ENGINEERING

CENC 1194
SUPPLEMENT
NO. 1

VOLUME II

LMFBR DEMONSTRATION PLANT STEAM GENERATING SYSTEM

AEC CONTRACT AT(11-1)-3031

FEBRUARY 1973

CE COMBUSTION DIVISION

COMBUSTION ENGINEERING, INC.

MASTER

DISCLAIMER

APPENDIX A.4

PRELIMINARY STRUCTURAL ANALYSIS

OF

TUBE-TO-TUBESHEET WELD

REPORT NO. ______

COMBUSTION ENGINEERING, INC.

CHATTANOOGA DIVISION

NUCLEAR COMPONENTS DEPARTMENT

TITLE

PRELIMINARY STRUCTURAL ANALYSIS
OF TUBE TO TUBESHEET WELD

	JANUARY 31, 1973
CALCULATION NO	DATED
D-51100	SODIUM S.G.
CONTRACT NO.	UNIT DESIGNATION

W. J. Heilker
W. J. HEILKER
PREPARED

S.R. Penfield, Jr.
CHECKED

Frank P. Hell...
REVIEWED

CHANGES

DATE	REV. BY	PAGES AFFECTED: REVISED	REMOVED	ADDED	DESCRIPTION OF CHANGE

COMBUSTION ENGINEERING, INC.
ENGINEERING DEPARTMENT, CHATTANOOGA, TENN.

NUMBER A.4
SHEET 1 OF 24

CHARGE NO.

DATE 1-31-73 BY HEILKER

DESCRIPTION TUBE TO TUBESHEET WELD

CHECK DATE 1-31-73 BY [signature]

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1. ABSTRACT

This analysis presents the preliminary structural evaluation of the pressure tube to high pressure tubesheet weld. Stresses considered are those resulting for steam side design pressure, sodium side pressure, and forced displacement of the tubeweld due to tubesheet deflection under design loading.

The analysis is made in accordance with the 1971 ASME Boiler and Pressure Vessel Code, Section III for Nuclear Power Plant Components and includes Addenda through Summer, 1972.

2. SIGNIFICANT RESULTS

All stresses are satisfactory and meet the appropriate allowables set forth in the ASME Boiler and Pressure Vessel Code, Section III for Nuclear Power Plant Components.

The highest value of local primary membrane stress intensity occurred at Cut 5-6 (See sheet 18). The stress intensity was 20.4 ksi which did not exceed the allowable of 1.5 Sm = 22.2 ksi.

The greatest range of primary local plus secondary stress intensity (for pressure and forced displacement only) was 26.4 ksi which was well below the allowable of 3 S_m = 44.4 ksi. This value occurred at Location 5 (See sheet 18). The effects of temperature on the range of stress will be considered in the fatigue evaluation.

3. GENERAL DISCUSSION

The forced displacement imposed upon the tubeweld was determined by modeling the tubesheet, sodium flange, high pressure shell, sodium shell, and support skirt (See sheet 9) with the "SAAS" Finite Element Computer Program (Reference 46) and extracting deflections in the area of the weld.

The above deflections together with steam side and sodium side pressure were loaded on the tubeweld model (See sheet 10) and evaluated with Wilson's Finite Element Computer Program (Reference 25).

The exaggerated deformed shapes shown on Sheets 14 and 15 give a qualitative representation of the type of stress (compressive or tensile) in various areas of the tubesheet and tubeweld. The iso-stress plots on Sheets 16 and 17 show the stress patterns that have formed in the tube to tubesheet weld.

Examination of the surface stresses tabulated on Sheet 19 will disclose that some surface yielding occurs, however, these areas will shake-down to elastic action after one cycle of loading.

4. REFERENCES

1. ASME Boiler and Pressure Vessel Code, Section III for Nuclear Power Plant Components.

25. Computer program, "Analysis of Axisymmetric Solids," E. L. Wilson, University of California, February, 1967.

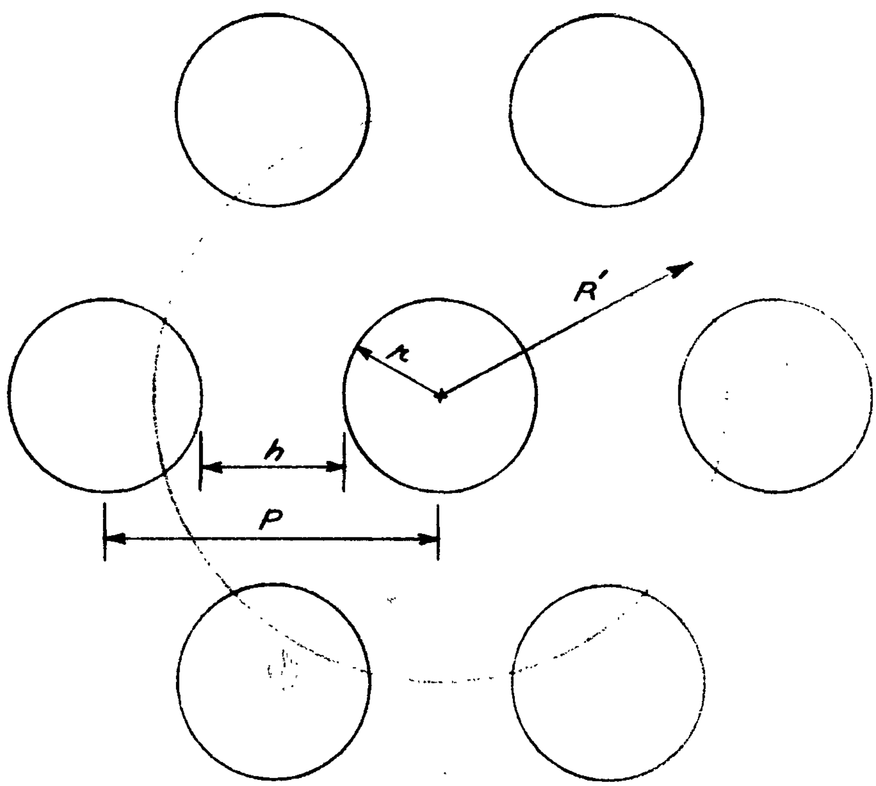
46. "SAAS II Finite Element Stress Analysis of Axisymmetric Solids with Orthotropic, Temperature-Dependent Material Properties," by Robert M. Jones and James G. Crose, September, 1968.

48. "Giant (Graphical Information Analysis Tool)," Computer program, by J. J. Diorio and S. E. Deabler, Combustion Engineering.

5. DETAILED ANALYSIS

a. GEOMETRY

HOLE PATTERN IN TUBESHEET



$$R' = P - \frac{1}{4}(P-h) = P - \frac{r}{2}$$

WHERE: $P = 2.0$ IN.
$h = 0.844$ IN.
$r = 0.578$ IN.

$$R' = 2.0 - \frac{0.578}{2} = 1.711 \text{ IN.}$$

5. DETAILED ANALYSIS

b. SYSTEM ALLOWABLES

1. THE LOCAL PRIMARY MEMBRANE STRESS INTENSITY DUE TO PRESSURE AND FORCED DISPLACEMENT, SHALL BE LESS THAN 1.5 Sm AT DESIGN TEMPERATURE.

2. THE PRIMARY PLUS SECONDARY STRESS INTENSITY RANGE SHALL BE LESS THAN 3 Sm.

c. SYSTEM LOADING

1. DESIGN CONDITIONS AS FOLLOWS:

$$\left.\begin{array}{l}\text{STEAM PRESSURE} = 2650 \text{ PSI} \\ \text{SODIUM SIDE PRESSURE} = 80 \text{ PSI}\end{array}\right\} @\ 675°F$$

2. FORCED DISPLACEMENT OF THE TUBEWELD DUE TO TUBESHEET DEFLECTION UNDER DESIGN LOADING (TUBEWELD ASSUMED TO BE AT THE CENTER OF THE TUBESHEET WHERE DEFLECTION IS MAXIMUM).

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CHARGE NO. ______

DESCRIPTION TUBE TO TUBESHEET WELD

NUMBER A.4

SHEET 9 OF 24

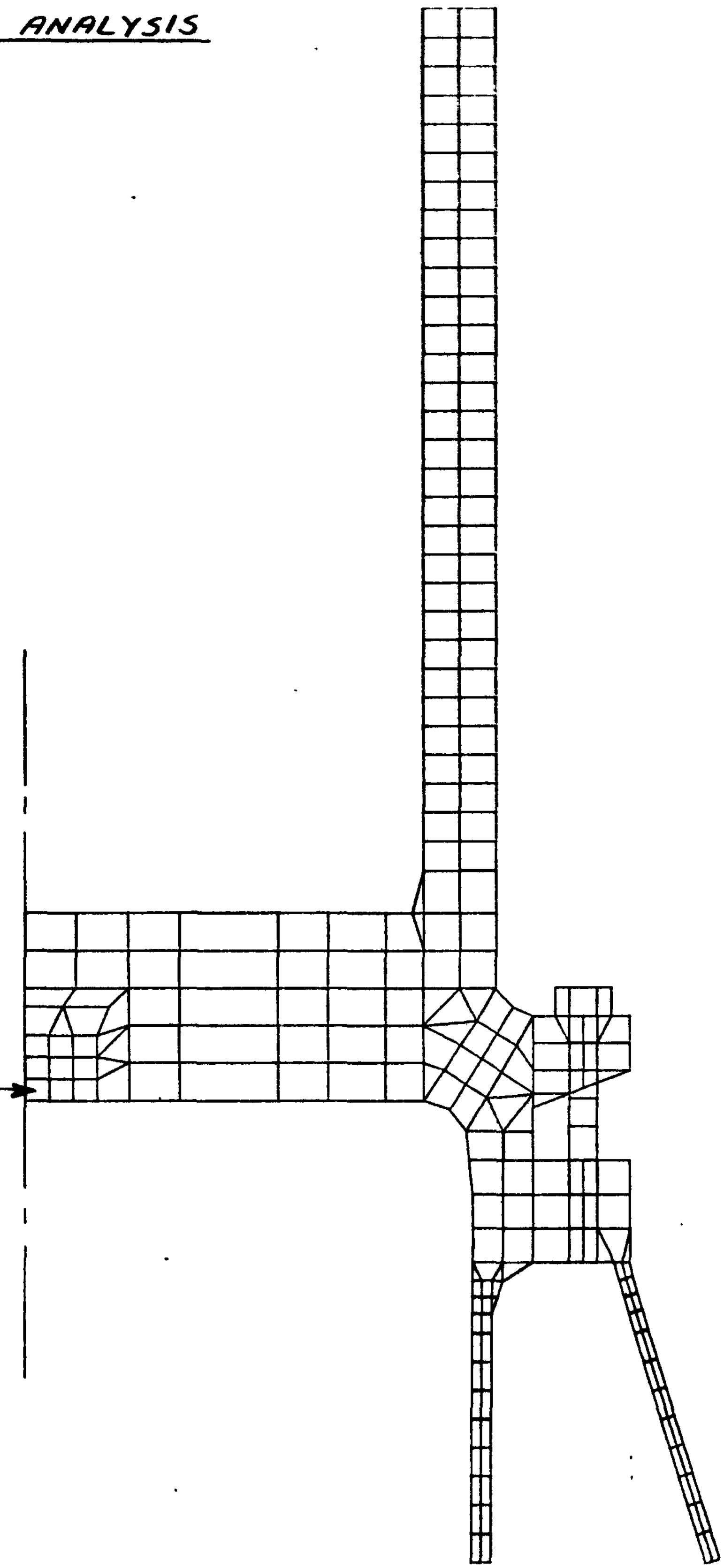
DATE 12-1-72 BY HEILKER

CHECK DATE 1-31-73 BY [illegible]

5. DETAILED ANALYSIS

d. MODEL

LOCATION OF TUBEWELD ENLARGEMENT



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CHARGE NO.

DESCRIPTION TUBE TO TUBESHEET WELD

NUMBER A.4

SHEET 10 OF 24

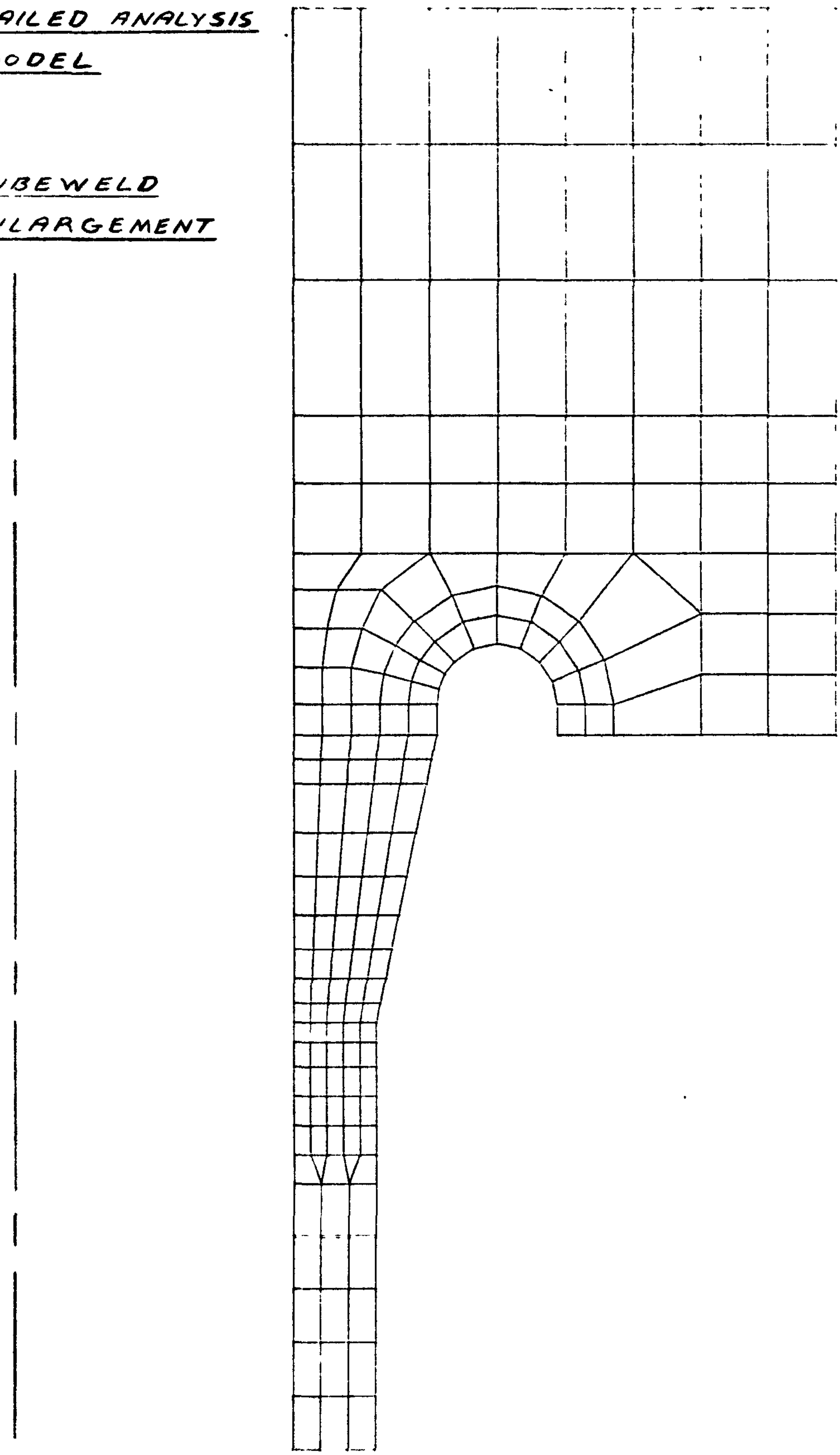
DATE 12-1-72 BY HEILKER

CHECK DATE 1-31-73 BY [illegible]

5. DETAILED ANALYSIS

d. MODEL

TUBEWELD ENLARGEMENT



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5.) DETAILED ANALYSIS

e. FINITE ELEMENT ANALYSIS

METHOD

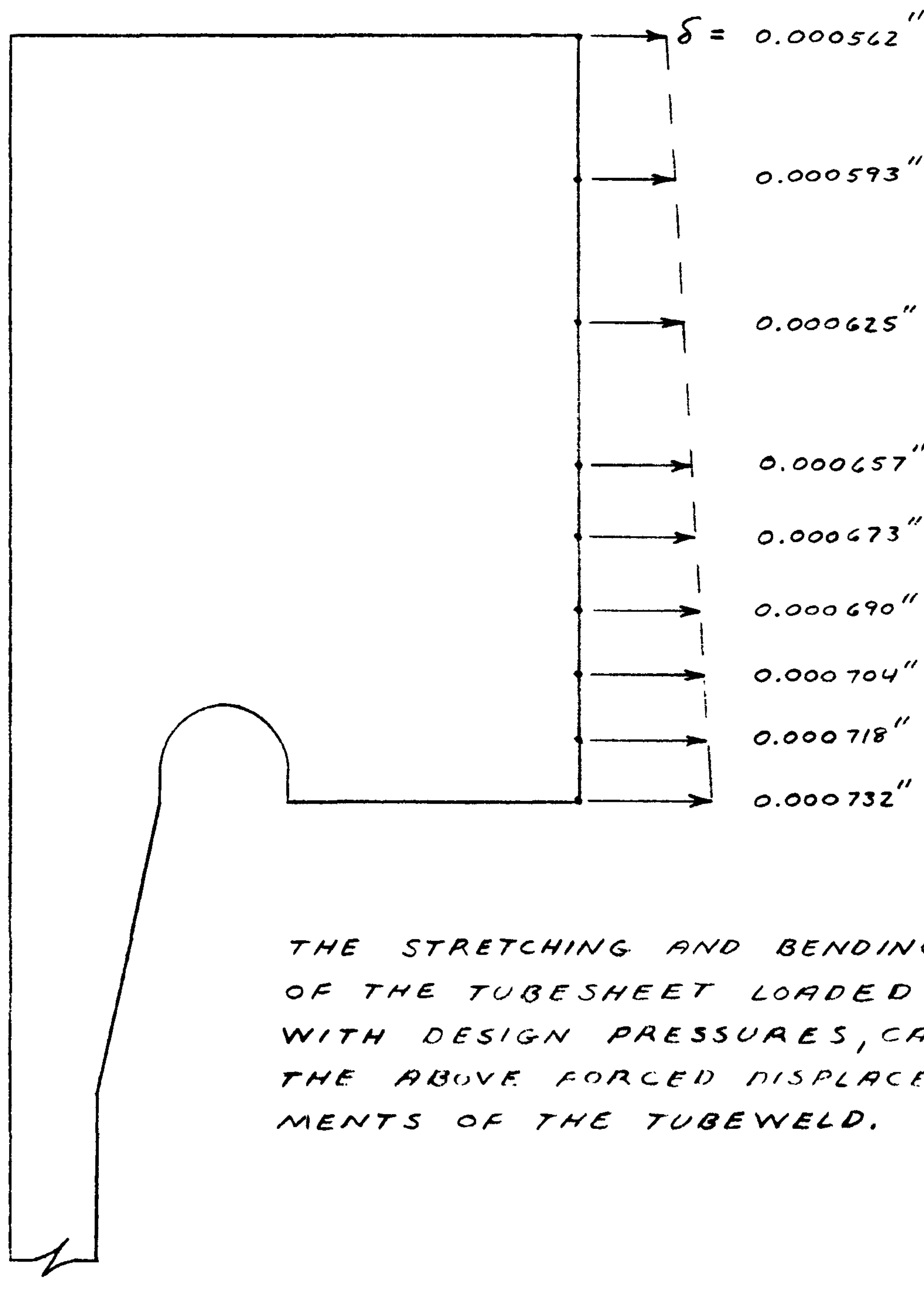
IN ORDER TO DETERMINE THE STRESS IN THE TUBEWELD, IT IS FIRST NECESSARY TO CALCULATE THE AMOUNT OF DEFORMATION IMPOSED ON THE WELD BY THE FLEXING OF THE TUBESHEET. THIS WAS ACCOMPLISHED BY CONSTRUCTING A FINITE ELEMENT MODEL ENCOMPASSING THE TUBESHEET, SODIUM FLANGE, SODIUM SHELL, SUPPORT SKIRT, HIGH PRESSURE SHELL, STUDS AND NUTS. ONE ELEMENT OF THIS MODEL REPRESENTS THE TUBEWELD GEOMETRY SHOWN IN 5.a. THE MODEL WAS ANALYZED USING THE COMPUTER PROGRAM SAAS II, REFERENCE 46.

THE NEXT STEP WAS TO CONSTRUCT A DETAILED MODEL OF THE TUBEWELD. AS PART OF THE BOUNDARY CONDITIONS FOR THIS DETAILED MODEL, DEFLECTIONS WERE EXTRACTED FROM THE TUBESHEET MODEL. THE TUBEWELD MODEL WAS THEN ANALYZED USING "WILSON'S" FINITE ELEMENT COMPUTER PROGRAM REFERENCE 25.

5. DETAILED ANALYSIS

e. FINITE ELEMENT ANALYSIS

TUBEWELD FORCED DISPLACEMENT



THE STRETCHING AND BENDING OF THE TUBESHEET LOADED WITH DESIGN PRESSURES, CAUSE THE ABOVE FORCED DISPLACEMENTS OF THE TUBEWELD.

5.) DETAILED ANALYSIS

e. FINITE ELEMENT ANALYSIS

MATERIAL PROPERTIES

2 1/4 Cr - 1 Mo

DESIGN TEMPERATURE = 675°F

$E = 25.2 \times 10^6$ PSI

$\nu = 0.3$

9 Cr - 1 Mo

DESIGN TEMPERATURE = 675°F

$E = 27.1 \times 10^6$ PSI

$\nu = 0.3$

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CHARGE NO.

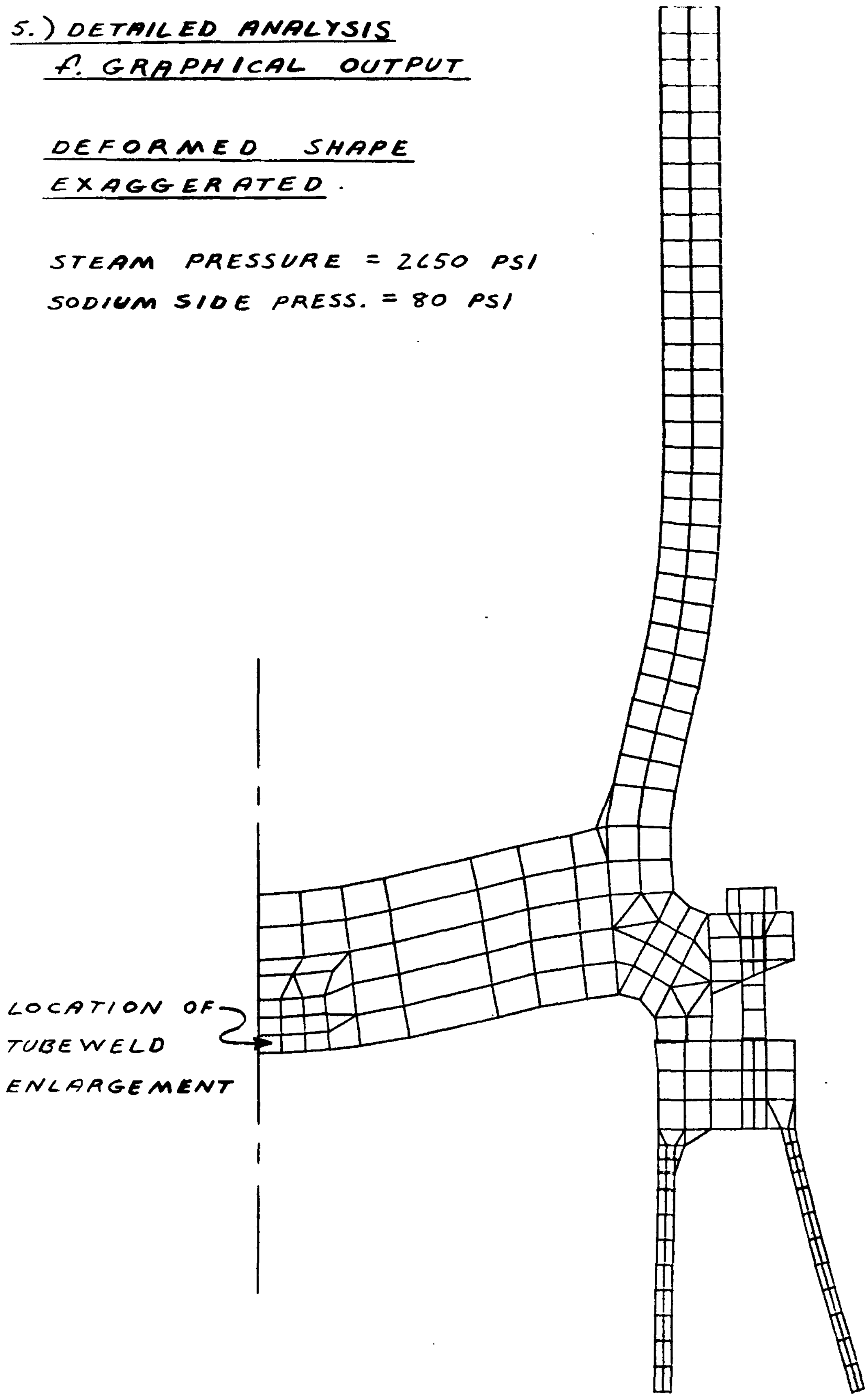
DESCRIPTION TUBE TO TUBESHEET WELD

NUMBER A.4
SHEET 14 OF 24
DATE 12-1-72 BY HEILKER
CHECK DATE 1-31-73 BY SMC

5.) DETAILED ANALYSIS
A. GRAPHICAL OUTPUT

DEFORMED SHAPE
EXAGGERATED.

STEAM PRESSURE = 2650 PSI
SODIUM SIDE PRESS. = 80 PSI



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ENGINEERING DEPARTMENT, CHATTANOOGA, TENN.

NUMBER A.4

SHEET 15 OF 24

CHARGE NO.

DATE 12-1-72 BY HEILKER

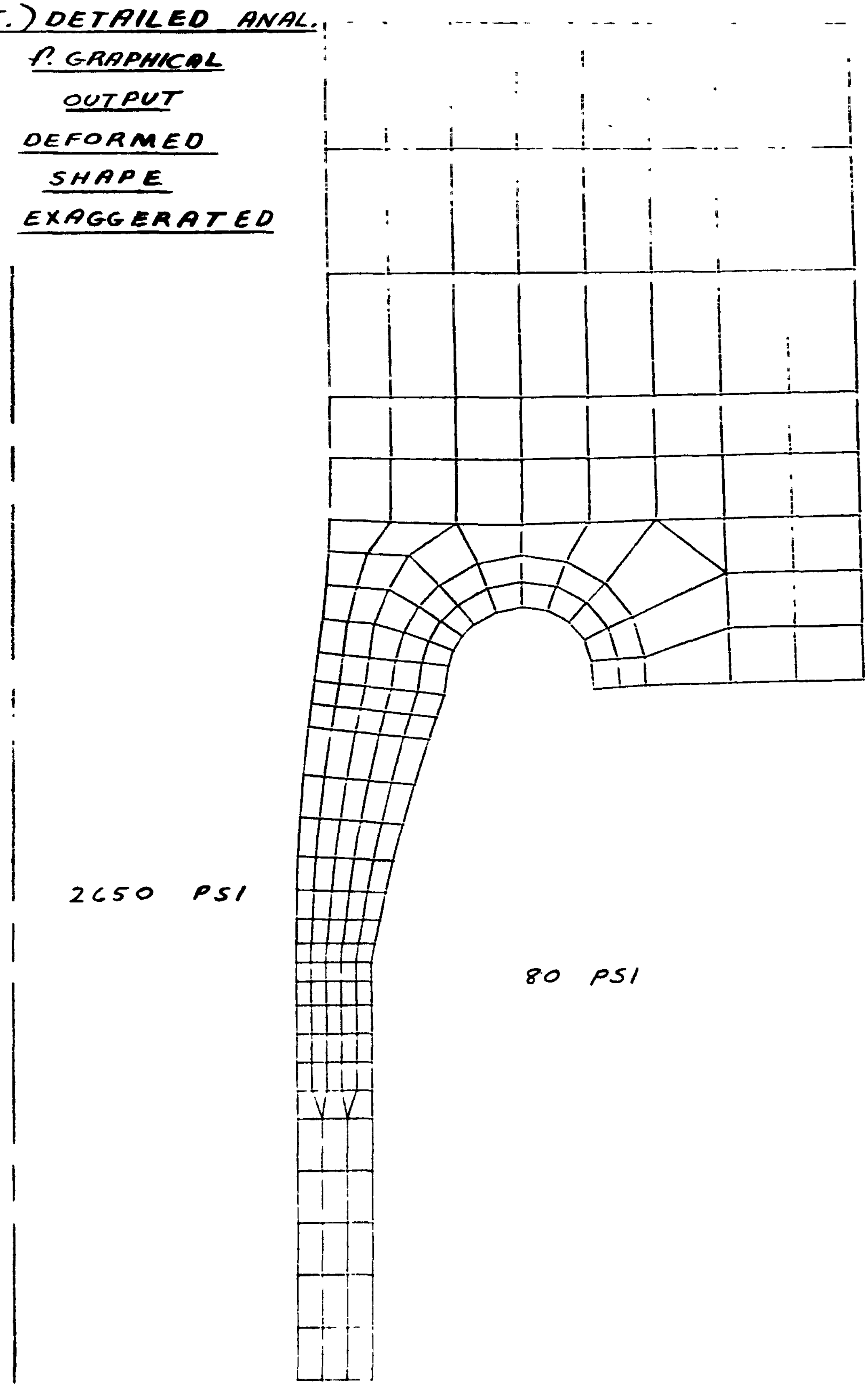
DESCRIPTION TUBE TO TUBESHEET WELD

CHECK DATE 1-31-73 BY SHP

5.) DETAILED ANAL.

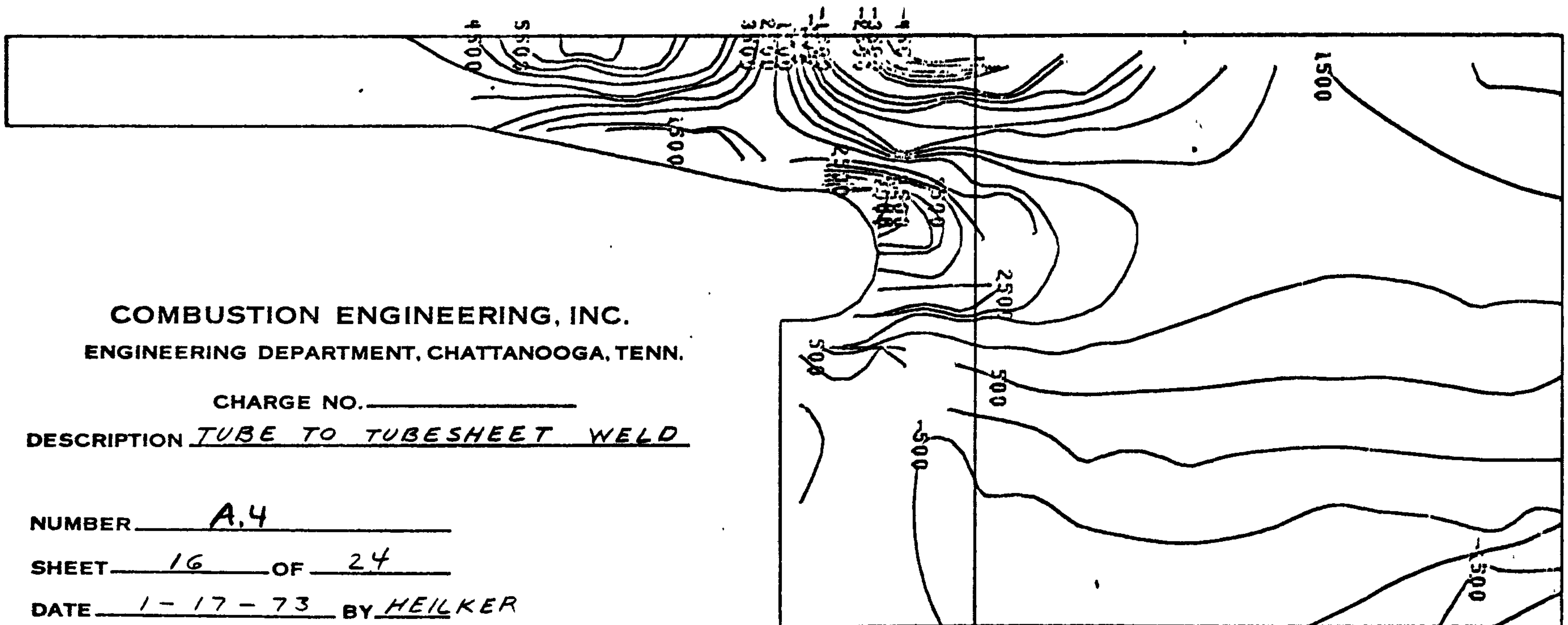
f. GRAPHICAL OUTPUT

DEFORMED SHAPE EXAGGERATED



SP 11461 A

SODIUM STEAM GENERATOR
EVAPORATOR
TUBEWELD
FULL POWER STEADY STATE
SIGMA Z



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CHARGE NO. ____________

DESCRIPTION *TUBE TO TUBESHEET WELD*

NUMBER *A.4*

SHEET *16* OF *24*

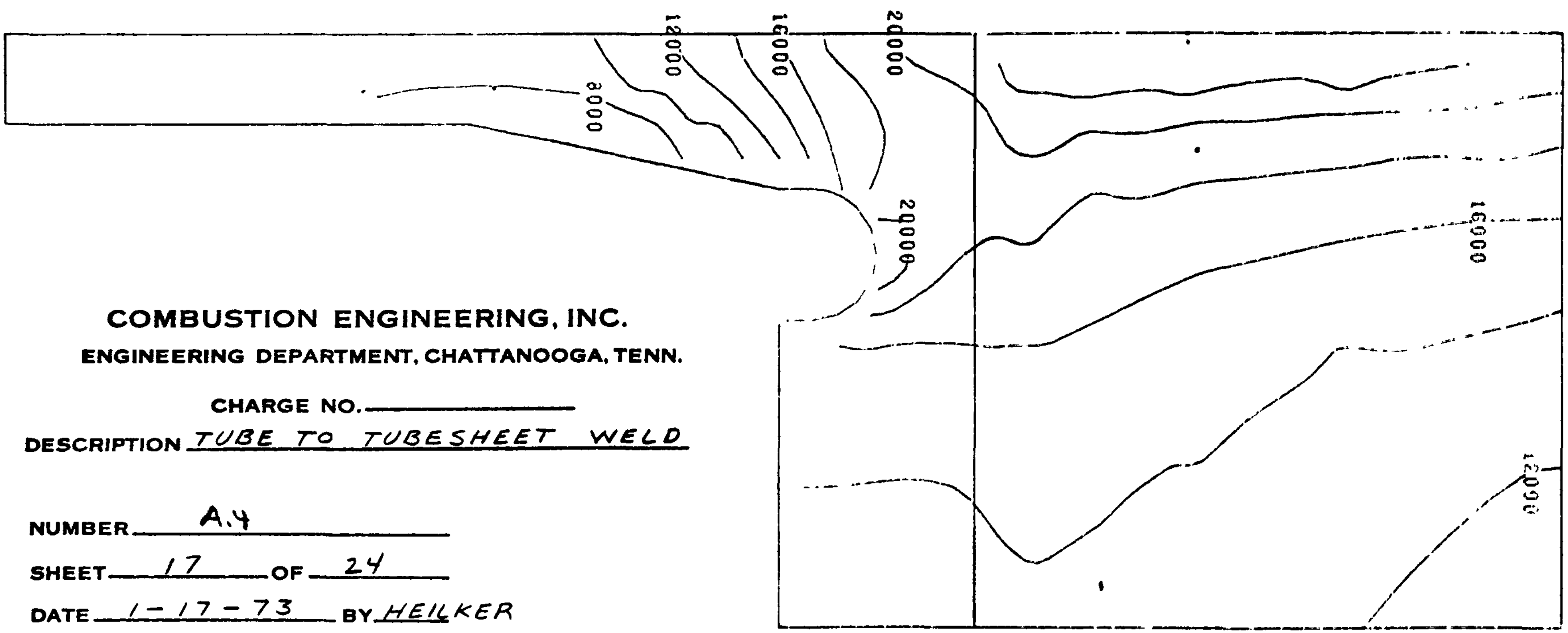
DATE *1-17-73* BY *HEILKER*

CHECK DATE *1-31-73* BY *[illegible]*

5.) DETAILED ANALYSIS

f. GRAPHICAL OUTPUT

SODIUM STEAM GENERATOR
EVAPORATOR
TUBEWELD
FULL POWER STEADY STATE
SIGMA T



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CHARGE NO. ____________

DESCRIPTION *TUBE TO TUBESHEET WELD*

NUMBER *A.4*

SHEET *17* OF *24*

DATE *1-17-73* BY *HEILKER*

CHECK DATE *1-21-73* BY [illegible]

5.) DETAILED ANALYSIS

f. GRAPHICAL OUTPUT

COMBUSTION ENGINEERING, INC.

ENGINEERING DEPARTMENT, CHATTANOOGA, TENN.

NUMBER A.4

SHEET 18 OF 24

CHARGE NO.

DATE 1-30-73 BY HEILKER

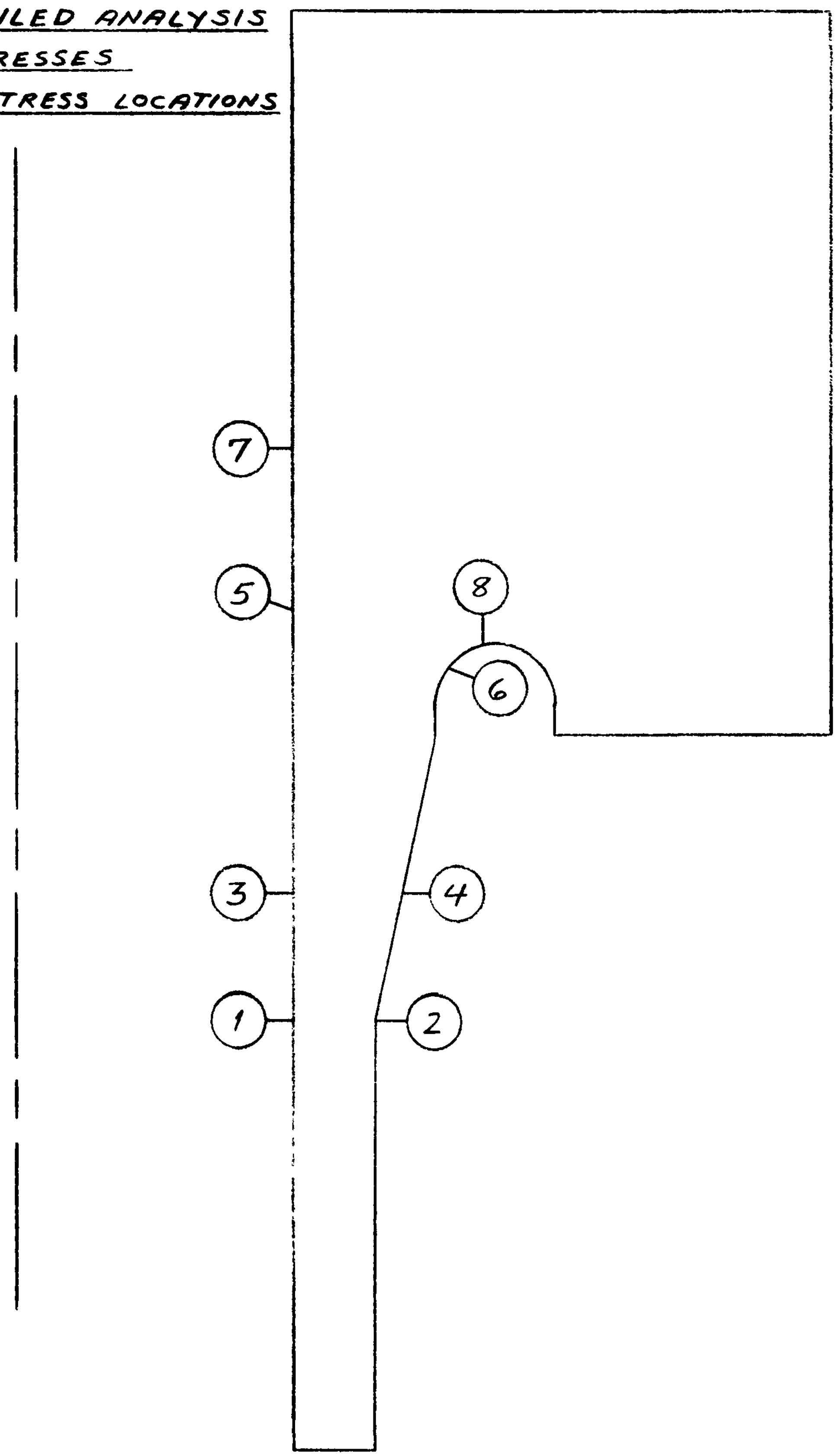
DESCRIPTION TUBE TO TUBESHEET WELD

CHECK DATE 1-31-73 BY SMP

5.) DETAILED ANALYSIS

g. STRESSES

1. STRESS LOCATIONS



SP 11461 A

5.) DETAILED ANALYSIS

g. STRESSES

2. SUMMARY

SURFACE STRESSES

INSIDE			OUTSIDE		
LOCATION	σ_{NORMAL}	σ_θ	LOCATION	σ_{NORMAL}	σ_θ
1	5.06	9.51	2	2.63	6.49
3	6.95	11.30	4	-1.02	5.61
5	-5.50	21.20	6	16.80	19.00
7	0.37	26.00	8	24.00	22.20

UNCONCENTRATED STRESS

LOCATION	INSIDE		MID-SURFACE		OUTSIDE	
	σ_{NORM}	σ_θ	σ_{NORM}	σ_θ	σ_{NORM}	σ_θ
1—2	4.77	9.43	3.76	7.81	2.75	6.19
3—4	6.66	10.99	2.93	8.20	-0.80	5.41
5—6	-6.26	20.09	3.51	19.01	13.29	17.93

COMBUSTION ENGINEERING, INC.
ENGINEERING DEPARTMENT, CHATTANOOGA, TENN.

CHARGE NO.
DESCRIPTION TUBE TO TUBESHEET WELD

NUMBER A.4
SHEET 20 OF 24
DATE 1-31-73 BY HEILKER
CHECK DATE 1-31-73 BY SMP

5.) DETAILED ANALYSIS

g. STRESSES

3. CALCULATIONS

UNCONCENTRATED STRESS

STRESSES OBTAINED FROM THE FINITE ELEMENT ANALYSIS INCLUDE THE EFFECT OF STRESS CONCENTRATION. IN ORDER TO CALCULATE PRIMARY LOCAL MEMBRANE STRESS AND SECONDARY BENDING STRESS, IT IS NECESSARY TO REMOVE STRESS CONCENTRATION. THE METHOD DESCRIBED BELOW WAS USED TO DETERMINE THE UNCONCENTRATED STRESSES.

F = TOTAL FORCE ACTING NORMAL TO CUT.

$$F = \sum_{i=1}^{n} \sigma_i A_i$$

WHERE: σ_i = ELEMENT STRESS NORMAL TO CUT
A_i = ELEMENT AREA AT CUT
n = NUMBER OF ELEMENTS AT CUT

M = TOTAL BENDING MOMENT ACTING ON CUT.

$$M = \sum_{i=1}^{n} \sigma_i A_i d_i$$

WHERE: d_i = DISTANCE FROM ℄ OF CUT TO CENTROID OF ELEMENT

USING THE TOTAL MOMENT AND TOTAL NORMAL FORCE ON THE CUT, THE UNCONCENTRATED STRESSES ARE CALCULATED.

$$\sigma = \frac{F}{t} \pm \frac{6M}{t^2}$$

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5.) DETAILED ANALYSIS

g. STRESSES

3. CALCULATIONS

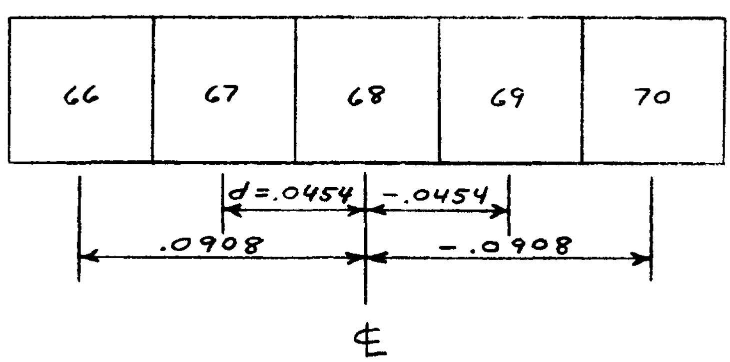
UNCONCENTRATED STRESS

EXAMPLE : 3—4

ELEMENT	σ_N	$\sigma_N A$	$\sigma_N A d$	σ_θ	$\sigma_\theta A$	$\sigma_\theta A d$
66	6.11	0.277	0.025	10.61	0.482	0.044
67	4.44	0.202	0.009	9.23	0.419	0.019
68	2.90	0.132	0	8.05	0.365	0
69	1.38	0.063	-0.003	7.02	0.319	-0.014
70	-0.22	-0.010	0.001	6.08	0.276	-0.025
	$\Sigma =$	0.664	0.032	$\Sigma =$	1.861	0.024
		F	M		F_θ	M_θ

$A_i = .0454$ TYP.

ASSUME UNIT THICKNESS



THICKNESS
$t = 0.227$

$$\sigma_N \text{ UNCONC.} = \frac{F}{t} \pm \frac{6M}{t^2} = 2.93 \pm 3.73 = 6.66 \quad (3) \text{ INSIDE}$$

$$= -0.80 \quad (4) \text{ OUTSIDE}$$

$$\sigma_\theta \text{ UNCONC.} = \frac{F_\theta}{t} \pm \frac{6M_\theta}{t^2} = 8.20 \pm 2.79 = 10.99 \quad (3) \text{ INSIDE}$$

$$5.41 \quad (4) \text{ OUTSIDE}$$

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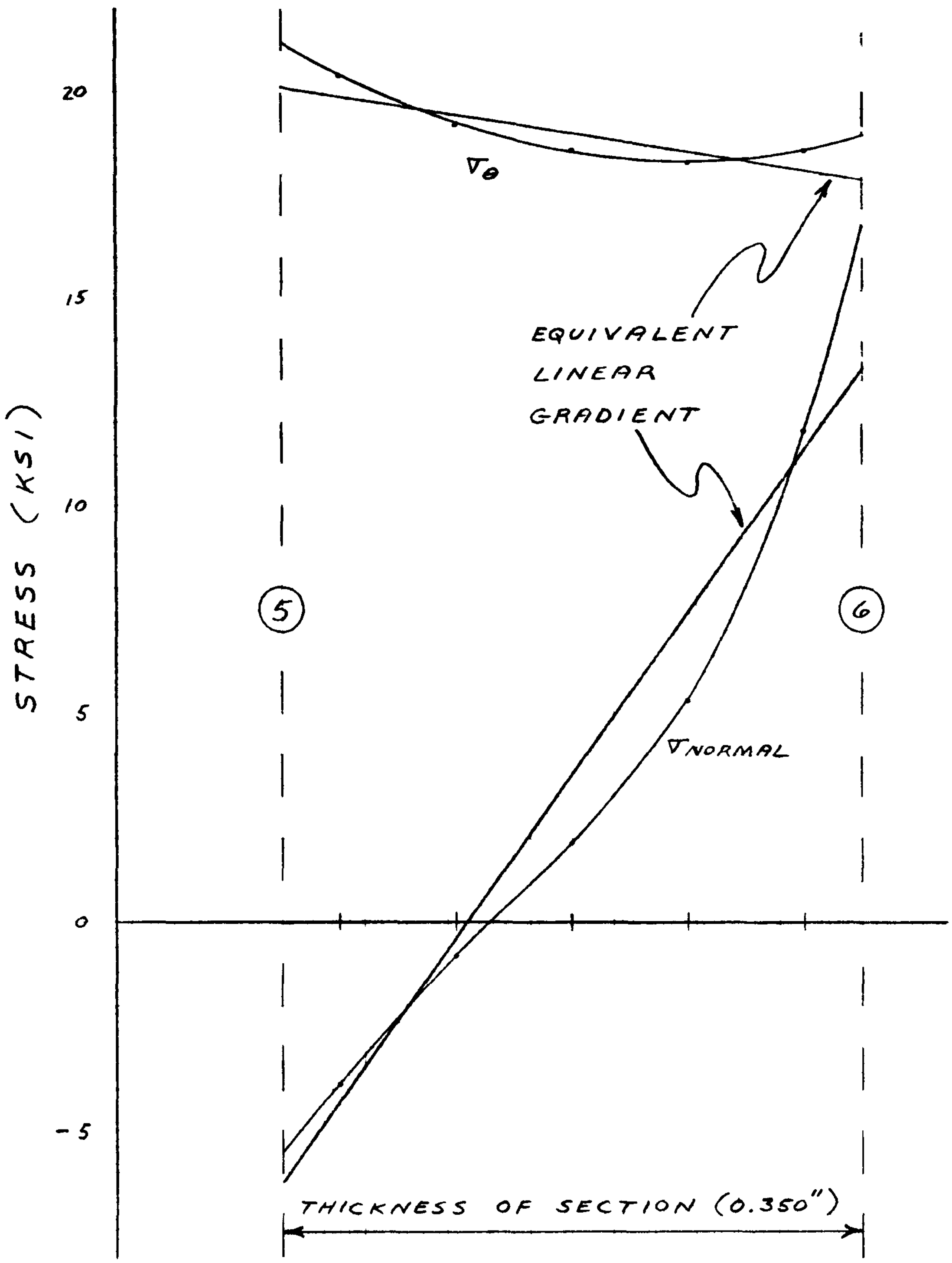
ENGINEERING DEPARTMENT, CHATTANOOGA, TENN.

NUMBER A.4
SHEET 22 OF 24
CHARGE NO.
DATE 1-31-73 BY HEILKER
DESCRIPTION TUBE TO TUBESHEET WELD
CHECK DATE 1-31-73 BY [illegible]

5.) DETAILED ANALYSIS
g. STRESSES
3. CALCULATIONS



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5.) DETAILED ANALYSIS

h. RESULTS

CRITERION 5.b.1

PRIMARY LOCAL MEMBRANE

LOCATION	σ_N	σ_θ	σ_r	STRESS INTENSITY		
				$\sigma_N - \sigma_\theta$	$\sigma_N - \sigma_r$	$\sigma_\theta - \sigma_r$
1—2	3.76	7.81	-1.37	-4.05	5.13	9.18
3—4	2.93	8.20	-1.37	-5.27	4.30	9.57
5—6	3.51	19.01	-1.37	-15.50	4.88	20.38

$S.I._{MAX} = 20.38 \text{ KSI} < 1.5 S_m = 22.2 \text{ KSI}$ @ 675°F

5.) DETAILED ANALYSIS .
h. RESULTS

CRITERION 5.b.2

PRIMARY LOCAL PLUS SECONDARY

LOCATION	σ_N	σ_θ	σ_n	STRESS INTENSITY		
				$\sigma_N - \sigma_\theta$	$\sigma_N - \sigma_n$	$\sigma_\theta - \sigma_n$
1	4.77	9.43	-2.65	-4.66	7.42	12.08
2	2.75	6.19	-0.08	-3.44	2.83	6.27
3	6.66	10.99	-2.65	-4.33	9.31	13.64
4	-0.80	5.41	-0.08	-6.21	-0.72	5.49
5	-6.26	20.09	-2.65	-26.35	-3.61	22.74
6	13.29	17.93	-0.08	-4.64	13.37	18.01

THE OPPOSING EXTREME FOR PURPOSES OF RANGE OF STRESS IS AMBIENT CONDITIONS WHERE ALL STRESS IS ZERO.

$$RANGE_{MAX} = 0 - (-26.35) = \underline{26.35 \text{ KSI} < 3S_m = 44.4 \text{ KSI}}$$

@ 675°F

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APPENDIX A.5

PRELIMINARY STRUCTURAL ANALYSIS OF HIGH PRESSURE TUBE AT THE SODIUM-ARGON INTERFACE (PARTIAL)

REPORT NO. ________

COMBUSTION ENGINEERING, INC.

CHATTANOOGA DIVISION

NUCLEAR COMPONENTS DEPARTMENT

TITLE

PRELIMINARY STRUCTURAL ANALYSIS OF THE
PRESSURE TUBE AT SODIUM-ARGON INTERFACE

A-5	December 12, 1972
CALCULATION NO	DATED
D-51100	SODIUM S.G.
CONTRACT NO.	UNIT DESIGNATION

W. J. Heilker
PREPARED

CHECKED

REVIEWED

CHANGES

DATE	REV. BY	PAGES AFFECTED REVISED	REMOVED	ADDED	DESCRIPTION OF CHANGE

COMBUSTION ENGINEERING, INC.

ENGINEERING DEPARTMENT, CHATTANOOGA, TENN.

CHARGE NO. D-51100

DESCRIPTION PRESSURE TUBE AT SODIUM-ARGON INTERFACE

NUMBER A-5

DATE 2-9-73 BY HEILKER

CHECK DATE ______ BY ______

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SUMMARY

The purpose of this analysis is to examine the high pressure tube at the sodium level in the steam generator, and determine if any inelastic deformation occurs. It is shown in this report that inelastic deformation does occur (See Sheet 14), and an attempt was made to calculate the incremental plastic strain using a simplified conservative method, the full relaxation Bree method. The Bree method proved, however, to be too conservative and the recommendation of this report is to perform a more rigorous inelastic calculation.

RESULTS

The stresses obtained from a seal-shell analysis (Sheet 7 through 10) indicate that the critical stress intensity is $\sigma_\theta - \sigma_r$ on the inside surface of the tube. Two locations were examined in detail, Node 14, the location of maximum thermal cycling between steady level and fluctuating level, and Node 22, the location of maximum stress. The effects of fatigue and secondary creep were found to be small but plastic deformation was shown to exist. Although satisfactory results were not obtained using the full relaxation Bree method, it should be pointed out that this method assumes ratcheting to exist while the figure on Sheet 14 clearly indicates the stresses to be in the shakedown regime.

REFERENCES

1. ASME Boiler and Pressure Vessel Code, Section III for Nuclear Power Plant Components

2. Interpretations of ASME Boiler and Pressure Vessel Code, Case 1331-5, August 4, 1971.

3. Commitments and Agreements of Steam Generator Design Transients Meeting held at LMEC, November 15, 1972.

4. "Metals Handbook," Volume 1, Properties and Selection, Eighth Edition, American Society for Metals.

5. "Elastic-Plastic Behavior of Thin Tubes Subjected to Internal Pressure and Intermittent High-Heat Fluxes with Application to Fast-Nuclear-Reactor Fuel Elements," by J. Bree.

6. "Seal-Shell-2, A Computer Program for the Stress Analysis of a Thick Shell of Revolution with Axisymmetric Pressure, Temperature and Distributed Loads," WAPD-TM-398, AEC Research and Development Report, 1963.

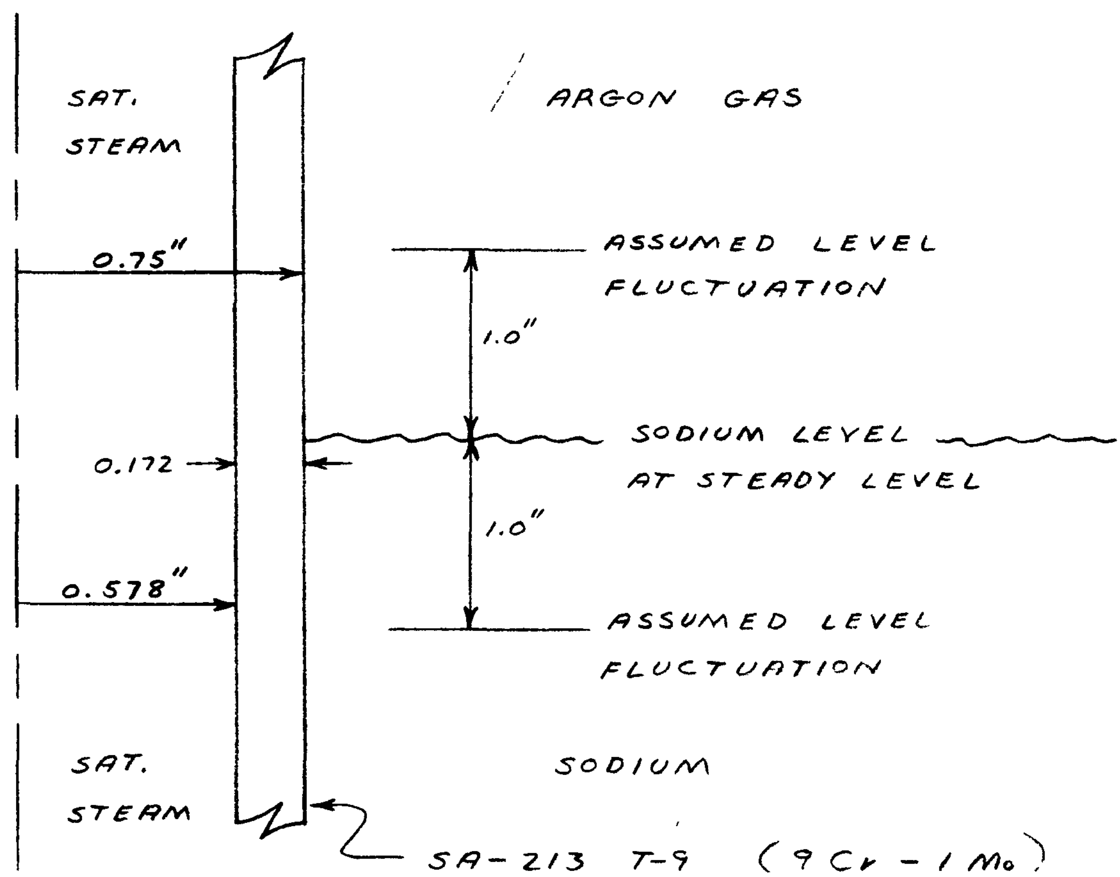
COMBUSTION ENGINEERING, INC.
ENGINEERING DEPARTMENT, CHATTANOOGA, TENN.

CHARGE NO. D-51100

DESCRIPTION PRESSURE TUBE AT SODIUM-ARGON INTERFACE

NUMBER A-5
SHEET 4 OF 20
DATE 12-12-72 BY HEILKER
CHECK DATE ______ BY ______

GEOMETRY



IT IS ASSUMED THAT DURING NORMAL OPERATION OF THE SODIUM HEATED STEAM GENERATOR, THERE WILL BE PERIODS WHEN THE SODIUM LEVEL AT THE SODIUM - ARGON INTERFACE WILL BE STEADY. IT IS ALSO ASSUMED THAT THERE WILL BE PERIODS WHEN THE SODIUM LEVEL IS FLUCTUATING AT SOME RATE. IT IS THEN NECESSARY TO SHOW THAT THE PRESSURE TUBE IS NOT OVERSTRESSED BY CYCLING BETWEEN THESE CONDITIONS. ASSUME 30,000 CYCLES OR APPROXIMATELY ONE CYCLE/8 HOURS. ASSUME 600 CYCLES OF NORMAL STARTUP.

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COMBUSTION ENGINEERING, INC.

ENGINEERING DEPARTMENT, CHATTANOOGA, TENN.

NUMBER A-5

SHEET 5 OF 20

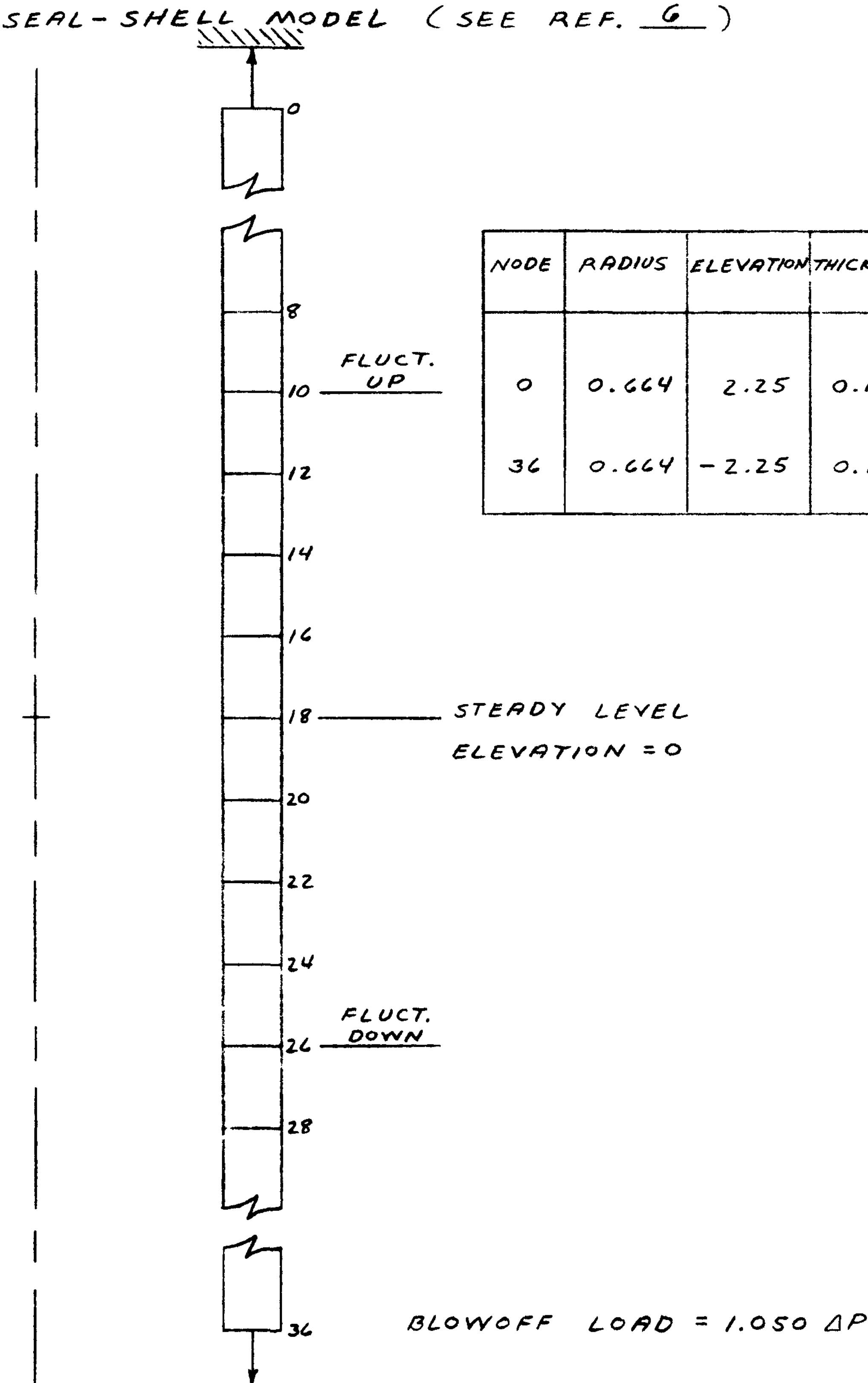
CHARGE NO. D-51100

DATE 12-12-72 BY HEILKER

DESCRIPTION PRESSURE TUBE AT SODIUM-ARGON INTERFACE

CHECK DATE ______ BY ______

SEAL-SHELL MODEL (SEE REF. 6)



NODE	RADIUS	ELEVATION	THICKNESS
0	0.664	2.25	0.172
36	0.664	-2.25	0.172

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LOADING

PRESSURE

FOR THE PURPOSES OF THIS ANALYSIS THE NORMAL OPERATING PRESSURES ARE COMBINED WITH THE THERMAL LOADINGS. THE PRESSURES USED ARE AS FOLLOWS:

STEAM PRESSURE = 2650 PSI

SODIUM PRESSURE = 80 PSI

THERMAL

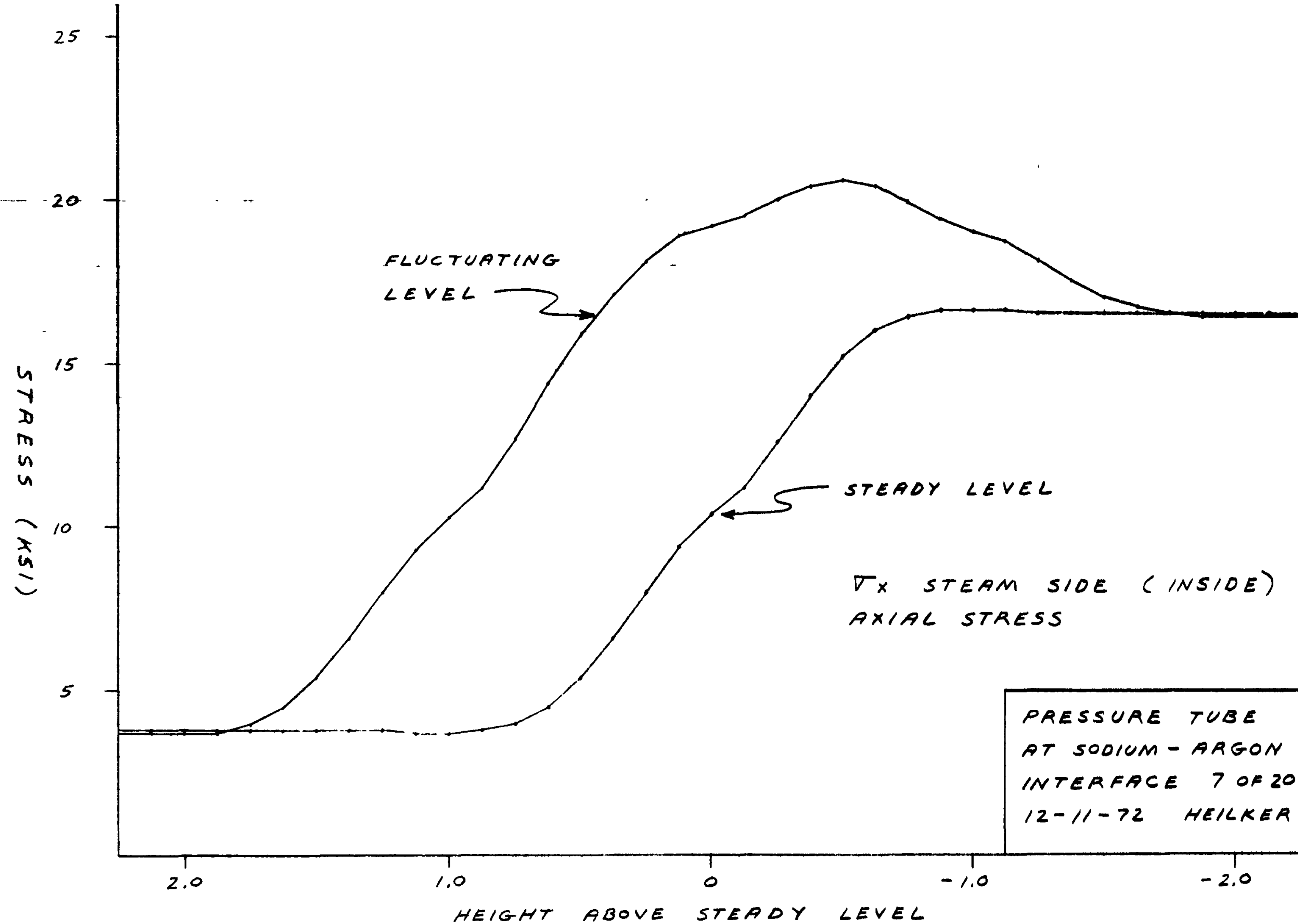
TEMPERATURE DATA IS TAKEN DIRECTLY FROM C.E. CALCULATION "HIGH PRESSURE TUBE SODIUM INTERFACE" DATED 11-2-72 BY DOUG SILVER WHICH APPEARED IN THE NOVEMBER 1972 PROGRESS REPORT.

CONDITION A - STEADY LEVEL

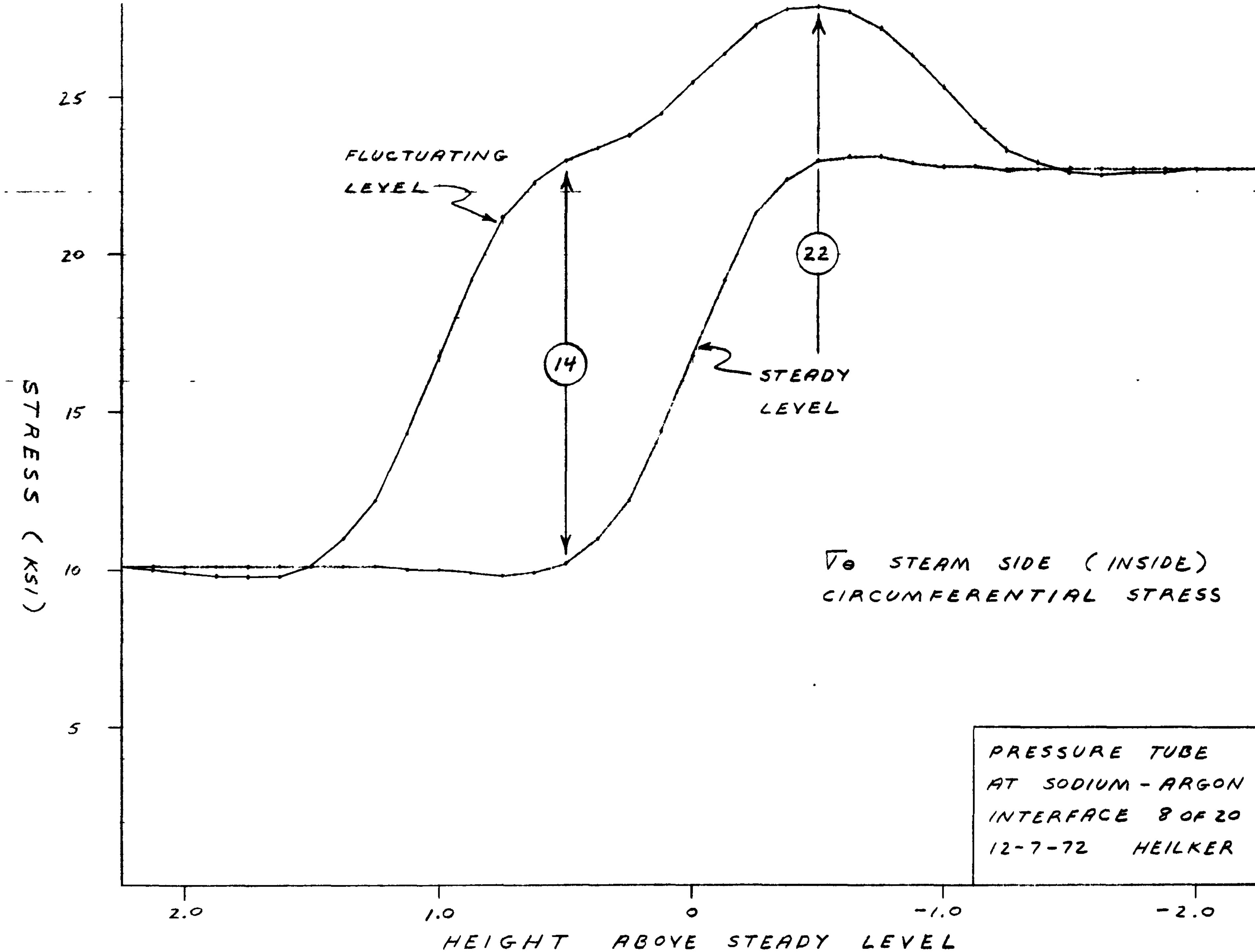
CONDITION B - FLUCTUATING LEVEL

IT WAS FOUND THAT AFTER A NUMBER OF FLUCTUATION, A STEADY TEMPERATURE DISTRIBUTION IS FORMED. REGUARDLESS OF THE FREQUENCY OF FLUCTUATION, THE SAME TEMPERATURE DISTRIBUTION IS FORMED.

CONDITION C - AMBIENT (ZERO STRESS)

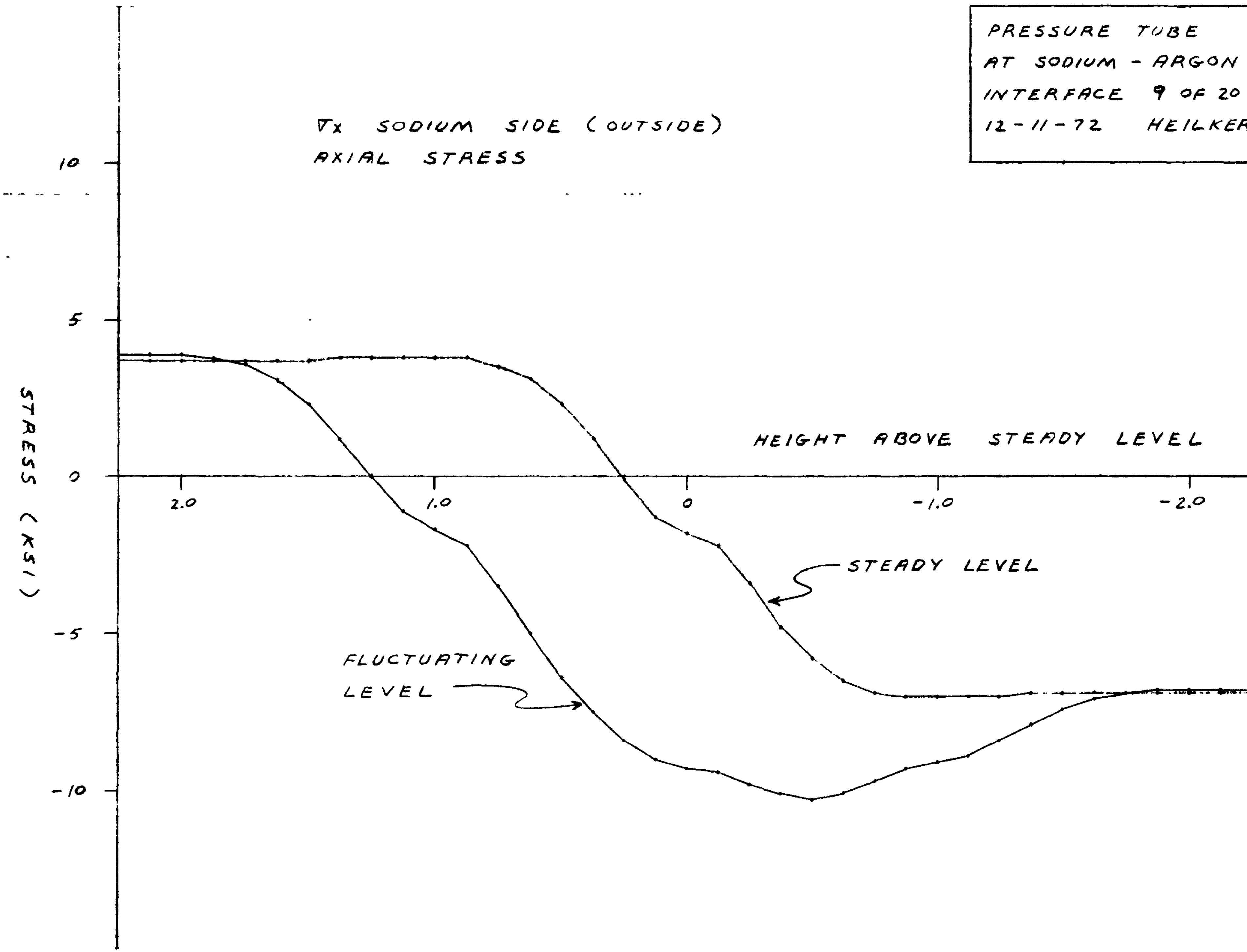


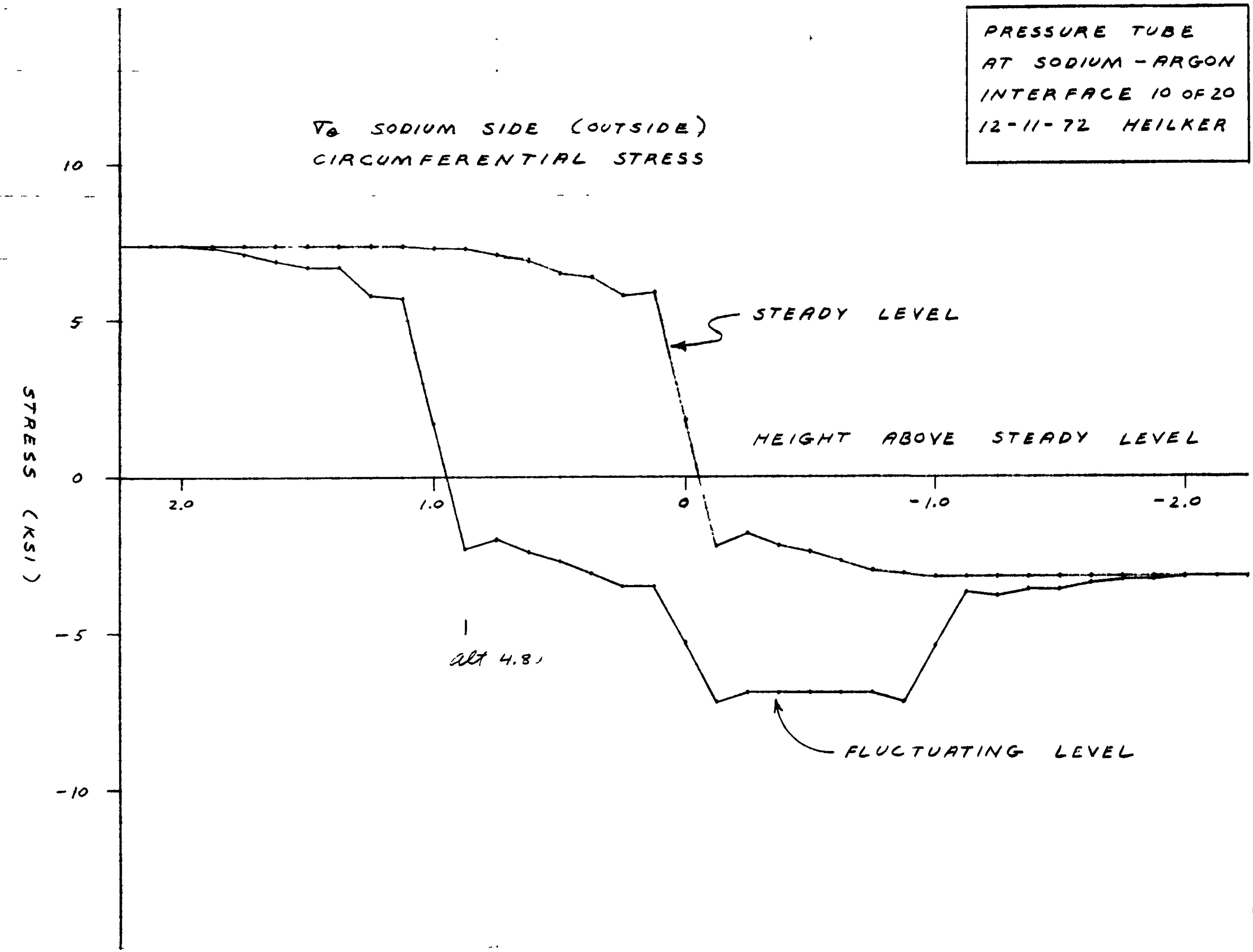
25
20
15
10
5
STRESS (KSI)
FLUCTUATING LEVEL
STEADY LEVEL
σx STEAM SIDE (INSIDE)
AXIAL STRESS
PRESSURE TUBE
AT SODIUM - ARGON
INTERFACE 7 OF 20
12-11-72 HEILKER
2.0
1.0
0
-1.0
-2.0
HEIGHT ABOVE STEADY LEVEL



FLUCTUATING LEVEL
14
22
STEADY LEVEL
STRESS (KSI)
25
20
15
10
5
σ_θ STEAM SIDE (INSIDE) CIRCUMFERENTIAL STRESS
PRESSURE TUBE
AT SODIUM - ARGON
INTERFACE 8 OF 20
12-7-72 HEILKER
2.0
1.0
0
-1.0
-2.0
HEIGHT ABOVE STEADY LEVEL

A-5


PRESSURE TUBE
AT SODIUM - ARGON
INTERFACE 9 OF 20
12-11-72 HEILKER
σ_x SODIUM SIDE (OUTSIDE)
AXIAL STRESS
STRESS (KSI)
10
5
0
-5
-10
HEIGHT ABOVE STEADY LEVEL
2.0
1.0
0
-1.0
-2.0
STEADY LEVEL
FLUCTUATING
LEVEL



PRESSURE TUBE
AT SODIUM - ARGON
INTERFACE 10 OF 20
12-11-72 HEILKER
σ_θ SODIUM SIDE (OUTSIDE)
CIRCUMFERENTIAL STRESS
STEADY LEVEL
HEIGHT ABOVE STEADY LEVEL
FLUCTUATING LEVEL
STRESS (KSI)
10
5
0
-5
-10
2.0
1.0
0
-1.0
-2.0
alt 4.8)

A-5

PRIMARY STRESS (LOAD CONTROLLED)

ASSUME A DESIGN PRESSURE OF 2700 PSI @ 868°F

$$\sigma_x = \frac{b^2 P}{2Rt} = 1.46\,P = 3.94 \text{ KSI}$$

$$\sigma_\theta = \frac{bP}{t} = 3.36\,P = 9.07 \text{ KSI}$$

$$\sigma_r = -\frac{P}{2} = -0.50\,P = -1.35 \text{ KSI}$$

$$S.I._{max} = \sigma_\theta - \sigma_r = 9.07-(-1.35) = \underline{10.42 \text{ KSI}}$$

FROM SEC. VIII, ALLOWABLE $S = \underline{12.3 \text{ KSI}}$ @ 868°F

PRIMARY + SECONDARY STRESS

FOR THE CONSIDERATION OF SECONDARY STRESSES, TWO LOCATIONS WILL BE EXAMINED, NODE 14 AND NODE 22 (SEE SHEET 5). EXAMINATION OF THE STRESS CURVES ON THE PRECEDING SHEET REVEALS THAT THE CRITICAL STRESS INTENSITY IS $(\sigma_\theta - \sigma_r)$ ON THE INSIDE SURFACE OF THE TUBE.

NODE 14 — LOCATION OF MAXIMUM RANGE OF SECONDARY STRESS BETWEEN CONDITION A AND CONDITION B.

NODE 22 — LOCATION OF MAXIMUM RANGE OF SECONDARY STRESS.

NOTE THAT THE PRIMARY STRESS IS THE SAME AT ALL LOCATIONS. $(P_L + P_B/K) = P_m$

PRIMARY + SECONDARY STRESS CONT'D

NODE 14 (INSIDE)

CONDITION A - STEADY

$T_m = 674\,°F$

$S_Y = 22.2$ KSI

$\sigma_\theta = 10.2$ KSI

$S.I. = \sigma_\theta - \sigma_r = 12.85$ KSI

$\sigma_m = 3.36\,(2.57) + 2.65$

$= 11.29$ KSI

$\sigma_T = 1.56$ KSI

CONDITION B - FLUCT.

$T_m = 734\,°F$

$S_Y = 21.8$ KSI

$\sigma_\theta = 23.0$ KSI

$S.I. = \sigma_\theta - \sigma_r = 25.65$ KSI

$\sigma_m = 11.29$ KSI

$\sigma_T = 14.36$

CONDITION C - AMBIENT

$\sigma = 0$

$$\left[\left(P_L + \frac{P_b}{K} \right)_{MAX} + Q_{RANGE} \right] \leq S_q$$

$$\left(\sigma_{m\,MAX} + \sigma_{T\,RANGE} \right) \leq S_{Y\,AVG.}$$

$$(11.29 + 14.36) \leq 22.0$$

BUT $\underline{25.65 > 22.0}$ THUS INELASTIC ANALYSIS IS REQUIRED

FOR CONDITION B

$x = \sigma_m / \sigma_Y = 0.518$

$y = \sigma_T / \sigma_Y = 0.659$ (SEE SHEET 14)

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DESCRIPTION PRESSURE TUBE AT SODIUM-ARGON INTERFACE
CHECK DATE ____ BY ____

PRIMARY + SECONDARY STRESS CONT'D

NODE 14 (INSIDE)

CONDITION A - STEADY

$T_m = 674\,°F$
$S_Y = 22.2$ KSI
$\sigma_\theta = 10.2$ KSI
$S.I. = \sigma_\theta - \sigma_r = 12.85$ KSI
$\sigma_m = 3.36(2.57) + 2.65$
$= 11.29$ KSI
$\sigma_T = 1.56$ KSI

CONDITION B - FLUCT.

$T_m = 734\,°F$
$S_Y = 21.8$ KSI
$\sigma_\theta = 23.0$ KSI
$S.I. = \sigma_\theta - \sigma_r = 25.65$ KSI
$\sigma_m = 11.29$ KSI
$\sigma_T = 14.36$

CONDITION C - AMBIENT

$\sigma = 0$

$$\left[\left(P_L + \frac{P_b}{K}\right)_{MAX} + Q_{RANGE}\right] \leq S_q$$

$$\left(\sigma_{m\,MAX} + \sigma_{T\,RANGE}\right) \leq S_{Y\,AVG.}$$

$$(11.29 + 14.36) \leq 22.0$$

BUT $25.65 > 22.0$ THUS INELASTIC ANALYSIS IS REQUIRED

FOR CONDITION B

$x = \sigma_m / \sigma_Y = 0.518$

$y = \sigma_T / \sigma_Y = 0.659$ (SEE SHEET 14)

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CHARGE NO. D-51100
DESCRIPTION PRESSURE TUBE AT SODIUM-ARGON INTERFACE

NUMBER A-5
SHEET 13 OF 20
DATE 2-2-73 BY HEILKER
CHECK DATE ______ BY ______

PRIMARY + SECONDARY STRESS CONT'D

NODE 22 (INSIDE)

CONDITION A-STEADY	CONDITION B - FLUCT.
$T_m = 734°F$	$T_m = 759°F$
$S_y = 21.8$ KSI	$S_y = 21.6$ KSI
$\sigma_\theta = 23.0$ KSI	$\sigma_\theta = 27.9$ KSI
$S.I. = \sigma_\theta - \sigma_r = 25.65$ KSI	$S.I. = \sigma_\theta - \sigma_r = 30.55$ KSI
$\sigma_m = 11.29$ KSI	$\sigma_m = 11.29$ KSI
$\sigma_T = 14.36$ KSI	$\sigma_T = 19.26$ KSI

CONDITION C - AMBIENT

$\sigma = 0$

$$\left[\left(P_L + \frac{P_b}{K}\right)_{MAX} + Q_{RANGE}\right] \leq S_q$$

$$\left(\sigma_{m\,MAX} + \sigma_{T\,RANGE}\right) \leq S_{Y\,AVG}$$

$$(11.29 + 19.26) \leq 21.7$$

BUT 30.55 > 21.7 THUS INELASTIC ANALYSIS IS REQUIRED

FOR CONDITION B

$X = \sigma_m / \sigma_Y = 0.523$

$Y = \sigma_T / \sigma_Y = 0.892$ (SEE SHEET 14)

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CHARGE NO. D-51100

DESCRIPTION PRESSURE TUBE AT SODIUM-ARGON INTERFACE

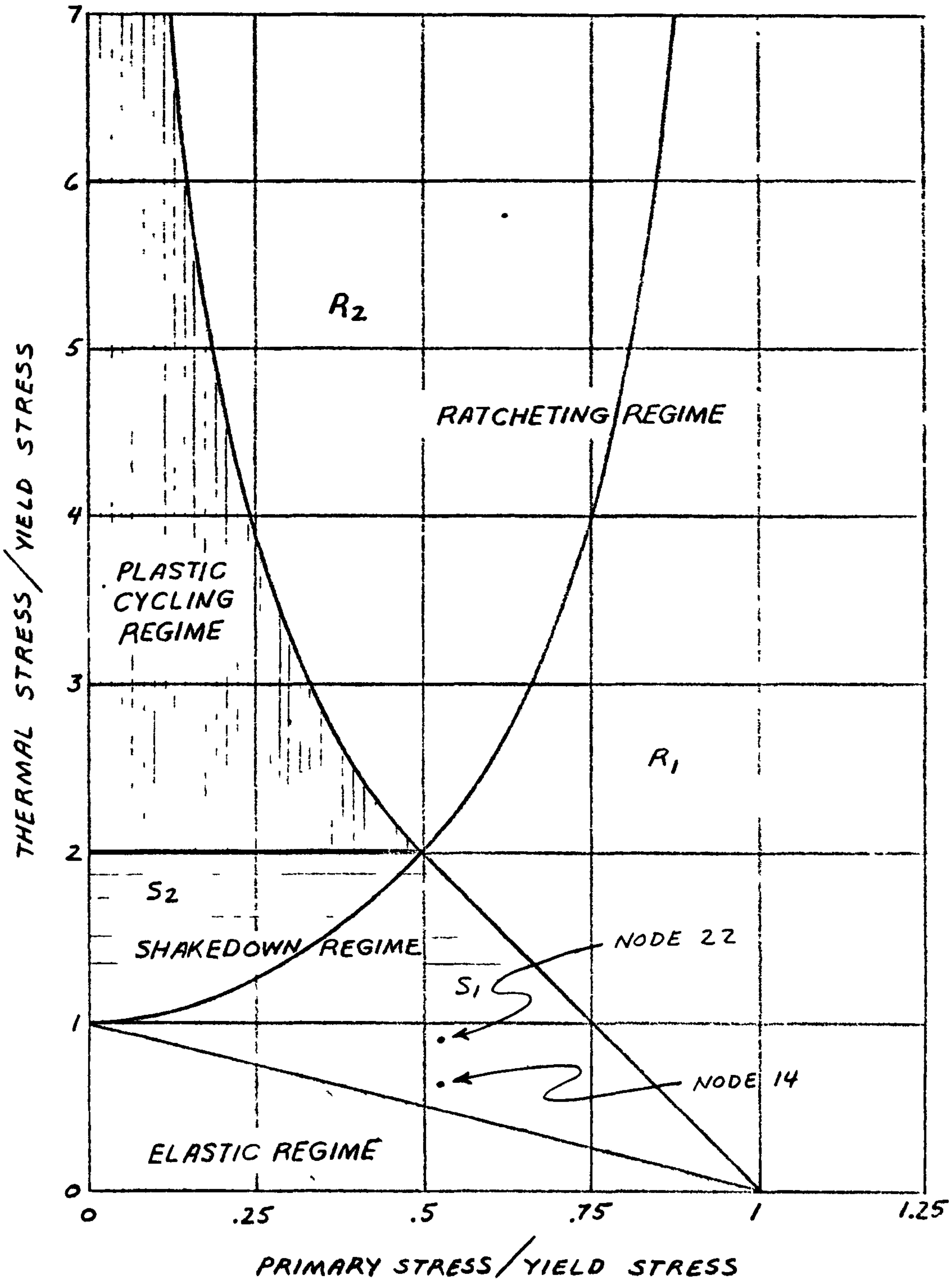
NUMBER A-5

SHEET 14 OF 20

DATE 2-2-73 BY HEILKER

CHECK DATE ______ BY ______

STRESS REGIMES FOR ONE-DIMENSIONAL ELASTIC-PERFECTLY-PLASTIC MODEL



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NUMBER A-5
SHEET 15 OF 20

CHARGE NO. D-51100

DATE 2-2-73 BY HEILKER

DESCRIPTION PRESSURE TUBE AT SODIUM-ARGON INTERFACE

CHECK DATE ____ BY ____

FULL RELAXATION BREE METHOD

IN THE FULL RELAXATION BREE METHOD, IT IS ASSUMED THAT SUFFICIENT CREEP STRAINS OCCUR, WHILE THE WALL IS SUBJECT TO THE MAXIMUM THERMAL GRADIENT, TO RELAX THE STRESSES TO σ_m THROUGHOUT THE WALL. NO RELAXATION TAKES PLACE IN ANY OTHER PART OF THE CYCLE. USING THIS METHOD, RATCHETING OCCURS IN ALL REGIONS, SHOWN IN THE FIGURE ON SHEET 14, EXCEPT THE ELASTIC REGIME. THIS METHOD GIVES CONSERVATIVE PREDICTIONS RELATIVE TO THE MORE RIGOROUS INELASTIC METHODS OF ANALYSIS.

THE PLASTIC STRAIN INCREMENT PER CYCLE = ϵ_g

$\epsilon_g = \frac{\sigma_Y}{E}\,\delta$ WHERE $E = 24.6 \times 10^3$ KSI @ 750°F

$X = \sigma_m / \sigma_Y$

$Y = \sigma_T / \sigma_Y$

$\delta = 1 - X + Y - 2\left[Y(1-X)\right]^{1/2}$ FOR (X, Y) IN S_1

SEE SHEET 14

FULL RELAXATION BREE METHOD CONT'D

NODE 14 (INSIDE)

FROM SHEET 12 :

$X = 0.518$

$Y = 0.659$

$$\delta = 1 - 0.518 + 0.659 - 2\left[0.659\,(1 - 0.518)\right]^{1/2}$$

$\delta = 0.0138$

$$\epsilon_g = \frac{21.8}{24.6 \times 10^3}(0.0138) = \underline{1.22 \times 10^{-5}} \text{ PER CYCLE}$$

NODE 22 (INSIDE)

FROM SHEET 13 :

$X = 0.523$

$Y = 0.892$

$$\delta = 1 - 0.523 + 0.892 - 2\left[0.892\,(1 - 0.523)\right]^{1/2}$$

$\delta = 0.0644$

$$\epsilon_g = \frac{21.6}{24.6 \times 10^3}(0.0644) = \underline{5.66 \times 10^{-5}} \text{ PER CYCLE}$$

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FULL RELAXATION BREE METHOD CONT'D

NODE 14 (INSIDE)

SINCE MOST OF THE THERMAL CYCLING OCCURS BETWEEN CONDITION A AND CONDITION B, IT WILL CONSERVATIVELY BE ASSUMED THAT THERE ARE 30,000 CYCLES OF THE PLASTIC STRAIN INCREMENT CALCULATED ON THE PRECEDING SHEET FOR NODE 14 INSIDE.

$$\epsilon_p = N \epsilon_g = 30{,}000 (1.22 \times 10^{-5}) = 0.366$$

$$\epsilon_p / \epsilon_{ALLOW} = \frac{0.366}{0.02} = 18.3 > 1.0$$

THEREFORE THIS METHOD OF CALCULATING THE INELASTIC PLASTIC STRAIN IS TOO CONSERVATIVE AND A MORE RIGOROUS METHOD IS REQUIRED.

NODE 22 (INSIDE)

SINCE THE STRESS RANGE BETWEEN CONDITION A AND CONDITION B IS SMALL, IT WILL BE ASSUMED THAT THE CYCLING OCCURS BETWEEN CONDITION B AND AMBIENT. THERE ARE 600 CYCLES OF THE PLASTIC STRAIN INCREMENT CALCULATED ON THE PRECEDING SHEET FOR NODE 22 INSIDE.

$$\epsilon_p = N \epsilon_g = 600 (5.66 \times 10^{-5}) = 0.034$$

$$\epsilon_p / \epsilon_{ALLOW} = \frac{0.034}{0.02} = 1.7 > 1.0$$

THEREFORE THIS METHOD OF CALCULATING THE INELASTIC PLASTIC STRAIN IS TOO CONSERVATIVE AND A MORE RIGOROUS METHOD IS REQUIRED.

COMBUSTION ENGINEERING, INC.
ENGINEERING DEPARTMENT, CHATTANOOGA, TENN.

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DESCRIPTION PRESSURE TUBE AT SODIUM-ARGON INTERFACE

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SHEET 18 OF 20
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LONG TERM CREEP (SECONDARY)

THE CREEP STRAIN WILL BE UNIFORM THROUGHOUT THE TUBE. IT IS CAUSED BY THE CONSTANT PRIMARY STRESS LOADING OVER THE UNIT'S 30 YEAR LIFETIME.

FOR A PRIMARY STRESS LEVEL OF 11.29 KSI,

AND A TEMPERATURE LEVEL OF 750°F,

THE SECONDARY CREEP IS NEGLIGIBLE.

THUS $\epsilon_c = 0$.

COMBUSTION ENGINEERING, INC.
ENGINEERING DEPARTMENT, CHATTANOOGA, TENN.

CHARGE NO. D-51100

DESCRIPTION PRESSURE TUBE AT SODIUM-ARGON INTERFACE

NUMBER A-5
SHEET 19 OF 20
DATE 2-2-73 BY HEILKER
CHECK DATE ______ BY ______

FATIGUE

NODE 14 (INSIDE)

(CONDITION B-FLUCT.) TO (AMBIENT)

STRESS RANGE = 25.65 − 0 = 25.65 KSI

ALTERNATING STRESS $= \frac{25.65}{2} = 12.83$ KSI

FROM REF. 1 FIG. I-9-1

$S_a = \frac{E_{30}}{E} \nabla = \frac{30}{24.6}(12.83) = 15.65$ KSI

NUMBER OF CYCLES = 300,000

$n/N_d = \frac{600}{300,000} = 0.0020$

(CONDITION B-FLUCT.) TO (CONDITION A-STEADY)

STRESS RANGE = 25.65 − 12.85 = 12.80 KSI

ALTERNATING STRESS $= \frac{12.80}{2} = 6.40$ KSI

$S_a = \frac{E_{30}}{E} \nabla = \frac{30}{26.4}(6.40) = 7.27$ KSI $<$ 13.0 KSI

$n/N_d = 0$

$\sum n/N_d = 0.0020 + 0 = \underline{0.0020}$

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FATIGUE CONT'D

NODE 22 (INSIDE)

(CONDITION B-FLUCT.) TO (AMBIENT)

STRESS RANGE = 30.55 − 0 = 30.55 KSI

ALTERNATING STRESS $= \frac{30.55}{2} = 15.28$ KSI

FROM REF. 1 FIG. I-9-1

$S_a = \frac{E_{30}}{E} \sigma = \frac{30}{24.6}(15.28) = 18.63$ KSI

NUMBER OF CYCLES = 120,000

$n/N_d = \frac{600}{120{,}000} = 0.0050$

(CONDITION B-FLUCT.) TO (CONDITION A-STEADY)

STRESS RANGE = 30.55 − 25.65 = 4.90 KSI

ALTERNATING STRESS $= \frac{4.90}{2} = 2.45$ KSI

$S_a = \frac{E_{30}}{E} \sigma = \frac{30}{24.6}(2.45) = 2.99$ KSI < 13 KSI

$n/N_d = 0$

$\sum n/N_d = 0.0050 + 0 = \underline{0.0050}$

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APPENDIX A-6

SODIUM WATER REACTION

DEMO PLANT

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REPORT NO. ________

COMBUSTION ENGINEERING, INC.

CHATTANOOGA DIVISION

NUCLEAR COMPONENTS DEPARTMENT

TITLE

SODIUM-WATER REACTION

A-6
CALCULATION NO

DATED

D-51100
CONTRACT NO.

DEMO PLANT
STEAM GENERATOR
UNIT DESIGNATION

Gordon G. Chen
PREPARED

CHECKED

REVIEWED

CHANGES

DATE	REV. BY	PAGES AFFECTED			DESCRIPTION OF CHANGE
		REVISED	REMOVED	ADDED	

TABLE OF CONTENTS

1. ABSTRACT

Tube leakage may occur sometime during the life of the sodium-heated steam generator. The reaction between sodium and water is a potential hazard to both personnel and equipment. The objective of the analysis is to determine the dynamic response of the vessel shell and the flow baffle of the evaporator and the superheater of the steam generator by numerical analysis of the deformation of these two elements under the reaction pressures resulting from the rupture of the tubes in the evaporator or in the superheater and then further determine the integrity of the steam generator. The effect of volume increase in the vessel where the sodium-water reaction takes place has not been considered in developing the dynamic pressures. However, the results are known to be conservative. This analysis has concluded a satisfactory result for both units of the steam generator under the given dynamic loads.

2. SIGNIFICANT RESULTS

As shown in Figures 5 and 6 and tables in Section 5.e., the maximum displacements of the vessel shells and the baffles are found to be as follows:

	Max. Displacement of the Flow Baffle	Max. Displacement of the Vessel Shell
Evaporator	Exceeded $r^* = 1.0898$	$1.01538 < r^* = 1.1157$
Superheater	$1.038409 < r^* = 1.0887$	---

It is noticed that the baffle in the evaporator has ruptured, however, the vessel shell sustained the load. The displacement of the baffle in the superheater stayed within the strain limit of 0.0887".

The calculated required thicknesses for the vessels under the maximum cover gas steady pressures of 450 psi and 150 psi are 0.9187" and 0.2253" for the evaporator shell and the superheater shell respectively. Both are less than the actual thickness of 1.25".

These results assure the integrity of these two vessels subjected to the sudden increase of internal pressure due to sodium-water reaction.

3. GENERAL DISCUSSION

The structure considered in this analysis is shown in Section 5.a. The shells and the flow baffles were analyzed for the dynamic pressure described in Section 5.b., which was developed from the BUG-3 Computer Code by NPD; only the highest peak pressure inside each vessel was considered.

The procedure used in evaluating the dynamic response of the vessels and the baffles was as follows:

(a) The equation of motion for the flow baffle was evaluated numerically by the Runge-Kutta method. The baffle was allowed to expand with time until it reached its maximum displacement or it exceeded the critical radius and ruptured.

(b) If the flow baffle had reached its maximum displacement before it ruptured, the analysis was concluded, no further study on the dynamic response of the shell would be needed.

(c) If the flow baffle had ruptured before reaching a maximum displacement then the equation of motion for the vessel shell was evaluated by the same numerical method, starting from the time when the baffle was ruptured. The shell was allowed to expand with time until either the shell reached its maximum displacement or it exceeded the radius at instability and ruptured.

The displacement versus time curves for the shell and the baffles are shown in Figures 5 and 6. The values are taken from tables in Section 5.e.

4. REFERENCES

1. Costantino, C. J., "The Strength of Thin-Walled Cylinders Subjected to Dynamic Internal Pressures," Journal of Applied Mechanics, March 1965, Transaction of ASME, Vol. 87, pp. 104-108.

2. Biggs, J. M., Introduction to Structural Dynamics, McGraw-Hill, Inc. 1964.

3. Smith, G. V., "Supplemental Report on the Elevated-Temperature Properties of Chromium-Molybdenum Steels," ASTM Data Series Publication DS 6S2-40006-002, March 1971.

4. ASME Boiler and Pressure Vessel Code, Section III for Nuclear Power Plant Components.

5. C. E. Drawing D. 63672679-3, D. 63672684-2.

6. Nielsen, Kaj L., Methods in Numerical Analysis, The MacMillan Company, New York, 1960.

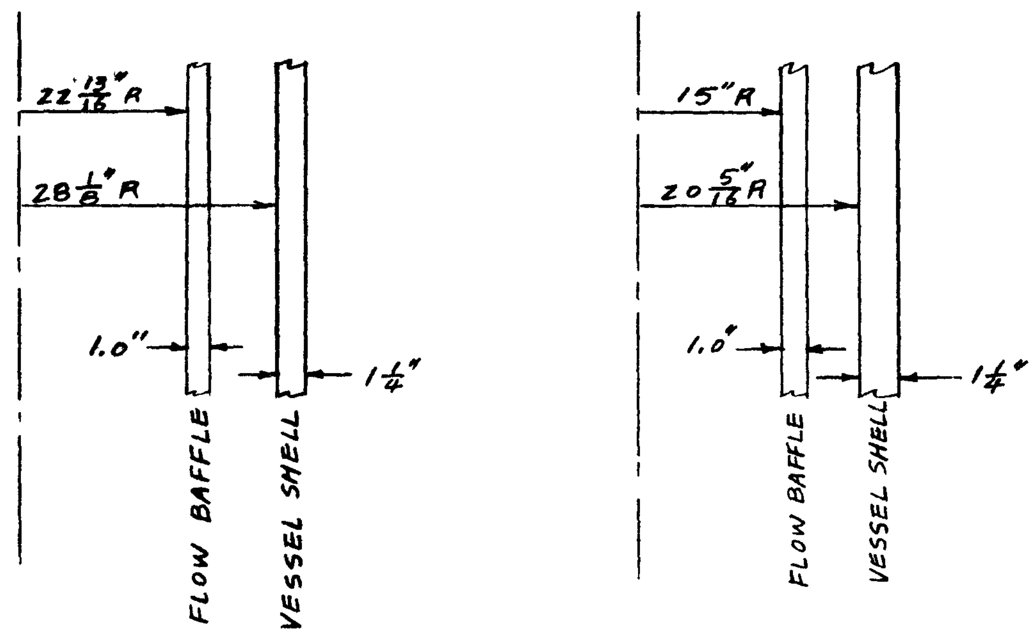
7. ASME Boiler and Pressure Vessel Code, Section I for Power Boilers.

8. ASME Boiler and Pressure Vessel Code, Section VIII for Pressure Vessels.

5.) DETAILED ANALYSIS

a) SYSTEM GEOMETRY

A CROSS SECTION OF THE VESSEL SHELL AND FLOW BAFFLE OF THE EVAPORATOR AND SUPERHEATER SECTIONS ARE SHOWN BELOW. CERTAIN DIMENSIONS ARE GIVEN TO FACILITATE THE ANALYSIS.



EVAPORATOR SECTION

SUPERHEATER SECTION

MATERIAL:

FLOW BAFFLE : SA-240 TYPE 405 MOD ½ Mo

VESSEL SHELL : SA-387 GR.D

MATERIAL:

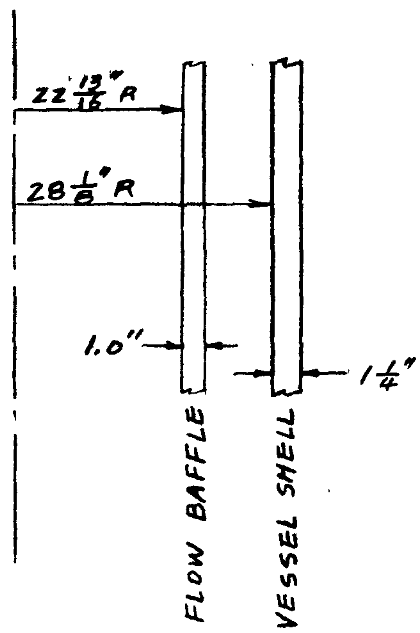
FLOW BAFFLE : SA-240 TYPE 405 MOD ½ Mo

VESSEL SHELL : SA-387 GR.D

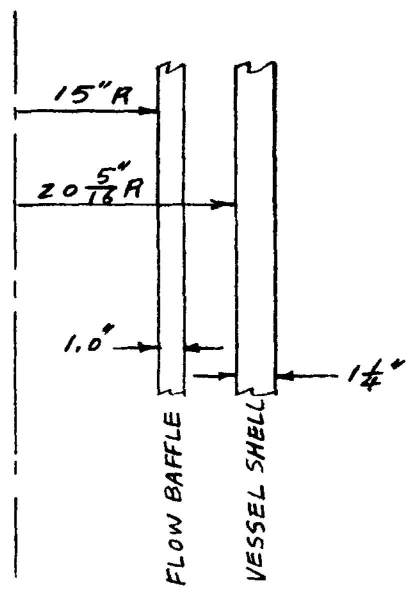
5.) DETAILED ANALYSIS

a) SYSTEM GEOMETRY

A CROSS SECTION OF THE VESSEL SHELL AND FLOW BAFFLE OF THE EVAPORATOR AND SUPERHEATER SECTIONS ARE SHOWN BELOW. CERTAIN DIMENSIONS ARE GIVEN TO FACILITATE THE ANALYSIS.



EVAPORATOR SECTION



SUPERHEATER SECTION

MATERIAL:

FLOW BAFFLE : SA-240 TYPE 405 MOD ½ Mo

VESSEL SHELL : SA-387 GR.D

MATERIAL:

FLOW BAFFLE : SA-240 TYPE 405 MOD ½ Mo

VESSEL SHELL : SA-387 GR.D

5.) DETAILED ANALYSIS

b) SYSTEM ALLOWABLE

THE OBJECTIVE OF THIS CALCULATION IS TO ASSURE THAT THE SYSTEM MEETS THE FOLLOWING REQUIREMENTS.

1. THE THICKNESSES OF THE VESSEL SHELLS ARE TO BE GREATER THAN $t_{REQUIRED} = \frac{PR_i}{S_m - .5P}$, WHERE P IS THE COVER GAS PRESSURE AFTER THE RUPTURE DISC BURSTS. (SEE I-110, REF. 5)

2. THE MAXIMUM DISPLACEMENTS OF THE VESSEL SHELLS OF THE EVAPORATOR AND THE SUPERHEATER SECTIONS DO NOT EXCEED THEIR CRITICAL VALUES.

5) DETAILED ANALYSIS

C) SYSTEM LOADING

THE DYNAMIC PRESSURE DURING THE SODIUM-WATER REACTION INSIDE THE VESSELS ARE PRESENTED IN FIGURES 1 AND 3 FOR EVAPORATOR AND SUPERHEATER RESPECTIVELY. INDICATED IN THE FIGURES ARE DYNAMIC PRESSURE VS. TIME CURVES OF DIFFERENT ELEVATIONS IN THE VESSELS. ONLY THE CURVE WITH THE HIGHEST PRESSURE, WHICH OCCURED AT THE BOTTOM OF THE VESSELS, ARE CHOSEN AS THE DYNAMIC LOADING FOR THE ANALYSIS.

FIGURES 2 AND 4 SHOW THE COVER GAS PRESSURES OF THE EVAPORATOR AND THE SUPERHEATER. FOR THE CONSIDERATION OF REQUIREMENT 1 OF SEC. 5.b. A VALUE OF 450 PSI IS DETERMINED FOR THE EVAPORATOR SECTION AND 150 PSI FOR THE SUPERHEATER AS DESIGN PRESSURES FROM THESE FIGURES.

5.) DETAILED ANALYSIS

c.) SYSTEM LOADING

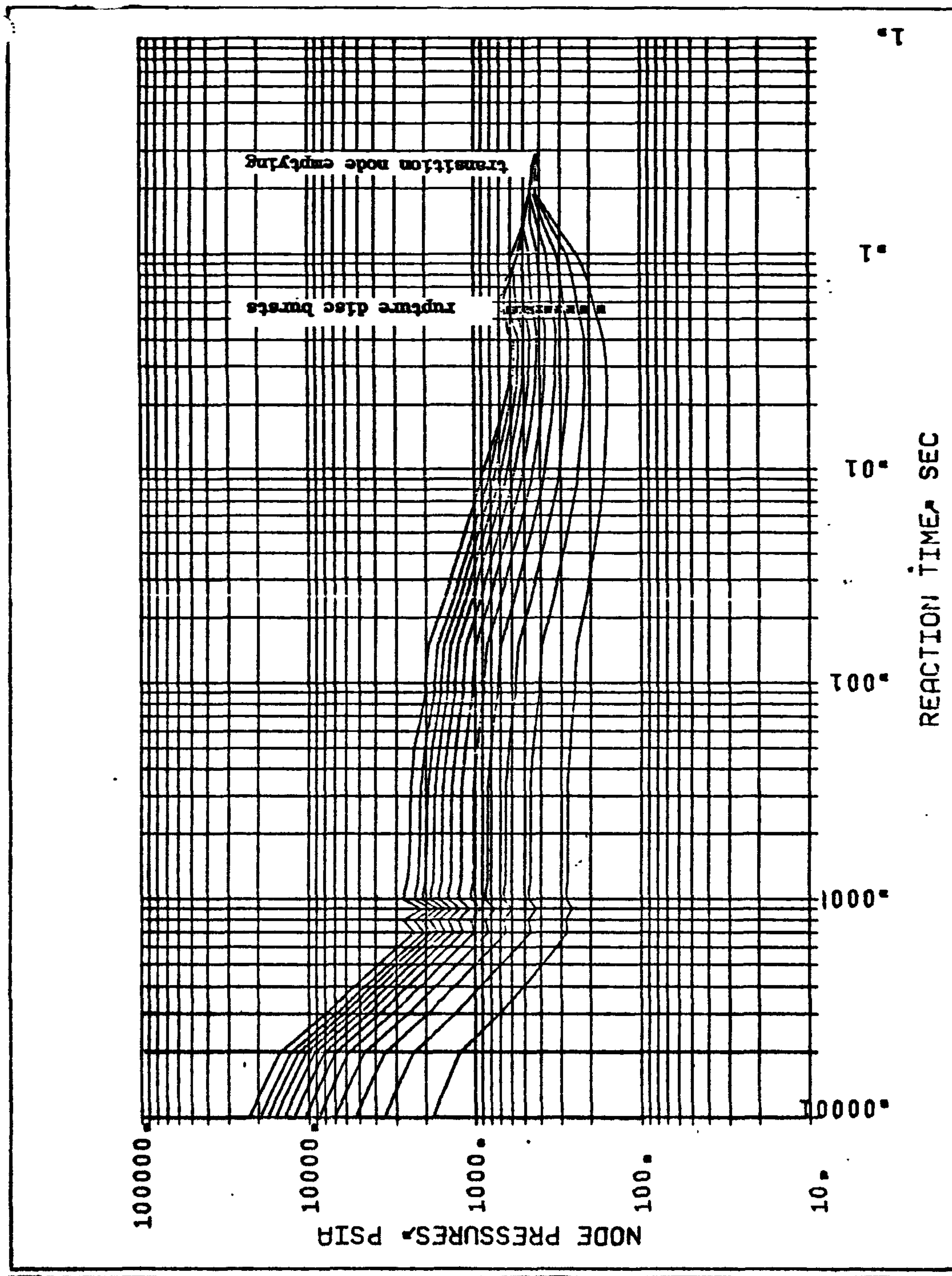


FIGURE 1 DYNAMIC PRESSURE IN EVAPORATOR SECTION

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5) DETAILED ANALYSIS

C) SYSTEM LOADING

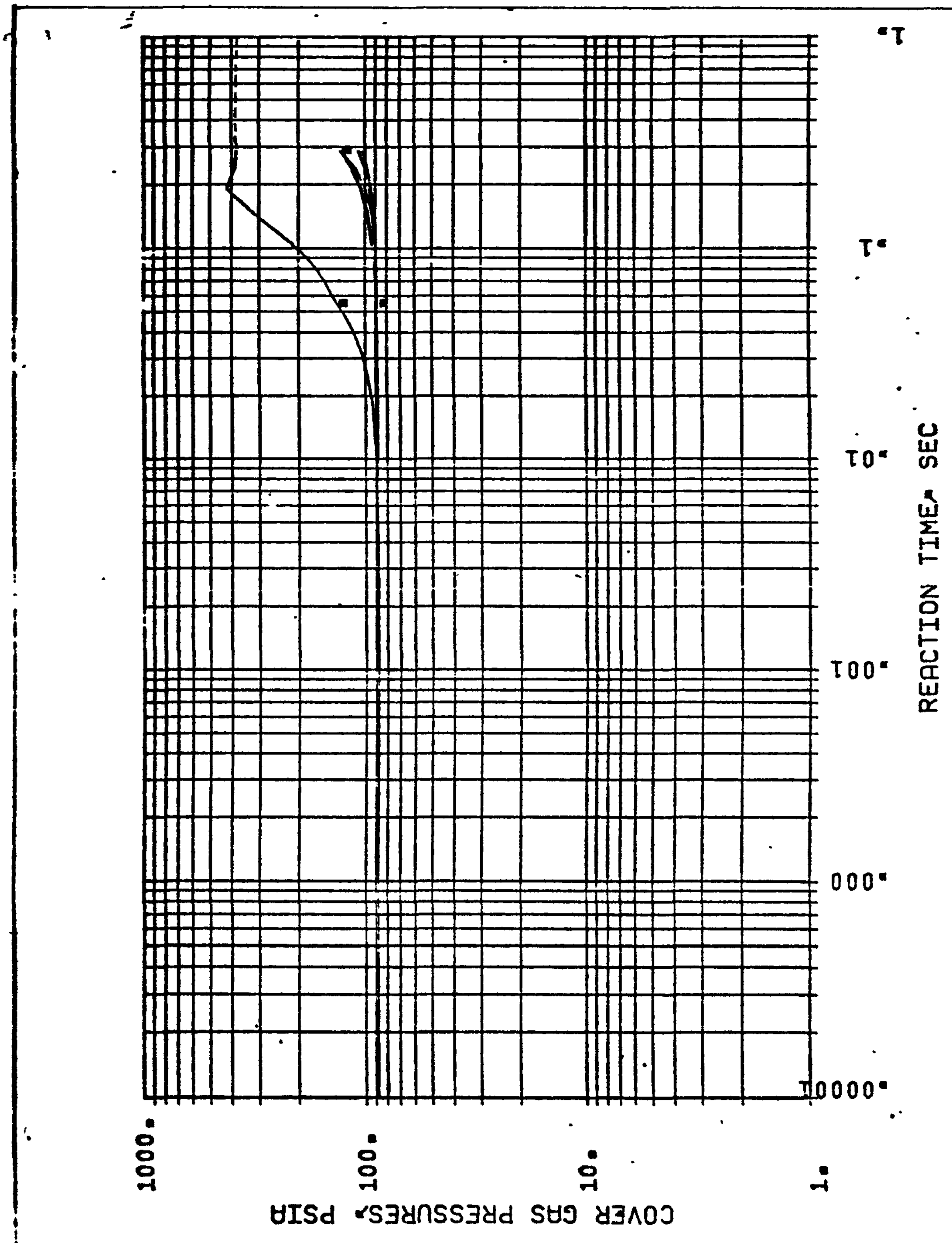

FIGURE 2 COVER GAS PRESSURE IN EVAPORATOR SECTION

5) DETAILED ANALYSIS

C) SYSTEM LOADING

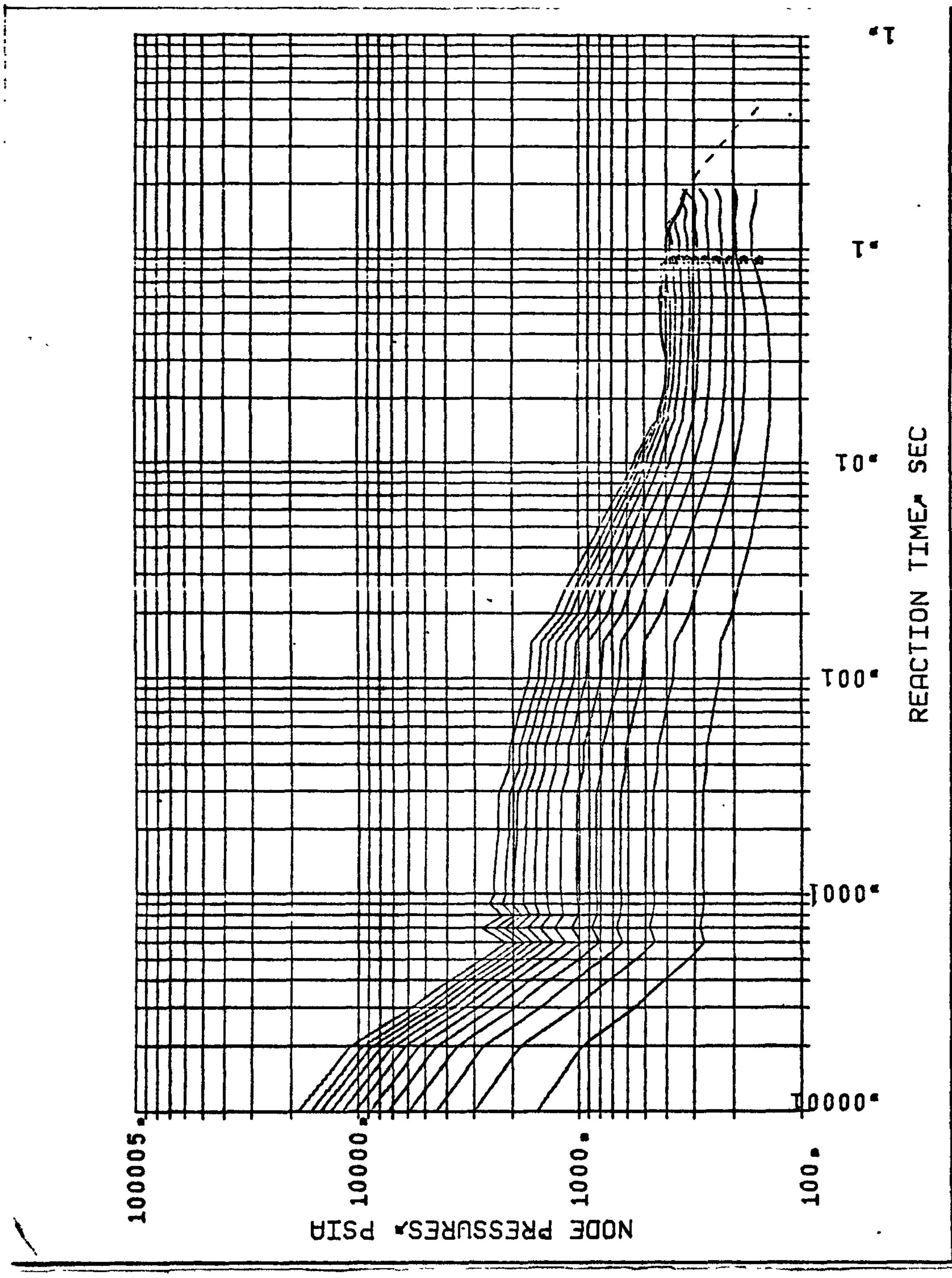

FIGURE 3 DYNAMIC PRESSURE IN SUPERHEATER SECTION

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5) DETAILED ANALYSIS

C) SYSTEM LOADING

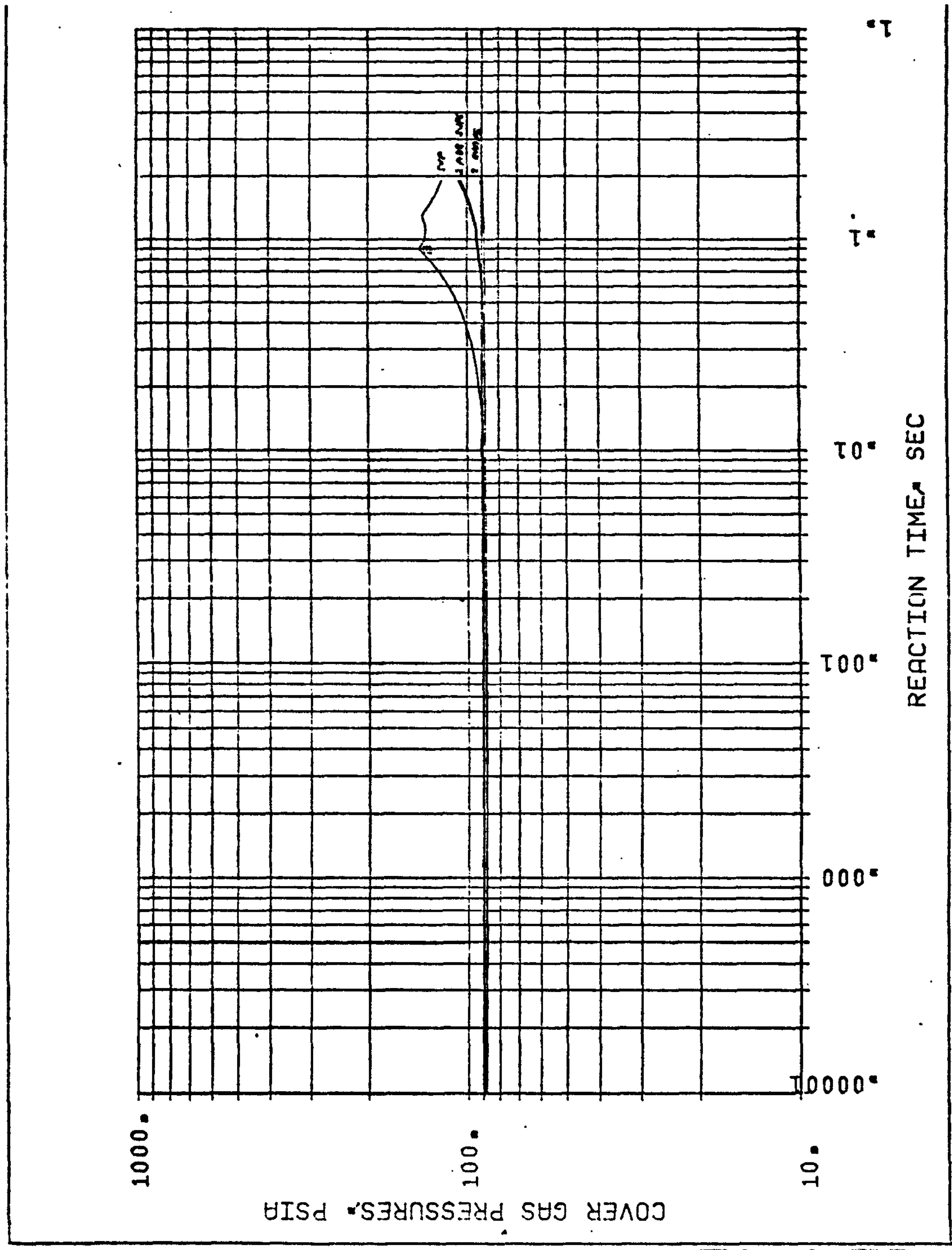

FIGURE 4 COVER GAS PRESSURE IN SUPERHEATER SECTION

5) DETAILED ANALYSIS

d) METHOD OF ANALYSIS

d-1) NOMENCLATURE

NOMENCLATURE

H_o = INITIAL SHELL THICKNESS, IN

$h = H/H_o$, DIMENSIONLESS SHELL THICKNESS

n = STRAIN-HARDEN EXPONENT

P_d = APPLIED DYNAMIC INTERNAL PRESSURE PSI

P_s = APPLIED STATIC INTERNAL PRESSURE PSI

$p_d = P_d R_o / \sigma_o H_o$ DIMENSIONLESS DYNAMIC PRESSURE RATIO

$p_s = P_s R_o / \sigma_o H_o$ DIMENSIONLESS INTERNAL PRESSURE RATIO

R = SHELL RADIUS, IN

R_o = INITIAL SHELL RADIUS, IN

$r = R/R_o$ DIMENSIONLESS SHELL RADIUS

t = TIME SCALE, SEC

$T = [\gamma R_o^2 / 2\sigma_o]^{1/2}$ TIME FACTOR, SEC.

t_o = INITIAL TIME, SEC

γ = UNIT MASS OF MAT'L

σ_o = STRENGTH MEASURE, PSI

$\tau = t/T$ DIMENSIONLESS TIME SCALE

V_o = INITIAL VELOCITY

r^* = INSTABILITY DIMENSIONLESS RADIUS

E = MODULUS OF ELASTICITY.

5) DETAILED ANALYSIS

d) METHOD OF ANALYSIS

d-2) ASSUMPTIONS

a) MATERIAL IS INCOMPRESSIBLE (ELASTIC DEFORMATION NEGLECTED).

b) PLANE STRAIN FOR LONG CYLINDERS ($\epsilon_{zz} = 0$)

c) LUDWISK POWER-LOW STRAIN HARDING ($\sigma = \sigma_0 \epsilon^n$)

d) VONMISES FLOW CRITERIA IS USED.

e) NO INTERACTION BETWEEN PRESSURE VERSUS TIME CURVE AND INCREASE OF VOLUME DUE TO DEFORMATION OF VESSEL SHELL IS CONSIDERED.

h) IN DETERMINING THE RESPONSE OF THE THIN CYLINDERS, THE HIGHEST INSTANTANEOUS PEAK PRESSURE IN THE SECTION IS TAKEN AND IS ASSUMED TO BE RADIALLY AND AXIALLY SYMMETRICAL.

5) DETAILED ANALYSIS

d) METHOD OF ANALYSIS

d-3) MATERIAL PROPERTIES

THE TYPES OF MATERIAL USED AND THEIR PROPERTIES ARE TABULATED AS FOLLOWS, BASED ON THE DESIGN TEMPERATURES SHOWN.

ELEMENT	TYPE OF MATERIAL	DESIGN TEMP. (F)	σ_0 KSI	γ lb/in^3	n	S_y KSI	E KSI
EVAPORATOR BAFFLE	SA-240 TYPE 405 MOD ½ Mo	866	65.4	0.000734	0.172	18.8	26.0×10^3
EVAPORATOR VESSEL SHELL	SA-387 GR. D	866	98.5	0.00073	0.219	20.92	24.55×10^3
SUPERHEATER BAFFLE	SA-240 TYPE 405 MOD ½ Mo	960	58.70	0.000734	0.170	16.95	25×10^3
SUPERHEATER VESSEL SHELL	SA-387 GR. D	960	101.92	0.00073	0.227	20.32	24.8×10^3

THE σ_0 & n VALUES ARE OBTAINED AS ILLUSTRATED IN THE FOLLOWING EXAMPLE FOR THE MATERIAL OF SA-240 TYPE 405 AT 866.

AT YIELD POINT $S_y = \sigma_1 = 18.8$ KSI $E = 26\times10^3$ KSI

$E = 26\times10^3$ KSI

$\epsilon_1 = \sigma_1/E = 0.723\times10^{-3}$

AT RUPTURE POINT $S_U = \sigma_2 = 50$ KSI

% ELONG. $= \epsilon_2 = .21$

SINCE $\sigma = \sigma_0 \epsilon^n$ [SEE. 5. C-2). ASSUMPTION C]

SOLVING SIMULTANEOUSLY: $18.8 = \sigma_0 (0.723\times10^{-3})^n$

$50 = \sigma_0 (0.21)^n$

WE HAVE $n = 0.172$ & $\sigma_0 = 65.4$ KSI

5.) DETAILED ANALYSIS

d) METHOD OF ANALYSIS

d-4) FORMULATION AND COMPUTATION

THIN-WALLED CYLINDRICAL SHELL SUBJECTED TO UNIFORM INTERNAL PRESSURE

THE EQUIVALENT STATIC PRESSURE-RADIUS RELATION FOR THIN WALLED CYLINDRICAL SHELL WRITTEN IN A DIMENSIONLESS FORM IS

$$P = \left(\frac{2}{\sqrt{3}}\right)^{n+1} \frac{\ln^n r}{r^2} \qquad \text{(REF. 1)} \qquad (1)$$

THE EQUATION OF MOTION FOR THE SYSTEM, IN A DIMENSIONLESS FORM, IS GIVEN AS

$$\frac{d^2 r}{d\tau^2} = \frac{r}{2}(P_d - P_s) \qquad \text{(REF. 1)} \qquad (2)$$

SUBSTITUTING P_s BY THE EXPRESSION FROM EQUATION (1) LEADS TO THE FOLLOWING RELATION

$$\frac{d^2 r}{d\tau^2} = \frac{r}{2}\left[P_d - \left(\frac{2}{\sqrt{3}}\right)^{n+1} \frac{\ln^n r}{r^2}\right] \qquad (3)$$

THE DIMENSIONLESS SHELL RADIUS AT INSTABILITY, r^*, IS FOUND BY SETTING THE DERIVATIVE OF P_s WITH RESPECT TO r EQUAL TO ZERO.

$$\frac{dP_s}{dr} = 0 = \left[2r(\ln r) - r^2 n(\ln r)^{n-1}\frac{1}{r}\right] / r^4$$

i.e. $2 \ln r = n$

OR $r^* = e^{n/2}$

5) DETAILED ANALYSIS

d) METHOD OF ANALYSIS

d-4) FORMULATION AND COMPUTATION

EQUATION (3) IS SOLVED BY NUMERICAL INTEGRATION USING RUNGE-KUTTA METHOD (REF. 6). THE RESULTS OF WHICH ARE PRESENTED IN SECTION 5.e.

THE VALUES OF INSTABILITY RADIUS OF THE ELEMENTS IN THIS ANALYSIS ARE LISTED AS FOLLOWS.

ELEMENT	R_o (IN)	n	r^*	UNIT
FLOW BAFFLE	23.3125	0.17	1.0898	EVAPORATOR
	15.500	0.172	1.0887	SUPERHEATER
VESSEL SHELL	28.75	0.219	1.1157	EVAPORATOR
	20.9375	0.227	1.1210	SUPERHEATER

IF A THIN CYLINDER FAILURE IS TO BE AVOIDED, THE RADIUS OF THE CYLINER CAN NOT EXCEED THE RADIUS AT INSTABILITY DEFINED ABOVE.

A COMPUTATIONAL FORM FOR A SECOND-ORDER DIFFERENCIAL EQUATION, OBTAINED FROM REF. 6 WITH SLIGHT CHANGE IN THE NOTATIONS, IS PRESENTED IN THE FOLLOWING PAGE FOR REFERENCE.

COMBUSTION ENGINEERING, INC.
ENGINEERING DEPARTMENT, CHATTANOOGA, TENN.
NUMBER A-6
CHARGE NO. D-51100
SHEET 18 OF 27
DATE 2-14-73 BY CHEN
DESCRIPTION SODIUM-WATER REACTION
CHECK DATE [illegible] BY [illegible]

5) DETAILED ANALYSIS

d) METHOD OF ANALYSIS

d-4) FORMULATION AND COMPUTATION

COMPUTATIONAL FORM OF THE RUNGE-KUTTA METHOD   D.E. : $\frac{d^2 r}{d\tau^2} = f(\dot r, r, \tau)$

INITIAL CONDITION : $r_0, \dot r_0, \tau_0$

τ	r	$\dot r$	$\ddot r$
$\tau_{11} = \tau_0$ $\tau_{12} = \tau_{11} + \Delta\tau/2$ $\tau_{13} = \tau_{11} + \Delta\tau/2$ $\tau_{14} = \tau_{11} + \Delta\tau$	$r_{11} = r_0$ $r_{12} = r_{11} + \dot r_{11} \Delta\tau/2$ $r_{13} = r_{11} + \dot r_{12} \Delta\tau/2$ $r_{14} = r_{11} + \dot r_{13} \Delta\tau$	$\dot r_{11} = \dot r_0$ $\dot r_{12} = \dot r_{11} + \ddot r_{11}$ $\dot r_{13} = \dot r_{11} + \ddot r_{12} \Delta\tau/2$ $\dot r_{14} = \dot r_{11} + \ddot r_{13} \Delta\tau$	$\ddot r_{11} = f(r_0, \dot r_0, \tau)$ $\ddot r_{12} = f(r_{12}, \dot r_{12}, \tau_{12})$ $\ddot r_{13} = f(r_{13}, \dot r_{13}, \tau_{13})$ $\ddot r_{14} = f(r_{14}, \dot r_{14}, \tau_{14})$
		$\Delta r_1 = \frac{\Delta\tau}{6}(\dot r_{11} + 2\dot r_{12} + 2\dot r_{13} + \dot r_{14})$	$\Delta \dot r_1 = \frac{\Delta\tau}{6}(\ddot r_{11} + 2\ddot r_{12} + 2\ddot r_{13} + \ddot r_{14})$
$\tau_{21} = \tau_{11} + \Delta\tau$ $\tau_{22} = \tau_{21} + \Delta\tau/2$ $\tau_{23} = \tau_{21} + \Delta\tau/2$ $\tau_{24} = \tau_{21} + \Delta\tau$	$r_{21} = r_{11} + \Delta r_1$ $r_{22} = r_{21} + \dot r_{21} \Delta\tau/2$ $r_{23} = r_{21} + \dot r_{22} \Delta\tau/2$ $r_{24} = r_{21} + \dot r_{23} \Delta\tau$	$\dot r_{21} = \dot r_{11} + \Delta \dot r_1$ $\dot r_{22} = \dot r_{21} + \ddot r_{21} \Delta\tau/2$ $\dot r_{23} = \dot r_{21} + \ddot r_{22} \Delta\tau/2$ $\dot r_{24} = \dot r_{21} + \ddot r_{23} \Delta\tau$	$\ddot r_{21} = f(r_{21}, \dot r_{21}, \tau_{21})$ $\ddot r_{22} = f(r_{22}, \dot r_{22}, \tau_{22})$ $\ddot r_{23} = f(r_{23}, \dot r_{23}, \tau_{23})$ $\ddot r_{24} = f(r_{24}, \dot r_{24}, \tau_{24})$
		$\Delta r_2 = \frac{\Delta\tau}{6}(\dot r_{21} + 2\dot r_{22} + 2\dot r_{23} + \dot r_{24})$	$\Delta \dot r_2 = \frac{\Delta\tau}{6}(\ddot r_{21} + 2\ddot r_{22} + 2\ddot r_{23} + \ddot r_{24})$
$\tau_{31} = \tau_{21} + \Delta\tau$	$r_{31} = r_{21} + \Delta r_2$	$\dot r_{31} = \dot r_{21} + \Delta \dot r_2$	$\ddot r_{31} = f(r_{31}, \dot r_{31}, \tau_{31})$

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5) DETAILED ANALYSIS

$R_o = 23.3125''$ $\gamma = 0.000734$ lb/IN³ $v_o = 0$ IN/sec

$H_o = 1.0''$ $n = 0.172$

$\sigma_o = 65{,}400$ psi $t_o = 0.0001$ sec.

e) SUMMARY

e-1) EVAPORATOR SECTION (FLOW BAFFLE)

INPUT DATA					OUTPUT DATA				COMMENTS
TIME SEC.	P_{i1} PSIA	P_{i2} PSIA	P_{i3} PSIA	P_{i4} PSIA	DISP. IN	VELOCITY IN/SEC	ACCELER IN/SEC²	R/R_o	
0.00001	23000	19500	19500	16000	23.3125	0.0	3.134×10^7	1.0	
0.00002	16000	12000	12000	8000	23.3139	261.70	2.095×10^7	1.0000598	
0.00003	8000	6700	6700	5400	23.3174	415.60	9.848×10^6	1.0002091	
0.00004	5400	4600	4600	3800	23.3220	495.71	6.180×10^6	1.0004058	
0.00005	3800	3350	3350	2900	23.3272	546.12	3.908×10^6	1.000630	
0.00006	2900	2650	2650	2400	23.3328	578.69	2.609×10^6	1.000872	
0.00007	2400	2550	2550	2700	23.3387	601.07	1.868×10^6	1.001125	
0.00008	2700	2500	2500	2300	23.3449	621.54	2.227×10^6	1.001387	
0.00009	2300	2500	2500	2700	23.3512	640.86	1.637×10^6	1.001658	
0.0001	2700	2650	2650	2600	23.3576	659.76	2.144×10^6	1.0019369	
0.00015	2600	2600	2600	2600	23.3932	759.47	1.854×10^6	1.003462	
0.0002	2600	2600	2600	2600	23.4334	849.22	1.740×10^6	1.005188	
0.00025	2600	2550	2550	2500	23.4780	933.92	1.650×10^6	1.007101	
0.0003	2500	2475	2475	2450	23.5267	1011.08	1.438×10^6	1.009188	
0.00035	2450	2425	2425	2400	23.5790	1079.62	1.305×10^6	1.0114316	
0.00040	2400	2350	2350	2300	23.6346	114.175	1.181×10^6	1.013815	

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ENGINEERING DEPARTMENT, CHATTANOOGA, TENN.
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DATE 2-14-73 BY CHEN
CHECK DATE [illegible] BY [illegible]

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5) DETAILED ANALYSIS

e) SUMMARY

e-1) EVAPORATOR SECTION (FLOW BAFFLE)

INPUT DATA					OUTPUT DATA				COMMENTS
TIME SEC.	P_{i1} PSIA	P_{i2} PSIA	P_{i3} PSIA	P_{i4} PSIA	DISP. IN	VELOCITY IN/SEC	ACCELER IN/SEC^2	R/R_0	
0.0005	2300	2275	2275	2250	23.7542	124.82	9.521×10^5	1.018950	
0.0006	2250	2225	2225	2200	23.8836	133.63	8.123×10^5	1.024447	
0.0007	2200	2150	2150	2100	24.0211	141.11	6.872×10^5	1.030394	
0.0008	2100	2050	2050	2000	24.1653	147.06	5.028×10^5	1.036582	
0.0009	2000	2000	2000	2000	24.3146	151.20	3.267×10^5	1.042985	
0.0010	2000	1995	1995	1990	24.4674	154.43	2.999×10^5	1.049539	
0.0011	1990	1985	1985	1980	24.6231	157.15	2.653×10^5	1.056220	
0.0012	1980	1975	1975	1970	24.7816	159.65	2.362×10^5	1.063016	
0.0013	1970	1965	1965	1960	24.9442	161.89	2.115×10^5	1.069913	
0.0014	1960	1955	1955	1950	25.1053	163.89	1.9075×10^5	1.076901	
0.0015	1950	1945	1945	1940	25.2701	165.71	1.73261×10^5	1.083971	
0.0016	1940	1910	1910	1880	25.4366	167.37	1.5863×10^5	1.091116	← $R/R_0 > r^* = 1.0898$, Rupture of the baffle occurs

COMBUSTION ENGINEERING, INC.
ENGINEERING DEPARTMENT, CHATTANOOGA, TENN.
CHARGE NO. D-51100
DESCRIPTION SODIUM-WATER REACTION
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DATE 2-14-73 BY CHEN
CHECK DATE ____ BY ____

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5) DETAILED ANALYSIS

e) SUMMARY

e-1) EVAPORATOR SECTION

(VESSEL SHELL)

$R_o = 28.75''$ $\quad \gamma = 0.00073$ lb/IN³ $\quad V_o = 0.0$ IN/sec

$H_o = 1.25''$ $\quad n = 0.219$

$\sigma_o = 98500$ PSI $\quad t_o = 0.0015$ sec.

INPUT DATA					OUTPUT DATA				COMMENTS
TIME SEC	P_{i1} PSIA	P_{i2} PSIA	P_{i3} PSIA	P_{i4} PSIA	DISP. IN	VELOCITY IN/SEC.	ACCELER IN/SEC²	R/R_o	
0.0015	1950	1945	1945	1940	28.75	0.0	2.137×10^{6}	1.0	
0.0016	1940	1910	1910	1880	28.759	168.26	1.166×10^{6}	1.00000	
0.0017	1880	1850	1850	1820	28.781	265.83	8.093×10^{5}	1.001087	
0.0018	1820	1790	1790	1760	28.811	333.15	5.474×10^{5}	1.002136	
0.0019	1760	1730	1730	1700	28.847	376.86	3.329×10^{5}	1.003377	
0.0020	1700	1680	1680	1670	28.889	400.74	1.490×10^{5}	1.004735	
0.0021	1670	1655	1655	1640	28.927	408.67	2.010×10^{4}	1.006146	
0.0022	1640	1625	1625	1610	28.967	404.95	-9.218×10^{4}	1.007564	
0.0023	1610	1595	1595	1580	29.007	390.67	-1.913×10^{5}	1.008951	
0.0024	1580	1565	1565	1550	29.045	367.05	-2.795×10^{5}	1.01027	
0.0025	1550	1535	1535	1520	29.080	335.07	-3.585×10^{5}	1.011495	
0.0026	1520	1505	1505	1490	29.1121	295.62	-4.293×10^{5}	1.012594	
0.0027	1490	1475	1475	1460	29.1394	249.45	-4.928×10^{5}	1.013543	
0.0028	1460	1445	1445	1430	29.1618	197.29	-5.495×10^{5}	1.014322	
0.0029	1430	1415	1415	1400	29.1787	139.77	-5.997×10^{5}	1.014910	
0.0030	1400	1395	1395	1380	29.1896	77.55	-6.437×10^{5}	1.01529	
0.0032	1380	1375	1375	1360	29.1921	-53.31	-6.683×10^{5}	1.01538	← MAX. DISPla.
0.0034	1360	1350	1350	1340	29.1681	-186.56	-6.670×10^{5}	1.01454	$< r^* = 1.1157$

5) DETAILED ANALYSIS

e) SUMMARY

e-1) EVAPORATOR SECTION

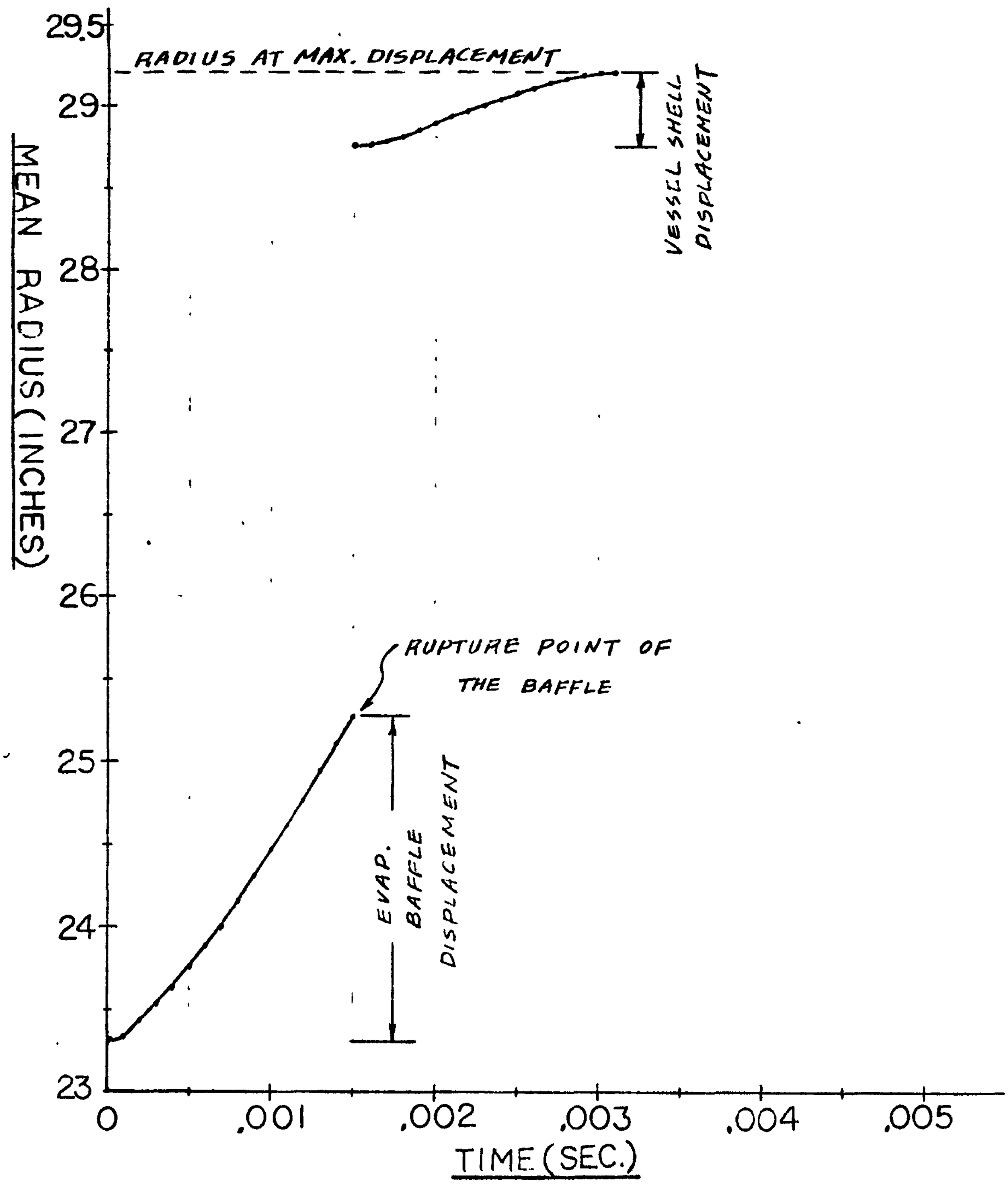

FIGURE 5. RESPONSE OF THE FLOW BAFFLE AND THE VESSEL SHELL FOR ONE TUBE FAILURE IN EVAPORATOR

5) DETAILED ANALYSIS

$A_o = 15.5''$ $\gamma = 0.000734$ lb/in³ $v = 0.0$ in/sec

$H_o = 1.0''$ $n = 0.170$

$\sigma_o = 58,700$ PSI $t_o = 0.00001$ sec

e) SUMMARY

e-2) SUPERHEATER SECTION (FLOW BAFFLE)

INPUT DATA					OUTPUT DATA				COMMENTS
TIME SEC.	P_{i1} PSIA	P_{i2} PSIA	P_{i3} PSIA	P_{i4} PSIA	DISP. IN.	VELOCITY IN/SEC	ACCELER IN/SEC²	R/R_o	
0.00001	18500	15000	15000	11500	15.5	0.0	2.520×10^7	1.0	
0.00002	11500	8600	8600	5700	15.501	198.74	1.446×10^7	1.000069	
0.00003	5700	4775	4775	3850	15.504	302.41	6.292×10^6	1.000336	
0.00004	3850	3325	3325	2800	15.507	351.82	3.602×10^6	1.000448	
0.00005	2800	2525	2525	2250	15.511	380.05	2.049×10^6	1.000685	
0.00006	2250	2500	2500	2750	15.515	396.31	1.205×10^6	1.000936	
0.00007	2750	2525	2525	2300	15.519	411.35	1.809×10^6	1.001196	
0.00008	2300	2400	2400	2500	15.523	426.03	1.127×10^6	1.001467	
0.00009	2500	2475	2475	2450	15.527	438.37	1.341×10^6	1.001746	
0.00010	2450	2425	2425	2400	15.532	451.18	1.221×10^6	1.002033	
0.00015	2400	2375	2375	2350	15.555	505.06	9.471×10^5	1.003578	
0.00020	2350	2350	2350	2350	15.582	546.81	7.294×10^5	1.005278	
0.00025	2350	2350	2350	2350	15.610	580.24	6.119×10^5	1.007098	
0.00030	2350	2300	2300	2250	15.640	608.[illegible]4	5.152×10^5	1.009016	
0.00035	2250	2175	2175	2100	15.671	628.57	2.959×10^5	1.011014	
0.00040	2100	2075	2075	2050	15.702	636.40	1.911×10^4	1.013058	

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5) DETAILED ANALYSIS

e) SUMMARY

e-2) SUPERHEATER SECTION (FLOW BAFFLE)

INPUT DATA					OUTPUT DATA				COMMENTS
TIME SEC.	P_{i1} PSIA	P_{i2} PSIA	P_{i3} PSIA	P_{i4} PSIA	DISP. IN	VELOCITY IN/SEC	ACCELER. IN/SEC^2	R/R_0	
0.0005	2050	2015	2015	1980	15.766	628.99	-1.619×10^{5}	1.017150	
0.0006	1980	1940	1940	1900	15.828	603.53	-3.435×10^{5}	1.021135	
0.0007	1900	1875	1875	1850	15.886	560.22	-5.199×10^{5}	1.024899	
0.0008	1850	1825	1825	1800	15.939	502.13	-6.398×10^{5}	1.028332	
0.0009	1800	1775	1775	1750	15.986	432.65	-7.481×10^{5}	1.031353	
0.0010	1750	1745	1745	1740	16.025	352.82	-8.471×10^{5}	1.033892	
0.0011	1740	1735	1735	1730	16.056	266.31	-8.819×10^{5}	1.035891	
0.0012	1730	1725	1725	1720	16.078	176.67	-9.0986×10^{5}	1.037322	
0.0013	1720	1715	1715	1710	16.092	84.54	-9.318×10^{5}	1.038166	
0.0014	1710	1705	1705	1700	16.095	-9.510	-9.482×10^{5}	1.038409	← REACHES MAX. DISPLACEMENT. $R/R_0 < r^* = 1.0887$
0.0015	1700	1700	1700	1700	16.090	-10.491	-9.590×10^{5}	1.03804	
0.0016	1700	1675	1675	1650	16.074	-20.040	-9.497×10^{5}	1.03705	

5) DETAILED ANALYSIS

e) SUMMARY

e-2) SUPERHEATER SECTION

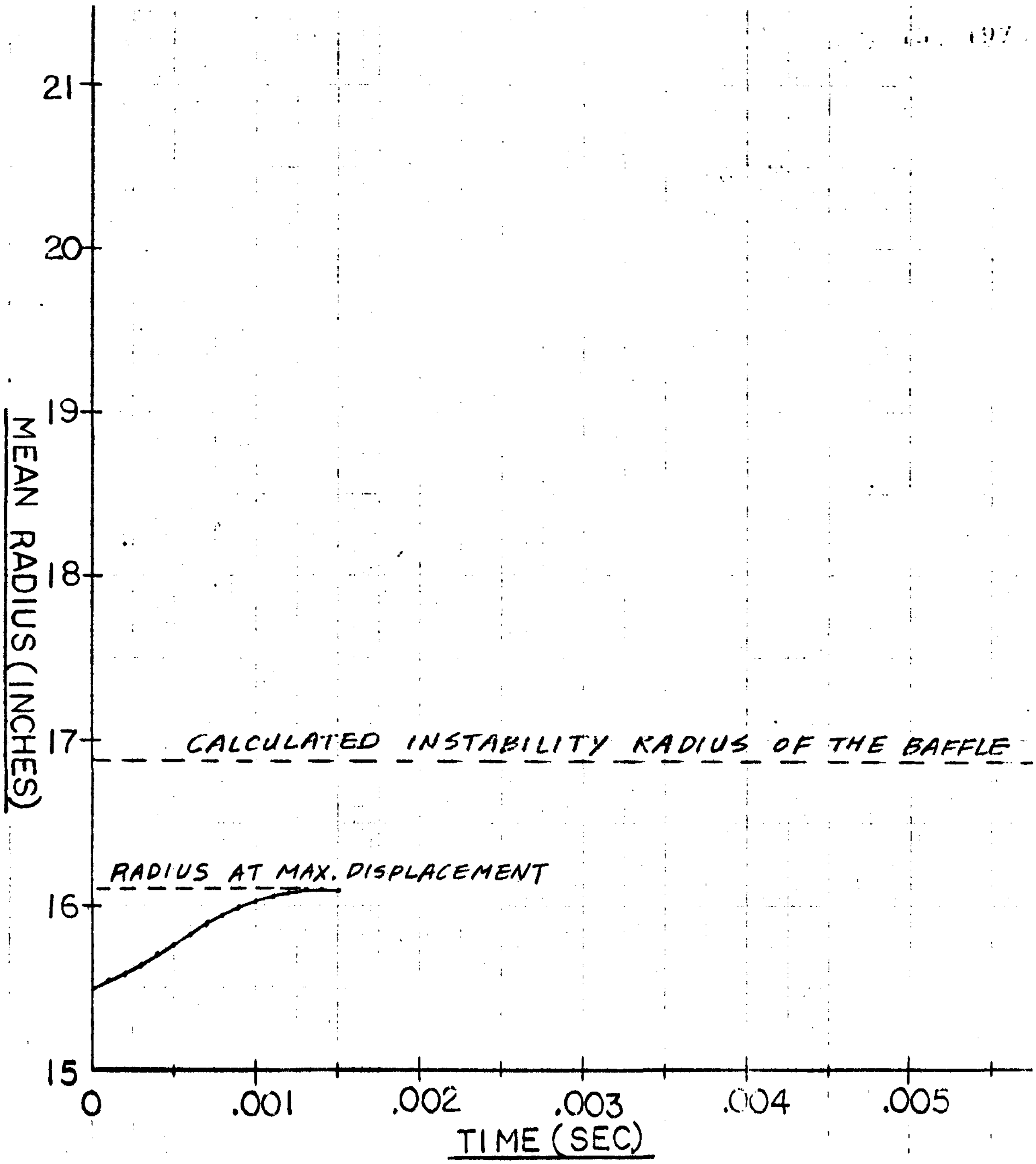

FIGURE 6. RESPONSE OF THE FLOW BAFFLE FOR ONE TUBE FAILURE IN SUPERHEATER

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5) DETAILED ANALYSIS

e) SUMMARY

e-3) EVALUATION OF THE RESULTS

CONSIDER REQUIREMENT 1

(A) EVAPORATOR

$$t_{REQUIRED} = \frac{PR_i}{S_m - .5P} = \frac{0.45 \times 28.125}{14 - 0.5 \times 0.45}$$

$$= 0.9187'' < t_{vessel\ shell} = 1.25''$$

(B) SUPERHEATER

$$t_{REQUIRED} = \frac{PR_i}{S_m - .5P} = \frac{0.15 \times 20.3125}{13.6 - 0.5 \times 0.15}$$

$$= 0.2253'' < t_{vessel\ shell} = 1.25''$$

CONSIDER REQUIREMENT 2

(A) EVAPORATOR

THE FLOW BAFFLE EXCEEDED ITS INSTABILITY RADIUS AT 0.0016 SEC. (SEE SHEET 20)

AFTER THE RUPTURE OF THE FLOW BAFFLE, THE VESSEL SHELL DISPLACEMENT REACHED ITS MAXIMUM AT 0.0032 SEC., WHICH WAS STILL BELOW THE CRITICAL VALUE.

$$R/R_0 = 1.01538 < r^* = 1.1157 \text{ (SEE SHEET 21)}$$

COMBUSTION ENGINEERING, INC.
ENGINEERING DEPARTMENT, CHATTANOOGA, TENN.

NUMBER A-6
SHEET 27 OF 27

CHARGE NO. D-51100
DATE 2-14-73 BY CHEN
DESCRIPTION SODIUM-WATER REACTION
CHECK DATE 2-16-73 BY SAC

5) DETAILED ANALYSIS

e) SUMMARY

e-3) EVALUATION OF THE RESULTS

CONSIDER REQUIREMENT 2

(B) SUPERHEATER

THE FLOW BAFFLE REACHED ITS MAXIMUM DISPLACEMENT AT 0.0014 SEC., WHICH WAS STILL BELOW THE CRITICAL VALUE.

$$R/R_0 = 1.038409 < r^* = 1.0887$$ (SEE SHEET 24)

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APPENDIX B

SPECIFICATIONS - DEMONSTRATION PLANT STEAM GENERATORS

REF: RDT STD E4-16T

DEMONSTRATION PLANT

MODIFICATIONS

EXCEPTIONS AND/OR INTERPRETATIONS TO

RDT E4-16T DATED MAY 1972

SODIUM HEATED STEAM GENERATORS

FOR LMFBR DEMONSTRATION PLANT

JANUARY 1973
COMBUSTION ENGINEERING, INC.

MODIFICATIONS

TABLE OF CONTENTS

MODIFICATION EXCEPTIONS AND/OR INTERPRETATIONS TO RDT E4-16T DATED MAY, 1972, FOR SODIUM HEATED STEAM GENERATORS

This exceptions/interpretations document modifies, supplements or clarifies RDT Standard E4-16T dated May, 1972, and in conjunction with the Ordering Data, can be used to procure sodium heated steam generators.

2.1 Reactor Development and Technology (RDT) Standards. Delete the following RDT Standards from this paragraph (not applicable).

RDT M 1 - 2T	3 - 11T
RDT M 1 - 6T	4 - 1T
2 - 1T	4 - 2T
2 - 4T	5 - 2T
2 - 7T	5 - 3T
2 - 8T	5 - .4T
2 - 15T	6 - 1T
3 - 4T	7 - 3T
3 - 5T	
3 - 6T	
3 - 9T	

Add the following RDT Standards to paragraph 2.1:

RDT M 1 - 10T
RDT M 7 - 4T

2.2 American Society of Mechanical Engineers (ASME) Codes Section XI rules for in-service inspection dated 1971 and addenda through Summer, 1972.

ASME Boiler and Pressure Vessel Code Case interpretations, code case 1331-7 Nuclear Vessels for high temperature service.

3.1.5 Seals - Internal by-pass seals are not goverened by this paragraph.

3.1.7.2 Delete last sentence.

3.3.2.1 Add paragraph as follows: The steam generator design pressures and temperatures shall be included in the Ordering Data.

3.4.8 Threaded Elements and Bolts - Item 9 - Delete this requirement and replace with the following: Threaded or bolted connections on the sodium side shall be held to a minimum. Where bolted connections are required, suitable analysis shall be provided.

3.5.3 Water Level Indicators - Delete this paragraph. (In a forced recirculating steam generator utilizing a separate steam drum, the water level will be established within the drum and level indicators will be required in this component.

3.11 Reports and Documentation - Add sentence: "Submittal of reports and Documentation shall be in accordance with Table IV".

REF: RDT STD E4-16T
DATED MAY 1972

DEMONSTRATION PLANT

ORDERING DATA

SUPPLEMENTS RDT STANDARD E4-16T DATED MAY 1972

"SODIUM HEATED STEAM GENERATOR"

FOR LMFBR DEMONSTRATION PLANT

JANUARY 1973
COMBUSTION ENGINEERING, INC.

SODIUM HEATED STEAM GENERATOR

LMFBR DEMONSTRATION PLANT

ORDERING DATA

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Note: This document supplements the Sodium-Heated Steam Generator Standard RDT E4-16T, dated May, 1972. The numbering system used in this document is the same as RDT E4-16T. When no additional requirements to RDT E4-16T are required, the paragraph is omitted.

1. SCOPE

This document contains the Ordering Data to be used for the design and fabrication of sodium heated steam generators for LMFBR Demonstration Plants.

1.1 Components and Services to be Provided - Components and services to be provided include the following items:

A complete steam generator as defined in 3.1.1, including:

A. Structural supports

B. Lifting lugs

C. Drain, vent, and relief connections

D. Preheat Requirements

E. Tests and mockups to back up the design

F. Drawings, plans and instruction manuals

G. Design, analysis, and performance reports

H. Quality Assurance procedures and documentation

I. Spare parts

J. Field services as defined in 3.9

K. Insulation Requirements

1.1 Components and Services to be Provided continued

L. Steam Generator Instrumentation

Associated equipment not covered by this specification: steam drum, pumps, piping, valves, sodium water reaction relief systems, control systems, isolation and dump systems.

3. TECHNICAL REQUIREMENTS

3.1 General Requirements

3.1.1 Steam Generator

3.1.1.2 Type and Description - The steam generator shall consist of a bayonet-evaporator module and a bayonet superheater module as shown in Figure 1. The steam generator shall operate as a forced recirculation system as shown in Figure 2.

Feedwater shall enter the evaporator through nozzle H and flow downward through the bayonet tubes to the base of the tubes, where it shall flow upward through an annulus, receiving heat from the sodium on the shell side in counterflow. The steam shall then exit through nozzle G and after passing through a separator (not part of this specification), enter the superheater module through nozzle C.

3.1.1.2 Type and Description continued

Entering the superheater, the steam shall flow downward through an annulus between the bayonet tube and pressure tube, receiving heat from the shell side sodium in counterflow. Upon reaching final superheat temperature at the base of the tube, the steam shall flow upward through the bayonet tube and exit the unit through nozzle D. The bayonet tube portion of the superheater tube assembly shall be made of double wall tubes to provide a stagnant space between the tubes for insulating purposes.

Sodium shall enter the lower portion of the superheater module through nozzle A and flow upward around the bayonet tubes and exit through nozzle B near the top of the bundle. The sodium shall then enter the upper part of the evaporator module through nozzle F and flow downward around the bayonet tubes and exit the evaporator at the lower end through nozzle E.

The steam generator shall be designed to incorporate an inert cover gas between the tubesheet and the sodium free surface in the evaporator and superheater modules. The cover gas systems in the modules are completely independent. Nozzles above and below the sodium level shall be incorporated to provide a means of measuring and controlling the sodium level in each module. A vent nozzle/s shall be provided at the highest points inside the module shells.

The steam generator design shall incorporate sodium-water reaction relief systems for each of the modules. Primary and secondary relief systems shall be connected to the modules at nozzles located in the gas space in each

3.1.1.2 Type and Description continued

of the modules.

3.1.1.3 Design Basis - The steam generator shall be designed in accordance with the latest edition and addenda of the ASME Boiler and Pressure Vessel Code, Section III, together with the applicable code cases including Code Case 1331-7, the high-temperature criteria of RDT F9-1T, and the requirements of this specification. The steam generator shall be classified as a Class I vessel and shall be Code stamped. The steam generator shall be designed for a service life of 30 years operation at the conditions defined in 3.6.

3.1.3.1 Inspection - Requirements for In-Service Inspection shall be in accordance with ASME Code Sec. XI,"Rules for Inservice Inspection".

3.1.6 Cover Gas - Argon cover gas shall be utilized above the surface of the sodium. The argon cover gas supply shall have the following characteristics:

Supply Inlet Temperature (^{o}F)	100
Working Pressure (psia)	0 - 150
Purity	99.996%

Impurities	PPM
Oxygen (tentative)	5
Hydrogen (tentative)	2
Nitrogen	15
Carbonaceous Gases	5
Water (D.P. - 84^{o}F)	6
Other	7

3.1.7.2 Drain lines will be 2" pipe size.

3.1.8 Size and Weight - The approximate size and weight of

3.1.8 Size and Weight continued

the steam generator are shown in Figure 1.

3.1.9.2 The heatup rate of the steam generator shall be 5°F/hr. from ambient to 350°F.

3.1.11 Performance Requirements - The steam generator shall be capable of continuous operation through the range of conditions specified in 3.6. The steam generator shall be designed for exposure to the following environmental conditions after installation and during operation:

Ambient Temperature	Later
Ambient Pressure	Later
Ambient Humidity	Later
Nuclear Radiation Intensity	Later
Neutron Radiation	Later

3.2 Thermal and Hydraulic Design Requirements

3.2.1 Design Objectives

3.2.1.1 The steam generator shall be designed to permit safe, stable, and predictable operation throughout the load range of the steady state and transient conditions presented in 3.6.

3.2.2 Type and Orientation - The steam generator shall be forced recirculating shell and bayonet tube heat exchanger with sodium on the shell side and water/steam on the tube side. The superheater and evaporator shall be contained in separate shells each mounted separately in a vertical position as shown in Figure 1.

3.2.3 Physical Properties and Purity of Sodium - For the steam generator design, the sodium properties as recommended in ANL-7323 shall be used.

3.2.4 Physical Properties and Purity of Feedwater - The feedwater purity limits for the steam generator model shall be as follows:

		PPB
1.	Total dissolved solids	50
2.	Total silica	20
3.	Total iron	10
4.	Total copper	2
5.	Conductivity after ion exchange	1
6.	PH	9.2 - 9.4

3.2.5.1 Excess Surface (Later)

3.2.5.2 The steam generator sodium level control system will be designed to maintain the sodium level with a permissable variation from the set point of plus or minus 3 inches.

3.2.7 Pressure Drop - The water/steam side and sodium side pressure drop shall not exceed the following:

Water/steam side, evaporator	50 psi
Water/steam side, superheater	125 psi
Sodium side, evaporator	20 psi
Sodium side, superheater	20 psi

3.2.11 Thermal Transients - An analysis shall be performed to evaluate the response of the steam generator to the inlet flow and temperature transients of water and sodium specified in Section 3.6.

3.3 Structural Design Requirements

3.3.2 Stress Limits - Allowable stresses for the steam generator shall be in accordance with Section III of the Code, Code case 1331-7 and RDT F9-1 for a service life of 30 years under the normal, upset, emergency, and faulted conditions specified in 3.6.

3.3.3 Seismic Considerations - Earthquake zone 2

DBE Acceleration	.2G Horizontal	acting
	.1G Vertical	simultaneously

3.3.4 Vibration

3.3.4.2 Flow rates for use in vibration analyses shall be 110% of the maximum flow rates specified in table II.

3.3.6 Thermals Transient Stresses - Regions of discontinuities such as nozzles, tube to tubesheet joints, tubesheet to shell regions, and thick sections such as tubesheets, nozzles, and shells shall be analyzed for thermal stresses resulting in past from the number of sodium and water-steam temperature transients specified in Section 3.6.

3.3.7 Nozzle Loads - Nozzle loads shall be in accordance with Figure 3.

3.3.9.2 Tube bundles, baffle and tube support thickness, and tube hole clearances shall meet the requirements of the specifications, codes and code cases contained in Section 2.0.

3.3.11 Sodium-Water Reaction - The steam generator shall be designed to contain the pressure and shock forces caused by the design basis leak (DBL) defined below:

3.3.11 Sodium-Water Reaction continued

One (tentative) guillotine failure of pressure tube at the worst location as determined by a sodium-water reaction pressure loading analysis.

3.3.14 Corrosion Allowance - Later

3.4 Connections, Accesses, and Appurtenances

3.4.2 Auxiliary Lines - See 3.4.7

3.4.3 Insulation Supports - The supplier shall provide insulation supports on the exterior surfaces of the steam generator evaporator and superheater shells. Integral clips or rings for supporting the insulation shall be provided prior to final heat treatment.

Requirements for insulation shall be established by the supplier based upon the following criteria and operating conditions. (Later)

3.4.6 Foundation and Support Structure - (Later)

3.4.7 Piping Connections - The interface location of the sodium and water/steam lines and auxiliary lines are shown in Figure 1. Piping sizes, end preparation and number required shall be as follows:

	No. Required	Pipe Size	End Preparation
Sodium inlet, superheater	1	18"	Butt Welded
Sodium inlet, Evaporator	2	12"	Butt Welded
Sodium outlet, evaporator	1	18"	Butt Welded
Sodium outlet, superheater	2	12"	Butt Welded
Evaporator recirc. water inlet	1	16"	Butt Welded
Evaporator saturated steam outlet	1	16"	Butt Welded
Superheater saturated steam inlet	1	14"	Butt Welded
Superheater steam outlet	1	14"	Butt Welded

3.4.7 Piping Connections continued

Auxiliary Lines (in each evap/S. H. module)

	No. Required	Pipe Size	End Prep.
Sodium Level Nozzles	Later		Butt Welded
Primary Relief Nozzles	Later		Butt Welded
Secondary Relief Nozzle	2	12"	Butt Welded
Cover Gas Makeup & Vent Nozzle	1	1"	Butt Welded
Vent (steam/water)	1	1"	Butt Welded
Instrumentation	1	2"	Butt Welded

3.4.9 Preheating System Supports - (Later)

3.5.1.1 The following instruments and provisions for instrumentation are considered to be a part of the steam generator and consist of the following:

a. Strain gages for measurement of shell stresses
b. Sodium level indicators
c. External sodium leak detection
d. Water-to-sodium leak detectors
e. Pressure indicator
f. Vibration instrumentation
g. Temperature indicator

Numbers and location of instrumentation (Later)

3.5.2 Sodium Level Indication - The steam generator shall be fitted with independent sodium level indicators mounted external to the evaporator and superheater shell modules.

3.5.4 Pressure Gages - The requirements of the pressure gages to measure sodium and water/steam pressures shall be determined and provided by the supplier for installation in the piping adjacent to the steam generator model.

3.5.5 Temperature Instrumentation - Requirements for the temperature

3.5.5 Temperature Instrumentation (continued)

instrumentation necessary to establish the thermal and structural performance of the steam generator model shall be determined by the supplier. Thermocouples shall be installed on the steam generator model to adequately measure the following:

a. Sodium side temperature
b. Water/steam side temperature at end points
c. Shell temperature

Requirements and location of temperature measuring devices shall be as follows: (Later)

3.5.6 Requirements for flow measurement shall be provided by the supplier. No flow measurement will be required within the steam generator shells.

3.5.7 Water-to-sodium Detection - The evaporator and superheater modules of the steam generator shall have provisions for two (2) independant means of monitoring for water-to-sodium leakage.

1. Diffusion tube Hydrogen detectors shall be located in the sodium piping as close as practicable to the superheater outlet nozzles and the evaporator outlet nozzle.

2. The cover gas space in the evaporator and superheater modules shall have provision for continuous monitoring of the cover gas by means of a gas chromatograph to detect hydrogen.

3.5.8 External Leakage Detection - (Later)

3.5.9 Instrumentation, Inspection, Maintenance, and Repair - Test procedures to determine the functional operation of instrumentation shall be provided by the purchaser.

3.5.10 Structural Instrumentation - The supplier shall determine the requirements of instrumentation necessary to determine the structural performance of critical areas of the steam generator. This instrumentation shall be provided as part of the steam generator. Procedures for installation of instrumentation will be provided by the supplier.

3.6 Operating Conditions

Note: (The information and criteria in this section is of a preliminary nature and is not in the form that will be contained in the final specification. Some of the notations and discussions are included for preliminary information only and will be deleted in the final spec. Tables will replace the curves shown in figures 4 through 11).

3.6.1 Types of Operating Conditions - The steam generator shall be designed and fabricated for satisfactory operations under the normal, upset, emergency and faulted conditions as defined in paragraph NB-3113 of Section III of the ASME Code and as specified below in Sections 3.6.2 and 3.6.3. These operating condition categories are defined in summary in the following paragraphs.

3.6.1.1 Normal Conditions - Normal conditions are any conditions experienced in the course of system startup, operation in the design power range, hot standby and system shutdown, other than Upset, Emergency, Faulted or Testing Conditions.

3.6.1.2 Upset Conditions - Any deviations from Normal Conditions anticipated to occur often enough that design should include a capability to withstand the conditions without operational impairment are

3.6.1.2 Upset Conditions continued

called Upset Conditions. The Upset Conditions include those transients which result from any single operator error or control malfunction, transients caused by a fault in a system component requiring its isolation from the system and transients due to loss of load or power. Upset Conditions include any abnormal incidents not resulting in a forced outage and also forced outages for which the corrective action does not include any repair of mechanical damage.

3.6.1.3 Emergency Conditions - Emergency Conditions are those deviations from Normal Conditions which require shutdown for correction of the conditions or repair of damage in the system. The conditions have a low probability of occurrence but are included to provide assurance that no gross loss of structural integrity will result as a concomitant effect of any damage developed in the system. The total number of postulated occurrences for such events shall not cause more than 25 stress cycles having an Sa value greater than that for 10^6 cycles from the applicable fatigue design curves.

3.6.1.4 Faulted Conditions - Faulted conditions are those combinations of conditions associated with extremely-low-probability, postulated events whose consequences are such that the integrity and operability of the nuclear energy system may be impaired to the extent that considerations of public health and safety are involved. Such considerations require compliance with safety criteria as may be specified by jurisdictional authorities.

3.6.2 Steady-State and Transient Operation - The steam generator shall perform under steady-state operational and environmental conditions, including transients induced by varying loads. It shall also operate under other operational transients which are not steady-state conditions. All of these operating conditions are defined in the following paragraphs.

3.6.2.1 Normal Conditions

1. Full Power Steady-State Operating Conditions - The steam generator full power steady state operating conditions are shown in Table II.

2. Part Load Steady-State Operating Conditions - The steam generator part load steady-state operating conditions of temperature flow and pressure are shown in Table II and the curves on Figures 4, 5 and 6.

3. Dry Heatup and Cooldown - Dry heatup and cooldown rates of the steam generator shell are shown in the following table. Heatup of the shell is accomplished by trace heating prior to filling with sodium.

Event	Location	Temp. °F Start	Temp. °F End	No. of Occurrences	Temperature Change
Dry Heatup	Shell	Ambient	350°F	10	+ 5°F/hr.
Dry	Shell	350°F	Ambient	10	- 5°F/hr.

4. Normal Startup and Shutdown - The normal startup and shutdown temperature transient conditions are specified in the following table. The temperature change rates shown apply specifically to the inlet

3.6.2.1 Normal Conditions continued

nozzles on both the sodium and water/steam sides.

Event	Location	Temp. °F Start	End	No. of Occurrences	Max. Rate of Temp. Change
Startup	Sodium Inlet Nozzle	350°F	924°F	600	+100°F/hr.
	FW Inlet Nozzle @ Drum	100°F	370°F		
Shutdown	Sodium Inlet Nozzle	924°F	350°F	250	-100°F/hr.
	FW Inlet Nozzle @ Drum	370°	100°F		

During the startup transient sodium flow is held constant at 30% to 35% power and then increased lineraly to 40% flow at 40% power. The flow and power transients during shutdown are the reverse of those given for startup. (Feedwater flow, temperature and pressure during startup and shutdown will be determined later).

5. Load changes Over the 40 - 100% Power Range - The steam generator will be subjected to two types of load changes, i.e. step changes in load and ramps

a. Normal Ramp Change - Normal ramp load changes will vary linearly between the end conditions at 40% and 100% full power with a power ramp rate of 3%/min. Temperature change rates are shown in the table below. Sodium and feedwater flows vary from 39.5% to 100% and 37% to 100% respectively over this power range. This information is summarized in Figure 7 for a normal load decrease from 100% to 40%. A load increase over the same range is simply the reverse of these curves.

Additional temperature ramps over different load ranges can be obtained from the part load operation curves shown in Figures 4, 5 and 6.

Event	Location	Temp., °F Start	Temp., °F End	No. of Occurrences	Max. Rate of Temp. Change
Normal Power Increase	Sodium Inlet Nozzle	924	960	9000	1.8°F/min.
	FW Inlet Nozzle @ Drum	370	460		4.5°F/min.
Normal Power Decrease	Sodium Inlet Nozzle	960	924	9000	-1.8°F/min.
	FW Inlet Nozzle @ Drum	460	370		-4.5°F/min.

*Temperature transients at other locations within the steam generator will be available later.

b. Step Changes - It is assumed that the demonstration plant must be capable of performing step changes in power which are at least equivalent to those provided for in current water reactor plants. Therefore, the step change is based on a ± 10% instantaneous change in power from any power level over the 40% - 100% power range. The response of the plant to these steps should be as quickly as possible. However, the rate of change of temperature, pressure and flow has not been determined and will be supplied later. The C-E PWR's specify that these step changes exhibit temperature and pressure fluctuations within the range of their normal specified plant variations. The normal demonstration plant variations are not available, therefore, this information will be supplied later. The number of occurrences is estimated to be 1500, equally divided between upswings and downswings.

c. Fast or Emergency Ramp Power Changes - (Later)

6. Other Normal Operating Conditions - There are several other operating conditions that are considered normal but are not included here because it is felt that they are similar in transient response to some of those already listed. These operating conditions would include such things as: Hot Restart after Scram, Shutdwon after Scram, or the Return to Power from Hot Standby or the reverse, i.e., going to Hot Standby condition from Power Operation. All of these transient conditions are similar to or a combination of either normal startup and normal power increases over the load range or the reverse, which is normal power decreases and normal shutdown. Therefore, they have been accounted for by increasing the number of occurrences of the appropriate normal operating condition.

3.6.2.2 Upset Conditions - The following transients are defined as upset conditions and are considered to be representative of most the group of possible upset events. Included are curves showing the expected thermal transients in the fluid at the sodium and feedwater inlet nozzles and their estimated number of occurrences. The thermal transients at other locations throughout the steam generator are currently being evaluated.

1. Normal Reactor Plant Scram - (Number of Lifetime Occurrences = 350) Reactor plant scram is an operating procedure for very quickly reducing the power output of the reactor to a low level of 2-5% as determined by fission product decay heat. This scram might

3.6.2.2 Upset Conditions continued

be a result of a false spurious electrical signal or a desired response from one of many protective system signals or an operator-initiated signal. The reactor scram primarily consists of reducing the reactor power by dropping all control rods, and simultaneously reducing the heat transfer system flows by tripping the primary and secondary sodium pumps and the feedwater pumps. The secondary sodium flow coasts down to 5% while the feedwater flow coasts down to 4%. These flow decay curves and the sodium and feedwater inlet temperature transients for the first 300 seconds following scram are shown in Figure 8. Also shown is the feedwater inlet pressure drop of 200 psi which varies with the square of the flow. Furthermore, when scram occurs, the turbine-generator is also tripped. After the 300 seconds shown on the curves, the transient is assumed to follow the normal shutdown transient.

2. Loss of Recirculation Pump Flow - (Number of lifetime occurrences = 20 pump) Definition of transient - (Later)

3. Other Upset Operating Conditions - It is recognized that there are a number of other operational events that are considered to be Upset Conditions. However, at this time since they all would very quickly cause a reactor scram, it is assumed that the transients would be similar to a scram transient. Therefore, allowance has been made for these events by appropriately adjusting the number of occurrences of the normal scram. Some of the other Upset Conditions include events such as: positive reactivity insertion with scram; loss of electrical power to one

3.6.2.2 Upset Conditions continued

primary, secondary or feedwater pump with scram, loss of plant power with scram, control rod drop with scram, and turbine trip with scram. It is recognized that the scram following some of these events might be delayed somewhat and therefore, momentary up or down temperature transients could be initiated. However, from past experience these have been found to be quite short in time and have very little effect on the overall temperature transient used for structural analysis. Therefore, since these delays were not identified at this time, no attempt was made to include them and all such events are assumed to be similar to a normal scram. It should be pointed out that as additional plant system design information becomes available, these Upset Conditions will be reconsidered and if this assumption appears invalid in any of the cases, the transient will be considered separately.

3.6.3 Abnormal Conditions - Abnormal conditions are classifiable as occurring after a failure or malfunction of a component or system. Although no longer stated as an operating category in the ASME Code, the events under this section in the RDT Standards include events which now fall under Faulted, Emergency and in some cases, Upset Condition categories of the ASME Code. Since some Upset Conditions were also listed under section 3.6.2 - "Steady-State and Transient Operation" of the RDT standards above, e.g., reactor scram, it was decided to include all Upset Conditions under that category. Therefore, only Emergency and Faulted

3.6.3 Abnormal Conditions continued

Conditions will be included in this section.

3.6.3.1 Emergency Conditions - The following transients are defined as Emergency Conditions and are representative of a group of possible events but do not include all such events. Included here, similar to what was done previously for Upset Conditions, are curves showing the expected thermal transients in the fluid at the sodium and feedwater inlet nozzles and their estimated number of occurrences. The transients at other locations throughout the steam generator are currently being determined and will be available in the near future.

1. Single Secondary Sodium Pump Seizure with Scram - (Number of Lifetime Occurrences = 3 per loop) The seizure of one secondary sodium pump is quite similar to a normal scram, with the exception of the flow coastdown in the affected loop. Since the pump shaft is assumed to seize quickly, the impellar stops rotating and therefore creates an additional resistance in the loop flow. This is assumed to cause a flow coastdown twice as fast as a normal loop coastdown. The sodium inlet temperature to the steam generator of the affected loop will remain constant over the first 300 seconds. Feedwater parameters are identical to those occurring during normal scram. All of these steam generator inlet parameters for the affected loop are shown in Figure 9. Parameters in the other two secondary sodium loops and associated feed-

3.6.3 Abnormal Conditions continued

water systems are identical to normal reactor scram, since scram is assumed to occur a few instants after pump seizure. After the 300 sec. shown on the curves, the transient is assumed to follow the normal shutdown transient.

2. Single Primary Sodium Pump Seizure with Scram - (Number of Lifetime Occurrences -3 loop) The steam generator inlet conditions for the first 300 seconds of this transient are shown in Figure 10. Everything is the same as normal scram with the important exception of sodium inlet temperature which eventually decreases at a rather high rate of -3°F/sec. This temperature ramp is caused by the stoppage of the primary flow in the seized loop caused by closure of the check valve. That is, once flow in that primary loop is essentially stopped, the secondary sodium loop, even at low decay heat flows, rapidly removes the stored heat in the IHX primary sodium. The reduction in primary sodium temperature causes a similar reduction in secondary sodium temperature which eventually levels out near the steam saturation temperature. Sodium and feedwater flows, temperatures and pressures in the two other main heat transfer loops are the same as for normal scram. Furthermore, after the 300 seconds shown on the curves of Figure 10, the transient is assumed to follow the normal shutdown transient.

3. Activation of Na/H_2O Protection System - (Number of Lifetime Occurrences - 2 per component - evaporator/superheater pair)

3.6.3 Abnormal Conditions continued

This event consists of an emergency isolation and dump of the water/steam side of an evaporator/superheater module pair during power operation. Following the initiation of such an event, a reactor plant scram will occur, probably triggered by a loss of pressure signal of the effected steam generator. For this transient, a delay of 5 seconds is assumed before scram occurs. The sodium inlet nozzle temperature and flow transients are therefore similar to normal scram and are illustrated in Figure 11. The feedwater parameters will also vary according to normal scram conditions with a 5 second delay; however, this transient is not influenced by feedwater conditions since the water/steam sides are isolated. Instead, when the isolation valves close and the dump valve opens, the water/steam mixtures will quickly blow down. This occurs in both the evaporator and superheater modules simultaneously since the protection system of each pair is tied together. These blowdown flows are currently being determined and are not yet available. Therefore, the internal temperature transients in the steam generator during and after blowdown have not been determined. They will be supplied in the near future. Again, as before, after the first 300 seconds, the transient is assumed to be similar to the normal scram transient.

4. Loss of Feedwater Flow Via Feedwater Control Valve Closure with Scram - (Number of Lifetime Occurrences = 2 valves) Although the operating philosophy for this event has not been completely defined yet, it is included and briefly discussed here because of the possible effects it might have on the steam drum, the evaporator, and recirculation pump designs. This event is different from the loss of feedwater pump power event in that the

3.6.3 Abnormal Conditions continued

closing of the feedwater control valve stops all feedwater flow whereas 4% natural convection flow still remained after loss of pump power. With the complete loss of feedwater flow to the steam drum, the drum level can drop, possibly to a point where the recirculation pumps began to cavitate. This condition could damage the pumps. Furthermore, with no low temperature feedwater entering the drum, the drum water temperatures will begin to rise as a function of the recirculation water temperature only. This will cause temperature transients on the drum as well as the evaporator. It is probably possible to eliminate this possible transients by proper operating procedures and/or a backup feedwater supply, however, this has not yet been determined. Therefore, the transient, if any, during this event will be defined later.

5. Sodium-Water Reaction Due to a Design Basis Leak - The pressure, flow and temperature transients during this event will be defined later.

6. Steam Line Rupture (Loss of Pressure) - This transient has not been defined yet and will be supplied later.

7. Other Emergency Operating Conditions - Several other events which might be considered to fall within this category by some designers are briefly discussed in this section. The RDT standard lists four additional events including: loss of primary sodium, loss of secondary sodium, loss of operating power, and scram with sodium flow continuing. The first two events could include two initiating conditions, both of which have been already

3.6.3 Abnormal Conditions continued

discussed and specified. This is, the loss of primary or secondary flow can be caused by power failure to the pumps which is similar in transient response to reactor scram and is included under Upset Conditions. This event could also be caused by seizure of a pump as was discussed above under Emergency Conditions. The third event, loss of operating power is similar in response to reactor scram and is included under Upset Conditions. The last event, i.e., scram with sodium flow continuing, has been precluded by the control philosophy of the demonstration plant. That is, sufficient backup trip circuits will be provided to insure that all pumps trip whenever scram occurs.

3.7.3 Material Samples - (Later)

3.8.1 Fabrication - Submittal of drawings and documents shall be in accordance with tables 3 and 4.

3.8.2.7 Tube Plug Welding - Tests to demonstrate tube plugging procedures and equipment shall be performed by the supplier. The type and number of these demonstration acceptance tests shall be submitted to the purchaser for approval.

Tube plugs to be furnished by supplier (Later).

3.8.3 Heat Treatment - Heat treatment procedures shall be submitted for purchaser approval prior to heat threatment. Six copies of heat treatment records shall be submitted. Number of copies is shown in Table 4.

3.8.11 Assembling - An inspection and assembly procedure shall be submitted by the supplier for purchaser approval.

3.8.13 Identification and Marking

3.8.13.3 Nameplates - Nameplates shall be installed on both modules on stand-offs in an accessible location to permit reading after insulation is applied. In addition to the ASME Code requirements for marking, the nameplates shall include the following data:

Part Name	Contract No.
Identifying No.	C-E Spec. No.

3.9 Installation and Field Services Requirements

3.9.1 Services to be Provided - The supplier shall provide the services of a field representative to monitor installation, pre-operational testing, repair, performance of test program and to provide technical advice during these operations. Specific requirements will be determined later.

3.9.4 Handling - The supplier shall provide all fixtures and equipment necessary to handle the steam generator components. These include:

- Shipping saddles, covers, tie downs, etc.
- Fixtures necessary for up-ending the components at the site
- Equipment for maintaining a positive pressure inert gas atmosphere on the interiors of the components during shipment and storage.

3.9.6 Installation Requirements - The supplier shall monitor the off-loading of the steam generator at the site and the up-ending, lifting and installation of the components in the facility.

The supplier shall monitor the installation of all instrumentation furnished by the supplier.

3.10.2.3 Requirements for pressure relieving devices - (Later)

3.12 Drawings - The time, number of copies, and approval require-

3.12 Drawings continued

ments for drawings shall be in accordance with Table 3.

3.12.6 Parts List - Spare tube plugs to be furnished - See paragraph 3.8.2.7.

4.6.2 Helium Leak Testing - Acceptance standards and specific requirements (Later).

5.1 Preparation - Items to be delivered (Later).

COMBUSTION ENGINEERING, INC.

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NUMBER ____________

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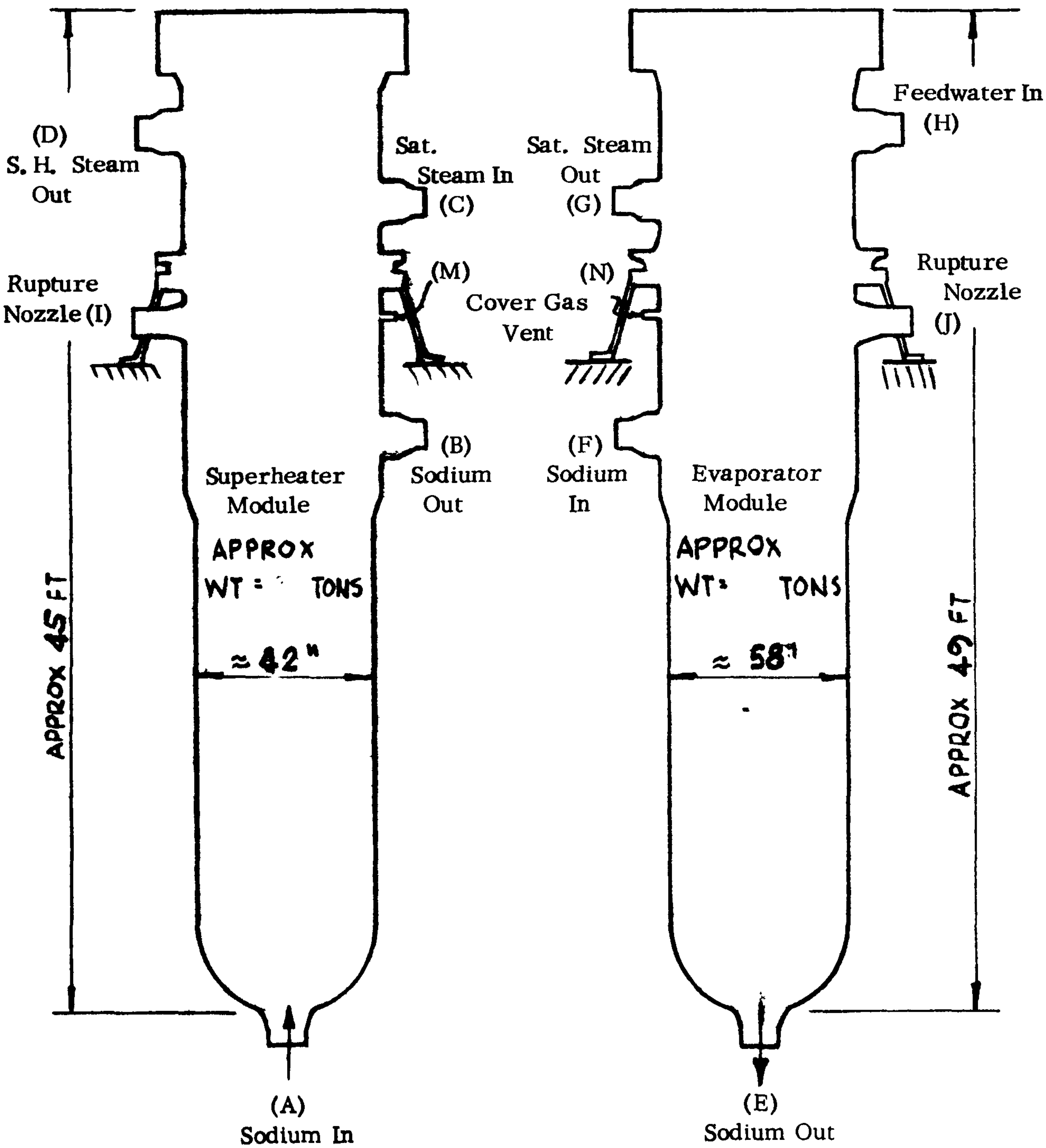

FIGURE I

COMBUSTION ENGINEERING, INC.

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SHEET ______ **OF** ______

DATE ______ **BY** ______

CHECK DATE ______ **BY** ______

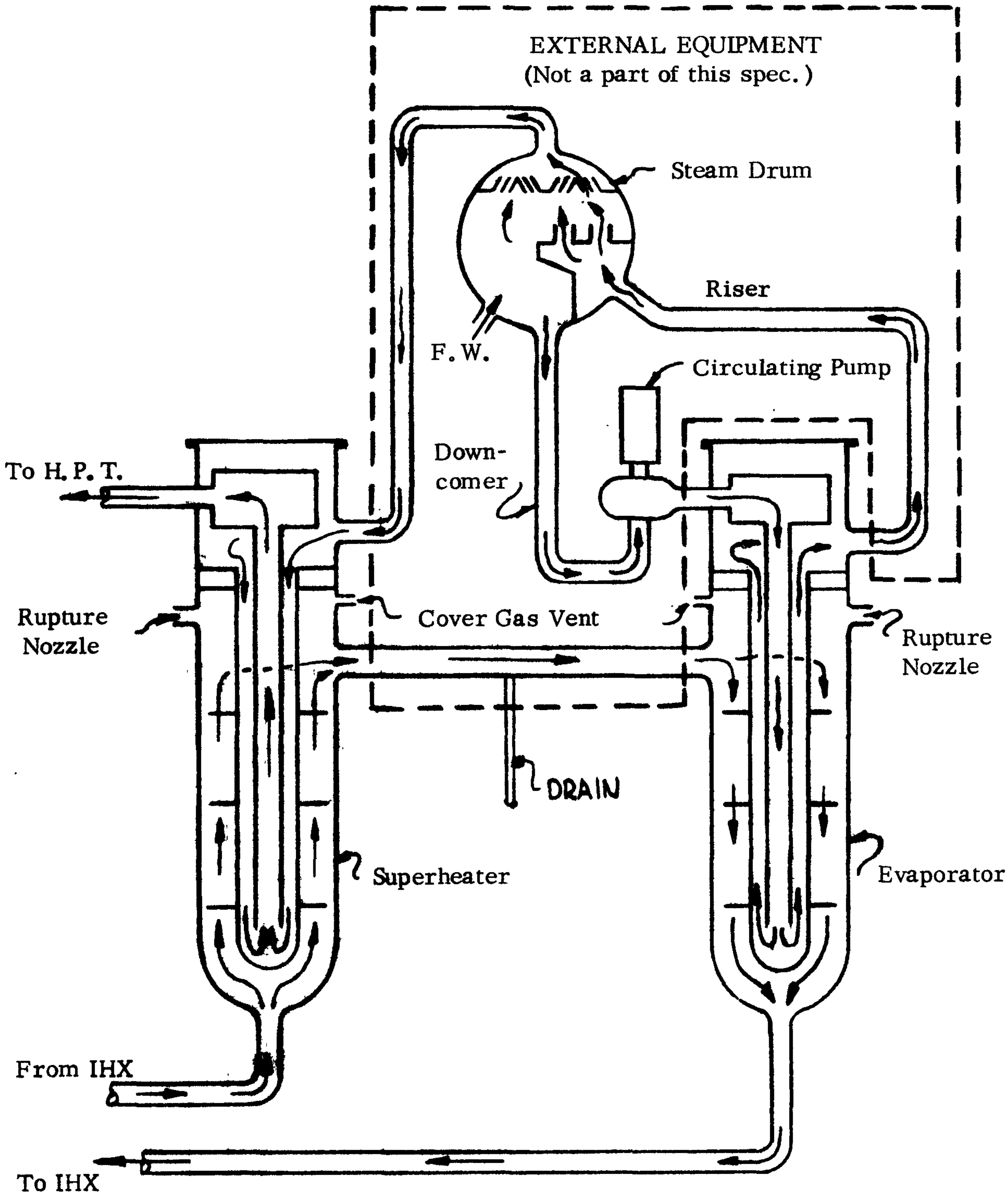
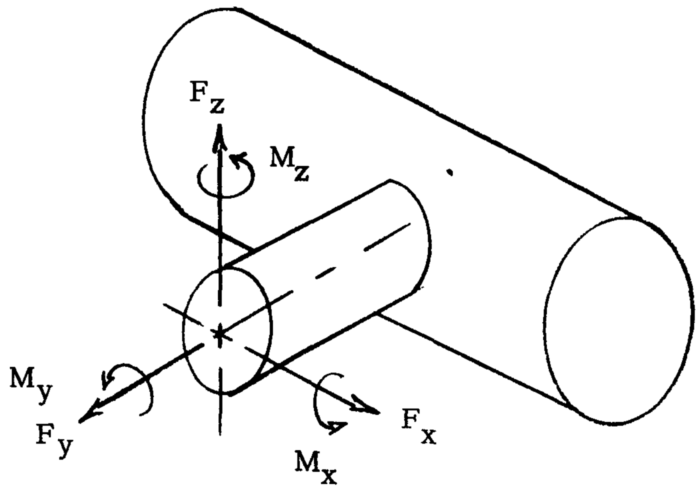


FIGURE 2
Schematic Arrangement
Bayonet Tube Evaporator Drum & Superheater

SP 11461 A



NOZZLE LOCATION	F_x (lb.)	F_y (lb.)	F_z (lb.)	M_x (in.-lb.)	M_y (in.-lb.)	M_z (in.-lb.)
		L A T E R				

NOZZLE LOADS

FIGURE 3

STEAM GENERATOR SYSTEM TEMPERATURES AS A FUNCTION OF POWER FOR AEC DEMO PLANT UNIT

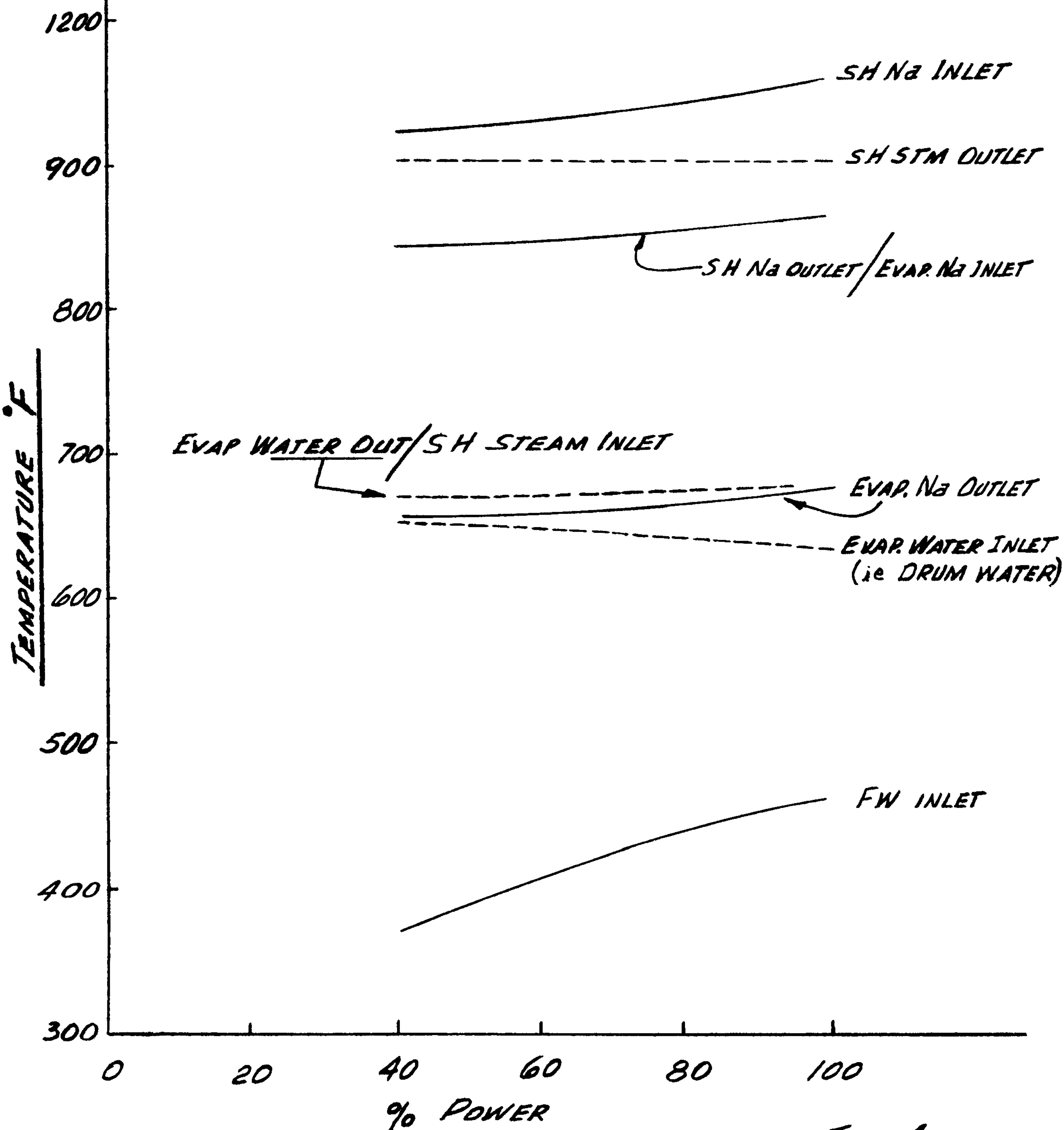
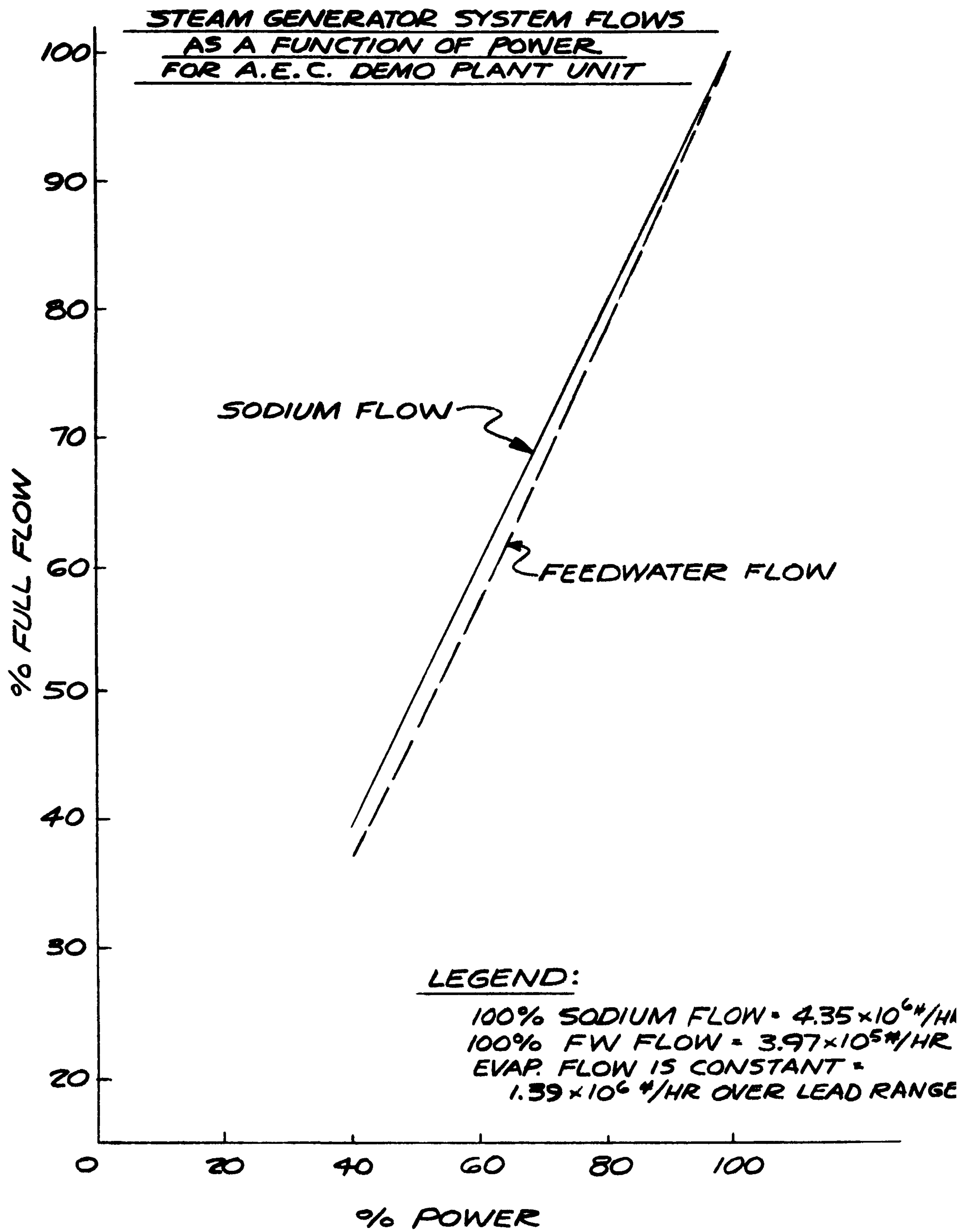


FIG. 4



STEAM GENERATOR SYSTEM FLOWS
AS A FUNCTION OF POWER
FOR A.E.C. DEMO PLANT UNIT
100
90
80
70
60
50
40
30
20
% FULL FLOW
SODIUM FLOW
FEEDWATER FLOW
LEGEND:
100% SODIUM FLOW = 4.35×10⁶ #/H
100% FW FLOW = 3.97×10⁵ #/HR
EVAP. FLOW IS CONSTANT =
1.39×10⁶ #/HR OVER LEAD RANGE
0
20
40
60
80
100
% POWER

FIG. 5

STEAM GENERATOR SYSTEM PRESSURES AS A FUNCTION OF POWER FOR A.E.C. DEMO PLANT UNIT

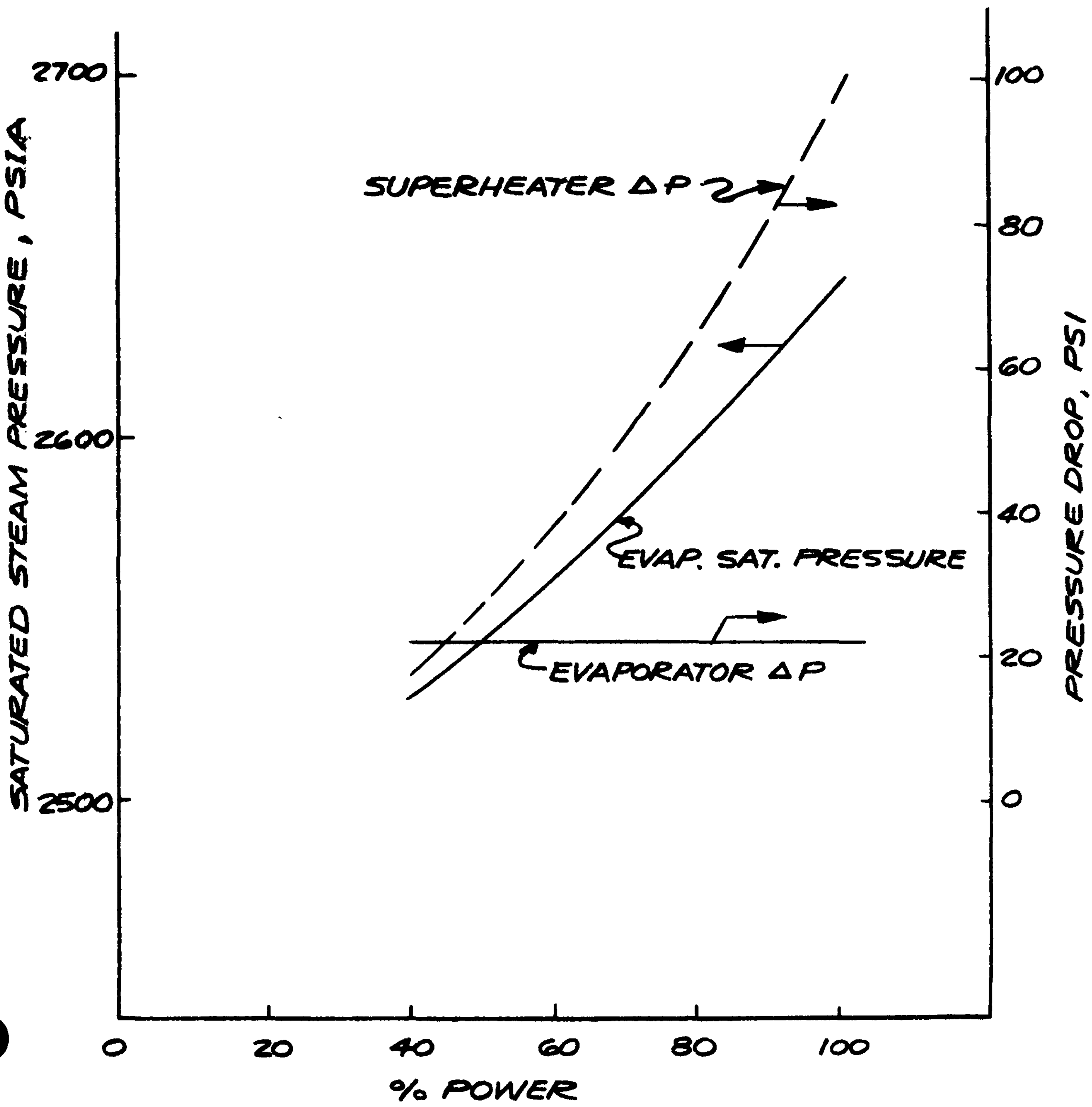


FIG. 6

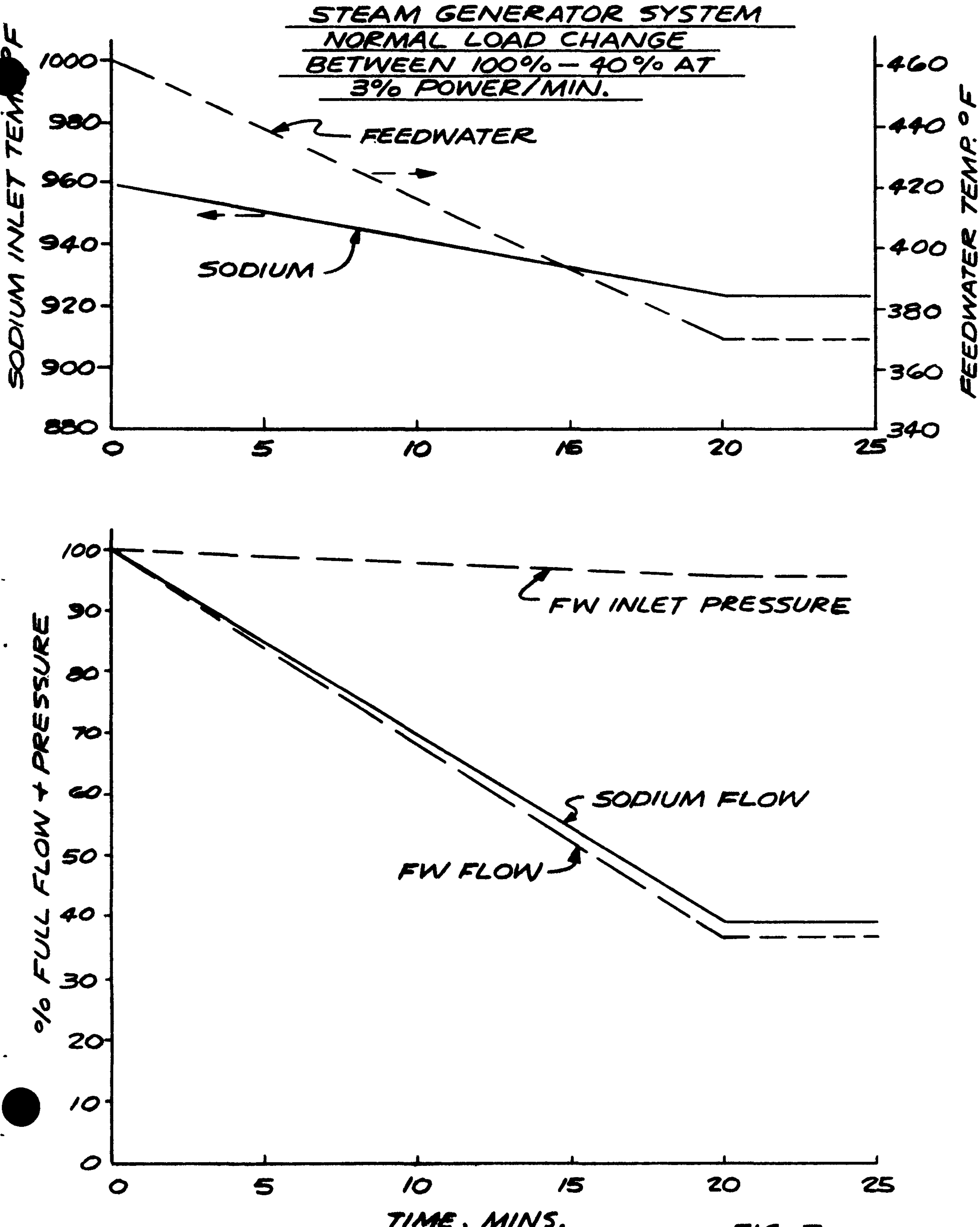
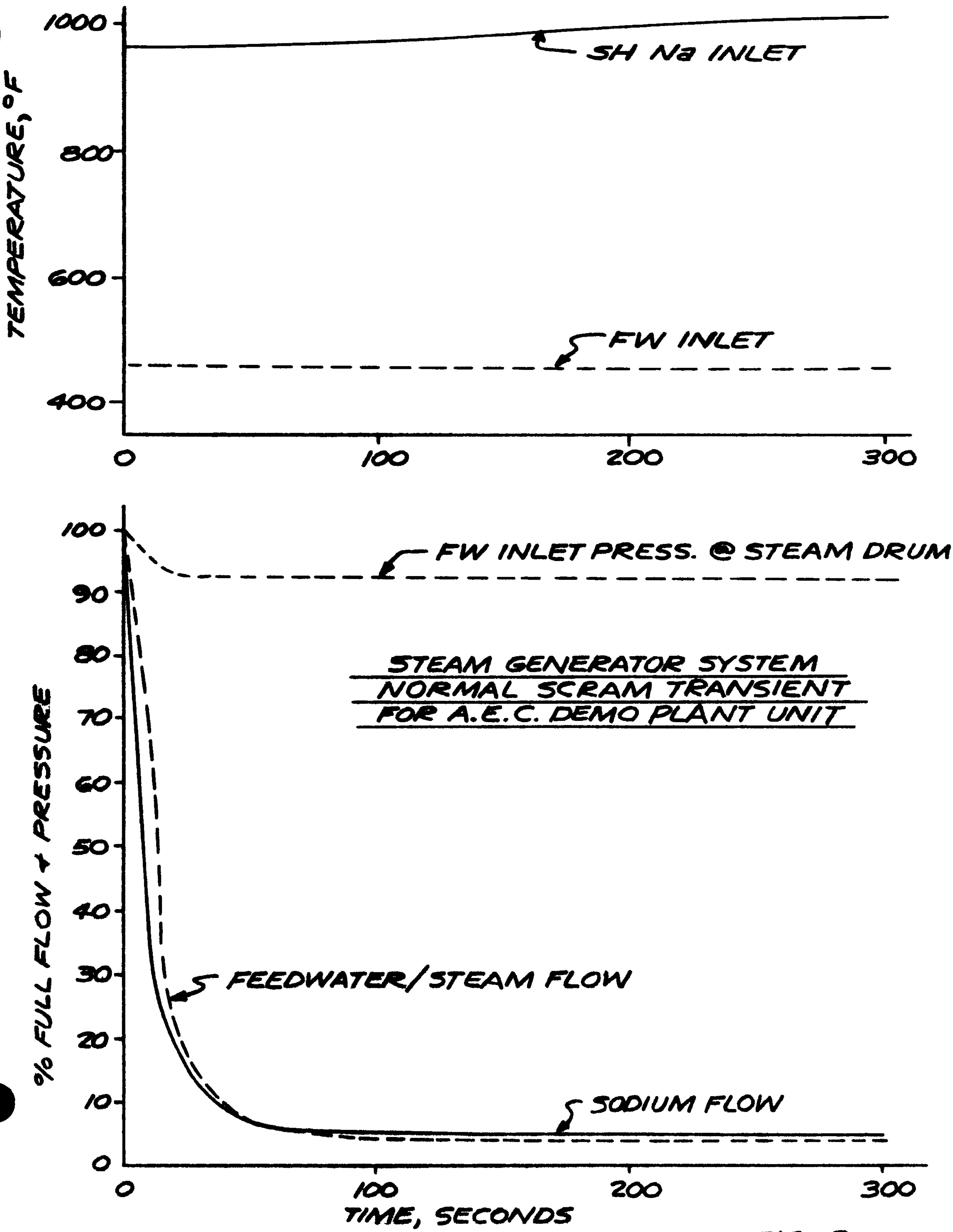

STEAM GENERATOR SYSTEM
NORMAL LOAD CHANGE
BETWEEN 100% – 40% AT
3% POWER/MIN.
SODIUM INLET TEMP. °F
FEEDWATER TEMP. °F
FEEDWATER
SODIUM
1000
980
960
940
920
900
880
460
440
420
400
380
360
340
0
5
10
15
20
25
% FULL FLOW + PRESSURE
FW INLET PRESSURE
SODIUM FLOW
FW FLOW
100
90
80
70
60
50
40
30
20
10
0
TIME, MINS.

FIG. 7



1000
800
600
400
TEMPERATURE, °F
SH Na INLET
FW INLET
0
100
200
300
100
90
80
70
60
50
40
30
20
10
0
% FULL FLOW + PRESSURE
FW INLET PRESS. @ STEAM DRUM
STEAM GENERATOR SYSTEM
NORMAL SCRAM TRANSIENT
FOR A.E.C. DEMO PLANT UNIT
FEEDWATER/STEAM FLOW
SODIUM FLOW
0
100
200
300
TIME, SECONDS

FIG. 8

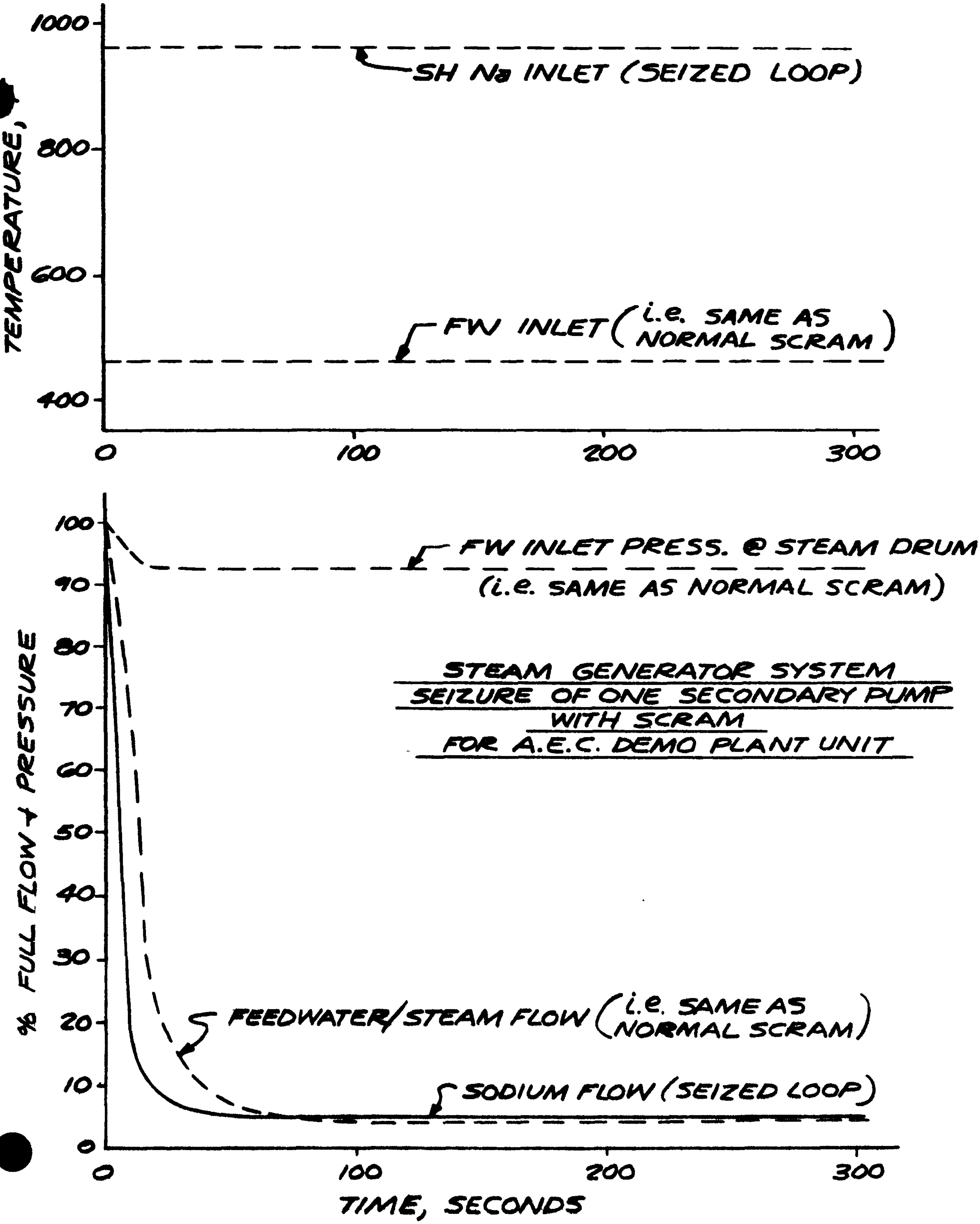
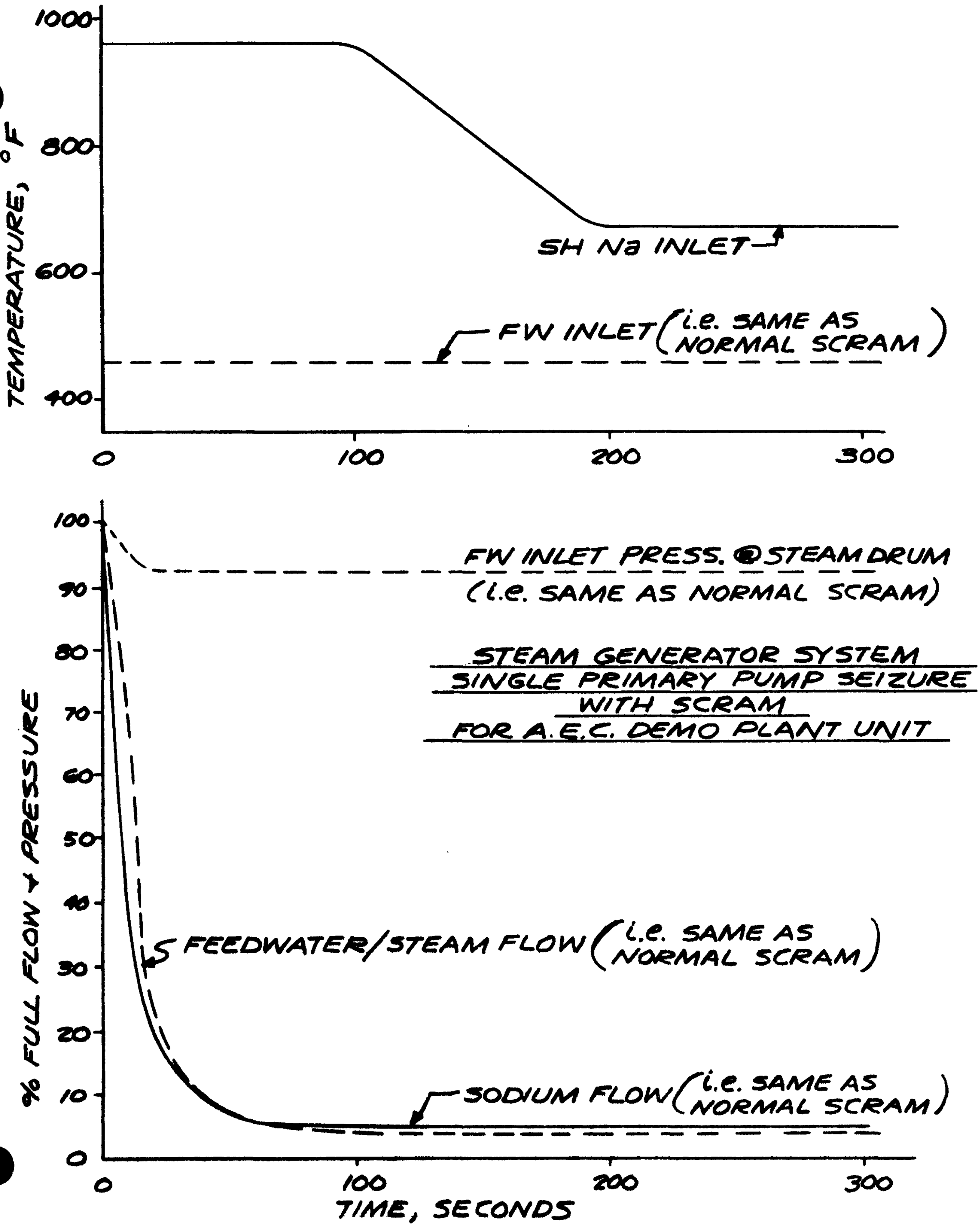
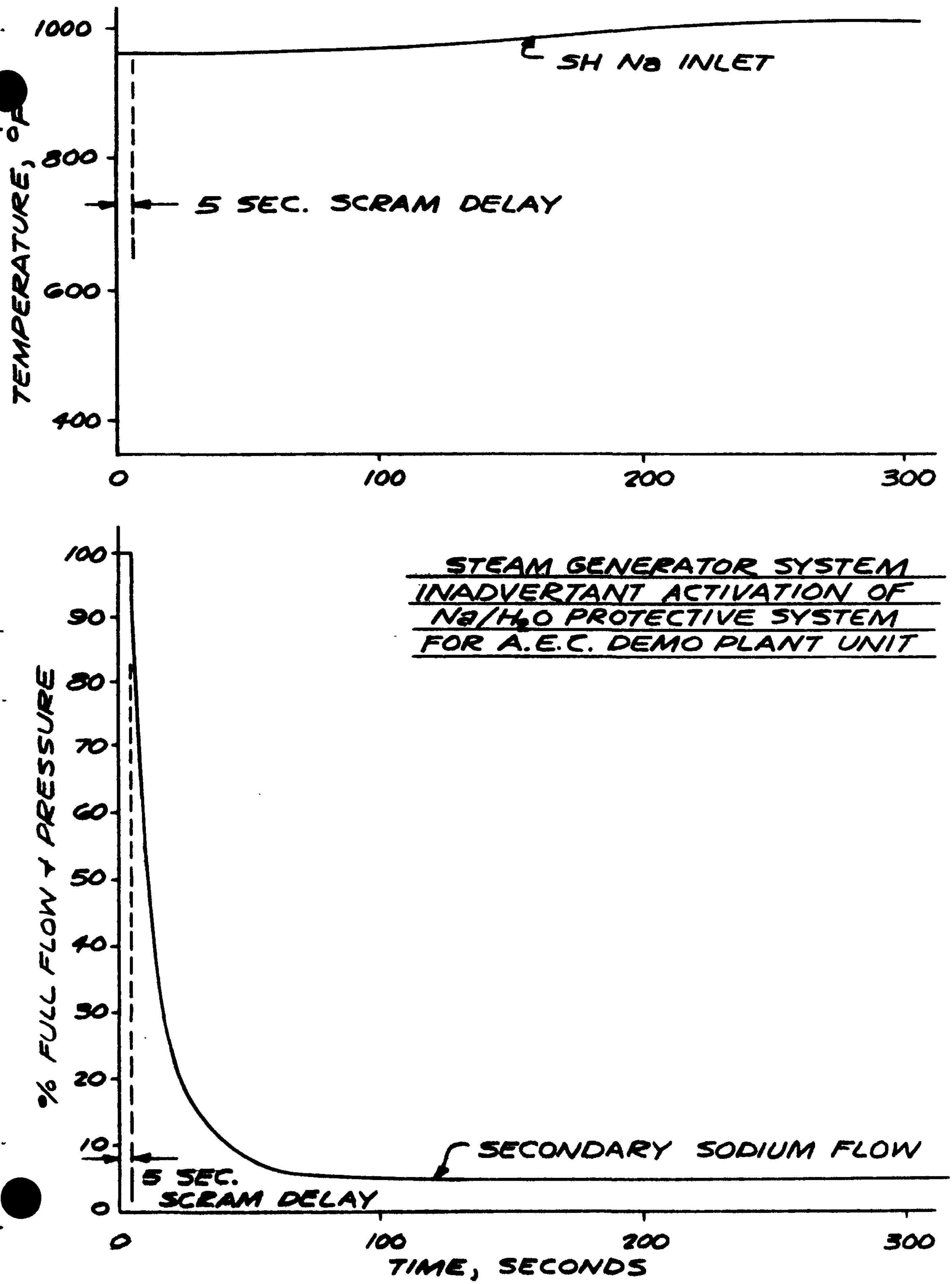

1000
800
600
400
TEMPERATURE, °F
SH Na INLET (SEIZED LOOP)
FW INLET (i.e. SAME AS NORMAL SCRAM)
0
100
200
300
100
90
80
70
60
50
40
30
20
10
0
% FULL FLOW + PRESSURE
FW INLET PRESS. @ STEAM DRUM (i.e. SAME AS NORMAL SCRAM)
STEAM GENERATOR SYSTEM
SEIZURE OF ONE SECONDARY PUMP
WITH SCRAM
FOR A.E.C. DEMO PLANT UNIT
FEEDWATER/STEAM FLOW (i.e. SAME AS NORMAL SCRAM)
SODIUM FLOW (SEIZED LOOP)
0
100
200
300
TIME, SECONDS

FIG. 9



1000
800
600
400
TEMPERATURE, °F
SH Na INLET
FW INLET (i.e. SAME AS NORMAL SCRAM)
0
100
200
300
100
90
80
70
60
50
40
30
20
10
0
% FULL FLOW + PRESSURE
FW INLET PRESS. @ STEAM DRUM (i.e. SAME AS NORMAL SCRAM)
STEAM GENERATOR SYSTEM
SINGLE PRIMARY PUMP SEIZURE
WITH SCRAM
FOR A.E.C. DEMO PLANT UNIT
FEEDWATER/STEAM FLOW (i.e. SAME AS NORMAL SCRAM)
SODIUM FLOW (i.e. SAME AS NORMAL SCRAM)
TIME, SECONDS

FIG. 10



1000
800
600
400
TEMPERATURE, °F
SH Na INLET
5 SEC. SCRAM DELAY
0
100
200
300
100
90
80
70
60
50
40
30
20
10
0
% FULL FLOW + PRESSURE
STEAM GENERATOR SYSTEM
INADVERTANT ACTIVATION OF
Na/H2O PROTECTIVE SYSTEM
FOR A.E.C. DEMO PLANT UNIT
SECONDARY SODIUM FLOW
5 SEC.
SCRAM DELAY
0
100
200
300
TIME, SECONDS

FIG. 11

TABLE II A

RANGE OF STEADY STATE OPERATING CONDITIONS

	CONDITION			
	100% Power	80	60	40
Heat Load, MWt	110	88	66	44
SODIUM SIDE				
Sodium Flow Rate (lb./hr.)	4.35×10^6	3.48×10^6	2.61×10^6	1.74×10^6
Sodium Inlet Temp. (^oF)	960	945	933	926
Sodium Outlet Temp. (^oF)	673	665	658	656
Sodium Inlet Pressure (psig)		Later		
Sodium Δ P (psi)	18	11.5	6.5	2.9
STEAM-WATER SIDE				
Feedwater Flow (lbs. hr.)	3.97×10^5	3.18×10^5	2.38×10^5	1.59×10^5
Feedwater Temp. (^oF)	460	440	412	370
Feedwater Pressure (psia)	2666	2624	2590	2563
Recirculation Ratio	3.5	4.0	6.0	9.0
Steam Outlet Temp. (^oF)	900	900	900	900
Steam Outlet Pressure (psig)	2500	2500	2500	2500

DESIGN CONDITIONS

TABLE II B

Item	Superheater Design Press/Temp.	Evaporator Design Press/Temp.
Heat transfer tubes	2600 psia/960°F	2700 psia/866°F
Tubesheet	2700 psia/750°F	2750 psia/750°F
Sodium shell	250 psia/960°F	250 psia/866°F
Channel shell	2700 psia/750°F	2750 psia/750°F

TABLE 3

DRAWING SUBMITTAL REQUIREMENTS

ITEM	DESCRIPTION	NO. OF REPRODUCIBLE DOCUMENTS REQUIRED	REF.	SUBMIT FOR APPROVAL INFO.	TIME OF SUBMITTAL
1.	Preliminary Outline Drawings Initial Submittal Final Submittal				
2.	Assembly Drawings Initial Submittal Final Submittal				
3.	Final Outline Drawings				
4.	Detail Drawings				
5.	Drawings & Spec. List				
6.	Paris List				
7.	Weights and Center of Gravity				

<u>TABLE 4</u>

<u>REPORTS AND DOCUMENTS SUBMITTAL</u>

ITEM	DESCRIPTION	NO. OF REPRODUCIBLE DOCUMENTS REQUIRED	REF.	SUBMIT FOR APPROVAL INFO.	TIME OF SUBMITTAL
1.	Design Reports				
2.	Interim Reports				
3.	Q.A. Program Plan				
4.	Inspection and Test Plan				
5.	Process Special Control & NDT Procedures				
6.	Inspection & Test Procedures				
7.	Nonconforming Item Documentation				
8.	Handling, Pres., Packing, & Storage Procedures				
9.	Proposed New Design Criteria				
10.	Design Description				
11.	Quality Records				
12.	Periodic and Progress Reports				
13.	Operations and Maintenance Manual				

PRELIMINARY DATA EVALUATION:

EFFECTS ON SIZING AND

RECOMMENDATIONS CONCERNING

FUTURE TESTING

CONTRACT 7670

C.E. LMFBR STEAM GENERATOR

DEVELOPMENT PROGRAM

SUBTASK 2.2

INSULATOR TUBE TEST

APPENDIX C

January 18, 1973

Prepared by: ____________

Approved by: ____________

Introduction

Over the preceding several weeks, Combustion Engineering's KDL PWR test loop facility has been utilized to experimentally verify the effectiveness of the bayonet assembly, which comprises a critical component of the CE LMFBR steam generator design.

The purpose of this interim report is to summarize the effect of those tests on the CE reference design, and to set forth CE recommendations with regard to future testing.

Only a limited amount of data, directly applicable to the CE design, has been considered in this preliminary study. The remainder of the considerable data available will be evaluated in the formal and complete report of the insulator tube test program.

Discussion

The CE reference steam generator concept utilizes a bayonet tube configuration in both the evaporator and superheater modules. This configuration, along with the flow arrangement of the two fluids, allows CE's steam generator to enjoy several advantages over competing designs such as unrestrained tubes, tubesheets which operate below the creep range and unexceeded accessability for in-service inspection and maintenance. Details of this concept are shown in Figure 1.

In order to utilize a bayonet arrangement, however, an effective barrier against regenerative heat transfer across the bayonet tube must be found. The CE design utilizes a double bayonet tube with an "insulating" gap between to provide the required barrier. Figure 2 is a schematic representation of this "bayonet assembly".

The purpose of the insulator tube test program was to verify the effectiveness of the bayonet assembly as a barrier to regenerative heat transfer.

A six foot test section similar to the bayonet assembly was operated in the PWR test loop at KDL. Both the evaporator and superheater configurations were simulated. Figures 3 and 4 are schematics of the test loop configurations for simulation of the evaporator and superheater respectively. Complete details of the test program may be found in Ref. (1).

Summary of Results - Evaporator Configuration

The test loop was first operated in the evaporator configuration . The results of this series indicated that regenerative heat transfer will cause a temperature rise in the downcomer of approximately 5^{o}F. This corresponds to a Nusselt number of approximately 3.5 to 4 for the insulator gap. The effect of this regenerative heat transfer is negligable, and the effect on sizing cannot be detected within the accuracy of our current methods.

Summary of Results - Superheater Configuration

While the effects of regenerative heat transfer in the evaporator were found to be negligable, the opposite case holds true in the superheater, within which much higher temperature differences across the insulation gap are found.

Figures 5, 6 and 7 represent a portion of the data from the superheater series at three pressure levels 1500, 1900 and 2600 psia. The data covers a range of flow rates and average insulator gap temperatures.

Although the data, to be best understood, must be reduced to non-dimensional terms, three general conclusions can be drawn from the general trend of the data. First, the flow rate has a preceptable, but not important, effect on the heat losses. This is expected due to the low overall contribution of the flow film resistances to the overall resistance. At lower flow ratings the resistance tends to be slightly greater. Secondly, operating pressure has a significant effect, with more heat being lost at the higher pressure levels. Third, the trend at the highest driving temperature differences ($>$ 300F) indicates a probable radiation component.

Some scatter appears in the data, with a few points showing unusually high losses. These few points are thought to result from data taken during periods where leaks occurred at the flanges in the test section. This meant that the actual flow rate in the test section was lower than measured, and hence, indicated heat loss(from WCp ΔT at 2 points in the bayonet tube) was much worse than actual. This will be further investigated as data reduction proceeds.

Effect on Superheater Sizing

Certain groups of data were directly applicable to the conditions in the CE LMFBR superheater, and these data points were used to reset the sizing of the Demonstration Plant Superheater.

Effect on Superheater Sizing - continued

The procedure for this evaluation was as follows:

1. Select data points relevant to the conditions in the superheater, and make a plot of the heat loss per foot of test section versus the temperature difference across the bayonet assembly. The applicable points chosen were those at 2600 psia, and flow rates above 1800 #/hr. These points were ploted in Figure 8 and a conservative design curve was determined therefrom.

2. The dimensions of the simulated bayonet assembly in the test are different from those of the present steam generator bayonet assemblies in that the diameters are larger in the steam generator (See Figure 9). A correction factor for the increased area per foot was determined as follows:

$$f = \frac{1/2\,(.916 + .636)}{1/2\,(.75 + .4925)} = 1.25$$

The differences in t_1 and t_2 were neglected.

3. An estimate of the superheater size was made and a plot of the temperature difference across the bayonet assembly versus length was created using the performance program (See Figure 10).

4. The superheater was divided into four foot lengths from the sodium level down, and using Figures 8 and 10, an estimate of the heat loss in each section made. The approximately four feet of bayonet assembly above the sodium level was also taken into account, assuming the highest temperature difference. Table 1 shows the computation of the heat loss in the superheater.

5. Knowing the flow rate in the superheater, an enthalpy balance was made to determine the temperature required at the entrance to the riser tube.

The flow rate in each tube is about 2337 #/hr. in a superheater with 170 tubes. A trial and error enthalpy balance yields a required inlet temperature of $\sim$ 925.6F (at 2525 psia) to result in conditions of 900°F at 2500 psia at the outlet.

6. The performance program was run with various multipliers on the insulator gap fluid conductivity to match the required conditions at the bottom of the tube. This resulted in a steam generator with a somewhat larger area than previously sized. A summary is presented in Table 2. The indicated multiplier on the conductivity (Nusselt number) was 4.7.

Comparison with Predicted Values

A value of the Nusselt number may be estimated by a correlation based on Jakob's study of convection heat transfer in vertical enclosed air spaces. The conclusion is not directly applicable, since the ratio of length to gap width in the insulator gap greatly exceeds the range of the correlation. A conservative estimate of the heat loss, however, should be obtained by assuming the ratio to be 42.2, the maximum investigated. Using this procedure, the Nusselt number is predicted to be about 3.1, as compared with 4.7 as indicated above. Details of the calculations are presented in Appendix 1.

Recommendations for Future Testing

In the light of the results of the insulator tube testing to date, a decision must now be made concerning continuation of the test program with a different (smaller) gap. The following facts are pertinent to such a decision.

1. The present configuration is adequate for the present demonstration plant steam generator design.

2. There appears to be an increasing radiation component at higher temperature differences ($> 300^{\circ}F$).

3. The aforementioned correlation for convection heat transfer predicts only small effects from variation of gap width. Specifically, with larger gaps, the value of n would become 1/3 ($Gr > 2.1 \times 10^5$) and the heat transferred would be essentially independent of gap width. With a smaller gap, some added resistance is predicted down to the point where conduction begins to predominate ($Gr < 2 \times 10^3$). For a gap half the size of the current test, the predicted resistance would be about 15% higher.

Based on the above facts, it is CE's view that another test with a different insulator gap would be desirable from a scientific aspect in that further information on the nature of heat transfer across such gaps would be obtained.

From a practical aspect, however, the current gap configuration is considered adequate for the intended use, and the available analytical tool predicts that little if any added resistance would be gained by variation. (Conversely, it predicts that the effectiveness of the gap will be relatively insensitive to manufacturing tolerances, etc). This viewpoint would tend to indicate that no further testing is needed.

Looking forward to future designs, it is estimated, based on this preliminary assessment, that the present insulator tube would be adequate for design steam temperatures up to 950^{o}F assuming saturation pressures on the order of 2500 psia. For steam generators incorporating even higher superheat temperatures at that pressure, an advanced insulator design, possibly with smaller multiple gaps may be required, and additional testing may be desirable at that time.

TABLE I HEAT LOSS ACROSS BAYONET ASSEMBLY IN SUPERHEATER

Increment No.	Length	T	Uncorrected Loss/Ft.	Uncorrected Loss - 4Ft.
1	4 ft.	223 F	1830 $\frac{BTU}{FT\ Hr}$	7320 BTU/HR
2		193	1510	6040
3		152	1140	4560
4		110	800	3200
5		74	530	2120
6		44	310	1240
7		20	140	560
8	2.5	5	30	75
Above Na	4	233	1950	7800
			Total	32,915

Correction for Area:

QL = 32915 x 1.25 = 41144 BTU/HR

TABLE 2 EFFECT ON SIZING OF SUPERHEATER

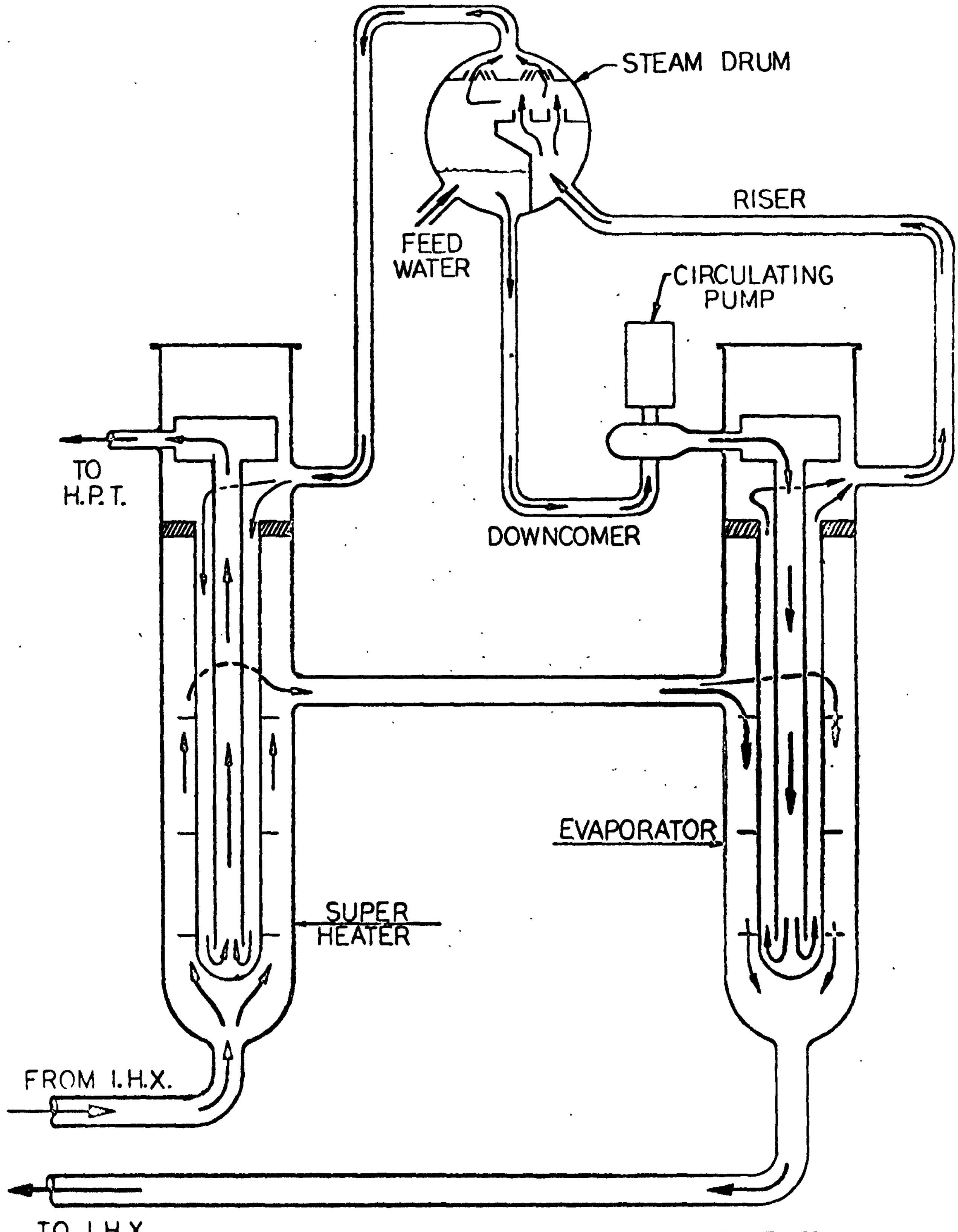
	Number Tubes*	Length	Area
Original Design	150	30.8 ft.	1814 ft^2
Present Design	170	31.1 ft.	2076 ft^2

*Does not include design margins.

REFERENCES

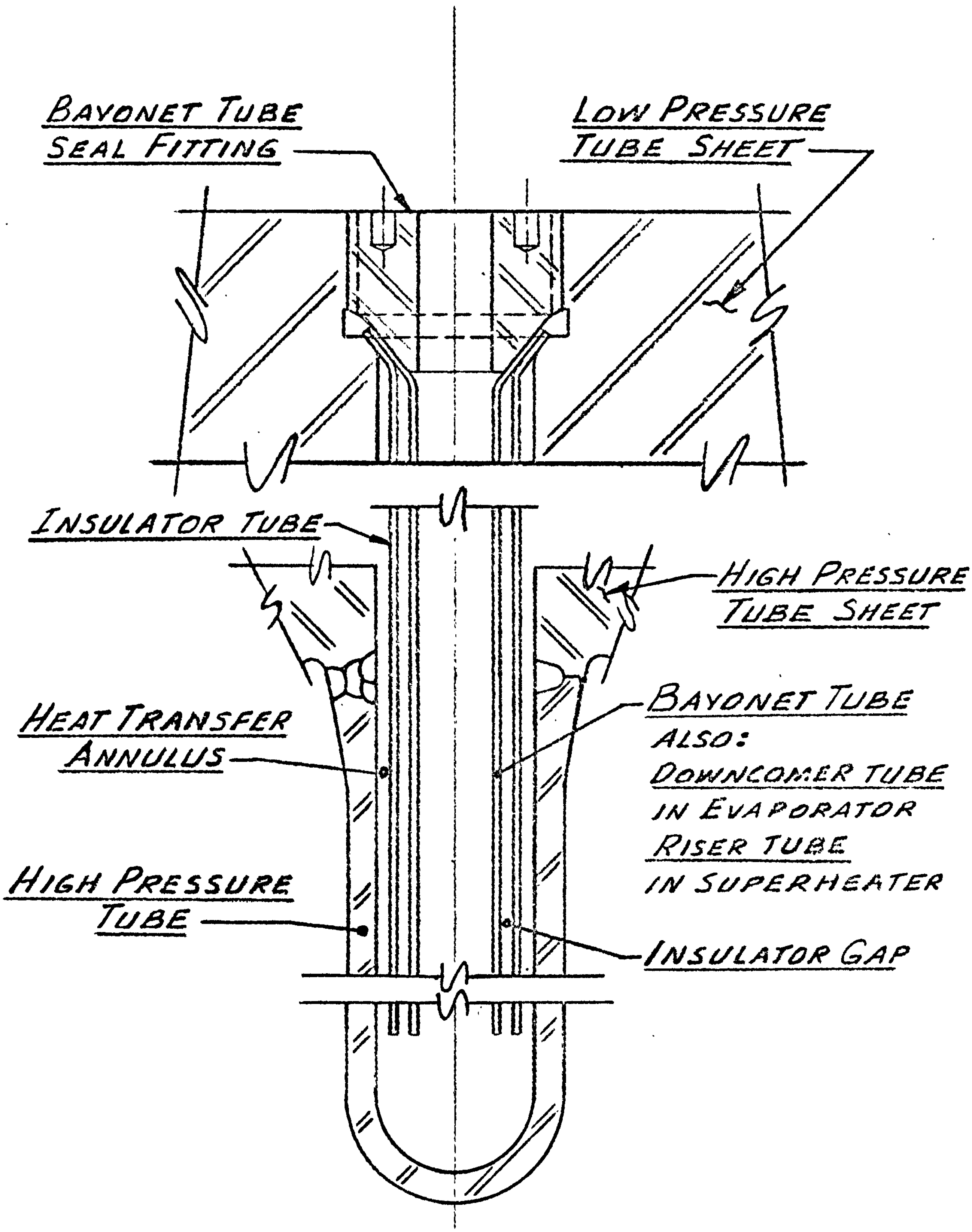
1. Letter submittal A. A. Tuzes to H. N. Miller, "Insulator Tube Test Program" dated 12/12/72.

2. McAdams, W. H., "Heat Transmission", McGraw-Hill Book Co., Inc., New York, (Third Edition - 1954) pp 181-2.



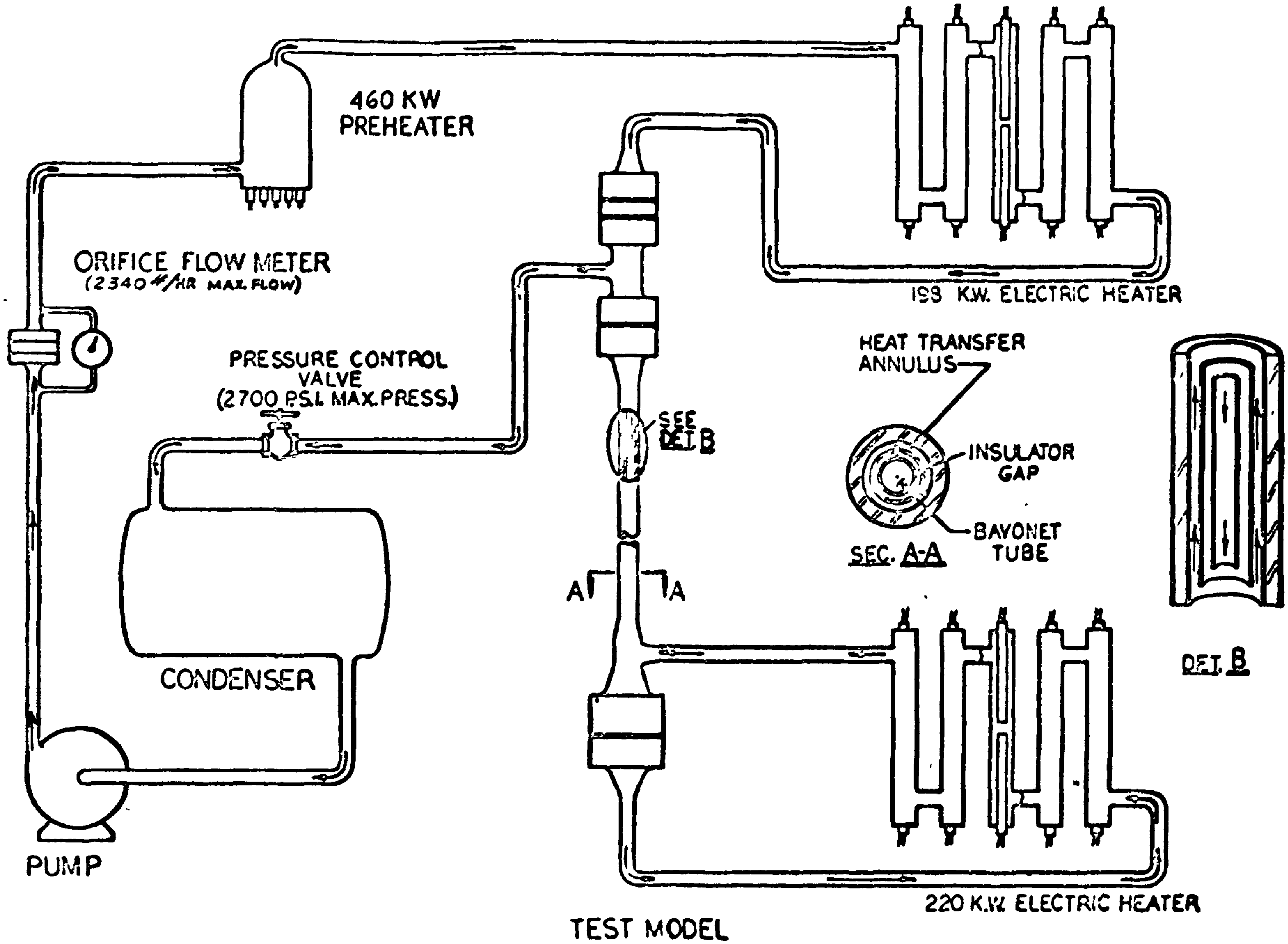
SCHEMATIC ARRANGEMENT
STEAM GENERATION SYSTEM

FIGURE I



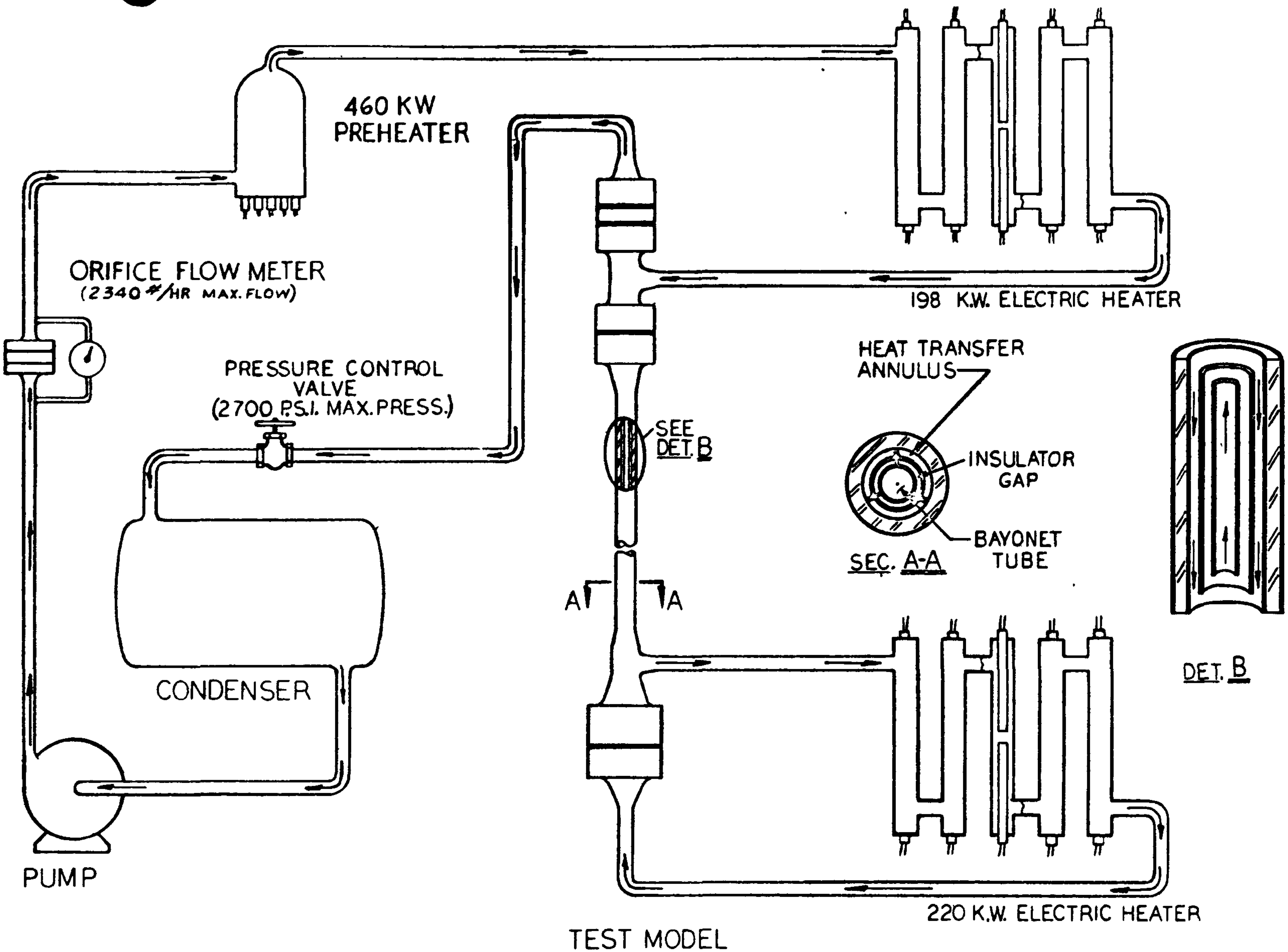
BAYONET TUBE ASSEMBLY

FIGURE 2



EVAPORATOR TEST LOOP FOR INSULATOR TUBE TEST

FIGURE 3



SUPERHEATER TEST LOOP FOR INSULATOR TUBE TEST

FIGURE 4

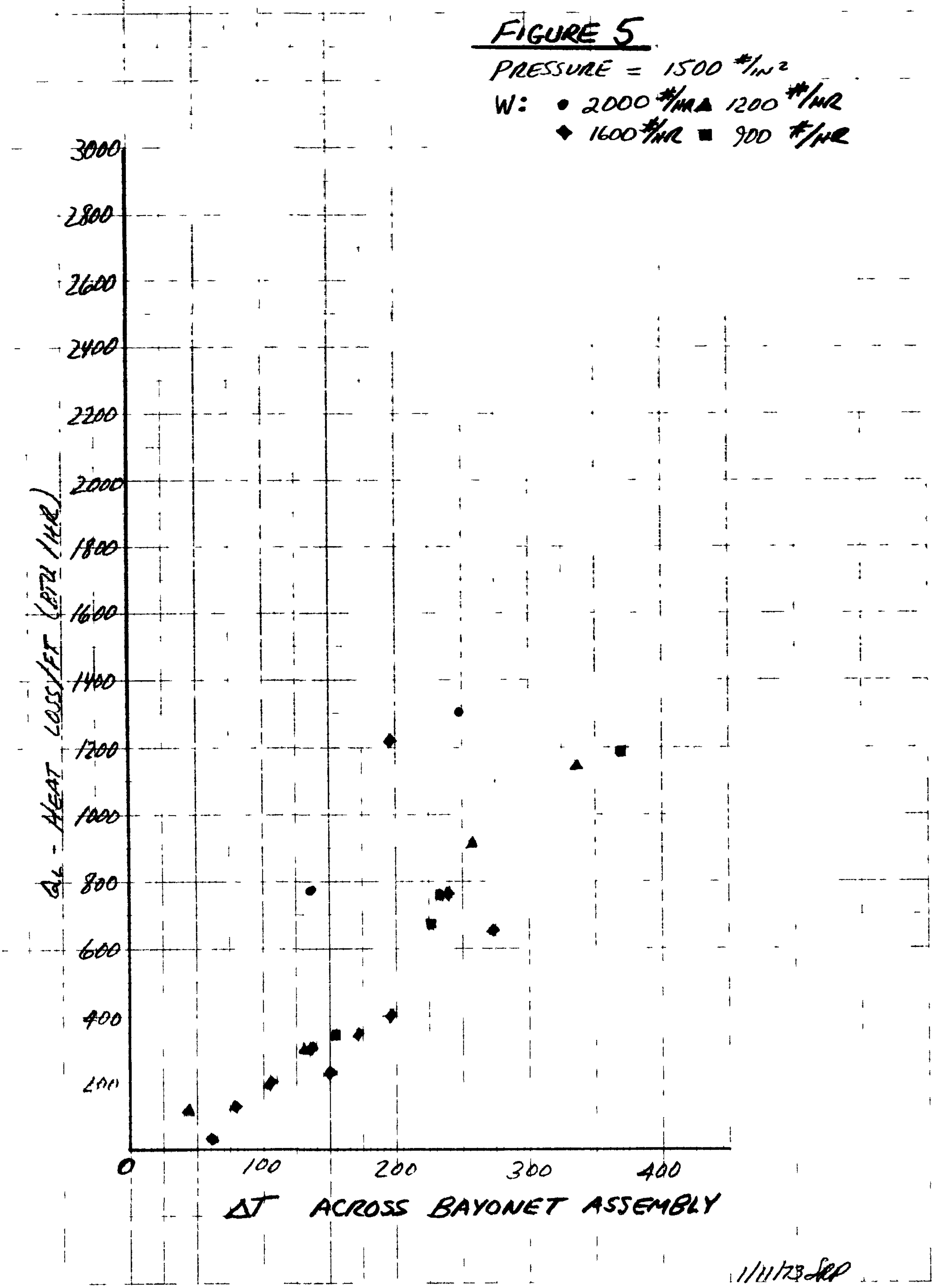


FIGURE 5
PRESSURE = 1500 #/in²
W: ● 2000 #/HR ▲ 1200 #/HR
◆ 1600 #/HR ■ 900 #/HR
QL - HEAT LOSS/FT (BTU/HR)
3000
2800
2600
2400
2200
2000
1800
1600
1400
1200
1000
800
600
400
200
0
100
200
300
400
ΔT ACROSS BAYONET ASSEMBLY
1/11/73 SRP

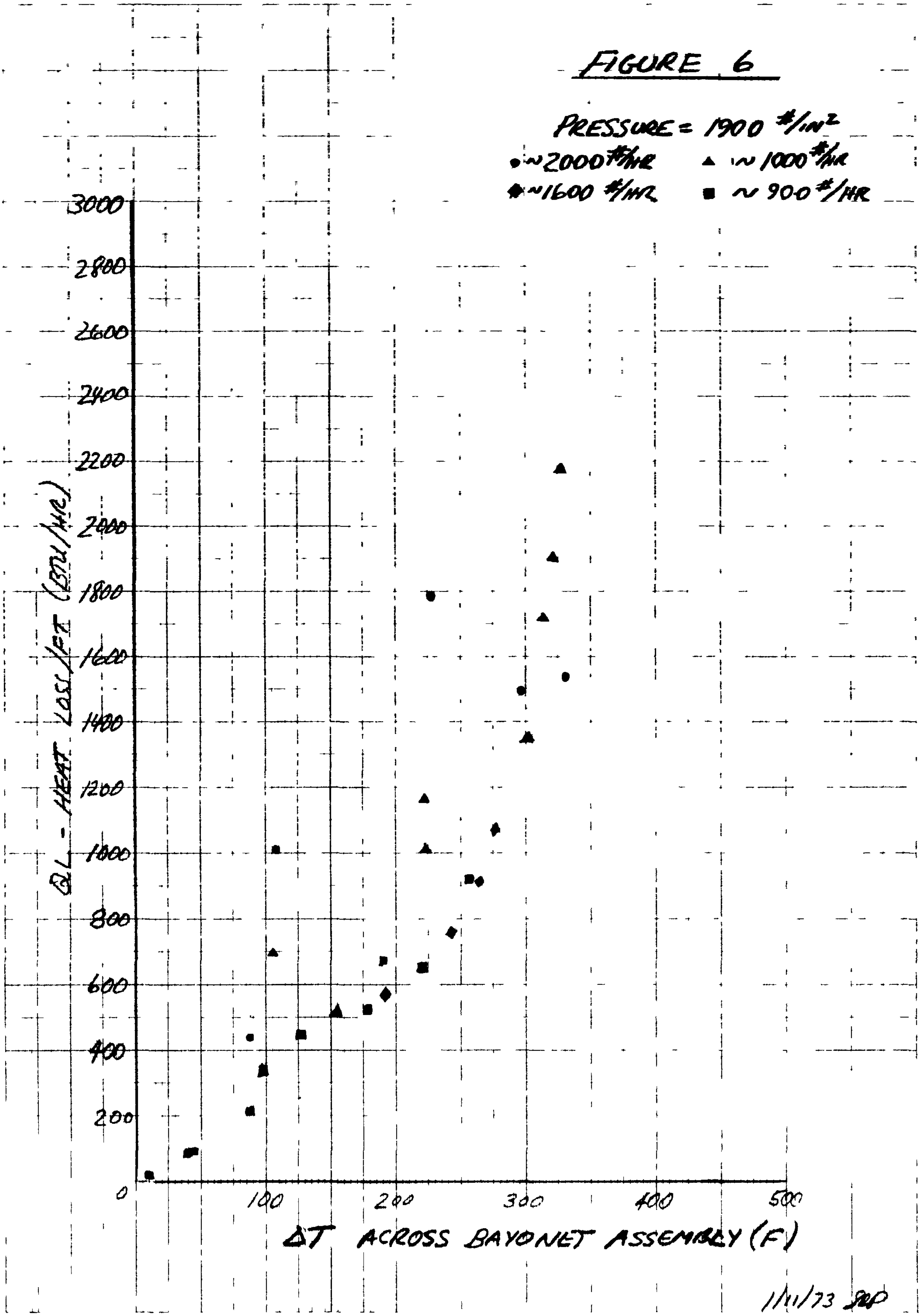


FIGURE 6
PRESSURE = 1900 #/in²
• ~2000 #/HR
▲ ~1000 #/HR
◆ ~1600 #/HR
■ ~900 #/HR
QL - HEAT LOSS/FT (BTU/HR)
3000
2800
2600
2400
2200
2000
1800
1600
1400
1200
1000
800
600
400
200
0
100
200
300
400
500
ΔT ACROSS BAYONET ASSEMBLY (F)
1/11/73

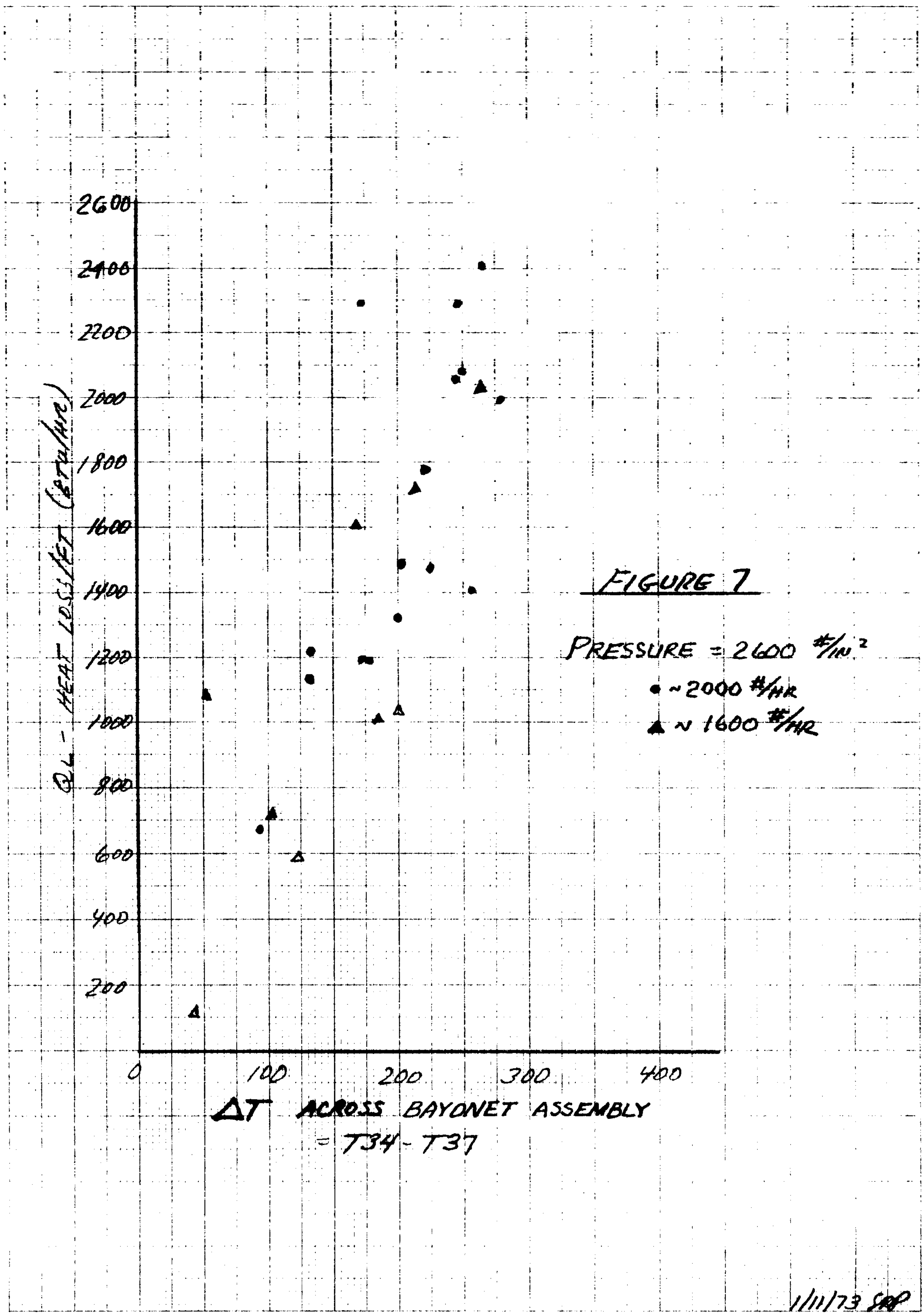


FIGURE 7
PRESSURE = 2600 #/in.²
• ~2000 #/hr
▲ ~ 1600 #/hr
QL - HEAT LOSS/FT (BTU/HR)
2600
2400
2200
2000
1800
1600
1400
1200
1000
800
600
400
200
0
100
200
300
400
ΔT ACROSS BAYONET ASSEMBLY
= T34 - T37
1/11/73 SAP

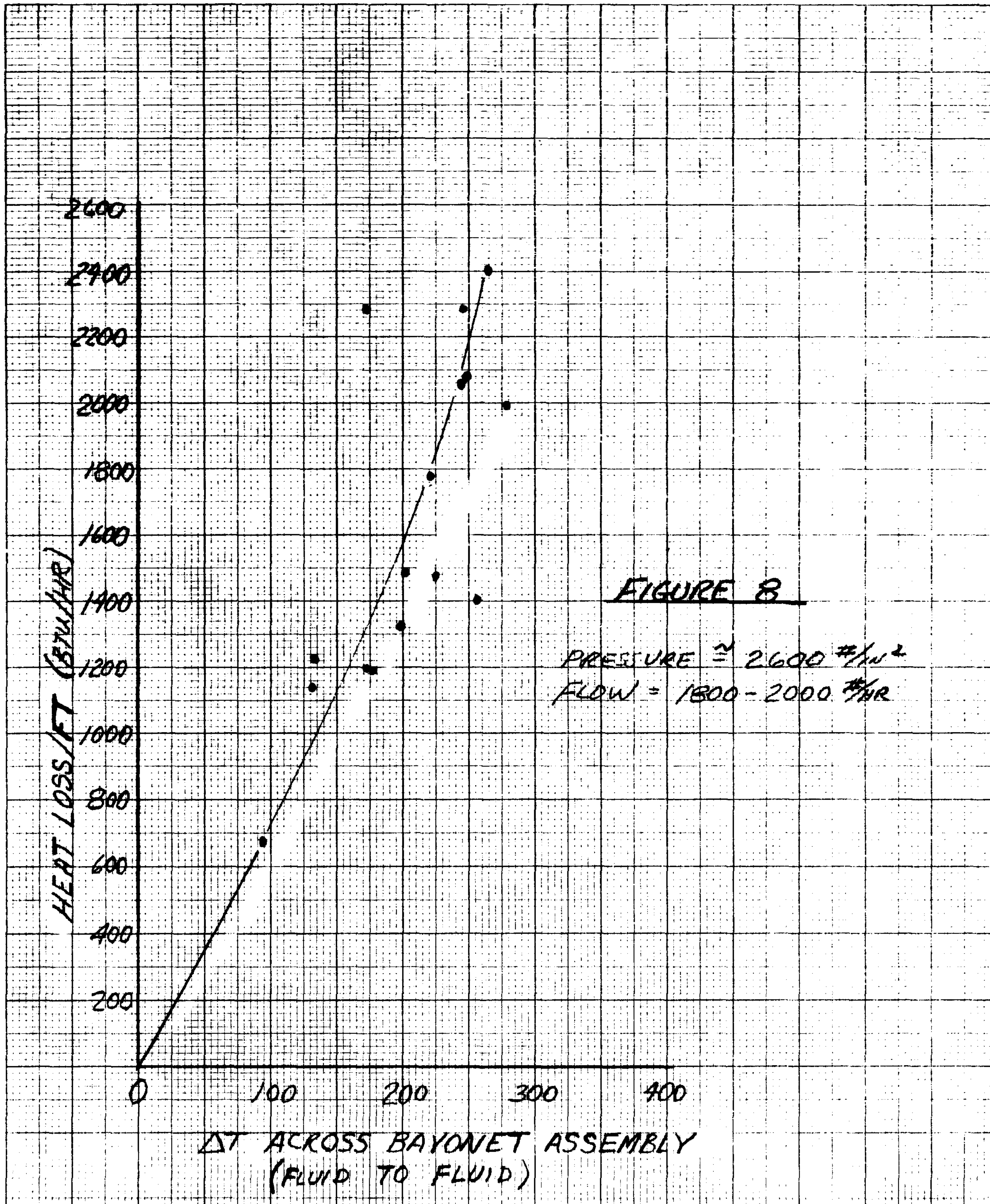


FIGURE 8

PRESSURE ≅ 2600 #/IN²
FLOW = 1800 - 2000 #/HR

1/12/73

COMBUSTION ENGINEERING, INC.

ENGINEERING DEPARTMENT, CHATTANOOGA, TENN.

CHARGE NO. 7670

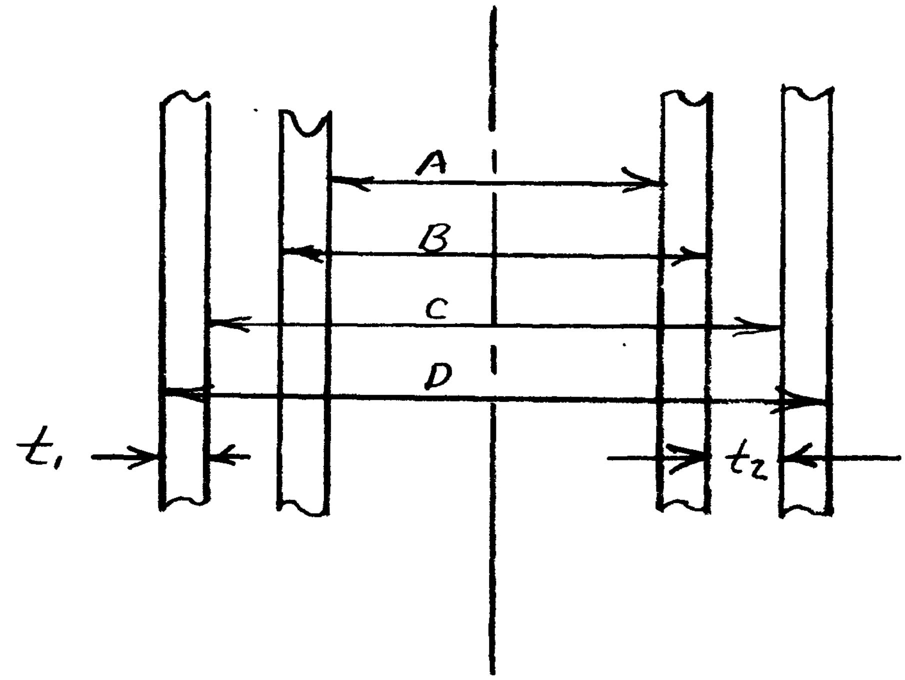
DESCRIPTION INSULATOR TUBE TEST

NUMBER

SHEET ____ OF ____

DATE 1/17/73 BY

CHECK DATE ____ BY ____



DIMENSION	A	B	C	D	t_1	t_2
INSULATOR TUBE TEST	.4925	.5625	.68	.75	.035	.0588
STEAM GEN.	.636	.716	.836	.916	.040	.060

FIGURE 9 COMPARISON OF INSULATOR TUBE TEST WITH STEAM GENERATOR BAYONET ASSEMBLY

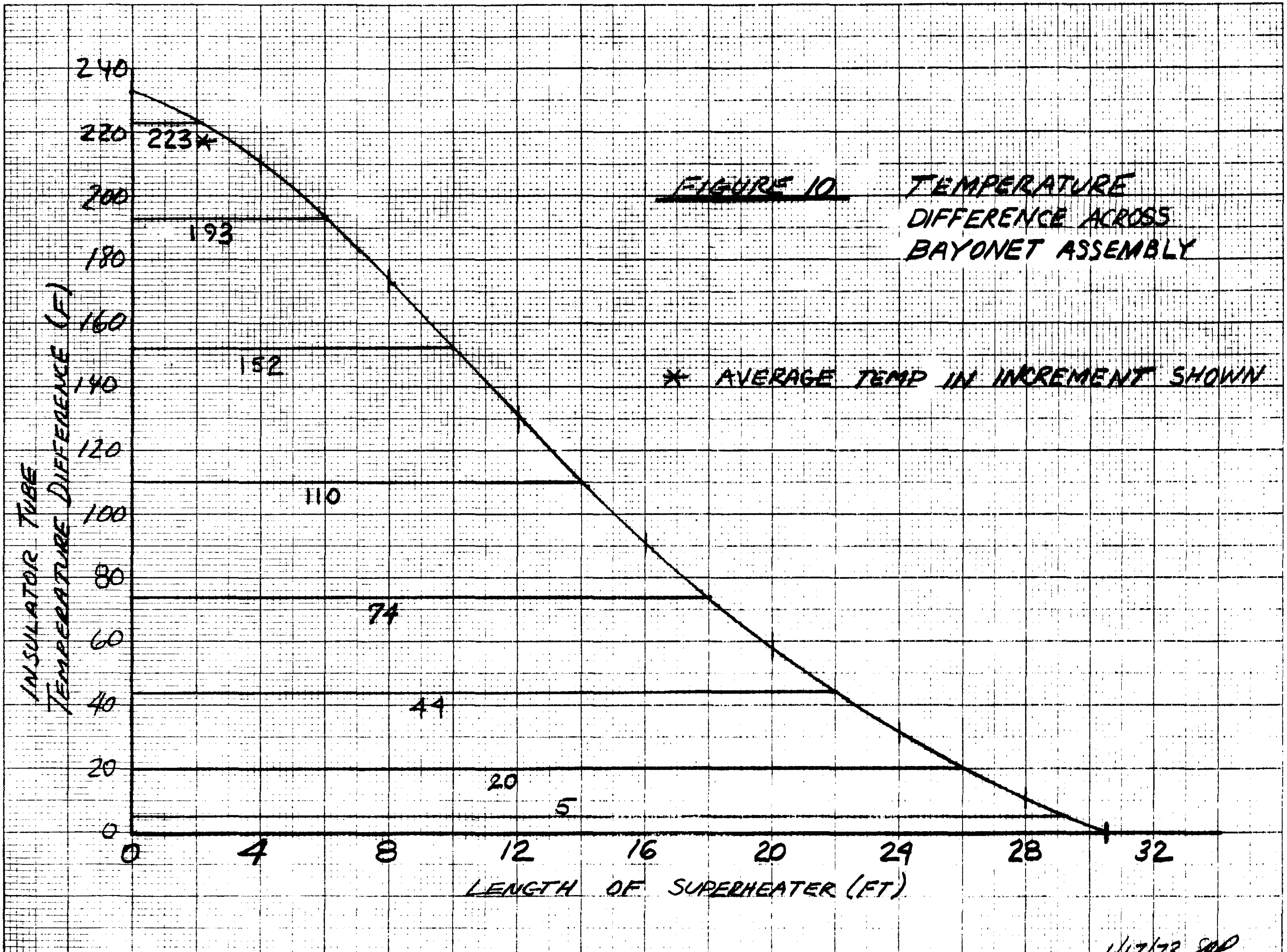


FIGURE 10 TEMPERATURE DIFFERENCE ACROSS BAYONET ASSEMBLY
* AVERAGE TEMP IN INCREMENT SHOWN
INSULATOR TUBE TEMPERATURE DIFFERENCE (F)
LENGTH OF SUPERHEATER (FT)
240
220
200
180
160
140
120
100
80
60
40
20
0
4
8
12
16
20
24
28
32
223*
193
152
110
74
44
20
5
1/17/73 SAP

Comparison with Predicted Values

A correlation may be found in Ref 2. for heat transfer between parallel plates.

$$\frac{h_c' x}{k_f} = \frac{C}{(L/x)^{1/9}}\left[(Gr)_f\,(Pr)_f\right]^n$$

where: $(Gr)_f = \dfrac{x^3 \rho_f^2 g \beta_f \Delta T}{\mu_f^2}$

$(Pr)_f$ – Prandtl Number evaluated under film conditions
x – Gap width
ρ_f – Density (film)
g – Acc. of gravity
β – Coeff. of thermal expansion (film)
ΔT – Temp across gap
μ_f – Viscosity (film)
L – Length
k_f – Fluid conductivity
h_c' – H.T. coefficient based on ΔT

For assumed gap conditions of:

$P = 2600$ #/in² $\quad T = 750°F$

$\rho_f = 5.238$ #/ft³ $\quad \beta_f = .003117/°F$

$\mu_f = 1.898 \times 10^{-5}$ #/ft-sec $\quad Pr = 1.59$

COMBUSTION ENGINEERING, INC.

ENGINEERING DEPARTMENT, CHATTANOOGA, TENN.

CHARGE NO. 7670

DESCRIPTION Appendix I

NUMBER

SHEET 2 OF 3

DATE 1/17/73 BY SMP

CHECK DATE 1/18/73 BY JWA

$$(Gr)_f = \frac{\left(\frac{.06}{12}\right)^3 (5.238)^2 (32.17)(.00317)(\Delta T)}{(1.898 \times 10^{-5})^2}$$

$$Gr = 955 \Delta T$$

FOR $\Delta T \cong 200$ $Gr \cong 1.9 \times 10^5$

FOR $2.0 \times 10^4 < Gr < 2.1 \times 10^5$

$C = .20$ $n = 1/4$

$$\therefore \frac{h'_c x}{k_f} = \frac{.20}{(L/x)^{1/9}} \left[(Gr)_f (Pr)_f \right]^{1/4}$$

NOTE THAT AS L IS INCREASED h'_c SLIGHTLY DECREASES, FOR CONSERVATISM TAKE THE MAX L/x INVESTIGATED = 42.2

$$\therefore \frac{h'_c x}{k_f} = \frac{.20}{(42.2)^{1/9}} \left[(1.9 \times 10^5)(1.59) \right]^{1/4}$$

$$\frac{h'_c x}{k_f} = 3.1$$

NOTE THAT FOR A GAP OF $X = .03''$

$$Gr = \frac{1.9 \times 10^5}{8} = 2.375 \times 10^4$$

AGAIN LEAVING $L/X = 42.2$;

$$\frac{h'_c X}{K_f} = 3.1 \left(\frac{2.375}{19} \right)^{1/4} = 1.8$$

BUT COMPARING h'_c FOR BOTH ;

$$\frac{\left(\frac{h'_c X}{K_f} \right)_{.06}}{\left(\frac{h'_c X}{K_f} \right)_{.03}} = \frac{3.1}{1.8}$$

$$\frac{h'_{c(.06)}}{h'_{c(.03)}} = \frac{3.1}{1.8} \times \frac{.03}{.06} = .86$$

APPENDIX D

EFFECT OF STEAM PRESSURE ON DESIGN

EFFECT OF STEAM PRESSURE ON DESIGN

1.0 Introduction

While the work presented in this steam generator conceptual design package has been based on a steam outlet pressure of 2500 psia (2400 at the turbine), it has been recognized throughout the period of this effort, however, that alternate, lower, pressures are being considered. The 2500 psia design pressure was selected by CE as presenting the greatest challenge to our design, analytical and fabrication efforts. Extrapolation to lower pressures was considered to be relatively straight forward, while extrapolation to higher pressures was seen to pose some difficulties.

This summary has been prepared to serve two purposes. First, to assist those responsible for setting plant parameters and, secondly, to indicate the capabilities of the CE generator design.

Portions of this summary are quantitative reflecting the results of development work already performed. Other portions, where specific information is not yet available, are addressed in qualitative terms. Where quantitative information was available, comparisons were made at the three pressures which are currently considered candidates: 1400, 1900 and 2400 psia.

2.0 Steam Generator Sizing

Due to the changing properties of water and steam, and effects on the mechanisms of heat transfer, the water/steam side pressure has a significant effect on the physical size of the steam generator.

2.1 Tube Thickness

As design pressure increases, the thickness of the high pressure tube, constituting part of the sodium water boundry, correspondingly increases. The resistance of the tube is a significant protion of the overall resistance to heat transfer and a reduction in design pressure, and hence thickness, will result in a notable size decrease in both the evaporator and superheater.

2.2 Saturation Temperature

For given inlet conditions, a change in the design pressure, and hence saturation temperature, can have a marked effect on the available LMTD, and therefore, generator size. Lower pressures, with accompanying lower saturation temperatures, result in a larger LMTD. The required heat transfer surface is therefore reduced.

2.3 Insulator Effectiveness

The effectiveness of the insulator gap as a barrier to regenerative heat transfer, particularly in the superheater, is dependent upon the thermal conductivity of the fluid. As the pressure is increased, the conductivity of the steam in the superheater increases and thus the efficiency of the insulator gap decreases. This is partially offset since temperatures differences across the insulator gap are lower at higher pressures.

2.4 Circulation Ratio

If the effect of pressure on the DNB heat flux-quality relationship were significant, this aspect of design could affect steam generator size, since a lower allowable circulation ratio would result in a more efficient economizer section.

The design circulation ratio was determined for 2500, 1900 and 1500 psia saturation pressures. The two higher pressure CR's were selected using pripriętary CE data as outlined in Section 3.1.5.3. The CR at 1500 was determined by an extrapolation of the data of BAW 3238-13. There were no significant differences indicated and a CR of 3.5 would be selected for all three cases.

2.5 Effect on Generator Size

A preliminary study was made to quantitatively evaluate the effects of the above mechanisms on generator size. Not included in this study were the results of the insulator tube testing (Appendix C), which were available at a later date.

The study was accomplished using the CE recirculation sizing program. The sodium outlet temperature was optimized for each steam outlet pressure to achieve a minimum weighted surface area in the generator and IHX combination. The following ground rules were used in the study.

1. As sodium temperature out of the steam generator is reduced, steam generator size increases and IHX size decreases; therefore, an optimum (minimum) size exists for each steam outlet pressure. Optimum sodium outlet temperature was defined as the temperature at which the sum of the steam generator heated surface area and 2/3 of the associated intermediate heat exchanger heated surface area reached a minimum. The 2/3 factor accounted for the estimated relative cost of the units. (See Section 3.1.5.2)
2. A number of tubes was chosen in the superheater and evaporator so that reasonable pressure drops were obtained (100 psi or less in the superheater and 50 psi or less in the evaporator).
3. Three steam generator outlet pressures were considered:

Turbine Pressure	Steam Generator Outlet Pressure
1400 psia	1500 psia
1800 psia	1900 psia
2400 psia	2500 psia

4. The following parameters were held constant for the three cases:

1.	Total heat transferred	- 110 MWt
2.	Sodium inlet temperature	- 960^{o}F
3.	Steam outlet temperature	- 900^{o}F
4.	Feedwater temperature	- 460^{o}F
5.	Tube flow geometry	
	High pressure tube OD	- 1.5 in.
	Heat transfer annulus, evaporator	- .1875 in.
	Heat transfer annulus, superheater	- .1000 in.
	Insulation gap (evaporator & superheater)	- .06 in.

5. The following parameters were varied as dictated by operating requirements:

   1. Tube wall thickness - established by design pressure and temperature.

   2. Fluid Properties.

The results of the sizing study are summarized in table 2.5-1

TABLE 2.5-1
SUMMARY OF PRESSURE EFFECTS

Steam Outlet Pressure (psia)	2500	1900	1500
Superheater			
No. of tubes	150	134	134
Length - ft.	30.78	23.70	20.00
Heated area - ft^2	1813.00	1247.00	1052.00
P - psid	99.70	101.00	97.50
Evaporator			
No. of tubes	380	250	250
Length - ft.	32.45	30.90	23.50
Heated area - ft^2	4842.00	3034.00	2307.00
P - psid	21.00	26.30	17.00
CR (required)	2.82	3.22	2.94
CR (actual)	3.50	3.50	3.50
T_{NA} OUT - ^oF	673.00	650.00	635.00

As the table indicates, the total surface area required in the steam generator increases with water-side pressure; this is also true of the IHX. This increase is primarily due to two mechanisms. First, is the decrease in available log mean temperature difference (LMTD) as the saturation temperature in the evaporator approaches closer to the sodium temperature. Second, is an increase in tube wall thickness required to contain the higher pressure. The incremental size increase (slope of the curve in Figure 1) increases with pressure.

Within the range of this study, there is no reason in terms of practicality, with respect to sizing, to choose one outlet pressure over another. Ultimately, the decision is an economic one, weighing capital cost versus plant performance over the expected lifetime.

3.0 Performance

In addition to effects on steam generator size, the effect of design pressure on performance must be considered. Three areas which were evaluated in this study were stability, sodium-water reaction effects and reliability.

3.1 Stability

A meaningful quantitative evaluation of the effects of design steam pressure on stability would involve an extensive effort and probably experimental verification.

The following references, however, can lend some qualitative insight into the problem:

-Nanavandi, A. N., and Von Hollen, R. F., "Flow Stability in Large Vertical Steam Generators ASME 64-WA/AUT-11

-Boure', J. A. Bergles, A. E., and Tong, L. S., "Review of Two Phase Flow Instability" ASME 71 HT-42

Both of these references indicate that, in a boiling system, increasing the pressure tends to make the system more stable.

This is not to say, however, that either a 1400 psia or 2400 psia design would be stable or unstable, only that the trend is favorable toward higher pressures.

3.2 Effect of System Pressure on Sodium-Water Reactions

The magnitude of reaction loading imposed on the steam generator shell is affected by the rate of blowdown of water into sodium. The investigation of blowdown rate for severed tube requires four discrete methods of analysis. These are:

Subcooled pressurized liquid blowdown through short tubes
Subcooled pressurized liquid blowdown through long tubes
Saturated steam/water blowdown
Superheated steam blowdown

The following is a discussion of these methods, and comparisons of blowdown rates.

3.2.1 Subcooled Pressurized Liquid Blowdown Through Short Tubes Using Zaloudek's Model

Zaloudek's [1] data demonstrates that compressed subcooled water blowdown for $0 - \frac{L}{D} - 20$ can be represented by:

$$G = C\sqrt{\frac{2g}{V_f}\,(P_{up} - P_{sat})}\ \text{lbs/ft}^2 - \text{sec}$$

G	=	mass velocity during upstream choking,	lbs/ft^2 - sec
P_{up}	=	upstream pressure	lbs/ft^2
P_{sat}	=	saturation pressure	lbs/ft^2
V_f	=	specific volume of saturated water	$\text{ft}^3\text{/lb}$
C	=	0.95, contraction coefficient	dimensionless

The data were correlated by equation (1) with a maximum deviation of 10%.

Since a tube rupture at the tubesheet bottom would have an $\frac{L}{D}$ greater than 20 (L= 12.5" for tubesheet thickness, bayonet tube I.D. = .501"), equation (1) (which assumes entrance acceleration only with no momentum loss due to flashing) cannot be applied. It is clear that momentum losses must be accounted for in the computation of critical mass flux, where flashing does occur. Where longer tubes are ruptured, the initially supersaturated liquid core has more time to break up. Extremely long tubes (such as a bottom tip rupture with an L/D 800) allow thermodynamic equilibrium to be approached, where bubbly two-phase flow breaks down into a nearly homogeneous mixture. The pressure drop is then primarily caused by momentum losses due to changes in specific volume and wall shear flow retardation.

---

(1) F. R. Zaloudek, "Steam-Water Critical Flow from High Pressure Systems," Hanford Atomic Products Operation, HW-80535, January 1964

TABLE 3.2-1

A COMPARISON OF BLOWDOWN RATE VS. PRESSURE

	Case 1	Case 2	Case 3
P_{up}, psia	1400	2000	2600
Subcool, ^oF	100	100	100
Temp., ^oF	487	536	574
P_{sat}, psia	606	931	1270
V_f, ft^3/lb	.0201	.0213	.0226
G, lb/ft^2-sec	18,175	20,487	22,184

Since V_f is essentially constant, blowdown rate increased by 22% for the 2600 psia system over the 1400 psia system.

3.2.2 Subcooled Pressurized Blowdown Through Long Tubes

R. E. Henry [(2)] developed a non-equilibrium model to describe the two phase critical discharge of initially saturated and subcooled liquid through long tubes (L/D ≥ 12). His solutions are based on the upstream and stagnation fluid conditions. Henry shows that

$$G_c^2 = (Pc - Pt) \Big/ \left[\frac{V}{2gc^2} + \frac{X_t}{g} \left(V_{gt} - V_{\ell o} \right) \right] \qquad (2)$$

where
- v = specific volume, lb/ft^3
- P = pressure, lb/ft^2
- g = gravitational constant, ft/sec^2
- x = quality

(2) R. E. Henry, "The Two-Phase Critical Discharge of Initially Saturated or Subcooled Liquid," Nuclear Science & Engrg., Vol. 41, No. 3., Sept. 1970

where G_c = critical mass flux, lb/ft^2 - sec

C = contraction coefficient = .95

and the subscript 0 refers to initial stagnation conditions

t refers to throat (exit plane) conditions

ℓ refers to saturated liquid properties

g refers to saturated vapor properties

For an analytical solution, Henry's equation requires an estimate of the exit plane quality and critical pressure ratios. His results are plotted as critical mass flux vs. L/D ratio for saturated water and are attached as Firgure 3.2-1, with initial stagnation pressure varying from 28.4 to 2600 psia (dotted lines indicate cross-plotting and/or extrapolation from other graphs in reference 2). Figure 3.2-2 is a plot of critical mass flux vs. initial stagnation pressure for various L/D ratios. These curves can be used to obtain maximum steady state blowdown rates through long tubes assuming the liquid to be saturated; however, where a considerable degree of subcooling may exist, solution of equation (2) is essential, or non-conservative results may be obtained. Subcooling causes a lower outlet quality and, therefore, less momentum pressure drop resulting in a higher blowdown rate. Firgue 3.2-3 (critical pressure ratio vs. L/D) from reference (2) is attached to use as an estimate for P_T for obtaining throat quality.

TABLE 3.2-2

A COMPARISON OF BLOWDOWN RATE VS. PRESSURE

	Case 1	Case 2	Case 3
P_0, psia	1400	2000	2600
Subcool, ^oF	100	100	100
P_{sat}, psia	606	931	1270
v	.0201	.0213	.0226
h_0	473.	532.	581.
n, critical pressure ratio	.66	.63	.60
x_t	.06	.09	.11
P_t	400	586	762
v_{gt}	1.16	.78	.60
G_c	7632	9044	10,580

The above calculations show a blowdown rate increase of 39% where the system pressure was increased from 1400 psia to 2600 psia, keeping 100° subcooled water.

3.2.3 Saturated Steam/Water Mixture Blowdown Using Moody's Model

F. J. Moody[3] developed a model for maximum blowdown of a two-phase mixture through pipes in terms of upstream stagnation properties and pipe geometry. The exact mechanism of the model need not be described here, since Moody's grapical output allows the user to obtain an immediate solution.

Graphs are given for values of FL/D = 0, 1, 2,3,4,5,10, 20, 50 and 100.

TABLE 3.2-3

A COMPARISON OF BLOWDOWN RATE VS. PRESSURE

	Case 1			Case 2			Case 3		
P_o, psia	1400			2000			2600		
quality	.30			.30			.30		
f, friction factor	.01			.01			.01		
Pipe L/D	100	300	500	100	300	500	100	300	500
G_c lb/ft^2 -sec	5200	3700	2900	7000	5100	4100	8700	6800	5600

The above calculations show blowdown rate increases of 67% for L/D = 100, 78% for L/D = 300, and 93% for L/D = 500 where the steam pressure was increased from 1400 psia to 2600 psia with constant quality.

3.2.4 Superheated Steam Blowdown Using Lapple's Model

C.E. Lapple[4] presents graphs of $P_{ambient} / P_o$ vs. G/G_{crit} with fL/D as

(3) F. J. Moody, "Maximum Two-Phase Vessel Blowdown from Pipes, "General Electric Co., APED 4827, April 1965.

(4) C. E. Lapple, Fluid and Particle Dynamics, Univ. of Del., March 1951, pp. 61

the independent parameter. With this method, superheated steam can be treated as an ideal gas. Lapple presents graphs for values of k (ratio of specific heats, C_p/C_v of 1.0, 1.4 and 1.8. For steam, the slightly conservative value of k = 1.4 is used.

TABLE 3.2-4

A COMPARISON OF BLOWDOWN RATE VS. PRESSURE

	Case 1			Case 2			Case 3		
P_o, psia	1400			2000			2600		
Superheat, ^oF	300			300			300		
T_o, ^oF	887			936			974		
h_o, btu/lb	1430			1433			1438		
G_{cni}, lb/ft^2 - sec	2512			3494			4453		
Pipe L/D	100	300	500	100	300	500	100	300	500
G/G_{cni}	.6	.42	.33	.6	.42	.33	.6	.42	.33
G	1507	1055	829	2096	1467	1153	2672	1870	1469

where $G_{cni} = P_o \sqrt{\frac{g_c M}{eRT_o}}$, maximum mass velocity of fluid under frictionless, isothermal flow.

The above calculations show blowdown rate increases of 77% where the system pressure was increased from 1400 psia to 2600 psia keeping degree of superheat constant.

3.2.5 Conclusions

Since results with CEBUG have shown that the specific impulse on the shell ($\int$ Pdt) during a sodium/water reaction is proportional to (leak rate)$^{1/2}$, an increase in system pressure will raise the load on the shell during the reaction. The following results are for comparisons with generating tubes of the same diameter (if larger diameter tubes were to be used, the specific impulse will increase in proportion to the diametric increase):

TABLE 3.2-5

INCREASE IN SHELL IMPULSE FOR:

Regime	L/D	1400 psi	2600 psi	2600 psi
Subcooled short tube	0-12	Base	6%	10.5%
Subcooled long tube	12-300	"	9%	18 %
Saturated mixture (30% quality)	100	"	15%	29 %
Saturated mixture (30% quality)	300	"	17%	33 %
Saturated mixture (30% quality)	500	"	19%	39 %
Superheated steam	all	"	18%	33 %

3.3 Reliability

With respect to the tubesheet, channel shell and channel cover, the reliability of the generator should not be affected by steam generator pressure. All of these are sized to the same standards for each pressure and the margins of safety should be similar.

In the case of the tubing, however, where relatively small defects could comprise a significant portion of the tube wall, the thicker wall sections at a design pressure of 2400 could possibly be considered an advantage.

4.0 Manufacturing Aspects

An evaluation was made in qualitative terms with respect to the effect of steam generator design pressure on the manufacturing aspects of producing a steam generator. These are discussed in the following sections.

4.1 Manufacturing Costs

In addition to the effects on generator size which were discussed above, there are several factors which would tend to slightly increase the cost of a higher pressure unit when compared to a lower pressure design.

The materials required for the tubesheet, tubing, and channel shell and cover would be thicker thus adding to material costs.

Additional welding would be required on the thicker sections.

Cost of Non-Destructive Testing, particularly radiography would be somewhat increased for the thicker sections.

There would be little if any effect on the other aspects of manufacture such as forming, fitup, bundling, etc.

It is our best judgement that the net effects of these items on the overall manufacturing cost would be insignificant, especially when compared to other effects such as variation of number of tubes and/or heated length as previously reported.

4.2 Welding and NDT

No particular effect on welding and NDT is anticipated except as mentioned above with respect to costs.

5.0 Effect of Steam Pressure on Water/Steam Recirculating Equipment

5.1 Recirculating Pumps and Seals

Recirculating pumps over the range of pressures have been sucessfully incorporated with C-E fossil boiler applications. As such, C-E considers pumps and seals to be within the state-of-the-art.

5.3 Steam Drums - Effect on Costs and Size

Using the C-E demonstration plant steam generator steam drum as a reference (2400 psig system), the following can be noted with respect to a lower pressure steam drum:

a) For effective steam separation, a longer steam drum would be required due to the higher specific volume (approximately 25% longer maintaining the same drum diameter when comparing the 2400 and 1400 psia systems). Also, thinner plate would be used on the drum for the lower pressure systems.

b) Costs for the 1450 psia steam drum when compared to the 2500 psia would be approximately 10% lower. This is essentially due to the lower costs of material due to the thinner plate. The labor and welding costs, etc. would not be appreciably changed.

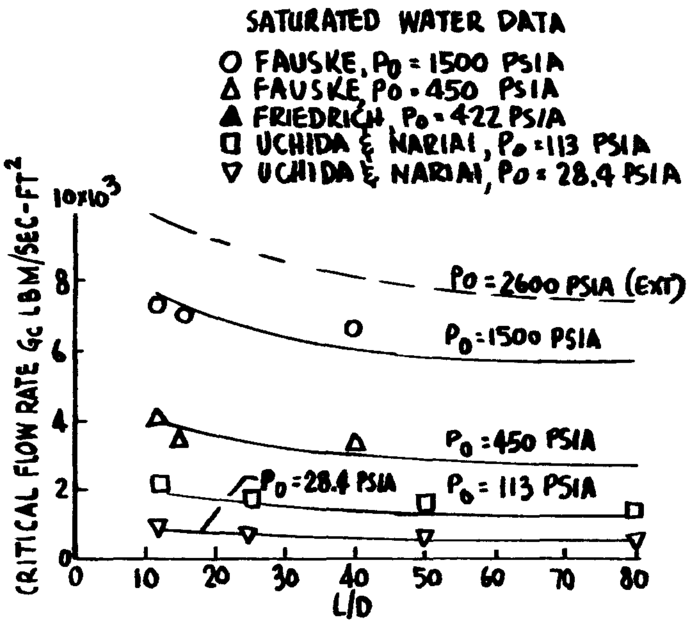


FIGURE 3.2-1

CRITICAL FLOW RATE G_C, Vs L/D

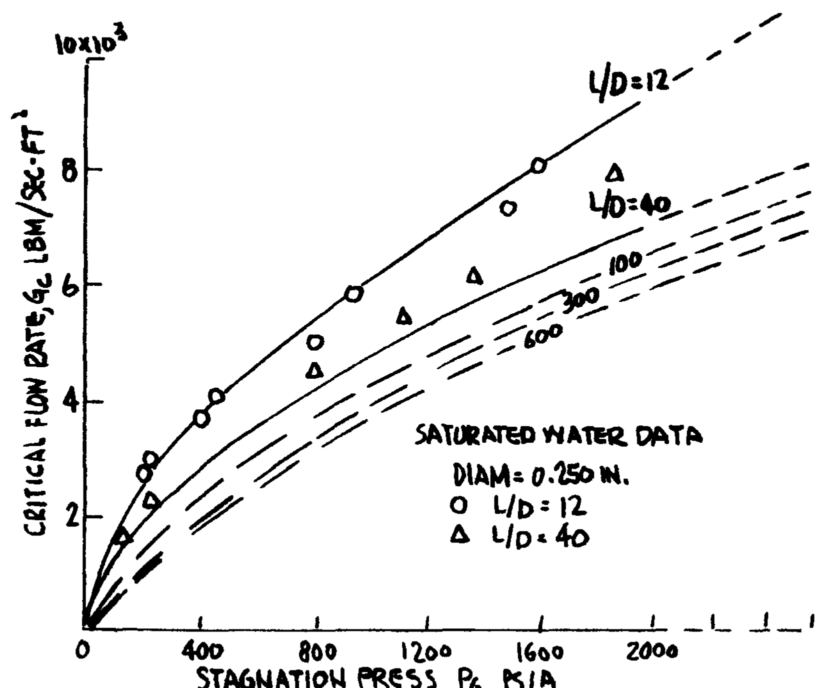


FIGURE 3.2-2

CRITICAL FLOW RATE, G_c, Vs STAGNATION PRESSURE, P_o.

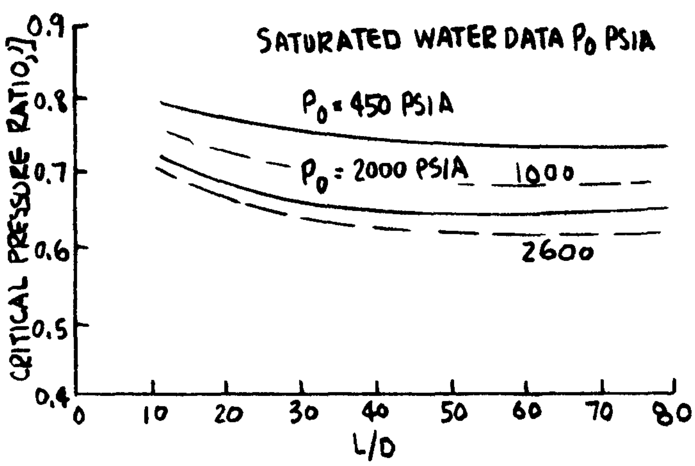


FIGURE 3.2-3

CRITICAL PRESSURE RATIO η VS L/D

APPENDIX E

SHELL SIDE DISTRIBUTION AND PRESSURE LOSS CALCULATIONS

E.1 Axial ΔP Calculations for LMFBR Superheater Tube Bundle

E.2 Crossflow ΔP Calculations for the LMFBR Superheater

E.3 Axial ΔP Calculations for LMFBR Evaporator Tube Bundle

E.4 Crossflow ΔP Calculations for the LMFBR Evaporator

TITLE: Axial Δ P Calculations for LMFBR Superheater Tube Bundle

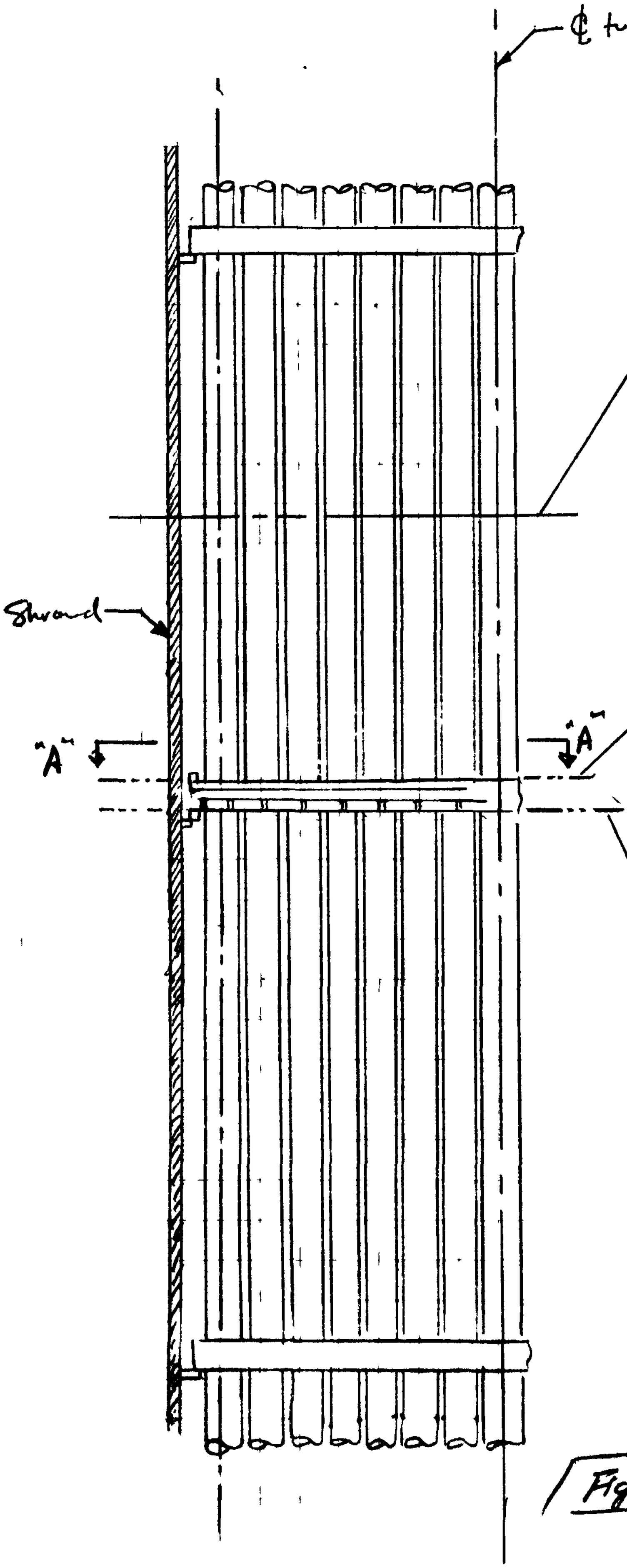
OBJECTIVE: To determine the relative magnitude of pressure drops expected within the tube bundle of the LMFBR Steam Generator (below the crossflow region) in order to assess whether or not flow maldistribution problems exist in this area.

ASSUMPTIONS: Assuming that the effects of the wall (shroud inside diameter) will be felt to a distance of 2 tube rows into the tube bundle.

CONCULSIONS: This calculation indicates negligible pressure drop differentials between the peripherial region of the tube bundle and the central region; therefore, it is concluded that no maldistribution problems will exist in the tube bundle.

Prepared by: D. D. DeFur 1/25/73

Checked by: S. R. Penfield 1/29/73



Plane "A" (between tube supports)

A_{A1} = Total Free Flow Area in zone #1 = 307.726 in^2

A_{A2} = Total Free Flow Area in zone #2 = 153.548 in^2

(for Area calculations, See page 4)

Plane "B" (at top support bars)

A_{B1} = Total Free Flow Area in zone #1 = 207.626 in^2

A_{B2} = Total Free Flow Area in zone #2 = 114.964 in^2

(for Area calculations, See page 4)

Plane "C" (at bottom support bars)

A_{C1} = Total Free Flow Area in zone #1 = 129.261 in^2

A_{C2} = Total Free Flow Area in zone #2 = A_{B2} = 114.964 in^2

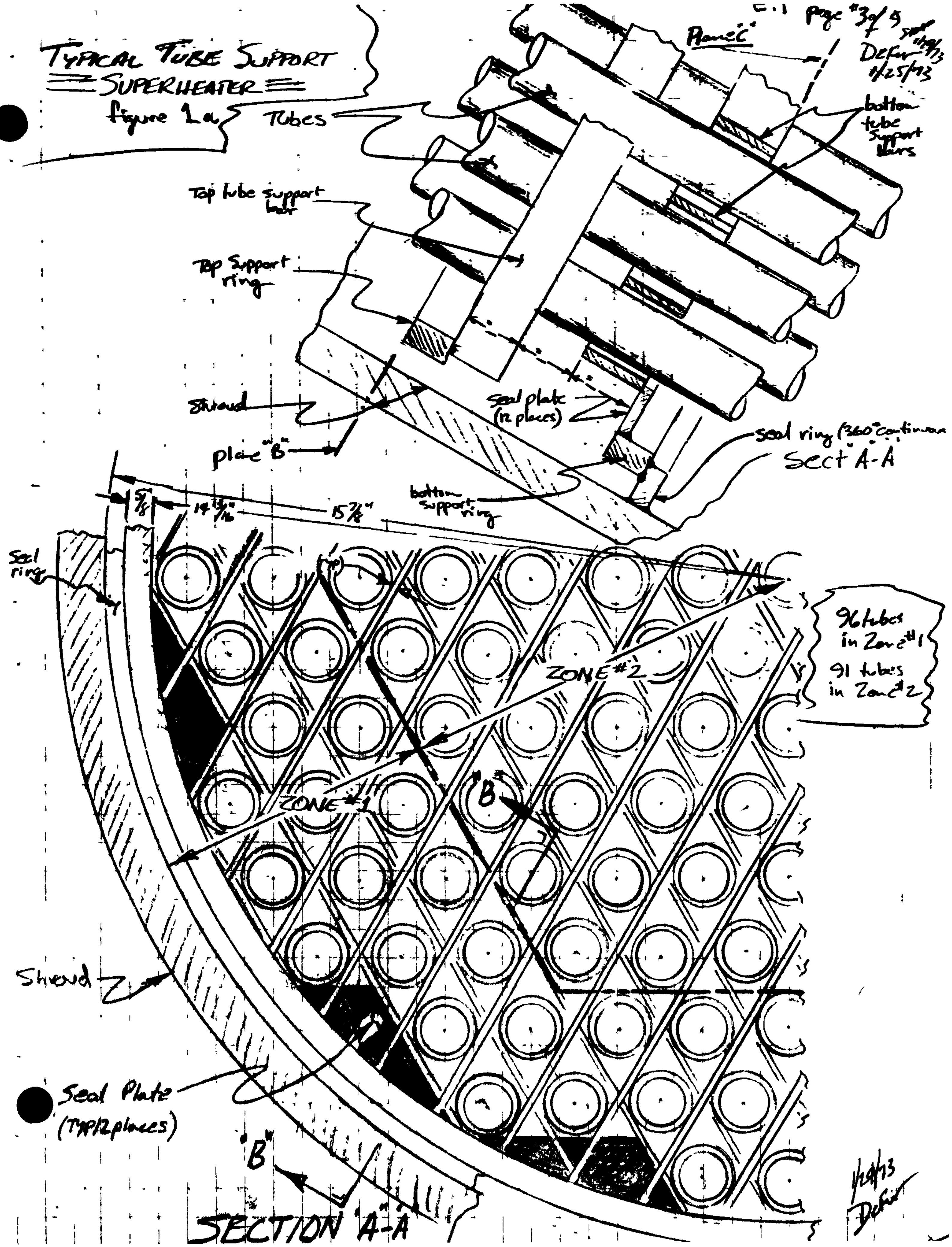
(for Area Calculations, See page 4)

Figure 1

E.1 page #3 of 9

Defur 1/25/93

TYPICAL TUBE SUPPORT
SUPERHEATER
figure 1a



Geometry:

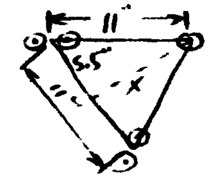
R_{et} = radius of extreme tube in bundle = 7.0002 P = 7.0002(2.000) = 14.0004 in

R_{ir} = inner radius of support rings = $R_{et} + \frac{\text{tube dia}}{2} + .0625$ = 14.8129 ≈ 14 13/16 in

R_{or} = outer radius of support rings = R_{ir} + .625 = 15 7/16 in

R_{is} = inner radius of shroud = R_{or} + .4375 = 15 7/8 in

Areas:



$x = \sqrt{11^2 - 5.5^2}$
= 9.526

$A_\Delta = \frac{1}{2}(11 \times 9.526)$
= 52.393

A_{si} = Total Area inside shroud = $\pi(15.875)^2$ = 791.730 in²

A_{cr} = Total Area inside zone #2 = $6(A_\Delta)$ = 6(52.393)
= 314.358 in²

A_{pr} = Total Area inside zone #1 = $A_{si} - A_{cr}$ = 477.372 in²

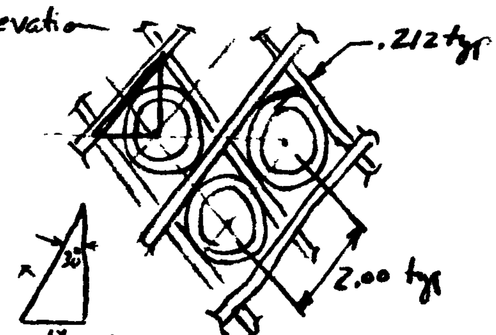
A_{t1} = Total Area of tubes in zone #1 = $96(\pi/4)(1.50)^2$ = 169.646 in²

A_{t2} = Total Area of tubes in zone #2 = $91(\pi/4)(1.500)^2$ = 160.810 in²

A_{A1} = Total Free Flow Area in zone #1 at plane "A" = $A_{PT} - A_{t1}$ = 307.726 in²

A_{A2} = Total Free Flow Area in zone #2 at plane "A" = $A_{cr} - A_{t2}$ = 153.548 in²

A_b = Area of tube support bars/tube/elevation
= 2(2.000)(2.12/2)
= .424 in²



A_{b2} = total area of bars in zone #2
= No. tubes × A_b = (91)(.424) = 38.584 in²

A_{b1} = total area of bars in zone #1
= 96(.424) = 40.704 in²

$1/x = \sin 30° = .5000$
$x = \frac{1}{.5000} = 2.00$

A_r = area of top & bottom support rings = $\pi(R_{or}^2 - R_{ir}^2) = \pi(15.4375^2 - 14.8125^2)$
= 59.396

A_{SR} = Area of seal ring = $\pi(R_{is}^2 - R_{or}^2) = \pi(15.875^2 - 15.4375^2)$ = 40.037 in²

A_{sp} = Area of seal plates = $12(2.994)^*$ = 35.328 in²

A_{B1} = Total Free Flow Area in zone #1 At plane "B" = $A_{A1} - A_{b2} - A_r$ = 207.626 in²

A_{B2} = Total Free Flow Area in zone #2 at plane "B" = $A_{A2} - A_{b2}$ = 114.964 in²

A_{C1} = Total Free Flow Area in zone #1 at plane "C" = $A_{B1} - A_{SR} - A_{SP}$ = 129.261

$A_{C2} = A_{B2}$ = 114.964 in²

* Area determined from scaled drawing.

D_{A1} = hydraulic Diameter for zone #1 at plane "A" $= \frac{4A_{A1}}{WP} = \frac{4(307.726)}{26(\pi(1.5))+2\pi(15.875)}$
$= 2.229 \text{ in}$

D_{A2} = hydraulic Diameter for zone #2 at plane "A" $= \frac{4A_{A2}}{WP} = \frac{4(153.548)}{91(\pi(1.5))}$
$= 1.432 \text{ in}$

N_{Re2} = Reynold's number for zone #2 $= \frac{\rho V_2 D_{A2}}{\mu} = \frac{\rho\left(\frac{W_2}{\rho A_{A2}}\right) D_{A2}}{\mu} = \frac{W_2 D_{A2}}{A_{A2}\mu}$

μ = viscosity $= \frac{.60 \#}{hr\text{-}ft}\left(\frac{ft}{12 in}\right) = .050 \#/hr\text{-}in$

$$N_{Re2} = \frac{(1.432)(3600)}{(.050)(153.548)} = 671.477\, W_2$$

$$\frac{\epsilon}{D_{A2}} = \frac{(.000005)(12)}{1.432} \simeq .00004$$

for a range of values of W_2, enter Moody's diagram with ϵ/D_{A2} and N_{Re2} and determine values of friction factor (f).

$$\Delta P_{friction\,2} = \frac{fL}{D_{A2}}\frac{\rho V_{A2}^2}{2g} = \frac{fL}{D_{A2}}\frac{\rho\left(\frac{W_2^2}{\rho^2 A_{A2}^2}\right)}{2g} = \frac{fL}{D_{A2}}\frac{W_2^2}{\rho A_{A2}^2\, 2g}$$

where $L = 32.6 \text{ ft} = 391.2 \text{ in}$ & $\rho = 52.3 \#/ft^3$

$$\Delta P_{friction\,2} = \frac{(391.2)(144)\, f\, W_2^2}{(1.432)(2)(32.2)(52.3)(153.548)^2} = 4.953 \times 10^{-4} f\, W_2^2$$

utilizing the values of "f" and the values of W_2, calculate $\Delta P_{friction\,2}$

W_2 (#/sec)	N_{Re2}	f	$\Delta P_{friction\,2}$ (#/in²)
300	2.01×10^5	.0158	.704
402.3	2.70×10^5	.0150	1.202
600	4.03×10^5	.0140	2.496
800	5.37×10^5	.0135	4.279
1000	6.71×10^5	.0130	6.439

$$\Delta P_{tube\ support\ 2} = N K_T \frac{\rho V^2}{2g} = N K_T \frac{w^2 \rho}{\rho^2 A^2 2g}$$

where N = Number of tube supports = 10
and $K_T = 2(K_{contraction} + K_{expansion})$

$$K_{contraction} = .165^{*}$$

$$K_{expansion} = \left(1 - \frac{114.96^2}{153.598}\right)^2 = .063$$

$$K_T = 2(.165 + .063) = .456$$

$$\Delta P_{tube\ supports\ 2} = \frac{10(.456)(144)\ w_2^2}{52.3(114.969)^2 \times 2(32.2)} = .00001475\ w_2^2$$

Calculate $\Delta P_{tube\ supports\ 2}$ for the range of values of w_2 and Add $\Delta P_{friction\ 2}$ to determine values of $\Delta P_{total\ 2}$

w_2 (#/sec)	$\Delta P_{tube\ supports\ 2}$ (#/in²)	$\Delta P_{total\ 2}$ (#/in²)
300	1.328	2.032
402.3	3.387	3.589
600	5.310	7.806
800	9.440	13.719
1000	14.750	21.189

N_{Re_1} = Reynolds number for zone #1 $= \frac{w_1 D_{A1}}{A_{A1}\ \mu} = \frac{(2.229)(3600)}{(.050)(307.726)} w_1$

$= 521.529\ w_1$

$$\epsilon/D_{A1} = \frac{(.000005)(12)}{2.229} \approx .00003$$

* ref fig 1A-39 SAE AEROSPACE APPLIED THERMODYNAMICS MANUAL

$$\Delta P_{friction\,1} = \frac{fL}{D_{A1}}\,\frac{W_1^2}{\rho A_{A1}^2\, 2g} = \frac{(391.2)(144)\, f W_1^2}{(2.231)(2)(32.2)(52.3)(307.726)^2}$$

$$= .00007917\, f W_1^2$$

W_1 (#/sec)	N_{Re}	f	$\Delta P_{friction\,1}$ (#/in²)
300	1.56×10^5	.0164	.117
400	2.09×10^5	.0159	.201
600	3.13×10^5	.0148	.422
806.9	4.21×10^5	.0140	.722
1000	5.22×10^5	.0136	1.077

$$\Delta P_{tube\ supports\,1} = \frac{N W_1^2}{2g\rho}\left(\frac{K_{TOT\,B}}{A_{B1}^2} + \frac{K_{TOT\,C}}{A_{C1}^2}\right)$$

where $K_{TOT\,B}$ = Kcontraction at plane "B" + Kexpansion at plane "B"
and $K_{TOT\,C}$ = Kcontraction at plane "C" + Kexpansion at plane "C"

Kcontraction at plane "B" = .215* Based on $\frac{207.626}{307.726} = .675$
Kexpansion at plane "B" = $\left(1-\frac{207.626}{307.726}\right)^2 = .106$

Kcontraction at plane "C" = .370* Based on $\frac{129.261}{307.726} = .420$
Kexpansion at plane "C" = $\left(1-\frac{129.261}{307.726}\right)^2 = .336$

$K_{TOT\,B} = .215 + .106 = .321$
$K_{TOT\,C} = .370 + .336 = .706$

$$\Delta P_{tube\ supports\,1} = \frac{10(144)}{52.3(2)(32.2)}\left(\frac{.321}{207.626^2} + \frac{.706}{129.261^2}\right) W_1^2 = .00002\, W_1^2$$

* ref fig 1A-39 SAE AEROSPACE APPLIED THERMODYNAMIC MANUAL

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CHARGE NO.

DATE 1/25/73 BY DcFw

DESCRIPTION Axial ΔP-Superheater

CHECK DATE 1/29/73 BY [illegible]

W_1 (#/hr)	$\Delta P_{\text{tube supports 1}}$	$\Delta P_{\text{total 1}}$ (#/in)
300	1.800	1.917
400	3.200	3.401
600	7.200	7.622
806.9	13.022	13.744
1000	20.000	21.077

Plot $\Delta P_{TOT\,1}$ & $\Delta P_{TOT\,2}$ VS W_1 & W_2 respectively (see attached plot)

By inspection of the two plots, one can quickly deduce that for a given ΔP the difference in W_1 & W_2 is negligible.

For purposes of being able to read total ΔP for a given total flow rate, read values of W_1 & W_2 for several values of ΔP and prepare a plot of ΔP_{total} VS W_{TOT} where $W_{TOT} = W_1 + W_2$ (See attached plot)

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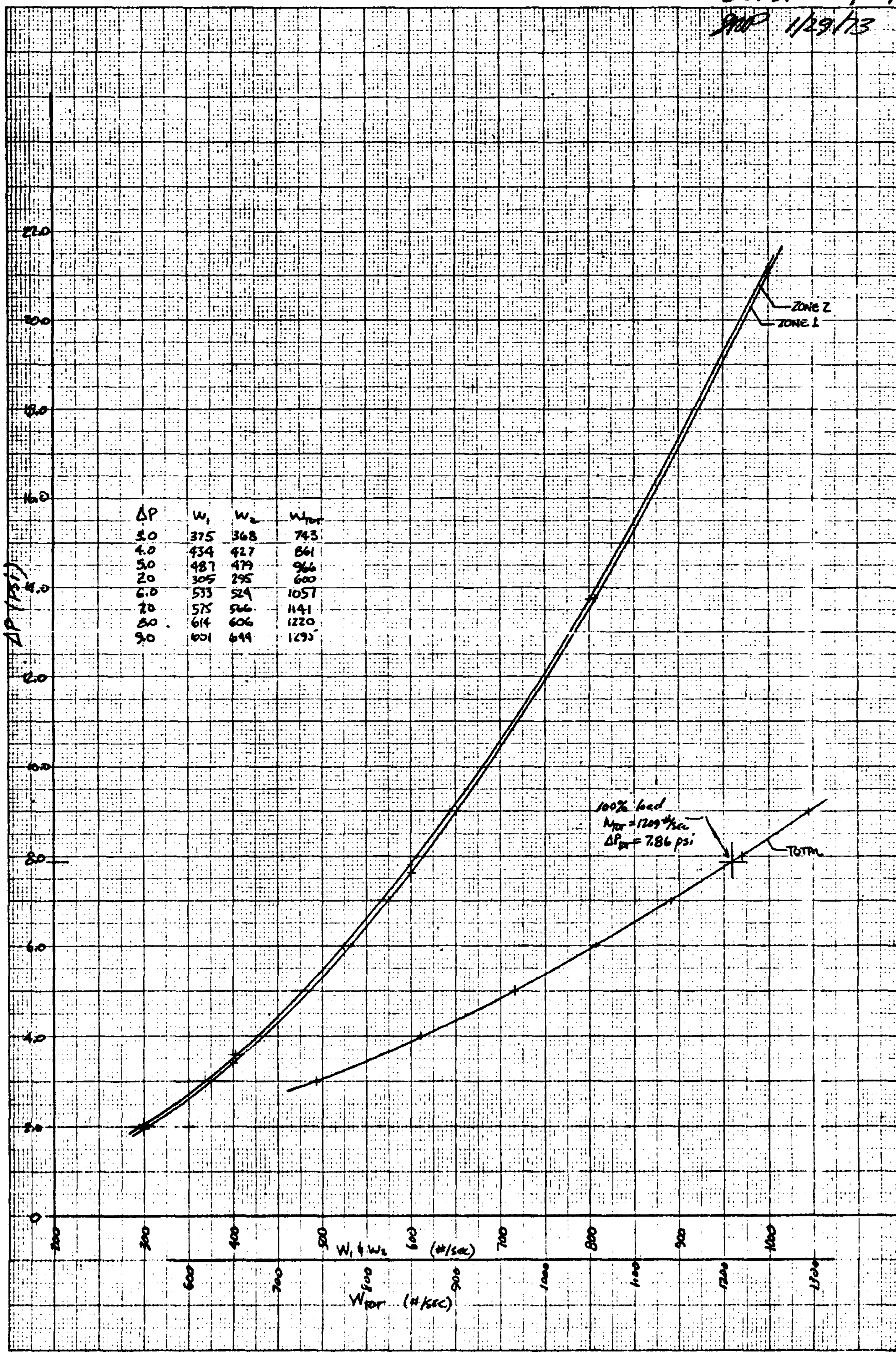
Defur 1/25/73
1/29/73



ΔP	W_1	W_2	W_{TOT}
3.0	375	368	743
4.0	434	427	861
5.0	487	479	966
2.0	305	295	600
6.0	533	524	1057
7.0	575	566	1141
8.0	614	606	1220
9.0	651	644	1295

TITLE: Crossflow Δ P Calculations for the LMFBR Superheater

OBJECTIVE: To determine the magnitude of pressure drops expected in the crossflow region of the LMFBR Steam Generator (the region where fluid flows from the openings in the flow baffle radially into the tube bundle) in order to assess whether or not flow maldistribution problems exist in that area.

CONCLUSION: This calculation indicates a pressure drop which is negligible compared with total pressure drop down through the tube bundle; therefore, it is concluded that no maldistribution problems will exist in this region.

Prepared by: D. D. DeFur 1/23/73

Checked by: S. R. Penfield 1/29/73

E.2

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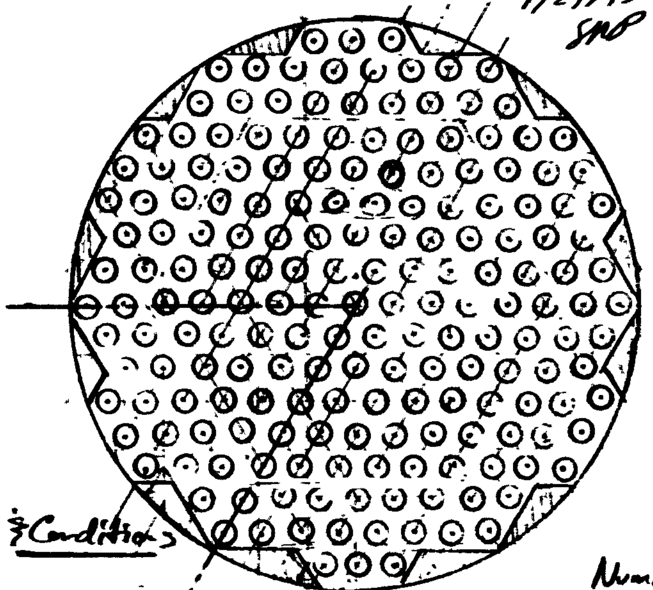
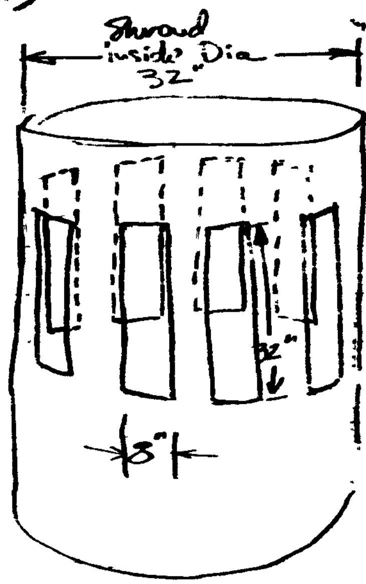
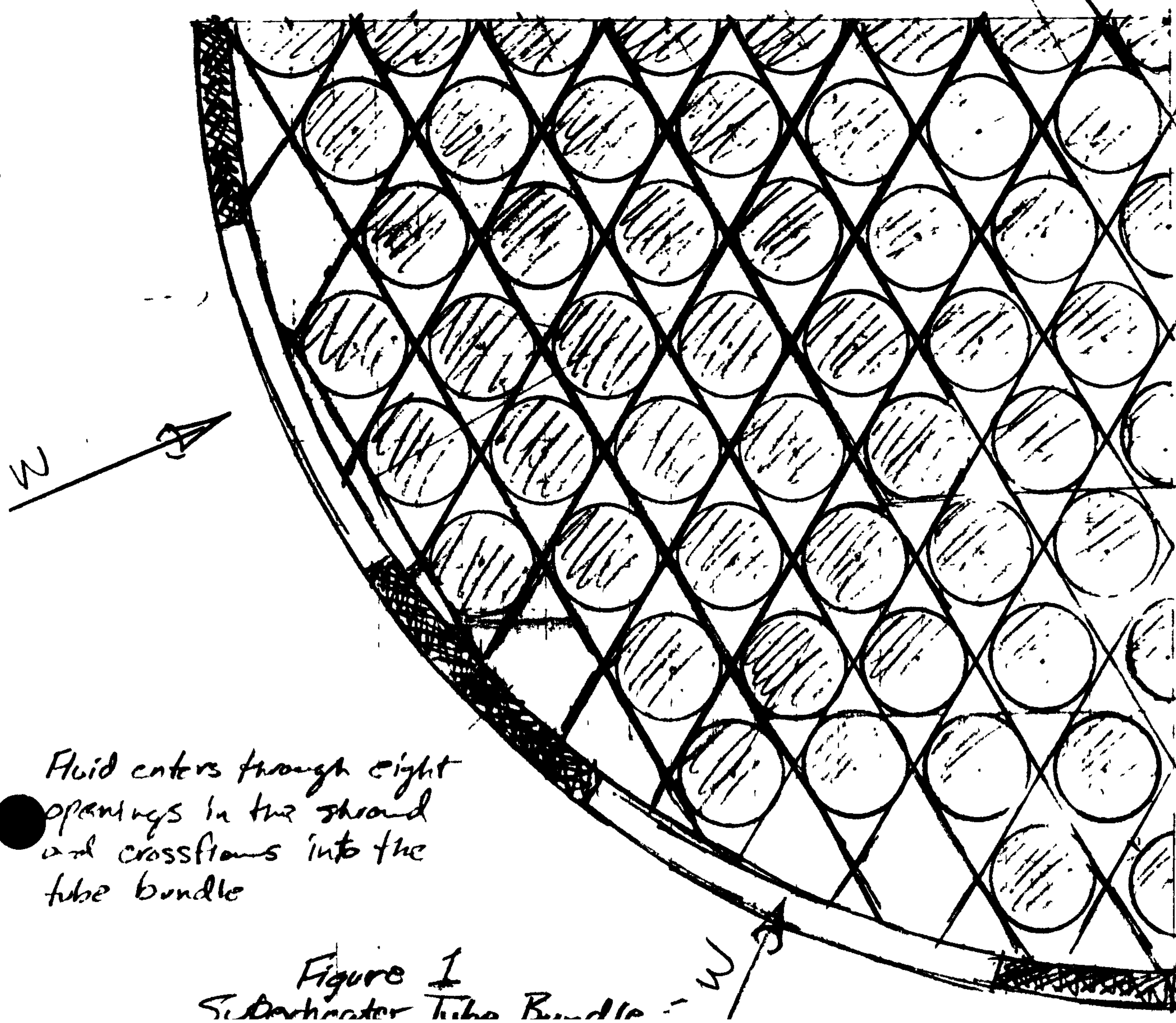




Fluid properties & Conditions
Sodium
Main Stream temp 875°F
Tube wall temp 860°F
Flow rate = 1209.2 #/sec

Number of tubes = 187
Tube diameter = 1.50 inches
Tube locations = 2.00 triangular pitch



Fluid enters through eight openings in the shroud and crossflows into the tube bundle

Figure 1
Superheater Tube Bundle

CROSSFLOW PRESSURE DROP – SUPERHEATER

This calculation is based on a procedure presented in a paper entitled "A General Correlation of Friction Factors for Various Types of Surfaces in Crossflow" by A.Y. Gunter and W.A. Shaw

The basic equation is stated as follows:

$$f/2 = \frac{\Delta P \, g_c \, D_v \, \rho}{G^2 L} \left(\frac{\mu}{\mu_w}\right)^{.14} \left(\frac{D_v}{S_T}\right)^{-.4} \left(\frac{S_L}{S_T}\right)^{-.6} \qquad EQ\#1$$

where $f/2 = .96 \left(N_{Re}\right)^{-.145}$
g_c = acceleration of gravity
D_v = volumetric hydraulic diameter
ρ = fluid density
G = fluid mass velocity based on minimum flow area
L = fluid flow length
μ = absolute viscosity at avg main stream temp
μ_w = absolute viscosity at surface wall temp
S_T = transverse tube to tube pitch
S_L = longitudinal tube to tube pitch
d_t = tube diameter

Since the direction of flow relative to the tube bundle is not known, an average of cases #I & #II (See figures 2a and 2b) will be used.

for case #1

$S_L = 2/12 = .1667$ ft $\qquad d_T = 1.50/12 = .1250$ ft

$S_T = 2/12 = .1667$ ft

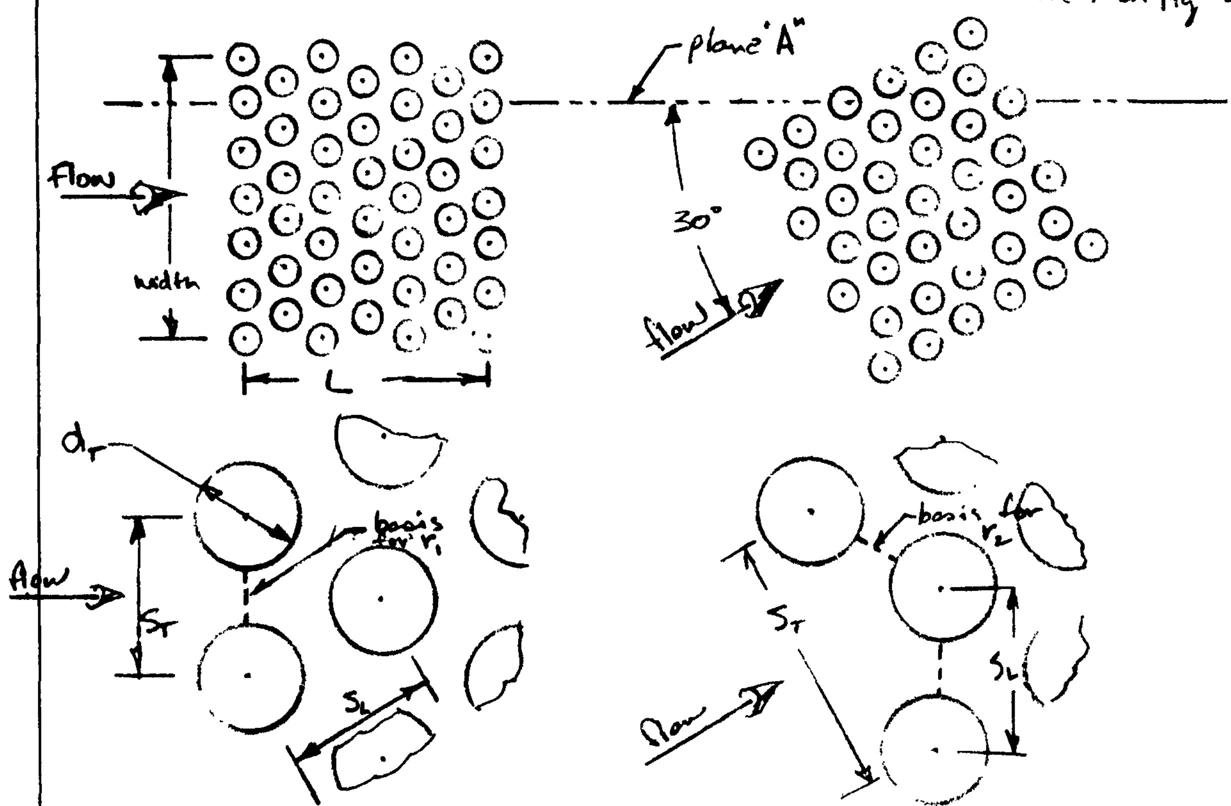
$r_1 = \frac{S_T - d_T}{S_T} = \frac{.1667 - .1250}{.1667} = .250$ = Area ratio based on Minimum area indicated by dashed line of Fig 2a.

for Case II

$S_L = 2/12 = .1667$

$S_T = 2(.86603)(2.0)/12 = .2887$

$r_2 = \frac{2(S_L - d_T)}{S_T} = \frac{2(.1667 - .125)}{.2887} = .2889$ = area ratio based on minimum area indicated by dashed lines on fig 2b.



CASE I
Flow Parallel to Plane "A"
Fig 2a

CASE II
Flow 30° oblique to Plane "A"
Fig 2b

$G = \frac{W}{rA}$ where r is r_1 & r_2 for cases I & II respectively

and $A = \pi D_A H$

where D_A = the diameter of a circle which divides the area being calculated into two equal area parts

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and H = the crossflow height

$$G^2 = \frac{w^2}{r^2 A^2}$$

as previously shown:

$$f/2 = .96 \left(N_{Re}\right)^{-.145}$$

where N_{Re} = Reynolds Number $= \dfrac{D_v G}{\mu}$

or

$$f/2 = .96 \left(\frac{D_v G}{\mu}\right)^{-.145} = .96 \left(\frac{D_v w}{r A \mu}\right)^{-.145}$$

substituting back into the basic equation (equation #1)

$$.96 \left(\frac{D_v w}{r A \mu}\right)^{-.145} = \frac{\Delta P\, g\, D_v\, \rho}{\frac{w^2}{r^2 A^2}\, L} \left(\frac{\mu}{\mu_w}\right)^{.14} \left(\frac{D_v}{S_T}\right)^{-.4} \left(\frac{S_L}{S_T}\right)^{-.6}$$

solving for ΔP

$$\Delta P = \frac{.96 \left(\frac{D_v w}{r A \mu}\right)^{-.145} \frac{w^2}{r^2 A^2}\, L \left(\frac{\mu_w}{\mu}\right)^{.14} \left(\frac{D_v}{S_T}\right)^{.4} \left(\frac{S_L}{S_T}\right)^{.6}}{\rho g D_v}$$

assuming $\left(\frac{\mu_w}{\mu}\right)^{.14} \approx 1$ and collecting terms, the above equation reduces to the following

$$\Delta P = \frac{.96}{\rho g} \left(\frac{1}{\pi H}\right)^{1.855} \mu^{.145} \underbrace{\left[\frac{S_L^{.6}}{r^{1.855} S_T}\right]}_{\text{use avg of case 1 \& 2 for value of [ ]}} \frac{L}{D_v^{.745}} \left(\frac{w}{D_A}\right)^{1.855} \qquad \text{EQ \#2}$$

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DESCRIPTION Crossflow ΔP - Superheater

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Determine avg [ ] value

for case #I

$$\frac{S_L^{.6}}{r_1^{1.855} S_T} = \frac{(.1667)^{.6}}{(.250)^{1.855}\,.1667}$$

$$\log .1667 = 9.22194 - 10$$
$$\times .6$$
$$5.533164 - 6$$
$$\bar{1}.5332$$
$$(.1667)^{.6} = .3414$$

$$\log .250 = 999.39794 - 1000$$
$$\times 1.855$$
$$1853.8832 \quad -1855$$
$$\bar{2}.8832$$
$$(.250)^{1.855} = .07642$$

$$\frac{S_L^{.6}}{r_1^{1.855} S_T} = \frac{.3414}{(.07642)(.1667)} = 26.7991$$

for Case #II

$$\frac{S_L^{.6}}{r_2^{1.855} S_T} = \frac{(.1667)^{.6}}{(.2888)^{1.855}(.2887)}$$

$$\log .2888 \quad 999.46060 - 1000$$
$$\times 1.855$$
$$1853.99941 - 1855$$
$$\bar{2}.99941$$
$$(2.888)^{1.855} = .09987$$

$$\frac{S_L^{.6}}{r_2^{1.855} S_T} = \frac{.3414}{(.09987)(.2887)} = 11.8408$$

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DESCRIPTION Crossflow ΔP - Superheater

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* Avg [ ] value = $\frac{26.799 + 11.8408}{2}$ = 19.320

$\mu = .62/3600 \;\; .0001722$

* $\mu^{.145} = (.0001722)^{.145} = .2846$

$H = 32/12 = 2.6667$

$\frac{1}{\pi H} = \frac{1}{\pi(2.6667)} = .1194$

* $\left(\frac{1}{\pi H}\right)^{1.855} = .01940$

* $\rho = 52.5$

log .0001722 = 996.23603 - 1000

$\times .145$

144.45422 - 145

$\bar{1}.45422$

$(.0001722)^{.145} = .2846$

log .1194 = 999.07700 - 1000

1.855

1853.28783 - 1855

$\bar{2}.28783$

$(.1194)^{1.855} = .01940$

Incorporating * items above into EQ#2 yields

$$\Delta P = \frac{.96}{(52.6)(32.2)}(.01940)(.2846)\left[19.320\right] \frac{L}{D_2^{.745}}\left(\frac{W}{D_A}\right)^{1.855}$$

$$= \frac{(9.6)(1.940)(2.846)(1.932)}{(5.26)(3.22)} \times 10^{-5} \frac{L}{D_2^{.745}}\left(\frac{W}{D_A}\right)^{1.855}$$

$$\Delta P = 6.043 \times 10^{-5} \frac{L}{D_2^{.745}}\left(\frac{W}{D_A}\right)^{1.855} \qquad EQ\#3$$

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Assuming an average case where one half of the total flow exits the region through each of two equal areas divided by a cylindrical plane of diameter D_A; therefore, one half of the total flow passes through the cylindrical plane. For conservatism, the flow passing through the plane is assumed to traverse tubes for a distance equal to the tube bundle radius.

$* \; L = \text{Shroud Dia}/2 = 2.66/2 = 1.33 \text{ ft}$

$W_{TOT} = 1209.2 \text{ \#/sec} \qquad W = \frac{1209.2}{2} = 604.6 \text{ \#/sec}$

$$D_v = 4 \times \frac{\text{Net free Volume}}{\text{friction Surface}} = 4 \times \frac{\frac{32}{12}\left[\pi/4\,(2.66)^2 - 187\left(\pi/4\left(\frac{1.5}{12}\right)^2\right)\right]}{187\left(\pi\left(1.5/12\right)\right)\frac{32}{12}}$$

$$= 4\;\frac{5.557 - 2.295}{73.435} = 4(.0444)$$

$D_v = .1760$

$* \; D_v^{.745} = .2741$

$\log .1760 = 999.24551 - 1000$

$\times .745$

$744.43790 - 745$

$\bar{1}.43790$

$.1760^{.745} = .2741$

$D_A = 1.881$

Outer Area = inner area

outer Area $= \pi/4\,(D_{Shroud}^2 - D_A^2)$

inner Area $= \pi/4\, D_A^2$

$\pi/4\,(D_{Shroud}^2 - D_A^2) = \pi/4\, D_A^2$

$D_{Shroud}^2 = 2 D_A^2$

$D_A = \sqrt{D_{Shroud}^2/2} = \sqrt{\frac{2.66^2}{2}} = 1.881$

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CHARGE NO. ____

DATE 1/23/73 BY DeFur

DESCRIPTION Crossflow ΔP – Superheater

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$$\frac{W}{D_A} = \frac{604.6}{1.881} = 321.424$$

$$* \left(\frac{W}{D_A}\right)^{1.855} = 4.473 \times 10^{4}$$

$$\log 321.424 = 2.50707$$

$$\times\ 1.855$$

$$4.65061$$

$$321.424^{1.855} = 44730.$$

incorporate * in eq # 3

$$\Delta P = 6.043 \times 10^{-5} \; \frac{1.33}{.2741} \left(4.473 \times 10^{4}\right)$$

$$\Delta P = 13.116 \text{ lb/ft}^2 = .091 \text{ lb/in}^2$$

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TITLE: Axial ΔP Calculations for LMFBR Evaporator Tube Bundle

OBJECTIVE: To determine the relative magnitude of pressure drops expected within the tube bundle of the LMFBR Steam Generator (below the crossflow region) in order to assess whether or not flow maldistribution problems exist in this area.

ASSUMPTIONS: Assuming that the effects of the wall (shroud inside diameter) will be felt to a distance of 3 tube rows into the tube bundle.

CONCULSIONS: This calculation indicates sodium maldistribution between tubes near the shell wall and central tubes.

Prepared by: J. W. Aderholdt 2/7/73

Checked by: S. R. Penfield 2/7/73

AREAS - (REFER TO FIGURE 1)

TOTAL, INSIDE TUBE BUNDLE LINER

$$A = \frac{\pi}{4}(46.125)^2 = 1671 \text{ IN.}^2$$

TOTAL, INTERIOR REGION

$$A = \text{TOTAL AREA/TUBE} \times \text{NO. OF TUBES}$$

$$A = (2 \times \cos 30°)\ 2 \times 271. = 938.74 \text{ IN.}^2$$

TOTAL, PERIPHERAL REGION

A = INSIDE TUBE BUNDLE LINER AREA − INTERIOR REGION AREA

$$A = 1671. - 938.74 = 732.26 \text{ IN.}^2$$

SODIUM FLOW AREA, INTERIOR REGION BETWEEN TUBE SUPPORTS.

A = INTERIOR REGION TOTAL AREA − TUBE AREA

$$A = 938.74 - \frac{\pi}{4}(1.5)^2(271) = 459.84 \text{ IN.}^2$$

SODIUM FLOW AREA, INTERIOR REGION AT UPPER AND LOWER TUBE SUPPORTS.

A = SODIUM FLOW AREA BETWEEN TUBE SUPPORTS − TUBE SUPPORT AREA

$$A = 459.84 - 2(.218)(271.) = 341.68 \text{ in.}^2$$

SODIUM FLOW AREA, PERIPHERAL REGION BETWEEN TUBE SUPPORTS.

A = PERIPHERAL REGION AREA − TUBE AREA

$$A = 732.26 - \frac{\pi}{4}(1.5)^2\, 150 = 467.19 \text{ in}^2$$

SODIUM FLOW AREA, PERIPHERAL REGION AT LOWER TUBE SUPPORT

A = SODIUM FLOW AREA BETWEEN TUBE SUPPORTS − SUPPORT RING AREA − TUBE SUPPORT AREA

$$A = 467.19 - \frac{\pi}{4}\left(46.125^2 - 44^2\right) - 2(.218)\,150$$

$$A = 251.37 \text{ IN.}^2$$

SODIUM FLOW AREA, PERIPHERAL REGION AT UPPER TUBE SUPPORT

A = SODIUM FLOW AREA, PERIPHERAL REGION AT LOWER TUBE SUPPORT − AREA OF SEAL PLATES

$$A = 251.37 - 44.2 = 207.2 \text{ IN.}^2$$

SODIUM FLOW AREA REDUCTION RATIO

PERIPHERAL, UPPER TUBE SUPPORT $= \frac{207.2}{467.19} = .44$

PERIPHERAL, LOWER TUBE SUPPORT $= \frac{251.37}{467.19} = .54$

INTERIOR, BOTH TUBE SUPPORTS $= \frac{341.68}{459.84} = .743$

HYDRAULIC DIAMETER

PERIPHERAL, BETWEEN TUBE SUPPORTS

$$D_h = \frac{4A}{P} = \frac{4(467.19)}{\pi(1.5)(150) + \pi(46.125)} = 2.19 \text{ IN.}$$

INTERIOR, BETWEN TUBE SUPPORTS

$$D_h = \frac{4A}{P} = \frac{4(459.84)}{\pi(1.5)(271)} = 1.44 \text{ IN.}$$

SODIUM FLOW @ 100% LOAD = 1209 #/SEC.

CALCULATE PRESSURE DROP THROUGH INTERIOR REGION FOR A SERIES OF SODIUM FLOWS

REYNOLDS NO. = $\frac{W\ D_h}{\mu\ A} = \frac{W\ (1.44)(3600)}{\frac{.72}{12}(459.8)} = W\,(187.9)$

$\frac{\varepsilon}{D_h} = \frac{.000005 \times 12}{1.44} = .000042$

FROM MOODY DIAGRAM,

W #/SEC	Re	f
400	75,160.	.019
600.7	112,871.	.018
800.	150,320.	.017

CALCULATE VELOCITIES

$V = \frac{W}{\rho A}$

V_1 = BETWEEN THE TUBE SUPPORTS
V_2 = AT THE TUBE SUPPORT

$V_1 = \frac{W\ (144)}{53.8\ (459.8)} = W\,(.00582)$

$V_2 = \frac{V_1}{.743}$

W #/SEC	V_1 ft/SEC.	V_2 ft/SEC.	
400	2.33	3.13	
600.7	3.49	4.70	
800.	4.65	6.26	

$$\Delta P_{friction} = \frac{fL}{D_h}\frac{\rho V^2}{2g} = \frac{f\,(32.5)}{\frac{1.44}{12}}\;\frac{53.8\;V_1^2}{64.4\,(144)}$$

$$= 1.5712\; f\; V_1^2$$

w #/SEC.	$\Delta P_{friction}$ PSI
400.	.162
600.7	.344
800.	.58

K LOSS FACTORS FOR TUBE SUPPORTS

ASSUME $K_{TUBE\ SUPPORT} = (K_{EXPANSION} \times 2) + (K_{CONTRACTION} \times 2)$

AREA RATIO = .743

FROM SAE AERO-SPACE APPLIED THERMODYNAMICS MANUAL PAGE A-46

$K_{EXPANSION} = .07 \times 2 = .14$
$K_{CONTRACTION} = .17 \times 2 = .34$

$K_{TUBE\ SUPPORT} = 0.48$

N = NO. OF TUBE SUPPORTS = 10

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$$\Delta P_{\text{TUBE SUPPORTS}} = N\,K\,\frac{\rho V_2^2}{2g} = \frac{10(.48)(53.8)\,V_2^2}{64.4(144)}$$

$$= .02785\,V_2^2$$

W #/SEC	$\Delta P_{\text{TUBE SUPPORTS}}$ PSI
400.	.273
600.7	.615
800.	1.091

$$\Delta P_{\text{TOTAL}} = \Delta P_{\text{FRICTION}} + \Delta P_{\text{TUBE SUPPORTS}}$$

W #/SEC	ΔP_{TOTAL}
400.	.435
600.7	.959
800.	1.671

PLOT IN FIGURE 2

CALCULATE PRESSURE DROP THROUGH PERIPHERAL REGION FOR A SERIES OF SODIUM FLOWS.

$$\text{REYNOLDS NO.} = \frac{W\,D_h}{\mu\,A} = \frac{W\,(2.19)\,(3600)}{\frac{.72}{12}\,(467.19)}$$

$$= 281.3\ W$$

$$\frac{\varepsilon}{D_h} = \frac{.000005 \times 12}{2.19} = .000027$$

FROM MOODY DIAGRAM

W #/sec	Re	f
400.	112,500.	.0175
608.3	171,100.	.0165
800.	225,000.	.0160

CALCULATE VELOCITIES

$$V = \frac{W}{\rho A}$$

V_1 = VELOCITY BETWEEN TUBE SUPPORTS
V_2 = " AT THE UPPER TUBE SUPPORT
V_3 = " " " LOWER " "

$$V_1 = \frac{W\,(144)}{53.8\,(467.19)} = W\,(.005729)$$

$$V_2 = \frac{V_1}{.44}$$

$$V_3 = \frac{V_1}{.54}$$

W #/SEC.	V_1 ft/SEC.	V_2 ft/SEC.	V_3 ft/SEC.
400.	2.29	5.19	4.24
608.3	3.49	7.9	6.5
800.	4.59	10.43	8.5

$$\Delta P_{friction} = \frac{f L}{D_h} \frac{\rho V_1^2}{2g} = f V_1^2 \frac{32.5 \quad (53.8)}{\frac{2.19}{12} \quad 64.4 (144)}$$

$$= 1.0331 \quad f V_1^2$$

W #/SEC.	$\Delta P_{friction}$ PSI
400	.095
608.3	.2076
800.	.348

K LOSS FACTORS FOR TUBE SUPPORTS, SUPPORT RING AND SEAL PLATES.

K_{LOSS} @ UPPER TUBE SUPPORT = $K_{EXPANSION}$ + $K_{CONTRACTION}$ @ .442 AREA RATIO.

K_{LOSS} @ LOWER TUBE SUPPORT = $K_{EXPANSION}$ + $K_{CONTRACTION}$ @ .537 AREA RATIO.

K_{UPPER} = .31 + .36 = .67

K_{LOWER} = .21 + .30 = .51

N = NO. OF SUPPORTS = 10

$$\Delta P_{\substack{TUBE\\SUPPORTS}} = N \; K_{UPPER} \frac{\rho V_2^2}{2g\,144} + N K_{LOWER} \frac{\rho V_3^2}{2g\,144}$$

$$= \frac{10\,(.67)(53.8)}{64.4(144)} V_2^2 + \frac{(10).51\,(53.8)}{64.4(144)} V_3^2$$

$$= .03887\; V_2^2 + .02959\; V_3^2$$

W #/SEC.	$\Delta P_{\substack{TUBE\\SUPPORTS}}$ PSI
400.	1.58
608.3	3.68
800.	6.37

$$\Delta P_{TOTAL} = \Delta P_{FRICTION} + \Delta P_{TUBE\ SUPPORTS}$$

W #/SEC	ΔP_{TOTAL} PSI
400.	1.68
608.3	3.89
800.	6.72

PLOT IN FIGURE 2

FROM FIG. 2 READ THE FLOW DISTRIBUTION BETWEEN PERIPHERAL AND INTERIOR TUBES.

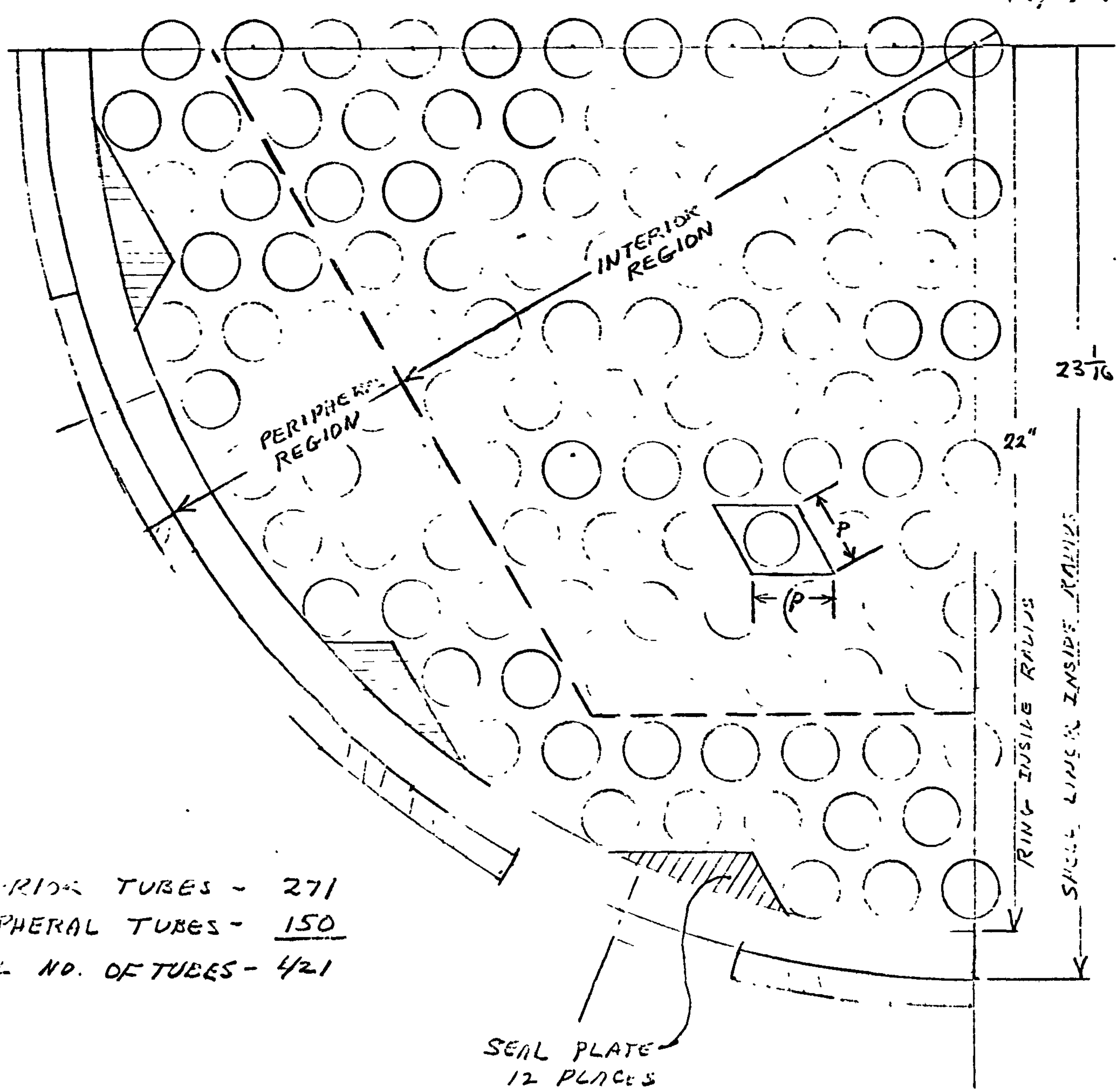
INTERIOR − $\dfrac{809 \text{ \#/SEC.}}{271 \text{ TUBES}} = 2.985$ #/SEC-TUBE

PERIPHERAL − $\dfrac{400 \text{ \#/SEC}}{150 \text{ TUBES}} = 2.667$ #/SEC-TUBE

TOTAL − $\dfrac{1209 \text{ \#/SEC}}{421 \text{ TUBES}} = 2.871$ #/SEC-TUBE

INTERIOR MALDISTRIBUTION $= \dfrac{(2.985 - 2.871) \times 100}{2.877} = +3.97\ \%$

PERIPHERAL MALDISTRIBUTION $= \dfrac{(2.667 - 2.871) \times 100}{2.871} = -7.1\ \%$

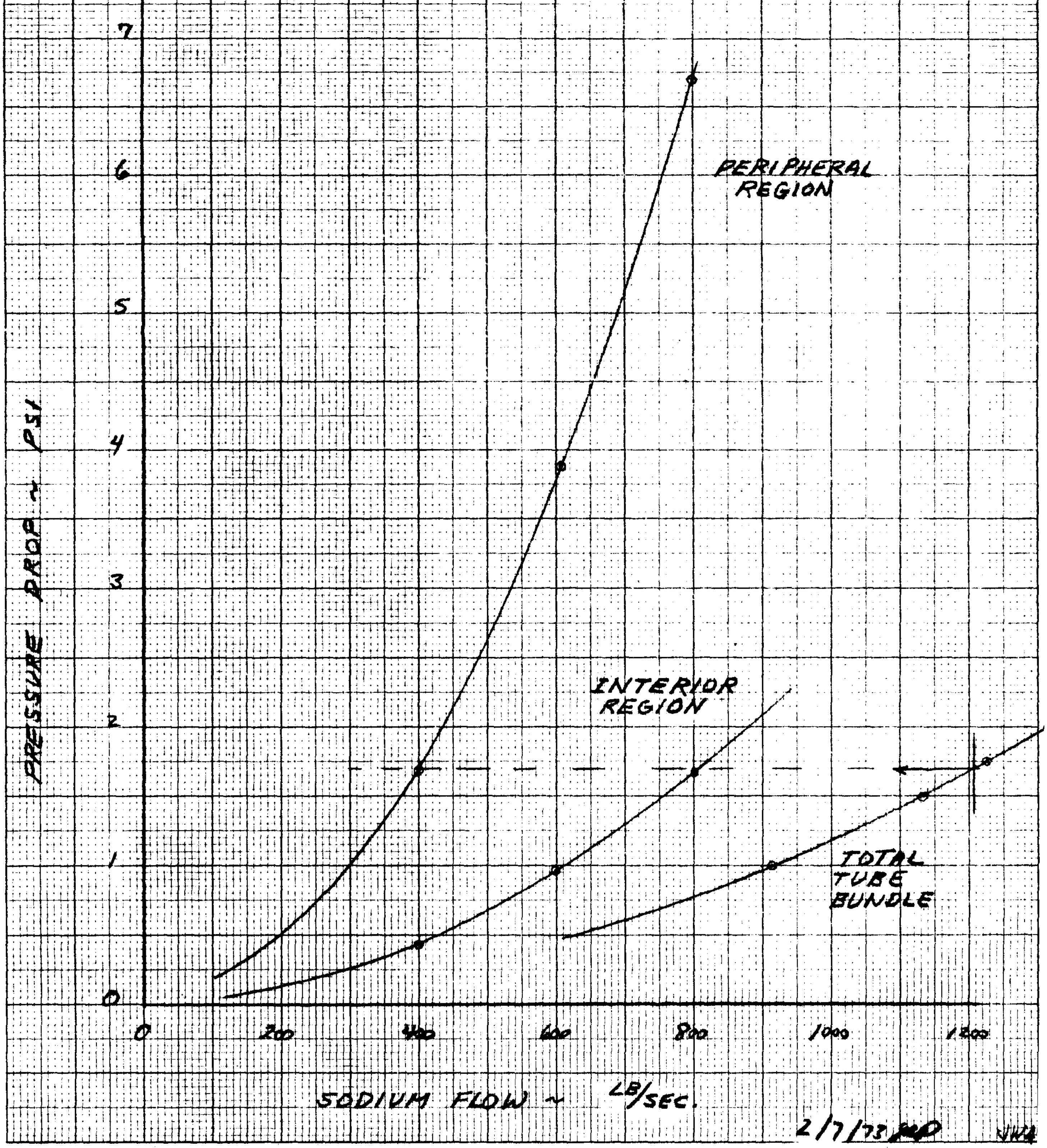


INTERIOR TUBES - 271
PERIPHERAL TUBES - 150
TOTAL NO. OF TUBES - 421

FIGURE 1
EVAPORATOR TUBE BUNDLE

FIGURE 2

SODIUM PRESSURE DROP



TITLE: Crossflow ΔP Calculations for the LMFBR Evaporator

OBJECTIVE: To determine the magnitude of pressure drops expected in the crossflow region of the LMFBR Steam Generator (the region where fluid flows from the openings in the flow baffle radially into the tube bundle) in order to access whether or not flow maldistribution problems exist in that area.

CONCLUSION: This calculation indicates a pressure drop which is negligible compared with total pressure drop down through the tube bundle; therefore, it is concluded that no maldistribution problems will exist in this region.

Prepared by:	J. W. Aderholdt	2/7/73
Checked by:	S. R. Penfield	2/7/73

CALCULATE CROSSFLOW ΔP IN THE EVAPORATOR IN THE REGION AT THE TOP OF THE TUBE BUNDLE.

FROM PREVIOUS CALCULATION FOR THE SUPERHEATER, THE FOLLOWING EQUATION FOR PRESSURE DROP WAS DEVELOPED

$$\Delta P = 6.043 \times 10^{-5} \frac{L}{D_N^{.745}} \left(\frac{W}{D_A} \right)^{1.855}$$

THIS EQUATION IS APPLICABLE WHERE:

1. THE TUBE PITCH IS TRIANGULAR ON 2" CENTERS AND TUBE DIAMETER IS 1.5".
2. THE HEIGHT OF THE FLOW WINDOWS IS 32".
3. SODIUM PROPERTIES ARE ASSUMED THE SAME.
4. W = SODIUM FLOW ~ LB/SEC.
5. D_A = DIAMETER THAT DIVIDES THE AREA INSIDE THE SHELL LINER INTO 2 EQUAL AREAS.

6. L = LENGTH OF FLOW PATH ACROSS THE TUBE BUNDLE

7. D_V = VOLUMETRIC HYDRAULIC DIAMETER

ASSUME "WORST CASE" CONDITIONS AS FOLLOWS:

1. ONE-HALF THE TOTAL SODIUM FLOWS ACROSS THE TUBE BUNDLE A DISTANCE FROM THE WINDOW TO THE CENTER OF THE TUBE BUNDLE THROUGH AN AVERAGE AREA BASED ON D_A

$$L = \frac{ID_{SHROUD}}{12 \times 2} = \frac{46.125}{12 \times 2} = 1.922 \text{ ft.}$$

$$D_A = \sqrt{\frac{4}{\pi}\left(\frac{A_{SHROUD\ ID}}{2}\right)} = \sqrt{\frac{4}{\pi}\left(\frac{\pi\,(46.125)^2}{4 \times 2}\right)}$$

$$= 32.6 \text{ IN.} = 2.72 \text{ ft}$$

$$D_V = 4 \times \frac{\text{NET FREE VOLUME}}{\text{FRICTION SURFACE}}$$

$$= \frac{4\left[\frac{\pi}{4}\left(\frac{46.125}{12}\right)^2 - 421\left(\frac{\pi}{4}\right)\left(\frac{1.5}{12}\right)^2\right]}{421\,(\pi)\,\frac{1.5}{12}}$$

$$= .155 \text{ ft.}$$

$$D_V^{.745} = .249$$

COMBUSTION ENGINEERING, INC.

ENGINEERING DEPARTMENT, CHATTANOOGA, TENN.

CHARGE NO.

DESCRIPTION

NUMBER E.4

SHEET 4 OF 5

DATE 2-7-73 BY

CHECK DATE 2-7-73 BY

$$\Delta P = 6.043 \times 10^{-5} \frac{L}{D_N^{.745}} \left(\frac{W}{D_A}\right)^{1.855}$$

$$\Delta P = 6.043 \times 10^{-5} \times \frac{1.922}{.249} \left(\frac{1209}{2 \times 2.72}\right)^{1.855}$$

$$= 10.52 \text{ PSF} = .073 \text{ PSI}.$$

2-7-73 SMP
2-7-73 JNA

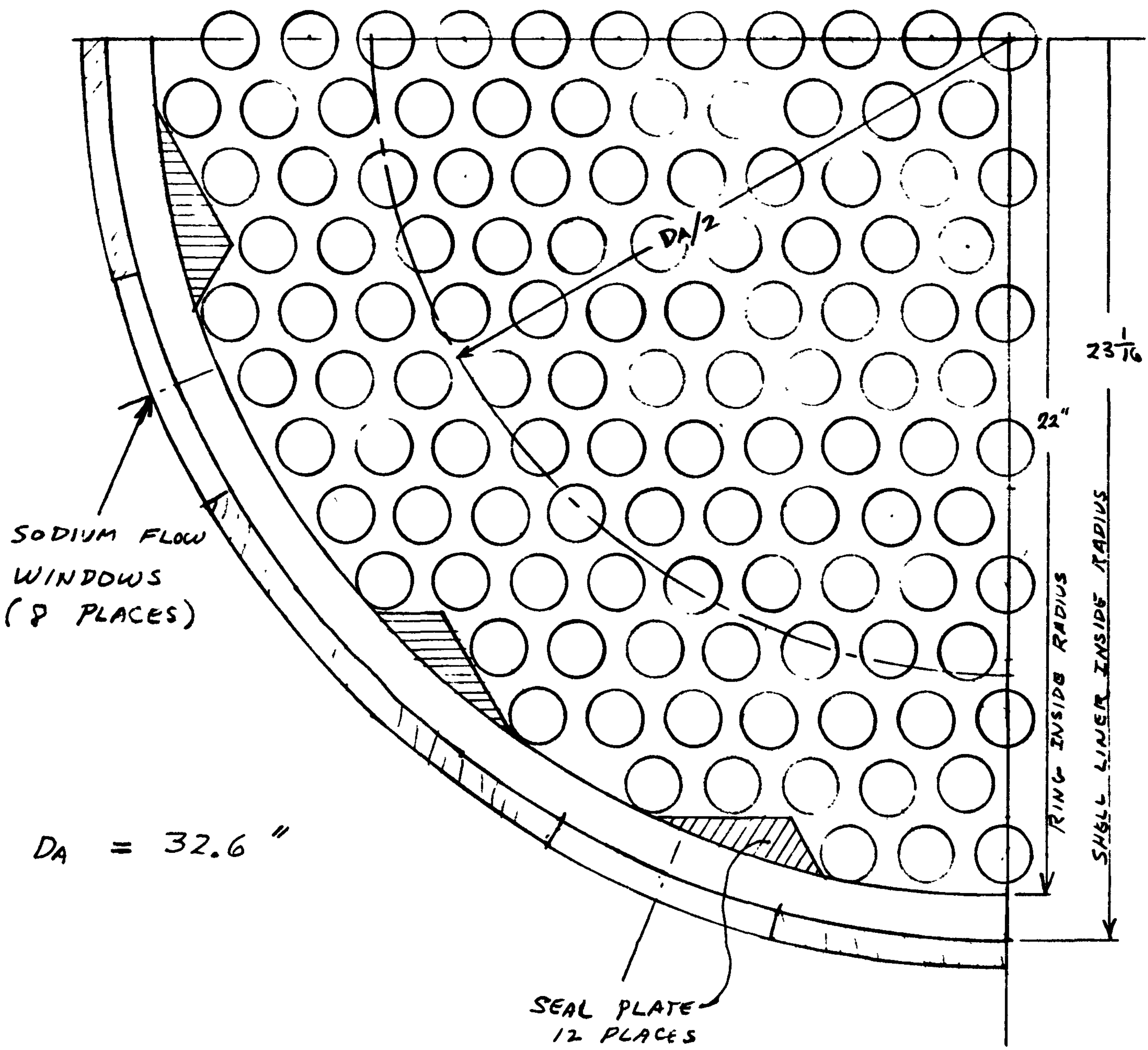


FIGURE 1
EVAPORATOR TUBE BUNDLE