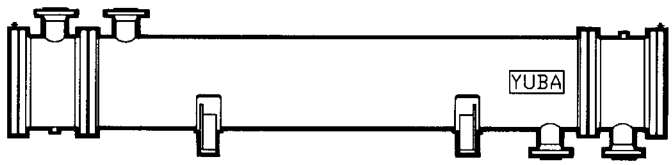
SHELL AND TUBE EXCHANGERS
TECHNICAL MANUAL



Yuba®

INDEX

SHELL AND TUBE EXCHANGERS

APPROXIMATE HEAT EXCHANGER SIZING

Very often, in the course of preparing a heat exchanger specification, there is a need to know the approximate size of the equipment in advance of quotations. This information may be used for preliminary budgeting or may influence the selection of bidders, the choice of materials, or the type of exchanger selected. Moreover, the size, if much larger or smaller than anticipated, may influence the design or arrangement of associated equipment. The following data and procedure can be applied to quickly establish approximate heat transfer surface requirements.

Basic Heat Transfer Equation

$$Q = UA\Delta T \quad \text{or} \quad A = \frac{Q}{U\Delta T}$$

Q = Heat Transferred, BTU/Hour
U = Overall Heat Transfer Coefficient, BTU/Hour/Square Foot/°F
A = Square Feet of Heat Transfer Surface
ΔT = Corrected Logarithmic Mean Temperature Difference, °F

Of these quantities, A is the desired unknown; Q is known; ΔT can be quickly and accurately calculated; U can be approximated from average values shown in Table 1. The equation is then solved for A.

Calculation of ΔT

Logarithmic Mean Temperature Difference:

The LMTD is determined by the relationship of the fluid temperature differences at the Terminals of the heat exchanger. The terminal temperatures of the fluids are designated below:

T_1 = hot fluid inlet temperature.
T_2 = hot fluid outlet temperature.
t_1 = cold fluid inlet temperature.
t_2 = cold fluid outlet temperature.

Counter-current flow is assumed. The heat exchanger terminal termperature differences may be expressed as below:

$T_1 - t_2$ = hot terminal temperature difference
$T_2 - t_1$ = cold terminal temperature difference.

The greater of these two differences is designated as GTTD and the lesser as LTTD. The logarithmic mean temperature difference may be calculated as follows:

$$\text{LMTD} = \frac{\text{GTTD} - \text{LTTD}}{\text{Log}_e \dfrac{\text{GTTD}}{\text{LTTD}}} = \frac{\text{GTTD} - \text{LTTD}}{2.303\ \text{Log}_{10} \dfrac{\text{GTTD}}{\text{LTTD}}}$$

The LMTD may also be read directly from Figure 1.

If flow is true counter-current, the LMTD is the ΔT for use in the basic heat transfer equation.

If flow is not true counter-current (more tube passes than shell passes), the LMTD must be corrected. The correction factor F may be obtained from Figure 2 or 3, based on the following calculated values:

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

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

Example:

Stream A temperatures — 250°F inlet, 125°F outlet
Stream B temperatures — 85°F inlet, 115°F outlet

$T_1 - t_2 = 250 - 115 = 135$°F
$T_2 - t_1 = 125 - 85 = 40$°F
GTTD = 135°F
LTTD = 40°F

LMTD = 78 (from Figure 1)

Assuming multi-pass tube side and single pass shell side design, the LMTD correction factor is calculated below.

$$P = \frac{t_2 - t_1}{T_1 - t_1} = \frac{115-85}{250-85} = .182$$

$$R = \frac{T_1 - T_2}{t_2 - t_1} = \frac{250-125}{115-85} = 4.17$$

$F = .88$ (from Figure 2)

Note:
If the correction factor is below 0.8 in Figure 2, then more than one shell pass is required. Figure 3 provides correction factors for two pass shell flow. If the correction factor in Figure 3 is below 0.8, several shell passes are required or consideration should be given to using a design that provides true counter-current flow.

Corrected ΔT

Compute the corrected ΔT by multiplying the LMTD by the correction factor F.

$\Delta T = LMTD \times F = 78 \times .88 = 69.$

Overall Heat Transfer Coefficient, U

The overall coefficient, U, is computed from the following equation:

$$\frac{1}{U} = \frac{1}{h_o} + \frac{1}{h_{io}} + r_f + r_w$$

h_o = Shell side heat transfer coefficient. Obtain from Table 1.
h_{io} = Tube side heat transfer coefficient (referred to outside). Table 1.
r_f = Fouling resistance, arbitrarily assigned. Varies from 0.001 to 0.010.
A good average value to use is 0.003. Use 0.0015 for clean services, 0.005 for dirty services.
r_w = Tube metal-wall resistance to heat flow. Assumed as constant at 0.0004.

Example: Shell side — Kerosene: $h_o = 150$
Tube side — Water: $h_{io} = 700$

$$\frac{1}{h_o} = 0.0067$$

$$\frac{1}{h_{io}} = 0.0014$$

$r_f = 0.003$ (assumed)
$r_w = 0.0004$ (assumed constant for all cases)

$$\frac{1}{U} = 0.0067 + 0.0014 + 0.003 + 0.0004 = 0.0115$$

$U = 87$

Heat Transfer Surface:

Using the ΔT and U from the example above and assuming that Q has been set at 5,000,000 BTU/Hour, the surface, A, required is:

$$A = \frac{5{,}000{,}000}{87 \times 69} = 833 \text{ square feet.}$$

TABLE 1 — APPROXIMATE HEAT TRANSFER COEFFICIENTS

Heat transfer coefficients in this table are based upon "average" conditions of temperature, pressure, corresponding fluid properties, pressure drop allowed and construction features. Coefficients can vary appreciably with even small departures from these average conditions. However, under most conditions accuracy will be within 25 percent for a single exchanger and will improve with a number of units. Notes at the conclusion of the table may assist in evaluating unusual conditions.

LIQUIDS	Shell Side	Tube Side
Oils, 20° API		
200°F Average temperature	40 - 50	15 - 25
300°F Average temperature	70 - 85	20 - 35
400°F Average temperature	80 - 100	65 - 75
Oils, 30° API		
150°F Average temperature	70 - 85	20 - 35
200°F Average temperature	80 - 100	50 - 60
300°F Average temperature	110 - 130	95 - 115
400°F Average temperature	130 - 155	120 - 140
Oils, 40° API		
150°F Average temperature	80 - 100	50 - 60
200°F Average temperature	120 - 140	115 - 135
300°F Average temperature	150 - 170	140 - 160
400°F Average temperature	180 - 200	175 - 195
Heavy Oils, 8 - 14° API		
300°F Average temperature	20 - 30	10 - 20
400°F Average temperature	40 - 50	20 - 30
Diesel Oil	115 - 130	95 - 115
Kerosene	145 - 155	140 - 150
Heavy Naphtha	145 - 155	130 - 140
Light Naphtha	180	180
Gasoline	200	200
Light Hydrocarbons	250	250
Alcohols, most organic solvents	200	200
Water, ammonia	700	700
Brine, 75% water	500	500

	Shell or Tube Sides				
VAPORS	10 PSIG	50 PSIG	100 PSIG	300 PSIG	500 PSIG
Light Hydrocarbons	25	60	100	170	200
Medium HC's, Organic Sol.	25	70	105	180	220
Light Inorganic Vapors	14	30	60	100	120
Air	13	25	50	85	100
Ammonia	14	30	55	95	110
Steam	15	30	50	90	135
Hydrogen — 100%	40	105	190	350	420
Hydrogen — 75% (vol.)	35	80	150	280	340
Hydrogen — 50% (vol.)	30	70	130	240	310
Hydrogen — 25% (vol.)	25	55	100	180	270

TABLE 1 (Continued)

VAPORS CONDENSING	**Shell or Tube Sides**
Steam	1500
Steam, 10% non-condensables	600
Steam, 20% non-condensables	400
Steam, 40% non-condensables	220
Pure light hydrocarbons	250 - 300
Mixed light hydrocarbons	175 - 250
Gasoline	150 - 220
Gasoline-Steam mixtures	200
Medium hydrocarbons	100
Medium hydrocarbons with steam	125
Pure organic solvents	250
Ammonia	600
LIQUIDS BOILING	
Water	1500
Water solutions, 50% water or more	600
Light hydrocarbons	300
Medium hydrocarbons	200
Freon	400
Ammonia	700
Propane	400
Butane	400
Amines, alcohols	300
Glycols	200
Benzene, toluene	200

NOTES:

1. Where a range of coefficients is given for liquids, the lower values are for cooling and the higher are for heating. Coefficients in cooling, particularly, can vary considerably depending upon actual tube wall temperature.

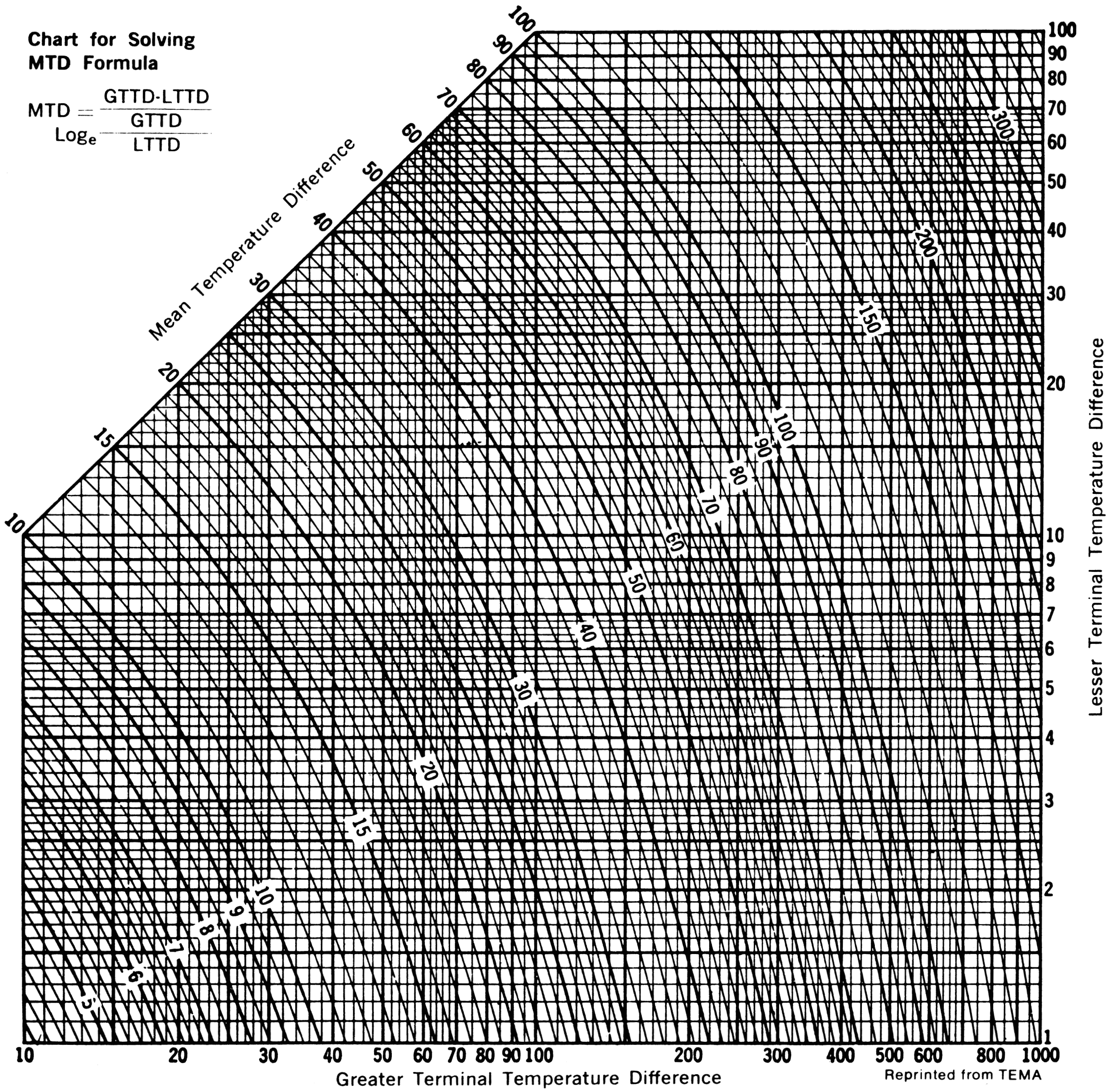
2. Tube side coefficients are based on ¾" diameter tubes. Adjustment to other diameters may be made by multiplying by 0.75/actual outside diameter. Shell side coefficients are also based upon ¾" diameter. Precise calculations would require adjustment to other diameters. The accuracy of this procedure does not warrant it.

3. Coefficients can vary widely under any one or combination of the following:
   a. Low allowable pressure drop.
   b. Low pressure condensing applications, particularly where condensation is not isothermal.
   c. Cooling of viscous fluids, particularly with high coefficient coolants and large LMTD's.
   d. Condensing with wide condensing temperature ranges — 100°F and larger.
   e. Boiling, where light vapor is generated from viscous fluid.
   f. Conditions where the relative flow quantities on shell and tube sides are vastly different (usually evidenced by difference in temperature rise or fall on shell and tube sides).
   g. Wide temperature ranges with liquids. (May be partly in streamline flow.)

Logarithmic Mean Temperature Difference Chart

Chart for Solving MTD Formula

$$MTD = \frac{GTTD - LTTD}{\log_e \frac{GTTD}{LTTD}}$$



NOTE—For points not included on this sheet multiply Greater Terminal Temperature Difference and Lesser Terminal Temperature Difference by any multiple of 10 and divide resulting value of curved lines by same multiple.

FIGURE 1 — BULLETIN 1

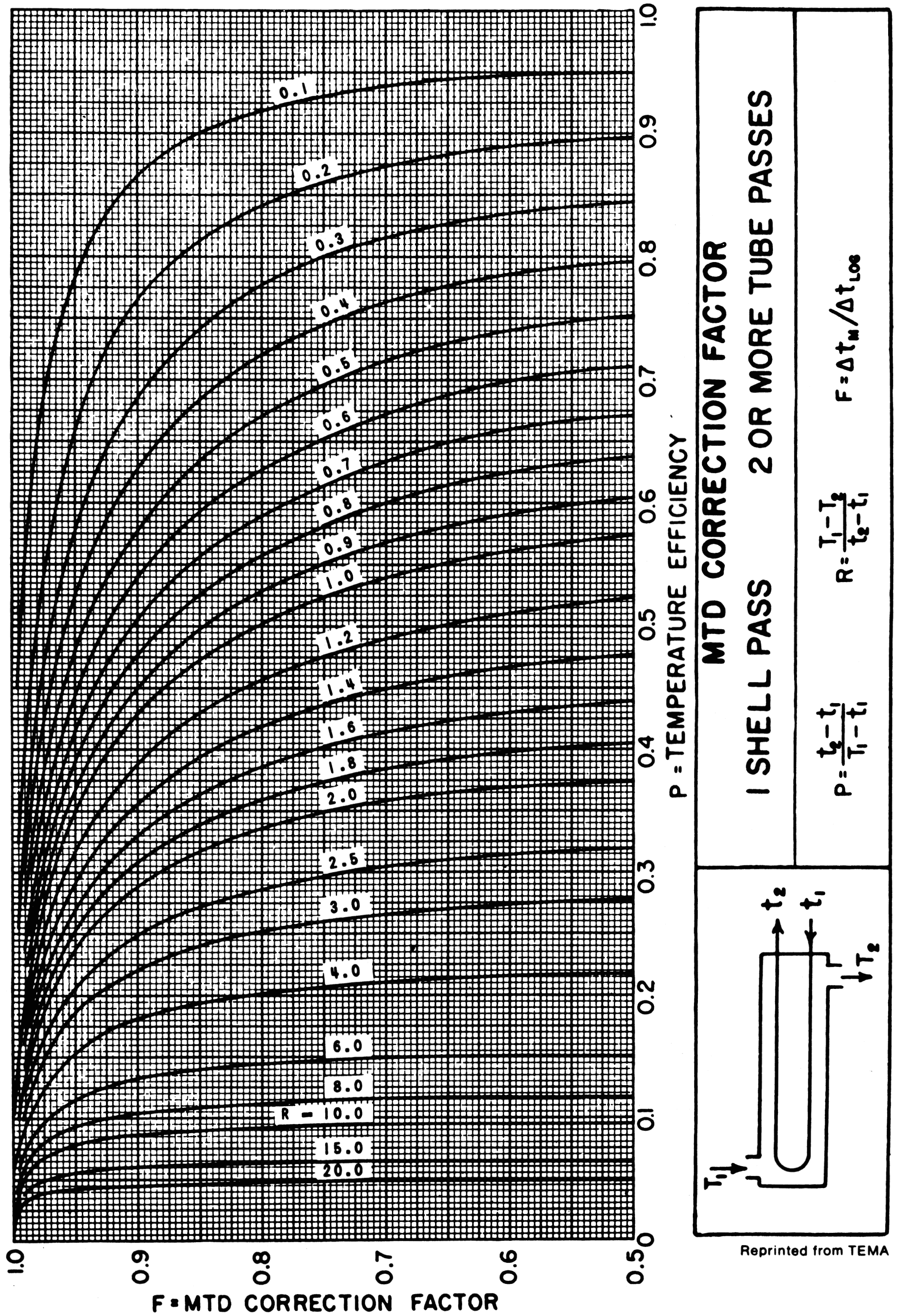


FIGURE 2 — BULLETIN 1

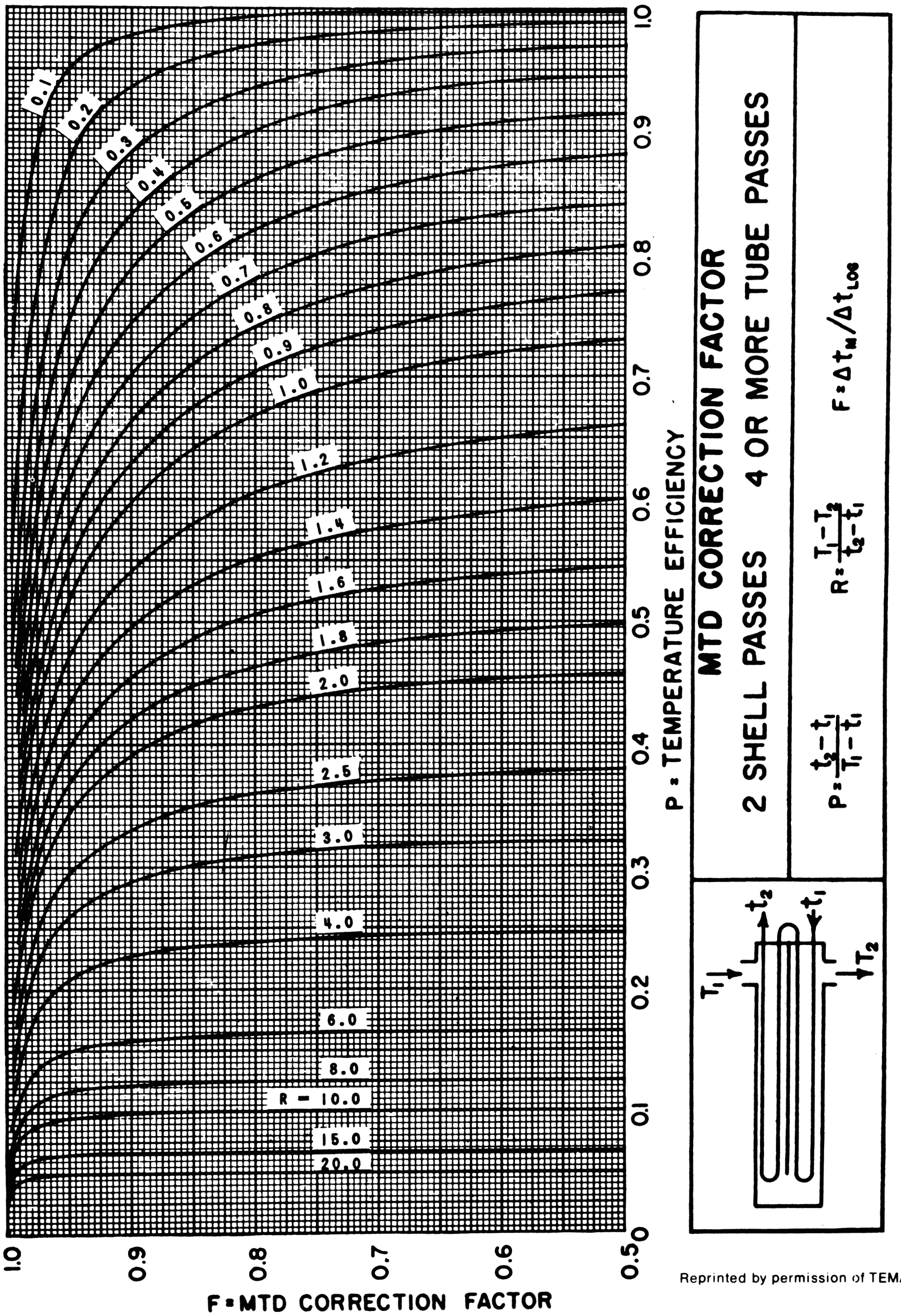


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FIGURE 3 — BULLETIN 1

FACTS ON FOULING FACTORS

How much fouling factor shall we specify? — a common question when heat exchanger specifications are being prepared. A fouling factor (fouling resistance) is really a safety factor to avoid reduced performance or too frequent cleaning. In most engineering applications safety factors are proportional to load and approximately proportional to costs. This is not true with fouling factors. A twenty-five per cent increase in fouling factor may result in a fifty per cent or more increase in the cost of the heat exchanger.

The data presented below provide some guidelines for determining the effect of fouling factors. From here on we use the term fouling resistance since it is truly a resistance to heat transfer.

Fouling resistances are applied to other heat transfer resistances to compute the overall heat transfer coefficient according to the following abbreviated equation:

$$\frac{1}{U_d} = \frac{1}{h_o} + \frac{1}{h_{io}} + r_w + r_f$$

U_d = service or design heat transfer coefficient

$\frac{1}{h_o}$ = film resistance outside tubes

$\frac{1}{h_{io}}$ = film resistance inside tubes (referred to outside surface)

r_w = tube wall resistance

r_f = sum of outside and inside fouling resistance (referred to outside surface).

When r_f is omitted from this equation, U_d becomes U_c, the clean, unfouled coefficient. Figure 1 is a plot of clean versus design coefficients at different fouling resistances. The equation for Figure 1 is:

$$\frac{1}{U_c} = \frac{1}{U_d} - r_f$$

Since U is directly proportional to heat transfer surface, the ratio of U_c/U_d represents the ratio of clean surface required to total surface including fouling allowance. The ratio is:

$$\frac{U_c}{U_d} = 1 + U_c r_f$$

The term $U_c r_f$, multiplied by 100, is the percentage excess surface. This is the surface above that needed clean to satisfy the added fouling resistance.

The significance of the ratio U_c/U_d and the percentage excess surface can be realized more quickly by tabulating some results. Table 1 shows the percentage of excess surface that must be added at various values of service coefficient, U_d, and fouling resistance, r_f.

TABLE 1 — BULLETIN 2

Service Coefficient	Fouling Resistance				
U_d	.001	.002	.003	.004	.005
20	2.0	4.2	6.4	8.7	11.1
40	4.2	8.7	13.6	19.0	25.0
60	6.4	13.6	22.0	31.6	42.9
80	8.7	19.0	31.6	47.1	66.7
100	11.1	25.0	42.9	66.7	100.0
120	13.6	31.6	56.3	92.3	150.0
140	16.3	38.9	72.4	127.3	233.3

Table 1 has been divided into three groups, A, B, and C. In Group A the excess surface is below 10%. Most manufacturers will not size exchangers with less than about 10% excess surface regardless of the fouling factors. If a particular service falls in Group A, then it is appropriate to specify a fouling resistance that will provide at least 10% excess surface as the manufacturer will provide it anyway.

In Group B. the percentage of excess surface is within normal range. There is little reason to spend time in considering adjustments of specified fouling resistance unless there is reason to suspect that the service in question is quite dirty, or conversely, very clean.

Excess surface added in Group C becomes costly and warrants careful attention. Just the difference of one point in the specified fouling resistance will have a considerable effect on the size and cost of the equipment. And very often a fouling resistance on the high side leads to lowered overall efficiency. Velocities are lower, pressure drop is not used efficiently, and the result is a greater degree of fouling that would have occurred had a lower resistance been specified at the outset.

These are some of the questions that are worth reviewing if a particular service falls in the Group C category:

1. Is there a previous record of severe fouling in this service?
2. If there is a previous record, was it attributable to the exchanger design, service conditions, flow characteristics, temperatures? Can these be avoided in the case in question?
3. Does the reduced frequency of cleaning (assuming it is reduced) and associated costs warrant the added cost of excess surface for fouling?
4. Can the equipment be removed from service without upsetting production?

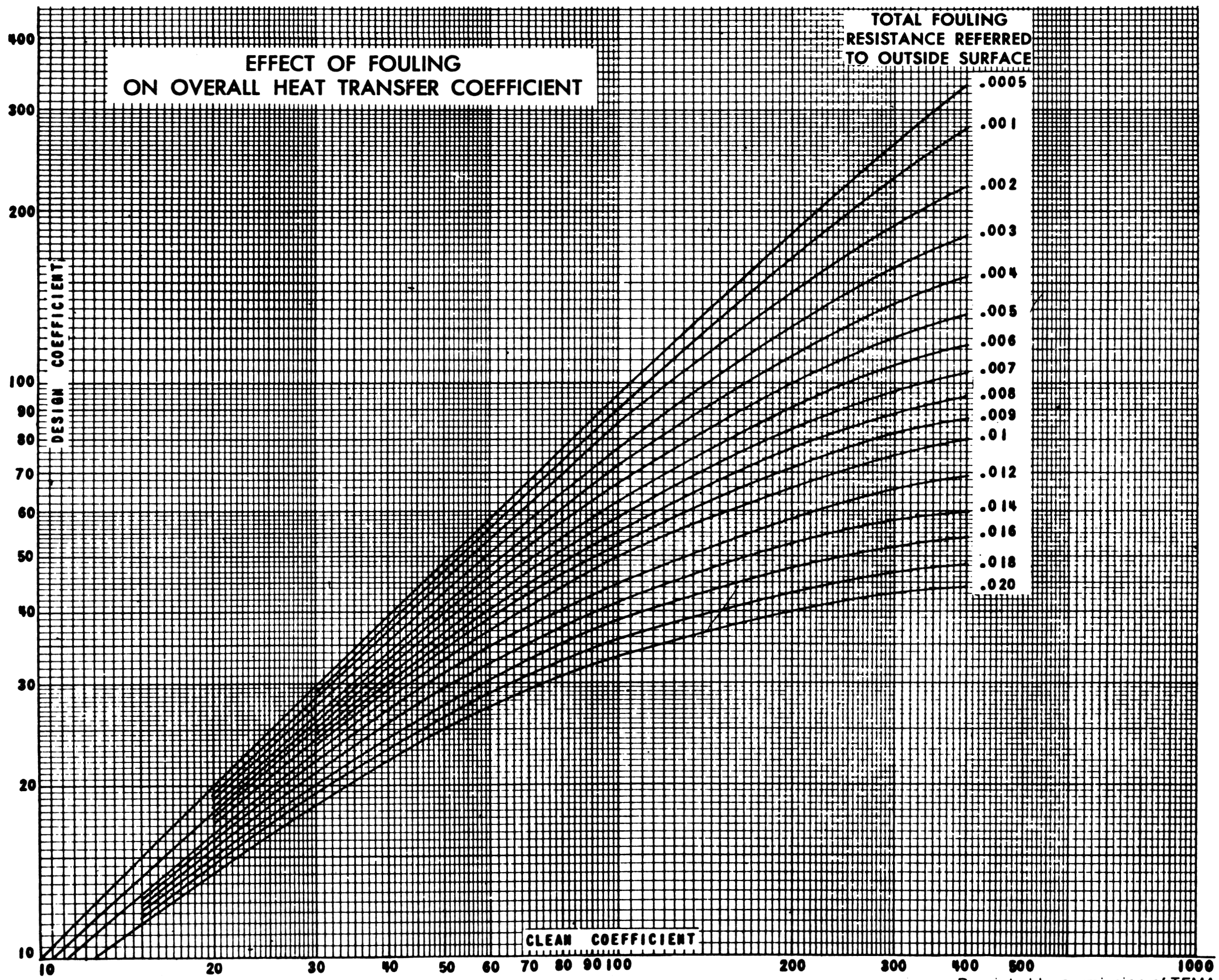

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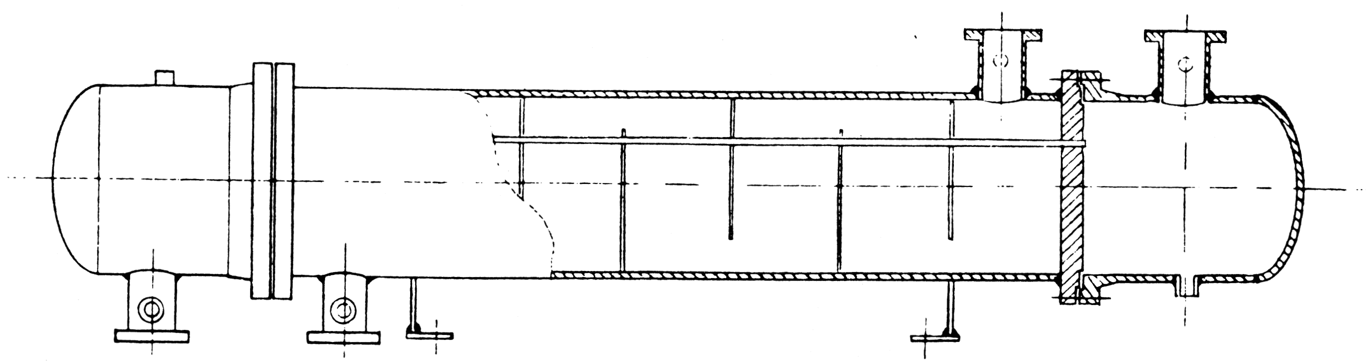
FIGURE 1 — BULLETIN 2

FIXED TUBESHEET EXCHANGERS DON'T STRIKE BACK
. . . if they're used correctly

Of the several types of basic shell and tube exchanger designs available, the fixed tubesheet type is probably the most often incorrectly specified and incorrectly utilized. The result: some painful and costly experiences, followed by condemnation of the type. In fact, some specifications are very carefully written to exclude fixed tubesheet design. Yet in many services, the fixed tubesheet exchanger is the logical choice. And it costs less, too. Some guidelines for determining these services are given below.

Construction Features

The following sketch illustrates basic features of the design.



Options are available for bonnet or flat covers at either end. Flat covers are generally specified when frequent tube cleaning is expected. Single pass tube side is illustrated. Multipass is available with two, three, four, five passes or more. Shell expansion joint (not shown) may or may not be required.

Pluses and minuses of the fixed tubesheet design are more apparent when compared with a conventional floating head heat exchanger.

Advantages:

1. Permits single pass flow tube side without the use of packed joints, internal expansion joints or other special construction features. True countercurrent flow is readily achieved — particularly advantageous when there is a temperature crossover between streams. In some cases, one fixed tubesheet exchanger may do the job when two or more floating head units would be needed.
2. Construction features are simple with fewer parts, a minimum of bolted joints, and no internal bolted joints to cause possible inter-stream leakage.
3. Straight tubes, easily cleaned inside and quickly accessible through two bolted, external joints.
4. Better flow characteristics on the shell side — fewer "dead" areas, less tubefield bypass area.
5. Tubes are effective stays for the tubesheet, thus usually reducing required tubesheet thickness. This advantage is most applicable where shell side pressure is high relative to tube side pressure and is always above tube side pressure under any operating condition.
6. Generally less expensive. Besides having fewer parts, size is slightly smaller and construction much simpler.

Disadvantages:

1. No access to shell side for mechanical cleaning. Chemical cleaning is sometimes unsatisfactory. Limited access for inspection.
2. Difficult and expensive to maintain if corrosion occurs. Retubing in place is expensive, time-consuming and sometimes impractical, as is exchanger removal for repair. No spare tube bundles, of course.
3. May require use of a relatively large expansion joint in the shell to avoid high stresses resulting from tube and shell differential thermal expansion. Cost of the expansion joint may quickly wash out price advantage over other types.

Avoid Fixed Tubesheet Design When . . .

a) With materials chosen, there is a question of corrosion occurring with either flowing stream.

b) The stream in the shell may be fouling or may form deposits — under any predictable operating circumstance — unless fouled surfaces can readily be cleaned chemically.

c) A shell expansion joint is required and must operate under severe conditions of pressure or temperature, or both. In fact, if there is any doubt about the integrity of an expansion joint, another exchanger type should be considered.

Consider Fixed Tubesheet Design When . . .

a) Shell fluid is non-fouling, clear of solids, and both shell and tube fluids are known without question to be non-corrosive to materials selected.

b) Shell fluid has high heat transfer characteristics relative to the tube side fluid. This often eliminates the need for an expansion joint. Steam, non-fouling water are good examples. Even so, an expansion joint may be needed if there are dissimilar metals involved.

c) True countercurrent flow is required. However, watch flow quantities in this case. Low tube side velocities may generate uneconomical heat transfer rates. Allow the manufacturer some latitude in selecting tube size and length, if possible.

Specifications:

As with other exchanger types, the manufacturer needs the customary performance data and material requirements. When specifying fixed tubesheet design, data should also be provided covering the most severe operating condition for differential expansion between shell and tubes. This may be a startup condition, a shutdown condition, an operational upset, or loss of shell side flow. When established at the outset, responsibilities are clear cut. In cases where differential expansion appears significant and an expansion joint may be needed under adverse conditions, it is wise to request alternate quotations for other types.

PERMIT DESIGN FREEDOM — YOU'LL GET A BETTER EXCHANGER

The "Exchanger Specification Sheet" is a commonly used means of communicating a need for equipment to a manufacturer. It serves a valuable purpose in presenting data needed for design purposes. The same type of specification sheet also provides a means for the manufacturer to submit data in return about the exchanger selection he has made to meet the required need. This sounds good. But there is a basic fault. Historically, the exchanger user tends to over-specify. As over-specification approaches extremes, the manufacturer's freedom to develop designs decreases to the point where engineering ability becomes secondary to production capability. In the long run, over-specification is costly to both user and manufacturer. The discussion below is intended to assist in preparation of a specification that will avoid this problem.

SPECIFICATION TYPE

The Tubular Exchanger Manufacturers Association, Inc. (TEMA), has presented a form for specification purposes which is used by most purchasers and manufacturers, perhaps with variations. The form used by Yuba is shown on page 4. A ninimum completion of data on this form will challenge the designer to select what he considers the most economical equipment. A maximum completion of data is very restrictive to his selection. Unfortunately, the use of a form such as TEMA suggests, reduces the specification to the mechanics of filling in spaces, with little thought to the results. If, in the completion of the specification, the user gives some thought to the "maximum" and the "minimum" that can be provided, then he can produce a specification that will give him the most chance of an optimum result. He will effectively employ the manufacturer's designer to achieve this end.

MINIMUM SPECIFICATION

An exchanger can be designed if the following minimum information is available:

1. Flowing quantities, both streams.
2. Terminal temperatures of the two streams.
3. A description of the flowing streams if the physical properties are well known, or alternatively, physical properties: specific heats, densities, viscosities, thermal conductivities, and sometimes properties such as latent heat, molecular weight, and pressure-volume-temperature relations if a change of phase occurs in the exchanger.
4. Allowable pressure drop and operating pressures.
5. Minimum Design Metal Temperature (MDMT). This is defined as the minimum metal temperature at full design pressure determined through consideration of the lowest operating temperature, operational upsets, auto refrigeration, atmospheric temperature and any other sources of cooling.

With this minimum information, the exchanger can be designed for the required performance. If the designer does some optimizing, the cost will be minimum, too.

MAXIMUM SPECIFICATION

The "maximum" specification reduces the freedom available to the designer to the extent that he selects only the number of tubes, number of tube passes, establishes shell diameter based on these, and determines shell side baffle and flow arrangement. Excluding details covering fabrication, inspection and testing, the maximum specification (divided into Categories A and B) will provide:

Category A:
- a) All of the data in the "minimum" specification.
- b) Design pressures and temperatures.
- c) Fouling factors.
- d) Materials of construction and corrosion allowances.
- e) Construction requirements, codes, standards, etc.
- f) Tube gauge.

Category B: g) Type of exchanger construction permitted — floating head, U-tube, fixed tubesheet.
h) Diameter and length of tubes.
i) Stream location (shell or tube side).
j) Tube layout and pitch.
k) Connection sizes.

OPTIMUM SPECIFICATION

The exchanger user customarily wants to specify items falling in Category A because of his exclusive and intimate knowledge of the service requirements. However, if he is able to offer some freedom in Category B, within limits, then he will increase his chances of obtaining the most economical exchanger selection that will still meet his particular needs. The freedom which might be offered will be more apparent from the following comments about each of the items in Category B:

g) Construction Type: Three basic types are available, with many variations — floating head, fixed tubesheet, U-tube.

g.1 If floating head is the indicated choice, consider fixed tubesheet (clean, non-corrosive fluid shell side), with or without shell side and tube side streams reversed.

g.2 If fixed tubesheet is the indicated choice, consider U-tube with streams reversed and tube layout altered from triangular to square.

g.3 If U-tube is the indicated choice, consider fixed tubesheet with streams reversed.

g.4 If more than one shell is required (in series) because of a temperature crossover condition, consider permitting a two-pass shell.

g.5 With floating head construction and one tube pass, consider use of a packed joint rather than an expansion joint at the rear end.

g.6 Removable channel cover is often specified without thought to cleaning frequency. Consider bonnet type removable channel. Similarly, non-removable shell covers are often satisfactory with floating head pull-through construction.

h) Diameter and length of tubes: Selection of tube dimensions is frequently affected by the user's practice in inventorying replacement tubes. The extension of these practices to alloy tubes without careful thought can create unneeded expense.

h.1 Consider use of ⅝″ tube diameters where internal cleaning is not required. (⅝″ is considered the minimum commercial size.).

h.2 Permit use of larger tubes than are specified. Pressure drop may be controlling.

h.3 Limit U-tube and floating head tube bundle length based on bundle handling facilities.

h.4 Limit fixed tubesheet tube length only on available space or cleaning needs.

h.5 Allow tubes shorter than the specified length, or make the tube length optional with a maximum.

i) Stream location: If streams can be switched between exchanger sides, even with change of tube layout (usually from triangular to square) this provides a valuable freedom to the designer. The need for material changes under these conditions is understood.

j) Tube layout: Normally, tube layout will be specified based on cleaning requirements, and could not, for example, be changed from square to triangular. Usually there would be no objection to changing from triangular to square or to increasing pitch between tubes if this is necessary owing to pressure drop restrictions.

k) Nozzle size: Since the exchanger manufacturer must guarantee pressure drop as well as thermal performance, recommended practice is to permit the manufacturer to size nozzles.

Quite clearly, the above comments are not all-inclusive; they are intended to be thought-provoking. Most shell and tube heat exchangers will be cylindrical, cigar shaped. You visualize them this way. But if one comes along that is square, oblong, or elliptical, you would not object, if it meets all of your requirements.

ALTERNATES

When suggestions are advanced that added freedom should be permitted in design, the first thought may be to request alternative offerings. Under circumstances where an evaluation must be made, this is justified. In most cases, the selection of the most economical unit must be left to the manufacturer who knows his costs and the relative costs of alternate designs. Requests for extensive alternates can be extremely burdensome to both user and manufacturer.

SPECIFICATION OF PHYSICAL PROPERTIES

Far too frequently, the most significant physical properties of the streams in a heat exchanger are omitted from the specification. The two most important properties are viscosity and thermal conductivity. Specific heat and density are also important but are usually known with reasonable accuracy. The results produced by an exchanger designer depend upon the completeness of the data he is working with. For example:

A. A ten percent error in thermal conductivity will produce a seven percent error in calculation of an individual film heat transfer coefficient.

B. A ten percent error in viscosity will produce a four percent error in calculation of an individual film heat transfer coefficient.

C. A combination of errors in viscosity-thermal conductivity, each in the direction of reducing the coefficient, will produce roughly the same percentage change as the percentage of error in the computed heat transfer coefficient. Errors of this magnitude may well justify some laboratory determination of these properties if they are uncertain.

YUBA HEAT TRANSFER

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SHELL AND TUBE SPECIFICATION SHEET

1	CUSTOMER:		REFERENCE NO.:	
2	ADDRESS:		PROPOSAL NO.:	
3	PLANT LOCATION:		ENGR: DATE:	REV.
4	SERVICE OF UNIT:		ITEM NO.	
5	SIZE: TYPE:	POSITION:	CONNECTED IN: PARALLEL	SERIES
6	SURF/UNITS EFF. SQ FT	SHELLS/UNIT	SURF/SHELL	EFF. SQ FT
7	PERFORMANCE OF ONE UNIT			
8	FLUID ALLOCATION		SHELL SIDE	TUBE SIDE
9	FLUID NAME			
10	FLUID QUANTITY, TOTAL	#/HR.		
11	VAPOR (IN)	#/HR.		
12	LIQUID	#/HR.		
13	STEAM	#/HR.		
14	FLUID VAPZD./COND.	#/HR.		
15	NON-CONDENSABLES	#/HR.		
16	TEMPERATURE (IN/OUT)	DEG F.		
17	SPECIFIC GRAVITY			
18	VISCOSITY - LIQUID	Cp.		
19	MOLECULAR WEIGHT-VAPORS			
20	MOLECULAR WEIGHT, NONCONDENSABLE			
21	SPECIFIC HEAT	BTU/LB DEG F.		
22				
23	OPERATING PRESSURE	Psig		
24	VELOCITY	FT/SEC		
25	PRESSURE DROP, ALLOW/CALC.	PSI	/	/ CALC
26	FOULING RESISTANCE (MIN)			
27	HEAT EXCHANGED		BTU/HR;MTD (CORRECTED)	DEG F.
28	TRANSFER RATE, SERVICE		CLEAN	BTU/HR SQ. FT.DEG. F.
29	CONSTRUCTION OF ONE SHELL			
30	DESIGN/TEST PRESSURE	PSIG	/	/
31	DESIGN TEMPERATURE / MDMT	DEG F	/	/
32	NO. PASSES PER SHELL			
33	CORROSION ALLOWANCE	IN.		
34	CONNECTIONS (SIZE & RATING)	IN		
35		OUT		
36		INTERMEDIATE		
37	TUBE NO. OD IN.	AVG BWG	STR.LGTH-IN.	PITCH
38	TUBE TYPE		MATERIAL SPEC.	
39	SHELL ID-IN.	MATL	SHELL COVER	
40	CHANNEL OR BONNET		CHANNEL COVER	
41	TUBESHEET-STATIONARY		TUBESHEET-FLOATING	
42	FLOATING HEAD COVER		IMPINGEMENT PROTECTION	
43	BAFFLES-CROSS	TYPE	DIAM/AREA % CUT	SPACING c/c
44	BAFFLES-LONG		TYPE	
45	SUPPORTS-TUBE	U-BEND	TYPE	
46	BYPASS SEAL ARRANGEMENT		TUBE-TUBESHEET JOINT	
47	EXPANSION JOINT			
48	GASKETS-SHELL SIDE		TUBE SIDE	
49	-FLOATING HEAD			
50	CODE REQUIREMENTS	ASME SECTION VIII	TEMA CLASS	
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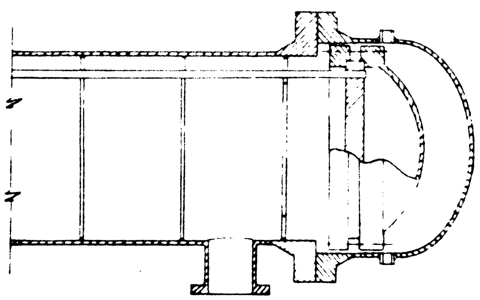
FLOATING HEADS — SPLIT RING OR PULL THROUGH?

Floating head shell and tube exchangers may be provided with either pull-through or with split-ring floating heads. The Tubular Exchanger Manufacturers Association, in the latest issue of TEMA standards, designates split-ring and pull-through floating heads as Types S and T, respectively. The description ends with this letter designation. No discussion is included about the two basic types nor comments about the relative value of either. Most exchanger manufacturers have standardized on one of the two types depending on their particular feelings. However, all of them manufacture the pull-through type because there are some circumstances when the split-ring type cannot be used. In some cases this quasi-standardization has led to suggestions that one type is superior to the other. This is obviously not true as both types continue to be manufactured. Points discussed below will assist the user in selecting one or the other type for his service requirements.

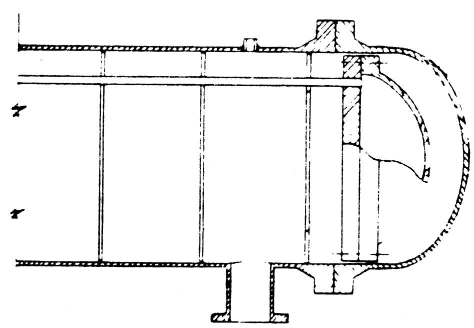
Construction Features

Figures 1 and 2 illustrate the two floating head types.



SPLIT-RING FLOATING HEAD

FIGURE 1 — BULLETIN 5



PULL-THROUGH FLOATING HEAD

FIGURE 2 — BULLETIN 5

Construction Differences

Split-Ring Type:

1. The construction type is always evident from an external view of the exchanger because the shellcover diameter is larger than the shell diameter, usually by about three or four inches.
2. The floating tubesheet extends beyond the face of the rear shell flange by a distance sufficient to accommodate the split rings and bolting. The shellcover is always removeable.
3. If tube bundle removal is required, the shellcover, split rings, and floating head must be removed first.
4. The construction type permits tubes to be close to the inside diameter of the shell. Thus, for a given number of tubes, the shell diameter is about two inches smaller than with pull-through type.

Pull-Through Type:

1. The shellcover and the shell are the same diameter. The rear face of the tubesheet customarily extends beyond the face of the rear shell flange but may be within the shell proper. The shellcover need not be removable.
2. The floating tubesheet is drilled with bolt holes to accommodate similar drilling for the floating head cover and is "through-bolted" similar to most flanged joints.
3. Since sufficient space must be provided around the periphery of the floating tubesheet to accommodate the bolting, the shell diameter must be larger than with split-ring construction for the same number of tubes.

Cost Differences

The split-ring type of construction is looked upon by many exchanger users as the lesser expensive of the two types. Perhaps because of this opinion, it is viewed as the less desirable. But, in fact, the pull-through type is sometimes less expensive. And the split-ring type is more efficient in some circumstances. While hard and fast rules cannot be made, some guidelines can be drawn:

Split-ring construction is generally more expensive than pull-through construction when:

a) Design pressures are on the high side — say over 300 PSIG, and at the same time the exchanger will be medium size or larger — say over 2,500 square feet. Under these circumstances the rear shell flange geometry becomes unwieldy and the split rings and floating head flange unduly heavy. In fact, split-ring construction should not be used for over 600 PSIG tube side design pressure due to the increased possibility of leakage problems at the floating head joint.

b) Floating head construction is needed but removable shellcover is not required. Many users overlook the savings possible by elimination of the removable shellcover.

c) The need for alloy construction on the shell side requires that floating head, split rings and bolting be alloy also.

d) Large shell inlet and outlet nozzles are required. These large nozzles will eliminate many more tubes (to provide for adequate entrance and exit areas) from a split-ring type than from a pull-through type.

Pull-through construction is generally more expensive than split-ring construction when:

a) Alloys of high cost are used for tubebundle parts including tubesheets, baffles, and floating head. The smaller shell diameter with split-ring construction pays off when expensive materials are involved.

b) In small sizes where the use of bypass preventers (sealing strips) is necessary to block bypass flow areas.

Exchanger Maintenance

From a maintenance standpoint the pull-through type rates a distinct edge. Tube bundle removal can be accomplished without breaking the shellcover joint.

Performance

Current studies of shell side flow and heat transfer point up the significance of leakage paths on both pressure loss and heat transfer. Leakage paths stem from three sources:

1. Leakage through tube-hole-to-tube-clearance in crossflow baffles.
2. Leakage around baffles through baffle-to-shell clearances.
3. Bypassing of the tube field because of peripheral clearances between tubes and shell.

Of these three leakage sources the most damaging to efficient heat transfer is Item 3.

Clearances between outermost tubes and shell, by the nature of the two designs, are greater in pull-through floating head exchangers. Figure 3 is a plot showing the percentage of bypass area versus shell diameter for both split-ring and pull-through types. In a 12″ shell, for example, the leakage area is 60 percent of the total crossflow area for pull-through type while only 30 percent for split-ring type. In a 38″ shell these figures are much lower at 31 percent for pull-through type and 18 percent for split-ring type.

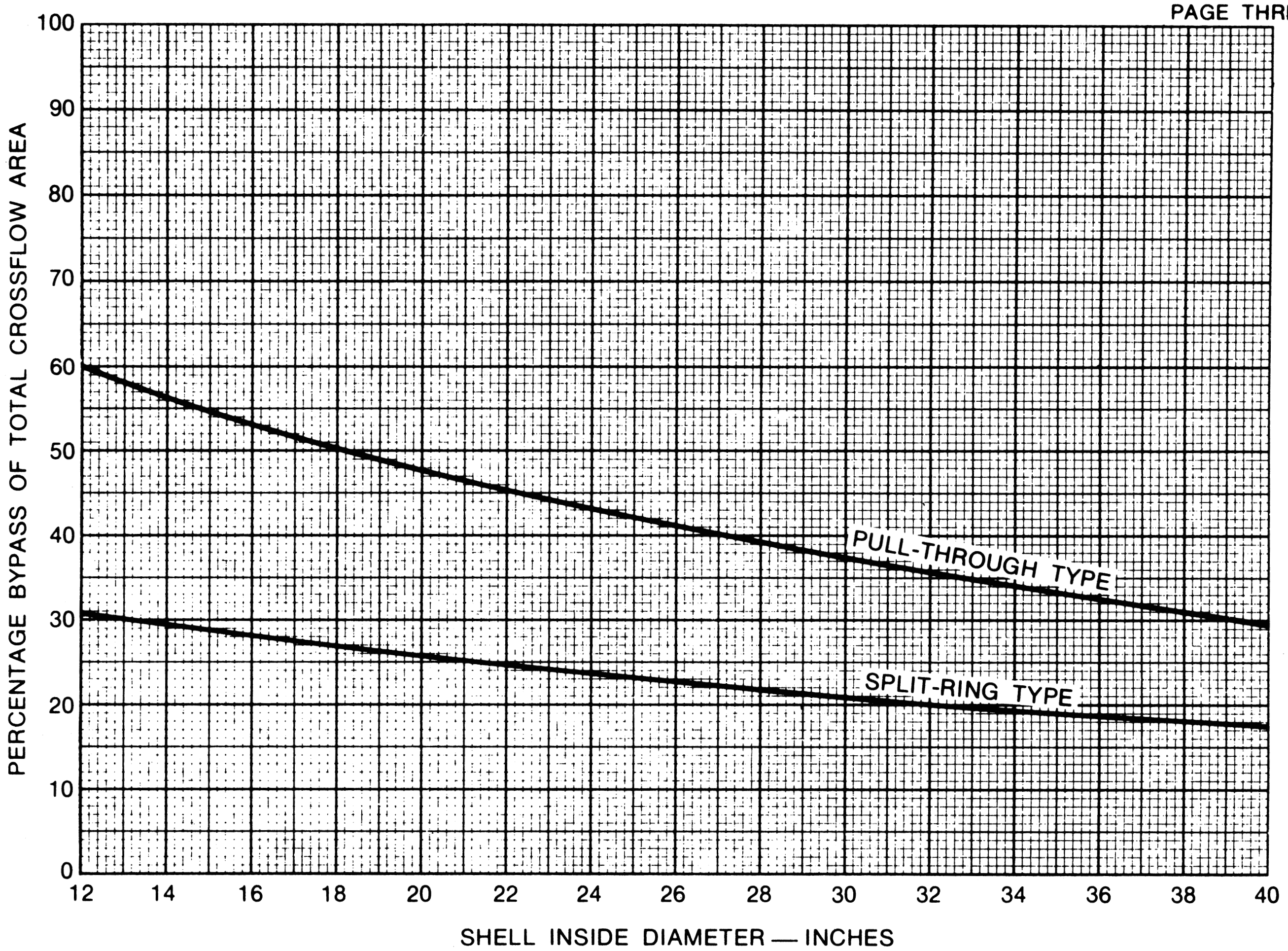


FIGURE 3 — BULLETIN 5

Figure 3 is based on triangular tube layout. However, the differences are still significant with square pitch, being 46 percent versus 20 percent for a 12″ shell, and 21 percent versus 11 percent for a 38″ shell.

Reliable manufacturers are aware of the efficiency loss caused by excessive bypass area and install sealing strips to block flow in these areas. But the effectiveness of such methods decreases as the amount of area to be blocked increases. A small diameter shell with pull-through floating head may be less efficient than a split-ring type in spite of the use of sealing strips to block bypass areas. This efficiency evaluation is also affected by the number of tubes dropped out of a tube field at the shell inlet and outlet nozzles. The loss of these tubes (the number of tubes eliminated is dependent on the nozzle sizes) creates flat spots at the top and bottom of the tube field which means more bypass area.

Suggested Approach

In specifying exchanger types consideration should be given to two points when there is a choice between either split-ring or pull-through types. First, cost difference points should be reviewed as outlined above. Second, the following:

a) Permit the manufacturer to determine whether the exchanger should have split-ring or pull-through type floating head.

b) In any event, insist that bypass areas be properly blocked, whether in pull-through or in split-ring type.

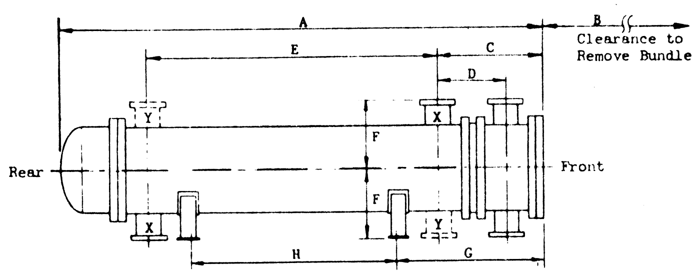
KEY TO OUTLINE DRAWINGS

Once an order has been placed for a shell and tube exchanger, the most important need from the purchaser's viewpoint is the outline (dimensional) drawing. Until this is available, final details of civil, structural, and piping work necessary to integrate the exchanger into the system cannot be completed. Preparation of the outline drawing by the manufacturer requires complete detailed design of the exchanger — a time-consuming job. Usually, the outline drawing will be available two to three weeks after order. Sometimes this period extends to a month or more. During this time the purchaser can complete preliminary layout work, estimating exchanger dimensions by reference to the exchanger specification sheet.

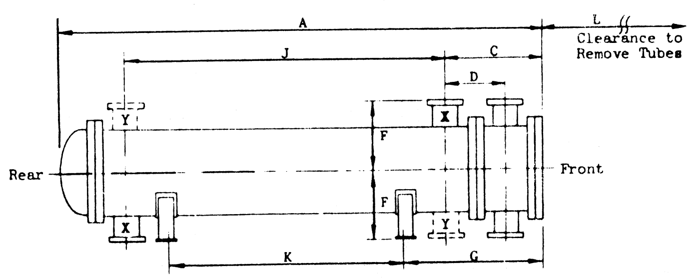
The data on the exchanger specification sheet which establish the general configuration of the exchanger are: (1) the exchanger type, (2) the tube length, and (3) the shell diameter. With these and a knowledge of permissible nozzle arrangements and orientations, a quick estimate of exchanger dimensions can be made.

Important Dimensions

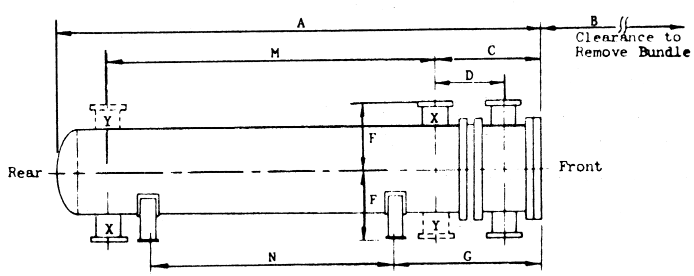
The sketches below illustrate important layout dimensions for floating head, fixed tubesheet, and U-tube exchangers.



FLOATING HEAD EXCHANGER (TEMA AES OR AET)



FIXED TUBESHEET EXCHANGER (TEMA AEM)



U-TUBE EXCHANGER (TEMA AEU)

FIGURE 1 — BULLETIN 6

Estimating Dimensions

Using tube length and shell diameter, dimensions can be approximated by formulas below. The abbreviation T.L. stands for tube length; S.D. for shell diameter.

Formula	Possible Error
$A = T.L. + \frac{S.D.}{2} + 26''$	$+12''$ $-6''$
$B = T.L.$	—
$C = \frac{S.D.}{2} + 24''$	$\pm 4''$
$D = \frac{S.D.}{4} + 19''$	$\pm 4''$
$E = T.L. - \frac{S.D.}{2} - 20''$	$\pm 8''$
$F = \frac{S.D.}{2} + 9''$	$\pm 2''$*
$G = \frac{A}{4} - 1''$	$\pm 3''$
$H = \frac{A}{2}$ (Rounded off to next smaller foot dimension)	*
$J = T.L. - 24''$	$\pm 8''$
$K = \frac{A}{2}$ (Rounded off to next smaller foot dimension)	*
$L = T.L.$	—
$M = T.L. - 24''$ (Not valid if a nozzle is located beyond end of bundle)	$\pm 8''$
$N = \frac{A}{2}$ (Rounded off to next smaller foot dimension)	*

(*) Span between supports and projection of supports can vary considerably depending on standards established by the manufacturer.

Note: For U-tube units, manufacturers customarily show the straight length of the tubes (to the tangent point of the U-bends) as a part of the size designation. For U-tube units the tube length (T.L.) is this straight length plus one-half the shell diameter.

These formulas are based on exchangers with 8″ nozzles. Corrections may be applied for nozzles of different size as follows: (Note that CNS stands for the larger channel nozzle size; FSNS stands for the front shell nozzle size; RSNS stands for the rear shell nozzle size).

Dimension A — Add: $CNS - 8''$
Dimension C — Add: $(CNS - 8'') + \frac{1}{2}(FSNS - 8'')$
Dimension D — Add: $\frac{1}{2}(FSNS - 8'') + \frac{1}{2}(CNS - 8'')$
Dimensions E, J, M — Subtract: $\frac{1}{2}(FSNS - 8'') + \frac{1}{2}(RSNS - 8'')$

Unless there is a large difference in size, these corrections are not required.

Other Exchanger Types

Three basic exchanger types are illustrated on Page 1. Flat, removable channel covers (TEMA A) are shown for each type. Dimensions are very nearly the same if bonnet covers (TEMA B) are used, providing nozzles are arranged as shown (perpendicular to the exchanger horizontal axis).

If fixed tubesheet construction is used with a single tube pass rather than multiple passes as illustrated, the overall length, A, is increased by 16″. The distance between front and rear channel nozzles is approximately $J + 2 \times D$.

Chillers, vaporizers, and reboilers represent a special case if they are kettle type (TEMA K). The overall length will apply if S.D. is the larger diameter shown on the exchanger specification sheet. Add surge section length to this if applicable. There is considerable latitude for nozzle location except that the outlet vapor nozzle (shell side) should normally split the tube bundle.

In exchanger types with divided or split flow (single inlet and two outlets or vice versa, and single inlet and outlet centrally located), the dimension between the channel nozzle centerline and the single shell nozzle is $D + \frac{1}{2}E$ (or J or L).

Formulas are not applicable to special exchanger types and to high pressure exchangers. However, data provided by the manufacturer is usually sufficient to determine at least limiting dimensions.

Space Requirements

The space "envelope" needed for the exchanger can be developed from the formulas:

Overall Length Required = $A + B$ (or $A + L$ with fixed tubesheet type)
Overall Height Required = $2 \times F$ (Excluding foundation height)
Overall Width Required = $2 \times F$ maximum ($2 \times F - 8''$ average)

Nozzle Location and Orientation

In many cases there is more than one choice of nozzle location or orientation. The manufacturer's outline drawing will show suggested location. A knowledge of the options available will permit the purchaser to either approve the manufacturer's suggested location or to select an arrangement more suited to his needs.

Shell Nozzle Location and Orientation

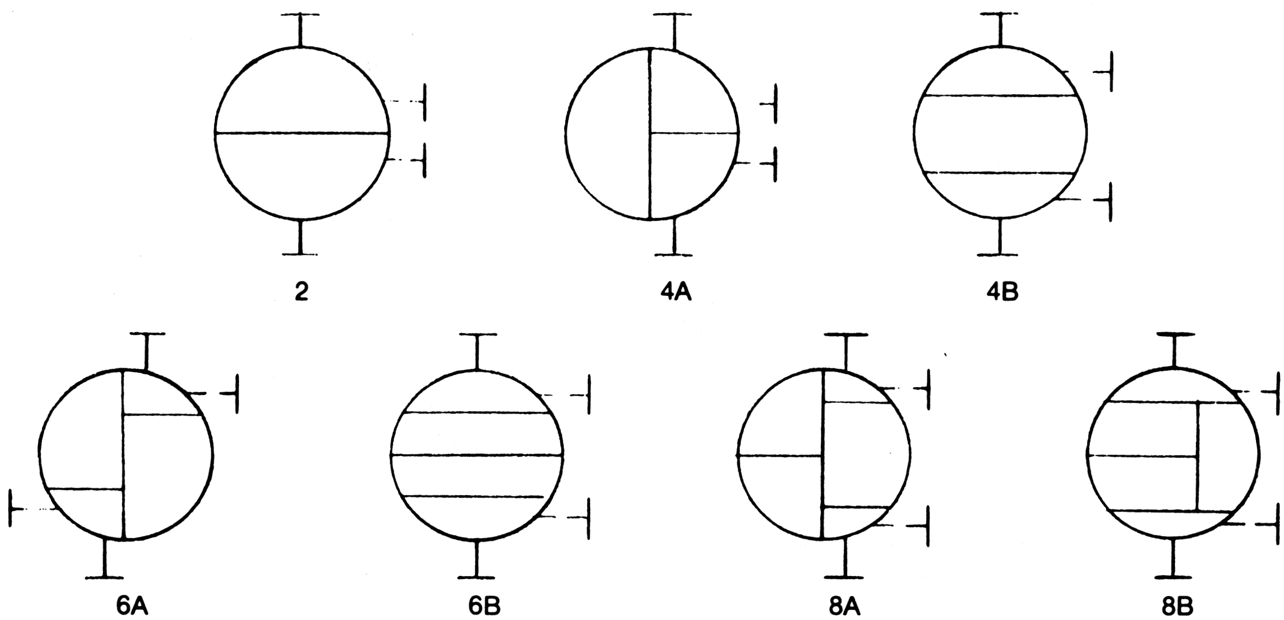
With a single pass shell, nozzles can be located either in positions X-X or positions Y-Y as shown on the sketches.

With a two pass shell, nozzles are on the same centerline, located at the front of the shell.

Normal orientation of shell nozzles is top and bottom. When no change of phase occurs as with gases or liquids, nozzles, can be side mounted, or inlet may be top or bottom as desired. When condensation occurs, the inlet is at the top and outlet at the bottom. When vaporization occurs, the inlet is at the bottom and the outlet at the top.

Tubeside Nozzle Location and Orientation

Nozzle arrangements with two to eight passes are as follows:



Nozzle orientations shown dashed are also available, but less common than those shown. Additionally, the orientations shown above may be reversed to provide mirror images. Many variations of nozzle arrangement are possible with two tube passes.

Arrangements 4B, 6B, and 8B have fewer tubes (or heat transfer surface) for a given shell diameter than their counterparts, 4A, 6A, and 8A. For this reason their use is often restricted to exchangers having appreciable temperature differences, pass to pass, or to other special cases.

With condensation or vaporization, nozzles are normally located at, or as close as possible, to top and bottom centerlines. When no change of phase occurs, the above orientations may be rotated 90 degrees. Nozzles are usually installed with flanges either horizontal or vertical.

The use of a two-pass shell is more restrictive on channel nozzle arrangement. The longitudinal baffle dividing the shell must be horizontal. This limits 2, 4, 6B, and 8-pass arrangements to those illustrated above or to mirror images. And the 6A pass arrangement must be rotated 90 degrees.

Summary

Guidelines above can be used with reasonable accuracy in predicting dimensions that will be produced by the manufacturer on his outline drawings. The formulas should not be used blindly with the expectation that the results will be totally accurate and that the manufacturer, therefore, should be able to comply with estimated dimensions. The purchaser is in a prime position to know what nozzle locations, orientations, etc., best suit his needs. Very often, there is a tendency to accept the manufacturer's suggested arrangement as final and unalterable. This should be the case only where dimensions are concerned. Where nozzle location and orientation are concerned, the purchaser should be aware of the options available and exercise them to his advantage.

HEAT EXCHANGER LEAK TESTING

Shell and tube heat exchangers are special kinds of pressure vessels. As such they are usually built and tested according to TEMA Standards and the ASME Code. The ASME Code is essentially a set of rules developed for safety purposes. As a final step in qualifying an item of equipment to the Code, a hydrostatic (water) pressue test is required. Successful passing of this test qualifies the equipment for a Code stamp (assuming other Code requirements for design and construction have been met previously).

The Code has become so ingrained in our thinking that a paradox has developed: The purchaser of equipment **assumes** a condition of structural soundness. The ASME Code requires proof by test. The purchaser looks upon the pressure test as his guarantee of leak tightness. The Code looks upon the pressure test as a measure of safety. The result: structurally sound equipment, but not necessarily leak tight in the intended service.

Would you accept a room temperature hydrostatic test as satisfactory proof that leakage will not occur when the equipment is operated at 2,200 PSIG and 900°F? This is an extreme example. But many purchasers *will* accept a hydrostatic test. They may require a number of quality control measures during fabrication which far exceed Code requirements. Yet the final hydrostatic test is their proof of leak tightness.

Only the purchaser is in a position to specify what testing is to be done beyond the basic Code requirement. The manufacturer cannot include special testing if he is to maintain his competitive position. Yet the purchaser seems often to be unaware of what constitutes an adequate test for his service needs. And equally unaware of the costs involved. The discussion below describes some of the tests available.

Leak Testing Quantitatively

The determination of what leak detection method best suits the service needs depends upon a knowledge of the sensitivity of different methods available. A given size defect may be tight to a water test yet leak a considerable quantity of gas under equal test conditions. The quantity of fluid a defect will pass is dependent on the size of the defect, the fluid pressure across the defect, and the characteristics of the fluid. Gases, particularly those of low molecular weight such as hydrogen or helium will leak more readily than air or nitrogen. And even heavy gases will leak more readily than liquids. Units for leakage rates are based on gas flow. Conversions are made to account for differences in gas composition and for differences in pressure or temperature — so that a comparable base is established.

The physical measure of leakage is in rather unusual units:

$$\frac{\text{Torr} \times \text{liters}}{\text{second}}$$

The unit "Torr" is a measure of pressure slightly less than one millimeter of mercury. This unit of leakage is equivalent to more easily understood units as follows:

$$1 \frac{\text{Torr} \times \text{liters}}{\text{second}} = 1.32 \frac{\text{Cubic Centimeters}}{\text{second}} = \frac{0.0000493 \text{ Cubic Feet}}{\text{second}}$$

It would appear from these units that the unit Torr × liters/second is rather small as a unit of leakage. Yet a defect that will produce one Torr × 1/second at a pressure difference of one Torr, would produce 0.123 cubic feet per second at a pressure difference of 500 PSI — a rather substantial leak.

Of more significance in leak testing is the smallest detectable leakage rate using different techniques, and the relative sensitivity of these tests:

Technique	**Smallest Detectable Leak** **Torr** × 1/second	Cubic Foot	Sensitivity
Hydrostatic Test	5×10^{-1}	0.47	1
Isotope Test	7×10^{-2}	3.35	7.1
Fluorescence Test	1×10^{-2}	23.5	50
Soap Bubble Test	1×10^{-3}	235	500
Chemical Test	8×10^{-4}	293	625
Halogen Test	1×10^{-5}	23,470	50,000
Helium Test	1×10^{-4} to 1×10^{-11}	2,347+	5,000+

Note that a simple soap bubble test is 500 times more sensitive than a hydrostatic test. Keep in mind though, that these sensitivities are at equal pressures. If a hydrostatic test is conducted at 1,000 PSI and an air test at 100 PSI, the relative sensitivity is reduced by a factor of 10. The sensitivity becomes 50 rather than 500.

Hydrostatic Tests of Heat Exchangers

Leak testing of shell and tube heat exchangers, even with simple hydrostatic tests, requires a much more extensive procedure than with the common category of pressure vessels. There are many possible leakage paths, both internal and external. Each tube joint is a possible leaker. Many flanged joints are present, each with potential leakage. A knowledge of typical testing procedures using hydrostatic tests provides a better understanding of the procedures which might be followed using other testing methods.

Except in rare cases, a hydrostatic test precedes other leak-testing methods which may be employed. This accomplishes two purposes. First, the requirements of the ASME Code are satisfied so that subsequent testing can be undertaken at less than required Code test pressures. Second, any leaks which occur under hydrostatic test will have been corrected thus minimizing time required for other more sensitive tests.

The sequence used in testing heat exchangers varies according to the type of unit and the manufacturer's or user's preference. Tube bundles are usually not tested outside of the shell because of the need for special fixtures. The following procedure for a floating head unit is one of several acceptable methods:

a) The heat exchanger is completely assembled with all bolted joints made up except that the shell cover is not installed. Tube side test pressure is applied. Checks are made for leakage at the floating head cover, at tube joints in the floating tubesheet, and at the channel.

b) The shell cover is installed and the channel cover removed. Shell side test pressure is applied followed by leakage inspection of tube joints at the front tubesheet, all exposed bolted joints, weld seams, etc.

c) The channel cover is reinstalled and the tube side pressure test repeated to establish integrity of the channel cover bolted joint.

d) Test pressures, with the exception of the final check of the channel cover bolted joint are held for a sufficient period to assure tightness. Steady pressure gauge readings are mandatory.

Testing procedures such as the one described above have several disadvantages:

1. Leakage through the floating head tube joints must be observed at the back face of the tubesheet and the tubes partially obstruct the view.
2. Faults along the length of the tubes are covered by the shell. Very small leaks are often not discovered.
3. The tube side design and test pressures are often much higher than the shell side pressures. Yet the tube joints at the front tubesheet are open for inspection only on shell side test. The reverse is true at the floating tubesheet.

For these reasons some exchanger purchasers specify that tube bundles be tested outside of the shell. Alternatively, they may purchase a test fixture which permits a shell side test with the floating tubesheet exposed for leakage inspection. Such a test fixture can be used for subsequent field testing to allow visual inspection of tube ends to determine if leaks have developed during service.

Leak Testing Methods

Testing with more sensitive methods than hydrostatic tests may follow each step of the typical procedure outlined above. Usually, however, the more sensitive tests are limited to leakage paths which are of importance to operation — inter-stream leakage or flange leakage, for example. There are many test methods. The more important are described below.

1. **Fluorescence Test** This is a variation of the hydrostatic test. Fluorescent chemicals are added to the test water. Inspection for leaks is made with "black" light under hoods or in darkness. Leaks often not detected with hydrostatic testing are evident with this method.
2. **Soap Bubble Test** Air pressure is applied and leakage detected by application of soap solution. Surfaces being tested must be reasonably clean and free of oil to assure effective foaming of the soap solution. More care must be used in this test to avoid overlooking defects than in other methods where a positive detector is utilized.
3. **Helium Testing** Helium testing is the most sophisticated test presently available. Test equipment and testing are relatively expensive. Leak detection is by means of a mass spectrometer detector, with a probe for use with positive pressure testing. Because of the high sensitivity of the test, the procedure normally followed includes an air soap bubble test prior to the helium test. Helium testing has taken the place of halogen testing for most heat exchanger manufacturers.
4. **Other Testing Methods** Many other testing methods are available. Most are not usually employed in heat exchanger testing. Included in these are chemical testing of varied types, isotope testing, high voltage spark testing, etc. Most heat exchanger manufacturers are not equipped or prepared to execute these tests.

Leak Testing Costs

The cost of leak testing beyond the normal hydrostatic tests is highly variable as it depends on equipment size, complexity, and the extent of testing. The cost of hydrostatic testing an "average" heat exchanger will be in the range of roughly $1,200 to $1,600. With this figure in mind, the following approximate factors may be applied to arrive at an idea of the added cost for further testing:

Hydrostatic Test = Base Cost

Fluorescence Test = Base Cost x 1.5

Helium Test = Base Cost x 2

These factors are based on a complete exchanger test. If added testing is limited to certain key areas, such as tube joints, the costs are much reduced.

Conclusion

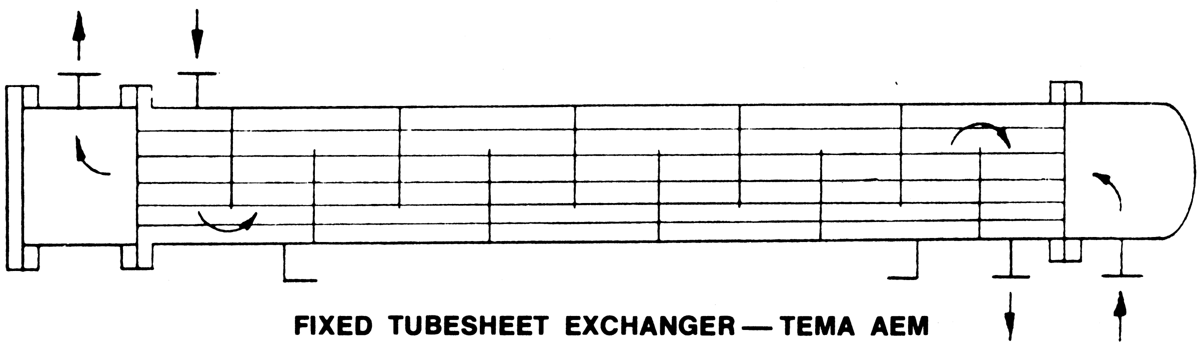
More attention needs to be given to leak testing beyond the usual hydrostatic test. This is particularly important with current trends to high temperature high pressure equipment and high severity service. Methods currently available, properly applied to specific "need areas" are not expensive relative to equipments costs or to costs of leakage repair as it occurs during equipment service. The specification of added testing may in certain cases emphasize the need for more careful design to satisfy these requirements. This is putting the cart where it belongs.

EXCHANGERS FOR TEMPERATURE CROSSOVERS

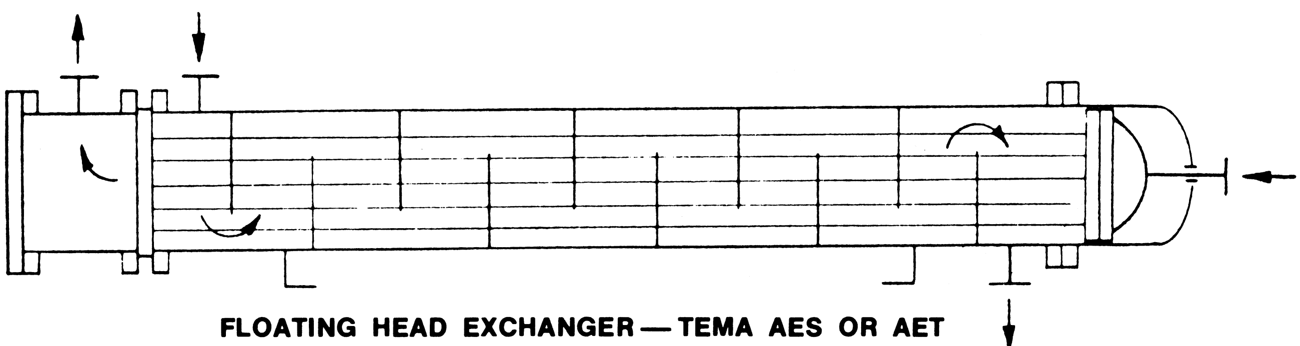
There are some heat exchanger services that literally cry for a "true" counter-current flow heat exchanger — an exchanger having an equal number of shell and tube passes. This need becomes economically of greater importance as the "temperature crossover" increases (examples below). There are several types of shell and tube exchangers which will provide true countercurrent flow. However, each of the types available has certain restrictive limitations, partly because of inherent flow characteristics and partly because of basic design features. The selection of the best type for a given service is correspondingly more demanding.

Exchanger Types Available

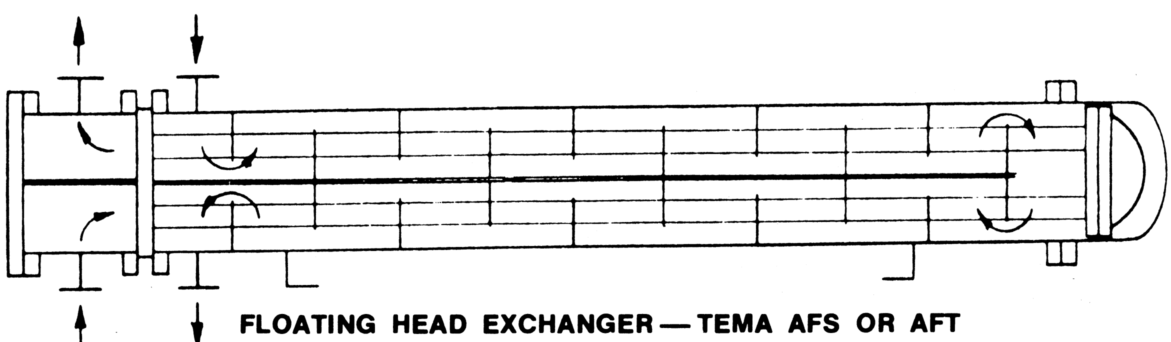
Sketches below illustrate the three basic types of shell and tube exchangers available with true countercurrent flow arrangement (hereinafter termed counterflow):



FIXED TUBESHEET EXCHANGER — TEMA AEM
(Single Pass Shell, Single Pass Tubes)



FLOATING HEAD EXCHANGER — TEMA AES OR AET
(Single Pass Shell, Single Pass Tubes)



FLOATING HEAD EXCHANGER — TEMA AFS OR AFT
(Two Shell Passes, Two Tube Passes)

Variations of these basic types are also available. For example, the fixed tubesheet exchanger (TEMA AEM) above may be provided with two shell passes (TEMA AFM) though this is uncommon; the TEMA AES or AET may be fitted with an expansion joint between the floating head cover and the shellcover, or with a packed joint through the shellcover, or with an outside packed floating head (AEP); the TEMA AFS or AFT may be U-tube (TEMA AFU). And of course other variations are possible.

Temperature Crossover

An exchanger with a single pass shell and multiple tube passes (Parallel-Counterflow Type) can provide an "even" temperature approach — that is, the outlet temperature of the two streams in the exchanger are equal. If there is a temperature crossover, the outlet temperature of the colder stream is higher than the outlet temperature of the warmer stream, and without true counterflow more than one shell must be used, arranged for flow in series.

The economics of parallel-counterflow as compared with true counterflow are quickly evident from calculation of the log mean temperature difference (LMTD). With true counterflow, no correction factor is applied. With parallel-counterflow, the LMTD must be corrected. The magnitude of the correction factor, F, depends on the number of shells used in series. But the number of shells used must not permit a temperature crossover in any individual shell. The following examples are selected to illustrate this.*

Example 1 (even approach):

Stream A: 300°F 100°F
Stream B: 85°F 100°F

LMTD (true counterflow) = 71.5°F
LMTD × F (parallel-counterflow) = 57.2°F (one shell with multiple tube passes)

Example 2 (temperature crossover = 75°F):

Stream A: 350°F 225°F
Stream B: 200°F 300°F

LMTD (true counterflow) = 36°F
LMTD × F (parallel-counterflow) = 28.8°F (three shells in series, each with multiple tube passes)

Example 3 (temperature crossover = 240°F):

Stream A: 700°F 380°F
Stream B: 280°F 620°F

LMTD (true counterflow) = 89.7°F
LMTD × F (parallel-counterflow) = 75.3°F (four shells in series, each with multiple tube passes)

Since heat transfer surface is inversely proportional to the LMTD, (and to the correction factor), the added surface needed with parallel-counterflow as compared with true counterflow is readily figured. It is 25% more for Example 1, 25% more for Example 2, and 19% more for Example 3. Usually the LMTD correction factor cannot be less than 0.8 without adding more shells in series. Hence, the normal maximum difference is 25%.

Economics often cannot be represented by a simple comparison of surface requirements. If, in Example 2, a single shell with 3,000 square feet of surface would satisfy performance needs with true counterflow, then 3,750 square feet would be needed with parallel-counterflow — in three, 1,250 square-foot units. Since the cost per square foot in the three smaller shells is roughly 20 percent higher than in the single shell, the total cost of the three shells is 50 percent higher than the single shell. If, in any of the above examples, the number of shells is the same, either because performance can be achieved in one shell (Example 1) or because surface needs require more shells anyway, then costs are more directly comparable on a surface basis.

Flow Characteristics and Heat Transfer

The complexity of selection and economics does not end with a comparison of LMTD's or of the number of shells needed. Flowing quantities on both shell and tube sides together with friction losses allowed, heat transfer rates, and surface requirements are all inter-related and must be properly balanced in the right exchanger selection. A single shell pass, single tube pass fixed tubesheet exchanger may look like the perfect choice for a given service. Yet the tube side fluid mass velocity may be so low with a reasonable tube size and length that fixed tubesheet design is uneconomical. This same service may fit nicely into a floating head exchanger with two shell and two tube passes, or into a two-shell pass unit with more than two tube passes.

Another service may appear ideal for a two-pass shell but pressure drop restrictions cannot be met in the shell side because of restricted flow area as compared with a single pass shell. Here, a single pass, single pass exchanger may be the choice.

Limitations

Each of the exchanger types illustrated are subject to limitations which must be considered during specification and design.

Fixed Tubesheet Exchangers

1. There must be adequate resolution of the problem of differential expansion between the shell and the tubes. Usually, an expansion joint is required in the shell. The size-pressure-temperature relationships of the expansion joint must be appropriate to proper design integrity and cost. A 20″ expansion joint designed for 150 PSIG at 300°F for moderate movement is inexpensive and proven. A 48″ expansion joint designed for moderate movement at 450 PSIG and 300°F would be expensive and most likely would rule out use of fixed tubesheet construction.
2. The lack of shell-side access must be considered. Refer to Bulletin No. 3.

Single Tube-Pass Floating Head Exchangers

1. If a packed joint is used to seal off leakage of the shell fluid, the consequences of possible leakage must be considered. Normally, packed joints are not used above 300 PSIG and 500°F.
2. If an expansion joint is used rather than a packed joint, the integrity of the joint must be considered, as with fixed tubesheet design. In this usage, with the design configuration commonly used, the joint is subject to external pressure. The joint is hidden, and failure will cause interstream leakage.

Two-Pass Shell Exchangers

1. Some leakage across the longitudinal baffle dividing the shell must be expected even with the fairly effective sealing design currently being used. The tolerance for leakage is a function of temperature differences across the longitudinal baffle, pressure drop, quantity of fluid flowing and the nature of the fluid, and the size of the shell. Possible leakage indicates that two-pass shells should not be used under the following circumstances:
   a. When shell diameter is small and shell flow quantities result in tightly packed baffles. A small shell will have the same leakage area (with the same tube length) as a large shell.
   b. When the shell fluid is heated or cooled over a wide range. Here a small amount of leakage or bypass from inlet to outlet nozzle will quickly spoil performance, particularly where large temperature crossovers are involved.
   c. Where nozzle to nozzle pressure drop exceeds about 7 PSI. This figure is rule-of-thumb but generally applies both because of leakage and because of structural limitations of the longitudinal baffle.
2. The usual pass-to-pass limitations of temperature difference (usually about 300°F) maximum apply also to two-pass shells. However, in two-pass shells there is also a heat transfer effect across the longitudinal baffle which must be offset by using added tube surface.

Sealing Baffles in Two-Pass Shells

Two-pass shells developed a bad reputation years ago when many of them failed to meet specified performance requirements because of baffle leakage. Moreover the sealing designs then used were difficult and expensive to maintain.

The sealing design currently used by most manufacturers is simple, relatively inexpensive, and easily maintained or replaced. The design is variously called "multiflex", "leaf", "lamiflex", etc. It is a flexible spring seal, orientated so that pressure drop in the shell tends to improve the seal. The flexible "spring" consists of a multiple number of light gauge alloy strips usually about 1½ to 2" wide, 0.002 to 0.004" thick, and the length of the baffle. These spring strips are bolted to the longitudinal edges of the baffle and are curved up or down against the shell as the tube bundle is inserted.

The seal achieved with the flexible spring design has proven quite effective. Leakage does occur but to a much lesser extent than with previously used designs. Assuming limitations itemized above are observed, the stigma previously attached to two-pass shells is no longer deserved.

Another type of sealing baffle design used by many manufacturers is an integral design wherein the sealing baffle is welded to the inside surface of the shell cylinder.

Conclusions

Exchanger services with temperature crossovers require special consideration both in design and in economics. Economics, in particular, are difficult to assess without actually sizing the exchanger, often using different types so that a comparison is possible. Limitations itemized above must also be considered. It is important that the purchaser be as liberal as possible in specifying exchanger types permitted so as to allow the designer freedom in his selection of the most economical type.

*(Note: Examples are based on a straight-line variation of temperature with heat transferred. This is sometimes not the case, particularly where a change in phase occurs. The analysis then becomes more complex.)

EXCHANGER BOLTED JOINTS

Most shell and tube exchangers have at least one main bolted flanged joint. The commonly used floating head exchanger with removable channel cover usually has four such joints, one hidden from view. These flanged connections are needed to permit inspection, cleaning, and repair. They appear to be simple and dependable, and they usually are. However, they are subjected to a surprising number of interacting forces. If they are abused, or improperly made up, leakage may result. The following discussion may assist the user in obtaining better service from his equipment and in understanding the need for careful attention to exchanger flanged joints.

Bolted Flange Design — Past and Present

The ASME Code, under which nearly all heat exchangers are fabricated, devotes an entire appendix to "Rules for Bolted Flanged Connections." These rules were formulated more than forty years ago. The fact that they are virtually unchanged today is proof that the rules have provided adequate flange designs over the years. However, advances in the technical use of the rules has tended to place more of a burden on users to carefully observe proper bolting-up procedures.

Past Design For several years after the Code rules were established, flanges were hand designed using slide rules or desk calculators. The design procedure is rather complex, and a trial and error method of solution is necessary. To reduce design time, most exchanger manufacturers standardized sizes and pressure classes (150, 300, 450 PSI designs). But even standardized designs were not optimum. Two points are significant: First, the lack of optimization was usually conservative, providing a more rugged flange than was needed. Second, the use of standard pressure class designs provided added safety factor if the design pressure was lower than the standard class. The net result was a certain amount of over-design in nearly all exchanger flanged connections.

Present Design Currently, exchanger flanges are computer designed. The design is optimum. Standardization has disappeared. It was for this reason that standard flange tables that had appeared in the 1952 Edition of Standards of Tubular Exchanger Manufacturers Association (TEMA), were not present in the 1959 Edition. Most manufacturers changed over to computer designed flanges during the middle 1950's. since these were optimized designs, there was a smaller margin for abuse or misapplication of flanges.

Exchanger Flanges and Piping Flanges Compared

Although configuration is the same, piping flanges and exchanger flanges differ in three major ways:

1. Exchanger flanges, on the average, are considerably larger in diameter.
2. Exchanger flanges in multiple-pass units are subjected to variations in temperature around the flange. Piping flanges are at uniform temperature except in unusual circumstances.
3. Exchanger flanges are normally designed for actual service conditions. Piping flanges are selected from pressure classes that often provide a stronger flange than is required.

This comparison is made simply to emphasize the fact that flanged joints in exchanger service deserve somewhat more care than their cousins in piping service.

Gaskets and Flange Facing

Part of the Code rules for flange design are devoted to gaskets and flange contact facings. This portion of the Code is not mandatory as it is impossible to cover the multiplicity of gasket forms and shapes available. Choice of gasket material and type and corresponding flange facing is a subject that needs more research. "Rule of thumb" methods are often used for selecting gasket material and flange facing type. In mild service, the selection is not too important as most gasket types are adequate. At higher pressure, in corrosive service, and where temperatures are high or where fluctuations may be encountered, the choice is difficult and important. Some guidelines for selection follow:

a) Choose a gasket material that is softer than mating surfaces. This may seem obvious; surprisingly, it is often violated.

b) Regardless of the gasket type selected, but particularly with solid metal gaskets, specify no splices of any kind in the gasket, welded or otherwise.

c) In material choice, particularly at elevated temperatures, look first for materials having expansion coefficients about the same as the flange material.

d) Do not be highly concerned about corrosion, within limits. Gaskets are subject to edge corrosion only, rarely fail from corrosion, and are easily replaceable. Consider galvanic action more seriously than corrosion.

e) Avoid complex mating surfaces, such as serrated or phonograph finishes. These are easily damaged and their ability to seal better than smooth surfaces is over-rated. A nubbin is often needed for proper gasket seating, particularly with solid metal gaskets. Even this type of surface is subject to damage in normal maintenance.

f) **Where stainless steel gaskets are needed or considered to be needed because of corrosive or other conditions, consider, first, using soft iron (see Paragraph d above) or non-ferrous alloy with the idea of replacing the gasket during regular maintenance programs. If stainless steel must be used, specify spiral wound or double jacketed non-asbestos in preference to solid metal. Solid stainless steel gaskets are sometimes difficult to seat.**

g) When purchasing replacement gaskets, seek quality first.

Bolt Tension and Torque Relationships

Bolts in a flanged joint must be designed for the limiting of two conditions. First, bolting must be adequate to withstand the internal pressure plus maintain a certain minimum sealing pressure on the gasket. Second, bolting must be adequate to "seat" the gasket during boltup. During gasket seating, the gasket yields and some plastic flow occurs to create a seal against mating surfaces.

In relatively low pressure designs, the bolt load to seat the gasket is generally the controlling load. In moderate to high pressure designs the bolt load to sustain pressure and maintain gasket pressure controls. Excessive tightening in this instance will crush the gasket, and if continued, will overstress bolts and flanges.

The need to achieve proper bolt loading in flanged joints cannot be overemphasized. Too low a bolting load may cause leakage. Too high a loading may damage flanges, gasket and bolts as pointed out above. The ASME Code rule requiring pressure testing at 1.3 times the design pressure tends to magnify the need for careful bolt preloading. Theoretically, bolting must be loaded to about one-and-one-half times the design stress. The yield point of commonly used alloy studs is nearly always far above the yield point of flange material. Permanent damage can be done to flanges and gaskets long before bolts reach their elastic limit.

The determination of the proper torque to develop the required bolt tension depends on the friction coefficient between nut-and-stud bearing surfaces and the friction coefficient between the nut-and-flange bearing surfaces. Most torque tables are based on a rule-of-thumb value of about 0.15 for this coefficient. The coefficient ranges from 0.15 to 0.25 without lubrication. Recent data indicate that the actual coefficient with good surface condition of threads and a good lubricant can be as low as 0.06 and sometimes slightly less than this. The following table shows torque values to develop a bolt stress of 30,000 PSI at friction coefficients of 0.06, 0.12, and 0.18.

Nominal Stud Diameter*	Thread Root Area**	Compression Load Pounds***	Torque, foot-pounds, for 30,000 PSI Stress Frict. Coeff. = 0.06	Frict. Coeff. = 0.12	Frict. Coeff. = 0.18
5/8"	0.202	6,060	30	54	78
3/4"	0.302	9,060	51	54	137
7/8"	0.419	12,570	82	150	217
1"	0.551	16,530	122	223	324
1 1/8"	0.728	21,840	179	326	475
1 1/4"	0.929	27,870	253	457	664
1 3/8"	1.155	34,650	343	619	900
1 1/2"	1.405	42,150	448	811	1,171
1 5/8"	1.608	50,400	573	1,050	1,514
1 3/4"	1.980	59,400	716	1,320	1,916
1 7/8"	2.304	69,120	886	1,646	2,370
2"	2.652	79,560	1,075	2,040	2,893

(*) National Coarse series below 1" and eight-thread series 1" and above.
(**) Square inches.
(***) Per bolt.

Assuming that torque were applied based on a coefficient of 0.12 but the actual coefficient were 0.06, the bolt stress would be about 55,000 PSI rather than the 30,000 PSI of the table.

The proper torque to apply becomes a matter of judgment, based on the condition of studs and nuts. In most instances, it seems unlikely that studs and nuts would be in the "mint" condition necessary to obtain friction coefficients lower than about 0.10. Under these circumstances, table values based on a coefficient of 0.12 can usually be safely used.

Recommended Flange Bolting-Up Procedure

The following procedure is recommended for bolting-up of flanged joints where accurate control of bolt loading is required:

1. Inspect, clean and repair gasket surfaces as needed. Remachining may be required if surfaces are damaged or corroded.
2. Use a new gasket of first-rate quality. Make certain the gasket is centered and correctly located.
3. Before bolt and nut installation, examine threads in the engagement area for defects or burrs. Clean threads thoroughly of rust and foreign material and previous lubricant.
4. Apply lubricant to bolt and nut threads and to surfaces contacted by the nuts on the back face of the flanges. The lubricant used should be of uniform consistency, thoroughly mixed.
5. After hand tightening of nuts apply torque with a steady, even pull of the torque wrench to the desired reading. Wrench size should be such that the reading is between 1/4 and 7/8 of the scale range, and the wrench should be accurately calibrated.
6. Tighten bolts in a sequence that will provide uniform loading. Final torque should be reached in a stepwise procedure, using one or two intermediate values.
7. During stepwise tightening, check flanges carefully for tipping or cupping, particularly if flanges are low-pressure design and if torque values are based on high friction coefficients.

Bolt Tensioners

Large stud bolts are difficult to tighten accurately or sufficiently to give the proper bolt preloading. In such cases, the use of a bolt tensioner should be considered. Bolt tensioners operate on the principle of a hydraulic jack and put a direct tension pull on the stud to be tightened. Certain things should be kept in mind if bolt tensioners are to be used to tighten stud bolts on a heat exchanger:

1. The load capacity of the tensioner should be generously sized. A tensioner sized to stress bolting to 50,000 PSI is not too large because this reserve capacity may be needed since the bolt tension is relaxed somewhat when the tensioner is removed.
2. Clearance is needed to apply the tensioner. Usually, channel and shell nozzles are in the way unless the exchanger manufacturer is aware that a bolt tensioner is to be used, and has been advised of the clearance required.
3. Stud bolts must be approximately one nut height longer than normal.

Recommendations of the bolt tensioner manufacturer regarding proper tightening procedures should be closely followed. Usually, the bolt tensioners are used in set of four.

Operation Effects

The most critical periods to which a flanged joint is subjected are those which occur during transient conditions of temperature and/or pressure. These usually occur during startup or shutdown or during operational upsets. Generally, these are not known by the manufacturer during the design stage. Some of the conditions which occur are described as follows:

a) Flange bolting temperature lags behind flange metal temperature during exchanger warm-up and vice-versa. The bolt tension varies accordingly.

b) Exchanger flanges, particularly in multipass units, may be subjected to considerably higher temperature differences around their periphery than are considered during design at "steady state" conditions.

c) Gasket temperatures respond quickly to contacting fluid temperatures while adjacent thicker parts lag behind.

On occasions one or a combination of these conditions may result in leakage. Competent manufacturers are aware of these problems and attempt to minimize them by careful design and careful selection of the proper heat exchanger type. Flange peripheral temperature differences, for example, are restricted to certain maximums that have proven satisfactory by experience. And material selections are made to avoid serious differential expansion problems at bolted joints. Nonetheless, proper usage is all important and this burden can fall only on the user. Heat-up and cool-down rates must be controlled. Insulation must be applied for best effectiveness. And the user may sometimes find it necessary to adjust bolt loading as exchangers are brought up to operating temperatures.

Pre-Operation

Generally, several months elapse between the time a new exchanger has been shop tested and the time when final field tests are undertaken. Occasionally, leaks occur in bolted joints during field tests in spite of the fact that units were leak-tight during shop testing. It occurs because of rough handling during shipment, weather conditions, forces exerted by piping, and the like. Sound policy on the user's part is to go over all bolted joints at the time of field testing to make certain bolts are properly tightened.

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