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for
SHELL AND TUBE HEAT EXCHANGERS

FIFTH EDITION

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HEAT EXCHANGE INSTITUTE, INC.

STANDARDS for **SHELL & TUBE HEAT EXCHANGERS**

FIFTH EDITION

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FOREWORD

The HEI has changed the name of Standards for Power Plant Heat Exchangers to Standards for Shell and Tube Heat Exchangers. The fifth edition of these standards has been developed by the Shell and Tube Heat Exchanger Section of the Heat Exchange Institute, Inc. The technical information in these standards combines present industry standards, typical Purchaser requirements, and Manufacturers' experience and outlines the important design criteria for power plant heat exchangers.

These standards provide practical information on nomenclature, dimensions, design, testing, construction and performance. Use of the standards will ensure a minimum of misunderstanding between Manufacturer and Purchaser and will assist in the proper selection of equipment best suited to the requirements of the application.

These standards represent the collective experience of the Section members and provide a guide in the writing of specifications and in the selection of heat exchangers for power plant use.

In the preparation of these standards, consideration has been given to the work of other organizations, such as the American National Standards Institute, the American Society of Mechanical Engineers and others. Credit is hereby given to all those whose standards may have been helpful in this work.

The publication of the fifth edition of the Standards for Shell and Tube Heat Exchangers represents another step in the Heat Exchange Institute's continuing program to provide standards which reflect the latest technological advancements in the field of heat exchange equipment. The Standards for Shell and Tube Heat Exchangers are continually reviewed by the Technical Committee at scheduled meetings under the direction of the Shell and Tube Heat Exchanger Section. Suggestions for improvement of these Standards are welcome and should be sent to the Heat Exchange Institute, Inc., 1300 Sumner Avenue, Cleveland, Ohio 44115-2185, or via telephone at 216-241-7333, via fax at 216-241-0105, or email the Heat Exchange Institute, Inc. at hei@heatexchange.org. Additional information about the Heat Exchange Institute, Inc. can be found at www.heatexchange.org.

1.0 SCOPE AND PURPOSE

1.1 Scope

These Standards are intended to apply to shell-and-tube type heat exchangers containing bare or extended surface tubes used primarily in power plants.

Shell and tube heat exchangers transfer heat indirectly from one fluid to another by means of a number of tubes arranged in an optimized pattern enclosed by a cylindrical shell. In general, one fluid flows through the tubes and the other flows over and along the length of the exchanger.

Almost every exchanger is custom designed and constructed. Requirements can vary over a wide range of pressures and temperatures. For each basic construction, there are many possible variations in configurations to suit special operating needs such as designing for desired heat transfer, lower pressure drop, corrosion, fouling, erosion etc. They are the most versatile of any other heat transfer equipment in terms of materials used for the construction and size. Maintenance such as cleaning, replacement of failure prone components etc. can be made relatively easy.

Depending on the state of the working fluids, exchangers can be classified into liquid-liquid exchangers, gas-liquid exchangers, and gas-gas exchangers.

Some of the commonly used names for the heat exchangers to which these Standards apply are listed below. It is not intended that this list be all-inclusive or that it limit the use of these Standards to only those heat exchangers named.

Auxiliary Steam Generators
Bearing Water Coolers
Blowdown Exchangers
Bypass Condensers
Cleanup Exchangers

Component Cooling Water Exchangers
Condensate Coolers
Fuel Oil Heaters
Fuel Pool Coolers
Fuel Reprocessing Exchangers
Geothermal Units
Glycol/Glycol-Water Heaters
HTGR Exchangers
Jacket Water Coolers
Liquid Metal Exchangers
Lube Oil Coolers
Preheaters
Radwaste Treatment Exchangers
Reactor Building Exchangers
Reboilers and Evaporators
Residual Heat Removal Exchangers
Turbine Building Exchangers

It is not intended that these Standards be applied to heat exchange equipment covered by other HEI Standards, such as feedwater heaters, condensers, etc.

1.2 Purpose

These Standards have been developed to be used by heat exchanger Purchasers and Manufacturers to delineate some of the pertinent thermal, hydraulic, and mechanical design features and requirements for heat exchangers used in power plants.

It is intended that these Standards provide a basis for a mutual understanding and interpretation of heat exchanger requirements between the Purchaser and Manufacturer and assist in specifying, designing, and fabricating heat exchangers.

Most of the heat exchangers covered by these Standards may also be required to conform to the Design Specification and the ASME Boiler and Pressure Vessel Code, Section III, Division 1, Class 1, 2, or 3, or Section VIII, Division 1 or 2.

2.0 DEFINITIONS

2.1 Cleanliness Factor

The cleanliness factor is the ratio of the overall heat transfer coefficient to the clean overall heat transfer coefficient.

2.2 Code

For the purpose of these Standards, the Code refers to the ASME Boiler and Pressure Vessel Code, Section III, Division 1, Class 1, 2, or 3, or Section VIII, Division 1 or 2.

2.3 Design Point

The design point is the set of operating conditions and constraints which are to be satisfied by the heat exchanger.

2.4 Design Pressures

The design pressures are the pressures for which the shell and tube sides of the exchanger are structurally designed.

2.5 Design Temperatures

The design temperatures are the temperatures for which the shell and tube sides of the exchanger are structurally designed.

2.6 Effective Surface

The effective surface in the heat exchanger is the external tube surface used for heat transfer.

2.7 Fouling Resistance

Fouling resistance is a resistance to heat flow caused by the deposition of corrosive products, dirt, or other foreign material on the inside or outside surface of the tubes.

2.8 Gross Surface

The gross surface in the heat exchanger is the total external tube surface.

2.9 Heat Exchanger Approach Temperature

The heat exchanger approach temperature is the temperature difference between the hotter fluid exit temperature and the colder fluid entrance temperature.

2.10 Heat Exchanger Boundaries

For the purpose of these Standards, the boundaries of the heat exchanger extend from the inlet nozzles to the outlet nozzles on both the shell side and the tube side. The boundaries also include foundation supports welded to the heat exchanger pressure parts.

2.11 Heat Exchanger Duty

Heat exchanger duty is the heat transferred per unit of time from one fluid to another.

2.12 Logarithmic Mean Temperature Difference (LMTD)

Logarithmic mean temperature difference is a mathematical relationship expressing the integrated thermal driving potential for transferring heat between the shell side and tube side fluids in true counterflow or parallel flow heat exchangers. Formula can be found in Appendix B, (see B1.0).

2.13 LMTD Correction Factor

The logarithmic mean temperature difference correction factor is used to adjust for deviation from true counterflow within multipass or various other flow arrangements of counterflow heat exchangers (see B2.0).

2.14 Mean Temperature Difference (MTD)

The mean temperature difference is the integrated thermal driving potential for transferring heat between the shell side and tube side fluids in heat exchangers. In counterflow heat exchangers with multipass or various other flow arrangements, a correction factor is applied to the logarithmic mean temperature difference to obtain the mean temperature difference; it is the product of the LMTD times the correction factor (see B2.0).

2.15 Operating Pressures

The pressures for which the shell and tube sides of the exchanger are thermally and hydraulically rated.

2.16 Operating Temperatures

The temperatures for which the shell and tube sides of the exchanger are thermally and hydraulically rated.

2.17 Overall Heat Transfer Coefficient

The overall heat transfer coefficient is the average heat transfer rate between the tube side and shell side fluids under specified fouling conditions. The overall heat transfer coefficient is commonly referred to as the service rate and is the inverse of the sum of all of the heat transfer resistances (including fouling resistances). The inclusion of fouling resistances (or application of a cleanliness factor) increases the amount of required effective surface so that if fouling occurs over time, the increased surface will allow acceptable performance for some additional amount of time (see 3.2). When the overall heat transfer coefficient does not include fouling resistances (or application of a cleanliness factor), it is commonly referred to as the clean rate. These rates are shown on line 33 of the heat exchanger specification sheet (see Appendices F-1, F-2, and F-3).

2.18 Pressure Loss

The pressure loss of a fluid traveling through the heat exchanger tube side or shell side consists of the irrecoverable loss in operating pressure as the fluid stream travels from one boundary of the heat exchanger to the other.

The tube side pressure loss includes the loss through the inlet and outlet nozzles, the channels, and the tubes. The shell side pressure loss includes the loss through the inlet and outlet nozzles and the bundle. The tube or shell side pressure loss does not include any change in static head.

3.0 HEAT EXCHANGER PERFORMANCE

3.1 Exchanger Performance

Although heat exchangers may be operated under a number of different conditions, the design should be predicated on one specific set of operating conditions termed the "design point". For the respective flow rates and inlet temperatures, the heat transfer requirements must be satisfied by meeting the respective heat exchanger duty and the outlet temperatures. For the respective flow rates, the maximum allowable pressure losses must not be exceeded.

The procedures of the ASME Power Test Code for the measurement of temperature, pressure, and flow may be followed in evaluating the performance capability of any heat exchanger built to these Standards.

3.1.1 Minimum Data Required to be Supplied by the Purchaser

- (1) General information
 - Plant location:
 - Service of unit/item number:
 - Position: (horizontal or vertical)
 - Arrangement: (single or multiple stream)
 - Space limitations: (overall length or overall length plus withdrawal clearance)
 - Unit type: (U-tube, floating head, removable bundle, fixed tubesheets, etc.) See Appendix M
 - Heat exchanger duty: (if outlet temperatures are not specified) Btu/hr

Cleanliness factor: (if fouling resistances are not specified)

(2) Tube Side and Shell Side Parameters

Fluid name... _____
Fluid composition... _____
Fluid flow rate... _____ lbm/hr
Fluid temperature-in... _____ °F
Fluid temperature-out... _____ °F
(if duty is not specified)
Fluid velocity-maximum @ °F... _____ ft/sec
Fluid pressure loss-maximum... _____ psi
Fluid connection sizes... _____ in
Design pressure... _____ psig
Minimum design temperature... _____ °F
Maximum design temperature... _____ °F
Operating pressure... _____ psia
Fouling resistance... _____ hr-ft²-°F/Btu
(if cleanliness factor is not specified)
Corrosion allowance... _____ in
Applicable Code Section/Division/Class... _____
Material requirements... _____
Steam quality (if applicable)... _____ mass %
Blowdown (if applicable)... _____ lbm/hr

Thermodynamic properties, including density, viscosity, specific heat, thermal conductivity, and latent heat, should be provided for fluids where data is not readily available.

(3) Overload and Abnormal Conditions

It is possible that severe loads (either hydraulic or thermal) may occur when the exchanger is operated at other than the design point conditions. To ensure that all factors are taken into consideration in the design of a heat exchanger, the following information shall be provided by the Purchaser to enable the Manufacturer to perform a comprehensive fatigue and operability analysis.

Mode of Operation

Tube side and shell side fluid parameters
(see 3.1.1 (2))

Maximum allowable pressure losses for abnormal operating conditions

Transients (thermal and hydraulic)

Chemical cleaning thermal conditions, if any

When such data is not provided, the Manufacturer's design shall be limited to steady state conditions.

3.2 Fouling Resistance and Cleanliness Factor

It is recommended that fouling resistances be applied to both the inside and outside tube surfaces, as all heat transfer fluids cause fouling to some degree. Fouling resistances are more difficult to quantify than other thermal parameters since they depend on a number of factors. The purchaser shall specify the fouling resistances or cleanliness factor (see 3.3.1). The fouling resistance is responsible for specifying material suitable for the fluid chemistry, pressure, and temperature to avoid erosion/corrosion, stress corrosion cracking, galvanic action, etc.

3.2.1 Types of Fouling

Most types of fouling which occur in power plant heat exchangers can be classified as follows:

- (1) **Corrosion.** This usually occurs in the form of an oxide layer. It is more prevalent with carbon steel tubing.
- (2) **Crystallization.** This is the formation of a salt scale, especially calcium carbonate on the tubes as a result of minerals in the water in excess of the saturation point.
- (3) **Solids.** These are in the form of silt, suspended dust particles, corrosion particles, etc.
- (4) **Biological Growth.** This is caused by a number of organisms that can attach to the tubing, such as algae, mussels, etc. They can build up rapidly, reducing the heat transfer rate and in some cases severely restricting the flow.
- (5) **Hydrocarbon deposits.** When hydrocarbons are exposed to high temperatures, a hard crust can form on the tubing.

3.2.2 Ways to Minimize Fouling

Although it is not always practical, periodic cleaning can be used to substantially reduce fouling. This can be accomplished by mechanical cleaning (cleaning balls, brushes, etc.). Chemicals can be introduced into either or both of the heat transfer fluids to effect a partial removal of foulants from the tubing. Backwashing can be used to sweep away loose particles. Thermal shocking can be used to break up mineral deposits.

It is recommended that the fluid which fouls most rapidly be circulated through the tubes, thereby avoiding the accumulation of particles in stagnant areas.

Tubeside velocities less than 2 fps should be avoided, with velocities above 3 fps preferable. Likewise, excessively low shell side velocities should be avoided.

Untreated water should be avoided since it may contain an appreciable amount of minerals, microorganisms, silt, etc.

3.2.3 Fouling Resistance Versus Cleanliness Factor

If the Purchaser specifies a cleanliness factor in lieu of fouling resistances, the clean overall heat transfer coefficient shall be multiplied by the cleanliness factor to determine the overall heat transfer coefficient. Figure 1 is provided to illustrate the relationship between fouling resistance, cleanliness factor and overall heat transfer coefficient. For example, a heat exchanger with a total fouling resistance of 0.001 hr-ft²-°F/Btu and an overall heat transfer coefficient of 200 Btu/hr-ft²-°F has a cleanliness factor of 80%. If the overall heat transfer coefficient increases to 400 Btu/hr-ft²-°F, the cleanliness factor will be 60%.

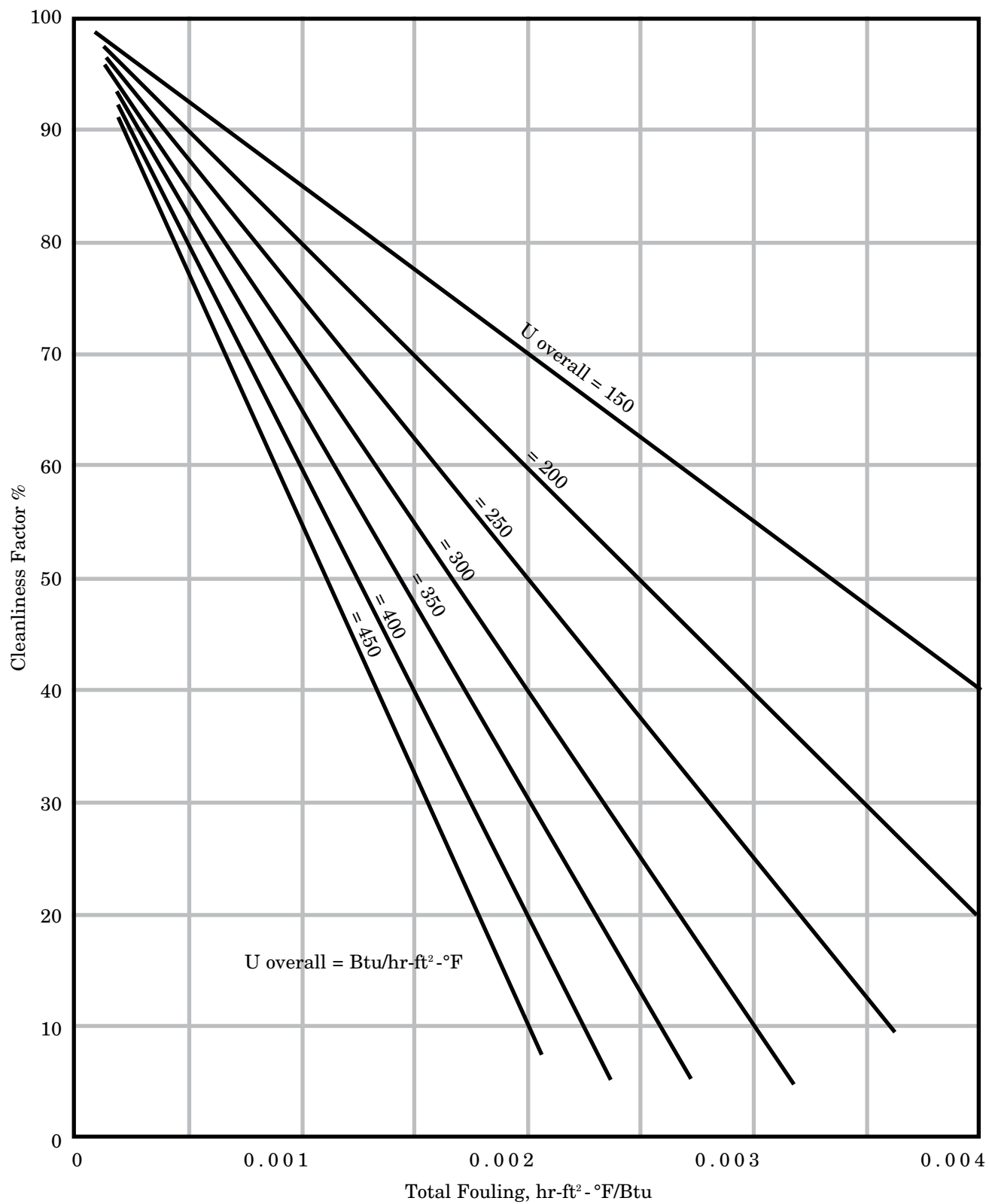


Figure 1
CLEANLINESS FACTOR-TOTAL FOULING COMPARISON

3.2.4 Performance Monitoring

If the Purchaser will monitor the increase of fouling with time, then it is recommended that baseline performance testing be done while the amount of fouling is minimal. This should be done as soon as possible after the installation of the heat exchanger. Any subsequent deterioration in performance will be attributable to an increase of fouling, and the amount of increase in fouling resistance can readily be calculated.

3.2.5 Representative Fouling Resistances

Table 1 presents fouling resistances typically used in power plant heat exchangers.

Table 1
Representative Fouling Resistances
hr-ft²-°F/Btu

Fluid	Fouling Resistances (Range)
Cooling Tower Water (treated)	0.0005 to 0.0015
Demineralized Water	0.0005 to 0.001
Treated Condensate	0.0003 to 0.001
Sea Water	0.0005 to 0.003
Brackish Water	0.001 to 0.005
River Water	0.0005 to 0.003
Boiler Blowdown	0.001 to 0.003
Oil-free Steam	0.000 to 0.0005
Oil-bearing Steam	0.0005 to 0.0015
Number 6 Fuel Oil	0.002 to 0.020
Number 2 Fuel Oil	0.002 to 0.020
Lube Oil	0.001 to 0.005
Ethylene Glycol Solutions	0.0005 to 0.0015
Industrial Heat Transfer Fluids	0.0005 to 0.002

3.3 Heat Exchanger Approach Temperature

The Purchaser, by stipulating the design point, specifies the heat exchanger approach temperature. Generally, as the approach temperature decreases, the required heat exchanger surface increases.

The selection of the approach temperature affects the hot and cold fluid flows which, in turn, affects plant operating costs. Care should be taken to consider capital costs versus operating costs.

When multipass arrangements are used, care should be taken to ensure that the exchanger does not operate in a thermally unstable region; that is, the LMTD correction factor should not be subject to large fluctuations with small changes in inlet parameters.

3.4 Tube Velocity

The fluid velocity through the tubes at the average temperature for the design point should not exceed the values contained in Table 2. These velocities are applicable to water of boiler feed quality. Lower velocities should be considered when erosive fluids are present.

Table 2
Maximum Tube Velocity

Tube Material	Fluid Velocity ft/sec
Stainless Steel, Nickel Alloys, Titanium	10.0
Copper-Nickel (70-30, 80-20, 90-10)	9.0
Low Alloy Steel	9.0
Admiralty, Copper, Aluminum-Brass	8.5
Carbon Steel	8.0

3.5 Pressure Loss

The allowable shell side and tube side pressure losses shall be specified by the Purchaser. By specifying as high a shell side pressure loss as economically justifiable, the Purchaser allows the designer to minimize the baffle pitch and thus, minimize the unsupported tube length. By minimizing the unsupported tube length, the potential for detrimental tube vibration is reduced. Also, a shorter baffle pitch normally contributes to a higher cross-flow velocity on the shell side which improves the heat transfer coefficient. Generally, as the allowable shell side and tube side velocities increase, the heat exchanger surface and the potential for fouling decrease.

3.5.1 Tube Side Pressure Loss

Below is a method of determining the tube side pressure losses from and including the channel inlet and outlet nozzles (pressure losses are calculated for friction, nozzles, tube entrance, exit and turning). This method is applicable to either straight or U-tubes.

This method is only applicable to clean smooth tubes with turbulent flow ($Re \geq 3,000$) and no change of phase. It is a condensed method to check pressure losses in the evaluation of the equipment. In the event of multiple tube gauges, the nominal I.D. is the mean effective value.

Inside Tubes

$$\Delta P_T = \frac{8.705 \times 10^{-6}}{\rho} \left(\frac{w}{A_t} \right)^2 N_f \frac{L}{d}$$

$$\text{where: } f = 0.0014 + 0.125(Re)^{-0.32}$$

$$Re = \frac{wd}{0.2014\mu A_t}$$

Nozzle Losses

$$\Delta P_{NI} = \frac{0.1995 \times 10^{-6}}{\rho} \left(\frac{w}{A_N} \right)^2$$

$$\Delta P_{NO} = \frac{0.1270 \times 10^{-6}}{\rho} \left(\frac{w}{A_N} \right)^2$$

Tube Entrance, Exit and Turn Losses

$$\Delta P_E = \frac{0.1814 \times 10^{-6}}{\rho} K_t N \left(\frac{w}{A_t} \right)^2$$

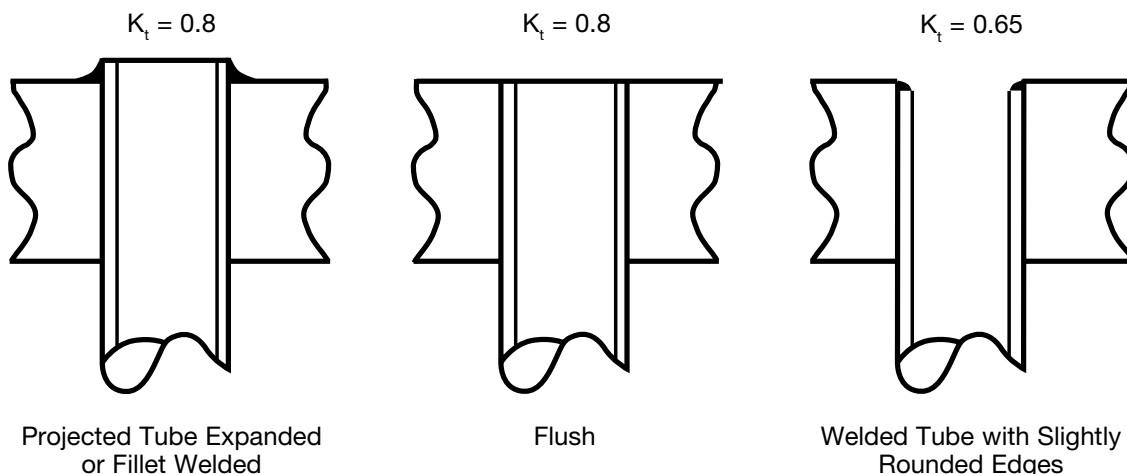


Figure 2
LOSS CORRECTION FACTOR (K_t) FOR MULTIPLE PASSES

Total Pressure Loss

$$\Delta P_{TOT} = \Delta P_T + \Delta P_{NI} + \Delta P_{NO} + \Delta P_E$$

NOTE: CONSTANTS SHOWN INCLUDE A 5% SAFETY FACTOR.

Definitions

ΔP_{TOT} = Total nozzle-to-nozzle tube side pressure loss, psi

ΔP_T = Pressure loss through tubes, psi

ΔP_{NI} = Pressure loss through channel inlet nozzle, psi

ΔP_{NO} = Pressure loss through channel outlet nozzle, psi

ΔP_E = Tube entrance, exit, and turn losses, psi

w = Tube side flow, lbm/hr

ρ = Effective tube side density, lbm/ft³

μ = Viscosity, cp

L = Tube length per pass, ft

A_t = Flow area of tubes per pass, in²

d = Nominal inside diameter of tube, (See Appendix H or I)

f = Friction factor

A_N = Tube side nozzle area, in²
Note: For tapered nozzles use mean area.

K_t = Loss correction factor for tube configuration. For a single pass, $K_t = 0.9$. For multiple passes, (see Figure 2)

N = Number of tube passes

Re = Reynolds number

3.6 Nozzle Sizes

It is recommended that the nozzle sizes be selected so that the criteria in Table 3 will not be exceeded at the design point. It is necessary that the flow entering the exchanger be uniform across the nozzle cross-section. The Purchaser shall design the piping to ensure that the exchanger is not subject to high local velocities due to a reducer, elbow, valve, or other fitting close to the nozzle. If a standard pipe reducer is utilized on external piping to the heat exchanger in order to meet the maximum criteria listed below, it is recommended that the reducer be located at least 10 times the larger diameter from the point of attachment to the heat exchanger. Piping configurations which produce non-uniform flow patterns may result in accelerated wear on the internal components of the heat exchanger.

Table 3
Nozzle Size Criteria

	Maximum G^2/ρ in Nozzle⁽¹⁾			
	Tube Side Nozzles		Shell Side Nozzles	
	Inlet	Outlet	Inlet	Outlet
Liquids (Subcooled)	6200	6200	4000	4000
Liquids (Near Saturation Point)	1000	250	1000	250
Gases/Dry Vapors	2000	2000	2000	2000

⁽¹⁾ G = Mass velocity, lbm/sec-ft²

ρ = Density, lbm/ft³

At the discretion of the designer, these values should be lowered to account for the effect of fluids containing entrained droplets, bubbles, foreign matter, etc.

3.7 Shell Inlet Area with Impingement Devices

The use of an impingement device at the shell inlet nozzle is dependent upon the Manufacturer's design of the heat exchanger and is a function of the fluid inlet velocity and the fluid properties. All heat exchangers containing erosive fluids require an impingement device at the shell inlet nozzle.

The impingement device shall be sized assuming a minimum angle of diffusion of 15 degrees from the point at which the nozzle penetrates the shell (see Figure 3).

When an impingement device is used, it shall be located such that the unrestricted flow area between the inside diameter of the shell at the nozzle and the top of the impingement device along with any open area through the impingement device is equal to or greater than the area calculated using the allowable value of G^2/ρ from Table 3.

3.7.1 Shell Inlet Area with Impingement Plate

The unrestricted flow area is the radial surface area of the volume described by the projection of the nozzle into the shell (shown as B in Figure 3).

3.7.2 Shell Inlet Area with Impingement Rods

The unrestricted flow area is the radial surface area of the volume described by the projection of the nozzle into the shell plus the open area between the first row of the impingement rods within the radial surface area (shown as A in Figure 4). A minimum of two rows of staggered impingement rods is required as shown in Figure 4.

3.7.3 Shell Inlet Area with Perforated Impingement Plate

The unrestricted flow area is the radial surface area of the volume described by the projection of the nozzle into the shell plus the open area in the holes on the top perforated impingement plate within the radial surface area (shown as B in Figure 3). The holes between the plates must be staggered and the area of the holes on the lower plate must equal the area of the holes on the top plate. A minimum gap distance of perforated hole diameter divided by 4 must separate the plates.

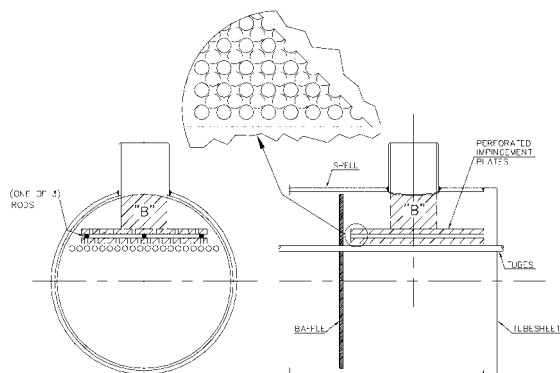


Figure 3

SHELL INLET AREA WITH PERFORATED IMPINGEMENT PLATES

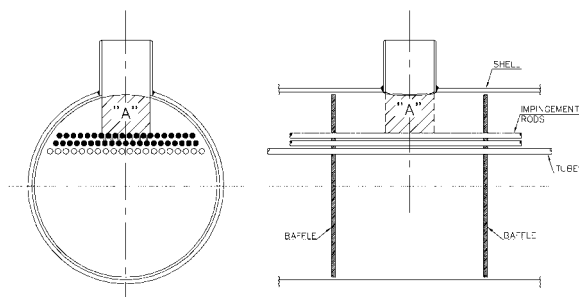


Figure 4
SHELL INLET AREA WITH IMPINGEMENT RODS

3.8 Shell Inlet or Outlet Area without Impingement Plate

When an impingement plate is not used, the centerline of the row of tubes closest to the nozzle shall be located such that the unrestricted flow area described by the projection of the nozzle into the shell is equal to or greater than the area calculated using the allowable value of G^2/ρ from Table 3. This unrestricted flow area can include the flow area between the tubes described by the projection of the nozzle on the outermost row of tubes in addition to the radial surface area under the nozzle (shown as B in Figure 5)

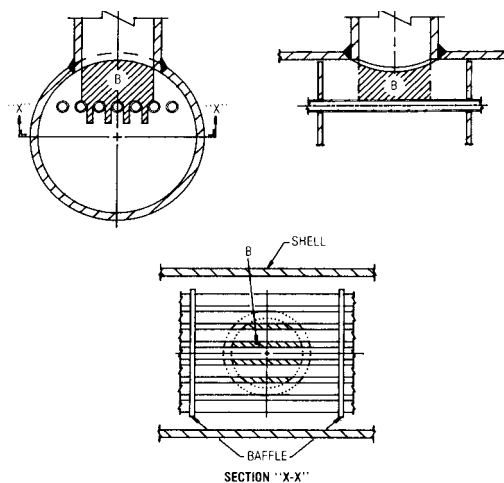


Figure 5
SHELL INLET OR OUTLET AREA WITHOUT IMPINGEMENT PLATE

3.9 Bundle Entrance and Exit Areas

The bundle entrance or exit area is the section of the tube bundle between the adjacent baffles or the adjacent baffle and tubesheet at the shell inlet or outlet nozzle. The unrestricted flow area for the fluid entering or exiting the tube bundle shall be equal to or greater than the area calculated using the allowable value of G^2/ρ from Table 3. The unrestricted flow area is the cross-sectional area between the first full row of tubes closest to the nozzle. This area cannot include the cross-sectional area between the portion of tubes encompassed by an impingement plate, unless the impingement plate is located at least two tube diameters away from the outermost row of tubes.

The unrestricted flow area is shown as **C** or **D** in Figure 6, depending upon the use and placement of an impingement plate.

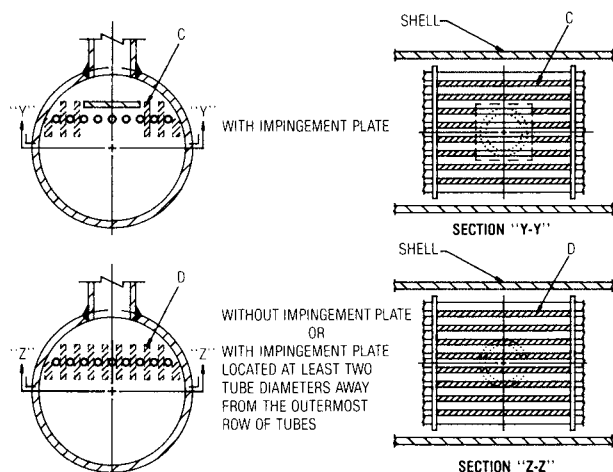


Figure 6
BUNDLE ENTRANCE AND EXIT AREAS

3.10 Vent and Drain Connections

All high and low points on the shell and tube sides of an exchanger, not otherwise vented or drained by nozzles, shall be provided with connections, as required.

3.11 Heat Exchanger Operating Modes

When heat exchangers are designed for series or parallel operation or when pumps operate in parallel, there exists the potential for operating a heat exchanger in excess of its design point. The flow rates may increase to a point which will cause malfunction or damage to the operating unit. Listed below are three situations which can result in an overload or an abnormal operating mode as a result of flow conditions:

- (1) Removing a heat exchanger from service that is designed for parallel flow operation without throttling flow to the heat exchanger remaining in service.
- (2) Removing a heat exchanger from service that is designed for series flow operation without adjusting the flow rates to the heat exchanger remaining in service.
- (3) Operating a heat exchanger with increased pumping capacity; for example, with three half-capacity cooling water pumps operating in parallel.

When such operation is anticipated, it should be referenced in the Design Specification in order that the effect can be considered and the internals properly designed (see 3.1.1(3)).

If the design limits are exceeded, accelerated erosion and failure may occur. There are no definitive guidelines presently available that can adequately determine the relationship of erosion to length of time at overload or abnormal operating conditions.

4.0 MATERIALS OF CONSTRUCTION

4.1 General

The materials used for pressure parts and for external supports, where applicable, shall be in accordance with the Code, as required by the Design Specification.

The Purchaser is responsible for specifying materials suitable to withstand the radiation levels specified in the Design Specification. The Purchaser is responsible for specifying material suitable for the fluid chemistry, pressure, and temperature to avoid erosion/corrosion, stress corrosion cracking, galvanic action, etc.

Some materials which are permitted for use in Section VIII heat exchangers may not be permitted by the Code for use in the construction of Section III exchangers. Also, materials which may be permit-

ted in Section III, Class 3 exchangers may not be permitted in Section III, Class 1 or Class 2 exchangers. Furthermore, the required tests and inspections differ depending on the applicable section of the Code. For example, certain sections of the Code may require impact or ultrasonic testing of the materials being used.

It would be impractical to list all the materials that may be used in Code constructed units; however, some of the more commonly used materials and the parts for which they are used are given in Table 4. It should be noted that the specification number indicated may not be acceptable for use in all classes of Section III heat exchangers (refer to ASME Section II, Part D).

Table 4
Materials of Construction

	ASME Specification Number	Primary Material	Baffles	Bolts and Studs	Channels	Covers	Flanges	Nozzles	Nuts	Shells	Spacers	Supports	Tie Rods	Tubes	Tubesheets
Bar	Commercial SA-36 SA-479	CS SS											X X X		
Bolting	SA-193 SA-194 SA-320*	LA LA LA		X X					X						
Forging	SA-105 SA-182 SA-266 SA-350*	CS SS CS CS			X X X X	X X X X	X X X X	X X X X							X X X X
Pipe	Commercial SA-53 SA-106 SA-312 SA-333*	CS CS SS CS			X X X X			X X X X		X X X X	X X X X	X X X X			
Plate	Commercial SA-36 SA-204 SA-240 SA-285 SA-387 SA-515 SA-516* SB-127 SB-169 SB-171	CS LA SS CS LA CS CS NI CU CU	X X X X X X X X		X X X X X X X X	X X X X X X X X	X X X X X X X X	X X X X X X X X			X X X X X X X X	X X X X X X X X			X X X X X X X X
Tube	Commercial SA-179 SA-210 SA-213 SA-214 SA-249 SA-268 SA-334* SA-556 SA-557 SA-688 SB-111 SB-163 SB-338 SB-395	CS CS SS CS SS SS CS CS CS SS CU NI TI CU									X X X X X X X X X X X X X			X X X X X X X X X X X X X	

*These specifications are suggested when impact testing is required.

Legend: CS = Carbon Steel
SS = Stainless Steel

LA = Low Alloy Steel
CU = Copper and Copper Alloys

NI = Nickel and High Nickel Alloys
TI = Titanium and Titanium Alloys

4.2 Gaskets and Packing

The choice of a suitable gasket material depends upon the conditions of service and, unless otherwise specified by the Purchaser, will be in accordance with the standards of the heat exchanger Manufacturer. The Design Specification should stipulate special conditions such as thermal shocks, corrosive fluids, pulsating pressures, etc., since these factors influence the gasket design and material selection.

Some of the more commonly used gasket materials are copper, nickel-copper, stainless steel, carbon steel, and rubber. The gasket type may be flat solid material, metal jacketed, spiral wound or O-rings. Graphite or acrylic fibers may be used for flat solid materials and as fillers for metal jacketed gaskets. For spiral wound gaskets, graphite, ceramic, or chlorite mineral fillers may be used.

Braided or solid packing with a variety of binders may also be used to perform the gasket function. Graphite or organic polymer fibers may be used for packing.

4.3 Floating Head Split Backing Rings and Bolting

Floating head split backing rings and associated bolting shall be considered pressure parts and shall have corrosion resistances similar to the material of the shell.

4.4 Gaskets for Internal Floating Heads

Flat solid material, metal jacketed, or solid metal gaskets may be used for internal floating heads. For tube or shell side design pressures greater than 200 psig or for design temperatures greater than 300°F, flat solid material gaskets of the compressed fiber type should not be used.

4.5 Halogenated Compounds

Halogenated compounds are generally not acceptable for use with austenitic stainless steel due to the possibility of stress corrosion cracking.

4.6 Stainless Steel

Austenitic stainless steel pressure parts used in nuclear power plant exchangers shall meet an acceptable criteria for preventing susceptibility to intergranular corrosion attack.

4.7 Nonmetallic Coatings and Liners

As an alternative to using materials of construction that inhibit corrosion or erosion, consideration may be given to using coatings and liners when handling very active fluids.

Coatings such as epoxy, ceramic, coal tar, neoprene, and paint can be used to protect pressure parts.

The maintenance and initial fabrication and assembly must be done with care to avoid damage to the coatings since small defects may create severe failures of the coatings or liners, thus exposing the underlying metal to corrosive attack.

5.0 MECHANICAL DESIGN STANDARDS

5.1 Code Requirements

The shell and tube sides of the heat exchanger are considered separate pressure vessels and may be constructed to separate Code Sections, Divisions and Classes, as specified by the Purchaser. The applicable Code edition and addenda shall be determined in accordance with current Code rules. Code Cases may also be used.

5.2 Design Pressures

The Purchaser shall specify separate design pressures for the shell and tube sides. This shall include any vacuum or external pressure conditions which may be applicable.

5.3 Design Temperatures

The Purchaser shall specify separate design temperatures for the shell and tube sides. Particular attention should be given to both minimum and maximum design temperatures. The most severe design temperature, whether shell or tube side, should be used to design parts (tubesheets, tubes, floating heads) which come into contact with both fluids, unless a less severe temperature can be justified.

5.4 Hydrostatic Tests

The shell and tube sides are to be hydrostatically tested in accordance with the Code. The test should be so conducted as to facilitate visual inspection for tube joint leakage from at least one side, preferably the tube side. The temperature of the hydrostatic

test medium should be high enough to preclude the possibility of damage due to brittle fracture.

5.5 Corrosion Allowances

Corrosion allowances shall be specified by the Purchaser and should apply to all surfaces of the pressure retaining parts which contact the corrosive fluid(s), except as follows:

- (1) Flange faces
- (2) Floating head backing rings
- (3) Internal bolting
- (4) Tubes, unless otherwise specified by the Design Specification

On parts which are grooved for pass partitions, the depth of the groove may be considered available for corrosion allowance.

5.6 Tubes

The useful life of a tube is normally affected by the conditions of service, such as fluid chemistry, operating temperatures, and fluid velocities, as well as the effects of short and long term shutdowns. These factors should be taken into consideration by the Purchaser when making a tube material selection.

5.6.1 Tube Diameters

The minimum outside tube diameter should be $\frac{3}{8}$ inch nominal. These Standards cover outside tube diameters up to 2 inches nominal; however, larger diameters may be used.

5.6.2 Tube Wall Thickness

Average wall or minimum wall tubes are equally acceptable providing that, in the case of average wall tubes, the calculated thickness for pressure takes into consideration the tolerance in wall thickness. The minimum recommended tube wall thicknesses are shown in Table 5.

Table 5
Minimum Recommended Tube Wall Thicknesses

Tube Material	Wall Thickness
Austenitic Stainless Steel (Straight Tubes)	22 BWG Avg. Wall
Austenitic Stainless Steel (U-tubes)	20 BWG Avg. Wall
Nickel Alloy	18 BWG Avg. Wall
Copper and Copper Alloy	18 BWG Avg. Wall
Titanium (Straight Tubes)	22 BWG Avg. Wall
Carbon Steel	0.050 in Avg. Wall

5.6.3 Tube Pitch and Layout

The tubes may be laid out in any one of the following patterns:

- (1) Triangular
- (2) Rotated triangular
- (3) Square
- (4) Rotated square

Tubes should have a nominal center-to-center spacing which is no less than that shown in Table 6. When square or rotated square pitch is necessary to provide a cleaning lane, the nominal distance between tubes should be no less than $\frac{1}{4}$ inch.

Table 6
Minimum Recommended Tube Pitches

Nominal Tube Outside Diameter in	Nominal Tube Pitch in
$\frac{3}{8}$	$\frac{1}{2}$ ⁽¹⁾
$\frac{1}{2}$	$\frac{5}{8}$ ⁽¹⁾
$\frac{5}{8}$	$\frac{3}{4}$
$\frac{3}{4}$	$\frac{15}{16}$
$\frac{7}{8}$	$1\frac{3}{16}$
1	$1\frac{1}{2}$
$1\frac{1}{4}$	$1\frac{9}{16}$
$1\frac{1}{2}$	$1\frac{7}{8}$
2	$2\frac{1}{2}$

⁽¹⁾ These pitches should be increased when the tube holes are grooved.

5.6.4 Extending Tube Length

Circumferential welding of tubes to extend their lengths is not recommended.

5.6.5 U-Tubes

The following formula should be used to determine the minimum required thickness of the tube wall before bending:

$$t = t_s \left(1 + \frac{d_o}{4R} \right)$$

where

t = Minimum required tube wall thickness before bending, in

t_s = The greater of the following, in:

- (1) The minimum required tube wall thickness of a straight tube calculated for internal pressure:

$$t_s = \frac{Pd_o}{2S + 0.8P}, \text{ or}$$

- (2) The minimum required tube wall thickness of a straight tube calculated for external pressure in accordance with the Code

d_o = Outside diameter of tube, in

P = Design pressure, psig

R = Radius of bend at centerline of tube, in

S = Allowable design stress, psi

All U-tubes shall be pressure tested after bending, but prior to assembly.

5.6.6 Heat Treatment of U-Bends

Cold work in forming U-bends may increase susceptibility to stress corrosion cracking in certain materials and environments. The Purchaser shall specify if heat treatment of the U-bends is required.

5.6.7 Tube Joint Temperature

When tubes are to be installed in carbon steel tubesheets, Table 7 should be used to determine whether the maximum operating metal temperature for the specified tube material is in a range low enough to permit a tube joint which is expanded only. Welded tube joints should be used when temperatures range from the values in Table 7 to the maximum metal temperatures permitted by the Code.

Table 7
Maximum Recommended Metal
Temperature of Expanded Tube Joints
in Carbon Steel Tubesheets

Tube Material	Code P Number	Alloy Designation	Joint Temperature °F
Austenitic Stainless Steel	8	ALL	500
Nickel-Copper 70-30	42	400	550
Nickel-Chromium-Iron	43	600	500
Admiralty Types B,C,D	32	443,444,445	350
Aluminum-Brass	—	687	350
Copper-Nickel 90-10	34	706	400
Copper-Nickel 80-20	34	710	450
Copper-Nickel 70-30	34	715	500
Titanium	51,52	ALL	400
Carbon Steel	1	ALL	650

5.6.8 Shop Tube Plugging

Occasionally, defective tubes may not be discovered until the final test. In cases where the defective tube(s) cannot be feasibly replaced, the Manufacturer shall plug the tube(s) in accordance with an appropriate tube plugging procedure. Documentation specifying the number and location of the plugged tube(s) is required, and the Purchaser shall be appropriately informed. The Manufacturer remains responsible for the performance of the heat exchanger.

5.6.9 Low Fin Tubing

In certain circumstances, integral low fin tubing may provide a more economical or compact design. The enhanced tube surface on the OD of the tube can become advantageous when either the shell side heat transfer film coefficient is controlling, or when a shell side phase change is occurring. Consideration to fouling should be considered.

5.7 Tubesheets

5.7.1 Thickness

More factors influence the design of a tubesheet than most other heat exchanger parts. These factors include the following:

- (1) tube diameter, thickness, pitch, and length
- (2) tube layout
- (3) number of tubes
- (4) outer tube limit radius
- (5) shell and channel cylinder thicknesses
- (6) method of attachment of tubesheet to shell, channel, and pass partition plates
- (7) shell and tube side design pressures and metal temperatures
- (8) shell and tube side material properties

To consider the effect of the above variables, it is necessary to employ either a finite element analysis or an analysis using classical axisymmetric shell-and-plate theory. The tubesheet should be modeled as a perforated plate with an unperforated rim. The

perforated plate analysis is normally performed using effective elastic material properties.

An analytical solution requires that interaction analyses be performed between the perforated region of the tubesheet and the unperforated rim, and between the unperforated rim and any ring element outboard of the unperforated rim. Modeling of the behavior of such an outboard ring requires that consideration be given to its interaction with the shell and channel (gasketed or integral).

The analysis should consider the following:

- (1) tube side and shell side design pressures
- (2) tube side pressure in the tube holes
- (3) flange bolt loads
- (4) gasket loads
- (5) differential thermal expansion between shell and tubes in fixed tubesheet units

The analysis should yield tubesheet radial, tangential, and shear stresses, channel and shell stresses, and tube stresses which are to be within the applicable Code allowable stresses.

5.7.2 Tube Hole Diameters and Tolerances

Tube holes in tubesheets should be finished to the sizes and tolerances shown in Table 8. Ninety-six percent (96%) of the tube holes must not exceed the value for standard over-tolerance and the remainder must not exceed the value for maximum over-tolerance. The tube holes shall be smooth, and burrs shall be removed to prevent damage to the tubes.

5.7.3 Ligament Widths and Tolerances

Table 9 tabulates the widths for nominal ligament, minimum standard ligament, and minimum permissible ligament for the tube pitches shown. Ninety-six percent (96%) of the ligaments must be at least equal to the value for minimum standard ligament width and the remainder must be at least equal to the value for minimum permissible ligament width. The heaviest recommended tube gages for the pitches shown are also given in Table 9.

5.7.4 Pass Partition Grooves

Pass partition gasket seating surfaces in tubesheets should have pass partition grooves whose depth is greater than or equal to the gasket thickness.

5.7.5 Clad Tubesheets

When required by the Design Specification or deemed necessary by the Manufacturer, clad tubesheets shall be used. In calculating the minimum required thickness of clad tubesheets, credit shall not be taken for the clad material except where permitted by the Code. Nonintegral tubesheet facings are not permitted, and only cladding which is integral or deposited by welding should be used. When cladding is required on either face of the tubesheet, the nominal thickness of the clad material should be no less than $\frac{3}{16}$ inch. The thickness of the clad material under gasket seating surfaces should be $\frac{1}{8}$ inch minimum.

Grooved tube joints or welded tube joints, or both, should be considered when corrosive fluids are in contact with the cladding.

Table 8
Tube Hole Diameters and Tolerances for Tubesheets

Nominal Tube O.D. (in)	Nominal Tube Hole Diameter and Under-Tolerance, in				Over-Tolerance, in	
	Standard Fit		Close Fit		Standard	Maximum
	Nominal Diameter	Under- Tolerance	Nominal Diameter	Under- Tolerance		
$\frac{3}{8}$	0.384	0.004	0.382	0.002	0.002	0.007
$\frac{1}{2}$	0.510	0.004	0.508	0.002	0.002	0.008
$\frac{5}{8}$	0.635	0.004	0.633	0.002	0.002	0.010
$\frac{3}{4}$	0.760	0.004	0.758	0.002	0.002	0.010
$\frac{7}{8}$	0.885	0.004	0.883	0.002	0.002	0.010
1	1.012	0.004	1.010	0.002	0.002	0.010
$1\frac{1}{4}$	1.264	0.006	1.261	0.003	0.003	0.010
$1\frac{1}{2}$	1.518	0.007	1.514	0.003	0.003	0.010
2	2.022	0.007	2.018	0.003	0.003	0.010

Table 9
Tubesheet Drilling Tolerances and Maximum Recommended Tube Gages

Nominal Tube O.D. in.	Nominal Tube Pitch in.	Heaviest ⁽¹⁾ Recommended Tube Gage B.W.G.	Nominal Ligament Width in.	Minimum Standard Ligament Width, in. ⁽²⁾								Minimum Permissible Ligament in.
				Tubesheet Thickness, in.								
				1	1½	2	2½	3	4	5	6	
⅜	½ 17/32	18	0.116	0.087	0.083	0.079	0.075	0.070	0.062	—	—	0.060
		18	0.147	0.118	0.114	0.110	0.106	0.101	0.093	0.084	0.076	0.075
½	⅝ 21/32 11/16	18	0.115	0.089	0.085	0.082	0.079	0.076	0.069	0.063	—	0.060
		16	0.146	0.120	0.116	0.113	0.110	0.107	0.100	0.094	0.088	0.075
		16	0.178	0.152	0.148	0.145	0.142	0.139	0.132	0.126	0.120	0.090
⅝	25/32 13/16 7/8	15	0.146	0.111	0.108	0.106	0.103	0.101	0.096	0.090	0.085	0.075
		14	0.178	0.143	0.140	0.138	0.135	0.133	0.128	0.122	0.117	0.090
		14	0.240	0.205	0.202	0.200	0.197	0.195	0.190	0.184	0.179	0.120
¾	15/16 1 1½ 11/8	13	0.178	0.144	0.142	0.139	0.137	0.135	0.131	0.127	0.122	0.090
		12	0.240	0.206	0.204	0.201	0.199	0.197	0.193	0.189	0.184	0.120
		12	0.302	0.268	0.266	0.263	0.261	0.259	0.255	0.251	0.246	0.150
		12	0.365	0.331	0.329	0.326	0.324	0.322	0.318	0.314	0.309	0.185
7/8	13/16 11/8	12	0.209	0.175	0.174	0.172	0.170	0.168	0.164	0.161	0.157	0.105
		12	0.240	0.206	0.205	0.203	0.201	0.199	0.195	0.192	0.188	0.120
1	1½ 15/16 13/8	10	0.238	0.205	0.203	0.202	0.200	0.198	0.195	0.192	0.189	0.120
		9	0.300	0.267	0.265	0.264	0.262	0.260	0.257	0.254	0.251	0.150
		9	0.363	0.330	0.328	0.327	0.325	0.323	0.320	0.317	0.314	0.185
1¼	19/16	9	0.298	0.265	0.264	0.263	0.262	0.260	0.258	0.255	0.253	0.150
1½	17/8	8	0.357	0.325	0.324	0.323	0.322	0.321	0.318	0.316	0.314	0.180
2	2½	6	0.478	—	0.446	0.445	0.444	0.443	0.442	0.440	0.438	0.250

⁽¹⁾ Heavier gages may be used for full strength welded and partial strength welded tube-to-tubesheet joints.

⁽²⁾ The above Table of Minimum Standard Ligament Width is based on a ligament tolerance not exceeding the sum of twice the drill drift tolerance plus 0.020" for tubes less than $\frac{5}{8}$ " O.D. and 0.030% for tubes $\frac{5}{8}$ " O.D. and larger. Drill drift tolerance = 0.0016 x (thickness of tubesheet in tube diameters), in.

5.7.6 Removable Tube Bundles

In exchangers where the tube bundle is removable and the tubesheet is bolted between two flanges (three element bolted joint), it is recommended that provisions be made such that the shell side or tube side joint can be independently disassembled from the tubesheet without breaking the other joint.

5.7.7 Tube Joints

5.7.7.1 Expanded

Tubes should be expanded into the tubesheet for a distance of at least 2 inches or within $\frac{1}{8}$ inch of the shell side face, whichever is less. Caution should be taken to ensure that the expansion does not extend beyond the shell side face of the tubesheet. The tubes may protrude beyond the tubesheet surface by no more than $\frac{1}{4}$ inch or be recessed by no more than $\frac{1}{16}$ inch. Tubes should not extend above the top tubesheet in vertical exchangers.

5.7.7.2 Grooved

When required, each tube hole should be grooved (rectangular or curved) with two $\frac{1}{8}$ inch wide by $\frac{1}{64}$ inch deep grooves.

5.7.7.3 Welded

Welded tube-to-tubesheet joints should be used where additional tube joint sealing or strength is required.

Tube-to-tubesheet welds are classified as full strength welds, partial strength welds, and seal

welds. These welds shall be defined and sized in accordance with UW-20 of ASME Section VIII, Division 1.

5.7.7.4 Welded and Expanded

A welded and expanded tube joint is typically constructed and tested as follows:

- (1) Groove tube holes, if required
- (2) Clean tubes, tube holes, and face of tubesheet
- (3) Insert tubes and fit up as required
- (4) Weld tubes to tubesheet
- (5) Perform leak test
- (6) Expand tubes
- (7) Liquid penetrant examine tube-to-tubesheet welds

5.8 Tube Bundles

All baffles and support plates in the tube bundle should be securely held in place by tie rods and spacers or an equivalent construction.

5.8.1 Cross Baffles and Support Plates

Cross baffles and support plates (baffles) should be designed with consideration given to both thermal and mechanical requirements. This relationship is especially important in limiting tube vibration.

Support plates are primarily designed for supporting tubes. Cross baffles provide flow direction for heat transfer in addition to supporting tubes. See Figure 7 for a representation of typical baffles and support plates. Other types may be considered.

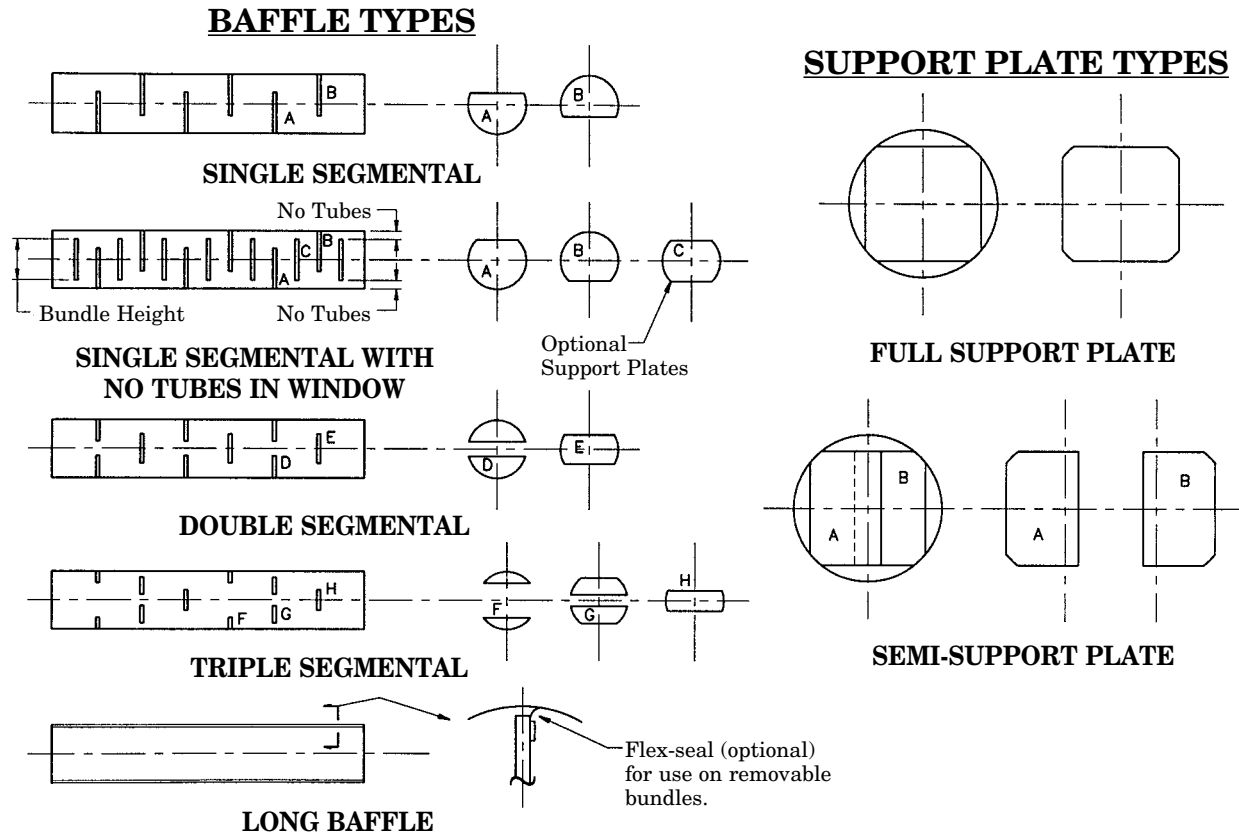


Figure 7
TYPICAL BAFFLES AND SUPPORT PLATES

5.8.1.1 Tube Hole Diameters

For unsupported tube lengths up to and including 30 inches, tube hole diameters should be the nominal tube OD plus $\frac{1}{32}$ inch. When the unsupported tube length is greater than 30 inches and the nominal tube OD is less than or equal to $1\frac{1}{4}$ inches, tube hole diameters should be the nominal tube OD plus $\frac{1}{64}$ inch. All tube holes shall have a maximum over-tolerance of 0.010 inch. The tube holes shall be smooth, and burrs shall be removed to prevent damage to the tubes.

5.8.1.2 Baffle Thickness

The nominal thickness of baffles should be no less than that given in Table 10. When the floating head end of a tube bundle is to be supported by a full or partial support plate, the nominal thickness of the plate should not be less than that given in the column for unsupported tube lengths over 60 inches.

Table 10
Cross Baffle and Support Plate
Thicknesses, in

Nominal Shell Inside Diameter in	Unsupported Tube Length, in				
	Up to 24	Over 24 to 36	Over 36 to 48	Over 48 to 60	Over 60
4-14	$\frac{1}{4}$ ⁽¹⁾	$\frac{5}{16}$ ⁽¹⁾	$\frac{3}{8}$ ⁽¹⁾	$\frac{3}{8}$	$\frac{3}{8}$
>14-27	$\frac{5}{16}$ ⁽¹⁾	$\frac{3}{8}$ ⁽¹⁾	$\frac{3}{8}$	$\frac{1}{2}$	$\frac{1}{2}$
>27-39	$\frac{3}{8}$ ⁽¹⁾	$\frac{3}{8}$ ⁽¹⁾	$\frac{3}{8}$	$\frac{1}{2}$	$\frac{5}{8}$
>39-59	$\frac{3}{8}$ ⁽¹⁾	$\frac{3}{8}$	$\frac{1}{2}$	$\frac{5}{8}$	$\frac{3}{4}$
Over 59	$\frac{3}{8}$	$\frac{1}{2}$	$\frac{5}{8}$	$\frac{3}{4}$	$\frac{3}{4}$

⁽¹⁾ For alloy baffles, deduct $\frac{1}{8}$ inch.

5.8.1.3 Baffle Spacing

The nominal baffle spacing should be no less than $\frac{1}{5}$ the nominal shell diameter or 2 inches, whichever is greater. Baffles should be spaced so that the nominal unsupported tube length will be no greater than that given in Table 11.

Each leg of all U-bends should be supported close to the point of tangency. All U-bent tubes, wherein the diameter of the bend plus the length of each leg from the tangent point to the last baffle exceeds the applicable value in Table 11, should have adequate provision in the U-bend area for the support of such tubes.

Special consideration should be given to the unsupported tube span between the tubesheet and adjacent baffle, between adjacent baffles, and at the U-bends to avoid detrimental tube vibration (see 5.14).

Table 11
Maximum Unsupported Tube Length

Nominal Tube Outside Diameter in	Maximum Unsupported Tube Length in
$\frac{3}{8}$	28
$\frac{1}{2}$	36
$\frac{5}{8}$	48
$\frac{3}{4}$	54
$\frac{7}{8}$	57
1	60
$1\frac{1}{4}$	70
$1\frac{1}{2}$	80
2	100

5.8.1.4 Baffle Cuts

The baffle cut is the ratio of open window area to total area and cuts may be horizontal, vertical or rotated. Baffle cuts shall overlap sufficiently to provide adequate bundle rigidity. The minimum baffle cut for single segmental baffles should be 15%. Double segmental and triple segmental baffles are typically cut at 40% and 60% respectively with permissible variations. Segmental baffles with no tubes in the window area should be limited to 9 ft/sec. liquid velocity through the open window. For steam service, support plates shall be cut to prevent entrapment of non-condensibles.

5.8.1.5 Baffle Diameters

Baffle diameters should be no less than that calculated using the applicable design baffle-to-shell diametral clearance given in Table 12. If baffle diameters are to be rounded off, it should not result in a value less than that determined using Table 12. The diametral clearances given in Table 12 are maximums and may be reduced where increased thermal performance is desired. In any case, the actual design baffle-to-shell diametral clearance should be considered in the thermal design. The baffle edges along the outside diameter shall have a machined finish.

Table 12
Maximum Design Diametral Clearances
Between Shell and Baffle

Nominal Shell Inside Diameter in	Diametral Clearance in
4-9	$\frac{3}{16}$
>9-23	$\frac{1}{4}$
>23-39	$\frac{3}{8}$
>39-59	$\frac{7}{16}$
>59-79	$\frac{1}{2}$
Over 79	$\frac{5}{8}$

5.8.2 Tie Rods and Spacers

5.8.2.1 Tie Rods

Each baffle segment shall be supported by a minimum of three tie rods. The number of tie rods and nominal tie rod diameter should be no less than that given in Table 13. The number of tie rods and tie rod diameters may be varied provided an equivalent metal area is maintained.

Table 13
Minimum Tie Rod Parameters

Nominal Shell Inside Diameter in	Number of Tie Rods	Nominal Tie Rod Diameter in
4-14	4	$\frac{3}{8}$
>14-29	6	$\frac{1}{2}$
>29-43	8	$\frac{1}{2}$
>43-59	10	$\frac{5}{8}$
Over 59	12	$\frac{5}{8}$

5.8.2.2 Spacers

Spacers should be cut in a manner that provides proper baffle alignment. Spacer wall thickness shall be adequate to withstand buckling loads caused by tie rod nut torque and, in the case of vertical heat exchangers, additional dead weight baffle load.

5.8.3 Longitudinal Baffles

The nominal thickness of longitudinal baffles shall be adequate to meet the design conditions, but in no case less than that given in Table 14.

Table 14
Minimum Longitudinal Baffle Thicknesses

Nominal Shell Inside Diameter in	Nominal Thickness in
4-25	$\frac{1}{4}$
>25-39	$\frac{3}{8}$
>39-59	$\frac{1}{2}$
>59-79	$\frac{5}{8}$
Over 79	$\frac{3}{4}$

The preferred method of sealing longitudinal baffles is to weld them to the shell. When longitudinal baffles are not welded, flexible seals may be used to prevent leakage between the shell and the longitudinal baffle. Special design consideration should be given to cases where there is a large differential pressure across the longitudinal baffle, and the use of flexible seals may be undesirable.

5.8.4 Impingement Plates

When an impingement plate is used, the nominal thickness should be no less than $\frac{1}{4}$ inch for alloys and $\frac{3}{8}$ inch for carbon steel (see 3.7).

5.8.5 Sealing Devices

Where by-pass of shell fluid around or through the tube bundle must be limited to provide adequate thermal performance, sealing bars or dummy tubes should be provided. Dummy tubes should be plugged at one end and adequately secured. Tie rods and spacers may also be located so as to function as sealing devices.

5.9 Shells and Shell Covers

5.9.1 Diameters

Although the tables in these Standards reference certain nominal shell diameters, this should not be construed as a restriction on the minimum or maximum shell diameter. In specifying shell diameters, nominal inside diameters are used in these Standards and are generally used throughout the industry.

5.9.2 Thickness

The nominal shell and shell cover thicknesses should be no less than that shown in Table 15.

Table 15
**Minimum Cylinder and Formed Head
Thicknesses**

Nominal Shell Inside Diameter in	Nominal Thickness	
	Carbon Steel in	Alloys in
4-7	$\frac{1}{4}^{(1)}$	0.120
>7-9	$\frac{1}{4}^{(2)}$	0.120
>9-12	$\frac{5}{16}^{(2)}$	$\frac{3}{16}$
>12-19	$\frac{3}{8}$	$\frac{3}{16}$
>19-29	$\frac{3}{8}$	$\frac{1}{4}$
>29-39	$\frac{7}{16}$	$\frac{5}{16}$
>39-59	$\frac{1}{2}$	$\frac{3}{8}$
Over 59	$\frac{5}{8}$	$\frac{1}{2}$

⁽¹⁾ If pipe is used, the nominal wall thickness should be Standard Weight.

⁽²⁾ If pipe is used, the nominal wall thickness should be Schedule 30.

5.9.3 Expansion Joints

The Purchaser shall specify all the operating conditions required by 3.1.1 to enable the Manufacturer to assess the need for an expansion joint. Under some conditions, excessive longitudinal stresses in either the shell or tubes of a heat exchanger having fixed tubesheets and straight tubes can arise. These stresses result from a combination of loads generated by fluid pressure and those due to differential thermal expansion. They are generated by the interaction between the shell, tubes, and tubesheets, all acting as structural members.

A shell expansion joint should be considered if, in its absence, the shell or tube longitudinal stress exceeds the allowable value; the tube joint load is excessive; or the tube buckling load is excessive. The likelihood of any of these conditions is increased when there is an extreme temperature differential or high pressure on either the shell side or tube side.

Shell expansion joints shall be constructed in accordance with the Code and analyzed for each operating condition. The data for each operating condition shall consist of the shell side design conditions, the total design axial movement of the expansion joint (+ for joint extension, – for joint compression), and the required cycle life, as a minimum. To ensure that the expansion joint is designed for its intended function, it is important that the required cycle life be a realistic indication of the cycles that the heat exchanger is expected to experience over its design life.

Bellows expansion joints should be supplied with shipping brackets which shall be removed or disconnected after the heat exchanger is installed. In addition, removable covers shall be supplied on bellows expansion joints which are to be insulated and where it is necessary to protect the bellows element. The Manufacturer shall advise the bellows expansion joint manufacturer of the required weld end material and the acceptable flexible element materials.

When expansion joints are used, the tubes may no longer act as stays for the tubesheets; hence, the tubesheet thicknesses may have to be increased to reduce the stresses to allowable values.

5.10 Channels, Bonnets, and Floating Heads

5.10.1 Thickness

The nominal channel, bonnet, and floating head thicknesses should be no less than that shown in Table 15.

5.10.2 Interpass Flow Velocity

The length of channels, bonnets, and floating heads should be designed such that the nominal interpass flow velocity will be no greater than 0.7 times the mean velocity in one tube pass.

5.10.3 Pass Partition Plates

The thickness of pass partition plates should be calculated in accordance with the equation below, but in no case should the nominal thickness be less than that given in Table 16.

$$t = W \sqrt{\frac{KP}{S_y}}$$

where

- S_y = Yield stress at design temperature, psi
- t = Pass partition plate thickness (excluding corrosion allowance), in
- P = Calculated differential pressure across the pass partition plate at the maximum overload flow specified, psi
- K = Shape factor from Figure 8
- W = Width of pass partition (short edge), in
- L = Length of pass partition (long edge), in

The edge of the pass partition which contacts the gasket may be tapered to the minimum pass partition thickness for alloys as shown in Table 16.

Table 16
Minimum Pass Partition Thicknesses

Nominal Channel Inside Diameter in	Nominal Thickness	
	Carbon Steel in	Alloys in
4-12	1/4	1/4
>12-25	3/8	1/4
>25-39	1/2	3/8
>39-59	5/8	1/2
Over 59	3/4	5/8

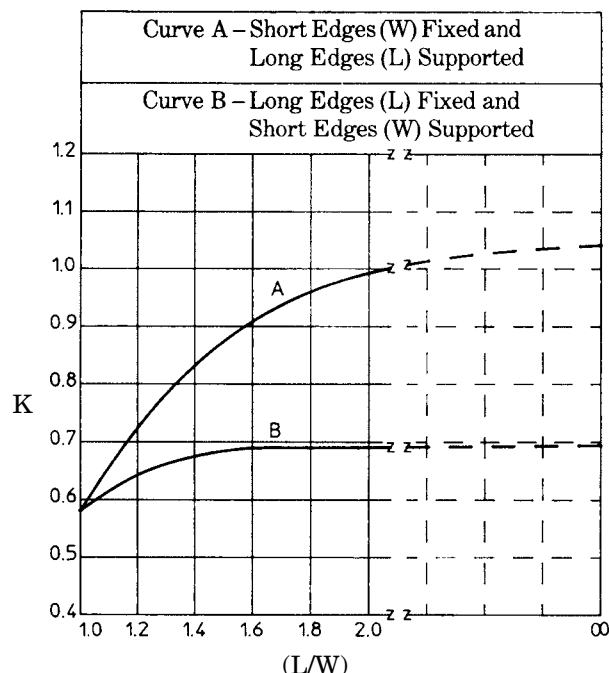


Figure 8
PASS PARTITION SHAPE FACTOR

5.11 Bolted Covers

Bolted covers should be designed in accordance with the applicable Code rules; however, in some cases, it may be desirable to use covers thicker than required by Code rules to minimize deflection and the resulting leakage across pass partitions.

5.11.1 Cover Thickness

The following equation may be used to estimate the thickness required to limit the deflection at the center of a plate subjected to a pressure load and a uniform moment applied at the gasket diameter. The deflection to be used in the equation should be selected by the designer based on the location of the partitions, the thickness and resilience of the gasket material, the pressure differential across the partitions, and the consequences of interpass leakage.

Table 17
Packed Joint Parameters

Nominal Shell Inside Diameter in	Nominal Packing Ring Size in	Packed Tubesheet ⁽¹⁾			Packed Tubesheet with Lantern Ring		
		Minimum Number of Rings	Maximum Pressure psig	Maximum Temp. °F	Minimum Number of Rings Each Side	Maximum Pressure psig	Maximum Temp. °F
4-19	3/8	2	300	600	1	300	400
>19-39	1/2	2	300	600	1	150	400
>39-59	5/8	2	300	600	1	75	400
Over 59	Packed Joints Not Recommended In These Sizes						

⁽¹⁾ Can be applied to a packed nozzle with diameters equivalent to the respective shell diameters.

$$T = \left[\frac{3G^2(1-\nu)}{2E\delta} \left(\frac{PG^2}{128} (5+\nu) + \frac{Wh}{\pi G} \right) \right]^{1/3}$$

where

- E = Modulus of elasticity of cover material, psi (see Appendix J)
- G = Diameter at location of gasket load reaction, in
- h = Distance (radial) from bolt centerline to gasket load reaction location, in
- P = Design pressure, psig
- T = Thickness, in
- W = Total bolt load, lbf
- δ = Deflection at center of cover, in
- ν = Poisson's ratio for cover material

In cases where there are no partitions and deflection is not a consideration, only Code requirements need to be considered.

5.11.2 Pass Partition Grooves

Pass partition gasket seating surfaces in bolted covers should have pass partition grooves whose depth is greater than or equal to the gasket thickness.

5.12 Packed Joints

5.12.1 Service Restrictions

Packed joints shall not be used in exchangers containing radioactive, lethal, or flammable fluids.

Exchangers designed using a packed floating tubesheet with a lantern ring shall only be used for water, steam, air, or lubricating oil services.

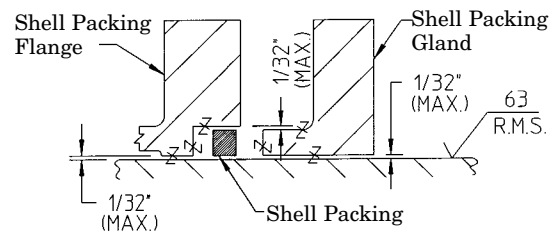
5.12.2 Design Restrictions

Table 17 should be used as a guide for designing packed joints. The parameters in Table 17 may be modified when the number of packing rings is increased.

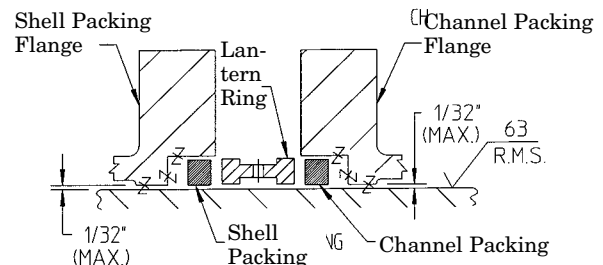
5.12.3 Construction Requirements

The recommended clearances and surface finishes for packed joints designed per Table 17 are shown in Figure 9.

Floating tubesheet skirts should extend toward the tube side or be designed to prevent the formation of stagnant areas on the shell side.



PACKED TUBESHEET



PACKED TUBESHEET WITH LANTERN RING

- Z - Indicate machined surface, but not a particular surface finish.
- $\sqrt{\quad}$ - Indicate particular machined surface finish.

Figure 9
PACKED JOINT CONSTRUCTION REQUIREMENTS

5.13 Nozzles and Supports

5.13.1 Nozzles

Nozzle projections shall be in accordance with the Manufacturer's normal practice, unless otherwise specified by the Purchaser. The bolt holes of flanged nozzles should straddle the planes of the exchanger centerlines.

5.13.2 Supports

5.13.2.1 Design

Each exchanger should have supports designed to support the heat exchanger in the specified position and to resist all other specified external loads. The supports should be designed such that the exchanger is restrained from movement in all lateral directions; however, only one support should restrain movement in the longitudinal direction, while the remaining supports permit longitudinal movement. Supports will not be designed to lift the exchanger, unless otherwise specified by the Purchaser.

5.13.2.2 Welded Supports

When a support is welded directly to a pressure boundary part, the support material shall be the same type as the part to which it is being welded or made compatible by suitable overlay. When a support is attached to a pad which is welded directly to a pressure boundary part, the pad material shall be the same type as the part to which it is being welded.

5.13.3 Nozzle Load and Support Analysis

All nozzle and support loads, which are to be taken into consideration in the design of the exchanger, shall be included as part of the Design Specification. The Purchaser shall identify the nozzle and support loads and their combinations with the appropriate Service Limits as defined by Section III, Division 1 of the Code, or the appropriate Load Combination as defined by Section VIII, Division 2 of the Code. When nozzle and support loads are specified for Section VIII, Division 1 exchangers, the Manufacturer may use the yield strength as the stress limit for the various combinations and magnitudes of loadings.

5.13.3.1 Nozzle Loads

When the Purchaser requires a nozzle load analysis, it shall be his responsibility to specify the magnitude and direction of the forces and moments which act at the nozzle-to-shell juncture. These are shown in Figure 10. In addition to determining the stresses at the nozzle-to-shell juncture, the Manufacturer should consider the effect of the nozzle loads on the exchanger's gasketed joints and bellows-type expansion joints.

The Purchaser may need to know the allowable forces and moments at the nozzle in order to determine the piping configuration and generate the actual loads. The determination of allowable nozzle loads is a complex problem involving the interaction of external forces and moments applied at the vessel wall. These loads are functions of the piping mechanical and thermal design.

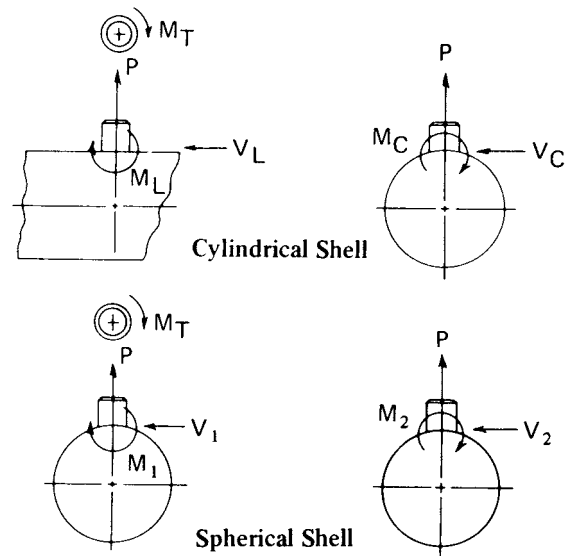


Figure 10
NOZZLE LOAD NOMENCLATURE

Cylindrical Shell

- V_C = Shear force in the circumferential direction
- V_L = Shear force in the longitudinal direction
- P = Axial force
- M_C = Bending moment in the circumferential direction with respect to the shell
- M_L = Bending moment in the longitudinal direction with respect to the shell
- M_T = Torsional moment

Spherical Shell

- V_1, V_2 = Shear force in two orthogonal directions
- P = Axial force
- M_1, M_2 = Bending moment in two orthogonal directions
- M_T = Torsional moment

5.13.3.2 Procedure for Calculating Nozzle External Forces and Moments in Cylindrical Vessels

The procedure given in Appendix C permits estimating nozzle loads for cylindrical shells. The procedure is based in part on the design data included in Welding Research Council Bulletin 107⁽¹⁾. The allowable loads have been linearized to show the interaction between the maximum permitted external radial load and the maximum permitted applied moment vector.

The procedure represents a simplification of the method of WRC 107, and users of the procedure included in these Standards are cautioned that more exact analysis is required to verify the adequacy of the final design. The stresses considered in developing the procedure have been defined as secondary stresses with stress limits established according to that definition.

Although the effect of internal pressure has been included in the combined stresses, the effect of pressure on nozzle thrust has not been included and requires combination with other radial loads.

Loads exceeding those calculated by the method in Appendix C usually require additional reinforcement. The Purchaser is cautioned that the higher allowable loads obtained through design modifications may require the strengthening of other parts, such as flanged joints, supports, supporting structures, and floors. It should be understood by the Purchaser that the exchangers are not intended to serve as anchor points for the piping and that every effort should be made to minimize the reactions to the exchanger nozzles.

5.13.3.3 Support Analysis

The supports, including the anchor bolts, shall be designed to support the exchanger and resist all specified nozzle loadings, seismic forces, and all other specified external loads.

When the Purchaser requires a seismic analysis, the floor response accelerations and the type of analysis (static or dynamic) shall be specified by the Purchaser.

For exchangers where the primary natural frequency is calculated to be above 33 hertz, a static analysis is generally acceptable. When the natural frequency is calculated to be equal to or below 33 hertz, it is recommended that a dynamic analysis be performed. When a dynamic analysis is required, the Purchaser shall supply the Manufacturer with the floor response spectra or acceleration time history.

5.14 Tube Vibration

All heat exchangers should be reviewed for susceptibility to detrimental tube vibration and designed to ensure the adequacy of the tube bundle. When deemed necessary by the designer or required by the Design Specification, a vibration analysis

shall be performed. Numerous geometric and hydrodynamic factors play significant roles in initiating flow-induced vibration in tube bundles. A definitive analysis for actual heat exchangers remains intractable; hence, the analytical method given herein is presented as a tentative guideline and shall be used with due engineering judgment.

5.14.1 Areas of Consideration

The design of the entire tube bundle, especially those areas with high local velocities or long unsupported tube spans, should be reviewed for potential detrimental tube vibration. The U-bends and the inlet and outlet areas of the tube bundle, particularly in the vicinity of an impingement plate, should be analyzed. Although the flow at the central portion of the tube bundle may be more evenly distributed, this portion of the tube bundle should also be analyzed.

5.14.2 Tube Support Condition

All tubes should be considered fixed at the tubesheet and simply supported at the baffles. Where a U-bend support has been provided, the tubes should be assumed to be simply supported at that point (see Figure 11).

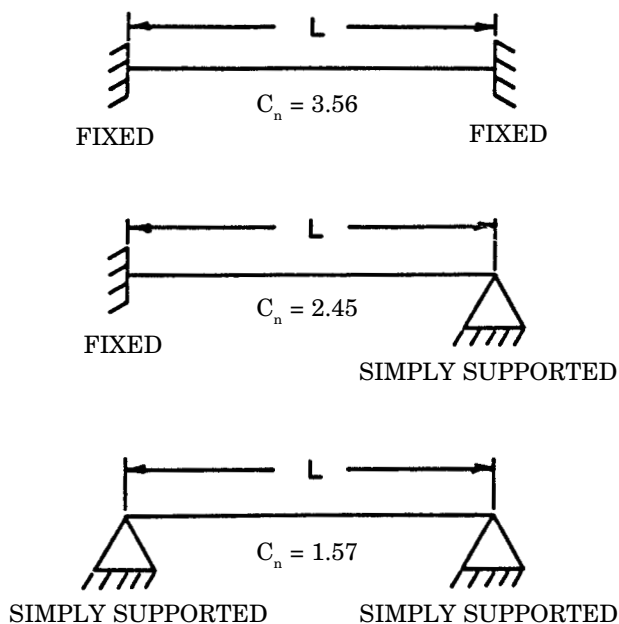


Figure 11
METHODS OF SUPPORT
FOR THE UNSUPPORTED
TUBE SPAN UNDER CONSIDERATION

⁽¹⁾ Local Stresses in Spherical and Cylindrical Shells due to External Loadings, K.R. Wichman, A.G. Hopper and J.L. Mershon-Welding Research Council, Bulletin 107/August, 1965—Revised Printing—March, 1979

5.14.3 Design Criteria

The tube bundle should be designed so that the velocity at the tube span under consideration satisfies the following relationship:

$$V_{act} < V_{crit}$$

where

V_{act} = Maximum cross flow gap velocity at the tube row under consideration, ft/sec

V_{crit} = Critical velocity, ft/sec

5.14.4 Method of Calculation

NOTE: Other valid calculation methods may be used in lieu of this method.

5.14.4.1 Calculate V_{act}

5.14.4.2 Calculate $\left[f_n \sqrt{m} \right]$

$$\left[f_n \sqrt{m} \right] = K \frac{C_n}{L^2} \sqrt{EIg + \frac{gFL^2}{\pi^2}}$$

where

C_n = Constant applicable to the method of support for the unsupported tube span under consideration as follows (see Figure 11):

= 3.56 for both ends fixed

= 2.45 for one end fixed, one end simply supported

= 1.57 for both ends simply supported

d_o = Outside diameter of the tube, in

d_i = Inside diameter of the tube, in (see Appendix H or I)

E = Modulus of elasticity, psi (see Appendix J)

F = Axial force in tubes, tensile (+), compressive (-), lbf

f_n = Calculated natural frequency of the unsupported tube span under consideration using the applicable end fixity conditions (C_n), 1/sec

g = Gravitational constant = 386 lbf-in/lbf-sec²

I = Moment of inertia, in⁴ (see Appendix H or I)
= $\pi(d_o^4 - d_i^4)/64$

K = 1.0 for straight portion of tubes
= 0.866 for U-bend portion of tubes

L = Unsupported tube span under consideration, in (for U-bends, L = the full developed length)

m = Effective weight of the tube per unit length, lbf/in

= $W_e + W_t + W_s$

W_e = lbf/in of tube metal

W_t = lbf/in of fluid inside tube

W_s = lbf/in of shell side fluid displaced by the tube

5.14.4.3 Calculate V_{crit}

$$V_{crit} = \left[f_n \sqrt{m} \right] \left[\beta \sqrt{\frac{12\delta}{\rho}} \right]$$

where

β = Instability Constant from Figure 12

p = Tube pitch, in (see Figure 12)

δ = Logarithmic decrement of damping is a measure of the decay of vibration amplitude with time of a tube vibrating in a still fluid.

ρ = Density of the shell side fluid, lbf/ft³

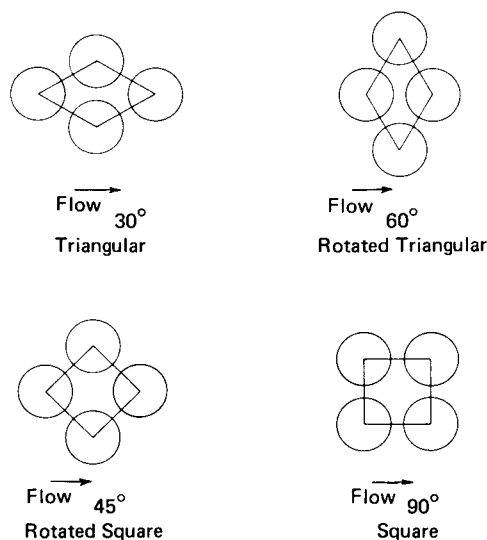
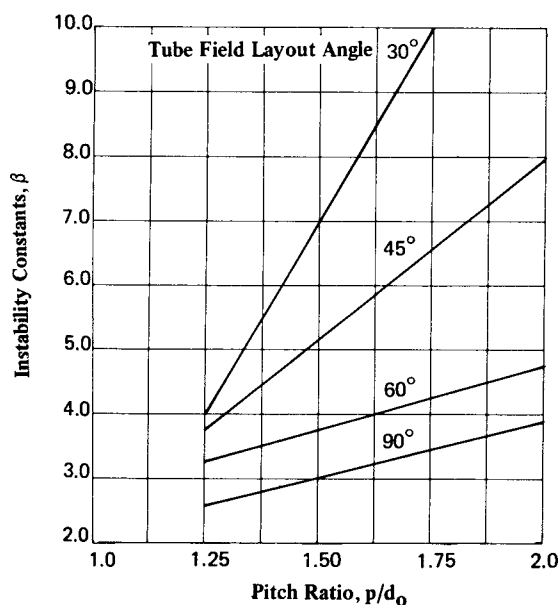


Figure 12
INSTABILITY CONSTANTS FOR CRITICAL VELOCITY

6.0 HEAT EXCHANGER PROTECTION

6.1 Safety Requirements

The Code specifies a variety of measures for the protection of heat exchangers against over-pressure. The Purchaser shall install protective devices in the system to prevent thermal and mechanical transients from exceeding those conditions for which the heat exchanger is designed.

6.2 Relief Valves

6.2.1 Specification

Relief valves are normally beyond the scope of the heat exchanger Manufacturer's responsibility. Pressure and temperature relief requirements are most appropriately specified for the entire piping loop including the heat exchanger. However, should the Purchaser require independent relief valves specifically for the heat exchanger, it shall be so stated in the Design Specification including any special requirements.

6.2.2 Installation Criteria

The following recommendations are offered to assist in the proper location and installation of relief valves:

- (1) Valves should be installed vertically.
- (2) If two or more relief valves are attached to the same nozzle, the flow area of the nozzle should be at least equal to the combined flow area of the relief valve inlet connections.
- (3) The discharge piping connected to the relief valve exit should be equal or larger in diameter than the valve exit opening. There should be no intervening valves or obstructions in this line.
- (4) The discharge piping should be designed to minimize the stress of the valve body and the heat exchanger.

6.2.3 Design Criteria

Conditions that should be considered in sizing the orifice area of the relief valves are given below:

- (1) Thermal Expansion Relief: The relief valve should be designed to relieve the pressure caused by thermal expansion of the entrapped fluid when the isolation valves are closed.
- (2) Tube Rupture: Relief valve flow shall be based on the clean rupture of one exchanger tube resulting in two flow conduits.

The flow velocity through each tube end can be estimated using an iterative procedure as outline below. The required relief valve flow rate is then determined using this velocity.

- (i) Calculate the flow velocity using the following equation:

$$V = \sqrt{\frac{2(144g\Delta p)}{k\rho}}$$

where

V = Rupture flow velocity, ft/sec

g = Gravitational Constant = 32.2 lbm-ft/lbf-sec²

Δp = Net difference in the design pressures

between the shell and tube sides, psi

ρ = Density of the discharging fluid, lbm/ft³

k = Resistance coefficient

In the case where the shell side design pressure is less than the tube side design pressure, the resistance coefficient is constant at 1.2. Use this and proceed to the last step. When the shell side design pressure is greater than the tube side design pressure, use a resistance coefficient of 0.44 as a first guess and proceed.

- (ii) Using the calculated value of V from step (i), determine the following:

$$Re = \frac{124dV\rho}{\mu}$$

$$f = 0.049(Re)^{-0.2}$$

$$k = 0.44 + 1.5(4f) + 0.048 \sqrt{4f}$$

where

Re = Reynolds number associated with the discharging stream

V = Rupture flow velocity, ft/sec

μ = Dynamic viscosity of discharging fluid, cP

f = Friction Factor

d = Nominal inside diameter of tube, in
(see Appendix H or I)

- (iii) Using the above calculated value of k, recalculate the flow velocity in step (i) and repeat the above procedure until the calculated value of k matches the assumed value.

- (iv) Using the calculated flow velocity, calculate the required relief valve flow rate.

$$Q = 4.9Vd^2$$

where

Q = Relief valve flow rate, gal/min

For compressible fluids, the calculated velocity used for valve sizing shall not exceed sonic velocity.

6.3 Cathodic Protection of Carbon Steel Channels

For heat exchangers with carbon steel channels that use sea water or brackish water as the tube side cooling medium, cathodic protection of the carbon steel components exposed to the cooling water should be considered. The saline solution acts as an electrolyte so that galvanic corrosion of carbon steel internals, having a lower electrochemical potential than the nonferrous tubing, occurs. Even though the channel and cover may be protected by a coating, there is a danger that a pinhole or discontinuity in the coating may occur, exposing a small portion of the steel to the cooling water, in which case the rate of corrosion is especially high.

Cathodic protection is most commonly provided by anodes made of zinc, magnesium, or some material having a relatively low electro potential with respect to carbon steel. These anodes act in a sacrificial capacity, thus sparing the carbon steel components from galvanic attack. These anodes are attached to the channel cover, if possible, with the size and quantity depending on the size of the unit.

An alternative, but more costly, form of protection is the application of an impressed current system, where the potential of the carbon steel components is maintained within an acceptable range by means of an electrical current.

6.4 Shop Cleaning

Internal surfaces of the exchanger should be cleaned to remove weld spatter, slag, burrs, loose scale, etc. The shell should be cleaned prior to bundle assembly.

Baffles, tie rods, and spacers should be cleaned of loose dirt or oil, using solvent if necessary. The tubesheet should be cleaned with special attention to the tube holes. The holes may be cleaned by swabbing or blowing aspirated solvent through them, followed by wiping. The solvent may be acetone or alcohol of suitable volatility (to dry by evaporation after wiping). Halogenated solvents shall not be used for cleaning austenitic stainless steel surfaces. The external surface of the tubes, especially the tube segment to be expanded or welded to the tubesheet, should be carefully cleaned.

When specified by the Purchaser, the interior of the tubes shall be cleaned by blowing solvent-soaked felt plugs through the tubes.

6.5 Corrosion Protection

An effort shall be made to remove the moisture from the internals of the exchanger after hydrostatic test. Hot air or other means of moisture removal may be used. Consideration should be given to using a rust inhibitor in the hydrostatic test fluid to reduce corrosion on carbon steel surfaces.

It is recommended that exchangers with carbon steel internals be kept reasonably dry. An acceptable way to maintain dryness is by placing desiccants at suitable locations. A superior alternative is to thoroughly dry the exchanger internals utilizing the vacuum drying technique. The drying technique should be such that freezing of entrapped moisture does not occur. When the desired degree of dryness is achieved, the vacuum should be broken with a dry

gas such as nitrogen, and then the space should be filled and pressurized with the gas to 5 to 10 psig. It is beneficial to provide a system to maintain and check the pressure during shipment and storage. See 6.6 for additional information.

6.6 Protection During Shipment and Storage

6.6.1 Shipment Protection

Externals of the exchanger most susceptible to in-transit damage, such as butt weld nozzle ends, nozzle flange gasket seating surfaces, etc., shall be properly protected. The unit shall be securely mounted on the transportation vehicle and fastened to eliminate shifting during shipment.

6.6.2 Storage Protection

The Purchaser shall carefully adhere to an appropriate maintenance program during storage and installation. It is suggested that exchangers stored outdoors be kept in a fire resistant, weatherproof enclosure. The unit should be mounted on skids such that no part of the exchanger makes contact with the ground and should be kept in a well-drained area. Provision shall be made to allow air circulation around the exterior of the exchanger. All exposed surfaces of the heat exchanger shall be periodically examined and recoated by the Purchaser when necessary.

Exchangers containing carbon steel internals or those subject to long-term outdoor storage should have a properly monitored moisture prevention program.

6.6.3 Inert Gas Blankets

Those exchangers using inert gas blanketing should be checked on receipt of the shipment and monitored periodically thereafter to ensure that proper blanket pressure is maintained.

To avoid personal injury, heat exchangers utilizing inert gas blanketing should be relieved of pressure before, and remain vented during, the removal of nozzle and inspection covers.

6.7 Inservice Inspection

The requirements of inservice inspection as given in Section XI of the Code are responsibilities of the Owner. The Purchaser shall determine whether the details of the design are consistent with the Owner's inservice inspection program.

6.8 External Surface Painting

The external surfaces, except machined surfaces, should be given a coat of shop primer for short-term protection during shipment and storage.

Before applying a shop primer, the surfaces shall be prepared by hand and/or power tool cleaning. The external surfaces are to be free from loose scale and weld splatter, grease and oil, and other foreign material.

All exposed machined surfaces shall be coated with an easily removable rust preventative.

7.0 SITE INSTALLATION, INSPECTION, MAINTENANCE, AND CLEANING

7.1 General

The Manufacturer's instructions, if provided, should be consulted in conjunction with the following subsections.

7.2 Installation

Heat exchangers should be installed with sufficient clearance to allow convenient and proper maintenance of the units without disturbing adjacent equipment. Installation should be made so that it enables the use of cranes or hoists installed in the plant to service the exchangers. Ample space should be provided for the disassembly of removable parts, such as shells and channel covers, bundles, etc., and for the retightening of all bolted joints. Similarly, for exchangers with welded joints, space should be provided to permit disassembly and rewelding of all joints.

Shipping brackets restraining bellows expansion joints shall be removed or disconnected after the heat exchanger is installed.

7.3 Installation and Operation Under Freezing Conditions

The Purchaser shall provide and maintain proper protection to prevent freezing of the equipment before, during, and after installation. Heat exchangers that are not in service and exposed to freezing conditions shall be drained or otherwise protected to prevent damage from freezing. Experience has shown that tubes in a horizontal position may not drain sufficiently by gravity alone to preclude freezing damage.

7.4 Inspection

Heat exchangers shall be inspected periodically for any evidence of corrosion or other abnormal conditions, such as tube leaks, etc., that may affect the performance and the life of the equipment (see 6.7).

7.5 Cleaning

It is suggested that provisions be made so that heat exchangers can be cleaned periodically. The removal of foulants from the tube surfaces is required to maintain the thermal performance of the heat exchanger. The Purchaser shall select a cleaning method (mechanical, chemical, etc.) which is appropriate for the conditions of service and the configuration of the heat exchanger.

7.6 Initial Startup Precautions

The bolts should be retightened shortly after the heat exchanger has been put in service for the first time.

It is important that all bolted joints be tightened uniformly and in a diametrically staggered pattern as illustrated in Figure 13; however, the instructions of the Manufacturer should be followed for special closures and spiral wound gaskets.

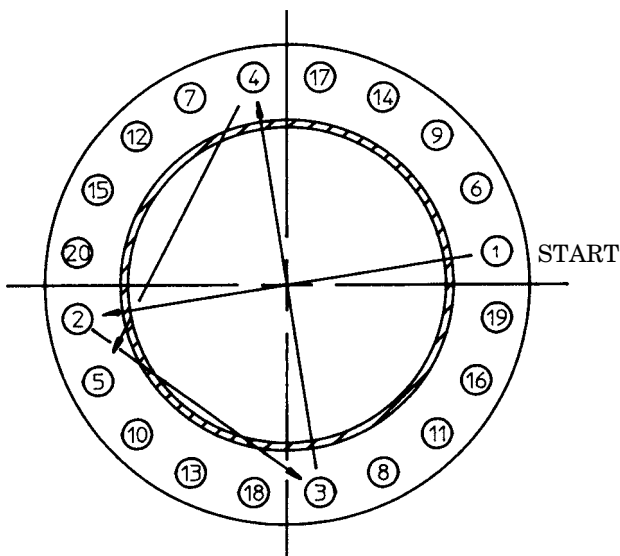
Periodic checks should be made during the first six months of operation to ensure that all bolted joints remain tight. When major bolted connections are insulated, it is recommended that this insulation be removable in order to facilitate periodic retightening as described in Figure 13. Retightening of bolts and periodic verification of bolt torque should only be done when the vessel is not pressurized. Refer to the Manufacturer's instruction manual for additional precautions and specific operating procedures.

7.7 Startup and Shutdown of Fixed Tubesheet Exchangers

Fluids should be introduced in such a manner to minimize differential expansion between the shell and tubes.

7.8 Alterations and Repairs

It is recommended that any alterations or repairs be made in accordance with the Manufacturer's procedures and direction and with the approval of the Authorized Inspection Agency having jurisdiction at the plant, as applicable. Plans should be reviewed with the local Authorized Inspection Agency prior to the work.



Method of Tightening Bolted Joint.

- (1) Tighten all bolts hand tight.
- (2) Tighten bolts, one flat at a time in pattern shown.
- (3) Continue until joint is tight.

Figure 13
BOLT TIGHTENING SEQUENCE

7.9 Spare Parts and Special Tools

The following list of typical spare parts and special tools should be considered by the purchaser of heater exchangers. The specific parts and quantities should be listed in the specifications. In the preparation of the specification the purchaser should consider pre-operational and post-operational spares.

7.9.1 Spare Parts

The recommended spare parts for heat exchangers are listed below:

Spare Parts	Typical Quantity	Comments
Tube Plugs	10% of tube holes	To include special welding supplies if welded plugs are used.
Bolting: Manway Cover, Channel Cover, or Pass Partition Cover (if required)	10% of sets	A set implies a bolt and nut.
Gaskets	2 Sets	This set should include gaskets for pass partition cover (if required). NOTE: Proper storage procedures must be observed since some gasket materials can deteriorate in a short time if improperly stored.
Accessories (when supplied by the heat exchanger manufacturer).	----	As recommended by the accessory manufacturer.
Pass Partition Nuts	1 Set	

7.9.2 Special Tools

The recommended special tools for heater exchangers are listed below:

Special Tools	Typical Quantity	Comments
Tube Expanders	1 set roller expanders for each tube diameter and gage	Drivers optional Spare rolls optional

APPENDIX A

HEAT TRANSFER EQUATIONS

A1.0 Basic Heat Transfer Relation

$$Q = UA_o \text{ [MTD]}$$

where

- Q = Heat exchanger duty, Btu/hr
- U = Overall heat transfer coefficient, Btu/hr-ft²-°F (external surface)
- A_o = Effective external surface, ft²
- MTD = Mean temperature difference, °F

A2.0 Determination of Overall Heat Transfer Coefficient

A2.1 Overall Heat Transfer Coefficient

The overall heat transfer coefficient, U, including fouling, shall be calculated as follows:

$$U = \frac{1}{\left[\left(\frac{1}{h_o} + r_o \right) \left(\frac{1}{E_f} \right) + r_w + r_i \left(\frac{A_o}{A_i} \right) + \frac{1}{h_i} \left(\frac{A_o}{A_i} \right) \right]}$$

where

- U = Overall heat transfer coefficient (fouled), Btu/hr-ft²-°F (external surface)
- h_o = Film coefficient of fluid outside of tubes, Btu/hr-ft²-°F (external surface)
- h_i = Film coefficient of fluid inside of tubes, Btu/hr-ft²-°F (internal surface)
- r_o = Fouling resistance on outside of tubes, hr-ft²-°F/Btu (external surface)
- r_i = Fouling resistance on inside of tubes, hr-ft²-°F/Btu (internal surface)
- r_w = Resistance of tube wall referred to outside surface, including extended surface, if present, hr-ft²-°F/Btu (external surface)
- A_o = Effective external surface, ft²
- A_i = Effective internal surface, ft²
- E_f = Fin efficiency (equals one for bare tubes and less than one for finned tubes)

A2.2 Tube Wall Resistance

A2.2.1 Bare Tubes

$$r_w = \frac{d_o}{24k} \left[\ln \left(\frac{d_o}{d_o - 2t} \right) \right]$$

A2.2.2 Integral Circumferentially Finned Tubes

$$r_w = \frac{t}{12k} \frac{[d_o + 2Nz(d_o + z)]}{(d_o - t)}$$

where

- d_o = O.D. of bare tube or root diameter of fin, in
- z = Fin height, in
- t = Tube wall thickness, in
- N = Number of fins per inch
- k = Tube wall thermal conductivity, Btu-ft/hr-ft²-°F

A3.0 Heat Balance

$$Q = W_h \left[\overbrace{[C_{phv} (T_1 - T_s)]}^{(1)} + \overbrace{\lambda_h}^{(2)} + \overbrace{C_{phl} (T_s - T_2)]}^{(3)} \right]$$

$$Q = W_c \left[\overbrace{[C_{pcv} (t_2 - t_s)]}^{(1)} + \overbrace{\lambda_c}^{(2)} + \overbrace{C_{pcl} (t_s - t_1)]}^{(3)} \right]$$

where

- Q = Total heat exchanger duty, Btu/hr
- T₁, T₂ = Hot fluid inlet, outlet temperature, °F
- t₁, t₂ = Cold fluid inlet, outlet temperature, °F
- T_s, t_s = Hot fluid, cold fluid saturation temperature, ° F
- W_h, W_c = Hot fluid, cold fluid mass flow rate, lbm/hr
- C_{phv}, C_{pcv} = Hot fluid, cold fluid vapor mean heat capacity, Btu/lbm-°F
- C_{phl}, C_{pcl} = Hot fluid, cold fluid liquid mean capacity, Btu/lbm-°F
- λ_h, λ_c = Hot fluid, cold fluid latent heat of vaporization, Btu/lbm

Note: Term (1) applies when fluid is in superheated region.

Term (2) applies when fluid changes state.

Term (3) applies when fluid is in subcooled region.

For partial condensation or vaporization, the appropriate latent heat is multiplied by the fluid fraction which changes state.

APPENDIX B

LMTD CORRECTION FACTORS AND TEMPERATURE EFFICIENCIES

B1.0 Logarithmic Mean Temperature Difference

B1.1 Parallel Flow

$$\text{LMTD} = \frac{[T_1 - t_1] - [T_2 - t_2]}{\ln \left[\frac{T_1 - t_1}{T_2 - t_2} \right]}$$

B1.2 True Counterflow

$$\text{LMTD} = \frac{[T_1 - t_2] - [T_2 - t_1]}{\ln \left[\frac{T_1 - t_2}{T_2 - t_1} \right]}$$

where

LMTD = Logarithmic mean temperature difference, °F

T_1, T_2 = Hot fluid inlet, outlet temperature, °F

t_1, t_2 = Cold fluid inlet, outlet temperature, °F

B2.0 LMTD Correction Factors

$$[\text{MTD}] = [\text{LMTD}] F$$

F, the correction factor to adjust for deviation from true counterflow, is a function of R and P and can be obtained from Figures B-1 thru B-9, as applicable.

MTD = Mean temperature difference, °F

LMTD = Logarithmic mean temperature difference for true counterflow, °F

$$R = \frac{T_1 - T_2}{t_2 - t_1}$$

$$P = \frac{t_2 - t_1}{T_1 - t_1}$$

When using Figures B-1 thru B-5,

T_1, T_2 = Hot fluid inlet, outlet temperature, °F

t_1, t_2 = Cold fluid inlet, outlet temperature, °F

When R is greater than 1.0, it may be difficult reading F values off of Figures B-1 thru B-5. If this is the case, R and P may be recalculated using T_1 and T_2 interchanged with t_1 and t_2 , respectively.

When using Figures B-6 thru B-9,

T_1, T_2 = Shell side inlet, outlet temperature, °F

t_1, t_2 = Tube side inlet, outlet temperature, °F

In these cases, the temperatures are not interchangeable.

B3.0 Temperature Efficiency

The outlet temperatures T_2 and t_2 may be calculated as follows:

$$\begin{aligned}t_2 &= t_1 + P [T_1 - t_1] \\T_2 &= T_1 - R [t_2 - t_1]\end{aligned}$$

P, the temperature efficiency, is a function of R and NTU and can be obtained from Figures B-10 thru B-12, as applicable.

$$R = \frac{T_1 - T_2}{t_2 - t_1} = \frac{W_c C_{pc}}{W_h C_{ph}}$$

$$NTU = \frac{UA_o}{W_c C_{pc}}$$

where

- T_1, T_2 = Hot fluid inlet, outlet temperature, °F
- t_1, t_2 = Cold fluid inlet, outlet temperature, °F
- W_h, W_c = Hot fluid, cold fluid mass flow rate, lbm/hr
- C_{ph}, C_{pc} = Hot fluid, cold fluid heat capacity, Btu/lbm - °F
- U = Overall heat transfer coefficient (fouled), Btu/hr-ft² - °F (external surface)
- A_o = Effective external surface, ft²
- NTU = Number of Transfer Units

When R is greater than 1.0, it may be difficult reading NTU values off of Figures B-10 thru B-12. If this is the case, R, NTU, t_2 , and T_2 may be recalculated using W_h , C_{ph} , and T_1 interchanged with W_c , C_{pc} , and t_1 , respectively.

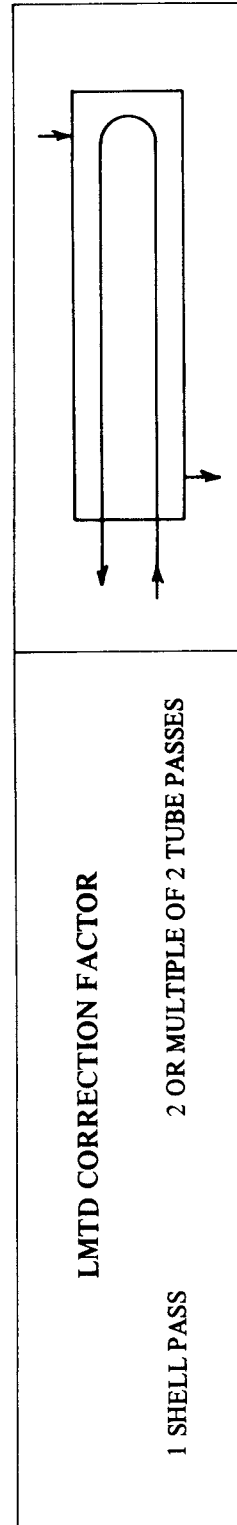
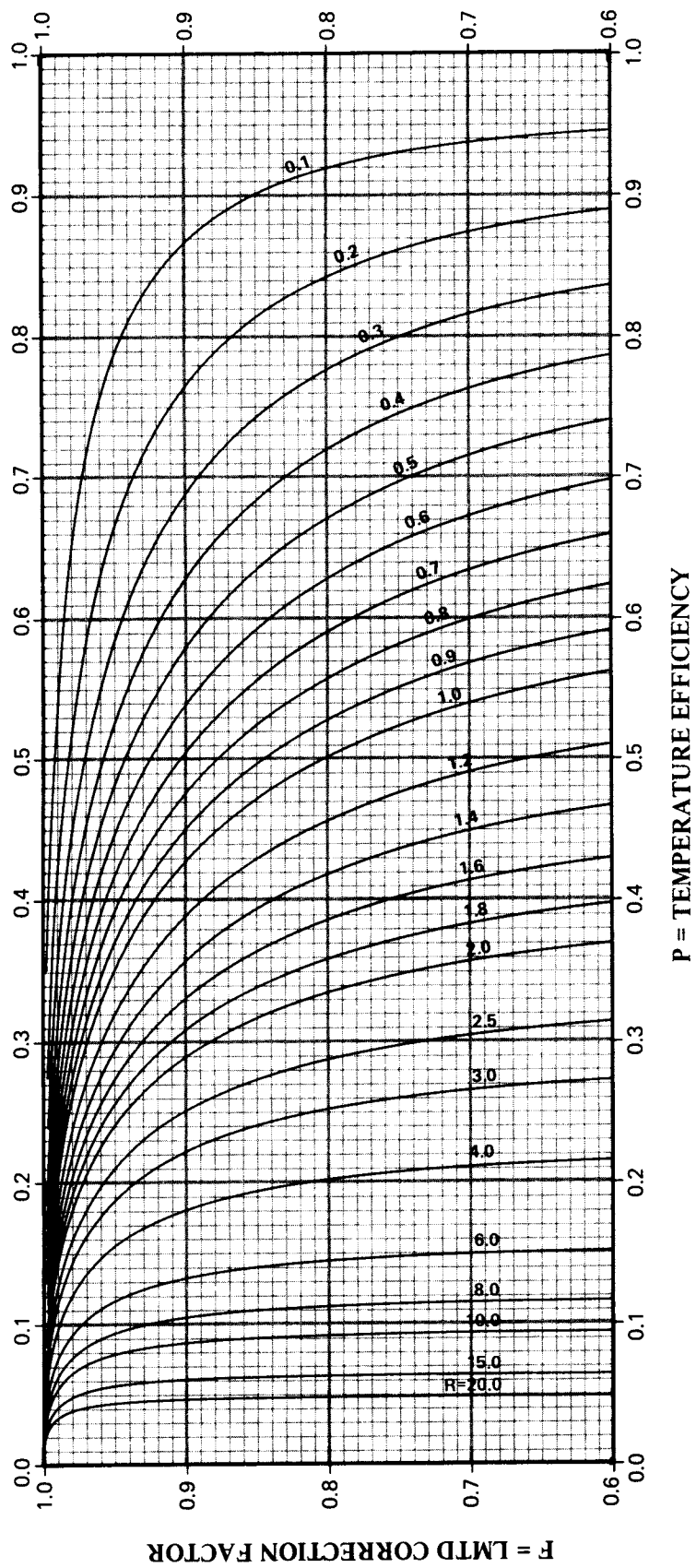
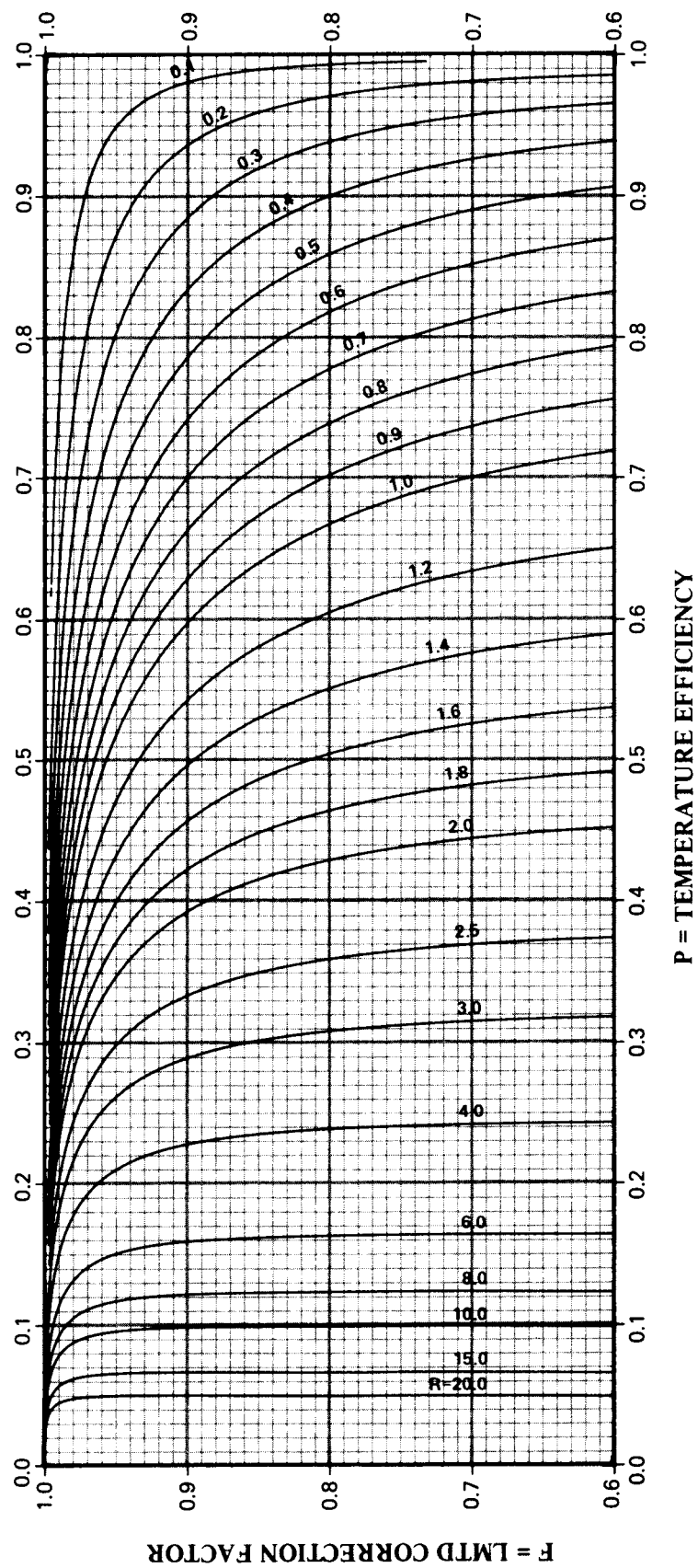


Figure B-1



P = TEMPERATURE EFFICIENCY

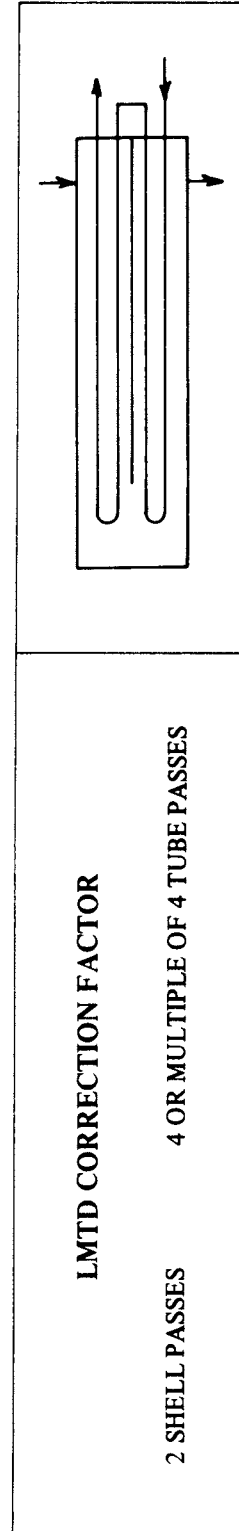


Figure B-2

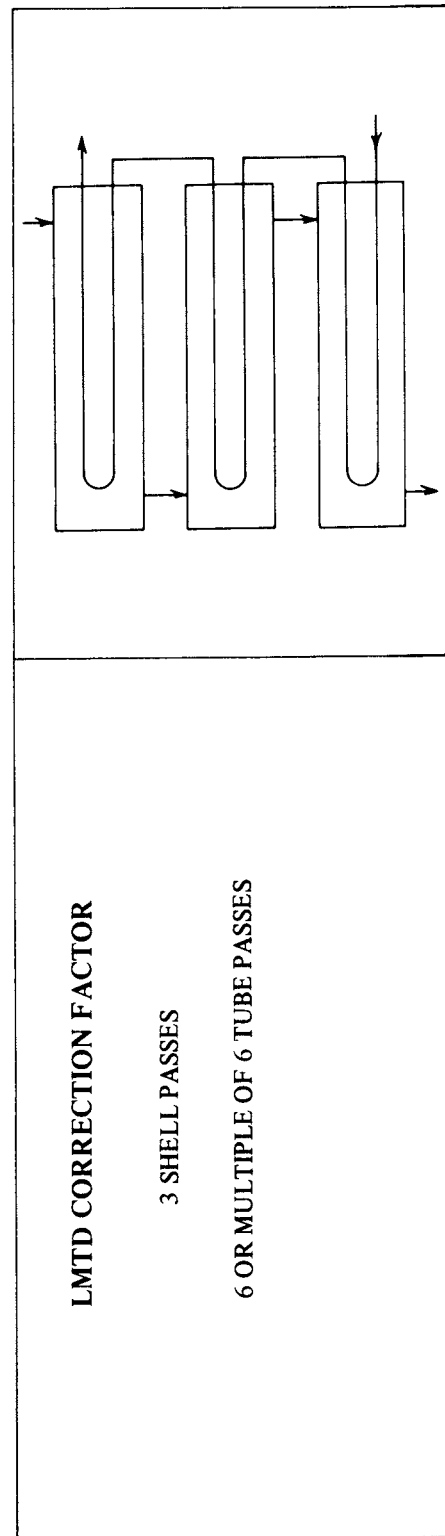
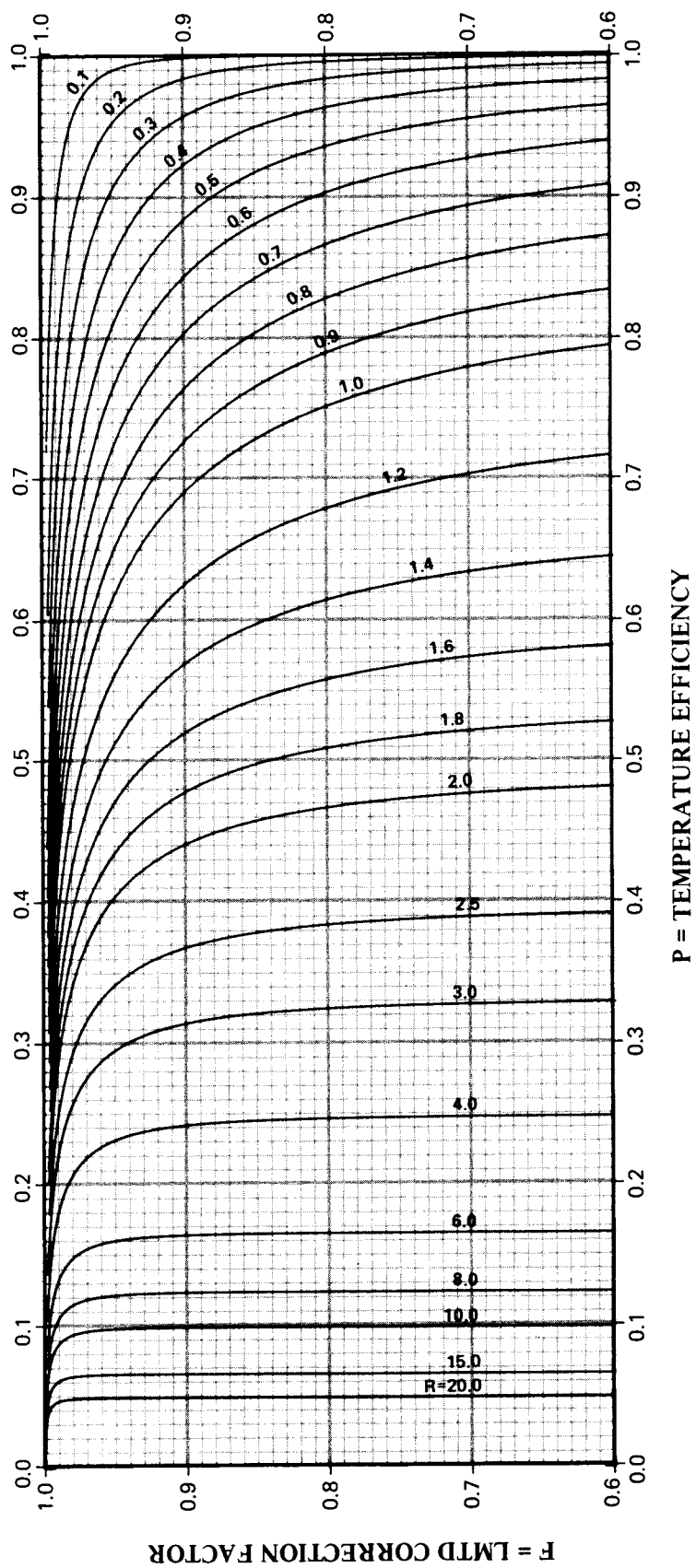


Figure B-3

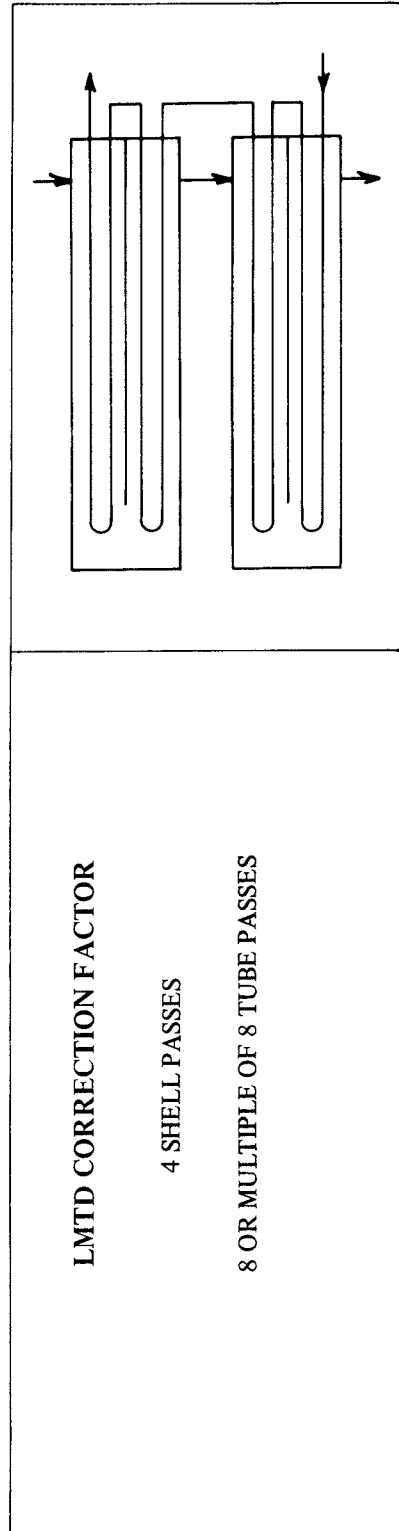
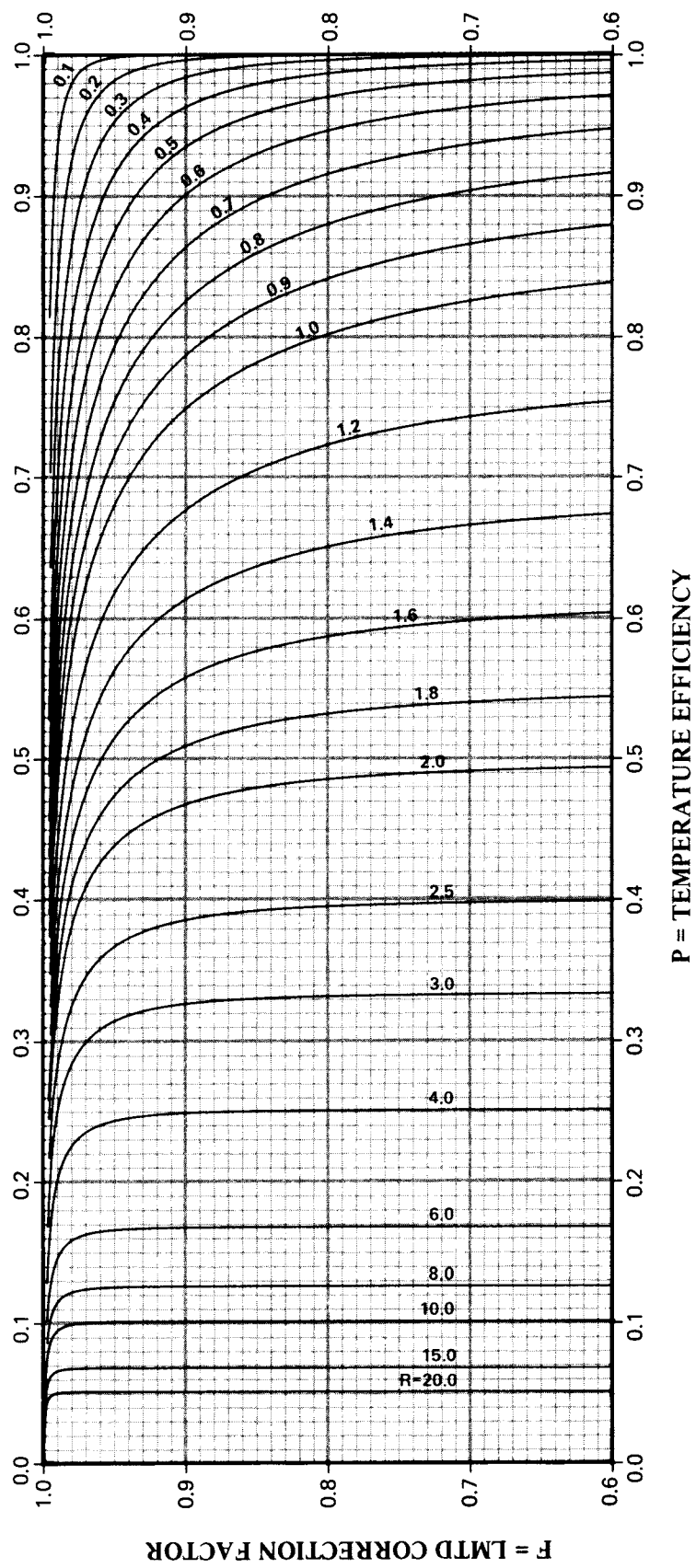


Figure B-4

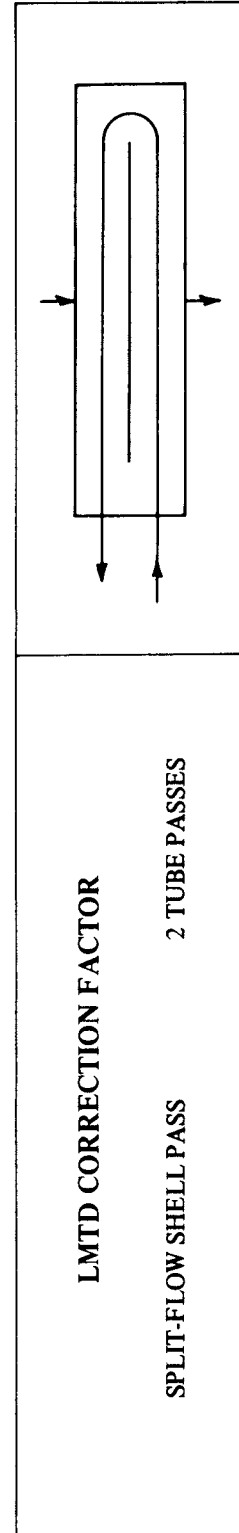
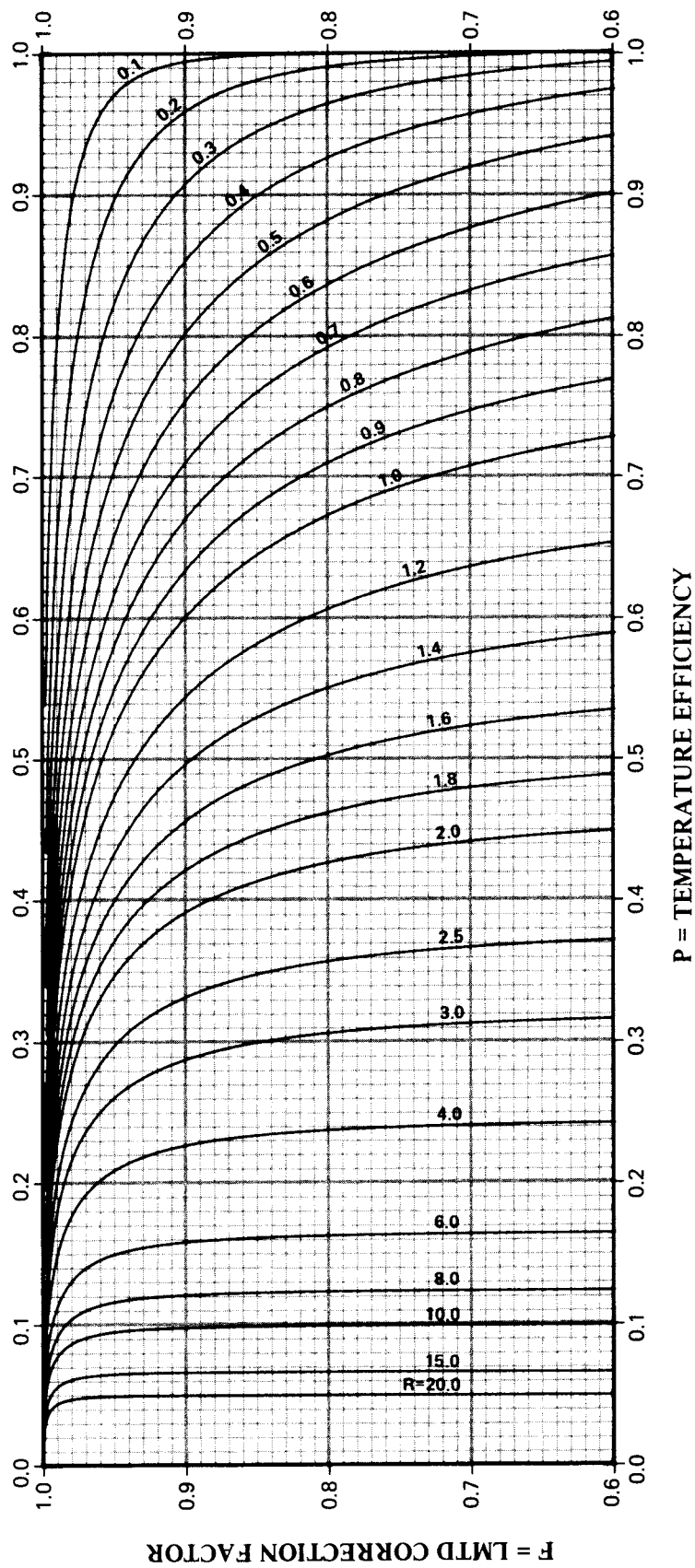


Figure B-6

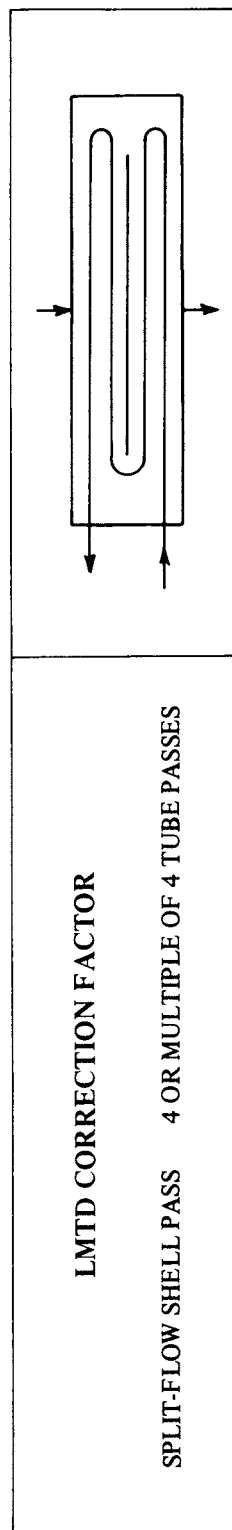
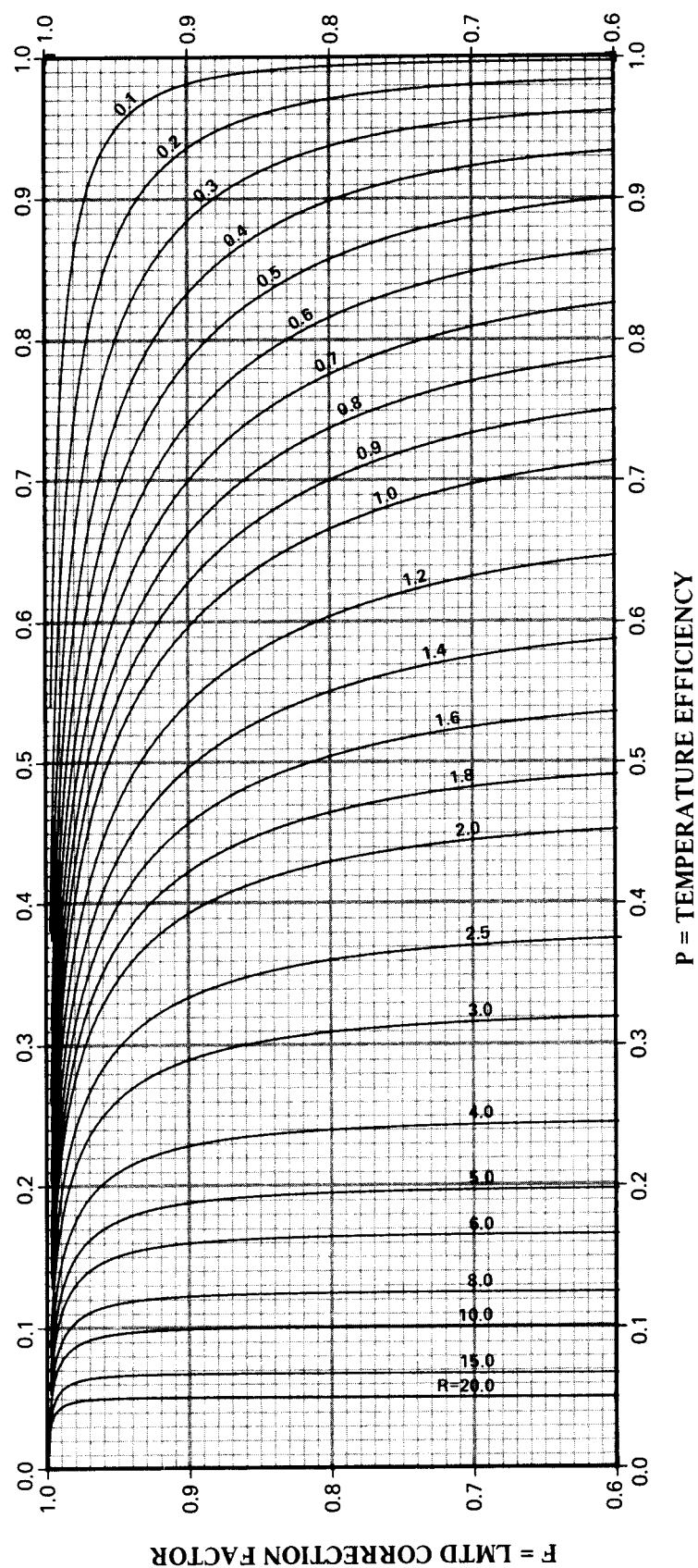


Figure B-7

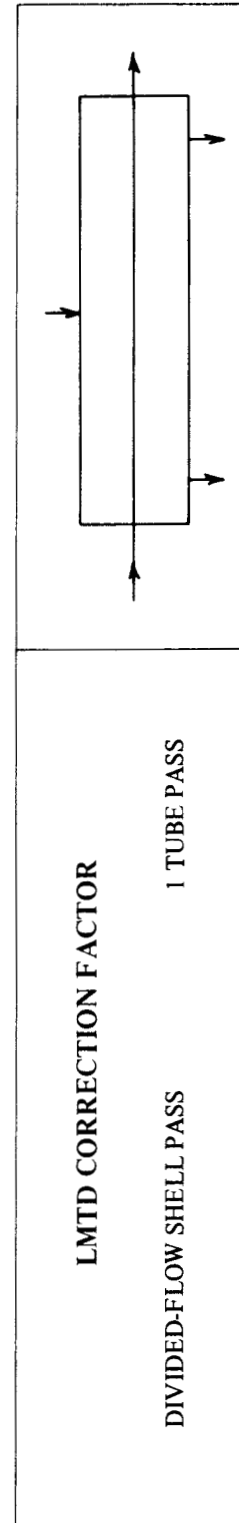
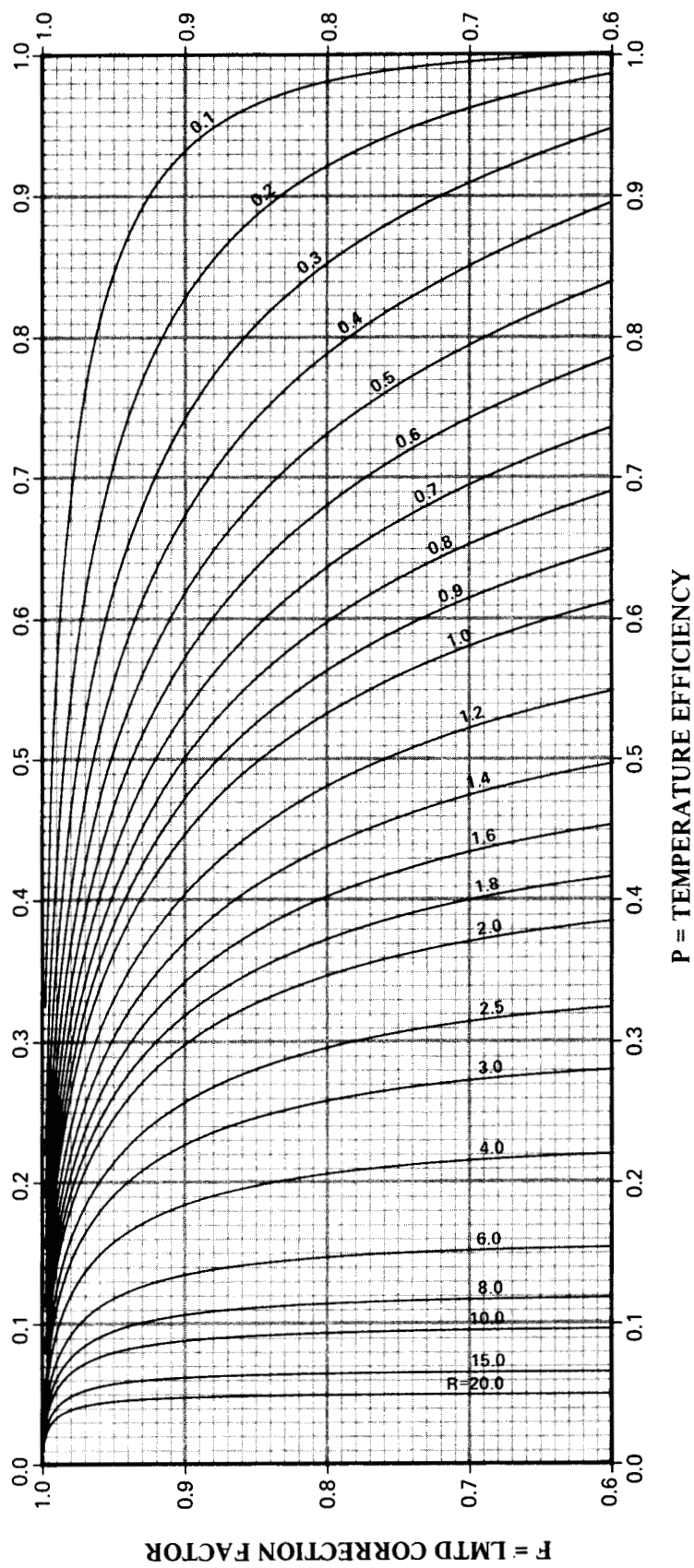


Figure B-8

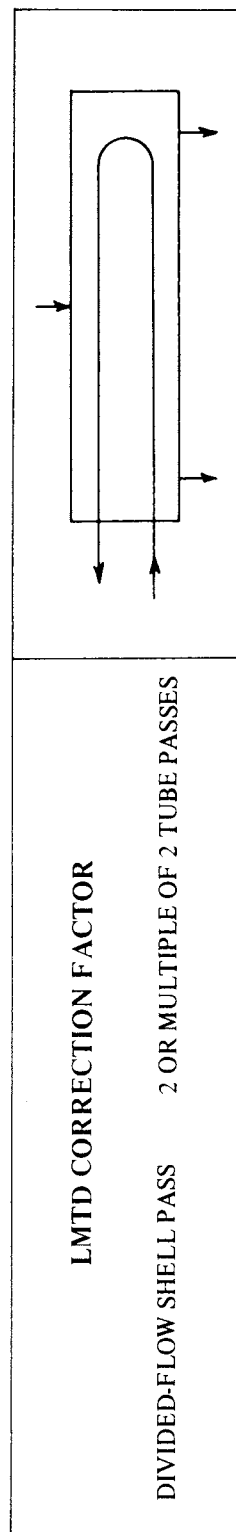
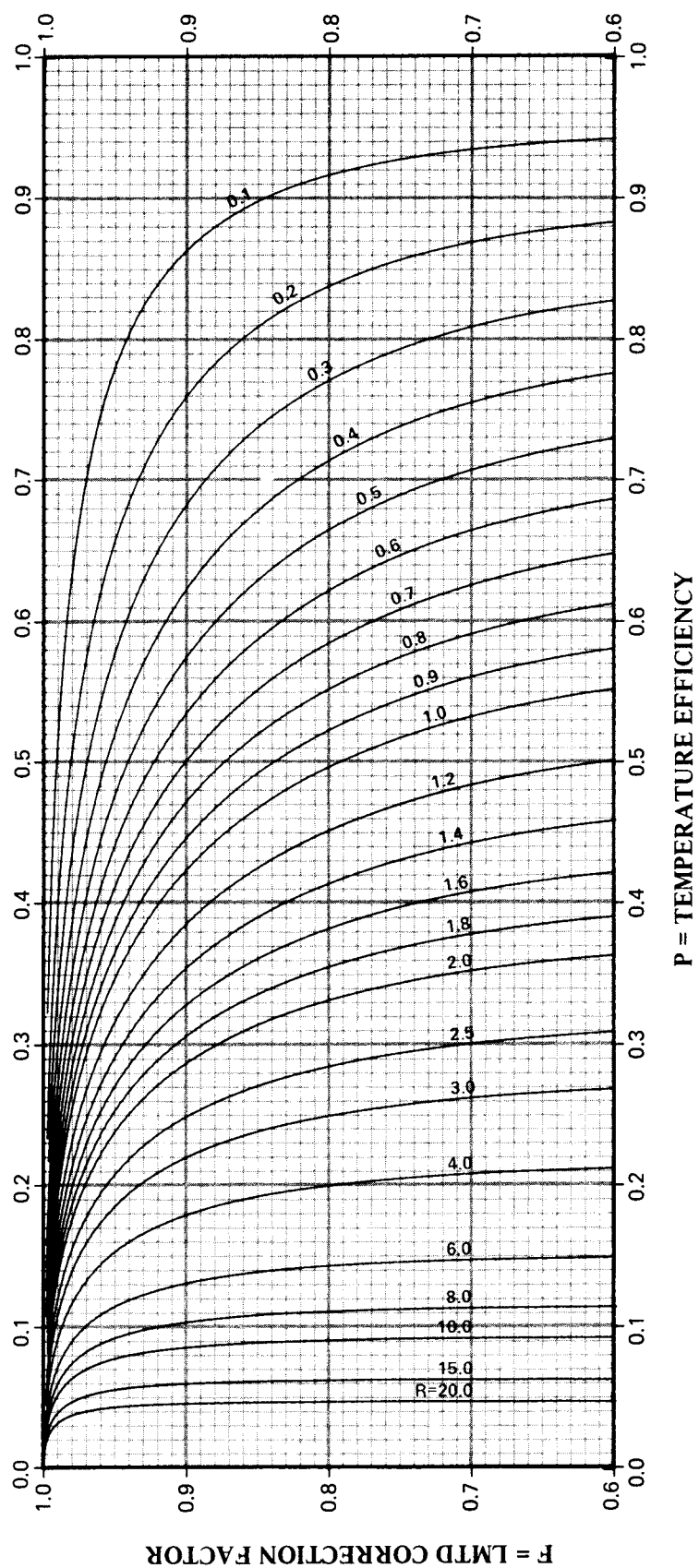


Figure B-9

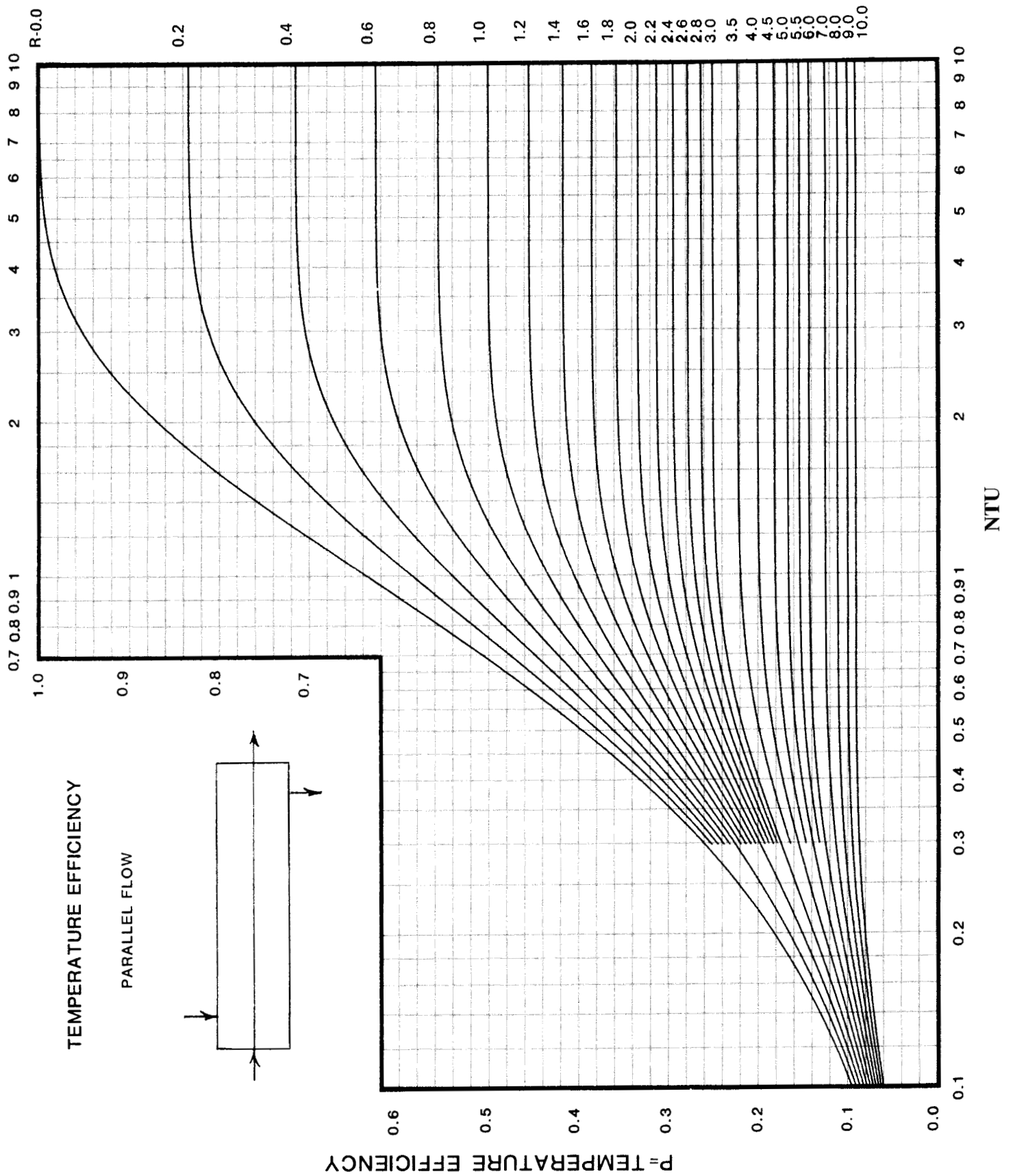


Figure B-10

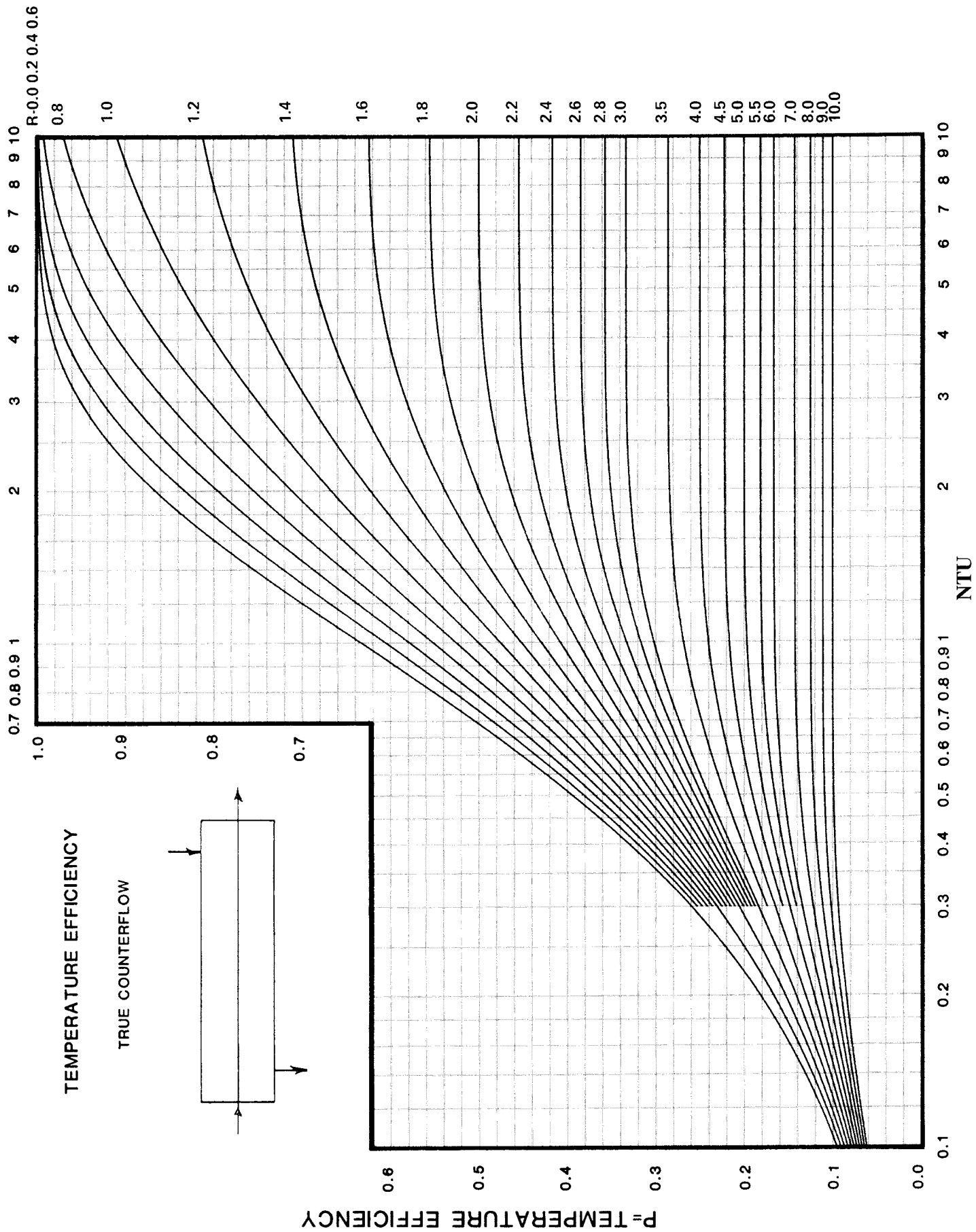


Figure B-11

APPENDIX C

PROCEDURE FOR CALCULATING ALLOWABLE NOZZLE EXTERNAL FORCES AND MOMENTS IN CYLINDRICAL VESSELS ⁽¹⁾

C1.0 Nomenclature

- P = Design Pressure, psi
 r_o = Nozzle Outside Radius, inches (see Figure C-1)
 R_m = Mean Radius of Shell, inches (see Figure C-1)
 T = Shell Thickness, inches (see Figure C-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 of Shell material 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 due to design pressure at design temperature, psi
 Section VIII, Division 1: Maximum Allowable Stress Value of Shell material, 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 C-1)
 M_{RCM} = Maximum Resultant Circumferential Moment, in-lbf* (see Figure C-1)
 M_{RLM} = Maximum Resultant Longitudinal Moment, in-lbf* (see Figure C-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.

* Use absolute values.

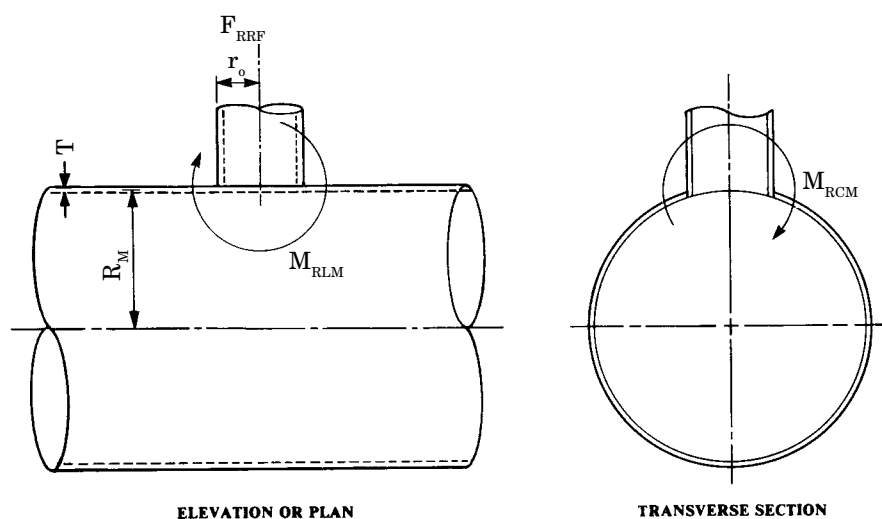


Figure C-1
NOMENCLATURE

C2.0 External Forces and Moments

To calculate the maximum force and moment, first evaluate β and γ . Then determine α , Σ and Δ from Figures C-2, C-3 and C-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 C-2, C-3 and C-4.

Calculate Pressure Stress, σ .

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

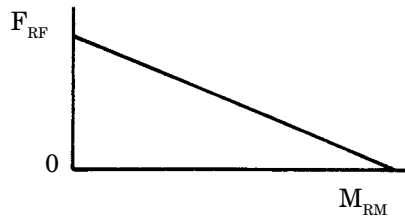
If σ is greater than S_a , then use S_a as the stress due to design pressure:

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

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

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

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} .



C3.0 Sample Problem

Determine Resultant Force and Moment

$$R_m = 37.5 \text{ in} \quad S_y = 31,500 \text{ psi @ } 460^\circ\text{F}$$

$$r_o = 15 \text{ in} \quad S_a = 17,500 \text{ psi}$$

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

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

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

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

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

From Figure C-3, $\Sigma = 1,100$

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

Calculate Pressure Stress

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

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

$$14,850 \text{ psi} < S_a = 17,500 \text{ psi}$$

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

Calculate Allowable Forces and Moments

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

$$\frac{(37.5)^2}{440} (31,500 - 14,850) = 53,214 \text{ lbf}$$

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

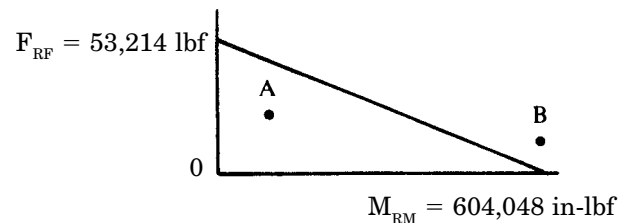
$$\frac{(37.5)^2 (15) (31,500)}{1,100} = 604,048 \text{ in-lbf}$$

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

$$\frac{(37.5)^2 (15)}{340} (31,500 - 14,850) =$$

$$1,032,973 \text{ in-lbf}$$

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 $M = 604,000^*$ in-lbf would not be allowable (point B).

*Note: Use absolute values in the graph.

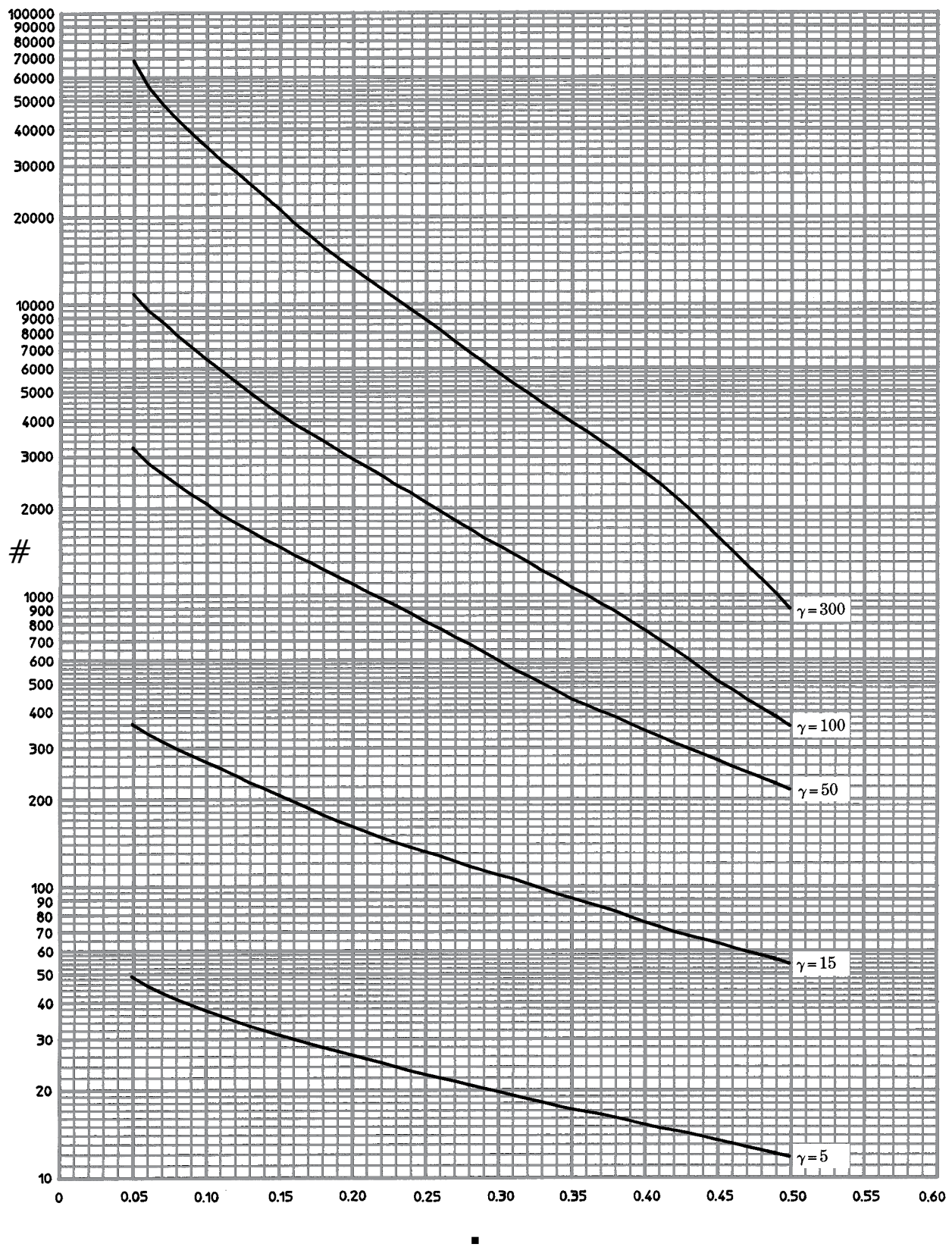


Figure C-2
ALLOWABLE NOZZLE LOADS

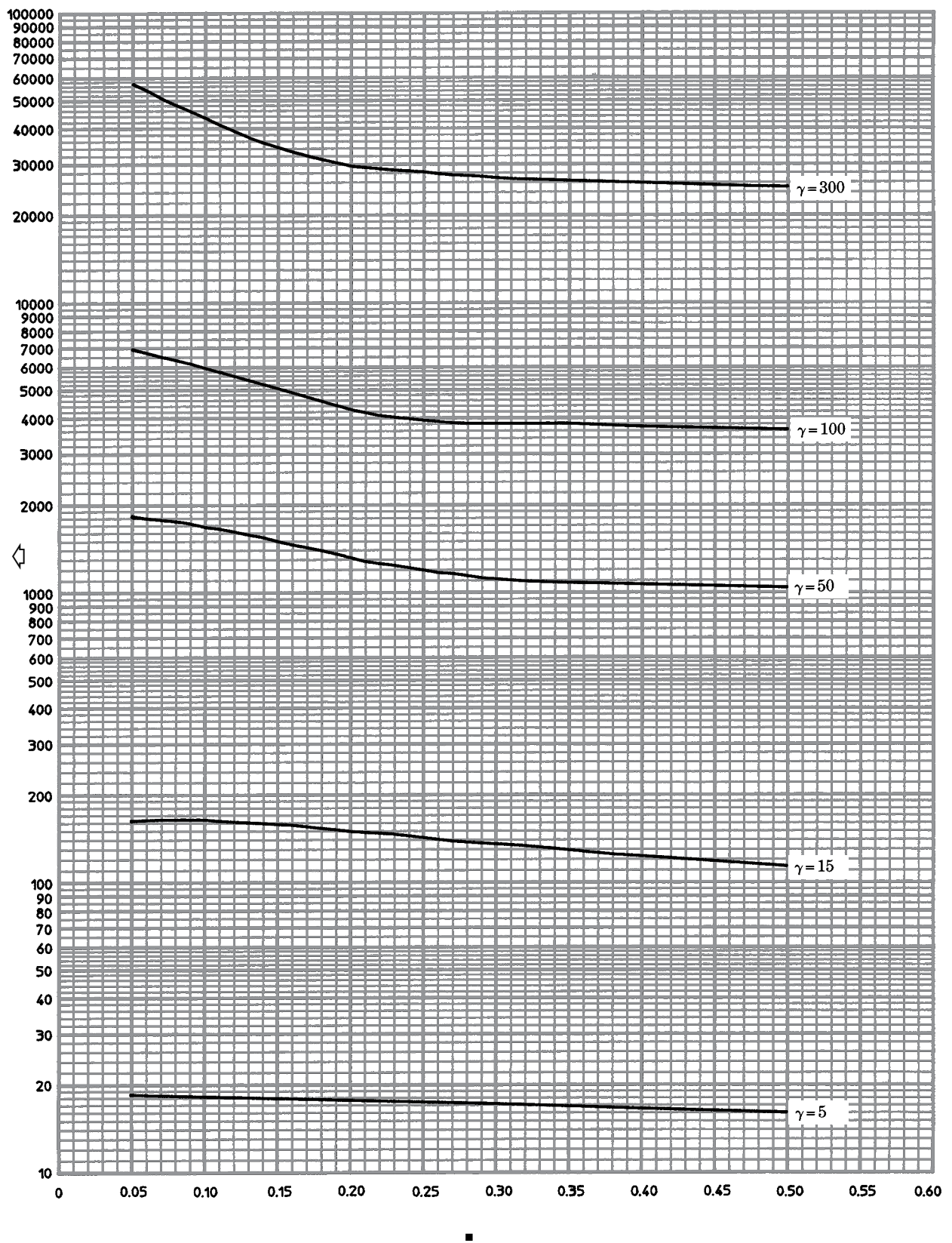


Figure C-3
ALLOWABLE NOZZLE LOADS

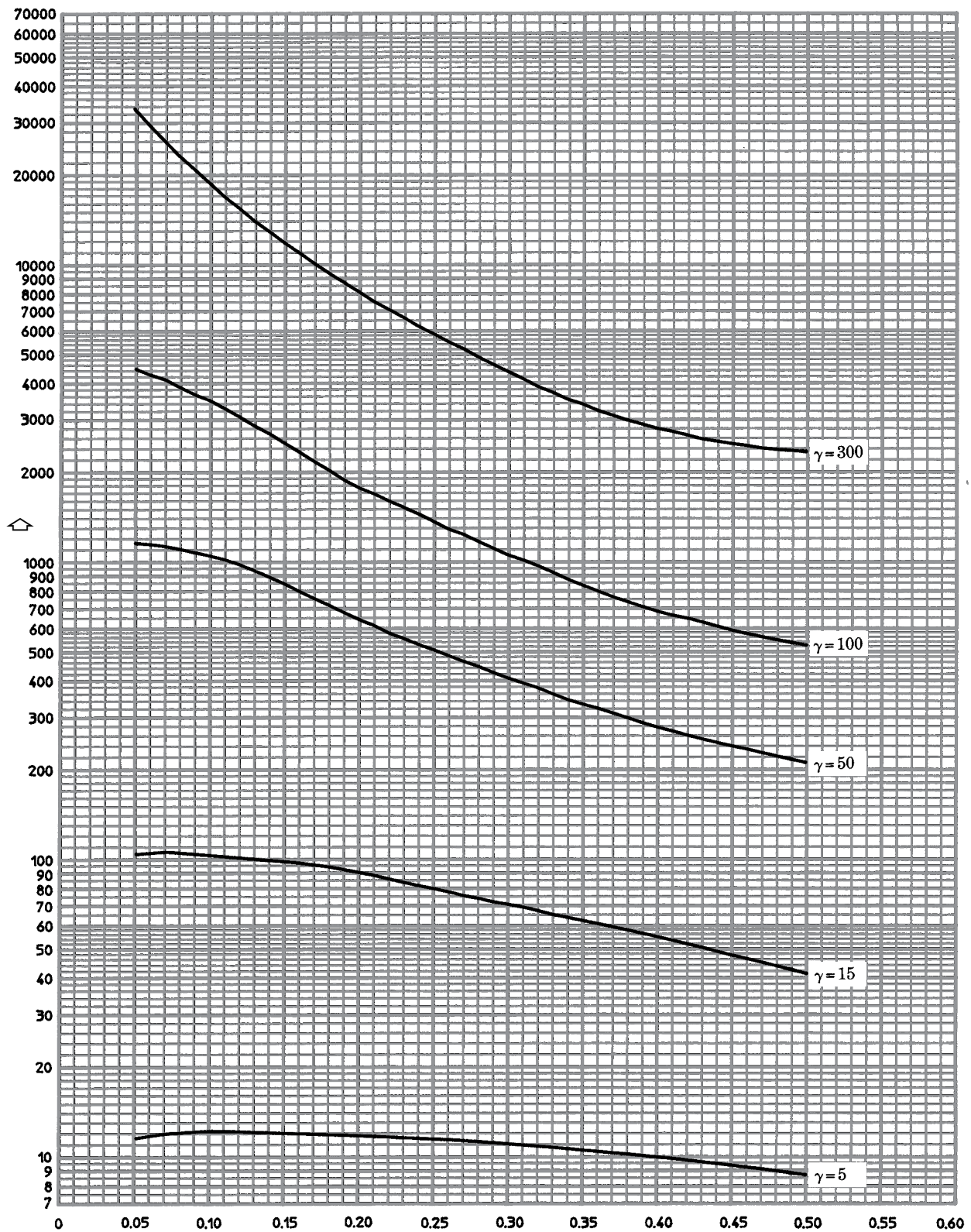


Figure C-4
ALLOWABLE NOZZLE LOADS

APPENDIX D

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	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	0.450	0.34278
.002	.00012	.051	.01512	.101	.04148	.151	.07459	.201	.11262	.251	.15441	.301	.19908	.351	.24593	.401	.29435	.451	.34378
.003	.00022	.052	.01556	.102	.04208	.152	.07531	.202	.11343	.252	.15528	.302	.20000	.352	.24689	.402	.29533	.452	.34477
.004	.00034	.053	.01601	.103	.04269	.153	.07603	.203	.11423	.253	.15615	.303	.20092	.353	.24784	.403	.29631	.453	.34577
.005	.00047	.054	.01646	.104	.04330	.154	.07675	.204	.11504	.254	.15702	.304	.20184	.354	.24880	.404	.29729	.454	.34676
.006	.00060	.055	.01691	.105	.04391	.155	.07747	.205	.11584	.255	.15789	.305	.20276	.355	.24976	.405	.29827	.455	.34776
.007	.00078	.056	.01737	.106	.04452	.156	.07819	.206	.11665	.256	.15876	.306	.20368	.356	.25071	.406	.29926	.456	.34876
.008	.00095	.057	.01783	.107	.04514	.157	.07892	.207	.11746	.257	.15964	.307	.20460	.357	.25167	.407	.30024	.457	.34975
.009	.00113	.058	.01830	.108	.04576	.158	.07965	.208	.11827	.258	.16051	.308	.20553	.358	.25263	.408	.30122	.458	.35075
.010	.00133	.059	.01877	.109	.04638	.159	.08038	.209	.11908	.259	.16139	.309	.20645	.359	.25359	.409	.30220	.459	.35175
.011	.00153	.060	.01924	.110	.04701	.160	.08111	.210	.11990	.260	.16226	.310	.20738	.360	.25455	.410	.30319	.460	.35274
.012	.00175	.061	.01972	.111	.04763	.161	.08185	.211	.12071	.261	.16314	.311	.20830	.361	.25551	.411	.30417	.461	.35374
.013	.00197	.062	.02020	.112	.04826	.162	.08258	.212	.12153	.262	.16402	.312	.20923	.362	.25647	.412	.30516	.462	.35474
.014	.00220	.063	.02068	.113	.04889	.163	.08332	.213	.12235	.263	.16490	.313	.21015	.363	.25743	.413	.30614	.463	.35573
.015	.00244	.064	.02117	.114	.04953	.164	.08406	.214	.12317	.264	.16578	.314	.21108	.364	.25839	.414	.30712	.464	.35673
.016	.00268	.065	.02166	.115	.05016	.165	.08480	.215	.12399	.265	.16666	.315	.21201	.365	.25936	.415	.30811	.465	.35773
.017	.00294	.066	.02215	.116	.05080	.166	.08554	.216	.12481	.266	.16755	.316	.21294	.366	.26032	.416	.30910	.466	.35873
.018	.00320	.067	.02265	.117	.05145	.167	.08629	.217	.12563	.267	.16843	.317	.21387	.367	.26128	.417	.31008	.467	.35972
.019	.00347	.068	.02315	.118	.05209	.168	.08704	.218	.12646	.268	.16932	.318	.21480	.368	.26225	.418	.31107	.468	.36072
.020	.00375	.069	.02366	.119	.05274	.169	.08779	.219	.12729	.269	.17020	.319	.21573	.369	.26321	.419	.31205	.469	.36172
.021	.00403	.070	.02417	.120	.05338	.170	.08854	.220	.12811	.270	.17109	.320	.21667	.370	.26418	.420	.31304	.470	.36272
.022	.00432	.071	.02468	.121	.05404	.171	.08929	.221	.12894	.271	.17198	.321	.21760	.371	.26514	.421	.31403	.471	.36372
.023	.00462	.072	.02520	.122	.05469	.172	.09004	.222	.12977	.272	.17287	.322	.21853	.372	.26611	.422	.31502	.472	.36471
.024	.00492	.073	.02571	.123	.05535	.173	.09080	.223	.13060	.273	.17376	.323	.21947	.373	.26708	.423	.31600	.473	.36571
.025	.00523	.074	.02624	.124	.05600	.174	.09155	.224	.13144	.274	.17465	.324	.22040	.374	.26805	.424	.31699	.474	.36671
.026	.00555	.075	.02676	.125	.05666	.175	.09231	.225	.13227	.275	.17554	.325	.22134	.375	.26901	.425	.31798	.475	.36771
.027	.00587	.076	.02729	.126	.05733	.176	.09307	.226	.13311	.276	.17644	.326	.22228	.376	.26998	.426	.31897	.476	.36871
.028	.00619	.077	.02782	.127	.05799	.177	.09384	.227	.13395	.277	.17733	.327	.22322	.377	.27095	.427	.31996	.477	.36971
.029	.00653	.078	.02836	.128	.05866	.178	.09460	.228	.13478	.278	.17823	.328	.22415	.378	.27192	.428	.32095	.478	.37071
.030	.00687	.079	.02889	.129	.05933	.179	.09537	.229	.13562	.279	.17912	.329	.22509	.379	.27289	.429	.32194	.479	.37171
.031	.00721	.080	.02943	.130	.06000	.180	.09613	.230	.13646	.280	.18002	.330	.22603	.380	.27386	.430	.32293	.480	.37270
.032	.00756	.081	.02998	.131	.06067	.181	.09690	.231	.13731	.281	.18092	.331	.22697	.381	.27483	.431	.32392	.481	.37370
.033	.00791	.082	.03053	.132	.06135	.182	.09767	.232	.13815	.282	.18182	.332	.22792	.382	.27580	.432	.32491	.482	.37470
.034	.00827	.083	.03108	.133	.06203	.183	.09845	.233	.13900	.283	.18272	.333	.22886	.383	.27678	.433	.32590	.483	.37570
.035	.00864	.084	.03163	.134	.06271	.184	.09922	.234	.13984	.284	.18362	.334	.22980	.384	.27775	.434	.32689	.484	.37670
.036	.00901	.085	.03219	.135	.06339	.185	.10000	.235	.14069	.285	.18452	.335	.23074	.385	.27872	.435	.32788	.485	.37770
.037	.00938	.086	.03275	.136	.06407	.186	.10077	.236	.14154	.286	.18542	.336	.23169	.386	.27969	.436	.32887	.486	.37870
.038	.00976	.087	.03331	.137	.06476	.187	.10155	.237	.14239	.287	.18633	.337	.23263	.387	.28067	.437	.32987	.487	.37970
.039	.01015	.088	.03387	.138	.06545	.188	.10233	.238	.14324	.288	.18723	.338	.23358	.388	.28164	.438	.33086	.488	.38070
.040	.01054	.089	.03444	.139	.06614	.189	.10312	.239	.14409	.289	.18814	.339	.23453	.389	.28262	.439	.33185	.489	.38170
.041	.01093	.090	.03501	.140	.06683	.190	.10390	.240	.14494	.290	.18905	.340	.23547	.390	.28359	.440	.33284	.490	.38270
.042	.01133	.091	.03559	.141	.06753	.191	.10469	.241	.14580	.291	.18996	.341	.23642	.391	.28457	.441	.33384	.491	.38370
.043	.01173	.092	.03616	.142	.06822	.192	.10547	.242	.14666	.292	.19086	.342	.23737	.392	.28554	.442	.33483	.492	.38470
.044	.01214	.093	.03674	.143	.06892	.193	.10626	.243	.14751	.293	.19177	.343	.23832	.393	.28652	.443	.33582	.493	.38570
.045	.01255	.094	.03732	.144	.06963	.194	.10705	.244	.14837	.294	.19268	.344	.23927	.394	.28750	.444	.33682	.494	.38670
.046	.01297	.095	.03791	.145	.07033	.195	.10784	.245	.14923	.295	.19360	.345	.24022	.395	.28848	.445	.33781	.495	.38770
.047	.01339	.096	.03850	.146	.07103	.196	.10864	.246	.15009	.296	.19451	.346	.24117	.396	.28945	.446	.33880	.496	.38870
.048	.01382	.097	.03909	.147	.07174	.197	.10943	.247	.15095	.297	.19542	.347	.24212	.397	.29043	.447	.33980	.497	.38970
.049	.01425	.098	.03968	.148	.07245	.198	.11023	.248	.15182	.298	.19634	.348	.24307	.398	.29141	.448	.34079	.498	.39070
.050	.01468	.099	.04028	.149	.07316	.199	.11102	.249	.15268	.299	.19725	.349	.24403	.399	.29239	.449	.34179	.499	.39170
.051	.01512																.500		.39270

D1.0 Calculation of Area

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

where

D = Diameter, in

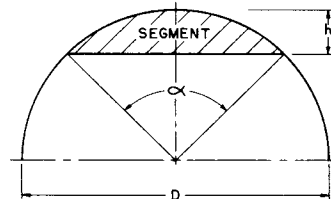
h = Segment height, in

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

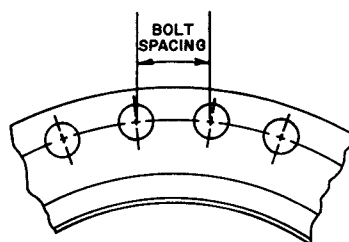
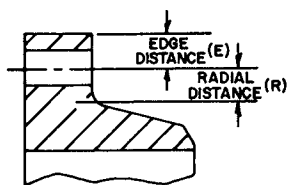
$$\alpha = 2 \cos^{-1} \left(1 - \frac{2h}{D} \right)$$



APPENDIX E

BOLTING DATA

Nominal Bolt Size (in)	Thread Data			Heavy Hex Nut Dimensions		Minimum Dimensions		
	No. of Threads per in n	Root Area (in ²) A _b	Pitch Diameter (in) D _i	Across Flats (in) D _o	Across Corners (in)	Bolt Spacing (in)	Radial Distance (in) R	Edge Distance (in) E
1/2	13	0.126	0.4485	0.875	0.969	1 1/4	1 3/16	5/8
5/8	11	0.202	0.5644	1.062	1.175	1 1/2	1 5/16	3/4
3/4	10	0.302	0.6832	1.250	1.382	1 3/4	1 1/8	1 3/16
7/8	9	0.419	0.8009	1.438	1.589	2 1/16	1 1/4	1 5/16
1	8	0.551	0.9168	1.625	1.796	2 1/4	1 3/8	1 1/2
1 1/8	8	0.728	1.0417	1.812	2.002	2 1/2	1 1/2	1 1/8
1 1/4	8	0.929	1.1667	2.000	2.209	2 3/16	1 3/4	1 1/4
1 3/8	8	1.155	1.2916	2.188	2.416	3 1/16	1 7/8	1 3/8
1 1/2	8	1.405	1.4166	2.375	2.622	3 1/4	2	1 1/2
1 5/8	8	1.680	1.5416	2.562	2.828	3 1/2	2 1/8	1 5/8
1 3/4	8	1.980	1.6665	2.750	3.035	3 3/4	2 1/4	1 3/4
1 7/8	8	2.304	1.7915	2.938	3.242	4	2 3/8	1 7/8
2	8	2.652	1.9165	3.125	3.449	4 1/4	2 1/2	2
2 1/4	8	3.423	2.1664	3.500	3.862	4 3/4	2 3/4	2 1/4
2 1/2	8	4.292	2.4164	3.875	4.275	5 1/4	3 1/16	2 3/8
2 3/4	8	5.259	2.6663	4.250	4.688	5 3/4	3 3/8	2 5/8
3	8	6.324	2.9162	4.625	5.102	6 1/4	3 5/8	2 7/8
3 1/4	8	7.487	3.1662	5.000	5.515	6 3/4	3 7/8	3
3 1/2	8	8.749	3.4162	5.375	5.928	7 1/8	4 1/8	3 1/4
3 3/4	8	10.108	3.6661	5.750	6.341	7 5/8	4 3/8	3 1/2
4	8	11.566	3.9161	6.125	6.755	8 1/8	4 5/8	3 5/8



NUT DIMENSIONS ARE PER ANSI B18.2.2
THREAD DIMENSIONS ARE PER ANSI B1.1

(Continued on following page)

E1.0 Calculation of Applied Torque on Lubricated Studs and Bolts

Caution—The torque values derived from the following equations are not intended to be those for gasket seating. User should refer to the Manufacturer's operating instruction manual for the proper gasket seating torque.

The following equation may be used to calculate the applied torque on the nut to develop the stress in the bolts:

$$T = \text{Torque} = \frac{SA_b F}{12}, \text{ ft-lbf}$$

where

S = Bolt stress, psi

$$F = \frac{U_1 D_i}{2} + \frac{1}{2\pi n} + \frac{\pi U_2}{12A_n} (D_o^3 - D_i^3), \text{ in}$$

$$A_n = \frac{\pi}{4} (D_o^2 - D_i^2), \text{ in}^2$$

U_1 = Factor for friction between nut and stud

U_2 = Factor for friction between nut surface and bearing surface

Typical value of the factor for friction for lubricated surfaces is 0.15; however, this value may vary between Manufacturers.

Other variables in the equations above are found in the preceding Bolting Data Table.

APPENDIX F-1

HEAT EXCHANGE INSTITUTE INC.

HEAT EXCHANGER SPECIFICATION SHEET

English Units

	No	Date	By	Ckd.
Revision				

1	CUSTOMER	DATE
2	ADDRESS	CUSTOMER NO.
3	PLANT LOCATION	PROPOSAL NO.
		JOB NO.
4	SERVICE OF UNIT	ITEM NO.
5	SIZE TYPE	POSITION (HOR) (VERT)
6	NO. UNITS SHELLS/UNIT	CONNECTED IN
7	EFFECTIVE GROSS SURFACE/UNIT – ft ²	SURFACE/SHELL – ft ²
8	PERFORMANCE OF ONE UNIT	
9		SHELL SIDE TUBE SIDE
10	FLUID CIRCULATED	
11	TOTAL FLUID ENTERING lbm/hr	
12	VAPOR lbm/hr	
13	LIQUID lbm/hr	
14	STEAM lbm/hr	
15	NON-CONDENSABLES lbm/hr	
16	FLUID VAPORIZED OR CONDENSED lbm/hr	
17	STEAM CONDENSED lbm/hr	
18	DENSITY lbm/ft ³	
19	VISCOSITY cp	@ °F
20	VISCOSITY cp	@ °F
21	SPECIFIC HEAT Btu/lbm-°F	
22	THERMAL CONDUCTIVITY Btu-ft/hr-ft ² -°F	
23	LATENT HEAT Btu/lbm	
24		
25	TEMPERATURE IN °F	
26	TEMPERATURE OUT °F	
27	OPERATING PRESSURE (abs) psia	
28	NUMBER OF PASSES PER SHELL	
29	VELOCITY ft/sec	
30	PRESSURE LOSS psi	
31	FOULING RESISTANCE hr-ft ² -°F/Btu	
32	HEAT EXCHANGER DUTY – Btu/hr	MTD-°F
33	SERVICE RATE – Btu/hr-ft ² -°F	CLEAN
34		
35	CONSTRUCTION OF ONE SHELL	
36	DESIGN PRESSURE psig	
37	TEST PRESSURE psig	
38	DESIGN TEMPERATURE (max/min) °F	/
39	GASKETS	
40	NOZZLES – INLET, TYPE in	
41	– OUTLET, TYPE in	
42	CORROSION ALLOWANCE in	
43	CODE REQUIREMENTS	
44	TUBES NO. OD BWG (avg/min) LENGTH PITCH	
45	TUBESHEET	TUBE JOINT TYPE
46	SHELL EXP. JOINT	SHELL COVER (INT/REMOVABLE)
47	CHANNEL OR BONNET	CHANNEL COVER
48	BAFFLE – IMPINGEMENT	FLOATING HEAD
49	BAFFLES – CROSS TYPE	SPACING CUT
50	BAFFLE – LONG TYPE	TUBE SUPPORTS
51	WEIGHTS – EMPTY BUNDLE	FULL OF WATER
52	ACCESSORIES:	
53	NOTES:	
54		
55		

APPENDIX F-2

HEAT EXCHANGE INSTITUTE INC.

HEAT EXCHANGER SPECIFICATION SHEET

SI Units

Revision	No	Date	By	Ckd.

1	CUSTOMER	DATE
2	ADDRESS	CUSTOMER NO.
3	PLANT LOCATION	PROPOSAL NO.
		JOB NO.
4	SERVICE OF UNIT	ITEM NO.
5	SIZE TYPE	POSITION (HOR) (VERT)
6	NO. UNITS SHELLS/UNIT	CONNECTED IN
7	EFFECTIVE GROSS SURFACE/UNIT – m ²	SURFACE/SHELL – m ²
8	PERFORMANCE OF ONE UNIT	
9		SHELL SIDE TUBE SIDE
10	FLUID CIRCULATED	
11	TOTAL FLUID ENTERING kg/s	
12	VAPOR kg/s	
13	LIQUID kg/s	
14	STEAM kg/s	
15	NON-CONDENSABLES kg/s	
16	FLUID VAPORIZED OR CONDENSED kg/s	
17	STEAM CONDENSED kg/s	
18	DENSITY kg/m ³	
19	VISCOSITY mPa • s @ °C	@ °C
20	VISCOSITY mPa • s @ °C	@ °C
21	SPECIFIC HEAT kJ/kg • °C	
22	THERMAL CONDUCTIVITY W/m • °C	
23	LATENT HEAT kJ/kg	
24		
25	TEMPERATURE IN °C	
26	TEMPERATURE OUT °C	
27	OPERATING PRESSURE (abs) kPa	
28	NUMBER OF PASSES PER SHELL	
29	VELOCITY m/s	
30	PRESSURE LOSS kPa	
31	FOULING RESISTANCE m ² • °C/W	
32	HEAT EXCHANGER DUTY – MW	MTD-°C
33	SERVICE RATE – W/m ² • °K	CLEAN
34		
35	CONSTRUCTION OF ONE SHELL	
36	DESIGN PRESSURE kPag	
37	TEST PRESSURE kPag	
38	DESIGN TEMPERATURE (max/min) °C	/ /
39	GASKETS	
40	NOZZLES – INLET, TYPE mm	
41	– OUTLET, TYPE mm	
42	CORROSION ALLOWANCE mm	
43	CODE REQUIREMENTS	
44	TUBES NO. OD BWG (avg/min) LENGTH PITCH	
45	TUBESHEET	TUBE JOINT TYPE
46	SHELL EXP. JOINT	SHELL COVER (INT/REMOVABLE)
47	CHANNEL OR BONNET	CHANNEL COVER
48	BAFFLE – IMPINGEMENT	FLOATING HEAD
49	BAFFLES – CROSS TYPE SPACING CUT	
50	BAFFLE – LONG TYPE	TUBE SUPPORTS
51	WEIGHTS – EMPTY BUNDLE	FULL OF WATER
52	ACCESSORIES:	
53	NOTES:	
54		
55		

APPENDIX F-3

HEAT EXCHANGE INSTITUTE INC.

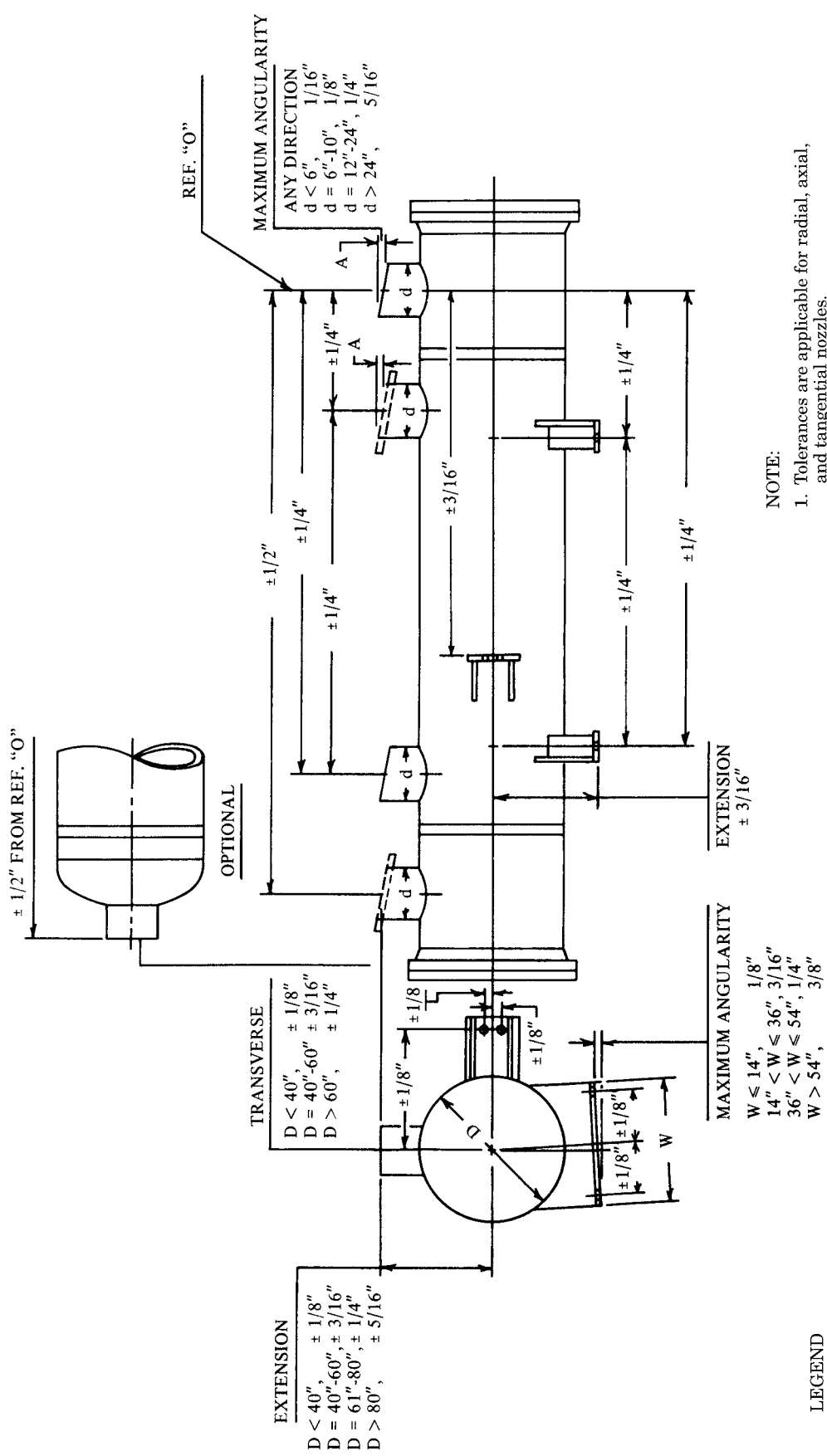
HEAT EXCHANGER SPECIFICATION SHEET

MKH Units

	No	Date	By	Ckd.
Revision				

1	CUSTOMER	DATE
2	ADDRESS	CUSTOMER NO.
3	PLANT LOCATION	PROPOSAL NO.
		JOB NO.
4	SERVICE OF UNIT	ITEM NO.
5	SIZE TYPE	POSITION (HOR) (VERT)
6	NO. UNITS SHELLS/UNIT	CONNECTED IN
7	EFFECTIVE GROSS SURFACE/UNIT – m ²	SURFACE/SHELL – m ²
8	PERFORMANCE OF ONE UNIT	
9		SHELL SIDE TUBE SIDE
10	FLUID CIRCULATED	
11	TOTAL FLUID ENTERING kg/h	
12	VAPOR kg/h	
13	LIQUID kg/h	
14	STEAM kg/h	
15	NON-CONDENSABLES kg/h	
16	FLUID VAPORIZED OR CONDENSED kg/h	
17	STEAM CONDENSED kg/h	
18	DENSITY kg/m ³	
19	VISCOSITY cp @ °C	@ °C
20	VISCOSITY cp @ °C	@ °C
21	SPECIFIC HEAT kcal/kg • °C	
22	THERMAL CONDUCTIVITY kcal/h • m • °C	
23	LATENT HEAT kcal/kg	
24		
25	TEMPERATURE IN °C	
26	TEMPERATURE OUT °C	
27	OPERATING PRESSURE (abs) kg/cm ²	
28	NUMBER OF PASSES PER SHELL	
29	VELOCITY m/s	
30	PRESSURE LOSS kg/cm ²	
31	FOULING RESISTANCE h • m ² • °C/kcal	
32	HEAT EXCHANGER DUTY – kcal/h	MTD-°C
33	SERVICE RATE – kcal/h • m ² • °K	CLEAN
34		
35	CONSTRUCTION OF ONE SHELL	
36	DESIGN PRESSURE kg/cm ² g	
37	TEST PRESSURE kg/cm ² g	
38	DESIGN TEMPERATURE (max/min) °C	/ /
39	GASKETS	
40	NOZZLES – INLET, TYPE mm	
41	– OUTLET, TYPE mm	
42	CORROSION ALLOWANCE mm	
43	CODE REQUIREMENTS	
44	TUBES NO. OD BWG (avg/min) LENGTH PITCH	
45	TUBESHEET	TUBE JOINT TYPE
46	SHELL EXP. JOINT	SHELL COVER (INT/REMOVABLE)
47	CHANNEL OR BONNET	CHANNEL COVER
48	BAFFLE – IMPINGEMENT	FLOATING HEAD
49	BAFFLES – CROSS TYPE	SPACING CUT
50	BAFFLE – LONG TYPE	TUBE SUPPORTS
51	WEIGHTS – EMPTY BUNDLE	FULL OF WATER
52	ACCESSORIES:	
53	NOTES:	
54		
55		

APPENDIX G-1 **STANDARD TOLERANCES FOR NOZZLES AND SUPPORTS – English Units**



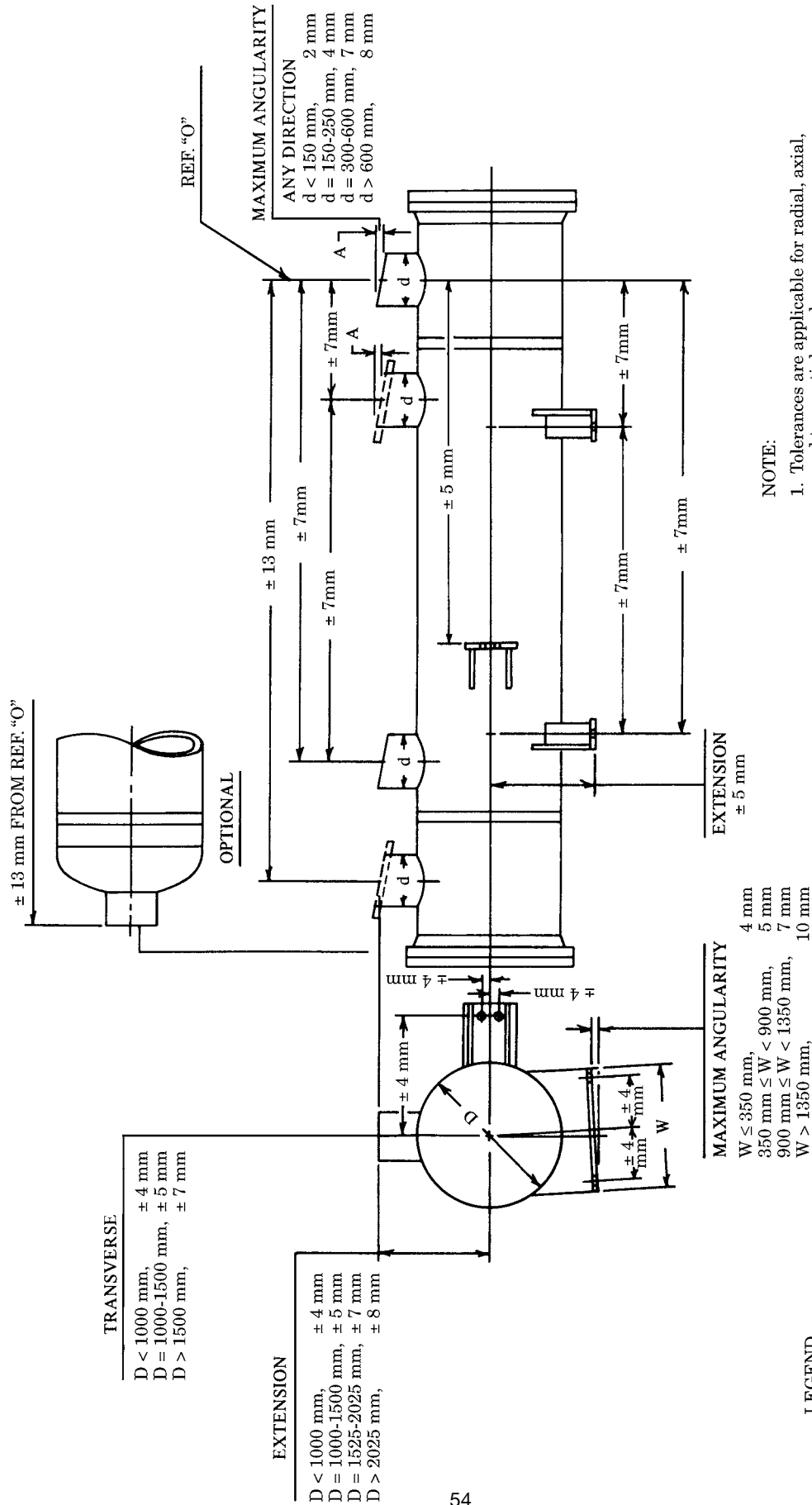
NOTE:

1. Tolerances are applicable for radial, axial, and tangential nozzles.
2. Tolerances for support maximum angularity, are applicable to saddle supports and lug supports
3. All diameters are nominal dimension

LEGEND

d = NOMINAL PIPE SIZE
D = SHELL O.D.
W = SUPPORT WIDTH

APPENDIX G-2 **STANDARD TOLERANCES FOR NOZZLES AND SUPPORTS – SI Units**



LEGEND

d = NOMINAL PIPE SIZE
D = SHELL O.D.
W = SUPPORT WIDTH

APPENDIX H

MECHANICAL CHARACTERISTICS OF STEEL TUBING

Nominal Tube OD (in)	External Surface per Ft of Tube (ft ²)	Nominal Thick- ness (in)	Nominal Tube ID (in)	Internal Area (in ²)	Ratio OD/ID	Constant C*	Wt/Ft (Steel) (lbm/ft)	Transverse Metal Area (in ²)	Moment of Inertia (in ⁴)
5/8	0.1636	0.150	0.325	0.0830	1.923	129.	0.762	0.2238	0.006942
		0.135	0.355	0.0990	1.761	154.	0.707	0.2078	0.006711
		0.125	0.375	0.1104	1.667	172.	0.668	0.1963	0.006519
		0.110	0.405	0.1288	1.543	201.	0.606	0.1780	0.006169
		0.105	0.415	0.1353	1.506	211.	0.584	0.1715	0.006034
		0.095	0.435	0.1486	1.437	232.	0.538	0.1582	0.005733
		0.085	0.455	0.1626	1.374	254.	0.491	0.1442	0.005386
		0.075	0.475	0.1772	1.316	276.	0.441	0.1296	0.004991
		0.065	0.495	0.1924	1.263	300.	0.389	0.1144	0.004543
		0.060	0.505	0.2003	1.238	312.	0.362	0.1065	0.004298
		0.055	0.515	0.2083	1.214	325.	0.335	0.0985	0.004037
		0.050	0.525	0.2165	1.190	338.	0.307	0.0903	0.003761
3/4	0.1963	0.150	0.450	0.1590	1.667	248.	0.962	0.2827	0.013519
		0.135	0.480	0.1810	1.562	282.	0.888	0.2608	0.012926
		0.125	0.500	0.1963	1.500	306.	0.835	0.2454	0.012464
		0.110	0.530	0.2206	1.415	344.	0.753	0.2212	0.011658
		0.105	0.540	0.2290	1.389	357.	0.724	0.2128	0.011358
		0.095	0.560	0.2463	1.339	384.	0.665	0.1955	0.010704
		0.085	0.580	0.2642	1.293	412.	0.604	0.1776	0.009977
		0.075	0.600	0.2827	1.250	441.	0.541	0.1590	0.009170
		0.065	0.620	0.3019	1.210	471.	0.476	0.1399	0.008278
		0.060	0.630	0.3117	1.190	486.	0.443	0.1301	0.007790
		0.055	0.640	0.3217	1.172	502.	0.409	0.1201	0.007296
		0.050	0.650	0.3318	1.154	518.	0.374	0.1100	0.006769
7/8	0.2291	0.150	0.575	0.2597	1.522	405.	1.163	0.3416	0.023408
		0.135	0.605	0.2875	1.446	448.	1.068	0.3138	0.022198
		0.125	0.625	0.3068	1.400	479.	1.002	0.2945	0.021284
		0.110	0.655	0.3370	1.336	526.	0.900	0.2644	0.019739
		0.105	0.665	0.3473	1.316	542.	0.864	0.2540	0.019174
		0.095	0.685	0.3685	1.277	575.	0.792	0.2328	0.017966
		0.085	0.705	0.3904	1.241	609.	0.718	0.2110	0.016648
		0.075	0.725	0.4128	1.207	644.	0.641	0.1885	0.015212
		0.065	0.745	0.4359	1.174	680.	0.563	0.1654	0.013653
		0.060	0.755	0.4477	1.159	698.	0.523	0.1536	0.012824
		0.055	0.765	0.4596	1.144	717.	0.482	0.1417	0.011962
		0.050	0.775	0.4717	1.129	736.	0.441	0.1296	0.011066
1	0.2618	0.150	0.700	0.3848	1.429	600.	1.363	0.4006	0.037301
		0.135	0.730	0.4185	1.370	653.	1.248	0.3669	0.035147
		0.125	0.750	0.4418	1.333	689.	1.169	0.3436	0.033556
		0.110	0.780	0.4778	1.282	745.	1.047	0.3076	0.030918
		0.105	0.790	0.4902	1.266	765.	1.005	0.2952	0.029968
		0.095	0.810	0.5153	1.235	804.	0.919	0.2701	0.027957
		0.085	0.830	0.5411	1.205	844.	0.831	0.2443	0.025791
		0.075	0.850	0.5674	1.176	885.	0.742	0.2179	0.023463
		0.065	0.870	0.5945	1.149	927.	0.650	0.1909	0.020965
		0.060	0.880	0.6082	1.136	949.	0.603	0.1772	0.019650
		0.055	0.890	0.6221	1.124	970.	0.556	0.1633	0.018289
		0.050	0.900	0.6362	1.111	992.	0.508	0.1492	0.016881
1 1/8	0.2945	0.150	0.825	0.5346	1.364	834.	1.563	0.4595	0.055889
		0.135	0.855	0.5741	1.316	896.	1.429	0.4199	0.052396
		0.125	0.875	0.6013	1.286	938.	1.336	0.3927	0.049854
		0.110	0.905	0.6433	1.243	1003.	1.194	0.3508	0.045701
		0.105	0.915	0.6576	1.230	1026.	1.145	0.3365	0.044221
		0.095	0.935	0.6866	1.203	1071.	1.046	0.3074	0.041113
		0.085	0.955	0.7163	1.178	1117.	0.945	0.2777	0.037798
		0.075	0.975	0.7466	1.154	1165.	0.842	0.2474	0.034269
		0.065	0.995	0.7776	1.131	1213.	0.737	0.2165	0.030516
		0.060	1.005	0.7933	1.119	1238.	0.683	0.2007	0.028552
		0.055	1.015	0.8091	1.108	1262.	0.629	0.1849	0.026529
		0.050	1.025	0.8252	1.098	1287.	0.575	0.1689	0.024445

*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 I

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)
$\frac{3}{8}$	0.0982	10	0.134	0.107	0.0090	3.505	14.	0.345	0.1015	0.000964
		11	0.120	0.135	0.0143	2.778	22.	0.327	0.0961	0.000954
		12	0.109	0.157	0.0194	2.389	30.	0.310	0.0911	0.000941
		13	0.095	0.185	0.0269	2.027	42.	0.284	0.0836	0.000913
		14	0.083	0.209	0.0343	1.794	54.	0.259	0.0761	0.000877
		15	0.072	0.231	0.0419	1.623	65.	0.233	0.0685	0.000831
		16	0.065	0.245	0.0471	1.531	74.	0.215	0.0633	0.000794
		17	0.058	0.259	0.0527	1.448	82.	0.197	0.0578	0.000750
		18	0.049	0.277	0.0603	1.354	94.	0.171	0.0502	0.000682
		19	0.042	0.291	0.0665	1.289	104.	0.150	0.0439	0.000619
		20	0.035	0.305	0.0731	1.230	114.	0.127	0.0374	0.000546
		21	0.032	0.311	0.0760	1.206	119.	0.117	0.0345	0.000512
		22	0.028	0.319	0.0799	1.176	125.	0.104	0.0305	0.000462
		23	0.025	0.325	0.0830	1.154	129.	0.093	0.0275	0.000431
		24	0.022	0.331	0.0860	1.133	134.	0.083	0.0244	0.000389
		25	0.020	0.335	0.0881	1.119	137.	0.076	0.0223	0.000359
$\frac{1}{2}$	0.1309	10	0.134	0.232	0.0423	2.155	66.	0.524	0.1541	0.002926
		11	0.120	0.260	0.0531	1.923	83.	0.487	0.1433	0.002844
		12	0.109	0.282	0.0625	1.773	97.	0.456	0.1339	0.002758
		13	0.095	0.310	0.0755	1.613	118.	0.411	0.1209	0.002615
		14	0.083	0.334	0.0876	1.497	137.	0.370	0.1087	0.002457
		15	0.072	0.356	0.0995	1.404	155.	0.329	0.0968	0.002280
		16	0.065	0.370	0.1075	1.351	168.	0.302	0.0888	0.002148
		17	0.058	0.384	0.1158	1.302	181.	0.274	0.0805	0.002001
		18	0.049	0.402	0.1269	1.244	198.	0.236	0.0694	0.001786
		19	0.042	0.416	0.1359	1.202	212.	0.206	0.0604	0.001598
		20	0.035	0.430	0.1452	1.163	227.	0.174	0.0511	0.001390
		21	0.032	0.436	0.1493	1.147	233.	0.160	0.0470	0.001294
		22	0.028	0.444	0.1548	1.126	242.	0.141	0.0415	0.001160
		23	0.025	0.450	0.1590	1.111	248.	0.127	0.0373	0.001075
		24	0.022	0.456	0.1633	1.096	255.	0.112	0.0330	0.000963
		25	0.020	0.460	0.1662	1.087	259.	0.103	0.0302	0.000886

*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

The above weights are for carbon steel with a density of 0.2833 lbm/in³.

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 C44300-1.088	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

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5/8	0.1636	10	0.134	0.357	0.1001	1.751	156.	0.703	0.2067	0.006693
		11	0.120	0.385	0.1164	1.623	182.	0.648	0.1904	0.006412
		12	0.109	0.407	0.1301	1.536	203.	0.601	0.1767	0.006143
		13	0.095	0.435	0.1486	1.437	232.	0.538	0.1582	0.005733
		14	0.083	0.459	0.1655	1.362	258.	0.481	0.1413	0.005311
		15	0.072	0.481	0.1817	1.299	283.	0.426	0.1251	0.004863
		16	0.065	0.495	0.1924	1.263	300.	0.389	0.1144	0.004543
		17	0.058	0.509	0.2035	1.228	317.	0.352	0.1033	0.004195
		18	0.049	0.527	0.2181	1.186	340.	0.302	0.0887	0.003704
		19	0.042	0.541	0.2299	1.155	359.	0.262	0.0769	0.003285
		20	0.035	0.555	0.2419	1.126	377.	0.221	0.0649	0.002833
		21	0.032	0.561	0.2472	1.114	386.	0.203	0.0596	0.002628
		22	0.028	0.569	0.2543	1.098	397.	0.179	0.0525	0.002345
		23	0.025	0.575	0.2597	1.087	405.	0.160	0.0471	0.002164
		24	0.022	0.581	0.2651	1.076	414.	0.142	0.0417	0.001932
		25	0.020	0.585	0.2688	1.068	419.	0.129	0.0380	0.001773
3/4	0.1963	10	0.134	0.482	0.1825	1.556	285.	0.882	0.2593	0.012882
		11	0.120	0.510	0.2043	1.471	319.	0.808	0.2375	0.012211
		12	0.109	0.532	0.2223	1.410	347.	0.747	0.2195	0.011600
		13	0.095	0.560	0.2463	1.339	384.	0.655	0.1955	0.010704
		14	0.083	0.584	0.2679	1.284	418.	0.592	0.1739	0.009822
		15	0.072	0.606	0.2884	1.238	450.	0.522	0.1534	0.008912
		16	0.065	0.620	0.3019	1.210	471.	0.476	0.1399	0.008278
		17	0.058	0.634	0.3157	1.183	492.	0.429	0.1261	0.007601
		18	0.049	0.652	0.3339	1.150	521.	0.367	0.1079	0.006661
		19	0.042	0.666	0.3484	1.126	543.	0.318	0.0934	0.005874
		20	0.035	0.680	0.3632	1.103	567.	0.268	0.0786	0.005036
		21	0.032	0.686	0.3696	1.093	577.	0.246	0.0722	0.004661
		22	0.028	0.694	0.3783	1.081	590.	0.216	0.0635	0.004145
		23	0.025	0.700	0.3848	1.071	600.	0.194	0.0569	0.003815
		24	0.022	0.706	0.3915	1.062	611.	0.171	0.0503	0.003398
		25	0.020	0.710	0.3959	1.056	618.	0.156	0.0459	0.003114
7/8	0.2291	10	0.134	0.607	0.2894	1.442	451.	1.061	0.3119	0.022110
		11	0.120	0.635	0.3167	1.378	494.	0.969	0.2846	0.020793
		12	0.109	0.657	0.3390	1.332	529.	0.893	0.2623	0.019628
		13	0.095	0.685	0.3685	1.277	575.	0.792	0.2328	0.017966
		14	0.083	0.709	0.3948	1.234	616.	0.703	0.2065	0.016370
		15	0.072	0.731	0.4197	1.197	655.	0.618	0.1816	0.014758
		16	0.065	0.745	0.4359	1.174	680.	0.563	0.1654	0.013653
		17	0.058	0.759	0.4525	1.153	706.	0.507	0.1489	0.012484
		18	0.049	0.777	0.4742	1.126	740.	0.433	0.1272	0.010882
		19	0.042	0.791	0.4914	1.106	767.	0.374	0.1099	0.009558
		20	0.035	0.805	0.5090	1.087	794.	0.314	0.0924	0.008161
		21	0.032	0.811	0.5166	1.079	806.	0.288	0.0847	0.007539
		22	0.028	0.819	0.5268	1.068	822.	0.254	0.0745	0.006689
		23	0.025	0.825	0.5346	1.061	834.	0.227	0.0668	0.006147
		24	0.022	0.831	0.5424	1.053	846.	0.201	0.0590	0.005465
		25	0.020	0.835	0.5476	1.048	854.	0.183	0.0537	0.005003

*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

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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	0.2618	10	0.134	0.732	0.4208	1.366	657.	1.241	0.3646	0.034994
		11	0.120	0.760	0.4536	1.316	708.	1.129	0.3318	0.032711
		12	0.109	0.782	0.4803	1.279	749.	1.038	0.3051	0.030731
		13	0.095	0.810	0.5153	1.235	804.	0.919	0.2701	0.027957
		14	0.083	0.834	0.5463	1.199	852.	0.814	0.2391	0.025339
		15	0.072	0.856	0.5755	1.168	898.	0.714	0.2099	0.022732
		16	0.065	0.870	0.5945	1.149	927.	0.650	0.1909	0.020965
		17	0.058	0.884	0.6138	1.131	957.	0.584	0.1716	0.019111
		18	0.049	0.902	0.6390	1.109	997.	0.498	0.1464	0.016594
		19	0.042	0.916	0.6590	1.092	1028.	0.430	0.1264	0.014529
		20	0.035	0.930	0.6793	1.075	1060.	0.361	0.1061	0.012367
		21	0.032	0.936	0.6881	1.068	1073.	0.331	0.0973	0.011411
		22	0.028	0.944	0.6999	1.059	1092.	0.291	0.0855	0.010106
		23	0.025	0.950	0.7088	1.053	1106.	0.260	0.0766	0.009275
		24	0.022	0.956	0.7178	1.046	1120.	0.230	0.0676	0.008236
		25	0.020	0.960	0.7238	1.042	1129.	0.209	0.0616	0.007533
1 ¹ / ₈	0.2945	10	0.134	0.857	0.5768	1.313	900.	1.420	0.4172	0.052150
		11	0.120	0.885	0.6151	1.271	960.	1.289	0.3789	0.048516
		12	0.109	0.907	0.6461	1.240	1008.	1.184	0.3479	0.045409
		13	0.095	0.935	0.6866	1.203	1071.	1.046	0.3074	0.041113
		14	0.083	0.959	0.7223	1.173	1127.	0.925	0.2717	0.037110
		15	0.072	0.981	0.7558	1.147	1179.	0.811	0.2382	0.033167
		16	0.065	0.995	0.7776	1.131	1213.	0.737	0.2165	0.030516
		17	0.058	1.009	0.7996	1.115	1247.	0.662	0.1944	0.027750
		18	0.049	1.027	0.8284	1.095	1292.	0.564	0.1656	0.024021
		19	0.042	1.041	0.8511	1.081	1328.	0.486	0.1429	0.020982
		20	0.035	1.005	0.8742	1.066	1364.	0.408	0.1199	0.017818
		21	0.032	1.061	0.8841	1.060	1379.	0.374	0.1099	0.016423
		22	0.028	1.069	0.8975	1.052	1400.	0.328	0.0965	0.014525
		23	0.025	1.075	0.9076	1.047	1416.	0.294	0.0864	0.013317
		24	0.022	1.081	0.9178	1.041	1431.	0.259	0.0762	0.011814
		25	0.020	1.085	0.9246	1.037	1442.	0.236	0.0694	0.010797
1 ¹ / ₄	0.3272	10	0.134	0.982	0.7574	1.273	1182.	1.599	0.4698	0.074195
		11	0.120	1.010	0.8012	1.238	1250.	1.450	0.4260	0.068762
		12	0.109	1.032	0.8365	1.211	1305.	1.330	0.3907	0.064164
		13	0.095	1.060	0.8825	1.179	1377.	1.173	0.3447	0.057871
		14	0.083	1.084	0.9229	1.153	1440.	1.035	0.3043	0.052064
		15	0.072	1.106	0.9607	1.130	1499.	0.907	0.2665	0.046392
		16	0.065	1.120	0.9852	1.116	1537.	0.823	0.2420	0.042602
		17	0.058	1.134	1.0100	1.102	1576.	0.739	0.2172	0.038667
		18	0.049	1.152	1.0423	1.085	1626.	0.629	0.1849	0.033389
		19	0.042	1.166	1.0678	1.072	1666.	0.542	0.1594	0.029110
		20	0.035	1.180	1.0936	1.059	1706.	0.455	0.1336	0.024673
		21	0.032	1.186	1.1047	1.054	1723.	0.417	0.1224	0.022722
		22	0.028	1.194	1.1197	1.047	1747.	0.366	0.1075	0.020075
		23	0.025	1.200	1.1310	1.042	1764.	0.327	0.0962	0.018390
		24	0.022	1.206	1.1423	1.036	1782.	0.289	0.0849	0.016301
		25	0.020	1.210	1.1499	1.033	1794.	0.263	0.0773	0.014891

*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

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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	0.3927	10	0.134	1.232	1.1921	1.218	1860.	1.957	0.5750	0.135418
		11	0.120	1.260	1.2469	1.190	1945.	1.770	0.5202	0.124781
		12	0.109	1.282	1.2908	1.170	2014.	1.621	0.4763	0.115911
		13	0.095	1.310	1.3478	1.145	2103.	1.427	0.4193	0.103943
		14	0.083	1.334	1.3977	1.124	2180.	1.257	0.3695	0.093054
		15	0.072	1.356	1.4441	1.106	2253.	1.099	0.3230	0.082543
		16	0.065	1.370	1.4741	1.095	2300.	0.997	0.2930	0.075582
		17	0.058	1.384	1.5044	1.084	2347.	0.894	0.2628	0.068405
		18	0.049	1.402	1.5438	1.070	2408.	0.760	0.2234	0.058851
		19	0.042	1.416	1.5748	1.059	2457.	0.655	0.1924	0.051161
		20	0.035	1.430	1.6061	1.049	2505.	0.548	0.1611	0.043240
		21	0.032	1.436	1.6196	1.045	2527.	0.502	0.1476	0.039774
		22	0.028	1.444	1.6377	1.039	2555.	0.441	0.1295	0.035083
		23	0.025	1.450	1.6513	1.034	2576.	0.394	0.1158	0.032100
		24	0.022	1.456	1.6650	1.030	2597.	0.347	0.1022	0.028419
		25	0.020	1.460	1.6742	1.027	2611.	0.316	0.0930	0.025939
2	0.5236	10	0.134	1.732	2.3561	1.155	3675.	2.673	0.7855	0.343663
		11	0.120	1.760	2.4328	1.136	3795.	2.412	0.7087	0.314398
		12	0.109	1.782	2.4940	1.122	3891.	2.203	0.6475	0.290403
		13	0.095	1.810	2.5730	1.105	4014.	1.935	0.5685	0.258551
		14	0.083	1.834	2.6417	1.091	4121.	1.701	0.4999	0.230047
		15	0.072	1.856	2.7055	1.078	4221.	1.484	0.4361	0.202917
		16	0.065	1.870	2.7465	1.070	4284.	1.345	0.3951	0.185142
		17	0.058	1.884	2.7877	1.062	4349.	1.204	0.3539	0.166964
		18	0.049	1.902	2.8413	1.052	4432.	1.022	0.3003	0.142989
		19	0.042	1.916	2.8832	1.044	4498.	0.879	0.2584	0.123865
		20	0.035	1.930	2.9255	1.036	4564.	0.735	0.2161	0.104317
		21	0.032	1.936	2.9437	1.033	4592.	0.673	0.1978	0.095808
		22	0.028	1.944	2.9681	1.029	4630.	0.590	0.1735	0.084339
		23	0.025	1.950	2.9865	1.026	4658.	0.528	0.1551	0.077050
		24	0.022	1.956	3.0049	1.022	4687.	0.465	0.1367	0.068111
		25	0.020	1.960	3.0172	1.020	4706.	0.423	0.1244	0.062105

*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 J

MODULUS OF ELASTICITY E OF MATERIALS FOR GIVEN TEMPERATURES

	Modulus of Elasticity E = Value Given x 10 ⁶ psi. for Temp. °F of												
	-325	-200	-100	70	200	300	400	500	600	700	800	900	1000
Stainless Steel (Austenitic)	30.3	29.7	29.1	28.3	27.6	27.0	26.5	25.8	25.3	24.8	24.1	23.5	22.8
Carbon Steel (C≤0.30%)	31.4	30.8	30.2	29.5	28.8	28.3	27.7	27.3	26.7	25.5	24.2	22.4	20.4
Carbon Steel (C>0.30%)	31.2	30.6	30.0	29.3	28.6	28.1	27.5	27.1	26.5	25.3	24.0	22.3	20.2
Nickel Alloy 20Cb-3 (Cr-Ni-Fe-Mo-Cu-Cb)	30.0	29.4	28.8	28.0	27.3	26.9	26.6	26.2	25.9	25.5	24.9	24.4	23.8
Nickel Alloy 400/405 (Ni-Cu)	27.8	27.3	26.8	26.0	25.4	25.0	24.7	24.3	24.1	23.7	23.1	22.6	22.1
Nickel Alloy 600 (Ni-Cr-Fe)	33.2	32.6	31.9	31.0	30.2	29.9	29.5	29.0	28.7	28.2	27.6	27.0	26.4
Nickel Alloy 625 (Ni-Cr-Mo-Cb)	32.1	31.5	30.9	30.0	29.3	28.8	28.5	28.1	27.8	27.3	26.7	26.1	25.5
Nickel Alloy 800/800H (Ni-Fe-Cr)	30.5	29.9	29.4	28.5	27.8	27.4	27.1	26.6	26.4	25.9	25.4	24.8	24.2
Nickel Alloy 825 (Ni-Fe-Cr-Mo-Cu)	30.0	29.4	28.8	28.0	27.3	26.9	26.6	26.2	25.9	25.5	24.9	24.4	23.8
Nickel Alloy C-276 (Ni-Mo-Cr)	31.9	31.7	30.7	29.8	29.1	28.6	28.3	27.9	27.6	27.1	26.5	25.9	25.3
Muntz Metals (Alloys 280/365)	15.9	15.6	15.4	15.0	14.6	14.4	14.1	13.8	13.4	12.8	—	—	—
Admiralty Metals-B/C/D (Alloys 443/444/445)	16.9	16.6	16.5	16.0	15.6	15.4	15.0	14.7	14.2	13.7	—	—	—
Naval Brass (Alloys 464/465/466/467)	15.9	15.6	15.4	15.0	14.6	14.4	14.1	13.8	13.4	12.8	—	—	—
Aluminum Bronze D (Alloy 614)	18.0	17.7	17.5	17.0	16.6	16.3	16.0	15.6	15.1	14.5	—	—	—
Aluminum-Nickel Bronze (Alloy 630)	18.5	18.2	18.0	17.5	17.1	16.8	16.5	16.1	15.6	15.0	—	—	—
Aluminum Brass-B (Alloy 687)	16.9	16.6	16.5	16.0	15.6	15.4	15.0	14.7	14.2	13.7	—	—	—
90-10 Copper Nickel (Alloy 706)	19.0	18.7	18.5	18.0	17.6	17.3	16.9	16.6	16.0	15.4	—	—	—
80-20 Copper-Nickel (Alloy 710)	21.2	20.8	20.6	20.0	19.5	19.2	18.8	18.4	17.8	17.1	—	—	—
70-30 Copper Nickel (Alloy 715)	23.3	22.9	22.7	22.0	21.5	21.1	20.7	20.2	19.6	18.8	—	—	—
Unalloyed Titanium Grades 1, 2, 3, & 7	—	—	—	15.5	15.0	14.6	14.0	13.3	12.6	11.9	11.2	—	—

Reference: ASME Section II, Part D

APPENDIX K

THERMAL CONDUCTIVITY OF MATERIAL FOR GIVEN TEMPERATURES

	Thermal Conductivity, K, Btu-ft/hr-ft ² -°F for Temp. °F of														
	70	100	150	200	250	300	350	400	450	500	550	600	650	700	750
Stainless Steel 304 (UNS S304XX)	8.6	8.7	9.0	9.3	9.6	9.8	10.1	10.4	10.6	10.9	11.1	11.3	11.6	11.8	12.0
Stainless Steel 316/317 (UNS S316XX/S317XX)	7.7	7.9	8.2	8.4	8.7	9.0	9.2	9.5	9.8	10.0	10.3	10.5	10.7	11.0	11.2
Stainless Steel 439 (UNS S43035)	–	–	12.3	12.5	12.6	12.8	12.9	13.1	13.3	13.4	13.6	13.8	13.9	14.1	–
Stainless Steel 29-4 (UNS S44735)	–	–	10.1	10.3	10.5	10.7	10.8	11.0	11.2	11.3	11.5	11.7	11.8	12.0	–
Carbon Steel	30.0	29.9	29.6	29.2	28.9	28.4	28.0	27.6	27.1	26.6	26.1	25.6	25.1	24.6	24.0
Nickel Alloy 400/405 (UNS N04400/N04405)	12.6	12.9	13.4	13.9	14.5	15.0	15.6	16.1	16.6	17.0	17.5	17.9	18.4	18.9	19.4
Nickel Alloy 600 (UNS N06600)	8.6	8.7	8.9	9.1	9.3	9.6	9.8	10.1	10.3	10.6	10.8	11.1	11.3	11.6	11.8
Nickel Alloy 20Cb-3 (UNS N08020)	–	6.9	7.2	7.5	7.8	8.0	8.3	8.6	8.8	9.1	9.4	9.7	10.0	10.2	10.5
Nickel Alloy AL6XN (UNS N08367)	–	–	6.7	7.1	7.4	7.7	8.0	8.3	8.6	8.9	9.2	9.5	9.8	10.1	–
Admiralty Metals - B/C/D (UNS C44300/C44400/C44500)	64.0	65.4	67.7	70.1	72.4	74.7	77.1	79.4	–	–	–	–	–	–	–
Aluminum Brass - B (UNS C68700)	58.0	59.4	61.6	63.9	66.2	68.5	70.7	73.0	–	–	–	–	–	–	–
90-10 Copper-Nickel (UNS C70600)	26.0	26.9	28.4	29.9	31.5	33.0	34.5	36.0	37.5	39.1	40.6	42.1	–	–	–
80-20 Copper-Nickel (UNS C71000)	21.0	21.5	22.3	23.1	23.9	24.7	25.5	26.3	27.1	27.9	28.7	29.5	30.3	31.1	–
70-30 Copper-Nickel (UNS C71500)	17.0	17.5	18.3	19.1	19.9	20.8	21.6	22.4	23.2	24.0	24.8	25.7	26.5	27.3	–
Unalloyed Titanium Grades 1/2/3/7 (UNS R50250/R50400/R50550/R52400)	12.68	12.52	12.25	12.00	11.85	11.72	11.60	11.45	11.35	11.29	11.23	11.20	11.17	11.15	11.18
	800	850	900	950	1000	1050	1100	1150	1200	1250	1300	1350	1400	1450	1500
Stainless Steel 304 (UNS S304XX)	12.2	12.5	12.7	12.9	13.2	13.4	13.6	13.8	14.0	14.3	14.5	14.7	14.9	15.1	15.3
Stainless Steel 316/317 (UNS S316XX/S317XX)	11.5	11.7	12.0	12.2	12.4	12.7	12.9	13.1	13.3	13.6	13.8	14.0	14.2	14.4	14.6
Stainless Steel 439 (UNS S43035)	–	–	–	–	–	–	–	–	–	–	–	–	–	–	–
Stainless Steel 29-4 (UNS S44735)	–	–	–	–	–	–	–	–	–	–	–	–	–	–	–
Carbon Steel	23.5	23.0	22.5	21.9	21.4	20.8	20.2	19.6	19.0	18.3	17.6	16.8	16.2	15.7	15.6
Nickel Alloy 400/405 (UNS N04400/N04405)	19.8	20.4	20.9	21.5	22.0	–	–	–	–	–	–	–	–	–	–
Nickel Alloy 600 (UNS N06600)	12.1	12.3	12.6	12.9	13.2	13.5	13.8	14.0	14.3	14.6	14.9	15.2	15.5	15.8	16.0
Nickel Alloy 20Cb-3 (UNS N08020)	10.8	11.0	11.3	11.6	11.9	–	–	–	–	–	–	–	–	–	–
Nickel Alloy AL6XN (UNS N08367)	–	–	–	–	–	–	–	–	–	–	–	–	–	–	–
Admiralty Metals - B/C/D (UNS C44300/C44400/C44500)	–	–	–	–	–	–	–	–	–	–	–	–	–	–	–
Aluminum Brass - B (UNS C68700)	–	–	–	–	–	–	–	–	–	–	–	–	–	–	–
90-10 Copper-Nickel (UNS C70600)	–	–	–	–	–	–	–	–	–	–	–	–	–	–	–
80-20 Copper-Nickel (UNS C71000)	–	–	–	–	–	–	–	–	–	–	–	–	–	–	–
70-30 Copper-Nickel (UNS C71500)	–	–	–	–	–	–	–	–	–	–	–	–	–	–	–
Unalloyed Titanium Grades 1/2/3/7 (UNS R50250/R50400/R50550/R52400)	11.20	11.23	11.30	11.36	11.43	11.51	11.58	–	–	–	–	–	–	–	–

References: ASME Section II, Part D, except for the following materials: • Aluminum Brass-B: Copper Development Association
 • Admiralty and all Copper-Nickels: @ 70 °F, Scovill; @ maximum temperature, HTRI • SS439, SS29-4, and AL6XN: Allegheny Ludlum

APPENDIX L

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	
kelvin	(SI)	K	Note 5.
degree Celsius	(SI)	°C	Note 5.
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
s ³			
megawatt		MW	
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.
5. For temperature interval, 1K = 1°C exactly.

APPENDIX L – Continued

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 (SI)	4.535924	$\times 10^{-1}$	kg

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

TEMPERATURE			
K = (°F + 459.67)/1.8		K = (°C + 273.15)	
°C = (°F – 32)/1.8		°C = (K – 273.15)	
°F = 1.8 °C + 32		°F = 1.8 K – 459.67	

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

APPENDIX L – Continued

POWER (ENERGY/TIME)			
MULTIPLY	BY	TO OBTAIN	
Btu/hr	2.930711×10^{-1}	W	(SI)
Btu/hr	2.930711×10^{-7}	MW	
Btu/hr	2.519958×10^{-1}	kcal/h	

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	
ftH ₂ O (39.2°F)	2.98898×10^3	Pa	(SI)
ftH ₂ O (39.2°F)	2.98898	kPa	
ftH ₂ O (39.2°F)	3.047915×10^2	kgf/m ²	

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	

APPENDIX L – Continued

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 • °C)	(SI)
Btu/(lbm • °F)	4.186800	kJ/(kg • °C)	
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 • °C)	(SI)
Btu • in/(hr • ft ² • °F)	1.240137×10^{-1}	kcal • m/(h • m ² • °C)	
Btu • ft/(hr • ft ² • °F)	1.730735	W/(m • °C)	(SI)
Btu • ft/(hr • ft ² • °F)	1.488164	kcal • m/(h • m ² • °C)	

DYNAMIC VISCOSITY (MASS/TIME-LENGTH OR FORCE-TIME/AREA)

MULTIPLY	BY	TO OBTAIN	
cp	1.000000×10^{-3}	Pa • s	(SI)
cp	1.000000	mPa • s	
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 • °C)	(SI)
Btu/(hr • ft ² • °F)	4.882428	kcal/(h • m ² • °C)	

FOULING RESISTANCE (TIME-AREA-TEMPERATURE/ENERGY)

MULTIPLY	BY	TO OBTAIN	
hr • ft ² • °F/Btu	1.761102×10^{-1}	m ² • °C/W	(SI)
hr • ft ² • °F/Btu	2.048161×10^{-1}	h • m ² • °C/kcal	

APPENDIX M

TYPICAL SHELL AND CHANNEL ARRANGEMENTS

M1.0 SCOPE

This appendix provides a detailed expression for accurately describing the construction of a heat exchanger.

M2.0 GENERAL EXPRESSION

V W X Y Z

This expression breaks down as follows:

V represents the front tube side closure
W represents the front tubesheet arrangement
X represents the shell side arrangement
Y represents the rear tubesheet arrangement
Z represents the rear tube side closure

M3.0 PARAMETERS

M3.1 V (see Figure M-1)

V = **C** for a channel with a bolted **C**over

V = **B** for a channel with an integral (welded) cover (**B**onnet)

V = **R** for a channel with a **R**educer

M3.2 W (see Figure M-2)

W = **1** for a stationary tubesheet which is gasketed (bolted) on both sides

W = **2** for a stationary tubesheet which is integral (welded) on the tube side and gasketed (bolted) on the shell side

W = **3** for a stationary tubesheet which is gasketed (bolted) on the tube side and integral (welded) on the shell side

W = **4** for a stationary tubesheet which is integral (welded) on both sides

M3.3 X (see Figure M-3)

X = **E** for a **O**ne-pass shell

X = **T** for a **T**wo-pass shell

X = **S** for a **S**plit-flow shell

X = **D** for a **D**ivided-flow shell

X = **X** for a **C**ross-flow shell

X = **K** for a **K**ettle-type shell

M3.4 Y (see Figures M-2 and M-4)

Y = **1,2,3,4** for the stationary tubesheet arrangements described in paragraph M3.2 above

Y = **5** for a pull-through floating tubesheet which is extended for bolting and gasketed on the tube side

Y = **6** for a floating tubesheet which is not extended for bolting (sandwiched between split backing ring and floating head flange) and gasketed on the tube side only

Y = **7** for a tubesheet which is packed on the shell side and integral (welded) on the tube side

Y = **8** for a tubesheet which is packed on both sides with a lantern ring

M3.5 Z (see Figures M-1 and M-5)

Z = **C, B, R** for the tube side closures described in paragraph M3.1 above

Z = **F** for a **F**loating head assembly

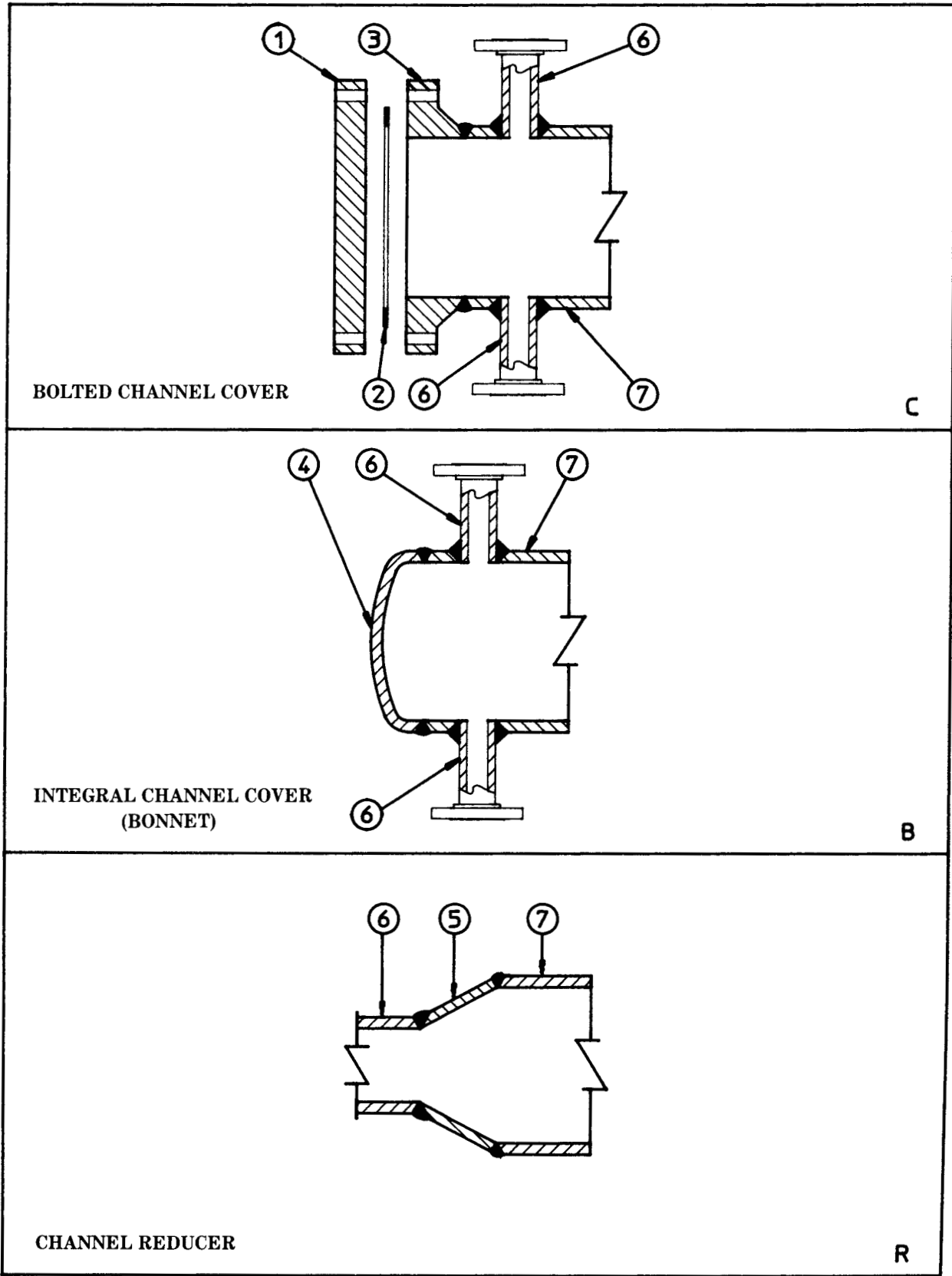
Z = **P** for a **P**acked channel cover assembly

Z = **U** for **U**-tubes

APPENDIX M (cont'd)
TYPICAL SHELL AND CHANNEL ARRANGEMENTS

M4.0 NOMENCLATURE

1. CHANNEL COVER
2. CHANNEL COVER GASKET
3. CHANNEL COVER FLANGE
4. CHANNEL HEAD
5. CHANNEL REDUCER
6. CHANNEL NOZZLE (FLANGED OR WELD END)
7. CHANNEL CYLINDER
8. CHANNEL TUBESHEET FLANGE
9. CHANNEL TUBESHEET GASKET
10. STATIONARY TUBESHEET
11. SHELL TUBESHEET GASKET
12. SHELL TUBESHEET FLANGE
13. SHELL CYLINDER
14. SHELL LONGITUDINAL BAFFLE
15. SHELL NOZZLE (FLANGED OR WELD END)
16. SHELL FRONT CYLINDER
17. SHELL FRONT REDUCER
18. SHELL REAR REDUCER
19. SHELL REAR CYLINDER
20. SHELL COVER FLANGE
21. SHELL COVER GASKET
22. SHELL HEAD FLANGE
23. SHELL HEAD
24. FLOATING TUBESHEET SPLIT BACKING RING
25. FLOATING TUBESHEET
26. FLOATING TUBESHEET GASKET
27. FLOATING HEAD FLANGE
28. FLOATING HEAD
29. SHELL PACKING FLANGE
30. SHELL PACKING
31. SHELL PACKING GLAND
32. LANTERN RING
33. CHANNEL PACKING
34. CHANNEL PACKING FLANGE
35. PACKED TUBESHEET
36. PACKED CHANNEL CYLINDER
37. PACKED CHANNEL COVER FLANGE
38. PACKED CHANNEL SPLIT SHEAR RING
39. PACKED CHANNEL COVER GASKET
40. PACKED CHANNEL COVER
41. TUBES
42. TIE RODS AND SPACERS
43. BAFFLES AND SUPPORT PLATES
44. PASS PARTITION PLATE



**Figure M-1
Tube Side Closures**

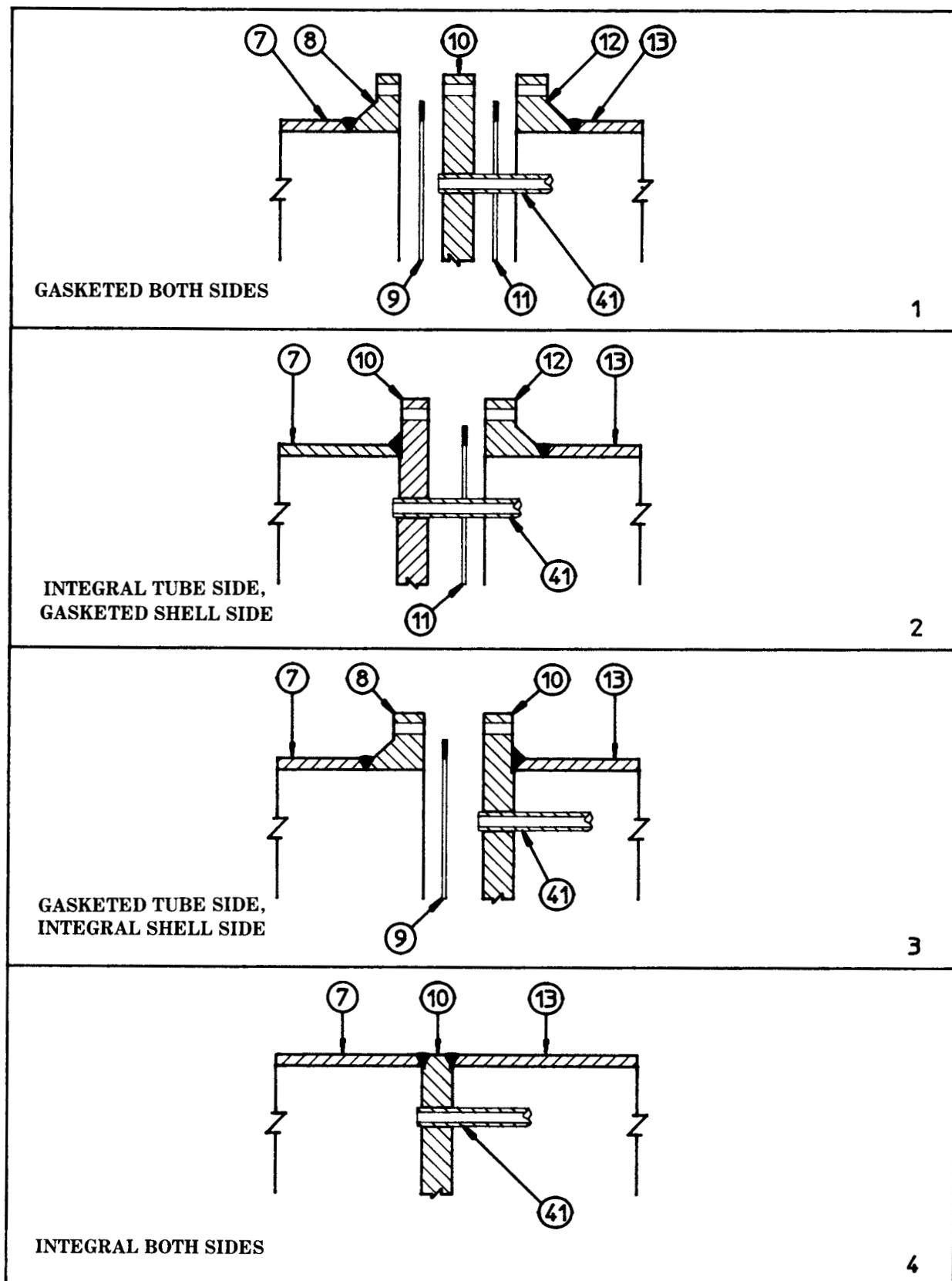


Figure M-2
Stationary Tubesheet Arrangements

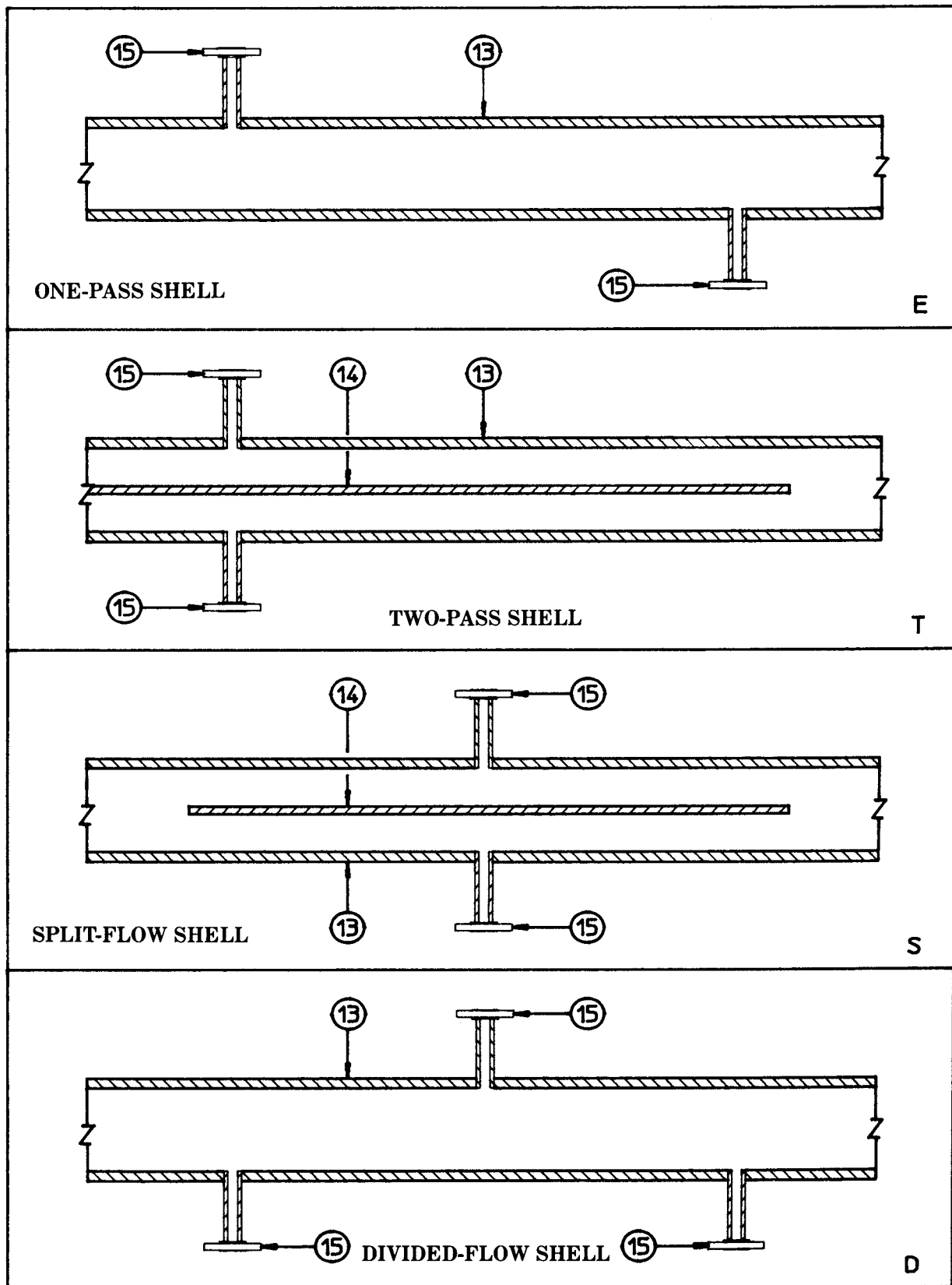


Figure M-3
Shell Side Arrangements

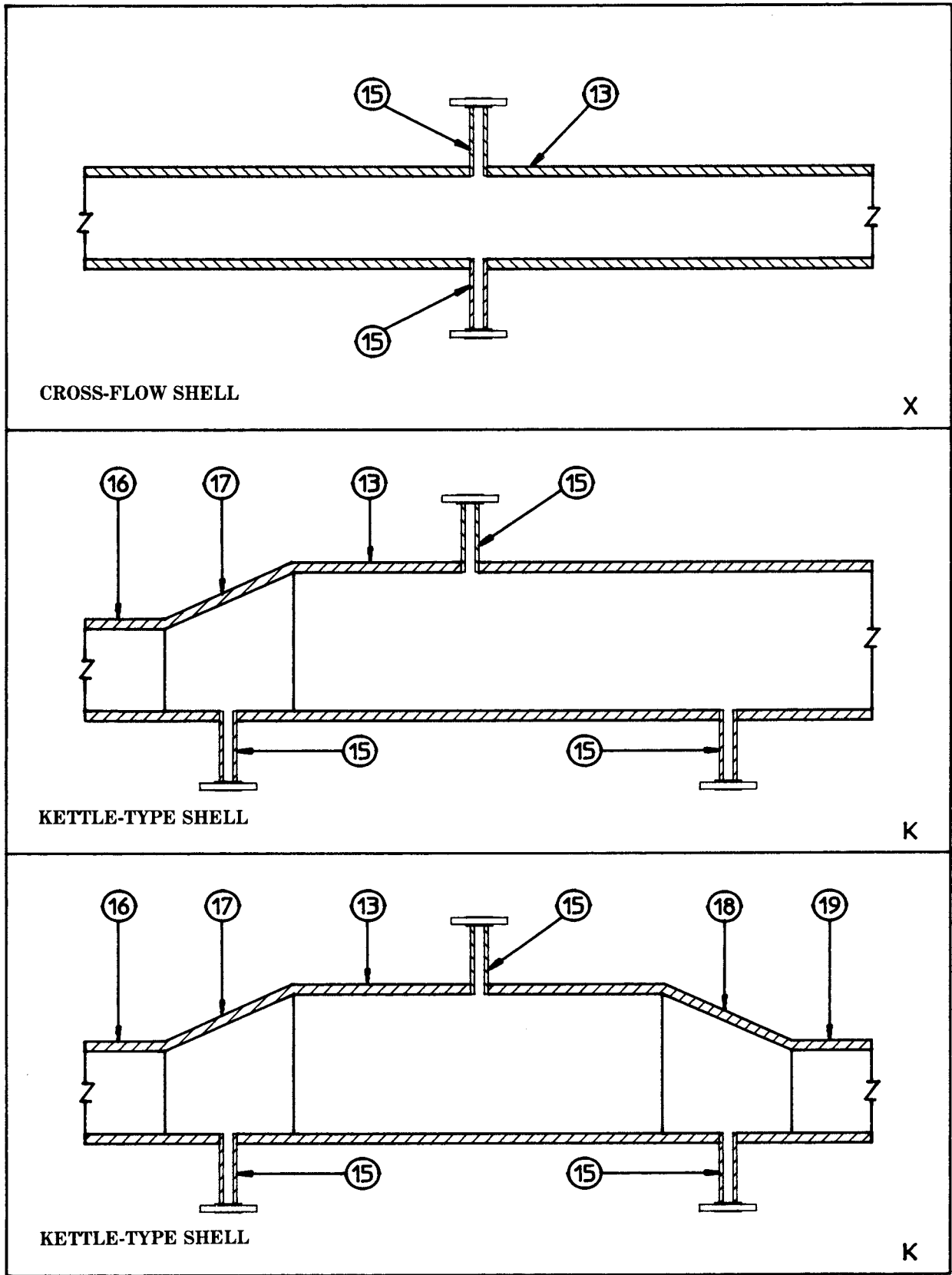


Figure M-3 (cont'd.)
Shell Side Arrangements

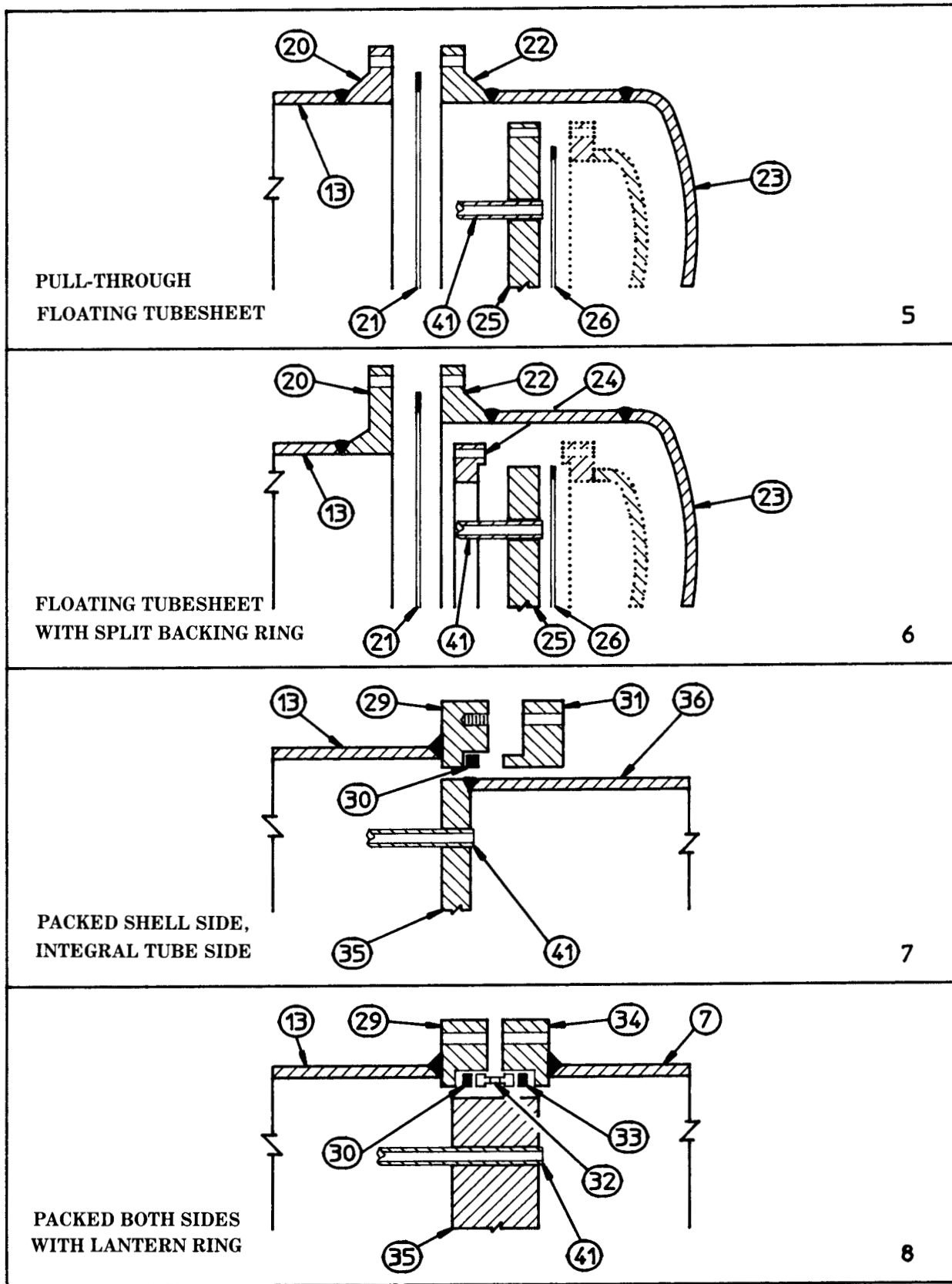


Figure M-4
Floating and Packed Tubesheet Arrangements (Rear Only)

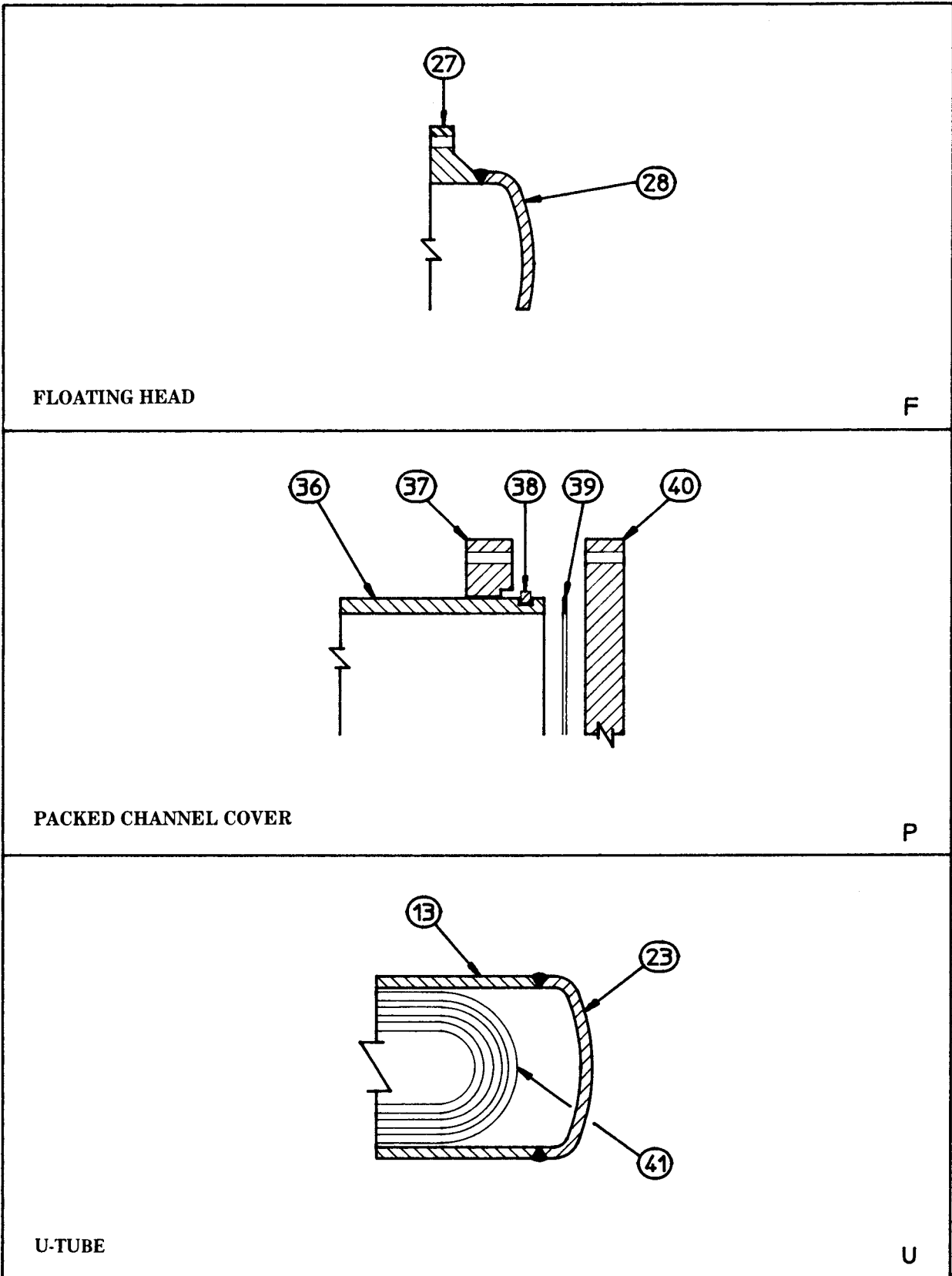
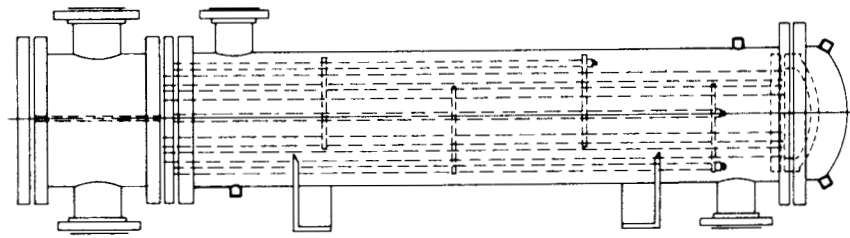
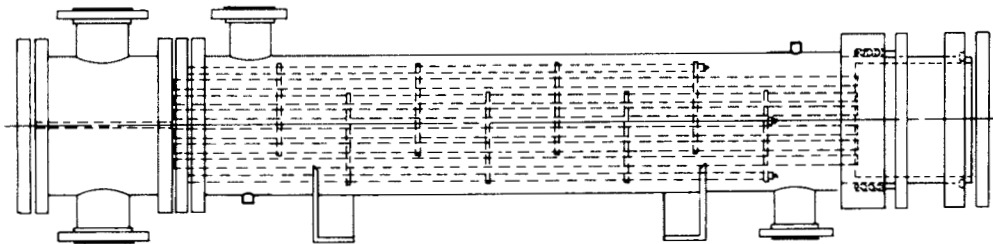


Figure M-5
Tube Side Closures (Rear Only)



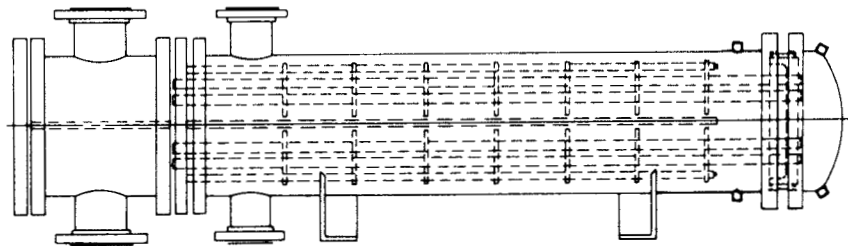
C1E5F

Exchanger with a one-pass shell, channel cover, pull-through floating tubesheet, and floating head.



C1E7P

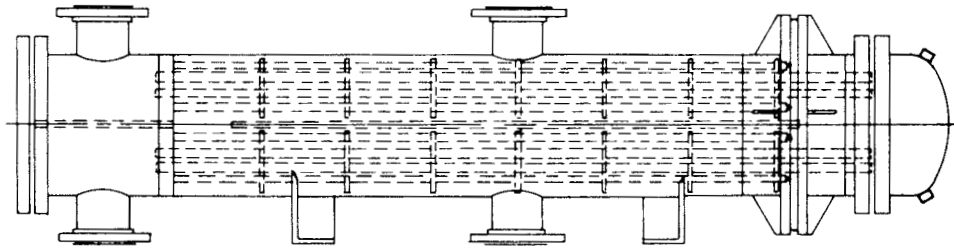
Exchanger with a one-pass shell, channel cover on front end, packed tubesheet, and packed channel cover on rear end.



C1T8B

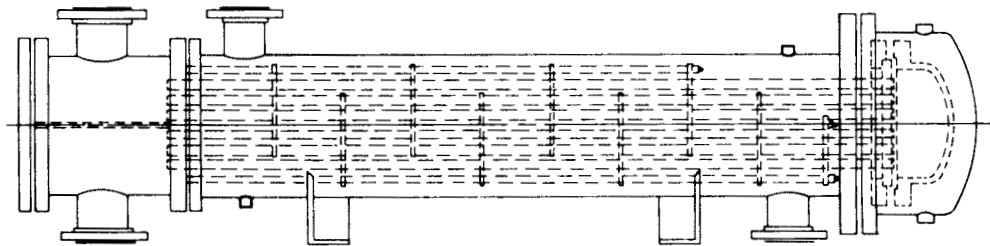
Exchanger with a two-pass shell, channel cover on front end, packed tubesheet with lantern ring, and bonnet on rear end.

Figure M-6
Typical Arrangements



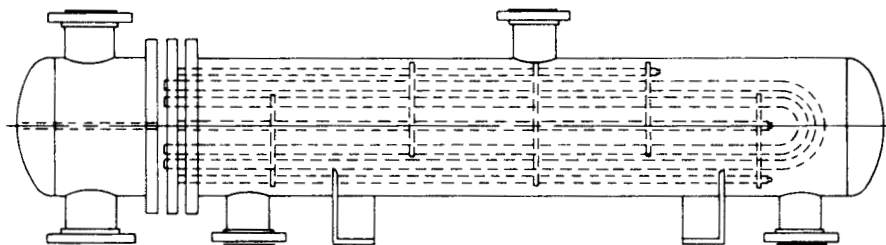
C4S3B

Exchanger with a split-flow shell, bellows expansion joint, channel cover on front end, and bonnet on rear end.



C2E6F

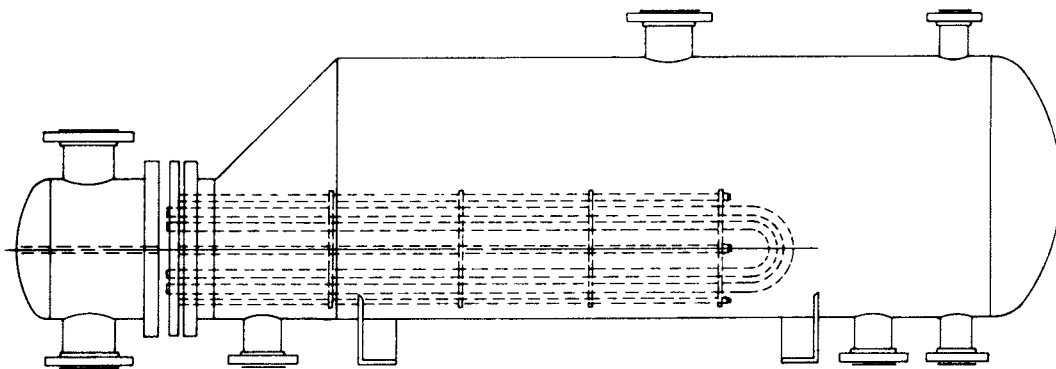
Exchanger with a one-pass shell, channel cover, floating tubesheet with split backing ring, and floating head.



B1DU

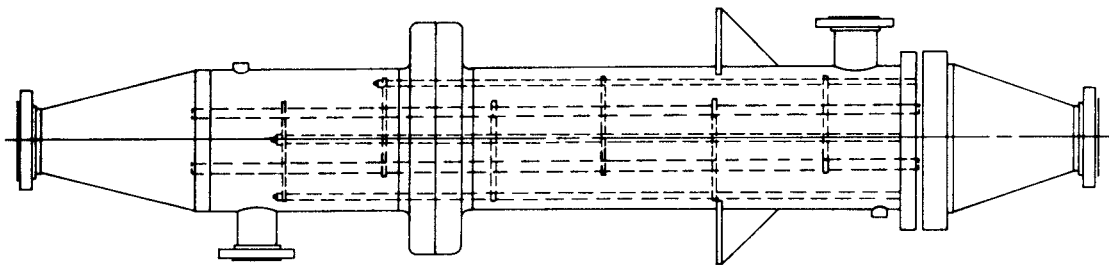
Exchanger with a divided-flow shell, bonnet, and U-tubes.

**Figure M-6 (cont'd.)
Typical Arrangements**



B1KU

Kettle with bonnet and U-tubes.

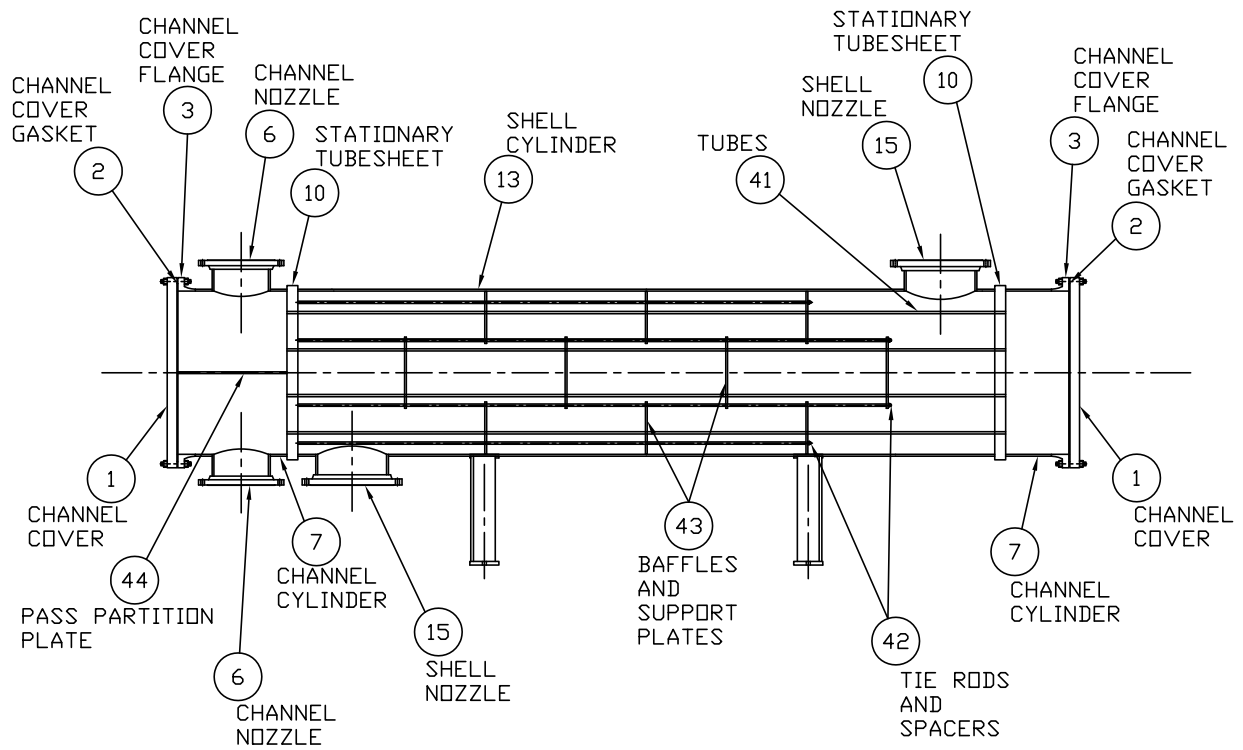


R4E3R

Exchanger with a one-pass shell, flanged and flued expansion joint, and channel reducers on both sides.

**Figure M-6 (cont'd.)
Typical Arrangements**

TYPICAL CROSS-SECTION DETAIL



TWO-PASS TUBE SIDE, ONE-PASS SHELL SIDE, TYPE C4E4C
WITH BOLTED CHANNEL COVERS AND INTEGRAL STATIONARY TUBESHEETS.

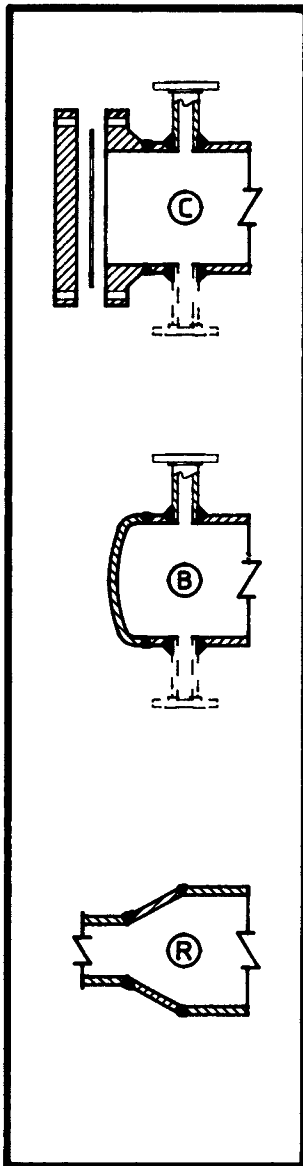
VARIOUS OPTIONAL CHANNEL, SHELL AND TUBESHEET ARRANGEMENTS
(WITH NOMENCLATURE ITEM NUMBERS FROM SECTION M4.0) ARE
DETAILED ON THE PAGES THAT FOLLOW.

Figure M-7
Typical Cross Section Detail

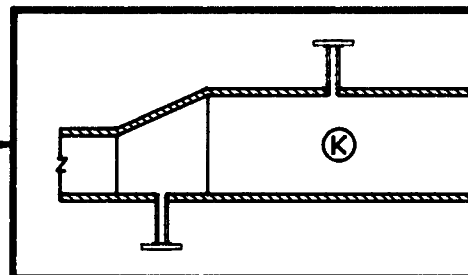
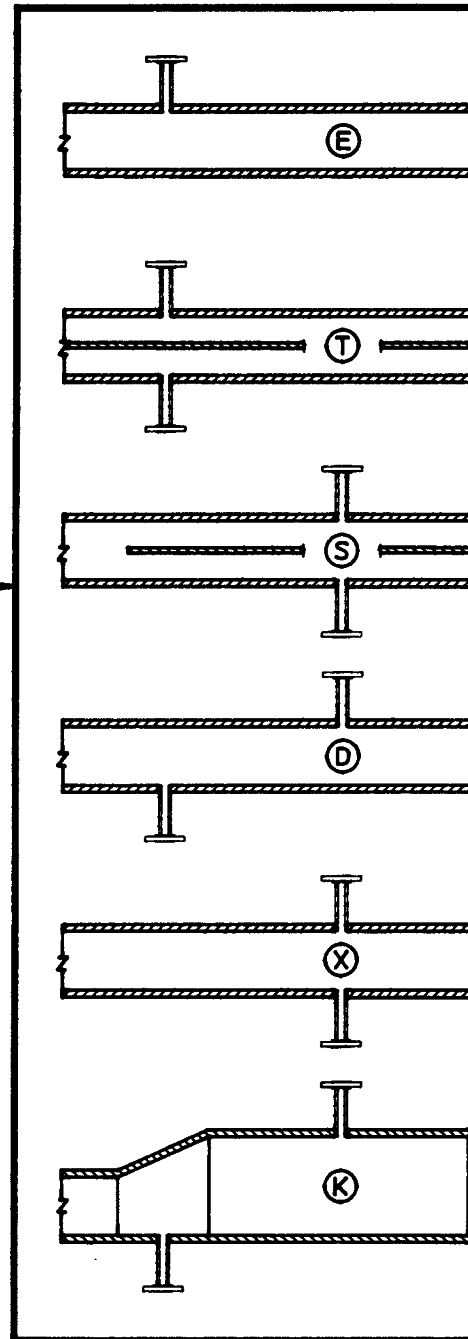
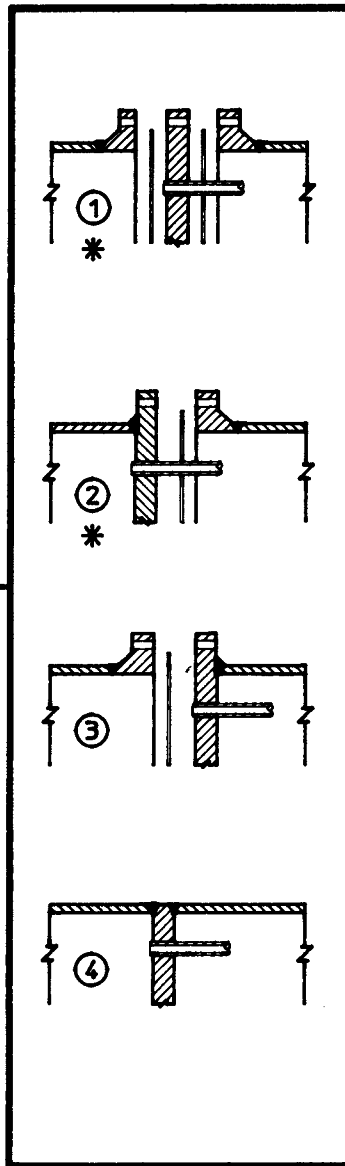
TYPICAL SHELL AND CHANNEL AND
STATIONARY TUBESHEET (FRONT & R

SHELLSIDE ARRANGEMENT

FRONT TUBESIDE
CLOSURES

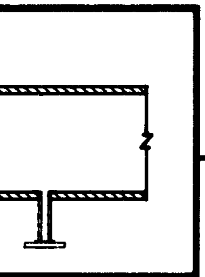
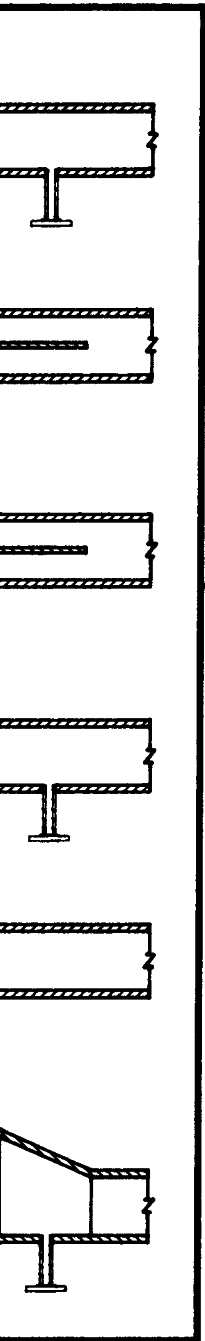


FRONT TUBESHEET
ARRANGEMENTS

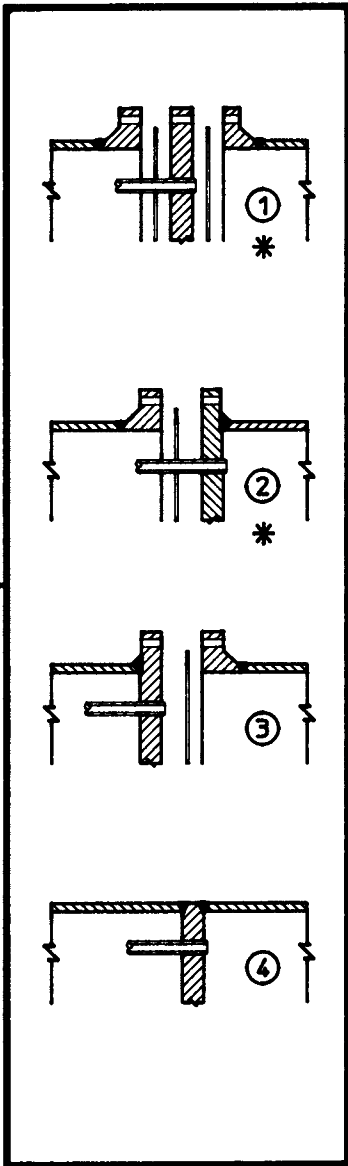


***CAUTION:** IF THESE CONFIGURATIONS ARE USED, THE
SHELL TUBESHEET GASKET CANNOT BE
REPLACED WITHOUT REMOVING THE TUBES

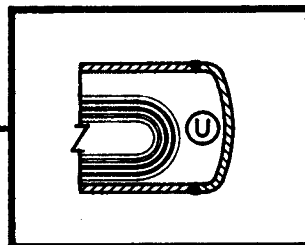
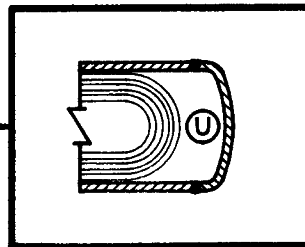
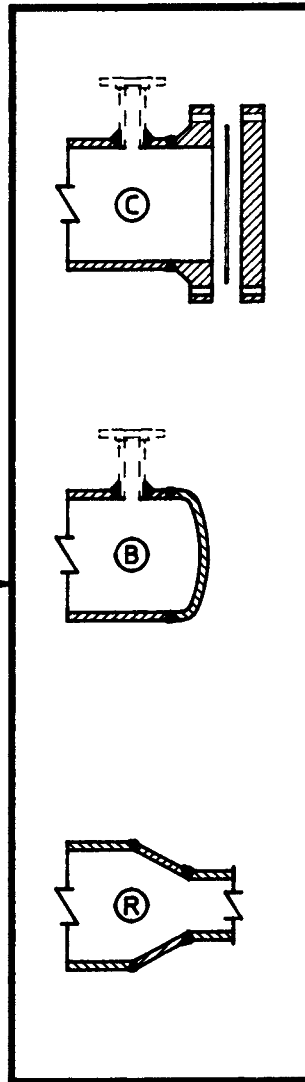
ARRANGEMENTS
(REAR) OR "U" TUBE
NTS



REAR TUBESHEET
ARRANGEMENTS

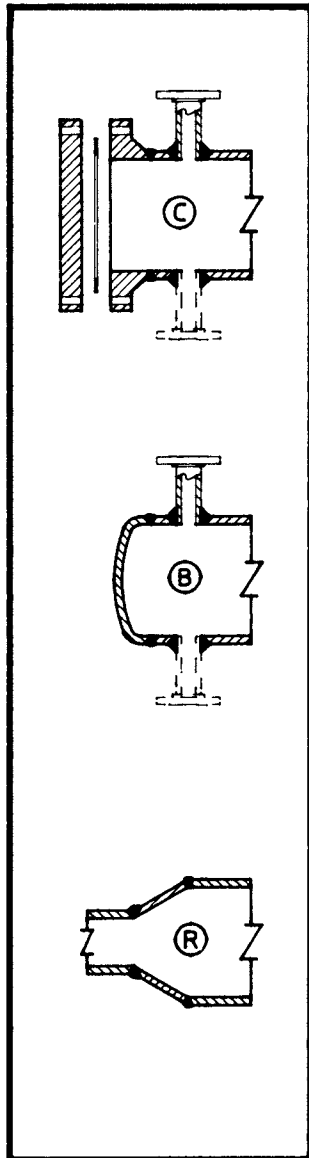


REAR TUBESIDE
CLOSURES

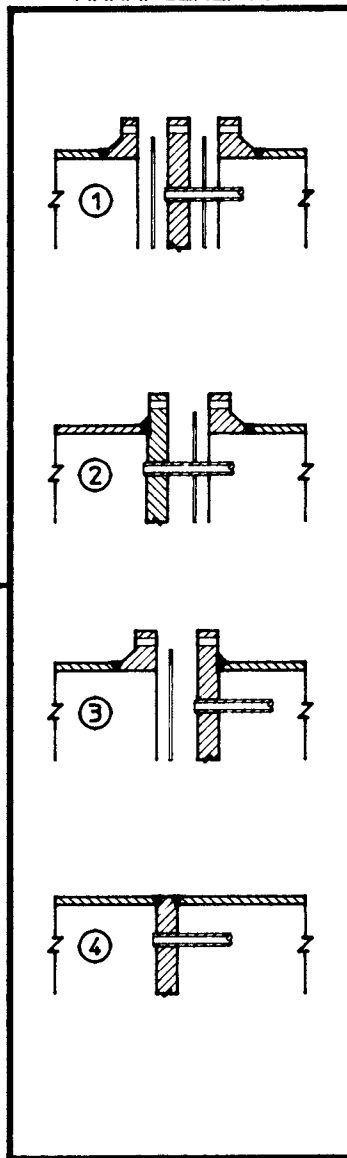


TYPICAL SHELL AND CHANNEL ARRANGEMENTS
STATIONARY TUBESHEET (FRONT) WITH FLOATING CLOSURES

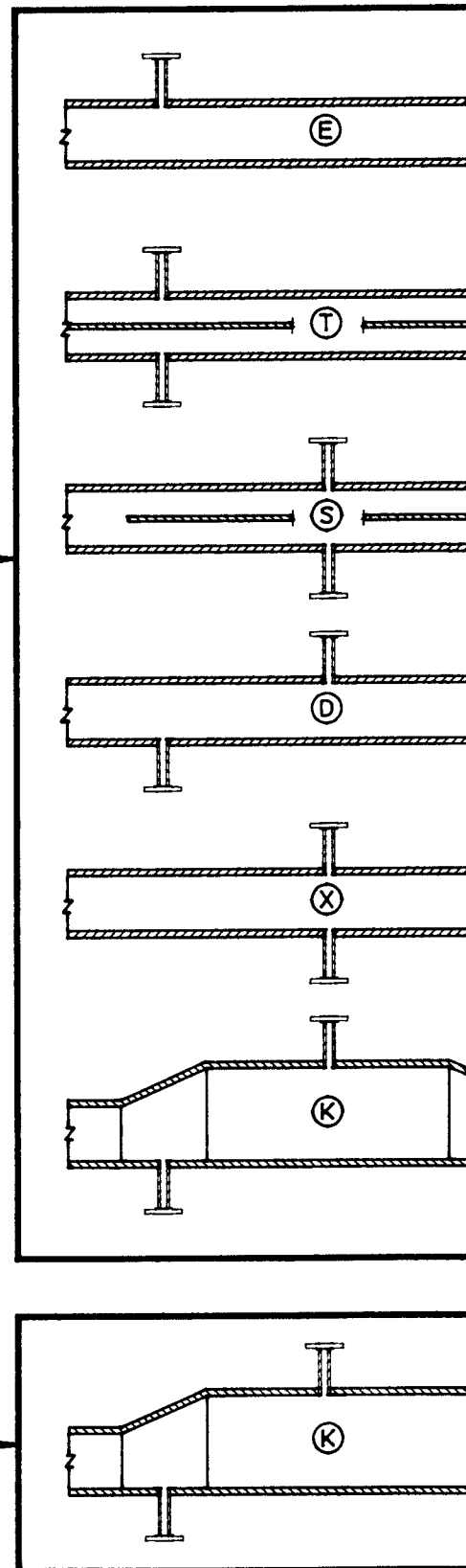
FRONT TUBESIDE
CLOSURES



FRONT TUBESHEET
ARRANGEMENTS



SHELLSIDE ARRANGEMENTS

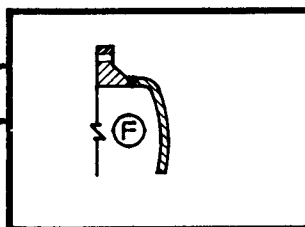
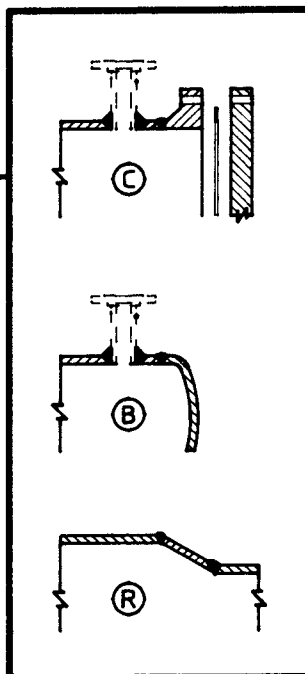
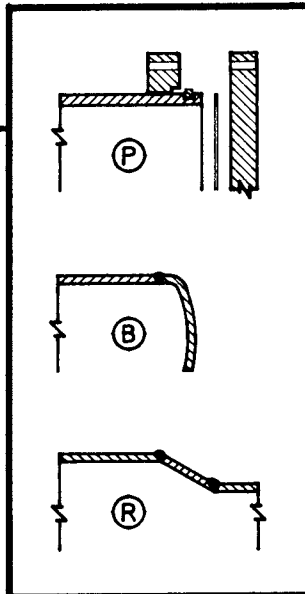
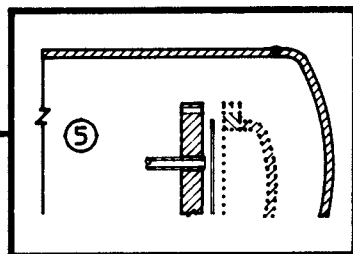
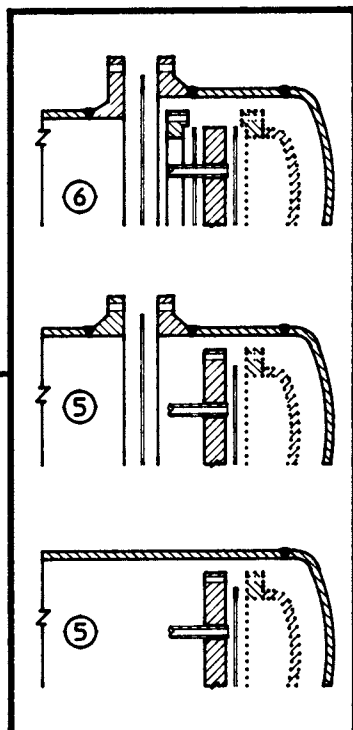
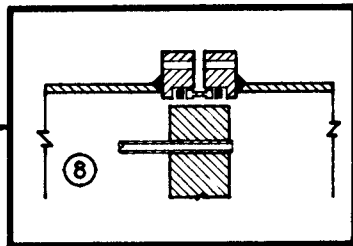
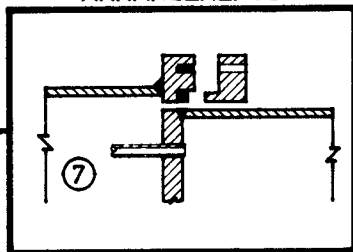
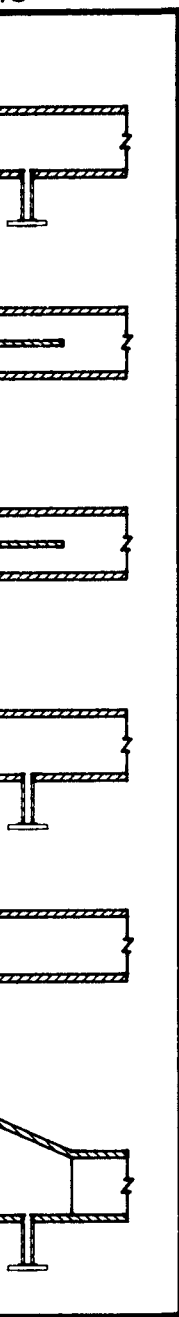


ARRANGEMENTS FOR PACKED TUBESHEET (REAR)

TS

REAR TUBESHEET ARRANGEMENTS

REAR TUBESIDE CLOSURES



APPENDIX N

TROUBLESHOOTING GUIDE

This troubleshooting guide has been prepared to assist operators of power plant heat exchangers. 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 heat exchanger manufacturer; however, these items do affect operation and must be considered by operators.

Please submit all questions and inquiries to the HEI at hei@heatexchange.org, or visit the HEI website at www.heatexchange.org.

Symptoms	Possible Causes	Possible Solutions
Gasket Leaks	Improper bolt torque sequence at installation	Replace gasket and consult operating manual for proper sequence of bolt tightening
	Replacement gasket is not compatible with original design	Replace gasket with compatible gasket shown in operating manual
	Gasket surface has been eroded due to previous leaks	Remove cover, repair gasket surface by welding and/or machining and replace gasket
	Bolting has galled due to improper tightening and/or lubrication	Replace gasket and bolting
Corrosion	High fluid velocities sweep away protective oxide layer or coating	Reduce velocity, reduce turbulence
	Dissimilar metals in the presence of an electrolytic solution	Provide epoxy lining, or coating, provide cathodic protection (Reference section 6.3 in the <i>HEI Standards for Shell & Tube Heat Exchangers</i> .)
	Damage to the oxide layer or to the protective coating	Consult plant chemist
	Cavitation	Eliminate cavitation
	Acidity or oxygen building up in cracks and crevices	Remove crevice, consult plant chemist
Tube Leak(s) (Exhibited by increased flow, pressure losses, and contamination on the lower pressure side. In severe cases, tube leaks will result in relief valve or rupture disk activation.)	Tube to tubesheet joint failures	Inspect overlaid tubesheets for cracking or separation. Inspect tube to tubesheet joint for damage. If damaged, contact supplier for repair procedure
	Damage from corrosion and/or erosion	Check operations to prevent any possible temperature shocks
	Mechanical damage	Inspect tubes for corrosion and/or erosion damage. Consult the plant chemist and the supplier for possible repairs which may include retubing, replacement, sleeving, or linings
		Inspect the tube and shell sides for foreign or loose internal parts. Remove any foreign parts and repair damage. Consult with the supplier if any loose internal parts are found

APPENDIX N

TROUBLESHOOTING GUIDE

Symptoms	Possible Causes	Possible Solutions
Tube Leak(s) continued	Vibration	Identify all tube leak locations on the tube layout drawing. An effort should be made to determine the location of the leak from the front tubesheet. The supplier of the equipment should be notified and provided all pertinent operating conditions along with the plug map identifying leak locations Possible solutions may involve the following: • Tube plugging (after first using capture rods to stabilize loose tube ends.) • Insurance plugging • Staking of tube bundles where possible • Limiting certain modes of operation
Pass Partition Leaks (Exhibited by reduced tubeside temperature rise and higher TTD.)	Pass partition gasket failure	Consult supplier for appropriate replacement
	Pass partition nut failure due to cycling operation	Consult supplier over stud and nut material selections, and consider tack welding replacement nuts, self locking nuts, or tension controlled washers
	Pass partition cracking due to cycling	Consult supplier over possible repairs
	Water hammer or abnormal operating conditions	Make repairs as required and review operating procedures
	Erosion damage from tubeside inlet nozzle	Consult supplier and review pass partition plate material

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