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PM-3A NUCLEAR POWER PLANT
DESIGN ANALYSIS REPORT

Volume II - Book I

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PM-3A-NUCLEAR POWER PLANT
DESIGN ANALYSIS REPORT

Volume II - Book I

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INTRODUCTION

This volume of the PM-3A Design Analysis Report contains a review of the stress analyses of major piping systems and components of the PM-3A Plant. The analyses were done at the request of the U.S. Atomic Energy Commission and the Navy Nuclear Power Unit. The components which were investigated are listed in the Table of Contents.

The object of this review was to confirm the structural adequacy of the system components for operation at rated conditions. The results may also be used to determine the reliability and safety of components in the event of a change on system operating pressure and temperature due to increases in plant generating capacity.

In preparing the report, all available pertinent literature concerning the stress analyses of the PM-3A Plant components was reviewed. References to supporting documents are cited for those calculations and analyses which were found to be applicable to the present plant condition and which were deemed admissable from the viewpoint of accuracy, and adequacy of mathematical representation of the physical system. Included herein, also, are original analyses and computations made by Burns and Roe, Inc. This new work supercedes or supplements the original design work.

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BY: R. J. DOBBS

TITLE: _____ SHEET 1 OF 46

PM-3A

REACTOR PRESSURE VESSEL

STRESS ANALYSIS

ITEM 1.0

ITEM-1

INTRODUCTION

THIS REVIEW CONSISTS OF A CHECK OF THE REACTOR PRESSURE VESSEL STRESS ANALYSIS PRESENTED IN THE MARTIN COMPANY REPORT MND-M-1864 "STRESS ANALYSIS OF REACTOR PRESSURE VESSEL", DEC. 1961.

THE REACTOR PRESSURE VESSEL WAS ANALYZED IN ACCORDANCE WITH ASME BOILER AND PRESSURE VESSEL CODE SECTION VII UNFIRED PRESSURE VESSELS, 1962 AND THE TENTATIVE STRUCTURAL DESIGN BASIS FOR REACTOR PRESSURE VESSELS AND DIRECTLY ASSOCIATED COMPONENTS, U.S. DEPT. OF COMMERCE, PB-151987 (NAVY CODE), 1 DEC 1958.

THE BASIC DESIGN CONDITIONS CONSIDERED ARE AS FOLLOWS;

MECHANICAL LOADINGS

- 1- INTERNAL DESIGN PRESSURE OF 1500 PSI
- 2- EXTERNAL LOADS & MOMENTS IMPOSED ON NOZZLES
- 3- DEAD WEIGHT INCLUDING SHIPPING LOAD FACTORS

THERMAL LOADINGS

- 1- DESIGN TEMPERATURE OF 600°F
- 2- INTERNAL GAMMA HEATING
- 3- TRANSIENT POWER FLUCTUATIONS
- 4- TEMPERATURE LAGS FROM STARTUP AND SHUTDOWN RATE OF 106°F/HR

SPECIFIC DETAILS OF THE DESIGN CONDITIONS CAN BE FOUND IN REFERENCE #1.

SUMMARY

AN INDIVIDUAL SUMMARY IS GIVEN BELOW FOR THE VARIOUS MAJOR ANALYSES PRESENTED IN REFERENCE #1.

1- ASME CODE ANALYSIS

THE PRESSURE VESSEL AND ALL PRESSURE CONTAINING COMPONENTS WERE ANALYZED IN ACCORDANCE WITH SECTION VIII OF THE ASME CODE. IN ALL CASES THE ACTUAL MINIMUM THICKNESS OF THE COMPONENT IS GREATER THAN THAT REQUIRED BY CODE AND THEREFORE ACCEPTABLE. REINFORCEMENT OF OPENINGS, SUCH AS PRIMARY COOLANT NOZZLES WAS PROPERLY INVESTIGATED AND THE REQUIRED METAL AREA WAS SUPPLIED. THE REACTOR PRESSURE VESSEL SATISFIES THE REQUIREMENTS OF THE ASME CODE SECTION VIII.

2- FLANGE TRANSITION SECTION

A DETAILED MECHANICAL PLUS THERMAL DISCONTINUITY ANALYSIS OF THE SHELL FLANGE TRANSITION SECTION WAS PERFORMED. THE RESULTS OF THE MECHANICAL DISCONTINUITY ANALYSIS WERE COMBINED WITH THE THERMAL EFFECTS, INCLUDING TRANSIENTS FOR A FATIGUE EVALUATION OF THE FLANGE TRANSITION SECTION.

FOR PURPOSE OF ANALYSIS, THE TRANSITION SECTION WAS BROKEN UP INTO FAMILIAR SHAPES BY CONSIDERING THE FLANGE AS A RING, THE HUB AS A SHORT CYLINDER, THE TRANSITION AS A TRUNCATED CONE AND THE SHELL AS A SEMI-INFINITE CYLINDER.

SUMMARY - 2 (CONT)

THE DEFORMATION OF EACH INDIVIDUAL SECTION WAS CALCULATED FOR PRESSURE AND THERMAL EFFECTS. CONTINUITY AT THE JOINTS WAS THEN RESTORED BY REDUNDANT MOMENTS AND SHEARS.

THE BASIC ANALYTICAL MODEL, ASSUMPTIONS MADE AND REFERENCES USED IN THE ANALYSIS ARE ALL ACCEPTABLE.

ONE MINOR DISCREPANCY IS THAT THE REF REPORT INDICATES AN ALLOWABLE PRIMARY PLUS SECONDARY STRESS INTENSITY OF 21500 PSI FOR 347 STAINLESS STEEL. ACTUALLY FOR 347SS AT 600°F THE "NAVY CODE" * LISTS 21000 PSI AS THE ALLOWABLE. THIS VALUE WILL BE USED IN ANY FURTHER DISCUSSION.

THE MAXIMUM CALCULATED STRESS INTENSITY OCCURS AT THE HUB - FLANGERING JUNCTION AND IS EQUAL TO 20900 PSI. SINCE THIS IS LESS THAN THE ALLOWABLE OF 21000 PSI, IT IS ACCEPTABLE.

THE RESULTS OF THE FATIGUE ANALYSIS SHOW THAT THE TOTAL USAGE FACTOR (U) IS WELL BELOW THE ALLOWABLE TOTAL USAGE FACTOR OF 0.8. THE TOTAL USAGE FACTOR IS DEFINED AS THE SUM OF THE INDIVIDUAL USAGE FACTORS FOR THE VARIOUS TRANSIENTS CONSIDERED; THE FACTOR FOR EACH TRANSIENT BEING EQUAL TO THE RATIO OF ACTUAL NUMBER OF CYCLES TO ALLOWABLE NUMBER OF CYCLES.

THE SHELL TRANSITION SECTION IS THEREFORE ACCEPTABLE FOR THE DESIGN CONDITIONS INVESTIGATED.

* REF. #9

SUMMARY-(CONT)3- NOZZLE

A DETAILED DISCONTINUITY ANALYSIS AND FATIGUE EVALUATION OF THE PRIMARY COOLANT NOZZLE DESIGN WAS PERFORMED. THE ANALYSIS CONSIDERS THE EFFECT OF PRIMARY PIPE LOADS AND MOMENTS APPLIED TO THE NOZZLES AS WELL AS THE DESIGN PRESSURE OF ISCOPSI AND THERMAL TRANSIENTS.

FOR THE ANALYSIS THE NOZZLE IS BROKEN UP INTO THREE BASIC COMPONENTS: THE SHELL WALL, REINFORCING RING AND PIPE (CYLINDER). THE BASIC ANALYTICAL MODEL, ASSUMPTIONS MADE, AND REFERENCES USED IN THE ANALYSIS ARE ALL ACCEPTABLE.

THE MAXIMUM CALCULATED STRESS-INTENSITIES DUE TO MECHANICAL EFFECTS ARE ALL BELOW THE ALLOWABLE OF 21000 LBS/IN² AND THEREFORE ACCEPTABLE. THE EFFECT OF THERMAL TRANSIENTS WAS SMALL. A FATIGUE EVALUATION CONSIDERING THE START-UP AND SHUT-DOWN TRANSIENT, INDICATES A TOTAL USAGE FACTOR OF ONLY 0.011. THIS IS WELL BELOW THE TOTAL ALLOWABLE OF 0.80, IN ACCORDANCE WITH REF # 9.

THE PRIMARY COOLANT NOZZLES ARE THEREFORE ACCEPTABLE FOR THE DESIGN CONDITIONS INVESTIGATED.

SUMMARY (CONT)4. TOP HEAD

A DETAILED DISCONTINUITY ANALYSIS AND FATIGUE EVALUATION WAS MADE OF THE REACTOR PRESSURE VESSEL TOP HEAD. FOR PURPOSE OF ANALYSIS THE TOP HEAD WAS DIVIDED INTO THREE BASIC COMPONENTS; FLANGE RING, SHORT TRUNCATED CONE AND THICK FLAT PLATE. THE BASIC ANALYTICAL MODEL, ASSUMPTIONS AND REFERENCES USED ARE ACCEPTABLE.

THE REVIEW OF THE ANALYSIS INDICATED SOME DISCREPANCIES IN THE DISCONTINUITY ANALYSIS PRESENTED IN REF #1. SPECIFICALLY ON PAGE 4.12 IN REF #1 THE VALUE OF FACTOR (H_4) WAS INCORRECTLY COMPUTED AS $(0.978 H_4^*)$. USING THE VALUES GIVEN IN REF #1 TO COMPUTE (H_4), YIELDED THE VALUE OF $(0.801 H_4^*)$. THIS AMOUNTS TO AN ERROR OF APPROXIMATELY 19%. SINCE THIS FACTOR WOULD CHANGE COEFFICIENTS IN EACH OF THE CHARACTERISTIC EQUATIONS USED TO OBTAIN THE DISCONTINUITY MOMENTS AND SHEARS, FURTHER INVESTIGATION WAS IN ORDER.

AN INDEPENDENT ANALYSIS OF THE TOP HEAD BY BURNS AND ROE (SEE PAGES 10 TO 44) INDICATES THAT THE MAXIMUM CALCULATED STRESS INTENSITY AT THE OUTER EDGE OF THE TRUNCATED CONE AT THE FLANGE RING JUNCTION IS 26690 psi. THIS IS GREATER THAN THE ALLOWABLE OF 21000 psi AND NOT ACCEPTABLE. FURTHER INVESTIGATION SHOWS THAT AN INCREASE IN THICKNESS FROM $2\frac{3}{4}$ " TO $3\frac{1}{4}$ " IS REQUIRED TO REDUCE THE STRESS TO A VALUE BELOW THE ALLOWABLE OF 21000 psi.

IT SHOULD BE NOTED THAT, ALTHOUGH THE EXISTING DESIGN DOES NOT SATISFY THE DESIGN BASIS (REF #9) USED WHEN THE REACTOR PRESSURE VESSEL WAS ORIGINALLY DESIGNED, IT DOES SATISFY THE REQUIREMENTS OF ASME BOILER

SUMMARY - 4 (CONT)

AND PRESSURE VESSEL CODE SECTION III NUCLEAR VESSELS. THEREFORE, THE CALCULATED STRESS INTENSITY OF 26690 psi WOULD BE ACCEPTABLE IF THE PM-3A REACTOR PRESSURE VESSEL WAS BEING DESIGNED TODAY. FOR EXAMPLE, IF WE COMBINE THE MECHANICAL STRESS OF 26690 psi WITH THE THERMAL DISCONTINUITY STRESS OF 4960 psi (REF #1 p. 4.26) WE OBTAIN A TOTAL STRESS OF 31650 psi. THIS TOTAL (Q) STRESS WOULD BE COMPARED WITH A SECTION III CODE ALLOWABLE OF $3(5m)$ WHICH IS EQUAL TO $3(18400) = 55200$ psi (FOR 347 SS AT 600°F).

BASED ON THE ABOVE DISCUSSION, THE STRUCTURAL INTEGRITY OF THE TOP HEAD CAN BE ASSURED.

THE HIGHER STRESSES CALCULATED WILL HAVE A NEGLIGIBLE EFFECT ON THE FATIGUE EVALUATION OF THE TOP HEAD, THEREFORE NO FURTHER CALCULATIONS BY BURNS AND ROE IS WARRANTED.

THE FATIGUE EVALUATION PRESENTED IN REF #1 REPORT WAS NOT COMPLETED. THE LAST PAGE IN THIS SECTION (4.31) CALCULATES THE ALTERNATING STRESS INTENSITIES BUT DOES NOT GIVE THE ALLOWABLE NUMBER OF CYCLES WHICH IS DETERMINED FROM REF. #9. POSSIBLY A PAGE IS MISSING FROM THE REPORT. USING THE ALTERNATING AND MEAN STRESS OF 35065 psi AND 1935 psi RESPECTIVELY YIELDS AN ALLOWABLE NUMBER OF CYCLES OF APPROXIMATELY 19000. BASED ON A SPECIFIED DESIGN FOR 750 CYCLES OF STARTUP-SHUTDOWN TRANSIENT, THE USAGE FACTOR IS WELL BELOW THE TOTAL USAGE FACTOR OF 0.8 ALLOWED BY REF. #9.

BASED ON THE ABOVE, THE STRUCTURAL INTEGRITY OF THE TOP HEAD IS ASSURED FOR THE DESIGN CONDITIONS INVESTIGATED.

SUMMARY (CONT)5- FIXED ELLIPSOIDAL HEAD

THE ANALYSIS OF THE REACTOR PRESSURE VESSEL BOTTOM ELLIPSOIDAL HEAD CONSIDERS THE PRIMARY AND DISCONTINUITY EFFECTS OF INTERNAL PRESSURE TOGETHER WITH THE EFFECT OF INTERNAL GAMMA HEATING.

THE CALCULATED STRESSES FOR THE LOADINGS, INDICATED ABOVE, ARE NOT CRITICAL AND THEREFORE THE BOTTOM HEAD IS ACCEPTABLE FOR THE DESIGN CONDITIONS INVESTIGATED.

6. ORIFICE PLATE

THE INTERNAL ORIFICE PLATE, LOCATED JUST BELOW THE WATER BOX, IS PERFORATED BY 252 $\frac{1}{2}$ INCH DIAMETER HOLES. THIS PLATE MUST BE CAPABLE OF SUPPORTING THE REACTOR CORE. IT MUST ALSO SUPPLY THE REACTION TO THE HOLDDOWN SPRINGS. IN THE ANALYSIS A CONSERVATIVE DESIGN LOAD OF 5000 LBS AT A LOAD FACTOR OF 3 G'S WAS ASSUMED.

THE ANALYTICAL MODEL, ASSUMPTIONS, REFERENCES USED AND THE CALCULATED STRESSES ARE ALL ACCEPTABLE. THE ORIFICE PLATE IS CAPABLE OF RESISTING THE SPECIFIED DESIGN LOADS.

SUMMARY (CONT)7- THERMAL SHIELDS

THE THERMAL SHIELDS ARE SEGMENTED TO ALLOW FOR REMOVAL FROM THE PRESSURE VESSEL. SINCE THEY ARE NOT PRESSURE CONTAINING COMPONENTS, THE PRIME CONCERN WAS TO INVESTIGATE THEM FOR THE EFFECT OF THERMAL GRADIENTS.

THE CALCULATED THERMAL STRESSES ARE NOT CRITICAL, AND THEREFORE THE THERMAL SHIELDS ARE STRUCTURALLY ACCEPTABLE.

8- REACTOR SUPPORTS

AN ANALYSIS WAS MADE TO INSURE THAT THE REACTOR PRESSURE VESSEL SUPPORT CAN SAFELY SUSTAIN SPECIFIED DESIGN LOADS OF 8 G'S FORWARD AND AFT AND 1 G. VERTICAL, WHICH MIGHT OCCUR DURING SHIPMENT.

THE RESULTS INDICATE THAT THE REACTOR SHIPPING SUPPORT CAN SAFELY RESIST THE SPECIFIED DESIGN LOADS.

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SHEET 10 OF 46

REACTOR PRESSURE VESSEL

TOP HEAD

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* NOTE: MINIMUM THICKNESS (h_2) IS INCREASED TO 3.25"

TOP HEAD(I) ANALYSIS FOR MECHANICAL LOADS(A) DEVELOPMENT OF EQUATIONS1. RIGID RING

THE TOP HEAD FLANGE IS ASSUMED TO ACT AS A RIGID BODY.

ROTATION (θ)

$$E\theta_{c.g.} = \frac{M_{c.g.} r_{im}^2}{I_{NA}} \quad Ref \# 2$$

THE MOMENT ABOUT THE CENTER OF GRAVITY IS,

$$\begin{aligned} M_{c.g.} = & W(r_w - r_{im}) \frac{r_w}{r_{im}} + G_1(r_{im} - r_{ci}) \frac{r_{ci}}{r_{im}} + G_2(r_{im} - r_{ce}) \frac{r_{ce}}{r_{im}} \\ & - N_{IH}(x_2) \frac{r_{em}}{r_{im}} + N_{IH}(r_{im} - r_{em}) \frac{r_{em}}{r_{im}} - M_1 \frac{r_{em}}{r_{im}} + Q_1(x_2) \frac{r_{em}}{r_{im}} \\ & - p(l_1) \left(\frac{l_1}{2} - x_2 \right) + p(r_{e2} - r_{ci}) \left(r_{im} - \frac{r_{e2} + r_{ci}}{2} \right) \left(\frac{r_{e2} + r_{ci}}{2} \right) \left(\frac{1}{r_{im}} \right) \end{aligned}$$

ALSO, $N_{IV} = \frac{p r_{ci}^2}{2 r_{em}} ; N_{IH} = N_{IV} \tan \alpha$

FOR $\alpha = 45^\circ$, $\tan \alpha = 1.0$

SUBSTITUTING INTO EQUATION FOR $E\theta$ YIELDS,

TOP HEADI-A-1 (CONT.)

$$\begin{aligned}
 E\theta_{c.g.} = & \frac{W}{I_{NA}} (r_w - r_m) r_w r_m + \frac{Q_1 (r_m - r_o)}{I_{NA}} (r_o r_m) \\
 & + \frac{Q_2 (r_m - r_{o2})}{I_{NA}} (r_{o2} r_m) - \frac{M_1 r_{2m} r_m}{I_{NA}} + \frac{Q_1 (x_2) r_{2m} r_m}{I_{NA}} \\
 & + \frac{p}{I_{NA}} \left\{ - \frac{x_2 (r_{2m}) r_m r_{2i}^2}{2 r_{2m}} + (r_m - r_{2m}) \frac{r_{2m} r_m r_{2i}^2}{2 r_{2m}} - r_1 r_m^2 \left(\frac{r_1}{2} - x_2 \right) \right. \\
 & \left. + r_m (r_{o2} - r_{2i}) \left(r_m - \frac{r_{o2} + r_{2i}}{2} \right) \left(\frac{r_{o2} + r_{2i}}{2} \right) \right\} \\
 & \equiv K_1
 \end{aligned}$$

DISPLACEMENT (δ)

THE DISPLACEMENT (δ) OF THE RING AT JUNCTION ① IS $\delta_{c.g.} + \theta x_2$ For small angles

$$\therefore E\delta_1 = E\delta_{c.g.} + E\theta_{c.g.} x_2$$

$$E\delta_1 = \frac{Q_{c.g.} r_m^2}{Area} + E\theta_{c.g.} x_2$$

TOTAL FORCE AT CENTER OF GRAVITY IS,

$$Q_{c.g.} = -N_H \frac{r_{2m}}{r_m} + Q_1 \frac{r_{2m}}{r_m} + p r_1 \left(\frac{r_{2i} + r_{1i}}{2} \right) \left(\frac{1}{r_m} \right)$$

TOP HEADA-1 (CONT)

$$E\delta_1 = -\frac{N_H r_{im} r_{em}}{A} + \frac{Q_1 r_{im} r_{em}}{A} + \frac{p r_i}{A} \frac{(r_{em} + r_{im})}{2} r_{im} + E\theta_{ex} r_e$$

WHERE (A) = CROSS-SECTIONAL AREA OF RING.

THE NUMERICAL CONSTANTS FOR THE RING
EQUATIONS WILL NOW BE DETERMINED.

NOTE: $\frac{N_H r_{im} r_{em}}{A} = \frac{p r_i^2 \tan \alpha}{2 r_{em}} \frac{r_{im} r_{em}}{A} = \frac{r_i^2}{2A} r_{im}(p)$

For $\alpha = 45^\circ$

TOP HEAD (CONT)I-A-1 (CONT)RING CONSTANTS

THE CENTROID OF THE RING
WILL BE DETERMINED FIRST.

$$X_2 = \frac{h_1}{3} \left(\frac{2a+b}{a+b} \right) \quad \text{Ref: 43}$$

$$X_2 = \frac{6.625}{3} \left[\frac{2(6.58) + 10.18}{6.58 + 10.18} \right]$$

$$X_2 = 154.6 / 50.28 = 3.07" \quad , \quad X_1 = 3.56"$$

$$Y_2 = \frac{\sum AY}{\sum A} = \frac{6.58(6.625)(3.29) + 0.5(6.625)(3.60)(1.78)}{6.58(6.625) + 0.5(6.625)(3.60)}$$

$$Y_2 = 236.2 / 55.5 = 4.25"$$

$$r_{im} = 20.0 - 4.25 = 15.75"$$

$$h_1 = 6.58 + \left(\frac{3.60}{6.625} \right) (3.56) = 6.58 + 1.93 = 8.51"$$

TOP HEADI-A-1 (CONT)MOMENT OF INERTIA (I_{NA})

$$I_{NA} = \frac{l^3 (a^2 + 4ab + b^2)}{36(a+b)}$$

$$= \frac{(6.625)^3 \{ 6.58^2 + 4(6.58)(10.18) + (10.18)^2 \}}{36(6.58 + 10.18)}$$

$$= 120611 / 603.3 = 200 \text{ in}^4$$

$$\frac{(r_w - r_m)}{I_{NA}} r_w r_m = \frac{(18.00 - 15.75)(15.75)(18.00)}{200} = 3.189$$

$$\frac{(r_m - r_1)}{I_{NA}} r_1 r_m = \frac{(15.75 - 15.58)(15.58)(15.75)}{200} = 0.209$$

$$\frac{(r_m - r_2)}{I_{NA}} r_2 r_m = \frac{(15.75 - 14.14)(14.14)(15.75)}{200} = 1.793$$

$$\frac{r_m r_m}{I_{NA}} = \frac{11.8(15.75)}{200} = 0.929$$

$$\frac{r_m r_m}{I_{NA}} (X_2) = 3.07(0.929) = 2.862$$

$$\frac{(r_m - r_2) r_m r_2^2}{2 I_{NA}} = \frac{(15.75 - 11.8)(15.75)(9.82)^2}{2(200)} = 15.000$$

$$\frac{r_1 r_m^2}{I_{NA}} \left(\frac{r_1}{2} - X_2 \right) = \frac{6.625(15.75)^2 (3.312 - 3.07)}{200} = 1.989$$

$$\frac{K_1}{I_{NA}} = \frac{15.75(14.14 - 9.82)(15.75 - 12.0)(12.0)}{200} = 15.309$$

TOP HEADI-A-1 (CONT)MOMENT OF INERTIA (I_{NA})

$$I_{NA} = \frac{l^3 (a^2 + 4ab + b^2)}{36(a+b)}$$

$$= \frac{(6.625^3) \{ 6.58^2 + 4(6.58)(10.18) + (10.18)^2 \}}{36(6.58 + 10.18)}$$

$$= 120611 / 603.3 = 200 \text{ in}^4$$

$$\frac{(r_w - r_m)}{I_{NA}} r_w r_m = \frac{(18.00 - 15.75)(15.75)(18.00)}{200} = 3.189$$

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$$\frac{r_m r_m}{I_{NA}} = \frac{11.8(15.75)}{200} = 0.929$$

$$\frac{r_m r_m}{I_{NA}} (x_2) = 3.07(0.929) = 2.852$$

$$\frac{(r_m - r_m) r_m r_2^2}{2 I_{NA}} = \frac{(15.75 - 11.8)(15.75)(9.82)^2}{2(200)} = 15.000$$

$$\frac{r_m^2}{I_{NA}} \left(\frac{l}{2} - x_2 \right) = \frac{6.625(15.75)^2 (3.312 - 3.07)}{200} = 1.989$$

$$\frac{K_1}{I_{NA}} = \frac{15.75(14.14 - 9.82)(15.75 - 12.0)(12.0)}{200} = 15.309$$

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$$\frac{r_m r_m}{A} = \frac{11.8 (15.75)}{55.5} = 3.349$$

$$\frac{\lambda_i (r_{2i} + r_{1i}) (r_m)}{2(A)} = \frac{6.625 (9.82 + 13.42) (15.75)}{2(55.5)} = 21.846$$

$$\frac{x_2 r_m r_{2i}^2}{2 I_{NA}} = \frac{3.07 (15.75) (9.82)^2}{2(200)} = 11.656$$

$$\frac{3.349 (r_{2i}^2)}{2 r_m} = \frac{3.349 (9.82)^2}{2(11.8)} = 13.684$$

TOP HEADI-(A) CONT2 - FLAT PLATE

$$2h = h_3 = 4.79''$$

DISPLACEMENT DUE TO PRESSURE(P)

THE DISPLACEMENT OF POINT ② DUE TO PRESSURE IS OBTAINED BY USING EQUATIONS FOR A THICK PLATE SUPPORTED AT THE EDGE. THE EQUATIONS INCLUDE THE EFFECT OF SHEAR DEFORMATION.

FROM REF # 4 p 481 AND SUBSTITUTING δ FOR u , r FOR x AND v FOR y WE OBTAIN THE DISPLACEMENT AT $r = r_m$,

$$\delta_{r_m} = \frac{p r_m}{E} \left[\frac{v}{2} - \frac{3(1-v)}{32} \left(\frac{z}{h} \right) \left\{ (3+v) r_m^2 - (1+v) r_m^2 \right\} + \frac{2}{40} \left(\frac{z}{h} \right) (2+9v-v^2) - \frac{1}{8} \left(\frac{z}{h} \right) (2-v-v^2) \right]$$

δ = DISPLACEMENT (HORIZONTAL)

w = DEFLECTION

SOLVING FOR THE DISPLACEMENT AT THE INTERFACE POINT ② ($z=h$) YIELDS,

TOP HEADI-A-2 (CONT)

$$E \delta_{\beta m, z=h} = p \beta_m \left[\frac{\nu}{2} - \frac{3(1-\nu)}{32 h^2} \left\{ (3+\nu) \beta_m^2 - (1+\nu) \beta_m^2 \right\} + \frac{3}{40} (2+9\nu-\nu^2) - \frac{1}{8} (2-\nu-\nu^2) \right]$$

$$E \delta_{\beta m, z=h} = p \beta_m K_z$$

ROTATION DUE TO PRESSURE (P)

THE DEFLECTION (W) OF A THICK PLATE IS (FROM REF # 4 p 481)

$$w = -\frac{1}{8} p \frac{r^2}{D} (a^2 - r^2) \left\{ \frac{1}{8} \left(\frac{5+\nu}{1+\nu} (a^2) - r^2 \right) + \frac{1}{9} \left(\frac{8+\nu+\nu^2}{1-\nu^2} \right) (h^2) \right\}$$

$$+ p \frac{z}{E} \left[-\frac{1}{2} + \frac{3\nu z}{32 h^3} \left\{ (3+\nu) a^2 - 2(1+\nu) r^2 \right\} - \frac{3}{40} \left(\frac{z}{h} \right) (5+2\nu+\nu^2) + \frac{1}{16} \frac{z^3}{h^3} (1+\nu)^2 \right],$$

THE ROTATION CAN BE DETERMINED BY DIFFERENTIATING WITH RESPECT TO r

$$\theta = \frac{dw}{dr} = \frac{p r (a^2 - r^2)}{32 D} + \frac{p r}{4 D} \left\{ \frac{1}{8} \left(\frac{5+\nu}{1+\nu} (a^2) - r^2 \right) + \frac{1}{9} \left(\frac{8+\nu+\nu^2}{1-\nu^2} \right) (h^2) \right\} - \frac{3}{8} \frac{p z^2 \nu (1+\nu) r}{E h^3}$$

TOP HEADI-A-2 (CONT)FOR POINT ②AT $r=a=r_m$ AND $z=h$ WE OBTAIN

$$\theta = 0 + \frac{p r_m}{4D} \left\{ \frac{1}{8} (r_m)^2 \left(\frac{5+v}{1+v} - 1 \right) + \frac{1}{5} \left(\frac{8+v+v^2}{1-v^2} \right) (h^2) \right\} \\ - \frac{3}{8} \frac{p r_m v (1+v)}{E h}$$

WHERE $D = \frac{E (2h)^3}{12(1-v^2)}$

FINALLY,

$$E\theta_2 = p \left[\frac{r_m (3 \times 1 - v^2)}{(8) h^3} \left\{ \frac{(r_m)^2}{8} \left(\frac{5+v}{1+v} - 1 \right) + \frac{(h^2)}{5} \left(\frac{8+v+v^2}{1-v^2} \right) \right\} \right. \\ \left. + \frac{3 r_m v (1+v)}{8 h} \right]$$

$$E\theta_2 = p K_3 \quad (\text{AT } r=r_m, z=h) \quad \text{Note: } 2h = h_3 = 4.75''$$

TOP HEADI-A-2 (CONT)DISPLACEMENT & ROTATION DUE TO DISCONTINUITYMOMENTS & FORCES

ROTATION OF EDGE AT MID-PLANE OF PLATE

(M) at mid-plane = $M_m = M_2 - (Q_2 + N_{2H})h$, where $h = (h_3/2)$

FROM REF #5 p.62 WE FIND,

$$w = \frac{M}{2D(1+\nu)} (a^2 - r^2)$$

$$\theta = -\frac{dw}{dr} = + \frac{2Mr}{2D(1+\nu)}$$

$$D = \frac{Eh_3^3}{12(1-\nu^2)}$$

$$\theta = \frac{12(1-\nu^2)Mr}{Eh_3^3(1+\nu)} = \frac{12(1+\nu)(1-\nu)Mr}{E(h_3)^3(1+\nu)}$$

AT $r = r_{3M}$

$$E\theta_2 = \frac{12(M_2 - [Q_2 + N_{2H}][h_3/2]) r_{3M} (1-\nu)}{(h_3)^3}$$

$$E\theta_2 = \frac{12 r_{3M} (1-\nu)}{(h_3)^3} (M_2 - [Q_2 + N_{2H}][\frac{h_3}{2}]) \quad \text{--- (DUE TO } M_2 \text{ \& } Q_2 \text{ ONLY)}$$

TOP HEADI-A-2 (CONT)DISPLACEMENT (S)

$$S_m(\text{OF MID-PLANE}) = \frac{(1-\nu) Q_m \bar{f}_m}{E h_3}$$

$$ES_2 = ES_m - E\theta_2 \left(\frac{h_3}{2} \right)$$

$$= \frac{(Q_2 + N_{2M})(1-\nu) \bar{f}_m}{h_3} - \frac{6 \bar{f}_m (1-\nu)}{(h_3)^3} (M_2 - [Q_2 + N_{2M}] \left[\frac{h_3}{2} \right])$$

$$ES_2 = \frac{4 \bar{f}_m (1-\nu) (Q_2 + N_{2M})}{h_3} - \frac{6 \bar{f}_m (1-\nu)}{(h_3)^3} (M_2)$$

Due To M_2 & Q_2
ONLY

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BY: R. J. DOBBSTITLE: REACTOR PRESSURE VESSEL - PM-3A SHEET 23 OF 46TOP HEADI-A-2 (CONT)FLAT PLATE CONSTANTS

$$h = h_{1/2} = 2.37$$

$$f_{3m} K_2 = 7.7 \left[\frac{0.30}{2} - \frac{3(1-0.3)}{32(2.37)^3} \left\{ 3.3(7.7)^2 - 1.3(7.7)^2 \right\} \right. \\ \left. + \frac{3(2 + 2.7 - 0.09)}{40} - \frac{1}{8}(2 - 0.30 - 0.09) \right] \\ = 7.7 \left[0.150 - \underbrace{1.987}_{-1.090} + 0.346 - 0.201 \right]$$

$$f_{3m} K_2 = -8.393$$

$$K_3 = - \frac{7.7(3)(1-0.09)}{(8)(2.37)^3} \left\{ \frac{(7.7)^2}{8} \left(\frac{5.3}{1.3} - 1 \right) + \frac{(2.37)^2}{9} \left(\frac{8+0.30+0.09}{0.91} \right) \right\} \\ - \frac{3(7.7)(0.30)(1.30)}{8(2.37)}$$

$$= 0.197 \left\{ 22.804 + 10.557 - 0.475 \right\}$$

$$K_3 = 6.439$$

$$\frac{12 f_{3m} (1-v)}{(h_0)^3} = \frac{12(7.7)(0.70)}{(4.75)^3} = 0.604$$

$$0.604 \left(\frac{h_2}{2} \right) = 0.604(2.37) = 1.433$$

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BY: R. J. DOBBSTITLE: REACTOR PRESSURE VESSEL - PM-3ASHEET 24 OF 46TOP HEADI-A-2 (CONT)FLAT PLATE CONSTANTS (CONT)

$$\frac{4 f_{3m} (1-\nu)}{h_3} = \frac{4(7.7)(0.70)}{4.75} = 4.539$$

$$\frac{6 f_{3m} (1-\nu)}{(4.75)^2} = \frac{6(7.70)(0.70)}{22.562} = 1.433$$

$$N_{2H} = \frac{p f_{3i}^2}{2 f_{3m}} = \frac{(5.72)^2}{2(7.7)} p = 2.124 p$$

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BY: R. J. DOBBSTITLE: REACTOR PRESSURE VESSEL - PM-3ASKETCH 25 OF 46TOP HEADI-A. (3) SHORT TRUNKATED CONE

THE METHOD OF REFERENCE #6 WILL BE USED TO DETERMINE THE EFFECT OF END MOMENTS AND SHEARS. THE EFFECT OF PRESSURE WILL BE DETERMINED IN ACCORDANCE WITH REF. #7.

ROTATION AND DISPLACEMENT OF JUNCTION ① AND JUNCTION ② OF CONE SEGMENT

NOTATION OF REF #6 IS USED

$$r_2 = 7.7''$$

$$r_1 = 11.8''$$

$$h = h_2 = 2.75'' \text{ CONSTANT}$$

$$r_m = \frac{r_2 + r_1}{2} = \frac{19.5}{2} = 9.75''$$

$$C \approx \frac{-L^2}{4r_m / \cos \phi}$$

$$C \approx \frac{-(2.90)^2}{4(9.75)/0.707} = -0.152''$$

$$zL = 9.80''$$

C = X LOCATION TO VIRTUAL CENTER

$$\therefore r_0 = r_m - C \cos \phi = 9.75 - 0.152(0.707) = 9.64''$$

$$\text{Attenuation Length } (l) = 0.778 \sqrt{r_0 h_0 / \sin \phi}$$

$$l = 0.778 \sqrt{37.5} = 4.76''$$

TOP HEADI-A-3 (CONT)

$$f = l/D_o, \quad D_o = Eh_o^3 / (12(1-\nu^2)), \quad \nu = 0.30$$

$$= \frac{E(2.75)^3}{10.08} = 1.904E$$

$$f = \frac{4.76}{1.904E} = \frac{2.50}{E}$$

$$l_A = l \sin \phi = 4.76 \sin 49^\circ = 3.57$$

$$\text{FACTOR 'N', } 2Nl = \int_{-L}^{+L} \sqrt{\frac{r_o}{r}} dx_m$$

$$Q = \sin \phi / \sin \phi_o = 1.0$$

$$r = r_o = \frac{r_m + x_m \cos \phi}{\cos \phi}$$

$$\eta = h/h_o = 1.0$$

$$2Nl = \int_{-L}^{+L} \sqrt{\frac{r_o}{r_m + x_m \cos \phi}} dx_m = \sqrt{r_o} \int_{-L}^{+L} \frac{1}{\sqrt{r_m + \cos \phi x_m}} dx$$

$$= \frac{2\sqrt{r_o}}{\cos \phi} \left[\sqrt{r_m + \cos \phi(L)} - \sqrt{r_m - \cos \phi(L)} \right]$$

$$\therefore N = \frac{\sqrt{r_o}}{l \cos \phi} \left[\sqrt{\dots} - \sqrt{\dots} \right] = \frac{\sqrt{261}}{3.57} \left[\sqrt{9.79 + 2.90(0.707)} - \sqrt{9.79 - 2.90(0.707)} \right]$$

$$N = 0.923 \left[\underset{0.68}{3.43} - 2.78 \right] = 0.600$$

TOP HEADI-A-3 (CONT)

EQUATIONS FOR DISPLACEMENT & ROTATION ARE (Ref #6),

$$S_u^*/f l_A = 2G_2 M_2^* - \frac{1}{2} J_2 M_u^* + G_1 l_A H_2^* + \frac{1}{2} J_1 l_A H_u^*$$

$$X_u^*/f = -2G_3 M_2^* + J_3 M_u^* - 2G_2 l_A H_2^* - \frac{1}{2} J_2 l_A H_u^*$$

$$S_2^*/f l_A = -\frac{1}{2} J_2 M_2^* + 2G_2 M_u^* - \frac{1}{2} J_1 l_A H_2^* - G_1 l_A H_u^*$$

$$X_2^*/f = -J_3 M_2^* + 2G_3 M_u^* - \frac{1}{2} J_2 l_A H_2^* - 2G_2 l_A H_u^*$$

THE POWER-SERIES EXPRESSIONS OF G_K AND J_K ARE

AS FOLLOWS,

$$N = 0.60$$

$$4NG_1 = 1 - 0.228571 N^4 + 0.032162 N^6 \dots$$

$$G_1 = \frac{1}{4(0.6)} [0.971] = 0.405$$

$$16N^2 G_2/3 = 1 - 0.330159 N^4 + 0.049284 N^6 \dots$$

$$G_2 = \frac{3.0}{16(0.36)} [0.958] = 0.500$$

$$16N^3 G_3/3 = 1 - 0.685714 N^4 + 0.11249 N^6 \dots$$

$$G_3 = \frac{3}{16(0.216)} [0.913] = 0.793$$

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TOP HEADI-A-3 (CONT)

$$NJ_1 = 1 + 0.152381N^4 - 0.016653N^6 \dots$$

$$J_1 = \frac{1}{0.60} [1.019] = 1.699$$

$$4N^2J_2/3 = 1 + 0.558730N^4 - 0.052304N^6 \dots$$

$$J_2 = \frac{3}{4(0.36)} [1.0719] = 2.232$$

$$8N^3J_3/3 = 1 + 1.980952N^4 - 0.124543N^6 \dots$$

$$J_3 = \frac{3}{8(0.216)} [1.299] = 2.178$$

SUBSTITUTING INTO THE "STARRED" EQUATIONS YIELDS,

$$f_{LA} = 8.425/E, \quad f = 2.50/E, \quad l_A = 3.37$$

$$E \delta_u^* = 8.425 M_2^* - 9.402 M_u^* + 11.499 H_2^* + 24.119 H_u^*$$

$$E \chi_u^* = -3.965 M_2^* + 5.445 M_u^* - 8.425 H_2^* - 9.402 H_u^*$$

$$E \delta_2^* = -9.402 M_2^* + 8.425 M_u^* - 24.119 H_2^* - 11.499 H_u^*$$

$$E \chi_2^* = -5.445 M_2^* + 3.965 M_u^* - 9.402 H_2^* - 8.425 H_u^*$$

TOP HEADI-A-3 (CONT)

THE EQUATIONS WILL NOW BE TRANSFORMED INTO ACTUAL DISPLACEMENT & ROTATION EQUATIONS FOR THE TRUNCATED CONE.

FROM REF #6

$$\delta_M = [] \delta_u^* = \left(\frac{r_u}{r_o} \right)^{1/4} \delta_u^* = \left(\frac{11.8}{9.64} \right)^{1/4} \delta_u^* = 1.052 \delta_u^*$$

$$\delta_L = [] \delta_L^* = \left(\frac{r_L}{r_o} \right)^{1/4} \delta_L^* = \left(\frac{7.7}{9.64} \right)^{1/4} \delta_L^* = 0.944 \delta_L^*$$

$$\chi_u = \left(\frac{r_o}{r_u} \right)^{1/2} \left(\frac{r_u}{r_o} \right)^{1/4} \chi_u^* = \left(\frac{r_o}{r_u} \right)^{1/4} \chi_u^* = \left(\frac{9.64}{11.8} \right)^{1/4} \chi_u^* = 0.948 \chi_u^*$$

$$\chi_L = \left(\frac{r_o}{r_L} \right)^{1/4} \chi_L^* = \left(\frac{9.64}{7.7} \right)^{1/4} \chi_L^* = 1.058 \chi_L^*$$

$$M_u = \left(\frac{r_o}{r_u} \right) \left(\frac{r_u}{r_o} \right)^{1/4} M_u^* = \left(\frac{r_o}{r_u} \right)^{3/4} M_u^* = \left(\frac{9.64}{11.8} \right)^{3/4} M_u^* = 0.859 M_u^*$$

$$M_L = \left(\frac{r_o}{r_L} \right)^{3/4} M_L^* = \left(\frac{9.64}{7.7} \right)^{3/4} M_L^* = 1.183 M_L^*$$

$$H_u = \left(\frac{r_o}{r_u} \right)^{3/2} \left(\frac{r_u}{r_o} \right)^{1/4} H_u^* = \left(\frac{r_o}{r_u} \right)^{5/4} H_u^* = \left(\frac{9.64}{11.8} \right)^{5/4} H_u^* = 0.777 H_u^*$$

$$H_L = \left(\frac{r_o}{r_L} \right)^{5/4} H_L^* = \left(\frac{9.64}{7.7} \right)^{5/4} H_L^* = 1.324 H_L^*$$

CONVERTING TO NOTATIONS OF THIS REPORT REQUIRE THE FOLLOWING ADDITIONAL SUBSTITUTIONS.

$$E\delta_z = E\delta_L$$

$$E\theta_z = E\chi_L$$

$$M_z = M_L$$

$$Q_z = H_L$$

$$E\delta_1 = E\delta_u$$

$$E\theta_1 = E\chi_u$$

$$M_1 = M_u$$

$$Q_1 = -H_u$$

TOP HEADI-A-3 (CONT)

THE EQUATIONS FOR (δ) AND (θ) DUE TO
(M) AND (Q) BECOME,

$$E\delta_1 = 7.492 M_2 - 11.514 M_1 + 9.137 Q_2 - 32.655 Q_1$$

$$E\theta_1 = -3.177 M_2 + 6.009 M_1 - 6.032 Q_2 + 11.471 Q_1$$

$$E\delta_2 = -7.500 M_2 + 9.259 M_1 - 17.197 Q_2 + 13.970 Q_1$$

$$E\theta_2 = -4.870 M_2 + 4.883 M_1 - 7.513 Q_2 + 11.472 Q_1$$

TOP HEADI-A-3 (CONT)DISPLACEMENT & ROTATION DUE TO PRESSURE

REFERENCE # 7

$$E\delta_1 = p \frac{r_{2i}}{h_2 \cos \alpha} \left(r_{2m} - \frac{1}{2} r_{2i} \right)$$

$$E\theta_1 = - \frac{3 r_{2m} \tan \alpha}{2 h_2 \cos \alpha} p$$

$$E\delta_2 = p \frac{r_{2i}}{h_2 \cos \alpha} \left(r_{2m} - \frac{1}{2} r_{2i} \right)$$

$$E\theta_2 = - \frac{3 r_{2m} \tan \alpha}{2 h_2 \cos \alpha} p$$

CONSTANTS :

$$\frac{101.411}{1.969} \frac{r_{2i}}{h_2 \cos \alpha} \left(r_{2m} - \frac{1}{2} r_{2i} \right) = \frac{982}{2.75(0.707)} (11.8 - 0.15(9.82)) = 92.166$$

$$\frac{3 r_{2m} \tan \alpha}{2 h_2 \cos \alpha} = \frac{3(11.8)(1.0)}{2(2.75)(0.707)} = 9.105$$

9.278

$$\frac{r_{2i}}{h_2 \cos \alpha} \left(r_{2m} - \frac{1}{2} r_{2i} \right) = \frac{5.72(7.70 - 0.15(4.72))}{2.75(0.707)} = 20.132$$

$$\frac{3 r_{2m} \tan \alpha}{2 h_2 \cos \alpha} = \frac{3(7.70)(1.0)}{2(2.75)(0.707)} = 5.941$$

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BY: R. J. DOBBS

TITLE: REACTOR Pressure Vessel - PH-34 SHEET 32 OF 46

TOP HEAD

I-ALCOULT

TABLE 1 - EDGE DEFORMATIONS IN TERMS OF EDGE LOADS AND MECHANICAL LOADS

		M_1	Q_1	M_2	Q_2	P	G_1	G_2	W
R	$E\theta =$	(-) 0.929	2.852	0	0	16.650	0.209	1.793	3.189
	$E\delta_1 =$	(-) 2.852	12.105	0	0	59.277	0.642	5.905	9.790
SHORT CORNER	$E\theta_1 =$	6.009	11.471	(-) 3.177	(-) 6.032	(-) 9.105			
	$E\delta_1 =$	(-) 11.514	(-) 32.635	7.492	9.137	52.166			
	$E\theta_2 =$	4.883	11.472	(-) 4.870	(-) 7.513	(-) 5.941			
	$E\delta_2 =$	9.259	13.970	(-) 7.500	(-) 17.197	20.132			
PLATE	$E\theta_2 =$			0.604	(-) 1.433	3.395			
	$E\delta_2 =$			(-) 1.433	4.539	1.248			

* EQUATIONS REVISED - SEE TABLE - 3

0

TOP HEADI-B SOLUTION OF EQUATIONS

THE FOLLOWING EQUATIONS OF CONTINUITY
ARE USED TO SOLVE FOR THE UNKNOWN DISCONTINUITY
MOMENTS (M) AND SHEARS (Q)

$$(1) \quad E\theta_{RING} = E\theta_{CONE}$$

$$(2) \quad ES_{RING} = ES_{CONE}$$

$$(3) \quad E\theta_2_{CONE} = E\theta_2_{PLATE}$$

$$(4) \quad ES_2_{CONE} = ES_2_{PLATE}$$

THE ABOVE RELATIONSHIPS YIELD THE
FOUR CHARACTERISTIC EQUATIONS GIVEN
IN TABLE - 2 .

TOP HEAD
I-B (CONT)

TABLE (2) - CHARACTERISTIC EQUATIONS FOR TOP HEAD

	M_1	Q_1	M_2	Q_2		p	G_1	G_2	W
(1)	(-)	(-)	(-)	(-)		(-)	(-)	(-)	(-)
	6.938	8.619	3.177	6.032		25.755	0.209	1.793	3.189
(2)	8.662	44.760	7.492	9.137	=	(-)	(-)	(-)	(-)
			(-)	(-)		7.111	0.642	5.505	9.790
(3)	4.883	11.472	5.474	6.080		9.366	0	0	0
(4)	9.259	13.970	6.067	21.736		(-)	0	0	0
			(-)	(-)		18.884			

(1) $E\theta_{RING} = E\theta_{CONE}$

(2) $E\delta_{RING} = E\delta_{CONE}$

(3) $E\theta_{2CONE} = E\theta_{2PLATE}$

(4) $E\delta_{2CONE} = E\delta_{2PLATE}$

ABOVE EQUATIONS HAVE BEEN REVISED - SEE TABLE - 4

TOP HEAD (CONT)I-(B) (CONT)

THE SOLUTION OF THE EQUATIONS
YIELDS THE FOLLOWING RESULTS,

$$M_1 = 7.25(p) + 0.07(G_1) + 0.63(G_2) + 1.13(W)$$

$$Q_1 = -1.07(p) - 0.03(G_1) - 0.23(G_2) - 0.41(W)$$

$$M_2 = -1.63(p) + W_{EG} - 0.07(G_2) - 0.13(W)$$

$$Q_2 = 3.72(p) + W_{EG} + 0.14(G_2) + 0.25(W)$$

TOP HEAD (CONT)II- STRESSES DUE TO MECHANICAL LOADSDESIGN CONDITIONS $p = 1500 \text{ psi}$, $T = 600^\circ\text{F}$ BOLT & GASKET LOADS $G_1 = G_2 = 2700 \text{ lb/in}$

$$W = 12788 \text{ lb/in}$$

THE DISCONTINUITY FORCES BECOME,

$$M_1 = 7.25(1500) + 0.07(2700) + 0.63(2700) + 1.13(12788) = 27,200 \text{ lb-in}$$

$$Q_1 = -1.07(1500) - 0.03(2700) - 0.23(2700) - 0.41(12788) = -7950 \text{ lb}$$

$$M_2 = -1.63(1500) + 0 - 0.07(2700) - 0.13(12788) = -4300 \text{ lb-in}$$

$$Q_2 = 3.72(1500) + 0 + 0.14(2700) + 0.25(12788) = 9160 \text{ lb}$$

$$N_{1v} = N_{1h} = 4.09 p = 4.09(1500) = 6135 \text{ lb/in}$$

$$N_{2v} = N_{2h} = 2.12 p = 2.12(1500) = 3180 \text{ lb/in}$$

THE MINUS SIGNS INDICATE THAT THE DIRECTION OF THE FORCE WAS ASSUMED INCORRECTLY.

TOP HEAD (CONT)II-(CONT)(A) STRESSES IN CONE AT JUNCTION ①AXIAL

$$N_x = N_H \sin \alpha + N_V \cos \alpha + Q_1 \sin \alpha = 14000 \text{ LBS}$$

$$\sigma_x = \frac{N_x}{h_2} \pm \frac{6 M_1}{h_2^2} = \frac{14000}{2.75} \pm \frac{6(21200)}{(2.75)^2}$$

$$= 5090 \pm 21600 = +26690 \frac{\text{LBS}}{\text{IN}^2} \text{ OUTER}$$

$$-16510 \frac{\text{LBS}}{\text{IN}^2} \text{ INNER}$$

THE MAXIMUM STRESS OF 26690 $\frac{\text{LBS}}{\text{IN}^2}$ IS ABOVE THE ALLOWABLE $S_p = 21000$ (FOR 347 $\frac{\text{LBS}}{\text{IN}^2}$ AT 600°F). THE ACTUAL THICKNESS AT JUNCTION ① IS SLIGHTLY GREATER THAN 2 $\frac{3}{4}$ " (BY APPROX $\frac{1}{8}$ "). THIS OCCURS DUE TO THE FACT THAT THE INSIDE SURFACE OF THE TOP HEAD IS SPHERICAL AND THE 2 $\frac{3}{4}$ " ACTUALLY OCCURS ONLY AT ONE POINT (I.E. AT 45° ANGLE). HOWEVER, THE ADDITIONAL MATERIAL IS NOT SUFFICIENT TO REDUCE THE TOTAL STRESS TO THE ALLOWABLE VALUE. THEREFORE, A MODIFICATION IS REQUIRED.

THE THICKNESS OF THE HEAD AT THE CONICAL SECTION WILL BE INCREASED TO 3 $\frac{1}{4}$ " AT THE MINIMUM SECTION. THIS WILL PROVIDE A THICKNESS OF 3.38" AT THE JUNCTION.

THE NEW DISCONTINUITY MOMENT & SHEARS WILL NOW BE DETERMINED.

TOP HEAD (CONT)II (CONT)

THE DISCONTINUITY FORCES WILL BE RECALCULATED FOR THE NEW MINIMUM THICKNESS OF $3\frac{1}{4}"$ FOR THE CONICAL SEGMENT. THE EFFECT OF CHANGE OF RADIUS IS SMALL AND IS THEREFORE NEGLECTED. ONLY THE EQUATIONS FOR THE CONICAL SEGMENT ARE AFFECTED.

NEW FACTORS;

$$f = \frac{l}{D_o}$$

$$, l = \sqrt{\left(\frac{3.25}{2.75}\right)} (4.76) = 5.19$$

$$D_o = E h_o^3 / (2(1-\nu^2)) = \left(\frac{3.25}{2.75}\right)^3 1.904E = 3.14E$$

$$f = 5.19 / 3.14E = 1.65/E, \quad l_A = 5.19(0.707) = 3.66$$

$$f l_A = \frac{6.04}{E}$$

THE EQUATIONS FOR DISPLACEMENT & ROTATION

DUE TO M & Q WILL BE CORRECTED AS FOLLOWS,

Ratio of $(f l_A \text{ new}) / (f l_A \text{ old})$

$$\left. \begin{array}{l} \text{NEW EQUATIONS} \\ \text{FOR } h_o = 3.25 \end{array} \right\} ES'_1 = ES_1 \left(\frac{6.04}{8.425} \right) = 0.717 ES_1$$

$$EO'_1 = EO_1 \left(\frac{1.65}{2.50} \right) = 0.660 EO_1$$

$$ES'_2 = 0.717 ES_2$$

$$EO'_2 = 0.660 EO_2$$

TOP HEAD (CONT)II. (CONT)

THE EFFECT DUE TO PRESSURE WILL ALSO CHANGE
AS FOLLOWS

$$E\delta_1 = \frac{2.75}{3.25} (52.166) p = 44.132 p$$

$$E\theta_1 = \frac{2.75}{3.25} (-9.105) p = -7.703 p$$

$$E\delta_2 = 0.846 (20.132) p = 17.032 p$$

$$E\theta_2 = 0.846 (-5.941) p = -5.026 p$$

THE EQUATIONS FOR (δ) AND θ FOR THE CONE BECOME

TABLE - 3

	M_1	Q_1	M_2	Q_2	p
$E\delta_1 =$	-8.256	-23.414	5.372	6.551	44.132
$E\theta_1 =$	3.966	7.970	-2.097	-3.981	-7.703
$E\delta_2 =$	6.639	10.016	-5.377	-12.330	17.032
$E\theta_2 =$	3.223	7.572	-3.214	-4.958	-5.026

THE CHARACTERISTIC EQUATIONS BECOME,

TABLE - 4

	M_1	Q_1	M_2	Q_2
(1)	(-) 4.895	(-) 4.718	2.097	3.981
(2)	5.404	55.519	(-) 5.372	(-) 6.551
(3)	3.223	7.572	(-) 3.818	(-) 3.925
(4)	6.639	10.016	(-) 3.944	(-) 16.869

=

p	G_1	G_2	W
(-) 24.353	(-) 0.209	(-) 1.793	(-) 3.189
(-) 15.145	(-) 0.642	(-) 5.505	(-) 9.790
8.421	0	0	0
(-) 15.784	0	0	0

TOP HEAD (CONT)11- (CONT)

THE SOLUTION OF THE CHARACTERISTICS EQUATIONS YIELDS THE FOLLOWING,

$$M_1 = \overset{P}{9.09(1900)} + \overset{G_1}{0.08(2700)} + \overset{G_2}{0.706(2700)} + \overset{W}{1.25(12788)} = 31700 \text{ IN-LB/IN}$$

$$Q_1 = -\overset{P}{1.15(1900)} - \overset{G_1}{0} - \overset{G_2}{0.237(2700)} - \overset{W}{0.422(12788)} = -7760 \text{ LB/IN}$$

$$M_2 = -\overset{P}{0.445(1900)} - \overset{G_1}{0} - \overset{G_2}{0} - \overset{W}{0} = -700 \text{ IN-LB/IN}$$

$$Q_2 = \overset{P}{3.93(1900)} + \overset{G_1}{0} + \overset{G_2}{0.137(2700)} + \overset{W}{0.244(12788)} = 9400 \text{ LB/IN}$$

$$N_{1V} = N_{1H} = 4.09 p = 4.09(1900) = 6100 \text{ LB/IN}$$

$$N_{2V} = N_{2H} = 2.12 p = 2.12(1900) = 3200 \text{ LB/IN}$$

THE MINUS SIGN INDICATES THAT THE DIRECTION OF THE FORCE WAS ASSUMED INCORRECTLY.

THE SHEARS AND MOMENTS ARE SHOWN CORRECTLY ON THE NEXT PAGE.

TOP HEAD ANALYSIS (CONT)

FREE BODY DIAGRAM

FOR MECHANICAL LOADS

FOR NEW THICKNESS
OF CONICAL SECTION

* NEW MINIMUM THICKNESS

TOP HEAD (CONT)II STRESSES (CONT)(A) STRESSES IN CONE AT INTERFACE JUNCTION ①AXIAL

$$\sigma_x = \frac{N_x}{h} \pm \frac{6 M_1}{h^2} \quad , \quad N_x = N_H S_{WH} + N_W C_{WD} + Q_1 S_{WD} = 14100 \text{ LBS/IN}$$

$$= \frac{14100}{3.38} \pm \frac{6(31700)}{11.43} = 4170 \pm 16600 = 20770 \text{ LBS/IN}^2 \text{ (OUTER)}$$

$$- 12430 \text{ LBS/IN}^2 \text{ (INNER)}$$

SURFACECIRCUMFERENTIAL

$$\sigma_\theta = \frac{E \epsilon_1}{r_m} \pm \nu \frac{6 M_1}{h^2}$$

FROM TABLE-3

$$E \epsilon_1 = 44000 \quad ,$$

$$r_m = 12.12''$$

$$\sigma_\theta = \frac{44000}{12.12} \pm 0.3(16600)$$

$$= 3600 \pm 5000 = 8600 \text{ LBS/IN}^2 \text{ (OUTER)}$$

$$- 1400 \text{ LBS/IN}^2 \text{ (INNER)}$$

RADIAL $\sigma_r = 0 \text{ OUTER, } -1500 \text{ INNER}$

THE MAXIMUM STRESS INTENSITY OCCURS AT THE OUTER SURFACE $S_{MAX} = \sigma_x - \sigma_r = 20770 - 0 = 20770 \text{ LBS/IN}^2$.

THE ALLOWABLE $S_p = 21000 \text{ LBS/IN}^2$ (FOR 347 SS AT 600°F).

THEREFORE, THE CONDITION IS ACCEPTABLE, SINCE

$$\text{ALLOWABLE } S_p > S_{ACTUAL}$$

TOP HEAD (CONT)II (CONT)(B) STRESSES IN RING

$$\text{MOMENT (M}_{c.g.}) = 2.97(W) + 0.17(G_1) + 1.44(G_2) - 0.749 M_1 \\ + 2.30(Q_1) + 13.4(p)$$

$$M_{c.g.} = 2.97(12788) + 0.17(2700) + 1.44(2700) - 0.749(31700) \\ - 2.30(7760) + 13.4(1500) = 15720 \text{ in.-lb/in}$$

$$E\delta_{c.g.} = -3.349 Q_1 + (21.846 - 13.684)p \\ = -3.349(7760) + 8.162(1500) = 13745$$

$$\sigma_{\theta} = \pm M_{c.g.} \frac{r_{im} \left(\frac{c}{I} \right) + \frac{E\delta_{c.g.}}{r_{im}}}$$

AT UPPER SURFACE

$$\sigma_{\theta} = \frac{15720(3.07)(15.75)}{200} + \frac{13745}{15.75} \\ = 3800 + 900 = 4700 \text{ lbs/in}^2$$

AT LOWER SURFACE

$$\sigma_{\theta} = -3800 \left(\frac{3.56}{3.07} \right) + 900 = -4400 + 900 = -3500 \text{ lbs/in}^2$$

$$\sigma_r \text{ (AT INSIDE SURFACE)} = -1500 \text{ lbs/in}^2$$

TOP HEAD (CONT)II-(B) CONT.

THE MAXIMUM STRESS INTENSITY OCCURS AT
INNER EDGE OF UPPER SURFACE,

$$S_{MAX} = \sigma_{\theta} - \sigma_r = 4700 - (-1500) = 6200 \text{ LB}_3/\text{IN}^2$$

$$\text{ALLOWABLE } S_p = 21000 \text{ LB}_3/\text{IN}^2 \quad (\text{OK})$$

(C) STRESSES IN FLAT PLATE AT JUNCTION ②

$$M_{JA} = M_2 + \frac{4.75}{2}(Q_2 + J_{2H}) = 700 + \frac{4.75}{2}(9400 + 3200) = 30560 \frac{\text{IN-LB}}{\text{IN}}$$

$$F = Q_2 + J_{2H} = 12600 \text{ LB}_3/\text{IN}$$

THE STRESS (σ) IS,

$$\sigma = \frac{F}{h} \pm \frac{6 M_{JA}}{h^2} = \frac{12600}{4.75} \pm \frac{6(30560)}{(4.75)^2}$$

$$= 2600 \pm 8100 = 10700 \frac{\text{LB}_3}{\text{IN}^2} \text{ LOWER SURFACE}$$

$$-5500 \frac{\text{LB}_3}{\text{IN}^2} \text{ UPPER SURFACE}$$

$$\text{NORMAL STRESS } \sigma_H = -1500 \text{ LB}_3/\text{IN}^2 \text{ LOWER}$$

$$0 \text{ UPPER}$$

$$\text{MAX. STRESS INTENSITY EQUALS } 10700 - (-1500) = 12200 \text{ LB}_3/\text{IN}^2$$

$$(\text{OK}) \text{ SINCE ALLOWABLE } S_p = 21000 \text{ LB}_3/\text{IN}^2$$

REFERENCES

- #1. "STRESS ANALYSIS OF REACTOR PRESSURE VESSEL" MND-M-1864, DECEMBER 1961
- #2. "STRENGTH OF MATERIALS" - PART II - BY TIMOSHENKO
2ND EDITION.
- #3. "FORMULAS FOR STRESS AND STRAIN" - BY ROARK
3RD EDITION
- #4. "THEORY OF ELASTICITY" BY A.E.H. LOVE
- #5. "THEORY OF PLATES AND SHELLS" - BY TIMOSHENKO - 1ST EDITION
- #6. "ANALYSIS OF SHORT THIN AXISYMMETRICAL SHELLS
UNDER AXISYMMETRICAL EDGE LOADING" BY
G. HORVAY, ETAL, JOURNAL OF APP. MECH. - MARCH 1956
- #7. "THEORETICAL ELASTIC STRESS DISTRIBUTIONS ARISING
FROM DISCONTINUITIES AND EDGE LOADS IN SEVERAL
SHELL TYPE STRUCTURES" TECH. REPT. R-103 BY
JOHNS & ORANGE, LEWIS RESEARCH CENTER, CLEVE.,
OHIO.
- #8. ASME BOILER AND PRESSURE VESSEL CODE
SECTION VII - UNFIRED PRESSURE VESSELS

W. O. NO. 2288-12

BURNS AND ROE, INC.
NEW YORK, N. Y.

DATE: 8/66

DRAWING NO. _____

BY: P. J. DOBBS

TITLE: PM-3A STRESS ANALYSIS - REACTOR PRESSURE VESSEL SHEET 46 OF 46

REFERENCES (CONT)

- *9 "TENTATIVE STRUCTURAL DESIGN BASIS FOR REACTOR PRESSURE VESSELS AND DIRECTLY ASSOCIATED COMPONENTS" U.S. DEPT. OF COMMERCE PB-151987 (NAVY CODE) 1 DEC. 1958.

FORM U-1 MANUFACTURERS' DATA REPORT FOR UNFIRED PRESSURE VESSELS

As required by the Provisions of the ASME Code Rules

1. Manufactured by A. O. Smith Corporation, Milwaukee, Wisconsin
(Name and address of Manufacturer)

2. Manufactured for The Martin Company, Baltimore, Maryland
(Name and address of Purchaser)

MV 80017A-2 Antarctica

Type Vert. Kind Tank Vessel No. () (Mfrs. Serial) (State & State No) Natl. Bd. No. 9956 Yr. Built 1961
(Horiz or Vert) (Tank, Jacketed, Heat Exch)

Items 4-9 incl. to be completed for single wall vessels (such as air tanks), jackets of jacketed vessels, or shells of heat exchangers.

4. SHELL: Material SA240 T347 T.S. 75,000 Nominal 2 3/16 Corrosion 11 5/8 3 15/16
(Kind and Spec. No.) (Fig. or F.B. & Spec. Min T.S.) Thickness In. Allowance In. Diam. 2 Ft. In. Length 9 Ft. In.

5. SEAMS: Long Welded Dbl. Butt T. No X.R. Complete Sectioned No Efficiency 100 %
(Welded, Dbl, Single, Lap, Butt) (Yes or No) (Spot or Complete) (Yes or No)

Girth Welded Dbl. Butt T. No X.R. Complete Sectioned No No. of Courses 1

6. HEADS (a) Material SA182 Gr F347 T.S. 75,000 (b) Material SA 240 Type 347 T.S. 75,000
Location Thickness Crown Radius Knuckle Radius Elliptical Ratio Conical Apex Angle Hemispherical Radius Flat Diameter Side to Pressure
(Top, bottom, ends) (Convex or Concave)

(a) Top 2 3/4 13 3/4 Concave

(b) Bottom 2 2:1 Concave

If removable, bolts used A.L. A286-146, 000-2 1/2"-20 Other fastening _____
(Material, Spec. No., T.S., Size, Number) (Describe or Attach Sketch)

7. STAYBOLTS: (Material) _____ If hollow (Size of Hole) Attachment (Threaded, Welded) Pitch (Horiz.) X (Vert.) Diam. (Nominal)

8. JACKET CLOSURE: _____
(Describe as ogee & weld, bar, etc. If bar, give dimensions, if bolted, describe or sketch)

9. Constructed for max. allowable working press. 1500 psi at max. temp. 600 °F. less than -20° Min. temp. (when Hydrostatic Pneumatic or Test Press 2250 psi.
°F. Combination }
Test Press

Items 10 and 11 to be completed for tube sections.

10. TUBE SHEETS: Stationary. Material (Kind & Spec. No.) Diam. In. Thickness In. Attachment (Welded, Bolted)

Floating. Material (Kind & Spec. No.) Diam. In. Thickness In. Attachment

11. TUBES: Material (Kind & Spec. No.) O.D. In. Thickness In. or Gage Number Type (Straight or U)

Items 12-15 incl. to be completed for inner chambers of jacketed vessels, or channels of heat exchangers.

12. SHELL Material (Kind and Spec. No.) T.S. _____ Nominal Thickness _____ Corrosion _____ In. Allowance _____ In. Diam. _____ Ft. _____ In. Length _____ Ft. _____ In.

13. SEAMS: Long (Welded, Dbl., Single, Lap, Butt) H.T. _____ X.R. _____ Sectioned _____ Efficiency _____ %
(Yes or No) (Spot or Complete) (Yes or No)

Girth H.T. _____ X.R. _____ Sectioned _____ No. of courses _____

14. HEADS (a) Material _____ T.S. _____ (b) Material _____ T.S. _____ (c) Material _____ T.S. _____
Location Thickness Crown Radius Knuckle Radius Elliptical Ratio Conical Apex Angle Hemispherical Radius Flat Diameter Side to Pressure
(Top, bottom, ends) (Convex or Concave)

(b) Channel _____

(c) Floating _____

If removable, bolts used (a) _____ (b) _____
(Material, Spec. No., T.S., Size, Number)

(c) _____ Other fastening _____
(Describe or Attach Sketch)

15. Constructed for max. allowable working press. _____ psi at max. temp. _____ °F. less than -20° Min. temp. (when Hydrostatic Pneumatic or Test Press _____ psi.
°F. Combination }
Test Press

Items below to be completed for all vessels where applicable.

16. SAFETY VALVE OUTLETS: Number _____ Size _____ Location _____

17. NOZZLES

Purpose (Inlet, Outlet, Drain)	Number	Diam. or Size	Type	Material	Thickness	Reinforcement Material	How Attached
Bundle Port	1	4 1/8"					Drilled Hole
Actuator Ports	6	1 3/8"					Drilled Hole
Inlet	1	6"	Stub	(ASTM A182 Gr. F347, SA403)	.432		Welded
Outlet	1	6"	Stub	Gr. WP347, SA312 Type 347	.432		Welded

¹ If postweld heat-treated.

² List under remarks other internal or external pressures with coincident temperature when applicable.

FORM U-1 (back)

18. INSPECTION Manholes, No. _____ Size _____ Location _____
 OPENINGS: Handholes, No. _____ Size _____ Location _____
 Threaded, No. _____ Size _____ Location _____
 19. SUPPORTS: Skirt Yes Lugs _____ Legs _____ Other _____ Attached Welded to
 (Yes or No) (Number) (Number) (Describe) bottom head
 20. REMARKS: Reactor Pressure Vessel - PM3A Nuclear Power Plant

(Brief description of purpose of the vessel, as Air Tank, After Cooler, Jacketed Cooker, etc. State contents of each part.)

We certify that the statements made in this report are correct and that all details of design, material, construction, and workmanship of this vessel conform to the ASME Code for Unfired Pressure Vessels.

Date April 20, 19 61 Signed A. O. Smith Corporation By s/John L. Iverson
 (Manufacturer)

Certificate of Authorization Expires December 31, 1961

Inspection Agency's Serial No. 1CC 16553

CERTIFICATE OF SHOP INSPECTION

VESSEL MADE BY A. O. Smith Corporation at Milwaukee, Wisconsin

I, the undersigned, holding a valid commission issued by the National Board of Boiler and Pressure Vessel Inspectors and/or the State of National Board State and employed by Maryland Casualty Company of Baltimore, Md.

have inspected the pressure vessel described in this manufacturer's data report on April 20, 1961, and state that to the best of my knowledge and belief, the manufacturer has constructed this pressure vessel in accordance with the applicable sections of the ASME Boiler and Pressure Vessel Code.

By signing this certificate neither the Inspector nor his employer makes any warranty, expressed or implied, concerning the pressure vessel described in this manufacturer's data report. Furthermore, neither the Inspector nor his employer shall be liable in any manner for any personal injury or property damage or a loss of any kind arising from or connected with this inspection.

Date April 20, 19 61

s/Calvin J. Phom

Inspectors Signature

Commissions

3671

Nat'l Board or State and No.

CERTIFICATE OF FIELD ASSEMBLY INSPECTION

I, the undersigned, holding a valid commission issued by the National Board of Boiler and Pressure Vessel Inspectors and/or the State of _____ and employed by _____ of _____

have compared the statements in this manufacturer's data report with the described pressure vessel and state that parts referred to as data items _____, not included in the certificate of shop inspection have been inspected by me and that to the best of my knowledge and belief

the manufacturer has constructed and assembled this pressure vessel in accordance with the applicable sections of the ASME Boiler and Pressure Vessel Code. The described vessel was inspected and subjected to a hydrostatic test of _____ psi.

By signing this certificate neither the Inspector nor his employer makes any warranty, expressed or implied, concerning the pressure vessel described in this manufacturer's data report. Furthermore, neither the Inspector nor his employer shall be liable in any manner for any personal injury or property damage or a loss of any kind arising from or connected with this inspection.

Date _____ 19 _____

Inspector's Signature

Commissions

Nat'l Board or State and No.

W. O. NO. 2283-12

BURNS AND ROE, INC.
NEW YORK, N. Y.

DATE: 2/66

DRAWING NO. _____

BY: M. CIDEAN

TITLE: _____

SHEET 1 OF 21

PM - 3A STRESS ANALYSIS

STEAM GENERATOR CODE CALCULATIONS

ITEM 2.0

SUMMARY

THE STEAM GENERATOR WAS ANALYZED IN ACCORDANCE WITH ASME BOILER AND PRESSURE VESSEL CODE SECTION VIII UNFIRED PRESSURE VESSELS.

THE SPECIFIED DESIGN CONDITIONS FOR THE PRIMARY SIDE ARE INTERNAL PRESSURE OF 1500 PSIG AND TEMPERATURE OF 600°F. THE SECONDARY SIDE WAS DESIGNED FOR 600 PSIG INTERNAL PRESSURE AND TEMPERATURE OF 600°F.

THE ACTUAL THICKNESS OF THE VARIOUS STEAM GENERATOR COMPONENTS (SHELL, HEAD AND TUBE SHEET) IS IN EACH CASE GREATER THAN THE CALCULATED MINIMUM THICKNESS REQUIRED BY CODE AND THEREFORE ACCEPTABLE.

REINFORCEMENT CALCULATIONS CLEARLY INDICATE THAT REQUIREMENTS OF THE CODE HAVE BEEN SATISFIED.

REINFORCEMENT OF THE PRIMARY MANHOLE, SECONDARY MANHOLE AND STEAM OUTLET NOZZLE WERE NOT CALCULATED DUE TO LACK OF DETAIL DRAWINGS OF THE ABOVE ITEMS.

INTRODUCTION

THE STEAM GENERATOR IS A VERTICAL U-TUBE WATER TO WATER HEAT EXCHANGER BUILT IN ACCORDANCE WITH THE ASME CODE FOR UNFIRED PRESSURE VESSELS, SECTION VIII. THE VESSEL IS $31\frac{1}{4}$ " IN DIAMETER BY 17' LONG. THE SECONDARY SIDE SHELL IS FABRICATED FROM $\frac{5}{8}$ " THICK CARBON STEEL PLATE WELDED TO A $\frac{7}{16}$ " THICK HEMISPHERICAL HEAD OF THE SAME MATERIAL. THE BASIC SECONDARY SIDE CONNECTIONS CONSIST OF A 6" STEAM OUTLET NOZZLE IN THE TOP HEAD AND A 15" I.D. MANHOLE IN THE SHELL. IN ADDITION THERE ARE A FEEDWATER NOZZLE, BLOWDOWN CONNECTIONS, LIQUID LEVEL CONTROL AND INDICATOR CONNECTIONS.

THE PRIMARY INLET AND OUTLET NOZZLES ARE LOCATED IN THE BOTTOM HEAD OF THE VESSEL. THE HEAD IS DIVIDED IN HALF INTERNALLY BY THE PARTITION PLATE WHICH SEPERATES THE INLET AND OUTLET NOZZLE. THE HEAD IS MADE OF 1" THICK CARBON STEEL CLAD WITH $\frac{1}{4}$ " OF STAINLESS STEEL.

THE PRIMARY AND SECONDARY SIDE OF THE STEAM GENERATOR ARE SEPERATED BY A $5\frac{5}{16}$ " THICK TUBE SHEET CLAD WITH INCONEL ON THE PRIMARY SIDE.

SECONDARY SIDE SHELL:

P = DESIGN PRESSURE, 600 PSI

R = RADIUS OF SHELL, 15.06 IN.

S = ALLOWABLE MEMBRANE STRESS, 17,500 PSI.

E = JOINT EFFICIENCY, 100%

$$t = \frac{PR}{SE - 0.6P} \quad (\text{PAR. UG-27 REF. \#2})$$

$$t = \frac{600(15.06)}{(17500) - 0.6(600)}$$

$$t = \frac{9036}{17140} = 0.525$$

WITH 1/16" CORR. ALLOWANCE, MINIMUM THICKNESS
REQUIRED = 0.525 + 0.063 = 0.588 IN.

$$t_{\text{ACT}} = 0.625 \text{ IN.} > t_{\text{REQ.}} = 0.588 \text{ IN.}$$

SHELL SIDE HAND HOLE:

$$P = 600 \text{ PSIG.}$$

$$T = 600^\circ \text{F}$$

$$\text{GASKET S.S. FLEXITALLIC } 5 \frac{3}{16} \text{ I.D.} \times 6 \frac{3}{16} \text{ O.D.}$$

$$m = 3.00$$

$$y = 4500 \text{ PSI (TABLE UA-49.1 REF. #2)}$$

$$b_o = \frac{N}{2} = \frac{0.50}{2} = 0.25" \quad (b = b_o \text{ WHEN } b_o \leq 0.25) \text{ (TABLE UA-49.2)}$$

$$H = 0.7854 G^2 P = 0.7854 (5.6875)^2 (600) = 15,200 \text{ #}$$

$$H_p = 2b \pi G m P = 2(0.25)(3.14)(5.6875)(3)(600) = 16,100 \text{ #}$$

$$W_{M1} = 31,300$$

$$W_b = \frac{A_b (20000)}{2} + \frac{A_f (20000)}{2}$$

$$= \frac{A_b (20000)}{2} + \frac{W_{M1} (20000)}{20000 \cdot 2}$$

$$= \frac{A_b (20000) + W_{M1}}{2}$$

A_b = AREA OF BOLTS IN FLANGE

A_f = MINIMUM REQUIRED BOLT AREA

BOLTS: 8 - 5/8" (MINIMUM SECTION AREA = 0.202 IN.²)

$$A_{\text{BOLTS}} = 8(0.202) = 1.616 \text{ IN.}^2$$

$$\text{ALLOWABLE STRESS} = 20,000 \text{ PSI @ } 600^\circ\text{F}$$

$$W_b = 1.616(20,000) = 32,300 \text{ \#}$$

$$W = \frac{32300 + 31300}{2} = 31,800 \text{ \#}$$

SHELL SIDE HANDHOLE COVER:

$$t_{\text{reqd.}} = d \sqrt{\frac{CP}{S} + \frac{1.78 W h_g}{S d^3}} \quad (\text{PAGE 19 REF. \#1})$$

$$t_{\text{reqd}} = 5.625 \sqrt{\frac{.3(600)}{17500} + \frac{1.78(.843)(31800)}{17500(5.625)^3}} = 0.90''$$

0.062 CORR.
0.962"

$$t_{\text{AVAIL.}} = 1.375'' > 0.962''$$

CHECK GASKET WIDTH:

$$N_{\text{MW.}} = \frac{W_b}{2\pi y G} = \frac{32300}{2\pi (4500)(5.6875)} = 0.201'' < N_{\text{ACT}} = 0.500''$$

BOLT SPACING:

$$L = \frac{3.14(7.375)}{8} = 2.90'' > L_{\text{MIN.}} = 1.50'' (\text{TABLE 1, PAGE 27 REF. \#3})$$

$$\text{BOLT SPACING (MAX)} = 2a + \frac{6t}{(m+0.5)} \quad (\text{PAGE 12 REF. \#3})$$

$$= 2\left(\frac{5}{8}\right) + \frac{6(1.375)}{3+0.5} = 3.60''$$

SECONDARY HEAD:

$$t_{reqd} = \frac{PL}{2SE - 0.2P} + c \quad (\text{PAR. UG-32 REF. \#2})$$

P = DESIGN PRESSURE, 600 PSI.

L = RADIUS OF SHELL, 15.06 IN.

S = ALLOWABLE MEMBRANE STRESS, 17500 PSI.

E = JOINT EFFICIENCY, 100%

$$t = \frac{600(15.06)}{2(17500) - 0.2(600)} = 0.258 \text{ IN.}$$

WITH $\frac{1}{16}$ " CORR. ALLOWANCE, MINIMUM THICKNESS
REQUIRED = $0.258" + 0.063" = 0.321"$

$$t_{act.} = 0.438" > t_{reqd} = 0.321"$$

W. O. NO. 2258-12BURNS AND ROE, INC.
NEW YORK, N. Y.DATE: 2/44

DRAWING NO. _____

BY: M. GIDEN

TITLE: _____

SHEET 8 OF 21CHANNEL:

P = DESIGN PRESSURE, 1500 PSIG.

L = RADIUS OF HEAD, 15 IN.

E = JOINT EFFICIENCY, 100%

S = ALLOWABLE MEMBRANE STRESS, 17,500 P.S.I.

$$t_{req.} = \frac{PL}{2SE - 0.2P} \quad (\text{PAR. UG-32 REF. \#2})$$

$$t_{req.} = \frac{1500(15)}{2(17500) - 0.2(1500)} = 0.650"$$

CARBON STEEL THICKNESS OF 1" > 0.650" REQD.

PRIMARY HANDHOLE:

$$P = 1500 \text{ PSIG} \quad T = 600^\circ \text{F}$$

$$G = 6.125"$$

$$h_g = 1.5'$$

$$\text{FLEXITALLIC GASKET} \quad m = 1.00 \quad y = 4500 \text{ PSI (STAINLESS STEEL)}$$

$$\text{TABLE UA-4) 1 PAGE 183 REF. 2}$$

$$N = .5625"$$

$$b_o = N/2 = \frac{.5625}{2} = .2812 > 1/4$$

$$b = \frac{\sqrt{.2812}}{2} = .265$$

$$H = .785 G^2 P = .785 (6.125)^2 (1500) = 44200 \#$$

$$H_p = 2 b \pi \phi m P = 2 (.265) (3.14) (1.0) (3) (1500) = \frac{45800 \#}{90000 \#}$$

W. O. NO. 2288-12

BUENS AND ROE, INC.

DATE: 9/66

DRAWING NO. _____

NEW YORK, N. Y.

BY M. GIDENTITLE: PM-3A STEAM GENERATORSHEET 10 OF 21PRIMARY HANDHOLE COVER:

$$t = d \sqrt{\frac{CP}{S} + \frac{1.78 W h_s}{S d^3}}$$

PAGE 19 REF. 1

$$t = 6.125 \sqrt{\frac{.30(1500)}{17500} + \frac{1.78(95250)(1.25)}{17500(6.125)^3}} = 6.125 \sqrt{0.0257 + 0.0527}$$

$$t = 6.125(.282) = 1.72 \text{ IN.}$$

$$t_{ACT.} = 1.875" > t_{req.} = 1.72 \text{ IN.}$$

W.O. NO. 2289-12BURNS AND ROE, INC.
NEW YORK, N. Y.DATE: 9/66

DRAWING NO. _____

BY: M. GIDENTITLE: PM-3A STEAM GENERATORSHEET 11 OF 21PRIMARY HANDHOLE BOLTS:12 - $7/8"$ BOLT:MINIMUM ROOT AREA = 0.419 IN.²

FOR SA-193 B. BOLTS, ALLOWABLE STRESS = 20,000 PSI.

$$W_{\text{BOLTS}} = 12(0.419)(20,000) = 100,500 \#$$

GASKET WIDTH:

$$W_{\text{GASKET}} = \frac{W_{\text{BOLTS}}}{2\pi G_y} = \frac{100,500}{2\pi \left(\frac{4500}{173000} \right) (6.125)} = 0.581 \text{ USE } 5/8" \text{ GASKET}$$

BOLT SPACING:

$$B_s = 2(4.312) \sin 15^\circ = 2.23 \text{ IN.}$$

MINIMUM SPACING = $2 \frac{1}{16}"$

W. O. NO. 2288-13

BURNS AND ROE, INC.
NEW YORK, N. Y.

DATE: 9/66

DRAWING NO. _____

BY: M. GIDON

TITLE: STEAM GENERATOR

SHEET 12 OF 21

TUBE SHEET: t_r = REQD THICKNESS OF SHELL FOR PRESSURE = 0.65" (PAGE 8) t_s = ACTUAL THICKNESS OF SHELL = 1.0"

$$m = \frac{t_r}{t_s} = \frac{0.65}{1.0} = 0.65$$

FROM FIG. UG-34, PAGE 21, FIG. f $C = \frac{1}{2}m = .325$

$$t_{min.} = d \sqrt{\frac{CP}{S}} = 30 \sqrt{\frac{.325(1500)}{17500}} = 5.0"$$

$$t_{ACT.} = 5.3125" + .375" \text{ S.S. CLAD}$$

PRESSURE STRESS: (DESIGN)PRIMARY SIDE:

$$P = 1500 \text{ PSIG.}$$

$$T = 600^{\circ}\text{F}$$

SECONDARY SIDE:

$$P = 600 \text{ PSIG.}$$

$$T = 500^{\circ}\text{F}$$

CHANNEL:

$$P = \frac{2SEt}{R - 0.4t} \quad \text{PAR. UG.-27}$$

$$S = 17500 \text{ PSI (ALLOWABLE)}$$

$$t = 1.0''$$

$$R = 15''$$

$$P = 1500 \text{ PSIG. DESIGN}$$

$$P = 1300 \text{ PSI OPERATING}$$

$$S = \frac{P(R - 0.4t)}{2Et} = \frac{1500(15 - 0.4)}{2(1)} = 10,950 \text{ PSI. } \checkmark S_{ALL} = 17,500 \text{ PSI.}$$
$$= 9480 \text{ PSI (OPERATING)}$$

SHELL: (SECONDARY SIDE)

$$P = \frac{SEt}{R + 0.6t} \quad \therefore S = \frac{P(R + 0.6t)}{Et} \quad P = 300 \text{ PSI. OPERATING}$$

$$S = 17,500 \text{ PSI (ALLOWABLE)}$$

$$t = 0.625 - 0.0625 \text{ COR.} = 0.5625''$$

$$R = 15.063''$$

$$E = \text{ALLOWABLE JOINT EFFICIENCY} = 100\%$$

$$S = \frac{600(15.063 + 0.337)}{0.5625} = 16,400 \text{ PSI. } < S_{ALLOWABLE} = 17,500 \text{ PSI.}$$
$$= 8,200 \text{ PSI OPERATING}$$

SECONDARY HEAD:

$$P = 600 \text{ PSI.}$$

$$t = 0.4375 - 0.0625" = 0.375"$$

$$R = 15"$$

$$S = 17500 \text{ PSI. ALLOWABLE}$$

$$E = 100 \%$$

$$P = \frac{2Set}{R + 0.2t} \quad \therefore S = \frac{P(R + 0.2t)}{2Et}$$

$$S = \frac{600(15 + 0.075)}{0.75} = 12,000 \text{ PSI} < S_{\text{ALLOWABLE}} = 17,500 \text{ PSI.}$$

$$= 6,000 \text{ PSI (OPERATING)}$$

TUBE SHEET:

$$t_{\min.} = d \sqrt{\frac{cP}{S}}$$

$$P = 1500 \text{ PSI.}$$

$$d = 30"$$

$$c = 0.325 \text{ (PAGE 12)}$$

$$t = 5.3125"$$

$$S = \frac{d^2 c P}{t^2} = \frac{(30)^2 (0.325)(1500)}{(5.3125)^2} = 15,600 \text{ PSI} < S_{\text{ALL.}} = 17,500 \text{ PSI}$$

$$= 13,500 \text{ PSI OPERATING}$$

W.O. NO. 2288-12

DRAWING NO. 497C-392

TITLE:

BURNS AND ROE, INC.
NEW YORK, N. Y.

DATE: 9/66

BY: M. GIDEN

SHEET 15 OF 21

FEDERATER NOZZLE REINFORCEMENT

15" R TO STEAM GENERATOR

MATERIAL ASTM A-105 GR. II

T.S. = 65,000 PSI.

DESIGN PRESSURE 600 PSI

DESIGN TEMP. 600°F

NOZZLE THICKNESS

$$t_{reqd} = \frac{PR}{SE - 0.6P} + CORR. \quad (PAR. UG-27 REF. \# 2)$$

P = DESIGN PRESSURE, 600 PSI

R = NOZZLE RADIUS, 2 IN.

S = ALLOWABLE MEMBRANE STRESS = 17,500 PSI.

E = JOINT EFFICIENCY, 100%

$$t_{reqd} = \frac{600(2)}{17500 - 0.6(600)} + 0.0625 = 0.07 + 0.0625 = 0.1325"$$

$$AREA OF REINF. REQD, A = d t_r = 4(.525) = 2.10 \text{ IN.}^2$$

$$SHELL AVAILABLE FOR REINF. A_1 = d(t - t_r) = 4(0.625^{0.10} - 0.525) = 0.40 \text{ IN.}^2$$

$$NOZZLE \quad " \quad " \quad A_2 = 5t(t_n - t_{rn}) = 5(.5625)(.8125 - .07) = 2.08 \text{ IN.}^2$$

$$AREA OF WELD \quad A_4 = 2 \times \frac{1}{2} \times \left(\frac{3}{8}\right)^2 = 0.141 \text{ IN.}^2$$

$$A_1 + A_2 + A_4 = 0.40 + 2.08 + 0.141 = 2.621 \text{ IN.}^2 > 2.22 \text{ IN.}^2 \text{ REQD.}$$

W. O. NO. 2388-12

DRAWING NO. 442 ISSI

TITLE: _____

DURNS AND ROE, INC.
NEW YORK, N. Y.

DATE: 9/66

BY: M. GIDEN

SHEET 17 OF 21

PRIMARY NOZZLE

PRIMARY NOZZLE

$$t = \frac{PR}{SE - 0.6P} \quad (\text{PAR. UG-27, PAGE 10 REF. #2})$$

P = DESIGN PRESSURE, 1500 PSI.

R = RADIUS OF SHELL, $2\frac{3}{4}$ " IN.

E = EFFICIENCY, 100%

S = ALLOWABLE MEMBRANE STRESS, 17500 PSI.

$$t = \frac{1500(2.75)}{17500 - 900} = \frac{4130}{16600} = 0.248"$$

NOZZLE REINFORCEMENT (FIG. UA-280 REF. #2)AREA OF REINFORCEMENT REQD, $A = dt_r = 5.5(0.65) = 3.57 \text{ IN.}^2$ SHELL AVAILABLE FOR REINF. $A_1 = d(t - t_r) = 5.5(1 - .65) = 1.92 \text{ IN.}^2$ NOZZLE " " " $A_2 = st_n(t_n - t_{rn}) = 5(.625)(.625 - .248) = 1.18 \text{ IN.}^2$ AREA OF WELD $A_4 = 2 \times \frac{1}{2} \times \frac{1}{2} \times \frac{3}{4} = 0.375 \text{ IN.}^2$

$$A_1 + A_2 + A_4 = 1.92 + 1.18 + 0.375 = 3.475 \text{ IN.}^2$$

REINFORCEMENT OF SHELL SIDE HANDHOLE

AREA OF REINF. REQD. $A = d t_r = 5.25(0.525) = 2.75 \text{ in}^2$

NOZZLE THICKNESS:

$$t_{\text{reqd}} = \frac{PR}{SE - 0.6P} + \text{CORR. (PAR. UG-27 REF. #2)}$$

P = DESIGN PRESSURE, 600 PSI.

R = NOZZLE RADIUS, $2\frac{5}{8}$ IN.

S = ALLOWABLE MEMBRANE STRESS, 17500 PSI.

E = JOINT EFFICIENCY, 100%

$$t_{\text{reqd}} = \frac{600(2.625)}{17500 - 0.6(600)} + 0.0625 = \frac{1570}{17140} + 0.0625 = 0.0916 + 0.0625$$

$$t_{\text{reqd}} = 0.153"$$

W.O. NO. 2282-12BURNS AND ROE, INC.
NEW YORK, N. Y.DATE: 9/66

DRAWING NO. _____

BY: M. GIDENTITLE: _____ SHEET 34 OF 21SHELL AVAILABLE FOR REINF. $A_1 = d(t - t_r) = 5.25(.562 - .525) = 0.194 \text{ in}^2$ PAD AVAILABLE FOR REINF. $A_3 = 2(8.875 - 5.34)1.375 = 4.85 \text{ in}^2$ AREA OF WELDS, $A_4 = 2 \times \frac{1}{2} \times (3/8)^2 = 0.141 \text{ in}^2$

$$A_1 + A_3 + A_4 = 0.194 + 4.85 + 0.141 = 5.185 \text{ in}^2 > 2.75 \text{ in}^2$$

W. O. NO. 2288-1L

BURNS AND ROE, INC.
NEW YORK, N. Y.

DATE: 9/66

DRAWING NO. _____

BY: M. GIDEN

TITLE: _____

SHEET 21 OF 21

REFERENCES

1. TECHNICAL MANUAL 1440-CSS, VERTICAL STEAM
GENERATOR FOR THE MARTIN CO.
PM-3A NUCLEAR POWER PLANT
WESTINGHOUSE ELECTRIC CORP, ESSINGTON, Pa.
2. ASME BOILER & PRESSURE VESSEL CODE, SECTION VIII
UNFIRED PRESSURE VESSELS.
3. MODERN FLANGE DESIGN, BULLETIN #502 1964 ED.
TAYLOR FORGE

FORM U-1 MANUFACTURERS' DATA REPORT FOR UNFIRED PRESSURE VESSELS

As required by the Provisions of the ASME Code Rules

1. Manufactured by Westinghouse Electric Corporation, Lester, Pa.
(Name and address of Manufacturer)

2. Manufactured for The Martin Co. Baltimore 3, Maryland
(Name and address of Purchaser)

Type Vertical Kind Steam Generator Vessel No. 16-A-9037-1 (None) Natl. Bd. No. 1960-61
(Horiz. or Vert.) (Tank, Jacketed, Heat Exch.) (Mfrs. Serial) (State & State No)

Items 4-9 incl. to be completed for single wall vessels (such as air tanks), jackets of jacketed vessels, or shells of heat exchangers.

4. SHELL: Material ASTM A-212 Gr. B T.S. 70,000 Nominal Thickness 5/8 In. Allowance 1/16 In. Diam. 2 Ft. 6 In. Length 13 Ft. 4-9/16 In.
(Kind and Spec. No.) (Fig. or F.B. & Spec. Min. T.S.)

5. SEAMS: Long Dbl. Butt Welded H.T. Yes X.R. Compl. Sectioned No Efficiency 100 %
(Welded, Dbl., Single, Lap, Butt) (Yes or No) (Spot or Complete) (Yes or No)

If riveted describe seams fully on reverse side of form.

6. HEADS (a) Material Girth Single Butt-Back-Up Strip-Yes T.S. 70,000 (b) Material Yes Sectioned No No. of Courses One
(Top, bottom, ends) Location Thickness Crown Radius Knuckle Radius Elliptical Ratio Conical Apex Angle Hemispherical Radius Flat Diameter Side to Pressure (Convex or Concave)

(a) Top 7/16 15 In. Concave

(b) _____

If removable, bolts used _____ Other fastening _____
(Material, Spec. No., T.S., Size, Number) (Describe or Attach Sketch)

7. STAYBOLTS: _____ If hollow _____ Attachment _____ Pitch _____ X _____ Diam. _____
(Material) (Size of Hole) (Threaded, Welded) (Horiz.) (Vert.) (Nominal)

8. JACKET CLOSURE: _____
(Describe as ogee & weld, bar, etc. If bar, give dimensions, if bolted, describe or sketch)

9. Constructed for max. allowable working press. 600 psi at max. temp. 600 °F. Min. temp. (when less than -20°) _____ °F. Hydrostatic Pneumatic or Test Press 900 psi.
Combination

Items 10 and 11 to be completed for tube sections.

10. TUBE SHEETS: Stationary. Material A-266 Gr. II, 65000 Diam. 30 In. Thickness 5-5/16 In. Attachment welded
(Kind & Spec. No.) (Subject to Pressure) (Welded, Bolted)

Floating. Material _____ Diam. _____ In. Thickness _____ In. Attachment _____
(Kind & Spec. No.)

11. TUBES: Material Inconel ASTM B-163 O.D. 1/2 In. Thickness 19 BWG Inches or Gage Number 465 U's Type U
(Kind & Spec. No.) (Straight or U)

Items 12-15 incl. to be completed for inner chambers of jacketed vessels, or channels of heat exchangers.

12. SHELL Material _____ T.S. _____ Nominal Thickness _____ Corrosion _____ In. Allowance _____ In. Diam. _____ Ft. _____ In. Length _____ Ft. _____ In.
(Kind and Spec. No.) (Fig. or F.B. & Spec. Min. T.S.)

13. SEAMS: Long _____ H.T. _____ X.R. _____ Sectioned _____ Efficiency _____ %
(Welded, Dbl., Single, Lap, Butt) (Yes or No) (Spot or Complete) (Yes or No)

If riveted describe seams fully on reverse side of form

14. HEADS (a) Material ASTM A-212 Gr. B T.S. 70,000 Roll Clad with 1/4" Stainless Steel
(Top, bottom, ends) Location Thickness Crown Radius Knuckle Radius Elliptical Ratio Conical Apex Angle Hemispherical Radius Flat Diameter Side to Pressure (Convex or Concave)

(a) Top 1 Plus 1/4" cladding 15 In. Concave

(b) Channel _____

(c) Floating _____

If removable, bolts used (a) _____ (b) _____
(Material, Spec. No., T.S., Size, Number)

(c) _____ Other fastening _____
(Describe or Attach Sketch)

15. Constructed for max. allowable working press. 1500 psi at max. temp. 600 °F. Min. temp. (when less than -20°) _____ °F. Hydrostatic Pneumatic or Test Press 2250 psi.
Combination

Items below to be completed for all vessels where applicable.

16. SAFETY VALVE OUTLETS: Number None Size _____ Location _____

17. NOZZLES

Purpose (Inlet, Outlet, Drain)	Number	Diam. or Size	Type	Material	Thickness	Reinforcement Material	How Attached
Coolant In&Out	2	6" Sch. 80	Weld End Type	316SS	0.432	Steel	Welded
Steam Outlet	1	6" Sch. 40	Weld End	Carbon Stl.	0.280	Steel	Welded
Feedwater Inlet	1	2-1/2" 80	Weld End	Carbon Stl.	0.276	Steel	Welded
Vent, Drains, Etc.	8	1" Sch. 80	Weld End	Carbon Stl.	0.179	Steel	Welded
Phosphate Inlet	1	1/2" Sch. 80	Weld End	Carbon Stl.	0.147	Steel	Welded

¹ If postweld heat-treated.

² List under remarks other internal or external pressures with coincident temperature when applicable.

FORM U-1 (back)

18. INSPECTION Manholes, No. One Size 15" Location Top of shell
 OPENINGS: Handholes, No. 4 Size 5-1/2 Location Bottom Head-(Tube Side)
 Handholes Threaded, No. 2 Size 5-1/8 Location Top of Shell
 19. SUPPORTS: Skirt _____ Lugs _____ Legs 3 Other _____
 (Yes or No) (Number) (Number) (Describe)

Welded to tube
 Attached plate
 (Where & How)

20. REMARKS: Stm. Generator for Nuclear Power Plant. Pressurized Water (coolant) on tube side,
steam and water on shell side. Meets code case 1270 N. Type 316 st. stl. extension welded
to coolant nozzles governed by piping code.

(Brief description of purpose of the vessel, as Air Tank, After Cooler, Jacketed Cooker, etc. State contents of each part.)

We certify that the statements made in this report are correct and that all details of design, material, construction, and workmanship of this vessel conform to the ASME Code for Unfired Pressure Vessels.

Date October 7 19 60 Signed Westinghouse Electric Corp. By s/Peter F. Sokolowski
 (Manufacturer)

Certificate of Authorization Expires December 1961

Inspection Agency's Serial No. LMC 16-A-9037-1

CERTIFICATE OF SHOP INSPECTION

VESSEL MADE BY Westinghouse Electric Corp. at Lester, Pa.

I, the undersigned, holding a valid commission issued by the National Board of Boiler and Pressure Vessel Inspectors and/or the State of Pennsylvania and employed by Lumbermens Mutual Casualty Co. of Chicago, Ill. have inspected the pressure vessel described in this manufacturer's data report on June 2 19 61, and state that to the best of my knowledge and belief, the manufacturer has constructed this pressure vessel in accordance with the applicable sections of the ASME Boiler and Pressure Vessel Code.

By signing this certificate neither the Inspector nor his employer makes any warranty, expressed or implied, concerning the pressure vessel described in this manufacturer's data report. Furthermore, neither the Inspector nor his employer shall be liable in any manner for any personal injury or property damage or a loss of any kind arising from or connected with this inspection.

Date June 2, 19 61

s/W. M. Nuss

Inspectors Signature

Commissions Pa-1406

Nat'l Board or State and No.

CERTIFICATE OF FIELD ASSEMBLY INSPECTION

I, the undersigned, holding a valid commission issued by the National Board of Boiler and Pressure Vessel Inspectors and/or the State of _____ and employed by _____ of _____

_____ have compared the statements in this manufacturer's data report with the described pressure vessel and state that parts referred to as data items _____, not included in the certificate of shop inspection have been inspected by me and that to the best of my knowledge and belief the manufacturer has constructed and assembled this pressure vessel in accordance with the applicable sections of the ASME Boiler and Pressure Vessel Code. The described vessel was inspected and subjected to a hydrostatic test of _____ psi.

By signing this certificate neither the Inspector nor his employer makes any warranty, expressed or implied, concerning the pressure vessel described in this manufacturer's data report. Furthermore, neither the Inspector nor his employer shall be liable in any manner for any personal injury or property damage or a loss of any kind arising from or connected with this inspection.

Date _____ 19 _____

Inspector's Signature

Commissions _____

Nat'l Board or State and No.

W. O. NO. 2288-12

BURNS AND ROE, INC.
NEW YORK, N. Y.

DATE: 9/56

DRAWING NO. _____

BY: M. GIDEN

TITLE: _____ SHEET 1 OF 20

PM-3A STEAM GENERATOR

STRESS ANALYSIS

OF

TUBE SHEET-SHELL-CHANNEL ASSEMBLY

ITEM 2.1

INTRODUCTION

A detailed mechanical plus thermal discontinuity analysis of the PM-3A steam generator tube sheet-shell-channel assembly was performed.

The design basis used is the ASME Boiler and Pressure Vessel Code Section III Nuclear Vessels 1965.

The stresses were calculated for the following operating and design conditions:

	Operating	Design
Primary side pressure (p_o) psi	300	600
Secondary side pressure (p_i) psi	1300	1500
Temperature drop across tube sheet (ΔT), $^{\circ}F$	46	100

The following is a description of the analysis with a list of the basic assumptions used.

The tube sheet-shell-channel assembly was considered comprised of the following parts:

1. Spherical shell segment.
2. Short cylindrical shell.
3. Long cylindrical shell.
4. Rectangular ring (solid portion of tube sheet).
5. Perforated circular plate (perforated portion of tube sheet).

Figure 2 shows the free body diagram of the assembly, with the various unknown forces and moments acting. Table 1 gives the rotation and displacements of the various parts in terms of the applied loadings. From the relationships of continuity (equations 1-8 below Table 3) the 8 characteristic equations in Table 3 were determined. These were solved by electronic computer, yielding the unknown moments and forces at the top of Table 4, which in turn, permits calculation of the rotations and deflections tabulated below them. Stresses were determined from these rotations and displacements; these stresses are tabulated in Table 5 in terms of pressure and ΔT across the tube sheet. Specific design and operating conditions were then imposed and the resulting stresses calculated.

The following assumptions and simplifications have been made in this analysis:

1. All material is homogeneous, isotropic and elastic.
2. Thin shell and small deflection theories are used.
3. The short cylindrical shell component is taken as having a constant thickness at its mean value. Actual thicknesses are used to find stresses.

W. O. NO. 2283-12

BURNS AND ROE, INC.
NEW YORK, N. Y.

DATE: 9/66

DRAWING NO. _____

BY: M. SIDEN

TITLE: _____ SHEET 3 OF 20

4. The perforated portion of the tube sheet is considered isothermal at the average primary temperature (T_o).
5. The temperature drop across the rectangular solid ring component is assumed to be linear.
6. The influence of the divider plate is not considered.
7. Stress concentration factors have not been included.
8. Strength of cladding was not included.

SUMMARY

THE RESULTS OF THE DISCONTINUITY ANALYSIS ARE SUMMARIZED ON TABLES 5, 6 AND 7. TABLE 5 INDICATES THE AXIAL, RADIAL AND TANGENTIAL MEMBRANE AND DISCONTINUITY STRESSES AT THE CRITICAL LOCATIONS IN TERMS OF PRESSURE p_0, p_1 AND TEMPERATURE GRADIENT ΔT . TABLE 6 GIVES THE ACTUAL VALUE OF THE COMBINED STRESSES FOR THE OPERATING CONDITION OF PRESSURE AND TEMPERATURE ($p_0 = 1300$ PSI, $p_1 = 300$ PSI, $\Delta T = 46^\circ\text{F}$) WHILE TABLE 7 GIVES THE VALUES OF THE STRESSES FOR THE DESIGN CONDITIONS OF $p_0 = 1500$ PSI, $p_1 = 600$ PSI, $\Delta T = 100^\circ\text{F}$.

THE STRESSES ARE COMBINED IN ACCORDANCE WITH REF. # 5. THE ALLOWABLE PRIMARY + SECONDARY STRESS INTENSITY FOR SA-105 GR II AT 600°F IS $3(S_M) = 3(17800) = 53400$ PSI

THE MAXIMUM STRESS INTENSITY OCCURS AT THE SHORT SHELL - TUBE SHEET JUNCTION AND IS EQUAL TO 38,300 PSI (PAGE 19). SINCE THIS VALUE IS BELOW THE ALLOWABLE STRESS INTENSITY OF REF. # 5 THE STEAM GENERATOR IS SATISFACTORY FOR THE DESIGN CONDITIONS INVESTIGATED.

PHYSICAL CONSTANTS & EXPRESSIONS USED IN ANALYSISFOR SPHERICAL SHELL SEGMENT

$$\lambda^4 = 3(1-\nu^2) \left(\frac{a_m}{h_1} \right)^2$$

FOR SHORT SHELL

$B_{11}, B_{12}, B_{22}, G_{11}, G_{12}, G_{22}$ ARE SHORT SHELL COEFFICIENTS

$$\beta_2^4 = 3(1-\nu^2) / R_m h_2^2$$

$$D_2 = E h_2^3 / 12(1-\nu^2)$$

$$K_1 = \frac{B_{12} E}{2\beta_2^2 D_2} \left(\frac{R_1^2 \cot \phi}{2R_m} \right)$$

$$K_2 = -\frac{B_{11} E}{2\beta_2^3 D_2} \left(\frac{R_1^2 \cot \phi}{2R_m} \right) + \frac{R_1}{h_2} \left(R_m - \frac{\nu}{2} R_1 \right)$$

$$K_3 = -\frac{G_{12} E}{2\beta_2^2 D_2} \left(\frac{R_1^2 \cot \phi}{2R_m} \right)$$

$$K_4 = -\frac{G_{11} E}{2\beta_2^3 D_2} \left(\frac{R_1^2 \cot \phi}{2R_m} \right) + \frac{R_1}{h_2} \left(R_m - \frac{\nu}{2} R_1 \right)$$

FOR SOLID RING

$$K_5 = \frac{12\Gamma_2^2}{t h^3} \left[(R_m - \Gamma_2) \frac{R_1^2}{2\Gamma_2} + (\Gamma_1 - \Gamma_1) \left(\Gamma_2 - \frac{\Gamma_1 + R_1}{2} \right) \left(\frac{\Gamma_1 + R_1}{2\Gamma_2} \right) + \frac{\Gamma_1^2 t}{4\Gamma_2} \right]$$

$$K_6 = \frac{12\Gamma_2^2}{t h^3} \left[(\Gamma_m - \Gamma_2) \frac{\Gamma_3^2}{2\Gamma_2} + (\Gamma_3 - \Gamma_1) \left(\Gamma_2 - \frac{\Gamma_1 + \Gamma_3}{2} \right) \left(\frac{\Gamma_1 + \Gamma_3}{2\Gamma_2} \right) + \frac{\Gamma_1^2 t}{4\Gamma_2} \right]$$

W. O. NO. 2288-12

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DATE: 9/66

DRAWING NO. _____

BY: M. GIDEN

TITLE: _____

SHEET 6 OF 20

CONSTANTS & EXPRESSIONS (CONT.)

PERFORATED PLATE (TUBE SHEET)

ν^*

E^*

LONG SHELL

$$\beta_3^4 = 3(1-\nu^2)/E^2 h_3^2$$

$$D_3 = E h_3^3 / 12(1-\nu^2)$$

W. O. NO. 3288-12

BURNS AND ROE, INC.
NEW YORK, N. Y.

DATE: 9/66

DRAWING NO. _____

BY: M. GIDEN

TITLE: _____ SHEET 7 OF 20

PERFORATED PLATE: (TUBE SHEET)

$$\frac{H}{R} = \frac{.80}{1.30} = .615$$

$$\frac{E^*}{E} = 0.66$$

$$U^* = 0.29$$

EVALUATED PHYSICAL CONSTANTS1. SPHERICAL SHELL SEGMENT

$a_i = 15.25$

$a'_i = 15.00$

$a_m = 15.75$

$h_1 = 1.00$

$a_m/h_1 = 15.75$

$\phi = 78^\circ - 45'$

$\nu = 0.30$

$\sin \phi = .9805$

$\lambda^4 = 677.2106$

$\lambda^3 = 132.8852$

$\lambda^2 = 26.0406$

$\lambda = 5.103$

2. SHORT CYLINDRICAL SHELL

$R_1 = 15.25$

$R_m = 15.75$

$R_2 = 16.25$

$h'_2 = 1.00$

$h_2 = 1.125$

$h''_2 = 1.25$

$l = 3.1562$

$\beta_2 = .3068$

$\beta_2^2 = .09413$

$\beta_2^3 = .028$

$\beta l = .97$

$D_2 = E(.1304)$

$B_{11} = 2.0791$

$B_{12} = 3.2865$

$B_{22} = 7.2922$

$G_{11} = -1.0180$

$G_{12} = -3.1307$

$G_{22} = -6.8269$

3. PERFORATED PLATE

$h = 5.3125$

$r_1 = 13.00$

$\nu^* = .29$

$E^* =$

$E/E^* = .66$

$D' = 9.003E$

4. SOLID RING

$t = 3.25$

$r_2 = 14.625$

$r_3 = 16.25$

5. LONG CYLINDRICAL SHELL

$h_3 = 0.625$

$r_m = 15.3125$

$r_3 = 15.00$

$\beta_3 = .4156$

$\beta_3^2 = .1727$

$\beta_3^3 = .07177$

$D_3 = E(.02232)$

FIGURE 1
FREE BODY DIAGRAM
SHELL TUBE SHEET ASSEMBLY

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BY: M. GIDREN

TITLE:

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TABLE I

		M_1	H_1	M_2	H_2	M_3	H_3	M_4	N_4	p_0	p_1	$E_d \Delta T$	$E_d T_0$	$E_d T_1$
SPHERICAL SHELL	$E \theta_1$	$-\frac{4\lambda^3}{a_m h_1}$	$\frac{2\lambda^2 \sin \phi}{h_1}$											
	$E \delta_1$	$-\frac{2\lambda^2 \sin \phi}{h_1}$	$\frac{2a_m \lambda \sin \phi}{h_1}$							$\frac{(1-\nu) \rho_1^2}{2h_1}$			R_M	
SHORT SHELL	$E \theta_1$	$\frac{B_{22} E}{2\beta D}$	$\frac{B_{12} E}{2\beta^2 D}$	$\frac{G_{22} E}{2\beta D}$	$-\frac{G_{12} E}{2\beta^2 D}$					K_1				
	$E \delta_1$	$-\frac{B_{12} E}{2\beta^2 D}$	$-\frac{B_{11} E}{2\beta^3 D}$	$-\frac{G_{12} E}{2\beta^2 D}$	$\frac{G_{11} E}{2\beta^3 D}$					K_2			R_M	
	$E \theta_2$	$-\frac{G_{22} E}{2\beta D}$	$-\frac{G_{12} E}{2\beta^2 D}$	$-\frac{B_{22} E}{2\beta D}$	$\frac{B_{12} E}{2\beta^2 D}$					K_3				
	$E \delta_2$	$-\frac{G_{12} E}{2\beta^2 D}$	$-\frac{G_{11} E}{2\beta^3 D}$	$-\frac{B_{12} E}{2\beta^2 D}$	$\frac{B_{11} E}{2\beta^3 D}$					K_4			R_M	
TUBE SHEET	$E \theta$			$\frac{12F_2 R_M}{th^3}$	$\frac{6F_2 R_M}{th^2}$	$-\frac{12F_2 F_1}{th^3}$	$\frac{6F_2 F_1}{th^2}$	$\frac{12F_2 F_1}{th^3}$		K_5	K_6	$-\frac{F_1}{h}$ or $-\frac{F_2}{h}$		
	$E \delta$				$-\frac{F_2 R_M}{th}$		$\frac{F_2 F_1}{th}$		$-\frac{F_2 F_1}{th}$			$-\frac{F_1}{2}$	r	
	$E \delta_2$			$-\frac{6F_2 R_M}{th^2}$	$-\frac{4F_2 R_M}{th}$	$\frac{6F_2 F_1}{th^2}$	$-\frac{2F_2 F_1}{th}$	$-\frac{6F_2 F_1}{th^2}$	$-\frac{F_2 F_1}{th}$	$-\frac{h}{2} K_5$	$-\frac{h}{2} K_6$	$\frac{R_M - F_2}{2}$	R_M	
	$E \delta_3$			$\frac{6F_2 R_M}{th^2}$	$\frac{2F_2 R_M}{th}$	$-\frac{6F_2 F_1}{th^2}$	$\frac{4F_2 F_1}{th^2}$	$\frac{6F_2 F_1}{th^2}$	$-\frac{F_2 F_1}{th}$	$\frac{h}{2} K_5$	$\frac{h}{2} K_6$	$\frac{F_1 - F_2}{2}$		L
	$E \theta_4$							$-\frac{12(1-\nu) F_1}{E^* h^3}$		$\frac{2(1-\nu) E F_1^3}{2E^* h^3}$	$-\frac{3(1-\nu) E F_1^3}{2E^* h^3}$			
LONG SHELL	$E \delta_4$								$\frac{(1-\nu) F_1}{h}$				r	
	$E \theta_3$					$\frac{E}{\beta_3 D_3}$	$\frac{E}{2\beta_3^2 D_3}$							
	$E \delta_3$					$-\frac{E}{2\beta_3^2 D_3}$	$-\frac{E}{2\beta_3^3 D_3}$				$\frac{F_2(F_1 - \frac{3F_1^2}{2})}{h_3}$			L

FORM BR 8003-1

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TABLE 2

EDGE ROTATIONS & DISPLACEMENTS IN TERMS OF
EDGE LOADINGS, PRESSURE, & THERMAL LOADING

	M_1	H_1	M_2	H_2	M_3	H_3	M_4	N_4	P_0	P_1	$\alpha \Delta T$
SPHERICAL SHELL	$ED_1 -33.75$	$+51.06$									
	$ES_1 -51.06$	154.55							$+77.21$		
SHORT SHELL	$ED_1 +71.15$	$+133.87$	-79.086	$+127.5$					$+198.17$		
	$ES_1 -133.87$	-284.8	$+127.5$	-151.0					-221.4		
	$ED_2 +79.09$	$+127.5$	-91.15	$+133.87$					$+188.77$		
	$ES_2 +127.5$	139.45	-133.87	$+286.0$					$+414.3$		
TYPE SHEET PERFORATED PLATE	ED		$+5.673$	$+15.07$	-5.45	$+14.50$	$+4.68$		101.8	-83.28	-2.45 -2.35
	ES			-13.22		$+12.90$		-11.01			-6.5
	ED_2		-15.07	-52.89	14.50	-25.80	-12.44	-11.01	-270.14	221.5	-0.562
	ES_3		$+15.07$	25.94	-14.50	$+53.36$	$+12.44$	-11.01	270.14	-221.5	$+0.3138$
	ES_4						-1.118		$+23.64$	-23.64	
LONG SHELL	ED_3				107.574	129.7		$+1.74$			
	ES_3				-129.7	-312.5				$+313.44$	

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TABLE 3

CHARACTERISTIC EQUATIONS FOR TUBE SHEET ASSEMBLY

	M_1	H_1	M_2	H_2	M_3	H_3	M_4	N_4		P_0	P_1	$E \alpha \Delta T$
(1)	-124.9	-52.81	79.086	-127.5						198.17		
(2)	+82.81	+439.4	-127.5	+151.0						-298.6		
(3)	+79.09	+127.5	-96.82	+118.80	+5.45	-14.50	-4.68			-86.97	-83.28	-2.45
(4)	+127.5	+139.45	-118.80	+338.89	-14.50	+25.80	+12.44	+11.01		+684.44	221.21	-0.562
(5)			+5.673	+15.07	-113.02	-115.2	+4.68			-101.8	+83.28	2.75
(6)			+15.07	+25.94	+115.2	+365.86	+12.44	-11.01		-270.11	+534.65	-0.348
(7)			+5.673	+15.07	-5.45	+14.50	+5.798			-78.16	+59.64	2.75
(8)				-13.22		+12.90		-12.75				6.5

ABOVE EXPRESSIONS DERIVED BY EQUATING DEFLECTIONS AND ROTATIONS
OF ADJACENT PARTS OF THE SHELL TUBE SHEET ASSEMBLY:

$$\begin{aligned}
 (1) : E\theta_1 &= E\theta_1 & (2) : E\theta_1 &= E\theta_1 & (3) : E\theta_2 &= E\theta & (4) : E\theta_2 &= E\theta_L \\
 (5) : E\theta &= E\theta_1 & (6) : E\theta_1 &= E\theta_3 & (7) : E\theta &= E\theta_4 & (8) : E\theta &= E\theta_4
 \end{aligned}$$

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W. O. NO. 2288-12BURNS AND ROE, INC.
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BY: GIDENTITLE: _____ SHEET 13 OF 20TABLE 4SOLUTION OF MATRIX EDGE LOADINGS & DISPLACEMENTS
IN TERMS OF p_0 , p_1 & ΔT .

EDGE LOADINGS		p_0	p_1	$E \Delta T$	ΔT $E \Delta = 188$
	M_1	-0.7552	0.2248	0.00796	1.496
	H_1	-0.7352	0.2727	0.00334	0.722
	M_2	-4.074	2.3181	0.0234	4.399
	H_2	-2.864	1.0405	0.00419	0.788
	M_3	0.9486	-3.1325	0.0712	13.386
	H_3	-0.6076	2.4481	-0.0648	-12.182
	M_4	0.3619	-3.7534	0.6697	125.904
	H_4	6.50	-6.50	0	0
	N_1	7.527	0	0	0
	N_2	7.38	0	0	0
	N_3	0	7.35	0	0
	N_4	2.3551	1.3981	-0.5798	-109
	$E\theta_1$	-12.081	6.346	-0.07215	-13.565
ROTATIONS & DISPLACEMENTS	$E\delta_1$	2.14	30.67	0.187	35.16
	$E\theta_2$ $E\theta_3$	23.25	-19.45	3.81	597.8
	$E\delta_2$	-58.27	54.02	-0.383	-72
	$E\delta_3$	66.85	-45.62	11.02	2072
	$E\theta$ $E\theta_4$	23.24	-19.44	-0.749	-149
	$E\delta$ $E\delta_4$	4.10	2.43	-1.01	-189.88

NOTE H_4 , N_1 , N_2 & N_3 ARE STATISTICALLY DE SIGNIFICANT

STATICALLY DETERMINABLE QUANTITIES

$$N_2 = \frac{R_1^2 p_0}{2 R_M} = \frac{(15.25)^2 p_0}{2 (15.75)} = \frac{232.56 p_0}{31.50} \quad N_1 = \frac{N_2}{\sin \phi} = \frac{7.38 p_0}{.9805} = 7.527 p_0$$

$$N_3 = \frac{r_3^2 p_1}{2 r_M} = \frac{15^2 p_1}{2 (15.3125)} = \frac{225 p_1}{30.625} \quad H_4 = \frac{r_1}{2} (p_0 - p_1) = 6.5 (p_0 - p_1)$$

EQUATIONS FOR AXIAL (x) & RADIAL (r) STRESSES(A) MEMBRANE COMPONENTSSHORT SHELL:

$$(\sigma_M)_{1x} = N_2 / h_2 = \frac{7.38 p_0}{1.25} = 5.904 p_0$$

$$(\sigma_M)_{2x} = N_1 / h_2 = \frac{7.38 p_0}{1.0} = 7.38 p_0$$

LONG SHELL:

$$(\sigma_M)_{3x} = N_3 / h_3 = \frac{7.35 p_1}{0.625} = 11.76 p_1$$

TUBE SHEET:

$$(\sigma_M)_{4r} = (\sigma_M)_{5r} = \left(\frac{S_P}{S_P - d} \right) \frac{N_4}{h} = 1.551 \frac{M_4}{h}$$

(B) BENDING COMPONENT

$$S_P = 1.30''$$

$$d = \frac{1}{2} (0.500 + .424) = 0.462'' \quad (m + m)$$

SHORT SHELL:

$$(\sigma_b)_{1x} = 6 M_1 / h_2^2 = 3.840 M_1$$

$$(\sigma_b)_{2x} = 6 M_2 / h_2^2 = 6.000 M_2$$

LONG SHELL:

$$(\sigma_b)_{3x} = 6 M_3 / h_3^2 = 15.360 M_3$$

AXIAL & RADIAL STRESSES (CONT.)(B) BENDING STRESS (CONT.)TUBE SHEET, $h = 5.3125$

$$(\sigma_b)_{4r} = \left(\frac{S_P}{S_P - d} \right) \frac{6M_4}{h^2} = 1.551 \left(\frac{6M_4}{h^2} \right) = 0.3297 M_4$$

$$(\tau_b)_{5r} = \frac{Eh}{2} \left[\frac{\theta}{r_i} + \eta \frac{r_i^2 (P_0 - P_1)}{16D'} \right]$$

$$\eta = 0.75 \quad (\text{assumed})$$

$$(\sigma_b)_{5r} = 2.656 E \left[0.07692 \theta + 0.8799 (P_0 - P_1) \right]$$

FOR SECONDARY SIDE, THERMAL STRESS IS FOUND BY ADDING
A SKIN EFFECT:

$$(\sigma_b)_{5r} = \frac{E \alpha \Delta T}{1 - \nu} = 1.429 E \alpha \Delta T = 268.7 \Delta T$$

EQUATIONS FOR TANGENTIAL (θ) STRESS(A) MEMBRANE COMPONENT:SHORT SHELL:

$$(\sigma_M)_{1\theta} = \frac{E\delta_1}{R_M}$$

$$(\sigma_M)_{2\theta} = \frac{E\delta_2}{R_M}$$

LONG SHELL:

$$(\sigma_M)_{3\theta} = \frac{E\delta_3}{r_M}$$

(B) BENDING COMPONENT:SHORT SHELL

$$(\sigma_b)_{1\theta} = \pm \nu(\sigma_b)_{1x}$$

$$(\sigma_b)_{2\theta} = \nu(\sigma_b)_{2x}$$

LONG SHELL

$$(\sigma_b)_{3\theta} = \nu(\sigma_b)_{3x}$$

TUBE SHEET

$$(\sigma_t)_{4\theta} = \pm \frac{Eh\theta}{2r} - \pm 0.2043 E\theta$$

$$(\sigma_b)_{5\theta} = (\sigma_b)_{5r}$$

TABLE 5AXIAL, RADIAL & TANGENTIAL STRESS IN TERMS OF p_o , p_i & ΔT

		MEMBRANE			DISCONTINUITY		
		p_o	p_i	ΔT	p_o	p_i	ΔT
AXIAL	σ_{1x}	5.904	0	0	∓ 2.90	± 0.863	± 5.745
	σ_{2x}	7.38	0	0	∓ 24.44	± 13.91	± 26.39
	σ_{3x}	0	11.76	0	± 14.57	∓ 48.12	± 205.61
RADIAL	σ_{4r}	.687	.408	-31.81	$\mp .142$	± 1.47	± 219.26 ± 49.44
	σ_{5r}	.687	.408	-31.81	± 7.09	∓ 6.30	± 238.3 ± 30.4
TANGENTIAL	$\sigma_{1\theta}$	-136	-1.947	2.23	$\mp .97$	$\pm .29$	± 1.91
	$\sigma_{2\theta}$	-3.70	3.43	-4.57	∓ 8.14	± 4.63	± 8.79
	$\sigma_{3\theta}$	4.37	-2.98	135.3	± 4.85	∓ 16.02	± 68.47
	$\sigma_{4\theta}$	.687	.408	-31.81	± 4.75	∓ 3.97	∓ 30.44
	$\sigma_{5\theta}$	.687	.408	-31.81	± 7.03	∓ 6.30	± 238.3 ± 30.4

NOTE: FOR SHELL, UPPER SIGN REFERS TO OUTER SURFACE
 LOWER " " INNER SURFACE

FOR TUBE SHEET, UPPER SIGN REFERS TO SECONDARY SIDE
 LOWER " " PRIMARY SIDE

TABLE C
AXIAL, RADIAL AND TANGENTIAL STRESS
FOR OPERATING CONDITIONS

$p_o = 1300 \text{ PSI.}$ $p_i = 300 \text{ PSI.}$ $\Delta T = 46^\circ \text{ F}$			
LOCATION	F I B E R	AXIAL OR RADIAL (PSI)	TANGENTIAL (PSI)
SHORT SHELL AT SPHERICAL SHELL (σ_1)	O	4423	-223
	I	10917	1980
SHORT SHELL AT TUBE SHEET (σ_2)	O	-16700	-12780
	I	35900	4785
LONG SHELL AT TUBE SHEET (σ_3)	O	17570	15660
	I	-10530	6360
JUNCTION OF RING & PERF. PLATE (σ_4)	S	9895	3136
	R	1587	-4030
CENTER OF TUBE SHEET (σ_5)	S	17828	17828
	P	-6364	-6364

TABLE 7AXIAL, RADIAL AND TANGENTIAL STRESS FOR
DESIGN CONDITIONS

$p_0 = 1500 \text{ PSI}$ $p_1 = 600 \text{ PSI}$ $\Delta T = 100^\circ \text{ F}$			
LOCATION	F I B E R	AXIAL OR RADIAL	TANGENTIAL (PSI)
SHORT SHELL AT SPHERICAL SHELL (σ_1)	O	5593	505
	I	12107	2685
SHORT SHELL AT TUBE SHEET (σ_2)	O	-14600	-12578
	I	36800	4600
LONG SHELL AT TUBE SHEET (σ_3)	O	20603	22807
	I	-6491	13763
JUNCTION OF RING & PERF. PLATE (σ_4)	S	20690	-206
	P	2371	-3605
CENTER OF TUBE SHEET (σ_5)	S	28754	28754
	P	5706	5706

$$\text{MAX STRESS INTENSITY} = \sigma_{\text{max}} - \sigma_{\text{min}} = 36800 - (-1500) = 38300 \text{ psi}$$

$$\text{Radial stress } (\sigma_r) = -1500 \text{ psi (inside surface)}$$

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TITLE: _____ SKETCH 20 OF 20

REFERENCES

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PM-3A STRESS ANALYSIS

OF STEAM GENERATOR

FOR THERMAL TRANSIENTS

ITEM 2.2

INTRODUCTIONTRANSIENT ANALYSIS

THE EFFECT OF TRANSIENTS DUE TO BOTH NORMAL OPERATING CONDITIONS AND ACCIDENTAL SITUATIONS RESULTING FROM EQUIPMENT MALFUNCTIONS OR OPERATOR ERRORS WERE ANALYZED AND REPORTED IN REFERENCE 1 APPENDIX B-3. BASED ON THIS REPORT, THE FOLLOWING CRITICAL TRANSIENTS WERE SELECTED FOR EVALUATION.

1. RUN No. 2 LOAD CHANGE 100% TO 25% 0 SEC.
2. RUN No. 18 LOAD CHANGE 25% TO 100% 0 SEC.
3. RUN No. 41 ROD DROP WITHOUT SCRAM
4. RUN No. 42 ROD WITHDRAWAL AT POWER RUNS
5. RUN No. 63 LOSS OF PRIMARY COOLANT FLOW.
6. RUN No. 75 STEAM LINE RUPTURE
7. RUN No. 83 COLD WATER ACCIDENT

A FATIGUE EVALUATION OF THE STEAM GENERATOR IS MADE FOR THE ABOVE TRANSIENTS IN ACCORDANCE WITH REFERENCE # 2.

SUMMARY

BASED ON THE SEVEN TRANSIENT, THE MAXIMUM STRESS INTENSITY IN THE STEAM GENERATOR AT THREE CRITICAL LOCATION WERE CALCULATED. THE RESULTS OF THE ANALYSIS IS TABULATED IN TABLES 5 AND 6.

THESE STRESS INTENSITIES WERE ANALYZED ON A FATIGUE BASIS IN ACCORDANCE WITH REFERENCE #2. THE RESULTS OF THIS ANALYSIS IS TABULATED BELOW.

<u>TRANSIENT</u>	<u>ALLOWABLE NO. OF CYCLES</u>
1	∞
2	∞
3	∞
4	∞
5	35000
6	4600
7	100000

IT SHOULD BE NOTED HOWEVER, THAT THE ACTUAL NUMBER OF CYCLES OF OPERATION OF THE VARIOUS TRANSIENTS ANALYZED IS REQUIRED BEFORE A FINAL CONCLUSION CAN BE MADE. ONCE THE ACTUAL NUMBER OF CYCLES HAS BEEN DETERMINED, A USAGE FACTOR FOR EACH TRANSIENT CAN BE CALCULATED. THE USAGE FACTOR IS THE RATIO OF THE ACTUAL CYCLES TO THE ALLOWABLE CYCLES. A TOTAL USAGE FACTOR WHICH IS THE SUM OF THE INDIVIDUAL USAGE FACTORS CAN THEN BE CALCULATED. FOR AN ACCEPTABLE DESIGN, IT IS NECESSARY THAT THE TOTAL USAGE FACTOR ≤ 0.80 .

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BY: M. GIDENTITLE: _____ SHEET 4 OF 12TABLE 1SUMMARY OF TRANSIENTS PRESSURES AND TEMPERATURES

TRANSIENT	TIME (SEC.)	P_o	P_i	T_o	T_i	ΔT
1	0	1300	304	462.5	418	44.5
2	0	1300	418	466.8	449	17.8
3	0	1300	N.A.* 302	465.7	418	47.7
4	0	1300	304	462.5	N.A. + 418	44.5
5	0	1300	304	462.5	418	44.5
6	0	1300	350	463	463	0
7	0	1300	480	463	463	0

* VALUE OF P_i ASSUMED TO CORRESPOND TO (T_i) + " " T_i " " " " (P_i)

TABLE 1 IS A TABULATION OF THE PRESSURE AND TEMPERATURE CONDITIONS IN THE STEAM GENERATOR PRIOR TO THE START OF THE TRANSIENT ($T=0$). TABLE 2 PRESENTS THE SAME INFORMATION AFTER THE TRANSIENT HAD STARTED.

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BY: M. GIDENTITLE: _____ SHEET 5 OF 12TABLE 2SUMMARY OF TRANSIENT PRESSURES AND TEMPERATURES

TRANSIENT	TIME (SEC)	P_0	P_1	T_0	T_1	ΔT
1	75	1314	500	480	468	12
2	18	1317	278	455	414	41
3	32	1195	N.A. + (228)	417.5	393	24.5
4	30	1420	379	417.5	N.A. + (439)	-21.5
5	80	1220	24	357.5	268	89.5
6	13	1220	70	418	313	105
7	30	1270	55	403	343	60

* VALUE OF P_1 ASSUMED TO CORRESPOND TO (T_1) + " " T_1 " " " " T_0 (P_1)

TABLE 3

STRESSES ($\sigma_x, \sigma_\theta, \sigma_r$) AT CRITICAL JUNCTIONS (FOR TIME $t = 0$ SEC.)SHORT SHELL AT TUBE SHEET (σ_2)

TRANSIENT	ΔT °F	AXIAL STRESS (PSI.)		TANGENTIAL STRESS (PSI.)	
		OUTSIDE	INSIDE	OUTSIDE	INSIDE
1	44.5	-16810	36110	-12760	4820
2	17.8	-15910	35210	-11945	5040
3	47.7	-16740	36040	-12760	4780
4	44.5	-16810	36110	-12760	4820
5	44.5	-16810	36110	-12760	4820
6	0	-17340	36640	-12580	5360
7	0	-17340	34840	-11540	5200

LONG SHELL AT TUBE SHEET (σ_3)

TRANSIENT	ΔT °F	AXIAL STRESS (PSI.)		TANGENTIAL STRESS (PSI.)	
		OUTSIDE	INSIDE	OUTSIDE	INSIDE
1	44.5	16950	-9850	6315	6315
2	17.8	7360	2440	6015	6015
3	47.7	17700	-10600	6495	6495
4	44.5	17000	-9850	6315	6315
5	44.5	17000	-9850	6315	6315
6	0	6200	2100	3925	3925
7	0	1500	9900	5625	5625

CENTER OF TUBE SHEET (σ_5)

TRANSIENT	ΔT °F	RADIAL STRESS (PSI.)		TANGENTIAL STRESS (PSI.)	
		SECONDARY	PRIMARY	SECONDARY	PRIMARY
1	44.5	17475	-6340	AS STRESS	AS STRESS
2	17.8	11300	-5545		
3	47.7	18170	-6370		
4	44.5	17480	-6340		
5	44.5	17480	-6340		
6	0	8040	-5980	AS RADIAL	AS RADIAL
7	0	7260	-5100		

* STRESSES CALCULATED FROM TABLE 5 ITEM 2.1

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TITLE: _____ SHEET 7 OF 12

TABLE 4
TRANSIENT STRESSES

STRESSES ($\sigma_x, \sigma_\theta, \sigma_r$) AT CRITICAL JUNCTIONS (FOR TIME $t = t_1, 70$ SEC)

TRANSIENT	ΔT °F	AXIAL STRESS (PSI)		TANGENTIAL STRESS (PSI)	
		OUTSIDE	INSIDE	OUTSIDE	INSIDE
1	12	-15230	34630	-11620	5030
2	41	-17590	37000	-13190	4970
3	24.5	-16650	34315	-12260	4700
4	-21.5	-19610	40590	-13880	6130
5	89.5	-18200	36200	-13930	5200
6	105	-17155	35160	-13500	3940
7	60	-19380	38120	-14400	-2520

LONG SHELL AT TUBE SHEET (σ_3)

TRANSIENT	ΔT	AXIAL STRESS (PSI)		TANGENTIAL STRESS (PSI)	
		OUTSIDE	INSIDE	OUTSIDE	INSIDE
1	12	3420	8530	5060	5700
2	41	17500	-10950	14990	5730
3	24.5	14150	-8800	11660	4045
4	-21.5	2380	6520	1510	2830
5	89.5	27430	-34710	29000	8520
6	105	36810	-35150	31330	15585
7	60	28800	-27510	22860	4110

CENTER OF TUBE SHEET (σ_r)

TRANSIENT	ΔT °F	RADIAL STRESS (PSI)		TANGENTIAL STRESS (PSI)	
		SECONDARY	PRIMARY	SECONDARY	PRIMARY
1	12	9730	-5070	RADIAL STRESS	TANGENTIAL STRESS
2	41	17000	-6620		
3	24.5	13015	-6150		
4	-21.5	4310	-6590		
5	89.5	27860	-7765		
6	105	30790	-7480		
7	60	21940	-7850		

THE STRESS INTENSITY FOR THREE CRITICAL LOCATIONS OF THE STEAM GENERATOR WERE CALCULATED AND THE RESULTS ARE PRESENTED IN TABLE 5. THE STRESS INTENSITY IS DEFINED BY:

$$S = \sigma_x - \sigma_\lambda, \quad \sigma_x - \sigma_\theta, \quad \sigma_\theta - \sigma_\lambda$$

FOR TRANSIENT # 1 (OUTSIDE SURFACE) TABLE # 3

$$\sigma_x = -16810 \text{ PSI} \quad \sigma_\lambda = 0 \quad \sigma_\theta = -12760 \text{ PSI}$$

$$S = \sigma_x - \sigma_\lambda = -16810 - 0 = -16810 \text{ PSI.}$$

THE DRAWINGS AVAILABLE DID NOT INDICATE THE FILLET RADIUS AT THE JUNCTION OF BOTH THE SHORT AND LONG SHELL TO TUBE SHEET JUNCTION. THEREFORE FOR THIS ANALYSIS, A STRESS CONCENTRATION OF 2 WAS ASSUMED FOR THE INSIDE SURFACE.

TABLE 5STRESS INTENSITIES ($S = \sigma_t - \sigma_c$) FOR TIME $t = 0$ SEC.STRESS INTENSITY SHORT SHELL AT TUBE SHEET

TRANSIENT	OUTSIDE	INSIDE *
1	- 16810	73520
2	- 15910	71710
3	- 16740	73380
4	- 16810	73520
5	- 16810	73520
6	- 17340	74580
7	- 17340	70980

STRESS INTENSITY LONG SHELL AT TUBE SHEET

TRANSIENT	OUTSIDE	INSIDE *
1	16950	26015
2	7360	5298
3	17700	27695
4	17000	26015
5	17000	26015
6	6200	4550
7	15000	20280

STRESS INTENSITY CENTER OF TUBE SHEET

TRANSIENT	SECONDARY	PRIMARY
1	17780	- 5040
2	11720	- 4245
3	18470	- 5070
4	17780	- 5040
5	17780	- 5040
6	8400	- 4680
7	7740	- 3800

TABLE 6STRESS INTENSITIES ($S = \sigma_x - \sigma_y$) FOR TIME $t = t_1, 70 \text{ SEC.}$ STRESS INTENSITY SHORT SHELL AT TUBE SHEET

TRANSIENT	OUTSIDE ($\sigma_x - \sigma_y$)	INSIDE * ($\sigma_x - \sigma_y$) OR ($\sigma_x - \sigma_\theta$)
1	- 15230	70574
2	- 17590	75317
3	- 16650	69825
4	- 19610	82600
5	- 18200	73620
6	- 17155	71540
7	- 19380	77760

STRESS INTENSITY LONG SHELL AT TUBE SHEET

TRANSIENT	OUTSIDE	INSIDE *
1	5060	17560
2	17500	27630
3	14150	21645
4	2380	13419
5	29000	77940
6	26810	85885
7	28800	59130

TRANSIENT	SECONDARY	PRIMARY
1	10230	- 6385
2	17280	- 7940
3	13240	- 7345
4	4690	- 8010
5	27885	- 8985
6	30860	- 8700
7	22000	- 9120

FATIGUE ANALYSIS LONG SHELL AT TUBE SHEET (INSIDE)

TRANSIENT No.	(S_{max}) MAX. STRESS TABLE (S) OR (L)	(S_{min}) MIN. STRESS TABLE (S) OR (L)	(S_a) ALT. STRESS (PSI.)	No. OF CYCLES*
1	26015	17560	4228	∞
2	27630	5298	11166	∞
3	27695	21645	3025	∞
4	26015	13419	6298	∞
5	77940	26015	25963	35000
6	85885	4550	40668	4600
7	59130	20280	19425	100000

* FIG. N-415(A) REF. #2

A COMPARISON OF TABLES 5 AND 6 INDICATES THAT THE CRITICAL LOCATION IS THE LONG SHELL AT THE TUBE SHEET JUNCTION. THEREFORE THE FATIGUE EVALUATION WAS NECESSARY ONLY AT THIS LOCATION.

$$ALT. STRESS (S_a) = \frac{(S_{max} - S_{min.})}{2}$$

W.O. NO. 22288-13
DRAWING NO. _____
TITLE: _____

BURNS AND ROE, INC.
NEW YORK, N.Y.

DATE: 10/66
BY: M. GIDEN
SHEET 11 OF 12

W.O. NO. 2288-12

BURNS AND ROE, INC.
NEW YORK, N. Y.

DATE: 10/66

DRAWING NO. _____

BY: M. GIDEN

TITLE: _____

SHEET 13 OF 12

REFERENCES

1. PM-3A NUCLEAR POWER PLANT DESIGN ANALYSIS REPORT, VOL. I, BOOK I PREPARED FOR THE AEC UNDER CONTRACT AT-30-3280 BY BURNS & ROE INC. DEC. 1965
2. ASME BOILER AND PRESSURE VESSEL CODE SECTION III NUCLEAR VESSELS. 1965.

W. O. NO. 2288-12

BURNS AND ROE, INC.
NEW YORK, N. Y.

DATE: _____

DRAWING NO. _____

BY: R. J. DOBBS

TITLE: _____

SHEET 1 OF 8

PM-3A STRESS ANALYSIS

ANALYSIS OF STEAM GENERATOR AT
PRIMARY COOLANT NOZZLES
FOR APPLIED PIPING LOADS

ITEM 2.3

ITEM 2.4

INTRODUCTION

THE EFFECT OF PIPING MOMENTS AND LOADS APPLIED TO THE PRIMARY COOLANT NOZZLES IN THE STEAM GENERATOR IS DETERMINED.

THE NOZZLES CONSIST OF A FORGED NECK $7\frac{3}{8}$ " OUTSIDE DIAMETER WELDED INTO THE PRIMARY COOLANT CHANNEL. THE CHANNEL IS A 1" THICK HEMISPHERICAL HEAD WITH A 15" INSIDE RADIUS.

IN LIEU OF ACTUAL PIPING LOADS THE ANALYSIS IS BASED ON USING LOADS THAT WOULD CAUSE YIELDING TO TAKE PLACE IN THE ATTACHED PRIMARY PIPING. THIS IS AN EXTREMELY CONSERVATIVE APPROACH. HOWEVER, IF THE ANALYSIS INDICATES THAT THE SYSTEM CANNOT TOLERATE LOADS OF THIS MAGNITUDE, THEN THE MAGNITUDE OF THE LOADS THAT CAN BE RESISTED IS EASILY DETERMINED. THE RESULTS CAN THEN BE COMPARED WITH ACTUAL CALCULATED PIPING LOADS, OR ANY FUTURE REVISED LOADS TO DETERMINE IF THE SYSTEM IS STRUCTURALLY ADEQUATE.

THE MAXIMUM STRESSES DUE TO APPLIED PIPE LOADS ARE CALCULATED AND THEN COMBINED WITH THE STRESSES DUE TO THE INTERNAL DESIGN PRESSURE OF 1500 PSI AT 600°F, IN ACCORDANCE WITH THE REQUIREMENTS OF THE "TENTATIVE STRUCTURAL DESIGN BASIS FOR REACTOR PRESSURE VESSELS AND DIRECTLY ASSOCIATED COMPONENTS" (NAVY CODE).

ITEM 2.4

SUMMARY

THE RESULTS OF THE ANALYSIS INDICATE THAT THE APPLICATION OF AXIAL LOAD AND MOMENT THAT WOULD CAUSE YIELDING IN THE PRIMARY PIPING CANNOT BE ADEQUATELY RESISTED BY THE STEAM GENERATOR. THE MAGNITUDE OF THE AXIAL LOAD (P) AND MOMENT (M) THAT INDEPENDENTLY CAN BE RESISTED ARE;

$$\begin{aligned}\text{AXIAL LOAD (P)} &= 36600 \text{ LBS} \\ \text{BENDING MOMENT (M)} &= 87600 \text{ IN-LBS}\end{aligned}$$

SINCE THE ABOVE LOADS ARE THE MAXIMUM INDEPENDENT LOADS THAT CAN SAFELY BE RESISTED, THE FOLLOWING EQUATION MUST BE USED TO DETERMINE IF A SPECIFIC COMBINATION OF ACTUAL AXIAL LOAD AND MOMENT CAN BE RESISTED BY THE STEAM GENERATOR CHANNEL;

$$\frac{P_{\text{ACTUAL (LBS)}}}{36600} + \frac{M_{\text{ACTUAL (IN-LB)}}}{87600} \leq 1.0$$

FURTHER INVESTIGATION OF AVAILABLE INFORMATION^{*} INDICATES THAT THE ACTUAL AXIAL LOAD AND MOMENT THAT EXIST AT THE PRIMARY COOLANT NOZZLES, SATISFY THE ABOVE EQUATION. THE CHANNEL-NOZZLE DESIGN IS THEREFORE ACCEPTABLE FOR THE CONDITIONS INVESTIGATED.

* SEE REF. 2

PRIMARY COOLANT NOZZLE: DWG. 662557

$$t = 1.00''$$

$$R = 15.75''$$

$$\lambda_0 = 3.6875''$$

$$S = 1.82 \frac{\lambda_0}{R} \sqrt{\frac{R}{t}}$$

(REF. 1)

$$S = \frac{1.82(3.6875)}{15.75} \sqrt{\frac{15.75}{1.00}} = 1.685$$

FROM FIG. A5-2 REF 1

$$\frac{\sigma t^2}{P} = .30$$

$$\frac{\sigma t^2}{\frac{M}{R} \sqrt{\frac{R}{t}}} \cos \theta = 0.50$$

I. NOZZLE LOADS:

THE SHELL AND NOZZLE SHALL BE CHECKED FOR THE FOLLOWING LOADS APPLIED TO THE PIPE NOZZLE.

1. TORSIONAL MOMENT (M_T) = $\sigma_Y (S)$
2. BENDING MOMENT (M_B) = $\sigma_Y (S)$
3. AXIAL THRUST (P) = $\sigma_Y (A)$

(σ_Y) IS THE 0.2% OFFSET YIELD STRENGTH OF THE MATERIAL AT DESIGN TEMPERATURE, (S) IS THE SECTION MODULUS OF THE CONNECTING PIPING AND (A) IS THE CROSS-SECTIONAL AREA.

THE ABOVE LOADS ARE APPLIED SEPERATELY AND THE STRESSES PRODUCED ARE NOT SUPERIMPOSED. HOWEVER, THE INDIVIDUAL STRESSES ARE COMBINED WITH THE STRESSES DUE TO INTERNAL DESIGN PRESSURE.

THE ALLOWABLE STRESS FOR THE ABOVE CONDITION IS THE STRESS INTENSITY " S_p " FOR THE MATERIAL AT DESIGN TEMPERATURE. " S_p " IS EQUAL TO 90% OF THE 0.2% OFFSET YIELD STRENGTH.

THE ATTACHED PIPING IS 6" SCH. 80 TYPE 316
 $A = 8.40 \text{ IN}^2$ $S = 12.23 \text{ IN}^3$ $\sigma_Y = 17,100 \text{ PSI @ } 300^\circ \text{ F}$

$$M_T = 17100(12.23) = 210,000 \text{ IN.}\cdot\text{#}$$

$$M_B = 17100(12.23) = 210,000 \text{ IN.}\cdot\text{#}$$

$$P = 17100(8.40) = 144,000 \text{ #}$$

STRESS DUE TO AXIAL LOAD (P)

$$\sigma_x = \sigma_a = \frac{.2(14400)}{(1)^2} = 43,200 \text{ psi}$$

STRESS DUE TO MOMENT (M_B)

$$\sigma_x = \sigma_b = \frac{0.5(210,000)(3.97)}{15.7(1)^4} = 26,466 \text{ psi}$$

STRESS DUE TO TORSION (M_T)

$$\sigma_T = \frac{M_T}{\pi(R_o)^2 t} = \frac{210,000}{2\pi(3.69)^2(1)} = 2,460 \text{ psi}$$

THE ABOVE STRESSES ARE ABOVE THE ALLOWABLE
S_y VALUE FOR THE SHELL MATERIAL (S_y = 24,000 psi)
THEREFORE WE CANNOT DESIGN FOR PIPE FILLED LONG.

PIPE JOINT STRESS = 12,000

$$S = \frac{P(R_o + t)}{2t} = \frac{1500(15+1)}{2} = 12,000 \text{ psi} < S_m = 18,000 \text{ psi}$$

$$Y = \frac{R_o + t}{R_i} = \frac{15+1}{11} = 1.065 \quad Y^3 = 1.21$$

INSIDE SURFACE =

$$\sigma_x = \sigma_a = \frac{P}{2} \left[\frac{Y+1}{Y-1} \right] = \frac{1500}{2} \left[\frac{3.21}{.21} \right] = 11,500 \text{ psi}$$

$$\sigma_T = -1500 \text{ psi}$$

OUTSIDE SURFACE:

$$\sigma_r = 0$$

$$\sigma_x = \sigma_\theta = \frac{3P}{2} \left[\frac{1}{r^{3.1}} \right] = \frac{1.8(1500)}{0.21} = 10700 \text{ PSI}$$

DETERMINATION OF MAXIMUM PIPE LOAD THAT WILL
LIMIT SHELL STRESS TO "S_p".

S = MAXIMUM STRESS INTENSITY

$$S = \sigma_x - \sigma_r = \sigma_{x \text{ PIPE}} + \sigma_{x \text{ PRESS.}} - \sigma_r$$

$$24000 = \sigma_{x \text{ PIPE}} + 11500 - (-1500)$$

$$\sigma_{x \text{ PIPE}} = 11000 \text{ PSI}$$

FOR AXIAL LOAD:

$$\sigma_x = \frac{-3(-P)}{(1)^2} = 11000 \text{ PSI}$$

$$\therefore P = 36600 \text{ \#}$$

FOR MOMENT:

$$\sigma_x = \frac{0.5(M) 3.97}{15.75 (1)^2} = 11000$$

$$M = 87,600 \text{ IN. \#}$$

W. O. NO. 2288-12

BURNS AND ROE, INC.
NEW YORK, N. Y.

DATE: 7-6-66

DRAWING NO.

BY: M. GIDEN

TITLE: SHEET 8 OF 9

REFERENCES:

1. "TENTATIVE STRUCTURAL DESIGN BASIS FOR
REACTOR PRESSURE VESSELS AND DIRECTLY
ASSOCIATED COMPONENTS." FEB. 1959 REV.
2. REPORT OF SUBTASK 9.2 CONTRACT AT(30-1)-3345
MARTIN CO. NUCLEAR PROGRAM, BALTIMORE MARYLAND

W. O. NO. 2288-12

BURNS AND ROE, INC.
NEW YORK, N. Y.

DATE: 7-7-44

DRAWING NO. _____

BY: M. GIDEN

TITLE: _____ SHEET 1 OF 9

PM-3A STRESS ANALYSIS

ANALYSIS OF STEAM GENERATOR AT

STEAM OUTLET NOZZLE

FOR APPLIED PIPING LOADS

ITEM 2.4

INTRODUCTION

THE EFFECT OF PIPING MOMENTS ~~AND~~ LOADS APPLIED TO THE STEAM OUTLET NOZZLE IN THE STEAM GENERATOR IS DETERMINED. THE NOZZLE CONSIST OF A FORGED NECK $9\frac{3}{4}$ " OUTSIDE DIAMETER WELDED INTO THE SECONDARY HEAD. THE HEAD IS A $7\frac{1}{16}$ " HEMISPHERICAL SHELL WITH A 15" INSIDE RADIUS.

IN LIEU OF ACTUAL PIPING LOADS THE ANALYSIS IS BASED ON USING LOADS THAT WOULD CAUSE YIELDING TO TAKE PLACE IN THE ATTACHED STEAM PIPING. THIS IS AN EXTREMELY CONSERVATIVE APPROACH. HOWEVER, IF THE ANALYSIS INDICATES THAT THE SYSTEM CANNOT TOLERATE LOADS OF THIS MAGNITUDE, THEN THE MAGNITUDE OF THE LOADS THAT CAN BE RESISTED IS EASILY DETERMINED. THE RESULTS CAN THEN BE COMPARED WITH ACTUAL CALCULATED PIPING LOADS, OR ANY FUTURE REVISED LOADS TO DETERMINE IF THE SYSTEM IS STRUCTURALLY ADEQUATE.

THE MAXIMUM STRESS DUE TO APPLIED PIPE LOADS ARE CALCULATED AND THEN COMBINED WITH THE STRESSES DUE TO INTERNAL DESIGN PRESSURE OF 600 PSI AT 600°F, IN ACCORDANCE WITH THE REQUIREMENTS OF THE "TENTATIVE STRUCTURAL DESIGN BASIS FOR REACTOR PRESSURE VESSELS AND DIRECTLY ASSOCIATED COMPONENTS" (NAVY CODE).

SUMMARY

THE RESULTS OF THE ANALYSIS INDICATE THAT THE APPLICATION OF AXIAL LOAD AND MOMENT THAT WOULD CAUSE YIELDING IN THE STEAM OUTLET PIPING CANNOT BE ADEQUATELY RESISTED BY THE STEAM GENERATOR. THE MAGNITUDE OF THE AXIAL LOAD (P) AND MOMENT (M) THAT INDEPENDENTLY CAN BE RESISTED ARE;

$$\begin{aligned}\text{AXIAL LOAD (P)} &= 21,300 \text{ \#} \\ \text{BENDING MOMENT (M)} &= 35,700 \text{ IN.-\#}\end{aligned}$$

SINCE THE ABOVE LOADS ARE THE MAXIMUM INDEPENDENT LOADS THAT CAN SAFELY BE RESISTED, THE FOLLOWING EQUATION MUST BE USED TO DETERMINE IF A SPECIFIC COMBINATION OF ACTUAL AXIAL LOAD AND MOMENT CAN BE RESISTED BY THE SECONDARY SIDE HEAD.

$$\frac{P_{\text{ACTUAL}} (\text{\#})}{21,300 \text{ \#}} + \frac{M_{\text{ACTUAL}} (\text{IN.-\#})}{35,700 \text{ IN.-\#}} \leq 1.0$$

W. O. NO. 2282-12BURNS AND ROE, INC.
NEW YORK, N. Y.DATE: 7-8-66

DRAWING NO. _____

BY: M. GIDENTITLE: _____ SHEET 4 OF 9

THE ACTUAL LOADS ON THE STEAM OUTLET NOZZLE
REPORTED IN REF. 2 ARE AS FOLLOWS.

$$M_x = 396.2 \text{ FT. *}$$

$$M_y = 242.67 \text{ FT. *}$$

$$M_z = -274.45 \text{ FT. *}$$

$$F_x = 19.77 \#$$

$$F_y = -45.38 \#$$

$$F_z = -11.80 \#$$

THESE VALUES ARE BASED ON THE COLD MODULUS OF
ELASTICITY WITH NO ALLOWANCE FOR COLD SPRING.

IT IS EVIDENT THAT THE ABOVE LOADS SATISFY THE
EQUATION ON PAGE 3. THEREFORE THE STEAM OUTLET
NOZZLE DESIGN IS ACCEPTABLE.

W. O. NO. 2288-12

BURNS AND ROE, INC.
NEW YORK, N. Y.

DATE: 7-7-46

DRAWING NO. _____

BY: M. GIDEN

TITLE: STEAM OUTLET NOZZLE

SHEET 5 OF 9

WESTINGHOUSE DWG 6615197

DESIGN TEMP. = 600°F

DESIGN PRESSURE = 600 PSI.

$$t = .4375''$$

$$R_M = 15.2187''$$

$$\lambda_0 = 4.375''$$

$$S = 1.82 \frac{\lambda_0}{R_M} \sqrt{\frac{R_M}{t}} \quad (\text{REF. 1})$$

$$S = \frac{1.82(4.375)}{15.2187} \sqrt{\frac{15.2187}{.4375}} = 3.09$$

FROM FIG. A5-2 REF. 1

$$\frac{\sigma t^2}{P} = .13$$

$$\sigma t^2 / \frac{M}{R} \sqrt{\frac{R}{t} \cos \theta} = .20$$

NOZZLE LOADS:

THE SHELL AND NOZZLE SHALL BE CHECKED FOR THE FOLLOWING LOADS APPLIED TO THE PIPE NOZZLE.

1. TORSIONAL MOMENT (M_T) = $\sigma_Y (S)$
2. BENDING MOMENT (M_B) = $\sigma_Y (S)$
3. AXIAL THRUST (P) = $\sigma_Y (A)$

(σ_Y) IS THE 0.2% OFFSET YIELD STRENGTH OF THE MATERIAL AT DESIGN TEMPERATURE. (S) IS THE SECTION MODULUS OF THE CONNECTING PIPE AND (A) IS THE CROSS-SECTIONAL AREA.

THE ABOVE LOADS ARE APPLIED SEPARATELY AND THE STRESSES PRODUCE ARE NOT SUPERIMPOSED. HOWEVER, THE INDIVIDUAL STRESSES ARE COMBINED WITH THE STRESS DUE TO INTERNAL DESIGN PRESSURE.

THE ALLOWABLE STRESS FOR THE ABOVE CONDITION IS THE STRESS INTENSITY " S_p " FOR THE MATERIAL AT DESIGN TEMPERATURE. " S_p " IS EQUAL TO 90% OF THE 0.2% OFF SET YIELD STRENGTH.

THE ATTACHED PIPING IS 6" SCH. 40. (A-106 GRB ASSUMED)
 $A = 5.58 \text{ IN}^2$ $S = 8.50 \text{ IN}^3$ $\sigma_Y = 26,000 \text{ PSI @ } 600^\circ\text{F}$

$$M_T = 26000 (8.5) = 221,000 \text{ IN.}\cdot\text{#}$$

$$M_B = 26000 (8.5) = 221,000 \text{ IN.}\cdot\text{#}$$

$$P = 26000 (5.58) = 145,000 \text{ #}$$

STRESS DUE TO AXIAL LOAD (P)

$$\sigma_x = \sigma_\theta = \frac{.13(145000)}{(.4375)^2} = 98,500 \text{ PSI} *$$

STRESS DUE TO MOMENT (MB)

$$\sigma_x = \sigma_\theta = \frac{.20(221000)5.9}{15.218(.4375)^2} = 89,600 \text{ PSI} *$$

STRESS DUE TO TORSION (MC)

$$\sigma_r = \frac{MT}{2\pi(r_o)^2t} = \frac{221000}{2(3.14)(.4375)^2(.4375)} = 4,200 \text{ PSI}$$

PRESSURE STRESSES:

$$S = \frac{P(R_i + t)}{2t} = \frac{600(15 + 0.4375)}{2(.4375)} = 10,600 \text{ PSI} < S_H = 18000 \text{ PSI @ } 600^\circ\text{F}$$

$$Y = \frac{R_i + t}{R_i} = \frac{15 + 0.4375}{15} = 1.03 \quad Y^3 = 1.09$$

INSIDE SURFACE:

$$\sigma_x = \sigma_\theta = \frac{P}{2} \left[\frac{Y^3 + 2}{Y^3 - 1} \right] = \frac{300(1.09 + 2)}{(1.09 - 1)} = 10,300 \text{ PSI}$$

$$\sigma_r = -600 \text{ PSI}$$

* THE ABOVE STRESSES ARE ABOVE THE ALLOWABLE S_p VALUES FOR THE SHELL MATERIAL ($S_p = 25,000 \text{ PSI @ } 600^\circ\text{F}$) THEREFORE WE CANNOT DESIGN FOR PIPE YIELD LOADS.

OUTSIDE SURFACE:

$$\sigma_r = 0$$

$$\sigma_x = \sigma_\theta = \frac{3P}{2} \left[\frac{1}{Y^{3-1}} \right] = 1.5(600) \left[\frac{1}{.09} \right] = 10,000 \text{ psi}$$

DETERMINATION OF MAXIMUM PIPE LOAD THAT WILL
LIMIT SHELL STRESS TO "S_p",

'S' = MAXIMUM STRESS INTENSITY

$$S = \sigma_x - \sigma_A = \sigma_{x \text{ PIPE}} + \sigma_{x \text{ PRES.}} - \sigma_A$$

$$25000 = \sigma_{x \text{ PIPE}} + 9940 - (-600)$$

$$\sigma_{x \text{ PIPE}} = 25000 - 10540 = 14,460 \text{ psi.}$$

FOR AXIAL LOAD:

$$\sigma_{x \text{ PIPE}} = \frac{.13P}{(.4375)^2} = 14460$$

$$P = \frac{.1915(14460)}{.13} = 21,300 \#$$

FOR MOMENT:

$$\sigma_x = \frac{.20M(5.9)}{15.218(.4375)^2} = 14460$$

$$M = 35,700 \text{ in. lb}$$

W. O. NO. 2288-12

BURNS AND ROE, INC.
NEW YORK, N. Y.

DATE: 7-7-66

DRAWING NO.

BY: M. GIDEN

TITLE: STEAM OUTLET NOZZLE

SHEET 9 OF 2

REFERENCES

1. "TENTATIVE STRUCTURAL DESIGN BASIS FOR REACTOR PRESSURE VESSELS AND DIRECTLY ASSOCIATED COMPONENTS," FEB. 1959 REV.
2. REPORT OF SUBTASK 9.2 CONTRACT AT (30-1)-3345
MARTIN CO. NUCLEAR PROGRAM, BALTIMORE, MARYLAND

W. O. NO. 2283-12

BURNS AND ROE, INC.
NEW YORK, N. Y.

DATE: 7-8-51

DRAWING NO. _____

BY: M. GORDON

TITLE: _____ SHEET 1 OF 10

PM-3/4 STRESS ANALYSIS

ANALYSIS OF STEAM GENERATOR AT

FEEDWATER NOZZLE FOR APPLIED

PIPING LOADS

ITEM 2.5

INTRODUCTION

THE EFFECT OF PIPING MOMENTS AND LOADS APPLIED TO THE FEEDWATER NOZZLE IN THE STEAM GENERATOR IS DETERMINED.

THE NOZZLE CONSISTS OF A FORGED NECK $5\frac{5}{8}$ " OUTSIDE DIAMETER WELDED INTO THE $9\frac{1}{16}$ " SECONDARY SIDE SHELL.

IN LIEU OF ACTUAL PIPING LOADS THE ANALYSIS IS BASED ON USING LOADS THAT WOULD CAUSE YIELDING TO TAKE PLACE IN THE ATTACHED PRIMARY PIPING. THIS IS AN EXTREMELY CONSERVATIVE APPROACH. HOWEVER, IF THE ANALYSIS INDICATES THAT THE SYSTEM CANNOT TOLERATE LOADS OF THIS MAGNITUDE, THEN THE MAGNITUDE OF THE LOADS THAT CAN BE RESISTED IS EASILY DETERMINED. THE RESULTS CAN THEN BE COMPARED WITH ACTUAL CALCULATED PIPING LOADS, OR ANY FUTURE REVISED LOADS TO DETERMINE IF THE SYSTEM IS STRUCTURALLY ADEQUATE.

THE MAXIMUM STRESSES DUE TO APPLIED PIPE LOADS ARE CALCULATED AND THEN COMBINED WITH THE STRESSES DUE TO INTERNAL DESIGN PRESSURE OF 600 PSI. AT 600°F, IN ACCORDANCE WITH THE REQUIREMENTS OF THE "TENTATIVE STRUCTURAL DESIGN BASIS FOR REACTOR PRESSURE VESSELS AND DIRECTLY ASSOCIATED COMPONENTS" (NAVY CODE).

W. O. NO. 2288-12BURNS AND ROE, INC.
157 VERN, N. Y.DATE: 7-8-66

DRAWING NO. _____

BY: M. GLOTTINTITLE: _____ SHEET 2 OF 10SUMMARY

THE RESULTS OF THE ANALYSIS INDICATE THAT THE APPLICATION OF AXIAL LOAD AND MOMENT THAT WOULD CAUSE YIELDING IN THE FEEDWATER PIPING CANNOT BE ADEQUATELY RESISTED BY THE STEAM GENERATOR. THE MAGNITUDE OF THE AXIAL LOAD (P) AND THE MOMENT (M) THAT INDEPENDENTLY CAN BE RESISTED ARE;

AXIAL LOAD (P) = 4100 #

BENDING MOMENT (M) = 8550 IN. - #

SINCE THE ABOVE LOADS ARE THE MAXIMUM INDEPENDENT LOADS THAT CAN SAFELY BE RESISTED, THE FOLLOWING EQUATION MUST BE USED TO DETERMINE IF A SPECIFIC COMBINATION OF ACTUAL AXIAL LOAD AND MOMENT CAN BE RESISTED BY THE STEAM GENERATOR CHANNEL.

$$\frac{P_{ACTUAL}(\#)}{4100 \#} + \frac{M_{ACTUAL}(IN. - \#)}{8550 IN. - \#} \leq 1.0$$

W. O. NO. 2288-12

BURNS AND ROE, INC.
NEW YORK, N. Y.

DATE: 7-8-66

DRAWING NO. _____

BY: M. GIDEN

TITLE: FEEDWATER NOZZLE SHEET 4 OF 10

FEED WATER NOZZLE

WESTINGHOUSE DWG. 661 J 197

NOZZLE LOADS:

THE SHELL AND NOZZLE SHALL BE CHECKED FOR THE FOLLOWING LOADS APPLIED TO THE PIPE NOZZLE.

1. TORSIONAL MOMENT (M_T) = $\sigma_Y(S)$
2. BENDING MOMENT (M_B) = $\sigma_Y(S)$
3. AXIAL THRUST (P) = $\sigma_Y(A)$

(σ_Y) IS THE 0.2% OFFSET YIELD STRENGTH OF THE MATERIAL AT DESIGN TEMPERATURE. (S) IS THE SECTION MODULUS OF THE CONNECTING PIPE AND (A) IS THE CROSS-SECTIONAL AREA.

THE ABOVE LOADS ARE APPLIED SEPARATELY AND THE STRESSES PRODUCED ARE NOT SUPERIMPOSED. HOWEVER, THE INDIVIDUAL STRESSES ARE COMBINED WITH THE STRESS DUE TO INTERNAL DESIGN PRESSURE.

THE ALLOWABLE STRESS FOR THE ABOVE CONDITION IS THE STRESS INTENSITY S_p FOR THE MATERIAL AT DESIGN TEMPERATURE. S_p IS EQUAL TO 90% OF THE 0.2% OFFSET YIELD STRENGTH.

THE ATTACHED PIPING IS 2 1/2" SCH. 80

$$D = 2.254 \text{ IN.} \quad S = 1.339 \text{ IN.}^3 \quad \sigma_Y = 26000 \text{ PSI @ } 600^\circ\text{F}$$

$$M_T = 26000 (1.339) = 34,814 \text{ IN. LB}$$

$$M_B = 26000 (1.339) = 34,814 \text{ IN. LB}$$

$$P = 26000 (2.254) = 58,604 \text{ LB}$$

$$\beta = \frac{0.875 \lambda_0}{a} = \frac{.875(2.812)}{15.28} = .161$$

$$\gamma = a/t = \frac{15.28}{.5625} = 27.2$$

WHERE

a = RADIUS OF THE MIDDLE SURFACE OF THE SHELL

λ_0 = IS THE OUTER RADIUS OF THE ATTACHMENT

t = THE THICKNESS OF THE SHELL

MAXIMUM STRESS DUE TO RADIAL LOADS:

FIG. A.5-4 REF. 1

$$\frac{M\phi}{P} = .10$$

$$\frac{N\phi}{P/a} = 3.9$$

$$\sigma_{\phi} = \frac{M\phi}{I} \times \frac{CP}{t^2} + \frac{N\phi}{P/a} \times \frac{P}{a} + \sigma_{\phi(PRESS.)}$$

$$\sigma_{\phi} = \frac{.10(6)(58000)}{.5625} + \frac{3.9(58000)}{15.28(.5625)} + \sigma_{\phi(PRESS.)}$$

$$\sigma_{\phi} = 62000 + 26400 + \sigma_{\phi(PRESS.)} = 88400 + \sigma_{\phi(PRESS.)} > S_P$$

STRESS DUE TO PRESSURE:

$$\gamma = \frac{R_i + t}{R_i} = \frac{15.5625}{15} = 1.035 \quad \gamma^2 = 1.07$$

INSIDE SURFACE:

$$\sigma_{\phi} = P \left[\frac{Y^2+1}{Y^2-1} \right] = 600 \left[\frac{2.07}{.07} \right] = 17750 \text{ PSI (PAGE 53 REF. 1)}$$

OUTSIDE SURFACE:

$$\sigma_{\phi} = \frac{2P}{Y^2-1} = \frac{2(600)}{.07} = 17150 \text{ PSI} < S_M = 18000 @ 600^{\circ}\text{F}$$

DETERMINATION OF MAXIMUM PIPE LOAD THAT WILL
LIMIT SHELL STRESS TO "S_p".

"S" MAXIMUM STRESS INTENSITY = 24000 PSI @ 600^oF

$$S = \sigma_{\phi} - \sigma_{\lambda} = \sigma_{\phi \text{ PIPE}} + \sigma_{\phi \text{ PRES}} - \sigma_{\lambda \text{ PRES.}}$$

$$\sigma_{\phi \text{ PRES.}} - \sigma_{\lambda \text{ PRES.}} = 17150 - (-600) = 17750 \text{ PSI (INSIDE SURFACE)}$$

$$24000 = \sigma_{\phi \text{ PIPE}} + 17750$$

$$\sigma_{\phi \text{ PIPE}} = 6250 \text{ PSI}$$

AXIAL LOAD:

$$6250 = \frac{.10(6)P}{.5625} + \frac{3.9P}{15.28(.5625)}$$

$$6250 = 1.069P + .455P$$

$$P = \frac{6250}{1.524} = 4100 \#$$

EXTERNAL MOMENT: (LONGITUDINAL)

$$\frac{M_x}{M/a_f} = .085$$

(FIG. A.5-6 TO A.5-9 REF.1)

$$\frac{M_\phi}{M/a_f} = .05$$

$$\frac{N_x}{M/a^2 \rho} = .325$$

$$\frac{N_\phi}{M/a^2 \rho} = 0.90$$

EXTERNAL MOMENT: (CIRCUMFERENTIAL)

$$\frac{M_\phi}{M/a_f} = .088$$

(FIG. A.5-11 TO A.5-13 REF.1)

$$\frac{M_x}{M/a_f} = .048$$

$$\frac{N_x}{M/a^2 \rho} = 2.1$$

$$\frac{N_\phi}{M/a^2 \rho} = 1.2$$

BY INSPECTION, THE EXTERNAL CIRCUMFERENTIAL MOMENT WILL CONTROL SINCE $\sigma_{\phi \text{ PRES}} = 2 \sigma_{x \text{ PRES}}$ AND THE COEFFICIENTS DUE TO THE EXTERNAL MOMENT ARE ALMOST EQUAL.

STRESS DUE TO EXTERNAL MOMENT:

$$\sigma_f = \frac{M_f}{M/a^3} \times \frac{CM}{a^3 t^2} + \frac{N_f}{M/a^3} \times \frac{M}{t a^2}$$

$$\sigma_f = \frac{.088(6)M}{15.23(.161)(.062)^2} + \frac{1.2M}{(.0625)(15.28)^2(.161)}$$

$$\sigma_f = .676M + .056M$$

$$\sigma_f = .732M$$

$$\sigma_f \text{ AVAILABLE PRESS.} = 6250 \text{ PSI (PAGE 6)}$$

$$\therefore M = \frac{6250}{.732} = 8550 \text{ IN. LB}$$

STRESS DUE TO TORSION:

$$\sigma_T = \frac{M_T}{2\pi(r_o)^2 t} = \frac{34300}{2(3.14)(2.81)^2(.0625)} = 1250 \text{ PSI}$$

W. O. NO. 27,38-12

BURNS AND ROE, INC.
NEW YORK, N. Y.

DATE: 7-11-66

DRAWING NO.

BY: M. J. DEN

TITLE: FEEDWATER NOZZLE

SHEET 10 OF 10

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W. O. NO. 2288-12

BURNS AND ROE, INC.
NEW YORK, N. Y.

DATE: 10/6

DRAWING NO. _____

BY: M. W. D. E. D.

TITLE: _____ SHEET 1 OF 10

STRESS ANALYSIS

FEEDWATER NOZZLE THERMAL-

TRANSIENT ANALYSIS

PM - 3A

ITEM 2.6

W. O. NO. 2285-12

DRAWING NO. _____

TITLE: _____

BURNS AND ROE, INC.
NEW YORK, N. Y.

DATE: 10/6

BY: M. G. E.

SHEET 2 OF 10

INTRODUCTION AND SUMMARY

An investigation was made of the effect of the cold water transient on the feed water inlet nozzle. The nozzle, shown on Sheet 3, is protected with an internal thermal sleeve which is located in the vicinity of the steam generator shell. The critical nozzle section investigated is Section AA, which is in direct contact with the feed water. A fatigue evaluation, to determine the allowable number of cycles of operation of the transient was made for two possible initial conditions. The first assumes that the feed water valve is closed. The ambient temperature of the nozzle would then approach the maximum normal shell side operating temperature of 463°F. Feed water is then injected into the system and drops in temperature to 80°F in one second.

The second condition assumes the feed water at a temperature of 343°F is being injected into the system and then dropped to 80°F in one second. For this condition the ambient temperature is taken as 343°F.

The results of the analysis indicates the following:

For condition 1, the number of cycles of operation allowed during the expected life of the system is 250,000.

For condition 2, the number of cycles is 350,000.

Based on the results stated above, an over-all evaluation of the effect of the cold water transient on plant operation can be made.

MATERIAL ASTM A-105 GR. II

OPERATING PRESSURE = 300 PSI.

PRESSURE STRESS SECTION A-A (REF. 1)

$$Y = \frac{R_o + t}{R_i} = \frac{1.16 + 0.963}{1.16} = 1.83 \quad Y^2 = 3.35$$

$$\sigma_{HOOP} = P \left[\frac{Y^2 + 1}{Y^2 - 1} \right] = 300 \left[\frac{4.35}{2.35} \right] = 555 \text{ PSI (INSIDE)}$$

$$\sigma_{HOOP} = \frac{2P}{Y^2 - 1} = \frac{2(300)}{2.35} = 255 \text{ PSI (OUTSIDE)}$$

$$\sigma_{AXIAL} = \frac{P}{Y^2 - 1} = \frac{300}{2.35} = 127.5 \text{ PSI (INSIDE & OUTSIDE)}$$

THERMAL STRESS AT SECTION A-A

THE THERMAL STRESS AT SECTION A-A THE CRITICAL SECTION OF THE NOZZLE WILL BE INVESTIGATED FOR TWO TRANSIENT CONDITIONS.

THE FIRST CONDITION ASSUMES THAT THE FEEDWATER IS NOW SHUT OFF AND THE NOZZLE WALL IS AT AN AMBIENT TEMPERATURE OF 463°F, WHICH CORRESPONDS TO THE MAXIMUM NORMAL OPERATING TEMPERATURE OF THE SHELL SIDE OF THE STEAM GENERATOR. AT THIS TIME FEEDWATER IS INJECTED INTO THE SYSTEM AND DROPS THE TEMPERATURE TO 80° IN ONE SECOND.

THE SECOND CONDITION ASSUMES THAT THE FEEDWATER IS BEING SUPPLIED TO THE SYSTEM AT A TEMPERATURE OF 343°F AND THEN THE FEEDWATER DROPS TO 80°F IN ONE SECOND. THE AMBIENT TEMPERATURE OF THE NOZZLE FOR THIS CONDITION IS TAKEN AS 343°F.

THE PRIMARY DIFFERENCE BETWEEN THE TWO CONDITIONS ANALYZED IS THE AMBIENT TEMPERATURE OF THE NOZZLE WALL.

AREA OF FEEDWATER NOZZLE = 0.02938 FT.²

FLOW RATE = 30,000 #/HR.

DIA. OF NOZZLE = 0.1936 FT.

$$R.N. = \frac{G}{\mu}$$

R.N. = REYNOLDS No.

G = MASS FLOW RATE / UNIT AREA

μ = VISCOSITY

$$G = \frac{m}{A} = \frac{36000 \text{ lb/hr}}{0.02938 \text{ ft}^2} = 1.235 \times 10^6 \text{ lb/ft}^2\text{-hr} = 340.3 \text{ lb/ft}^2\text{-sec.}$$

$$\mu (280^\circ\text{F}) = 0.578 \times 10^{-3} \text{ lb/ft-sec. (REF. 2 PAGE 537)}$$

$$R.N. = \frac{340.3 \text{ lb/ft}^2\text{-sec.} \times 0.1936 \text{ ft.}}{0.578 \text{ lb/ft-sec.} \times 10^{-3}} = 1.14 \times 10^5$$

$$\frac{hD}{k} = 0.023 \left[\frac{GD}{\mu} \right]^{0.8} \times \left[\frac{\mu C_p}{k} \right]^{0.4} \quad (\text{REF. 3})$$

h = heat transfer coefficient

k = thermal conductivity = 0.353 BTU/hr-ft-°F

$$\left(\frac{\mu C_p}{k} \right) = 5.89 \text{ @ } 280^\circ\text{F (REF. 2)}$$

$$\frac{hD}{k} = 0.023 (1.14 \times 10^5)^{0.8} \times (5.89)^{0.4} = 5.19 \times 10^2$$

$$h = \frac{5.19 \times 10^2 (0.353)}{0.1936} = 946.3 \text{ BTU/hr-ft}^2\text{-°F}$$

WHEN A VESSEL WALL IS INSULATED ON THE OUTSIDE AND IS IN CONTACT WITH A FLUID ON THE INSIDE, AND THE TEMPERATURE OF THE FLUID CHANGES LINEARLY WITH TIME, THE STRESS DUE TO THE CHANGE IN FLUID TEMPERATURE IS:

$$\text{INSIDE } \sigma_{\text{HOOP}} = \sigma_{\text{AXIAL}} = \frac{E \alpha \Delta T_f}{1-\nu} \quad (\text{REF. \#1})$$

$$\text{OUTSIDE } \sigma_{\text{HOOP}} = \sigma_{\text{AXIAL}} = \frac{E \alpha \Delta T_f}{1-\nu} ?$$

WHERE :

$$E_d = 186 \text{ PSI (FIG. 4.1-3 REF. 1)}$$

 ΔT_f = CHANGE IN FLUID TEMPERATURE

$$\nu = \text{POISSONS RATIO} = 0.3$$

$$\frac{12K}{ht}$$

K = THERMAL CONDUCTIVITY OF NOZZLE WALL = 25.8 BTU/HR-FT-°F

t = WALL THICKNESS = 0.963"

h = HEAT TRANSFER COEFFICIENT BETWEEN FLUID & WALL = 946.3

$$\frac{12(25.8)}{946.3(0.963)} = 0.341$$

$$\lambda = \frac{\epsilon \theta}{t^2}$$

 ϵ = THERMAL DIFFUSIVITY, .018 IN²/SEC. θ = TIME FOR FLUID CHANGE, 1 SEC.

t = WALL THICKNESS, = 0.963"

$$\lambda = \frac{.018(1)}{(0.963)^2} = .01945$$

FROM FIG. A.3-5 (REF. #1)

$$\eta_i = 0.43$$

FROM FIG. A.3-6

$$\eta_o = 0.10$$

THERMAL STRESS SECTION A

$$\text{INSIDE: } \sigma_H = \sigma_A = \frac{E \alpha \Delta T_f \gamma_c}{1-\nu} = \frac{186(0.42) \Delta T_f}{0.7} = 114 \Delta T_f$$

$$\text{OUTSIDE: } \sigma_H = \sigma_A = \frac{E \alpha \Delta T_f \gamma_o}{1-\nu} = \frac{186(0.10) \Delta T_f}{0.7} = 26.6 \Delta T_f$$

CONDITION-1

$$\Delta T_f = 463 - 180 = 283^\circ \text{F}$$

$$\text{INSIDE: } \sigma_H = \sigma_A = 114(283) = 32,300 \text{ PSI.}$$

$$\text{OUTSIDE: } \sigma_H = \sigma_A = 26.6(283) = 7530 \text{ PSI.}$$

CONDITION-2

$$\Delta T_f = 343 - 180 = 163$$

$$\text{INSIDE: } \sigma_H = \sigma_A = 114(163) = 18,600 \text{ PSI}$$

$$\text{OUTSIDE: } \sigma_H = \sigma_A = 26.6(163) = 4330 \text{ PSI.}$$

COMBINED STRESS, PRESSURE + THERMALCONDITION-1

$$\sigma_H = 32,300 + 1420 = 33,720 \text{ PSI (INSIDE)}$$

$$\sigma_H = 7530 + 1120 = 8650 \text{ PSI (OUTSIDE)}$$

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CONDITION - 2

$$\sigma_H = 30000 + 1420 = 31420 \text{ PSI (INSIDE)}$$

$$\sigma_H = 7000 + 1120 = 8120 \text{ PSI (OUTSIDE)}$$

FATIGUE ANALYSIS

CONDITION - 1

$$S_{MAX.} = 33720 + 300 = 34020 \text{ PSI.}$$

$$S_{MIN.} = 1420 + 300 = 1720 \text{ PSI}$$

$$S_{ALT.} = \frac{34020 - 1720}{2} = 16150 \text{ PSI}$$

N = 250,000 CYCLES (FIG. N-415(A) REF #4

CONDITION - 2

$$S_{MAX} = 31420 + 300 = 31720 \text{ PSI.}$$

$$S_{MIN} = 1420 + 300 = 1720 \text{ PSI.}$$

$$S_{ALT} = \frac{31720 - 1720}{2} = 15000 \text{ PSI}$$

N = 350,000 CYCLES (FIG N-415(A) REF. # 4

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NUCLEAR VESSELS. 1955.

W. O. NO. 2288-12

BURNS AND ROE, INC.
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BY R.J. Duggan

TITLE: _____ SHEET 1 OF 10

STRESS ANALYSIS

REACTOR COOLANT PUMP

PM-3A

ITEM 3.0

W. O. NO. 2288-12

BURNS AND ROE, INC.
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DATE: 9/66

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BY: B. J. DOBBS

TITLE: PM-3A STRESS ANALYSIS

SHEET 2 OF 10

INTRODUCTION

THE REACTOR COOLANT PUMP IS AN INTEGRALLY DESIGNED PUMP AND MOTOR UNIT. EACH UNIT CONSISTS OF A SINGLE STAGE CENTRIFUGAL PUMP, DRIVEN BY A CANNED MOTOR. THE PUMP SUCTION AND DISCHARGE NOZZLES ARE WELDED INTO THE PRIMARY PIPING SYSTEM.

THE MATERIALS OF CONSTRUCTION OF THE BASIC PUMP COMPONENTS ARE: PUMP CASE WALL IS ASTM-A351 CF8M STEEL CASTING FOR HIGH TEMPERATURE, STATOR SHELL IS ASTM A-362 GRCF-B CENTRIFUGAL STEEL CASTING, MOTOR COVER IS ASTM A-336 CLFB STEEL FORGING AND THE COOLING COIL IS ASTM A-312 TYPE 304 SS TUBING.

THE DESIGN BASIS USED FOR THE ANALYSIS IS THE ASME BOILER AND PRESSURE VESSEL CODE SECTION VIII UNFIRED PRESSURE VESSELS.

THE SPECIFIED DESIGN CONDITIONS ARE PRESSURE OF 1485 PSIG AND TEMPERATURE OF 600°F.

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NEW YORK, N. Y.

BY: R. J. DOBBS

TITLE: PM-3A STRESS ANALYSIS

SHEET 3 OF 10

SUMMARY

THE REACTOR COOLANT PUMP WAS ANALYZED IN ACCORDANCE WITH ASME BOILER AND PRESSURE VESSEL CODE SECTION VIII UNFIRED PRESSURE VESSELS.

THE MINIMUM REQUIRED THICKNESS OF THE CRITICAL PUMP COMPONENTS (CASE WALL, STATOR SHELL, MOTOR COVER AND COOLING COIL) WAS CALCULATED FOR THE SPECIFIED DESIGN CONDITIONS. IN EACH CASE THE ACTUAL THICKNESS IS GREATER THAN THE MINIMUM REQUIRED AND THEREFORE ACCEPTABLE.

THE MOTOR COVER AND PUMP CASE BOLTING WAS CHECKED AND FOUND TO BE ADEQUATE.

THE REACTOR COOLANT PUMP IS ACCEPTABLE FOR THE DESIGN CONDITIONS INVESTIGATED.

ANALYSIS
OF

PM3A REACTOR COOLANT PUMP STRESSES

I CASE WALL THICKNESS (ITEM 1 DWG IF-4454-A)

CONSIDER THE CASE WALL A CYLINDRICAL SHELL. (THIS IS A POINT AT WHICH THE WALLS ARE THINNEST)

FROM ASME SECTION VIII UG 27

$$t = \frac{PR}{SE - 0.6P}$$

Where: $P = 1485$ psig
 $t_{sat} = 596.23^\circ F \approx 600^\circ F$

$R = 7.75$ in

$S = 15,350$ psi (Table UHA-23 for CF8M, A351, @ $600^\circ F$)

$E = 1.0$ (UG-24(5))

$$t = \frac{1485 \times 7.75}{15350 \times 1 - 0.6 \times 1485} = 0.797 \quad \text{which is less than } 1.25''$$

Disregarding the double volute wall and extending to the outer walls

$R = 9.75$

$$t = \frac{1485 \times 9.75}{15350 \times 1 - 0.6 \times 1485} = 1.00 \quad \text{This also is satisfactory.}$$

The above method has proved to be sufficient in pump case designs now in service.

PREPARED BY D.P.W.

CHECKED BY deagles 7/22/55

REVISED BY _____

ANALYSIS
OF

II STATOR SHELL THICKNESS (ITEM 9-4 DWG IP 4454-A)

The material of construction is A362 Gr CF-8. A362 is a specification for Tubular Centrifugal castings which in turn references a more general casting specification, A296, for chemicals and physicals. Since neither A362 nor A296 are listed in ASME Section VIII a comparison of the chemical and physical properties with a listed material follows:

	A351 CF8	A296 CF8	Actual
Carbon (max.%)	0.08	0.08	.05
Manganese (max.%)	1.50	1.50	.88
Silicon (max.%)	2.00	2.00	1.24
Phosphorus (max.%)	0.04	0.05	.018
Sulphur (max.%)	0.04	0.05	.015
Chromium (%)	18.0-21.0	18.0-21.0	20.00
Nickel (%)	8.0-11.0	8.0-11.0	9.50
Tensile Strength (min psi)	70,000	70,000	*
Yield point (min psi)	28,000	28,000	*
Elongation in 2 in. (min %)	35.0	35.0	*

* Physicals not reported on BJ Metallurgical Laboratory Test Report in Martin files. However, the heat no. is, and further checking could determine these values.

The above comparison is to justify the use of allowable stress values for CF8 A351 from Table UHA-23 of ASME Section VIII in the formula

$$t = \frac{PR}{SE - 0.6P}$$

Where:

$$P = 1485 \text{ psig}$$

ANALYSIS
OF

$$R = 5.00 \text{ in}$$

$$S = 14,350 \text{ psi @ } 600^\circ\text{F (1)}$$

$$E = 1.00 \text{ (2)}$$

$$t = 0.668 \text{ in.}$$

$$D = 10.00 \text{ in.}$$

$$t = \frac{1485 \times 5.00}{14350 \times 1 - 0.6 \times 1485} = .551 \text{ in. which is less than } 0.668 \text{ in.}$$

NOTES

(1) This cylinder would only be so stressed after failure of the Inconel stator can. [No credit is taken for the motor punchings or the cooling coils about the OD of this cylinder] Since failure would destroy the pump's electrical characteristics, the higher allowable stress value, which allows for slightly greater deformation, was used.

(2) It was felt that the requirements of UG-24 (5) and Appendix VII were met. The material was dye penetrant inspected to a level satisfying UA-80(a)(2) and since it is a centrifugal casting the critical sections of UA-80(a)(1) are non-existent.

III MOTOR COVER THICKNESS (ITEM 26 DWG IF 4454-H)

Assume a flat head design

$$t = D \sqrt{\frac{C \cdot P}{S}} \quad \text{From UG 34(c)(2) of ASME VIII}$$

Where:

$$D = 9.00 \text{ in } \phi \text{ of gasket}$$

$$C = 0.25 \quad \text{From Fig. UG-34(p)}$$

$$P = 1485 \text{ psig}$$

$$S = 11,600 \text{ psi for A336 Cl. F 8}$$

@ 600°F [THE LOWER ALLOWABLE]

PREPARED BY D.P.W.

CHECKED BY du Gas 7/27/65

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ANALYSIS
OF

III (Cont.)

$$t = 9.00 \sqrt{\frac{2.5 \times 1485}{11,600}} = 1.610 \text{ in which is less than } 1.750 \text{ in.}$$

IV COOLING COIL THICKNESS (HEM 9-C DWG IF 4454-A)

$$t = \frac{PR}{SE - 0.6P} \quad \frac{7}{8} \text{ OD } 5 - 0.120 \text{ in}$$

Where:

$$P = 1485 \text{ psig}$$

$$R = .875 \text{ in} / 2 - .120 \text{ in} = .318 \text{ in}$$

$$S = 11,600 \text{ psi for A 312 TP 304 @ 600°F [LOWER VALUE]$$

$$E = 1.0 \text{ Seamless}$$

$$t = \frac{1485 \times .318}{11,600 \times 1 - 0.6 \times 1485} = 0.044 \text{ in which is less than } 0.22 \text{ in.}$$

V MOTOR COVER BOLTING (ITEM 30 DWG IF 4454-A)

$$W_{m1} = H + H_p = 0.785 G^2 P + (2b \times 3.14 G_m P) \quad \text{UA-49(1)}$$

Where:

$$G = 9.00 \text{ in } \text{FROM DRAWING}$$

$$P = 1485 \text{ psig}$$

$$b = .353 \quad (b = \frac{\sqrt{b_o}}{2} = \frac{\sqrt{.5}}{2} = \frac{.707}{2})$$

$$m = 3.00$$

TABLE UA 49.1

$$W_{m1} = 0.785 \times 9.00^2 \times 1485 + (2 \times .353 \times 3.14 \times 9.00 \times 3.00 \times 1485)$$

$$W_{m1} = 94,500 + 89,000 = 183,500 \text{ lbs.}$$

PREPARED BY DAW CHECKED BY du Gas 7/22/15 REVISED BY _____

ANALYSIS
OF

V (Cont.)

$$W_{m2} = 3.14 b G y \quad \text{UA 49(a)(2)}$$

Where $y = 4500$ from Table UA 491

$$W_{m2} = 3.14 \times .353 \times 9.00 \times 4500 = 45,000 \text{ lb}$$

$$A_{m1} = \frac{W_{m1}}{S_b} \quad \cdot \quad A_{m2} = \frac{W_{m2}}{S_a}$$

Where

$$S_a = S_b = 20,000 \text{ psi} \quad -20 \text{ to } 650^\circ\text{F from Table UCS-23 for SA 193 B7}$$

$$A_{m1} = \frac{183,500}{20,000} = 9.175 \text{ in}^2$$

$$A_{m2} = \frac{45,000}{20,000} = 2.25 \text{ in}^2$$

Minimum Stud diameter = $1\frac{3}{8}$ " and there are 10 studs

$$A = 1.375^2 \times \frac{\pi}{4} \times 10 = 14.82 \text{ in}^2 \quad \text{which is greater than } 9.175 \text{ in}^2$$

$$\text{Actual stud stress} = \frac{183,500}{14.82} = 12,400 \text{ psi}$$

VI Pump Case Boltting (Item 35 DWG 1F4154-A)

The constants are:

$$G = 14.00 \text{ in } \phi \text{ Gasket}$$

$$b = .390 \quad (b = \frac{\sqrt{6.25}}{2})$$

$$m = 3.00 \quad \left. \begin{array}{l} \\ \end{array} \right\} \text{Table UA 49.1}$$

$$y = 4500 \quad \left. \begin{array}{l} \\ \end{array} \right\}$$

$$S_a = S_b = 20,000 \text{ psi}$$

Bolt Minimum Diameter = $1\frac{5}{8}$ " and there are 16 of them

PREPARED BY DWY

CHECKED BY du Gas 7/27/55

REVISED BY _____

ANALYSIS
OF

VI (Cont.)

$$W_{m1} = 0.785 \times 14.00^2 \times 1485 + (2 \times 3.14 \times 3.14 \times 14.00 \times 3.00 \times 1485)$$

$$W_{m1} = 228,500 + 155,000 = 383,500 \text{ lbs.}$$

$$A_{m1} = \frac{383,500}{20,000} = 19.175 \text{ in}^2$$

$$A_{m2} = \frac{3.14 \times 3.14 \times 14 \times 450}{20,000} = 5.92 \text{ in}^2$$

$$A = 1.625^2 \frac{\pi}{4} \times 16 = 33.2 \text{ in}^2 \text{ which is greater than } 19.18 \text{ in}^2$$

$$\text{Actual bolt stress} = \frac{383,500}{33.2} = 11,600 \text{ psi}$$

Conclusions

While a minimum of areas have been renewed, these areas constitute the highest stressed areas. The remaining areas are deemed satisfactory by inspection and service.

W. O. NO. 2288-12

BURNS AND ROE, INC.
NEW YORK, N. Y.

DATE: 9/66

DRAWING NO.

BY: R. J. DOBBS

TITLE: PM-3A STRESS ANALYSIS

SHEET 10 OF 10

REFERENCES

- 1 - ASME BOILER AND PRESSURE VESSEL CODE
SECTION VIII UNFIRED PRESSURE VESSELS.
1965.

W. O. NO. 2283-12

BURNS AND ROE, INC.
NEW YORK, N. Y.

DATE: 12/66

DRAWING NO. _____

BY: M. GIDEN

TITLE: _____ SHEET 1 OF 19

STRESS ANALYSIS

REACTOR COOLANT PIPING

PM - 3A

ITEM 4.0

INTRODUCTION AND SUMMARY

THE PRIMARY COOLANT PIPING WAS ANALYZED FOR THE EFFECT OF THERMAL EXPANSION. THE ANALYSIS WAS PERFORMED TO CHECK THE INTEGRITY OF THE PIPING AND TO DETERMINE THE THERMAL LOADS ACTING ON THE STEAM GENERATOR AND REACTOR PRIMARY NOZZLES.

THE CRITICAL PORTION OF THE PRIMARY PIPING IS THE STEAM GENERATOR OUTLET TO REACTOR INLET. THIS PORTION WAS ANALYZED AS TWO INDEPENDENT PROBLEMS. ONE A VERTICAL LOOP BETWEEN THE DISCHARGE NOZZLE OF THE STEAM GENERATOR AND THE PUMP SUCTION NOZZLE. THE OTHER IS A HORIZONTAL RUN BETWEEN THE PUMP DISCHARGE NOZZLE AND THE REACTOR INLET NOZZLE.

THE MARTIN ANALYSIS FOR THE STEAM OUTLET TO PUMP INLET ASSUMES FULL FIXITY AT THE PUMP AND STEAM GENERATOR. THIS IS CONSERVATIVE, SINCE THE STEAM GENERATOR AND PUMP ARE ON ROLLER SUPPORTS WHICH ALLOW AXIAL MOTION TO TAKE PLACE. THE RESULTS INDICATE THAT THE PIPING IS ACCEPTABLE AND THEREFORE THERE IS NO NEED FOR A MORE EXACT ANALYSIS. HOWEVER THIS APPROACH IS TOO CONSERVATIVE FOR DETERMINING THE LOADS ACTING ON THE STEAM GENERATOR NOZZLES.

INCLUDED WITH THE ORIGINAL MARTIN ANALYSIS WAS AN INDEPENDENT COMPUTER PROGRAM ANALYSIS OF THE PRIMARY PIPING LOOP (REF. P. 16 TO 18). THEREFORE,

FOR APPLIED STEAM GENERATOR NOZZLE LOADS THE RESULTS OBTAINED FROM THE PIPING PROGRAM (SHEET 17) SHOULD BE USED, SINCE THIS ANALYSIS CORRECTLY CONSIDERED THE MOVABLE SUPPORT. THE STEAM GENERATOR MAXIMUM NOZZLE LOADS WOULD THEREFORE BE APPROXIMATELY EQUAL TO:

$$\left. \begin{array}{ll} M_x = 5783 & \text{FT. \# (TORSION)} \\ M_y = +2 & \text{FT. \# (BENDING)} \\ M_z = -297 & \text{FT. \# (BENDING)} \end{array} \right\} \begin{array}{l} \text{REF SHEET 17} \\ \text{POINT \# 4} \end{array}$$

$$F_x = +1\#$$

$$F_y = -1395\#$$

THE MARTIN ANALYSIS FOR THE PUMP TO REACTOR INLET DID CONSIDER THE FACT THAT THE PUMP IS ON SLIDING SUPPORTS. THEREFORE, THE APPROACH USED IS CORRECT. THE PRIMARY EFFECT CONSIDERED IS THE VERTICAL GROWTH OF THE REACTOR PRESSURE VESSEL. A CHECK OF THIS ANALYSIS INDICATES THAT THE PIPING IS ACCEPTABLE. THE CALCULATED REACTOR NOZZLE LOADS ARE:
 $M_L = 137,000 \text{ IN. \#}$, TORQUE = $79,500 \text{ IN. \#}$, SHEAR = $3280\#$
 (REF. P 15) WERE USED IN THE ANALYSIS OF THE REACTOR PRESSURE VESSEL.

ANALYSIS
OF

REACTOR COOLANT PIPING

ST'M GEN OUTLET TO PUMP

(FULLY RESTRAINED)

PREPARED BY J. E. J.

CHECKED BY _____

REVISED BY _____

ANALYSIS
OF

$$M_0 \sum_{s=0}^{13} \frac{y \Delta s}{I} + F_x \sum_{s=0}^{13} \frac{y^2 \Delta s}{I} + F_y \sum_{s=0}^{13} \frac{xy \Delta s}{I} = E \alpha \Delta T L_x$$

$$M_0 \sum_{s=0}^{13} \frac{x \Delta s}{I} + F_x \sum_{s=0}^{13} \frac{xy \Delta s}{I} + F_y \sum_{s=0}^{13} \frac{x^2 \Delta s}{I} = -E \alpha \Delta T L_y$$

$$M_0 \sum_{s=0}^{13} \frac{\Delta s}{I} + F_x \sum_{s=0}^{13} \frac{y \Delta s}{I} + F_y \sum_{s=0}^{13} \frac{x \Delta s}{I} = 0$$

$$\Delta T = (470 - 40) = 430$$

$$\alpha = 10^{-5}$$

$$E = 28 \times 10^6 \text{ PSI}$$

$$E \alpha = 280$$

$$L_x = 40$$

$$L_y = 10$$

$$178.56 M_0 + 8978.19 F_x + 4000.93 F_y = 4,816,000$$

$$88.24 M_0 + 4000.93 F_x + 3143.56 F_y = -1,204,000$$

$$4.22 M_0 + 178.56 F_x + 88.24 F_y = 0$$

SOLUTION OF THESE EQUATIONS YIELDS:

$$M_0 = -121,309.$$

$$F_x = 3,702.$$

$$F_y = -1,689.$$

PREPARED BY

J. G. B.

CHECKED BY

REVISED BY

GRAPHICAL INTEGRATION - RECTANGULAR RULE

①	②	③	④	⑤	⑥	⑦	⑧	⑨	⑩	⑪
ELEM	ΔS	I	$\frac{\Delta S}{I}$	X	Y	$X \frac{\Delta S}{I}$	$Y \frac{\Delta S}{I}$	$X^2 \frac{\Delta S}{I}$	$Y^2 \frac{\Delta S}{I}$	$XY \frac{\Delta S}{I}$
			③ ÷ ④			⑤ ④	⑥ ④	⑤ ⑦	⑥ ⑦	⑥ ⑦
1	10	40.5	.2469	0	5	0	1.2345	0	6.1725	0
2	10	40.5	.2469	0	15	0	3.7035	0	55.5525	0
3	10	40.5	.2469	0	25	0	6.1725	0	154.3125	0
4	10	40.5	.2469	0	35	0	8.6415	0	302.4525	0
5	10	2500	.0400	0	45	0	1.80	0	81.0	0
6	64	73.1	.8755	4.5	57.5	3.9398	50.3413	17.7291	2874.6248	226.5395
7	8	105.7	.0757	16.	62	1.2112	4.6934	19.3792	290.9708	75.0944
8	8	105.7	.0757	24.	62	1.8168	4.6934	43.6032	290.9708	112.6416
9	86	73.1	1.1765	35.5	57.5	41.7658	67.6481	152.8859	3289.8060	2401.5335
10	10	40.5	.2469	40.	45	9.8760	11.1105	395.0400	499.9725	444.42
11	10	40.5	.2469	40	35	9.8760	8.6415	395.0400	302.4525	345.66
12	10	40.5	.2469	40	25	9.8760	6.1725	395.0400	154.3125	246.9
13	10	40.5	.2469	40	15	9.8760	3.7035	395.0400	55.5525	148.14
$\Sigma 256$			$\Sigma 4.2186$			$\Sigma 88.2376$	$\Sigma 178.5569$	$\Sigma 3143.5574$	$\Sigma 8971.1924$	$\Sigma 4000.928$

PREPARED BY

CHECKED BY

REVISED BY

ANALYSIS
OF

ASSUME EQUIVALENT TO A PIPE OF
FOLLOWING DIMENSIONS.

$$OD = 9 \quad ID = 5.761$$

$$I = \frac{\pi}{4} (R_o^4 - R_i^4) = \frac{\pi}{4} (4.5^4 - 2.88^4)$$

$$I = \frac{\pi}{4} (410 - 68) = \frac{\pi}{4} (342) = 268 \text{ IN}^2$$

USE I OF 250 IN⁴ TO SIMULATE FLANGE

PREPARED BY _____ CHECKED BY _____ REVISED BY _____

ANALYSIS
OF

$$\begin{aligned} 178.56 M_o + 8978.19 F_x + 4000.93 F_y &= 4,816,000 \\ 88.24 M_o + 4000.93 F_x + 8143.56 F_y &= -1,204,000 \\ 4.22 M_o + 178.56 F_x + 88.24 F_y &= 0 \end{aligned}$$

$$\begin{aligned} -2.03357 \\ 178.56 M_o + 8978.19 F_x + 4000.93 F_y &= 4,816,000 \\ -178.56 M_o - 8096.16 F_x - 6361.21 F_y &= 2,436,378 \\ \hline 882.03 F_x - 2360.28 F_y &= 7,252,378 \end{aligned}$$

$$\begin{aligned} -42.31271 \\ 178.56 M_o + 8978.19 F_x + 4000.93 F_y &= 4,816,000 \\ -178.56 M_o - 7555.37 F_x - 3733.68 F_y &= 0 \\ \hline 1422.82 F_x + 267.25 F_y &= 4,816,000 \end{aligned}$$

$$\begin{aligned} -61992 \\ 882.03 F_x - 2360.28 F_y &= 7,252,378 \\ -882.03 F_x - 165.67 F_y &= -2,985,535 \\ \hline -2525.95 F_y &= 4,266,843 \end{aligned}$$

$$F_y = -1,689,203$$

$$\begin{aligned} 8.83173 \\ 882.03 F_x - 2360.28 F_y &= 7,252,378 \\ 12565.96 F_x + 2360.28 F_y &= 42,533,612 \\ \hline 13447.99 F_x &= 49,785,990 \end{aligned}$$

$$F_x = 3702.114$$

$$\begin{aligned} -2.24425 \\ 178.56 M_o + 8978.19 F_x + 4000.93 F_y &= 4,816,000 \\ -191.01 M_o - 8978.19 F_x - 7054.23 F_y &= 2,701,806 \\ -19.45 M_o - 3053.30 F_y &= 7,517,806 \end{aligned}$$

$$\begin{aligned} -50.281012 \\ 178.56 M_o + 8978.19 F_x + 4000.93 F_y &= 4,816,000 \\ -212.19 M_o - 8978.19 F_x - 4436.80 F_y &= 0 \\ -33.63 M_o - 435.87 F_y &= 4,816,000 \end{aligned}$$

PREPARED BY _____ CHECKED BY _____ REVISED BY _____

ANALYSIS
OF

$$-7.00507$$

$$-19.45 M_o - 2053.30 F_y = 7,517,806$$

$$+235.58 M_o + 2053.30 F_y = -33,736,417$$

$$216.13 M_o$$

$$-26,218,611$$

$$M_o = -121,309,448$$

PREPARED BY _____

CHECKED BY _____

REVISED BY _____

ANALYSIS
OF

PUMP REACTIONS

- BASED ON $E_c = 28 \times 10^6$

$$M_{PUMP} = 121,309 + (40 \times 1689) - (10 \times 3702)$$

$$M_{PUMP} = 121,309 + 67560 - 37020$$

$$M_{PUMP} = 151,789 \text{ IN-LBS}$$

WA: 250,000 LBS
FOR 8" PIPE.

BENDING STRESS

$$\sigma = \frac{M}{Z} = \frac{151,789}{12.23} = 12,400 \text{ PSI.}$$

PREPARED BY _____

CHECKED BY _____

REVISED BY _____

ANALYSIS
OF

$$E_C = 28 \times 10^6 \text{ psi.}$$

$$E_{HOT} = 26$$

$$\frac{E_b}{E_C} = \frac{26}{28} = .93$$

$C = 0$ NO COLD SPRING

$$R_H = \left(1 - \frac{2}{3}C\right) \frac{E_b}{E_C} R = .93 R$$

$$F_x = .93 \times 3702 = 3440 \text{ Lb}$$

$$F_y = .93 \times 1689 = 1570 \text{ Lb.}$$

$$M_{PUMP} = .93 \times 151,781 = 141,000 \text{ IN-Lb}$$

$$M_{STGEN} = .93 \times 121,309 = 113,000 \text{ IN-Lb}$$

$$\sigma_E = \frac{M_i}{Z} = \frac{141,000 \times 1.3}{12.23} = 15,000 \text{ PSI.}$$

PREPARED BY _____

CHECKED BY _____

REVISED BY _____

ANALYSIS
OF
PRIMARY

REACTOR
TO
PUMP

VIRTUAL LENGTH OF ELBOWS

$$L = \frac{.58}{2} = 29 \text{ INCHES}$$

$$\begin{aligned} S &= \sum (M_P - P_i) \times \frac{\Delta S}{EI} & \theta &= \sum (M_P - P_i) \frac{\Delta S}{EI} \\ &= M_P \sum \frac{\Delta S}{EI} - P \sum \frac{\Delta S}{EI} & \theta &= M_P \sum \frac{\Delta S}{EI} - P \sum \frac{\Delta S}{EI} \end{aligned}$$

ANALYSIS
OF

DIFFERENTIAL EXPANSION

$$\Delta = \alpha \Delta T L = 10^{-5} \times 400 \times 60 = 0.240$$

$$\begin{aligned} \Delta &\approx 48(480 - 70)(10^{-5}) + 20(120 - 70)(10^{-5}) \\ &= 0.192 + 0.010 \\ &= .202'' \end{aligned}$$

PREPARED BY _____

CHECKED BY _____

REVISED BY _____

ANALYSIS
OF

$EI \sim \text{CONSTANT}$

ELEM	ΔS	X	X^2	$\Delta S X$	$\Delta S X^2$
1	29	3.2	10.24	92.8	296.96
2	5	8.5	72.25	42.5	361.25
3	5	12.2	148.84	61.0	744.20
4	29	17	289.0	493	8381
5	8	25	625	200	5000
6	8	33	1089	264	8712
7	8	41	1681	328	13448
8	8	49	2401	392	19208
9	8	57	3249	456	25992
10	8	65	4225	520	33800
11	8	73	5329	584	42632
	124.			3,433.3	158,575

$$-.207 = M_p \frac{3433.3}{EI} - \frac{158575 P}{EI}$$

$$0 = M_p \frac{124}{EI} - \frac{3433.3 P}{EI}$$

$$M_p = \frac{3433.3 P}{124} = 27.6 P$$

$$-.207EI = 3433.3 \times 27.6 P - 158,575 P$$

$$= 94,759 P - 158,575 P$$

PREPARED BY 28

CHECKED BY _____

REVISED BY _____

ANALYSIS
OF

$$P = \frac{.20 EI}{63816} = \frac{(.207)(27.5)(10^6)(40.5)}{63816}$$

$$P = 3600 \#$$

$$M = 27.6 \times 3600 = 99,000 \text{ "H} \text{ PUMP}$$

$$M_{\text{REACTOR}} = (3600)(76) - 99000$$

$$= 274,000 - 99000$$

$$M_{\text{REACTOR}} = 185,000 \text{ "H}$$

$$M_{\text{LONGITUDINAL}} = .866 \times 159000$$

$$= 137000 \text{ "H}$$

$$\text{TORQUE} = 0.5 \times 159000$$

$$= 79500 \text{ "H}$$

$$\text{SHEAR} = 3280 \#$$

$$S_{\text{MAX}} = \frac{183000}{12.2} = \underline{15000 \text{ PSI}}$$

PREPARED BY _____ CHECKED BY _____ REVISED BY _____

MAIN LOOP PIPING PM-1 NUCLEAR POWER PLANT

SHEET 16
PROB. 60003
60004
60005

RATIO STRESSES
DOWN BY $\frac{.207}{.274}$

CONDITIONS

TEMPERATURES - REACTOR TO STM. GEN. - 479°F $\epsilon = .0475"/\text{FT}$
STM. GEN. TO REACTOR - 447°F $\epsilon = .0435"/\text{FT}$

STEAM GENERATOR AND PUMP ANCHOR TO ROLLER SUPPORT WHICH ALLOWS THEM TO MOVE FREELY IN THE "Z" DIRECTION.

SLOTTED HOLES IN PUMP SUPPORT ALLOW PUMP TO MOVE FREELY IN "Z" DIRECTION WITH RESPECT TO STEAM GENERATOR

PIPING SCHED. 80 SEAMLESS
STAINLESS A376 TYPE 316

MAX. ALLOWABLE STRESS = 27700 PSI
(AS PER PIPING CODE)

PROBLEM 60003
REACTOR TO STM. GEN

LOCATION	<u>MOMENTS</u>		
	M_x	M_y	M_z
0 (NOTE 1)	-8279	-8	-631
1	+329	-2	-631
2	+2058	0	+1098
3	+5783	+2	+1098
4 (NOTE 2)	+5783	+2	-297

$$F_x = +1 \quad \text{FORCES} \quad F_y = -1395 \quad \left\{ \begin{array}{l} \text{NOTE 1} \\ \text{AND NOTE 2} \end{array} \right.$$

NOTE 1 - REACTION OF PIPING ON REACTOR
NOTE 2 - REACTION OF STM. GEN. ON PIPING

$$S_{MAX} = 8110 \text{ PSI (POINT 0)}$$

THE ABOVE DOES NOT INCLUDE
THE EFFECT OF ANY FRICTIONAL
FORCE ACTING AT THE STEAM
GENERATOR DUE TO THE ROLLER
SUPPORTS.

PROBLEM 60004
STM. GEN. TO PUMP

MOMENTS

LOCATION	M_x	M_y	M_z
0 (NOTE 1)	-2100	-200	+278
1	-1823	+284	+707
2	+641	+284	+150
3	+1998	-301	-742
4 (NOTE 2)	-344	-301	-213

FORCES

$$F_x = -111 \quad F_y = +493 \quad F_z = -493 \quad \left\{ \begin{array}{l} \text{NOTE 1} \\ \text{AND} \\ \text{NOTE 2} \end{array} \right.$$

NOTE 1 - REACTION OF PIPING ON STM. GEN.

NOTE 2 - REACTION OF PUMP ON PIPING.

$$S_{\max} = \underline{\underline{2100 \text{ PSI}}}$$

THE ABOVE DOES NOT INCLUDE ANY EFFECT
DUE TO THE FRICTION OF THE ROLLER
SUPPORT FOR STM. GEN. AND PUMP.

IT WAS ASSUMED THAT LOOP PIPING
BETWEEN STM. GEN. AND PUMP WOULD
ABSORB ALL THE DIFFERENTIAL EXPANSION
BETWEEN RUN FROM REACTOR AND RUN
TO REACTOR.

MAIN LOOP PIPING
PM-1 NUCLEAR POWER PLANT

SHEET 19
PROB. 60005

PROBLEM 60005
PUMP TO REACTOR
MOMENTS

	LOCATION	M_x	M_y	M_z
(NOTE 1)	0	-24352	0	0
	1	-17213	0	0
	2	-9191	0	0
(NOTE 2)	3	+26184	0	0

$\begin{matrix} 2140 \text{ PSI} \\ 2954 \\ \hline 3091 \end{matrix}$

FORCES

$$F_y = +8022 \quad \left\{ \begin{array}{l} \text{NOTE 1 AND} \\ \text{NOTE 2} \end{array} \right.$$

NOTE 1 REACTION OF PIPING ON PUMP
NOTE 2 REACTION OF REACTOR ON PIPING

$$S_{\max} = \frac{19,400}{26554} \text{ PSI (PT. 3)}$$

THE ABOVE DOES NOT INCLUDE
THE EFFECT OF ANY FRICTIONAL
FORCE ACTING AT THE PUMP DUE
SLIDING OF PUMP ON ITS SUPPORT.

W. O. NO. 2283-12

BURNS AND ROE, INC.
NEW YORK, N. Y.

DATE: 9/66

DRAWING NO. _____

BY: M. GIDEN

TITLE: _____

SHEET 1 OF 9

STRESS ANALYSIS

PRESSURIZER

PM-3A

ITEM 5.0

W. O. NO. 2288-12

BURNS AND ROE, INC.
NEW YORK, N. Y.

DATE: 9/66

DRAWING NO.

BY: M. GIDEN

TITLE: SHEET 3 OF 9

INTRODUCTION & SUMMARY

THE PM-3A PRESSURIZER IS A VERTICAL CYLINDRICAL TANK 18 IN. DIAMETER AND 107 IN. LONG. THE HEADS ARE HEMISPHERICAL.

THE BASIC MATERIALS OF CONSTRUCTION ARE ASTM-A 240 TYPE 316 STAINLESS STEEL FOR THE SHELL AND HEADS WITH TYPE 316 STAINLESS STEEL NOZZLES.

THE VESSEL WAS ANALYZED IN ACCORDANCE WITH THE ASME BOILER CODE SECTION VIII UNFIRED PRESSURE VESSELS.

THE SPECIFIED DESIGN CONDITIONS ARE PRESSURE OF 1500 PSIG AND TEMPERATURE OF 600°F.

THE ACTUAL THICKNESS OF SHELL AND HEADS IS GREATER THAN THE CALCULATED MINIMUM THICKNESS REQUIRED BY CODE AND THEREFORE ACCEPTABLE.

THE 3 IN. NOZZLE IS INTEGRALLY REINFORCED, AND THE 1½" AND SMALLER NOZZLES DO NOT HAVE TO BE ANALYZED IN ACCORDANCE WITH REF. #2 PAR: UG-36-C-3a.

W. O. NO. 2288-12

BURNS AND ROE, INC.
NEW YORK, N. Y.

DATE: 9/6

DRAWING NO. _____

BY: M. GIDEN

TITLE: PM-3A PRESSURIZER

SHEET 3 OF 9

HEAD

MATERIAL: SA-240 TYPE 316 (T.S. = 75,000 PSI)

DESIGN PRESSURE: 1500 PSI

DESIGN TEMP.: 600°F

MEMB. ALLOWABLE 17,100 PSI (TABLE UHA-23 REF. 2)

EFFICIENCY 100%

$$\text{REQ'D MINIMUM THICKNESS } t = \frac{PL}{2SE - 0.2P} \quad (\text{PAR. UG-32 REF. 2})$$

P = DESIGN PRESSURE, PSI.

L = INSIDE RADIUS OF HEAD, IN.

S = MEMBRANE ALLOWABLE STRESS, PSI.

E = JOINT EFFICIENCY

$$t_{\text{REQ'D}} = \frac{1500(9.156)}{2(17100) - 0.2(1500)} = 0.405" < t_{\text{ACT.}} = 0.422"$$

W. O. NO. 2288-12BURNS AND ROE, INC.
NEW YORK, N. Y.DATE: 9/66

DRAWING NO. _____

BY: M. SIDENTITLE: PM-3A PRESSURIZER SHEET 4 OF 9SHELL

MATERIAL: SA-240 TYPE 316 S.S.

DESIGN PRESSURE: 1500 PSI

DESIGN TEMP: 200°F

MEMBRANE ALLOWABLE: 17,100 PSI

EFFICIENCY: 100%

$$\text{REQD MINIMUM THICKNESS } t = \frac{PR}{SE - 0.6P} \quad (\text{PAR. UG 27 REF. 4})$$

P = DESIGN PRESSURE, PSI.

R = RADIUS OF SHELL, IN.

S = MEMBRANE ALLOWABLE STRESS, PSI.

E = JOINT EFFICIENCY

$$t = \frac{1500(9)}{17100 - 0.6(1500)} = 0.835" < t_{\text{ACT.}} = 0.844"$$

HEATER WELLS

MATERIAL : SA-213 TYPE 316 SS.

7/8" x 0.148" MINIMUM WALL TUBING

MEMBRANE ALLOWABLE STRESS = 17,100 PSI (TABLE UHA-23 REI.)

DESIGN PRESSURE = 1500 PSI.

DESIGN TEMPERATURE = 600° F

FROM CHART FIG. UG-31 REF. #2 FOR P=1500 PSI
AND DESIGN STRESS = 17,100 PSI

$$t/D_o = 0.092$$

$$\therefore t = 0.092(0.875) = 0.081" \text{ REQD MIN. WALL} < t_{ACT.} = 0.148"$$

HEATER SHEATH

O.D. = 0.562 TO 0.567"

MINIMUM WALL = 0.052"

$$t_{REQD} = 0.0925(0.567) = 0.052" = t_{ACT.}$$

SHELL PENETRATION: AT HEATER WELLS.33 - $\frac{57}{64}$ " DIAMETER PENETRATION IN SHELL FORGING

* DWG. 387-2825004

W. O. NO. 2288-12

BURNS AND ROE, INC.
NEW YORK, N. Y.

DATE: 9/66

DRAWING NO.

BY: M. GIDEN

TITLE: SHEET 6 OF 9

HEATER WELL FORGING

LIGAMENT EFFICIENCY (PAR. UG-53)

$$\frac{p'}{p} = 1$$

$$\text{LONGITUDINAL EFF.} = \frac{p-d}{p} = \frac{2.25 - .8906}{2.25} = .605$$

DIAGONAL EFFICIENCY (FIG. UG-53.6)

$$\theta = 60^\circ$$

$$p'/d = \frac{2.25}{.8906} = 2.53$$

$$\text{EFF} = .85$$

W. O. NO. 2283-12BURNS AND ROE, INC.
NEW YORK, N. Y.DATE: 9/66

DRAWING NO. _____

BY: M. GIDENTITLE: _____ SHEET 8 OF 9

$$\text{REQD MINIMUM THICKNESS } t = \frac{PR}{SE - 0.6P} \quad (\text{PAR. UG 27 REF. 2})$$

P = DESIGN PRESSURE

R = RADIUS OF SHELL

S = MEMBRANE ALLOWABLE STRESS

E = JOINT EFFICIENCY

$$t_{\text{reqd}} = \frac{1500(9)}{(17100, 605 - 900)} = \frac{13500}{9450} = 1.43" < t_{\text{ACT}} = 2.50"$$

PRESSURIZER WEIGHTS

THE DRY WEIGHT OF THE PRESSURIZER IS 2440 #

THE WEIGHT WEIGHT OF THE PRESSURIZER IS 2750 #

W.O. NO. 3388-12

BURNS AND ROE, INC.
NEW YORK, N. Y.

DATE: 9/66

DRAWING NO. _____

BY: M. GUDEN

TITLE: PM-31A PRESSURIZER

SHEET 9 OF 9

REFERENCES

1. FORM U-1A MANUFACTURES' DATA REPORT
FOR UNFIRED PRESSURE VESSELS. GENERAL
AMERICAN TRANSPORTATION CORP. SHARON, Pa.
2. ASME BOILER AND PRESSURE VESSEL CODE, SECT. VIII
UNFIRED PRESSURE VESSELS. 1965

FORM U-1A MANUFACTURERS' DATA REPORT FOR UNFIRED PRESSURE VESSELS
Alternate Form for Single Chamber Completely Shop-Fabricated Vessels Only

1. Manufactured by General American Transportation Corporation, Sharon, Penna.
(Name and address of Manufacturer)

Manufactured for The Martin Company, Baltimore, Md.
(Name and address of Purchaser)

3. Type Vert. Vessel No. TS-8556 (Mfrs. Serial) (State & State No.) Natl. Bd. No. 733 Yr. Built 1961
(Horiz. or Vert.) (See Below) (Nom. 27/32 Corr. I.S.)

4. SHELL: Matl. SA-240-316 T.S. 75,000 Thk. 75,000 In. Allow 75,000 In. Diam. 1 Ft. 6 In. Length 8 Ft. 11 In. Hds. I.S.
(Kind and Spec. No.) (Fig. or F.B. & Spec. Min. T.S.)

5. SEAMS: Long Wld. Dbl. Butt H.T. No X.R. Complete Sectioned No Efficiency 100 %
(Welded, Dbl., Single, Lap, Butt) (Yes or No) (Spot or Complete) (Yes or No)

Girth Wld. Dbl. Butt H.T. No X.R. Complete Sectioned No No. of Courses 2

6. HEADS: (a) Material SA-240-316 T.S. 75,000 (b) Material SA-240-316 T.S. 75,000
Location (Top, bottom, ends) Thickness 27/64 Crown Radius 27/64 Knuckle Radius 27/64 Elliptical Ratio 27/64 Conical Apex Angle 9-5/32" Hemispherical Radius 9-5/32" Flat Diameter 9-5/32" Side to Pressure (Convex or Concave) Concave

(a) Top (b) Bot

If removable, bolts used 27/64 (Material, Spec. No., T.S., Size, Number) Other fastening 27/64 (Describe or Attach Sketch)

7. Constructed for max. allowable working press. 1500 psi at max. temp. 600 °F. Min. temp. (when less than -20°) 600 °F. Hydrostatic Pneumatic or Combination Test Press 2250 psi.

8. SAFETY OR RELIEF VALVE OUTLETS: Number 1 Size 1-1/2 Location Top Head

9. NOZZLES: (8) Nozzles - (1)-3", (1)-1-1/2", (2)-1", (1)-3/4", (3)-1/2"

Purpose (Inlet, Outlet, Drain)	Number	Diam. or Size	Type	Material	Thickness	Reinforcement Material	How Attached
Nozzles machine formed of SA-182-F-316 material welded in complete penetration. Reinforcement integral in nozzles.							

10. INSPECTION Manholes, No. 1 Size 4 Location In shell stud type flange
OPENINGS: Handholes, No. 1 Size 4 Location Forging A-182-F-316
Threaded, No. 1 Size 4 Location Forging A-182-F-316

11. SUPPORTS: Skirt No Lugs 4 Legs 4 Other Welded on shell
(Yes or No) (Number) (Number) (Describe) (Where & How)

12. REMARKS: Vessel is constructed to ASME Code Section VIII & in accordance with Case No. 1270-N for nuclear service. Vessel is a pressurized tank in primary coolant circuit. This vessel terminates at the welding end connection of each welded in nozzle, except for the 3" primary loop connection in bottom head which terminates at face of welding near flange.

4. Shell: A semi-circular forging (SA-182-F-316) forms a portion of the shell. (33) heater wells are welded in the forging and extend into the vessel. Wells are SA-213, Type 316, Tubing 7/8" X .148" Wall X 1'-9-1/2" inner ends closed with caps machine from SA-182-F-316 material.

(Brief description of purpose of the vessel as Air Tank, Water Tank, L.P.G., Etc.—State Contents.)
List other internal or external pressures with coincident temperature when applicable.

We certify that the statements made in this report are correct and that all details of design, material, construction, and workmanship of this vessel conform to the ASME Code for Unfired Pressure Vessels.

Date March 14, 19 61 Signed Gen. Amer. Transp. Corp. By s/S. R. Johnsen
(Manufacturer)

Certificate of Authorization Expires December 31, 1961

HSB-1916		CERTIFICATE OF SHOP INSPECTION	
VESSEL MADE BY <u>General American Transp. Corp.</u> at <u>Petroleum, Trumbull City, Ohio</u>			
I, the undersigned, holding a valid commission issued by the National Board of Boiler and Pressure Vessel Inspectors and/or the State of <u> </u> and employed by <u>Hartford Steam Boiler Insp. & Ins. Co.</u> of <u> </u>			
have inspected the pressure vessel described in this manufacturer's data report on <u>March 10</u> 19 <u>61</u> and state that to the best of my knowledge and belief, the manufacturer has constructed this pressure vessel in accordance with the applicable sections of the ASME Boiler and Pressure Vessel Code.			
By signing this certificate neither the Inspector nor his employer makes any warranty, expressed or implied, concerning the pressure vessel described in this manufacturer's data report. Furthermore, neither the Inspector nor his employer shall be liable in any manner for any personal injury or property damage or a loss of any kind arising from or connected with this inspection.			
Date <u>March 14,</u> 19 <u>61</u>		Commissions <u>NATL BD 588</u>	
Inspector's Signature <u>s/B. G. Murphy</u>		Nat'l Board or State and No. <u> </u>	

W. O. NO. 2288-12

BURNS AND ROE, INC.
NEW YORK, N. Y.

DATE: 9.16.62

DRAWING NO. _____

BY: L. CLAPP

TITLE: _____ SHEET 1 OF 4

STRESS ANALYSIS

PM - 3A

PRESSURIZER SPRAY LINE

ITEM 5.1

W. O. NO. 2288-12

BURNS AND ROE, INC.
NEW YORK, N. Y.

DATE: 9/66

DRAWING NO. _____

BY: J. CLAPP

TITLE: _____ SHEET 2 OF 4

SUMMARY

THERMAL PIPE STRESS ANALYSIS HAS BEEN PERFORMED FOR THE 1-INCH (SCHEDULE 40) PRESSURIZER SPRAY LINE FOR THE TEMPERATURE RANGE OF 390 °F. (70 °F. TO 460 °F.), UTILIZING THE BURNS AND ROE, INC. COMPUTER PROGRAM FOR PIPING FLEXIBILITY ANALYSIS.

RESULTS OF ANALYSIS INDICATE THAT A MAXIMUM THERMAL STRESS OF 6,180 PSI IS DEVELOPED IN THE LINE, AND OCCURS AT POINT ① OF THE ISOMETRIC DRAWING ON PAGE 3.

THE ALLOWABLE THERMAL STRESS FOR ASTM A312 TYPE 316 STAINLESS STEEL, BASED ON THE FORMULA $S_A = f(1.25 S_c + 0.25 S_h)$ OF REF. 1, IS 27,700 PSI.

THE FORCE AND MOMENT LOADINGS ACTING AT THE TERMINAL CONNECTIONS OF THE LINE (REACTOR COOLANT LOOP AND PRESSURIZER INLET NOZZLE) ARE OF NEGLIGIBLE MAGNITUDE.

ANALYSIS
OF
PM-3A PIPING

PRESSURIZER SPRAY LINE

REF DWG. 387-3725002, SH 3

1" SCH 40 - TY 316 STAINLESS STEEL

PREPARED BY duGAS 6/2/65 CHECKED BY _____ REVISED BY _____

W. O. NO. 2288-12

BURNS AND ROE, INC.
NEW YORK, N. Y.

DATE: 9/66

DRAWING NO. _____

BY: J. CLAPP

TITLE: _____ SHEET 4 OF 4

REFERENCE

1. CODE FOR PRESSURE PIPING ASA B31.1-1955
THE AMERICAN SOCIETY OF MECHANICAL
ENGINEERS .

W. O. NO. 2288-12

BURNS AND ROE, INC.
NEW YORK, N. Y.

DATE: 12/20

DRAWING NO. _____

BY: M. GIDEN

TITLE: _____ SHEET _____ OF _____

PM-3A

STRESS ANALYSIS PRESSURIZER RELIEF LINE

ITEM 5.2

cc: SBarn
HZizza
✓JClapp
File
db

BURNS AND ROE, INC.
ENGINEERS AND CONSTRUCTORS
160 WEST BROADWAY
NEW YORK 13, N. Y.
TEL RECTOR 2-7000

November 23, 1965

Subject: FM-#A Reactor
W.O. 2288-12
Contract AT(30-1)-3280
Stress Analysis, Pressurizer relief line.

U.S. Atomic Energy Commission
376 Hudson Street
New York, New York 10014

Attention: Mr. D. Streins
Project Engineer

Gentlemen:

The results of the stress analysis on the pressurizer relief line are discussed below, and a copy of the ILM Service Bureau Corporation Piping Flexibility Analysis is included for your records.

1. Pipe flexibility stress analysis was performed for the pressurizer relief line, which runs from the pressurizer to the expansion tank, for the design temperature of 550°F. Results of analysis are as follows:

a. The maximum stress in the system is 17,100 psi and occurs at point 13 in the 2-inch vertical run (Reference Enclosure 1.)

b. The stresses at the relief valve connections are here tabulated: (Reference Enclosure 1.)

Relief Valve (VS1)

Point 31: 3,456 psi @ 2-inch connection.
Point 30: 4,363 psi @ 1-1/2-inch connection.

Relief Valve (VS2)

Point 21: 3,573 psi @ 2-inch connection.
Point 22: 2,719 psi @ 1-1/2 inch connection.

U.S. Atomic Energy Commission
Attention: Mr. D. Streinz

November 23, 1965
Page -2-

2. The maximum stress in the system of 17,100 psi is well below the allowable stress of 26,400 psi for stainless-steel (304) pipe material.

3. The low stresses at the relief valve connections correspond to low bending moments listed at these points in Enclosure 1, as Mx, My, and Mz. These moments carried through the relief valve body are insufficient to cause distortion and leakage at the valve seat.

4. A possible source of high mechanical loading, independent of the thermal loading on the pressurizer relief line, is worthy of consideration. The relief valves, connected in parallel, form a closed loop with the 2-inch and 1-1/2-inch pipe legs. During initial installation, or subsequent replacement of the relief valves, the presence of any "as-built" misalignment between the mating flanges of the valves and the pipe legs would result in the development of bending and torsional moments throughout the loop. Because of the high degree of stiffness inherent in the compact loop, high bending and torsional moments can be developed with relatively small misalignments.

It should be noted also, that because the valves are bolted to the loop, the high mechanical advantage associated with screw-threads would enable one to overcome "as-built" misalignment in the process of bolting. As a result, the loop would be subjected to high "locked-in" mechanical loads prior to, and independent of, the thermal loading on the pressurizer relief line.

5. In the absence of any "as-built" misalignment between the mating flanges of the relief valves and pipe legs described in 4 above, it appears that the leakage at the valve seat of the relief valve may be due to the temperature to which the valve is exposed. The relief valve may be temperature sensitive, such that warpage of the seat or loss of mechanical spring-load takes place.

Very truly yours,

A. Andonedis
Project Manager

cc: Mr. J. Angelo, NNPU

Encl: Burns & Roe 1310
SBC Analysis 11/4/65

The Service Bureau Corporation

**AN EXPLANATION OF
PIPING FLEXIBILITY ANALYSIS**

GENERAL

The method of solution used by the SBC Piping Flexibility Analysis Program follows this general procedure:

- a. Determine the flexibility of the various piping elements.
- b. Combine these elements to give flexibility of sub-configurations and of the entire configuration.
- c. Calculate the thermal expansion of the piping.
- d. Incorporate the free thermal motions of anchors, terminals, and constraints.
- e. Calculate the forces and moments exerted by the piping on the anchors and constraints.
- f. Calculate the forces and moments internal to the piping system.
- g. According to standard procedures and conventions, calculate the expansion stress range (S_E) at various points in the piping configuration.
- h. Calculate displacements and rotations for various points in the piping configuration.

In determining flexibility, the usual procedures of Strength of Materials (implying the validity of the Euler-Bernoulli moment - curvature relation) are employed. An additional "flexibility factor", as specified in the ASA Code for Pressure Piping, is employed in figuring both in-plane and out-of-plane bending characteristics of elbows and pipe bends. Even though the influence of direct axial strain and cross shearing strain is small in most configurations, these effects are also taken into account. To determine cross shearing strain, the SBC Program makes use of a more precise evaluation of the so-called "shear distribution factor" than the customary (and usually quite satisfactory) approximation 2.0.

SIGN CONVENTION

All signed data are related to the Cartesian coordinate system employed to represent the piping configuration. The right handed system is the standard coordinate system usually chosen and is defined as follows:

With the thumb of the right hand pointed in the positive X-direction, and the index finger pointed in the positive Y-direction, the middle finger will point in the positive Z-direction.

The positive sense of an angular rotation or moment about an axis in a right handed system is defined by the direction in which the fingers of the right hand curl around this axis as the thumb points along the positive direction of the axis.

PIPING FLEXIBILITY ANALYSIS REPORT

Part I Edited Input

This is a reproduction of the input data describing the piping system, as edited and checked by the Program. Special symbols used are defined in the Piping Flexibility Users Manual. Differences that may exist between the computer edited input data and the original input data submitted by a customer fall into two general categories:

- a. The heading information (HED) is modified to indicate SBC's estimate and job number for filing and accounting reasons.
- b. Self explanatory identification symbols may be introduced by the Program. The presence of three asterisks (***) indicates information extracted from the Program's tables. This usually describes the properties of "standard" materials that correspond to those specified in the ASA Code for Pressure Piping. These "standard" materials are identified by the mnemonic code as listed in the Users Manual. This information includes:

E(C) - Cold Modulus of Elasticity
E(H) - Hot Modulus of Elasticity
ALPHA - Linear Thermal Expansion
NU - Cold Poisson's ratio

The tables of material properties contain data reported by an ASA Piping Code Task Force. Editing and checking of input data takes several forms:

- a. The geometry of the system is examined to detect any inadvertent errors.

- b. Input data is examined to insure that physical and engineering concepts are not violated.
- c. The format of the input data is examined for consistency with the requirements of the Program.

Part II Anchor and Constraint Reactions

This part presents data corresponding to the reaction ranges "R" as defined in the ASA Code.

This part contains the forces and moments exerted BY the piping system ON the anchors and constraints in going from 70 deg. F. to the specified temperature conditions. These values, as well as all others in the report, correspond to the full expansion range and are computed using cold moduli (70°F). The effects of bending, torsion, axial force and shear are taken into account.

Part III Force, Moment and Stress Range

This part contains forces, moments and expansion stresses, S_E , at all numbered points on the piping system. (Tangent intersection points at bends and elbows are not considered to be points on the piping system.) Two cases must be considered for Part III.

Case A: Systems NOT Containing Closed Loops

If the piping system is cut at any point and the portion of the system NOT connected to the basic anchor discarded, then the forces and moments at that point represent the action of the discarded portion ON the portion connected to the basic anchor. It should also be noted that the cuts are made infinitely close to branch points for each direction of approach. Similarly, cuts are made at both sides of constraint points as well as points at which concentrated forces or moments are applied.

Case B: Systems Containing Closed Loops

The computer detects a closed loop during input data editing whenever the piping closes on a previously defined branch point. A cut is automatically made at the branch point in the section of pipe which closed the loop. This "cutting" is performed systematically for all closed loops and results, in effect, in a system having no closed loops. After "cutting" equal but opposite forces and moments are applied to both sides of the cut.

In the diagram of a system with a closed loop, forces, moments and expansion stresses will appear in Part III at all points indicated by an "X".

Expansion Stress Range, S_E

The expansion stress range, S_E , is calculated for straight pipe, curved pipe and elbows using formulas specified in the ASA Code for Pressure Piping. In the case of curved pipe and elbows, the appropriate stress intensification factor as stipulated in the Code is calculated and applied. However, stress intensification factors appropriate to special joints (such as lap joint flanges), tees, nozzles, bifurcations, or other special fabrications are not introduced by the Program. For such cases, the customer must make his own calculations of S_E , employing the appropriate stress intensification factor and the pertinent force and moment (range) values given in the flexibility report. With these conventions, the program computes S_E at each end of each piece, and for every point records the larger of the two values appropriate to the point. RIGID pieces are considered to have zero stress in them, since special computations would in all cases be required to determine stress in any piece stiff enough or massive enough to be designated as RIGID. Since no stress intensification factors are introduced for bifurcations, the reported values of S_E pertain to the pipe emerging from the bifurcation point.

Part IV: Displacement and Rotations

This section contains the cold-to-hot displacements and rotations for each numbered point ON the piping system.

NEW YORK DATA PROCESSING CENTER

PIPING FLEXIBILITY ANALYSIS

ESTIMATE NO	JOB NO	REQUESTED BY	DATE
0298	1310	BURNS + ROE INC.	11/4/65

TYPE	FROM	TO	DELTA X	DELTA Y	DELTA Z	U.D.	W.T.	METAL	TEMP
CUM									
BAN		1							
		2		1-10.7415		2.375	0.218	AUS	550
***			EIC)=28,300,000	E(H)=25,750,000	ALFA=5.62	NU=0.300			
TIN		3		0-1.2585					
		4	-0-0.5387	0-0.8818	0-0.7183				
		5	-1-4.9226	1-7.4884	1-10.5834				
TIN		6	-0-0.5387	0-0.8818	0-0.7183				
		7		0-1.2585					
		8		0-10.989					
TIN		9		0-1.2525					
		10	-0-0.1324	0-0.8806	-0-0.8805				
		11	-0-4.7352	2-7.4898	-2-7.489				
TIN		12	-0-0.1324	0-0.8806	-0-0.8805				
		13		0-1.2525					
BKA		14		0-4.7475					
CAP		14	X=-1-11	Y=14-2.0	Z=-0-4.250				
		15	-0-2.1213		0-2.1213				
TIN		16	-0-2.1213		0-2.1213				
		17		0-3.000					
		18		0-2.500					

[illegible]

ANCHOR AND CONSTRAINT REACTION RANGES - R

ANCHOR -----	FORCES IN POUNDS			F --	..	MOMENTS IN INCH-POUNDS			R --
	F(X) -----	F(Y) -----	F(Z) -----			M(X) -----	M(Y) -----	M(Z) -----	
1	59	-513	22	517		8.284	1,131	2,491	8,705
40	-59	513	-22	517		2.887	312	1,202	3,125

FORCE , MOMENT AND STRESS RANGES

POINT -----	FORCES IN POUNDS			MOMENTS IN INCH-POUNDS			STRESSES IN PSI
	F(X) -----	F(Y) -----	F(Z) -----	M(X) -----	M(Y) -----	M(Z) -----	S(E) -----
40	59	-513	22	-2,867	-312	-1,202	7,588
39	59	-513	22	-1,126	-37	559	3,119
37	59	-513	22	-304	72	678	2,334
26	59	-513	22	2,320	376	878	6,093
26	38	-430	45	1,032	125	1,303	4,047
25	38	-430	45	-107	-95	164	558
23	38	-430	45	-890	-227	-434	3,019
22	38	-430	45	-1,057	-227	-290	2,719
21	38	-430	45	2,594	99	-290	3,573
20	38	-430	45	3,453	175	-290	6,262
18	38	-430	45	4,875	290	-405	8,851
17	38	-430	45	4,964	290	-481	9,023
15	38	-430	45	6,008	465	315	9,817
14	38	-430	45	6,920	641	1,226	9,655
14	59	-513	22	9,292	1,175	759	12,856
13	59	-513	22	9,393	1,175	478	17,080
11	59	-513	22	8,990	1,126	419	16,386
10	59	-513	22	-6,479	-636	984	11,890
8	59	-513	22	-6,864	-685	925	12,591
7	59	-513	22	-5,069	-685	-3,991	11,699
5	59	-513	22	-4,653	-631	-3,841	10,940
4	59	-513	22	7,351	1,076	3,688	14,990

POINT -----	F(X) -----	F(Y) -----	F(Z) -----	M(X) -----	M(Y) -----	M(Z) -----	S(E) -----
2	59	-513	22	7,767	1,131	3,638	15,736
1	59	-513	22	8,264	1,131	2,491	11,910
14	-21	84	23	-2,372	-534	407	3,388
36	-21	84	23	-2,549	-531	290	4,100
34.1	-21	84	23	-2,795	-527	49	5,104
34	-21	84	23	-2,840	-527	7	5,184
32	-21	84	23	-2,857	-464	-56	4,876
31	-21	84	23	-2,490	-422	-50	3,456
30	-21	84	23	-1,779	-244	-50	4,563
29	-21	84	23	-1,694	-244	23	5,112
27	-21	84	23	-1,510	-247	203	4,108
26	-21	84	23	-1,289	-251	425	3,351

MAXIMUM STRESS RANGE AT POINT 13 = 17,080 PSI.

DEFLECTION RANGES

POINT	DISPLACEMENTS IN INCHES			.	ROTATIONS IN RADIANS		
	U(X)	U(Y)	U(Z)		R(X)	R(Y)	R(Z)
40	0.000	0.000	0.000		0.00000	0.00000	0.00000
39	0.016	0.003	-0.016		0.00097	0.00008	0.00004
37	0.019	0.004	-0.024		0.00120	0.00007	-0.00009
28	0.018	0.010	-0.048		0.00074	-0.00004	-0.00062
25	0.006	0.011	-0.035		0.00059	-0.00004	-0.00088
23	0.000	0.022	-0.026		0.00107	0.00010	-0.00066
22	0.002	0.040	-0.022		0.00140	0.00020	-0.00054
21	0.001	0.051	-0.061		0.00140	0.00020	-0.00054
20	0.000	0.054	-0.071		0.00115	0.00019	-0.00051
18	-0.001	0.061	-0.084		-0.00128	0.00006	-0.00035
17	-0.002	0.032	-0.081		-0.00168	0.00003	-0.00031
15	0.008	0.010	-0.082		-0.00425	-0.00052	-0.00060
14	0.019	-0.001	-0.091		-0.00519	-0.00059	-0.00059
13	0.016	-0.023	-0.062		-0.00699	-0.00088	-0.00071
11	0.014	-0.026	-0.040		-0.00962	-0.00115	-0.00084
10	-0.060	0.235	0.529		-0.01194	-0.00142	-0.00230
8	-0.066	0.235	0.557		-0.01002	-0.00132	-0.00254
7	-0.177	-0.155	0.507		0.01018	0.00169	0.00264
5	-0.169	-0.153	0.480		0.01160	0.00178	0.00374
4	-0.040	0.108	0.108		0.00960	0.00161	0.00400
2	-0.031	0.106	0.085		0.00742	0.00136	0.00275
1	0.000	0.000	0.000		0.00000	0.00000	0.00000
14	0.019	-0.001	-0.091		-0.00519	-0.00059	-0.00059

<u>POINT</u>	<u>U(X)</u>	<u>U(Y)</u>	<u>U(Z)</u>	<u>R(X)</u>	<u>R(Y)</u>	<u>R(Z)</u>
36	0.030	-0.013	-0.100	-0.00483	-0.00052	-0.00069
34.1	0.043	-0.010	-0.121	-0.00358	-0.00051	-0.00052
34	0.044	0.000	-0.128	-0.00335	-0.00045	-0.00052
32	0.044	0.021	-0.122	-0.00178	-0.00025	-0.00056
31	0.044	0.024	-0.113	-0.00157	-0.00022	-0.00055
30	0.042	0.037	-0.073	-0.00157	-0.00022	-0.00055
29	0.040	0.020	-0.068	-0.00098	-0.00011	-0.00055
27	0.031	0.010	-0.060	0.00017	-0.00012	-0.00043

TITLE: PRESSURIZER RELIEF SYSTEM

BURNS AND ROE, INC.
NEW YORK, N. Y.

DATE: 10-28-55

BY: 14 G1754

SHEET 1 **OF** _____

SEC 5.2

304 S.S.

DESIGN TEMP. = 550°F

W. O. NO. 2288-12

BURNS AND ROE, INC.
NEW YORK, N. Y.

DATE: 9/66

DRAWING NO. _____

BY: R. J. DOBBS

TITLE: _____

SHEET 1 OF 6

STRESS ANALYSIS

PURIFICATION DEMINERALIZER

PM-3A

ITEM - 6.0

W. O. NO. 2288-12

BURNS AND ROE, INC.
NEW YORK, N. Y.

DATE: 9/66

DRAWING NO.

BY: R. J. DOBBS

TITLE: PM-3A STRESS ANALYSIS

SHEET 2 OF 6

INTRODUCTION & SUMMARY

THE COOLANT PURIFICATION DEMINERALIZER IS A VERTICAL CYLINDRICAL TANK 11" DIAMETER AND 29" LENGTH. THE HEADS ARE ASME STANDARD 2:1 ELLIPTICALLY DISHED.

THE BASIC MATERIALS OF CONSTRUCTION ARE ASTM-A240 TYPE 304 STAINLESS STEEL FOR THE SHELL AND HEADS WITH TYPE 304 AND TYPE 316 STAINLESS STEEL NOZZLE NECKS.

THE VESSEL WAS ANALYZED IN ACCORDANCE WITH THE ASME BOILER AND PRESSURE VESSEL CODE SECTION VIII UNFIRED PRESSURE VESSELS.

THE SPECIFIED DESIGN CONDITIONS ARE PRESSURE OF 1500 PSIG AND TEMPERATURE OF 500°F.

THE ACTUAL THICKNESS OF SHELL AND HEADS IS GREATER THAN THE CALCULATED MINIMUM THICKNESS REQUIRED BY CODE AND THEREFORE ACCEPTABLE.

THE INLET AND OUTLET NOZZLES ARE 2 INCH PIPE SIZE OR SMALLER AND THEREFORE REINFORCEMENT REQUIREMENTS ARE WAIVED IN ACCORDANCE WITH REF. # 2 PARA. UG-36-C-3a.

BY NCH DATE 5/27/65
CHKD. BY _____ DATE _____

CREAM CITY Fabricators

BOILER COMPANY • MILWAUKEE 46, WIS.

SHEET NO. 3 OF 6

JOB NO. C.C.B. Co.

No. H-87761

CALCULATIONS

COOLANT PURIFICATION DEMINERALIZER
CP02-IE1 CONTRACT No. AT(30-1)2700

ELLIPTICAL HDS: PAR. UG-32(d)

$$t = \frac{PD}{RSE - 0.2P} = \frac{1500 \times 9.75}{2 \times 15,100 \times 1 - 0.2 \times 1500} = 0.49''$$

SHELL: PAR. UG-27(c)

$$t = \frac{PR}{SE - 0.6P} = \frac{1500 \times 4.875}{15,100 \times 1 - 0.6 \times 1500} = 0.515$$

NO REINFORCEMENT REQUIRED ON
NOZZLE OPENINGS PER PAR. UG-36(C)(3)(a)

REF. THE MARTIN COMPANY
BALTIMORE, MARYLAND
DRWG. E-387-2876021

FORM U-1 MANUFACTURERS' DATA REPORT FOR UNFIRED PRESSURE VESSELS

4 Of 6

As required by the Provisions of the ASME Code Rules

1. Manufactured by CREAM CITY BOILER CO. - MILWAUKEE, WIS.
(Name and address of Manufacturer)

2. Manufactured for ILLINOIS WATER TREATMENT CO. - ROCKFORD, ILL.
(Name and address of Purchaser)

Type Vent Kind Tank Vessel No. (H-87761) Natl. Bd. No. NB530 Yr. Built 1961
(Horiz. or Vert.) (Tank, Jacketed, Heat Exch.) (Mfrs. Serial) (State & State No.)

Items 4-9 incl. to be completed for single wall vessels (such as air tanks), jackets of jacketed vessels, or shells of heat exchangers.

4. SHELL: Material A240T304 T.S. 75000 Nominal Thickness 5/8 In. Allowance 0 In. Diam. 0 Ft. 11 In. Length 2 Ft. 5 In.
(Kind and Spec. No.) (Fig. or F.B. & Spec. Min. T.S.)

5. SEAMS: Long Wld. Dbl. Butt H.T. No X.R. Complete Sectioned no Efficiency 100 %
(Welded, Dbl., Single, Lap, Butt) (Yes or No) (Spot or Complete) (Yes or No)

If riveted describe seams fully on reverse side of form.

Girth Wld. dbl. butt H.T. No X.R. No Sectioned No No. of Courses 1

6. HEADS (a) Material A240T304 T.S. 75000 (b) Material _____ T.S. _____
Location Thickness Crown Radius Knuckle Radius Elliptical Ratio Conical Apex Angle Hemispherical Radius Flat Diameter Side to Pressure
(Top, bottom, ends) (Convex or Concave)

(a) top&btm. 1/2" _____ _____ _____ _____ _____ _____ _____ concave

(b) _____ _____ _____ _____ _____ _____ _____ _____

If removable, bolts used _____ Other fastening _____
(Material, Spec. No., T.S., Size, Number) (Describe or Attach Sketch)

7. STAYBOLTS: _____ If hollow _____ Attachment _____ Pitch _____ X _____ Diam. _____
(Material) (Size of Hole) (Threaded, Welded) (Horiz.) (Vert.) (Nominal)

8. JACKET CLOSURE: _____
(Describe as ogee & weld, bar, etc. If bar, give dimensions, if bolted, describe or sketch)

9. Constructed for max. allowable working press. 1500 psi at max. temp. 500° F. Min. temp. (when less than -20°) _____ °F. Hydrostatic Pneumatic or Test Press 2800 psi.
Combination

Items 10 and 11 to be completed for tube sections.

10. TUBE SHEETS: Stationary. Material _____ Diam. _____ In. Thickness _____ In. Attachment _____
(Kind & Spec. No.) (Subject to Pressure) (Welded, Bolted)

Floating. Material _____ Diam. _____ In. Thickness _____ In. Attachment _____
(Kind & Spec. No.)

11. TUBES: Material _____ O.D. _____ In. Thickness _____ Inches or Gage Number _____ Type _____
(Kind & Spec. No.) (Straight or U)

Items 12-15 incl. to be completed for inner chambers of jacketed vessels, or channels of heat exchangers.

12. SHELL Material _____ T.S. _____ Nominal Thickness _____ In. Allowance _____ In. Diam. _____ Ft. _____ In. Length _____ Ft. _____ In.
(Kind and Spec. No.) (Fig. or F.B. & Spec. Min. T.S.)

13. SEAMS: Long _____ H.T. _____ X.R. _____ Sectioned _____ Efficiency _____ %
(Welded, Dbl., Single, Lap, Butt) (Yes or No) (Spot or Complete) (Yes or No)

Girth _____ H.T. _____ X.R. _____ Sectioned _____ No. of courses _____

If riveted describe seams fully on reverse side of form.

14. HEADS (a) Material _____ T.S. _____ (b) Material _____ T.S. _____ (c) Material _____ T.S. _____

Location Thickness Crown Radius Knuckle Radius Elliptical Ratio Conical Apex Angle Hemispherical Radius Flat Diameter Side to Pressure
(Top, bottom, ends) (Convex or Concave)

(a) Top, bottom, ends _____ _____ _____ _____ _____ _____ _____ _____

(b) Channel _____ _____ _____ _____ _____ _____ _____ _____

(c) Floating _____ _____ _____ _____ _____ _____ _____ _____

If removable, bolts used (a) _____ (b) _____
(Material, Spec. No., T.S., Size, Number)

(c) _____ Other fastening _____
(Describe or Attach Sketch)

15. Constructed for max. allowable working press. _____ psi at max. temp. _____ °F. Min. temp. (when less than -20°) _____ °F. Hydrostatic Pneumatic or Test Press _____ psi.
Combination

Items below to be completed for all vessels where applicable.

16. SAFETY VALVE OUTLETS: Number _____ Size _____ Location _____

17. NOZZLES

Purpose (Inlet, Outlet, Drain)	Number	Diam. or Size	Type	Material	Thickness	Reinforcement Material	How Attached
Inlet	1	2"	Pipe	316	Sch. 80		Welded
"	1	1/2"	"	304	"		Welded
Outlet	1	2"	"	316	"		Welded
"		1/2"	"	304	"		"

¹ If postweld heat-treated.² List under remarks other internal or external pressures with coincident temperature when applicable.

FORM U-1 (back)

18. INSPECTION Manholes, No. _____ Size _____ Location _____
 OPENINGS: Handholes, No. _____ Size _____ Location _____
 Threaded, No. _____ Size _____ Location _____
 19. SUPPORTS: Skirt No (Yes or No) Lugs 4 (Number) Legs _____ (Number) Other _____ (Describe) Attached Weld to (Where & How) Shell

20. REMARKS: This vessel is a coolant purification demineralizer with an external housing to hold 6" of lead all around per the Martin Co.
Dwg. E387-2876021

(Brief description of purpose of the vessel, as Air Tank, After Cooler, Jacketed Cooker, etc. State contents of each part.)

We certify that the statements made in this report are correct and that all details of design, material, construction, and workmanship of this vessel conform to the ASME Code for Unfired Pressure Vessels.

Date Jan. 23 19 61 Signed Cream City Boiler Co. (Manufacturer) By Belbert L. Pellman V.P.

Certificate of Authorization Expires Dec. 31, 1961

CERTIFICATE OF SHOP INSPECTION

VESSEL MADE BY CREAM CITY BOILER CO. at MILWAUKEE, WIS.

I, the undersigned, holding a valid commission issued by the National Board of Boiler and Pressure Vessel Inspectors and/or the State of NB States and employed by LUMBERMENS MUTUAL CASUALTY CO. of CHICAGO have inspected the pressure vessel described in this manufacturer's data report on Jan. 23, 19 61 and state that to the best of my knowledge and belief, the manufacturer has constructed this pressure vessel in accordance with the applicable sections of the ASME Boiler and Pressure Vessel Code.

By signing this certificate neither the Inspector nor his employer makes any warranty, expressed or implied, concerning the pressure vessel described in this manufacturer's data report. Furthermore, neither the Inspector nor his employer shall be liable in any manner for any personal injury or property damage or a loss of any kind arising from or connected with this inspection.

Date Jan. 23 19 61

Inspector's Signature F.G. Wurf Commissions NB#2169 Nat'l Board or State and No.

CERTIFICATE OF FIELD ASSEMBLY INSPECTION

I, the undersigned, holding a valid commission issued by the National Board of Boiler and Pressure Vessel Inspectors and/or the State of _____ and employed by _____ of _____ have compared the statements in this manufacturer's data report with the described pressure vessel and state that parts referred to as data items _____, not included in the certificate of shop inspection have been inspected by me and that to the best of my knowledge and belief the manufacturer has constructed and assembled this pressure vessel in accordance with the applicable sections of the ASME Boiler and Pressure Vessel Code. The described vessel was inspected and subjected to a hydrostatic test of _____ psi.

By signing this certificate neither the Inspector nor his employer makes any warranty, expressed or implied, concerning the pressure vessel described in this manufacturer's data report. Furthermore, neither the Inspector nor his employer shall be liable in any manner for any personal injury or property damage or a loss of any kind arising from or connected with this inspection.

Date _____ 19 _____

Inspector's Signature _____ Commissions _____ Nat'l Board or State and No.

W. O. NO. 2288-12

BURNS AND ROE, INC.
NEW YORK, N. Y.

DATE: 9/66

DRAWING NO. _____

BY: M. GIDEN

TITLE: PM-3A STRESS ANALYSIS

SHEET 6 OF 6

REFERENCES

1. FORM U-1 MANUFACTURES' DATA REPORT
FOR UNFIRED PRESSURE VESSELS.
CREAM CITY BOILER Co. 1961
2. ASME BOILER AND PRESSURE VESSEL CODE
SECTION VIII. UNFIRED PRESSURE VESSELS 1965.

W. O. NO. 2288-12

BURNS AND ROE, INC.
NEW YORK, N. Y.

DATE: 9-6-66

DRAWING NO. _____

BY: M. GILLEN

TITLE: _____ SHEET 1 OF 8

STRESS ANALYSIS

PM - 3A

PURIFICATION DEMINERALIZER

AND DECAY HEAT PIPING

ITEM 6.1

W.O. NO. 2288-12

BURNS AND ROE, INC.
NEW YORK, N. Y.

DATE: 9-6-66

DRAWING NO. _____

BY: M. GIDEN

TITLE: _____ SHEET 2 OF 8

SUMMARY

THE THREE LOOPS OF THE PURIFICATION AND DECAY HEAT PIPING SYSTEM WERE ANALYZED FOR A 390° TEMPERATURE RISE UTILIZING THE BURNS & ROE COMPUTER PROGRAM FOR PIPING FLEXIBILITY ANALYSIS

THE MAXIMUM STRESS (21,657 PSI) OCCURS IN LOOP No. 1. THIS IS BELOW THE MAXIMUM ALLOWABLE STRESS RANGE (27,770 PSI) OF REF. 1.

THE FORCES AND MOMENTS ON THE EQUIPMENT ARE TABULATED IN TABLE I.

FOR THE DESIGN CONDITIONS, THE PURIFICATION AND DECAY HEAT PIPING SYSTEM IS SATISFACTORY.

TITLE:

TABLE I

PM-3A PURIFICATION AND DECAY HEAT PIPING FORCES AND MOMENTS

	LOOP No. 1		LOOP No. 2		LOOP No. 3	
	POINT 4	POINT 7	POINT 7	POINT 13	POINT 27	POINT 29
F_x (LBS)	320	-320	-1	1	-18	18
F_y (LBS)	370	-370	35	-35	83	-83
F_z (LBS)	0	0	10	-10	1	-1
M_x (FT.-LBS)	0	0	-83	118	-4	75
M_y (FT.-LBS)	0	0	-3	-11	-1	13
M_z (FT.-LBS)	318	-513	4	49	96	173

LOOP No.	MAX. STRESS	LOCATION
1	21,650 PSI.	MEMBER 1-0
2	6,490 PSI.	POINT 12
3	8,740 PSI.	POINT 20

DESIGN TEMPERATURE RISE 70°F TO 460°F

$$S_A (\text{ALLOWABLE STRESS, REF * 1}) = f(1.25S_c + 0.25S_h)$$

f = CYCLIC FACTOR = 1.0

S_c = ALLOWABLE STRESS AT COLD TEMP. (PSI)

S_h = ALLOWABLE STRESS AT HOT TEMP. (PSI)

FOR 316 S.S. PIPE

$$S_A = 1.0 [1.25(18750) + 0.25(17300)] = 27700 \text{ PSI}$$

POSITIVE SIGN CONVENTION

ANALYSIS
OF
PM-3A PIPING

PURIFICATION & DECAY HEAT PIPING FLEXIBILITY

REF DWG 387-3725002, SH. 2.

LOOP #1 $1\frac{1}{2}$ SCH 80 - TY 316 STAINLESS STEEL

DISTANCE PT. (A) TO PT. (B) = $(1.0' + 0.17' + 4.75')^{\dagger} = 4.85 \text{ FT.}$

PREPARED BY duGas
6/27/66

CHECKED BY _____

REVISED BY _____

ANALYSIS
OF
PM-3A PIPING

PURIFICATION & DECAY HEAT PIPING FLEXIBILITY
REF. DWG. 387-3725002, SH. 2
LOOP #2 1 1/2" SCH 80 - TY 316 STAINLESS STEEL

$$\text{DISTANCE PT. ⑥ TO PT. ⑬} = \left(\overline{6.42}^2 + \overline{0.75}^2 + \overline{16.83}^2 \right)^{\frac{1}{2}} = 18.0 \text{ FT.}$$

PREPARED BY du Gas
5/27/65

CHECKED BY _____

REVISED BY _____

ANALYSIS
OF
PM-3A PIPING

PURIFICATION & DECAY HEAT PIPING FLEXIBILITY

REF. DWG 387-3725002, SH 2

LOOP # 3 1 1/2 SCH 80 - TY 316 STAINLESS STEEL

$$\text{DISTANCE FROM PT. } \textcircled{E} \text{ TO PT. } \textcircled{F} = \left(6.92^2 + 3.0^2 + 12.58^2 \right)^{\frac{1}{2}} = 14.63 \text{ FT.}$$

PREPARED BY duGas
5/27/65

CHECKED BY _____

REVISED BY _____

W. O. NO. 2288-12

BURNS AND ROE, INC.
NEW YORK, N. Y.

DATE: 9-6-66

DRAWING NO. _____

BY: M. GIDEN

TITLE: _____ SHEET 8 OF 8

REFERENCES

1. CODE FOR PRESSURE PIPING ASA B 31.1-1955
THE AMERICAN SOCIETY OF MECHANICAL
ENGINEERS.

W. O. NO. 2288-12

BURNS AND ROE, INC.
NEW YORK, N. Y.

DATE: 9/66

DRAWING NO. _____

BY: R. J. DOBBS

TITLE: _____

SHEET 1 OF 12

STRESS ANALYSIS

ECONOMIZER

PM-3A

ITEM 6.2

INTRODUCTION & SUMMARY

THE ECONOMIZER IS A U-TUBE HEAT EXCHANGER. THE VESSEL IS FABRICATED FROM 4 INCH SCH 40 PIPE ON THE SHELL SIDE AND 4 INCH SCH 80 ON THE TUBE SIDE.

THE BASIC MATERIALS OF CONSTRUCTION ARE ASTM-A312-TYPE 316 STAINLESS STEEL OR EQUIVALENT.

THE VESSEL WAS ANALYZED IN ACCORDANCE WITH ASME BOILER AND PRESSURE VESSEL CODE SECTION VIII UNFIRED PRESSURE VESSELS.

THE SPECIFIED DESIGN CONDITIONS FOR BOTH SHELL AND TUBE SIDES ARE INTERNAL PRESSURE OF 1500 PSIG AND TEMPERATURE OF 500°F.

THE ACTUAL THICKNESS OF THE VARIOUS ECONOMIZER COMPONENTS (SHELL, END CAP, TUBE SHEET, TUBING) IS IN EACH CASE GREATER THAN THE CALCULATED MINIMUM THICKNESS REQUIRED BY CODE AND THEREFORE ACCEPTABLE.

REINFORCEMENT CALCULATIONS CLEARLY INDICATE THAT REQUIREMENTS OF THE CODE HAVE BEEN SATISFIED.

A CHECK WAS ALSO MADE OF THE 3 INCH DIAMETER BLIND FLANGE AND FOUND TO BE ACCEPTABLE. THE STRESSES BEING LESS THAN THOSE ALLOWED BY CODE

NOTE THAT THE CALCULATIONS PRESENTED IN THIS REPORT ARE FOR THE PM-1 ECONOMIZER. IT WAS ASSUMED THAT THE PM-3A ECONOMIZER IS IDENTICAL.

ANALYSIS
OF
PM-1 ECONOMIZER

REF DWG 372-2876053
SHEETS 3

ASME CODE ANALYSIS: SECTION VIII

BASIC DESIGN CHARACTERISTICS

DESIGN PRESSURE - 1500 PSI
DESIGN TEMPERATURE - 500 °F
OPERATING PRESSURE - 1300 PSI

VESSEL IS FABRICATED FROM 4" SCHEDULE 40
PIPE ON SHELL SIDE

4" SCHEDULE 80 ON TUBE SIDE.

END FLANGE - 3" BLIND FLANGE FOR 900 PSI.
NOZZLES 1/2" SCHEDULE 40 PIPE.

ALL MATERIAL SA 213 TYPE 316 STAINLESS
STEEL OR EQUIVALENT

ALLOWABLE WORKING PRESSURES FOR SA 312
TYPE 316 STAINLESS STEEL PIPES (@ 500 °F)
WERE OBTAINED FROM TUBE TURN BULLETIN
TT 600, PAGE 45 AND ARE AS FOLLOWS:

4" SCHEDULE 40 - 1650 PSI

4" SCHEDULE 80 - 2380 PSI

1/2" SCHEDULE 40 - 4300 PSI

PRESSURE TEMPERATURE RATING FOR
3" BLIND FLANGE 900 LB. PER LADISH
CAT. 55, PAGE 267.

3" 900 LB. 316 STAINLESS FLANGE AT 500 °F
SERVICE PRESSURE RATING 1500 PSI

PREPARED BY C.H.

11/11/60

CHECKED BY

REVISD BY

ANALYSIS
OF
PM-1 ECONOMIZER

END CAP

REF. SUBSECTION A

MATERIAL SA 240 TYPE 316
S = 17,200 @ 500° F.

$$t = d \sqrt{\frac{CP}{S}} \quad \text{REF. FP. UG-34} \quad C = .050 \quad \text{FIG UG-34 f}$$

$$t = 3.826 \sqrt{\frac{.50 \times 1500}{17200}}$$

$$t = 3.826 \sqrt{.0436}$$

$$t = 3.826 \times .204$$

$$t = .780 \text{ REQUIRED}$$

$$t_{ACT} = 15/16 \text{ (ALLOWANCE MADE FOR GROOVE DEPTH } 1/8 \text{)}$$

PREPARED BY LEN

CHECKED BY [Signature]

REVISED BY _____

11/11/60

ANALYSIS
OF
PM-1 ECONOMIZER

REINFORCEMENT OF NOZZLES

REF: SUBSECTION A

SHELL AND NOZZLE SA 240 TYPE 316 S.S.
 $S = 17200 @ 500^{\circ}F$

SHELL

$$t_r = \frac{PR}{SE - .6P} = \frac{1500 \times 1.913}{17200 - .6 \times 1500} \quad \text{REF. PP. UG - 27}$$

$$t_r = .174''$$

NOZZLE

$$t_r = \frac{PR}{SE - .6P} = \frac{1500 \times .311}{17200 - .6 \times 1500}$$

$$t_r = .028''$$

PREPARED BY E.H.
4/11/60

CHECKED BY J. G. Smith

REVISIO BY _____

ANALYSIS
OF

PM-1 ECONOMIZER

REINFORCEMENT OF NOZZLES (CONTINUED)

FILLET WELD

$$t_c \sim \text{SMALLER OF } 1/4 \text{ OR } .7 t_m \quad t_m = .109''$$

$$t_c \sim .076''$$

REF. UW-16-6

$$\Delta \text{ FILLET REQUIRED} = 1.414 \times .076 = .107$$

USE $1/8$ FILLET

AREA OF REINFORCEMENT REQUIRED

$$A = d t_r = .622 \times .174 = .1083 \text{ IN.}^2$$

REF. UA-280

AREA AVAILABLE

$$\text{SHELL } A_1 = .622 (.337 - .174)$$

$$= .622 (.163) = .1013$$

$$\text{NOZZLE } A_2 = 2 \times \pi (4.35) (.109 - .029) = .070$$

$$\text{WELD } A_3 = 1/2 (.125)^2 (2) = .015$$

$$\text{TOTAL AREA PROVIDED } .186 \text{ IN.}^2$$

REINFORCEMENT IS ADEQUATE:

$$\begin{aligned} & * 2 1/2 (\text{NOZZLE WALL}) + (\text{VESSEL } t_{ACT} - t_{REQ'D}) \\ & (2.5)(.109) + .174 = 0.446 \end{aligned}$$

PREPARED BY JH

11/11/60

CHECKED BY [Signature]

11

REVISED BY _____

ANALYSIS
OF

PM-1 ECONOMIZER

LOAD CARRIED BY WELDS

$$W = (A - A_1) S$$

$$W = (.1083 - .1013) 17200 = 120^{\#}$$

UNIT STRESSES

SHEAR IN FILLET

$$.49 \times 17200 = 8400 \text{ PSI}$$

TENSION IN GROOVE WELD

$$.74 \times 17200 = 12700 \text{ PSI}$$

SHEAR IN NOZZLE WALL

$$.70 \times 17200 = 12040 \text{ PSI}$$

STRENGTH OF CONNECTING ELEMENTS

FILLET WELD IN SHEAR

$$1.57 \times .840 \times .125 \times 8400 = 1385 \text{ LBS.}$$

NOZZLE NECK IN SHEAR

$$1.57 \times .731 \times .109 \times 12040 = 1510 \text{ LBS.}$$

GROOVE WELD IN TENSION

$$1.57 \times .840 \times .125 \times 12700 = 2040 \text{ LBS.}$$

ALL PARTS ARE STRONGER THAN 120[#] LOAD
WHICH MUST BE CARRIED BY WELD.

PREPARED BY L.H. CHECKED BY W. H. H. - REVISED BY U
H/A/60

ANALYSIS
OF

PM-1 ECONOMIZER

TUBE SHEET

REF SUBSECTION A

HOLES
PITCH .375"
DIA .250"

3/8
HOLE SPACING

$$\text{LIGAMENT EFF.} = \frac{P-d}{P} = \frac{.375-.250}{.375} = .333 \checkmark$$

$$E = 33\%$$

REF. UG-53

MATERIAL - SA 240 TYPE 316
S = 17200 PSI @ 500°F

C = .25 REF. UG-340

$$t_{REQD} = d \sqrt{\frac{CP}{SE}}$$

$$t = 3.826 \sqrt{\frac{.25 \times 1500}{.333 \times 17200}}$$

$$t = 3.826 \sqrt{\frac{375}{5733}} = 3.826 \sqrt{.0654}$$

$$t = 3.826 \times .256 \checkmark$$

$$t_{REQD} = .979" \checkmark$$

$$t_{ACT} (\text{THINNEST SECTION}) = 1.625 - .531 = 1.094"$$

PREPARED BY KIL
11/11/20

CHECKED BY [Signature]

REVISED BY _____

ANALYSIS
OF
PM-1 ECONOMIZER

TUBE SHEET (CONT'D)

JOINT STRESS

$$t = \frac{PR}{SE - 0.6P}$$

FOR "E" USE .70 PER
PP. UW-12

$$t = \frac{1500 \times 1.913}{17200 \times .70 - .6 (1500)}$$

$$t = \frac{2869.5}{12040 - 900} = \frac{2869.5}{11140}$$

$$t = .257" \checkmark$$

$$t_{ACT.} = .337"$$

JOINT STRESS AT JUNCTION OF 4" SCHEDULE 40
TO 4" SCHEDULE 80 PIPE

$$t = \frac{PR}{SE - 0.6P}$$

$$t = \frac{1500 \times 1.963}{17200 \times .85 - .6 (1500)}$$

$$t = \frac{2744.5}{13720} \checkmark$$

$$t_{REQ'D} = .200" \checkmark$$

$$t_{ACT.} = .237"$$

WELD RADIOGRAPHED
USE E = .85

PREPARED BY div

CHECKED BY J. Brooks

REVISED BY _____

11/11/60

ANALYSIS
OF
PM-1 ECONOMIZER

HAIRPIN TUBING

1/4 O.D. X .035 W.T.
AISI 304 STAINLESS

DESIGN PRESSURE 1500 PSI
DESIGN TEMPERATURE 500°F

ALLOWABLE S FOR 304 STAINLESS
@ 500°F = 12500

$$t_{REQ'D} = \frac{PR}{SE - .6P}$$

REF: UG-27C

$$t_{REQ'D} = \frac{1500 \times .125}{12500 - 900} = .01620$$

$$t_{ACTUAL} = .023"$$

MIN TUBE
WALL .023

W. O. NO. 2288-12

BURNS AND ROE, INC.
NEW YORK, N. Y.

DATE: 9/66

DRAWING NO.

BY: R. J. DOBBS

TITLE: PM-3A STRESS ANALYSIS

SHEET 12 OF 12

REFERENCES

- 1 - ASME BOILER AND PRESSURE VESSEL
CODE SECTION VIII UNFIRED PRESSURE
VESSELS - 1965.

W. O. NO. 2288-12

BURNS AND ROE, INC.
NEW YORK, N. Y.

DATE: 9/66

DRAWING NO. _____

BY: R. J. DOBBS

TITLE: _____ SHEET 1 OF 17

STRESS ANALYSIS

COOLER

PM-3A

ITEM-6.3

W. O. NO. 2288-12

BURNS AND ROE, INC.
NEW YORK, N. Y.

DATE: 9/66

DRAWING NO.

BY: R. J. DOBBS

TITLE: PM-3A STRESS ANALYSIS

SHEET 2 OF 17

INTRODUCTION & SUMMARY

THE COOLER IS A U-TUBE HEAT EXCHANGER. THE VESSEL IS FABRICATED FROM 6 INCH SCH. 40 PIPE ON THE SHELL SIDE AND 8 INCH PIPE 1/2 INCH THICK ON THE TUBE SIDE.

THE BASIC MATERIALS OF CONSTRUCTION ARE ASTM-A 213 TYPE 304 AND ASTM-A 240 TYPE 304 STAINLESS STEEL OR EQUIVALENT.

THE VESSEL WAS ANALYZED IN ACCORDANCE WITH ASME BOILER AND PRESSURE VESSEL CODE SECTION VIII UNFIRED PRESSURE VESSELS.

THE SPECIFIED DESIGN CONDITIONS ARE:

SHELL SIDE - 95 PSIG PRESSURE AT 250°F

TUBE SIDE - 1500 PSIG PRESSURE AT 500°F

THE ACTUAL THICKNESS OF THE VARIOUS COOLER COMPONENTS WAS IN EACH CASE LESS THAN THE CALCULATED MINIMUM REQUIRED BY CODE AND THEREFORE ACCEPTABLE.

REINFORCEMENT AND FLANGE CALCULATIONS WERE CHECKED AND FOUND TO BE ACCEPTABLE

NOTE THAT THE CALCULATIONS PRESENTED IN THIS REPORT ARE FOR THE PM-1 COOLER. IT WAS ASSUMED THAT THE PM-3A COOLER IS IDENTICAL.

ANALYSIS
OF
PM-1 COOLER

REF DW'G 372-2876052
SHEETS 2

DESIGN CONDITIONS

DESIGN PRESSURE - 1500 PSI
DESIGN TEMPERATURE - 500°F
OPERATING PRESSURE - 1300 PSI

- HIGH PRESSURE
SIDE

DESIGN PRESSURE - 95 PSI
DESIGN TEMPERATURE - 250°F

- LOW PRESSURE
SIDE

END CAP

6" SCHEDULE 40
SA 213 TYPE 304 STAINLESS STEEL OR
EQUIVALENT.

REINFORCEMENT FOR NOZZLE PENETRATION

$$t = \frac{PLM}{2SE - 0.2P} \quad \frac{L}{r} = 5 \quad \text{REF. UR-4.4}$$

$$t_{REQ'D} = \frac{95 \times 5.31 \times 1.31}{2 \times 15000 \times .85 - 0.2(95)}$$

$$t_{REQ'D} = \frac{660.82}{25480} = .025" /$$

$$t_{ACT} = .280"$$

S FOR 304 = 15000 @ 300°

ANALYSIS
OF
PM-1 COOLER

NOZZLE t REQUIRED

$$t_{REQ'D} = \frac{PR}{SE - 0.6P} \quad \text{REF. UG - 27}$$

$$t_{REQ'D} = \frac{95 \times .311}{15000 \times .70 - .6 (95)} = \frac{29.51}{10496}$$

$$t_{REQ'D} = .030"$$

$$t_{ACT.} = .109"$$

AREA OF REINFORCEMENT REQUIRED

$$A = d t_r = .622 \times .025 = .0155 \text{ in.}^2$$

AREA PROVIDED

$$A_1 = .622 (.280 - .025) = (.622 \times .255) = .158 \text{ in.}^2$$

REINFORCEMENT IS ADEQUATE.

ANALYSIS
OF

TUBE SHEET

PITCH = .5625
DIA. = .360

$$\text{LIGAMENT EFF.} = \frac{P-d}{P} = \frac{.5625 - .360}{.5625} = .360 \quad \text{REF UG-53}$$

MATERIAL = SA 240 TYPE 304

S = 15100 PSI @ 500 °F

$$t_{req'd} = d \sqrt{\frac{C \cdot P}{S \cdot U}} \quad C = .25 \text{ REF. UG-34D}$$

$$t = 7.625 \sqrt{\frac{.25 \times 1500}{.36 \times 17200}} = 7.625 \sqrt{.0606}$$

$$= 7.625 (.246)$$

$$= 1.876 \text{ IN.}$$

$$t_{act} = 2.4375 - .5 = 1.9375 \text{ IN.}$$

(INCLUDES 1/2 IN. REDUCTION DUE TO SLOT)

PREPARED BY _____

CHECKED BY _____

REVISED BY _____

ANALYSIS
OF
PM-1 COOLER

END CAP

$$h_g = \frac{Q - G}{2} = 1.16875$$

$$d = G = 5.6875$$

$$p = 1500 \text{ PSI } T = 500$$

MATERIAL TYPE 316 STAINLESS STEEL

$$C = .30 + \frac{1.40 W h_g}{H_d} \leftarrow \text{REF. FIG. UG 34-h}$$

$$W = \frac{\pi}{4} G^2 p + (2b)(3.14) G m p$$

$$m = 3.0$$

$$W = 38000 + 26800$$

$$p = 1500$$

$$W = 64800$$

$$b = .25$$

REF. UA PP. 472

$$H = \frac{\pi}{4} G^2 p = 38000$$

$$C = .30 + \frac{1.4 (64800) (1.16875)}{38000 \times 5.6875}$$

$$C = .30 + .49 = .79$$

$$t = d \sqrt{\frac{C p}{S}} = 5.6875 \sqrt{\frac{.79 \times 1500}{17200}}$$

REF.
PP. UG 34

$$t = 1.5 \text{ INCHES}$$

$$t_{\text{ACTUAL}} = 2.0''$$

ANALYSIS
OF

COOLER NOZZLE - HIGH PRESSURE SIDE

DESIGN CONDITIONS

$P = 1500 \text{ PSI}$
 $T = 500 \text{ }^{\circ}\text{F}$

MATERIAL

SA 240 TYPE 304 S.S.

ALLOWABLE STRESS

$S = 15000 \text{ PSI}$

WELDS FULLY RADIOGRAPHED $E = 1$

PREPARED BY _____ CHECKED BY _____ REVISED BY _____

ANALYSIS
OF

SHELL

$$\begin{aligned} t_{ry} &= \frac{PR}{SE - .6p} \\ &= \frac{1500 \times \frac{7.625}{2}}{15000 - .6 \times 1500} \\ &= .406 \text{ IN} \end{aligned}$$

NOZZLE

$$\begin{aligned} t_{rn} &= \frac{PR}{SE - .6p} \\ &= \frac{1500 \times .75}{15000 - 900} \\ &= .080 \text{ IN.} \end{aligned}$$

PREPARED BY _____ CHECKED BY _____ REVISED BY _____

ANALYSIS
OF

AREA REMOVED

$$A = d t_r = 1.5 \times .406 = .609 \text{ IN.}^2$$

AREA REMOVED IN SHELL

$$\begin{aligned} A_1 &= d (t_{ACT} - t_r) \\ &= 1.5 (.5 - .406) \\ &= .141 \text{ IN.}^2 \end{aligned}$$

AREA REMOVED IN NOZZLE

$$\begin{aligned} A_2 &= 2 a (t_N - t_{rN}) \\ a &= 2.5 \times .375 + .094 = 1.03'' \\ A_2 &= 2 (1.03) (.375 - .080) = .608 \text{ IN.}^2 \end{aligned}$$

AREA IN WELD

$$A_3 = 2 \times \frac{1}{2} \times \left(\frac{1}{4}\right)^2 = .063 \text{ IN.}^2$$

TOTAL AREA REPLACED

$$A_1 + A_2 + A_3 = .812 \text{ IN.}^2$$

PREPARED BY _____ CHECKED BY _____ REVISED BY _____

ANALYSIS
OF
PM-1 COOLER

PIPE JOINT STRESS

JUNCTION OF END FLANGE TO PIPE

FOR E USE .85 PER
PR. UW-12

$$t = \frac{PR}{SE - 0.6P}$$

$$t = \frac{1500 \times 3.812}{17200 \times .85 - .6(1500)}$$

$$t = \frac{5720}{14600 - 900} = \frac{5720}{13700} = .417 \text{ in.}$$

$$t_{REQ'D} = .417 \text{ in.}$$

$$t_{ACT.} = .500 \text{ in.}$$

PREPARED BY SM

CHECKED BY 11/11/60

REVISED BY _____

ANALYSIS
OF
PM-1 COOLER

HAIRPIN TUBING

3/8 O.D. X .049 WALL THICK.
AISI 304 STAINLESS

DESIGN PRESSURE - 1500 PSI
DESIGN TEMPERATURE - 500°F

ALLOWABLE S FOR 304 STAINLESS
@ 500°F = 12500

$$S = 12500$$

$$t = .033"$$

$$D = .375$$

$$t_{REQ'D} = \frac{PR}{SE - .6P} = \frac{1500 \times .1875}{12500 - 900}$$

$$t_{REQ'D} = .024"$$

$$t_{ACT} = .033"$$

.033" MIN
W.T.

PREPARED BY PLH
11/1/60

CHECKED BY PLH

REVISED BY _____

ANALYSIS
OF
COOLER FLANGE

ASME CODE ANALYSIS - FLANGE REF. PP

UA-47

A = 10.5"
B = 5.125"
C = 8.125"
G = 5.6875"
E = 7.675

GASKET DETAILS

m = 3.0
y = 4500 PSI
b = b₀ = .25

$$W_{m1} = (2b)(3.14)GmP + \frac{\pi}{4}G^2P$$

$$W_{m1} = (2)(.25)(3.14)(5.6875)(3.0)(1500) + \frac{(3.14)(5.6875)^2}{4}(1500)$$

$$W_{m1} = 26800 + 39000$$

$$= 64800 \#$$

$$W_{m2} = (3.14)bGy = 3.14 \times .25 \times 5.6875 \times 4500$$

$$= 20200 \#$$

MATERIAL TYPE 316 STAINLESS STEEL

PREPARED BY J. Goller 11/1/60 CHECKED BY J. S. REVISED BY _____

ANALYSIS
OF

FLANGE MOMENTS

OPERATING

$$R_G = \frac{C-G}{2} = \frac{8.125 - 5.6875}{2} = 1.16875''$$

$$R_T = R_G + \frac{G-B}{2} = 1.16875 + .28125 = 1.45''$$

$$R_O = \frac{C-E}{2} = \frac{8.125 - 7.625}{2} = .25''$$

$$H_O = \frac{\pi}{4} \rho E^2 = (.785)(1500)(7.625)^2 = 68500 \#$$

$$H_T = \frac{\pi}{4} \rho (G^2 - B^2) = (.785)(1500)(5.6875^2 - 5.125^2) = 7100 \#$$

$$H_G = 26800 \#$$

$$H_P = H_O - \left(\frac{\pi}{4}\right) B^2 \rho = 68500 - (.785)(5.125)^2(1500)$$

$$H_P = 37600 \#$$

$$M_O = H_O R_O = 68500 \times .25 = 17100 \text{ "}\#$$

$$M_T = H_T R_T = 7100 \times 1.45 = 10300$$

$$M_G = H_G R_G = 26800 \times 1.16875 = 31100$$

$$M_P = -H_P R_P = 37600 \times 1.5 = -56500$$

$$M_O = M_O + M_T + M_G + M_P = 2000 \text{ "}\#$$

PREPARED BY

J. J. [Signature]
11/11/60

CHECKED BY

REVISED BY

ANALYSIS
OF

FLANGE MOMENTS (BOLTING UP)

$$M = W_{m2} \times R_g$$

$$= 20200 \times 1.16875 = 23600 \text{ "H}$$

BOLTING UP IS CRITICAL

SHAPE FACTORS

FOR A CONSERVATIVE ANALYSIS ASSUME THE FLANGE IS CUT AWAY AS SHOWN BY THE DOTTED LINE ON THE PREVIOUS PAGE. THEN $g_1 = g_0 = .5$

$$\frac{A}{B} = \frac{10.5}{5.125} = 2.05$$

$$\left. \begin{array}{l} T = 1.6 \\ Z = 1.8 \\ Y = 3.0 \\ U = 3.2 \end{array} \right\} \text{REF UA 51.2}$$

$$\frac{g_1}{g_0} = 1 \quad g_1 = g_0 = .5$$

$$F = 0.909 \quad \text{REF UA 51.2}$$

$$R_0 = \sqrt{B g_0} = \sqrt{5.125 \times .5} = 1.6$$

$$e = \frac{F_0}{R_0} = \frac{0.909}{1.6} = .567$$

$$f = 1 \quad \text{REF UA 51.6}$$

PREPARED BY

J. D. [Signature]
11/1/60

CHECKED BY

REVISED BY

ANALYSIS
OF

$$V = .550 \quad \text{REF UA 51.2}$$

$$d = \frac{U}{V} R_o q_o^2 = \left(\frac{3.2}{.550} \right) (1.6) (1.5)^2 = 1.42$$

$$L = \frac{tc+1}{T} + \frac{t^3}{d} = \frac{(1.5)(.567)+1}{1.6} + \frac{1.5^3}{1.42}$$

$$L = 1.16 + 2.38$$

$$L = 3.54$$

CALCULATED FLANGE
STRESS

$$S_H = \frac{f M_o}{L q_o^2 B} = \frac{(1)(23600)}{(3.54)(1.5)^2(5.125)}$$

$$S_H = \underline{5200 \text{ PSI}}$$

RADIAL FLANGE STRESS

$$S_R = \frac{\left(\frac{4}{3} tc + 1 \right) M_o}{L t^2 B}$$

$$S_R = \frac{\left[\left(\frac{4}{3} \right) (1.5) (.567) + 1 \right] 23600}{(3.54)(1.5)^2(5.125)}$$

$$S_R = \underline{1240 \text{ PSI}}$$

ANALYSIS
OF

TANGENTIAL FLANGE STRESS

$$S_T = \frac{Y M_o}{t^2 B} - Z S_R$$

$$= \frac{3.0 \times 23600}{1.5^2 \times 5.125} - (1.8)(1240)$$

$$= 6150 - 2230$$

$$S_T = 3920 \text{ PSI}$$

ALL FLANGE STRESSES ARE LESS THAN
ALLOWABLE ATMOSPHERIC STRESS ALLOWABLE
FOR TYPE 316 STAINLESS STEEL 1211750 PSI.

HEAD BOLTS - COOLER

$$W_{m1} = 64800 \text{ "}$$

$$W_{m2} = 20,200 \text{ "}$$

No. OF BOLTS - 8
SIZE OF BOLTS 1 1/8 - 7 UNC - 3 A
ROOT AREA = .7627 IN.²
TOTAL AREA = 8 X .7627 = 6.10 IN.²

$$(A_R)_{MIN} = \text{GREATER OF } \frac{W_{m2}}{S_{OP}} \text{ OR } \frac{W_{m2}}{S_{ATM}}$$

MATERIAL - SA 325
S = 18750

$$(A_R)_{MIN} = \frac{64800}{18750} = 3.46 \text{ IN}^2$$

$$\text{ACTUAL AREA} = 6.10 \text{ IN}^2$$

PREPARED BY

CHECKED BY

REVISED BY

11/11/60

W. O. NO. 2288-12

BURNS AND ROE, INC.
NEW YORK, N. Y.

DATE: 9/66

DRAWING NO. _____

BY: R. J. DOBBS

TITLE: PM-3A STRESS ANALYSIS

SHEET 17 OF 17

REFERENCES

- 1- ASME BOILER AND PRESSURE VESSEL
CODE SECTION VIII UNFIRED PRESSURE
VESSELS - 1965

W. O. NO. 2288-12

BURNS AND ROE, INC.
NEW YORK, N. Y.

DATE: 4-28-66

DRAWING NO. _____

BY: R. J. D.

TITLE: _____ SHEET 1 OF 15

STRESS ANALYSIS

PM-3A

CONTROL ROD PRESSURE THIMBLE

ITEM 7.0

W. O. NO. 2288-12BURNS AND ROE, INC.
NEW YORK, N. Y.DATE: 4-28-66

DRAWING NO. _____

BY: R. J. D.TITLE: CONTROL ROD PRESSURE THIMBLESHEET 2 OF 15

INTRODUCTION

THE CONTROL ROD DRIVE PRESSURE THIMBLE, WHICH IS FLANGED AND BOLTED TO THE REACTOR VESSEL TOP HEAD, CONSISTS OF FOUR MAJOR SECTIONS. THEY ARE VALVE AND MECHANICAL STOP, SCRAM SPRING HOUSING, ARMATURE HOUSING AND THE LOWER SECTION WHICH IS A SPOOL PIECE. THE SECTIONS WHICH ARE CYLINDRICAL IN SHAPE, ARE THREADED ONE INTO THE OTHER AND THEN SEAL WELDED.

THE SKETCH ON SHEET 5 OF THE ANALYSIS INDICATES THE VARIOUS SECTIONS, THE MATERIALS OF CONSTRUCTION AND THE REFERENCE DRAWINGS FROM WHICH DETAILS CAN BE OBTAINED.

SINCE THE CONTROL ROD DRIVE THIMBLE CONTAINS PRIMARY FLUID THE INTERNAL DESIGN PRESSURE IS EQUAL TO THE REACTOR VESSEL DESIGN PRESSURE OF 1500 PSIA. THE DESIGN TEMPERATURE IS 500°F.

THIS INVESTIGATION IS TO CHECK THE STRUCTURAL INTEGRITY OF THE AS-BUILT COMPONENTS.

THE BASIC ANALYSIS CONSISTS IN DETERMINING THE THICKNESS REQUIRED FOR THE VARIOUS SECTIONS AND COMPARING THEM AGAINST ACTUAL. THE DESIGN BASIS FOR THIS PORTION OF THE ANALYSIS IS THE ASME CODE FOR UNFIRED PRESSURE VESSELS SECTION VIII.

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BURNS AND ROE, INC.
NEW YORK, N. Y.

DATE: 4-28-66

DRAWING NO. _____

BY: R. J. D.

TITLE: CONTROL ROD PRESSURE THIMBLE

SHEET 3 OF 15

INTRODUCTION (CONT)

THE INVESTIGATION ALSO CONSIDERED PRESSURE DISCONTINUITY STRESSES. ONE CRITICAL AREA WAS ANALYZED. THIS IS AT THE TOP PORTION OF THE SPOOL PIECE, AT SECTION D-D (REF SKETCH ON SHEET 5).

THE DESIGN BASIS FOR THE DISCONTINUITY ANALYSIS IS THE "TENTATIVE STRUCTURAL DESIGN BASIS FOR REACTOR PRESSURE VESSELS AND DIRECTLY ASSOCIATED COMPONENTS" OFTEN REFERRED TO AS THE "NAVY CODE".

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BURNS AND ROE, INC.
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DATE: 4-28-66

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BY: R. J. D.

TITLE: CONTROL ROD PRESSURE THIMBLE

SHEET 4 OF 15

SUMMARY

THE RESULTS OF THE ANALYSIS INDICATE THAT THE BASIC THICKNESS OF ALL THE SECTIONS OF THE PRESSURE THIMBLE ARE ACCEPTABLE. IN EACH CASE THE ACTUAL THICKNESS IS EQUAL TO OR GREATER THAN THAT REQUIRED BY CODE.

THE RESULTS OF THE PRESSURE DISCONTINUITY ANALYSIS, AT SECTION D-D OF THE SPOOL PIECE, INDICATES A MAXIMUM STRESS INTENSITY "SP" (DEFINED AS TWICE THE MAXIMUM SHEAR) EQUAL TO 17,000 LBS/IN². THIS IS LESS THAN THE ALLOWABLE "SP" WHICH IS EQUAL TO 18100 LBS/IN², AND THEREFORE ACCEPTABLE.

— THE CONTROL ROD PRESSURE THIMBLE IS THEREFORE ACCEPTABLE FOR THE SPECIFIED DESIGN CONDITIONS.

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BURNS AND ROE, INC.
NEW YORK, N. Y.

DATE: 4-28-66

DRAWING NO.

BY: R. J. DOBBS

TITLE: PM-3A CONTROL ROD PRESSURE THIMBLE

SHEET 5 OF 15

W. O. NO. 2288-12BURNS AND ROE, INC.
NEW YORK, N. Y.DATE: 4-28-66

DRAWING NO. _____

BY: R. J. DOBBSTITLE: PM-3A CONTROL ROD PRESSURE THIMBLE SHEET 6 OF 15(I) - DESIGN CONDITIONS

INTERNAL PRESSURE = 1500 psia

TEMPERATURE = 500°F

(II) ALLOWABLE STRESSES (AT DESIGN TEMPERATURE)

<u>MATERIAL</u>	BASIC MEMBRANE ALLOWABLE (S_m)	STRESS INTENSITY ALLOWABLE (S_p)
ASTM A-276 TYPE 304 STAINLESS STL [ANNEAL & WATER QUENCH TENSILE MIN = 75000 YIELD MIN = 30000]	12500 [*] psi	17,500 [*] psi
ASTM A-276 TYPE 403 STAINLESS STL [HARDEN & TEMPER TENSILE MIN. = 110000 psi YIELD MIN. = 90000 psi]	29000 [⊗] psi	51,500 [⊗] psi
ASTM A-182 TYPE F-316 STAINLESS STL [ANNEAL & WATER QUENCH TENSILE MIN = 75000 YIELD = 30000]	17200 [*] psi	18100 [✕] psi

* REFERENCE #1

✕ REFERENCE #2

⊗ REFERENCE #2 (BASED ON 410 S.S. DATA)

✕ BASED ON 90% OF YIELD STRENGTH, IN ACCORDANCE WITH
REFERENCE #2

(III) BASIC THICKNESS CALCULATIONS(A) HOUSING - SCRAM SPRING

CHECK THICKNESS REQ'D AT SECTION A-A (REF PAGE 5). ACTUAL MIN. THICK. OCCURS AT THIS SECTION.

MAT'L = ASTM A-276 TYPE 304 SS

$$t_r \text{ (REQ'D)} = \frac{p R_i}{S E - 0.6 P} + \text{CORROSION} \quad \text{REF. \#1}$$

$$t_r = \frac{1500(1.017)}{12500 - 0.6(1500)} + 0$$

E = 100% Joint Efficiency
p = Design Pressure
S = ALLOWABLE "S_m"
SEE P. 6.

$$t_r = \frac{1520}{11600} = 0.1312''$$

$$t_a \text{ (actual)} = 1.225 - 1.017 = 0.208''$$

$$(\text{OK}) \quad t_{\text{actual}} > t_{\text{req'd}}$$

SINCE CHECK WAS MADE FOR THE MINIMUM SECTION, ALL OTHER SECTIONS OF THE SCRAM SPRING - HOUSING ARE ACCEPTABLE.

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BY: R J. DOBBS

TITLE: PM-3A CONTROL ROD PRESSURE THIMBLE

SHEET 8 OF 15

III (CONT)(B) HOUSING - ARMATURE

CHECK THICKNESS REQ'D AT SECTION
B-B (REF. p 5). ACTUAL MIN. THICKNESS OCCURS
AT THIS SECTION.

MAT'L = ASTM A-276
TYPE 403 SS.

$$t_r = \frac{p R_i}{S E - 0.6 p} = \frac{1500 (1.185)}{29000 - 0.6 (1500)}, \text{ ZERO CORROSION!}$$

$$t_r = \frac{1780}{28100} = 0.0635"$$

$$t_a (\text{ACTUAL}) = 1.285 - 1.185 = 0.100"$$

$$(\text{OK}) \quad t_{\text{actual}} > t_{\text{req'd}}$$

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DRAWING NO. _____

BY: R. J. DOBBSTITLE: PM-3A CONTROL ROD PRESSURE THIMBLESHEET 9 OF 15III (CONT)(C) SPOOL THIMBLE

CHECK THICKNESS REQD AT SECTION C-C.

ACTUAL MIN. THICKNESS OCCURS AT THIS SECTION.

$$t_r = \frac{p R_i}{SE - 0.6p} = \frac{1500(1.815)}{29000 - 0.6(1500)} + \text{ZERO CORROSION}$$

$$t_r = \frac{2720}{28100} = 0.0968"$$

$$t_{ACT.} = 2.02765 - 1.815 = 0.21265 \quad \text{O.K. } t_{ACT} > t_{req'd}$$

(IV) PRESSURE DISCONTINUITY ANALYSIS

FOR THE ANALYSIS OF THE PRESSURE DISCONTINUITY STRESS AT SECTION D-D, THE FEMALE THREADED PORTION OF THE SPOOL THIMBLE WILL BE TREATED AS A SHORT CYLINDER WITH FULL END FIXITY PROVIDED BY THE HEAVY ADJACENT SECTION OF THE SPOOL THIMBLE.

FOR THE CYLINDER:

$$t_{avg} = 2.094 - 1.517 = 0.577''$$

$$D = \frac{Et^3}{12(1-\nu^2)} = \frac{28.5 \times 10^6 (.577)^3}{10.92} = 501,000$$

$$k = \sqrt[4]{\frac{3(1-\nu^2)}{R_m^2 t^2}} = \sqrt[4]{\frac{2.73}{(1.706)^2 (.577)^2}} = \sqrt[4]{2.82} = 1.296$$

$$\frac{G}{k} = \frac{6}{1.296} = 4.65 \quad \text{SINCE } L = 2.07'' < \frac{L}{k} \quad \text{THE ASSUMPTION}$$

OF A SHORT CYLINDER IS VALID

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BURNS AND ROE, INC.

DATE: 4-28-66

DRAWING NO. _____

NEW YORK, N. Y.

BY: M. GIDEN

TITLE: PM-3A CONTROL ROD PRESSURE THIMBLE

SHEET 10 OF 15

(IV) PRESSURE DISCONTINUITY ANALYSIS

FOR THE ANALYSIS OF THE PRESSURE DISCONTINUITY STRESS AT SECTION D-D, THE FEMALE THREADED PORTION OF THE SPOOL THIMBLE WILL BE TREATED AS A SHORT CYLINDER WITH FULL END FIXITY PROVIDED BY THE HEAVY ADJACENT SECTION OF THE SPOOL THIMBLE.

FOR THE CYLINDER:

$$t_{avg} = 2.094 - 1.517 = 0.577''$$

$$D = \frac{Et^3}{12(1-\nu^2)} = \frac{28.5 \times 10^6 (.577)^3}{10.92} = 501,000$$

$$k = \sqrt[4]{\frac{3(1-\nu^2)}{R_m^2 t^2}} = \sqrt[4]{\frac{2.73}{(1.706)^2 (.577)^2}} = \sqrt[4]{2.82} = 1.296$$

$\frac{L}{k} = \frac{2.07}{1.296} = 1.597$ SINCE $L = 2.07'' < \frac{L}{k}$ THE ASSUMPTION OF A SHORT CYLINDER IS VALID

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BY: M. GIDENTITLE: PM-3A CONTROL ROD PRESSURE THIMBLESHEET 11 OF 15

SHORT CYLINDER COEFFICIENTS: REF. ROARK PG. 277 - REF. #3

 $XL = 1.296(2.07) = 2.68$ INTERPOLATE FOR COEFFICIENT. $\delta, C_3 = 1.08$ END SHEAR (CASE I) $\theta, C_4 = 1.14$ $\delta, C_5 = 1.13$ END MOMENT (CASE II) $\theta, C_6 = 1.05$ BOUNDARY CONDITIONS AT FIXED END:

$$\delta = 0, \quad \theta = 0$$

DEFLECTION OF SHELL AT FIXED END:

$$\text{DUE TO PRESSURE, } \delta = \frac{p R_M^2 (1-\nu)}{E t} = \frac{.85 p R_M^2}{E t}$$

$$\text{DUE TO AXIAL FORCE, } \delta = \frac{V p a R_M}{2 E t}$$

$$\text{DUE TO SHEAR, } \delta = \frac{C_3 Q}{20 \lambda^3}$$

$$\text{DUE TO MOMENT, } \delta = \frac{C_5 M}{20 \lambda^2}$$

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NEW YORK, N. Y.

BY: M. GIDEN

TITLE: PM-3A CONTROL ROD PRESSURE THIMBLE

SHEET 12 OF 15

ROTATION OF SHELL AT FIXED END:

$$\text{DUE TO SHEAR, } \theta_Q = \frac{C_4 Q}{2D \lambda^3} = \frac{1.14 Q}{2(501000)(1.296)^3} = .677 Q \times 10^{-6}$$

$$\text{DUE TO MOMENT } \theta_M = \frac{C_6 M}{\lambda D} = \frac{1.05 M}{1.296(501000)} = 1.62 M \times 10^{-6}$$

$$\text{DUE TO AXIAL FORCE } \theta_M = \frac{C_6 (pae/2)}{\lambda D} = \frac{1.05 (.218 p)}{1.296(501000)} = .353 p$$

$$R_M = 1.80$$

$$a = 1.517$$

$$e = 0.288$$

$$t = .577$$

$$\frac{pae}{2} = \frac{p(1.517)(.288)}{2} = 0.218 p$$

$$\delta_Q = \frac{C_3 Q}{2D \lambda^3} = \frac{1.08 Q}{2(501,000)(1.296)^3} = .495 Q \times 10^{-6}$$

$$\delta_M = \frac{C_5 M}{2D \lambda^2} = \frac{1.13(M - pae/2)}{2(501,000)(1.296)^2} = .671(M - pae/2) \times 10^{-6}$$

$$\delta_p = \frac{.85 p (R_M)^2}{Et} = \frac{.85 p (1.80)^2}{28.5 \times 10^6 (.577)} = .168 p \times 10^{-6}$$

$$\delta_{p_1} = \frac{V p a e R_M}{2 E t} = \frac{.3 p (1.517)(1.80) 10^{-6}}{2(28.5)(.577)} = .0249 p 10^{-6}$$

FOR DEFLECTION:

$$\delta_{p_1} - \delta_p + \delta_Q - \delta_M = 0$$

$$.0249 p 10^{-6} - .168 p 10^{-6} + .495 Q 10^{-6} - .671(M - .218 p) 10^{-6} = 0$$

FOR ROTATION:

$$\theta_Q - \theta_M = 0$$

$$.677 Q 10^{-6} - 1.62 M 10^{-6} + .353 p 10^{-6}$$

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DATE: 4-28-66

DRAWING NO.

NEW YORK, N. Y.

BY: M. GIDEN

TITLE: PM-3A CONTROL ROD PRESSURE THIMBLE

SHEET 13 OF 15

SOLVING THE EQ. FOR δ & θ

$$\text{FOR } \delta, .1431p + .495p - .671M + .146p = 0$$

$$\text{FOR } \theta, .677Q - 1.62M = -.353P$$

$$.495Q - .671M = -.0029P$$

$$.677Q - 1.62M = -.353P$$

$$Q - 1.36M = -.00585p$$

$$Q - 2.393M = -.521p$$

$$-1.033M = -.5207p$$

$$M = .504p$$

$$Q - 1.36(.504p) = -.00585p$$

$$Q - .685p = -.0063p$$

$$Q = .678p$$

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DATE: 4-28-66

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NEW YORK, N. Y.

BY: M. GIDEN

TITLE: PM-3A CONTROL ROD PRESSURE THIMBLE

SHEET 14 OF 15

SHELL STRESS:

$$\sigma_{\text{AXIAL}} = \frac{p_a}{2t} = \frac{1.517p}{2(.577)} = 1.314p$$

$$\sigma_{\text{HOOP}} = \frac{p_a}{t} = 2.628p$$

$$\sigma_{\text{BENDING}} = \frac{6M}{t^2} = \frac{6(.504p)}{(.577)^2} = 9.05p$$

$$\text{LONG STRESS} = 9.05p + 1.314p = 10.36p$$

$$\text{MAX. STRESS INTENSITY} = 10.36p - (-p) = 11.36p$$

FOR DESIGN CONDITIONS $p = 1500 \text{ PSI @ } 500^\circ\text{F}$

$$\sigma_{\text{MAX}} = 1500(11.36) = 17,000 \text{ PSI} < S_{\text{ALL}} = 18,100 \text{ PSI}$$

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BURNS AND ROE, INC.
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DATE: 4-28-66

DRAWING NO. _____

BY: R. J. D.

TITLE: PM-3A CONTROL ROD PRESSURE THIMBLE

SHEET 15 OF 15

REFERENCES

- 1 - ASME BOILER & PRESSURE VESSEL CODE
SECTION VIII - UNFIRED PRESSURE VESSELS
1965 EDITION
- 2 - TENTATIVE STRUCTURAL DESIGN BASIS FOR
REACTOR PRESSURE VESSELS AND DIRECTLY
ASSOCIATED COMPONENTS.
1-DEC-1958 REVISION. U.S. DEPT. OF COMMERCE
BULLETIN PB-151987.
- 3 - FORMULAS FOR STRESS AND STRAIN BY
ROARK - 3RD EDITION MCGRAW-HILL PUB.

W. O. NO. 2288-17

BURNS AND ROE, INC.
NEW YORK, N. Y.

DATE: 4-29-66

DRAWING NO. _____

BY: M. GIDEN

TITLE: _____ SHEET 1 OF 17

STRESS ANALYSIS

PM-3A

CONTROL ROD DRIVE TRANSMITTER & ARMATURE HOUSING

ITEM 7.1

W. O. NO. 2288-12

BURNS AND ROE, INC.
NEW YORK, N. Y.

DATE: 4-29-66

DRAWING NO.

BY: M. GIDEN

TITLE: CONTROL ROD DRIVE HOUSING

SHEET 2 OF 17

INTRODUCTION

THE CONTROL ROD DRIVE TRANSMITTER HOUSING CONSISTS OF WATER TIGHT HOUSING OF APPROXIMATELY CYLINDRICAL SHAPE 7 INCHES IN DIAMETER WITH A SHELL THICKNESS OF $\frac{1}{8}$ INCH, THE MATERIAL OF CONSTRUCTION IS 304 STAINLESS STEEL.

THE ARMATURE HOUSING IS A 6 INCH DIAMETER 304 STAINLESS STEEL SHELL, 0.120 INCHES THICK. THE DESIGN CONDITIONS USED IN THIS ANALYSIS ARE 51 PSI EXTERNAL PRESSURE AND 100°F TEMPERATURE.

THE DESIGN PRESSURE OF 51 PSI IS THE SUM OF A NORMAL OPERATING PRESSURE OF 6 PSI DUE TO STATIC HEAD OF LIQUID IN THE CONTAINMENT VESSEL PLUS 45 PSI WHICH WOULD OCCUR DURING LEAK TEST OF THE CONTAINMENT SYSTEM

THE SYSTEM HAS BEEN INVESTIGATED THAT FOR OPERATING CONDITIONS, THE STRESSES ARE BELOW THE ALLOWABLE VALUES OF THE CODE, AND DURING LEAK TEST OF THE CONTAINMENT VESSEL, BUCKLING OF THE CAUS IS NOT A PROBLEM.

W. O. NO. 2288-12

BURNS AND ROE, INC.
NEW YORK, N. Y.

DATE: 4-29-46

DRAWING NO.

BY: M. GIDEN

TITLE: CONTROL ROD DRIVE HOUSING

SHEET 3 OF 12

SUMMARY

THE RESULTS OF THE ANALYSIS INDICATE THAT THE BASIC MEMBRANE STRESS IN THE COMPONENT, SATISFY THE ASME CODE SECTION VIII, THE DISCONTINUITY STRESSES AT THE JUNCTION OF 003 & 011 INDICATES THAT THE MAXIMUM STRESSES ARE IN ACCORDANCE WITH THE ALLOWABLE IN TENTATIVE STRUCTURAL DESIGN BASED.

AN INVESTIGATION FOR BUCKLING, INDICATES THAT BUCKLING IS NOT A PROBLEM. THEREFORE, THE SYSTEM IS ACCEPTABLE FOR THE SPECIFIED DESIGN CONDITIONS.

W. O. NO. 2288-12

BURNS AND ROE, INC.
NEW YORK, N. Y.

DATE: 4-29-66

DRAWING NO.

BY: M. GIDEN

TITLE: CONTROL ROD DRIVE HOUSING

SHEET 4 OF 17

CONCLUSIONS

STRESS ANALYSIS OF THE PM-3A CONTROL ROD DRIVE HOUSING, INDICATES THAT THE DESIGN PRESSURE OF 51 P.S.I. WILL NOT CAUSE BUCKLING OF THE TRANSMITTER OR ARMATURE COIL HOUSINGS.

FOR NORMAL OPERATING CONDITIONS (6 PSI, HYDROSTATIC HEAD) THE MAXIMUM STRESS IS 3030 PSI, WHICH IS BELOW THE ALLOWABLE STRESS (18750 PSI.) OF THE 304 S.S. HOUSING MATERIAL.

W. O. NO. 2288-12

BURNS AND ROE, INC.

DATE: 4-29-66

DRAWING NO. _____

NEW YORK, N. Y.

BY: M. H. GIDEN

TITLE: CONTROL ROD DRIVE HOUSING

SHEET 5 OF 17

FOR 14' OF WATER ABOVE REACTOR, $p = \frac{14(62.4)}{144} = 6 \text{ PSI.}$

FOR $p = 45 \text{ PSI (TEST)} + 6 \text{ PSI (HYDROSTATIC)} = 51 \text{ PSI}$

AT 100°F FOR 304 S.S. $S_{ALL} = 18,750 \text{ PSI @ } 100^\circ\text{F}$

$S_{YIELD} = 30,000 \text{ PSI @ } 100^\circ\text{F}$

W. O. NO. 2288-12

DRAWING NO.

BURNS AND ROE, INC.
NEW YORK, N. Y.

DATE: 9-29-64

BY: M. GARDNER

TITLE: TRANSMITTER HOUSING

SHEET 6 OF 17

TRANSMITTER HOUSING

DWG. 387-2271036

W. O. NO. 2288-13

BURNS AND ROE, INC.
NEW YORK, N. Y.

DATE: 4-29-66

DRAWING NO. _____

BY: M. GIDEN

TITLE: _____ SHEET 7 OF 17

VIEW A-A

TRANSMITTER HOUSING

DWG. 387-2271036

W. O. NO. 2288-12

BURNS AND ROE, INC.
NEW YORK, N. Y.

DATE: 4-29-66

DRAWING NO.

BY: M. GREN

TITLE: DETAILS TRANSMITTER HOUSING PM-3A

SHEET 8 OF 17

DETAIL - 001

DETAIL - 003

DETAIL - 007

DWG. 387-2271035

W. O. NO. 2288-12BURNS AND ROE, INC.
NEW YORK, N. Y.DATE: 4-29-66

DRAWING NO. _____

BY: M. GIDENTITLE: TRANSMITTER HOUSINGSHEET 9 OF 17ITEM 001

ASSUME THAT ITEM 001 CAN BE TREATED AS A CYLINDRICAL SHELL.

$$P = \frac{K t^2}{L R} \sqrt{\frac{t}{R}}$$

REF. 2

$$K = 23.2 \times 10^6$$

$$t = .125$$

$$R = 3.488$$

$$L = 37.69$$

$$P_{cal.} = \frac{23.2 \times 10^6 (.125)^2}{37.69 (3.488)} \sqrt{\frac{.125}{3.488}} = 5250 \text{ PSI} > P_{max} = 51 \text{ PSI.}$$

SHELL STRESS:

$$\sigma = \frac{p R}{t} = \frac{3.488 \cancel{p}}{.125} = 27.9 \cancel{p}$$

$$\sigma = 27.9(51) = 1423 \text{ PSI.} < S_{ALL.} = 18,750 \text{ PSI}$$

FOR NORMAL OPERATING CONDITIONS,

$$\sigma = 27.9(6) = 167 \text{ PSI.}$$

W. O. NO. 2233-12

BURNS AND ROE, INC.
NEW YORK, N. Y.

DATE: 4-29-66

DRAWING NO. _____

BY: M. GIDEN

TITLE: TRANSMITTER HOUSING PM-3A

SHEET 10 OF 17

DETAIL 003

CONSIDER A ONE INCH STRIP OF DETAIL 003 TO BE A
FIXED END ARCH UNDER UNIFORM PRESSURE (REF. 3)

$$P_{cr} = \frac{EI}{R^3} (k^2 - 1) \quad \text{ENGINEERING DESIGN, J. FAUPEL}$$

JOHN WILEY 1964

$$R = 3.488$$

$$k = 3 \quad (\text{FIG. 8.44})$$

$$I = \frac{1}{12} b t^3 = \frac{1}{12} (1) (.125)^3 = 163 \times 10^{-6} \text{ in}^4$$

$$E = 28.2 \times 10^6$$

$$P_{cr} = \frac{28.2 (163)}{(3.488)^3} (9 - 1) = 865 \text{ PSI} > P_{max} = 51 \text{ PSI}$$

STRESS IN SHELL:

REF. 2 PAGE 163 CASE 30

W. O. NO. 2282-12BURNS AND ROE, INC.
NEW YORK, N. Y.DATE: 4-29-66

DRAWING NO. _____

BY: M. GIBENTITLE: TRANSMITTER HOUSING PM-3ASHEET 11 OF 17

$$H = \rho R \left[\frac{\frac{1}{4} \left(\frac{s^2 c}{\theta} - s \right) + \frac{s^3}{6} + d \left(\frac{1}{4} \theta - \frac{1}{2} \theta c^2 + \frac{1}{4} s c \right)}{\frac{(\theta - s)^2}{\theta} - \frac{3}{2} \theta + 2s - \frac{s c}{2} - d \left(\frac{\theta}{2} + \frac{s c}{2} \right)} \right]$$

$$S = \sin \theta$$

$$C = \cos \theta$$

$$\theta = \pi/2 = 1.57$$

$$d = \frac{I}{AR^2}$$

$$R = 3.488$$

$$d = \frac{\frac{1}{12} b t^3}{b t R^2} = \frac{t^2}{12(R^2)} = \frac{(.125)^2}{12(3.488)^2} = .000107 \text{ NEGLIGIBLE}$$

$$S = \sin 90^\circ = 1$$

$$C = \cos 90^\circ = 0$$

$$H_1 = 3.488 \rho \left[\frac{\overset{-.00033}{-.25 + .1667}}{\underset{-.1481}{.2069 - 2.355 + 2}} \right] = +1.962 \rho$$

$$V_1 = \rho R s = 3.488 \rho$$

$$M_1 = \rho R^2 \left(\frac{1}{2} s^2 - \frac{1}{4} \cdot \frac{1}{4} \frac{s c}{\theta} \right) - H R \left(\frac{s}{\theta} - c \right)$$

$$M_1 = 12.166 \rho \left(.50 \overset{.25}{-} .25 \right) - 6.84 \rho (.6369)$$

$$M_1 = 3.0415 \rho - 4.3586 \rho = -1.317 \rho$$

$$\text{FOR } \rho = 51 \text{ PSI.}$$

$$M_1 = -1.317(51) = -67$$

$$V_1 = 3.488(51) = +178$$

$$H_1 = 1.962(51) = +100$$

$$x = R - R \sin \theta$$

$$y = R \cos \theta$$

$$M = -M_1 + VR(1 - \sin \theta) - HR \cos \theta - \frac{\gamma R^2}{2}(1 - \sin \theta)^2$$

AT CROWN, $\theta = 0$

$$M = -M_1 + VR - HR - \frac{\gamma R^2}{2}$$

$$M = +67 + 3.488(178) - 3.488(100) - \frac{51}{2}(3.488)^2$$

$$M = 67 + 621 - 349 - 310 = +29$$

WHEN $\theta = 45^\circ$, $\sin \theta = \cos \theta = .707$

$$M = 67 + 621(.293) - 349(.707) - 310(.293)^2$$

$$M = 67 + 182 - 247 - 27 = -25$$

WHEN $\theta = 30^\circ$ $\sin \theta = .50$ $\cos \theta = .866$

$$M = 67 + 621(.5) - 349(.866) - 310(.5)^2$$

$$M = 67 + 310 - 302 - 78 = +3$$

WHEN $\theta = 60^\circ$ $\sin \theta = .866$ $\cos \theta = .50$

$$M = 67 + 621(.134) - 349(.50) - 310(.134)^2$$

$$M = 67 + 83 - 175 - 6 = -31$$

W. O. NO. 2288-12BURNS AND ROE, INC.
NEW YORK, N. Y.DATE: 4-29-66

DRAWING NO. _____

BY: M. GIDENTITLE: TRANSMITTER HOUSING PM-3ASHEET 13 OF 17STRESS IN SHELL:

$$\sigma = \frac{C M_1}{t^2} = \frac{C (67)}{(.125)^2} = 25,700 \text{ PSI}$$

FOR NORMAL OPERATING CONDITIONS, $P = 6 \text{ PSI}$

$$\sigma = \frac{C}{S_1} (25770) = 3030 \text{ PSI} < S_{\text{ALLOWABLE}} = 18,750 \text{ PSI}$$

$$\sigma_{\text{MEMBRANE}} = \frac{P}{t} = \frac{51 (3.48)}{(.125)} = 1400 \text{ PSI}$$

$$\sigma_c = 25,700 + 1400 = 27,100 \approx S_p (90\% \text{ of } S_y) \text{ REF. 4}$$

W. O. NO. _____

BURNS AND ROE, INC.

DATE: 4-29-66

DRAWING NO. 387-2271019

NEW YORK, N. Y.

BY: M. J. GIDEN

TITLE: CAN ASSEMBLY ARMATURE COIL

SHEET 14 OF 17

MAT. 304 S S. TUBE

DWG. 387-2271019

W. O. NO. 2288-12

BURNS AND ROE, INC.
NEW YORK, N. Y.

DATE: 4-29-56

DRAWING NO.

BY: M. GIDEN

TITLE: ARMATURE COIL HOUSING

SHEET 15 OF 17

MAXIMUM EXTERNAL WORKING PRESSURE: FIG. UHA-28.1

$$\frac{L}{D_o} = \frac{31.7}{6} = 5.3$$

$$\frac{D_o}{t} = \frac{6}{.120} = 50$$

FROM FIG UHA-28.1 REF 1

$$P(D_o/t) = 7500$$

$$P = \frac{7500}{50} = 150 \text{ PSI} > \text{DESIGN PRESSURE} = 51 \text{ PSI}$$

L = LENGTH OF VESSEL SECTION

D_o = OUTSIDE DIAMETER

t = SHELL THICKNESS

P = MAXIMUM WORKING PRESSURE

W. O. NO. 2288-12

BURNS AND ROE, INC.
NEW YORK, N. Y.

DATE: 4-29-66

DRAWING NO. _____

BY: M. GIBEN

TITLE: ARMATURE COIL CAN

SHEET 16 OF 17

SHELL STRESS:

$$\sigma = \frac{pR}{t} = \frac{2.94(12)}{.120} = 24.4$$

$$\sigma = 24.4(51) = 1244 \text{ PSI.} < S_{ALL} = 18,750 \text{ PSI.}$$

FOR NORMAL OPERATING CONDITIONS, (6 PSI)

$$\sigma = 24.4(6) = 146 \text{ PSI.} < S_{ALL} = 18,750 \text{ PSI.}$$

W. O. NO. 2288-12

BURNS AND ROE, INC.
NEW YORK, N. Y.

DATE: 4-22-66

DRAWING NO.

BY: M. GIDEN

TITLE: CONTROL ROD HOUSING

SHEET 17 OF 17

REFERENCES

1. ASME BOILER & PRESSURE VESSEL CODE SECTION VIII
UNFIRED PRESSURE VESSELS 1965 EDITION.
2. FORMULAS FOR STRESS & STRAIN BY ROARK 3RD EDITION
Mc GRAW HILL
3. ENGINEERING DESIGN. J. FAUPLE, J WILEY 1964
4. TENTATIVE STRUCTURAL DESIGN BASIS FOR REACTOR
PRESSURE VESSELS & DIRECTLY ASSOCIATED
COMPONENTS. 1 DEC. 1958

W. O. NO. 2288-12

BURNS AND ROE, INC.
NEW YORK, N. Y.

DATE: 4-29-66

DRAWING NO. _____

BY: R. J. D.

TITLE: _____ SHEET 1 OF 8

STRESS ANALYSIS

PM-3A

COOLANT DISCHARGE & VENT SYSTEM

EXPANSION TANK

ITEM 8.0

W. O. NO. 2288-12BURNS AND ROE, INC.
NEW YORK, N. Y.DATE: 4-29-66

DRAWING NO. _____

BY: R. J. D.TITLE: EXPANSION TANKSHEET 2 OF 8

INTRODUCTION

THE COOLANT DISCHARGE AND VENT SYSTEM EXPANSION TANK CONSISTS OF A 20 INCH DIA BY $\frac{1}{2}$ INCH THICK SHELL WITH $\frac{3}{8}$ INCH THICK HEMI-SPHERICAL HEADS. THE TANK PENETRATIONS ARE A 2 INCH DIAMETER INLET, A $1\frac{1}{2}$ INCH DIAMETER DRAIN AND A $\frac{1}{2}$ INCH DIAMETER INSTRUMENT CONNECTION. THE MATERIAL OF CONSTRUCTION IS TYPE 304 STAINLESS STEEL.

THE LISTED DESIGN CONDITIONS ARE 600.PSI INTERNAL PRESSURE AND 550. $^{\circ}$ F TEMPERATURE.

THE INVESTIGATION HAS BEEN MADE TO CHECK THE CONFORMANCE OF THE EXPANSION TANK WITH THE REQUIREMENTS OF THE ASME CODE FOR UNFIRED PRESSURE VESSELS SECTION VIII. THIS CONSISTS IN DETERMINING THE THICKNESS OF THE SHELL AND HEADS REQUIRED IN ACCORDANCE WITH THE CODE FOR THE DESIGN CONDITIONS. THESE CALCULATED VALUES ARE THEN COMPARED WITH THE ACTUAL THICKNESSES SUPPLIED.

THE CODE REQUIREMENT FOR REINFORCING PADS AT VESSEL NOZZLE PENETRATIONS IS ALSO INVESTIGATED.

W. O. NO. 2288-12

BURNS AND ROE, INC.
NEW YORK, N. Y.

DATE: 4-29-66

DRAWING NO. _____

BY: R. J. D.

TITLE: EXPANSION TANK

SHEET 3 OF 8

SUMMARY

THE RESULTS OF THE ANALYSIS INDICATE THAT THE ACTUAL THICKNESS OF THE EXPANSION TANK SHELL AND HEADS IS GREATER THAN THAT REQUIRED BY CODE.

THE REQUIREMENT FOR REINFORCING AROUND THE 2 INCH INLET, 1 1/2 INCH DRAIN AND 1/2 INCH INSTRUMENT CONNECTION IS NOT APPLICABLE SINCE THE CODE DOES NOT REQUIRE REINFORCING FOR PENETRATIONS 2 INCH AND SMALLER.

THE EXPANSION TANK IS THEREFORE ACCEPTABLE FOR THE SPECIFIED DESIGN CONDITIONS.

W. O. NO. 2288-12

BURNS AND ROE, INC.

DATE: 4-29-66

DRAWING NO.

NEW YORK, N. Y.

BY: M. H. GIDEN

TITLE: EXPANSION TANK COOLANT DISCHARGE SYSTEM SHEET 4 OF 8

CONCLUSIONS

THE COOLANT DISCHARGE AND VENT SYSTEM EXPANSION TANK WAS ANALYZED IN ACCORDANCE WITH SECTION VIII OF THE ASME BOILER CODE.

THE THICKNESS OF THE VESSEL SHELL & HEAD WAS CALCULATED FOR DESIGN CONDITIONS (600 PSI, 550°F). THE CALCULATED THICKNESS OF THE SHELL IS 0.461" AND THE HEAD IS 0.203".

THE AS BUILT THICKNESS OF THE SHELL IS 0.500" AND THE HEAD IS 0.375". SINCE THESE VALUES ARE GREATER THAN THE THICKNESSES REQUIRED BY CODE, THE EXPANSION TANK IS SATISFACTORY FOR THE DESIGN CONDITIONS.

W. O. NO. 2288-12

BURNS AND ROE, INC.
NEW YORK, N. Y.

DATE: 4-29-66

DRAWING NO. _____

BY: M. H. SIDEN

TITLE: EXPANSION TANK

SHEET 5 OF 8

EXPANSION TANK:

MATERIAL OF CONSTRUCTION:

SHELL - 304 S.S. 1/2" THICK SA 240

HEAD - 304 S.S. 3/8" THICK SA 240

DESIGN CONDITIONS:

PRESSURE : 600 PSI.

TEMPERATURE : 550 °F

REF: #1 FOR ABOVE DATA

W.O. NO. 2288-12BURNS AND ROE, INC.
NEW YORK, N. Y.DATE: 4-29-66

DRAWING NO. _____

BY: M. H. GIDENTITLE: COOLANT SYSTEM EXPANSION TANKSHEET 6 OF 8SHELL THICKNESS:

$$t = \frac{PR}{SE - 0.6P} \quad \text{PAR UG-32 REF. 2}$$

$$P = 600 \text{ PSI}$$

$$R = 10''$$

$$S = 14850 \text{ PSI. (TABLE UHA-23)}$$

$$E = 90\%$$

$$t = \frac{600(10)}{.90(14850) - .6(600)} = .461'' < t_{ACT} = .500''$$

HEAD THICKNESS:

$$t = \frac{PL}{2SE - .2P}$$

$$P = 600 \text{ PSI}$$

$$R_0 = 10''$$

$$S = 14850 \text{ PSI}$$

$$E = 100\%$$

$$t = \frac{600(10)}{2(14850) - .2(600)} = .203'' < t_{ACT} = .375''$$

NO CORROSION ALLOWANCE REQD SINCE THE VESSEL IS 304 S.S.

W. O. NO. 2288-12

BURNS AND ROE, INC.
NEW YORK, N. Y.

DATE: 4-29-66

DRAWING NO. _____

BY: M. GIDEN

TITLE: EXPANSION TANK

SHEET 7 OF 8

PENETRATION ANALYSIS:

2" INLET

1½" DRAIN

1½" INSTRUMENT CONN.

REINFORCEMENT OF THESE NOT REQUIRED, PER. PAR. UG 36-36
REF. 2

W. O. NO. 2288-12

BURNS AND ROE, INC.
NEW YORK, N. Y.

DATE: 4-29-66

DRAWING NO. _____

BY: R. J. D.

TITLE: EXPANSION TANK

SHEET 8 OF 8

REFERENCES

- 1 - FORM U-1 MANUFACTURER'S DATA REPORT FOR UNFIRED PRESSURE VESSELS - B. H. HUBBERT AND SONS - 1961.
- 2 - ASME BOILER & PRESSURE VESSEL CODE SECTION VIII - UNFIRED PRESSURE VESSELS 1965 EDITION.

FORM U-1 MANUFACTURERS' DATA REPORT FOR UNFIRED PRESSURE VESSELS

As required by the Provisions of the ASME Code Rules

1. Manufactured by B. H. Hubbert & Son, Inc., 1311 S. Penca St., Baltimore 24, Maryland
(Name and address of Manufacturer)

2. Manufactured for The Martin Company, Baltimore 3, Maryland
(Name and address of Purchaser)

Type Kind Vessel No. 17778 () () Natl. Bd. No. 2972 Yr. Built 1961
(Horiz. or Vert.) (Tank, Jacketed, Heat Exch.) (Mfrs. Serial) (State & State No.)

Items 4-9 incl. to be completed for single wall vessels (such as air tanks), jackets of jacketed vessels, or shells of heat exchangers.

4. SHELL: Material SA240(304) T.S. 75,000 Nominal Thickness .50 In. Allowance 0 In. Diam. 1 Ft. 8 In. Length 1 Ft. 0 In.
(Kind and Spec. No.) (Fig. or F.B. & Spec. Min. T.S.)

5. SEAMS: Long Dbl. Butt H.T. No X.R. Complete Sectioned No Efficiency 90 %
(Welded, Dbl., Single, Lap, Butt) (Yes or No)¹ (Spot or Complete) (Yes or No)

If riveted describe seams fully on reverse side of form.

Girth Dbl. Butt H.T. No X.R. Complete Sectioned No No. of Courses

6. HEADS (a) Material SA240 (304) T.S. 75,000 (b) Material T.S.
Location Thickness Crown Radius Knuckle Radius Elliptical Ratio Conical Apex Angle Hemispherical Radius Flat Diameter Side to Pressure
(Top, bottom, ends) (Convex or Concave)
(a) Top .375 10" Concave
(b) Bottom .375 10" Concave

If removable, bolts used (Material, Spec. No., T.S., Size, Number) Other fastening (Describe or Attach Sketch)

7. STAYBOLTS: If hollow Attachment Pitch X Diam.
(Material) (Size of Hole) (Threaded, Welded) (Horiz.) (Vert.) (Nominal)

8. JACKET CLOSURE:
(Describe as ogee & weld, bar, etc. If bar, give dimensions, if bolted, describe or sketch)

9. Constructed for max. allowable working press.² 600 psi at max. temp. 550 °F. Min. temp. (when less than -20°) °F. Hydrostatic Pneumatic or } Test Press 900 psi.
Combination

Items 10 and 11 to be completed for tube sections.

10. TUBE SHEETS: Stationary. Material Diam. In. Thickness In. Attachment
(Kind & Spec. No.) (Subject to Pressure) (Welded, Bolted)
Floating. Material Diam. In. Thickness In. Attachment
(Kind & Spec. No.)

11. TUBES: Material O.D. In. Thickness Inches or Gage Number Type
(Kind & Spec. No.) (Straight or U)

Items 12-15 incl. to be completed for inner chambers of jacketed vessels, or channels of heat exchangers.

12. SHELL Material T.S. Nominal Thickness In. Allowance In. Diam. Ft. In. Length Ft. In.
(Kind and Spec. No.) (Fig. or F.B. & Spec. Min. T.S.)

13. SEAMS: Long H.T. X.R. Sectioned Efficiency %
(Welded, Dbl., Single, Lap, Butt) (Yes or No)¹ (Spot or Complete) (Yes or No)

If riveted describe seams fully on reverse side of form.

Girth H.T. X.R. Sectioned No. of courses

14. HEADS (a) Material T.S. (b) Material T.S. (c) Material T.S.
Location Thickness Crown Radius Knuckle Radius Elliptical Ratio Conical Apex Angle Hemispherical Radius Flat Diameter Side to Pressure
(Top, bottom, ends) (Convex or Concave)
(b) Channel
(c) Floating

If removable, bolts used (a) (Material, Spec. No., T.S., Size, Number) (b)

(c) Other fastening (Describe or Attach Sketch)

15. Constructed for max. allowable working press.² psi at max. temp. °F. Min. temp. (when less than -20°) °F. Hydrostatic Pneumatic or } Test Press psi.
Combination

Items below to be completed for all vessels where applicable.

16. SAFETY VALVE OUTLETS: Number Size Location

Purpose (Inlet, Outlet, Drain)	Number	Diam. or Size	Type	Material	Thickness	Reinforcement Material	How Attached
Inlet	1	2" IPS	Coupling	304	Sch. 80	None	Welded
Drain	1	1-1/2" IPS	Coupling	304	Sch. 80	None	Welded
Inst. Conn.	1	1/2" IPS	Coupling	304	3000#	None	Welded

¹ If postweld heat-treated. ² List under remarks other internal or external pressures with coincident temperature when applicable.

FORM U-1 (back)

18. INSPECTION Manholes, No. _____ Size _____ Location _____
 OPENINGS: Handholes, No. _____ Size _____ Location _____
 Threaded, No. _____ Size _____ Location _____
 19. SUPPORTS: Skirt _____ Lugs _____ Legs _____ Other _____ Attached _____
 (Yes or No) (Number) (Number) (Describe) (Where & How)

20. REMARKS: This is an Expansion Tank built in accordance with 1959 ASME Code Section VII
Case 1270 H.

(Brief description of purpose of the vessel, as Air Tank, After Cooler, Jacketed Cooker, etc. State contents of each part.)

We certify that the statements made in this report are correct and that all details of design, material, construction, and workmanship of this vessel conform to the ASME Code for Unfired Pressure Vessels.

Date May 8, 19 61 Signed B. H. Hubbert & Son, Inc. By s/A. J. Panusho
 (Manufacturer)

Certificate of Authorization Expires December 31, 1961

CERTIFICATE OF SHOP INSPECTION

VESSEL MADE BY B. H. Hubbert & Son, Inc. at Baltimore, Maryland

I, the undersigned, holding a valid commission issued by the National Board of Boiler and Pressure Vessel Inspectors and/or the State of Maryland and employed by Lumbermens Mutual Casualty Co. of Chicago have inspected the pressure vessel described in this manufacturer's data report on _____ 19 _____, and state that to the best of my knowledge and belief, the manufacturer has constructed this pressure vessel in accordance with the applicable sections of the ASME Boiler and Pressure Vessel Code.

By signing this certificate neither the Inspector nor his employer makes any warranty, expressed or implied, concerning the pressure vessel described in this manufacturer's data report. Furthermore, neither the Inspector nor his employer shall be liable in any manner for any personal injury or property damage or a loss of any kind arising from or connected with this inspection.

Date May 8, 19 61

s/L. K. Styles

Inspectors Signature

Commissions NB 2568 PA 1413

Nat'l Board or State and No.

CERTIFICATE OF FIELD ASSEMBLY INSPECTION

I, the undersigned, holding a valid commission issued by the National Board of Boiler and Pressure Vessel Inspectors and/or the State of _____ and employed by _____ of _____

_____ have compared the statements in this manufacturer's data report with the described pressure vessel and state that parts referred to as data items _____, not included in the certificate of shop inspection have been inspected by me and that to the best of my knowledge and belief the manufacturer has constructed and assembled this pressure vessel in accordance with the applicable sections of the ASME Boiler and Pressure Vessel Code. The described vessel was inspected and subjected to a hydrostatic test of _____ psi.

By signing this certificate neither the Inspector nor his employer makes any warranty, expressed or implied, concerning the pressure vessel described in this manufacturer's data report. Furthermore, neither the Inspector nor his employer shall be liable in any manner for any personal injury or property damage or a loss of any kind arising from or connected with this inspection.

Date _____ 19 _____

Inspector's Signature

Commissions _____

Nat'l Board or State and No.

W. O. NO. 2288-12

BURNS AND ROE, INC.
NEW YORK, N. Y.

DATE: 4-29-66

DRAWING NO. _____

BY: R J.D.

TITLE: _____ SHEET 1 OF 8

STRESS ANALYSIS

PM-3A

COOLANT DISCHARGE & VENT SYSTEM

DRAIN COOLER

ITEM 8.1

W. O. NO. 2288-12BURNS AND ROE, INC.
NEW YORK, N. Y.DATE: 4-29-66

DRAWING NO. _____

BY: R. J. D.TITLE: DRAIN COOLERSHEET 2 OF 8

INTRODUCTION

THE COOLANT DISCHARGE AND VENT SYSTEM DRAIN COOLER CONSISTS OF A 3 INCH SCHEDULE 40 - 304 STAINLESS STEEL PIPE WITH SWAGED ENDS AND AN INTERNAL $\frac{3}{8}$ INCH DIA X 0.035 INCH THICK COIL. THE SHELL SIDE AND COIL SIDE CONNECTIONS ARE $1\frac{1}{2}$ INCH 2000 LB COUPLINGS FABRICATED OF TYPE 316 STAINLESS STEEL.

THE LISTED DESIGN CONDITIONS ARE 60 PSI PRESSURE AND 486°F TEMPERATURE FOR THE SHELL AND 600 PSI AND 486°F FOR THE COIL.

THE INVESTIGATION HAS BEEN MADE TO CHECK THE CONFORMANCE OF THE DRAIN COOLER WITH THE REQUIREMENTS OF THE ASME CODE FOR UNFIRED PRESSURE VESSELS SECTION VIII. THIS CONSISTS IN COMPARING THE REQUIRED MINIMUM THICKNESS AGAINST ACTUAL THICKNESS.

THE CODE REQUIREMENT FOR REINFORCING PADS AT VESSEL NOZZLE PENETRATIONS IS ALSO INVESTIGATED

W. O. NO. 2288-12

BURNS AND ROE, INC.
NEW YORK, N. Y.

DATE: 4-29-66

DRAWING NO. _____

BY: R. J. D.

TITLE: DRAIN COOLER

SHEET 3 OF 8

SUMMARY

THE RESULTS OF THE ANALYSIS INDICATE THAT THE ACTUAL THICKNESS OF THE DRAIN COOLER SHELL IS WELL ABOVE THAT REQUIRED BY CODE.

THE MEMBRANE PRESSURE STRESS IN THE COIL TUBE WALL WAS CALCULATED TO BE APPROXIMATELY 3500 PSI. THIS IS WELL BELOW THE CODE ALLOWABLE STRESS OF 15150 PSI.

THE REQUIREMENT FOR REINFORCING AROUND THE 1 1/2 INCH DIAMETER PENETRATIONS IS NOT APPLICABLE SINCE THE ASME CODE DOES NOT REQUIRE REINFORCING FOR PENETRATIONS 2 INCH AND SMALLER.

THE DRAIN COOLER IS, THEREFORE, ACCEPTABLE FOR THE SPECIFIED DESIGN CONDITIONS.

W. O. NO. 2288-12

DRAWING NO.

TITLE:

BURNS AND ROE, INC.
NEW YORK, N. Y.

DATE: 9-6-66

BY: M. GUDEN

SHEET 3 OF 8

INTRODUCTION

THE PURIFICATION DEMINERALIZER AND DECAY HEAT PIPING SYSTEM CONSISTS OF THREE LOOPS OF 1 1/2" SCH. 80 TYPE 316 STAINLESS STEEL PIPING.

LOOP NUMBER 1 CONNECTS THE DECAY HEAT PUMP INLET TO THE REACTOR COOLANT LOOP. THE OUTLET OF THE DECAY HEAT PUMP CONNECTS TO THE INLET OF THE PURIFICATION COOLER (LOOP NO. 2). LOOP NO. 3 CONNECTS THE COOLER OUTLET TO THE REACTOR COOLANT LOOP.

THE THREE LOOPS HAVE BEEN ANALYZED TO INSURE THAT THE PIPING IS NOT OVERSTRESSED DUE TO THERMAL EXPANSION OF THE SYSTEM.

W. O. NO. 2233-12

BURNS AND ROE, INC.
NEW YORK, N. Y.

DATE: 4-29-65

DRAWING NO. _____

BY: M. H. GIDEN

TITLE: DRAIN COOLER

SHEET 4 OF 8

DRAIN COOLER:

MATERIAL OF CONSTRUCTION:

SHELL & HEAD S/S SA 240 .216" THICK (Nom.)
 TYPE 304 .189" THICK (min.)

DESIGN CONDITIONS:

PRESSURE: 60 PSI.

TEMPERATURE: 486°F

REF #1 FOR ABOVE DATA

W. O. NO. 2288-12

BURNS AND ROE, INC.
NEW YORK, N. Y.

DATE: 4-29-66

DRAWING NO.

BY: M. H. GIDEN

TITLE: DRAIN COOLER

SHEET 5 OF 8

SHELL THICKNESS:

$$t = \frac{PR}{SE - 0.6P} \quad \text{PAR. UG 32 REF. 2}$$

$$P = 60 \text{ PSI,}$$

$$R = 1.75'$$

$$S = 15149 \text{ PSI (TABLE UHA-23)}$$

$$E = .45$$

$$t = \frac{60(1.75)}{(15149)(.45) - .6(60)} = .0153" < t_{min} = .189"$$

CONICAL HEAD:

$$t = \frac{PD}{2Cwd(SE - .6P)}$$

$$P = 60 \text{ PSI}$$

$$D = 3.122"$$

$$Cwd = .95 \text{ (ESTIMATED)}$$

$$S = 15149 \text{ PSI}$$

$$E = 100\%$$

$$t = \frac{60(3.122)}{1.9(15149 - 36)} = .0124" < t_{min} = .189"$$

W. O. NO. 2288-12BURNS AND ROE, INC.
NEW YORK, N. Y.DATE: 4-29-66

DRAWING NO. _____

BY: M. H. WIDENTITLE: DRAIN COOLERSHEET 6 OF 8COIL: TUBE PRESSURE = 600 PSI. $T = 486^{\circ}\text{F}$ CONSIDER THE COIL AS A TORUS

$$S = \frac{p b}{t} \left(\frac{2a - b}{2a - 2b} \right) \quad \text{PAGE 274 CASE 21 REF. 3}$$

$$a = 1.25''$$

$$b = .1875''$$

$$t = .035''$$

$$S = \frac{600(.1875)}{.035} \left[\frac{2.5 - .1875}{2.5 - .375} \right] = 3497 < S_{ALL} = 15,149 \text{ PSI.}$$

W. O. NO. 2288-12

BURNS AND ROE, INC.
NEW YORK, N. Y.

DATE: 4-29-66

DRAWING NO.

BY: M. H. GIDEN

TITLE: DRAIN COOLER

SHEET 7 OF 8

PENETRATION ANALYSIS:

ALL OPENINGS $1\frac{1}{2}$ " REINFORCING NOT REQUIRED
PER PAR. UG-34 REF. 2,

W. O. NO. 2288-12

BURNS AND ROE, INC.
NEW YORK, N. Y.

DATE: 4-29-66

DRAWING NO. _____

BY: R. J. D.

TITLE: DRAIN COOLER

SHEET 8 OF 8

REFERENCES

- 1- FORM U-1 MANUFACTURER'S DATA REPORT FOR UNFIRED PRESSURE VESSELS. WHITLOCK MANUF. CO. DEC 31 1961.
- 2- ASME BOILER & PRESSURE VESSEL CODE SECTION VIII - UNFIRED PRESSURE VESSELS 1965 EDITION
- 3- FORMULAS FOR STRESS AND STRAIN BY ROARK - 3rd EDITION - MCGRAW HILL PUB.

W. O. NO. 2233-12

BURNS AND ROE, INC.
NEW YORK, N. Y.

DATE: 12/66

DRAWING NO. _____

BY: M. GIDEN

TITLE: _____

SHEET 1 OF 6

PM-3A

MAIN CONDENSER AND CONDENSATE
SYSTEM EXHAUST PIPING

ITEM 9.0

W. O. NO. 2288-12

BURNS AND ROE, INC.
NEW YORK, N. Y.

DATE: 12/20

DRAWING NO. _____

BY: M. SIDEN

TITLE: _____ SHEET 2 OF 6

INTRODUCTION AND SUMMARY

AN INVESTIGATION WAS MADE OF THE TURBINE EXHAUST PIPING SYSTEM.

AN ANALYSIS TO DETERMINE THE STRESSES DUE TO THERMAL EXPANSION WAS NOT NECESSARY SINCE EXPANSION JOINTS WERE USED TO ABSORB THE THERMAL EXPANSION.

THE INFORMATION SHOWN ON SHEET 3 WAS CHECKED AND EVALUATED. BASED ON THIS REVIEW, THE PIPING SYSTEM IS ACCEPTABLE. THE EXPANSION JOINTS ADEQUATELY TAKE CARE OF THE THERMAL EXPANSION OF THE PIPING SYSTEM.

A REVIEW OF THE EXPANSION JOINTS THEMSELVES WAS NOT POSSIBLE SINCE COMPLETE DETAILS WAS NOT AVAILABLE.

TURBINE EXHAUST PIPING
PM-3A NUCLEAR POWER PLANT

SHEET 3 OF 6

CONDITIONS

PIPING — CARBON STEEL (SIZE SHOWN ON SKETCH)

DESIGN TEMP. — 360°F.

DESIGN PRESS. — 150 PSIG

OPERATING TEMP. — 240°F.

OPERATING PRESS. — 5" H₂O ABS. TO 12 PSIG.

AMBIENT TEMP. — 40°F.

EXPANSION RATE — 1.42"/100 FT.

TURBINE EXHAUST PIPING

Exp. 125/100,
Press. 2.46 Abs. 7.15 psia.
Or Temp. 540°F.
Pipe 15" & larger to be 25 lb. Wt.

STEAM STRESS ANALYSIS
PROBLEM C390-B, MAIN
FOR POINTS 57, 58, 59, 60, 61, 62

BRISQ ENGINEERS, INCORPORATED
CHARLOTTE, N.C.

DESIGN	DATE	REV	BY	CHK	APP
8	1-13-61	1	W. H. H.	W. H. H.	W. H. H.
1	1-13-61	1	W. H. H.	W. H. H.	W. H. H.

DUE TO THE ARRANGEMENT OF EXPANSION JOINTS AND PIPE SUPPORTS, THERE ARE NO STRESSES IN THE SYSTEM DUE TO THERMAL EXPANSION. THERE ARE SOME, HOWEVER, DUE TO THE "BLOWING OUT" (OR IN) OF THE EXPANSION JOINTS, THE FRICTIONAL RESISTANCE OF SLIDING SUPPORTS, AND THE RESISTANCE OF THE EXPANSION JOINTS.

BLOWING OUT DIAGRAM

MOMENT AT 'A'

$$M_A = 1560 \times 36 = 56,300 \text{ in-in}$$

STRESS ON 12" PIPE

$$S_A = \frac{56,300}{43.8} = \underline{\underline{1290 \text{ PSI}}}$$

MOMENT AT 'B'

$$M_B = 3120 \times 37 = 115,300 \text{ in-in}$$

$$S_B = \frac{115,300}{89.6} = \underline{\underline{1290 \text{ PSI}}}$$

THE 24" EXP. JOINT WILL NOT HAVE ANY BLOWING OUT EFFECT SINCE THE SOLID HANGER SHOWN ON DWG. B-6380-102 WILL TAKE THE FORCE.

TURBINE EXHAUST PIPING
PM-3A NUCLEAR POWER PLANT

SHEET 6 of 6

THE FRICTIONAL FORCE OF THE HANGERS SHOWN ON DAG.
B-6380-60 WILL CAUSE A BENDING MOMENT ON THE 18" PIPE,
BUT ONLY WHEN THE PIPE IS EXPANDING

EXAMPLE

COEFF. OF
FRICTION
 $F_c = .25 \times \text{LOAD ON HANGER}$

MULT $F \times 53"$

$$S_H = \frac{F \times 53"}{Z \text{ (SECT. MOD. OF 18" PIPE)}}$$

$$Z_H = 89.7 \text{ in}^3$$

THESE STRESSES WOULD OBVIOUSLY BE LESS IN MAGNITUDE
THAN $1/6$ THE LOAD ON THE RESPECTIVE HANGERS

W. O. NO. 2288-12

BURNS AND ROE, INC.
NEW YORK, N. Y.

DATE: 9/66.

DRAWING NO. _____

BY: J. CLAPP

TITLE: _____

SHEET 1 OF 5

STRESS ANALYSIS

PM-3A

HYDROGEN ADDITION VESSEL

ITEM 10.0

W.O. NO. 2288-12

BURNS AND ROE, INC.
NEW YORK, N. Y.

DATE: 9/66

DRAWING NO. _____

BY: J. CLAPP

TITLE: PM-3A HYDROGEN ADDITION VESSEL SHEET 2 OF 5

ITEM 10.1

SUMMARY

THE HYDROGEN ADDITION VESSEL WAS ANALYZED IN ACCORDANCE WITH SECTION VIII OF THE ASME BOILER AND PRESSURE VESSEL CODE.

REQUIRED MINIMUM THICKNESSES WERE DETERMINED FOR THE CYLINDRICAL SHELL AND FLAT HEADS FOR THE DESIGN CONDITIONS OF PRESSURE (1700 PSIG.) AND TEMPERATURE (135 °F.).

RESULTS OF ANALYSIS INDICATE THAT THE REQUIRED MINIMUM THICKNESSES OF THE SHELL AND HEADS ARE 0.381 IN. AND 1.445 IN., RESPECTIVELY.

SINCE THE ACTUAL MINIMUM THICKNESSES OF THE SHELL AND HEADS ARE 0.4375 IN. AND 1.500 IN., RESPECTIVELY, THE VESSEL IS ADEQUATELY SIZED FOR THE DESIGN CONDITIONS INVESTIGATED.

NOZZLE REINFORCEMENT IS WAIVED IN ACCORDANCE WITH PAR. UG-36, REF. 2.

W. O. NO. 2288-12

BURNS AND ROE, INC.
NEW YORK, N. Y.

DATE: 9/66

DRAWING NO. _____

BY: J. LAPE

TITLE: PM-3A HYDROGEN ADDITION VESSEL

SHEET 3 OF 5

INTRODUCTION

THE HYDROGEN ADDITION VESSEL SERVES TO CHARGE THE REACTOR COOLANT SYSTEM WITH A CONTROLLED HYDROGEN CONCENTRATION. THE VESSEL HAS AN 8-INCH NOMINAL SEAMLESS SHELL, 3'-5" LENGTH, AND IS CONSTRUCTED WITH FLAT HEAD ENDS. SHELL PENETRATIONS ARE FOR TWO $\frac{1}{2}$ -INCH CONNECTIONS.

THE MATERIAL FOR FABRICATION OF THIS VESSEL IS STAINLESS STEEL TYPE 304.

DESIGN CONDITIONS OF PRESSURE AND TEMPERATURE ARE 1700 PSIG. AND 135 °F, RESPECTIVELY.

THE ANALYSIS HAS BEEN MADE TO CHECK CONFORMANCE WITH SECTION VIII OF THE ASME BOILER AND PRESSURE VESSEL CODE. THE REQUIRED MINIMUM THICKNESSES FOR THE SHELL AND HEADS ARE DETERMINED, AND THEN COMPARED TO THE ACTUAL THICKNESSES USED.

W. O. NO. 2285-12

BURNS AND ROE, INC.

DATE: 9/66

DRAWING NO. _____

NEW YORK, N. Y.

BY: J. CLAPPTITLE: PM-3A HYDROGEN ADDITION VESSEL SHEET 4 OF 5CYLINDRICAL SHELL (PAR. UG-27, REF. 2)

THE MANUFACTURERS DATA REPORT INDICATES SEAMLESS STAINLESS STEEL TYPE 304 VESSEL WITH 0.500 IN. NOMINAL WALL THICKNESS. IT IS THEREFORE ASSUMED THAT AN 8-INCH NOMINAL PIPE SCHEDULE 80 WAS USED FOR FABRICATION. (THE 80% JOINT EFFICIENCY CALLED-FOR IN THE DATA REPORT IS ONLY APPLICABLE TO THE ROUND SEAMS OF THE VESSEL.)

∴ FOR 8-INCH NOM. PIPE (SCHEDULE 80):

$$O.D. = 8.625'' ; I.D. = 7.625'' ; t_s = 0.500''$$

$$t = \frac{PR}{SE - 0.6P}$$

$$= \frac{1700 \times 3.813}{18,000(1.0) - 0.6(1700)}$$

$$= \frac{6482}{16,980}$$

$$= 0.381 \text{ IN.} < t_s = 0.875(0.500) = 0.4375 \text{ IN.}$$

$$P = 1700 \text{ PSIG.}$$

$$R = 3.813 \text{ IN.}$$

$$S = 18,000 \text{ PSI.}$$

$$E = 1.0$$

SHELL TEMP. (135 °F.)

FLAT HEAD

(PAR. UG-34, REF. 2)

$$t = d \sqrt{\frac{CP}{S}}$$

$$= 7.625 \sqrt{\frac{0.381 \times 1700}{18,000}}$$

$$= 7.625(0.1895)$$

$$= 1.445 \text{ IN.} < t_h = 1.500 \text{ IN.}$$

$$P = 1700 \text{ PSIG.}$$

$$d = 7.625 \text{ IN.}$$

$$S = 18,000 \text{ PSI.}$$

$$C = 0.5 M$$

$$M = \frac{t_r}{t_s} = \frac{0.381}{0.500} = 0.762$$

$$C = 0.5(0.762) = 0.381$$

HEAD TEMP. (135 °F.)

W. O. NO. 2288-12

BURNS AND ROE, INC.
NEW YORK, N. Y.

DATE: 9/66

DRAWING NO. _____

BY: J. CLAPP

TITLE: PM-3A HYDROGE ADDITION VESSEL SHEET 5 OF 5

REFERENCES

1. FORM U-1 MANUFACTURERS' DATA REPORT
FOR UNFIRED PRESSURE VESSELS.
ELLICOTT BRANDT, INC.
2. ASME BOILER AND PRESSURE VESSEL CODE
SECTION VIII - UNFIRED PRESSURE VESSELS
1965 EDITION.

FORM U-1A MANUFACTURERS' DATA REPORT FOR UNFIRED PRESSURE VESSELS
Alternate Form for Single Chamber Completely Shop-Fabricated Vessels Only

1. Manufactured by ELLICOTT BRANDT INC. 700 Ridgely Street, Baltimore 30, Md.
(Name and address of Manufacturer)

Manufactured for MARTIN-MARIETTA CORP. MARYLAND
(Name and address of Purchaser)

3. Type Horiz. Vessel No. (8916) (Mfrs. Serial) (State & State No.) Natl. Bd. No. 25103 Yr. Built 1962
A. S. Murray Co.

4. SHELL: Matl. 304 S.S. T.S. 70,000 Thk. 500 In. Allow 500 In. Diam. 8 Ft. 5 In. Length 3 Ft. 5 In.
(Kind and Spec. No.) (Fig. or F.B. & Spec. Min. T.S.)

5. SEAMS: Long Seamless H.T. No X.R. No Sectioned No Efficiency 80 %
(Welded, Dbl., Single, Lap, Butt) (Yes or No)¹ (Spot or Complete) (Yes or No)

Girth Full Fillet H.T. No X.R. No Sectioned No No. of Courses 1
Steel Specialties

If riveted or brazed describe seams fully under remarks.

6. HEADS: (a) Material Steel Special 304 T.S. 70,000 (b) Material Steel Special 304 T.S. 70,000
Location (Top, bottom, ends) Thickness 1-1/2" Crown Radius 70,000 Knuckle Radius 70,000 Elliptical Ratio 70,000 Conical Apex Angle 70,000 Hemispherical Radius 70,000 Flat Diameter 70,000 Side to Pressure (Convex or Concave)
(a) End 1-1/2" 70,000 70,000 70,000 70,000 70,000 70,000 8" Flat
(b) End 1-1/2" 70,000 70,000 70,000 70,000 70,000 70,000 8" Flat

If removable, bolts used (Material, Spec. No., T.S., Size, Number) Other fastening (Describe or Attach Sketch)

7. Constructed for max. allowable working press. ²1700 psi at max. temp. 135 °F. Min. temp. (when less than -20°) 70,000 °F. Hydrostatic Pneumatic or Combination } Test Press 2550 psi.

8. SAFETY OR RELIEF VALVE OUTLETS: Number 70,000 Size 70,000 Location 70,000

9. NOZZLES:
Purpose (Inlet, Outlet, Drain) Number 2 Diam. or Size 1/2" Type Stainless Steel 304 Couplings Material 70,000 Thickness 70,000 Reinforcement Material 70,000 How Attached Welded

10. INSPECTION Manholes, No. 70,000 Size 70,000 Location 70,000
OPENINGS: Handholes, No. 70,000 Size 70,000 Location 70,000
Threaded, No. 70,000 Size 70,000 Location 70,000

11. SUPPORTS: Skirt (Yes or No) Lugs (Number) Legs (Number) Other (Describe) Attached (Where & How)

12. REMARKS: 1 Hydrogen Vessel

(Brief description of purpose of the vessel as Air Tank, Water Tank, L.P.G., Etc.—State Contents.
¹ If postweld heat-treated.
² List other internal or external pressures with coincident temperature when applicable.

We certify that the statements made in this report are correct and that all details of design, material, construction, and workmanship of this vessel conform to the ASME Code for Unfired Pressure Vessels.

Date December 7, 1962 Signed Ellicott Brandt, Inc. By E. Lockard
(Manufacturer)

Certificate of Authorization Expires 70,000

62EB 1639 CERTIFICATE OF SHOP INSPECTION

VESSEL MADE BY ELLICOTT BRANDT INC. at Baltimore 30 Md.

I, the undersigned, holding a valid commission issued by the National Board of Boiler and Pressure Vessel Inspectors and/or the State of Maryland and employed by Mutual Boiler & Machinery Ins. Co. of Waltham, Mass. have inspected the pressure vessel described in this manufacturer's data report on December 7, 1962, and state that to the best of my knowledge and belief, the manufacturer has constructed this pressure vessel in accordance with the applicable sections of the ASME Boiler and Pressure Vessel Code.

By signing this certificate neither the Inspector nor his employer makes any warranty, expressed or implied, concerning the pressure vessel described in this manufacturer's data report. Furthermore, neither the Inspector nor his employer shall be liable in any manner for any personal injury or property damage or a loss of any kind arising from or connected with this inspection.

Date December 7, 1962
s/ J. N. Kruhmin Commissions N.B. 3171
Inspector's Signature Nat'l Board or State and No.

W. O. NO. 2288-12

BURNS AND ROE, INC.
NEW YORK, N. Y.

DATE: 9-66

DRAWING NO. _____

BY: R. J. Dabbs

TITLE: _____ SHEET 1 OF 7

STRESS ANALYSIS

SHIELD WATER AIR BLAST COOLER

PM-3A

ITEM 11.0

W. O. NO. 2288-12BURNS AND ROE, INC.
NEW YORK, N. Y.DATE: 9-66

DRAWING NO. _____

BY: R. J. DOBBSTITLE: PM-3A STRESS ANALYSISSHEET 2 OF 7

INTRODUCTION

THE SHIELD WATER AIR BLAST COOLERS ARE "AIR TO WATER" FORCED FLOW HEAT EXCHANGERS. EACH UNIT CONSISTS OF TWO COOLING COILS, AN ELECTRIC DRIVEN FAN AND A SET OF MOTOR DRIVEN EXHAUST AND RECIRCULATION DAMPERS.

THE DESIGN CONDITIONS FOR THE COOLERS ARE PRESSURE OF 126 PSIG. AND TEMPERATURE OF 350°F. THE DESIGN PRESSURE CORRESPONDS TO THE PRIMARY CONTAINMENT TANK DESIGN PRESSURE.

THE COOLING COILS CONSIST OF $\frac{5}{8}$ " O.D. X 0.049" WALL TYPE 304 SS TUBES AND $1\frac{1}{2}$ " SCH 40S TYPE 304 SS HEADERS. THE HEADER INLET AND OUTLET CONNECTIONS ARE $1\frac{1}{2}$ " - 150 LB ASA STANDARD RAISED FACE FLANGES, FABRICATED OF TYPE 304 SS.

W. O. NO. 2288-12

BURNS AND ROE, INC.
NEW YORK, N. Y.

DATE: _____

DRAWING NO. _____

BY: R. J. DOBBS

TITLE: PM-3A STRESS ANALYSIS

SHEET 3 OF 7

SUMMARY

THE SHIELD WATER AIR BLAST COOLING COIL TUBES AND HEADERS WERE ANALYZED IN ACCORDANCE WITH THE ASA CODE FOR PRESSURE PIPING B31.1-1959 AMERICAN SOCIETY OF MECHANICAL ENGINEERS.

IN EACH CASE THE ACTUAL THICKNESS IS GREATER THAN THAT REQUIRED BY CODE AND THEREFORE ACCEPTABLE.

NO HEADER FLANGE CALCULATIONS WERE REQUIRED SINCE, THE $1\frac{1}{2}$ "-150^{LB} FLANGES HAVE AN ALLOWABLE PRESSURE RATING OF 195 psig AT 350°F WHICH IS GREATER THAN THE DESIGN PRESSURE OF 126 psig.

THE REINFORCEMENT CALCULATIONS NORMALLY REQUIRED AT THE HEADER BRANCH CONNECTIONS WERE ALSO NOT NECESSARY. THE HEADERS HAVE SUFFICIENT EXCESS METAL IN THEIR BASIC WALL THICKNESS TO COMPENSATE FOR ANY BRANCH CONNECTION OPENING. THIS IS OBVIOUS FROM INSPECTION BY COMPARING THE CODE REQUIRED MINIMUM THICKNESS OF 0.0083" WITH THE ACTUAL THICKNESS OF 0.145".

THE SHIELD WATER AIR-BLAST COOLERS ARE ACCEPTABLE FOR THE DESIGN CONDITIONS INVESTIGATED.

ANALYSIS
OF

STRESS ANALYSIS

SHIELD WATER AIR BLAST COOLER

DESIGN DATA:

DESIGN TEMPERATURE - - - - - 350 °F

DESIGN PRESSURE - - - - - 126 PSIG

(NOTE: THIS CORRESPONDS TO PRIMARY
CONTAINMENT TANK DESIGN PRESSURE)

TUBES:

MATERIAL - - - TYPE 304 STAINLESS STEEL

SIZE - - - 5/8" OD X .049 WALL

HEADERS:

MATERIAL - - - TYPE 304 STAINLESS STEEL

SIZE - - - 1 1/2" SCH 40 S PIPE

FLANGES - - - 1 1/2" - 150# ASA ST'D,
ASTM 182 S.S. - TYPE 304.

REFERENCE DWG:

TRANE DWG: A-2813-1188 "COIL ASSEMBLY

TRANE DWG A-2813-6292 "COIL ASSEMBLY

TRANE FLUID COOLER DATA SHEET NO 1473

PREPARED BY _____ CHECKED BY _____ REVISED BY _____

ANALYSIS
OF

TUBE ANALYSIS: (COOLER SECTION)

$$t_m = \frac{PD}{2S + 2YP} + C \quad \text{REF. (1) Pg 13.}$$

$$P = 126 \text{ PSI}$$

$$D = .625" \text{ (OUTSIDE DIA.)}$$

$$Y = .4 \text{ (AUSTENITIC STEEL BELOW 900°F)}$$

$$S = 14325 \text{ PSI [REF 1 TABLE 12 FOR SEAMLESS S.S. PIPE]}$$

$$C = 0$$

$$t_m = \frac{126 \times .625}{2(14325) + 2(.4)(126)}$$

$$= .0028"$$

$$t_{act} = .049"$$

HEADERS

$$t_m = \frac{PD}{2S + 2YP} + C$$

$$D = 1.90 \text{ (OD OF 1.5 PIPE)}$$

$$Y = .4$$

$$S = 14325 \text{ PSI}$$

$$C = 0$$

$$t_m = \frac{126 \times 1.90}{2(14325) + (2)(.4)(126)} + 0$$

$$t_m = .0083" \checkmark$$

PREPARED BY _____

CHECKED BY _____

REVISED BY _____

ANALYSIS
OF

HEADERS - CONTINUED

$t_{ACT} = .145"$

FLANGES: - (HEADER INLET & OUTLET)

1 1/2" - 150 LB ASA STD TYPE 304
1" RAISED FACE (CLASS A)
16

MAXIMUM SERVICE PRESSURE RATING AT
350°F IS 195 PSIG

REF: (2) PAGE 162

BRANCH CONNECTIONS: - (HEADERS)

IT IS OBVIOUS FROM THE PREVIOUS CALCULATIONS
THAT THERE IS MORE THAN SUFFICIENT EXCESS
WALL THICKNESS IN THE HEADERS TO COMPENSATE
FOR THE WEAKENING EFFECT OF BRANCH LINES.
THEREFORE BRANCH LINE REINFORCEMENT CALCULATIONS
ARE NOT NECESSARY.

PREPARED BY _____ CHECKED BY _____ REVISED BY _____

ANALYSIS
OF

REFERENCES:

- (1) ASA B 31.1-1955, "Code For Pressure Piping",
PUBLISHED BY ASME.
- (2) CATALOG "FABRICATED PIPING DATA", PITTSBURGH
PIPING AND EQUIPMENT COMPANY, 1956.

PREPARED BY _____

CHECKED BY _____

REVISED BY _____

W. O. NO. 2288-12

BURNS AND ROE, INC.
NEW YORK, N. Y.

DATE: 4-29-66

DRAWING NO. _____

BY: M. GIDEN

TITLE: _____ SHEET 1 OF 12

STRESS ANALYSIS

PM-3A

DECAY HEAT PUMP

ITEM 12.0

W. O. NO. 2288-12

BURNS AND ROE, INC.
NEW YORK, N. Y.

DATE: 4-29-66

DRAWING NO.

BY: M. H. GIDEN

TITLE: DECAY HEAT PUMP

SHEET 2 OF 12

INTRODUCTION

THE PM-3A DECAY HEAT PUMP CONSISTS OF A 304 STAINLESS STEEL HERMETICALLY SEALED CENTRIFUGAL PUMP. THE LISTED DESIGN CONDITIONS ARE 1500 PSI. AND 260°F TEMPERATURE

THE INVESTIGATION HAS BEEN MADE TO CHECK THE CONFORMANCE OF THE PUMP WITH THE REQUIREMENTS OF THE ASME CODE FOR UNFIRED PRESSURE VESSELS SECTION VIII. THIS CONSISTS IN DETERMINING THE THICKNESS OF THE VARIOUS STRUCTURAL COMPONENTS FOR THE DESIGN CONDITIONS, THESE CALCULATED VALUES ARE THEN COMPARED WITH THE ACTUAL THICKNESSES SUPPLIED.

W. O. NO. 2288-12

BURNS AND ROE, INC.
NEW YORK, N. Y.

DATE: 4-29-66

DRAWING NO.

BY: M. GIDEN

TITLE:

SHEET 3 OF 12

SUMMARY

THE FOLLOWING SUMMARIZES THE RESULTS OF THE ANALYSIS.

1. PUMP CASING AND LUG SATISFY THE REQUIREMENTS OF THE ASME CODE, SECTION VIII AND ARE ACCEPTABLE.
2. THE THICKNESS REQUIREMENT FOR THE PUMP COVER IN ACCORDANCE WITH ASME CODE SECTION VIII IS 1.65 INCHES. THIS IS GREATER THAN THE ACTUAL MINIMUM THICKNESS SUPPLIED 1.156 INCHES, AND THEREFORE UNACCEPTABLE. A SEPERATE ANALYSIS TREATING THE COVER AS A FLAT PLATE INDICATES A MAXIMUM STRESS OF 20,800 PSI. FOR THE DESIGN CONDITION. THIS MAXIMUM STRESS IS ALSO IN EXCESS OF WHAT IS ALLOWED BY CODE.
3. THE ANALYSIS ALSO INDICATES THAT FOR DESIGN CONDITIONS, THE ACTUAL BOLTING AREA IS LESS THAN THAT REQUIRED BY CODE AND UNACCEPTABLE.
4. FOR OPERATING CONDITIONS, 750 PSI, CALCULATIONS SHOW THAT BOTH THE BOLTS AND COVER MEET THE CODE REQUIREMENTS.

W.O. NO. 2288-12

BURNS AND ROE, INC.
NEW YORK, N. Y.

DATE: 4-29-66

DRAWING NO. _____

BY: M. H. GIDEN

TITLE: DECAY HEAT PUMP DHQ2-PPI SHEET 4 OF 12

DECAY HEAT PUMP
DWG. 387-2876059

W.O. NO. 2282-12

BURNS AND ROE, INC.
NEW YORK, N. Y.

DATE: 4-29-66

DRAWING NO.

BY: M. H. GIDEN

TITLE: DECAY HEAT PUMP

SHEET 5 OF 12

MATERIAL OF CONSTRUCTION

BOLTS - STEEL 10-7/8" - 9 N.C.

PUMP HOUSING & COVER 304 S.S

DESIGN CONDITIONS:

DESIGN PRESSURE = 1500 PSI

DESIGN TEMPERATURE = 260°F

W.O. NO. 2282-12

BURNS AND ROE, INC.
NEW YORK, N. Y.

DATE: 4-29-66

DRAWING NO.

BY: M. GIDEN

TITLE: DECAY HEAT PUMP

SHEET 6 OF 12

PRESSURE STRESS IN PUMP CHAMBER:

REF. CHEMPUMP DWG. D-10760

REQD THICKNESS OF SHELL FOR PUMP CHAMBER

$$t = \frac{PR}{SE - .6P} \quad (\text{REF. 1})$$

$$P = 1500 \text{ PSI.}$$

$$R = 2.4687$$

$$S = 15660 \text{ PSI}$$

$$t_{\text{REQD}} = \frac{1500(2.4687)}{15660 - .6(1500)} = .254" \quad t_{\text{ACT.}} = 2" \text{ PUMP CASING O.K.}$$

STRESS ON LUG:

ASSUME THAT THE TOTAL PRESSURE IN THE PUMP CHAMBER IS TAKEN BY THE LUG.

$$W = .7854 d^2 p$$

$$d_o = 4.9375"$$

$$p = 1500 \text{ PSI.}$$

$$d_i = 2.9375"$$

$$W = .7854 (4.9375)^2 1500 = 28,600 \#$$

$$\text{AREA OF LUG} = .7854 (d_o^2 - d_i^2) = .7854 (12.34 - 8.63) = 8.411 \text{ in}^2$$

$$\frac{W}{A} = \frac{28,600}{8.411} = 3,400 \#/\text{in}^2$$

W.O. NO. 2282-12

BURNS AND ROE, INC.
NEW YORK, N. Y.

DATE: 4-29-6-

DRAWING NO.

BY: M. GIDEN

TITLE: FRONT COVER DECAY HEAT PUMP

SHEET 8 OF 12

FRONT COVER

CHEMPUMP DWG. C-10010-2

COVER PLATE - CONTINUED:

d = DIAMETER OF 4 OF GASKET = 5.875"

C = FACTOR = 0.30 (FIG. UG-34-J)

S = MAXIMUM ALLOWABLE STRESS = 15,660 PSI (TABLE UHA-2)

P = DESIGN PRESSURE = 1500 PSI.

W = FLANGE DESIGN BOLT LOAD

h_g = GASKET MOMENT ARM = 0.875"

G = DIA. OF GASKET REACTION = 5.70"

$$t = d \sqrt{CP/S + 1.78Wh_g/Sd^2} \text{ PAR. UG-34} \quad (\text{REF. 1})$$

$$W = .785 G^2 P + (2b \times 3.14 G m P) \text{ PAR. UA-99 EQ. 1 (REF. 1)}$$

$$W = .785 (5.7)^2 1500 + (.80 \times 3.14 \times 5.70 \times 3 \times 1500)$$

$$W_{1500} = 38,250 + 64,430 = 102,680 \#$$

$$W_{750} = 51,340 \#$$

$$t = 5.875 \sqrt{\frac{.3(1500)}{15660} + \frac{1.78(102680).875}{15660(5.875)^2}} = 5.875 \sqrt{.0287 + .050}$$

$$t_{1500} = 5.875(.281) = 1.65" > t_{ACT.} = 1.156"$$

FOR 750 PSI.

$$t_{750} = 5.875 \sqrt{\frac{.3(750)}{15660} + \frac{1.78(51340).875}{15660(5.875)^2}} = 5.875 \sqrt{.01435 + .025}$$

$$t = 5.875(.1973) = 1.16" \text{ O.K. } t_{MIN.} = 1.16"$$

W. O. NO. 2288-12

BURNS AND ROE, INC.
NEW YORK, N. Y.

DATE: 4-29-66

DRAWING NO. _____

BY: M. GIDEN

TITLE: FRONT COVER DECAY HEAT PUMP

SHEET 10 OF 12

CONSIDER THE COVER AS A SIMPLY SUPPORTED PLATE

$$a = 3.875''$$

$$a^2 = 15$$

$$t = 1.156$$

$$t^2 = 1.336$$

CASE I PAGE 194 REF. 2

$$S_t = S_c = \frac{3W}{8\pi m t^2} (3m+1) \quad (\text{AT CENTER})$$

$$S_t = S_c = \frac{3 p a^2}{8 m t^2} (3m+1) = \frac{3 p (15) (10.99)}{8 (3.33) (1.336)} = 13.9 p$$

$$S_t = S_c = 13.9 (1500) = 20,850 \text{ PSI} > S_{all} = 15,600 \text{ PSI.}$$

W.O. NO. 2288-12

BURNS AND ROE, INC.

DATE: 9-29-66

DRAWING NO. _____

NEW YORK, N. Y.

BY: M. GIDENTITLE: COVER RETAINING BOLTSSHEET 11 OF 12REQD BOLT AREA:FOR CARBON STEEL BOLTS, $S = 20,000 \text{ PSI}$ at 260°F $W_H = 102,680 \text{ LBS}$ DESIGN (1500 PSI) PAGE 9 $W_H = 51,340 \text{ LBS}$ OPERATING (750 PSI)

$$A_H = \frac{W_H}{S} = \frac{102,680}{20,000} = 5.14 \text{ IN}^2 \quad (\text{REF. 1})$$

 $A_H \text{ AVAILABLE} = 4.62 \text{ IN}^2 \quad (10 - \frac{3}{4} - \text{S.N.C. BOLTS})$

BASED ON THE MAXIMUM ALLOWABLE STRESS FOR CARBON STEEL BOLTS (SECT. VIII ASME BOILER CODE). THE BOLTS ARE OVERSTRESSED FOR DESIGN CONDITIONS. THE BOLTS ARE FABRICATED OF HIGH STRENGTH CARBON STEEL WITH A YIELD STRENGTH OF $150,000 \text{ PSI}$, AND AN ULTIMATE STRENGTH OF $165,000 \text{ PSI}$. IF THE NORMAL BASIS WAS USED FOR ESTABLISHING ALLOWABLE STRESS, ($1/4$ ULTIMATE) THE BOLTS WOULD BE SATISFACTORY FOR DESIGN CONDITIONS.

BASED ON THE ABOVE CALCULATION, THE BOLTS ARE SATISFACTORY FOR OPERATING CONDITIONS.

W. O. NO. 2288-12

BURNS AND ROE, INC.
NEW YORK, N. Y.

DATE: 4-29-66

DRAWING NO. _____

BY: M. GIDEN

TITLE: DECAY HEAT PUMP

SHEET 12 OF 12

REFERENCES

1. ASME BOILER & PRESSURE VESSEL CODE SECTION III
UNFIRED PRESSURE VESSELS 1965 EDITION.
2. FORMULAS FOR STRESS & STRAIN BY ROARK 3RD EDITION
MCGRAW-HILL.

W. O. NO. 2288-12

BURNS AND ROE, INC.
NEW YORK, N. Y.

DATE: 9/66

DRAWING NO. _____

BY: J. CLAPP

TITLE: _____ SHEET I. OF 8

STRESS ANALYSIS

PM-3A

MAIN & AUXILIARY STEAM SYSTEM

ITEM 13.0

W. O. NO. 2288-12

BURNS AND ROE, INC.

DATE: 7/66

DRAWING NO. _____

NEW YORK, N. Y.

BY: J. CLAPPTITLE: PM-3A - MAIN & AUXILIARY STEAM SYSTEM SHEET II OF 8

INTRODUCTION & SUMMARY

THE PIPING FLEXIBILITY ANALYSIS PERFORMED FOR THE MAIN & AUXILIARY STEAM SYSTEM CONSIDERS THE 6-INCH SCHEDULE 40 LINE-RUN FROM THE STEAM GENERATOR TO THE TURBINE, WITH BRANCH CONNECTIONS TO THE TURBINE EXHAUST LINE AND TO THE FEEDWATER PUMP. THE SYSTEM CONTAINS AN INTERMEDIATE ANCHOR AND TWO RESTRAINTS. THE SYSTEM IS ANALYZED AS TWO INDEPENDENT PROBLEMS, DESIGNATED AS 6380 A & 6380 B, WITH THE INTERMEDIATE ANCHOR ACTING AS A COMMON TERMINAL POINT FOR EACH PROBLEM.

RESULTS OF ANALYSIS INDICATE A MAXIMUM THERMAL STRESS OF APPROXIMATELY 1,000 PSI @ POINT 1 OF PROBLEM 6380 A; AND A MAXIMUM THERMAL STRESS OF APPROXIMATELY 6,930 PSI @ POINT 31 OF PROBLEM 6380 B.

THE ALLOWABLE THERMAL STRESS FOR CARBON STEEL, CONSERVATIVELY ASSUMING THE USE OF A-106 A MATERIAL, IS 18,000 PSI.

FORCE AND MOMENT LOADING AT TERMINAL POINTS ARE LISTED ON SHEET 4 FOR PROBLEM 6380 A. FORCE AND MOMENT LOADING AT TERMINAL POINTS, AND LOADS AT RESTRAINTS ARE LISTED ON SHEET 5 FOR PROBLEM 6380 B. (IT SHOULD BE NOTED THAT FORCES, MOMENTS, AND RESTRAINT LOADS LISTED ARE BASED ON THE COLD MODULUS OF ELASTICITY, AND SINCE THERE IS NO INDICATED COLD-SPRING FOR THE SYSTEM, THESE VALUES SHOULD BE MULTIPLIED BY THE RATIO OF THE HOT TO COLD MODULI TO ATTAIN REDUCED VALUES FOR THE HOT, OR OPERATING, CONDITION. HOWEVER,

W. O. NO. 2289-12

BURNS AND ROE, INC.
NEW YORK, N. Y.

DATE: 9/66

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BY: J. CLAPP

TITLE: PM-3A - MAIN & AUXILIARY STEAM SYSTEM SHEET III OF 8

THIS CORRECTION IS ACADEMIC, SINCE THE VALUES OF FORCE, MOMENT, AND RESTRAINT LOAD ARE QUITE LOW.)

EACH OF THE TWO PROBLEMS PRESENTED BY THE MARTIN COMPANY WERE RE-RUN AND VERIFIED ON THE BURNS AND ROE IBM 1620 COMPUTER PROGRAM FOR PIPING FLEXIBILITY ANALYSIS.

$$\begin{array}{r} 0.05 \cdot TW \cdot 200.0 - 0.1 \cdot 250.0 = 0.0 \cdot 39.6 \\ \hline 0.05 \cdot TW \cdot 200.0 - 0.1 \cdot 250.0 = 0.0 \cdot 39.6 \\ \hline 0.05 \cdot TW \cdot 200.0 - 0.1 \cdot 250.0 = 0.0 \cdot 39.6 \\ \hline 0.05 \cdot TW \cdot 200.0 - 0.1 \cdot 250.0 = 0.0 \cdot 39.6 \\ \hline 0.05 \cdot TW \cdot 200.0 - 0.1 \cdot 250.0 = 0.0 \cdot 39.6 \end{array}$$

MAN STEAM PIPING

BRAC ENGINEERS, INCORPORATED
CHARLESTON, PA.

1-0802-A	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23	24	25	26	27	28	29	30	31	32	33	34	35	36	37	38	39	40	41	42	43	44	45	46	47	48	49	50	51	52	53	54	55	56	57	58	59	60	61	62	63	64	65	66	67	68	69	70	71	72	73	74	75	76	77	78	79	80	81	82	83	84	85	86	87	88	89	90	91	92	93	94	95	96	97	98	99	100
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CONDITIONS

PIPING — CARBON STEEL
SCHEDULE 40 (SIZE SHOWN ON SKETCHES)

DESIGN TEMPERATURE — 500° F

DESIGN PRESSURE — 585 PSIG

OPERATING TEMPERATURE — 417° F

OPERATING PRESSURE — 285 PSIG

AMBIENT TEMPERATURE — 40° F

EXPANSION RATE @ OPERATING TEMPERATURE • 9.07"/100'

CALCULATION BASED ON COLD MODULUS OF ELASTICITY
WITH NO ALLOWANCE FOR COLD SPRING.

MOMENTS IN FOOT POUNDS

FORCES IN POUNDS

$$\text{MAX. STRESS} = \sqrt{S_{\text{BENDING}}^2 + 4 S_{\text{TORSIONAL}}^2}$$

TOTAL MOMENTS & FORCES

LOCATION	M _x	M _y	M _z	F _x	F _y	F _z
A-1	396.20	242.67	-274.45	19.77	-45.38	-11.80
A-2	114.94	91.23	538.41	-19.77	45.38	11.80

NOTE:

ALL REACTIONS ARE THE ACTION OF THE PIPING ON THE ANCHOR POINTS

TABLE OF STRESSES & DEFLECTIONS

POINT	STRESS	Δ_x	Δ_y	Δ_z
0	762.19	-0.625	0.750	0.000
1	996.48	-0.779	0.759	0.007
2	786.80	-0.756	0.720	0.175
3	628.67	-0.583	1.048	0.288
4	448.08	-0.471	0.919	0.562
5	425.93	-0.103	0.577	0.408
6	371.44	-0.260	0.394	0.350
7	929.60	0.049	0.168	0.266
8	836.52	-0.030	0.004	0.255
9	702.63	-0.041	0.001	0.000
10	788.23	0.000	0.000	0.000

STRESS MAX = 996.48 $\frac{1}{w}$ @ POINT 1

MAIN STEAM PIPING
PM3A NUCLEAR POWER PLANT

SHEET 5 OF 6
PROBLEM 6380-B

TOTAL MOMENTS & FORCES

LOCATION	M _x	M _y	M _z	F _x	F _y	F _z
A1	325.91	-638.02	74.85	-46.53	-10.86	33.79
A2	-770.97	-1475.37	-541.48	54.21	32.05	-295.33
R1	—	—	—	-175.11	4.43	—
R2	—	—	—	306.55	-150.81	—
A3	1748.94	-2247.36	-289.08	240.70	198.90	890.62
A4	-2635.68	4066.65	749.83	-379.82	-73.71	-629.08

NOTE:

ALL REACTIONS ARE THE ACTION OF THE PIPING ON THE ANCHORS OR RESTRAINTS.

TABLE OF STRESSES & DEFLECTIONS

LOCATION	STRESS	Δx	Δy	Δz
0	1017.46	0.000	0.000	0.000
1	669.83	0.321	0.013	0.059
2	569.82	0.353	0.002	0.126
3	568.68	0.309	0.029	0.185
4	633.20	0.405	0.476	0.439
5	633.74	0.381	0.440	0.504
6	439.92	0.383	0.349	0.457
7	1335.08	1.150	0.224	0.471
8	1357.62	1.166	0.289	0.489
9	946.60	-0.087	0.502	-0.562
10	441.64	0.271	0.291	-0.872
11	219.43	0.432	0.168	-0.639
12	276.35	0.199	0.304	-0.478
13	321.52	0.167	0.250	-0.513
13A	1058.96	—	—	—
14	491.78	-0.016	0.389	-0.746
15	1984.61	-0.187	0.456	-0.605
16	1098.08	-0.225	0.245	0.024
17	1366.91	-0.127	0.223	0.076
18	1544.03	-0.151	0.067	0.114
19	2472.56	-0.062	0.062	0.125
13B	1337.14	—	—	—
20	2586.09	-0.000	-0.000	-0.662
21	2146.65	-0.000	0.000	-0.466
22	2275.21	0.264	0.165	-0.631
23	2097.39	0.213	0.179	-0.611
23A	4869.46	—	—	—
24	2296.10	0.033	0.025	-0.347
25	4042.97	-0.000	0.000	-0.281
23B	4349.70	—	—	—
26	2560.04	0.224	0.219	-0.569
27	3972.54	0.143	0.128	-0.626
28	3157.05	0.150	0.072	-0.510
29	4743.29	0.009	0.034	-0.058
30	6061.47	0.021	0.001	-0.007
31	6926.39	-0.000	-0.000	-0.000

MAX STRESS: 6926.39 #/in² @ POINT 31

W. O. NO. 2288-12

BURNS AND ROE, INC.
NEW YORK, N. Y.

DATE: 9/66

DRAWING NO. _____

BY: R. J. DOBBS

TITLE: _____ SHEET 1 OF 13

STRESS ANALYSIS

FEEDWATER DEAERATOR

PM-3A

ITEM 14.0

W. O. NO. 2288-12

DRAWING NO. _____

TITLE: _____

BURNS AND ROE, INC.
NEW YORK, N. Y.

DATE: 9/66

BY: R. J. DOBBS

SHEET 2 OF 13

INTRODUCTION

THE FEEDWATER DEAERATOR IS A CYLINDRICAL VESSEL 4^{FT} INSIDE DIAMETER WITH AN OVERALL LENGTH OF 8^{FT}, INCLUDING TWO ASME TORISPHERICAL HEADS. BOTH THE HEADS AND SHELL ARE 5/16 INCH THICK ASTM A-285 GR C CARBON STEEL WITH A 20% TYPE 304 STAINLESS STEEL CLADDING ON THE INSIDE SURFACE.

THE PRIMARY DEAERATOR NOZZLES ARE; 10 INCH DIA INLET, 4 INCH DIA OUTLET, 4 INCH DIA OVERFLOW OUTLET, 6 INCH DIA STEAM INLET AND A 12X18 INCH ELLIPTICAL MANHOLE.

THE DESIGN BASIS USED FOR THE DEAERATOR IS THE ASME BOILER AND PRESSURE VESSEL CODE SECTION VIII UNFIRED PRESSURE VESSELS.

THE SPECIFIED DESIGN CONDITIONS ARE PRESSURE OF 30 PSIG AND TEMPERATURE OF 450 °F.

SUMMARY

THE FEEDWATER DEAERATOR WAS ANALYZED IN ACCORDANCE WITH ASME CODE SECTION VIII.

THE ACTUAL THICKNESS OF THE SHELL AND HEADS IS GREATER THAN THE MINIMUM REQUIRED BY CODE AND THEREFORE ACCEPTABLE

A REVIEW OF THE REINFORCEMENT CALCULATIONS AT THE VARIOUS NOZZLE PENETRATIONS PRESENTED IN REFERENCE #1 (AND INCLUDED IN THIS REPORT) INDICATES SOME ERRORS AND INCONSISTENCIES. CORRECTION OF THE ERRORS DOES NOT AFFECT THE FINAL CONCLUSION, THAT THE PENETRATIONS ARE ADEQUATELY REINFORCED IN ACCORDANCE WITH THE REQUIREMENTS OF THE CODE. THE FOLLOWING ITEMS INDICATE THE NATURE OF THE ERRORS AND INCONSISTENCIES;

1. THE ORIGINAL CALCULATION FOR THE MINIMUM WALL THICKNESS REQUIRED FOR THE 10 INCH DIA NOZZLE USED AN INCORRECT RADIUS. A WELD EFFICIENCY OF 70% WAS ALSO USED IN THE CALCULATION. THE EFFICIENCY IS NOT APPLICABLE SINCE THE NOZZLE NECK IS FABRICATED OF SEAMLESS PIPE WITHOUT ANY LONGITUDINAL WELDS. THIS CALCULATION WAS RE-DONE BY BURNS AND ROE AND IS INCLUDED AS PAGE 8 .
2. THE ANALYSIS AT THE 12 X 18 INCH MANHOLE NOZZLE USED A CORROSION ALLOWANCE FOR THE SHELL. THIS IS NOT APPLICABLE SINCE THE SHELL IS CLAD WITH STAINLESS STEEL. (SEE P. 9)

W. O. NO. 2288-12

BURNS AND ROE, INC.
NEW YORK, N. Y.

DATE: _____

DRAWING NO. _____

BY: R. J. DOBBS

TITLE: PM-3A STRESS ANALYSIS

SHEET 4 OF 13

SUMMARY (CONT)

3. THE ANALYSIS AT THE 6 INCH DIA STEAM INLET NOZZLE USED A CORROSION ALLOWANCE FOR THE SHELL, AND A WELD EFFICIENCY OF 85% IN THE CALCULATION FOR THE REQUIRED THICKNESS OF THE SEAMLESS PIPE NOZZLE NECK. NEITHER OF THESE IS APPLICABLE FOR REASONS STATED IN ITEM 1 AND 2 ABOVE.

THE FEEDWATER DEAERATOR IS, HOWEVER, ACCEPTABLE FOR THE DESIGN CONDITIONS INVESTIGATED.

ANALYSIS
OF
FEEDWATER DEAERATOR

ASME SECTION VIII ANALYSIS

GENERAL DATA

MATERIALS OF CONSTRUCTION

HEADS & SHELL : $\frac{5}{16}"$ THK. CARBON STEEL (ASTM-A285, GR C)
WITH 20% STAINLESS STEEL CLAD (304 SS) ON
INSIDE SURFACE.

DESIGN CONDITIONS

DESIGN PRESSURE : 30 psig
DESIGN TEMPERATURE : 450 °F

PREPARED BY duGAS 7/1/65 CHECKED BY _____ REVISED BY _____

ANALYSIS
OF
FEEDWATER DEAERATOR

SHELL ANALYSIS

REQ'D THICKNESS:

$$t = \frac{PR}{SE - 0.6P} \quad (\text{PARA. UG-27})$$

WHERE

$$P = 30 \text{ psig}$$

$$R = 24 \text{ in.}$$

$$S = 13,750 \text{ psi} \quad (\text{TABLE UCS-23})$$

$$E = 70\% \quad (\text{TABLE UW-12})$$

$$t_{\text{REQ'D}} = \frac{30 \times 24}{13,750(.70) - 0.6(30)}$$

$$= \underline{0.075 \text{ in.}}$$

$$t_{\text{ACT}} = \underline{0.3125 \text{ in.}}$$

> O.K.

NO CORROSION ALLOWANCE REQUIRED SINCE
THE SHELL IS LINED WITH 20% 304 S.S. CLAD

ANALYSIS
OF
FEEDWATER DEAERATOR

HEAD ANALYSIS

REQ'D THICKNESS :

$$t = \frac{0.885 PL}{SE - 0.1P} \quad (\text{PARA UG-32})$$

WHERE

$$P = 30 \text{ psig}$$

$$L = 48 \text{ in}$$

$$S = 13,750 \text{ psi} \quad (\text{TABLE UCS-23})$$

$$E = 70\% \quad (\text{TABLE UW-12})$$

$$t_{\text{REQ'D}} = \frac{0.885 \times 30 \times 48}{13,750(.70) - 0.1(30)}$$

$$= \underline{0.133 \text{ in.}}$$

$$t_{\text{ACT}} = \underline{0.3125 \text{ in.}} > \text{O.K.}$$

NO CORROSION ALLOWANCE REQ'D SINCE THE HEADS
ARE CLAD WITH 20% STAINLESS STEEL (304 S.S.)

PREPARED BY duGas
7/1/65

CHECKED BY _____

REVISED BY _____

PENETRATION ANALYSIS10" ϕ FEEDWATER INLET NOZZLEREQD NOZZLE THICKNESS (t_{rn})

$$\begin{aligned}
 t_{rn} &= \frac{pr}{SE - 0.6p} \\
 &= \frac{30(5.01)}{13750 - 0.6(30)} \\
 &= 0.011"
 \end{aligned}$$

$$d = 10.02 \text{ in.} \quad t_n = 0.365 \text{ in.} \quad t = 0.3125 \text{ in.}$$

$$b = 10.02/2 - 0.365 = 4.645 \text{ in.}$$

$$a = 2.5(0.3125) = 0.781 \text{ in.} \quad t_r = 0.133 \text{ in. pg. 7}$$

$$A_{req'd} = d(t_r) = 10.02(0.133) = 1.33 \text{ in}^2$$

$$\begin{aligned}
 A_{avail} &= \left[(0.3125 - 0.133)10.02 + 2(0.781)(0.365 - 0.011) \right] \\
 &= 1.80 + 0.55 = 2.35 \text{ in}^2 \quad (\text{OK})
 \end{aligned}$$

ANALYSIS
OF
FEEDWATER DEAERATOR

12"x18" ELLIPTICAL MANHOLE NOZZLE

MAT'L: ASTM A285, Gr C

CORROSION ALLOWANCE

$$C = \frac{1}{6}(t_r)$$

$$d = 16.12 \text{ in.}$$

$$t_N = .75 \text{ in.}$$

$$t_r = 0.133 \text{ p.7}$$

$$t = 0.3125 \text{ in.}$$

$$a = 2.5 \left(0.3125 - \frac{0.133}{6} \right)$$

$$= 0.725 \text{ in.}$$

$$A_{REQ'D} = d t_r = 16.12 (.133) = \underline{2.14 \text{ sq. in.}}$$

$$A_{AVAIL} = \left[16.12 \left(.3125 - .133 - \frac{.133}{6} \right) + .75 \left(.725 \right)^2 \right] = \underline{3.62 \text{ sq. in.}}$$

OK.

NO CREDIT TAKEN FOR WELDS OR PART OF MANHOLE RING
INSIDE VESSEL. NO ALLOWANCE FOR CORROSION OF RING
OUTSIDE VESSEL, IN CALCULATIONS, BUT SUFFICIENT EXCESS
MATERIAL AVAILABLE.

PREPARED BY du Gas 7/2/65 CHECKED BY _____ REVISED BY _____

ANALYSIS
OF
FEEDWATER DEAERATOR

6" STEAM INLET PIPING

VERTICAL OFFSET PT. ① = 16.0 IN.

" " PT. ② = 16.0 + 3.312 = 19.312 IN.

" " PT. ③ = 16.0 - 3.312 = 12.688 IN.

ANGULAR OFFSET PT. ② = $\sin^{-1} \frac{19.312}{24} = 53.8^\circ$

" " PT. ③ = $\sin^{-1} \frac{12.688}{24} = 32.0^\circ$

$\Theta = 53.8 - 32.0 = 21.8^\circ$

$\frac{a}{r} = 0.378 \Rightarrow a (\text{CHORD}) = 0.378(24) = 9.07 \text{ IN.}$

7/2/65

ANALYSIS
OF
FEEDWATER DEAERATOR

6" STEAM INLET PIPING (CONT'D)

$$a = 9.07 \text{ in (pg. 10)} \quad t = 0.312 \text{ in.} \quad t_R = 0.075 \text{ in. (pg. 6)}$$

$$d = 9.07 - 2(.280) = 8.51 \text{ in.} \quad b = 2.5(.280) = 0.70 \text{ in.}$$

$$t_{RN} = \frac{Pr}{SE - .6P} = \frac{30 \times 3.032}{15,000(.85) - .6(30)} = 0.0071 \text{ in.}$$

$$A_{REQD} = 8.51 \left(0.075 + \frac{.075}{6} \right) = \underline{\underline{0.745 \text{ sq. in.}}}$$

$$A_{AVAIL} = 8.51 \left(.312 - .075 - \frac{.075}{6} \right) + 4 \left[.70 \left(.280 - .0071 - \frac{.0071}{6} \right) \right] = \underline{\underline{2.662 \text{ sq. in.}}} > 0.$$

PREPARED BY duGas
7/6/65

CHECKED BY _____

REVISED BY _____

ANALYSIS
OF
FEEDWATER DEAERATOR

4" ϕ FEEDWATER OUTLET (TYPICAL FOR OVERFLOW OUTLET)

MAT'L: ASTM-A285 Gr.C

$$\left. \begin{array}{l} t = 0.312'' \\ t_r = 0.075'' \end{array} \right\} \text{REF. PG. 6}$$

$$A_{REQ'D} = 9 \times 0.075 = 0.675 \text{ sq. in.}$$

$$A_{AVAIL} = (9 - 4) 1.625 = 8.12 \text{ sq. in.}$$

> O.K.

PREPARED BY duGAS 7/7/65 CHECKED BY _____ REVISED BY _____

W. O. NO. 2288-12

BURNS AND ROE, INC.
NEW YORK, N. Y.

DATE: 7/66

DRAWING NO. _____

BY: R J DOBBS

TITLE: PM-3A STRESS ANALYSIS

SHEET 13 OF 13

REFERENCES

- 1 - REPORT OF SUB-TASK 9.2 CONTRACT AT(30-1)-3345
MARTIN CO. NUCLEAR PROGRAM, BALTIMORE MARYLAND.
- 2- ASME BOILER AND PRESSURE VESSEL CODE
SECTION VIII UNFIRED PRESSURE VESSELS

FORM U-1A MANUFACTURERS' DATA REPORT FOR UNFIRED PRESSURE VESSELS
Alternate Form for Single Chamber Completely Shop-Fabricated Vessels Only

1. Manufactured by MOHAWK MANUFACTURING CORP., WHITEHOUSE STATION, N.J.
(Name and address of Manufacturer)

Manufactured for The Permutit Company New York 36, N. Y.
(Name and address of Purchaser)

3. Type Horiz. Vessel No. (F-961) (Mfrs. Serial) (State & State No.) Natl. Bd. No. 1208 Yr. Built 1961
(Horiz. or Vert.)

4. SHELL: Matl. SA-285C fl T.S. 55,000 Nom. 3125 Corr. Thk. In. Allow In. Diam. 4 Ft. 0 In. Length 8 Ft. 0 In.
(Kind and Spec. No.) (Fig. or F.B. & Spec. Min. T.S.)

5. SEAMS: Long DBFW H.T. No X.R. No Sectioned No Efficiency 70 %
(Welded, Dbl., Single, Lap, Butt) (Yes or No)¹ (Spot or Complete) (Yes or No)
Girth DBFW H.T. No X.R. No Sectioned No No. of Courses 1

If riveted or brazed describe seams fully under remarks.

6. HEADS: (a) Material SA-285-C fig. T.S. 55,000 (b) Material SA-285-C fig. T.S. 55,000
Location (Top, bottom, ends) Thickness Crown Radius Knuckle Radius Elliptical Ratio Conical Apex Angle Hemispherical Radius Flat Diameter Side to Pressure (Convex or Concave)
(a) ends .3125 48" ASME concave

(b)
If removable, bolts used (Material, Spec. No., T.S., Size, Number) Other fastening (Describe or Attach Sketch)

7. Constructed for max. allowable working press. ² 30 psi at max. temp. 450 °F. Min. temp. (when less than -20°) °F. Hydrostatic Pneumatic or Combination } Test Press 45 psi.

8. SAFETY OR RELIEF VALVE OUTLETS: Number 1 Size 3/4" Location Top of Shell

9. NOZZLES:
Purpose (Inlet, Outlet, Drain) Number Diam. or Size Type Material Thickness Reinforcement Material How Attached
Inlet 1 6" flange steel steel welded
Outlet 1 10" " " " "

10. INSPECTION Manholes, No. 1 Size 12 X 16 Location Head
OPENINGS: Handholes, No. (2) 4" stud Size Conns. (1) Location 2-1/2" flg. - various
Threaded, No. (5) 3/4" Size (6) 1" Location various

11. SUPPORTS: Skirt (Yes or No) Lugs (Number) Legs (Number) Other (Describe) Attached (Where & How)

12. REMARKS: Type "B" Deaerator as per Permutit Company drawing E-10143.

(Brief description of purpose of the vessel as Air Tank, Water Tank, L.P.G., Etc.—State Contents.
³ If postweld heat-treated.
⁴ List other internal or external pressures with coincident temperature when applicable.

We certify that the statements made in this report are correct and that all details of design, material, construction, and workmanship of this vessel conform to the ASME Code for Unfired Pressure Vessels.

Date Feb. 8 19 61 Signed Mohawk Manufacturing Corp. By s/R. Meacham
(Manufacturer)

Certificate of Authorization Expires December 31, 1961 3773

CERTIFICATE OF SHOP INSPECTION

VESSEL MADE BY Mohawk Manufacturing Corp. at Whitehouse Station, J. J.

I, the undersigned, holding a valid commission issued by the National Board of Boiler and Pressure Vessel Inspectors and/or the State of N. J. and employed by The State of

 have inspected the pressure vessel described in this manufacturer's data report on Feb 8, 1961 and state that to the best of my knowledge and belief, the manufacturer has constructed this pressure vessel in accordance with the applicable sections of the ASME Boiler and Pressure Vessel Code.

By signing this certificate neither the Inspector nor his employer makes any warranty, expressed or implied, concerning the pressure vessel described in this manufacturer's data report. Furthermore, neither the Inspector nor his employer shall be liable in any manner for any personal injury or property damage or a loss of any kind arising from or connected with this inspection.

Date Feb. 8 19 61
A. G. Hauell Jr. Commissions Natl. Bd. 1757
Inspector's Signature Nat'l Board or State and No.

W. O. NO. 2288-12

BURNS AND ROE, INC.
NEW YORK, N. Y.

DATE: 9/66

DRAWING NO. _____

BY: R. J. DOBBS

TITLE: _____ SHEET 1 OF 8

STRESS ANALYSIS

FEEDWATER HEATER

PM-3A

ITEM-15.0

W. O. NO. 2288-12

BURNS AND ROE, INC.
NEW YORK, N. Y.

DATE: 9/66

DRAWING NO.

BY: R. J. DOBBS

TITLE: PM-3A STRESS ANALYSIS

SHEET 2 OF 8

INTRODUCTION & SUMMARY

THE FEEDWATER HEATER IS A HORIZONTAL U-TUBE HEAT EXCHANGER. IT IS APPROXIMATELY 18" DIAMETER AND 52" LENGTH.

THE HEATER WAS ANALYZED IN ACCORDANCE WITH ASME BOILER AND PRESSURE VESSEL CODE SECTION VIII UNFIRED PRESSURE VESSELS.

THE SPECIFIED DESIGN CONDITIONS ARE:

SHELL SIDE - 195 PSIG PRESSURE AT 400°F

TUBE SIDE - 600 PSIG PRESSURE AT 450°F

THE ACTUAL THICKNESS OF THE VARIOUS COMPONENTS IN ALL CASES IS GREATER THAN THE CALCULATED MINIMUM THICKNESS REQUIRED BY CODE AND THEREFORE ACCEPTABLE.

REINFORCEMENT REQUIREMENTS SHOULD BE CHECKED AT THE 3" DIA CHANNEL NOZZLES AND THE 4" DIA SHELL NOZZLE. HOWEVER, MORE SPECIFIC DETAILS ARE REQUIRED BEFORE AN ACCEPTABLE CONCLUSION CAN BE REACHED. FORM U-1 (PAGE) LISTS THE NOZZLES AS 3"-3000" ASA-R.F. AND 4"-3000" ASA-R.F. IT APPEARS THAT A TYPING ERROR WAS MADE IN LISTING THE NOZZLES AS 3000". FURTHER INVESTIGATION IN THIS AREA IS REQUIRED.

W.O. NO. 2288-12BURNS AND ROE, INC.
NEW YORK, N. Y.DATE: 9/66

DRAWING NO. _____

BY: M. GIDENTITLE: PM-3A STRESS ANALYSISSHEET 5 OF 8FEEDWATER HEATER TUBES

MATERIAL SB-III

 $\frac{1}{2}$ " BWG 18 $t = 0.049$ " $P = 600$ PSI $T = 450^{\circ}\text{F}$ $S = 8300$ PSI (TABLE UNF-23 REF. 2)MINIMUM THICKNESS REQD. $t = \frac{PR}{SE - 0.6P}$ (PAR. UG-27 REF. 2)

$$t = \frac{600(0.201)}{8300 - .6(600)} = \frac{120.6}{7940} = 0.0152" < t_{\text{ACT}} = 0.0492"$$

ϕ of δ

As required by the Provisions of the ASME Code Rules

1. Manufactured by YUBA HEAT TRANSFER DIVISION, HONESDALE, PA.
(Name and address of Manufacturer)

2. Manufactured for The Martin Company - Nuclear Division - Project PM-3A
(Name and address of Purchaser)

Type Horiz Kind Heat Exch Vessel No. H-1081971 (----) Natl. Bd. No. 949 Yr. Buil 1961
(Horiz. or Vert.) (Tank, Jacketed, Heat Exch.) (Mrs. Serial) (State & State No.)

Items 4-9 incl. to be completed for single wall vessels (such as air tanks), jackets of jacketed vessels, or shells of heat exchangers.

4. SHELL: Material SA-106-B T.S. _____ Nominal _____ Corrosion 1/16 _____
(Kind and Spec. No.) (Fig. or F.B. & Spec. Min. T.S.) Thickness 3/8 In. Allowance _____ In. Diam. _____ Ft. _____ In. Length _____ Ft. _____ In.

5. SEAMS: Long Seamless H.T. No X.R. Spot Sectioned No Efficiency 100 %
(Welded, Dbl., Single, Lap, Butt) (Yes or No)¹ (Spot or Complete) (Yes or No)

If riveted describe seams fully on reverse side of form.

6: HEADS (a) Material SA-285-C T.S. 18"-1-1/8" (b) Material SA-285-C T.S. 18"-1-1/8"

Location (Top, bottom, ends)	Thickness	Crown Radius	Knuckle Radius	Elliptical Ratio	Conical Apex Angle	Hemispherical Radius	Flat Diameter	Side to Pressure (Convex or Concave)
(a) End	3/8	18	1-1/8					Concave

(b) [REDACTED]

If removable, bolts used _____ Other fastening _____
(Material, Spec. No., T.S., Size, Number) (Describe or Attach Sketch)

7. STAYBOLTS: None If hollow Attachment Pitch X Diam.
(Material) (Size of Hole) (Threaded, Welded) (Horiz.) (Vert.) (Nominal)

8. JACKET CLOSURE: None
(Describe as ogee & weld, bar, etc. If bar, give dimensions, if bolted, describe or sketch)

9. Constructed for max. allowable working press ² 195 psi at max. temp. 400 °F. Min. temp. (when less than -20°) _____ °F.

hydrostatic Pneumatic or Combination	}	Test Press	<u>293</u> psi.
--	---	---------------	--------------------

Items 10 and 11 to be completed for tube sections.

10. TUBE SHEETS: Stationary: Material SA-212 Gr. B F.B Diam. 18 In. Thickness 1-3/4 In. Attachment Bolted
(Kind & Spec. No.) (Subject to Pressure) (Welded, Bolted)

Floating. Material None - "U" Tubes Diam. _____ In. Thickness _____ In. Attachment _____
(Kind & Spec. No.)

11. TUBES: Material Adm-SB-111 1 1/2" In. Thickness 17&18 BWG Inches or Gage Number 156 Type "U"
(Kind & Spec. No.) (Straight or U)

ems 12-15 incl. to be completed for inner chambers of jacketed vessels, or channels of heat exchangers.

Channel 1
12. ~~STEEL~~ Material SA-106-B T.S. Nominal Thickness 3/4 Corrosion Allowance 1/16 In. Diam. 1 Ft. 6 In. Length 8 Ft. 8 In.
(Kind and Spec. No.) (Fig. or F.B. & Spec. Min. T.S.)

13. SEAMS: Long Seamless H.T. Yes X.R. --- Sectioned No Efficiency 100 %
(Welded, Dbl., Single, Lap, Butt) (Yes or No) (Spot or Complete) (Yes or No)

If riveted describe seams fully on reverse side of form.

14. **Cover** Girth D.B. Welded **H.T.** Yes **X.R.** Spot **Sectioned** No **No. of courses** One **form.**
HEADS (a) Material SA-212-B **T.S.** _____ **(b) Material** _____ **T.S.** _____ **(c) Material** _____ **T.S.** _____

Location	Thickness	Crown Radius	Knuckle Radius	Elliptical Ratio	Conical Apex Angle	Hemispherical Radius	Flat Diameter	Side to Pressure (Convex or Concave)
----------	-----------	--------------	----------------	------------------	--------------------	----------------------	---------------	--------------------------------------

(b) Channel covers 2-1/2" Flat 18"

(c) Floating

If removable, bolts used (a) _____ (Material, Spec. No., T.S., Size, Number) (b) SA-193 Gr. 7

(c) _____ Other fastening _____
(Describe or Attach Sketch)

15. Constructed for max.. 600 psi at max. temp. 450 °F. Min. temp. (when less than -20°) _____ °F. Pneumatic or Test Press 900 psi. Combination

Items below to be completed for all vessels where applicable.

16. SAFETY VALVE OUTLETS: Number Two (2) Size One @ 3/4" Location Shell-Channel

17. NOZZLES

Purpose (Inlet, Outlet, Drain)	Number	Diam. or Size	Type	Material	Thickness	Reinforcement Material	How Attached
Chan-Inlet-Outlet	2	2"-3000#	ASA-R.F.	SA-106-B	Sch. 80	None	Welded
Shell - In	1	4"-3000#	ASA-R.F.	SA-106-B	Sch. 80	None	Welded
Shell-Out	1	2"-N.P.T.		SA-106-B	Sch. 80	None	Welded

¹ If postweld heat-treated.

² List under remarks other internal or external pressures with coincident temperature when applicable.

FORM U-1 (back)

18. INSPECTION Manholes, No. None Size _____ Location _____
 OPENINGS: Handholes, No. None Size _____ Location _____
 Threaded, No. _____ Size _____ Location _____
 19. SUPPORTS: Skirt No Lugs No Legs TWO Other _____ Attached Shell-Weld
 (Yes or No) (Number) (Number) (Describe) (Where & How)

20. REMARKS: _____

(Brief description of purpose of the vessel, as Air Tank, After Cooler, Jacketed Cooker, etc. State contents of each part.)

We certify that the statements made in this report are correct and that all details of design, material, construction, and workmanship of this vessel conform to the ASME Code for Unfired Pressure Vessels.

Date Feb 10 19 61 Signed YUBA HEAT TRANSFER DIVISION
 (Manufacturer)

Certificate of Authorization Expires 12-31-61

CERTIFICATE OF SHOP INSPECTION

VESSEL MADE BY YUBA HEAT TRANSFER DIVISION HONESDALE, PA.

I, the undersigned, holding a valid commission issued by the National Board of Boiler and Pressure Vessel Inspectors and/or the State of Pa. & Nat'l Bd. and employed by The Hartford Steam Boiler Ins. & Ins. Co.
Hartford, Conn.

_____ have inspected the pressure vessel described in this manufacturer's data report on Feb 10 1961, and state that to the best of my knowledge and belief, the manufacturer has constructed this pressure vessel in accordance with the applicable sections of the ASME Boiler and Pressure Vessel Code.

By signing this certificate neither the Inspector nor his employer makes any warranty, expressed or implied, concerning the pressure vessel described in this manufacturer's data report. Furthermore, neither the Inspector nor his employer shall be liable in any manner for any personal injury or property damage or a loss of any kind arising from or connected with this inspection.

Date Feb 10 19 61

 Inspector's Signature

Commissions Penna; 1833 Nat'l Bd 4659
 Nat'l Board or State and No.

CERTIFICATE OF FIELD ASSEMBLY INSPECTION

I, the undersigned, holding a valid commission issued by the National Board of Boiler and Pressure Vessel Inspectors and/or the State of _____ and employed by _____ of _____

_____ have compared the statements in this manufacturer's data report with the described pressure vessel and state that parts referred to as data items _____, not included in the certificate of shop inspection have been inspected by me and that to the best of my knowledge and belief the manufacturer has constructed and assembled this pressure vessel in accordance with the applicable sections of the ASME Boiler and Pressure Vessel Code. The described vessel was inspected and subjected to a hydrostatic test of _____ psi.

By signing this certificate neither the Inspector nor his employer makes any warranty, expressed or implied, concerning the pressure vessel described in this manufacturer's data report. Furthermore, neither the Inspector nor his employer shall be liable in any manner for any personal injury or property damage or a loss of any kind arising from or connected with this inspection.

Date _____ 19 _____

 Inspector's Signature

Commissions _____
 Nat'l Board or State and No.

W. O. NO. 2288-12

BURNS AND ROE, INC.
NEW YORK, N. Y.

DATE: 9/66

DRAWING NO. _____

BY: M. GIDEN

TITLE: PM-3A STRESS ANALYSIS

SHEET 8 OF 8

REFERENCES

1. FORM U-1 MANUFACTURED DATA FOR UNFIRED PRESSURE VESSELS. YUBA Co.
2. ASME BOILER AND PRESSURE VESSEL CODE SECTION VIII, UNFIRED PRESSURE VESSELS, 1965 ED.

W. O. NO. 2288-12

BURNS AND ROE, INC.
NEW YORK, N. Y.

DATE: 9/66

DRAWING NO. _____

BY: R. J. DOBBS

TITLE: _____ SHEET 1 OF 9

STRESS ANALYSIS

EVAPORATOR

PM-3A

ITEM 16.0

W. O. NO. 2288-12

BURNS AND ROE, INC.
NEW YORK, N. Y.

DATE: 9/66

DRAWING NO.

BY: R. J. DOBBS

TITLE: PM-3A STRESS ANALYSIS

SHEET 2 OF 9

INTRODUCTION & SUMMARY

THE EVAPORATOR IS A HORIZONTAL U-TUBE STEAM TO WATER HEAT EXCHANGER. IT IS APPROXIMATELY 26" DIAMETER AND 72" LENGTH.

THE EVAPORATOR WAS ANALYZED IN ACCORDANCE WITH ASME BOILER AND PRESSURE VESSEL CODE SECTION VIII UNFIRED PRESSURE VESSELS.

THE SPECIFIED DESIGN CONDITIONS ARE:

SHELL SIDE - 50 PSIG AND FULL VACUUM AT 450°F
TUBE SIDE - 150 PSIG AT 440°F.

THE ACTUAL THICKNESS OF THE VARIOUS COMPONENTS IN ALL CASES IS GREATER THAN THE CALCULATED MINIMUM THICKNESS REQUIRED BY CODE.

REINFORCEMENT REQUIREMENTS ARE WAIVED IN ACCORDANCE WITH REF. #1 PARA. UG 36-C-3a WITH THE EXCEPTION OF THE 6" DIA. HANDHOLE IN THE RETURN END PLATE. ADDITIONAL DETAILS OF THE ACTUAL CONSTRUCTION AT THE HANDHOLE NOZZLE CONNECTION ARE REQUIRED BEFORE AN ACCEPTABLE CONCLUSION CAN BE REACHED.

IT SHOULD BE NOTED THAT AN INCONSISTENCY EXISTS IN THE DATA SHEETS FOR THE MATERIALS OF CONSTRUCTION OF THE END PLATE AT RETURN END AND END PLATE AT TUBE SHEET END. THE EVAPORATOR SPECIFICATION SHEET (PAGE 8) AND THE CALCULATION SHEET (PAGE 3) ARE CONSISTENT AND LIST THE MATERIAL OF THE PLATE AT RETURN END AS ASTM-A-285 GRC AND THE PLATE AT TUBE SHEET END AS ASTM-A-212 GR-B. THE FORM U-1 (PAGE 6) HOWEVER, LISTS ASTM-A-212-GR-A MATERIAL FOR BOTH PLATES. THIS ITEM SHOULD BE CHECKED, POSSIBLY AGAINST AN AS-BUILT DRAWING, TO DETERMINE WHICH MATERIAL IS CORRECT, SINCE IT COULD AFFECT THE CALCULATED RESULTS.

EVAPORATOR

YUBA CO. JOB NO. V10820

ITEM NO. 1

SIZE UNIT 10-27-36

FLANGE NO. 1:

FLANGE NO. 3:

FLANGE NO. 5:

FLANGE NO. 2:

FLANGE NO. 4:

NO. OF PASSES:

SHELL ~~SR~~ FR ~~FULL~~ SPOT

MAT'L P.S. (COPPER BEARING)

SPEC. SA-285

GR. C

INTERNAL PRESSURE

$$t = \frac{PR}{SE - 0.6P} + CORR. = \frac{50 \times 13.063}{13750 \times .85 - 30} + .063 = .119$$

EXTERNAL PRESSURE

$$PAR. UG-28 \quad \frac{L}{D_o} = \frac{1.9}{1} ; \frac{D_o}{i} = \frac{85}{1}$$

$$B = 10,000 ; P_o = \frac{B}{D_o/i} = \frac{10,000}{85} = 118 \text{ PSI}$$

SKIRT OR SHELL COVER BARREL ~~SR~~ FR ~~FULL~~ SPOT

MAT'L

SPEC.

$$t = \frac{PR}{SE - 0.6P} + CORR. = \frac{50 \times 13.063}{13750 \times .85 - 30} + .063 = .119$$

EXTERNAL PRESSURE

$$PAR. UG-28 \quad \frac{L}{D_o} = \frac{1.9}{1} ; \frac{D_o}{i} = \frac{85}{1}$$

$$B = 10,000 ; P_o = \frac{B}{D_o/i} = \frac{10,000}{85} = 118 \text{ PSI}$$

DISHED HEAD (2:1 ELLIP) ~~SR~~ FR ~~FULL~~ SPOT

MAT'L R.S. (CU-BEAR'G)

SPEC. SA-285

GR. C

END PLATE @ RETURN END

$$t = \frac{PR}{SE - 0.6P} + CORR. = \frac{50 \times 13.063}{13750 \times .85 - 30} + .063 = .119$$

EXTERNAL PRESSURE

$$PAR. UG-28 \quad \frac{L}{D_o} = \frac{1.9}{1} ; \frac{D_o}{i} = \frac{85}{1}$$

$$B = 10,000 ; P_o = \frac{B}{D_o/i} = \frac{10,000}{85} = 118 \text{ PSI}$$

DISHED HEAD (TORISPHERICAL) ~~SR~~ FR ~~FULL~~ SPOT

MAT'L R.S. (CU-BEAR'G)

SPEC. SA-212

GR. B

END PLATE @ TUBE SHEET END

$$t = \frac{PR}{SE - 0.6P} + CORR. = \frac{50 \times 13.063}{13750 \times .85 - 30} + .063 = .119$$

EXTERNAL PRESSURE

$$PAR. UG-28 \quad \frac{L}{D_o} = \frac{1.9}{1} ; \frac{D_o}{i} = \frac{85}{1}$$

$$B = 10,000 ; P_o = \frac{B}{D_o/i} = \frac{10,000}{85} = 118 \text{ PSI}$$

CHANNEL BARREL ~~SR~~ FR ~~FULL~~ SPOT

MAT'L S.M.L. STL.

SPEC. SA-106

GR. B

END PLATE @ TUBE SHEET END

$$t = \frac{PR}{SE - 0.6P} + CORR. = \frac{50 \times 13.063}{13750 \times .85 - 30} + .063 = .119$$

EXTERNAL PRESSURE

$$PAR. UG-28 \quad \frac{L}{D_o} = \frac{1.9}{1} ; \frac{D_o}{i} = \frac{85}{1}$$

$$B = 10,000 ; P_o = \frac{B}{D_o/i} = \frac{10,000}{85} = 118 \text{ PSI}$$

CHANNEL COVER ~~SR~~ FR ~~FULL~~ SPOT

MAT'L R.S.

SPEC. SA-285

GR. C

END PLATE @ TUBE SHEET END

$$t = \frac{PR}{SE - 0.6P} + CORR. = \frac{50 \times 13.063}{13750 \times .85 - 30} + .063 = .119$$

EXTERNAL PRESSURE

$$PAR. UG-28 \quad \frac{L}{D_o} = \frac{1.9}{1} ; \frac{D_o}{i} = \frac{85}{1}$$

$$B = 10,000 ; P_o = \frac{B}{D_o/i} = \frac{10,000}{85} = 118 \text{ PSI}$$

NOTE: FOR STANDARD CHANNEL COVERS, SEE E702.30

THE YUBA COMPANY

HONESDALE, PA.

PRESSURE VESSEL COMPUTATIONS

ASME CODE 1159

EVAPORATOR.

TUBE SHEETS MAT'L. _____ R.S. (COPPER BEARING) _____ SPEC. SA-285 C

STATIONARY

$t = F_t d \sqrt{\frac{0.25 P}{S}} + CORR = 1.25 \times 13.125 \sqrt{\frac{0.25 \times 150}{13750}} + .125 = .982$

USE: $1\frac{3}{4}" \phi \times 1" T.K.$

FLOATING HEAD (USE SAME "Y" AS STATIONARY TUBE SHEET)

USE:

NOTE: FOR STANDARD TUBE SHEETS, SEE E103.3

$F_t = 1.25$
 $d = 13.125$
 $P = 150$
 $S = 13750 @ 450^\circ F$

TUBES MAT'L. _____ SPEC. _____

INTERNAL PRESSURE (FOR U-TUBES, SEE E102.14 AND E102.15)

$t = \frac{PR}{SE - 0.6P} = \frac{PR}{SE - 0.6P}$

USE:

EXTERNAL PRESSURE

1. API-ASME CODE-PAR. EW-300
 2. ASME CODE-PAR. UQ-31

$P =$ _____
 $R =$ _____
 $S =$ _____
 $E =$ _____

FLOATING HEAD COVER

HOGGED OUT HEAD MAT'L. _____ SPEC. _____

$T = t + G =$

$t = D_1 \sqrt{\frac{0.25 P}{S}} + CORR = \sqrt{\frac{0.25}{S}}$

$G = \frac{Z \times G_1}{W_2} + 1/4" =$

NOTE: FOR STANDARD HOGGED OUT HEADS, SEE E 501.9

$D_1 =$ _____
 $P =$ _____
 $S =$ _____
 $A =$ _____
 $G_1 =$ _____
 $W_2 =$ _____

JOB V 10820 TCM

FORM NO. 2-5713

REV.	DATE	MADE	CHKD.	APPR.	REV.	ISSUE
1	10/27/16	Slone			1	PRINT 11-25-60

THE YUBA COMPANY
 HONESDALE, PA.
 PRESSURE VESSEL COMPUTATIONS
 1959 ASME CODE

W.O. NO. 2288-12

BURNS AND ROE, INC.
NEW YORK, N. Y.

DATE: 9/66

DRAWING NO.

BY: M. GIDEN

TITLE: PM-3A STRESS ANALYSIS

SHEET 5 OF 9

EVAPORATOR TUBES

MATERIAL SB-III

1" BWG 18

O.D. = 1.00"

t = 0.049"

P = 150 PSI.

T = 440°F

S = 8,300 PSI (TABLE UNF-23 REF. #2)

MINIMUM THICKNESS REQ'D $t = \frac{PR}{SE - 0.6P}$ (PAR. UG-27 REF. 2)

$$t = \frac{150(0.451)}{8300 - 0.6(150)} = 0.00825" < t_{ACT.} = 0.049"$$

FORM U-1 MANUFACTURERS' DATA REPORT FOR UNFIRED PRESSURE VESSELS

As required by the Provisions of the ASME Code Rules

6 of 9

1. Manufactured by YUBA HEAT TRANSFER DIVISION, HONESDALE, PA.
(Name and address of Manufacturer)2. Manufactured for The Martin Co. - Plant 1 OE-1703 Middle River, Maryland
(Name and address of Purchaser)Type Horiz. Kind Evaporator Vessel No. V-10820 (#1) Natl. Bd. No. 948 Yr. Built 1961
(Horiz. or Vert.) (Tank, Jacketed, Heat Exch.) (Mfrs. Serial) (State & State No.)

Items 4-9 incl. to be completed for single wall vessels (such as air tanks), jackets of jacketed vessels, or shells of heat exchangers.

4. SHELL: Material SA-285-C T.S. 55,000 Nominal Thickness 3/8 Corrosion Allowance 1/16 In. Diam. 2 Ft. 2-3/4 In. Length 5 Ft. 3-1/4 In.
(Kind and Spec. No.) (Fig. or F.B. & Spec. Min. T.S.)5. SEAMS: Long D.B. Welded H.T. No X.R. Spot Sectioned No Efficiency 85 %
(Welded, Dbl., Single, Lap, Butt) (Yes or No) (Spot or Complete) (Yes or No)

If riveted describe seams fully on reverse side of form.

Girth Fillet H.T. No X.R. No Sectioned No No. of Courses 1
6. HEADS (a) Material SA-212-A T.S. 65,000 (b) Material SA-212-A T.S. 65,000
(Top, bottom, ends) Location Thickness Crown Radius Knuckle Radius Elliptical Ratio Conical Apex Angle Hemispherical Radius Flat Diameter Side to Pressure (Convex or Concave)(a) End Pl. 1-7/8" 26-3/4"(b) " " Return 1-1/4" 26-3/4"If removable, bolts used (Material, Spec. No., T.S., Size, Number) Other fastening (a) (b) Welded to shell ends.
(Describe or Attach Sketch)7. STAYBOLTS: (Material) If hollow Attachment Pitch (Horiz.) X (Vert.) Diam. (Nominal)
(Size of Hole) (Threaded, Welded)

8. JACKET CLOSURE: (Describe as ogee & weld, bar, etc. If bar, give dimensions, if bolted, describe or sketch)

9. Constructed for max. allowable working press 30/Vac. psi at max. temp. 450 °F. Min. temp. (when less than -20°) °F. Hydrostatic Pneumatic or Test Press 75 psi.
Combination

Items 10 and 11 to be completed for tube sections.

10. TUBE SHEETS: Stationary. Material SA-285-C Copper Diam. 10 In. Thickness 1 In. Attachment Bolted
(Kind & Spec. No.) (Subject to Pressure) (Welded, Bolted)

Floating. Material (Kind & Spec. No.) Diam. In. Thickness In. Attachment

Cu. N1. 90:1011. TUBES: Material SB-111 O.D. 1 In. Thickness 18 BWG or Gage Number 10 Type "U"
(Kind & Spec. No.) (Straight or U)

Items 12-15 incl. to be completed for inner chambers of jacketed vessels, or channels of heat exchangers.

12. ~~SEAM~~ Material SA-106-B T.S. 60,000 Nominal Thickness 365 In. Allowance 1/16 In. Diam. 0 Ft. 10-3/4 In. Length 0 Ft. 7-1/2 In.
(Kind and Spec. No.) (Fig. or F.B. & Spec. Min. T.S.)13. SEAMS: Long Seamless H.T. Yes X.R. No Sectioned No Efficiency 100 %
(Welded, Dbl., Single, Lap, Butt) (Yes or No) (Spot or Complete) (Yes or No)

If riveted describe seams fully on reverse side of form.

Girth Full Fillet H.T. Yes X.R. No Sectioned No No. of courses 1
14. HEADS (a) Material SA-285-CT.S. T.S. 55,000 (b) Material SA-285-C T.S. 55,000 (c) Material SA-285-C T.S. 55,000

(a) Top, Flange Location Thickness Crown Radius Knuckle Radius Elliptical Ratio Conical Apex Angle Hemispherical Radius Flat Diameter Side to Pressure (Convex or Concave)

(b) Channel Cover 7/8" 14-3/4" 0. D.(c) Floating 10"If removable, bolts used (a) A-193B7 125,000 16-3/4 (b) Welded in channel barrel.
(Material, Spec. No., T.S., Size, Number)Flange full fillet welded to chan. barrel

(c) Other fastening (Describe or Attach Sketch)

15. Constructed for max. allowable working press 2150 psi at max. temp. 440 °F. Min. temp. (when less than -20°) °F. Hydrostatic Pneumatic or Test Press 225 psi.
Combination

Items below to be completed for all vessels where applicable.

16. SAFETY VALVE OUTLETS: Number 1 Size 2" Pipe & W.N. Flange Location in shell

17. NOZZLES

Purpose (Inlet, Outlet, Drain)	Number	Diam. or Size	Type	Material	Thickness	Reinforcement Material	How Attached
Inlet	1	1-1/2"-6000#	P.T. Cplg.	SA-105-2			Welded
Outlet	1	2"-6000#	P.T. Cplg.	SA-105-2			Welded
Drain	1	2"-6000#	P.T. Cplg.	SA-105-2			Welded

¹ If postweld heat-treated.² List under remarks other internal or external pressures with coincident temperature when applicable.

FORM U-1 (back)

18. INSPECTION Manholes, No. _____ Size _____ Location _____
 OPENINGS: Handholes, No. 1 Size 6" Dia. Location In return end plate
 Threaded, No. _____ Size _____ Location _____
 19. SUPPORTS: Skirt _____ Lugs (2) Welded Supports in shell Attached _____
 (Yes or No) (Number) (Number) (Describe) (Where & How)
 20. REMARKS: 10-27-36 "U" Tube Make U Evaporator built in accordance with
ASME Code 1959 - - Steam in tubes, water and vapor in shell.

(Brief description of purpose of the vessel, as Air Tank, After Cooler, Jacketed Cooker, etc. State contents of each part.)

We certify that the statements made in this report are correct and that all details of design, material, construction, and workmanship of this vessel conform to the ASME Code for Unfired Pressure Vessels.

Date Feb. 10, 1961 19 _____ Signed YUBA HEAT TRANSFER DIVISION By _____
 (Manufacturer)

Certificate of Authorization Expires 12-31-61

CERTIFICATE OF SHOP INSPECTION -

VESSEL MADE BY YUBA HEAT TRANSFER DIVISION HONESDALE, PA.

I, the undersigned, holding a valid commission issued by the National Board of Boiler and Pressure Vessel Inspectors and/or the State of Pa. & Nat'l. Bd. and employed by The Hartford Steam Boiler & Ins. Co. of Hartford, Conn. have inspected the pressure vessel described in this manufacturer's data report on Feb. 10, 1961 19 _____, and state that to the best of my knowledge and belief, the manufacturer has constructed this pressure vessel in accordance with the applicable sections of the ASME Boiler and Pressure Vessel Code.

By signing this certificate neither the Inspector nor his employer makes any warranty, expressed or implied, concerning the pressure vessel described in this manufacturer's data report. Furthermore, neither the Inspector nor his employer shall be liable in any manner for any personal injury or property damage or a loss of any kind arising from or connected with this inspection.

Date Feb. 10, 1961 19 _____

Inspectors Signature _____

Commissions Penna. 1833 Nat'l. Bd. 4659
 Nat'l Board or State and No.

CERTIFICATE OF FIELD ASSEMBLY INSPECTION

I, the undersigned, holding a valid commission issued by the National Board of Boiler and Pressure Vessel Inspectors and/or the State of _____ and employed by _____ of _____

_____ have compared the statements in this manufacturer's data report with the described pressure vessel and state that parts referred to as data items _____, not included in the certificate of shop inspection have been inspected by me and that to the best of my knowledge and belief the manufacturer has constructed and assembled this pressure vessel in accordance with the applicable sections of the ASME Boiler and Pressure Vessel Code. The described vessel was inspected and subjected to a hydrostatic test of _____ psi.

By signing this certificate neither the Inspector nor his employer makes any warranty, expressed or implied, concerning the pressure vessel described in this manufacturer's data report. Furthermore, neither the Inspector nor his employer shall be liable in any manner for any personal injury or property damage or a loss of any kind arising from or connected with this inspection.

Date _____ 19 _____

Inspector's Signature _____

Commissions _____
 Nat'l Board or State and No.

W.O. NO. 2288-12

BURNS AND ROE, INC.
NEW YORK, N. Y.

DATE: 9/66

DRAWING NO. _____

BY: M. GIDEN

TITLE: PM-3A STRESS ANALYSIS

SHEET 9 OF 9

REFERENCES

1. ASME BOILER AND PRESSURE VESSEL CODE
SECTION VIII UNFIRED PRESSURE VESSELS 1965 ED.

W. O. NO. 2288-12

BURNS AND ROE, INC.
NEW YORK, N. Y.

DATE: 9-2-66

DRAWING NO. _____

BY: M. GIDEN

TITLE: _____ SHEET 1 OF 6

STRESS ANALYSIS

PM-3A

INSTRUMENT AIR DRYER VESSEL

ITEM 17.0.

W. O. NO. 2288-12

BURNS AND ROE, INC.
NEW YORK, N. Y.

DATE: 9-2-66

DRAWING NO. _____

BY: M. GIDEN

TITLE: _____ SHEET 2 OF 6

SUMMARY

THE AIR DRYER VESSELS WERE ANALYZED IN ACCORDANCE WITH SECTION VIII OF THE ASME BOILER AND PRESSURE VESSEL CODE.

REQUIRED MINIMUM THICKNESSES WERE DETERMINED FOR THE CYLINDRICAL SHELL AND FLAT HEADS FOR THE DESIGN CONDITIONS (PREFILTER 250 PSI, 650°F) (TOWER 150 PSI, 450°F).

RESULTS OF THE ANALYSIS INDICATE THAT THE REQUIRED MINIMUM THICKNESS OF THE TOWER SHELL (SA-106-B) AND HEADS (SA-212-B) ARE 0.116" AND 0.586" RESPECTIVELY. THE THICKNESS OF THE PREFILTER SHELL (SA-106-B) AND HEADS (SA-285-C) ARE 0.115" AND 0.539" RESPECTIVELY.

SINCE THE ACTUAL MINIMUM THICKNESS OF THE SHELL AND HEAD FOR BOTH VESSELS ARE GREATER THAN THE REQUIRED MINIMUM THICKNESS, THE VESSELS ARE ADEQUATELY SIZED FOR THE DESIGN CONDITIONS INVESTIGATED.

NOZZLE REINFORCEMENT IS WAIVED IN ACCORDANCE WITH PAR. UG-36, REF. 2.

W. O. NO. 2288-12

BURNS AND ROE, INC.
NEW YORK, N. Y.

DATE: 9-2-66

DRAWING NO. _____

BY: M. GIDEN

TITLE: _____ SHEET 3 OF 6

INTRODUCTION

THE INSTRUMENT AIR DRYER CONSISTS OF A
PREFILTER VESSEL AND A DRYER TOWER.

THE PREFILTER VESSEL CONSISTS OF A 5" STANDARD PIPE
5'-2" LONG, WITH $\frac{5}{8}$ " THICK FLAT HEAD AT EACH END,

THE DRYER TOWER CONSISTS OF A 8" SCH. 30 PIPE
6'-6 $\frac{1}{2}$ " LONG WITH A 1" THICK FLAT TOP HEAD AND A
 $\frac{3}{4}$ " THICK BOTTOM HEAD.

THE VESSELS HAVE BEEN ANALYZED TO INSURE THAT
THEY CONFORM TO THE REQUIREMENTS OF THE
ASME CODE FOR THE DESIGN CONDITIONS.

W. O. NO. 2288-12

BURNS AND ROE, INC.
NEW YORK, N. Y.

DATE: 9-2-66

DRAWING NO.

BY: M. GIDEN

TITLE: PM-3A AIR DRYER PREFILTER

SHEET 4 OF 6

DESIGN CONDITIONS:

$$P = 250 \text{ PSI.}$$

$$T = 450^\circ \text{F}$$

MATERIAL:

SHELL SA-106-B (T.S. = 60,000 PSI)

HEAD SA 285-C (T.S. = 55,000 PSI)

$$\text{EFFICIENCY} = 100\%$$

1/16" CORR. ALLOWANCE

CYLINDRICAL SHELL: (PAR. UG-27 REF. 2)

$$t = \frac{PR}{SE - CP} + \text{CORR.} = \frac{250(2.5)}{15000 - .6(250)} + 0.0625 = .042 + 0.0625 = 0.1145"$$

$$t_{ACT.} = 0.226" > 0.1145"$$

FLAT HEAD: (PAR. UG-34

$$t = d \sqrt{\frac{CP}{S}} + \text{CORR.}$$

ASSUME $C = 0.50$ (FIG. UG-34 REF. 2)

SINCE DETAILS OF VESSEL

CONSTRUCTION ARE NOT AVAILABLE

$$t = 5 \sqrt{\frac{0.5(250)}{13750}} + 0.0625 = 0.476" + 0.0625" = 0.5385"$$

$$t_{ACT.} = 0.625" > 0.5385"$$

W. O. NO. 2382-12

BURNS AND ROE, INC.
NEW YORK, N. Y.

DATE: 9-2-66

DRAWING NO. _____

BY: M. J. DEAN

TITLE: PM-3A AIR DRYER TOWER

SHEET 5 OF 6

DESIGN CONDITIONS

$$P = 150 \text{ PSI.}$$

$$T = 450^\circ \text{F}$$

MATERIAL

SHELL: SA-106-B (T.S. = 60,000 PSI.)

HEAD: SA-212-B (T.S. = 70,000 PSI.)

EFFICIENCY = 100%

1/16" CORR. ALLOWANCE

CYLINDRICAL SHELL: (PAR. UG-27, REF. 2)

$$t = \frac{PR}{SE - 0.1} + \text{CORR.} = \frac{150(4)}{15000 - 0.1(150)} + 0.0625" = 0.043 + 0.0625 = 0.1055"$$

$$t_{\text{ACT.}} = 0.242" > 0.106"$$

FLAT HEAD:

$$t = d \sqrt{\frac{CP}{S}} + \text{CORR.} \quad \text{ASSUME } C = 0.5 \text{ (FIG. UG-24 REF. 1)}$$

SINCE DETAILS OF VESSEL CONSTRUCTION ARE NOT AVAILABLE.

$$d = 8"$$

$$S = 17,500 \text{ PSI.}$$

$$P = 150 \text{ PSI.}$$

$$t = 8 \sqrt{\frac{0.5(150)}{17500}} + 0.0625" = 0.523 + 0.0625 = 0.5855"$$

$$t_{\text{ACT.}} = 0.75" > 0.586"$$

W. O. NO. 2288-12

BURNS AND ROE, INC.
NEW YORK, N. Y.

DATE: 9-2-66

DRAWING NO. _____

BY: M. GIDEN

TITLE: _____ SHEET 6 OF 6

REFERENCES

1. FORM U-1 MANUFACTURES DATA FOR UNFIRED
PRESSURE VESSELS, INGERSOLL-RAND
2. ASME BOILER & PRESSURE VESSEL CODE SECTION III
UNFIRED PRESSURE VESSELS.
1965 EDITION,

FORM U-1A MANUFACTURERS' DATA REPORT FOR UNFIRED PRESSURE VESSELS
Alternate Form for Single Chamber Completely Shop-Fabricated Vessels Only

1. Manufactured by ELLICOTT FABRICATORS, INC. 1700 Ridgely Street, Blatimore 30, Md.
(Name and address of Manufacturer)

Manufactured for C. M. KEMP MFG. MARYLAND
(Name and address of Purchaser)

3. Type Vert. Vessel No. (14849-1) (Mfrs. Serial) () (State & State No.) Natl. Bd. No. 21774 Yr. Built 1961
J. & L. (Horiz. or Vert.)

4. SHELL: Matl. 106-B T.S. 60.000 Nom. Thk. 274n. Allow 1/16n. Diam. 8 Ft. 8 In. Length 6 Ft. 6-1/2n.
(Kind and Spec. No.) (Fig. or F.B. & Spec. Min. T.S.)

5. SEAMS: Long Seamless H.T. No X.R. No Sectioned No Efficiency 100 %
(Welded, Dbl., Single, Lap, Butt) (Yes or No)¹ (Spot or Complete) (Yes or No)
Girth Full-Fillet H.T. No X.R. No Sectioned No No. of Courses 1

If riveted or brazed describe seams fully under remarks.

6. HEADS: (a) Material <u>Beth. 212-B</u> T.S. <u>70.000</u>		(b) Material <u>Beth. 212-B</u> T.S. <u>70.000</u>						
Location (Top, bottom, ends)	Thickness	Crown Radius	Knuckle Radius	Elliptical Ratio	Conical Apex Angle	Hemispherical Radius	Flat Diameter	Side to Pressure (Convex or Concave)
(a) <u>Top</u>	<u>1"</u>						<u>8"</u>	<u>Flat</u>
(b) <u>Bottom</u>	<u>3/4"</u>						<u>8"</u>	<u>Flat</u>

If removable, bolts used _____ Other fastening _____
(Material, Spec. No., T.S., Size, Number) (Describe or Attach Sketch)

7. Constructed for max. allowable working press. ² 150 psi at max. temp. 450 °F. Min. temp. (when less than -20°) _____ °F. Hydrostatic Pneumatic or Combination } Test Press 300 psi.

8. SAFETY OR RELIEF VALVE OUTLETS: Number	Size	Location					
<u>1</u>	<u>1"</u>	<u>106-S Sch. 80 Pipe</u>					
9. NOZZLES: Purpose (Inlet, Outlet, Drain)	Number	Diam. or Size	Type	Material	Thickness	Reinforcement Material	Attachment
	<u>3</u>	<u>2"</u>	<u>106-B Sch. 40 Pipe</u>				<u>Welded</u>
	<u>3</u>	<u>1/2"</u>	<u>EH. Couplings</u>				<u>Welded</u>
	<u>2</u>	<u>1"</u>	<u>EH. Couplings</u>				<u>Welded</u>
	<u>1 Ea.</u>	<u>2"-3/4"</u>	<u>EH. Couplings</u>				<u>Welded</u>

OPENINGS: Handholes, No. _____ Size _____ Location _____
Threaded, No. _____ Size _____ Location _____

11. SUPPORTS: Skirt _____ Lugs _____ (Number) _____ Legs _____ (Number) _____ Other _____ (Describe) _____ Attached _____ (Where & How)

12. REMARKS: 1 TOWER

(Brief description of purpose of the vessel as Air Tank, Water Tank, L.P.G., Etc.—State Contents.
¹ If postweld heat-treated.
² List other internal or external pressures with coincident temperature when applicable.

We certify that the statements made in this report are correct and that all details of design, material, construction, and workmanship of this vessel conform to the ASME Code for Unfired Pressure Vessels.

Date January 9, 1961 Signed Ellicott Fabricators, Inc. By _____
(Manufacturer)

Certificate of Authorization Expires December 31, 1961

EFA

CERTIFICATE OF SHOP INSPECTION

VESSEL MADE BY ELLICOTT FABRICATORS, INC. at Baltimore 30, Maryland

I, the undersigned, holding a valid commission issued by the National Board of Boiler and Pressure Vessel Inspectors and/or the State of Maryland and employed by Mutual Boiler Insp. & Ins. Co. of Waltham 54, Mass. have inspected the pressure vessel described in this manufacturer's data report on January 9, 1961, and state that to the best of my knowledge and belief, the manufacturer has constructed this pressure vessel in accordance with the applicable sections of the ASME Boiler and Pressure Vessel Code.

By signing this certificate neither the Inspector nor his employer makes any warranty, expressed or implied, concerning the pressure vessel described in this manufacturer's data report. Furthermore, neither the Inspector nor his employer shall be liable in any manner for any personal injury or property damage or a loss of any kind arising from or connected with this inspection.

Date January 9, 1961

Inspector's Signature _____

Commissions _____

NB-3714

Natl Board or State and No.

FORM U-1A MANUFACTURERS' DATA REPORT FOR UNFIRED PRESSURE VESSELS
Alternate Form for Single Chamber Completely Shop-Fabricated Vessels Only

1. Manufactured by ELLICOTT FABRICATORS, INC., 1700 Ridgely Street, Baltimore 30, Md.
(Name and address of Manufacturer)

Manufactured for C.M. Kemp Mfg. Maryland
(Name and address of Purchaser)

3. Type Vert. Vessel No. (14850) () Natl. Bd. No. 21781 Yr. Built 1961
(Horiz. or Vert.) (Mfrs. Serial) (State & State No.)

4. SHELL: Matl. 106-B T.S. 60.000 Nom. 258 In. Allow 1/16 In. Diam. 5 Ft. 8 In. Length 2 Ft. 8 In.
(Kind and Spec. No.) (Fig. or F.B. & Spec. Min. T.S.)

5. SEAMS: Long Seamless H.T. No X.R. No Sectioned No Efficiency 100 %
(Welded, Dbl., Single, Lap, Butt) (Yes or No)¹ (Spot or Complete) (Yes or No)

Girth Full-Fillet H.T. No X.R. No Sectioned No No. of Courses 1

6. HEADS: (a) Material Beth. 285-C T.S. 55.000 (b) Material Beth. 285-C T.S. 55.000
(Top, bottom, ends) Thickness 5/8" Crown Radius Elliptical Ratio Conical Apex Angle Hemispherical Radius Flat Diameter 5" Side to Pressure (Convex or Concave)

(a) Top 5/8" 5" Flat

(b) Bottom 5/8" 5" Flat

If removable, bolts used (Material, Spec. No., T.S., Size, Number) Other fastening (Describe or Attach Sketch)

7. Constructed for max. allowable working press. 250 psi at max. temp. 650 °F. Min. temp. (when less than -20°) °F. Hydrostatic Pneumatic or Test Press 500 psi.
Combination

8. SAFETY OR RELIEF VALVE OUTLETS: Number Size Location

9. NOZZLES:

Purpose (Inlet, Outlet, Drain)	Number	Diam. or Size	Type	Material	Thickness	Reinforcement Material	How Attached
<u> </u>	<u>2</u>	<u>1/2"</u>	<u>EH. Couplings</u>	<u> </u>	<u> </u>	<u> </u>	<u>Welded</u>
<u>1 Ea.</u>	<u>1"</u>	<u>& 2"</u>	<u>EH. Couplings</u>	<u> </u>	<u> </u>	<u> </u>	<u>Welded</u>

10. INSPECTION Manholes, No. Size Location

OPENINGS: Handholes, No. Size Location

Threaded, No. Size Location

11. SUPPORTS: Skirt Lugs Legs Other Attached
(Yes or No) (Number) (Number) (Describe) (Where & How)

12. REMARKS: 1 Prefilter

(Brief description of purpose of the vessel as Air Tank, Water Tank, L.P.G., Etc.—State Contents.)
¹ If postweld heat-treated.
² List other internal or external pressures with coincident temperature when applicable.

We certify that the statements made in this report are correct and that all details of design, material, construction, and workmanship of this vessel conform to the ASME Code for Unfired Pressure Vessels.

Date January 6, 1961 Signed Ellicott Fabricators, Inc. By s/ E. Lockard
(Manufacturer)

Certificate of Authorization Expires December 31, 1961

EFA 61EFA 46 CERTIFICATE OF SHOP INSPECTION

VESSEL MADE BY ELLICOTT FABRICATORS, INC. at Baltimore 30, Md.

I, the undersigned, holding a valid commission issued by the National Board of Boiler and Pressure Vessel Inspectors and/or the State of Maryland and employed by Mutual Boiler Insp. & Ins. Co. of Waltham 54, Mass. have inspected the pressure vessel described in this manufacturer's data report on January 6, 1961 and state that to the best of my knowledge and belief, the manufacturer has constructed this pressure vessel in accordance with the applicable sections of the ASME Boiler and Pressure Vessel Code.

By signing this certificate neither the Inspector nor his employer makes any warranty, expressed or implied, concerning the pressure vessel described in this manufacturer's data report. Furthermore, neither the Inspector nor his employer shall be liable in any manner for any personal injury or property damage or a loss of any kind arising from or connected with this inspection.

Date January 6, 1961

Inspector's Signature Commissions NB-3714
Nat'l Board or State and No.

W. O. NO. 2288-12

BURNS AND ROE, INC.
NEW YORK, N. Y.

DATE: 7-18-66

DRAWING NO. _____

BY: M. GIDEN

TITLE: _____ SHEET 1 OF 5

STRESS ANALYSIS

PM-3A

INSTRUMENT AIR RECEIVER

ITEM 17.1

W. O. NO. 2288-12

BURNS AND ROE, INC.
NEW YORK, N. Y.

DATE: _____

DRAWING NO. _____

BY: J. CLAPP

TITLE: PM-34 DIESEL STARTING AIR TANK

SHEET 2 OF 5

ITEM 18.1

SUMMARY

THE DIESEL STARTING AIR TANK WAS ANALYZED IN ACCORDANCE WITH SECTION VIII OF THE ASME BOILER AND PRESSURE VESSEL CODE.

REQUIRED MINIMUM THICKNESSES WERE DETERMINED FOR THE CYLINDRICAL SHELL AND SEMI-ELLIPSOIDAL HEADS FOR THE DESIGN CONDITIONS OF PRESSURE (203 PSIG.) AND TEMPERATURE (650 °F.).

RESULTS OF ANALYSIS INDICATE THAT THE REQUIRED MINIMUM THICKNESSES OF THE SHELL AND HEADS (SA-285 C) ARE 0.256 IN. AND 0.177 IN., RESPECTIVELY.

SINCE THE ACTUAL MINIMUM THICKNESSES OF THE SHELL AND HEADS ARE 0.281 IN. AND 0.250 IN., RESPECTIVELY, THE TANK IS ADEQUATELY SIZED FOR THE DESIGN CONDITIONS INVESTIGATED.

NOZZLE REINFORCEMENT IS WAIVED IN ACCORDANCE WITH PAR. UG-36, REF. 2.

W. O. NO. 2284-12

BURNS AND ROE, INC.
NEW YORK, N. Y.

DATE: 7-19-66

DRAWING NO. _____

BY: A.J.S.

TITLE: AIR TANK - DIESEL STARTING

SHEET 3 OF 5

INTRODUCTION

THE VESSEL SERVES AS STORAGE FOR DIESEL ENGINE STARTING AIR ENERGY. IT CONSISTS OF A 24 INCH DIA. BY $\frac{9}{32}$ " THICK SHELL BY 66" LONG WITH ELLIPTICAL HEADS $\frac{1}{4}$ " THICK.

VESSEL PENETRATIONS ARE: DRAIN ONE $\frac{1}{2}$ ",
OUTLETS TWO 3", ONE $\frac{3}{4}$ ", ONE $\frac{1}{4}$ " AND TWO 2".

DESIGN CONDITIONS ARE: 203 PSI & 650°F

MATERIAL FOR FABRICATION IS SA-285-C

THE INVESTIGATION HAS BEEN MADE TO CHECK THE CONFIRMANCE OF THE AIR TANK WITH THE REQUIREMENTS OF THE ASME CODE, SECTION VIII FOR UNFIRED PRESSURE VESSELS. REQUIRED THICKNESSES ARE CALCULATED AND THEN COMPARED WITH THE ACTUAL THICKNESSES FURNISHED.

W. O. NO. 2288-12

BURNS AND ROE, INC.

DATE: 8/66.

DRAWING NO. _____

NEW YORK, N. Y.

BY: J. CLAPPTITLE: PM-3A INSTRUMENT AIR RECEIVER SHEET 4 OF 5CYLINDRICAL SHELL (PAR. UG-27, REF. 2)

$$t = \frac{PR}{SE - 0.6P}$$

$$= \frac{125 \times 10}{12,650(0.80) - 0.6(125)}$$

$$= \frac{1250}{10045}$$

$$= 0.124 \text{ IN. } < t_{\text{ACTUAL}} = 0.188 \text{ IN.}$$

$$P = 125 \text{ PSIG.}$$

$$R = 10 \text{ IN.}$$

$$S = 12,650 \text{ PSI.}$$

$$E = 80 \%$$

SHELL TEMP. (MAX.) 250 °F.

SHELL MAT'L: SA-283 C

(T.S. - 55,000 PSI.)

NOM. THICK. = $\frac{3}{16}$ IN.ELLIPSOIDAL HEAD (PAR. UG-32, REF. 2)

$$t = \frac{PD}{2SE - 0.2P}$$

$$= \frac{125 \times 20}{2(13,750)(1.00) - 0.2(125)}$$

$$= \frac{2500}{27,475}$$

$$= 0.091 \text{ IN. } < t_{\text{ACTUAL}} = 0.179 \text{ IN.}$$

$$P = 125 \text{ PSIG.}$$

$$D = 20 \text{ IN.}$$

$$S = 13,750 \text{ PSI.}$$

$$E = 100 \%$$

SHELL TEMP. (MAX.) 250 °F.

SHELL MAT'L: SA-414 C

(T.S. - 55,000 P.S.I.)

THICK. = 0.179 IN.

W.O. NO. 2288-12

BURNS AND ROE, INC.
NEW YORK, N. Y.

DATE: 7-19-65

DRAWING NO. _____

BY: M. WIDEN

TITLE: _____ SHEET 5 OF 5

REFERENCES

1. FORM U-1 MANUFACTURES DATA FOR UNFIRED
PRESSURE VESSELS. WESTINGHOUSE AIR BRAKE DIV.
2. ASME BOILER & PRESSURE VESSEL CODE SECTION III
UNFIRED PRESSURE VESSELS.
1965 EDITION.

FORM U-1A MANUFACTURERS' DATA REPORT FOR UNFIRED PRESSURE VESSELS
Alternate Form for Single Chamber Completely Shop-Fabricated Vessels Only

1. Manufactured by THE J. B. BEAIRD COMPANY, INC., SHREVEPORT, LOUISIANA
(Name and address of Manufacturer)

Manufactured for INGERSOLL RAND COMPANY, PAINTED POST, NEW YORK
(Name and address of Purchaser)

3. Type Horizontal Vessel No. (W-7955-14) (Mfrs. Serial) (State & State No.) Natl. Bd. No. 71273 Yr. Built 1960
(Horiz. or Vert.)

4. SHELL: Matl. A-283-C T.S. 55,000# Nom. Thk. 3/16 In. Allow 0 In. Diam. 1 Ft. 8 In. Length 4 Ft. 4-3/4 In.
(Kind and Spec. No.) (Fig. or F.B. & Spec. Min. T.S.)

5. SEAMS: Long Dbl. Butt H.T. No X.R. No Sectioned No Efficiency 80 %
(Welded, Dbl., Single, Lap, Butt) (Yes or No) (Spot or Complete) (Yes or No)

Girth Fig. UW-13K H.T. No X.R. No Sectioned No No. of Courses 1

If riveted or brazed describe seams fully under remarks.

6. HEADS: (a) Material A-414-C-Flg. T.S. 55,000# (b) Material Conical T.S. Flat
Location (Top, bottom, ends) Thickness .179 Crown Radius 2:1 Elliptical Ratio Apex Angle Hemispherical Radius Side to Pressure
(Convex or Concave)

(a) Ends (b) Concave

If removable, bolts used (Material, Spec. No., T.S., Size, Number) Other fastening (Describe or Attach Sketch)

7. Constructed for max. allowable working press. 125 psi at max. temp. 250 °F. Min. temp. (when less than -20°) Hydrostatic Test Press 188 psi.
Pneumatic or Combination

8. SAFETY OR RELIEF VALVE OUTLETS: Number Size Location

9. NOZZLES:
Purpose (Inlet, Outlet, Drain) Number Diam. or Size Type Material Thickness Reinforcement Material How Attached
(2) 2" Phoenix Ser. 153 Curved Flg. A-181-1: (1) 1/8" 6000# Coupling A-105-II Welded
(3) 1" Phoenix Ser. 150 Hvy. Curved Flg. A-181-1 Welded
(1) 3/4" (1) 1/4" Phoenix Ex. Hvy. Flat Flg. A-181-1 Welded

OPENINGS: Handholes, No. Size Location
Threaded, No. Size Location

11. SUPPORTS: Skirt (Yes or No) Lugs (Number) Legs (Number) Other (Describe) Attached (Where & How)

12. REMARKS: One 80 W.G. Horizontal Air Receiver, Per Beaird Dwg. W-7955

(Brief description of purpose of the vessel as Air Tank, Water Tank, L.P.G., Etc.—State Contents.
1 If postweld heat-treated.
2 List other internal or external pressures with coincident temperature when applicable.

We certify that the statements made in this report are correct and that all details of design, material, construction, and workmanship of this vessel conform to the ASME Code for Unfired Pressure Vessels. 1956 Edition

Date January 19 60 Signed THE J. B. BEAIRD CO., INC. By s/G. S. Stores
(Manufacturer)

Certificate of Authorization Expires DECEMBER 31, 1961

OCEAN W-7955-14 CERTIFICATE OF SHOP INSPECTION

VESSEL MADE BY THE J. B. BEAIRD CO., INC. at SHREVEPORT, LOUISIANA

I, the undersigned, holding a valid commission issued by the National Board of Boiler and Pressure Vessel Inspectors and/or the State of N. B. and employed by The Ocean Accident & Guarantee Corp. LTD of New York

have inspected the pressure vessel described in this manufacturer's data report on January 1960 and state that to the best of my knowledge and belief, the manufacturer has constructed this pressure vessel in accordance with the applicable sections of the ASME Boiler and Pressure Vessel Code.

By signing this certificate neither the Inspector nor his employer makes any warranty, expressed or implied, concerning the pressure vessel described in this manufacturer's data report. Furthermore, neither the Inspector nor his employer shall be liable in any manner for any personal injury or property damage or a loss of any kind arising from or connected with this inspection.

Date January 19 60

s/F. A. Zonkel
Inspector's Signature

Commissions

OHIO COMM.
PA. NO. WC 468
N. B. COMM. 2660
Natl Board or State and No.

FORM U-1A MANUFACTURERS' DATA REPORT FOR UNFIRED PRESSURE VESSELS
Alternate Form for Single Chamber Completely Shop-Fabricated Vessels Only

1. Manufactured by THE J. B. BEAIRD COMPANY, INC., SHREVEPORT, LOUISIANA
(Name and address of Manufacturer)

Manufactured for INGERSOLL RAND COMPANY, PAINTED POST, NEW YORK
(Name and address of Purchaser)

3. Type Horizontal Vessel No. (W-7955-108) (Mfr. Serial) (State & State No.) Natl. Bd. No. 71367 Yr. Built 1960
(Horiz. or Vert.)

4. SHELL: Matl. A-283-C T.S. 55,000# Nom. 3/16 Corr. 0 In. Allow 0 In. Diam. 1 Ft. 8 In. Length 4 Ft. 4-3/4 In.
(Kind and Spec. No.) (Fig. or F.B. & Spec. Min. T.S.) (Thk. In.)

5. SEAMS: Long Dbl. Butt H.T. No X.R. No Sectioned No Efficiency 80 %
(Welded, Dbl., Single, Lap, Butt) (Yes or No)¹ (Spot or Complete) (Yes or No)
Girth Fig. UW-13K H.T. No X.R. No Sectioned No No. of Courses 1

If riveted or brazed describe seams fully under remarks.

6. HEADS: (a) Material A-414-C-Fig. T.S. 55,000# (b) Material Conical Hemispherical Flat Side to Pressure
(Top, bottom, ends) Thickness .179 Crown Radius 2:1 Elliptical Ratio 2:1 Apex Angle Concave
(a) Ends (b) Concave

If removable, bolts used (Material, Spec. No., T.S., Size, Number) Other fastening (Describe or Attach Sketch)

7. Constructed for max. allowable working press. ² 125 psi at max. temp. 250 °F. Min. temp. (when less than -20°) 0 °F. Hydrostatic Pneumatic or Combination } Test Press 188 psi.

8. SAFETY OR RELIEF VALVE OUTLETS: Number Size Location

9. NOZZLES:
Purpose (Inlet, Outlet, Drain) Number Diam. or Size Type Material Thickness Reinforcement Material How Attached
(2) 2" Phoenix Ser. 153 Curved Flg. A-181-1; (1) 1/8" 6000# Coupling A-105-II Welded
(3) 1" Phoenix Ser. 150 Hvy. Curved Flg. A-181-1 Welded
(1) 3/4" (1) 1/4" Phoenix Ex. Hvy. Flat Flg. A-181-1 Welded

OPENINGS: Handholes, No. Size Location
Threaded, No. Size Location

11. SUPPORTS: Skirt Lugs Legs Other Attached
(Yes or No) (Number) (Number) (Describe) (Where & How)

12. REMARKS: One 80 W.G. Horizontal Air Receiver, Per Beard Dwg. W-7955

(Brief description of purpose of the vessel as Air Tank, Water Tank, L.P.G., Etc.—State Contents.)

¹ If postweld heat-treated.

² List other internal or external pressures with coincident temperature when applicable.

We certify that the statements made in this report are correct and that all details of design, material, construction, and workmanship of this vessel conform to the ASME Code for Unfired Pressure Vessels. 1956 Edition

Date January 19 60 Signed THE J. B. BEAIRD CO., INC. By s/G.S. Stores
(Manufacturer)

Certificate of Authorization Expires DECEMBER 31, 1961

OCEAN W-7955-108 CERTIFICATE OF SHOP INSPECTION

VESSEL MADE BY THE J. B. BEAIRD CO., INC. at SHREVEPORT, LOUISIANA

I, the undersigned, holding a valid commission issued by the National Board of Boiler and Pressure Vessel Inspectors and/or the State of N. B. and employed by The Ocean Accident & Guarantee Corp. LTD of New York have inspected the pressure vessel described in this manufacturer's data report on January 1960 and state that to the best of my knowledge and belief, the manufacturer has constructed this pressure vessel in accordance with the applicable sections of the ASME Boiler and Pressure Vessel Code.

By signing this certificate neither the Inspector nor his employer makes any warranty, expressed or implied, concerning the pressure vessel described in this manufacturer's data report. Furthermore, neither the Inspector nor his employer shall be liable in any manner for any personal injury or property damage or a loss of any kind arising from or connected with this inspection.

Date January 19 60

s/F. A. Zonkel

Inspector's Signature

Commissions

OHIO COMM.

PA. NO. WC 468

N. B. COMM. 2660

Natl Board or State and No.

W. O. NO. 2288-12

BURNS AND ROE, INC.
NEW YORK, N. Y.

DATE: 7-19-66

DRAWING NO. _____

BY: M. GIDEN

TITLE: _____ SHEET 1 OF 5

STRESS ANALYSIS

PM-3A

DIESEL STARTING AIR TANK

ITEM 18.0

W. O. NO. 2288-12

BURNS AND ROE, INC.
NEW YORK, N. Y.

DATE: _____

DRAWING NO. _____

BY: J. CLAPP

TITLE: PM-3A INSTRUMENT AIR-RECEIVER SHEET 2 OF 5

ITEM 17.1

SUMMARY

THE INSTRUMENT AIR-RECEIVER WAS ANALYZED IN ACCORDANCE WITH SECTION VIII OF THE ASME BOILER AND PRESSURE VESSEL CODE.

REQUIRED MINIMUM THICKNESSES WERE DETERMINED FOR THE CYLINDRICAL SHELL AND SEMI-ELLIPSOIDAL HEADS FOR THE DESIGN CONDITIONS OF PRESSURE (125 PSIG.) AND TEMPERATURE (250 °F.).

RESULTS OF ANALYSIS INDICATE THAT THE REQUIRED MINIMUM THICKNESSES OF THE SHELL (SA-283 C) AND HEADS (SA-414 C) ARE 0.124 IN. AND 0.091 IN., RESPECTIVELY.

SINCE THE ACTUAL MINIMUM THICKNESSES OF THE SHELL AND HEADS ARE 0.188 IN. AND 0.179 IN., RESPECTIVELY, THE RECEIVER IS ADEQUATELY SIZED FOR THE DESIGN CONDITIONS INVESTIGATED.

NOZZLE REINFORCEMENT IS WAIVED IN ACCORDANCE WITH PAR. UG-36, REF. 2.

W. O. NO. 2282-12

BURNS AND ROE, INC.
NEW YORK, N. Y.

DATE: 7-13-66

DRAWING NO.

BY: EJS

TITLE: RECEIVER - INSTRUMENT AIR

SHEET 3 OF 5

INTRODUCTION

THE VESSEL SERVES AS STORAGE AND PULSATION DAMPENER FOR INSTRUMENT AIR. IT CONSISTS OF A 20. INCH DIA, BY $\frac{3}{16}$ " THICK SHELL BY $4'-4\frac{3}{4}"$ LONG WITH CONCAVE ENDS 0.179" THICK.

VESSEL PENETRATIONS ARE: TWO 2", ONE 1", ONE $\frac{3}{4}"$, ONE $\frac{1}{4}"$ & ONE $\frac{1}{8}"$.

MATERIAL FOR FABRICATION IS SA-283-C & -414-C

DESIGN CONDITIONS ARE: 125. PSI & 250°F.

THE INVESTIGATION HAS BEEN MADE TO CHECK THE CONFORMANCE OF THE VESSEL WITH THE REQUIREMENTS OF THE ASME CODE, SECTION VIII FOR UNFIRED PRESSURE VESSELS. REQUIRED THICKNESSES ARE CALCULATED AND THEN COMPARED WITH THE ACTUAL THICKNESSES FURNISHED.

W. O. NO. 2288-12BURNS AND ROE, INC.
NEW YORK, N. Y.DATE: 3/52

DRAWING NO. _____

BY: J. CLAPPTITLE: PM-3A DIESEL STARTING AIR TANK SHEET 4 OF 5CYLINDRICAL SHELL (PAR. UG-27, REF. 2)

$$t = \frac{PR}{SE - 0.6P}$$

$$= \frac{203 \times 12}{13,750(0.70) - 0.6(203)}$$

$$= \frac{2436}{9503}$$

$$= 0.256 \text{ IN.} < t_{\text{ACTUAL}} = 0.281 \text{ IN.}$$

$$P = 203 \text{ PSIG.}$$

$$R = 12 \text{ IN.}$$

$$S = 13,750 \text{ PSI.}$$

$$E = 70\%$$

SHELL TEMP (MAX) 650 °F.
SHELL MATL: SA-285C
(T.S. - 55,000 PSI.)

$$\text{NOM. THICK.} = \frac{9}{32} \text{ IN.}$$

ELLIPSOIDAL HEAD (PAR. UG-32, REF. 2)

$$t = \frac{PD}{2SE - 0.2P}$$

$$= \frac{203 \times 24}{2(13750)(1.00) - 0.2(203)}$$

$$= \frac{4872}{27,457}$$

$$= 0.177 \text{ IN.} < t_{\text{ACTUAL}} = 0.250 \text{ IN.}$$

$$P = 203 \text{ PSIG.}$$

$$D = 24 \text{ IN.}$$

$$S = 13,750 \text{ PSI.}$$

$$E = 100\%$$

HEAD TEMP MAX, 650 °F.
HEAD MATL: SA-285C
(T.S. - 55,000 PSI.)

$$\text{NOM. THICK.} = \frac{1}{4} \text{ IN.}$$

W. O. NO. 2-288-12

BURNS AND ROE, INC.
NEW YORK, N. Y.

DATE: 7-19-66

DRAWING NO. _____

BY: M. GIDEN

TITLE: _____ SHEET 5 OF 5

REFERENCES

1. FORM U-1 MANUFACTURES DATA FOR UNFIRED
PRESSURE VESSELS. INGERSOLL-RAND
2. ASME BOILER & PRESSURE VESSEL CODE SECTION VIII
UNFIRED PRESSURE VESSELS.
1965 EDITION

FORM U-1A MANUFACTURERS' DATA REPORT FOR UNFIRED PRESSURE VESSELS
Alternate Form for Single Chamber Completely Shop-Fabricated Vessels Only

1. Manufactured by Westinghouse Air Brake Division, Westinghouse Air Brake Co., Wilmerding, Pa.
(Name and address of Manufacturer)

2. Manufactured for Martin Marietta Corp. Middle River, Maryland
(Name and address of Purchaser)

3. Type Ver. Vessel No. (193603) (Mfrs. Serial) (State & State No.) Natl. Bd. No. 193603 Yr. Built 1963
(Horiz. or Vert.)

4. SHELL: Matl. SA-285-C T.S. 55000 Nom. Thk 9/32 In. Corr. Allow In. Diam. 24 In. Length 66 In.
(Kind and Spec. No.) (Fig. or F.B. & Spec. Min. T.S.)

5. SEAMS: Long Dbl. Butt H.T. X.R. No Sectioned Efficiency 70 %
(Welded, Dbl., Single, Lap, Butt) (Yes or No)¹ (Spot or Complete) (Yes or No)

Girth Dbl. Butt H.T. X.R. No Sectioned No. of Courses 1

6. HEADS: (a) Material SA-285-C T.S. 55000 (b) Material T.S.
Location (Top, bottom, ends) Thickness 1/4" Crown Radius Knuckle Radius Elliptical Ratio 2-1 Conical Apex Angle Hemispherical Radius Flat Diameter Side to Pressure (Convex or Concave) Concave

(a) Ends (b)

If removable, bolts used (Material, Spec. No., T.S., Size, Number) Other fastening (Describe or Attach Sketch)

7. Constructed for max. allowable working press.² 203 psi at max. temp. 650 °F. Min. temp. (when less than -20°) °F. Hydrostatic Pneumatic or Combination } Test Press 406 psi.

8. SAFETY OR RELIEF VALVE OUTLETS: Number Size Location

9. NOZZLES:

Purpose (Inlet, Outlet, Drain)	Number	Diam. or Size	Type	Material	Thickness	Reinforcement Material	How Attached
Drain	<u>1</u>	<u>1/2"</u>	<u> </u>	<u> </u>	<u> </u>	<u> </u>	<u>Welded</u>
Outlet	<u>1</u>	<u>1/4"</u>	<u>1</u>	<u>2</u>	<u>3"</u>	<u> </u>	<u>Welded</u>

10. INSPECTION Manholes, No. Size Location

OPENINGS: Handholes, No. Size Location

Threaded, No. 2 Size 2" Location SHELL

11. SUPPORTS: Skirt (Yes or No) Lugs (Number) Legs 3 (Number) Other (Describe) Attached Shell Welded (Where & How)

12. REMARKS: Air Tank
Radiographing not required as per ASME Boiler and Pressure Vessel
Code 1962 Edition Paragraph UW12(c)

(Brief description of purpose of the vessel as Air Tank, Water Tank, L.P.G., Etc.—State Contents.
¹ If postweld heat-treated.
² List other internal or external pressures with coincident temperature when applicable.

We certify that the statements made in this report are correct and that all details of design, material, construction, and workmanship of this vessel conform to the ASME Code for Unfired Pressure Vessels.

Date November 4 19 63 Signed Air Brake Division, Westinghouse By s/ Stewart C. Rowles
(Manufacturer)

Certificate of Authorization Expires December 31, 1964

193603 CERTIFICATE OF SHOP INSPECTION

VESSEL MADE BY Westinghouse Air Brake Div. at Wilmerding, Pa.

I, the undersigned, holding a valid commission issued by the National Board of Boiler and Pressure Vessel Inspectors and/or the State of Penn. and employed by Lumdermens Mutual Casualty Co. of Chicago, Ill. have inspected the pressure vessel described in this manufacturer's data report on Nov. 4 19 63 and state that to the best of my knowledge and belief, the manufacturer has constructed this pressure vessel in accordance with the applicable sections of the ASME Boiler and Pressure Vessel Code.

By signing this certificate neither the Inspector nor his employer makes any warranty, expressed or implied, concerning the pressure vessel described in this manufacturer's data report. Furthermore, neither the Inspector nor his employer shall be liable in any manner for any personal injury or property damage or a loss of any kind arising from or connected with this inspection.

Date November 4 19 63
R. W. Brown Commissions NB-498
Inspector's Signature Nat'l Board or State and No.