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THERMAL ANALYSIS OF LOFT CONTAINMENT VESSEL
PENETRATIONS S-5E AND S-6A

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TECHNICAL REPORT
NUCLEAR ENGINEERING AND ANALYSIS BRANCH
APPLIED STRUCTURAL MECHANICS SECTION

SUBJECT		REPORT NO.
THERMAL ANALYSIS OF LOFT CONTAINMENT VESSEL PENETRATIONS S-5E & S-6A		A-83
AUTHOR K. G. Condie	APPROVAL C. A. Moore	DATE 4-21-70
ACTIVITY OR GROUP LOFT & PBF Thermal Hydraulics Group		

SUMMARY

Thermal analysis of the LOFT containment vessel penetration S-6A was conducted for steady-state operation and blowdown transients. Penetration S-6A is a 20-inch pipe through which passes a 10-inch pipe from the steam generator containing 600°F steam^[1, 2]. Recommendations for penetration S-5E were based on the analysis of penetration S-6A. Penetration S-5E is an 18-inch pipe through which pass a 4-inch pipe containing the steam generator feedwater at 430°F and a 0.5-inch pipe containing sample water at 600°F. The thermal analysis is necessary to determine the maximum temperature gradient in the containment vessel around the penetrations. The outside of the penetrations protrude into a room adjacent to the containment vessel which will be maintained at a minimum temperature of 50°F. The steady-state temperature inside the containment vessel was assumed to be 100°F. Table I of LOFT Containment Vessel Specification S-1 was used for inside transient temperatures.

CONCLUSION

The maximum temperature in the containment vessel adjacent to penetration S-6A will be significantly below 200°F; therefore, no auxiliary cooling will be required.

RECOMMENDED ACTION

The following recommendations are made concerning penetrations S-5E and S-6A:

- (1) Use the temperature gradients obtained from this analysis to calculate the thermal stresses in the area of penetrations S-5E and S-6A.
- (2) Completely fill the annular area (10.75-in. ID x 19.0-in. OD) between the 10-inch steam line and the 20-inch pipe in penetration S-6A with Owens-Corning "Kaylo" insulation^[3] or some other insulation which has an equivalent or lower thermal conductivity over the temperature range

150°F to 600°F.* The 3/4-inch thick urathane insulation now specified for the inside of penetration S-6A will be eliminated.

- (3) Completely fill the area around the two pipes inside the 18-inch penetration S-5E with blanket or other type insulation with a thermal conductivity at 400°F of 0.50 Btu/hr-ft²-°F/in. or lower. The 3/4-inch thick urathane insulation specified for the inside of penetration S-5E may be retained.

(*) The thermal conductivity of "Kaylo" insulation for various temperatures is:

<u>Temp (°F)</u>	<u>K (Btu/hr-ft²-°F/in.)</u>
200	.360
300	.405
400	.455
500	.50
600	.55

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THERMAL ANALYSIS OF LOFT CONTAINMENT VESSEL PENETRATIONS S-5E and S-6A

I. SCOPE OF INVESTIGATION

A steady state and transient thermal analysis of the 20-inch containment vessel penetration S-6A was conducted using the TOODEE computer code^[4] to determine the maximum thermal gradients in the containment vessel around the penetration. A maximum temperature of 200°F in the containment shell is believed to result in acceptable stress levels^[5]. Several steady state runs were made to determine the required thermal conductivity of the insulation necessary to maintain the temperature of the containment shell below 200°F. A transient run was then made using the value of the thermal conductivity obtained above. From the results of the analysis on penetration S-6A the required insulation for penetration S-5E was estimated. Figures 1 and 2 show penetrations S-5E and S-6A, respectively. Secondary coolant lines 1/2"-SC-37-B containing sample water at 600°F and 4"-SC-16-B containing feedwater at 430°F pass through penetration S-5E. Line 10"-SC-1-B containing 600°F steam from the steam generator passes through penetration S-6A.

The thermal model of penetration S-6A used in this analysis is shown in Figure 3.

II. PURPOSE OF INVESTIGATION

This analysis was requested by the LOFT Facility Branch^[5] to determine the temperature distributions around the LOFT containment vessel penetrations S-5E and S-6A. The temperature gradients are necessary to calculate the maximum thermal stresses in this area.

III. METHOD OF INVESTIGATION

The steady state and transient temperature distributions around penetration S-6A were determined by using the TOODEE computer code. Since penetration S-6A is located adjacent to several other penetrations^[6] and since axisymmetric cylindrical (R-Z) geometry was used, it was necessary to determine an equivalent circular area at the outside surface of the containment shell for the rejection of the heat. Figure 4 shows the location of penetration S-6A with respect to other penetrations close to it and the heat rejection area used for this analysis. The penetrations are mounted in a 2-inch thick steel plate which is in turn welded to the 1-3/16-inch thick containment shell. A circular area 54 inches in diameter was used for the 2-inch thick steel plate and a 20-ft diameter area was used for the 1-3/16-inch thick containment shell.

The use of these areas for heat rejection is conservative since the outside of the penetration extends into a 2-ft wide annular area around the containment building which extends about 22-ft below the center of the penetration and about 10-ft above it[7]. The penetrations adjacent to S-6A add no significant amount of heat to the shell.

The expansion joints on either end of the penetration were not included in this analysis. The material in the expansion joints is thin enough that essentially no heat flows axially along the joint to the containment shell.

The following material properties were used in the TOODEE code for this analysis:

Thermal Conductivity (Btu/hr-ft-°F)

Carbon Steel	$K = 36.6 - 0.01T$
Concrete	$K = 0.70$
Insulation	$K = 0.0151 \exp (.001052T)^{[3]}$

Volumetric Heat Capacity (Btu/ft³-°F)

Carbon Steel	$\rho c_p = 51.4 + 0.03124 (T-459.6)$
Concrete	$\rho c_p = 28.0$
Insulation	$\rho c_p = 0.25$

where T is in degrees Rankine.

The heat transfer coefficient on the outside of the containment shell was conservatively assumed to be 0.2 Btu/hr-ft²-°F. The heat transfer coefficients on the inside during steady state and transient were 0.69[8] and 200[9] Btu/hr-ft²-°F, respectively. The heat transfer coefficient between the steam and inside of pipe was assumed infinite. Table I shows the temperature inside the containment vessel as a function of time during blowdown as used in this analysis. Table I in this report is identical to Table I in the LOFT Containment Vessel Specification S-I[9].

IV. DISCUSSION OF RESULTS

Tables II and III show the temperature distributions in penetration S-6A and the surrounding area as obtained in this analysis for the steady state case, and for 1200 seconds after blowdown, respectively. At 1200 seconds after blowdown, the maximum temperature gradient exists in the 20-inch pipe. The temperature gradient in the containment shell shows no significant change from steady state during blowdown due to the large mass of concrete which effectively insulates it.

V. REFERENCES

1. Kaiser Engineers Drawing, LOFT 650-P-64, Revision 8 (July 10, 1969).
2. Pittsburgh-Des Moines Steel Company Drawing, 2-04-012 (July 18, 1967).
3. Sweet's Plant Engineering Catalog File, PE-1-INS-1/OW, Sweet's Industrial Information Systems, New York, New York (1968) pp 1-38.
4. John A. McClure, TOODEE, A Two-Dimensional Time Dependent Heat Conduction Program, IDO-17227 (April 1, 1967).
5. E. W. Kendall, General Work Authorization W-31205 (January 23, 1970).
6. Kaiser Engineers Drawing, LOFT 650-S-10, Revision 12 (April 2, 1969).
7. Kaiser Engineers Drawing, LOFT 650-S-29 Sheet 1, Revision 12 (November 11, 1969).
8. Letter, J. B. Fussell to C. W. Gay, "Temperature Variations in LOFT Containment Vessel Due to Pipe Inclusions", JBF-2-69 (August 28, 1969).
9. LOFT Containment Vessel Specification S-1, Revision 10 (August 6, 1964).

VI. ACKNOWLEDGEMENT

The author wishes to thank Mr. G. L. Ondricek for the preparation of the figures and tables for this report.

Fig. 3 THERMAL MODEL OF LOFT CONTAINMENT VESSEL PENETRATION S-6A

Fig. 4 Layout Showing Location of LOFT Containment Vessel Penetration S-6A
With Respect to Penetrations Adjacent to it & Heat Transfer Surface Area

TABLE I

LOFT CONTAINMENT VESSEL TEMPERATURES vs. TIME

TIME SEC	INSIDE TEMP DEG F
0.0	1.000000000 02
6.000000000 00	2.600000000 02
8.520000000 01	2.540000000 02
1.212000000 02	2.526000000 02
1.572000000 02	2.510000000 02
1.932000000 02	2.498000000 02
2.292000000 02	2.484000000 02
2.652000000 02	2.470000000 02
3.012000000 02	2.456000000 02
3.192000000 02	2.445000000 02
3.732000000 02	2.430000000 02
7.332000000 02	2.318000000 02
7.692000000 02	2.390000000 02
8.052000000 02	2.435000000 02
8.412000000 02	2.423000000 02
8.772000000 02	2.410000000 02
1.093200000 03	2.357000000 02
1.453200000 03	2.284000000 02
1.813200000 03	2.226000000 02
2.173200000 03	2.180000000 02
2.533200000 03	2.141000000 02
2.893200000 03	2.106000000 02
3.613200000 03	2.049000000 02
4.506000000 03	1.988000000 02
5.406000000 03	1.938000000 02
7.206000000 03	1.857000000 02
1.087800000 04	1.733000000 02
1.447800000 04	1.637000000 02
1.807800000 04	1.528000000 02
2.167800000 04	1.412000000 02
2.527800000 04	1.312000000 02
2.887800000 04	1.224000000 02
3.247800000 04	1.143000000 02
3.564600000 04	1.076000000 02

TABLE II

STEADY-STATE TEMPERATURE IN LOFT CONTAINMENT VESSEL PENETRATION S-6A

TEMPERATURE GRID IN DEGREES FAHRENHEIT
BLOCK BOUNDARY LOCATIONS INDICATED BY -X-
THE X OR R DIRECTION IS HORIZONTAL THE Y, Z, OR THETA DIRECTION IS VERTICAL

POINT LOC	1-1	2-2	3-0	4-0	5-3	6-4	7-0	8-5	9-0	10-0	11-0	12-0	13-6	14-0	15-0	16-0	17-0	18-0	19-0	20-0	21-0	22-0	23-7
19-0 600	600	600	530	378	211	131	130	130	50														
18-9 600	600	600	530	378	211	131	130	130	50														
17-0 600	600	600	530	378	210	130	130	130	50														
16-8 600	600	600	530	376	207	125	50	50	50														
15-0 600	600	600	529	373	201	119	50																
14-0 600	600	600	528	370	194	110	50																
13-0 600	600	600	527	367	188	101	50	50	50	50	50	50	50	50									
12-7 600	600	600	527	366	185	96	93	89	86	83	79	74	69	50	50	50	50	50	50	50	50	50	50
11-6 600	600	600	527	365	184	95	93	89	86	83	79	75	70	71	73	73	73	73	73	73	73	73	73
10-5 600	600	600	527	366	185	96	93	89	86	83	79	75	70	72	73	73	73	73	73	73	73	73	73
9-4 600	600	600	527	367	188	101	100	97	93	90	86	81	77	76	75	74	74	74	74	74	74	74	74
8-0 600	600	600	528	370	194	110	109	107	104	100	96	91	86	82	76	75	75	75	75	75	75	75	75
7-0 600	600	600	529	373	200	117	116	113	109	105	101	96	91	86	78	77	77	77	77	77	77	77	77
6-0 600	600	600	529	375	204	122	119	114	110	107	103	98	93	88	81	79	79	79	79	79	79	79	79
5-3 600	600	600	530	377	207	126	100	100	100	100	100	100	100	100	100	100	100	100	100	100	100	100	100
4-0 600	600	600	530	378	209	129	100	100	100														
3-2 600	600	600	530	378	211	131	131	131	100														
2-0 600	600	600	530	379	211	131	131	131	100														
1-1 600	600	600	530	379	211	131	131	131	100														

OUTSIDE OF CONTAINMENT VESSEL

INSIDE OF CONTAINMENT VESSEL

TABLE III

TEMPERATURE DISTRIBUTION IN LOFT CONTAINMENT VESSEL PENETRATION S-6A
1200 SEC AFTER BLOWDOWN

THE CURRENT TIME = 0.120000 04 SEC 2400 TIME STEPS HAVE BEEN CALCULATED

TEMPERATURE GRID IN DEGREES FAHRENHEIT
 BLOCK BOUNDARY LOCATIONS INDICATED BY -X-

THE X OR R DIRECTION IS HORIZONTAL

THE Y, Z, OR THETA DIRECTION IS VERTICAL

POINT LOC	1-1	2-2	3-0	4-0	5-3	6-4	7-0	8-5	9-0	10-0	11-0	12-0	13-6	14-0	15-0	16-0	17-0	18-0	19-0	20-0	21-0	22-0	23-7
19-0	600	600	600	530	378	210	129	129	129	50													
18-9	600	600	600	530	378	210	129	129	129	50													
17-0	600	600	600	530	377	209	129	129	129	50													
16-8	600	600	600	530	376	206	124	50	50	50													
15-0	600	600	600	529	373	200	118	50															
14-0	600	600	600	528	370	194	109	50															
13-0	600	600	600	527	367	187	101	50	50	50	50	50	50	50									
12-7	600	600	600	527	366	185	96	93	89	85	82	78	74	69	50	50	50	50	50	50	50	50	50
11-6	600	600	600	527	366	184	95	93	89	86	82	79	74	70	71	73	73	73	73	73	73	73	73
10-5	600	600	600	527	366	186	97	93	89	86	82	79	74	70	72	73	73	73	73	73	73	73	73
9-4	600	600	600	527	368	190	105	100	96	92	89	85	81	77	76	74	74	74	74	74	74	74	74
8-0	600	600	600	529	374	203	122	111	106	103	99	95	90	86	82	76	75	75	75	75	75	75	75
7-0	600	600	600	532	383	222	147	124	114	110	107	103	98	93	88	80	79	79	79	79	79	79	79
6-0	600	600	600	536	399	252	185	157	142	138	135	132	128	125	121	115	114	114	114	114	114	114	114
5-3	600	600	600	540	413	284	227	233	233	233	233	233	233	233	233	233	233	233	233	233	233	233	233
4-0	600	600	600	542	418	291	233	233	233	233													
3-2	600	600	600	542	418	292	235	235	235	233													
2-0	600	600	600	542	418	292	235	235	235	233													
1-1	600	600	600	542	418	292	235	235	235	233													

OUTSIDE OF CONTAINMENT VESSEL

INSIDE OF CONTAINMENT VESSEL

LOFT Report No. LOFT 1217-1
Technical Report No. 132 (A-97)
Date August 21, 1970
Work Request No. W-31209

ANALYSIS OF LOFT CONTAINMENT VESSEL PIPING PENETRATIONS

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TECHNICAL REPORT
NUCLEAR ENGINEERING AND ANALYSIS BRANCH
APPLIED STRUCTURAL MECHANICS SECTION

SUBJECT		REPORT NO.
ANALYSIS OF LOFT CONTAINMENT VESSEL PIPING PENETRATIONS		A-97
AUTHOR (See Cover)	APPROVAL C. A. Moore	DATE
ACTIVITY OR GROUP Applied Structural Mechanics Section		

SUMMARY

The flanges and seal plates of LOFT containment vessel penetrations Type I-A, II-A, Type III-A, III-B, II-B, IV-A, and IV-B were analyzed for stresses. Results were compared to ASME code allowable. Some of the conclusions reached are presented as "recommended action."

RECOMMENDED ACTION

The investigation presented herein for the subject nozzles led the authors to following recommendations:

1. The 150 lb ASA blind flanges and seal plates are adequate except for nozzle S-8A. For this flange, a 300 lb ASA blind flange and seal plate should be used.
2. Blind flanges and seal plates should have the same thickness.
3. Full penetration welds should be used on all flange/pipe and flange/nozzle joints.
4. The bolting material should be ASTM A-193 Gr. B7 or approved equivalent.
5. The nozzle loads must either be transferred across all bellows or transferred to pipe supports. One acceptable method is that shown in Reference 8. However, four more nuts than shown should be added to the inside (i.e. between bellows flanges) of each limit rod. Two of the nuts are lock nuts. The limit rod material should be per Reference 9. Finally, the design shown in reference 8 (or some satisfactory alternate method of transferring loads across bellows) should be applied to all remaining nozzles which have bellows.
6. The flange gaskets should be designed so that bolt stresses can be calculated. It appears that some of the bolts may have to be changed before the number of cycles listed in Reference 5 is reached (see Table IX).
7. More analysis is required. Specifically:
 - (a) A piping analysis should be conducted to determine nozzle loads. These loads should be checked against design loads - higher loads and/or more fatigue cycles indicate that reanalysis of the nozzles considered herein may be required.

- (b) After (a) is completed, the pipe stresses should be checked against allowable values. The stresses in some of the pipes were found to be over allowable values.
 - (c) After gaskets are chosen, the bolt stresses should be calculated to ascertain that they are within allowable values.
 - (d) The nozzles omitted from the present investigation should be checked when (a) is completed.
8. Lack of funds prevented additional calculations but it appears that thermal stresses in the following nozzles should be checked, S-5D, S-5E, S-7C, S-17B. It is recommended that this be done after 7(a) is accomplished.

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ANALYSIS OF LOFT CONTAINMENT VESSEL PIPING PENETRATIONS

I. SCOPE OF INVESTIGATION

The flanges and seal plates of LOFT containment vessel penetrations types I-A, II-A, III-A, III-B, II-B, IV-A, and IV-B were analyzed for stresses. The loads were induced by applied forces, pressures and transient temperatures. The temperatures were obtained by a separate analysis and are included in Appendix C of this report. Not all cases were investigated for thermal stresses but instead only certain cases were checked. The flange and bolt stresses were compared against ASME code allowables. A bolting analysis was conducted mainly to see if a bolting problem exists.

II. PURPOSE OF INVESTIGATION

The purpose of the present investigation is to size for thickness, flanges and seal plates on certain of the LOFT nozzles.

III. INTRODUCTION

The work statement of Reference 1 states:

"Analyze containment vessel penetrations types I-A, II-A, III-A, III-B, II-B, IV-A, and IV-B. The analysis will include the following:

1. Calculate discontinuity stresses (compare with Section III allowables).
2. Size weld and flange thickness."

Overall descriptions of the penetrations may be found in References 2 and 3.

The LOFT containment vessel penetrations have been analyzed by Dynatech and the results presented in the Dynatech 2011 Submittal (Reference 4). The static design loads of Reference 5 were employed. These analysis are valid for the case where the nozzles are terminated with blind flanges. Further analysis was conducted on the blind flanges by PDM in Reference 6. The LOFT vessel is an ASME Section III, Class B Vessel, but the nozzle configurations analyzed herein are not within the boundaries of code jurisdiction (see Reference 7, Section N-150). However, a "Section III-type" analysis was conducted. This means that the system considered to be a portion of a Class A vessel. The rules of Class A were employed primarily because of the relative ease of applying these rules as compared to those of Class B.

The loading on the nozzles consists of pipe-applied mechanical loads, thermal loads from pipes and from the containment environment, internal pressure in the pipes, and internal pressure in the containment vessel. Nozzle loads are given in Reference 5, but it is not known exactly where the values came from. It appears that they are estimates based upon past experience. Per the specifications of Paragraph S1-04 M4 of Reference 5, these loads were applied directly to the flanges. It was assumed that the loads in Reference 5 are the total loads and thus, include, for example, earthquake loads. The calculations presented herein are preliminary since the piping system has not yet been completely designed. In particular, pipe supports have not been located and flange gaskets have not been chosen. After the piping is designed, the response of the piping system to its loads may be determined. Then the nozzle loads may be obtained from the analysis. If these loads are less than or equal to the loads used herein, then the flanges considered herein are satisfactory. If the loads are greater, than additional checks and perhaps design will be required.

All configurations were not analyzed for thermal loads. Instead "worst-cases" were analyzed. To help decide which were the worst cases, some simple analysis were conducted - See Appendix A. Due to the many variables involved, i.e. nozzle sizes, loads and number and size of the penetrations, it is difficult to apply results from the nozzles analyzed to the other nozzles.

It should be noted that the bellows have not been analyzed, the gaskets have not been chosen and many nozzle configurations have not been considered - see Reference 2 for all the configurations. Certain of these configurations appear to have stresses about as high as the cases analyzed and thus should be checked. However, this effort is not included in the work statement. For the purposes of checking the flanges, it was assumed that the flanges carry all of the load. Somehow this load must be transferred to the nozzle. This may be accomplished by letting the bellows transfer the load or by letting limit rods transfer the load or by employing pipe supports. For nozzles S-5, S-6, and S-19, limit rods were employed - see Reference 8. However, four more nuts (than shown in Reference 8) should be added to the inside (i.e. between the bellows flanges) of each limit rod. The limit rods will then transfer bellows compressive loads. The rod material should conform to the requirements of Reference 9.

The nozzle analysis were accomplished by the combined effort of the authors. The work was divided as:

<u>Nozzle Type</u>	<u>Individual</u>
I-A, IV-B	R. M. Fors
II-B, III-B, IV-A	R. G. Rahl
II-A, III-A	P. I. Wilterdink

while S. Silverman provided overall coordination. The above general nozzle types break down into the following specific nozzles*:

I-A	S-5B, S-5C, S-5D, S-5E, S-6A
II-A	S-18A, S-19A
III-A	S-3F, S-3L
III-B	S-2A
II-B	S-1A, S-7A, S-7B, S-7C, S-8A, S-11C, S-17A, S-17B
IV-A	S-12A, S-13B, S-13C, S-13D
IV-B	S-11A, S-11B, S-11D

Many of the nozzle configurations drawings show fillet welds at joints which are considered to be a part of the real pressure boundary (see Reference 2). These welds are interpreted as being of Category C of Section N-1320 and therefore, all welds on the blind flanges, seal plates, and H & V ducting should be full penetration welds.

THE CODE

The analysis was guided by the code requirements for a Class A vessel (see Reference 7). The following approach was employed. Compensation requirements for the blind flanges and seal plates were verified to be met. These requirements are presented in Section N-450 of Reference 7. Satisfaction of the requirements assures satisfaction of the requirements of N-414.1, N-414.2, N-414.3. It was assumed that the requirements of

*Nozzles 18A and 7B have been deleted so that no analyses were conducted for these two. Also, subsequent to the present analyses, it was learned that nozzle S-1A was deleted.

N-414.4 are satisfied if the sum of the secondary bending stresses and thermal stresses is less than $1.5 S_m$. Section N-415.2 was followed to check satisfaction of the cyclic loading requirements. The procedure outlined in Section N-416 was followed to analyze the bolting for fatigue.

IV ANALYSIS OF NOZZLE S-1A (NOZZLE TYPE II-B)

A detailed analysis of nozzle S-1A follows. Results are simply presented in tabular form for the remaining nozzles.

LOADS

The general nozzle configuration may be seen in Figures 1 and 2. Three pipes penetrate the nozzle. The loads as obtained from Reference 5, Section S1-04, are as follows. Per paragraph S1-04 M 4, these loads are to be applied directly to the blind flanges.

TABLE I

LOADS FOR NOZZLE S-1A

Fatigue Cycles	F_x lb	F_y lb	F_z lb	M_x in-lb	M_y in-lb	M_z in-lb
500	1000	1000	1000	20,000	20,000	20,000

PRIMARY STRESS (COMPENSATION)

The design requirement of Section N-450 (Reference 7) will be followed in order to meet the compensation requirements for a Class A vessel. The flange thickness, t_r , required for no penetration, must be determined. The flange loads employed to calculate t_r are containment vessel internal pressure, force F_x , and moment M_x .

This thickness was determined by means of a computer solution for the stress intensity in the flange. The edge of the circular flange was assumed simply supported. The formulas employed are as follows:

<u>Loading</u>	<u>Reference or Formula</u>
Pressure (= 36 psi)	Ref. 10, p. 216, Case No. 2
F_x	Ref. 10, p. 216, Case No. 3
M_x	$\frac{M_x}{2\pi r_0^2 t} = \tau$
Where	r_0 = Average diameter of pipe which applies the load.
	t = Flange thickness
	τ = Torsional stress

The stress intensity was determined from (see N-411 of Reference 7)

$$SI = \sqrt{(\sigma_r - \sigma_t)^2 + 4\tau^2}$$

where σ_r and σ_t are the radial and tangential stresses in the flange. For ASTM A-105 Gr. II steel, which is called out in Reference 11 for all the flanges, the allowable stress is (Reference 12, Table UCS-23)

$$S_m = 17,500 \text{ psi}^*$$

The fact that the plate has three holes in it and that the mechanical and certain of the thermal loads arise from the pipes penetrating these holes pose a difficult analytical problem. It was decided to obtain the required flange thickness by using "equivalent" hole based upon the average pipe OD's and ID's. Thus,

$$\text{Equivalent Outside Area} = (\pi/4) OD_{\text{equiv}}^2 = (\pi/4) (8.625^2 + 2.375^2 + 3.5^2)$$

$$\text{Equivalent Inside Area} = (\pi/4) ID_{\text{equiv}}^2 = (\pi/4) (7.81^2 + 2.07^2 + 3.5^2)$$

$$\text{and } OD_{\text{equiv}} = 9.6"$$

$$ID_{\text{equiv}} = 8.6"$$

The required plate thickness was determined to within about $\pm 0.03"$ by means of the computer program (see formulas above). The pertinent data is

$a = 18"$, radius of plate = radius of flange bolt circle

$r_p = 16.5"$, radius over which pressure acts = radius of flange raised face.

$r_o = 4.8"$, outside radius of equivalent pipe

$$W = \pi r_p^2 p = 30,790 \text{ lb}$$

$$F_x = 1000 \text{ lb}$$

$$M_x = 20,000 \text{ in-lb}$$

With this data t_r was found to be $.77"$.

Compensation requirements will be checked using the single equivalent hole and using the actual holes. From Paragraph N-453 of Reference 7, the required cross-sectional area of compensation is given by

$$A_r = .5 dt_r$$

which becomes

$$A_r = .5 \times 8.6 \times .77 = 3.31 \text{ in}^2$$

for the equivalent hole. The available area is given by

$$A_a = ID_{\text{equiv}} X(t_f - t_r) = 11.65 \text{ in}^2$$

where t_f is the thickness of the flange. Thus, $A_a > A_r$.

Denoting A, B, C, and O as the midpoints of the 8", 3", 2", and 30" pipes, the distances between these points are:

$$\begin{array}{ll} \overline{OA} = 7.04" & \overline{AB} = 14.33" \\ \overline{OB} = 10.68" & \overline{BC} = 11.54" \\ \overline{OC} = 10.38" & \overline{CA} = 17.38" \end{array}$$

*For conservatism, this allowable stress was taken from Reference 12 instead of Reference 7 which lists $S_m = 21,300 \text{ psi}$.

The inside radius of the nozzle is 14.5". Since the following inequalities are satisfied, full compensation can be obtained for all holes. Essentiality these inequalities state mathematically that no area will be used twice and that the centers of all the holes are greater than one hole diameter from the edge of the pipe.

$$\begin{aligned} OA + 8.625/2 &< 14.5 \\ OB + 3.5/2 &< 14.5 \\ OC + 2.375/2 &< 14.5 \\ (8.625 + 3.5)/2 &< 14.33" \\ (3.5 + 2.375)/2 &< 11.54" \\ (2.375 + 8.625)/2 &< 17.38" \end{aligned}$$

Since compensation requirements are satisfied (the above inequalities are satisfied) so are the requirements of N-414.1, N-414.2, N-414-3*. The requirements of N-414.4 will be satisfied if the sum of the thermal stress and secondary stress is less than $1.5 S_m$.

SECONDARY STRESS

Secondary bending stresses are introduced by the moments M_y and M_z . The forces F_y and F_z are considered to introduce insignificant stresses in the flange and are therefore ignored†. The resultant M_{yz} is

$$M_{yz} = \sqrt{M_y^2 + M_z^2} = 28,289 \text{ in-lb}$$

The stresses in the plate are then found from Case 5, page 217 of Reference 10. Thus,

<u>Variable</u>	<u>Seal Plate</u>	<u>Flange</u>
a	15"	18.
r_o	4.8"	4.8
t_f	2.125"	2.125
σ_r	1730 psi	1690

where it has been assumed that M_{yz} is carried by first the seal plate and then the flange.

The thermal stresses for this nozzle were obtained by means of two computer codes. The first code is a two-dimensional, time-dependent heat conduction program TOODEE[13]. It was used to generate the transient temperature distribution in the flange/pipe system. The second code is the finite element stress analysis program entitled GAS. It was used to calculate the stress distribution at various instants of time (Reference 14).

The system was idealized as follows. The 8-inch low pressure steam line was judged to induce the largest thermal stresses - see Appendix A for the bases of this judgment. The other two pipes were ignored and the 8-inch line was "analytically" placed at the center flange. It was further assumed that the fluid temperature in the pipe changed as a step function from 50°F to 245°F, the temperature of 15 psig steam. The thermal distribution was determined out to time 500 sec. The distributions at 100 seconds, 300 seconds, and 500 seconds were then used as input in the stress analysis code. The internal pressure in the pipe was also included. The results and further discussion may be found in Appendix B.

*All the flanges do not satisfy Section N-415.1 of Reference 7 so that a fatigue analysis is required.

†These stresses should be calculated in any subsequent analysis.

FATIGUE OF FLANGES

Section N-415 of Reference 7 contains information on analysis for cyclic operation. Since the operation does not meet all the requirements of N-415.1, a fatigue analysis shall be made in accordance with N-415.2. For the nozzles considered herein, it is assumed that the principal stresses σ_1 and σ_2 are the same and equal to three times the material allowable. The reasoning for this is that the sum of the primary and secondary stresses satisfies the $3 S_m$ criterion. The third stress is assumed to be zero. Following the procedure of N-415.2, yields

$$S_{12} = 0$$

$$S_{23} = 3 S_m$$

$$S_{31} = -3 S_m$$

The absolute magnitude of the range is

$$S_{rij} = 3 S_m$$

and

$$S_{alt} = 1.5 S_m$$

Letting $S_a = S_{alt}$, the allowable number of cycles is

$$N = 3.5 \times 10^4 \text{ cycles}$$

Since all nozzles are to be subjected to 10,000 fatigue cycles or less, fatigue of the flanges is no problem.

BOLTING

Section N-416 of Reference 7 contains design requirements for bolting. However, specific gaskets have not been designed for the flanges. Therefore, the bolts cannot be analyzed according to the rules of Reference 7. The following analysis was conducted to obtain an idea if there may be a bolting problem. Consider the bolt circle to be a beam having a cross section consisting of the bolt cross sections. Then the moment of inertia is:

$$I = \left(\frac{n}{4\pi}\right) A_t^2 + A_t \left(\frac{BC}{2}\right)^2 \left[\sin^2 \left(\frac{360^\circ}{n}\right) + \sin^2 \left(\frac{2 \times 360}{n}\right) + \sin^2 \left(\frac{3 \times 360}{n}\right) + \dots \right]$$

Which can be reduced to

$$I = \left(\frac{n}{4\pi}\right) A_t^2 + \left(\frac{n}{8}\right) (A_t) (BC)^2$$

In this formula

A_t = Tensile area of single bolt, in².

n = Number of bolts

BC = Diameter of the bolt circle, in.

The bolt loads are comprised of internal pressure, stress resultants F_x , M_x , F_{yz} , and M_{yz} , and bolt preload. In addition, fatigue must be considered. The stresses in the bolts are given by the following equations:

$$\sigma_t = \frac{F_x + W}{nA_t} + \frac{M_{yz} (BC)}{2I} \quad (\text{Tension})$$

$$\tau_l = \frac{F_{yz}}{nA_t} \quad (\text{lateral shear})$$

$$\tau_2 = \frac{2 M_x}{n A_t (BC)} \quad (\text{Torsional Shear})$$

where

$$W = \pi r_p^2 p$$

For this nozzle, $n = 28$, $BC = 36"$, and $A_t = .969 \text{ in}^2$ (1-1/4" bolts used. See, for example, Reference 15 for the dimensions of the flanges). For the loading given in a previous paragraph, and $p = 40 \text{ psi}$ (this slightly conservative value was used instead of 36 psi) the stresses are

$$\sigma_t = 1290 \text{ psi}$$

$$\tau_1 + \tau_2 = 93 \text{ psi}$$

The preload P must also be considered. From Reference 16, the torque T necessary to obtain the preload is given by $T = .2 D_n P$ where D_n is the nominal diameter. The preload necessary to resist the forces tending to open the joint is estimated as

$$P = 2 \times \sigma_t \times A_t$$

where the factor of two is a safety factor against leakage. Thus, $T = .4 D_n \sigma_t A_t$ and for the 1-1/4" bolts, $T = 52 \text{ ft-lb}$.

This is a reasonable value and will assure that no leakage will occur. With this preload the stresses and stress intensity become:

$$\sigma_t = 3 \times 1290 = 3870 \text{ psi}$$

$$\tau = 93 \text{ psi}$$

$$SI = 3880 \text{ psi}$$

The intensity is well below bolt allowables so that with the possible exception of fatigue no problems should be encountered with excessive stresses in these bolts. For ASTM A-193 Gr. B7 the allowable stress is 31,400 for bolts less than 2-1/2 inches in diameter and less than 300°F.

The forces on the gasket per inch of circumference induced by the preload is

$$G = \frac{P_n}{\pi (BC)} = 618 \text{ lb/in.}$$

For an ethylene propylene gasket (Parker Seal Co.), the approximate sealing force is 150 lb/in. so that gasket seating is assured.

FATIGUE OF BOLTING

The fatigue analysis of the bolts is based on Section N-416.2 of Reference 7. Since the true stress in the bolts cannot yet be determined (gasket design is not yet complete), the analysis will be based upon the bolt stresses previously found. Following the rules of Reference 7,

$$S_{alt} = (1/2)(10,400) = 5200 \text{ psi}$$

$$S_{mean} = (4) \times 2600 = 10,400 \text{ psi}$$

$$S_{alt} + S_{mean} = 15,600$$

where it has been assumed that the stress cycles as shown in Figure 1*. Since 15,600 psi is less than the yield stress, $S_{\text{mean}} = 10,400$ psi.

The equivalent stress is

$$S_{\text{eq}} = \frac{5200}{1 - \frac{10,400}{125,000}} = 5670 \text{ psi}$$

The allowable number of cycles is given by

$$N = \left[\frac{10^6 + .08 \times 10^6}{5.67 \times 10^3 - 25 \times 10^3} \right]^2 = 3130 \text{ cycles}$$

This is greater than the 500 cycles stated in Reference 5 so that bolt fatigue is no problem with this nozzle.

V. RESULTS FOR REMAINING NOZZLES

The remaining nozzles were handled in the same manner as S-1A. The results are presented in the following tables. Data on nozzle S-1A is included for comparative purposes.

* The value 1300 psi was used instead of 1290 psi to calculate S_{alt} .

TABLE II

LOADS (After Reference 2)

Nozzle	Fatigue Cycles	F _x , lb	F _{yz} , lb	M _x , in-lb	M _{yz} , in-lb	Nozzle Type
S-1A	500	1,000.	1,414.	20,000.	28,300.	II-B
S-7A	500	1,000.	1,414.	20,000.	28,300.	II-B
S-7C	2000	2,000.	1,414.	20,000.	28,300.	II-B
S-8A	500	1,000.	1,414.	20,000.	28,300.	II-B
S-11C	10,000	5,000.	707.	8,000.	11,300.	II-B
S-17A	500	1,000.	1,414.	20,000	28,300	II-B
S-17B	500	1,000	1,414.	20,000	28,300.	II-B
S-2A	1,000	2,000	1,414	20,000	28,300	III-B
S-12A	10,000	+62,500	-2,000.	-----	-70,000.	IV-A
S-13B	10,000	+40,000	-15,000.	-----	-40,000.	IV-A
S-13C	10,000	+51,000	-4,000.	-----	-80,000.	IV-A
S-13D	10,000	+21,000.	-3,000.	-----	-70,000	IV-A
S-5B	500	+3,000.	+2,240.	+20,000.	-31,600.	I-A
S-5C	5,000	+5,000.	+3,600.	+30,000.	+58,300.	I-A
S-5D	5,000	+2,000.	+4,470.	-----	+5,000.	I-A
S-5E	5,000.	+2,000.	+4,470.	-----	+5,000.	I-A
S-6A	5,000.	2,000.	-4,000.	-----	4,000.	I-A
S-11A	10,000.	5,000.	707.	8,000.	11,300.	IV-B
S-11B	10,000	5,000.	707.	8,000.	11,300.	IV-B
S-11D	10,000	5,000.	707.	8,000.	11,300.	IV-B
S-19A	500.	1,000.	1,414.	20,000.	28,300.	II-A
S-3F	-----	-----	-----	-----	-----	III-A
S-3L	-----	-----	-----	-----	-----	III-A

TABLE III

BLIND FLANGE REQUIRED THICKNESS CALCULATIONS

<u>Nozzle</u>	a, in.	r _p , in.	r _o , in.	t _r , in.
S-1A	18.	16.5	4.8	.77
S-7A	8.5	7.5	4.31	.38
S-7C	5.875	5.313	1.75	.36
S-8A	4.75	4.25	1.1875	.31
S-11C	8.5	7.5	.93	.61
S-17A	8.5	7.5	2.25	.36
S-17B	8.5	7.5	2.25	.36
S-2A	14.75	13.625	9.79	.66
S-12A	No required thicknesses needed since no penetrations through these nozzles.			
S-13B				
S-13C				
S-13D				
S-5B	10.625	9.25	3.03	.55
S-5C	10.625	9.25	3.09	.62
S-5D	10.625	9.25	1.436	.56
S-5E	11.375	10.5	2.25	.58
S-6A	12.500	11.5	5.375	.58
S-11A	No required thicknesses needed since no penetrations through these nozzles.			
S-11B				
S-11D				
S-19A	8.5	7.5	3.312	.413
S-3F	No required thickness needed since no appreciable loads are exerted by the penetrations.			
S-3L				

TABLE IV

AREA REINFORCEMENT CALCULATIONS

Nozzle [1]	CD _{equiv} = 2 r _o in.	ID _{equiv} in.	t _f , in. [3] Flange Thickness	t _r , in. [2] Required Thickness	Area, in. ² 5 x ID _{equiv} x t _r	Area Available [4] ID _{equiv} (t _f - t _r), in. ²	Area Available [5]
S-1A	9.6	8.6	2.125	.77	3.31	11.65	17.55
S-7A	8.625	7.81	1.25	.38	1.48	4.30	4.30
S-7C	3.50	3.07	1.125	.36	.55	2.35	1.28
S-8A	2.375	2.07	1.0	.31	.32	1.43	1.43
S-11C	1.86	1.47	1.25	.61	.45	.94	1.34
S-17A	4.50	4.03	1.25	.36	.725	3.59	3.59
S-17B	4.50	4.03	1.25	.36	.725	3.59	3.59
S-2A	9.79	8.79	1.875	.66	2.90	10.68	5.20
S-12A	No area reinforcement calculations required since no penetrations through these nozzles.						
S-13B							
S-13C							
S-13D							
S-5B	7.86	7.1	1.438	.55	1.95	6.3	2.22
S-5C	6.18	5.46	1.438	.62	1.69	4.76	1.69
S-5D	2.87	2.40	1.438	.56	.671	2.11	.71
S-5E	4.50	4.07	1.563	.58	1.18	4.00	4.00
S-6A	10.75	9.56	1.688	.58	2.77	11.3	11.3
S-11A	No area reinforcement calculations required since no penetrations through these nozzles.						
S-11B							
S-11D							
S-19A	6.625	6.065	1.250	.413	1.25	5.07	
S-3F	No area reinforcement calculations required since no appreciable loads exerted by the penetration.						
S-3L							

NOTES: [1] See Reference 2 for details of nozzles and penetrations.

[2] Obtained from computer code for the plate subjected to the loading described in the previous section.

[3] See Reference 2 for nozzle size and Reference 15 for flange dimensions.

[4] Based upon single equivalent hole.

[5] Smallest area based upon actual holes and dimensions. Overlapping and location of penetration with respect to the nozzle wall were considered.

TABLE V

SECONDARY STRESSES IN FLANGES DUE TO FORCES [1,2]

Nozzle	Inside Vessel				Outside Vessel ^[3]			
	a, in.	r _o , in.	t _f , in.	σ _r , psi	a	r _o	t _f	σ _r , psi
S-1A	18.	4.8	2.125	1,690.	15.	4.8	2.125	1,730.
S-7A	8.5	4.31	1.25	5,640.	6.375	4.31	1.25	4,950.
S-7C	5.875	1.75	1.125	16,800.	4.3125	1.75	1.125	17,300.
S-8A	4.75	1.188	1.00	30,530.	3.3125	1.188	1.00	32,000.
S-8A [4]	5.313	1.188	1.4375	14,512.	3.3125	1.188	1.4375	15,481.
S-11C	8.5	.93	1.25	8,800.	6.375	.93	1.25	9,160.
S-17A	8.5	2.25	1.25	10,400.	6.375	2.25	1.25	10,790.
S-17B	8.5	2.25	1.25	10,400.	6.375	2.25	1.25	10,790.
S-2A	14.75	9.79	1.875	987.	12.0	9.79	1.875	657.
S-12A	Heating and Ventilation Ducting See Table VI for Stresses.							
S-13B								
S-13C								
S-13D								
S-5B	10.625	3.93	1.4375	5,255.	10.625	3.93	1.4375	5,255.
S-5C	10.625	3.09	1.4375	11,976.	10.625	3.09	1.4375	11,976.
S-5D	10.525	1.43	1.4375	1,970.	10.625	1.435	1.4375	1,970.
S-5E	11.375	2.25	1.5625	1,125.	11.375	2.25	1.5625	1,125.
S-6A	12.500	5.37	1.6875	355.	12.500	5.375	1.6875	355.
S-11A	Heating and Ventilation Ducting. See Table VI for Stresses.							
S-11B								
S-11D								
S-19A	8.5	3.31	1.25	7,394.	8.5	3.31	1.25	7,394.
S-3F	No stresses in flanges since		1.875				1.875	
S-3L			1.875				1.875	
there are no loads.								

[1] Stresses obtained from Case 5, Page 217 of Reference 10, the forces are M_y and M_z .

[2] See References 2 and 15 for dimensional data.

[3] The thicknesses of the seal plates could not be obtained and therefore it was assumed that they are the same thickness as the flanges.

[4] Since the stresses in the 150 lb flange were rather high, the calculations were repeated for a standard 300 lb flange.

TABLE VI
STRESSES IN HEATING & VENTILATION DUCTING^[1]

Pipe	S-12A	Nozzle ^[4]				
		S-13B S-13C [3]	S-13D	S-11A	S-11B	S-11D
OD, in.	36.		24.	12.75	12.75	12.75
Wall, in.	.375		.375	.406	.406	.406
I, in. ⁴	3382.		994.	.566	596.	596.
A, in. ²	21.		14.	15.6	15.6	15.6
σ_x , psi	2960.		1500.	320.	320.	320.
τ , psi	100.		214.	45.3	45.3	45.3
σ_b , psi	370.		845.	121.	121.	121.
SI, psi	3410.		2384.	450.	450.	450.

NOTES: [1] in this table:

$$I = \frac{\pi}{64} (OD^4 - ID^4) \quad \text{Moment of inertia of the pipe}$$

$$A = \frac{\pi}{4} (OD^2 - ID^2) \quad \text{Cross-sectional area of the pipe}$$

$$\sigma_x = \frac{F_x}{A} \quad \text{Axial stress in pipe}$$

$$\tau = \frac{F_{yz}}{A} \quad \text{Average shear stress in pipe}$$

$$\sigma_b = \frac{M_{yz} (OD/2)}{I} \quad \text{Maximum bending stress in pipe}$$

[2] Ducting satisfies the requirements of Reference 17.

[3] By comparing the loads for this nozzle with those of S-12A, it is apparent that the stresses for this nozzle are much less than allowable.

[4] Stresses are zero in nozzles S-3F and S-3L since loads are negligible.

VI. DISCUSSION OF RESULTS

The primary results are contained in Tables VII, VIII, IX, B-1 and B-2. From these results and from discussion contained in previous sections, the following conclusions are drawn:

1. The number of allowable fatigue cycles for any flange is 35,000. Since this is greater than the largest design number of 10,000 cycles, fatigue of the flanges poses no problem.
2. The 150 lb. ASA blind flanges and seal plates are adequate except for nozzle S-8A. For this flange, a 300 lb. ASA blind flange and seal plate should be used.
3. Blind flanges and seal plates should have the same thickness.
4. Full penetration welds should be used on all flange/pipe and flange/nozzle joints.
5. The bolting material should be ASTM A-193 Gr. B7 or approved equivalent.
6. The nozzle loads must be either transferred across all bellows or transferred to pipe supports. The design shown in Reference 8 or some satisfactory alternate method of transferring loads across bellows should with one exception, be applied to all remaining nozzles which have bellows. The exception arises when the bellows itself is capable of transferring the load. However, four more nuts than shown should be added to the inside (i.e. between bellows flanges) of each limit rod. Two of the nuts are lock nuts. The limit rod material should be per Reference 9.
7. The flange gaskets should be designed so that bolt stress can be calculated. It appears that certain of the bolts may have to be changed before the number of cycles listed in Reference 5 is reached (see Table IX).
8. More analysis is required. Specifically:
 - (a) A piping analysis should be conducted to determine nozzle loads. These loads should be checked against design loads - higher loads and/or fatigue cycles indicate that re-analysis of the nozzles considered herein may be required.
 - (b) After (a) is completed, the pipe stresses should be checked against allowable values. The stress in certain of the pipes were found to be over allowable values.
 - (c) After gaskets are chosen, the bolt stresses should be calculated to ascertain that they are within allowable values.
 - (d) The nozzles omitted from the present investigation should be checked when (a) is completed.
9. Lack of funds prevented additional calculation but it appears that thermal stresses in the following nozzles should be checked: S-5D, S-5E, S-7C, S-8A, S-17B. It is recommended that this be done after 8 (a) is accomplished.

TABLE VII

SECONDARY STRESSES IN FLANGES, Q

Nozzle	σ_r , psi ^[2]		Max. Thermal Stress Intensity, psi ^[3]	Approximate Max. Stress Intensity, psi ^[4]
	Inside	Outside		
S-1A	1,690	1,730	22,900	24,630.
S-7A	5,640	4,950	Pipe temp. 150°F	
S-7C	16,800	17,300	Pipe temp. 190°F	
S-8A [1]	14,512	15,481	Pipe temp. 150°F	
S-11C	8,800	9,160		
S-17A	10,400	10,790	Pipe temp. 100°F	
S-17B	10,400	10,790	Pipe temp. 190°F	
S-2A	987	-----	Pipe temp. ambient	40,215
S-12A	3,410	-----	Pipe temp. 160°F	
S-13B	-----	-----	Pipe temp. 160°F	
S-13C	-----	-----	Pipe temp. 160°F	
S-13D	2,384	-----	Pipe temp. 160°F	
S-5B	5,255	5,255	Pipe temp. ambient	
S-5C	11,976	11,976	Pipe temp. ambient	
S-5D	1,970	1,970	Pipe temp. 350°F	22,308
S-5E	1,125	1,125	Pipe temp. 600°F	
S-6A	355	355	39,860	
S-11A	450	450	Pipe temp. ambient	
S-11B	450	450	Pipe temp. ambient	
S-11D	450	450	Pipe temp. 160°F	
S-19A	7,394	7,394	14,914	
S-3F	-----	-----	Pipe temp. ambient	
S-3L	-----	-----	Pipe temp. ambient	

- NOTES: [1] Data for 300 lb flange
 [2] See Tables V and VI
 [3] Stresses from Appendix B; Temperature notes from Appendix A.
 [4] Obtained by adding σ_r and the maximum thermal stress intensity.

TABLE VIII

BOLT STRESSES, $p = 36$ psi

Nozzle	Nominal Bolt Dia., in. D	Bolt Area, in ² A _t	Number of Bolts, N	Diameter of Bolt Circle in. BC	NO PRELOAD			WITH PRELOAD			Gasket Force lb/in.
					σ_t psi	τ psi	Torque T, ft-lb	σ_t psi	τ psi	SI psi	
S-1A	1.25	.969	28	36	1290.	93.	52.	3,870.	93.	3,920	618.
S-7A	.875	.462	12	17	2530.	680.	34.	7,590.	680.	7,680.	525.
S-7C	.750	.334	8	11.75	5540.	1800.	46.	16,620.	1800.	17,000	800.
S-8A	.750	.334	8	9.50	5580.	2105.	47.	16,740.	2105.	17,300	1000.
S-11C	.875	.462	12	17.	2530.	300.	34.	7,590.	300.	7,610.	525.
S-17A	.875	.462	12	17.	2530.	680.	34.	7,590.	680.	7,710.	525.
S-17B	.875	.462	12	17.	2530.	680.	34.	7,590.	680.	7,710.	525.
S-2A											
S-12A											
S-13B											
S-13C											
S-13D											
S-5B	1.0	.606	16	21.25	1920.	424.8	38.8	5,760.	424.8	5,810	557.7
S-5C	1.0	.606	16	21.25	2644.7	663.1	53.4	7,940.	663.1	8,050.	768.
S-5D	1.0	.606	16	21.25	1301.3	461.	26.3	3,900.	461.	4,000.	378.
S-5E	1.125	.763	16	22.75	1257.1	366.	24.4	3,770.	366.	3,840.	429.
S-6A	1.125	.763	20	25.	1153.	262.	29.3	3,460.	262.	3,500.	448.
S-11A											
S-11B											
S-11D											
S-19A	.875	.462	12	17.	2526.8	679.5	38.9	7,580.	679.5	7,700.	504.
S-3F	1.25	.969	20	29.5	*						
S-3L	1.25	.969	20	29.5	*						

*Use sufficient preload to obtain a gasket force greater than that required to seal the gasket.

TABLE IX

FATIGUE ANALYSIS OF BOLTS

Nozzle	S _{alt} psi	S' _{mean} psi	S _{mean} psi	S _{eq} psi	Allowable Cycles	Required Cycles
S-1A	5,200.	10,400.	10,400.	5,670.	3,130.	500
S-7A	10,100.	20,200	20,200	12,060.	6,950	500
S-7C	22,400.	44,800	44,800	35,000.	11,700.	2,000
S-8A	22,300.	44,600	44,600	35,000.	11,700.	500
S-11C	10,100	20,200	20,200	12,050.	6,950.	10,000[1]
S-17A	10,100	20,200	20,200	12,050.	6,950.	500
S-17B	10,100	20,200	20,200	12,050.	6,950.	500
S-2A						
S-12A						
S-13B	All welded joints so that there are no bolts.					
S-13C						
S-13D						
S-5B[2]	3,835.	19,200	19,200	4,530.	2,780.	500
S-5C[2]	5,290.	26,447	26,447	6,710.	3,470.	5,000[1]
S-5D[2]	2,600.	13,013	13,013	2,900.	2,380.	5,000[1]
S-5E[2]	2,510.	12,571	12,571	2,890.	2,380.	5,000
S-6A	4,610	9,220	9,220	4,980.	2,860.	5,000[1]
S-11A						
S-11B	All welded joints so that there are no bolts.					
S-11D						
S-19A	10,100.	20,200.	20,200.	12,050.	6,950.	500
S-3F						
S-3L	No cyclic requirements for these bolts.					

NOTES:

- [1] Note that the allowable cycles is less than that required by LOFT Spec. S-1 (Reference 5). Thus, these bolts will have to be changed before the required number of cycles has occurred.
- [2] Because the loads for these nozzles are in only one direction instead of $\pm$, it was assumed that the stress varied from the preload value, up to a maximum, down to the preload value, etc.

VII. REFERENCES

1. INC General Work Authorization, W-31209 (2-25-70).
2. "Containment Building Penetration and Nozzle Designation, Identification, and Control Data", Prepared by Interface Control of the System Branch, INC (12-22-69).
3. W. J. Dodson, Kaiser Engineers, LOFT SDD No. 1.2.17, "Preliminary System Design Description for the Piping Penetrations Through Containment Vessel and Membrane Nozzle System" (10-24-69).
4. LOFT End Plate Stress Analysis, Main Report, G. J. Antonucci, Jr., J. L. D'Agata, and R. B. Boin, Prepared for PDM per Order No. 724667 by Dynatech Corp., Submittal 2011 (10-68).
5. LOFT Specification S-1, Containment Vessel, Rev. 11 (1-30-70).
6. G.J.R., PDM Submittal 3419, Pittsburgh-Des Moines Steel Company "LOFT Containment Vessel Nozzle Blind Flanges" (1-23-70).
7. American Society of Mechanical Engineers, ASME Boiler and Pressure Vessel Code, Section III, "Nuclear Vessels" (1965).
8. INC Drawing L-W31207, T. Kinnaman, "LOFT Containment Vessel, Piping Penetration, Expansion Joints"
9. Letter, S. Silverman to P. S. Lanham, "Bolt Sizes for Expansion Joint on Piping Penetrations of LOFT", SS-18-70 (6-3-70).
10. R. J. Roark, Formulas for Stress and Strain, 4th Edition, McGraw-Hill (1965).
11. LOFT Specification P-4, "Piping Materials", Rev. 13 (9-11-69).
12. American Society of Mechanical Engineers, ASME Boiler and Pressure Vessel Code, Section VIII, Division 1, "Pressure Vessels" (1968).
13. John A. McClure, "TOODEE, A Two-Dimensional Time-Dependent Heat Conduction Program", Phillips Petroleum Co., IDO-17227, AEC R & D Report, T1D-4500 (4-67).
14. R. E. Narum, Phillips Petroleum Co., A Guide for Input Preparation for the Gas Computer Program, (1-68).
15. Tube Turn, Catalog 311, 2900 W. Broadway, Louisville, Kentucky 40201, (7-69).
16. "How to Calculate and Design for Stresses in Preloaded Bolts", A. G. Hopper and G. V. Thompson, Product Engineering, (9-14-64).
17. LOFT Specification HV-3, Ductwork, Plenums, Hoods and Accessories, Rev. 5 (2-9-70).
18. Theory of Plates and Shells, S. Timoshenko and S. Woinowsky-Krieger, McGraw-Hill, Second Edition (1959)

Fig. 1. Configuration of Nozzles S-1A (After Reference 1).

Fig. 2 Cross Section of Nozzle S-1A (After Reference 1)

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Fig. 3 Assumed Fatigue Cycle for Bolts in Nozzle S-1A

APPENDIX A

DETERMINATION OF THE "WORST-CASES" OF THERMAL LOADS

APPENDIX A

DETERMINATION OF THE "WORST-CASES" OF THERMAL LOADS

The pipe/flange system will be idealized as a thin, pressurized, heated, shell restricted from radial expansion at one point along the axis, by a rigid flange. The stresses in the various flange configurations will then be investigated to determine which cases appear to be worst.

Consider a long cylindrical shell subjected to internal pressure and a temperature change. From Reference 18, Page 475, the radial deflection due to the pressure acting alone is

$$\delta = \frac{pa^2}{Eh}$$

where p is the pressure, a the shell radius, E the modulus of elasticity, and h the shell thickness. The radial deflection caused by a temperature change T is

$$\delta_T = \alpha Ta$$

where α is the linear coefficient of expansion. The total deflection due to temperature and pressure is then

$$\delta_{\text{tot.}} = \alpha Ta + \frac{pa^2}{Eh}$$

Now consider a long cylinder subjected to a ring load of magnitude P . The deflection under the load is

$$\delta_P = \frac{Pa^2\beta}{2Eh}$$

where

$$\beta^4 = \frac{Eh}{4a^2D} = \frac{(1.65)^2}{a^2h^2} \quad \text{for } \nu = .3$$

and it has been assumed that the moduli of elasticity of the pipe and flange are equal. Since the pipe is much weaker than the flange surrounding the pipe, it is reasonable to assume that

$$\delta_P = \delta_{\text{tot}}$$

which yields

$$\frac{P}{t_f} = \frac{2Eh}{a\beta t_f} \left(\alpha T + \frac{pa}{Eh} \right) \quad (\text{A-1})$$

where t_f is the thickness of the flange. This is the desired relation for radial pressure acting on the flange. The larger the value of P/t_f , the larger will be the stresses in the flange. It is seen that many parameters enter the problem. Generally, the stress in the flange increases with the temperature change, pressure, and the parameter

$$\frac{h^{3/2} a^{1/2}}{t_f}$$

Reference 2 supplemented with information about the temperatures and pressures in the various pipes contains sufficient data to enable the solution of Equation A-1 for P/t_f . The relative values of P/t_f are then indicative of the stresses in the plate. Larger values indicate larger stresses. Since the pressure and temperature information is not readily available, it is presented in Table A-1. The data was verbally obtained from Mr. G. A. Dinneen of Idaho Nuclear Corporation.

Six cases which appear to be bad, were investigated. The results are summarized in Table A-II.

It is seen from Table A-II that by far the worst case is nozzle S-6A. The 1500 psi 600°F secondary coolant induces extremely high flange stresses. This observation is confirmed by computer analysis of the flange. Indeed, a steam trap is necessary to assure that this pipe/flange system is heated slowly. Discussion with N. E. Pace indicates that he is satisfied that the present design which incorporates a steam trap is satisfactory and will prevent rapid heating of the flange. Thus, the case analyzed is much more severe than can physically be obtained. The next worse case is nozzle S-19A, followed by S-2A and S-1A. These cases (except for S-2A) were investigated by computer analysis - see the appropriate section of the report and Appendix B.

TABLE A-I
PIPE TEMPERATURES & PRESSURES

Nozzle	Pipe Schedule Number*	Approximate Max. Temp. of Fluid in Pipe, °F	Approximate Maximum Gauge Pressure in Pipe, psi
S-1 A	8"-LS-7-A, Sch. 40	245	17
	2"-HS-96-A, Sch. 40	350	125
	3"-DS-14-E, Sch. 40	190	---
S-2 A	2"-IAV-252-A, Sch. 40	Ambient	---
	8"-IW-28-A, Sch. 40	"	150
	3"-DW-18-E, Sch. 40	"	---
	1½"-IA-20-A, Sch. 40	"	---
S-3 F	Various	"	---
S-3 L	½"-DD-18-Z	"	---
S-5 B	2"-IA-9-A, Sch. 40	"	---
	3"-IA-17-A, Sch. 40	"	---
	6"-IW-45-A, Sch. 40	"	---
S-5 C	2"-DW-36-E, Sch. 40	"	---
	3"-DW-1-E, Sch. 40	"	---
	4"-TW-1-A, Sch. 40	"	---
S-5 D	½"-PC-30-C, Sch. 80	130	2400
	1"-PC-43-C, Sch. 160	Ambient	---
	2"-PC-43-C, Sch. 160	350	---
S-5 E	½"-SC-37-B, Sch. 80	600	1130
	4"-SC-16-B, Sch. 80	450	---
S-6 A	10"-SC-1-B, Sch. 80	600	1500
S-7 A	8"-WW-45-A, Sch. 40	150	---
S-7 B	1½"-WWV-93-E, Sch. 40	180	---
S-7 C	3"-WW-100-E, Sch. 40	190	---
S-8 A	2"-WHV-148-E, Sch. 40	150	---
S-11 C	½"-PR-46-E, Sch. 40	260	---
	½"-PR-13-E, Sch. 40	260	---
G-17 A	4"-WW-104 E, Sch. 40	100	---
S-17 B	4"-WH-128-E, Sch. 40	190	---
S-18 A	2"-PCV-5-A, Sch. 40	150	---
S-19 A	6"-IW-144-E, Sch. 40	60	90
S-11 A	12"-HV-Duct	Ambient	---

TABLE A-I (cont'd.)

Nozzle	Pipe Schedule Number*	Approximate Max. Temp. of Fluid in Pipe, °F	Approximate Maximum Gauge Pressure in Pipe, psi
S-11 B	12"-HV-Duct	Ambient	---
S-11 D	12"-HV-Duct	160	---
S-12 A	36"-HV-Duct	160	---
S-13 B	36"-HV-Duct	160	---
S-13 C	36"-HV-Duct	160	---
S-13 D	24"-HV-Duct	100	---

*

Pipe Code Letter

Material Specification; Pipe Schedule

A	ASTM A-53 or A-120; Schedule 40 for pipe 10" and under, Schedule 20 for pipe 12" and over.
B	ASTM A-106, Gr. B; Schedule 80.
C	ASTM A-106, Gr. B; Schedule 80 for 1/2" pipe and under, Schedule 160 for 3/4" through 2" pipe.
E	ASTM A-132, Gr. TP-304 or TP-304L, ASA B 36.19; Schedule 40
HV Duct, Type B	ASTM A-139, Gr. A or ASTM A-135, Gr. A for steel pipe to 28". ASTM A-134 for sizes 30" and larger or alternatively all steel pipe shall conform to ASTM A-211 with steel conforming to ASTM A-245. Square or rectangular ducts shall be fabricated from 12 gauge steel sheets conforming to ASTM A-366.
HV Duct, Type C	ASTM A-516, Gr. 70 and ASTM A-300 for ducts larger than 24". Wall thickness shall be 3/8". ASTM A-333, Gr. 6 standard weight for ducts 24" and smaller.

TABLE A-II
FLANGE STRESSES [1]

Nozzle & Pipe	Pressure, psi	Maximum Temperature, °F	Estimated Temperature Change, T, °F [2]	Thickness h, inches	radius, a, inches	Thickness, t _f , inches	Pressure, $\frac{P}{t_f}$ (from Eq. A-1)
S-1A 2"-HS-96-A	125	350	250	.154	1.11	2.125	2100.
S-1A 8"-LS-7-A	17	250	175	.322	4.65	2.125	2140
S-6A 10"-SC-1-B	1500	600	420	.594	5.08	1.6875	17800
S-5D ½"-PC-30-C	2400	130	90	.147	.346	1.4375	795.
S-2A 8"-IW-28-A	150	260 [3]	182	.322	4.15	1.875	2800
S-19A 6"-IW-144-E	90	260 [3]	182	.280	3.17	1.25	3790

NOTES

[2] It was assumed for all cases that $T = .7 \times (\text{Initial Temperature})$.

[1] For all calculations, it was assumed that $E = 30 \times 10^6$ psi and
 $\alpha = 6.5 \times 10^{-6}$ in./in./°F.

[3] The case of interest here is that of a cold pipe hit by the 260°F LOFT containment internal temperature. The stresses for this case will be identical to those calculated for a "hot"-260°F pipe and a cold flange.

APPENDIX B

THERMAL STRESS ANALYSIS OF NOZZLES

APPENDIX B

THERMAL STRESS ANALYSIS OF NOZZLES

The computer code called GAS^[14] was employed to calculate the stresses in nozzles S-1A, S-6A, and S-19A arising from the flange/pipe thermal loads and pipe internal pressure. The calculations of the thermal distributions are described in Appendix C. The finite element grid for the GAS program for S-1A is shown in Figure B-1. A representative temperature distribution at time = 100 seconds is shown in Figure B-2. The input for S-1A at time = 100 seconds is described below. Results for nozzles A-6A and S-19A are presented. Details of those inputs are omitted since they are similar to those of S-1A.

Input, Nozzle S-1A

No. of node points = 560
No. of elements = 480
No. of different materials = 2
No. of pressure cards = 35
No. of approximations = 1
Half band width = 37
Reference temperature = 70°F
Axial acceleration = 0
Angular velocity = 0

For material No. 1:

No. of temperature cards = 6
Mass density = 0
Modulus ratio = 0
SCODE = 0

The data on the first temperature card for material No. 1 is:

Temperature = 70°F
E(RZ) = 27.9×10^6 psi (modulus of elasticity in RZ plane)
NU(RZ) = .3 (Poisson's ratio in RZ plane)
E(T) = 27.9×10^6 psi
NU(T) = .3
ALPHA(RZ) = 6.07×10^{-6} in./in./°F (linear coefficient of expansion in RZ plane)
ALPHA(T) = 6.07×10^{-6} in./in./°F
YIELD STRESS = 0

The other five temperature cards for material No. 1 are similar except the modulus of elasticity, E, and linear coefficient of thermal expansion, α , change slightly with temperature.

The six temperature cards for material No. 2 are the same as for material No. 1 except that it was assumed that the steel composition for the flange had above .3 percent carbon instead of less than .3 percent. This changes E and α very slightly - see Tables N-426 and N-427 of Reference 7.

All the R loads and Z loads were assumed zero for the nodal point cards. The temperature distribution for these cards was obtained from the transient thermal analysis. The distribution is shown for a few representative points in Figure B-2.

The pressures for the pressure boundary conditions are all 21.0 psi for the locations shown in Figure B-1. This value is the difference between the pipe operation pressure of 15 psi and containment internal pressure of 36 psi. The input for the 300 sec and 505 sec is identical except that the temperature distribution is different.

RESULTS, NOZZLE S-1A

Results for the highest stressed locations are presented in the following table.

TABLE B-I

STRESS INTENSITY IN NOZZLE S-1A

<u>Location</u>	<u>Coordinate, Inches</u>		<u>Stress Intensity, SI x 10⁻⁴, psi</u>		
	<u>R</u>	<u>Z</u>	<u>Time Sec</u>		
			<u>100 sec</u>	<u>300 sec</u>	<u>500.5 sec</u>
Pipe	3.951	3.812	2.85	2.93	2.92
Pipe	3.951	3.937	3.22	3.298	3.27
Pipe	3.951	4.106	3.34	3.403	3.37
Pipe	3.951	5.806	3.31	3.386	3.34
Flange	4.376	4.106	2.087	2.235	2.29
Flange	4.376	4.743	2.088	2.237	2.25
Flange	4.376	4.956	2.095	2.28	2.29

It can be seen that the peak stress intensity in the pipe is greater than $1.5 S_m = 26,250$ psi. Although the stresses in the flange are still rising at time = 500 sec, they are nearly peaked out as indicated by the small changes between 300 sec and 500 sec.

It is concluded that the thermal stresses in the 8-inch pipe are excessive. The sum of the thermal plus secondary stresses in the flange must also satisfy Section III of the ASME Code. This is checked in the main body of the report.

RESULTS FOR NOZZLES S-6A & S-19A

Results for nozzles S-6A and S-19A were also obtained by means of GAS computer code. The idealized flange/pipe configurations may be seen in Figures B-3 and B-4. The pertinent data have been summarized in the following table:

TABLE B-II

STRESS INTENSITY IN NOZZLES S-6A & S-19A

<u>Nozzle</u>	<u>Location</u>	<u>Coordinate, in.</u>		<u>Max. Stress Intensity SI x 10⁻⁴, psi</u>	<u>Idealization</u>
		<u>R</u>	<u>Z</u>		
S-6A	Pipe	4.856	5.325	79,672	10" Sch. 80 pipe penetrating a 150 lb x 25" OD blind flange
	Flange	5.514	4.12	39,860	
S-19A	Pipe	3.067	5.16	22,836	6" Sch. 40 pipe penetrating a 150 lb x 17" OD blind flange
	Flange	3.415	5.16	14,914	

It is concluded that the 6" pipe of nozzle S-19A is satisfactory since the maximum stress is less than $1.5 S_m = 26,250$ psi. In the main body of the report the flange stresses shown are added to the secondary stresses due to forces and checked against allowable values. However, the flange and pipe of nozzle S-6A are severely overstressed for the assumed thermal loading. This is not surprising since a preliminary investigation indicated that this nozzle is by far the worst nozzle. It appears that the only reasonable alternative is to require that the nozzle be heated up gradually by the secondary coolant. This requirement is satisfied in the present design by a steam trap. See Appendix A for additional discussion.

Fig. B-1 Finite Element Grid for Stress Analysis of Nozzle S-1A

Fig. B-2 Temperature Distribution at 100 sec in Nozzle S-1A, °F

Fig. B-3 Finite Element Grid for Stress Analysis of Nozzle S-6A

Fig. B-4 Finite Element Grid for Stress Analysis of Nozzle S-19A

APPENDIX C

THERMAL ANALYSIS

APPENDIX C

THERMAL ANALYSIS

SCOPE OF INVESTIGATION

Thermal analyses were made on the pipe to nozzle flange closures on nozzles S-1A, S-6A, and S-19A. These three nozzles were chosen to be representative of the range of nozzles in the containment and are expected to exhibit the highest thermal stresses caused from temperature differences between the nozzle flange and the pipes welded into them, as shown in Figure C-1.

Two types of thermal transients were considered: (1) a hot fluid being induced into a cold pipe and adjacent nozzle flange, and (2) a cold nozzle flange and fluid filled pipe subjected to the steam-air atmosphere resulting from a loss of coolant experiment (LOCE). The thermal analyses of the three nozzles considered the following:

- (1) Nozzle S-1A: A 30 inch nozzle containing an insulated 8-inch low pressure steam pipe (8"-LS-7-A). The transient considered was rapid heating from an ambient temperature of 50°F to 245°F (the saturation temperature of 15 psig steam) which simulates opening the steam supply valve to this line.
- (2) Nozzle S-6A: A 20-inch nozzle containing an insulated 10-inch secondary coolant steam pipe (10"-SC-1-B). The transient considered for this analyses was hypothetical and represented the worst thermal transient possible for this nozzle closure. That is, a rapid thermal transient from 50°F ambient temperature to 600°F, the maximum secondary coolant system operating temperature at zero reactor power. This case could only exist if a slow secondary coolant system heatup was undertaken and sufficient condensate collected in the pipe to completely fill it in the area of nozzle S-6A. With the pipe filled with condensate, the pipe temperature could eventually reach the ambient temperature of 50°F. With this condition established a rapid (almost step change) temperature change would be initiated by the opening of the downstream flow control valves.

To remove the possibility of this severe thermal transient requires only that the steam trap upstream of the steam flow control valves be maintained in operation during all reactor system heatup cycles.

- (3) Nozzle S-19A: A 12-inch nozzle containing an uninsulated 6-inch industrial water pipe (6"-IW-144-E). The thermal transient considered results from an LOCE where the initial temperature of the pipe and nozzle flange was 60°F and the containment atmosphere increases rapidly from 60°F to 260°F due to flowing industrial water.

DISCUSSION

Method of Investigation

The transient temperature distributions in nozzles S-1A, S-6A, and S-19A were determined by using the TOODEE computer code (Reference 13) using axisymmetric cylindrical (R-Z) geometry. The models developed for the TOODEE code are shown in Figures C-2, C-3, and C-4 for Nozzles S-1A, S-6A, and S-19A, respectively.

The material thermal properties used were:

Thermal Conductivity (Btu/hr-ft-°F)

Insulation $K = 0.04$

Carbon Steel $K = 36.6 - 0.01T$

Volumetric Heat Capacity (Btu/ft³-°F)

Insulation $\rho c_p = 0.0168$

Carbon Steel $\rho c_p = 51.4 + 0.03124(T-459.6)$

where T is in degrees Rankine.

The initial temperature assumed for nozzles S-1A and S-6A was 50°F. The thermal transient was assumed to be a step temperature change from 50°F to 245°F and 600°F, respectively, for nozzles S-1A and S-6A. The heat transfer coefficient on the inside of the containment vessel was assumed to be 1.0 Btu/hr-ft²-°F. The heat transfer coefficients on the inside of the steam pipes were assumed to be 2000 and 1500 Btu/hr-ft²-°F for S-1A & S-6A, respectively.

The initial temperature assumed for nozzle S-19A was 70°F with flowing industrial water inside the pipe at a constant 60°F. The thermal transient investigated was that due to a LOCE where the temperature inside the containment increases from 100°F to 260°F in 6 seconds and then decreases slowly as specified in Table C-1[5]. The heat transfer coefficient inside the containment vessel was assumed to be (200 Btu/hr-ft²-°F) at the initiation of the LOCE and remain constant. The heat transfer coefficient inside the industrial water pipe remained at the initial value (1000 Btu/hr-ft²-°F) and the water temperature remained at 60°F.

RESULTS

The thermal analysis results for nozzle S-1A are given in Tables C-II, C-III and C-IV for 100, 300, and 500 seconds after transient initiation. These tables consist of the temperature map as it is printed out by the computer.

Table C-V shows the temperature distribution in nozzle S-6A, 101 seconds at the transient initiation.

Table C-VI shows the temperature distribution in nozzle S-19A, 1180 seconds after LOCE initiation.

The times selected for including the temperature distributions of these three nozzles represent times where the stresses were maximum or a range of times where stresses were determined to determine their maximums. The computer printout for these problems will be retained by the Applied Structural Mechanics Section for several years for future reference.

Fig. C-1 Typical Nozzle Showing Pipe & Flange Area Analyzed

Fig. C-2 Nozzle S-1A TOODEE Model

Fig. C-3 Nozzle S-6A TOODEE Model

GLO-99-70

Fig. C-4 Nozzle S-19A TOODEE Model

TABLE C-I

TEMPERATURES INSIDE LOFT CONTAINMENT
FOLLOWING AN LOCE

<u>Time</u> <u>sec.</u>	<u>Temperature</u> <u>°F</u>
0.0	100.0
6.0	260.0
85.2	254.0
121.2	252.6
157.2	251.0
193.2	249.8
229.2	248.4
265.2	247.0
301.2	245.6
319.2	244.5
373.2	243.0
733.2	231.8
769.2	239.0
805.2	243.5
841.2	242.3
877.2	241.0
1093.2	235.7
1453.2	228.4
1813.2	222.6
2173.2	218.0
2533.2	214.1
2893.2	210.6
3613.2	204.9
4506.0	198.8
5406.0	193.8
7206.0	185.7
10878.0	173.3
14478.0	163.7
18078.0	152.8
21678.0	141.2
25278.0	131.2
28878.0	122.4
32478.0	114.3
35646.0	107.6

TABLE C-II

OTIME STEPS HAVE BEEN CALCULATED

TEMPERATURE GRID IN DEGREES FAHRENHEIT

THE Y, Z, OR THETA DIRECTION IS VERTICAL

1-1 2-0 3-2 4-0 5-0 6-3 7-0 8-0 9-0 10-0 11-0 12-0 13-0 14-0 15-0 16-4 17-0 18-0 19-5 20-0

[illegible]

TABLE C-III
TEMPERATURE DISTRIBUTION IN NOZZLE S-1A
AT 300 SEC. AFTER TRANSIENT INITIATION

THE CURRENT TIME = 0.30000D 03 SEC 250TIME STEPS HAVE BEEN CALCULATED

TEMPERATURE GRID IN DEGREES FAHRENHEIT

BLOCK BOUNDARY LOCATIONS INDICATED BY -X-

THE X OR R DIRECTION IS HORIZONTAL

THE Y, Z, OR THETA DIRECTION IS VERTICAL

POINT
LOC

1-1 2-0 3-2 4-0 5-0 6-3 7-0 8-0 9-0 10-0 11-0 12-0 13-0 14-0 15-0 16-4 17-0 18-0 19-5 20-0

[illegible]

TEMPERATURE DISTRIBUTION IN NOZZLE S-1A AT 500 SEC. AFTER TRANSIENT INITIATION

TEMPERATURE GRID IN DEGREES FAHRENHEIT
BLOCK BOUNDARY LOCATIONS INDICATED BY -X-
THE X OR R DIRECTION IS HORIZONTAL THE Y, Z, OR THETA DIRECTION IS VERTICAL

1-1 2-0 3-2 4-0 5-0 6-3 7-0 8-0 9-0 10-0 11-0 12-0 13-0 14-0 15-0 16-4 17-0 18-0 19-5 20-0

[illegible]

TABLE C-VI
TEMPERATURE DISTRIBUTION IN NOZZLE S-19A
AT 1180 SEC. AFTER LOCE INITIATION

~~THE CURRENT TIME = 0.118000 04 SEC 2160 TIME STEPS HAVE BEEN CALCULATED~~

TEMPERATURE GRID IN DEGREES FAHRENHEIT
BLOCK BOUNDARY LOCATIONS INDICATED BY -X-

THE X OR R DIRECTION IS HORIZONTAL

THE Y, Z, OR THETA DIRECTION IS VERTICAL

POINT

~~LOC~~ ~~1-1~~ ~~2-0~~ ~~3-2~~ ~~4-0~~ ~~5-0~~ ~~6-0~~ ~~7-0~~ ~~8-0~~ ~~9-0~~ ~~10-3~~ ~~11-0~~ ~~12-0~~ ~~13-0~~ ~~14-0~~ ~~15-0~~ ~~16-0~~ ~~17-0~~ ~~18-4~~ ~~19-0~~

[illegible]