



STANDARDS
for
STEAM SURFACE CONDENSERS

ELEVENTH EDITION

Heat Exchange Institute, Inc.

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7th Edition, 2012

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CONDENSER TECH SHEETS

Tech Sheet # 101: Operational Alert on Steam Pumps

Tech Sheet # 113: Condenser Basics

Tech Sheet # 117: Waterbox Coating

Tech Sheet # 122: Condenser Modular Replacement vs. Retube

Tech Sheet # 123: Steam Inlet Expansion Joints

Tech Sheet # 124: Relief Values vs. Rupture Discs

Tech Sheet # 125: Condenser Tube Cleaning

Tech Sheet # 131: Vacuum Breaker Valve

All condenser tech sheets are available for download on the HEI web site: www.heatexchange.org



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HEAT EXCHANGE INSTITUTE, INC.

STANDARDS for **STEAM SURFACE CONDENSERS**

ELEVENTH EDITION

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CONTENTS

1.0 NOMENCLATURE.....	Page 1
2.0 DEFINITIONS	3
2.1 Absolute Pressure	3
2.2 Circulating Water Velocity	3
2.3 Cleanliness Factor	3
2.4 Condensate Temperature Depression (Subcooling).....	3
2.5 Condenser Duty	3
2.6 Condenser Heat Transfer Coefficient	3
2.7 Condenser Pressure.....	3
2.8 Condensing Steam Temperature	3
2.9 Effective Surface	3
2.10 Effective Tube Length.....	3
2.11 Hotwell Capacity	3
2.12 Initial Temperature Difference.....	3
2.13 Logarithmic Mean Temperature Difference	3
2.14 Static Pressure	3
2.15 Temperature Rise	3
2.16 Terminal Temperature Difference	3
3.0 SYMBOLS AND UNITS	4-5
4.0 CONDENSER PERFORMANCE	6
4.1 General Considerations	6
4.2 Heat Transfer Rates	6
4.3 Oxygen Content of Condensate.....	13
4.4 Performance Curves	15
4.5 Hydraulic Loss-Circulating Water Pressure Loss	15
4.6 Condensate Temperature Depression	18
4.7 Geothermal Applications	25
5.0 SERVICE CONNECTIONS	26
5.1 General Considerations	26
5.2 Flow Data	26
5.3 Connection Locations	26
5.4 Connection Design Guidelines	27
5.5 Turbine Bypass Guidelines	28
6.0 VENTING EQUIPMENT CAPACITIES	29
6.1 Venting Requirements	29
6.2 Design Suction Pressure	29
6.3 Design Suction Temperature	29
6.4 Calculation of Water Vapor Load Component	29
6.5 Minimum Recommended Capacities	29
6.6 Rapid Evacuation Equipment	30
7.0 ATMOSPHERIC RELIEF DEVICES	35
7.1 General	35
7.2 Atmospheric Relief Valves	35
7.3 Rupture Devices	35
8.0 CONSTRUCTION	36
8.1 General	36
8.1.1 Design Philosophy	36
8.1.2 Materials of Construction.....	36
8.1.3 Design Pressures	36
8.1.4 Hydrostatic Testing	36
8.1.5 Corrosion Allowances	37
8.2 Design And Construction Methods	38
8.2.1 Design Factors of Safety	38
8.2.2 Design By More Exact Analyses and By Empirical Formula and Testing	38
8.2.3 Shell Design	38
8.2.4 Support Plate Design Guidelines	40
8.2.5 Water Box Thickness Design Guidelines	44

CONTENTS (continued)

8.2.6	Design Procedures For Flanges and Bolting	46
8.2.7	Tubesheet Design Guidelines	46
8.2.8	Condenser Tube Ends	50
8.2.9	Tubesheet and Support Plate Hole Criteria	50
8.3	Welding	52
8.4	Lagging for Extraction Lines and Feedwater Heaters	52
8.5	Fabrication for Geothermal Service.....	54
8.6	Condenser Support Systems	55
9.0	INSPECTION, QUALITY, TRANSPORTATION, AND FIELD INSTALLATION	56
9.1	Inspection and Quality of Welding Standards	56
9.2	Surface Preparation Requirements	56
9.3	Painting, Coating, Linings, and Corrosion Protection	60
9.4	Quality Assurance	60
9.5	Dimensional Tolerances	61
9.6	Shipping and Site Storage	61
9.7	Field Installation	61
9.8	Erection Superintendent Duties	62
9.9	Post Erection Walk Down	62
APPENDICES		
APPENDIX A	Typical Specification for Steam Surface Condensers.....	66
APPENDIX B	Metric Conversion Factors	71
APPENDIX C	Areas of Circular Segments	75
APPENDIX D	Procedure for Calculating Allowable Nozzle External Forces and Moments in Cylindrical Vessels	76
APPENDIX E	Air and Water Vapor Mixture Data (Dalton's Law)	81
APPENDIX F	Mechanical Characteristics of Tubing	83
APPENDIX G	Troubleshooting Guide	87
APPENDIX H	HEI Surface Condenser Data Sheet	91
APPENDIX I	Condenser Tubes Stress Values	92
APPENDIX J	Condenser Material Stress Values.....	93
APPENDIX K	Tubes Material Properties	94
APPENDIX L	Condenser Performance	95
TABLES		
TABLE 1	Uncorrected Heat Transfer Coefficients U_1	7
TABLE 2	Inlet Water Temperature Correction Factor F_w	9
TABLE 3	Tube Material and Gauge Correction Factors F_M	11
TABLE 4	Venting Capacity and Oxygen Content.....	13
TABLE 5	Gauge Correction Factor for Friction Loss R_2	18
TABLE 6	Rapid Evacuation Equipment Dry Air Capacities	31
TABLE 6A	Venting Equipment Capacities: One Condenser Shell	32
TABLE 6B	Venting Equipment Capacities: Two Condenser Shells	33
TABLE 6C	Venting Equipment Capacities: Three Condenser Shells.....	34
TABLE 7	Atmospheric Relief Valve Sizes	35
TABLE 8	Typical Materials of Construction	37
TABLE 9	Correction Factor K_1	45
TABLE 10	Correction Factor K_2	45
TABLE 11	Correction Factor K_3	45
TABLE 12	Support Plate Hole Size Limits	50
TABLE 13	Tubesheet Hole Size Limits	50
TABLE 14	Weld Acceptance Criteria.....	58
TABLE 15	Condenser Surface Preparation Requirements	63
FIGURES		
FIGURE 1	Uncorrected Heat Transfer Coefficients U_1	8
FIGURE 2	Inlet Water Temperature Correction Factor F_w	10
FIGURE 3	Absolute Pressure Limit Curves for Oxygen Content	14
FIGURE 4	Sample Performance Curve	15
FIGURE 5	Absolute Pressure Limit Curves.....	16
FIGURE 6	Friction Loss for Water Flowing in 18 BWG Tubes R_T	19
FIGURE 7	Temperature Correction for Friction Loss in Tubes R_1	20
FIGURE 8	Water Box and Tube End Losses Single Pass Condensers R_E	21

CONTENTS (continued)

FIGURE 9	Water Box and Tube End Losses Two Pass Condensers R_E	22
FIGURE 10	Water Box and Tube End Losses Three Pass Condensers R_E	23
FIGURE 11	Water Box and Tube End Losses Four Pass Condensers R_E	24
FIGURE 12-13	Point Support – Pipe	38
FIGURE 14	Point Support – Double Clips	39
FIGURE 15	Point Support – Single Clips	39
FIGURE 16	Ribs	39
FIGURE 17	Design Nozzle Loading on Flat Plate	39
FIGURE 18	Spacing of Longitudinal Stiffeners	40
FIGURE 19	Cylindrical Condenser Shell Thickness	41
FIGURE 20	Stiffening Rings Required Moment of Inertia	42
FIGURE 21	Determination of L_U	43
FIGURE 22	Rib Supported Panels	44
FIGURE 23-24	Bolting of Flat Faced Flanges	46
FIGURE 25	Gasket Seating Pressure	46
FIGURE 26-28	Required Flange Thickness	46
FIGURE 29	Idealized Representation of Tubesheet Loading	50
FIGURE 30	Tubesheet Showing Beam-Strip Locations	51
FIGURE 31	Beam-Strip for a Tube Pattern of Triangular Pitch	51
FIGURE 32	Beam-Strip for a Laned Tube Pattern of Triangular Pitch	51
FIGURE 33	Section AA through Beam-Strip of Figure 32	51
FIGURE 34	Structural Model for Beam-Strip of Figure 33	51
FIGURE 35	Moment and Deflection Curves for Beam-Strip of Figure 32	51
FIGURE 36-42	Typical Condenser Welds	52
FIGURE 43	Weld Geometries	53
FIGURE 44	Welding Nomenclature	57
FIGURE 45	Standard Tolerances for Interfaces and Supports – English Units	64
FIGURE 45M	Standard Tolerances for Interfaces and Supports – Metric Units	65

FOREWORD

The Eleventh Edition of the “Standards for Steam Surface Condensers” represents another step in the Heat Exchange Institute’s continuing program to provide Standards that reflect the latest technological advancement in the field of condensing equipment.

The Eleventh Edition of “Standards for Steam Surface Condensers” has incorporated several new revisions since the Tenth Edition, such as new sample calculations for oxygen content and tubeside pressure drop, a new Section 4.6 on Condensate Temperature Depression, information on clad tubesheets, and several new Appendices. A listing of all HEI standards and condenser related technical articles is also listed on the inside cover of the standard for your convenience. Please visit the HEI website, www.heatexchange.org, for more information.

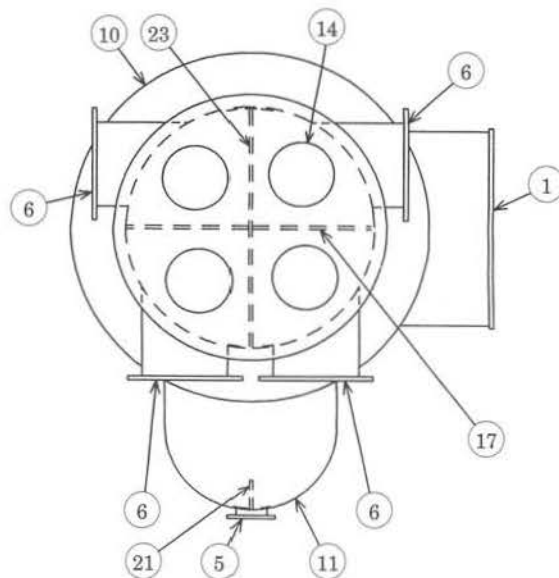
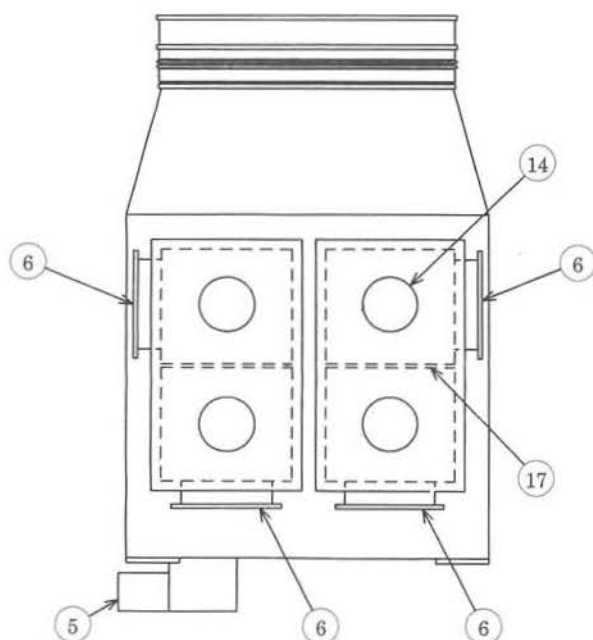
The Heat Exchange Institute anticipates a continuing program to extend and amplify the coverage presented in these Standards and this may require the periodic issuance of addenda to these Standards. As a result, users of these Standards should make sure that they are in possession of all such addenda by enquiry to the Heat Exchange Institute offices.

The Heat Exchange Institute solicits comments from all interested parties regarding areas where further treatment or more detailed treatment is desired or felt necessary. Contact the Institute at 1300 Sumner Ave., Cleveland, OH, 44115, or visit the HEI website at www.heatexchange.org.

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OPTIONAL SPRING SUPPORT
IN LIEU OF EXHAUST NECK
EXPANSION JOINT

OPTIONAL SPRING SUPPORT IN LIEU OF EXHAUST NECK EXPANSION JOINT



1. STEAM INLET CONNECTION
2. EXTENSION NECK
3. TRANSITION PIECE
4. VENT OUTLET CONNECTION
5. CONDENSATE OUTLET CONNECTION
6. CIRCULATING WATER INLET OR OUTLET
7. TUBES

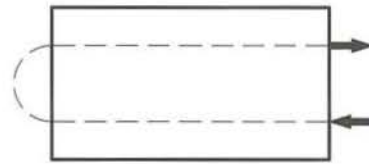
8. INLET-OUTLET WATER BOX
9. RETURN WATER BOX
10. SHELL
11. HOTWELL
12. TUBESHEETS
13. TUBE SUPPORT PLATES
14. ACCESS OR INSPECTION OPENINGS
15. SHELL EXPANSION JOINT

16. EXHAUST NECK EXPANSION JOINT
17. WATER BOX PASS PARTITION
18. SPRING SUPPORTS
19. SUPPORT FEET
20. SOLE PLATES
21. ANTI-VORTEX BAFFLE
22. WATER BOX COVER PLATE
23. WATER BOX DIVISION PLATE

TUBE AND SHELL CIRCUIT SCHEMATICS



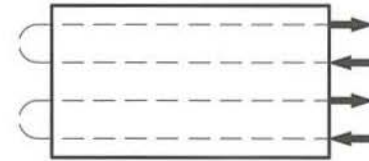
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- ONE PASS • NON DIVIDED



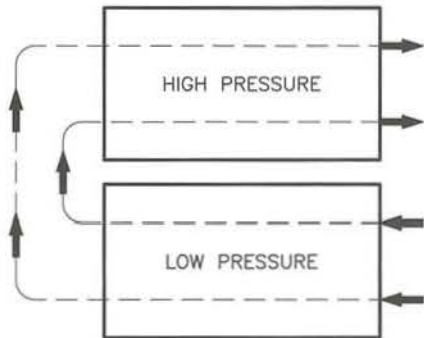
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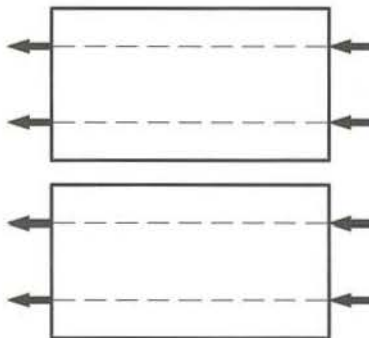
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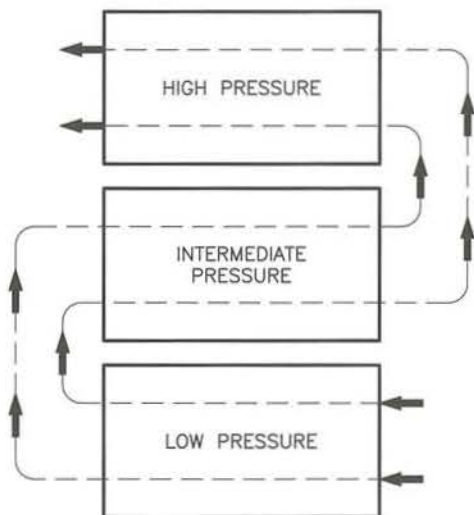
- ONE SHELL • SINGLE PRESSURE
- TWO PASS • DIVIDED



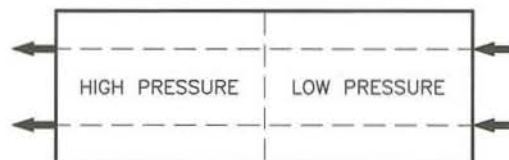
- TWO SHELLS • MULTI PRESSURE
- ONE PASS W/CROSSOVER • DIVIDED



- TWO SHELLS • SINGLE PRESSURE
- ONE PASS • DIVIDED



- THREE SHELLS • MULTI PRESSURE
- ONE PASS W/CROSSOVER • DIVIDED



- ONE SHELL • MULTI PRESSURE
- ONE PASS • DIVIDED

2.0 DEFINITIONS

2.1 Absolute Pressure

Absolute pressure is the pressure measured from absolute zero.

2.2 Circulating Water Velocity

Circulating water velocity is the average velocity of circulating water through the tubes.

2.3 Cleanliness Factor

Cleanliness factor is the ratio of the condenser heat transfer coefficient to the clean heat transfer coefficient.

2.4 Condensate Temperature Depression (Sub-Cooling)

Condensate depression is the difference between the condensing steam temperature and the temperature of the condensate in the hotwell.

2.5 Condenser Duty

Condenser duty consists of the net heat transferred to the circulating water. Unless otherwise specified, condenser duty is assumed to be the quantity of steam, in pounds per hour, entering the condenser multiplied by 950 Btu per pound for turbine service, or 1000 Btu per pound for engine service.

2.6 Condenser Heat Transfer Coefficient

Condenser heat transfer coefficient is the average rate of heat transfer from the steam to circulating water.

2.7 Condenser Pressure

Condenser pressure is the absolute static pressure maintained within the condenser shell at locations not greater than one foot from the first tube. The distribution of measurement points shall conform with ASME PTC 12.2, Steam Condensing Apparatus, latest edition.

2.8 Condensing Steam Temperature

Condensing steam temperature is the saturation temperature corresponding to the absolute static pressure of the steam.

2.9 Effective Surface

Effective surface is the total surface measured on the outside of the tubes between the inside surfaces of the tube sheets and includes internal and/or external air cooler surfaces.

2.10 Effective Tube Length

Effective tube length is the distance between inside surfaces of the tube sheets.

2.11 Hotwell Capacity

Hotwell capacity is condensate storage volume. The minimum recommended hotwell capacity is the volume sufficient to contain all of the condensate produced in the condenser in a period of one minute under conditions of design steam load.

2.12 Initial Temperature Difference

Initial temperature difference is the difference between the condensing steam temperature and the inlet circulating water temperature.

2.13 Logarithmic Mean Temperature Difference

Logarithmic mean temperature difference is the ratio of the temperature rise to the natural logarithm of the ratio of initial temperature difference to terminal temperature difference.

2.14 Static Pressure

Static pressure is the pressure of a fluid at rest.

2.15 Temperature Rise

Temperature rise is the difference between outlet and inlet circulating water temperatures.

2.16 Terminal Temperature Difference

Terminal temperature difference is the difference between the condensing steam temperature and the outlet circulating water temperature.

3.0 SYMBOLS AND UNITS

A_1	Inside cross-sectional area of a single tube	in ²	M_a, M_c, M_g, M_h	Moments	in-lb
A_D	Minimum Required Flow Area	in ²	M_R	Resultant Moment	in-lb
A_E	Turbine Exhaust Flow Area	ft ²	MW_{NC}	Molecular Weight of Non-Condensable Gas	—
A_S	Surface Area	ft ²	MW_{wv}	Molecular Weight of Water Vapor	—
A_T	Inside Tube Flow Area	ft ² /pass	N	Number of Bolts	—
BWG	Tube Gauge	—	N_p	Number of Tube Side Passes	—
C	Geometric Constant	—	N_{pp}	Number of Tubes Per Pass	—
CA	Corrosion Allowance	in	N_T	Total number of Tubes	—
C_C	Column Slenderness Ratio	—	P	Beam Load	lb
CFM	Gas Flow	ft ³ /min	P_A	Relieving Pressure	psia
C_P	Specific Heat	Btu/lb•°F	P_C	Column Load	lb
D	Tube Outside Diameter	in	P_D	Design Pressure	psig
D_1	Tube Inside Diameter	in	P_E	End Load on Beam Strip	lb
D_P	Pipe Diameter	in	P_G	Pressure Required to Compress Gasket	psi
E	Modulus of Elasticity	psi	P_h	Hydrostatic Test Pressure	psig
F	Force	lb	P_S	Saturation Pressure	in HgA
F_C	Correction Factor for Cleanliness	—	P_T	Test Pressure	psig
F_M	Correction Factor for Material and Gauge	—	P_t	Absolute "Total" Pressure at Condenser Vent Outlet	in HgA
F_R	Resultant Force	lb	P_w	Absolute "Water Vapor" Pressure Corresponding to Temperature at Condenser Vent Outlet	in HgA
FS	Factor of Safety	—	P_1	Saturation Pressure at Sonic Strata	psia
F_W	Correction Factor for Water	—	Q	Heat Duty	Btu/hr
F_1, F_2, F_3	Force Loading	lb/in	R	Radius	in
G	Cutoff Point	in HgA	R_E	Friction Loss (Water Box and Tube Ends)	ft of water
H	Enthalpy	Btu/lb	R_T	Friction Loss (Tubes)	ft of water/ ft length
I	Moment of Inertia	in ⁴	R_{TT}	Friction Loss (Total)	ft of water
ITD	Initial Temperature Difference	°F	R_1	Correction Factor (Water Temperature)	—
J	Zero Load Back Pressure	in HgA	R_2	Correction Factor (Tube O.D. and Gauge)	—
K	Column End Condition Factor	—	SCFM	Gas Flow at Standard Conditions of Pressure and Temperature	ft ³ /min
K_D	Discontinuity Factor (Geometry Dependent)	—	S	Stress	psi
K_1	Pressure, O.D. and Gauge Correction Factor	—	S_A	Allowable Stress	psi
K_2	O.D. and Pitch Correction Factor	—	S_{BOLTS}	Total Bolt Stress	psi
K_3	Material Correction Factor	—	S_G	Specific Gravity	—
K_4	Flow Coefficient	—	S_U	Ultimate Strength	psi
L_C	Column Height (Unsupported)	ft	S_Y	Yield Strength	psi
L_E	Effective Tube Length	ft	T	Temperature	°F
LMTD	Logarithmic Mean Temperature Difference	°F	TD	Temperature of depression	°F
L_n	Natural Logarithm	—	TR	Temperature Rise	°F
L_b	Beam Length	in	TTD	Terminal Temperature Difference	°F
L_U	Uncorrected Support Plate Spacing	in	T_1	Inlet Water Temperature	°F
L_S	Shell Unsupported Length	in			
L_{SP}	Support Plate Span	in			
L_{SP1}	Intermediate Support Plate Spacing	in			
L_{SP2}	End Support Plate Spacing	in			
L_T	Total Tube Length	ft			
L_1	Tube Length Between Tubesheet and First Support Plate	in			

T_2	Outlet Water Temperature	°F	a_c	Metal Area of Column	in ²
T_S	Saturation Temperature (Steam)	°F	a_G	Area of Gasket	in ²
T_{CHP}	Temperature of Condensate Leaving High Pressure Shell	°F	a_M	Area	in ²
T_{SHP}	Saturation Temperature Higher/Highest Pressure Shell	°F	a_F	Tube Flow Area	in ²
T_{SIP}	Saturation Temperature Intermediate Pressure Shell	°F	a_s	External Tube Surface Area Per Unit Length	ft ² /ft
T_{SLP}	Saturation Temperature Lower/Lowest Pressure Shell	°F	cc/L	Cubic Centimeter Per Liter	—
U	Heat Transfer Coefficient	Btu/hr•ft ² •°F	d_1	Diameter	in
U_1	Uncorrected Heat Transfer Coefficient	Btu/hr•ft ² •°F	d_H	Tube Hole Diameter	in
V_S	Velocity of Steam	ft/sec	e	Efficiency Factor (Welds)	—
V_W	Velocity of Water	ft/sec	e_s	Ligament Efficiency	—
W	Pounds of Water Vapor per Pound of Noncondensable Gas	—	f_G, f_H	Correction Factors	—
W_e	Weight Per Unit Length of the Tube	lb/in	g	Acceleration of Gravity	ft/sec ²
W_m	Weight Per Unit Length of the Tube Material	lb/in	h	Tube Ligament	in
W_t	Weight Per Unit Length of the Tube Side Fluid	lb/in	k	Thermal Conductivity	Btu/hr•ft ² •°F/ft
W_S	Steam Flow	lb/hr	k_s, k_T	Spring Constants	lb/in
W_G	Water Flow	gpm	n	Integer	—
W_{LP}	Total Fluids Entering Lower/Lowest Pressure Condenser Shell	lb/hr	p	Tube Pitch	in
W_{IP}	Total Fluids Entering Intermediate Pressure Condenser Shell	lb/hr	ppb	Parts per Billion	—
W_{HP}	Total Fluids Entering Higher/Highest Pressure Condenser Shell	lb/hr	r	Radius of Gyration	in
Z	Section Modulus	in ³	t_p, t_R	Thickness (No Corrosion Included)	in
a_{BOLTS}	Tensile Area of Bolts	in ²	t_s	Thickness of Support Plate	in
			t_w	Tube Wall Thickness	in
			v	Specific Volume	ft ³ /lb
			w	Width	in
			a_1, b_1, c_1	Linear Dimensions and Measure	in
			g_1, h_1, l_1	Linear Dimensions and Measure	in
			e_1, e_2	Linear Dimensions and Measure	in
			x_1, y_1	Linear Dimensions and Measure	in
			α	Coefficient of Thermal Expansion	in/in-°F
			ρ	Density	lb/in ³
			v	Poisson's Ratio	—
			ψ	Reduced Geometry Factor	—
			δ	Deflection	in

4.0 CONDENSER PERFORMANCE

4.1 General Considerations

4.1.1 It is recognized that the performance of a condenser cannot be exactly predicted under each one of a number of possible operating conditions. Consequently, curves or tabulations of condenser performance data are only approximate, except for one specific condition termed the "Design Point." Performance checks should be made only when the system has been stabilized and reproducible values are attainable.

4.1.2 Commercial operating conditions are recognized as involving uncontrollable variations in air and gas tightness of the condenser and its related system under vacuum. These variations, while negligible under some conditions, render the exact prediction of condenser performance impractical where the terminal temperature difference is less than 5°F. In addition, terminal temperature differences of less than 5°F are not considered sufficient to give determinative and predictable heat transfer performance and are not recommended.

4.1.3 Condenser tube water velocities under 3 feet per second do not build up resistance sufficient to insure a uniform quantity of water through all the tubes; therefore, condenser performance under such conditions cannot be exactly predicted and such predictions are not recommended.

4.1.4 As a general rule and within the degree of accuracy expected in steam condensers, the effect of sea or brackish water as opposed to fresh water is comparatively insignificant with respect to performance. If environmental laws require strict limitation on the water temperature discharged from condensers to natural sea water or brackish water sources, it may be necessary to allow for the effect of such waters on the circulating water temperature rise through condensers in borderline cases. In instances where this is necessary or where it is otherwise considered necessary, the following allowance for corrected specific heat and specific gravity of such circulating water may be made. The Purchaser shall furnish specific weight flow or specific gravity and specific heat.

$$W_G = \frac{Q}{500 \times S_G \times C_P \times TR}$$

$$V_w = \frac{Q}{A_T \times 3600 \times 62.4 \times S_G \times C_P \times TR}$$

4.1.5 Due to its effect on condenser performance, the location of heaters and/or extraction piping should be subject to the condenser Manufacturer's approval after the turbine flow distribution diagram has been made available.

4.1.6 Performance information as generated from these standards is based on venting equipment having a capacity at one inch mercury absolute pressure of

not less than that listed in Section 6, and the actual air and non-condensibles being removed from the system not exceeding 50% of those values.

4.1.7 It should be recognized that at reduced duties, a terminal temperature difference less than 5°F will unpredictably affect condenser performance.

4.1.8 HEI has established a condenser rating program, for further information please visit the HEI website.

4.2 Heat Transfer Rates

4.2.1 The design of a steam surface condenser must consider the effects of noncondensable gases which are present in the condenser, pressure drop of the steam as it flows around and through the tube bundle, and tube inundation as condensate falls through the bundle. Due to these effects, the heat transfer coefficient of a typical, commercial operating condenser is less than that attainable in laboratory tests.

The heat transfer rates published by the HEI are **OVERALL TUBE BUNDLE "U" VALUES** to be obtained by the condenser under actual operating conditions and not single tube "U" values. Because these values take into account parameters other than the basic heat transfer across the wall of the tube, they are not meant to be used by designers as specific individual tube "U" values.

The Heat Exchange Institute has conducted tests for the purpose of arriving at heat transfer coefficients for surface condensers. The following is the Heat Exchange Institute's method for calculating condenser heat transfer coefficients. Other methods of calculating heat transfer coefficients are available.

This method includes an allotment for the steamside effects described above. It is the responsibility of the condenser designer to develop tube bundle and shell configurations which result in the heat transfer coefficients calculated by this Standard.

The general heat transfer equations are:

$$Q = U \times A_s \times LMTD$$

$$Q = (H_{\text{steam}} - H_{\text{condensate}}) \times W_s + \text{Auxiliary heat load}$$

$$U = U_1 \times F_w \times F_M \times F_C$$

U_1 - Figure 1 or Table 1

F_w - Figure 2 or Table 2

F_M - Table 3

F_C - Cleanliness Factor

$$LMTD = \frac{TR}{\ln \left(\frac{ITD}{TTD} \right)}$$

$$TR = T_2 - T_1$$

$$ITD = T_s - T_1$$

$$TTD = T_s - T_2$$

4.2.2 Table 1 and Figure 1 are based on clean 18 BWG Admiralty metal tubes with 70°F inlet circulating water temperature.

4.2.3 For inlet circulating water temperatures other than 70°F, the basic heat transfer coefficients should be multiplied by the corresponding design correction factors shown in Figure 2 or Table 2.

4.2.4 For any tube gauge or material other than 18 BWG Admiralty, basic heat transfer coefficients should be multiplied by the appropriate correction factors from Table 3.

4.2.5 In actual operation, both the circulating water and condensing steam will produce heat transfer resistance films on the tube surfaces which will have

characteristics related to the type of fluid. A design cleanliness factor should be selected by the Purchaser that suitably reflects the probable operating condition the tubes will experience in service. Non-copper bearing tube materials are more susceptible to bio-fouling than tubes with high copper content.

U_1
UNCORRECTED HEAT TRANSFER COEFFICIENTS BTU/hr x ft² x F

TUBE DIAMETER, in	TUBE VELOCITY, ft/sec								
	3.0	3.5	4.0	4.5	5.0	5.5	6.0	6.5	7.0
0.625 & 0.75	462.5	499.5	534.0	566.4	597.0	626.2	654.0	680.7	706.4
0.875 & 1.00	455.0	492.0	526.0	557.9	588.1	616.8	644.2	670.5	695.8
1.125 & 1.25	448.6	484.5	518.0	549.4	579.1	607.4	634.4	660.3	685.2
1.375 & 1.50	441.7	477.1	510.0	540.9	570.2	598.0	624.6	650.1	674.7
1.625 & 1.75	434.7	469.6	502.0	532.5	561.3	588.6	614.8	639.9	664.1
1.875 & 2.00	427.8	462.1	494.0	524.0	552.3	579.8	605.0	629.7	653.5

TUBE DIAMETER, in	TUBE VELOCITY, ft/sec									
	7.5	8.0	8.5	9.0	9.5	10.0	10.5	11.0	11.5	12.0
0.625 & 0.75	731.2	755.2	775.5	795.3	814.1	831.9	848.9	865.2	880.7	895.6
0.875 & 1.00	720.3	743.9	763.9	783.2	801.6	819.0	835.6	851.5	866.6	881.1
1.125 & 1.25	709.3	732.6	752.0	770.7	788.4	805.3	821.4	836.7	851.3	865.3
1.375 & 1.50	698.3	721.2	740.4	758.7	776.1	792.6	808.3	823.2	837.5	851.2
1.625 & 1.75	687.4	709.9	727.8	745.7	762.7	778.8	794.1	808.8	822.7	836.0
1.875 & 2.00	676.4	698.6	716.8	734.4	751.0	766.8	781.8	796.2	809.8	822.9

Table 1

U_1 UNCORRECTED HEAT TRANSFER COEFFICIENTS

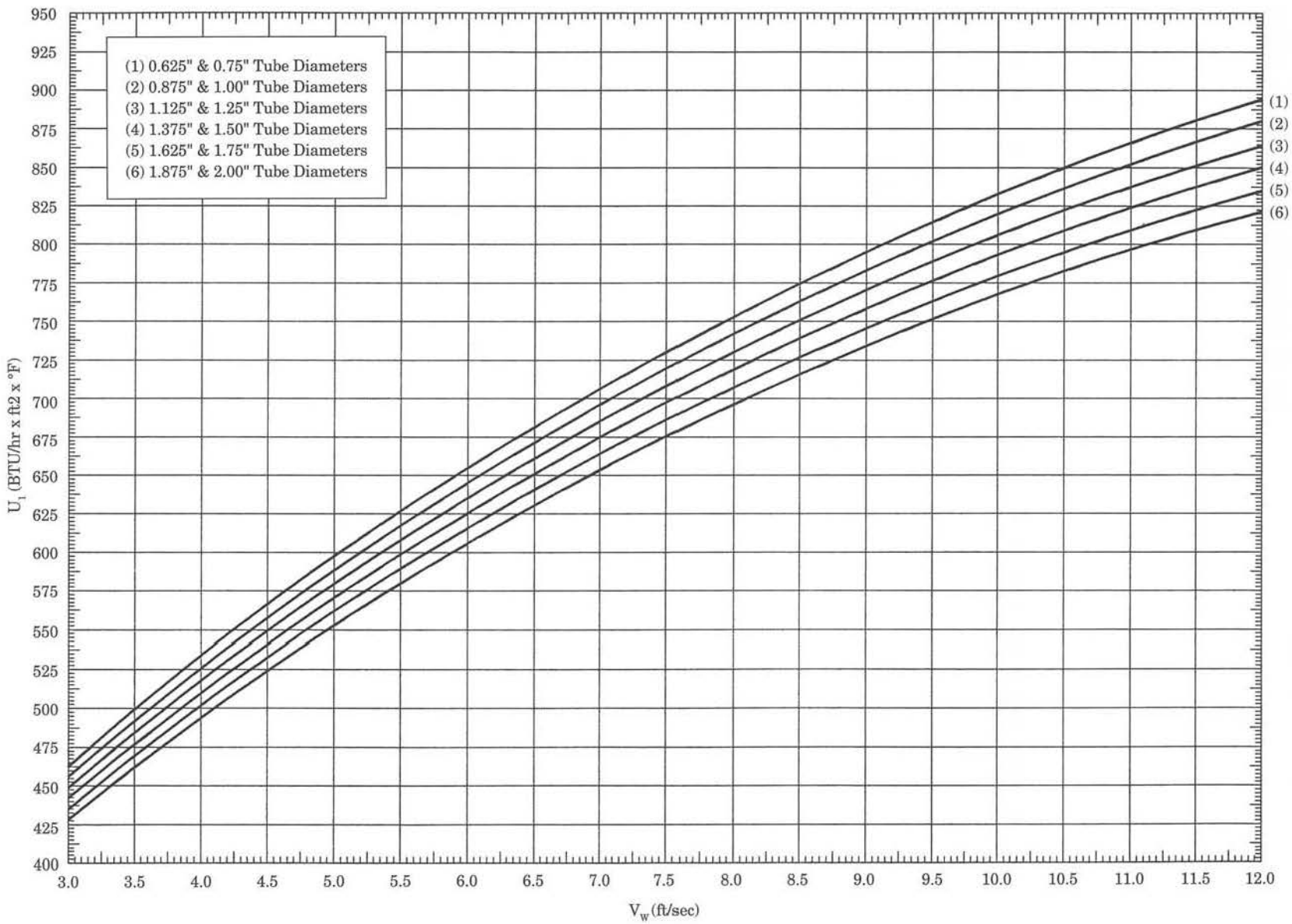


Figure 1
 V_w Velocity through tubes, ft/sec

F_w
INLET WATER TEMPERATURE CORRECTON FACTOR

Inlet Water °F	F _w	Inlet Water °F	F _w	Inlet Water °F	F _w
30	0.650	60	0.923	90	1.075
31	0.659	61	0.932	91	1.078
32	0.669	62	0.941	92	1.080
33	0.678	63	0.950	93	1.083
34	0.687	64	0.959	94	1.085
35	0.696	65	0.968	95	1.088
36	0.706	66	0.975	96	1.090
37	0.715	67	0.982	97	1.092
38	0.724	68	0.989	98	1.095
39	0.733	69	0.994	99	1.097
40	0.743	70	1.000	100	1.100
41	0.752	71	1.005	101	1.103
42	0.761	72	1.010	102	1.105
43	0.770	73	1.015	103	1.108
44	0.780	74	1.020	104	1.110
45	0.789	75	1.025	105	1.113
46	0.798	76	1.029	106	1.115
47	0.807	77	1.033	107	1.117
48	0.816	78	1.037	108	1.119
49	0.825	79	1.041	109	1.121
50	0.834	80	1.045	110	1.123
51	0.843	81	1.048	111	1.125
52	0.852	82	1.051	112	1.127
53	0.861	83	1.054	113	1.129
54	0.870	84	1.057	114	1.131
55	0.879	85	1.060	115	1.133
56	0.888	86	1.063	116	1.135
57	0.897	87	1.066	117	1.137
58	0.905	88	1.069	118	1.139
59	0.914	89	1.072	119	1.141
				120	1.143

Table 2

F_w
INLET WATER TEMPERATURE CORRECTION FACTOR

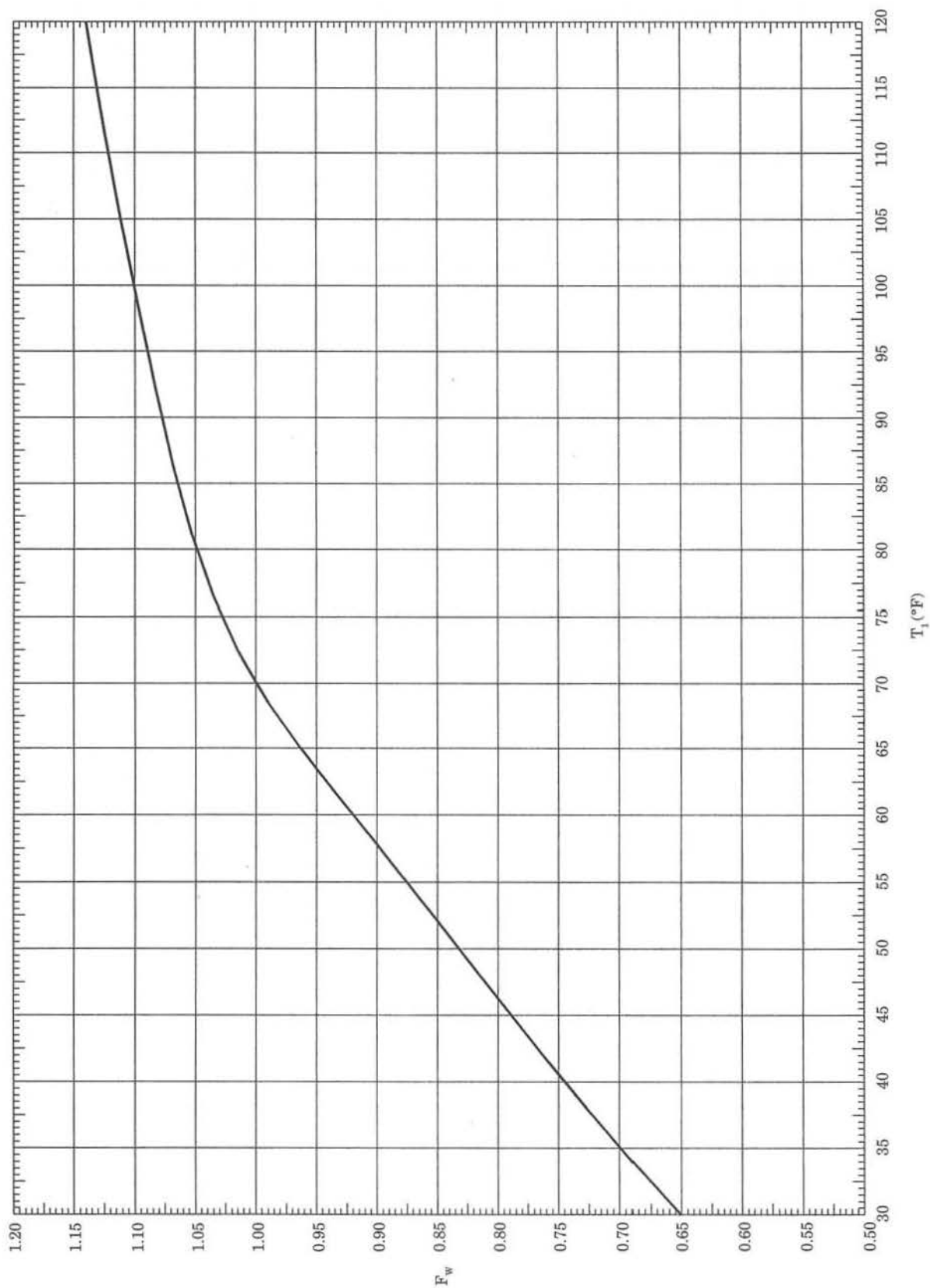


Figure 2

F_M TUBE MATERIAL AND GAUGE CORRECTION FACTORS

Tube Material	k	Tube Wall Gauge (BWG) & Wall Thickness t _w (in)								
		25	24	23	22	20	18	16	14	12
		0.020	0.022	0.025	0.028	0.035	0.049	0.065	0.083	0.109
Cu Fe 194	150	1.042	1.041	1.039	1.038	1.034	1.028	1.020	1.010	0.997
Arsenical Cu	112	1.038	1.037	1.035	1.033	1.029	1.020	1.010	0.997	0.979
Admiralty	64	1.029	1.027	1.024	1.021	1.013	0.998	0.981	0.961	0.932
Al Brass	58	1.027	1.025	1.021	1.018	1.010	0.993	0.974	0.952	0.921
Al Bronze	46	1.021	1.018	1.014	1.009	0.999	0.979	0.956	0.930	0.892
Carbon Steel	27.5	1.002	0.998	0.990	0.983	0.967	0.936	0.901	0.863	0.810
Cu Ni 90-10	26	1.000	0.995	0.987	0.980	0.963	0.930	0.893	0.854	0.800
Cu Ni 70-30	17	0.974	0.967	0.957	0.946	0.922	0.876	0.828	0.777	0.710
SS (UNS S43035)	14.0	0.959	0.951	0.938	0.926	0.898	0.846	0.792	0.736	0.664
Titanium Grades 1 & 2	12.7	0.951	0.942	0.928	0.915	0.885	0.830	0.772	0.714	0.640
SS (UNS S44660)	10.5	0.932	0.922	0.906	0.891	0.857	0.795	0.732	0.669	0.591
SS (UNS S44735)	10.1	0.928	0.917	0.901	0.886	0.851	0.787	0.723	0.659	0.581
SS TP 304	8.6	0.910	0.897	0.879	0.862	0.823	0.754	0.685	0.619	0.539
SS TP 316 / 317	8.2	0.904	0.891	0.872	0.854	0.815	0.744	0.674	0.607	0.527
SS (UNS N08367)	6.8	0.879	0.864	0.843	0.823	0.779	0.702	0.628	0.558	0.477

Table 3

4.2.6 Sample Thermal Calculation

The following is a sample thermal calculation using these methods. Based on the sample data provided

below, the required surface area of a condenser will be calculated.

Design Information:

Condenser Pressure, P_s	1.177" H _g (a)
Condenser Temperature, T_s	84.01 °F
Condenser Heat Duty, Q	1032.8 MM BTU/hr
Turbine Exhaust Steam Flow Rate, W_s	1,064,000 lb/hr
Circulating Water Flow Rate, W_G	253,900 GPM
Circulating Water Inlet Temperature, T_1	60.0 °F
Tube Water Velocity, V_w	9.0 ft/s
Cleanliness Factor, F_c	0.80
Tube O.D., D	1.00 inch
Tube I.D., D_i	0.944 inch (22 BWG Tubes)
Tube Material	A249-316 (Stainless Steel 316)
Circulating Water Type	Fresh Water
Circulating Water Density, ρ	62.4 lb/ft ³
Circulating Water Specific Heat, C_p	1.00 BTU /lb °F

Determine Circulating Water Outlet Temperature:

$$T_2 = \frac{Q}{W_G \cdot \rho \cdot C_p} + T_1 = \frac{1032.8 \cdot 10^6 \cdot \text{BTU/hr}}{\left(\frac{253,900 \cdot \text{Gal}}{\text{min}}\right) \cdot \left(\frac{60 \cdot \text{min}}{1 \cdot \text{hr}}\right) \cdot \left(\frac{1 \cdot \text{ft}^3}{7.48 \cdot \text{Gal}}\right) \cdot \left(\frac{62.4 \cdot \text{lb}}{1 \cdot \text{ft}^3}\right) \cdot \left(\frac{1.0 \cdot \text{BTU}}{\text{lb} \cdot ^\circ\text{F}}\right)} + 60^\circ\text{F} = 68.1^\circ\text{F}$$

Determine the Log Mean Temperature Difference:

$$LMTD = \frac{TR}{\ln \left(\frac{ITD}{TTD} \right)} = \frac{(T_2 - T_1)}{\ln \left(\frac{T_s - T_1}{T_s - T_2} \right)} = \frac{(68.1^\circ\text{F} - 60.0^\circ\text{F})}{\ln \left(\frac{84.01^\circ\text{F} - 60.0^\circ\text{F}}{84.01^\circ\text{F} - 68.1^\circ\text{F}} \right)} = 19.7^\circ\text{F}$$

Calculate the Overall Heat Transfer Coefficient:

From Section 4.2;

$U_1 = 783.2 \text{ BTU/ft}^2 \cdot ^\circ\text{F} \cdot \text{hr}$ (Table 1, Page 7)

$F_w = 0.923$ (Table 2, Page 9)

$F_M = 0.854$ (Table 3, Page 11)

$$U = U_1 \cdot F_w \cdot F_M \cdot F_c = \frac{783.2 \cdot \text{BTU}}{\text{ft}^2 \cdot ^\circ\text{F} \cdot \text{hr}} \cdot 0.923 \cdot 0.854 \cdot 0.80 = 493.9 \cdot \frac{\text{BTU}}{\text{ft}^2 \cdot ^\circ\text{F} \cdot \text{hr}}$$

Calculate the surface area of the condenser:

$$A_s = \frac{Q}{U \cdot LMTD} = \frac{1032.8 \cdot 10^6 \cdot \text{BTU/hr}}{\frac{493.9 \cdot \text{BTU}}{\text{ft}^2 \cdot ^\circ\text{F} \cdot \text{hr}} \cdot 19.7^\circ\text{F}} = 106,148 \cdot \text{ft}^2$$

4.3 Oxygen Content of Condensate

4.3.1 Under practical operating conditions, the condenser can be expected to produce condensate with an oxygen content not exceeding 42 parts per billion. With certain conditions of stable operation and suitable construction, as the application may require, an oxygen content not exceeding 14 parts per billion or as low as 7 parts per billion may be obtained as follows:

4.3.1.1 Condenser pressures should not be lower than the values shown on the curves in Figure 3, Curve A for 7 parts per billion and Curve B for 14 parts per billion.

4.3.1.2 The ratio of the actual non-condensable load removed from the system to the design capacity of the venting equipment should be no greater than the values in Table 4.

4.3.1.3 There should be zero air leakage directly into the condensate below the condensate level in the hotwell. The arrangement and location of all entrance points into the condenser for water vapor or other gases should be subject to the approval of the Manufacturer.

Examples of the potential sources of air are as follows:

4.3.1.3.1 Leakage into the vacuum side of the system through leaks in welds, packing glands, gauge glasses, salinity cells, instrumentation leads, etc.

4.3.1.3.2 Low pressure heater condensate drains and vents, particularly when operating below atmospheric pressure.

4.3.1.3.3 Make up, which is usually saturated with oxygen.

4.3.1.3.4 Condensate surge tank, when utilized in closed cycles.

4.3.1.4 Total water introduced into the condenser shell at a temperature lower than the inlet steam temperature should not be more than 5% of the steam being condensed for 14ppb or more than 3% for 7ppb.

4.3.2 Where condensate from processing systems and/or cogeneration systems is introduced to the condenser, it shall be assured that the oxygen content of the returned condensate is no greater than that specified for hotwell condensate. If this is not the case, special internal deaerating provisions may be required and/or returns shall be deaerated externally prior to being returned to the condenser. The specific oxygen level in returning condensate and the quantity of condensate being returned must be specified for the Manufacturer's considerations.

4.3.3 Sample Oxygen Content Calculation

In order to determine the oxygen content of the condensate at different operating cases (off design operating cases), the following procedure shall be followed:

Step 1: Determine the condenser shellside pressure based on the circulating water inlet temperature and condenser duty. This information may be found by using the performance curves provided by the manufacturer.

Step 2: Using Figure 3 from Page 14 of the HEI Standards for Steam Surface Condensers, locate the circulating water temperature on the horizontal axis.

Step 3: Once this temperature is found, move vertically (straight) up the figure until you intersect Curve "B".

Step 4: Move horizontally to the left to find the corresponding pressure (in inches of Hg).

Step 5: In order to achieve an oxygen content of 14 PPB the actual condenser shellside operating pressure

VENTING CAPACITY AND OXYGEN CONTENT

Venting Equipment Design Capacities (SCFM) ^(a)	Actual Load/ Design Capacity Ratio ^(b)	Expected Oxygen Content In Condensate ppb (cc/L)
0-20	0.50	42 (0.03)
	0.35	14 (0.01)
	0.25	7 (0.005)
20-40	0.50	42 (0.03)
	0.24	14 (0.01)
	0.15	7 (0.005)
Greater than 40	See note (c)	42 (0.03)
		14 (0.01)
		7 (0.005)

Notes:

a. The design capacity of the venting equipment should be in accordance with Section 6.

b. These ratios are for venting equipment rated at 1 in. HgA. The venting equipment in operation should also have a minimum capacity of 40% of the free dry air (stated in Section 6) at 0.5 in. HgA suction pressure and a temperature of 51.3°F when operation is lower than 1 in. HgA.

c. For venting equipment with design capacity exceeding 40 SCFM, the non-condensibles removed should not exceed the following definitive values:

20 SCFM for 42 ppb

10 SCFM for 14 ppb

6 SCFM for 7 ppb

Table 4

ABSOLUTE PRESSURE LIMIT CURVES FOR OXYGEN CONTENT

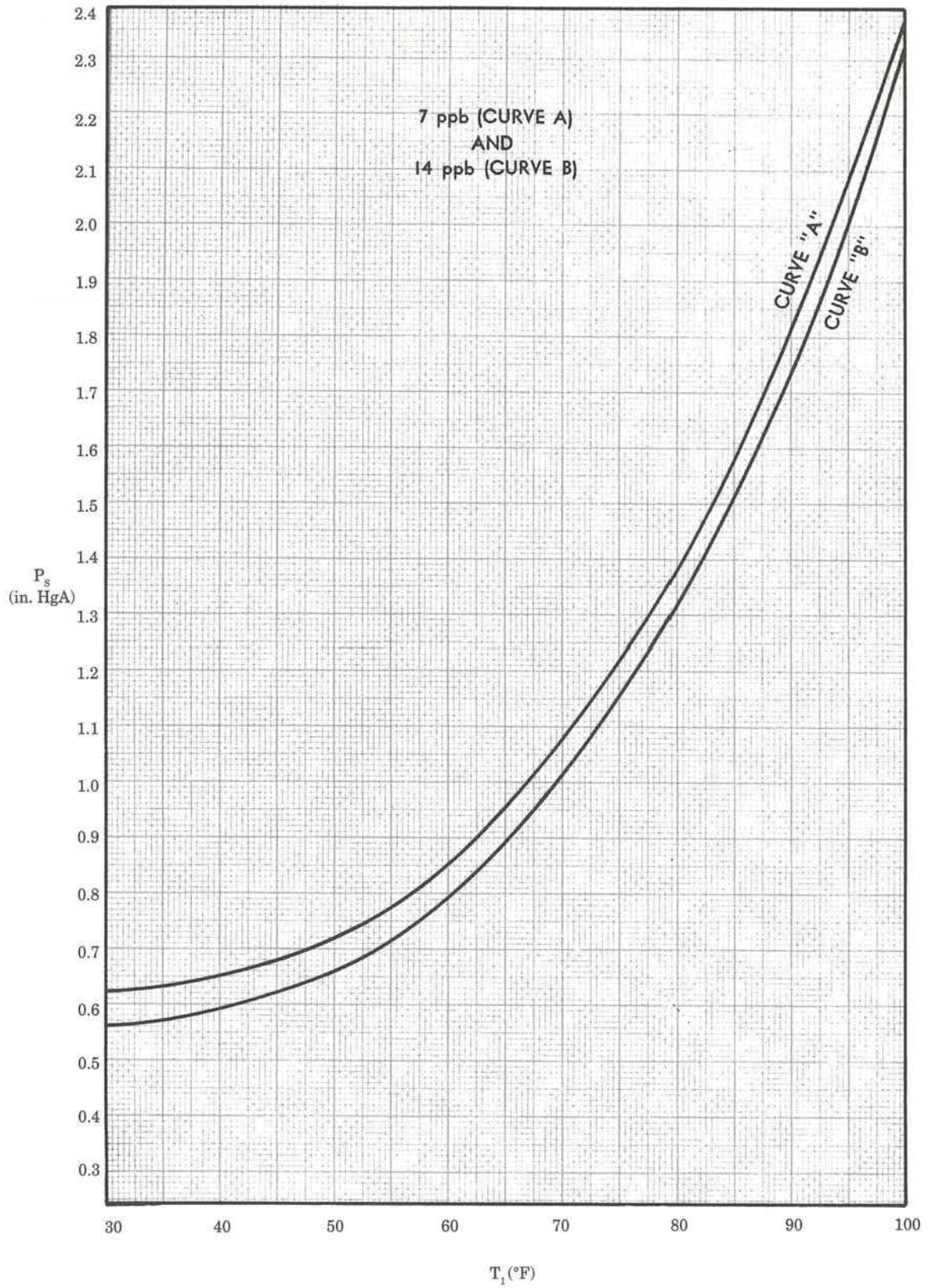


Figure 3

(found in Step 1) must be equal or greater than the pressure found in Step 4.

Step 6: The above step can be repeated using Curve "A" to determine if the oxygen content of the condensate is 7 PPB.

The oxygen content values shown in Figure 3 are only valid if the provisions from Section 4.3.1.1 to 4.3.1.4 maintained.

4.3.4 In the case of nuclear power cycles in which additional non-condensable gases such as oxygen and hydrogen are present in the condenser, the expected oxygen content of the condensate will be appreciably higher than those power cycles where air is the only non-condensable present in the condenser. The Heat Exchange Institute has conducted a field survey of a number of condensers for Boiling Water Reactor power plants and has reached the conclusion that condensate oxygen levels of 10-50 ppb over a fairly wide range of operation are to be expected with this type of plant.

4.3.5 It is recognized that a subcooled liquid has greater potential for dissolving gases that might be present in the hotwell reheat area. This factor increases the importance of eliminating sources of noncondensable gases in the hotwell area (Par. 4.3.1.3). The restrictions of paragraph 4.3.1.4 are not applicable to condensate cascaded from the lower pressure shell to higher pressure shell since this condensate has been effectively deaerated in its respective shell prior to being cascaded.

4.4 Performance Curves

4.4.1 Having established the overall heat transfer coefficient for a given condenser, it is then possible to plot performance curves showing absolute pressures for varying condenser duties and inlet circulating water temperatures. A sample performance curve is shown (Figure 4).

4.4.2 It is recognized that at lower heat duties the curves must be modified due to the limitations of the venting equipment. This modification begins at Point J and proceeds as a straight line to Point G. Point J is determined from Figure 5, (Curve A) and is commonly referred to as the cut-off point. Point G is the minimum absolute pressure zero duty and is provided by Figure 5, (Curve B).

4.4.3 It should be recognized that a terminal temperature difference less than 5°F will unpredictably affect condenser performance.

4.5 Hydraulic Loss - Circulating Water Pressure Loss

The circulating water pressure loss through the condenser is calculated using the following equations.

$$R_{TT} = L_T (R_T \times R_2 \times R_1) + \Sigma R_E$$

R_{TT} = Total Loss

L_T = Tube Length

*Multiply by number of tube passes.

R_T = Tube Loss, Figure 6

Or use:

$$R_2 \times R_T = \frac{0.00642 V_w^{1.75}}{D_i^{1.25}}$$

SAMPLE PERFORMANCE CURVE

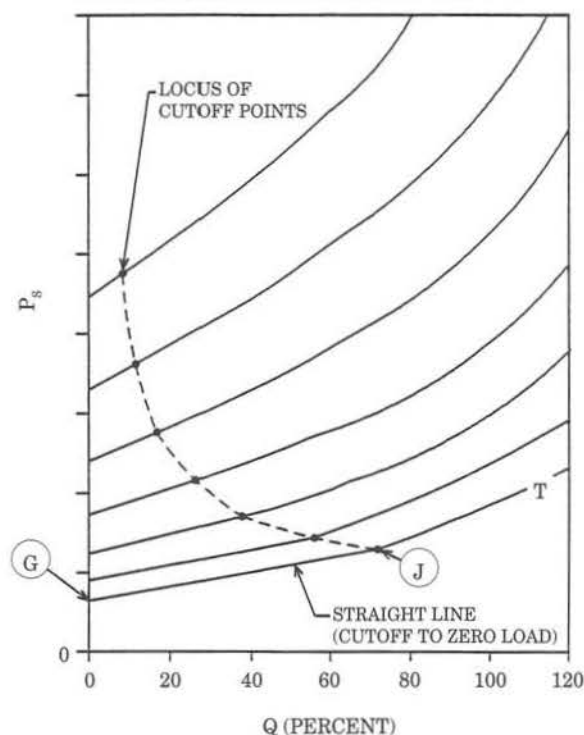


Figure 4

(Note: Correct V_w for Average Water Temperature)

R_1 = Temperature Correction Factor, Figure 7

R_2 = Tube O.D. & Gauge Correction Factor, Table 5

R_E^{**} = Water Box and Tube End Losses

**See Figures 8, 9, 10, and 11 for appropriate number of water passes.

Figures 8 and 9 cover the head losses to be expected in waterboxes and tube entrances and exits of single pass and two pass condensers, respectively. For single pass condenser, the inlet and outlet waterbox losses should be determined from the curves in Figure 8 using the actual nozzle water velocity in each case. The tube inlet and outlet losses are combined in one curve in Figure 8 and the value for these losses should be taken directly from the curve using the actual water velocity in the tubes.

For two pass condensers, the above procedure should be followed using the curves of Figure 9. It should be noted that the tube inlet and outlet loss is double that of Figure 8 and the value obtained therefrom should only be used once in the head loss computations. Similar procedures should be used for three and four pass condensers.

The values given by Figure 6 are based on a clean 18 BWG tube with an average cooling water inlet temperature of 70°F with a 15°F temperature rise. Factors should be adjusted using this as a base.

ABSOLUTE PRESSURE LIMIT CURVES

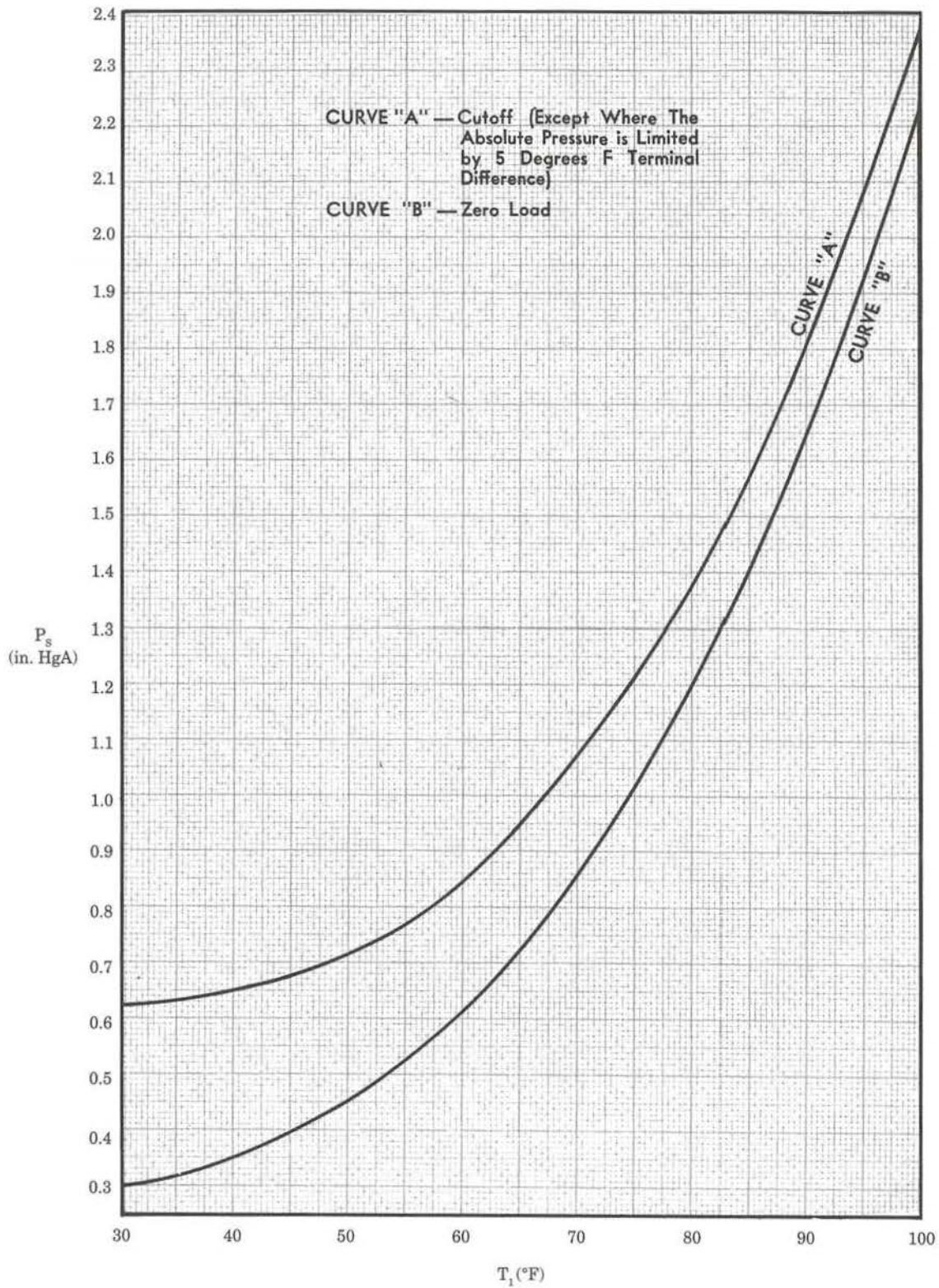


Figure 5

4.5.1 Sample Tubeside Pressure Drop Calculation

Design Information:

Circulating Water Type ¹	Fresh Water
Circulating Water Inlet Temperature ¹ , T _i	60.0 °F
Circulating Water Outlet Temperature	68.1 °F
Circulating Water Flow Rate ¹ , W _G	253,900 GPM
Tube Water Velocity ¹ , V _w	9.0 ft/s
Tube O.D. ¹ , D	1.00 inch
Tube I.D. ¹ , D _i	0.944 inch (22 BWG Tubes)
Surface Area ¹ , A _s	106,148 ft ²
Number of Tube Side Passes ² , N _p	1

Determine the Internal Cross-Sectional Area of a single Condenser Tube:

A_i See Appendix F, Mechanical Characteristics of Tubing

Determine the External Surface Area (per length) for a Condenser Tube a_s (See Appendix F)

Determine the Inside Tube Flow Area For The Specified Flow And Velocity:

$$A_T = \frac{W_G}{V_w} \cdot \left(\frac{.134 \cdot ft^3}{1 \cdot gal} \right) \cdot \left(\frac{1 \cdot min}{60 \cdot s} \right) = \frac{253,900 \cdot GPM}{9.0 \cdot ft/s} \cdot \left(\frac{.134 \cdot ft^3}{1 \cdot gal} \right) \cdot \left(\frac{1 \cdot min}{60 \cdot s} \right) = 62.863 \cdot ft^2$$

Determine the Total Number of Tubes Per Pass and Total Number of Tubes:

$$N_{pp} = \frac{A_T}{a_i} \cdot \left(\frac{144 \cdot in^2}{1 \cdot ft^2} \right) = \frac{62.863 \cdot ft^2}{.700 \cdot in^2} \cdot \left(\frac{144 \cdot in^2}{1 \cdot ft^2} \right) = 12,931$$

$$N_T = N_{pp} \cdot N_p = 12,931 \cdot 1 = 12,931$$

Determine the Length of the Tubes:

$$L_T = \frac{A_s}{N_p \cdot N_{pp} \cdot a_s} = \frac{106,148 \cdot ft^2}{1 \cdot 12,931 \cdot (.2618 \cdot ft^2)} = 31.355 \cdot ft$$

Determine the total Head Loss of tube and waterboxes.

$$R_1 = 1.042 \text{ (Figure 7, Page 20)}$$

$$R_2 = .94 \text{ (Table 5, Page 18)}$$

$$\Sigma R_E = 3.0 \text{ ft. of water (0.39 + 1.24 + 1.41) (Figure 8, Page 21)}$$

$$R_T = .34 \cdot \frac{\text{ft of water}}{\text{ft of Length}} \text{ (Figure 6, Page 19)}$$

$$R_{TT} = L_T \cdot (R_T \times R_2 \times R_1) + \Sigma R_E = 31.355 \cdot ft \cdot \left(.34 \cdot \frac{\text{ft of water}}{\text{ft of Length}} \times .94 \times 1.042 \right) + 3.00 \cdot \text{ft of water} = 13.43 \cdot \text{ft of water}$$

Reference:

1. 4.26 Sample Thermal Calculation

2. Assumed to have one pass

R_2
GAUGE CORRECTION FACTOR FOR FRICTION LOSS

Tube O.D. (in.)	12 BWG	14 BWG	16 BWG	18 BWG	20 BWG	22 BWG	23 BWG	24 BWG	25 BWG
0.625	1.38	1.21	1.10	1.00	0.94	0.91	0.90	0.89	0.88
0.750	1.28	1.16	1.06	1.00	0.95	0.93	0.92	0.90	0.90
0.875	1.25	1.13	1.06	1.00	0.96	0.94	0.93	0.92	0.91
1.000	1.19	1.11	1.05	1.00	0.96	0.94	0.94	0.93	0.93
1.125	1.16	1.09	1.04	1.00	0.97	0.95	0.94	0.94	0.93
1.250	1.14	1.08	1.04	1.00	0.97	0.96	0.95	0.94	0.94
1.375	1.13	1.07	1.03	1.00	0.97	0.96	0.95	0.94	0.95
1.500	1.12	1.06	1.03	1.00	0.97	0.96	0.96	0.95	0.95
1.625	1.10	1.05	1.02	1.00	0.97	0.96	0.96	0.95	0.95
1.750	1.10	1.05	1.02	1.00	0.98	0.97	0.96	0.96	0.96
1.875	1.09	1.05	1.02	1.00	0.98	0.97	0.97	0.96	0.96
2.000	1.08	1.04	1.02	1.00	0.98	0.97	0.97	0.96	0.96

Table 5

4.6 Condensate Temperature Depression

4.6.1 Single-Pressure Units- Condensation droplets, as they fall from the tubes, are reheated to saturation temperature under ideal conditions, however, longer tube residence time can produce sub-cooled droplet temperatures. When operating at or near full load, condensers will produce very little sub-cooling (temperature depression). Sub-cooling represents an inefficient condensing process with the possibility of air re-absorption by the colder droplets leading to higher oxygen content in the condensate. Both sub-cooling and resultant higher oxygen levels are undesirable. The distance from hotwell high water level to the bottom tubes will be recommended by the manufacture, which will allow main exhaust steam to effectively reheat the falling droplets, thereby returning their temperature as close to saturation conditions as possible.

4.6.2 Multi-Pressure Units- Multi-pressure condenser designs are created using circulating water flow arranged in series circuits. As the cooling water passes through each shell, it becomes hotter, condensing efficiency decreases, hence the steam side absolute pressure in subsequent shells will be higher than that produced in the initial cold water shell. Chapter 1.0 nomenclature provides illustrations of tube and shell circuit skematics for these arrangements. By cascading from the lower to the higher pressure shell, condensate can be heated to the saturated thermodynamic conditions of that shell. Cascading is normally accomplished through the use of a loop seal that overcomes shell differential pressures. A well designed reheat system

should, under design conditions of operation, be capable of achieving a reheat rate of 80% or better of the temperature difference between the respective pressure zones. The sub-cooling effects of multipressure designs are similar to those of a single pressure design.

4.6.3 Sub-cooling- Sub-cooling can be estimated for single pressure designs in the 0.5 to 1.0°F range and multipressure designs can be estimated using the following equations:

Two Pressure Designs:

$$T_{CHP} = \frac{W_{LP} [(T_{SLP} + 0.8 (T_{SHP} - T_{SLP}))] + W_{HP}(T_{SHP})}{W_{LP} + W_{HP}}$$

$$T_D = T_{SHP} - T_{CHP}$$

Multiple (nth) Three Pressure Designs:

$$T_{CHP} = \frac{\sum \{W_n (T_n + .8 (T_{SHP} - T_n))\} + W_{HP}(T_{SHP})}{\sum W_n}$$

$$T_D = T_{SHP} - T_{CHP}$$

Where:

T_D (°F) Degrees of Temperature depression
n (no units) Denotes an nth shell

R_T
**FRICTION LOSS FOR WATER FLOWING
 IN 18 BWG TUBES**

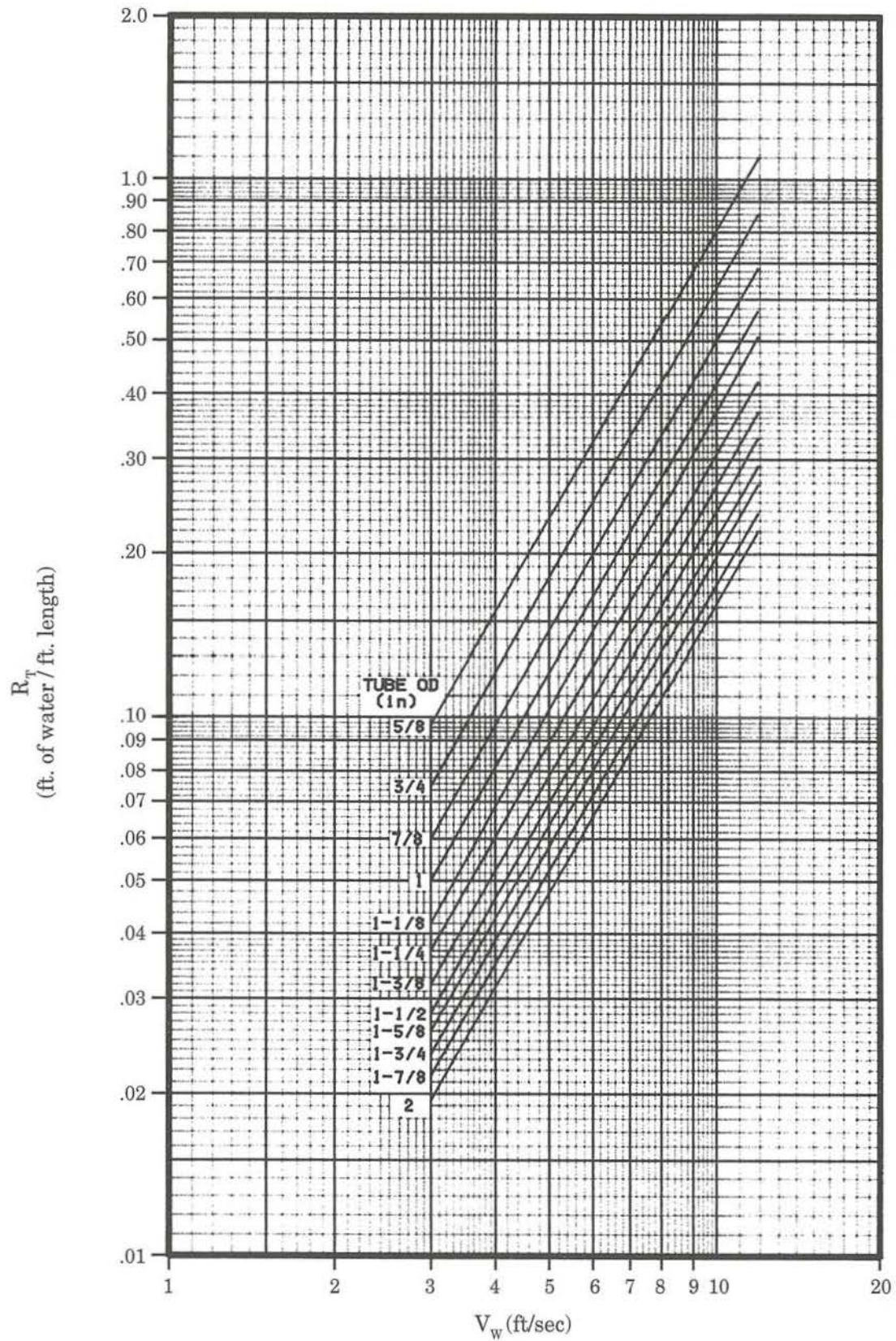


Figure 6
 V_w Velocity Through Tubes ft/sec

R_1
TEMPERATURE CORRECTION FOR FRICTION
LOSS IN TUBES

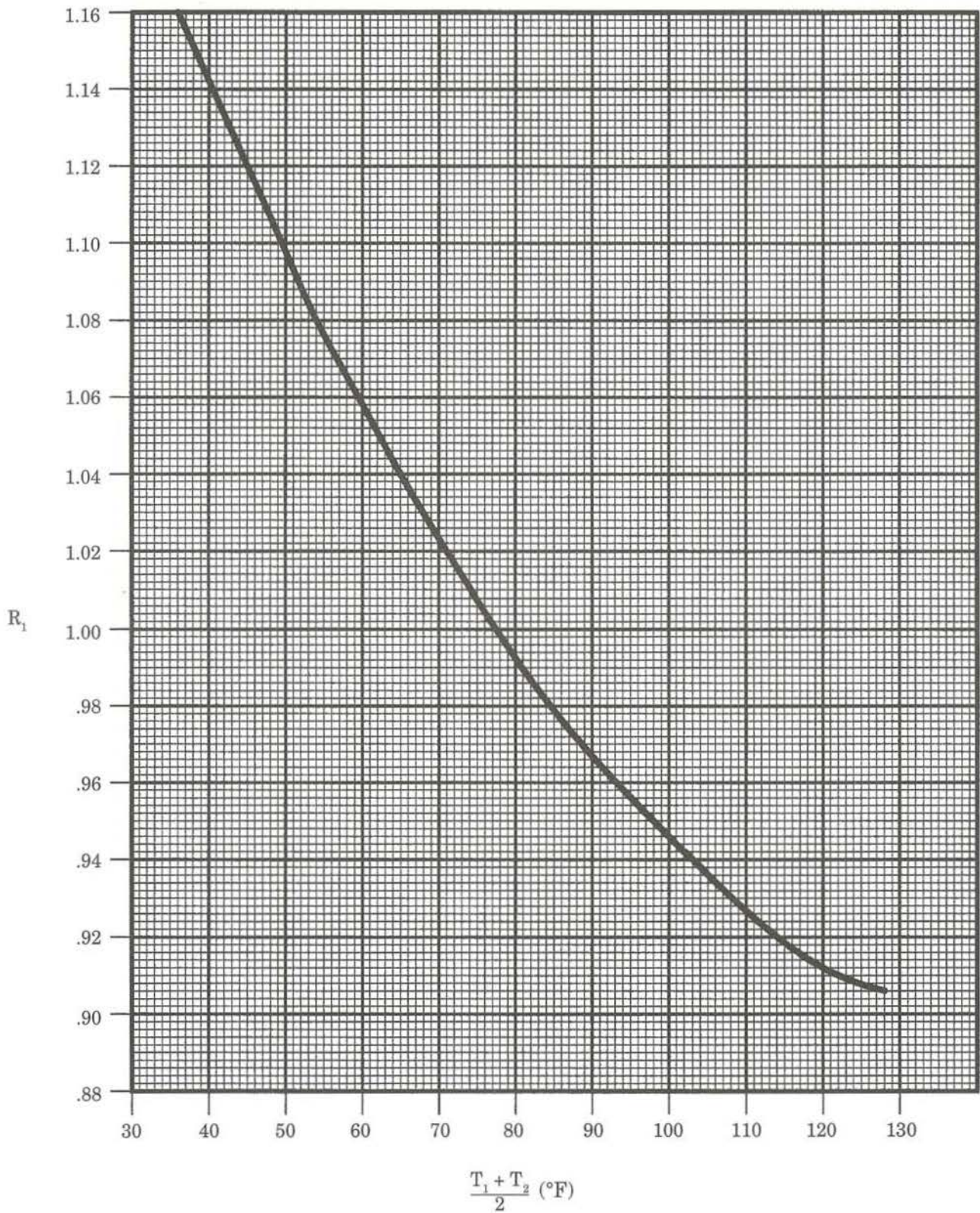


Figure 7

R_E
WATER BOX AND TUBE END LOSSES
SINGLE PASS CONDENSERS

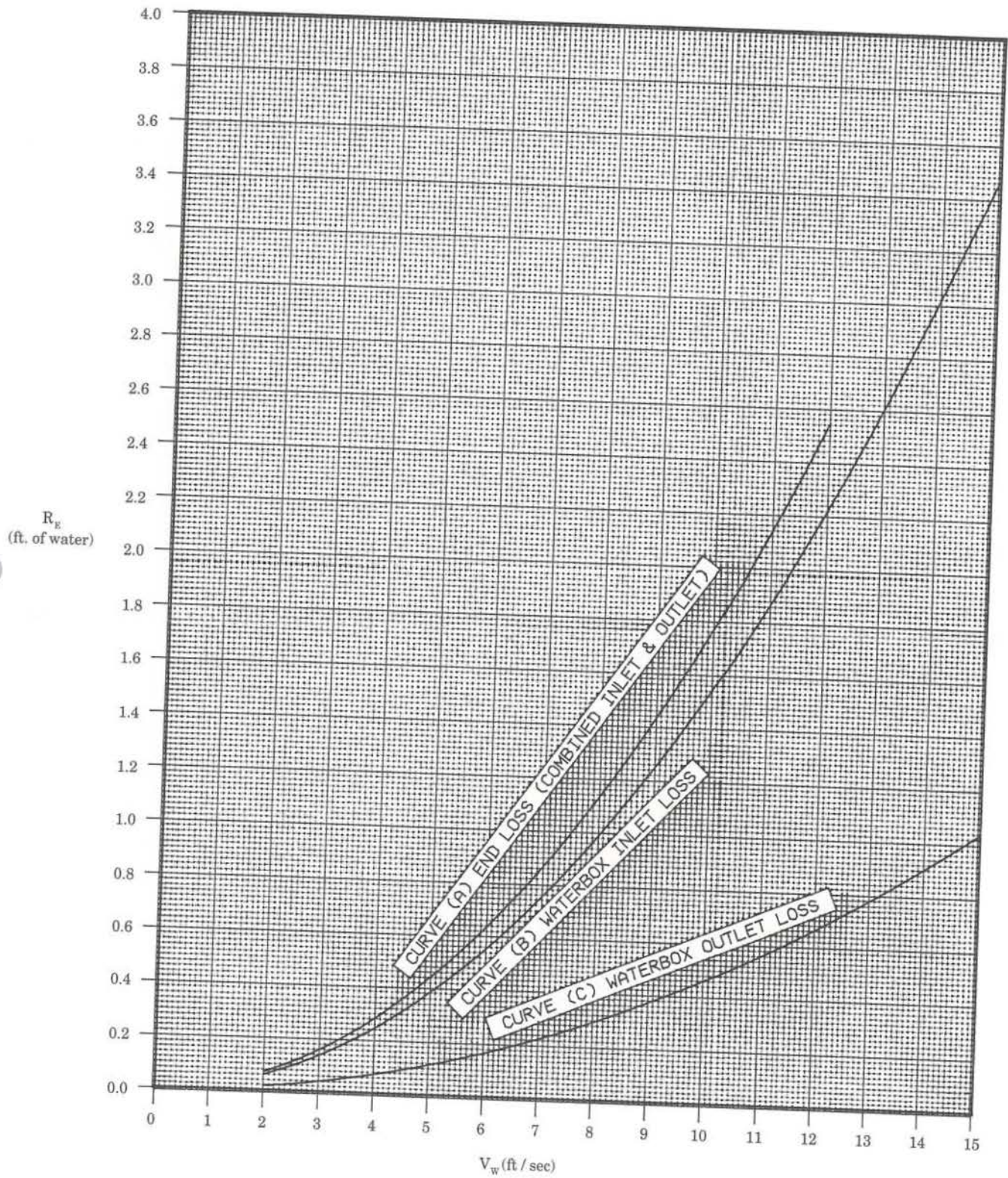


Figure 8
 V_w Velocity Through Tubes ft/sec

R_E
WATER BOX AND TUBE END LOSSES
TWO PASS CONDENSERS

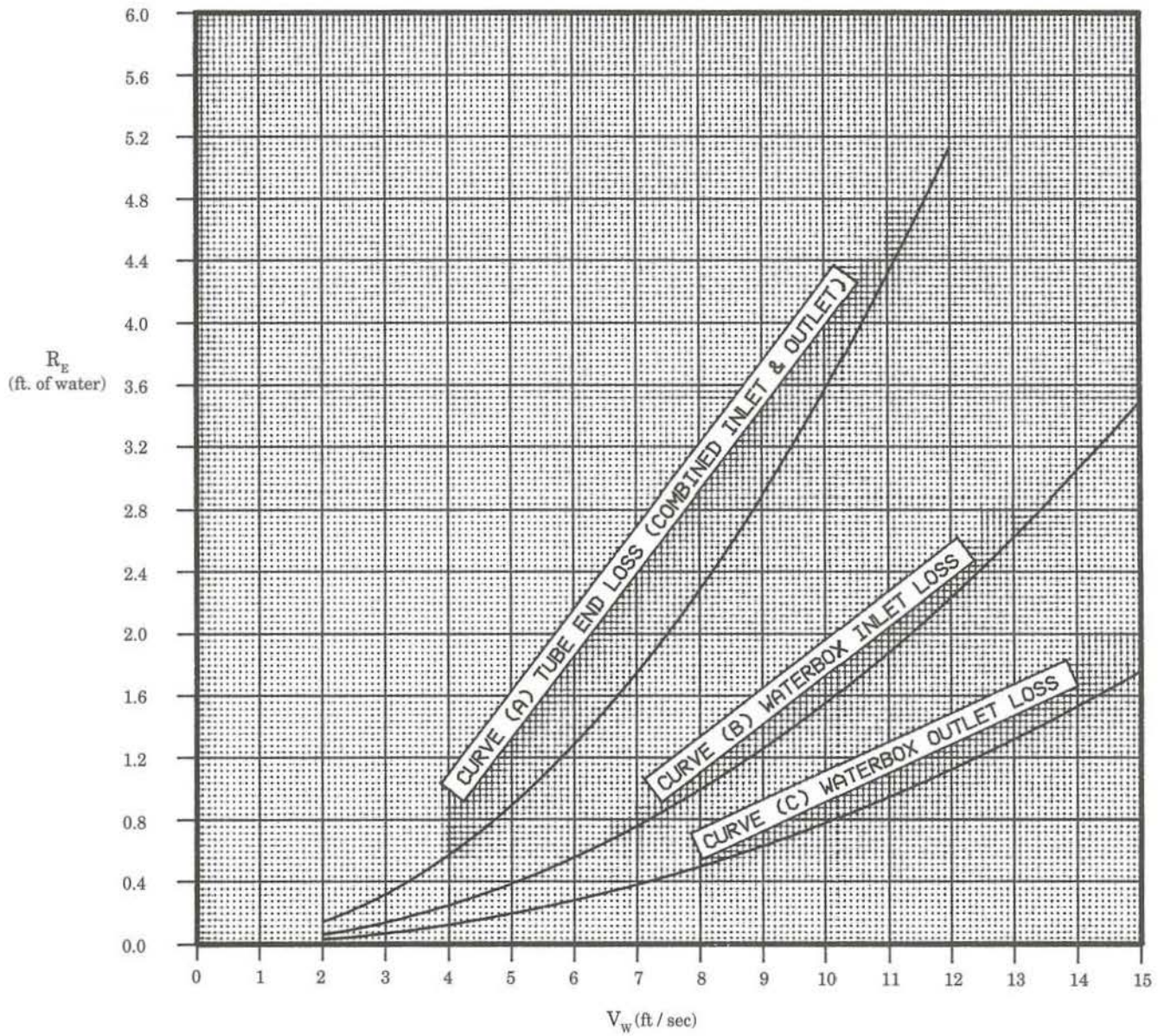


Figure 9
 V_w Velocity Through Tubes ft/sec

R_E
WATER BOX AND TUBE END LOSSES
THREE PASS CONDENSERS

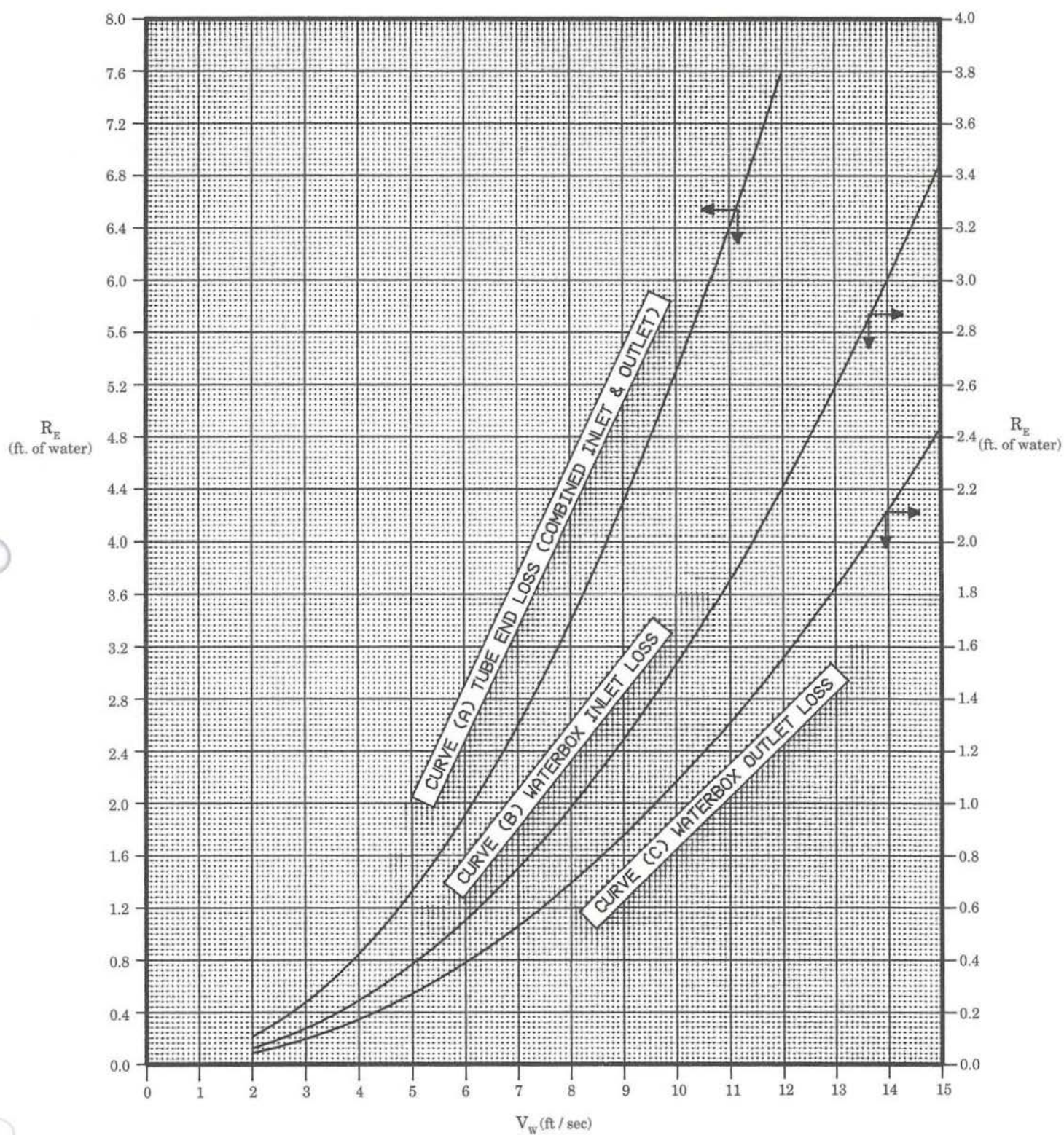


Figure 10
 V_w Velocity Through Tubes ft/sec

R_E
WATER BOX AND TUBE END LOSSES
FOUR PASS CONDENSERS

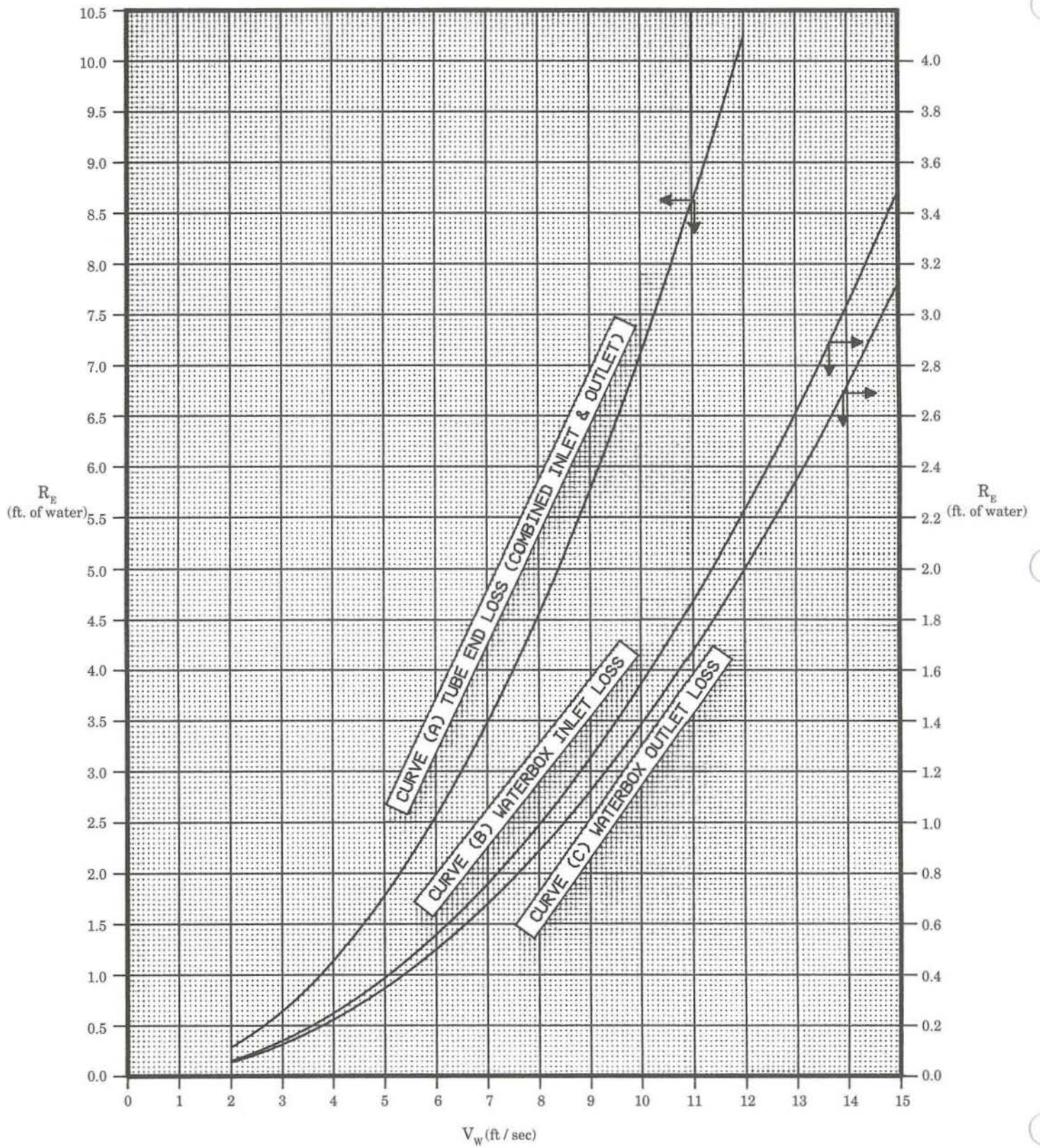


Figure 11
 V_w Velocity Through Tubes ft/sec

4.7 Geothermal Applications

4.7.1 Condensers which are intended for use with steam from geothermal sources require special considerations in design due to substantial differences between geothermal steam and the steam in conventional equipment. The most significant differences are listed below.

4.7.1.1 Conventional units have gas fractions of less than 0.005% by weight while geothermal units have gas fractions which can be several orders of magnitude greater.

4.7.1.2 Geothermal fluids contain elements and chemical compounds in gaseous, dissolved, and particulated forms which can be aggressive in both chemical and mechanical actions in promoting corrosion and erosion. The same constituents may also create fouling films or scaling on both the outside and inside of tubing.

Due to these complexities, the HEI is not in a position to establish design criteria for such equipment. However, the following sections will provide guidelines and information helpful in the selection, rating, and construction of geothermal condensers.

4.7.2 Thermal Design and Rating – The heat transfer coefficients which are established from paragraph 4.2 are not considered valid for geothermal applications because of the high noncondensable gas fractions. Use the following guidelines to correct for the presence of noncondensibles.

4.7.2.1 Cleanliness Factor Correction Method – When operating experience is available, test data can determine an overall fouling equivalency or overall heat transfer coefficient. When such data are available, the Purchaser should specify the factor or overall rate for design. This method is suggested only when the equipment and conditions are exact or near duplicate of an existing condenser.

4.7.2.2 Analytical Procedures – When gas fractions are high or test information is not available, the effect upon condensation in the presence of the gases must be treated by more sophisticated heat and mass transfer analysis. Methods have been developed by various Manufacturers which have been demonstrated to provide reasonable solutions of the heat transfer phenomenon. Any of the analytical methods must be supplemented by a Purchaser specified factor to

account for tube OD and ID fouling. The analytical procedures should account for gradients in condensing temperature caused by the variation in partial pressure of the condensing vapor.

4.7.3 Venting Equipment

4.7.3.1 Capacity of Gas Removal Equipment – The capacity of venting equipment should be based upon actual gas content analysis of the source of the geothermal steam, plus an allowance for air leakage into the system. Also, there is evidence that geothermal wells will have an increasing gas content with age; therefore, the Purchaser should specify a suitable design factor for this condition.

4.7.3.2 Design Suction Pressure and Temperature – The large amounts of noncondensibles in geothermal condensers require special treatment of the noncondensable-vapor mixture. Condensing and cooling of the vented gases and vapor can be accomplished internally or externally to the condenser. The suction pressure and temperature at the vent outlet should be based on the system operating conditions and not on the guidelines of Sections 6.2 and 6.3.

4.7.4 Other Performance Related Characteristics

4.7.4.1 Condensate Temperature – The high gas fraction in geothermal steam will result in significant sub-cooling of the condensate below the apparent saturation temperature of the condenser. Generally, a geothermal hotwell will not provide reheating of the condensate. Condensate temperature depression of 3°F, or more, can be expected.

4.7.4.2 Dissolved Gas in Condensate – The amount of noncondensibles in the condensate is dependent on the amount and composition of the initial gas, the pH of the condensate, the degree of sub-cooling, and the operating pressure. In addition, both mechanical entrainment and chemical combinations contribute to the gas content and gas partitioning. Data is not available in the condenser manufacturing industry to permit prediction of the dissolved gas content.

4.7.5 Performance levels shall be mutually agreed upon between the Purchaser and Manufacturer.

5.0 SERVICE CONNECTIONS

5.1 General Considerations

5.1.1 This section serves as a guide to provide information on the location and design of the various types of connections on a steam surface condenser to permit the dispersion of fluid energies at steady state operation without causing detrimental effects on the unit's internals, especially tubes.

5.1.2 Specific recommendations are provided since each connection will have different flow and energy levels in order to achieve the most effective dispersion. Required connection service will range from high energy, large volume steam dumps (in some cases requiring multi-stage breakdowns and desuperheating) to relatively low flow and low energy level connections.

5.2 Flow Data

5.2.1 It is imperative that the condenser Manufacturer be furnished with reliable flow data required for designing the internal headers and/or baffles. The energy levels and flows will have a bearing on the acceptable location of connections. Incorrect or incomplete information can result in improper location and possible attendant operation problems. Similar comments may be made concerning service (e.g., start-up, continuous) conditions. An example of the latter would be where a condenser Manufacturer may, due to available space, design and locate a connection intended for start-up service in a specific area while the actual service is continuous. This can cause problems as the severity of service is increased over that originally intended.

5.2.2 All thermal and hydraulic design conditions in

the connections provided to the Manufacturer shall be at the connection on the condenser proper (not upstream of control valve, etc.).

5.3 Connection Locations

5.3.1 Locating connections on the condenser shell must be given high priority and be integrated into the plant layout during preparation of the specifications to avoid compromising condenser performance. This basically involves designing the turbine foundation to accomplish two major requirements. First, sufficient space in the foundation must be provided so that proper distribution of connections may be made on the condenser to avoid crowding problems. The second and more important requirement is supplying sufficient volume in the foundation so that the condenser will have required corresponding volume, not only for the internal distribution of headers and baffles, but to provide for the effective dispersion of fluids after resultant expansion within the condenser spaces. Inadequate volume for the latter requirement will result in compromising condenser operation with associated potential for damage within the condenser.

5.3.2 In order to ensure that all connections on the shell side of the condenser are located such that the integrity of the unit is not affected, and to ensure that required deaeration and reheating of flows is obtained, the following requirements on the placement of connections and acceptable conditions of flows in the connections shall be provided. The following paragraphs indicate the preferred locations for some categories of connections usually installed on condenser shells. Numbers indicate the order of preference, space permitting.

	Above Top Tube or in Transition Section	Between Top and Bottom Tube (d)	Below Bottom Tube and Above Maximum Water Level
5.3.2.1 Low Temperature ^(a) Drains Requiring Deaeration	1 (e)	2	NOT RECOMMENDED
5.3.2.2 Low Temperature ^(a) Drains Not Requiring Deaeration	2	3	1
5.3.2.3 Make-up	1	2	NOT RECOMMENDED
5.3.2.4 Condensate Recirculation	1 (e)	2	3
5.3.2.5 Boiler Feed Pump Turbine Exhaust	1	3	2
5.3.2.6 Gland Seal Drain	1	2	NOT RECOMMENDED
5.3.2.7 High Temperature Steam Drains (Higher Flows) (Lower Flows)	1 1	2 2	1 (c) 1
5.3.2.8 High Temperature Water Drains	1	2	1
5.3.2.9 Steam Dumps	1	2	1 (c)
5.3.2.10 Continuous Feedwater Heater Drains (location dependent upon flashing and deaeration capability)	1 (e)	2	NOT RECOMMENDED
5.3.2.11 Water Dumps	1 (e)	2	NOT RECOMMENDED
5.3.2.12 Control or Instrument Connections	AS REQUIRED	AS REQUIRED	AS REQUIRED
5.3.2.13 Miscellaneous Drains and Vents	DETERMINE LOCATION FROM SIMILAR APPLICATION ABOVE	DETERMINE LOCATION FROM SIMILAR APPLICATION ABOVE	DETERMINE LOCATION FROM SIMILAR APPLICATION ABOVE

Note:

- a. Low Temperature refers to flows with a maximum temperature of 100°F over T_s (saturation)
- b. 1 = first preference, 2 = second, 3 = third
- c. When using this location, consideration must be given to available space and to the effect of water level disturbances and water pick-up
- d. Any drains requiring deaeration should be located above the tube bundle centerline.
- e. Locate approximately in the lower 1/3 of exhaust neck

5.4 Connection Design Guidelines

5.4.1 Complete conditions (pressure, temperature, enthalpy and flow) must be provided at each connection. In addition, service conditions shall be supplied (i.e., continuous, intermittent, start-up, etc.).

5.4.2 Limit the enthalpy of entering steam to no more than 1225 Btu/lb. Acceptance of flows with enthalpy higher than 1225 Btu/lb may be considered depending upon specific conditions of service.

5.4.3 Limit pressures to a maximum of 250 psia. Pressures should be lower, where possible, especially for liquid flows. Special considerations for higher pressures should be reviewed with individual Manufacturers.

5.4.4 Ventilator valve (and other high energy short duration sources) discharges should be to the atmosphere; however, if they are directed to the condenser, limitations as described above will apply.

5.4.5 Where conditions exceed the above requirements, external desuperheating must be provided by the purchaser for both the higher flow connections and for the lower flow connections that are in operation when exhaust steam flow is absent. Desuperheating shall be accomplished in a manner that ensures that the condition of the fluid at the condenser wall has 25 - 75°F of superheat.

5.4.6 It is recommended that drains requiring deaeration have a pressure of at least 5 psia.

5.4.7 Design of condenser connections and/or locations should be such that the steam release volumes from the additional steam loading will not result in velocities in excess of 500 ft/sec anywhere on the periphery of the tube bundle or on lagging. The latter will be determined from fluid conditions and available space in the area of connection (i.e., space between support plates, etc.).

5.4.8 Thermal sleeves should be provided on connections designed for flow conditions of 450°F and higher, except for instrumentation connections.

5.4.9 Under no circumstances should steam flashing drains be admitted to the condenser unless circulation water flow is established and non-condensable gas removal equipment is in operation.

5.4.10 Consideration should be given, in divided waterbox condensers, to the possibility that operation with one bundle out of service could cause high temperature steam drains to impinge on the non-operating bundle and cause severe thermal and mechanical problems. Provisions such as locating drains where this condition cannot occur are recommended.

5.4.11 Connections should not be located below the water level in the hot well and at, or near, support plate lines, field weld lines, any flexible diaphragms (exhaust steam inlet expansion joints, shell diaphragm, heater diaphragm, etc.) or corners.

Discharge should not be directed into separate condenser compartments such as the areas below false bottoms in multi pressure units unless this is considered in the design.

5.4.12 Do not locate a series of connections, except gauge and control, in close proximity so that high flow concentrations and/or interferences from discharges from all of the connections will result. High energy drain effluent lines must be kept away from liquid return lines to prevent moisture pick-up and associated erosion.

5.4.13 If insufficient volume is available within the condenser for the introduction of steam dump flows, a separate external steam dump condenser should be considered.

5.4.14 The use of external tanks is recommended for high temperature, high pressure drain flows prior to being admitted to the condenser. This would usually apply to units where a large number of small connections with higher energy levels exist. Minor steam drains or vents may exceed specified conditions in paragraphs 5.4.2 and 5.4.3 provided flow from the main turbine exists and the locations are acceptable to the Manufacturer.

5.4.15 Piping upstream of all flowing connections shall be properly trapped and drained to prevent damaging water slugs being introduced into connections.

5.4.16 The external location shall be such that re-routing of internal piping is not required, since internal piping will interfere with normal steam distribution within the condenser.

5.5 Turbine Bypass Guidelines

5.5.1 General

5.5.1.1 Complete evaluation of the design parameters for main steam bypass lines is important for the safe operation of the condenser. Operating requirements and special customer requirements could affect the condenser design. It is imperative that customers cooperate with the condenser Manufacturer to assure all conditions are examined prior to the final design.

5.5.1.2 Operation of turbine bypass should occur with 100% circulating water flow. Other circulating water operating modes are possible. Careful design and planning are essential, and customer specifications must clearly outline all expected operational modes.

5.5.1.3 The total amount of energy released to the condenser can vary over a wide range. Condenser Manufacturers do not guarantee performance for this service, but rather make accommodations for the safe dispersion of the incoming fluid.

5.5.1.4 Noise abatement measures such as the use of a special noise attenuating valve or noise attenuating insulation should be considered by plant designers. Such valves may reduce noise levels to below plant operating background levels. No noise guarantee can be made by condenser Manufacturers during bypass operations.

5.5.2 Steam Conditioning

5.5.2.1 Steam inlet design values are not to exceed 1225 Btu/lb and 250 psia to ensure the discharge is a vapor, and not a moisture laden mixture capable of creating impingement problems on internal components. External desuperheating devices that reduce enthalpy to 1225 Btu/lb must be located sufficiently upstream of the condenser to ensure adequate mixing of the attemperature fluid, such that when the steam reaches the condenser, super-heated conditions are maintained. Superheat within the dispersion device should be in the 25-75°F range. Wet steam is not permitted.

5.5.2.2 Occasionally turbine Manufacturers set specific guidelines for maximum temperature at the interface of the turbine with the condenser. Main expansion joint suppliers may also have temperature limits, which need to be taken into account. When such limitations are encountered, a cooling water spray curtain may be required within the condenser transition area to reduce local temperature excursions. The water spray should reduce temperatures below 200°F. System delivery rate, pressure, and connection size must be coordinated with the condenser Manufacturer.

5.5.3 Condenser Operations

5.5.3.1 The condenser Manufacturer must be provided with total flow, pressure, temperature, enthalpy and duration of the discharge. A complete understanding of all relevant information such as simultaneous discharges of main exhaust flow and HP, IP, LP bypasses is essential for proper condenser design.

5.5.3.2 The preferred location for bypass discharge is in the transition section. Discharge shall not be directed toward the steam turbine exhaust opening. A secondary location may be the hotwell, but this area may not be large enough to accept the total quantity of steam. The discharge should maximize distribution inside the condenser to allow rapid reduction of the steam velocities.

5.5.3.3 The entrance points of turbine bypass to the condenser should be discussed with the condenser supplier. Consideration must be given to high steam velocity regions, internal impingement, and tube protection.

5.5.3.4 When the condenser is multi-shell and/or multi-pressure configuration, it may be necessary to split the bypass flows between shells and/or pressure zones so that differential pressure/temperature limitations are not exceeded.

5.5.3.5 Design philosophy for the steam bypass dispersion device will differ for each condenser Manufacturer. However, all Manufacturers must take care to ensure a safe distance is maintained between discharge of the spray pipe and the condenser tubes in order to reduce the potential for tube vibration and erosion. Condenser neck height shall be sized to ensure safe bypass operation. In general, the steam should be discharged to avoid direct impingement on the tubes. Tubing can be protected with grating, impingement rods, etc.

5.5.4 Dispersion Device

5.5.4.1 Internal piping should be designed to simplify the bypass pipe support structure and allow for thermal expansion. The piping should have a minimum number of bends and fittings. Where more than one connection is used, the connections should be located so as to ensure proper steam distribution inside the condenser.

5.5.4.2 If the inlet flow temperature is 450 °F or above a thermal sleeve should be provided. See Section 5.4.8.

The pipe size is dependent on the desuperheating device and allowable velocities of the incoming steam. Normal steam velocities within the pipe are in the 200-400 ft/sec range.

5.5.4.3 Dispersion device design pressures are to be established such that blockage of the main turbine exhaust flow is minimized. Maximum line pressure shall be 250 psia.

5.5.4.4 Typical hole size range is 1/4" - 1" in diameter depending on steam flow rate. Hole spacing is a function of line pressure and available space inside the condenser.

5.5.4.5 The condenser Manufacturer will provide adequate drain provisions, internal supports, thermal sleeves, and other specified design details to meet plant design needs.

5.5.4.6 Piping upstream of all flowing connections shall be properly trapped and drained to prevent damaging water slugs being introduced into connections.

6.0 VENTING EQUIPMENT CAPACITIES

6.1 Venting Requirements

6.1.1 Venting equipment must be capable of removing all non-condensibles and associated water vapor from the condenser to produce the minimum steam condensing pressure consistent with physical dimensions and heat transfer and to provide for deaeration of condensate. The sources of the non-condensibles to be removed include, but are not limited to:

6.1.1.1 Air leakage into all system components operating at sub-atmospheric pressure.

6.1.1.2 Gases released from feedwater drains and vents admitted to the condenser.

6.1.1.3 Gases released from make up admitted to the condenser.

6.1.1.4 Condensate surge tank, when utilized in a closed cycle.

6.1.1.5 Disassociation of feedwater into oxygen, hydrogen, and other non-condensibles in certain types of nuclear fueled cycles.

6.1.2 In addition to non-condensibles, an adequate amount of associated water vapor must be vented to insure proper performance of the condenser and to produce reasonable velocities to minimize steam side corrosion within the condenser.

6.2 Design Suction Pressure – In order to coordinate the performance of the venting equipment to be installed with a surface condenser serving a turbine, it is recommended that the design suction pressure be in accordance with the following:

6.2.1 Electric Generating Service – The venting equipment design pressure is 1.0 inch HgA or the condenser design pressure, whichever is lower. Final selection should consider compatible operation of the condenser and its venting equipment over the full range of anticipated condenser operating pressures. In addition, the physical location of the equipment should be considered when the design suction pressure is selected.

6.2.2 Pumps, Compressors, and Other Mechanical Drives – The venting equipment design pressure is that for which the condenser is designed minus 1.0 inch Hg or the lowest required operating pressure, whichever is lower. Minimum is to be 1.0 inch HgA.

6.3 Design Suction Temperature – The saturation temperature of the gas vapor mixture shall be considered as the steam temperature corresponding to the design pressure of the venting equipment less the greater of the following:

$$\begin{array}{c} \text{either} \\ 0.25 (T_s - T_1) \\ \text{or} \\ 7.5^\circ\text{F} \end{array}$$

The 7.5°F temperature differential and the 25% factor are design values utilized to physically size the venting equipment. The actual temperature of the vapor at the vent outlet during operation is influenced

by the operating characteristics, the non-condensable load, and the capacity characteristics of the venting equipment and may not necessarily be equal to the 7.5°F and/or the 25% differential.

6.4 Calculation of Water Vapor Load Component

– The amount of water vapor to saturate the non-condensibles can be calculated from the following formula:

$$W = \frac{18}{MW_{NC}} \times \frac{P_w}{P_t - P_w}$$

When the non-condensable is dry air (molecular weight = 29), the weight of the water vapor can be obtained from Appendix E.

6.5 Minimum Recommended Capacities – It is recommended that the capacity of the venting equipment be not less than the values shown in Tables 6A, 6B, and 6C at the design suction pressure to insure adequate removal capacity under commercial operating conditions.

6.5.1 Procedure for Sizing Venting Equipment

6.5.1.1 Determine the total steam flow of the unit by adding the main turbine exhaust flow and any auxiliary turbine exhaust flow entering all shells of the condenser.

6.5.1.2 Determine the total number of MAIN turbine exhaust openings of all shells. Do not include auxiliary turbine exhaust openings.

6.5.1.3 Divide flow obtained in 6.5.1.1 by exhaust opening number obtained in 6.5.1.2. The resultant number is the *EFFECTIVE STEAM FLOW EACH MAIN EXHAUST OPENING*.

6.5.1.4 Enter the appropriate section of Tables 6A, 6B, or 6C based on whether unit is a single shell, twin shell or triple shell condenser and locate the flow obtained in Step 6.5.1.3 in the left vertical column.

6.5.1.5 Determine TOTAL NUMBER OF EXHAUST OPENINGS for all shells by adding the total number of main turbine exhaust openings to the total number of auxiliary turbines exhausting into the condenser. Split auxiliary turbine exhaust ducts coming from one auxiliary turbine count as one auxiliary turbine exhaust.

6.5.1.6 Locate the appropriate column and capacity using the number obtained in 6.5.1.5.

6.5.1.7 If independent venting systems are utilized for each shell of a multi-shell condenser, the capacity of each system is determined by dividing the total capacity obtained from the appropriate table by the number of independent venting systems.

6.5.1.8 The following is an example of sizing the venting equipment

6.5.1.8.1 Example No. 1: The condenser design parameters are the following:

- Total Steam Flows From Main Turbine Exhausts = 1,600,000 lb/hr
- Total Steam Flows From Turbine Auxiliary Exhausts = 0 lb/hr
- Number of Main Turbine Exhaust Openings = One (1)

- Number of Auxiliary Turbine Exhaust Openings = Zero (0)
- Number of Condenser Shells = One (1)

6.5.1.1 The total steam flow of the unit is the sum of the main turbine exhaust and auxiliary exhausts. [This value is 1,600,000 lb/hr.]

6.5.1.2 The number of main turbine openings is one (1).

6.5.1.3 Divide 1,600,000 lb/hr by one (1). The result is 1,600,000 lb/hr which is the effective steam flow for each main exhaust opening.

6.5.1.4 Enter Table 6A since only one condenser shell is used. Use the row listed for the Effective Steam Flow Each Main Exhaust Opening of 1,000,001 to 2,000,000 lb/hr.

6.5.1.5 The total number of exhaust opening is one (1). This is determined by the sum of the total number main exhaust openings and auxiliary turbine openings. Since there is only one (1) main turbine exhaust opening, use the values from the column marked "1".

6.5.1.6 The intersection of this column and row results in a venting capacity of 15 SCFM.

6.5.1.8.2 Example No. 2: The condenser design parameters are the following:

- Total Steam Flows From Main Turbine Exhausts = 950,000 lb/hr
- Total Steam Flows From Turbine Auxiliary Exhausts = 200,000 lb/hr
- Number of Main Turbine Exhaust Openings = Four (4)
- Number of Auxiliary Turbine Exhaust Openings = Two (2)
- Number of Condenser Shells = Two (2)

6.5.1.1 The total steam flow of the unit is the sum of the main turbine exhaust and auxiliary exhausts. [This value is 1,150,000 lb/hr.]

6.5.1.2 The number of main turbine openings is four (4).

6.5.1.3 Divide 1,150,000 lb/hr by four (4). The result is 287,500 lb/hr which is the effective steam flow for each main exhaust opening.

6.5.1.4 Enter Table 7B since two condenser shells are used. Use the row listed for the Effective Steam Flow Each Main Exhaust Opening of 250,001 to 500,000 lb/hr.

6.5.1.5 The total number of exhaust opening is six (6). This is determined by the sum of the total number main exhaust openings and auxiliary turbine openings. Since there is a total of six (6) turbine exhaust openings, use the values from the column marked "6".

6.5.1.6 The intersection of this column and row results in a venting capacity of 30 SCFM.

6.5.2 Single Pressure Multiple Shell Units – Operating conditions may require that each shell in a single pressure multiple shell condensing plant have its own independent venting equipment for normal operation. In a multiple shell condensing plant with a single venting system, unequal air leakage into one of the condenser shells, unequal tube side fouling, unequal tube side water flows, and unbalanced pressure loss in the piping between either of the condenser shells and their single unit venting system will result in a partial non-condensable pressure build-up in the condenser shell operating at a lower pressure. Pressure in all of the shells will equalize at the pressure level of the poorest performing shell, causing a build-up of oxygen level in the condensate from the combined multiple shell condensing system.

6.5.3 Multi-Pressure Units

6.5.3.1 Multi-Pressure Units, Single Shell – The venting capacity of multi-pressure condensers with pressure stages contained within a single shell should be considered the same as a single shell.

6.5.3.2 Multi-Pressure Units, Multiple Shells – The venting capacity of multi-pressure condensers with pressure stages in separate shells shall be in accordance with paragraph 6.5.1.

6.5.3.3 Consideration should be given to the application of independent venting equipment or other means to insure adequate venting.

6.5.3.4 When steam jet ejectors are used, the temperature of the condensate entering the ejector condensers should correspond to the pressure in the highest pressure shell.

6.5.4 Nuclear Plant Units – The selection of venting equipment to be used with condensers for nuclear power cycles in which additional non-condensable gases are present should be carried out in accordance with Section 6.0 and Tables 6A, 6B, or 6C with allowance for the quantity of such gases specified.

6.5.5 Steam Dump (Bypass) Application – When sustained steam dump operation is required, venting equipment must also be suitable to handle the design quantities of non-condensibles saturated at a temperature 7.5°F below that corresponding to the saturation steam pressures at the highest condensing pressure likely to occur with full steam dump load with all or partial number of circulating water pumps operating.

6.6 Rapid Evacuation Equipment – When starting a turbine, it is desirable to reduce the condenser pressure from atmospheric to some lower value. This can be done by means of single stage ejector or mechanical vacuum pump. The capacity of the device is dependent on the effectiveness of the turbine gland seals, the volume of the condenser shells, turbine casings, and associated ducting as well as the time desired for such reduction. Where specific values are not listed, refer to Table 6.

RAPID EVACUATION EQUIPMENT DRY AIR CAPACITIES

TOTAL STEAM CONDENSED (lb/hour)			DESIGN SUCTION DRY AIR (At a design suction pressure of 10" HgA)	
			(SCFM)	(lb/hour)
UP	to	100,000	50	225
100,001	to	250,000	100	450
250,001	to	500,000	200	900
500,001	to	1,000,000	350	1,575
1,000,001	to	2,000,000	700	3,150
2,000,001	to	3,000,000	1,050	4,725
3,000,001	to	4,000,000	1,400	6,300
4,000,001	to	5,000,000	1,750	7,875
5,000,001	to	6,000,000	2,100	9,450
6,000,001	to	7,000,000	2,450	11,025
7,000,001	to	8,000,000	2,800	12,600
8,000,001	to	9,000,000	3,150	14,175
9,000,001	to	10,000,000	3,500	15,750

NOTE: In the range of 500,000 lbs/hr steam condensed and greater, the above table provides evacuation of the air in the condenser and LP turbine from atmospheric pressure to 10" HgA in about 30 minutes if the volume of the condenser and LP turbine is assumed to be 26 cu ft/1,000 lb/hr of condensed steam.

Table 6

VENTING EQUIPMENT CAPACITIES
A. One Condenser Shell

Effective Steam Flow <i>Each</i> Main Exhaust Opening lbs/hr	Total Number of Exhaust Openings									
	1	2	3	4	5	6	7	8	9	
Up to 25,000 *SCFM	3.0	4.0	5.0	5.0	7.5	7.5	7.5	10.0	10.0	
Dry Air lbs/hr	13.5	18.0	22.5	22.5	33.8	33.8	33.8	45.0	45.0	
Water Vapor lbs/hr	29.7	39.6	49.5	49.5	74.4	74.4	74.4	99.0	99.0	
Total Mixture lbs/hr	43.2	57.6	72.0	72.0	108.2	108.2	108.2	144.0	144.0	
25,001 to 50,000 *SCFM	4.0	5.0	7.5	7.5	10.0	10.0	10.0	12.5	12.5	
Dry Air lbs/hr	18.0	22.5	33.8	33.8	45.0	45.0	45.0	56.2	56.2	
Water Vapor lbs/hr	39.6	49.5	74.4	74.4	99.0	99.0	99.0	123.6	123.6	
Total Mixture lbs/hr	57.6	72.0	108.2	108.2	144.0	144.0	144.0	179.8	179.8	
50,001 to 100,000 *SCFM	5.0	7.5	10.0	10.0	12.5	12.5	15.0	15.0	15.0	
Dry Air lbs/hr	22.5	33.8	45.0	45.0	56.2	56.2	67.5	67.5	67.5	
Water Vapor lbs/hr	49.5	74.4	99.0	99.0	123.6	123.6	148.5	148.5	148.5	
Total Mixture lbs/hr	72.0	108.2	144.0	144.0	179.8	179.8	216.0	216.0	216.0	
100,001 to 250,000 *SCFM	7.5	12.5	12.5	15.0	17.5	20.0	20.0	25.0	25.0	
Dry Air lbs/hr	33.8	56.2	56.2	67.5	78.7	90.0	90.0	112.5	112.5	
Water Vapor lbs/hr	74.4	123.6	123.6	148.5	173.1	198.0	198.0	247.5	247.5	
Total Mixture lbs/hr	108.2	179.8	179.8	216.0	251.8	288.0	288.0	360.0	360.0	
250,001 to 500,000 *SCFM	10.0	15.0	17.5	20.0	25.0	25.0	30.0	30.0	35.0	
Dry Air lbs/hr	45.0	67.5	78.7	90.0	112.5	112.5	135.0	135.0	157.5	
Water Vapor lbs/hr	99.0	148.5	173.1	198.0	247.5	247.5	297.0	297.0	346.5	
Total Mixture lbs/hr	144.0	216.0	251.8	288.0	360.0	360.0	432.0	432.0	504.0	
500,001 to 1,000,000 *SCFM	12.5	20.0	20.0	25.0	30.0	30.0	35.0	40.0	40.0	
Dry Air lbs/hr	56.2	90.0	90.0	112.5	135.0	135.0	157.5	180.0	180.0	
Water Vapor lbs/hr	123.6	198.0	198.0	247.5	297.0	297.0	346.5	396.0	396.0	
Total Mixture lbs/hr	179.8	288.0	288.0	360.0	432.0	432.0	504.0	576.0	576.0	
1,000,001 to 2,000,000 *SCFM	15.0	25.0	25.0	30.0	35.0	40.0	40.0	45.0	50.0	
Dry Air lbs/hr	67.5	112.5	112.5	135.0	157.5	180.0	180.0	202.5	225.0	
Water Vapor lbs/hr	148.5	247.5	247.5	297.0	346.5	396.0	396.0	445.5	495.0	
Total Mixture lbs/hr	216.0	360.0	360.0	432.0	504.0	576.0	576.0	648.0	720.0	
2,000,001 to 3,000,000 *SCFM	17.5	25.0	30.0	35.0	40.0	45.0	50.0	55.0	60.0	
Dry Air lbs/hr	78.7	112.5	135.0	157.5	180.0	202.5	225.0	247.5	270.0	
Water Vapor lbs/hr	173.1	247.5	297.0	346.5	396.0	445.5	495.0	544.5	594.0	
Total Mixture lbs/hr	251.8	360.0	432.0	504.0	576.0	648.0	720.0	792.0	864.0	
3,000,001 to 4,000,000 *SCFM	20.0	30.0	35.0	40.0	45.0	50.0	55.0	60.0	65.0	
Dry Air lbs/hr	90.0	135.0	157.5	180.0	202.5	225.0	247.5	270.0	292.5	
Water Vapor lbs/hr	198.0	297.0	346.5	396.0	444.5	495.0	544.5	594.0	613.5	
Total Mixture lbs/hr	288.0	432.0	504.0	576.0	648.0	720.0	799.2	864.0	936.0	

*14.7 psia at 70°F

Note: These tables are based on air leakage only and the air vapor mixture at 1 inch HgA and 71.5°F.

Table 6A

**VENTING EQUIPMENT CAPACITIES
B. Two Condenser Shells**

Effective Steam Flow <i>Each</i> Main Exhaust Opening lbs/hr		Total Number of Exhaust Openings												
		2	3	4	5	6	7	8	9	10	11	12	13	14
100,000 to 250,000	*SCFM	15.0	20.0	20.0	20.0	25.0	25.0	30.0	30.0	35.0	35.0	40.0	40.0	40.0
	Dry Air lbs/hr	67.5	90.0	90.0	90.0	112.5	112.5	135.0	135.0	157.5	157.5	180.0	180.0	180.0
	Water Vapor lbs/hr	148.5	198.0	198.0	247.5	247.5	297.0	297.0	346.5	346.5	396.0	396.0	396.0	396.0
	Total Mixture lbs/hr	216.0	288.0	288.0	288.0	360.0	360.0	432.0	432.0	504.0	504.0	576.0	576.0	576.0
250,001 to 500,000	*SCFM	20.0	20.0	25.0	30.0	30.0	35.0	40.0	40.0	50.0	50.0	50.0	60.0	60.0
	Dry Air lbs/hr	90.0	90.0	112.5	135.0	135.0	157.5	180.0	180.0	225.0	225.0	225.0	270.0	270.0
	Water Vapor lbs/hr	198.0	198.0	247.5	297.0	297.0	346.5	396.0	396.0	495.0	495.0	495.0	594.0	594.0
	Total Mixture lbs/hr	288.0	288.0	360.0	432.0	432.0	504.0	576.0	576.0	720.0	720.0	720.0	864.0	864.0
500,001 to 1,000,000	*SCFM	25.0	25.0	30.0	35.0	40.0	50.0	50.0	50.0	60.0	60.0	70.0	70.0	70.0
	Dry Air lbs/hr	112.5	112.5	135.0	157.5	180.0	225.0	225.0	225.0	270.0	270.0	315.0	315.0	315.0
	Water Vapor lbs/hr	247.5	247.5	297.0	346.5	396.0	495.0	495.0	495.0	594.0	594.0	693.0	693.0	693.0
	Total Mixture lbs/hr	360.0	360.0	432.0	504.0	576.0	720.0	720.0	720.0	864.0	864.0	1008.0	1008.0	1008.0
1,000,001 to 2,000,000	*SCFM	30.0	35.0	40.0	40.0	50.0	50.0	60.0	60.0	70.0	70.0	80.0	80.0	90.0
	Dry Air lbs/hr	135.0	157.5	180.0	180.0	225.0	225.0	270.0	270.0	315.0	315.0	360.0	360.0	405.0
	Water Vapor lbs/hr	297.0	346.5	396.0	396.0	495.0	495.0	594.0	594.0	693.0	693.0	792.0	792.0	891.0
	Total Mixture lbs/hr	432.0	504.0	576.0	576.0	720.0	720.0	864.0	864.0	1008.0	1008.0	1152.0	1152.0	1296.0
2,000,001 to 3,000,000	*SCFM	35.0	40.0	40.0	50.0	60.0	60.0	70.0	70.0	80.0	80.0	90.0	100.0	100.0
	Dry Air lbs/hr	157.5	180.0	180.0	225.0	270.0	270.0	315.0	315.0	360.0	360.0	405.0	450.0	450.0
	Water Vapor lbs/hr	346.5	396.0	396.0	495.0	594.0	594.0	693.0	693.0	792.0	792.0	891.0	990.0	990.0
	Total Mixture lbs/hr	504.0	576.0	576.0	720.0	864.0	864.0	1008.0	1008.0	1152.0	1152.0	1296.0	1440.0	1440.0
3,000,001 to 4,000,000	*SCFM	40.0	50.0	50.0	60.0	70.0	70.0	80.0	80.0	90.0	100.0	100.0	110.0	120.0
	Dry Air lbs/hr	180.0	225.0	225.0	270.0	315.0	315.0	360.0	360.0	405.0	450.0	450.0	495.0	540.0
	Water Vapor lbs/hr	396.0	495.0	495.0	594.0	693.0	693.0	792.0	792.0	891.0	990.0	990.0	1089.0	1188.0
	Total Mixture lbs/hr	576.0	720.0	720.0	864.0	1008.0	1008.0	1152.0	1152.0	1296.0	1440.0	1440.0	1584.0	1728.0

*14.7 psia at 70°F

Note: These tables are based on air leakage only and the air vapor mixture at 1 inch HgA and 71.5°F.

Table 6B

**VENTING EQUIPMENT CAPACITIES
C. Three Condenser Shells**

Effective Steam Flow <i>Each</i> Main Exhaust Opening lbs/hr	Total Number of Exhaust Openings											
	3	4	5	6	7	8	9	10	11	12	13	14
250,000 to 500,000 *SCFM	30.0	30.0	37.5	37.5	37.5	45.0	52.5	52.5	60.0	60.0	75.0	75.0
Dry Air lbs/hr	135.0	135.0	168.8	168.8	168.8	202.5	236.3	236.3	270.0	270.0	337.5	337.5
Water Vapor lbs/hr	297.0	297.0	371.4	371.4	371.4	445.5	519.9	519.9	594.0	594.0	742.5	742.5
Total Mixture lbs/hr	432.0	432.0	540.2	540.2	540.2	648.0	756.2	756.2	864.0	864.0	1080.0	1080.0
500,001 to 1,000,000 *SCFM	30.0	37.5	45.0	45.0	52.5	52.5	60.0	75.0	75.0	75.0	90.0	90.0
Dry Air lbs/hr	135.0	168.8	202.5	202.5	236.3	236.3	270.0	337.5	337.5	337.5	405.0	405.0
Water Vapor lbs/hr	297.0	371.4	445.5	445.5	519.9	519.9	594.0	742.5	742.5	742.5	891.0	891.0
Total Mixture lbs/hr	432.5	540.2	648.0	648.0	756.2	756.2	864.0	1080.0	1080.0	1080.0	1296.0	1296.0
1,000,001 to 2,000,000 *SCFM	37.5	45.0	52.5	52.5	60.0	75.0	75.0	75.0	90.0	90.0	105.0	105.0
Dry Air lbs/hr	168.8	202.5	236.3	236.3	270.0	337.5	337.5	337.5	405.0	405.0	472.5	472.5
Water Vapor lbs/hr	371.4	445.5	519.9	519.9	594.0	742.5	742.5	742.5	891.0	891.0	1039.5	1039.5
Total Mixture lbs/hr	540.2	648.0	756.2	756.2	864.0	1080.0	1080.0	1080.0	1296.0	1296.0	1512.0	1512.0
2,000,001 to 3,000,000 *SCFM	45.0	52.5	60.0	75.0	75.0	75.0	90.0	90.0	105.0	105.0	120.0	120.0
Dry Air lbs/hr	202.5	236.3	270.0	337.5	337.5	337.5	405.0	405.0	472.5	472.5	540.0	540.0
Water Vapor lbs/hr	445.5	519.9	594.0	742.5	742.5	742.5	891.0	891.0	1039.5	1039.5	1188.0	1188.0
Total Mixture lbs/hr	648.0	756.2	864.0	1080.0	1080.0	1080.0	1296.0	1296.0	1512.0	1512.0	1728.0	1728.0
3,000,001 to 4,000,000 *SCFM	52.5	60.0	75.0	75.0	90.0	90.0	105.0	105.0	120.0	120.0	135.0	135.0
Dry Air lbs/hr	236.3	270.0	337.5	337.5	405.0	405.0	472.5	472.5	540.0	540.0	607.5	607.5
Water Vapor lbs/hr	519.9	594.0	742.5	742.5	891.0	891.0	1039.5	1039.5	1188.0	1188.0	1336.5	1136.5
Total Mixture lbs/hr	756.2	864.0	1080.0	1080.0	1296.0	1296.0	1512.0	1512.0	1728.0	1728.0	1944.0	1944.0

*14.7 psia at 70°F

Note: These tables are based on air leakage only and the air vapor mixture at 1 inch HgA and 71.5°F.

Table 6C

7.0 ATMOSPHERIC RELIEF DEVICES

7.1 General

7.1.1 The size of atmospheric relief devices is dependent upon the local operating conditions. It is always understood that they must be of sufficient size to pass all of the steam which can be admitted to a condenser through any openings, except from the lines which are already protected by relief devices set to open at pressures not exceeding 10 psig.

7.1.2 The size and location of atmospheric relief devices should be based on the following criteria:

7.1.2.1 Device size and associated piping should be selected to prevent pressure in condenser from exceeding 10 psig.

7.1.2.2 Relief devices should be located and installed so they are readily accessible for inspection and repair. The protective devices need not be directly installed on the condenser but may be installed on the turbine exhaust hood.

7.1.3 Exhaust from all relief devices must be properly vented by purchaser to avoid injury to personnel or damage to equipment.

7.1.4 Relief devices should be supported by purchaser and provisions made to keep discharge thrust and/or thermal expansion forces from being transmitted to condenser shell.

7.2 Atmospheric Relief Valves

7.2.1 Install a water seal around the valve disc of ample depth to ensure proper sealing of the seat with provision for adequate drainage.

7.2.2 Valve should be equipped with a manual lifting or opening device for maintenance purposes.

7.2.3 For valve size selection see Table 7. Sizes with flows listed are for guidelines only.

7.3 Rupture Devices

7.3.1 A rupture disc is a non-reclosing pressure relief apparatus actuated by static pressure and designed to function by the bursting of a pressure containing disc.

7.3.2 Every rupture disc shall have a burst pressure tagged in accordance with the design requirements.

7.3.3 Rupture discs may be located on the condenser for ease of replacement. A removable protective cage or an equivalent design must be installed by purchaser to protect plant personnel and avert accidental disc damage.

7.3.4 The following equation may be used to size rupture discs based on dry saturated steam and compliance with the interface piping requirements of ASME Section VIII, DIV. 1, UG-127:

$$A_d = \frac{70 W_s}{K_d P_A 3600}$$

Where,

A_d = minimum required flow area

W_s = discharge flow rate

K_d = flow coefficient, use value of 0.62

P_A = relieving pressure

7.3.5 Rupture discs shall be designed to operate satisfactorily, and without leakage under full vacuum.

ATMOSPHERIC RELIEF VALVE SIZES

Size	Maximum Relief Flow (pounds per hour)
6	7,500
8	20,000
10	30,000
12	45,000
14	62,000
16	82,000
18	120,000
20	170,000
24	250,000
30	380,000
36	550,000

Sizes with flows listed are for reference only. Valve supplier shall determine size for the maximum flow conditions.

Table 7

8.0 CONSTRUCTION

8.1 General

8.1.1 Design Philosophy – This standard contains general rules for the structural design of surface condenser shells, waterboxes, and tubesheets. In those instances where complete details and procedures are not specified, it is intended that the Manufacturer will utilize design and construction procedures which have been demonstrated as being adequate for the service intended and conform to accepted engineering practices. Because of their unique structure, large steam surface condenser components are usually designed by application of conventional elastic design procedures. The methods utilized are those that have been developed and/or applied based on the experience of the various Manufacturers. The steam side of large steam surface condensers is under an essentially static loading condition, and is constructed using ductile materials; the structural design lends itself alternatively to the application of limit analysis techniques which may supplement and/or replace conventional elastic design procedures.

8.1.2 Materials of Construction – Table 8 indicates typical acceptable construction materials for condenser shells and water boxes. Application of a design procedure based on conventional elastic analysis of the structure, together with the allowable stress value (S_A), will provide a factor of safety (FS) against extensive yielding.

$$FS = \frac{S_Y}{S_A}$$

Except where specifically noted, the design formulae presented in these standards are to be used in conjunction with the allowable stress values taken from Section II, Part D of the latest ASME Pressure Vessel Code.

8.1.3 Design Pressures

8.1.3.1 Condenser Shell – The design pressure of the shell shall be 30 in Hg vacuum and suitable for an emergency internal pressure of 15 psig with an allowance, if necessary, for static head developed during hydrostatic test (refer to Section 8.1.4) of the shell when the units exceed 34 feet in height. If limit analysis is employed as a design technique, subject to the requirements of Section 8.2.2, then the limit design pressure shall be no less than 15 psig times the load factor of safety as defined in Section 8.2.2.2. A similar definition of the limit load shall apply where the hydrostatic test condition governs the design.

8.1.3.2 Water Box – The water box design pressure at the bottom of the box shall be specified by the Purchaser. It is defined as the pressure to be used in the design of the water box for the purpose of determining the minimum permissible thickness and/or structural characteristic of the component. The Purchaser in determining the pressure should include consideration for, but not be limited to, normal operating pressure and/or vacuum and temperature at which the component will function. The design pres-

sure should include the range of operating pressures as affected by system characteristics; e.g., consideration of static heads, pump shut off heads, pressure surges, etc. The Purchaser, however, shall design the circulating water system to eliminate pressure surges such as water hammer.

The Purchaser shall specify the magnitude and direction of external loadings on the water box nozzles and shall design the circulating water piping so that unacceptable loads are not imposed upon the water box. Consideration shall be given to the use of ties across the circulating water nozzle expansion joints, or an equivalent load limiting device giving due consideration to thermal movements. The resultant load distribution shall be mutually agreed upon by Manufacturer and Purchaser.

8.1.4 Hydrostatic Testing – All shop hydrostatic tests shall be performed prior to applying any paint, coatings or linings to pressure boundary joints. Duration of test shall be that established by Manufacturer's Quality Assurance Department as necessary to determine leakage or deficiency. It is recommended that the shell and/or water box not be subjected to hydrostatic test conditions where the material temperatures will be below 60°F. If the Purchaser anticipates lower test temperatures, he shall also specify the material to be used for the shell and/or water box.

8.1.4.1 Condenser Shell

8.1.4.1.1 One-piece, shop-tubed condensers shall be tested by filling with clean water.

8.1.4.1.2 Field assembled condensers shall be tested in the installed position by filling with clean water. The water level shall be maintained approximately one foot above final joint of condenser exhaust neck to turbine. The temperature of the water used to test the shell shall not be below 60°F unless materials of construction have sufficiently low Nil Ductility Transition Temperature. If the total height of the unit exceeds 34 feet, a suitable design and/or test procedure should be agreed upon by Manufacturer and Purchaser. The shell side and the water side tests shall not be conducted simultaneously.

8.1.4.1.3 Side exhaust units require special hydrostatic field test procedures because of possible damage to the turbine. The test procedure shall be as agreed upon by the Manufacturer and Purchaser.

8.1.4.2 Water Box

8.1.4.2.1 Hydrostatic test pressure for water boxes shall be 1.3 times design pressure except that the minimum shall be 25 psig. If analyses as described in Section 8.2.2 are used, special consideration should be given to establishing hydrostatic test pressure.

8.1.4.2.2 Hydrostatic test shall be performed in the field after completion of the condenser erection. The field hydrostatic test pressure shall be the operating pressure of the circulating water circuit if the waterboxes have been tested in the shop. Without shop testing, waterbox test pressure shall be in accordance

TYPICAL MATERIALS OF CONSTRUCTION

Component	Materials***	Specification
SHELL AND WATER BOXES	Carbon Steel Plates	ASTM A 36 ASTM A 283 ASTM A 285 ASTM A 515 ASTM A 516
	Stainless Steel Plates 70/30 CuNi Plate and Sheet 90/10 CuNi Plate and Sheet Structural Shapes Bars and Forgings	ASTM A 240 ASTM B 402 ASTM B 402 ASTM A 36 ASTM A 36 ASTM A 105 ASTM A 266 ASTM A 675
	Pipe (Structural Application)	AISI 1020 to 1030** ASTM A 53 ASTM A 106
	Cast Iron *	ASTM A 48 ASTM A 278
BOLTING	Carbon Steel	ASTM A 193
TUBESHEETS	Muntz Metal Aluminum Bronze Naval Brass 70/30 Copper Nickel 90/10 Copper Nickel Silicon Bronze(Copper Silicon Alloy) Titanium Stainless Steel Carbon Steel	ASTM B 171 ASTM B 171 ASTM B 171 ASTM B 171 ASTM B 171 ASTM B 96 ASTM B 265 ASTM A 240 ASTM A 283 ASTM A 285 ASTM A 515 ASTM A 516
TUBES		Refer to Appendix J, K

* Allowable Stress = $0.1 \times S_t$

** Stress values to be obtained using the American Iron and Steel Institute Standards, specifically .92 ($S_u/4$), with maximum temperature of 650°F.

*** Selection of materials is the responsibility of Purchaser. The Manufacturer assumes no responsibility for deterioration of any part or parts of the equipment due to erosion or corrosion or any other causes.

Table 8

with the contract specification. The temperature of the water used to test waterboxes shall not be below 60°F unless the materials of construction have sufficiently low Nil Ductility Transition Temperature.

8.1.4.2.3 With single pass or bottom water inlet condenser, the pressure gauge should be located at the bottom of the water box. With two pass top water inlet condensers, the pressure gauge shall be located at the bottom of the inlet pass. Shop hydrostatic testing shall be specified by the Purchaser if desired. The test should give a pressure loading at least equivalent to the field hydrostatic test pressure loading. If it is impractical to shop test the water boxes in the same position as installed in the field, then the pressure gauge shall be located by mutual agreement between the Purchaser and Manufacturer.

8.1.5 Corrosion Allowances – Values given below are minimum values. If additional allowances are desired, they shall be specified by the Purchaser.

8.1.5.1 Shell – Shell pressure boundary plates, support plates, and welds shall be provided with a 1/32" corrosion allowance on each wetted side. Carbon steel air hoods and/or check baffles shall have minimum thickness of 3/16". Where specific drains and/or high velocity steam flows impact on the shell boundary or structural members, consideration of erosion shall be given.

8.1.5.2 Water Boxes – Water boxes and water box welds shall have a 1/16" corrosion allowance on each wetted side when material is such that a corrosion allowance is indicated. Where cast iron water boxes are specified the Purchaser shall specify corrosion allowance.

8.1.5.3 Tubesheets – Tubesheets shall include a shell side and/or water side corrosion allowance when material is such that a corrosion allowance is indicated. Corrosion allowance shall be mutually agreed

upon by Purchaser and Manufacturer.

8.2 Design and Construction Methods

8.2.1 Design Factors of Safety

8.2.1.1 Factors of safety on stress or load are specified at levels which reflect sophistication and completeness of the analysis. Test data from model tests, proof tests, and/or full scale performance is also permitted to demonstrate the applicability of empirical formulations for configurations which are not amenable to analytical analysis.

8.2.1.2 In these standards explicit design formulae are presented for the analysis of isolated elements of the condenser structure. The allowable stress values to be used with these design procedures are as referenced in paragraph 8.1.2.

8.2.2 Design by More Exact Analyses and by Empirical Formula and Testing

8.2.2.1 In lieu of the explicit design formulae and methods presented in these standards, more complete and exacting elastic analysis of any or all of the structural components is permitted. If a more exact analysis is undertaken and is demonstrated to be applicable, then the higher allowable stresses of the ASME Boiler and Pressure Vessel Code, Section VIII, Div. 2, may be used. These higher allowable stresses, S_A , are the lesser of ($S_U/3$ or $2S_y/3$). Higher allowable values shall be based on classification of the stresses into primary and secondary categories as given in that code. **In no case are these higher allowable stress values to be used with the design formulae in Sections 8.2.3 through 8.2.6.**

8.2.2.2 If limit design techniques are used in lieu of conventional elastic analysis, the factor of safety for the analysis is defined as:

$$FS = \frac{\text{LIMIT LOAD}}{\text{DESIGN LOAD}}$$

The load factor of safety can be no less than 1.67. The use of limit design procedures is restricted to ductile materials satisfying $S_y/S_U \leq 0.8$.

8.2.2.3 Where configurations are such that analytical solutions are not feasible, test data from model tests, proof tests, and/or full scale in service performance under conditions which duplicate the operating environment may be used to demonstrate the structural integrity and validity of empirical design formulations for the subject components.

8.2.2.4 In any structural areas subject to significant cyclic loading, consideration shall be given to fatigue effects when a design procedure is used which permits allowable stress values higher than those referenced in paragraph 8.1.2.

8.2.2.5 The use of any procedures which allow use of increased allowable stress values over those referenced in paragraph 8.2.1.2 shall be undertaken only when it can be demonstrated that the Manufacturer employs a quality assurance program validated by comprehensive documentation and implementation.

8.2.2.6 Where more complete design procedures

which permit the use of higher allowable stress values are employed, such procedures may be subject to independent review.

8.2.3 Shell Design

8.2.3.1 Design Formulae for Condenser Shell Thickness (Rectangular Shells) – Condenser shells are usually comprised of stayed flat rectangular plates which form a basically rectangular box-type vessel. The final geometry may include curved sections. The sections are usually joined by electric arc welding, but may also include sections attached by mechanical fasteners such as bolted flanges. The design thickness of the plate or the pitch of the associated point support stays or ribs can be established by use of the following formulae. In the formulae listed for flat plate sections where load is resisted by bending, the maximum fiber stress is taken as $1.5S_A$. This factor is incorporated in the constant and should not be reapplied.

8.2.3.1.1 Point Support – Pipe

POINT SUPPORT – PIPE

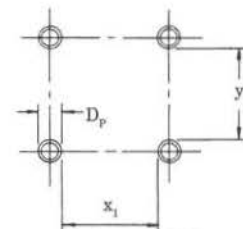


Figure 12

$$t_p \geq \sqrt{\frac{P_D x_1 y_1}{2.8 S_A}}$$

$$\text{where: } .8 \leq \frac{x_1}{y_1} \leq 1.25$$

$$D_p \geq 0.10 \sqrt{x_1 y_1}$$

where: t_p = nominal plate thickness-inches, not including corrosion allowance.

At corners and/or adjacent to ribbed sections or shell side or end plates:

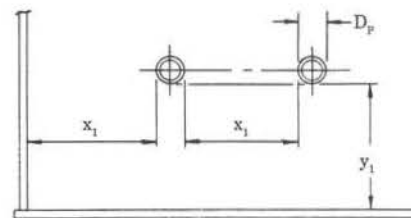


Figure 13

8.2.3.1.2 Alternate Point Support – Alternate plate or structural shape type clips or stays may be used where x_1 or y_1 are defined as shown. When supports other than pipes are used, consideration must be given to minimizing stress concentration effects.

POINT SUPPORT - DOUBLE CLIPS

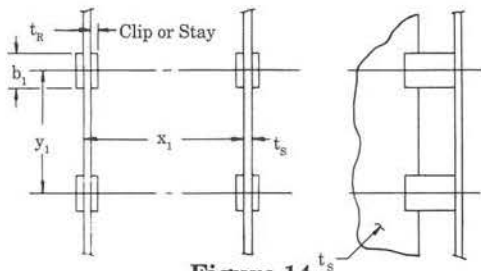


Figure 14

$$t_p \geq \sqrt{\frac{P_D x_1 y_1}{2.8 S_A}}$$

$$\text{where: } .8 \leq \frac{x_1}{y_1} \leq 1.25$$

$$t_R = \frac{3}{8}'' \text{ min.}$$

$$b_1 \geq 0.1 \sqrt{x_1 y_1}$$

POINT SUPPORT - SINGLE CLIPS

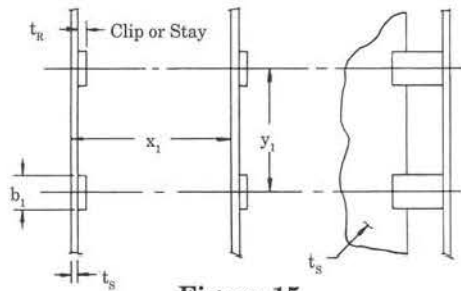


Figure 15

$$t_p \geq \sqrt{\frac{P_D x_1 y_1}{2.8 S_A}}$$

$$\text{where: } .8 \leq \frac{x_1}{y_1} \leq 1.25$$

$$t_R \geq t_s$$

$$b_1 \geq 0.1 \sqrt{x_1 y_1}$$

(Note: Bending loads and/or eccentrically applied loads should be considered in design.)

8.2.3.1.3 Ribs – For continuous ribs or support plates extending to shell wall, unsupported span may be designed as follows:

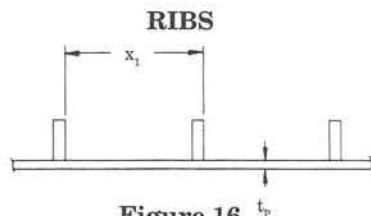


Figure 16

$$t_p \geq \sqrt{\frac{P_D x_1^2}{3 S_A}}$$

Where ribs are used to support the shell plate, the rib depth and thickness shall be designed in accordance with established beam formulae using elastic concepts or, when applicable, limit design concepts.

The rib section modulus may be established based on an effective section considering that a maximum of 30 nominal shell plate thicknesses, less corrosion allowance, acts in concert with the rib to resist bending. When considering part of shell plate as the flange of an effective section, in no case can the effective flange area be greater than the chosen cross sectional area of the added rib.

8.2.3.2 Design Formula for Concentric Compression Stays – When column elements are subjected to significant compression loads, column instability becomes a design consideration. The following formula insures that structural integrity is maintained.

$$\text{IF: } \frac{KL_c}{r} > C_c$$

$$\frac{P_c}{a_c} \leq \frac{\pi^2 E}{1.92 \left(\frac{KL_c}{r} \right)^2}$$

$$\text{IF: } \frac{KL_c}{r} < C_c$$

$$\frac{P_c}{a_c} \leq \frac{S_y}{FS} \left[1 - \left(\frac{KL_c}{\sqrt{2} C_c} \right)^2 \right]$$

$$FS = \frac{5}{3} + \frac{3}{8} \left(\frac{KL_c}{r C_c} \right) - \frac{1}{8} \left(\frac{KL_c}{r C_c} \right)^3$$

Where:

K = end condition factor:

0.65 – fixed ends

1.0 – pinned ends

0.75 – one end fixed

one end pinned

$$C_c = \pi \sqrt{\frac{2E}{S_y}} = \text{slenderness ratio}$$

(Note: Bending loads and/or eccentrically applied loads may necessitate the use of more sophisticated formulation than is shown.)

8.2.3.3 Design Nozzle Loading on Flat Plate

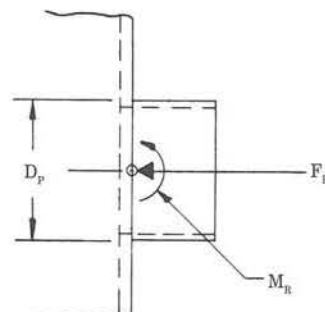


Figure 17

Typically, no reinforcement is provided at shell penetrations; however, consideration must be given to the effect of external forces or moments acting on the nozzle.

Allowable forces and moments can be provided by the condenser manufacturer. Forces and moments in excess of allowables provided and all additional questions shall be reviewed with the condenser manufacturer.

8.2.3.4 Design nozzle loads on cylindrical shell. Refer to Appendix D for design information.

8.2.3.5 Design Formulae for Shell Thickness (Cylindrical Sections) – Where a cylindrical section joins a flat plate section, staying, if required, should be accomplished along the line of tangency. The minimum cylindrical shell thickness for fabricated condensers subjected to external pressure shall be determined from Figure 19. The required corrosion allowance, (1/32") must be added to the thickness determined from these curves or by:

$$t_p \geq \frac{P_D R}{S_A}$$

When a condenser has been designed with an unsupported length L , dependent upon the use of circumferential stiffening rings (either internal or external), the required moment of inertia of such rings shall be determined from Figure 20. In the case of internal rings, the section providing the required moment of inertia must have corrosion allowance (1/32") added to all wetted surfaces.

8.2.4 Support Plate Design Guidelines

8.2.4.1 Thickness

8.2.4.1.1. Support plate design thickness recognizes that the support plates typically support attached pressure boundary members and act as plates subject to edge compressions. In addition, they must support tube bundle static weight and the weight of any heaters or piping and support structure attached to them. They are also subject to bending loads imposed by the tube bundle and other internal structural forces. In most instances, the entire weight of the condenser, including water weights that are present during testing or accidental flooding of unit, must be carried by support plates to the side walls of the condenser shell and via the sidewalls to the support feet.

8.2.4.1.2 The support plate thickness may be determined by:

$$t_s = \frac{30 p L_{SP1}}{S_A (p - D)}$$

This formula incorporates the vacuum pressure loading of 15 psi.

Where: A 1/32" corrosion allowance should be supplied on each face.

Minimum total thickness 1/2".

8.2.4.2 Buckling Considerations – To insure against support plate buckling, appropriate criteria must be applied. Since the support plate acts as a primary

compression member when supporting the shell plate against the vacuum load, longitudinal stiffening may be required (See Figure 18). The spacing of longitudinal stiffeners shall not exceed:

$$h_1 = \frac{220 t_s}{3.5}$$

Any air hoods, check baffles, or other bracing required in the design for other purposes can be considered to act as longitudinal stiffening. Credit may also be taken for longitudinal stiffening provided by the tubes when the Manufacturer can demonstrate by appropriate analysis and/or test that such support does indeed exist in the structure.

SPACING OF LONGITUDINAL STIFFENERS

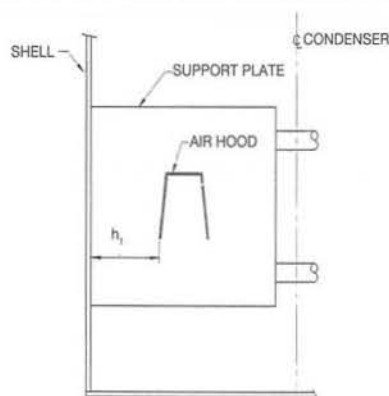


Figure 18

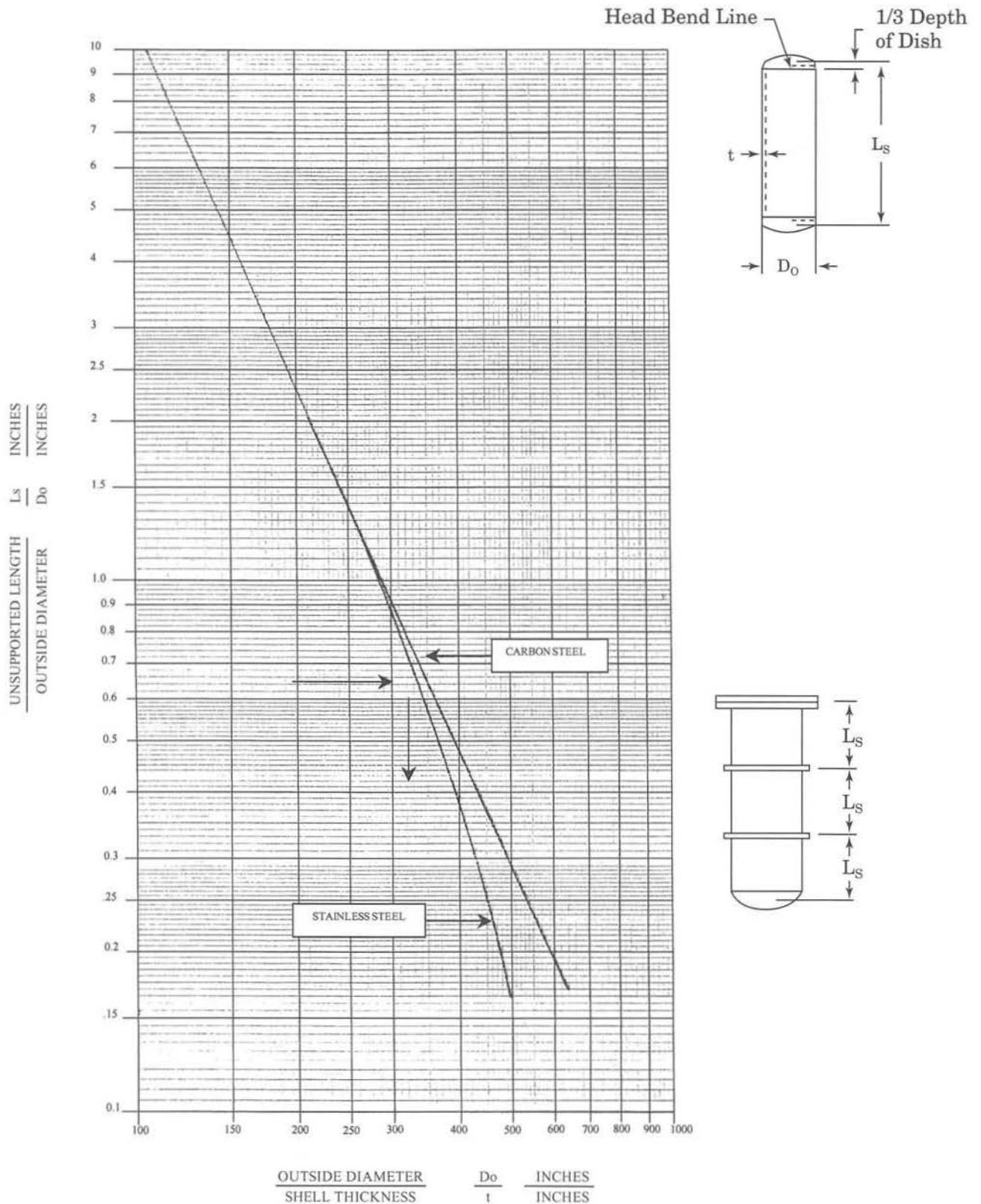
8.2.4.3 Support Plate Spacing

8.2.4.3.1 Due to the difficulty in establishing the actual flow characteristics of steam entering the condenser and other influencing geometry, and fluid flow factors which can contribute to vibration, the HEI includes two design methods for information. By using these or similar methods, spans so calculated have resulted in support plate spacing which is sufficient to provide units free from damaging tube vibration. Other more rigorous methods are available for examining vibration mechanisms like fluidelastic instability, turbulent buffeting, and vortex shedding and can be utilized by designers when they feel it is necessary.

8.2.4.3.2 The maximum intermediate and end support plate spacing shall be the shorter of that obtained by calculating according to the two methods contained in 8.2.4.3.3 and 8.2.4.3.4.

8.2.4.3.3 This method considers tube material, tube geometry, and a deflection limitation based on calculation method equal to approximately 1/3 of the nominal distance between tubes and assumes that sonic velocity is present. The method establishes the spans for the design point of the condenser. In some instances, the condenser is required to operate with one bundle out of service. Under this condition, changes in the support span may be required. If the operating condition with one bundle out of service has an exhaust flow of greater than 1/2 of the design flow or changes in any other parameters which could cause

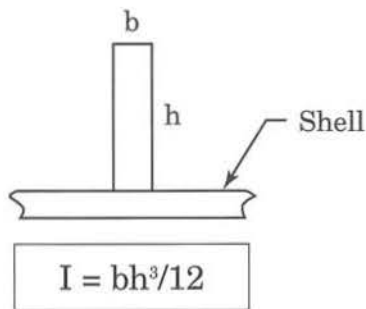
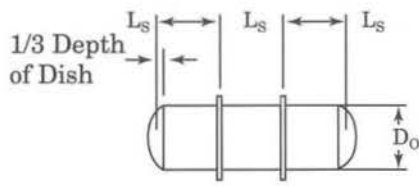
CYLINDRICAL CONDENSER SHELL THICKNESS



Note: The above curves are based on 15 psig external pressure and 300° F metal temperature. The material yield strength utilized for carbon steel was 30,000 – 38,000 psi. The material yield strength for stainless steel was 30,000 psi.

Figure 19

STIFFENING RINGS REQUIRED MOMENT OF INERTIA



Note: The above curve is based on 15 psig external pressure and 300° F metal temperature. The material yield strength utilized for carbon steel was 30,000 to 38,000 psi.

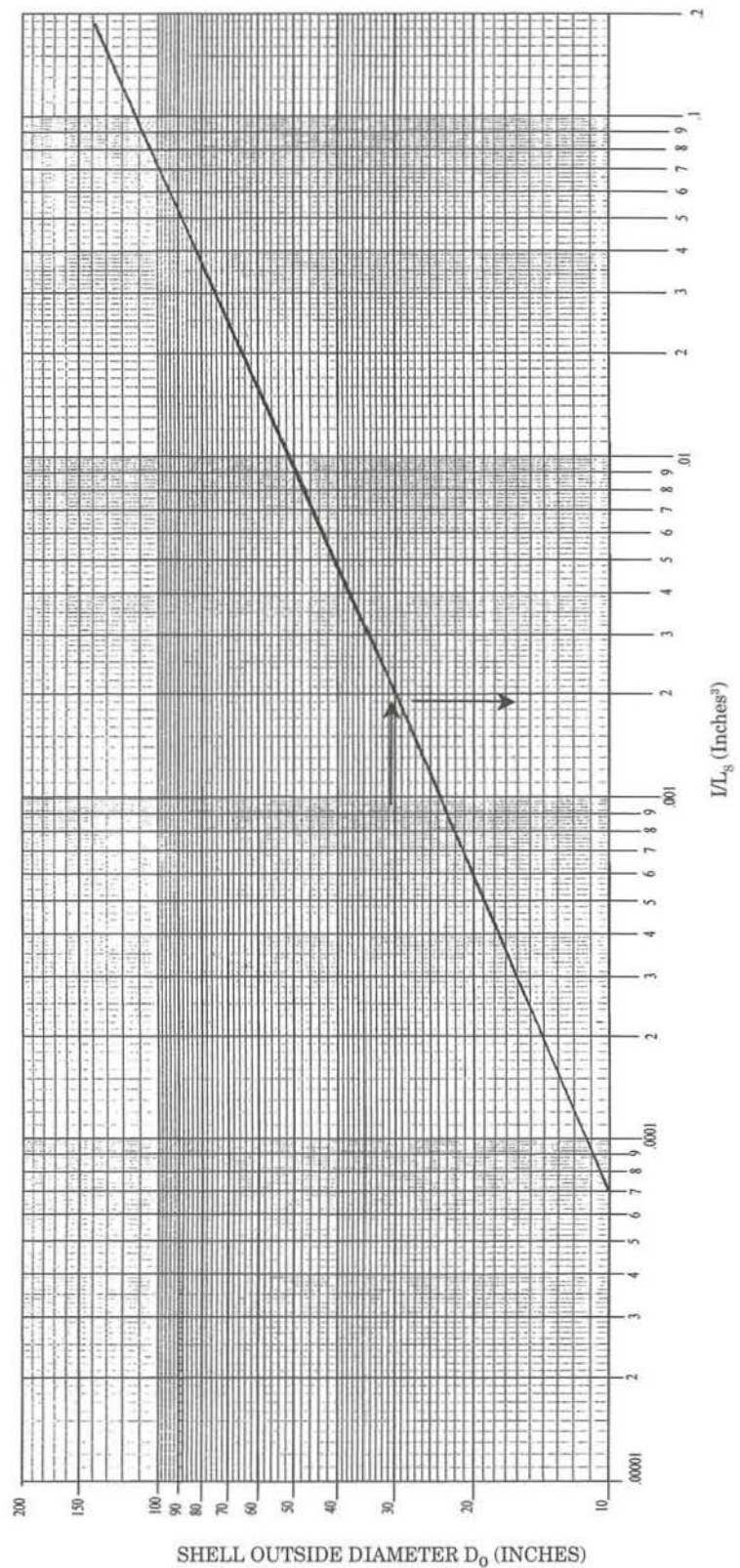
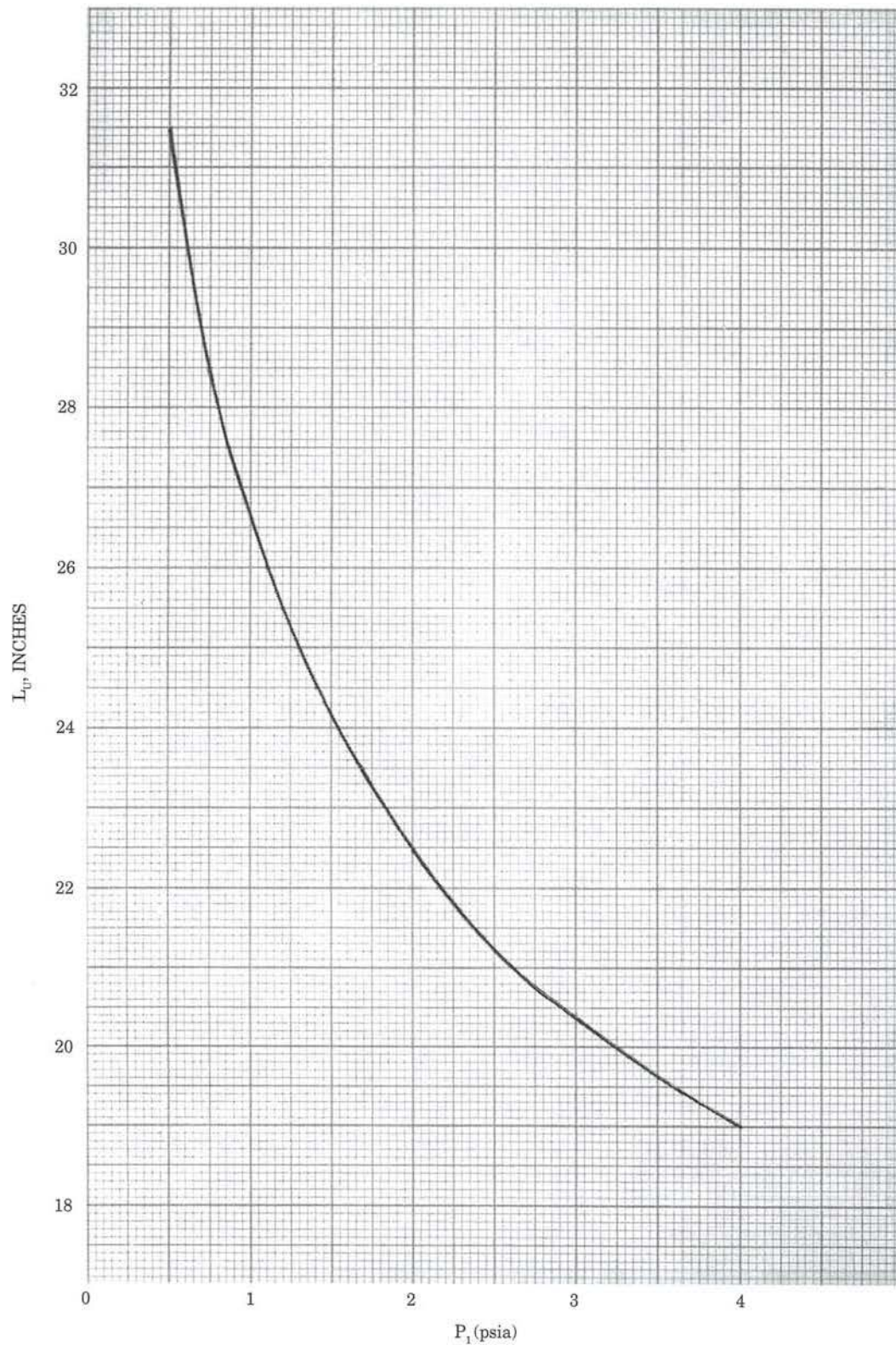


Figure 20

DETERMINATION OF L_U



Note: The value for L_U for pressures less than 0.5 is identical to the value for a pressure of 0.5.

Figure 21

velocities, pressures, volumetric flows, etc., which deviate from the design point, the Purchaser must specify these conditions. The condenser spans will then be established for the most conservative set of conditions.

(1) Calculate the specific volume (v) of the steam when sonic strata conditions of flow occur in the turbine exhaust.

$$v = 1.46 \times 10^6 A_E / W_s$$

(2) Determine saturation pressure P_1 from steam tables corresponding to v from (1). This is the maximum pressure at which sonic velocities exist.

(3) Read L_U from curve in Figure 21 corresponding to P_1 from (2). Note: The value of L_U for pressures less than 0.5 is identical to the value for a pressure of 0.5.

(4) From Tables 12, 13, and 14 determine K_1 , K_2 , and K_3 correction factors.

(5) Calculate support plate span L_{SP} .

$$L_{SP} = L_U \times K_1 \times K_2 \times K_3$$

(6) Maximum intermediate support plate spacing shall not exceed L_{SP1}

$$L_{SP1} = 1.05 L_{SP}$$

(7) Maximum end support plate spacing shall not exceed L_{SP2} .

$$L_{SP2} = 1.18 L_{SP}$$

The above approach is similar to that presented by "Sebal & Nobles," "Control of Vibration in Large Steam Surface Condensers." Volume XXIV Proceedings of the American Power Conference, 1962; and may be generally compared with the approach presented by Coit, Peake and Loheimer "Design & Manufacture of Large Surface Condensers" Volume XXVIII Proceedings of the American Power Conference, 1966, using a severity factor of 1 and other conservative modifications to the original method as defined in the paper "Some Reliability Considerations for Large Surface Condenser" by Peake, Gerstenkorn and Arnold, Volume XXXVII Proceedings of the American Power Conference, 1975.

NOTE: The method defined above calculates allowable span on basis of deflection equivalent to approximately 1/3 nominal ligament. Since various ligament to tube diameter ratios may be used, tube stress in bending may limit allowable span. After the maximum span has been determined, tube bending stress for a deflection of approximately 1/3 ligament should be made and the span reduced if the stress is excessive. For cases where intermediate support plates are added in existing designs, the loading applicable to the original design should be the sole basis for determining tube stresses.

8.2.4.3.4 This method, which is extracted from the MacDuff and Fegler equation, produces results based on limiting the minimum natural frequency of the tube to minimize the effects of vortex shedding and fluidelastic whirling.

$$L_{SP1}^2 = .023 \sqrt{\frac{EI g_c}{W_e}}$$

Where

- $I = \pi(D_o^4 - D_i^4)/64$, sectional moment of inertia
- $g_c =$ gravitational constant, 386.04 in/sec²
- $W_e = W_m + W_f$, effective weight per unit length of the tube
- $W_m =$ weight per unit length of the tube material
- $W_f =$ weight per unit length of the tube side fluid
- $E =$ modulus of elasticity of the tube material

End Spacing not to exceed $1.12 L_{SP1}$

8.2.5 Water Box Thickness Design Guidelines

8.2.5.1 Internal Pressure – The following relationship may be used to determine the basic plate thickness of cylindrical or conical sections:

$$t_p = \frac{P_D K_D R}{S_A e}$$

Where:

- $K_D =$ Factor for discontinuity effects, depending on geometry.
- $e =$ weld efficiencies for welds transverse to hoop stress (if present) = 0.75.

8.2.5.2 External Pressure – If any external pressure loading acts on a major curved portion of the water box plate, the plate thickness must also satisfy the requirements of paragraph 8.2.3.4.

8.2.5.3 Flat Panel Sections – Flat panel widths or plate thickness shall be determined in accordance with the following formula. Where panel widths become too wide to be self supporting, internal (stays, etc.) or external (tee sections, etc.) members shall be supplied for bracing the panels. The design of these members shall be in accordance with applicable procedures and shall consider bending, shear, etc.

8.2.5.3.1 Rib Supported Panels

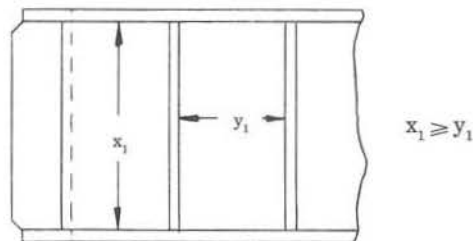


Figure 22

$$\frac{t_p}{y_1} \geq \sqrt{\frac{CP_D}{S_A}}$$

K₁
CORRECTION FACTOR

		PRESSURE RANGE, PSIA						PRESSURE RANGE, PSIA			
O.D.	BWG	0-1	1-2	2-3	3-4	O.D.	BWG	0-1	1-2	2-3	3-4
5/8"	16	0.949	0.950	0.951	0.951	1-3/8"	16	1.461	1.469	1.471	1.471
"	18	0.905	0.904	0.904	0.904	"	18	1.378	1.383	1.384	1.384
"	19	0.879	0.878	0.878	0.877	"	19	1.333	1.336	1.337	1.337
"	20	0.849	0.847	0.845	0.845	"	20	1.281	1.283	1.283	1.283
"	22	0.811	0.808	0.807	0.807	"	22	1.218	1.218	1.219	1.219
"	23	0.792	0.788	0.788	0.787	"	23	1.186	1.186	1.186	1.186
"	24	0.771	0.767	0.766	0.766	"	24	1.152	1.152	1.151	1.151
"	25	0.754	0.750	0.750	0.749	"	25	1.126	1.125	1.125	1.125
3/4"	16	1.052	1.055	1.055	1.055	1-1/2"	16	1.529	1.539	1.541	1.542
"	18	1.000	1.000	1.000	1.000	"	18	1.441	1.447	1.449	1.449
"	19	0.971	0.970	0.969	0.969	"	19	1.393	1.398	1.398	1.399
"	20	0.936	0.934	0.933	0.933	"	20	1.339	1.341	1.342	1.342
"	22	0.893	0.890	0.889	0.889	"	22	1.273	1.274	1.274	1.274
"	23	0.871	0.868	0.867	0.867	"	23	1.239	1.240	1.240	1.240
"	24	0.847	0.843	0.843	0.843	"	24	1.203	1.204	1.204	1.204
"	25	0.829	0.825	0.824	0.824	"	25	1.176	1.176	1.176	1.176
7/8"	16	1.146	1.150	1.150	1.151	1-5/8"	16	1.594	1.605	1.607	1.609
"	18	1.087	1.087	1.088	1.088	"	18	1.501	1.509	1.510	1.511
"	19	1.054	1.053	1.053	1.053	"	19	1.451	1.456	1.458	1.458
"	20	1.015	1.013	1.013	1.013	"	20	1.394	1.398	1.398	1.399
"	22	0.967	0.965	0.965	0.964	"	22	1.325	1.327	1.327	1.327
"	23	0.943	0.940	0.940	0.939	"	23	1.290	1.291	1.292	1.292
"	24	0.917	0.914	0.913	0.913	"	24	1.252	1.253	1.253	1.254
"	25	0.897	0.893	0.893	0.892	"	25	1.224	1.224	1.225	1.225
1"	16	1.232	1.237	1.238	1.238	1-3/4"	16	1.656	1.669	1.671	1.673
"	18	1.167	1.168	1.169	1.169	"	18	1.559	1.568	1.569	1.570
"	19	1.130	1.131	1.131	1.131	"	19	1.506	1.513	1.514	1.515
"	20	1.088	1.087	1.087	1.087	"	20	1.447	1.452	1.453	1.453
"	22	1.036	1.034	1.033	1.033	"	22	1.374	1.378	1.398	1.399
"	23	1.010	1.007	1.007	1.007	"	23	1.338	1.341	1.341	1.341
"	24	0.981	0.977	0.977	0.977	"	24	1.299	1.301	1.302	1.302
"	25	0.960	0.957	0.956	0.956	"	25	1.270	1.271	1.271	1.272
1-1/8"	16	1.312	1.318	1.319	1.320	1-7/8"	16	1.715	1.730	1.733	1.734
"	18	1.241	1.244	1.244	1.245	"	18	1.614	1.624	1.626	1.627
"	19	1.202	1.203	1.203	1.203	"	19	1.559	1.567	1.569	1.570
"	20	1.156	1.156	1.156	1.156	"	20	1.497	1.503	1.505	1.505
"	22	1.100	1.099	1.099	1.099	"	22	1.422	1.427	1.428	1.428
"	23	1.072	1.070	1.070	1.070	"	23	1.385	1.388	1.389	1.389
"	24	1.042	1.040	1.039	1.039	"	24	1.344	1.347	1.348	1.348
"	25	1.018	1.016	1.016	1.015	"	25	1.314	1.313	1.360	1.317
1-1/4"	16	1.388	1.395	1.397	1.398	2"	16	1.773	1.789	1.792	1.794
"	18	1.311	1.315	1.316	1.316	"	18	1.668	1.679	1.682	1.683
"	19	1.269	1.271	1.272	1.272	"	19	1.611	1.620	1.622	1.623
"	20	1.220	1.221	1.221	1.221	"	20	1.546	1.554	1.555	1.556
"	22	1.161	1.160	1.160	1.160	"	22	1.516	1.522	1.523	1.523
"	23	1.131	1.130	1.130	1.129	"	23	1.429	1.434	1.435	1.435
"	24	1.098	1.097	1.097	1.097	"	24	1.388	1.392	1.392	1.393
"	25	1.074	1.072	1.072	1.072	"	25	1.356	1.359	1.360	1.360

Table 9

K₂
CORRECTION FACTOR

Ligament, inches	K ₂
3/16	1.000
1/4	1.075
5/16	1.136
3/8	1.189
7/16	1.236
1/2	1.278

Table 10

K₃
CORRECTION FACTOR

Tube Material	K ₃
Admiralty	1.000
Arsenical Copper	1.015
Aluminum Brass	1.000
Aluminum Bronze	1.023
90-10 Cu Nickel	1.029
70-30 Cu Nickel	1.082
Carbon Steel	1.171
Stainless Steel (304/316/317)	1.151
Titanium B338 Gr. 1 and Gr. 2	0.987
Alloy 194 B 543	1.022
UNS N08367	1.140
UNS S43035	1.171
UNS S44660	1.181
UNS S44735	1.171

Table 11

Where:

$\frac{x_1}{y_1}$	1.0	1.2	1.4	1.5	1.6	1.8	2.0	3.0	4.0
C	.205	.256	.290	.302	.312	.325	.332	.334	.334

Note: These factors apply to water box flat panels only and assume the maximum fiber stress to be equal to $1.5 S_A$.

8.2.5.3.2 Pipe Supported Panels – Panel sections shall be determined in accordance with paragraph 8.2.3.1.

8.2.6 Design Procedures for Flanges and Bolting

8.2.6.1 The following typical procedure, or manufacturer's proven design, may be used to size flat faced gasketed flanges and determine bolting requirements.

8.2.6.2 Bolting of Flat Faced Flanges

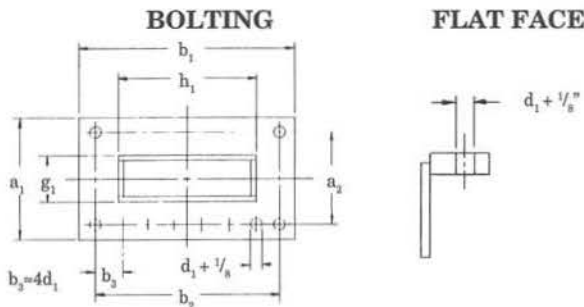


Figure 23

Figure 24

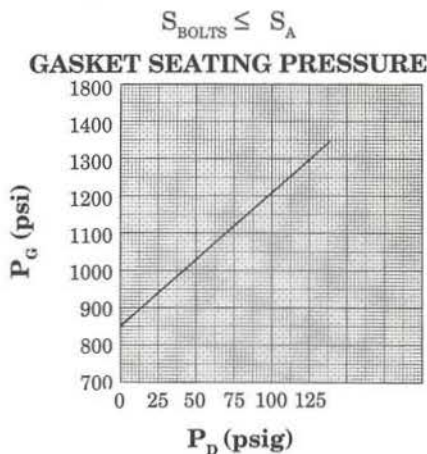
$$F_{\text{BOLTS}} = F_{\text{GASKET}} + F_{\text{HYDRAULIC}} \\ = P_G a_G + P_D a_2 b_2$$

$$= P_G \left[a_1 b_1 - g_1 h_1 - \left\{ N \frac{\pi}{4} \left(d_1 + \frac{1}{8} \right)^2 \right\} \right] + P_D a_2 b_2$$

AND

$$S_{\text{BOLTS}} = \frac{F_{\text{BOLTS}}}{N \times a_{\text{BOLTS}}}$$

Where: P_G = Pressure required to compress gasket (See Figure 25)



Note: The above curve is based on average gasket seating pressure derived from test data with 1/16" cloth insert rubber gasket.

Figure 25

In the hydrostatic test condition, the allowable bolt stress need only satisfy the condition: $S_b \leq 0.9 S_Y$

Where: S_Y = bolt material yield stress

Acceptable bolting materials may be obtained from ASME Code, Section VIII, Division 1.

8.2.6.3 Required Flange Thickness

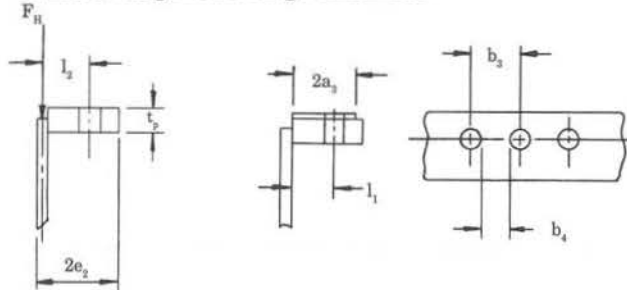


Figure 26

Figure 27

Figure 28

$$t_P = \left[\frac{6 (M_G + M_H)}{S_A \times b_4} \right]^{1/2}$$

Where: M_G = bending moment due to gasket load

$$M_G = \frac{F_G \times l_1 \times f_G}{4N}$$

Where: f_G = gasket geometric correction factor

$f_G = 1$ except when $l_1 > a_3$; Then

$$f_G = 0.9 + \frac{l_1 - a_3}{a_3}$$

Where: M_H = bending moment due to hydraulic load

$$M_H = \frac{F_H \times l_2}{N} \times f_H$$

Where: f_H = hydraulic geometric correction factor

$f_H = 1$, except when $l_2 > 0.6 (2e_2)$; Then

$$f_H = 0.9 + \frac{l_2 - e_2}{e_2}$$

8.2.7 Tubesheet Design Guidelines – The design of condenser tubesheets involves a complex interaction of the tubesheet, tubes, water box, and shell. In addition to hydrostatic pressure, loadings which must be considered include forces and moments imposed on the water box by piping, by dead weight of the water box and its contents, and by effects of differential thermal expansion. It does not appear practical to provide exact tube sheet design methods of broad applicability in HEI Construction Standards because of the great variety of condenser construction, which may include various tube layouts, multiple tube bundle arrangements, different water box types and methods of attachment to the tubesheet, the presence of large

pipng connections of varied orientation, etc.

The structural integrity of the tubesheet and tubes shall be demonstrated by using one of the following methods or its equivalent:

- (1) Interaction analysis using plate and shell formulae.
- (2) Beam strip on elastic foundation (single or multiple strips).
- (3) Finite element analysis (elastic or elastic-plastic).
- (4) Experimental modeling techniques or prior service.

Some of the above methods are described in more detail in the following sections.

8.2.7.1 Model Testing or Prior Service – In lieu of any analysis, a new design may be qualified by testing or prior service as described in Section 8.2.2.3. Prior service, in the present section, means that a duplicate or near duplicate, (structurally equivalent), unit has successfully passed testing at a pressure no less than the hydrostatic test pressure specified for the new unit.

8.2.7.2 Allowable Stresses for Tubesheet and Tube Design by Analysis – The tubesheet design methods of this section follow the design by analysis procedures of the ASME Code, Section VIII, Division 2, subject to the following limitations:

Subsections 8.2.2.5 and 8.2.2.6 of this HEI Standard, dealing with quality assurance and independent review shall apply.

The direct tensile load in any tube under hydrostatic test pressure shall not exceed 0.75 times the average pullout load as determined by a tube pullout test. Without a tube pullout test, the direct tensile load at design pressure is limited to a maximum of 1.5 times the allowable tube-to-tubesheet joint load calculated in accordance with Appendix A of Section VIII, Division 1, of the ASME Boiler and Pressure Vessel Code.

When tubesheet design is based on the approximate beam-strip analysis of subsection 8.2.7.4, the maximum extreme fiber bending stress under hydrostatic test conditions is limited to 1.25 times the yield stress for the tubesheet material.

8.2.7.3 Finite Element Computer Analysis – For any given water box-tubesheet-tube configuration, it is always possible to obtain a reasonably accurate solution of the design problem by applying a suitable finite element computer program to the complete assembly of components. In such a case, where modeling of the entire structure allows proper determination of load paths and boundary conditions, structural integrity may be demonstrated using the procedures of appendices 4 and 5 of the ASME Code, Section VIII, Division 2 (design based on stress and fatigue analysis).

Note that the water box hydrostatic test pressure of this standard is 1.3 times the design pressure, while the hydrostatic test pressure called for in ASME Section VIII, Division 2, is only 1.25 times design pressure. Therefore, the special stress limits for vessels under hydrostatic test given in paragraph AD-151.1 of Section VIII, Division 2 should be checked.

8.2.7.4 Tubesheet Design Using Beam Strip Approximation – The following sub-sections pres-

ent an approximate tubesheet design method which requires certain assumptions as to analytical model used, boundary conditions and loading. This design procedure may be used to determine tubesheet thickness and maximum tube loads.

A condenser tubesheet is a partially perforated rectangular plate supported by its tubes, which act as an elastic foundation. Figure 29 is a somewhat idealized representation of a tubesheet and tube bundle. Because of typically irregular tube patterns and considerable variation in edge boundary conditions and loading, the exact solution of this problem is difficult.

An approximate solution can be obtained by avoiding explicit consideration of the water box and just treating narrow strips of tubesheet and their supporting tubes and beams on an elastic foundation. Some further assumptions must be made to determine loadings on the beam strips and the degree of restraint or end fixity provided at the end of the beam-strips by the water box and shell structure. It is then possible to determine the major factors influencing tubesheet design: maximum bending stress in the tubesheet, and maximum tube loading, which occurs at the outermost tubes. The assumptions as to loading, end fixity and the choice of a particular beam-strip model are influenced by too many variables to permit other than general guidance here. The designer must determine these factors using accepted engineering methods in order to carry out the design procedures detailed below.

Once the beam-strip model and its loading are established, it is possible to achieve a solution either analytically or by a number of widely available computer programs.

A general discussion and explanation of the various steps in the method is given next, followed by detailed procedures and an example.

8.2.7.4.1 Beam-Strip Loading – The loads on the beam-strips are the hydrostatic pressure on the strip, and a bending moment and direct force applied by the water box to the end of the strip. Note that the effective hydrostatic pressure on the strip is reduced in the tubed region because of the tube holes.

The following loadings must be considered:

- (1) Design pressure (psi)
- (2) Hydrostatic test pressure (psi)
- (3) End load (lbs) acting on water box (may include nozzle loads in addition to hydrostatic forces)
- (4) Resultant moment (in-lbs) imposed on water box by dead weight of water box and contents, by hydrostatic pressure, and by piping including the effects of any expansion joints.

No general method for the determination of the force and moment to be applied to the end of the strip can be offered because of the great number of variables involved. In arriving at these loads, the designer must first consider the resultant of all pressures, moments, forces and weights acting on the water box.

A determination must then be made of the distribution of loading around the perimeter of the tube sheet caused by all the forces acting on the water box. This distribution is not necessarily uniform. A larger share of the load is taken by those borders of the tubesheet where the tubes extend out to the very edge. One means of estimating this load distribution is by

determining an effective load-carrying perimeter of the tube bundle, where the outer tubes are close to the water box. In distributing the water box loading over this effective perimeter, due account must be taken not only of the direct force but also of any moment resultant applied by the water box to the tubesheet.

8.2.7.4.2 Choice of the Beam-Strip Models – As described above, the loading from the water box will not be uniformly distributed around the perimeter of the tubesheet. A larger share of this load will be carried by the edges of the tubesheet where tubes are present all the way out to the water box flange. Thus, the choice of beam-strip models should include those regions where the distance between the water box flange and the outermost tubes is a minimum. It is also known that high bending stresses occur near mid-height and mid-width of rectangular plates, at locations such as shown in Figure 30. It is the responsibility of the designer to investigate a sufficient number of beam strips to determine the most severe loading conditions for both tubesheet and tubes.

The beam-strip width shall be chosen as the width of a repeating tube pattern. The width may change around the periphery of the tube sheet due to the omission of tube rows. Examples are illustrated in Figure 31 and 32. The analysis method for perforated beam-strips utilizes the concept of an equivalent solid beam with reduced elastic properties which account for the weakening effect of tube holes. These reduced properties are functions of a "ligament efficiency" which must be defined for the particular configuration under study. An example of the application of ligament efficiency to determine reduced elastic properties is given in Section 8.2.7.4.3.

Forces and moments imposed on the tubesheet by the water box are resisted by the tubes in the outer zone of the tube bundle (unless alternate load paths are provided). Tube loading and tubesheet bending fall off rapidly away from the water box flange due to the elastic foundation action of the tubes. Accordingly, the length of the beam-strip used in the design analysis is not an important factor, so long as it is sufficient to demonstrate that the effect of water box loading is no longer of concern.

A further assumption required by the designer is an estimate of the edge restraint against rotation provided to the beam-strip by the water box flange and shell structure. For instance, a heavy flange with welded stiffeners on the water box provides a relatively high degree of restraint, or edge fixity, while an unstiffened flangeless water box welded directly to the tubesheet provides much less restraint. Variations in edge fixity affect resultant tube loads and tubesheet bending stress. The solution of the beam problem for any assumed degree of edge fixity may be achieved by first solving the fully clamped case (i.e., permitting no end rotation) loaded by water box pull and hydrostatic pressure. The resultant end moment is then by definition the 100 percent fixed end moment.

Any desired portion of this moment, corresponding to the edge fixity assumed, may be combined with the edge load from the water box and the hydrostatic pressure on the beam strip to complete the loading to be considered on the beam.

8.2.7.4.3 Beam Strip Sample Calculations – For an

illustrative example, the beam-strip shown in Figure 32 and 33 is chosen. The problem will be defined for solution by a structural analysis computer program, with the following data given:

(A) Tubesheet

$$E = 15 \times 10^6$$

$$\nu = 0.3$$

$$S_y = 20,000$$

$$d_H = 1.0$$

$$\text{Equilateral triangular tube pitch } p = 1.25$$

$$\text{Tube ligament } h = 0.25$$

Laned arrangement as shown in Fig. 32.

Unperforated length of beam-strip from line of action of water box loading to centerline of first tube, $a_1 = 2$

$$\text{Thickness } t_p = 1.25$$

$$\text{Width of beam-strip, } w = 3p = 3.75$$

(B) Tubes

$$E = 27 \times 10^6$$

$$S_A = 18,700$$

$$S_y = 30,000$$

$$D = 1.0$$

$$\text{Tube wall } t = .049$$

$$\text{Tube metal area } a_m = 0.1464$$

$$\text{Tube flow area } a_f = 0.639$$

$$\text{Moment of inertia } I = 0.0166$$

$$\text{Section modulus } Z = 0.0332$$

$$\text{Tube length between tubesheets } L_R = 40$$

$$\text{Tube length between tubesheet and first support plate } L_1 = 36$$

(C) Loads

$$P_D = 46$$

$$P_h = 60$$

It is assumed that all loading from the water box results in a load per unit width on this beam strip of $F_1 = 1000$ at the hydrostatic test condition.

(D) Calculation of Tube Spring Constant

The structural model of the beam-strip to be analyzed is shown in Figure 34. Each row of tubes across the width w is simulated by a discrete spring having spring constant k_T .

$$k_T = \frac{nEa_m}{\left(\frac{L_E}{2}\right) \times 12}$$

$$n = 2, \text{ (number of tubes across width)}$$

$$k_T = \frac{2 \times 27 \times 10^6 \times 0.1464}{240} = 32,940$$

Note that the axial stiffness of the tubes is based on a length equal to half the distance between condenser tubesheets, not the full distance. This is because each half of the condenser presents separately the problem of a water box, tubesheet and supporting tubes, and a plane normal to the tubes at condenser mid-length can be treated as a fixed reference plane. The bending stiffness of the tubes is neglected in this calculation. To account for this stiffness an additional rotational spring (a function of L_1) could be added at each tube location.

(E) Calculation of Beam Stiffness

Since the beam-strip is part of a plate, the plate bending stiffness will be used. In the unperforated region, the bending stiffness is $\bar{E} \times I$ where

$$\bar{E} = \frac{E}{(1 - \nu^2)} = \frac{15 \times 10^6}{1 - 0.3^2} = 16.48 \times 10^6$$

$$I = \frac{w(t_p)^3}{12} = \frac{3.75 \times 1.25^3}{12} = 0.6104$$

In the perforated region, the bending stiffness is

$\bar{E}^* \times I$ where

$$\bar{E}^* = \frac{E^*}{(1 - \nu^{*2})}$$

E^* , ν^* are effective elastic properties which account for the weakening effect of the tube holes. There is no one convenient source for these reduced effective properties for the various tube patterns and arrangements used in condensers, but several references provide guidance on this subject for common arrangements.** The user is cautioned that some of the reported results are of limited applicability. In this sample calculation, the effective elastic constants E^* , ν^* are based on the net section ligament efficiency (taken across the width of the beam strip) and the curves provided in Section VIII, Division 2. From Figure 32, the effective ligament efficiency, e_s , is found for this strip as:

$$e_s = \frac{3p - 2D}{3p} = \frac{3.75 - 2(1)}{3.75} = 0.47$$

Note that by using the tube outer diameter D in the above formula, the contribution of the tube wall to the strength of the ligament has been neglected. Using the curves in Article 4-9 of Section VIII, Division 2, with a ligament efficiency = 0.47, gives:

$$\frac{E^*}{E} = 0.454; \nu^* = 0.303$$

$\bar{E}^*$ can now be determined as:

$$\bar{E}^* = \frac{E^*}{(1 - \nu^{*2})} = \frac{0.454 (15 \times 10^6)}{1 - (0.303)^2} = 7.498 \times 10^6$$

(F) Calculation of Loadings

End load on beam-strip

$$P_E = F_1 w = F_1 \times (3p) = 1000 \times 3.75 = 3750$$

Uniform loading on unperforated portion of beam strip

$$F_2 = P_h w = 60 \times 3.75 = 225$$

The uniform loading F_3 on the perforated portion of the beam strip is $F_3 = \psi F_2$, where ψ is a factor which accounts for the reduced area of beam strip subject to hydrostatic pressure because of the tube holes. For this example:

$$\psi = \frac{(3p \times p \cos 30^\circ) - (2a_p)}{(3p \times p \cos 30^\circ)} = \frac{(3.75 \times 1.0825) - 2(0.639)}{3.75 \times 1.0825}$$

$$= 0.6852$$

$$\text{Therefore, } F_3 = 225 \times 0.6852 = 154$$

(G) Input for Structural Analysis

For calculation of bending moments and deflections, the following data are used in a structural analysis computer program:

P_E , F_2 , F_3 , k_T , $\bar{E}$, $\bar{E}^*$, I , a_1 , individual span $p \cos 30^\circ$, and total length of beam L_b . In this sample problem, 16 tube rows are modeled so that,

$$L_b = 15 p \cos 30^\circ + a_1 = 16.24 + 2.00 = 18.24$$

The boundary condition at the innermost tube spring is taken as fixed against rotation. The computer solution is obtained first with full fixity against rotation at the outer edge of the beam (point A in Figure 33).

The bending moment obtained in this case at point A is, by definition, the fixed end moment. For this sample problem, the water box is then assumed to offer 50% edge fixity to the beam. A second computer run is then made with 50% of the fixed end bending moment applied at point A together with the pressure loads F_2 and F_3 and the end load P_E . In this second run, point A is not restrained against rotation. The value of the bending moment imposed at point A is 6345 in.-lbs.

(H) Solution

The moment and deflection diagrams for the case of 50% edge fixity are shown in Figure 35. Using these results, tubesheet and tube stresses are now determined at locations of interest by conventional means:

Maximum tubesheet stress in unperforated region (point A)

$$S = \frac{M_A t_p}{2I} = \frac{6345 \times 1.25}{2 \times 0.6104} = 6500$$

Maximum tubesheet stress in perforated region (point C)

$$S = \frac{M_C t_p}{2e_s I} = \frac{4765 \times 1.25}{2 \times 0.47 \times 0.6104} = 10380$$

Maximum tube load (at point B)

$$F = \frac{(k_T \delta_B)}{n} = \frac{32,940 \times 0.032825}{2} = 541$$

δ_B = deflection at point B

These results are then compared with allowable values.

**References on Effective Properties of Tubesheets:

1. ASME Boiler and Pressure Vessel Code, Section VIII, Division 2, Article 4-9.

2. ASME Boiler and Pressure Vessel Code, Section III, Division I, Appendix A, Article A-8000.

3. O'Donnell, W.J., "Effective Elastic Constants for the Bending of Thin Perforated Plates with Triangular and Square Penetration Patterns," Journal of Engineering for Industry, Trans. ASME, Series B, Vol. 95, 1973, p. 121.

4. Soler, A.I., and Hill, W.S. "Effective Bending Properties for Stress Analysis of Rectangular Tubesheets," ASME Paper 76-WA/PWR-1, published in the Trans. ASME, Journal of Engineering for Power, Series A, Vol. 99, 1977.

IDEALIZED REPRESENTATION OF TUBESHEET LOADING

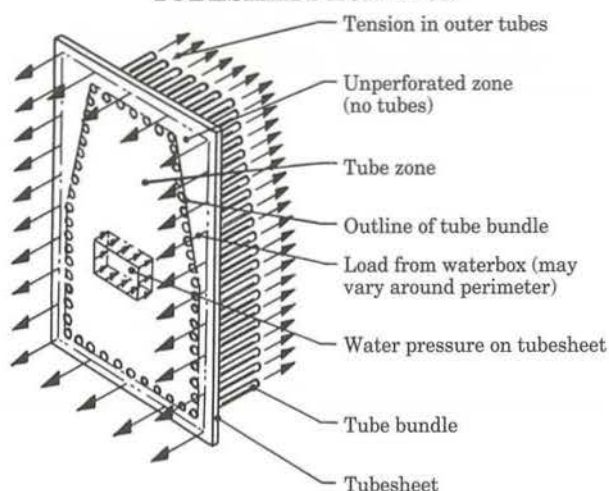


Figure 29

8.2.8 Condenser Tube Ends

8.2.8.1 Tube ends after rolling and/or rolling and flaring at inlet end of tube should present a uniform appearance and be nominally flush with face of tube sheet. By use of appropriate tube rolling procedures, ends of tubes at inlet end will not extend beyond face of tube sheet more than 1/16". Sanding and/or grinding flush is unnecessary.

8.2.8.2 Flaring or belling of inlet ends of tubes is acceptable for copper alloys. For alloys which have an inherent resistance to inlet end erosion (i.e., stainless steel and titanium, etc.) inlet end flaring geometry may be detrimental.

8.2.8.3 The normal construction tolerances used in condenser fabrication plus specified mill tolerances on tube length will result in projection of tubes beyond the face of the outlet end tubesheet. A projection of up to one tube diameter is acceptable.

8.2.8.4 Some design considerations (i.e., reverse flow, welded tube ends, certain types of "on-line" cleaning systems) may require special tube end geometry. In these instances, further treatment of tube end projection should be specified by the Purchaser.

8.2.9 Tubesheet and Support Plate Hole Criteria

8.2.9.1 Support Plate Holes

SUPPORT PLATE HOLE SIZE LIMITS

Nominal Tube O.D. (in)	Lower (in)	Upper (in)
5/8	0.634	0.649
3/4	0.760	0.775
7/8	0.885	0.900
1	1.010	1.025
1 1/8	1.139	1.157
1 1/4	1.264	1.282
1 3/8	1.389	1.407
1 1/2	1.514	1.532
1 5/8	1.640	1.660
1 3/4	1.765	1.785
1 7/8	1.890	1.910
2	2.017	2.037

Table 12

Note: Table 12 was based on ASTM standards for O.D. tolerances. If tolerances other than these are used, hole sizes may have to be adjusted to reflect these variations.

Hole finish is to be 500 RMS. Sharp edges are to be broken and all burrs removed. An over tolerance up to a maximum of 0.006" is permitted on 4% of holes (see 8.2.9.3)

8.2.9.2 Tubesheet Holes

TUBESHEET HOLE SIZE LIMITS

Nominal Tube O.D. (in)	Lower (in)	Upper (in)
5/8	0.632	0.641
3/4	0.758	0.767
7/8	0.883	0.892
1	1.008	1.018
1 1/8	1.138	1.148
1 1/4	1.263	1.273
1 3/8	1.388	1.398
1 1/2	1.513	1.523
1 5/8	1.639	1.651
1 3/4	1.764	1.776
1 7/8	1.889	1.901
2	2.016	2.028

Table 13

Hole finish is to be as follows:

- (1) 500 RMS for carbon steel and stainless steel tubesheets
- (2) 250 RMS for non-ferrous tubesheets

There are to be no axial or spirally oriented grooves visible to the unaided eye that are greater than the

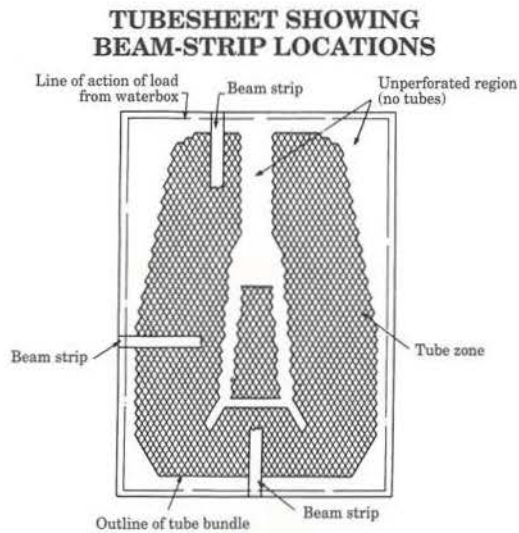


Figure 30

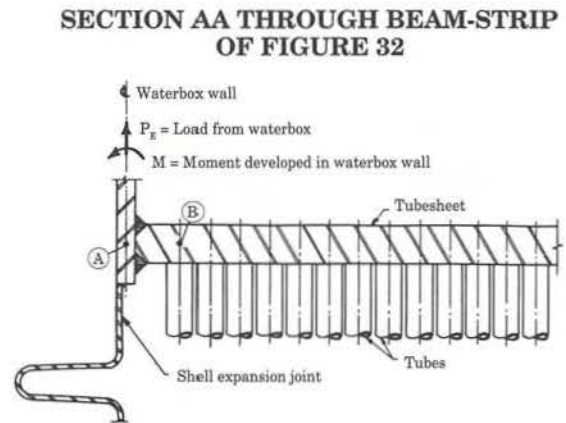


Figure 33

**BEAM-STRIP FOR A TUBE PATTERN
OF TRIANGULAR PITCH**

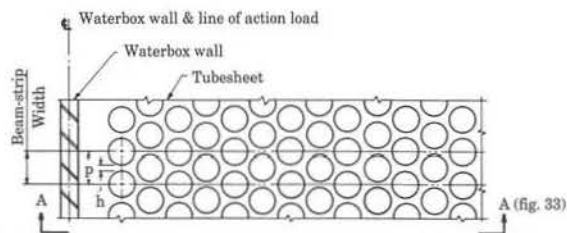


Figure 31

**STRUCTURAL MODEL FOR
BEAM-STRIP OF FIGURE 33**

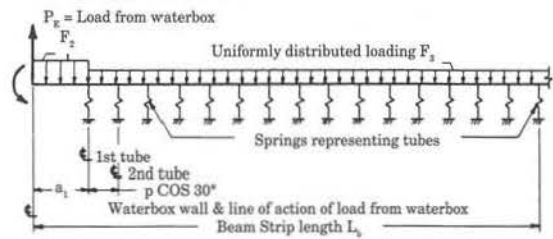


Figure 34

**BEAM-STRIP FOR A LANED TUBE
PATTERN OF TRIANGULAR PITCH**

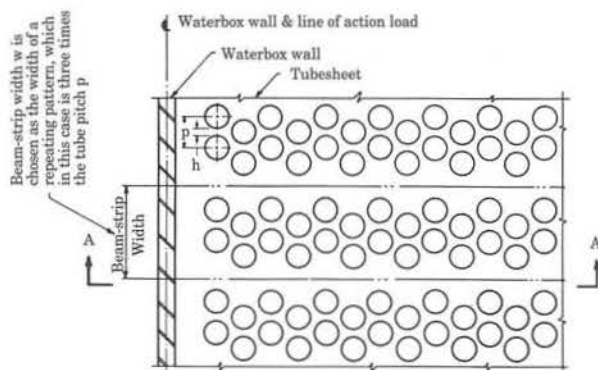


Figure 32

**MOMENT AND DEFLECTION CURVES FOR
BEAM-STRIP OF FIGURE 32**

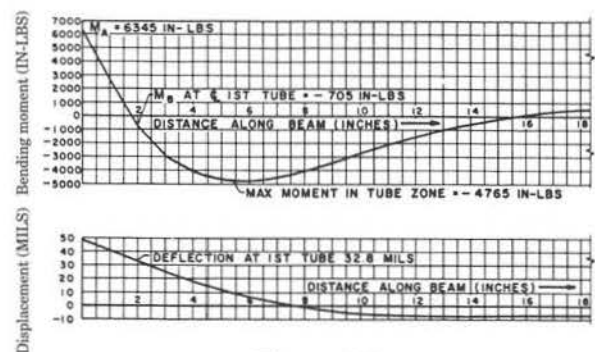


Figure 35

TYPICAL CONDENSER WELDS

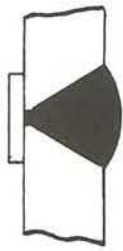


Figure 36



Figure 37

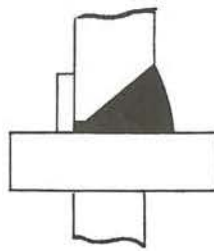


Figure 38



Figure 39



Figure 40

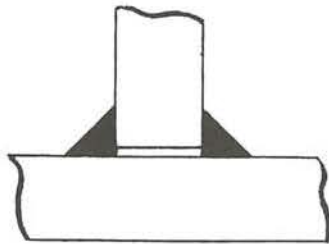


Figure 41

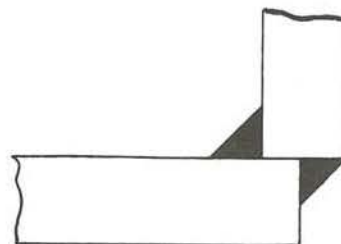


Figure 42

normal hole finish RMS. An over tolerance up to a maximum of 0.006" is permitted on 4% of holes (see 8.2.9.3).

8.2.9.3 Hole variations in excess of those indicated in 8.2.9.1 and 8.2.9.2 will be as permitted by the Manufacturer's Quality Assurance Manual.

8.2.10 Clad Tubesheets

8.2.10.1 Dimensional, thickness and flatness tolerances shall be per Manufacturer's Standard and/or Purchaser Specifications.

8.2.10.2 Carbon steel base plate and clad material should be specified using ASME/ASTM material specifications.

8.2.10.3 Tubesheets shall be ultrasonically tested in accordance with ASTM B 898.

8.2.10.4 Bonding must extend to the edges of the plate.

8.2.10.5 If the tubesheet is so large that the clad plate must be spliced:

- a) The splice should not be in the tube field
- b) The splice weld must be ground flush and level with the clad plate.
- c) The weld splice should be 100% UT or RT.

8.2.10.6 Grooves in tube holes, if and when required, shall be located in the base plate area, unless otherwise specified.

8.2.10.7 Repair of disbonding in the condenser manufacturer's shop shall be per manufacturer's standard.

8.3 Welding

8.3.1 Pressure boundary welds may be either full penetration or properly designed fillet welds. Where fillet welds are utilized, the weld configuration must be such as to be adequate to support whatever bending loads

are present to the joint. If welded from one side only, a backing strip of compatible material shall be used.

8.3.2 For design guidance in calculating weld stresses, refer to Figure 43. Typical welds used in condenser construction are shown in Figures 36 to 42. Main structural welds which support the pressure boundary or which form part of the main foundation supports shall have a minimum leg length of 5/16 inch unless otherwise justified by the Manufacturer's design criteria. All other structural welding (except welds which are seal welds) shall have a leg length of $0.75 \times$ the thickness of the thinnest part being welded or 3/16 inch whichever is larger. Seal welds shall be 1/8 inch minimum. Main structural welds which support the pressure boundary or which form a part of the main foundation supports shall not be single fillet welded if any bending loads can be expected. Other structural members such as Categories II and III of Paragraph 9.1.3 may be of the single fillet weld style.

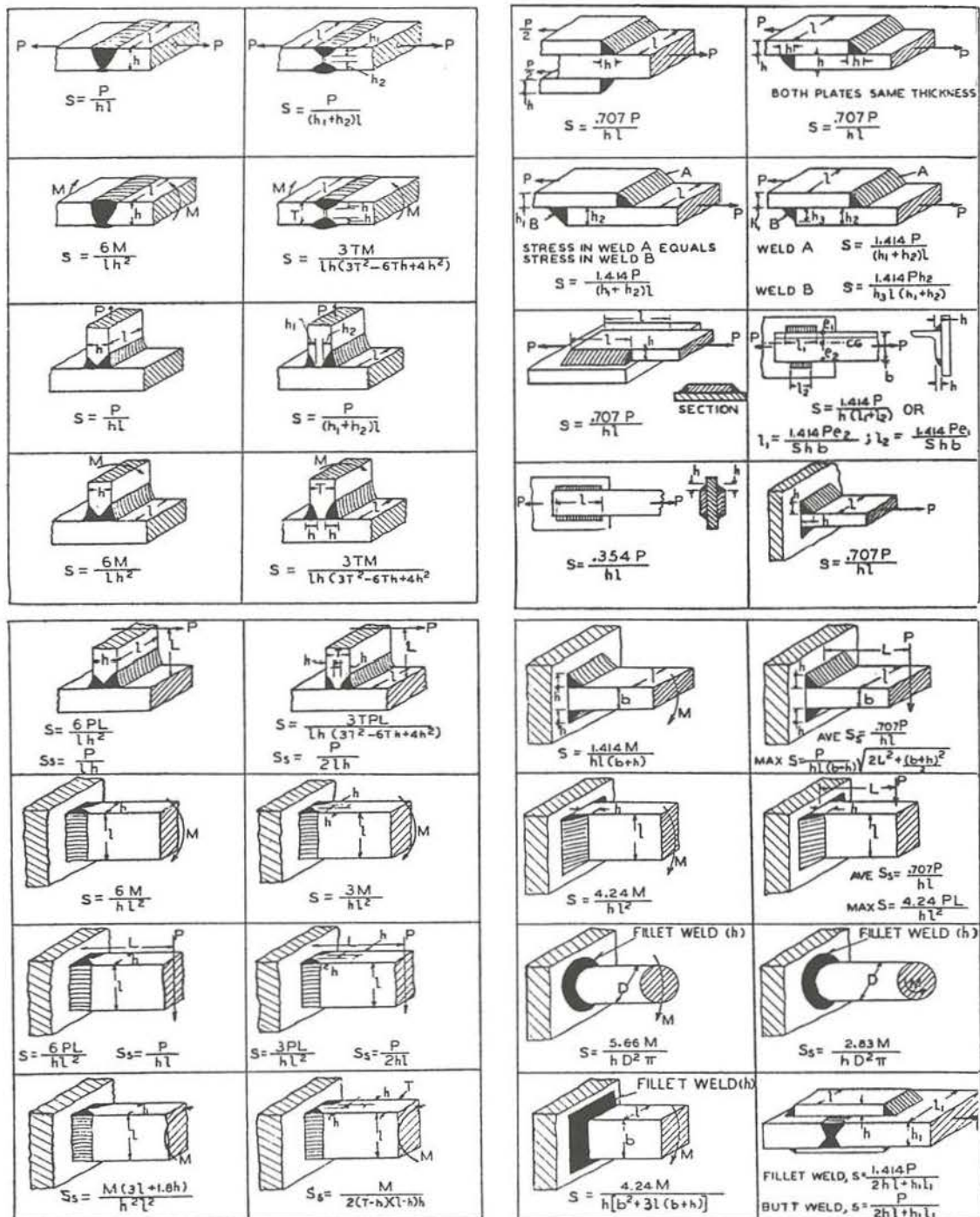
8.3.3 All such welds shall be of sufficient size to transfer these loads with an allowable stress level that is no greater than that of the base material. For fillet welds, the stress shall not exceed 0.55 times allowable (S_A) stress of base material based upon minimum leg length ($0.55S_A = 0.77S_A \times \text{Sine } 45^\circ$).

8.3.4 Welding of condenser components will be subject to visual inspection and dimension check in accordance with Section 9.1. All butt-type field welds used for assembly of the shell pressure boundary will be full depth welds. When the weld is made without a backing strip, it must be welded from both sides unless otherwise qualified.

8.4 Lagging for Extraction Lines and Feedwater Heaters

8.4.1 This section contains rules for the design, fabrication and field installation of lagging around extraction piping and feedwater heaters within the condenser exhaust neck.

WELD GEOMETRIES



S for fillet weld construction is equal to .77 S_u ; S for butt weld construction is equal to S_u .

Figure 43

8.4.2 Materials for Sheathing

8.4.2.1 Stainless Steel-300 Series (Austenitic)-Type 304; 16 gauge minimum.

8.4.2.2 Stainless Steel-400 Series (Ferritic)-Type 410S; 16 gauge minimum.

8.4.2.3 Ferrous – Commercial Carbon Steel similar to A-36 or A-285 C; 10 gauge minimum.

8.4.3 Standoff Design

8.4.3.1 Continuous standoffs shall be spaced a maximum of 24 inches for longitudinal spans and at each end, with not less than 0.5 inches standoff distance

8.4.3.2 Intermittent standoffs shall be spaced a maximum of 24 inches for longitudinal spans and circumferentially the spacing shall be approximately 8 inches for extraction lines and approximately 12 inches for feedwater heaters. Standoff distance shall not be less than 0.5 inch.

8.4.3.3 Standoff material should be compatible with the pipe material if applied to the piping and with the lagging material if applied to the lagging.

8.4.3.4 If the lagging system utilizes standoffs welded to the pipe or the vessel, all Code requirements or special welding controls associated with the piping system or vessel must be complied with. This may require certification of welders and Code stamping as may be applicable.

8.4.4 Design Considerations

8.4.4.1 Consideration for differential thermal expansions shall be incorporated into the design.

8.4.4.2 Stiffening shall be provided as necessary to limit vibration.

8.4.4.3 Drainage shall be provided at the lowest point in each assembly. However, where continuous standoffs are provided, drain holes shall be provided between each pair of standoffs. Minimum drain hole size shall be 0.5 inch.

8.4.5 General Considerations

8.4.5.1 Design shall not interfere with expansion joints or other appurtenances in the line.

8.4.5.2 The lagging will be designed to minimize field modification and installation effort; however, installation tailored to suit as constructed dimensions must be considered part of the normal installation requirements and is not considered part of the Manufacturer's supply.

8.4.5.3 Lagging installer should be knowledgeable in this craft. Additional securing as may be indicated by experience will be the responsibility of the installer as will field corrections and/or trimming for structural interferences, piping supports, instrument connections, etc., and variations as may be caused by tolerance build up.

8.4.5.4 Circumferential and longitudinal parting line joints shall be lapped a minimum of one inch. The lap shall be in a direction to avoid generation of a steam flow between lagging and vessel lagged.

8.4.6 Method of Attachment

8.4.6.1 Continuous welding is preferred. If intermittent welding is used, it shall be 3 inches every 6 inches. Intermittent welding is not permitted at branch connections.

8.4.6.2 Mechanical fastening is acceptable using banding, screws, etc. If banding is used, it shall be a minimum of 1 inch wide and the same thickness and material as the basic lagging material.

8.4.6.2.1 Where mechanical fastening is used, fasteners must be of sufficient length and strength and incorporate some method of locking such as tack welding to preclude loosening.

8.4.6.2.2 Banding shall be applied directly over standoffs. Bands shall also be tack welded (or mechanically secured) to prevent slippage during operation.

8.4.7 Quality Control

8.4.7.1 Extraction piping and heater lagging are often located in high velocity flow areas and are subject to thermal movement and flow induced vibration. Following completion of lagging, a visual inspection shall be performed and acceptance criteria should be as defined in Section 9.1.

8.4.7.2 In addition to these requirements, care must be taken to avoid denting or damaging caused by walking on or by supporting temporary platforms, etc., on the lagging.

8.4.7.3 Any major deformation of the lagging shall be reworked to the contour of the surrounding lagging.

8.5 Fabrication For Geothermal Service – The fabrication of geothermal condensers should be in accordance with the provisions of this Standard, except as noted below:

8.5.1 Materials should be specified by the Purchaser. Materials should be compatible with the expected chemical composition of the geothermal fluids and any combinations that can be expected to occur in the condensing process.

8.5.2 Corrosion allowances shall be as specified by the Purchaser.

8.5.3 Post weld heat treatment of condensers should not be considered for equipment of this size.

8.5.4 Welding processes for Series 300 Stainless Steels should be qualified to a suitable delta Ferrite range.

8.5.5 Welding processes for Series 300 Stainless Steels should limit interpass temperatures to a maximum of 350°F.

8.6 Condenser Support Systems

8.6.1 General – There are a variety of condenser support methods that can be used in power plant installations. Each design has many variables but depend primarily on the turbine casing allowable loads. Other influencing factors are; seismic criteria, down exhaust, side exhaust, circulating water pipe layout, etc. In all cases a complete evaluation of the turbine/condenser interface loads must be accomplished. It is incumbent upon the turbine manufacturer, condenser manufacturer and foundation designers to exchange sufficient information such that an acceptable arrangement is selected and agreed to by all parties.

8.6.2 Condenser hard mounted to concrete base

8.6.2.1 Turbine / condenser expansion joint – When an expansion joint is installed between the turbine and condenser (for isolation purposes) the vacuum unbalance load is significant due to the exhaust size. This load is calculated by multiplying the expansion joint effective area times atmospheric pressure. The direction can be horizontal or vertical depending on the installation. The condenser manufacturer calculates the net uplift and/or moments as the arrangement dictates and resultant loads are presented to the plant foundation designers for approval. The turbine manufacturer must evaluate a similar load but opposite in direction.

8.6.2.2 Circulating water expansion joint – The placement of expansion joints in piping adjacent to the condenser is common practice. When joints are used, they normally require control rods with compression sleeves to prevent large unbalanced forces from over loading the condenser and piping components. Control rods and sleeves can not be eliminated unless the condenser and its anchorage are designed to accommodate the unbalanced loads.

8.6.2.3 Water test loads – Condensers are hydro tested (steam side filled with water) after field installation. The condenser manufacturer evaluates structural integrity and advises the plant foundation designer of all loads acting at the supports.

8.6.3 Condenser hard mounted to LP turbine exhaust casing

8.6.3.1 Thermal considerations – When a condenser is bolted or welded directly to the turbine casing, they move together in thermal harmony. It is important to identify the origins of this movement so all parties can add appropriate thermal guides and seismic restraints as required by plant design.

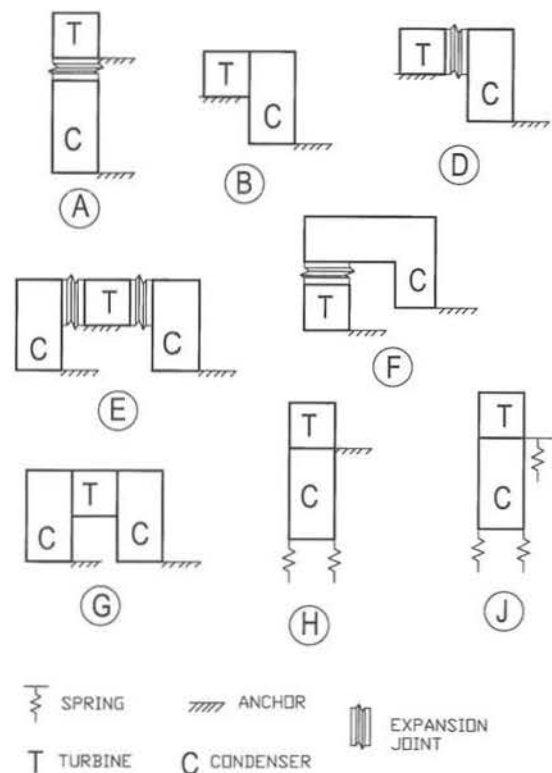
8.6.3.2 Equipment loads:

8.6.3.2.1 Turbine loads acting on condenser – All turbine loads (therefore weight, static, dynamic, etc.) must be presented to the condenser manufacturer for incorporation into the design to insure adequate condenser supports are furnished. Depending on the magnitude of the forces, the condenser manufacturer may offer special provisions to support these loads.

8.6.3.2.2 Condenser loads acting on the LP turbine – Sometimes, by virtue of arrangement, the condenser manufacturer is obligated to use the turbine casing as partial support for the condenser. When this occurs, the condenser manufacturer must present all loads to the plant designer for evaluation by the turbine manufacturer.

8.6.3.3 Spring support considerations – Condenser springs are used in some installations. The turbine manufacturer must provide data concerning the load carrying ability of the casing and depending on the turbine allowable loads, the condenser can be either partially or fully supported by springs. Almost always the condenser will need additional cribbing during field hydro test (steam side filled with water).

8.6.4 Support Systems Arrangements



9.0 INSPECTION, QUALITY, TRANSPORTATION, AND FIELD INSTALLATION

9.1 Inspection and Quality of Welding Standards

9.1.1 Inspection Standards for Welds – This section establishes minimum standards for visual inspection of condensers in three specific categories based on the expected severity of service requirements of the various zones within and/or attachments to the structure. The visual acceptance criteria are developed using recognized codes and standards such as ASME codes, ANSI standards, AWWA, and AWS as a guide. More stringent requirements may be specified by the Purchaser and will take precedence.

9.1.1.1 Non-destructive examination is not performed except by special arrangement between the Manufacturer and the Purchaser.

9.1.1.2 The welding shall be performed using welders and written weld procedures which have been qualified in a manner comparable to that defined in Section IX of the ASME Unfired Pressure Vessel Code.

9.1.1.3 Welds shall be examined in the “as welded” condition preceded only by normal cleaning.

9.1.2 Weld Inspection Methods and Equipment

9.1.2.1 Personnel performing inspections shall be qualified to eye examinations in accordance with Society of Nondestructive Testing (SNT) requirements.

9.1.2.2 Inspections are performed with the naked eye or corrective lenses as required to obtain the visual acuity required by SNT. Magnifying glasses, microscopes, and other optical devices shall not be used.

9.1.2.3 Any required measurement of welds shall be made using instruments such as scales, rules, calipers, gauges (height and depth), fillet gauges, and dial-type equipment. Micro-measuring and vernier-type equipment is not required.

9.1.2.4 Accuracy of measurement equipment shall be to 1/64". All measuring equipment shall be maintained and calibrated in accordance with the Manufacturer's approved quality control manuals and procedures.

9.1.3 Weld Categories – The following categories are established considering the service requirements of specific types of welds.

These criteria apply to shop welds and to field welds in the apparatus except for pipe welds made to connection stubs.

9.1.3.1 Category I includes pressure boundary and main support welds.

Weld Type A. Those welds which provide a separation of atmospheric pressure and condenser internal pressure.
B. Cantilever or outrigger type main support welds to shell.
C. Extraction piping welds.
D. Heater and extraction pipe supports to shell.

E. Steam and water dumping spray systems designed for operating at 100 psi or greater.

9.1.3.2 Category II includes structural welds.

Weld Type A. Tube supports or members connecting supports to the shell and support members to each other.
B. Joints in main bracing members and ribs and attachment of main bracing and ribs to the shell.
C. End attachments of longitudinal bracing used for control of hydraulic loads.
D. Plates or members separating internal pressure zones.
E. Condenser bottom supports and anchors.
F. Impact baffle plates.
G. Spray and distribution systems operating at less than 100 psi.

9.1.3.3 Category III includes all other welds.

Weld Type A. Internal partitions such as air removal systems, hotwell dividers and retention systems, drain trays, and leak detection systems.
B. Screens, erosion shields, dirt collars, vortex breakers, and internal sampling systems.
C. Extraction pipe and heater shielding lagging and stand-offs.
D. Personnel grating, ladder rungs and grab bars.
E. Instrument and accessory support.
F. Temporary fabricating and shipping members.
G. Nameplates and brackets.

9.1.4 Acceptance Levels – Acceptance levels for various types of welds in each of Categories I, II, III are tabulated in Table 14. Refer to Figure 44 for nomenclature.

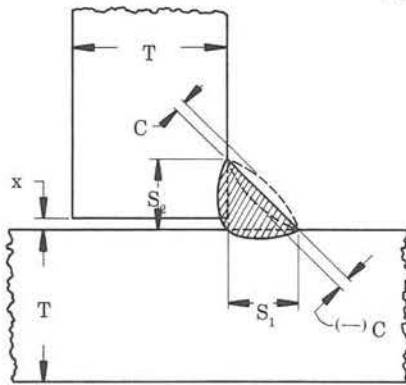
9.2 Surface Preparation Requirements

9.2.1 General Requirements – Surfaces of condensers shall be prepared by the Manufacturer to assure that the equipment will be acceptable from the following aspects:

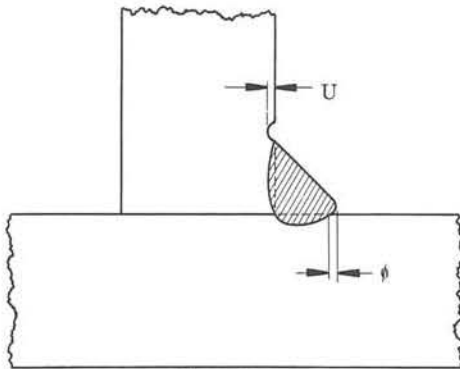
9.2.1.1 Surfaces to be painted will be suitably free from deleterious materials which may affect the adhesion of prime paint coatings.

9.2.1.2 Surfaces such as water boxes which are to be lined or coated with heavy duty corrosion protection coatings are free from irregularities which could cause air pockets, protrusions which could cause local thick spots in thin film coatings, and deleterious materials which would prevent adequate bonding of the corrosion liners. The surface finish shall meet the requirements of the coating or lining system to be used.

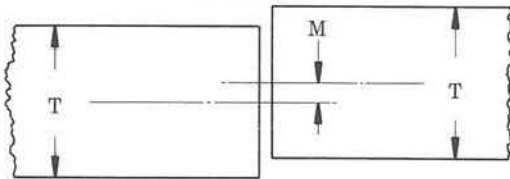
WELDING NOMENCLATURE



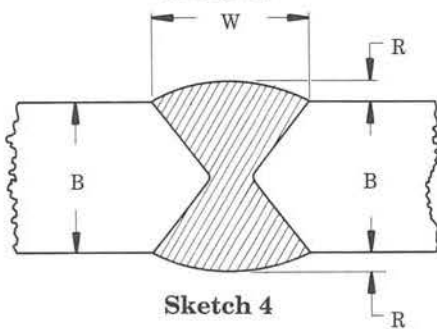
Sketch 1



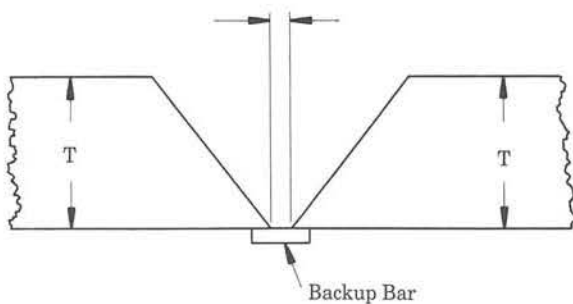
Sketch 2



Sketch 3



Sketch 4



Sketch 5

- S - Min. Specified Weld Leg
- S_1 - Weld Leg Size
- S_2 - Weld Leg Size
- x - Weld Gap (Excluding Local Gouges)
- T - Material Thickness of the Thinnest Member
- C - Convexity
- (-) C - Concavity
- L_w - Length of Weld Described by the Weld Symbol

- ϕ - Overlap
- U - Undercut

- M - Misalignment

- B - Size of Butt Weld
- R - Reinforcement
- L_w - Length of Weld Described by the Weld Symbol
- W - Width of Cap Pass

- G - Root Gap

Figure 44

WELD ACCEPTANCE CRITERIA*

SUBJECT	REFERENCE	ACCEPTANCE CRITERIA		
		CATEGORY I	CATEGORY II	CATEGORY III
Fillet Weld Fit-Up	S Sketch 1	$x \leq \frac{S}{4}$ (1/8" Maximum) See Notes 4 and 5.	For $T < 3"$ $\times \leq 3/16"$ For $T \geq 3"$ $\times \leq 3/16"$ See Notes 4 and 5.	
Fillet Weld Size Minimum	S Sketch 1	$S_{Min} = S - 0"$ See Note 5.	$S_{Min} = S - 1/16"$ For $.1 L_w$ See Note 5.	$S_{Min} = S - 1/16"$ For $.25 L_w$ See Note 5.
Fillet Weld Size Maximum	S Sketch 1	$S_{Max} = S + T$ or $S + 3/8"$ (Whichever is greater) See Note 5.		
Fillet Weld Concavity	(-) C Sketch 1	Concavity shall not encroach upon minimum required weld throat.	$(-)C \leq 1/16"$ for $.1 L_w$ (Not to exceed 3" in any foot.)	$(-)C \leq 1/16"$ for $.25 L_w$ (Not to exceed 6" in any foot.)
Fillet Weld Convexity	C Sketch 1	$C \leq \frac{S}{6} + 1/32"$	$C \leq \frac{S}{6} + 1/16"$	
Weld Overlap	ϕ Sketch 2	None Allowed	$\phi \leq 3/32"$ Depth for a $.1 L_w$ Length of ϕ not to exceed 6" in any 60" length of weld. The fusion line must be visible.	
Butt Weld Fit-Up Misalignment	T Sketch 3	$M = \frac{T}{4}$ (1/4" Maximum)		$M = \frac{T}{4}$ or $1/8"$ (Whichever is greater)
Taper Required for Butt Weld Misalignment	M Sketch 3	3 to 1, Width of weld to M (Both Sides)	3 to 2, Width of weld to M, both sides only when M exceeds $\frac{T}{8}$	Not Required
Butt Weld Root Open with Backup Bar	G Sketch 5	$G_{Min} =$ as qualified The weld geometry should be such as to permit complete joint penetration where required.		

*See Notes

Table 14

WELD ACCEPTANCE CRITERIA*

SUBJECT	REFERENCE	ACCEPTANCE CRITERIA		
		CATEGORY I	CATEGORY II	CATEGORY III
Butt Weld Root Opening Without Backup Bar	G Sketch 5	G_{Min} = as qualified Oversize gaps shall be welded with a technique which will close the gap without excessive bridging of the open root. The weld geometry should be such as to permit complete joint penetration where required.		
Butt Weld Size	B Sketch 4	$B \geq T$ See Note 5.		$B = T - \frac{T}{4}$ ($\frac{T}{4}$ not to exceed 1/8") See Note 5.
Butt Weld Capout Pass Excluding Weld Metal Used For Tapering and Blending	W Sketch 4	W_{Min} = Equal to Width of Weld Geometry $W_{Max} \leq 2-1/2 \times$ thickness of the thicker of the members being joined. Note: Capout may consist of multiple passes.		
Butt Weld Reinforcement on Both Sides	R Sketch 4	When $T \leq 1/2"$ $R = 1/8"$ Max. $T > 1/2"$ or $\leq 1"$, $R = 3/16"$ Max. $T > 1"$ or $\leq 2"$, $R = 1/4"$ Max. $T > 2"$, $R = 5/16"$ Max. Where T is the thinner of the members being joined.		
Undercut	U Sketch 2	1/32" Continuous	1/32" undercut is permitted for full length weld; up to 1/16" for .25 L_w .	
Visible Porosity (Defined as a Hole 1/32" in Diameter or Greater)		No Visible pore 1/32" is allowed.	The sum of the diameters shall not exceed 3/8" in any linear in. or 3/4" in any 12" length of weld. the maximum single pore diameter is 1/8". Linear porosity must be separated by a minimum of one pore diameter.	
Visible Cracks		None Allowed		
Visible Cracks		None Allowed		

*NOTES:

- Highly localized concavity or weld reinforcement such as caused by weld restarts or where welds join or cross another weld are not to be considered unacceptable under these criteria.
- Weld ripples transverse to the direction of electrode travel are permitted providing that the crest-to-trough height is not greater than 3/22" and providing that the weld throat dimension is within acceptable range. Valleys between successive weld beads shall not exceed 1/8" providing that minimum weld size is maintained.
- On weldments made by more than one welder and when combinations of stringer and weave techniques are used, the quality of the weld shall be judged by the visual acceptance criteria defined.
- If fit-up root gap of fillet welds in any category exceeds 1/16", the fillet weld size shall be locally increased (in the area where fit-up is beyond this criteria) by an amount equal to the size of the separation.
- Fit-ups and weld sizes not within these criteria shall be resolved by the Manufacturer and/or as provided in the contract documents.

Table 14 (Continued)

9.2.1.3 Heavy or loose scales, spatter, or other materials which could come off and cause damage or fouling of pumps, filters and valves shall be removed by suitable mechanical methods.

9.2.1.4 Surfaces will have a workmanlike appearance and freedom from scars and protrusions which could cause bodily injury.

9.2.1.5 Preparations for field-welded joints will be reasonably smooth and free of coarse irregularities.

9.2.1.6 The preparations required by this section may be performed at any time in the manufacturing cycle. Rust which develops during manufacture shall be removed prior to painting if it would be detrimental to the paint application. Rust on non-painted surfaces need not be removed. Pre-cleaned material such as pre-blasted plates may be painted prior to fabrication. All accessible paint scars and blemishes shall be retouched prior to shipment. It must be recognized that some touch-up will be required after unloading or installation.

9.2.2 Specific Requirements

9.2.2.1 Table 15 contains the recommended acceptable preparations for various areas and components of the condenser. Each area is evaluated on the basis of preparation required for coatings as well as the ultimate destination of the contained fluids and any particles which may be carried with the flow.

9.2.2.2 The requirements as written apply to the preparation of components and assemblies as built in the Manufacturer's facilities. Final assembly of the apparatus by the erection contractor should meet the applicable sections of Table 15.

9.2.2.3 Purchaser should assure that parts of the components supplied by other than the condenser Manufacturer but which are connected to or installed in the condenser are prepared in similar fashion.

9.2.3 Special Requirements – The requirements of this section represent good practices recommended by the condenser Manufacturer and the paint and coating manufacturers and applicators and, in general, meet the intent of specifications by engineering firms, owners, and Purchasers of this equipment. There will exist, however, exceptions requiring special preparation. There are two basic groups of special requirements.

9.2.4 Purchaser Specified Requirements – If the Purchaser or his agent desire any preparation more stringent (i.e., abrasive blasting) than this Standard, it must be clearly stated in the procurement documents. Guidance for the special finishes can be found in the Structural Steel Painting Council publications.

9.2.5 Manufacturer Specified Requirements

9.2.5.1 The Manufacturer may at any time prepare the equipment in a manner superior to the requirements of Table 15. This improvement is discretionary and could be done to suit the Manufacturer's economic evaluation and/or his processing equipment and schedules.

9.2.5.2 As a minimum, the Manufacturer is required to provide preparation as dictated by the requirements of the painting, coating, or sheet lining process.

9.3 Painting, Coating, Linings and Corrosion Protection

9.3.1 Before leaving the factory, carbon steel condenser external surfaces are to be cleaned and painted with one coat of primer. Touch-up of the primer and application of the finish paint are performed after final field installation by Purchaser.

9.3.2 Condenser internal surfaces do not require primer, paint or rust inhibitors for normal shipment and storage. Rusting of these surfaces is acceptable and is to be expected.

9.3.3 These Standards do not cover the application of any coatings. All such applications shall be done to the requirements of the applicable process.

9.3.4 Cathodic Protection The Purchaser should consider the use of cathodic protection for condenser components that are exposed to cooling water with a relatively high conductivity, particularly sea water or brackish water.

There are two methods to cathodically protect a condenser: sacrificial anodes (galvanic coupling) and impressed current (external power supply). The selection of a system must be made by the Purchaser based on the installation site conditions as well as design and economic considerations. Particular attention should be placed on the condenser materials of construction and the cooling water quality.

The cooling water acts as an electrolyte which promotes galvanic corrosion of the condenser internals when dissimilar materials are present. The parts which have a lower electrochemical potential are subject to attack.

9.3.4.1 The use of a coating to protect these parts may not be sufficient on its own. If a pinhole or discontinuity in the coating occurs exposing a small portion of the base metal to the cooling water, the rate of corrosion at this discontinuity will be much higher than it would be if the entire surface were exposed.

9.3.4.2 A sacrificial cathodic protection system uses anodes made of zinc, magnesium, aluminum, carbon steel, or some other metal having a relatively low electrical potential when compared to the base metal to be protected. The consumption of the anodes spares the base metal components from attack. The anodes should be connected to the parts they are intended to protect. The material, quantity, and size of the anodes is dependent on the materials of construction and size of the condenser and the cooling water quality.

9.3.4.3 An impressed current system uses an external DC power supply with an inert anode to maintain an electrical potential which protects the base metal from attack. Proper operation of the system is essential and is the responsibility of the Purchaser.

9.4 Quality Assurance – The Manufacturer shall have an organized and operational Quality Assurance program for steam surface condensers. This program shall be outlined in a Quality Assurance Manual which will be available to the Purchaser and his

representatives upon request. The system shall provide for control of quality in both the Manufacturer's plant and that of any subcontractor fabricating parts. Field Quality Assurance is the responsibility of the Purchaser and/or his agent. The party responsible for the field installation should have a quality assurance program comparable to the seller. Review of this quality assurance program shall be the responsibility of the Purchaser.

The Quality Assurance program shall provide for assurance of compliance with, but not limited to, the Manufacturer's and HEI Standards, which provide as a minimum:

- (1) Base material control (e.g., plate and weld wire).
- (2) Controlled distribution of material and weld wire.
- (3) Control of purchasing requirements.
- (4) Written weld procedures which have been qualified in a manner comparable to that defined in Section IX of ASME Unfired Pressure Vessel Code.
- (5) Qualification of welders and control of welder assignments to assure personnel doing welding are qualified.
- (6) System for audit of control of procedures.
- (7) Detail drawings contain definition of size and applicable weld procedures.

9.5 Dimensional Tolerances – Refer to Figure 45 and 45M for details.

9.6 Shipping and Site Storage

9.6.1 Preparation for Shipment

9.6.1.1 Condenser shipments are of different sizes and shapes, therefore, some components may not be covered specifically in this guideline.

9.6.1.2 Condenser connections shall be protected for transportation and site storage.

9.6.1.3 Tube bundles that are exposed to the elements shall be protected for transportation and site storage.

9.6.1.4 Machined surfaces should be coated with a suitable protective coating and covered to prevent corrosion and damage.

9.6.1.5 Small parts shall be properly packaged, skidded, boxed, or containerized to prevent damage while in transit and during site storage.

9.6.1.6 Skids, boxes, containers, etc. shall have adequate marking for unit weight and lift points.

9.6.1.7 Overseas shipments may require special packaging and protection. Manufacturer will coordinate this with the Purchaser to assure safe delivery of the components.

9.6.2 Transportation

9.6.2.1 Typical transportation means are by truck, rail, barge, or ocean freight.

9.6.2.2 Components must be properly tied down in accordance with the Manufacturer and transport com-

pany's procedures prior to equipment movement.

9.6.2.3 Extra heavy loads and oversized components may require special escort and special permits during ground transportation. This work should be coordinated with the transport company prior to delivery.

9.6.3 Site Offloading and Storage

9.6.3.1 Appropriate lift points provided by the Manufacturer must be used for offloading of components at the site. Special spreader beams may be required to handle extra heavy and oversized loads. Purchaser is responsible for this item and should consult with the Manufacturer for proper handling procedures. Lifting arrangement should be in accordance with Manufacturer guidelines.

9.6.3.2 Purchaser must inspect all components upon receipt at the site and must immediately report any damage to the Manufacturer.

9.6.3.3 Components shall be stored in a secure lay down or storage area and shall be adequately supported.

9.6.3.4 Accessories and miscellaneous parts shall be stored indoors in proper warehouses to protect the equipment from elements or environment. These parts must be placed on suitable cribbing.

9.6.3.5 Components which are stored outdoors must have proper cribbing to protect from any water damage. Lay down area must have adequate load carrying capacity with proper drainage provisions.

9.6.3.6 Exposed tube bundles shall be covered to prevent water and outside elements from entering the bundles. Exercise additional cares when freezing temperatures are expected.

9.6.3.7 Care must be taken to prevent damage during site storage and to protect all components from contamination.

9.6.3.8 If the Purchaser intends to store the condenser for long periods of time, proper procedures must be in place for routine inspection. Consult with the Manufacturer for any special requirements.

9.7 Field Installation

9.7.1 This section provides general information on the field installation of a steam surface condenser. Due to jobsite space limitations, available erection facilities, and shipping constraints, it is not possible to provide an exact erection sequence. The Purchaser may consult with the Manufacturer for general guidelines.

9.7.2 The Purchaser must ensure that all condenser components can be safely erected at the jobsite.

9.7.3 The Purchaser must follow directions and precautions included in Manufacturer's installation, operation, and maintenance manuals, drawings, and other documents furnished by the Manufacturer, and auxiliary equipment suppliers.

9.7.4 The condenser should be erected on a clean level surface as close as possible to the condenser foundation to minimize handling.

9.7.5 The condenser should not be lifted or jacked in any area unless designated by the Manufacturer.

9.7.6 During the erection of the condenser, all components must be protected against damage. Particular attention should be given to protection of tubes and tubesheets from arc strikes, falling objects, flame cutting, debris, etc.

9.7.7 Waterbox internal coatings must be protected at all times. No welding should be performed on waterboxes if an internal coating has been applied.

9.7.8 The condenser internals should be thoroughly cleaned after all erection work has been completed.

9.8 Erection Superintendent Duties – The Manufacturer may provide the services of an Erection Superintendent to advise the Purchaser in the proper installation and start up of the surface condenser and accessories in accordance with the erection drawings and procedures.

The Erection Superintendent will comply with all jobsite regulations. Unless otherwise provided for

in the purchase order, the Erection Superintendent shall not be responsible for the following:

- The supervision of the erection crew
- Fit up and weld quality
- The health and safety of the erection crew
- The schedule of erection and work progress

In the event of any conflict between the Manufacturer's requirements and site practice, the Erection Superintendent will bring such conflicts to the attention of the Purchaser's Designated Representative.

9.9 Post Erection Walk Down – The Manufacturer may provide the services of the Manufacturer's Representative to perform a post installation walk down of the surface condenser and accessories.

Unless otherwise provided for in the purchase order, the Manufacturer's Representative will be responsible for visually inspecting the installation of the surface condenser and accessories for fit-up, weld quality and general compliance with Manufacturer's erection drawings.

Unless otherwise provided for in the purchase order, the Manufacturer's Representative shall not be responsible for the following:

- NDE of field welds
- Examination of welds, weldments and components not readily accessible or well lighted

CONDENSER SURFACE PREPARATION REQUIREMENTS*

Condition	Component	Pressure Boundary		Water Box Internal and Circulating Water Piping			Non-Pressure Boundary	Shielding and Extraction Pipe		Field Weld Preparations	
		External	Internal	Uncoated	Sheet Material Coated and Heavy Film-Coatings	High Build Epoxy Type Coatings	Surfaces	Internal	External		
I	Weld Surfaces	As Welded (Note 3)			As Required by Process	As Required by Process	As Welded	As Welded	As Welded	N/A	
II	General Surface Condition	Mechanically Hand-Cleaned to SSPC-2					Same as Pressure Boundary	Same as Pressure Boundary	Same as Pressure Boundary (Note 7)	Mechanically De-Slagged	
III	Indentations (Note 1)	Depth to be the Smaller of 0.2T or 1/8" Does not apply to tubesheet and main expansion members			Depth to be Smaller of 0.2T or 1/8". Blend into Adjacent Metal	Depth to be Smaller of 0.1T or 1/16". Blend Sharp Outside Edges and Corners	Depth to be the Larger of 0.2T or 1/2".	Max. Depth is 1/32"		Max. Depth is 1/3T. Enclosed angle must be > 120°	
IV	Residual Weld Metal and Protrusions (Note 1)	Max. Height = 1/8". Dress as necessary to assure good paint coverage.	Max. Height not to exceed 7/2.		Blend into Adjacent Material		Max. Height not to exceed 7/2.	Max. Height not to exceed 7/2.		N/A	
V	Arc Strikes	Remove All Arc Strikes from High Alloy Materials									N/A
VI	Weld Spatter	Spatter Not Removable by Manual Scraping Wire Brushing Is Acceptable			Remove As Required by the Applicable Process Cleaning to SSPC-2 is Acceptable.		Spatter Not Removable by Manual Scraping or Wire Brushing Is Acceptable			N/A	
VII	Mill Scale	Mill Scale Not Removable by Manual Cleaning to SSPC-2 is Acceptable.					Mill Scale Not Removable by Manual			N/A	
VIII	General Condition of Components or Sub-Assemblies	Loose dirt, particles, excessive, rust, oils, and general contaminants shall be removed by brushing, air blowing, and/or suitable solvents to produce a workmanlike appearance.							N/A		

***NOTES:**

1. Maximum length of any single indentation is 2 inches. Maximum width is 1/8 inch.
2. Maximum length of any single protrusion is 4 inches. Maximum width is 1/2 inch. Sharp protrusions which may cause injury must be dressed regardless of size.
3. "As welded" means meeting the visual inspection criteria contained elsewhere in this Standard.
4. Blasting when specified in this table or when performed at the manufacturer's option shall be done with shot, sand or grit. Special blasting media may be used for specialty coatings, liners, or paints only with the approval of the coater and purchaser. When blasting surfaces in contact with steam or condensate only silica free shot or grit is permitted.
5. Materials purchased are expected to have imprints from the raw material processing. Indentations, lumps, etc., from roller, forging, and other forming operations are not objectionable and will be accepted providing the raw material specification is not violated.
6. Discoloration from the heat of welding, thermal cutting, and heat treatments are acceptable. Heavy oxides and scaling caused by these processes will be removed by manual wire brushing or similar processes.
7. Stainless steel shielding requires only mechanical cleaning or solvent cleaning. Blasting and pickling are not required.
8. Solvent cleaning as referenced in many sections of the SSPC Standard is not required as a general cleaning method. It shall be used only for local cleaning of oily type contaminants.
9. The thickness of the base material in Table 15 is referred to as "T".

Table 15

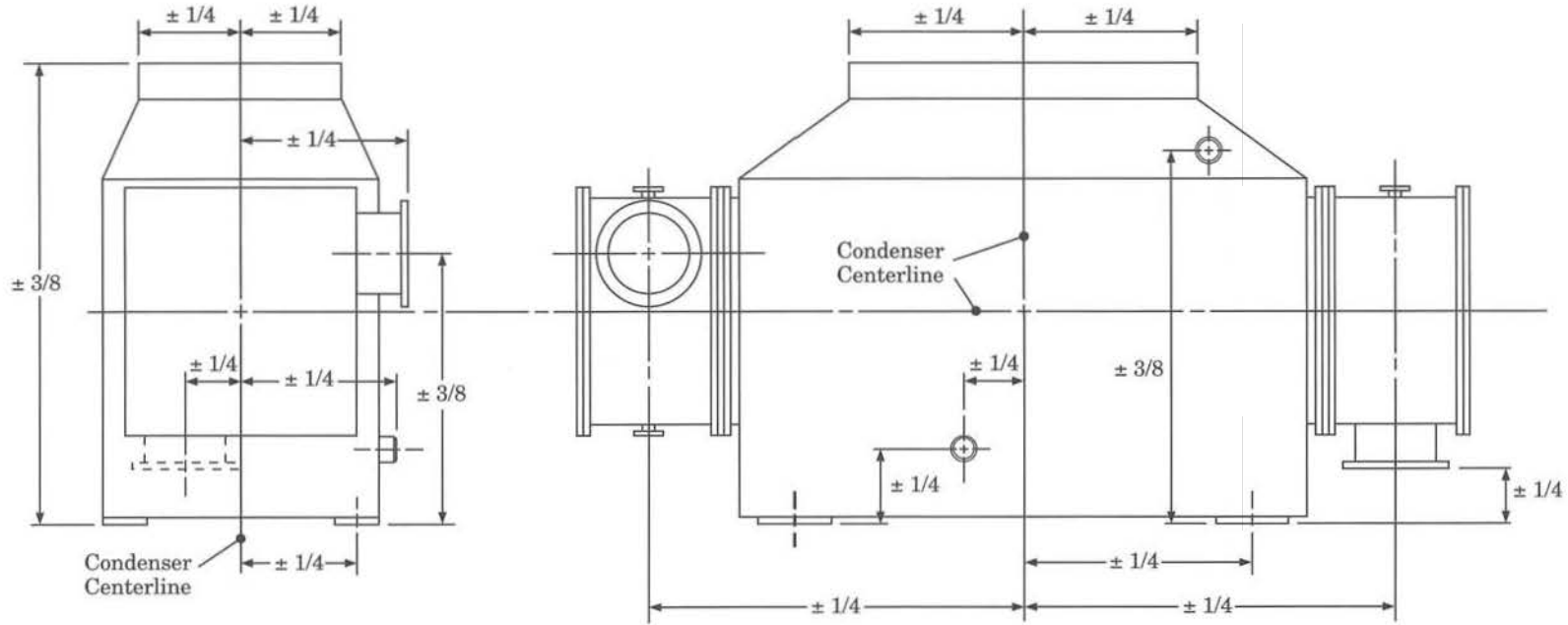
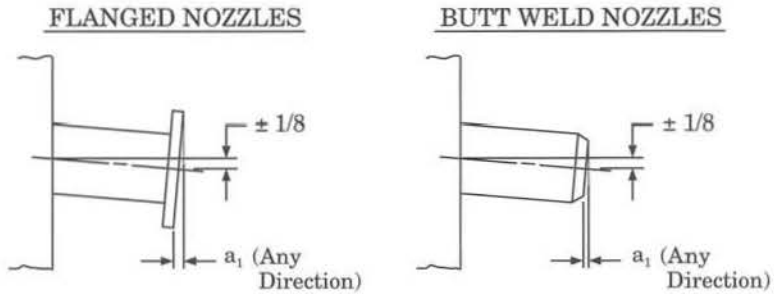


Figure 45



	<u>a_1 max.</u>
$D_p \leq 4$	1/8
$4 < D_p \leq 14$	3/16
$14 < D_p \leq 36$	1/4
$36 < D_p$	5/16
D_p = Nominal pipe size	

General Notes:

1. All dimensions in inches.
2. Vertical dimensional tolerances from bottom of condenser support feet.

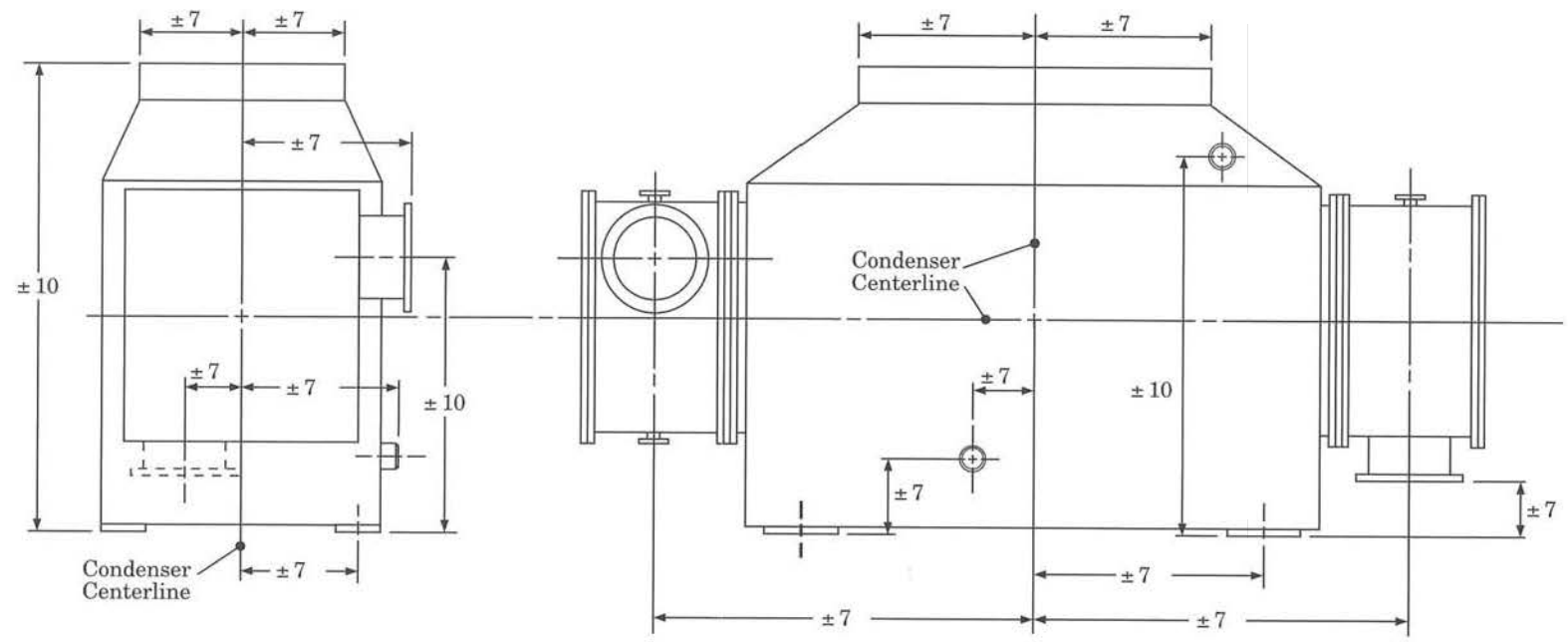
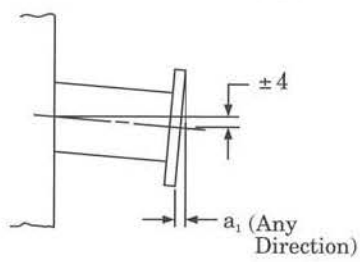
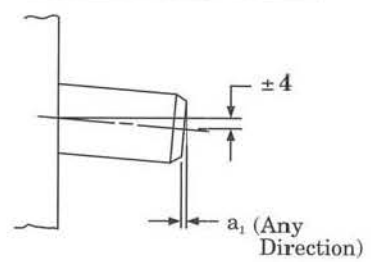


Figure 45M

FLANGED NOZZLES



BUTT WELD NOZZLES



	<u>a_1 max.</u>
$D_p \leq 100$	4
$100 < D_p \leq 350$	5
$350 < D_p \leq 900$	7
$900 < D_p$	8

D_p = Nominal pipe size

General Notes:

1. All dimensions in mm.
2. Vertical dimensional tolerances from bottom of condenser support feet.

APPENDIX A TYPICAL SPECIFICATION FOR STEAM SURFACE CONDENSERS

A1.0 GENERAL COMMENTS

A1.1 This Appendix is intended solely as a guide to assist in writing specifications for steam surface condensers. It is for the convenience of Purchasers who wish to incorporate parts or all of it in their own specifications.

A1.2 In the procurement of condensers, it is very important that the Purchaser set forth the conditions to be met by the unit so that:

1. The equipment is of the proper capacity, type, and quality to perform successfully the functions for which it is specified
2. The performance characteristics are proper to meet the Purchaser's specific service conditions

3. The accessories to be included are adequate to meet the requirements of service for which the unit is specified
4. The highest degree of safety in operation and continuity of service is obtained

A1.3 The design and construction of air removal equipment shall conform to the latest *Heat Exchange Institute Standards for Steam Jet Vacuum Systems* and *Performance Standards for Liquid Ring Vacuum Pumps*.

A1.4 The design capacity of air removal equipment shall conform to the latest *Heat Exchange Institute Standards for Steam Surface Condensers*.

A2.0 CONDENSER DESCRIPTION

A2.1 Type and Arrangement – The condenser shall be of the horizontal tube surface type. Materials of construction should be in accordance with those referenced by the latest *Heat Exchange Institute Standards for Steam Surface Condensers*.

A2.2 Shell – The design shall be adequate for the condition under which the unit shall be operated and shall remain tight under any operating vacuum. The shell shall be hydrostatically tested in accordance with *Heat Exchange Institute Standards for Steam Surface Condensers*. Suitable supporting feet shall be provided. There shall be openings for admission or removal of steam, air, vents, and returns as specified. Where impingement on the tubes may cause erosion, internal protection shall be provided. Suitable means shall be incorporated in the design to provide for the differences in thermal expansion between shell and tubes.

A2.3 Water Boxes – Water inlets and outlets and other openings shall be sized and located in accordance with good engineering practice and plant layout. The water boxes and cover plates if required shall have provisions for lifting. A sufficient number of openings shall be provided of suitable size and location to provide access for maintenance and inspection. Water boxes shall be designed for the specified pressure and hydrostatically tested in accordance with *Heat Exchange Institute Standards for Steam Surface Condensers*.

A2.4 Tube Support Plates – The tube support plates shall be of sufficient number and proper spacing as indicated by the *Heat Exchange Institute*

Standards for Steam Surface Condensers to provide adequate support for the tubes. Holes for the tubes shall be drilled in accordance with paragraph 8.2.9.1.

A2.5 Tubesheets – The tubesheets shall be drilled and prepared to receive the tubes in accordance with paragraph 8.2.9.2.

A2.6 Hotwell – The hotwell, with capacity as specified, of the same material as the shell, may be furnished as an integral part of or connected to the bottom of the condenser shell.

A2.7 Miscellaneous – The Manufacturer shall provide connections on the condenser for all instruments as specified.

A2.8 Transition – The arrangement shall be adequate to provide a smooth transition between the turbine exhaust and the condenser tube bundles. Provisions shall be provided for feedwater heaters, boiler feed pump turbine exhausts, turbine by-pass piping or other devices as specified. Materials of construction shall be of the same material as the shell.

A2.9 Tubes – Tube material and gauge is as specified by the Purchaser. Tubes shall be shop (field) installed.

A2.10 Steam Inlet Expansion Joint – The preferred type expansion joint is (Stainless steel, Rubber belt, Rubber U, other) as specified.

A3.0 SPECIFICATION CONTENT

A3.1 In view of the complexity of most specifications it is important that sufficient time (a minimum of four weeks) be allowed for preparation of designs at the proposal stage. Proposals shall be submitted on or before _____ in hard copy or electronic format and addressed to:

A3.2 The intent of this specification is to state the requirements and conditions for the furnishing and delivery of steam surface condensers as hereinafter specified.

A3.3 The design and construction of the condenser shall conform to the latest standards of the *Heat Exchange Institute Standards for Steam Surface Condensers*.

A3.4 The Manufacturer shall submit, with his proposal, drawings with approximate dimensions and weights, as well as descriptive material and data sheets which adequately describe the proposed design.

A3.5 The materials and workmanship used in the construction of all apparatus shall be of industry standard quality.

A3.6 Purchaser shall be responsible for supply of the following:

- Concrete foundations
- Anchor bolts, washers, sole plates and grouting
- All necessary building openings through which equipment is to be passed, as well as shoring of floors and items of similar building alteration which may be necessary because of the size and weight of the condenser equipment
- Labor and material for unloading, handling, and installation of equipment
- All piping, valves, and fittings, including those between condenser and auxiliaries
- All electrical wiring and motor controls
- All instruments and gauges
- All supplies and operating personnel necessary for condenser startup
- Circulating water chemistry with material selection
- Turbine allowable loads

A3.7 The equipment shall be installed at:

plant name: _____

Unit number: _____

Turbine capability: _____ mw _____
Plant truck delivery
address: _____

Plant rail delivery address: _____

Delivering railroads: _____

A railroad siding does (does not) extend into the building and is (is not) under the station crane. The equipment must be moved _____ feet (miles) from the railroad cars to the _____ foundation. The method of unloading and movement of the condenser components is as follows:

A3.8 Shipping Terms shall be: _____
(Incoterms 2000 or other).....

(i.e. Ex Works, FCA Delivered, FAS Port of Export, DDU, etc.) **A3.9** After order award, the Manufacturer shall provide the Purchaser with outline drawings of the condenser for approval. Also, the Purchaser will provide copies of power plant reference drawings to the Manufacturer for use in the design and fabrication of the condenser. These will include drawings such as: foundations, turbine exhaust details, circulating water piping, turbine bypass pipe, etc.

A4.0 CONDENSER THERMAL DATA

Number of shells / Number of exhaust openings		/
Pressure - (Single) (multiple)		
Steam condensed - design		lb/hr
Steam condensed - maximum		lb/hr
Heat rejected to circulating water (design total)		Btu/hr
Turbine bypass conditions at condenser connection		
HP:		lb/hr, Btu, °F, psia
LP:		lb/hr, Btu, °F, psia
Condenser pressure (absolute)		in HgA
Inlet circulating water temperature (design)		°F
Inlet circulating water temperature (range)	/	°F / °F
Circulating water flow - design		gpm
Circulating water nozzle size		in
Circulating water velocity in tubes - design		ft/sec
Circulating water source:		
(River) (Cooling tower) (Ocean) (Well) (Other)		
Circulating water chemistry		
Tube side pressure drop		psi
Cleanliness factor		%
Oxygen in condensate (maximum) (42) (14) (7)		ppb
Condenser surface		ft²
Number of water passes - (one) (two) (other)		
Tubes: Furnished by (Purchaser) (Manufacturer)		
Outside Diameter		in
Length - effective / total	/	ft / ft
Direction of tube withdrawal		
Tube end preparation (rolled) (welded)		
Tubes material / wall thickness:		
Gas Cooler	/	/BWG
Main Bank	/	/BWG
Periphery	/	/BWG
Additional requirements		

A5.0 CONDENSER MECHANICAL DATA

Tube sheet material		
Shell material		
Water box type (divided) (non-divided)		
Water box material		
Water box internal coating (epoxy) (neoprene) (none) (other)		
Water box pressure design / test	/	psig
For a siphon arrangement of the circulating water system:		
Minimum elevation of discharge water level		ft
Hotwell arrangement (divided) (non-divided)		
Hotwell capacity		gal
Hotwell retention time		min
Hotwell special requirements		
Condensate outlets - quantity / size	/	# / in
List of all service connections including: size, quantity, material, flow, pressure		
enthalpy, temperature, frequency		
LP Turbine information:		
Exhaust connection (flanged) (welded)		
Turbine exhaust dimensions	x	ft
Allowable exhaust load X / Y / Z	/ /	lbs
Movement - lateral / axial	/	in / in
Condenser tube axis referred to turbine shaft axis:		
(parallel) (perpendicular)		
Elevation and dimensional information:		
Turbine room floor elevation		ft
Turbine exhaust connection elevation		ft
Basement floor or condenser pit elevation		ft
Turbine foundation beam depth		ft
Condenser top tube elevation (maximum)		ft
Condenser bottom tube elevation (minimum)		ft
Condenser Width (maximum allowable)		ft
Condenser Height (maximum allowable)		ft
Condenser Length (maximum allowable)		ft
Additional special requirements		

A6.0 ASSOCIATED EQUIPMENT

A6.1 LP Turbine / condenser expansion provisions:

(Spring supported) (expansion joint) (solid mounted)

A6.2 Turbine / condenser expansion joint:

(rubber) (stainless steel)

A6.3 Provisions for feed water heaters located in transition section:

Number / Diameter: _____ / _____

(Supports) (Closing plates) (Lagging)

A6.4 LP Turbine extraction:

Pipes: number / size _____ / _____

(Lagging) (Supports) (Expansion joints)

A6.5 Air removal equipment:

(vacuum pumps) (steam jet air ejectors) (hybrid pump / ejector system)

A6.6 Pressure relief device:

(rupture disc) (atmospheric relief valve)

A6.7 Vacuum breaker valve

Actuated (manual) (electric motor) (pneumatic)

A6.8 Instrumentation, provide a complete list

A6.9 Water box accessories:

Circulating water expansion joints (rubber arch type)

Gauge glasses

Cathodic protection

Continuous tube cleaning system

Priming system

Air release valves

A6.10 Additional equipment or accessories:

APPENDIX B

METRIC CONVERSION FACTORS

NOMENCLATURE

NAME		SYMBOL	OTHER UNITS
inch/inches		in	
foot/feet		ft	
meter	(SI)	m	
millimeter		mm	
square inch		in ²	
square foot		ft ²	
square meter	(SI)	m ²	
square centimeter		cm ²	
square millimeter		mm ²	
cubic inch		in ³	
cubic foot		ft ³	
gallon (US liquid)		gal	
cubic meter	(SI)	m ³	
liter		L	
pound mass (avoirdupois)		lbm	
kilogram	(SI)	kg	
pound force (avoirdupois)		lbf	
kilogram force		kgf	
newton	(SI)	N	m · kg/s ²
degree Fahrenheit		°F	
degree Kelvin	(SI)	K	
degree Celsius	(SI)	°C	
British thermal unit (International Table)		Btu	
kilocalorie (International Table)		kcal	
joule	(SI)	J	N · m, m ² · kg/s ²
kilojoule		kJ	
second (customary)		sec	
second	(SI)	s	
minute		min	
hour (customary)		hr	
hour (metric)		h	
watt	(SI)	W	J/s, N · m/s, m ² · kg/s ³
pound force/square inch		psi	lbf/in ²
inches of mercury		in Hg	
feet of water		ft H ₂ O	
pascal	(SI)	Pa	N/m ² , kg/(m · s ²)
kilopascal		kPa	
bar		bar	
millimeter of mercury		mmHg	
torr		torr	
centipoise		cp	

Notes:

1. (SI) Denotes an "International System of Units" unit.
2. Pressure should always be designated as gage or absolute.
3. The acceleration of gravity, *g*, is taken as 9.80665 m/s².
4. One gallon (U S liquid) equals 231 in³

PREFIXES DENOTING DECIMAL MULTIPLES OR SUBMULTIPLES

PREFIX	SYMBOL	MULTIPLICATION FACTOR
micro	μ	$0.000\ 001 = 10^{-6}$
milli	m	$0.001 = 10^{-3}$
centi	c	$0.01 = 10^{-2}$
deci	d	$0.1 = 10^{-1}$
deca	da	$10 = 10^1$
hecto	h	$100 = 10^2$
kilo	k	$1\ 000 = 10^3$
mega	M	$1\ 000\ 000 = 10^6$
giga	G	$1\ 000\ 000\ 000 = 10^9$

CONVERSION FACTORS

LENGTH			
MULTIPLY	BY		TO OBTAIN
in	2.540	$\times 10^{-2}$	m (SI)
in	2.540	$\times 10^1$	mm
ft	3.048	$\times 10^{-1}$	m (SI)
ft	3.048	$\times 10^2$	mm

AREA			
MULTIPLY	BY		TO OBTAIN
in ²	6.451600	$\times 10^{-4}$	m ² (SI)
in ²	6.451600	$\times 10^2$	mm ²
ft ²	9.290304	$\times 10^{-2}$	m ² (SI)
ft ²	9.290304	$\times 10^4$	mm ²

VOLUME			
MULTIPLY	BY		TO OBTAIN
in ³	1.638706	$\times 10^{-5}$	m ³ (SI)
in ³	1.638706	$\times 10^{-2}$	L
ft ³	2.831685	$\times 10^{-2}$	m ³ (SI)
ft ³	2.831685	$\times 10^1$	L
gal	3.785412	$\times 10^{-3}$	m ³ (SI)
gal	3.785412		L

MASS			
MULTIPLY	BY		TO OBTAIN
lbm	4.535924	$\times 10^{-1}$	kg (SI)

FORCE			
MULTIPLY	BY		TO OBTAIN
lbf	4.448222		N (SI)
lbf	4.535924	$\times 10^{-1}$	kgf
kgf	9.806650		N (SI)

TEMPERATURE

$$K = (^{\circ}F + 459.67)/1.8 \quad (SI)$$

$$^{\circ}C = (^{\circ}F - 32)/1.8 \quad (SI)$$

$$^{\circ}F = 1.8 K - 459.67$$

$$^{\circ}F = 1.8 ^{\circ}C + 32$$

ENERGY, WORK OR QUANTITY OF HEAT			
MULTIPLY	BY		TO OBTAIN
Btu	1.055056	$\times 10^3$	J (SI)
Btu	2.519958	$\times 10^{-1}$	kcal
ft·lbf	1.355818		J (SI)
ft·lbf	3.238316	$\times 10^{-4}$	kcal

POWER (ENERGY/TIME)			
MULTIPLY	BY	TO OBTAIN	
Btu/hr	2.930711×10^{-1}	W	(SI)

PRESSURE OR STRESS (FORCE/AREA)			
MULTIPLY	BY	TO OBTAIN	
psi	6.894757×10^3	Pa	(SI)
psi	6.894757	kPa	
psi	6.894757×10^{-2}	bar	
psi	7.030696×10^{-2}	kgf/cm ²	
lbf/ft ²	4.788026×10^1	Pa	(SI)
lbf/ft ²	4.788026×10^{-2}	kPa	
lbf/ft ²	4.882428	kgf/m ²	
inHg (32°F)	3.38638×10^3	Pa	(SI)
inHg (32°F)	3.38638	kPa	
inHg (32°F)	3.38638×10^{-2}	bar	
inHg (32°F)	3.45315×10^{-2}	kgf/cm ²	
inHg (32°F)	2.540×10^1	mmHg	
torr (0°C)	1.33322×10^2	Pa	(SI)
torr (0°C)	1.0	mmHg	
ft water (39.2°F)	2.98898×10^3	Pa	(SI)
ft water (39.2°F)	2.98898	kPa	
ft water (39.2°F)	3.047915×10^2	kgf/m ²	
ft water (39.2°F)	2.98898×10^{-2}	bar	

VELOCITY (LENGTH/TIME)			
MULTIPLY	BY	TO OBTAIN	
ft/sec	3.048000×10^{-1}	m/s	(SI)
ft/min	5.080000×10^{-3}	m/s	(SI)

MASS FLOW RATE (MASS/TIME)			
MULTIPLY	BY	TO OBTAIN	
lbm/hr	1.259979×10^{-4}	kg/s	(SI)
lbm/hr	4.535924×10^{-1}	kg/h	

VOLUME FLOW RATE (VOLUME/TIME)			
MULTIPLY	BY	TO OBTAIN	
ft ³ /min	4.719474×10^{-4}	m ³ /s	(SI)
ft ³ /min	1.699011	m ³ /h	
gal/min	6.309020×10^{-5}	m ³ /s	(SI)
gal/min	2.271247×10^{-1}	m ³ /h	
gal/min	3.785412	L/min	

MASS VELOCITY (MASS/TIME-AREA)			
MULTIPLY	BY	TO OBTAIN	
lbm/(hr · ft ²)	1.35623×10^{-3}	kg/(s · m ²)	(SI)
lbm/(hr · ft ²)	4.882428	kg/(h · m ²)	
lbm/(sec · ft ²)	4.882428	kg/(s · m ²)	(SI)

SPECIFIC VOLUME (VOLUME/MASS)			
MULTIPLY	BY	TO OBTAIN	
ft ³ /lbm	6.242797×10^{-2}	m ³ /kg	(SI)
ft ³ /lbm	6.242797×10^1	L/kg	
gal/lbm	8.345406×10^{-3}	m ³ /kg	(SI)
gal/lbm	8.345406	L/kg	

DENSITY (MASS/VOLUME)			
MULTIPLY	BY	TO OBTAIN	
lbm/in ³	2.767990×10^4	kg/m ³	(SI)
lbm/in ³	2.767990×10^1	kg/L	
lbm/ft ³	1.601846×10^1	kg/m ³	(SI)
lbm/ft ³	1.601846×10^{-2}	kg/L	
lbm/gal	1.198264×10^2	kg/m ³	(SI)
lbm/gal	1.198264×10^{-1}	kg/L	

ENTHALPY (ENERGY/MASS)			
MULTIPLY	BY	TO OBTAIN	
Btu/lbm	2.326000×10^3	J/kg	(SI)
Btu/lbm	2.326000	kJ/kg	
Btu/lbm	5.555556×10^{-1}	kcal/kg	

HEAT CAPACITY AND ENTROPY (ENERGY/MASS-TEMPERATURE)			
MULTIPLY	BY	TO OBTAIN	
Btu/(lbm · °F)	4.186800×10^3	J/(kg · K)	(SI)
Btu/(lbm · °F)	4.186800	kJ/(kg · K)	
Btu/(lbm · °F)	1.000000	kcal/(kg · °C)	

THERMAL CONDUCTIVITY (ENERGY-LENGTH/TIME-AREA-TEMPERATURE)			
MULTIPLY	BY	TO OBTAIN	
Btu · in/(hr · ft ² · °F)	1.442279×10^{-1}	W/(m · K)	(SI)
Btu · in/(hr · ft ² · °F)	1.240137×10^{-1}	kcal · m/(h · m ² · K)	
Btu · ft/(hr · ft ² · °F)	1.730735	W/(m · K)	(SI)
Btu · ft/(hr · ft ² · °F)	1.488164	kcal · m/(h · m ² · K)	

DYNAMIC VISCOSITY (MASS/TIME-LENGTH OR FORCE-TIME/AREA)			
MULTIPLY	BY	TO OBTAIN	
cp	1.000000×10^{-3}	Pa · s	(SI)
lbm/(hr · ft)	4.133789×10^{-4}	Pa · s	(SI)
lbm/(hr · ft)	4.133789×10^{-1}	cp	
lbm/(sec · ft)	1.488164	Pa · s	(SI)
lbm/(sec · ft)	1.488164×10^3	cp	
lbf · sec/ft ²	4.788026×10^1	Pa · s	(SI)
lbf · sec/ft ²	4.788026×10^4	cp	

HEAT FLUX DENSITY (ENERGY/TIME-AREA)			
MULTIPLY	BY	TO OBTAIN	
Btu/(hr · ft ²)	3.154591	W/m ²	(SI)
Btu/(hr · ft ²)	2.712460	kcal/(h · m ²)	

HEAT TRANSFER COEFFICIENT (ENERGY/TIME-AREA-TEMPERATURE)			
MULTIPLY	BY	TO OBTAIN	
Btu/(hr · ft ² · °F)	5.678263	W/(m ² · K)	(SI)
Btu/(hr · ft ² · °F)	4.882428	kcal/(h · m ² · K)	

FOULING RESISTANCE (TIME-AREA-TEMPERATURE/ENERGY)			
MULTIPLY	BY	TO OBTAIN	
hr · ft ² · °F/Btu	1.761102×10^{-1}	m ² · K/W	(SI)
hr · ft ² · °F/Btu	2.048161×10^{-1}	h · m ² · K/kcal	

APPENDIX C **AREAS OF CIRCULAR SEGMENTS**

h/D	C	h/D	C	h/D	C	h/D	C	h/D	C	h/D	C	h/D	C	h/D	C	h/D	C
0.001	0.00004	0.050	0.01468	0.100	0.04087	0.150	0.07387	0.200	0.11182	0.250	0.15355	0.300	0.19817	0.350	0.24498	0.400	0.29337
.002	.00012	.051	.01512	.101	.04148	.151	.07459	.201	.11262	.251	.15441	.301	.19908	.351	.24593	.401	.29435
.003	.00022	.052	.01556	.102	.04208	.152	.07531	.202	.11343	.252	.15528	.302	.20000	.352	.24689	.402	.29533
.004	.00034	.053	.01601	.103	.04269	.153	.07603	.203	.11423	.253	.15615	.303	.20092	.353	.24784	.403	.29631
.005	.00047	.054	.01646	.104	.04330	.154	.07675	.204	.11504	.254	.15702	.304	.20184	.354	.24880	.404	.29729
.006	.00062	.055	.01691	.105	.04391	.155	.07747	.205	.11584	.255	.15789	.305	.20276	.355	.24976	.405	.29827
.007	.00078	.056	.01737	.106	.04452	.156	.07819	.206	.11665	.256	.15876	.306	.20368	.356	.25071	.406	.29926
.008	.00095	.057	.01783	.107	.04514	.157	.07892	.207	.11746	.257	.15964	.307	.20460	.357	.25167	.407	.30024
.009	.00113	.058	.01830	.108	.04576	.158	.07965	.208	.11827	.258	.16051	.308	.20553	.358	.25263	.408	.30122
.010	.00133	.059	.01877	.109	.04638	.159	.08038	.209	.11908	.259	.16139	.309	.20645	.359	.25359	.409	.30220
.011	.00153	.060	.01924	.110	.04701	.160	.08111	.210	.11990	.260	.16226	.310	.20738	.360	.25455	.410	.30319
.012	.00175	.061	.01972	.111	.04763	.161	.08185	.211	.12071	.261	.16314	.311	.20830	.361	.25551	.411	.30417
.013	.00197	.062	.02020	.112	.04826	.162	.08258	.212	.12153	.262	.16402	.312	.20923	.362	.25647	.412	.30516
.014	.00220	.063	.02068	.113	.04889	.163	.08332	.213	.12235	.263	.16490	.313	.21015	.363	.25743	.413	.30614
.015	.00244	.064	.02117	.114	.04953	.164	.08406	.214	.12317	.264	.16578	.314	.21108	.364	.25839	.414	.30712
.016	.00268	.065	.02166	.115	.05016	.165	.08480	.215	.12399	.265	.16666	.315	.21201	.365	.25936	.415	.30811
.017	.00294	.066	.02215	.116	.05080	.166	.08554	.216	.12481	.266	.16755	.316	.21294	.366	.26032	.416	.30910
.018	.00320	.067	.02265	.117	.05145	.167	.08629	.217	.12563	.267	.16843	.317	.21387	.367	.26128	.417	.31008
.019	.00347	.068	.02315	.118	.05209	.168	.08704	.218	.12646	.268	.16932	.318	.21480	.368	.26225	.418	.31107
.020	.00375	.069	.02366	.119	.05274	.169	.08779	.219	.12729	.269	.17020	.319	.21573	.369	.26321	.419	.31205
.021	.00403	.070	.02417	.120	.05338	.170	.08854	.220	.12811	.270	.17109	.320	.21667	.370	.26418	.420	.31304
.022	.00432	.071	.02468	.121	.05404	.171	.08929	.221	.12894	.271	.17198	.321	.21760	.371	.26514	.421	.31403
.023	.00462	.072	.02520	.122	.05469	.172	.09004	.222	.12977	.272	.17287	.322	.21853	.372	.26611	.422	.31502
.024	.00492	.073	.02571	.123	.05533	.173	.09080	.223	.13060	.273	.17376	.323	.21947	.373	.26708	.423	.31600
.025	.00523	.074	.02624	.124	.05600	.174	.09155	.224	.13144	.274	.17465	.324	.22040	.374	.26805	.424	.31699
.026	.00555	.075	.02676	.125	.05666	.175	.09231	.225	.13227	.275	.17554	.325	.22134	.375	.26901	.425	.31798
.027	.00587	.076	.02729	.126	.05733	.176	.09307	.226	.13311	.276	.17644	.326	.22228	.376	.26998	.426	.31897
.028	.00619	.077	.02782	.127	.05799	.177	.09384	.227	.13395	.277	.17733	.327	.22322	.377	.27095	.427	.31996
.029	.00653	.078	.02836	.128	.05866	.178	.09460	.228	.13478	.278	.17823	.328	.22415	.378	.27192	.428	.32095
.030	.00687	.079	.02889	.129	.05933	.179	.09537	.229	.13562	.279	.17912	.329	.22509	.379	.27289	.429	.32194
.031	.00721	.080	.02943	.130	.06000	.180	.09613	.230	.13646	.280	.18002	.330	.22603	.380	.27386	.430	.32293
.032	.00756	.081	.02998	.131	.06067	.181	.09690	.231	.13731	.281	.18092	.331	.22697	.381	.27483	.431	.32392
.033	.00791	.082	.03053	.132	.06135	.182	.09767	.232	.13815	.282	.18182	.332	.22792	.382	.27580	.432	.32491
.034	.00827	.083	.03108	.133	.06203	.183	.09845	.233	.13900	.283	.18272	.333	.22886	.383	.27678	.433	.32590
.035	.00864	.084	.03163	.134	.06271	.184	.09922	.234	.13984	.284	.18362	.334	.22980	.384	.27775	.434	.32689
.036	.00901	.085	.03219	.135	.06339	.185	.10000	.235	.14069	.285	.18452	.335	.23074	.385	.27872	.435	.32788
.037	.00938	.086	.03275	.136	.06407	.186	.10077	.236	.14154	.286	.18542	.336	.23169	.386	.27969	.436	.32887
.038	.00976	.087	.03331	.137	.06476	.187	.10155	.237	.14239	.287	.18633	.337	.23263	.387	.28067	.437	.32987
.039	.01015	.088	.03387	.138	.06545	.188	.10233	.238	.14324	.288	.18723	.338	.23358	.388	.28164	.438	.33086
.040	.01054	.089	.03444	.139	.06614	.189	.10312	.239	.14409	.289	.18814	.339	.23453	.389	.28262	.439	.33185
.041	.01093	.090	.03501	.140	.06683	.190	.10390	.240	.14494	.290	.18905	.340	.23547	.390	.28359	.440	.33284
.042	.01133	.091	.03559	.141	.06753	.191	.10469	.241	.14580	.291	.18996	.341	.23642	.391	.28457	.441	.33384
.043	.01173	.092	.03616	.142	.06822	.192	.10547	.242	.14666	.292	.19086	.342	.23737	.392	.28554	.442	.33483
.044	.01214	.093	.03674	.143	.06892	.193	.10626	.243	.14751	.293	.19177	.343	.23832	.393	.28652	.443	.33582
.045	.01255	.094	.03732	.144	.06963	.194	.10705	.244	.14837	.294	.19268	.344	.23927	.394	.28750	.444	.33682
.046	.01297	.095	.03791	.145	.07033	.195	.10784	.245	.14923	.295	.19360	.345	.24022	.395	.28848	.445	.33781
.047	.01339	.096	.03850	.146	.07103	.196	.10864	.246	.15009	.296	.19451	.346	.24117	.396	.28945	.446	.33880
.048	.01382	.097	.03909	.147	.07174	.197	.10943	.247	.15095	.297	.19542	.347	.24212	.397	.29043	.447	.33980
.049	.01425	.098	.03968	.148	.07245	.198	.11023	.248	.15182	.298	.19634	.348	.24307	.398	.29141	.448	.34079
		.099	.04028	.149	.07316	.199	.11102	.249	.15268	.299	.19725	.349	.24403	.399	.29239	.449	.34179
																.500	.39270

D1.0 Calculation of Area

$$\text{AREA} = D^2 \times C$$

where

D = Diameter, in

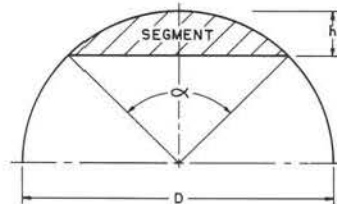
h = Segment height, in

C = value taken from Table for h/D; or

= $\pi/4 (\alpha/360 - \sin \alpha/2\pi)$, α in degrees, or

= $(\alpha - \sin \alpha)/8$, α in radians

$$\alpha = 2 \cos^{-1} \left(1 - \frac{2h}{D} \right)$$



APPENDIX D

PROCEDURE FOR CALCULATING ALLOWABLE NOZZLE EXTERNAL FORCES AND MOMENTS IN CYLINDRICAL VESSELS

D1.0 Nomenclature

- P = Design Pressure, psig
 r_o = Nozzle Outside Radius, inches (see Figure D-1)
 R_m = Mean Radius of Shell, inches (see Figure D-1)
 T = Shell Thickness, inches (see Figure D-1)
 S_y = Maximum Code allowable stress due to design pressure and nozzle loads at design temperature, psi
 Section VIII, Division 1: Value of yield strength associated with the applicable load combination or stress combination.
 Section VIII, Division 2: Allowable stress intensity associated with the applicable load combination or stress combination.
 Section III, Division 1: Allowable stress intensity associated with the applicable service limit or stress combination.
 σ = Calculated Stress Due to Design Pressure, psi
 S_a = Maximum Code allowable stress for shell due to design pressure at design temperature, psi
 Section VIII, Division 1: Maximum Allowable Stress Value, S
 Section VIII, Division 2: Design Stress Intensity Value, S_m
 Section III, Division 1: Design Stress Intensity Value, S_m or Allowable Stress Value, S , as applicable.
 β = Dimensionless Numbers
 γ = Dimensionless Numbers
 α = Dimensionless Numbers
 Σ = Dimensionless Numbers
 Δ = Dimensionless Numbers
 F_{RRF} = Maximum Resultant Radial Force, lbf (see Figure D-1)*
 M_{RCM} = Maximum Resultant Circumferential Moment, in-lbf (see Figure D-1)*
 M_{RLM} = Maximum Resultant Longitudinal Moment, in-lbf (see Figure D-1)*
 F_{RF} = Maximum Resultant Force, lbf*
 M_{RM} = Maximum Resultant Moment, in-lbf*

(1) This procedure is not applicable to nozzles that utilize reinforcing pads.

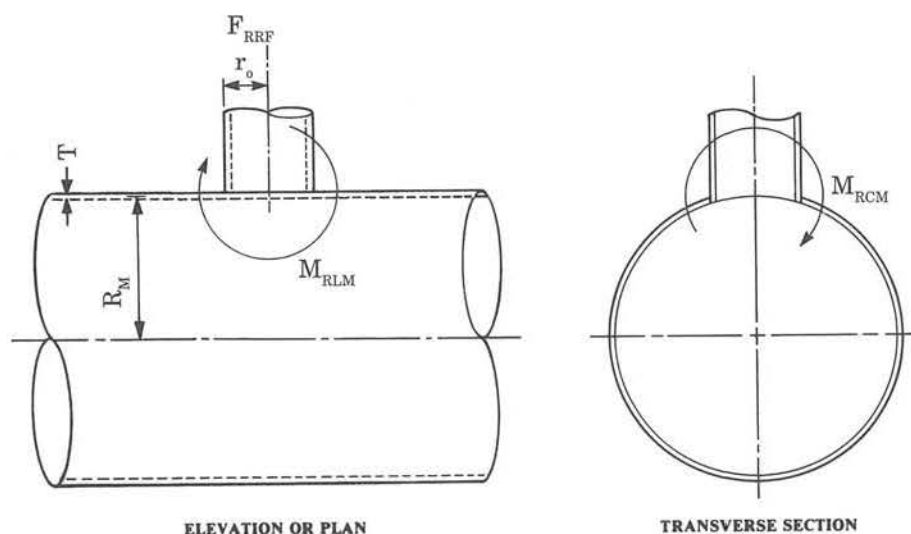


Figure D-1
NOZZLE LOAD NOMENCLATURE

APPENDIX D

D2.0 External Forces and Moments

To calculate the maximum force and moment, first evaluate β and γ . Then determine α , Σ and Δ from Figures D-2, D-3 and D-4 for the specified β and γ , substitute into the equations below and calculate F_{RRF} , M_{RCM} and M_{RLM} .

$$\beta = .875 \left(\frac{r_o}{R_m} \right); \quad \gamma = \frac{R_m}{T}$$

Determine α , Σ , and Δ from Figures D-2, D-3 and D-4.

Calculate Pressure Stress, σ .

$$\sigma = \frac{2P}{T} \left(R_m - \frac{T}{2} \right)$$

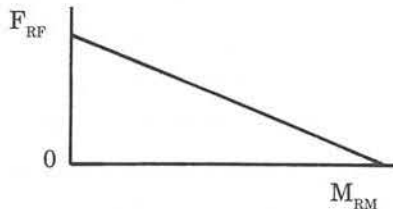
If σ is greater than S_a , then use S_a as the stress due to design pressure:

$$F_{RRF} = \frac{R_m^2}{\alpha} (S_y - \sigma)$$

$$M_{RCM} = \frac{R_m^2 r_o S_y}{\Sigma}$$

$$M_{RLM} = \frac{R_m^2 r_o}{\Delta} (S_y - \sigma)$$

Plot the value of F_{RRF} as F_{RF} and the smaller of M_{RCM} and M_{RLM} as M_{RM} . The allowable nozzle loads are bounded by the area F_{RF} , 0, M_{RM} .



D3.0 Sample Problem

Determine Resultant Force and Moment

$$R_m = 37.5 \text{ in} \quad S_y = 31,600 \text{ psi @ } 460^\circ\text{F}$$

$$r_o = 15 \text{ in} \quad S_a = 20,000 \text{ psi}$$

$$T = .75 \text{ in}$$

$$P = 150 \text{ psig}$$

$$\beta = .875 \left(\frac{r_o}{R_m} \right) = .875 \left(\frac{15}{37.5} \right) = .35$$

$$\gamma = \left(\frac{R_m}{T} \right) = \frac{37.5}{.75} = 50$$

From Figure D-2, $\alpha = 440$

From Figure D-3, $\Sigma = 1,070$

From Figure D-4, $\Delta = 340$

Calculate Pressure Stress

$$\sigma = \frac{2P}{T} \left(R_m - \frac{T}{2} \right) =$$

$$\frac{2(150)}{.75} \left(37.5 - \frac{.75}{2} \right) =$$

$$14,850 \text{ psi} < S_a = 20,000 \text{ psi}$$

Use $\sigma = 14,850$ psi in the equations for calculating F_{RRF} and M_{RLM} .

Calculate Allowable Forces and Moments

$$F_{RRF} = \frac{R_m^2}{\alpha} (S_y - \sigma) =$$

$$\frac{(37.5)^2}{440} (31,600 - 14,850) = 53,533 \text{ lbf}$$

$$M_{RCM} = \frac{R_m^2 r_o S_y}{\Sigma} =$$

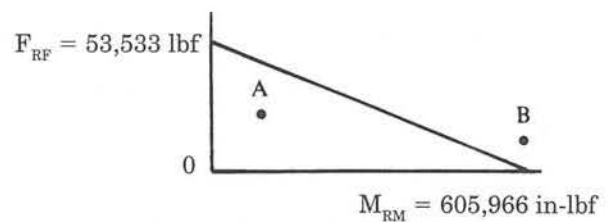
$$\frac{(37.5)^2 (15) (31,600)}{1,100} = 605,966 \text{ in-lbf}$$

$$M_{RLM} = \frac{R_m^2 r_o}{\Delta} (S_y - \sigma) =$$

$$\frac{(37.5)^2 (15)}{340} (31,600 - 14,850) =$$

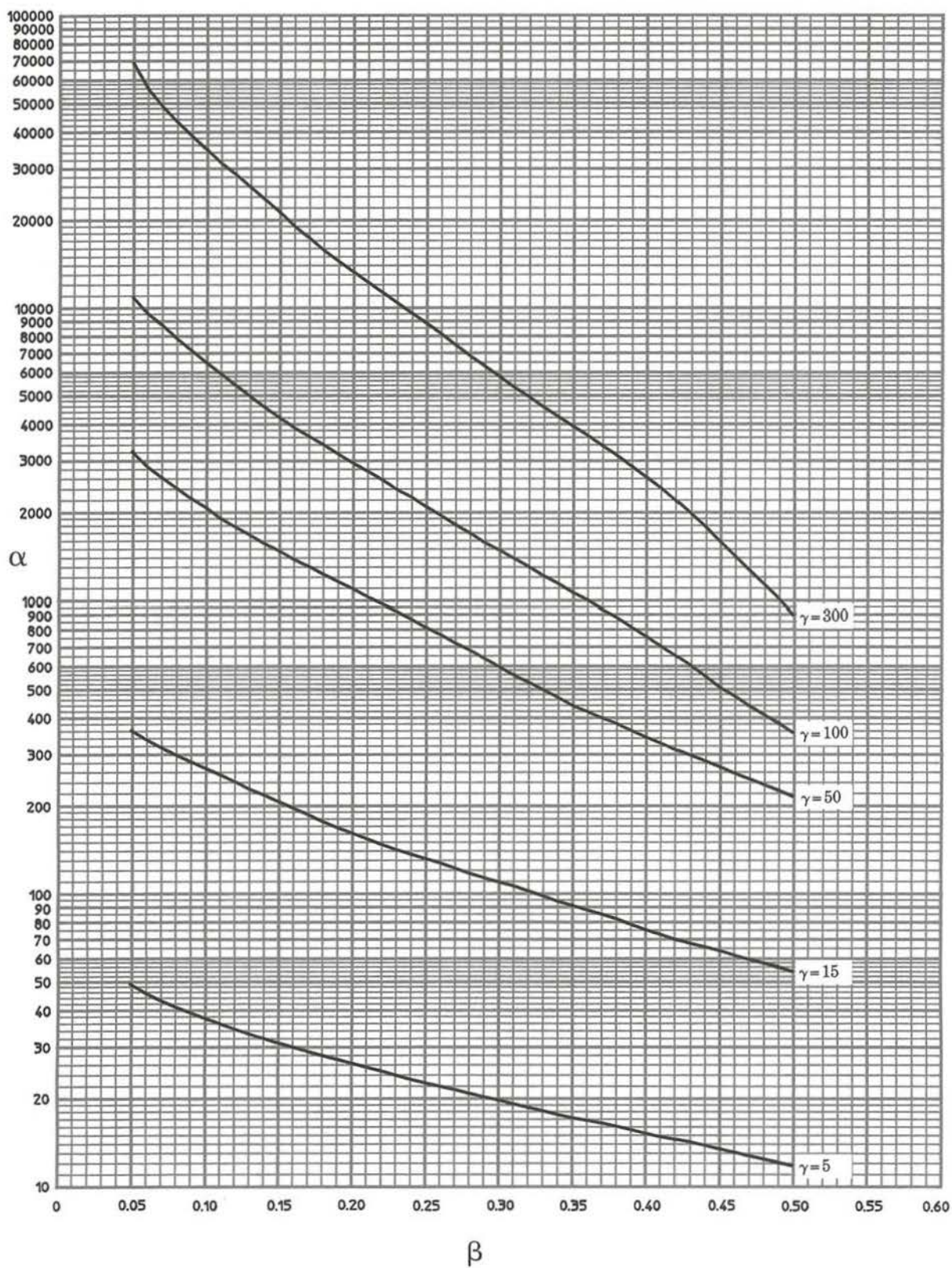
$$1,039,177 \text{ in-lbf}$$

Plot the value of F_{RRF} as F_{RF} and the smaller of M_{RCM} and M_{RLM} as M_{RM} . The allowable nozzle loads are bounded by the area of F_{RF} , 0, and M_{RM} .



Therefore, a nozzle reaction of $F = 20,000$ lbf and $M = 100,000$ in-lbf would be allowable (point A) but a nozzle reaction of $F = 5,000$ lbf and $M = 605,000$ in-lbf would not be allowable (point B).

*Note: Use absolute values in the graph.



β
Figure D-2
ALLOWABLE NOZZLE LOADS

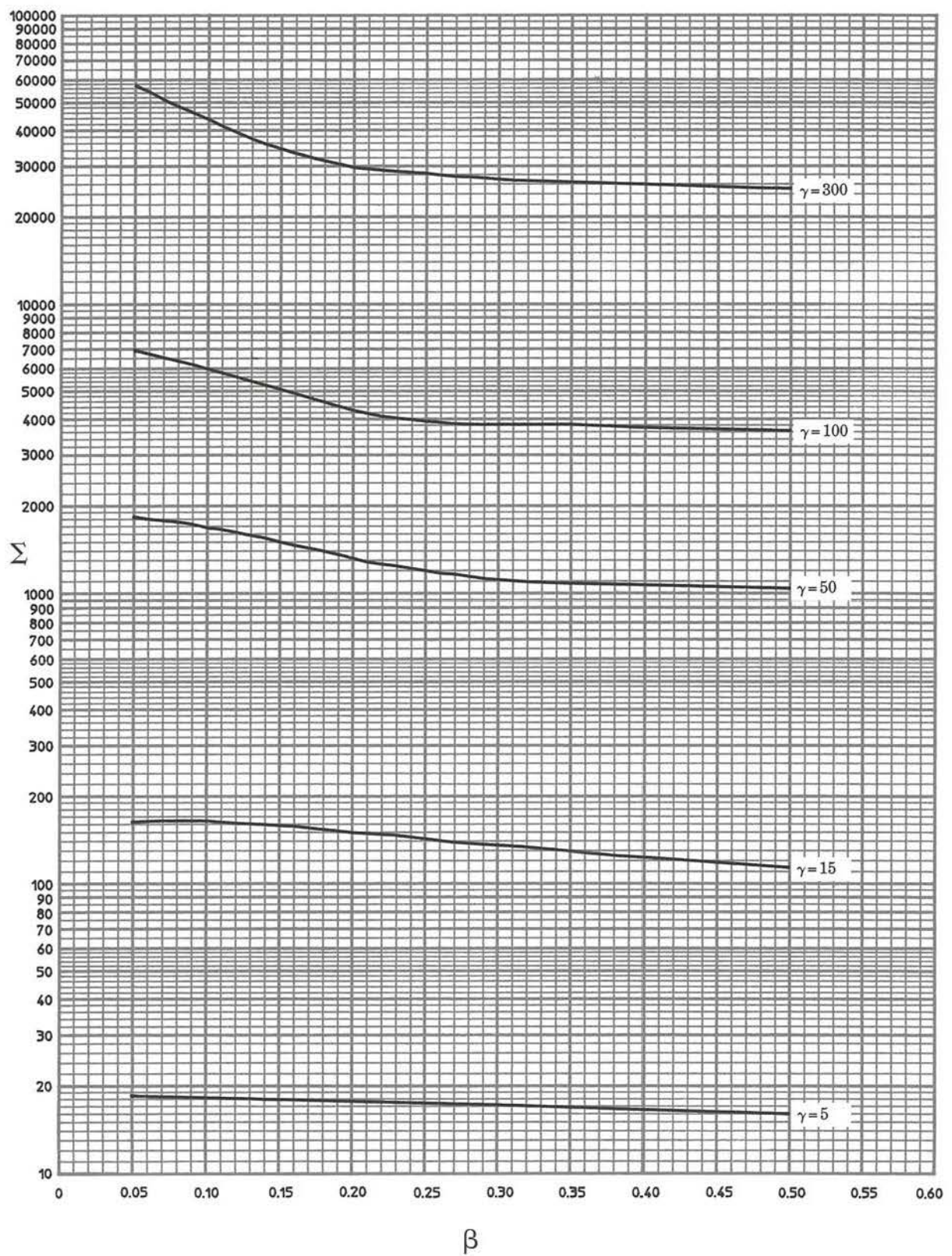
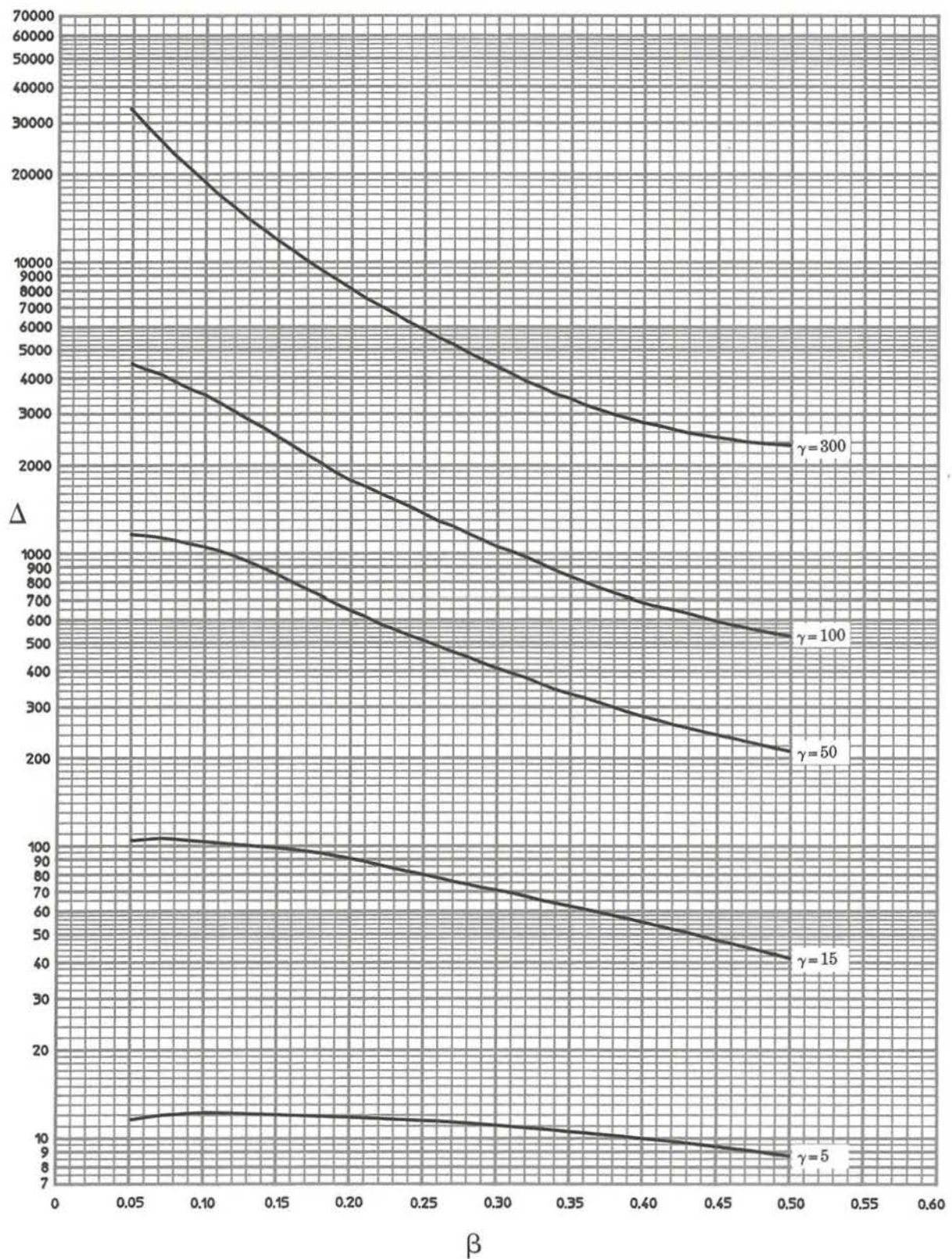


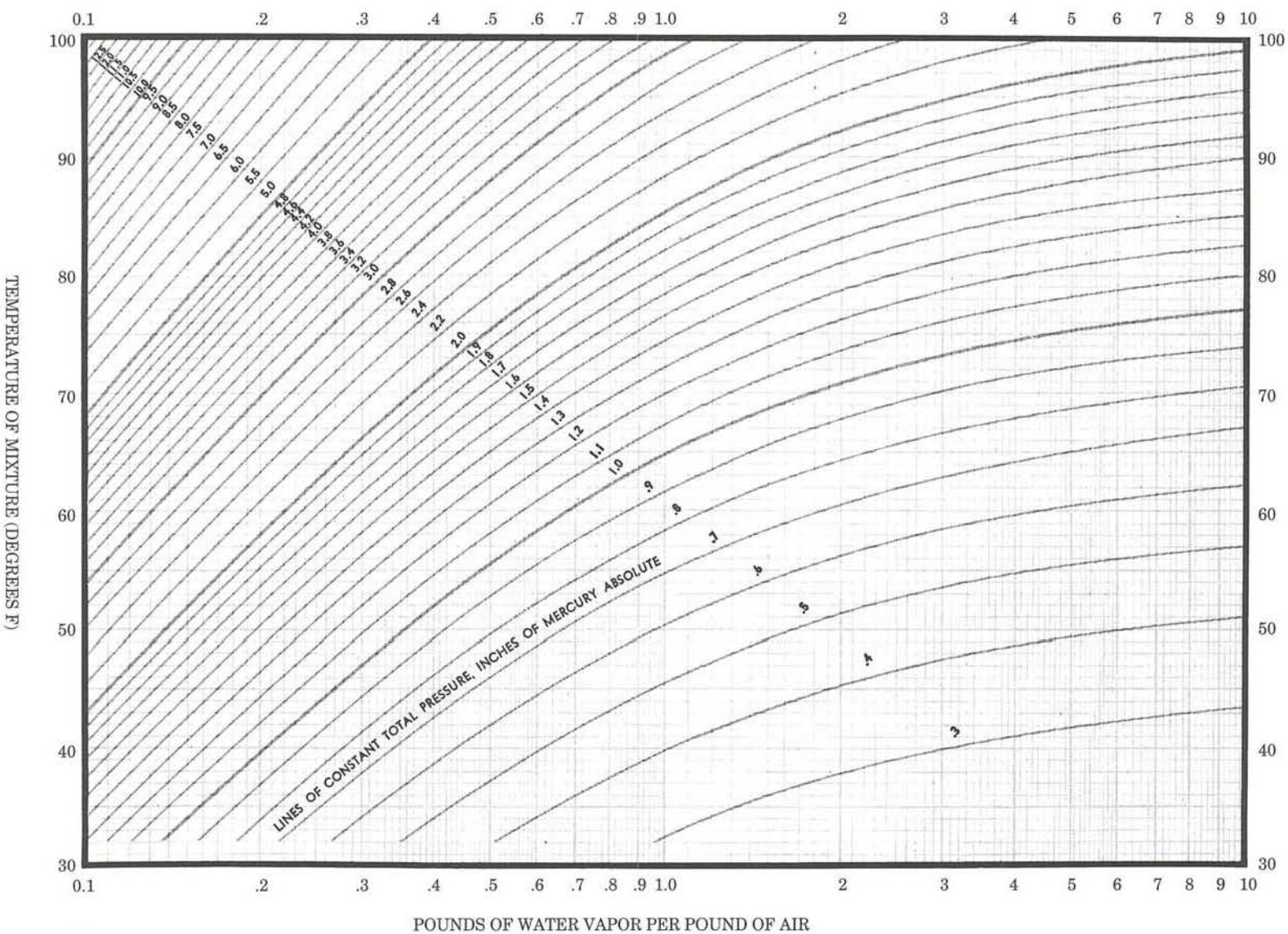
Figure D-3
ALLOWABLE NOZZLE LOADS



β
Figure D-4
ALLOWABLE NOZZLE LOADS

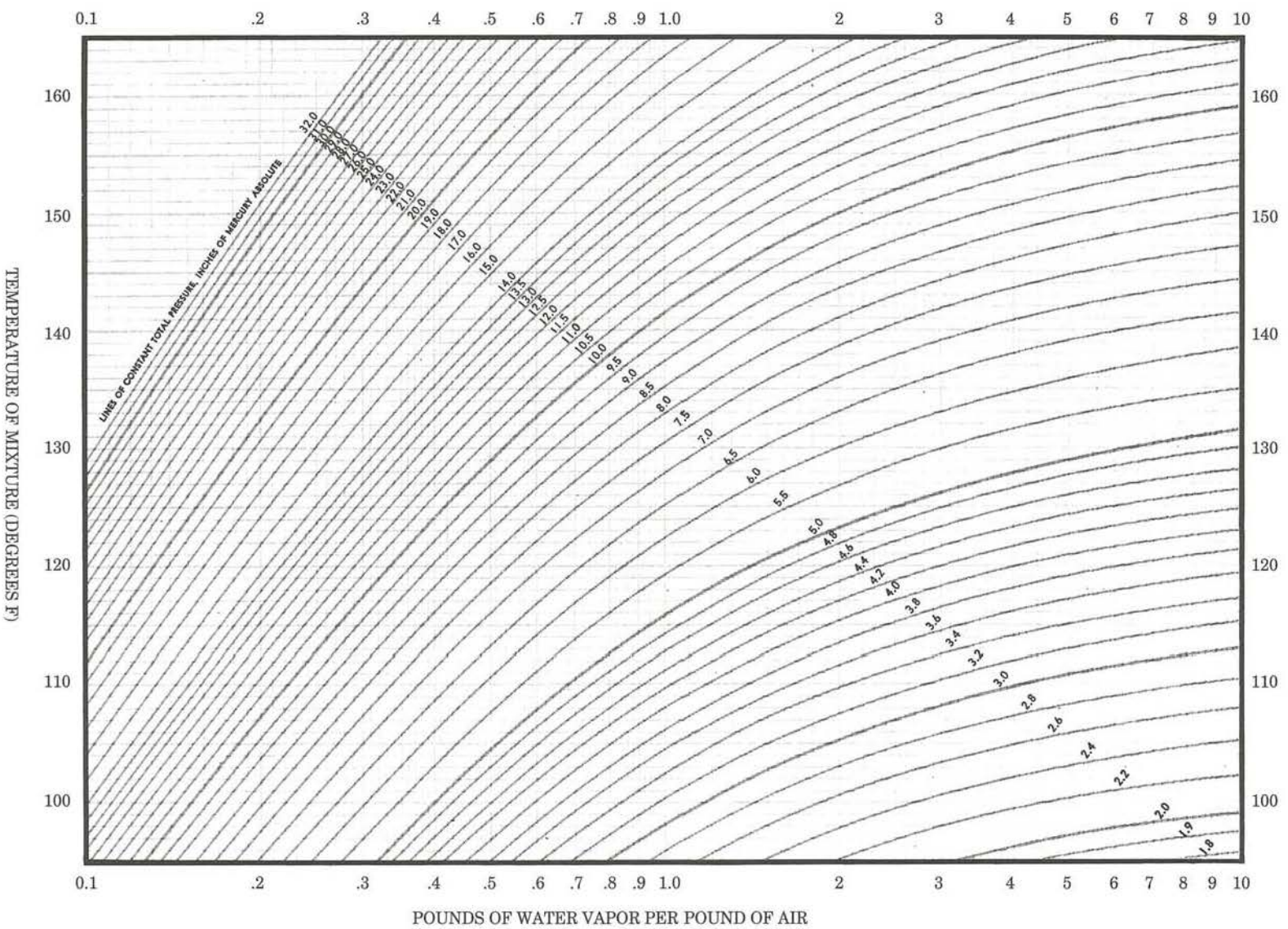
APPENDIX E

AIR AND WATER VAPOR MIXTURE DATA (DALTON'S LAW)



APPENDIX E

AIR AND WATER VAPOR MIXTURE DATA (DALTON'S LAW)



APPENDIX F

MECHANICAL CHARACTERISTICS OF TUBING

Tube OD (in)	Nominal Surface per Ft. of Tube (ft ²)	External BWG Gauge	Thick-ness (in)	Nominal Tube ID (in)	Nominal Internal Area (in ²)	Ratio OD/ID	Constant C*	Wt/Ft (Steel) (lbm/ft)	Trans-verse Metal Area (in ²)	Moment of Inertia (in ⁴)
3/8	.0982	10	.134	.107	.0090	3.505	14.	.345	.1015	.000964
		11	.120	.135	.0143	2.778	22.	.327	.0961	.000954
		12	.109	.157	.0194	2.389	30.	.310	.0911	.000941
		13	.095	.185	.0269	2.027	42.	.284	.0836	.000913
		14	.083	.209	.0343	1.794	54.	.259	.0761	.000877
		15	.072	.231	.0419	1.623	65.	.233	.0685	.000831
		16	.065	.245	.0471	1.531	74.	.216	.0633	.000794
		17	.058	.259	.0527	1.448	82.	.197	.0578	.000750
		18	.049	.277	.0603	1.354	94.	.171	.0502	.000682
		19	.042	.291	.0665	1.289	104.	.150	.0439	.000619
		20	.035	.305	.0731	1.230	114.	.127	.0374	.000546
		21	.032	.311	.0760	1.206	119.	.117	.0345	.000512
		22	.028	.319	.0799	1.176	125.	.104	.0305	.000462
		23	.025	.325	.0830	1.154	129.	.093	.0275	.000431
		24	.022	.331	.0860	1.133	134.	.083	.0244	.000389
		25	.020	.335	.0881	1.119	137.	.076	.0223	.000359
1/2	.1309	10	.134	.232	.0423	2.155	66.	.524	.1541	.002926
		11	.120	.260	.0531	1.923	83.	.487	.1433	.002844
		12	.109	.282	.0625	1.773	97.	.456	.1339	.002758
		13	.095	.310	.0755	1.613	118.	.411	.1209	.002615
		14	.083	.334	.0876	1.497	137.	.370	.1087	.002457
		15	.072	.356	.0995	1.404	155.	.329	.0968	.002280
		16	.065	.370	.1075	1.351	168.	.302	.0888	.002148
		17	.058	.384	.1158	1.302	181.	.274	.0805	.002001
		18	.049	.402	.1269	1.244	198.	.236	.0694	.001786
		19	.042	.416	.1359	1.202	212.	.206	.0604	.001598
		20	.035	.430	.1452	1.163	227.	.174	.0511	.001390
		21	.032	.436	.1493	1.147	233.	.160	.0470	.001294
		22	.028	.444	.1548	1.126	242.	.141	.0415	.001160
		23	.025	.450	.1590	1.111	248.	.127	.0373	.001075
		24	.022	.456	.1633	1.096	255.	.112	.0330	.000963
		25	.020	.460	.1662	1.087	259.	.103	.0302	.000886

*Liquid velocity in feet/second = $\frac{\text{pounds per tube per hour}}{C \times \text{specific gravity of liquid}}$ Specific gravity of water at 60 deg. F = 1.0

For weights of other materials, multiply carbon steel weights by the following factors:

90-10 CuNi UNS C70600-1.140	Titanium per ASTM B338-.573
70-30 CuNi UNS C71500-1.140	Stainless Steel UNS S30400-1.013
Arsenical Cu UNS C14200-1.140	Stainless Steel UNS S31600-1.013
Admiralty UNS C444300-1.088	Stainless Steel UNS S43035-0.989
Al Brass UNS C68700-1.060	Stainless Steel UNS N08367-1.025
Al Bronze UNS C60800-1.042	Stainless Steel UNS S44735-0.989
Copper Iron UNS C19400-1.119	Stainless Steel UNS S44660-0.989

APPENDIX F

MECHANICAL CHARACTERISTICS OF TUBING

Nominal Tube OD (in)	External Surface per Ft. of Tube (ft ²)	BWG Gauge	Nominal Thick- ness (in)	Nominal Tube ID (in)	Internal Area (in ²)	Ratio OD/ID	Constant C*	Wt/Ft (Steel) (lbm/ft)	Trans- verse Metal Area (in ²)	Moment of Inertia (in ⁴)
5/8	.1636	10	.134	.357	.1001	1.751	156.	.703	.2067	.006693
		11	.120	.385	.1164	1.623	182.	.648	.1904	.006412
		12	.109	.407	.1301	1.536	203.	.601	.1767	.006143
		13	.095	.435	.1486	1.437	232.	.538	.1582	.005733
		14	.083	.459	.1655	1.362	258.	.481	.1413	.005311
		15	.072	.481	.1817	1.299	283.	.426	.1251	.004863
		16	.065	.495	.1924	1.263	300.	.389	.1144	.004543
		17	.058	.509	.2035	1.228	317.	.352	.1033	.004195
		18	.049	.527	.2181	1.186	340.	.302	.0887	.003704
		19	.042	.541	.2299	1.155	359.	.262	.0769	.003285
		20	.035	.555	.2419	1.125	377.	.221	.0649	.002833
		21	.032	.561	.2472	1.114	386.	.203	.0596	.002528
		22	.028	.569	.2543	1.098	397.	.179	.0525	.002345
		23	.025	.575	.2597	1.087	405.	.160	.0471	.002164
		24	.022	.581	.2651	1.076	414.	.142	.0417	.001932
		25	.020	.585	.2688	1.068	419.	.129	.0380	.001773
3/4	.1963	10	.134	.482	.1825	1.556	285.	.882	.2593	.012882
		11	.120	.510	.2043	1.471	319.	.808	.2375	.012211
		12	.109	.532	.2223	1.410	347.	.747	.2195	.011600
		13	.095	.560	.2463	1.339	384.	.655	.1955	.010704
		14	.083	.584	.2679	1.284	418.	.592	.1739	.009822
		15	.072	.606	.2884	1.238	450.	.522	.1534	.008912
		16	.065	.620	.3019	1.210	471.	.476	.1399	.008278
		17	.058	.634	.3157	1.183	492.	.429	.1261	.007601
		18	.049	.652	.3339	1.150	521.	.367	.1079	.006661
		19	.042	.666	.3484	1.126	543.	.318	.0934	.005874
		20	.035	.680	.3632	1.103	567.	.268	.0786	.005036
		21	.032	.686	.3696	1.093	577.	.246	.0722	.004661
		22	.028	.694	.3783	1.081	590.	.216	.0635	.004145
		23	.025	.700	.3848	1.071	600.	.194	.0569	.003815
		24	.022	.706	.3915	1.062	611.	.171	.0503	.003398
		25	.020	.710	.3959	1.056	618.	.156	.0459	.003114
7/8	.2291	10	.134	.607	.2894	1.442	451.	1.061	.3119	.022110
		11	.120	.635	.3167	1.378	494.	.969	.2846	.020793
		12	.109	.657	.3390	1.332	529.	.893	.2623	.019628
		13	.095	.685	.3685	1.277	575.	.792	.2328	.017966
		14	.083	.709	.3948	1.234	616.	.703	.2065	.016370
		15	.072	.731	.4197	1.197	655.	.618	.1816	.014758
		16	.065	.745	.4359	1.174	680.	.563	.1654	.013653
		17	.058	.759	.4525	1.153	706.	.507	.1489	.012484
		18	.049	.777	.4742	1.126	740.	.433	.1272	.010882
		19	.042	.791	.4914	1.106	767.	.374	.1099	.009558
		20	.035	.805	.5090	1.087	794.	.314	.0924	.008161
		21	.032	.811	.5166	1.079	806.	.288	.0847	.007539
		22	.028	.819	.5268	1.068	822.	.254	.0745	.006689
		23	.025	.825	.5346	1.061	834.	.227	.0668	.006147
		24	.022	.831	.5424	1.053	846.	.201	.0590	.005365
		25	.020	.835	.5476	1.048	854.	.183	.0537	.005003

*Liquid velocity in feet/second = $\frac{\text{pounds per tube per hour}}{C \times \text{specific gravity of liquid}}$ Specific gravity of water at 60 deg. F = 1.0

APPENDIX F **MECHANICAL CHARACTERISTICS OF TUBING**

Nominal Tube OD (in)	External Surface per Ft. of Tube (ft ²)	BWG Gauge	Nominal Thick- ness (in)	Nominal Tube ID (in)	Internal Area (in ²)	Ratio OD/ID	Constant C*	Wt/Ft (Steel) (lbm/ft)	Trans- verse Metal Area (in ²)	Moment of Inertia (in ⁴)
1	.2618	10	.134	.732	.4208	1.366	657.	1.241	.3646	.034994
		11	.120	.760	.4536	1.316	708.	1.129	.3318	.032711
		12	.109	.782	.4803	1.279	749.	1.038	.3051	.030731
		13	.095	.810	.5153	1.235	804.	.919	.2701	.027957
		14	.083	.834	.5463	1.199	852.	.814	.2391	.025339
		15	.072	.856	.5755	1.168	898.	.714	.2099	.022732
		16	.065	.870	.5945	1.149	927.	.650	.1909	.020965
		17	.058	.884	.6138	1.131	957.	.584	.1716	.019111
		18	.049	.902	.6390	1.109	997.	.498	.1464	.016594
		19	.042	.916	.6590	1.092	1028.	.430	.1264	.014529
		20	.035	.930	.6793	1.075	1060.	.361	.1061	.012367
		21	.032	.936	.6881	1.068	1073.	.331	.0973	.011411
		22	.028	.944	.6999	1.059	1092.	.291	.0855	.010106
		23	.025	.950	.7088	1.053	1106.	.260	.0766	.009275
		24	.022	.956	.7178	1.046	1120.	.230	.0676	.008236
		25	.020	.960	.7238	1.042	1129.	.209	.0616	.007533
1 1/8	.2945	10	.134	.857	.5768	1.313	900.	1.420	.4172	.052150
		11	.120	.885	.6151	1.271	960.	1.289	.3789	.048516
		12	.109	.907	.6461	1.240	1008.	1.184	.3479	.045409
		13	.095	.935	.6866	1.203	1071.	1.046	.3074	.041113
		14	.083	.959	.7223	1.173	1127.	.925	.2717	.037110
		15	.072	.981	.7558	1.147	1179.	.811	.2382	.033167
		16	.065	.995	.7776	1.131	1213.	.737	.2165	.030516
		17	.058	1.009	.7996	1.115	1247.	.662	.1944	.027750
		18	.049	1.027	.8284	1.095	1292.	.564	.1656	.024021
		19	.042	1.041	.8511	1.081	1328.	.486	.1429	.020982
		20	.035	1.005	.8742	1.066	1364.	.408	.1199	.017818
		21	.032	1.061	.8841	1.060	1379.	.374	.1099	.016423
		22	.028	1.069	.8975	1.052	1400.	.328	.0965	.014525
		23	.025	1.075	.9076	1.047	1416.	.294	.0864	.013317
		24	.022	1.081	.9178	1.041	1431.	.259	.0762	.011814
		25	.020	1.085	.9246	1.037	1442.	.236	.0694	.010797
1 1/4	.3272	10	.134	.982	.7574	1.273	1182.	1.599	.4698	.074195
		11	.120	1.010	.8012	1.238	1250.	1.450	.4260	.068762
		12	.109	1.032	.8365	1.211	1305.	1.330	.3907	.064164
		13	.095	1.060	.8825	1.179	1377.	1.173	.3447	.057871
		14	.083	1.084	.9229	1.153	1440.	1.035	.3043	.052064
		15	.072	1.106	.9607	1.130	1499.	.907	.2665	.046392
		16	.065	1.120	.9852	1.116	1537.	.823	.2420	.042602
		17	.058	1.134	1.0100	1.102	1576.	.739	.2172	.038667
		18	.049	1.152	1.0423	1.085	1626.	.629	.1849	.033389
		19	.042	1.166	1.0678	1.072	1666.	.542	.1594	.029110
		20	.035	1.180	1.0936	1.059	1706.	.455	.1336	.024673
		21	.032	1.186	1.1047	1.054	1723.	.417	.1224	.022722
		22	.028	1.194	1.1197	1.047	1747.	.366	.1075	.020075
		23	.025	1.200	1.1310	1.042	1764.	.327	.0962	.018390
		24	.022	1.206	1.1423	1.036	1782.	.289	.0849	.016301
		25	.020	1.210	1.1499	1.033	1794.	.263	.0773	.014891

*Liquid velocity in feet/second = $\frac{\text{pounds per tube per hour}}{C \times \text{specific gravity of liquid}}$ Specific gravity of water at 60 deg. F = 1.0

APPENDIX F **MECHANICAL CHARACTERISTICS OF TUBING**

Nominal Tube OD (in)	External Surface per Ft. of Tube (ft ²)	BWG Gauge	Nominal Thick- ness (in)	Nominal Tube ID (in)	Internal Area (in ²)	Ratio OD/ID	Constant C*	Wt/Ft (Steel) (lbm/ft)	Trans- verse Metal Area (in ²)	Moment of Inertia (in ⁴)
1½	.3927	10	.134	1.232	1.1921	1.218	1860.	1.957	.5750	.135418
		11	.120	1.260	1.2469	1.190	1945.	1.770	.5202	.124781
		12	.109	1.282	1.2908	1.170	2014.	1.621	.4763	.115911
		13	.095	1.310	1.3478	1.145	2103.	1.427	.4193	.103943
		14	.083	1.334	1.3977	1.124	2180.	1.257	.3695	.093054
		15	.072	1.356	1.4441	1.106	2253.	1.099	.3230	.082543
		16	.065	1.370	1.4741	1.095	2300.	.997	.2930	.075582
		17	.058	1.384	1.5044	1.084	2347.	.894	.2628	.068405
		18	.049	1.402	1.5438	1.070	2408.	.760	.2234	.058851
		19	.042	1.416	1.5748	1.059	2457.	.655	.1924	.051161
		20	.035	1.430	1.6061	1.049	2505.	.548	.1476	.043240
		21	.032	1.436	1.6196	1.045	2527.	.502	.1611	.039774
		22	.028	1.444	1.6377	1.039	2555.	.441	.1295	.035083
		23	.025	1.450	1.6513	1.034	2576.	.394	.1158	.032100
		24	.022	1.456	1.6650	1.030	2597.	.347	.1022	.028419
		25	.020	1.460	1.6742	1.027	2611.	.316	.0930	.025939
2	.5236	10	.134	1.732	2.3561	1.155	3675.	2.673	.7855	.343663
		11	.120	1.760	2.4328	1.136	3795.	2.412	.7087	.314398
		12	.109	1.782	2.4940	1.122	3891.	2.203	.6475	.290403
		13	.095	1.810	2.5730	1.105	4014.	1.935	.5685	.258551
		14	.083	1.834	2.6417	1.091	4121.	1.701	.4999	.230047
		15	.072	1.856	2.7055	1.078	4221.	1.484	.4361	.202917
		16	.065	1.870	2.7465	1.070	4284.	1.345	.3951	.185142
		17	.058	1.884	2.7877	1.062	4349.	1.204	.3539	.166964
		18	.049	1.902	2.8413	1.052	4432.	1.022	.3003	.142989
		19	.042	1.916	2.8832	1.044	4498.	.879	.2584	.123865
		20	.035	1.930	2.9255	1.036	4564.	.735	.2161	.104317
		21	.032	1.936	2.9437	1.033	4592.	.673	.1978	.095808
		22	.028	1.944	2.9681	1.029	4630.	.590	.1735	.084339
		23	.025	1.950	2.9865	1.026	4658.	.528	.1551	.077050
		24	.022	1.956	3.0049	1.022	4687.	.465	.1367	.068111
		25	.020	1.960	3.0172	1.020	4706.	.423	.1244	.062105

*Liquid velocity in feet/second = $\frac{\text{pounds per tube per hour}}{C \times \text{specific gravity of liquid}}$ Specific gravity of water at 60 deg. F = 1.0

APPENDIX G

TROUBLESHOOTING GUIDE

This troubleshooting guide has been prepared to assist operators of steam surface condensers. The guide provides general guidance, and operators are advised to consult with the manufacturer when necessary for specific instructions regarding their equipment. Many of the items listed below are not in the scope of the condenser manufacturer; however, these items do affect operation and must be considered by operators.

HEI has produced a computer rating program to assist operators in evaluating the performance of their condenser. Contact HEI at hei@heatexchange.org, or visit the HEI website at www.heatexchange.org for further details.

Symptoms	Possible Causes	Possible Solutions
Air In-Leakage	Gasket failure	Replace gasket
	Expansion Joint Failure	Replace or repair expansion joint
	Weld failure	Locate weld failure and repair
	Leaks from incoming vents and drains	Check all valves on vent and drain lines
	Instrumentation	Check all flanged connections and valves of instrumentation
	LP Turbine	Check all LP turbine seals
	Gasket seating surface damaged	Repair gasket seating surface
Condensate Chemistry (High Conductivity)	Tube leak	Locate leak and repair
	Corrosion products in Condenser	Clean condenser hotwell
	Incoming drains	Check drain sources
False Instrumentation Reading	Instruments out of calibration	Check calibration
	Damaged Instrument	Repair or replace as necessary
	Improper instrument air/electrical supply	Check manufacturer's recommendations and correct supply
	Improper installation	Check manufacturers manuals and comply
	Improper or damaged wiring	Repair or replace wiring
	Incorrect range	Check process requirements and ensure instrument can meet conditions
	No pigtail on steam line pressure gauges	Install pigtail siphon and replace gauge if heat damage has occurred
	Corroded switch contacts	Clean and/or replace contacts as needed
	Isolated instrument connection	Check that connection is open to instrument

APPENDIX G

TROUBLESHOOTING GUIDE

Symptoms	Possible Causes	Possible Solutions
Fluctuating or Unstable Hotwell Levels	Introduction of drains exceeding design parameters	Evaluate internal dispersion design. Install stilling chambers where applicable.
	Malfunctioning level controller/control valves	Check to ensure that level controller and control valves are functioning properly. Ensure that control valves are correctly sized.
	Condensate pump malfunction	Check condensate pump operating manual
	Transient operating conditions	Evaluate after steady state is reached
	False instrumentation readings	See False Instrumentation Reading section
High Absolute Back Pressure	Air in-leakage	See Air In-Leakage section
	Insufficient Cooling Water	Validate water flow and temperature. See Improper Venting of Waterbox section
	Waterbox not full of water	See Improper Venting of Waterbox section
	Excessive tube fouling or blockage	Clean tubes, tubesheets, and/or waterboxes
	Excessive number of plugged tubes	Replace plugged tubes
	False instrumentation readings	See False Instrumentation Reading section
	Vacuum equipment failure	See HEI Vacuum Equipment Troubleshooting Guide
	High condensate level	Check condensate levels and adjust to design
High Dissolved O₂ in Condensate	Air In-Leakage	See Air In-Leakage section
	High dissolved O ₂ in makeup water	Check makeup water source.
	High dissolved O ₂ in return condensate streams	Check return condensate stream sources.
	Vacuum equipment failure	See HEI Vacuum Equipment Troubleshooting Guide
	High condensate level resulting in sub-cooling of the condensate	Check condensate levels and adjust to design
	Insufficient sparging steam in the hot-well	Restore sparging steam to design conditions
	Low load operation	Consult HEI design standards

APPENDIX G

TROUBLESHOOTING GUIDE

Symptoms	Possible Causes	Possible Solutions
High Tube Side Pressure Loss	Severe fouling inside the tubes	Clean the inside of the tubes
	A large percentage of plugged tubes	Replace plugged tubes
	A restriction or debris located in the tubes or in the waterbox	Remove the obstructions in the tubes or in the waterbox
	Improperly vented waterbox	See Improper Venting of Waterbox section
	Increased circulating water flow rate	Restore to design conditions
Improper Venting of Waterbox	Blocked air vents	Inspect and repair vent opening as required.
	Corroded connections	Replace with corrosion resistant material.
	Insufficient air vent capacity.	Calculate piping system air release and check vent size.
	Insufficient cooling water pump capacity	Check cooling water piping and tubes for blockage. Check cooling water pumps for wear. Review overall system hydraulics.
	Missing automatic air release valve	Check high points in cooling water system for automatic air release valves.
	Missing vacuum check valve	Where vacuum exists in top of boxes and piping high points (once through siphon systems) include vacuum check valve on air release valves.
Lagging Failure (For Extraction Piping and Condenser Neck Mounted Heater)	Improper attachment of lagging	Inspect lagging attachment prior to plant startup, and during plant outages to make sure all lagging parts are securely installed.
	High velocity steam impingement/erosion	Evaluate flow distribution and local velocity profile in the area of failed lagging. Protect lagging from direct impingement from incoming drains.
	Physical impact from failed expansion joint	Repair failed expansion joint and replace lagging section.
Local Overheat (Hot Spots)	Operating conditions exceeding design parameters	Re-evaluate internal dispersion design

APPENDIX G

TROUBLESHOOTING GUIDE

Symptoms	Possible Causes	Possible Solutions
Low Hotwell Temperature (Condensate Sub-cooling)	Insufficient freefall (distance from bottom tube to liquid level)	Hotwell level is above design or normal level. Operate at design level. Check instrumentation.
	Make up to condenser exceeding design parameters	Check make up conditions and adjust to design
	Operation off design/low load operation	Expected at low load. Possibly alleviated with steam sparging in hotwell
	Low circulating water temperature	Expected when due to seasonal variation. Possibly alleviated with steam sparging in hotwell
Unusual Sounds	Broken or loose components inside the condenser	Repair broken or loose components
	Water hammer impact inside the waterbox	Check for proper operation of cooling water system
Tube Failures	Worn or damaged internal connection baffles or spray pipes	Inspect connections for damage. Review actual operating conditions with manufacturer. Repair connection, plug or replace tubes.
	Excessive tube vibration due to abnormal operating conditions.	Correct abnormal conditions. Contact manufacturer for re-evaluation.
	Excessive tube vibration due to location of service connections.	Re-evaluate connections for baffles, shields, or impingement protection.
	Chloride attack – stainless steel tubing	Review water chemistry and tube compatibility.
	Stagnant water during extended shutdown	Tubes may need to be rinsed and dried for extended shutdown.
	Microbiologically Induced Corrosion (MIC)	Check with plant water chemistry specialist
	Ammonia attack – copper alloy tubing	Check with plant water chemistry specialist
	Maintenance or construction damage	Plug or replace as required.
	Tube inlet erosion	Check water flow and adjust to design. Check water quality, especially suspended solids.

APPENDIX H

HEI SURFACE CONDENSER DATA SHEET

1	Manufacturer:					
2	Customer / Project:					
3	Plant Location:					
4	Specification No:			Item:		
5	Date:			Revision:		
Design Conditions (Thermal)						
6	Shell Side			Tube Side		
7	Total Duty:	MMBtu/hr	Source of Water:			
8	Steam Inlet:	lbs/hr	Cooling Water Flow:	USGPM		
9	Non Condensables:	lbs/hr	Temp. Water In:	°F		
10	U - Service:	Btu/hr-ft ² -°F	Temp. Water Out:	°F		
11	Saturation Temp.:	°F*	Number of Passes:			
12	Condensate Temp.:	°F*	Water Velocity:		ft/sec	
13	LMTD :	°F	Pressure Drop:		ft water	
14	Operating Pressure:	in-Hga*	Cleanliness Factor:			
15	Eff. Surface Area:	ft ²	Specific Heat:			
16			Specific Gravity:			
17	Remarks: * - Guarantee Point, Oxygen Guarantee per HEI					
Construction (Mechanical)						
18	Shell side			Tube Side		
19	Design Pressure:	psig	Design Pressure:	psig		
20	Design Temperature:	°F	Design Temperature:	°F		
21	Test Pressure:	psig	Test Pressure:	psig		
22	Water Box					
23	Type:	Number per shell:				
24	Plate Material (ASTM):					
25	Thickness:	in.	Corrosion allowance:	in.		
26	Manways (Qty):	Size:		in.		
27	Circwater Inlet (Qty):	Size:		in.		
28	Circwater Outlet (Qty):	Size:		in.		
29	Drains (Qty):	Size:		in.		
30	Vents (Qty):	Size:		in.		
31	Waterbox Internal Coating:		Cathodic Protection Type:			
32	Hotwell					
33	Thickness:	in.	Plate Material (ASTM):			
34	Normal Liquid Level:	in.	Capacity:	ft ³		
35	Maximum Liquid Level:	in.	Capacity:	ft ³		
36	Condensate Outlet Size:	in.	Condensate Outlet Number:			
37	Shell					
38	Thickness:	in.	Plate Material (ASTM):			
39	Air offtake (Qty):	Size:		in.		
40	Transition / Extended Neck					
41	Thickness:	in.	Plate Material (ASTM):			
42	Exhaust Opening Dimensions:					in.
43	Expansion Joint Type:					
44	Tubes	Quantity	Material (ASTM)	Dia. (in.)	Gauge	T.T.L. (ft)
45	Condensing Zone					
46	Air Cooling Zone					
47	Impingement Zone					
48	Tubesheet	Quantity	Material (ASTM)	Thk. (in)		
49						
50	Support Plates	Quantity	Material (ASTM)	Thk. (in)	Spacing Max. / Min. (in.)	
51						
52	Weights (lbs.)					
53	Empty:	Operating:		Shell Side Hydrotest:		
54	Notes:					
55						
56						
57						
58						

APPENDIX I

CONDENSER MATERIAL STRESS VALVES

		ASTM	UNS	S _u	S _y	S _y /1.5	S _A @ deg F	
SHAPES	CARBON STEEL	A 36	K02600	58	36	24.0	16.6	650
PIPE	CARBON STEEL	A 53 Gr B	K03005	60	35	23.3	14.6	650
FORGING	CARBON STEEL	A 105	K03504	70	36	24.0	20.0	400
PIPE	CARBON STEEL	A 106 Gr B	K03006	60	35	23.3	17.1	650
PLATE	STAINLESS STEEL	A 240 TP 304	S30400	75	30	20.0	18.9	250
PLATE	STAINLESS STEEL	A 240 TP 304L	S30403	70	25	16.7	16.7	250
PLATE	STAINLESS STEEL	A 240 TP 316	S31600	75	30	20.0	20.0	250
PLATE	STAINLESS STEEL	A 240 TP 316L	S31603	70	25	16.7	16.7	250
PLATE	STAINLESS STEEL	A 240 TP 317	S31700	75	30	20.0	20.0	250
PLATE	STAINLESS STEEL	A 240 TP 317L	S31703	75	30	20.0	19.6	250
PLATE	CHROME - NICKEL	A 240 TP 2205	S32205	95	65	43.3	27.1	250
FORGING	CARBON STEEL	A 266 Gr 2	K03506	70	36	24.0	20.0	400
PLATE	CARBON STEEL	A 283 Gr C	K02401	55	30	20.0	15.7	500
PLATE	CARBON STEEL	A 285 Gr C	K02801	55	30	20.0	15.7	500
PIPE	CHROME - MOLY	A 335 Gr P22	K21590	60	32	21.3	16.6	250
PLATE	CARBON STEEL	A 515 Gr 70	K03101	70	38	25.3	20.0	500
PLATE	CARBON STEEL	A 516 Gr 70	K02700	70	38	25.3	20.0	500
BAR	CARBON STEEL	A 675 Gr 60	na	60	30	20.0	17.1	400
BAR	CARBON STEEL (HOT ROLLED)	AISI 1020/1030		62 avg	35 avg	note 8	(note 6)	
BOLTING	ALLOY	A 193 Gr B7 ≤ 2.5"	G41400	125	105	70.0	25.0	700
TUBE SHEET	COPPER - SILICON	B 96	C65500	50	18	12.0	12.0	100
TUBE SHEET	MUNTZ	B 171 ≤ 2"	C36500	50	20	13.3	13.3	300
TUBE SHEET	ALUMINUM - BRONZE	B 171 ≤ 2"	C61400	70	30	20.0	20.0	100
TUBE SHEET	NAVAL BRASS	B 171 ≤ 3"	C46400	50	20	13.3	13.3	300
TUBE SHEET	70 / 30 COPPER - NICKEL	B 171 ≤ 2.5"	C71500	50	20	13.3	13.3	100
TUBE SHEET	90 / 10 COPPER - NICKEL	B 171 ≤ 2.5"	C70600	40	15	10.0	10.0	100
TUBE SHEET	TITANIUM	B 265 Gr 2	R50400	50	40	26.7	14.3	100
TUBE SHEET	NICKEL - CHROME	B 625	N08904	71	31	20.7	20.3	100
TUBE SHEET	NICKEL - CHROME	B 688	N08367	95	45	30.0	27.1	200

NOTES:

- 1) SELECTION OF CONDENSER MATERIALS IS THE RESPONSIBILITY OF PURCHASER.
- 2) MATERIALS LISTED ARE TYPICAL AND SHOULD NOT LIMIT THE PURCHASERS SELECTION OF OTHER SUITABLE CHOICES.
- 3) MANUFACTURER ASSUMES NO RESPONSIBILITY FOR THE DETERIORATION OF MATERIALS DUE TO EROSION, CORROSION OR ANY OTHER CAUSES.
- 4) S_u ULTIMATE STRENGTH S_y YIELD STRENGTH S_A ALLOWABLE STRESS (lb/in² x 1,000)
- 5) ALL STRESS VALUES TAKEN FROM THE 2007 EDITION OF THE ASME B&PV CODE.
- 6) FOR THESE SPECIFIC MATERIALS, S_A IS ESTABLISHED BASED ON THE FORMULA: S_A = .92 (S_u/4) @ 250 °F
- 7) S_A VALUES PROVIDED ARE BASED ON TEMPERATURES SHOWN.

APPENDIX J

CONDENSER TUBE STRESS VALVES

	ASTM	UNS NUMBER	TEMPER		S _U	S _Y	S _Y /1.5	S _A @ deg F
ADMIRALTY	B 111 (note 5)	C44300	061	ANNEALED	45	15	10.0	10.0 300
ADMIRALTY	B 111	C44400	061	ANNEALED	45	15	10.0	10.0 300
ADMIRALTY	B 111	C44500	061	ANNEALED	45	15	10.0	10.0 300
ALUMINUM BRASS	B 111	C68700	061	ANNEALED	50	18	12.0	12.0 100
ALUMINUM BRONZE	B 111	C60800	061	ANNEALED	50	19	12.7	12.7 100
70-30 COPPER NICKEL	B 111	C71500	061	ANNEALED	52	18	12.0	12.0 100
70-30 COPPER NICKEL	B 111	C71500	HR50	DRAW, STRESS RELIEVED	72	50	33.3	20.6 300
90-10 COPPER NICKEL	B 111	C70600	061	ANNEALED	40	15	10.0	10.0 100
90-10 COPPER NICKEL	B 111	C70600	H55	LIGHT DRAWN	45	35	23.3	10.0 100
PHOSPHORIZED ARSENICAL	B 111	C14200	H55	LIGHT DRAWN	36	30	20.0	10.3 250
PHOSPHORIZED ARSENICAL	B 111	C14200	H80	HARD DRAWN	45	40	26.7	12.9 250
ADMIRALTY	B 543 (note 6)	C44300	W061	ANNEALED	45	15	10.0	8.5 300
ADMIRALTY	B 543	C44400	W061	ANNEALED	45	15	10.0	8.5 300
ADMIRALTY	B 543	C44500	W061	ANNEALED	45	15	10.0	8.5 300
ALUMINUM BRASS	B 543	C68700	W061	ANNEALED	50	18	12.0	10.2 100
70/30 COPPER NICKEL	B 543	C71500	W061	ANNEALED	52	18	12.0	10.2 100
70/30 COPPER NICKEL	B 543	C71500	WC55	LIGHT COLD WORKED	54	35	23.3	(note 8)
90/10 COPPER NICKEL	B 543	C70600	W061	ANNEALED	40	15	10.0	8.5 100
90/10 COPPER NICKEL	B 543	C70600	W061	LIGHT COLD WORKED	45	35	23.3	8.5 100
COPPER IRON	B 543	C19400	W061	ANNEALED	45	15	10.0	8.5 100
COPPER IRON	B 543	C19400	WC55	LIGHT COLD WORKED	45	22	14.7	10.9 100
STAINLESS STEEL	A 213 TP 304	S30400		SEAMLESS	75	30	20.0	20.0 100
STAINLESS STEEL	A 213 TP 304L	S30403		SEAMLESS	70	25	16.7	16.7 100
STAINLESS STEEL	A 213 TP 316	S31600		SEAMLESS	75	30	20.0	20.0 100
STAINLESS STEEL	A 213 TP 316L	S31603		SEAMLESS	70	25	16.7	16.7 100
STAINLESS STEEL	A 213 TP 317	S31700		SEAMLESS	75	30	20.0	(note 8)
STAINLESS STEEL	A 213 TP 317L	S31703		SEAMLESS	75	30	20.0	(note 8)
STAINLESS STEEL	A 249 TP 304	S30400		WELDED	75	30	20.0	17.0 100
STAINLESS STEEL	A 249 TP 304L	S30403		WELDED	70	25	16.7	14.2 100
STAINLESS STEEL	A 249 TP 316	S31600		WELDED	75	30	20.0	17.0 100
STAINLESS STEEL	A 249 TP 316L	S31603		WELDED	70	25	16.7	14.2 100
STAINLESS STEEL	A 249 TP 317	S31700		WELDED	75	30	20.0	17.0 100
STAINLESS STEEL	A 249 TP 317L	S31703		WELDED	75	30	20.0	17.0 100
CHROME STEEL	A 268 TP 439	S43035		WELDED & SEAMLESS	60	30	20.0	14.6 200
CHROME STEEL	A 268	S44735		WELDED & SEAMLESS	75	60	40.0	18.2 100
CHROME STEEL	A 268 26-3-3	S44660		WELDED & SEAMLESS	85	65	43.3	20.6 300
NICKEL CHROME STEEL	B 676 CL 2	N08367		WELDED / SOLUTION ANNEALED	100	45	30.0	24.3 100
CARBON STEEL	A 179	K01200		SEAMLESS	47	26	17.3	13.4 500
CARBON STEEL	A 214	K01807		WELDED	47	26		11.4 500
TITANIUM	B 338 GR 2	R50400		ANNEALED, SEAMLESS	50	40	26.7	12.1 100
TITANIUM	B 338 GR 2	R50400		ANNEALED, WELDED	50	40	26.7	14.3 100

NOTES:

- 1) SELECTION OF TUBE MATERIALS IS THE RESPONSIBILITY OF PURCHASER.
- 2) MATERIALS LISTED ARE TYPICAL AND SHOULD NOT LIMIT THE SELECTION OF OTHER SUITABLE CHOICES.
- 3) MANUFACTURER ASSUMES NO RESPONSIBILITY FOR THE DETERIORATION OF MATERIALS DUE TO EROSION, CORROSION OR ANY OTHER CAUSES.
- 4) S_U ULTIMATE STRENGTH S_Y YIELD STRENGTH S_A ALLOWABLE STRESS (lb/in² x 1,000)
- 5) ASTM 111 DESIGNATES A SEAMLESS TUBE.
- 6) ASTM 543 DESIGNATES A WELDED TUBE.

APPENDIX K

TUBE MATERIAL PROPERTIES

	ASTM	UNS NUMBER	ρ	k (note 5)	$\alpha \times 10^{-6}$ (note 4)	$E \times 10^{+6}$
ADMIRALTY	B 111	C44300				
ADMIRALTY	B 111	C44400	.308	64.0 (a)	11.2 (a)	16.0
ADMIRALTY	B 111	C44500				
ALUMINUM BRASS	B 111	C68700	.301	58.0 (a)	10.3 (a)	16.0
ALUMINUM BRONZE	B 111	C60800	.295 (a)	46.0 (a)	10.0 (a)	17.5
70-30 COPPER NICKEL	B 111	C71500	.323	17.0 (a)	9.0 (a)	22.0
90-10 COPPER NICKEL	B 111	C70600	.323 (a)	26.0 (a)	9.5 (a)	18.0
90-10 COPPER NICKEL	B 111	C70600				
PHOSPHORIZED ARSENICAL	B 111	C14200	.323	112.0 (a)	9.8 (a)	17.0
ADMIRALTY	B 543	C44300				
ADMIRALTY	B 543	C44400	.308	64.0 (a)	11.2 (a)	16.0
ADMIRALTY	B 543	C44500				
ALUMINUM BRASS	B 543	C68700	.301	58.0 (a)	10.3 (a)	16.0
70/30 COPPER NICKEL	B 543	C71500	.323	17.0	9.0 (a)	22.0
90/10 COPPER NICKEL	B 543	C70600	.323	26.0 (a)	9.5 (a)	18.0
COPPER IRON	B 543	C19400	.322 (a)	150.0 (a)	9.8 (a)	17.5 (a)
STAINLESS STEEL	A 213 TP 304	S30400	.290	8.6 (5)	8.5	28.3
STAINLESS STEEL	A 213 TP 304L	S30403				
STAINLESS STEEL	A 213 TP 316	S31600				
STAINLESS STEEL	A 213 TP 316L	S31603	.290	8.2 (5)	8.6	28.3
STAINLESS STEEL	A 213 TP 317	S31700				
STAINLESS STEEL	A 213 TP 317L	S31703				
STAINLESS STEEL	A 249 TP 304	S30400	.290	8.6 (5)	8.5	28.3
STAINLESS STEEL	A 249 TP 304L	S30403				
STAINLESS STEEL	A 249 TP 316	S31600				
STAINLESS STEEL	A 249 TP 316L	S31603	.290	8.2 (5)	8.6	28.3
STAINLESS STEEL	A 249 TP 317	S31700				
STAINLESS STEEL	A 249 TP 317L	S31703				
CHROME STEEL	A 268 TP 439	S43035	.278 (c)	14.0 (c)	5.6 (c)	29.0 (c)
CHROME STEEL	A 268	S44735	.277 (e)	9.9 (e)	5.2 (e)	30.0 (e)
CHROME STEEL	A 268 26-3-3	S44660	.279 (f)	9.2 (5,f)	5.4 (f)	31.0 (f)
NICKEL CHROME STEEL	B 676 CL 2	N08367	.291 (d)	6.8 (d)	8.5 (d)	28.3 (d)
CARBON STEEL	A 179	K01200	.283 (b)	27.5 (5)	7.3	29.5
CARBON STEEL	A 214	K01807				
TITANIUM	B 338 GR 2	R50400	.163	12.7 (5)	4.6	15.5

NOTES:

- 1) SELECTION OF TUBE MATERIALS IS THE RESPONSIBILITY OF PURCHASER.
- 2) MATERIALS LISTED ARE TYPICAL AND SHOULD NOT LIMIT THE SELECTION OF OTHER SUITABLE CHOICES.
- 3) MANUFACTURER ASSUMES NO RESPONSIBILITY FOR THE DETERIORATION OF MATERIALS DUE TO EROSION, CORROSION OR ANY OTHER CAUSES.
- 4) ALL PROPERTY VALUES TAKEN FROM THE 2007 EDITION OF THE ASME B&PV CODE EXCEPT AS NOTED BELOW:
 - (a) COPPER DEVELOPMENT ASSOCIATION (www.copper.org) JAN 27, 2006
 - (b) STRUCTURAL ALLOYS HANDBOOK MECHANICAL PROPERTIES DATA CENTER , 1976
 - (c) ALLEGHENY LUDLUM (TECHNICAL DATA BLUE SHEET) TYPE 439/AL, 1999
 - (d) ALLEGHENY LUDLUM (TECHNICAL DATA BLUE SHEET) AL6XN, 2002
 - (e) ALLEGHENY LUDLUM (TECHNICAL DATA BLUE SHEET) AL 29-4C, 2010
 - (f) PLYMOUTH TUBE (DATA SHEET) SEA CURE, 2005
- 5) THERMAL CONDUCTIVITY VALUES ARE AT 68°F EXCEPT THOSE DENOTED WITH (5), WHICH WERE DETERMINED AT 70°F.

APPENDIX L

CONDENSER PERFORMANCE

ABSTRACT

The Heat Exchange Institute promulgates standards for mechanical design, performance and testing for various types heat exchange, vacuum producing and related equipment. This article concerns itself with steam surface condenser performance. It is a brief history of the HEI followed by design considerations for both the condenser as a whole and tube bundles individually that effect heat transfer rate. Finally, the HEI heat transfer rate "U" is defined.

HISTORY

The Heat Exchange Institute was formed in 1933 as part of the National Recovery Act program to bring order to the market segment that included steam surface condensers, and low level jet and barometric condensers. Prior to this time, there were varying claims of heat transfer rates based on development work by various manufacturers. As a result, the end user was unable to determine the viability of the various offerings. This same uncertainty of what performance could be purchased also hampered preliminary plant design efforts and planning.

Consequently, the HEI's first goal was to provide a practical performance standard that the entire industry could rely on. The HEI immediately concentrated its efforts on publishing overall heat transfer rates for condensers based on data derived from actual field tests. By 1940 this approach was abandoned because of the extreme variations in operating conditions of the various field installations.

Since the field testing data would not allow a satisfactory and reliable standard, the HEI contracted with Lehigh University in 1941 to develop calorimeter testing procedures. Extensive laboratory testing proceeded and the working arrangement was in effect for over 40 years. Information from this testing was correlated with the actual field performance data to produce technically accurate data for publication in the Condenser Standards. These standards have been periodically updated as new data and information has become available.

The resulting HEI condenser performance standard achieved its intended mission. The overall tube bundle heat transfer rates insured the users that they could rely on the results in their plant design and were assured they could purchase the performance from competent manufacturers.

CONDENSER OPERATION AND DESIGN CONSIDERATIONS

To understand the HEI performance, it is necessary to understand the condenser as a whole unit and not just a single tube or tube bundle. The condenser has many functions that it is required to perform beyond the primary function of condensing steam.

As we look at these functions and how they affect thermal performance, the following example will be used.

Steam load (lb/hr)	1,008,735
Heat Rejected (BTU/lb)	978.55
Heat Load (BTU/hr)	987,095,000
Abs press (" Hga)	2.84
Steam temperature (°F)	113.13
Inlet Water temperature (°F)	92.0
Initial Temp Diff (°F)	21.13
Number of passes	single pass vertically divided
Effective tube length (ft-in)	34'-0"
Tube OD - BWG	3/4" - 18
Tube material	Al Brass
HEI Correction factors:	
Water Temp	1.08
Percent Clean/100	.85
Material & Gauge	.993
Velocity (fps)	6.15
Temp Rise (°F)	14.96
Terminal Temp Diff (° F)	6.17
Cooling Water (GPM)	132,000
Number of tubes	19,765
Surface (Sq Ft)	131,916

This example is single pass and down turbine exhaust rectangular configuration for ease of discussion only. All information contained herein is applicable regardless of the number of passes, exhaust orientation, or geometry.

The body of the condenser isolates the condenser steam space from the surrounding atmosphere. The section (Figures 1 & 1A) below the bundle is the "hot-well" whose primary function is to store the condensate in a quantity meets the system requirements.

The section that contains the tube bundles is the "shell" whose primary function is support of the tubes and provide proper distribution of steam to the tube bundle.

The "transition piece" provides the geometry that connects the turbine exhaust. The primary function of the transition piece is to distribute the steam appropriately to the tube bundles.

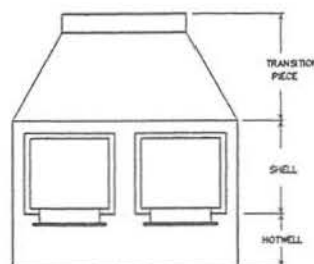


FIGURE 1

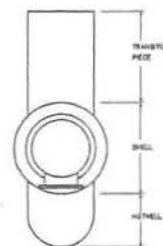


FIGURE 1-A

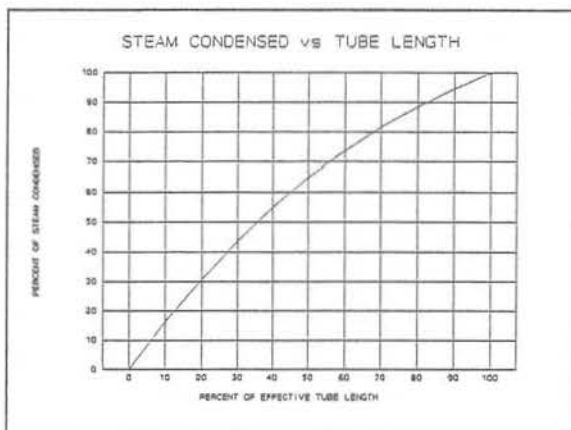
APPENDIX L

CONDENSER PERFORMANCE

Examining the conditions at the inlet to the condenser, it should be noted that the turbine exhaust steam flow is neither uniform nor dry. The moisture content of the incoming steam normally ranges from 6-12% and in some cases such as nuclear or geothermal cycles may be up to 15-20%.

The flow pattern from any turbine, is not uniform. Not only does the flow pattern vary between turbine manufacturers and turbine types, but streak areas can be 2 to 2-1/2 times the average velocity. It is under these high velocity streak areas of the turbine exhaust that erosion, if it is to occur, will show up on the tube bundle.

A natural characteristic of any condenser is to condense more steam in the first half of the tube length. Typically this first half/second half breakdown is in the 60-70% / 40-30% range. The example depicted on Graph A condenses 65% in the first half of the tube length.



GRAPH A

The transition piece must also function to provide for distribution of steam to the ends of the tube bundles with a minimum of pressure loss. The example has a distribution pattern as shown in Figure 2. The longer a condenser is, the more severe the requirement to distribute steam to the ends is while keeping the pressure loss to a minimum.

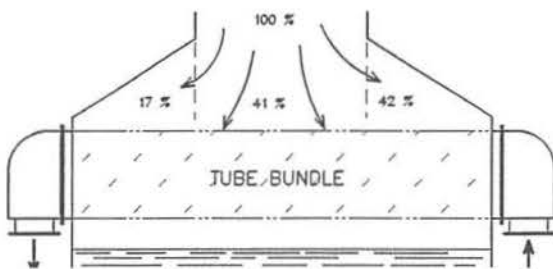


FIGURE 2

The loss associated with distributing steam to the ends of the condenser and flow around feedwater heaters, extraction piping, bracing, etc. all tend to interfere with proper steam distribution to the tube bundles. This affecting the steam temperature and pressure at the entrance to the tube bundle at various positions along the length. The effect is a lengthwise variation from uniform heat transfer along the tube length.

The shell, which contains and support the tubes, must also provide adequate area for the distribution of the steam around the tube bundle as depicted in Figures 3 and 3-A:

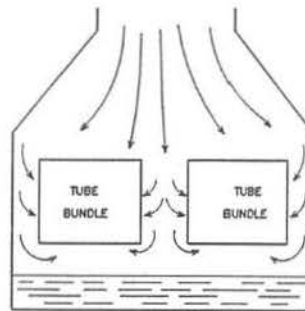


FIGURE 3



FIGURE 3-A

Open channels at the side walls and between tube bundles must allow steam to be distributed at a minimum of flow loss.

In addition to the primary function of condensing steam, the condenser is required to deaerate and reheat the condensate. This requires sufficient space below the bottom tube bundle and above the condensate level. The result of this extra space reduces steam velocities to the lower part of the tube bundle.

The condenser is also the lowest pressure point in the power plant. As such, all drains return to this point. The drains vary from subcooled liquid to flashing liquid to superheated steam. Due to the inlet drain conditions and their proximity to the tube bundle, the steam entering a given area may be different than the assumed saturation conditions that the thermal calculations were based on. Take, for example, the plant makeup connection which usually is subcooled water. To meet the requirements of deaeration and reheat it is normally introduced in a spray header in the transition piece.

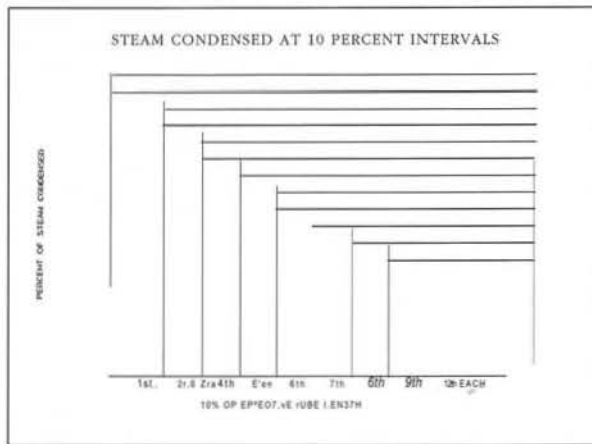
The operation and design conditions of the condenser operating within a system varies from installation to installation. Since actual conditions are virtually impossible to determine design, any prudent standard for heat transfer must take into consideration all of these effects.

APPENDIX L

CONDENSER PERFORMANCE

TUBE BUNDLE OPERATION AND DESIGN CONSIDERATIONS

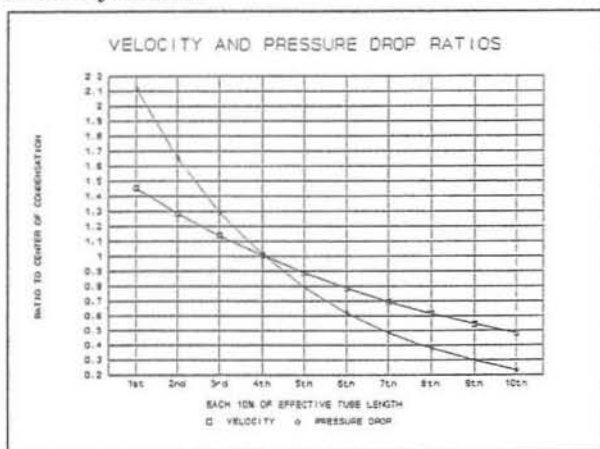
Thermal characteristics of condenser cause varying amounts of steam to be condensed along the length of the tubes. If we take our example unit and divide the tube length into ten discrete sections, it is evident from Graph B that the cold (inlet) section of the condenser will condense about three times more than the hot (outlet) section.



GRAPH B

Any condenser tube bundle design will have a fixed geometry throughout its entire length. The tube sheet pattern fixes the steam flow area into the tube bundle.

Knowing that the geometry is fixed and considering the variation in steam condensed along the length, it can be shown that velocities and losses into and through the tube bundle will also vary. The example has a center of condensation at approximately 35% from the cold end of the condenser. Graph C relates velocities into and pressure loss through the tube bundle to the center of condensation (located in the fourth section) for each of the ten condenser sections previously chosen.



GRAPH C

The variation in the steam condensed in any section coupled with the corresponding variation in the velocity in that section directly relates to the condensate boundary layer. Variation in the condensate boundary layer thickness will also be different for each design and each set of operating conditions.

Figure 4 illustrates a typical tube sheet design that could be selected for the example condenser. The tube sheet shown is used to illustrate the design considerations and does not represent an actual design,

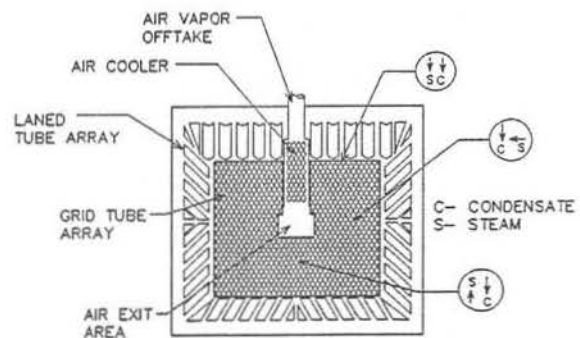


FIGURE 4

Tube configurations will dictate different flows to the various sections of the tube bundle. The example could be arranged to have 40% of the steam entering the top, 20% on each side and 20% in the bottom. It is the designer's responsibility to produce a tube pattern that has equal pressure loss from each quadrant to the air cooler to ensure that the noncondensibles can be properly evacuated from the tube bundle. The designer varies the lanes, grids pattern, and air exit area to accomplish this goal.

The upper part of the tube bundle is typified by concurrent condensate and steam flow, the side sections by perpendicular flows of condensate and steam, while the bottom sees counter current flow. Each part of the bundle will therefore have different heat transfer characteristics.

The lower tubes are exposed to more condensate from the upper tubes adding to the condensate boundary layer. The larger the tube bundle and/or the higher the condensing rate, the more significant the effect of condensate flooding.

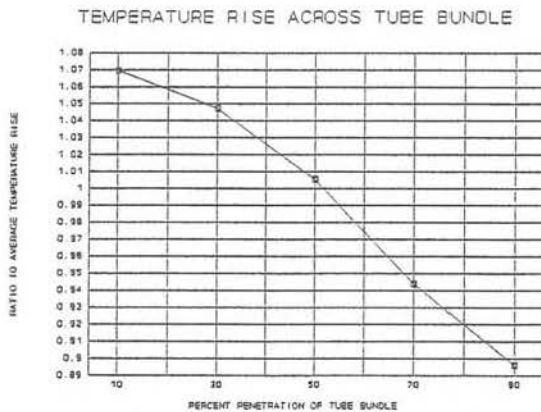
As the steam travels from the outside of the tube bundle to the air cooler, (a) the flow volume is reduced and the velocity diminishes, (b) the absolute pressure decreases with an associated decrease in steam temperature, and (c) the percentage of noncondensibles increases with flow penetration. As these considerations are taken into account, it can be seen that the heat transfer rates for various sections of the tube

APPENDIX L

CONDENSER PERFORMANCE

bundle will be different. Generally the top outside tubes have the highest heat transfer rate with the inside tubes near the air cooler having the lowest rate.

The change in heat transfer rate can be detected on any condenser by thermally probing cooling water temperature rise changes across the outlet tube sheet. The preliminary design estimate taken from the example would suggest that this would range from a little over 16°F temperature rise for the outer most tube to a little under 13°F temperature rise for the innermost tube, compared to the calculated 15°F average temperature rise. The ratio of this change is depicted on Graph D.



GRAPH D

Every tube bundle has a variation in heat transfer rate across its depth and along its length for the factors previously delineated. Consequently the performance of any tube array cannot be determined until the actual tube array is complete.

HEI HEAT TRANSFER RATE "U"

The heat transfer rate ("U" value) published by the HEI is an OVERALL TUBE BUNDLE "U" VALUE to be obtained by the condenser under actual operating conditions. Because this "U" value takes into account parameters other than the basic heat transfer across the wall of the tube, it is not meant to be used as the specific individual tube "U" value.

The mandate of the HEI is to promulgate a heat transfer standard that every purchaser can rely on when issuing specifications for surface condensers and one that every competent designer can meet. The

standard is workable over a wide range of conditions and over a wide range of applications, taking into consideration information from tube calorimeter tests and actual field performance tests.

For end users to be able to rely on performance to the HEI "U" value, the designer/ fabricator must be able to fabricate a condenser that will operate to this overall "U" value. The condenser designer must know the characteristics of the tube arrays used under varying conditions within the tube bundle. Competent designers will have embarked on their own research and testing program to determine the effects of these characteristics. Values determined from testing programs are then used to arrange a tube bundle assuring that the HEI requirements are met. In this way the designer/fabricator can provide performance guarantees that are essential to plant design.

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