1 INTRODUCTION TO HEAT EXCHANGERS

Heat exchangers are used in many different services in a typical chemical plant. The following lists some of the services along with the common terms used to describe the heat exchangers services.

**Chiller**
An exchanger which uses a refrigerant to cool a fluid to a temperature below that which is achievable with water.

**Condenser**
Condenses a vapour or mixture of vapours either alone or in the presence of non-condensable gases.

**Cooler**
Cools liquids or gases normally using water.

**Exchanger**
Cools one fluid while heating the other fluid.

**Heater**
Imparts sensible heat to a liquid or gas by condensing steam or a heat transfer fluid.

**Reboiler**
Reboilers generate vapour to drive fractional distillation separation. The heating can be achieved via condensing a heat transfer fluid or sensible heat from a fluid.

**Thermosyphon Reboiler**
Natural circulation of a boiling liquid is maintained by providing sufficient head to provide circulation

**Forced Circulation Reboiler**
A pump is used to circulate fluid through the reboiler

**Superheater**
Heats a vapour to a temperature above it's boiling point

**Vaporizer**
A heat exchanger which vaporizes part or all of a liquid stream.
The preliminary Process Flow Diagram project indicates that many heat exchangers are required.

The question is: what exchanger size, type and orientation is best for each service?

The following sections describe the various types of exchangers and give guidelines for selection and design.

2 HEAT EXCHANGERS TYPES

There are several different styles of heat exchanger equipment in common use. These include:

- Double pipe heat exchangers
- Hairpin heat exchangers (multitube double pipe heat exchangers)
- Shell and tube heat exchangers
- Plate fin exchangers
- Plate & frame heat exchangers
- Spiral tube heat exchangers
- Spiral plate heat exchangers
- Air-cooled heat exchangers

Other types of heat transfer equipment not discussed here are:

- Tank jackets and coils
- Cooling Towers
- Fired heaters & Boilers

By far the most common is the shell and tube design. However, other styles are often suitable or even preferable in specific applications.

The following describes each of these styles and indicates when they are suitable or not recommended.
2.1 DOUBLE PIPE HEAT EXCHANGERS

Double pipe heat exchangers are the simplest of all types. They are fabricated from two pieces of pipe – one inside the other. One fluid flows through the inner pipe while the second fluid flows through the annulus between the pipes.

Flow inside double pipe heat exchangers can be co-current or countercurrent.

![Countercurrent Flow](image1.png) ![Co-current Flow](image2.png)

**Advantages**

- Inexpensive
- True countercurrent or co-current flow
- Easily designed for high pressure service

**Disadvantages**

- Difficult to clean on shell side.
- Only suitable for small sizes. They are generally not economical if UA > 50,000 Btu/hr.°F.
- Thermal expansion can be an issue.

**Typical Applications**

1. Single phase heating and cooling when the required heat transfer area is small.

2. Can be used for heating using condensing steam if fabricated with elbows to allow expansion.
2.2 HAIRPIN HEAT EXCHANGERS

The hairpin heat exchanger design is similar to that of double pipe heat exchangers with multiple tubes inside one shell. The design provides the flexibility of a U-tube design with an extended shell length that improves the exchanger’s ability to achieve close temperature approaches.

Advantages

- Good countercurrent or co-current flow – good temperature approach.
- Can be designed with removable shell to allow cleaning & inspection.
- Use of finned tubes results in compact design for shellside fluids with low heat transfer coefficients.
- Easily designed for high pressure service.
- Able to handle large temperature difference between the shell and tube sides without using expansion joints.
- All connections are at one end of the exchanger.

Disadvantages

- Designs are proprietary – limited number of manufacturers.
- Relatively expensive.
- Limited size – Not economical if UA > 150,000 Btu/hr-°F.

Typical Applications

Single phase heating and cooling when the required heat transfer area is relatively small. Often found in high pressure services and where there is a large temperature difference between the shell and tubeside fluids.
2.3 PLATE & FRAME HEAT EXCHANGERS

A plate and frame heat exchanger is a compact heat exchanger where thin corrugated plates are stacked in contact with each other, and the two fluids flow separately along adjacent channels in the corrugation. The closure of the stacked plates may be by clamped gaskets, brazed (usually copper brazed stainless steel), or welded (stainless steel, copper, titanium), the most common type being the first, for ease of inspection and cleaning.

Advantages

- Very compact design
- High heat transfer coefficients (2 – 4 times shell & tube designs)
- Expandable by adding plates
- Ease of maintenance
- Plates manufactured in many alloys
- All connections are at one end of the exchanger
Good temperature approaches
Fluid residence time is very short
No dead spots
Leakage (if it should occur) is generally to the outside – not between the fluids
Low fouling due to high turbulence

Disadvantages

- Designs are proprietary – limited number of manufacturers
- Gaskets limit operating pressures and temperatures & require good maintenance
- Typical maximum design pressures are 150-250 psig.
- Gasket compatible with fluids are not always available
- Poor ability to handle solids – due to close internal clearances
- High pressure drop
- Not suitable for hazardous materials
- Not suitable in vacuum service.

Typical Applications

Low pressure and temperature single phase heating and cooling when fluids are not hazardous, a high pressure drop can be tolerated and alloys are required for the fluids being handled.

Courtesy: Alfa Laval Inc.
2.4 SPIRAL PLATE HEAT EXCHANGERS

Spiral plate heat exchangers are fabricated from two metal plates that are wound around each other. One process fluid stream enters the exchanger at the centre and flows outwards while the second fluid enters on the outside and flows inward. This creates almost a true countercurrent flow. See attached information from Alfa Laval for more information.

Advantages

- Single flow paths reduce fouling rates associated with fluids containing solids.
- Ability to handle two highly fouling fluids
- No dead spots for solids to collect inside exchanger
- Countercurrent flow
- Manufactured in many alloys
- Very low pressure drop

Disadvantages

- Designs are proprietary – limited number of manufacturers
- Generally more expensive than shell & tube designs

Typical Applications

1. Liquid/liquid heating, cooling or heat recovery, where one or both of the fluids may cause fouling.
2. Vapour/liquid condensing, particularly at very low pressure and/or high-volume flow.
2.5 SPIRAL TUBE & HELIFLOW HEAT EXCHANGERS

Spiral tube type heat exchangers are fabricated from coiled tubing. In some cases the tubing is installed inside a fabricated bundle to provide a compact stand alone heat exchanger.

These exchangers are used primarily for small services such as pump seal fluid and sample coolers.

See attached article "Graham Spiral Flow Heat Exchangers.pdf" for a more detailed description.

Advantages

- Compact very inexpensive exchanger for small applications
- Can handle high pressures

Disadvantages

- Designs are proprietary – limited number of manufacturers
2.6 AIR COOLED HEAT EXCHANGERS

Air cooled heat exchangers use ambient air for cooling and condensing. They are typically used in locations where there is a shortage of cooling water.

Air-cooled heat exchangers are usually used when the heat exchanger outlet temperature is at least 20 °F above the maximum expected ambient air temperature. They can be designed for closer approach temperatures, but often become expensive compared to a combination of a cooling tower and a water-cooled exchanger.

Air cooled heat exchangers use electrically driven fans to move air across a bank of tubes. There are two basic arrangements:

- Induced draft: Fans draw air through the tube banks.
- Forced draft: Fans blow air through the tube banks.

Air cooled exchangers are expensive compared to water cooled exchangers due to their large size, low heat transfer coefficients on the air size, and structural and electrical requirements. In addition, air cooler exchangers require large plot areas and must be designed to handle diurnal and seasonal changes in air temperature.
The very low heat transfer coefficient associated with air on the outside of the tubes is partially overcome through extensive use of finned tubes to increase the outside surface area.

Changes in ambient air temperatures are often handled by using variable speed or pitch fans to adjust the air flow. In cold climates, it may be necessary to design in the ability to recirculate air to prevent freezing in the process.

Smaller units (similar to radiators) are available and commonly used for small duty applications.

**Advantages**

- Do not use water for cooling

**Disadvantages**

- Requires large plot area
- Expensive
- Fins can plug in "dirty" environments
- Fans can be noisy

**Typical Applications**

Cooling and condensing where cooling water is unavailable or is uneconomical to use.
Shell and tube heat exchangers are known as the work-horse of the chemical process industry and represent the most widely used vehicle for transfer of heat in industrial applications. In essence, a shell and tube exchanger is a pressure vessel with many tubes inside of it. One process fluids flows through the tubes of the exchanger while the other fluid flows outside of the tubes within the shell. The tube side and shell side fluids are separated by a tube sheet.

Shell and tube heat exchangers can be configured for liquid-liquid, gas-liquid, condensing, or vaporizing heat transfer.

Shell and tube heat exchangers have the ability to transfer large amounts of heat in relatively low cost, serviceable designs. They can provide large amounts of effective heat transfer surface while minimizing the requirements of floor space, liquid volume and weight. Shell and tube heat
Exchangers are available in a wide range of sizes and configurations. They have been used in industry for over 150 years, so the thermal technologies and manufacturing methods are well defined. Tube surfaces from standard to exotic metals with plain or enhanced surface characteristics are widely available. They often provide the least costly mechanical design for the flows, liquids and temperatures associated with a particular process.

Shell and tube heat exchangers have the following advantages:

- Relatively inexpensive
- Easy to clean
- Available in many sizes
- Compact design
- Available in many different materials
- Can be designed for high pressures without excessive cost
- Design principles well known
- Many different manufacturers

3.1 SHELL & TUBE HEAT EXCHANGER COMPONENTS

The components of a shell and tube heat exchanger are illustrated below.
The principle components are:

- shell
- shell cover
- tubes
- channel
- channel cover
- tubesheet
- baffles, and
- nozzles

Other components include tie-rods and spacers, pass partition plates (channel partitions), impingement plates, longitudinal baffles, sealing strips, and supports.

### 3.2 TUBES

The tubes are the basic component of a shell and tube heat exchanger, providing the heat transfer surface between the fluid flowing through the inside of the tubes and the other fluid flowing across
the outside of the tubes. Tubes are available in a variety of diameters, wall thicknesses, lengths and materials of construction. Common materials of construction include carbon steel, copper and stainless steel. Many other alloys of nickel, titanium or aluminium are available for specific requirements.

The tubes may be either bare or have extended or enhanced surfaces – usually on the outside. Extended surfaces such as finned tubes can increase the heat transfer area on the outside of the tubes by two to four times that of a bare tube and this area ratio helps to offset a lower outside heat transfer coefficient.

![Finned Tubes](image)

Tube sizes refer to the outside diameter. Standard exchanger tube diameters are 3/4 inch (19.05 mm) or 1 inch (25.4 mm). Larger (1-1/4, 1-1/2 or 2 inch) are sometimes used where a low pressure drop is required; e.g. vacuum condensers. 1/2 or 5/8 inch tubes are sometimes used with off-the-shell vendor’s standard design units for clean services. It is difficult to clean tubes smaller than ¾ inch and therefore these tubes are not normally used in process service.

Tube wall thicknesses are normally given in terms of Birmingham Wire Gauge (BWG) as follows:

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<th>BW Gauge</th>
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<th>Thickness, mm</th>
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Carbon steel tubes will normally be 12 or 14 gauge which includes an allowance for corrosion. Alloy tubes will normally be 18 gauge. 16 gauge tubes may be used if the process fluid contains solids that may cause erosion, if required for high pressure or if the tubes are to be regularly cleaned mechanically.

A listing of common heat exchanger tube dimensions is included on the next page.

Tubes may be seamless or welded. Typically there is no difference in performance between seamless and welded tubes and the decision on type of tube is made based on availability and cost.

Tubes lengths are normally specified in 2 foot increments up to 24 feet long.
### Heat Exchanger Tube Dimensional Data

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Heat Exchanger Tube Dimensional Data

Courtesy of Kern

### 3.3 TUBE LAYOUT
Tubes are installed in the heat exchanger in a specific pattern, the most common being triangular although a square pattern is sometimes used. In addition, the tubes are spaced at equal intervals—the distance from tube centre to tube centre is called the tube pitch. TEMA requires that the ratio of tube pitch to the outside diameter be 1.25 or greater. In practice, the minimum pitch is usually used to keep the shell diameter as small as possible. However, larger pitches are sometimes used to reduce shell side pressure drop or to facilitate cleaning of the outer tube surfaces.

Triangular layouts are either 30° or 60° as shown below. Square layouts are either 90° or 45°. Triangular layouts give a higher heat transfer coefficient and pressure drop than square layouts, particularly for sensible heating and cooling of single-phase fluids and for condensing. The difference is not great for vaporization and as a result square layouts are often used for shell side vaporizers—particularly when the process is such that the shell side of the exchanger requires frequent cleaning.

Tube Layouts

Tube sheet layout charts providing tube counts for various shell diameters are provided on the following pages. Note that these charts assume that the shell is completely filled. This may not always be the case, especially if impingement protection as described later is required.
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<p>| Tube Layouts for Square Pitch | Courtesy of Kern |</p>
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<th>Tube Layouts for Triangular Pitch</th>
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<td><strong>3/8 in. OD tubes on 1-in. triangular pitch</strong></td>
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<th>1 3/4 in. OD tubes on 1 3/4-in. triangular pitch</th>
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3.3 TUBESHEETS

Tube Layouts for Triangular Pitch
Courtesy of Kern
Tubesheets are the plates that support the ends of the tubes. The tube to tubesheet joint must be mechanically strong enough to resist the forces that would tend to separate the tube from the tubesheet during operation and it must be leak tight. Typically tubes are “rolled” or mechanically expanded into grooves that have been cut inside the tube holes. Often, tubes are also “seal welded” at the face of the tubesheet to prevent leakage. Sometimes, a deeper penetration “strength weld” is specified to provide additional mechanical integrity.

The tubesheet is normally a single round plate drilled in the appropriate pattern to accept the tubes, tie-rods, spacers and gaskets.

The tube sheet, in addition to its mechanical requirements must (like the tubes) be capable of withstanding corrosive attack by both fluids in the heat exchanger. Sometimes, to save costs, tube sheets are fabricated with carbon steel and then faced with a more expensive material.

### 3.4 TEMA DESIGNATIONS

Because of the number of variations in mechanical designs for the front and rear heads and shells, TEMA (Tubular Exchanger Manufacturers Association, [www.tema.org](http://www.tema.org)) has designated a system of notations that correspond to each major type of front heat, shell style and rear head. The TEMA standard notation system is shown on the next page.
<table>
<thead>
<tr>
<th>Stationary Head Types</th>
<th>Shell Types</th>
<th>Rear Head Types</th>
</tr>
</thead>
<tbody>
<tr>
<td>A Removable Channel and Cover</td>
<td>E One-Pass Shell</td>
<td>L Fixed Tube Sheet Like &quot;A&quot; Stationary Head</td>
</tr>
<tr>
<td>B Bonnet (Integral Cover)</td>
<td>F Two-Pass Shell with Longitudinal Baffle</td>
<td>M Fixed Tube Sheet Like &quot;B&quot; Stationary Head</td>
</tr>
<tr>
<td>C Integral With Tubesheet Removable Cover</td>
<td>G Split Flow</td>
<td>N Fixed Tube Sheet Like &quot;C&quot; Stationary Head</td>
</tr>
<tr>
<td>D Special High-Pressure Closures</td>
<td>H Double Split Flow</td>
<td>P Outside Packed Floating Head</td>
</tr>
<tr>
<td></td>
<td>J Divided Flow</td>
<td>S Floating Head with Backing Device</td>
</tr>
<tr>
<td></td>
<td>K Kettle-Type Reboiler</td>
<td>T Pull-Through Floating Head</td>
</tr>
<tr>
<td></td>
<td>X Cross Flow</td>
<td>U U-Tube Bundle</td>
</tr>
<tr>
<td></td>
<td></td>
<td>W Extremally Sealed Floating Tubesheet</td>
</tr>
</tbody>
</table>
3.5 SHELL

The shell is simply the container for the shell side fluid. The shell is normally a cylinder fabricated from a pipe (for smaller diameter shells) or by rolling a plate (for larger diameter shells).

The shell side of a shell and tube exchanger contains the most metal and therefore, for reasons of economy, the fluid that requires the most expensive metallurgy is placed on the tube side whenever possible.

Shells are specified by their inside diameter. However, standard pipe sizes are used for shells where practical. For example a standard 23 inch shell may be a section of 24-inch pipe with a wall thickness of 3/8-inch (schedule 20) with an actual inside diameter of 23.25 inches.

Since a large part of the cost of fabricating an exchanger is related to rolling the shell, it is desirable to keep the shell diameter as small as possible. Long “skinny” exchangers are usually less expensive than short “fat” exchangers of the same area. However, the layout or placement of the exchanger amongst other process equipment must be considered in this decision. For example if the exchanger has a removable bundle, the space that must be reserved for the exchanger is over twice the length.

The single pass TEMA “E” shell is by far the most common type used. A shell expansion joint may be required for fixed tubesheet exchangers (TEMA type BEM or AEL exchangers). The decision as to whether an expansion joint is required is part of the mechanical design of the exchanger. However, the process engineer must take into consideration that the type of exchanger selected may result in the need for an expansion joint which will increase the cost of the exchanger.
The TEMA “F” shell in theory offers the possibility of approaching countercurrent flow in a two-pass exchanger. However, in practice, there may be substantial thermal leakage across the longitudinal baffle in an F shell exchanger and it is seldom that the extra costs associated with an F shell can be justified by superior performance.

There are a number of shell styles that are used for special cases. The “J” shell is sometimes used for shell side condensation where a low pressure drop is required.

The “X” shell, or cross flow exchanger, gives even lower pressure drops and is often used for vacuum condensation.

The “K” shell is used for kettle reboilers; the oversized shell allows space for disengagement of vapour from liquid.
### 3.6 BAFFLES

Baffles serve two functions.

The first function is to support the tubes in the proper position during assembly and operation of the heat exchanger. This support is necessary to prevent flow induced vibration in the tubes which can quickly lead to tube failure.

The second function is to guide the shell side flow back and forth across the tubes (as illustrated below), increasing the velocity and therefore the heat transfer coefficient.

---

The most common baffle shape is single segmental. The percent of the inside diameter that is removed is referred to as the baffle cut and must be less than 50% to ensure that adjacent baffles overlap at least one full tube row (otherwise the centre tubes would be unsupported along their full length). When a liquid is flowing on the shell-side of a heat exchanger, baffle cuts of 20 to 25% of the diameter are common. Baffle cuts of 40-45% are common for condensation of steam and other pure component vapours as well as for sensible heat transfer in low pressure gases where low pressure drops are required.
For many high velocity gas flows, use of a single segmental baffle configuration results in high shell-side pressure drops. One way to reduce the pressure drop is to use double segmental baffles. Note that use of double segmental baffles will reduce velocities in the shell-side of the exchanger which will in turn reduce the heat transfer coefficient.

Baffle orientation refers to the positioning of the baffle relative to the shell inlet nozzle. “Horizontal segmental” means that the baffle edges are perpendicular to the nozzle centre line. This is the normal arrangement.

“Vertical segmental” baffles (baffle edges parallel to the inlet nozzle centreline) are used for horizontal shell side condensers, where the condensate will collect in the bottom of the shell. Note that with this arrangement, notches are normally cut in the baffles at the lowest points to facilitate liquid drainage from the exchanger.

The baffle spacing is the distance between adjacent baffles. In general, the baffle spacing is chosen to make the free flow areas through the “window” (the area between the baffle edge and the shell) and the area across the tube bank roughly equal. This normally gives a baffle spacing equal to 40-60% of the shell diameter. Closer spacings give a higher pressure drop and promote bypassing, while wider spacings allow “dead zones” and recirculation downstream of the baffles. Baffle spacings at the ends of the bundle are determined by the nozzle sizes.
3.7 SHELL SIDE NOZZLES AND IMPINGEMENT PROTECTION

From a thermal designer’s point of view the main requirement of the nozzles is that they be large enough to avoid significant pressure drop. When rating an exchanger, one should always look at the nozzle pressure drops as a percentage of the total. Where the total pressure drop is significant, the nozzles should contribute at most 15-20% of the total. Nozzle sizing is particularly important in vacuum services.

The fluid entering the heat exchanger on the shell side can cause the tubes to vibrate. The vibrations can cause tube failure due to fatigue and/or wear where the tubes strike each other or contact the baffles. For this reason an impingement plate or other means of protection is almost always required under the shell side inlet nozzle when the shell side fluid is a gas or a condensing vapour. Impingement protection is also required for liquids unless the velocity is low enough so that $\rho V^2 < 3500$ lb/ft-$^2$ (\(\rho\) is density in lbs/ft$^3$, \(V\) is the velocity in ft/sec). For small heat exchangers it is often worthwhile to increase the nozzle diameter to avoid the need for impingement protection.

Impingement protection often takes the form of a plate located below the inlet nozzle. An alternative is for three or more rows of solid rods the same diameter as the tubes to be installed under the inlet nozzle. This is preferable to an impingement plate in larger exchangers as it permits the shell side fluid to assume its normal velocity distribution without the disruption of flow around the plate. In addition the pressure drop in the inlet region is lower and the possibility of localized high velocities impacting the tubes around the edge of the plate is eliminated.

Another design feature used to prevent damage due to flow induced vibration is the "no tubes in the window" design. With this arrangement every tube is supported by every baffle and there are no tubes installed in the baffle cut area.

It is important to note that provision for an impingement plate requires removal of tubes under the nozzle which will result in a larger diameter shell being required for the same heat transfer area.
### 3.8 TUBE SIDE CHANNELS

The tube-side channels (commonly referred to as the heads) control the flow of the tube-side fluid into and out of the tubes. Since the tube side fluid is generally the most corrosive, these components are often constructed of or lined with alloys.

Bonnets are the least costly heads (TEMA type BEM) and are normally used on small exchangers. Bonnets have the disadvantage of making it necessary to disconnect piping and completely remove the head to allow access to the tube sheet for cleaning or maintenance. Channels with bolted and gasketed covers (TEMA type AEL) permit access to the tubes by simply removing the channel covers; thereby eliminating the need to disturb the piping. This is an advantage for larger exchangers with heavy piping.

---

**Pass Divider**

**Bonnet Type Head**

**Channel with Gasketed Cover**

Pass dividers are installed in the tube-side channels to allow multiple tube-side passes. The pass dividers are arranged to direct the tube flows in such a way as to create equal velocities (i.e. an equal number of tubes) in each pass. The pass dividers are fabricated from the same materials as the channels.

Illustrated below are TEMA head designs which allow thermal expansion of the bundle within the shell. Use of these head types may avoid using an expansion joint on the shell. Note that the TEMA T arrangement is basically a U-tube bundle with an internal head.

**TEMA W**
- Externally Sealed
- Floating Tubesheet

**TEMA T**
- Pull Through
- Floating Head

**TEMA S**
- Floating Head with Backing Device
4 SHELL & TUBE HEAT EXCHANGER ARRANGEMENTS

There are three basic types of shell and tube exchangers; floating head, fixed tubesheet and U-tube.

4.1 FLOATING HEAD HEAT EXCHANGERS

The floating head unit has a tube bundle that can be removed for cleaning and replacing while the shell remains in place.

4.2 FIXED TUBESHEET HEAT EXCHANGERS

The fixed tubesheet exchangers do not have removable bundles as both tube sheets are welded to the shell. Note that the shellside of the exchanger is not accessible for mechanical cleaning.
The fixed tubesheet design cost about 20% less than a floating head design and has less operating problems. However, the fixed tubesheet design is difficult to clean on the shell-side and may require (expensive) expansion joints be installed in the shell if there is a large temperature difference between the shell and tubes.

4.3 U-TUBE HEAT EXCHANGERS

U-Tube

The U-tube bundle can be configured both in a fixed tubesheet (TEMA N) or as a removable bundle (TEMA C) as illustrated above. The use of a U-tube bundle eliminates one of the tubesheets making this arrangement about 10% less expensive than the fixed tubesheet design.

4.4 HEAT EXCHANGER ORIENTATIONS

Shell and tube heat exchangers can be installed horizontally, vertically and occasionally on an incline. The orientation depends on the following:

a) All things being equal it is desirable to orient shell and tube heat exchangers horizontally. This allows easy access for removal of the heads and tube bundles for inspection and cleaning.

b) With sensible heat transfer (i.e. no phase change) the orientation has no affect on the heat transfer and the heat exchanger can be installed in any orientation.

c) Condensing heat transfer coefficients are affected by the flow patterns of the condensed liquid on the tube walls. This is in turn affected by the orientation of the tubes. In general the best heat transfer coefficients for condensing pure components are achieved by condensing on the outside of the tubes with the condenser in a horizontal position. In cases where subcooling of the condensed liquids is desired or when condensing vapour...
from steams containing inerts the best performance is normally achieved by condensing inside the tubes with downward flow and the exchanger in the vertical position.

d) Heat transfer inside vaporizers and reboilers is normally limited by maximum heat fluxes. The orientation of the exchanger has little affect on the heat transfer.

e) Thermosyphon reboilers can be designed to be installed either horizontally or vertically.

f) Kettle type reboilers are always horizontal.
# Selection of Heat Exchanger Type & Orientation

## 5.1 Selection Chart

The following chart gives guidelines as to the type and configuration of heat exchanger that is suitable for a particular application.

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<thead>
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<th>Condensing Duty</th>
<th>Evaporating Duty</th>
<th>UA &gt; 50 k Btu/h°F</th>
<th>UA &gt; 100 k Btu/h°F</th>
<th>High Pressure Service - 1 in. S.I. - 75 gpm</th>
<th>High Pressure Service - 1 in. S.I. - 150 gpm</th>
<th>Flow Rates</th>
<th>Disk Fluid on 1 side</th>
<th>Disk Fluid on Both Sides</th>
<th>Viscous Fluids</th>
<th>Shell &amp; Tube: Can be Accepted for Gaskets</th>
<th>High/Low Bypass Fluids</th>
<th>Close Temperature Approach Required</th>
<th>Low Pressure Drop Required - 1 in. S.I.</th>
<th>Corrosion Design Required</th>
<th>Heat Transfer Across, As Is Specified</th>
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<td></td>
</tr>
</tbody>
</table>

**Notes:**
1. Place dirty fluid on tube side
2. Specify with removable tube bundle
3. Requires use of multiple exchangers in parallel
4. Probably an uneconomical choice
5. Place High Pressure fluid on tube side
6. Place corrosive fluid on tubeside
7. Expansion Joint may be required
8. Special design considerations may be required
9. Coil only
10. Room may be required to pull bundle
11. Only available with alloy tubes
12. Rarely used in this service
13. See separate selection chart for condensers
14. See separate selection chart for refrigerators
15. Horizontal orientation is usually preferable for maintenance reasons
16. Place viscous fluid on shellside
17. Some floating head types have internal gaskets which can leak

**Legend:**
- Green: A good choice
- Orange: Could be used in some situations
- Red: A poor or unacceptable choice
5.2 HEAT EXCHANGER TYPE

- Shell and tube exchangers can be designed for almost any service. All other exchanger types are compared to shell and tube.
- Hairpin type exchangers are a good choice when one fluid is at a very high pressure.
- Hairpin exchangers are a good choice when there is a large difference in temperature between the two fluids.
- Plate and frame exchangers are a good choice for liquid-liquid exchangers when the operating pressure is low and a low pressure drop across the exchanger is not required.
- Plate and frame exchangers should not be used for hazardous or flammable fluids.
- Plate and frame exchangers should not be used in vacuum service.
- Spiral Plate exchangers are a good choice for two dirty fluids.
- Spiral Plate exchangers are a good choice for vacuum service.

5.3 FLUID PLACEMENT

- Whenever possible place corrosive fluids on the tubeside to reduce the exchanger cost.
- Place the highest pressure fluid on the tubeside to reduce exchanger cost.
- Place dirty fluids on the tubeside as it is much easier to clean the tubeside than the shellside.
- Place viscous fluids on the shellside. Pressure drops are easier to minimize on the shellside.

5.4 SELECTION OF REBOILER TYPE & ORIENTATION

The various types of reboiler configurations are illustrated on the following pages.
Stab-In Reboilers are the least expensive type. They are limited in size by the physical space available inside the column and are not suitable for viscosities above 1 cP.

Kettle Reboilers used when a high turndown or a high quality vapour is required. They are also used when large heat transfer surfaces are needed. Kettle reboilers are expensive due to the shell design but are able to handle large differences in temperature between the fluids due to their U-tube design.

Vertical Recirculating Thermosyphon Reboilers are applicable when process rates are fairly constant, the viscosity of the fluid is low and the column height can be increased to accommodate the head requirements. Their operation requires a fixed static head. Vertical Thermosyphon reboilers are generally the least costly heat exchanger type (excluding stab ins) due to their high heat transfer rates and low fouling tendencies.
Horizontal Recirculating Thermosyphon Reboilers are applicable when process rates are fairly constant, the viscosity of the fluid is low and the column height cannot be increased to accommodate the head requirements. Their operation requires a fixed static head.

Once Through Reboilers are used when the feed to the exchanger cannot be recirculated. The orientation can be horizontal or vertical. This design provides a low residence time on hot surfaces which is important in some applications. However, this design has a very narrow range of flows in which operation is stable and careful consideration is required to produce a successful design.

Forced Circulation Reboilers use a pump to move the fluid through the exchanger and are applicable when handling viscous liquids or a particulate laden liquid or when it is desirable to heat the liquid and then carry out the vaporization downstream of the exchanger. Any arrangement of shell side or tubeside boiling, vertical or horizontal may be used. Suppressed vaporization operation requires a throttling valve in the reboiler outlet line.
Fired Heater Reboilers are used when the required temperatures are higher than what can be achieved with other utilities.

The main considerations in the selection of a reboiler type are the viscosity of the fluid, the turndown required and the physical layout.

The selection chart is provided below provides guidance in the selection of a reboiler type.
Determination of Kettle Reboiler Shell Diameters

In kettle type reboilers the shell diameter is chosen to allow vapour-liquid separation to take place in the vapour space above tube bundle. The vapour space can be determined using vapour-liquid separation calculations. Ludwig provides the following guidelines:

<table>
<thead>
<tr>
<th>Heat Flux Btu/ft²-hr</th>
<th>Ratio Shell diameter over Bundle Diameter</th>
</tr>
</thead>
<tbody>
<tr>
<td>20,000</td>
<td>1.9 – 2.5</td>
</tr>
<tr>
<td>15,000</td>
<td>1.8 – 2.1</td>
</tr>
<tr>
<td>12,000</td>
<td>1.5 – 1.7</td>
</tr>
<tr>
<td>8,000</td>
<td>1.3 – 1.5</td>
</tr>
</tbody>
</table>

5.4 SELECTION OF CONDENSER TYPE & ORIENTATION

The main consideration in the selection of condenser types and orientations is the type of fluid being condensed.

- For condensation of pure components use a horizontal exchanger and place the process on the outside of the tubes.
- If there are significant quantities of noncondensibles present; condense inside the tubes with a vertical orientation and downward flow.
- Spiral Plate exchangers are often used as condensers especially in vacuum service. They can be mounted directly on top of a column.
6 SHELL & TUBE HEAT EXCHANGERS – THERMAL DESIGN

6.1 FILM THEORY OF HEAT TRANSFER

The film theory assumes that the resistance to heat, mass or momentum transfer at a surface is confined to a thin film adjacent to the surface.

Consider the diagram above which illustrates heat transfer across a tube wall.

There are five layers through which the heat must flow:

1. The inside boundary layer
2. The inside fouling layer
3. The tube wall
4. The outside fouling layer
5. The outside boundary layer

The quantity of heat flow is the same through each layer.
Therefore (assuming that the outside area is equal to the inside area):

\[ q = \frac{Q}{A} = h_i(T_H - T_1) = h_{fl}(T_1 - T_2) = h_w(T_2 - T_3) = h_{fo}(T_3 - T_4) = h_o(T_4 - T_C) \]

Where:
- \( q \) = heat flux, Btu/hr/ft\(^2\) or W/m\(^2\)
- \( Q \) = Total Heat Transfer, Btu/hr or W
- \( A \) = Heat Transfer Area, ft\(^2\) or m\(^2\)
- \( h \) = Heat Transfer Coefficient, Btu/hr-ft\(^o\)F or W/m\(^o\)K
- \( T \) = Temperature, \(^o\)F or \(^o\)K

The thermal resistance to heat flow, \( r \) is the reciprocal of the heat transfer coefficient.

i.e. \[ r = \frac{1}{h} \]

Where:
- \( r \) = thermal resistance, hr-ft\(^o\)F/Btu or m\(^o\)K/W

Therefore the above equation can be modified as follows:

\[ q = \frac{Q}{A} = h_i(T_H - T_1) = \frac{(T_1 - T_2)}{r_{fi}} = \frac{(T_2 - T_3)}{r_w} = \frac{(T_3 - T_4)}{r_{fo}} = \frac{(T_4 - T_C)}{h_o} \]

In terms of temperature differences one can write:

\[ T_H - T_1 = \frac{q}{h_i} \]
\[ T_1 - T_2 = q \frac{1}{r_{fi}} \]
\[ T_2 - T_3 = q \frac{1}{r_w} \]
\[ T_3 - T_4 = q \frac{1}{r_{fo}} \]
\[ T_4 - T_C = q \frac{1}{h_o} \]

Substituting one gets,

\[ T_H - T_C = \Delta T = q\left(\frac{1}{h_i} + \frac{1}{r_{fi}} + \frac{1}{r_w} + \frac{1}{r_{fo}} + \frac{1}{h_o}\right) \]

Since \( q = \frac{Q}{A} \), the above equation can be rearranged to give the following:

\[ Q = \frac{A \ (T_H - T_C)}{(1/h_i + r_{fi} + r_w + r_{fo} + 1/h_o)} \]

The term \( 1/(1/h_i + r_{fi} + r_w + r_{fo} + 1/h_o) \) is referred to as the overall Heat Transfer Coefficient, \( U \).

Where:
- \( U \) = Overall heat transfer coefficient, Btu/hr-ft\(^o\)F or W/m\(^o\)K
Previously it had been assumed that the outside area is equal to the inside area. However, this is not the case when tubes are being used. Modifying the above equation to take into account differences in areas gives:

$$U = \frac{1}{1/h_o + A_o/A_i h_i + r_w + r_{fo} + A_o r_i/A_i}$$

Therefore, as developed above the two main heat transfer equations are as follows:

$$Q = U A \Delta T$$

And

$$U = \frac{1}{1/h_o + A_o/A_i h_i + r_w + r_{fo} + A_o r_i/A_i}$$

Normally it is desired to calculate the heat transfer area required using:

$$A = \frac{Q}{U \Delta T}$$

The following sections discuss how each of the terms in the above equations is determined.
6.2 AN OVERVIEW OF HEAT EXCHANGER CALCULATION PROCEDURES

The calculations carried out to design heat exchangers are far from exact. In fact Perry’s refers to the design of heat exchangers as being an art as much as a science.

These days almost all heat exchanger designs are carried out using software specifically written for the purpose. The most accepted software used for heat exchanger design is that produced by Heat Transfer Research Inc. (HTRI). Although this, and other company’s software are invaluable in carrying out the actual calculations they are limited in their ability to give guidance to the designer as to what is the best exchanger type, fluid placement and orientation for the particular design being considered.

Therefore, the designer must have a good understanding of where to start and how to adjust the various parameters to achieve the optimum design.

Even when using design software the heat exchanger design procedure (often called rating) is iterative. The procedure is illustrated on the following flowchart. Basically the procedure is to choose a design and then test the selected design against the requirements. If the design meets the requirements in all respects the design is complete. If the design does not meet some or all of the requirements then the design is adjusted and the procedure repeated.

To achieve a proper design in a reasonable amount of time the designer should proceed as follows:

1. Obtain good basic data. If the duties, process flows and fluid physical properties used in the design are incorrect or constantly changing (due to the basic process design being incomplete) then the design will be wrong. Check the data before starting the design to ensure that it makes sense and is up to date.

2. Specify realistic allowable pressure drops. Often a low specified pressure drop will dictate that the exchanger heat transfer area is much larger than would be the case if more pressure drop were allowed. Take the time to determine what the maximum allowable pressure drop can be.

3. Take time before you start the design to take into account all the requirements for the heat exchanger. Are the steady state conditions from the process simulation the proper basis for the design or are there startup, shutdown or upset conditions that need to be considered?

4. Review the service versus the various heat exchanger types and determine which exchanger type is must suitable for the application. Not all types may be eliminated but at least the field will be narrowed.

5. Review the service versus recommended heat exchanger fluid placement (shell or tube) and exchanger orientation (horizontal or vertical) and select the arrangement that is must suitable for the application. Not all arrangements may be eliminated but at least the field will be narrowed.

6. Use a Heat Exchanger data sheet as illustrated on the next page as a guide as to what information is required as well as to document the information in an organized fashion.

7. Carry out the rating calculations either by hand or by using design software. In both cases take the time to understand the limitations of the methods being used. If the calculations are being carried out by hand understand the limitations of the correlations being used and make...
sure they are applicable to your design. If you are using software, make sure that the software is recognized as being accurate by those for whom you are preparing the design.

8. Stay organized. It is very easy, especially when using design software, to head off in all directions instead of taking a stepwise approach towards determining the optimum design. Understanding and paying attention to the affect of the various variables which can be used to optimize the design (as described later) will save a lot of time.
### HEAT EXCHANGER SPECIFICATION SHEET

<table>
<thead>
<tr>
<th>Company</th>
<th>Plant:</th>
</tr>
</thead>
<tbody>
<tr>
<td>Location</td>
<td></td>
</tr>
<tr>
<td>Service of Unit</td>
<td></td>
</tr>
<tr>
<td>Equipment Number</td>
<td></td>
</tr>
<tr>
<td>Date</td>
<td>Revision</td>
</tr>
<tr>
<td>Size</td>
<td>Type</td>
</tr>
<tr>
<td>Surface Area</td>
<td>$\text{ft}^2$</td>
</tr>
</tbody>
</table>

### PERFORMANCE OF ONE UNIT

<table>
<thead>
<tr>
<th>SHELL SIDE</th>
<th>TUBE SIDE</th>
</tr>
</thead>
<tbody>
<tr>
<td>Fluid Name</td>
<td></td>
</tr>
<tr>
<td>Fluid Quantity, Total</td>
<td>$\text{lbs/hr}$</td>
</tr>
<tr>
<td>Vapour (In/Out)</td>
<td>$\text{lbs/hr}$</td>
</tr>
<tr>
<td>Liquid, In / Out</td>
<td>$\text{lbs/hr}$</td>
</tr>
<tr>
<td>Noncondensables</td>
<td>$\text{lbs/hr}$</td>
</tr>
<tr>
<td>Temperature (In/Out)</td>
<td>$^\circ\text{F}$</td>
</tr>
<tr>
<td>Dew/Bubble Point</td>
<td>$^\circ\text{F}$</td>
</tr>
<tr>
<td>Density</td>
<td>$\text{lbs/ft}^3$</td>
</tr>
<tr>
<td>Viscosity</td>
<td>$\text{cP}$</td>
</tr>
<tr>
<td>Molecular Weight, Vapour</td>
<td></td>
</tr>
<tr>
<td>Molecular Weight, Noncondensables</td>
<td></td>
</tr>
</tbody>
</table>

| Specific Heat | $\text{BTU/(lb} \cdot \text{°F)}$ |
| Thermal Conductivity | $\text{BTU/(ft} \cdot \text{hr} \cdot \text{°F)}$ |
| Latent Heat | $\text{BTU/lb}$ |
| Pressure | $\text{psia}$ |
| Velocity | $\text{ft/sec}$ |
| Pressure Drop, Allowable / Calculated | $\text{psi}$ |
| Fouling Resistance | $\text{ft}^2\cdot\text{hr} \cdot \text{°F} \cdot \text{BTU}$ |
| Heat Exchanged | $\text{BTU/hr}$ |
| MTD, Corrected | $\text{°F}$ |
| Heat Transfer Rate, Service | $\text{BTU/hr} \cdot \text{°F}^2$ |

### CONSTRUCTION

<table>
<thead>
<tr>
<th>SHELL SIDE</th>
<th>TUBE SIDE</th>
</tr>
</thead>
<tbody>
<tr>
<td>Design / Test Pressure</td>
<td>$\text{psig}$</td>
</tr>
<tr>
<td>Design Temperature, Max/Min</td>
<td>$^\circ\text{F}$</td>
</tr>
<tr>
<td>Number of Passes</td>
<td></td>
</tr>
<tr>
<td>Corrosion Allowance</td>
<td></td>
</tr>
<tr>
<td>Connections, In</td>
<td></td>
</tr>
<tr>
<td>Cut</td>
<td></td>
</tr>
<tr>
<td>Number of Tubes</td>
<td>$\text{Tube Length} \times \text{Tube Pitch} \times \text{Pattern}$</td>
</tr>
<tr>
<td>Tube Type</td>
<td>$\text{Tube Gauge}$</td>
</tr>
<tr>
<td>Shell, Material</td>
<td>$\text{Shell OD}$</td>
</tr>
<tr>
<td>Channel / Bonnet</td>
<td></td>
</tr>
<tr>
<td>Tubesheet</td>
<td></td>
</tr>
<tr>
<td>Floating Head Cover</td>
<td></td>
</tr>
<tr>
<td>Baffles, Crossing</td>
<td>$\text{Type} \times % \text{Cut} \times \text{Horizontal Spacing}$</td>
</tr>
<tr>
<td>Baffles, Longitudinal</td>
<td></td>
</tr>
<tr>
<td>Tube Supports</td>
<td></td>
</tr>
<tr>
<td>Tube to Tubesheet Joint</td>
<td></td>
</tr>
<tr>
<td>Expansion Joint</td>
<td></td>
</tr>
<tr>
<td>Gaskets - Tube Side</td>
<td></td>
</tr>
<tr>
<td>Gaskets - Shell Side</td>
<td></td>
</tr>
<tr>
<td>Code Requirements</td>
<td>$\text{TEMA Class}$</td>
</tr>
<tr>
<td>Weight</td>
<td></td>
</tr>
<tr>
<td>Remarks</td>
<td></td>
</tr>
</tbody>
</table>
6.3 BASIC DATA REQUIRED FOR HEAT EXCHANGER CALCULATIONS

The first step in preparing a heat exchanger design is assembly of the required data.

The minimum requirements for basic data are as follows:

**Process Fluid Design Flow Rates**

This is normally the maximum flow rate for the process fluid at full flowsheet rates. However, there may be temporary higher rates during startup or shutdown or during periods of process upset that the exchanger must perform properly with.

**Process Fluid Heat Duties**

For heating or cooling a single phase fluid this may be expressed in terms of the temperature change. For vaporizing or condensing, one must know the quality (weight fraction vapour) and the heat duty versus the equilibrium temperature.

**Process Inlet Conditions**

The inlet temperature, pressure and allowable pressure drop for the process fluids.

As will be seen later, the allowable pressure drop can have a major influence on the design of a heat exchanger. The designer should pay close attention to assigning an allowable pressure drop and ensure that it is realistic.

**Fouling Nature of Process Fluids**

This includes sufficient information so that the designer can make a decision as to how much fouling allowance to provide for in the exchanger design.

**Physical Properties of the Process Fluids**

Physical properties including at a minimum density, viscosity, specific heat and thermal conductivity are required at the inlet and outlet temperatures.

In cases where there are changes of state then dew points and latent heats are also required.

**Materials of Construction**

What materials of construction are suitable for construction of the heat exchanger.
6.4 DETERMINATION OF DUTY, Q

Often, the duty of each of the heat exchange steps will be determined using simulation software. In cases where a computer simulation has not been prepared, the duty can be calculated using the following procedures:

For sensible heat transfer, the heat duty is calculated by:

\[ Q = W \int C_p \Delta T \]

Where:
- \( C_p \) = Heat Capacity, Btu/lb-\(^o\)F
- \( W \) = mass flow, lb/hr
- \( T \) = Temperature, \(^o\)F
- \( Q \) = Duty, Btu/hr

The integration is taken over the range from the inlet temperature (\( T_i \)) to the outlet temperature (\( T_o \)). Usually the specific heat function is nearly linear with temperature and the above equation becomes:

\[ Q = W C_{pavg} \Delta T \]

Where the average specific heat is evaluated at the mean of the inlet and outlet temperatures.

For condensing or boiling, the heat duty is given by

\[ Q = W (H_{Vo} + H_{Lo} - H_i) \]

Where:
- \( H_{Vo} \) = Enthalpy of vapour leaving the heat exchanger, Btu/lb
- \( H_{Lo} \) = Enthalpy of liquid leaving the heat exchanger, Btu/lb
- \( H_i \) = Enthalpy of the entering stream, Btu/lb

In the case of a condenser, the condensing temperature of the heating fluid is normally specified and then the mass flow of the condensing vapour is calculated.

In the case of a condenser, either the flow rate of the coolant is specified and the outlet temperature of the coolant is calculated or vice versa.

Selection of Cooling Water Outlet Temperatures

For heat exchanger sizing purposes, the cooling water inlet temperature is usually taken as the maximum summer temperature.

However, the cooling water outlet temperature can be varied. Theoretically there is an optimum cooling water flow rate (and corresponding cooling water outlet temperature) for each exchanger design. Clearly, there is a maximum temperature which the exiting cooling water cannot exceed. For example, returning water to a cooling water at too high a temperature can cause excessive vaporization of the water which will cause deposition of minerals in the cooling tower packing. Similarly, cooling water returned to lakes or rivers must be at temperatures that are not harmful to the environment. Also, tube wall temperatures above 50 \(^o\)C in contact with many cooling water
streams will result in deposition of calcium deposits on the heat exchangers surfaces. Any temperature under these maximums are theoretically allowable.

From the standpoint of only finding the lowest cost design for the heat exchanger, the water exit temperature which will almost always result in the lowest heat transfer area (and therefore cost) is the lowest water exit temperature possible - as this will provide the highest temperature difference. However, this design will require the highest water pumping rate which will increase the cost of pumps and piping carrying the water to and from the exchanger.

In the big picture, on a strictly economic basis, the best water exit temperature is normally as close as possible to the maximum allowed.
6.5 TYPICAL OVERALL HEAT TRANSFER COEFFICIENTS

Typical overall heat transfer coefficients are given below. These "typical" heat transfer coefficients can be used to estimate the performance of a heat exchanger.

Typical heat transfer coefficients are normally published as a range of an "all-in" numbers which includes the inside and outside heat transfer coefficients, the fouling factors and the tube wall resistances.

It must be noted that the typical heat transfer coefficients correspond to the performance of properly designed heat exchangers in known services.

Typical heat transfer coefficients can be used to estimate the heat transfer area for the purpose of preparing a cost estimate for a conceptual design. However, it must be recognized that the results will be ballpark estimates only – especially if the process fluids do not exactly correspond to those that form the basis for the typical number.

As described later, the performance of heat exchangers very much depends on the flow regime at the tube walls. Use of the typical heat transfer coefficients without later confirming that the heat transfer coefficient is valid can lead to an improperly designed heat exchanger.

In other words a typical heat transfer coefficient should be used as a starting point only.

### TYPICAL OVERALL HEAT TRANSFER COEFFICIENTS

<table>
<thead>
<tr>
<th>Service</th>
<th>( U_0 ), W/m(^2\cdot)°K</th>
</tr>
</thead>
<tbody>
<tr>
<td>Water – Water</td>
<td>1000 - 2000</td>
</tr>
<tr>
<td>Water – Steam</td>
<td></td>
</tr>
<tr>
<td>Water – Oil</td>
<td>500 - 850</td>
</tr>
<tr>
<td>Steam – Oil</td>
<td></td>
</tr>
<tr>
<td>Water – Organic Liquid</td>
<td>500 - 1000</td>
</tr>
<tr>
<td>Steam – Organic Liquid</td>
<td></td>
</tr>
<tr>
<td>Water – Gas</td>
<td>25 - 150</td>
</tr>
<tr>
<td>Steam – Gas</td>
<td></td>
</tr>
<tr>
<td>Water – Condensing Organic</td>
<td>500 - 1000</td>
</tr>
<tr>
<td>Water – Condensing Organic</td>
<td></td>
</tr>
<tr>
<td>With inerts</td>
<td>150 - 500</td>
</tr>
<tr>
<td>Steam – Boiling Organic</td>
<td>750 - 1500</td>
</tr>
</tbody>
</table>
6.6 TEMPERATURE DIFFERENCE, $\Delta T$

In most instances the temperature difference between the hot stream and the cold stream will vary in different portions of the heat exchanger. As a result, an effective average value must be used in the rate equation. The appropriate average depends on the configuration of the exchanger. For simple countercurrent and co-current exchangers the Log Mean Temperature Difference (LMTD) applies.

The LMTD is calculated using: 

$$\text{LMTD} = \frac{\text{GTTD} - \text{LTTD}}{\ln(\text{GTTD}/\text{LTTD})}$$

Where: The GTTD is the Greatest Terminal Temperature Difference and the LTTD is the Least Terminal Temperature Difference where terminal refers to the first or last point of heat exchange in the heat exchanger.

As illustrated in the diagram below, the configuration of most shell and tube heat exchangers does not allow true countercurrent or co-current flow to be achieved.
For exchanger configurations with flow passes arranged to be partially countercurrent and partially co-current a correction factor is applied to the LMTD.

\[ \text{CMTD} = (F) \text{LMTD} \]

Values of the LMTD correction factor, F depend on the exchanger configuration and the stream temperatures. Correction charts have been published for all exchanger configurations. A copy is included on the next page.

In general if the value for F is found to be less than 0.8, it is a signal that the selected exchanger configuration is not suitable, and an arrangement that allows a closer approach to countercurrent flow should be sought.

Heat Exchange With Non-linear Temperature Profiles

The above Corrected Log Mean Temperature Difference (CMTD) is based only on the inlet and outlet temperatures and assumes a linear relation between the duty and stream temperature change. In some cases this is not the case. Examples are multi-component condensing or boiling and cases where the streams undergo large temperature changes which significantly alter the physical properties.

It is the Engineer's responsibility to be aware that non-linear relationships may occur in heat transfer equipment. If simulation software is being used the presence of this situation can easily be checked by printing a T versus Q table or plot for the heat transfer equipment. It should be noted that standard exchanger data sheets only specify the conditions at the inlet and outlet of an exchanger and therefore the heat exchanger designer may not be aware of the non-linear situation. This can lead to an improperly sized heat exchanger.
LMTD correction factors for heat exchangers. In all charts, \( R = (T_1 - T_2)/(T_1 - T_2) \) and \( S = (T_1 - T_2)/(T_1 - T_2) \). (a) One shell pass, two or more tube passes. (b) Two shell passes, four or more tube passes. (c) Three shell passes, six or more tube passes. (d) Four shell passes, eight or more tube passes. (e) Six shell passes, twelve or more tube passes. (f) Cross-flow, one shell pass, one or more parallel rows of tubes. (g) Cross-flow, two passes, two rows of tubes; for more than two passes, use \( F_s = 1.0 \). (h) Cross-flow, one shell pass, one tube pass, both fluids counter.

LMTD Correction Factors
Courtesy of Perry’s
Non-linear situations can be handled in two ways:

1. Divide the heat exchanger into zones. Carry out calculations to determine the heat transfer area required for each zone of the exchanger and then add the areas together to give the total area, or

2. By dividing the heat exchanger into zones, each which is treated individually with respect to the linear $Q$ versus $T$ assumption. The overall exchanger performance is then estimated using weighted average performance of the zones in the overall rate equations. The following equations may be taken as the rate equations for overall exchanger and for the $n^{th}$ zone of the exchanger.

\[ Q_{Total} = U_{wtd}A_{Total}(WTD) \]

\[ Q_n = U_nA_n(LMTD)_n \]

Then the weighted temperature difference can be defined as:

\[ WTD = \frac{\sum [U_n A_n(LMTD)_n]}{\sum [U_n A_n]} = \frac{Q_{Total}}{\sum [Q_n/(LMTD)_n]} \]

And the weighted overall heat transfer coefficient becomes:

\[ U_{wtd} = \frac{Q_{Total}}{A_{Total}(WTD)} = \frac{\sum [Q_n/(LMTD)_n]}{A_{Total}} \]
6.7 **METAL RESISTENCE IN TUBES,** $r_w$

The metal resistance, $r_w$ is calculated using:

$$
r_w = \frac{D_o}{24k_w} \ln \left(\frac{D_o}{D_i}\right)
$$

Where: $D = \text{diameter in feet}$

Typical values for metal thermal conductivities are given in the table below.

**Typical Metal Thermal Conductivities, $k_w$**

<table>
<thead>
<tr>
<th>Material</th>
<th>200°F</th>
<th>400°F</th>
<th>600°F</th>
<th>800°F</th>
</tr>
</thead>
<tbody>
<tr>
<td>Aluminum, 3003 Tempered</td>
<td>96</td>
<td>98</td>
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<td>21% Cr, 1% Mo Steel</td>
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<td>11</td>
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<td>79</td>
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<tr>
<td>Copper</td>
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<td>—</td>
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<tr>
<td>90-10 CuNi</td>
<td>30</td>
<td>34</td>
<td>42</td>
<td>49</td>
</tr>
<tr>
<td>70-30 CuNi</td>
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<tr>
<td>Nickel</td>
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<tr>
<td>NiFeCrMoCu (Alloy 825)</td>
<td>7.1</td>
<td>8.1</td>
<td>9.1</td>
<td>10</td>
</tr>
<tr>
<td>Titanium, Gr 3</td>
<td>11.3</td>
<td>11.5</td>
<td>11.5</td>
<td>11.5</td>
</tr>
</tbody>
</table>

*Excerpt from TEMA Standards

In reality, other than in exceptional circumstances, the metal used for heat transfer will have little affect on the total heat transfer area required.
6.9 FOULING RESISTENCES

Fouling resistances depend on:

- The amount and type of dissolved and/or suspended solids in the stream which may deposit on the tube wall.
- The susceptibility to thermal decomposition which will form a thermal barrier on the tube walls.
- The velocity of the fluid.
- The temperature of the fluid.

The fouling resistance specified for a particular design is usually selected based on previous experience with similar streams. An excellent discussion on fouling (including typical fouling resistances) can be found at:


A Few Notes About Fouling

The allowance for fouling will increase the heat transfer area of a typical heat exchanger by up to 30%. If the area attributable to fouling is more than 30% of the total area then the designer should take a close look at the assumptions used.

Fouling is usually assigned a value before the exchanger is designed. This is technically not correct. The applicable fouling factor depends on the velocity and temperature of the fluids in the exchanger both of which are unknown before the design is started. This is especially applicable to heat exchangers using cooling water. The designer should revisit his/her assumptions used to estimate the initial fouling factor vis-à-vis the velocities and temperatures determined during the design to ensure that the initial assumptions were valid.

Fouling is included in the design to account for degradation of performance over time. When the exchanger is first placed in service it is clean and there is no fouling. A clean exchanger will have a larger heat transfer capacity than a fouled exchanger. In some cases this fact can lead to safety issues. For example a clean exchanger in vaporizing service can generate much more vapour than the fouled exchanger. If the pressure relief system (i.e. safety relief valves) are only designed to handle the duty of the fouled exchanger then the system can be overpressured.
Heat Exchanger Cleaning

Heat exchangers in fouling service require periodic cleaning.

They may be cleaned mechanically or chemically. Mechanical cleaning may be with high pressure water or through mechanical scouring. These methods require that the exchanger be taken out of service for cleaning.

Scraped tube heat exchangers are sometimes used in high fouling applications. These exchangers use externally driven scrapers to continuously clean the inside of the tubes while the exchanger is in operation.
7. ESTIMATION OF FILM COEFFICIENTS

Film Coefficients can be divided into three types:

- Sensible Film Coefficients (No Phase Change)
- Condensing Film Coefficients
- Boiling (or Vaporizing) Film Coefficients

In addition the film can be either on the inside or the outside of the tubes. Therefore there are six situations to be considered.

1. Sensible Heat Transfer – Flow on Outside of Tubes
2. Sensible Heat Transfer – Flow on Inside of Tubes
3. Condensing on Outside of Tubes
4. Condensing on Inside of Tubes
5. Vaporizing on the Outside of Tubes
6. Vaporizing on the Inside of Tubes

Of course these 6 film coefficients can be combined with each other in all possible combinations.

The following sections provide the information required to manually film coefficients using the methods described in Kern.

7.1 SENSIBLE HEAT TRANSFER TUBESIDE FILM COEFFICIENTS, $h_i$

Tubeside film coefficients are determined using the Sieder and Tate relation between the heat transfer coefficient, mass velocity and fluid physical properties of a single phase fluid flowing inside tubes.

The relationships are given graphically on the next page. Cooling water is commonly used inside tubes. Correlations using physical properties of water are given on the following page.

The viscosity correction factor, $(\mu/\mu_w)^{0.14}$ is only significant for very viscous fluids and is normally assumed to be 1.

The physical properties are evaluated at the average of the inlet and outlet temperatures. That is the heat transfer coefficient is a function of the physical properties of the fluid which in turn are a function of the temperature. Since the temperature changes as the fluid moves through the exchangers the heat transfer coefficient also changes. By using average temperatures an average heat transfer coefficient is calculated.
Sieder and Tate Relationship
Tubeside Film Coefficients – Single Phase
Courtesy of Kern
Tubeside Film Coefficients – Water
Courtesy of Kern
It is important to note that the film coefficient is strongly influenced by turbulence at the tube wall. Therefore it is desirable to achieve velocities in the tubes high enough to achieve turbulent flow.

The velocities in the tubes will normally be limited by the allowable pressure drop that forms the basis for the design of the heat exchanger. In addition to pressure drop the velocity in the tubes may be limited to erosional velocities as follows:

**Water**

<table>
<thead>
<tr>
<th>Material</th>
<th>Maximum Velocity</th>
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<tbody>
<tr>
<td>Carbon Steel Tubes</td>
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<tr>
<td>Stainless Steel Tubes</td>
<td>15 ft/sec</td>
</tr>
<tr>
<td>Aluminum Tubes</td>
<td>6 ft/sec</td>
</tr>
<tr>
<td>Copper Tubes</td>
<td>6 ft/sec</td>
</tr>
</tbody>
</table>

**Liquids Other Than Water**

\[ V_{\text{max}} = V_{\text{max(water)}} \left( \frac{\rho_{\text{water}}}{\rho_{\text{liquid}}} \right)^{1/2} \text{ ft/sec} \]

**Gases and Dry Vapours**

\[ V_{\text{max}} = \frac{1800}{(P_a \text{ MW})^{1/2}} \text{ ft/sec} \]

\( P_a = \) Absolute Pressure, psia
7.2 SENSIBLE HEAT TRANSFER SHELLSIDE FILM COEFFICIENTS, $h_i$

Methods for the computation of shellside film coefficients in general use today are the proprietary methods developed by Heat Transfer Research Inc. (HTRI) and Heat Transfer and Fluid Flow Services (HTFS). These methods have been experimentally tested against data from commercial scale exchangers for most exchanger geometries. However, they are available only to members and subscribers to the above organizations.

Methods available in the open literature have not been given the same degree of experimental validation but have been used successfully over many years.

In general the heat transfer coefficients calculated using open literature methods will be lower than those calculated using HTRI and HTFS methods. The result is that heat exchangers designed by hand are generally more conservative than those designed using HTRI and HTFS software methods.

A graphical solution based on the methods described by Kern is given below.
Shellside Sensible Heat Transfer With Finned Tubes

Often Finned tubes are used to increase the heat transfer on the shell side of heat exchangers when the film coefficient on the outside of the tubes is very low, for example when the fluid flowing on the outside of the tubes is a gas. The graph below (from Ludwig) can be used as a guideline for determining when the use of finned tubes should be considered.

Low-finned tubes (16 & 19 fins per inch) are adaptable to conventional heat exchanger designs – i.e. they can be used to directly replace plain tubes without altering the design. Higher finned tubes require that the tube spacings be altered and require specialized design procedures.

A table giving physical data for low finned tubes follows.

The design procedure is exactly the same as for plain tubes – except the outside area of the tube is increased.

[Graph showing overall coefficient fouled and fouling factor inside tubes for predicting economical use of finned tubes in shell and tube units. (By permission, R. B. Williams and D. L. Katz, "Performance of Finned Tubes and Shell and Tube Heat Exchangers," University of Michigan, 1951.)]
### Low-Finned Tube Data

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<tr>
<th>Nominal Size</th>
<th>Plain Section Dimensions</th>
<th>Finned Section Dimensions</th>
<th>Approx. Wt/Lf. Lbs. (Copper)</th>
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<td>Inside Diameter</td>
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<th>Nominal Size</th>
<th>Plain Section Dimensions</th>
<th>Finned Section Dimensions</th>
<th>Approx. Wt/Lf. Lbs. (Copper)</th>
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</table>

Low-Finned Tube Data
7.3 CONDENSING FILM COEFFICIENTS

The primary function of a condenser is to remove latent heat, although it is sometimes necessary to remove sensible heat as well. Examples include:

- distillation columns
- reactors
- steam heaters, reboilers
- power plants
- refrigeration systems

Condensers are typically shell and tube exchangers with multiple tube passes. The heat is removed by contacting vapor with a cold surface (the tube wall). The liquid then flows off the tube under the influence of gravity, collects, and flows out of the exchanger. In some cases, vapor flow rates may be high enough to sweep the liquid off the tubes. This is called vapor shear and is a concern when liquid is condensing inside a tube.

Condensing vapor may be a single component or a mixture, with or without the presence of noncondensibles. Usually, mixed vapors are condensed inside tubes, while single components are condensed on the outside of tubes.

Under similar conditions, horizontal tubes tend to have larger condensing heat transfer coefficients than vertical tubes (5x for film type condensation). Vertical tubes are preferred when substantial subcooling of the condensate is required.

In calculations, it is common to assume that the vapor-liquid interface is at thermodynamic equilibrium at the vapor temperature. Liquid adjacent to the cold surface is assumed to be at the surface temperature. It is also common to treat condensers as constant pressure systems, since the total friction losses through an exchanger are usually small.

Condensation Mechanisms

There are two main mechanisms of condensation:

1. If the condensate "wets" the surface, a film forms as the drops coalesce. The condensate forms a continuous layer that flows over the tube (gravity flow) in film type condensation. The primary heat transfer resistance is in the film.

2. If the condensate does not wet the surface, drops form at nucleation sites (pits, dust, etc.) and remain separated until carried away by gravity or vapor flow. Only then do they coalesce, prior to falling off the tube. This is dropwise condensation. Most of the tube surface remains uncovered by liquid, so there is little heat transfer resistance and very high transfer rates.

In both cases, nucleation is typically the rate limiting step, rather than heat transfer. Most industrial applications are based on film mechanisms, since it is tricky and expensive to build non-wetting surfaces.

After condensation, the liquid flows down the tube surface under the influence of gravity (unless vapor rates are high enough to produce vapor shear). The flow may be laminar or turbulent, depending on the fluid, rate of condensation, tube size, etc. The film tends to thicken as it flows to
the bottom of the tube, and the weight of the fluid may cause ripples to form. These will cause deviations from pure laminar flow.

Superheated Vapors

Before a vapor can condense, any sensible heat must be removed. For steam, sensible heat is usually much less than latent and hence is sometimes considered negligible, but this is not true for all vapors.

Practically, one can assume the entire heat load (sensible and latent) is transferred across the condensing film resistance, but if superheat (or subcooling) is substantial, the calculation should be separated into parts -- a desuperheater (gas cooler) and condenser -- and the areas determined separately.

Noncondensibles

The presence of even small amounts of noncondensible gases drastically reduces heat transfer. It has been suggested that only 1-2% air in steam can reduce heat transfer by 75%.

Every condenser requires a venting system for removal of noncondensibles even if just during startup. If noncondensibles are not removed, they will collect in the condenser, "blanketing" the tubes and preventing heat transfer.

Multicomponent Condensation

The above correlations are based on condensing pure components. When a pure component condenses, the temperature at the interface between the vapour and liquid is determined by the local pressure. I.e. the local pressure is the fluid's vapour pressure at the condensing temperature. Generally the pressure drop through a condenser is very small and as a result the condensing temperature of a single component mixture is assumed to be the same throughout the condenser.

When a vapour mixture condenses this relationship is no longer valid. Now the condensing temperature is a function of the vapour and liquid concentrations as well as the pressure. As the least volatile components of the mixture preferentially condense, the condensing temperature will continuously decrease. The relationship between the heat duty, the mass vapour fraction and the equilibrium condensing temperature is known as the “condensing curve”. Generation of the condensing curve by hand is a very time consuming affair and use of the condensing curve data to design a heat exchanger in a stepwise fashion is even more onerous. Thus, design of condensers for multicomponent mixtures is an task best done using software specifically designed to handle this situation. The HTRI CST-2 program is one such program.

Condensing Immiscible Condensate Mixtures

Another aspect to the condensation of vapour mixtures is the possibility of forming immiscible condensate mixtures such as water and hydrocarbons. When this occurs the condensate layer may be a hydrocarbon layer with water droplets suspended in it or a water layer with hydrocarbon droplets suspended in it. Since the condensing coefficient depends on the properties of the condensate layer, the designer must decide which material is continuous and which is dispersed. This is very difficult to do, and in fact the regime may change in different parts of the exchanger. As a result the recommended procedure is to use the physical properties of the material with the worst heat transfer characteristics (the hydrocarbon in this case) for the calculations.
Condensing Coefficients

Graphical solutions to determine condensing coefficients developed by Kern and Akers are presented below.
Tubeside Film Coefficients for Condensation inside Vertical Tubes
Courtesy of Akers & Deans
7.4 VAPORIZERS AND REBOILERS

Boiling Film Coefficients

Heat transfer in vaporizing operations is limited by maximum heat flux. The maximum flux occurs at a critical temperature difference. Beyond this critical temperature difference both the coefficient and flux decrease due to the formation of a layer of gas on the tubes. It is the phenomenon of vapour blanketing that poses the principle difficulty in designing vaporizing exchangers.

The flux is defined as $Q/A$.

The maximum allowable heat flux for forced circulation vaporizers and reboilers vaporizing organics is 20,000 Btu/hr-ft$^2$.

The maximum allowable heat flux for natural circulation vaporizers and reboilers vaporizing organics is 12,000 Btu/hr-ft$^2$.

The maximum allowable heat flux for vaporization of water or aqueous solutions of low concentration (forced or natural circulation) is 30,000 Btu/hr-ft$^2$.

The maximum allowable vaporizing film coefficient for forced or natural circulation vaporization of organics is 300 Btu/hr-ft$^2$-°F.

The maximum allowable vaporizing film coefficient for forced or natural circulation vaporization of water and aqueous solutions of low concentration is 1000 Btu/hr-ft$^2$-°F.

The method for estimating the film coefficients is as follows:

a) Estimate the film coefficient using the previously described methods for sensible heat transfer.

b) If the film coefficient estimated in a) above is higher than the maximum allowable given above use the maximum allowable.
c) If the film coefficient calculated in a) above is lower than the maximum allowable use the lower value.

The figure below can be used to determine the maximum heat flux from the sensible heat transfer film coefficient.

![Maximum Heat Flux versus ΔT](image)

**Maximum Heat Flux versus ΔT**

Courtesy of Kern

**Design of Thermosyphon Reboilers**

The sketch below illustrates the principle for operation of a thermosyphon reboiler. The same principles apply both for horizontal and vertical thermosyphon reboilers.

The basic principle is that the constant vaporization of a portion of the liquid in the reboiler produces a two phase mixture with an average density lower than the liquid being fed to the reboiler. The heavier column of liquid on the reboiler inlet side then pushes the lighter material out of the reboiler causing circulation.

The design requires careful consideration of the hydraulics of the system. Pressure losses in the piping systems to and from the reboiler as well as the pressure loss through heat exchanger must be low enough to allow the circulation to take place.
The procedure for a vertical reboiler is a trial and error procedure as follows:

1. Estimate the surface area required using the guidelines above.

2. Assume a tube length, usually 8 to 16 feet. Calculate the number of tubes required.

3. Assume a recirculation ratio - typically 4:1. The recirculation ratio is ratio of vapour to liquid. Therefore a recirculation ratio of 4:1 means that 20% of the liquid entering the reboiler will be vaporized.

4. Calculate the specific volumes of the liquid and vapour, $v_{\text{liq}}$ and $v_{\text{vap}}$ both in ft$^3$/lb. The specific volume of the liquid is the same as the liquid feed to the reboiler, $v_i$

5. Calculate the liquid flowrate = recirculation ratio x the vapour generation rate.

6. Calculate the total volume out of the reboiler = liquid flowrate x $v_{\text{liq}}$ + vapour generation rate x $v_{\text{vap}}$.

7. Calculate the specific volume of the mixture, $v_o$ = Total volume out of reboiler divided by the total mass flow out of the reboiler.
8. Calculate the static head of the two phase mixture in the reboiler.

\[
\text{Static Head} = \frac{2.3 \times H \times \log \left( \frac{v_o}{v_i} \right)}{144(v_o - v_i)} \text{ psi}
\]

Where \( H \) = height from bottom of heat exchanger tubesheet to centre of reboiler return nozzle on the tower.

9. Calculate the pressure drop through the exchanger using the procedure outlined in the next session.

10. The total resistance is the static head + the exchanger pressure drop + piping pressure drop. Assume that the exchanger piping loop will be designed to keep the pressure drop in this piping below 0.25 psi (or do detailed pressure drop calculations).

11. Calculate the driving force, i.e. the head of the liquid on the inlet.

\[
\text{Driving Force} = \frac{z \rho_l}{144} \text{ psi}
\]

Where \( z \) is the head as shown on the previous diagram.

12. If the driving force calculated in 11 is exactly equal to the resistances calculated in 10, then the recirculation rate will be as assumed. If they are not equal, adjust the exchanger design and repeat the calculation.
8. PRESSURE DROP

8.1 SHELL SIDE PRESSURE DROP

The pressure drop through the shell of an exchanger is proportional to the number of times the fluid crosses the bundle between baffles. It is also proportional to the distance across the bundle each time it is crossed.

The isothermal equation for pressure drop for the shellside flow of a fluid being heated or cooled and including the entrance and exit losses is:

\[ \Delta P_s = \frac{f G_s^2 D_s (N+1)}{5.22 \times 10^{10} D_e s \phi_s} \text{ lbs/in}^2 \]

Where:
- \( \Delta P_s \) = Pressure drop across the shell, lbs/ft\(^2\)
- \( f \) = Friction factor, ft\(^2\)/in\(^2\)
- \( G_s \) = Shell mass velocity, lb/hr-ft\(^2\)
- \( D_s \) = Shell inside diameter, ft
- \( N \) = Number of baffles
- \( D_e \) = Equivalent diameter, ft
- \( s \) = Specific gravity
- \( \phi_s \) = The viscosity ratio \((\mu/\mu_w)^{0.14}\)

![Shellside Friction Factors](image)

Courtesy of Kern
8.2 TUBE SIDE PRESSURE DROP

The tubeside pressure drop is the sum of the pressure drop through the tubes plus the pressure drop through the channels:

\[ \Delta P_t = \frac{\varepsilon G_t^2 L n}{5.22 \times 10^{10} D_e \phi_s} + \frac{4n V^2}{s 2g'} \text{ lbs/in}^2 \]

Where:
- \( \Delta P_t \) = Pressure drop across the tubeside, lbs/ft²
- \( \varepsilon \) = Friction factor, ft²/in²
- \( G_t \) = Tube mass velocity, lb/hr-ft²
- \( L \) = Tube length, ft
- \( n \) = Number of tube passes
- \( D_e \) = Equivalent diameter, ft
- \( s \) = Specific gravity = density, lbs/ft³ / 62.4
- \( \phi_s \) = The viscosity ratio \((\mu/\mu_w)^{0.14}\)
- \( g' \) = Acceleration due to gravity = 32.2 ft/sec²
Reasonably good results are obtained by using the total weight flow and the average specific gravity between the inlet and outlet. This is applicable for change of state situations as well.

This method can be further simplified, for cases of total condensers, by using taking one-half the conventional pressure drop computed based on the inlet conditions. These methods apply to both shell and tubeside calculations.
9 THERMAL DESIGN – OTHER TYPES OF EXCHANGERS

9.1 THERMAL DESIGN – PLATE & FRAME EXCHANGERS

Plate and Frame exchangers are usually sized by the vendors using proprietary methods and correlations for heat transfer and pressure drop specifically developed for their plate channels. Computer programs for rating shell and tube heat exchangers are available from HTRI and HTFS which use generic correlations for heat transfer and pressure drop. Of course these programs are less accurate than the vendor’s programs since they do not refer to the specific plate designs.

It should be noted that Plate and Frame heat exchangers are less prone to fouling than shell and tube types in the same service. In addition they have much higher heat transfer coefficients. Thus, when preparing specifications for a plate and frame exchanger the designer should be careful about specifying the fouling resistance. For example, the clean U for a water-water service shell and tube heat exchanger may be 400 Btu/hr-ft²-°F. If one specified a fouling resistance of 0.002 for both sides of this exchanger the area would be increased by 80% relative to a clean exchanger. However, if the U for the same service plate and frame exchanger were 1000 Btu/hr-ft²-°F then specifying the same fouling factors would increase the area by 200%.

9.2 THERMAL DESIGN – SPIRAL PLATE EXCHANGERS

Spiral Plate heat exchangers are sized by the vendors. However, the heat transfer coefficients are similar in magnitude to those obtained for shell and tube equipment.

9.3 THERMAL DESIGN – DOUBLE PIPE & HAIR-PIN HEAT EXCHANGERS

Double pipe and Hair Pin heat exchangers are sized using the same calculation procedures used for shell and tube heat exchangers.
9.4 THERMAL DESIGN – AIR COOLED HEAT EXCHANGERS

Air cooler heat exchangers are normally designed by the vendors. However their size can be estimated using the following procedure:

1. Select a value for the overall heat transfer coefficient from the table below:

### APPROXIMATE OVERALL HEAT TRANSFER COEFFICIENTS FOR AIR COOLED HEAT EXCHANGERS BASED ON BARE TUBE SURFACE (W/m²·°K)

#### Condensers

- Steam – 100%: 790-850
- Steam – 10% Noncondensibles: 570-620
- Steam – 20% Noncondensibles: 540-570
- Steam – 40% Noncondensibles: 400-425
- Pure Organic Solvents: 425-450
- Ammonia: 570-620
- Mixed Light Hydrocarbons: 370-425
- Medium Hydrocarbons: 250-280

#### Liquid Coolers

- Water: 680-790
- Ammonia: 570-680
- Alcohols and most industrial solvents: 400-425
- Diesel Oil: 250-310
- Light Hydrocarbons: 425-450

#### Vapour Coolers

- 2 bar: 45-55
- 35 bar: 55-85
- Steam: 85-110
- Light Hydrocarbons: 110-170
- Hydrogen: 250-280


2. Choose a bare tube size and material to satisfy the process conditions. Normal size tube sizes are 19.05 mm or 25.4 mm outside diameter for liquid cooling or condensing under pressure or 38.1 mm to 50.8 mm for condensing under vacuum.
3. Pick a design value for the inlet air temperature. This is usually the 95% dry bulb temperature (i.e. the maximum daily temperature does not exceed this value 95% of the time). Choose an outlet air temperature typically 55 °C.

4. Calculate the mean temperature difference. Be sure to use the proper F correction factor for crossflow with the number of tube passes you think might be employed.

5. Calculate \( A_0 \) the bare tube heat transfer surface required.

6. Calculate the air flow rate from the heat balance equation. Calculate the volumetric air flow rate, \( \text{m}^3/\text{sec} \) at the inlet conditions.

7. Using a face velocity of 3 m/sec for air, compute the face area of the bundle.

8. Pick the number of tube rows and tube length and pitch to satisfy both the bundle face area and the required heat transfer area.

9. Choose the number of tube passes to obtain a process side velocity in a reasonable range (1.0 to 2.0 m/sec for liquid cooling).
10 REFERENCES


