PDHonline Course K116 (3 PDH)

Specifying a Liquid-Liquid Heat Exchanger

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Specifying a Liquid-Liquid Heat Exchanger

Christopher Haslego, B.S. ChE

COURSE CONTENT

Heat Transfer Resources

Although most engineers who are asked to specify a heat exchanger may have the appropriate background in heat transfer knowledge, there are cases when the engineer could benefit from a refresher on the basics of heat transfer and the equipment types involved. Here are some resources that will help you review the basics of industrial heat transfer:

Industrial Heat Transfer Basics:
http://www.cheresources.com/heat_transfer_basics.shtml

Design Considerations for Shell and Tube Exchangers
http://www.cheresources.com/designexzz.shtml

Overall Heat Transfer Coefficients in Heat Exchangers
http://www.cheresources.com/uexchangers.shtml

Correlations for Convective Heat Transfer
http://www.cheresources.com/convection.shtml

Shell and Tube Heat Exchanger Design Manual
http://www.wlv.com/products/databook/databook.pdf

Recognizing and Evaluating the Duty Requirements

The first step in specifying any heat exchanger is to properly evaluate and identify the necessary heat transfer duty requirements. In other words, "what do you need the exchanger to do once it's installed?"

A useful tool in evaluating heat transfer duty requirements is the T-Q diagram. This visual tool can help the specifying engineer easily determine what is possible in a given heat exchanger. Let's begin with a simple example.

Due to a process change, one of the plant's main products is exiting the process unit 30 °F higher than before. Sending the product to the storage tank at this elevated temperature may cause safety concerns. As the plant engineer, you've been tasked with specifying a product cooler for this new requirement. The total product stream flow rate is 500,000 lb/h

Previously, the product stream was sent to storage at approximately 130 °F. Now, it's exiting the processing unit at 160 °F. The new product cooler must be able to cool the product stream back down to 130 °F for safe operation. The product stream has physical properties that are very close to those of phenol. For the initial heat balance examination, we'll check the heat capacity of phenol at the midpoint of the cooling duty which is 145 °F to get an average heat capacity through the exchanger. At 145 °F, the heat capacity of phenol is reported as 0.529 Btu/lb °F. Using the following equations:

$$Q_H = m_H \; Cp_H \; (T_{inH} - T_{outH}) \qquad (1)$$

$$Q_C = m_C \; Cp_C \; (T_{outC} - T_{inC}) \qquad (2)$$

Where:
Q = heat transferred in thermal unit per time (Btu/h or kW)
M = mass flow rate
T = temperature
Cp = heat capacity or specific heat of fluid
Subscript "H" = hot fluid
Subscript "C" = cold fluid

Solving Equation 1, we find that the heat transfer duty is:

Q_H = (500,000 lb/h) x (0.529 Btu/lb °F) x (160 – 130 °F) = 7,935,000 Btu/h

Now, we make the following assumption:

1. The heat capacity of the cooling tower water is 1.0 Btu/lb °F
2. Cooling tower water is available at 88 °F during the warmest summer month
3. $Q_H = Q_C$ (perfect heat transfer, a typical assumption)
4. The tower water can undergo a 20 °F temperature rise in the exchanger

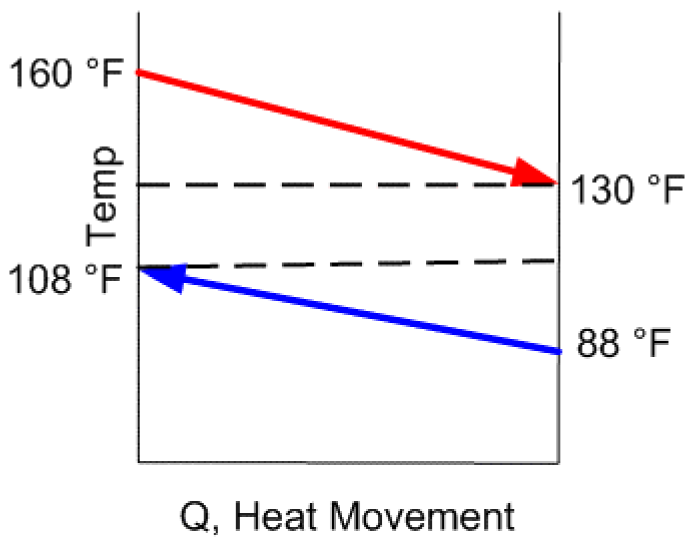
Then, we solve Equation 2 for m_C.

m_C = (7,935,000 Btu/h) / (1.0 Btu/lb °F x 20 °F) = 396,750 lb/h

This is converted to gallon per minute as follows:

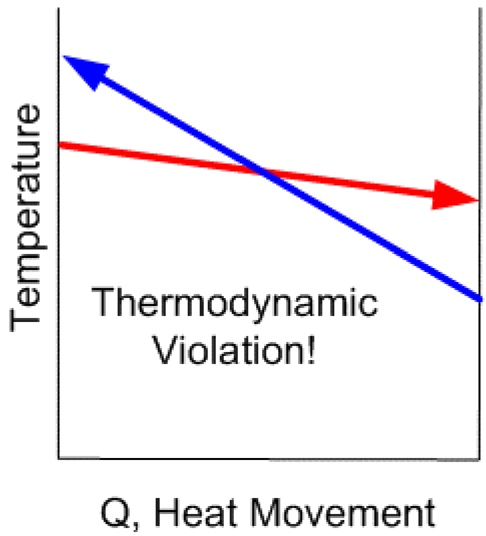
(396,740 lb/h) / (8.27 lb/gal) / (60 min/h) = 800 GPM (nearly a factor of 500, actually 496)

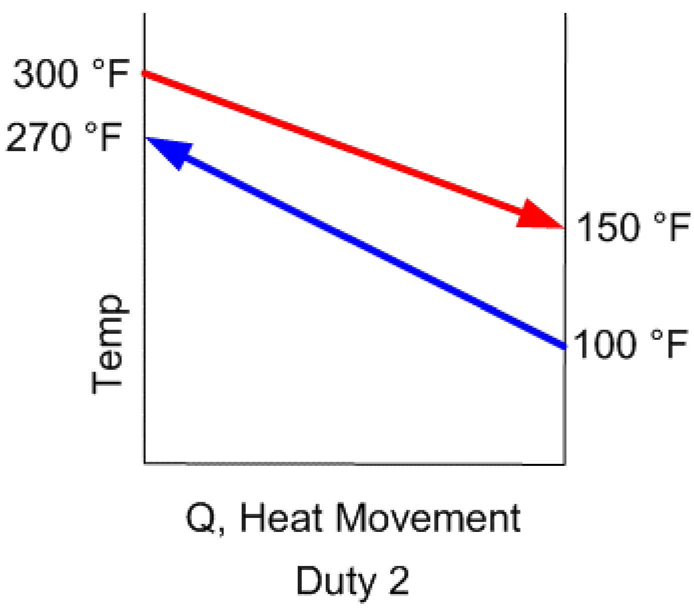
Now, we can construct our T-Q diagram for our system:



Now, we have the basis for what our heat exchanger needs to perform and we've begun to identify the utility requirements for the duty. At this time, we need to note of couple of items. Firstly, as defined, our heat exchanger may require as much as 800 GPM of cooling tower water to perform the cooling task. An investigation should be made to determine if 800 GPM of cooling tower water is actually available. If not, the duty must be re-examined. In this situation, the engineer finds that he has up to 1000 GPM of water available, so this will not be a concern.

Secondly, we note that our duty does not contain any thermodynamic violations and it does not contain a temperature cross. There are two cases are illustrated below:





Notice the T-Q diagram that shows a thermodynamic violation. The cold side is being heated to a temperature that is above the inlet temperature of the hot side. Suppose that in our example, the engineer found that there were only 100 GPM of water available. His analysis would have shown the water would have exited the exchanger far above the 160 °F hot side inlet temperature. In short, this is not enough water to accomplish the duty. At that point, he would have to investigate other utility options.

In the second image above, the T-Q diagram shows what is know as a temperature cross. The cold side outlet temperature is higher than the hot side outlet temperature. It's important to note whether or not your duty contains a temperature cross as it will have a significant impact on the type and number of heat exchangers that may be required to perform the duty.

As the engineer is examining a new heat transfer duty, the concept of NTU or Number of Transfer Units should be used to help guide the specification. A good rule of thumb is that a single shell and tube heat exchanger should be designed with a minimum temperature approach of 10 °F. The "temperature approach" is defined as the temperature difference between the hot side outlet temperature and the cold side outlet temperature. In our example above, the approach temperature is 130 °F-108°F = 22 °F. This duty can easily be accomplished in a single shell and tube heat exchanger.

Now, consider the following duty shown in "Duty 2" above. This unit has a deep "temperature cross". This is where the concept of NTU can be helpful. For Duty 2 above, we calculate the NTU for the hot side and the cold side as follows:

$$NTU_{HOT} = \frac{T_{HOT\ IN} - T_{HOT\ OUT}}{LMTD} \qquad (3)$$

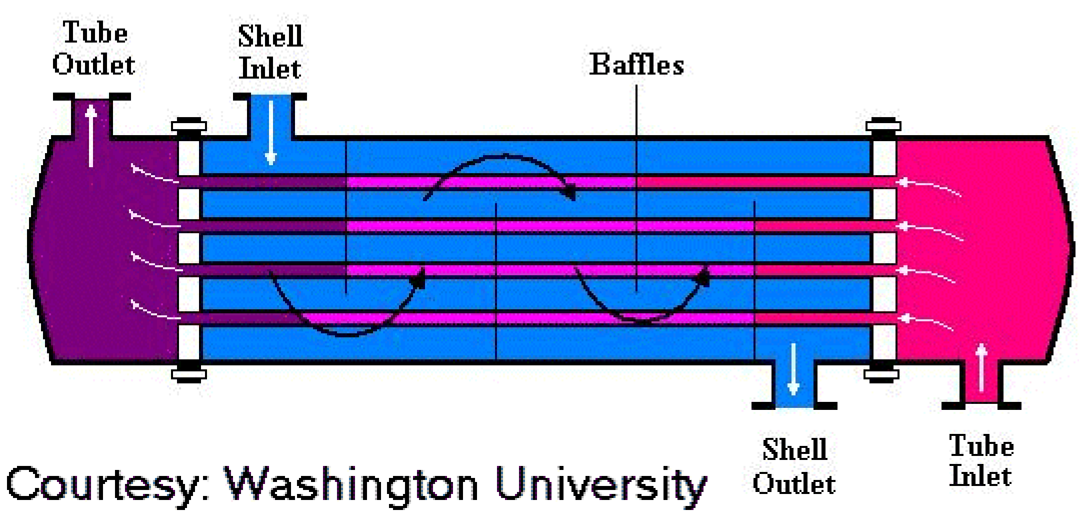
$$NTU_{COLD} = \frac{T_{COLD\ OUT} - T_{COLD\ IN}}{LMTD} \qquad (4)$$

$$LMTD = \frac{(T_{Hot\,in} - T_{Cold\,out}) - (T_{Hot\,out} - T_{Cold\,in})}{LN\left(\frac{T_{Hot\,in} - T_{Cold\,out}}{T_{Hot\,out} - T_{Cold\,in}}\right)} \qquad (5)$$

LMTD = 39.15 °F
NTU_{HOT} = 150 / 39.15 = 3.83
NTU_{COLD} = 170 / 39.15 = 4.34

The NTU can be translated into the approximate number of shell and tube heat exchangers in series that will be required to perform a given duty. The engineer must realize that if it is necessary to perform Duty 2 just as it is shown, it will be an expensive proposition in terms of purchased equipment costs, installation costs, and maintenance costs over the life of the shell and tube heat exchangers.

Shell and tube heat exchanger do a relatively poor job of "temperature crossing" due to their lack of purely countercurrent flow as shown here:



The shell side of the exchanger is almost always baffled so that a reasonable heat transfer coefficient can be obtained. Shell side baffles increase the shell side velocity, minimize fouling, maximize the heat transfer coefficient, and provide support for the tubes. The tube side flow in this image shows a single tube pass. While this is possible, it's not very common. The tube side velocity is the key to the tube side heat transfer coefficient and the ability to mitigate fouling. For these reasons, multiple tube passes are typically used in shell and tube exchangers. The result of these flow patterns is a lack of countercurrent flow. In fact, the LMTD or Log Mean Temperature Difference in shell and tube heat exchangers has be corrected for these flow patterns. Typically, the calculated LMTD has to be multiplied by a factor of 0.70 to 0.90 to account for the flow patterns.

If Duty 2 is required, the engineer may want to consider a heat exchanger type with truly countercurrent flow such as a pipe-in-pipe (commonly called a hairpin exchanger) or a plate heat exchanger. These devices, with their truly countercurrent flow patterns, can perform duties with temperature crosses in a single unit rather than requiring multiple units in series.

Regardless of the heat exchanger type chosen, the engineer **must** be aware of this scenario during the initial specification stage for the heat exchanger.

At this point, the engineer has established the basic parameters for the heat exchanger and now other factors need to be investigated prior to the specification process.

Exploring Other Considerations in Heat Exchanger Specification

Prior to completing the heat exchanger specification data sheet, the engineer should answer questions such as:

Are there any phase changes expected to occur in the heat exchanger?
Are there any dissolved gases in either stream?
Are there any dissolved or suspended solids in either stream?
What are the operating pressures of the streams?
How much pressure loss is available in the exchanger (for existing pumps)?
What are the fouling tendencies of the fluids involved?
Are either of the fluids non-newtonian?
Are either of the fluids corrosive? What material of constructions are required?
What types of elastomers and/or compression gaskets are compatible with the fluids?
Are either of the fluids considered lethal to plant personnel?
Is mechanical cleaning expected to be necessary for one or both fluids?
Is there a cleaning solution that may be effective for the exchanger?
How much room is required for maintenance of the new exchanger?

Having answers to these questions can help ensure that your heat exchanger specification, and ultimately the heat exchanger that you purchase, is right for your heat transfer duty.

Phase Changes

Even in liquid-liquid heat transfer duties, it's important to recognize the potential for phase changes inside the heat exchanger. For example, if a process stream is available to 350 °F to plant water from 100 °F to 300 °F, it's very important to note that both the inlet and outlet pressure of the plant water stream. If the water stream is not under enough pressure, it may undergo partial vaporization. In this case, the vapor pressure of water at 300 °F is about 52 psig. So, for a heat exchanger with a nominal pressure loss of 10 psig, the process water should enter the exchanger at a minimum pressure of around 70 psig.

Dissolved Gases

While not always common, there are several instances where a mostly liquid process stream may pickup dissolved gases. It's important to recognize that dissolved gases can have a profound negative impact on liquid heat transfer as the dissolved gases serve as a significant resistance to heat transfer. If you suspect that dissolved gases may be present, the best bet is to run the process stream through a separation vessel to allow for vapor disengagement prior to transferring heat to or from the stream. Trying to design for a stream with dissolved gases is very difficult and often times uneconomical. A classic example of this can often be found in the interchangers in amine units used to treat natural gases. Through the absorption column, the amine solutions can pick up significant quantities of carbon dioxide that should be removed prior to transferring heat with the amine solution.

Dissolved and Suspended Solids

Dissolved solids can be very common in the chemical processing industry. In fact, a general rule of thumb for cooling tower water is that it should not be heated to temperatures in excess of 120 °F. The reason behind this guideline is that at temperatures above 120 °F, some of the common water treatment chemicals used in the water can quickly plate out onto heat transfer surfaces. Usually, these are carbonate salts which follow an inverse solubility curve. This means that as temperature increases, the salts actually become less soluble in water rather than more soluble as is usually the case. Inverse soluble salts can be found in other processing stream as well and are not confined only to cooling tower water.

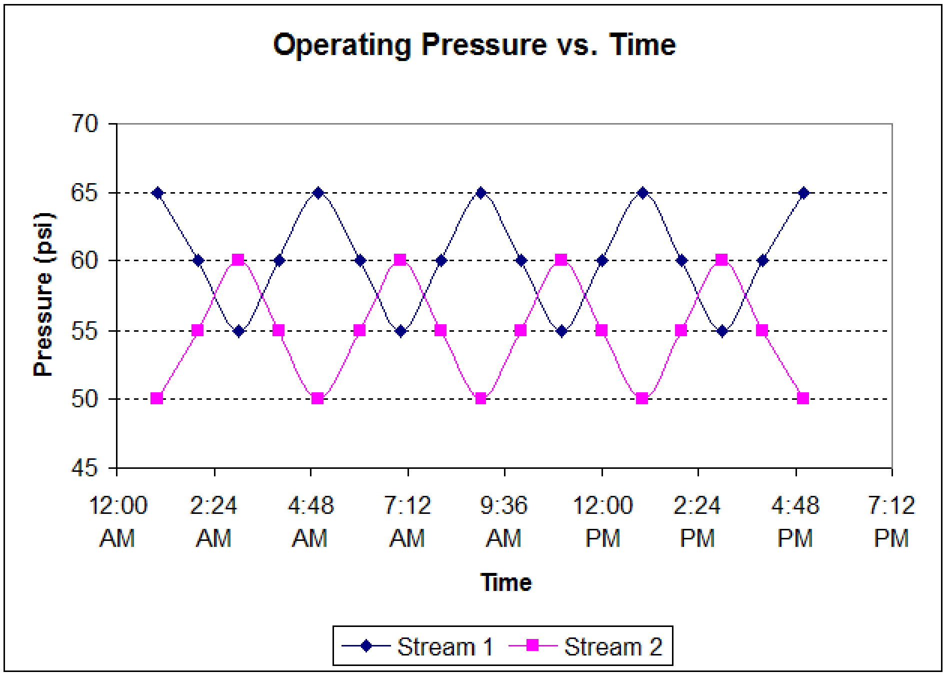
Suspended solids can pose obvious problems for heat exchangers. Aside from the common problem of pluggage, suspended solids can also cause erosion of the heat transfer surfaces if the velocity in the heat exchanger is too high. If suspended solids are present, it's advisable to obtain a particle size distribution and chemical analysis to determine the relative hardness of the particles. For example, hard particles that range in size from a few microns to up to 0.20 mm would have to be addressed differently than a slurry of relatively soft particles such as a powdered solid. Usually, a plant will have a well documented history of what type of exchangers work well with a solid-laden stream and often times, plants will establish velocity limitations for pipelines and other processing equipment.

Operating Pressure

Examine the operating pressures of each stream that are to enter the heat exchanger. Many companies have policies dictating that the design pressure of a process vessel must be a certain factor above the highest operating pressure. For example, if the highest operating pressure for a heat exchanger is to 100 psi, then a reasonable design pressure may be 150 psi or 200 psi. Remember, that the higher the design pressure, the more expensive the exchanger will become as the wall becomes thicker.

Additionally, give some thought to what will happen if a leak occurs within the heat exchanger. The higher pressure fluid will leak to the lower pressure fluid. For example, consider a process stream at 50 psi and cooling tower water at 75 psi. If a leak occurs, the cooling tower water will leak into the process stream. The engineer must evaluate the consequences of such leakage and determine which fluid should be at the higher pressure.

As with any piece of chemical process equipment, it can be subject to mechanical fatigue. Consider a heat exchanger where the two streams have operating pressures that are very close to one another. Without extreme pressure stability (which is often difficult to maintain), a run chart of the pressure versus time may look like the chart below:



Notice how the stream can be allowed to actually cross pressures in the heat exchanger if the two streams are close in operating pressures. This scenario, if extreme enough, can cause flexing of thinner material surfaces inside of heat exchangers and lead to premature failures.

Available Pressure Loss

If preparing to install a heat exchanger in an existing process system, the engineer should examine any pumps in the system to determine how much pressure loss is available. Generally speaking, most heat exchangers should need between 5 and 15 psi of pressure loss to operate effectively. For known fouling fluids, a higher pressure loss (corresponding to a higher velocity) will help keep the exchanger clean for a longer period of time. Also remember that pressure loss is proportional to the fluid viscosity. Specifying a pressure loss of 5 psi for a process fluid with a viscosity of 300 cP may result in a very large heat exchanger.

If your pumping system cannot handle the necessary additional pressure loss to obtain a good heat exchanger design, then an impeller change out, a new pump, or an additional pump in series may be justified. When utilizing a shell and tube exchanger, you can expect the pressure loss on the tube side to be higher than the shell side in most cases.

Fouling Tendencies of the Fluids

The engineer should also be aware of the fouling tendencies of the fluids involved. Through personal experience, interviews with other plant personnel, or investigation into other heat exchangers, the engineer can usually determine how quickly a particular fluid may foul an exchanger. Many plants will have a library of shell and tube fouling factors for various process duties.

Probably one of the most common errors made in specifying a new heat exchanger is overdesign. Anticipating fouling is smart, overdesigning too far however will ensure that fouling will occur. Choose your fouling coefficient carefully. Remember, that specifying too large a fouling factor will often result in more tube or parallel channels. This will lower the velocity in the exchanger and actually promote the fouling. This is a balancing act that is well worth a little time and effort.

When considering a fouling factor, it's very important to note the type of equipment that may be used in the service. Another common mistake during heat exchanger specification is to apply fouling factor information from one type of equipment to a completely different type of equipment.

Remember that shell and tube fouling factors have been compiled over decades through experience and temperature measurement. Also, realize that typical overall heat transfer coefficients for shell and tube may range from 150 to 400 Btu/h ft^2 °F while compact heat transfer technologies can easily obtain overall heat transfer coefficients ranging from 600 to 1000 Btu/h ft^2 °F. If we examine the equation:

$$\frac{1}{U} = \frac{1}{h_H} + \frac{1}{h_C} + R_f \qquad (6)$$

for a shell and tube exchanger and for a compact heat exchanger, we'll see how the difference can impact designs. If the engineer were to specify a fouling factor of 0.001 h ft^2 °F/Btu independent of the type of heat exchanger used, the result would look like this:

$$\frac{1}{U_{S\&T}} = \frac{1}{250} + \frac{1}{300} + 0.001 = 120 \text{ Btu/h ft}^2 \text{ °F}$$

$$\frac{1}{U_{compact}} = \frac{1}{800} + \frac{1}{1000} + 0.001 = 308 \text{ Btu/h ft}^2 \text{ °F}$$

So, the U-value for the shell and tube went from 136 to 120 Btu/h ft^2 °F through the fouling coefficient. The U-value for the compact exchanger went from 445 to 308 Btu/h ft^2 °F through the fouling coefficient. Therefore, the shell and tube overdesign is about 12% while the compact exchanger overdesign is over 40%.

The specifying engineer must realize where the fouling factor information is derived from and apply it properly in the future. While shell and tube exchangers have long used fouling factors, compact heat exchangers generally utilize a "heat transfer margin" that is typically 10-25% over the clean heat transfer coefficient. This change in language was designed to avoid confusion as shown above and to bring the overdesign between the two technologies onto even ground to avoid problems. Also realize that overdesigning in compact heat exchangers is even more detrimental to performance than in a shell and tube heat exchanger.

Non-Newtonian Fluids

While most fluids in the chemical processing industry are Newtonian in their flow behavior, some are not. In short, a Newtonian fluid is one whose viscosity in NOT dependent on the forces acting upon it (shear stress in heat exchangers), only on the fluid's temperature. Some fluids, known as being non-Newtonian, have flow characteristics such that they can actually become more or less viscous depending on the forces acting on the fluid. Confirming that a fluid is Newtonian during the design stage can save the engineer from procuring a heat exchanger that is vastly over or under sized later.

Corrosion Potential and Materials of Construction

Specifying materials on construction for the heat exchanger is an extremely important part of the overall process. Again, most plants have some history regarding what metals are appropriate for their process fluids. Typically, if one fluid requires a higher metallurgy than another, then that fluid is placed on the tube side of a shell and tube exchanger to minimize costs as cladding a shell can become quite expensive.

It's important to consider temperature and pH when deciding on a material of construction for your exchanger. If you're not sure what metal you need, consult with a corrosion expert as this is one aspect of heat exchanger design that no one can afford to get wrong.

If your duty does require an expensive alloy, then a compact heat exchanger may cost significantly less considering their higher overall heat transfer coefficients. Another point to remember is that just because a fluid is compatible with a stainless steel tube for example, it may not be compatible with a stainless steel plate that has been pre-stressed (during the pressing process). Pre-stressing of metals can make them susceptible to pitting corrosion such as chloride attack. Consult with manufacturers of compact equipment. While they will seldom take the legal responsibility for choosing a material of construction, then can point you in the right direction and save you from making a costly mistake.

Elastomers and Compression Gasket Compatibility

Depending on the type of heat transfer technology that is being considered for the application, a check of gasket compatibility may be required. Elastomer gaskets are most commonly offered in materials such as EPDM, Nitrile, PTFE, and FKMG (a generic form of Viton-G from Dupont). Elastomer gaskets can seldom be rated for temperatures in excess of 320 °F. Generally speaking, the engineer should seek a recommendation from the heat exchanger manufacturer as they usually have extensive databases that show the best gasket choice for a given application.

For compression gaskets, such as those used on the heads of shell and tube exchangers, there are a couple of rules of thumb to keep in mind. In addition to the need for the gasket to be compatible with the process or service fluid, the engineer may need to decide between a metallic or non-metallic compression gasket. Consider this guideline:

Find the value of: Operating Pressure (psig) x Operating Temperature (°F)
If this value exceeds 250,000, the use of a metallic gasket should be strongly considered. Additionally, non-metallic gaskets are seldom used at pressures in excess of 1200 psig and temperatures in excess of 850 °F.

Lethal Service Requirements

The ASME pressure vessel code stipulates very specific pressure vessel requirements for heat transfer service that are qualified as "lethal". If the service requires an ASME "L" stamp, be sure to specify this to the heat exchanger manufacturer.

Cleaning Considerations

Some process fluids can leave fouling deposits that can be especially difficult to remove. Sometimes, these deposits can be removed by chemical cleaning. Chemical cleaning of heat exchangers, in general, is popular in industries that utilize sanitary protocols (food, pharmaceutical, etc.) and chemical cleaning is widely accepted in the chemical process industry in Europe. Chemical cleaning requires additional equipment, cleaning chemicals, and a method of disposing of the chemical cleaning agent. Chemical cleaning can be a good choice in the following instances:

1. The fouling deposit can be easily dissolved and removed by a readily available cleaning agent.
2. The heat exchanger fouls quickly and must be cleaned fairly often (4 or more times a year)
3. The heat exchanger to be cleaned has a relatively small hold up volume so that chemical cleaning equipment and the volume of cleaning agents can be minimized.

For heat transfer duties where chemical cleaning does not seem like the best choice, the engineer must be sure that the fouling fluid is placed on a side of the heat exchanger that is readily accessible for mechanical cleaning. Mechanical cleaning usually consists of a high pressure water spray of the affect area, although additional scraping can sometimes be necessary. Floating head shell and tube heat exchangers, gasketed plate exchangers, spiral heat exchangers, and some welded plate heat exchangers allow good access for mechanical cleaning.

A final consideration for mechanical cleaning is the space required around the heat exchanger. When choosing an installation location, be sure that the necessary maintenance space is available for proper and safe maintenance of the new equipment.

Shell and Tubes – Where Should I Put the Fluids?

In shell and tube heat exchangers, the specifying engineer has to decide whether each fluid should be placed on the shell side or the tube side. In general, fluids that exhibit these characteristic are preferred for the tube side:

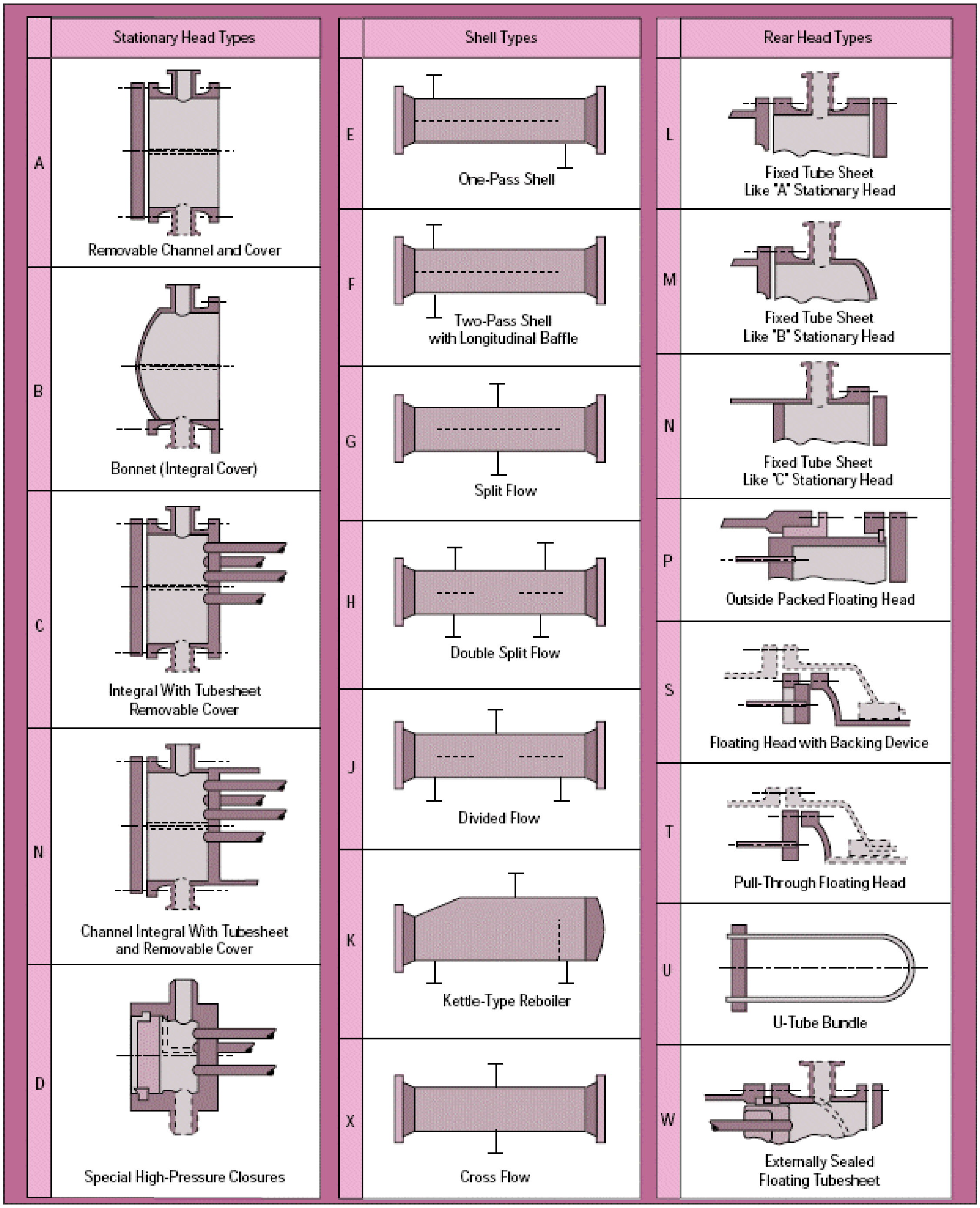
1. High pressure fluids
2. Corrosive fluids
3. Fouling fluids
4. Viscous fluids
5. Slurries or fluids with significant solid loading

Placing the high pressure fluid in the tubes will minimize the cost associated with the exchanger because the cost of thicker tube walls is generally less expensive than a thick shell. Corrosive fluids that require a higher alloy are also best placed in the tubes so that the shell does not have to be cladded with or fabricated from an expensive material. It is a "must" to place the most fouling fluid inside the tubes. The shell sides of shell and tube heat exchangers are notoriously difficult to clean. Viscous fluids are certainly good candidates for tube side flow as well. The heat transfer coefficient in an exchanger with a viscous fluid will almost certainly be limited by the viscous fluid. The heat transfer coefficient of a viscous fluid will be higher on the tube side than the shell side.

There may be situations where the engineer would prefer both fluids be on the tube side. In such cases, the engineer will have to consider each fluid carefully. In some cases, a shell and tube heat exchanger may not be the best choice and another heat transfer technology may have to be considered.

TEMA Designations for Liquid-Liquid Heat Exchangers

Shell and tube heat exchangers are available in a wide range of configurations as defined by the Tubular Exchanger Manufacturers Association (TEMA, www.tema.org). 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 flows outside of the tubes within the shell. The tube side and shell side fluids are separated by a tube sheet.



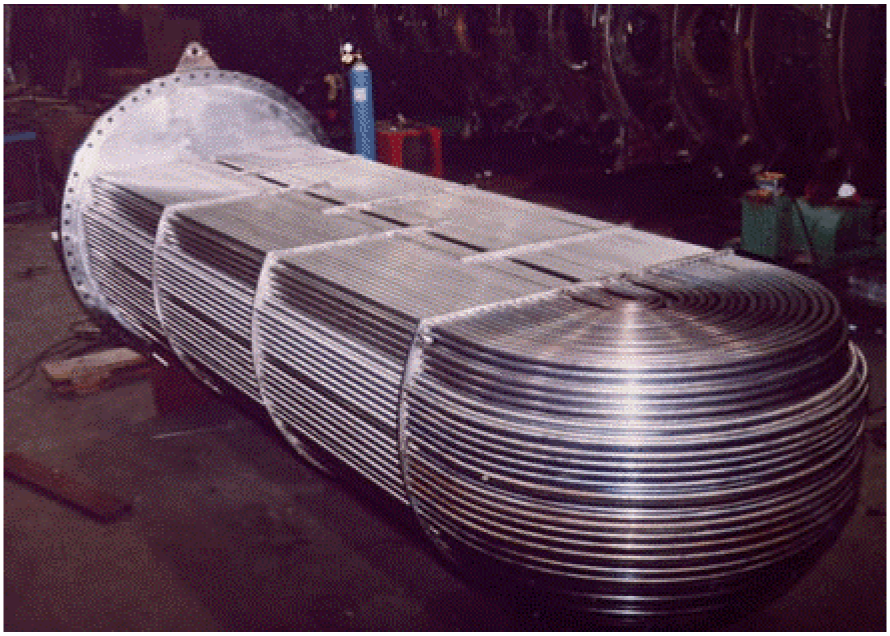
Source:
CHEMICAL ENGINEERING PROGRESS • FEBRUARY 1998

Each shell and tube exchanger is designated by a three (3) letter code. The letters refer to the specific type of stationary head at the front, the shell type, and rear head type. Fixed tube sheet style exchangers with TEMA designations of BEM, AEM, or NEN are fairly common for liquid-liquid heat transfer duties. While fixed head arrangements have the advantage of being inexpensive and avoiding gaskets or packing to contain the shell side fluid, they do not allow for mechanical cleaning of the shell side.

For liquid-liquid heat transfer duties where the ability to mechanically clean the shell side as well as the tube side is a requirement, a floating head design should be chosen. With a floating head design, the tube bundle can be pulled out of the shell to allow access to the shell side. Some of the most commonly used floating head designs for liquid-liquid duties include AES and BES. Another advantage of the floating head design is the ability to accommodate larger temperature differentials between the hot side and cold side fluids.

All of the designations discussed so far (BEM, AEM, NEN, AES, and BES) allow for multiple tube passes which are usually required for liquid-liquid duties so that the tube side velocities can be manipulated during the design stage of the exchanger.

The final type of shell and tube exchanger commonly used for liquid-liquid duties is commonly referred to as the “U-tube” type. Common TEMA designations are BEU and AEU. In this arrangement the tubes are bent into a series of concentrically tighter U-shapes with the end of the tubes being attached to the tube sheet.



The U-tube bundle can be removed to access the shell side for mechanical cleaning. The U-tube design is preferred for services with temperature or pressure cycling, intermittent service, and when there is a large temperature differential between the shell side and tube side fluids.

Since the “U” bend of the tubes cannot be accessed for mechanical cleaning, the tube side fluid should be clean or a suitable chemical cleaning agent should be identified.

Methods of Estimating Physical Properties

When specifying a liquid-liquid heat exchanger, the specifying engineer must be able to provide physical properties that are as accurate as possible. For liquid-liquid duties, the following data should be provided for each fluid: density, thermal conductivity, specific heat (sometimes called heat capacity), and viscosity. Ideally, these properties should be provided for each fluid at both the inlet and outlet temperature of the exchanger.
If data is limited, there are some estimating rules that be of assistance. The physical properties that will impact the design of the exchangers the most are the viscosity and the specific heat. Recall that the specific heat of a fluid is required in order to accurately specify the exchanger. Now, we’ll examine estimation methods for each of the physical properties. While actual plant data or experimentally determined data is preferred, these methods can be used when no other

data is available. For our estimation methods, we'll assume that typical process fluids are made up of mixtures of components and that the data for each component is available. This is almost always the case.

Specific Heat

For fluid mixtures where there is no known heat of mixing, a weighted average can be used:

$$C_{pmix} = \sum W_1 C_{pL1} + W_2 C_{pL2} + \ldots \quad (7)$$

Where:
C_{pmix} = Heat capacity of the mixture in consistent units
W_1 = Weight fraction of component one
C_{pL1} = Heat capacity of component one in consistent units
W_2 = Weight fraction of component two
C_{pL2} = Heat capacity of component two in consistent units

If using the above method, look up the heat capacity of the components at the average temperature through the heat exchanger. It's not uncommon to provide heat exchanger designers with a single heat capacity point for each fluid. Be aware that the heat capacity of most liquids will increase with temperature.

Viscosity

While it's not critical to supply a physical property point at the inlet and outlet temperature for other properties, it's very beneficial to do for the viscosity of the fluids. Using only a single viscosity point will affect both the heat transfer and pressure drop calculation of any heat exchanger.

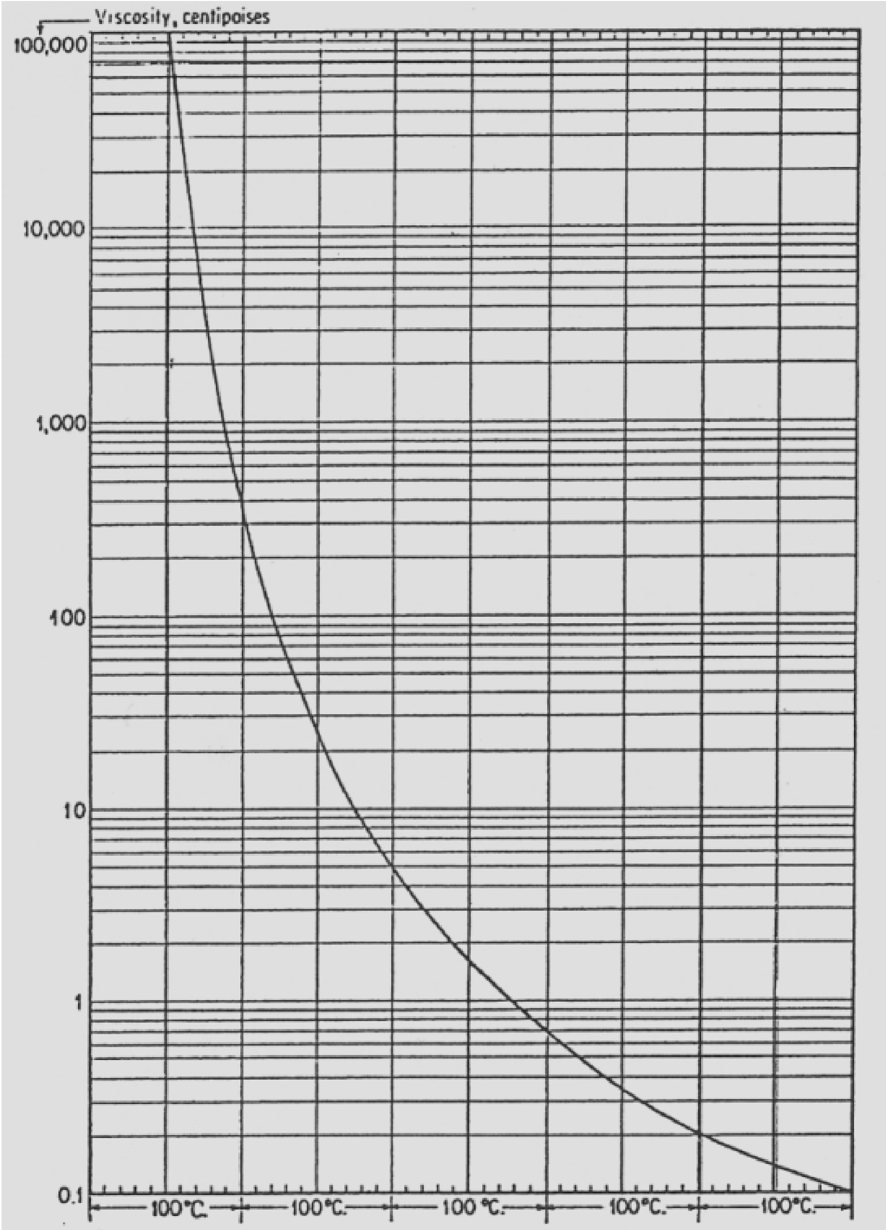
For non-polar mixtures, the following has shown to provide viscosity estimates to within +/- 5 - 10%:

$$\ln \mu_{mix} = \sum W_1 \ln \mu_1 + W_2 \ln \mu_2 + \ldots \quad (8)$$

Where:
μ_{mix} = Viscosity of mixture in centipoise
W_1 = Weight fraction of component one
μ_1 = Viscosity of component one in centipoise
W_2 = Weight fraction of component two
μ_2 = Viscosity of component two in centipoises

For polar fluids, electrolyte solutions, and non-newtonian fluids, it is highly advisable to either find reliable data or have an outside lab perform testing. There is an estimation method available for polar mixtures called the "Method of Grunberg and Nissan". This method is detailed in *Properties of Liquids and Gases*, Edition 4 by Reid et al. (ISBN 0070517991, see page 474).

If viscosity data is available at one temperature, the following correlation chart that has been used for years to estimate the viscosity at a second temperature. This chart is used by finding the single available viscosity point on the Y-axis and move left to meet the curve. Then, on the X-axis, adjust the temperature difference up or down by the temperature change required. Next, move up to hit the curve again and read the resulting viscosity from the Y-axis.



Thermal Conductivity

The thermal conductivities of mixtures can be estimated via the same weighted average method shown in Equation 9 for specific heat calculations. The calculation is the same, just replace the component specific heats with the component thermal conductivities. In cases where there is a complete absence of data, keep the following ranges in mind:
Water based mixtures, thermal conductivity range is about 0.28 to 0.35 Btu/ h ft °F
Hydrocarbon based mixtures, thermal conductivity range is about 0.055 to 0.080 Btu/ h ft °F

Density

The density of mixtures represents another case where a weighted average method is usually adequate as an estimate for heat exchanger design.

Avoiding Specifications That Are Too Specific

One final pitfall that the specifying engineer should avoid is making the heat exchanger specification too rigid. The engineers that design the heat exchangers are the experts, give them as much freedom as possible and allow them to present you with the best option(s) that will work well for your application.

As an example, consider the following:

A process stream requires Alloy C-276 material to guard against corrosion. The stream needs to be cooled with cooling water before being sent to storage. The metallurgy makes the process stream an immediate candidate for the tube side of a shell and tube heat exchanger. The cooling water is available at 80 ^{0}F and must be returned at a temperature no higher than 115 ^{0}F. The process engineer realizes that with the water flow being placed on the shell side, larger flow rates will enhance the heat transfer coefficient. The basis for the heat exchanger quotation was specified as follows:

	Tube side	Shell side
Flowrate (GPM)	500	1800
Temperature in (^{0}F)	280	80
Temperature out (^{0}F)	150	92
Allowable pressure drop (psig)	15	15

According to the engineer's calculations, these basic parameters should provide a good shell and tube design with a minimum amount of Alloy C-276 material (an expensive alloy). The completed specification sheet is forwarded to many manufacturers, including those that could easily quote plate and frame or another compact technology. A typical plate and frame unit designed to meet this specification would have about 650 ft^2 of area compared to about 420 ft^2 for a shell and tube exchanger. A plate and frame unit designed to the above specification is limited by the allowable pressure drop on the cooling water. If the cooling water flow is reduced to 655 GPM and the outlet water temperature rose to 115 ^{0}F, the plate and frame heat exchanger would contain about 185 ft^2 of area. The unit is smaller, less expensive, and uses less water. The load being transferred to the cooling tower is the same.

The theory that applied to the shell and tube heat exchanger (increasing water flow will minimize heat transfer area), works in exactly the opposite direction for compact technologies. The larger water flow actually drives the cost of the unit upward. Rather than supplying a rigid specification to all heat exchanger manufacturers, the engineer should have explained his goal in regards to the process stream. Then he could have stated the following:

"The process stream is to be cooled with cooling water. Up to 2000 GPM of water is available at 80 ^{0}F. The maximum return temperature is 115 ^{0}F."

This simple statement could result in vastly different configurations when compared with the designs that would result from the original specification.

Completing the TEMA Specification Sheet

The TEMA specification sheet shown below has been color coded to help explain which information should be provided by the specifying engineer and which information should be provided by the designer/manufacturer. Green cells are to be completed by the specifying engineer, yellow cells by the designer/manufacturer, and gray cells could be completed by either party.

The Chemical Engineers' Resource Page
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Courtesy of: www.cheresources.com	TUBULAR HEAT EXCHANGER SPECIFICATION SHEET	Page 1 of 2
Package No.	Doc. No.	Rev.

Tag No.			
Unit		Number of Units	
Service		Series/Parallel	
Area/Unit		Inquiry No.	
		Quote No.	
Size & Type		P.O. No.	
Supplier		Job No.	
Manufacturer		Serial No.	

PERFORMANCE PER PARALLEL TRAIN OR PER UNIT

		Shell Side	Tube Side
Fluid Circulated			
Total Fluid Entering	GPM - lb/h	-	-
Vapor	lb/h		
Liquid	GPM - lb/h	-	-
Steam	lb/h		
Non-Condensibles	lb/h		
Fluid Vaporizer/Cond	lb/h		
Steam Condensed	lb/h		

		Shell Side	Tube Side
Temperature In	°F		
Temperature Out	°F		
Operating Pressure	psig		
Number of Pass Per Shell			
Velocity	ft/s		
Pressure Drop	psig		
Fouling Coefficient	ft^2 °F h/Btu		

Heat Exchanged : Btu/h LMTD (corrected) : °F

"U" Value - Clean/Service : / Btu/h ft^2 °F Surface Area/Unit : ft^2

PHYSICAL PROPERTIES

		Shell Side	Tube Side
Liquid Properties			
Density	lb/ft^3	In / Out	In / Out
Heat Capacity	Btu/lb °F	In / Out	In / Out
Viscosity	cP	In / Out	In / Out
Thermal Cond.	Btu/h ft °F	In / Out	In / Out
Vapor Properties			
Density	lb/ft^3	In / Out	In / Out
Heat Capacity	Btu/lb °F	In / Out	In / Out
Viscosity	cP	In / Out	In / Out
Thermal Cond.	Btu/h ft °F	In / Out	In / Out
Critical Temp.	°F		
Critical Press.	psia		
Vapor Pressure	psia	In / Out	In / Out

*For condensing duties, supply heat release curve if available and latent heats

**For non-Newtonian fluids, supply apparent viscosity data or consistency index and flow behavior index

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CONSTRUCTION

Design Pressure psig :
Test Pressure psig :
Design Temperature °F :

Tubes ____ No. ____ OD ____ BWG ____ Length ____ Pitch ____
Shell ____ ID ____ OD ____

Shell Cover ____ Floating Head Cover ____
Channel ____ Channel Cover ____
Tube Sheets-Stationary ____ Floating ____
Baffles-Cross ____ Type ____
Baffles-Long ____ Type ____
Tube Supports ____
Gaskets ____
Connections :
Shell : In ____ Out ____ Type ____ Rating ____ psi
Channel : In ____ Out ____ Type ____ Rating ____ psi
Corrosion Allowance : Shell Side ____ Tubeside ____
Code Requirements ____ TEMA Class ____
Weights :
Each Shell ____ lbs. Bundle ____ lbs. Full of Water ____ lbs
Stress Relieved Parts :

Radiographed Parts :

SKETCH

REMARKS

The information on the first page of the specification is essentially a description of the problem as has been covered herein. On the second page, the specifying engineer is asked to indicate the design pressure and temperature. The test pressure is generally accepted as 1.3 times the design pressure as specified by the latest version of the ASME Pressure Vessel code. Some times, however, the specifying engineer may request a higher design pressure (perhaps 1.5 times design pressure). The engineer is also expected to specify materials or choices of materials for the heads, shell, tube sheets, tubes, and the compression gaskets (if applicable).

The specifying engineer can also indicate any required corrosion allowances required as well as any special mechanical or non-destructive testing that may be required for an exchanger to be installed in a particular duty.

If the installation of the exchanger would be simplified by a particular nozzle arrangement or maximum overall length, this type of information can be provided in the sketch box or in the remarks section. Use the remarks section to convey any other pertinent information to the designer.