MASTER

PAE No. 41
Volume I

INTERMEDIATE HEAT EXCHANGER PRELIMINARY DESIGN

ALCO

Submitted to
UNITED STATES
ATOMIC ENERGY COMMISSION

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ALCO PRODUCTS, INC.
Research And Development

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APAE No. 41
Volume I
Of 3 Volumes

INTERMEDIATE HEAT EXCHANGER

PRELIMINARY DESIGN

Vol. I IHX Preliminary Design
Vol. II Steam Generator Preliminary Design
Vol. III IHX & Steam Generator Concept Study

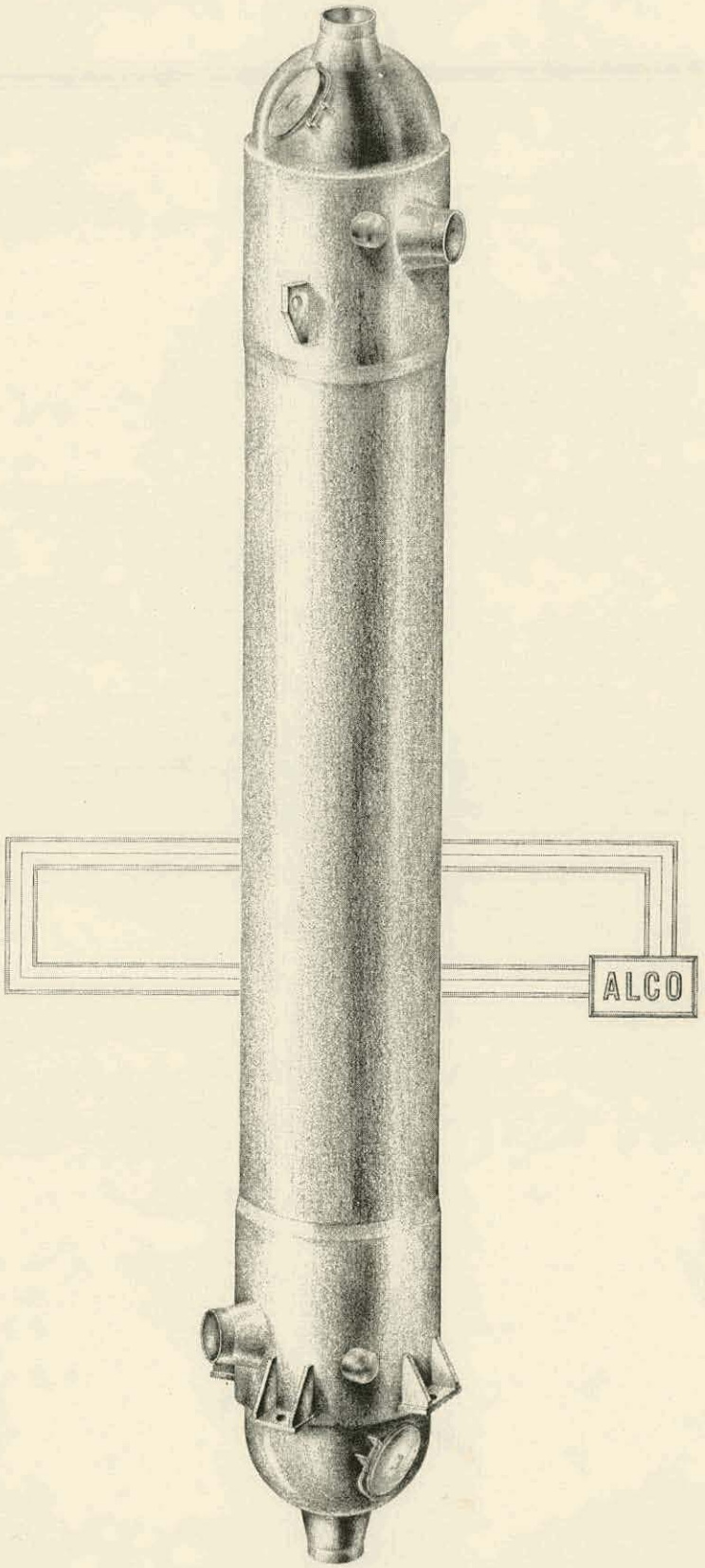
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CHICAGO OPERATIONS OFFICE
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Contract No. AT(11-1)-666
February 28, 1959

ALCO PRODUCTS, INC.
Research and Development
Schenectady, New York

367 001



967 002





967 003

PREFACE

The intermediate heat exchanger is designed for operation in a nuclear power plant using liquid sodium as the primary and secondary coolant. Since the primary fluid coming from the reactor is radioactive, the purpose of the IHX is to transfer heat to a nonradioactive fluid which then goes to a steam generator. Because of this activity the unit will be enclosed in a concrete pit and will not be accessible during periods of operation. Immediately after shut down it will be necessary to allow time for radioactive decay before the unit will be accessible to personnel. Because of inaccessibility and possible long periods allowed for decay time, it is imperative that the unit give trouble free operation. During periods of shut down, the internals should have easy access for inspection and repair if necessary so that down time is held to a minimum.

The basic design is a shell and tube heat exchanger with fixed tube sheets and sine wave tubes. This tube design offers the advantage of operating essentially independently of the shell or adjacent tubes. This means each tube can lengthen or shorten according to the surrounding temperature without being affected by expansion or contraction of the shell or other tubes.

Numerous investigations during this design period have pointed out that a critical area exists in every shell and tube design, which is the tube to tube sheet welds. This area becomes most critical during transient conditions when relatively cold fluid starts to flow through the tubes. As the cold fluid flows through the tubes it cools the thin metal walls and the tubes grow smaller in diameter. Because of the mass of material in the tube sheet it does not cool as rapidly as the tubes. Since the tubes are welded to the tube sheet, a high stress concentration area develops at the welds because the tubes decrease in diameter but the tube sheet remains unchanged. This phenomenon is less critical during transient conditions when fluid of increasing temperature flows through the tubes because the tube tends to expand against the tube sheet holes. In order to reduce the stress concentration at the tube to tube sheet welds a trepanning operation is performed on the tube sheet which makes a nipple at each tube penetration. The nipple extends approximately 1/2 inch above the tube sheet and is the same thickness as the tube wall at the weld. The nipple thickness increases over its extension and is flared into the face of the tube sheet. This design reduces stress concentration at the weld in two ways. The external pressure tends to squeeze the nipple to the tube and fluid flowing through the tube spills over the nipple and since the thickness at the weld is the same as the tube, the two metal temperatures will be essentially the same. In this manner the stress concentration is moved away from the weld to a point where the continuity of the metal is not upset by welding and where the material is thicker.

The channel end of the heat exchanger is hemispherical in order to reduce material thickness to a minimum. Included in each closure is a manway which provides access to the tube sheets for inspection and repairing or sealing off tubes if necessary. Inspection may be accomplished with a minimum of cutting of piping and headers.

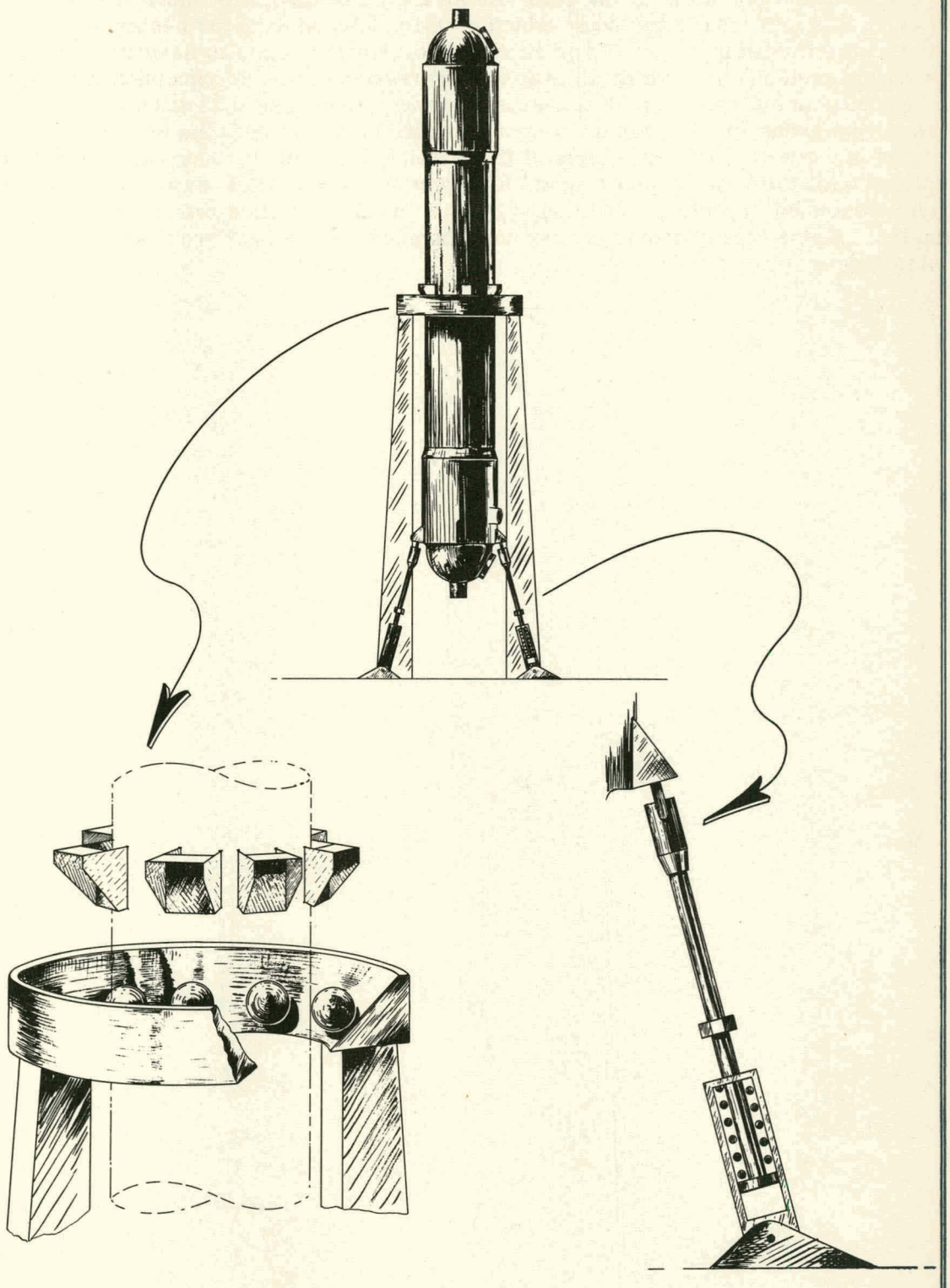
Experience has shown that sodium heat exchangers rapidly go isothermal at low flow rates and at shutdown conditions. Rapid stratification of the sodium is often experienced prior to this isothermal condition. In the present design a vertical shell with close overlapping disc and doughnut baffles is used to minimize these undesirable conditions.

The general arrangement of this heat exchanger presents a conventional design utilizing known materials and existing methods of fabrication. In further consideration of all concepts, designs and analyses developed during this period of the program, it is felt that this preliminary design will provide an intermediate sodium heat exchanger of lower cost and more reliable operation.

007 004

In the preliminary design of the heat exchanger, conventional mounting brackets have been placed near the bottom of the unit. As an alternate method of mounting, some thought has been given to a universal type mount which would be placed near the center of the unit as indicated on the following sketch. A preliminary evaluation points to several advantages over the conventional method of mounting as follows: (1) freedom of shell expansion (2) equalization of pipe reactions and (3) freedom of movement in any plane. The hold down units shown at the bottom of the exchanger are simply a spring loaded device to hold the exchanger down if pipe reactions are greater than the weight of the vessel. Detailed engineering, of course, cannot be completed until the location and magnitude of pipe reactions are known. Other factors such as angular movement expected, vibration, cost etc. must be studied before its potential can be evaluated. If this type of mount proves advantageous for the heat exchanger, it could serve for the steam generator as well.

967 005



—UNIVERSAL MOUNT—

967 006

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SECTION 1

INTRODUCTION

INTRODUCTION

In conjunction with the development of sodium cooled reactor systems for producing economic electric power from nuclear energy, a research and development program has been sponsored by the U. S. Atomic Energy Commission to develop less expensive and more reliable sodium components. In support of this program a conceptual study has been carried on leading to a preferred concept and preliminary design of an intermediate sodium heat exchanger. A detailed description of the preliminary design and analysis is presented in the following sections.

967 009

SECTION 2

DESCRIPTION OF DESIGN AND ANALYSIS

A - THERMAL DESIGN

General

The basic size and arrangement of the intermediate sodium heat exchanger is primarily dependent upon thermal design considerations. This includes a literature survey to establish the thermal design criteria, a review of the design operating conditions, a concept study including the preliminary analysis, and a complete thermal analysis of the preliminary design. In brief, the thermal analysis includes the complete calculations for predicting heat transfer, fluid flow and thermal performance.

Thermal Design Criteria

A literature survey and evaluation was made in order to obtain the most applicable heat transfer and fluid flow design criteria. In general, this survey included a review of current engineering handbooks and text books, supplemented by some recently published technical periodicals and papers. A brief summary is presented in the following paragraphs which includes the preferred analytical expressions for predicting sodium heat transfer and fluid flow.

Heat transfer to sodium flowing inside of tubes is best represented for design purposes by the "Lubarsky and Kaufman Equation".

$$Nu = 0.625 (Re\ Pr)^{0.4}$$

Ref. NACA Report 1270

This equation is the best representation of the experimental results of 20 observations using various liquid metals. Most of the data for alkali metals indicate some conservatism in this equation.

For predicting heat transfer to sodium flowing on the shell side of tubes the only available source of information is the 3rd edition (Sodium - NaK Supplement) of the "Liquid-Metals Handbook". Four empirical relations are presented from the actual test results of a few baffled and unbaffled sodium exchangers of small and large sizes. The following expressions relating to large size exchangers are recommended for present application to this design.

(1) Baffled (cross flow) $Nu = .212 (D_E\ Re\ Pr)^{0.6}$
(2) Unbaffled (axial flow) $Nu = .106 (D_E\ Re\ Pr)^{0.6}$ (*)

where D_E is the equivalent diameter of shell in inches.

Since these expressions do not consider the type of baffle arrangement, tube pitch, or flow configuration there is need for more applicable equations.

Included in the fluid flow analysis of the thermal design calculations is the evaluation of the tube and shell side pressure drops. The over-all nozzle to nozzle pressure drop for both sides of the exchanger was considered as a series of component pressure drops. Each component was either calculated by utilizing conventional fluid flow equations or estimated from previous experience. For the tube side, from inlet nozzle to outlet nozzle, the series of pressure drops include:

(*) Refer to page 285, "Liquid-Metals Handbook", and note that the above equation is the unbaffled modification of equation (4) which was made similar to the modification of equation (1) to get equation (5). This was done because of the difficulty encountered in using equations (1) and (5).

067 011

(1) Expansion, inlet line to closure.
(2) Contraction, closure into tubes.
(3) Axial flow through tubes.
(4) Expansion, tubes into closure.
(5) Contraction, closure to outlet line.

For the shell side, from inlet nozzle to outlet nozzle, the series of pressure drops include:

(1) Entrance; includes expansion, impact and turns.
(2) Axial flow along length of sine wave bend, includes cross-flow.
(3) Cross-flow between baffles.
(4) Baffle window – outside disc, includes turns.
(5) Baffle window – inside doughnut, includes turns.
(6) Exhaust, includes contraction and expansion.

The conventional fluid flow equations that are recommended for the analysis are as follows:

(1) Axial flow for Re of 5000 to 200,000

$$\Delta P_L = f_L \left(\frac{G^2}{2g_c \rho}\right) \frac{L}{r_h} \left(\frac{1}{144}\right)$$

where: $r_h = \frac{D_h}{4}$

$$f_L = \frac{.046}{(Re)^{0.2}}$$

Ref.: "Heat Transmission", Chapter 6 by W. McAdams.

(2) Cross flow for Re_x of 2000 to 40,000 and staggered tubes of X_T of 1.5 to 4.0.

$$\Delta P_X = 4 \; f_x \, N_{tc} \left(\frac{Gmax^2}{2g_c \rho}\right) \frac{1}{144}$$

where: $Re_x = \frac{D_o \, G_{max}}{12 \, \mu_f}$

$$X_T = \frac{\text{Transverse Pitch}}{\text{Tube Diameter}}$$

$$f_x = \left[0.23 + \frac{0.11}{(X_T - 1)^{1.08}}\right] (Re_x)^{-0.15}$$

Ref.: "Heat Transmission", Chapter 6 by W. McAdams

007 012

(3) Sudden Expansion

$$\Delta Pe = K_e\left(\frac{G_1^2}{2g_c\rho}\right)\frac{1}{144}$$

where: $K_e = \left(1 - \frac{A_1}{A_2}\right)^2$, Loss coefficient

A_1, A_2 = cross sectional flow area in order of flow.

Ref.: "Reactor Handbook", Volume 2, Chapter 1.5.

(4) Sudden Contraction for Re > 10,000

$$\Delta P_c = K_c\left(\frac{G_2^2}{2g_c\rho}\right)\frac{1}{144}$$

where: K_c = Loss coefficient, Figure 1 on the following page.

Ref.: "Reactor Handbook", Volume 2, Chapter 1.5.

In order to apply the preceding heat transfer and fluid flow equations to the actual design certain physical properties of sodium must be evaluated at various temperatures. The best presentation of these properties is given in the 3rd Edition (Sodium - NaK Supplement) of the "Liquid-Metals Handbook". Plots of density, viscosity, thermal conductivity and specific heat versus temperature for liquid sodium are included in the thermal design calculations of the Appendix.

Design Operating Conditions

A preliminary consideration of the design and operation of the primary and secondary sodium components was required at the beginning of the design period in order to establish the primary and secondary sodium flow rates. From the temperature limits as given in the functional specifications, Contract AT(11-1)-666, for the primary outlet and secondary inlet to the intermediate heat exchanger the corresponding limits of sodium flow rate were obtained. For the primary loop the limits of the flow rate are 1.59×10^6 to 2.66×10^6 lb./hr. and likewise for the secondary loop the limits are 1.62×10^6 to 2.74×10^6 lb./hr. If the systems were limited to a balanced flow arrangement the corresponding limits would be 1.62×10^6 to 2.66×10^6 lb./hr. or a possible variation in design flow rate of 1 to 1.65. These considerations assume that the secondary sodium outlet temperature from the heat exchanger is 1140 F or as originally specified.

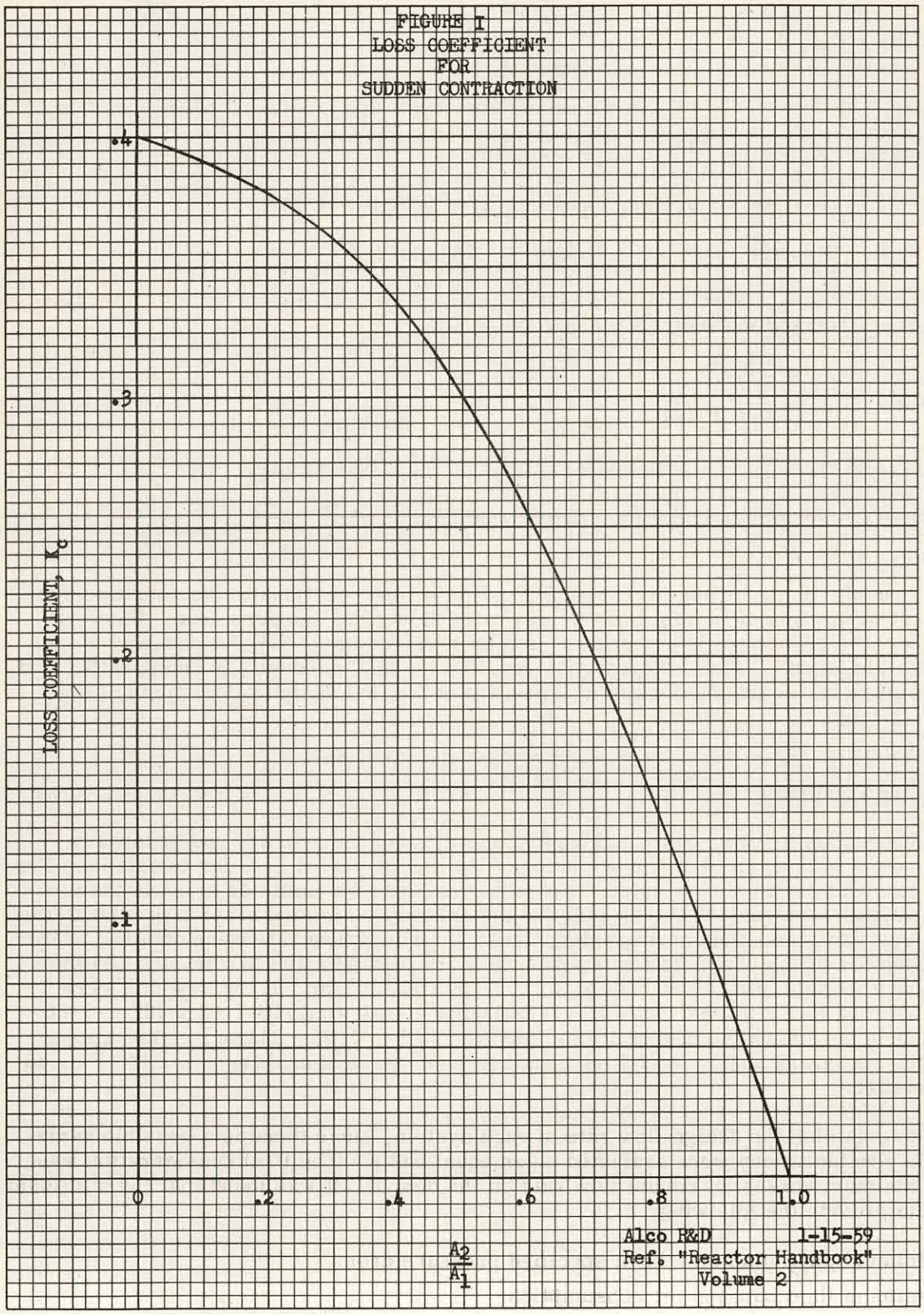
In order to establish the design flow rate of each system, since there is a possible variation, two main considerations were evaluated on an equal basis. The first consideration was the effect of minimum versus maximum flow rate on the design. It was found that the minimum flow rate would provide the following:

(1) Minimum shell diameter for a specified pressure drop.

(2) Minimum fluid horsepower consumption.

(3) Maximum temperature change which gives a lower sodium temperature at boiler outlet for a possible control point.

867 013


FIGURE I
LOSS COEFFICIENT
FOR
SUDDEN CONTRACTION
LOSS COEFFICIENT, K_c
.4
.3
.2
.1
0
.2
.4
.6
.8
1.0
$\frac{A_2}{A_1}$
Alco R&D 1-15-59
Ref. "Reactor Handbook"
Volume 2

967 014

The second consideration was a comparison of balanced versus unbalanced flow rates in the primary and secondary systems. It was found that a balanced flow arrangement would provide the following:

(1) Less complex control system for variable load operation.

(2) Lower temperature gradients within the tube sheet material to give lower thermal stresses.

From these considerations it seems evident that a balanced flow arrangement with a minimum flow rate would present the most preferred design.

From the concept study of the over-all system a recirculation type of steam boiler was preferred for the secondary loop. With this type of unit preheating of the water takes place in the steam drum causing the lower temperature of the boiler inlet water to increase to approximately saturation. For this design the sodium outlet temperature of the boiler must be increased to above its lower limit. For a reasonable boiler design a terminal temperature difference of 15^{o} F was preferred (Reference Volume II of this report). This establishes the design temperature of the secondary sodium inlet to the heat exchanger at 665 deg. F and the design flow rate of both sodium loops at 1.665×10^{6} lb./hr. For the analytical determination of these conditions refer to the thermal design calculations presented in the Appendix.

The optimization of the division of heat transfer surface between the intermediate heat exchanger and steam generator was considered in conjunction with the design conditions. Although an analytical analysis did not present conclusive proof that the optimum design conditions were established, it did show trends in support of the above selection.

Concept Study

In order to establish the preferred design a concept study including a preliminary analysis was made. The preliminary analysis and evaluation of the basic types of shell and tube heat exchangers resulted in the selection of a preferred concept. Further analysis of this concept which considered variations in shell diameter, tube size and pitch, baffle arrangement, shell and tube velocities, and fluid horsepower consumption resulted in a economic evaluation and selection of the preferred design. Volume III of this report presents a summary of the concept study and preliminary analysis which includes the designs considered with reasons for their rejection.

The preferred design can be described as a vertical fixed tube sheet type of unit with prebent sine wave tubes to take up thermal expansion and close overlapping disc and doughnut baffles to minimize sodium stratification.

Thermal Analysis

During this design program certain problem areas were considered which are reflected in the preliminary design. A brief discussion of the thermal problems considered and how the design was affected is presented as follows:

(1) Sodium Stratification

Experience has shown that a sodium heat exchanger will rapidly go isothermal during low flow rates and at shutdown conditions. Poorly positioned segmental type baffles inside a horizontal shell together with conventional manufacturing tolerances contribute to this phenomenon. From this consideration a vertical shell with close overlapping disc and doughnut baffles is felt to be a positive method for reducing these effects.

867 015

(2) Fluid Distribution

Bypassing and poor fluid distribution within a heat exchanger has considerable effect on reducing the heat transfer capabilities of the unit. In the present design close overlapping baffles is felt to provide a more predictable flow distribution. In addition manufacturing tolerances and clearances are maintained at a minimum to reduce bypassing of shell fluid.

(3) Flow Control

To provide for a less complex control system and simplicity in operation, a balanced flow arrangement is recommended. This system will also provide lower temperature gradients within tube sheet material.

(4) Heat Transfer Data

Basic heat transfer data is very meager and considered inadequate for application to large scale sodium exchangers. There is considerable variation between the experimental and predicted tube side heat transfer correlations, and there is need for more applicable shell side correlations. The effect on shell side heat transfer from variations in baffling and flow configuration cannot be evaluated with present available correlations. To account for inaccuracies in the heat transfer analysis, a conservative approach was taken by using conservative correlations where possible and including some excess heat transfer surface in the design.

To complete the thermal analysis of the preliminary design a detail analytical evaluation of heat transfer, pressure drop and thermal performance was made. One basic design change was introduced during the final thermal analysis. In the preferred concept a center cylindrical spacer and outer cylindrical shroud were included for sine wave tubes and shell uniformity. Upon consideration of the voided volume of the unit it was decided that these regions could be used for fluid turning in a completely overlapped baffle arrangement. With this feature the heat transfer is more effective, the flow distribution is more predictable and the internal natural circulation of sodium at low flow rates is isolated in layers.

The final thermal design calculations of the preliminary design are included in the appendix of this report.

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B - MECHANICAL DESIGN

The Intermediate Heat Exchanger is mechanically designed in accordance with the A.S.M.E. Unfired Pressure Vessel Code, Section VIII, 1956 Edition, including 1957 Addenda and T.E.M.A. Class R Standards where applicable.

Materials and calculations are in accordance with the above codes. The calculations reflect minimum design thicknesses of the various components using code stresses suitable for the design temperatures.

All design drawings reflect the final thicknesses resulting from transients, loadings, pipe moments and gross structural discontinuities.

Mechanical design calculations are included in the appendix section of this report.

967 017

C - STRESS ANALYSIS

Structural Design Basis:

The U.S. Navy Code "Tentative Structural Design Basis for Reactor Pressure Vessels and Directly Associated Components" dated 1 April 1958, SSN/S51(551B) Ser. 551B-596, has been selected as a method for handling stress analysis of combined mechanical hydrostatic and thermal stresses, and conditions for using creep and stress rupture properties.

The structural materials used will be 2 1/4% Cr, 1% Mo low alloy steel (Croloy 2 1/4) for the boiler and Type 316 stainless steel for all other parts. Tables of allowable stresses up to 1200 deg. F for these two materials are attached. For membrane stresses and for stresses due to gross discontinuities these are based upon figures given in the latest revisions of the Unfired Pressure Vessels Code as modified by Ruling 1234. Values of yield strength and of ultimate strength are based upon the best available data, as given in tabular form in the Appendix.

The allowable stress for pipe reaction stresses plus membrane and gross discontinuity stresses is taken as 90% of the yield strength.

Values for thermal stress and stress concentrations are taken from the Navy Code up to 700 deg. F, which is as far as it goes. Beyond this, the values are assumed to vary with the ultimate strength. This assumption is in accord with such experimental results as to date have been made available. Further experimental work is desirable.

Thermal stresses are treated as governed by Para. 5.1.4 of the Navy Code; "No stress limitation need be considered with regard to steady-state thermal stresses. All thermal stresses shall be considered as transient conditions and treated in accordance with Para. 5.2."

Major stress problems are cared for as follows:

Differential Expansion Between Tubes and Shell:

To provide for freedom of differential expansion between tubes and shell and between different tubes, all tubes in the heat exchanger are provided with a pre-bent sine wave: i.e., a curved portion 48" long with 10" maximum offset, placed adjacent to the upper tube sheet. In order to furnish the necessary clearance for this sine wave, the tubes are placed in concentric circles, and the wave is given a double curvature to fit these circles. An area in the center of the tube sheet is left without tubes. This area serves a useful purpose in providing a turning area for the flow between the disk and doughnut baffles.

Under the extreme condition of an isothermal cold, these waves will be required to provide for a differential contraction of 1.06". This corresponds to a longitudinal bending stress of 53,600 psi - as compared to an allowable stress at 640 deg. F of 66,000 psi. Other stresses simultaneously present will be small. Under an isothermal hot, the differential expansion will be 0.86", and the bending stress 36,300 as compared to an allowable stress at 1200 deg. F of 48,000 psi. This is a total expected movement from ± 0.86" to -1.06", or a range of 1.92". When tested experimentally at room temperature, a 48" sine-wave 1/2" dia. stainless steel tube withstood successively 25 cycles each at 1.5", 2.0", 2.5", 3.0", and 4.0" range, showing no signs of damage whatever.

Statement of theory and detailed computations are given in the Appendix.

967 018

GTD 239

MAXIMUM ALLOWABLE STRESS INTENSITIES
Croloy (2¼% Cr 1% Mo low alloy steel)
Thousands of Pounds per Sq. In.

Metal Temp °F	U F P V		S P		Sy Yield Strength	Su Ultimate Strength	Thermal Stress ≠ Stress Concentration				
	Plate or Tubing	Forgings	Membrane ≠ Discontinuities	≠ Pipe Reactions			SS Steady State Component	SO Endurance Limit	Sa I 100000	SA II 2500	SA III 500'
RT	15.0	17.5		27.0	30.0	60.0	50.	24.	24.	45.	66.
400	"	"		"	"	"	"	"	"	"	"
600	"	"		"	"	"	"	"	"	"	"
700	"	"		"	"	"	47	"	"	"	"
750	"	"		"	"	"	46	23	23	42	63
800	"	"		"	"	"	44	22	22	40	60
850	14.4	16.0		"	"	"	40	20	20	38	57
900	13.1	14.0	50% above UFPV Code	27	30	60	36	19	19	36	54
950	11.0	11.0					33	18	18	34	51
1000	7.8	7.8		24.3	27	50	30	17	17	32	48
1050	5.8	5.8						16	16	29	44
1100	4.2	4.2		21.6	24	40	24	14.5	14.5	26	40
1150	3.0	3.0						12.8	12.8	23	35
1200	2.0	2.0		16.2	18.0	30	18	11.0	11.0	20	30

90% of Yield Strength

MAXIMUM ALLOWABLE STRESS INTENSITIES
TYPE 316 STAINLESS

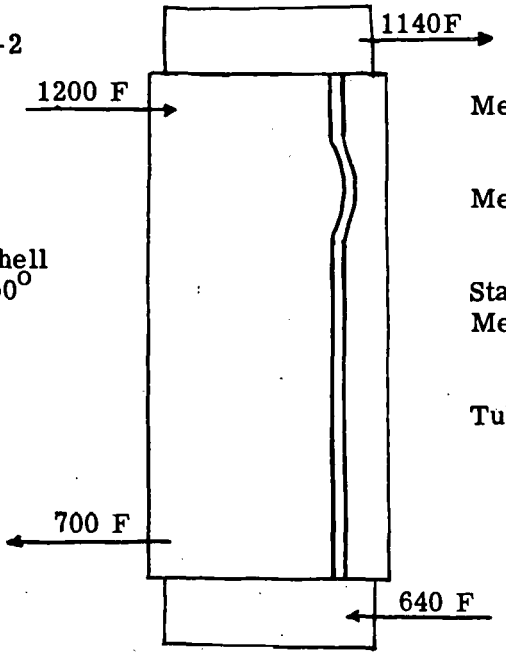
Metal Temp °F	UFPV Plate or Tubing & Forgings	Sp Membrane ≠ Discontinuities	Sp ≠ Pipe Reactions	Sy Yield Stress	Su Ultimate Strength	Thermal Stress ≠ Stress Concentrations: S_s Steady State Component	S_o Endurance Limit	Sa I 100,000	Sa II 2,500	Sa III 500
RT	18750	50% above UFPV Code	27000	30000	75000	52000	24000	24000	45000	66000
400	17500		24750	27500			"	"	"	"
600	17100		22500	25000			"	"	"	"
700	17000		21600	24000	72000	45000	"	"	"	"
750	16900									
800	16750		20700	23000	72000	43000	23000	23000	43000	63000
850	16500						"	"		
900	16000		19800	22000	70000	41500	22000	22000	41000	60000
950	15100									
1000	14000		18900	21000	68000	40000	21000	21000	39000	57000
1050	12200									
1100	10400		18500	20500	62000	37000	20000	20000	37000	54000
1150	8500									
1200	6800		18000	20000	55000	34000	18000	18000	33000	48000
1250	5300			19500		-	-	-	-	-
1300	4000		17100	19000		-	-	-	-	-

90% of Yield Strength

Pre-bent Sine Wave Tubes

Heat Exchanger Dwg. 666-1-2

48" wave 10" offset



Differential Change in Length

	Temperature Differential	Differential Elongation
Full Load Operation	$920^{o}-950^{o} = -30^{o}$	-.103"
Isothermal Hot	$1200^{o}-950^{o} = +250^{o}$	+.858"
Isothermal Cold	$640^{o}-950^{o} = -310^{o}$	-1.064"

Bending Stress in Tubes 1/2" ϕ. $S = \frac{E\delta}{480}$

	Temp.	E	Stress	Permissible
Normal Operation	920^{o}	22.2×10^{6}	-4760	59,000 psi
Isothermal Hot	1200^{o}	20.3×10^{6}	+36,300	48,000
Isothermal Cold	640^{o}	24.2×10^{6}	-53,600	66,000

Tube Sheet Pressure Stresses:

The tube sheet is treated as a composite disk, composed of a central core without tube holes, surrounded by an annular area with tube holes 1/2" dia. on 1" circular pitch, in turn attached to a solid ring, to which the shell and barrel are welded. Horvay's analysis is used for stresses and deflections in the annular area.

Stresses have been computed separately on the basis of 125 psi pressure within the tubes and channel and 100 psi pressure within the shell. The maximum stress occurred in the radial ligaments at the outside tube row under the 125 psi pressure, and amounted to 7180 psi.

967 021

These stresses are summarized on the accompanying sheet. This sheet also lists the bending and direct stresses in channel and shell adjacent to the tube sheet.

Nozzle Reaction Stresses:

The maximum stress in the shell adjacent to the primary nozzles, computed by the Bijlaard Method, for a circumferential momentum of 40,000 ft. lb., amounts to 15,800 psi. To this should be added a pressure circumferential stress of 1790 psi making an equivalent combined stress of 17,580 psi. This is slightly less than the allowable 18,000 psi at 1200 deg. F. Under the more likely assumption that the nozzle moment acts longitudinally, the expected equivalent combined stress is 7680 psi, well below the allowable.

The shell thickness has been kept at 1 3/8" for a distance of 32" beyond the center line of the nozzle, by which distance the nozzle reaction stresses will have considerably decreased.

Heat Exchanger Tube Sheets and Pipe Reactions (SS-type 316)

Summary:

Allowable stress @ 1200°F = 6800 x 1.50 = 10,200 psi
" " with pipe reactions = 18,000 psi

Barrel Pressure:	125 psi	0 psi
Shell Pressure:	0 "	100 "

Tube Sheet:

Bending stress in ligaments

Inside row of tubes	7180 psi	6440 psi
Outside row of tubes	6340 "	4660 "

Barrel Wall Adjacent to Tube Sheet:

Longitudinal Tension	1000 psi	0 psi
Circumferential Tension	2000 "	0 "
Bending Stress	7500 "	6670 "

Equivalent Combined Stress = 7500 + 1000 = 8500 psi Tension
or 7500 - 1000 + 2000 = 8500 " Compression

Shell Wall Adjacent to Tube Sheet:

Longitudinal Tension	0 psi	890 psi
Circumferential Tension	0 "	1780 "
Bending Stress	5830 "	9000 "

Equivalent Combined Stress = 9000 + 890 = 9890 psi Tension

Nozzles on Hemispherical Barrels:

Bending Stress - 4275 + 250 = 4525 psi
Pressure Stress 1030 "
5555 psi

Nozzles on Shell: ~~Sheet 14A~~

Circumferential Moment: Bending Stress - 14,550 + 1246 = 15,796 psi
Pressure Stress 1,782 "
17,578 psi

Longitudinal Moment---------- 5900 + 1780 = 7680 psi

967 022

Transient Thermal Stresses (General Statement)

The very rapid temperature changes expected in this heat exchanger indicated the need for careful study of methods of determining transient temperatures.

The finite difference method of Schmidt was first used, modified to provide for the temperature differential between the face of the plate and the body of the liquid, to obtain temperature gradients in a stainless steel plate of indefinite thickness, resulting from the specified temperature drop of 35 deg. per second for 14 seconds, from 1140 deg. to 650 deg. in a flow of sodium in contact with the plate, followed by flow at the 650 deg. temperature. A film coefficient h of 2700 BTU/hr ft^2 ^oF was chosen but it was later found that due to the method of computation used, the effective coefficient varied with time, averaging more nearly 3200. The resulting charts are included in the following pages. Within limitations, they can also be applied to determine temperatures in plates of finite thickness, using a reflection method to make corrections.

For more general problems, other charts developed by Alco Products Corporation were used. Six of these charts are included in the following pages. Three of these charts deal with the temperatures at outer face and inner face, and mean temperature, in plates of various thicknesses, at various intervals of time following a sudden change, T_f, in the temperature of the fluid flowing past the plate. The other three deal with the same three temperatures in a plate, during the period while the fluid flowing past the plate is changing in temperature at a uniform rate. The intermediate case of a uniform change followed by a dwell may be handled by approximating between these two cases.

Two of the three charts last mentioned correspond to Figures A3-5 and A3-6 in the Navy Structural Design Basis for Reactor Pressure Vessels, considerably extended and revised.

Derivations connected with these charts are given in an Alco Supplement to this report.

Transient Thermal Stress - Shell, Channel and Tube Sheet

Assume	h = 2700 BTU/hr ft^2 ^oF	Sodium
Conductivity	k = 12.1 BTU/hr ft ^oF	SS316
Diffusivity	α' = .006078 in^2/sec	SS316

Startup and Shutdown @ 100°/hr. to 1100° Total

Barrel Wall and Shell Wall near nozzles 1.375" thick.
See charts for linear change.

$$\frac{12k}{hx} = \frac{12 \times 12.1}{2700 \times 1.375} = .0391$$

$$\frac{\alpha t}{x^2} = \frac{.006078 \times 11 \times 3600}{(1.375)^2} = 127.2$$

This comes outside the range of the chart, and

$$\frac{T_o - T_{ave}}{\Delta T_f} = \frac{0.32}{N_{FO}} = \frac{0.32\,x^2}{\alpha' t}$$

$$T_o - T_{ave} = \frac{0.32 \times (1.375)^2}{.006078} \times \frac{1100}{11 \times 3600} = 2.76^o$$

UCT 023

CROLOY ($2\frac{1}{4}$% CR 1% MO) — INCHES — FOR $h = 3115$ BTU/HR.FT^2 °F

316 STAINLESS STEEL — INCHES — FOR $h = 2700$ BTU/HR.FT^2 °F

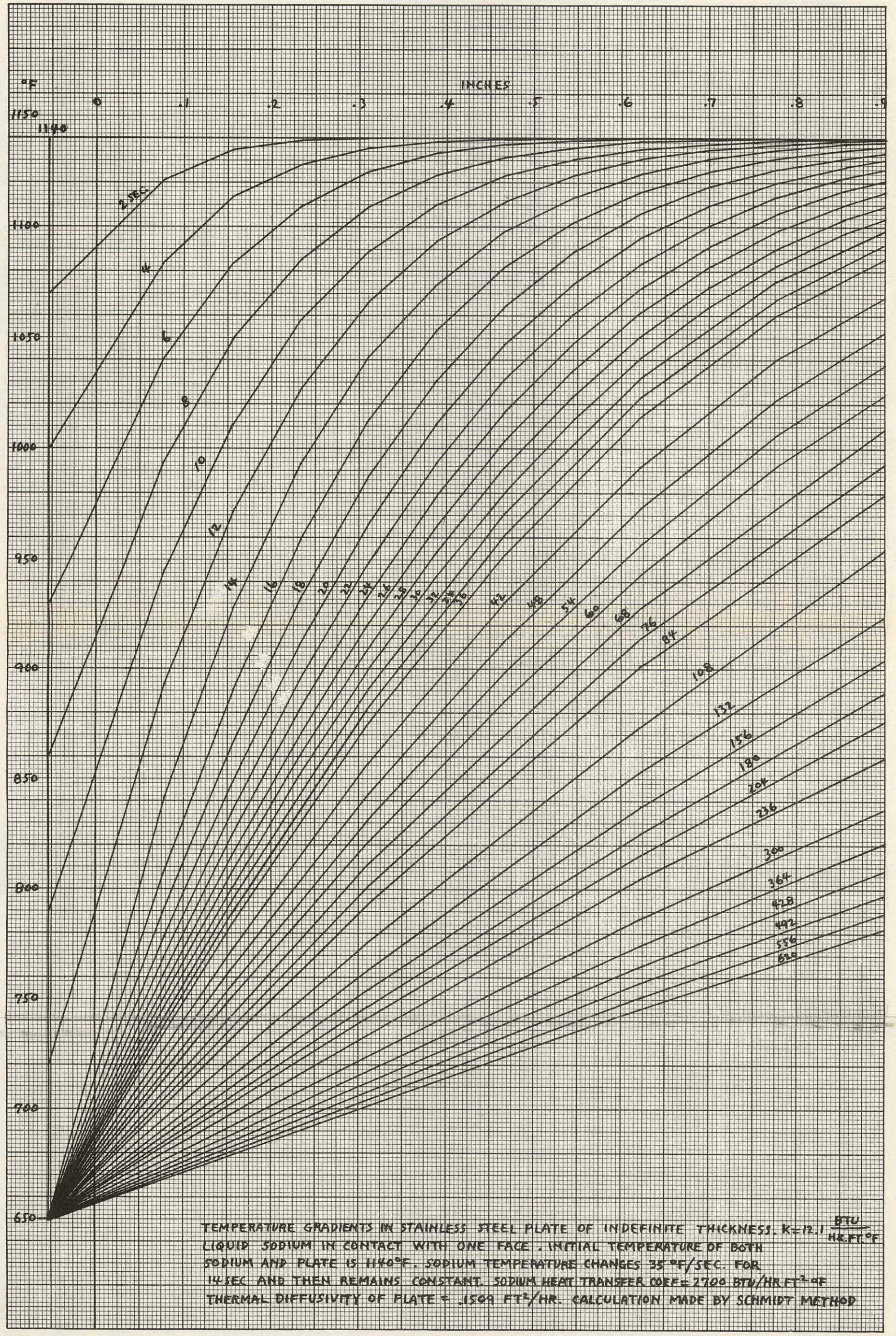
°F

1140 1100 1050 1000 950 900 850 800 750 700 650

14 SEC. 30 60 100 140 180 226 300 364 428 492 556 620

TEMPERATURE GRADIENTS IN STAINLESS STEEL PLATE OF INDEFINITE THICKNESS. $k = 12.1 \frac{BTU}{HR.FT.°F}$ LIQUID SODIUM IN CONTACT WITH ONE FACE. INITIAL TEMPERATURE OF BOTH SODIUM AND PLATE IS 1140°F. SODIUM TEMPERATURE CHANGES 35°F/SEC. FOR 14 SEC. AND THEN REMAINS CONSTANT. SODIUM HEAT TRANSFER COEF. = 2700 BTU/HR.FT^2 °F THERMAL DIFFUSIVITY OF PLATE = .1509 FT^2/HR. CALCULATION MADE BY SCHMIDT METHOD SCALE IS ADDED FOR CROLOY PLATE WITH SODIUM HEAT TRANSFER COEF = 3115 BTU/HR.FT^2 °F CROLOY K = 16.5 BTU/HR.FT.°F AND DIFFUSIVITY = .2109 FT^2/HR

720 024 967 495



TEMPERATURE GRADIENTS IN STAINLESS STEEL PLATE OF INDEFINITE THICKNESS. k=12.1 $\frac{BTU}{HR.FT.^{\circ}F}$
LIQUID SODIUM IN CONTACT WITH ONE FACE. INITIAL TEMPERATURE OF BOTH SODIUM AND PLATE IS 1140°F. SODIUM TEMPERATURE CHANGES 35 °F/SEC. FOR 14 SEC AND THEN REMAINS CONSTANT. SODIUM HEAT TRANSFER COEF = 2700 BTU/HR FT² °F
THERMAL DIFFUSIVITY OF PLATE = .1509 FT²/HR. CALCULATION MADE BY SCHMIDT METHOD

967 025

Temperature of Flat Plate with one surface in contact with Fluid and other surface insulated. Initially the Plate and Fluid are at the same temperature. The Fluid temperature is suddenly changed and held constant.

#1 OF 3 CHARTS

SUDDEN CHANGE

ΔT_f = Sudden Temperature change °F
T_o = Temperature of surface in contact with Fluid °F
T_e = Temperature of insulated surface °F
T_{ave} = Average temperature of Plate °F

t = time, seconds
X = Plate Thickness, inches
k = Conductivity Btu/hr ft °F
h = Film Coef. Btu/hr ft² °F
α = Diffusivity ft²/sec
Reciprocal of Biot Number is

$$N_{Bi}^{-1} = \frac{\text{Conductivity}}{\text{Film Coef.} \times \text{thickness}}$$

Fourier Number is

$$N_{Fo} = \frac{\text{Diffusivity} \times \text{time}}{(\text{thickness})^2}$$

$N_{Bi}^{-1} = \frac{12k}{hX}$

$(T_o - T_{ave}) / \Delta T_f$

1.0, .9, .8, .7, .6, .5, .4, .3, .2, .1

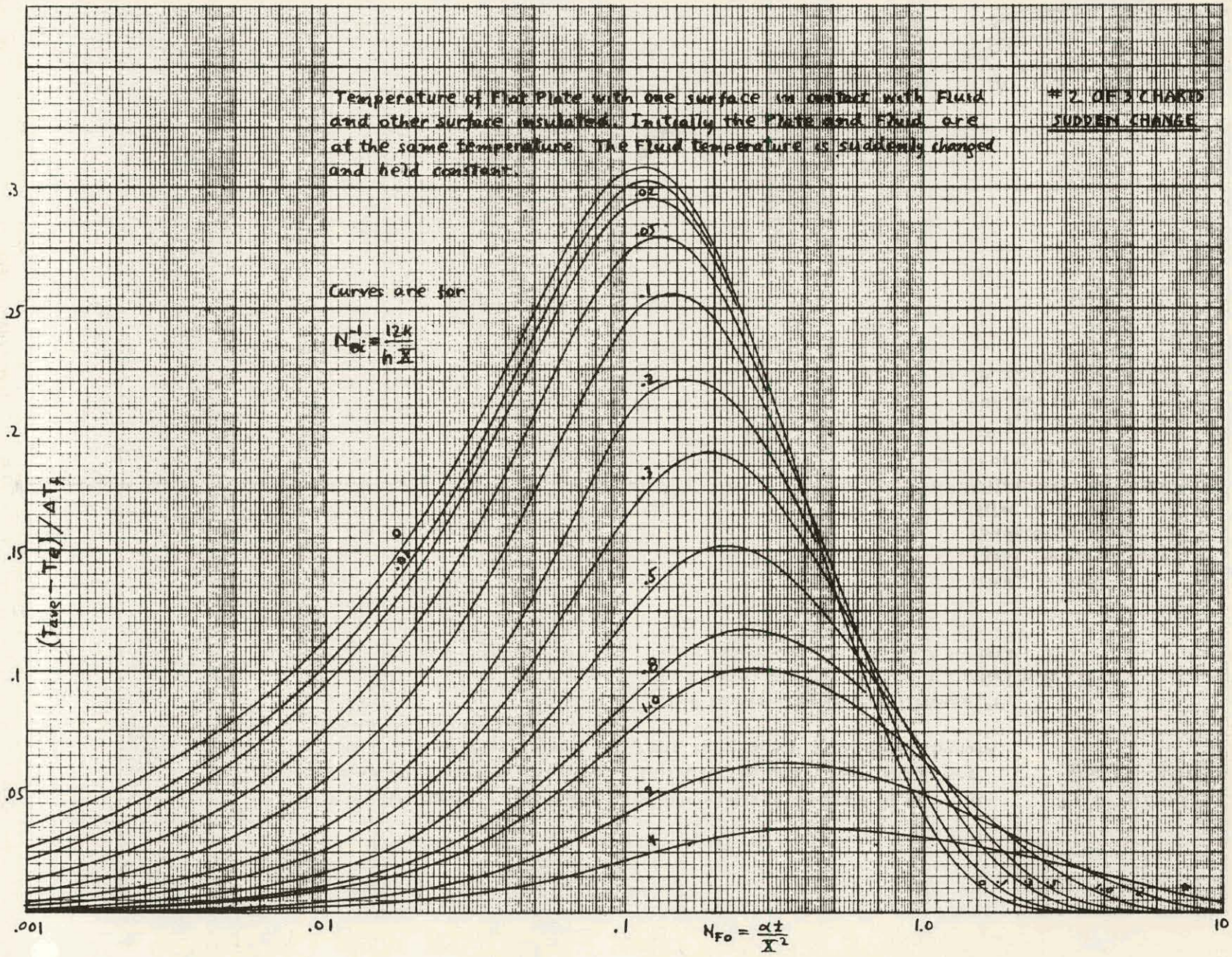
.01, .02, .05, 0, .1, .2, .3, .5, .8, 1.0, 2, 4

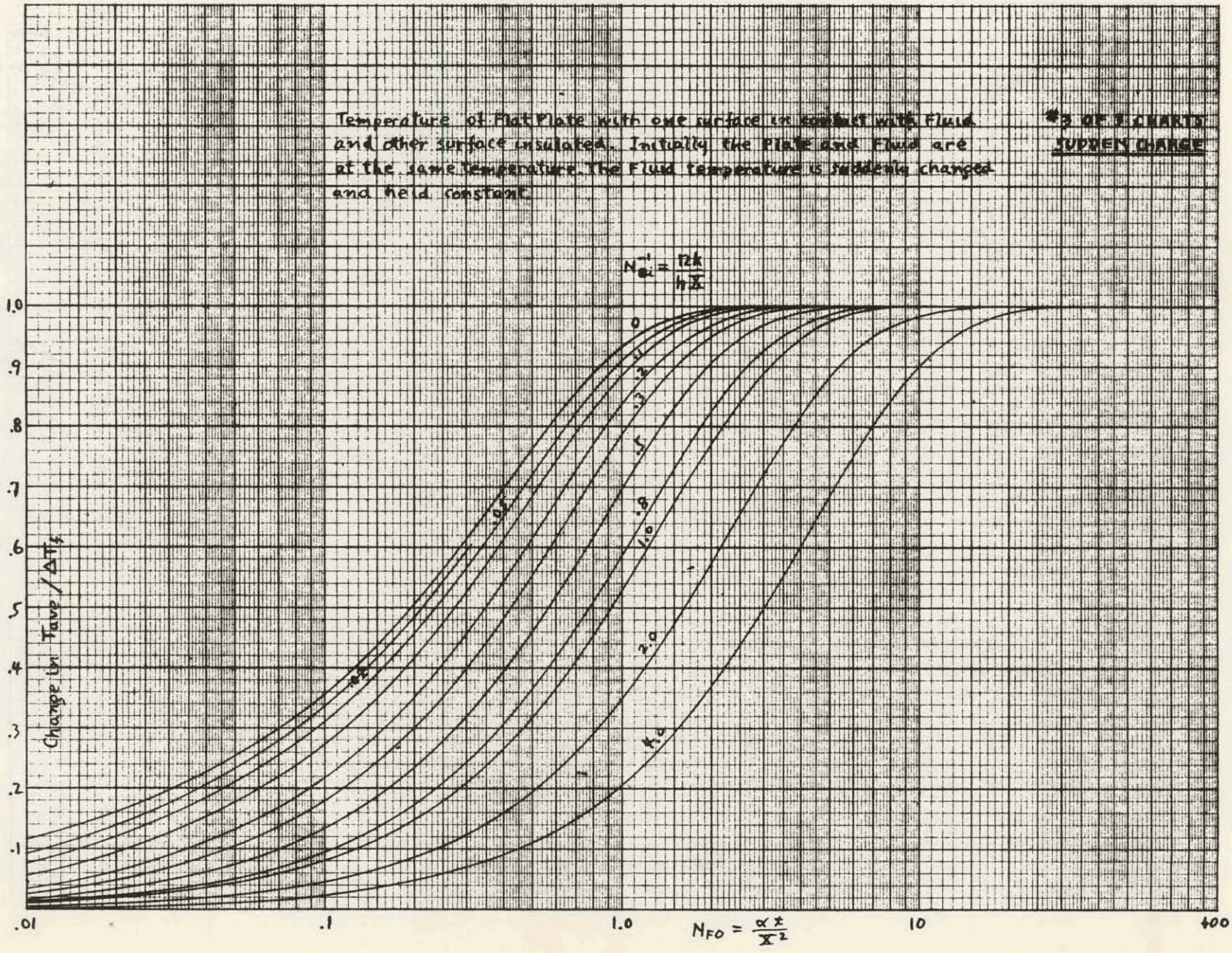
0, .1, .3, .5, 1, 2, 4

.001, .01, .1, 1.0, 10

$N_{Fo} = \frac{\alpha t}{X^2}$

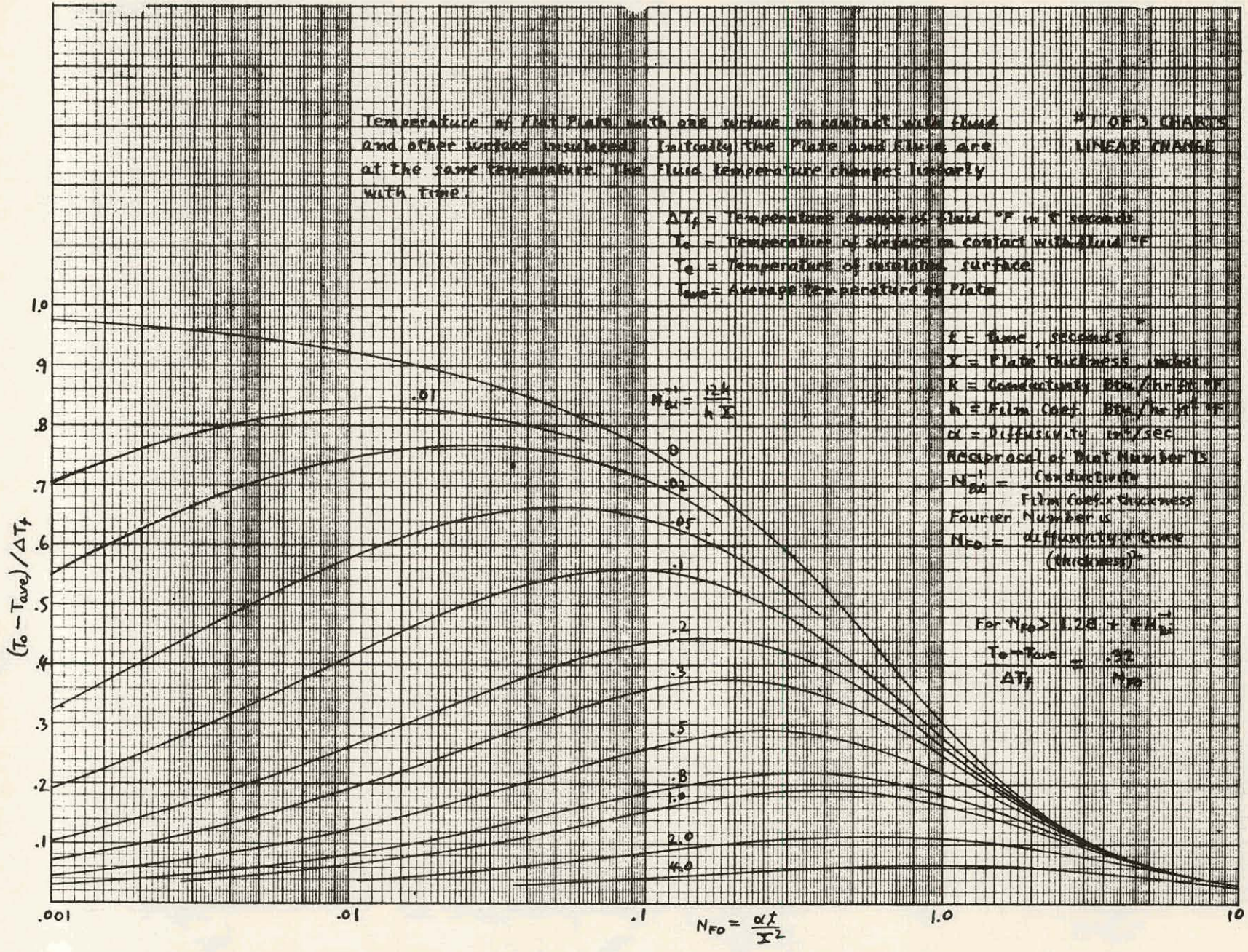
2-C9 920 496

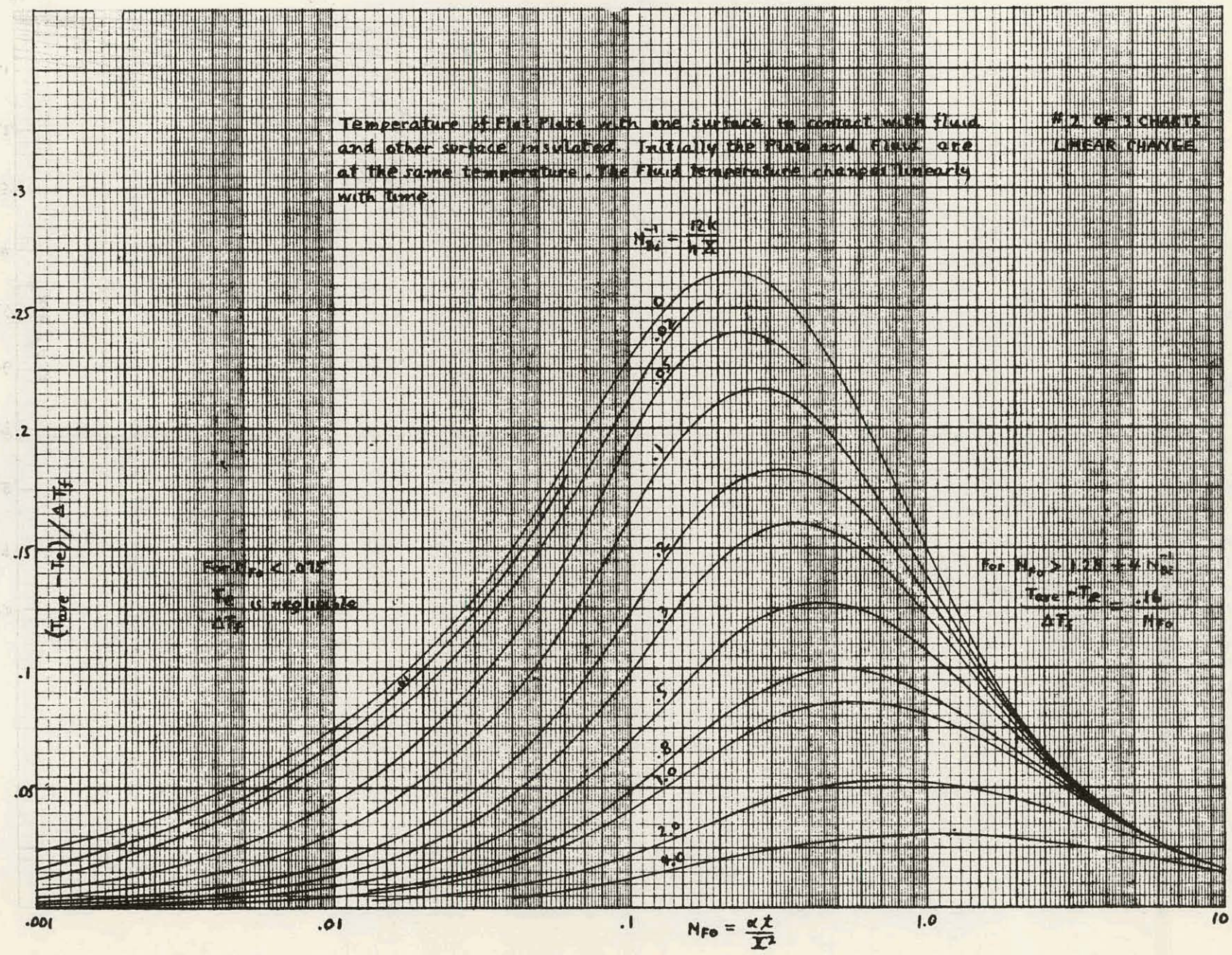

Temperature of Flat Plate with one surface in contact with Fluid and other surface insulated. Initially the Plate and Fluid are at the same temperature. The Fluid temperature is suddenly changed and held constant.
2 OF 3 CHARTS
SUDDEN CHANGE
Curves are for
$N_{Bi}^{-1} = \frac{12k}{h\frac{X}{2}}$
$(T_{ave} - T_f)/\Delta T_f$
.3
.25
.2
.15
.1
.05
.02
.05
.1
.2
.3
0
.5
.8
1.0
2
4
.001
.01
.1
1.0
10
$N_{Fo} = \frac{\alpha t}{X^2}$


Temperature of Flat Plate with one surface in contact with Fluid and other surface insulated. Initially the Plate and Fluid are at the same temperature. The Fluid temperature is suddenly changed and held constant.
#3 OF 3 CHARTS
SUDDEN CHANGE
$N_{Bi}^{-1} = \frac{12k}{h\Delta x}$
0
.1
.3
.5
.8
1.0
2.0
∞
Change in T_{ave} / ΔT_f
1.0
.9
.8
.7
.6
.5
.4
.3
.2
.1
.01
.1
1.0
10
100
$N_{FO} = \frac{\alpha t}{\Delta x^2}$

807 028

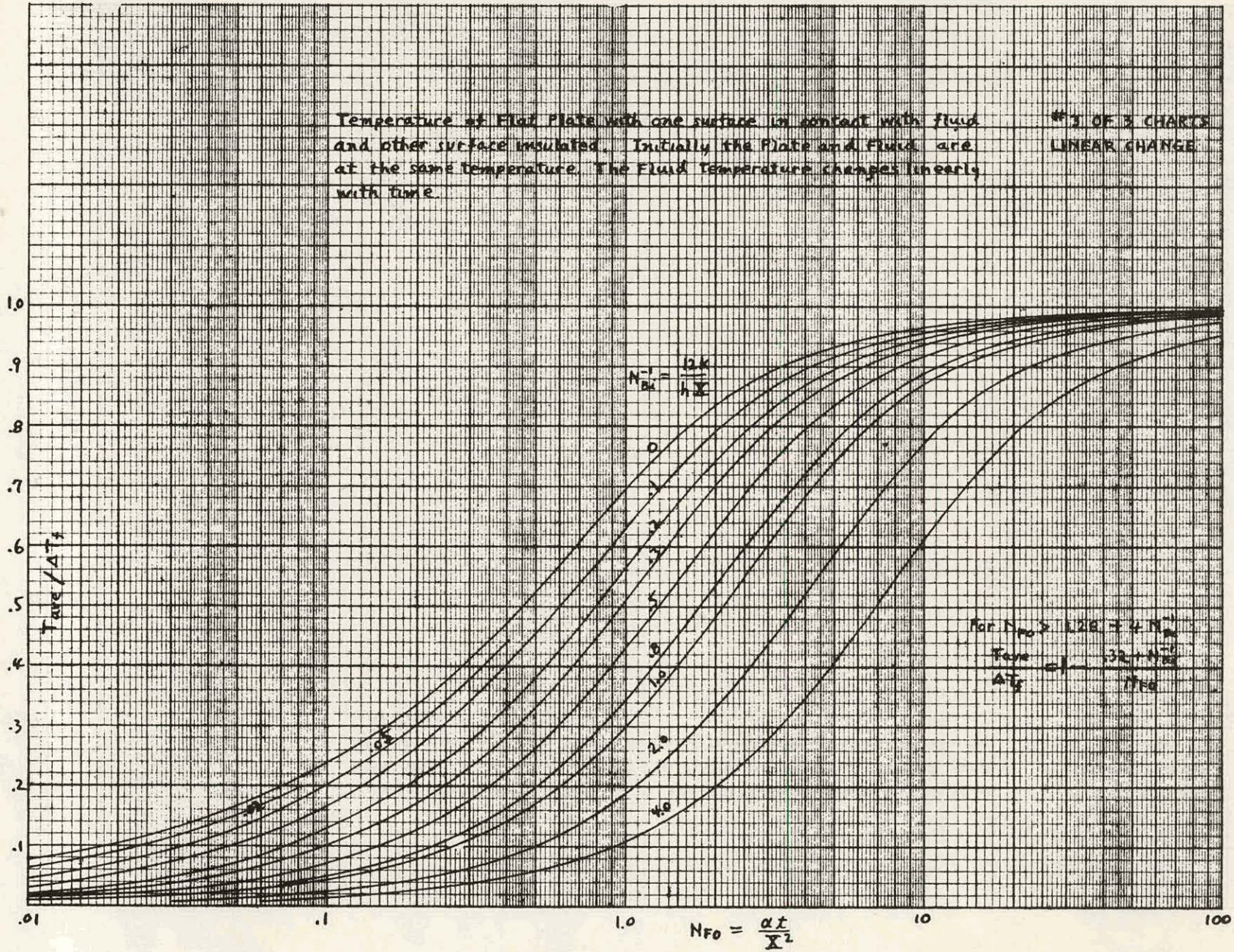
620 496


Temperature of Flat Plate with one surface in contact with fluid and other surface insulated. Initially the Plate and Fluid are at the same temperature. The Fluid temperature changes linearly with time.
1 OF 3 CHARTS
LINEAR CHANGE
ΔT_f = Temperature change of fluid °F in t seconds
T_o = Temperature of surface in contact with fluid °F
T_i = Temperature of insulated surface
T_{ave} = Average Temperature of Plate
t = time, seconds
X = Plate thickness inches
K = Conductivity Btu/hr/ft/°F
h = Film Coef. Btu/hr/ft²/°F
α = Diffusivity in²/sec
Reciprocal of Biot Number is
$N_{Bi}^{-1} = \frac{\text{Conductivity}}{\text{Film Coef.} \times \text{Thickness}}$
Fourier Number is
$N_{Fo} = \frac{\text{diffusivity} \times \text{time}}{(\text{thickness})^2}$
$N_{Bi}^{-1} = \frac{12k}{hX}$
For $N_{Fo} > 1.28 + .8 N_{Bi}^{-1}$
$\frac{T_o - T_{ave}}{\Delta T_f} = \frac{.32}{N_{Fo}}$
$(T_o - T_{ave})/\Delta T_f$
1.0
.9
.8
.7
.6
.5
.4
.3
.2
.1
.01
0
.02
.05
.1
.2
.3
.5
.8
1.0
2.0
4.0
.001
.01
.1
1.0
10
$N_{Fo} = \frac{\alpha t}{X^2}$


Temperature of Flat Plate with one surface in contact with fluid and other surface insulated. Initially the Plate and Fluid are at the same temperature. The Fluid Temperature changes linearly with time.
#2 OF 3 CHARTS
LINEAR CHANGE
$N_{Bi}^{-1} = \frac{hL}{k}$
0
.02
.05
.1
.2
.5
1.0
2.0
4.0
∞
$\frac{(T_{ave} - T_i)}{\Delta T_f}$
.3
.25
.2
.15
.1
.05
.001
.01
.1
1.0
10
$N_{Fo} = \frac{\alpha t}{L^2}$
For $N_{Fo} < .075$
$\frac{T_{ave} - T_i}{\Delta T_f}$ is negligible
For $N_{Fo} > 1.28 + 4 N_{Bi}^{-1}$
$\frac{T_{ave} - T_i}{\Delta T_f} = \frac{.16}{N_{Fo}}$

004 030

030 031 967


Temperature at Flat Plate with one surface in contact with fluid and other surface insulated. Initially the Plate and Fluid are at the same temperature. The Fluid temperature changes linearly with time.
#3 OF 3 CHARTS
LINEAR CHANGE
$N_{Bi}^{-1} = \frac{12k}{hX}$
0
.1
.2
.3
.5
.8
1.0
2.0
4.0
∞
$T_{ave}/\Delta T_i$
1.0
.9
.8
.7
.6
.5
.4
.3
.2
.1
.01
.1
1.0
10
100
$N_{Fo} = \frac{\alpha t}{X^2}$
For $N_{Fo} > 1.28 + 4 N_{Bi}^{-1}$
$\frac{T_{ave}}{\Delta T_i} = 1 - \frac{.32 + N_{Bi}^{-1}}{N_{Fo}}$

Similarly, for the Tube sheet, X = 3.50

$$T_o - T_{ave} = \frac{0.32 \times (3.5)^2}{.006078} \frac{1100}{11x3600} = 17.9^o$$

The equivalent stress in the surface of the metal: -

$$\text{Barrel Wall} = \frac{E\alpha\Delta T}{0.7} \quad \frac{240x2.76}{0.7} = 950 \text{ psi.}$$

$$\text{Tube Sheet} \quad \frac{240x17.9}{0.7} = 6160 \text{ psi.}$$

These stresses are negligible.

Effect of Rapid Isothermal Hot or Cold

Assume a change in sodium temperature of 35°F per sec. for 14 seconds. Maximum temperature differences will be present immediately at the end of this change -

Channel Wall, and Shell Wall near nozzles, 1.375" thick.

Then $\frac{12k}{hx} = \frac{12x12.1}{2700x1.375} = .0391$

$$\frac{\alpha t}{x^2} = \frac{.006078x14}{(1.375)^2} = .045$$

From charts for linear change: -

$$T_o - T_{ave} = 0.71 \times 14x35 = 348^oF$$

Surface siress, at 710°F = $\frac{E\alpha\Delta T}{0.7} \quad \frac{265x348}{0.7} = 131600$ psi.

This indicates the need for shielding.

Turning to chart for temperature gradients in stainless steel plates of indefinite thickness with this rate of temperature change -

Assume 1/8" thick shield separated by 1/8" of sodium from wall - and assume the sodium is equivalent to one quarter its thickness of steel:

	At 14 Sec.	At 30 Sec.
Surface of shield	726°F	688 F
Back of shield (.125")	893°F	779 F
Face of wall (.156")	928°F	798 F
Back of wall (1.531")	1140°F	1138 F
Mean temp. of wall	1100°F	1052 F

Mean Temperature of shield $\frac{726+893}{2} = 811.5$

Stress in shield at 14 sec. = $\frac{(811.5-726)x264}{0.7} = 32,300$ psi.

007 032

Stress in face of wall at 30 sec.

$$= \frac{(1052-798)\times 254}{0.7} = 91600 \text{ psi.}$$

$$S_{alt} = \frac{91600}{2} = 45800 \text{ psi. for } 25 \sim$$

At this instant the face of the wall will have changed by (1140-798) = 352° from its initial temperature.

Allowable stresses are:

Lower shell wall. Temp. 700°+352° = 1052°
S_a III 55500 psi.

Upper shell wall 1200°-352° = 848°
S_a III 61500 psi

This allows a satisfactory margin of safety.

Channel wall temperatures will be slightly lower, and values of Sa III slightly higher.

Nozzle stresses are dealt with elsewhere.

Tube Sheet: -

Central zone, not penetrated by tubes: - Both faces of the lower tube sheet and the upper face of the upper tube sheet, should be protected by an 1/8" shield,as for the shell and barrel walls.

Temperature changes for surface and back of shield, and for the face of the tube sheet, will be the same as those already computed. Due to its greater thickness, the mean temperature will change by say 25° instead of 40°.

$$\text{Max Stress in shield} = \frac{811.5-726}{0.7} \times 264 = 32300 \text{ psi.}$$

$$\text{Stress in face of tube sheet} = \frac{(1115-928)\times 254}{0.7} = 67800 \text{ psi.}$$

$$S_{alt} = 33900 \text{ psi. for } 25 \sim.$$

This allows an adequate margin of safety.

In the outer ring of the tube sheet, the shell side between shell wall and tubes needs an 1/8" thick shield. Stresses will be as for the central zone.

The distance between Channel wall and tube is too short to permit a shield.

Surface stresses in the tube hole zone have been computed in connection with nipple design. Under the pessimistic assumption that argon gas has displaced the sodium in all the gaps between tubes and tube sheet ligaments. The stresses developed in the tube nipples on the upper tube sheet during an isothermal cold are the most serious stresses in the heat exchanger. They should even then be satisfactory for 25 cycles. If these gaps are even partially filled with sodium, the stresses become less.

This indicates the desirability, when filling the unit, of drawing a vacuum over the top of the sodium in the heat exchanger, so as to decrease the weight of gas in these gaps. The unit should then be completely filled with sodium, which would be placed under its normal pressure of 100 psi. through argon gas in the reactor. This would fill the gaps with sodium, which would tend to be held in place by surface tension.

In normal operation, it is intended that the heat exchanger shell be completely filled with primary sodium. Argon gas below the upper tube sheet should be present only as a carry over from the reactor blanket of argon.

Under the opposite assumption that these gaps are completely filled with sodium, the tube hole zone of the tube sheets would change in temperature during an isothermal hot or cold much more rapidly than either the central zone or the outside ring. This would cause differential radial expansions and radial compressions or tensions, which would be largest at the junction between tube hole zone and outer ring. Due to the low "effective modulus" of elasticity of the tube hole zone, these stresses will not be unduly severe.

Stresses At Welds, Tube to Tube Sheets

Under stoppage of the primary pump, the secondary sodium passing through the tubes in the heat exchanger is assumed by specification to drop in temperature at 35^{o} F per second for 14 seconds, from 1140^{o} F to 650^{o} F.

Argon gas is likely to carry over from the reactor to the heat exchanger and collect in a thin layer below the upper tube, penetrating into the gaps between tube walls and tube sheet. This would insulate the tube walls from the tube sheet. The thin tube walls would therefore follow the sodium temperature very closely, while the thicker tube sheet, which is heated from the upper and lower faces only, will lag far behind. Computations given in the Appendix indicate that for this condition, for a tube wall thickness of .049" (18 gage) a stable condition would be reached after two seconds, with the wall lagging 10^{o} below the sodium temperature.

At one inch depth in the tube sheet, the change in temperature will barely begin to be felt after 14 seconds. At .05" depth, in the ligament midway between tube welds, the computed temperature at 14 seconds is 800^{o} F. In the tube, slightly below this, the temperature would be $1140^{o} - 14 \times 35 + 10 = 660^{o}$ F.

The distance center to center of tube holes will change only as the central mass of the tube sheet cools, or as it bends.

Assuming an effective tube sheet temperature of 1130^{o} F, the tube radius will have shortened with respect to this by

$$\alpha R(T_o - T_t) = 11 \times 10^{-6} (1130-660)R = 5170 \times 10^{-6} R$$

The half ligament thickness will have decreased by

$$\alpha (T_o - T_L) \left(\frac{P}{2} - R\right) = 11 \times 10^{-6} (1130-800) (P/2-R) = 3630 \times 10^{-6} (P/2-R)$$

For 1/2" tubes on 1" pitch, this will produce a mismatch of

$$(5170 \times 1/4 + 3630 \times 1/4)\ 10^{-6} = .00220"$$

967 034

Treating the tube wall as a beam on an elastic foundation with this much forced offset:

$$M_o = \frac{K}{Z\beta^2} y_o$$

Where: $K = \frac{Et}{R^2} = \frac{25\,(10)^6\,(.049)}{(.250 - .0245)^2} = 24.1 \times 10^6$

$$\beta = \frac{1.285}{\sqrt{.2255 \times .049}} = 12.24$$

$$M_o = \frac{24 \times 10^6}{2(12.24)^2} \times .00220 = 177 \text{ inch lb. per inch}$$

Computed bending stress:

$$\frac{I}{C} = \frac{.049^2}{6} = .00040 \qquad S = \frac{177}{.00040} = 442,500 \text{ psi.}$$

This exaggerates the stress. Bending will occur in the weld, producing a favorable initial angle, and the sudden contraction in radius will be eased off by a temperature gradient over the first few hundredths of an inch. The stress will however remain severe.

Three methods of relieving this stress are possible:

1. Insure that the gap between tubesheet and tube is filled with molten sodium. This will give good thermal conductivity between the tube and the body of the tube sheet so that the body of the tube sheet will cool off more rapidly. The maximum temperature differential will become more nearly 125°, dividing the stress by about three.

2. Trepan out a nipple on the lower face of the tube sheet, and weld the tube on the lower end of this nipple so that it does not enter the tube sheet at all. This reduces the stress by giving a gradually tapering connection between tube and tube sheet, and by cooling the tube sheet.

   On the other hand, the weld is difficult to make and to inspect; it cannot be reached for repair once adjacent tubes have been connected; and if a crack at the weld occurred, it would allow much more rapid leakage of fluid than if the leakage had to occur through the narrow passage between tube and tube sheet.

3. Trepan out nipples on the upper face of the tube sheet. Cut off the tube flush with the nipple, and weld upper ends of tube and nipple together.

   The upper end of this nipple will cool and decrease in diameter at the same rate as the tube. This will transfer the stress from the weld down to the lower end of the nipple.

   The resulting bending stresses in four different nipple cross sections have been studied, using a process of numerical integration. These shapes are shown on a 1 to 20 scale on graphs included in the Appendix. Values at points 0 and 1 should be ignored. The maximum stress is about the same for all four, and under pessimistic assumptions, is equal to or less than:

   2200 x 56 = 123200 psi compression.

967 035

This extends over a width of 1/20 inch, dropping off rapidly beyond this, and may occur 25 times during the working life, and at a temperature of 900-950° F. Unfortunately it is accompanied by a tension at right angles to it due to the cooling of the tube sheet face which may reach

$$E\alpha(T-T) = 250\ (1130-930) - 50000 \text{ psi tension.}$$

On the basis of shear theory of failure, this represents an equivalent stress range from 0 to 173200, equivalent to ± 86600 psi, occurring not over 25 times. This however is a compressive rather than a tensile stress. On the other face of the nipple the stress is 123200 psi tension, not additive to the 50000 psi tension, or ± 61600 psi as compared to an allowable of 66000 psi.

This pessimistic value assumes no heat transfer between tube wall and ligament or nipple, except at the weld. Actually such contact will be present, at least over part of the total area.

If this gap is filled with sodium instead of argon, the temperature differential between tube wall and ligament is smaller. The temperature distribution which could be expected at the end of the 14 seconds for tube with .049" wall and .093" wall is as follows:

	Sodium Filled Gap		Argon Filled Gap
	.049" Wall	.093" Wall	
Body of ligaments	950°F	990°F	1130° F
Tube Wall	758°	795°	660°
Difference	192°	195°	470°

The expected bending stress with the sodium filled gap will change by somewhat less than the temperature difference ratio. Nipples would still be needed on the upper tube sheet of the heat exchanger, but they will be less heavily stressed.

In the lower tube sheet, the gap between tube wall and tube sheet will necessarily be filled with sodium. An isothermal hot casualty would cause a rapid increase in sodium temperature from 650° to 1140° F, causing the tubes to expand and tend to bear against the tube walls rather than to draw away from them. The resulting stresses will be much less severe than at the upper tube sheet, but nipples are provided here also to guard against overstress.

Detailed computations for the nipples are given in the Appendix.

Heat Exchanger Sodium Nozzles

The accompanying drawing shows the shielding proposed for these nozzles, to care for stresses which could result from the specified sudden change of 490° F which could occur under casualty conditions.

Expected stresses are as follows: -
In the 3/8" thick pipe wall: -

Pressure stress	circumferential	1765 psi T
	longitudinal	882 " T

967 036

Specified Nozzle reaction
longitudinal 9260 psi T or C.

Thermal stress upper nozzle 69000 psi T
lower nozzle 65000 psi C.

Resulting values of attenuating stress:

	Upper Nozzle	Lower Nozzle
S_{alt}	43760	41760
Allowed (S_a III)	66000	53200

Steady state stress is negligible.

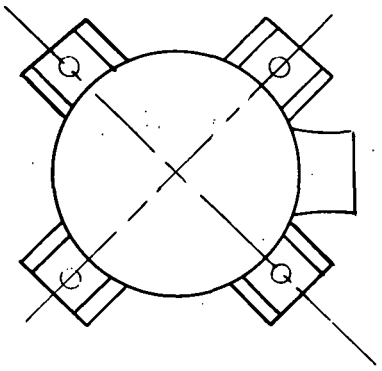
In the nozzle itself, as shielded, thermal stresses are: -

Surface of metal at E, immediately above the shield - 100000 psi T
Shield, below E - longitudinally 84900 - 31700 = 53200 psi tension.
Circumferentially 84900 x 0.3 + 31700 = 57200 psi tension.
Stresses elsewhere are lower. Detailed computations are given in Appendix F.

Load On Support Brackets

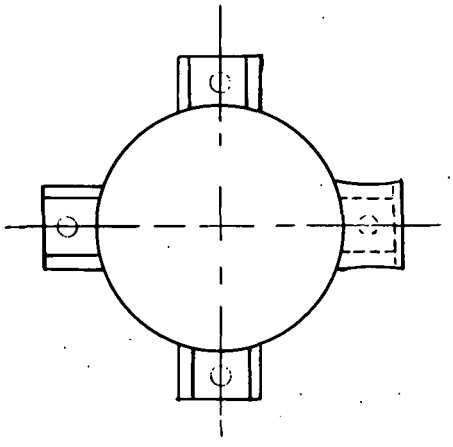
Assume for worst possible condition that the full specified values of nozzle loads, 4000 lb. of thrust and 40,000 ft. lb. of moment, act at the four nozzles simultaneously, thereby producing a total cumulative overturning moment of 4,704,000 in-lb. Then for each of the following arrangements, the load borne by the worst-loaded bracket and the corresponding bending stresses in the shell wall and in the bracket proper are as follows:

A.



	Bracket Load	Shell	Bracket
nozzles	65,400 lb.	9440 psi	7310 psi

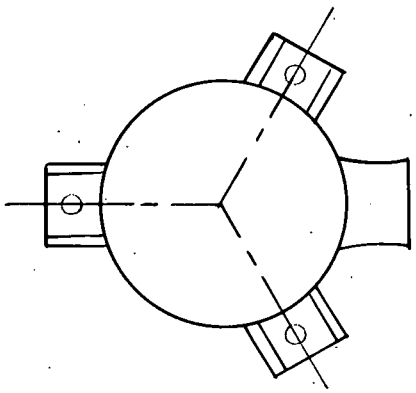
B.



	Bracket Load	Shell	Bracket
nozzles	100,200 lb.	14,470 psi	11,200 psi

037

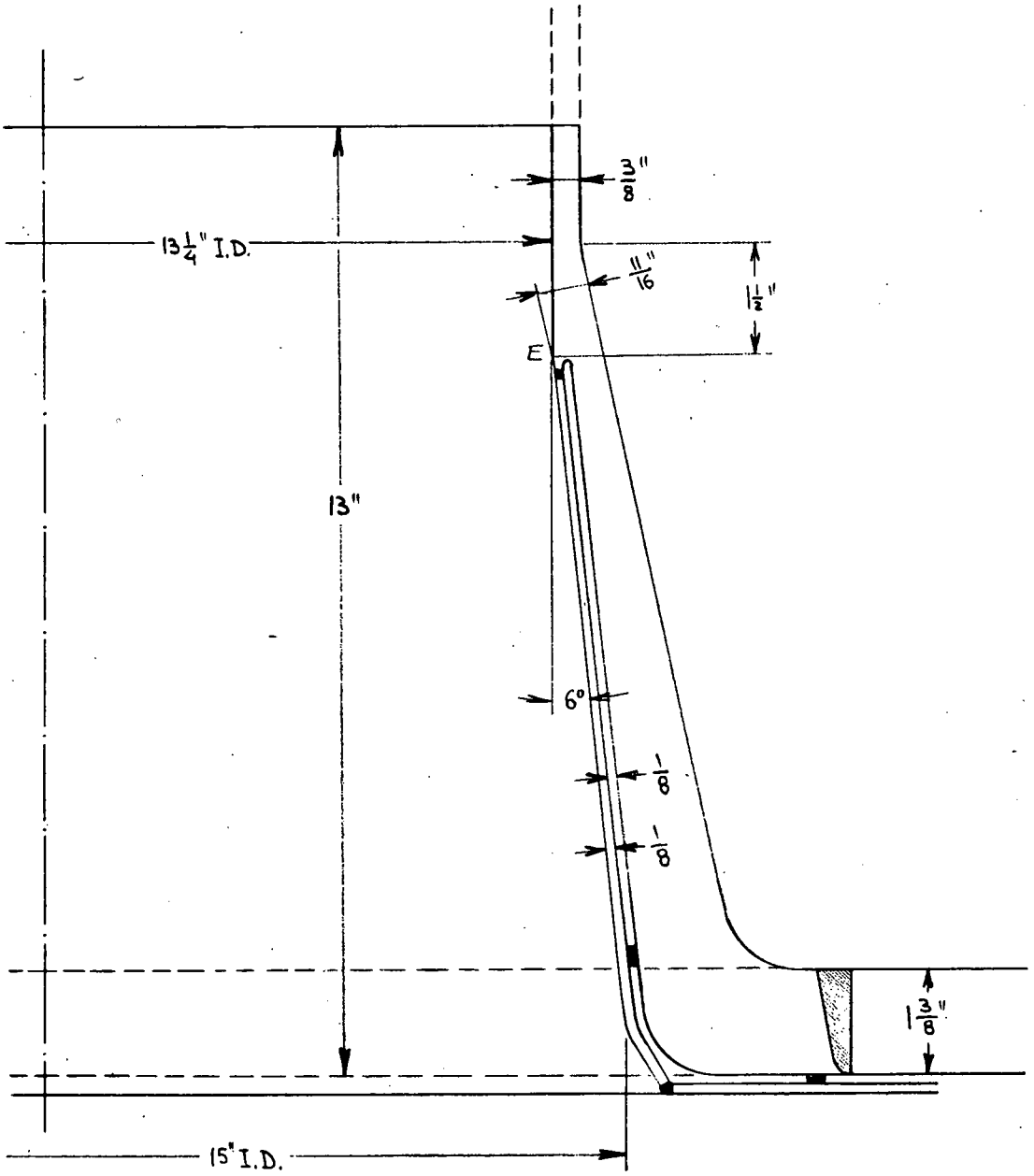
C.



	Bracket Load	Shell	Bracket
nozzles	116,100 lb.	16,780 psi	13,000 psi

967 038

Nozzle Shielding



007 039

D - CHEMICAL ANALYSIS

Consideration Of Operational Hazards And Safety Requirements For IHX Design

The intermediate heat exchanger serves to avoid the hazard of chemical reaction between water and radioactive sodium in the event of an internal boiler leak and to assure containment of all radioactivity in the reactor building. It is thus imperative to insure that there is no undetected radioactive contamination of the secondary sodium in the event of an internal IHX leak. It is recommended that the secondary coolant system be maintained at a higher pressure than the primary coolant system at all adjacent locations in the IHX to assure that any leakage in the intermediate heat exchanger will be from the secondary system to the primary system. In addition, the secondary sodium system should be continuously monitored for gamma radiation as an added precaution.

Double wire electrical systems (R-1) or an equivalent detection system should be utilized around all primary sodium piping for indication of sodium-to-air leaks. It is also advisable to consider a secondary containment around all primary system piping and vessels to avoid excessive loss of radioactive sodium in the event of a leak.

The only operating malfunction which could result in a dangerous primary system pressure rise would be a water leak in the steam generator with transmission of the pressure surge to the intermediate heat exchanger where a leak between the secondary and primary systems could result. Although this possibility is remote because of secondary system relief provisions, it is recommended that a relief device be provided on the primary system piping near the heat exchanger vessel with no valves between the relief device and vessel. This relief device should be designed to preclude any possibility of heat exchanger equipment explosions because of overpressure.

It is essential to thoroughly analyze all possible radiation hazards in the primary system design and to take all measures necessary for protection of personnel. This will include a means for remotely controlling and extinguishing any primary coolant fires that may occur within the shielded enclosure.

Sodium Cover Gas System

Argon or helium will be satisfactory as a cover gas for the primary sodium system. Nitrogen is not recommended because of the nitriding effect on stainless steel at high temperature (R-2). Nitriding will reduce the strain fatigue strength of stainless steel at high strains (R-3). Argon is recommended as the preferable choice because it has a greater density than air and will form a cover over sodium in the presence of air. This will allow internal access to equipment containing sodium with a minimum of oxygen contamination to the system. However, some radioactivity will be induced in argon and facilities for safe handling of the radioactive gas must be made available. For this reason, helium is preferred in some sodium cooled nuclear plants. All cover gas must be passed through NaK bubblers for removal of water vapor and oxygen prior to use over sodium.

The intermediate heat exchanger is designed to operate full of sodium without a gas blanket. A resistance type liquid level indicator will be mounted on the side of the vessel to measure the liquid level as it approaches the top tube sheet. Two pipe lines will join the level indicator to the heat exchanger shell below the operating liquid level so that thermal circulation will prevent freezing in the indicator. A third vent line will be located through the side of the top tube sheet to allow equalization of vessel and indicator liquid levels to that point. The level indicator will serve as an aid for filling the system and will show if the vessel is full during operation.

(R-1) References included at the end of this section.

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Epstein (R-4) reports the solubility of helium in liquid sodium to be 1.63×10^{-14} and 1.48×10^{-10} mole fraction at 450 deg. and 900 deg. F. respectively. Assuming that argon has the same behavior in sodium of greater solubility at higher temperatures, there is a possibility of solubility induced mass transfer of the cover gas from the reactor to the heat exchanger vessel. If this occurs, the cover gas will tend to collect around the top tube sheet since the primary sodium temperature is reduced in the heat exchanger. The magnitude of this effect will be determined by the temperature difference in solubilities from the cover gas location to the primary coolant exit from the heat exchanger, the sodium mass flow rate, and the degree of solubility from the cover gas in relation to incomplete mixing in the system. The quantity of gas mass transfer cannot be predicted without experimental observation because of the dependence on degree of gas-sodium contact for the sodium to become saturated.

One of the outer tubes in the heat exchanger bundle will be open to the primary sodium to allow automatic removal of any gas which collects around the top tube sheet. This tube sheet will be under cut and the tube will be tack welded to it flush with the lower surface. The bottom of the open tube will be passed into the primary sodium exit pipe line. The lower pressure drop through the tube in relation to the primary sodium path around the baffles in the heat exchanger will force a continual flow of primary sodium through this path of least resistance. Any gas collecting around the top of the tube sheet will also flow through the open tube and will be entrained in the exit sodium. The entrained gas will be carried back to the nearest cover gas where it will be trapped. The inside diameter of the tube will be altered if necessary to give the pressure drop and flow rate desired.

Vent lines will be provided from the heat exchanger shell thru the side of the top tube sheet to facilitate filling and draining operations.

Sodium Purity Control Recommendations

Sodium oxide is the most prevalent impurity in sodium systems. Excessive amounts of this impurity can be troublesome because of danger of precipitation and plugging in cooler portions of the system. In addition, this compound has a well known accelerating effect on sodium corrosion of stainless steel and other structural materials. The detrimental effects of sodium oxide can be combated successfully if the oxygen level is maintained below 50 parts per million. Such control is not uncommon in large scale sodium systems.

Carbon will react with stainless steel in high temperature sodium to give a brittle carburized case which reduces the ductility of the alloy. System sodium in contact with stainless steel should therefore be kept free of contact with carbonaceous material in areas where the carbon is likely to go into sodium solution (R-5).

Other impurities in commercially available sodium will be negligible if the sodium is filtered prior to charging the system.

Methods of Controling Oxygen

In the following discussion, "hot trapping" refers to system oxygen control by diffusion of oxygen from the sodium at high temperature into a "gettering agent" such as zirconium. A large surface area of zirconium foil is exposed to flowing sodium to promote reduction of the sodium oxide. Cold trapping refers to removal of sodium oxide from a sodium system by cooling a stream of sodium to the point where the oxide solubility is very low. The sodium oxide precipitate is trapped and allowed to accumulate in a packed vessel known as the "cold trap" (R-6).

967 041

The sodium primary system is amenable to "hot trapping" for oxygen removal because of the high operating temperatures. Zirconium hot traps are normally operated at 1200 deg. F. The IHX inlet sodium is thus under ideal conditions for hot trapping, without the expense of external heating.

It is recommended that a 170 deg. C. cold trap be included in the system for initial sodium purification. This is theoretically capable of reducing the system oxygen content to less than 20 parts per million. To prolong the effective life of the zirconium gettering material, the system should initially be cold trapped to an oxygen concentration of about 30 parts per million. The cold trap should then be closed off from the system to prevent it from becoming an oxygen source after hot trapping is started. The "hot trap" should then be used continuously to maintain the system within operating specifications.

Satisfactory purity can be attained by cold trapping only. However, the cold trap temperature should be lowered to 145 deg. C. if this is the only means of removing oxygen from the system. Although the system can be maintained within recommended oxygen specification limits by this technique, the oxygen level will be somewhat higher with a greater corrosion rate in comparison to hot trap purification.

The hot trap walls will also act as a sink for any carbon in the sodium system.

In summary, the following recommendations are made concerning primary sodium system purity.

1. Filter commercial grade sodium before charging to the system.

2. Avoid the use of materials which will introduce carbon into system sodium.

3. The system cannot be operated continuously at oxygen levels in excess of 50 parts per million without danger of excessive corrosion of system materials.

4. Removal of oxygen can be accomplished by (a) cold trapping at 170 deg. C. until the oxygen level is down to approx. 30 ppm and subsequently hot trapping continuously in a zirconium hot trap at 1200 deg. F.; or (b) continuous cold trapping at 145 deg. C. The initial method is preferable.

Analytical Procedures

A plugging indicator (R-6) is recommended for continuous monitoring of sodium system oxygen content. This consists of an orifice placed in a by-pass stream, usually in series with the cold trap. The by-pass stream is cooled until a definite flow restriction is noted on a flow meter. The oxide solubility vs. temperature curve can then be used to determine the system oxygen content at saturation temperature. The sodium oxide can be redissolved by heating the indicator sodium stream. An oxide indicator and cold trap system are described in MSA Technical Proposal No. 1098 (R-7).

Sodium samples should be taken periodically for laboratory analyses as a check of the plugging indicator and to determine impurities other than oxygen. The vacuum-distillation flame spectrometer method originated by J. R. Humphrey's of Argonne (R-8) is recommended as the best analytical technique for oxygen in sodium. In this method, a representative sodium sample is drawn into a metal-foil-lined metal receiver contained in a sampler which can be an integral part of the sodium system. The sampler can be surrounded by a lead box shield for sampling of the radioactive primary sodium. The sodium is removed by vacuum distillation and the residue for analyses can then be handled in air. The cap assembly can be removed by a simple pneumatic piston mechanism and the cup liner plus residue can then be removed and withdrawn through the shielding. This technique is potentially sensitive to a range of ± 1 ppm from zero

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contamination up (R-9). ANL has found that any unreacted sodium in the sample is distilled off and that nonvolatile materials are not carried off during distillation. Analyses at hourly intervals are entirely within the present state of the art.

The mercury amalgamation technique of Pepkowitz and Judd (R-10) and the butyl bromide method of White, Ross, and Rowan (R-11) have been more widely practiced in past applications of oxygen determination in sodium. Each of these methods is reputedly sensitive to ± 10 ppm of oxygen. They are somewhat cumbersome, time consuming, and present problems in sampling of radioactive sodium. However, consideration should be given to these tenchiques if analytical personnel employed in the plant are experienced and in accord with either technique. Silverman and Shideler (R-12) have suggested modifications in application of the butyl bromide method.

Analysis for total carbon may be accomplished by the use of the conventional Van Slyke reagent, a chromic acid oxidant. (R-13).

List of References

(R-1) Final Report Liquid Metal Technology, Project No. NR 031-364, Mine Safety Appliances Co., Callery, Pa., March 29, 1955.

(R-2) KAPL-1618, S1G/S2G Program Progress Report, pg. 89-90, August and Sept., 1956.

(R-3) KAPL-1654, S1G/S2G Program Progress Report, pg. 75, Oct. and Nov., 1956.

(R-4) Epstein, L. F., The Solubility of Helium Gas in Liquid Sodium, Report No. Memo LFE-10, Knolls Atomic Power Laboratory, Jan. 9, 1952.

(R-5) Gratton, J. G., Solubility of Carbon In Sodium at Elevated Temperatures, KAPL-1807, June 30, 1957.

(R-6) I. L. Gray, R. L. Neal, and B. G. Voorhees, Control of Oxygen in Sodium Heat Transfer Systems, Liquid Metals Technology, Part I, Chemical Engineering Progress Symposium. Series No. 20, Vol. 53, AIChE, 1957.

(R-7) Technical Proposal No. 1098, Liquid Metal Oxide Control and Indicating System, MSA Research Corp., Callery, Pa.

(R-8) J. R. Humphreys, Jr., Sampling and Analysis for Impurities in Liquid Sodium Systems, Liquid Metals Technology, Part I., Chemical Engineering Progress Symposium Series No. 20, Vol. 53, AIChE, 1957.

(R-9) Private Correspondence, J. H. Stang of Battelle Memorial Institute to L. E. Phillips of Alco Products, Inc., Sept. 25, 1958.

(R-10) L. P. Pepkowitz and W. C. Judd, Anal. Chem. 22, 1283 (1950).

(R-11) J. C. White, W. J. Ross, and R. Rowan, Jr., Anal. Chem. 26, 210-3 (1954).

(R-12) Louis Silverman and Mary Shideler, Experimental Techniques for the Determination of Oxygen in Sodium and in NaK by the Butyl Bromide Method, NAA-SR-1509, Atomics International, June 15, 1956.

(R-13) L. P. Pepkowitz and W. D. Moak, Analytical Chemistry 26, 1022 (1954).

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E - METALLURGICAL ANALYSIS

The inherent physical, chemical and mechanical properties of a material will in most cases determine its suitability for a particular application. A thorough analysis of the intended service is essential in regards to imposed stresses, temperatures and environment conditions. For specific applications such as heat exchangers, additional attention must be given to the thermal conductivity and thermal expansion characteristics of the material. Material requirements for sodium heat exchangers for nuclear reactors are particularly stringent as a failure would result in radioactive contamination, necessitating a shutdown and costly repairs.

The material requirements for the sodium heat exchanger must be compatible to the service conditions. This unit will operate at 1200 deg. F. in contact with radioactive and non-radioactive sodium. The outer shell must resist oxidation by air at 1200 deg. F. To perform satisfactorily under these conditions, a material must resist mass transfer at 1200 deg. F., have sufficient strength and oxidation resistance at this temperature and present a minimum of fabrication problems.

All of the commercially available materials were considered for construction of this equipment. The aluminum, copper and titanium alloys and noble metals are not recommended for extended sodium service at elevated temperatures, Ref. (Table 111-2, P 178 Liquid Metals Handbook, NaK supplement). A survey of the literature indicates that high nickel alloys have a tendency to mass transfer at 1200 deg. F. and above. If there is a remote possibility that these nickel base alloys would mass transfer, they are therefore not reliable. The high strength precipitation hardening iron and nickel base alloys appear to possess very desirable properties, however attendant fabrication problems, service experience in sodium and low ductility in the 1200 deg. F. range precludes their use. The low alloy ferritic steels will decarburize at 1200 deg. F. and in turn carburize austenitic stainless steels in the system in temperatures in excess of 1000 deg. F. Differences of opinion exist in regard to the effect of decarburization of ferritic steels on mechanical properties. From an overall standpoint of resistance to sodium, high temperature strength, oxidation resistance fabrication etc., the austenitic stainless steels appear to possess the most desirable properties as the construction material for this equipment.

As a result of the material evaluation study, type 316 stainless steel has been selected as the material of construction for the intermediate heat exchanger. The selection was made on the basis of its superior high temperature strength, oxidation resistance, resistance to attack by sodium and the extensive fabrication experience with this material.

Of the conventional 300 series austenetic stainless steels, type 316 has the superior high temperature strength properties and the highest allowable ASME Boiler Code stress. Although type 316 is not a stabilized grade it is believed that if this material is sensitized during service no adverse effect will result due to contact with sodium. * Results of tests have indicated that sensitized stainless steels are unaffected by sodium at 1100 deg. F. A slight strengthening of this material is even produced by sensitization possibly by the precipitation of chromium carbide within crystallographic grains.

Surface temperatures approaching 1200 deg. F. will exist on the outer shell of the heat exchanger. While this shell will be protected with an insulation blanket these high temperatures would result in excessive scaling of carbon steel. Type 316 stainless steel offers additional advantage over ferrite steels in that scaling will not become appreciable with this grade until 1600 deg. F. is exceeded. Low alloy ferritic steel will oxidize excessivily at 1100 deg. F. Therefore if ferritic steels were to be used for this application, some special protection in the form of cladding or an inert gas blanket would be necessary to lessen the scaling problem. By using type 316 stainless steel, this protection is not required.

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In order for the heat exchanger to withstand the drastic transient conditions anticipated during the operation of this unit, optimum high temperature properties are essential. Thermal stresses will also be induced during welding of the tubes to the tube sheets. Type 316 stainless steel possesses slightly superior ductility at 1200 deg. F. than type 347 and is one reason for its selection over the latter material. Sufficient ductility at 1200 deg. F. can be produced in type 316 weldments by annealing the base metal at 1900 deg. - 2300 deg. F. A similar preweld anneal of type 347 will not effect the 1200 deg. F. ductility and a post weld anneal at 1900 deg. F. is required. Room temperature ductility is also decreased in type 347 when heated to welding temperatures, however, 316 is apparently not affected. Type 347 is therefore more susceptable to structural damage and possible cracking at welding temperatures. In this regard, type 316 appears to be the more reliable material.

* The corrosion testing of various materials in sodium, R. F. Dudek and M. K. Furguson, the Babcock & Wilcox Company, Report No. 7020, April 26, 1957.

007 045

SECTION 3

OPERATION AND CONTROL

967 046

3 - OPERATION AND CONTROL

Start up and Shut Down of System

Preheat primary sodium loop and Intermediate Heat Exchanger from ambient to 300 F at maximum rate of 100 F per hour.

Fill primary loop with sodium at approximately 300 F and circulate at low flow.

Bring primary sodium flow rate up to full flow.

Start reactor and raise sodium temperature to approximately 400 F at rate of 100 F per hour.

Reheat balance of secondary loop from ambient to 300 F at maximum rate of 100 F per hour.

Fill secondary loop with sodium at approximately 300 F and circulate at low flow.

Bring secondary sodium flow rate up to full flow maintaining temperature of approximately 300 F.

Fill boiler drum to low level with boiler water. During this period steam will be generated, raising pressure. Excess steam will be dumped. Maximum temperature differential through drum wall should not exceed 150 F.

Raise reactor outlet temperature at rate of 100 F per hour by reducing primary sodium flow rate until full temperature range is reached at approximately 15% flow rate.

Raise secondary sodium outlet temperature from Intermediate Heat Exchanger by reducing sodium flow rate until full temperature range is reached at approximately 15% flow rate.

Raise pressure in steam generator so that the maximum temperature differential through drum wall does not exceed 150°F.

Adjust power and flows until full temperature and pressure conditions are met for approximately 15% flow.

Roll Turbine and pick up approximately 15% load.

Shut down procedures are the inverse of the start up procedures.

Design Criteria of Instrumentation and Control System

The sodium conponent power plant control system is to be designed for maximum safety precautions with economical plant operation. The plant shall operate at essentially constant steam pressure of 2200 psi and a minimum steam temperature of 1050°F. The primary sodium loop shall operate at maximum 1200°F and 100 psi (temperatures may go below 1200°F but must maintain steam line temperature of 1050°F), and the secondary sodium loop shall operate at 1140°F and 100 psi. The temperatures and pressure shall be maintained as constant as possible for maximum operating efficiency. The instrument control system shall provide adquate operating control of the plant, control and warning of any operating malfunctions, and safe plant shutdown in the event of an abnormal operating conditions.

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The instrumentation system for the primary and secondary sodium loop is designed to react to the requirements of each loop to satisfy economical steam load conditions. When the load requirements change at the turbine, it reacts throughout the system immediately so as to keep an economical steam generation load balance. The system control shall operate instantly on signal impulses for correction or changes in load requirements for each sodium loop and for the steam loop.

Description of Instrument System

The controls and instrumentation shall contain a controlled feedwater pump, a control system for each sodium pump and shall control the following:

1. Feedwater pump
2. Sodium pump No. 1
3. Sodium pump No. 2
4. Reactor control rods

The steam loop shall be controlled by a system of instruments and controls to control the following:

1. Steam flow, pressure and temperature compensated
2. Steam generator water level
3. Feedwater flow

The steam flow, steam generator water level and feed water flow sensing elements are connected to the feed water flow control unit. The feed water control unit determines the feedwater system requirements as the generator load varies in order to maintain water level and maintain steam generation. The heat requirements to the steam generator, due to generator load variation, are transmitted to the sodium pump No. 2 controller.

The sodium pump No. 2 controller receives signal inputs of actual temperature from the steam main, actual temperature from the secondary sodium loop, actual and set temperature from the sodium pipe between boiler and superheater, biased temperature set and pressure from the steam main. The controller for the secondary sodium loop will maintain sodium pump speed, sodium flow, and sodium temperature requirements to satisfy the steam generator load demand. The heat load requirements of the secondary sodium loop are transmitted to the primary sodium loop pump speed controller.

The primary sodium loop pump controller receives signal inputs of advanced ΔT temperature requirements from the secondary sodium loop and also flow in the secondary sodium loop. The controller for the primary sodium loop will maintain the primary sodium pump speed to satisfy the heat requirements of the secondary sodium loop.

The reactor ΔT temperature determines the maximum reactor discharge temperature at maximum flow for the maximum reactor discharge temperature at maximum flow for the maximum reactor power output. Negative temperature coefficient should be allowed to control the reactor power output if possible.

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Instrumentation Safety Control Features

The two sodium loops and the steam loop shall have safety features which will be actuated in event of any malfunctions because of the reaction of sodium in contact with moisture. If the control rod system switches to a scram position because of malfunction; the turbine shall operate at "no-load" to expend the steam, the two sodium loops shall circulate long enough to remove the decay heat and upon removal of the decay heat, circulation will cease and valves close to prevent thermal circulation. Should there be a large leak between the sodium and water-steam systems, the emergency scram control shall scram the rods, and trigger the safety valve to dump the steam system immediately. The steam generator is isolated by closing valves and the primary system allowed to circulate to remove decay heat. The feed water pump system is stopped to prevent additional water from entering the steam generator, and the valve on the feed water line is closed. After the steam in the generator has decreased in pressure, and decay heat removed, the secondary sodium system is manually drained to a holding tank for storage. The holding tank will have a blanket of Argon gas and will be at atmospheric pressure. A drain valve is located in the primary sodium loop with piping to a separate radioactive sodium storage tank.

In the event of a small leak between the sodium and water-steam systems, the rise in gas pressure will sound an alarm. The high and low sodium level indicatores located on the steam generator, heat exchanger, and in the sodium pump-chamber systems will sound an alarm in the event of any malfunctions. The reactor will be manually shut down, the turbine operated at "no-load" to expend the steam while the sodium pumps are operated to remove the decay heat, after which circulation will cease and valves will close to prevent thermal circulation. A rupture disc is located in the superheater and in the boiler section of the steam generator to relieve extreme pressures in the shell side. In the event of a disc rupture, the sodium is piped to a holding tank with an Argon gas blanket.

In the boiler section sphere is located a safety valve and an automatic pressure relief valve; the safety valve is to dump the steam in an emergency, being actuated by automatic emergency control and the automatic relief valve is a must to meet boiler safety requirements. An alarm is sounded in the area immediately when an emergency scram is announced.

Primary and secondary sodium piping will be wound with double wire electrical systems which will alarm a sodium to air leak. The secondary sodium system will be continuously monitored for gamma radiation as a precaution to guard against contamination from a radioactive sodium leak. The secondary system will be maintained at a higher pressure than the primary to prevent inleakage of primary radioactive sodium in the event of an intermediate heat exchanger leak. The leak will be detected by a drop in pressure and liquid level on the secondary side and a rise in primary pressure and liquid levels.

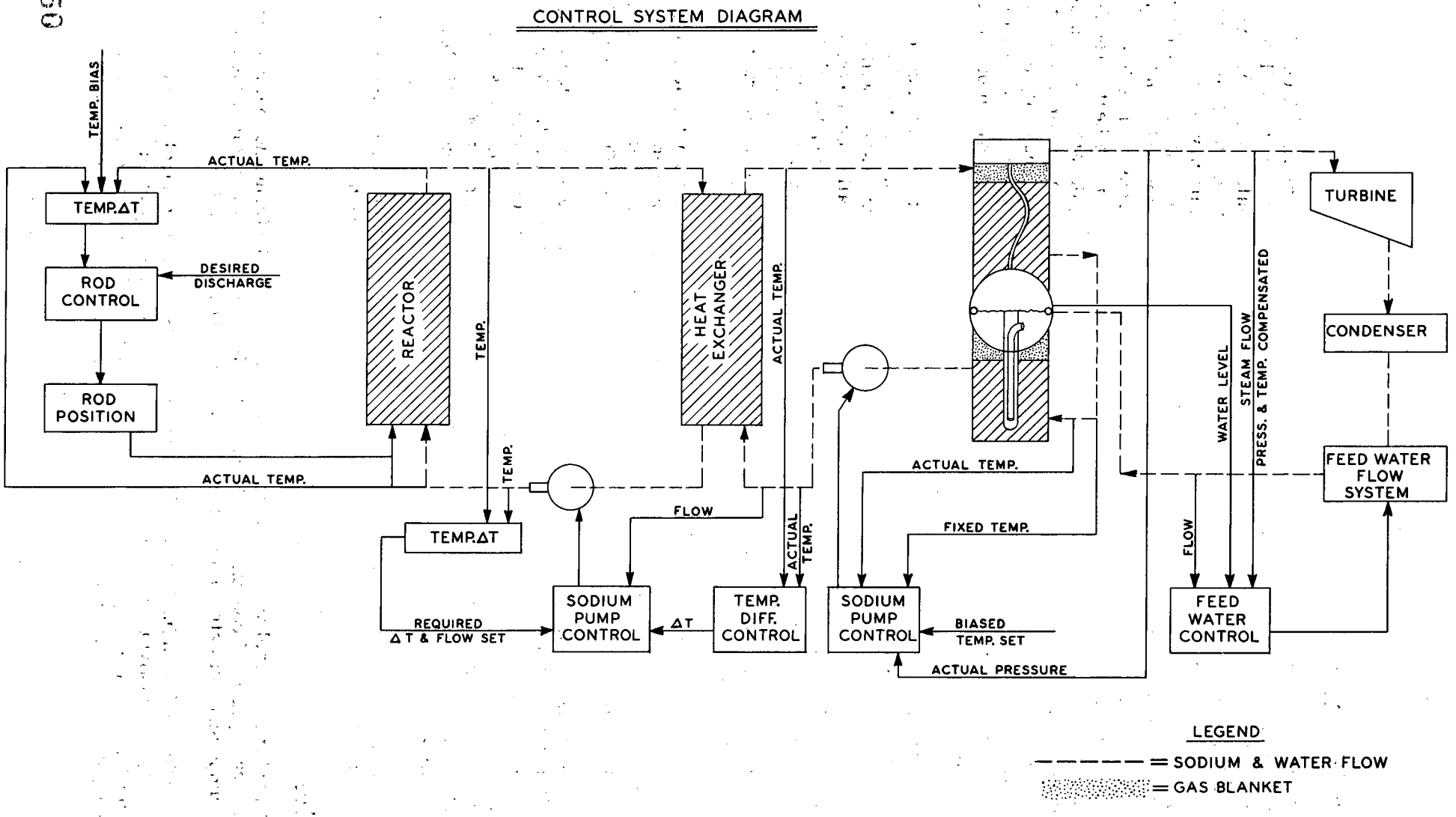
Liquid relief valves will be located in primary and secondary sodium piping systems between the intermediate heat exchanger and the nearest valve on each loop.

SUMMARY

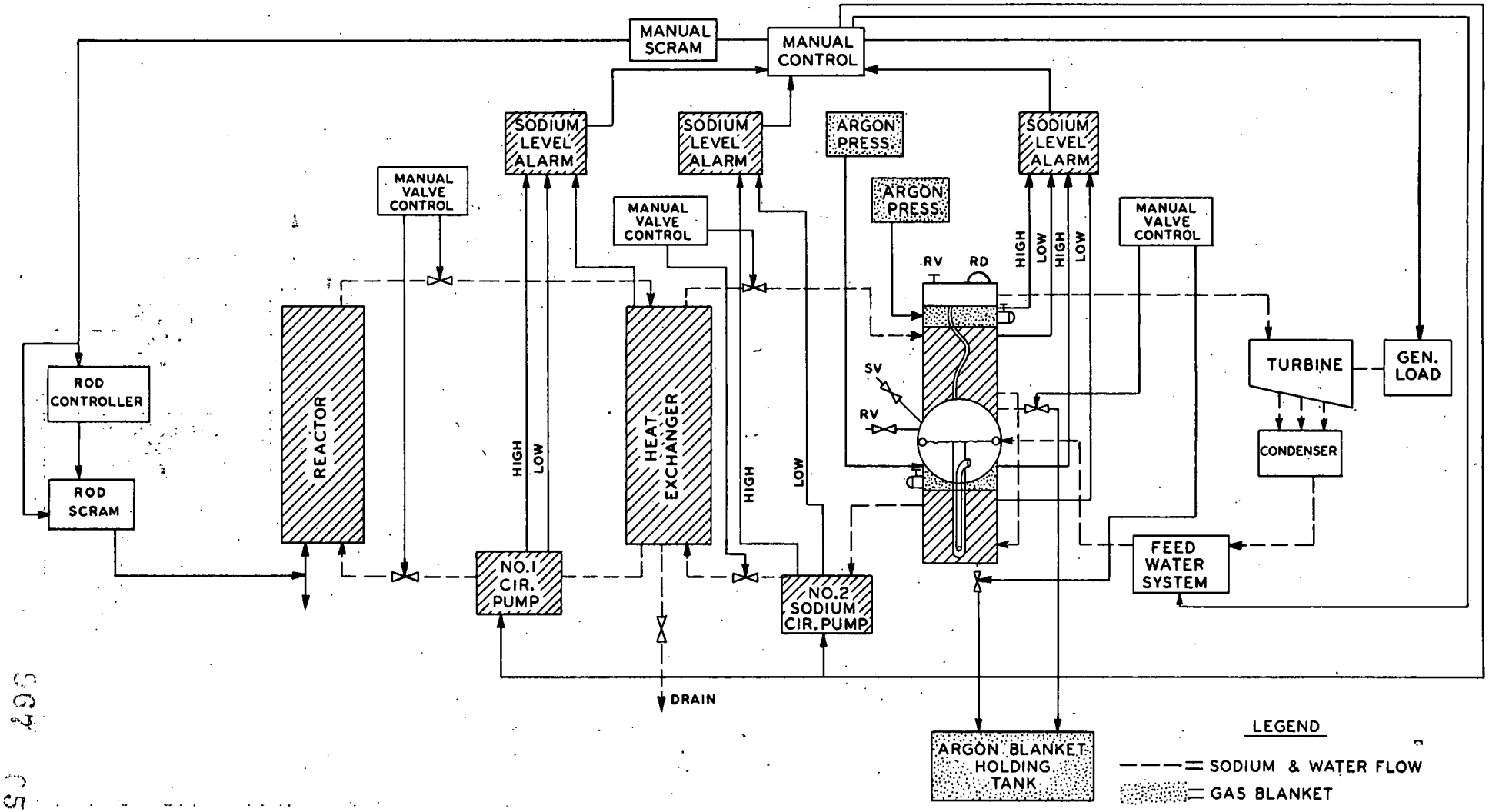
A preliminary analysis indicates these controls can be instruments that are readily available for power plants of conventional design. The instrument most readily adaptable for control of pumps and feedwater is the three element feedwater controller.

In order to determine time and load requirements for each control system, a complete study and system analysis must be made. After a complete study is made, a control system with immediate control response must be incorporated in the design so the required pressures and temperatures are maintained for maximum operating efficiency and safety.

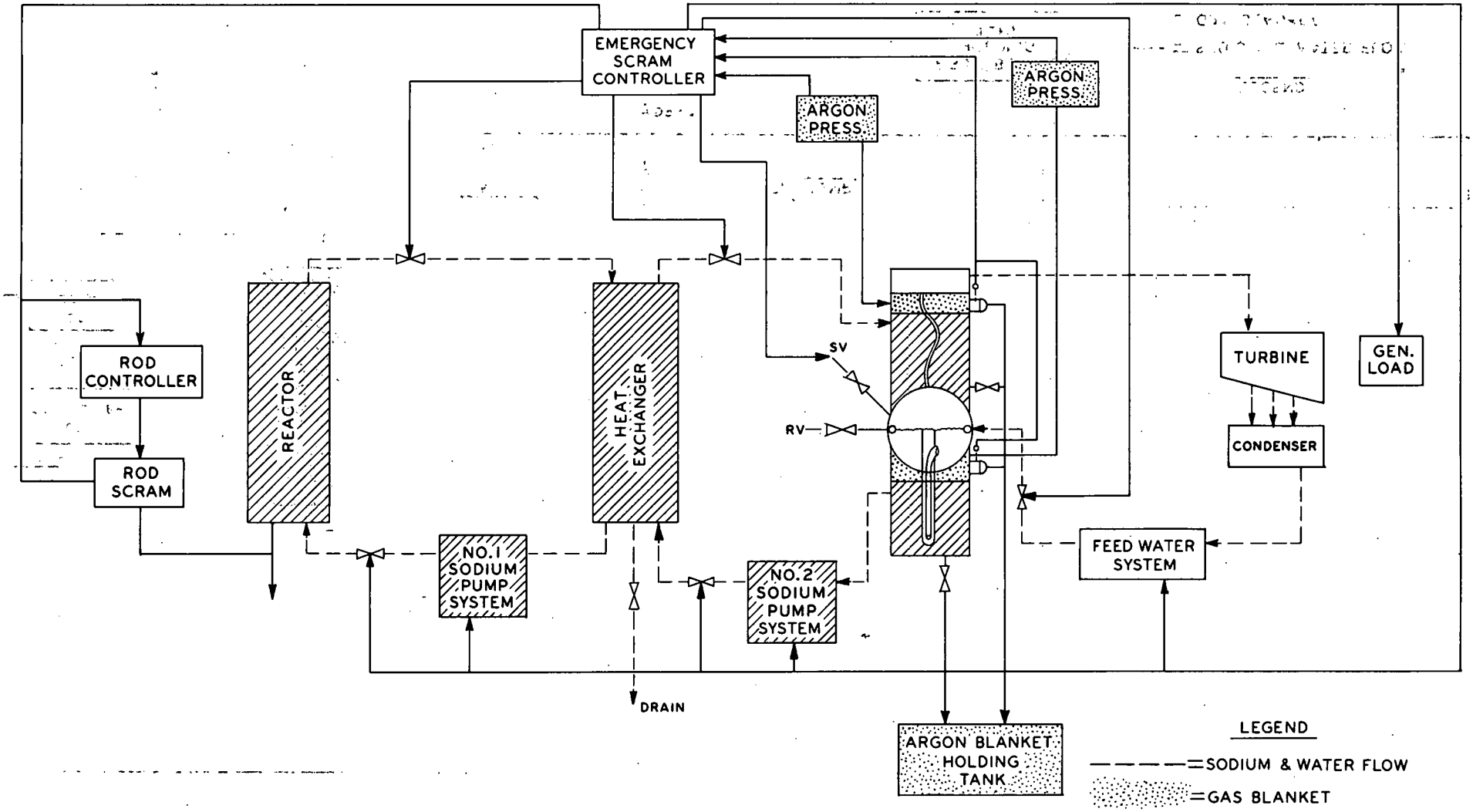
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CONTROL SYSTEM DIAGRAM
TEMP. BIAS
ACTUAL TEMP.
TEMP.ΔT
ROD CONTROL
DESIRED DISCHARGE
ROD POSITION
REACTOR
HEAT EXCHANGER
TURBINE
CONDENSER
FEED WATER FLOW SYSTEM
WATER LEVEL
STEAM FLOW
PRESS. & TEMP. COMPENSATED
TEMP.
ACTUAL TEMP.
FLOW
FIXED TEMP.
REQUIRED ΔT & FLOW SET
SODIUM PUMP CONTROL
ΔT
TEMP. DIFF. CONTROL
BIASED TEMP. SET
ACTUAL PRESSURE
FEED WATER CONTROL
LEGEND
= SODIUM & WATER FLOW
= GAS BLANKET

SAFETY CONTROL SYSTEM FOR SMALL LEAK



SAFETY CONTROL SYSTEM FOR LARGE LEAK



SECTION 4

INSPECTION AND MAINTENANCE

4 - INSPECTION AND MAINTENANCE

Inspection

The intermediate heat exchanger is of all welded design, and the primary and secondary operating fluids will be liquid sodium. Since the unit is of all stainless steel construction, corrosion in a liquid sodium atmosphere is considered negligible. On this basis, inspection periods will not be scheduled periodically or on estimated corrosion rates. Inspection of this unit is recommended only after there is evidence of a leak or an indication of mixing of primary and secondary fluids. Subsequent inspection periods will be based similarly on leaks or other evidence of faulty operation of the unit. While the unit is open for repairs, the following inspections shall be made:

A. All tubeside surfaces of the tubesheets, including tube welds and nipples, shall be visually inspected for cracks or other defects. The shielding within the channel and vent and drain openings should also be inspected.

B. External inspection will include vessel supports, insulation and the general condition of the vessel.

C. Inspection is not required or considered necessary on the shell side of this unit.

Repairs required as a result of the inspection shall be of first quality workmanship and,where necessary, according to acceptable practices under Section VIII, of the ASME Boiler and Pressure Vessel Code. Complete records of inspection and repairs shall be filed as permanent records.

Maintenance

In the event there is evidence of a leak between the primary and secondary fluids, the unit shall be completely drained and isolated. After draining, the unit shall be steam cleaned and flushed with high purity water. When the unit is accessible, the following procedures shall be followed:

1. Remove upper channel manway cover nuts and open cover.

2. Using cutting tool, cut seal weld and remove seal.

3. Pressurize the shell side of the unit with an inert gas, and check for the leak by sound, feel, or bubble test. Pressure not to exceed 150 psig.

4. If gas is escaping through a tube, the leak may be in a tube between tubesheets, or at the tube connections in the lower tubesheet. To determine the exact location of the leak in this instance it will be necessary to remove the lower manway cover and seal. After location of the leak, release gas pressure on the shell side.

5. If the leak is at a tube weld, repair the weld as follows:

   a. Coverings shall be used to protect surrounding area from foreign matter.

   b. Using a tube end cutter, remove the existing weld.

   c. Reweld tube to nipple with a two pass weld, using inert gas shielded tungsten arc process with filler rod addition. Minimum depth of weld to be 1.5 T (T = tube wall thickness).

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d. After the reweld is completed, check with dye-penetrant test, then proceed with a bubble test by pressurizing the shell side with inert gas. Test pressure to be 150 psig.

e. After test is completed, clean dye test material from weld and remove coverings. Using a vacuum cleaner with soft brush on hose, clean entire channel surface. Remove lower manway cover and seal (if not already removed) and run a lint-free cloth through the repaired tube. After the tube is clean, vacuum lower channel area and close unit. If the seal diaphragm is in good condition, it can be rewelded into place. Install new manway cover gaskets and bolt on covers.

f. Completely dry and purge unit, then refill.

6. If a defective tube is found, it will be necessary to plug the tube at the upper and lower tubesheet weld as follows:

a. Coverings shall be used to protect surrounding area from foreign matter.

b. Drill the tube so as to reduce wall thickness to approximately .015 inch to a point below the tubesheet. Using a puncture tool, punture the tube just inside the upper and lower tubesheet surface on the shell side.

Then drill out the entire tube for a distance of approximately 1-1/2 inches below the tubesheet weld so that the plug will fit tightly in the tubesheet. The plug length should correspond to tubesheet thickness, in order to eliminate a sodium hold up pocket in the lower tubesheet. Drive fit a stepped, stainless steel plug (Type 316) into upper and lower tube ends and weld.

c. Plugs shall be welded using the inert gas shielded tungsten arc process with filler rod addition. Two weld passes shall be required.

d. Check weld with dye-penetrant test, then make a bubble test by pressurizing the shell side with inert gas. Test pressure to be 150 psig.

e. After test is completed, clean dye test material from weld and remove coverings. Using a vacuum cleaner with soft brush on hose, clean entire upper and lower channel area.

f. Close unit, purge and refill.

7. In the event of a defect in a welded joint or shell material, the defect will be repaired in accordance with Section VIII of the ASME Boiler and Pressure Vessel Code, Paragraph UA-821.

037 055

TOOLS AND ACCESSORIES

1. Wrenches for manway cover nuts.
2. Manway cover gasket (3 spares each manway).
3. Cutting rool for manway seals.
4. Tube end cutter.
5. Vacuum cleaner with hose and soft brush.
6. Welder:

   Recommended: AC-DC rectifier, with high frequency unit and current slope control, 300 amp. capacity.
7. Electric drill — 1/2 inch chuck.
8. 15/32 inch drill, with 12 inch extension.
9. 1/2 inch tube puncture tool.
10. Manway cover studs (10% extra for spares)
11. 9/16" drill.

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SECTION 5

PROPOSED RESEARCH AND DEVELOPMENT

967 057

5 - PROPOSED RESEARCH AND DEVELOPMENT

General Approach and Recommendations

In our effort to contribute to the prime objectives of the Commission's sodium component development program as a whole, we recommend a series of tests in the mechanical field, followed by testing of models designed for 10% full heat load of full size units, in order to develop data in thermal, operational and control areas. This we feel will provide maximum benefit at minimum cost and insure dependability of the large units. This approach, as against building and testing full scale models, will fall short primarily in developing the elements of economy inherent in full size units. While it is generally felt that heat transfer, fluid flow and distribution data, permissable of accurate extrapolation to full size units cannot all be obtained from small test models, much pertinent and valuable information can be so obtained.

The recommended program includes tests of mechanical components and a comprehensive operational test of models designed for 10% of full heat load. The test models would duplicate as many features of the submitted design as possible, such as size and thickness of tubes, so that the relative resistances will be maintained as closely as possible to those of original design values. The test models are not scaled down in all physical dimensions, but are small units made up of a lesser number of similar components. In this way it is felt that the heat transfer data obtained would be quite reliable, but because of the size effect it would not be possible to obtain fluid flow, distribution and pressure drop data susceptible of accurate extrapolation.

Of vital importance to the final design are problems associated with operation and control. A very important consideration is that of performance of the boiler section of the steam generator. By maintaining the same size tubes and reducing the steam drum to correspond to the quantity of steam generated for 10% full load it will be possible to accurately check operation of the bayonet tube and steam drum internals. These data will include the recirculation rate in the bayonet tube as well as the preheating capacity of the contacting tray and the quality of steam leaving the steam drum. In conjunction with the steam generator test, certain water problems such as chemical carryover and deposition can be checked. The model test will also provide much information with regard to control, instrumentation and response as well as operational limits and procedures.

Even though the size effect will influence operational characteristics, model test data will be very helpful in the final design considerations of full units.

Full advantage will be made of Alco's experience in analogue simulations of thermal systems for response to transient and steady state conditions, such as are currently being performed for the design of APPR-1b.

In order to obtain maximum correlation between model test and analogue simulation data with actual performance data of full size units, test data should be obtained from such full size units when built, operating at full flow conditions and at full load or at least at 50% load.

Mechanical Tests

Consideration was given to the sequence of the following tasks. It is recommended that the welding development program be performed prior to the mechanical and stress tests of the tube and tubesheets. In this way, the welding procedures can be developed prior to the fabrication of subsequent test units.

LDT 058

Welding Development

1. Inconel Overlay

   a. The objective of this work is to develop the procedure for overlaying Inconel weld weld metal to type 316 stainless steel on all surfaces exposed to steam in the superheater section of the steam generator.

   b. The design of the superheater will require risers to be machined on the steam side of the tube sheets at the tube weld location. The 316 stainless steel tube sheets must therefore be overlayed with Inconel to a maximum depth of 1-3/8" and the trepanned risers machined from the overlayed deposit.

   c. A procedure for overlaying 1-3/8" of Inconel on a flat type 316 stainless steel plate will be developed, using either the inert gas shield arc or the submerged arc processes whichever is most suitable. This procedure will then be extended and modified, if necessary, for overlaying the concave surface of a type 316 stainless steel 22" radius hemisphere.

   d. The following laboratory tests will be conducted on samples taken from the flat test plate:

      1. Macroscopic examination of the overlay cross section to determine the depth of deposit, porosity, defects or cracks, etc.

      2. Microstructure of the overlay will be examined at 100X and 500X.

      3. Chemical analysis of overlay to determine extent of dilution of the Inconel into the base metal.

      4. Side and face bend tests.

      5. Other tests that may be necessary to determine quality of the overlay.

2. Welding of Bi-metallic Tubes to Inconel Trepanned Overlay

   a. The objective of this work is to develop the procedures for welding type 316 stainless steel and Inconel bimetallic tubes to the trepanned Inconel overlayed tube sheet.

   b. This will be accomplished by a manual welding process with Inconel filler metal addition. Possibly, an initial automatic inert gas-shielded tungsten arc weld pass will be used to obtain a seal weld between the type 316 and outer portion of the tube and the Inconel overlay. If the latter process is used, a second pass of Inconel filler metal will be required to cover all stainless steel portions of the tubes.

   c. The welds will be evaluated by the following laboratory tests:

      1. Tensile pull tests to determine strength of weld joint.

      2. Several welds will be sectioned through the diameter of the tube and examined metallographically. A macro-examination of the cross section will reveal weld thickness, leak paths, defects or cracks, macroporosity, etc. The microstructure will be examined at magnifications of 100X and 500X.

3. Welding 2-1/4% Cr Steel Tubes to 2 1/4% Cr Steel Tubesheets

a. The objective of this work is to develop the procedures for welding 2-1/4% Cr Steel tubes to 2-1/4% Cr. Steel trepanned tube sheets.

b. The tube to tubesheet joints will be welded, using the manual metal arc process or other suitable manual or semi-automatic processes. In order to develop a suitable procedure, it will be necessary to establish preheat and post-heat times and temperatures and convenient and practical means of applying this heating. These heat treatments will be necessary to eliminate any undesirable microstructures in the tubes and heat affected areas that will be deleterious to the weld joint.

c. Laboratory testing in conjunction with this study will include the following:

1. Tensile pull tests to determine strength of the weld joint.

2. Macroexamination of cross sections of the weld joint will be examined for porosity, defects or cracks, weld thickness, length of leak paths, etc.

3. Microstructure will be examined at 100X and 500X for martensite or other transformation products.

4. Hardness measurements of weld areas and heat affected zones.

4. 2-1/4% Cr Steel to Stainless Steel, Transition Weld

a. The object of this work is to develop a procedure for welding a 5" thick 54" diameter circular section of 2-1/4% Cr. steel to a similar section of type 316 stainless steel.

b. The following work will be conducted to satisfy the objectives of this investigation:

1. Overlay of a transition metal on the 2-1/4% Cr steel.

a) Investigate automatic deposition procedures for applying type 316 overlay on the 2-1/4% Cr. steel.

b) Investigate manual welding overlay of Inconel (Inco A) or other alloys to the 2-1/4% Cr steel followed by type 316.

c) Evaluate the influence of preheating and post-heating temperatures.

2. In conjunction with the above work, the following laboratory tests will be conducted:

a) Metallographic examinations of weld metal, fusion and heat affected zones.

b) Mechanical tests such as side and bend tests.

c) Hardness surveys of weld and heat affected areas.

d) Chemical analysis.

e) Other tests that may be necessary to evaluate the quality of the transition metal overlay.

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3. Testing of procedures for joining the transition overlay to the type 316 plate material.

   a) Automatic submerged arc welding of 5" thick material to be tested under restrained conditions.

   b) Hand welding procedures may require checking, depending on 3-a above.

   c) Quality control methods will be evaluated; i. e., X-ray of partial welds and other non-destructive methods.

   d) Metallographic evaluation of welds, fusion and heat affected zones.

   e) Mechanical tests will be conducted to satisfy ASME Code and other requirements deemed necessary to meet the quality and strength conditions according to the service requirements.

4. Cyclic thermal fatigue tests.

   a) A literature survey will be conducted to collect all available details on work by the utility companies on the welding and thermal fatigue properties of ferritic to austenitic welded joints.

   b) A sufficient number of prototype test specimens will be fabricated to evaluate the thermal fatigue properties of the joint.

   c) Thermal fatigue tests will be run.

   d) The results of the above (c) will be evaluated by non-destructive, destructive and metallographic techniques.

Mechanical Test of Components

1. Mechanical Behavior of Sine Wave Tubes

   a. At room temperature and at elevated temperatures.
      1. The objectives of this test are to determine the effect of tube row curvature on pre-bent sine wave tubes and to further study the true fatigue strength of sine wave tubes.
      2. This test would be a continuation of the previous testing performed by this company. In brief, the sine wave tube is held in a long metal lathe and longitudinally flexed by use of the automatic feed mechanism. The first series of tests would be to flex several tubes pre-bent for various tube row curvatures (75^{o} to 180^{o}.) under normal deflection and cycles. This test would be performed with both thin and thick tubes as utilized in the preliminary design. Further testing of standard tubes should be performed by similar flexing with increased longitudinal deflection until fatigue occurs.
      3. Similar tests will be run, modified to produce the additional results of high steady state and cyclic temperatures.

2. Tube to Tube Sheet Welded Connection

   a. The objective of this program is to determine and verify the advantages of tube to tube sheet connections using trepanned risers.

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b. A series of tests would be performed on small test units simulating each sodium component. Each test unit will consist of two small diameter tube sheets with one row of sine wave tubes. Tube to tube sheet connections with and without risers will be included in each unit so that exact conditions will be imposed upon each type and shape of connection. In this way various welded joints and riser shapes can be compared. The tube and tube sheet unit will be installed into a flanged shell common to all three test units. This unit will then be cycled hot to cold through approximately 500 F for 50 cycles. The heating of the tubes and tube sheets can be done by superheated low pressure (150 psig) steam at 660 F and suddenly cooled by water at 60 F. In this way critical thermal stresses as expected during thermal transients can be imposed upon the tube to tube sheet connection for each component.

c. After completion of 50 cycles at the maximum temperature differential expected during operation, the temperature differential would be increased until failure occurs.

3. Metallurgical Examination of Bi-metallic Tubes

a. The objective of this task is to determine the integrity of stressed bi-metallic tubes.

b. This test includes a complete metallurgical examination of bi-metallic tubes after being thermally stressed or fatigued from previous component tests of this program. This examination would include X-ray, dye-penetrant, ultrasonic or other methods as required.

4. Superheater to Steam Drum, Transition Weld

a. The objective of this test is to determine the reliability of the transition weld in the superheater and steam drum connection as experienced in actual operation.

b. This test will be performed on a full size joint as fabricated under the welding development of this program. The test sample will be restrained by attachment to restraining blocks and cycled in an automatic furnace or other suitable means between 650 F and ambient temperature. An internal pressure of 2500 psi should be maintained during the heating portion of the cycle. The unit should be subjected to 1000 cycles with X-ray inspection of the joint after every 50 cycles.

5. Decarburization of 2-1/4% Cr - 1 Mo Steel

a. Objective: Present data on decarburization of 2-1/4 Cr steel in sodium indicate that the material will decarburize at temperatures approaching 1100 F. Other data indicate that no decarburization occurs at temperatures of approximately 930 F. Metallurgical evaluation of the two-tube Griscom Russell evaporator operated at MSA, as reported in KAPL-1884 and KAPL-1826, shows that no decarburization occurred on the NaK surface of that unit after 3856 hours of steaming time. Normal NaK inlet temperature was 850 F but there was one 400 hour creep test with a NaK inlet temperature of 1010 F and several short time incidents with NaK temperatures as high as 1300 F. In view of the dearth of data to indicate the temperature at which decarburization begins and the marginal operating temperature of the proposed 2-1/4 Cr steel boiler, a corrosion test program is recommended to determine the temperature at which the boiler material is in danger of decarburization.

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b. Test loops of 2-1/4 Cr steel will be tested in circulating sodium to determine the temperature at which decarburization begins. Since pure sodium will be necessary for the test, it is recommended that a circulating purification loop be utilized as the source of sodium. The purification loop will consist of a sump tank, an electromagnetic pump and flow meter, a hot trap section wrapped with nichrome heaters and packed with an oxide gettering agent such as titanium, and adequate temperature controls. The loop will be designed for a hot trap temperature of 1200 F during sodium purification operation. Sodium will be loaded into the sump tank through a micro metallic filter and subsequently purified by circulation through the hot trap prior to exposure of test material. This purification procedure will insure satisfactory sodium purity and eliminate the need for costly oxygen analyses.

Loops of the test material will be constructed as by-pass lines from the main purification loop. The number of test loops and velocity or range of velocities of sodium flow through them will be determined at a later date. For initial testing, the circulating sodium temperature will be maintained at a predetermined temperature such as 1000 F. When the loop sodium has been adequately purified, valves will be opened to allow sodium flow through the individual test loops. At a set time, say 1000 hours, one of the loops will be valved off and cut up for metallographic examination. If decarburization has occured, the rest of the loops under test will be removed for examination. Fresh loops would then be tested at a stepped down temperature of 980 F.

If no decarburization has occurred when the initial loop is examined, the other loops will be further tested at the same temperature until an adequate time has lapsed to insure that no decarburization will occur. This procedure will continue until the temperature at which decarburization begins is satisfactorily pinpointed.

Test loops can easily be removed and replaced by new test loops. Other materials can be investigated for sodium corrosion if desired. The main loop material will be type 316 stainless steel so that any accelerating effect of this material on decarburization of 2-1/4 Cr steel in the same system will be present.

Periodic sodium analyses for carbon content will be desirable.

6. Acceptance Testing for Materials

a. The objective of this portion of the testing is the development of acceptable procedures for inspection of materials, such as tubing, forgings and sheet, and outlining the limits and basis for rejection.

b. The procedure is to determine the limits for chemical, metallurgical and material soundness specification for the various types and forms of material to be used as well as the methods of inspection to be used.

c. A project shall be initiated in which close laison will be maintained between the supplier of raw materials and the contractor. Existing and new testing techniques will be studied and samples of rejectable and acceptable material according to the vendor's best manufacturing practice will be tested by destructive metallographic procedures. In such cases where material may be declared acceptable according to the vendor s inspection procedure but destructive inspection proves otherwise, more refined inspection techniques must be used. For example inspection of bi-metallic tubes must be sufficient to determine the integrity of the bond between stainless steel and inconel.

Model Test

Thermal, Operation and Control Test

This testing is based on using a scaled down system duplicating as many features of the presented design as possible and operating at approximately 10% of full design flow capacity. All elements, such as tubes, are to be duplicates of design elements. The relative resistances are to be maintained as close as possible to design values. The actual size and capacity will be designed to utilize, as much as possible, any existing accessories such as sodium pumps and valves, etc.

It is not certain that results obtained from a small scale test can be directly extrapolated to the full size unit, but by maintaining the same method of thermal analysis the results will be more nearly consistant with the full size unit configuration than the generalized extrapolated data used in the preliminary calculations. Since the size and performance of the final unit depends mostly on the heat transfer factor the scale model will be designed to maintain the heat transfer effects rather than pressure drops or distribution effects. Temperature and pressure conditions will be maintained as closely as possible to the full size design values.

The individual sections of the system (heat exchanger, boiler, boiler and superheater) will be tested completely in the small scale sodium loop for their heat transfer characteristics, and then the combined system will be tested to check out the operation and control mechanisms. The control mechanisms will be identical to those recommended for the full size units.

The full size effects, dynamic and transient responses of equipment and fluid flow characteristics will vary radically from the model to the full size unit but it is anticipated that certain evaluations and criteria for analysis may be obtained.

The heat exchanger test may be performed on a single sodium "figure eight" loop with an auxiliary cooler between the shell and tube side sodium stream. The cooler shall be capable of transferring enough heat so that extreme variations in flow and temperature may be imposed in the exchanger. A system of controllable by-passes around the primary and secondary sides of the exchanger will give sufficient flexibility to simulate most operating and transient conditions.

The boiler test may be performed on a single sodium loop and high pressure water system with an auxiliary means of condensing the steam generated. The characteristics of the boiler under varying sodium temperature levels and approaches and the operating characteristics of the preheating and separating portions of the drum will be determined. Tests of the control and relief mechanisms may also be performed in this test.

The boiler and superheater test is to determine the operating characteristics of the combined boiler-drum-superheater system. The boiler and superheater are connected in series in a single sodium flow loop. A high pressure feed water system and an auxiliary means of condensing steam are required to complete the steam generation system. Testing of the characteristics of the system and the effects on steam superheating at various sodium flow rates and temperature levels during normal operating fluctuations are to be determined.

Some of the major areas to be tested are outlined as follows:

1. Steaming Capacity of Bayonet Tubes

   a. The objective of this portion of the testing is to confirm the steaming rate and circulation rate in the bayonet tubes.

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b. Since the operation of the boiler and its response to system requirements depends on the ability of the bayonets to follow sodium temperatures and controls, the elements will be full size and full scale but fewer in number than the submitted design. The water side conditions will be held at design values except for gross flow. The tubes must be able to react quickly enough to prevent excessive pressure surges and hunting.

2. Preheating Capacity of Contacting Tray

a. The objective of this portion of the testing is to evaluate the capacity of the steam drum internals to preheat the incoming feed water to saturation at loads ranging from full rated load to the point of "weeping" of the tray and then to also determine what effect, if any, the "weeping" will have on the low load operation. The scrubbing effect of the contacting tray on the exit steam may also be checked.

3. Steam Separation in the Steam Drum

a. The objective of this portion of the testing is to investigate:

1 The effect of steam bubbles in the reservoir volumes and downcomer tubes on circulation.

2 The quality of steam from the drum at various operating conditions and levels.

b. While the steam drum has been emperically designed to avoid the entrainment of steam bubbles in the downcomer tube entrance regions and to give maximum quality and purity of steam to the superheater, a test will be made for verification of these methods.

4. Testing of Control Methods and Systems

a. The objective of this test is to verify the selection of the control systems chosen and their operation.

b. Controls duplicating the type selected for use in the submitted design will be installed and tested under all conditions likely to be imposed on the final units.

5. Chemical Carryover and Deposition

a. The objective of this portion of the test is to determine the effect of the various methods and locations of boiler chemical injection, taking into consideration the separating and scrubbing capabilities of the steam drum and contacting tray.

b. Although the quality of steam expected to the superheater is high, it is possible that boiler chemical carryover could deposit in the lower superheater tubes. Also according to some conventional boiler theories the deposits formed may be soluble or insoluble depending on the pressure range in operation. For instance, in the 500 to 900 psi range the normal deposits are usually water soluble, in the 1000 to 1500 psi range silica deposits predominate and are not easily removed, and with pressures of 2000 psi or over the insoluble deposits tend to diminish.

6. Operational Limits and Procedures

a. The objective of this test is to determine the operational limits of the units and fix operating procedures. These would include:

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1 Liquid fill levels for starting up.

2 Maximum safe rate of change of load.

3 Maximum safe rate of raising and lowering pressure.

4 Maximum safe rate of shutdown under normal emergency conditions.

7. Analog Simulation of Transient Response

a. The objective of this program is to derive a valid analog simulation to predict the transient response of a 70 MW heat exchanger and steam generator.

b. This will be accomplished by first deriving a general analog simulation of the transient behavior of the heat exchanger and steam generator. The simulation will be a mathematical representation of the units, consisting of simultaneous differential equations, which describe the various physical phenomenon and can be represented by analog computer circuitry. The validity of the general simulation will be proven by application to the specific test models, and correlations of analog response and test response to similar perturbations. These general analog simulations will then be applied to the 70 MW units to predict their transient behavior.

Experimental Evaluation of Prototype Units

In general it is recommended that the heat exchanger and steam generator be finally tested as full size units. The mechanical and thermal performance may be evaluated in a sodium loop or plant of a capacity such that full rated flow may be obtained. It is desirable but not necessary that this test loop develop full rated power. A sodium test loop of approximately 30-35 MW and rated flow would enable the system to be tested at approximately 50% performance and also by operating at full flow, part power the extrapolation of performance to full power would be reasonably accurate. The emphasis in this testing would be placed on obtaining full flow condition.

The heat exchanger could be tested to approximately 50% of design point and sufficient data obtained to extrapolate to 100% design.

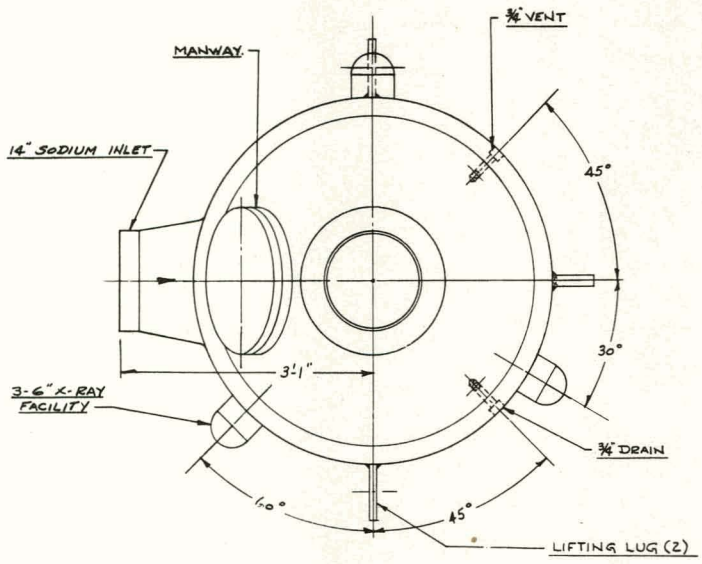
The boiler section could be tested to approximately full load without the superheater connected or to approximately 50% of design with the superheater in the system.

The individual components having been evaluated separately the final test should be of the complete system including the heat exchanger and steam generator with a secondary sodium loop so that casualty situations and exact response of equipment and control mechanisms can be evaluated under operating conditions. The results of the full size tests should be compared with data from the model tests and analogue simulations.

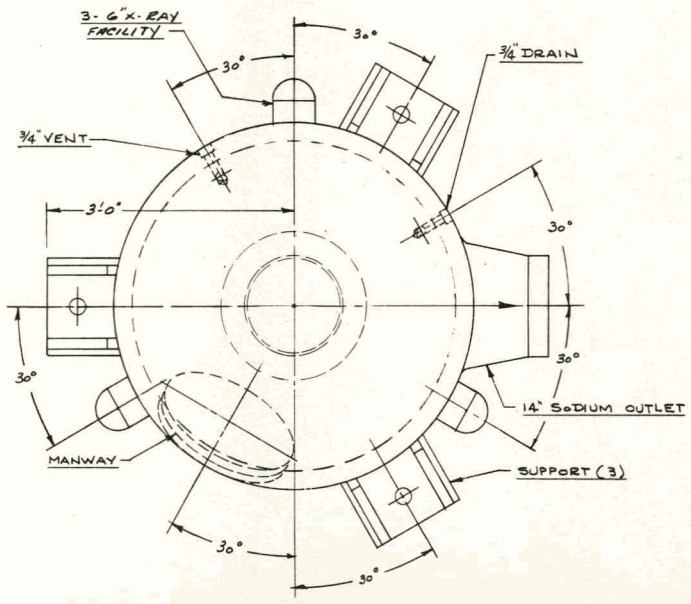
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SECTION 7

APPENDIX



SECTION A-A
SCALE 1" = 1'-0"

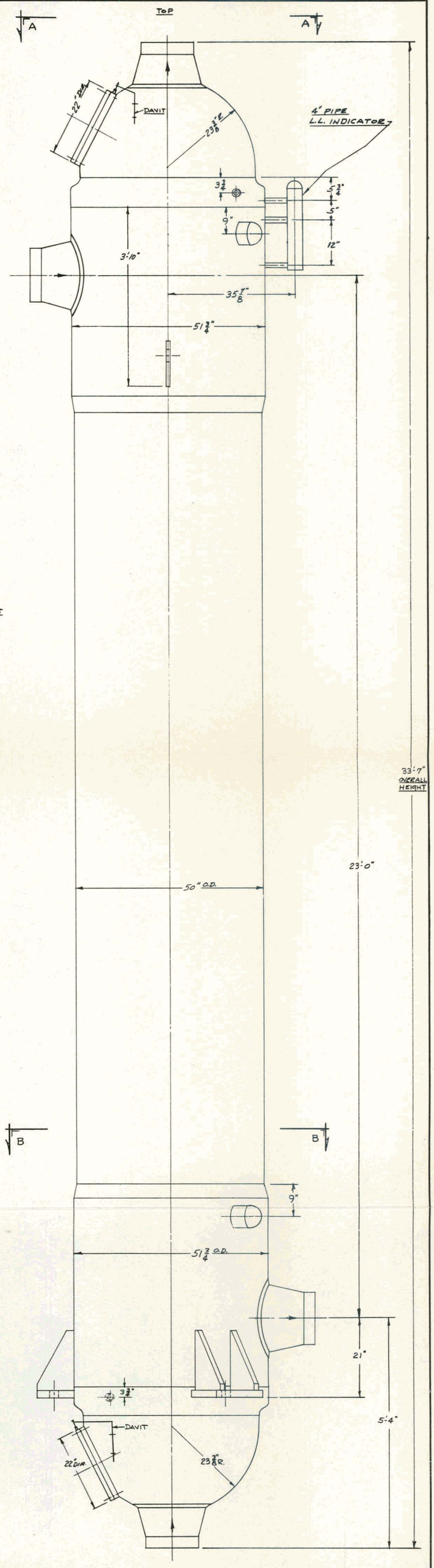


SECTION B-B
SCALE 1" = 1'-0"

CALCULATED WEIGHTS (LBS)	
ENTIRE UNIT (DRY)	35,000
ENTIRE UNIT (OPERATING)	53,500
ENTIRE UNIT (WITH WATER	57,200
MANHOLE COVER EACH	180

	DESIGN PRESS.	DESIGN TEMP.	TEST PRESS (CODE)
SHELL SIDE	100 P.S.I.	1200°F	413 P.S.I.
TUBE SIDE	150 P.S.I.	1200°F	620 P.S.I.

REFERENCE DRAWINGS	
CROSS SECTION	D-666-1-2
TUBE STUDY	D-666-1-3
NOZZLE CONNECTION	B-666-1-4



DIMENSIONAL TOLERANCE UNLESS OTHERWISE NOTED
FRACTIONAL ± 1/16"
DECIMAL ± .005
ANGULAR ± 1°

ISSUE	DATE	BY	CHD	REVISION

REVISIONS

ALCO

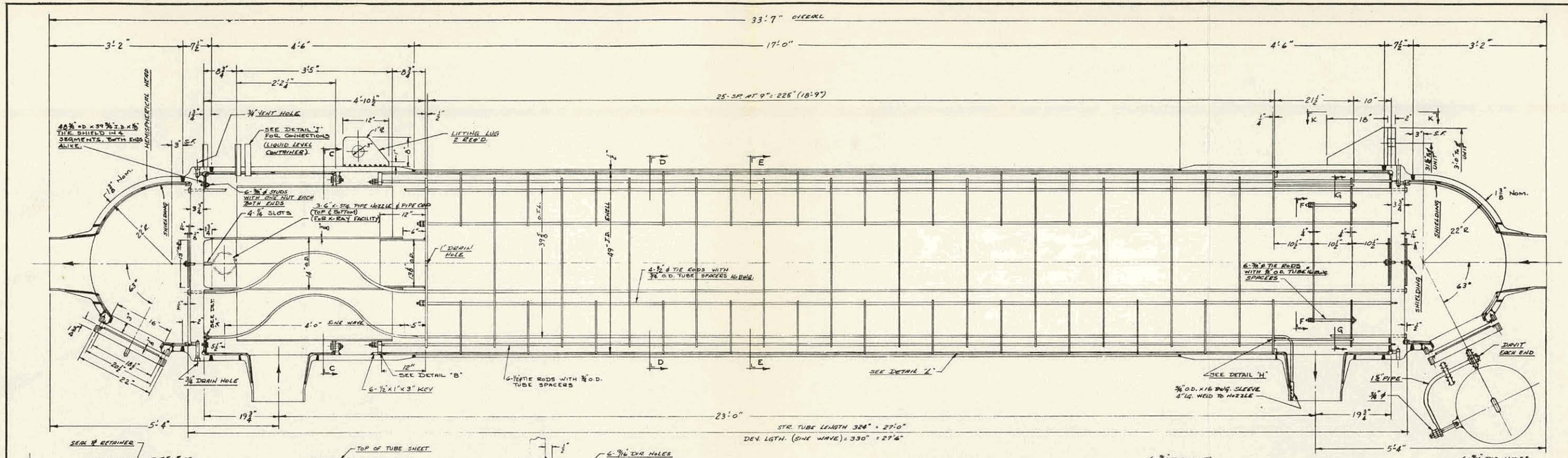
ALCO PRODUCTS, INCORPORATED
RESEARCH & DEVELOPMENT
SCHENECTADY, NEW YORK

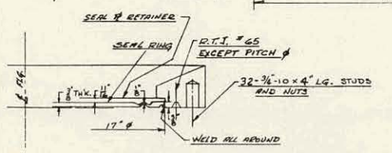
INTERMEDIATE HEAT EXCHANGER
OUTLINE

SCALE	MADE	CHK'D	CERT.	DATE	DRAWING NO.	ITEM NO.	IDENT. NO.
				3/26/57	D 666	1	1

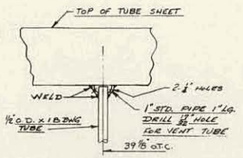
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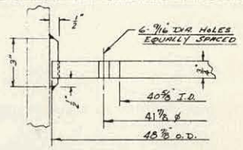




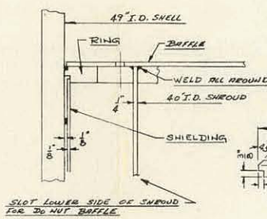
DETAIL OF MANWAY FLANGE AND SEAL.
SCALE 3" = 1'-0"



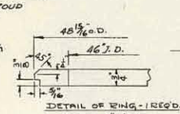
CONTINUOUS INERT GAS VENT TUBE
DETAIL "A"
SCALE 3" = 1'-0"
1 REQ'D.

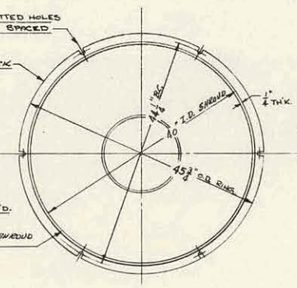


DETAIL "B"
RETAINING RING
1 REQ'D
SCALE 3" = 1'-0"

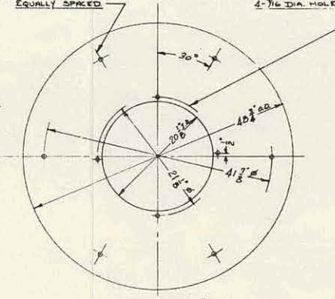


DETAIL "H"
SHOWING SHROUD AND RING ASSEMBLY
SCALE 3" = 1'-0"

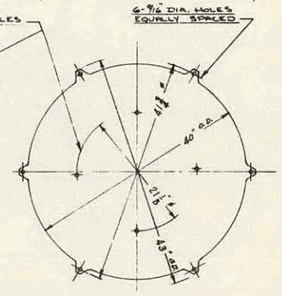




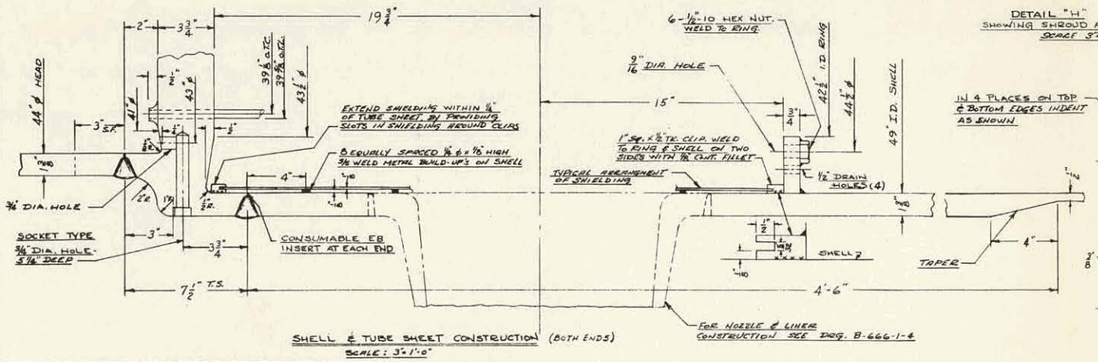
SECTION "C"
SHROUD & RING
SCALE 1" = 1'-0"



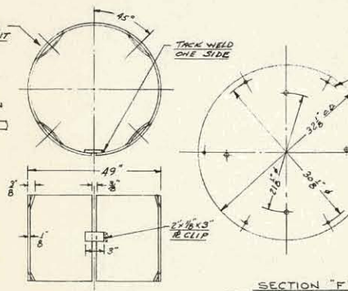
SECTION "D"
DO-NUT BAFFLE - 13 REQ'D.
SCALE 1" = 1'-0"



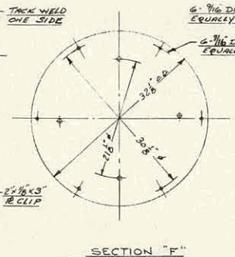
SECTION "E"
DISC BAFFLE - 13 REQ'D.
SCALE 1" = 1'-0"



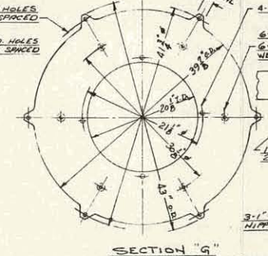
SHELL & TUBE SHEET CONSTRUCTION (BOTH ENDS)
SCALE: 3" = 1'-0"



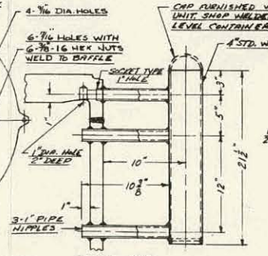
DETAIL "L"
TYPICAL SHIELD AT BAFFLE



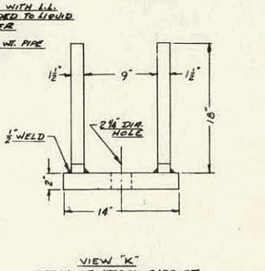
SECTION "F"
DISC BAFFLE - 1 REQ'D.
SCALE 1" = 1'-0"



SECTION "G"
DO-NUT BAFFLE - 1 REQ'D.
SCALE 1" = 1'-0"



DETAIL "J"
LIQUID LEVEL INDICATOR CONNECTIONS & CONTAINER
SCALE 1-1/2" = 1'-0"



VIEW "K"
DETAIL OF VESSEL SUPPORT
SCALE 1-1/2" = 1'-0"
4 - REQ'D.

MATERIAL SPECIFICATIONS	1" OR. PITCH 1" BND. PITCH
TUBES: 1088 1/2 O.D. x 18 BWG. SEAMLESS 316 SA 213 GR TP-316 (LIKEL. VENT TUBE)	
SHELL & CHANNEL, SEAL RING & RETAINER: 316 SA 240 GR M TP-316	
MANHOLE COVER, VESSEL SUPPT: 316 SA 240 GR M TP-316	
MANHOLE RING & TUBE SHEETS: FG'D 316 SA 182 GR F 316	
NOZZLE BODY: FG'D. 316 SA 182 GR F 316	
CENTER PIPE: 14" O.D. STD. WT. SEAMLESS 316 TP-304 COMMERCIAL	
SHROUD, BAFFLES & INT. SUPP'T.: 316 R. TP-304 COMMERCIAL	
SHROUD SUPPORT: SHIELDS & BACKING UP STRIP: 316 R TP 304 COMMERCIAL	
NUTS: SA-194 GR 8C TP-347	
STUDS: SA-193 GR B 8C TP-347	
GASKET: R.T.J. SEAMLESS 316 TP-316 COMMERCIAL ANNEALED	
SPACERS: TUBING 316 TP-304 COMMERCIAL	
TIE RODS: 316 BAR TP-304 COMMERCIAL	
MANHOLE DAVIT PART: TP-304 COMMERCIAL	
PIPE NOZZLES 316 SA-312 GR TP 316	

GENERAL NOTES:

CONSTRUCTION & WELDING SHALL BE IN ACCORDANCE WITH ASME CODE SECTION VIII (1956 ED) & T.E.M.A. SPECIFICATIONS.

STRESS RELIEVE NO.

CORROSION ALLOWANCE - NONE
CODE STAMP APPLIED
RADIOGRAPH YES IN ADDITION THE CLOSING SEAMS DYE PENETRANT INSPECTION ROOT PASS EVERY 3/8 OF WELD DEPOSIT & COVER PASS.

REFERENCE DRAWINGS

OUTLINE	D-666-1-1
TUBE STUDY	D-666-1-3
NOZZLE CONNECTION	B-666-1-4

CALCULATED WEIGHTS (LBS.)		
ENTIRE UNIT (DRY)	35,000	DIMENSIONAL TOLERANCE UNLESS OTHERWISE NOTED
ENTIRE UNIT (OPERATING - SODIUM)	53,500	FRACTIONAL ±1/16
ENTIRE UNIT (WITH WATER)	57,200	DECIMAL ±.005
MANHOLE COVER EACH	180	ANGULAR ±1°

	DES. PRESS.	DES. TEMP.	TEST. PRESS. CODE
SHELL SIDE	100 P.S.I.	1200° F	413 P.S.I.
TUBE SIDE	150 P.S.I.	1200° F	620 P.S.I.

D-666-1-2

ISSUE	DATE	BY	CKD	REVISION	CERT

REVISIONS

ALCO

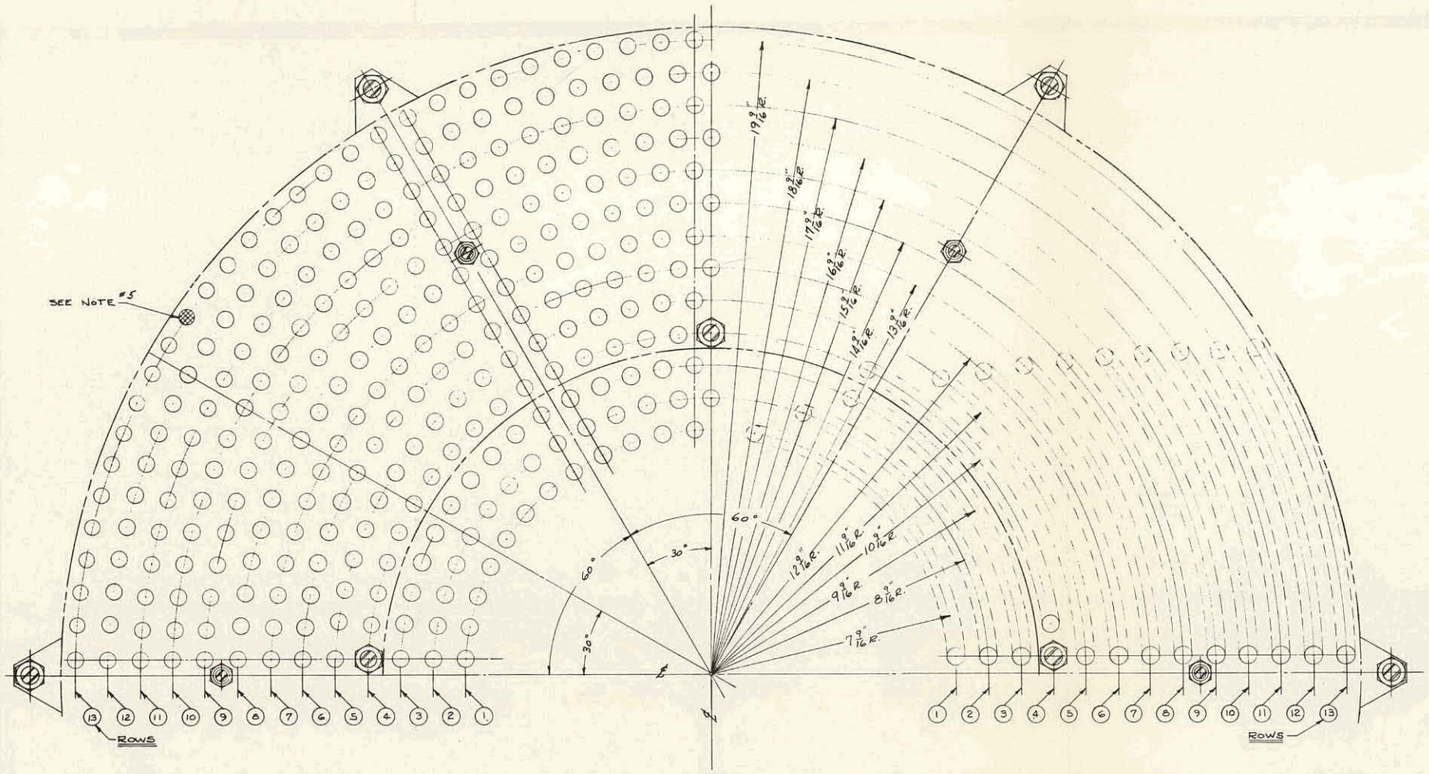
ALCO PRODUCTS, INCORPORATED
RESEARCH & DEVELOPMENT SCHENECTADY, NEW YORK

INTERMEDIATE HEAT EXCHANGER
CONTRACT AT(11-1)666

SCALE	MADE	CHK'D	CERT.	DATE	DRAWING NO. ORDER No.	ITEM No.	IDENT. No.
1/2"-0"	Farrell	WJM		12/1/61	D 666	1	2

967 069

7-A3



VIEW SHOWING TUBE & TIE ROD ARRANGEMENT
SCALE 6"=1'-0"

VIEW FROM TOP END AFTER 2ND BENDING OPERATION

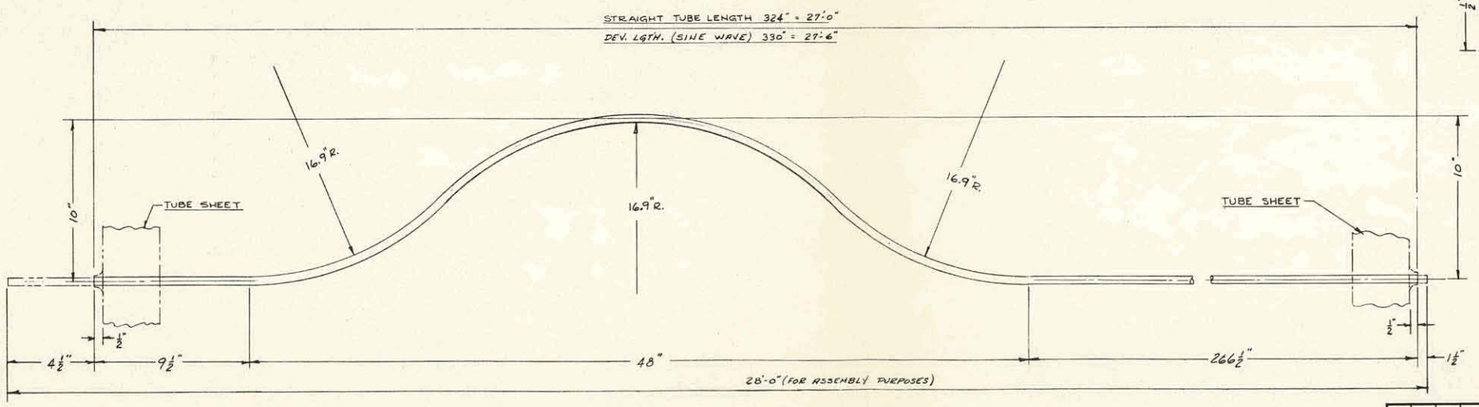
TUBE SHEET DRILLING

ROW No.	CIRCLE DIA.	EQUAL SPAC.	CIR. Ø	NO. OF PITCHES	No. TUBES	
1	15 1/8"	.9899	47.517	48	48	
2	17 1/8"	.9963	53.800	54	54	
3	19 1/8"	1.001	60.083	60	60	
4	21 1/8"	1.005	66.366	66	62+	4 TIE RODS
5	23 1/8"	1.009	72.649	72	72	
6	25 1/8"	1.012	78.933	78	78	
7	27 1/8"	1.014	85.216	84	84	
8	29 1/8"	1.016	91.499	90	90	
9	31 1/8"	1.018	97.782	96	96	
10	33 1/8"	1.020	104.065	102	102	
11	35 1/8"	1.021	110.348	108	108	
12	37 1/8"	1.023	116.632	114	114	
13	39 1/8"	1.025	122.915	120	119+	1 VENT TUBE

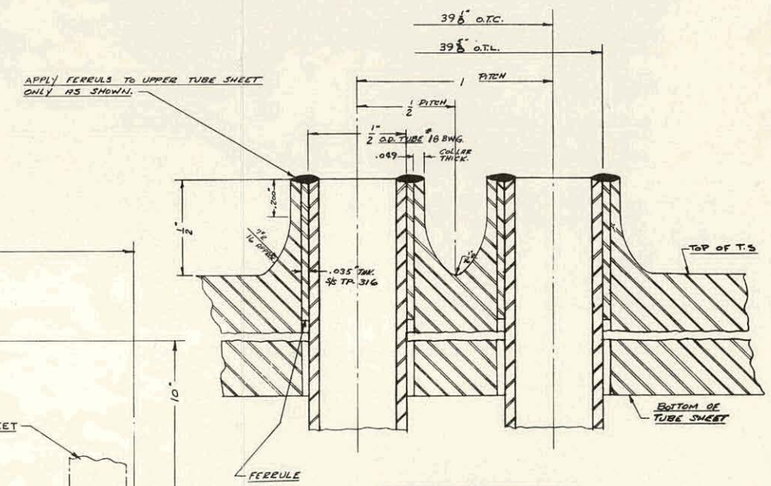
1087 + 1 VENT = 1088 TOTAL

NOTE:

1. ROW No. 4 - 4 TUBES WERE ELIMINATED TO ACCOMODATE 1" O.D. TIE RODS & SPACERS AS SHOWN.
2. BETWEEN ROW 8 & 9 AT 30 1/16 DIA. 6-3/8 O.D. TIE RODS & SPACERS AS SHOWN.
3. AT 41 7/8 DIA. 6-1" O.D. TIE RODS & SPACERS AS SHOWN.
4. TOTAL No. OF TUBES: 1088. DRILL & REAM TUBE SHEET FOR 1087 3/4 O.D. TUBES ON 1" CIR. & 1" RAD. PITCH AS SHOWN.
5. ONE TUBE MARKED THUS ⊗ TO BE USED AS INERT GAS VENT PIPE. DO NOT DRILL TUBE SHEET, SEE DRG 1E-1005-1



TUBE DETAIL
AFTER FIRST BENDING OPERATION
SCALE 1/4"=1'-0"



TUBE WELDING ARRANGEMENT
SCALE 3"=1"

REFERENCE DRG.
D-666-1-2 INT. HEAT EXCHANGER (X-SECTION)

D-666-1-3

ISSUE	DATE	BY	CKD	REVISION	CERT
				REVISIONS	

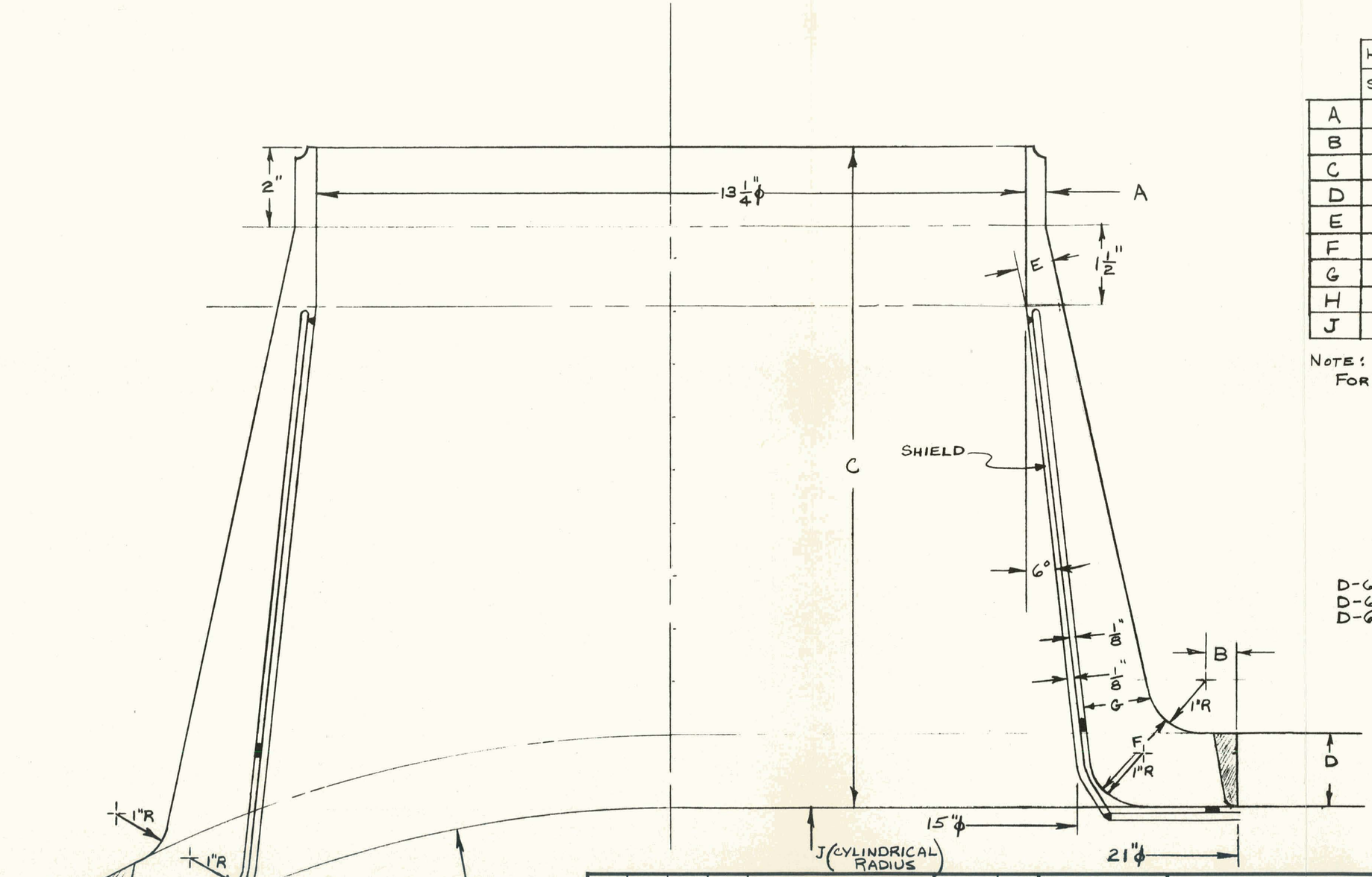
ALCO

ALCO PRODUCTS, INCORPORATED
RESEARCH & DEVELOPMENT SCHENECTADY, NEW YORK

HEAT EXCH. TUBE STUDY, TUBE WELDING AND BENDING DETAILS

SCALE	MADE	CHK'D	CERT.	DATE	DRAWING NO.		
	W. Jones / W. Farrell	W.J.H.		1/6/69	D	ORDER No. 666	ITEM No. 1	IDENT. No. 3

067 070



	HEAT EXCH.		BOILER		SUPER H.
	SHELL	HEAD	SHELL	HEAD	
A	3/8"	3/8"	3/8"	3/8"	3/8"
B	9/16"	9/16"	9/16"	9/16"	9/16"
C	13"	13"	13"	13"	13"
D	1 3/8"	1 3/8"	1 3/8"	1"	1 3/8"
E	11/16"	11/16"	11/16"	11/16"	11/16"
F	1 3/4"	1 3/4"	1 3/4"	1 1/2"	1 3/4"
G	1 1/4"	1 1/4"	1 1/4"	1 1/4"	1 1/4"
H	—	22"	—	40 1/2"	—
J	24 1/2"	—	40 1/2"	—	22 1/2"

NOTE:
FOR MATERIAL SEE X-SECTION DRAWING

REFERENCE DWGS.
D-666-1-2 INT. HEAT EXCH. (X-SECTION)
D-666-2-2 STEAM GEN. (BOILER SECTION)
D-666-2-3 STEAM GEN. (SUPERHTR. SECTION)

B-666-1-4

ISSUE	DATE	BY	CKD	REVISION	CERT

REVISIONS

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ALCO PRODUCTS, INCORPORATED
RESEARCH & DEVELOPMENT — SCHENECTADY, NEW YORK

14" O.D. NOZZLE CONNECTION
CONTRACT - AT(11-1)666

SCALE	MADE	CHK'D	CERT.	DATE	DRAWING NO.		
					ORDER No.	ITEM No.	IDENT. No.
6" = 1'	RDK	WJM.		12/18/56 B	666	1	4

967 071

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B - SPECIFICATIONS

PREFACE

The specifications included herein are of a preliminary nature and may be revised or amended as the need arises. The intent of the specification in regard to the quality of materials and fabrication techniques, is to maintain standards equal to or better than similar equipment manufactured for the Naval and Civil nuclear facilities.

907 072

SPECIFICATIONS FOR SODIUM TO SODIUM INTERMEDIATE HEAT EXCHANGER

A. SCOPE

1. This specification covers the mechanical design data, operating conditions, materials of construction, fabrication requirements and other necessary information relative to the construction of a sodium to sodium heat exchanger.

2. This 70 megawatt intermediate heat exchanger shall be an all stainless steel unit of single tube construction and shall be capable of transferring heat from a sodium reactor coolant to a non-radioactive sodium coolant.

B. DESIGN DATA

1. DESIGN CONCEPTS

a. The heat exchanger shall be a vertically designed unit consisting of fixed tube sheets welded to a cylindrical shell. The channels (tubeside) shall be hemispherical heads welded to the tube sheets.

b. Each channel shall be furnished with a 16" I. D. manway complete with cover, gasket, studs and cover davit. The manway opening shall be provided with a thin corrugated seal plate welded to the manway ring.

c. Each tube sheet shall be trepanned on the tube side to form tapered risers (nipples). Seamless tubes shall be expanded into the tube sheet and welded to the trepanned risers.

d. One end of the seamless tubes shall be prepared with a sine wave of sufficient length to allow for shell thermal expansion.

e. The shell and channel interior surface shall be provided with the necessary shielding to cover thermal transients.

f. The shell shall be provided with two welded inspection openings consisting of an 8" nominal pipe and welding cap.

g. Internal design consists of disc and do-nut baffles, internal shroud at inlet and outlet nozzles together with a center pipe (dummy) and necessary baffles, tie rods and spacers.

h. Sufficient vent and drain connections shall be provided at low and high points of shell and tube side.

i. Three liquid level connections shall be installed.

j. Four bracket type vessel supports and two lifting lugs for field installation shall be furnished.

073

k. Reference drawings for these specifications are as follows:

1.	666-1-1	Outline
2.	666-1-2	Cross Section
3.	666-1-3	Tube Layout
4.	666-1-4	Nozzle Layout

2. STEADY STATE DESIGN OPERATING CONDITIONS

	Shell Side	Tube Side
Total Fluid Entering	1,665,000 #/Hr	1,665,000 #/Hr
Temperature, In	1200° F	665° F
Temperature, Out	723° F	1140° F
Operating Pressure	100 psig	100 psig
Maximum Pressure Drop	15 psi	15 psi
Heat Transfer Duty	239,000,000 BTU/Hr	

3. MECHANICAL DESIGN CONDITION

	Shell Side	Tube Side
Temperature	1200 F	1200 F
Pressure	100 psig	150 psig

C. MATERIALS OF CONSTRUCTION

The major components of the sodium heat exchanger shall be constructed of A.I.S.I. Type 316 stainless steel. Components made of other materials shall be as specified. ASTM specifications and Section II of the ASME Boiler and Pressure Vessel Code shall apply to these materials. The reliability and intended service conditions for some of the components of this unit, however, may require modification of these specifications. While internal quality is not a requirement set forth in the ASTM specifications, the material must meet ultrasonic and liquid dye penetrant inspection set forth herein. Failure to do so will be basis for rejection, even though the material meets with the ASTM specification requirements.

1. TUBES

ASTM A-213-55T, Grade TP-316, Seamless

2. TUBE SHEETS AND FORGED NOZZLES

ASTM A-182-56T, Grade F-316

Requirements in Section 4 (b) and (d) of the above specification shall also apply. Under Section II (c), the test specimens, when possible, shall be cut from the production forging. The location of the test specimens shall be indicated on the sketches

907 074

submitted to meet Section 4 (d). Location of test bars to be approved by the fabricator and contractor prior to release of the order. The test specimens are to be heat treated as an integral part of the forging.

3. SHELL BARREL, HEMISPHERICAL HEAD AND VESSEL SUPPORTS

ASTM A-240-57T, Grade M-316, Finish Hot Rolled, Annealed, Pickled.

Under Section 10 (a), the tensile test specimen shall be cut parallel with the major dimension of the plate, which represents the circumferential dimension of the finished shell barrel.

Section 12 shall apply.

4. PIPE NOZZLES

ASTM A-312-56T, Gr. TP-316, Seamless

5. NUTS

ASTM A-194-56T, Grade 8-C (TP-347)

6. STUDS

ASTM A-193-56T, B-8C (TP-347)

7. SHROUD, BAFFLES, SHIELDS, INTERNAL SUPPORTS, SPACERS, TIE RODS, MANHOLE DAVIT PARTS, CENTER PIPE AND BACKING UP STRIPS

Commercial gradeof AISI Type 304 or 316 stainless steel shall be acceptable.

8. GASKET

Type 316 stainless, seamless annealed

D. INSPECTION OF MATERIALS

1. All tubing shall be 100% ultrasonically inspected at the mill for internal defects. Tubing examined longitudinally and in two directions circumferentially shall have no defect in excess of 0.004 inches or 3% of the wall thickness, whichever is greater, or more than 3/8 inch long. A notarized inspection report shall accompany the mill test report, and the manufacturer must guarantee the quality.

2. Ultrasonic inspection shall also be made on all forgings, according to ALCO Spec. MTS-100, Rev. 2, All additional pipe and plate shall be inspected according to applicable ASTM specifications.

007 075

3. Radiographic inspection shall be required on all tubing, pipe forgings and plate where ultrasonic inspection is not feasible, or results are unsatisfactory. Radiographic inspection shall be performed in accordance with practices set forth by ASTM E-94-52R and the ASME Code. All indications of defects shall be within the limits set forth by the ASME Code, or as defined in these specifications. A dye penetrant check shall be made on all tubes, pipe, forgings and plate to determine surface quality and magnitude of any surface defects. All surface imperfections indicated by the dye penetrant inspection shall be considered as defects and may be removed by grinding, provided the thickness of the material is decreased to less than that permitted by the applicable ASTM specification or tolerances specified on the engineering drawings for the particular part. After conditioning, material shall be reinspected.

4. Ultrasonic, radiographic or liquid dye penetrant inspection shall not be required on any material designated as commercial quality.

E. FABRICATION

1. GENERAL

a. All fabrication, processing and inspection shall conform to the latest edition of Section VIII of the ASME Boiler and Pressure Vessel Code, including addenda and T.E.M.A. Class R Standards, or as otherwise shown on the Engineering Drawings or contained in this specification. Code Stamp shall be applied.

b. Three copies of the welding procedures shall be submitted by the fabricator to the contractor before welding is actually begun for each procedure covered by the Code. One copy, approved or approved as amended, shall be returned to the fabricator prior to commencement of fabrication.

c. External piping and connections shall be properly beveled as specified on the Engineering Drawings to permit field butt-welding. The unit shall be fabricated with single wall tubes, with no overlays or clad material and shall be all-welded construction.

d. All materials of construction shall be properly identified as set forth in the ASME Code for Unfired Pressure Vessels. The heat and lot number of weld metal deposited in each major pressure weld and the welder or welders making welds shall be identified.

e. To eliminate possible stress raisers, or other adverse effects that may be caused by metal die stamping, all identifications required shall be clearly and accurately recorded on applicable sub assembly or final assembly drawings.

507 076

f. Weld defects may be removed by grinding or chipping by a competent operator to the satisfaction of a qualified inspector. All repair welds shall be reinspected by all techniques as specified and conducted on the initial welds.

g. All surfaces exposed to the primary and secondary sodium shall be machined or ground to a 250 rms surface finish. Where welds in the "as-deposited" condition meet the intent of the 250 rms surface finish, a dye penetrant inspection shall be made. Persistent bleeding that indicates cracks, folds, porosity, or other injurious defects shall be sufficient cause for repairing the deficiency and reinspection in accordance with these and other applicable specifications.

2. TUBE-TO-TUBESHEET WELD JOINTS

a. The tube-to-tubesheet joints shall be prepared from a trepanned type tube sheet, as illustrated in the attached and engineering drawings. The minimum ligament thickness of the trepanned area at the weld shall be as shown on the engineering drawings.

b. The procedure to be used for welding tube-to-tubesheet joints shall be submitted in triplicate by the fabricator to the contractor prior to qualification and fabrication of tube-to-tubesheet joints. Subsequent to approval of tube-to-tubesheet welding qualification, one copy of the fabricating procedure shall be returned to fabricator approved, or approved as amended.

c. Prior to assembly, the tubes and tubesheets shall be thoroughly cleaned of oil, grease, dirt, or other contaminators detrimental to effective tube rolling and welding.

d. A minute amount of rolling lubricant, such as "KS Lube-a-Tube", or distilled water, may be used for tube rolling. The rolling lubricant shall be free of chlorides. A letter of conformance to this effect shall be considered a part of this specification. Upon completion of rolling, and prior to welding, the rolling lubricant on the inside surfaces of the tube shall be removed by means of a clean, lint-free cloth.

e. The temperature of the tube sheet during welding shall not be less than 60° F, nor more than 300° F.

f. A minimum of two weld passes is required for manual welding procedures.

A single pass weld shall be permitted when automatic welding is performed using a properly qualified procedure and equipment.

g. The size of the tube-to-tube sheet weld, as measured from the

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point of fusion of tube-to-tube sheet projection, shall be a minimum of 1-1/2T (T = tube wall thickness).

h. Welds made by the inert gas shielded tungsten arc process may be left with an "as welded" surface, provided it is smooth (with normal ripples) and is free of any crevices as indicated by dye penetrant inspection.

i. Welds made by the metal arc process that do not meet the dye penetrant inspection shall require conditioning. In no case, however, shall the welds be ground below the 1-1/2T dimension, or in such a way as to remove the two pass leak path for manually made welds.

j. The weld deposit shall be sound and free of foreign matter, including tungsten particles. No weld metal shall be deposited over slag, cracks, gross porosity, or other conditions considered detrimental to weld quality. The weld deposits shall be free of cracks, incomplete fusion, crevices or other significant deficiencies that may be apparent by visual or liquid penetrant inspection. Excessive roll over shall not be acceptable.

k. The procedure for tube-to-tubesheet welding required by this specification shall include the manner in which repairs of defective welds shall be made. A repaired weld shall be a two pass weld with filler metal added as required.

F. QUALIFICATION OF TUBE TO TUBE SHEET WELD JOINTS

1. Procedure and operator qualification will be required for tube to tube sheet welds. Position used in qualification shall be used on the production unit. Procedure and operator qualification may be performed on the same test sample. Approval of qualification may be performed on the same test sample. Approval of qualification shall be required prior to initiation of production fabrication pertinent to tube to tube sheet weld joint design and welding.

2. Qualification and test sample shall consist of at least four rows of tube-to-tube sheet joints with a mimimum of five joints in each row. These joints shall have the same design, tube hole pitch and be of the same heat, if possible, or equivalent composition as materials to be used on the production heat exchanger.

3. The sample tube sheet, after completion of welding, shall be sectioned in planes parallel to the longitudinal axis of the tubes. One section shall be cut to include at least four completed joints and shall be made so as to contain approximately half circle tube sections. Those sections shall be prepared for examination at magnifications up to 30 diameters.

4. At least two other sections containing two weld cross sections

007 078

shall be made for microscopical examination at suitable higher magnifications. In addition, the contractor's representative may require examination at suitable higher magnification of any joint showing questionable defects or characteristics examined at magnification up to 30 diameters. All welds, other than those sectioned, shall be available for visual examination.

5. Approval of qualification by the authorized contractor's representative will be based upon meeting the requirements of tube-to-tube sheet joints as stated in Section E-2 of these specifications.

G. MATERIAL CLEANING

1. All material to be fabricated into pipe, fittings and equipment shall be cleaned of all mill scale, rust, or adherent dirt by shot blasting, or by mechanical brushing, using the same type of bristle material as that of the part being descaled. Internal surfaces of sub-assemblies or component parts which cannot be thoroughly cleaned and inspected after subsequent assembly shall be cleaned prior to assembly so as to insure removal of all contamination, including oils, greases, lubricants, rust, scale, weld spatter, grinding particles, fingerprints, and all other foreign matter. Such items shall be maintained in this same state of cleanliness after further assembly or fabrication.

2. It is essential that cleanliness and freedom from contamination be obtained and maintained for all final assemblies. Adequate inspection techniques shall be prescribed to insure the specified degree of cleanliness. All equipment shall be thoroughly dried and inspected subsequent to cleaning. No visible film shall appear on the cleaned surface upon drying.

3. Within the past year, Military Specification MIL-C-19874 (Ships), "Cleaning Requirements for Nuclear Primary Coolant Equipment Including Piping Systems", has become a generally accepted basis for cleaning primary system components for pressurized water reactors. This specification, MIL-C-19874 (including Amendment 1), is incorporated in, and forms a part of, the fabrication requirements for this heat exchanger.

4. Inasmuch as MIL-C-19874 is a general specification, the requirements listed therein are to be supplemented by the following procedures specific to this design.

5. The requirements of this specification shall apply to metal surfaces exposed to both primary and secondary sodium.

6. Since the inside surfaces of the tubes in this heat exchanger cannot be adequately inspected by the unaided eye, the wipe technique shall be used to evaluate tube cleanliness. Clean swabs are to be run through each tube by means of snakes. The swabs are to be visually inspected for the presence of foreign material by

the unaided eye and are to be examined under ultra violet light for the detection of oil and grease.

7. During fabrication, assembly, and testing, it is very important that all surfaces be protected from pitting or corrosion products. Considerable judgment will be necessary during this period to determine exactly what procedures and precautions should be taken to prevent or inhibit corrosion and contamination. Precautions fall into three general categories:

    a. Corrosion may be minimized by supplying sufficient heat to the unit to prevent the condensation of water, by using such items as strip heaters, hot air blowers, etc. Adequate consideration should be given to ill effects which may result from differential thermal expansion.

    b. Use of a preservative on certain components where the presence of the preservative does not interfere with the assembly operations or does not contaminate areas which will not be subsequently accessible for cleaning.

    c. The same results may be obtained by carrying out the assembly operations in an area which is sufficiently heated to prevent condensation.

8. Since lubricating oils will probably be used during various stages of fabrication and assembly, and particularly during final machining of the tube-to-tube sheet welds, the completed heat exchanger shall be chemically cleaned by vapor degreasing or alkaline immersion, or both. Water used in preparing the alkaline cleaning solution shall have a total solids content of less than 10 ppm. The unit shall be thoroughly rinsed with water having a pH of 10.5, obtained by adding trisodium phosphate. Excluding additions for pH control, the water shall have a total solids content of less than 10 ppm. The drying procedure shall be started within thirty minutes after the rinsing or draining operation in accordance with the procedures in 3.5.

9. When the surfaces of the assembly or component are in contact with water, such as during cleaning, rinsing, or testing, and water is drained from the unit, procedures shall be in accordance with the following:

    a. Small parts with all surfaces accessible shall be dried with clean, lint-free cloths, with filtered hot air, or by heating in an oven. Parts are considered dry when there is no visual evidence of water or moisture when the items have cooled to room temperature.

    b. The unit shall be dried in an oven maintained at a temperature between 250 and 300° F. Inlet and outlet pipe or suitable attachments shall be provided on the

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unit so that it may be flushed with an inert gas (-40 deg. F dew point) such as a nitrogen, after the unit has reached the temperature of the oven. Sufficient gas should be introduced into the unit in order to completely replace the moisture laden air. Following the completion of the flushing operation, the oven shall be turned off and the unit allowed to remain sealed for eight hours, after which a dew point reading shall be taken on the gas in the unit. If the dew point of the gas in the unit is equal to or less than -40 deg. F., the unit is considered dry. However, if the dew point determined is above -40 deg. F., additional drying shall be employed until this requirement is met.

Adequate consideration shall be given to ill effects which may result from thermal expansion.

H. INSPECTION (Fabrication)

1. A 100% radiographic inspection shall be required on all welds made in conformance to Section VIII of the ASME Boiler and Pressure Vessel Code, Part UW-51. On all welded joints where suitable X-ray inspection cannot be made, a dye liquid penetrant check may be substituted. Butt-welded joints requiring several passes, shall be inspected after the first pass, at the intermediate point and after the final pass. A minimum of three dye penetrant checks shall be required for each weld. All imperfections indicated by the dye penetrant inspection will be considered defects. Defective welds as determined by the radiographic dye penetrant imspection must be removed, repaired and then reinspected.

2. The completed unit shall also be subjected to a hydrostatic test on both the shell and the tube sides in conformance to the latest edition of Section VIII of the ASME Boiler and Pressure Vessel Code, Part UG-99K. Commercial distilled or deionized water shall be used for this test. Upon completion of the test, the vessel shall be dried and all traces of water shall be removed.

3. The vessel shall be subjected to a helium mass spectrometer leak test subsequent to the hydrostatic test. The vessel must be evacuated to an absolute pressure not to exceed 15 microns of mercury. No leak shall be detected by a Consolidated Engineering Co. Detector Model #24-101A or equivalent set at maximum sensitivity.

I. PREPARATION FOR SHIPMENT

1. After approval of cleanliness, unit is to be thoroughly flushed with an inert gas, and openings are to be sealed internally with expanding rubber plugs capable of maintaining an internal gas pressure of 2-4 psig.

2. Sealed unit is to be filled with an inert gas to a pressure of

4 psig.

3. Pipe openings which have been prepared for welding shall be protected from damage during shipment by end caps securely fastened.

4. A means of introducing inert gas into the vessel or pipe, shall be included in the plug device. (The cap shall be removable.)

5. All equipment shall be prepared with the necessary fixtures for shipment.

907 082

Material Test Specification - MTS-100 Rev. 2

ULTRASONIC TESTING
FORGINGS, FORGED & ROLLED RINGS

1.0 Scope

This standard is applicable to all steel grades; carbon, low and high alloy and the stainless steels in the form of ring forgings, pancake or disc forgings, head-channel forgings (water boxes,) shaft and pinion forgings.

2.0 Purposes

This specification sets forth certain basic requirements for acceptance testing; also, outlines the degree of sensitivity required and method of reporting findings.

3.0 Applicable specifications

3.1 MIL-Std. 271 Non-Destructive Testing Requirements for Metals

3.2 KAPL - Spec. #O-KPMO-44 Rev. 4

3.3 ASTM Specification E-114-55T

4.0 General Requirements

4.1 Surface Finish - Surfaces of material tested shall be clean and free from dirt, oil, and rough or loose scale. Regular shaped or semi-finished material shall have a surface finish of 250 micro-inches RMS or better.

4.2 When possible all ultrasonic testing shall be performed on material before special shapes such as grooves, channels and tapers are cut.

4.3 Material shall be tested by the contact method using the Sperry Reflectoscope with proper search units of conventional crystal size of 3/4 to 1-1/8" diameter. Angle crystals shall be used when applicable or necessary.

4.4 Inspection shall be performed in such manner as to provide complete coverage of material. This does not imply that all surfaces (inside diameters of Rings and Forgings) shall be scanned. The description of testing procedure for each type of forging is given under its respective section. (Sections 6.0, 7.0, 8.0, 9.0).

4.5 In scanning, the path of successive passes shall overlap by 10%. Scanning speed of crystal shall not exceed 6" per second.

4.6 Whenever possible or practical the test piece shall be rotated on a lathe, horizontal boring machine, or other suitable equipment to facilitate ultrasonic inspection.

4.7 When special testing procedures, not included in this specification, are recommended by the purchaser, these special procedures shall be agreed upon in advance by the vendor and purchaser.

5.0 Calibration Notch

5.1 When a calibration notch is necessary it shall be a notch formed by grinding or other methods which produce sides that are essentially parallel to each other and perpendicular to the test surface.

5.2 Calibration Notch Dimensions:

Depth - 3% of material thickness

Length - One inch

Width - Any convenient width not to exceed twice the depth. Minimum notch width in all cases shall be 1/32".

5.3 No calibration notches shall remain in the finished parts. Allowances shall be made in material dimensions for removal of notches after testing.

6.0 Ring Forgings

6.1 Testing Procedure - All ring forgings shall be tested by the longitudinal and angle beam methods.

6.1.1 Angle Beam - Testing shall be done from the outside diameter (tread) of the ring forging. The crystal shall be oriented in such a manner that the sound beam enters the ring forging in a circumferential direction.

6.1.2 Longitudinal Beam - Testing shall be done from both faces (ends) and the outside diameter (tread) of the ring forging. In all cases the sound beam will enter the material in a direction perpendicular to the test surface.

6.2 Calibration (Angle Beam)

6.2.1 Ring forgings with a wall thickness less than 20% of the I.D. - A test notch shall be formed axially along the outer surface of the ring parallel to the axis of rotation of the forgings. The search unit shall be placed on the outer surface and scanned until the notch indication is

at a maximum magnitude on either the first or second outside bounce, whichever is best seen on the scope. The instrument shall be adjusted until the indication from the test notch occupies at least 1/2 of the screen height. The magnitude of the notch indication shall be marked on the screen. Sound transmission into the ring shall be confirmed during the testing, either by permanent, steady reflection of the first revolution reflection on the screen or if this is not feasible, then the observation of the general screen pattern.

6.2.2 Ring forgings with a wall thickness of 20% or more of the I.D. - In addition to the O.D. test notch as described in Section 6.2.1, a similarly oriented test notch shall be formed on the I.D. of the ring forging. The I.D. test notch should be at least 90° away from the O.D. test notch. The search unit shall be placed on the outside surface of the ring and scanned until the maximum indications from the I.D. and O.D. test notches have been determined. The instrument shall be adjusted until the weakest test notch indication occupies one half or more of the screen height. The magnitude of the notch indications shall be marked on the screen. Sound transmission shall be confirmed as outlined in 6.2.1.

6.3 Interpretation of Test Results (Angle Beam) - Ring forgings shall be subject to review if the angle beam shows a defect whose indication exceeds in magnitude the indication from the calibrated notch. When both I.D. and O.D. test notches are used, the ring forging shall be subject to review when the angle beam shows a defect whose indications exceeds in magnitude the weakest indication from either the I.D. or O.D. test notches.

6.4 Calibration (Longitudinal Beam) - The transmission of sound into the ring shall be confirmed by observing the first end reflection at the right hand side of the sweep. The sensitivity of the instrument shall be adjusted until the height of this reflection occupies at least 3/4 of the screen height. When testing, a frequency of 2.25 mo shall be used, unless (1) grain coarseness is such that a lower frequency must be used or (2) the thickness of the material is such that the pulse length at 2.25 mo extends over 50% of the thickness of the material. In such cases a frequency of 5 mo shall be used. If at 5 mo the pulse length is still over 1/2 the thickness of the material, then the thickness is too small to be tested with conventional equipment.

6.5 Interpretation of Test Results (Longitudinal) - Ring forgings shall be subject to review if the longitudinal wave at the appropriate frequency, as outlined in 6.4 shows a defect whose indication exceeds 5% of the end reflection indication.

7.0 Disc or Pancake Forgings

7.1 Testing Procedure - Disc forgings shall be tested by the longitudinal wave method. Testing shall be done from the outside diameter and both faces.

7.2 Calibration - Sound transmission into the forging shall be confirmed by observing the first end reflection at the right hand side of the sweep. The sensivity of the instrument shall be adjusted until the height of this reflection occupies at least 3/4 of the screen height. When testing a frequency of 2.25 mo shall be used, unless (1) grain coarseness is such that a lower frequency must be used, or (2) the thickness of the material is such that the pulse length at 2.25 mo extends over 50% of the material thickness. In such cases a frecuency of 5 mc shall be used. If at 5 mo the pulse length is still over 1/2 the material thickness; then the thickness is too small to be tested with conventional equipment.

7.3 Interpretation of Test Results - Disc or pancake forgings shall be subject to review if the longitudinal beam at the appropriate frequency as outlined in 7.2 shows a defect whose indication exceeds 5% of the end reflection indication.

8.0 Water Boxes, Combination Channel and Tube Sheet Forgings (Combination of Disc and Ring Type Forging)

8.1 Inspection of material shall be performed as outlined in Sections 6.0 and 7.0.

8.1.1 Channel End - The testing procedure as outlined in Section 6.0 on Ring Forgings shall be followed when inspecting the Channel portion of the Forging.

8.1.2 Interpretation of test results shall be indentical to those as outlined in Section 6.0 under Ring Forgings.

8.1.3 Head End - The testing procedure as outlined in Section 7.0 on disc forgings shall be followed when inspecting the head portion of the forging.

8.1.4 Interpretation of test results of the head end shall be identical to those as outlined in Section 7.0 under Disc Forgings.

9.0 Shaft and Pinion Forgings

9.1 Testing Procedure - All shaft and pinion forgings shall be tested by the longitudinal beam method. The ultrasonic beam shall be introduced into the test material in a direction perpendicular to the longitudinal axis and perpendicular to the flat surfaces.

9.2 Testing Speed - The shaft forging shall be rotated continuously during testing at the approximate speeds in the Table below:

Diameter In.	Speed R.P.M.
51 - 60"	2.0
41 - 50"	2.5
31 - 40"	3.0
26 - 30"	4.0
21 - 25"	5.0
20" and less	6.0

9.3 Calibration (Longitudinal wave) - Transmission of the sound beam into the shaft shall be confirmed by observing the first back reflection or bore reflection at the right hand side of the sweep. The sensitivity of the instrument shall be adjusted until the height of this reflection occupies at least 3/4 of the screen height. When testing a frequency of 2.25 mc shall be used. Sensitivity settings shall be made at rotational speeds of Section 9.2.

9.4 Interpretation of Test Results - Shaft and pinion forgings shall be subject to review if the longitudinal beam at 2.25 mc shows a defect whose indication exceeds 5% of the back reflection. Indications shall be identified as to type whether traveling or stationary. A shaft forging with a loss of back reflection exceeding 20% in an indication free area shall also be subject to review.

10.0 Disposition

Any material subject to review as defined above if not accepted by the purchaser may be examined by the vendor or purchaser to determine actual extent and magnitude of defects. If examination reveals the defects to be inconsequential or non-detrimental in nature, the order will be considered complete and the cost of the examination and replacement shall be borne by the purchaser.

If examination reveals the material to be of rejectable quality, the responsibility for replacement shall be upon the vendor.

C- RATING SHEET

Included in this section is a commercial thermal rating sheet with specifications for the preliminary design.

Note that complete thermal calculations are included in APPENDIX 7-D.

967 088

REV'N.	DATE

ALCO PRODUCTS, INC.

EXCHANGER SPECIFICATION SHEET

			ALCO ORDER NO.
CUSTOMER	US ATOMIC ENERGY COMMISSION		CUST. REF. NO.
~~ADDRESS~~	CONTRACT AT(11-1)-666		ALCO INQUIRY NO.
PLANT LOCATION	UNKNOWN		DATE 1-15-59
SERVICE OF UNIT	INTERMEDIATE SODIUM HEAT EXCHANGER		ITEM NO. 1
SIZE 49-324	TYPE W1-21-80(SW)-V		CONNECTED IN SERIES PARALLEL
SURFACE PER UNIT 3750 EFF.	SHELLS PER UNIT 1		SURFACE PER SHELL 3750 EFF.

DESIGN CONDITION **PERFORMANCE OF ONE UNIT**

	SHELL SIDE	TUBE SIDE
FLUID CIRCULATED	PRIMARY SODIUM	SECONDARY SODIUM
TOTAL FLUID ENTERING LBS./HR.	1665000	1665000
VAPOR		
LIQUID	1665000	1665000
STEAM		
NON-CONDENSABLES		
FLUID VAPORIZED OR CONDENSED		
STEAM CONDENSED		
GRAVITY-LIQUID		
VISCOSITY-LIQUID		
MOLECULAR WEIGHT-VAPORS		
SPECIFIC HEAT-LIQUIDS	B.T.U./LB./°F.	B.T.U./LB./°F.
LATENT HEAT-VAPORS	B.T.U./LB.	B.T.U./LB.
TEMPERATURE IN	1200 °F	665 °F
TEMPERATURE OUT	723 °F	1140 °F
OPERATING PRESSURE – DESIGN	100 P.S.I.	125 P.S.I.
NUMBER OF PASSES PER SHELL	1	1
VELOCITY	2.3-4.5 FT./SEC.	9.27 FT./SEC.
PRESSURE DROP	13.98 P.S.I.	8.48 P.S.I.
FOULING RESISTANCE	—	—

HEAT EXCHANGED – B.T.U./HR. 239 000 000 M.T.D. (CORRECTED) 59.0

TRANSFER RATE – SERVICE 1080 CLEAN 1180

CONSTRUCTION

DESIGN PRESSURE	100 P.S.I.	125 P.S.I.
TEST PRESSURE	413 P.S.I.	517 P.S.I.
DESIGN TEMPERATURE	1200 °F.	1200 °F.

TUBES SA-213-TP316 NO. 1087* O.D. ½" BWG. 18 MIN. LENGTH 324" (STRAIGHT) PITCH ⊗ △ ◁ □ ◇

SHELL SA-240-M TYPE 316 (X-R) I.D. 49" THICKNESS .500"

SHELL COVER — FLOATING HEAD COVER —

CHANNEL SA-240-M TYPE 316 (X-R) HEMISPHERICAL 44" I.D. CHANNEL ~~COVER~~ MANHOLE SA-182-F316

TUBE SHEETS – STATIONARY SA-182-F316 FLOATING —

BAFFLES – CROSS DISC & DO-NUT TYPE — THICKNESS 1/4" (TOP DISC 1/2")

BAFFLES – LONG — TYPE — THICKNESS —

TUBE SUPPORTS — THICKNESS —

GASKETS RTJ SEAMLESS TYPE 316

CONNECTIONS – SHELL-IN 14" O.D. OUT 14" O.D. SERIES W.E. STD. WT.

CHANNEL-IN 14" O.D. OUT 14" O.D. SERIES W.E. STD. WT.

CORROSION ALLOWANCE – SHELL SIDE NONE TUBE SIDE NONE

CODE REQUIREMENTS ASME 1956 VIII TEMA CLASS R

WEIGHTS – EACH SHELL 35000 DRY BUNDLE FIXED FULL OF WATER 57200

NOTE: – INDICATE AFTER EACH PART WHETHER STRESS RELIEVED (S.R.) AND WHETHER RADIOGRAPHED (X-R)

REMARKS: – THICKNESS INDICATED IS SUBJECT TO CHANGE UPON FINAL ENGINEERING CHECK

* PLUS 1 TUBE FOR GAS BLEED

⊗ PITCH 1" X 1" CIRCULAR

ALSO – PROVIDE VENTS, DRAINS AND LIQ. LEVEL CONNECTIONS

7-C2

SALES DEPARTMENT WILL QUOTE ABOVE ON FORM 26A FORM 154 REV. C

057 089

ALCO ORDER NO.
ALCO ITEM NO.
CUST. ITEM NO.

EXCHANGER RATING SHEET

ENGINEER B.F. LOVE DATE 1-15-59 INQUIRY NO. US ATOMIC ENERGY COMMISSION CONTRACT AT(11-1)-666

SHELL SIDE FLUID PROPERTIES					MTD CALCULATION			TUBE SIDE FLUID PROPERTIES				
°API =		CALC. MEAN TEMP. = 962			1200 T_{s_1}	T_{t_2} 1140	60	CALC. MEAN TEMP. = 903		°API =		
TEMP.		CALC. MEAN		Tw	723 T_{s_2}	T_{t_1} 665	58	Tw		CALC. MEAN		TEMP.
SP. GR. LB./CF		51.5			477	TEMP. RANGE 475				52.0		SP. GR. LB./CF
SP. HT.		.301		$(z/z_w)^{.14}$	R =	P =	F =	$(z/z_w)^{.14}$		.302		SP. HT.
VISC. CP.		.562	(Lb/Ft Hr)		X = 59	CORR. LOG. MTD. ARITH.				.591	(Lb/Ft Hr)	VISC. CP.
K		38.2		$(z/z_w)^m$				$(z/z_w)^{.25}$		39.0		K
$K(\frac{cz}{k})^{1/3}$					TOTAL LBS. PER HR.							$K(\frac{cz}{k})^{1/3}$
SHELL SIDE DUTY					1665000		1665000			TUBE SIDE DUTY		
ARITH. MEAN TEMP. 962			SP. HT. .301				ARITH. MEAN TEMP. 903			SP. HT. .302		
1665000 x .301 x 477 = 239 000 000							1665000 x .302 x 475 = 239 000 000					

HEAT TRANSFER

SHELL SIDE				TUBE SIDE		
F.D.-	Dv-	A-	N.F.A.-	TUBES/PASS- 1087	FT/SEC- 9.27	L/D- 650

Gx = ——— Gl = ———

$G = \frac{1665000 \times 144}{1087 \times .127 \times 3600} = 482$ Lb/Ft²Sec

R_{e_x} = ——— Gm = $\sqrt{\quad}$

$R_e = \frac{.402 \times 482 \times 3600}{12 \times .591} = 98{,}300$ (1)

R_{e_m} = — SEE CALCULATIONS IN APPENDIX —

~~Pr~~ $Pr = \frac{.591 \times .302}{39.0} = .00457$ (1)

hs = SUMMARY

$Nu = 0.625 (98{,}300 \times .00457)^{.4}$

$Nu = 7.18$ (1)

$h_t = \frac{12 \times 7.18 \times 39.0}{.402} = 8370$

$r_t = \frac{1}{8370} \times \frac{.500}{.402} = .000149$

REGION	EFF SURFACE	U
1	712	750
2	2670	1285
3	368	1250
TOTAL	3750	1180 (2) AVG.

SHELL SIDE P.D. B=

ΔP_x = — SEE CALCULATIONS — SUMMARY

ΔP_t — REGION - 1 0.77; REGION - 2 11.19; REGION - 3 2.02

h_s - SEE CALCS. h_t - 8370

U CLEAN CALC. - 1180 (2)

U CLEAN REQ'D. - 1080

U OVERALL FOULING - —

U - 1080 SERVICE RATE

$S = \frac{239000000}{59.0 \times 1080} = 3750$

EFF. SHELL SURFACE	3750	EFF. UNIT SURFACE	3750
GR. SHELL SURFACE	3750	GR. UNIT SURFACE	3750
15.0	P.D. ALLOW		15.0
13.98	P.D. ~~CALC.~~		8.48

TUBE SIDE P.D.

FRICTION FACTOR - $.046/(Re)^{0.2} = .00462$

~~Outside~~ $K_e = .82$ $K_c = .39$

$\Delta P_i = .487\,K = .487(.82 + .39) = .59$

$\Delta P_t = 1530\,f = 1580(.00462) = 7.30$

$\Delta P_o = .487\,K = .487(.82 + .39) = .59$

NO. CROSS BAFFLES	SPACING	~~NFA~~	BAFFLE CUT
14 DISC ~~SEMI~~	9.0"	40" O.D.	CIR. ~~HORIZ.~~
14 DO-NUT ~~QUARTER~~	9.0"	20⅛" I.D.	~~VERT.~~

SEAL STRIPS NO. NONE SIZE ¼" x ——— MATL. ———

NO. TUBE SUPPORTS (— FULL — SF — HALF — QUART. + — FLTG. HD. U-BEND)

TUBE SHEET TK. @ 1200 °F METAL t = REF. MECH. DES. + = $3\frac{1}{2}$" BASIC ~~QUOTE~~

SHELL TK. @ 1200 °F METAL t = REF. MECH. DES. + = $\frac{1}{2}$" BASIC ~~QUOTE~~

SPECIAL REQUIREMENTS & NOTES

TUBE LAYOUT CIR ~~RTHN~~ ~~SPECIAL~~ 39 5/8 OTL

TUBE METAL RESISTANCE

$K_m = 11.97$ @ $T_m = 932$ F

$r_m = \frac{.500 - .402}{2 \times 12 \times 11.97} \times \frac{2 \times .500}{.500 + .402}$

$r_m = .000378$

NOTES

(1) DIMENSION LESS

(2) AVERAGE CALC. RATE BASED ON SURFACE

TEST RING

STACKING

( ) SETS SPARE GASKETS

CHKD.	PREL.			THERM.			FINAL		

T-C8

D- THERMAL DESIGN CALCULATIONS

INTRODUCTION

In general these calculations confirm and refine the basic size and configuration selected during the concept study of this program. This includes in detail a heat transfer rating and fluid pressure drop analysis for full load operation (design conditions), together with the predicted thermal performance at part load operation.

007 091

NOMENCLATURE

A_E	Equivalent heat transfer surface (ft^2)
A_i	Inside area (in^2)
A_F	Flow area (in^2)
C_p ①②	Specific heat at constant pressure ($Btu/lb\ ^oF$)
D_1	Outside diameter, center spacer (in)
D_2	Inside diameter, vertical shroud (in)
D_b, D_R	Outside diameter of disc & ring type baffles (in)
D_E	Equivalent diameter of tube bundle (in)
D_{MTC}, D_{OTC}	Tube circle diameter, minimum & maximum (in)
D_o, D_i	Tube outside & inside diameter (in)
D_S	Inner shell diameter (in)
f_L	Friction factor, axial-flow
f_x	Friction factor, cross-flow
G ①②	Mass velocity, ($Lb/Ft^2 sec$)
G_x	Max. mass velocity in cross-flow, ($Lb/Ft^2 sec.$)
g_c	Gravity constant, $32.2\ ft/sec^2$
h ②	Film heat transfer coefficient, ($Btu/Ft^2 Hr\ ^oF$)
k ②	Thermal conductivity, ($Btu/Ft\ Hr\ ^oF$) (k_m = tube metal)
K_e, K_c	Pressure loss coefficient, expansion & contraction
L' ③	Effective tube length (in.)
L_F	Total flow length between tubes in a row (in)
N.F.A.	Net free area of tube bundle (in^2)
N_T	Number of tubes
N_{TC}	Number of tube rows crossed
NTU	Number of thermal units
Nu ②	Nusselt Number, dimensionless
P_1, P_2	Tube pitch, radial & circumferential (in)
P_B	Baffle pitch (in)
P_r ②	Prandtl Number, dimensionless
ΔP ②	Pressure drop, psi
ΔP_n	Pressure drop, psi

$n = e$, expansion $n = c$, contraction $n = L$, axial flow
$n = x$, cross flow $n = wi$, inside baffle window
$n = wo$, outside window

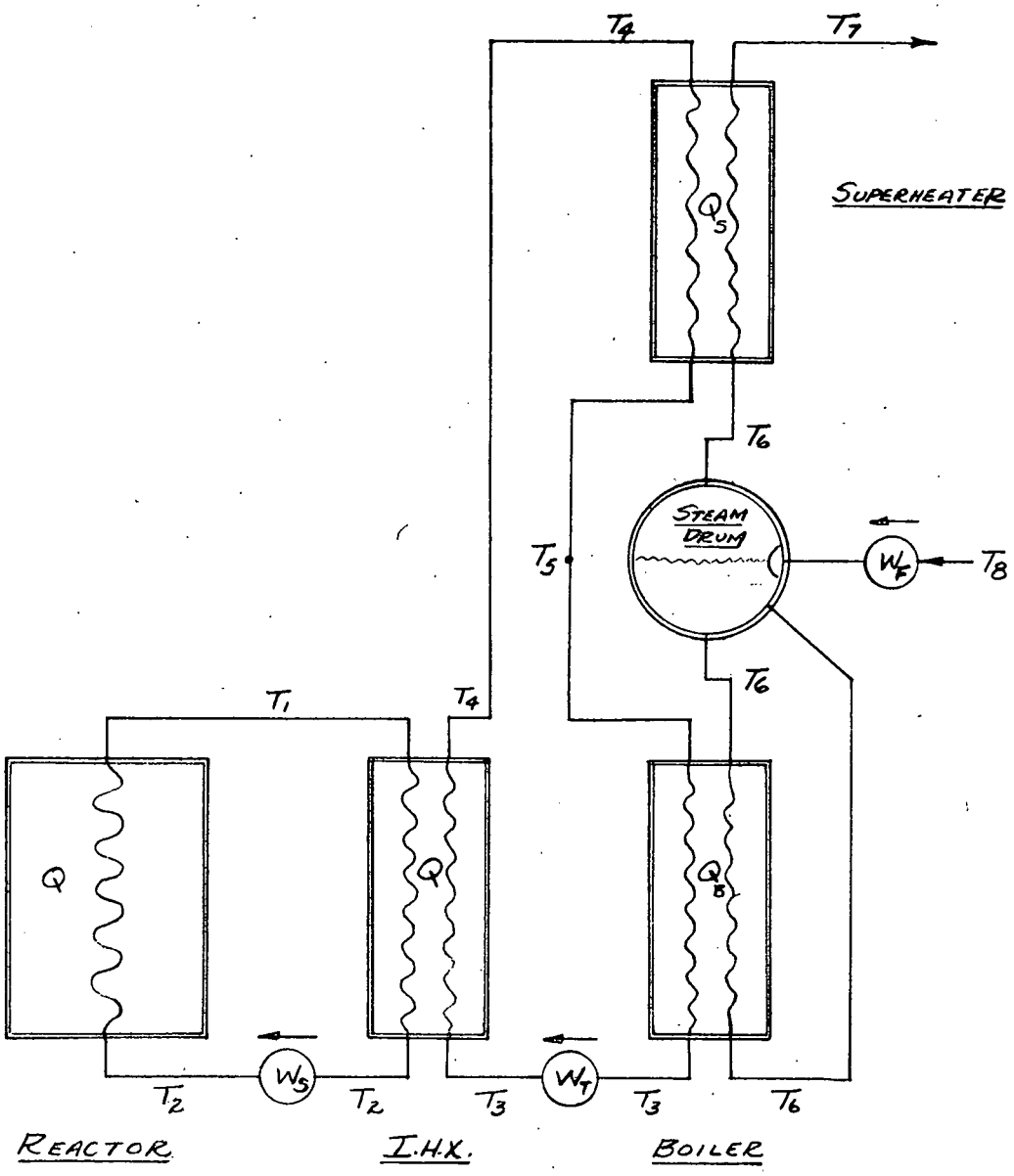
OOT 092

Q	③	Heat load, (Btu/Hr)
		(Q_S = Superheater, Q_B = Boiler)
q		Local heat transfer, (Btu/Hr)
f_r	②	Film resistance, ($Ft^2 Hr\,^oF/Btu$)
		(r_m = tube metal)
Σr		Summation of heat resistances in series
Re	①	Reynolds Number, dimensionless
Re_x		Reynold's Number, normal cross flow, dimensionless
S_E	③	Total effective heat transfer surface (ft^2)
T	②	Fluid temperature (F)
T_m	①②	Arith mean fluid temperature (F)
ΔT	②	Fluid temperature difference (F)
$(\Delta T)_m$		Log mean temperature difference (F)
U	③	Overall heat transfer coefficient ($Btu/Ft^2 Hr\,^oF$)
		(U_{avg} - average, U_{req} - required, U_E - equivalent)
V	①	Fluid velocity (Ft/sec)
W	①②	Flow rate (lb/hr)
		(W_F - inlet water flow)
ϵ		Effectiveness
ρ	②	Density (lb/Ft^3)
μ	②	Absolute viscosity ($lb/Ft\,sec$)

REF: ① Ref. Fluid Locations - FLOW DIAGRAM
② S - shell, T - Tube
③ 1, 2, 3 REGIONS - INTERNAL CONFIGURATION

907 093

FLOW DIAGRAM



7-D4

THERMAL DESIGN CONDITIONS

1. CONTRACT SPECIFICATIONS, AT(11-1)-666

$T_1 = 1200\ F$
$T_2 = 700 - 900\ F$
$T_3 = 650 - 850\ F$
$T_4 = 1140\ F$
$T_7 = 1050\ F$ @ 2200 psig.
$T_8 \cong 600\ F$

$Q \cong 2.39 \times 10^8 \frac{Btu}{Hr}$

2. DESIGN CONDITIONS (100% LOAD)

a. Steam Conditions: Ref. "Thermodynamic Properties of Steam" by Keenan & Keyes

Pressure psia	Sat Temp °F	Enthalpy Sat. Vapor	Enthalpy @ 1050	Enthalpy 600 F Liq.
2200	649.46	1119.2	1500.0	614.5
2214.7	650.41	1117.9	1499.6	613.5
2300	655.91	1110.4	1497.1	612.1

Thus: $T_6 = 650\ F$

b. From Basic Analysis of Steam Generator: Ref. Vol. II

$T_3 = 665\ F$

NOTE: This temperature gives a 15 F terminal difference at boiler feedwater inlet and sodium outlet.

967 095

c. For $W_S = W_T$

$T_2 = 723\ F$

Thus:

$$W_S = \frac{Q}{(C_p)_{1-2}\ (\Delta T)_{1-2}}$$

$(T_m)_{1-2} = 961.5\ F$

$\therefore (C_p)_{1-2} = .301\ \frac{Btu}{Lb\ ^\circ F}$

$(\Delta T)_{1-2} = 477\ F$

So: $W_S = 1.665 \times 10^6\ \frac{Lb.}{Hr.}$

And

$$W_T = \frac{Q}{(C_p)_{3-4}\ (\Delta T)_{3-4}}$$

$(T_m)_{3-4} = 902.5\ F$

$\therefore (C_p)_{3-4} = .302\ \frac{Btu}{Lb\ ^\circ F}$

$(\Delta T)_{3-4} = 475\ F$

So $W_T = 1.665 \times 10^6\ \frac{Lb.}{Hr.}$

d. To determine W_F

Enthalpy Rise of Water $= 1499.6 - 613.5 = 886.1\ \frac{Btu}{Lb.}$

$$W_F = \frac{Q}{886.1}$$

$\therefore W_F = 2.70 \times 10^5\ \frac{Lb}{Hr}$

OCT 096

e. To Determine Q_B & Q_{SH}

Enthalpy Rise of Water Through Boiler = $1117.9 - 613.5 = 504.4 \frac{Btu}{Lb}$

$$Q_B = \frac{504.4}{886.1} Q$$

$$\therefore Q_B = 1.36 \times 10^8 \frac{Btu}{Hr}$$

Enthalpy Rise of Steam Through Superheater = $1499.6 - 1117.9 = 381.7 \frac{Btu}{Lb}$

$$Q_{SH} = \frac{381.7}{886.1} Q$$

$$\therefore Q_{SH} = 1.03 \times 10^8 \frac{Btu}{Hr}$$

f. To Determine T_5

$$Q_{SH} = W_T (Cp)_{4-5} (\Delta T)_{4-5}$$

$$\text{or } T_5 = T_4 - \frac{Q_{SH}}{W_T (Cp)_{4-5}}$$

$T_4 = 1140$ F

$(Cp)_{4-5} = .300 \frac{Btu}{Lb\ ^\circ F}$

$(T_m)_{4-5} = 1037$ F

$$\therefore T_5 = 934 \text{ F}$$

g. To Check T_3

$$T_3 = T_5 - \frac{Q_B}{W_T (Cp)_{5-3}}$$

$(Cp)_{5-3} = .304 \frac{Btu}{Lb\ ^\circ F}$

$(T_m)_{5-3} = 800$ F

$$\therefore T_3 = 665 \text{ F}$$

967 097

h. Log Mean Temperature Difference (COUNTER-FLOW)

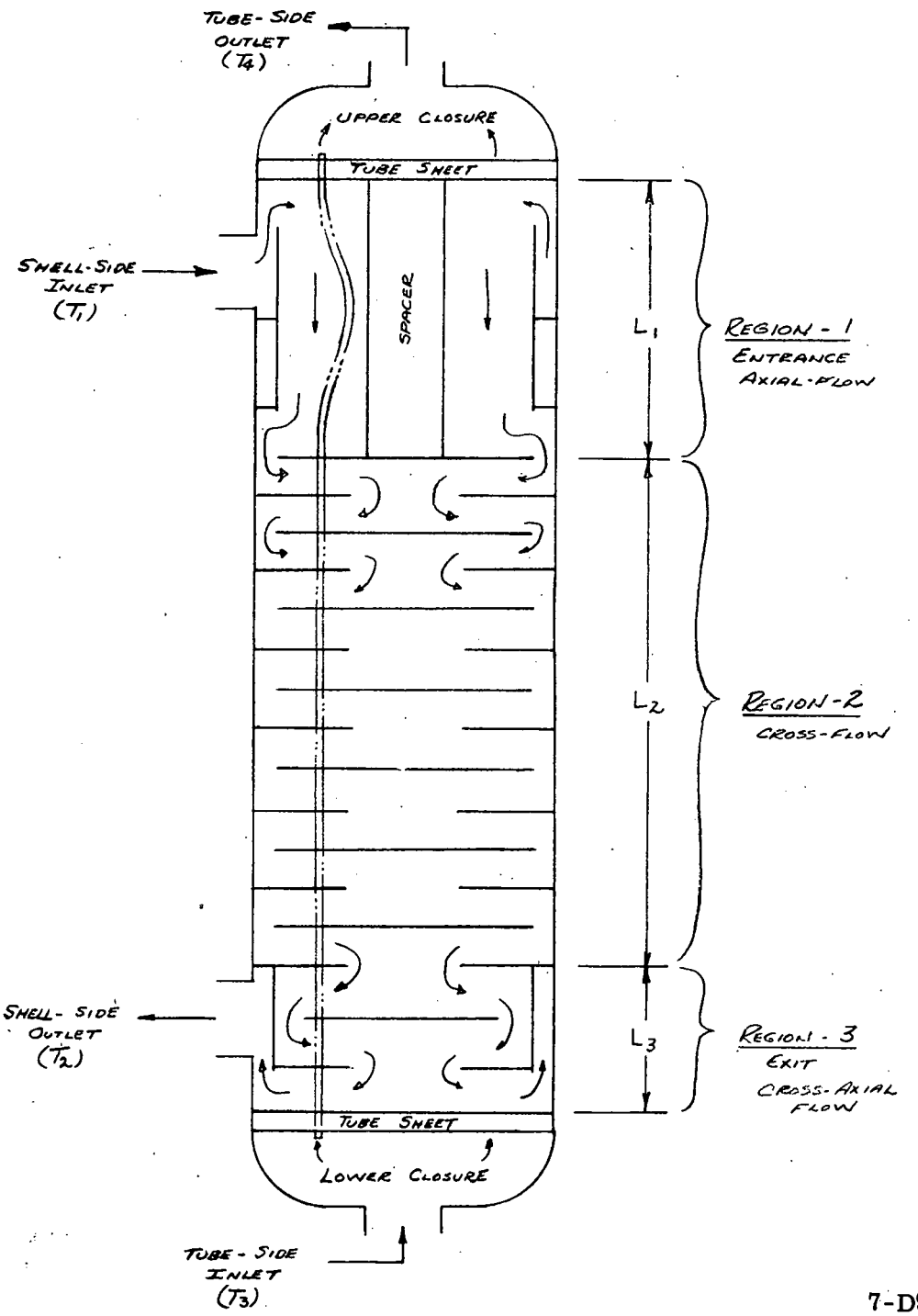
$$(\Delta T)_m = \frac{(T_1 - T_4) - (T_2 - T_3)}{\ln\left[\frac{T_1 - T_4}{T_2 - T_3}\right]}$$

$$(\Delta T)_m = \frac{(1200-1140)-(723-665)}{\ln\left[\frac{1200-1140}{723-665}\right]} = \frac{2.0}{\ln 1.0345} = \frac{2.0}{.0339}$$

$\therefore$ $(\Delta T)_m = 59$ F

037 098

INTERNAL CONFIGURATION



7-D9

SCT 099

FLUID PROPERTIES

READ FROM FOLLOWING GRAPHS.
SHEET NOS. 13–16

1. SHELL - SIDE SODIUM

$T_1 = 1200\ F$, $T_2 = 723\ F$

@ $(Tm)_{1-2} = 962\ F$

a. $\rho_S = 51.5\ Lb/Ft^3$

b. $\mu_S = 0.156 \times 10^{-3}\ Lb/Ft.Sec.$

c. $k_S = 38.2\ Btu/Ft\,Hr\,°F$

d. $Cp_S = 0.301\ Btu/Lb\,°F$

2. TUBE - SIDE SODIUM

$T_3 = 665\ F$, $T_4 = 1140\ F$

@ $(Tm)_{3-4} = 903\ F$

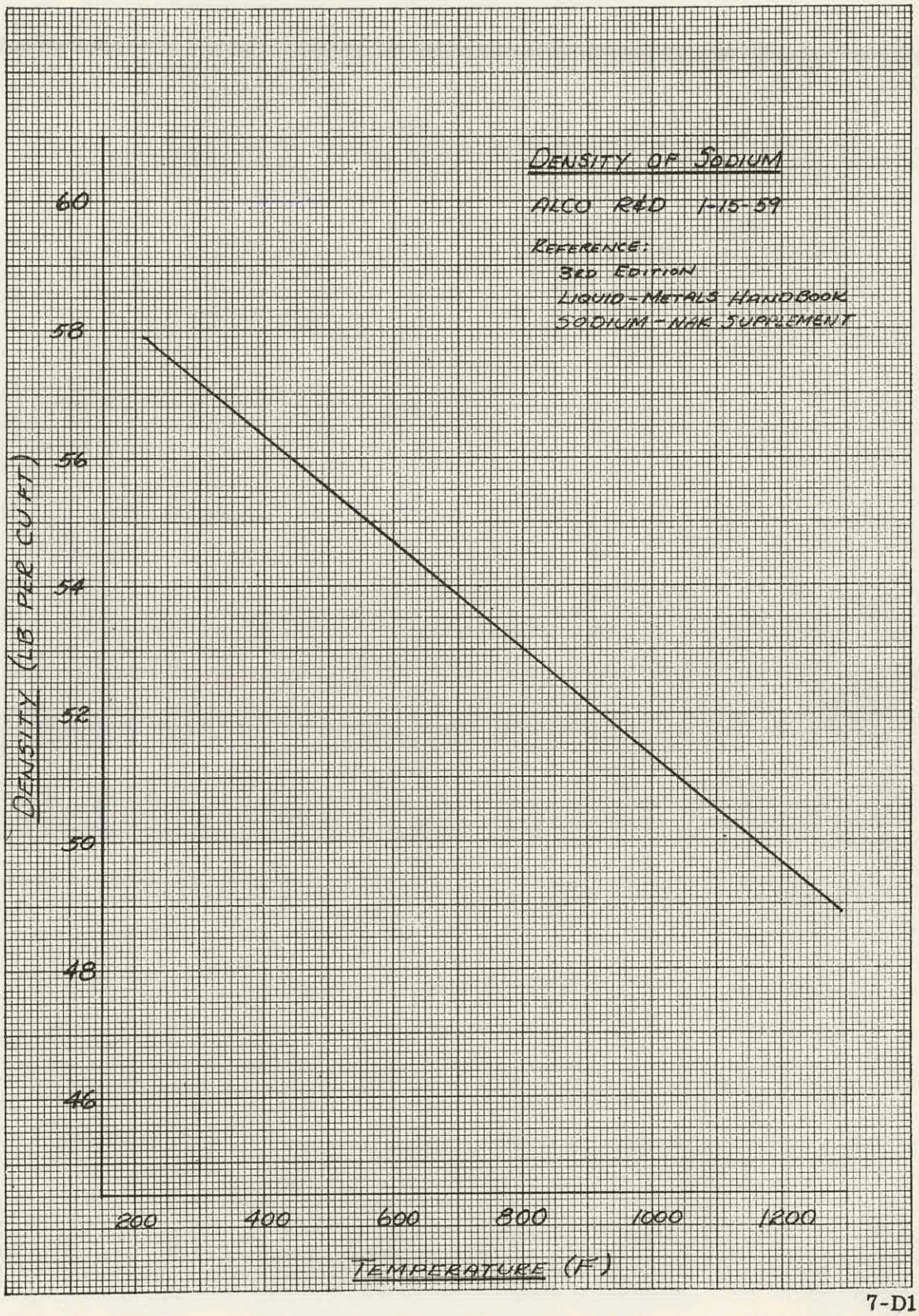
a. $\rho_T = 52.0\ Lb/Ft^3$

b. $\mu_T = 0.164 \times 10^{-3}\ Lb/ft.Sec.$

c. $k_T = 39.0\ Btu/Ft.Hr\,°F$

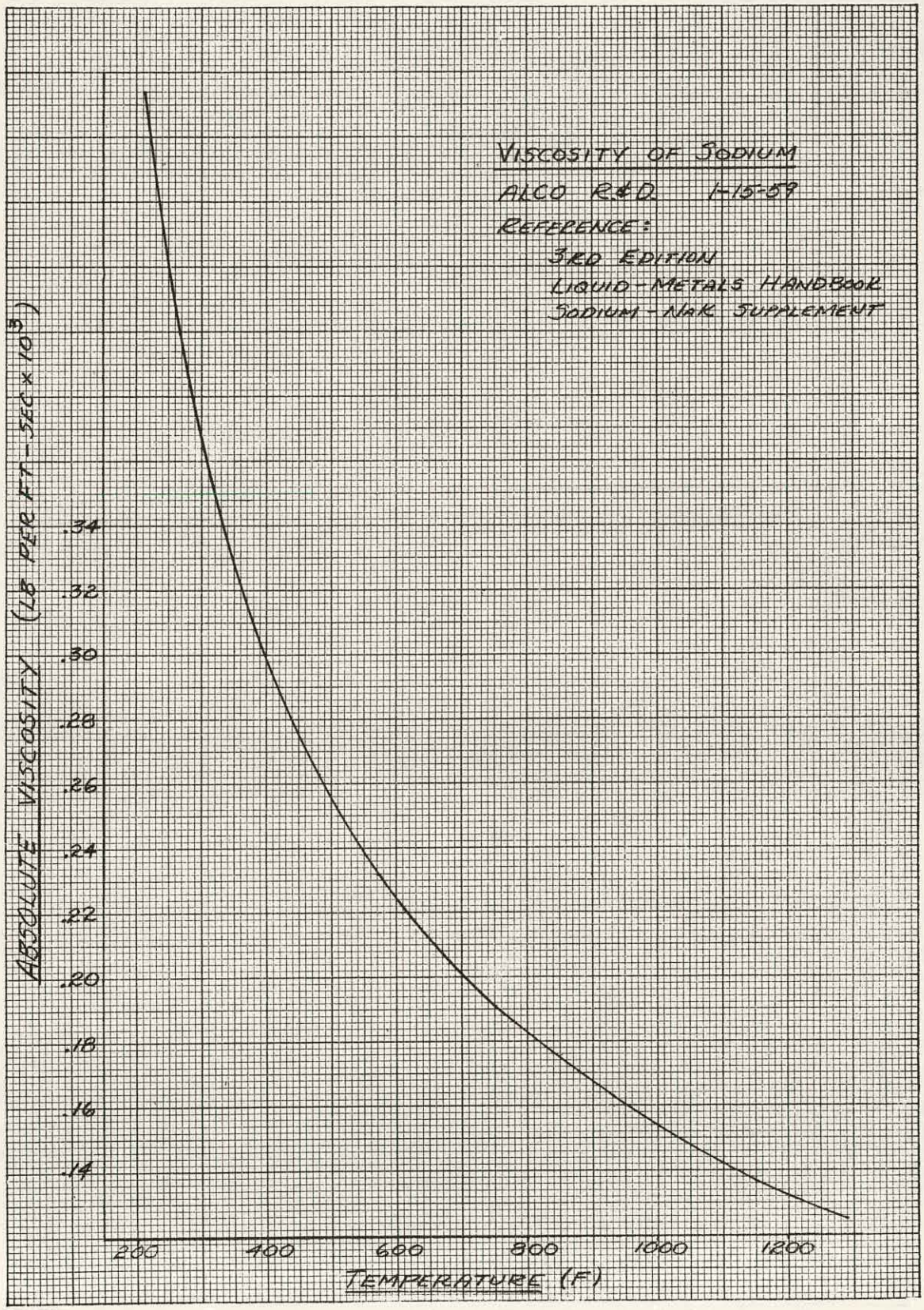
d. $Cp_T = 0.302\ Btu/Lb\,°F$

CSK 100


DENSITY OF SODIUM
ALCO R&D 1-15-59
REFERENCE:
3RD EDITION
LIQUID-METALS HANDBOOK
SODIUM-NAK SUPPLEMENT
60
58
56
54
52
50
48
46
DENSITY (LB PER CU FT)
200
400
600
800
1000
1200
TEMPERATURE (F)

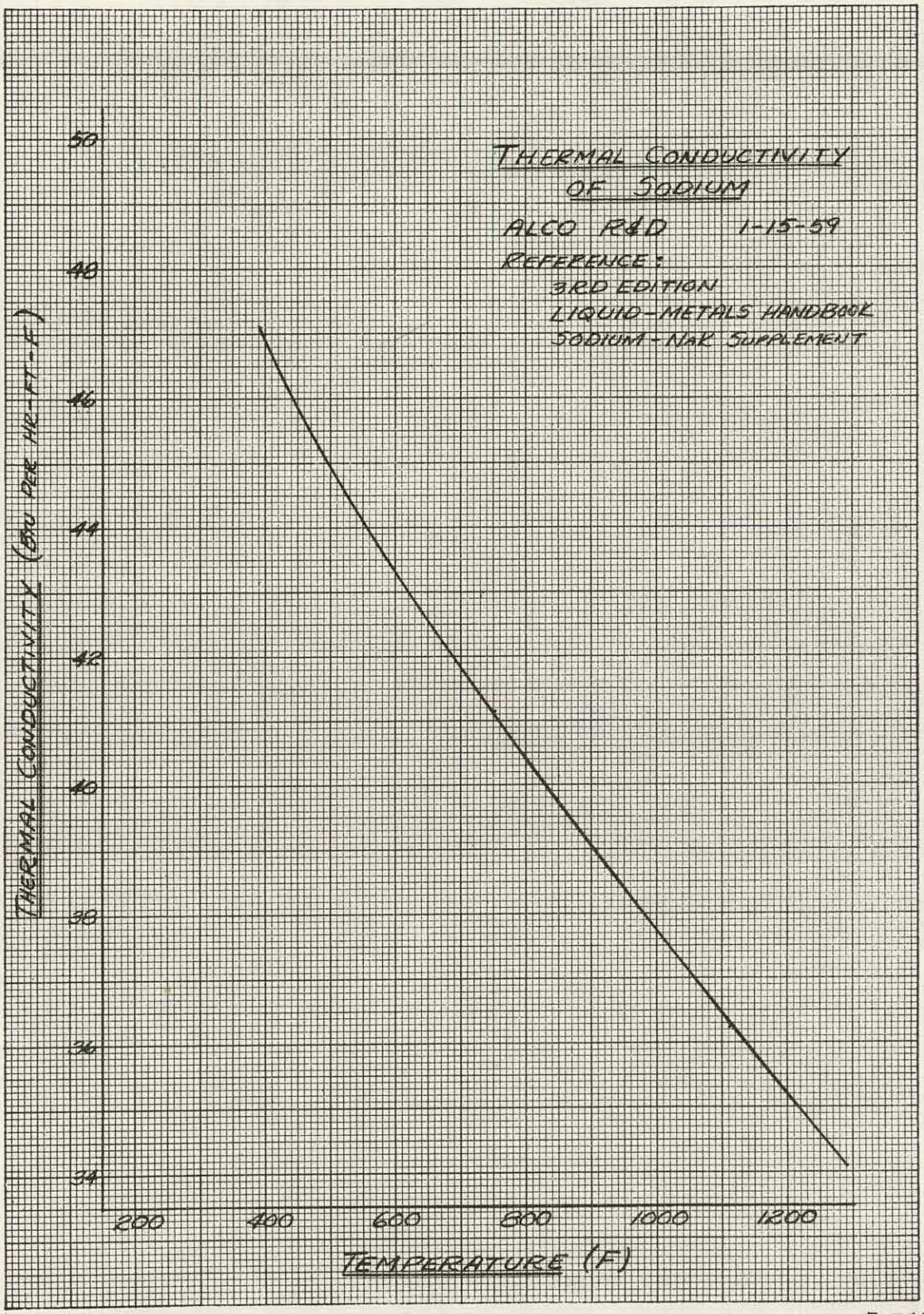
7-D11

967 101

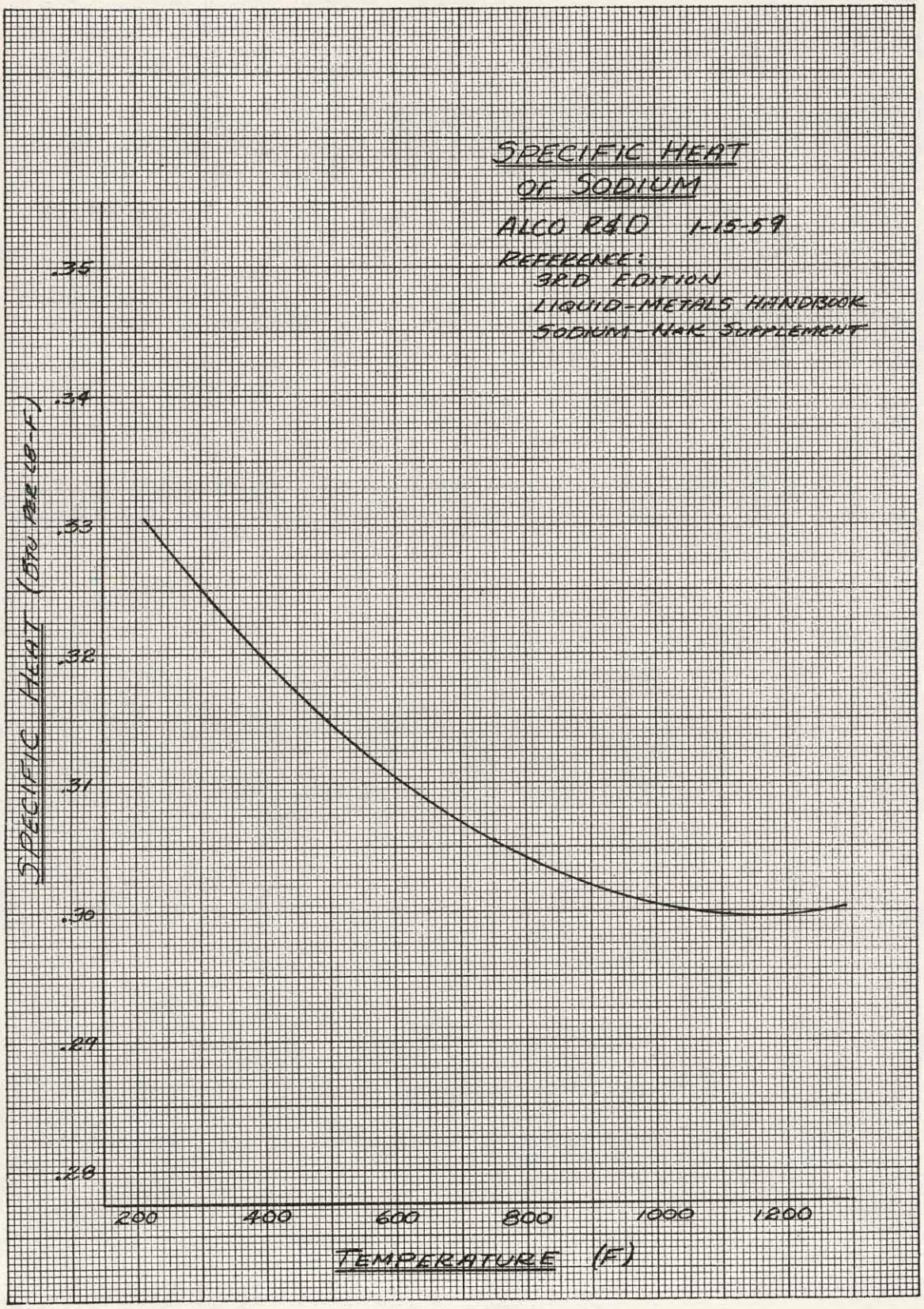

VISCOSITY OF SODIUM
ALCO R&D 1-15-59
REFERENCE:
3RD EDITION
LIQUID-METALS HANDBOOK
SODIUM-NAK SUPPLEMENT
ABSOLUTE VISCOSITY (LB PER FT-SEC x 10^5)
.34
.32
.30
.28
.26
.24
.22
.20
.18
.16
.14
200
400
600
800
1000
1200
TEMPERATURE (F)

7-D12

967 102


THERMAL CONDUCTIVITY OF SODIUM
ALCO R&D 1-15-59
REFERENCE:
3RD EDITION
LIQUID-METALS HANDBOOK
SODIUM-NaK SUPPLEMENT
THERMAL CONDUCTIVITY (BTU PER HR-FT-F)
50
48
46
44
42
40
38
36
34
200
400
600
800
1000
1200
TEMPERATURE (F)

967 103


SPECIFIC HEAT
OF SODIUM
ALCO R&D 1-15-59
REFERENCE:
3RD EDITION
LIQUID-METALS HANDBOOK
SODIUM-NAK SUPPLEMENT
SPECIFIC HEAT (BTU PER LB-F)
.35
.34
.33
.32
.31
.30
.29
.28
200
400
600
800
1000
1200
TEMPERATURE (F)

7-D14

HEAT TRANSFER ANALYSIS

1. TUBE-SIDE RESISTANCE

ESTABLISHED CONFIGURATION (CONCEPT STUDY, VOL. III)

TUBE SIZE: $D_o = .500"$

$D_i = .402"$

CIRCULAR PITCH: CIRCUMFERENTIAL $P_1 = 1.00"$

RADIAL $P_2 = 1.00"$

MINIMUM TUBE CIRCLE: $D_{MTC} = 15.125"$

OUTER TUBE CIRCLE: $D_{OTC} = 39.125"$

NUMBER OF TUBES: $N_T = 1087$ EFFECTIVE

$$A_i = \frac{\pi (D_i)^2}{4} = 0.127 \text{ in}^2$$

$$G_T = \frac{W_T}{25\, A_i\, N_T} = \frac{1.665 \times 10^6}{25(0.127)1087} = 482 \text{ Lb/Ft}^2 \text{ Sec.}$$

$$V_T = \frac{G_T}{\rho_T} = \frac{482}{52.0} = 9.27 \text{ Ft/Sec.}$$

$$Re_T = \frac{D_i\, G_T}{12\, \mu_T} = \frac{.402(482)}{12\,(.164 \times 10^{-3})} = 98{,}300$$

$$Pr_T = 3600\, \frac{\mu_T\, C_{pT}}{k_T} = 3600\, \frac{.164 \times 10^{-3}\,(.302)}{39.0} = .00457$$

$$Nu_T = 0.625\, (Re_T\, Pr_T)^{.4}$$ "LUBARSKY & KAUFMAN EQUATION"

REF. NACA REPT. 1270 (1956)

$$Nu_T = 0.625\,[(98{,}300 \times .00457)]^{.4} = 7.18$$

$$h_T = \frac{12\, Nu_T\, k_T}{D_i} = 8370 \text{ Btu/Ft}^2 \text{ Hr } ^\circ\text{F}$$

$$r_T = \frac{1}{h_T}\left(\frac{D_o}{D_i}\right) = \frac{1}{8370}\left(\frac{.500}{.402}\right)$$

$$\therefore\ r_T = .000149 \text{ Ft}^2 \text{ Hr } ^\circ\text{F/Btu}$$

7-D15

2. Tube Metal Resistance

Tube size: $D_o = .500''$

$D_i = .402''$

Tube material - 316 S.S.

$k_m = 11.97$ Btu/Ft Hr °F @ 932 F (Mean metal temp)

$$r_m = \frac{D_o - D_i}{2(12)\,k_m}\left(\frac{2D_o}{D_o + D_i}\right)$$

or $$r_m = \frac{D_o}{12\,k_m}\left(\frac{D_o - D_i}{D_o + D_i}\right) = \frac{.500}{12(11.97)}\left(\frac{.500 - .402}{.500 + .402}\right)$$

$$\therefore \quad r_m = .000378 \quad Ft^2\,Hr\,°F/Btu$$

3. Shell-Side Resistances

a. Region-1 (Axial-Flow)

Center Spacer (O.D.); $D_1 = 14.00''$

Vertical Shroud (I.D.); $D_2 = 40.00''$

$$N.F.A. = \frac{\pi}{4}\left[(D_2)^2 - (D_1)^2 - (N_T + 1)(D_o)^2\right]$$

$$N.F.A. = \frac{\pi}{4}\left[(40)^2 - (14)^2 - 1088\,(.500)^2\right] = 900\ in^2$$

$$D_E = \frac{4\ N.F.A.}{N_T\,\pi\,D_o} = \frac{4\,(900)}{1087\,\pi(.500)} = 2.10\ in.$$

$$G_s = \frac{W_s}{25\ N.F.A.} = \frac{1.665 \times 10^6}{25\,(900)} = 74\ Lb/Ft^2\,Sec.$$

$$V_s = \frac{G_s}{\rho_s} = \frac{74}{51.5} = 1.44\ Ft/sec.$$

$$Re_s = \frac{D_E\,G_s}{12\,\mu_s} = \frac{2.10\,(74)}{12\,(.156 \times 10^{-3})} = 83{,}000$$

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3 a. (CONT.)

$$Pr_s = 3600 \frac{C_{p_s} \mu_s}{k_s} = 3600 \frac{.301(.156 \times 10^{-3})}{38.2} = .00443$$

$$Nu_s = 0.106 \left(D_E Re_s Pr_s\right)^{.6}$$

UNBAFFLED SHELL-SIDE HEAT TRANSFER - LARGE SODIUM EXCHANGER

REF. "LIQUID-METALS HANDBOOK" SODIUM-NaK SUPPLEMENT

NOTE: Refer to page 285, "LIQUID-METALS HANDBOOK", the above equation is unbaffled modification of equation (4) for a relatively large exchanger. This modification was made similar to the modification of equation (1) to get equation (5).

$$Nu_s = 0.106\left[(2.10)(83{,}000)(.00443)\right]^{.6} = 5.72$$

$$h_s = \frac{12\, Nu_s\, k_s}{D_E} = \frac{12(5.72)\,38.2}{2.10} = 1250 \ \text{Btu/Ft}^2\text{Hr}^\circ\text{F}$$

$$r_s = \frac{1}{h_s} = \frac{1}{1250}$$

$$\therefore \quad r_s = .000800 \ \text{Ft}^2\text{Hr}^\circ\text{F/Btu}$$

b. REGION - 2 (CROSS-FLOW)

DISC & DOUGHNUT TYPE BAFFLES

DISC O.D. = D_o = 40.0"

RING I.D. = D_R = 20.125"

BAFFLE THICKNESS = 0.25"

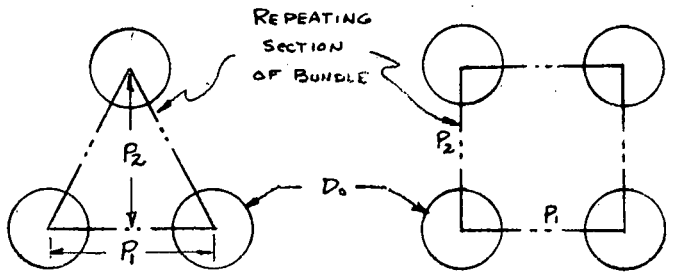
BAFFLE PITCH = P_B = 9.0"

SHELL: O.D. = 50", I.D. = 49"

INNER SHIELD I.D. = D_s = 48.50"

3.6 (CONT.)

TUBE ARRANGEMENT FULL-STAGGER TO IN-LINE



$P_1 = 1.0"$

$P_2 = 1.0"$

$D_o = .500'$

$N.F.A._1 = .402\ in^2$ $\qquad$ $N.F.A._2 = .804\ in^2$

$$D_E = \frac{4\ N.F.A.}{\pi\ D_o/2} \qquad D_E = \frac{4\ N.F.A._1}{\pi\ D_o}$$

$\therefore\ D_E = 2.05"$ $\qquad$ $\therefore\ D_E = 2.05"$

$$Re_s = \frac{D_E\ G_s}{12\ \mu_s}\ ;\quad G_s = \frac{W_s}{25\ L_F(P_B - .25)}$$

where: L_F = Total Flow Length Between Tubes in a Row (inches)

P_B = Baffle-Pitch (inches)

So

$$Re_s = \frac{D_E\ W_s}{300\ \mu_s\ L_F\ (P_B - .25)}$$

$$Re_s = \frac{(2.05)(1.665 \times 10^6)}{300\ (.156 \times 10^{-8})}\ \frac{1}{L_F(P_B - .25)}$$

$$\therefore\ Re_s = 7.30 \times 10^7\ \frac{1}{L_F(P_B - .25)}$$

3.6 (CONT.)

$$Pr_s = 3600 \frac{C_{p_s} \mu_s}{k_s} = 3600 \frac{(.301)(.156 \times 10^{-3})}{38.2}$$

$$\therefore Pr_s = .00443$$

$$Nu_s = \frac{h_s D_E}{12 k_s} = \frac{2.05}{12(38.2)} h_s = .00447 h_s$$

BUT: $Nu_s = .212 \left(D_E Re_s Pr_s \right)^{.6}$ BAFFLED SHELL-SIDE HEAT TRANSFER.

REF. "LIQUID-METALS HANDBOOK", SODIUM-NAK SUPPLEMENT

BY SUBSTITUTION:

$$h_s = \frac{.212}{.00447} \left[2.05 (7.30 \times 10^7) \frac{1}{L_F (P_B - .25)} (.00443) \right]^{.6}$$

$$h_s = 1.47 \times 10^5 \left[\frac{1}{L_F (P_B - .25)} \right]^{.6}$$

$$r_s = \frac{1}{h_s}$$

$$\therefore r_s = 6.80 \times 10^{-6} \left[L_F (P_B - .25) \right]^{.6}$$

FOR $P_B = 9.0''$

$$r_s = 2.5 \times 10^{-5} (L_F)^{.6}$$

C. REGION-3 (CROSS-AXIAL-FLOW)

FOR $P_B = 10.50''$, FROM ABOVE

$$r_s = 2.79 \times 10^{-5} (L_F)^{.6}$$

4. Overall Rates

a. Region -1

$r_T = .000149$
$r_m = .000378$
$r_s = .000800$

$\Sigma r = .001327$

$$U_1 = \frac{1}{\Sigma r} = 750 \ \text{Btu/Ft}^2\text{Hr}^\circ\text{F}$$

b. Region -2

$r_T = .000149$
$r_m = .000378$

$\Sigma r = .000527 + r_s$
$r_s = 2.5 \times 10^{-5} (L_F)^{.6}$

Row	N_T (EFF)	L_F	$(L_F)^{.6}$	$r_s \times 10^6$	$\Sigma r \times 10^6$	U	$N_T U \times 10^2$
1	48	23.5	6.65	DOUGHNUT OPENING		1365	655
2	54	26.8	7.20			1365	737
3	60	30.1	7.70			1365	820
4	66 (62)	33.3	8.20	205	732	1365	845
5	72	36.7	8.70	218	745	1340	965
6	78	39.9	9.15	229	756	1320	1030
7	84	43.2	9.60	240	767	1305	1095
8	90	46.5	10.00	250	777	1285	1160
9	96	49.8	10.40	260	787	1270	1220
10	102	53.0	10.80	270	797	1255	1280
11	108	56.3	11.20	280	807	1240	1340
12	114	59.8	11.60	290	817	1225	1395
13	120 (119)	63.0	12.00	300	827	1210	1440
	(1087)						13,982

$$(U_2)_{avg.} = \frac{13,982}{1087} = 1285 \ \text{Btu/Ft}^2\text{Hr}^\circ\text{F}$$

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4. C. REGION - 3

$r_T = .000149$

$r_m = .000378$

$\Sigma r = .000527 + r_3$

$r_3 = 2.79 \times 10^{-5} (L_F)^{.6}$

ROW	N_T (EFF)	$(L_F)^{.6}$	$r_3 \times 10^6$	$\Sigma r \times 10^6$	U	$N_T U \times 10^2$
1	48	6.65	DOUGH NUT OPENING		1320	633
2	54	7.20			1320	713
3	60	7.70			1320	792
4	66 (62)	8.20	229	756	1320	818
5	72	8.70	242	769	1300	935
6	78	9.15	255	782	1280	987
7	84	9.60	268	795	1260	1060
8	90	10.00	279	806	1240	1115
9	96	10.40	290	817	1225	1175
10	102	10.80	DISC OPENING		1225	1250
11	108	11.20			1225	1325
12	114	11.60			1225	1398
13	120 (119)	12.00			1225	1460
	(1087)					13,661

$$(U_3)_{avg} = \frac{13661}{1087} = 1250 \quad Btu/Ft^2\,Hr\,^oF$$

5. Required Tube Length

a. Region - 1

To provide for Sine wave bend

$L_1 \geq 60''$

$$\text{Thus, } SE_1 = \frac{N_T \pi D_o L_1}{144} = \frac{1087(\pi).5(60)}{144}$$

$$SE_1 = 712 \text{ Ft}^2$$

Since, $U_1 = 750 \text{ Btu/Ft}^2\text{Hr}^\circ\text{F}$

$(\Delta T)_m = 59\text{F}$

$$Q_1 = U_1 SE_1 (\Delta T)_m = 750(712)59$$

$$Q_1 = 3.15 \times 10^7 \text{ Btu/Hr}$$

$\therefore L_1 \geq 60''$

b. Region-3

To provide for exhaust annulus

$L_3 \geq 30''$

$$\text{Thus, } SE_3 = \frac{N_T \pi D_o L_3}{144} = \frac{1087(\pi).5(30)}{144}$$

$$SE_3 = 356 \text{ Ft}^2$$

Since, $U_3 = 1250 \text{ Btu/Ft}^2 \text{ Hr}^\circ\text{F}$

$(\Delta T)_m = 59\text{F}$

$$Q_3 = U_3 SE_3 (\Delta T)_m = 1250(356)59$$

$$Q_3 = 2.62 \times 10^7 \text{ Btu/Hr}$$

$\therefore L_3 \geq 30''$

5. c. REGION -2

FOR REQUIRED HEAT TRANSFER

$$Q_2 = Q - (Q_1 + Q_3)$$
$$Q_2 = 2.39 \times 10^8 - 5.77 \times 10^7$$
$$Q_2 = 1.813 \times 10^8 \text{ Btu/Hr}$$

Since, $Q_2 = U_2 S_{E_2} (\Delta T)_m$

$$S_{E_2} = \frac{1.813 \times 10^8}{(1285)(59)} = 2400 \text{ Ft}^2$$

$$L_2 = \frac{144 S_{E_2}}{N_T \pi D_o} = \frac{144(2400)}{1087 \pi (.50)}$$

$$\therefore L_2 \cong 202''$$

6. PRELIMINARY DESIGN RATING

a. CALCULATED OVERALL RATE (AVERAGE)

$L_1 = 60''$	$L_2 = 225''$	$L_3 = 31''$	$L = 316''$
$S_{E_1} = 712$	$S_{E_2} = 2670$	$S_{E_3} = 368$	$S_E = 3750$
$U_1 = 750$	$U_2 = 1285$	$U_3 = 1250$	

$$\Sigma S_E U = (533 + 3430 + 460) \times 10^3$$
$$S_E = 3750 \text{ Ft}^2$$

$$U_{avg.} = \frac{4423 \times 10^3}{3750} = 1180 \text{ Btu/Ft}^2 \text{ Hr } ^\circ\text{F}$$

b. REQUIRED OVERALL RATE (SERVICE)

$$U_{REQ} = \frac{Q}{S_E (\Delta T)_m} = \frac{2.39 \times 10^8}{3750 (59)} = 1080 \text{ Btu/Ft}^2 \text{ Hr } ^\circ\text{F}$$

FLUID FLOW ANALYSIS

1. TUBE-SIDE PRESSURE DROP, (FOR SUMMARY SEE 1-f)

a. EXPANSION, inlet line to closure

Assume: Sudden Expansion, Ref. "Reactor Handbook" Volume 2

$$k_e = \left(1 - \frac{A_1}{A_2}\right)^2, \text{ Loss coefficient}$$

$A_1 = .785(13.25)^2 = 138\ in^2$, Inlet line Flow Area

$A_2 = .785(44 - .50)^2 = 1485\ in^2$, Closure

$$\therefore \quad k_e = \left(1 - \frac{138}{1485}\right)^2 = .822$$

$$\Delta P_e = k_e \left(\frac{G_1^2}{2 g_c \rho}\right) \frac{1}{144}$$

$$G_1 = \frac{W_S}{25\, A_F} = \frac{1.665 \times 10^6}{25(138)} = 482\ \#/sec\ ft^2$$

$$\Delta P_e = \left[\frac{(482)^2}{2 \times 32.2 \times 515}\right] \frac{1}{144}\ k_e$$

$$\therefore \quad \Delta P_e = .487\ k_e = .487(.822) = .401\ psi$$

1. b. CONTRACTION, closure into tubes

For sudden contraction, Ref. "Reactor Handbook" Volume 2

$A_1 = 1485 in^2$, closure

$A_2 = 1087 (A_1) = 1087(.127) = 138 in^2$, tubes

From FIGURE 1, Section 2-A

$\therefore$ FOR $\frac{A_2}{A_1} = .093$ $\quad K_c = .39$

$$\Delta P_c = K_c \left(\frac{G_T^2}{2 g_c \rho} \right) \frac{1}{144}$$

Since: $G_T = 482 \ lb/sec\ ft^2$

$\Delta P_c = .487 K_c$ From 1a.

$\therefore \Delta P_c = .190 \ psi$

C. AXIAL-FLOW THROUGH TUBES

Friction Factor

$$f_L = \frac{.046}{(Re_T)^{0.2}}$$ Ref. "Heat Transmission" by W.H. McAdams

From: HEAT TRANSFER RATING, TUBE-SIDE RESISTANCE

$Re_T = 98,300 \quad (Re_T)^{0.2} = (98,300)^{0.2} = 9.95$

SO: $f_L = \frac{.046}{9.95} = .00462$

1 c. (CONT.)

$$\Delta P_L = f_L \left(\frac{G_T^2}{2g_c\rho_T}\right) \frac{L_T}{D_L/4} \frac{1}{144}$$

Ref. "Heat Transmission" by W. McAdams

From BEFORE:

$G_T = 482$ Lb/Ft²Sec.

$L_T = 316(EFF) + 8(T.S.) + 6(S.W.)$ { EFF - Effective, T.S. - Tube Sheet, S.W. - Sine Wave

$L_T = 330"$

$D_L = .402"$

$\rho_T = 52.0$ Lb/Ft³

By Substitution:

$$\Delta P_L = .00462 \left[\frac{(482)^2}{2 \times 32.2 \times 52.0}\right]\left(\frac{330}{.402} \times \frac{4}{144}\right)$$

$\therefore \Delta P_L = 7.30$ psi

d. EXPANSION, tubes to closure

Same as 1-a. (inlet line to closure)

$\Delta P_e = .401$ psi

e. CONTRACTION, closure to outlet line

Same as 1-b (closure into tubes)

$\Delta P_c = .190$ psi

1. f. TUBE-SIDE PRESSURE DROP SUMMARY

(1) EXPANSION, inlet line to closure --- .401 psi

(2) CONTRACTION, closure into tubes --- .190 psi

(3) AXIAL-FLOW THROUGH TUBES ----- 7.30 psi

(4) EXPANSION, tubes to closure ------- .401 psi

(5) CONTRACTION, closure to outlet line-- .190 psi

TOTAL $\Delta P_T = 8.5$ psi

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2. Shell-Side Pressure Drop, (For Summary See 2-g)

a. Entrance; includes nozzle expansion, shroud impact and turns.

From previous experience & preliminary analysis assume 1.5 velocity heads @ inlet velocity are loss.

Inlet line = 14" O.D. STD PIPE
Wall Thickness = .375"
Thus I.D. = 13.25"
or $A_i = 138\ in^2$

$G_i = 482\ lb/sec.ft^2$ From - Tube - Side Pressure Drop

For $\rho_s = 51.5\ lb/ft^3$
$V_i = 9.37\ fps.$

$$\Delta P_e = 1.5\left(\frac{G_i^2}{2g_c\rho_s}\right)\frac{1}{144}$$

$$\therefore \Delta P_e = 1.5\left[\frac{(482)^2}{2\times 32.2\times 51.5}\right]\frac{1}{144} = .730\ psi.$$

b. Axial-Flow along length of sine wave bend, includes some cross-flow:

(1) Cross-flow

For Row 13 $L_p = 63.0"$
Shroud to T.S. = 8.75"

$$G_x = \frac{W_s}{25\,A_F} = \frac{1.665\times 10^6}{25(63\times 8.75)} = 121\ lb/sec\,ft^2$$

$$Re_x = \frac{D_o G_x}{12\,\mu_s} = \frac{(.50)\,121}{12(.156\times 10^{-3})} = 32{,}300$$

$$f_x = \left[.023 + \frac{0.11}{(X_T-1)^{1.08}}\right](Re_x)^{-.15} = .0715$$

where: $X_T = 2.0$

2 b. (CONT)

(1) CROSS-FLOW

$$\Delta P_x = 4 f_x N_{Tc} \frac{(G_x)^2}{2 g_c \rho_s} \frac{1}{144}$$

N_{Tc} = No of Tubes crossed

Assume $\frac{1}{2}$ rows @ both ends of Region-1

So $N_{Tc} = 13$

$$\Delta P_x = 4(.0715)\, 13 \left[\frac{(121)^2}{2 \times 32.2 \times 51.5}\right] \frac{1}{144}$$

$$\Delta P_x = .0285 \text{ psi}$$

(2) AXIAL-FLOW

$$f_L = \frac{.046}{(Re_s)^{0.2}}$$

FROM: SHELL-SIDE RESISTANCE — HEAT TRANSFER:

$Re_s = 83{,}000$

OR $(Re_s)^{0.2} = 9.63$

$$f_L = \frac{.046}{9.63} = .00477$$

$$\Delta P_L = f_L \left[\frac{(G_s)^2}{2 g_c \rho_s}\right]\left(\frac{L_1}{D_E/4}\right)\frac{1}{144}$$

FROM BEFORE:

$G_s = 74$ Lb/sec ft²

$L_1 = 60"$

$D_E = 2.05"$

$$\Delta P_L = .00477 \left[\frac{(74)^2}{2 \times 32.2 \times 51.5}\right] \frac{60}{2.05} \times \frac{4}{144}$$

$$\Delta P_L = .00625 \text{ psi}$$

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2 C. CROSS-FLOW BETWEEN BAFFLES

$$A_F = (P_B - .25) L_F = 8.75 L_F$$

$$G_s = \frac{W_s}{.25 A_F} = 6.66 \times 10^4 \frac{1}{A_F}$$

$$Re_x = \frac{D_o G_s}{12 \mu_s} = \frac{.50}{12(.156 \times 10^{-3})} = 267.1\, G_s$$

$$f_x = \frac{0.34}{(Re_x)^{.15}}$$

$$\Delta P_x = 4 f_x N_{TC} \frac{G_s^2}{2 g_c \rho_s} \times \frac{1}{144} = \frac{4}{64.4(51.5)} \frac{1}{144} f_x G_s^2$$

$$\Delta P_x = \frac{f_x G_s^2}{119{,}500}$$

Row	L_F	A_F	G_s	Re_x	$Re_x^{.15}$	f_x	ΔP_x
1	23.5	206	323	86,500	5.50	.0618	.0543
2	26.8	234	284	76,100	5.40	.0630	.0511
3	30.1	263	253	67,600	5.30	.0642	.0412
4	33.3	291	229	60,900	5.22	.0651	.0339
5	36.7	321	207	55,300	5.15	.0660	.0283
6	39.9	349	191	51,000	5.08	.0670	.0244
7	43.2	378	176	47,000	5.02	.0677	.0210
8	46.5	407	164	43,800	4.96	.0685	.0184
9	49.8	436	153	40,900	4.92	.0691	.0162
10	53.0	463	144	38,500	4.88	.0697	.0145
11	56.3	492	135	36,100	4.82	.0705	.0129
12	59.8	523	127	33,900	4.78	.0711	.0115
13	63.0	551	121	32,300	4.75	.0716	.0105

TOTAL $\Delta P_x = 0.338$ psi per crossing

007 120

2 d. BAFFLE WINDOW – OUTSIDE DISC, includes turning

From Previous Experience & Preliminary analysis assume 1.5 velocity heads loss @ window velocity.

configuration: $I.D._{Shell\ Shield} = 48.5''$
$O.D._{Disc} = 40''$
6-Tie Rods @ $1\ in^2 = 6\ in^2$

$$A_F = .785\left[(48.5)^2 - (40)^2\right] - 6 = 584\ in^2$$

$$V = \frac{1.665 \times 10^6}{25(51.5)584} = 2.21\ fps.$$

$$\Delta P_{wo} = 1.5\left(\frac{V^2\rho}{2g_c}\right)\frac{1}{144} = 1.5\left[\frac{(2.21)^2}{2\times 32.2}\right]\frac{51.5}{144}$$

$$\therefore \Delta P_{wo} = .0407\ psi\ per\ disc$$

FOR LOWER WINDOW (Region-3)

$$A_F = .785\left[(40)^2 - (32.125)^2 - 444(.5)^2\right]$$
$$A_F = 360\ in^2$$

$$V = \frac{1.665 \times 10^6}{25(51.5)360} = 3.60\ fps$$

$$\therefore \Delta P_{wo} = 1.5\left[\frac{(3.60)^2}{64.4}\right]\frac{51.5}{144} = .108\ psi$$

C. BAFFLE WINDOW - INSIDE DO-NUT, includes turning

As above assume 1.5 velocity heads @ window velocity

$$A_F = .785\left[(20.125)^2 - 162(.5)^2\right] = 286\ in^2$$

$$V = \frac{1.665 \times 10^6}{25(51.5)286} = 4.52\ fps.$$

$$\therefore \Delta P_{wi} = 1.5\left[\frac{(4.52)^2}{64.4}\right]\frac{51.5}{144} = .170\ psi\ per\ do\text{-}nut$$

2 f. EXHAUST, includes contraction and expansion

From previous experience & preliminary analysis assume 1.5 velocity heads @ outlet velocity. are loss

AS FOR ENTRANCE $\Delta P_o = .730$ psi

g. SHELL-SIDE PRESSURE DROP SUMMARY

REGION-1 :	ENTRANCE --	0.730 psi
	CROSS-FLOW --	0.029 psi
	AXIAL-FLOW --	0.006 psi
		0.77 psi

REGION-2 :	25 - CROSSINGS @ 0.338 =	8.45 psi
	13 - OUTER WIND. @ 0.0407 =	0.53 psi
	13 - INNER WIND @ 0.170 =	2.21 psi
		11.19 psi

REGION-3 :	3 - CROSSINGS @ 0.338 =	1.01 psi
	1 - OUTER WINDOW ---	.11 psi
	1 - INNER WINDOW ---	.17 psi
	EXHAUST ----	.73 psi
		2.02 psi

∴

REGION-1	0.77 psi
REGION-2	11.19 psi
REGION-3	2.02 psi
ΔP_s =	13.98 psi

OCT 122

PART LOAD THERMAL PERFORMANCE

REF. BOILER THERMAL CALCULATIONS, VOL. II

1. HEAT TRANSFER RATES

A_E = 3100 Ft^2 EQUIVALENT SURFACE

% LOAD	r_s	r_m	r_T	U_E
100	.000250	.000378	.000149	1285
80	286	378	163	1210
60	340	378	183	1110
40	434	378	215	973
20	659	378	284	735

2. PART LOAD TEMPERATURES

$$\epsilon = \frac{NTU}{1+NTU} = \frac{T_1 - T_2}{T_1 - T_3} = \frac{T_4 - T_3}{T_1 - T_3}$$

$$NTU = \frac{A_E U_E}{W C_p}$$

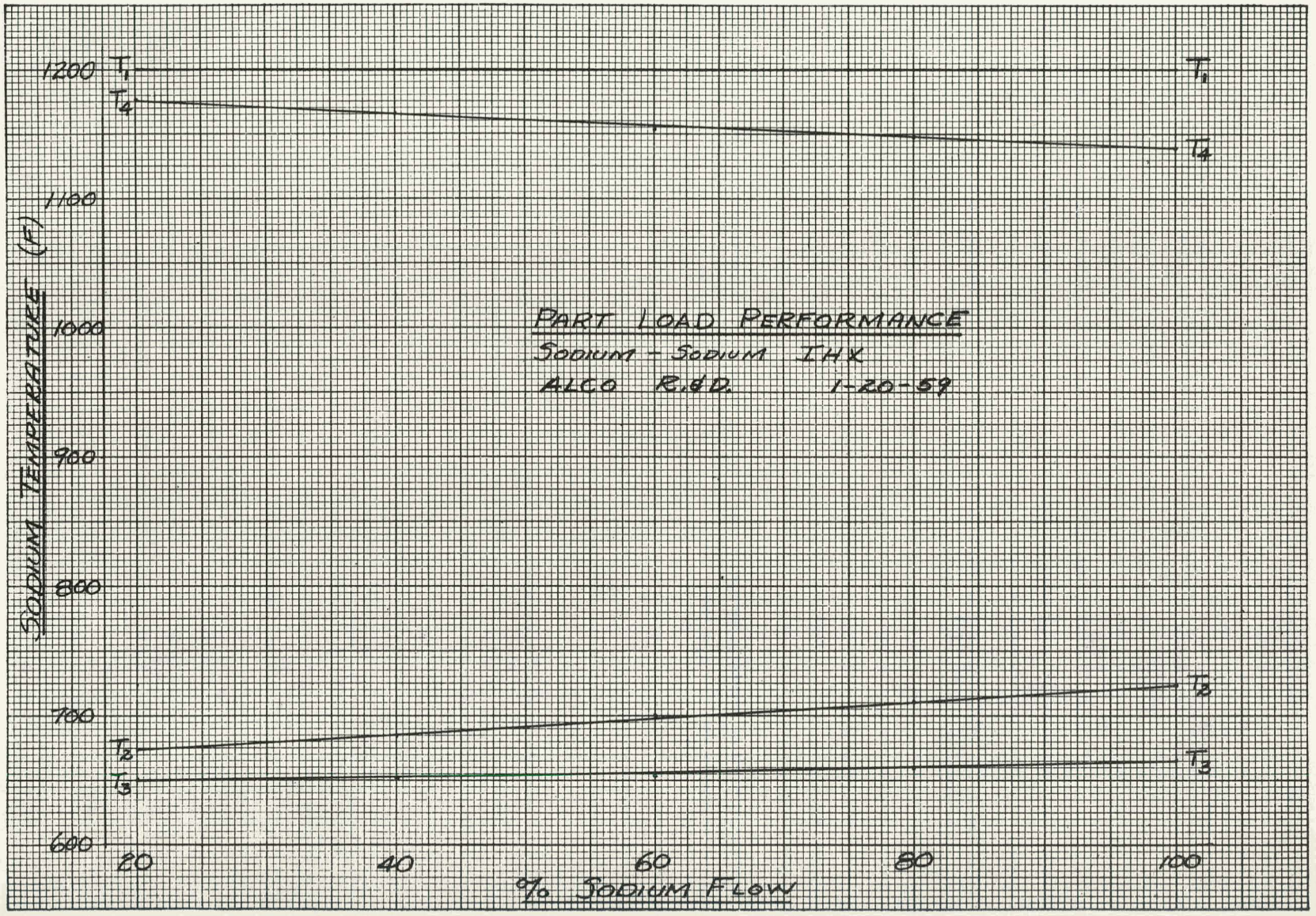
% LOAD	NTU	ϵ	T_1 *	T_2	T_3	T_4
100	7.85	.886	1200	724	666	1140
80	9.29	.902	1200	711	659	1148
60	11.33	.919	1200	700	654.6	1154.6
40	14.91	.936	1200	685	651.4	1166.4
20	22.50	.957	1200	674	650.2	1176.2

* T_1 TO BE HELD CONSTANT

OCT 123

907 124

7-D34


PART LOAD PERFORMANCE
SODIUM - SODIUM IHX
ALCO R.&D. 1-20-59
SODIUM TEMPERATURE (°F)
1200
1100
1000
900
800
700
600
20
40
60
80
100
% SODIUM FLOW
T1
T4
T2
T3

E - MECHANICAL DESIGN CALCULATIONS

49" I.D. SHELL — $P \leq 0.385SE = 0.385(6800)(.80) = 2095$ PSI

PL MAT'L.: S/S SA 240 GR. M (TP.-316)

DESIGN PRESS. 100 P.S.I. - DES. TEMP. 1200°F
ALLOW. STRESS = 6800 P.S.I. EFF. 80%

$$t = \frac{P \times D}{2SE - 1.2P} = \frac{100 \times 49}{\underset{10,880}{2 \times 6800 \times .80} - \underset{-120}{1.2 \times 100}} = \frac{4900}{10,760}$$

$= .4555$ SAY $\frac{1}{2}$"

44" I.D. CHANNEL (CYLINDRICAL SECTION) $.385SE = 2357$ PSI

PL MAT'L.: S/S SA 240 GR.M TP.-316

DESIGN PRESS. 150 P.S.I. - DES. TEMP. 1200°F
ALLOW. STRESS = 6800 P.S.I. EFF. = 90% X-RAY

$$t = \frac{P \times D}{2SE - 1.2P} = \frac{150 \times 44}{2 \times 6800 \times .90 - 1.2 \times 150} = \frac{6600}{\underset{12,060}{12,240} - 180} = .548$$

SAY $\frac{9}{16}$"

$\frac{1}{2}$" O.D. TUBES

MAT'L.: S/S SA 213 GR. (TP. 316)

INT. DESIGN PRESS. 150 P.S.I. - DES. TEMP. 1200°F
EXT. DESIGN PRESS. 100 P.S.I.
ALLOW. STRESS = 6800 P.S.I. EFF. = 100%

$$t = \frac{P \times D}{2SE - 1.2P}$$

$$t = \frac{150 \times .402}{\underset{13,600}{2 \times 6800 \times 1.0} - \underset{-180}{1.2 \times 150}}$$

$13,420$

$$= \frac{60.3}{13,420} = .0045$$

.049	.500
.049	.098
.098	.402 I.D.

USE 18 GAGE WITH $.049 = t$

USE FIG. UG-31 TO CHECK EXTERNAL PRESS.

$\frac{t}{D_o} = \frac{.049}{.5} = .098$ ∴ EXT. PRESS. = 675 P.S.I. CAN BE USED

GCP 125

TUBESHEET BENDING FORMULA — USING STANDARDS OF FEEDWATER HEATER MANUFACTURERS ASSOCIATION, INC. – 2ND. ED. 1958

ACTUAL CHANN. $\frac{t}{D_i} = \frac{1.375}{44} = .03123$ ACTUAL SHELL I.D. $t_o = \frac{1.375}{49} = .0281$

SH. PRES. = 100 PSI
CH. PRES. = 150 PSI
TEMP. = 1200°F

F = 1.151 (CHAN.) F = 1.175 (SHELL) S = 6800 PSI

FG'D. MAT'L: S/S SA-182 GR. F-316 (TP. 316)

$$\text{CHAN. } T = \frac{FG}{2}\sqrt{\frac{P}{S}} = \frac{1.151 \times 44}{2}\sqrt{\frac{150}{6800}} = \frac{1.151 \times 44}{2}\sqrt{.1485} = 3.76 \text{ SAY } 3\tfrac{3}{4}\text{" MIN.}$$

$$\text{SHELL } T = \frac{1.175 \times 49}{2}\sqrt{\frac{100}{6800}} = 3.49 \text{ SAY } 3\tfrac{1}{2}\text{"}$$

SHEAR FORMULA – TEMA PAR. R-7.122 .02205

$$T = \frac{0.31 D_L}{\left(1 - \frac{d}{p}\right)}\left(\frac{P}{S}\right) = \frac{0.31(39.125)}{\left(1 - \frac{.5}{1}\right)}\left(\frac{150}{6800}\right)$$

$$= .536 \text{ SAY } \tfrac{5}{8}\text{"}$$

TUBE SHEET SKIRT (TUBE SIDE)

$$t = \frac{44 \times 150}{2 \times 6800 \times 1.0 - 1.2 \times 150} = .492 \text{ SAY } \tfrac{1}{2}\text{"}$$

13,600 − 180 = 13,420

(SHELL SIDE)

$$t = \frac{49 \times 100}{2 \times 6800 \times 1.0 - 1.2 \times 100} = .364 \text{ SAY } \tfrac{3}{8}\text{"}$$

13,480

(TUBE SIDE) INTERNAL TEST PRESSURE CALC.

$$\frac{18,750}{6800} \times 150 \times 1.5 = 620 \text{ P.S.I.}$$

(SHELL SIDE) INTERNAL TEST PRESS. CALC.

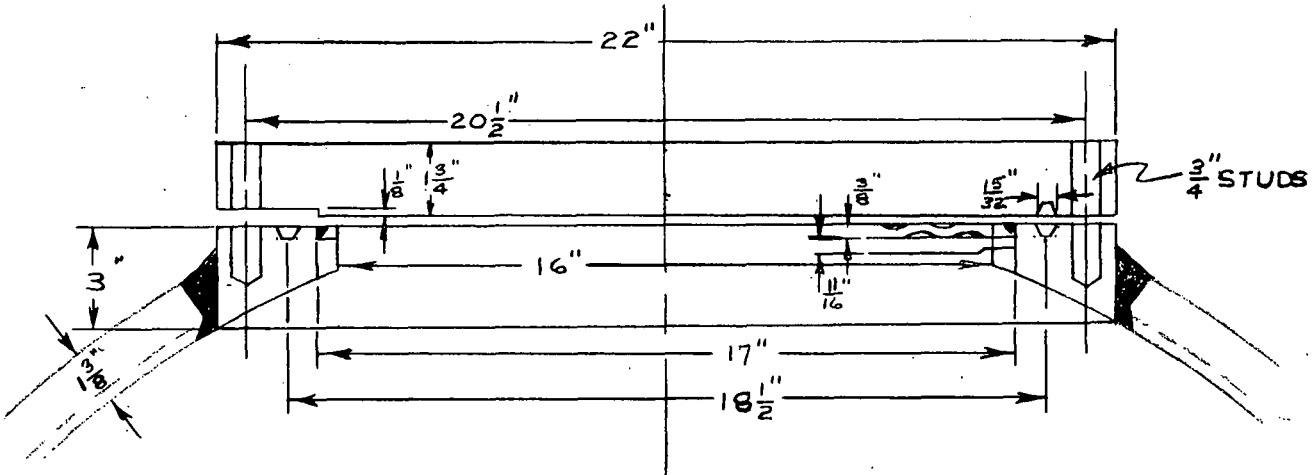
$$\frac{18,750}{6800} \times 100 \times 1.5 = 413 \text{ PSI.}$$

CHANN. HEMISPHERICAL HEAD

$$t = \frac{PD}{4SE - 0.4P} = \frac{150 \times 44}{4(6800)(.90) - .4(150)} = \frac{6600}{24,480 - 60} = \frac{6600}{24,420} = .271 \text{ SAY } \tfrac{5}{16}\text{"}$$

7-E2

MANWAY COVER & GASKET



$H_{GY} = b\pi G Y = .0547 \times 3.14 \times 18.5 \times 26{,}000$

$= \underline{\underline{82{,}750}}$

$H_{GP} = 2b\pi G P m = .1094 \times 3.14 \times 18.5 \times 150 \times 6.5$

$= \underline{\underline{6{,}190}}$

$H = \frac{\pi}{4} G^2 P = \frac{3.14}{4}(18.5)^2 \times 150 = \underline{\underline{40{,}300}}$

$W_{OP} = H + H_{GP} = \underline{\underline{46{,}490}}$

$A_{B\,MIN} = \frac{82{,}750}{15{,}000} = 5.52$

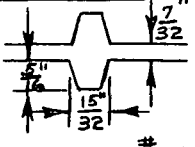
$= \frac{46{,}490}{5{,}000} = 9{,}298$

$\frac{9.298}{.302} = 30.60$ SAY 32 STUDS

$32 \times .302 = \underline{\underline{9.664}} = A_{ACT.}$

$W = \frac{9.664 + 9.298}{2} \times 5{,}000 = 47{,}405$

GASKET GROOVE:



300# CLASS A - RATING

RTJ - #65 EXCEPT - 18 1/2" PITCH ϕ

$b_o = \frac{.4375}{8} = .0547 = b$

$Y = 26{,}000$

$m = 6.5$

STUDS - S/S - 194 GR. B8C TP. 347

$S_{OP} = 5000$ PSI

$S_{ATM} = 15000$ PSI

SPACING REQ'D.: 1.75" MIN.
3.0" PREFF.

$\frac{20.5 \times 3.14}{32} = 2.01$ STUD SPACING

COVER

$c = .3 + \frac{1.4 \times 47{,}405 \times 1}{40{,}300 \times 18.5} = .389$

$t = 18.5 \sqrt{\frac{.389 \times 150}{6800}}$ $\quad .00859$

$= 18.5 \times .0925 = 1.71$ SAY $1\frac{3}{4}$"

7-E3

REINFORCEMENT FOR MANWAY

THICKNESS REQ'D:

$$t = \frac{150 \times 16}{2 \times 6800 \times 1.0 - 1.2 \times 150} = \frac{2400}{13{,}600 - 180} = \frac{2400}{13{,}420} = .1788 \text{ SAY } \frac{1}{4}"$$

$\frac{5}{16}$" THICK SPHERICAL REQ'D REINFORCEMENT: $16 \times \frac{5}{16} = 5.0"$

AVAIL: $22 - 16.5 = 5.5 \times (2.5 - 1.00) = 8.525$

LESS: $2 \times .75 \times 2 = 3.000$

5.525 MIN. AVAIL.

14" O.D. NOZZLE - SHELL & CHANNEL INLET AND OUTLET

ASSUME CUSTOMERS PIPING EQUAL TO SCH. 10 = .25" THK:

$.25 \times .875 = .21875$

DESIGN 150 P.S.I. AT 1200°F
CUSTOMER MAT'L. SA - 213 GR. TP316 PIPE
NOZZLE MAT'L. SA - 182 GR. F316 FORGING

.21875	14.0000
.21875	.4375
.43750	13.5625

$$t_N = \frac{13.5625 \times 150}{2 \times 6800 \times 1.0 - 1.2 \times 150} = \frac{13.5625 \times 150}{13{,}420} = .1515$$ USE WT. THICK $\frac{3}{8}$" AND TAPER IF NECESSARY TO SUIT CUSTOMERS PIPING.

SEAL & SEAL PL RETAINER

EXTERNAL PRESS. PLUS VACUUM ON SEAL RETAINER = 30 P.S.I.

FLAT PLATE FORMULAS:

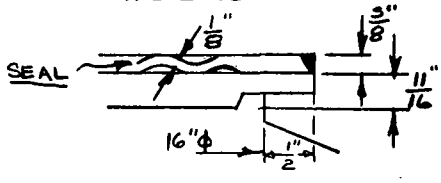
TEMP. = 1200°F S/S TP 316
ALLOWABLE STRESS = 6800 PSI

EDGE SUPPORTED

$$1.24 \times 30 \times \frac{8.5^2}{.6875^2} = 5{,}666 \text{ P.S.I. (STRESS)}$$

DEFLECTION - SEAL PLATE

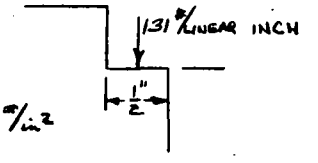
$$.17 \times \frac{150}{22.4 \times 10^6} \times \frac{8.5^4}{.375^3} \quad (5{,}210;\ .0528) = .17 \times \frac{781{,}000}{118.2 \times 10^4} = .1124"$$



SHEAR STRESS CHECK:

$W = 17^2 \times .7854 \times 30 = 6800\#$

$$\frac{6800}{16.5 \times 3.14} = 131 \#/\text{LINEAR INCH}$$



$$\frac{bd^2}{6} = \frac{.5 \times .625^2}{6} = .0325$$

ALL. SHEAR STRESS = $.8 \times 6800 = 5{,}440 \#/in^2$

$$\frac{131}{.0325} = 4{,}025 \#/in^2$$

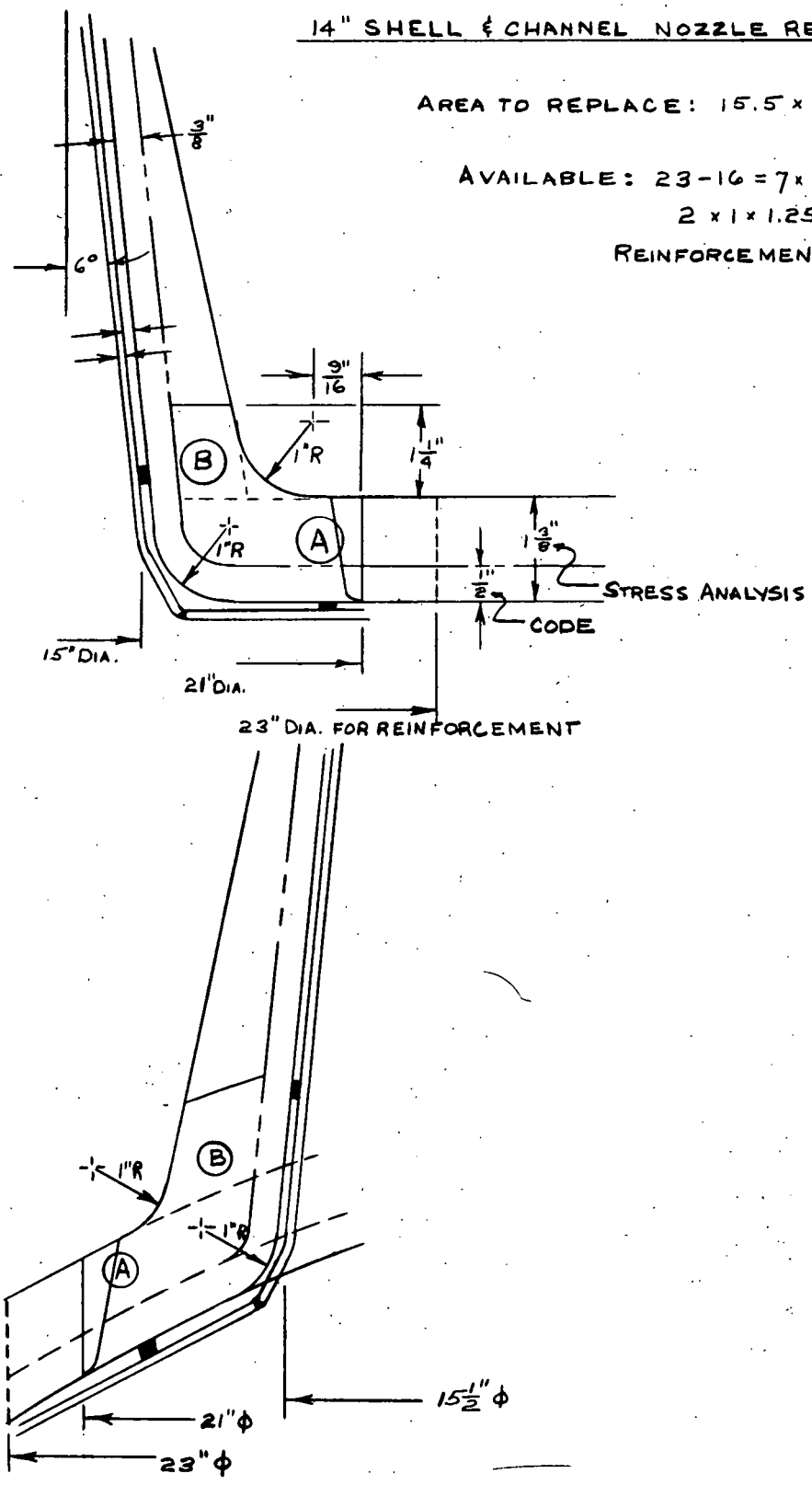
14" SHELL & CHANNEL NOZZLE REINFORCEMENT

AREA TO REPLACE: 15.5 x .5 = 7.75 D"

AVAILABLE: 23 - 16 = 7 x .875 = 6.125 ← (A)

2 x 1 x 1.25 = 2.5 ← (B)

REINFORCEMENT = 8.625 D"



7-E5

WEIGHT CALCULATIONS - DRY

$S/S = .283\ ^{\#}/cu.in.$

MANWAY COVERS

$22"\phi = 107.69 \times 1.75 = 188.5$ SAY $180^{\#}/COVER \times 2 = \underline{360^{\#}}$

MANWAY FLANGE

$22"\phi = 107.69$

$16"\phi = \underline{56.96}$

$50.73 \times 2.5 = 125^{\#}$ EACH $\times 2 = \underline{250^{\#}}$

CHANNEL

$$\frac{\left[\frac{4}{3}\pi\left((23.375)^3 - (22)^3\right)\right]}{2} \times .283 = 1250^{\#}\ EACH$$

(12,750 and 10,648)

HOLE FOR NOZZLE

$15"\phi = 50.06 \times 1.375 = 68.9$

HOLE FOR MANWAY

$22"\phi = 107.69 \times 1.375 = 147.69$

216.59 LESS 216

$1035^{\#}$ EACH $\times 2 = \underline{2070^{\#}}$

NOZZLES

APPROX:

$14\frac{7}{8}"\phi = 49.23$

$13\frac{1}{4}"\phi = \underline{39.06}$

$10.17 \times 13 = 132.5^{\#}$ EACH $\times 2 = \underline{265^{\#}}$

TUBESHEET

CHAN. SKIRT

$46\frac{3}{4}"\phi = 486.3$

$44"\phi = \underline{430.8}$

$55.5 \times 2 = 111^{\#} \times 2 = 222^{\#}$

SHELL SKIRT

$51\frac{3}{4}"\phi = 595.9$

$49"\phi = \underline{534.2}$

$61.7 \times 1.75 = 108.0^{\#}$ EACH $\times 2 = 216^{\#}$

TUBESHEET

$51\frac{3}{4}"\phi = 595.9 \times 3\frac{3}{4} = 2235^{\#}$ EACH $\times 2 = 4470^{\#}$

222

216

4908

TUBE HOLES

$(.402)^2 \times \frac{3.14}{4} \times 3.75 \times 1088 = 518\ in^3$

$518 \times .283 = 145^{\#}/_{TUBESHEET} \times 2 =$ 290

4618

213

NIPPLES

$595.9 \times .5 \times .75 = 213.2$

TOTAL WEIGHT: 4831 #

SHELL

$1\frac{3}{8}"$ SECTIONS

$61.7 \times 54" = 3330^{\#}$ EACH $\times 2 = 6660^{\#}$

HOLES FOR NOZZLE OPENINGS = $68.9 \times 2 =$ -130

6530 #

NOZZLES

APPROX. = 265 #

$\frac{1}{2}"$ SECTION

$50"\phi = 556.3$

$49"\phi = 534.2$

$22.1 \times 204 = 4,510$

SHELL & NOZZLE LINERS

$48\frac{3}{4}"\phi = 528.8$

$48\frac{1}{2}"\phi = 523.4$

$5.4 \times 25 = 135^{\#} \times 2 = 270^{\#}$

SHIELDS

$48\frac{3}{4} = 528.78$

$48\frac{1}{2} = 523.4$

$5.38 \times 304 = 1635^{\#}$

$43\frac{3}{4} = 43,612$

$43\frac{1}{2} = 43,099$

513 CU.IN. $\times .283 =$ 145 #

SHROUD

$40\frac{1}{2}"\phi = 364.95$

$40"\phi = 355.95$

$9.0 \times 63 = 570^{\#}$

DAVITS & LUGS

$3.6 \times 2 = 7.2$

10.0

2.8

$20.0 \times 2 = 40$

60

100 #

LIFTING LUGS = $30 \times 2 = 60$

7-E7

DISC BAFFLES

$32\frac{1}{8}"\phi = 229.63 \times \frac{1}{4} \times 1 = 57^{\#}$

$40"\phi = 356 \times 12 \times \frac{1}{4} = 1100$

DO-NUT BAFFLES

$48\frac{3}{4}"\phi = 528.78$

$20\frac{1}{8}"\phi = 90.12$

$438.66 \times \frac{1}{4} \times 13 = 1425$

$39\frac{7}{8}"\phi = 353.79$

$20\frac{1}{8}"\phi = 90.12$

$263.67 \times \frac{1}{4} \times 1 = 66$

$1491^{\#}$

CENTER PIPE

$14"\phi = 43.62$

$13\frac{1}{4}"\phi = 39.06$

$5.56 \times 65 = 300^{\#}$

TIE RODS

$\frac{1}{2}"\phi = 10 \times .0555 \times 272 = 152$

$\frac{3}{8}"\phi = 6 \times .032 \times 14 = 3$

$155^{\#}$

TIE ROD SPACERS

$10 \times .367 \times 22.0 = 81$

$6 \times .236 \times 1 = 3$

$84^{\#}$ SAY $85^{\#}$

TUBES

$1088 - \frac{1}{2}"$ O.D. $\times$ 18 GA. $= .236^{\#}/FT$

$27.5 \times 1088 \times .236 = 7050^{\#}$

STUDS, SEAL & NUTS.

$\frac{28 \times 4.5}{12} \times 1.5 = 14 \times 2 = 28^{\#}$

22

$50^{\#}$

SHROUD RINGS

$48\frac{7}{8}"\phi = 531.5$

$40\frac{5}{8}"\phi = 367.2$

$164.3 \times .75 = 124$

$49"\phi = 534.22$

$42\frac{1}{2}"\phi = 401.89$

$132.33 \times .75 = 100$

$49"\phi = 599.22$

$46"\phi = 470.82$

$63.40 \times .75 = 50$

$45\frac{3}{4}"\phi = 465.72$

$40\frac{1}{2}"\phi = 364.95$

$100.77 \times .50 = 51$

124
100
50
51
$325^{\#}$

$\frac{1}{2}"$ BAFFLE

$40"\phi = 356 \times .5 \times 1 = 178^{\#}$

1100
57
$1335^{\#}$

360
250
2070
265
4831
6530
265
4510
270
1635
145
570
100
50
325
1335
1490
300
155
85
7050
425

33,016

MISC. 1,984

TOTAL WEIGHT (DRY) 35,000

7-E8

SUPPORT

$2 \times 9 \times 12 = 216$

$1.5 \times 8 \times 13 = 156$

$372 \times .283 = 105.2^{\#} \times 4 = \underline{425^{\#}}$

LIQUID VOLUME - SODIUM & WATER WEIGHTS

VOLUME OF CHANNEL HEMISPHERES

$\frac{4}{3} \frac{\pi (22)^3}{2} = 22{,}300 \text{ IN.}^3$ (10,648)

$\frac{\pi}{4} (44)^2 \times 5 = 7{,}600 \text{ IN.}^3$ (1936)

$29{,}900 \text{ IN.}^3$ EACH $\times 2 = \underline{59{,}800 \text{ IN}^3}$

VOLUME OF NOZZLES

APPROX. AVE. I.D. = 14"

$(14)^2 \times \frac{3.14}{4} \times 13 = 2000 \times 4 = \underline{8000 \text{ IN}^3}$ FOR FOUR NOZZLES (196)

VOLUME OF SHELL

$(49)^2 \times \frac{3.14}{4} \times 315.5 = \underline{594{,}000 \text{ IN}^3}$ (2409)

VOLUME OF SHROUD SUPPORTS:

$\left[(40\tfrac{7}{8})^2 - (40\tfrac{5}{8})^2\right] \times \frac{3.14}{4} \times 2.5 = 1450 \text{ IN.}^3$ (2390 − 1655 = 735)

VOLUME OF SHROUD:

$40.25 \times 3.14 \times \frac{1}{4} \times 63 = 1990 \text{ IN}^3$

VOLUME OF LINERS:

$48.5 \times 3.14 \times .125 \times 304 = 5800 \text{ IN.}^3$

VOLUME OF DISC. BAFFLES:

$40^2 \times .785 \times .25 \times 14 = 4400 \text{ IN.}^3$

VOLUME OF DO-NUT BAFFLES:

$\left[(48\tfrac{3}{4})^2 - (20\tfrac{1}{8})^2\right] \times \frac{3.14}{4} \times \frac{1}{4} \times 14 = 5050 \text{ IN.}^3$ (2380 − 405 = 1975)

VOLUME OF CENTER PIPE:

$$\left[(14)^2 - (13.25)^2\right] \times \frac{3.14}{4} \times 65 = 1020 \text{ IN.}^3$$

196 − 176
20

METAL VOLUME OF TUBES:

$$\left[(.5)^2 - (.402)^2\right] \times \frac{3.14}{4} \times 330 \times 1088 = 24{,}900 \text{ IN.}^3$$

.25 − .162
.088

VOLUME OF TIE RODS:

$$\left[(.75)^2\right] \times \frac{3.14}{4} \times 264 \times 10 = 1{,}165$$

.5625

45,780
SAY 45,800

TOTAL LIQUID VOLUME OF SHELL:

594,000
45,800
548,200

TOTAL LIQUID VOLUME

548,200
59,800
8,000
616,000 in.^{3}

WEIGHT OF SODIUM

$$\frac{616{,}000}{1728} \times 51.8 = 18{,}500^{\#}$$

AVE. SODIUM = 51.8 #/ft.^{3}

WEIGHT OF WATER

$$\frac{617{,}000}{1728} \times 62.35 = 22{,}200^{\#}$$

35,000
18,500
53,500 # SODIUM-OPERATING

35,000
22,200
57,200 FULL OF WATER

---

007 134

F - STRESS CALCULATIONS

Physical Properties of SS316, Croloy (2¼Cr. 1Mo.) & Sodium

The following tabulation of data were extracted from various manufacturers literature and technical publications. Values may be the results of one test or the average of several tests. Manufacturing variables, chemistry, grain size, heat treatment and other conditions that would influence the test results are not known and these data should, therefore, be used with caution.

CROLOY (2¼ Cr. 1 Mo) Density Approx. 0.283 #/in³ At RT

Temperature °F	70	600	800	900	1000	1100	1200	1300	Ref.
High Temp. Tensile Strength 1,000 PSI	71.5	-	66.3	63.5	54.2	45.0	33.5	-	1
	69.9	-	66.8	63.8	49.8	39.1	31.3	-	2
	72.9	-	—	-	55.5	45.3	32.9	-	3
	70	-	67	62.5	53.5	43	32.5	-	9
High Temp. Yield Strength 0.2 %, 1,000 PSI	43.0	-	31.0	31.0	28.2	25.5	20.0	-	1
	45.0	-	-	-	-	-	-	-	2
	41.5	-	-	-	28.3	26.6	18.3	-	3
	42.5	-	33.0	31.5	24.0	25.0	22.0	-	9
Thermal Expansion (Mean) in/in/°F x 10^{-6} Room Temp. to Indicated	-	7.21	7.54	-	7.86	-	8.09	-	2
	-	-	-	-	7.72	-	7.84	-	3
* B&W Data	-	7.21	7.54	-	7.86	-	8.09	-	*
Thermal Expansion (Instantaneous) in/in/°F x 10^{-6}	-	-	-	-	9.38	9.72	10.07	10.6	2
Poisson's Ratio	.288	-	.298	.301	.304	.308	.315	-	3
Modulus of Elasticity 1x10^6 PSI	31	-	26.3	25.6	24.9	24.1	23.2	-	3
Thermal Conductivity BTU/Sq. Ft./in/Hr./°F	-	198.6	199.8	-	199.2	-	199.5	- interpolated	6

Hours					100	1,000	10,000	100,000	Ref.
Stress Rupture - 1200°F - 1,000PSI					14.0	8.5	6.2	2.5	1
					12.0	7.7	4.5	4.5	2
					-	8.5	6.0	-	3
					-	8.3	6.0	-	9
Stress for 1% Creep 1200°F - 1,000 PSI					-	-	4.3	2.3	1
					-	-	4.18	2.5	2
					-	-	4.2	2.5	6
					-	-	4.2	2.3	9

TYPE 316 SS - DENSITY APPROX. 0.29 #/in³ AT RT

Temperature °F	70	600	800	900	1000	1100	1200		Ref.
High Temp. Tensile Strength	85.5	–	71.7	71	68.7	66.5 *	56.0	* 1110°F	1
1,000 PSI	103.9	–	–	–	73.5	69.1 *	60.5		2
	75 min.	–	74.3	72.1	70.8	63.8 *	54.4		3
	82.5	–	71.5	70.0	67.5	63.0 *	56.5		5
	76	–	71.7	71	69.2	66.5 *	57.7		7
High Temp. Yield Strength	38.5	–	23.0	22.0	21.2	20.5	20.0		1
0.2% - 1,000PSI	36.8	–	–	–	20.3	19.6	20.5		2
	31	–	23	22	21	20.5	–		7
Thermal Expansion (Mean)	–	9.75	10.6	–	10.3	–	10.5		1
in/in/°F x 10^{-6} Room Temp. to Indicated	–	–	9.86	–	10.7	–	10.29		2
	–	–	–	–	–	–	10.80		3
	–	9.0	–	–	9.7	–	10.3		5
	–	–	10.2	–	10.5	–	10.8	* B&W	6
	–	9.37	9.81	–	–	–	10.28		8
Poisson's Ratio	.284	–	.312	.316	.320	.324	.328	*347 SS	3*
Modulus of Elasticity 1x10^6 PSI	28.9	–	24.1	23.4	22.8	22	21.4	*347 SS	2*
	28	–	24	–	22	–	21		6
	–	–	–	–	23.4	22.8	22.4		7
Thermal Conductivity	–	–	–	149 *	–	–	–	* 932°F	4
BTU/Sq.Ft./in/Hr./°F	–	–	–	145*	147	–	–		5
	–	–	–	–	164	–	175	(18-8) ←	6
	–	–	–	–	141.6	–	151.2		8

TYPE 316 SS (Contd.)

Hours	100	1,000	10,000	100,000	Ref.
Stress Rupture - 1200°F - 1000 PSI	-	24.7	18.2	-	1
	31	25.0	18.0	12.5	2
	-	25.0	-	11.0	5
	-	25	18	16	6
	-	24	20	-	7
Stress For 1% Creep - 1200°F - 1000 PSI	-	-	12.7	6.6	1
	-	-	8.2	5.0	3
	-	-	11.0	6.0	5
	-	-	14.0	6.0	6
	-	-	12	-	7

List of References

1. "Steels for elevated Temperature Service", U.S. Steel Corp., 1952.
2. "Properties of carbon and Alloy Seamless Steel Tubing for High Temperature and Pressure Service", The B & W Tube Co., Technical Bulletin No. 6-E, 1948.
3. "Digest of Steels for High Temperature Service", 6th Ed., 1957, The Timken Roller Bearing Co.
4. "Stainless Steels", Allegheny-Ludlum Steel Corp.
5. "Metals Handbook", 1948 and 1954 supplement.
6. "Metal Data", S.L. Hoyt, 1952.
7. "The Reactor Handbook", Vol. 3, Sect. I, General Properties of Materials, AEC, 1955.
8. ASTM Special Technical Publication No. 227.
9. ASTM Special Technical Publication No. 100.

Material data at elevated temperatures for liquid sodium, 316 stainless steel alloy, and Croloy - $2\frac{1}{4}$% Cr 1% Mo steel alloy are given on the accompanying three charts.

Thermal diffusivity $\alpha = k/\omega C_p$

316 Stainless steel

If $k = 12.1$ Btu/hr.ft.°F

$\omega = .29$ lb/in^3

$C_p = .16$ Btu/lb °F

$$\alpha = \frac{12.1}{.29 \times 1728 \times .16} = .1509 \frac{ft^2}{hr}$$

$$= .1509 \times \frac{144}{3600} = .006,036 \frac{in^2}{sec}$$

This is approximately the value at 890 °F

Croloy - $2\frac{1}{4}$% Cr 1% Mo steel alloy

If $k = 16.5$ Btu/hr ft °F

$\omega = .283$ lb/in^3

$C_p = .16$ Btu/lb °F

$$\alpha = \frac{16.5}{.283 \times 1728 \times .16} = .2109 \frac{ft^2}{hr}$$

$$= .2109 \times \frac{144}{3600} = .008436 \frac{in^2}{sec}$$

This is approximately the value at 890°F

Typical values of film coefficient h for heat flow between liquid sodium and 316 stainless steel

For temperature change from 1140°F to 650°F in 10" d. nozzle with mean velocity of sodium = $18.8 \frac{ft}{sec} = 67,680 \frac{ft}{hr}$.

Compute the Péclot Number $N_{pe} = \frac{d u w C_p}{k}$

Calculate values of the Nusselt Number from

$N_{Nu} = 7 + .025 N_{pe}^{.8}$ (theoritical value within 10%)

or $N_{Nu} = .625 N_{pe}^{.4}$ (average of test values)

Then from the definition of the Nusselt Number

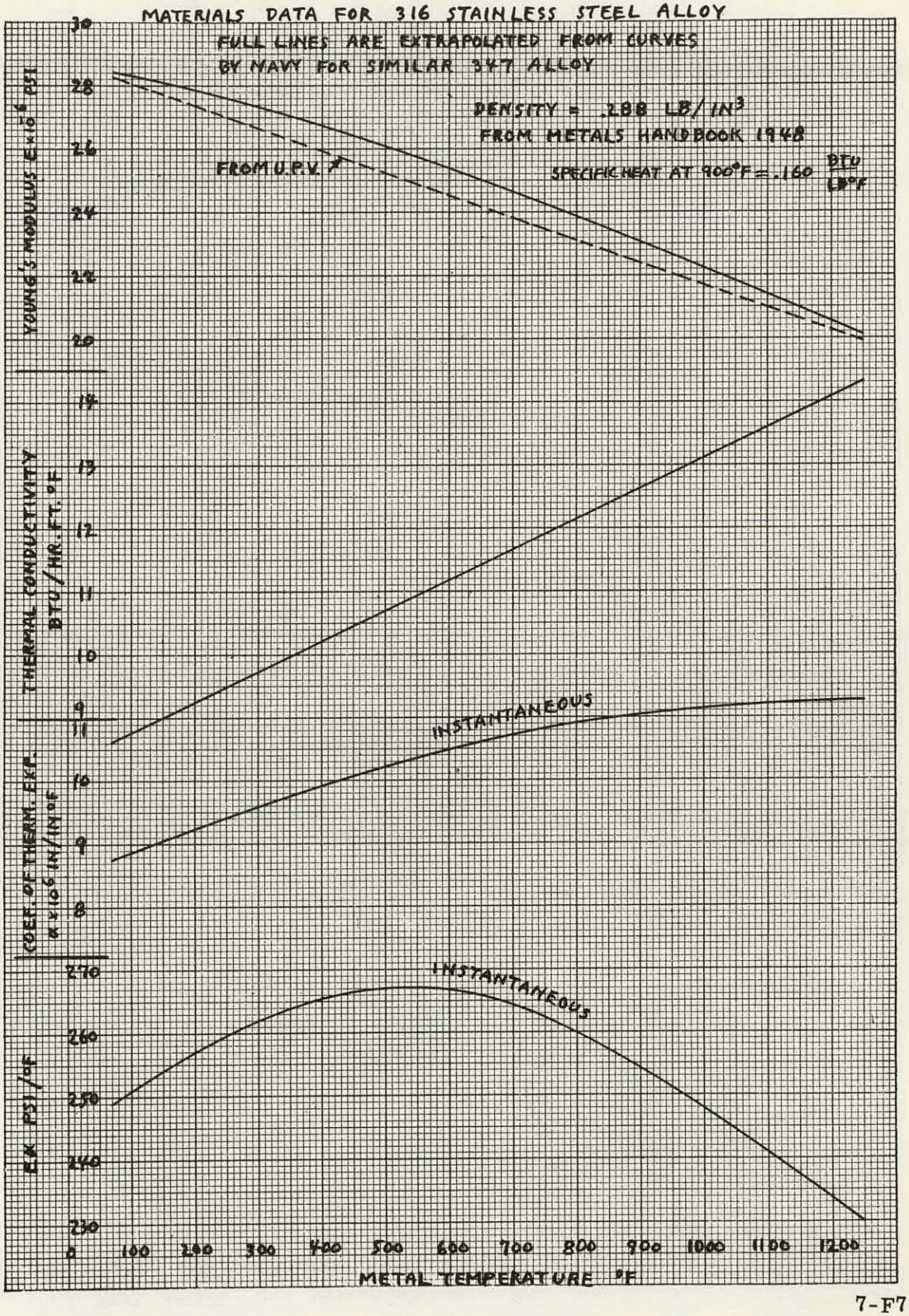
$h = N_{Nu} \frac{K}{d}$

Sodium Temp. °F	w $\frac{lb}{ft^3}$	C_p $\frac{Btu}{lb\,°F}$ From Curves	K $\frac{Btu}{hr\,ft\,°F}$	Péclot No. $N_{pe} = \frac{d u w C_p}{K}$ $\left(du = 56400 \frac{ft^2}{hr}\right)$	Nusselt No. From Formula	$h = N_{Nu} \frac{K}{d}$ $\frac{Btu}{hr\,ft^2\,°F}$
1140	50.0	.2991	36.3	23236	Theor 84.78	3693
					Ave 34.86	1519
895	52.1	.3016	39.2	22608	Theor 83.10	3909
					Ave 34.48	1622
650	54.2	.3083	42.6	22123	Theor 81.78	4181
					Ave 34.18	1747
					Average	2778

For same temperature change in tubes $d = .527$ in $= .043916$ ft with mean velocity of sodium $u = 3.323 \frac{ft}{sec} = 11,763 \frac{ft}{hr}$

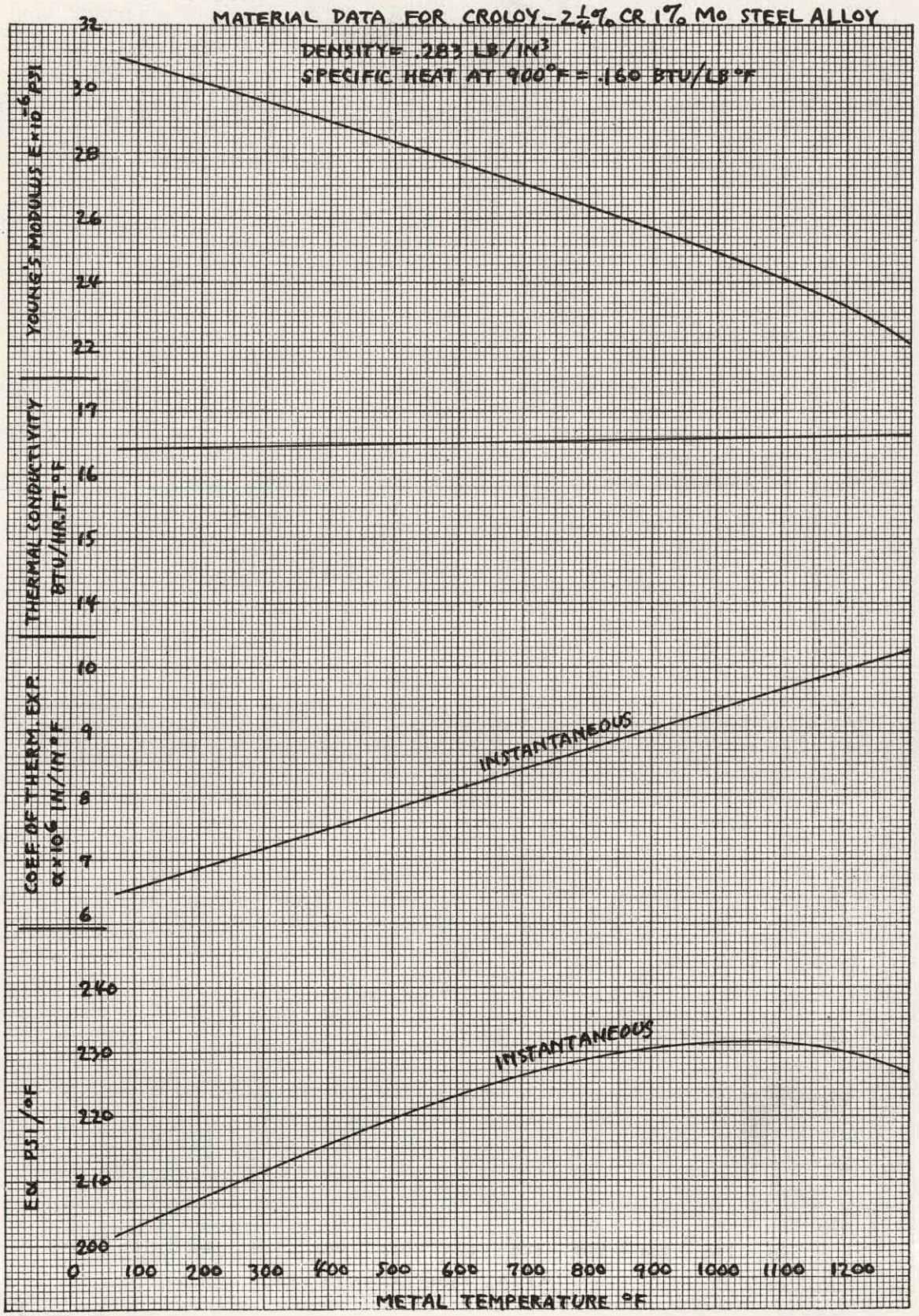
Sodium Temp °F	w $\frac{lb}{ft^3}$	C_p $\frac{Btu}{lb\,°F}$ From Curves	K $\frac{Btu}{hr\,ft\,°F}$	Peclot No. $N_{pe} = \frac{d u w C_p}{K}$ $\left(du = 515.38 \frac{ft^2}{hr}\right)$	Nusselt No. From Formula	$h = N_{Nu} \frac{K}{d}$ $\frac{Btu}{hr\,ft^2\,°F}$
1140	50.0	.2991	36.3	216.5	Theor 8.84	7307
					Ave 5.37	4439
89.5	52.1	.3016	39.2	210.6	Theor 8.80	7855
					Ave 5.31	4740
650	54.2	.3083	42.6	206.1	Theor 8.78	8517
					Ave 5.27	5112
					Average	6330

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MATERIALS DATA FOR 316 STAINLESS STEEL ALLOY
FULL LINES ARE EXTRAPOLATED FROM CURVES
BY NAVY FOR SIMILAR 347 ALLOY
DENSITY = .288 LB/IN³
FROM METALS HANDBOOK 1948
SPECIFIC HEAT AT 900°F = .160 BTU/LB°F
FROM U.P.V.
YOUNG'S MODULUS E×10⁶ PSI
THERMAL CONDUCTIVITY BTU/HR.FT.°F
COEF. OF THERM. EXP. α ×10⁶ IN/IN°F
E/α PSI/°F
INSTANTANEOUS
INSTANTANEOUS
METAL TEMPERATURE °F
0
100
200
300
400
500
600
700
800
900
1000
1100
1200
30
28
26
24
22
20
18
17
12
11
10
9
11
10
9
8
270
260
250
240
230

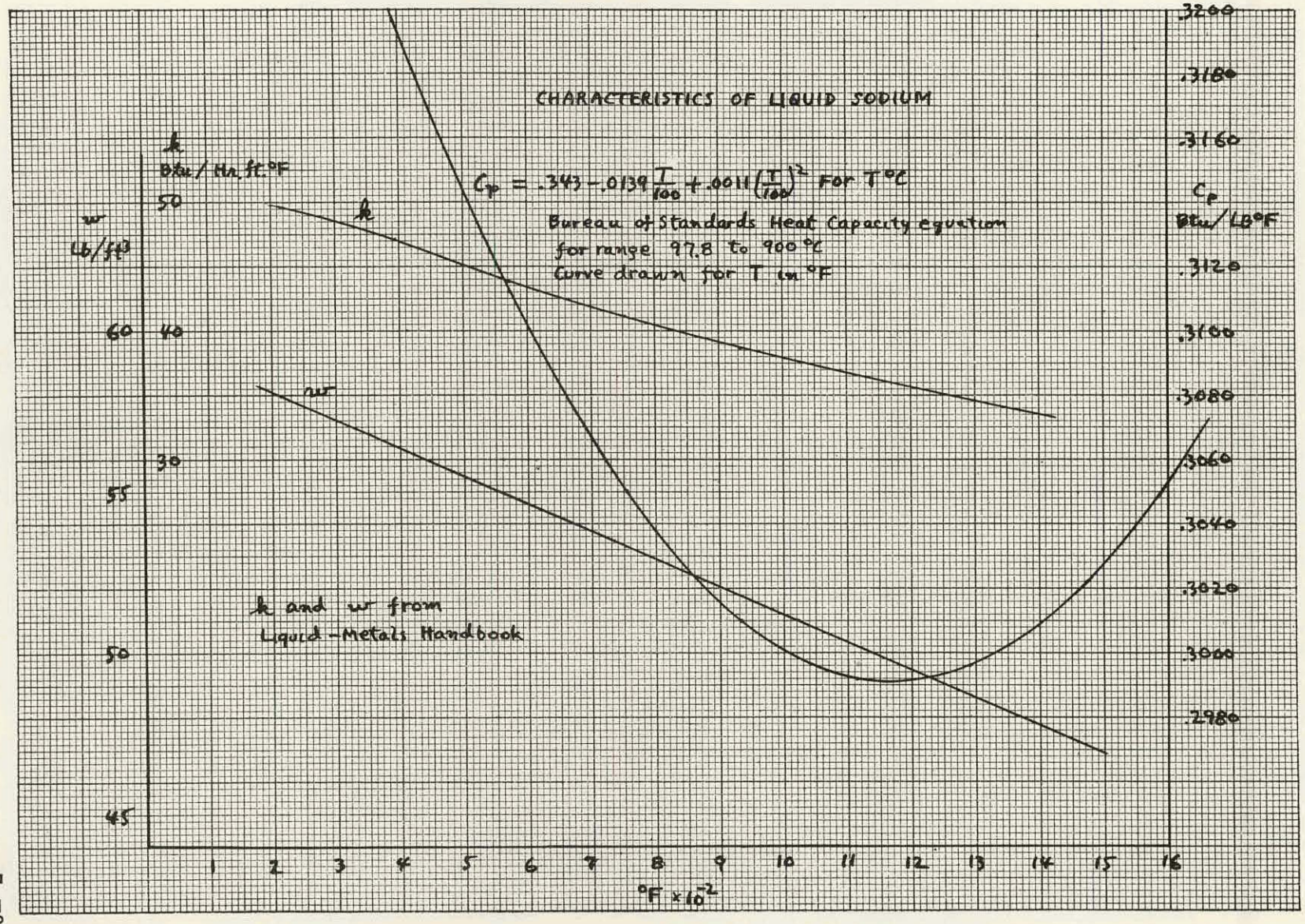
7-F7

367 141


MATERIAL DATA FOR CROLOY-2¼% CR 1% MO STEEL ALLOY
DENSITY= .283 LB/IN³
SPECIFIC HEAT AT 900°F = .160 BTU/LB °F
YOUNG'S MODULUS E x10⁻⁶ PSI
32
30
28
26
24
22
THERMAL CONDUCTIVITY BTU/HR.FT.°F
17
16
15
14
COEF OF THERM. EXP. α x10⁻⁶ IN/IN °F
10
9
8
7
6
INSTANTANEOUS
Ex PSI/°F
240
230
220
210
200
INSTANTANEOUS
0
100
200
300
400
500
600
700
800
900
1000
1100
1200
METAL TEMPERATURE °F

7-F8

967 142


CHARACTERISTICS OF LIQUID SODIUM
$C_p = .343 - .0139\frac{T}{100} + .0011\left(\frac{T}{100}\right)^2$ For T °C
Bureau of Standards Heat Capacity equation
for range 97.8 to 900 °C
Curve drawn for T in °F
k
Btu / Hr. ft. °F
w
Lb/ft³
C_p
Btu/LB°F
k and w from
Liquid-Metals Handbook
°F ×10⁻²

BASIC EQUATIONS

Tube sheet with central solid Core, reinforced by channel and shell walls - Horvay Method.

The Horvay Method takes into account stress concentrations in the ligaments.

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Circular Disk - Uniform Load: q (psi.)

After Timoshenko: $\phi = -\frac{qx^3}{16D} + \frac{C_1 x}{2} + \frac{C_2}{x}$ ____ page 95 - (92) _____ (A)

(Strength of Materials — Part II. — Third Edition)

$M_1 = D\left(\frac{d\phi}{dx} + \mu\frac{\phi}{x}\right)$ _____ page 93 - (88)

$M_2 = D\left(\frac{\phi}{x} + \mu\frac{d\phi}{dx}\right)$ _____ page 93 - (89)

$$\frac{d\phi}{dx} = -\frac{3qx^2}{16D} + \frac{C_1}{2} - \frac{C_2}{x^2}$$

therefore: $M_1 = D\left[-\frac{3qx^2}{16D} + \frac{C_1}{2} - \frac{C_2}{x^2} + \mu\left(-\frac{qx^2}{16D} + \frac{C_1}{2} + \frac{C_2}{x^2}\right)\right] =$

$$= -\frac{3qx^2}{16} + \frac{C_1 D}{2} - \frac{C_2 D}{x^2} - \frac{\mu q x^2}{16} + \frac{D\mu C_1}{2} + \frac{D\mu C_2}{x^2} =$$

$$M_1 = -\frac{(3+\mu)}{16}qx^2 + \frac{(1+\mu)}{2}DC_1 - (1-\mu)\frac{DC_2}{x^2} \quad \text{---------} \quad (B)$$

$$M_2 = D\left[-\frac{qx^2}{16D} + \frac{C_1}{2} + \frac{C_2}{x^2} + \mu\left(-\frac{3qx^2}{16D} + \frac{C_1}{2} - \frac{C_2}{x^2}\right)\right] =$$

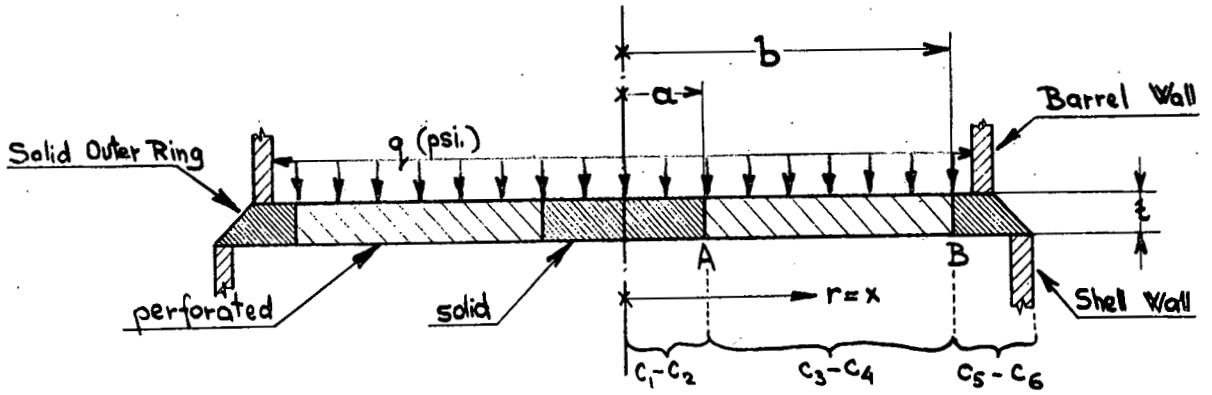
$$= -\frac{qx^2}{16} + \frac{C_1 D}{2} + \frac{C_2 D}{x^2} - \frac{3\mu q x^2}{16} + \frac{D\mu C_1}{2} - \frac{D\mu C_2}{x^2} =$$

$$M_2 = -\frac{(1+3\mu)}{16}qx^2 + \frac{(1+\mu)}{2}DC_1 + (1-\mu)\frac{DC_2}{x^2} \quad \text{-------} \quad (C)$$

For a Disk with solid center, radius "a" and annular perforated area, radius "a" to "b"; equations (A) (B) and (C) apply to area radius "0" to "a", constant "C_2" becomes zero.

For annular area μ becomes μ^* and D becomes D^* (Horvay's notation)

7-F11



(1) $\phi_A = -\frac{qa^3}{16D} + \frac{C_1 a}{2}$

(2) $\phi_A = -\frac{qa^3}{16D^*} + \frac{C_3 a}{2} + \frac{C_4}{a}$

(3) $\phi_B = -\frac{qb^3}{16D^*} + \frac{C_3 b}{2} + \frac{C_4}{b}$

(4) $M_{1A} = -\frac{3.3}{16} qa^2 + 0.65\ DC_1$

(5) $M_{1A} = -\frac{(3+\mu^*)}{16} qa^2 + \frac{(1+\mu^*)}{2} D^* C_3 - (1-\mu^*)\frac{D^* C_4}{a^2}$

(6) $M_{1B} = -\frac{(3+\mu^*)}{16} qb^2 + \frac{(1+\mu^*)}{2} D^* C_3 - (1-\mu^*)\frac{D^* C_4}{b^2}$

From properties of outside solid ring:

(7) $M_{1B} = C_5 \phi_B + C_6$

Using Timoshenko equation – page 139 – (126):

$\phi_B = \frac{M_t R^2}{E I_x}$; where M_t = Ext. Twisting Moment Loading the Outer Ring
R = Radius of Centroid of Outer Ring
E = Modulus of Elasticity
I_x = Moment of Inertia of Outer Ring

$D = \frac{Et^3}{12(1-\mu^2)}$; where $\mu = 0.3$

Beam on an Elastic Foundation.

Basic Equations

$$y = e^{-\beta x}(C\cos\beta x + D\sin\beta x) \quad (1)$$

$$\frac{dy}{dx} = -\beta e^{-\beta x}\left[(C-D)\cos\beta x + (C+D)\sin\beta x\right] \quad (2)$$

$$\frac{d^2y}{dx^2} = \frac{M}{EI} = \beta^2 e^{-\beta x}\left[-2D\cos\beta x + 2C\sin\beta x\right] \quad (3)$$

$$\frac{d^3y}{dx^3} = \frac{V}{EI} = -\beta^3 e^{-\beta x}\left[(2C-2D)\sin\beta x - (2C+2D)\cos\beta x\right] \quad (4)$$

$$\frac{d^4y}{dx^4} = \frac{q_c}{EI} = -\frac{k}{EI}y = -4\beta^4 y \quad (5)$$

For a tube - Stiffness factor $k = \frac{Et}{R^2}$

$$\beta = \sqrt[4]{\frac{k}{4EI}} \qquad \text{where } I = \frac{t^3}{12(1-\mu)} = \frac{t^3}{10.92}$$

$$\beta = \frac{1.285}{\sqrt{Rt}} \quad (6)$$

$$2EI\beta^2 = 2E\frac{t^3}{10.92}\cdot\frac{(1.285)^2}{Rt} = 0.3025\frac{Et^2}{R} \quad (7)$$

R = mean radius. t = wall thickness.

At $x = 0$, these equations become:-

$$y_0 = C \quad (8)$$

$$\theta_0 = (D-C)\beta \quad (9)$$

$$M_0 = -2EI\beta^2 D \quad (10)$$

$$V_0 = 2(C+D)EI\beta^3 \quad (11)$$

967 147

Rearranging eq. (8) to (11):-

$$M_o = -2EI\beta^2\left(y_o + \frac{\theta_o}{\beta}\right) = -.3025\frac{Et^2}{R}\left(y_o + \frac{\theta_o}{\beta}\right) \quad (12)$$

$$V_o = 2EI\beta^2\left(2\beta y_o + \theta_o\right) = .3025\frac{Et^2}{R}\left(2y_o\beta + \theta_o\right) \quad (13)$$

$$y_o = \frac{M_o + V_o/\beta}{2EI\beta^2} \quad (14)$$

$$\theta_o = -\frac{2\beta M_o + V_o}{2EI\beta^2} \quad (15)$$

Bending Stress at origin $= \frac{M_o c}{I}$

$$= \frac{1.815E}{R}\left(y_o + \frac{\theta_o}{\beta}\right) \quad (16)$$

At a distance x from the origin, using Hetenyi's Tables - for A_x, B_x, C_x and D_x:-

$$M = A_x M_o + B_x \frac{V_o}{\beta} \quad (17)$$

$$V = -2B_x\beta M_o + C_x V_o \quad (18)$$

Also. $$M = -2EI\beta^2\left[C_x y_o + D_x \frac{\theta_o}{\beta}\right] \quad (19)$$

$$V = 2EI\beta^2\left[2\beta D_x y_o + A_x \theta_o\right] \quad (20)$$

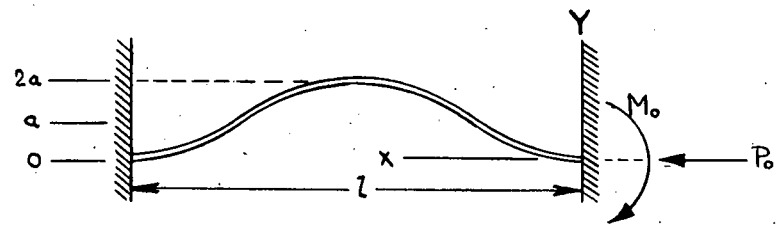
Pre-bent Sine Wave Tubes.

Theory.

To reduce the stresses in the tubes and shell of a fixed tube sheet heat exchanger under differential expansions, the tubes may have part of their length prebent to a sine wave shape or an approximation to that shape.

Forces and moments in the tubes may be computed as follows:

By Theorem of Castigliano:



Strain energy of flexure

$$U = \int \frac{M^2}{2EI}\,dx \equiv \int_0^l \frac{M_x^2}{2EI}\,dx$$

where $M_x = M_o + P_o y$

deflection $\delta = \frac{\partial U}{\partial P_o} = \frac{1}{EI}\int_0^l (M_o + P_o y)\,y\,dx$

slope $\Theta = \frac{\partial U}{\partial M_o} = \frac{1}{EI}\int_0^l (M_o + P_o y)\,dx$

where slope = 0

$$\frac{1}{EI}\int_0^l (M_o + P_o y)\,dx = 0$$

$$\int_0^l M_o\,dx + \int_0^l P_o y\,dx = 0$$

$$M_o l + P_o \int_0^l y\,dx = 0$$

$$M_o = -\frac{P_o}{l}\int_0^l y\,dx$$

$$\delta = \frac{1}{EI}\left[\int_0^l M_o y\,dx + \int_0^l P_o y^2\,dx\right] =$$

$$= \frac{1}{EI}\left[M_o\int_0^l y\,dx + P_o\int_0^l y^2\,dx\right]$$

Assume bend of tube takes shape $y = a\left(1-\cos\frac{2\pi x}{l}\right)$

then $\int_0^l y\,dx = a\int_0^l \left(1-\cos\frac{2\pi x}{l}\right)dx = a\left[\int_0^l dx - \int_0^l \cos\frac{2\pi x}{l}\,dx\right] =$

$$= a\left[(x)_0^l - \left(\frac{l}{2\pi}\sin\frac{2\pi x}{l}\right)_0^l\right] =$$

$$= a\left[(1-0)-(0-0)\right] =$$

$$= al$$

and $\int_0^l y^2\,dx = \int_0^l a^2\left(1-\cos\frac{2\pi x}{l}\right)^2 dx = a^2\int_0^l \left(1-2\cos\frac{2\pi x}{l}+\cos^2\frac{2\pi x}{l}\right)dx =$

$$= a^2\left[\int_0^l dx - \int_0^l 2\cos\frac{2\pi x}{l}\,dx + \int_0^l \cos^2\frac{2\pi x}{l}\,dx\right] =$$

$$= a^2\left[(x)_0^l - \left(\frac{2l}{2\pi}\sin\frac{2\pi l}{x}\right)_0^l + \left(\frac{l}{4\pi}\left\{\frac{2\pi x}{l}+\sin\frac{2\pi x}{l}\times\cos\frac{2\pi x}{l}\right\}\right)_0^l\right] =$$

$$= a^2\left[(1-0)-(0-0)+\left(\frac{1}{4\pi}\left\{2\pi+0\times 1\right\}-\frac{1}{4\pi}\left\{0+0\times 1\right\}\right)\right] =$$

$$= a^2\times\frac{3l}{2} =$$

$$= \frac{3a^2 l}{2}$$

then since $M_o = -\frac{P_o}{l}\int_0^l y\,dx = -\frac{P_o}{l}\times al = -P_o a$

and since $\delta = \frac{1}{EI}\left[M_o\int_0^l y\,dx + P_o\int_0^l y^2\,dx\right] =$

$$= \frac{1}{EI}\left[-P_o a\times al + P_o\times\frac{3a^2 l}{2}\right] =$$

$$= \frac{1}{EI}\left[-P_o a^2 l + \frac{3}{2}P_o a^2 l\right] =$$

$$= \frac{P_o a^2 l}{2EI}$$

CCT 150

Maximum bending moment at mid point and at ends of wave, will equal $P_o a$.

$$\text{whence } M_{max.} = \frac{2EI}{al}\delta$$

$$\text{max. bending stress} = \frac{Mc}{I} = \frac{2Ec}{al}\delta = \frac{Er_o}{al}\delta$$

where r_o = outside radius of tube
r_i = inside radius of tube

$$\text{Stress due to the direct load } P_o = \frac{2EIS}{a^2 l \times \text{metal area}}$$

$$\text{But } I = \frac{\pi}{4}\left(r_o^4 - r_i^4\right)$$

$$\text{metal area} = \pi\left(r_o^2 - r_i^2\right)$$

$$\text{Stress} = \frac{2E\delta}{a^2 l} \times \frac{r_o^2 + r_i^2}{4} = \text{approx } \frac{E r_o r_i}{a^2 l}\delta$$

Combining bending and direct stress:

$$S = \frac{Er_o}{al}\delta + \frac{Er_o r_i}{a^2 l}\delta =$$

$$= \frac{Er_o}{al}\left(1 + \frac{r_i}{a}\right)\delta$$

The tube could alternatively be shaped as a succession of circular arcs without appreciably changing these values.

Pre-bent Sine Wave Tubes.

Deflection $\delta = \frac{P_o a^2 l}{2EI}$

by strain energy method

$M_{max} = P_o a = \frac{SI}{c}$

whence $\delta = \frac{Sal}{E d_o}$

$a = \frac{10}{2} = 5$

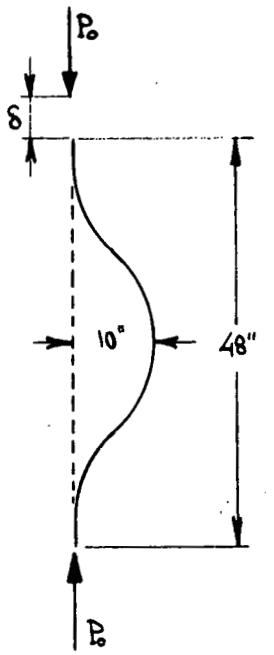
$l = 48$

S = bending stress

d_o = O.D. of tubes

$c = \frac{d_o}{2}$

E = modulus of elasticity



48" Wave – 10" Offset:

For 3/4" tube – $\delta = \frac{S \times 5 \times 48}{0.75\,E} = \frac{320\,S}{E}$

" 5/8" tube – $\delta = \frac{S \times 5 \times 48}{0.625\,E} = \frac{384\,S}{E}$

" 1/2" tube – $\delta = \frac{S \times 5 \times 48}{0.5\,E} = \frac{480\,S}{E}$

Values of E:

	Temp.	# 316	Croloy
Isothermal Hot	1200°	$20.5\ (10)^6$	$23.2\ (10)^6$
Normal Operation	1150°	$21.0\ (10)^6$	$23.6\ (10)^6$
Isothermal Cold	640°	$25.0\ (10)^6$	$27.4\ (10)^6$
Mean value of α		$11.0\ (10)^{-6}$	$9.0\ (10)^{-6}$

60" Wave – 12 1/2" Offset:

For 1/2" tube – $\delta = \frac{S \times 6.25 \times 60}{0.50\,E} = \frac{750\,S}{E}$ $\qquad S = \frac{E\delta}{750}$

36" Wave – 7 1/2" Offset:

For 1/2" tube – $\delta = \frac{S \times 3.75 \times 36}{0.50\,E} = \frac{270\,S}{E}$ $\qquad S = \frac{E\delta}{270}$

Force $P = \frac{E}{a^2 L/2I} \times$ deflection

Moment of Inertia :

	O.D.	I.D.	$(O.D)^2$	$(ID)^2$	I	$\frac{a^2 L}{2I}$ 48" Wave 10" Offset	$\frac{24\,(10)^6}{a^2 L/2I}$ Thrust for 1" Deflection
½"ϕ 18 gage	0.500	0.402	0.250	0.1615	0.001788	335 000	71.7 lb
¾" ϕ 17 "	0.750	0.634	0.5625	0.4020	0.00760	79 000	304 lb.
15 "	"	0.606	"	0.3673	0.00891	67 300	356 "
12 "	"	0.532	"	0.2830	0.01160	51 700	464 "
Thick wall	"	0.232	"	0.0538	0.01540	39 000	615 "

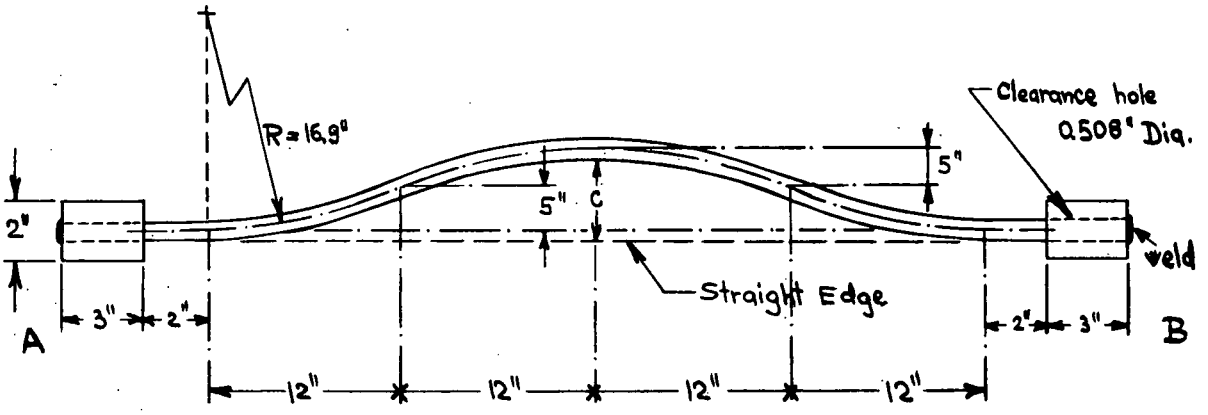
Lateral Movement = 1.108 × Vertical Movement

½"ϕ 20 gage	0.500	0.430	0.250	0.1849	0.001390	432 000	@ 700°F 55.5 lb.

@ 70°F 64.8 lb.

Endurance Test on Sine Wave Tubing.

Stainless Steel Tube – ½" outside dia. –20 Gage.
Bent to form wave 48" long – 10" maximum offset – as shown.



Brass block A was clamped in head stock of lathe. Block B was held in tool stock, in line with A, with tube wave in equilibrium position. Offset C as measured from straight edge held against tube at ends, equalled 9 7/8".

To simulate behavior of tube wave under extreme service conditions, tool stock was moved to right a distance D = 3/4". Offset C = 9 1/16". Tool stock was moved to equilibrium position and beyond by 3/4". Offset distance = 10 9/16" Tool stock was then cycled back and forth as shown the accompanying table:

	Offset C Measurement			Movement		Computed
	Right	Neutral	Left	Right	Left	Bending Stress
1 Original Measurement		9.87				
2 3/4" deflection	9.06		10.56	–.81	+.69	
3 After 25 cycles at 3/4"	9.06		10.56	–.81	+.69	± 44 000 psi.
4 " 25 " at 1.0"	8.81		10.81	–1.06	+.94	± 58 000 psi.
5 " 25 " at 1.25"	8.50		11.06	–1.37	+1.19	± 73 000 psi.
6 " 25 " at 1.50"	8.25		11.37	–1.62	+1.50	± 87 000 psi.
7 " 25 " at 2.00"	7.62		11.87	–2.25	+2.00	± 117 000 psi.

From the final compressed position 2.0" left, the tool stock was brought back to its mid position, and the tube released. The tube sprang back 0.44" toward the left, indicating a final "permanent set" in length of that amount. Offset C measurement after release = 10.31", or a set of 0.44" laterally.

After 125 cycles, ranging up to double that expected unde maximum causalty conditions, the tube showed no damage whatever.

7-F20

Intermediate Heat Exchanger with Hemispherical Barrel and with Sine Wave Tubes.

Dimensions: I.D. of Shell _____ = 49"
I.D. of Barrel (at bottom, where is cylindrical) _____ = 44"
Wall Thicknesses: Tube Sheet _____ = $3\frac{1}{2}$" = 3.500"
Shell _____ = $1\frac{3}{8}$ = 1.375"
Barrel _____ = $1\frac{3}{8}$ = 1.375"
½" O.D. Tubes — 1" radial pitch — 1" circular pitch
Extreme Inside Radius of Tubes _____ a = 7.3125"
Extreme Outside Radius of Tubes _____ b = 19.8125"

Assume (a) internal pressure at the tube side: $p = q_1 = 125$ psi.
at the shell side: $p = 0$
(b) center pipe not load bearing
(c) ignoring the various reaction forces of tubes
(d) the effect of Barrel Wall and Shell Wall is acting in the level of outer surfaces of Tube Sheet.

Base data: Horvay's Ratio at perforated portion: $\eta = \frac{2h}{2r} = \frac{0.50}{1.00} = 0.50$

$$D = \frac{Et^3}{12(1-\mu^2)} = \frac{(21)(10)^6 (3.500)^3}{(10.92)} = 82.4\ (10)^6$$

$\mu = 0.30$

$\mu^* = 0.40$

$D = 82.4\ (10)^6$

$D^* = 0.42\ D = (0.42)(82.4)(10)^6 = 34.60\ (10)^6$

$E = 21\ (10)^6$

Shell Wall Effect:

Shell wall = 1.375
Mean Radius = 25.1875
Tube sheet acting arm = 1.750

Assuming only radial movement of shell wall edge!

total radial displacement: $y_o = 1.750\ \phi_B$

After Hetényi: $M_s = \frac{k}{2\beta^2}\left(y_o + \frac{\phi_B}{\beta}\right)$

$Q_s = \frac{k}{2\beta}\left(2y_o + \frac{\phi_B}{\beta}\right)$

$\beta = \frac{1.285}{\sqrt{(25.1875)(1.375)}} = 0.2182$ $\qquad k = \frac{(21)(10)^6(1.375)}{(25.1875)^2} = 45{,}550$

$M_s = \frac{45{,}550}{0.09524}\left(1.750\ \phi_B + \frac{1}{0.2182}\phi_B\right) = \frac{45{,}550}{0.09524}\left(6.330\ \phi_B\right) = \underline{3.028\ (10)^6\ \phi_B}$

$Q_s = \frac{45{,}550}{0.4364}\left(3.500\ \phi_B + \frac{1}{0.2182}\phi_B\right) = \frac{45{,}550}{0.4364}\left(8.080\ \phi_B\right) = \underline{0.843\ (10)^6\ \phi_B}$

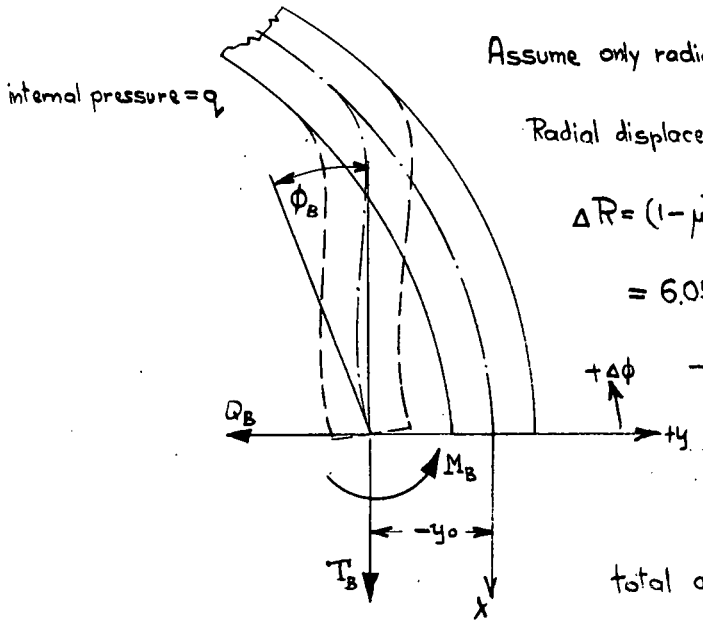
Effect of Hemispherical Barrel Wall:

Ignore the short cylindrical portion and assume direct connection of hemispherical barrel wall to the tube sheet!

Symbols and data:

Outside radius of barrel _____: $r_o = 23.375"$
Mean —"— —"— _____: $R = 22.6875"$
Inside —"— —"— _____: $r_i = 22.000"$
Thickness of barrel wall _____: $t = 1.375"$
Tube sheet acting arm _____ = 1.750"

q = Internal pressure in barrel
M_B = Bending moment
Q_B = Shearing force
T_B = Longitudinal Tensional Force



Assume only radial displacement of barrel wall edge! $\Delta X = 0$

Radial displacement due to barrel internal pressure:

$$\Delta R = (1-\mu)\frac{r_i R}{2Et}q = 0.7\,\frac{(22.000)(22.6875)}{2(21)(1.375)(10)^6}q =$$

$$= 6.050\,(10)^{-6}q$$

Therefore the total radial displacement of barrel wall edge:

$$y_o = 1.750\,\phi_B + 6.050\,(10)^{-6}q$$

total angular movement = ϕ_B

Equations are from "Formulas for Stress and Strain" - Roark (page 272-273, Ref. 14)
Radial displacement and angular movement of wall at edge are positive when outward!

$$\beta = \sqrt[4]{3(1-\mu^2)\left(\frac{R}{t}\right)^2} = \sqrt[4]{3(1-0.09)\left(\frac{22.6875}{1.375}\right)^2} = 5.220$$

(1) $$-y_o = -\frac{2Q_B}{Et}(\beta R) + \frac{M_B}{Et}(2\beta^2) = -\frac{2\beta R}{Et}Q_B + \frac{2\beta^2}{Et}M_B$$

(2) $$+\phi_B = -\frac{Q_B}{Et}(2\beta^2) + \frac{M_B}{Et}\left(\frac{4\beta^3}{R}\right) = -\frac{2\beta^2}{Et}Q_B + \frac{4\beta^3}{EtR}M_B$$

from (2): $$\frac{2\beta^2}{Et}Q_B = -\phi_B + \frac{4\beta^3}{EtR}M_B$$

$$\therefore Q_B = \frac{Et}{2\beta^2}\left(-\phi_B + \frac{4\beta^3}{EtR}M_B\right) = -\frac{Et}{2\beta^2}\phi_B + \frac{2\beta}{R}M_B$$

Substituting to (1):

$$-y_o = -\frac{2\beta R}{Et}\left(-\frac{Et}{2\beta^2}\phi_B + \frac{2\beta}{R}M_B\right) + \frac{2\beta^2}{Et}M_B$$

$$-y_o = \frac{R}{\beta}\phi_B - \frac{4\beta^2}{Et}M_B + \frac{2\beta^2}{Et}M_B = \frac{R}{\beta}\phi_B - \frac{2\beta^2}{Et}M_B$$

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Therefore: $M_B = \frac{Et}{2\beta^2}\left(\frac{R}{\beta}\phi_B + y_o\right)$

$$M_B = \frac{(21)(10)^6(1.375)}{(2)(5.220)^2}\left[\frac{22.6875}{5.220}\phi_B + 1.750\,\phi_B + 6.050\,(10)^{-6}q_v\right] =$$

$$= 0.5295\,(10)^6\left[6.094\,\phi_B + 6.050\,(10)^{-6}q_v\right]$$

$$\underline{\underline{M_B = 3.230\,(10)^6\phi_B + 3.206\,q_v}}$$

(1) $$-1.750\,\phi_B - 6.050\,(10)^{-6}q_v = -\frac{2\beta R}{Et}Q_B + \frac{2\beta^2}{Et}\left[3.230\,(10)^6\phi_B + 3.206\,q_v\right]$$

$$\frac{2\beta R}{Et}Q_B = 1.750\,\phi_B + 6.050\,(10)^{-6}q_v + \frac{2(5.220)^2}{(21)(1.375)}(10)^{-6}\left[3.230\,(10)^6\phi_B + 3.206\,q_v\right] =$$

$$= 1.750\,\phi_B + 6.050\,(10)^{-6}q_v + 6.095\,\phi_B + 6.045\,(10)^{-6}q_v =$$

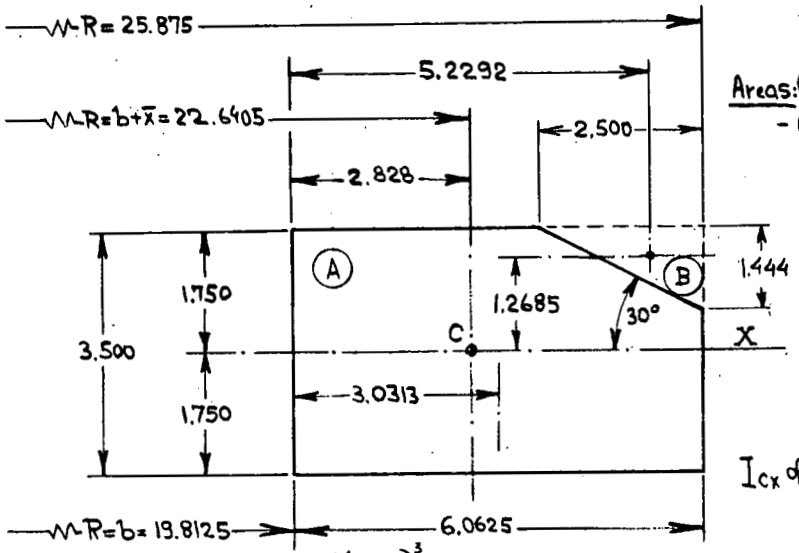
$$= 7.845\,\phi_B + 12.095\,(10)^{-6}q_v$$

$$Q_B = \frac{Et}{2\beta R}\left[7.845\,\phi_B + 12.095\,(10)^{-6}q_v\right] = \frac{(21)(10)^6(1.375)}{2(5.220)(22.6875)}\left[7.845\,\phi_B + 12.095\,(10)^{-6}q_v\right]$$

$$\underline{\underline{Q_B = 0.956\,(10)^6\phi_B + 1.475\,q_v}}$$

$$\underline{\underline{T_B = \frac{q\,r_i}{2t}t = \frac{r_i}{2}q_v = \frac{22.000}{2}q_v = 11.000\,q_v}}$$

Cross section of outer solid ring:



Ⓐ' represents the total assumed square cross section

Areas: Ⓐ' = (6.0625)(3.500) = 21.225

− Ⓑ = −(1.250)(1.444) = − 1.805

Ⓐ = 19.420 in.²

Ⓐ' × (x̄_A') = (21.225)(3.0313) = 64.350

− Ⓑ × (x̄_B) = −(1.805)(5.2292) = −9.438

54.912

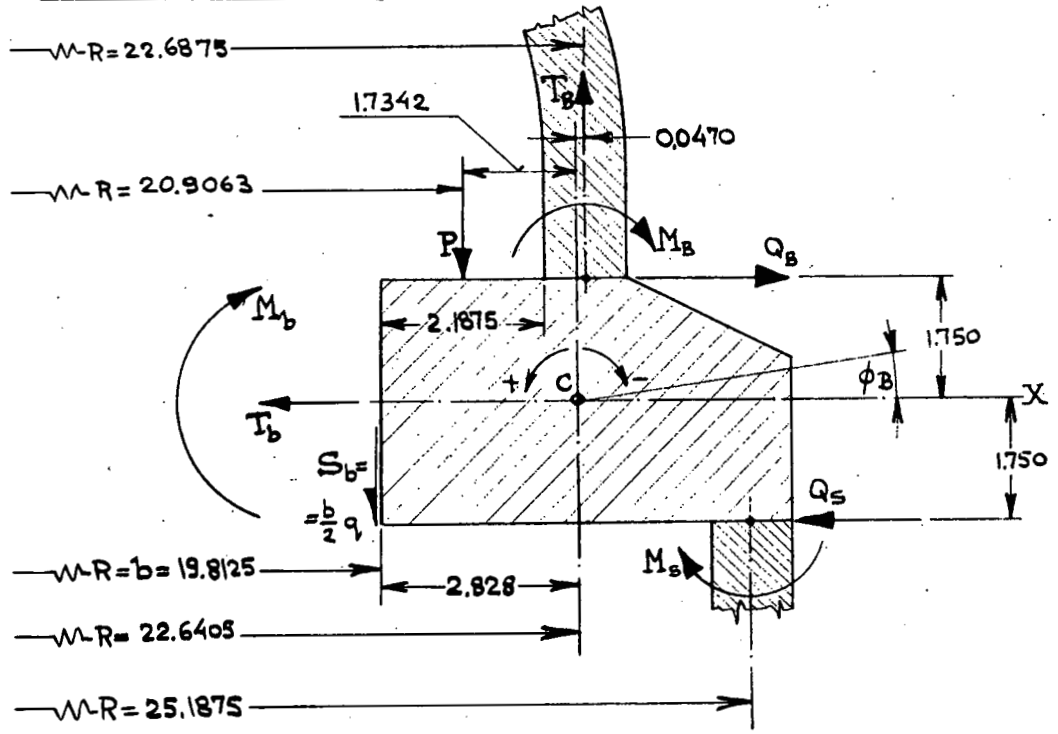
$\bar{x} = \frac{54.912}{19.420} = 2.828$ in.

I_{cx} of Ⓐ' $= \frac{(6.0625)(3.500)^3}{12} = 21.670$

I_{cx} of Ⓑ $= \frac{(2.500)(1.444)^3}{36} + (1.805)(1.2685)^2 = 0.210 + 2.902 = 3.112$

I_{cx} of Ⓐ $= 21.670 - 3.112 = 18.558$ in.⁴

Forces and Moments acting of outer solid ring:



7-F25

External twisting Moment "M_t" per unit length at center "C":

due to $M_b = -M_b \frac{19.8125}{22.6405} = -0.8748\ M_b$

" " $S_b = +3.9063\,q\,(2.828)\frac{19.8125}{22.6405} = +24.500\ q$

" " $P = +2.1875\,q\,(1.7342)\frac{20.9063}{22.6405} = +3.502\ q$

" " $T_B = +11.000\,q\,(0.0470)\frac{22.6875}{22.6405} = +0.518\ q$

" " $M_B = -\left[3.230\,(10)^6\,\phi_B + 3.206\,q\right]\frac{22.6875}{22.6405} = -3.236\,(10)^6\,\phi_B - 3.211\ q$

" " $Q_B = -\left[0.956\,(10)^6\,\phi_B + 1.475\,q\right](1.750)\frac{22.6875}{22.6405} = -1.675\,(10)^6\,\phi_B - 2.584\ q$

" " $M_s = -3.028\,(10)^6\,\phi_B\,\frac{25.1875}{22.6405} = -3.367\,(10)^6\,\phi_B$

" " $Q_s = -0.843\,(10)^6\,\phi_B\,(1.750)\frac{25.1875}{22.6405} = -1.640\,(10)^6\,\phi_B$

After summation: $M_t = -0.8748\ M_b - 9.918\,(10)^6\,\phi_B + 22.725\ q$

Now, $\phi_B = \frac{M_t R^2}{E I_x}$

$\therefore \phi_B = \left(-0.8748\ M_b - 9.918\,(10)^6\,\phi_B + 22.725\ q\right)\frac{(22.6405)^2}{(21)(18.558)}(10)^{-6} =$

$= \left(-0.8748\ M_b - 9.918\,(10)^6\,\phi_B + 22.725\ q\right)(1.316)\,(10)^{-6} =$

$= -1.151\,(10)^{-6}\ M_b - 13.050\ \phi_B + 29.910\,(10)^{-6}\ q$

$\therefore \phi_B = \frac{-1.151\,(10)^{-6}\ M_b + 29.910\,(10)^{-6}\ q}{14.050} = -0.08195\,(10)^{-6}\ M_b + 2.129\,(10)^{-6}\ q$

$\therefore M_b = -\frac{1}{0.08195}(10)^6\,\phi_B + \frac{2.129}{0.08195}\ q = -12.200\,(10)^6\,\phi_B + 25.97\ q$

$M_{1B} = -12.200\,(10)^6\,\phi_B + 25.97\ q$

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Using paper of "Basic Equations" – Oct. 28, 1958

(1) $\phi_A = -\frac{a^3}{16D} q + \frac{a}{2} C_1$

(2) $\phi_A = -\frac{a^3}{16 D^*} q + \frac{a}{2} C_3 + \frac{1}{a} C_4$

(3) $\phi_B = -\frac{b^3}{16 D^*} q + \frac{b}{2} C_3 + \frac{1}{b} C_4$

(4) $M_{1A} = -\frac{(3.3) a^2}{16} q + 0.65 D C_1$

(5) $M_{1A} = -\frac{(3+\mu^*) a^2}{16} q + \frac{(1+\mu^*) D^*}{2} C_3 - \frac{(1-\mu^*) D^*}{a^2} C_4$

(6) $M_{1B} = -\frac{(3+\mu^*) b^2}{16} q + \frac{(1+\mu^*) D^*}{2} C_3 - \frac{(1-\mu^*) D^*}{b^2} C_4$

(7) $M_{1B} = C_5 \phi_B + C_6$

$a = 7.3125$
$a^2 = 53.45$
$a^3 = 390.8$

$b = 19.8125$
$b^2 = 392.4$
$b^3 = 7775$

$D = 82.4\ (10)^6$
$D^* = 34.6\ (10)^6$
$\mu^* = 0.40$

$\frac{a^3}{16 D} = \frac{390.8}{16(82.4)} (10)^{-6} = 0.2963\ (10)^{-6}$

$\frac{a}{2} = 3.656$

$\frac{a^3}{16 D^*} = \frac{390.8}{16(34.6)} (10)^{-6} = 0.7059\ (10)^{-6}$

$\frac{1}{a} = 0.1368$

$\frac{b^3}{16 D^*} = \frac{7775}{16(34.6)} (10)^{-6} = 14.040\ (10)^{-6}$

$\frac{b}{2} = 9.906$

$\frac{(3.3) a^2}{16} = \frac{(3.3)(53.45)}{16} = 11.024$

$\frac{1}{b} = 0.05048$

$\frac{(3+\mu^*) a^2}{16} = \frac{(3.4)(53.45)}{16} = 11.355$

$0.65\ D = 53.57\ (10)^6$

$\frac{(3+\mu^*) b^2}{16} = \frac{(3.4)(392.4)}{16} = 83.39$

$\frac{(1+\mu^*) D^*}{2} = 24.22\ (10)^6$

$\frac{(1-\mu^*) D^*}{a^2} = \frac{(0.6)(34.6)}{53.45} (10)^6 = 0.3884\ (10)^6$

$\frac{(1-\mu^*) D^*}{b^2} = \frac{(0.6)(34.6)}{392.4} (10)^6 = 0.0529\ (10)^6$

7-F27

After substituting: (1) $\phi_A = -0.2963\,(10)^{-6} q_1 + 3.656\, C_1$

(2) $\phi_A = -0.7059\,(10)^{-6} q_1 + 3.656\, C_3 + 0.1368\, C_4$

(3) $\phi_B = -14.040\,(10)^{-6} q_1 + 9.906\, C_3 + 0.05048\, C_4$

(4) $M_{1A} = -11.024\, q_1 + 53.57\,(10)^6\, C_1$

(5) $M_{1A} = -11.355\, q_1 + 24.22\,(10)^6\, C_3 - 0.3884\,(10)^6\, C_4$

(6) $M_{1B} = -83.39\, q_1 + 24.22\,(10)^6\, C_3 - 0.0529\,(10)^6\, C_4$

(7) $M_{1B} = -12.200\,(10)^6\, \phi_B + 25.97\, q_1$

(1) = (2) $-0.2963\,(10)^{-6} q_1 + 3.656\, C_1 + 0.7059\,(10)^{-6} q_1 - 3.656\, C_3 - 0.1368\, C_4 = 0$

(8) $0.4096\,(10)^{-6} q_1 + 3.656\, C_1 - 3.656\, C_3 - 0.1368\, C_4 = 0$, $\left[K = \frac{53.57}{3.656}\, 10^6 = 14.652\,(10)^6\right]$

(4) = (5) $-11.024\, q_1 + 53.57\,(10)^6\, C_1 + 11.355\, q_1 - 24.22\,(10)^6\, C_3 + 0.3884\,(10)^6\, C_4 = 0$

(9) $0.331\, q_1 + 53.57\,(10)^6\, C_1 - 24.220\,(10)^6\, C_3 + 0.3884\,(10)^6\, C_4 = 0$

(8)x(-k) $-6.000\, q_1 - 53.57\,(10)^6\, C_1 + 53.57\,(10)^6\, C_3 + 2.0030\,(10)^6\, C_4 = 0$ (8)'

(8)' + (9) $-5.669\, q_1 + 29.350\,(10)^6\, C_3 + 2.3914\,(10)^6\, C_4 = 0$ (10)

(6)(7)(3) $-83.39\, q_1 + 24.22\,(10)^6\, C_3 - 0.0529\,(10)^6\, C_4 =$

$= 25.97\, q_1 - 12.200\,(10)^6 \left[-14.040\,(10)^{-6} q_1 + 9.906\, C_3 + 0.05048\, C_4\right]$

$= 25.97\, q_1 + 171.250\, q_1 - 120.850\,(10)^6\, C_3 - 0.616\,(10)^6\, C_4$

(11) $280.61\, q_1 - 145.07\,(10)^6\, C_3 - 0.5631\,(10)^6\, C_4 = 0$ $\left[z = \frac{145.07}{29.35} = 4.942\right]$

(10)x(z) $-28.03\, q_1 + 145.07\,(10)^6\, C_3 + 11.825\,(10)^6\, C_4 = 0$ (10)'

(10)' + (11) $252.58\, q_1 = -11.2619\,(10)^6\, C_4$

$$C_4 = -\frac{252.58}{11.2619}(10)^{-6} q_1 = -22.42\ (10)^{-6} q_1$$

from (10) $29.350\ (10)^6 C_3 = 5.669\ q_1 + (2.3914)(22.42)\ q_1 = (5.669 + 53.650)\ q_1 = 59.319\ q_1$

$$C_3 = \frac{59.319}{29.350}(10)^{-6} q_1 = 2.021\ (10)^{-6} q_1$$

$$\boxed{C_3 = 2.021\ (10)^{-6} q_1 \quad ; \quad C_4 = -22.42\ (10)^{-6} q_1}$$

Using paper of "Basic Equations"

Moments in perforated portion at any $a \le x \le b$

$$M_{1x} = -\frac{(3+\mu^*)}{16} x^2 q_1 + \frac{(1+\mu^*)}{2} D^* C_3 - (1-\mu^*) D^* C_4 \frac{1}{x^2}$$

$$M_{2x} = -\frac{(1+3\mu^*)}{16} x^2 q_1 + \frac{(1+\mu^*)}{2} D^* C_3 + (1-\mu^*) D^* C_4 \frac{1}{x^2}$$

$\frac{(3+\mu^*)}{16} = \frac{3.40}{16} = 0.2125$, $\frac{(1+\mu^*)}{2} D^* C_3 = (0.7)(34.60)(2.021)\ q_1 = 48.97\ q_1$

$\frac{(1+3\mu^*)}{16} = \frac{2.20}{16} = 0.1375$, $(1-\mu^*) D^* C_4 = (0.6)(34.60)(-22.42)\ q_1 = -465.7\ q_1$

Therefore the Moments: $M_{1x} = -0.2125\ x^2 q_1 + 48.97\ q_1 + \frac{465.7}{x^2} q_1$

$$M_{2x} = -0.1375\ x^2 q_1 + 48.97\ q_1 - \frac{465.7}{x^2} q_1$$

Moments at the middle radius of first inside tubes: $x = 7.5625''$ = radius ; $x^2 = 57.20$

$$M_1 = (-12.16 + 48.97 + 8.14)\ q_1 = +44.95\ q_1$$
$$M_2 = (-7.86 + 48.97 - 8.14)\ q_1 = +32.97\ q_1$$

Moments at the middle radius of first outside tubes: $x = 19.5625''$ = radius ; $x^2 = 382.7$

$$M_1 = (-81.32 + 48.97 + 1.22)\ q_1 = -31.13\ q_1$$
$$M_2 = (-52.60 + 48.97 - 1.22)\ q_1 = -4.85\ q_1$$

Max. ligament stresses without Horvay's factor, when the barrel pressure - q = 125 psi.

$$\sigma_{inside} = \frac{6 M_1}{t^2} = \frac{6(44.95)}{(3.500)^2} = 22.02\, q = (22.02)(125) = 2750 \text{ psi.}$$

$$\sigma_{outside} = \frac{6 M_1}{t^2} = \frac{6(31.13)}{(3.500)^2} = 15.25\, q = (15.25)(125) = 1910 \text{ psi.}$$

Ratio of principal stresses: Inside : $\frac{+32.97}{+44.95} = +0.733$

Outside : $\frac{-4.85}{-31.13} = +0.155$

Horvay's stress multiplier (at $\eta = 0.50$) : $Y_{inside} = 2.61$

$Y_{outside} = 3.32$

Therefore the max. stresses: $\sigma_{inside} = (2,750)(2.61) = 7,180$ psi. (maximum)

$\sigma_{outside} = (1,910)(3.32) = 6,340$ psi.

At any other portion of perforated part the stresses are smaller!

Controling result of moments:

Bending Moment at $x=b=19.8125$, $x^2=392.4$

$$M_{1B}=(-83.38+48.97+1.19)\,q_1=\underline{\underline{-33.22\,q_1}}$$

On the other hand the bending moment at $x=b$ is from equation of twisted outer solid ring:

$$M_{1B}=-12.200\,(10)^6\,\phi_B+25.97\,q_1$$

where: $\phi_B=-14.040\,(10)^{-6}\,q_1+9.906\,C_3+0.05048\,C_4=$

$$=-14.040\,(10)^{-6}\,q_1+(9.906)(2.021)(10)^{-6}\,q_1+(0.05048)(-22.42)(10)^{-6}\,q_1=$$

$$=(-14.040+20.020-1.132)(10)^{-6}\,q_1=$$

$$=4.848\,(10)^{-6}\,q_1$$

Therefore:

$$M_{1B}=-12.200\,(4.848)\,q_1+25.97\,q_1=(-59.13+25.97)\,q_1=$$

$$=\underline{\underline{-33.16\,q_1}}$$

Difference = 0.2 %

Analysis of Working Stresses in barrel wall due to internal pressure in barrel and the tube sheet effect. Using in this calculation the actual cylindrical barrel wall connection.

Barrel internal pressure: $q_1 = 125$ psi.

Bending Moment: $M_B = 3.230\,(10)^6 \phi_B + 3.206\, q_1$

$= (3.230)(4.848)\, q_1 + 3.206\, q_1 = (15.66 + 3.206)\, q_1 =$

$= 18.866\, q_1$

Shearing Force: $Q_B = 0.956(10)^6 \phi_B + 1.475\, q_1$

$= (0.956)(4.848)\, q_1 + 1.475\, q_1 = (4.635 + 1.475)\, q_1 =$

$= 6.110\, q_1$

Therefore: Bending Stress:

$$\sigma_M = \frac{6 M_B}{t^2} = \frac{6(18.866)}{(1.375)^2} q_1 = 59.90\, q_1 = \underline{7,500 \text{ psi.}}$$

Shearing Stress:

$$\sigma_Q = \frac{Q_B}{t} = \frac{6.110}{1.375} q_1 = 4.45\, q_1 = \underline{560 \text{ psi.}}$$

Longitudinal Tension:

$$\sigma_L = \frac{R}{2t} q_1 = \frac{22.000}{2.750} q_1 = 8.00\, q_1 = \underline{1,000 \text{ psi.}}$$

Circumferential Tension:

$$\sigma_C = \frac{R}{t} q_1 = \frac{22.000}{1.375} q_1 = 16.00\, q_1 = \underline{2,000 \text{ psi.}}$$

$\sigma_{max.} = 8,500$ psi.

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Analysis of Working Stresses in shell wall due to tube sheet effect:

when pressure in shell equal to zero!

Bending Moment: $M_S = 3.028\,(10)^6 \phi_B = (3.028)(4.848)\,q_1 =$

$= 14.69\,q_1$

Shearing Force: $Q_S = 0.843\,(10)^6 \phi_B = (0.843)(4.848)\,q_1 =$

$= 4.09\,q_1$

Therefore: Bending Stress:

$$\sigma_M = \frac{6 M_S}{t^2} = \frac{6(14.69)}{(1.375)^2}\,q_1 = 46.62\,q_1 = \underline{5{,}830 \text{ psi.}}$$

Shearing Stress:

$$\sigma_Q = \frac{Q_S}{t} = \frac{4.09}{1.375}\,q_1 = 2.975\,q_1 = \underline{372 \text{ psi.}}$$

Similar Intermediate Heat Exchanger with Hemispherical Barrel and Sine Wave Tubes.

Change in Assumption: now internal pressure at the tube side: $p=0$
at the shell side: $p=q_1=100$ psi.

Shell Wall Effect:

Inside Radius of Shell ___ = 24.500"
Mean Radius of Shell ____ = 25.1875"
Thickness of Shell Wall ___ = 1.375"
Tube Sheet acting arm ___ = 1.750"
Internal Pressure _____ = q_1

Assume only radial movement of shell wall edge!

$$\text{hoop stress} = \frac{R}{t} q_1 = \frac{24.500}{1.375} q_1 = 17.820\, q_1$$

$$\varepsilon_R = \frac{17.820 q_1}{E}\left(1-\frac{\mu}{2}\right) = \frac{(17.820)(0.85)}{21}(10)^{-6} q_1 = 0.7215\,(10)^{-6} q_1$$

$\therefore$ the radial displacement as shell pressure effect:

$$\Delta R = (25.1875)(0.7215)(10)^{-6} q_1 = 18.170\,(10)^{-6} q_1$$

$\therefore$ the total radial displacement: $y_o = 1.750\,\phi_B + 18.170\,(10)^{-6} q_1$

Hetényi: $M_S = \frac{k}{2\beta^2}\left(y_o + \frac{\phi_B}{\beta}\right)$, where: $k = 45{,}550$, $\beta = 0.2182$ } see page ③

$$Q_S = \frac{k}{2\beta}\left(2y_o + \frac{\phi_B}{\beta}\right)$$

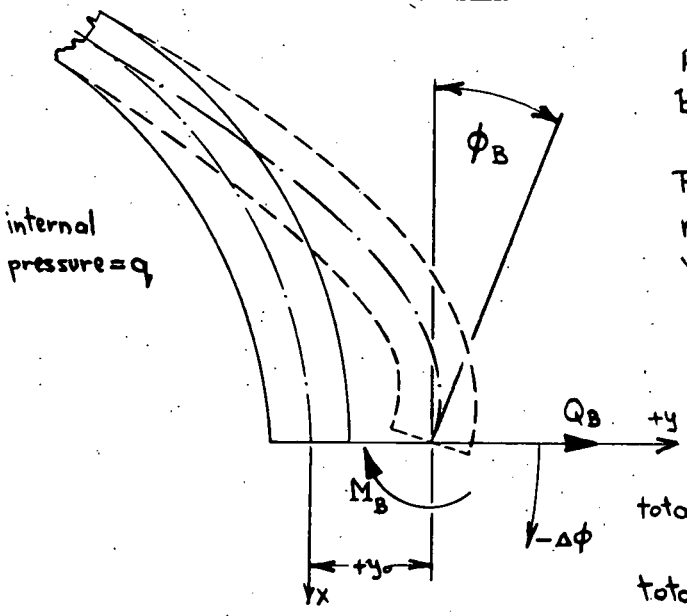
$$M_S = \frac{45{,}550}{0.09524}\left(1.750\,\phi_B + 18.170\,(10)^{-6} q_1 + \frac{1}{0.2182}\phi_B\right) = \underline{3.028\,(10)^6\,\phi_B + 8.692\, q_1}$$

$$Q_S = \frac{45{,}550}{0.4364}\left(3.500\,\phi_B + 36.340\,(10)^{-6} q_1 + \frac{1}{0.2182}\phi_B\right) = \underline{0.843\,(10)^6\,\phi_B + 3.792\, q_1}$$

$$T_S = \frac{q_1 R}{2t} t = \frac{R}{2} q_1 = \frac{24.500}{2} q_1 = \underline{12.250\, q_1}$$

7-F34

Effect of Hemispherical Barrel Wall:



Assume only radial displacement of barrel wall edge! $\Delta x = 0$

Radial displacement and angular movement of wall at edge are positive when outward !

Similar equations as page ③

total radial displacement $= y_o = +1.750\,\phi_B$

total angular movement $= -\phi_B$

Sign of M_B is negative because it causes a negative $\Delta\phi$
Sign of Q_B is positive because it causes a positive Δy

$\beta = 5.220$
$R = 22.6875"$
$t = 1.375"$
$E = 21\,(10)^6$

} see page ② and ③

$$(1)\quad +y_o = \frac{2Q_B}{Et}(\beta R) - \frac{M_B}{Et}(2\beta^2) = \frac{2\beta R}{Et}Q_B - \frac{2\beta^2}{Et}M_B$$

$$(2)\quad -\phi_B = \frac{Q_B}{Et}(2\beta^2) - \frac{M_B}{Et}\left(\frac{4\beta^3}{R}\right) = \frac{2\beta^2}{Et}Q_B - \frac{4\beta^3}{EtR}M_B$$

$$\text{from (2)}: \quad \frac{2\beta^2}{Et}Q_B = -\phi_B + \frac{4\beta^3}{EtR}M_B$$

$$\therefore\ Q_B = \frac{Et}{2\beta^2}\left(-\phi_B + \frac{4\beta^3}{EtR}M_B\right) = -\frac{Et}{2\beta^2}\phi_B + \frac{2\beta}{R}M_B$$

Substituting to (1):

$$y_o = \frac{2\beta R}{Et}\left(-\frac{Et}{2\beta^2}\phi_B + \frac{2\beta}{R}M_B\right) - \frac{2\beta^2}{Et}M_B$$

$$y_o = -\frac{R}{\beta}\phi_B + \frac{4\beta^2}{Et}M_B - \frac{2\beta^2}{Et}M_B = -\frac{R}{\beta}\phi_B + \frac{2\beta^2}{Et}M_B$$

Therefore: $M_B = \frac{Et}{2\beta^2}\left(\frac{R}{\beta}\phi_B + y_o\right)$

$$M_B = \frac{(21)(10)^6\,(1.375)}{(2)(5.220)^2}\left[\frac{22.6875}{5.220}\phi_B + 1.750\,\phi_B\right] =$$

$$= 0.5295\,(10)^6\,(6.094\,\phi_B)$$

$$\underline{\underline{M_B = 3.230\,(10)^6\,\phi_B}}$$

(1) $1.750\,\phi_B = \frac{2\beta R}{Et}Q_B - \frac{2\beta^2}{Et}\left(3.230\,(10)^6\,\phi_B\right)$

$$\frac{2\beta R}{Et}Q_B = 1.750\,\phi_B + \frac{2\beta^2}{Et}3.230(10)^6\,\phi_B =$$

$$= 1.750\,\phi_B + \frac{(2)(5.220)^2}{(21)(1.375)}(10)^{-6}\,3.230\,(10)^6\,\phi_B =$$

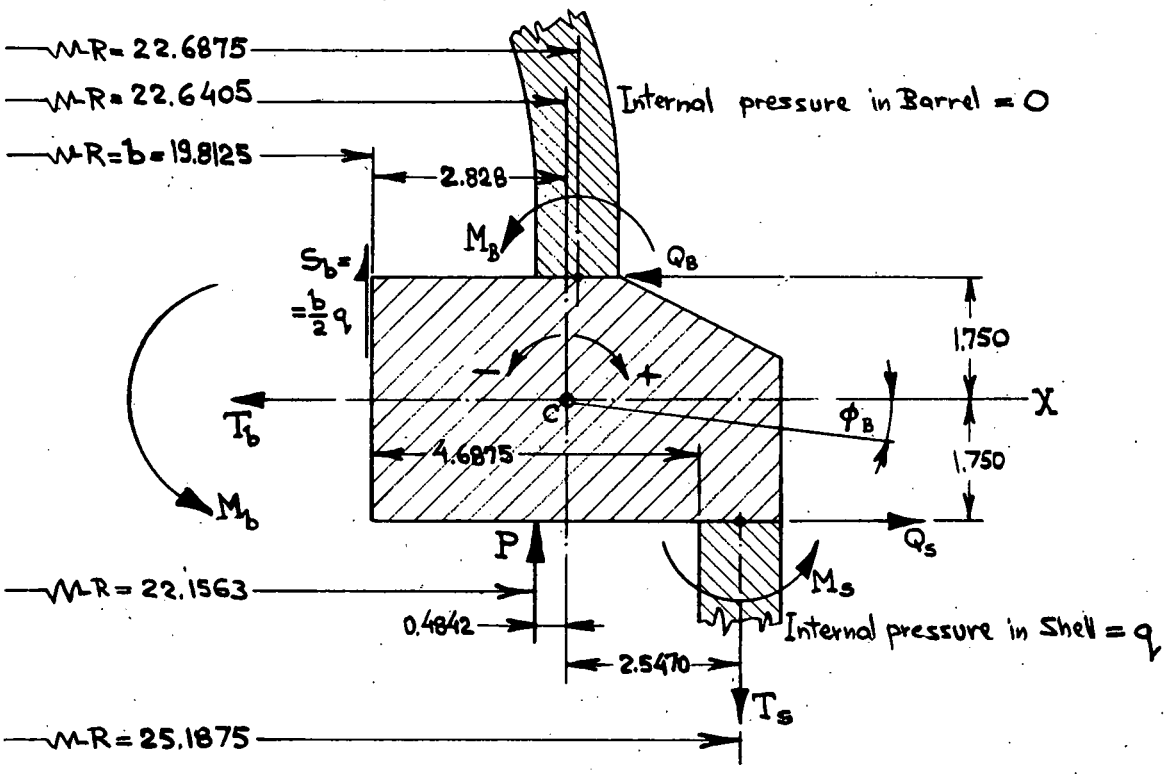
$$= 1.750\,\phi_B + 6.095\,\phi_B = 7.845\,\phi_B$$

$$Q_B = \frac{Et}{2\beta R}7.845\,\phi_B = \frac{(21)(10)^6(1.375)}{2(5.220)(22.6875)}7.845\,\phi_B$$

$$\underline{\underline{Q_B = 0.956\,(10)^6\,\phi_B}}$$

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Forces and Moments acting at outer solid ring:



External twisting Moment "M_t" per unit length at center "C":

due to $M_b = -M_b \frac{19.8125}{22.6405} = -0.8748\ M_b$

" " $S_b = +9.9063\, q\, (2.828) \frac{19.8125}{22.6405} = +24.500\, q$

" " $P = +4.6875 q\, (0.4842) \frac{22.1563}{22.6405} = +2.221\, q$

" " $M_B = -3.230\ (10)^6 \phi_B \frac{22.6875}{22.6405} = -3.236\ (10)^6\ \phi_B$

" " $Q_B = -0.956\ (10)^6 \phi_B\ (1.750) \frac{22.6875}{22.6405} = -1.675\ (10)^6\ \phi_B$

" " $M_s = -\left[3.028\ (10)^6 \phi_B + 8.692\, q\right] \frac{25.1875}{22.6405} = -3.367\ (10)^6\ \phi_B - 9.662\, q$

" " $Q_s = -\left[0.843\ (10)^6 \phi_B + 3.792\, q\right] (1.750) \frac{25.1875}{22.6405} = -1.640\ (10)^6 \phi_B - 7.376\, q$

" " $T_s = +12.250\, q\, (2.5470) \frac{25.1875}{22.6405} = +34.680\, q$

7-F37

After summation : $M_t = -0.8748\, M_b - 9.918\,(10)^6 \phi_B + 44.363\, q$

Inertia Moment of outer solid ring (see page ⑤) : $I_{cx} = 18.558$ in.^{4}

$$\text{Now, } \phi_B = \frac{M_t R^2}{E I_x}$$

$$\therefore \phi_B = \left(-0.8748\, M_b - 9.918\,(10)^6 \phi_B + 44.363 q\right)\frac{(22.6405)^2}{(21)(18.558)}\,(10)^{-6} =$$

$$= \left(-0.8748\, M_b - 9.918\,(10)^6 \phi_B + 44.363 q\right)(1.316)\,(10)^{-6} =$$

$$= -1.151\,(10)^{-6}\, M_b - 13.050\, \phi_B + 58.380\,(10)^{-6} q$$

$$\therefore \phi_B = \frac{-1.151\,(10)^{-6} M_b + 58.380\,(10)^{-6} q}{14.050} = -0.08195\,(10)^{-6} M_b + 4.156\,(10)^{-6} q$$

$$\therefore M_b = -\frac{1}{0.08195}(10)^6 \phi_B + \frac{4.156}{0.08195} q = -12.200\,(10)^6 \phi_B + 50.700\, q$$

$M_{1B} = -12.200\,(10)^6 \phi_B + 50.700\, q$ → to equation No.(7)

Since the all of data are same, using similar equations as at page ⑦ and ⑧, it changes only the equation of (7) :

(10) $\quad -5.669\, q + 29.350\,(10)^6 C_3 + 2.3914\,(10)^6 C_4 = 0$

(6)(7)(3) $\quad -83.39\, q + 24.22\,(10)^6 C_3 - 0.0529\,(10)^6 C_4 =$

$$= +50.700 q - 12.200\,(10)^6 \left[-14.040\,(10)^{-6} q + 9.906\, C_3 + 0.05048\, C_4\right]$$

$$= +50.700\, q - 120.850\,(10)^6 C_3 + 171.250\, q - 0.616\,(10)^6 C_4$$

(11) $\quad 305.340\, q - 145.070\,(10)^6 C_3 - 0.5631\,(10)^6 C_4 = 0$, $\left[k = \frac{145.07}{29.350} = 4.942\right]$

(10)x(k) $\quad -28.03\, q + 145.070\,(10)^6 C_3 + 11.825\,(10)^6 C_4 = 0 \qquad (10)'$

(10)' + (11) $\quad 277.310\, q = -11.2619\,(10)^6 C_4$

$$C_4 = -\frac{277.310}{11.2619}(10)^{-6} q_1 = -24.62\,(10)^{-6} q_1$$

from (10) $\quad 29.350\,(10)^{6} C_3 = 5.669\, q_1 + (2.3914)(24.62) q_1 = (5.669 + 58.900) q_1 = 64.569\, q_1$

$$C_3 = \frac{64.569}{29.350}(10)^{-6} q_1 = 2.199\,(10)^{-6} q_1$$

$$\boxed{C_3 = 2.199\,(10)^{-6} q_1 \quad ; \quad C_4 = -24.62\,(10)^{-6} q_1}$$

Using paper of "Basic Equations" — see page No. (9)

$$\frac{(1+\mu^*)}{2} D^* C_3 = (0.7)(34.60)(2.199) q_1 = 53.25\, q_1$$

$$(1-\mu^*) D^* C_4 = (0.6)(34.60)(-24.62) q_1 = -511.30 q_1$$

Therefore the Moments: $M_{1x} = -0.2125\, x^2 q_1 + 53.25\, q_1 + \frac{511.30}{x^2} q_1$

$$M_{2x} = -0.1375\, x^2 q_1 + 53.25\, q_1 - \frac{511.30}{x^2} q_1$$

Moments at the middle radius of first inside tubes: $x = 7.5625'' = \text{radius} \; ; \; x^2 = 57.20$

$$M_1 = (-12.16 + 53.25 + 8.94) q_1 = +50.03\, q_1$$
$$M_2 = (-7.86 + 53.25 - 8.94) q_1 = +36.45\, q_1$$

Moments at the middle radius of first outside tubes: $X = 19.5625'' = \text{radius} \; ; \; x^2 = 382.7$

$$M_1 = (-81.32 + 53.25 + 1.34) q_1 = -26.73\, q_1$$
$$M_2 = (-52.60 + 53.25 - 1.34) q_1 = -0.69\, q_1$$

Max. ligament stresses without Horvay's factor, when the shell pressure:

$$q_1 = 100 \text{ psi.}$$

$$\sigma_{inside} = \frac{6 M_1}{t^2} = \frac{6(50.03)}{(3.500)^2} q_1 = 24.50\, q_1 = 2450 \text{ psi.}$$

$$\sigma_{outside} = \frac{6 M_1}{t^2} = \frac{6(26.73)}{(3.500)^2} q_1 = 13.10\, q_1 = 1310 \text{ psi.}$$

Ratio of principal Stresses: Inside: $\frac{+36.45}{+50.03} = +0.729$

Outside: $\frac{-0.69}{-26.73} = +0.026$

Horvay's stress multiplier (at $\eta = 0.50$): $Y_{inside} = 2.63$

$Y_{outside} = 3.56$

Therefore the max. stresses: $\sigma_{inside} = (2450)(2.63) = \underline{6440 \text{ psi. (maximum)}}$

$\sigma_{outside} = (1310)(3.56) = \underline{4660 \text{ psi.}}$

At any other portion of perforated part the stresses are smaller!

Controling result of moments:

Bending Moment at $x=b=19.8125$; $x^2=392.4$

$$M_{1B}=(-83.38+53.25+1.30)q_v=\underline{\underline{-28.83\,q_v}}$$

On the other hand the bending moment at $x=b$ is from equation of twisted outer solid ring:

$$M_{1B}=-12.200\,(10)^6\phi_B+50.700\,q_v$$

where: $\phi_B=-14.040\,(10)^{-6}q_v+9.906\,C_3+0.05048\,C_4=$

$$=\left[-14.040+(9.906)(2.199)+(0.05048)(-24.62)\right](10)^{-6}q_v=$$

$$=(-14.040+21.780-1.243)\,(10)^{-6}q_v=$$

$$=6.497\,(10)^{-6}q_v$$

Therefore:

$$M_{1B}=-(12.200)(6.497)\,q_v+50.700\,q_v=(-79.25+50.70)\,q_v=$$

$$=\underline{\underline{-28.55\,q_v}}$$

Difference = 1.0 %

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Analysis of Working Stresses in shell wall due to the internal pressure and the tube sheet effect.

Internal pressure in Shell = 100 psi.

Bending Moment: $M_S = 3.028\,(10)^6\,\phi_B + 8.692\,q_1 =$
$= (3.028)(6.497)\,q_1 + 8.692\,q_1 = (19.68 + 8.692)\,q_1 =$
$= 28.372\,q_1$

Shearing Force: $Q_S = 0.843\,(10)^6\,\phi_B + 3.792\,q_1 =$
$= (0.843)(6.497)\,q_1 + 3.792\,q_1 = (5.480 + 3.792)\,q_1 =$
$= 9.272\,q_1$

Therefore: $M_s = 2837$ in.-lb.
$Q_s = 927$ lb.

Bending stress: $\sigma_M = \frac{6\,M_s}{t^2} = \frac{6\,(2837)}{(1.375)^2} = 9000$ psi.

Shearing stress: $\tau_Q = \frac{Q_s}{t} = \frac{927}{1.375} = 674$ psi.

Longitudinal Tension: $\sigma_L = \frac{R}{2t}\,q = \frac{24.500}{2(1.3750)}\,(100) = 890$ psi.

Circumferential Tension: $\sigma_c = 2\,\sigma_L = 1780$ psi.

Max. stress is given from combination of bending and longitudinal tension:

$$\sigma_{max} = 9890 \text{ psi.}$$

Analysis of Working Stresses in Hemispherical Barrel Wall due to tube sheet effect:

When pressure in barrel equal to zero!

Bending Moment: $M_B = 3.230\ (10)^6\ \phi_B = (3.230)(6.497)\ q_1 = 21.00\ q_1$

$= 2100$ in.-lb.

Shearing Force: $Q_B = 0.956\ (10)^6\ \phi_B = (0.956)(6.497)\ q_1 = 6.21\ q_1$

$= 621$ lb.

Therefore: Bending Stress:

$$\sigma_M = \frac{6\,M_B}{t^2} = \frac{6(2100)}{(1.375)^2} = \underline{6{,}670 \text{ psi.}}$$

Shearing Stress:

$$\tau_Q = \frac{Q_B}{t} = \frac{621}{1.375} = \underline{452 \text{ psi.}}$$

Analysis of Working Stresses due to Nozzle Loads at the top of hemispherical barrel.
(Bijlaard's Theory)

External Loads of Nozzle: Nozzle Thrust : $P = 4{,}000$ lb. (radial)
Nozzle Moment : $M = 480{,}000$ in.-lb.

Mean Radius of barrel wall : $R = 22.6875$"
Thickness of barrel wall : $t = 1.375$"
Acting outer radius of nozzle : $r_o = 9.000$"

Parameter of diagram: $s = 1.82 \frac{r_o}{R}\sqrt{\frac{R}{t}} = 1.82 \frac{9.000}{22.6875}\sqrt{\frac{22.6875}{1.375}} = 2.933$

Fig. A.5-2: $\frac{\sigma t^2}{P} = 0.118$ and $\frac{\sigma t^2}{\frac{M}{R}\sqrt{\frac{R}{t}}} = 0.094$

Therefore: $\sigma = \frac{(0.118)P}{t^2} = \frac{(0.118)(4{,}000)}{(1.375)^2} = 250$ psi.

$$\sigma = \frac{(0.094)M}{Rt^2}\sqrt{\frac{R}{t}} = \frac{(0.094)(480{,}000)}{(22.6875)(1.375)^2}\sqrt{\frac{22.6875}{1.375}} = 4{,}275 \text{ psi.}$$

Pressure Stress: $\sigma_p = \frac{qR}{2t} = \frac{(125)(22.6875)}{2(1.375)} = 1{,}030$ psi.

Combined Stress from Nozzle Thrust and internal pressure:

$$\sigma = 1{,}030 + 250 = \underline{1{,}280 \text{ psi.}}$$

Combined Stress from Nozzle Moment and internal pressure:

$$\sigma = 1{,}030 + 4{,}275 \cong \underline{5{,}300 \text{ psi.}}$$

Intermediate Heat Exchanger with Sine Wave Tubes.

External Nozzle Moment: $M = 480{,}000$ in.-lb.

Internal pressure in Shell side: $p = 100$ psi.
in Tube side: $p = 125$ psi.

Shell: Inside Dia. of Shell = 49"
Thickness of Shell Wall = $1\frac{3}{8}$" = 1.375"
Acting O.Dia. of Nozzle = 18"

$a = 25.1875$
$t = 1.375$
$r_o = 9.000$

$$\beta = \frac{0.815\, r_o}{a} = \frac{(0.875)(9.000)}{25.1875} = 0.3129$$

$$\gamma = \frac{a}{t} = \frac{25.1875}{1.375} = 18.31$$

Circumferential Stress as maximum stress due to external circumferential Nozzle Moment:

Fig. A.5-11: $\frac{M\varphi_2}{M/a\beta} = 0.0752$ Fig. A.5-13: $\frac{N\varphi_2}{M/a^2\beta} = 0.708$

Therefore: $M\varphi_2 = \frac{M\,0.0752}{a\beta} = \frac{(480{,}000)(0.0752)}{(25.1875)(0.3129)} = 4{,}582$

$$N\varphi_2 = \frac{M\,0.708}{a^2\beta} = \frac{(480{,}000)(0.708)}{(25.1875)^2(0.3129)} = 1{,}714$$

$$\sigma_5 = \frac{N\varphi_2}{t} + \frac{6\,M\varphi_2}{t^2} = \frac{1{,}714}{1.375} + \frac{6(4{,}582)}{(1.375)^2} = 1{,}246 + 14{,}550$$

$$\text{Pressure Stress} = \frac{(100)(24.500)}{(1.375)} = 1{,}782$$

Max. Combined Stress: $\sigma_{5\,max.} = 1{,}246 + 14{,}550 + 1{,}782 = 17{,}578 \cong 17{,}600$ psi.

vs 18,000 psi. Allowable at 1200°F

7-F45

Stress due to Nozzle Thrust – P = 4000 lb.

Fig. A.5-4: $\frac{M\phi}{P} = 0.055$ and Fig. A.5-5: $\frac{N\phi}{P/a} = 1.8$

Therefore: $M\phi = P(0.055) = (4000)(0.055) = 220$

$$N\phi = \frac{P(1.8)}{a} = \frac{(4000)(1.8)}{(25.1875)} = 286$$

$$\sigma_1 = \frac{286}{1.375} + \frac{6(220)}{(1.375)^2} = 208 + 698 = 906 \text{ psi.}$$

Factors for the Stresses due to Nozzle Moment – M = 480 000 in.-lb.

$\frac{M_n}{M/a\beta} = x$, therfore $M_n = \frac{Mx}{a\beta}$; $\frac{N_n}{M/a^2\beta} = y$, therefore $N_n = \frac{My}{a^2\beta}$

Bending Stress due to Nozzle Moment: $\sigma = \frac{6M_n}{t^2} = \frac{6Mx}{a\beta t^2} = x\frac{6M}{a\beta t^2} = x F_B$

Membrane Stress due to Nozzle Moment: $\sigma = \frac{N_n}{t} = \frac{My}{a^2\beta t} = y\frac{M}{a^2\beta t} = y F_M$

$$F_B = \frac{6M}{a\beta t^2} = \frac{6(480000)}{(25.1875)(0.3129)(1.375)^2} = 193\,400$$

$$F_M = \frac{M}{a^2\beta t} = \frac{480000}{(25.1875)^2(0.3129)(1.375)} = 1760$$

Longitudinal Stress due to Longitudinal Nozzle Moment:

Fig. A.5-6: $\frac{M_x}{M/a\beta} = 0.028$ and Fig. A.5-8: $\frac{N_{x1}}{M/a^2\beta} = 0.58$

$$\sigma_2 = (1760)(0.58) + (193400)(0.028) = 1000 + 5420 = 6420 \text{ psi.}$$

Circumferential Stress due to Longitudinal Nozzle Moment:

Fig. A.5-7: $\frac{M\varphi_1}{M/a\beta} = 0.019$ and Fig. A.5-9: $\frac{N\varphi_1}{M/a^2\beta} = 1.26$

$$\sigma_3 = (1760)(1.26) + (193400)(0.019) = 2220 + 3680 = 5900 \text{ psi.}$$

997 180

Longitudinal Stress due to Circumferential Nozzle Moment:

Fig. A.5—10: $\frac{M_{x2}}{M/a\beta} = 0.034$ and Fig. A.5-12: $\frac{N_{x2}}{M/a^2\beta} = 1.40$

$\sigma_4 = (1760)(1.40) + (193400)(0.034) = 2460 + 6580 = 9040$ psi.

Summary:—

Shell stresses:		Nozzle Bending psi.	Nozzle Membrane psi.	Pressure Tension psi.	Total psi.
Nozzle Thrust of 4000 lb.	σ_1	698	208	1780	2686
Nozzle Moment of 480 000 in.-lb.					
Acting Longitudinally:					
Longitudinal Stress	σ_2	5420	1000	890	7310
Circumferential Stress	σ_3	3680	2220	1780	7680
Acting Circumferentially:					
Longitudinal Stress	σ_4	6580	2460	890	9930
Circumferential Stress	σ_5	14550	1250	1780	17580

Heat Exchanger Sodium Nozzles

The upper nozzle, entering the shell, normally operates at a temperature of 1200° F, but may be subjected under casualty conditions to a drop in temperature of 490° within 14 seconds. The lower nozzle normally operates at 700° F but during an isothermal hot may be subject to a rapid increase in sodium temperature of 490° in 14 seconds. Resulting stresses in the two cases will be very nearly alike.

The accompanying charts show the temperature transients in a stainless steel plate of indefinite thickness under these conditions. They are computed as for a sodium film transfer coefficient of 2700 BTU/Hr.ft.², but due to the method of computation, more nearly represent a coefficient of 3200 BTU/Hr.ft².

By proper handling they can also be made to apply to plates of finite thickness, and also to other metals than stainless steel.

Maximum stress in pipe wall 3/8" thick, — after 14 seconds:

Surface — 0.000"	726°	−7	= 719	×0.5	359
0.075"	836°	−11	= 825	×1	825
0.150"	920°	−19	= 901	×1	901
0.225"	986°	−30	= 956	×1	956
0.300"	1035°	−47	= 988	×1	988
0.375"	1068°	−72	= 996	×0.5	498
					4528

(60° should be added to these temperatures) $\frac{4528}{5} = 905$

Mean temperature = 905°

Difference between surface and mean temperature = 719 − 905 = −186°

Tensile stress in surface = $\frac{(T_m - T_o) E \alpha}{0.7} = \frac{(186)(260)}{0.7}$ = 69 000 psi. in the upper nozzle during an isothermal cold. In this nozzle, no stress would result from an isothermal hot.

In the lower nozzle during an isothermal hot, the surface temperature would have risen rather than dropped. Difference between surface and mean temperature again will equal 186° F.

007 182

The surface temperature would be 1114° F.

Compressive stress in surface $= \frac{(186)(245)}{0.7} = 65000$ psi.

In both cases Stress due to pressure $= \frac{(100)(6.625)}{0.375} = 1765$ psi.

Nozzle reaction – 4000 lb. plus 40 000 ft.-lb.
Assuming this to act in full on the pipe;

Area of metal $= \pi\left(7.000^2 - 6.625^2\right) = \pi(5.12) = 16.09$ in.^{2}

Moment of inertia $= \frac{\pi}{4}\left(7.000^4 - 6.625^4\right) = \frac{\pi}{4}(475) = 373$ in.^{4}

Stress in pipe due to nozzle reactions $= \frac{4000}{16.09} + \frac{(480{,}000)(7.000)}{373} =$

$= 250 + 9010 = 9260$ psi.

Under the most unfavorable circumstances, the nozzle reaction stresses in the pipe wall, acting longitudinaly may range from 9260 psi. tension to 9260 psi. compression. The thermal transient stress, occuring a maximum of 25 times may range from zero to 69 000 psi. tension in the upper nozzle and from zero to 65000 psi. compression in the lower nozzle, acting both longitudinally and circumferentially. The pressure stress is 1765 psi. tension acting circumferentially and half that much or 882 psi. acting longitudinally.

For the upper nozzle, the equivalent steady state stress is 882 psi. and the stress alternations are from – 9260 to plus 78260 psi. Temperature at the maximum stress point is 769° F, corresponding to a value for S_b of 24000 psi. and for S_a III of 66 000 psi. $S_{alt.} = (9260 + 78260)(0.5) = 43760$ psi.

Since in this case $S_a > S_b$ and exceeds $S_{alt.}$, the condition is safe! See Navy Code Par. A 1.8.5.

For the lower nozzle, the temperature at the point of maximum stress = 1114° F. $S_b = 19700$ psi. and S_a III $= 53200$ psi.; $S_{alt.} = (9260 + 74260)(0.5) = 41760$ psi.

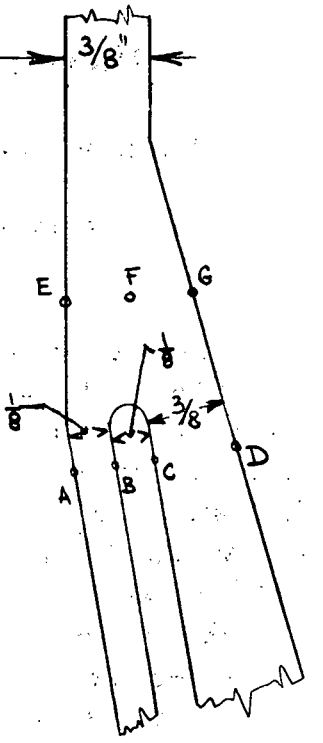
Again S_a III $> S_b$ and S_a III $> S_{alt.}$, and the condition is safe.

The secondary sodium nozzles should be similarly protected.

Stresses at weld of shield to nozzle wall following a rapid change in sodium temperature of 490°

h for sodium assumed 2700 k for SS 316 = 11×10^{-6}

Temperatures given are based on a drop of 490° F from 1140° F, but with proper values of E, the stresses also apply to a drop from 1200° or to a rise from 700° F. The greatest stresses, occuring at the end of this pdrop in temperature, are shown below:



After 14 second drop @ 35°/sec. (see Graphs)

				Mean:	Difference:
0 '	Temperature at	A	726°	809°	-83°
.125'	—"—	B	892°		+83°
.156"	—"—	C	928°	991°	-63°
.531"	—"—	D	1054°		+63°

Treat .125" thickness of sodium, B–C, as equivalent to 1/4 its thickness of stainless

E	0.0"	726°	x ½	= 363
	0.15"	920°	x 1	= 920
F	0.30"	1035°	x 1	= 1035
	0.45"	1086°	x 1	= 1086
G	0.60"	1100°	x ½	= 550
			4	3954 / 4 = 989

Mean temperature = 989°

Differential – wall at EG to shield at AB = 989 – 809 = 180°

$\Delta R = 180 \times 11 \times 10^{6} \times 6.687 = 0.01324$

Stress in shield due to temperature differential across shield:

$$\sigma = \frac{E \alpha \Delta T}{(0.7)} = \frac{(267)(83)}{(0.7)} = 31{,}700 \text{ psi.}$$

Bending moment at weld:

$$M = 2EI\beta^2 y_o = 0.2904 \frac{Et^2}{R} y_o = \frac{(0.2904)(24.6)(10)^6(0.125)^2}{(6.687)}(0.01324) =$$

$$= (16{,}700)(0.01324) = 221 \text{ in.-lb.}$$

$$\text{Bending Stress} = \frac{(221)6}{(0.125)^2} = 84{,}900 \text{ psi.}$$

This acts in the opposite direction to the stress due to the temperature differential.

Primary sodium inlet nozzle or secondary sodium outlet nozzle:

Combined stress on inside face:

Longitudinal : $84{,}900 - 31{,}700 = 53{,}200$ psi. tension

Circumferential : $(84{,}900)(0.3) + 31{,}700 = 57{,}200$ psi. tension

On the face next the nozzle wall, these stresses are reversed and become compressive.

Surface thermal stress at E:

$$\frac{E \alpha \Delta T}{(0.7)} = \frac{(267)(989 - 726)}{(0.7)} = \frac{(267)(263)}{(0.7)} = 100{,}000 \text{ psi. tension}$$

Surface thermal stress at C, behind the shield:

$$\frac{E \alpha \Delta T}{(0.7)} = \frac{(267)(991 - 928)}{(0.7)} = \frac{(267)(63)}{(0.7)} = 24{,}000 \text{ psi. tension}$$

For the primary sodium outlet nozzle during an isothermal hot, the temperature starting at 700° rises to 1190° F.

E changes from from $24.6(10)^6$ to $20.6(10)^6$,

and therefore $E\alpha$ from 267 to 234

Making these changes:
Stress in shield due to temperature differential across shield =

$$= \frac{(234)(83)}{(0.7)} = 27{,}700 \text{ psi.}$$

Bending stress in shield near weld =

$$= 84{,}900 \frac{20.6}{24.6} = 71{,}100 \text{ psi.}$$

Combined stresses on inner face:

Longitudinal : $71{,}100 - 27{,}700 = 43{,}400$ psi. compression

Circumferential : $(71{,}100)(0.3) + 27{,}700 = 49{,}000$ psi. compression

Surface thermal stress at E = $\frac{(234)(263)}{(0.7)} = 87{,}900$ psi. compression

Surface thermal stress at C = $\frac{(234)(63)}{(0.7)} = 21{,}000$ psi. compression

These values also apply to the secondary sodium inlet nozzle.

Other Stresses: Internal pressure of 100 psi. :

Pipe is 0.375" thick — $\frac{(100)(6.625)}{(0.375)} = 1765$ psi.

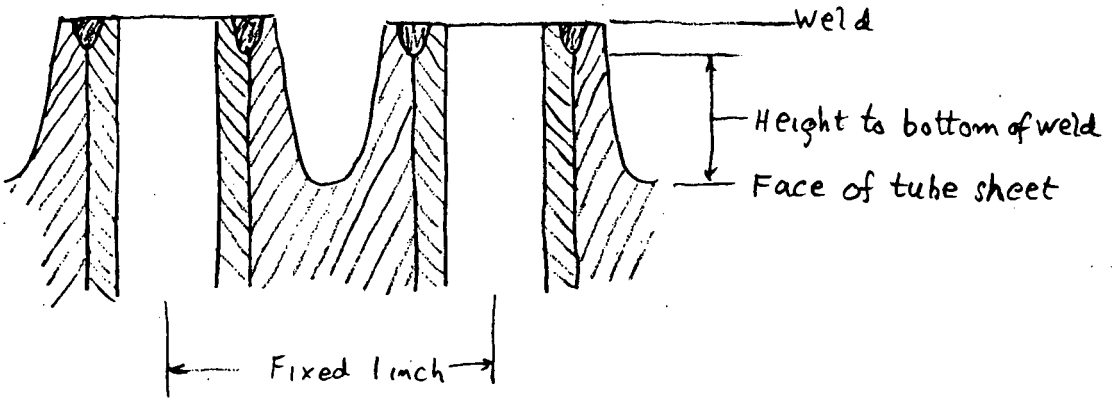
Pipe junction to shield — $\frac{(100)(6.625)}{(0.60)} = 1105$ psi.

Shield — none.

Thermal stresses during start up and shut down, at say 200° F/hour : — For thickness of 1" of stainless these amount to 2000 psi. — For smaller thicknesses they are negligible.

Thermal stress due to rapid change of 50° in sodium temperature during normal operation. — Stresses vary essentially directly with the total temperature drop, and would be $50/490 = 10\%$ of those just given.

Nipples machined on top of tube sheet for welding tubes at a position above the surface of the tube sheet.



Temperature of face of tube sheet, nipples, and tubes 740°F
Temperature of most of tube sheet 1140°F
Mean coef. of thermal expansion = 11.05×10^{-6} IN/IN °F
Separation of tube wall and nipple at face of tube sheet due to 400 °F temperature difference between them and most of the tube sheet equals

$$400 \times 11.05 \times 10^{-6} \times .5 = .00221 \text{ inches}$$

This distance must be equal to the combined deflection of the nipple and tube, Calculations for these deflections may be made by treating the nipple and tube as beams on elastic foundations

Beams on Elastic Foundation

Differential equations (Signs depend on sign convention used)

$$EI \frac{d^2y}{dx^2} = M$$

$$\frac{dM}{dx} = V$$

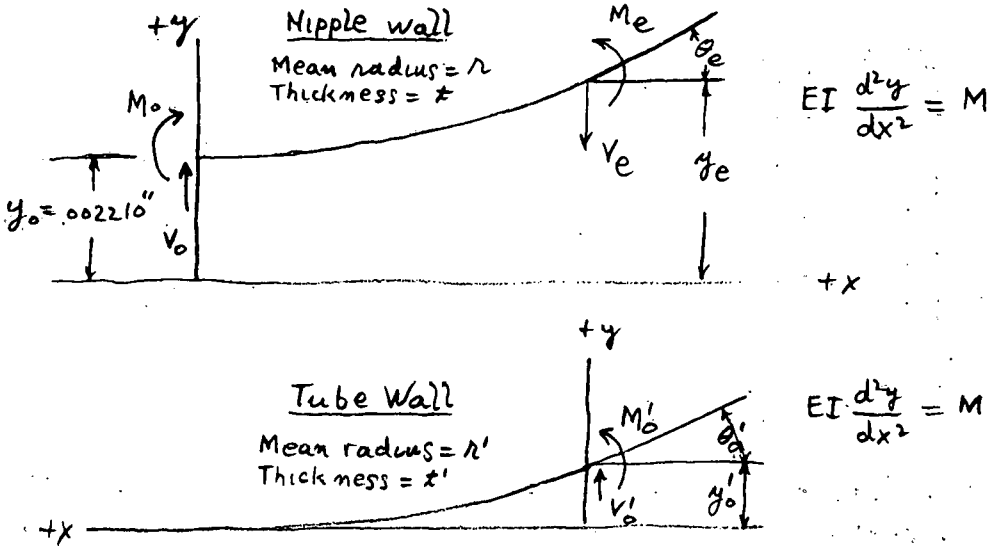
$$\frac{dV}{dx} = -ky - w$$

These are the same as the standard beam formulas except that ky in $\frac{dV}{dx}$ represents the reaction per unit length due to the elastic foundation.

In the solution for a tube, a unit wide longitudinal element of the tube is considered a beam on an elastic foundation with $k = \frac{Et}{r^2}$ where r and t are mean radius and thickness of wall and $I = \frac{t^3}{12(1-\nu^2)}$ where ν is Poisson's ratio.

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Sign convention used in the solution for the deformation of the tube sheet nipples and tubes



At the weld between the nipple and tube

$$\left.\begin{aligned} y_e &= y'_o \\ \theta_e &= \theta'_o \end{aligned}\right\} \text{ condition of continuity}$$

$$\left.\begin{aligned} r V_e &= r' V'_o \\ r M_e &= -r' M'_o \end{aligned}\right\} \text{ condition of equilibrium}$$

For the tube

$$V'_o = 2EI\beta^2(2\beta y'_o + \theta'_o) \qquad \text{where } \beta = \frac{\sqrt[4]{3(1-\nu^2)}}{\sqrt{r't'}} = \frac{1.285}{\sqrt{r't'}}$$

$$M'_o = -2EI\beta^2\left(y'_o + \frac{\theta'_o}{\beta}\right)$$

Therefore,

$$V_e - 4EI\beta^3\frac{r'}{r}y_e - 2EI\beta^2\frac{r'}{r}\theta_e = 0$$

$$M_e - 2EI\beta^2\frac{r'}{r}y_e - 2EI\beta\frac{r'}{r}\theta_e = 0$$

These are sufficient to determine V_e and M_e. They may be arranged differently as follows: Multiply the second by β and subtract from the first.

$$V_e - 2EI\beta^3\frac{r'}{r}y_e - \beta M_e = 0$$

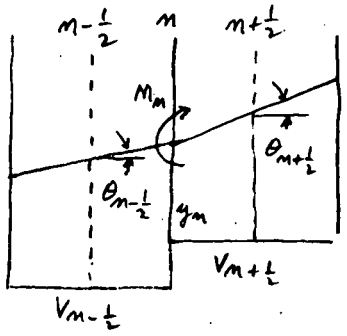
Subtract this from the first and divide by $-2EI\beta^2\frac{r'}{r}$

$$\theta_e + \beta y_e - \frac{1}{2EI\beta\frac{r'}{r}}M_e = 0$$

The last two equations are also sufficient to determine V_e and M_e

The tube sheet nipple may be tapered to obtain reduced stresses. In this case the equivalent beam will have variable fexural rigidity (EI) and variable foundation modulus (k).

The calculation may be made by a step-by-step tabular method devised by Prof. Sayre. Divide the length of the beam into a number of equal increments Δx and denote the section at $x = n\Delta x$ by the subscript n.



Let y_n and $\theta_{n-\frac{1}{2}}$ be the deflection and moment at $x = n\Delta x$ and $V_{n-\frac{1}{2}}$ and $\theta_{n-\frac{1}{2}}$ be the shear and slope at $x = (n-\frac{1}{2})\Delta x$

The differential equations in finite difference form are

$$\left.\begin{aligned} \frac{\Delta\theta}{\Delta x} &= \frac{M}{EI} \\ \frac{\Delta M}{\Delta x} &= V \\ \frac{\Delta V}{\Delta x} &= -ky - w \\ \frac{\Delta y}{\Delta x} &= \theta \end{aligned}\right\} \text{ or } \quad \begin{aligned} \Delta\theta &= \frac{M}{EI}\Delta x \\ \Delta M &= V\Delta x \\ \Delta V &= -(ky+w)\Delta x \\ \Delta y &= \theta\Delta x \end{aligned}$$

THerefore

$$V_{n+\frac{1}{2}} = V_{n-\frac{1}{2}} - (ky_n + w_n)\Delta x$$

$$M_{n+1} = M_n + V_{n+\frac{1}{2}}\Delta x$$

$$\theta_{n+\frac{1}{2}} = \theta_{n-\frac{1}{2}} + \frac{M_n}{EI_n}\Delta x$$

$$y_{n+1} = y_n + \theta_{n+\frac{1}{2}}\Delta x$$

These formulas may be used to increase all subscripts by unity so that by repeated use y_e, M_e, $V_{e+\frac{1}{2}}$, $\theta_{e+\frac{1}{2}}$ may be obtained.

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At the beginning of the beam, two of the four terms y_o, M_o, $V_{-\frac{1}{2}}$, $\theta_{-\frac{1}{2}}$ are determined by the type of restraint at this end. The other two are left implicit to be determined by the type of restraint at the end of the beam.

SOLUTION OF COMBINATION OF NIPPLE AND TUBE

Since the step-by-step solution for nipple gives y_e, M_e, $V_{e+\frac{1}{2}}$, $\theta_{e+\frac{1}{2}}$ at its end, the following equalities will represent the conditions at the joint.

$$y_e = y'_o$$

$$\theta_{e+\frac{1}{2}} = \theta'_{\frac{1}{2}}$$

$$\Lambda V_{e+\frac{1}{2}} = \Lambda' V'_{\frac{1}{2}}$$

$$\Lambda M_e = -\Lambda' M'_o$$

where the prime refers to the tube with its origin at the joint.

From the basic equations for a beam on an elastic foundation

$$\theta'_{at\,x} = -\beta e^{-\beta x}\left[C(\sin\beta x + \cos\beta x) - D(\cos\beta x - \sin\beta x)\right]$$

$$= -\beta\left[C A_x - D C_x\right] \qquad \text{(For } A_x \text{ and } C_x \text{ refer to Hetenyi tables)}$$

$$V'_{at\,x} = -2EI\beta^3 e^{-\beta x}\left[C(\sin\beta x - \cos\beta x) - D(\sin\beta x + \cos\beta x)\right]$$

$$= 2EI\beta^3\left[C C_x + D A_x\right]$$

where $C = y'_o$, $D = -\dfrac{M'_o}{2EI\beta^2}$

Therefore, two simultaneous equations to solve for the two unknowns $\theta_{e+\frac{1}{2}}$ and $V_{e+\frac{1}{2}}$ at the end of the nipple may be written as

$$\theta_{e+\frac{1}{2}} + \beta A_x y_e - \frac{C_x}{2EI\beta\frac{\Lambda'}{\Lambda}} M_e = 0$$

$$V_{e+\frac{1}{2}} - 2EI\beta^3\frac{\Lambda'}{\Lambda} C_x y_e - \beta A_x M_e = 0$$

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NIPPLE .375 × $\frac{1}{16}$ R FOR $\frac{1}{2}$" TUBE

PHYSICAL CONSTANTS

x	t	r	t^3	r^2	$EI^* = \frac{Et^3}{10.92}$ $= 2,216,000\,t^3$	$K^* = \frac{Et}{r^2}$
0	.2590	.3665	.017,373,98	.134 322	38 500.74	46662,500
.04	.2130	.3495	.009,663,60	.122 150	21414.54	42198,900
.08	.1745	.3335	.005,313,57	.111 222	11774.87	37968,200
.12	.1415	.3195	.002,833,15	.102 080	6278.26	33,545,300
.16	.1150	.3080	.001,520,88	.094,864	3370.27	29,336,700
.20	.0925	.2985	.000,791,453	.089102	1753.86	25,122,900
.24	.0750	.2900	.000,421,875	.084,100	934.88	21,581,500
.28	.0615	.2840	.000232,608	.080656	515,46	18,452,400
.32	.0540	.2800	.000,157,464	.078,400	348.94	16668,400
.36	.0490	.2785	.000,117,649	.077 562	260.71	15,288,400
.40	.0490	.2785				
.44	.0490	.2785				
.48	.0490	.2785				
.52	.0490	.2785				
.56	.0490	.2785				

* E For mean T of 750 °F is taken as 24.2×10^6

X	$\frac{\Delta x}{EI} \times 10^6$ ($\Delta x = .04$)	$K\Delta x \times 10^{-6}$ ($\Delta x = .04$)	t^2	$\frac{6}{t^2}$		
0	1.03894	1.86650	.067,081	89.444		
.04	1.86789	1.68796	.045,369	132.249		
.08	3.39707	1.51873	.030,450,2	197.043		
.12	6.37119	1.34181	.020,022,2	299.67		
.16	11.8685	1.17347	.013,225,0	452.69		
.20	22.8068	1.00492	.008,556,2	701.25		
.24	42.7862	.86326	.005,625,0	1066.67		
.28	77.6006	.73810	.003,7822	1586.38		
.32	114.633	.66674	.002916 =	2057.6		
.36	153.427	.61154	.002,401,6	2499.		
.40	"	"		"		
.44	"	"		"		
.48	"	"		"		
.52	"	"		"		
.56	"	"		"		

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NIPPLE .375 × $\frac{1}{16}$R FOR $\frac{1}{2}$ TUBE

First page of step-by-step solution

n	$y \times 10^6$	$-K\Delta x \times 10^{-6}$	$\Delta V = -K\Delta x\, y$	V $(V+\Delta V)$	$\Delta M = V\Delta x = .04V$
0	1000	-1.86650	-1866.50	V_0	
$\frac{1}{2}$	✓		$V_{\frac{1}{2}} = V_0 + \frac{3}{8}\Delta V_0 + \frac{1}{8}\Delta V_1 \rightarrow$	-910,932,5 + V_0 -.004,384,222,906 M_0	-36.4373 +.04V_0 -.000,175,368,916,2 M_0
1	1000 +020,778,8 M_0	-1.68796	-1687.96 -.035,073,783,25 M_0	✓ -2598.8925 + V_0 -.039,458,006,16 M_0	-103.9557 +.04V_0 -.001,578,320,246 M_0
2	997.277,565,3 +.002,988,624 V_0 +.116,260,097,2 M_0	-1.51873	-1514.595,357 -004538,912,928 V_0 -.176,567,697.4 M_0	-4113.487.857 +.995461.087.1 V_0 -.216,025,703,6 M_0	-164.539,514,3 +.039.818.443.48 V_0 -.008,641,028,144 M_0
3	975.478,136,6 +.016,847,872 V_0 +.347,385,898,2 M_0	-1.34181	-1308.906,310 -.022,606,643,13 V_0 -.466,125.872,1 M_0	-5422.394,175 +.972.854,440 V_0 -.682,151,575.7 M_0	-216.895767.0 +.038,914,177,76 V_0 -.027,286 063,03 M_0
4	875.967.388.5 +.061,242,562,76 V_0 +.830,710,230,6 M_0	-1.17347	-1027.921.451 -.071,866,310,12 V_0 -.974.813.534.3 M_0	-6450,315626 +.900,988,133,9 V_0 -1.656,965,110 M_0	-258.012,625.0 +.036039,525,36 V_0 -.066,278,604,40 M_0
5	528.723,882,1 +.180,993,978,2 V_0 +1.770,885,989 M_0	-1.00492	-531.325.203,6 -.181 824 468,6 V_0 -1.779,598.748 M_0	-6981.640,830 +.719,103,665,3 V_0 -3.436,563,858 M_0	-279.265633.2 +.028,764,146,61 V_0 -.137,462 554,3 M_0
6	-529.946,647,9 +.478,430,569,3 V_0 +3.528,494,512 M_0	-.86326	457.481,743,3 -.413,009,973,3 V_0 -3.046,008,172 M_0	-6524.159,087 +.306,093,692,0 V_0 -6.482,572,030 M_0	-260.966363.5 +.012,243,747,68 V_0 -.259,302,881,2 M_0
7	-3401.222,947 +1.158,437,902 V_0 +6.584,369,940 M_0	-.73810	2510,442,657 -.855,043,015,5 V_0 -4.859,923,453 M_0	-4013.716,430 -.548,949,323.5 V_0 -11.342,495,48 M_0	-160.548,657,2 -.021,957,972,94 V_0 -.453,699,819.2 M_0
8	-10,370.037,22 +2.570.312 140 V_0 +11.190,009,51 M_0	-.66674	6914.118.616 -1.713,729.916 V_0 -7,460.825,607 M_0	-2900.402,196 -2,262,679,240 V_0 -18.803,321,09 M_0	116.016,0878 -.090.507,169,60 V_0 -.752,132,843,6 M_0
9	-24,127.975,15 +4.962,628,983 V_0 +16.004,622,69 M_0	-.61154	14755.22192 -3.034,846,128 V_0 -9.787,466,960 M_0	17,655.62410 -5.297,525,368 V_0 -28.590,788,05 M_0	706.224,9640 -.211,901.014.7 -1.143,631,522
10	-46260.60604 +8.111,739,222 V_0 +16.483,038.07 M_0				
11	-72433.770 78 +10.717.189,38 V_0 +5.606,695,51 M_0				

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NIPPLE .375 × 1/16 R FOR 1/2 TUBE

First page continued

n	$y \times 10^6$	$-K\Delta x/10^{-6}$	$\Delta V = -K\Delta x\, y$	V $(V+\Delta V)$	$\Delta M = V\Delta x = .04V$
9	-24,127.97515 + 4,962,628,983 Vo +16,004,622,69 Mo	-,61154	14755.22192 -3,034,846,128 Vo -9,787,466,960 Mo	-17655.62410 -5,297,525,368 Vo -28,580,788,05 Mo	706,224,964.0 -.211,901,014,7 -1,143,631,522
10	-46260.60604 + 8,111,739,222 Vo +16,483,038,07 Mo	-,61154	28290.21102 - 4,960,653,604 Vo - 10,080,037,10 Mo	45945.83512 -10258,17837 Vo -38,670,825,15 Mo	1837,833,405 -.410,327,134,8 Vo -1,546,833,006 Mo
11	-72433.77078 +10,717,189,38 Vo + 5,606,695,51 Mo	-,61154	44296.14818 -6,553,989,993 Vo -3,428,718,572 Mo	90241.98330 -16,812,168,36 Vo -42,099,543,72 Mo	3609,679,332 -.672,486,734,4 Vo -1,683,981,749 Mo
12	-91368.53874 10,260,769,00 Vo -26,117,442,89 Mo	-,61154	55875.51618 - 6,274,870,674 Vo 15,971,861,02 Mo	146117.4995 -23,087,039,03 Vo -26,127,682,70 Mo	5844,699,980 -.923,481,561,2 Vo -1,045,107,308 Mo
13	-80912.01909 2,615,373,192 Vo -89,024,107,85 Mo	-,61154	49480.93615 -1,599,405,322 Vo 54,441,802,91 Mo	195598.4356 -24,686,444,36 Vo 28,314,120,21 Mo	7823,937,424 -,987,457,777,4 Vo +1,132,564,808 Mo
14	-5194.82049 -17,886,478,26 Vo -189,527,206,6 Mo	-,61154	3176,840,522 +10,938,296,92 Vo +115,903,467,9 Mo	198775.276,1 -13,748,147,44 Vo +144,217,588,1 Mo	

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NIPPLE .375 x 1/16 R FOR 1/2 TUBE

Second page of step-by-step solution

n	M (M+ΔM)	$\frac{\Delta x}{EI} \times 10^6$	$\Delta\theta \times 10^6 = \frac{M\Delta x}{EI} \times 10^6$	$\theta \times 10^6$ $(\theta + \Delta\theta) \times 10^6$	$\Delta y \times 10^6 = \theta\Delta x \times 10^6$ $= .04\,\theta \times 10^6$
0	M_o	1.03894	$1.03894\,M_o$	0	
1/2				$.519,470\,M_o$	$.020,778,8\,M_o$
1	$-36,437,3 + .04V_o$ $+.999,824,631\,1M_o$	1.86789	$-68,060,868,30$ $+.074,715,6\,V_o$ $+1,867,562,430M_o$	0 $-68,060,868,30$ $+.074,715,6\,V_o$ $+2,387,032,430\,M_o$	$-2,722,434,732$ $+.002,988,624\,V_o$ $+.095,481,297,20\,M_o$
2	$-140,393 + .08\,V_o$ $+.998,246,310,9\,M_o$	3.39707	$-476.924,848,5$ $+.271,765,6\,V_o$ $+3,391,112,595\,M_o$	$-544.985,716,8$ $+.346,481,2\,V_o$ $+5.778,145,025\,M_o$	$-21.799,428,67$ $+.013,859,248\,V_o$ $+.231,125,801,0\,M_o$
3	$-304,932,514,3$ $+.119,818,443,5\,V_o$ $+.989,605,282,8\,M_o$	6.37119	-1942.782986 $+.763,386,069,0\,V_o$ $+6,304,963,282\,M_o$	-2487.768703 $+1.109,867,269\,V_o$ $+12.083.108,31\,M_o$	$-99,510,748,12$ $+.044,394,690,76\,V_o$ $+.483,324332,4\,M_o$
4	$-521,828,281.3$ $+.158,732,621,2\,V_o$ $+.962.319,219,7\,M_o$	11.8685	$-6193.318.957$ $+1.883,918,115\,V_o$ $+11.421,28566\,M_o$	$-8681.087,660$ $+2.993,785,384\,V_o$ $+23.504,393,97\,M_o$	$-347,243,506,4$ $+.119,751,415,4\,V_o$ $+.940,175,758,8\,M_o$
5	$-779,840,906,3$ $+.194772,146,6\,V_o$ $+.896,040,615,3\,M_o$	22.8068	$-17785.675,58$ $+4.442,129,393\,V_o$ $+20,435,819,11\,M_o$	$-26,466.763,24$ $+7.435,914.777\,V_o$ $+43.940,213,08\,M_o$	$-1058.670,530$ $+.297,436,591,1\,V_o$ $+1.757,608,523\,M_o$
6	$-1059,106,540$ $+.223,536,293,2\,V_o$ $+.758,578,061,0\,M_o$	42.7862	$-45,315.14424$ $+9.564,268,548\,V_o$ $+32,456.672,63\,M_o$	$-71,781.907,48$ $+17,060,183.32\,V_o$ $+76.396,885,71\,M_o$	$-2871,276,299$ $+.680,007,332,8\,V_o$ $+3,055,875,428\,M_o$
7	$-1320,072,904$ $+.235,780,040,9\,V_o$ $+.499,275,179,8\,M_o$	77.6006	$-102,438.449,4$ $+18.296,672,64\,V_o$ $+38.744,053,52\,M_o$	$-174,220.356,9$ $+35.296,855,96\,V_o$ $+115,140,939,2\,M_o$	$-6968,814,276$ $+1.411,874,238\,V_o$ $+4.605,637,568\,M_o$
8	$-1480,621,561$ $+.213,822,067,9\,V_o$ $+.045,575,360,6\,M_o$	114.633	$-169,728.091,4$ $+24.511,065,11\,V_o$ $+5.224,44031\,M_o$	$-343,948.4483$ $+59.807,921,07\,V_o$ $+120.365,379,5\,M_o$	$-13757.937,93$ $+2.392,316,843\,V_o$ $+4.814,615,180\,M_o$
9	$-1364,605,473$ $+.123,314,898,3\,V_o$ $-.706,557,483,0\,M_o$	153.427	$-209,367.3239$ $+18.919,83490\,V_o$ $-108.404,994,9\,M_o$	$-553,315.772,2$ $+78.727,75597\,V_o$ $+11.960,384,6\,M_o$	$-22132,630,89$ $+3,149,110,239\,V_o$ $+.478,415,384\,M_o$
10	$-658.380,509,2$ $-.088,586,116,4\,V_o$ $-1.850,189,005\,M_o$	153.427	$-101,013.346,4$ $-13.591,502,08\,V_o$ $-283.868,948,5\,M_o$	$-654,329.1186$ $+65.136,253,89\,V_o$ $-271.908,563,9\,M_o$	$-26173,16474$ $+2.605,450,156\,V_o$ $-10.876,342,56\,M_o$
11					

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NIPPLE .375 × 1/16 R FOR 1/2 TUBE

Second page continued

n	M (M+ΔM)	$\frac{\Delta x}{EI} \times 10^6$	$\Delta\theta \times 10^6 = \frac{M\Delta x}{EI} \times 10^6$	$\theta \times 10^6$ $(\theta + \Delta\theta) \times 10^6$	$\Delta y \times 10^6 = \theta \Delta x 10^6$ $= .04\theta \times 10^6$
9					
	-658.380,509,2		-101,013,346,4		
10	-.088,586,116,4 Vo	153,427	-13.591,502,08 Vo		
	-1.850,189.005 Mo		-283.868,948,5 Mo	-654,329.1186	-26173,16474
				+65.136,253 89 Vo	+2.605.450,156 Vo
	1179.452 896		180,959.919 5	-271.908,563 9 Mo	-10,876,342,56 Mo
11	-.498913,251,2 Vo	153,427	-76.546,763.39 Vo		
	-3.397,022,011 Mo		-521.194,896,1 Mo	-473 369,1991	-18934.76796
				- 11.410,509,50 Vo	-.456,420,380.0 Vo
	4789.132 228		734,782.190,3	- 793.103,460,0 Mo	-31.724,138,40 Mo
12	-1.171,399,986 Vo	153,427	-179.724 385,7 Vo		
	-5.081,003,760 Mo		-779.563,163.9 Mo	261,412,9912	10456.519 65
				-191.134,895,2 Vo	- 7.645,395,808 Vo
	10633. 832,21		1,631,516.974	-1572.666,624 Mo	-62,906,664,96 Mo
13	- 2.094,881.547 Vo	153,427	-321.411,391,1 Vo		
	-6.126,111,068 Mo		-939.910,842,8 Mo	1,892,929.965	75717.19860
				- 512.546,286,3 Vo	- 20,501,851.45 Vo
	18 457,769.63		2831,920.221	-2512.577,468 Mo	-100.503,098,7 Mo
14	- 3.082 339.322 Vo	153,427	-472.914,075,2 Vo		
	- 4.993,546,260 Mo		-766.144,822.0 Mo	4,724 850.186	
				- 985.460,361,5 Vo	
				- 3278.722,289 Mo	

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NIPPLE .375 x $\frac{1}{16}$ R FOR $\frac{1}{2}$ D TUBE

Tube $\frac{1}{2}$ D $\quad t' = .049$

$r' = .25 - .0245 = .2255$

$$\theta_{e+\frac{1}{2}} \times 10^6 + \beta A_x y_e \times 10^6 - \frac{10^6 C_x}{2EI\beta \frac{\Lambda'}{\Lambda}} M_e = 0$$

$$V_{e+\frac{1}{2}} - \frac{2EI\beta^3 \frac{\Lambda'}{\Lambda} C_x}{10^6} y_e \times 10^6 - \beta A_x M_e = 0$$

$$\beta = \frac{\sqrt[4]{3(1-\nu^2)}}{\sqrt{r' t'}} = \frac{1.285}{\sqrt{.049 \times .2255}} = 13.509$$

$$I = \frac{t'^3}{12(1-\nu^2)} = \frac{(.049)^3}{10.92} = 10.774 \times 10^{-6}$$

$E = 24.2 \times 10^6$ for $T = 740°F$

$$2EI\beta \frac{\Lambda'}{\Lambda} = 2 \times 24.2 \times 10.774 \times 13.509 \frac{.2255}{.2785} = 5704.$$

$$2EI\beta^3 \frac{\Lambda'}{\Lambda} = 5704 (13.509)^2 = 1,040,901$$

$\beta x = 13.509 \times .04 = .54036$

$A_x = e^{-\beta x}(\cos \beta x + \sin \beta x) = .799,2242$

$C_x = e^{-\beta x}(\cos \beta x - \sin \beta x = .199,8566$

$\beta A_x = 13.509 \times .799,2242 = 10,796,720$

$$\frac{10^6 C_x}{2EI\beta \frac{\Lambda'}{\Lambda}} = \frac{10^6 \times .199,8566}{5704} = 35,037,973$$

$$\frac{2EI\beta^3 \frac{\Lambda'}{\Lambda} C_x}{10^6} = 1,040,901 \times .199,8566 = .208,030,9$$

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NIPPLE – CURVE .375 × 1/16 R FOR 1/2 D TUBE

Solution with 1/2 D Tube welded to end of nipple

$\beta A_x = 10.796720$, $\frac{10^6 C_x}{2EI\beta\frac{\Lambda'}{\Lambda}} = 35,037.973$, $\frac{2EI\beta^3\frac{\Lambda'}{\Lambda}C_x}{10^6} = .208.030,9$

$\theta_{e+\frac{1}{2}}\cdot 10^6 = 4724.850.186 - 985.460,361,5V_o - 3278.722,289\,M_o$

$\beta A_x y_e 10^6 = -\ 56,087.022 - 193.115,297,6V_o - 2046.272.182\,M_o$

$-\frac{10^6 C_x}{2EI\beta\frac{\Lambda'}{\Lambda}} M_e = -\ 646,722.834 + 107.998,921,9V_o + 174.963,739\,M_o$

$V_{e+\frac{1}{2}} = 198,775.2761 - 13.748,147,44V_o + 144,217,588,1\,M_o$

$-\frac{2EI\beta^3\frac{\Lambda'}{\Lambda}C_x}{10^6} y_e 10^6 = 1080.6832 + 3.720,940\,17V_o + 39.427,515,4\,M_o$

$-\beta A_x M_e = -199,283.3705 + 33.279,154,60V_o + 53.913,920,8\,M_o$

$4,022,040.330 - 1070.576,737\,V_o - 5150.030,732\,M_o = 0$

$572.5888 + 23.251,947,33V_o + 237,559,024,3\,M_o = 0$

$185\,527.4320 - 49.383\,232,51V_o - 237.559,024,3\,M_o = 0$

$186,100.0208 - 26.131,285,18V_o = 0$

$V_o = 7121.732,418$

$572.5888 + 165,594.1471 + 237.559,024,3\,M_o = 0$

$M_o = -699.475,5782$

Nipple independent of tube , $M_e = 0$, $V_{e+\frac{1}{2}} = 0$

$198,775.2761 - 13.748,147,44V_o + 144,217,588,1\,M_o = 0$

$18,457.76963 - 3.082,339,322V_o - 4.993,546,260M_o = 0$

$6,882.61085 - 4,476,030,775V_o + 4.993,546,259\,M_o = 0$

$25,340.38048 - 3,558,370,097\,V_o = 0$

$V_o = 7121.344,826$

$18,457.76963 - 21,950.401,18 - 4.993,546,260M_o = 0$

$M_o = -699.429,097$

NIPPLE .375 x 1/16 R FOR 1/2 TUBE

Me = 0 , Ve+1/2 = 0

V_o = 7.121,344,826 , M_o = -.699,429,097

n				$y \times 10^6$	650 IBM
0	1.	—	—	1	1.
1	1.	—	-.014,533,297,3	.985,466,702,7	.985,466,70
2	.997,277,5653	.021,283,022,1	-.081,315,694,8	.937,244,892,6	.937.244,89
3	.975,478,1366	.119979,506,1	-.242,971,805,1	.852,485,837,6	.852,485,84
4	.875,967.3885	.436,129,407,4	-.581,022,906,5	.731,073,889,4	.731,073,91
5	.528,723,882,1	1,288,920,530	-1.238,609,188	.579,035,224	.579,035,29
6	-.529,946,647,9	3.407,069,059	-2,467,931,730	.409,190,681	.409,190,84
7	-3,401,222,947	8,249,635760	-4,605,299,921	.243,112,892	.243,113,24
8	-10,370,037.22	18,304079,06	-7.826,616,848	.107,424.99	.107.42570
9	-24.127975,15	35,340,592,23	-11.194,098,80	.018,518,28	.018,519,540
10	-46,260,606,04	57.766,492.14	-11.528.716,43	-.022830.33	-.022,828,480
11	-72.433,770,78	76.320,801,14	-3.921,485,98	-.034,455,62	-.034,453,513
12	-91.368,538,74	73 070,474,23	+18,267,299.50	-.030,765,01	-.030.763,637
13	-80,912,019.09	18.624,974,35	62,266,051,36	-.020.993,38	-.020,994,672
14	- 5,194820.49	-127.375,779,4	132.560.843,0	-.009756,9	-.009,764,1152

n				M	650 IBM
0	—	—	-.699,429.097	-.699429.097	-.699,429.08
1	-.036,4373	.284,853,793,0	-.699,306,439	-.450,889,946	-.450,889.92
2	-.140,393	.569,707,586,1	-.698,202,5158	-.268,887,929.7	-.268,887,90
3	-.304,932,5143	.853,268,452,7	-.692,158,729,3	-.143,822,7909	-.143,822,75
4	-.521,828,281,3	1.130,389,731	-.673,074,062,9	-.064,512,613	-.064,512.566
5	-.779,840,9063	1,387,039,618	-.626,716,881,2	-.019,518,170	-.019,518,144
6	-1.059,166,580	1.591,879,025	-.530,571,568,2	.002,200,917	.002,200,972,4
7	-1.320,072,904	1,679,070,974	-.349,207,588,2	.009,790,482	.009,790,535,6
8	-1.480,621,561	1.522,700,677	-.031,876,733,3	.010,202,383	.010,202,423
9	-1.364,605,473	.878,167,913	.494,186,862	.007,749,302	.007,749,310,5
10	-.658,380,509,2	-.630,852,281.7	1.294,076,025	.004,843,234	.004,843,180.5
11	1.179,452,896	-3.552,933,300	2,375,976,038	.002,495,634	.002,495,471.6
12	4.789,132,228	-8,341.943,229	3.553,801,872	.000,990,871	.000,990,550,76
13	10,633,832,21	-14.918,373,87	4,284,780,332	.000,238.67	.000,238,157,68
14	18,457,769,63	-21.950,401,18	3.492,631,551	.000,000,00	.000,000,672,16

NIPPLE .375 x 1/16 R FOR 1/2 D TUBE

Me = 0 , $V_{e+\frac{1}{2}} = 0$

$V_0 = 7,121,344,826$, $M_0 = -.699,429,097$

n				V	650 IBM
0	——	7.121,344,826	——	7.121,344826	7.121,345,7
1/2	-.910,932.5	7.121,344826	.003,066,453	6.213,478,779	6.213,479.7
1 1/2	-2,598,892.5	7.121,344,826	.027,598,078	4.550,050,704	4,550,050,5
2 1/2	-4.113,487.857	7.089,021,662	.151,094,663	3,126,628,468	3,126,628,6
3 1/2	-5,422,394,175	6.928.031,961	.477,116,661	1,982,756,872	1.986,754,6
4 1/2	-6.450,315,626	6,416,247,186	1.158 929,611	1,124,861,171	1,124,861,3
5 1/2	-6.981,640,830	5.120,985,166	2,403,632756	.542,977,092	.542,977,16
6 1/2	-6.524,159.087	2,179.798,730	4,534,099,501	.189,739,144	.189,739.08
7 1/2	-4,013,716,430	-3,909.257,425	7,933,271,371	.010,297,516	.010,297,200
8 1/2	2.900,402,196	-16.113,319,10	13.151,589,89	-.061,327,01	-.061,327,811
9 1/2	17,655,624,10	-37.725,504,87	19.997.229,07	-.072,651,70	-.072,653,250
10 1/2	45.945,835,12	-73,052,025,46	27.047,500,31	-.058,690,03	-.058,692,722
11 1/2	90.241 983,30	-119.725,248,2	29.445,645.85	-.037,619,0	-.037.623.021
12 1/2	146,117,499.5	-164410,765,9	18.274,461,52	-.018,804,9	-.018,809.827
13 1/2	195.598,435,6	-175.800,682,8	-19,803,719,95	-.005,967,2	-.005,970,7460
14 1/2	198.775,276,1	-97.905,298,64	-100.869,977,4	.000,000,0	.000000,401

n				$\theta \times 10^6$	650 IBM
0	0	0	0	0	0
1/2	——	——	-.363 332,4330	-.363 332.4330	-.363,332,42
1 1/2	-.068,060,86830	.532,075.551,5	-1.669,559,937	-1.205.545.254	-1.205,545,2
2 1/2	-.544,985.716,8	2,467,412,101	-4,041,402,757	-2.118,976,373	-2.118,976,2
3 1/2	-2.487,768,703	7.903,747,534	-8.451,277534	-3.035,298,703	-3.035 298,2
4 1/2	-8.681,087,660	21.319,778,05	-16439,657,05	-3,800,966,66	-3.800,965,5
5 1/2	-26.466,763,24	52.953,713,22	-30.733,063,56	-4.235,113,58	-4,246,111,2
6 1/2	-71.781,907,48	121.064,167,5	-53.434,204,79	-4,151,944,8	-4,151,939,9
7 1/2	-174.220356,9	251.361,082,6	-80.532,923,13	-3.392,197,4	-3.392,1884
8 1/2	-343,948,448,3	425.912,829.3	-84.187,048,69	-2.222,667,7	-2.222,6540
9 1/2	-553,315,772,2	560,647,497,6	- 8.365,441 00	-1.033 715,6	-1.033,700,5
10 1/2	-654,329,118,6	463,857,724,6	190.180,7613	-.290,632,7	-.290,625.84
11 1/2	-473.369,199,1	-81.258,172,79	554,719,636,9	.092,265,0	.092,246,881
12 1/2	261.412,991,2	-1361.137,497	1099.968,797	.244291	.244,224,11
13 1/2	1892,929,965	-3650.018,844	1757.369,790	.280,911	.280,763,92
14 1/2	4724.850,186	-7017.803,047	2293.233770	.280 909	.280,660,79

NIPPLE .375 × 1/16 R. FOR 1/2 D TUBE

650 IBM Solutions #1001

PHYSICAL CONSTANT ON PAGE 22

	μ × 10^6		M	
η	SPECIAL START	STANDARD START	SPECIAL START	STANDARD START
0	1	1	-.699 429 08	-.743,998,01
1	.985,466,70	1	-.450,889,92	-.484,646,08
2	.937,244,89	.963,789,37	-.268,887,90	-.292,812,55
3	.852,485,84	.887,790,56	-.143,822,75	-.159,528,45
4	.731,073,91	.771,136,31	-.064,512,566	-.073,894,206
5	.579,035,29	.619,401,53	-.019.518,144	-.024,456,178
6	.409,190,84	.445,356,06	.002,200,972,4	.000 083,890,8
7	.243,113,24	.271,454,17	.009,790,535,6	.009,245,636,8
8	.107,425,70	.126,250,96	.010,202,423	.010,392,970
9	.018,519,540	.028,702,848	.007 749 310,5	.008,173,240,6
10	-.022,828,480	-.018,685,432	.004,843,180,5	.005,251,393,6
11	-.034,453,513	-.033,845,489	.002,495,471,6	.002,786,622,2
12	-.030,763,637	-.031,903,822	.000,990,550,76	.001,149,765,6
13	-.020,994,672	-.022,905,952	.000,238,157,68	.000,293,327,52
14	-.009.764,115,2	-.012,107,908	.000,000 672,16	-.000,002,79436

	V		Θ × 10^6	
η	SPECIAL START	STANDARD START	SPECIAL START	STANDARD START
0	7.121,345,7		0	
1/2	6.213,479,7	6.483,7981	-.363,332,42	0
1 1/2	4.550,050,5	4.795,838,1	-1.205,545,2	-.905,265,56
2 1/2	3.126,628,6	3.332,102,3	-2,118,976,2	-1.899,970,2
3 1/2	1.986,754.6	2,140,856,1	-3.035 298,2	-2.916,356,2
4 1/2	1.124,861,3	1.235,950,7	-3.800,965,5	-3.793,369,5
5 1/2	.542,977,16	.613,501,72	-4.246,111,2	-4.351,136,6
6 1/2	.189,739,08	.229,043,65	-4,151,939,9	-4.347,547,2
7 1/2	.010,297,200	.028,683,330	-3.392,188,4	-3.630,080,2
8 1/2	-.061,327,811	-.055,493,235	-2,222,6540	-2.438,702,8
9 1/2	-.072,653,250	-.073,046,174	-1.033,700 5	-1.184,707,0
10 1/2	-.058,692,722	-.061,619,285	-.290,625,84	.379,001,43
11 1/2	-.037,623,021	-.040,921.415	.092,246,881	.048,541,654
12 1/2	-.018,809,827	-.021,410,952	.244,224,11	.224,946.74
13 1/2	-.005,970,746,0	-.007,403,047,0	.280,763 92	.269,951,10
14 1/2	.000,000,401	.000,001,423,0	.280,660,79	.269,522,36

CDC 260

NIPPLE .300 × 1/16 R FOR 1/2" TUBE — IBM-650 - PROBLEM 1002

PHYSICAL CONSTANTS

Subscript for A & B	x	t	r	t^3	r^2	$EI = \frac{Et^3}{10.92}$ * $= 2216.000\ t^3$	$K = \frac{EI}{r^2}$ *
1	0	.259	.3665	.017,373,98	.134 322	38,500.74	46,662,500
2	.04	.207	.346	.008,869,74	.119,716	19,655.34	41,844,000
3	.08	.165	.329	.004,492,12	.108,241	9,954.54	36,889,900
4	.12	.129	.314	.002,146,69	.098,596	4757.07	31,662,500
5	.16	.101	.303	.001,030,30	.091,809	2283.14	26,622,700
6	.20	.080	.293	.000,512,00	.085,849	1134.59	22,551,200
7	.24	.065	.286	.000,274,625	.081,796	608.57	19,230,800
8	.28	.054	.281	.000,157,464	.078,961	348.940	16,549,900
9	.30	.050	.279	.000,125,000	.077,841	277.000	15,544,500
10	.36	.049	.2785	.000,117,649	.077,562	260.710	15,288,400
11	.40	.049	.2875				
12	.44	.049	.2875				
13	.48	.049	.2875				
14	.52	.049	.2875				
15	.56	.049	.2785				

* E For mean T of 750°F is taken as 24.2×10^6

Subscript for A & B	x	A $\frac{\Delta x}{EI} \times 10^6$ ($\Delta x = .04$)	B $K \Delta x \times 10^{-6}$ ($\Delta x = .04$)	t^2	$\frac{6}{t^2}$		
1	0	1.03894	1.866,50	.067,081	89,444		
2	.04	2.03507	1.67376	.042,849	140,027		
3	.08	4.01827	1.47560	.027,225	220.39		
4	.12	8.40855	1.26650	.016,641	360.56		
5	.16	17.5197	1.06491	.010,201	588,18		
6	.20	35.2550	.90205	.006,400	937.50		
7	.24	65.7279	.76923	.004225	1420.12		
8	.28	114.633	.662 00	.002,916	2057.6		
9	.32	144,404	.62178	.002,500	2400,0		
10	.36	153.427	.61154	.002,401	2499.		
11	.40	"	"				
12	.44	"	"				
13	.48	"	"				
14	.52	"	"				
15	.60	"	"				

7-F67

NIPPLE .240 × 1/16 R FOR 1/2" TUBE - IBM 650 - PROBLEM 1003

PHYSICAL CONSTANTS

Subscript for A & B	X	t	r	t^3	r^2	$EI = \frac{Et^3}{10,92} = 2,216,000\,t^3$ *	$K = \frac{Et}{r^2}$ *
1	0	.247	.356	.015,069,22	.126736	33,393.39	47,164,200
2	.04	.193	.336	.007,189,06	.112896	15,930.96	41,370,800
3	.08	.147	.321	.003,176,52	.103,041	7,039.17	34,524,100
4	.12	.113	.307	.001,442,90	.094,249	3,197.47	29,014,600
5	.16	.086	.295	.000,636,06	.087,025	1,409.51	23,915,000
6	.20	.067	.287	.000,300,76	.082,369	666.48	19,684,600
7	.24	.054	.281	.000,157,464	.078,961	348.940	16,549,900
8	.28	.049	.2785	.000,117,649	.077,562	260,710	15,288,400
9	.30	.049	.2785				
10	.36	.049	.2785				
11	.40	.049	.2785				
12	.44	.049	.2785				
13	.48	.049	.2785				
14	.52	.049	.2785				
15	.56	.049	.2785				

* E For mean T of 750°F is taken as 24.2×10^6

Subscript for A & B	X	A $\frac{\Delta x}{EI} \times 10^6$ ($\Delta x = .04$)	B $K\Delta x \times 10^6$ ($\Delta x = .04$)	t^2	$\frac{6}{t^2}$
1	0	1.19784	1.886.57	.061,009	98.346
2	.04	2.51083	1.65483	.037,249	161.078
3	.08	5.68249	1.38096	.021 609	277.66
4	.12	12.5099	1.16058	.012,769	469.89
5	.16	28.3787	.95660	.007,396	811.25
6	.20	60,0168	.78738	.004,489	1336.60
7	.24	114.633	.66200	.002,916	2057.61
8	.28	153.427	.61154	.002,401	2499.
9	.32	"	"		
10	.36	"	"		
11	.40	"	"		
12	.44	"	"		
13	.48.	"	"		
14	.52	"	"		
15	60	"	"		

NIPPLE .197 x 0 R FOR $\frac{1}{2}$" TUBE - IBM 650 - PROBLEM 1004

PHYSICAL CONSTANTS

subscript for A & B	X	t	r	t^3	r^2	$EI^* = \frac{Et^3}{10,92} = 2,216,000\,t^3$	$k^* = \frac{Et}{r^2}$
1	0	.236	.349	.013,144,26	.121,801	29,127.68	46,889,600
2	.04	.176	.329	.005,451,78	.108,241	12,081.14	39,349,200
3	.08	.130	.313	.002,197,00	.097,969	4868.55	32,112,200
4	.12	.097	.299	.000,912,67	.089,401	2022,48	26,257,000
5	.16	.073	.289	.000,389,02	.083,521	862.07	21,151,600
6	.20	.057	.282	.000,185,193	.079,524	410.39	17,345,70
7	.24	.050	.279	.000,125,000	.077,841	277,000	15,544,500
8	.28	.049	.2785	.000,117649	.077,562	260.710	15,288,400
9	.32	.049	.2785				
10	.36	.049	.2785				
11	.40	.049	.2785				
12	.44	.049	.2785				
13	.48	.049	.2785				
14	.52	.049	.2785				
15	.56	.049	.2785				

* E For mean T of 750° F is taken as 24.2×10^6

Subscript for A & B	X	A $\frac{\Delta x}{EI} \times 10^6$ ($\Delta x = .04$)	B $k\Delta x \times 10^{-6}$ ($\Delta x = .04$)	t^2	$\frac{6}{t^2}$		
1	0	1.373,26	1.87558	.055,696	107.728		
2	.04	3.31095	1.57397	.030,976	193.698		
3	.08	8.21600	1.28449	.016,900	355.03		
4	.12	19.7727	1.05028	.009,409	637.69		
5	.16	46.4000	.84606	.005,329	1125.91		
6	.20	97.4683	.69383	.003,249	1846.72		
7	.24	144.404	.62178	.002,500	2400.		
8	.28	153.427	.61154	.002,401	2499.		
9	.32						
10	.36						
11	.40						
12	.44						
13	.48						
14	.52						
15	.56						

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650 IBM Solutions

$y \times 10^{-6}$

n	1001	1002	1003	1004
0	1	1	1	1
1	1	1	1	1
2	.963 789 37	.965,065,21	.964 084,08	.961,020,53
3	.887,790,56	.889 503,79	.882,063 06	.869,494,19
4	.771,136,31	.769,099,10	.749,341,20	.718,762,69
5	.619,401,53	.607 380,81	.570,359,17	.519,193,93
6	.445,35606	.421,208,80	.370,115,65	.311,388,23
7	.271.454,17	.241,231,89	.193, 147,33	.147,934,41
8	.126,250,96	.102,949.47	.072,688,402	.044,163.690
9	.028702,848	.019 205 026	.005,100,470,0	-.009,572,346,0
10	-.018,685.432	-.020, 166 466	-.024,167,047	-.029,552,090
11	-.033.845,489	-.031.620.915	-.030,430,427	-.030,616,905
12	-.031.903.822	-.028,586,783	-.025,377,924	-.023,171,711
13	-.022.905952	-.019,745,512	-.016,129,491	-.013,025,131
14	-.012 107.908	-.009,487,0160	-.005,995,275,4	-.002,507,187 8

M

n	1001	1002	1003	1004
0	-.743.998,01	-.672,507,55	-.578,574.72	-.491 711,61
1	-.484,646,08	-.429,159.56	-.357,610,00	-.294,322,35
2	-.292,812,55	-.252,761,97	-.202,838,48	-.159,891,89
3	-.159,528.45	-.133,326,39	-.101,321.42	-.074 838,286
4	-.073,894,206	-.058,953,070	-.040,752,552	-.026,313,178
5	-.024.456.178	-.017,340,606	-.008,856,475,6	-.002,112,724,4
6	.000,083,890,8	.002,356,344.0	.005,076,024 8	.007,678,436,4
7	.009,245,6368	.009,093,036.4	.009,207 862,8	.009.724,998,4
8	.010,392.970	.009,441,908,4	.008,615,008,4	.008,152 848,0
9	.008,173,240,6	.007,230,303,5	.006,244,078,4	.005,500,383,1
10	.005,251,393,6	.004,548,913,0	.003,748,38447	.003,082,073,1
11	.002,786,622,2	.002,360,826,5	.001,843,8546	.001,386,654,5
12	.001,149,765,6	.000,946,238,18	.000,683,701,48	.000,440,174,46
13	.000,293,327,52	.000,230,928,30	.000,144,332,96	.000,060,511,540
14	-.000,007,794,36	.000,001,37478	-.000,000,482,404	-.000,000,535,836

650 IBM Solutions

V				
η	1001	1002	1003	1004
$\frac{1}{2}$	6.483,798 1	6.083,699,7	5.524,117,9	4.934,731,3
$1\frac{1}{2}$	4.795,838,1	4.409,939,7	3.869,287,9	3.360,761,3
$2\frac{1}{2}$	3,332,102,3	2.985.889,5	2.537,926,4	2.126,340,1
$3\frac{1}{2}$	2.140,856,1	1.859,333.0	1.514,221,7	1.213,127,7
$4\frac{1}{2}$	1.235,950,7	1.040,311,6	.797.401,91	.605,011,34
$5\frac{1}{2}$	.613,501,72	.492,423,75	.348.312,51	.244,779,02
$6\frac{1}{2}$	.229,043,65	.168,417,31	.103,295,95	.051,164,050
$7\frac{1}{2}$	.028.683,330	.008,721,800,0	-.014,821,360	-.039,303,759
$8\frac{1}{2}$	-.055,493,235	-.055,290,121	-.059,273,225	-.066,311,621
$9\frac{1}{2}$	-.073,046,174	-.067.034,762	-.062,392,366	-.060,457,748
$10\frac{1}{2}$	-.061,619,285	-.054,702,162	-.047,613,251	-.042,385,463
$11\frac{1}{2}$	-.041,921,415	-.035,364,708	-.029,003,828	-.023,662,001
$12\frac{1}{2}$	-.021,410,952	-.017,882,747	-.013,484,213	-.009,491,573,0
$13\frac{1}{2}$	-.007,403,047,0	-.005,807,577,0	-.003,620,384,1	-.001,526,184,4
$14\frac{1}{2}$	.000,001,423,0	-.000,005,887,3	.000,045,966,6	.000 007,061,2

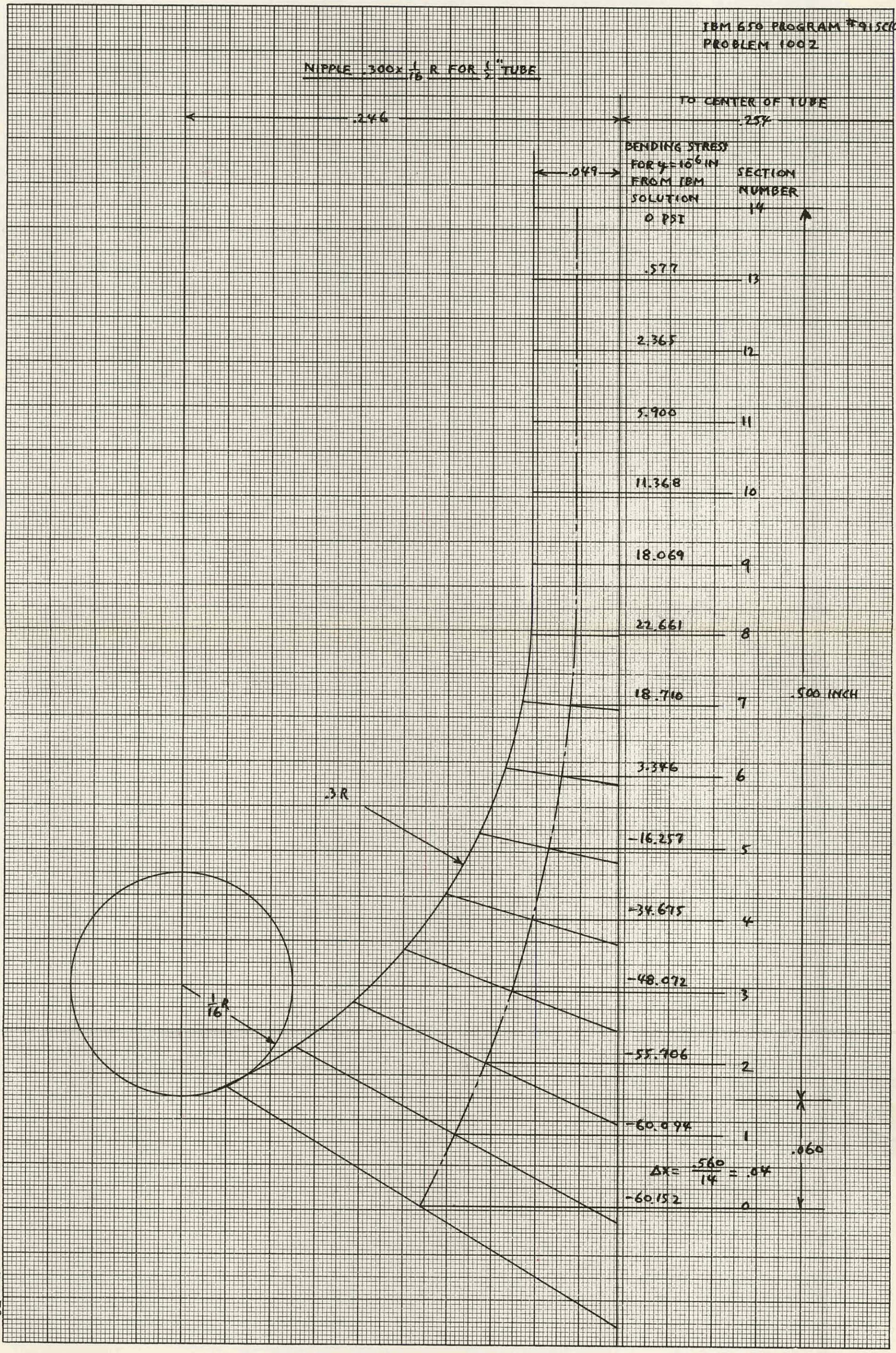
$\theta \times 10^6$				
η	1001	1002	1003	1004
$\frac{1}{2}$	0	0	0	0
$1\frac{1}{2}$	-.905,265,56	-.873,369,74	-.897,897,91	-.974,486,58
$2\frac{1}{2}$	-1.899,970.2	-1.889,035,5	-2.050,525,5	-2.288,158,3
$3\frac{1}{2}$	-2.916,356,2	-3.010,117,1	-3.318,046,3	-3.768,287,4
$4\frac{1}{2}$	-3.793,369,5	-4.042,957,2	-4.474,550,7	-4.989,218,8
$5\frac{1}{2}$	-4.351,136,6	-4.654,300,2	-5,006.088,0	-5.195,142,4
$6\frac{1}{2}$	-4,347,547,2	-4.499,422,6	-4.424,208,0	-4.086,345,4
$7\frac{1}{2}$	-3.630,080.2	-3.457.060,5	-3.011,473,2	-2.594.268,0
$8\frac{1}{2}$	-2.438,702,8	-2.093,611,1	-1.689698,3	-1.343,400,9
$9\frac{1}{2}$	-1.184,707,0	-.984.287,32	-.731,687,93	-.499,493,62
$10\frac{1}{2}$	.379,001,43	-.286,361,24	-.156,584,51	-.026,620,391
$11\frac{1}{2}$	.048,541,654	.075,853,287	.126,312.56	.186,129,84
$12\frac{1}{2}$	.224.946,74	.221.031,77	.231,210,82	.253,664.48
$13\frac{1}{2}$	.269,951,10	.256,462,40	.253,355,39	.262,948,58
$14\frac{1}{2}$	.269,522,36	.256,251,47	.253,281,37	.262,866,36

7-F71

SECTION	BENDING STRESS FOR $\psi = 1 \times 10^{-6}$				
	1001 SPECIAL START	1001	1002	1003	1004
0	-62.560	-66.546	-60.152	-56.901	-52.971
1	-59.630	-64.094	-60.094	-57.603	-57.010
2	-52.982	-57.697	-55.706	-56.320	-56.766
3	-43.099	-47.806	-48.072	-47.610	-47.724
4	-29.204	-33.451	-34.675	-33.061	-29.626
5	-13.687	-17.150	-16.257	-11.838	-3.902
6	2.348	.089	3.346	10.444	18.428
7	15.532	14.667	18.710	23.010	24.303
8	20.993	21.385	22.661	21.529	20.374
9	19.366	20.425	18.069	15.604	13.745
10	12.103	13.123	11.368	9.367	7.702
11	6.236	6.964	5.900	4.608	3.465
12	2.475	2.873	2.365	1.709	1.000
13	.595	.733	.577	.361	.151
14	.002	-.007	.003	-.001	-.001

IBM 650 PROGRAM #9150100
PROBLEM 1002

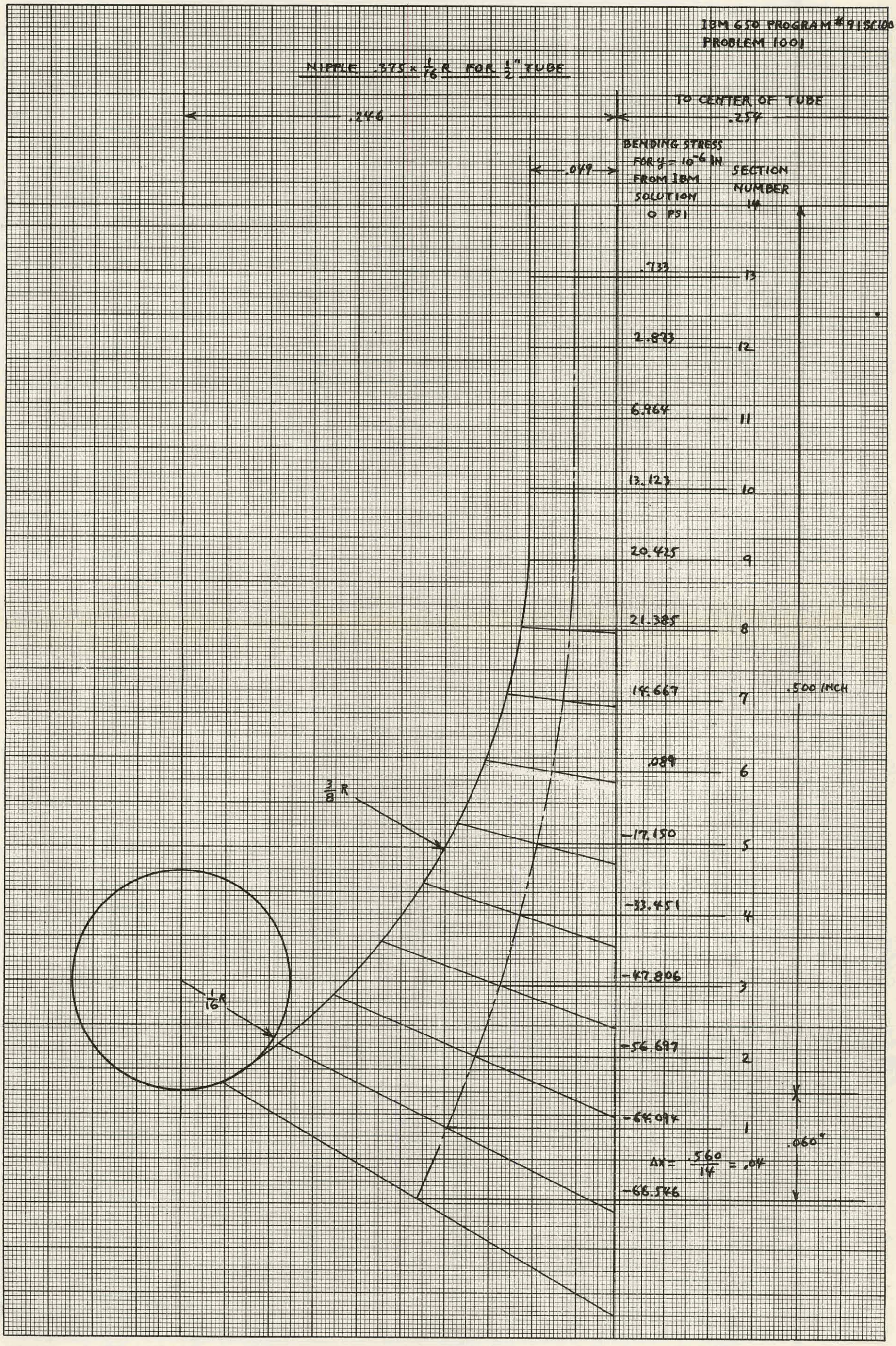
NIPPLE .300 x 1/16 R FOR 1/2" TUBE



7-FT13

IBM 650 PROGRAM #91SC100
PROBLEM 1001

NIPPLE .375 × $\frac{1}{16}$R FOR $\frac{1}{2}$" TUBE

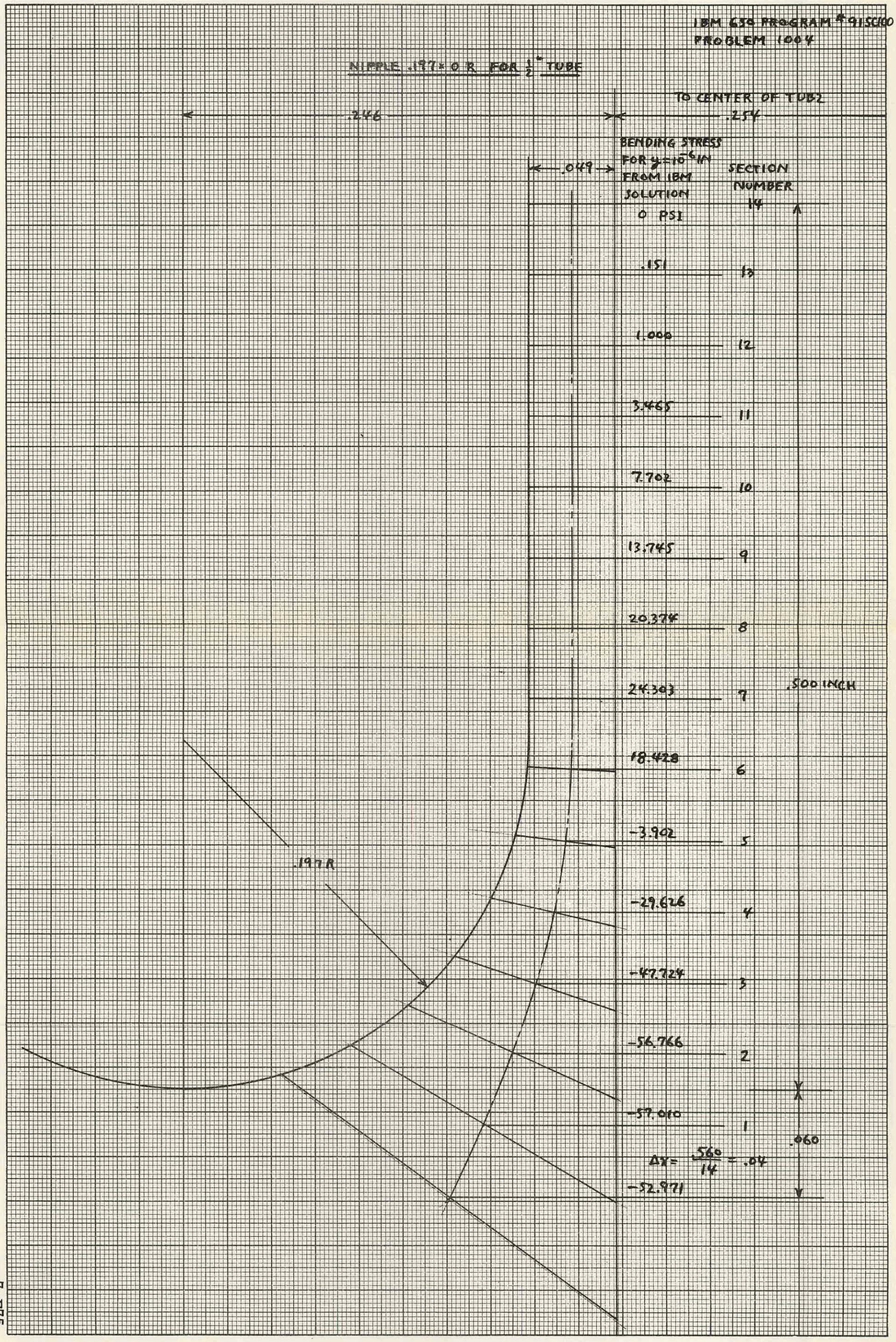


7-F74

208 208

IBM 650 PROGRAM #91SC00
PROBLEM 1004

NIPPLE .197 × 0 R FOR $\frac{1}{2}$" TUBE

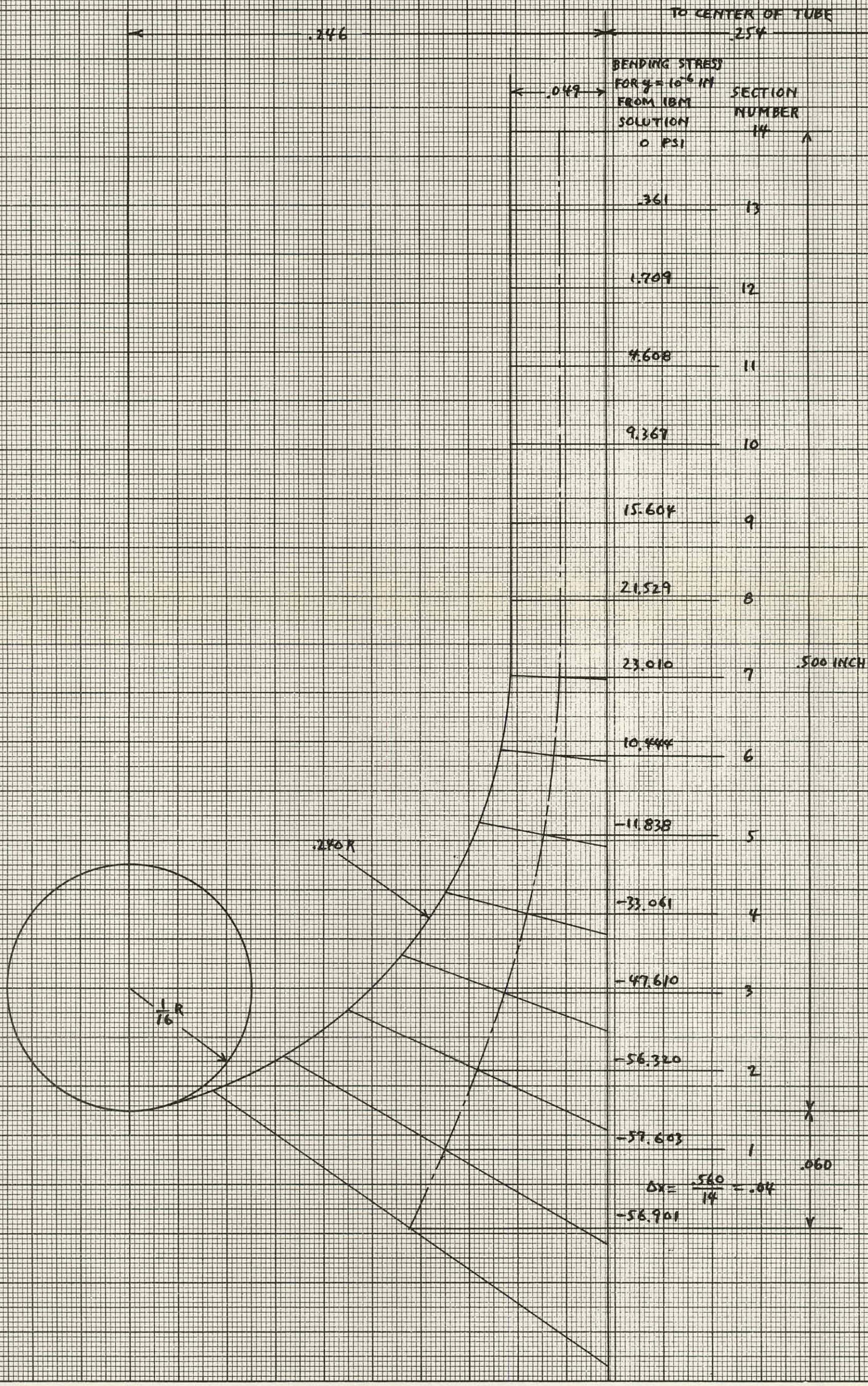


7-F75

602 406

IBM 650 PROGRAM # 91SC00
PROBLEM 1003

NIPPLE .240 × 1/16 R FOR 1/2" TUBE



97-F76

SUPPORT BRACKETS

Computations : Overturning moment

Primary sodium inlet, lever arm = 24'-8" = 296"

Due to thrust —	4000 × 296 =	1,184,000 in-lb
" " moment —		480,000

Primary sodium outlet, lever arm = 20"

Due to thrust —	4000 × 20 =	80,000
" " moment —		480,000

Secondary sodium inlet, lever arm = 32"

Due to thrust —	4000 × 32 =	128,000
" " moment —		480,000

Secondary sodium outlet, lever arm = 29'-0" = 348"

Due to thrust —	4000 × 348 =	1,392,000
" " moment —		480,000
	Total =	4,704,000 in-lb

Bracket loads

Distance of anchor bolt from center line of unit = 31.875"
Weight of unit with sodium = 53,000 lb.

Arrangement A: $2P(31.875 \times \sqrt{2}) - 53{,}000\left(\frac{31.875}{\sqrt{2}}\right) - 4{,}704{,}000 = 0$

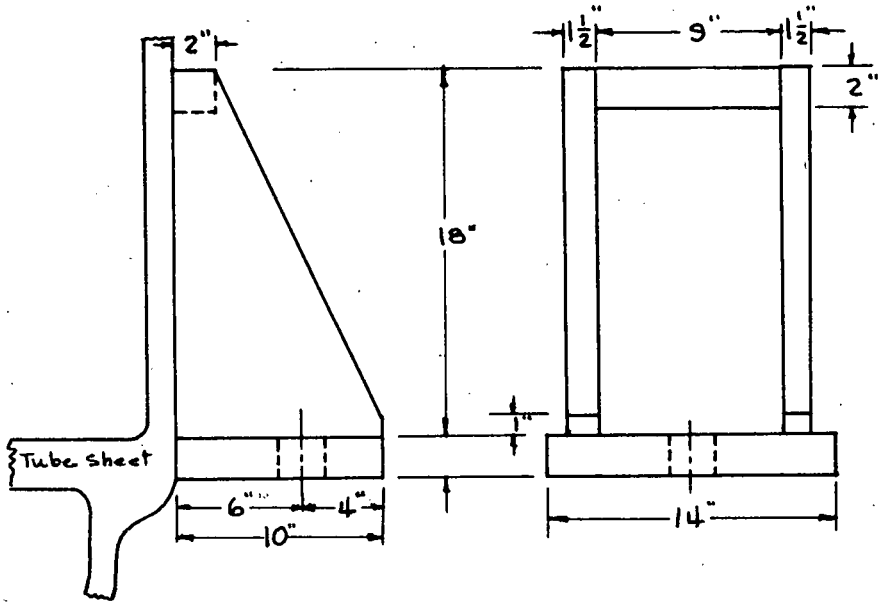
$P = 65{,}400$ lb.

Arrangement B: $P(31.875 \times 2) - 53{,}000\,(31.875) - 4{,}704{,}000 = 0$

$P = 100{,}200$ lb.

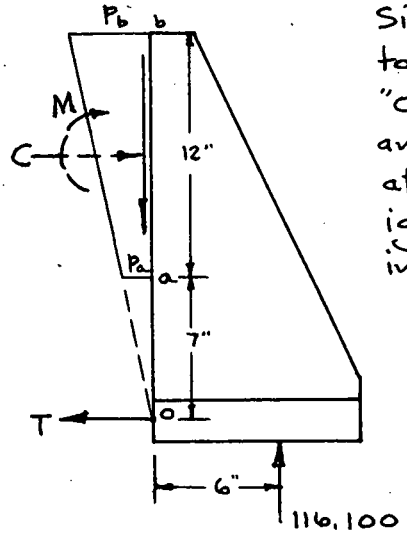
Arrangement C: $P(31.875 \times 1.5) - 53{,}000\left(\frac{31.875}{2}\right) - 4{,}704{,}000 = 0$

$P = 116{,}200$ lb.



Computations: Pressures between shell and bracket.

Arrangement C.



Since tube sheet is very rigid compared to shell, assume bracket pivots around "O" and that pressure between shell and bracket varies linearly from zero at "O" to a maximum at "b". Further, ignore contribution of pressure forces in region O-a.

Thus, $\frac{p_b}{p_a} = \frac{19}{7}$ (1)

$$\Sigma M_o = 12 p_a (13) + \frac{p_b - p_a}{2}(12)\left(7 + \tfrac{2}{3}\times 12\right) - 116{,}100\,(6) = 0 \quad (2)$$

(1) in (2): $156\left(\frac{7}{19}\right)p_b + \frac{12}{19}p_b(6)(15) - 696{,}600 = 0$

whence $p_b = \frac{696{,}600}{114.3} = 6090$ lb/in.

and $p_a = 2240$ lb/in.

Then $C = \frac{P_b + P_a}{2}(12) = 50{,}000$ lb. (3)

$M = \frac{P_b - P_a}{2}\left(\frac{6}{2}\right)\left(\frac{3}{3}\times 12\right) = 12(P_b - P_a) = 46{,}200$ in-lb (4)

Computations: stresses in shell wall and bracket proper

Consider C and M as nozzle loads on shell, on a loading pad, say, 12" x 12".

After Bijlaard (Welding J., Dec., 1954):

$a = 24.5 + 0.6875 = 25.1875$"

$c = 6$" $\quad t = 1.375$"

$\beta = \frac{c}{a} = 0.238 \quad \gamma = \frac{a}{t} = 18.3$

Due to thrust: $\frac{M_\phi}{P} = .08$; $\frac{M_x}{P} = .05$; $\frac{N_\phi}{P/a} = 2$; $\frac{N_x}{P/a} = 2.5$

$\therefore M_\phi = 50{,}000(.08) = 4{,}000$ in-lb/in

$N_\phi = \frac{50{,}000}{25.1875}(2) = 3{,}975$ lb/in

$M_x = 50{,}000(.05) = 2{,}500$ in-lb/in

$N_x = \frac{50{,}000}{25.1875}(2.5) = 4{,}960$ lb/in.

Due to longitudinal moment:

$\frac{M_\phi}{M/a\beta} = .03$; $\frac{M_x}{M/a\beta} = .05$; $\frac{N_\phi}{M/a^2\beta} = 2$; $\frac{N_x}{M/a^2\beta} = .7$

$\therefore M_\phi = .03\frac{46{,}200}{25.1875(.238)} = 232$ in-lb/in

$N_\phi = 2\frac{46{,}200}{(25.1875)^2(.238)} = 613$ lb/in

$M_x = .05\frac{46{,}200}{25.1875(.238)} = 385$ in-lb/in

$N_x = .7\frac{46{,}200}{(25.1875)^2(.238)} = 214$ lb/in

Then $\sigma = 6\frac{M_\phi}{t^2} + \frac{N_\phi}{t} = 6\frac{4000+232}{(1.375)^2} + \frac{3975+613}{1.375} = 16{,}780$ psi

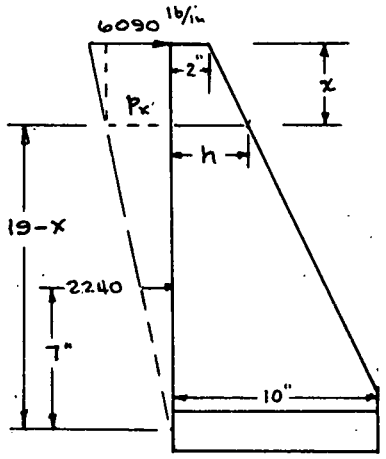
or $\sigma = 6\frac{M_x}{t^2} + \frac{N_x}{t} = 6\frac{2500+385}{(1.375)^2} + \frac{4960+214}{1.375} = 12{,}910$ psi

$\therefore$ maximum $\sigma = 16{,}780$ psi.

Max. radial tension stress transmitted to tube sheet

$$= \frac{50{,}000}{14(2)} = \underline{\underline{1790 \text{ psi}}}$$

To find bending stresses in bracket:



$$P_x = \frac{2240}{7}(19-x)$$

$$M_x = \left[6090 - \frac{2240}{7}(19-x)\right]\frac{x}{2}\left(\frac{2x}{3}\right) + \frac{2240}{7}(19-x)\,x\left(\frac{x}{2}\right)$$

$$= +320\frac{x^3}{3} + 3045x^2 - 320\frac{x^3}{2}$$

$$= 3045x^2 - 53.3x^3$$

$$h = 2 + \frac{8}{17}x$$

$$\frac{I}{c} = \frac{bh^2}{6} = \frac{3}{6}\left(2+\frac{8}{17}x\right)^2$$

Since $\sigma = \frac{M}{I/c}$ varies with x, bracket is not uniformly stressed. Therefore, maximum bending stress may be expected to occur at base, where $x = 18"$. Hence,

$$\sigma_{max} = \frac{6(50{,}000\times 12 + 46{,}200)}{(3)(10)^2}$$

$$= \frac{6(646{,}200)}{3(100)} \cong \underline{\underline{13{,}000 \text{ psi}}}$$

Summary (Arrangement C):

For worst possible conditions, these are the maximum stresses –

shell wall: 16,780 psi (bending)

bracket proper: 13,000 (")

bracket base: 1790 (tension)

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For Arrangements A and B, p_b, p_a, C, M, M_ϕ, M_x, N_ϕ, N_x, and δ are all linearly proportional to the load on bracket. Therefore,

for Arrangement A: $\sigma_{shell} = 16,780 \frac{65,400}{116,100} = 9440$ psi

$\sigma_{bracket} = 13,000 \frac{65,400}{116,100} = 7310$ psi

for Arrangement B: $\sigma_{shell} = 16,780 \frac{100,200}{116,100} = 14,470$ psi

$\sigma_{bracket} = 13,000 \frac{100,200}{116,100} = 11,200$ psi

SECTION - 8

ALCO SUPPLEMENT

ALCO SUPPLEMENT

Six charts dealing with transient temperature problems which have been developed by ALCO PRODUCTS INC. have been included in this report.

This supplement contains the derivations upon which these charts were based. Two of these charts correspond to Figures A 3-5 and A 3-6 in the Navy Structural Design Basis for Reactor Pressure Vessels, considerably extended and revised.

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Transient Heat Flow

List of Symbols

T = temperature °F
T_o = temperature of heated surface of plate °F
T_e = temperature of insulated surface of plate °F
T_{ave} = average temperature of plate °F
T_L = temperature of fluid °F
ΔT_L = temperature change of fluid during Δt
ΔT_f = total temperature change of fluid during time t
t = time
Q = quantity of heat Btu
q = quantity of heat per unit time
C_p = specific heat Btu/lb °F
w = density
x = distance in direction of heat flow
X or l = thickness of plate
h = coefficient of heat transfer, or film coefficient Btu/hr ft² °F
k = thermal conductivity Btu/hr ft °F
α = thermal diffusivity = $k/w C_p$

Dimensionless Numbers

$M = \frac{\Delta x^2}{\alpha \Delta t}$ Modulus in finite-difference solution

N_{Bi} = Biot Number = $\frac{\text{Film Coef.} \times \text{thickness}}{\text{Conductivity}} = \frac{h X}{k}$

N = Biot Number of Δx increment = $\frac{h \Delta x}{k}$

N_{Fo} = Fourier Number = $\frac{\text{diffusivity} \times \text{time}}{(\text{thickness})^2} = \frac{\alpha t}{X^2} = \frac{\alpha t}{l^2}$

Transient Heat Flow in one Direction

Analytical Solutions

Thin Plate: The solution is made for change of temperature considered positive. The actual temperatures are the initial plus or minus the change. One surface is in contact with fluid, the other surface is insulated. $T_L = \Delta T_f$ = change of fluid temperature, T = change of plate temperature. The heat transferred from the fluid equals heat stored in plate, so that

$$dQ = h(T_L - T)\,dt = w C_p X \frac{dT}{dt}\,dt$$

$$\frac{dT}{dt} = A(T_L - T) \qquad \text{where } A = \frac{h}{w C_p X} = \frac{h\alpha}{kX}$$

Sudden change of fluid temperature, T_L = constant

$$\frac{dT}{T_L - T} = A\,dt$$

$$-\log_e(T_L - T) = At + \text{Const.}$$

$T = 0$ at $t = 0$ so that $\text{Const} = -\log_e T_L$

$$\log_e \frac{T_L - T}{T_L} = -At$$

This gives $\frac{T_L - T}{T_L} = e^{-At}$ or $\frac{T}{T_L} = 1 - e^{-At}$

Linear change, or fluid temperature increasing linearly with time

$T_L = at$

$$\frac{dT}{dt} = A(at - T)$$

Let $v = A(at - T)$, $\frac{dv}{dt} = Aa - A\frac{dT}{dt}$

$$\frac{dv}{dt} - Aa = -Av$$

$$\frac{dv}{a - v} = A\,t$$

$$-\log_e(a - v) = At + \text{Const}$$

$T = 0$ at $t = 0$ so that $\text{Const} = -\log_e a$

$$\log_e \frac{a - v}{a} = -At$$

This gives $\frac{T}{T_L} = \frac{1}{At}\left(At - 1 + e^{-At}\right)$ (Note: $At = N_{Bi} N_{Fo}$)

Finite Plate : Sudden change of fluid temperature

Change of temperatures in a flat plate with one surface in contact with a fluid and the other surface insulated. Initially, all temperatures are the same. The fluid temperature is suddenly changed and held constant.

An analytical solution is given P.100, Carslaw and Jaeger, Conduction of Heat in Solids, Oxford University Press, 1947.
Let T = change of plate temperature considered positive, so that the actual temperature values = initial plus or minus the change.

$$A_m = \frac{2N_{Bi}}{\beta_m^2 + N_{Bi} + N_{Bi}^2} \quad \text{and} \quad \beta_m = \text{positive roots of } \beta \tan\beta = N_{Bi}$$

$$\frac{T}{\Delta T_f} = 1 - \sum_{m=1}^{\infty} \frac{A_m}{\cos\beta_m} e^{-\beta_m^2 N_{Fo}} \cos\beta_m \frac{x}{\ell} \quad , \quad \frac{1}{\cos\beta_m} = (-1)^{m+1} \sqrt{1 + \frac{N_{Bi}^2}{\beta_m^2}}$$

where the origin for x is at the insulated surface

$$\frac{T_o}{\Delta T_f} = 1 - \sum_{m=1}^{\infty} A_m e^{-\beta_m^2 N_{Fo}}$$

$$\frac{T_e}{\Delta T_f} = 1 - \sum_{m=1}^{\infty} \frac{A_m}{\cos\beta_m} e^{-\beta_m^2 N_{Fo}}$$

$$\frac{T_{ave}}{\Delta T_f} = 1 - \sum_{m=1}^{\infty} \frac{N_{Bi} A_m}{\beta_m^2} e^{-\beta_m^2 N_{Fo}}$$

N_{Bi}^{-1}	β_1	β_2	β_3	β_4	β_5	β_6
.01	1.555,245	4.665,765	7.776,374	10.887,130	13.998,090	17.109,307
.02	1.540,006	4.620,246	7.701,159	10.783,164	13.866,633	16.951.884
.05	1.496.129	4.491,480	7.495,412	10,511,670	13.541,977	16.586,394
.1	1.428,870	4.305,801	7.228,110	10,200,263	13.214,186	16.259,361
.2	1.313,838	4.033,568	6.909,596	9.892,753	12,935,221	16.010,658
.3	1.219.952	3.854,598	6.742,334	9.754,073	12,820,734	15.914,432
.5	1.076,874	3.643,597	6.578,334	9,629,560	12.722,299	15.833,611
.8	.930,757	3.485,897	6.473,921	9.554,863	12,664,751	15.786,978
1.0	.860,334	3.425,619	6,437.298	9.529,334	12,645,287	15.771.285
2.0	.653,271	3.292,310	6.361,623	9.477,485	12.606,013	15.739,719
4.0	.480,094	3.219,099	6.322,705	9.451,223	12,586,231	15.723,861

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The number of terms of the series required for 5 decimal place accuracy in $T/\Delta T_f$ is about as follows:

N_{Fo}	Number of terms
.03 to .045	6
.045 to .07	5
.07 to .1	4
.1 to .3	3
.3 to 1.0	2
1.0 up	1

The relative change in temperature of the insulated surface compared to that of the heated surface and average is given in the following table.

N_{Bi}	$\frac{T_e}{T_o}$				$\frac{T_e}{T_{ave}}$			
	N_{Fo}=.03	.045	.0625	.075	.03	.045	.0625	.075
0	.00009	.00172	.00936	.01965	.00046	.00717	.03316	.06358
.01	.00008	.00157	.00879	.01865	.00041	.00667	.03154	.06105
.02	.00007	.00146	.00831	.01779	.00037	.00626	.03017	.05885
.1	.00005	.00101	.00617	.01369	.00026	.00467	.02396	.04826
.5	.00003	.00068	.00425	.00957	.00019	.00346	.01816	.03718
1.0	.00003	.00062	.00388	.00872	.00017	.00322	.01698	.03478
4.0	.00002	.00057	.00357	.00802	.00015	.00302	.01599	.03275

If $N_{Bi}^{-1} = 0$ or $N_{Bi} = \infty$, $T_o/\Delta T_f = 1$

$$\frac{T_e}{\Delta T_f} = 1 - \frac{4}{\pi}\sum_{n=1}^{\infty}\frac{(-1)^{n+1}}{2n-1}e^{-\frac{\pi^2}{4}(2n-1)^2 N_{Fo}}$$

$$\frac{T_{ave}}{\Delta T_f} = 1 - \frac{8}{\pi^2}\sum_{n=1}^{\infty}\frac{1}{(2n-1)^2}e^{-\frac{\pi^2}{4}(2n-1)^2 N_{Fo}}$$

$$\frac{T}{\Delta T_f} = 1 - \frac{4}{\pi}\sum_{n=1}^{\infty}\frac{(-1)^{n+1}}{2n-1}e^{-\frac{\pi^2}{4}(2n-1)^2 N_{Fo}}\cos\frac{\pi}{2}(2n-1)\frac{x}{\ell}$$

where the origin for x is at the insulated surface
If x' is the length variable with the origin at the heated surface

$$\frac{x}{\ell} = 1 - \frac{x'}{\ell}, \quad (-1)^{n+1}\cos\frac{\pi}{2}(2n-1)x = \sin\frac{\pi}{2}(2n-1)\frac{x'}{\ell}$$

and

$$\frac{T}{\Delta T_f} = 1 - \frac{4}{\pi}\sum_{n=1}^{\infty}\frac{e^{-\frac{\pi^2}{4}(2n-1)^2 N_{Fo}}}{2n-1}\sin\frac{\pi}{2}(2n-1)\frac{x'}{\ell}$$

Semi-infinite Solid. Sudden change of fluid temperature

Change of temperature in a semi-infinite solid with one plane surface in contact with a fluid. Initially, the temperatures are all the same. The fluid temperature is suddenly changed and held constant.

An analytical solution is given P.53, Carslaw and Jaeger. Let T = change of temperature considered positive so that the actual temperature values = initial plus or minus the change.

$$\frac{T}{\Delta T_f} = \operatorname{erfc} \frac{x}{2\sqrt{\alpha t}} - e^{\frac{h}{K}x + \frac{h^2}{K^2}\alpha t} \operatorname{erfc}\left[\frac{x}{2\sqrt{\alpha t}} + \frac{h}{K}\sqrt{\alpha t}\right]$$

where the origin for x is at the surface, and the definition for $\operatorname{erfc} x$ is

$$\operatorname{erfc} x = 1 - \operatorname{erf} x = \frac{2}{\sqrt{\pi}} \int_x^\infty e^{-\xi^2} d\xi$$

and

$$\operatorname{erf} x = \frac{2}{\sqrt{\pi}} \int_0^x e^{-\xi^2} d\xi$$

Values of $\operatorname{erf} x$ and $\operatorname{erfc} x$ may be obtained from published tables of the Probability Integral or Error Function.

If N_{Bi} and N_{Fo} are the Biot and Fourier numbers for the first part of the solid between $x=0$ and ℓ, the expression for $\frac{T}{\Delta T_f}$ may be written

$$\frac{T}{\Delta T_f} = \operatorname{erfc} \frac{x/\ell}{2\sqrt{N_{Fo}}} - e^{N_{Bi}x/\ell + N_{Bi}^2 N_{Fo}} \operatorname{erfc}\left(\frac{x/\ell}{2\sqrt{N_{Fo}}} + N_{Bi}\sqrt{N_{Fo}}\right)$$

This formula may equally well be applied to any plate of finite thickness, ℓ, if the value of N_{Fo} is such that the change in temperature has not yet been appreciably felt at the insulated surface. The table on the preceding page indicates the values of N_{Fo} for which this can be considered as true.

The average temperature between $x = 0$ and ℓ is

$$\frac{T_{ave}}{\Delta T_f} = \int_0^1 \frac{T}{\Delta T_f} \, d\frac{x}{\ell}$$

Let $Z = \dfrac{x/\ell}{2\sqrt{N_{Fo}}}$

$$\frac{T}{\Delta T_f} = \operatorname{erfc} Z - e^{2N_{Bi}\sqrt{N_{Fo}}Z + N_{Bi}^2 N_{Fo}} \operatorname{erfc}\left(Z + N_{Bi}\sqrt{N_{Fo}}\right)$$

$$\frac{T_{ave}}{\Delta T_f} = 2\sqrt{N_{Fo}} \int_0^{\frac{1}{2\sqrt{N_{Fo}}}} \frac{T}{\Delta T_f} \, dZ$$

$$= 2\sqrt{N_{Fo}} \int_0^{\infty} \frac{T}{\Delta T_f} \, dZ$$ Approximately, since $\frac{T}{\Delta T_f}$ is negligible for large value of $\frac{1}{2\sqrt{N_{Fo}}}$

$$\frac{T_{ave}}{\Delta T_f} = 2\sqrt{N_{Fo}} \int_0^{\infty} \operatorname{erfc} Z \, dZ - 2\sqrt{N_{Fo}} \int_0^{\infty} e^{2N_{Bi}\sqrt{N_{Fo}}Z + N_{Bi}^2 N_{Fo}} \operatorname{erfc}\left(Z + N_{Bi}\sqrt{N_{Fo}}\right) dZ$$

$$2\int_0^{\infty} \operatorname{erfc} Z \, dZ = 1.1284$$

Let $Y = Z + N_{Bi}\sqrt{N_{Fo}}$

$$\frac{T_{ave}}{\Delta T_f} = 1.1284\sqrt{N_{Fo}} - 2\sqrt{N_{Fo}} \int_{N_{Bi}\sqrt{N_{Fo}}}^{\infty} e^{2N_{Bi}\sqrt{N_{Fo}}Y - N_{Bi}^2 N_{Fo}} \operatorname{erfc} Y \, dY$$

$$= 1.1284\sqrt{N_{Fo}} - \frac{1}{N_{Bi}} \int_{Y = N_{Bi}\sqrt{N_{Fo}}}^{\infty} \operatorname{erfc} Y \, de^{2N_{Bi}\sqrt{N_{Fo}}Y - N_{Bi}^2 N_{Fo}}$$

Integrating by parts

$$\frac{T_{ave}}{\Delta T_f} = 1.1284\sqrt{N_{Fo}} - \frac{1}{N_{Bi}} \left[e^{2N_{Bi}\sqrt{N_{Fo}}Y - N_{Bi}^2 N_{Fo}} \operatorname{erfc} Y \right]_{Y = N_{Bi}\sqrt{N_{Fo}}}^{\infty}$$
$$+ \frac{1}{N_{Bi}} \int_{Y = N_{Bi}\sqrt{N_{Fo}}}^{\infty} e^{2N_{Bi}\sqrt{N_{Fo}}Y - N_{Bi}^2 N_{Fo}} (-1) \frac{2}{\sqrt{\pi}} e^{-Y^2} dY$$

$$= 1.1284\sqrt{N_{Fo}} + \frac{1}{N_{Bi}} e^{N_{Bi}^2 N_{Fo}} \operatorname{erfc} N_{Bi}\sqrt{N_{Fo}} - \frac{2}{N_{Bi}\sqrt{\pi}} \int_{Y = N_{Bi}\sqrt{N_{Fo}}}^{\infty} e^{-\left(Y - N_{Bi}\sqrt{N_{Fo}}\right)^2} dY$$

$$= 1.1284\sqrt{N_{Fo}} + \frac{1}{N_{Bi}} e^{N_{Bi}^2 N_{Fo}} \operatorname{erfc} N_{Bi}\sqrt{N_{Fo}} - \frac{2}{N_{Bi}\sqrt{\pi}} \int_0^{\infty} e^{-Z^2} dZ$$

since $\frac{2}{\sqrt{\pi}} \int_0^\infty e^{-Z^2} dZ = 1$

$$\frac{T_{ave}}{\Delta T_f} = 1.1284 \sqrt{N_{Fo}} + \frac{1}{N_{Bi}} e^{N_{Bi}^2 N_{Fo}} \operatorname{erfc} N_{Bi}\sqrt{N_{Fo}} - \frac{1}{N_{Bi}}$$

The temperature at the heated surface is

$$\frac{T_o}{\Delta T_f} = 1 - e^{N_{Bi}^2 N_{Fo}} \operatorname{erfc} N_{Bi}\sqrt{N_{Fo}}$$

Therefore

$$\frac{T_{ave}}{\Delta T_f} = 1.1284 \sqrt{N_{Fo}} - \frac{1}{N_{Bi}} \frac{T_o}{\Delta T_f}$$

For large values of $N_{Bi}\sqrt{N_{Fo}} = a$

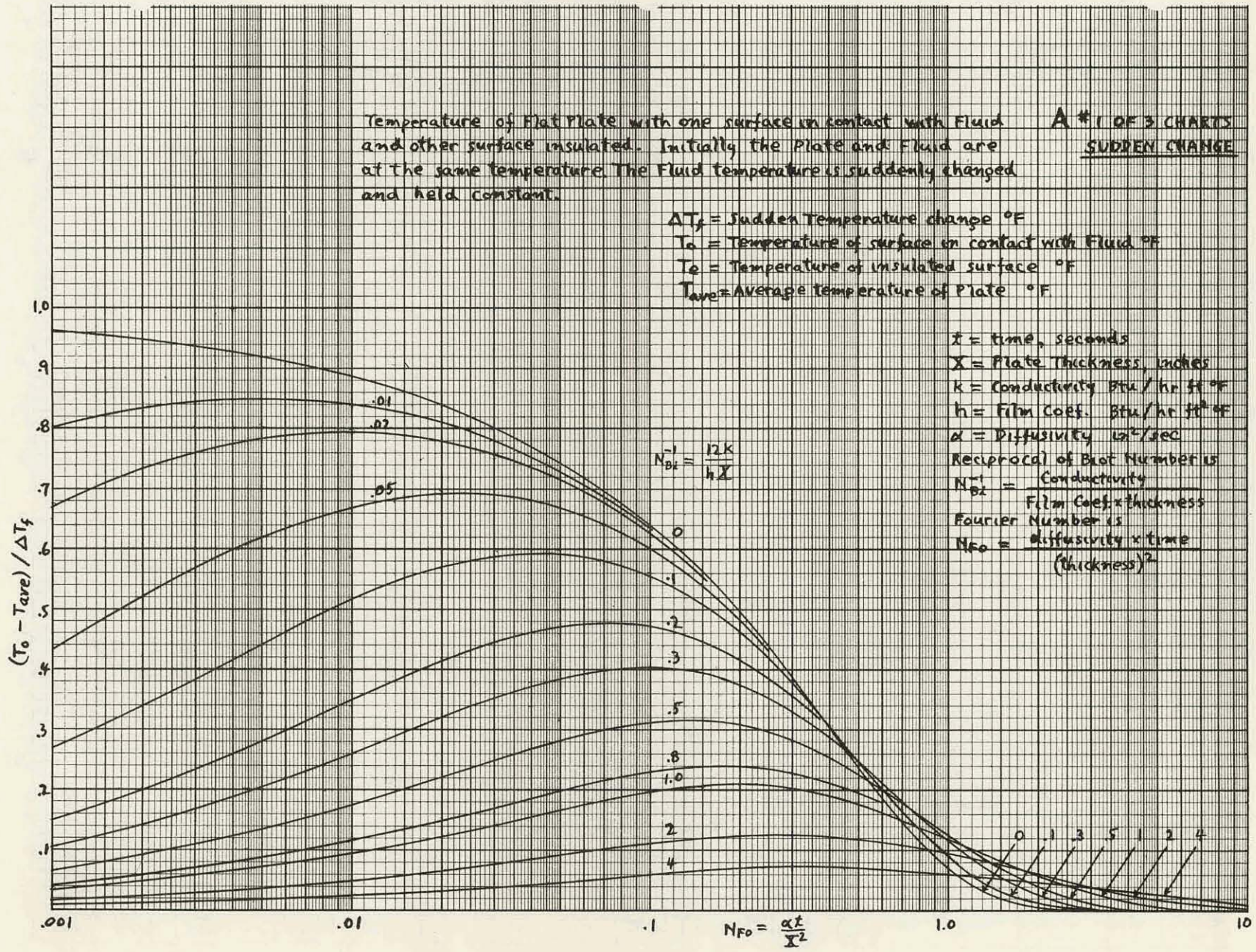
$$\frac{T_o}{\Delta T_f} = 1 - e^{a^2} \operatorname{erfc} a$$

$$= 1 - \frac{1}{a\sqrt{\pi}} \left[1 - \frac{1}{2a^2} + \frac{1 \cdot 3}{(2a^2)^2} - \frac{1 \cdot 3 \cdot 5}{(2a^2)^3} + \frac{1 \cdot 3 \cdot 5 \cdot 7}{(2a^2)^4} - \cdots \right]$$

where the sum of the series differs from the exact value required for the equality by less than the last term.

The analytical solutions described above were used to calculate the temperatures in a finite flat plate in contact with a fluid that has a sudden change of temperature. The results are summarized on the accompanying three charts labeled "Sudden Change".

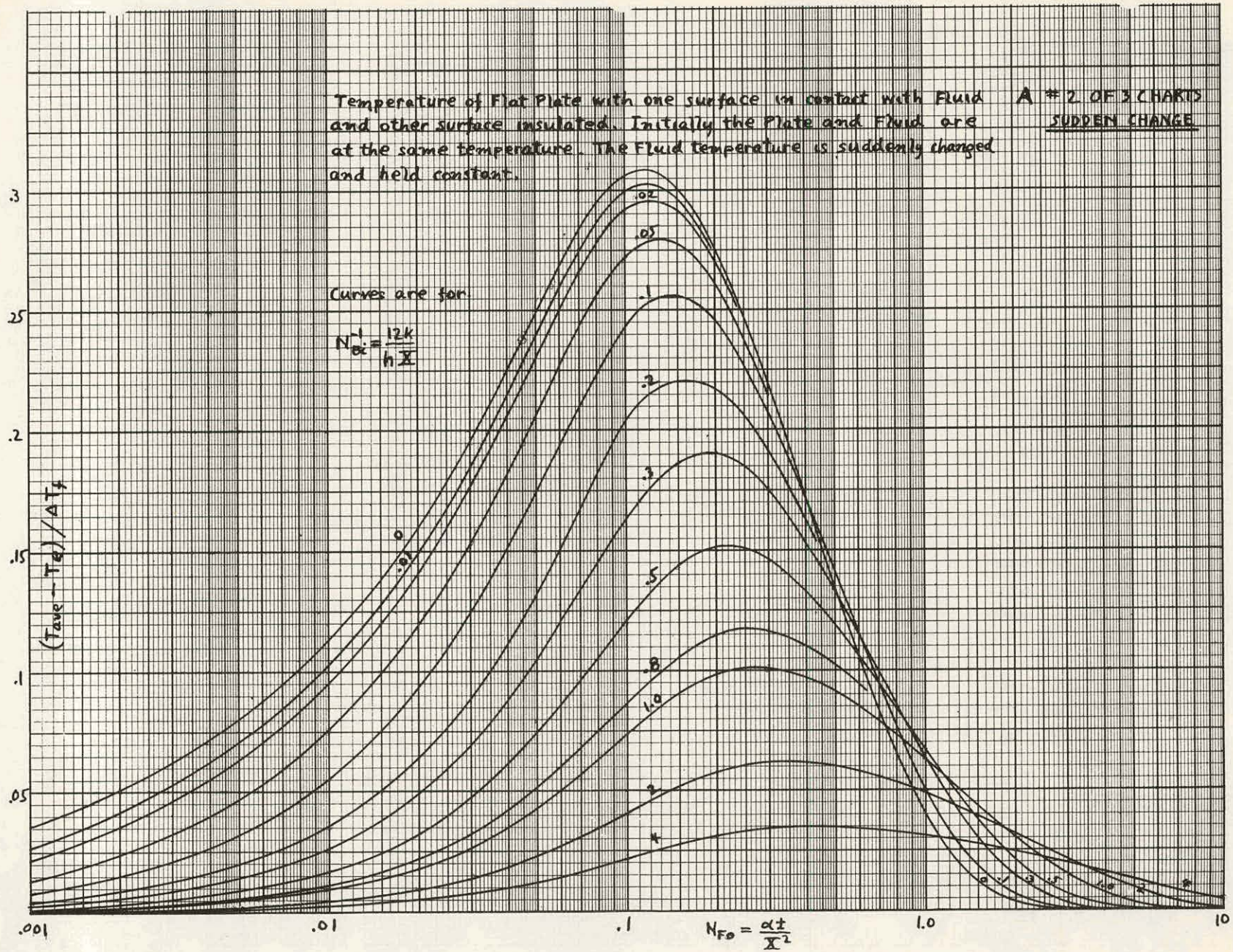
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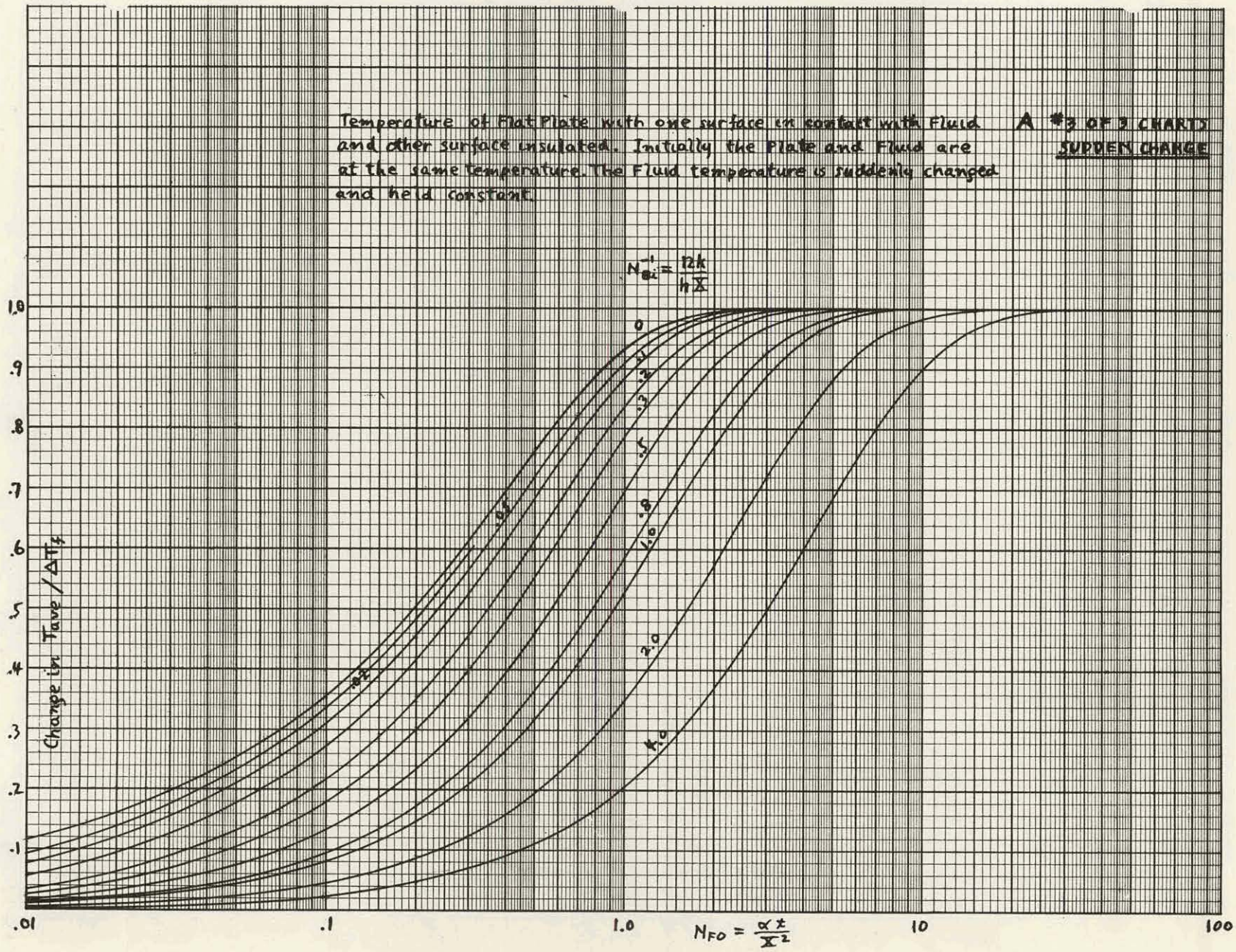

Temperature of Flat Plate with one surface in contact with Fluid and other surface insulated. Initially the Plate and Fluid are at the same temperature. The Fluid temperature is suddenly changed and held constant.
A #1 OF 3 CHARTS
SUDDEN CHANGE
ΔTf = Sudden Temperature change °F
To = Temperature of surface in contact with Fluid °F
Te = Temperature of insulated surface °F
Tave = Average temperature of Plate °F
t = time, seconds
X = Plate Thickness, inches
k = Conductivity Btu/hr ft °F
h = Film Coef. Btu/hr ft² °F
α = Diffusivity in²/sec
Reciprocal of Biot Number is
N⁻¹Bi = Conductivity / Film Coef. x thickness
Fourier Number is
NFo = diffusivity x time / (thickness)²
N⁻¹Bi = 12k / hX
(To - Tave) / ΔTf
NFo = αt / X²
1.0
.9
.8
.7
.6
.5
.4
.3
.2
.1
.01
.02
.05
0
.1
.2
.3
.5
.8
1.0
2
4
.001
.01
.1
1.0
10
0 .1 .3 .5 1 2 4

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Temperature of Flat Plate with one surface in contact with Fluid and other surface insulated. Initially the Plate and Fluid are at the same temperature. The Fluid temperature is suddenly changed and held constant.
A # 2 OF 3 CHARTS
SUDDEN CHANGE
Curves are for
$N_{\alpha}^{-1} = \frac{12k}{h\,X}$
$(T_{ave} - T_i)/(T_f - T_i)$
$N_{Fo} = \frac{\alpha t}{X^2}$
.3
.25
.2
.15
.1
.05
.001
.01
.1
1.0
10
0
.02
.05
.1
.2
.3
.5
.8
1.0
2
4
0
1
3
5
10
2
∞


Temperature of Flat Plate with one surface in contact with Fluid and other surface insulated. Initially the Plate and Fluid are at the same temperature. The Fluid temperature is suddenly changed and held constant.
A #3 OF 3 CHARTS
SUDDEN CHANGE
$N_{Bi}^{-1} = \frac{12k}{h\Delta}$
0
.1
.2
.3
.5
.8
1.0
2.0
4.0
∞
Change in Tave / ΔTf
1.0
.9
.8
.7
.6
.5
.4
.3
.2
.1
.01
.1
1.0
10
100
$N_{FO} = \frac{\alpha t}{\Delta^2}$

Finite-difference Method for Transient Heat Flow in one Direction

References: E. Schmidt, Thermodynamik, 5th Auflage, Springer, 1953
W. H. Giedt, Heat Transfer, D. van Nostrand, 1957
Dusinberre, Numerical Analysis of Heat Flow, McGraw-Hill, 1949
McAdams, Heat Transmission, 3rd Ed. McGraw-Hill, 1954
Schneider, Conduction Heat Transfer - Addison-Wesley, 1955

The Fourier equation of heat conduction in one direction is

$$\alpha \frac{\partial^2 T}{\partial x^2} = \frac{\partial T}{\partial t}$$

where T = temperature
t = time
x = distance in direction of heat flow
$\alpha = \frac{k}{w C_p}$ = thermal diffusivity
k = thermal conductivity
w = density
C_p = specific heat
h = coefficient of heat transfer

Consider a plate divided into a number of increments of equal thickness ΔX. Also, consider equal time intervals Δt. Let the temperature of the middle of the m-th increment of x at the beginning of the n-th time increment be $T_{m,n}$.

The Fourier equation in finite-difference form may be written

$$\alpha \frac{\frac{T_{m+1,n} - T_{m,n}}{\Delta x} - \frac{T_{m,n} - T_{m-1,n}}{\Delta x}}{\Delta x} = \frac{T_{m,n+1} - T_{m,n}}{\Delta t}$$

which may be reduced to

$$T_{m,n+1} = \frac{1}{M}\left[T_{m-1,n} + (M-2)\,T_{m,n} + T_{m+1,n}\right]$$

where $$M = \frac{\Delta x^2}{\alpha \Delta t}$$

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If the temperature of the first surface at $x=0$ is specified as a function of time, and if the initial temperatures of the plate are given, the subsequent temperatures in the plate may be calculated by repeated application of the above relation. The first surface may be taken as the middle of the 0-th Δx-increment, one half of which is fictitous.

To make a calculation, two of the three quantities, M, Δx, Δt must be assigned.

M may have any convenient value but should not be assigned a value much less than 2, since in this case the calculation may become unstable. (Schneider).

If $M = 2$

$$T_{m,n+1} = \frac{1}{2}\left(T_{m-1,n} + T_{m+1,n}\right)$$

and the temperatures may be obtained by a graphical construction called the "Schmidt Method".

If the plate has a finite thickness X with the last surface insulated, the temperatures are the same as occur in the first half of a plate $2X$ thick where the first and last surfaces have the same temperature-time specifications.

The insulated surface may be taken as the last surface of the e-th Δx-increment. This gives

$$\Delta x = \frac{X}{e+\frac{1}{2}}$$

and the temperature of the middle of the e-th increment is

$$T_{e,n+1} = \frac{1}{M}\left[T_{e-1,n} + (M-2)\,T_{e,n} + T_{e,n}\right]$$

Or, the insulated surface may be taken as the middle of the e-th Δx- increment, one half of which is fictitous. In this case

$$\Delta x = \frac{X}{e}$$

and the temperature of the insulated surface is

$$T_{e,n+1} = \frac{1}{M}\left[T_{e-1,n} + (M-2)\,T_{e,n} + T_{e-1,n}\right]$$

The values of $M, \Delta x, \Delta t$ may be changed at any time during the calculation by assigning new values to two of them and computing the third from $M = \Delta x^2 / \alpha \Delta t$. This change requires a reasonably good estimate for the temperatures at the middle of the new Δx-increments.

If the space increment is doubled and M is unchanged, the time increment is increased 4 times. In this case, if $T_{-2}, T_{-1}, T_0, T_1, T_2$ represent five of the original temperatures in succession, an approximate value for the new temperature at T_0 is

$$T_0' = \frac{1}{8}\left(-T_{-2} + 2T_{-1} + 6T_0 + 2T_1 - T_2\right)$$

This expression is an average of the two values that are obtained at T_0 by a line through the mid-points of the chords between $T_{-2}T_{-1}$ and $T_{-1}T_0$, and a line through the mid-points of the chords between T_0T_1 and T_1T_2.

If a calculation is made for a plate using constants indicated by the subscript o, the same temperatures at the same time intervals will occur in another plate at the middle of each Δx-increment if

$$\Delta x = \Delta x_0 \sqrt{\frac{\alpha}{\alpha_0}}$$

and

$$h = \frac{K}{K_0} \frac{h_0}{\sqrt{\frac{\alpha}{\alpha_0}}}$$

Various methods have been proposed to obtain the first surface temperature in the finite-difference calculation when the plate is in contact with a fluid at different temperature. These methods are based on the law of heat conduction

$$q = -K A \frac{dT}{dx}$$

and the equation defining the heat transfer coefficient

$$q = h A (T_\infty - T_s)$$

where q is the quantity of heat transferred per unit time from the fluid to the plate across area A, T_s is the surface temperature, and T_∞ is the ambient fluid temperature.

The assumption is made that the slope of the temperature curve at the first surface is the same as the slope of the chord across the first increment, or

$$\left(\frac{dT}{dx}\right)_0 = \frac{T_{0m} - T_{1m}}{\Delta x}$$

Let T_{Lm} be the temperature T_∞ at the beginning of the n-th time increment. Then

$$k\,\frac{T_{om}-T_{im}}{\Delta x} = h\,(T_{Lm}-T_{om})$$

or

$$\frac{T_{om}-T_{im}}{\Delta x} = \frac{T_{Lm}-T_{om}}{k/h}$$

This shows that if the temperature gradient from $x=0$ to $x=\Delta x$ is extended, its ordinate at a distance of k/h in front of the first surface will be the fluid temperature. Solving for T_{om} gives

$$T_{om} = \frac{T_{Lm} + T_{im}\frac{k}{h\Delta x}}{1+\frac{k}{h\Delta x}}$$

$$= T_{Lm} - (T_{Lm}-T_{im})\,\frac{\frac{k}{h\Delta x}}{1+\frac{k}{h\Delta x}}$$

or

$$T_{om} = T_{Lm}\,\frac{1}{1+\frac{k}{h\Delta x}} + T_{im}\left(1-\frac{1}{1+\frac{k}{h\Delta x}}\right)$$

This equation is the same as equ 12-21 Page 129 Dusinberre.

In the Schmidt Method described in his book "Thermodynamik" 1955, the middle of the Δx-increments are taken at $x=\frac{1}{2}\Delta x, \frac{3}{2}\Delta x, \ldots$ In this case, if the chord between $x=0$ and $x=\frac{1}{2}\Delta x$ is extended, its ordinate at a distance of k/h in front of the first surface is the fluid temperature. Therefore

$$\frac{T_{om}-T_{\frac{1}{2}m}}{\Delta x/2} = \frac{T_{Lm}-T_{om}}{k/h}$$

Solving for T_{om} gives

$$T_{om} = \frac{T_{Lm} + T_{\frac{1}{2}m}\,\frac{2k}{h\Delta x}}{1+\frac{2k}{h\Delta x}}$$

$$= T_{Lm} - (T_{Lm}-T_{\frac{1}{2}m})\,\frac{\frac{2k}{h\Delta x}}{1+\frac{2k}{h\Delta x}}$$

or

$$T_{om} = T_{Lm}\,\frac{1}{1+\frac{2k}{h\Delta x}} + T_{\frac{1}{2}m}\left(1-\frac{1}{1+\frac{2k}{h\Delta x}}\right)$$

This equation is the same as equ 12-23 Page 130 Dusinberre.

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Schmidt added an auxiliary plane at $x = -\frac{1}{2}\Delta x$ in front of the first surface and obtained an auxiliary temperature, $T_{-\frac{1}{2}n}$, at this surface by similar triangles, thus,

$$\frac{T_{L,n+1} - T_{0m}}{k/h} = \frac{T_{-\frac{1}{2}n} - T_{0n}}{\Delta x/2}$$

This gives $$T_{-\frac{1}{2}n} = T_{0m} + (T_{L,n+1} - T_{0m}) \frac{h\Delta x}{2k}$$

He then applied the Schmidt rule $T_{m,n+1} = \frac{1}{2}(T_{m-1,n} + T_{m+1,n})$ for $m = \frac{1}{2}, \frac{3}{2}, \frac{5}{2}, \ldots$ to obtain the temperatures at $t = (n+1)\Delta t$. The calculation is adapted for numerical computation or graphical procedure.

Finite difference calculations for the temperature in a plate in contact with a fluid where the fluid temperature was increasing linearly with time, failed to have a reasonable heat balance between the heat stored in the plate and that transferred from the fluid. At the 10-th line of the calculation, or after the computations for 10 time increments, there was about 30 % more heat stored in the plate than had been lost by the fluid, and at the 30-th line there was about 20 % more.

It appears likely that the fault in the method is due to the assumption that the slope of the temperature gradient at the first surface is the same as the slope of the chord across the first increment. Actually, the temperature gradient is curving so rapidly at the beginning of the plate that this assumption is very approximate.

The following method was developed to correct this fault. It is based on a heat balance between the transferred and stored heat at each step of the calculation.

The equation defining the heat transfer coefficient is

$$q = \frac{dQ}{dt} = h(T_L - T)$$

where q is the quantity of heat transferred per unit time from the fluid to the plate across a unit area, T is the surface temperature, and T_L is the ambient fluid temperature.

Integrating this equation, the heat transferred to the plate during the interval from t_a to t_b is

$$Q = h \int_{t_a}^{t_b} (T_L - T)\, dt$$

If $(T_{ave})_a$ and $(T_{ave})_b$ are the average temperatures of the plate at t_a and t_b, the heat stored in the plate during the interval from t_a to t_b is

$$Q_b - Q_a = w\, C_p\, \mathcal{I} \left[(T_{ave})_b - (T_{ave})_a \right]$$

Equating the heat stored to the heat transferred

$$(T_{ave})_b - (T_{ave})_a = \frac{h}{w C_p \mathcal{I}} \int_{t_a}^{t_b} (T_L - T)\, dt$$

This equation may be applied to the finite-difference solution of transient heat flow by obtaining a approximation for $\int_{t_a}^{t_b} T\, dt$ from the integration of parabolas through the tops of several consecutive values of T.

In the finite-difference solution, the plate is divided into a number of equal increments. The temperatures are obtained at the middle of these increments at equal time increments, and are identified by double subscripts, the first subscript indicating the space position and the second indicating the time. The first surface is taken as the middle of the zero-th increment, one half of which is fictitous.

The insulated surface may be taken as the last surface of the end or e-th increment, or as the middle of the e-th increment, one half of which is fictitous. The following solution applies to the former case. Similar formulas may be written for the latter.

Since $$\sum_{m=1}^{e} T_{s,m} = \frac{1}{M} \left[T_{0,n-1} - T_{1,n-1} + M \sum_{m=1}^{e} T_{m,n-1} \right]$$

$$(T_{ave})_n = \frac{1}{e+\frac{1}{2}} \left[\frac{T_{0n}}{2} + \sum_{m=1}^{e} T_{m,n} \right]$$

$$= \frac{1}{e+\frac{1}{2}} \left[\frac{T_{0n}}{2} + \frac{T_{0,n-1} - T_{1,n-1}}{M} + \sum_{m=1}^{e} T_{m,n-1} \right]$$

$$= \frac{1}{e+\frac{1}{2}} \left[\frac{T_{0n} - T_{0,n-1}}{2} + \frac{T_{0,n-1} - T_{1,n-1}}{M} + \frac{T_{0,n-1}}{2} + \sum_{m=1}^{e} T_{m,n-1} \right]$$

$$(T_{ave})_n - (T_{ave})_{n-1} = \frac{1}{e+\frac{1}{2}} \left[\frac{T_{0n}}{2} - \frac{M-2}{2M} T_{0,n-1} - \frac{T_{1,n-1}}{M} \right]$$

Let $N = \frac{h \Delta x}{K}$ = the Biot number of the increment Δx

$$\frac{h}{w C_p \mathcal{I}} = \frac{h \alpha}{K \mathcal{I}} = \frac{h \Delta x\, \alpha}{K (e+\frac{1}{2}) \Delta x^2} = \frac{N}{(e+\frac{1}{2}) M \Delta t}$$

A parabola of the second degree through the tops of three consecutive value, T_0, T_1, T_2, of the temperature of the first surface is as follows.

$$T = T_0 + \frac{-3T_0 + 4T_1 - T_2}{2\Delta t}\, t + \frac{T_0 - 2T_1 + T_2}{2\Delta t^2}\, t^2$$

and

$$\int_0^{\Delta t} T\,dt = \frac{\Delta t}{12}(5T_0 + 8T_1 - T_2)$$

$$\int_{\Delta t}^{2\Delta t} T\,dt = \frac{\Delta t}{12}(-T_0 + 8T_1 + 5T_2)$$

A parabola of the third degree through the tops of four consecutive values T_0, T_1, T_2, T_3 is as follows

$$T = T_0 + \frac{-11T_0 + 18T_1 - 9T_2 + 2T_3}{6\Delta t}\, t + \frac{2T_0 - 5T_1 + 4T_2 - T_3}{2\Delta t^2}\, t^2 + \frac{-T_0 + 3T_1 - 3T_2 + T_3}{6\Delta t^3}\, t^3$$

and

$$\int_0^{\Delta t} T\,dt = \frac{\Delta t}{24}(9T_0 + 19T_1 - 5T_2 + T_3)$$

$$\int_{\Delta t}^{2\Delta t} T\,dt = \frac{\Delta t}{24}(-T_0 + 13T_1 + 13T_2 - T_3)$$

$$\int_{2\Delta t}^{3\Delta t} T\,dt = \frac{\Delta t}{24}(T_0 - 5T_1 + 19T_2 + 9T_3)$$

3 Ordinate Parabola Solution

Initial temperatures a function of x.
Fluid temperature increasing linearly with time

Let ΔT_L represent the change of fluid temperature during one time increment, so that $T_L = T_{L0} + \frac{\Delta T_L}{\Delta t} t$

$$\int_{(n-1)\Delta t}^{n\Delta t} T_L \, dt = \left(T_{L0} + \frac{2n-1}{2} \Delta T_L\right) \Delta t$$

Assuming a parabola through $T_{0,0}$, T_{01}, T_{02}, the equation for the equality of the stored and transferred heat for the first two increments of t gives

$$(T_{ave})_1 - (T_{ave})_0 = \frac{N}{(e+\frac{1}{2})M}\left[T_{L0} + \tfrac{1}{2}\Delta T_L - \tfrac{1}{12}(5T_{00} + 8T_{01} - T_{02})\right]$$

$$(T_{ave})_2 - (T_{ave})_1 = \frac{N}{(e+\frac{1}{2})M}\left[T_{L0} + \tfrac{3}{2}\Delta T_L - \tfrac{1}{12}(-T_{00} + 8T_{01} + 5T_{02})\right]$$

Therefore, the following two simultaneous equations are obtained for T_{01} and T_{02}.

$$\frac{T_{01}}{2} - \frac{M-2}{2M}T_{00} - \frac{T_{10}}{M} = \frac{N}{M}\left[T_{L0} + \tfrac{1}{2}\Delta T_L - \tfrac{1}{12}(5T_{00} + 8T_{01} - T_{02})\right]$$

$$\frac{T_{02}}{2} - \frac{M-2}{2M}T_{01} - \frac{T_{11}}{M} = \frac{N}{M}\left[T_{L0} + \tfrac{3}{2}\Delta T_L - \tfrac{1}{12}(-T_{00} + 8T_{01} + 5T_{02})\right]$$

The solution of these gives

$$T_{01} = \frac{\left(5 + \frac{6M}{N}\right)\left\{12\left(T_{L0} + \frac{1}{2}\Delta T_L\right) - \left[5 - \frac{6(M-2)}{N}\right]T_{00} + \frac{12}{N}T_{10}\right\} + 12\left(T_{L0} + \frac{3}{2}\Delta T_L\right) + T_{00} + \frac{12}{N}T_{11}}{8 - \frac{6(M-2)}{N} + \left(8 + \frac{6M}{N}\right)\left(5 + \frac{6M}{N}\right)}$$

$$T_{02} = \left(8 + \frac{6M}{N}\right)T_{01} - \left\{12\left(T_{L0} + \frac{1}{2}\Delta T_L\right) - \left[5 - \frac{6(M-2)}{N}\right]T_{00} + \frac{12}{N}T_{10}\right\}$$

Assuming a parabola through $T_{0,n-2}, T_{0,n-1}, T_{0,n}$ $(n>1)$, the equation for the equality of the stored and transferred heat gives

$$(T_{ave})_n - (T_{ave})_{n-1} = \frac{N}{(e+\frac{1}{2})M}\left[T_{L0} + \frac{2n-1}{2}\Delta T_L - \frac{1}{12}\left(-T_{0,n-2} + 8T_{0,n-1} + 5T_{0n}\right)\right]$$

Therefore

$$\frac{T_{0n}}{2} - \frac{M-2}{2M}T_{0,n-1} - \frac{T_{1,M-1}}{M} = \frac{N}{M}\left[T_{L0} + \frac{2n-1}{2}\Delta T_L - \frac{1}{12}\left(-T_{0,n-2} + 8T_{0,n-1} + 5T_{0n}\right)\right]$$

This gives the following expression for T_{0n} $(n>1)$

$$\left(5+\frac{6M}{N}\right)T_{0n} = 12\left(T_{L0} + \frac{2n-1}{2}\Delta T_L\right) + T_{0,n-2} - \left[8 - \frac{6(M-2)}{M}\right]T_{0,n-1} + \frac{12}{N}T_{1,n-1}$$

Since $(T_{ave})_n - (T_{ave})_0 = (T_{ave})_1 - (T_{ave})_0 + \sum_{s=2}^{n}\left[(T_{ave})_s - (T_{ave})_{s-1}\right]$

a check formula may be obtained as follows,

$$(T_{ave})_n - (T_{ave})_0 = \frac{N}{(e+\frac{1}{2})M}\left\{T_{L0} + \frac{1}{2}\Delta T_L - \frac{1}{12}\left(5T_{00} + 8T_{01} - T_{02}\right)\right.$$

$$\left. + \sum_{s=2}^{n}\left(T_{L0} + \frac{2s-1}{2}\Delta T_L\right) - \frac{1}{12}\sum_{s=2}^{n}\left(-T_{0,s-2} + 8T_{0,s-1} + 5T_{0,s}\right)\right\}$$

After rearranging this becomes for $n>1$

$$(T_{ave})_n - (T_{ave})_0 = \frac{N}{(e+\frac{1}{2})M}\left[n\left(T_{L0} + \frac{n}{2}\Delta T_L\right) - \sum_{s=1}^{n}T_{0s} + \frac{1}{12}\left(-4T_{00} - 3T_{01} + T_{02} - T_{0,n-1} + 7T_{0n}\right)\right]$$

or, if $e+\frac{1}{2}$ is eliminated

$$\frac{T_{0n}}{2} + \sum_{m=1}T_{m,n} - \left(\frac{T_{00}}{2} + \sum_{m=1}T_{m,0}\right)$$

$$= \frac{N}{M}\left[n\left(T_{L0} + \frac{n}{2}\Delta T_L\right) - \sum_{s=1}^{n}T_{0s} + \frac{1}{12}\left(-4T_{00} - 3T_{01} + T_{02} - T_{0,n-1} + 7T_{0n}\right)\right]$$

where the left side is due to stored heat and the right side is due to transferred heat.

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4 Ordinate Parabola Solution

Initial temperatures a function of x

Fluid temperature increasing linearly with time

Assuming a parabola through $T_{00}, T_{01}, T_{02}, T_{03}$, the equation for the equality of the stored and transferred heat for the first three increments of t gives,

$$(T_{ave})_1 - (T_{ave})_0 = \frac{N}{(e+\frac{1}{2})M}\left[T_{L0} + \frac{1}{2}\Delta T_L - \frac{1}{24}(9T_{00} + 19T_{01} - 5T_{02} + T_{03})\right]$$

$$(T_{ave})_2 - (T_{ave})_1 = \frac{N}{(e+\frac{1}{2})M}\left[T_{L0} + \frac{3}{2}\Delta T_L - \frac{1}{24}(-T_{00} + 13T_{01} + 13T_{02} - T_{03})\right]$$

$$(T_{ave})_3 - (T_{ave})_2 = \frac{N}{(e+\frac{1}{2})M}\left[T_{L0} + \frac{5}{2}\Delta T_L - \frac{1}{24}(T_{00} - 5T_{01} + 19T_{02} + 9T_{03})\right]$$

Therefore, the following three simultaneous equations are obtained for T_{01}, T_{02}, T_{03}

$$\frac{T_{01}}{2} - \frac{M-2}{2M}T_{00} - \frac{T_{10}}{M} = \frac{N}{M}\left[T_{L0} + \frac{1}{2}\Delta T_L - \frac{1}{24}(9T_{00} + 19T_{03} - 5T_{02} + T_{03})\right]$$

$$\frac{T_{02}}{2} - \frac{M-2}{2M}T_{01} - \frac{T_{11}}{M} = \frac{N}{M}\left[T_{L0} + \frac{3}{2}\Delta T_L - \frac{1}{24}(-T_{00} + 13T_{01} + 13T_{02} - T_{03})\right]$$

$$\frac{T_{03}}{2} - \frac{M-2}{2M}T_{02} - \frac{T_{12}}{M} = \frac{N}{M}\left[T_{L0} + \frac{5}{2}\Delta T_L - \frac{1}{24}(T_{00} - 5T_{01} + 19T_{02} + 9T_{03})\right]$$

where $\frac{T_{12}}{M} = \frac{1}{M^2}\left[T_{01} + (M-2)T_{11} + T_{21}\right]$

These may be rearranged into the following three equations

$$\left(19 + \frac{12M}{N}\right)T_{01} - 5T_{02} + T_{03} = 24\left(T_{L0} + \frac{1}{2}\Delta T_L\right) - \left[9 - \frac{12(M-2)}{N}\right]T_{00} + \frac{24}{N}T_{10}$$

$$\left[13 - \frac{12(M-2)}{N}\right]T_{01} + \left(13 + \frac{12M}{N}\right)T_{02} - T_{03} = 24\left(T_{L0} + \frac{3}{2}\Delta T_L\right) + T_{00} + \frac{24}{N}T_{11}$$

$$-\left(5 + \frac{24}{MN}\right)T_{01} + \left[19 - \frac{12(M-2)}{N}\right]T_{02} + \left(9 + \frac{12M}{N}\right)T_{03} = 24\left(T_{L0} + \frac{5}{2}\Delta T_L\right) - T_{00} + \frac{24(M-2)}{MN}T_{11} + \frac{24}{MN}T_{21}$$

Assuming a parabola through $T_{0,n-3}$, $T_{0,n-2}$, $T_{0,n-1}$, T_{0m}, the equation for the equality of the stored and transferred heat gives

$$(T_{ave})_n-(T_{ave})_{n-1}=\frac{N}{(e+\frac{1}{2})M}\left[T_{L0}+\frac{2n-1}{2}\Delta T_L-\frac{1}{24}\left(T_{0,n-3}-5T_{0,n-2}+19T_{0,n-1}+9T_{0m}\right)\right]$$

Therefore

$$\frac{T_{0m}}{2}-\frac{M-2}{2}T_{0,n-1}-\frac{T_{1,M-1}}{M}=\frac{N}{M}\left[T_{L0}+\frac{2n-1}{2}\Delta T_L-\frac{1}{24}\left(T_{0,n-3}-5T_{0,n-2}+19T_{0,n-1}+9T_{0m}\right)\right]$$

This gives the following expression for T_{0m} $(n>2)$

$$\left(9+\frac{12M}{N}\right)T_{0m}=24\left(T_{L0}+\frac{2n-1}{2}\Delta T_L\right)-T_{0,n-3}+5T_{0,n-2}-\left[19-\frac{12(M-2)}{N}\right]T_{0,n-1}+\frac{24}{N}T_{1,n-1}$$

The check formula for $n>2$ is

$$(T_{ave})_n-(T_{ave})_0=\frac{N}{(e+\frac{1}{2})M}\left[n\left(T_{L0}+\frac{n}{2}\Delta T_L\right)-\sum_{s=1}^{n}T_{0s}+\frac{1}{24}\left(-9T_{00}-4T_{01}+T_{02}+T_{0,n-2}-4T_{0,n-1}+15T_{0n}\right)\right]$$

or, if $e+\frac{1}{2}$ is eliminated

$$\frac{T_{0n}}{2}+\sum_{m=1}T_{m,n}-\left(\frac{T_{00}}{2}+\sum_{m=1}T_{m0}\right)$$
$$=\frac{N}{M}\left[n\left(T_{L0}+\frac{n}{2}\Delta T_L\right)-\sum_{s=1}^{n}T_{0s}+\frac{1}{24}\left(-9T_{00}-4T_{01}+T_{02}+T_{0,n-2}-4T_{0,n-1}+15T_{0n}\right)\right]$$

where the left side is due to stored heat and the right side is due to transferred heat.

In general satisfactory results from the finite-difference method of calculation for transient heat flow depend on the stability of the calculation. Stability in this case refers to the inherent characteristic of the calculation to reduce small errors as the computations are carried through several time increments. This is a necessary characteristic since at best the solution for the temperatures at the fist few time increments is very approximate.

An unstable calculation tends to increase small errors. In this case the temperatures will show unnatural oscilations which tend to increase as the computations are carried through several time increments.

If the calculation is made for a sudden change of the surface temperature corresponding to an infinite Biot number, it is better to consider that half of the sudden change has occured at $t=0$ and the whole change at Δt. (Dusinberre) A satisfactory start in this case would be to use $M=4$ for the computation of several time increments and then drop back to $M=2$.

When the Biot number $N_{Bi} = \frac{h\,\Delta x}{k}$ is finite, calculations for the heat balance between the stored and transferred heat may be included in the finite difference method of calculation. Results may be obtained using the three ordinate parabola and $M=2$ if the calculation remains stable. However, the four ordinate parabola solution and $M=4$ will remain stable for a wider range of difficult cases.

If a calculation with the three ordinate parabola and $M=2$ has progressed to a certain time and it is desired to double or nearly double the space increment, it is difficult to determine the temperatures at the new space increments that will not cause some oscilations in the temperature of the first surface. In this case the computations for several time increments using the four ordinate parabola and $M=4$ will avoid the oscilations. The calculation may then be continued by droping back to the three ordinate parabola and $M=2$.

The accompanying three charts labeled "Linear Change" give the calculated results for the temperatures of a flat plate with one surface in contact with a fluid and the other surface insulated. Initially the plate and fluid are at the same temperature. The fluid temperature changes linearly with time. The calculations were made by the finite-difference method which included a heat balance between the stored and transferred heat.

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Temperature of Flat Plate with one surface in contact with fluid and other surface insulated. Initially the Plate and Fluid are at the same temperature. The Fluid temperature changes linearly with time.

B #1 OF 3 CHARTS

LINEAR CHANGE

ΔT_f = Temperature change of fluid °F in t seconds
T_o = Temperature of surface in contact with fluid °F
T_e = Temperature of insulated surface °F
T_{ave} = Average temperature of Plate °F

t = time, seconds
X = Plate thickness, inches
K = Conductivity Btu/hr ft °F
h = Film Coef. Btu/hr ft² °F
α = Diffusivity in²/sec

Reciprocal of Biot Number is

$$N_{Bi}^{-1} = \frac{\text{Conductivity}}{\text{Film Coef.} \times \text{thickness}}$$

Fourier Number is

$$N_{Fo} = \frac{\text{diffusivity} \times \text{time}}{(\text{thickness})^2}$$

For $N_{Fo} > 1.28 + 4N_{Bi}^{-1}$

$$\frac{T_o - T_{ave}}{\Delta T_f} = \frac{.32}{N_{Fo}}$$

$N_{Bi}^{-1} = \frac{12k}{hX}$

0, .02, .05, .1, .2, .3, .5, .8, 1.0, 2.0, 4.0, .01

$(T_o - T_{ave})/\Delta T_f$

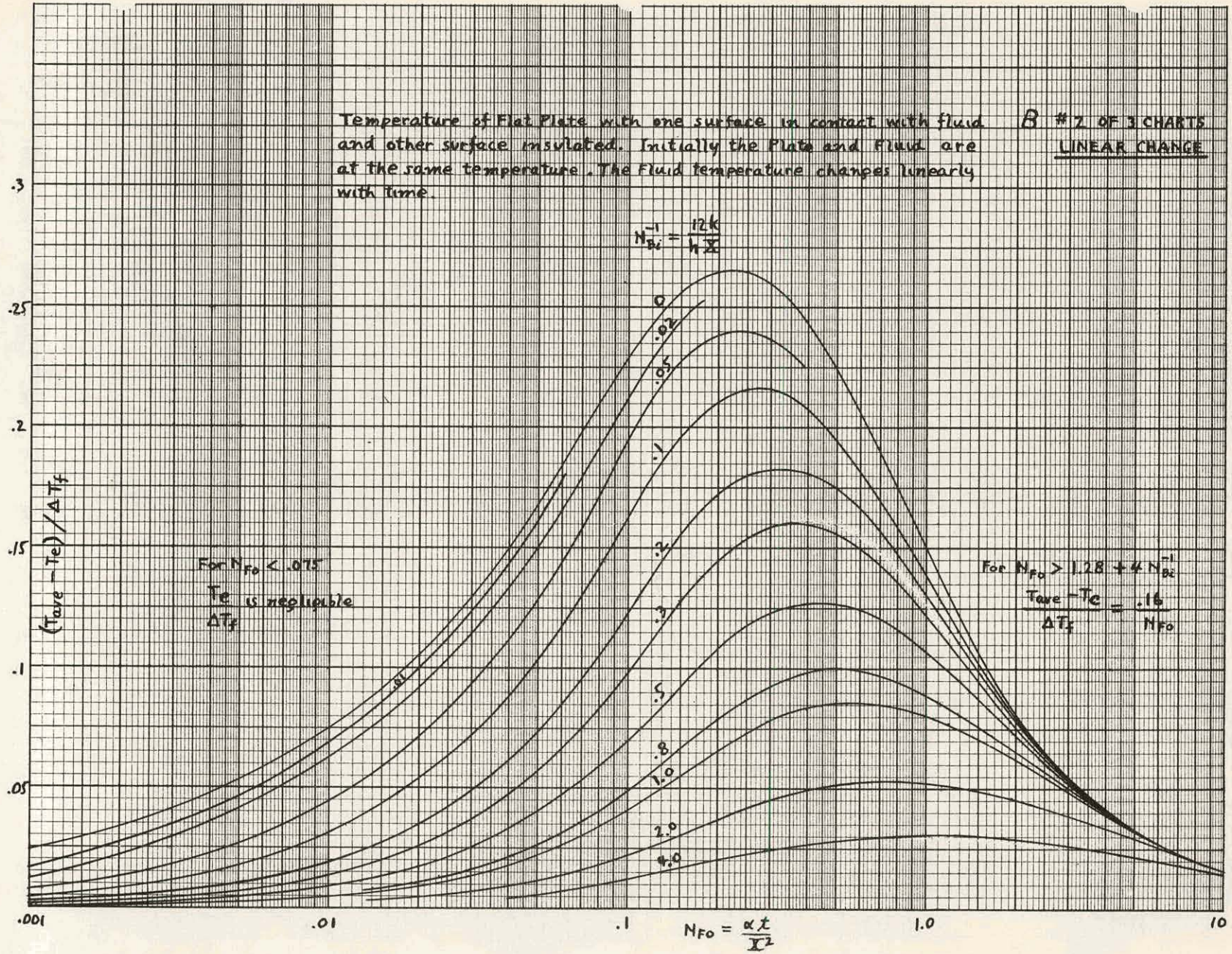
1.0, .9, .8, .7, .6, .5, .4, .3, .2, .1

.001, .01, .1, 1.0, 10

$N_{Fo} = \frac{\alpha t}{X^2}$

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Temperature of Flat Plate with one surface in contact with fluid and other surface insulated. Initially the Plate and Fluid are at the same temperature. The Fluid temperature changes linearly with time.
B #2 OF 3 CHARTS
LINEAR CHANGE
$N_{Bi}^{-1} = \frac{12k}{hX}$
0
.02
.05
.1
.2
.3
.5
.8
1.0
2.0
4.0
∞
$(T_{ave} - T_e)/\Delta T_f$
.3
.25
.2
.15
.1
.05
For $N_{Fo} < .075$
$\frac{T_e}{\Delta T_f}$ is negligible
For $N_{Fo} > 1.28 + 4 N_{Bi}^{-1}$
$\frac{T_{ave} - T_e}{\Delta T_f} = \frac{.16}{N_{Fo}}$
.001
.01
.1
1.0
10
$N_{Fo} = \frac{\alpha t}{X^2}$

Temperature of Flat Plate with one surface in contact with fluid and other surface insulated. Initially the Plate and Fluid are at the same temperature. The Fluid temperature changes linearly with time.

B #3 OF 3 CHARTS
LINEAR CHANGE

