of the carbon-carbon approach requires a more detailed design study than has

TABLE 2.6.2. NOZZLE INLET/THROAT LINER ALTERNATE MATERIALS.

MATERIAL	ADVANTAGES	DISADVANTAGES
• PAN BASED CLOTH/PHENOLIC	• REDUCED POTENTIAL FOR POCKETING EROSION • REDUCED EROSION RATE • COMPARABLE COST • MODERATE CHAR INCREASE • WILL FIT WITHIN CURRENT MOLD LINES	• SOME DATA BASE EXPANSION REQUIRED • SLIGHTLY REDUCED THERMAL MARGIN WITHIN CURRENT MOLD LINES
• PITCH BASED CLOTH/PHENOLIC	• REDUCED POTENTIAL FOR POCKETING EROSION • REDUCED EROSION RATE • COMPARABLE COST	• SOME DATA BASE EXPANSION REQUIRED • SIGNIFICANT CHAR INCREASE • WILL NOT FIT WITHIN CURRENT MOLD LINES • MAY NOT BE ABLE TO ACHIEVE REQUIRED THERMAL MARGIN WITHIN CURRENT CONCEPT
• CARBON/CARBON	• REDUCED POTENTIAL FOR POCKETING EROSION • REDUCED EROSION RATE • HIGH THERMO-STRUCTURAL CAPABILITY	• SIGNIFICANT DATA BASE EXPANSION REQUIRED • MAY NOT FIT WITHIN CURRENT MOLD LINES • INCREASED COST • INCREASED CHAR DEPTH IF DEGRADABLE BACK-UP USED • BACKSIDE PRESSURE BUILDUP FOR DEGRADABLE BACKUP • REDUCED THERMAL MARGIN IF NONDEGRADABLE BACKUP USED • HIGH R/T LEADS TO THERMAL BUCKLING CONCERN

been completed at this point. It is recommended, therefore, that this concept be carried along as an alternative through the end of the study program so that a full evaluation can be included in the final report. The most significant disadvantage faced by any alternative material is the overwhelming data base which exists for the current material. It is questionable whether any of the alternative's potential benefits overcomes this advantage in light of the requirement that any material used on the SRM must have an extensive data base comparable to the current materials.

The final material choice will be based on a trade study conducted, as shown in Table 2.6.3. The relative importance of the major study factors, reliability, performance, and cost, are shown by the associated weighting factors: 65, 15, and 20 percent, respectively. Reliability will be judged primarily on the reduced potential for pocketing erosion. Also considered will be the reproducibility of the material, and, in particular, process sensitivity which has been shown to be a problem with current material. Thermal margins and structural integrity will also be evaluated to minimize the possibility of introducing any new problems. The specific aspects of performance and cost, as shown in the table, are self-explanatory.

For the purposes of establishing a preliminary design, the anticipated material choice was determined to be the PAN based cloth/phenolic. This material appears to have increased resistance to pocketing erosion, based on 40 pound charge motor subscale nozzle firings, and also has reduced throat erosion which increases motor performance. No significant risks have been identified for this material, with the only major drawback being the significant test program required to provide the necessary data base. The final decision requires detail review of the referenced subscale firings and a cost estimate for the data base test program. It appears at this time that the char depth for the pitch based cloth/phenolic is sufficiently large to preclude it from practical consideration. As mentioned previously, carbon-carbon will continue to be studied as an alternative.

TABLE 2.6.3. NOZZLE INLET/THROAT LINER MATERIAL STUDY.

- RELIABILITY (65 PERCENT).
  - POTENTIAL FOR "POCKETING EROSION".
  - REPRODUCIBILITY AND PROCESS SENSITIVITY.
  - INSPECTABILITY.
  - THERMAL MARGINS.
  - STRUCTURAL INTEGRITY.
- PERFORMANCE (15 PERCENT).
  - INERT WEIGHT IMPACT.
  - THROAT EROSION.
- COST (20 PERCENT).
  - TEST PROGRAM TO DEVELOP DATA BASE.
  - FACILITIES IMPACT.
  - MATERIAL COST.
  - PROCESSING COST.
  - SCRAPPAGE.

2.7 TRANSPORTATION, HANDLING AND ASSEMBLY

2.7.1 TRANSPORTATION

2.7.1.1 SEGMENTED MOTOR

The shipment of SRM motor segments presents no particular problems. The railcars used for shipping are already existing and would require no capitalization. ARC assumes that the yearly usage (fixed lease) cost for each cross-country car will be $60,000. Estimated transportation costs are:

- Loaded SRM from Camden, Arkansas to KSC - $115,000/SRM
- Loaded SRM from Camden to MTI/Wasatch - $140,000/SRM

Table 2.7.1 is the proposed routing for the SRM segments. These routes were based on cost effectiveness and clearances (side and vertical). Special emphasis was placed on the conditions of the tracks and road bed such that the transportation acceleration limits specified for Shuttle SRM segments will not be exceeded. This routing is only preliminary, based on information received from the railroads' traffic managers. Minor changes may be required to improve cost and schedule at various times.

2.7.1.2 MONOLITHIC MOTOR

Two transportation modes were considered for the monolithic motor - rail and barge. Rail transport proved to be unsatisfactory. Rail shipping failed on several issues including safety, cost, and regulatory issues.

The length of the motors and their attending rail cars make curves, trackside obstacles, railyards, and traffic on adjacent tracks very difficult to negotiate. The integrity of the cargo would seriously be jeopardized by this.

The propulsive nature of the motor is reason for concern. The estimated range of the motor is 300 to 500 miles.

There are many regulatory uncertainties and obstacles due to the size, weight, hazardous nature, and special rail cars of the monolithic motor. The estimated

TABLE 2.7.1. PROPOSED RAIL ROUTING OF SRM HARDWARE.

CAMDEN, AR TO KSC

LOAD ON EAST CAMDEN AND HIGHLAND RR AND SWITCH TO SOUTHERN PACIFIC AT CAMDEN

SOUTHERN PACIFIC TO NEW ORLEANS VIA SHREVEPORT

FLORIDA EAST COAST RR TO VAB/KSC

CAMDEN, AR TO MTI/CORRINE, UTAH

LOAD ON EAST CAMDEN AND HIGHLAND RR AND SWITCH TO SOUTHERN PACIFIC AT CAMDEN

SOUTHERN PACIFIC TO KC SOUTHERN AT SHREVEPORT, LA

KC SOUTHERN TO N. PLATTE, NEB VIA KANSAS CITY

UNION PACIFIC FROM N. PLATTE TO CORINNE, UT

MTI/CORINNE, UT TO CAMDEN, AR

THE REVERSE OF THE ROUTING FROM CAMDEN TO MTI AS DESCRIBED ABOVE WILL BE UTILIZED time to get approval of the special rail cars would be two years. This is because the routes use more than one rail line. To facilitate rail transport of these motors would require the acquisition (by ARC or NASA) of custom-built rail cars. Estimates for these cars ranged from 6 to 10 million dollars apiece. Approximately 10 would be required.

In addition to the cost of rolling stock required, substantial improvement would be necessary for the rails, rail beds, and bridges along the route. This would be a multimillion dollar undertaking.

Waterborne transport is the only acceptable transportation mode for the monolithic motor. Custom-made barges would be necessary. The barges need to be able to load a rail car, dock with and off/on load to/from a Poseidon/Orion type seagoing transport and possibly be required to provide some degree of bullet-proof protection.

Routing from Camden to New Orleans would be along the Ouachita-Tensas-Mississippi Rivers. The barge would be off-loaded at New Orleans to a sea-going transport similar to the Orion or Poseidon and transported to KSC across the Gulf of Mexico. Round trip Camden-New Orleans would be an estimated 14 days. Round-trip New Orleans-KSC would be an estimated 12 days.

Although the safety record is much better for waterborne transportation than rail, there are still some serious safety concerns. The described route passes through fewer population centers than rail. It does, however, pass through Monroe, Baton Rouge, and New Orleans.

The use of water transport would require major construction at the riverside facilities of Camden and at KSC. In addition to this port facilitization cost, rail construction would be necessary for the approximately eight miles between the ARC plant and the Ouachita River. The large, expensive rail cars would still be necessary. The barges themselves would probably cost about four million dollars apiece. At least eight would be required to keep the flight schedule. It is unclear whether the Poseidon and Orion would be able to schedule SRM deliveries because of their ET delivery commitments. If not, that would be an additional large expenditure.

A monolithic motor would require an additional test stand because the two in Utah (one existing, one proposed) could not be accessed by barge. A likely site would be Complex 37 or Complex 34 at KSC.

2.7.2 HANDLING

2.7.2.1 SEGMENTED MOTOR

Handling motor segments presents no real problem. The tooling, facilities, equipment, and procedures are well known and established.

ARC's Camden plant would need to build lifting houses with 220-ton cranes. These would lift segments out of the casting pits and handle the tooling and load vehicles.

Much of the horizontal interior trafficking of motors would be handled by airborne palleting. Exterior movement would be done by rubber-tired vehicles.

Segment handling at KSC would require no additional expenditure. KSC currently handles motor segments without any problems.

2.7.2.2 MONOLITHIC MOTORS

Handling a monolithic motor the size of an SRM would require significant expenditure for equipment, tooling, and facilities. Moreover, the procedure for such an operation is not established and not well-defined.

The cranes to lift the motors out of the casting pits and break the motor over would have to have a 1000-ton capacity. The cost of the larger cranes is many times more than would be required for segmented motors. The lifthouse would necessarily be tall enough to accommodate an SRM in the vertical position. Such a building would be 13 to 14 stories tall.

It would be necessary to provide a strong back frame for the motor casing. This is to prevent it from deflecting during breakover operations. The additional weight and size would require larger cranes, casting pits, rail cars, etc.; thus, significantly increasing the cost.

Intraplant movement of the motor would require the use of the rail cars previously discussed. Rail cars of this magnitude would require 135-pound rails and comparable rail beds. Some handling would be done by airborne pallets.

Handling techniques and procedures would need to be developed and perfected. During the operation, lifting from the casting pit, the motor would be about 12 to 13 stories above the bottom of the pit. Many more safeguards would have to be employed.

Tooling used to handle all phases of monolithic motor production is many magnitudes larger and stouter. Again, this issue reflects itself in overall cost.

2.7.3 ASSEMBLY

2.7.3.1 SEGMENTED MOTOR

The production method for motor segments would be very similar to the methods and procedures currently employed. There is the possibility of making parts of the process more efficient than currently practiced. This could be achieved by combining some operations and even eliminating others. The general procedure is well known and proven.

2.7.3.2 MONOLITHIC MOTOR

The production of single grained motors is an entirely new procedure. Not only are there many uncertainties, but the overall cost for facilities is much more.

For example, the length of the motor would necessitate either a deep enough hole below grade or building above grade to accommodate the casting procedures. A pit that deep would need to be very heavily constructed because of the water seepage problems and weight of the walls required. Going above ground would require heavy steel structuring with much of the processing in the upper floors.

2.7.4 ENVIRONMENTAL REGULATIONS IMPACT

ARC considered three sites for SRM production in the early phases of this study. They were at KSC, a new Florida facility, and ARC's Highland Industries plant in Camden, Arkansas.

An assessment of environmental laws and concerns revealed several potential problems. An analysis of each follows:

- A permit under the Resource Conservation and Recovery Act (RCRA) is very difficult to acquire. There is currently no process by which a permit can be obtained for open pit burning of waste propellant.

- A permit under the Clean Air Act will be a problem in certain areas of Florida. Those areas have exceeded their allowable particulate levels.

- Permitting under the Clean Water Act covering water run-off will be difficult but not impossible.

- The disposal of liner and insulation material is a minor concern provided there are no hazardous materials involved.

- Florida's rigorous ground water protection regulations seriously hinder the feasibility of a Florida site.

ARC's facility has distinct advantages over the other two sites. The first two years of the project could be conducted under ARC's existing RCRA permit before exceeding the permit's disposal capacity. Likewise, the first year of production can be conducted under an existing air permit before exceeding permitted limits. The air permit can be modified without inordinate problems. These advantages should allow the Camden facility to come on line while the other two sites would be tied up in permitting delays. A new RCRA permit would take at least two and possibly as many as four years, depending on the progress of the new Subpart X regulation soon to be proposed. ARC should be able to increase its Clean Air Act permitted capacity in about eight months because the permit is current and would require only modifications.

2.7.5 CONCLUSIONS

Table 2.7.2 has been prepared as a comparison of tooling and facility cost between segmented and monolithic motors. The difference in facilities cost is approximately $308,225,000. There are serious transportation and manufacturing uncertainties. Paragraph 2.7.4 outlines some of the environmental issues involved in relocating from Arkansas. All of these factors overwhelmingly point to a segmented motor as the better choice.

TABLE 2.7.2 INCREASE IN NON-RECURRING COSTS FOR MONOLITHIC VS. SEGMENTED MOTORS.

	DIFFERENCE (K$)
MANUFACTURING	
FACILITIES	$ 61,460
TOOLING	48,840
TRANSPORTATION	
FACILITIES	$128,000
TOOLING	10,825
MISSION ASSEMBLY	
FACILITIES	$ 30,000
TOOLING	550
TEST FIRING	
FACILITIES	$ 26,500
TOOLING	2,050
TOTAL	
FACILITIES	$245,960
TOOLING	62,265
GRAND TOTAL DIFFERENCE	**$308,225**

3.0 DESIGN STUDIES

3.1 JOINT DESIGN STUDIES - STRUCTURAL ANALYSIS

This section addresses the detailed structural analysis of the ARC axial bolted joint design concept. For evaluation of the tang and clevis, the capture feature and the LARC in-line bolted configurations, the NASA/Langley structural analysis reports of References 1 and 2 were used. For completeness, the latest results from the NASA/Langley in-line bolted joint analysis are included.

Design Conditions

Loads and requirements are listed in Table 3.1.1. The listed bending moment of 68 x 10^6 in-lb is the updated value from Reference 3, and applies to the axial station of the upper field joint (851 inches).

Configuration

The sketch of Figure 3.1.1 shows the main features of the ARC joint. This first-cut design contains 160 one-inch diameter studs screwed into tapped holes in the lower segment flange. Metal between the stud holes is milled out as shown to form a fluted segment end. The top segment flange contains 160 alcoves and is very similar to the NASA/Langley in-line design. The membrane shells are tapered for approximatley seven inches leading into each flange, thus lessening the effect of the discontinuity. Standard hex nuts seated on spherical washers are used to minimize any bending of the studs induced by flange rotations. The lower segment flange is extended inboard to provide a seating surface for the two face seals and a shear lip. The latter ensures rounding during assembly. The seals are an Inconel jacketed, graphite foil filled gasket as a primary seal, and an Inconel c-ring as a backup seal. This arrangement ensures against leak check damage (the gasket will not shift position), and provides resistance to high temperature exposure.

Structural Model

Figure 3.1.2 shows the NASTRAN finite element model. Figure 3.1.2.A shows a PATRAN development of the model around part of the circumference of the booster

TABLE 3.1.1. JOINT AND SEAL DESIGN CONDITIONS.

- MEOP = 1004 PSIG

- ULTIMATE SAFETY FACTOR = 1.4

- YIELD SAFETY FACTOR = 1.2

- MAXIMUM MOMENT AT UPPER FIELD JOINT, $M = 68 \times 10^6$ IN-LB

- EQUIVALENT AXIAL LOAD:

  $$W_{eq} = P \pi R^2 + \frac{2M}{R}$$

  AT R = 72.0", P = 1004 PSIG

  $$W_{eq} = 18.26 \times 10^6 \text{ LB}$$

- MATERIAL

  CASE ULTIMATE = 195 KSI

  CASE YIELD = 180 KSI

  BOLT/STUD ULTIMATE = 260 KSI

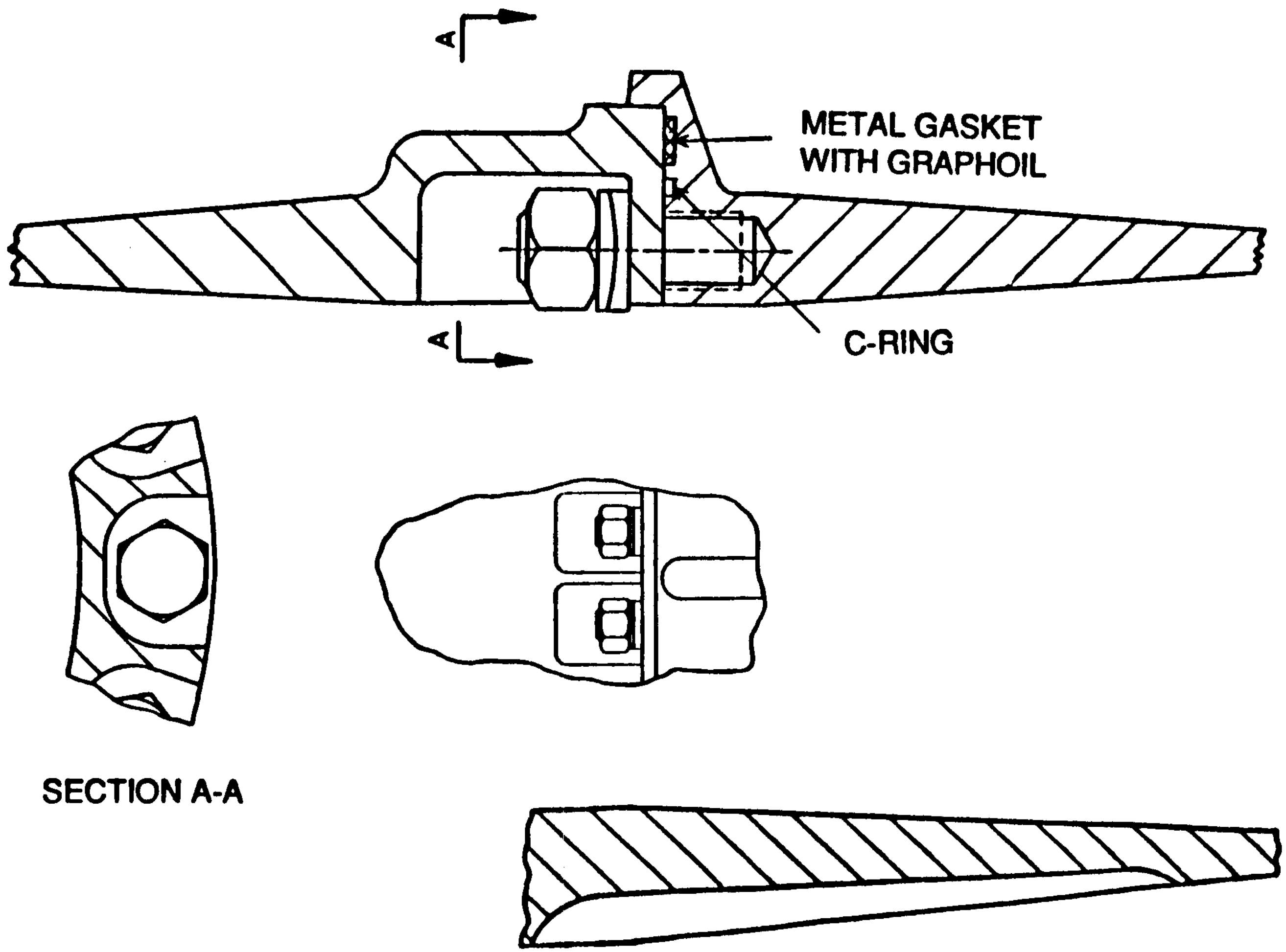


FIGURE 3.1.1. ARC AXIAL BOLTED JOINT.

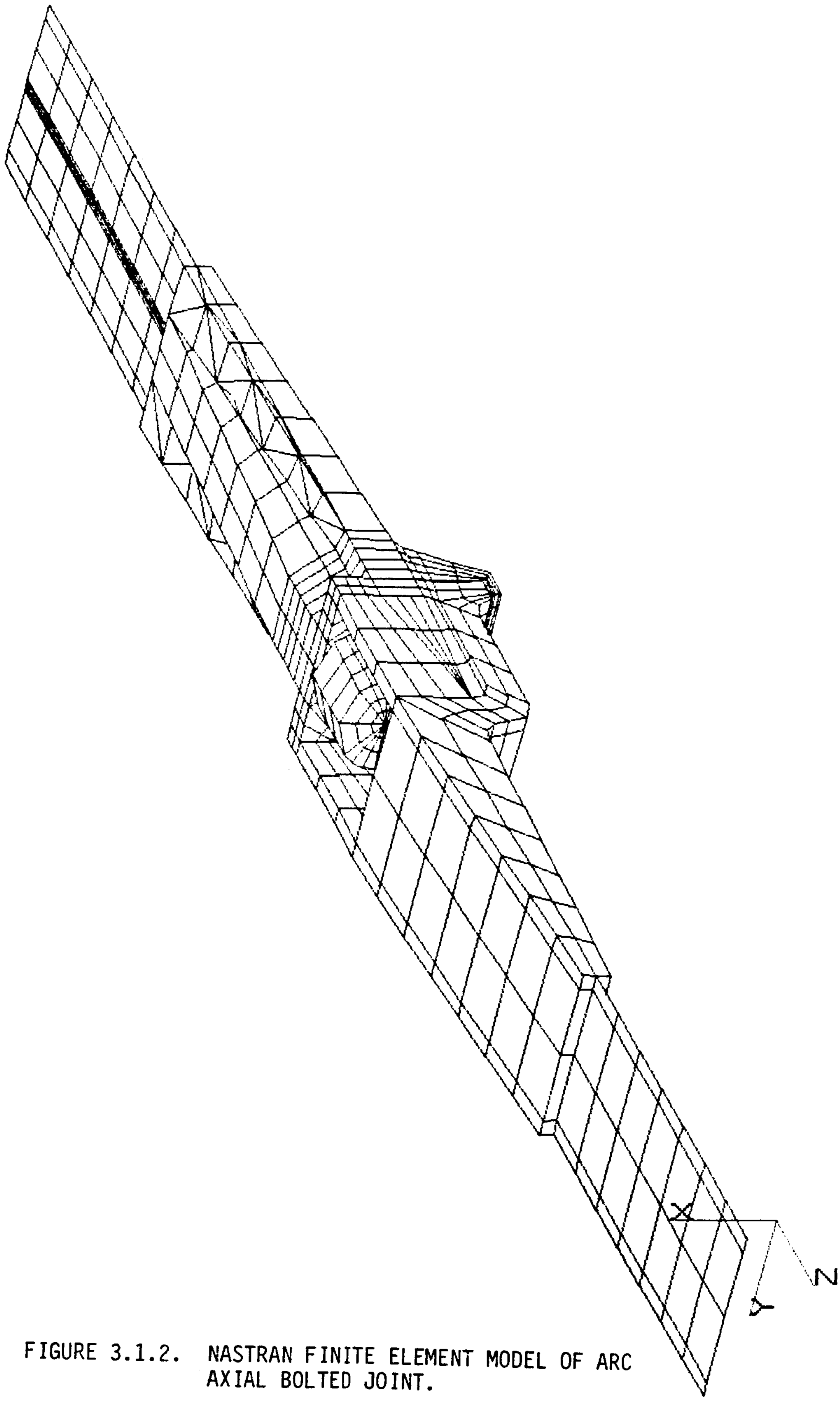


FIGURE 3.1.2. NASTRAN FINITE ELEMENT MODEL OF ARC AXIAL BOLTED JOINT.

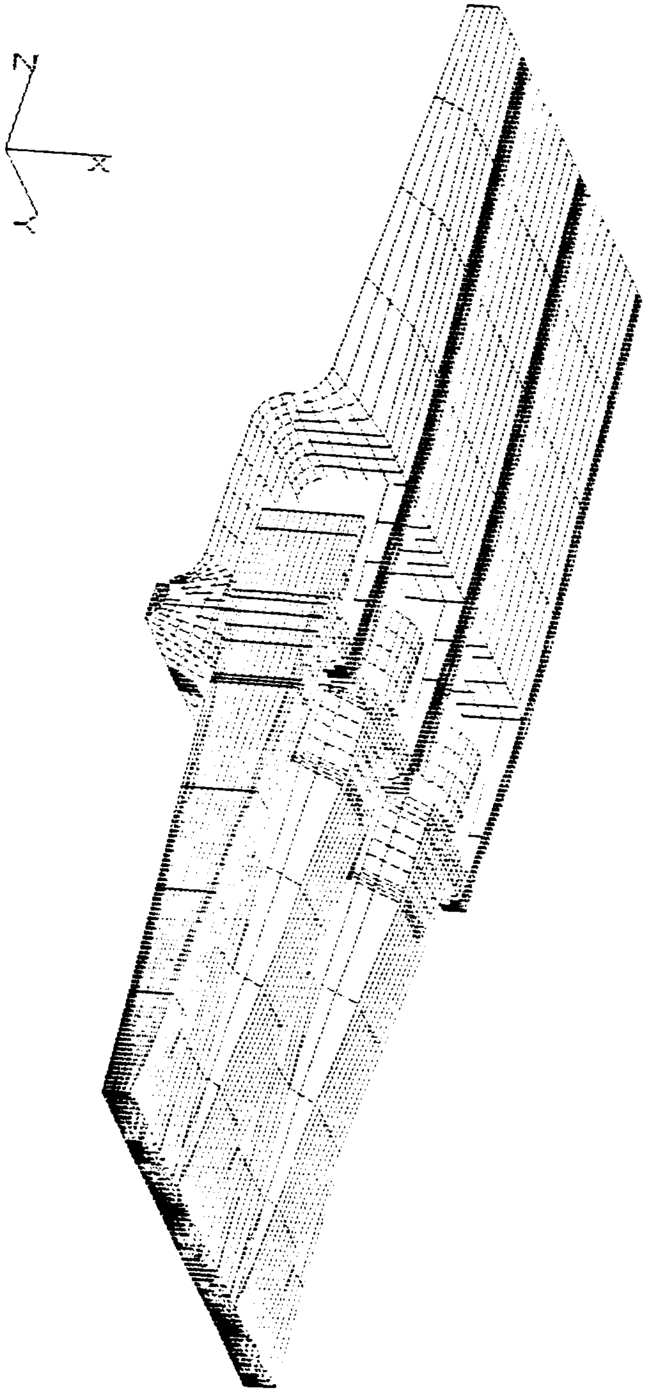


FIGURE 3.1.2-A. PATRAN DEVELOPED VIEW OF ARC AXIAL BOLTED JOINT - 5.625 DEGREES OF CIRCUMFERENCE.

case. The membrane portions of the upper and lower segments were modeled using shell elements, and tied into the solid brick elements using rigid bars. We note that the finite element model contains somewhat less material in and around the "alcove" gussets and roof than does the actual design of Figure 3.1.1. Therefore, the first-cut analysis will be conservative as regards deflection and stresses and nonconservative with respect to joint weight. The solution is nonlinear, insofar as this is a contact problem with contact elements between the nut and the alcove flange surface, and between the alcove flange lower surface and the lower segment flange surface.

Pressure and axial loads were applied to the model as per Table 3.1.1. Additionally, the elements representing the stud were assigned a thermal shrinkage which induced a preload equal to approximately 70 percent of the ultimate strength of the stud.

Results and Conclusions

Deflection results are shown in Figures 3.1.3 through 3.1.5. The gap at the inboard edge of the sealing surfaces was approximately .0248 inches at 1004 psig. Stress contour plots are shown in Figures 3.1.6 through 3.1.10. High bending stresses are evident on the upper segment flange's lower surface, and small areas of high stress appear in and around the alcove gussets and the lower segment flange surface. Further summary items and conclusions are listed in Table 3.1.2. Clearly increased thicknesses are called for in a design iteration, and in lieu of an analysis iteration(s) the inert weight and propellant loss were penalized, as per Table 3.1.2.

Current results of the LARC structural analysis are shown in Figures 3.1.11 through 3.1.16. We note that the current model's gap opening is held to only 2(.001) = .002 inches at the gasket locations (Figure 3.1.6). Table 3.1.3 summarizes the salient results.

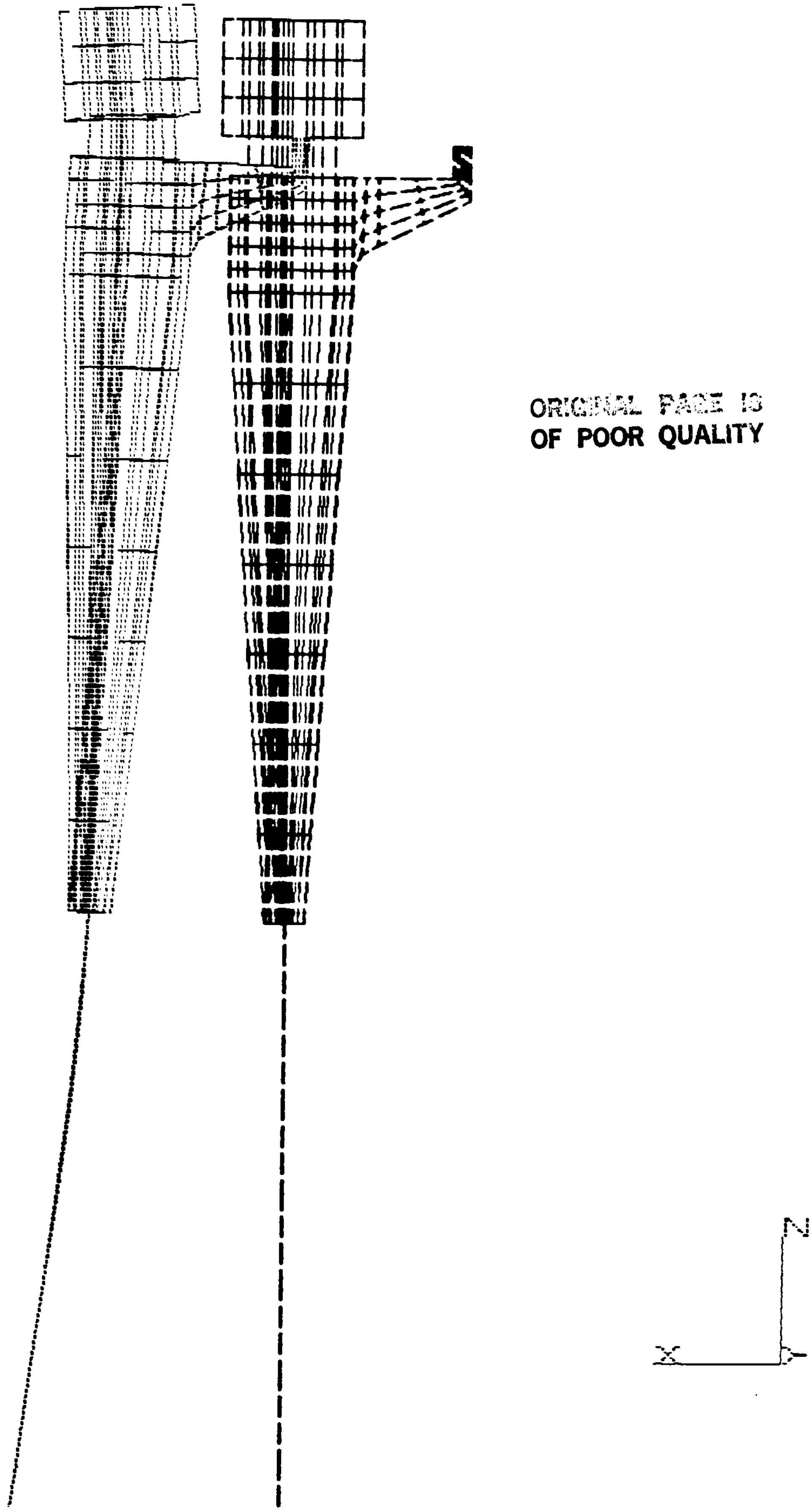


FIGURE 3.1.3. DEFORMED GRID - LOWER FLANGE AND SHELL.
INTERNAL PRESSURE AND AXIAL LOAD.

DEFORMATION NOT TO SCALE.

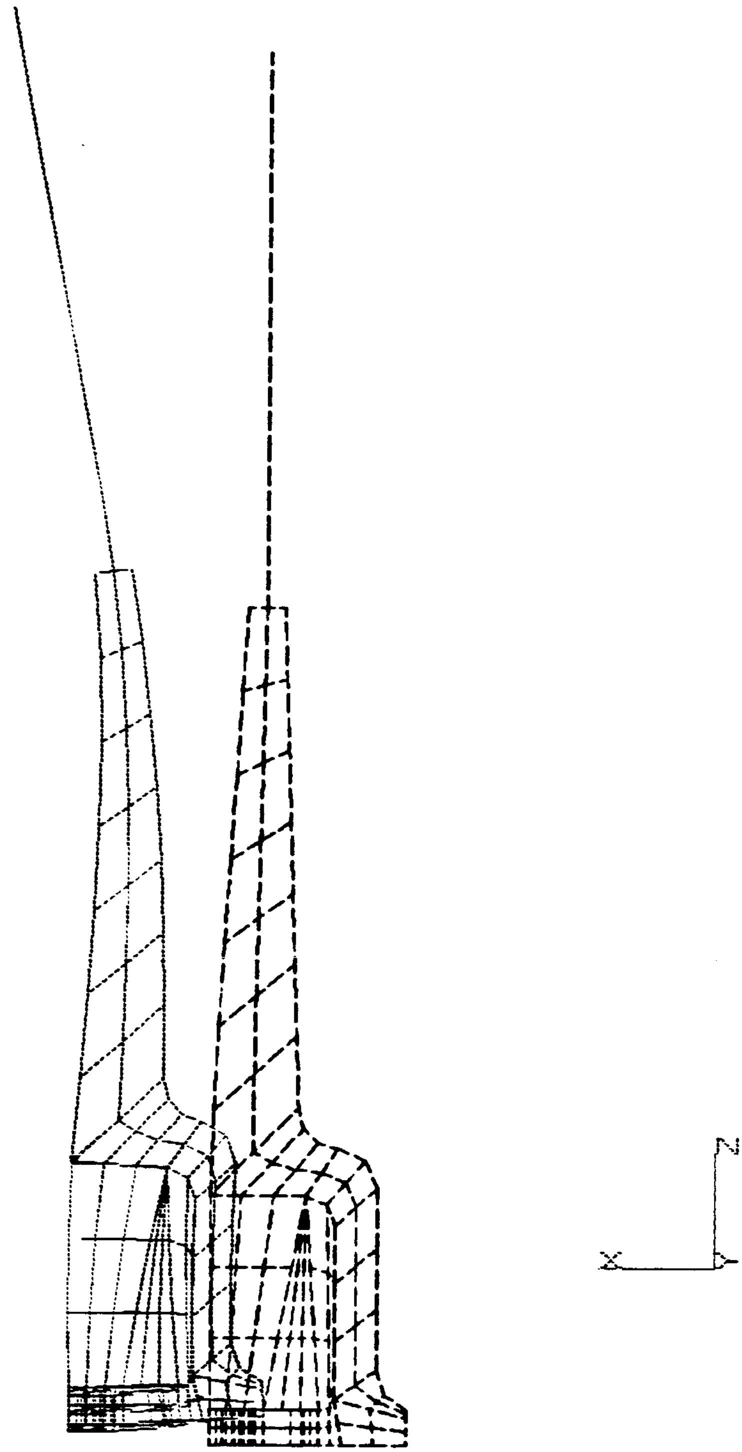


FIGURE 3.1.4. DEFORMED GRID - UPPER FLANGE AND SHELL
INTERNAL PRESSURE AND AXIAL LOAD
DEFORMATION NOT TO SCALE.

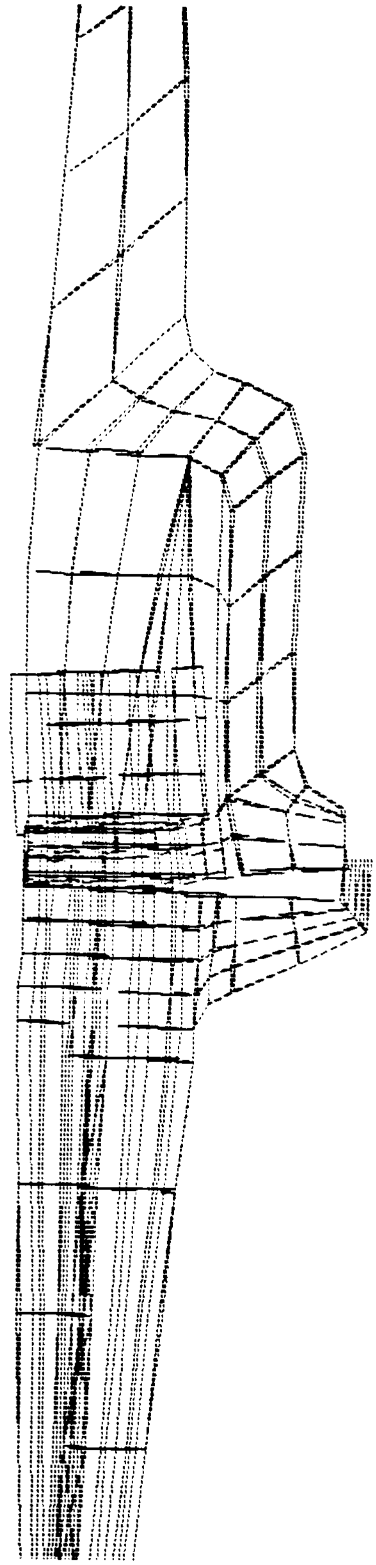


FIGURE 3.1.5. RELATIVE DEFORMATION OF SEATING SURFACES. INTERNAL PRESSURE AND AXIAL LOAD. DEFORMATION NOT TO SCALE.

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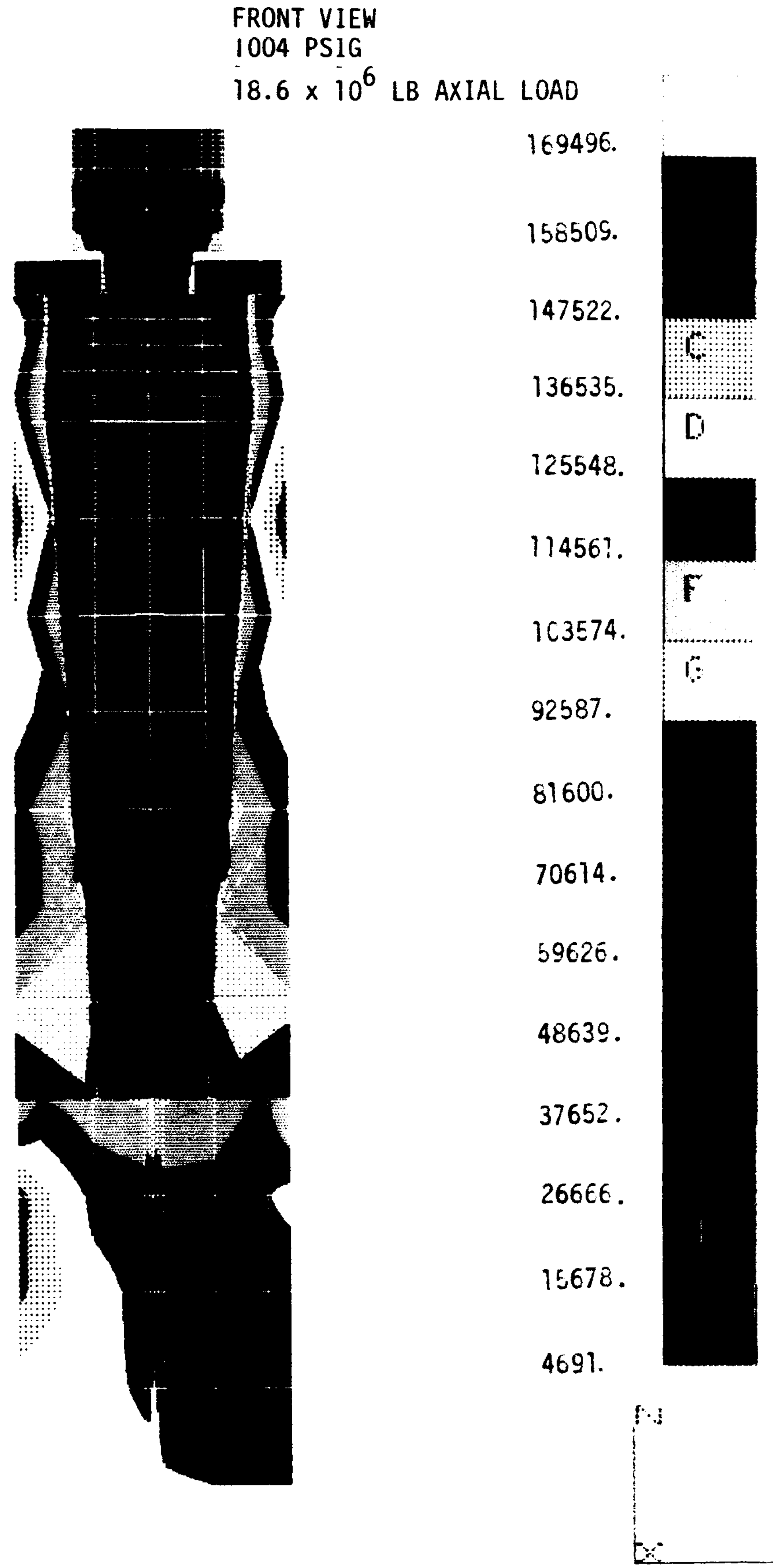

SLIGHT NONSYMMETRY DUE TO
NONSYMMETRIC SHELL-TO-SOLID LINK

FIGURE 3.1.6. STRESS CONTOURS ON LOWER FLANGE AND SHELL (VON MISES).

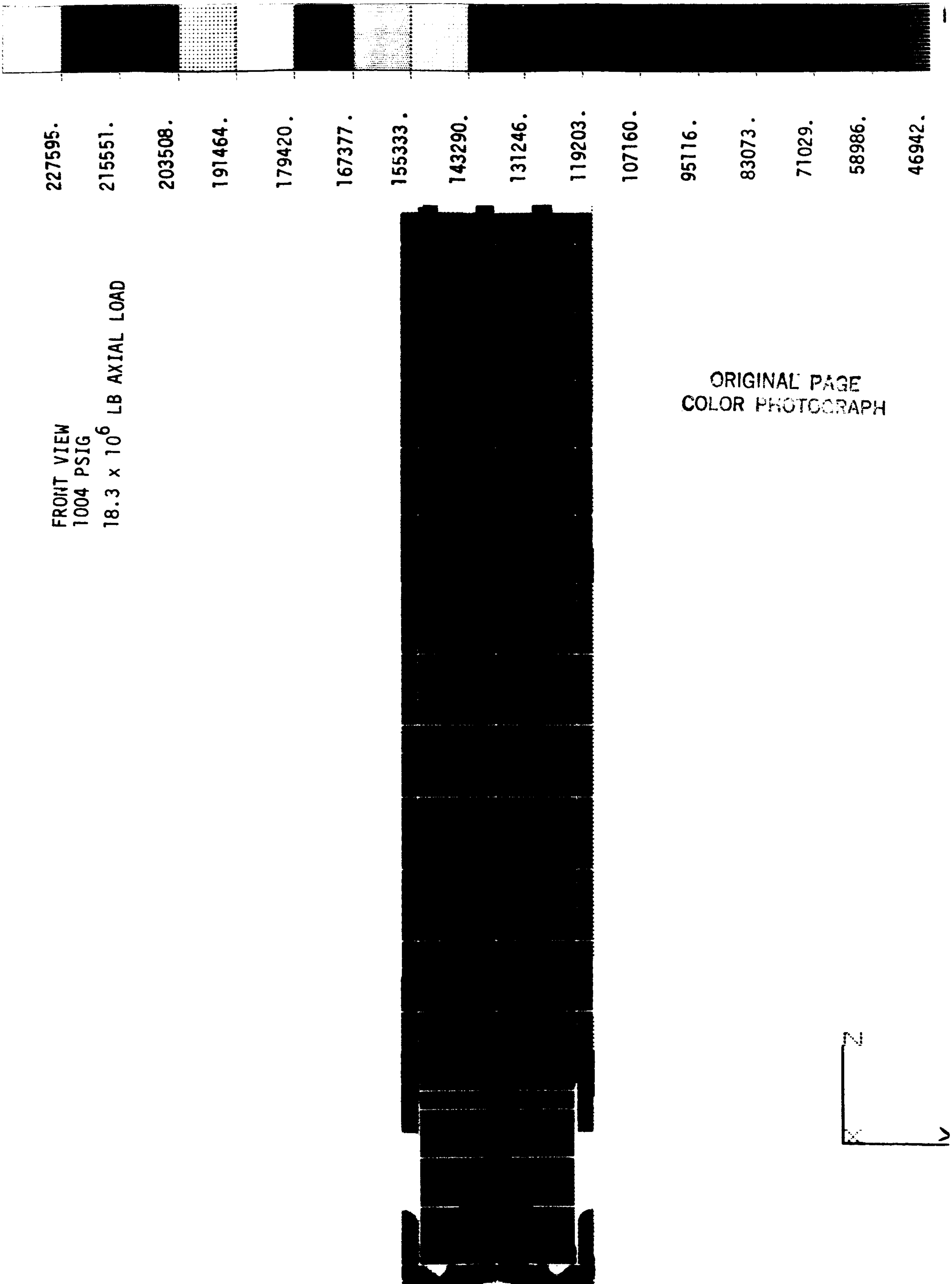

FIGURE 3.1.7. STRESS CONTOURS ON UPPER FLANGE AND SHELL (VON MISES).

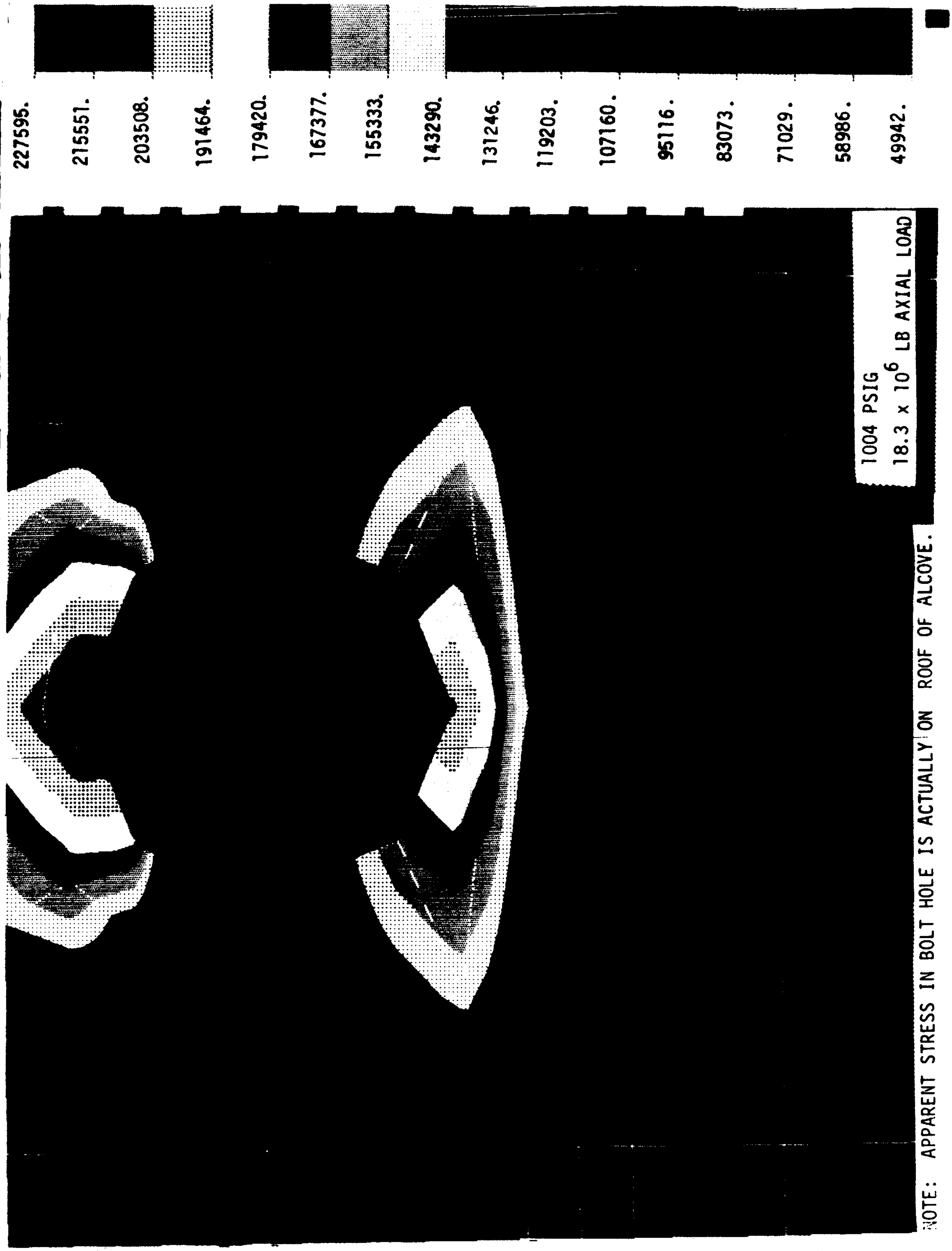

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FIGURE 3.1.8. STRESS CONTOURS ON LOWER SURFACE OF UPPER FLANGE (VON MISES).

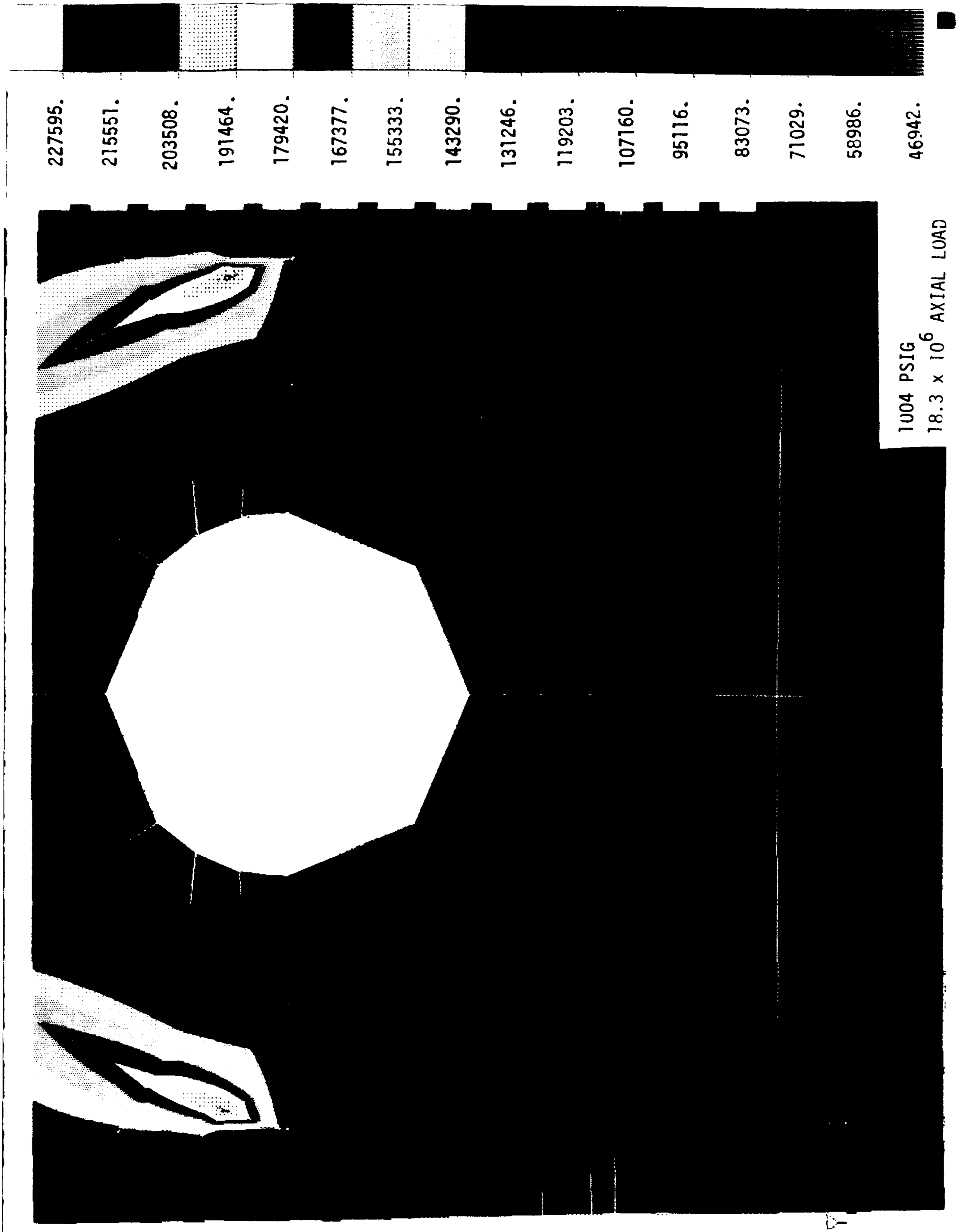

FIGURE 3.1.9. STRESS CONTOURS ON SURFACE OF UPPER FLANGE (VON MISES).

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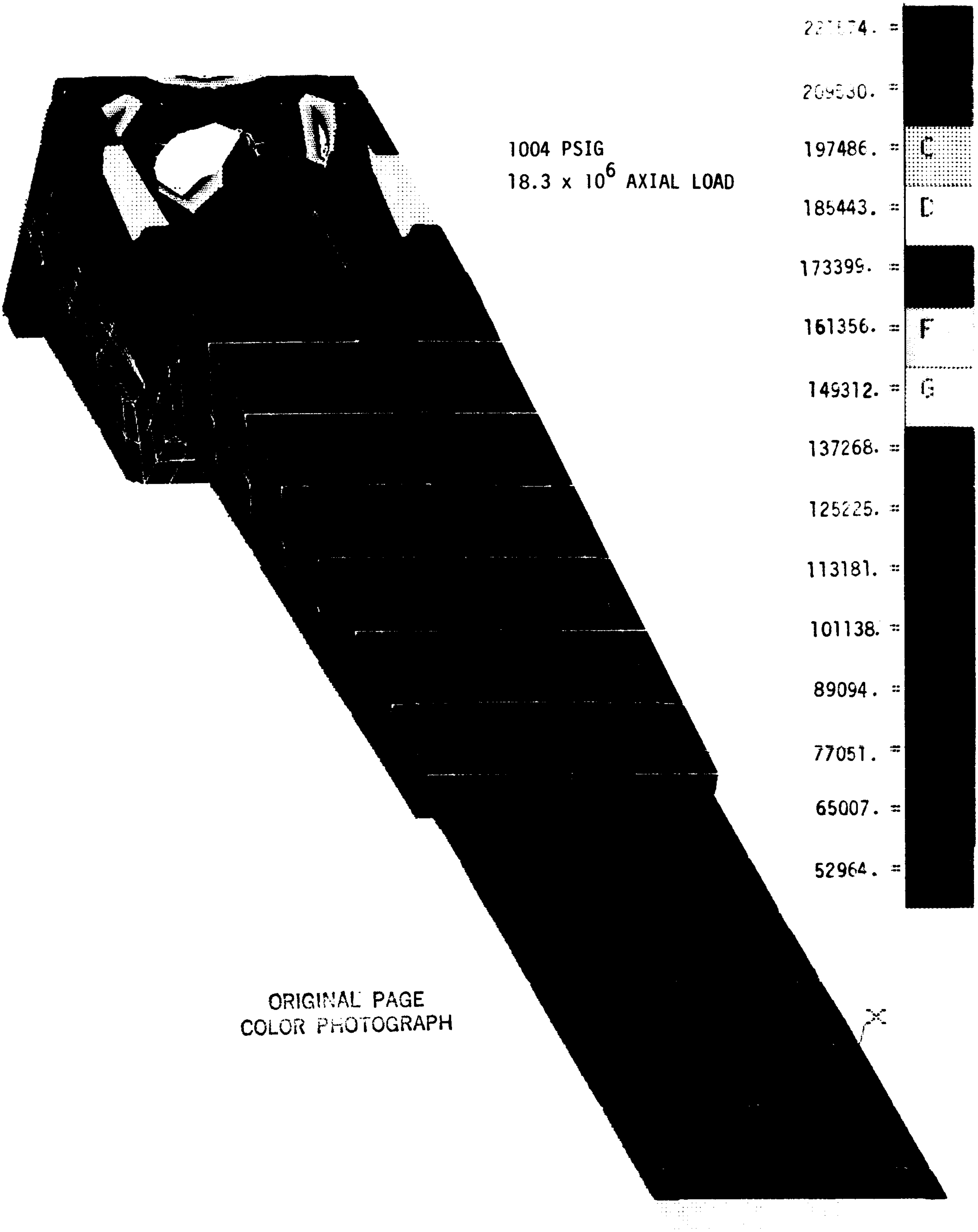

FIGURE 3.1.10. STRESS CONTOURS ON UPPER FLANGE AND SHELL - (VON MISES). ISOMETRIC VIEW

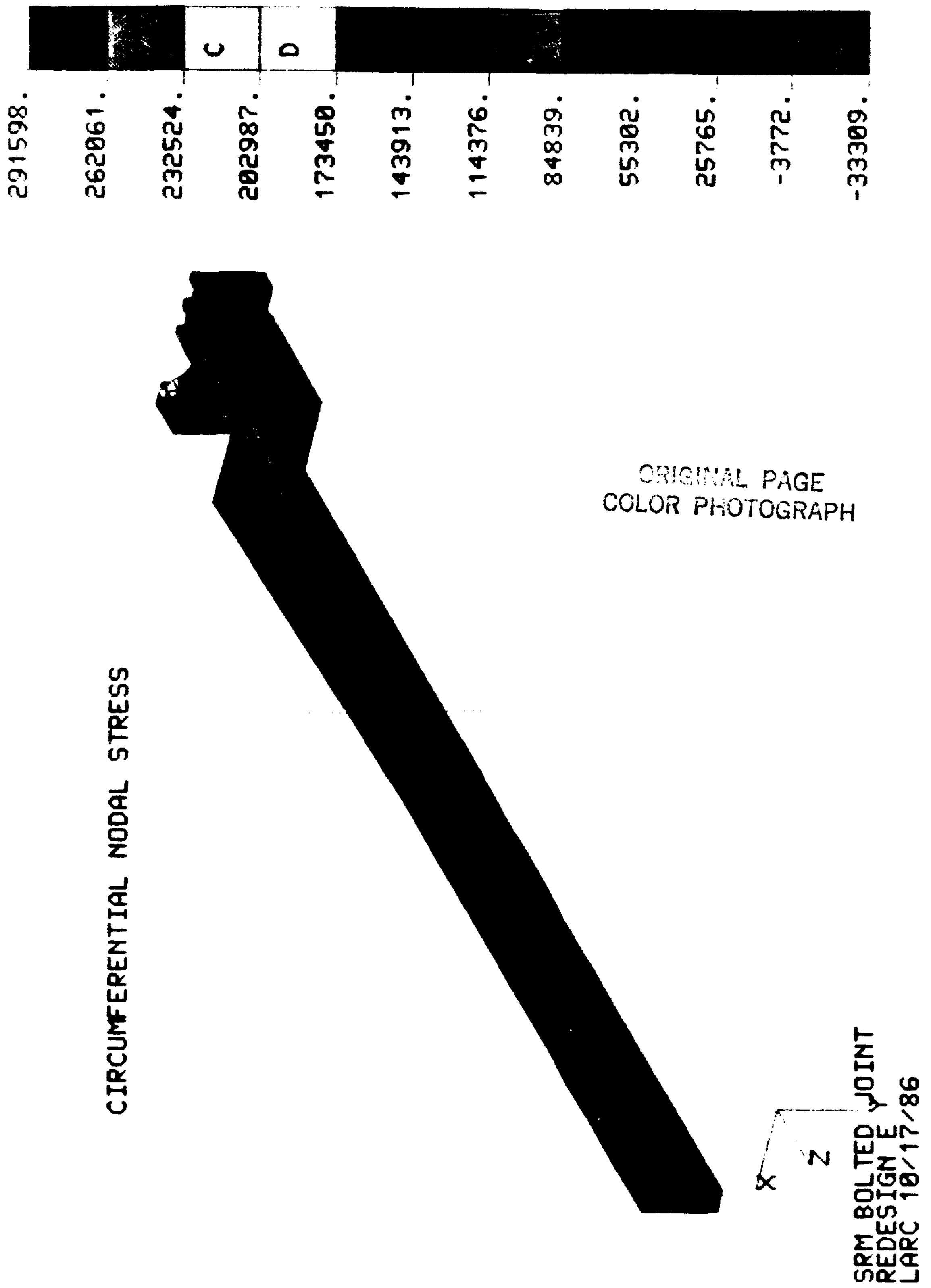

FIGURE 3.1.11

LARC JOINT CIRCUMFERENTIAL STRESS - INBOARD VIEW.

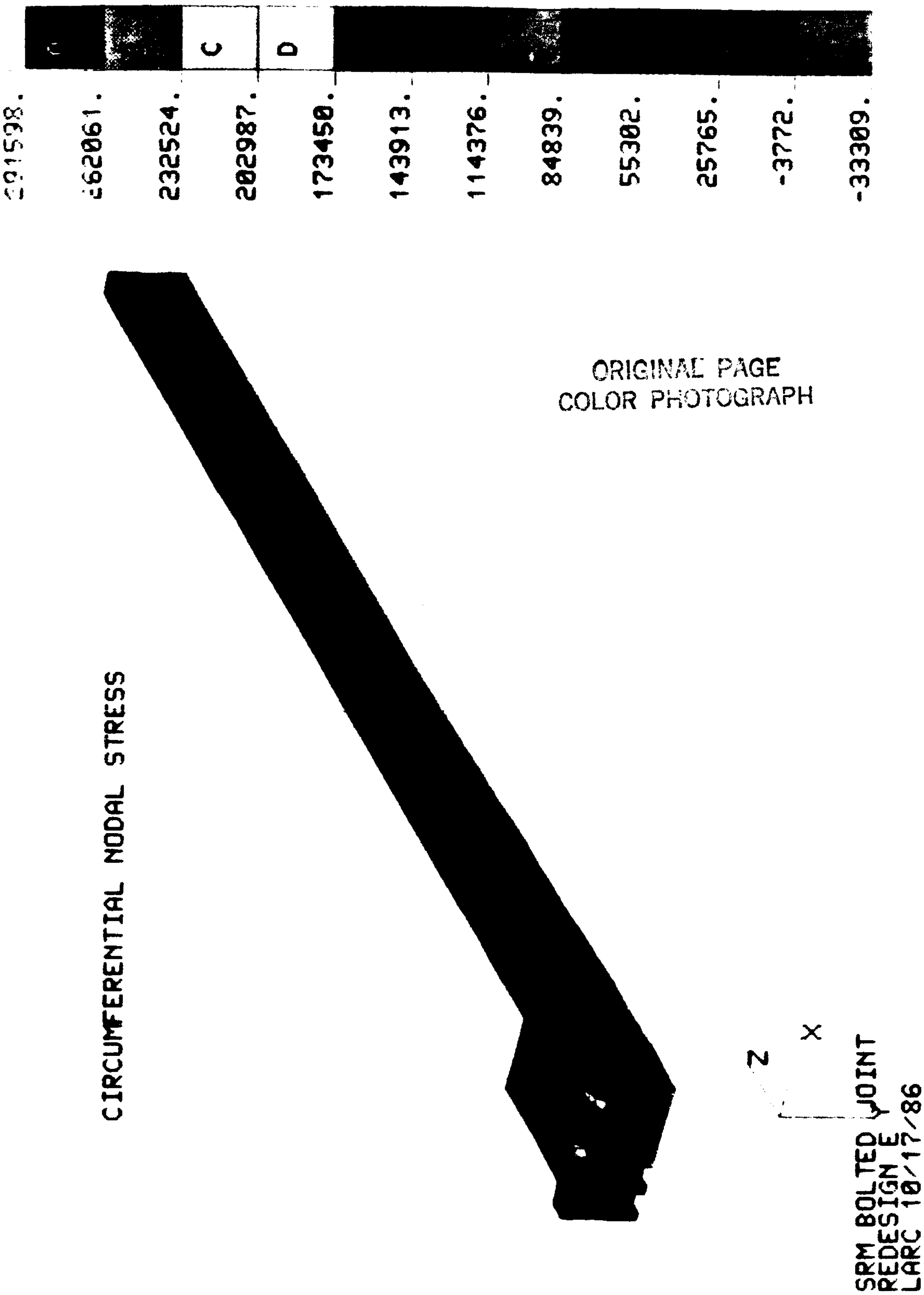

FIGURE 3.1.12

LARC JOINT CIRCUMFERENTIAL STRESS - OUTBOARD VIEW.

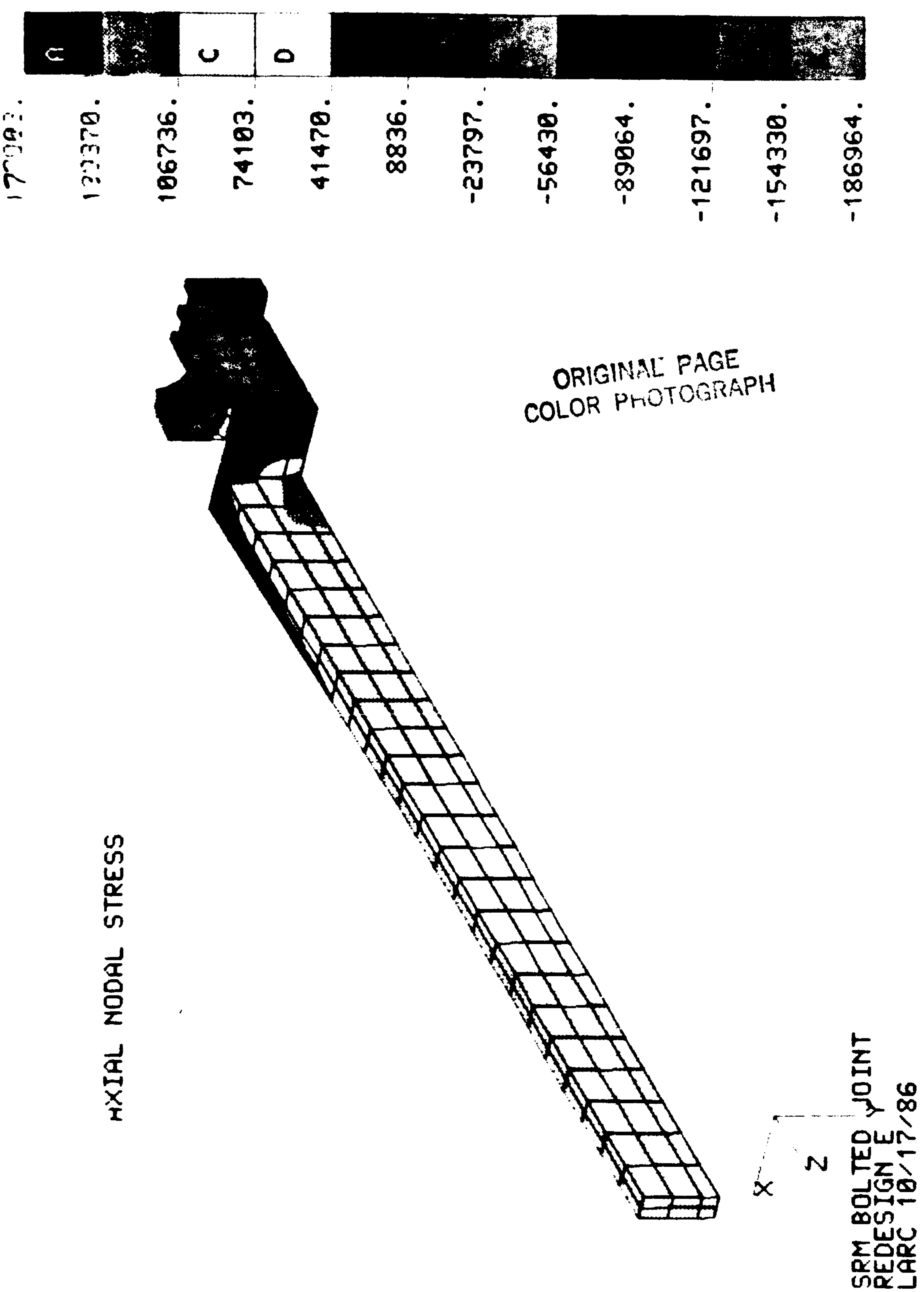

FIGURE 3.1.13
LARC JOINT AXIAL STRESS - INBOARD VIEW.

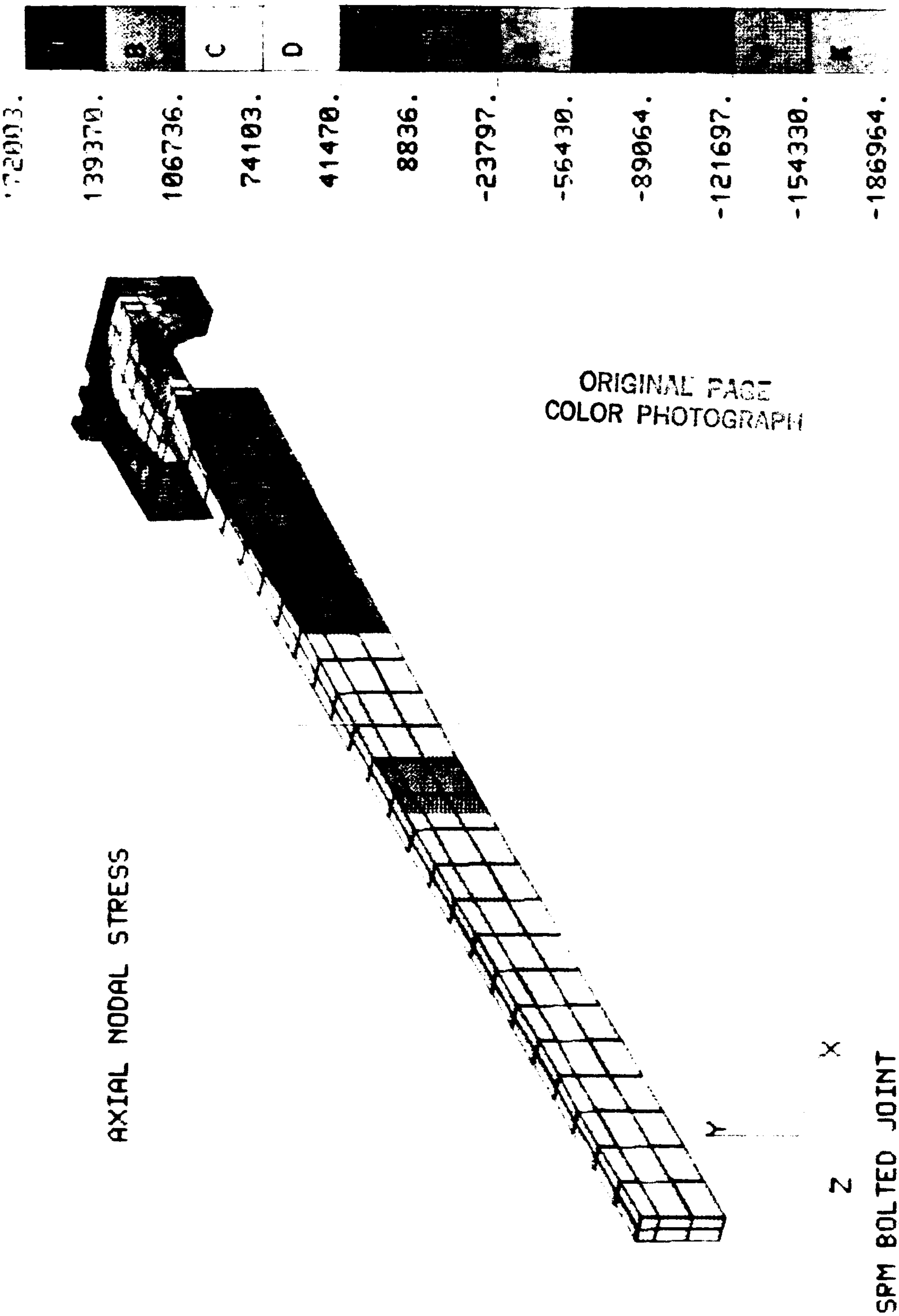

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FIGURE 3.1.14
LARC JOINT AXIAL STRESS - OUTBOARD VIEW, WEB UP.

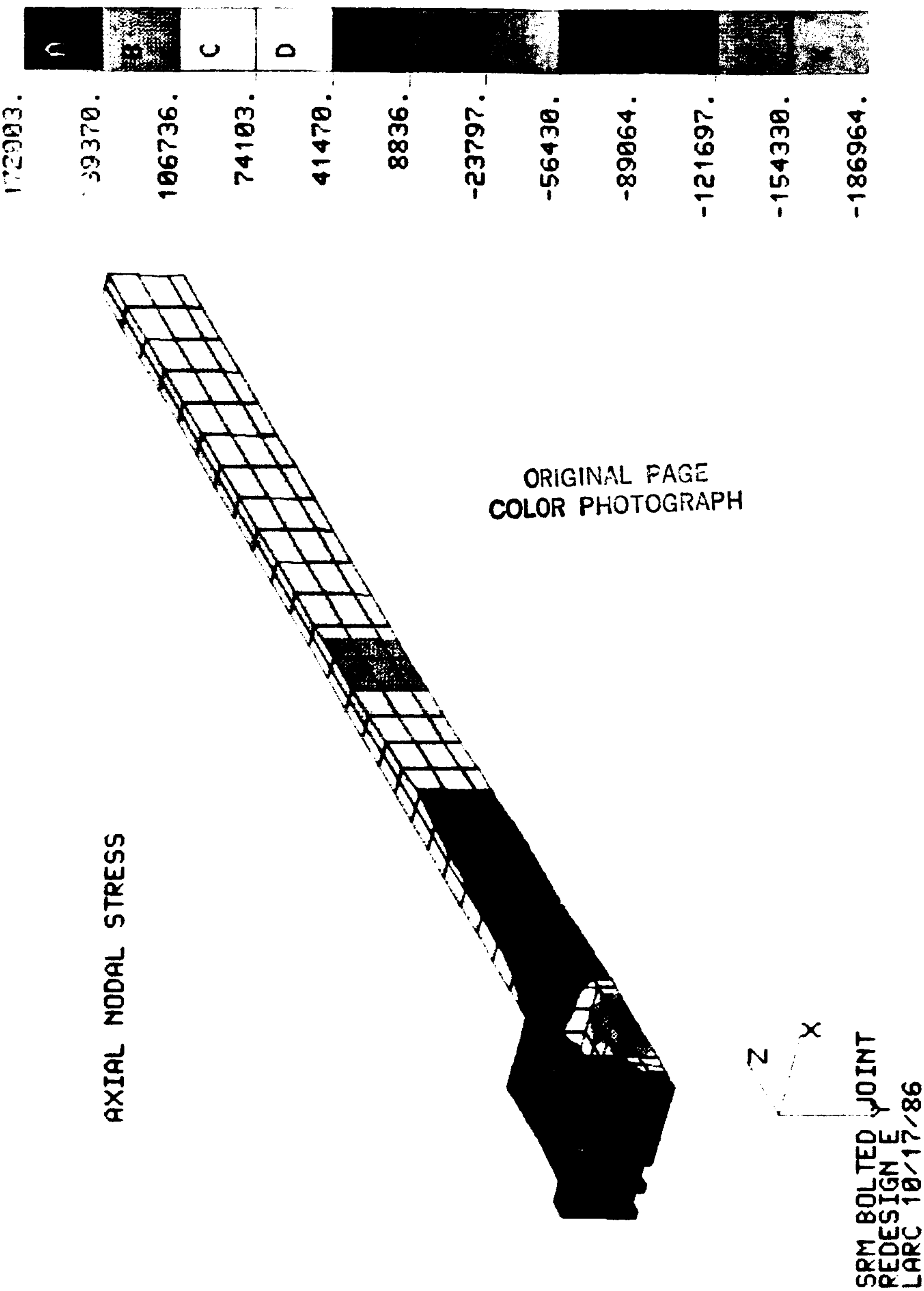


FIGURE 3.1.15
LARC JOINT AXIAL STRESS - OUTBOARD VIEW, BOLT HOLE UP.

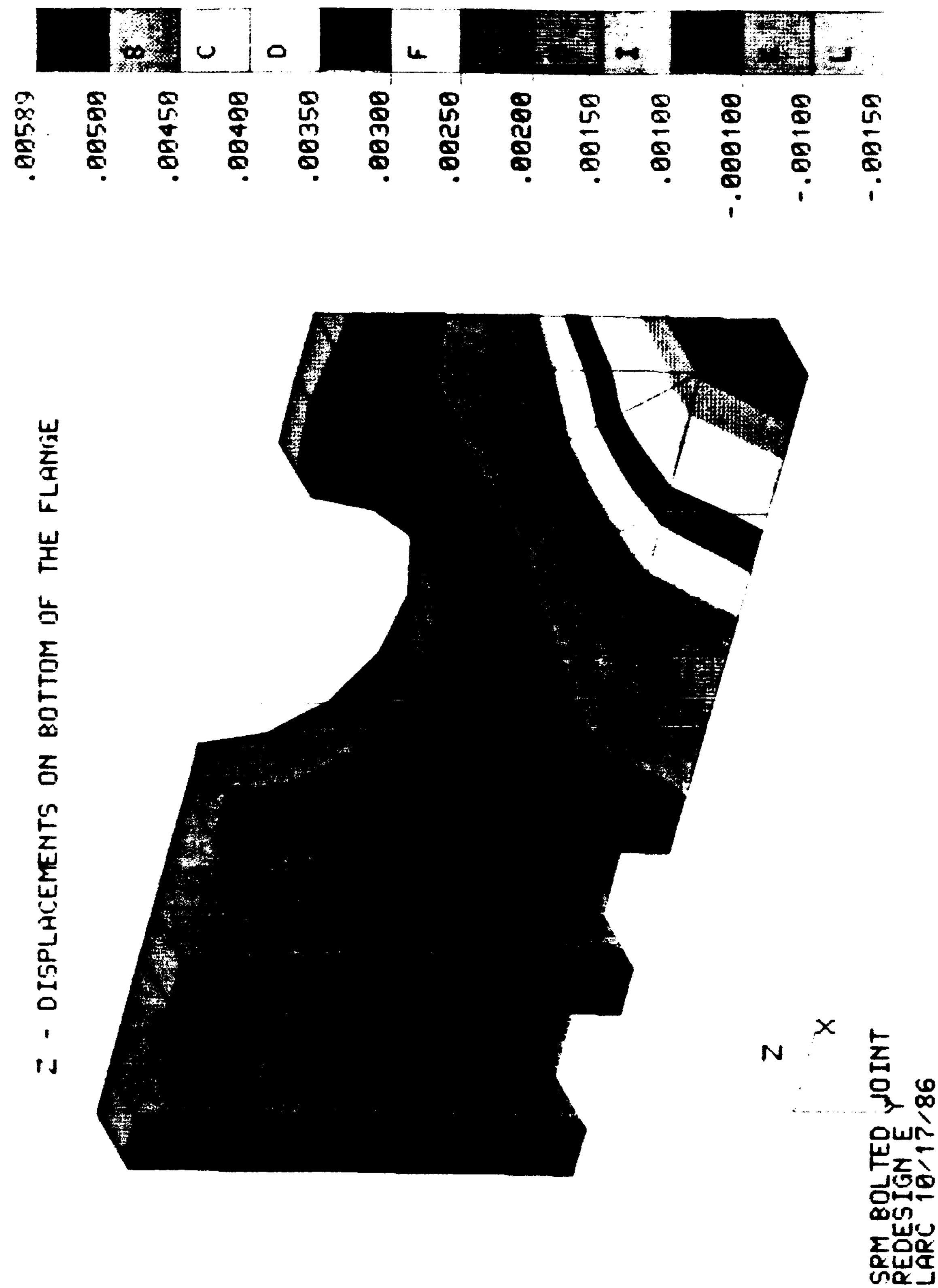


FIGURE 3.1.16
LARC SRM JOINT DEFLECTION.

TABLE 3.1.2. SUMMARY AND CONCLUSIONS.

FIRST CUT ANALYSIS

- FLANGE OPENING EXCESSIVE NEAR GASKET (.0248" AT 1004 PSIG)
- BENDING STRESSES IN UPPER FLANGE EXCESSIVE
- SOME AREAS IN GUSSETS OVERSTRESSED
- LOWER FLANGE AND FLUTED SHELL HAVE ACCEPTABLE STRESS LEVELS
- 160 ONE INCH DIAMETER STUDS ACCEPTABLE
- METAL WEIGHT INCREASED TO 20 PERCENT GREATER THAN ACTUAL MODEL WEIGHT FOR TRADE STUDIES
- PROPELLANT VOLUME LOSS INCREASED BY TEN PERCENT
- FURTHER ANALYSIS ITERATIONS ARE NECESSARY TO OPTIMIZE THIS DESIGN

TABLE 3.1.3. ANALYSIS RESULTS - LARC IN-LINE BOLTED JOINT.

- FLANGE OPENING PREDICTED TO BE <0.002 INCHES
- STRESSES ACCEPTABLE WITH SOME MODIFICATIONS
- 180 1.125 INCH DIAMETER STUDS ACCEPTABLE
- METAL WEIGHT INCREASE AND PROPELLANT VOLUME LOSS FROM ORIGINAL CONFIGURATION
- FURTHER ANALYSIS ITERATIONS ARE NECESSARY TO OPTIMIZE THIS DESIGN

REFERENCES

1. Greene, Knight, and Stockwell, "Structural Behavior of the Space Shuttle SRM Tang-Clevis Joint," NASA Technical Memorandum 89018, Langley Research Center, September 1986.

2. "Conceptual Design of Solid Rocket Booster In-Line Bolted Joint," NASA Technical Memorandum 89046, Langley Research Center, June 1986 (with updates).

3. Memo Bullock to Smith, "SRB Steel Motor Case Loads Update," ED2286-58, Marshall Space Flight Center, May 1985.

3.2 NOZZLE DESIGN STUDIES AND ANALYSES

3.2.1 ASSESSMENT OF CURRENT DESIGN

The current redesigned configuration for the nozzle-to-case joint (Figure 3.2.1) is a modification of the design used up to mission 51-L. Changes to the structure include the addition of 100 radial bolts, revised dimensions and tolerances to reduce the radial gap, and relocation of the primary seal further forward. These changes were made to increase the reliability of the joint, specifically its sealing characteristics. The radial bolts tie the nozzle and case components together so that gap opening at the primary seal location, due to the required loading, is reduced. By torquing these bolts upon installation the initial gap can be significantly reduced. NASA reports this initial installed gap to be zero. The gap opening upon loading is reported to be .000 due to "rounding" and .000/.003 due to joint rotation.

The addition of the 100 radial bolts, while enhancing the sealing capacity of the primary seal, does introduce some problem areas. These are summarized in Table 3.2.1 along with the advantages of the concept. The primary concern is the introduction of 100 potential leak paths, one at each of the radial bolts. Since these were added inside the secondary seal, the area under each bolt is sealed with a stato-seal washer. This arrangement is very sensitive to installation procedure. Omission of one small part, a stato-seal, could cause blowby, but more importantly proper torquing becomes critical. Since the structure deflects as the bolts are tightened, a bolt that was properly torqued initially could loose preload as the other bolts in the pattern are tightened. There is little tolerance for variation in radial bolt preload in this design. In addition, torquing of the radial bolts first could lead to sufficient friction forces to prevent the flange surfaces from seating properly when the axial bolts are tightened. The reverse situation is even more likely to occur if the axial bolts are tightened first.

Forces exerted by the preloaded radial bolts deform the nozzle and case structures to close any radial gap which is present at the primary seal location. While careful dimensioning and tolerancing can minimize this gap, the forces required to close even a small gap are not negligible. The stresses induced in the bolts, nozzle and motor case by these forces may be relatively small but they cannot be ignored since they exist in the parts throughout the storage life of the motor. Potential stress corrosion should be thoroughly evaluated especially because of potential salt water exposure during recovery from launch.

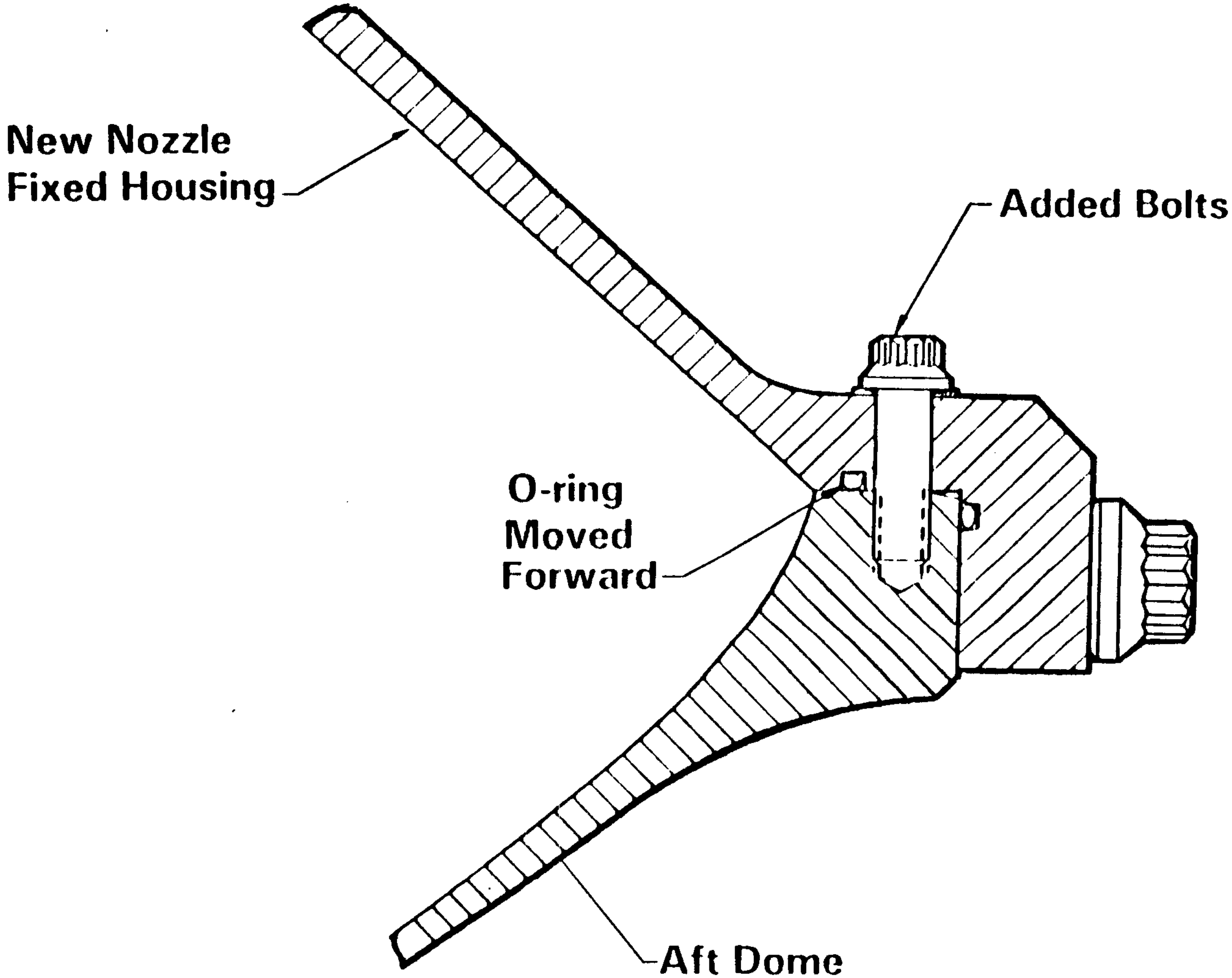


FIGURE 3.2.1. CURRENT REDESIGN OF NOZZLE-TO-CASE JOINT.

TABLE 3.2.1. ASSESSMENT OF CURRENT NOZZLE-TO-CASE JOINT.

- ADVANTAGES
  - TIES NOZZLE TO CASE IN RADIAL DIRECTION
  - PRIMARY SEAL IS PRELOADED
  - UTILIZES TWO TYPES OF SEALS (FACE AND BORE)
- PROBLEM AREAS
  - ADDITION OF 100 POTENTIAL LEAK PATHS
  - DEPENDENT ON PROPER PRELOAD AT ALL 100 RADIAL BOLTS
  - FRICTION MAY PREVENT SEALING SURFACES FROM SEATING PROPERLY
  - HIGH BOLTUP STRESSES MAY INTRODUCE STRESS CORROSION CONCERNS
  - ELASTOMERIC PRIMARY SEAL MAY BE EXPOSED TO DIRECT IMPINGEMENT OF HOT MOTOR GASES
  - REQUIRES CLOSE RADIAL TOLERANCES

Materials other than elastomerics should be considered for the seals. If the unvented bonded insulation joint were to fail and allow motor gases to directly impinge on the seals, a more heat resistant material is desirable.

An assessment of the nozzle-to-case insulation joint is included in Section 3.4.3.

The current redesigned configuration includes changes for the nozzle liner in the inlet and throat regions. In addition to material restrictions and processing changes previously incorporated, revisions to the carbon/phenolic liner ply wrap angles for the forward nose ring, the aft inlet ring and the throat ring are being introduced. This is being done to reduce the probability of pocketing erosion such as occurred in STS-8A. Several of the tests and analyses conducted to investigate this problem support this change. However, tests and analyses have also indicated that a different material, such as PAN-based graphite phenolic might have higher resistance to pocketing erosion. It is recommended that a backup material be developed in case the process sensitive current material again develops problems.

Specific data concerning other nozzle design changes and the associated problems has not been obtained and therefore no assessment can be provided.

3.2.2 NOZZLE SUBASSEMBLY JOINT SEALS

There are five joints in the nozzle subassembly. These joints are currently sealed with a single o-ring seal with no provision for leak check. These joints are necessary to join the large subcomponents of the nozzle into a single assembly. Some considerations that must be evaluated are: the ease of joint assembly and disassembly, the incorporation of a redundant seal, the ability to leak check the joint after assembly, and the temperature sensitivity effects of the seal material in its intended configuration.

The five joints have been studied individually for the above stated considerations. The joint locations are shown in Figure 3.2.2 and are numbered from one to five for identification purposes. The current joint seal designs are shown in Figure 3.2.3. Joint 1 joins the aft section of the nozzle exit cone to the main section of the nozzle exit cone. It currently utilizes a face seal to prevent gas leakage from passing through the nozzle joint. Joint 2 joins the flex bearing assembly to the nose section of the nozzle

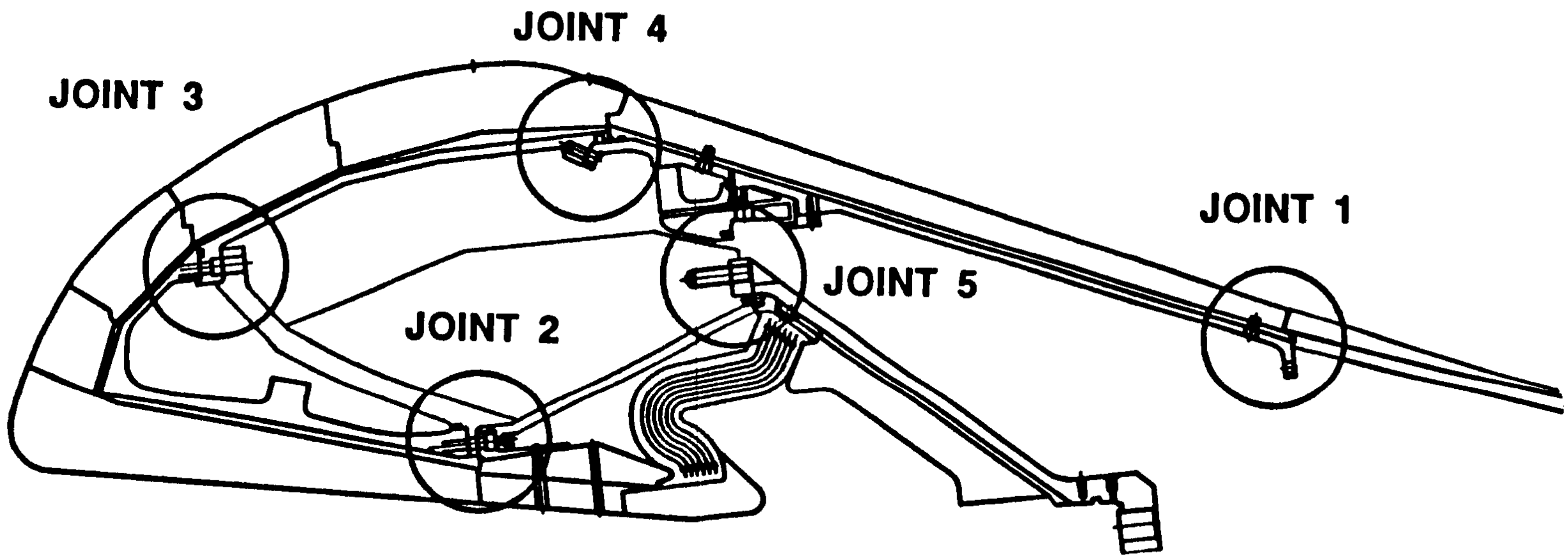


FIGURE 3.2.2. SRM NOZZLE JOINTS.

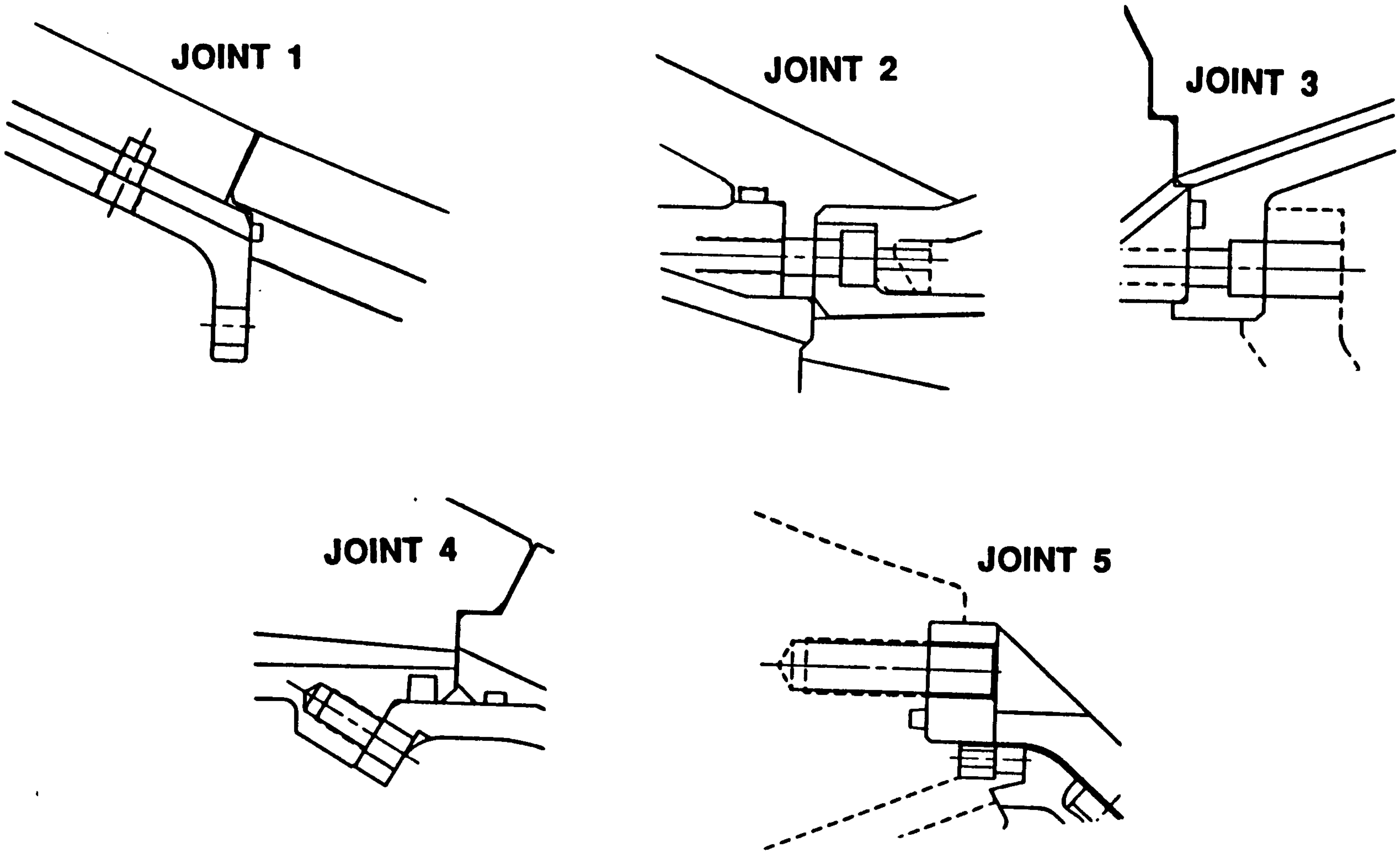


FIGURE 3.2.3. CURRENT NOZZLE JOINT SEAL CONCEPTS.

assembly. Unlike joint 1, joint 2 must withstand full motor operating pressure as well as being exposed to high velocity recirculation. The major purpose for this seal is to prevent gas flow in the structural members making up the nozzle nose shell and flexible bearing assembly. This joint is exposed to a high differential pressure that will cause damaging erosion. Joint 3 connects the nose piece with the nozzle throat section and is similar to joint 2 in operation. The joint 4 o-ring prevents gas leakage in the lower pressure exit cone region. Two seals are employed: one to prevent gas leakage into the flexible bearing area, and one to prevent leakage behind the exit cone insulation assembly. Joint 5 houses the mechanical interface between the flexible bearing and the nozzle support structure. It must withstand the loads imported by the flexible bearing torque while maintaining an adequate seal.

All joints were evaluated for seal design integrity using the following guidelines: Each seal must have redundancy, each seal must be verifiable with an external pressure check, and each joint and seal must be compatible with the assembly procedures over a defined temperature range. All joints resulted in three different sealing configurations. These seal configurations are shown in a representation by joint 3 in Figure 3.2.4. The first version consisted of two face seals and a pipe plug port located between two grooves for seal verification. The second option considered two types of seals: one face seal and one gland seal. The third concept incorporated a face seal and a gasketing material upstream of the primary o-ring seal.

The two face seals provide redundancy in number, but may not offer the intent of a redundant seal. If the operational failure of one of the seals occurs, then the probability of the redundant seal failing is also likely. A face seal in combination with a gland seal provides the high pressure advantage of a face seal with the assembly advantage of a gland seal. The two seal types operated differently and offer different sealing features. The face seal lowers the probability of o-ring extrusion due to control of mating port gaps. The gland seal can provide environmental protection for the face seal while acting as a backup for the face seal. The combination face seal and gasket utilizes the above mentioned face seal benefits with the gap filling feature of gasketing materials. The gasketing material could be a sacrificial insulation or an adhesive material that would comply with surface gaps and irregularities.

The selected joint/seal configuration is the face seal/gland seal combination. This combination best utilizes two seal types that have a demonstrated usage in

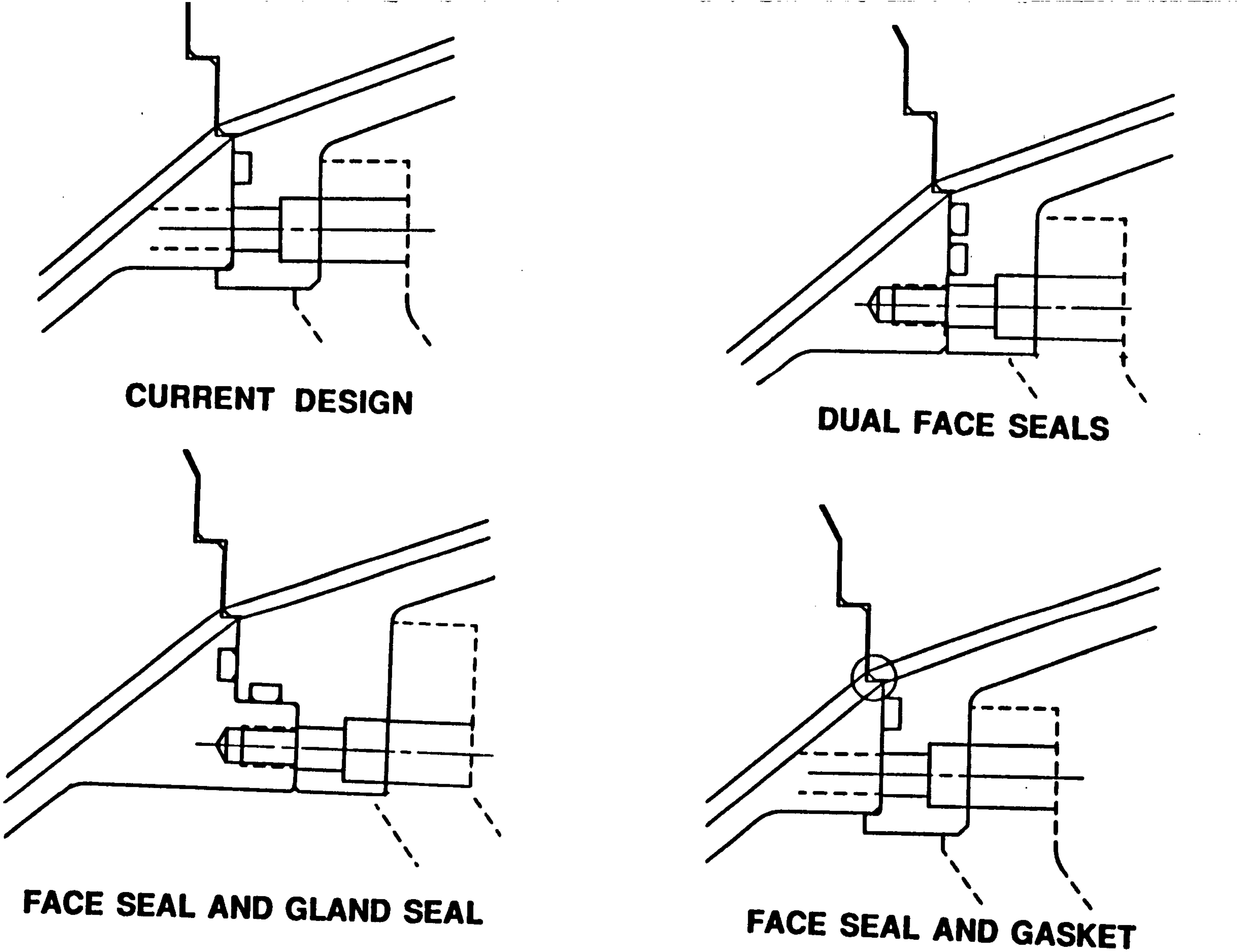


FIGURE 3.2.4. ALTERNATE JOINT SEAL CONCEPTS.

rocket motor sealing applications. The selected configuration is shown for each joint in Figure 3.2.5. Incorporation of an additional groove reduces the joint weight in joints 2 and 5 by 12.5 pounds, but increases the joint weight in the remaining joints by 260 pounds for a net weight increase of 247.5 pounds.

3.2.3 NEW DESIGN ANALYSES

3.2.3.1 NOZZLE-TO-CASE JOINT STRUCTURAL ANALYSES

This section describes the analysis completed to date in support of the design trade study of the nozzle-to-case joint.

As was stated in the joint trade study description (Section 2.6.2), an emphasis was placed on the design reliability; specifically controlling the gap opening between the nozzle and case components of the joint. This gap can be thought of as a sum of two induced gaps. The first being gap opening due to "rounding" or tolerances, and the second gap opening due to joint rotation.

The gap opening described as "rounding" in this report refers to relative radial motion of the nozzle and case attachment rings due to sliding along the mating flange interface. This type of motion occurs during rounding of the parts due to pressurization. For the 51-L design the range for this type of motion is from intimate contact at the mating cylindrical bore surfaces to contact of the bolt on the outer hole edge in the nozzle flange. The amount this can contribute to gap opening is a function of dimensioning and tolerancing. Several of the proposed concepts utilize a design feature to limit this type of motion and therefore require a dimensioning and tolerancing analysis to evaluate it. These studies have not been completed at this time. Results will be included in the final report.

To estimate the gap opening due to joint rotation, an elastic, axisymmetric finite element analysis of the joint assembly, using the TEXGAP 2-D computer program was done on the original joint design and the preliminary design choice, the dual face seal. The finite element mesh for the the dual face seal concept is shown in Figure 3.2.6. The reason for the analysis of the original design was to establish a comparison with the gap openings due to joint rotation reported by NASA.

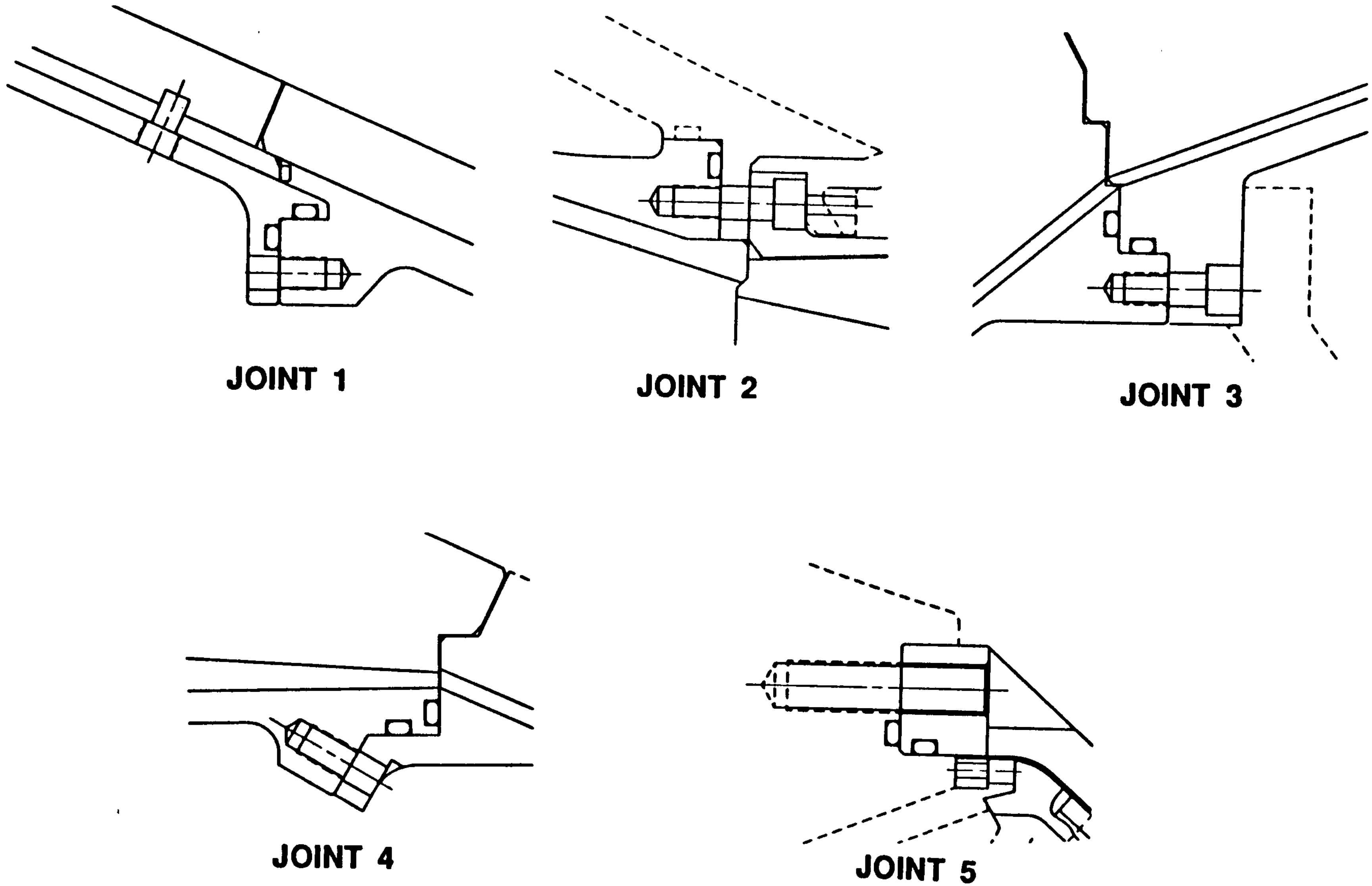


FIGURE 3.2.5. REVISED NOZZLE JOINT SEAL DESIGN.

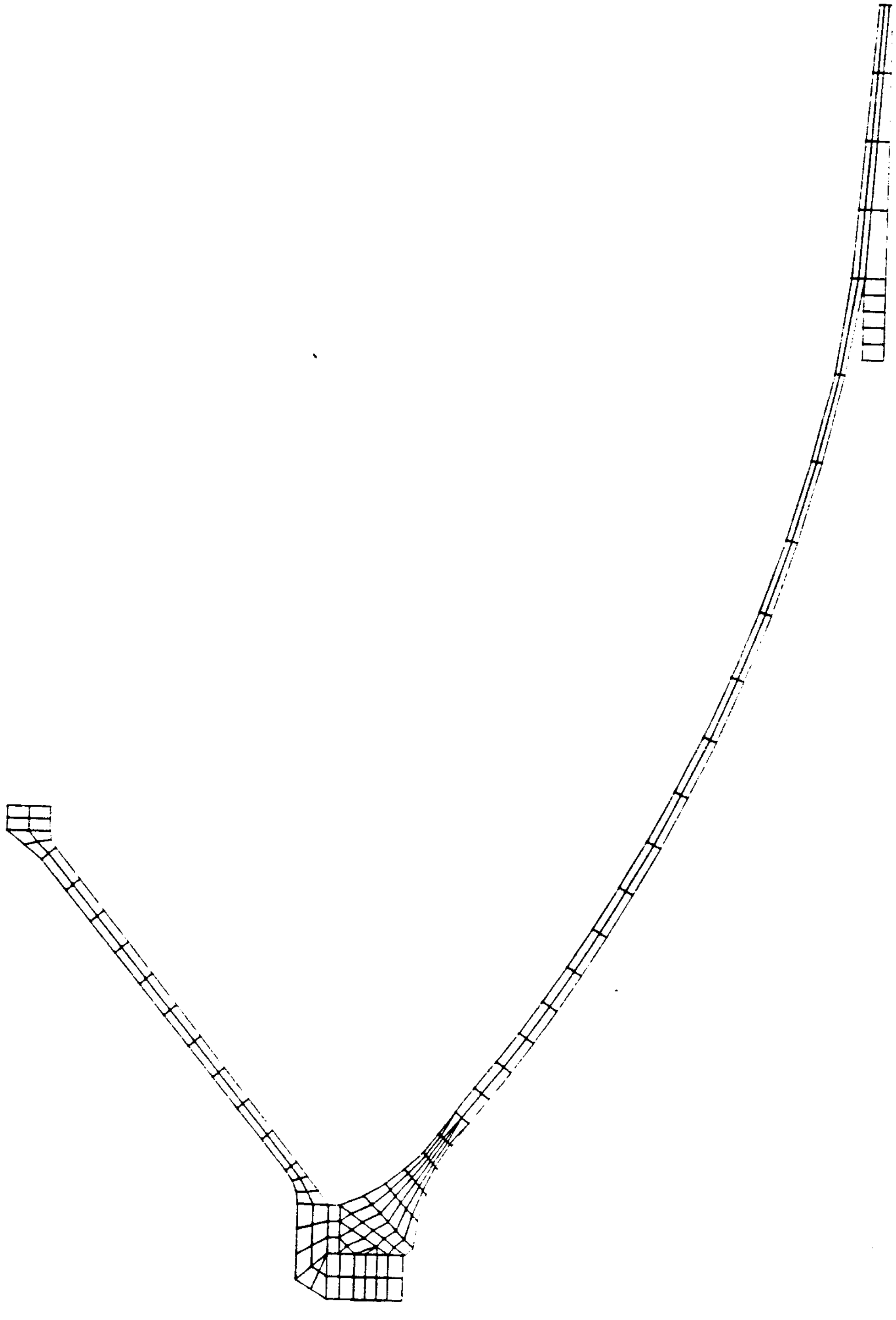


FIGURE 3.2.6. NOZZLE-TO-CASE JOINT PRELIMINARY DESIGN FINITE ELEMENT MODEL.

An initial finite element analysis was done on the nozzle and case components separately to determine their displacement response to pressurization. From this, the points of contact were identified and simulated in subsequent analyses. The bolted connection was modeled by incorporating elements that simulated the stiffness of the bolted interface. The effect of pre-loading the bolts was not modeled. This will have to be accounted for in future analyses.

With the above considerations, the finite element models were subjected to pressure loads, and the gaps between the two components evaluated. The results of this preliminary analysis are shown in Table 3.2.2 along with estimated weight impacts. The values for the redesigned baseline are taken from NASA presentation material. The ARC analysis result is shown for the 51-L design with the value from NASA presentation data included in a footnote. The values correlate well enough to provide a relative evaluation of the alternate designs. However, this analysis is not complete and the results shown are preliminary. In particular the gap opening due to joint rotation shown for Concept I does not include the effects of bolt preload. Preload is a significant contributor to the action of this joint and its inclusion should significantly reduce the gap opening. The preliminary joint selection was based on this anticipated result. This simplified 2-D analysis will be completed for all the concepts and the results will be given in the final report. In addition, it is expected that 3-D FEM analysis will be conducted for the chosen design to evaluate deflections and stresses between bolts.

3.2.3.2 NOZZLE LINER THERMAL ANALYSES

Alternate nozzle liner material candidates were thermally evaluated to quantify their ablative performance and to determine thermal margins for the current HPM nozzle liner configuration. Alternate materials fall into two generic categories: 1) phenolic ablatives and 2) carbon-carbon. Ablative phenolics are subdivided into groups using carbonized and graphitized cloths. The cloths are further divided into rayon, polyacrylonitorile (PAN), and pitch precursor groups. Carbon-carbon analyses evaluated a radially pierced 3-D carbon-carbon insulated with either a rayon precursor carbon cloth phenolic or a non-decomposing ceramic composite insulator.

Ablative performance relative to the current rayon precursor carbon cloth is summarized in Table 3.2.3. Pitch precursor graphite cloth phenolic has the lowest erosion rate due to its high thermal conductivity, which results in a low surface temperature. However, erosion improvement is gained at the expense of a much greater char

TABLE 3.2.2. SUMMARY OF NOZZLE-TO-CASE JOINT ANALYSIS.

	MAX "ROUNDING" GAP OPENING (IN)	GAP OPENING DUE TO JOINT ROTATION (IN)	WEIGHT IMPACT (LBS)
51-L DESIGN	0.038	.0099[1]	0.0
REDESIGN BASELINE[2]	0.000	0.000/0.003	--
CONCEPT I DUAL FACE SEAL	0.000	.0146[3]	+130.0
CONCEPT II SHEAR RETENTION LIP	TBD	.0057	>130.0
CONCEPT III CAPTURE LATCH	TBD	<.0057	>130.0

(1) ARC RESULT - NASA RESULT = 0.012.

(2) NASA PREDICTION.

(3) DOES NOT INCLUDE PRE-LOAD ON BOLTS. SHOULD DECREASE WITH PRE-LOAD.

TABLE 3.2.3. ALTERNATE ABLATIVE PHENOLIC NOZZLE LINER MATERIALS EVALUATION.

CLOTH PRECURSOR	CLOTH FORM	EROSION RATIO	CHAR THICKNESS RATIO	REQUIRED THICKNESS** RATIO
RAYON	CARBON	1.00*	1.00*	1.00*
PAN	CARBON	0.87	1.47	1.24
PAN	GRAPHITE	0.84	1.27	1.10
PITCH	GRAPHITE	0.81	4.47	1.96

*CURRENT MATERIAL - BASIS FOR RATIOS SHOWN.
**THICKNESS DEFINED AS 2x EROSION + 1.25 x CHAR THICKNESS.

depth. In order to meet current thermal margins (defined as 2.0 times the erosion plus 1.25 times the char thickness at action time), the pitch precursor carbon cloth phenolic would have to be approximately twice as thick as the current material. PAN precursor carbon and graphite cloth phenolics have a slightly lower erosion rate and higher char rate than rayon precursor carbon phenolic, with the graphite cloth being slightly superior. Required thickness of the PAN precursor graphite phenolic is approximately ten percent higher than for the current material. However, the HPM nozzle has a sufficient thermal margin to accommodate the use of this candidate material.

Thermal trade studies of a carbon/carbon liner insulated with a rayon precursor carbon phenolic were performed to determine the liner thickness which results in equivalent thermal margins to the current design. Required margins were assumed to be 2.0 times the erosion for the carbon/carbon liner and the same char depth as in the current nozzle for the carbon phenolic insulator. Analyses show the carbon/carbon erosion rate is reduced by a factor of 3.4 relative to the current carbon phenolic, which results in an erosion margin $[(\frac{\text{thickness}}{2 \times \text{erosion}} - 1) \times 100]$ of 42 percent. However, the carbon/carbon liner thickness must be limited to approximately 39 percent of the current liner thickness to obtain the equivalent char depth as exhibited in the current nozzle.

One of the major problems encountered when carbon/carbon liners are insulated with a decomposing insulator, such as phenolic, is the need to vent pyrolysis (decomposition) gases. For the configuration discussed above, the pyrolysis gas mass flux is approximately 130 lbm/ft^2hr. Unless these gases are adequately vented, high pressures will occur at the liner/insulator interface, which can result in buckling of the liner. The outgassing problem can be eliminated by the use of a non-decomposing ceramic/ceramic insulator. The primary disadvantage of these materials is the higher thermal diffusivity and specific gravity. A silicon carbide/silicon carbide composite was selected as the insulator for the carbon/carbon. Analyses assumed the total thickness of the carbon/carbon and ceramic/ceramic was equivalent to that of the current carbon phenolic liner. In-depth temperatures are predicted to be considerably higher than obtained in the current design. Interface temperatures between the ceramic/ceramic and the glass phenolic at action time are approximately 600°F. At splashdown, the steel nozzle housing temperature is over 750°F; which may impact the reusability of the metal components. Optimization of component thicknesses will reduce the temperatures to some degree, but preliminary indications show that there is not sufficient volume to obtain the same nozzle housing temperatures as in the current nozzle with a carbon/carbon liner and ceramic/ceramic insulator.

3.3 IGNITER

3.3.1 INTRODUCTION

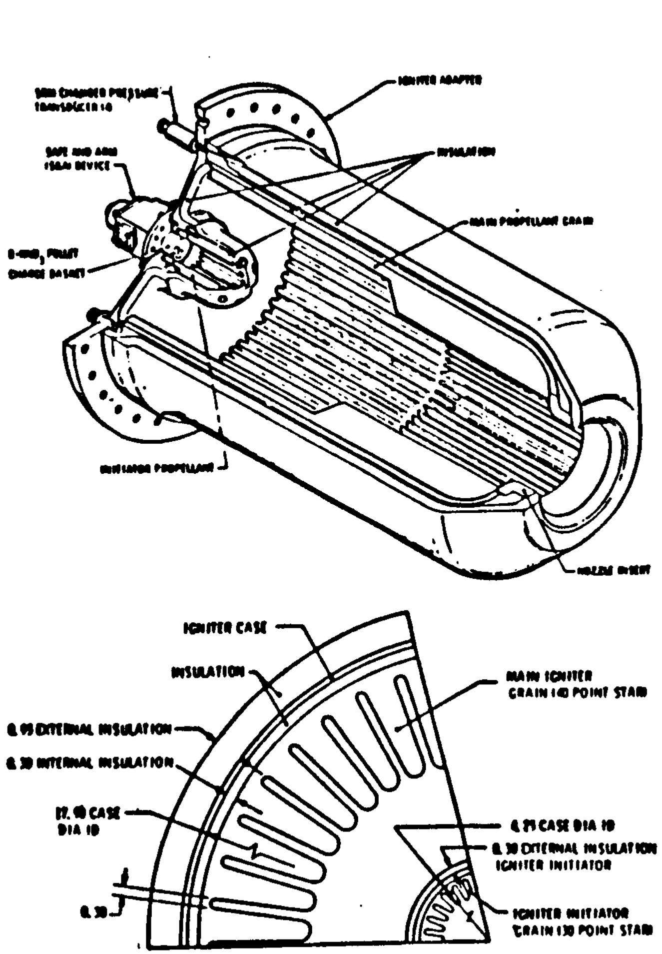
The primary purpose of the igniter design study is to minimize potential leak paths in the igniter-to-adapter and adapter-to-motor case interfaces. Secondary objectives are to evaluate igniter propellant and ballistic design, evaluate expendable versus reusable hardware, and replace the igniter insulation with an asbestos-free material.

The constraints on this design study are to maintain ignition performance and reproducibility while not degrading structural and thermal margins. In order to have a high degree of confidence in any design changes, the data base for the design change must exist or be created through a test program.

3.3.2 CURRENT SRM IGNITER DESIGN

The current shuttle igniter design is depicted in Figure 3.3.1. The SRM ignition system is a forward end, internally mounted solid rocket type (pyrogen) igniter and is approximately 44.5 inches long by 20 inches in diameter. The flight grain is a 40-point star configuration which is approximately 16.9 inches in diameter by 32.8 inches long. The propellant grain consists of approximately 137 pounds of a 10% aluminized PBAN propellant and it is cast into a D6AC steel case insulated internally and externally with asbestos and silica-filled NBR. A molded silica phenolic throat insert controls the igniter pressure and directs the igniter plume to the main SRM propellant grain.

The igniter chamber is bolted to the igniter adapter (D6AC steel) with 36 three-quarter inch bolts. Each bolt uses a special washer and pressure sealing packing. The main seal between the igniter chamber and the igniter adapter is a dual o-seal gasket. The adapter bolts to the main SRM chamber with 40 five-eighths bolts utilizing a washer and pressure sealing packing on each bolt. The primary seal between the adapter and the SRM chamber is also a dual o-seal gasket.



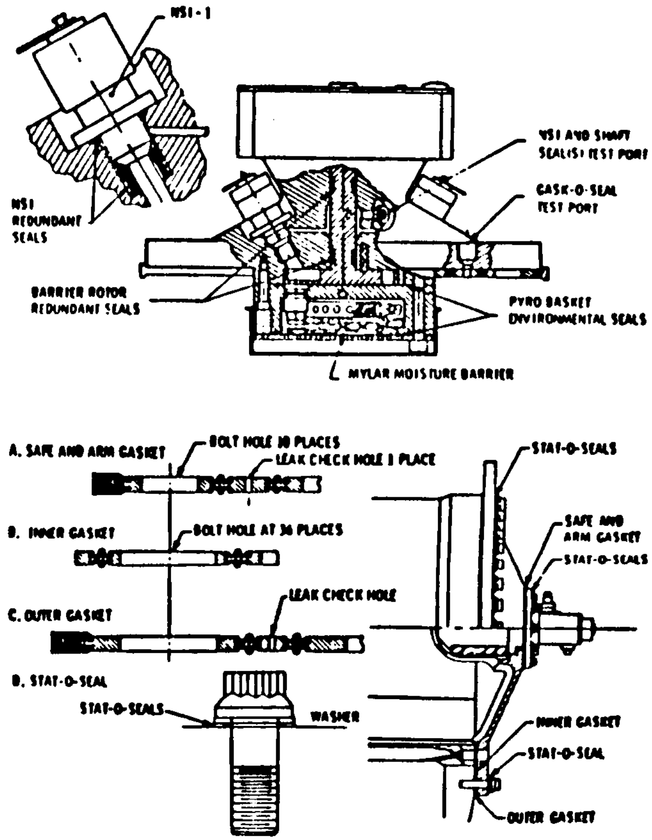


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FIGURE 3.3.1. CURRENT IGNITER AND SEALS.

The ignition initiator is a small, multinozzled asbestos- and silica-filled NBR insulated steel cased rocket motor containing 1.4 pounds of propellant in a 30-point star configuration. The initiator case and the safety and arming (S&A) device attach to the igniter adapter. The S&A is bolted to the adapter using 10 bolts. A dual o-seal gasket forms the dual redundant seals with special packing on each bolt as an environmental seal.

The S&A device consists of a reusable actuating and monitoring (A&M) and an expendable booster-barrier assembly containing a mixture of $BKNO_3$ pellets and granules. Two redundant NASA standard initiators (NSIs) provide positive ignition. The NSIs utilize dual redundant seals and the A&M uses dual o-ring seals on the barrier rotor shaft.

Totaling up the seals in the igniter/S&A, there are 6 primary seals, 42 secondary seals, and 88 environmental seals. The primary seals are the fundamental seals that hold igniter or motor gas pressure while secondary seals would seal against gas pressure only if the primary seals failed. The environmental seals are used for sealing out the environments except for the bolt seals on the bolts that attach the igniter chamber to the igniter adapter. These seals are secondary seals and environmental seals.

3.3.3 IMPROVED SRM IGNITER SYSTEM

The improved SRM igniter is depicted in Figure 3.3.2. This system is a forward end, internally mounted solid rocket type (pyrogen) igniter. The igniter is approximately 19 inches in diameter by 34 inches long, overall. The flight grain is a 40-point star configuration that is 16.4 inches in diameter by 21.6 inches long. The propellant grain consists of 119 pounds of 18% aluminized HTPB propellant cast into a 200 maraging steel case with an integral welded igniter adapter and a removable aft closure held in place using 36 high strength 3/4 inch bolts. The case is insulated internally and externally with Kevlar and silica-filled Hypalon. A molded silica phenolic throat insert controls the igniter pressure and directs the igniter plume to the main SRM propellant grain.

The igniter adapter is bolted to the main SRM chamber with 40 five-eighths bolts that have a washer and an environment seal on each bolt. The primary seal consists of a radially compressed aerospace G-T ring that seals against high pressures with larger clearances than an o-ring. This design is utilized for dynamic rod and piston seals and will not twist under installation. The secondary seal is a resilient metal c-ring mounted

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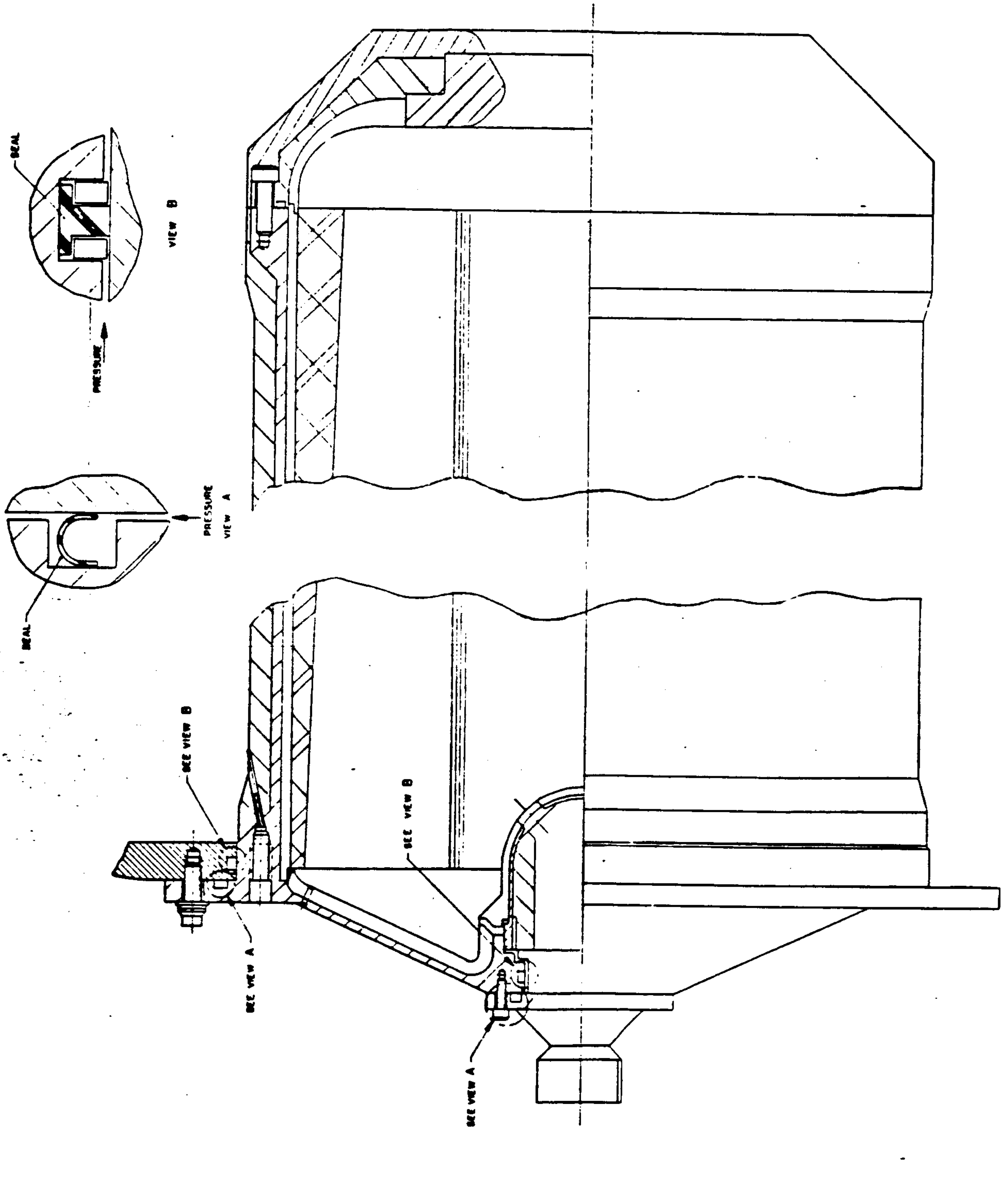


FIGURE 3.3.2. PROPOSED SRM IGNITER.

as a face seal between the adapter and the main SRM chamber. Metal c-ring seals are very high temperature seals (up to 2000°F) and they are resilient seals that maintain sealing in the event of flange separation caused by thermal or pressure shock or by bolt creep. Both types of seals are also much more compression set resistant than o-rings and have a higher recovery rate than o-rings, allowing them to always maintain contact with the sealing surfaces as the gap opens.

At the forward end of the igniter, mounted to the igniter adapter is the ignition initiator. The initiator utilizes the same hardware that is used in the current design but it is loaded with ARCADENE 360A HTPB propellant to maintain compatibility with the main igniter propellant. The grain design is also the same as in the current design since the 360A propellant is tailored to have the same burn rate as the TP1178. The initiator case will be insulated with Kevlar and silica-filled Hypalon to the same thickness that the abestos-silica NBR is applied on the current design.

The S&A device will remain the same as that used on the current design with the asbestos containing parts replaced with non-asbestos materials. The S&A clutch disc material will be replaced with Kevlar phenolic while the S&A commutator material will be replaced with ceramic phenolic.

The S&A is attached to the igniter adapter using 10 bolts with a washer and special packing used as an environmental seal. The S&A is sealed to the adapter using a radially squeezed G-T ring and a face sealing metal c-ring, the same as in the adapter to main SRM case. The dual o-ring seals on the A&M main rotor will remain the same as will the seals on the NSIs. For the external environmental seals, either an o-ring type seal or a formed in-place gasket material (i.e., RTV) can be used on the igniter adapter to SRM case and the S&A to igniter adapter.

Total weight savings for the improved versus the production igniter is approximately 110 pounds. The reduced grain length (32.8" vs. 21.6") and the thinner insulation on igniter case account for this weight reduction.

3.3.4 IGNITER PROPELLANT

The igniter propellant selected for the improved igniter is ARCADENE 360A, which is detailed in Table 3.3.1. ARCADENE 360A is an 88% solids-loaded HTPB

TABLE 3.3.1. ARCADENE 360A.

INGREDIENT	WT %
R-45 HT BINDER	10.0
DOA	2.0
Al POWDER	18.0
Fe_2O_3	1.5
AP (60/40 200μ/MA)	68.5
	100.0

TOTAL SOLIDS	88%
I°sps	260.7 lbf-sec/lbm
DENSITY	0.0655 lb/cu-in
EQUILIBRIUM T_c	3508°K
C*	5123 ft/sec
E	10.74
GAMMA	1.166
BURNING RATE (1000 PSI)	0.70
PRESSURE EXPONENT	0.48

MECHANICAL PROPERTIES (2 in/min x head)

	70°F	-40°F	+160°F
MAX. STRESS (PSI)	201	514	128
% STRAIN AT MAX. STRESS	30	43	26
TANGENT MODULUS (PSI)	1540	12,100	1010

propellant with a bimodal blend of ammonium perchlorate (AP) and 18% aluminum. This propellant is a variant of the MLRS propellant of which ARC has loaded over 28 million pounds into MLRS motors. The higher burn rate necessary for the SRM igniter application is achieved by increasing the percentage of iron oxide ($\tilde{1}$% vs. 1.5%) and by varying the percentage of fine versus course AP in the bimodal blend. This propellant is completely characterized for use through the MLRS program and represents very low risk in the SRM igniter application. Prior to casting the propellant, the insulated case will be barrier coated with EA-946 and then lined with ARL-151 liner. Both of these materials have been well characterized for use with this propellant in the MLRS program.

3.3.5 IGNITER GRAIN DESIGN

The igniter grain is shown in Figure 3.3.3 and described in Table 3.3.2. The grain design is a 40-point star design with the web between star points varying from 0.20 inches at the head end to 0.05 inches at the aft end. The star tip radii are 4.90 inches from the igniter centerline. The maximum nominal mass flow rate is approximately 65 percent of the DM-1 igniter firing. The molded cellulose phenolic nozzle has a throat diameter of 6.60 inches and will have a projected 0.030 inches total erosion on the diameter during the igniter firing. Maximum expected operating pressure (MEOP, $+3\sigma$ condition) is projected to be 1660 psia.

This grain design matches very closely the performance from the current production igniter. Igniter MEOP and mass flow rates compare very favorably with SRM firings QM-1 and QM-2 as shown in Figure 3.3.4. Igniter plume contact with the SRM will therefore match the current production igniter and ignition times for the SRM should remain the same.

3.3.6 IGNITER INSULATION

The improved igniter insulation selected for replacing the current asbestos and silica-filled NBR is Kevlar and silica-filled Hypalon. This selection comes from the extensive trade studies documented in Section 2.4. Preliminary estimates of insulation requirements, based upon reported Material Affected Rates (MARs) for the igniter in QM-2, using a 2.0 MAR safety factor, and reducing the igniter length results in a total igniter assembly insulation weight reduction of approximately 78 pounds. On the average, external igniter case insulation is reduced in thickness by 20 percent while the

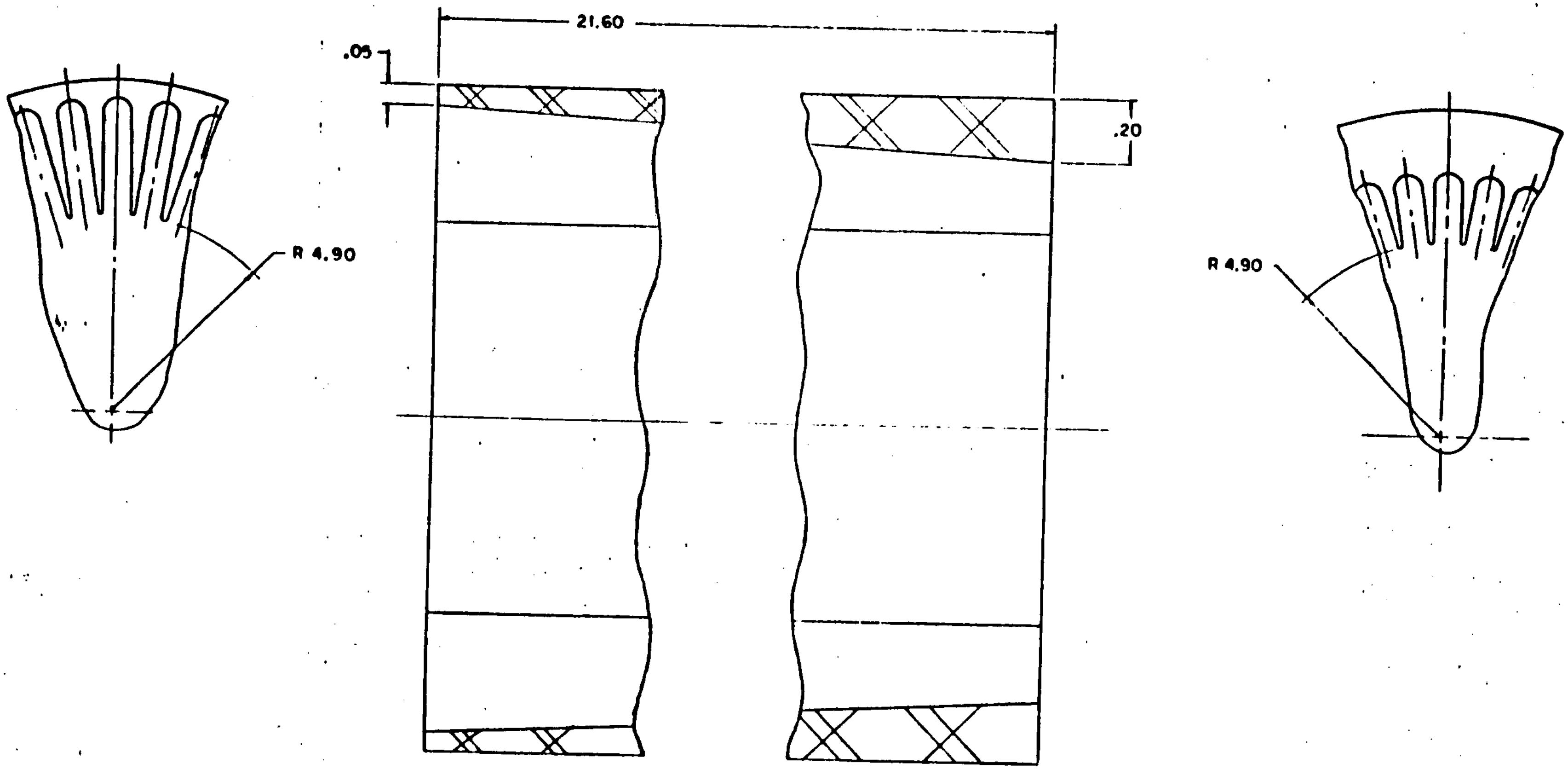


FIGURE 3.3.3. IMPROVED SRM IGNITER GRAIN.

TABLE 3.3.2. SRM IGNITION SYSTEM.

PROPELLANT

- HTPB/AP/Al
- r_{1000} = 0.70 ips @ 60°F

GRAIN CONFIGURATION

- 119 lb. GRAIN WEIGHT
- 40-POINT STAR, 21.6 IN. LONG BY 16.4 IN. OD
- 4.90 IN. RADIUS TO STAR TIPS
- 0.20 IN. TO 0.05 IN. WALL WEB TAPER FORWARD TO AFT

NOZZLE

- CELLULOSE PHENOLIC
  0.030 IN. TOTAL PREDICTED EROSION ON DIAMETER
- 6.60 IN. THROAT DIAMETER

PERFORMANCE

- 323 lbm/s MAXIMUM NOMINAL MASS FLOW RATE
- 1660 PSIA MAXIMUM EXPECTED OPERATING PRESSURE (90°F, +3σ CONDITIONS)

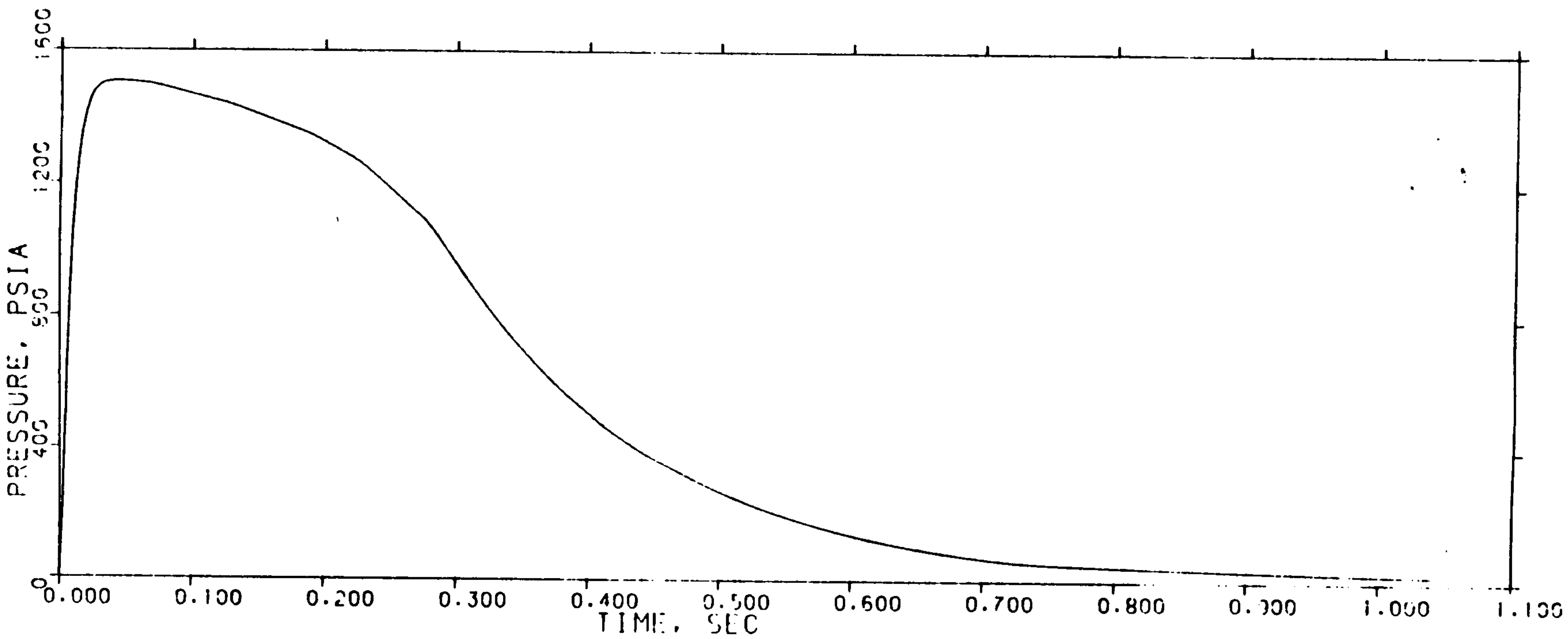


FIGURE 3.3.4. IMPROVED IGNITER PRESSURE TRACE.
SRM IGNITER NOMINAL PERFORMANCE @ 60°F.

internal case insulation is reduced by 33 percent. The igniter initiator insulation thickness was not changed.

The silica phenolic throat insert will be replaced with a lower cost, higher performance cellulose phenolic. Estimated cost savings of the cellulose versus the silica phenolic are approximately 85% for the raw material. The cellulose phenolic is also predicted to have 80% of the erosion experienced in the silica phenolic and 15% less weight due to a lower density.

These insulation thicknesses keep the igniter hardware below the required 300°F during SRM action time and 400°F following SRM web burnout per specification CPWI-3300.

3.3.7 IGNITER SEALS

The primary objective of this design study is to reduce potential exhaust gas leak paths from the igniter assembly. To achieve that goal, various hardware configuration concepts were examined to reduce the overall number of primary and secondary gas seals. In the current production igniter, there are 6 primary, 42 secondary, and 88 environmental seals as detailed in Table 3.3.3. In the proposed design, the case and adapter will be manufactured from 200 maraging steel and welded together, eliminating all case-to-adapter seals in the production design (Figure 3.3.2). Using this design, the total number of seals is reduced to 5 primary, 5 secondary, and 52 environmental. This provides a delta of 1 primary seal and 37 secondary seals. The secondary seals are drastically reduced due to the elimination of the bolts and the special bolt packing (stato-seals) in the igniter case to adapter joint.

A trade study was conducted to determine what type or types of primary and secondary seals to utilize in the improved igniter design. The trade study is presented in Table 3.3.4. The primary seals selected for use on the igniter adapter to SRM case and the S&A-to-igniter adapter are aerospace G-T rings. These rings will be radial squeeze seals and are commonly used to seal hydraulic fluid up to 8000 psi. These seals are very resistant to extrusion due built-in non-extrusion rings, provide a positive seal at zero or low pressures due to radial compression at installation and are not subject to rolling or spiral failures. These G-T ring seals permit sealing with larger gaps than o-rings under expansion of a pressure vessel as it is pressurized. Aerospace G-T rings are also designed

TABLE 3.3.3. IGNITER ASSEMBLY SEALS.

	PRESENT IGNITER			IMPROVED IGNITER		
	PRIMARY	SECONDARY	ENVIRONMENTAL	PRIMARY	SECONDARY	ENVIRONMENTAL
S&A						
• BARRIER-BOOSTER SHAFT	1	1	0	1	1	0
• NSIs	2	2	0	2	2	0
S&A TO IGNITER ADAPTER	1	1	11	1	1	11
IGNITER ADAPTER‡ TO IGNITER CASE	1	37*	37**	-	-	-
IGNITER ADAPTER TO SRM CASE	1	1	40	1	1	41
TOTALS	6	42	88	5	5	52

* 36 OF THESE 37 SEALS FUNCTION AS SECONDARY AND ENVIRONMENTAL SEAL

** INCLUDES 36 SEALS LISTED AS SECONDARY

‡ DOES NOT INCLUDE PRESSURE TRANSDUCER SEALS

TABLE 3.3.4. SEAL TRADE STUDY.

	O-RING		C-RING, METAL	STATIC FACE SEAL	FLEX METAL GASKET	G-T RING	LATHE CUT SEAL RING	GASK-O-SEAL
	RUBBER	METAL						
RESILIENCE (SPRINGBACK)	7	8	10	8	9	8	7	7
COMPRESSION SET RESISTANCE	5	10	10	7	9	7	7	5
TOUGHNESS (DAMAGE TOLERANCE)	7	4	5	8	7	8	8	7
RELAXATION MODULES	5	8	8	5	7	5	5	5
HIGH TEMPERATURE TOLERANCE	4	10	10	4	8	4	4	4
GAP SEALING CAPABILITY	5	7	10	9	4	9	6	5
MATING SURFACE FINISH/MACHINING	8	6	6	8	2	8	8	8
EXTRUSION RESISTANCE	5	10	10	8	9	8	7	6
TOTALS	46	63	69	57	55	57	52	47

RATED 1 THROUGH 10, WITH 10 HIGHEST

to fit any groove defined in specification MIL-G-5514F. Rubber compounds allow temperature coverage from -70°F to +450°F.

The secondary seals for the igniter adapter to SRM case and the S&A to igniter adapter are resilient metal c-rings. These rings will seal up to 9800 psi in a gland with a 32 RMS finish and can handle temperatures from cryogenic to 2200°F. A metal c-ring seals at low and no pressure due to compression from the flange joint. System pressure then supplements the sealing force by forcing the walls of the ring against its mating surfaces.

Both the G-T ring and metal c-ring seals are more compression set resistant than standard o-rings due to their basic designs. In dynamic loading situations where the gap between mating sealing surfaces tends to open, their resiliency assures that they will maintain contact with the sealing surfaces. By separating the seals into radial and face seals with different temperature capabilities for each, we are assured that no credible single event can cause a failure of both the primary and the secondary seal.

The aft closure will be sealed with a single static face seal that is similar in design to the G-T rings described above. It consists of an "L" shaped elastomeric sealing element and a mating non-extrusion ring. At low or zero pressure, the static face seal (SFS) seals like an o-ring. A pressure increase causes the elastomer to seal more tightly while the non-extrusion ring precludes extrusion. SFSs can seal with clearance gaps up to 0.015 inches and to pressures exceeding 10,000 psi. A single seal is utilized due to the fact that any leakage here would be into the SRM main chamber and should not compromise ignition (assuming leakage is not gross).

3.3.8 SUMMARY

The proposed igniter design utilizes a one-piece case/igniter adapter made from 200 maraging steel that is insulated internally and externally with Kevlar and silica-filled Hypalon. An aft closure bolts to the case and allows grain casting and mandrel pulling from the aft end of the igniter. An HTPB propellant, ARCADENE 360A, a variant on our well characterized MLRS propellant, will be utilized for the propellant grain. A combination of elastomeric and metal seal rings will be utilized to provide the minimum number of primary and secondary seals while providing superior sealing under all operating conditions of the SRMs.

3.4 INSULATION

3.4.1 CASE INSULATION

In Section 2.4.2 the insulation trade study identified some non-asbestos materials which were rated better than the current NBR material containing asbestos and silica. In studying the results of this effort, it became apparent that the top insulation candidates differed significantly in some of the rating categories with no one candidate excelling in all categories. In the design studies which have been performed to date, it became apparent that the ideal insulation system would logically be a combination of materials used in such a way that advantage could be taken of the best qualities of each.

Table 3.4.1 presents a design trade in which the top candidates and rankings from Section 2.4.2 have been reproduced and compared to two hybrid insulation systems. The design philosophy is to allow the low thermal diffusivity of the Hypalon/Kevlar/silica material to provide the optimum thermal protection next to the reusable case. A more erosion resistant material would be used near field joints and in areas which are exposed during propellant burn such as the aft case. As the design trade presented in Table 3.4.1 and illustrated in Figure 3.4.1 shows, the hybrid system using USR-3800 as the erosion resistant material outscores any of the individual materials. One of the advantages of the USR-3800 is the proven compatability of the NBR material with the propellant. The USR-3800 would also be used as the molded inhibitor material because of the erosion resistance required in these joint area.

The use of a hybrid system in the design poses no difficulty to the automated ribbon lay-up process which was described in Section 2.4.1. the second material can be wound directly onto the first with the resulting hybrid being vulcanized at the same time. The main concern in such a hybrid system is the quality of the bond which will be formed when the materials are vulcanized together. Initial tests have been performed at ARC in which an NBR material was vulcanized to a Hypalon/Kevlar/silica material. When pulled apart the failure occurred in the NBR rather than at the juncture of the materials. These data show that the interface is stronger than the materials themselves.

Complete thermal analyses with in-depth insulation sizing have not been completed. This work is planned for the remainder of the current program. The criteria for the trade and design studies to date were to select materials with equivalent or

TABLE 3.4.1. CASE INSULATION DESIGN STUDY.

		HYBRID/SINGLE MATERIAL COMPARISON				
RATING PARAMETER	WEIGHTING FACTOR	HYPALON SILICA/KEVLAR	EPDM SILICA/KEVLAR CELLULOSE	USR-3800 NBR/PHENOLIC BORIC ACID	HYBRID HYPALON USR-3800	HYBRID HYPALON EPDM/KEVLAR
THERMAL						
- MATERIAL AFFECTED RATE	0.18	1.26*	1.62	1.80	1.80	1.62
- THERMAL PROTECTION	0.07	0.70	0.63	0.35	0.70	0.70
MANUFACTURING/PROCESSING						
- RIBBON INTEGRITY - PLIABILITY	0.07	0.56	0.56	0.49	0.53	0.56
- GREEN TACK	0.09	0.63	0.45	0.36	0.50	0.54
- CURE REVERSION RATES	0.04	0.36	0.28	0.28	0.32	0.32
- VISCOSITY - SCORCH	0.10	0.80	0.40	0.60	0.70	0.60
COMPATIBILITY						
- VACUUM OUTGASSING	0.03	0.30	0.30	0.30	0.30	0.30
- AGING	0.07	0.70	0.70	0.56	0.63	0.70
MECHANICAL PROPERTIES						
- BOND	0.06	0.48	0.42	0.48	0.48	0.45
- STRAIN	0.03	0.12	0.09	0.24	0.18	0.11
- MODULUS	0.01	0.08	0.09	0.08	0.08	0.09
WEIGHT	0.15	0.75	1.20	1.05	0.90	0.98
COST						
- RECURRING	0.08	0.48	0.48	0.80	0.64	0.48
- NON-RECURRING	0.02	0.16	0.16	0.16	0.16	0.16
TOTAL RATING =		**7.38**	**7.38**	**7.55**	**7.92**	**7.61**

* RATING VALUE = WEIGHT FACTOR x RELATIVE PERFORMANCE

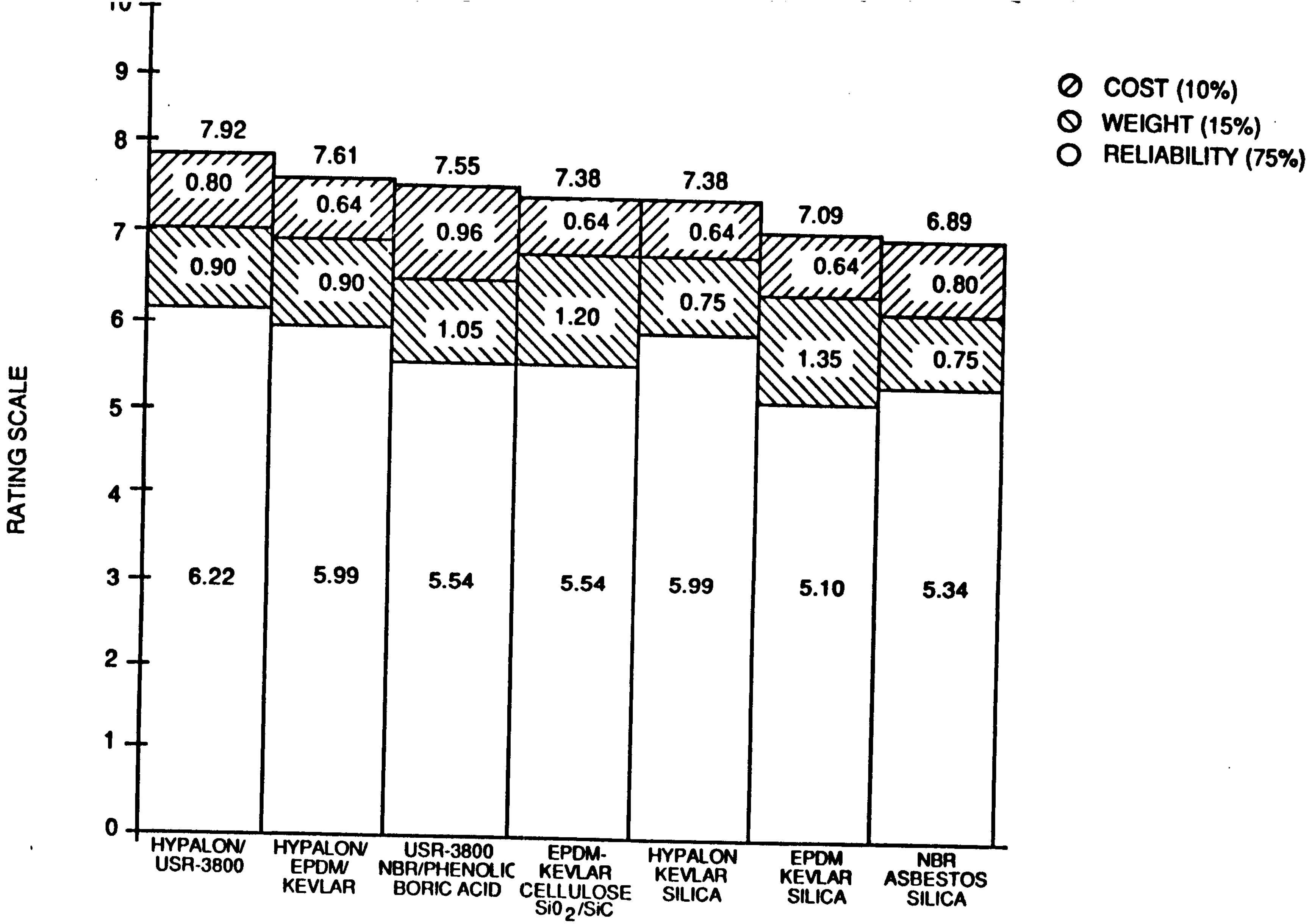


FIGURE 3.4.1. HYBRID INSULATION COMPARISON TO SINGLE MATERIAL DESIGN.

better thermal protection and erosion resistance when compared to the current insulation material. Insulation thicknesses for the Block II design will not therefore exceed those of the current HPM design. Because of the density differences (Section 3.4.4) between the proposed and current materials, the same volume of insulation (assuming 20% Hypalon and 80% USR-3800) shows a weight saving of 1600 pounds on the current HPM insulation weight of 18,670 pounds. The insulation sizing which will be performed in the remainder of this program will definitize the split between the two materials and consequently the weight savings.

3.4.2 CASE INSULATION JOINTS

Current case insulation field joints have inadequate reliability which stems from two major design flaws. The first is the use of zinc chromate putty in the gap between case insulation segments which provides a good static seal if it is applied correctly. However, zinc chromate putty does not have a sufficient sealing capability in a dynamic environment where case deflections are significant. Deflections combined with internal pressure loads can create flow paths through the putty which cause o-ring erosion if flow rates are sustained or if the o-ring is not seated. Secondly, reliance on pressure-actuated o-ring seals in conjunction with an unvented insulation joint increases the likelihood of failure. Loss of the insulation seal during motor operation can result in local erosion of the o-ring prior to it being properly seated. These problems illustrate that the selection of the insulation joint configuration must in part depend on the case seal design.

The selected case joint seal consists of a metal jacketed gasket and c-ring which are fixed in a compressive state by the use of bolted flanges. In this design, the insulation joint does not need to allow for pressure bleeding to actuate sealing. The insulation joint can be either a vented or unvented configuration. Initial studies evaluated the merits of vented versus unvented insulation joints. Evaluation criteria and their relative importance (denoted by weighting factors) are as follows:

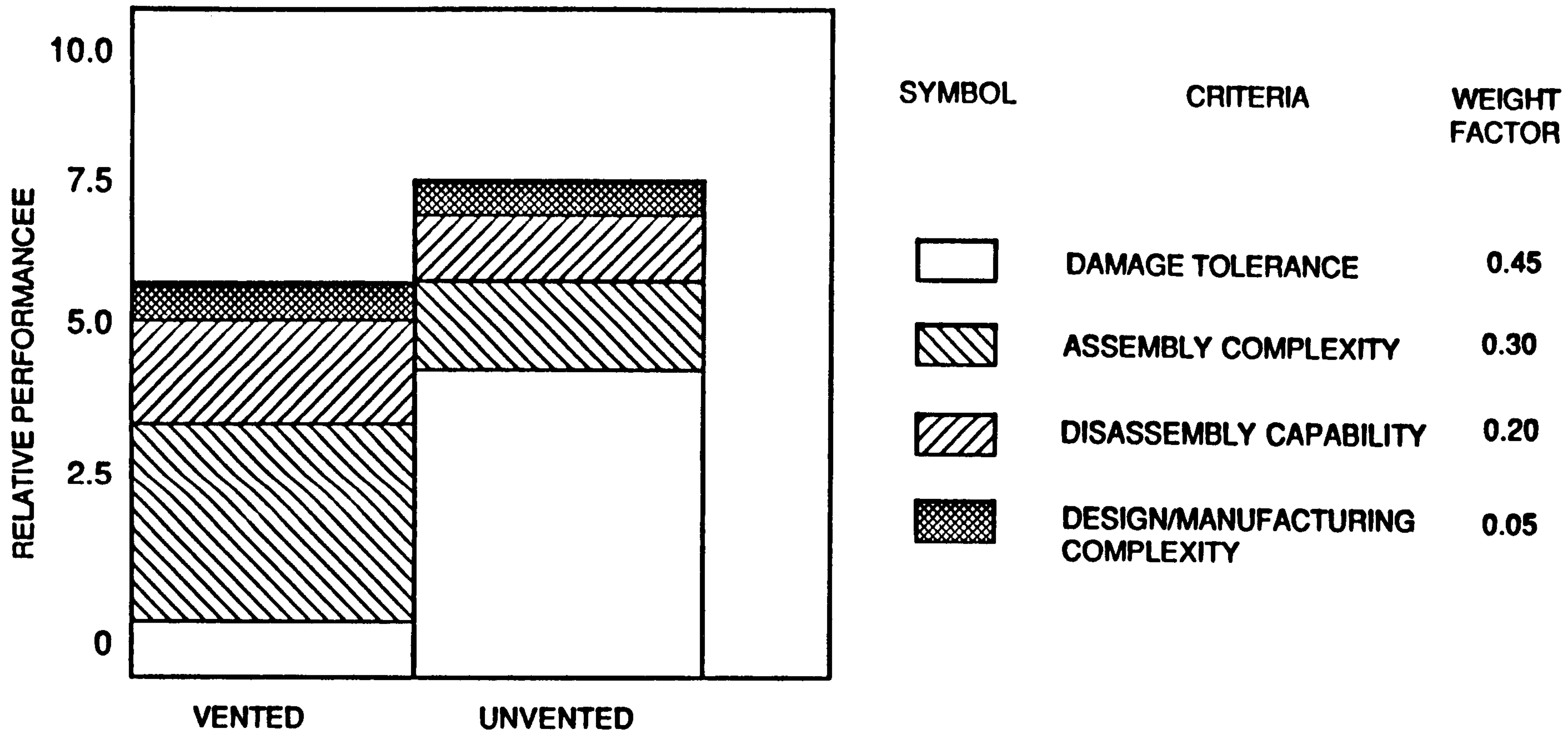
CRITERIA	WEIGHTING FACTOR
Thermal protection capability	0.45
Assembly complexity	0.30
Disassembly capability	0.20
Design/manufacturing complexity	0.05

Thermal protection capability (TPC) reflects the ability of the insulation joint to provide adequate thermal protection to the components outboard of the insulation seal (case joint, insulation/case bondline, etc.). It is the parameter which has the most impact on reliability. Long gas path flow lengths and configurations which can tolerate erosion result in large TPC ratings. Assembly complexity also affects reliability. Insulation joint designs which require a great deal of finesse to properly assemble are very susceptible to failure by being overly dependent on the workmanship of field technicians. Disassembly capability and design or manufacturing complexity are parameters which primarily impact cost and schedule. Comparisons of the relative performance between vented and unvented insulation joints, presented in Figure 3.4.2, clearly illustrate the inherent reliability of unvented joints.

Unvented insulation joint configurations used in the SRM industry are illustrated in Figure 3.4.3. Compressive sealing can only be designed to occur on one face of the joint, thereby mandating the use of an adhesive or sealant filler. These filler compounds should not be used outboard of the compressed insulation joint face since the likelihood of their extruding into and impairing the case seal is highly probable. There will, therefore, be an unfilled gap for each design which must be minimized to prevent heating of the case seals if the insulation joint fails. The estimated volume of the resulting unfilled gap was included with the previous criteria in evaluating the candidate insulation joints.

Figure 3.4.4 presents the relative performance of candidate insulation joints. In terms of reliability, the best three candidates are the labyrinth, buttress, and overlap joints. When disassembly and design/manufacturing complexity are factored in, the overlap and buttress joints are equivalent and rate slightly higher than the labyrinth joint.

The preliminary insulation joint concept selected is an overlap which utilizes an elastomeric open channel, stress relief component vulcanized within the case insulation (Figure 3.4.5). Radially oriented oval channels provide both compressive and tensile stress relief in the axial direction while prohibiting circumferential gas flow. The stress relief component is a low modulus silica/NBR which is preformed with pressure to final dimensions, tack adhered to the NBR/phenolic and vulcanized as an integral part of insulation assembly. The vertical outboard face of the overlap is toleranced to provide a nominal 0.070 inches of compression at static ambient conditions. A low tensile



*RELATIVE PERFORMANCE = Σ (WEIGHTING FACTOR X RATING), RATING = 1 TO 10 (BEST)

FIGURE 3.4.2. VENTED VERSUS UNVENTED CASE INSULATION JOINT EVALUATION.

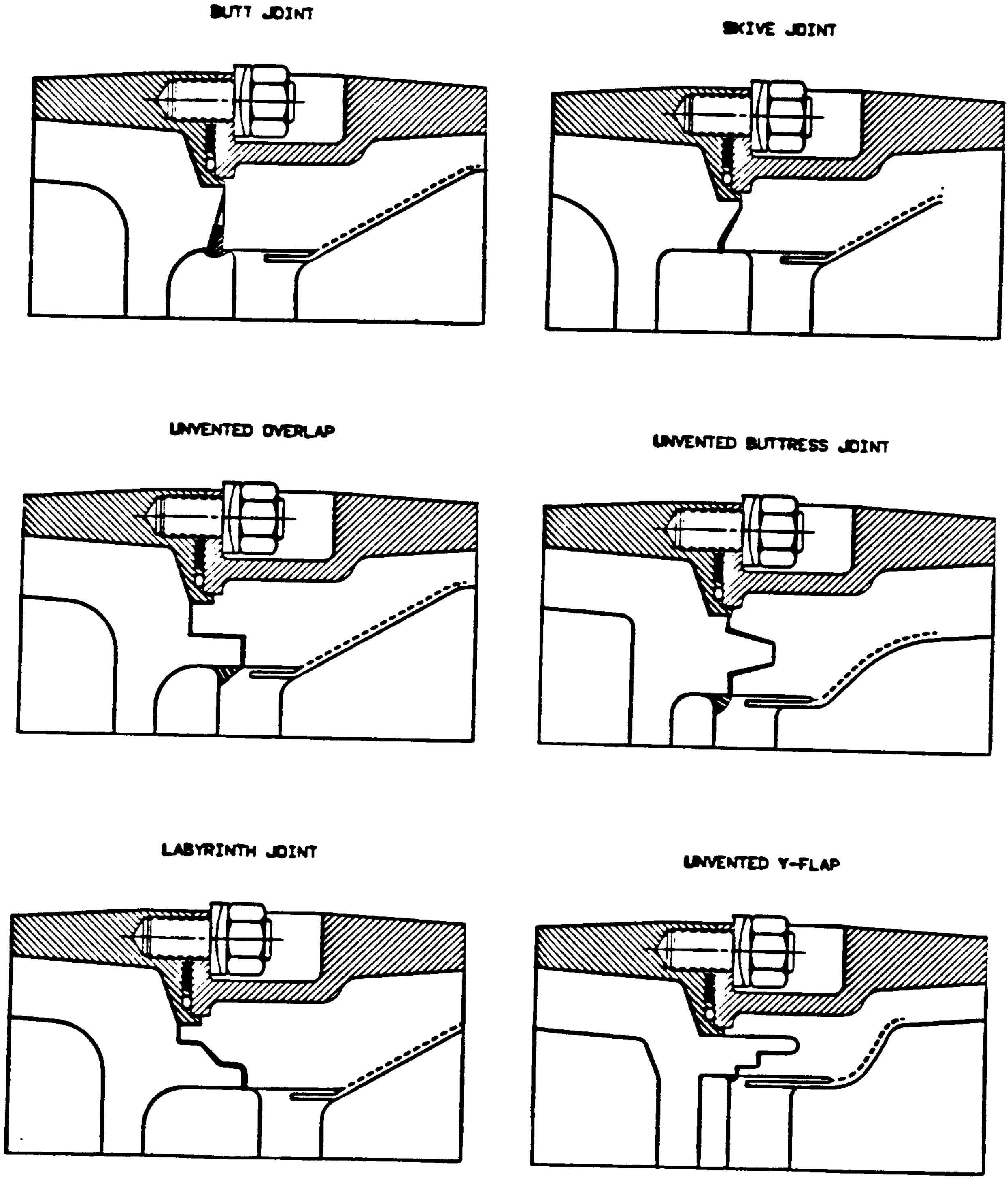


FIGURE 3.4.3. CASE INSULATION JOINT CANDIDATES.

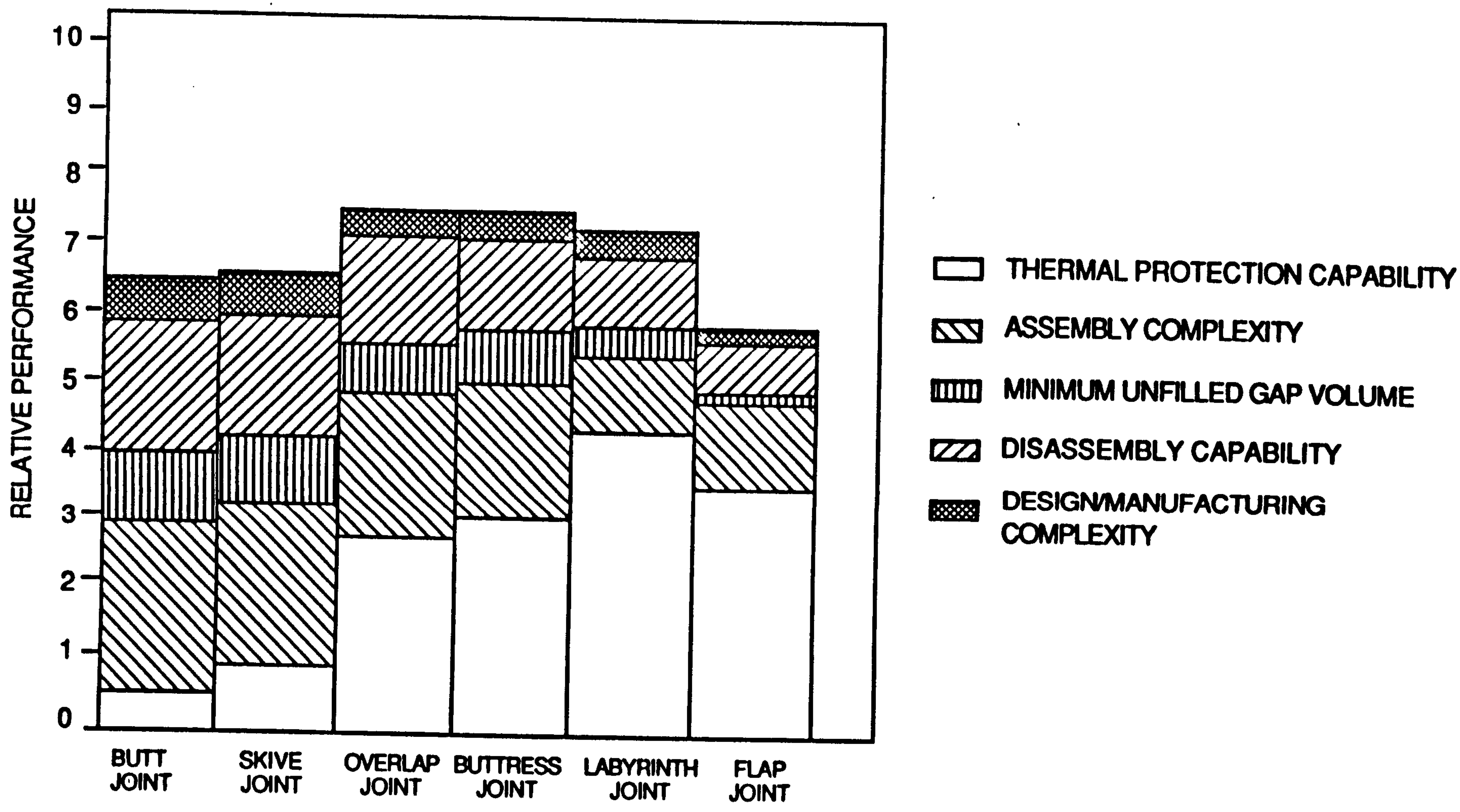


FIGURE 3.4.4. CASE INSULATION JOINT CONFIGURATION EVALUATION.

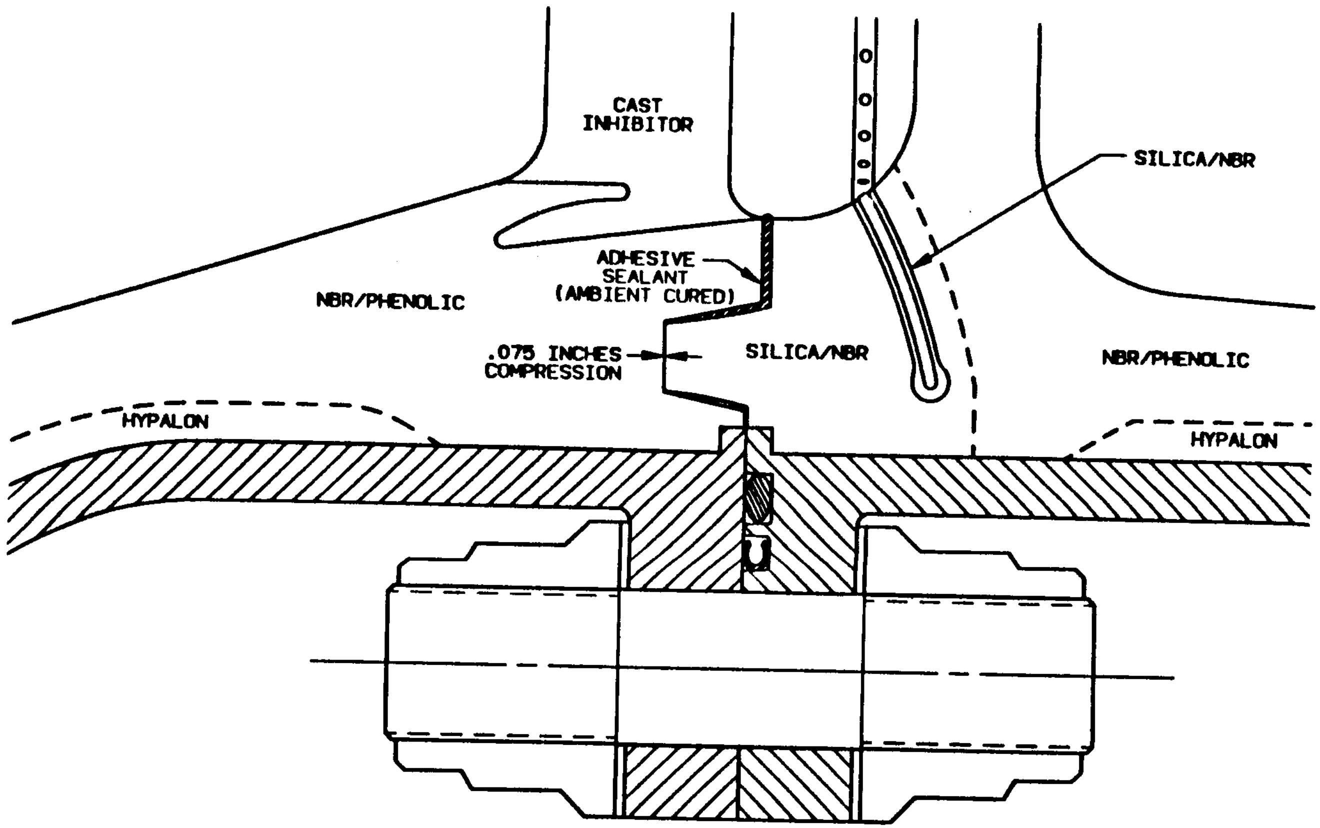


FIGURE 3.4.5. SELECTED CASE INSULATION JOINT.

strength, high strain, ambient cured, adhesive/sealant fills the gaps inboard of the compressed overlap faces. This filler is selected to have sufficient integrity for operation but will cohesively tear upon case disassembly. Pressure actuation of the stress relief channels increases the compressed length along the overlap. Case deflections or local pressure fluctuations result in flexing of the stress relief while maintaining a positive compressed insulation seal.

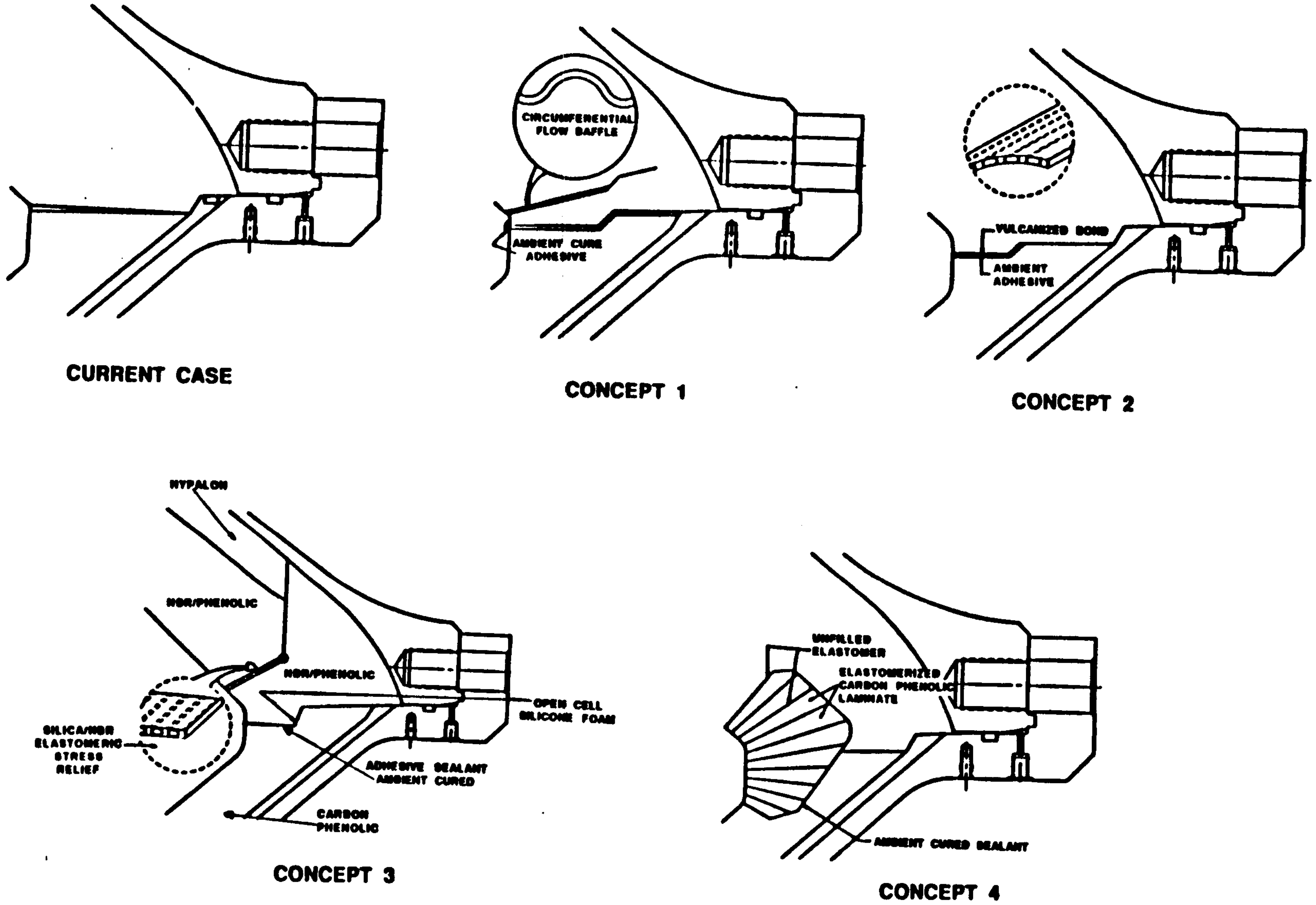
3.4.3 NOZZLE/CASE INSULATION JOINTS

The current nozzle/case insulation joint is subjected to rotation due to case and nozzle deflections during motor operation. This results in an open, vented gap which allows direct exposure of the silica phenolic insulator. In addition, the turbulent flow environment in the region sets up circumferential flow in the open gap which magnifies the potential for failure. The inherent reliability of an unvented gap resolves these problems. However, the unvented insulation joint must be capable of accommodating case and nozzle deflections.

Design trades for an unvented insulation joint primarily evaluate the capability in providing thermal protection to outboard components for various stress-relieved joint concepts. Evaluation criteria fall into three major categories: reliability, weight, and cost. Weighting factors and subdivisions of these criteria are:

CRITERIA	WEIGHTING FACTOR	
Reliability	0.75	
- Minimum cavity/gap volume		0.30
- Circumferential flow restrictions		0.25
- Assembly complexity		0.20
Weight	0.15	0.15
Cost	0.10	
- Producibility		0.05
- Disassembly capability		0.05

There are four basic stress-relieved joint concepts in addition to the current joint as shown in Figure 3.4.6. The first concept utilizes a vented, baffled, stress-relief slot in the case insulation adjacent to the joint. A second concept uses an elastomeric open channel stress relief at the joint. In the third concept, this stress relief system is vulcanized within the case insulation and open cell silicone foam is used on the vertical



NOTE: 51-L JOINT SHOWN FOR REFERENCE

FIGURE 3.4.6. UNVENTED NOZZLE/CASE INSULATION JOINT CONCEPTS.

leg of the overlap joint as a redundant flow restriction. The last concept utilizes an erosion resistant carbon phenolic and NBR laminate flex joint.

Figure 3.4.7 presents a summary of the trade study which shows the third concept to have a higher integrity and reliability. In this design (Figure 3.4.8), a NBR/phenolic is preformed under pressure to final dimensions and is tack adhered to the case. The vulcanized, silica/NBR, open channel stress relief is then similarly attached to the NBR/phenolic preform. Kevlar/silica Hypalon and NBR phenolic are ribbon wound to final dimensions and the completed aft case assembly is vulcanized. After cure, open cell silica foam is then bonded to the vertical side of the overlap. Ambient cured adhesive/sealant is placed on the inboard side of the case insulation joint prior to nozzle insertion. Pressurization of the stress relief channels allows the foam and sealant to remain in a compressed mode and rotate to an extent with the nozzle.

The first concept is the current NASA proposed redesign for this joint. While it is similar to the ARC proposed design, there are a few differences which led to the ARC design. An ambient cured adhesive is a part of both designs. As stated in the discussion in Section 3.4.1, it is necessary that this adhesive possess a low tensile strength and a high strain capability so that disassembly will result in cohesive failure of the adhesive rather than any failure of the insulation components. The NASA design incorporates a stress relief component which is bonded in place and resembles a sine wave in surface configuration. In the erosive environment in this area during motor burn, this bond might fail early inviting circumferential flow into the joint area. The bond failure itself would be in a peel mode. In the ARC design, the stress relief component is vulcanized into the insulation. This vulcanization as well as the open channel design would better resist the erosive circumferential flow as a tensile failure of the elastomeric stress relief would be required.

3.4.4 MATERIAL DATA BASE

A Hypalon/Kevlar/silica material is a recently developed material currently being used by ARC as an abestos replacement insulation in a surface-to-air missile system. This rocket motor design is currently in the qualification phase. Performance, bond, and aging data have been and are being generated in the program and will be available to aid in the Block II design. Thermal analysis material models calibrated with data from motor firings are already in existence. USR-3800 is not a newly developed material

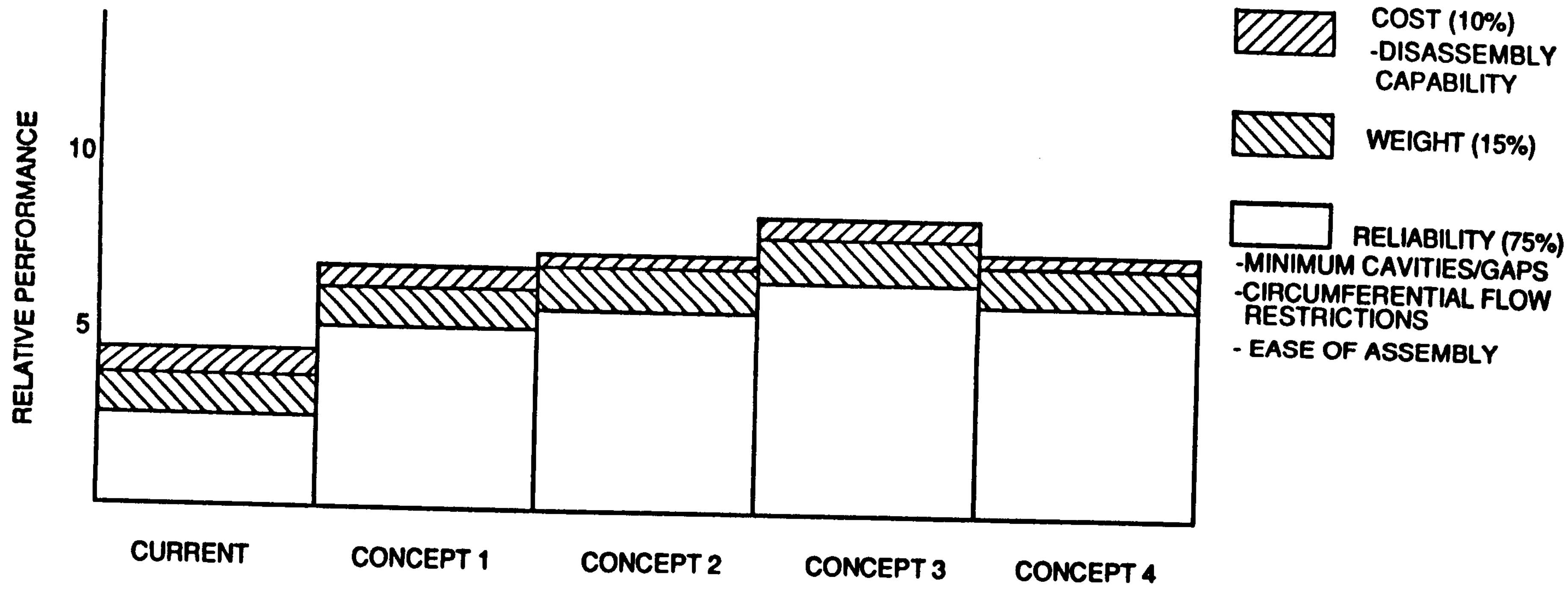


FIGURE 3.4.7. NOZZLE/CASE INSULATION JOINT EVALUATION.

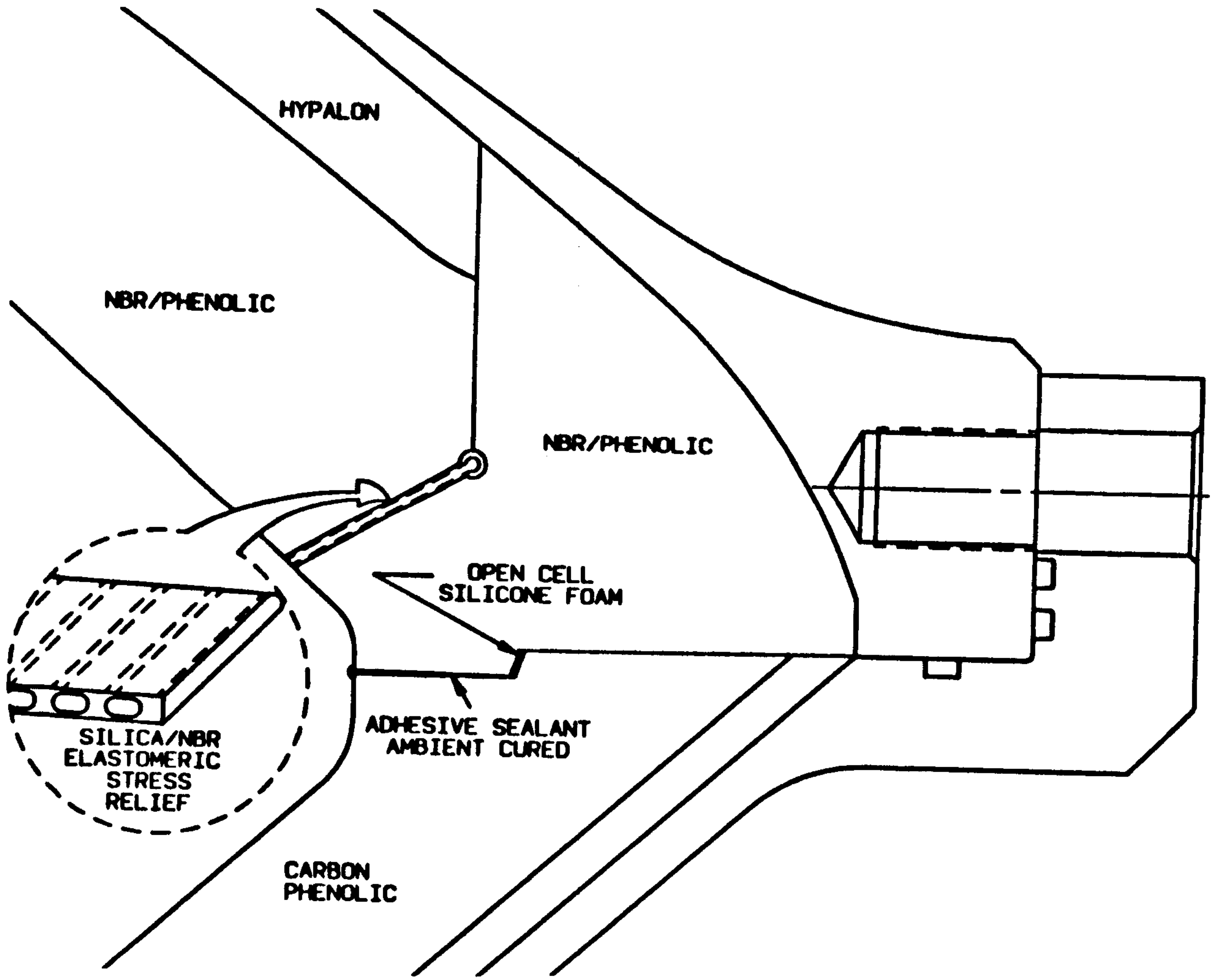


FIGURE 3.4.8. SELECTED CASE NOZZLE JOINT.

and consequently has a data base existing in the open literature. Table 3.4.2 presents a collection of some pertinent material properties for these two insulation materials as well as a comparison to the current insulation material.

In summary it is felt that both of the proposed insulation materials have been characterized to the extent necessary at this point in time to serve as viable non-asbestos insulation materials. Enough data are available for design purposes and to show that the reliability of these materials is high. The additional tests necessary to gain confidence in the Block II design will be presented in the Development and Verification Plan (Section 4.0).

3.5 PROPELLANT AND LINER

3.5.1 PROPELLANT

The current propellant forumulation (TPH-1148) has been selected for the Block II SRM design. This selection was favored from trade studies primarily because of proven reliability, performance, and experience. To our knowledge, no problems or malfunctions have ever been attributed to this propellant. However, in the event that higher performance is required or desired, offers a very credible approach for a reliable HTPB propellant with a vast data base. This propellant has been mass produced (720 million lbs) in high rate production.

3.5.2 LINER

The proposed liner composition consists of a minimum change from the demonstrated SRM liner, i.e., replacement of the asbestos with an alternate inert filler. Bonding and aging characteristics of a motor system are primarily dependent on the organic and polymeric constituents of the binder systems (propellant, liner, and insulation). Thus, if only the fibrons fillers in the liner and insulation are changed (and the propellant is unchanged), the system should remain the same in terms of bondline integrity. There will be no design difference. The major thrust of a required development effort will be to demonstrate acceptable processing and to verify that bondline performance is, indeed, unchanged.

TABLE 3.4.2. INSULATION MATERIAL DATA BASE.

MATERIAL	MECHANICAL PROPERTIES AT AMBIENT CONDITIONS: STRESS (PSI)	STRAIN (%)[1]	MODULUS (PSI)	MATERIAL AFFECTED RATE (MILS/SEC)[2]	SPECIFIC GRAVITY	THERMAL CONDUCTIVITY (BTU/ft-hr-F)[3]	DECOMPOSITION ONSET TEMPERATURE (°C)[4]
NBR/SILICA/ ASBESTOS	775	50	1,580	3.0	1.285	0.18	250
HYPALON/KEVLAR	1600	28	8,500	1.9	1.28	0.08	250
USR-3800	1100	200	26,000	1.1	1.146	0.10	250

NOTES:
1 IN FIBER DIRECTION
2 AT M = 0.1
3 AT 150°F
4 AT 10°C/MIN

Fibrons fillers to be examined in liner formations include PBI, carbon, Refrasil, Nextel, Wollastonite, Fiberfrax, and Franklin Fiber. Each will be evaluated in liner mixes on the basis of processibility, and those which process smoothly will be evaluated in bondline tests (tensile and peel).

3.6 BALLISTICS

Included as one of the design study areas was an investigation into the necessary SRM modifications to produce the "heads-up" thrust history. The required nominal burning rate thrust history bandwidth at 60°F was presented in Enclosure 22 of a letter from Larry Wear; subject "Responses to Block II SRM Requests." Simply stated, the heads-up thrust history requires a 10% increase in thrust level with a reduction in burning time to produce the same total impulse as required by Specification No. CPW1-3600. Figure 3.6.1 presents the required thrust band presented in CPW1-3600 while Figure 3.6.2 presents the heads-up thrust requirement.

Before heads-up modifications were investigated, a simple ballistic prediction model was established for the current High Performance Motor (HPM). Figure 3.6.3 presents the baseline predicted thrust which agrees well with the required bandwidth. Figure 3.6.4 compares the predicted pressure history with the nominal HPM curve. This simple prediction model does not include erosive burning behavior. This accounts for the low predicted pressure over the initial 15 seconds and the higher level during the 85 to 105-second time span. The predicted delivered vacuum specific impulse is 267.19 lbf-s/lbm.

Four different design parameters were investigated in an effort to produce the heads-up thrust history with minimum impact on the existing design. Combinations of propellant burning rate, propellant formulation, nozzle throat geometry, and propellant burning surface area versus web distance were studied. A number of different approaches were identified to produce the heads-up thrust history. Each approach also has a negative impact to some extent. For each approach, the impact of payload was calculated, and SRM components were identified that would be impacted by the approach. Table 3.6.1 summarizes the impact of the approaches that will be discussed.

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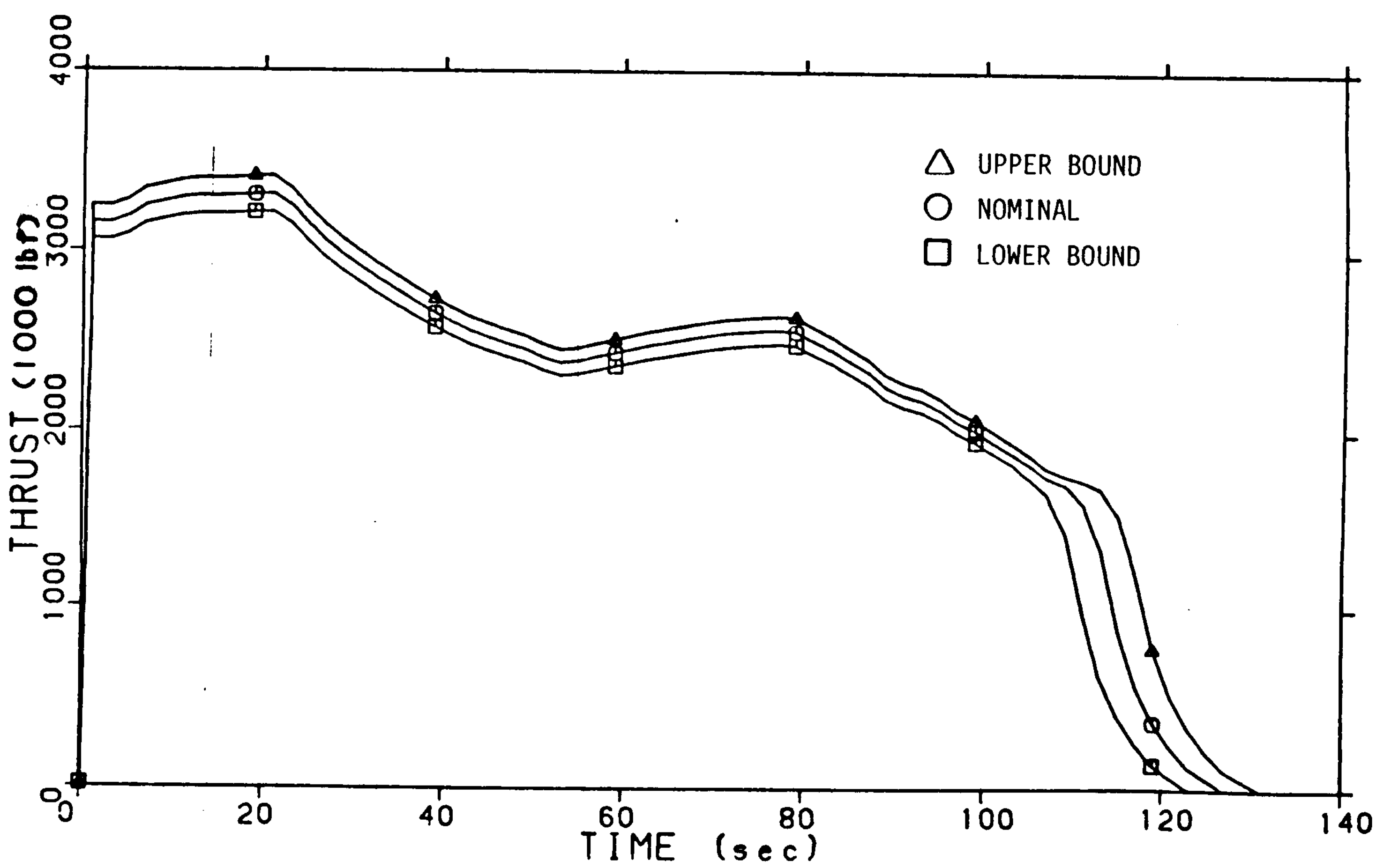


FIGURE 3.6.1. SPECIFICATION NO. CPW-3600 REQUIRED THRUST BAND.

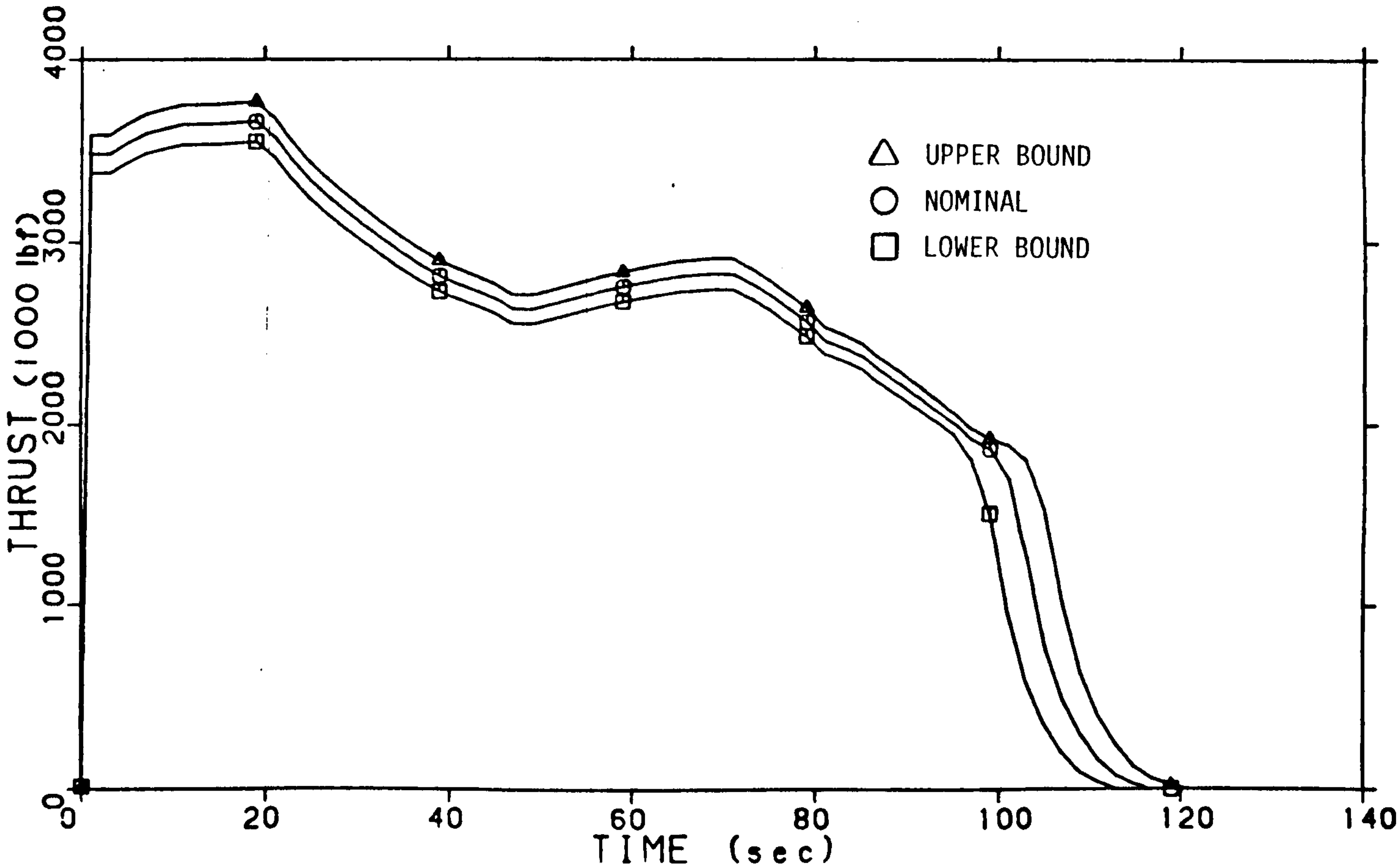


FIGURE 3.6.2 REQUIRED "HEADS-UP TRAJECTORY" THRUST BAND.

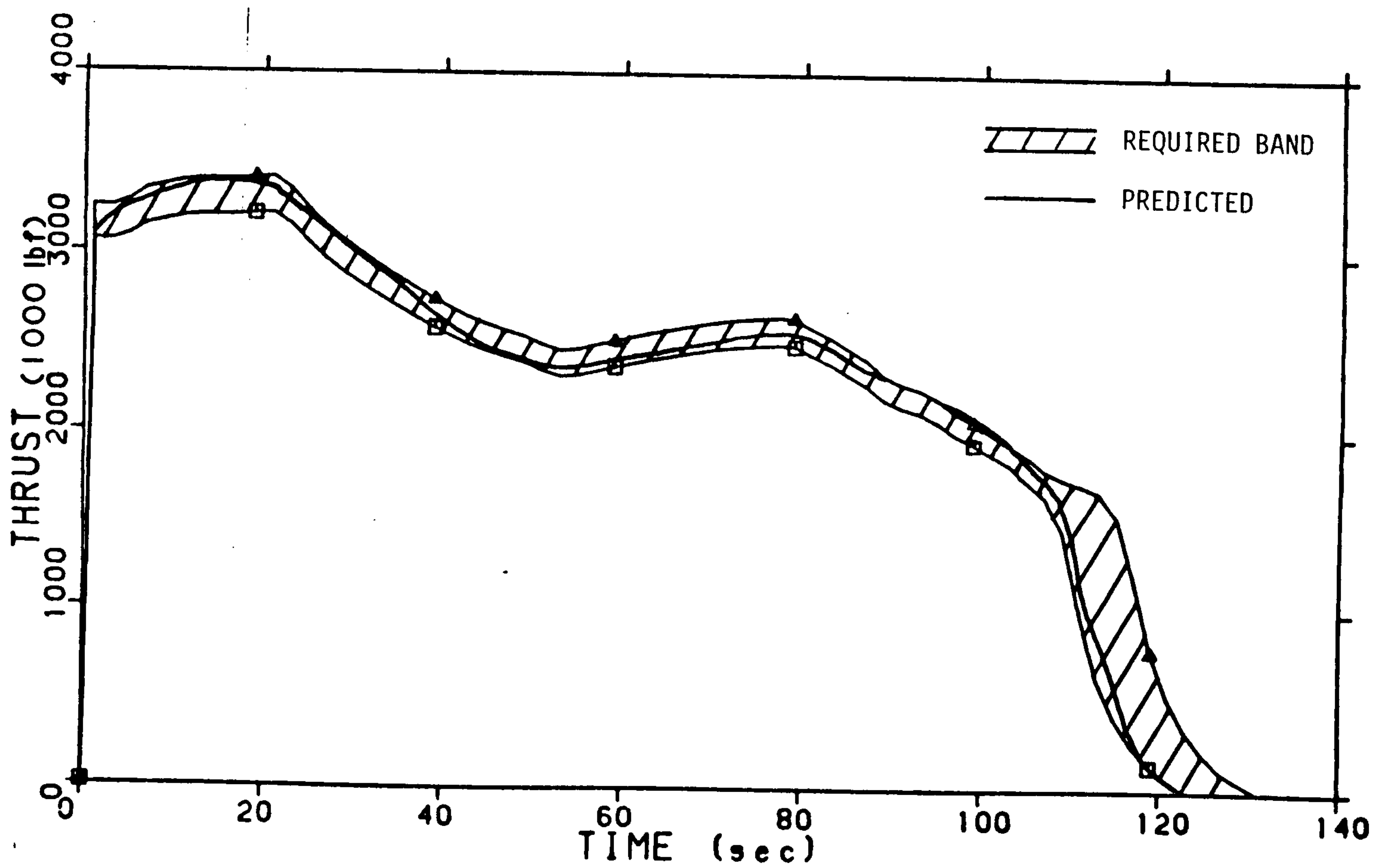


FIGURE 3.6.3. PREDICTED NOMINAL HPM THRUST HISTORY AT 60°F.

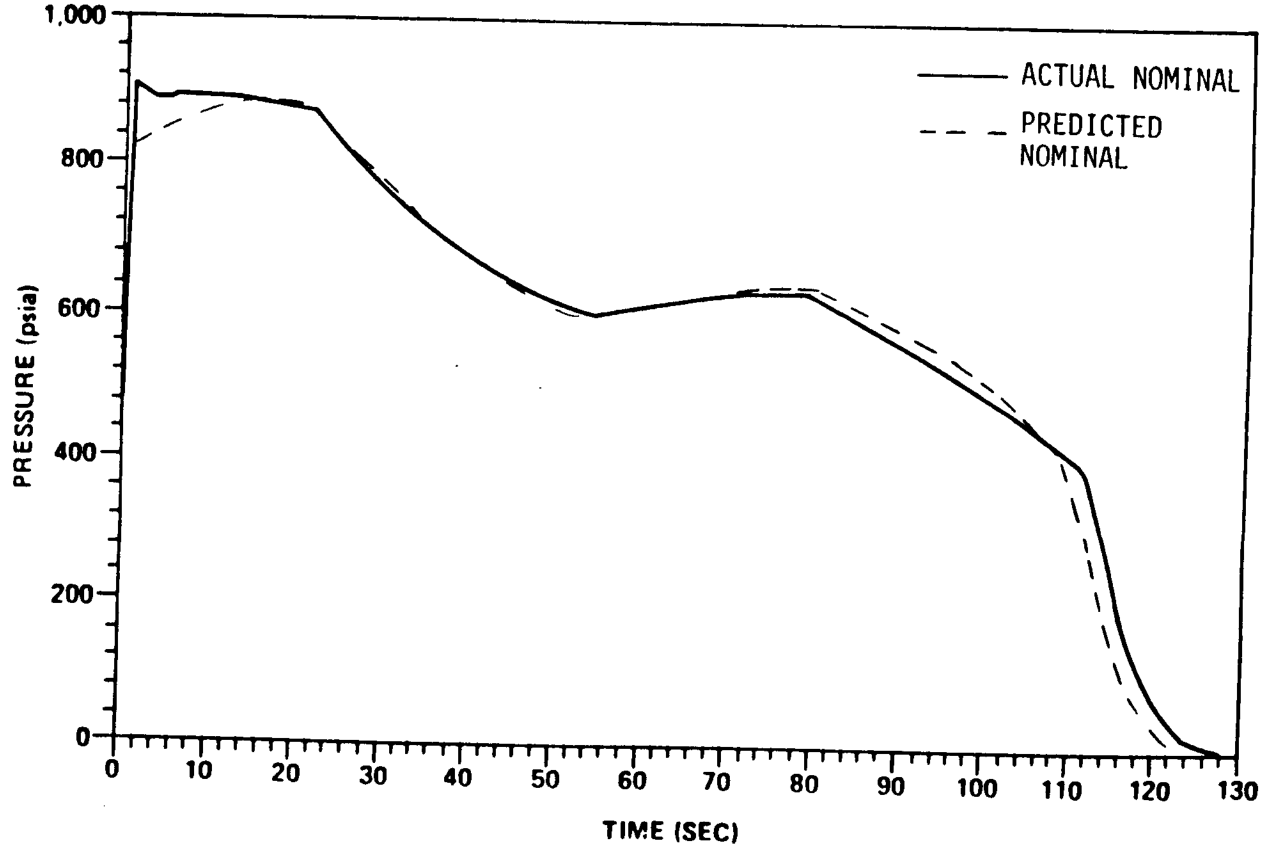


FIGURE 3.6.4. PREDICTED NOMINAL HPM THRUST PRESSURE HISTORY AT 60°F.

TABLE 3.6.1. SUMMARY OF HEADS-UP THRUST MODIFICATIONS.

	APPROACH			
	1	2	3	4
REQUIRED BURNING RATE, in/s	0.386	0.399	0.362	0.378
NOZZLE DIAMETER, in	53.86	56.49	51.00	53.86
MEOP, psia	1,120	1,015	1,215	1,120
PAYLOAD IMPACT W/D6AC CASE THICKNESS INCREASE, lbm	-1,800	-1,560	- 887	- 40
PAYLOAD IMPACT W/HIGHER STRENGTH STEEL @ CURRENT THICKNESS, lbm	0	-1,560	+2,576	+1,840
COMPONENT DEVELOPMENT/ REDESIGN REQUIRED*	NONE	NOZZLE STRUCTURE AND FLEX-BEARING	20% MEOP INCREASE LIKELY HIGH ENOUGH TO REQUIRE COMPLETE REDESIGN	HT PROPELLANT DEVELOPMENT

* ASSUMES CURRENT PBAN CAN BE TAILORED TO REQUIRED RATE.

The first approach investigated was to merely increase the burning rate of the current propellant. This can be accomplished by adjusting the amount of iron oxide (Fe_2O_3) and/or the ground to unground ratio of ammonium perchlorate (AP). The required base burning rate at 625 psia and 60°F was determined to be 0.386 in/s compared to the current 0.362 in/s baseline. Figure 3.6.5 presents the predicted thrust history based on a burning rate of 0.386 in/s.

While the required thrust band is achieved, this approach would increase MEOP to approximately 1,120 psia. The increase in MEOP would prohibit use of existing hardware. The required increase in case wall thickness if D6AC steel is retained would result in an approximate 1,800 lbm payload capability loss. If an alternate case material such as 250 maraging steel with a heat treat of 210 ksi or greater is used, the existing case wall thickness can be retained and essentially no payload loss would be required. This approach will involve no additional redesign or development beyond the concept currently proposed for the Block II SRM.

The second approach also involved an increase in burning rate, but nozzle throat area was also increased to retain MEOP at its current level. The required burning rate and initial throat diameter were determined to be 0.399 in/s and 56.490 in., respectively. Other minor nozzle modifications would include re-optimization of the nozzle exit cone contour and use of a carbon phenolic material with a fifteen percent lower nozzle erosion rate.

While satisfying, in general, the required thrust band, the resulting delivered specific impulse is reduced to 265.24 lbf-s/lbm from the current 267.19. The loss of 1.95 lbf-s/lbm delivered specific impulse results in a payload reduction of 1,560 lbm. Also, the increase in nozzle throat diameter is sufficiently large enough to require the redesign of the metal back-up structure of the nozzle.

The third approach investigated was a nozzle throat diameter reduction. Like the first approach, this method will increase MEOP. However, the nozzle diameter reduction will increase the delivered specific impulse. Also, the reduction will structurally allow the use of carbon-carbon as the nozzle throat material (lower erosion rate) within the envelope of the existing metal parts. The nozzle diameter required for this approach is 51.0 inches.

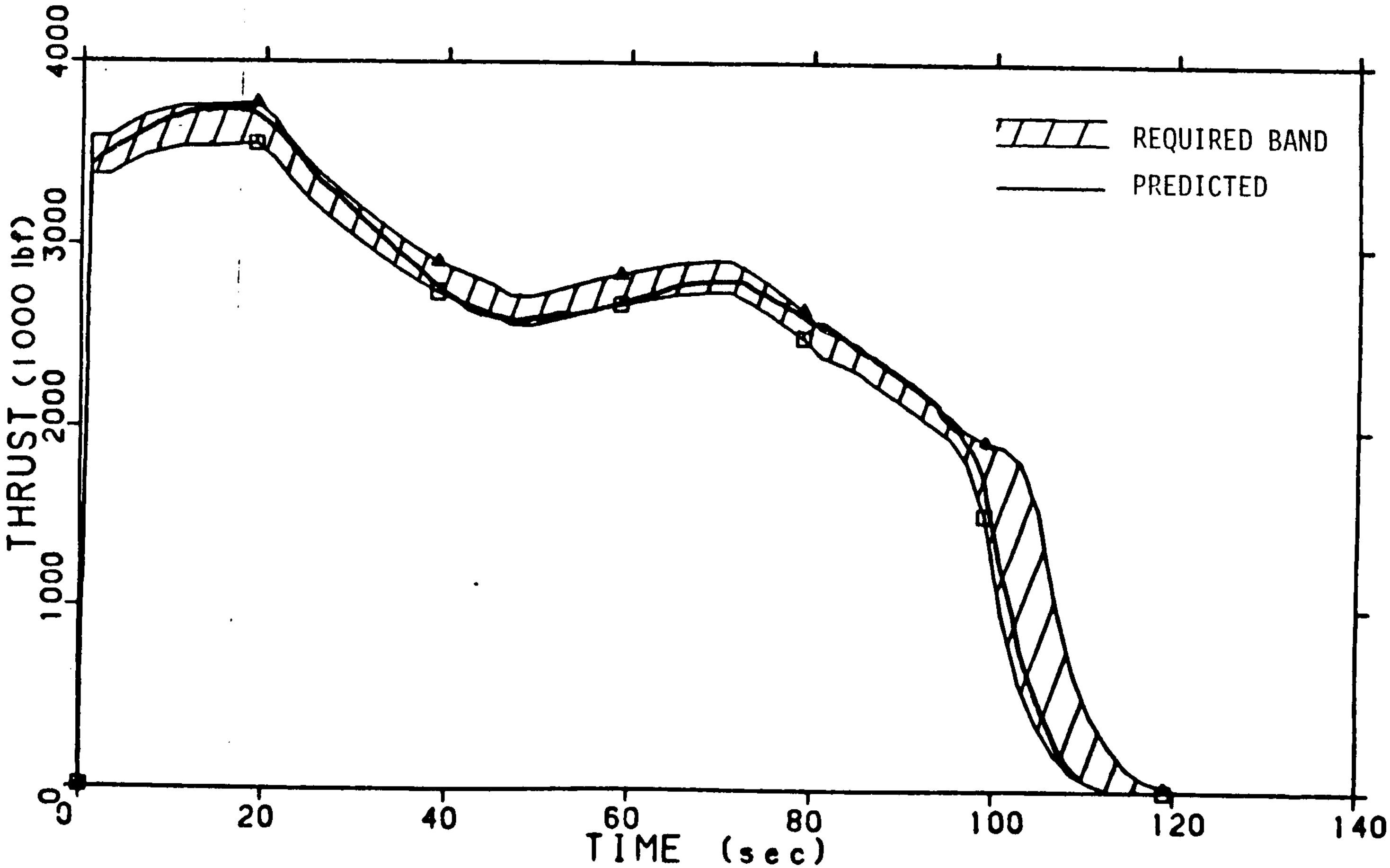


FIGURE 3.6.5. PREDICTED HEADS-UP THRUST HISTORY AT 60°F AND r_b = 0.386 IN/S.

Again, the desired thrust band is achieved. MEOP is 1,215 psia, but the delivered specific impulse is increased to 270.41 lbf-s/lbm. If the current case material is retained, the delivered I_{sp} and increased inert weight trade to yield a 887 lbm decrease in payload. If a higher strength steel is used for the case material allowing the case wall thickness to remain unchanged, the delivered I_{sp} increase will produce a payload increase of 2,576 lbm. It is likely that this great an increase in MEOP will cause other components besides the case wall to increase in weight. Therefore, the payload capability would likely decrease somewhat below the number just presented.

The fourth approach considered was the use of a higher energy and density HTPB propellant. The HTPB propellant will also likely have a pressure exponent which is somewhat higher than that of the PBAN propellant. This will require tailoring of the burning surface area versus web history such that the maximum burning surface area and the minimum surface area are closer to the web averaged mean. The required burning rate for the HTPB propellant would be 0.378 in/s at 625 psia and 60°F.

Again, this option results in raising MEOP to approximately 1,120 psia. If the current case material were used, the increased wall thickness would result in a net 40 lbm payload increase. The increased wall thickness is offset by a 0.6 lbf-s/lbm increase in theoretical specific impulse and 20,800 lbm extra propellant due to the higher propellant density. Also included is a 2,000 lbm increase in the insulation required for HTPB propellant use. If the wall thickness can be maintained, the net payload increase would be 1,840 lbm.

In addition to the four specific options discussed, a great number of other specific combinations of the four parameters exist. The specific approach chosen will largely be driven by the design details of the Block II SRM. It is clear however, that the current nozzle flex-bearing support structure cannot accommodate the increase in nozzle throat diameter (56.49 in.) required to maintain MEOP at its current level. Therefore, if the Block II SRM is to produce the heads-up thrust history, some increase in MEOP will result.

Based on the Block II SRM design studies to date, the best way to achieve the heads-up trajectory is the first approach (raise burning rate and maintain nozzle throat diameter). This is largely based on the minimum amount of additional development or redesign associated with this approach. The increase in burning rate will likely decrease the amount of erosive burning somewhat. This will help reduce the MEOP increase from 1,015 to a value somewhat below the 1,120 psia level which assumes the same erosive burning effect. Also, at this point, it is likely that a steel with a higher strength then D6AC will be suggested for the Block II design. These factors along with some minor burning surface area adjustments in the head-end segment should combine to significantly reduce the 1,800 lbm payload loss that was associated with the first approach.

4.0 DEVELOPMENT AND VALIDATION PLAN

Figure 4.0.1 provides a brief summary of the Preliminary Development and Validation (D&V) Plan which will be completed in the remaining half of this program. The primary features distinguishing the Block II from the current Morton Thiokol Corporation (MTC) Redesign were discussed in Sections 2.0 and 3.0. The Block II SRM D&V Schedule shown in Figure 4.0.2 is paced by the facility construction required to support a stretched case segment. During the first year of the program, detailed design and analysis of the Block II SRM components will be supported by component and subscale testing. Motor processing and handling facilities, equipment, and tooling will be designed and fabricated. Initial facility and equipment studies show promise for several cost effective improvements to current SRM manufacturing and handling methods.

4.1 ANALYSIS

Structural, thermal, and gas dynamic analysis will be conducted on all components and assemblies at worst-on-worst case conditions of SRM environment and manufacturing tolerances. Analytical methods will be validated against current NASA SRM data base.

Preference will be shown for analysis by similarity where appropriate. Since trade criteria was weighted heavily in favor of choices with existing data base to assure integrity in Block II components and materials, the existing data base will be carefully analyzed for validity of similarity in its SRM application.

4.2 COMPONENT TESTS

Table 4.2.1 summarizes the component tests recommended for the Block II Concept Design. These will be further defined during the next several weeks.

4.3 SYSTEMS LEVEL TESTS

Full-sized development and qualification testing is paced by availability of case hardware. The large case and nozzle simulators provide an excellent medium to test full-size motor joints cost effectively. ARC has chosen to use the same JES and NJES design for Block II SRM testing.

COMPONENTS/ ITEMS — ANALYSIS & TEST CONFIGURATION	CASE							NOZZLE				NON-ASBESTOS MATERIAL						IGNITER						ASSEMBLY	TOTAL SYSTEM
	SEALS	METAL JOINT	INSULATION JOINT	INSULATION	WELD	MATERIAL	FIELD SEALS	METAL JOINT	INSUL JOINT	FACTORY SEALS	COMPOSITES	CASE	INSULATION	INHIBITOR	LINER	S&A PARTS	CABLE TAPE	SEALS	METAL JOINTS	INSULATION	PROPELLANT	LINER	FUNCTION		
ANALYSIS																									
THERMAL	X	X	X	X		X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X			X
GAS DYNAMICS	X		X	X			X		X		X		X					X		X					
STRUCTURAL	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X
TEST																									
COMPONENT	X			X	X	X	X			X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	
SUBSCALE		X	X	X	X	X			X			X	X	X			X	X	X	X	X	X			
SIMULATORS																									
JES		X	X															X	X	X	X	X	X		
NJES							X	X	X									X	X	X	X	X	X		
NSTA							X	X		X	X														
REFEREE	X	X	X			X												X	X	X	X	X	X	X	
ATA	X	X	X			X																			
TPTA					X	X	X	X	X			X						X	X	X	X	X	X		
STA					X																				
FULL-SCALE																									
ETM	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	
DM-10	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	
DM-11	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	
FLIGHT CONFIG.																									
DM-12	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X
QM-9	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X
QM-10	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X
QM-11	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X

FIGURE 4.0.1. DEVELOPMENT AND VALIDATION PLAN.

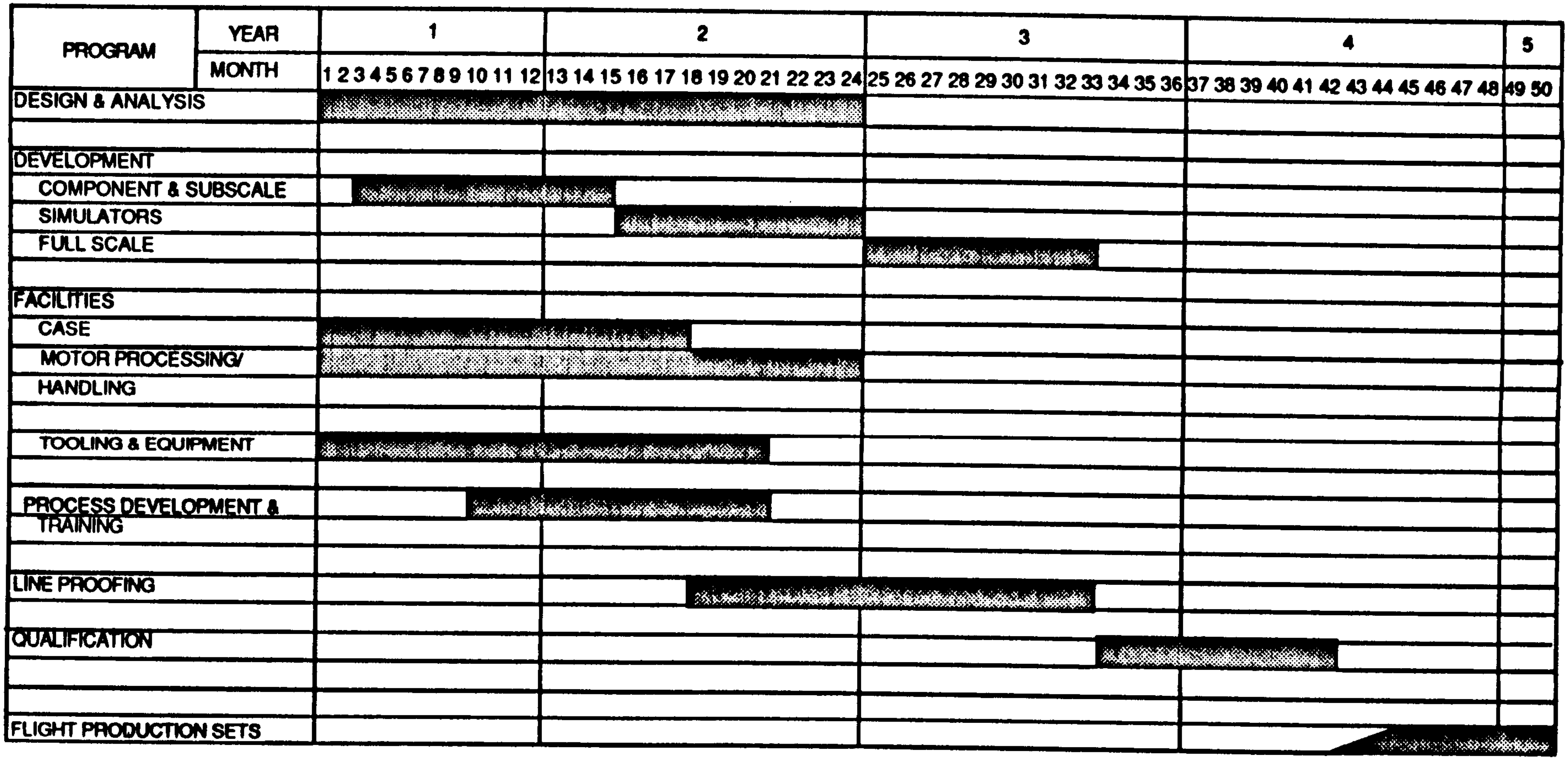


FIGURE 4.0.2. BLOCK II SRM DEVELOPMENT AND VALIDATION SCHEDULE.

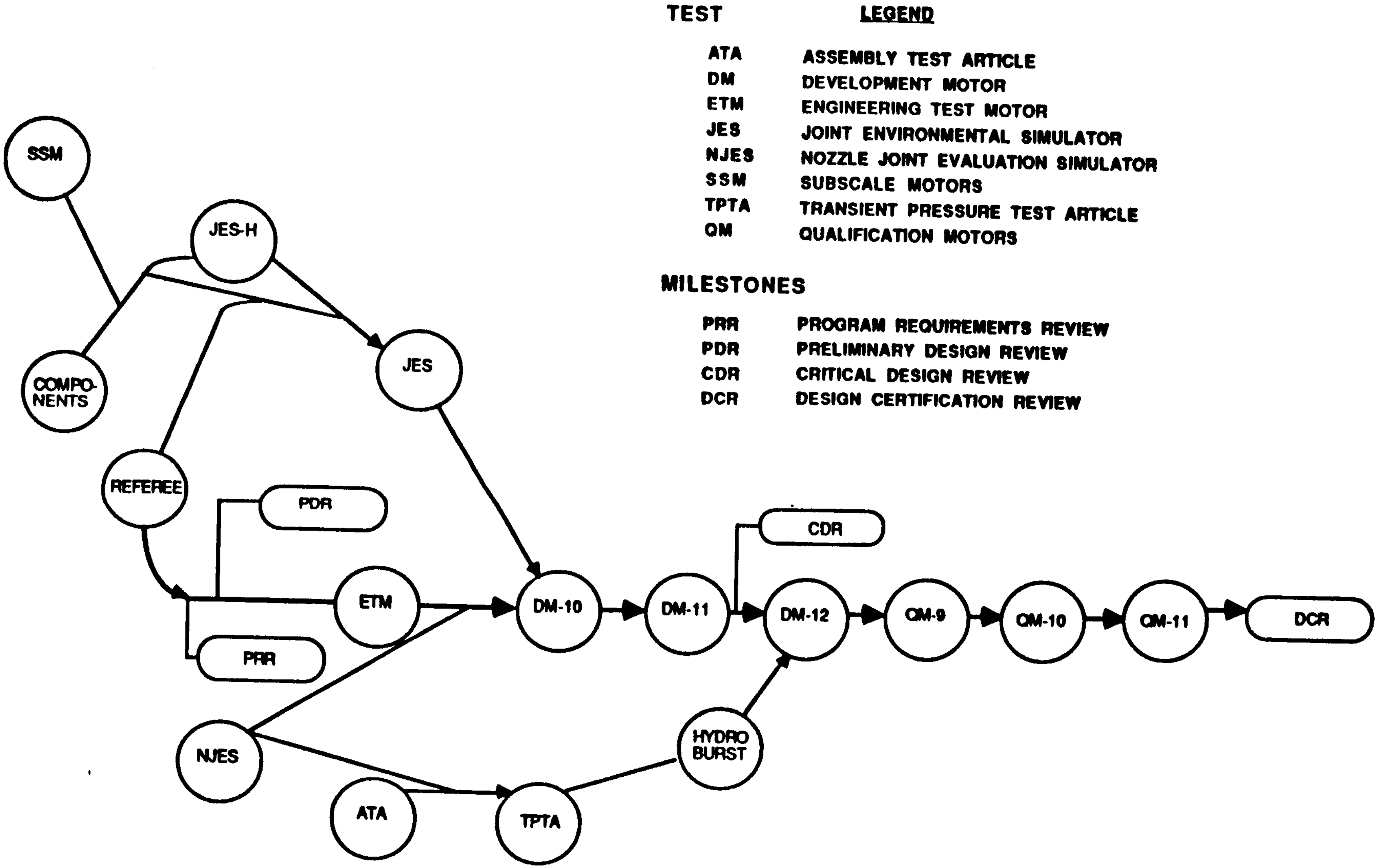


FIGURE 4.0.3. VERIFICATION TEST FLOW LOGIC.

TABLE 4.2.1. COMPONENT TESTS.

<u>CASE</u>

- MATERIAL PROPERTIES WITH FORCED RANGE OF COMPOSITION
- WELD CHARACTERIZATION - FLAT & CURVED PLATE, STANDARD SPECIMEN, SUBSCALE AND FULL DIAMETER, METALLUGICAL ANALYSIS
- FABRICATION RING TEST SERIES
- MANUFACTURING TOLERANCE TEST SERIES
- HANDLING EQUIPMENT/FABRICATION/MACHINERY FIXTURE FUNCTIONAL SERIES
- STRETCHED LENGTH SEGMENT HEAT TREAT
- HYDROTEST
- HYDROBURST

<u>CASE FIELD JOINT</u>

- FABRICAITON TESTS - FLAT/CURVED SECTIONS
- BOLT TORQUE TECHNIQUE
- BOLT STRESS CORROSION
- SEAL ENVIRONMENTAL TOLERANCE AND AGING
- SEAL HANDLING
- INSULATION STRESS RELIEF SUBSIDE FABRICATION 7 LOAD TESTS

<u>NOZZLE/CASE JOINT</u>

- SEAL ENVIRONMENTAL TOLERANCE AND AGING
- SEAL HANDLING
- INSULATION STRESS RELIEF SUBSIDE FABRICATION 7 LOAD TESTS

<u>NOZZLE JOINTS</u>

- SEAL ENVIRONMENTAL TOLERANCE AND AGING

<u>INSULATION</u>

- CHEMICAL/STRUCTURAL CHARACTERIZATION
- UNIAXIAL & BIOXIAL STRUCTURAL PROPERTIES
- AGING TESTS
- SUBSCALE MOTOR EROSION CHARACTERIZATION

<u>NOZZLE COMPOSITES</u>

- STRUCTURAL TESTING
- SUBSCALE MOTOR EROSION CHARACTERIZATION

<u>NON ASBESTOS LINES</u>

- BOND CHARACTERIZATION
- CHEMICAL AND STRUCTURAL CHARACTERIZATION
- AGING

<u>IGNITER</u>

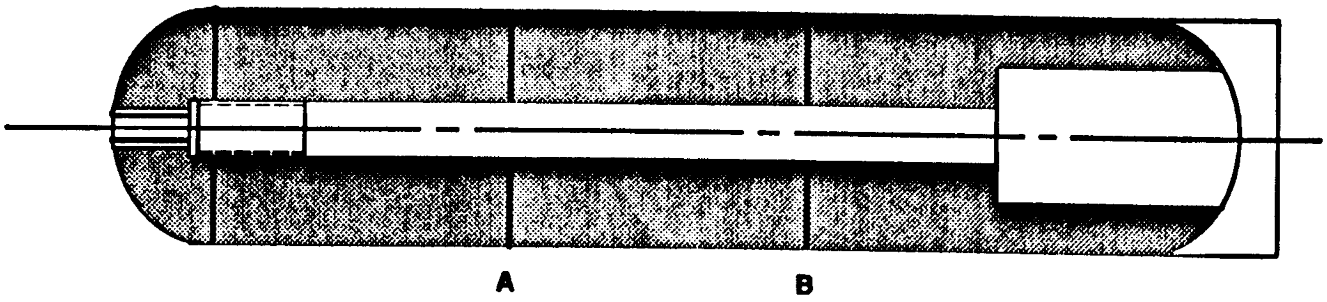
- SEAL ENVIRONMENTAL TOLERANCE AND AGING
- SEAL HANDLING
- PROPELLANT/LINER/INSULATION BOND CHARACTERIZATION
- HARDWARE HYDROBURST
- OPEN AIR IGNITER STATIC TESTS (HIGH FREQUENCY PC TRANSDUCERS)
- IGNITER ASSEMBLY ENVIRONMENTAL TESTS
- IGNITER QUALIFICATION
- MAIN MOTOR GRAIN IMPINGEMENT TEST

The case Joint Evaluation Simulator (JES) test series is shown in Figure 4.3.1. Figure 4.3.2 summarizes the Nozzle Joint Evaluation Simulator (NJES) testing. One "short stack" Assembly Test Article will be used to evaluate initial Bolted Joint Hardware at Camden, Arkansas. The Nozzle Structural Test Article (NSTA) will be used to evaluate Nozzle Process Development Units.

Ten (10) Transient Pressure Test Article assemblies will be provided to NASA for testing at MSFC. Two (2) Assembly Test Article sets of Bolted Joint Hardware will be shipped to Kennedy Space Center for functional assembly checkout.

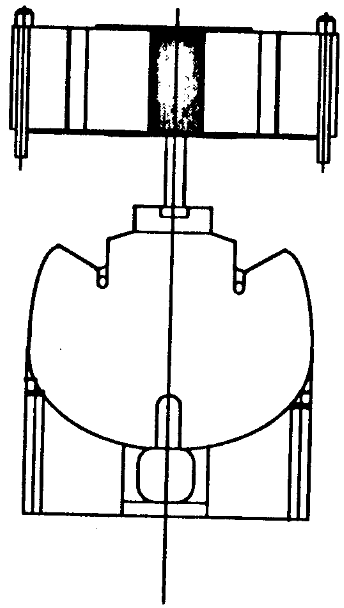
An Engineering Test Motor (ETM) will be manufactured using existing case hardware. All insulation joints, case insulation, igniter and other minor design features will represent the Block II SRM design. Use of existing F1-L type existing hardware will allow early full-size testing.

Development motor DM-10 and DM-11 will allow factory joints between new bolted field joint case segments, again to facilitate early testing. DM-12 will be flight-weight, complete Block II configuration and subjected to sequential environmental exposures prior to static test. Three (3) qualification motors have been selected to provide confidence in Block II design integrity.



CONFIG.	JES 1		JES 2		JES 3		JES 4		JES 5		JES 6		JES 7		JES 8		JES 9		JES 10	
JOINT	A	B	A	B	A	B	A	B	A	B	A	B	A	B	A	B	A	B	A	B
SEAL MATERIAL																				
● C-RING		X			X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X
● FLEXITALLIC GASKET		X		X	X		X		X	X	X	X	X	X	X	X	X			X
● ARCTIC NITRILE O RING	X		X	X		X		X												
MTC INSULATION DESIGN	X		X			X	X													
ARC INSULATION DESIGN																				
● MOD 1 STRESS RELIEF		X			X				X	X	X	X	X	X	X	X	X	X	X	X
● MOD 2 STRESS RELIEF				X				X												
● FULL BOND										X	X									
● PARTIAL BOND		X		X	X			X	X			X	X	X	X	X	X	X	X	X
SEAL DAMAGE																				
● PRIMARY												X								
● SECONDARY														X						
● INSULATION																X				
PRIMARY SEAL OMITTED																		X	X	
TEST JOINT TEMPERATURE																				
● 20^0F	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X				
● 0^0F																	X	X	X	X
● CAPTURE FEATURE JOINT	X		X																	
BOLTED JOINT		X		X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X	X

FIGURE 4.3.1. SRM CASE JOINT EVALUATION SIMULATOR TEST SERIES.



	NJES H-1	NJES H-2	NJES 1	NJES 2	NJES H-3	NJES 3	NJES 4	NJES 5	NJES 6	NJES 7	NJES 8	NJES H-4
INSULATION DESIGN												
● STRESS RELIEF #1	X		X		X	X	X	X	X	X	X	X
● STRESS RELIEF #2		X		X								
● PARTIALLY BONDED	X	X	X	X	X							
● FULLY BONDED												
JOINT												
● BASELINE	X	X	X	X			X	X	X	X	X	X
● MOD #1					X	X						
SEAL												
● C-RING	X	X	X	X	X	X	X	X	X	X	X	X
● ARCTIC NITRILE/SILICONE	X	X	X	X	X	X	X	X	X	X	X	X
SEAL DAMAGED												
● PRIMARY								X				
● SECONDARY									X			
● INSULATION										X		
PRIMARY SEAL												
● OMITTED											X	
TEST CONDITIONS												
● HYDROTEST	X	X			X							
● HYDROBURST												X
● HOT FIRING			X	X		X	X	X	X	X	X	

FIGURE 4.3.2. SRM NOZZLE JOINT EVALUATION SIMULATOR TEST SERIES.

5.0 STUDY COMPLETION PLAN

The remainder of the Block II SRM Concept Study Program will continue at its planned accelerated pace. The program effort shown on Figures 5.0.1 and 5.0.2 is currently on schedule and will continue to meet milestones and committments. The Final Report shall be divided into two parts with Tasks 1.0 and 2.0 in Part 1 and Tasks 3.0 and 4.0 in the second.

5.1 TASK 1.0 DESIGN

The preliminary concept design discussed in Section 2.0 and 3.0 will continue to be refined as supporting analysis is completed. Joints/seals and nozzle design and analysis will continue along with supporting analysis of the overall design. The Final Concept Design will be reviewed by the ARC Technical Review Committee on December 1, 1986, and the Preliminary CEI Specification will be delivered by December 12, 1986. The Conceptual Design Package will be completed and delivered before December 31, 1986.

5.2 TASK 2.0 D&V PLAN

The Preliminary D&V Plan will include component, subscale, simulator, and full-scale test description and quantities. D&V Program schedule and Logic Flow will be defined.

5.3 TASK 3.0 CAPABILITY ASSESSMENT

Assessment of existing and planned facilities and equipment will be completed and submitted by December 12, 1986. This task will include a review of handling, assembly, and transportation requirements and an assessment of environmental protection issues.

5.4 TASK 4.0 BUDGETARY COSTS

ARC will prepare budgetary costs for Development and Validation, Facilities, and Production Unit Costs within the Work Breakdown Structure defined in the contract.

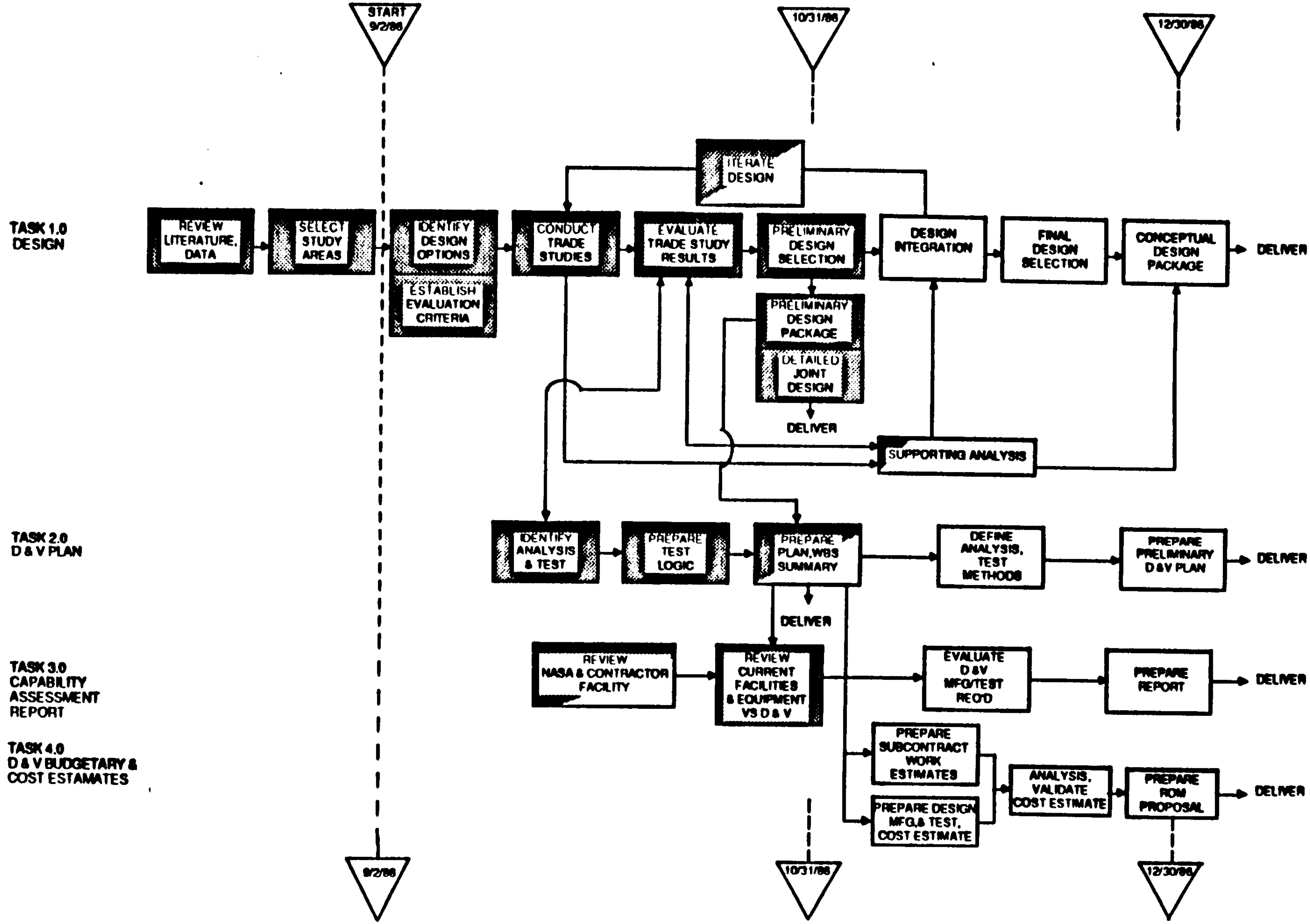


FIGURE 5.0.1. SRM BLOCK II STUDY FLOW DIAGRAM.

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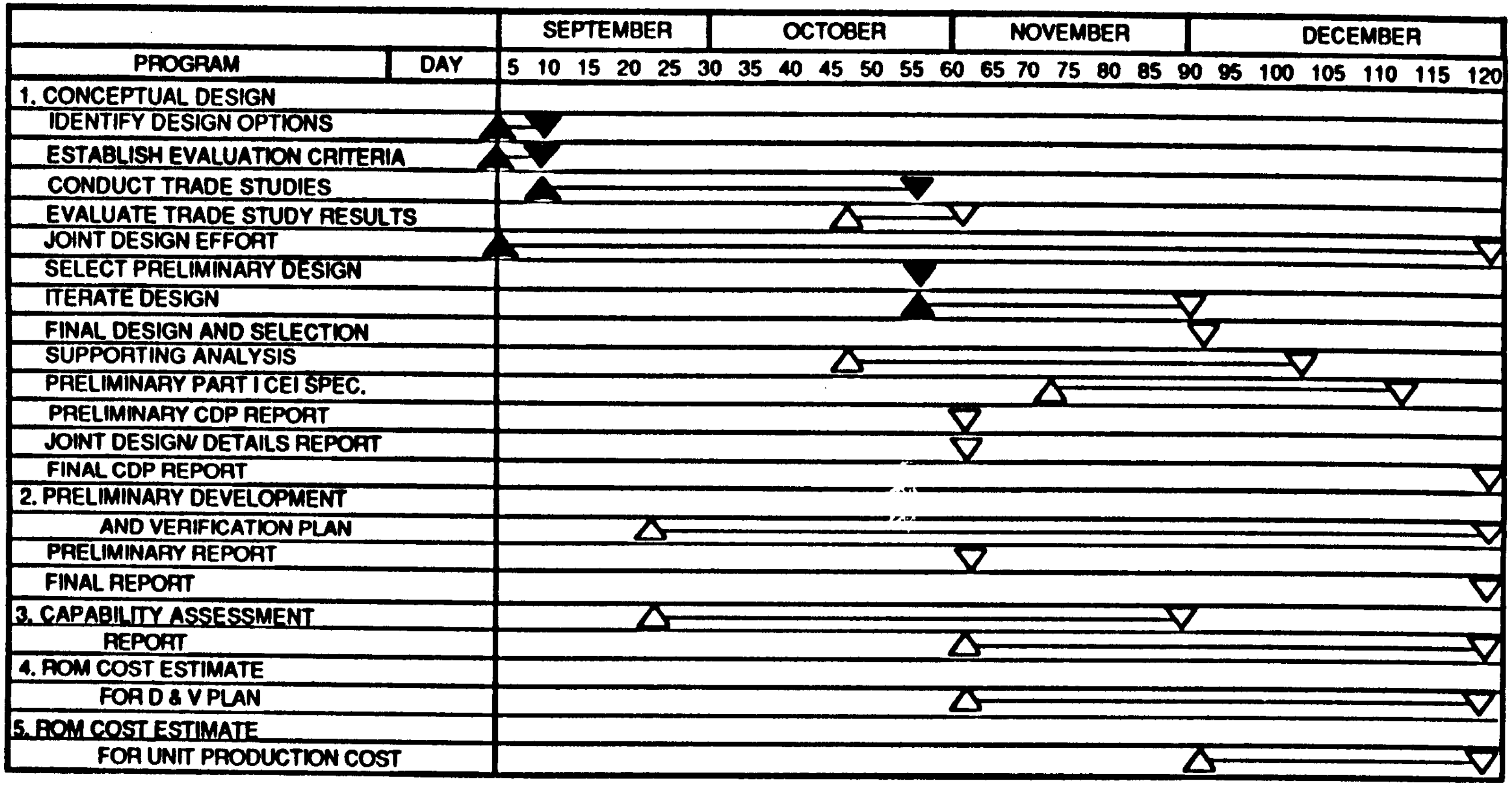


FIGURE 5.0.2. SRM BLOCK II MASTER MILESTONE SCHEDULE.

ADDENDUM TO

BLOCK II SOLID ROCKET MOTOR (SRM)
CONCEPTUAL DESIGN STUDY
CONTRACT NAS 8-37295

MID-TERM REPORT

NOZZLE LINER MATERIAL TRADE STUDY

Liners used on the current SRM nozzle are fabricated from rayon-based carbon cloth/phenolic tape. The performance of this liner material has been adequate with the exception of anomalous gouging and "pocketing erosion", which was first noted on STS-8. NASA and the current nozzle fabricator have spent considerable effort investigating the solutions to this problem and, as a result of stringent material/processing controls and revised processing/fabrication techniques, have virtually eliminated the incidence of pocketing erosion as demonstrated in the six flights preceding 51-L.

The goal of the program study in this area was to investigate the use of alternate state-of-the art liner materials to further reduce the possibility of pocketing erosion. A material trade study was, therefore, initiated and the preliminary results were given in Section 2.6 of the Mid-Term Report. This addendum presents the final results of the trade study, which differ somewhat from those previously reported.

The materials considered for this application were other tape-wrapped, cloth-reinforced phenolics and multidirectionally reinforced carbon-carbon advanced composites. In particular, the tape-wrapped materials were similar to the current material, except that the cloth fiber is based on either a PAN or pitch precursor. These materials are well known and have been evaluated in several alternate SRM nozzle material studies. The advantages and disadvantages of these materials were discussed in Section 2.3 of the Mid-Term Report. The carbon-carbon material is somewhat newer, however; thus, the following discussion of current carbon-carbon technology as it relates to the SRM nozzle liner has been included.

The propulsion industry is continually seeking nozzle materials to increase the performance of existing designs and to meet the demands of future vehicles. This has led to the development of multidirectionally reinforced (three-dimensional and four-dimensional) carbon-carbon composites, a material with good, uniform erosion resistance and excellent thermostructural properties. Use of this material and its derivatives as a single piece component in the nozzle throat and entrance has significantly elevated the reliability and performance of solid rocket nozzles. Three-dimensional cylindrical carbon-carbon integral throat and entrances (ITEs) are being used successfully on numerous solid propulsion vehicles. Employing this material from the 2-inch throat diameter of the Star 27 motors to the 15 inch throat diameter of the first stage Peacekeeper missile

TABLE 1. SOLID PROPULSION VEHICLES INCORPORATING CARBON-CARBON INTEGRAL THROAT AND ENTRANCES.

	VEHICLE	C-C SUPPLIER	TYPE
MX	STG 1	FMI, AVCO	3D
	STG 2	FMI, AVCO	3D
	STG 3	FMI, AVCO	3D
D5	STG 1	FMI, AVCO	3D
	STG 2	FMI, AVCO	3D
	STG 3	FMI	4D
IUS	SRM 1	FMI	3D
	SRM 2	FMI	3D
STAR	27	FMI	3D
	30	FMI	3D
	37XF	FMI	3D
	37FM	FMI	3D
	48	FMI	3D
	62	FMI	3D
	75	FMI	3D
SICBM	STG 1	TBD	3D
	STG 2	TBD	3D
	STG 3	TBD	3D
SRAM II		TBD	3D
TOMAHAWK EX 111		FMI, SEP	4D
MAGE 2 (FRENCH)		SEP	4D
ARIANE V		SEP	3D
IRIS (ITALIAN)		SEP, AEROSPATIALE	3D, 4D
SCOUT-ANTARES		SAI	4D

is established technology. Table 1 lists the solid propulsion systems presently incorporating three-dimensional carbon-carbon ITEs. Exceptional toughness, damage tolerance, and defect insensitivity inherent with three-dimensional carbon-carbon (unlike conventional two-dimensional laminates) have yielded ITEs with virtually 100-percent reliability.

The application of carbon-carbon ITEs has predominantly been ballistic missile and space propulsion systems. However, the upper stage of the NASA Scout launch vehicle incorporates a nozzle ITE of four-dimensional carbon-carbon, and the Ariane V solid rocket boosters will incorporate ITEs of three-dimensional carbon-carbon. The 31-inch throat diameter of the Ariane V booster will make it the largest carbon-carbon ITE employed to date.

The advent of automated preform manufacturing at both Fiber Materials, Inc. (FMI) and AVCO Speciality Materials (the leading three-dimensional carbon-carbon producers in the U.S.) offers further advantages of improved uniformity, producibility, and reproducibility. Autoweave™ facilities existing at AVCO are capable of fabricating three-dimensional pre-forms of up to 90 inches in diameter. Fifty four-inch diameter capability exists with FMI's Ultraloom™ facility. These woven pre-forms are typically densified with a coal tar pitch precursor to achieve the final graphite matrix. It is this stage of processing where scaleup issues must be addressed. Both FMI and AVCO subject billets to repetitive impregnation, high pressure carbonization (1,000 psi at AVCO and 5,000 to 15,000 psi at FMI), and 5,000°F graphitization. To accommodate nominal 54-inch throat diameter shuttle SRM ITEs, facility expansion or alternative processing must be realized. Also, the issue of a large R/T billet surviving the process pressure and temperatures warrants exploration.

Societe Europeenne de Propulsion (SEP) of France has taken a different approach to the fabrication of three-dimensional carbon-carbon ITEs for Ariane boosters. Utilizing conventional textile equipment and cost-effective chemical vapor infiltration (CVI) techniques for densification, SEP has achieved an ultrafine weave (an order of magnitude finer than conventional U.S. construction), three-dimensional carbon-carbon amenable to large, economically produced billets. This technology is presently being considered for transfer to the U.S. The largest U.S. graphite CVI capability (up to 93 inches in diameter) presently exists at B.F. Goodrich's SuperTemp facility. This novel three-dimensional carbon-carbon is presently receiving Air Force attention for potential application as nozzle components for the SICBM.

SEP was requested by ARC to investigate the usage of their advanced materials for use in the Block II SRM nozzle. In reply, SEP kindly submitted the design study below for use in this report.

SRM NOZZLE DESIGN WITH A CARBON-CARBON THROAT AND THERMOSTABLE INSULATORS

Submitted by: Societe Europeenne de Propulsion

The reliability of a nozzle is mainly a function of the materials used in the throat area. A significant improvement could have been obtained over the past ten years by the development of multidirectionally reinforced carbon-carbon, but the main weakness of these carbon-carbon throat nozzles is the phenolic insulators. Thermal expansion and outgassing of phenolics are responsible for most of the firing test problems; these two parameters are difficult to predict because they are a function of heating rate, ply-orientation, load state, process history, available vent path, and other complex boundary conditions. In order to replace phenolic insulators, SEP has developed new insulative materials using a thermostable matrix. These new materials are described in a paper presented by Paul Donguy in the 21st Joint Propulsion Conference (AIAA-85-1171). Different combinations of reinforcement and matrix are possible: ceramic-ceramic, ceramic-carbon, carbon-ceramic, and specially processed carbon-carbon. Thermostable insulators present the following advantages:

1. A low and predictible thermal expansion.
2. No outgassing.
3. Excellent structural properties allowing high temperature attachments, such as thread or pins.

Today, a significant number of nozzles of various sizes, using thermostable insulators, have been successfully tested at SEP.

Thermostable insulators are still more costly then phenolics, but the design can be often simplified, thus decreasing the nozzle assembly time; moreover, the nozzle behavior can be accurately modeled, eliminating failures in development.

Two versions of the SRM nozzle using thermostable insulators are presented.

For each version, a preliminary thermal analysis has been performed assuming safety factors of 2.8 on ablation and 1.0 on char (thermostable insulators do not char in firing).

The insulative thicknesses have been determined using the following design criteria:

Maximum temperature elevation of

metallic parts (+ 5°C after 130 sec)
(+230°C after 540 sec)

The first version (Figure 1) uses our current technology of thermostable insulator (heat diffusivity 2.5 10^{-6} m^2/s).

In this case, thermal thicknesses have been increased, compared to the current SRM design, and new metallic parts have been designed; however, the flex bearing remains the same.

The second version (Figure 2) uses a new material currently in development. This material already has been firing tested, but it has not yet been industrially manufactured.

In this case, the thermal diffusivity is 0.8 10^{-6} m^2/s and the same metallic parts can be used.

In each case, thermostable insulators are associated with a NOVOLTEX carbon-carbon throat, NOVOLTEX has been developed for exit cones; however, its use for ITE's has been already demonstrated; a NOVOLTEX ITEC has been recently demonstrated on a MINUTEMAN third stage under an RPL contact. This material is characterized by a fully automated preform construction, giving a three-dimensional fine spacing texture.

The densification is achieved by a chemical vapor deposition process; few cycles are needed to reach densities greater than 1.7 g/cc; the resulting matrix has good erosion and mechanical properties. A NOVOLTEX throat has been selected for the nozzle of the ARIANE five solid rocket booster (weight of propellant: 420,000 lbm). The manufacturing process has been already demonstrated at a three-fourths scale.

The carbon-carbon material and the alternative tape-wrapped materials were compared to the current rayon-based carbon cloth/phenolic using the trade study methodology described in Section 2.0 of the Mid-Term Report. The candidate materials were compared on the basis of relative reliability, performance, and cost. Weighting factors were assigned to each of these categories according to their relative importance as

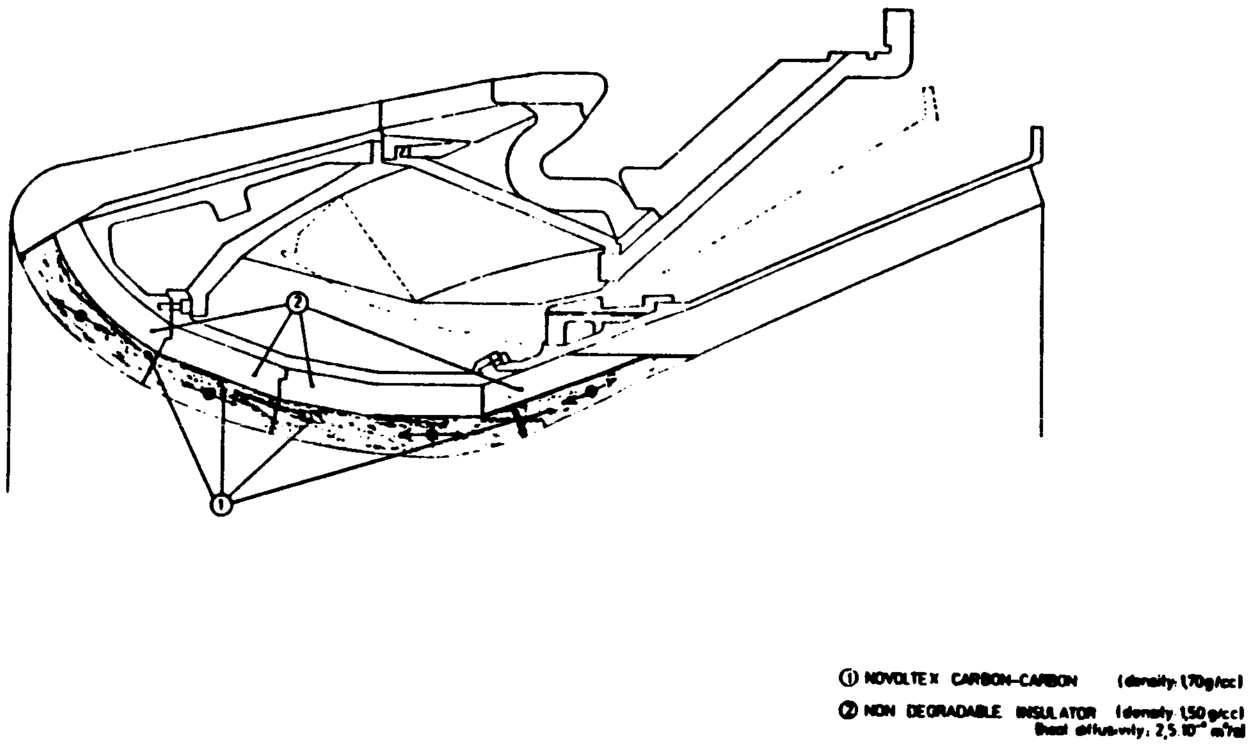


FIGURE 1. SEP DRAWING ABD 11070+AA.

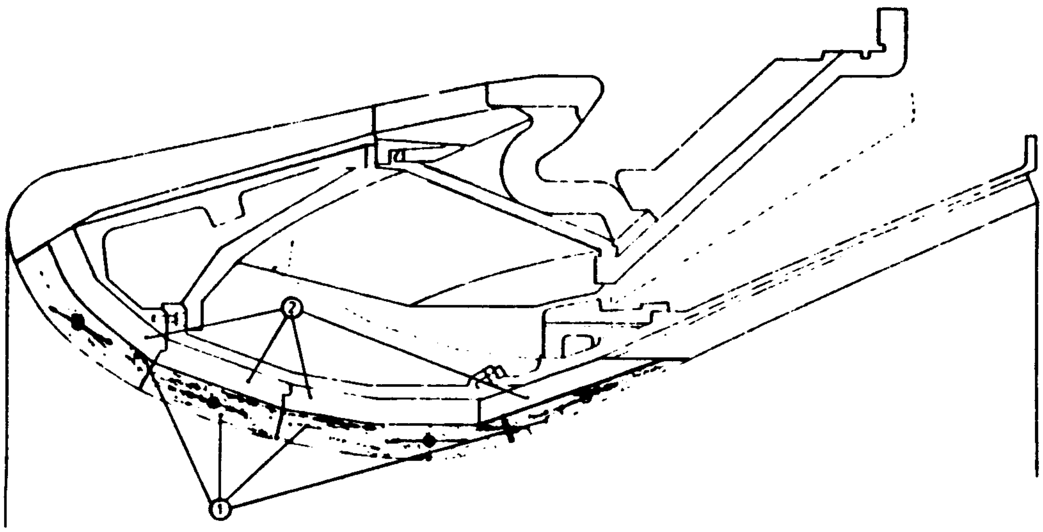


FIGURE 2. SEP DRAWING ABD 11071+AA.

follows: reliability, 0.65 (65 percent); performance, is 0.15 (15 percent); and cost, 0.20 (20 percent). The relative merit of each alternative was assessed, and a ranking score from 1 to 10 was assigned. Multiplying the raw rankings scores by the weighting factor in each category provided weighted scores that were summed to give a total for each material.

In most categories, the materials were ranked qualitatively. In evaluating the thermal margin category under reliability, the thermal analysis in Section 3.2.3.2 of the Mid-Term Report was used.

The results of the trade study are shown in Table 2. As the table indicates, the rayon-based and PAN-based materials received identical scores. Literature indicated that the PAN-based material had a somewhat reduced potential for pocketing erosion. However, this small advantage was offset by the less extensive database available for this material compared to rayon-based material, the increased developmental risk associated with a "new" material, and the slightly reduced thermal margin for the PAN-based material. The performance gain exhibited by the PAN material did not overcome the reduced reliability. With equal ranking, there is no reason to switch to the PAN material.

Results were similar for pitch-based material, except that it exhibited an even further reduced thermal margin to the point of impacting hardware design to meet the safety factor requirements.

The low score received by the carbon-carbon resulted primarily from the risks associated with replacing a well developed material in an existing design. The developmental risk and lack of an extensive data are the major drawbacks for reliability and, combined with the increased cost, more than offset the increased performance.

ARC's conclusion, then, is to retain the current nozzle liner material.

TABLE 2. NOZZLE LINER MATERIAL TRADE STUDY.

	WEIGHT FACTOR	RAYON BASED CLOTH/PHENOLIC RAW SCORE	RAYON BASED CLOTH/PHENOLIC WT SCORE	PAN BASED CLOTH/PHENOLIC RAW SCORE	PAN BASED CLOTH/PHENOLIC WT SCORE	PITCH BASED CLOTH/PHENOLIC RAW SCORE	PITCH BASED CLOTH/PHENOLIC WT SCORE	3-D CARBON-CARBON RAW SCORE	3-D CARBON-CARBON WT SCORE
RELIABILITY (65 PERCENT)									
POTENTIAL FOR POCKETING EROSION	0.20	7	1.40	8	1.60	8	1.60	9	1.80
THERMAL MARGINS	0.10	9	0.90	8	0.80	6	0.60	6	0.60
STRUCTURAL INTEGRITY	0.10	8	0.80	8	0.80	8	0.80	9	0.90
DEVELOPMENTAL RISK	0.10	9	0.90	8	0.80	8	0.80	6	0.60
EXISTING DATA BASE	0.15	9	1.35	8	1.20	8	1.20	6	0.90
SUBTOTAL			5.35		5.20		5.00		4.80
PERFORMANCE (15 PERCENT)									
INERT WT IMPACT	0.075	8	0.60	9	0.68	8	0.60	8	0.60
THROAT EROSION	0.075	7	0.53	8	0.60	8	0.60	9	0.68
SUBTOTAL			1.13		1.28		1.20		1.28
COST (20 PERCENT)									
FACILITIES IMPACT	0.05	10	0.50	10	0.50	10	0.50	5	0.25
MATERIAL COST	0.075	8	0.60	8	0.60	9	0.68	7	0.53
PROCESSING COST	0.075	8	0.60	8	0.60	8	0.60	6	0.45
SUBTOTAL			1.70		1.70		1.78		1.23
TOTALS			8.18		8.18		7.98		7.31

APPENDIX B

SEAL DESIGN/PERFORMANCE INFORMATION

FLUOROCARBON

Components Division
2620 The Boulevard, Columbia Industrial Park
P.O. Box 9889
Columbia, South Carolina 29290
803/783-1880

December 9, 1986

Mr. Nick Brown
Atlantic Research Corporation
7511 Wellington Road
Gainesville, VA 22065

Subject: Rocket Joint Seals

Dear Mr. Brown:

Thank you for contacting Fluorocarbon to aid in choosing a possible seal for your flange joint proposal. We contacted our Texas Division that was reviewing the Grafoil filled Inconel jacketed gasket and found they are declining to make a proposal at this time. See Houston's Mike Blake's recent letter to Mr. J.W. Chamlee for their comments.

We feel, however, that either metallic o-rings, c-rings or a combination should perform most satisfactorily and so we will make proposals using these materials.

Given the maximum temperatures and possible flange separation we feel, Inconel 718 o-rings would be the best choice. These would be silver plated and fully age hardened. Two rings 104 inches in diameter of .500 inch cross section would need a load of 1,650,000 pounds to seat the unrestrained rings. The load is somewhat higher in a groove because of contact with the groove but it should be well below the 12,000,000 pounds load expected. The groove should be .420 +/-.005 inches deep, at least .600 inches wide and have an 32 RMS finish. Retaining clips can be supplied for assembly ease if the o-rings are in the "upper" flange. One inch of space is needed on the ring I.D. for these clips (see page 11 of our o-ring catalog for clip possibilities).

FLUOROCARBON

The second possibility would be 2 o-rings of .375 inch cross section. These would take less room, narrower grooves, and about 80% of the load to compress the .500 cross section. Springback for the .375 cross section is .012 inches, well above the .0035 inches maximum expected flange separation.

The third possibility would be two (2) metallic c-rings. C-rings of this diameter should be at least .375 inches in cross section and silver plated as the metallic o-rings would be.

We feel the o-rings are the best choice here because they have a wider surface in contact with the flanges, and there is sufficient load to yield the plating and compress the rings. Please let us know your choice and we will supply detail drawings showing rings, grooves and retaining clip if you feel these are necessary.

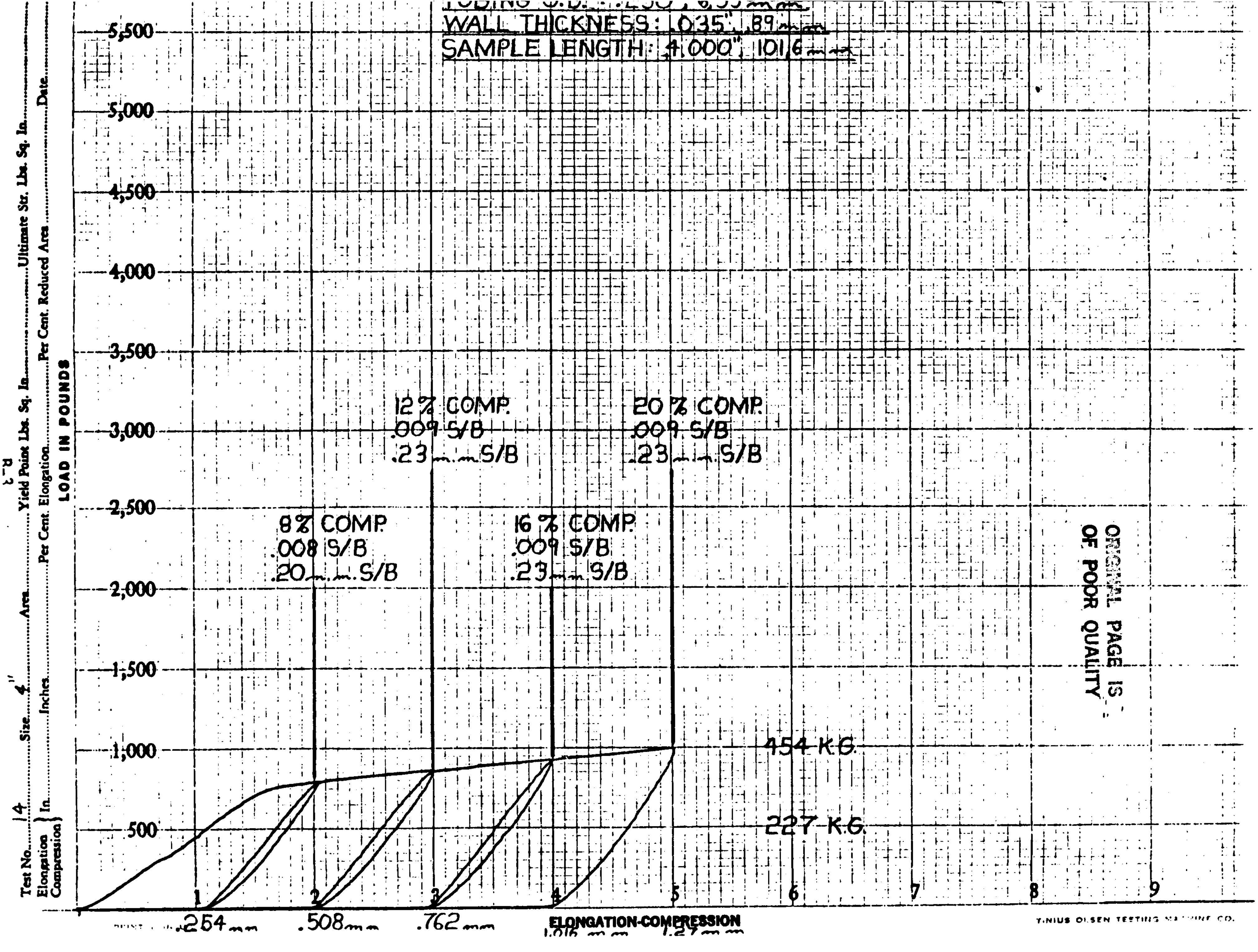
Thank you again for the chance to make a design proposal. Please advise quantities required, deliveries and of course type seal and we will supply pricing information.

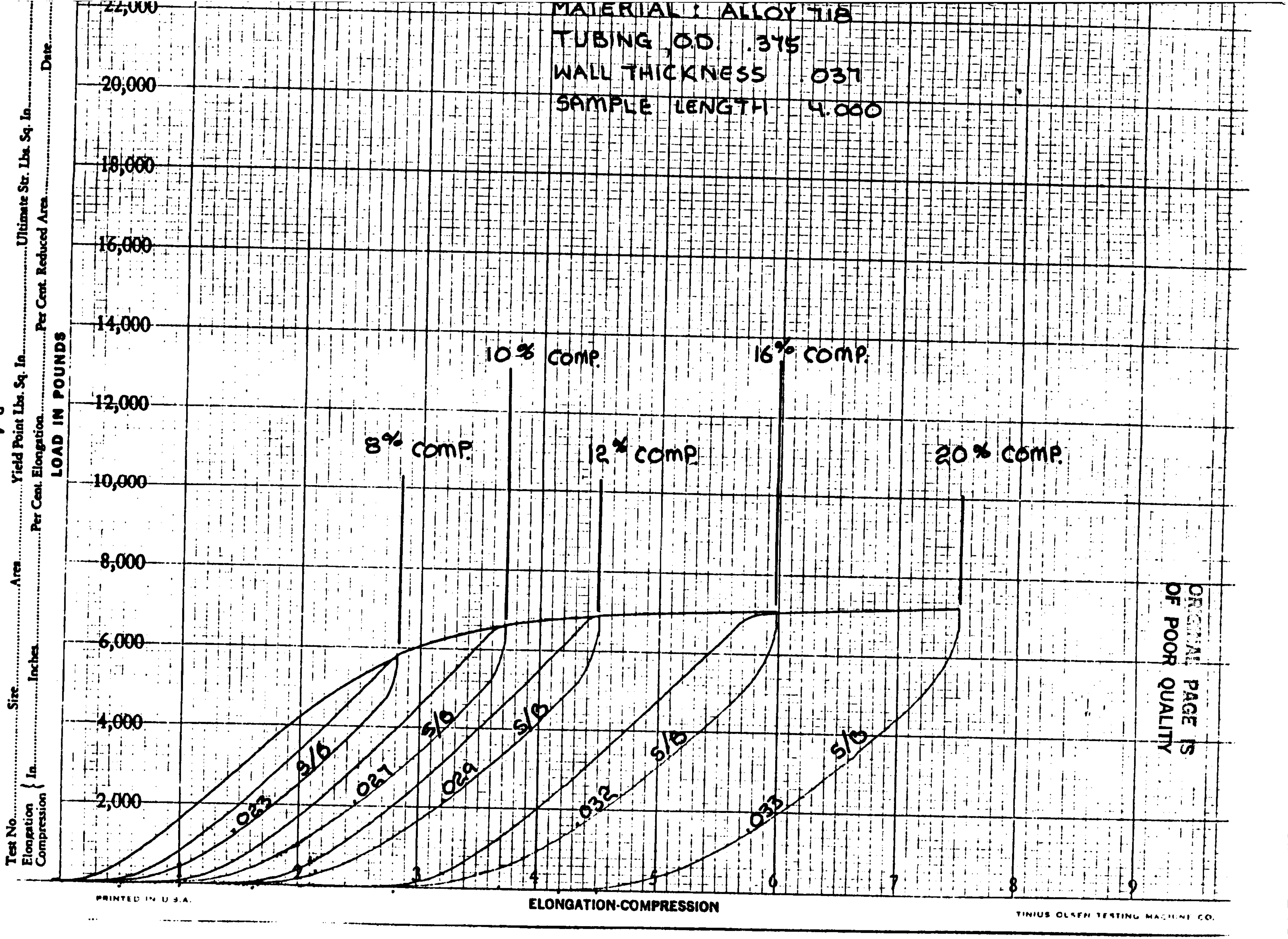
Best regards,

Kenneth Morales

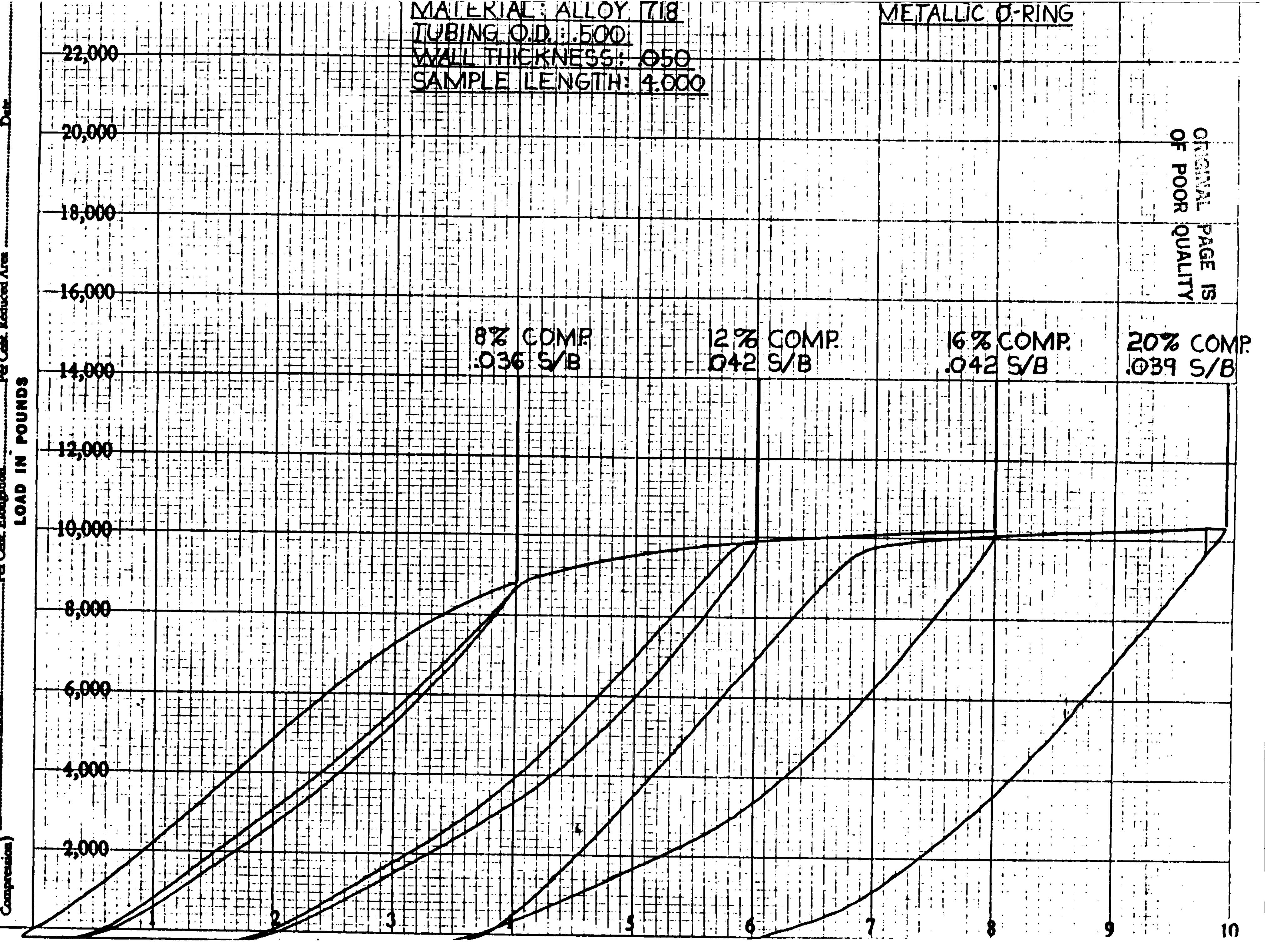
Kenneth Morales

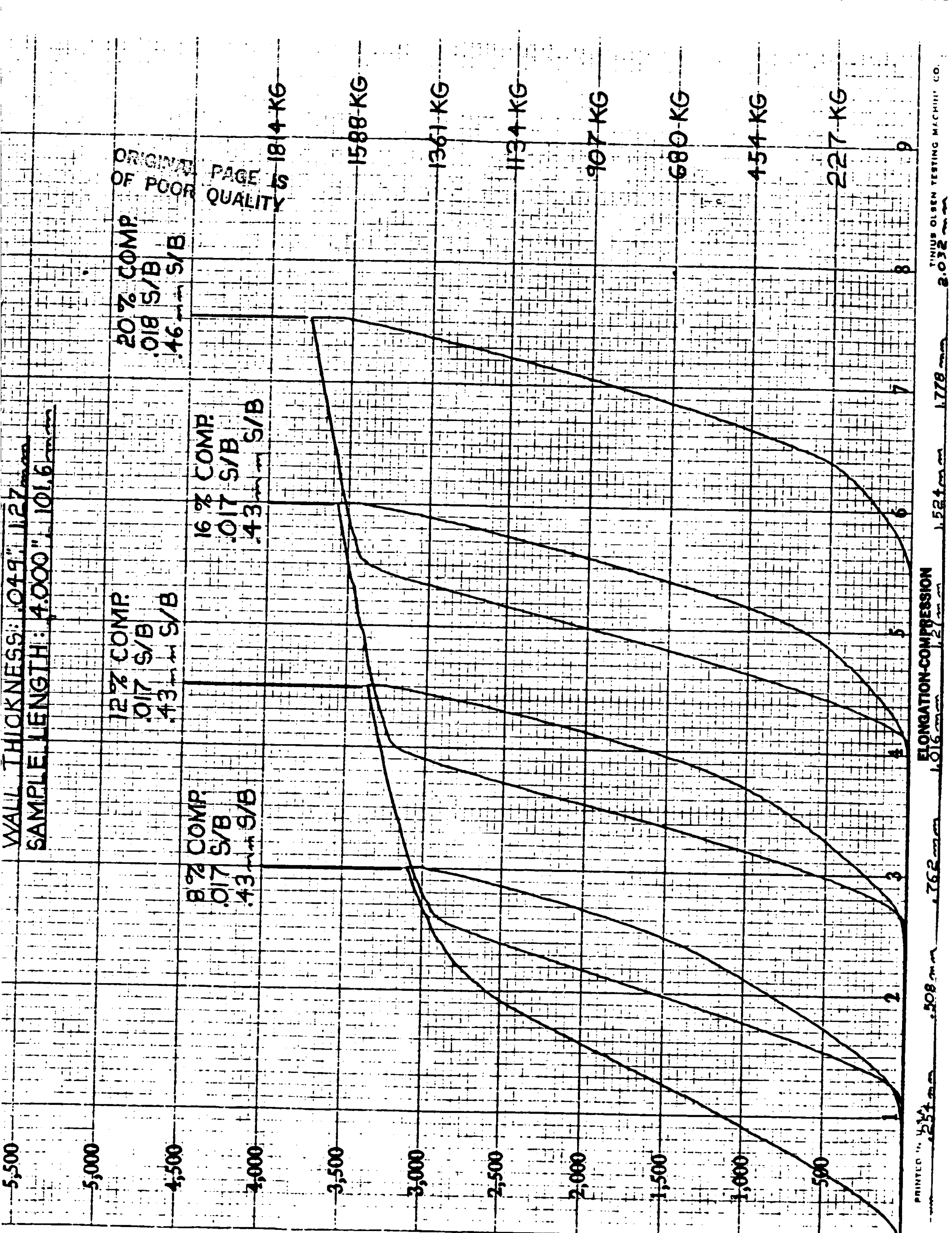
cc: J. McCrone
J. Swartz
B. Olafson


TUBING O.D.: .250", 6.35 mm
WALL THICKNESS: .035", .89 mm
SAMPLE LENGTH: 4.000", 101.6 mm
Test No. 14
Size 4"
Area
Yield Point Lbs. Sq. In.
Ultimate Str. Lbs. Sq. In.
Elongation / Compression In.
Inches
Per Cent. Elongation
Per Cent. Reduced Area
Date
LOAD IN POUNDS
5,500
5,000
4,500
4,000
3,500
3,000
2,500
2,000
1,500
1,000
500
8% COMP.
.008 S/B
.20 mm S/B
12% COMP.
.009 S/B
.23 mm S/B
16% COMP.
.009 S/B
.23 mm S/B
20% COMP.
.009 S/B
.23 mm S/B
454 K.G.
227 K.G.
1
2
3
4
5
6
7
8
9
.254 mm
.508 mm
.762 mm
1.016 mm
1.27 mm
ELONGATION-COMPRESSION
ORIGINAL PAGE IS OF POOR QUALITY
TINIUS OLSEN TESTING MACHINE CO.



MATERIAL: ALLOY 718
TUBING O.D. .375
WALL THICKNESS .037
SAMPLE LENGTH 4.000
ORIGINAL PAGE IS
OF POOR QUALITY
8% COMP.
10% COMP.
12% COMP.
16% COMP.
20% COMP.
.023 S/B
.027 S/B
.029 S/B
.032 S/B
.033 S/B
LOAD IN POUNDS
22,000
20,000
18,000
16,000
14,000
12,000
10,000
8,000
6,000
4,000
2,000
ELONGATION-COMPRESSION
1
2
3
4
5
6
7
8
9
Test No. Size Area Yield Point Lbs. Sq. In. Ultimate Str. Lbs. Sq. In.
Elongation Compression In. Inches Per Cent. Elongation Per Cent. Reduced Area Date
PRINTED IN U.S.A.
TINIUS OLSEN TESTING MACHINE CO.

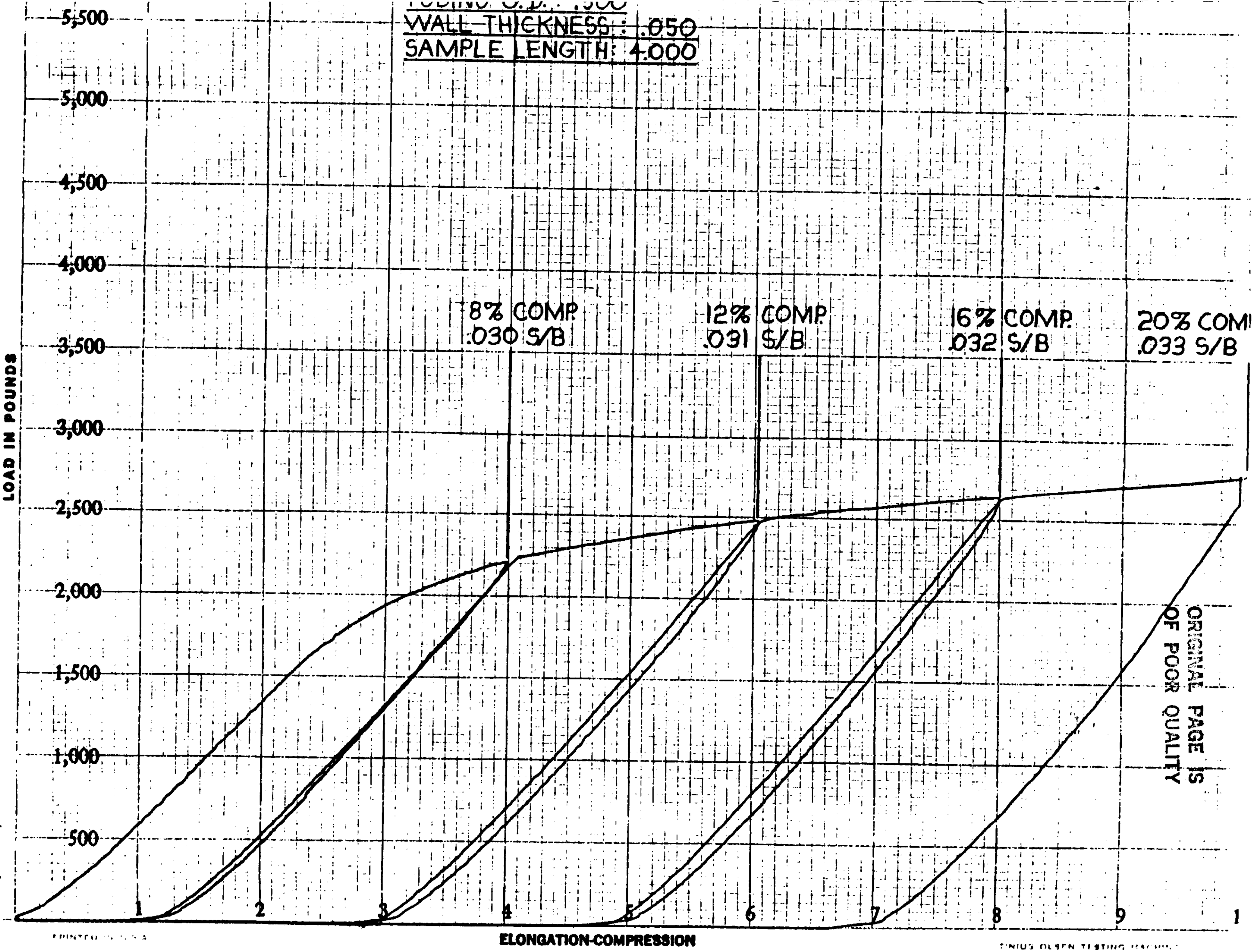

MATERIAL: ALLOY 718
TUBING O.D.: .500
WALL THICKNESS: .050
SAMPLE LENGTH: 4.000
METALLIC O-RING
8% COMP.
.036 S/B
12 % COMP.
.042 S/B
16 % COMP.
.042 S/B
20% COMP.
.039 S/B
22,000
20,000
18,000
16,000
14,000
12,000
10,000
8,000
6,000
4,000
2,000
LOAD IN POUNDS
1
2
3
4
5
6
7
8
9
10
Elongation
Compression
In
Inches
Per Cent Elongation
Per Cent Reduced Area
Date



WALL THICKNESS: .049" 1.27mm
SAMPLE LENGTH: 4.000" 101.6mm
ORIGINAL PAGE IS OF POOR QUALITY
20% COMP. .018 S/B .46mm S/B
16% COMP. .017 S/B .43mm S/B
12% COMP. .017 S/B .43mm S/B
8% COMP. .017 S/B .43mm S/B
1814 KG
1588 KG
1361 KG
1134 KG
907 KG
680 KG
454 KG
227 KG
5,500
5,000
4,500
4,000
3,500
3,000
2,500
2,000
1,500
1,000
500
LOAD IN POUNDS
ELONGATION-COMPRESSION
1 2 3 4 5 6 7 8 9
.254mm .508mm .762mm 1.016mm 1.27mm 1.524mm 1.778mm 2.032mm
TINIUS OLSEN TESTING MACHINE CO.
Test No. 17
Size 4.000
Area
Yield Point Lbs. Sq. In.
Ultimate Str. Lbs. Sq. In.
Elongation / Compression In. Inches
Per Cent. Elongation
Per Cent. Reduced Area
Date

Test No. 3 Size 4" Area Yield Point Lbs. Sq. In. Ultimate Str. Lbs. Sq. In.

Elongation / Compression } In. Inches Per Cent. Elongation Per Cent. Reduced Area Date



FLUOROCARBON

Components Division
2620 The Boulevard, Columbia Industrial Park
P.O. Box 9889
Columbia, South Carolina 29290
803/783-1880

November 21, 1986

Mr. J. W. Chamlee
Atlantic Research Corporation
Propulsion Division
7511 Wellington Road
Gainesville, VA 22065-1699

Re: Letter dated 11/13/86, Chamlee to McCrone

Dear Mr. Chamlee:

Thank you for your interest in our metallic seals. Fluorocarbon manufactures metallic seals for a wide variety of applications, including nuclear pressure vessels, aircraft engines, valves, pumps and other commercial applications.

Unfortunately, we do not have documented evidence that our seals have been used in rocket motors. Often, our seals are designed by our customers, and we may not know the end use for which the seal is intended. For this reason, it is possible that we may have supplied seals for rocket motors and not be aware of it.

As you are aware, we do supply large diameter seals to the nuclear power industry. These seals range in size from approximately 130 inches to 275 inches in diameter. Generally, Metal O-Rings are used to seal radioactive steam at approximately 600 degrees Fahrenheit and 1200 PSI. These seals are used to seal the reactor head to the vessel, and are used in pairs (one ring inside the other). Typically, the seals are made from Inconel 718 or Stainless Steel 304, and are silver plated for more leaktight sealing. This design is used in reactor vessels throughout the United States, Europe, and the Far East. The approximate cost of this type of seal is $5000.00 per seal.

The three sided groove design which you will be using is not a problem. Our only requirement is that your minimum groove width meet our catalog recommendations. For a 3/8" diameter cross section, we recommend a groove depth of 0.295/0.300", and a minimum width of 0.445". The Metal O-Ring would require approximately 2500 pounds per circumferential inch of load. A Metal C-Ring will require approximately one-half this amount. Because of the diameter of your requirement, we recommend that you consider using a larger diameter cross section, such as .455 or .500 inches. This will create a more stable ring and will improve the ease of manufacturing and handling.

Dedicated To Excellence

FLUOROCARBON

Our seals are TIG welded and the welds can be examined using Radiography or Liquid Penetrant examination. Immediately prior to installation, the rings should be examined visually for any signs of damage.

We have no experience with applications requiring a seal at 5500 degrees Fahrenheit.

Our seals have an unlimited shelf life when properly stored. The seals should be stored in a clean, dry environment. Lubrication of the seal cavity is not necessary during installation, but may be used if you desire.

As previously stated, the approximate cost of our Metal O-Rings in sizes similar to your application is $5000.00. A Metal C-Ring would be somewhat more expensive, although the exact price would depend on the seal cross-section, material, plating, tolerances, and other quality requirements. Normal delivery for these types of seals is 6-8 months after receipt of the order.

I hope this information will give you a better understanding of our products and will help you in determining your final seal design. If you have any questions, or need additional information, please don't hesitate to contact me at (803) 799-3606.

Very truly yours,

J Powell

Jim Powell
Sales Manager

cc: J. Swartz
J. McCrone

FLUOROCARBON

Metallic Gasket Division
P.O. Box 15639
Houston, Texas 77220

713/458-5830 Fax 713/458-0502 Telex 762-791
US 1-800-972-7638 Texas 1-800-833-0176

November 20, 1986

Mr. J. W. Chamlee
Program Manager
Space Shuttle SRM
Atlantic Research Corporation
Propulsion Division
7511 Wellington Road
Gainesville, Virginia 22065-1699

Dear Mr. Chamlee:

After carefully reviewing your correspondence of 13 Novmeber 1986 we are declining the opportunity to participate in the redesign of the seal for the SRM joint in question.

We manufacture to our customer's design and American Petroleum Institute Standards but do not have the resources available to respond to your requests.

Yours truly,

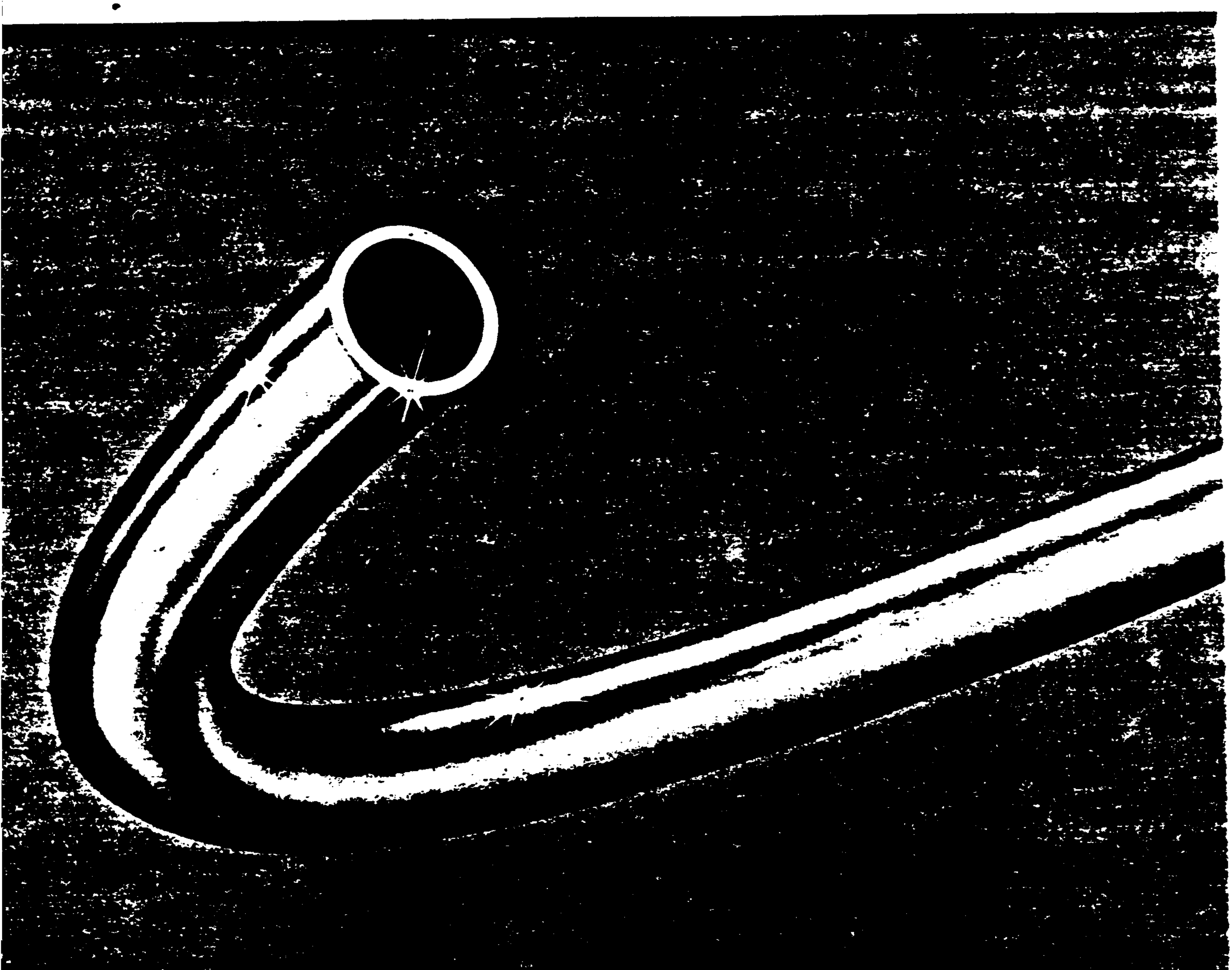
Mike Blake

Mike Blake
Sales and Marketing Manager
Fluorocarbon Metallic Gasket Division

MB/vac

FLUOROCARBON

METALLIC O-RINGS



Static, metal-to-metal seals
for confining gases or liquids
under adverse conditions of
pressure/temperature/ambience

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Fluorocarbon Metallic O-Rings

Fluorocarbon Metallic O-Rings are designed to prevent leakage of gases or liquids under adverse sealing conditions. These static, metal-to-metal seals can withstand pressures from high vacuum to 100,000 psi (6,804 atm). They can endure continuous temperatures from -425°F. up to 1,800°F. (-269°C. to 982°C.), or intermittent temperatures up to 3,000°F. (1,650°C.). They resist radiation, chlorides, corrosives, and other hostile environments. They will not deteriorate with age, either in use or in storage.

Design, Materials, Coatings, Sizes

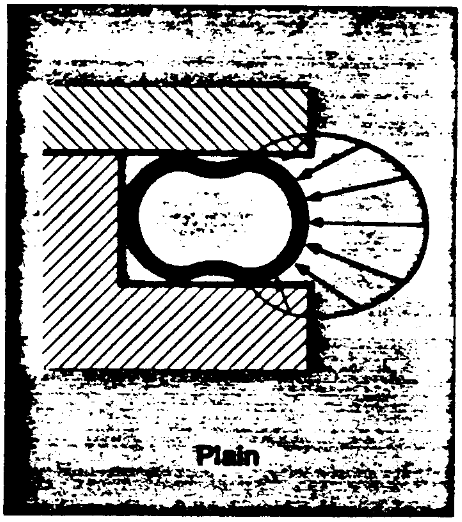
Fluorocarbon Metallic O-Rings, designated MOR, are made of metal tubing (or solid rod) which is formed into circular or other shapes and the two ends welded together. The O-Ring metal is stainless steel or other alloys. The O-Ring can be electroplated with silver, copper, indium, nickel, gold, lead or other metals, or it can be coated with Teflon. The flow of the finish material improves the sealing, especially under high pressure and/or vacuum. Since tensile strength and resilience of the seal are determined in part by metal temper, Fluorobarbon Components offers a choice of heat treating to material specification or tempering to customer specifications. Tubular or solid wire rings can be manufactured in sizes ranging up to 25 feet (7,6 m) or more in diameter, or as small as .250 inches (6,4 mm) OD.

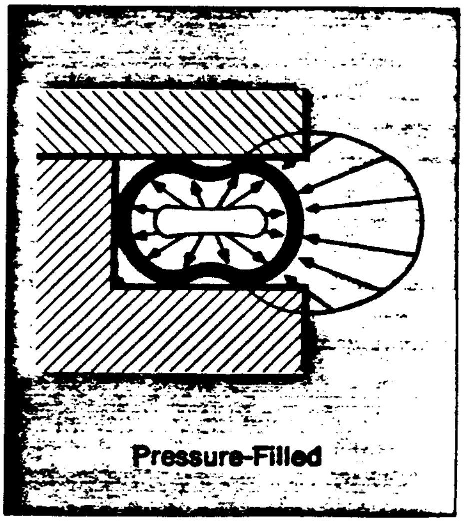
Application Characteristics

The typical application places a Metallic O-Ring in axial compression between parallel faces which are square to the fluid passage or vessel axis. The seal is usually located in an open or closed groove in one face. It can also be located in a retainer, which eliminates the need for machining a groove (see description of retainers on page 8).

Upon compression to a predetermined fixed height, the seal tubing buckles slightly, resulting in two contact areas on the seal face and maximum contact stress between the seal and the mating faces. When the flange faces are closed, the O-Ring is under compression and tends to spring back against the flanges, thus exerting a positive sealing force. If the O-Ring is the self-energizing type, the pressure of the gas or liquid on the vented side energizes the seal and further increases the sealing force by pushing the seal against the flange face.

Types of Metallic O-Rings





Plain

(Not Self-Energizing or Pressure-Filled)
Made of metal tubing (or solid rod) in most metals. This type is the most economical O-Ring. It is designed for low to moderate pressure and vacuum conditions.

Self-Energizing

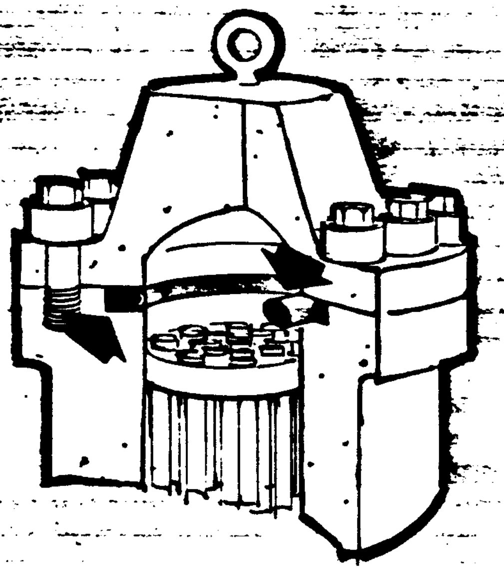
The inner periphery of the O-Ring is vented by small holes or a slot. The pressure inside the ring becomes the same as in the system. Increasing the internal pressure increases sealing effectiveness.

Pressure-Filled

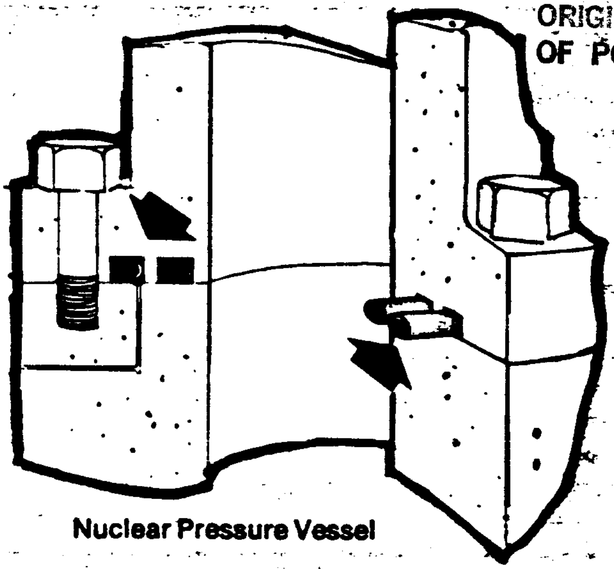
Pressure-filled O-Rings are designed for a temperature range of 800° F. to 2,000° F. (425° C. to 1093° C.). They cannot tolerate pressures as high as the self-energizing type. The ring is filled with an inert gas at about 600 psi (41 atm). At elevated temperatures, gas pressure increases, offsetting loss of strength in tubing and increasing sealing stress.

Typical Applications

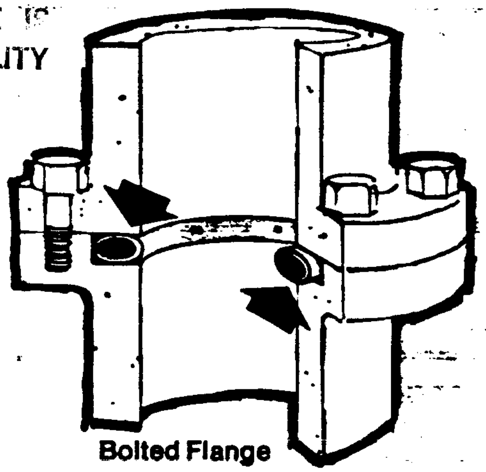
Metallic O-Rings have been used successfully in vacuum and high pressure systems, and in critical systems for hydraulic and lubricating oil, jet engine fuel, gasoline, rocket fuels, steam, liquid metals and combustion gas. They also provide positive, leak-proof seals in piping systems for chemical, petrochemical, oil and gas, and refining industries. Many reciprocating engines, gas turbines, compressors, heat exchangers, pressure vessels, injection molding machines, high pressure filters and other components rely on Metallic O-Rings for permanent, metal-to-metal seals. Several common applications are shown in the following illustrations.



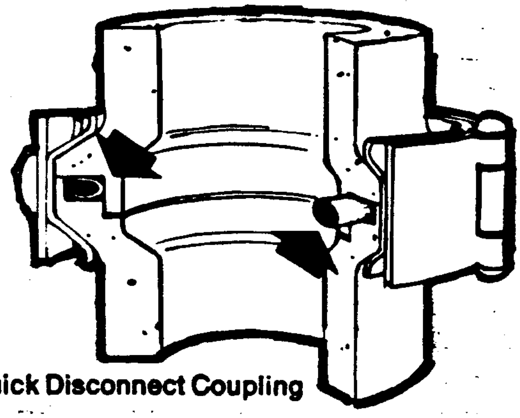
Heat Exchanger/Pressure Vessels



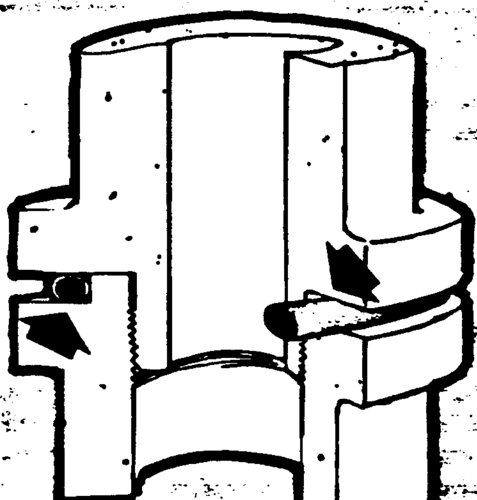
Nuclear Pressure Vessel



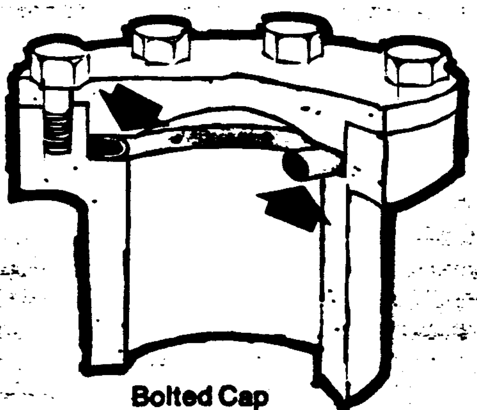
Bolted Flange



Quick Disconnect Coupling



External Pressure (Thread Joint)



Bolted Cap

Metallic O-Ring Selection Guide

To select the proper Metallic O-Ring for a particular application, it is necessary to determine system pressure, temperature, and kind of fluid to be sealed.

1. O-Ring Type

Pressure determines if O-Ring should be self-energizing.

Pressure	O-Ring Type
Vacuum to 100 psi (6.81 atm)	Self-energizing not required
100 psi (6.81 atm and above)	Self-energizing desirable

2. O-Ring Material

Temperature determines basic O-Ring material.

Temperature	O-Ring Material
Cryogenics to 500° F. (260° C.)	321 Stainless steel
to 800° F. (427° C.)	Alloy 600
to 1800° F. (982° C.)	Alloy X-750
above 1800° F. (982° C.)	Consult Factory

3. O-Ring Size

Tubing diameter is determined by ring OD, compression force desired, and available space. See complete data for O-Ring size selection on pages 6 and 7.

4. Seal Load vs. Seal Ring Diameter

Curves on page 7 show the seal load vs. seal ring diameter to various tubing outer diameters and wall thickness for stainless steel tubing. For tubing made of Alloy 600, multiply loads shown by 1.1. For Alloy X-750, multiply by 1.4.

5. O-Ring Wall Thickness

The wall thickness should be selected to provide the proper yield under compression. The data on pages 6 and 7 include the practical wall thickness dimensions that may be used for each tube diameter. If plating is used, wall thickness for seals made with .125 inch (3,2mm) tubing and smaller should cause yielding of the plating at a load of 400 lb/in (7,14 kg/mm). For tubing over .125 inch (3,2mm) diameter, 800 lb/in (14,28 kg/mm) should be required. Teflon coatings on rings will yield at 100 lb/in (1,78 kg/mm).

6. Groove Dimensions

The proper dimensions and surface finish of the groove are as important in achieving a seal as the O-Ring itself. As a general guide in the preparation of joint surfaces, the recommended groove dimensions for internal and external pressure applications are shown on page 5.

Should you need further guidance and our recommendations, submit the following information regarding your application: 1. Temperature and pressure ranges, 2. Space available, 3. Material, 4. Medium to be sealed, 5. Available compression load, 6. Sketch of proposed application.

7. Coating or Plating

Coating or plating of the O-Ring will provide adherence and ductility (softness) to conform to microscopic groove or flange irregularities.

For unplated seals, liquid leakage can be estimated by the following expression:

$$Q=\frac{5.0 \times 10^{-6} P}{\mu}$$

(Q=leakage cc/sec; P=pressure difference psi; and μ= liquid viscosity at operating conditions, centipoise.) If the resulting calculated leakage is 10^{-3} to 10^{-4} or less, actual leakage may be zero because of surface tension. If leakage occurs, it should be proportional to seal diameter, and in the above expression, multiplied by D/2. D=seal diameter. Actual leakage will probably be less than predicted.

For coated or plated seals, helium-leaktight joints may be made with proper O-Ring and coating or plating selections. Test results range from 10^{-6} to 10^{-9} cc/sec, and lower at one atmosphere differential. Recommended coating or plating materials are:

Temperature	Plating or Coating
Cryogenic to 500° F. (260° C.)	Teflon
to 1800° F. (982° C.)	Silver
to 2200° F. (1186° C.)	Nickel

See page 12 for other coatings and plating.

8. Sealing Surface Finish

The groove and mating flange face must have a surface finish of 16 μin. rms (0,4 μmm) for bare rings, and 32-100 μin. rms (0,8 μ-2,54 μ mm) for plated or coated rings.

For gas, vacuum and light liquid (water), a finish of 16 μin. (0,4 μmm) rms is recommended. For medium liquids (hydraulic oils) and heavy liquids (tar or polymers) a finish of 32 μ in. (0,8 μ mm) rms is recommended. Machining tool marks on groove or flange face must be concentric.

Seal surfaces should be free of dirt, grit or other foreign materials.

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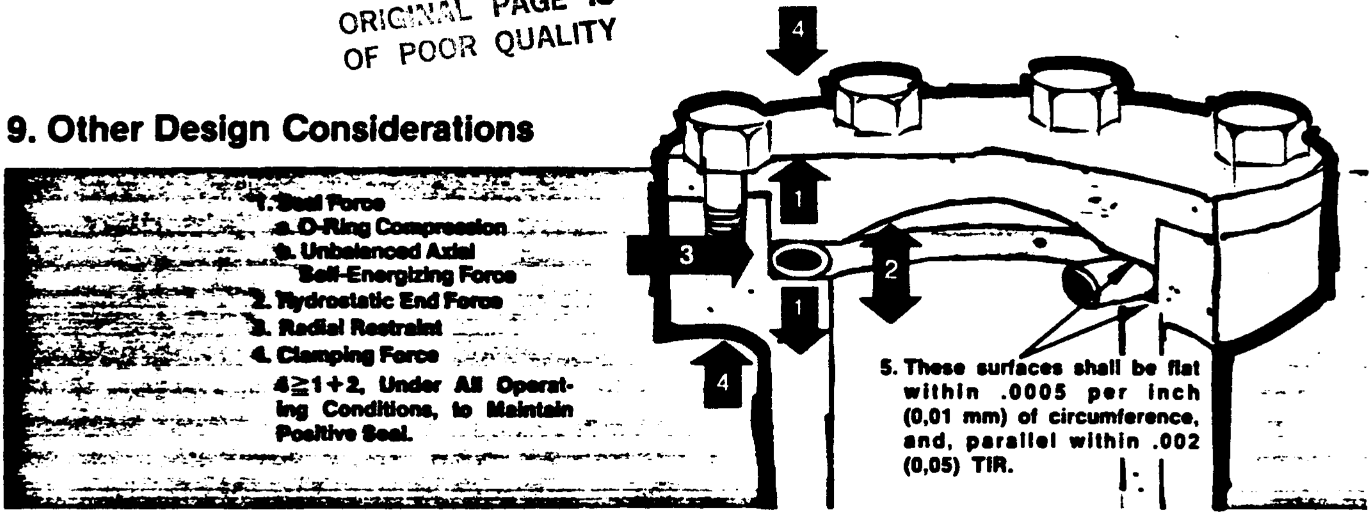
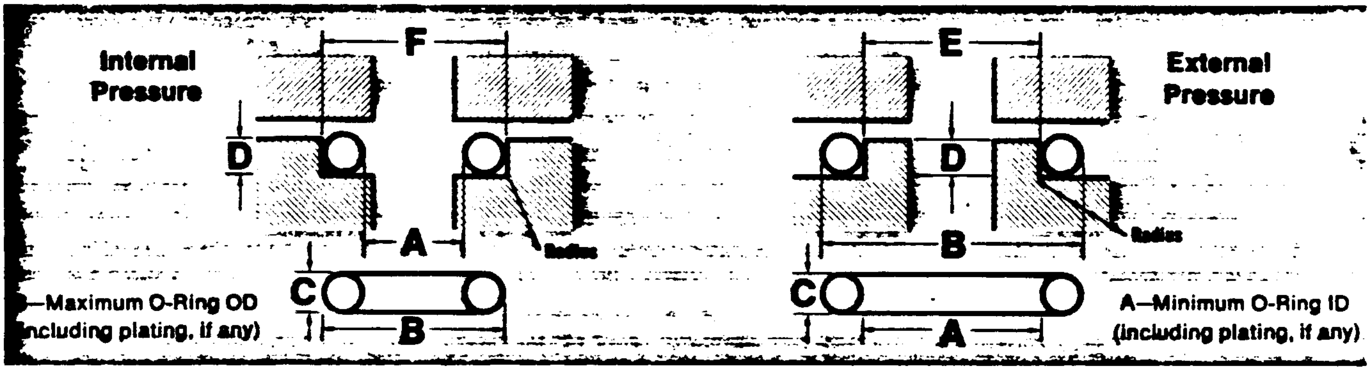
9. Other Design Considerations



Recommended Groove Dimension



Internal Pressure				External Pressure		
C Tube Diameter Inches/mm	F Groove OD Inches/mm	D Groove Depth Inches/mm	Ring Tolerance A - +.000 B + -.000 Inches/mm	E Groove ID Inches/mm	Minimum Groove Width Inches/mm	Springback * Inches/mm
.031	B+.004/.006	.020/.022	0.003	A—.004/.006	.042	.002
0,8	B+0,10/0,15	0,50/0,56	0,076	A—0,10/0,15	1,07	0,05
.063	B+.004/.006	.042/.045	0.003	A—.004/.006	.085	.002
1,6	B+0,10/0,15	1,07/1,14	0,076	A—0,10/0,15	2,16	0,05
.093	B+.005/.009	.065/.069	0.004	A—.005/.009	.112	.002
2,4	B+0,13/0.23	1,65/1,75	0,102	A—0,13/0,23	2,80	0,05
.125	B+.007/.012	.090/.095	0.005	A—.007/.012	.144	.003
3,2	B+0,18/0,30	2,29/2,41	0,127	A—0,18/0,30	3,66	0,08
.156	B+.008/.014	.115/.120	0.006	A—.008/.014	.182	.004
4,0	B+0,20/0,36	2,92/3,05	0,152	A—0,30/0,36	4,46	0,10
.188	B+.009/.015	.145/.150	0.007	A—.009/.015	.220	.004
4,8	B+0,23/0,38	3,68/3,8	0,178	A—0,23/0,38	5,59	0,10
.250	B+.011/.019	.195/.200	0.008	A—.011/.019	.290	.005
6,4	B+0,28/0,48	4,95/5,08	0,203	A—0,28/0,48	7,37	0,13
.375	B+.014/.029	.295/.300	0.012	A—.014/.029	.445	.009
9,5	B+0,36/0,74	7,49/7,62	0,305	A—0,36/0,74	11,3	0,23
.500	B+.020/.038	.415/.425	0.016	A—.020/.038	.645	.013
12,7	B+0,51/0,97	10,54/10,8	0,406	A—0,51/0,97	16,7	0,33
.625	B+.020/.038	.520/.530	0.016	A—.020/.038	.780	.017
15,9	B+0,51/0,97	13,21/13,46	0,406	A—0,51/0,97	19,8	0,43

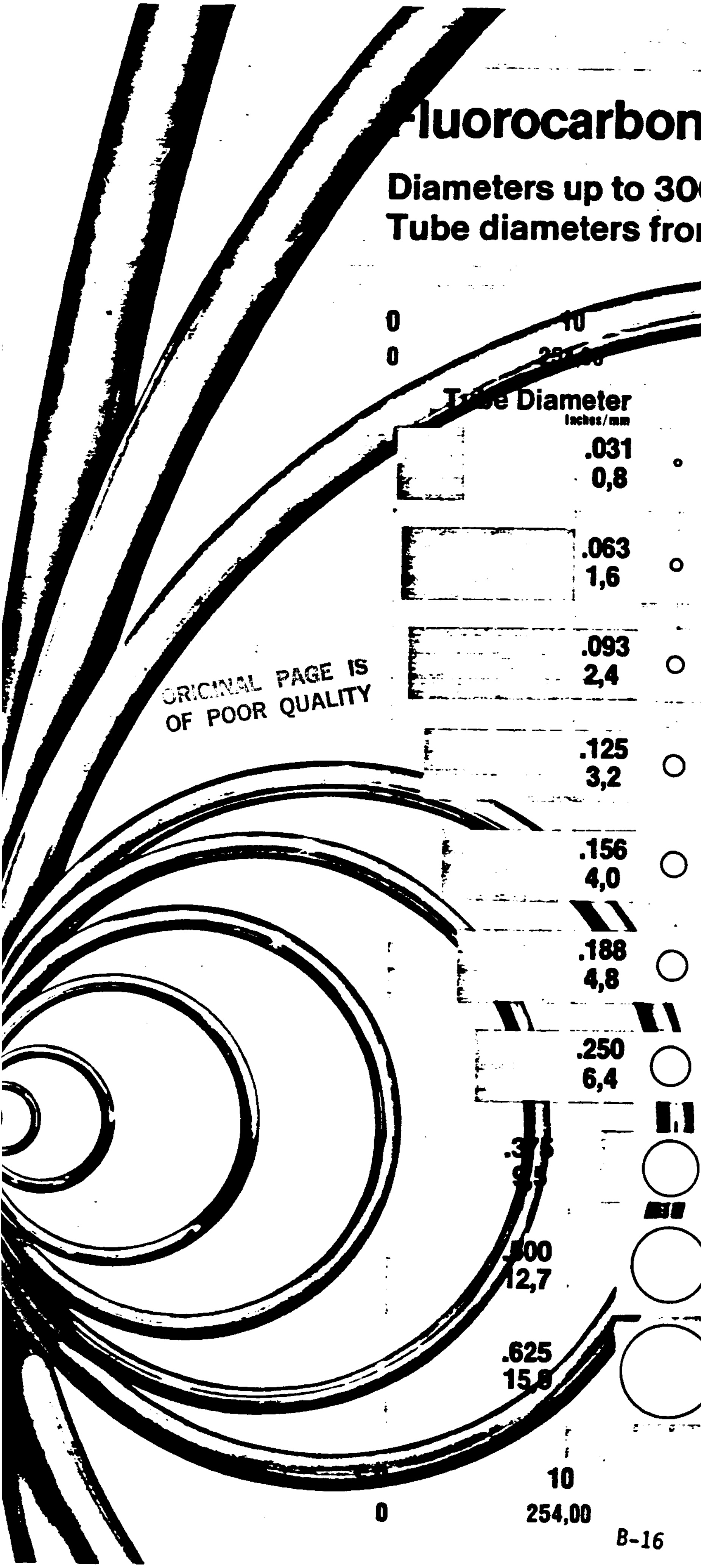
Dimensions in table above are for unplated rings. Increase groove depth for .031 inch (0,8mm) cross section rings by 2 times the plating or coating thickness of plated or coated rings.

Do not increase groove depth on plated or coated rings for cross section of .063 inch (1,6mm) and larger.

*Springback figures for tube diameters up to .250 inch (6,4 mm) are for stainless steel. Springback for .375, .500 and .625 inch (9,5, 12,7 and 15,9 mm) tube diameters are for precipitation hardened Alloy 718. Other values for different materials are available.

Fluorocarbon Metallic O-Rings

Diameters up to 300 inches (7620 mm) *
Tube diameters from .031 to .625 inches (0,8 to 15,9 mm) *



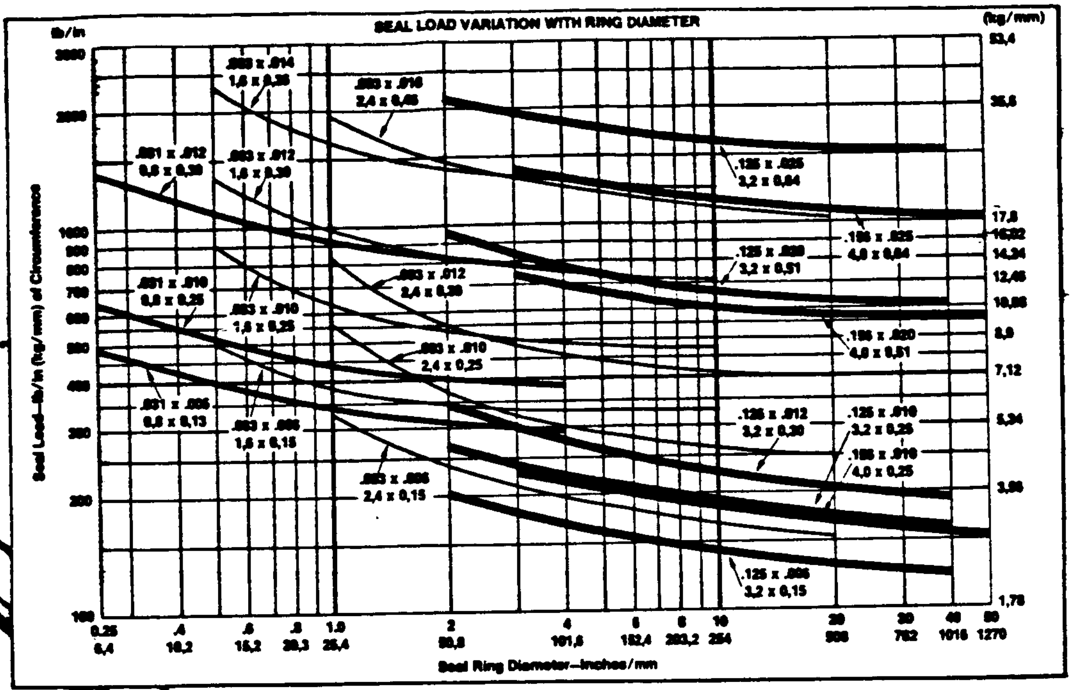
0 10 20 30 40
0 254,00 508,00 762,00 1016,0

Tube Diameter Inches/mm	Wall Thickness Inches/mm				
.031 0,8	.005 0,13	.010 0,25	.012 0,30		
.063 1,6	.006 0,15	.010 0,25	.012 0,30	.014 0,36	
.093 2,4	.006 0,15	.010 0,25	.012 0,30	.018 0,46	
.125 3,2	.006 0,15	.010 0,25	.012 0,30	.020 0,51	.025 0,64
.156 4,0	.010 0,25	.020 0,51	.025 0,64		
.188 4,8	.012 0,30	.020 0,51	.032 0,81		
.250 6,4	.012 0,30	.025 0,64	.032 0,81	.049 1,24	
.375 9,5	.035 0,89	.049 1,24			
.500 12,7	.050 1,27	.065 1,65			
.625 15,9	.063 1,60				

10 20 30 40
0 254,00 508,00 762,00 1016,00

Table I

50
1270,00



Tables I and II are for 321 stainless steel tubing.

For tubing made of Alloy 600, multiply loads shown by 1.1. For Alloy X-750, multiply by 1.4. For Alloy 718, multiply by 3

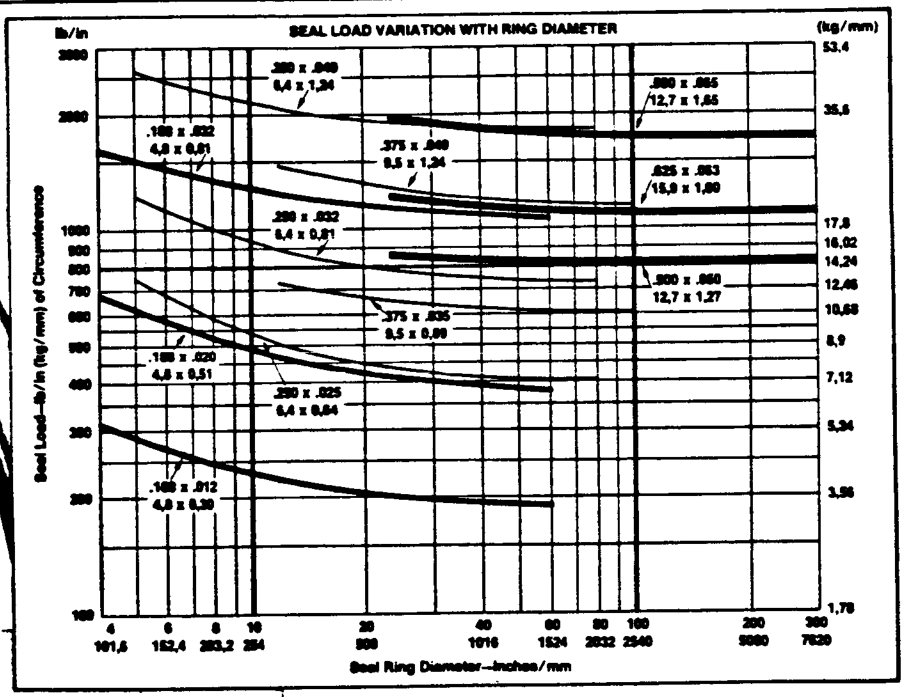


Table II

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1270,00	1524,00	1778,00	‾032,00	2286,00	2540,00	2794

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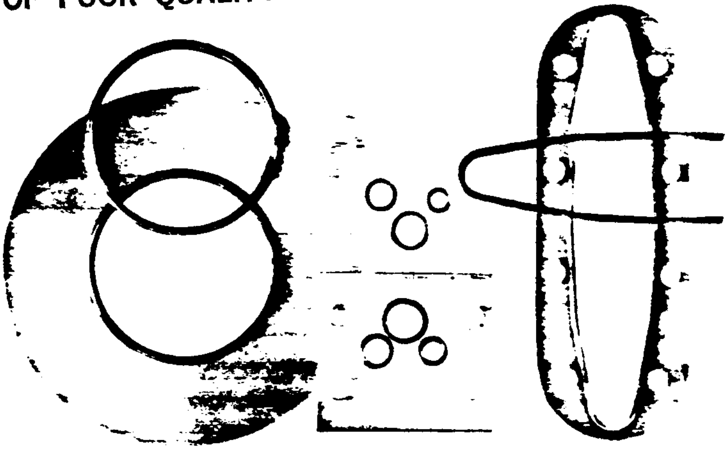
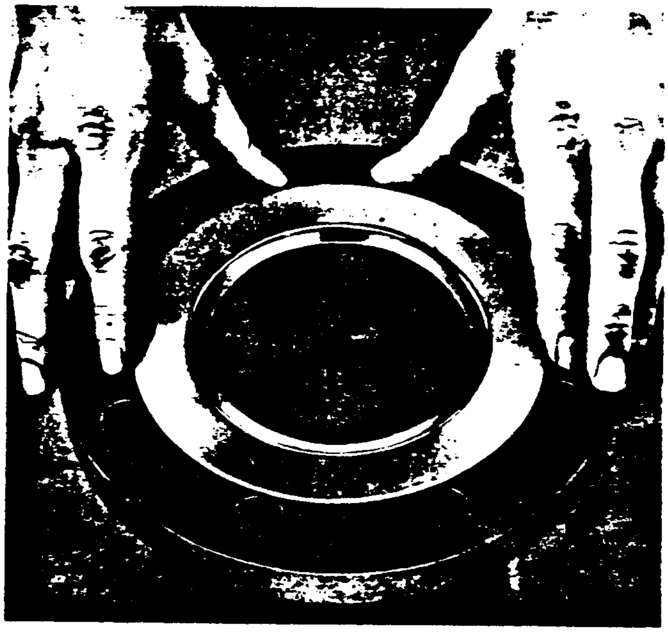
Retainer Assemblies



Metallic O-Rings can be used with a metal retainer plate for mechanical back-up that serves the same function as the machined groove wall in conventional installations. Retainer assemblies may incorporate several Metallic O-Rings into one all metallic assembly. The O-Rings are press-fitted without cross-section distortion, are secured against dropout and are easily handled during field assignment or retrofit programs. The retainer plate furnishes the O-Ring compression limit, controls hoop tension of the O-Ring, simplifies surface finish operation, permits interchangeability of flanges, and applies to single or multiple O-Ring requirements. A selection of several standard assemblies is described below:

ASA/API Pipe Flange Seals



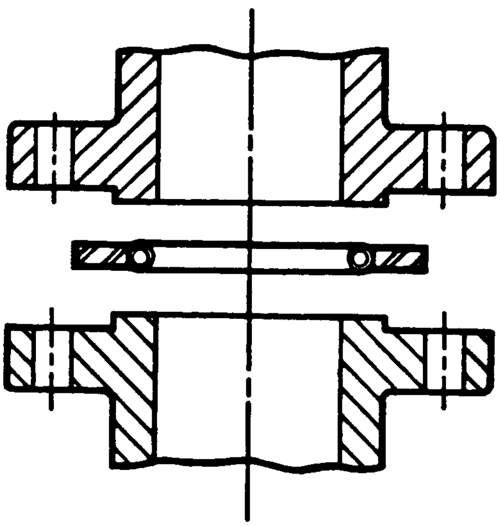
Metallic O-Rings offer static seal reliability and safety for installation or maintenance of piping. Over long periods of time, the all-metal construction of Fluorocarbon tubular Metallic O-Rings and retainer plates make them less susceptible to relaxation of sealing stresses—as compared to partially non-metallic gaskets.

In addition to their natural resilience characteristics, Metallic O-Rings provide the stability of a metal-to-metal pipe joint seal.

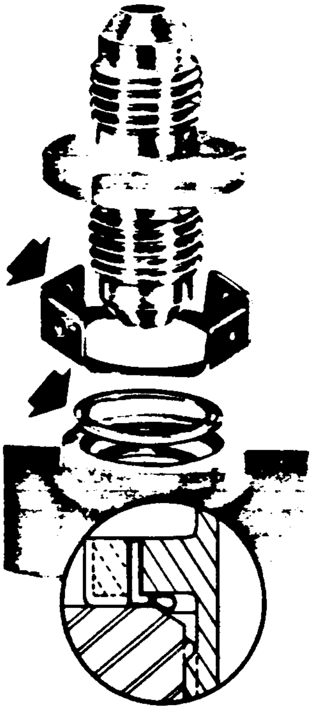
The natural springback of thin-wall metal tubing, and unique self-energizing design feature, create a balance of inside and outside forces which prevent collapse of the tube under pressure cycling. These same features allow Metallic O-Rings to respond to variations in sealing surface deflections without creep or cold flow, and to accommodate high and low temperature cycling. For process plant piping, they withstand temperatures from cryogenic to 1,800° F. (982° C.) and pressures from vacuum to 50,000 psi (3402 atm).

To maintain seal reliability, tubular Metallic O-Rings require less bolt stress than solid, fiber, flat metal, spiral wound or jacketed gaskets. Lower seal loads allow a greater bolt and flange safety factor for a given installation.

O-Rings and retainer plates are available for .250" to 24" (6,4 to 609,6 mm) pipe in all sizes of 150 to 2500 psi (10.2 to 170.1 atm) flat or raised face flanges.



Boss Seals

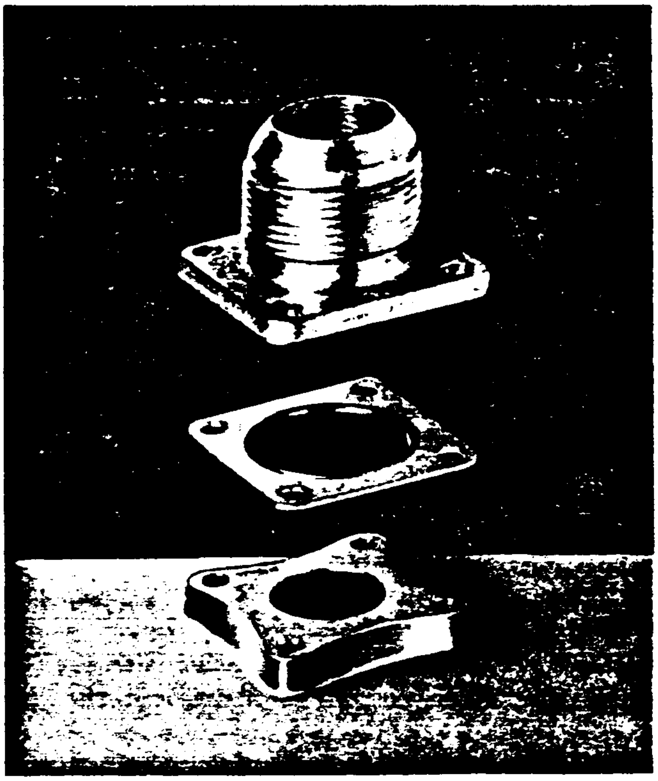


Fluorocarbon FIT-O-SEAL for boss joints combines a stainless steel retainer and a press fit Metallic O-Ring. The unit is self-positioning, controls ring compression, and can be reused. It won't deteriorate with age and is not affected by environment. Existing boss can be easily retrofitted. It can seal fuels and chemicals from high vacuum to 10,000 psi (680 atm) or higher, and will endure continuous temperatures of -452° F. (-269° C.) to 1,800° F. (982° C.). Standard seal assembly available for MS33656 fitting to MS33649 boss. Modifications available.

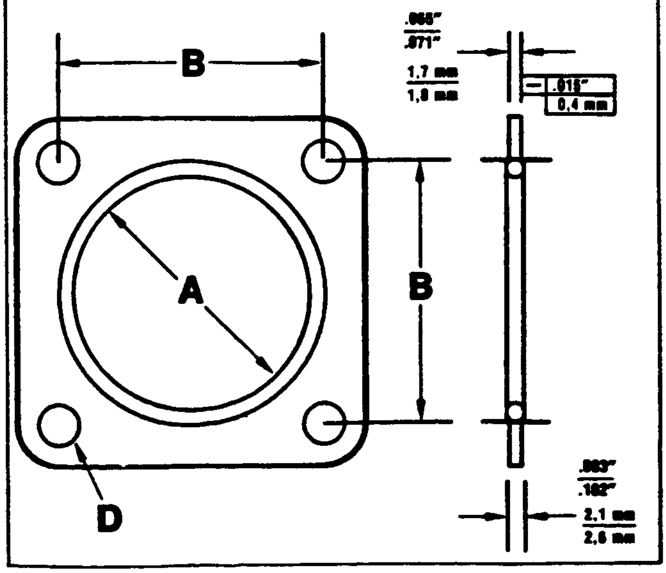
Flange-O-Seal

The Metallic O-Ring is semi-fastened into the metal retainer. The assembly is used for sealing jet engine fuel lines and exotic missile fuel lines from −452° F. (−269° C.) to 1,800° F. (982° C.).

It can be used for steel fittings MS20757 thru MS20762 and MS33786 fitting installation. The following assemblies are available from stock:



Part No. U-700490	Part No. U-700520	A Dia. Inches / mm ± .035 (0,89)	B Inches / mm ± .005 (0,13)	D Inches / mm ± .005 (0,13)
—12	—12	.863 21,92	1.156 29,36	.210 5,33
—16	—16	1.113 28,27	1.312 33,32	.210 5,33
—17	—17	1.113 28,27	1.414 35,92	.271 6,88
—20	—20	1.425 36,2	1.656 42,06	.271 6,88
—24	—24	1.613 40,97	1.812 46,02	.271 6,88
—32	—32	2.300 58,42	2.375 60,33	.333 8,46



Nuclear Pressure Vessel Seals

The principal applications of Fluorocarbon O-Rings in nuclear power plants is the sealing of reactor pressure vessel heads. They are also specified for sealing applications on valves, steam generators, condensers, pumps, piping and other equipment components throughout the nuclear flow chart.

Fluorocarbon O-Rings can easily meet the three major requirements of nuclear applications: temperature ratings, high pressure ratings, and larger than average ring diameters (see Page 2 for specifics). Fluorocarbon Metallic O-Rings offer other significant advantages in nuclear applications: they are not normally affected by damaging environments or corrosives; they don't deteriorate with age, even in storage, and they resist radiation and chlorides.

TABLE 1	O-Ring—Alloy 718—DEFLECTION and SPRINGBACK—Inches (mm)					
Load Force Unrestrained /linear inch ♦	.375 dia. x .038 wall (9,5 x 0,95)		.500 dia. x .050 wall (12,7 x 1,27)		.625 dia. x .063 wall (15,9 x 1,60)	
	2500 lb/in (45 kg/mm)		2500 lb/in (45 kg/mm)		4000 lb/in (71,5 kg/mm)	
Percentage	Deflection	Min. Springback	Deflection	Min. Springback	Deflection	Min. Springback
8%	.030 (0,76)	.009 (0,23)	.040 (1,02)	.013 (0,33)	.050 (1,27)	.017 (0,43)
10%	.037 (0,94)	.009 (0,23)	.050 (1,27)	.013 (0,33)	.062 (1,57)	.017 (0,43)
12%	.045 (1,14)	.009 (0,23)	.060 (1,52)	.013 (0,33)	.075 (1,91)	.017 (0,43)
16%*	**.060 (1,52)**	**.009 (0,23)**	**.080 (2,03)**	**.013 (0,33)**	**.100 (2,54)**	**.017 (0,43)**
17%	.064 (1,63)	.009 (0,23)	.085 (2,16)	.013 (0,33)	.106 (2,69)	.017 (0,43)

**Optimum compression percentage. 8 to 17% compression may be utilized with UAP Inconel 718. Load forces may vary slightly below 17% inch compression.*

Media to be Sealed

Media in the nuclear power plant which Fluorocarbon O-Rings can successfully seal include: ordinary (light) water, heavy water, boiling water, steam, borated water, carbon dioxide, helium, nitrogen, liquid metals including sodium, terphenyl and other phenyl fluids, and acids including boric acid.

Flange and Groove Details

Fluorocarbon O-Rings do not require expensive groove preparation and, being flexible, are easily installed. On pressure vessel head seals, a machined groove is required, the groove diameter being determined by the location of vessel rings so that minimum lift-off exists.

The O-Ring OD must be sufficiently large so that upon compression, the ring will expand and contact the groove outer wall. This limits hoop tension of the ring and provides a backup that restricts radial outward movement of the ring when the vessel is pressurized. Groove should be sufficiently wide so that the O-Ring ID does not contact the inside wall when the ring is compressed. Groove depth controls the amount of compression and the amount of load required to seat the ring. Table 1 shows the amount of flange load required to seat the seal.

The O-Ring and groove dimensions for internal and external pressure applications may be determined from the data on page 5.

Materials and Plating

Alloy 718 is the O-Ring material of choice on most nuclear sealing applications. Inconel 706 is also available. Alloy 718 used in Fluorocarbon O-Rings is annealed and age hardened, offers optimum strength and springback, and resists chlorides, radiation and corrosion. Type 304 stainless steel O-Rings are also offered for applications that are less critical and where a less expensive material will suffice.

Both Alloy 718 and Type 304 stainless steel O-Rings are available with silver plating of .004"—.006" (0,10 mm—0,15 mm) thickness. Ring OD can be controlled to .010" (0,25 mm) total tolerance after silver plating. The silver plating assures good adherence and ductility (softness) to conform to groove irregularities. Nickel plating is recommended when sealing sodium.

O-Ring Fabrication

Fluorocarbon Metallic O-Rings are fabricated by bending straight metal tubing into circular or other desired shapes. The two ends are welded together and the weld ground flush.

Where the proposed size of the fabricated O-Ring would prohibit shipping, the company offers on-site welding fabrication that meets the same quality standards as fabrication performed in our plant.

Tube and Ring Dimensions

The three most common tube diameters used for nuclear applications are shown below with the recommended relationship of tube diameter and wall thickness to the O-Ring diameter. Other tube diameters are also available for nuclear applications. See pages 6 and 7.

TABLE 2

Tube Diameter Inches/mm	Wall Thickness Inches/mm	O-Ring Diameter Inches/mm
.375 9.5	.038	Up to 180
.500 12.7	.050	120 to 260
.625 15.9	.063	220 and up

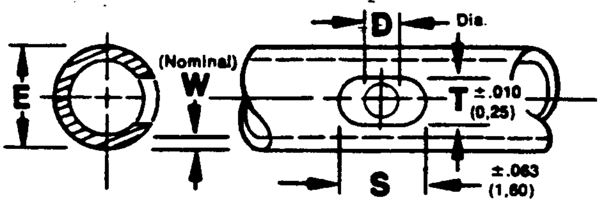
TABLE 3

O-RING DIAMETER inches (mm)	No. Slots or Holes
Up to 144 (3657.6)	8
144 (3657.6) and up.	12

**unless otherwise specified*

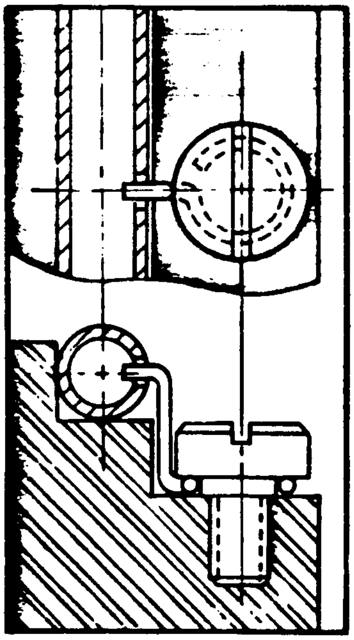
TABLE 4

SLOT or HOLE DIMENSIONS inches (mm)

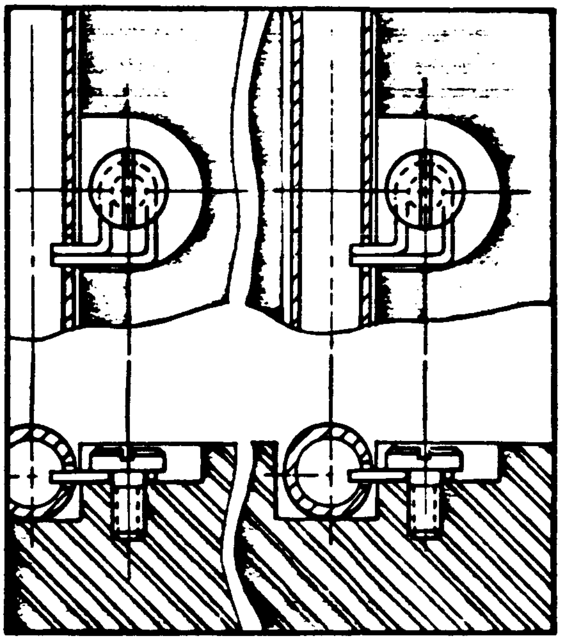


E	.375 (9,5)	.500 (12,7)	.625 (15,9)
W	.038 (1,0)	.050 (1,3)	.063 (1,6)
S	.281 (7,1)	.375 (9,5)	.438 (11,1)
T	.125 (3,2)	.205 (5,2)	.256 (6,5)
D	.070 (1,8)	.093 (2,4)	.125 (3,2)

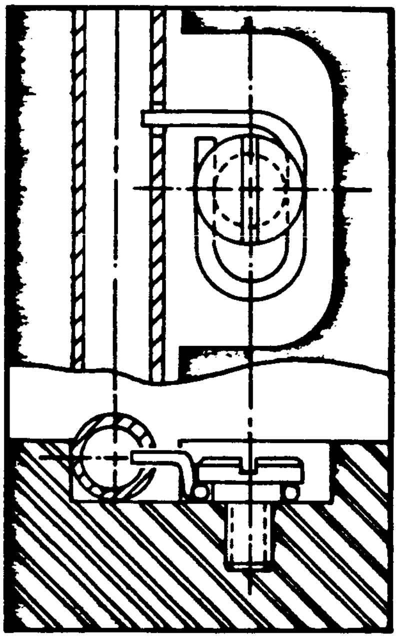
STYLE A



STYLE B



STYLE C



Retainer Clips

On nuclear pressure vessel heads, the rings are installed to the underside of the flange on the head. This requires clips to hold the rings in proper place and alignment during assembly of the head to the vessel. Slots are provided in the O-Ring to receive the retainer clips. In some instances the retainer clips are welded to the O-Ring. Instead of slots for retainer clips, drilled holes with additional self-energizing holes can be provided. The number of slots or holes and their size varies in relation to the ring and tube diameters (see Tables 3 and 4). The data shown assures installation without excessive O-Ring buckling in the groove and without endangering O-Ring strength. Different clipping methods are available, depending on vessel design, for both single and double ring applications (see drawings above—styles A, B and C).

How to Specify O-Rings

U 2 3 12 — 03 625 SE A

- **U** — Denotes Metallic O-Ring
- **2** — Materials
- **3** — Tubing OD (Thirty-Seconds)
- **12** — Wall Thickness (Thousandths)
- **03 625** — Metallic O-Ring OD (Inches) (Thousandths)
- **SE** — Type
- **A** — Coatings

Materials

1—Alloy 718	7—Stainless Steel 304
2—Stainless Steel 321	8—Stainless Steel 316
3—Aluminum	9—Stainless Steel 347
4—Copper	X—As Specified
5—Alloy 600	
6—Alloy X-750	

Type

SE—Self-energized on ID
PF—Pressure filled
NP—Not self-energized, not pressure filled
SO—Self-energized on OD
SX—Self-energized as spec.

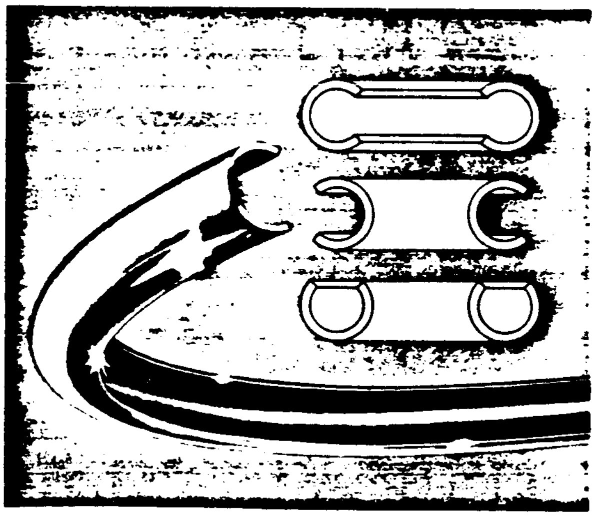
Example:

U2312-03625SEA

The above example, U2312-03625SEA, indicates a type 321 stainless steel O-Ring, 3/32" (2,38 mm) tube size, .012 (0,30 mm) wall thickness, 3.625" (92,08 mm) OD, self-energized (ID) and .001-.002" (0,03/0,05 mm) silver coating.

Coatings

A—Silver	.001/.002 (0,03/0,05)	N—None	
B—Silver	.002/.003 (0,05/0,08)	P—Lead	.001/.002 (0,03/0,05)
D—Teflon®	.001/.003 (0,03/0,08)	R—Indium	.001/.002 (0,03/0,05)
E—Teflon	.003/.004 (0,08/0,10)	T—Nickel	.001/.002 (0,03/0,05)
L—Copper	.001/.002 (0,03/0,05)	V—Gold	.0005/.001 (0,02/0,03)
X—As Specified			



Fluorocarbon Metallic C-Rings

Fluorocarbon Metallic C-Rings (designated MCR) are designed for static sealing on machinery or equipment and are available for internal pressure, external pressure, or axial pressure ID/OD applications. Because C-Rings are designed with an open side on the pressure side of the installation, the seal is self-energizing. Fluorocarbon C-Rings are offered in round or irregular shapes in a broad range of sizes from .126" (3,2 mm) OD x .032" (0,81 mm) free height to over 300" (7620 mm) OD x 2" (50,80 mm) free height. They are available in a wide variety of metal alloys and metallic or Teflon coatings. Sealing application temperature range is from cryogenic to 3,000°F. (1650°C.); pressure tolerances are from 10^{-10} torr to 100,000 psi (6,804 atm). Where customer requirements are large, the C-Ring provides the lowest unit price of any high performance seal on the market.

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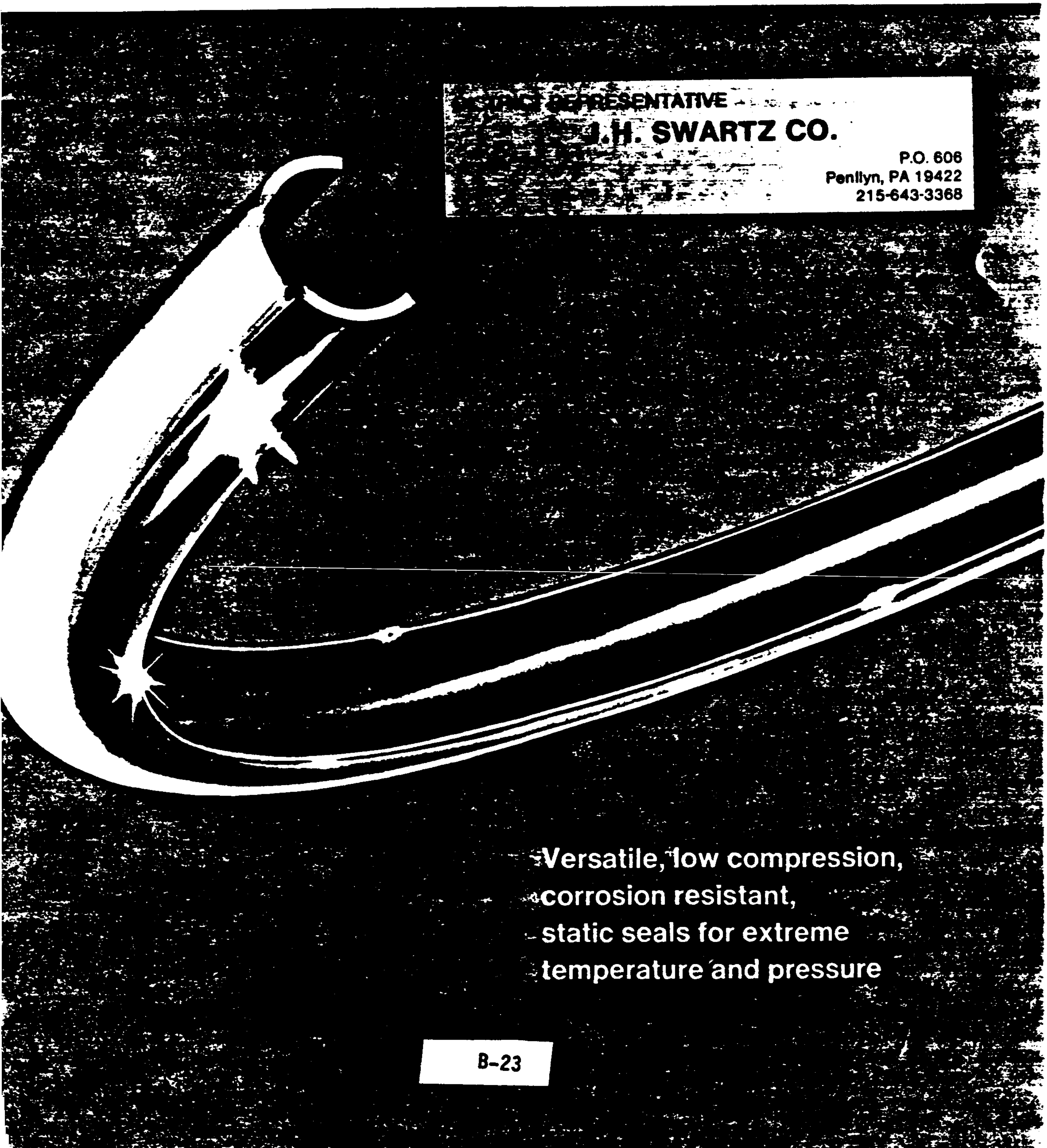
FLUOROCARBON

METALLIC C-RINGS



B-23

METALLIC

C-Rings

Metallic C-Rings solve gas or liquid sealing problems involving high or low temperature, high or low pressure, radiation, chemical corrosion, physical erosion, use and/or storage life. Metallic C-Rings (designated MCR) are designed for static sealing on machinery or equipment and are available for internal pressure, external pressure, or axial pressure ID/OD applications. They are offered in round or irregular shapes and in a broad range of sizes from .126" (3,2 mm) OD x .032" (0,81 mm) free height to over 300" (7620 mm) OD x .625" (15,88 mm) free height.

Materials, Coatings, Heat Treating

Metallic C-Rings are the most versatile of all metal seals. They can be manufactured from a wide variety of materials, including type 321 stainless steel, Inconel 600, Inconel 718, Inconel X-750, and other alloys. Depending on application, the Metallic C-Ring can be electroplated with silver, copper, indium, nickel, gold, lead and other metals, or it can be coated with Teflon. The flow of the finish material improves the sealing, especially under high pressure. Since tensile strength and resilience of the seal are determined in part by metal temper, Fluorocarbon offers a choice of heat treating to material specification or tempering to customer specifications.

Springback and Self-Energizing Characteristics

Metallic C-Rings are installed in a groove or a retainer plate, the height of which is normally held to 20% less than the free height of the seal. When the flange faces are closed, the seal is under compression and tends to spring back against the flanges, thus exerting a positive sealing force.

Because the ring is designed with an open side toward the pressure of the installation, the seal is self-energizing. The pressure of the gas or liquid on the open side energizes the seal and increases the sealing force by pushing the seal surfaces against the flange face.

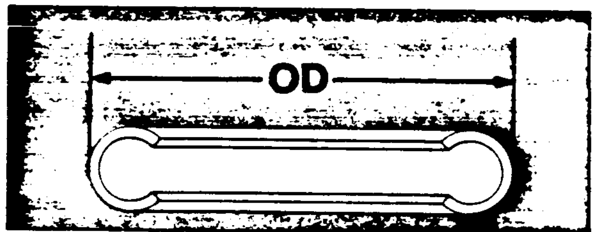
Operating Conditions

Metallic C-Rings are available for sealing applications ranging from cryogenic to 3000°F. (1650° C.) and at pressures from 10^{-10} torr to 100,000 psi (6,804 atm). The extremes of temperature and pressure may not necessarily be combined in the same seal. Provided the C-Ring material selected is compatible with the medium, Metallic C-Rings are not normally affected by radiation, corrosion, or other damaging environments.

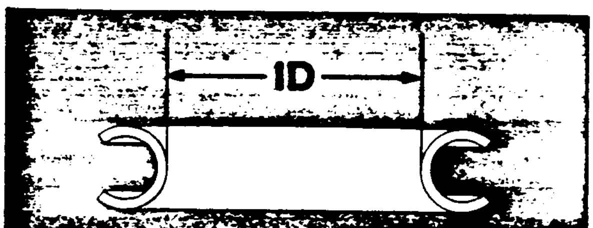
Cost/Performance Ratio

The design of the Metallic C-Ring lends itself to machine production in large quantities. Therefore it is relatively high priced in small quantities. But where quantity requirements are large, the C-Ring provides the lowest unit price of any high performance seal on the market.

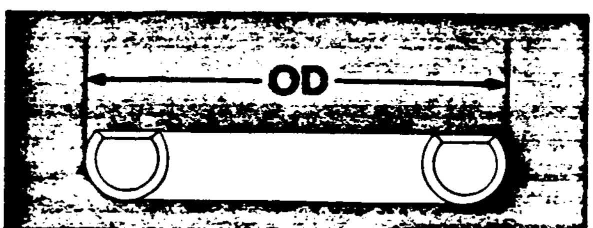
Types of C-RINGS



Type R - Internal Pressure



Type E - External Pressure

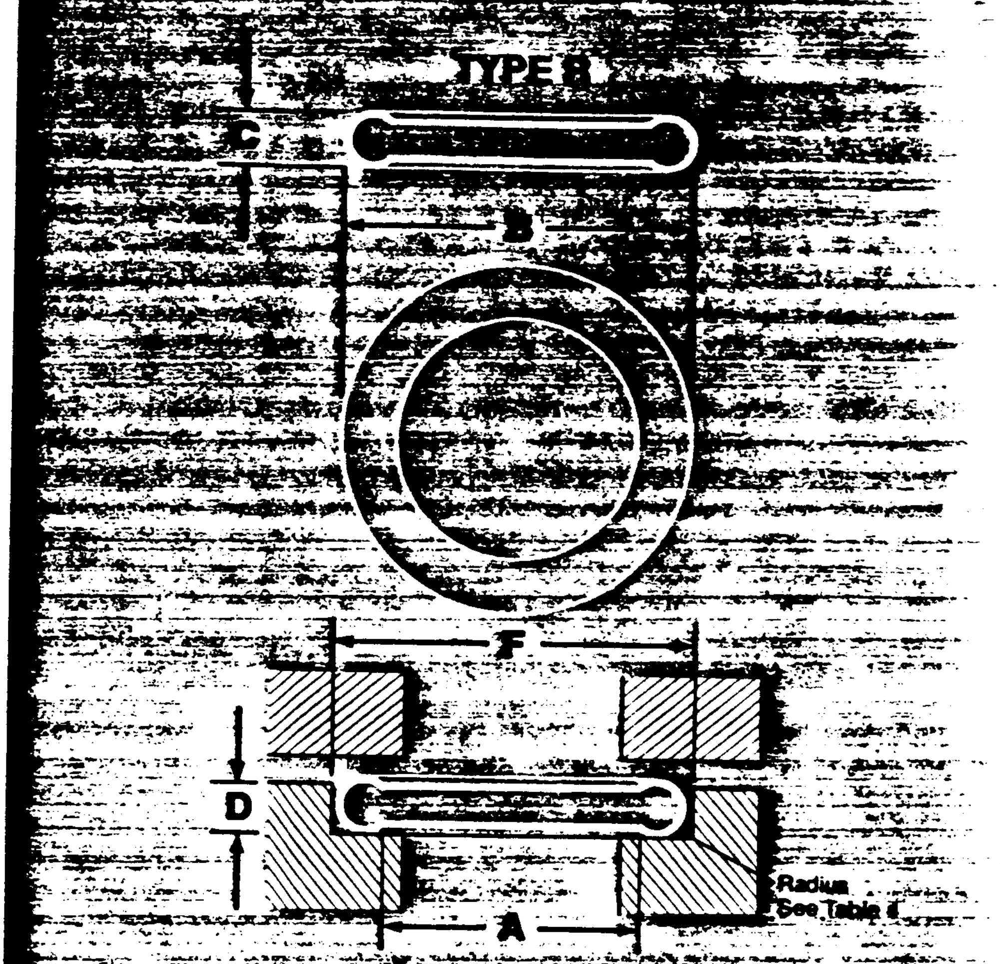


Type A - Axial Pressure ID/OD

ıternal 'ressure

›rnal pressure Metallic C-Ring dimen- ıs are referenced from the OD; thus ove dimensions are calculated from this ·ting point. Conversely, the C-Ring can ɔalculated from a known groove OD.

— C-Ring ID
— C-Ring OD
— C-Ring Free Height
— Groove Depth
— Groove OD



xternal ressure

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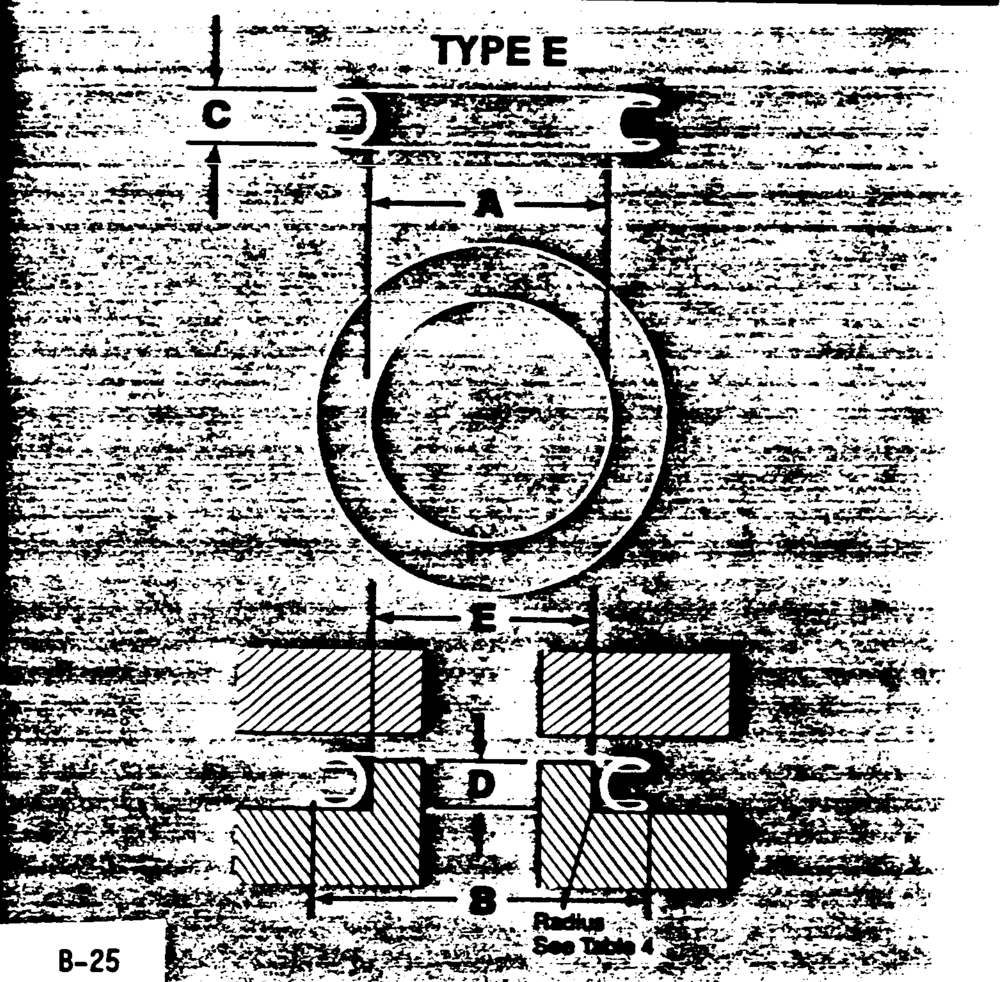
- C-Ring ID
- C-Ring OD
- C-Ring Free Height
- Groove Depth
- Groove ID



Internal Pressure C-Ring and Groove OD Sizes

Table 1

(TYPE R)

C-Ring OD/Tolerance B-Dimension (Inches/mm)					Groove OD/Tolerance* F-Dimension (Inches/mm)				
.126 3.2	to	1.000 25.40	+.000 +0.00	−.002 −0.05	.135 3.43	to	1.010 25.65	+.002 +0.05	−.000 −0.00
1.000 25.40	to	2.000 50.80	+.000 +0.00	−.002 −0.05	1.010 25.65	to	2.010 51.05	+.003 +0.08	−.000 −0.00
2.000 50.80	to	3.000 76.20	+.000 +0.00	−.003 −0.08	2.010 51.05	to	3.010 76.45	+.003 +0.08	−.000 −0.00
3.000 76.20	to	5.000 127.00	+.000 +0.00	−.004 −0.10	3.010 76.45	to	5.010 127.25	+.004 +0.10	−.000 −0.00
5.000 127.00	to	7.000 177.80	+.000 +0.00	−.006 −0.15	5.010 127.25	to	7.010 178.05	+.006 +0.15	−.000 −0.00
7.000 177.80	to	16.000 406.40	+.000 +0.00	−.010 −0.25	7.010 178.05	to	16.010 406.65	+.008 +0.20	−.000 −0.00
16.000 406.40	to	25.000 635.00	+.000 +0.00	−.015 −0.38	16.010 406.65	to	25.010 635.25	+.015 +0.38	−.000 −0.00

Example: 2.000" (50,8mm) OD C-Ring fits 2.010" (51,05mm) OD groove + .003"-.000" (+0,08mm −0,00mm)

* F-Dimensions in above table are for unplated rings.

Internal/External Pressure C-Ring ID And OD

Table 2

(TYPE R & E)

To Determine C-Ring ID For Type R Subtract The Number Below From The B-Dimension To Determine C-Ring OD For Type E Add The Number Below To The A-Dimension (Inches/mm)	Free Height C-Dimension (Inches/mm)	Standard Wall Thicknesses (Inches/mm)
.105 2.66	.063 1.6	.010 0.30
.160 4.06	.094 2.4	.015 0.38
.200 5.08	.126 3.2	.015 0.38
.300 7.62	.188 4.8	.020 0.51
.400 10.16	.252 6.4	.025 0.64

External Pressure C-Ring and Groove ID Sizes

Table 3

(TYPE E)

C-Ring ID/Tolerance A-Dimension (Inches/mm)					Groove ID/Tolerance* E-Dimension (Inches/mm)				
.126 3.2	to	1.000 25.40	+.002 +0.05	−.000 −0.00	.115 2.92	to	0.990 22.63	+.000 +0.00	−.002 −0.05
1.000 25.40	to	2.000 50.80	+.002 +0.05	−.000 −0.00	0.990 25.14	to	1.990 50.54	+.000 +0.00	−.003 −0.06
2.000 50.80	to	3.000 76.20	+.003 +0.08	−.000 −0.00	1.990 50.54	to	2.990 75.94	+.000 +0.00	−.003 −0.08
3.000 76.20	to	5.000 127.00	+.004 +0.10	−.000 −0.00	2.990 75.94	to	4.990 126.74	+.000 +0.00	−.004 −0.10
5.000 127.00	to	7.000 177.80	+.006 +0.15	−.000 −0.00	4.990 126.74	to	6.990 177.54	+.000 +0.00	−.006 −0.15
7.000 177.80	to	16.000 406.40	+.010 +0.25	−.000 −0.00	6.990 177.54	to	15.990 406.14	+.000 +0.00	−.008 −0.20
16.000 406.40	to	25.000 635.00	+.015 +0.38	−.000 −0.00	15.990 406.14	to	25.990 660.14	+.000 +0.00	−.015 −0.38

Example: 2.000" (50,8mm) ID C-Ring fits 1.990" (50,54mm) ID groove +.000" −.003" (+0,00mm −0,08mm)

* E-Dimensions in above table are for unplated rings.

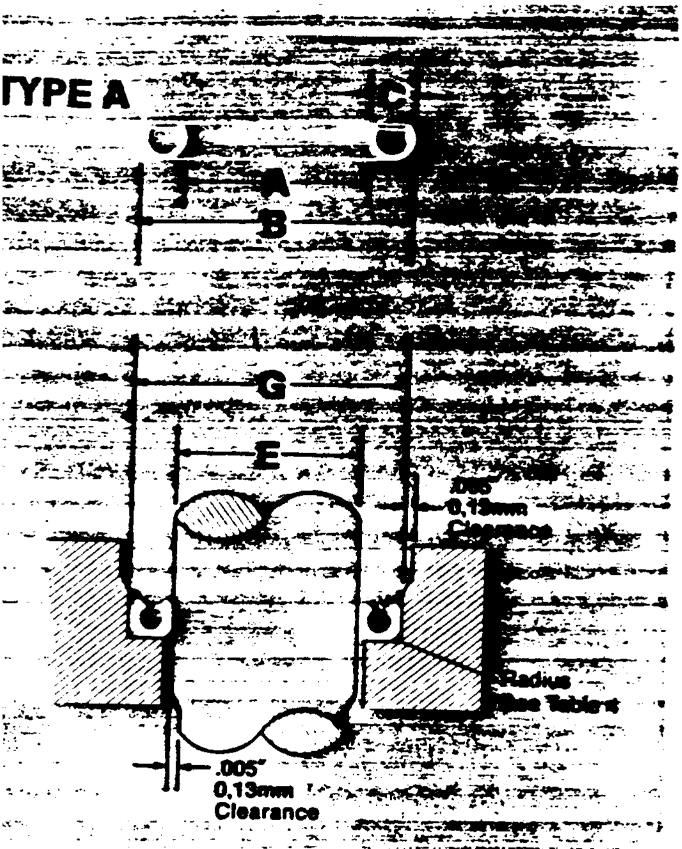
C-Ring Free Height and Groove Depth Dimensions

Table 4

(TYPE R & E)

C-Ring Free Height C-Dimension (Inches/mm)	Groove Depth for 20% Ring Compression D-Dimension (Inches/mm)	Corner Radius (Maximum) (Inches/mm)
.063 ±.002 1.6 ±0.05	.050 ±.001 1.27 ±0.03	.020 0.50
.094 ±.002 2.4 ±0.05	.074 ±.003 1.88 ±0.08	.030 0.76
.126 ±.003 3.2 ±0.08	.100 ±.003 2.54 ±0.08	.045 1.14
.188 ±.004 4.8 ±0.10	.151 ±.003 3.83 ±0.08	.070 1.77
.252 ±.004 6.4 ±0.10	.200 ±.004 5.08 ±0.10	.090 2.28

(For Internal or External Pressure)



— C-Ring ID
— C-Ring OD
— C-Ring Cross Section (Nominal)
— Cavity ID
— Cavity OD

Axial Pressure ID/OD

Table 5

(TYPE A)

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Axial pressure C-Ring Seals are installed by an interference fit at ID and OD. Dimensional data and tolerances are shown below.

Axial C-Ring			Seal Cavity	
C-Dim. Inches/mm Ref. *	B-Dim. Inches/mm	A-Dim. Inches/mm	G-Dim. Inches/mm	E-Dim. Inches/mm
.063 (.010) 1.6 (0.30)	.251 to 1.501 ±.001 6.37 to 38.12 ±0.03	.122 to 1.372 ±.001 3.09 to 34.84 ±0.08	B −.003 −.001 +.000 B −0.08 −0.03 +0.00	A +.003 −.000 +.001 A +0.03 −0.00 +0.03
.094 (.015) 2.4 (0.38)	.313 to 3.063 ±.001 7.95 to 77.80 ±0.03	.122 to 2.872 ±.001 3.09 to 72.94 ±0.03	B −.003 −.001 +.000 B −0.08 −0.03 +0.00	A +.003 −.000 +.001 A +0.08 −0.00 +0.03
.126 (.015) 3.2 (0.38)	.626 to 3.249 ±.001 15.90 to 82.52 ±0.025	.372 to 2.872 ±.001 9.44 to 72.94 ±0.025	B −.003 −.001 +.000 B −0.08 −0.03 +0.00	A +.003 −.000 +.001 A +0.08 −0.00 +0.03
.126 (.015) 3.2 (0.38)	3.249 to 5.749 ±.002 82.52 to 146.02 ±0.05	2.995 to 5.495 ±.002 76.07 to 139.57 ±0.05	B −.006 −.002 +.000 B −0.15 −0.05 +0.00	A +.006 −.000 +.002 A +0.15 −0.00 +0.05
.188 (.020) 4.8 (0.51)	3.374 to 5.874 ±.002 85.70 to 149.20 ±0.05	2.995 to 5.495 ±.002 76.07 to 139.57 ±0.05	B −.006 −.002 +.000 B −0.15 −0.05 +0.00	A +.006 −.000 +.002 A +0.15 −0.00 +0.05
.252 (.025) 6.4 (0.64)	4.374 to 5.999 ±.002 111.10 to 152.37 ±0.05	3.865 to 5.495 ±.002 98.17 to 139.57 ±0.05	B −.006 −.002 +.000 B −0.15 −0.05 +0.00	A +.006 −.000 +.002 A +0.15 −0.00 +0.05

Other sizes can be manufactured. Consult factory for dimensions and tolerances.
* Standard wall thicknesses shown in parentheses.

How to Specify C-Rings

Example For Ordering Type R C6415-02500RAB

In the sample below, C6415-02500RAB indicates a Metallic C-Ring (C) made with Inconel X-750 (6), .126" (3,2 mm) free height (4) and a .015" (0,38 mm) wall thickness (15). (02 500) indicates the Metallic C-Ring OD which is 2.500" (63,500 mm), internal pressure type (R), heat treated none (A), and silver plated to .002/.003" (0,05/0,08 mm) thick (B).

C — Denotes Metallic C-Ring

6 — **Materials**
1-Inconel 718
2-Stainless Steel 321
3-Aluminum
4-Copper
5-Inconel 600
6-Inconel X-750 (Standard)
7-Stainless Steel 304
8-Stainless Steel 316
9-Stainless Steel 347
X-As Specified

4 — Free Height (Thirty-Seconds)

15 — **Standard Wall Thickness (Expressed In Inches)** See Table 2, Page 4 & Table 5, Page 5.

02 500 — Metallic C-Ring Diameter (Inches/Thousandths)
See Ring Types Page 2

Note: This is the OD Dimension for Type R and A, or the ID Dimension for Type E.

R — **Types**
R-Internal Pressure
E-External Pressure
A-Axial Pressure ID/OD
See Ring Types, Page 2

A — **Heat Treatment**
A-None
B-4 Hour Aging
C-16 Hour Aging
D-Solution Annealing and Aging
E-As Specified

B — **Coatings**
A-Silver .001/.002 0,03/0,05
B-Silver .002/.003 0,05/0,08
D-Teflon .001/.003 0,03/0,08
E-Teflon .003/.004 0,08/0,10
L-Copper .001/.002 0,03/0,05
N-None
P-Lead .001/.002 0,03/0,05
R-Indium .001/.002 0,03/0,05
T-Nickel .001/.002 0,03/0,05
V-Gold .0005/.001 0,01/0,03

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THE ADVANCED PRODUCTS COMPANY

December 10, 1986

Atlantic Research Corporation
7511 Wellington Road
Gainesville, Virginia 22065

Attention: Mr. Nicholas Brown, Bldg. 300

Dear Mr. Brown:

We confirm our telephone conversation of today and discussion concerning a metal seal for the Space Shuttle Rocket Booster.

As a manufacturer of large silver plated head gaskets for nuclear reactors (up to 25 ft. dia.), we welcome an opportunity to work with you on this project.

We are looking forward to receiving your inquiry so we can submit specific recommendations.

Thank you for your interest.

Very truly yours,

D. F. Pfeiffer
Sales Manager

DFP/mnn
Enclosure

RESILIENT METAL SEALS

ORIGINAL PAGE IS
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ADVANCED PRODUCTS

DESIGN
MANUAL

B-29

Contents

For more than thirty years, The Advanced Products Company has served industry worldwide with high quality extreme environment seals and sealing systems.

Today, Advanced EnerRings are installed in most major aircraft engines, both military and commercial, in pumps and valves, in nuclear reactors and in many other special applications that demand seals of all sizes and shapes – from ¼" to 25 ft. in diameter.

Call us for design assistance and recommendations on your next extreme environment sealing problem. You'll find our phone nearest you listed on the back cover of this manual.

ADVANCED PRODUCTS COMPANY
VENDOR CODE: 04319

EnerRing®

Energized Static Metal Seals

This manual is a guide to assist Design and Project Engineers who must seal assemblies of all types under extreme environments . . . environments where conventional seals, such as rubber and elastomers, cannot function satisfactorily.

What are EnerRings?
EnerRings are resilient, all-metal, static seals that are energized by the system pressure – as with Metal-O-Rings (MOR), Spring-Energized-Rings (SER), Metal-C-Rings (MCR), and Metal-V-Rings (MVR); or are energized by internal pressurization – as with pressure-filled MOR.

Metal-Wire-Rings (MWR), although not resilient, energized seals, are also included in this EnerRing Design Manual.

Installation
For successful application, all EnerRings are installed in a counterbore, groove or retainer to control the deflection (compression) of the ring and obtain the optimum seal.

About Seating Loads and Resilience
The seating loads required to compress EnerRings for proper sealing vary, depending on the seal design, free height, wall thickness, material selected, and groove depth.

A principal performance feature of an EnerRing seal is its resilience. This causes the seal to "springback" against the loading faces, helping create – and maintain – a more effective seal at all times. This resilience is particularly important where flange separation – caused by pressure, temperature, or flange rotation – is a factor!

Because seating loads for EnerRings are significantly less than those required for crush-type gaskets, the strength and mass of the flanges can be reduced. This can be particularly important to designers concerned with down-sizing and down-weighting.

Seating loads and springback for the various EnerRing designs are shown in Section 4.

When To Use EnerRings
Frequently, much of today's technology precludes the use of rubber or elastomer seals. With the emphasis on higher performance levels under increasingly difficult ambient conditions, EnerRing's unique performance characteristics become even more important:

Vacuum – 10^{-11} Torr

High Pressure – to 100,000 PSI

Temperature – Cryogenics (– 423°F) to 2000°F

Radiation – Unlimited

To sum up, the sealing capabilities of EnerRings go far beyond those of rubber or elastomers. Advanced metal seals do not deteriorate over time due to compacting, outgasing or blowouts, characteristics attributable to composite asbestos and other organic gaskets.

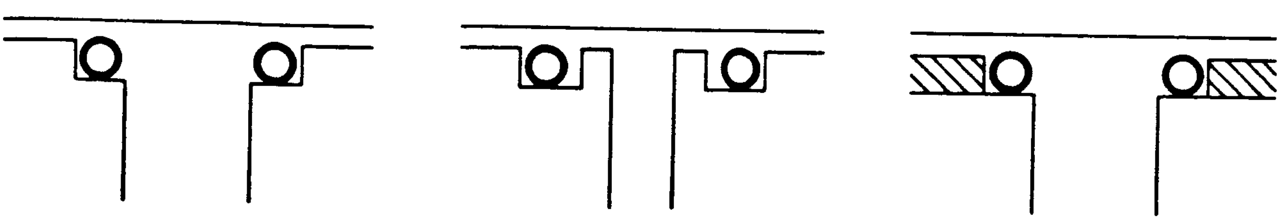
At the same time, it should be remembered that all-metal EnerRings carry a higher *initial* cost than conventional gaskets. This makes it particularly important for designers to thoroughly analyze their sealing problems, required performance levels, allowable failure rates and disassembly/overhaul intervals to assess their true requirements and factor those against the in-use cost – cost effectiveness – of EnerRings vs. conventional sealing devices.

How EnerRings Are Installed
EnerRings are installed between two flat, parallel surfaces. There are three basic installation methods: 1) in a counterbore, 2) in a groove, and 3) with a retainer plate.

In all cases, seals should be installed as close to the location of the clamping load as possible to minimize the possible effects of distortion, flange rotation or separation. This is particularly true for high pressure applications, where the diameter of the bolt circle is appreciably greater than the diameter of the bore to be sealed. Where conditions of extreme pressures or temperatures exist, recessed grooves rather than counterbores may be needed to protect the seal.

A retainer plate eliminates the need for a counterbore or groove. It is the simplest of the three methods because of the ease of preparing the mating surfaces to the specified finish. Often, it is the only practical way to retrofit existing equipment to use metal seals.

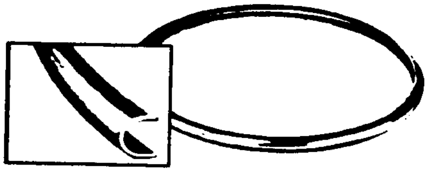
All EnerRings can be installed in these types of configurations.
(Metal-O-Rings Shown Below)



(1) Counterbore (2) Groove (3) Retainer Plate

Section 2—The EnerRing Seal Line

Metal-C-Rings (MCR)

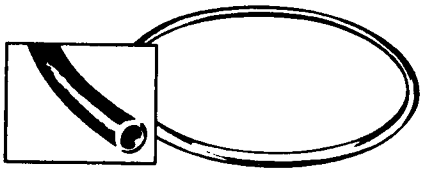


Available for internal, external, and axial applications, Metal-C-Rings are characterized by the "slot" running around the entire circumference. In this self-energizing design, the slot is always installed facing the high-pressure side of the system. For this reason, Metal-C-Rings are not suitable for applications that involve reversing high pressures. The system pressure itself supplements the effectiveness of the seal by forcing the walls of the Metal-C-Ring against the mating sealing surfaces. This energizing effect is proportional to the system pressure.

These seals are the best choice for applications that involve low seating loads and require high springback . . . and also for applications that combine a small OD with a large free height. For low vacuum systems in particular, and low pressure applications in general, the sealing characteristics are more than adequate.

Metal-C-Rings can be produced in circular and non-circular shapes—but the direction of a curve cannot be reversed. Available in Inconel X-750 and other alloys in a wide range of tempers and wall thicknesses, they are an excellent choice for a broad range of applications.

Metal-O-Rings (MOR)



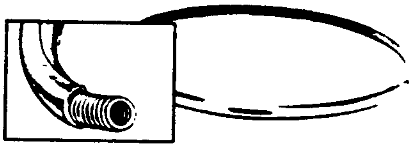
Metal-O-Rings are exceptionally versatile seals that have been proved effective in a very broad range of extreme-environment applications. They are fabricated from a variety of stainless steels, Inconels, and other alloys. In addition to round shapes, Metal-O-Rings can be produced in almost any non-round or irregular configuration, including reverse curves. They can be useo in many applications involving reversing pressures. Vented Metal-O-Rings are generally recommended where pressures exceed 1000 psi. Pressure-filled rings are advised and are available for certain high-temperature, low-pressure applications.

Because the Metal-O-Ring under compression deforms to conform to the mating flange surfaces, it is one of the most effective and "forgiving" of all the EnerRing seals in compensating for such flange conditions as waviness or minor deficiencies in flatness or parallelism.

Metal-Wire-Rings (MWR)

Full section wire rings, essentially crush-type gaskets, are used successfully where high flange loading is available and where springback is not important.

Metal Spring-Energized-Rings (SER)



EnerRing Metal Spring-Energized-Rings consist of a C-shaped outer metal jacket and a coiled spring. Employed for either internal or external pressures, their sealing effectiveness traces to the dual resistance of the jacket and its supporting spring against the seating load. Additionally, in high-pressure applications, the inherent resilience of the seal itself is reinforced by the system pressure.

Compared with other types of resilient seals, such as Metal-O-Rings, C-Rings, or V-Rings, Spring-Energized-Rings require higher seating loads. However, they generally produce better, more consistent sealing—because of this greater load and springback. While platings and coatings are available where required (under vacuum conditions or with poor flange finishes, for example), the Spring-Energized-Rings seal effectively without them in many applications. Additionally, their increased springback makes Spring-Energized-Rings the best EnerRing for maintaining a seal during flange separation.

EnerVac™ Aluminum SER Vacuum Seals

Designed for hard-vacuum and cryogenic applications, this special EnerRing design is essentially an unplated SER (see above). This seal is made of a soft aluminum outer jacket which itself tends to flow into the flange-face imperfections . . . rather than requiring an added cost, hard-to-handle outer plating . . . blocking potential leakage paths. A coiled stainless steel spring gives the seal strength and resilience.

Metal-V-Rings
Symmetrical (MVR)

Metal-V-Rings
Asymmetrical

Metal-V-Rings are particularly suitable for high reliability sealing in high-pressure or low-vacuum applications. Machined to close tolerances, they must be installed in precision grooves or a retainer plate between two flat, parallel surfaces. Available in both external and internal configurations, they are made only in circular shapes. The Metal-V-Ring is one of the most reliable EnerRings.

The asymmetrical configuration—1 long leg and 1 short leg—is designed specifically for sealing in an MS-33649 Boss and Union Fitting

Selecting the Optimum Seal

EnerRings are made of metals chosen specifically for their physical properties and for their characteristics and behavior under a broad range of extreme operating conditions. Essentially, this is why they are suitable over a wide spectrum of applications. The most effective seal is the one in which the metal selected and the seal design are carefully teamed to meet the primary requirements of the application – pressure, temperature, springback, seal life, etc.

The seal selection process is most always a compromise between design limitations, size, available load, springback, design needs . . . and cost. Following is a review of some of the major parameters to be considered.

Whatever you are sealing – pumps, valves, fuel nozzles, head gaskets, exhaust manifolds, pressure vessels, actuators, connectors, piping, reactor vessels – the parameters under which the component or system is to operate must be clearly established to ensure optimum design. Before proper seal selection can be made, operating factors and data, typified by the following, must be analyzed: Temperature; pressure-vacuum; seal life expectancy; rate of leakage allowed, if any; radioactivity, etc. Once these criteria are clearly established and their effect known, the seal selection process can begin.

Base Materials

The material selection table at the right offers a review of the various standard materials used in the manufacture of Advanced EnerRings. Included are the temperature limits for each of the materials, and a cross-reference to the applicable seal types for each.

Material	Max. Temp.	M-W-R	M-O-R	M-V-R	M-C-R	M-SE-R
Nickel	2200°F	•				
Haynes 25	2000°F		•			
Hastelloy	1800°F			•	•	
Gold	1700°F	•				
Copper	1700°F	•				
Inconel 718	1400°F		•		•	•
Inconel-X-750	1400°F	•	•	•	•	•
Inconel-600	1200°F	•	•			
300 Series-SS	800°F	•	•			•
Aluminum	400°F	•				•

Yield Strength (psi) for Various Materials at Various Temperatures (000 Omitted)

Material	Code	72°F. 20°C.	200°F. 93°C.	400°F. 204°C.	600°F. 316°C.	800°F. 427°C.	1000°F. 538°C.	1200°F. 649°C.	1400°F. 760°C.	1600°F. 871°C.	1800°F. 982°C.	2000°F. 1093°C.
Type 304SS	1	34	27	20	17	15	14	12	11	9	5	2
316SS	2	38	35	29	26	23	21	19	18	14	7	3
321SS	3	33	31	28	25	22	20	18	15	11	6	3
347SS	4	39	37	33	32	31	29	26	22	15	9	6
Monel 400	5	47	48	49	50	51	24	28	34	40	54	
Inconel 600	6	38	34	32	30	29	28	24	20	10	6	
Inconel X-750 Annealed	7	36	35	33	31	29	28	26	17	9	4	
Age Hardened	7	124	122	120	118	116	116	115	94	30	10	
Inconel 718 Age Hardened	14	164	162	160	158	154	148	142	106	42	15	6
Haynes 25	9	68	58	48	40	39	36	36	38	34	26	12

Pressures

While all EnerRings can seal at relatively high pressures, each seal design has its individual range . . . from high pressure to vacuum. The following table (pg.4) gives a breakdown of pressure ratings for each of the five types of EnerRings. Metal-O-Rings, for example, that are "vented" (self-energized) will seal considerably higher pressures than non-vented rings. "Venting" means that the Metal-O-Ring incorporates one or several holes on the pressure side, thus equalizing the pressure between the system and the inside of the ring.

ORIGINAL PAGE IS OF POOR QUALITY

Pressure	M-W-R	M-O-R	M-V-R	M-C-R	M-SE-R
100,000 psi		BV			
50,000		BV		B	A
25,000	A	BV	A	B	A
10,000	A	BV	A	B	A
5,000	A	BV	A	B	A
3,000	A	AV/B	A	A	A
1,000	A	AV/B	A	A	A
500	A	AV/B	A	A	A
15	A	A	A	A	A
10^{-1} Torr	A	A	A	A	A
10^{-2}	A	A	A	A	A
10^{-3}	A	A	A	A	A
10^{-4}	A	A	A	A	A
10^{-5}	A	A	A	A	A
10^{-6}	A	A	A	A	A
10^{-7}	A	B		B	A
10^{-8}	A	B		B	A
10^{-9}	A			B	A
10^{-10}	A			B	A
10^{-11}				B	A
10^{-12}					A
10^{-13}					A
See Also	NA	See Page 15	NA	See Page 12	NA

A = Standard Wall B = Heavy Wall V = Vented
See Section 4 – EnerRing Seal Selection
Tables (pgs. 6-9) for specific pressure ratings

Flange Finish
Depending on the application, EnerRings require flange finishes between 8 and 32 rms. Surface finishes of the mating flanges contribute much to the effectiveness of the seal. For example, by improving flange finish, it is possible, as the table shows, to compensate for some of the difficulties which can be encountered in sealing under extreme conditions.

Media/Pressure	Recommended Flange Finish
1. Hard and Ultra-High Vacuums	8 rms
2. Cryogenic/Hard Vacuum	8 rms
3. Helium, Hydrogen, Freon	8 rms
4. Cryogenic/Light Vacuum	16 rms
5. Steam	16 rms
6. *Nitrogen, Air, Argon, Fuel, Water	32 rms
7. *Hydraulic Oil, Polymer	32 rms

* If Pressures are higher than 10,000 PSI, improve finish to 16 rms

If Pressures are higher than 25,000 PSI, improve finish to 8 rms

Mating flanges require Circular Lay finish in the seal's footprint area, and must be free of dirt, grit or foreign matter.

Platings & Coatings
When an EnerRing is selected to seal air, water or gases, or is compressed with a threaded flange, the seal must be plated or coated. This plating or coating penetrates the imperfections in the mating surfaces; improves sealing capability; and lubricates, preventing galling of the metals.

The many standard plating and coating materials are shown in the table below. The selection process should take into consideration maximum temperature, compatibility with the medium to be sealed, and seating load. If the seal chosen has a required seating load of less than the recommended load range shown in the table below, then the plating/coating selected will not flow and, therefore, will not neutralize the mating surface irregularities. Plating or coating should also be applied if corrosion-resistance is desired.

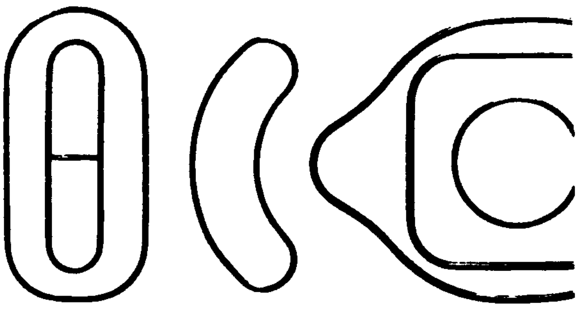
Maximum Working Temperature	Plating/Coating Type	Recommended Load Range for Sealing (lbs./circum. in.)
150° F	Indium	75-350
450° F	Teflon	150-450
300° F	Lead	100-400
1500° F	Silver	250 Up
1700° F	Copper	250 Up
1700° F	Gold	200 Up
2200° F	Nickel	400 Up

Dimensional Allowances for Plating and Coatings

Seals: The OD and free height dimensions will be increased 2 times the plating or coating thickness. The ID will be decreased by 2 times the plating or coating thickness.

Grooves: For internal pressure seals, increase groove OD by 2 times the maximum coating/plating thickness. For external pressure seals, decrease groove ID by 2 times the maximum coating/plating thickness. Do not change the groove depth.

Non-Circular EnerRings
All but the Metal-V-Ring can be manufactured circular or shaped.



The illustration above shows some of the many different shapes in which EnerRings can be made. This characteristic offers the designer flexibility of selection and installation, a flexibility not found in most other resilient metal seal designs. For applications as varied as fuel nozzle supports on aircraft gas turbines, or dies for extrusion of plastic film, the availability of non-circular EnerRings permits the equipment designer to make his design more productive and less costly than would be possible if only circular

The Metal-O-Ring is the most versatile of the non-circular EnerRing family as it may be made in complex shapes involving reverse curves. Metal-C-Rings and Metal Spring-Energized-Rings may be made in simpler non-circular shapes, but cannot accommodate reverse curves.

For any non-circular EnerRing, the width of its groove or seat must equal at least the minimum groove width specified in the appropriate Seal and Groove Specification Table (pgs. 10-22). In addition, any radius must be at least half the minimum recommended OD of a circular EnerRing of the same free height. (See table below.)

MINIMUM RADII (INCHES)

Free Height (Inches)	.035	.062	.094	.125	.156	.188	.250	.375	.500	.625
MOR	.125	.250	.500	1.000	2.000	3.000	4.000	8.000	12.000	16.000
MCR	.125	.158	.375	.500	.625	1.875	2.250	7.375	10.500	–
SER	–	–	.500	.500	.750	1.875	2.250	7.375	10.500	–

For appropriate design of a non-circular EnerRing, we suggest that you send us your layout or sketch so that we can review it and make a recommendation to you.

EnerRing Selection and Specification

STEP 1 In Section 4, which follows (pgs. 6 - 9), the general parameters discussed earlier are presented in chart form for each of a series of O.D. size and free-height ranges to simplify selection of a specific seal.

Select, by: Seal Design, O.D., Free-Height*, and Seal Material, the EnerRing most applicable.
*An important reminder: For maximum springback, specify the **largest** Free-Height for the space available.

STEP 2 In Section 5 (pgs. 10 - 22), turn to the appropriate (by seal design) SEAL & GROOVE SPECIFICATION TABLE, to determine groove and EnerRing dimensions and tolerances and apply this information – and that from STEP 1 – to the EnerRing Part Numbering System that follows each of the Seal & Groove Specification Tables.

For assistance in selecting – or specifying – an EnerRing, please call us at (203) 239-3341.

OD RANGE 16.000"/80.000" FREE HEIGHT .375"

Working Pressure Rating, psi	EnerRing Type	Material	Wall Thickness	Temper (Hardened)	Springback	Seating Load lb/in. circum
800	MOR-PLAIN	304	.038"	Work	.0040"	600
1100	MOR-PLAIN	304	.049"	Work	.0030"	1100
1100	MOR-PLAIN	718	.038"	Work	.0055"	850
1550†	MOR-PLAIN	718	.038"	Age	.0110"	1700
12000	MOR-VENT	304	.038"	Work	.0040"	600
18000	MOR-VENT	304	.049"	Work	.0030"	1100
12000	MCR	718	.038"	Work	.0150"	450
4000	SER-M10	SS or 718/E	–	Work	.0150"	1800
4000	SER-M20	SS or 718/E	–	Work	.0170"	2400
4000	SER-H10	SS or 718/E	–	Work	.0120"	4700
4000	SER-H20	SS or 718/E	–	Work	.0140"	7000
18000	MCR	718	.038"	Age*	.0180"	700
18000	MCR	718	.057"	Work	.0140"	1250
18000	MOR-VENT	718	.038"	Work	.0055"	850
24000	MCR	718	.057"	Age*	.0170"	1600
24000†	MOR-VENT	718	.038"	Age	.0110"	1700

†Used at 10% Nominal Compression *Max. OD-54"

OD RANGE 32.000"/180.000" FREE HEIGHT .500"

Working Pressure Rating, psi	EnerRing Type	Material	Wall Thickness	Temper (Hardened)	Springback	Seating Load lb/in. circum
800	MOR-PLAIN	304	.050"	Work	.0050"	800
1100	MOR-PLAIN	304	.065"	Work	.0040"	1600
1100	MOR-PLAIN	718	.050"	Work	.0070"	1150
1550†	MOR-PLAIN	718	.050"	Age	.0140"	2300
15000	MOR-VENT	304	.050"	Work	.0050"	800
21000	MOR-VENT	304	.065"	Work	.0040"	1600
15000	MCR	718	.050"	Work	.0200"	600
5000	SER-M10	SS or 718/E	–	Work	.0180"	2000
5000	SER-M20	SS or 718/E	–	Work	.0220"	2800
5000	SER-H10	SS or 718/E	–	Work	.0160"	5800
5000	SER-H20	SS or 718/E	–	Work	.0200"	9500
21000	MCR	718	.050"	Age*	.0250"	850
21000	MCR	718	.075"	Work	.0180"	1300
21000	MOR-VENT	718	.050"	Work	.0070"	1150
30000	MCR	718	.075"	Age*	.0200"	1850
30000†	MOR-VENT	718	.050"	Age	.0140"	2300

†Used at 10% Nominal Compression *Max. OD-54"

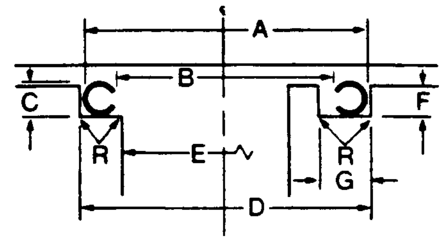
OD RANGE 60.000"/260.000" FREE HEIGHT .625"

Working Pressure Rating, psi	EnerRing Type	Material	Wall Thickness	Temper (Hardened)	Springback	Seating Load lb/in. circum
800	MOR-PLAIN	304	.063"	Work	.0060"	1000
1100	MOR-PLAIN	304	.093"	Work	.0050"	1900
1100	MOR-PLAIN	718	.063"	Work	.0085"	1400
1550†	MOR-PLAIN	718	.063"	Age	.0180"	2800
15000	MOR-VENT	304	.063"	Work	.0060"	1000
21000	MOR-VENT	304	.093"	Work	.0050"	1900
7000	MOR-VENT	718	.063"	Work	.0085"	1400
30000†	MOR-VENT	718	.063"	Age	.0180"	2800

†Used at 10% Nominal Compression

Section 5—Seal and Groove Specification Tables

Metal-C-Rings
Internal Pressure

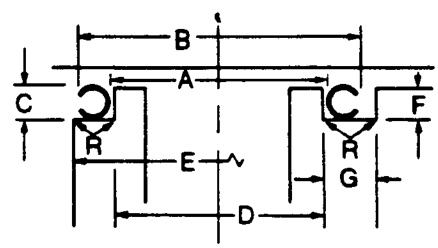


All dimensions are in inches and are shown before plating.

RING						GROOVE					
C		A		B (min)	Std Plating Thickness*	D		E (max)	F	G	R
Free Height	Tol	OD Range	Tol	A Minus	Min/Max	A Plus	Tol −.000	A Minus	Min/Max	Min	Max
.031	±.002	.250-1.000	See separate chart page 11	.056	.0010/.0015	.002	+.002	.070	.025/.027	.035	.010
.047	±.002	.315-1.375		.082	.0010/.0015	.002	+.003	.095	.037/.040	.050	.012
.062	±.002	.375-4.000		.106	.0010/.0015	.003	+.004	.120	.050/.054	.065	.015
.094	±.002	.500-6.000		.160	.0010/.0015	.004	+.004	.170	.075/.079	.095	.020
.125	±.003	1.000-8.000		.213	.0010/.0015	.004	+.005	.225	.100/.105	.125	.030
.156	±.003	1.250-12.000		.266	.0015/.0025	.005	+.006	.280	.125/.130	.160	.050
.188	±.004	3.750-24.000		.324	.0015/.0025	.006	+.006	.340	.151/.157	.190	.060
.250	±.004	4.500-36.000		.428	.0020/.0030	.008	+.008	.440	.200/.206	.250	.060
.375	±.004	14.750-80.000		.642	.0020/.0030	.010	+.010	.655	.300/.306	.380	.060
.500	±.005	21.000-112.000		.850	.0030/.0040	.010	+.010	.870	.400/.406	.500	.060

* To determine allowances for plating or coating thickness, see page 4.
NOTE: The inside C-section may not have the plating/coating thickness specified.

Metal-C-Rings
External Pressure



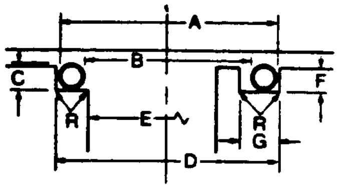
All dimensions are in inches and are shown before plating.

RING						GROOVE					
C		A		B (max)	Std Plating Thickness*	D		E (min)	F	G	R
Free Height	Tol	ID Range	Tol	A Plus	Min/Max	A Minus	Tol +.000	A Plus	Min/Max	Min	Max
.031	±.002	.195-1.000	See separate chart page 11	.056	.0010/.0015	.002	−.002	.070	.025/.027	.035	.010
.047	±.002	.225-1.375		.082	.0010/.0015	.002	−.003	.095	.037/.040	.050	.012
.062	±.002	.250-4.000		.106	.0010/.0015	.003	−.004	.120	.050/.054	.065	.015
.094	±.002	.300-6.000		.160	.0010/.0015	.004	−.004	.170	.075/.079	.095	.020
.125	±.003	.750-8.000		.213	.0010/.0015	.004	−.005	.225	.100/.105	.125	.030
.156	±.003	1.250-12.000		.266	.0015/.0025	.005	−.006	.280	.125/.130	.160	.050
.188	±.004	3.750-24.000		.324	.0015/.0025	.006	−.006	.340	.151/.157	.190	.060
.250	±.004	4.500-36.000		.428	.0020/.0030	.008	−.008	.440	.200/.206	.250	.060
.375	±.004	14.750-80.000		.642	.0020/.0030	.010	−.010	.655	.300/.306	.380	.060
.500	±.005	21.000-112.000		.850	.0030/.0040	.010	−.010	.870	.400/.406	.500	.060

* To determine allowances for plating or coating thickness, see page 4.
NOTE: The inside C-section may not have the plating/coating thickness specified.

Metal-O-Rings

Internal Pressure



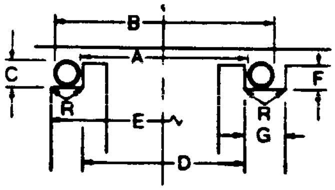
All dimensions are in inches and are shown before plating.

RING						GROOVE					
C		A		B (min)	Std Plating Thickness*	D		E (max)	F	G	R
Free Height	Tol	OD Range	Tol −.000	A Minus	Min/Max	A Plus	Tol −.000	A Minus	Min/Max	Min	Max
.035	+.003 −.001	.250-1.000	+.005	.080	.0010/.0015	.005	+.005	.090	.023/.027	.055	.010
.062	+.003 −.001	.500-5.000	+.005	.135	.0010/.0015	.005	+.005	.150	.045/.050	.090	.015
.094	+.003 −.001	1.000-8.000	+.005	.200	.0010/.0015	.010	+.005	.220	.074/.079	.125	.020
.125	+.003 −.001	2.000-10.000	+.005	.270	.0010/.0015	.010	+.005	.290	.100/.105	.160	.030
.156	+.004 −.000	4.000-18.000	+.005	.330	.0015/.0025	.015	+.005	.375	.125/.130	.200	.050
.188	+.005 −.000	6.000-30.000	+.005	.400	.0015/.0025	.015	+.005	.450	.150/.155	.250	.060
.250	+.005 −.000	8.000-40.000	+.008	.530	.0020/.0030	.020	+.008	.600	.200/.205	.350	.060
.375	+.005 −.000	16.000-80.000	+.010	.800	.0020/.0030	.020	+.010	.900	.300/.305	.500	.060
.500	+.006 −.000	32.000-180.000	+.010	1.050	.0030/.0040	.020	+.010	1.200	.400/.405	.650	.060
.625	+.006 −.000	60.000-260.000	+.010	1.300	.0030/.0040	.020	+.010	1.500	.500/.505	.810	.060

* To determine allowances for plating or coating thickness, see page 4.

Metal-O-Rings

External Pressure



All dimensions are in inches and are shown before plating.

RING						GROOVE					
C		A		B (max)	Std Plating Thickness*	D		E (min)	F	G	R
Free Height	Tol	ID Range	Tol −.000	A Plus	Min/Max	A Minus	Tol +.000	A Plus	Min/Max	Min	Max
.035	+.003 −.001	.180-1.000	+.005	.080	.0010/.0015	.000	−.005	.090	.023/.027	.055	.010
.062	+.003 −.001	.375-5.000	+.005	.135	.0010/.0015	.000	−.005	.150	.045/.050	.090	.015
.094	+.003 −.001	1.812-8.000	+.005	.200	.0010/.0015	.005	−.005	.220	.074/.079	.125	.020
.125	+.003 −.001	1.750-10.000	+.005	.270	.0010/.0015	.005	−.005	.290	.100/.105	.160	.030
.156	+.004 −.000	3.688-18.000	+.005	.330	.0015/.0025	.010	−.005	.375	.125/.130	.200	.050
.188	+.005 −.000	5.625-30.000	+.005	.400	.0015/.0025	.010	−.005	.450	.150/.155	.250	.060
.250	+.005 −.000	7.500-40.000	+.008	.530	.0020/.0030	.010	−.008	.600	.200/.205	.350	.060
.375	+.005 −.000	15.250-80.000	+.010	.800	.0020/.0030	.010	−.010	.900	.300/.305	.500	.060
.500	+.006 −.000	32.000-180.000	+.010	1.050	.0030/.0040	.010	−.010	1.200	.400/.405	.650	.060
.625	+.006 −.000	60.000-260.000	+.010	1.300	.0030/.0040	.010	−.010	1.500	.500/.505	.810	.060

* To determine allowances for plating or coating thickness, see page 4.

Metal-O-Ring PART NUMBERING/ORDERING SYSTEM

NOTE: Any part not in conformance with this numbering system will be assigned a unique 5-digit part number and drawing.

EXAMPLE
TO ORDER: A 2½" Metal-O-Ring, pressure on ID, with standard ID venting, .125" free height, made from 321 SS tubing with .010" wall thickness, silver plated to .0015"/.0025" thickness.

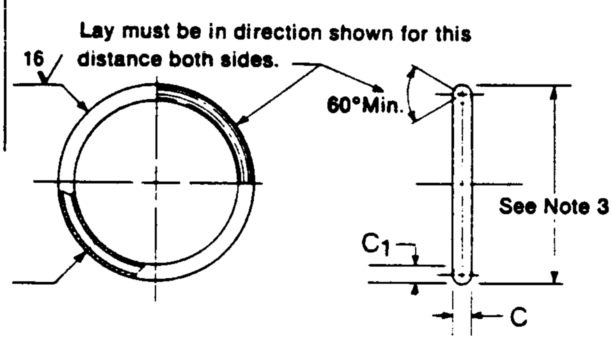
YOU WOULD SPECIFY PART NO: EOI–002500–07–03–1–SPC

SELECT PROPER CODE FROM TABLE BELOW

Measurement System	Product Code	Pressure Direction	Ring Diameter	Free Height × Wall Thickness Code	Material Code	Temper Code	Finish Code	Finish Thickness
E-English	O	N-Pressure on ID (OD Control) Plain Ring (not Vented) M-Pressure on OD (ID Control)-Plain Ring (not Vented) I-Pressure on ID (OD Control)-Standard ID Vent E-Pressure on OD (ID Control)-Standard OD Vent P-Pressure Filled (OD Control) Y-None of the above	(1) to 3 decimals (2) diameter is outside for internal pressure: inside for external pressure.	Inches 01-.035 × .006 **A** 02-.062 × .006 **C** 03- × .010 **A** 04-.094 × .006 **C** 05- × .010 **A** 06-.125 × .008 **C** 07- × .010 **A** 08-.062 × .014 **B** 09-.094 × .018 **B** 10-.125 × .020 **B** 11-.156 × .016 **A** 12- × .020 **B** 13-.188 × .020 **A** 14- × .025 **B** 15-.250 × .025 **A** 16- × .032 **B** 17-.375 × .038 **A** 18- × .049 **B** 19-.500 × .050 **A** 20- × .065 **B** 21-.625 × .063 **A** 22- × .093 **B** 23-.750 × .075 **A** 24- × .113 **B** 25-.125 × .012 **B** 26-.324 × .032 **A** 27-.435 × .044 **A** 28-.455 × .046 **A** 29-.047 × .007 **A** 30-.080 × .006 **C** 31-.062 × .012 **B**	01-304SS 02-316SS 03-321SS 04-347SS 05-Monel 400 06-Inconel 600 07-Inconel X-750 09-Haynes 25 (L605) 14-Inconel 718	1-Work hardened 2-Age hardened (short cycle) 4-Annealed	SP-Silver Plate IP-Indium plate LP-Lead plate NP-Nickel plate GP-Gold plate CP-Copper plate TC-Teflon Coat FOR SUPER FINISH PARTS SFX–Unplated SS–Silver SN–Nickel SG–Gold SC–Copper	Inches A- .0005 .0010 B- .0010 .0015 C- .0015 .0025 D- .0020 .0030 E- .0025 .0035 F- .0030 .0040 G- .0040 .0060 H- .0060 .0090 J- .0035 .0050 K- .0050 .0070 L- .0003 .0005

A = Standard Wall
B = Heavy Wall
C = Thin Wall

1. GENERAL DIMENSIONAL DATA



NOTES:
1. Finish weld flush with OD smooth blend. Dimensions at blend shall not be more than .002" below adjacent surfaces.
2. Ring shall be flat within Roundness or Flatness tolerance shown in the reference table for each tubing diameter. (pg. 24)
3. Ring shall be round within Roundness or Flatness tolerance (in free state). When restrained, diameter shall be within limits.

NOM. TUBE DIA.	C	C1
.035"	.034"/.038"	.029"/.038"
.062"	.061"/.065"	.056"/.067"
.094"	.093"/.098"	.089"/.101"
.125"	.124"/.128"	.120"/.132"
.156"	.156"/.160"	.151"/.160"
.188"	.188"/.193"	.183"/.193"
.250"	.250"/.255"	.245"/.255"
.375"	.375"/.380"	.370"/.380"
.500"	.500"/.506"	.495"/.506"
.625"	.625"/.631"	.620"/.631"

MILITARY STANDARDS FOR METAL-O-RINGS

Military Standard Number	Silver Plated	Nominal Tube Diameter	Tube Wall Thickness
9141	NO	.035	.005 .007
9142	NO	.062	.005 .007
9202	NO	.062	.009/.011
9203	NO	.094	.005 .007
9204	NO	.094	.009/.011
9205	NO	.125	.009/.011
9371	YES	.035	.005/.007
9372	YES	.062	.005/.007
9373	YES	.062	.009/.011
9374	YES	.094	.005/.007
9375	YES	.094	.009/.011
9376	YES	.125	.009/.011

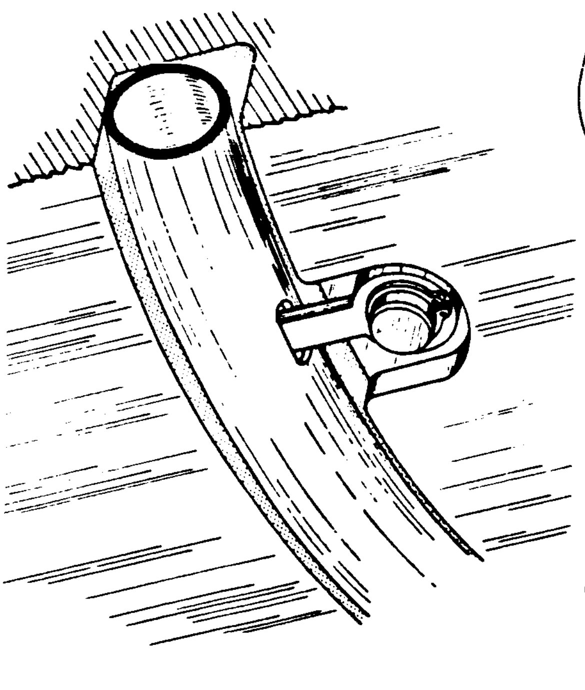
To find appropriate Dash Number, use MS Standards sheets or call

ORIGINAL PAGE IS
OF POOR QUALITY

Design Considerations For Nuclear Pressure Vessel Seals

The Advanced Products Company first introduced nuclear seals in 1960 for the Yankee Reactor in Massachusetts, Indian Point 1 in New York, and the merchantship NS, Savannah. Originally, type 304 stainless steel tubing in the annealed condition was used. Later this was changed to Inconel 718 in age hardened condition. The change resulted in much greater springback which, in turn, created a higher integrity seal. A second change is worth mentioning. At first, seals were accommodated in grooves on the top flange of the vessel. This was changed and both the inner and outer seals were attached to the underside of the head flange. To accomplish this, clips were installed in the head to hold the seals in place during assembly of the head to the vessel.

The critically located vent holes perform another function besides accommodating the clips and holding the seals in their inverted position. Vent holes permit the system's pressure to energize the inside of the hollow ring. The sketch below shows the arrangement.



Arrangement Retaining Metal-O-Ring in Groove

Vent holes are not so large as to relieve the seating force at each vent hole location. Because of the position of the clips, the location of vent holes is critical.

The selection of the correct number of vent holes depends on the ring diameter.

Number of Vent Holes

Less than 144" OD	– 12 vent holes
144" to 218" OD	– 18 vent holes
Larger than 218" OD	– 24 vent holes

Selection Of Free Height

Nuclear pressure vessel seals are manufactured in three standard free heights – .375", .500", and .625". Use the EnerRing Selection Tables on page 9 to choose your size. Springback and seating load requirements are also indicated there. Other free heights are available: see other tables in Section 4 for desired cross-section.

Groove Dimensions

Inconel 718 Metal-O-Rings are compressed only 10%; therefore, the Section 5 Seal and Groove Specification Tables, which are calculated for 20% compression seals, are not applicable for groove depth. Use the following dimensions instead

Free Height	Groove Depth
.375"	.340" + .005/ – .000
.500"	.460" + .005/ – .000
.625"	.570" + .005/ – .000

Summary

Advanced Products Company nuclear seals are manufactured, finished, plated, and inspected under the most stringent quality control conditions.

After the seals are inspected and approved, they are spiral-wrapped with poly foam. A protective polyethylene split tube is then installed over the base wrap. Finally, the longitudinal joint and butt joint of the polyethylene tube are sealed with polyethylene plastic pressure-sensitive tape.

The packaged rings are suspended at eight points in a wooden crate with bubble pack padding and hemp webbing straps.

With more than a quarter century of experience in manufacturing and shipping nuclear reactor seals, we can point to a proud record of dependability, quality, and service.

Parts are inspected to the tolerances given in this Design Manual. Parts are inspected in restrained conditions.

Consult factory for complete technical information.

ROUNDNESS & FLATNESS REFERENCE TABLE FOR MOR-MCR-MWR-SER

OD-Range (Inches)	Free Heights (Inches) .035	.062	.094	.125	.156	.188	.250
.250- 1.000	.020						
.500- 2.500		.030					
2.501- 4.000		.060					
1.000- 2.500			.030				
2.501- 5.000			.060				
5.001- 6.000			.090				
2.000- 2.500				.030			
2.501- 5.000				.060			
5.001- 8.000				.090			
4.000- 5.000					.060		
5.001-10.000					.090		
10.001-12.000					.125		
6.000-10.000						.090	
10.001-12.000						.125	
12.001-14.000						.150	
14.001-16.000						.175	
16.001-18.000						.200	
18.001-22.000						.250	
22.001-24.000						.500	
8.000-10.000							.090
10.001-12.000							.125
12.001-14.000							.150
14.001-16.000							.175
16.001-19.000							.200
19.001-22.000							.250
22.001-36.000							.500

Quality Control and Inspection
To ensure the highest product quality, all Advanced EnerRings are subject to rigorous in-process and final inspections.

All EnerRings are traceable, and can be certified by issuance of a Certificate of Conformance or Material, on request. Specifications that Advanced seals must meet are:

Mil-Q-9858A
Mil-I-45208A
3 — CK

Parts are inspected to the tolerances given in this Design Manual. Parts are inspected in restrained conditions.

EnerSeal™

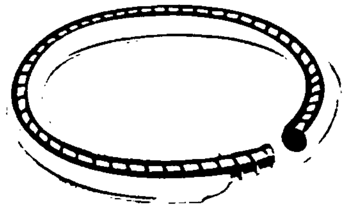
Spring-Energized Polymer Seals For Dynamic and Static Applications

The Advanced Products EnerSeal is a seal assembly consisting of a thin TFE jacket or shell made resilient by an internal spring of various configurations, of either metal or elastomer.

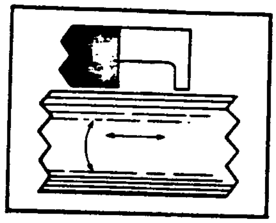
Although primarily a seal for dynamic applications–rotating, oscillating, reciprocating, or face-type–where there is relative motion between the faces–it is also an excellent seal for static, face-type applications, with either internal or external pressures.

Operating Ranges
EnerSeal is specifically designed for applications in which temperatures range from cryogenic to 600°F, and pressures from vacuum to 5,000 psi for dynamic applications, and higher for static installations.

EnerSeal™ MARK I
STANDARD

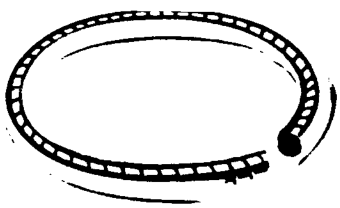


Helical internal spring. Good resilience and springback–with low required loading. Ideal for reciprocating, rotating and static installations.

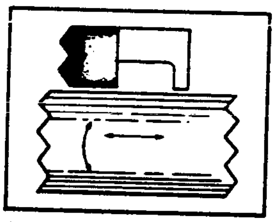


Reciprocating/Rotating

EnerSeal™ MARK I-H
HIGH PRESSURES

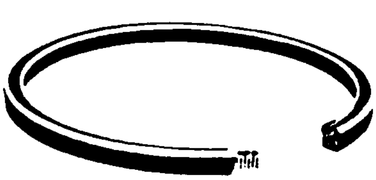


"Long tail" design for pressures greater than 5,000 psi prevents extrusion. Helical internal spring. Low loading requirement, high resilience. The choice for high pressure reciprocating, rotating and static applications. Recommended for vacuum service.

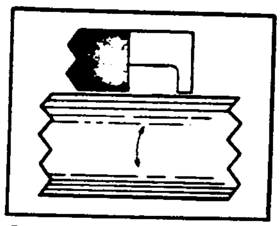


Reciprocating/Rotating

EnerSeal™ MARK IV
ROTATING



V-shaped internal spring. Higher resilience for most effective sealing–with lower flange loading. Greater deflection of spring-energizing element compensates for runout in dynamic applications. Engineered for dynamic rotary applications.

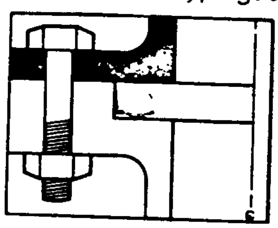


Rotating

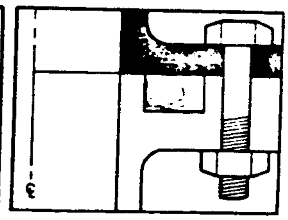
Applications
EnerSeals are designed for use where conventional seals and gaskets will not stand up to the operating and environmental conditions of the application.

Dynamic–in such applications as sealing rods, shafts and pistons. Because EnerSeal jackets are machined, they may be adapted to almost any gland, including such standard glands as MIL-G-5514 and those defined by other standards. They are interchangeable with many conventional seal types–O-rings, V-rings, chevrons, packings, etc.

Static–For internal or external pressure applications, as a face-type gasket.



Static, Internal Pressure



Static, External Pressure

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