MASTER

ENCLOSURE (1)

N8120R:72-022

NOTICE

This report is illegible to a degree
that precludes satisfactory reproduction

ENGINEERING OPERATIONS REPORT

PRESSURE VESSEL FORWARD CLOSURE JOINT

STRESS ANALYSIS

PROJECT 620, WORK STATEMENT 9a

5 JANUARY 1972

J. E. JELLISON

NOTICE

This report was prepared as an account of work sponsored by the United States Government. Neither the United States nor the United States Energy Research and Development Administration, nor any of their employees, nor any of their contractors, subcontractors, or their employees, makes any warranty, express or implied, or assumes any legal liability or responsibility for the accuracy, completeness or usefulness of any information, apparatus, product or process disclosed, or represents that its use would not infringe privately owned rights.

APPROVED:

K. SATO, MANAGER
ENGINEERING STAFF DEPARTMENT

DISTRIBUTION OF THIS DOCUMENT UNLIMITED

CLASSIFICATION CATEGORY	
Unclassified	
CLASSIFYING OFFICER	DATE
1/6/72

DISCLAIMER

This report was prepared as an account of work sponsored by an agency of the United States Government. Neither the United States Government nor any agency Thereof, nor any of their employees, makes any warranty, express or implied, or assumes any legal liability or responsibility for the accuracy, completeness, or usefulness of any information, apparatus, product, or process disclosed, or represents that its use would not infringe privately owned rights. Reference herein to any specific commercial product, process, or service by trade name, trademark, manufacturer, or otherwise does not necessarily constitute or imply its endorsement, recommendation, or favoring by the United States Government or any agency thereof. The views and opinions of authors expressed herein do not necessarily state or reflect those of the United States Government or any agency thereof.

DISCLAIMER

Portions of this document may be illegible in electronic image products. Images are produced from the best available original document.

TABLE OF CONTENTS

	<u>Page</u>
I. Introduction	1
II. Summary/Conclusions	1
III. Technical Discussion	3
IV. List of References	6

LIST OF APPENDICES

A. Material Properties	
B. Critical Part in Bolt Load Path at Ambient Temperature	
C. Finite Element Model	
D. Mechanical Loading and Thermal Environment	
E. Material Properties for Finite Element Axisymmetric Simulation of Non-Axisymmetric Parts	
F. Allowable Preload and Finite Element Simulation of Preload	
G. Bolt Loads, Stresses and Margins of Safety	
H. Flange Sealing	
I. Flange Stresses and Margins of Safety	
J. Cylinder Shell Stresses and Margins of Safety	

PRESSURE VESSEL FORWARD CLOSURE JOINT
STRESS ANALYSIS

I. INTRODUCTION

This report constitutes a steady state stress analysis of the NERVA Pressure Vessel Forward Closure Bolted Joint. The design analyzed incorporates 120 3/4 inch diameter bolts, a double flange seal, and a washer seal under each bolt head.

Publication of this report fulfills Project 620, Work Statement Item Number 9a.

II. SUMMARY/CONCLUSIONS

Appendices A through J present the detail stress analyses of the individual parts of the forward closure joint. Abstracting from the Appendices, the lowest margin of safety for each part results in the overall summary shown in Table I. In all cases, the loads, method of analyses, and assumptions are considered to be conservative.

Theoretical sealing is maintained by use of a nominal bolt preload of 37,000 lbs/bolt and all computed margins of safety are positive, based on the criteria of SNSO C-1, Ref. 1. Therefore, the analysis predicts a structurally adequate design for steady state operation.

Reliability analyses (outside the scope of this report) have been performed based on certain assumptions not entirely known to this author. However, ultimate verification of the closure joint reliability will be largely based on the ability to establish bolt preload accurately. Flange compression load variance will not exceed $\pm 25\%$. Some relief from this assumed variance can be achieved but how much is now unknown. It is felt that there is presently too little data on mean bolt load and variance on bolt load due to installation torque. It is therefore recommended that tests be conducted to provide this data. It is also recommended that tests be conducted to evaluate the retention, or percent retention, of the bolt load during "long time" periods of storage.

TABLE I
MINIMUM MARGINS OF SAFETY
P.V. FORWARD CLOSURE JOINT

P.V. Component	Material	Stress		M.S.	Reference	
		Magnitude (psi)	Mode & Condition		Appendix	Page
Cyl. Shell	7075-T73 AA	47,700	Hoop Tension 20.25" aft of joint during operation (T = -145°F)	+0.01	J	3
Cyl. Flange	7075-T73 AA	24,200	Thread shear due to max preload	+0.39	B	13
Insert	A-286	56,600	External thread shear due to max preload	+0.18	B	13
Bolt	A-286	135,000	Max principal stress due to tension & shear during installation	+0.00*	B&G	4&2
Clos. Flange	7075-T73 AA	73,000	Bearing under washer seal due to preload	+0.05	B	13
Clos. Shell	7075-T73 AA	36,650	Tension in shell inboard of bolt head counterbore due to operation	+0.58	I	3

*This margin of safety assumes +25% variance on preload. If this variance is controlled to within +3% and the nominal preload left at 37,000 lbs/bolt, a margin of safety of +0.25 would be predicted.

In support of reliability analyses, it is further recommended that either more theoretical work, or testing, be conducted to correlate flange compression load and variance of this load with bolt load for different bolt patterns, for flange load is the determining factor when checking against possible leakage. In the case considered here, 120 3/4 inch diameter bolts, theoretical work to date (Reference 2) indicates a much tighter variance on flange load than on bolt loads.

If it is established that +25% variance on flange preload is too conservative the bolt preload may be decreased and a higher margin of safety predicted.

There are other conditions, outside the scope of this report, which may be more critical. These are transient conditions, startup and shutdown, and space soak conditions. These conditions must be investigated in detail in the future.

III. TECHNICAL DISCUSSION

The analysis employs the use of an axisymmetric computerized finite element analysis which is reported in TM 23 by E. L. Wilson (Reference 3).

Geometry used was taken from Drawing No. 1138808, Revision B, and a 5:1 layout supplied by the Design Group. Minimum sizes were used throughout.

Material properties were taken from latest revision Data Release Memos when possible or Materials Data Book (Reference 4) when DRMs not available. Minimum strengths were used throughout while nominal values for moduli, coefficients of linear thermal expansion, and Poisson's Ratio were used. See Appendix A.

Initially the bolt head, bolt washer seal and insert were sized, in conjunction with the Design Group so that the full magnitude of the bolts ambient temperature tensile strength (more critical than operating temperature condition) would be developed. See Appendix B.

The structure was modeled for the finite element analysis as shown in Appendix B. The bolt is the only tension member which joins the closure

flange with the cylinder flange. Small elements of triangular shape are used to transmit compression loads which arise at the shear lip interface due to differential radial deflection and at the outboard faces of the mating flanges due to differential rotation.

Enough of the cylinder and closure shells were included in the model so that end conditions would not influence the flange deformations. It may appear that more than enough of the cylinder shell was included, but this was done because of varying thermal environment.

Steady state loads and thermal environments were taken from the latest available NETAP run (Reference 5) and thermal analyses supplied by the Thermophysics Section, (References 6, 7 and 8), respectively. See Appendix D.

Some parts of the forward closure joint are not continuous in the circumferential direction; e.g., bolts, inserts, bolt head washers, flange webs between bolts, etc. In order to model these parts for the axisymmetric finite element analysis a reduced modulus (very nearly zero) must be used in the circumferential direction while a slightly reduced modulus, one based on a ratio of axial stiffnesses (EA), is employed in the axial-radial plane. See Appendix E for details of method and actual values used.

The design is shown adequate since sealing is theoretically achieved without yielding any member of the joint parts. To prove sealing a pre-load was included on the bolt during the computer runs for the steady state operating conditions. Initially the magnitude of preload included was the theoretical minimum flange compression load for 120 - 3/4" diameter bolts, 4,777,951 lbs or the equivalent of approximately 39,800 lbs per bolt (Reference 2). This value of preload maintained theoretical flange compression during operation from approximately the bolt centerline outboard. It was felt that the theoretical spread or variance of flange compression load was too narrow.

$$[39,800 \leq P_{i_{\text{nominal}}} \leq 40,900]$$

It was therefore assumed that the flange compression load per bolt region would be equal to the bolt preload and that they could both vary about their nominal magnitudes by $\pm 25\%$. The maximum value was set at 46,280 lbs/bolt, the maximum allowable. Therefore the nominal value is:

$$\frac{46,280}{1.25} = 37,020 \text{ lbs/bolt, and the minimum is } .75(37,020) = 27,765 \text{ lbs/bolt.}$$

This minimum value was applied to the bolt and rerun. The results show flange compression is maintained from the outer edge of the insert outboard. This is felt to be sufficient to prove seal maintenance since both flange seals are outboard of the point of initiation of flange compression. Also this is achieved while pressure is assumed to be acting out to the outboard seal which theoretically increases the load which tends to promote leakage. The method of preload simulation is discussed in Appendix F while position of flange compression is shown in Appendix H.

The maximum magnitude of preload was then applied and operating conditions rerun. The resulting maximum bolt stress predicted was 130,000 psi. with the corresponding minimum margin of safety calculated to be +0.09. See Appendix G. It should be noted here that the ambient temperature maximum preload was set to give a zero margin of safety, therefore this condition is more critical for the bolt and also for the other bolt load path parts such as the inserts, cylinder flange threads, bolt washer seals and closure flange bearing. All these parts were checked and made less critical than the bolt as discussed before and shown in Appendix B.

Other flange stresses due to local shear load on shear lip and discontinuity loads at the flange to shell junctures are taken from the computer run results and shown in Appendix I.

A very conservative brief calculation of primary membrane stress in the midcylinder region is included in Appendix J. The results show that the cylinder is adequate structurally if a cooling efficiency factor of at least 1.2 is achieved. It should be stated here that from the thermal environment data to date the magnitude of the cooling efficiency factor which is actually achieved has no influence on the results of the closure

joint analysis. The minimum cylinder shell margin of safety is +.01 due to hoop stress 20.25 inches aft of the interface. This predicted margin is very conservative and independent of cooling efficiency factor, see Appendix J.

IV. LIST OF REFERENCES

1. NASA Specification SNSO-C-1, "NERVA Program Structural Design Requirements", December 1970
2. Memorandum, J. W. Carriker to W. I. Sanders, "Engineering Methods Development, Bolt Torque Final Report"
3. AGC Computer Program Ell401, "Finite Element Stress Analysis of Axisymmetric Bodies of Revolution and Plane Structures, Applicable to Compressible Solids with Non-Linear Material Properties", D. M. Campbell, 5 March 1970
4. ANSC Report 2275, NERVA Program Materials Properties Data Book, 30 March 1970
5. ANSC Computer Program NETAP ("NERVA Engine Transient Analysis Program) Printout for "State Point Conditions and Specification Extremes"
6. ANSC Engineering Operations Report N8110R:71-001, "Two-Dimensional Thermal Analyses of the 1138808B NERVA Pressure Vessel Annulus With Transverse Rib Flow Perturbators", W. R. Thompson, 30 June 1971
7. ANSC Engineering Operations Report N8110R:71-011, "Thermal Analysis of the Closure Joint of the 1138808B NERVA Pressure Vessel", W. R. Thompson, 31 August 1971
8. ANSC Engineering Operations Report N8110R:71-020, "Thermal Analysis of the Closure Shell of the 1138808B NERVA Pressure Vessel, D. T. Buchanan and W. R. Thompson, 29 October 1971

5 January 1972

AGCS-0800-11

REPORT NO.
N8120R:72-022

PAGE OF

DATE

WORK ORDER

BY

CHK. BY

DATE

APPENDIX A

MATERIAL PROPERTIES

TENSILE STRENGTH
Vs.
TEMPERATURE
FOR
7075-T73 AA FORGING
PER
DRM M-21 SUPP. #1 REV. #2 (12-18-70)

9-10-71

SUBJECT

DATE

8-30-70

WORK ORDER

BY

CHK. BY

DATE

SUBJECT

DATE

8-30-71

WORK ORDER

BY

CHK. BY

DATE

TENSILE STRENGTH OF A-286 BOLT MATERIAL

SINCE DRM gives
room temperature data only
strengths at other temperatures
are calculated by a direct
ratio of values for A-286
at temperature from Materials
Data Handbook

SUBJECT

DATE

8-30-71

WORK ORDER

BY

CHK. BY

DATE

MODULUS OF ELASTICITY
AND
COEFFICIENT OF LINEAR
THERMAL EXPANSION
FOR

A-286 MATERIAL

REF. MATERIALS
PROPERTIES DATA
BOOK

J. E. Jellison
5 January 1972

AGCS-0800-11

REPORT NO.
N8120R:72-022

PAGE OF

DATE

WORK ORDER

BY

CHK. BY

DATE

APPENDIX B

CRITICAL PART IN BOLT LOAD
PATH AT AMBIENT TEMPERATURE

SUBJECT

DATE

8-27-71

WORK ORDER

BY

CHK. BY

DATE

SKETCH OF BOLT INSERT & AI BOSS
SHOWING ENGAGEMENT LENGTHS

Useable engagement lengths are:

L_e (for bolt & internal thread of insert) = 0.621"

L_e (for external thread of insert & Boss) = 0.954"

Note: see pp. 3

SUBJECT

DATE

8-27-71

WORK ORDER

BY

CHK. BY

DATE

THREAD ENGAGEMENTSBolt to insert thread engagement

Bolt: EWB-0420-12-46 ← Note change from dwg.

Bolt length = 4.066 in. ← Ref SPS Bolt Catalog

Useful Engagement = Bolt length - seal thickness

- Boss thickness - Bolt end chamfer
- 2 threads - O'ore of insert
- below surface installation of insert

$$\therefore L_e = 4.066 - .10 - 3.000 - .09 - 2 \left(\frac{1}{16} \right) - .10 - .03$$

$$L_e = \underline{\underline{0.621 \text{ in.}}}$$

Insert to Boss thread engagement

KNH-1216 ← Note change from dwg.

Insert length = 1.12" ← Ref. "Keeksert" Catalog

$$L_e = 1.12" - 2 \text{ threads}$$

$$L_e = 1.12 - \frac{2}{12} = 1.12 - .166 = \underline{\underline{.954 \text{ in.}}}$$

SUBJECT

DATE

8-27-71

WORK ORDER

BY

CHK. BY

DATE

BOLT FWD CLOSURE JOINT

SPS, EWB 0420-12-46 A-286 mat'l.

Thread $3/4"-16$ UNJF-3ATensile Area 0.373 in.^2 ← Ref. HnBk H-28 $F_{TY \text{ min.}} = 159 \text{ Ksi. @ Rm Temp.}$ Ref. DRM 05.02BOLT TENSION:

$$\therefore \text{MAX. LOAD} = .85 F_{TY} A_{TH}$$

$$P_{\text{ALLOW.}} = .85 (159,000) (.373) = \underline{50,400 \text{ lbs.}}$$

For preload since combined with $\nearrow$ BOLT TENSION
 torsion $P_{\text{allow.}} = 46,280 \text{ lbs.}$
 shank dia. = .7480" to .7490"

$$\therefore \text{shank Area} = \frac{\pi}{4} (.7485)^2 = 0.44 \text{ in.}^2$$

$\therefore A_B = 0.44 \text{ in.}^2$ which is value to use
 for Bolt spring constant calc.

Note $A_B > A_{TH}$ $\therefore$ Thread Tension
 is more critical and above predicted
 load is correct bolt allowable for
 Tensile yield.

SUBJECT

DATE

8-27-71

WORK ORDER

BY

CHK. BY

DATE

BOLT THREAD SHEAR :

shear area of bolt thread:

$$A_s = \pi n L_e K_{n \max} \left[\frac{1}{2n} + .57735 (E_{s \min} - K_{n \max}) \right]$$

where: $n = 16$ = number of thread per inch $L_e = .621$ = length of thread engagement $K_{n \max} = .6908$ = max minor dia of internal thread (insert) $E_{s \min} = .7056$ = minimum pitch diameter of external thread (bolt)

$$A_s = \pi (16) (.621) (.6908) \left[\frac{1}{2(16)} + .57735 (.7056 - .6908) \right]$$

$$A_s = 0.857 \text{ in.}^2$$

$$P_{\text{ALLOW.}} = .8 F_{su} A_s$$

$$\text{Assume } F_{su} = .55 F_{tu} = .55 (182,000) = 100,000 \text{ psi.}$$

$$\therefore P_{\text{ALLOW.}} = .8 (100,000) (.857) = \underline{68,500 \text{ lbs.}}$$

BOLT THREAD SHEAR.

INSERT FWD. CLOSURE JOINT

KNH 1216

Internal thread Shear (3/4-16 UNF-3B)

$$A_s = \pi n L_e D_{s \min.} \left[\frac{1}{2n} + .57735 (D_{s \min.} - E_{n \max.}) \right]$$

$D_{s \min.} = .7406''$ = minimum major dia.
of external thread (bolt)

$E_{n \max.} = .7143''$ = max. pitch dia. of int.
thread (insert)

$$\therefore A_s = \pi (16)(.621)(.7406) \left[\frac{1}{2(16)} + .57735 (.7406 - .7143) \right]$$

$$A_s = 1.073 \text{ in.}^2$$

$$P_{\text{ALLOW.}} = .8 F_{su} A_s$$

Assume standard insert mat'l. of 303 CRES

& Assume $F_{su} = .55 F_{tu} = .55 (72,000) = 39,600 \text{ psi.}$

$$\therefore P_{\text{ALLOW.}} = .8 (39,600) (1.073) = 34,000 \text{ lbs.} \quad \text{Ref. mat'l.}$$

Since less than bolt tension load $H_n B_n$

call for A-286 insert mat'l. where $F_{tu} = 152 \text{ ksi.}$

again Assume $F_{su} = .55 F_{tu} = .55 (152,000) = 83,500 \text{ psi.}$

$$\therefore P_{\text{ALLOW.}} = .8 F_{su} A_s = .8 (83,500) (1.073) = 71,600 \text{ lbs.}$$

INTERNAL INSERT
THREAD SHEAR

SUBJECT

DATE

8-27-71

WORK ORDER

BY

CHK. BY

DATE

External thread Shear (insert) ($1\frac{1}{8}$ -12 UN 2A)

If A-286 mat'l. used o.k.
but check for 303 CRES, if ok. then
if A-286 insert becomes undesirable
a weight penalty may be paid by increasing
bolt length to get more engagement on
internal thread of insert. So,

$$A_s = \pi n L_e K_{n_{max}} \left[\frac{1}{2n} + 5.7735 (E_{s_{min}} - K_{n_{max}}) \right]$$

$$A_s = \pi (12)(.954)(1.053) \left[\frac{1}{2(12)} + 5.7735 (1.0631 - 1.053) \right]$$

$$A_s = 1.80 \text{ in.}^2$$

on external thread there are 4 kees
over $\frac{1}{4}$ of int. thread size. Therefore,
 A_s must be reduced due to loss of thread
surface.

$$\text{loss} = \left(\frac{1}{4} d_{\text{int.}} \right) \text{Number of kees} = \frac{1}{4} (.75)(4) = .75$$

$$\text{Useful fraction of } A_s = \frac{\text{Circ. ext.} - \text{loss}}{\text{Circ. ext.}}$$

$$= \frac{\pi (1.125) - .75}{\pi (1.125)} = \frac{3.535 - .75}{3.535} = \frac{2.785}{3.535} = .788$$

$$\therefore \text{Useful } A_s = 1.8(.788) = 1.42 \text{ in.}^2$$

$\therefore$ for 303 CRES $P_{\text{allow.}} = .8(39,600)(1.42) = 44,900$
this is still less than bolt tensile load

$\therefore$ Use A-286 insert

$$P_{\text{allow.}} = .8(83,500)(1.42) = 94,800 \text{ lbs.}$$

EXT. INSERT
THREAD SHEAR (See pp 9)

SUBJECT

DATE

8-27-71

WORK ORDER

BY

CHK. BY

DATE

ALUMINUM CYLINDER THREAD SHEAR

$$A_s = \pi n L_e D_{smin.} \left[\frac{1}{2n} + 1.57735 (D_{smin.} - E_{Amax.}) \right]$$

$$= \pi (12)(.954)(1.1118) \left[\frac{1}{2(12)} + 1.57735 (1.1118 - 1.0787) \right]$$

$$A_s = 2.43 \text{ in.}^2$$

$$\text{Usefull } A_s = .788 A_s = .788 (2.43) = 1.91 \text{ in.}^2$$

$$P_{\text{ALLOW.}} = .8 F_{su} A_s$$

F_{su} for 7075-T73 AA @ Rm Temp.
is calcd. as:

$$= \frac{F_{TU} \text{ (from DRM)}}{F_{TU} \text{ (from mat'l. HnBk.)}} \times F_{su \text{ mat'l. HnBk.}}$$

$$\therefore F_{su} = \frac{64.5}{66} \times 43 = 42 \text{ ksi.}$$

$$\therefore P_{\text{ALLOW.}} = .8 (42,000)(1.91) = \underline{64,000 \text{ lbs.}}$$

↗ BOSS SHEAR

SUBJECT

DATE

8-28-71

WORK ORDER

BY

CHK. BY

DATE

BOLT SPACING

$$S = \frac{\pi d_{e.c.}}{N} = \frac{\pi (56.25)}{120} = 1.4726"$$

Assume island of AA cyl. yields and will not transmit shear load over assume $\angle$ of 60° . Then insert external thread shear area is further decreased now useful fraction is:

$$\text{fraction} = \frac{360^\circ - 2(60^\circ)}{360^\circ} (\pi d_{ext.}^{1.125}) - 4(\frac{1}{4} d_{int.}^{.75})$$

$$= .667 - \frac{.75}{1.125\pi} = .667 - .212 = .455$$

assume KWS located outside island region

$$F_{net} = .455 A_s = .455 (1.8) = 0.818 \text{ in}^2$$

$$P_{allow.} = .8 F_{ult} F_{net} = .8 \left(\frac{54,600}{83,500} \right) (0.818) = \frac{25,900 \text{ lb} (200 \text{ crad})}{54,600 \text{ lb} (10 \cdot 216)}$$

With this load dist. the shear area of AA cyl. shown by red lines above is less critical than previously calculated. : OK

SUBJECT

DATE

8-28-71

WORK ORDER

BY

JF

CHK. BY

DATE

BOLT HEAD BEARING ON AA CLOSURE RING

$$A_{brg} = \frac{\pi}{4} \left[(d_{head} + .162)^2 - d_{seal \text{ I.D.}}^2 \right]$$

transfer through seal

$$= \frac{\pi}{4} \left[(1.190 + .162)^2 - (1.010)^2 \right]$$

$$= \frac{\pi}{4} \left[(1.352)^2 - (1.010)^2 \right] = 0.634 \text{ in.}^2$$

$$P_{ALLOW.} = .85 F_{BRY} A_{brg}$$

$$= .85 (90,000) (.634) = \underline{48,500 \text{ lbs.}}$$

BEARING ON AA CLOSURE RING →

See layout following.

LAYOUT OF BOLT, BOLT HEAD, SEAL
AND COUNTERBORE GEOMETRY

SUBJECT

DATE

8-28-71

WORK ORDER

BY

CHK. BY

DATE

SUMMARY:SUMMARY OF RESULTS OF BOLT
LOAD PATH FAILURE FOR AMBIENT
TEMPERATURE

FAILURE MECHANISM	MINIMUM PREDICTED FAILURE LOADS (lbs)	
Bolt Tension (Yield)	46,280	
Bolt Thread Shear	68,500	
Insert internal thread Shear	34,000	303 CRES
	71,600	A-286
Insert external thread Shear	25,900	303-CRES
	54,600	A-286
Cyl ring thread Shear	64,000	
Closing bearing (Yield)	48,500	

CONCLUSION:

USE of KNH 1216 T insert develops
full bolt tensile strength. ↑ A-286 mat'l.

SUBJECT

DATE
12-27-71

WORK ORDER

BY

JGJ

CHK. BY

DATE

Since ambient temperature preload conditions are critical and since the maximum preload has been set at the bolt allowable, (At least the assumption of $\pm 25\%$ variance on preload makes this the case) the following margins of safety will be calculated.

Bolt thread shear

$$M.S. = \frac{68,500}{46,280} - 1 = +0.48$$

Insert thread shear (external thread more critical than internal)

$$M.S. = \frac{54,600}{46,280} - 1 = +0.18$$

Cylinder thread shear

$$M.S. = \frac{64,000}{46,280} - 1 = +0.38$$

Closure Flange Bearing

$$M.S. = \frac{48,500}{46,280} - 1 = +0.05$$

J. E. Jellison

5 January 1972

AGCS-0800-11

REPORT NO.
N8120R:72-022

PAGE OF

DATE

WORK ORDER

DATE

BY

CHK. BY

APPENDIX C

FINITE ELEMENT MODEL

CHK. BY

DATE _____

BY 2

27.9.75
27.8.13
27.4.51
27.4.89

27.3.27
• 162.4p

27.165"R
(440)

(change p)

SUBJECT

REPORT NO.
N8120R:72-022

PAGE 6-2 OF 5

DATE
9-2-71

WORK ORDER

DATE

CHK. BY

BY

[Signature]

FINITE ELEMENT MODEL

DOUBLE SEAL CONCEPT

N8120R:72-022

Pa C-3 of 5

FINITE ELEMENT MODEL
DOUBLE SEAL CONCEPT
INCLUDING BOLT & INSERT

Q28

Pg. C-4 of 5

N8120R:72-022

J. E. Jellison
5 January 1972

AGCS-0800-11

REPORT NO
N8120R:72-022

PAGE OF

DATE

WORK ORDER

BY

CHK BY

DATE

APPENDIX D

MECHANICAL LOADING AND THERMAL ENVIRONMENT

SUBJECT

DATE

WORK ORDER

BY

CHK. BY

DATE

P.V. PRESSURE LOADING

SUBJECT

DATE
8-31-71

WORK ORDER

BY

CHK. BY

DATE

FINITE ELEMENT MODEL.

Model for mechanical loads

$$p' = \frac{P_{\text{gross}} \pi R_{\text{real}}^2 - \text{Thrust}}{2 \pi R_{\text{cyl}} t}$$

$$= \frac{1176 \pi (29.50)^2 - 73,000}{2 \pi (27.51)(.81)}$$

$$p' = 22,442.5$$

$$p'' = \frac{C a p}{t}$$

from structures manual
for 2:1 ellipse and

$$\frac{R}{a} = \frac{20.2}{27.57} = 0.734 :$$

$$C = 0.77$$

$$\therefore p'' = \frac{0.77 (27.57)(1176)}{0.81}$$

$$p'' = 30,800 \text{ psi.}$$

88

D-3 of 4

N8120R:72-022

88

822

N8120R:72-022

D-4 of 4

Flange & Forward Clos. Temperatures of

REF. REPORT No. N8110R:71-011
dtd 31 Aug, 1971

Case 7A.

J. E. Jellison
5 January 1972

AGCS-0800-11

REPORT NO.
N8120R:72-022

PAGE OF

DATE

WORK ORDER

DATE

SUBJECT

BY

CHK. BY

APPENDIX E

MATERIAL PROPERTIES FOR
FINITE ELEMENT AXISYMMETRIC
SIMULATION OF NON AXISYMMETRIC
PARTS

SUBJECT

DATE

10-13-71

WORK ORDER

BY

CHK. BY

DATE

SIMULATED STIFFNESS CALCULATIONS CYLINDER BOSS RING WEB STIFFNESS

$$S = \frac{2\pi R_{OC}}{N} = \frac{2\pi (28.125)}{120} = 1.4726 \text{ in.} \quad D = 1.10 \text{ in.}$$

$$t = D/3 = \frac{1.1}{3} = 0.367 \text{ in.}$$

$$E_{\textcircled{2} R-Z} = E_{AA} \left[\frac{D/3 (1.4726) - A_{\textcircled{2}}}{D/3 (1.4726)} \right]$$

$$A_{\textcircled{2}} = \frac{1}{2} \left(\frac{D}{2} \right)^2 [2\alpha - \sin 2\alpha]$$

$$\alpha = \cos^{-1} \frac{D/6}{D/2} = \cos^{-1} 0.333 = 70.54^\circ$$

$$2\alpha = 141.08^\circ = 141.08 \left(\frac{\pi}{180} \right) \text{ rad.} = 2.46 \text{ rad.}$$

DATE

10-13-71

WORK ORDER

BY

J. J.

CHK. BY

DATE

$$\sin 2\alpha \equiv \sin 38.92^\circ = 0.629$$

$$A_{\cup} = \frac{1}{2} \left(\frac{1.1}{2} \right)^2 [2.46 - 0.629]$$

$$A_{\cup} = 0.234 \text{ in}^2$$

$$E_{\textcircled{2} R-Z} = E_{AA} \left[1 - \frac{.234}{1.4726 (.367)} \right] = \underline{0.568 E_{AA}}$$

$$E_{\textcircled{3} R-Z} = E_{AA} \left[1 - \frac{A_{\square}}{1.4726 (.367)} \right]$$

$$A_{\square} = A_{\bigcirc} - 2 A_{\cup}$$

$$= \frac{\pi D^2}{4} - 2 (.234) = \frac{\pi (1.1)^2}{4} - .468 = .482 \text{ in}^2$$

$$E_{\textcircled{3} R-Z} = E_{AA} \left[1 - \frac{.482}{.541} \right] = \underline{0.11 E_{AA}}$$

SUBJECT

DATE
10-13-71

WORK ORDER

BY

JZJ

CHK. BY

DATE

BOLT STIFFNESS

make all three elements the same stiffness and make $D = D_{shank}$

$$E_{\textcircled{4} R-2} = \frac{\pi/4 D_s^2}{1.4726 D_s} E_{A-286}$$

$$= \frac{\pi (1.748)}{4(1.4726)} E_{A-286} = \underline{0.398 E_{A-286}}$$

SUBJECT

DATE

10-13-71

WORK ORDER

BY

CHK. BY

DATE

WASHER STIFFNESS

$$E_{(5) R-Z} = \frac{\frac{\pi}{4} [(1.352)^2 - (1.1)^2]}{(1.352 - 1.1) 1.4726} E_{A-286}$$

$$= \frac{0.487}{0.371} E_{A-286}$$

$$E_{(5) R-Z} = 1.31 E_{A-286}$$

DATE
10-13-71

WORK ORDER

BY

CHK. BY

DATE

INSERT STIFFNESS

$$E_{\textcircled{6} R-2} = \frac{\pi/4 [(1.1)^2 - (0.7)^2]}{(1.1 - 0.7)(1.4726)} E_{A-286}$$

$$= \frac{0.566}{0.589} E_{A-286} = 0.962 E_{A-286}$$

Since very nearly 1.0

Use $E_{\textcircled{6} R-2} = E_{A-286}$

SUBJECT

MAT'L PROPS FOR COMPUTER

REPORT NO.
N8120R:72-022

PAGE 5-6 of 6

DATE

WORK ORDER

BY

CHK. BY

DATE

ITEM No.	MAT'L.	TEMP. (°F)	$E_P - E_Z$ (psi $\times 10^6$)	ν_{R-Z}	E_0 (psi $\times 10^6$)	ν_0	$\alpha_R \& \alpha_Z$ (in/in $\times 10^{-5}$)	α_θ (in/in $\times 10^{-5}$)
1	7075 AL	-160	10.92	0.33	10.92	0.33	11.60	11.60
		-100	10.76	↑	10.76	↑	12.05	12.05
		-50	10.63	↑	10.63	↑	12.44	12.44
		0	10.50	↓	10.50	↓	12.81	12.81
		70	10.32	0.33	10.32	0.33	13.33	13.33
2	OVERLAY OFF	-160	6.20	0	0.001	0	11.60	0
	(.001)	-100	6.10	0	0.001	0	12.05	0
		70	5.84	0	0.001	0	13.33	0
3	OVERLAY ON	-160	1.20	0	0.001	0	11.60	0
	(.11)	-100	1.18	0	0.001	0	12.05	0
		70	1.14	0	0.001	0	13.33	0
4	BOLT (A-286)	-150	11.85	0	0.001	0	8.10	0
	(.11)	-100	11.75	0	0.001	0	8.30	0
		-50	11.70	0	0.001	0	8.51	0
		0	11.55	0	0.001	0	8.78	0
		70	11.40	0	0.001	0	9.10	0
5	WASHER	-150	30.05	0	0.001	0	8.10	0
	(A-286)	-100	33.70	0	0.001	0	8.30	0
	(.11)	70	37.65	0	0.001	0	9.10	0
6	INSERT	-150	29.78	0	0.001	0	8.10	0
	(A-286)	-100	29.50	0	0.001	0	8.30	0
	(.11)	70	28.67	0	0.001	0	9.10	0
7	△	-150	50.0	0	0.001	0	0	0
		70	50.0	0	0.001	0	0	0
8	FREEDOM SPRING	-150	11.85	0	0.001	0	10.0	0
		70	11.40	0	0.001	0	10.0	0
9	TENS. △	-150	0.001	0	0.001	0	0	0
		70	0.001	0	0.001	0	0	0

J. E. Jellison

5 January 1972

AGCS-0800-11

REPORT NO.
N8120R:72-022

PAGE OF

DATE

WORK ORDER

SUBJECT

BY

CHK. BY

DATE

APPENDIX F

ALLOWABLE PRELOAD AND
FINITE ELEMENT SIMULATION
OF PRELOAD

RESULTS OF BOLT - FLANGE PRELOAD
 CALCULATIONS FOR 120 - 3/4" dia. bolts
 using A-286 material with $F_{T \min.} = 159,000 \text{ psi.}$
 and $T_{\text{allow.}} = 135,000 \text{ psi.}$. Calculations
 consider teflon spray S-122 lubricant.
 Thread tensile area = 0.3777 in.^2

Single bolt preload due to
 recommended torque of 3770.9 in-lbs.

	<u>- 3 σ</u>	Mean	+ 3 σ
Torque	3544.7	3770.9	3997.2
$P_i \text{ bolt}$	34,431.7	40,357.2	46,282.7
$P_i \text{ flange}$ for 120 bolts.	4,777,951	4,842,864	4,907,772
$P_i \text{ flange/bolt}$	39,816	40,357.2	40,898

4842864
 avg.

120(40,357.2)
 = 4,842,864

Ref. Computer Analysis Results

Theory based on Memo J.W. Carriker to
 W.I. Sanders, "Engineering Methods Development,
 Bolt Torque Final Report"

SUBJECT

DATE

WORK ORDER

BY

CHK. BY

DATE

PRELOAD SIMULATION

The Total Flange Compression Load as determined by computer preload calculation for 120 - 3/4" dia bolts with $F_{allow.} = .85 F_{TY \min.} = .85(159,600) = 135,000 \text{ psi.}$ and teflon spray lubricant is:

$$\begin{aligned} +3\sigma &= 4,907,772 \text{ lbs.} \\ -3\sigma &= 4,777,951 \text{ lbs.} \end{aligned} \quad \left. \vphantom{\begin{aligned} +3\sigma &= 4,907,772 \text{ lbs.} \\ -3\sigma &= 4,777,951 \text{ lbs.} \end{aligned}} \right\} F$$

Therefore for modeling preload

$$P_{\text{bolt.}} = F / 120 \text{ (lbs/bolt)}$$

$$\therefore 39,816 \leq P_i \leq 40,898$$

Theoretical Bolt stretch due to preload

$$\Delta L_B = \frac{P_i L_B}{A_B E_B}$$

$$L_B = 3.0 + 0.08 + 0.13 = 3.21 \text{ in.}$$

$$A_B = 0.44 \text{ in.} \quad E_B = 28.67 (10)^6 \text{ psi.}$$

$$\Delta L_B = \frac{P_i (3.21)}{(0.44)(28.67)(10)^6}$$

$$\therefore .010,146'' \leq \Delta L_B \leq .010,421''$$

RELATED STRETCH OF BOLT BY
 USING LOCAL PIECE THERMALLY
 one line of nodal points and
 temperatures on two rows of elements
 being 0.25" long.

$$\Delta L_B = \alpha \Delta T l \equiv \Delta L_B$$

$$\Delta T_{req.} = \frac{\Delta L_B}{\alpha l}$$

SUBJECT

DATE

WORK ORDER

BY

CHK. BY

DATE

Let: $\alpha = 10 \times 10^{-6}$ and remain constant

$$l = 0.5''$$

$$\therefore \Delta T_{\text{req.}} = \frac{\Delta L_B}{10(10)^{-6}(0.5)} = 0.2 \Delta L_B (10)^6$$

$$\therefore -2084^\circ\text{F} \leq \Delta T_{\text{REQ.}} \leq -2029^\circ\text{F}$$

$$\& T_{\text{ELEM.}} = \Delta T + 70$$

$$\therefore -2014^\circ\text{F} \leq T_{\text{EL.}} \leq -1959^\circ\text{F}$$

$$T_{\text{N.P.}} = 2 T_{\text{EL.}}$$

$$\therefore -4028^\circ\text{F} \leq T_{\text{N.P.}} \leq -3918^\circ\text{F}$$

DATE

WORK ORDER

DATE

BY

CHK. BY

FOR PRELOAD ONLY RUN, REFERENCE
TEMPERATURE WILL BE 0°F . SINCE

$$E_B \text{ } 0^{\circ}\text{F} = 29.0 (10)^6 \text{ psi.}$$

$$\Delta L_B \text{ } 0^{\circ}\text{F} = \frac{28.67}{29.0} \Delta L_B \text{ } 70^{\circ}\text{F}$$

$$\therefore 1010,030'' \leq \Delta L_B \leq 1,019,303''$$

$$\& -2060.5^{\circ}\text{F} \leq \Delta T_{\text{REQ}} \leq -2006^{\circ}\text{F}$$

$$\text{SINCE } T_{\text{REF.}} = 0^{\circ}\text{F}$$

$$T_{\text{EL.}} = \Delta T$$

$$\& T_{\text{NP}} = 2 T_{\text{EL.}} = 2 \Delta T$$

$$\therefore -4121^{\circ}\text{F} \leq T_{\text{N.P.}} \leq -4012^{\circ}\text{F}$$

SUBJECT

DATE

WORK ORDER

BY

CHK. BY

DATE

CALCULATION OF BOLT STRESSES

Typical set of bolt elements

(F) Total load on bolts $= \sum (\sigma_z \text{ of element} \times \text{element Area})$

(P) Load per bolt $= F / \text{Number of bolts}$

(σ_b) Bolt stress $= P / \text{Area of bolt thread}$

$$\therefore F = \sum (\sigma_z \pi (R_o^2 - R_i^2))$$

$$= \pi \left[\sigma_{z_i} \left\{ (28.0)^2 - (27.751)^2 \right\} + \sigma_{z_c} \left\{ (28.25)^2 - (28)^2 \right\} + \sigma_{z_o} \left\{ (28.499)^2 - (28.25)^2 \right\} \right]$$

$$F = 43.612 \sigma_{z_i} + 44.179 \sigma_{z_c} + 44.392 \sigma_{z_o}$$

$$P = F / 120$$

$$\sigma_b = P / .3779$$

JH

F-7 of 9

BOLT PRELOAD SIMULATION RESULTS

Using $T_{AP} = -4121^{\circ}F$ to obtain $+30 F$
gave:

El. #	T_{AP}	R_o (inches)	R_i	$A = \pi(R_o^2 - R_i^2) = \frac{F}{\sigma}$	P_{BOL} $\frac{P}{EF/12}$
692	18,016	28.000	27.751	43.612	785,706
693	17,058	28.250	28.000	44.178	753,599
694	19,702	28.499	28.250	44.392	874,616
					2,413,921

note low
preload obtained
since calc. did not
include flange and
bolt interaction when
calculating ΔL_B

$$\frac{F_{REQ.}}{F_{OBTAINED}} = \frac{\Delta T_{REQ}}{\Delta T_{USED.}}$$

$$\therefore \Delta T_{REQ} = \Delta T_U \frac{F_R}{F_O}$$

$$\Delta T_{REQ} = -4121 \left(\frac{4,777,951}{2,413,921} \right) = -8156.83$$

$\therefore$ Use $\Delta T = -8157^{\circ}F$ to
obtain $-30 F$ of 4,777,951 lbs.

SUBJECT

DATE

WORK ORDER

BY

CHK. BY

DATE

BOLT PRELOAD SIMULATION RESULTS

$$\text{Using } T_{N.P.} = -8157^{\circ}\text{F}$$

$$F = 43.612 (35,665) + 44.179 (33,760)$$

$$\boxed{\text{El. 692} \rightarrow 694} + 44.392 (38,985) =$$

$$F = 1,555,422 + 1,491,483 + 1,730,622 = 4,777,527 \text{ lbs.}$$

$$F = 43.612 (40,383) + 44.179 (27,417)$$

$$\boxed{\text{El. 668} \rightarrow 670} + 44.392 (40,641) =$$

$$F = 1,761,183 + 1,211,256 + 1,804,135 = 4,776,575$$

$$\therefore F_{\text{OBTAINED}} \approx 4,777,000 \text{ O.K.}$$

$$P_i = \frac{4,777,000}{120} = 39,808 \text{ lbs/bolt.}$$

$$\sigma_{i \text{ min.}} = \frac{P_i}{.373} = \frac{39,808}{.373} = 107,000 \text{ psi.}$$

SUBJECT

DATE

WORK ORDER

BY

CHK. BY

DATE

J. E. Jellison
5 January 1972

AGCS-0800-11

REPORT NO.

N8120R:72-022

PAGE

OF

DATE

WORK ORDER

DATE

BY

CHK. BY

APPENDIX G

BOLT LOADS , STRESSES
AND MARGINS OF SAFETY

BOLT LOADS FOR OPERATING CONDITIONS

 $P_i + p + T$ DOUBLE SEALNOMINAL PRELOAD $\approx 39,800$ lbs/bolt

$$F = 43.612 (43,062) + 44.179 (29,419) \\ + 44.392 (41,600) =$$

EL. 665 $\rightarrow$ 670

$$= 1,875,020 + 1,299,702 + 1,846,707 = 5,024,429$$

$$F = 43.612 (42,947) + 44.179 (35,455) \\ + 44.392 (35,738) =$$

EL. 692 $\rightarrow$ 694

$$= 1,873,055 + 1,566,366 + 1,586,481 = 5,025,902$$

$$F_{TOTAL} = 5,025,140 \text{ lbs.}$$

$$P_B = \frac{5,025,140}{120} = 41,880 \text{ lbs/bolt}$$

This is the most probable bolt load
but for minimum M.S. calc. use $P_{i \text{ max.}}$ see
next page.

DATE
12-28-71

WORK ORDER

DATE

SUBJECT

BY

CHK. BY

BOLT LOAD CALCULATIONS

BASED ON MAXIMUM PRELOADED BOLT

$$\text{MAX. PRELOAD} = 46,280 \text{ lbs/bolt}$$

This gives M.S. on preload = 0 $\leftarrow$ P_{preload}

MAX. BOLT LOAD DUE TO OPERATION CONDITIONS

$$P_{B \text{ MAX.}} = 49,000 \text{ lbs./bolt.}$$

MAX. BOLT STRESS DUE TO OPER.

$$\sigma_{\text{max.}} = \frac{P_B}{A_B} = \frac{49,000}{.3779} = 130,000 \text{ psi.}$$

$$\text{M.S.} = \frac{.85 F_{TY}}{\sigma_{\text{max.}}} - 1 =$$

$$\text{A-286 BOLT } F_{TY \text{ min. @ ambient Temp.}} = 159,600 \text{ psi. Ref. DRM.}$$

$$\text{A-286 NORMAL } F_{TY @ 70^\circ\text{F}} = 84 \text{ ksi. } \& F_{TY @ -90^\circ\text{F}} = 88 \text{ ksi.}$$

$$\therefore F_{TY \text{ BOLT @ } -90^\circ\text{F}} = \frac{88}{84} (159,600) = 167,000 \text{ psi.}$$

$$\text{M.S.} = \frac{.85 (167,000)}{130,000} - 1 = +0.09 \leftarrow \text{Operation}$$

J. E. Jellison
5 January 1972

A-65-0800-11

REPORT NO
N8120R:72-022

PAGE OF

DATE

WORK ORDER

DATE

SUBJECT

BY

CHK. BY

APPENDIX H

FLANGE SEALING

J. F. Jellison
5 January 1972

N8120R:72-022

Page H-1 of 1

MAINTAINANCE OF FLANGE COMPRESSION FOR VARIOUS PRELOADS

J. E. Jellison
5 January 1972

AGCS-0800-11

REPORT NO.
N8120R:72-022

PAGE OF

DATE

WORK ORDER

DATE

BY

CHK. BY

APPENDIX I

FLANGE STRESSES

AND MARGINS OF SAFETY

DATE
12-28-71

WORK ORDER

BY

CHK. BY

DATE

CYLINDER FLANGE

MAX STRESS OTHER THAN THREAD SHEAR
WHICH IS CRITICAL FOR PRELOAD AND COVERED IN
APPENDIX B IS meridional tension

$\sigma = 22,300$ psi. in element 306
just inboard & aft of the aft end of
the insert

$$T = -151.5 \quad F_{TY} = 52,000 \text{ psi. Appendix A}$$

$$\therefore \text{M.S.} = \frac{.85(52,000)}{22,300} - 1 = +0.98$$

SUBJECT

DATE

12-28-71

WORK ORDER

BY

CHK. BY

DATE

CLOSURE FLANGE SHEAR LIP

MAX SHEAR OCCURS AT BASE
OF SHEAR LIP IN ELEMENT NO. 390

$$\tau_{max} = 12,400 \text{ psi}$$

$$T = -145^{\circ}\text{F} \quad \therefore F_{TY \text{ min. (ST)}} = 52,000 \text{ psi.} \quad \text{Appendix A}$$

$$M.S. = \frac{.85 (.55 F_{TY})}{\tau_{max}} - 1 = \frac{.85 (.55)(52,000)}{12,400} - 1 = +0.96$$

CLOSURE FLANGE UNDER BOLT HEAD
COMPRESSION:

$$\tau_{max} = -38,900 \text{ psi.} \quad T = -137.5^{\circ}\text{F}$$

$$\therefore F_{TY \text{ ST. min.}} = 51,800 \text{ psi.} \quad \text{assume!}$$

$$F_{cy} = F_{TY} \quad \text{then: } M.S. = \frac{.85 (51,800)}{38,900} - 1 = +0.13$$

For Ambient temp preload $\tau_{max} = -36,300 \text{ psi.}$

$$T = 70^{\circ} \quad F_{TY} = 47,800$$

$$\therefore M.S. = \frac{.85 (47,800)}{36,300} - 1 = +0.12$$

BENDING: SEE APPENDIX B

SUBJECT

DATE

12-28-71

WORK ORDER

BY

CHK. BY

DATE

CLOSURE SHELL AT FLANGE JUNCTURE

THE MAX STRESS IN SHELL AT
JOINT OCCURS AT I.D. INBOARD OF
BOLT HEAD COUNTER BORE REGION

MERIDIONAL STRESS IN ELEMENT NO. 574 IS:

$$\sigma = 36,650 \text{ psi.}$$

TEMP OF EL# 574 IS. -138°F

$\therefore F_{TY} = 58,300 \text{ psi.}$ from Appendix. A

$$\therefore \text{M.S.} = \frac{F_{TY}}{\sigma_{\max.}} - 1 = \frac{58,300}{36,650} - 1 = +0.58$$

J. E. Jellison
5 January 1972

AGCS-0800-11

REPORT NO
N8120R:72-022

PAGE OF

DATE

WORK ORDER

BY

CHK. BY

DATE

APPENDIX J

CYLINDER SHELL

STRESSES AND MARGINS OF SAFETY

SUBJECT

DATE
10/8/71

WORK ORDER

BY

CHK. BY

DATE

Mid cylinder Stresses and Margins of Safety

Hoop Stress

$$\sigma_H = \frac{p \cdot R}{t}$$

$$p_{max.} = 1245 \text{ psi.}$$

$$R_{max.} = \frac{52.320 + 1.010 + 0.022}{2} = 27.176 \text{ in.}$$

$$t_{min.} = 0.84 - 0.010 = 0.83 \text{ in.}$$

$$\sigma_{H,max.} = \frac{1245 (27.176)}{0.83} = 40,800 \text{ psi.}$$

$$T_{max.} = 100^\circ F \quad @ \text{ mid thickness}$$

$$\therefore F_{TY} = 51,600 \text{ psi.} \quad \leftarrow \text{Appendix A}$$

$$M.S. = \frac{0.85 F_{TY}}{\sigma_H} - 1 = \frac{0.85 (51,600)}{40,800} - 1 = +0.07$$

For M.S. vs. 1. Safety Factor

SUBJECT

DATE

10/8/71

WORK ORDER

BY

CHK. BY

DATE

MID CYLINDER MARGIN OF SAFETY AGAINST
PRIMARY HOOP STRESS YIELDING VERSUS COOLING
EFFICIENCY FACTORS

SUBJECT

DATE

12-28-71

WORK ORDER

DATE

CYLINDER SHELL STRESSES IN VICINITY OF

JOINT:

THE MAX HOOP STRESS OCCURS
AT 20.25" AFT OF THE INTERFACE
AT THE COLD POINT WHICH HAS WARMER
PARTS ON EITHER SIDE.

 $\sigma_H = 48,900 \text{ psi}$ from computer run

which uses $t_{min} = 0.81$ $t_{min} \text{ actually} = 0.83$

excluding ribs $\therefore \sigma_{H_{max}} = \frac{.81}{.83} (48,900) = 47,700 \text{ psi}$.

$T = -145^\circ \text{F}$ for element #53 $\therefore$ from Appendix A

$F_{T_{min}} = 56,800 \text{ psi}$.

$$\therefore \text{Min. M.S.} = \frac{.85 (56,800)}{47,700} - 1 = +0.01$$

this is very conservative however
it is fairly independent of coolant efficiency
factor.