

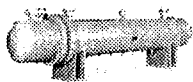
Heat Exchanger Design Course

By: Eng. Mohamed A. Ebeid

HEAT EXCHANGER DESIGN COURSE

By: Eng. Mohamed A. Ebeid

12/1/17



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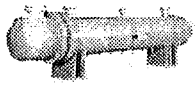
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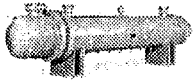
ATTACHEMENTS



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PART 1



I. INTRODUCTION

1. WHAT IS A HEAT EXCHANGER?

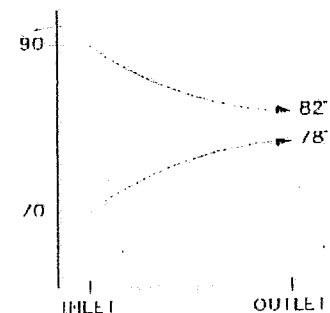
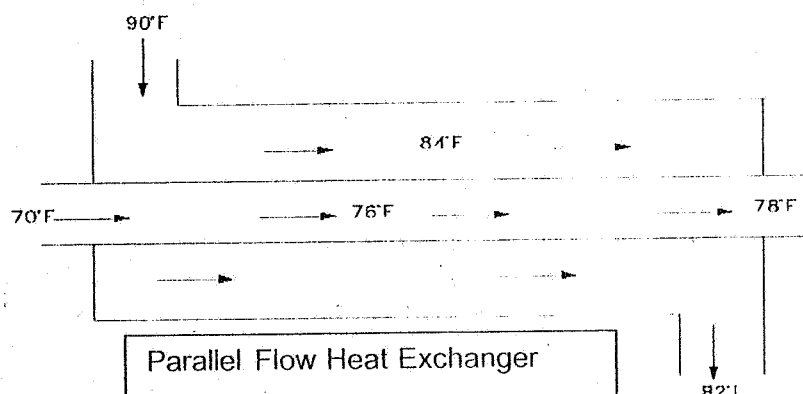
It is equipment that transfers heat energy from one fluid (or more) to another fluid (or more) without mixing it.

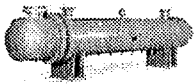
Example: Automotive radiators are a common example, where heat from the hot engine water is pumped through the radiator, while air is blown through the radiator fins. The hot engine water's heat energy is transferred to the air, thus keeping the water at the right temperature, to keep the engine from overheating. Most automotive radiators are liquid-to-air heat exchangers. Other types of heat exchangers are common in everyday equipment such as boilers, furnaces, refrigerators and air conditioning systems. In fact, every air conditioning system and refrigeration system has at least two heat exchangers one for the cooling side, and one to expel the heat. There are three main types of heat exchangers, defined by their construction or body types: shell and tube, plate, and air-cooled.

2. CLASSIFICATION OF A HEAT EXCHANGER ACCORDING TO FLOW

Because heat exchangers come in so many shapes, sizes, makes, and models, they are categorized according to common characteristics. One common characteristic that can be used to categorize them is the direction of flow the two fluids have relative to each other. The three categories are parallel flow, counter flow and cross flow.

Parallel flow exists when both the tube side fluid and the shell side fluid flow in the same direction. In this case, the two fluids enter the heat exchanger from the same end with a large temperature difference. As the fluids transfer heat, hotter to cooler, the temperatures of the two fluids approach each other. Note that the hottest cold-fluid temperature is always less than the coldest hot-fluid temperature.

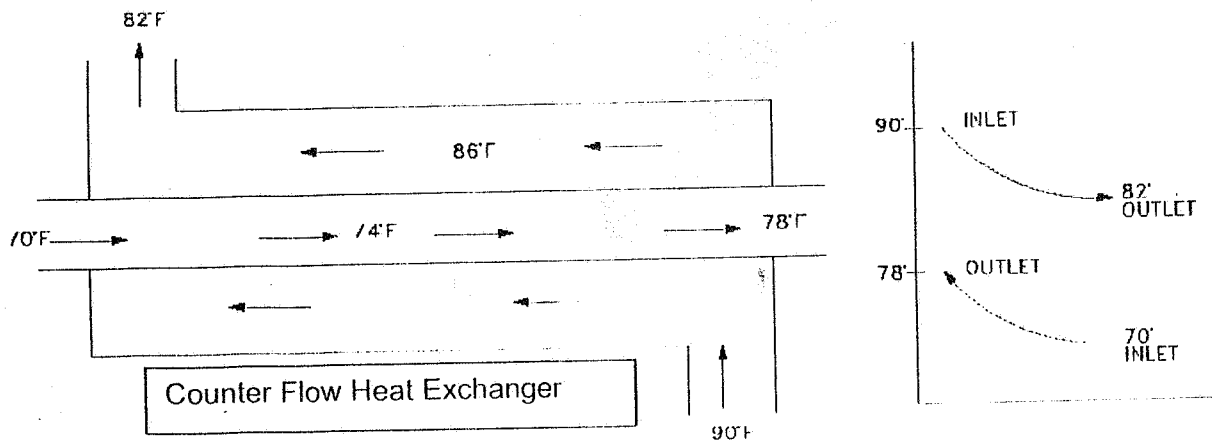




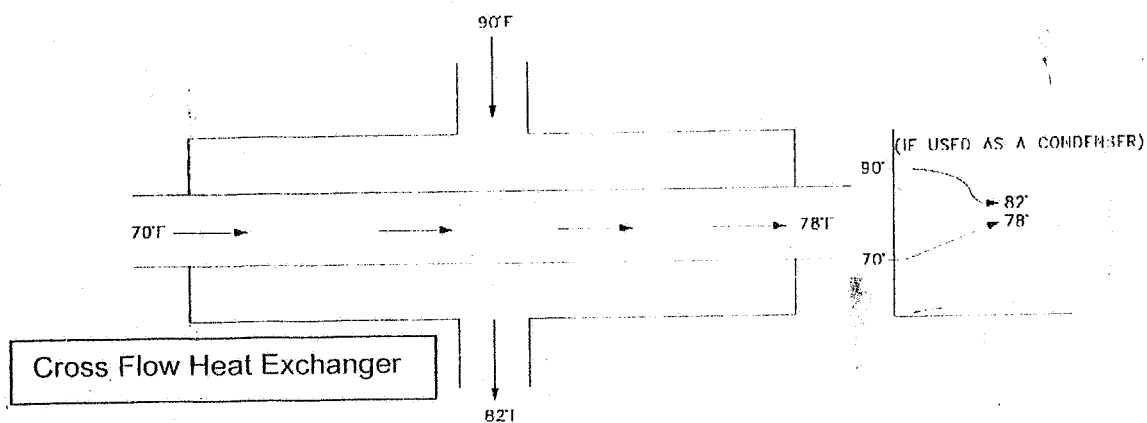
Heat Exchanger Design Course

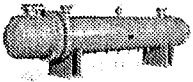
By: Eng. Mohamed A. Ebcid

Counter flow exists when the two fluids flow in opposite directions. Each of the fluids enters the heat exchanger at opposite ends. Because the cooler fluid exits the counter flow heat exchanger at the end where the hot fluid enters the heat exchanger, the cooler fluid will approach the inlet temperature of the hot fluid. Counter flow heat exchangers are the most efficient of the three types. In contrast to the parallel flow heat exchanger, the counter flow heat exchanger can have the hottest cold fluid temperature greater than the coldest hot-fluid temperature.



Cross flow exists when one fluid flows perpendicular to the second fluid; that is, one fluid flows through tubes and the second fluid passes around the tubes at 90° angle. Cross flow heat exchangers are usually found in applications where one of the fluids changes state (2-phase flow). An example is a steam system's condenser, in which the steam exiting the turbine enters the condenser shell side, and the cool water flowing in the tubes absorbs the heat from the steam, condensing it into water. Large volumes of vapor may be condensed using this type of heat exchanger flow.





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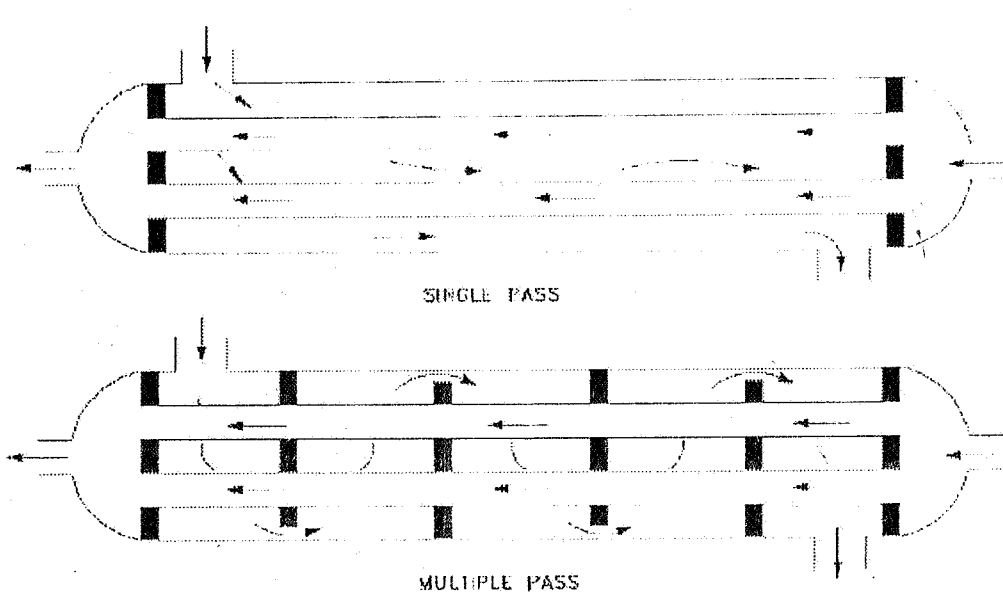
3. Comparison of the types of Flow

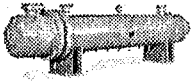
Each of the three types of heat exchangers has advantages and disadvantages. But of the three, the counter flow heat exchanger design is the most efficient when comparing heat transfer rate per unit surface area. The efficiency of a counter flow heat exchanger is due to the fact that the average T (difference in temperature) between the two fluids over the length of the heat exchanger is maximized. Therefore the log mean temperature for a counter flow heat exchanger is larger than the log mean temperature for a similar parallel or cross flow heat exchanger.

In actuality, most large heat exchangers are not purely parallel flow, counter flow, or cross flow; they are usually a combination of the two or all three types of heat exchangers. This is due to the fact that actual heat exchangers are more complex than the simple components shown in the idealized figures used above to depict each type of heat exchanger. The reason for the combination of the various types is to maximize the efficiency of the heat exchanger within the restrictions placed on the design. That is, size, cost, weight, required efficiency, type of fluids, operating pressures, and temperatures, all help determine the complexity of a specific heat exchanger.

One method that combines the characteristics of two or more heat exchangers and improves the performance of a heat exchanger is to have the two fluids pass each other several times within a single heat exchanger. When a heat exchanger's fluids pass each other more than once, a heat exchanger is called a multi-pass heat exchanger. If the fluids pass each other only once, the heat exchanger is called a single-pass heat exchanger. See Figure 6 for an example of both types.

Commonly, the multi-pass heat exchanger reverses the flow in the tubes by use of one or more sets of "U" bends in the tubes. The "U" bends allow the fluid to flow back and forth across the length of the heat exchanger. A second method to achieve multiple passes is to insert baffles on the shell side of the heat exchanger. These direct the shell side fluid back and forth across the tubes to achieve the multi-pass effect.





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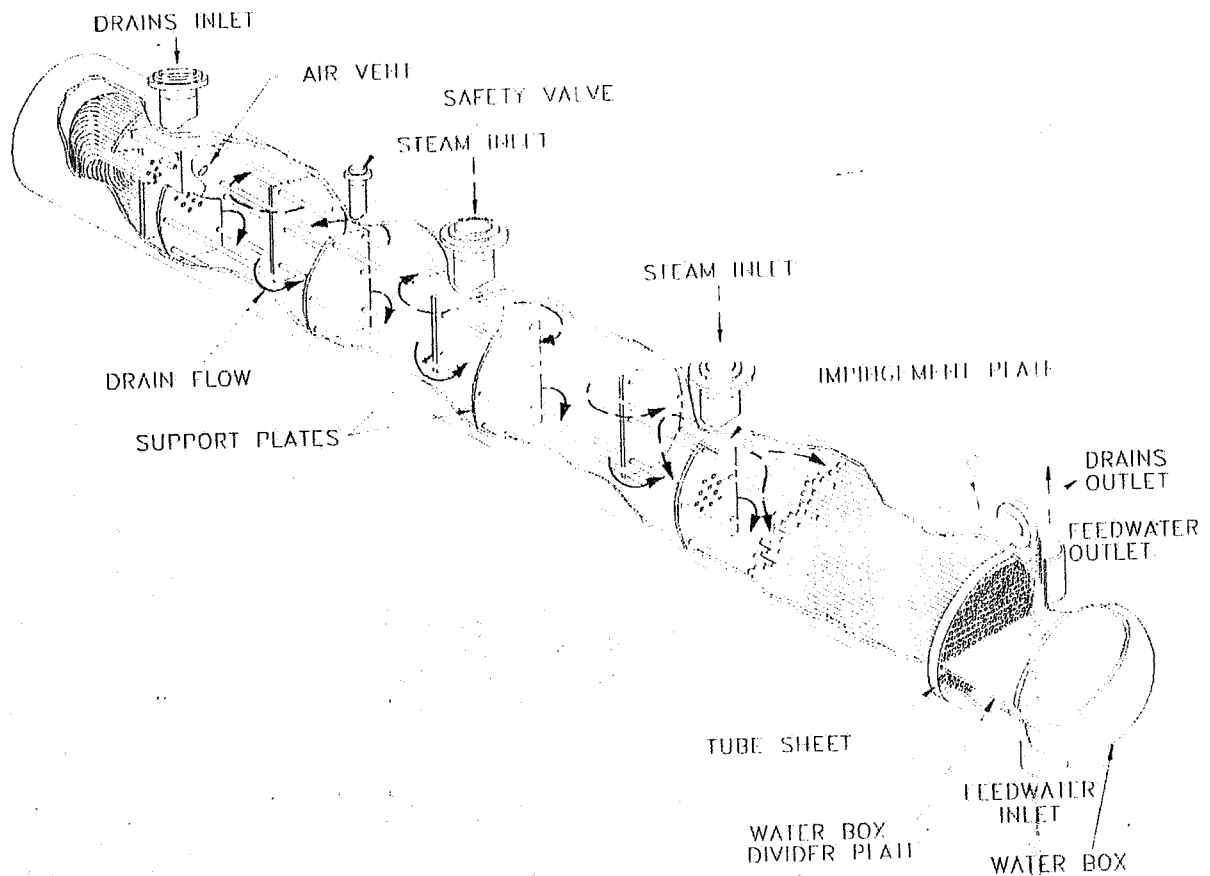
II. HEAT EXCHANGER TYPES

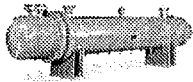
1. SHELL & TUBE HEAT EXCHANGER

The most basic and the most common type of heat exchanger construction is the tube and shell. This type of heat exchanger consists of a set of tubes in a container called a shell. The fluid flowing inside the tubes is called the tube side fluid and the fluid flowing on the outside of the tubes is the shell side fluid. At the ends of the tubes, the tube side fluid is separated from the shell side fluid by the tube sheet(s).

The tubes are rolled and press-fitted or welded into the tube sheet to provide a leak tight seal. In systems where the two fluids are at vastly different pressures, the higher pressure fluid is typically directed through the tubes and the lower pressure fluid is circulated on the shell side. This is due to economy, because the heat exchanger tubes can be made to withstand higher pressures than the shell of the heat exchanger for a much lower cost.

The support plates also act as baffles to direct the flow of fluid within the shell back and forth across the tubes.





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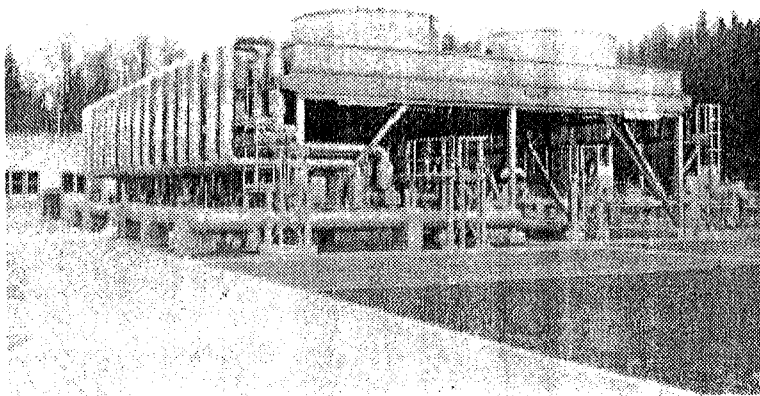
Shell and tube (or tubular) heat exchangers are used in applications where high temperature and pressure demands are significant. Tubular heat exchangers are also employed when fluid contains particles that would block the channels of a plate heat exchanger. There are several types of shell and tube heat exchangers including U-tube, straight, spiral, and finned tube.

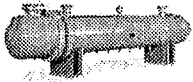
The U-tube design consists of straight length tubes bent into a U-shape. The bundle is fitted with tube supports or flow baffles, depending upon the fluid outside the tubes. The tube assembly is placed in a shell to contain the fluid on the outside of the tube bundle. A head assembly is bolted to the shell to direct the fluid into the tube bundle.

To handle heavy fouling fluids or applications where temperature cross conditions exist, straight-tube designs are available (a temperature cross occurs when the fluid being heated has an outlet temperature that falls between the inlet and outlet temperature of the heating medium). Because of the straight tubes, the head assemblies can be removed and the tubes can be mechanically cleaned. Depending upon the tube sheet configuration (fixed or floating) the ability to handle large temperature differences between the fluids may be limited.

2. AIR COOLED HEAT EXCHANGER

Air-cooled heat exchangers have an integral electric motor or fluid powered fan for cooling / heat removal of media. Some air-cooled models do not include the fan, but have cutouts for aftermarket fans. Air cooled heat exchanger are designed according to API-661 & ASME





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3. PLATE HEAT EXCHANGER

A plate type heat exchanger consists of plates instead of tubes to separate the hot and cold fluids. The hot and cold fluids alternate between each of the plates. Baffles direct the flow of fluid between plates. Because each of the plates has a very large surface area, the plates provide each of the fluids with an extremely large heat transfer area. Therefore a plate type heat exchanger, as compared to a similarly sized tube and shell heat exchanger, is capable of transferring much more heat. This is due to the larger area the plates provide over tubes. Due to the high heat transfer efficiency of the plates, plate type heat exchangers are usually very small when compared to a tube and shell type heat exchanger with the same heat transfer capacity. Plate type heat exchangers are not widely used because of the inability to reliably seal the large gaskets between each of the plates. Because of this problem, plate type heat exchangers have only been used in small, low pressure applications such as on oil coolers for engines. However, new improvements in gasket design and overall heat exchanger design have allowed some large scale applications of the plate type heat exchanger. As older facilities are upgraded or newly designed facilities are built, large plate type heat exchangers are replacing tube and shell heat exchangers and becoming more common.

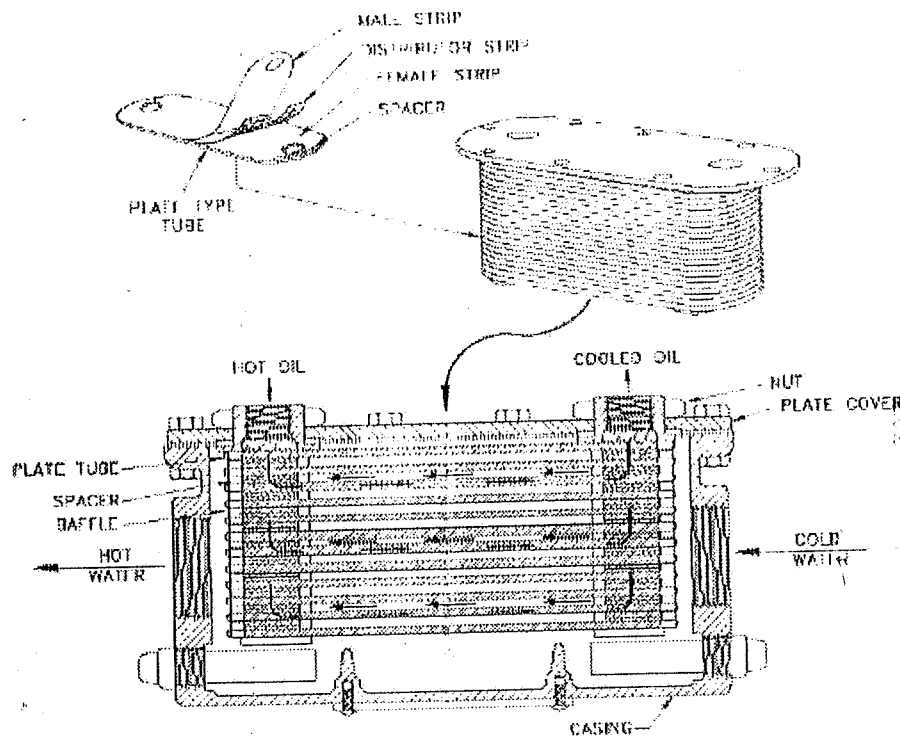
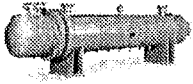


Plate heat exchangers are often used on low-viscous and clean applications with moderate demands on operating temperatures and pressures, typically below 150°C. Gasket material is chosen to withstand the operating temperature at hand and the properties of the processing fluid. There are several types of plate heat exchangers including gasketed, brazed, welded and semi-weld or hybrid types.

The most common of the plate-type heat exchangers is the gasketed plate-and-frame heat exchanger. Gasketed plate-and-frame heat exchangers consist of a series of channelled plates that are mounted



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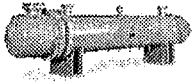
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on a frame and clamped together. Each plate is made from "pressable" materials (stainless steel, nickel, titanium, etc.) and is formed with a series of corrugations. Also included with each plate is an elastomer gasket. The gaskets contain the pressure and control the flow of each medium. The gasketed plates are assembled in a pack, mounted on upper and lower guide rails, and compressed between two end frames by compression bolts. The gasket arrangement of each plate distributes the hot and cold media into alternating flow channels throughout the plate pack. In all cases, hot and cold fluids flow counter to each other.

Brazed plate heat exchangers consist of specially formed plates, vacuum brazed together to form a heat transfer device.

Welded plate heat exchangers require no gaskets; they are fully welded and typically constructed of a single material (usually stainless steel).

Semi-weld or hybrid plate exchangers consist of pairs of plates that are laser welded together into cassettes. This allows one fluid to flow in a welded channel and the other to flow in a gasketed channel.



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III. CODES, STANDARDS, SPECIFICATIONS

Standards of the Tubular Exchanger Manufacturers Association

TEMA Eight Edition - 1999

The Standard Of The Brazed Aluminum Palte-Fin Heat Exchanger Manufacturer' Association

ALPEMA Second Edition - 2000

American Petroleum Institute (API)

API 660 Shell and tube heat exchanger for general refinery services
API 661 Air cooled exchanger for refinery services

Expansion Joint Manufacturers Association, Inc.

Standards, 5th Edition with 1985 Addenda and Practical Guide to Expansion Joints

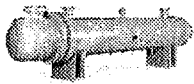
ASME Boiler and Pressure Vessel Code.

Sec II Material Specifications
Sec V Nondestructive Examination
Sec VIII Pressure Vessels — Division 1
Sec IX Welding and Brazing Qualifications

PIPING, FITTINGS, FLANGES, AND VALVES

ASME

B31.1 — 1992 Power Piping
B31.2 — 1968 Fuel Gas Piping
B31.3 — 1993 Chemical Plant and Petroleum Refinery Piping
B31.4 — 1989 Liquid Petroleum Transportation Piping Systems
B31.4 — 1992 Refrigeration Piping with 1978 Addenda
B31.8 — 1992 Gas Transmission and Distribution Piping Systems
B36.19-1976 Stainless Steel Pipe
B36.10 M-1985 Welded and Seamless Wrought Steel Pipe
B 16.25-1992 Butt welding Ends
B16.10-1992 Face-to-Face and End-to-End Dimensions of Ferrous Valves
B16.9-1993 Factory-Made Wrought Steel Butt welding Fittings
B16.14-1991 Ferrous Pipe Plugs, Bushings, and Locknuts with Pipe Threads
B16.11 -1991 Forged Steel Fittings, Socket-Welding and Threaded
B16.5 1988 Pipe Flanges and Flanged Sitlings, Steel, Nickel Alloy and Other Special Alloys
B 16.20-1993 Ring-Joint Gaskets and Grooves for Steel Pipe Flanges



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MATERIALS

The American Society for Testing and Materials (ASTM)
1989 Annual Book of ASTM Standards, Section 1 Iron and Steel Products

MISCELLANEOUS

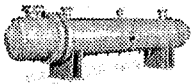
International Conference of Building Officials (ICBO)
Uniform Building Code — 1997

Steel Structures Painting Council (SSPC)
Steel Structures Painting Manual
Volume 1, Good Painting Practice
Volume 2, Systems and Specifications

Uniform Boiler and Pressure Vessel Laws Society
Synopsis of Boiler and Pressure Vessel Laws, Rules and Regulations by States, Cities, Counties and Provinces
(United States and Canada).

Environment Protection
Egyptian Law 4

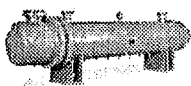
American Society of Civil Engineers (ASCE)
Minimum Design Loads for Buildings and Other Structures
ASCE 7-88 (Formerly ANSI A58.1)



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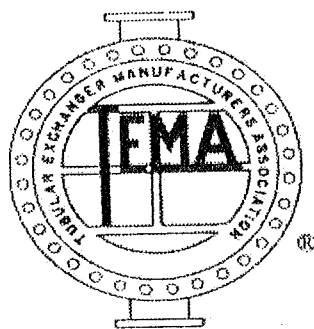
PART 2



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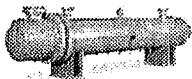
By: Eng. Mohamed A. Ebcid

STANDARDS OF THE TUBULAR EXCHANGER MANUFACTURERS ASSOCIATION



EIGHTH EDITION

TUBULAR EXCHANGER MANUFACTURERS ASSOCIATION, INC.
25 North Broadway
Tarrytown, New York 10591
Richard C. Byrne, Secretary
www.tema.org



TEMA CLASSES

1. TEMA CLASSES

1.1 TEMA CLASS R

The TEMA Mechanical Standards for Class "R" heat exchangers specify design and fabrication of unfired shell and tube heat exchangers for the generally severe requirements of petroleum and related processing applications.

1.2 TEMA CLASS C

The TEMA Mechanical Standards for Class "C" heat exchangers specify design and fabrication of unfired shell and tube heat exchangers for the generally moderate requirements of commercial and general process applications.

1.3 TEMA CLASS B

The TEMA Mechanical Standards for Class "B" heat exchangers specify design and fabrication of unfired shell and tube heat exchangers for chemical process service.

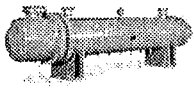
The most restrictive and includes the requirement for confined joints where recesses must be machined in the flanges and tubesheets. Spiral-wound gaskets with a ring construction also meet that TEMA R requirement. TEMA R also calls for greater minimum thickness for some components.

1.4 NOTES TO USERS OF THE TEMA STANDARDS

Three classes of Mechanical Standards, R, C and B, reflecting acceptable designs for various service applications are presented. The user should refer to the definition of each class and choose the one that best fits the specific need.

Corresponding subject matter in the three Mechanical Standards is covered by paragraphs identically numbered except for the prefix letter. Paragraph numbers preceded by RCB indicates that all three classes are identical. Any reference to a specific paragraph must be preceded by the class designation.

The Recommended Good Practice section has been prepared to assist the designer in areas outside the scope of the basic Standards. Paragraphs in the Standards having additional information in the RGP section are marked with an asterisk (*). The reference paragraph in the RGP section has the identical paragraph number, but with an "RGP" prefix.



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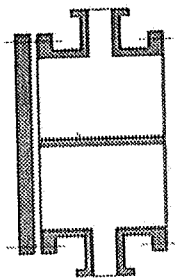
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2. TEMA TYPES

TEMA designations refer to the construction of the heat exchanger as follows:

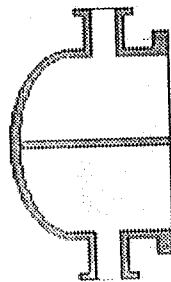
2.1 FRONT HEAD DESIGN

A



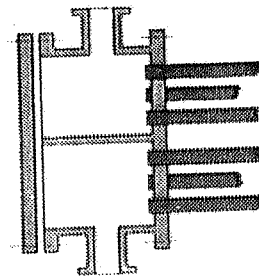
Channel and
Removable Cover

B



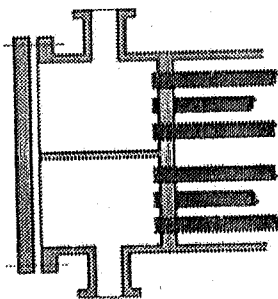
Bonnet
(Integral Cover)

C



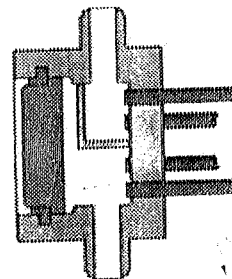
Channel Integral
with Tubesheet and
Removable Cove

N

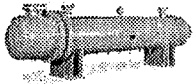


Channel Integral
with Tubesheet and
Removable Cove

D



Special High
Pressure Closure



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2.2 SHELL DESIGN

E



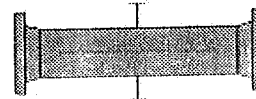
One Pass Shell

F



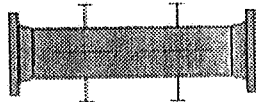
Two Pass Shell

G



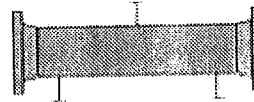
Split Flow

H



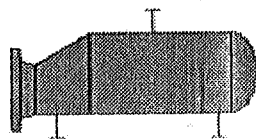
Double Split Flow

J



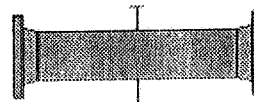
Divided Flow

K

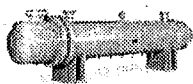


Kettle Type Reboiler

X



Cross Flow

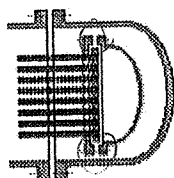


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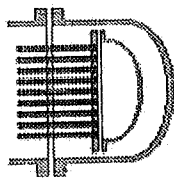
2.3 REAR HEAD DESIGN

S



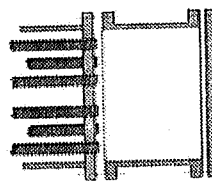
Floating Head
with backing device

T



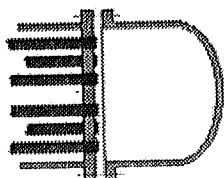
Pull Through
Floating Head

L



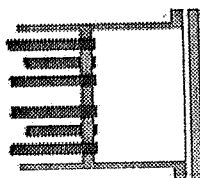
Fixed Tubesheet
Stationary Head

M



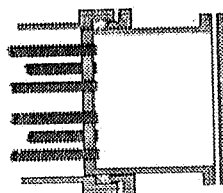
Fixed Tubesheet
Stationary Head

N



Fixed Tubesheet
Stationary Head

P



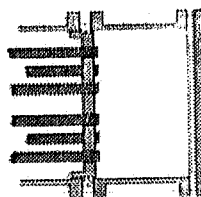
Outside Packed
Floating Head

U



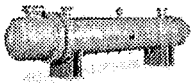
U-Tube Bundle

W



Externally Sealed
Floating Tubesheet

Example: TEMA Type BEM has a Type-B front head, a Type-E shell and a Type-M rear head design. TEMA types are chosen depending on the application. With high vapor flows, high pressure and temperature crossing, a combination of special TEMA features is advantageous. Example: Type-K shells allow for proper vapor disengagement in reboilers, while type-J and Type-H shells accommodate high vapor flow.



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3. SIZE NUMBERING AND TYPE DESIGNATION

It is recommended that heat exchanger size and type be designated by numbers and letters as described below.

3.1 SIZE

Sizes of shells (and tube bundles) shall be designated by numbers describing shell (and tube bundle) diameters and tube lengths, as follows:

3.2 NOMINAL DIAMETER

The nominal diameter shall be the inside diameter of the shell in inches (mm), rounded off to the nearest integer. For kettle reboilers the nominal diameter shall be the port diameter followed by the shell diameter, each rounded off to the nearest integer.

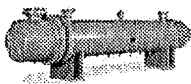
3.3 NOMINAL LENGTH

The nominal length shall be the tube length in inches (mm). Tube length for straight tubes shall be taken as the actual overall length. For U-tubes the length shall be taken as the approximate straight length from end of tube to bend tangent.

3.4 TYPE

Type designation shall be by letters describing stationary head, shell (omitted for bundles only), and rear head, in that order.

Example: Split-ring floating head exchanger with removable channel and cover, single pass shell, 23-1/4" (591 mm) inside diameter with tubes 16' (4877 mm) long. SIZE 23-192 (591-4877) TYPE AES.



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4. TYPES OF BUNDLES

4.1 NON REMOVABLE BUNDLE EXCHANGERS

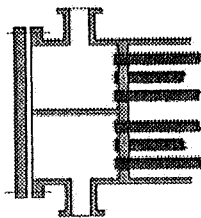
These types of units are often used in high pressure services and services where you wish to avoid leakage problems at gasketed joints. Another advantage is that they are generally more cost effective than removable bundle designs.

4.1.1 STRAIGHT TUBE, FIXED TUBESHEET: NEN

This TEMA type is the simplest. It's constructed without packed or gasketed joints on the shell side. The tubesheet is welded to the shell and the heads are bolted to the tubesheet. On the NEN heat exchanger, the shell and the head are welded to the tubesheet. Typically, a cover plate design is provided to make tube cleaning easier. This TEMA category, especially the NEN, is the least expensive TEMA design per square foot of heat transfer surface.

Front End
Stationary Head Type

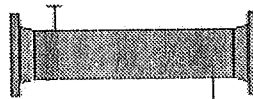
N



Channel Integral
with Tubesheet and
Removable Cove

Shell Type

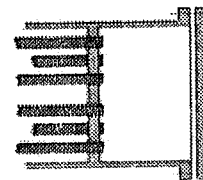
E



One Pass Shell

Rear End
Head Type

N



Fixed Tubesheet
Stationary Head

Advantages:

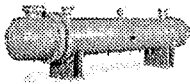
- Less costly than removable bundle designs;
- Provides maximum surface for a given shell and tube diameter;
- Allows for single and multiple tube passes for proper velocity;
- May be interchangeable with other manufacturers of the same TEMA type.

Limitations:

- Shell side can be cleaned only by chemical methods;
- No provision to allow for differential thermal expansion-- must use an expansion joint.

Applications:

- Oil coolers, liquid/liquid exchangers, vapor condensers, reboilers, gas coolers;
- More viscous and warmer fluids generally flow through the shell;
- Corrosive or high-fouling fluids should flow inside the tubes.



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4.2 REMOVABLE BUNDLE EXCHANGERS

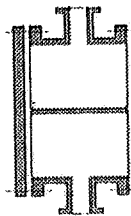
Removable bundle exchangers give the customer the ability to replace the tube bundle without replacing the shell or bonnets. They are generally less cost effective than non removable designs.

4.2.1 EXTERNALLY SEALED FLOATING TUBESHEET: AEW

This design allows for the removal, inspection and cleaning of the shell circuit and shell interior. The special floating tubesheet prevents the intermixing of fluids. In most cases, a straight tube design is more economical than a U-tube design.

Front End
Stationary Head Type

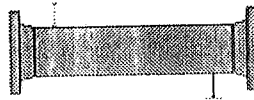
A



Channel and
Removable Cover

Shell Type

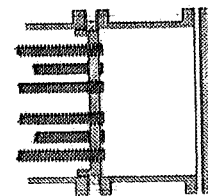
E



One Pass Shell

Rear End
Head Type

W



Externally Sealed
Floating Tubesheet

Advantages:

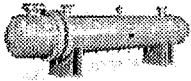
- Floating tubesheet allows for differential thermal expansion between the shell and the tube bundle;
- Shell circuit can be inspected, and steam cleaned or mechanically cleaned;
- The tube bundle can be repaired or replaced without disturbing shell pipe;
- Less costly than other floating head designs: BEP or BES;
- For removable bundles, it gives maximum surface area per given shell diameter;
- Tubes can be cleaned in AEW models without removing piping.

Limitations:

- Fluids in shell and tube circuits must be nonvolatile, nontoxic;
- Tube side passes are limited to single- or two-pass design;
- Two tubesheets are applied: front and back;
- Tubes cannot expand independently; thus, large temperature differentials should be avoided;
- Packing materials limit design pressure and temperature.

Applications are:

- Intercoolers and aftercoolers, air on tubeside;
- Coolers with water inside the tubes;
- Jacket water coolers;
- Hot side fluid through the shell with entry nearest the front



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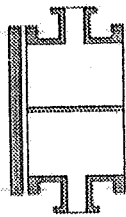
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4.2.2 OUTSIDE PACKED HEAD: AEP

This design allows for the easy removal, inspection and cleaning of the shell circuit and shell interior without taking off the floating head cover. A special floating tubesheet prevents the intermixing of fluids. The design incorporates the removal of straight tubes for cleaning. That is usually more costly than designs accommodating U-tubes.

**Front End
Stationary Head Type**

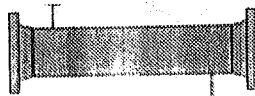
A



**Channel and
Removable Cover**

Shell Type

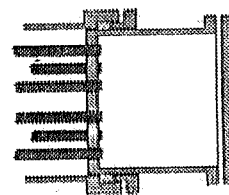
E



One Pass Shell

**Rear End
Head Type**

P



**Outside Packed
Floating Head**

Advantages:

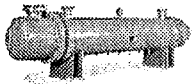
- Floating tube- sheet allows for differential thermal expansion between the shell and the tube bundle;
- Shell circuit can be inspected and steam cleaned. If the tube bundle has a square tube pitch, tubes can be mechanically cleaned by passing a brush between rows of tubes;
- The tube bundle can be repaired or replaced without disturbing shell piping;
- For AEP designs, tubes can be serviced without disturbing tubeside piping;
- Less costly than TEMA-type BES or BET designs;
- Only shell fluids are exposed to packing. Toxic or volatile fluids can be cooled in the tubeside circuit;
- Provides large bundle entrance area, reducing the need for entrance domes for proper fluid distribution.

Limitations:

- Shell fluids limited to nonvolatile, nontoxic materials;
- Packing limits shell-side design temperature and pressure;
- All tubes are attached to two tubesheets. Tubes cannot expand independently; thus, large thermal expansions should be avoided;
- Less surface per given shell and tube diameter than AEW or BEW.

Applications:

- Flammable or toxic liquids in the tube circuit;
- Good for high-fouling liquids in the tube circuit.



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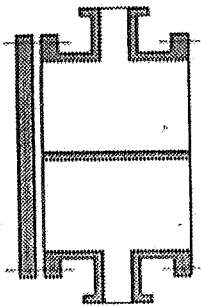
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4.2.3 INTERNAL SPLIT-RING FLOATING HEAD: AES

Exchangers with AES, BES, etc., designs are ideal for applications requiring frequent tube bundle removal, inspection and cleaning. The exchangers have a straight-tube design, suitable for large differential temperatures between the shell and tube fluids. They are more forgiving to thermal shock than AEW or BEW designs, and are suitable for cooling volatile or toxic fluids.

Front End
Stationary Head Type

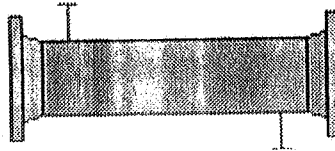
A



Channel and
Removable Cover

Shell Type

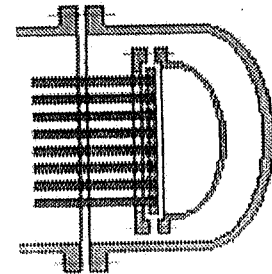
E



One Pass Shell

Rear End
Head Type

S



Floating Head
with backing device

Advantages:

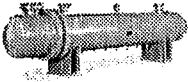
- Floating head design allows for differential thermal expansion between the shell and the tube bundle;
- Shell circuit can be inspected and steam cleaned. If it has a square tube layout, tubes can be mechanically cleaned;
- Higher surface per given shell and tube diameter than "pull-through" designs such as AET, BET, etc.;
- Provides multipass tube circuit arrangement.

Limitations:

- Shell-cover, split-ring and floating head cover must be removed to inspect the tube bundle. That results in higher maintenance cost than with the pull-through bundle;
- More costly per square foot of surface than fixed tube sheet or U-tube designs.

Applications:

- Chemical processing of toxic fluids;
- Special intercoolers and aftercoolers;
- General industrial uses.



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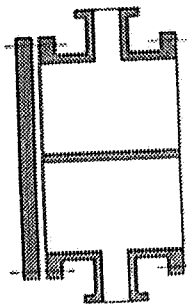
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4.2.4 PULL-THROUGH FLOATING HEAD: AET

Ideal for applications requiring frequent tube bundle removal for inspection and cleaning. The floating head is bolted to the floating tubesheet. That alleviates the need to remove the floating head to pull the tube bundle.

Front End
Stationary Head Type

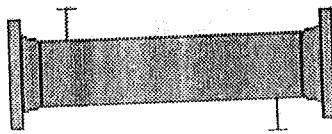
A



Channel and
Removable Cover

Shell Type

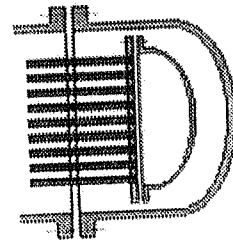
E



One Pass Shell

Rear End
Head Type

T



Pull Through
Floating Head

Advantages:

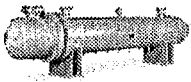
- Floating head design allows for differential thermal expansion between the shell and the tube bundle;
- Shell circuit can be inspected, steam or mechanically cleaned;
- Provides large bundle entrance area for proper fluid distribution;
- Provides multipass tube circuit arrangement;
- Suitable for toxic or volatile fluid cooling.

Limitations:

- For a given set of conditions, this TEMA style is the most expensive;
- Less surface per given shell and tube diameter than other removable designs.

Applications:

- Chemical processing applications for toxic fluids;
- Hydrocarbon fluid condensers;
- General industrial applications requiring frequent cleaning.



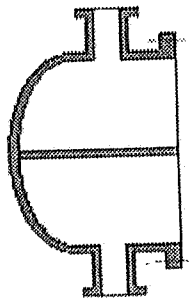
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4.2.5 U-TUBE: BEU

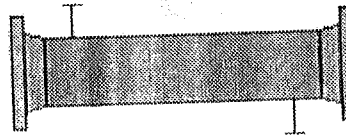
Especially good for severe performance requirements with maximum thermal expansion capability. Because each tube can expand and contract independently, the design works well with large temperature differentials. While the AEM and AEW are the least expensive designs, U-tube bundles are also economical.

Front End
Stationary Head Type
B



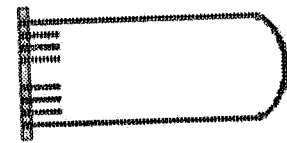
Bonnet
(Integral Cover)

Shell Type
E



One Pass Shell

Rear End
Head Type
U



U-Tube Bundle

Advantages:

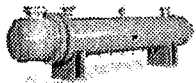
- U-tube design allows for maximum differential thermal expansion between the shell and the tube bundle—for individual tubes too;
- Shell circuit can be inspected, steamed or mechanically cleaned;
- Less costly than floating head or packed floating head designs;
- Provides multipass tube circuit arrangement;
- withstands thermal expansion applications;
- Bundle can be removed from one end for cleaning or replacement.

Limitations:

- Because of the 180° bend, U-tubes can be cleaned only by chemical means;
- U-tubes are difficult to replace;
- Neither single tube passes nor true countercurrent flow is possible;
- Tube wall thickness at the U-bend is thinner than for straight tubes;
- Tube circuits are difficult to drain.

Applications:

- Oil, chemical and water heating applications;
- Excellent in steam to liquid applications.



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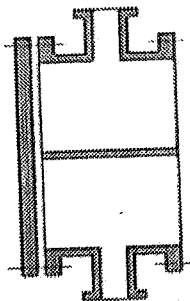
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4.2.6 KETTLE REBOILER

Kettle Reboiler is a prime example of a pool boiling unit. The tube bundle is usually U-tubes or has a separate head bolted directly to the floating tube sheet. The tube bundle is inserted into an enlarged shell which serves as a reservoir for column bottoms and as a disengaging space for the vapor. An overflow weir is normally provided to maintain the liquid level above the top of the tubes.

Front End
Stationary Head Type

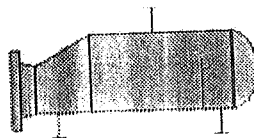
A



Channel and
Removable Cover

Shell Type

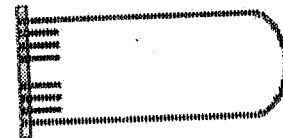
K



Kettle Type
Reboiler

Rear End
Head Type

U



U-Tube Bundle

Advantages:

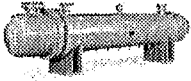
- The reboiler operation is insensitive to large changes in process operating conditions as there is no two phase fluid recirculating back to the column.
- High heat fluxes up to 20,000 BTU/hr.ft² are possible.
- Higher residence time in heating zone.
- Easy maintenance and cleaning.

Limitations:

- This type of reboiler tends to foul much more readily than other types and heavy residual materials accumulate continuously in the vaporization section.
- Expensive design.

Applications:

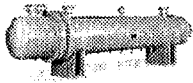
- Evaporation applications of Oil, chemical and water heating



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PART 3



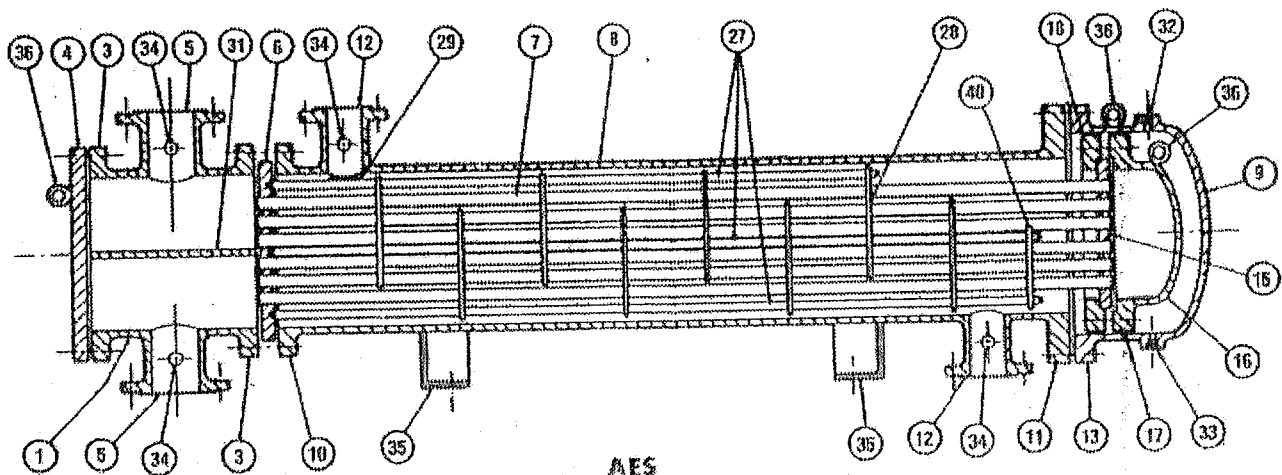
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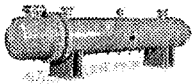
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I. COMPONENTS

For the purpose of establishing standard terminology, typical parts and connections are numbered for identification.

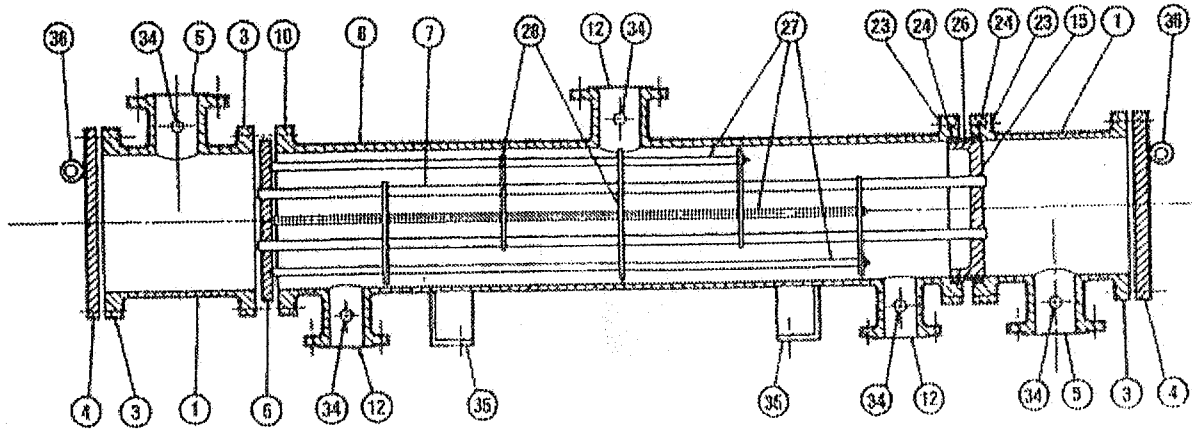
- | | |
|---|--|
| 1. Stationary Head-Channel | 21. Floating Head Cover-External |
| 2. Stationary Head-Bonnet | 22. Floating Tubesheet Skirt |
| 3. Stationary Head Flange-Channel or Bonnet | 23. Packing Box |
| 4. Channel Cover | 24. Packing |
| 5. Stationary Head Nozzle | 25. Packing Gland |
| 6. Stationary Tubesheet | 26. Lantern Ring |
| 7. Tubes | 27. Tie rods and Spacers |
| 8. Shell | 28. Transverse Baffles or Support Plates |
| 9. Shell Cover | 29. Impingement Plate |
| 10. Shell Flange-Stationary Head End | 30. Longitudinal Baffle |
| 11. Shell Flange-Rear Head End | 31. Pass Partition |
| 12. Shell Nozzle | 32. Vent Connection |
| 13. Shell Cover Flange | 33. Drain Connection |
| 14. Expansion Joint | 34. Instrument Connection |
| 15. Floating Tubesheet | 35. Support Saddle |
| 16. Floating Head Cover | 36. Lifting Lug |
| 17. Floating Head Cover Flange | 37. Support Bracket |
| 18. Floating Head Backing Device | 38. Weir |
| 19. Split Shear Ring | 39. Liquid Level Connection |
| 20. Slip-on Backing Flange | 40. Floating Head Support |



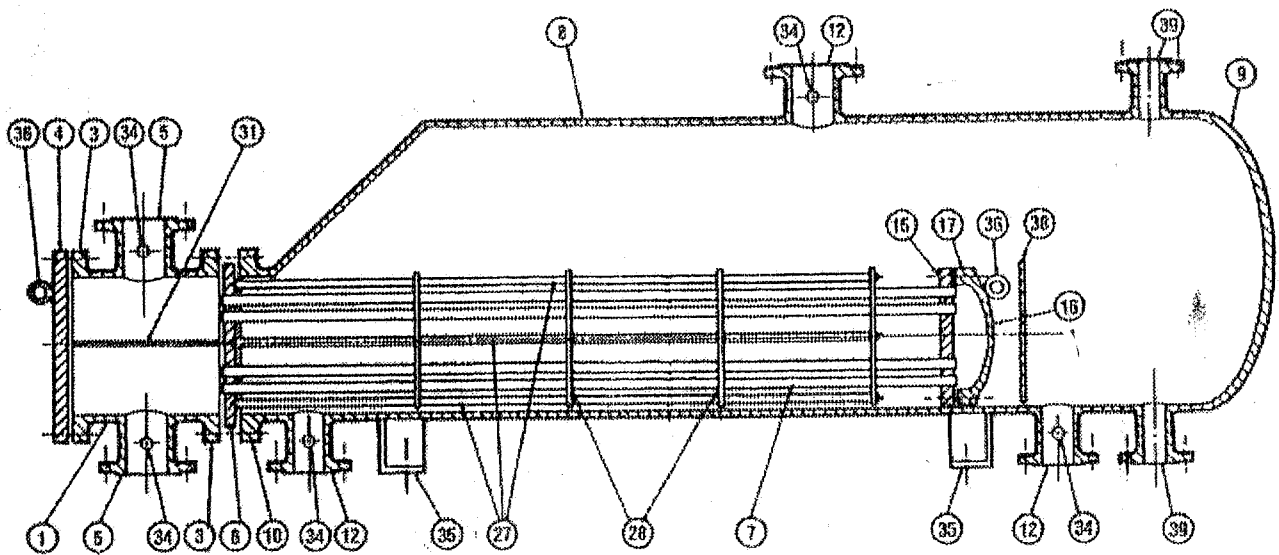


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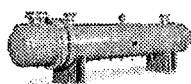
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AJW



AKT



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II. SCOPE AND GENERAL REQUIREMENTS

1. SCOPE OF STANDARDS

1.1 GENERAL

The TEMA Mechanical Standards are applicable to shell and tube heat exchangers which do not exceed any of the following criteria:

1. Inside diameters of 100 inches (2540 mm)
2. Product of nominal diameter, inches (mm) and design pressure, psi (kPa) of 100,000 (17.5x10⁶)
3. A design pressure of 3,000 psi (20684 kPa)

The intent of these parameters is to limit the maximum shell wall thickness to approximately 3 inches (76 mm), and the maximum stud diameter to approximately 4 inches (102 mm). Criteria contained in these Standards may be applied to units which exceed the above parameters.

1.2 CONSTRUCTION CODES

The individual vessels shall comply with the ASME (American Society of Mechanical Engineers) Boiler and Pressure Vessel Code, Section VIII, Division 1, hereinafter referred to as the Code. These Standards supplement and define the Code for heat exchanger applications. The manufacturer shall comply with the construction requirements of state and local codes when the purchaser specifies the plant location. It shall be the responsibility of the purchaser to inform the manufacturer of any applicable local codes. Application of the Code symbol is required, unless otherwise specified by the purchaser.

1.3 MATERIALS

For purposes of these Standards, "carbon steel" shall be construed as any steel or low alloy falling within the scope of Part UCS of the Code. Metals not included by the foregoing (except cast iron) shall be considered as "alloys" unless otherwise specifically named. Materials of construction, including gaskets, should be specified by the purchaser. The manufacturer assumes no responsibility for deterioration of parts for any reason.

Cast iron shall be used only for water service at pressures not exceeding 150 psi (1034 kPa).

Packed joints shall not be used when the purchaser specifies that the fluid in contact with the joint is lethal or flammable.

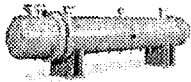
1.4 STANDARD CORROSION ALLOWANCES

The standard corrosion allowances used for the various heat exchanger parts are as follows, unless the conditions of service make a different allowance more suitable and such allowance is specified by the purchaser.

1.4.1 CARBON STEEL PARTS

All carbon steel pressure parts, except as noted below, are to have a corrosion allowance of 1/8" (3.2 mm).

Internal floating head covers are to have the corrosion allowance on all wetted surfaces except gasket seating surfaces. Corrosion allowance on the outside of the flanged portion may be included in the recommended minimum edge distance.



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Tubesheets are to have the corrosion allowance on each side with the provision that, on the grooved side of a grooved tubesheet, the depth of the gasketed groove may be considered as available for corrosion allowance.

Where flat external covers are grooved, the depth of the gasketed groove may be considered as available for corrosion allowance.

Corrosion allowance shall be applied only to the inside diameter of flanges where exposed to the fluids.

Non-pressure parts such as tie-rods, spacers, baffles and support plates are not required to have corrosion allowance.

Tubes, bolting and floating head backing devices are not required to have corrosion allowance.

Pass partition plates are not required to have corrosion allowance.

1.4.2 ALLOY PARTS

Alloy parts are not required to have corrosion allowance.

1.4.3 CAST IRON PARTS

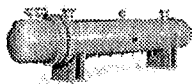
Cast iron pressure parts shall have a corrosion allowance of 1/8" (3.2 mm).

1.4.4 ANODES

Selection and placement of anodes is not the responsibility of the heat exchanger manufacturer. If a heat exchanger is to be furnished with anodes, when requesting a quotation, the purchaser is responsible for furnishing the heat exchanger manufacturer the following information:

1. Method of anode attachment.
2. Quantity of anodes required.
3. Size and manufacturer of the anodes.
4. Anode material.
5. Sketch of anode locations and spacing.

If the heat exchanger manufacturer chooses to install anodes for a customer, the manufacturer is not responsible for the suitability of the anodes for the service it is installed in, the life of the anodes, the corrosion protection provided by the anode, or any subsequent damage to the heat exchanger attributed to the anode, the method of anode installation, or the installed location of the anode in the heat exchanger.



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II. MECHANICAL DESIGN

1. GENERAL

Heat Exchanger design is one of the most complicated jobs for a Static Equipment Engineer; he must design the heat exchanger as an integrated item, not as individual parts.

2. TUBES (RCB-2)

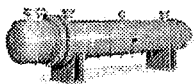
2.1 TUBES DIMENSIONS

The following tube lengths for both straight and U-tube exchangers are commonly used: 96 (2438), 120 (3048), 144 (3658), 192 (4877) and 240 (6096) inches (mm). Other lengths may be used.

Table RCB-2.21 lists common tube diameters and gages for bare tubes of copper, steel and alloy. Other diameters and gages are acceptable.

TABLE RCB-2.21
BARE TUBE DIAMETERS AND GAGES

O.D. Inches (mm)	Copper and Copper Alloys	Carbon Steel, Aluminum and Aluminum Alloys	Other Alloys
	B.W.G.	B.W.G.	B.W.G.
1/4 (6.4)	27 24 22	-	27 24 22
3/8 (9.5)	22 20 18	-	22 20 18
1/2 (12.7)	20 18	-	20 18
5/8 (15.9)	20 18 16	18 16 14	20 18 16
3/4 (19.1)	20 18 16	16 14 12	18 16 14
7/8 (22.2)	18 16 14 12	14 12 10	16 14 12
(25.4)	18 16 14	14 12	16 14 12
1-1/4 (31.8)	16 14	14 12	14 12
1-1/2 (38.1)	16 14	14 12	14 12
2 (50.8)	14 12	14 12	14 12



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When U-bends are formed, it is normal for the tube wall at the outer radius to thin. The minimum tube wall thickness in the bent portion before bending shall be:

$$t_o = t_1 [1 + d_o/4R]$$

Where

t_o = Original tube wall thickness, inches (mm)

t_1 = Minimum tube wall thickness calculated by Code rules for a straight tube subjected to the same pressure and metal temperature, inches (mm)

d_o = Outside tube diameter, inches (mm)

R = Mean radius of bend, inches (mm)

When U-bends are formed from tube materials which are relatively non-work-hardening and of suitable temper, tube wall thinning in the bends should not exceed a nominal 17% of original tube wall thickness.

Flattening at the bend shall not exceed 10% of the nominal tube outside diameter.

U-bends formed from tube materials having low ductility, or materials which are susceptible to work-hardening, may require special consideration. Refer to Paragraph RCB-2.33.

2.2 BEND SPACING (RCB-2.32)

The center-to-center dimensions between parallel legs of U-tubes shall be such that they can be inserted into the baffle assembly without damage to the tubes.

The assembly of bends shall be of workmanlike appearance. Metal-to-metal contact between bends in the same plane shall not be permitted.

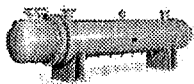
Cold work in forming U-bends may induce embrittlement or susceptibility to stress corrosion in certain materials and/or environments. Heat treatment to alleviate such conditions may be performed by agreement between manufacturer and purchaser.

2.3 TUBE PATTERN (RCB-2.4)

Standard tube patterns are:

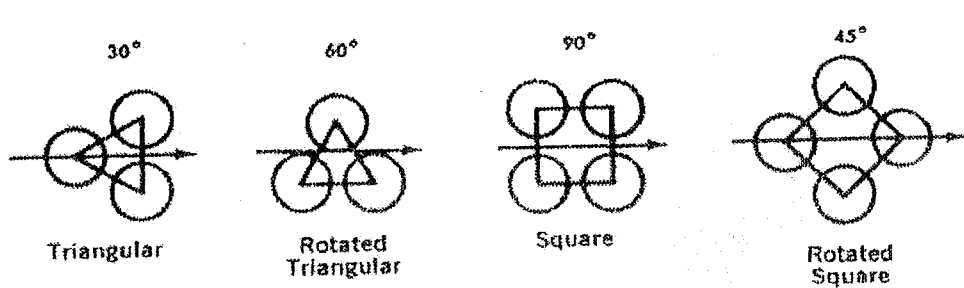
Square Pattern: In removable bundle units, when mechanical cleaning of the tubes is specified by the purchaser, tube lanes should be continuous.

Triangular Pattern: Triangular or rotated triangular pattern should not be used when the shell side is to be cleaned mechanically.



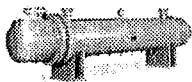
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2.4 TUBE PITCH (R-2.5)

Tubes shall be spaced with a minimum center-to-center distance of 1.25 times the outside diameter of the tube. When mechanical cleaning of the tubes is specified by the purchaser, minimum cleaning lanes of 1/4" (6.4 mm) shall be provided.



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3. SHELLS AND SHELL COVERS (RCB-3)

3.1 MINIMUM SHELL THICKNESS

Shell thickness is determined by the Code design formulas, plus corrosion allowance, but in no case shall the nominal thickness of shells be less than that shown in the applicable table. The nominal total thickness for clad shells shall be the same as for carbon steel shells.

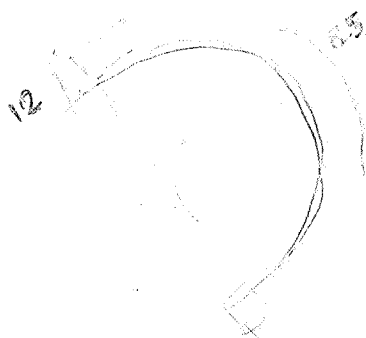
TABLE R-3.13
Minimum shell thickness dimensions in Inches (mm)

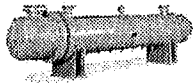
Nominal Shell Diameter	Minimum Thickness			
	Carbon Steel		Alloy *	
	Pipe	Plate		
6 (152)	SCH. 40	-	1/8 (3.2)	
8-12 (203-305)	SCH. 30	-	1/8 (3.2)	
13-29 (330-737)	SCH. STD	3/8 (9.5)	3/16 (4.8)	
30-39 (762-991)	-	7/16 (11.1)	1/4 (6.4)	
40-60 (1016-1524)	-	1/2 (12.7)	5/16 (7.9)	
61-80 (1549-2032)	-	1/2 (12.7)	5/16 (7.9)	
81-100 (2057-2540)	-	1/2 (12.7)	3/8 (9.5)	

*Schedule 5S is permissible for 6 inch (152 mm) and 8 inch (203 mm) shell diameters.

3.2 SHELL COVER THICKNESS

Nominal thickness of shell cover heads, before forming, shall be at least equal to the thickness of the shell as shown in the applicable table.





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4.5 SPACING OF BAFFLES AND SUPPORT PLATES

4.5.1 MINIMUM SPACING

Segmental baffles normally should not be spaced closer than $1/5$ of the shell ID or 2 inches (51 mm), whichever is greater. However, special design considerations may dictate a closer spacing.

4.5.2 MAXIMUM SPACING

Tube support plates shall be so spaced that the unsupported tube span does not exceed the value indicated in Table RCB-4.52 for the tube material used.

4.5.3 BAFFLE SPACING

Baffles normally shall be spaced uniformly, spanning the effective tube length. When this is not possible, the baffles nearest the ends of the shell, and/or tubesheets, shall be located as close as practical to the shell nozzles. The remaining baffles normally shall be spaced uniformly.

4.5.4 U-TUBE REAR SUPPORT

The support plates or baffles adjacent to the bends in U-tube exchangers shall be so located that, for any individual bend, the sum of the bend diameter plus the straight lengths measured along both legs from supports to bend tangents does not exceed the maximum unsupported span determined from Paragraph RCB-4.52. Where bend diameters prevent compliance, special provisions in addition to the above shall be made for support of the bends.

4.5.5 SPECIAL CASES

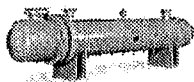
When pulsating conditions are specified by the purchaser, unsupported spans shall be as short as pressure drop restrictions permit. If the span under these circumstances approaches the maximum permitted by Paragraph RCB-4.52, consideration should be given to alternative flow arrangements which would permit shorter spans under the same pressure drop restrictions.

4.5.6 TUBE BUNDLE VIBRATION

Shell side flow may produce excitation forces which result in destructive tube vibrations. Existing predictive correlations are inadequate to insure that any given design will be free of such damage. The vulnerability of an exchanger to flow induced vibration depends on the flow rate, tube and baffle materials, unsupported tube spans, tube field layout, shell diameter, and inlet/outlet configuration. Section 6 of these Standards contains information which is intended to alert the designer to potential vibration problems. In any case, and consistent with Paragraph G-5, the manufacturer is not responsible or liable for any direct, indirect, or consequential damages resulting from vibration.

4.5.6 IMPINGEMENT BAFFLES AND EROSION PROTECTION

An impingement plate, or other means to protect the tube bundle against impinging fluids, shall be provided when entrance line values of $\sqrt{V^2}$ exceed the following: non-abrasive, single phase fluids, 1500 (2232); all other liquids, including a liquid at its boiling point, 500 (744). For all other gases and vapors, including all nominally saturated vapors, and for liquid vapor mixtures, impingement protection is required. V is the linear velocity of the fluid in feet per second (meters per second) and ρ is its density in pounds per cubic foot (kilograms per cubic meter). A properly designed diffuser may be used to reduce line velocities at shell entrance.



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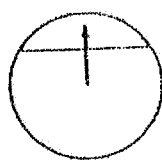
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4. BAFFLES AND SUPPORT PLATES (RCB-4)

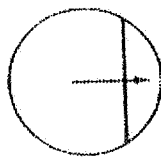
4.1 TYPE OF TRANSVERSE BAFFLES

The segmental or multi-segmental type of baffle or tube support plate is standard. Other type baffles are permissible. Baffle cut is defined as the segment opening height expressed as a percentage of the shell inside diameter or as a percentage of the total net free area inside the shell (shell cross sectional area minus total tube area). The number of tube rows that overlap for multi-segmental baffles should be adjusted to give approximately the same net free area flow through each baffle. Baffles shall be cut near the centerline of a row of tubes, of a pass lane, of a tube lane, or outside the tube pattern. Baffles shall have a workmanlike finish on the outside diameter. Typical baffle cuts are illustrated in Figure RCB-4.1. Baffle cuts may be vertical, horizontal or rotated.

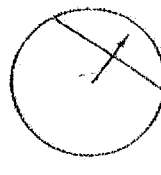
FIGURE RCB-4.1
BAFFLE CUTS FOR SEGMENTAL BAFFLES



Horizontal

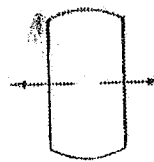
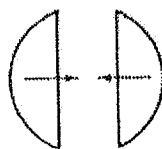
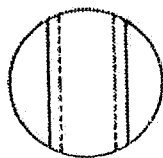


Vertical

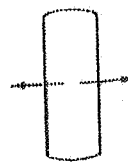
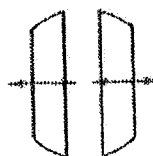
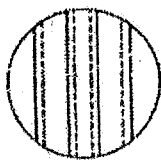


Rotated

BAFFLE CUTS FOR MULTI-SEGMENTAL BAFFLES



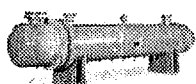
DOUBLE SEGMENTAL



TRIPLE SEGMENTAL

4.2 TUBE HOLES

Where the maximum unsupported tube length is 36 inches (914 mm) or less, or for tubes larger in diameter than 1 - 1/4 inches (31.8 mm) OD, standard tube holes are to be 1/32 inch (0.8 mm) over the OD of the tubes. Where the unsupported tube length exceeds 36 inches (914 mm) for tubes 1 - 1/4 inches (31.8 mm) diameter and smaller, standard tube holes are to be 1/64 inch (0.4 mm) over the OD of the tubes. For pulsating conditions, tube holes may be smaller than standard. Any burrs shall be removed and the tube holes given a workmanlike finish. Baffle holes will have an over-tolerance of 0.010 inch (0.3 mm) except that 4% of the holes are allowed an over-tolerance of 0.015 inch (0.4 mm).



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4.3 TRANSVERSE BAFFLE AND SUPPORT CLEARANCE

The transverse baffle and support plate clearance shall be such that the difference between the shell design inside diameter and the outside diameter of the baffle shall not exceed that indicated in Table RCB-4.3. However, where such clearance has no significant effect on shell side heat transfer coefficient or mean temperature difference, these maximum clearances may be increased to twice the tabulated values.

The design inside diameter of a pipe shell is defined as the nominal outside diameter of the pipe, minus twice the nominal wall thickness. The design inside diameter of a plate shell is the specified inside diameter. In any case, the design inside diameter may be taken as the actual measured shell inside diameter.

TABLE RCB-4.3
Standard Cross Baffle and Support Plate clearances
Dimensions in Inches (mm)

Nominal Shell ID	Design ID of Shell Minus Baffle OD
6 - 17 (152-432)	1/8 (3.2)
18 - 39 (457-991)	3/16 (4.8)
40 - 54 (1016-1372)	1/4 (6.4)
55 - 69 (1397-1753)	5/16 (7.9)
70 - 84 (1778-2134)	3/8 (9.5)
85 - 100 (2159-2540)	7/16 (11.1)

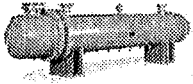
4.4 THICKNESS OF BAFFLES AND SUPPORT PLATES

The following tables show the minimum thickness of transverse baffles and support plates applying to all materials for various shell diameters and plate spacing.

The thickness of the baffle or support plates for U-tube bundles shall be based on the unsupported tube length in the straight section of the bundle. The U-bend length shall not be considered in determining the unsupported tube length for required plate thickness.

TABLE R-4.41
BAFFLE OR SUPPORT PLATE THICKNESS
Dimensions in Inches (mm)

Nominal Shell ID	Plate Thickness				
	Unsupported tube length between central baffles. End spaces between tubesheets and baffles are not a consideration.				
	24 (610) and Under	Over 24 (610) to 36 (914) Inclusive	Over 36 (914) to 48 (1219) Inclusive	Over 48 (1219) to 60 (1524) Inclusive	Over 60 (1524)
6 - 14 (152-356)	1/8 (3.2)	3/16 (4.8)	1/4 (6.4)	3/8 (9.5)	3/8 (9.5)
15 - 28 (381-711)	3/16 (4.8)	1/4 (6.4)	3/8 (9.5)	3/8 (9.5)	1/2 (12.7)
29 - 38 (737-965)	1/4 (6.4)	5/16 (7.5)	3/8 (9.5)	1/2 (12.7)	5/8 (15.9)
39 - 60 (991-1524)	1/4 (6.4)	3/8 (9.5)	1/2 (12.7)	5/8 (15.9)	5/8 (15.9)
61 - 100 (1549-2540)	3/8 (9.5)	1/2 (12.7)	5/8 (15.9)	3/4 (19.1)	3/4 (19.1)



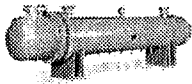
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Longitudinal baffles shall not be less than 1/4" (6.4 mm) nominal metal thickness.

Special consideration should be given to:

- Baffles and support plates subjected to pulsations.
- Baffles and support plates engaging finned tubes.
- Longitudinal baffles subjected to large differential pressures due to high shell side fluid pressure drop.
- Support of tube bundles when larger clearances allowed by RCB-4.3 are used.



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4.6 TIE RODS AND SPACERS RCB-4.7

Tie rods and spacers, or other equivalent means of tying the baffle system together, shall be provided to retain all transverse baffles and tube support plates securely in position.

Table R-4.71 shows suggested tie rod count and diameter for various sizes of heat exchangers. Other combinations of tie rod number and diameter with equivalent metal area are permissible; however, no fewer than four tie rods, and no diameter less than 3/8" (9.5 mm) shall be used. Any baffle segment requires a minimum of three points of support.

TABLE R-4.71 (TIE ROD STANDARDS)
Dimensions in Inches (mm)

Nominal Shell Diameter	Tie Rod Diameter	Minimum Number of Tie Rods
6-15 (152-381)	3/8 (9.5)	4
16-27 (406-686)	3/8 (9.5)	6
28-33 (711-838)	1/2 (12.7)	6
34-48 (864-1219)	1/2 (12.7)	8
49-60 (1245-1524)	1/2 (12.7)	10
61-100 (1549-2540)	5/8 (15.9)	12

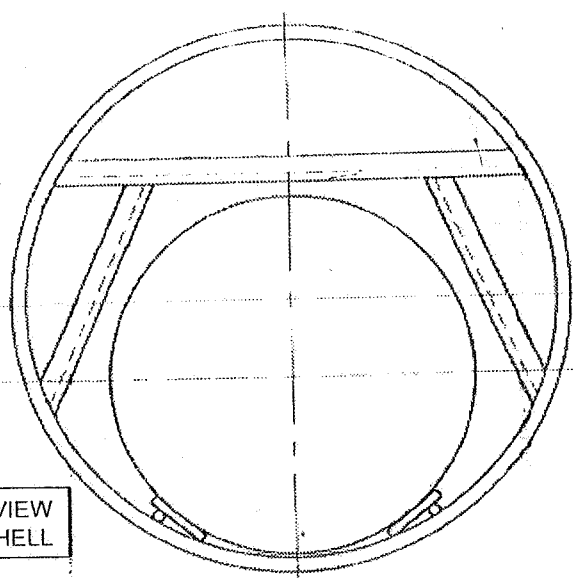
4.7 SEALING DEVICES RCB-4.8

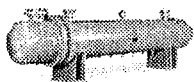
In addition to the baffles, sealing devices should be installed when necessary to prevent excessive fluid by-passing around or through the tube bundle. Sealing devices may be seal strips, tie rods with spacers, dummy tubes, or combinations of these.

4.8 KETTLE TYPE REBOILERS RCB-4.9

For kettle type reboilers, skid bars and a bundle hold-down may be provided. One method is shown in Figure RCB-4.9. Other methods which satisfy the intent are acceptable. Bundle hold-downs are not required for fixed tubesheet kettles.

CROSS-SECTION END VIEW
OF TUBE BUNDLE AND SHELL





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5. FLOATING END CONSTRUCTION RCB-5

5.1 INTERNAL FLOATING HEADS (TYPE S & T) RCB-5.2

5.1.1 MINIMUM INSIDE DEPTH OF FLOATING HEAD COVERS

For multipass floating head covers the inside depth shall be such that the minimum cross-over area for flow between successive tube passes is at least equal to 1.3 times the flow area through the tubes of one pass. For single pass floating head covers the depth at nozzle centerline shall be a minimum of one-third the inside diameter of the nozzle.

5.1.2 POSTWELD HEAT TREATMENT

Fabricated floating head covers shall be postweld heat treated when required by the Code or specified by the purchaser.

5.1.3 INTERNAL BOLTING

The materials of construction for internal bolting for floating heads shall be suitable for the mechanical design and similar in corrosion resistance to the materials used for the shell interior.

5.1.4 FLOATING HEAD BACKING DEVICES

The material of construction for split rings or other internal floating head backing devices shall be equivalent in corrosion resistance to the material used for the shell interior.

5.1.5 BACKING DEVICE THICKNESS (TYPE S)

The required thickness of floating head backing devices shall be determined by the following formulas or minimum thickness shown in Figure, using whichever thickness is greatest.

RCB-5.141 BACKING DEVICE THICKNESS (TYPE S)

The required thickness of floating head backing devices shall be determined by the following formulas or minimum thickness shown in Figure RCB-5.141, using whichever thickness is greatest.

BENDING

$$T = \left[\frac{(W)(H)(Y)}{(B)(S)} \right]^{1/2} \text{ For Style "A", "C" & "D", inches}$$

$$\text{Metric } T = \left[\frac{(W)(H)(Y)}{(B)(S)} \right]^{1/2} \times 10^{-3}, \text{ mm}$$

$$T = \left[\frac{2(W)(H)(Y)}{(B)(S)} \right]^{1/2} \text{ For Style "B", inches}$$

$$\text{Metric } T = \left[\frac{2(W)(H)(Y)}{(B)(S)} \right]^{1/2} \times 10^{-3}, \text{ mm}$$

SHEAR

$$t = \frac{W}{(\pi)(Z)(S_s)}, \text{ inches}$$

$$\text{Metric } t = \frac{W}{(\pi)(Z)(S_s)} \times 10^{-6}, \text{ mm}$$

where

A = Ring OD, inches (mm)

W = Design bolt load (as ref. in Code Appendix 2), lb. (kN)

B = As shown in Fig. RCB-5.141, inches (mm)

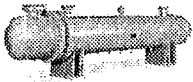
Y = From Code Fig. 2-7.1 using $K = A/B$

C = Bolt circle, inches (mm)

Z = Tubesheet OD, inches (mm)

H = (C - B) / 2, inches (mm)

L = Greater of T or t, inches (mm)



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S = Code allowable stress in tension (using shell design temperature), psi (kPa)

S_{br} = S of backing ring, psi (kPa)

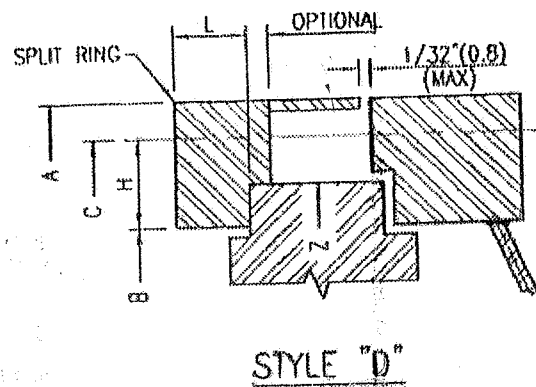
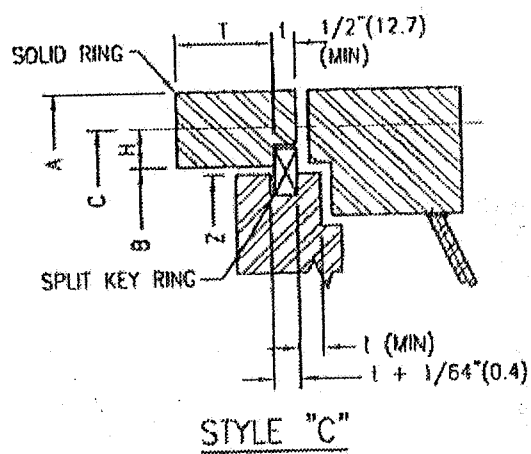
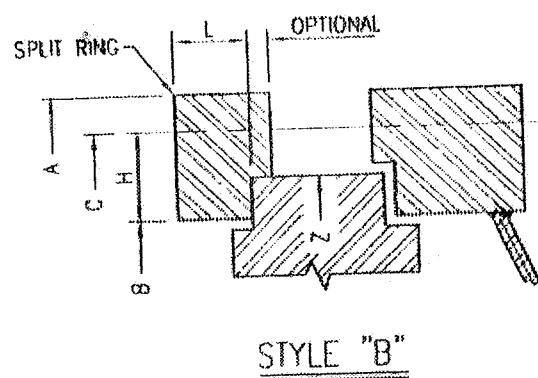
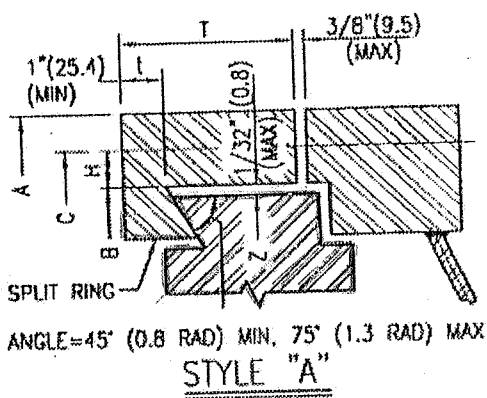
S_{kr} = S of split key ring, psi (kPa)

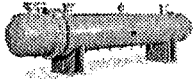
S_{ts} = S of tubesheet, psi (kPa)

$S_s = 0.8 S$, psi (kPa)

NOTES

1. All references above are to ASME Code Section VIII, Division 1.
2. Caution: For styles "A", "B" & "D" check thickness in shear of the tubesheet if $S_{ts} > S_{br}$
3. Caution: Style "C" check thickness in shear of the tubesheet if $5 S_{ts} > S_{kr}$





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5.1.6 TUBE BUNDLE SUPPORTS

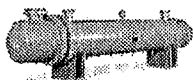
When a removable shell cover is utilized, a partial support plate, or other suitable means, shall be provided to support the floating head end of the tube bundle. If a plate is used, the thickness shall equal or exceed the support plate thickness specified in Table R-4.41 or CB-4.41 as applicable for unsupported tube lengths over 60 inches (1524 mm).

5.1.7 FLOATING HEAD NOZZLES

The floating head nozzle and packing box for a single pass exchanger shall comply with the requirements of Paragraphs RCB-5.21, RCB-5.22 and RCB-5.23.

5.1.8 PASS PARTITION PLATES

The nominal thickness of floating head pass partitions shall be identical to those shown in RCB-9.13 for channels and bonnets.



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5.2 OUTSIDE PACKED FLOATING HEADS (TYPE P) RCB-5.2

5.2.1 PACKED FLOATING HEADS

The cylindrical surface of packed floating head tubesheets and skirts, where in contact with packing (including allowance for expansion), shall be given a fine machine finish equivalent to 63 micro-inches.

5.2.2 PACKING BOXES

A machine finish shall be used on the shell or packing box where the floating tubesheet or nozzle passes through. If packing of braided material is used, a minimum of three rings of packing shall be used for 150 PSI (1034 kPa) maximum design pressure and a minimum of four rings shall be used for 300 PSI (2068 kPa) maximum design pressure. For pressures less than 150 PSI (1034 kPa), temperatures below 300° F (149° C), and non-hazardous service, fewer rings of packing may be used. Figure RCB-5.22 and Table RCB-5.22 show typical details and dimensions of packing boxes.

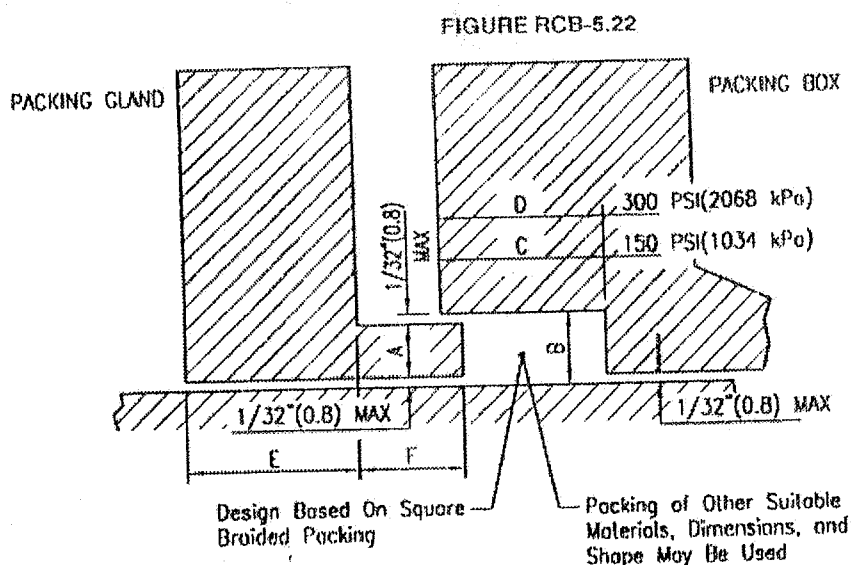
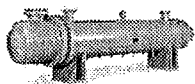


TABLE RCB-5.22
TYPICAL DIMENSIONS FOR PACKED FLOATING HEADS
150 PSI(1034 kPa) AND 300 PSI(2068 kPa) WITH 600 ° F (316 ° C) MAX. TEMP.

Dimensions in Millimeters

SIZE	A	B	C	D	E (MIN)	F	BOLTS	
							NO.	SIZE
152-203	9.53	11.11	31.75	41.28	25.40	19.05	4	M16
229-330	9.53	11.11	31.75	41.28	25.40	19.05	6	M16
356-432	9.53	11.11	31.75	41.28	25.40	19.05	8	M16
457-533	9.53	11.11	31.75	41.28	25.40	19.05	10	M16
559-584	9.53	11.11	31.75	41.28	25.40	19.05	12	M16
610-737	12.70	14.29	44.45	57.15	28.58	25.40	16	M16
762-838	12.70	14.29	44.45	57.15	28.58	25.40	20	M16
864-1092	12.70	14.29	44.45	57.15	28.58	25.40	24	M16
1118-1295	15.88	17.46	53.98	69.85	31.75	31.75	28	M16
1321-1524	15.88	17.46	53.98	69.85	31.75	31.75	32	M16

Note: Nominal size of packing is same as dimension "A"



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5.2.3 PACKING MATERIAL

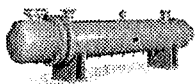
Purchaser shall specify packing material which is compatible with the shell side process conditions.

5.2.4 FLOATING TUBESHEET SKIRT

The floating tubesheet skirt normally shall extend outward. When the skirt must extend inward, a suitable method shall be used to prevent stagnant areas between the shell side nozzle and the tubesheet.

5.2.5 PASS PARTITION PLATES

The nominal thickness of floating head pass partitions shall be identical to those for channels and bonnets.



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5.3 EXTERNALLY SEALED FLOATING TUBESHEET (TYPE W) RCB-5.3

5.3.1 LANTERN RING

The externally sealed floating tubesheet using square braided packing materials shall be used only for water, steam, air, lubricating oil, or similar services. Design temperature shall not exceed 375 ° F (191 ° C). Design pressure shall be limited according to Table RB-5.31.

TABLE RB-5.31
MAXIMUM DESIGN PRESSURE FOR EXTERNALLY SEALED
FLOATING TUBESHEETS

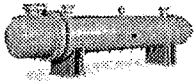
Nominal Shell Inside Diameter Inches (mm)		Maximum Design Pressure PSI (kPa)	
6-24	(152-610)	300	(2068)
25-42	(635-1067)	150	(1034)
43-60	(1092-1524)	75	(517)
61-100	(1549-2540)	50	(345)

5.3.2 LEAKAGE PRECAUTIONS

The design shall incorporate provisions in the lantern ring so that any leakage past the packing will leak to atmosphere. When endless packing rings are used, one ring of packing shall be used on each side of the lantern ring. For braided packing materials with a seam, a minimum of two rings of packing shall be used on each side of the lantern ring, with the seams staggered during assembly.

5.3.3 PACKING MATERIAL

Purchaser shall specify packing material which is compatible with the process conditions.



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6. GASKETS RCB-6

6.1 TYPE OF GASKETS

Gaskets shall be selected which have a continuous periphery with no radial leak paths. This shall not exclude gaskets made continuous by welding or other methods which produce a homogeneous bond.

6.2 GASKET MATERIALS

Metal jacketed or solid metal gaskets shall be used for internal floating head joints, all joints for pressures of 300 psi (2068 kPa) and over, and for all joints in contact with hydrocarbons. Other gasket materials may be specified by agreement between purchaser and manufacturer to meet special service conditions and flange design. When two gasketed joints are compressed by the same bolting, provisions shall be made so that both gaskets seal, but neither gasket is crushed at the required bolt load.

6.3 PERIPHERAL GASKETS

The minimum width of peripheral ring gaskets for external joints shall be $\frac{3}{8}$ " (9.5 mm) for shell sizes through 23 inches (584 mm) nominal diameter and $\frac{1}{2}$ " (12.7 mm) for all larger shell sizes.

The minimum width of peripheral ring gaskets for internal joints shall be $\frac{1}{4}$ " (6.4 mm) for all shell sizes.

Peripheral gasket contact surfaces shall have a flatness tolerance of $\pm \frac{1}{32}$ " (0.8 mm) maximum deviation from any reference plane. This maximum deviation shall not occur in less than a 20° (0.3 Rad) arc.

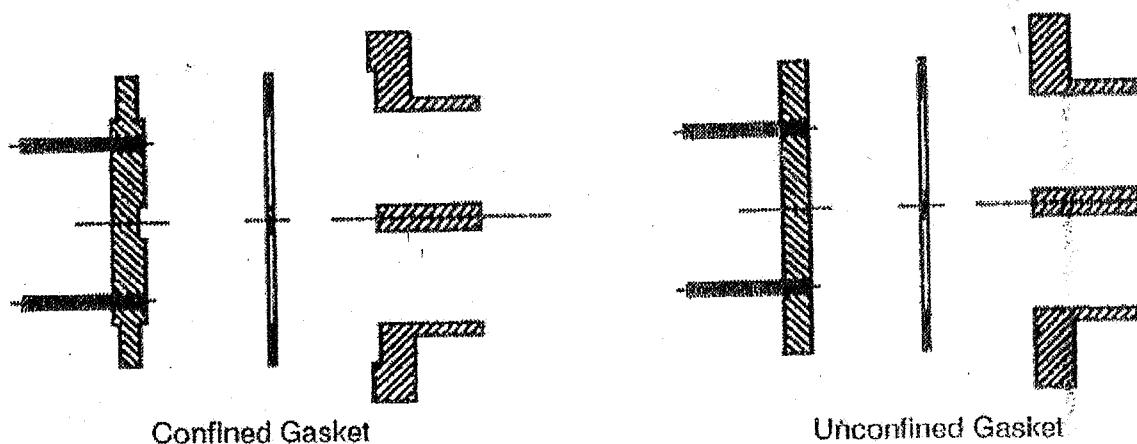
6.4 PASS PARTITION GASKETS

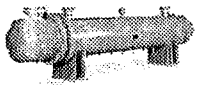
The width of gasket web for pass partitions of channels, bonnets, and floating heads shall be not less than $\frac{1}{4}$ " (6.4 mm) for shell sizes through 23 inches (584 mm) nominal diameter and not less than $\frac{3}{8}$ " (9.5 mm) for all larger shell sizes.

6.5 GASKET JOINT DETAILS

Gasketed joints shall be of a confined type.

FIGURE RCB-6.5





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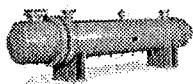
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7. TUBESHEETS

TUBESHEET THICKNESS

Effective Tubesheet Thickness shall be the thickness measured at the bottom of the tube side pass partition groove and/or shell side longitudinal baffle groove minus corrosion allowance in excess of the groove depths.

In no case shall the total thickness minus corrosion allowance, in the areas into which tubes are to be expanded, of any tubesheet be less than the outside diameter of tubes. In no case shall the total tubesheet thickness, including corrosion allowance, be less than 3/4" (19.1 mm).



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RCB-7.132 TUBESHEET FORMULA - BENDING

$$T = \frac{FG}{3} \sqrt{\frac{P}{\eta S}}$$

where

T = Effective tubesheet thickness, inches (mm).

S = Code allowable stress in tension, psi (kPa), for tubesheet material at design metal temperatures. (See Paragraph RCB-1.42).

For outside packed floating head exchangers (Type P), P shall be as defined in Paragraph RCB-7.141, psi (kPa).

For packed floating end exchangers with lantern ring (Type W), for the floating tubesheet, P shall be as defined in Paragraph RCB-7.142, psi (kPa).

For fixed tubesheet exchangers, P shall be as defined in Paragraph RCB-7.163, RCB-7.164 or RCB-7.165, psi (kPa).

For other type exchangers, P shall be the design pressure, shell side or tube side, corrected for vacuum when present on the opposite side, or differential pressure when specified by the purchaser, psi (kPa).

For U-tube tubesheets (Type U), where the tubesheet is extended as a flange for bolting to heads or shells with ring type gaskets, $P = P_s + P_b$ or $P_s + P_h$, depending upon the side under consideration.

$P =$

where

$$P_b = \frac{-6.2 M^*}{F^2 G^3}$$

and M^* is defined in Paragraph RCB-7.1342, psi (kPa).

For floating tubesheets (Type T), where the tubesheet is extended for bolting to heads with ring type gaskets, the effect of the moment acting upon the extension is defined in Paragraph RCB-7.162 in terms of equivalent tube side and shell side bolting pressures except G shall be the gasket G of the floating tubesheet. P psi (kPa) is given by the greatest absolute value of the following:

$$P = P_t + P_{Et}$$

$$\text{or } P = P_s - P_{Es}$$

$$\text{or } P = P_t$$

$$\text{or } P = P_s$$

G shall be either in the corroded or uncorroded condition, dependent upon which condition is under consideration.

For fixed tubesheet exchangers, G shall be the shell inside diameter.

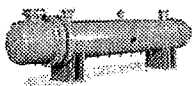
For kettle type exchangers, G shall be the port inside diameter.

For any floating tubesheet (except divided), G shall be the G used for the stationary tubesheet using the P as defined for other type exchangers.

Type T tubesheets shall also be checked using the pressure P defined above with bolting and using the actual gasket G of the floating tubesheet

$G =$ For a divided floating tubesheet, G shall be $1.41(d)$ where d is the length of the shortest span measured over centerlines of gaskets.

For other type exchangers, G shall be the diameter, inches (mm), over which the pressure under consideration is acting. (e.g.: Pressure acting on the gasketed side of a tubesheet, G = the diameter at the location of the gasket load reaction as defined in the Code. Pressure acting on an integral side of a tubesheet, G = the inside diameter of the integral pressure part.)



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$$\eta = \begin{cases} 1 - \frac{0.785}{\left(\frac{\text{Pitch}}{\text{Tube OD}}\right)^2} & \text{for square or rotated square tube patterns} \\ 1 - \frac{0.907}{\left(\frac{\text{Pitch}}{\text{Tube OD}}\right)^2} & \text{for triangular or rotated triangular tube patterns} \end{cases}$$

For integrally finned tubes, the OD of the tube in the tubesheet shall be used.

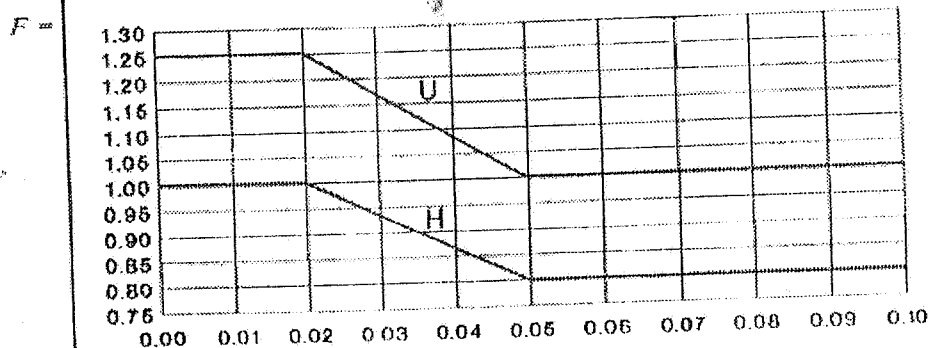
For unsupported tubesheets (e.g.: U-tube tubesheets) gasketed both sides, $F = 1.25$.

For supported tubesheets (e.g.: fixed tubesheets and floating type tubesheets) gasketed both sides, $F = 1.0$.

For unsupported tubesheets (e.g.: U-tube tubesheets) integral with either or both sides, F shall be the value determined by the curve U in Figure RCB-7.132.

For supported tubesheets (e.g.: fixed tubesheets and floating type tubesheets) integral with either or both sides, F shall be the value determined by the curve H in Figure RCB-7.132.

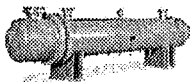
FIGURE RCB-7.132



Wall Thickness/ID Ratio For Integral Tubesheets

NOTE: If the tubesheet is integral with both the tube side and shell side, Wall Thickness and ID are to be based on the side yielding the smaller value of F .

See Table RCB-7.132 for illustration of the application of the above equations.



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TABLE RCB - 7.132

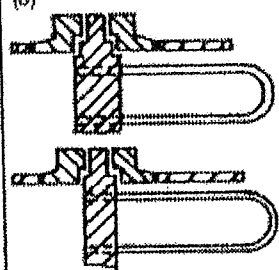
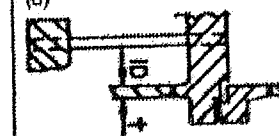
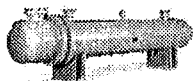
TUBESHEET THICKNESS FOR BENDING				
$T = \frac{FG}{3} \sqrt{\frac{P}{\eta S}}$		Note: Must be calculated for shell side or tube side pressure, whichever is controlling.		
For $\square$ $\diamond$ Tube pattern For integrally finned tubes, the OD of the tube in the tubesheet shall be used	$\eta = 1 - \left[\frac{0.785}{\left(\frac{P \times S}{F \times G} \right)^2} \right]$	For $\triangle$ $\triangle$ Tube pattern For integrally finned tubes, the OD of the tube in the tubesheet shall be used	$\eta = 1 - \left[\frac{0.907}{\left(\frac{P \times S}{F \times G} \right)^2} \right]$	S = Code allowable stress in tension, psi (kPa), for tubesheet material at design metal temperature. (See Paragraph RCB-1.42.)
	F	G		P
		Shell Side Pressure	Tube Side Pressure	
(a) 	1.0	Gasket G shell side See note 1	Gasket G tube side See note 1	Design pressure, psi (kPa), shell side or tube side, per Paragraph RCB-7.132 corrected for vacuum when present on opposite side or differential pressure when specified by customer.
(b) 	1.25	Gasket G shell side See note 1	Gasket G tube side See note 1	Design pressure, psi (kPa), shell side or tube side, per Paragraph RCB-7.132 corrected for vacuum when present on opposite side or differential pressure when specified by customer.
(c) 	See Figure RCB-7.132 $F = \frac{17 - 100 \left(\frac{1}{ID} \right)}{15}$ Note: F Max = 1.0 F Min = 0.8	Gasket G shell side See note 1	Channel ID	Design pressure, psi (kPa), shell side or tube side, per RCB-7.132 corrected for vacuum when present on opposite side or differential pressure when specified by customer, or fixed tubesheet type units, as defined in Paragraphs RCB-7.163 thru RCB-7.165
(d) 		Shell ID or port inside diameter for kettle type exchangers	Gasket G (shell ID if fixed tubesheet type unit) See note 1	
(e) 		Shell ID or port inside diameter for kettle type exchangers	Channel ID (shell ID if fixed tubesheet type unit)	

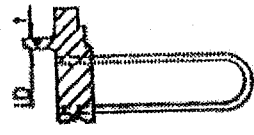
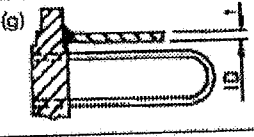
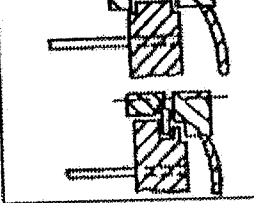
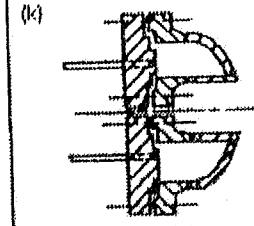
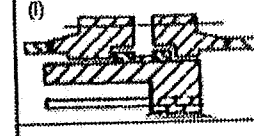
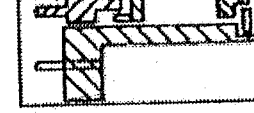
Table RCB - 7.132 continued next page



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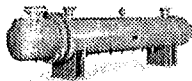
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TABLE RCB - 7.132 (Continued)

	<i>F</i>	<i>G</i>		<i>P</i>
		Shell Side Pressure	Tube Side Pressure	
(f) 	See Figure RCB-7.132 $F = \frac{17 - 100 \left(\frac{t}{id} \right)}{12}$ Note: <i>F</i> Max = 1.25 <i>F</i> Min = 1.00	Gasket <i>G</i> shell side See note 1	Channel ID	Design pressure, psi (kPa), shell side, or tube side, per Paragraph RCB-7.132 corrected for vacuum when present on opposite side, or differential pressure when specified by customer.
(g) 		Shell ID or port inside diameter for kettle type exchangers	Gasket <i>G</i> tube side See note 1	
(h) 		Shell ID or port inside diameter for kettle type exchangers	Channel ID	
(i) 	1.0	Same <i>G</i> as used for stationary tubesheet		Design pressure, psi (kPa), shell side, or tube side, per Paragraph RCB-7.132 corrected for vacuum when present on opposite side, or differential pressure when specified by customer.
(j) 	1.0	Same <i>G</i> as used for stationary tubesheet. Also check using gasket <i>G</i> of the floating tubesheet See note 1		See Paragraph RCB-7.132
(k) 	1.0	$G = 1.41 (cl)$ <i>cl</i> - Shortest span measured over center lines of gaskets.		Design pressure, psi (kPa), shell side, or tube side, per Paragraph RCB-7.132 corrected for vacuum when present on opposite side, or differential pressure when specified by customer.
(l) 	1.0	Same <i>G</i> as used for stationary tubesheet		Design pressure, psi (kPa), tube side per paragraph RCB-7.132 corrected for vacuum when present on the shell side.
(m) 	1.0	Same <i>G</i> as used for stationary tubesheet		Defined in Paragraph RCB-7.1411

Notes:

1. Gasket *G* = the diameter at the location of the gasket load reaction as defined in the Code.



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RCB-7.133 TUBESHEET FORMULA - SHEAR

$$T = \frac{0.31 D_t \left(\frac{P}{S} \right)}{\left(1 - \frac{d_o}{Pitch} \right)}$$

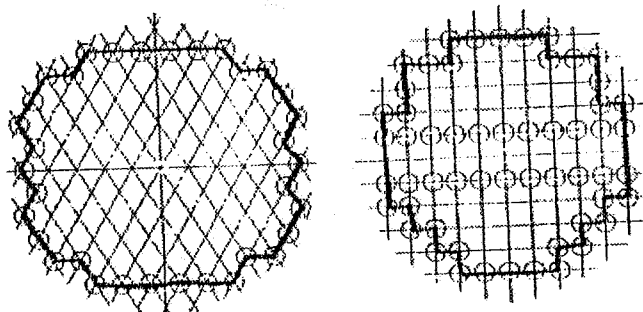
where

T = Effective tubesheet thickness, inches (mm)

$D_t = \frac{4A}{C}$ = Equivalent diameter of the tube center limit perimeter, inches (mm)

C = Perimeter of the tube layout measured stepwise in increments of one tube pitch from center-to-center of the outermost tubes, inches (mm). Figure RCB-7.133 shows the application to typical triangular and square tube patterns

FIGURE RCB-7.133



* C^* (perimeter) is the length of the heavy line

A = Total area enclosed by perimeter C , square inches (mm²)

d_o = Outside tube diameter, inches (mm), for integrally finned tubes, the OD of the tube in the tubesheet shall be used.

$Pitch$ = Tube center-to-center spacing, inches (mm)

For outside packed floating head exchangers (Type P), P shall be as defined in Paragraph RCB-7.141, psi (kPa).

P = For fixed tubesheet exchangers, P shall be as defined in Paragraphs RCB-7.163, RCB-7.164 or RCB-7.165, psi (kPa).

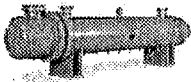
For other type exchangers, P shall be the design pressure, psi (kPa), shell side or tube side, corrected for vacuum when present on the opposite side, or differential pressure when specified by the purchaser.

S = Code allowable stress in tension, psi (kPa), for tubesheet material at design metal temperature. (See Paragraph RCB-1.42.)

NOTE: Shear will not control when

$$\frac{P}{S} < 1.6 \left(1 - \frac{d_o}{Pitch} \right)^2$$

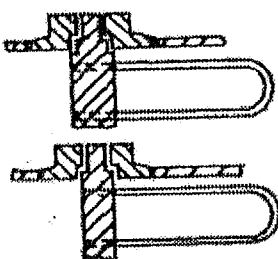
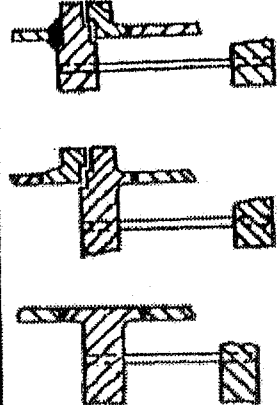
See Table RCB-7.133 for illustration of the application of the above equations.

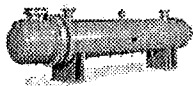


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TABLE RCB-7.133

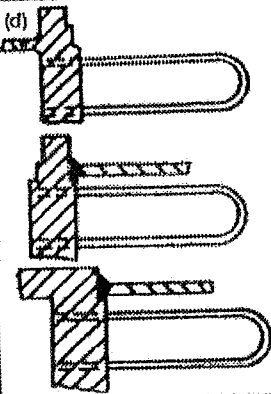
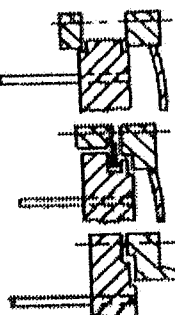
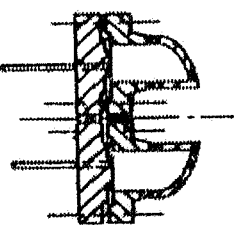
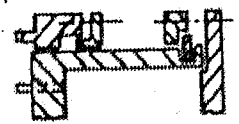
TUBESHEET THICKNESS FOR SHEAR		
$T = \left[\frac{0.31 D_L}{\left(1 - \frac{v_p}{E_p}\right)} \right] \left(\frac{P}{S} \right)$		Note: Must be calculated for shell side or tube side pressure, whichever is controlling.
d_o = Outside tube diameter, inches (mm). For integrally finned tubes, the OD of the tube in the tubesheet shall be used.	Pitch = Tube spacing, center-to-center, inches (mm).	S = Code allowable stress in tension, psi (kPa), for tubesheet material at design metal temperature. (See paragraph RCB-1.42.)
P		D_L
(a) 	Design pressure, psi (kPa), shell side or tube side, corrected for vacuum when present on opposite side, or differential pressure when specified by customer	$D_L = 4 \left(\frac{A}{C} \right)$
(b) 	Design pressure, psi (kPa), shell side or tube side, corrected for vacuum when present on opposite side, or differential pressure when specified by customer	C = Perimeter of tube layout measured stepwise in increments of one tube-to-tube pitch center-to-center of the outermost tubes, in inches (mm). See Figure RCB-7.133 A = total area enclosed by C in square inches (mm ²). See Figure RCB-7.133
(c) 	Design pressure, psi (kPa), shell side or tube side, corrected for vacuum when present on opposite side, or differential pressure when specified by customer, or for fixed tubesheet type units, as defined in paragraphs RCB-7.163 thru RCB-7.165	

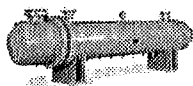


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TABLE RCB-7.133 Continued

	P	D_t
(d) 	Design pressure, psi (kPa), shell side or tube side, corrected for vacuum when present on opposite side, or differential pressure when specified by customer	$D_t = \sqrt{\frac{A}{C}}$
(e) 	Design pressure, psi (kPa), Shell side or tube side, corrected for vacuum when present on opposite side, or differential pressure when specified by customer	$C =$ Perimeter of tube layout measured stepwise in increments of one tube-to-tube pitch center-to-center of the outermost tubes, in inches (mm). See Figure RCB-7.133 $A =$ total area enclosed by C in square inches (mm ²). See Figure RCB-7.133
(f) 	Design pressure, psi (kPa), shell side or tube side, corrected for vacuum when present on opposite side, or differential pressure when specified by customer	
(g) 	Design pressure, psi (kPa), tube side, corrected for vacuum when present on the shell side	
(h) 	Defined in Paragraph RCB-7.1412	



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RCB-7.134 TUBESHEET FORMULA – TUBESHEET FLANGED EXTENSION

This paragraph is applicable only when bolt loads are transmitted, at the bolt circle, to the extended portion of a tubesheet. The peripheral portion extended to form a flange for bolting to heads or shells with ring type gaskets may differ in thickness from that portion inside the shell calculated in Paragraph RCB-7.132. The minimum thickness of the extended portion may be calculated from the following paragraphs.

RCB-7.1341 FIXED TUBESHEET OR FLOATING TUBESHEET EXCHANGERS

$$T_r = 0.98 \left[\frac{M (r^2 - 1 + 3.71 r^2 \ln r)}{S (A - G)(1 + 1.86 r^2)} \right]^{1/2}$$

where

T_r = Minimum thickness of the extended portion, inches (mm)

A = Outside diameter of the tubesheet, inches (mm)

$$r = \frac{A}{G}$$

M = the larger of M_1 or M_2 as defined in Paragraph RCB-7.162

Note: The moments may differ from the moments acting on the attached flange.

S and G are defined in Paragraph RCB-7.132

RCB-7.1342 U-TUBE TUBESHEET EXCHANGERS

$$T_r = 1.38 \left[\frac{M^* + M + 0.39 \frac{P}{S} G^2 w}{(A - G)} \right]^{1/2}$$

where

T_r = Minimum thickness of the extended portion, inches (mm)

$$M^* = \frac{\frac{0.669}{\eta} w F^2 P C^3 \left(\frac{T_r}{T} \right)^3 - M G - 0.39 w P C^3}{G + \frac{1.37}{\eta} \left(\frac{T_r}{T} \right)^3 w}$$

T = Effective tubesheet thickness calculated from Paragraph RCB-7.132, inches (mm)

$$w = \frac{(A - G)}{2}$$

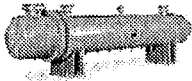
M = the larger of M_1 or M_2 as defined in Paragraph RCB-7.162

Note: The moments may differ from the moments acting on the attached flange.

F , G and η are defined in Paragraph RCB-7.132

P = P_s or P_r or maximum differential pressure, as applicable.

Note: See Paragraph RCB-7.13421 for procedure.



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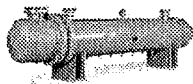
RCB-7.13421 ITERATIVE CALCULATION METHODS

Method 1

- (1) Calculate M^* assuming $T_r = T$.
- (2) Calculate P_b , then P from Paragraph RCB-7.132.
- (3) Calculate T from Paragraph RCB-7.132.
- (4) Calculate T_r from Paragraph RCB-7.1342.
- (5) Compare T and T_r ; if T is greater than T_r , calculation is terminated. Use T_r calculated. Do not proceed to Step (6).
- (6) If T_r is greater than T , or if it is desired to reduce T_r below T , select a new ratio of T_r/T that is less than 1 and repeat Steps (1) through (5). (Note: T_r/T ratio is calculated using actual corroded thickness of the part).

Method 2 - (ALTERNATIVE METHOD)

- (1) Set $M^* = -M$.
- (2) Calculate P_b , then P from Paragraph RCB-7.132.
- (3) Calculate T from Paragraph RCB-7.132.
- (4) Calculate T_r from Paragraph RCB-7.1342.
- (5) Recalculate $M^* = -M$ using values of T and T_r obtained in Steps (3) and (4) and as defined in Paragraph RCB-7.1342. (Note T_r/T must be ≤ 1).
- (6) If $|M^*|$ obtained in Step (5) is less than $|M|$ from Step (1), calculation is terminated. Use T_r calculated in Step (4). Do not proceed to Step (7).
- (7) If $|M^*|$ obtained from Step (5) is greater than $|M|$ from Step (1), repeat Step (2) using M^* calculated in Step (5). Then repeat Steps (3) through (5).
- (8) If last calculated $|M^*|$ is less than the previous $|M^*|$ used to calculate P_b , calculation is terminated. Use last calculated value of T_r .
- (9) If last calculated $|M^*|$ is greater than the previous $|M^*|$ used to calculate P_b , repeat Step (2) using last calculated M^* . Then repeat Steps (3) through (5). Continue this process until Step (8) is satisfied.



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RCB-7.14 PACKED FLOATING TUBESHEET TYPE EXCHANGERS EFFECTIVE PRESSURE

RCB-7.141 OUTSIDE PACKED FLOATING HEAD (TYPE P)

The thickness of tubesheets in exchangers whose floating heads are packed at the outside diameter of the tubesheet or a cylindrical extension thereof shall be calculated like stationary tubesheets using the formulas for P as defined below.

RCB-7.1411 EFFECTIVE DESIGN PRESSURE - BENDING

The effective design pressure to be used with the formula shown in Paragraph RCB-7.132 is given by:

$$P = P_i + P_s \left[\frac{1.25(D^2 - D_c^2)(D - D_c)}{DF^2G^2} \right]$$

where

P_i = Design pressure, psi (kPa), tube side
(For vacuum design, P_i is negative.)

P_s = Design pressure, psi (kPa), shell side
(For vacuum design, P_s is negative.)

D = Outside diameter of the floating tubesheet, inches (mm)

$D_c = \sqrt{\frac{4A}{\pi}}$ Equivalent diameter of the tube center limit perimeter, inches (mm), using A as defined in Paragraph RCB-7.133

F and G are as defined in Paragraph RCB-7.132

RCB-7.1412 EFFECTIVE DESIGN PRESSURE - SHEAR

The effective design pressure to be used with the formula shown in Paragraph RCB-7.133 is given by:

$$P = P_i + P_s \left(\frac{D^2 - D_c^2}{D_c^2} \right)$$

using terms as defined in Paragraph RCB-7.1411.

RCB-7.142 PACKED FLOATING TUBESHEET WITH LANTERN RING (TYPE W)

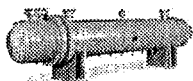
The thickness of floating tubesheets in exchangers whose floating tubesheets are packed at the outside diameter with return bonnet or channel bolted to the shell flange, shall be calculated as for gasketed stationary tubesheet exchangers, using P defined as the tube side design pressure, psi (kPa), corrected for vacuum when present on the shell side. It is incorrect to utilize the shell side pressure.

RCB-7.15 DOUBLE TUBESHEETS

Double tubesheets may be used where the operating conditions indicate their desirability. The diversity of construction types makes it impractical to specify design rules for all cases. Paragraphs RCB-7.154, RCB-7.155 and RCB-7.156 provide the design rules for determining the thickness of double tubesheets for some of the most commonly used construction types.

RCB-7.151 MINIMUM THICKNESS

Neither component of a double tubesheet shall have a thickness less than that required by Paragraph RCB-7.131.



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RCB-7.1542 INTERCONNECTING ELEMENT DESIGN - SHEAR

The radial shear stress (τ), psi (kPa), at attachment due to differential thermal expansion of tubesheets shall not exceed 80% of the lower Code allowable stress (S) of either of the tubesheet materials or the interconnecting element at their respective design temperature. The shear is defined as:

$$\tau = \frac{F_E}{t_E} \leq 0.8S$$

$$\text{(Metric)} \quad \tau = \frac{F_E}{t_E} \times 10^{-6} \leq 0.8S$$

t_E = Thickness of interconnecting element, inches (mm).

where

$$F_E = \left| \frac{(\alpha_1 \Delta T_1 - \alpha_2 \Delta T_2)(t_1 E_1)(t_2 E_2)}{(t_1 E_1) + (t_2 E_2)} \right|$$

$$\text{(Metric)} \quad F_E = \left| \frac{(\alpha_1 \Delta T_1 - \alpha_2 \Delta T_2)(t_1 E_1)(t_2 E_2)}{(t_1 E_1) + (t_2 E_2)} \right| \times 10^{-6}$$

where

F_E = Force per unit measure due to differential radial expansion, lbf/in (kN/mm).

E_1 = Modulus of Elasticity of tubesheet 1 at mean metal temperature, psi (kPa).

E_2 = Modulus of Elasticity of tubesheet 2 at mean metal temperature, psi (kPa).

α_1 = Coefficient of thermal expansion for tubesheet 1 at mean metal temperature, inches/inch/°F (mm/mm/°C).

α_2 = Coefficient of thermal expansion for tubesheet 2 at mean metal temperature, inches/inch/°F (mm/mm/°C).

ΔT_1 = Difference in temperature from ambient conditions to mean metal temperature for tubesheet 1, °F (°C).

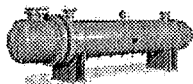
ΔT_2 = Difference in temperature from ambient conditions to mean metal temperature for tubesheet 2, °F (°C).

RCB-7.1543 INTERCONNECTING ELEMENT DESIGN - BENDING AND TENSILE

The combined stresses from bending due to differential thermal expansion of tubesheets and axial tension due to thermal expansion of tubes shall not exceed 1.5 times the Code allowable stress (S) of the interconnecting element.

The combined total stress of interconnecting element (σ_E), psi (kPa), is given by:

$$\sigma_E = \sigma_B + \sigma_{TE} \leq 1.5S$$



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RCB-7.152 VENTS AND DRAINS

Double tubesheets of the edge welded type shall be provided with vent and drain connections at the high and low points of the enclosed space.

RCB-7.153 SPECIAL PRECAUTIONS

When double tubesheets are used, special attention shall be given to the ability of the tubes to withstand, without damage, the mechanical and thermal loads imposed on them by the construction.

RCB-7.154 INTEGRAL DOUBLE TUBESHEETS

The tubesheets are connected in a manner which distributes axial load and radial thermal expansion loads between tubesheets by means of an interconnecting element capable of preventing individual radial growth of tubesheets. It is assumed that the element is rigid enough to mutually transfer all thermal and mechanical radial loads between the tubesheets. Additionally, it is understood that the tubes are rigid enough to mutually transfer all mechanical and thermal axial loads between the tubesheets.

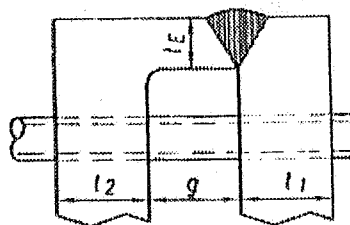


FIGURE RCB-7.154

RCB 7.1541 TUBESHEET THICKNESS

Calculate the total combined tubesheet thickness (T) per Paragraph RCB-7.13.

where

- T = Greater of the thickness, inches (mm), resulting from Paragraphs RCB-7.132 or RCB-7.133 using the following variable definitions:
- G = Per Paragraph RCB-7.13, inches (mm), using worst case values of shell side or tube side tubesheets at their respective design temperature.
- S = Lower of the Code allowable stress, psi (kPa), for either component tubesheet at its respective design temperature.
- F = Per Paragraph RCB-7.13, using worst case values of shell side or tube side tubesheets at their respective design temperature.

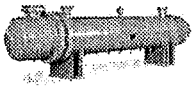
All other variables are per Paragraph RCB-7.13.

Establish the thickness of each individual tubesheet so that $t_2 + t_1 \geq T$ and the minimum individual tubesheet thicknesses (t_1 and t_2) shall be the greater of Paragraphs RCB-7.13 or RCB.7.134, as applicable.

where

t_1 = Thickness of tube side tubesheet, inches (mm).

t_2 = Thickness of shell side tubesheet, inches (mm).



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The stress due to axial thermal expansion of tubes (σ_{TE}), psi (kPa), is defined as:

$$\sigma_{TE} = \left| \frac{F_{TE}}{A_E} \right|$$

$$\text{(Metric)} \quad \sigma_{TE} = \left| \frac{F_{TE}}{A_E} \right| \times 10^6$$

where

$$F_{TE} = \frac{(\alpha_T \Delta T_T - \alpha_E \Delta T_E)(E_T A_T)(E_E A_E)}{(E_T A_T) + (E_E A_E)}$$

$$\text{(Metric)} \quad F_{TE} = \frac{(\alpha_T \Delta T_T - \alpha_E \Delta T_E)(E_T A_T)(E_E A_E)}{(E_T A_T) + (E_E A_E)} \times 10^{-6}$$

The stress due to bending caused by differential thermal expansion of tubesheets σ_B , psi (kPa), is defined as:

$$\sigma_B = \frac{6 M_B}{t_E^2}$$

$$\text{(Metric)} \quad \sigma_B = \frac{6 M_B}{t_E^2} \times 10^6$$

The bending moment is defined as:

$$M_B = \frac{F_{TE} g}{2}$$

where

M_B = Bending moment per unit measure acting on interconnecting element, inch-pounds per inch (mm-kN/mm).

g = Spacing between tubesheets, inches (mm). The spacing between tubesheets for an integral double tubesheet is left to the discretion of the manufacturer. For other types of double tubesheets, the minimum spacing is determined in accordance with Paragraphs RCB-7.1552 or RCB-7.1562, as applicable.

α_T = Coefficient of thermal expansion of tubes at mean metal temperature, inches/inch/°F (mm/mm/°C).

α_E = Coefficient of thermal expansion of interconnecting element at mean metal temperature, inches/inch/°F (mm/mm/°C).

ΔT_T = Difference in temperature from ambient conditions to mean metal temperature for tubes, °F (°C).

ΔT_E = Difference in temperature from ambient conditions to mean metal temperature for interconnecting element, °F (°C).

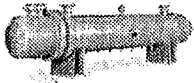
E_T = Modulus of Elasticity of tubes at mean metal temperature, psi (kPa).

E_E = Modulus of Elasticity of interconnecting element at mean metal temperature, psi (kPa).

A_T = Total cross sectional area of tubes between tubesheets, square inches (mm²).

A_E = Total cross sectional area of interconnecting element, square inches (mm²).

F_{TE} = Resultant force due to the difference in thermal expansion between tubes and element, lbf (kN).



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RCB-7.1544 TUBE STRESS CONSIDERATION – AXIAL STRESS

The axial stresses in the tubes due to thermal expansion and pressure load shall not exceed the Code allowable stress (S) of the tubes at design temperature.

The total combined stress of the tubes (σ_T), psi (kPa), is given by:

$$\sigma_T = \sigma_P + \sigma_{TE} \leq S$$

The axial stress due to pressure (σ_P), psi (kPa), is defined as:

$$\sigma_P = \frac{P \pi (G^2 - N d_o^2)}{4 A_T}$$

where

P = Greater of shell side or tube side design pressure, psi (kPa).

G = Per Paragraph RCB-7.13, inches (mm).

N = Number of tubes.

d_o = Tube OD between tubesheets, inches (mm).

The stress due to axial thermal expansion of tubes (σ_{TE}), psi (kPa), is defined by:

$$\sigma_{TE} = \frac{F_{TE}}{A_T}$$

$$\text{(Metric)} \quad \sigma_{TE} = \frac{F_{TE}}{A_T} \times 10^6$$

RCB-7.155 CONNECTED DOUBLE TUBESHEETS

The tubesheets are connected in a manner which distributes axial load between tubesheets by means of an interconnecting cylinder. The effect of the differential radial growth between tubesheets is a major factor in tube stresses and spacing between tubesheets. It is assumed the interconnecting cylinder and tubes are rigid enough to mutually transfer all mechanical and thermal axial loads between the tubesheets.

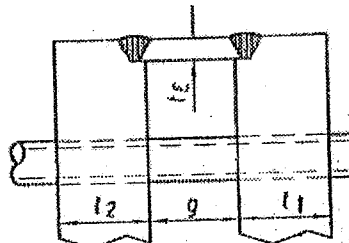
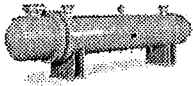


FIGURE RCB-7.155

RCB-7.1551 TUBESHEET THICKNESS

Calculate the total combined tubesheet thickness (T) per Paragraph RCB-7.13.



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where

T = Greater of the thickness, inches (mm), resulting from Paragraphs RCB-7.132 or RCB-7.133 using variables as defined in Paragraph RCB-7.1541.

Establish the thickness of each individual tubesheet so that $t_2 + t_1 \geq T$ and the minimum individual tubesheet thickness (t_1 and t_2) shall be the greater of Paragraph RCB-7.13 or RCB 7.134, when applicable.

where

t_1 = Thickness of tube side tubesheet, inches (mm).

t_2 = Thickness of shell side tubesheet, inches (mm).

RCB-7.1552 MINIMUM SPACING BETWEEN TUBESHEETS

The minimum spacing (g), inches (mm), between tubesheets required to avoid overstress of tubes resulting from differential thermal growth of individual tubesheets is given by:

$$g = \sqrt{\frac{d_o \Delta r E_T}{0.27 Y_T}}$$

where

d_o = Tube OD between tubesheets, inches (mm).

Y_T = Yield strength of the tube material at maximum metal temperature, psi (kPa).

Δr = Differential radial expansion between adjacent tubesheets, inches (mm). (Measured from center of tubesheet to D_{TL}).

$$\Delta r = \left(\frac{D_{TL}}{2} \right) (\alpha_2 \Delta T_2 - \alpha_1 \Delta T_1)$$

where

D_{TL} = Outer tube limit, inches (mm).

RCB-7.1553 INTERCONNECTING ELEMENT DESIGN - AXIAL STRESS

The interconnecting element axial stress (σ_{TE}), psi (kPa), due to the thermal expansion of the tubes shall not exceed the Code allowable stress (S) of the interconnecting element at design temperature. The axial stress is defined as:

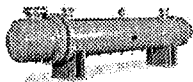
$$\sigma_{TE} = \frac{F_{TE}}{A_E}$$

$$\text{(Metric)} \quad \sigma_{TE} = \frac{F_{TE}}{A_E} \times 10^6$$

where

$$F_{TE} = \frac{(\alpha_T \Delta T_T - \alpha_E \Delta T_E)(E_T A_T)(E_E A_E)}{(E_T A_T) + (E_E A_E)}$$

$$\text{(Metric)} \quad F_{TE} = \frac{(\alpha_T \Delta T_T - \alpha_E \Delta T_E)(E_T A_T)(E_E A_E)}{(E_T A_T) + (E_E A_E)} \times 10^{-6}$$



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RCB-7.1554 TUBE STRESS CONSIDERATION – AXIAL STRESS

The axial stresses in the tubes due to thermal expansion and pressure load shall not exceed the Code allowable stress (S) of the tubes at design temperature.

The total combined stress of tubes (σ_T), psi (kPa), is given by:

$$\sigma_T = \sigma_P + \sigma_{TT} \leq S$$

The axial stress due to pressure (σ_P), psi (kPa), is defined as:

$$\sigma_P = \frac{P \pi (G^2 - N d_o^2)}{4 A_T}$$

where

P = Greater of shell side or tube side design pressure, psi (kPa).

G = Per Paragraph RCB-7.13, inches (mm).

N = Number of tubes.

d_o = Tube OD between tubesheets, inches (mm).

The stress due to axial thermal expansion of tubes (σ_{TT}), psi (kPa), is determined by:

$$\sigma_{TT} = \frac{F_{TE}}{A_T}$$

$$\text{(Metric)} \quad \sigma_{TT} = \frac{F_{TE}}{A_T} \times 10^6$$

RCB-7.156 SEPARATE DOUBLE TUBESHEETS

The tubesheets are connected only by the interconnecting tubes. The effect of differential radial growth between tubesheets is a major factor in tube stresses and spacing between tubesheets. It is assumed that no loads are transferred between the tubesheets.

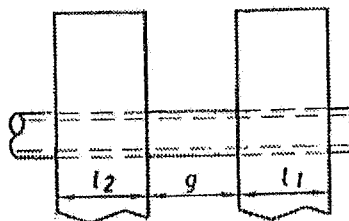
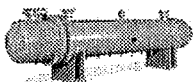


FIGURE RCB-7.156



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RCB-7.1561 TUBESHEET THICKNESS

Calculate tube side tubesheet thickness per Paragraph RCB-7.13. Use all variables as defined per TEMA, neglecting all considerations of shell side design conditions.

Calculate shell side tubesheet thickness per Paragraph RCB-7.13. Use all variables as defined per TEMA, neglecting all considerations of tube side design conditions.

RCB-7.1562 MINIMUM SPACING BETWEEN TUBESHEETS

The minimum spacing (g), inches (mm), between tubesheets required to avoid overstress of tubes resulting from differential thermal growth of individual tubesheets is given by:

$$g = \sqrt{\frac{d_o \Delta T E_T}{0.27 Y_T}}$$

RCB-7.16 FIXED TUBESHEET EFFECTIVE PRESSURE

This paragraph shall apply to exchangers having tubesheets fixed to both ends of the shell, with or without a shell expansion joint except as required or permitted by Paragraph RCB-7.3. Both tubesheets of fixed tubesheet exchangers shall have the same thickness, unless the provisions of Paragraph RCB-7.166 are satisfied.

For fixed tubesheet exchangers, the mutually interdependent loads exerted on the tubesheets, tubes, and shell are defined in terms of equivalent and effective design pressures in Paragraphs RCB-7.161 through RCB-7.165 for use in Paragraphs RCB-7.132 and RCB-7.133. These pressures shall also be used (with $J = 1$) in Paragraphs RCB-7.22, RCB-7.23 and RCB-7.25 to assess the need for an expansion joint. The designer shall consider the most adverse operating conditions specified by the purchaser. (See Paragraph E-3.2.)

RCB-7.161 EQUIVALENT DIFFERENTIAL EXPANSION PRESSURE

The pressure due to differential thermal expansion, psi (kPa), is given by:

$$P_d = \frac{4 J E_s t_s \left(\frac{\Delta T}{L} \right)}{(D_o - 3t_s) (1 + J K F_q)}$$

Note: Algebraic sign must be retained for use in Paragraphs RCB-7.163 through RCB-7.166, RCB-7.22 and RCB-7.23.

where

$J = 1.0$ for shells without expansion joints

$J = \frac{S_j L}{S_j L + \pi (D_o - t_s) t_s E_s}$ for shells with expansion joints. See Note (1)

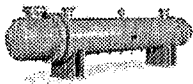
S_j = Spring rate of the expansion joint, lbs/inch (kN/mm)

$$K = \frac{E_s t_s (D_o - t_s)}{E_t t_t N (d_o - t_t)}$$

$$F_q = 0.25 + (F - 0.6) \left[\frac{300 t_s E_s \left(\frac{G}{T} \right)^3}{K L E} \right]^{1/4}$$

(Use the calculated value of F_q or 1.0, whichever is greater.)

F and G are as defined in Paragraph RCB-7.132.



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RCB-7.1554 TUBE STRESS CONSIDERATION – AXIAL STRESS

T = Tubesheet thickness used, but not less than 98.5% of the greater of the values defined by paragraph RCB-7.132 or RCB-7.133. (the value assumed in evaluating F_a must match the final computed value within a tolerance of $\pm 1.5\%$) See note (2).

L = Tube length between inner tubesheet faces, inches (mm).

ΔL = Differential thermal growth (shell - tubes), inches (mm) (See Section 7, Paragraph T-4.5).

L_t = Tube length between outer tubesheet faces, inches (mm).

E_s = Elastic modulus of the shell material at mean metal temperature, psi (kPa). (See Paragraph RCB-1.431). See Note (3).

E_t = Elastic modulus of the tube material at mean metal temperature, psi (kPa). (See Paragraph RCB-1.432).

E = Elastic modulus of the tubesheet material at mean metal temperature, psi (kPa). (See Paragraph RCB-1.432).

N = Number of tubes in the shell.

D_o = Outside diameter of the shell or port for kettle type exchangers, inches (mm).

d_o = Outside diameter of the tubes (for integrally finned tubes, d_o is root diameter of fin), inches (mm).

t_t = Tube wall thickness (for integrally finned tubes, t_t is wall thickness under fin), inches (mm).

t_s = Shell wall thickness, inches (mm).

Notes:

(1) J can be assumed equal to zero for shells with expansion joints where

$$S_j < \frac{(D_o - t_s)t_s E_s}{10L}$$

(2) Tubesheets thicker than computed are permissible provided neither shell nor tubes are overloaded. See Paragraph RCB-7.2.

(3) For Kettle type,

$$E_s = \frac{E_{sn} L}{(2L_p) + [(4L_c T_c D_p) / ((D_p + D_k) T_c)] + [(L_k T_p D_p) / (D_k T_k)]}$$

where

E_{sn} = Elastic modulus of the shell material at mean metal temperature, psi (kPa). (See Paragraph RCB-1.431).

L = Tube length between inner tubesheet faces, inches (mm).

L_p = Length of kettle port cylinder, inches (mm).

T_p = Kettle port cylinder thickness, inches (mm).

D_p = Mean diameter of kettle port cylinder, inches (mm).

L_k = Length of kettle cylinder, inches (mm).

T_k = Kettle cylinder thickness, inches (mm).

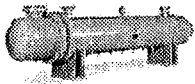
D_k = Mean diameter of kettle cylinder, inches (mm).

L_c = Axial length of kettle cone, inches (mm).

T_c = Kettle cone thickness, inches (mm).

RCB-7.162 EQUIVALENT BOLTING PRESSURE

When fixed tubesheets are extended for bolting to heads with ring type gaskets, the extension and that portion of the tubesheets inside the shell may differ in thickness. The extension shall be designed in accordance with Paragraph RCB-7.134. The effect



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of the moment acting upon the tubesheet extension shall be accounted for in subsequent paragraphs in terms of equivalent tube side and shell side bolting pressures which are defined as:

$$P_{Bt} = \frac{6.2}{F^2} \frac{M_1}{G^3}$$

$$P_{Bs} = \frac{6.2}{F^2} \frac{M_2}{G^3}$$

where

F and G are defined in Paragraph RCB-7.132.

M_1 = Total moment acting upon the extension under operating conditions, defined by the Code as M_o under flange design, inch-pounds (mm-kN).

M_2 = Total moment acting upon the extension under bolting-up conditions, defined by the Code as M_o under flange design, inch-pounds (mm-kN).

P_{Bt} = Equivalent bolting pressure when tube side pressure is acting, psi (kPa).

P_{Bs} = Equivalent bolting pressure when tube side pressure is not acting, psi (kPa).

RCB-7.163 EFFECTIVE SHELL SIDE DESIGN PRESSURE

The effective shell side design pressure is to be taken as the greatest absolute value of the following:

$$P = \frac{P_s' - P_d}{2}$$

or $P = P_s'$

or $P = P_{Bs}$

or $P = \frac{P_s' - P_d + P_{Bs}}{2}$

or $P = \frac{P_{Bs} + P_d}{2}$

or $P = P_s' + P_{Bs}$

where

$$P_s' = P_s \left[\frac{0.4 - J[1.5 + K(1.5 + f_s)] - \left[\left(\frac{1.4}{2} \right) \left(\frac{D_f^2}{G^2} - 1 \right) \right]}{1 + J - K - F_g} \right]$$

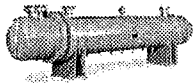
P_s = Shell side design pressure, psi (kPa) (For vacuum design, P_s is negative).

$$f_s = 1 - N \left(\frac{d_o}{G} \right)^2$$

G = Inside diameter of the shell, inches (mm).

D_f = Maximum expansion joint inside diameter, inches (mm) ($D_f = G$ when no expansion joint is present).

Other symbols are as defined under Paragraphs RCB-7.161 and RCB-7.162



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Notes:

- (1) Algebraic sign of P_s must be used above, and must be retained for use in Paragraphs RCB-7.164, RCB-7.165, RCB-7.166, RCB-7.22 and RCB-7.23.
- (2) When $J = 0$, formulae containing P_{st} will not control.
- (3) Delete the term P_{st} in the above formulae for use in Paragraph RCB-7.133.
- (4) For kettle type, G = port inside diameter.

RCB-7.164 EFFECTIVE TUBE SIDE DESIGN PRESSURE

The effective tube side design pressure is to be taken as the greatest absolute value of the following:

$$P = \frac{P_t' + P_{st} + P_d}{2}$$

$$\text{or } P = P_t' + P_{st}$$

When P_s is positive

$$P = \frac{P_t' - P_s' + P_{st} + P_d}{2}$$

$$\text{or } P = P_t' - P_s' + P_{st}$$

When P_s is negative

where

$$P_t' = P_t \left[\frac{1 + 0.4JK(1.5 + f_t)}{1 + JKF_g} \right]$$

P_t = Tube side design pressure, psi (kPa) (For vacuum design, P_t is negative).

$$f_t = 1 - N \left[\frac{d_o - 2t_t}{G} \right]^2$$

G = Inside diameter of the shell, inches (mm).

Other symbols are as defined under Paragraphs RCB-7.161, RCB-7.162, and RCB-7.163.

Notes:

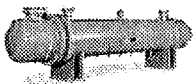
- (1) Algebraic sign of P_s must be used above, and must be retained for use in Paragraphs RCB-7.165, RCB-7.166, RCB-7.22 and RCB-7.23.
- (2) When $J = 0$:
 - a) Formulae containing P_{st} will not control.
 - b) When P_s and P_t are both positive the following formula is controlling:

$$P = P_t + \frac{P_s}{2} \left[\left(\frac{D_t}{G} \right)^2 - 1 \right] + P_{st}$$

- (3) Delete the term P_{st} in the above formulae for use in Paragraph RCB-7.133.
- (4) For kettle type, G = port inside diameter.

RCB-7.165 EFFECTIVE DIFFERENTIAL DESIGN PRESSURE

Under certain circumstances the Code and other regulatory bodies permit design on the basis of simultaneous action of both shell and tube side pressures. The effective differential design pressure for fixed tubesheets under such circumstances is to be taken as the greatest absolute value of the following:



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$$P = P_1' - P_2' + P_{B1}$$

$$\text{or } P = \frac{P_1' - P_2' + P_{B1} + P_d}{2}$$

$$\text{or } P = P_{B2}$$

$$\text{or } P = \frac{P_{B2} + P_d}{2}$$

$$\text{or } P = P_1' - P_2'$$

$$\text{or } P = \frac{P_1' - P_2' + P_d}{2}$$

$$\text{or } P = P_{B1}$$

where

P_d , P_{B2} , P_{B1} , P_2' and P_1' are as defined in Paragraphs RCB-7.161, RCB-7.162, RCB-7.163 and RCB-7.164.

Notes:

- (1) It is not permissible to use $(P_2 - P_1)$ in place of P_2 to calculate P_2' in Paragraph RCB-7.163, and it is not permissible to use $(P_1 - P_2)$ in place of P_1 to calculate P_1' in Paragraph RCB-7.164.
- (2) When $J = 0$, the formulae containing P_d will not control.
- (3) Delete the terms P_{B1} and P_{B2} in the above formulae for use in Paragraph RCB-7.133.

RCB-7.166 FIXED TUBESHEETS OF DIFFERING THICKNESSES

The rules presented in Paragraphs RCB-7.161 through RCB-7.165 and RCB-7.2 are intended for fixed tubesheet exchangers where both tubesheets are the same thickness. Conditions can exist where it is appropriate to use tubesheets of differing thicknesses. These conditions may result from significantly differing elastic moduli and/or allowable stresses. The following procedure may be used for such cases:

- (1) Separate the design parameters as defined in previous paragraphs for each tubesheet system by assigning subscripts *A* and *B* to each of the following terms:

T as T_A and T_B

L as L_A and L_B where $L_A + L_B = 2L$

E as E_A and E_B

F_q as F_{qA} and F_{qB}

Note: The values of M_1 , M_2 , F , G , ΔL , L_1 , D_o , t_s , d_s , t , E_s , E_1 , N and S , must remain constant throughout this analysis. If a fixed tubesheet exchanger has different bolting moments at each tubesheet, the designer should use the values of M_1 and M_2 that produce the conservative design.

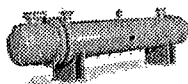
- (2) Calculate T_A per Paragraphs RCB-7.161 through RCB-7.165 assuming that both tubesheets have the properties of subscript *A* and $L_A = L$.



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LG	Long	REINF	Reinforcing
LG	Level Gage	PEPAD	Reinforcing Pad
Lin. ft.	Lineal Foot (Feet)	REQ'D	Required
LLA	Low Level Alarm	RF	Raised Face
LLC	Liquid Level Control	RJ	Ring Joint
LLSD	Low Level Shut Down	RTJ	Ring Type Joint
LR	Long Radius	RV	Relief Valve
LS	Low Stage	R	Radius
LWN	Long Welding Neck	Sht.	Service Sheet
N	North	SF	Straight Flange
N&C	New & Cold	SDV	Shutdown Valve
NLL	Normal Liquid Level	STD	Standard
NO	Number	STL	Steel
NOM	Nominal	SCH	Schedule
NPS	National Pipe Size	SCR	Screw
NPT	American National Taper Pipe Thread	SCR'D	Screwed
NTS	Not to Scale	STR	Straddle
MB	Machine Bolt	SUPT	Support
MK	Mark	SYM	Symmetrical
MAT'L	Material	SPEC	Specification
MAWP	Maximum Allowable Working Pressure	SP GR	Specific Gravity
MK'D	Marked	SQ	Square
mm	Millimeter	SR	Short Radius
MMSCF	Million Standard Cubic Feet	SS	Stainless Steel
MSCF	Thousand Standard Cubic Feet	SHT	Sheet
MW	Manway	SM	Seam
m	Meter	SMLS	Seamless
OA	Overall	SO	Slip On
OD	Outside Diameter	SPA	Spacing
OR	Outside Radius	TLE	Thread Large End
OSHA	Occupational Safety & Health Administration	TOC	Top of Concrete
oz	Ounce	TOS	Top of Steel
ozs	Ounces	TS	Tube Sheet
TSE	Thread Small End	VA	Valve
T-T	Tangent to Tangent	VOL	Volume
TW	Tack Weld	W	With
TW	Thermowell	WG	Water Gallon
T & B	Top & Bottom	WN	Welding Neck
TC	Temperature Control	W OUT	Without
TBE	Threaded Both Ends	WP	Working Pressure
TYP	Typical	WT	Weight
USAS	United States of America Standards Institute	XH	Extra Heavy
		XXH	Double Extra Heavy
		XX STG	Double Extra Strong



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(3) Calculate T_s per paragraph RCB-7.161 through RCB-7.165 assuming that both tube sheets have the properties of subscript B and $L_B = L$

(4) Calculate L_A and L_B as follows:

$$L = L_T - T_A - T_B$$

$$L_B = \frac{2L}{1 + \left(\frac{E_B}{E_A}\right)\left(\frac{T_B}{T_A}\right)^2}$$

$$L_A = 2L - L_B$$

(5) Recalculate T_A per Paragraphs RCB-7.161 through RCB-7.165 using the properties of subscript A and L_A from step 4.

(6) Recalculate T_B per Paragraphs RCB-7.161 through RCB-7.165 using the properties of subscript B and L_B from step 4.

(7) Repeat steps 4 through 6 until values assumed in step 4 are within 1.5% of the values calculated in step 5 for T_A and step 6 for T_B .

(8) Round T_A and T_B up to an appropriate increment and recalculate L_A and L_B per step 4.

(9) Calculate the shell and tube stresses and the tube-to-tubesheet joint loads per Paragraph RCB-7.2 for each tubesheet system using the appropriate subscripted properties.

Note: The shell and tube stresses and tube-to-tubesheet joint loads for each tubesheet system should theoretically be identical. Small differences may exist, however, because of rounding the calculated tubesheet thicknesses in step 8. The tube stress and the tube-to-tubesheet joint loads from the two systems should be averaged before comparing these values to the allowable values as calculated in Paragraph RCB-7.2.

* RCB-7.2 SHELL AND TUBE LONGITUDINAL STRESSES - FIXED TUBESHEET EXCHANGERS

Shell and tube longitudinal stresses, which depend upon the equivalent and effective pressures determined by Paragraphs RCB-7.161 through RCB-7.164, shall be calculated for fixed tubesheet exchangers with or without shell expansion joints by using the following paragraphs. The designer shall consider the most adverse operating conditions specified by the purchaser. (See Paragraph E-3.2.)

Note: The formulae and design criteria presented in Paragraphs RCB-7.23 through RCB-7.25 consider only the tubes at the periphery of the bundle, which are normally the most highly stressed tubes. Additional consideration of the tube stress distribution throughout the bundle may be of interest to the designer under certain conditions of loading and/or geometry. See the "Recommended Good Practice" section of these Standards for additional information.

RCB-7.21 HYDROSTATIC TEST

Hydrostatic test conditions can impose excessive shell and/or tube stresses. These stresses can be calculated by substituting the pressures and temperatures at hydrostatic test for the appropriate design pressures and metal temperatures in the paragraphs that follow and in Paragraphs RCB-7.161 through RCB-7.164 where applicable.

RCB-7.22 SHELL LONGITUDINAL STRESS

The effective longitudinal shell stress is given by:

$$S_s = \frac{C_s (D_o - t_s) P_s^*}{4t_s}$$

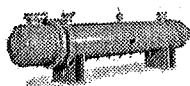
where

$$C_s = 1.0$$

except as noted below

$$P_s^* = P_1$$

Note (2)



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$$\begin{aligned} \text{or } P_s^* &= P_s^* && \text{Note (2)} \\ \text{or } P_s^* &= -P_d && \text{Note (1)} \\ \text{or } P_s^* &= P_1 + P_s^* && \\ \text{or } P_s^* &= P_1 - P_d && \text{Notes (1) and (2)} \\ \text{or } P_s^* &= P_s^* - P_d && \text{Notes (1) and (2)} \\ \text{or } P_s^* &= P_1 + P_s^* - P_d && \text{Note (1)} \end{aligned}$$

where

$$P_1 = P_1 - P_s^*$$

Other symbols are as defined in Paragraphs RCB-7.161, RCB-7.163 and RCB-7.164, using actual shell and tubesheet thicknesses and retaining algebraic signs.

Notes:

- (1) If the algebraic sign of P_s^* is positive, $C_s = 0.5$.
- (2) This formula is not applicable for differential pressure design per Paragraph RCB-7.165.

A condition of overstress shall be presumed to exist when the largest absolute value of S_s exceeds the Code allowable stress in tension for the shell material at design temperature, or 90% of yield stress at hydrostatic test, or when the greatest negative value of S_s exceeds the Code allowable stress in compression at design temperature.

RCB-7.23 TUBE LONGITUDINAL STRESS - PERIPHERY OF BUNDLE

The maximum effective longitudinal tube stress, psl (kPa), at the periphery of the bundle is given by:

$$S_t = \frac{C_t F_a P_t^* G^2}{4 N l_t (d_o - l_t)}$$

where

$$C_t = 1.0$$

except as noted below

$$P_t^* = P_2$$

Note (2)

$$\text{or } P_t^* = -P_3$$

Note (2)

$$\text{or } P_t^* = P_d$$

Notes (1) and (2)

$$\text{or } P_t^* = P_2 - P_3$$

$$\text{or } P_t^* = P_2 + P_d$$

Notes (1) and (2)

$$\text{or } P_t^* = -P_3 + P_d$$

Notes (1) and (2)

$$\text{or } P_t^* = P_2 - P_3 + P_d$$

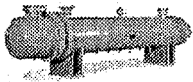
Note (1)

where

$$P_2 = P_1 - \left(\frac{I_t P_1}{F_a} \right)$$

$$P_3 = P_s^* - \left(\frac{I_s P_s^*}{F_a} \right)$$

* Other symbols are as defined in Paragraphs RCB-7.161, RCB-7.163 and RCB-7.164, using actual shell and tubesheet thicknesses and retaining algebraic signs.



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4.2 REMOVABLE BUNDLE EXCHANGERS

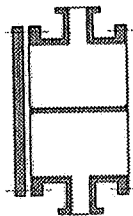
Removable bundle exchangers give the customer the ability to replace the tube bundle without replacing the shell or bonnets. They are generally less cost effective than non removable designs.

4.2.1 EXTERNALLY SEALED FLOATING TUBESHEET: AEW

This design allows for the removal, inspection and cleaning of the shell circuit and shell interior. The special floating tubesheet prevents the intermixing of fluids. In most cases, a straight tube design is more economical than a U-tube design.

Front End
Stationary Head Type

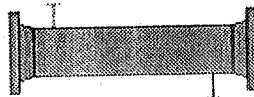
A



Channel and
Removable Cover

Shell Type

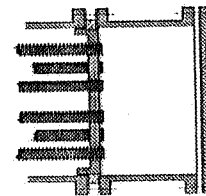
E



One Pass Shell

Rear End
Head Type

W



Externally Sealed
Floating Tubesheet

Advantages:

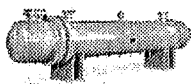
- Floating tubesheet allows for differential thermal expansion between the shell and the tube bundle;
- Shell circuit can be inspected, and steam cleaned or mechanically cleaned;
- The tube bundle can be repaired or replaced without disturbing shell pipe;
- Less costly than other floating head designs: BEP or BES;
- For removable bundles, it gives maximum surface area per given shell diameter;
- Tubes can be cleaned in AEW models without removing piping.

Limitations:

- Fluids in shell and tube circuits must be nonvolatile, nontoxic;
- Tube side passes are limited to single- or two-pass design;
- Two tubesheets are applied: front and back;
- Tubes cannot expand independently; thus, large temperature differentials should be avoided;
- Packing materials limit design pressure and temperature.

Applications are:

- Intercoolers and aftercoolers, air on tubeside;
- Coolers with water inside the tubes;
- Jacket water coolers;
- Hot side fluid through the shell with entry nearest the front



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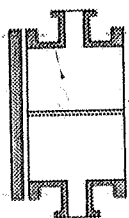
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4.2.2 OUTSIDE PACKED HEAD: AEP

This design allows for the easy removal, inspection and cleaning of the shell circuit and shell interior without taking off the floating head cover. A special floating tubesheet prevents the intermixing of fluids. The design incorporates the removal of straight tubes for cleaning. That is usually more costly than designs accommodating U-tubes.

**Front End
Stationary Head Type**

A



**Channel and
Removable Cover**

Shell Type

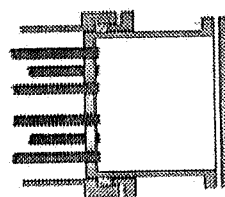
E



One Pass Shell

**Rear End
Head Type**

P



**Outside Packed
Floating Head**

Advantages:

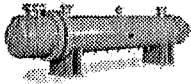
- Floating tube- sheet allows for differential thermal expansion between the shell and the tube bundle;
- Shell circuit can be inspected and steam cleaned. If the tube bundle has a square tube pitch, tubes can be mechanically cleaned by passing a brush between rows of tubes;
- The tube bundle can be repaired or replaced without disturbing shell piping;
- For AEP designs, tubes can be serviced without disturbing tubeside piping;
- Less costly than TEMA-type BES or BET designs;
- Only shell fluids are exposed to packing. Toxic or volatile fluids can be cooled in the tubeside circuit;
- Provides large bundle entrance area, reducing the need for entrance domes for proper fluid distribution.

Limitations:

- Shell fluids limited to nonvolatile, nontoxic materials;
- Packing limits shell-side design temperature and pressure;
- All tubes are attached to two tubesheets. Tubes cannot expand independently; thus, large thermal expansions should be avoided;
- Less surface per given shell and tube diameter than AEW or BEW.

Applications:

- Flammable or toxic liquids in the tube circuit;
- Good for high-fouling liquids in the tube circuit.



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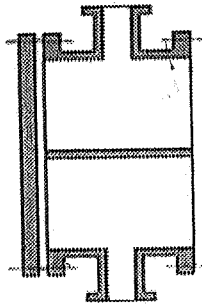
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4.2.3 INTERNAL SPLIT-RING FLOATING HEAD: AES

Exchangers with AES, BES, etc., designs are ideal for applications requiring frequent tube bundle removal, inspection and cleaning. The exchangers have a straight-tube design, suitable for large differential temperatures between the shell and tube fluids. They are more forgiving to thermal shock than AEW or BEW designs, and are suitable for cooling volatile or toxic fluids.

Front End
Stationary Head Type

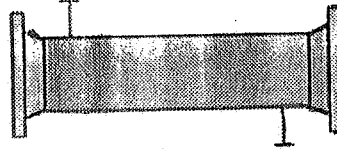
A



Channel and
Removable Cover

Shell Type

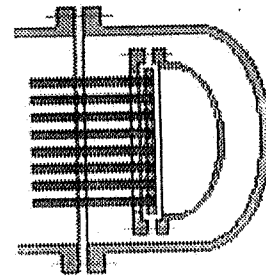
E



One Pass Shell

Rear End
Head Type

S



Floating Head
with backing device

Advantages:

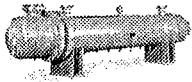
- Floating head design allows for differential thermal expansion between the shell and the tube bundle;
- Shell circuit can be inspected and steam cleaned. If it has a square tube layout, tubes can be mechanically cleaned;
- Higher surface per given shell and tube diameter than "pull-through" designs such as AET, BET, etc.;
- Provides multipass tube circuit arrangement.

Limitations:

- Shell-cover, split-ring and floating head cover must be removed to inspect the tube bundle. That results in higher maintenance cost than with the pull-through bundle;
- More costly per square foot of surface than fixed tube sheet or U-tube designs.

Applications:

- Chemical processing of toxic fluids;
- Special intercoolers and aftercoolers;
- General industrial uses.



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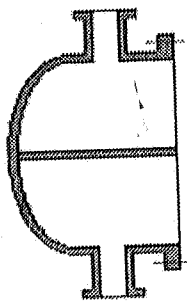
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4.2.5 U-TUBE: BEU

Especially good for severe performance requirements with maximum thermal expansion capability. Because each tube can expand and contract independently, the design works well with large temperature differentials. While the AEM and AEW are the least expensive designs, U-tube bundles are also economical.

Front End
Stationary Head Type

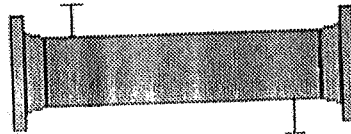
B



Bonnet
(Integral Cover)

Shell Type

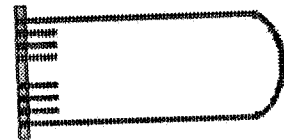
E



One Pass Shell

Rear End
Head Type

U



U-Tube Bundle

Advantages:

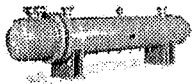
- U-tube design allows for maximum differential thermal expansion between the shell and the tube bundle--for individual tubes too;
- Shell circuit can be inspected, steamed or mechanically cleaned;
- Less costly than floating head or packed floating head designs;
- Provides multipass tube circuit arrangement;
- withstands thermal expansion applications;
- Bundle can be removed from one end for cleaning or replacement.

Limitations:

- Because of the 180° bend, U-tubes can be cleaned only by chemical means;
- U-tubes are difficult to replace;
- Neither single tube passes nor true countercurrent flow is possible;
- Tube wall thickness at the U-bend is thinner than for straight tubes;
- Tube circuits are difficult to drain.

Applications:

- Oil, chemical and water heating applications;
- Excellent in steam to liquid applications.



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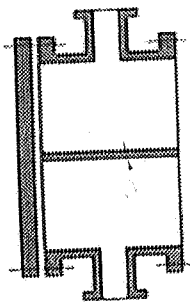
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4.2.4 PULL-THROUGH FLOATING HEAD: AET

Ideal for applications requiring frequent tube bundle removal for inspection and cleaning. The floating head is bolted to the floating tubesheet. That alleviates the need to remove the floating head to pull the tube bundle.

Front End
Stationary Head Type

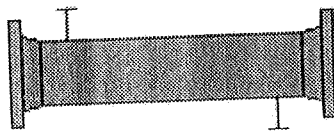
A



Channel and
Removable Cover

Shell Type

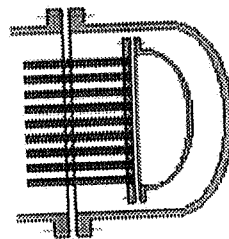
E



One Pass Shell

Rear End
Head Type

T



Pull Through
Floating Head

Advantages:

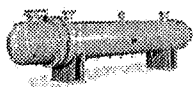
- Floating head design allows for differential thermal expansion between the shell and the tube bundle;
- Shell circuit can be inspected, steam or mechanically cleaned;
- Provides large bundle entrance area for proper fluid distribution;
- Provides multipass tube circuit arrangement;
- Suitable for toxic or volatile fluid cooling.

Limitations:

- For a given set of conditions, this TEMA style is the most expensive;
- Less surface per given shell and tube diameter than other removable designs.

Applications:

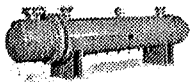
- Chemical processing applications for toxic fluids;
- Hydrocarbon fluid condensers;
- General industrial applications requiring frequent cleaning.



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PART 3



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Notes:

(1) If the algebraic sign of P_1^* is positive, $C_1 = 0.5$.

(2) This formula is not applicable for differential pressure design per Paragraph RCB-7.165.

A condition of overstress shall be presumed to exist when the largest positive value of S_t exceeds the Code allowable stress in tension for the tube material at design temperature, or 90% of yield stress at hydrostatic test, or when the greatest negative value of S_t exceeds the allowable compressive stress as determined in accordance with Paragraph RCB-7.24.

RCB-7.24 ALLOWABLE TUBE COMPRESSIVE STRESS - PERIPHERY OF BUNDLE

The allowable tube compressive stress, psi (kPa), for the tubes at the periphery of the bundle is given by:

$$S_c = \frac{\pi^2 E_t}{F_s \left(\frac{kl}{r} \right)^2} \quad \text{when } C_c \leq \frac{kl}{r}$$

$$\text{or } S_c = \frac{S_y}{F_s} \left[1 - \frac{\left(\frac{kl}{r} \right)}{2 C_c} \right] \quad \text{when } C_c > \frac{kl}{r}$$

where

$$C_c = \sqrt{\frac{2\pi^2 E_t}{S_y}}$$

S_y = Yield stress, psi (kPa), of the tube material at the design metal temperature. (See Paragraph RCB-1.42).

r = Radius of gyration of the tube, inches (mm), given by:

$$r = 0.25 \sqrt{d_o^2 + (d_o - 2t_i)^2} \quad (\text{See Table D-7}).$$

kl = Equivalent unsupported buckling length of the tube, inches (mm). The largest value considering unsupported tube spans shall be used.

l = Unsupported tube span, inches (mm).

$$k = \begin{cases} 0.6 & \text{for unsupported spans between two tubesheets} \\ 0.8 & \text{for unsupported spans between a tubesheet and a tube support} \\ 1.0 & \text{for unsupported spans between two tube supports} \end{cases}$$

F_s = Factor of safety given by:

$$F_s = 3.25 - 0.5 F_q$$

Note: F_s shall not be less than 1.25 and need not be taken greater than 2.0.

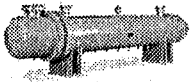
Other symbols are as defined in Paragraph RCB-7.161.

Note: The allowable tube compressive stress shall be limited to the smaller of the Code allowable stress in tension for the tube material at the design metal temperature (see Paragraph RCB-1.42) or the calculated value of S_c .

RCB-7.25 TUBE-TO-TUBESHEET JOINT LOADS - PERIPHERY OF BUNDLE

The maximum effective tube-to-tubesheet joint load, lbs. (kN), at the periphery of the bundle is given by:

$$W_j = \frac{\pi F_o P_1^* G^2}{4 N}$$



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where

$$P_t^* = P_2 \quad \text{Note (1)}$$

$$\text{or } P_t^* = -P_3 \quad \text{Note (1)}$$

$$\text{or } P_t^* = P_2 - P_3$$

P_2 and P_3 are as defined in Paragraph RCB-7.23. Other symbols are as defined in Paragraphs RCB-7.161, RCB-7.163 and RCB-7.164, using the actual shell and tubesheet thicknesses.

Note: (1) This formula is not applicable for differential pressure design per Paragraph RCB-7.165.

The allowable tube-to-tubesheet joint loads as calculated by the Code or other means may be used as a guide in evaluating l_v .

The tube-to-tubesheet joint loads calculated above consider only the effects of pressure loadings. The tube-to-tubesheet joint loads caused by restrained differential thermal expansion between shell and tubes are considered to be within acceptable limits if the requirements of Paragraph RCB-7.23 are met.

RCB-7.3 SPECIAL CASES

Special consideration must be given to tubesheet designs with abnormal conditions of support or loading. Following are some typical examples:

- (1) Tubesheets with portions not adequately stayed by tubes, or with wide untubed rims.
- (2) Exchangers with large differences in shell and head inside diameters; e.g. fixed tubesheets with kettle type shell.
- (3) The adequacy of the staying action of the tubes during hydrostatic test; e.g., with test rings for types S and T, or types P and W.
- (4) Vertical exchangers where weight and/or pressure drop loadings produce significant effects relative to the design pressures.
- (5) Extreme interpass temperature differentials.

Consideration may also be given to special design configurations and/or methods of analysis which may justify reduction of the tubesheet thickness requirements.

RCB-7.4 TUBE HOLES IN TUBESHEETS

RCB-7.41 TUBE HOLE DIAMETERS AND TOLERANCES

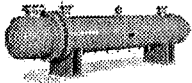
Tube holes in tubesheets shall be finished to the diameters and tolerances shown in Tables RCB-7.41 and RCB-7.41M, column (a). To minimize work hardening, a closer fit between tube OD and tube ID as shown in column (b) may be provided when specified by the purchaser.

RCB-7.42 TUBESHEET LIGAMENTS

Tables RCB-7.42 and RCB-7.42M give permissible tubesheet ligaments, drill drift and recommended maximum tube wall thicknesses.

*RCB-7.43 TUBE HOLE FINISH

The inside edges of tube holes in tubesheets shall be free of burrs to prevent cutting of the tubes. Internal surfaces shall be given a workmanlike finish.



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TABLE RCB-7.41

TUBE HOLE DIAMETER AND TOLERANCES

(All Dimensions in Inches)

(All Dimensions in Inches)

Nominal Tube OD	Nominal Tube Hole Diameter and Under Tolerance				Over Tolerance: 96% of tube holes must meet value in column (c). Remainder may not exceed value in column (d)	
	Standard Fit (a)		Special Close Fit (b)			
	Nominal Diameter	Under Tolerance	Nominal Diameter	Under Tolerance	(c)	(d)
1/4	0.259	0.004	0.257	0.002	0.002	0.007
3/8	0.364	0.004	0.362	0.002	0.002	0.007
1/2	0.510	0.004	0.508	0.002	0.002	0.008
5/8	0.635	0.004	0.633	0.002	0.002	0.010
3/4	0.760	0.004	0.758	0.002	0.002	0.010
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
1-1/4	1.264	0.006	1.261	0.003	0.003	0.010
1-1/2	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 RCB-7.41 M
TUBE HOLE DIAMETERS AND TOLERANCES
(All Dimensions in mm)

(All Dimensions in mm)

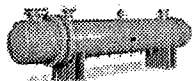
Nominal Tube OD	Nominal Tube Hole Diameter and Under Tolerance				Over Tolerance: 96% of tube holes must meet value in column (c). Remainder may not exceed value in column (d)	
	Standard Fit (a)		Special Close Fit (b)			
	Nominal Diameter	Under Tolerance	Nominal Diameter	Under Tolerance	(c)	(d)
6.4	6.58	0.10	6.53	0.05	0.05	0.18
9.5	9.75	0.10	9.70	0.05	0.05	0.18
12.7	12.95	0.10	12.90	0.05	0.05	0.20
15.9	16.13	0.10	16.08	0.05	0.05	0.25
19.1	19.30	0.10	19.25	0.05	0.05	0.25
22.2	22.48	0.10	22.43	0.05	0.05	0.25
25.4	25.70	0.10	25.65	0.05	0.05	0.25
31.8	32.11	0.15	32.03	0.08	0.08	0.25
38.1	38.56	0.18	38.48	0.08	0.08	0.25
50.8	51.36	0.18	51.26	0.08	0.08	0.25

RB-7.44 TUBE HOLE GROOVING

Tube holes for expanded joints for tubes 5/8" (15.9mm) OD and larger shall be machined with at least two grooves, for additional longitudinal load resistance, each approximately 1/8" (3.2mm) wide by 1/64" (0.4mm) deep. When integrally clad or applied tubesheet facings are used, all grooves should be in the base material unless otherwise specified by the purchaser. Strength welded tubes do not require grooves. Tubesheets with thicknesses less than 1" (25.4mm) may be provided with one groove. When utilizing hydraulic expansion, grooves shall be 1/4" (6.4mm) wide.

C-7.44 TUBE HOLE GROOVING

For design pressures over 300 psi (2068 kPa) and/or temperatures in excess of 350 °F (177 °C), the tube holes for expanded joints for tubes 5/8" (15.9 mm) OD and larger shall be machined with at least two grooves, for additional longitudinal load resistance, each approximately 1/8" (3mm) wide by 1/64" (0.4 mm) deep. When integrally clad or applied tubesheet facings are used, all grooves should be in the base material unless otherwise specified by the purchaser. Strength welded tubes do not require grooves. Tubesheets with thicknesses less than 1" (25.4mm) may be provided with one groove. When utilizing hydraulic expansion, grooves shall be 1/4" (6.4mm) wide.



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TABLE RCB-7.42
TABLE OF TUBESHEET LIGAMENTS AND RECOMMENDED HEAVIEST TUBE GAGES
(All Dimensions in Inches)

Tube Dia. d_o	Tube Pitch P	$\frac{P}{d_o}$	$P - d_o$	Heaviest Recommended Tube Gage BWG	Tube Hole Dia. Std. Fit	Nominal Ligament Width	Minimum Std. Ligaments (96% of ligaments must equal or exceed values tabulated below)								Minimum Permissible Ligament Width
							Tubesheet Thickness								
							t	1-1/2	2	2-1/2	3	4	5	6	
1/4	5/16 3/8	1.25 1.50	1/16 1/8	22 20	0.259	0.054 0.116	0.025 0.083	0.025 0.077	0.025 0.070	0.025 0.064	-- --	-- --	-- --	-- --	0.025 0.060
3/8	1/2 17/32	1.33 1.42	1/8 5/32	18 18	0.394	0.116 0.147	0.087 0.119	0.083 0.114	0.079 0.110	0.075 0.106	0.070 0.102	0.062 0.093	0.085	0.076	0.060 0.075
1/2	5/8 21/32 11/16	1.25 1.31 1.38	1/8 5/32 3/16	18 16 16	0.510	0.115 0.146 0.178	0.089 0.120 0.151	0.085 0.117 0.148	0.082 0.113 0.145	0.079 0.110 0.142	0.076 0.107 0.138	0.069 0.101 0.132	0.063 0.094 0.126	0.088 0.119	0.060 0.075 0.090
5/8	25/32 13/16 7/8	1.25 1.30 1.40	5/32 3/16 1/4	15 14 14	0.635	0.146 0.178 0.240	0.111 0.142 0.205	0.109 0.140 0.202	0.106 0.137 0.200	0.103 0.135 0.197	0.101 0.132 0.195	0.096 0.127 0.180	0.091 0.122 0.184	0.088 0.117 0.179	0.075 0.090 0.120
3/4	15/16 1 1-1/16 1-1/8	1.25 1.33 1.42 1.50	3/16 1/4 5/16 3/8	13 12 12 12	0.760	0.178 0.240 0.303 0.365	0.143 0.206 0.268 0.331	0.141 0.204 0.266 0.329	0.139 0.201 0.264 0.326	0.137 0.199 0.262 0.324	0.135 0.197 0.260 0.322	0.130 0.193 0.255 0.318	0.126 0.189 0.251 0.314	0.122 0.184 0.247 0.309	0.090 0.120 0.150 0.185
7/8	1-3/32 1-1/8 1-3/16 1-1/4	1.25 1.29 1.36 1.43	7/32 1/4 5/16 3/8	12 12 10 10	0.865	0.209 0.240 0.303 0.365	0.175 0.208 0.269 0.331	0.173 0.205 0.267 0.330	0.171 0.203 0.265 0.328	0.170 0.201 0.263 0.325	0.168 0.199 0.262 0.324	0.164 0.195 0.258 0.320	0.160 0.192 0.254 0.317	0.157 0.189 0.251 0.313	0.105 0.120 0.150 0.185
1	1-1/4 1-5/16 1-3/8	1.25 1.31 1.38	1/4 5/16 3/8	10 9 9	1.012	0.238 0.301 0.363	0.205 0.267 0.330	0.203 0.266 0.328	0.202 0.264 0.327	0.200 0.263 0.325	0.199 0.261 0.323	0.195 0.258 0.320	0.192 0.255 0.317	0.189 0.251 0.314	0.120 0.160 0.195
1 1/4	1-9/16	1.25	5/16	9	1.264	0.299	0.266	0.266	0.263	0.262	0.261	0.258	0.256	0.253	0.150
1 1/2	1-7/8	1.26	3/8	8	1.518	0.357	0.325	0.324	0.323	0.322	0.321	0.318	0.316	0.314	0.180
2	2-1/2	1.26	1/2	6	2.022	0.478	--	0.446	0.445	0.444	0.443	0.442	0.440	0.438	0.250

Notes: The above table of minimum standard ligaments is based on a ligament tolerance not exceeding the sum of twice the drill drift tolerance plus 0.020" for tubes less than 5/8" OD and 0.030" for tube holes 5/8" OD and larger.
Drill drift tolerance = 0.0016 (thickness of tubesheet in tube diameters), inches

*RCB-7.5 TUBE-TO-TUBESHEET JOINTS

RCB-7.51 EXPANDED TUBE-TO-TUBESHEET JOINTS

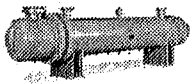
Expanded tube-to-tubesheet joints are standard.

RB-7.511 LENGTH OF EXPANSION

Tubes shall be expanded into the tubesheet for a length no less than 2" (50.8 mm) or the tubesheet thickness minus 1/8" (3.2 mm), whichever is smaller. In no case shall the expanded portion extend beyond the shell side face of the tubesheet. When specified by the purchaser, tubes may be expanded for the full thickness of the tubesheet.

C-7.511 LENGTH OF EXPANSION

Tubes shall be expanded into the tubesheet for a length no less than two tube diameters, 2" (50.8 mm), or the tubesheet thickness minus 1/8" (3.2 mm), whichever is smaller. In no case shall the expanded portion extend beyond the shell side face of the tubesheet. When specified by the purchaser, tubes may be expanded for the full thickness of the tubesheet.



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TABLE RCB-7.412 M

TABLE OF TUBESHEET LIGAMENTS AND RECOMMENDED HEAVIEST TUBE GAGES
(All Dimensions in mm)

Tube Dia. d_o	Tube Pitch p	$\frac{p}{d_o}$	$p - d_o$	Heaviest Recommended Tube Gage BWG	Tube Hole Dia. Std. Fit	Nominal Ligament Width	Minimum Std. Ligaments (96% of ligaments must equal or exceed values tabulated below)								Minimum Permissible Ligament Width
							Tubesheet Thickness								
							25.4	38.1	50.8	63.5	76.2	101.6	127.0	152.4	
6.4	7.94 9.53	1.25 1.50	1.59 3.18	22 20	6.579	1.361 2.951	0.635 2.108	0.635 1.956	0.635 1.778	0.635 1.626	-	-	-	-	0.635 1.524
9.5	12.70 13.49	1.33 1.42	3.17 3.96	18 18	9.754	2.946 3.736	2.210 3.023	2.108 2.896	2.007 2.794	1.905 2.692	1.778 2.591	1.575 2.362	2.159	1.930	1.524 1.905
12.7	15.88 16.67 17.48	1.25 1.31 1.37	3.18 3.97 4.76	16 16 16	12.954	2.926 3.716 4.506	2.261 3.048 3.835	2.159 2.972 3.759	2.083 2.870 3.683	2.007 2.794 3.607	1.930 2.718 3.505	1.753 2.555 3.353	2.368 3.200 3.023	2.235	1.905 2.286
15.9	19.84 20.64 22.23	1.25 1.30 1.40	3.96 4.76 6.35	15 14 14	16.129	3.711 4.511 6.101	2.819 3.607 5.207	2.709 3.556 5.131	2.682 3.480 5.060	2.616 3.429 5.004	2.565 3.353 4.953	2.438 3.226 4.801	2.311 3.099 4.674	2.184 2.972 4.547	1.905 2.286 3.048
19.1	23.81 25.40 26.09 28.59	1.25 1.33 1.42 1.60	4.76 6.35 7.94 9.53	13 12 12 12	19.304	4.506 6.090 7.686 9.276	3.632 5.232 6.807 8.407	3.581 5.182 6.756 8.357	3.531 5.105 6.708 8.280	3.480 5.055 6.655 8.230	3.429 5.004 6.604 8.179	3.302 4.902 6.477 8.077	3.200 4.801 6.375 7.976	3.099 4.674 6.274 7.849	2.286 3.048 3.810 4.609
22.2	27.78 28.58 30.16 31.75	1.25 1.29 1.36 1.43	5.56 6.35 7.93 9.52	12 12 10 10	22.479	5.301 6.101 7.681 9.271	4.445 5.232 6.833 8.407	4.384 5.207 6.782 8.382	4.343 5.156 6.731 8.331	4.318 5.105 6.680 8.280	4.267 5.055 6.655 8.230	4.166 4.953 6.553 8.125	4.064 4.877 6.452 8.052	3.988 4.775 6.375 7.950	2.607 3.048 3.810 4.609
25.4	31.75 33.34 34.93	1.25 1.31 1.38	6.35 7.94 9.53	10 9 9	25.705	6.045 7.635 9.225	5.207 6.782 8.382	5.156 6.756 8.331	5.131 6.706 8.306	5.080 6.680 8.255	5.029 6.629 8.204	4.953 6.553 8.128	4.877 6.477 8.052	4.801 6.375 7.976	3.048 3.810 4.609
31.8	39.69	1.25	7.94	9	32.106	7.584	6.756	6.731	6.680	6.655	6.629	6.553	6.502	6.426	3.810
38.1	47.63	1.25	9.53	8	38.557	9.073	8.255	8.230	8.204	8.179	8.153	8.077	8.026	7.976	4.572
50.8	63.50	1.25	12.70	6	51.359	12.141	-	11.328	11.303	11.278	11.252	11.227	11.176	11.125	6.350

Minimum ligament tolerance not exceeding the sum of twice the drill

Notes: The above table of minimum standard ligaments is based on a ligament tolerance not exceeding the sum of twice the drill drift tolerance plus 0.51mm for tubes less than 15.9mm OD and 0.76mm for tube holes 15.9mm OD and larger.
Drill drift tolerance = 0.041 (thickness of tubesheet in tube diameters), mm.

RCB-7.512 CONTOUR OF THE EXPANDED TUBE

The expanding procedure shall be such as to provide substantially uniform expansion throughout the expanded portion of the tube, without a sharp transition to the unexpanded portion.

RCB-7.513 TUBE PROJECTION

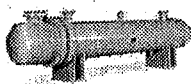
Tubes shall be flush with or extend by no more than one half of a tube diameter beyond the face of each tubesheet, except that tubes shall be flush with the top tubesheet in vertical exchangers to facilitate drainage unless otherwise specified by the purchaser.

RCB-7.52 WELDED TUBE-TO-TUBESHEET JOINTS

When both tubes and tubesheets, or tubesheet lacing, are of suitable materials, the tube joints may be welded.

RCB-7.521 SEAL WELDED JOINTS

When welded tube joints are used for additional leak tightness only, and tube loads are carried by the expanded joint, the tube joints shall be subject to the rules of Paragraphs RCB-7.4 through RCB-7.51.



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RCB-7.522 STRENGTH WELDED JOINTS

When welded tube joints are used to carry the longitudinal tube loads, consideration may be given to modification of the requirements of Paragraphs RCB-7.4 through RCB-7.51. Minimum tubesheet thicknesses shown in Paragraphs R-7.131, C-7.131 and B-7.131 do not apply.

RCB-7.523 FABRICATION AND TESTING PROCEDURES

Welding procedures and testing techniques for either seal welded or strength welded tube joints shall be by agreement between the manufacturer and the purchaser.

RCB-7.53 EXPLOSIVE BONDED TUBE-TO-TUBESHEET JOINTS

Explosive bonding and/or explosive expanding may be used to attach tubes to the tubesheets where appropriate. Consideration should be given to modifying the relevant parameters (e.g., tube-to-tubesheet hole clearances and ligament widths) to obtain an effective joint.

R-7.6 TUBESHEET PASS PARTITION GROOVES

Tubesheets shall be provided with approximately 3/16" (4.8 mm) deep grooves for pass partition gaskets.

CB-7.6 TUBESHEET PASS PARTITION GROOVES

For design pressures over 300 psi (2068 kPa), tubesheets shall be provided with pass partition grooves approximately 3/16" (4.8 mm) deep, or other suitable means for retaining the gaskets in place.

RCB-7.7 TUBESHEET PULLING EYES

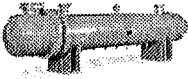
In exchangers with removable tube bundles having a nominal diameter exceeding 12" (305 mm) and/or a tube length exceeding 96" (2438 mm), the stationary tubesheet shall be provided with two tapped holes in its face for pulling eyes. These holes shall be protected in service by plugs of compatible material. Provision for means of pulling may have to be modified or waived for special construction, such as clad tubesheets or manufacturer's standard, by agreement between the manufacturer and the purchaser.

RB-7.8 CLAD AND FACED TUBESHEETS

The nominal cladding thickness at the tube side face of a tubesheet shall not be less than 5/16" (7.8 mm) when tubes are expanded only, and 1/8" (3.2 mm) when tubes are welded to the tubesheet. The nominal cladding thickness on the shell side face shall not be less than 3/8" (9.5 mm). Clad surfaces, other than in the area into which tubes are expanded, shall have at least 1/8" (3.2 mm) nominal thickness of cladding.

C-7.8 CLAD AND FACED TUBESHEETS

The nominal cladding thickness at the tube side face of a tubesheet shall not be less than 3/16" (4.8 mm) when tubes are expanded only, and 1/8" (3.2 mm) when tubes are welded to the tubesheet. The nominal cladding thickness on the shell side face shall not be less than 3/8" (9.5 mm). Clad surfaces, other than in the area into which tubes are expanded, shall have at least 1/8" (3.2 mm) nominal thickness of cladding.



8. SUPPORTS

All heat exchangers are to be provided with supports.

8.1 HORIZONTAL UNITS

The supports should be designed to accommodate the weight of the unit and contents, including the flooded weight during hydrostatic test.

For units with removable tube bundles, supports should be designed to withstand a pulling force equal to 1 - 1/2 times the weight of the tube bundle.

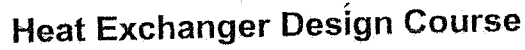
For purposes of support design, forces from external nozzle loadings, wind and seismic events are assumed to be negligible unless the purchaser specifically details the requirements. When these additional loads and forces are required to be considered, the combinations need not be assumed to occur simultaneously.

The references under Paragraph G-7.13 may be used for calculating resulting stresses due to the saddle supports.

Horizontal units are normally provided with at least two saddle type supports, with holes for anchor bolts. The holes in all but one of the supports are to be elongated to accommodate axial movement of the unit under operating conditions. Other types of support may be used if all design criteria are met, and axial movement is accommodated.

8.2 VERTICAL UNITS

Vertical units are to be provided with supports adequate to meet design requirements. The supports may be of the lug, annular ring, leg or skirt type. If the unit is to be located in a supporting structure, the supports should be of sufficient size to allow clearance for the body flanges.



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9. LIFTING DEVICES
Channels, bonnets, and covers which weigh over 60 lbs. (27.2 Kg) are to be provided with lifting lugs, rings or tapped holes for eyebolts. Unless otherwise specified, these lifting devices are designed to lift only the component to which they are directly attached.

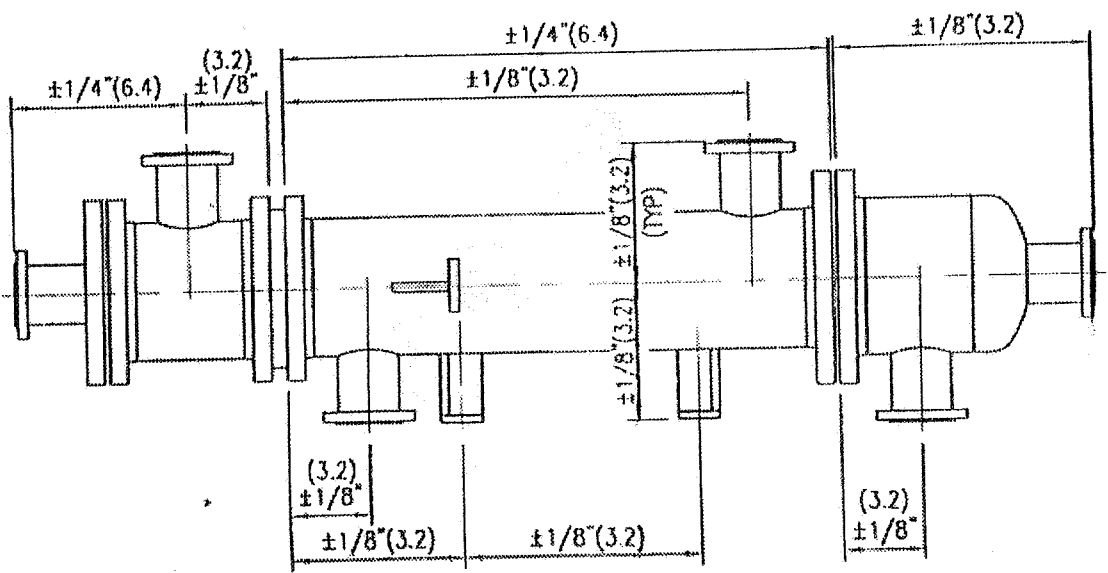
- The purchaser shall inform the manufacturer about the way in which the lifting device will be used. The purchaser shall be notified of any limitations of the lifting device relating to design or method of rigging.
- Liquid penetrant examination of the lifting device attachment weld should be considered on large heavy units.
- The design load shall incorporate an appropriate impact factor.
- Plate-type lifting lugs should be oriented to minimize bending stresses.
- The hole diameter in the lifting device must be large enough to accept a shackle pin having a load rating greater than the design load.
- The effect on the unit component to which the lifting device is attached should be considered. It may be necessary to add a reinforcing plate, annular ring or pad to distribute the load.
- The adequacy of the exchanger to accommodate the lifting loads should be evaluated.

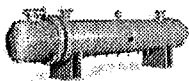
10. WIND & SEISMIC DESIGN

For wind and seismic forces to be considered in the design of a heat exchanger, the purchaser must specify in the inquiry the design requirements. The "Recommended Good Practice" section of these Standards provides the designer with a discussion on this subject and selected references for design application.

11. RECOMMENDED FABRICATION TOLERANCES

Fabrication tolerances normally required to maintain process flow nozzle and support locations are shown in Figure.

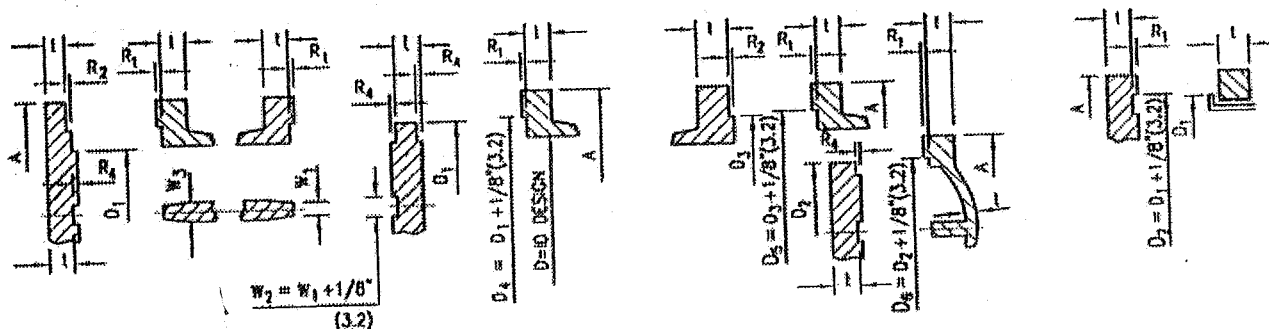




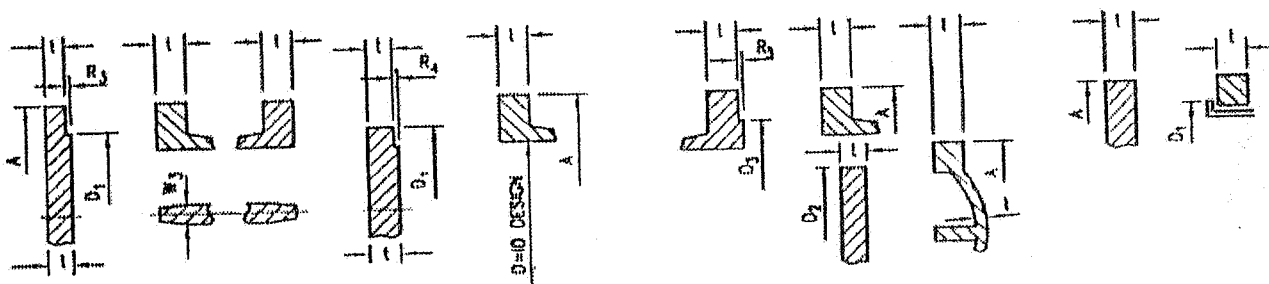
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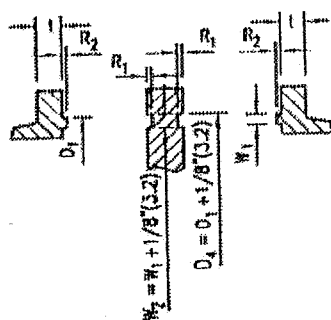
The standard clearances and tolerances applying to tubesheets, partitions, covers and flanges are shown in Figure.



STANDARD CONFINED JOINT CONSTRUCTION



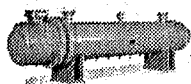
STANDARD UNCONFINED PLAIN FACE JOINT CONSTRUCTION



ALTERNATE
TONGUE AND GROOVE
JOINT

DIMENSIONS	TOLERANCES
A	$+1/4" -1/8"$ (+6.4 -3.2)
$D_1, D_2, D_3, D_4, D_5, D_6$	$\pm 1/32"$ (± 0.8)
l	$\pm 1/16"$ (± 1.6)
$R_1 = 3/16"$ (4.8)	$+0" -1/32"$ (+0 -0.8)
$R_2 = 1/4"$ (6.4) $R_3 = 1/4"$ (6.4)	$+1/32" -0"$ (+0.8 -0)
$R_4 = 3/16"$ (4.8)	$-1/32"$ (-0.8) (SEE NOTE 1)
W_1, W_2, W_3	$\pm 1/32"$ (± 0.8)

- SECTION 2 IS NOT INTENDED TO PROHIBIT UNMACHINED TUBESHEET FACES AND FLAT COVER FACES. THEREFORE NO PLUS TOLERANCE IS SHOWN ON R_4 .
- NEGATIVE TOLERANCES SHALL NOT BE CONSTRUED TO MEAN THAT FINAL DIMENSIONS CAN BE LESS THAN THAT REQUIRED BY DESIGN CALCULATIONS.
- FOR PERIPHERAL GASKETS, "CONFINED" MEANS "CONFINED ON THE OD".
- DETAILS ARE TYPICAL AND DO NOT PRECLUDE THE USE OF OTHER DETAILS WHICH ARE FUNCTIONALLY EQUIVALENT.
- FOR UNITS OVER 60" (1524) TO 100" (2540) DIAMETER, TOLERANCES "D" AND "W" MAY BE INCREASED TO $\pm 1/16"$ (1.6).



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12. NAME PLATES

12.1 MANUFACTURER'S NAME PLATE

A suitable manufacturer's name plate of corrosion resistant material shall be permanently attached to the head end or the shell of each TEMA exchanger. Name plates for exchangers manufactured in accordance with Classes "R" and "B" shall be austenitic (300 series) stainless. When insulation thickness is specified by the purchaser, the name plate shall be attached to a bracket welded to the exchanger.

12.2 NAME PLATE DATA

In addition to all data required by the ASME Code, a name plate shall also include the following (if provided):

User's equipment identification User's order number

12.3 SUPPLEMENTAL INFORMATION

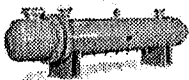
The manufacturer shall supply supplemental information where it is pertinent to the operation or testing of the exchanger. This would include information pertaining to differential design and test pressure conditions, restrictions on operating conditions for fixed tubesheet type exchangers, or other restrictive conditions applicable to the design and/or operation of the unit or its components. Such information can be noted on the name plate or on a supplemental plate attached to the exchanger at the name plate location.

12.4 PURCHASER'S NAME PLATE

Purchaser's name plates, when used, are to be supplied by the purchaser and supplement rather than replace the manufacturer's name plate.

12.5 TEMA REGISTRATION PLATE

The TEMA organization has adopted a voluntary registration system for TEMA members only. When a heat exchanger is registered with TEMA, a unique number is assigned to the heat exchanger. A TEMA registration plate, showing this number, is affixed to the heat exchanger and the ASME Code data report is placed on file at the TEMA office. By referencing this registration number, a copy of the ASME Code data report may be obtained by the purchaser from the TEMA office.



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13. DRAWINGS AND ASME CODE DATA REPORTS

13.1 DRAWINGS FOR APPROVAL AND CHANGE

The manufacturer shall submit for purchaser's approval three (3) prints of an outline drawing showing nozzle sizes and locations, overall dimensions, supports and weight. Other drawings may be furnished as agreed upon by the purchaser and the manufacturer. It is anticipated that a reasonable number of minor drawing changes may be required at that time.

Changes subsequent to receipt of approval may cause additional expense chargeable to the purchaser. Purchaser's approval of drawings does not relieve the manufacturer of responsibility for compliance with this Standard and applicable ASME Code requirements.

The manufacturer shall not make any changes on the approved drawings without express agreement of the purchaser. Shop detail drawings, while primarily for internal use by the fabricator, may be furnished to the purchaser upon request. When detail drawings are requested, they will only be supplied after outline drawings have been approved.

13.2 DRAWINGS FOR RECORD

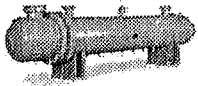
After approval of drawings, the manufacturer shall furnish three (3) prints or, at his option, a transparency of all approved drawings.

13.3 PROPRIETARY RIGHTS TO DRAWINGS

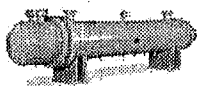
The drawings and the design indicated by them are to be considered the property of the manufacturer and are not to be used or reproduced without his permission, except by the purchaser for his own internal use.

13.4 ASME CODE DATA REPORTS

After completion of fabrication and inspection of ASME Code stamped exchangers, the manufacturer shall furnish three (3) copies of the ASME Manufacturer's Data Report.



PART 4



I. INSPECTION & TESTING

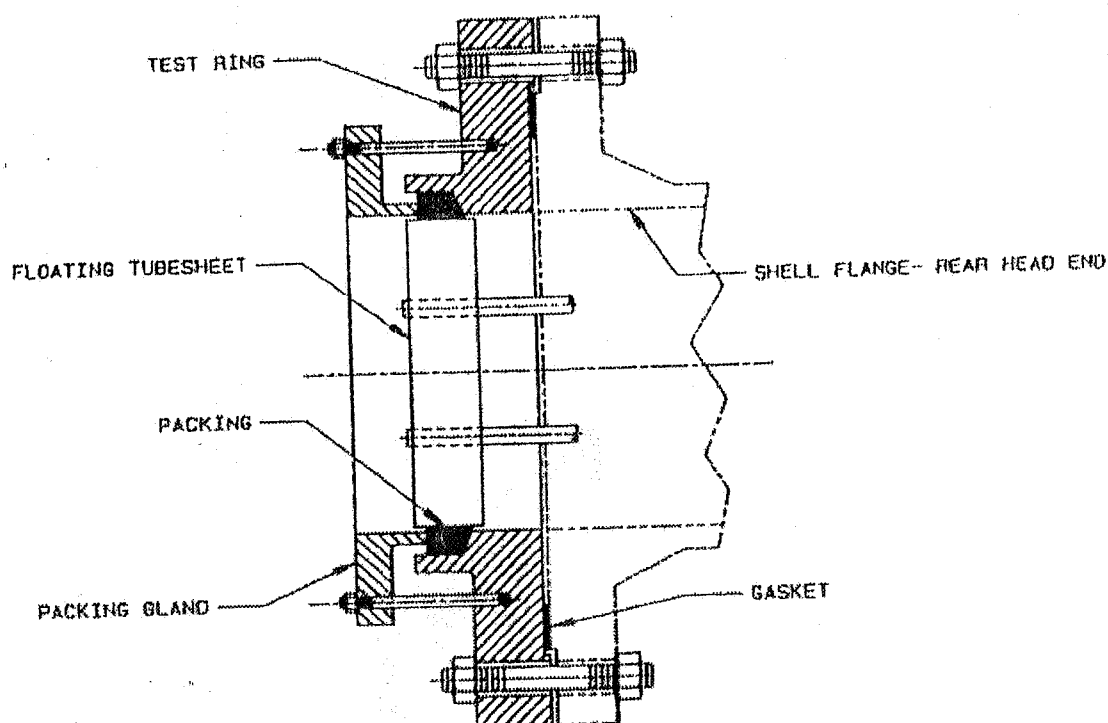
1. GENERAL

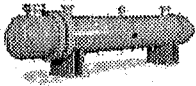
Inspection and testing of units will be provided by the manufacturer unless otherwise specified. The manufacturer shall carry out the inspections required by the ASME Code, and also inspections required by state and local codes when the purchaser specifies the plant location.

The purchaser shall have the right to make inspections during fabrication and to witness any tests when he has so requested. Advance notification shall be given as agreed between the manufacturer and the purchaser. Inspection by the purchaser shall not relieve the manufacturer of his responsibilities.

2. STANDARD TEST

The exchanger shall be hydrostatically tested with water. The test pressure shall be held for at least 30 minutes. The shell side and the tube side are to be tested separately in such a manner that leaks at the tube joints can be detected from at least one side. When the tube side design pressure is the higher pressure, the tube bundle shall be tested outside of the shell only if specified by the purchaser and the construction permits. Welded joints are to be sufficiently cleaned prior to testing the exchanger to permit proper inspection during the test. The minimum hydrostatic test pressure at room temperature shall be in accordance with the Code.





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3. OTHER LIQUID TESTS

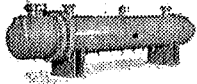
Liquids other than water may be used as a testing medium if agreed upon between the purchaser and the manufacturer.

4. PNEUMATIC TEST

When liquid cannot be tolerated as a test medium the exchanger may be given a pneumatic test in accordance with the Code. It must be recognized that air or gas is hazardous when used as a pressure testing medium. The pneumatic test pressure at room temperature shall be in accordance with the Code.

5. SUPPLEMENTARY AIR TEST

When a supplementary air or gas test is specified by the purchaser, it shall be preceded by the hydrostatic.



II. PREPARATION FOR SHIPMENT

1. CLEANING

Internal and external surfaces are to be free from loose scale and other foreign material that is readily removable by hand or power brushing.

2. DRAINING

Water, oil, or other liquids used for cleaning or hydrostatic testing are to be drained from all units before shipment. This is not to imply that the units must be completely dry.

3. FLANGE PROTECTION

All exposed machined contact surfaces shall be coated with a removable rust preventative and protected against mechanical damage by suitable covers.

4. THREADED CONNECTION PROTECTION

All threaded connections are to be suitably plugged.

5. DAMAGE PROTECTION

The exchanger and any spare parts are to be suitably protected to prevent damage during shipment.

6. EXPANSION JOINT PROTECTION

External thin walled expansion bellows shall be equipped with a protective cover which does not restrain movement.

7. BOLTING

Bolts and nuts shall be coated with waterproof lubricant.

8. MARKING

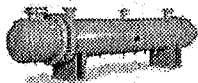
Heat exchangers shall be clearly identified by painting the order and item number in a conspicuous location on the Heat exchanger.

9. LOOSE ITEMS

Small parts which are to be shipped loose shall be bagged or boxed and marked with the order and item number of the Heat exchanger.

10. REMARK

Heat exchanger fabricator shall take all necessary precautions in loading by blocking and bracing the Heat exchanger and furnishing all necessary material to prevent damages.



III. GUARANTEE

1. GENERAL

Manufacturer shall guarantee that the Heat exchanger fulfills all conditions as stated in this Specification and that it is free from fault in design, workmanship and material. Should any defect develop during the first year of operation, the manufacturer agrees to make all necessary alterations, repairs and replacements free of charge.

2. PERFORMANCE

The purchaser shall furnish the manufacturer with all information needed for clear understanding of performance requirements, including any special requirements. The manufacturer shall guarantee thermal performance and mechanical design of a heat exchanger, when operated at the design conditions specified by the purchaser in his order, or shown on the exchanger specification sheet furnished by the manufacturer. This guarantee shall extend for a period of twelve (12) months after shipping date. The manufacturer shall assume no responsibility for excessive fouling of the apparatus by material such as coke, silt, scale, or any foreign substance that may be deposited. The thermal guarantee shall not be applicable to exchangers where the thermal performance rating was made by the purchaser.

3. THERMAL PERFORMANCE TEST

A performance test shall be made if it is established after operation that the performance of the exchanger is not satisfactory, provided the thermal performance rating was made by the manufacturer. Test conditions and procedures shall be selected by agreement between the purchaser and the manufacturer to permit extrapolation of the test results to the specified design conditions.

4. DEFECTIVE PARTS

The manufacturer shall repair or replace any parts proven defective within the guarantee period. Finished materials and accessories purchased from other manufacturers, including tubes, are warranted only to the extent of the original manufacturer's warranty to the heat exchanger fabricator.

5. CONSEQUENTIAL DAMAGES

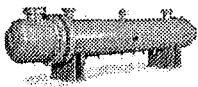
The manufacturer shall not be held liable for any indirect or consequential damage.

6. CORROSION AND VIBRATION

The manufacturer assumes no responsibility for deterioration of any part or parts of the equipment due to corrosion, erosion, flow induced tube vibration, or any other causes, regardless of when such deterioration occurs after leaving the manufacturer's premises.

7. REPLACEMENT AND SPARE PARTS

When replacement or spare tube bundles, shells, or other parts are purchased, the manufacturer is to guarantee satisfactory fit of such parts only if he was the original manufacturer. Parts fabricated to drawings furnished by the purchaser shall be guaranteed to meet the dimensions and tolerances specified.



IV. INSTALLATION

1. HEAT EXCHANGER SETTINGS

1.1 CLEARANCE FOR DISMANTLING

For straight tube exchangers fitted with removable bundles, provide sufficient clearance at the stationary head end to permit removal of the bundle from the shell and provide adequate space beyond the rear head to permit removal of the shell cover and/or floating head cover.

For fixed tubesheet exchangers, provide sufficient clearance at one end to permit withdrawal and replacement of the tubes, and enough space beyond the head at the opposite end to permit removal of the bonnet or channel cover.

For U-tube heat exchangers, provide sufficient clearance at the stationary head end to permit withdrawal of the tube bundle, or at the opposite end to permit removal of the shell.

1.2 FOUNDATIONS

Foundations must be adequate so that exchangers will not settle and impose excessive strains on the exchanger. Foundation bolts should be set to allow for setting inaccuracies. In concrete footings, pipe sleeves at least one size larger than bolt diameter slipped over the bolt and cast in place are best for this purpose, as they allow the bolt center to be adjusted after the foundation has set.

1.3 FOUNDATION BOLTS

Foundation bolts should be loosened at one end of the unit to allow free expansion of shells. Slotted holes in supports are provided for this purpose.

1.4 LEVELING

Exchangers must be set level and square so that pipe connections may be made without forcing.

2. CLEANLINESS PROVISIONS

2.1 CONNECTION PROTECTORS

All exchanger openings should be inspected for foreign material. Protective plugs and covers should not be removed until just prior to installation.

2.2 DIRT REMOVAL

The entire system should be clean before starting operation. Under some conditions, the use of strainers in the piping may be required.

2.3 CLEANING FACILITIES

Convenient means should be provided for cleaning the unit as suggested under "Maintenance of Heat Exchangers"

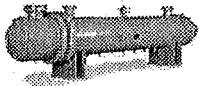
3. FITTINGS AND PIPING

3.1 BY-PASS VALVES

It may be desirable for purchaser to provide valves and by-passes in the piping system to permit inspection and repairs.

3.2 TEST CONNECTIONS

When not integral with the exchanger nozzles, thermometer well and pressure gage connections should be installed close to the exchanger in the inlet and outlet piping.



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3.3 VENTS

Vent valves should be provided by purchaser so units can be purged to prevent vapor or gas binding. Special consideration must be given to discharge of hazardous or toxic fluids.

3.4 DRAINS

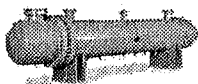
Drains may discharge to atmosphere, if permissible, or into a vessel at lower pressure. They should not be piped to a common closed manifold.

3.5 PULSATION AND VIBRATION

In all installations, care should be taken to eliminate or minimize transmission of fluid pulsations and mechanical vibrations to the heat exchangers.

3.6 SAFETY RELIEF DEVICES

The ASME Code defines the requirements for safety relief devices. When specified by the purchaser, the manufacturer will provide the necessary connections for the safety relief devices. The size and type of the required connections will be specified by the purchaser. The purchaser will provide and install the required relief devices.



V. OPERATION

1. DESIGN AND OPERATING CONDITIONS

Equipment must not be operated at conditions which exceed those specified on the name plate(s).

2. OPERATING PROCEDURES

Before placing any exchanger in operation, reference should be made to the exchanger drawings, specification sheet(s) and name plate(s) for any special instructions. Local safety and health regulations must be considered. Improper start-up or shut-down sequences, particularly of fixed tubesheet units, may cause leaking of tube-to-tubesheet and/or bolted flanged joints.

3. START-UP OPERATION

Most exchangers with removable tube bundles may be placed in service by first establishing circulation of the cold medium, followed by the gradual introduction of the hot medium. During start-up all vent valves should be opened and left open until all passages have been purged of air and are completely filled with fluid. For fixed tubesheet exchangers, fluids must be introduced in a manner to minimize differential expansion between the shell and tubes.

4. SHUT-DOWN OPERATION

For exchangers with removable bundles, the units may be shut down by first gradually stopping the flow of the hot medium and then stopping the flow of the cold medium. If it is necessary to stop the flow of cold medium, the circulation of hot medium through the exchanger should also be stopped. For fixed tubesheet exchangers, the unit must be shut down in a manner to minimize differential expansion between shell and tubes. When shutting down the system, all units should be drained completely when there is the possibility of freezing or corrosion damage. To guard against water hammer, condensate should be drained from steam heaters and similar apparatus during start-up or shut-down. To reduce water retention after drainage, the tube side of water cooled exchangers should be blown out with air.

5. TEMPERATURE SHOCKS

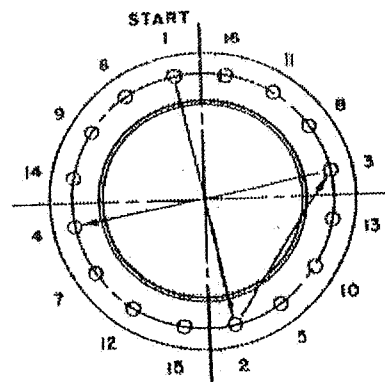
Exchangers normally should not be subjected to abrupt temperature fluctuations. Hot fluid must not be suddenly introduced when the unit is cold, nor cold fluid suddenly introduced when the unit is hot.

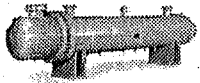
6. BOLTED JOINTS

Heat exchangers are pressure tested before leaving the manufacturer's shop in accordance with ASME Code requirements. However, normal relaxing of the gasketed joints may occur in the interval between testing in the manufacturer's shop and installation at the jobsite. Therefore, all external bolted joints may require retightening after installation and, if necessary, after the exchanger has reached operating temperature.

7. RECOMMENDED BOLT TIGHTENING PROCEDURE

It is important that all bolted joints be tightened uniformly and in a diametrically staggered pattern, as illustrated in Figure, except for special high pressure closures when the instructions of the manufacturer should be followed.





VI. MAINTENANCE

1. GENERAL

Satisfactory operation of heat exchangers can be obtained only from units which are properly designed and have built-in quality. Correct installation and preventive maintenance are user responsibilities.

2. PERFORMANCE FAILURES

The failure of heat exchanger equipment to perform satisfactorily may be caused by one or more factors, such as:

- Excessive fouling.
- Air or gas binding resulting from improper piping installation or lack of suitable vents.
- Operating conditions differing from design conditions.
- Maldistribution of flow in the unit.
- Excessive clearances between the baffles and shell and/or tubes, due to corrosion.
- Improper thermal design.

The user's best assurance of satisfactory performance lies in dependence upon manufacturers competent in the design and fabrication of heat transfer equipment.

3. INSPECTION OF UNIT

At regular intervals and as frequently as experience indicates, an examination should be made of the interior and exterior condition of the unit. Neglect in keeping all tubes clean may result in complete stoppage of flow through some tubes which could cause severe thermal strains, leaking tube joints, or structural damage to other components. Sacrificial anodes, when provided, should be inspected to determine whether they should be cleaned or replaced.

4. INDICATIONS OF FOULING

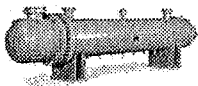
Exchangers subject to fouling or scaling should be cleaned periodically. A light sludge or scale coating on the tube greatly reduces its efficiency. A marked increase in pressure drop and/or reduction in performance usually indicates cleaning is necessary. The unit should first be checked for air or vapor binding to confirm that this is not the cause for the reduction in performance. Since the difficulty of cleaning increases rapidly as the scale thickness or deposit increases, the intervals between cleanings should not be excessive.

5. DISASSEMBLY FOR INSPECTION OR CLEANING

Before disassembly, the user must assure himself that the unit has been depressurized, vented and drained, neutralized and/or purged of hazardous material.

To inspect the inside of the tubes and also make them accessible for cleaning, the following procedures should be used:

- Stationary Head End
 - For type A, C, D & N, remove cover only
 - For type B, remove bonnet
- Rear Head End
 - For type L, N & P, remove cover only
 - For type M, remove bonnet
 - For type S & T, remove shell cover and floating head cover
 - For type W, remove channel cover or bonnet



6. LOCATING TUBE LEAKS

The following procedures may be used to locate perforated or split tubes and leaking joints between tubes and tubesheets. In most cases, the entire front face of each tubesheet will be accessible for inspection. The point where water escapes indicates a defective tube or tube-to-tubesheet joint.

- Units with removable channel cover: Remove channel cover and apply hydraulic pressure in the shell.
- Units with bonnet type head: For fixed tubesheet units where tubesheets are an integral part of the shell, remove bonnet and apply hydraulic pressure in the shell. For fixed tubesheet units where tubesheets are not an integral part of the shell and for units with removable bundles, remove bonnet, re-bolt tubesheet to shell or install test flange or gland, whichever is applicable, and apply hydraulic pressure in the shell. See Figure E-4.13-1 for typical test flange and test gland.
- Units with Type S or T floating head: Remove channel cover or bonnet, shell cover and floating head cover. Install test ring and bolt in place with gasket and packing. Apply hydraulic pressure in the shell. When a test ring is not available it is possible to locate leaks in the floating head end by removing the shell cover and applying hydraulic pressure in the tubes. Leaking tube joints may then be located by sighting through the tube lanes. Care must be exercised when testing partially assembled exchangers to prevent over extension of expansion joints or overloading of tubes and /or tube-to-tubesheet joints.

7. TUBE BUNDLE REMOVAL AND HANDLING

To avoid possible damage during removal of a tube bundle from a shell, a pulling device should be attached to eyebolts screwed into the tubesheet. If the tubesheet does not have lapped holes for eyebolts, steel rods or cables inserted through tubes and attached to bearing plates may be used. The bundle should be supported on the tube baffles, supports or tubesheets to prevent damage to the tubes.

Gasket and packing contact surfaces should be protected.

8. CLEANING TUBE BUNDLES

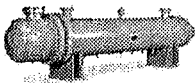
8.1 CLEANING METHODS

The heat transfer surfaces of heat exchangers should be kept reasonably clean to assure satisfactory performance. Convenient means for cleaning should be made available.

Heat exchangers may be cleaned by either chemical or mechanical methods. The method selected must be the choice of the operator of the plant and will depend on the type of deposit and the facilities available in the plant. Following are several cleaning procedures that may be considered:

- Circulating hot wash oil or light distillate through tubes or shell at high velocity may effectively remove sludge or similar soft deposits.
- Some salt deposits may be washed out by circulating hot fresh water.
- Commercial cleaning compounds are available for removing sludge or scale provided hot wash oil or water is not available or does not give satisfactory results.
- High pressure water jet cleaning.

Employ services of a qualified organization that provides cleaning services. These organizations will check the nature of the deposits to be removed, furnish proper solvents and/or acid solutions containing inhibitors, and provide equipment and personnel for a complete cleaning job.



8.2 CLEANING PRECAUTIONS

- Tubes should not be cleaned by blowing steam through individual tubes since this heats the tube and may result in severe expansion strain, deformation of the tube, or loosening of the tube-to-tubesheet joint.
- When mechanically cleaning a tube bundle, care should be exercised to avoid damaging the tubes.
- Cleaning compounds must be compatible with the metallurgy of the exchanger.

9. TUBE EXPANDING

A suitable tube expander should be used to tighten a leaking tube joint. Care should be taken to ensure that tubes are not over expanded.

10. GASKET REPLACEMENT

Gaskets and gasket surfaces should be thoroughly cleaned and should be free of scratches and other defects. Gaskets should be properly positioned before attempting to retighten bolts. It is recommended that when a heat exchanger is dismantled for any cause, it be reassembled with new gaskets. This will tend to prevent future leaks and/or damage to the gasket seating surfaces of the heat exchanger. Composition gaskets become dried out and brittle so that they do not always provide an effective seal when reused. Metal or metal jacketed gaskets, when compressed initially, flow to match their contact surfaces. In so doing they are work hardened and, if reused, may provide an imperfect seal or result in deformation and damage to the gasket contact surfaces of the exchanger. Bolted joints and flanges are designed for use with the particular type of gasket specified. Substitution of a gasket of different construction or improper dimensions may result in leakage and damage to gasket surfaces. Therefore, any gasket substitutions should be of compatible design. Any leakage at a gasketed joint should be rectified and not permitted to persist as it may result in damage to the gasket surfaces.

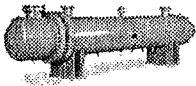
Metal jacketed type gaskets are widely used. When these are used with a tongue and groove joint without a nubbin, the gasket should be installed so that the tongue bears on the seamless side of the gasket jacket. When a nubbin is used, the nubbin should bear on the seamless side.

11. SPARE AND REPLACEMENT PARTS

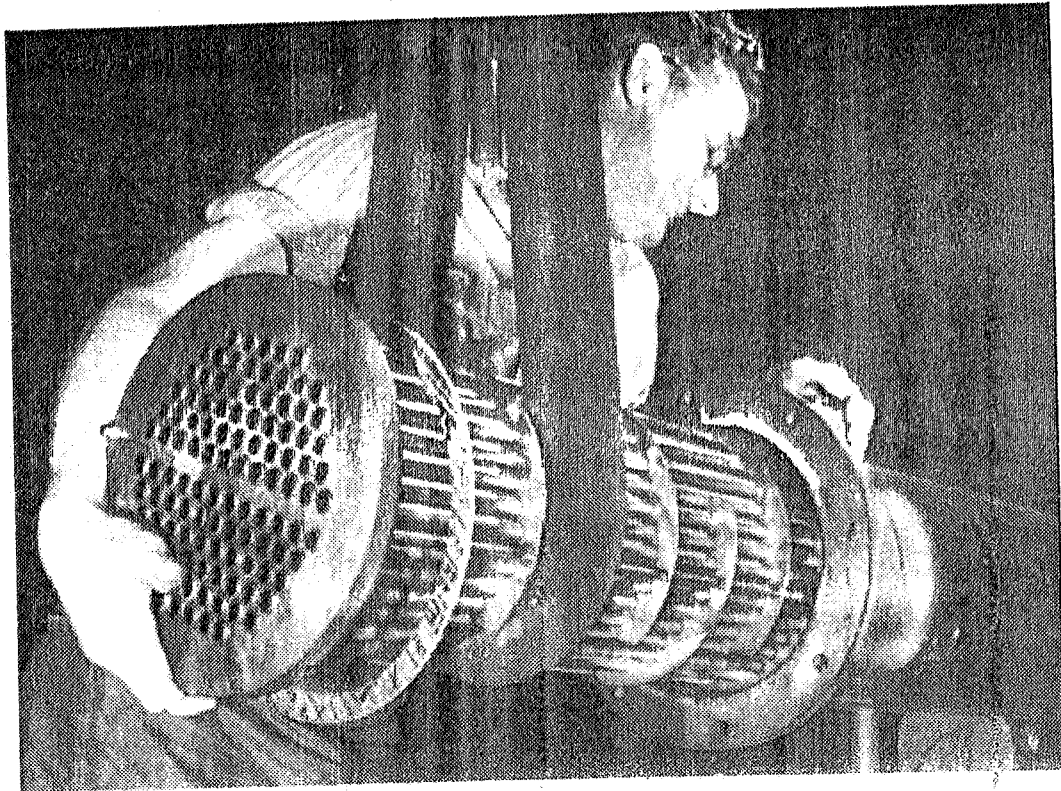
The procurement of spare or replacement parts from the manufacturer will be facilitated if the correct name for the part, as shown in Section 1, Table N-2, of these Standards is given, together with the serial number, type, size, and other information from the name plate. Replacement parts should be purchased from the original manufacturer.

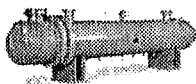
12. PLUGGING OF TUBES

In U-tube heat exchangers, and other exchangers of special design, it may not be feasible to remove and replace defective tubes. Defective tubes may be plugged using commercially available tapered plugs with ferrules or tapered only plugs which may or may not be seal welded. Excessive tube plugging may result in reduced thermal performance, higher pressure drop, and/or mechanical damage. It is the user's responsibility to remove plugs and neutralize the bundle prior to sending it to a shop for repairs.



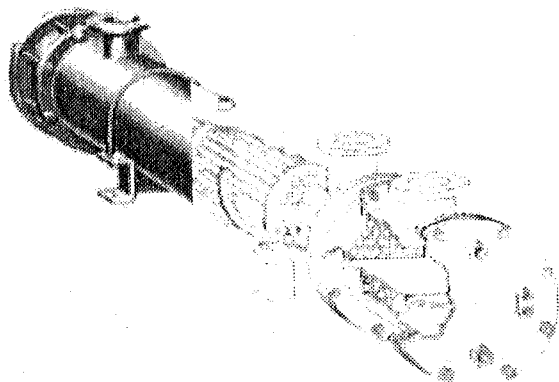
SAMPLE OF MANUFACTURER INSTRUCTION MANUAL FOR INSTALLATION, OPERATION & MAINTENANCE





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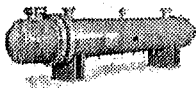


INSTALLATION

1. Provide sufficient clearance at the stationary tube sheet end of the unit to permit removal of tube bundles from shells. On the packed floating tube sheet end, a space of 3 or 4 feet should be provided to permit the removal of the rear head, packing and retainer rings.
2. Provide valves and by-passes in the piping systems so that both the shells and tube bundles may be by-passed to permit cutting out the unit for inspection or repairs.
3. Provide thermometer wells and pressure gauge connections in all piping to and from the unit and located as near the unit as possible.
4. Provide convenient means for frequently cleaning the unit as suggested under "Maintenance."
5. Provide necessary air vent cocks for units so they can be purged to prevent or relieve vapor or gas binding of either the tube or the shell sides.
6. Foundations must be adequate so that exchangers will not settle and cause piping strains. Foundation bolts should be set to allow for settling inaccuracies. In concrete footings, pipe sleeves at least one size larger than bolt diameter slipped over the bolt and cast in place are best for this purpose, as they allow the bolt center to be adjusted after the foundation has set.
7. Loosen foundation bolts at one end of unit to allow free expansion of shells. Oval holes in foundation brackets are provided for this purpose.
8. Set exchangers level and square so that pipe connections may be made without forcing.
9. Inspect all openings in exchanger for foreign material. Remove all wooden plugs and shipping pads just before installing. Do not expose units to the elements with pads or other covers removed from nozzles or other openings since rain water may enter the unit and cause severe damage due to freezing.
10. Be sure the entire system is clean before starting operation to prevent plugging of tubes with sand or refuse. The use of strainers in settling tanks in pipe lines leading to the unit is recommended.
11. Drain connections should not be piped to a common closed manifold.
12. Steam hammer can cause serious damage to the tubes of any heat exchanger. A careful consideration of the following points before an installation is made can prevent costly repairs which may be caused by steam hammer.
 - a. A vacuum breaker and/or vent, should be used in accordance with the type of steam system installed
 - b. The proper trap for the steam system installed should be used.
 - c. The trap and the condensate return line to the trap should be properly sized for the total capacity of the converter.
 - d. The trap should be sized for the pressure at the trap, not the inlet pressure to the steam controller.
 - e. Condensate should be piped and pitched to a condensate receiver, condensate return pump or drain at an elevation below the heat exchanger.



CAUTION: During times of shutdown, volumetric expansion can occur. We recommend the installation of a properly sized relief valve on both sides of the heat exchanger.



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OPERATION

1. When placing a unit in operation, open the vent connections and start to circulate the cold medium only. Be sure that the passages in the exchanger are entirely filled with the cold fluid before closing the vents. The hot medium should then be introduced gradually until all passages are filled with liquid, close vents and slowly bring the unit up to temperature.
2. Start operation gradually. Do not admit hot fluid to the unit suddenly when empty or cold. Do not shock unit with cold fluid when unit is hot.

CAUTION: Fluids must be gradually introduced to the unit. Failure to do so can cause damage to the heat exchanger.

3. In shutting down, flow of hot medium should be shut off first. If it is necessary to stop circulation of cooling medium the circulation of hot medium should also be stopped by by-passing or otherwise.
4. Do not operate equipment under conditions in excess of those specified on nameplate.

WARNING: Failure to operate the heat exchanger within the design pressure and temperature on the nameplate may result in damage to the heat exchanger and potential injury to adjacent personnel.

5. Drain all fluids when shutting down to eliminate possibility of freezing and corrosion. To guard against water hammer, condensate should be drained from steam heaters and similar apparatus both when starting up and when shutting down.
6. In all installations there should be no pulsation of fluids since this causes vibration and strain with resulting leaks.
7. All gasketed joints should be checked after starting for leaks and tightened if necessary.

MAINTENANCE

1. Do not open heads until all pressure is off equipment and the unit is drained.
2. Do not blow out heat exchangers with air when operating fluids are of a flammable or otherwise hazardous nature.

WARNING: Proper precautions must be taken (special clothing, equipment, etc.) to protect personnel from injury due to escaping fluids.

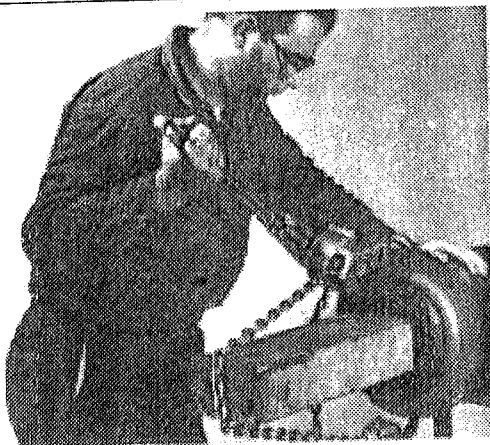


Illustration No. 1 - The method employed to push tube bundle out of shell.

3. Provide convenient means for frequently cleaning heat exchangers as suggested below:

- a. Circulating hot wash oil or light distillate through tubes or shell at good velocity will effectually remove sludge or other similar soft deposits.
- b. Soft salt deposits may be washed out by circulating hot fresh water.
- c. Some cleaning compounds on the market, such as "Oakite" may be used to advantage for removing sludge or coke, provided hot wash oil or water, as described above, does not give satisfactory results.
- d. If none of the above described methods are effective for the removal of hard scale or coke a mechanical means may be used. The interior of the tubes may be rodded.

WARNING: Care must be exercised when handling certain fluids. Follow manufacturers instructions. Use eye and skin protection. Wear a respirator when required.

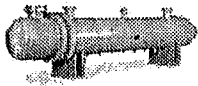
4. To clean or inspect inside of tubes, remove channel cover and rear head. On exchangers having bonnet type heads (without channel cover), piping must be disconnected and both heads removed.
5. Do not attempt to clean tubes by blowing steam through individual tubes. This overheats the tube and results in tube expansion strains and sometimes leaking tubes.
6. Frequently and at regular intervals, observe interior and exterior condition of all tubes and keep them clean. Frequency of cleaning should be according to scale build up.

CAUTION: Neglect in keeping all tubes clean may result in complete stoppage of flow through some tubes with consequent overheating of these tubes, resulting in severe expansion strains, leaking tube joints, and damage to the heat exchanger.

7. Exchangers subject to fouling or scaling should be cleaned periodically. A light sludge or scale coating on the tube greatly reduces its effectiveness. A marked increase in pressure drop and/or reduction in performance usually indicates cleaning is necessary. If the unit has been checked for air or vapor binding and this has been found not to be the cause, since the difficulty of cleaning increases rapidly as the scale thickens or deposits increases, the interval between cleanings should not be excessive.

B. Tube bundle removal

- a. During bundle removal, the dead weight of bundle should never be supported on individual tubes since the tubes are small and of relatively thin metal. Rest the bundle on the tube sheet, support plates, or wood blocks cut to fit periphery of the bundle.
- b. Be sure there is a soft wood filler between ends of tubes and steel bearing plate when pushing bundle out. On smaller bundles of 12" diameter and under, it is permissible to use hard wood block in place of wood filler and steel bearing plates. See Illustration No. 1.
- c. Tube bundles may be raised using slings formed by bending light plates into a "U" form and attaching lifting lugs to the ends of the sheets. Baffles can be easily bent and damaged if dragged over rough surfaces.



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CAUTION: Failure to follow the procedures mentioned in steps 8a, b, and c may result in damage to the tubes or tube joints leading to premature failure of the heat exchanger.

9. Removing the tube bundle:

Refer to drawings on pages 6 and 7 giving part names of models MEAH, OC, GC and ACA. The model MEAH tube bundles are more difficult to remove than the other models. The MEAH exchanger has the horizontal baffles (N) extending from the stationary tube steel (B) to about 5 inches of the packed floating tube sheet (E). Two rubber baffle strips (O) are adhered to the horizontal steel baffle (N). The rubber baffle strips seal the space between the shell and horizontal baffle, and prevent liquid by-passing. These rubber baffle strips (O) in the MEAH act as brakes against the shell (C) making bundle removal more difficult.

- If the exchanger is difficult to work upon, hoist it into the open after disconnecting piping.
- Remove front (A) and rear (F) heads. See Illustration No. 2.

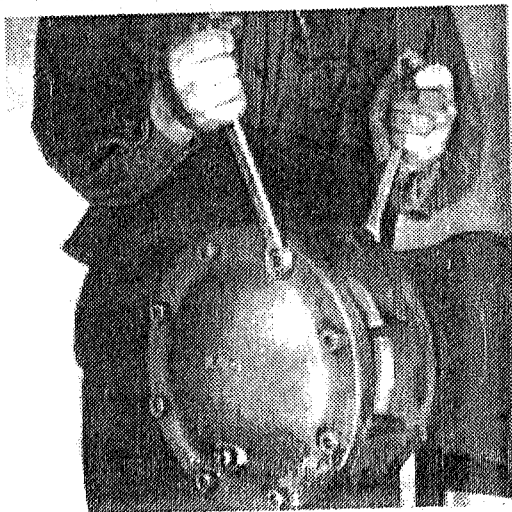


Illustration No. 2 - Removing rear head (completely remove front and rear heads).

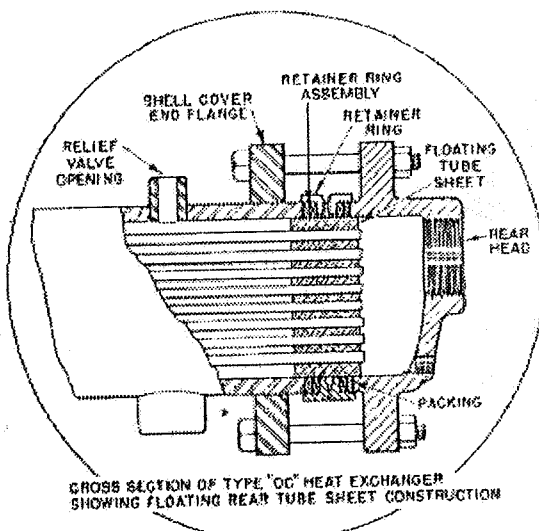


Illustration No. 3 - Cross section of type "OC" Heat Exchanger showing rear tube sheet construction. (See packing retainer detail)

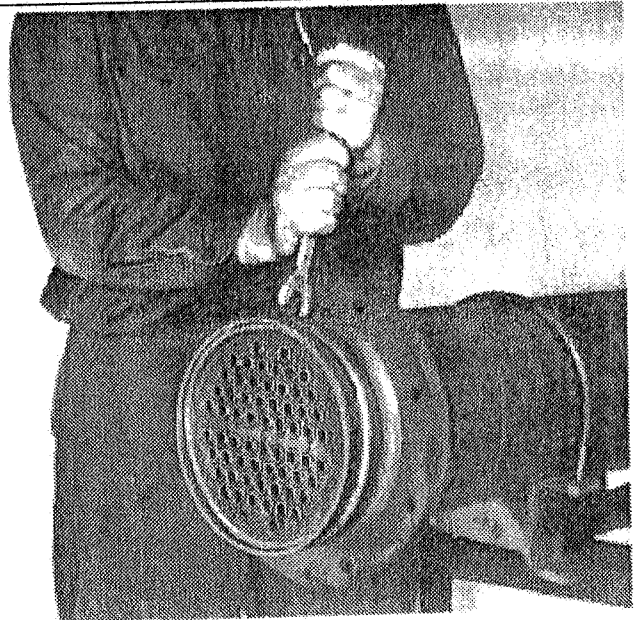


Illustration No. 4 - Prying off retainer rings.

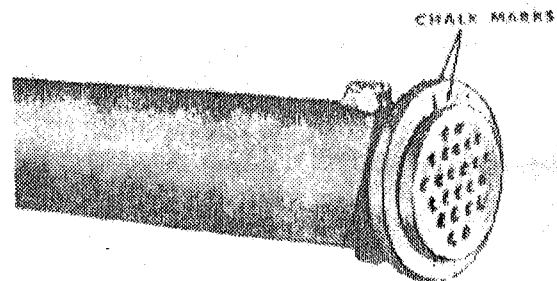
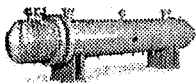


Illustration No. 5 - Marking of tube sheet and shell flange.

- Remove packing (G) and retaining rings (H). It may be necessary to tap lightly on the retainer rings for starting their removal. More detail of floating tube sheet is given in Illustration No. 3. Illustration No. 4 shows the method of prying retainer rings off. If tapping is necessary, be sure to tap evenly about the circumference so the retainer ring does not bind on the floating tube sheet.
- Mark tube sheet (B) and shell flange for later re alignment. Illustration No. 5
- Use chain jack. One suitable is Coffing Hoist, Model MA-15 1/4 ton capacity. Place short piece of 4x4 hardwood (oak) between chain of jack and floating tube sheet (E). See Illustration No. 1. Use longer 4x4 pieces as bundle is removed. Support the bundle by lifting or resting on the stationary tube sheet (B), as bundle is pushed out. Illustration No. 6



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10. Replacing the tube bundle:

The tube bundles for models ACA, GC, MEA, MEAH and OG (the packed floating tube sheet units) can be replaced using the tools and reverse procedure given for bundle removal. Usually, the bundles ACA, GC, MEA and OG, can be shoved back into the shell (C) manually without the use of the chain jack.

It may be necessary to use the chain jack when replacing MEAH bundles.



CAUTION: Model MEAH (which has the long steel horizontal baffle (N) with the rubber baffle sealing strips (O) adhered to the horizontal baffle)

The rubber baffle strips (O) are sealing strips between the shell (C) and horizontal baffle (N). The rubber baffle strips (O) must cut along the shell (C) towards the shell's liquid inlet connection. The liquid pressure forces the rubber baffle strips against shell and seals it. This is shown in illustration No. 7.



Illustration No. 6 - Removal of tube bundle showing method of tube bundle support.

NOTE: When replacing heads use a torque wrench.

- On heads (front heads) with gaskets, tighten $\frac{1}{2}$ " diameter bolts to 40 ft. lbs. and $\frac{3}{4}$ " diameter bolts to 80 ft. lbs. If the gasket joint still leaks, tighten in 5 ft. lbs. increments until leak stops.
- On heads with packing (rear heads), tighten initially to 50 ft. lbs. If the packing still leaks tighten in 5 ft. lbs. increments until leak stops. Do not exceed 100 ft. lbs. on the bolts for the packed heads.

*The above torque values apply to well lubricated nut bearing surfaces.

All bolted joints should be tightened uniformly and in a diametrically staggered pattern as illustrated below:

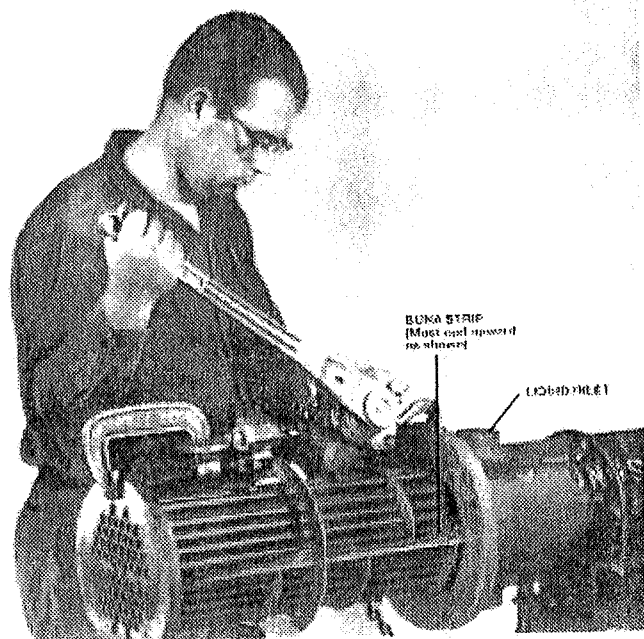
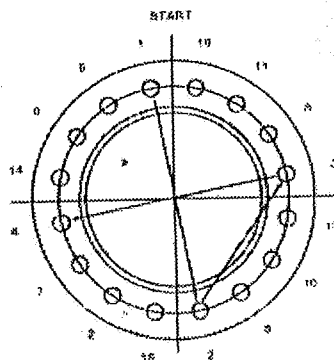
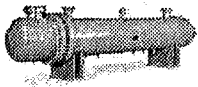
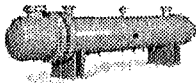


Illustration No. 7 - Position of rubber baffle strips and liquid inlet in relation to each other when assembling unit.



PART 5

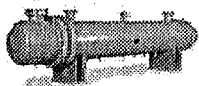


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FIGURE G-5.2M HEAT EXCHANGER SPECIFICATION SHEET

Customer				Job No.			
Address				Reference No.			
Plant Location				Proposal No.			
Service of Unit				Date		Rev.	
Size				item No.			
Type		(Hor/Vert)		Connected in		Parallel	
Surf/Unit (Gross/Eff.)		Sq m; Shells/Unit		Surf/Shell (Gross/Eff.)		Series	
PERFORMANCE OF ONE UNIT							
Fluid Allocation				Shell Side		Tube Side	
Fluid Name							
Fluid Quantity Total				kg/Hr			
Vapor (In/Out)							
Liquid							
Steam							
Water							
Noncondensable							
Temperature (In/Out)				°C			
Specific Gravity							
Viscosity, Liquid				Cp			
Molecular Weight, Vapor							
Molecular Weight, Noncondensable							
Specific Heat				J/kg °C			
Thermal Conductivity				W/m °C			
Latent Heat				J/kg@°C			
Inlet Pressure				kPa(abs.)			
Velocity				m/sec			
Pressure Drop, Allow. /Calc.				kPa			
Fouling Resistance (Min.)				Sq m °C / W			
Heat Exchanged				W:MTD (Corrected)		°C	
Transfer Rate, Service				Clean		W/Sq m °C	
CONSTRUCTION OF ONE SHELL							
Design / Test Pressure				kPag		Sketch (Bundle/Nozzle Orientation)	
Design temp. Max/Min				°C			
No. Passes per Shell							
Corrosion Allowance				mm			
Connections		In					
Size & Rating		Out					
		Intermediate					
Tube No.		OD		mm; Thk (Min/Avg)		mm; Pitch	
Tube Type		ID		OD		mm	
Shell						Material	
Channel or Bonnet						(Integ.) (Remov.)	
Tubesheet-Stationary						Channel Cover	
Floating Head Cover						Tubesheet-Floating	
Baffles-Cross		Type				Impingement Protection	
Baffles-Long						%Cut (Diam/Area)	
Supports-Tube		U-Bend				Spacing: c/c	
Bypass Seal Arrangement						Inlet	
Expansion Joint						mm	
pv-Inlet Nozzle		Bundle Entrance					
Gaskets-Shell Side						Tube Side	
Floating Head							
Code Requirements						TEMA Class	
Weight / Shell		Filled with Water				Bundle	
Remarks						kg	



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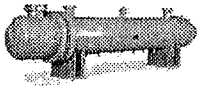
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COMPARISON BETWEEN DIFFERENT HEAT EXCHANGER TYPES

Type of Design	AEU BEU	AEW BEW	AEP BEP	AES BES	AET BET	CEU	NEU	AEM BEM AEL	NEN
Relative Cost(1 is lowest, 9 is highest)	3	6	7	8	9	2	1	5	4
Removable Bundle	Yes	Yes	Yes	Yes	Yes	Yes	No	No	No
Replaceable Bundle	Yes	Yes	Yes	Yes	Yes	Yes(1)	Yes(2)	No	No
Removable Bonnets/ Channel	Yes	Yes	Yes	Yes	Yes	Yes	No	Yes	No
Individual Tube Replacement	No	Yes	Yes	Yes	Yes	No	No	Yes	Yes
Tube side Cleaning	Chemical	Chemical & Mechanical				Chemical	Chemical	Chemical & Mechanical	
Shell side Cleaning	Chemical & Mechanical						Chemical	Chemical	Chemical
Double Tubesheet Design	Available	No	Available	No	No	Available			
Number of Tube side Passes	Any even number	1-2	Any	Any even number				Any	Any
Provision for Thermal Expansion	Each tube expands freely	Floating Tubesheet		Floating Head		Each tube expands freely		Expansion Joint when Applicable	

Chemical Cleaning Includes chemical cleaning, water blast and steam cleaning

- 1) Tube Bundle replaced with channel
- 2) Bundle can be replaced, however tube tubesheet would have to be cut off and re-welded.



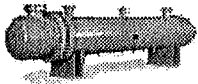
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PROPERTIES OF MATERIALS CARBON & LOW ALLOY STEEL

PROPERTIES OF MATERIALS (contd.)

Form	Specifications		P Number	Tensile Strength 1,000 psi	Yield Point 1,000 psi .	
	Number	Grade				
PLATE	SA - 283	C	1	55.0	30 .0	
	SA - 285	C	1	55.0	30 .0	
	SA - 515	55	1	55.0	30.0	
	SA - 515	60	1	60 .0	32.0	
	SA - 515	65	1	65.0	35.0	
	SA - 515	70	1	70.0	38.0	
	SA - 516	55	1	55.0	30.0	
	SA - 516	60	1	60.0	32.0	
	SA - 516	65	1	65.0	35.0	
	SA - 516	70	1	70.0	38.0	
FLANGE AND FITTING	SA - 105		1	70.0	36.0	
	SA - 181	I	1	60.0	30.0	
	SA - 350	LF 1 LF 2	1	60.0 70.0	30.0 36.0	
Seamless PIPE	SA - 53	B	1	60.0	35.0	
	SA - 106	B	1	60.0	35.0	
BOLTING	SA - 193	B7		125.0	105.0	DIAM> 2 1 / 2 in and < 4 in
	SA - 194	2H	-	55.0	-	
	SA - 307	B	-	55.0	-	

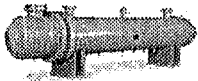


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Properties of Materials Carbon & Low Alloy Steel Maximum Allowable stress Values in Tension 1000 psi*												
Specification		For Metal temperature Not Exceeding Deg. F.										
Number	Grade	-20 to 650	700	750	800	850	900	950	1050	1100	1150	1200
SA - 283	C	12.7	-	-	-	-	-	-	-	-	-	-
SA - 285	C	13.8	13.3	12.1	10.2	8.4	6.5	-	-	-	-	-
SA - 515	55	13.8	13.3	12.1	10.2	8.4	6.5	4.5	2.5	-	-	-
SA - 515	60	15.0	14.4	13.0	10.8	8.7	6.5	4.5	2.5	-	-	-
SA - 515	65	16.3	15.5	13.9	11.4	9.0	6.5	4.5	2.5	-	-	-
SA - 515	70	17.5	16.6	14.8	12.0	9.3	6.5	4.5	2.5	-	-	-
SA - 516	55	13.8	13.3	12.1	10.2	8.4	6.5	4.5	2.5	-	-	-
SA - 516	60	15.0	14.4	13.0	10.8	8.7	6.5	4.5	2.5	-	-	-
SA - 516	65	16.3	15.5	13.9	11.4	9.0	6.5	4.5	2.5	-	-	-
SA - 516	70	17.5	16.6	14.8	12.0	9.3	6.5	4.5	2.5	-	-	-
SA - 105		17.5	16.6	14.8	12.0	9.3	6.5	4.5	2.5	-	-	-
SA - 181	I	15.0	14.4	13.0	10.8	8.7	6.5	4.5	2.5	-	-	-
SA - 350	LF 1	15.0	14.4	13.0	10.8	7.8	5.0	3.0	1.5	-	-	-
	LF 2	17.5	16.6	14.8	12.0	7.8	5.0	3.0	1.5	-	-	-
SA - 53	B	15.0	14.4	13.0	10.8	8.7	6.5	-	-	-	-	-
SA - 106	B	15.0	14.4	13.0	10.8	8.7	6.5	4.5	2.5	-	-	-
SA - 193	B7 $\frac{1}{2}$	25.0	25.0	23.6	21.0	17.0	12.5	8.5	4.5	-	-	-
SA - 194	2H	-	-	-	-	-	-	-	-	-	-	-
SA - 307	B	-	-	-	-	-	-	-	-	-	-	-

* The stress Values in this table may be interpolated to determine values for intermediate temperatures.



FLANGES

Flange Facing Finish

In pressure vessel construction only gasket seats of flanges, studed openings, etc. require special finish beyond that afforded by turning, grinding or milling.

The surface finish for flange facing shall have certain roughness regulated by Standard ASME B16.5. The roughness is repetitive deviation from the nominal surface having specified depth and width.

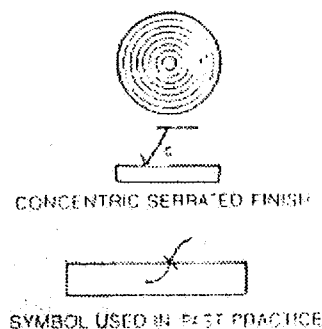
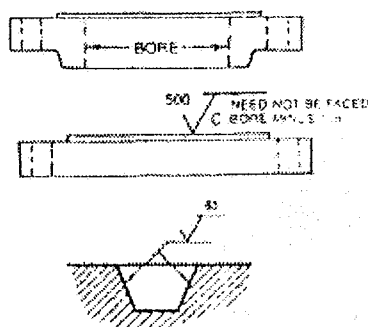
Raised faced flange shall have serrated finish having 24 to 40 grooves per inch. The cutting tool shall have an approximate 0.06 in. or larger radius resulting 500 microinch approximate roughness (ASME B16.5 - 6.3.4.1)

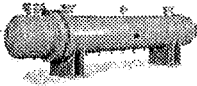
The side wall surface of gasket groove of ring joint flange shall not exceed 63 microinch roughness.(ASME B16.5 - 6.3.4.3)

Other finishes may be furnished by agreement between user and manufacturer.

The finish of contact faces shall be judged by visual comparison with Standard ASME B46-1. The center part of blind flanges need not to be finished within a diameter which equals or less than the bore minus one inch of the Joining flange, (ASME B16.5 - 6.3.3)

Surface symbol used to designate roughness $\sqrt{\quad}$, is placed either on the line indicating the surface or on a leader pointing to the surface as shown below, The numbers: 500 and 63 indicate the height of roughness; letter "c" the direction of surface pattern: "concentric-serrated" .





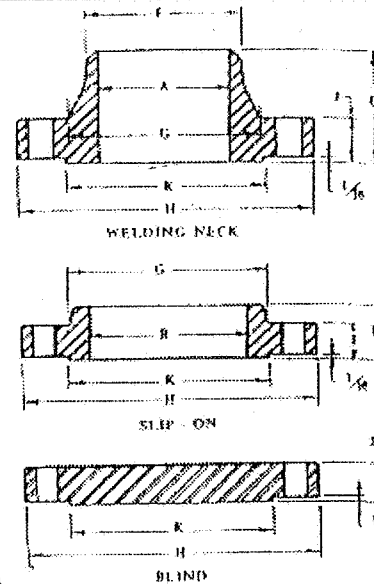
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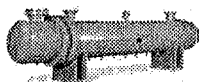
150 FLANGES STANDARD ASME B 16.5

1. All dimensions are in inches.
2. Material most commonly used, forged steel SA 181. available also in stainless steel SA 181 and non-ferrous metal.
3. The 1/16 in. raised face is included in dimensions C, D and J.
4. The lengths of stud bolts do not include the height of crown.
5. Bolt holes are 1/8 in. larger than bolt diameters.
6. Flanges bored to dimensions shown unless otherwise specified.
7. Flanges for pipe sizes 22, 26, 28 and 30 are not covered by ASME B16.5.

SEE FACING PAGE FOR DIMENSION K
AND DATA ON BOLTING.

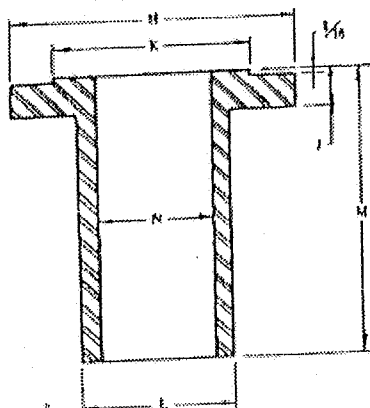


Nominal Pipe Size	Diameter of Bore		Length Hub Through		Diameter of Hub at Point of welding	Diameter of Hub at Base	Outside Diameter of Flange	Thickness of Flange
	A	B	C	D	E	G	H	J
1/2	.62	.88	1 7/8	5/8	.84	1 3/16	3 1/2	7/16
3/4	.82	1.09	2 1/16	5/8	1.05	1 1/2	3 7/8	1/2
1	1.05	1.36	2 3/16	11/16	1.32	1 15/16	4 1/4	9/16
1 1/4	1.38	1.70	2 1/4	13/16	1.66	2 5/16	4 5/8	5/8
1 1/2	1.61	1.95	2 7/16	7/8	1.90	2 9/16	5	11/16
2	2.07	2.44	2 1/2	1	2.38	3 1/16	6	3/4
2 1/2	2.47	2.94	2 3/4	1 1/8	2.88	3 9/16	7	7/8
3	3.07	3.57	2 3/4	1 3/16	3.50	4 1/4	7 1/2	15/16
3 1/2	3.55	4.07	2 13/16	1 1/4	4.00	4 13/16	8 1/2	15/16
4	4.03	4.57	3	1 5/16	4.50	5 5/16	9	15/16
5	5.05	5.66	3 1/2	1 7/16	5.56	6 7/16	10	15/16
6	6.07	6.72	3 1/2	1 9/16	6.63	7 9/16	11	1
8	7.98	8.72	4	1 3/4	8.63	9 11/16	13 1/2	1 1/8
10	10.02	10.88	4	1 15/16	10.75	12	16	1 3/16
12	12.00	12.88	4 1/2	2 3/16	12.75	14 3/4	19	1 1/4
14	13.25	14.14	5	2 1/4	14.00	15 3/4	21	1 3/8
16	15.25	16.16	5	2 1/2	16.00	18	23 1/2	1 7/16
18	17.25	18.18	5 1/2	2 11/16	18.00	19 7/8	25	1 9/16
20	19.25	20.20	5 11/16	2 7/8	20.00	22	27 1/2	1 11/16
22	21.25	22.22	5 7/8	3 1/8	22.00	24 1/4	29 1/2	1 13/16
24	23.25	24.25	6	3 1/4	24.00	26 1/8	32	1 7/8
26	To	26.25	5	3 3/8	26.00	28 1/2	34 1/4	2
28	be	28.25	5 1/16	3 7/16	28.00	30 3/4	36 1/2	2 1/16
30	specified	30.25	5 1/8	3 1/2	30.00	32 3/4	38 3/4	2 1/8



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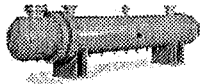
150 lb.

LONG WELDING NECK

1. All dimension are in inches.
2. Material most commonly used, forged steels SA 181. Available also in stainless steel, alloy steel and non-ferrous metal.
3. The 1/16 in. raised face is included in dimension J and M.
4. The length of bolts do not include the height of crown.
5. Bolt holes are 1/8 in. larger than bolt diameters.
6. Dimensions, M (length of welding necks) are based on data of major manufactures. Long welding necks with necks longer than listed are available on special order.

SEE PREVIOUS PAGE FOR DIMENSION J.

SEE PREVIOUS PAGE FOR DIMENSIONS									
Outside Diameter of Raised Face	No. of Holes	Diam. Of bolts	Bolt Circle	Length of Bolts		Outside Diameter	Length	Diameter of Bore	Nominal Pipe Size
K				1/16 Raised Face	Ring Joint	L	M	N	
1 3/8	4	1/2	2 3/8	2 1/2	--	2 2 3/8 2 5/8 3 1/4 3 1/2 4 4 1/4 4 7/8 5 1/2 6 1/2 7 3/4 9 3/4	9	Same as nominal size	1/2
1 11/16	4	1/2	2 3/4	2 1/2	--				3/4
2	4	1/2	3 1/8	2 3/4	3 1/4				1
2 1/2	4	1/2	3 1/2	2 3/4	3 1/4				1 1/4
2 7/8	4	1/2	3 7/8	3	3 1/2				1 1/2
3 5/8	4	5/8	4 3/4	3 1/4	3 3/4				2
4 1/8	4	5/8	5 1/2	3 1/2	4				2 1/2
5	4	5/8	6	3 3/4	4 1/4				3
5 1/2	8	5/8	7	3 3/4	4 1/4				3 1/2
6 3/16	8	5/8	7 1/2	3 3/4	4 1/4				5
7 5/16	8	3/4	8 1/2	4	4 1/2	6 1/2	5		
8 1/2	8	3/4	9 1/2	4	4 1/2	7 3/4	6		
10 5/8	8	3/4	11 3/4	4 1/4	4 3/4	9 3/4	8		
12 3/4	12	7/8	14 1/4	4 3/4	5 1/4	12	10		
15	12	7/8	17	4 3/4	5 1/4	14 3/8	12		
16 1/4	12	1	18 3/4	5 1/4	5 3/4	16	14		
18 1/2	16	1	21 1/4	5 1/2	6	18	16		
21	16	1 1/8	22 3/4	6	6 1/2	20	18		
23	20	1 1/8	25	6 1/4	6 3/4	22	20		
	20	1 1/4	27 1/4	6 1/2	7		22		
27 1/4	20	1 1/4	29 1/2	7	7 1/2	26 1/4	24		
29 1/4	24	1 1/4	31 3/4	7	--	28 1/2	26		
31 1/4	28	1 1/4	34	7	--	30 1/2	28		
33 3/4	28	1 1/4	36	7 1/4	--	32 1/2	30		



Heat Exchanger Design Course

By: Eng. Mohamed A. Ebcid

PRESSURE – TEMPERATURE RATING FOR STEEL PIPE FLANGES AND FLANGED FITTINGS

American National Standard ASME B16.5 – 1981

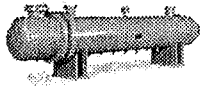
CLASS	150 lb	300 lb	400 lb	600 lb	900 lb.	1500 lb.	2500 lb.
Hydrostatic TEST PRESSURE, PSIG	450	1125	1500	2225	3350	5575	9275
Temperature F	MAXIMUM ALLOWABLE NON-SHOCK PRESSURE PSIG.						
-20 to 100	285	740	990	1480	2220	3705	6170
200	260	675	900	1350	2025	3375	5625
300	230	655	875	1315	1970	3280	5470
400	200	635	845	1270	1900	3170	5280
500	170	600	800	1200	1795	2995	4990
600	140	550	730	1095	1640	2735	4560
650	125	535	715	1075	1610	2685	4475
700	110	535	710	1065	1600	2665	4440
750	95	505	670	1010	1510	2520	4200
800	80	410	550	825	1235	2060	3430
850	65	270	355	535	805	1340	2230
900	50	170	230	345	515	860	1430
950	35	105	140	205	310	515	860
1000	20	50	70	105	155	260	430

Rating apply to materials:

SA-105^{1,2} SA-515-70² SA-516-70² SA-181-70^{1,2} SA-350-LF2
SA-537-C1.1³ SA-216-WCB²

Notes:

- For service temperatures above 850 F it is recommended that killed steels containing not less than 0.10% residual silicon be used.
- Upon prolonged exposure to temperatures above 800 F, the carbide phase of carbon steel may be converted to graphite.
- The material shall not be used in thickness above 2½ in.
- Flanges of ASME B16.5 shall not be used for higher ratings except where it is justified by the design methods of the Code.
- Ratings are maximum allowable non-shock working pressures expressed as gage pressure, at the tabulated temperatures and may be interpolated between temperatures shown.
- Temperatures are those on the inside of the pressure-containing shell of the flange. In general, it is the same as that of the contained material.



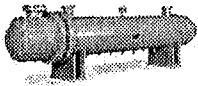
Heat Exchanger Design Course

By: Eng. Mohamed A. Ebeid

Dimensions and Masses of Steel Pipe

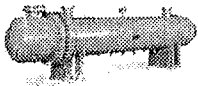
Referring to table, the upper figures in each square denote wall thickness in mm and the lower figures denote mass per meter, in kilograms.

Nom. pipe size imp. units	Equiv. nom. pipe size MM	Outside diam.	SCHEDULE														Double Extra strong
			10	20	30	Std. wall	40	60	Extra strong	80	100	120	140	160			
1/2	12.70	21.34	----	----	----	2.77 1.26	2.77 1.26	----	3.73 1.61	3.73 1.61	----	----	----	4.75 1.92	2.47 2.53		
3/4	19.05	26.67	----	----	----	2.87 1.67	2.87 1.67	----	3.91 2.17	3.91 2.17	----	----	----	5.54 2.87	7.02 3.61		
1	25.4	33.40	----	----	----	3.38 2.48	3.38 2.48	----	4.35 3.21	4.35 3.21	----	----	----	6.35 4.20	9.09 5.41		
1 1/4	31.75	42.16	----	----	----	3.56 3.36	3.56 3.36	----	4.85 4.43	4.85 4.43	----	----	----	6.35 5.57	9.70 7.70		
1 1/2	38.10	40.26	----	----	----	3.68 4.02	3.68 4.02	----	5.08 5.37	5.08 5.37	----	----	----	7.14 7.10	10.16 9.07		
2	50.80	60.33	----	----	----	3.91 5.39	3.91 5.39	----	5.54 7.42	5.54 7.42	----	----	----	8.71 11.00	11.07 13.35		
2 1/2	63.50	73.03	----	----	----	5.16 8.56	5.16 8.56	----	7.01 11.32	7.01 11.32	----	----	----	9.53 14.79	14.02 20.25		
3	76.20	88.90	----	----	----	5.49 11.20	5.49 11.20	----	7.62 15.15	7.62 15.15	----	----	----	11.13 21.16	15.24 27.46		
3 1/2	88.90	101.60	----	----	----	5.74 13.46	5.74 13.46	----	8.08 18.49	8.08 18.49	----	----	----	11.13 21.16	16.15 33.77		
4	101.60	114.30	----	----	----	6.02 15.95	6.02 15.95	----	8.56 23.14	8.56 23.14	----	11.13 20.10	----	13.49 30.27	17.37 40.70		
5	127.00	141.30	----	----	----	6.55 21.61	6.55 21.61	----	9.53 30.71	9.53 30.71	----	12.70 39.97	----	15.83 40.71	19.05 56.98		
6	152.40	160.28	----	----	----	7.11 20.04	7.11 20.04	----	10.97 42.23	10.97 42.23	----	14.27 53.70	----	18.24 66.95	21.95 78.57		
8	203.20	219.08	----	6.35 33.05	7.04 26.51	8.18 42.20	8.18 42.20	10.31 52.60	12.70 64.13	12.70 64.13	15.06 75.10	18.24 83.61	20.62 100.15	23.01 110.39	27.23 111.47		
10	250.40	273.05	----	6.35 41.44	7.80 50.51	9.27 59.87	9.27 59.87	12.70 60.91	12.70 60.91	15.06 95.08	18.24 113.17	21.41 131.04	25.40 153.90	28.58 178.93	33.32 190.00		
12	304.80	323.85	----	6.35 49.34	8.38 64.69	9.53 73.25	10.31 79.16	14.27 100.13	12.70 96.69	17.45 130.82	21.41 158.44	25.40 185.47	28.58 206.45	33.32 216.88	39.32 240.00		
14	355.60	355.60	6.35 50.26	7.92 67.52	9.53 80.65	9.53 80.65	11.13 93.66	15.06 125.60	12.70 106.55	19.05 156.04	23.00 192.22	27.76 222.89	31.75 251.59	35.71 279.52	40.46 300.00		
16	406.40	406.40	6.35 62.15	7.92 77.39	9.53 92.49	9.53 92.49	12.70 122.33	16.56 158.09	12.70 122.33	21.41 201.69	26.19 243.62	30.94 284.70	36.57 330.33	40.46 367.27	45.24 400.00		
18	457.20	457.20	6.35 70.04	7.92 87.25	11.13 121.28	9.53 104.33	14.27 154.82	19.05 204.22	12.70 138.12	23.60 252.37	29.36 307.36	34.93 350.84	39.67 405.31	45.24 455.98	50.00 500.00		
20	508.00	508.00	6.35 77.93	9.53 116.17	12.70 153.90	9.53 116.17	15.06 181.66	20.62 245.94	12.70 153.90	26.19 308.71	32.54 378.52	38.10 438.03	44.45 504.15	49.99 560.18	55.00 600.00		
24	609.60	609.60	6.35 93.72	9.53 139.85	14.27 208.10	9.53 139.85	17.45 252.99	24.59 341.33	12.70 185.74	30.94 438.02	38.09 543.02	46.02 634.66	52.37 714.07	59.51 800.99	65.00 700.00		
30	762.00	762.00	8.03 117.48	12.70 232.83	15.88 289.81	9.53 175.36	----	----	12.70 732.83	----	----	----	----	----	----		



DEFINITIONS

1. Baffle is a device to direct the shell side fluid across the tubes for optimum heat transfer.
2. Baffle and Support Plate Tube Hole Clearance is the diametral difference between the nominal tube OD and the nominal tube hole diameter in the baffle or support plate.
3. Consequential Damages are indirect liabilities lying outside the heat exchanger manufacturer's stated equipment warranty obligations.
4. Double Tubesheet Construction is a type of construction in which two (2) spaced tubesheets or equivalent are employed in lieu of the single tubesheet at one or both ends of the heat exchanger.
5. Effective Shell and Tube Side Design Pressures are the resultant load values expressed as uniform pressures used in the determination of tubesheet thickness for fixed tubesheet heat exchangers and are functions of the shell side design pressure, the tube side design pressure, the equivalent differential expansion pressure and the equivalent bolting pressure.
6. Equivalent Bolting Pressure is the pressure equivalent resulting from the effects of bolting loads imposed on tubesheets in a fixed tubesheet heat exchanger when the tubesheets are extended for bolting as flanged connections.
7. Equivalent Differential Expansion Pressure is the pressure equivalent resulting from the effect of tubesheet loadings in a fixed tubesheet heat exchanger imposed by the restraint of differential thermal expansion between shell and tubes.
8. Expanded Tube Joint is the tube-to-tubesheet joint achieved by mechanical or explosive expansion of the tube into the tube hole in the tubesheet.
9. Expansion Joint "J" Factor is the ratio of the spring rate of the expansion joint to the sum of the axial spring rate of the shell and the spring rate of the expansion joint.
10. Flange Load Concentration Factors are factors used to compensate for the uneven application of bolting moments due to large bolt spacing.
11. Minimum and Maximum Baffle and Support Spacings are design limitations for the spacing of baffles to provide for mechanical integrity and thermal and hydraulic effectiveness of the bundle. The possibility for induced vibration has not been considered in establishing these values.
12. Normal Operating Conditions of a shell and tube heat exchanger are the thermal and hydraulic performance requirements generally specified for sizing the heat exchanger.
13. Pulsating Fluid Conditions are conditions of flow generally characterized by rapid fluctuations in pressure and flow rate resulting from sources outside of the heat exchanger.
14. Seismic Loadings are forces and moments resulting in induced stresses on any member of a heat exchanger due to pulse mode or complex waveform accelerations to the heat exchanger, such as those resulting from earthquakes.
15. Shell and Tube Mean Metal Temperatures are the average metal temperatures through the shell and tube thicknesses integrated over the length of the heat exchanger for a given steady state operating condition.
16. Shut-Down Conditions are the conditions of operation which exist from the time of steady state operating conditions to the time that flow of both process streams has ceased.
17. Start-Up Conditions are the conditions of operation which exist from the time that flow of either or both process streams is initiated to the time that steady state operating conditions are achieved.
18. Support plate is a device to support the bundle or to reduce unsupported tube span without consideration for heat transfer.
19. Tubesheet Ligament is the shortest distance between edge of adjacent tube holes in the tube pattern.
20. Welded Tube Joint is a tube-to-tubesheet joint where the tube is welded to the tubesheet.



Heat Exchanger Design Course

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Abrasion - The removal of surface material from any solid through the frictional action of another solid, a liquid, or a gas or combination thereof.

Absolute Pressure - The pressure above the absolute zero value of pressure that theoretically obtains in empty space or at the absolute zero of temperature, as distinguished from gage pressure.

Alloy - Any of a large number of substances having metallic properties and consisting of two or more elements; with few exceptions, the components are usually metallic elements.

Angle Joint - A joint between two members located intersecting planes between zero (a butt joint) and 90 deg. (a corner joint). (Code UA-60)

Angle Valve - A valve, usually of the globe type, in which the inlet and outlet are at right angles.

Annealing - Annealing generally refers to the heating and controlled cooling of solid material for the purpose of removing stresses, making it softer, refining its structure or changing its ductility, toughness or of her properties. Specific heat treatments covered by the term annealing include black annealing, blue annealing, box annealing, bright annealing, full annealing, graphitizing, maleabilizing and process annealing.

Arc Welding - A group of welding processes wherein coalescence is produced by heating with an electric arc, with or without the application of pressure and with or without the use of filler metal.

Automatic Welding - Welding with equipment which performs the entire welding operation without constant observation and adjustment of the controls by an operator. The equipment may or may not perform the loading and unloading of the work.

Backing - Material backing up the joint during welding to facilitate obtaining a sound weld at the root. Backing Strip is a backing in a form of a strip.



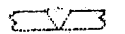
Brittle Fracture - The tensile failure with negligible plastic deformation of an ordinary ductile metal.

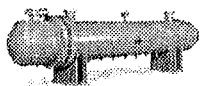
Brittleness - Materials are said to be brittle when they show practically no permanent distortion before failure.

Bushing - A pipe fitting for connecting a pipe with a female fitting of larger size. It is a hollow plug with internal and external threads.

Butt Weld - A weld joining two members lying approximately in the same plane. Butt welded joints in pressure vessel construction shall have complete penetration and fusion.

Types of butt welded joints: Single or Double Beveled Joint, Square Butt Joint, Full Penetration, Partial Penetration Butt Joints. Butt Joints with or without backing strips.





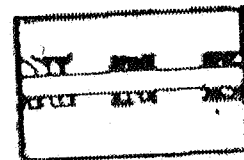
Heat Exchanger Design Course

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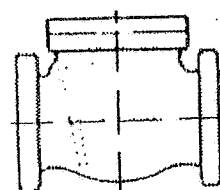
Centroid or an Area (Center or Gravity of Its Area)

- That point in the plane of the area about any axis through which the moment of the area is zero; it coincides with the center of gravity of the area materialized as an infinitely thin homogeneous and uniform plate.

Chain Intermittent fillet Welds - Two lines of intermittent fillet welding in a tee or lap joint, in which the increments of welding in one line are approximately opposite to those in the other line.



Check Valve - A valve designed to allow a fluid to pass through in one direction only. A common type has a plate so suspended that the reverse flow aids gravity in forcing the plate against a seat, shutting off reverse flow.



Chipping - One method of removing surface defects such as small fissures or seams from partially worked metal. If not eliminated, the defects might carry through to the finished material. If the defects are removed by means of a gas torch the term "deseaming" or "scarfing" is used.

Clad Vessel - A vessel made from plate having a corrosion resistant material integrally bonded to a base of less resistant material. (Code UA-60)

Complete fusion - Fusion which has occurred over the entire base-metal surfaces exposed for welding.

Complete Penetration - Penetration which extended completely through the joint.

Corner Joint - A welded joint at the junction of two parts located approximately at right angles to each other.

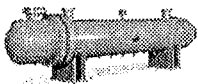
Corrosion - Chemical erosion by motionless or moving agents. Gradual destruction of a metal or alloy due to chemical processes such as oxidation or the action of a chemical agent.

Corrosion Fatigue - Damage to or failure of a metal due to corrosion combined with fluctuating fatigue stresses.

Coupling - A threaded sleeve used to connect two pipes. They have internal threads at both ends to fit external threads on pipe.

Creep - Continuous increase in deformation under constant or decreasing stress. The term is usually used with reference to the behavior of metals under tension at elevated temperatures. The similar yielding of a material under compressive stress is usually called plastic flow or flow.

Damaging Stress - The least unit stress, of a given kind and for a given material and condition of service, that will render a member unfit for service before the end of its normal life. It may do this by producing excessive set, or by causing creep to occur at an excessive rate. Or by causing fatigue cracking, excessive strain hardening, or rupture.



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Deformation (Strain) - Change in the form or in the dimension of a body produced by stress. Elongation is often used for tensile strain, compression or shortening for compressive strain, and detrusion for shear strain. Elastic deformation is such deformation as disappears on removal of stress; permanent deformation IS such deformation as remains on removal of stress.

Design Pressure - The pressure used in determining the minimum permissible thickness or physical characteristics of the different parts of the vessel. (Code UA-60)

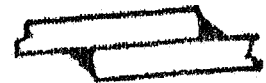
Design Temperature - The mean metal temperature (through the thickness) expected under operating conditions for the part considered. (Code UG-20)

Discontinuity, Gross Structural - A source of stress or strain intensification which affects a relatively large portion of a structure and has a significant effect on the overall stress or strain pattern or on the structure as a whole. Examples of gross structural discontinuities are head-to-shell and flange-to-shell junctions, nozzles, and junctions between shells of different diameters or thicknesses.

Discontinuity, Local Structural - A source of stress or strain intensification which affects a relatively small volume of material and does not have a significant effect on the overall stress or strain pattern or on the structure as a whole. Examples are small fillet radii, small attachments, and partial penetration welds.

Double-Welded Butt Joint - A butt joint welded from both side.

Double-Welded Lap Joint - A lap joint in which the overlapped edges of the members to be joined are welded along the edges of both members.



Ductility - The ability of a metal to stretch and become permanently deformed without breaking or cracking. Ductility is measured by the percentage reduction in area and percentage elongation of test bar.

Eccentricity - A load or component of a load normal to a given cross section of a member is eccentric with respect to that section if it does not act through the centroid. The perpendicular distance from the line of action of the load to either principal central axis is the *eccentricity* with respect to that axis.

Efficiency of a Welded Joint - The efficiency of a welded joint is expressed as a numerical quantity and is used in the design of a joint as a multiplier of the appropriate allowable stress value. (Code UA-60)

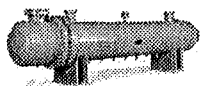
Elastic - Capable of sustaining stress without permanent deformation; the term is also used to denote conformity to the law of stress-strain proportionality. An elastic stress or elastic strain is a stress or strain within the elastic limit.

Elastic Limit The least stress that will cause permanent set.

Electroslag Welding - A welding process in which consumable electrodes are fed into a joint containing flux; the current melts the flux, and the flux in turn melts the faces of the joint and the electrodes, allowing the weld metal to form a continuously cast ingot between the joint faces. Used in pressure vessel construction when back of the welding is not accessible. All butt welds joined by electroslag welding shall be examined radio graphically for their full length. (Code UW-11) (a) (6)

Endurance Limit (Fatigue Strength) - By endurance limit of a material is usually meant the maximum stress which can be reversed an indefinitely large number of times without producing fracture.

Erosion-Corrosion - Attack on a metal surface resulting from the combined effects of erosion and corrosion.



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Expansion Joint - A joint whose primary purpose is not to join pipe but to absorb that longitudinal expansion in the pipe line due to heat.

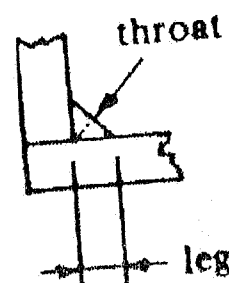
Factor of Safety - The ratio of the load that would cause failure of a member or structure, to the load that is imposed upon it in service.

Fatigue - Tendency of materials to fracture under many repetitions of a stress considerably less than the ultimate static strength.

Fiber Stress - A term used for convenience to denote the longitudinal tensile or compressive stress in a beam or other member subject to bending. It is sometimes used to denote this stress at the point or points most remote from the neutral axis, but the term stress in extreme fiber is preferable for this purpose. Also, for convenience, the longitudinal elements or filaments of which a beam may be imagined as composed are called fibers.

Fillet Weld

A weld of approximately triangular cross section joining two surfaces approximately at right angles to each other. The effective stress-carrying area of a fillet weld is assumed to be the product of the throat dimension and the length of the weld. Fillet welds are specified by their leg dimension.



The throat dimension of an equal legged fillet weld is 0.707 times the leg dimension. Fillet welds may be employed as strength welds for pressure parts of vessels within the limitations given in Table UW-12 of the Code. The allowable load on fillet welds shall equal the product of the weld area (based on minimum leg dimension), the allowable stress value in tension of the material being welded, and a joint efficiency of 55%. (Code UW.3) The allowable stress values for fillet welds attaching nozzles and their reinforcements to vessels are (in shear) 49 % of stress value for the vessel material. (Code (UW - 15)

Filler Metal - Material to be added in making a weld.

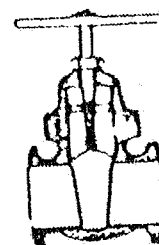
Full Fillet Weld - A fillet weld whose size is equal to the thickness of the thinner member joined.

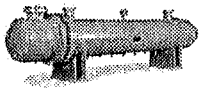
Gage Pressure - The amount by which the total absolute pressure exceeds the ambient atmospheric pressure.

Galvanizing - Applying a coating of zinc to ferrous articles. Application may be by hot dip process or electrolysis.

Gas Welding - A group of welding processes wherein coalescence is produced by heating with a gas flame with or without application of pressure and with or without the use of filler metal.

Gate Valve - A valve employing a gate, often wedge-shaped, allowing fluid to flow when the gate is lifted from the seat. Such valves have less resistance to flow than globe valves.

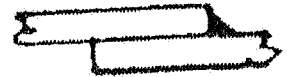




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Lap Joint - A welded joint in which two overlapping metal parts are joined by means of a fillet plug or slot welds.



Layer or Laminated Vessel - A vessel having a shell which is made up of two or more separate layers. (Code UA-60)

Leg - See under Fillet Weld.

Lethal Substances - Poisonous gases or liquids of such a nature that a very small amount of the gas or of the vapor of the liquid is dangerous to life when inhaled. It is the responsibility of the user of the vessel to determine that the gas or liquid is lethal. (Code UW-2)

Ligament - The section of solid material in a tube sheet or shell between adjacent holes.

Lined Vessel - A vessel having a corrosion resistant lining attached intermittently to the vessel wall. (Code UA-60)

Liquid Penetrant Examination (PT) - A method of nondestructive examination which provides for the detection of discontinuities open to the surface in ferrous and nonferrous materials which are nonporous. Typical discontinuities detectable by this method are cracks, seams, laps, cold shuts, and laminations. (Code UA-60)

Loading - Loadings (loads) are the results of various forces. The loadings to be considered in designing a vessel: internal or external pressure, impact loads, weight of the vessel, superimposed loads, wind and earthquake, local load, effect of temperature gradients. (Code UG-22)

Low-Alloy Steel - A hardenable carbon steel generally containing not more than about 1% carbon and one or more of the following alloyed components: < (less than) 2% manganese, < 4% nickel, < 2% chromium, 0.6% molybdenum, and < 0.2% vanadium.

Magnetic Particle Examination (MT) - A method of detecting cracks and similar discontinuities at or near the surface in iron and the magnetic alloys of

Malleable Iron - Cast iron heat-treated to reduce its brittleness. The process enables the material to stretch to some extent and to stand greater shock.

Material Test Report - A document on which the material manufacturer records the results of tests examinations, repairs, or treatments required by the basic material specification to be reported. (Code UA-60)

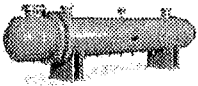
Maximum Allowable Stress Value - The maximum unit stress permissible for any specified material that may be used in the design formulas given in the Code. (UG-23)

Maximum Allowable Working Pressure - The maximum gage pressure permissible at the top of a completed vessel in its operating position for a designated temperature. This pressure is based on the weakest element of the vessel using nominal thicknesses exclusive of allowances for corrosion and thickness required for loadings other than pressure. (Code UA-60)

Membrane Stress - The component of normal stress which is uniformly distributed and equal to the average value of stress across the thickness of the section under consideration.

Metal Arc Welding - An arc welding process in which the electrode supplies the filler metal to the weld.

Modulus of Elasticity (Young's Modulus) - The rate of change of unit tensile or compressive stress



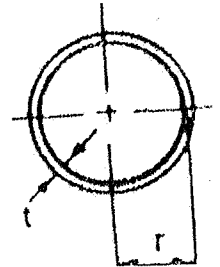
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with respect to unit tensile or compressive strain for the condition of uniaxial stress within the proportional limit. For most, but not all materials, the modulus of elasticity is the same for tension and compression. For nonisotropic materials such as wood, it is necessary to distinguish between the moduli of elasticity in different directions.

Modulus of Rigidity (Modulus of Elasticity In Shear) - The rate of change of unit shear stress with respect to unit shear strain, for the condition of pure shear within the proportional limit.

Moment of Inertia of an Area (Second Moment of an Area) The moment of inertia of an area with respect to an axis is the sum of the products obtained by multiplying each element of the area by the square of its distance from the axis. The Moment of Inertia (I) for thin walled cylinder about its transverse axis; $I = \pi r^3 t$



where r = mean radius of cylinder
 t = wall thickness

Needle Valve - A valve provided with a long tapering point in place of the ordinary valve disk. The tapering point permits fine graduation of the opening.

Neutral Axis - The line of zero fiber stress in any given section of a member subject to bending; it is the line formed by the intersection of the neutral surface and the section.

Neutral Surface - The longitudinal surface of zero fiber stress in a member subject to bending; it contains the neutral axis of every section.

Nipple - A tubular pipe fitting usually threaded on both ends and under 12 inches in length. Pipe over 12 inches long is regarded as cut pipe.

Non-Pressure Welding - A group of welding processes in which the weld is made without pressure.

Normalizing - Heating to about 1000 F. above the critical temperature and cooling to room temperature in still air. Provision is often made in normalizing for controlled cooling at a slower rate, but when the cooling is prolonged the term used is annealing.

Notch Sensitivity - A measure of the reduction in strength of a metal caused by the presence of a notch.

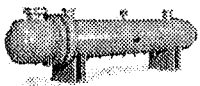
Notch Strength - The ratio of maximum tensional load required to fracture a notched specimen to the original minimum cross sectional area.

Notch Test - A tensile or creep test of a metal to determine the effect of a surface notch.

Operating Pressure - The pressure at the top of a pressure vessel at which it normally operates. It shall not exceed the maximum allowable working pressure and it is usually kept at a suitable level below the setting of the pressure relieving devices to prevent their frequent opening. (Code UA-60)

Operating or Working Temperature - The temperature that will be maintained in the metal of the part of the vessel being considered for the specified operation of the vessel (see UG-20 and UG-23). (Code UA-60)

Oxidation or scaling of metals occurs at high temperatures and access of air. Scaling of carbon steels from air or steam is negligible up to 1000 °F. Chromium increases scaling resistance of carbon steels. Decreasing oxidation resistance makes austenitic stainless steels unsuitable for operating temperatures above 1500°F.



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P-Number - The number of welding procedure-group. The classification of materials based on hardenability characteristic and the purpose of grouping is to reduce the number of weld procedures. (Code Section IX)

All carbon steel material listed in the Code (with the exception of SA-612) are classified as P-No. 1.

Pass - The weld metal deposited by one progression along the axis of a weld.

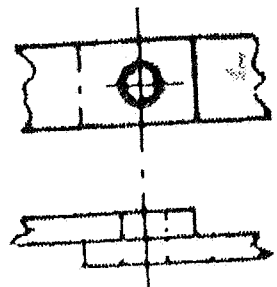
Plasticity - The property of sustaining appreciable (visible to the eye) permanent deformation without rupture. The term is also used to denote the property of yielding or flowing under steady load.

Plug Valve - One with a short section of a cone or tapered plug through which a hole is cut so that fluid can flow through when the hole lines up with the inlet and outlet, but when the plug is rotated 90°, flow is blocked.

Plug Weld - A weld made in a circular hole in one member of a lap joint. The hole may or may not be partially or completely filled with weld metal.

For pressure vessel construction plug welds may be used in lap joints in reinforcements around openings, in non pressure structural attachments (Code UW-17) and for attachment of heads with certain restrictions. (Code Table UW-12)

Pneumatic Test - The completed vessel may be tested by air pressure in lieu of hydrostatic test when the vessel cannot safely be filled with water or the traces of testing liquid cannot be tolerated (in certain services). The pneumatic test pressure shall be 1.1 times the maximum allowable working pressure to be stamped on the vessel. (Code UG-100)



Poisson's Ratio - The ratio of lateral unit strain to longitudinal unit strain, under the condition of uniform and uniaxial longitudinal stress within the proportional limit.

Porosity - Gas pockets or voids in metal. (Code UA-60)

Postweld Heat Treatment - Heating a vessel to a sufficient temperature to relieve the residual stresses which are the result of mechanical treatment and welding.

Heat exchanger and parts shall be postweld heat treated:

When the vessels are to contain lethal substances, (Code UW -2)
Unfired Steam Boilers (UW -2)

When the carbon (P-No. 1) material thickness exceeds 1½ in. at welded connections and attachments (see Code Table UCS-S6 for exceptions).

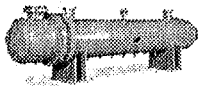
Preheating - Heat applied to base metal prior to welding operations.

Pressure Relief Valve - A valve which relieves pressure beyond a specified limit and recloses upon return to normal operating conditions.

Pressure Vessel - A metal container generally cylindrical or spheroid, capable of withstanding various loadings.

Pressure Welding - A group of welding processes wherein the weld is completed by use of pressure.

Primary Stress - A normal stress or a shear stress developed by the imposed loading which is necessary to satisfy the simple laws of equilibrium of external and internal forces and moments. The basic characteristics of a primary stress is that it is not self-limiting. Primary stresses which considerably exceed the strength will result in failure or at least gross distortion. A thermal stress is not classified as a primary stress. Primary membrane stress divided into "general" and "local" categories. A general



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primary membrane stress is one which is so distributed in the structure that no redistribution of load occurs as a result yielding. Examples of primary stress are general membrane stress in a circular cylindrical or a spherical shell due to internal pressure or to distributed live loads; bending stress in the central portion of a flat head due to pressure.

Quench Annealing - Annealing an austenitic ferrous alloy by heating followed by quenching from solution temperatures. Liquids used for quenching are oil, fused salt or water, into which a material is plunged.

Radiographing - The process of passing electronic radiations through an object and obtaining a record of its soundness upon a sensitized film. (Code UA-60)

Radius of Gyration - The radius of gyration of an area with respect to a given axis is the square root of the quantity obtained by dividing the moment of inertia of the area with respect to that axis by the area.

Random Lengths - A term indicating no specified minimum or maximum length with lengths falling within the range indicated.

Refractory - A material of very high melting point with properties that make it suitable for such uses as high-temperature lining.

Residual Stress - Stress remaining in a structure or member as a result of thermal or mechanical treatment, or both.

Resistance Welding - A pressure welding process wherein the heat is produced by the resistance to the flow of an electric current.

Root of Weld - The bottom of the weld.

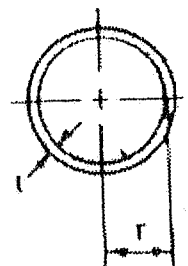
Scale - An iron oxide formed on the surface of hot steel, sometimes in the form of large sheets which fall off when the sheet is rolled.

Scarf - Edge preparation; preparing the contour on the edge of a member for welding.

Seal Weld - Seal weld used primarily to obtain tightness.

Secondary Stress - A normal stress or a shear stress developed by the constraint of adjacent parts or by self constraint of a structure. The basic characteristic of a secondary stress is that it is self-limiting. Local yielding and minor distortions can satisfy the conditions which cause the stress to occur and failure from one application of the stress is not to be expected. Examples of secondary stress are: general thermal stress; bending stress at a gross structural discontinuity.

Section Modulus - The term pertains to the cross section of a beam. The section modulus with respect to either principal central axis is the moment of inertia with respect to that axis divided by the distance from that axis to the most remote point of the section. The section modulus largely determines the flexural strength of a beam of given material.

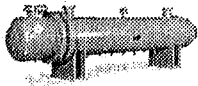


Section Modulus (Z) of a thin walled cylinder ($r > 10t$) about its transverse axis:

$$Z = p r^2 t$$

where r = mean radius of cylinder, in.

t = wall thickness, in.



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Shell - Structural element made to enclose some space. Most of the shells are generated by the revolution of a plane curve.

In the terminology of this book shell is the cylindrical part of a vessel or a spherical vessel is called also a spherical shell.

Shear Stress - The component of stress tangent to the plane of reference.

Shielded Metal-Arc Welding - An arc welding process wherein coalescence is produced by heating with an electric arc between a covered metal electrode and the work. Shielding is obtained from decomposition of the electrode covering. Pressure is not used and filler metal is obtained from the electrode.

Single-Welded Butt Joint - A butt joint welded from one side only.

Single-Welded Lap Joint - A lap joint in which the overlapped edges of the members to be joined are welded along the edge of one member.

Size of Weld - Groove Weld: The depth of penetration. Equal Leg Fillet Weld: the leg length of the largest isosceles right-triangle which can be inscribed within the fillet weld cross section.

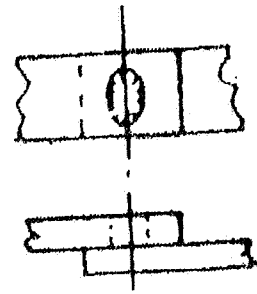
Unequal Leg Fillet Weld: The leg length of the largest right triangle which can be inscribed within the fillet weld cross section.



Slag - A result of the action of a flux on nonmetallic constituents of a proceed ore, or on the oxidized metallic constituents that are undesirable. Usually consist of combinations of acid oxides and basic oxides with neutral oxides added to aid fusibility.

Slenderness Ratio - The ratio of the length of a uniform column to the least radius of gyration of the cross section.

Slot Weld - A weld made in an elongated hole (slot) in one member of a lap Joint, joining that member to that portion of the surface of the other member which is exposed through the hole. The hole may or may not be filled completely with weld metal.

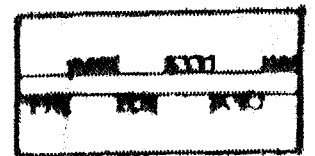


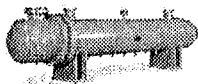
Specific Gravity - The ratio of the density of a material to the density of some standard material, such as water at a specified temperature, for example, 4°C or 60°F. or (for gases) air at standard conditions of pressure and temperature.

Spot Welding - Electric resistance welding in which fusion is limited to a small area directly between the electrode tips.

Stability of Vessels - (Elastic Stability) The strength of a vessel to resist buckling or wrinkling due to axial compressive stress. The stability of a vessel is severely affected by out of roundness.

Staggered Intermittent fillet Welds - Two lines of intermittent fillet welding in a tee or lap joint, in which the increments of welding in one line are staggered with respect to those in the other line.





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Static Head - The pressure of liquids that is not moving, against the vessel wall, is due solely to the "Static Head", or height of the liquid. This pressure shall be taken into consideration designing vessels.

Strain - Any forced change in the dimensions of a body. A stretch is a tensile strain; a shortening is a compressive strain; an angular distortion is a shear strain. The word strain is commonly used to connote unit strain.

Stress - Internal force exerted by either of two adjacent parts of a body upon the other across an imagined plane of separation. When the forces are parallel to the plane, the stress is called *shear stress*; when the forces are normal to the plane, the stress is called *normal stress*; when the normal stress is directed toward the part on which it acts it is called *compressive stress*; when it is directed away from the part on which it acts it is called *tensile stress*.



Stresses In Pressure Vessels - Longitudinal (meridional) S_1 stress, stress Circumferential (hoop) S_2 stress S_2 and S_2 called membrane (diaphragm) stress for vessels having a figure of revolution
Bending stress
Shear stress
Discontinuity Stress at an abrupt change in thickness or shape of the vessel.

Stud - A threaded fastener without a head, with threads on one end or both ends, or threaded full length. (Code UA-60)

Submerged Arc Welding - An arc welding process wherein coalescence is produced by heating with an arc or arcs between a bare metal electrode or electrodes and the work. The welding is shielded by a blanket of granular, fusible material on the work. Pressure is not used and filler metal is obtained from the electrode and sometimes from a supplementary

Tack Weld - A weld made to hold parts of a weldment in proper alignment until the final welds are made.

Tee Joint - A welded joint at the junction of two parts located approximately at right angles to each other in the form of a T.

Tensile Strength - The maximum stress a material subjected to a stretching load can withstand without tearing.

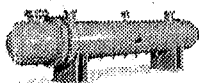
Tensile Stress - Stress developed by a material bearing tensile load.

Test Trial to prove that the vessel is suitable for the design pressure. See Hydrostatic test, Pneumatic test.

Test Pressure - The requirements for determining the test pressure based on calculations are outlined in UG-99(c) for the hydrostatic test and in UG-100(b) for the pneumatic test. The basis for calculated test pressure in either of these paragraphs is the highest permissible internal pressure as determined by the design formulas, for each element of the vessel using nominal thicknesses with corrosion allowances included and using the allowable stress values for the temperature of the test. (Code UA-60)

Thermal Fatigue - The development of cyclic thermal gradients producing high cyclic thermal stress and subsequent local cracking of material.

Thermal Stress - A self-balancing stress produced by a non-uniform distribution of temperature or, by differing thermal coefficients of expansion. Thermal stress is developed in a solid body whenever a volume of material is prevented from assuming the size and shape that it normally should under a change in temperature.



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Thickness of Vessel Wall

1. The "required thickness" is that computed by the formulas in this Division, before corrosion allowance is added (see UG-22).
2. The "design thickness" is the sum of the required thickness and the corrosion allowance (see UG-25).
3. The "nominal thickness" is the thickness selected as commercially available, and as supplied to the manufacturer; it may exceed the design thickness. (Code UA-60)

Throat - See under Fillet Weld.

Tolerances - For plates the maximum permissible under tolerance is the smaller value of 0.01 in. or 6% of the design thickness. (Code UG-16).

The manufacturing under tolerance on wall thickness of heads: pipes and pipefitting shall be taken into account and the next heavier commercial wall thickness may then be used.

U.M. Plate - Universal Mill Plate or plate rolled to width by vertical rolls as well as to thickness by horizontal rolls.

Ultrasonic Examination (UT) - a nondestructive means for locating and identifying internal discontinuities by detecting the reflections they produce of a beam of ultrasonic vibrations (Code UA-60)

Undercut - A groove melted into the base metal adjacent to the toe of a weld and left unfilled by weld metal.

Unit Strain - Unit tensile strain is the elongation per unit length; unit compressive strain is the shortening per unit length; unit shear strain is the change in angle (radians) between two lines originally at right angles to each other.

Unit Stress - The amount of stress per unit of area.

Vessel - A container or structural envelope in which materials are processed, treated, or stored; for example, pressure vessels, reactor vessels, agitator vessels, and storage vessels (tanks).

Weaving - A technique of depositing weld metal in which the electrode is oscillated from side to side.

Weld - A localized coalescence of metal produced by fusion with or without use of filler metal, and with or without application of pressure.

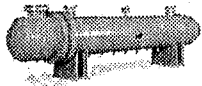
Weld Metal - The metal resulting from the fusion of the base metal and the filler metal.

Welding - The metal joining process used in making welds.
In the construction of vessels the welding processes are restricted by the Code (UW-27) as follows:

1. Shielded metal arc, submerged arc, gas metal arc, gas tungsten arc, plasma arc, atomic hydrogen metal arc, oxyfuel gas welding, electroslog, and electron beam.
2. Pressure welding processes: flash, induction, resistance, pressure thermit, and pressure gas.

Welding Procedure - The materials, detailed methods and practices involved in the production of a welded joint.

Welding Rod - Filler metal. in wire or rod form, used in the gas welding process, and in those arc welding processes wherein the electrode does not furnish the deposited metal.

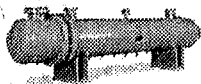


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Wrought Iron - Iron refined to a plastic state in a puddling furnace. It is characterized by the presence of about 3 per cent of slag irregularly mixed with pure iron and about 0.5 per cent carbon.

Yield Point - The lowest stress at which strain increases without increase in stress. For some purposes it is important to distinguish between the *upper* yield point, which is the stress at which the stress-strain diagram first becomes horizontal, and the *lower* yield point, which is the somewhat lower and almost constant stress under which the metal continues to deform. Only a few materials exhibit a true yield point; for some materials the term is sometimes used as synonymous with yield strength.



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ABBREVIATIONS

COMPILED from 1. ASA Z32.13-1950 ABRIVIATIONS FOR USE ON DRAWINGS
2.ASA Z10.1-1941 ABRIVIATIONS FOR SCIENTIFIC & ENGINEERING TERMS

ADDED: ABBREVIATIONS GENERALLY USED ON VESSEL & PIPNG DRAWINGS

AB	Anchor Bolt	DET	Detail
AISC	American Institute of Steel Construction	DIA	Diameter
ALLOW	Allowance - Allowable	DIAM	Diameter
ASME	American National Standards Institute	DIM	Dimension
ASA	American Standard Association	DP	Design Pressure
API	American Petroleum Institute	EA	Each
APPROX	Approximately	EH	Extra Heavy
ASB	Asbestos	EL	Elevation
ASME	American Society of Mechanical Engineers	ELEV	Elevation
ASTM	American Society for Testing Mat'ls.	ELL	Elbow
AVG	Average	ELLIP	Ellipse, Elliptical, Ellipsoid
ALLOW	Corrosion Allowance	EQ	Equal, Equally
bbl	Barrel	ETC	Et Cetera
BC	Bolt Circle	EXT	External
BEV	Bevel	F	Fahrenheit
BLD	Blind	F-F	Face to Face
BOP	Bottom of Pipe	F&G	Flanged & Dished
BOT	Bottom	FF	Flat Face
BRKT	Bracket	FIG	Figure
btu	British Thermal Unit	FIN	Finish
BW	Bevel Weld	FLG	Flange
BWG	Birmingham Wire Gauge	FS	Far Side, Forged Steel
CCW	Counter Clockwise	ft	Foot, Feet
cfm	Cubic Foot per Minute	ft ³	Cubic Foot
CFW	Continuous Fillet Weld	FW	Fillet Weld
CG	Commercial Grade	gpd	Gallon per Day
CG	Center of Gravity	gpm	Gallon per Minute
Cm	Centimeter	g	Gram
C _L	Centerline	GA	Gage
C _L to C _L	Centerline to Centerline	GALV	Galvanized
CO	Company	gal	Gallon
CONC	Concentric	GG	Gage Glass
CPLG	Coupling	GOL	Gage of Outstanding
C	Degree Centigrade	HLA	High Level Alarm
CA	Corrosion Allowance	HLSD	High Level Shut Down
DT'L	Detail	HR	Hot Rolled
DWG	Drawing	HT	Heat Treatment
DC	Down comer	HLL	High Liquid Level
DEH	Double Extra Heavy	ID	Inside Diameter
in	Inches	P	Pressure
INCL	Including, Included	PBE	Plain Both Ends
INS	Inspection	PC	Pressure Control
INT	Internal	PCS	Pieces
JE	Joint Efficiency	PCV	Pressure Control Valve
kg	Kilogram	PI	Pressure Indicator
l	Liter	PL	Plate
lb	Pound	PROJ	Projection
lbf	Pound Force	PSE	Plain Small End
lbs	Pounds	Psi	Pound per Square Inch
LC	Level Control	RAD	Radial
LCV	Liquid Control Valve	REF	Reference