

HEX HEAT EXCHANGER DESIGN HANDBOOK

4

Mechanical design of heat exchangers

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HEDH

GENERAL PREFACE

Ernst U. Schlünder

■ The idea to create a Heat Exchanger Design Handbook originated at the seminar Recent Developments in Heat Exchangers, organized by the International Centre for Heat and Mass Transfer (ICHMT) in Trogir, Yugoslavia in 1972.

Even though much heat transfer data and correlations were available, this information was scattered throughout the open literature or buried in the files of proprietary organizations. For manufacturers, vendors, and users of heat transfer equipment it was becoming more and more difficult not only to keep up with the continually growing heat transfer information but also to compare data and correlations from various sources. In addition, these sources were often inconsistent, and it was difficult for users to decide which data and correlations would serve their own purposes best. Moreover, over the years many heat transfer publications had become more analytical and more academic, which delayed their immediate application by practicing heat transfer engineers.

Because of this situation, a group of heat transfer specialists decided to collect heat transfer information available in the existing open literature and also-as far as possible-from proprietary sources in order to create a heat transfer information base. Recommended correlations based on selected experimental material would then be provided for all heat exchanger design data.

Although there was great enthusiasm for this project, not much progress was made until 1974, at the Fifth International Heat Transfer Conference in Tokyo. The idea was formulated then of publishing the information base under the auspices of the ICHMT by Hemisphere Publishing Corporation, the publisher of the

Centre. Eight representatives of universities, industry, and proprietary organizations formed the Editorial Board, which immediately started working on the scope and organization of this Handbook. As a result of these joint efforts the following live parts were planned:

Part 1 presents the theory of heat exchangers defining and explaining all parameters and concepts needed for thermal and hydraulic design and rating.

Part 2 contains recommended correlations for design and rating parameters based on evaluated available information. As far as possible experimental data are presented together with the recommended correlations in order to reveal both the consistency of the data as well as the reliability of the correlations.

Part 3 shows how to apply the material given in Parts 1 and 2 when solving actual heat exchanger design and rating problems. Because of the volume of material, it would be impossible for this part to cover all known kinds of heat exchangers. Nevertheless, most of the commonly used types of equipment are discussed in detail together with related installations such as cooling towers and combustion chambers.

Part 4 is devoted to the mechanical design of heat exchangers, which actually cannot be separated from thermal and hydraulic design. A useful comparison of some major national standards is included.

Part 5 contains physical property data that are needed in design calculations in the other four parts.

For each of the five parts, a member of the Editorial Board agreed to act as Lead Editor. Although each Lead Editor accepted additional administrative responsibility for his particular part, the whole Handbook has been a

cooperative effort with each editor sharing the reviewing and editorial work for all the material.

A large number of authors—well-known experts in their fields—have contributed to this Handbook. Each manuscript went through a review procedure and many of them were discussed during the regular meetings of the Editorial Board, which have been held every six months since 1975.

Nevertheless, this Handbook must be considered as a first step rather than as a final result. Though the concept was clear, the realization has not been easy and there is still room for improvement. Additionally, because new heat transfer data and correlations are published each year, a permanent updating procedure has been established to keep this Handbook current. This supplementary material will be offered regularly to the

subscribers. To ensure that any future material (including personal remarks and comments) can be added easily, a loose-leaf ring binder format was chosen for the Handbook.

The users of this Handbook are urged to communicate with us about what is missing and to make suggestions for improving the contents of the book and its applicability. This kind of interaction and cooperation between users and editors not only should improve the quality of the Handbook but also should stimulate further research work guided by the requirements of heat exchanger design.

The Editorial Board wishes to thank the publisher as well as the contributing authors for their most valuable cooperation.

2

Preface

Frank W. Schmidt

■ The thermal design engineer is forced to base his or her design on published or proprietary correlations or on experimental results. It is recognized that there is uncertainty in the accuracy of many of these correlations. The primary objective of this part of the Handbook is to evaluate and present the best available correlations. Whenever possible, the expected accuracy of the correlations is indicated by comparison with reliable experimental results.

A number of authors have included a discussion of the basic phenomena, recognizing that in many instances an engineer unfamiliar with a technical area will refer to the Handbook for design assistance. Words of caution are also given when special considerations must be given to avoid an undesirable or faulty design. Although space limitations required that only the most general correlations be given, most authors have included an extensive list of references which will assist users of the Handbook in their search for more detailed information.

Heat exchangers take so many varied forms that their design requires correlations that are applicable over a wide range of transfer processes and exchanger geometries. The Editorial Board, recognizing this, attempted to cover as completely as possible all the significant areas associated with thermal transfer devices. As a result, this part will be of considerable value not only to engineers concerned with the design and rating of heat exchangers but to those interested in all aspects of thermal design.

Section 2.1 reviews the fundamentals of heat, mass, and momentum transfer. Single phase fluid flow in ducts, in banks of plain and finned tubes, and over immersed bodies is discussed in Sec. 2.2 for both Newtonian and non-Newtonian fluids. A discussion of flow in fixed and fluidized beds is also presented in this section.

Extensive coverage of multiphase fluid flow and pressure drop is given in Sec. 2.3. A discussion of the fundamentals of these types of flows is followed by a detailed discussion of gas-liquid, solid-gas, and solid-liquid flows.

Heat conduction is covered in Sec. 2.4. Steady-state and transient heat conduction with and without changes in phase are discussed. A section on contact resistance is also included.

Section 2.5 presents correlations for single phase convective heat transfer. These include free, forced, and combined free and forced convection. Heat transfer in Newtonian, non-Newtonian, and liquid metals systems are discussed. Techniques used to augment heat transfer are also described.

A very complete section on condensation of pure vapor, vapor mixtures, and vapor mixtures forming immiscible liquids is presented in Sec. 2.6. Dropwise condensation, fogging, and enhancement devices are also discussed in this section.

Boiling and evaporation is the subject of Sec. 2.7. The techniques required to predict heat transfer and vapor formation rate in pool boiling, boiling inside and outside tubes, and tube bundles of single component liquids are presented. Boiling of binary and multicomponent mixtures is also discussed.

Section 2.8 discusses heat transfer to gas-solid systems in fixed beds with and without gas movement, fluidized beds, and agitated packed beds.

A very thorough discussion of radiation heat transfer is presented in Sec. 2.9. Surface radiation characteristics, radiation heat transfer between specular and diffuse surfaces and gaseous radiation are discussed in detail in the section.

HEDH NOMENCLATURE

■ The most frequently used symbols for quantities in heat and mass transfer are listed here according to the following principles:

1. Capital Roman letters (not italic) denote absolute quantities expressed in either **SI** basic units or derived units, e.g., $N = \text{kg m/s}^2$, $J = \text{N m}$, $W = \text{J/s}$.

Exceptions: Time (t), variables, coordinates (x, y, z, r), characteristic lengths in dimensionless numbers (l, d, s).

2. Lowercase Roman letters (not italic) denote the corresponding specific quantities per unit mass, mole, length, area, or volume (but not time). A dot over the symbol denotes an absolute or specific quantity per unit time.

Exception: Velocity (u, v, w).

A function will be indicated by $\{ \dots \}$.

Molar quantities are denoted by a tilde ($\sim$) over the symbol.

Exceptions: Gas constant ($\tilde{R}$), Loschmidt number ($\tilde{L}$), molecular weight ($\tilde{M}$), chemical potential (μ).

3. Lowercase Greek letters are used for coefficients and physical properties.

Exceptions: Overall heat transfer coefficient (U), heat capacity (c_p, c_v).

Capital Greek letters refer to eddy (turbulent) quantities; e.g., A denotes eddy thermal conductivity.

4. Dimensionless variables should be denoted by a superscript $^+$ following the corresponding symbol, e.g., y^+ for a dimensionless coordinate.

5. Decimal fractions or multiples of the **SI** units may be denoted by the corresponding symbol preceding the **SI** unit, e.g., $MW = 10^6 \text{ W}$.

Basic Quantities

Quantity	Symbol	Unit
Heat rate	$\dot{Q}$	W
Heat quantity	$Q = \int \dot{Q} dt$	J
Heat flux	$\dot{q} = \dot{Q}/A$	W/m ²
Time	t	s
Frequency	f	1/s
Mass	M	kg
Mass flow rate	$\dot{A}$	kg/s
Mass velocity (mass flow per unit area)	$\dot{m} = \dot{M}/S$	kg/m ² s
Mass flux	$\dot{m}_n = \dot{M}_n/A$	kg/m ² s
Number of moles	N	mol
Molar mass flow rate	$\dot{N}$	mol/s
Molar mass velocity	$\dot{n} = \dot{N}/S$	mol/m ² s
Molecular weight	$\tilde{M}$	g/mol
Density	ρ	kg/m ³
Concentration	c	kg/m ³
Molar density	$\tilde{\rho}$	mol/m ³
Molar concentration	$\tilde{c}$	mol/m ³
Mass fraction	x_i, y_i	kg _i /kg
Mass loading	X_i, Y_i	kg _i /kg _j
Molar mass fraction	$\tilde{x}_i, \tilde{y}_i$	mol _i /mol
Molar mass loading	$\tilde{X}_i, \tilde{Y}_i$	mol _i /mol _j
Volume	V	m ³
Volume flow rate	$\dot{V}$	m ³ /s
Velocity	u, v, w	m/s
Surface area	A	m ²
Cross-sectional area	S	m ²
Length or diameter	L, D, l, d, s	m
Coordinate	$\psi, \varphi, r, x, y, z$	—
Void fraction	ϵ	—
Fraction of volume flow rate	$\dot{\epsilon}$	—
Quality	x	—
Flow quality	$\dot{x}$	—
Work	w	J

Transport coefficients

Quantity	Symbol	Unit
Heat transfer coefficient	α	W/m ² K
Mass transfer coefficient	β	m/s
Overall heat transfer coefficient	U	W/m ² K
Drag coefficient	ξ	—
Friction factor	f	—
Emissivity	ϵ	—
View factor	φ_{12}	—
Thermal conductivity	λ	W/m K
Diffusivity	δ	m ² /s
Kinematic viscosity	ν	m ² /s
Dynamic (absolute) viscosity	η	kg/m s
Thermal diffusivity	$\kappa \equiv \lambda/\rho c_p$	m ² /s
Surface tension	σ	N/m
Shear stress	τ	Pa (N/m ²)
Pressure	p	Pa (N/m ²)
Pressure drop	Δp	Pa (N/m ²)

Thermodynamic quantities

Quantity	Symbol*	Unit
Enthalpy (specific)	h	J/kg
Internal energy (specific)	u	J/kg
Gibbs function free enthalpy (specific)	$g = h - Ts$	J/kg
Free energy (specific)	$f = u - Ts$	J/kg
Entropy (specific)	s	J/kg K
Chemical potential	μ	J/kg
Specific heat capacity	c_p, c_v	J/kg K
Temperature	T	K, °C
Mean logarithmic temperature difference	ΔT_{LM}	K, °C

*For totals, letters are capitalized, **H**, **U**.

Chemical reactions

Quantity	Symbol	Unit
Reaction enthalpy; phase-change enthalpy; latent heat	Ah	J/kg
Energy of activation	AE	J/kg
Equilibrium constant	K	Various
Rate constant	k	Various
Rate of reaction	$\dot{r}$	mol/m³s
Activity coefficient	γ_l	
Fugacity coefficient	γ_g	—
Stoichiometric factors	ν_i	—

Physical constants

Quantity	Symbol	Unit
Gas constant	R	8.3 14 J/mol K
Loschmidt number	$\tilde{L}$	6.0252 $\cdot 10^{23}$
Avogadro number	$\tilde{L}$	1/mol
Stefan-Boltzmann constant	C_s	5.6697 $\cdot 10^{-8}$ W/m ² K ⁴
Acceleration of gravity	g	m/s ²
Standard acceleration of gravity	$g_n = 9.8 \dots$	m/s ²

Subscripts

Quantity	Symbol
Wall	w
Bulk	b
Mean	m
Solid	s
Liquid	l
Gas	g
Inlet	in
Outlet	out

Dimensionless numbers (most frequently used)

Name	Symbol and definition
Archimedes number	$Ar = g l^3 \Delta \rho / \nu^2 \rho = g l^3 \rho \Delta \rho / \eta^2$
Biot number	$Bi = \alpha_g l / \lambda_s$ or $\alpha_l l / \lambda_s$
Effectiveness	E
Euler number	$Eu = \Delta p / (\rho u^2 / 2)$
Fourier number	$Fo = \kappa t / l^2 = \lambda t / \rho c_p l^2$
Froude number	$Fr = u^2 / gl$
Galileo number	$Ga = l^3 g / \nu^2 = l^3 g \rho^2 / \eta^2$
Grashof number	$Gr = g l^3 \Delta \rho / \nu^2 \rho = g l^3 \rho \Delta \rho / \eta^2$
Graetz number	$Gz = u d^2 / \kappa l = u d^2 \rho c_p / \lambda l = Re \ Pr \ d / l$
Knudsen number	$Kn = l_0 / l$, where l_0 is the molecular mean free path
Lewis number	$Le = \kappa / \delta = \lambda / \rho c_p \delta$
Mach number	$Ma = u / u_{\text{sound}}$
Number of transfer units	NTU
Nusselt number	$Nu = \alpha l / \lambda$
Péclet number	$Pe = ul / \kappa = ul \rho c_p / \lambda = Re \ Pr$
Phase change number	$Ph = c_p \Delta T / Ah_{fg}$
Prandtl number	$Pr = \nu / \kappa = \eta c_p / \lambda$
Rayleigh number	$Ra = Gr Pr$
Reynolds number	$Re = ul / \nu = ul \rho / \eta = \dot{m} l / \eta$
Schmidt number	$Sc = \nu / \delta = \eta / \rho \delta$
Sherwood number	$Sh = \beta l / \delta$
Stanton number	$St = \alpha / \rho u c_p = Nu / Re \ Pr$
Strouhal number	$Sr = fl / u$
Weber number	$We = u^2 \rho l / \sigma$

Dimensionless mass transfer numbers are distinguished from dimensionless heat transfer numbers by a prime ('), e.g., $Bi' = \beta l / \delta$. Exceptions: $Nu' = \beta l / \delta = Sh$ and $Pr' = \nu / \delta = \eta / \rho \delta = Sc$

HEDH

INTERNATIONAL SYSTEM OF UNITS (SI): RULES, PRACTICES, AND CONVERSION CHARTS

J. Taborek

● HEDH is written in SI units because this system has been officially adopted by all countries and is replacing the older metric engineering system, the cgs system, as well as the U.S. customary system. In this section we shall review the basic rules for use of the SI. However, recognizing the fact that many references are written in other unit systems, comprehensive unit conversion charts are included.

A. Basic SI units

SI is absolute in the sense that there are only seven fundamental, precisely defined basic units from which all others are derived. The following five units are important to heat transfer and fluid flow.

length (meter, m): defined in terms of krypton-86 wavelength

mass (kilogram, kg): the mass of an international prototype, approximately equal to the weight of 1 liter H_2O at $4^\circ C$

time (second, s): defined in terms of radiation frequency of the cesium-133 atom

temperature (kelvin, K): defined in terms of the triple point of water (ice point = 273.15 K)

amount of substance (mole, mol): amount of substance that contains as many elementary entities as there are atoms in 0.012 kg of carbon-12. The elementary entities must be identified when the mole is used (atoms, molecules, ions).

B. Derived SI units with special names

Because SI is defined by the basic units only, there is a need for establishing definitions for the most frequently used composite or derived units, mainly for the purpose of simplifying unit notation. Thus, for example, newton, N, replaces $kg\ m/s^2$.

There are a number of derived units that have special names, signifying their importance: newton, joule, watt, pascal, and many others. While the unit symbols are capital letters (N, J, W, Pa), the spelled out unit names begin with lowercase letters, that is, newton, joule, watt, and pascal.

The most frequently used derived SI units with special names in the area of heat transfer and fluid flow are shown in Table 1.

Table 1 Most frequently used derived SI units

Quantity	Unit	Symbol	Dimension
Force	newton	N	$kg\ m/s^2$
Energy, work, heat (also power $\times$ time)	joule	J	$N\ m$ or $kg\ m^2/s^2$ $W\ s = J\ s/s$
Power	watt	W	$J/s = kg\ m^2/s^3$
Pressure	pascal	Pa	$N/m^2 = kg/m\ s^2$
Frequency	hertz	Hz	$1/s$
Dynamic viscosity	poise centipoise	P cP	$0.1\ Pa\ s$ $mPa\ s$

C. Prefixes and specification rules

The magnitude of the basic units may often not fit the practice of specific applications without using a large number of decimal digits or power of ten multipliers. SI provides for this by assigning prefixes to the basic units, as shown in Table 2.

Thus, for example, to express very high pressures (stresses) in Pa, the megapascal, **MPa**, is used instead of 10^6 Pa. The preferred graduation is in powers of 10^3 , and there are only a few exceptions to this rule, such as for units that are deeply ingrained in common use, such as centimeter. Some of these are discussed later.

(a) Punctuation and spelling rules

The following general rules should be carefully followed to maintain SI consistency.

1. **Prefixes** are attached directly to the base unit symbol or a derived unit symbol with a special name, such as km for kilometer or MW for megawatt.

2. Prefixed units are spelled out in text as one word, without hyphen, such as meganewton, not **mega-newton**.

3. The multiplication of base units can be expressed by leaving a space between the symbols, such as N s for newton X second. However, for better clarity, a center dot is often used, for example, N•s. This usage becomes especially important when the symbol m is used, which can stand for milli or for meter. Thus **mN** (millinewton) is clearly distinguished from **m•N** (meter X newton).

4. A prefix applied to a base unit constitutes a new unit. When a power is applied to such a unit, the exponent operates on the prefixed unit as an entity, i.e., $\text{mm}^2 = (\text{mm})^2$.

5. The unit of length in SI is correctly spelled “metre” (French spelling). This rule will be hard to enforce in U.S. literature, however, and meter will probably be more commonly used.

Table 2 SI prefixes

Factor	Prefix	Symbol	Status
10^{12}	tera	T	Preferred
10^9	giga	G	Preferred
10^6	mega	M	Preferred
10^3	kilo	k	Preferred
10^2	hecto	h	Accepted
10	deka	da	Accepted
10	deci	d	Accepted
10^{-2}	centi	c	Accepted
10^{-3}	milli	m	Preferred
10^{-6}	micro	μ	Preferred
10^{-9}	nano	n	Preferred
10^{-12}	pico	p	Preferred

6. Multiple prefixes are prohibited. This presents some problems with the unit kilogram (kg), where the k stands for the prefix kilo and is one of the very few inconsistencies in SI (a carryover from the older cgs system). Thus k (for kilo) cannot be used in connection with kg, as this would result in the unacceptable term kkg. Equally incorrect is the use of Mg (megagram), as gram is not a basic SI unit. Thus, the unit used is 10^3 kg or the equivalent unit, metric ton (see below).

Similarly, mkg is not an acceptable way to express 10^{-3} kg, and the more common alternate unit of gram (g) is used.

D. Alternate units and supplementary comments

There are as many derived units as the technology demands, as SI is particularly suitable for this purpose by the clearly defined unit relations. The Nomenclature of this handbook contains practically all such units used in heat transfer and fluid flow, showing the appropriate **SI** dimensions. Consequently, these are not repeated here.

In some cases, strict SI rules are relaxed to respect generally accepted and deeply established practices. The following list presents most frequently used exceptions, together with some additional comments to alternate units, their definition, and their usage.

length: The unit centimeter (cm) is a very practical measure for everyday usage (person's height, 6 ft $\approx$ 183 cm), and will probably remain in perpetual use.

mass: The unit of kilogram is often too small for expressing quantities of mass experienced in many engineering applications. One of the few weaknesses of SI is the lack of a well-defined alternate unit of mass. As explained above, the prefixed units (kkg, Mg) are not acceptable. Instead, an alternate unit for large quantities of mass is introduced, designated metric ton (symbol t), $t = 1\,000$ kg. While this unit has not yet been accepted as a recognized SI unit, it is very much in common use.

In the U.S. customary lb.-based system, several definitions of a unit of mass similar in magnitude to the metric ton exist. In engineering practice, the following two definitions of the lb.-based ton are most frequently used:

short ton, which is equal to 2 000 lb.,,
(exact) and therefore approximately = 907.2 kg
long ton, which is equal to 2 240 lb,
(exact), and therefore approximately = 1 016 kg

It is important to be aware of these differences in the inconsistently defined U.S. customary system.

For small quantities of mass, gram (g) is also often used in place of 10^{-3} kg.

time: The unit second (s) should be used whenever possible, as the alternates of minutes, hours, and days are nondecimal entities, and thus complicate the system.

However, to express large quantities of mass per unit of time, the unit hour (h), 3 600 seconds, or the unit day (d), 86 400 seconds, must be used sometimes. The time unit minute (min), often used in the U.S. customary system (**gal/min**), has no real justification, except that it is ingrained in some industrial practices. Otherwise, the differential quantity of a minute, as compared to a second, can be readily expressed by prefixes in SI units, to obtain a number with reasonable digits.

force: The newton is a very small unit of force, approximately equal to the weight of water in a small glass. Thus the more practical units of force are often the **kN** or **MN**.

volume: The liter, defined as dm^3 , is a very popular unit of volume for everyday use, such as for water and gasoline. Liter (L) is accepted as a common alternate unit.

amount of substance: The frequently used units of the metric engineering system, gram-mol and kg-mol, are designated in SI as follows: 1 gram-mol = 1 mol (SI), and 1 kg-mol = 1 kmol (SI). Thus 1 mol = 6.023×10^{23} molecules.

pressure: The basic unit of pressure, **pascal** ($\text{Pa} = \text{N/m}^2$), is a very small quantity, equal to the pressure exerted by about 0.1-mm layer of water over 1m^2 . Thus in most engineering applications **kPa** or **MPa** is more useful.

Many derived units of pressure are firmly established in various branches of engineering and will probably remain in use indefinitely. The most important metric-based units are:

bar: has exact relationship to SI ($\approx 100\text{ kPa}$)
 torr = **1 mmHg** = 0.133 kPa : used for gas pressure measurements
 mm water = 9.8067 Pa : used for very small gas pressure measurements
 at (technical atmosphere) = $\text{kg}_f/\text{cm}^2 = 98.066\text{ kPa}$
 atm (normal or barometric atmosphere) = $760\text{ mmHg} = 101.3\text{ kPa}$

temperature: In many engineering applications it is more convenient to use degrees Celsius rather than the absolute unit kelvin. The use of Celsius is permitted as an alternate, with the relationship
 $T(^{\circ}\text{C}) = T(\text{K}) - 273.16$ and $\Delta T(^{\circ}\text{C}) = \Delta T(^{\circ}\text{K})$

Table 3 Unit conversion

Physical quantity	Given in Gives	Multiplied by Divided by	Gives Given in	Approximate or useful relationship
Length	ft	0.3048	m	$3\frac{1}{4}\text{ ft} \approx 1\text{ m}$
	in	25.4 (exact)	mm	$1\text{ in} \approx 25\text{ mm}$
	mil	0.0254	mm	
	yard	0.9144	m	
	mile (mi)	1 609.3	m	$1\text{ mi} \approx 1.6\text{ km}$
	km	0.621388	mi	
Area	ft^2	0.092903	m^2	$100\text{ ft}^2 \approx 9\text{ m}^2$
	in^2	645.16	mm^2	$1\text{ in}^2 \approx 650\text{ mm}^2$
	acre	4 047.0	m^2	
Volume	ft^3	0.028317	m^3	$35\text{ ft}^3 \approx 1\text{ m}^3$
	U.S. gal	0.003785	m^3	$260\text{ gal} \approx 1\text{ m}^3$
	U.S. gal	3.785	liter (L)	$1\text{ gal} \approx 3\frac{3}{4}\text{ L}$
	L (liter)	0.2642	U.S. gal	$1\text{ L} \approx 0.26\text{ gal}$
	Brit. gal	0.004546	m^3	
	U.S. gal	0.13368	ft^3	
	barrel (U.S. pet.)	0.15898	m^3	
	barrel (U.S. pet.)	42	U.S. gal	
Velocity	ft/s^a	0.3048	m/s	$10\text{ ft/s} \approx 3\text{ m/s}$
	m/s	3.2808	ft/s	
	ft/min	0.00508	m/s	$100\text{ ft/min} \approx 0.5\text{ m/s}$
	mi/h	1.6093	km/h	$30\text{ mi/h} \approx 48\text{ km/h}$
	km/h	0.6214	mi/h	$50\text{ km/h} \approx 31\text{ mi/h}$
	knots	1.852	km/h	

(See footnotes on page xxix.)

Table 3 Unit conversion (*Continued*)

Physical quantity	Given in Gives	Multiplied by Divided by	Gives Given in	Approximate or useful relationship
Mass	lb_m kg metric ton ton (2 000 lb.)	0.45 359 2.2046 2 204.6 907.18	kg lb_m lb_m kg	1 lb, $\approx$.45 kg 1 kg $\approx$ 2.2 lb _m metric ton = 10 ³ kg
Force	lb_f lb_f kg_f kg_f dyne	4.44822 0.45359 2.2046 9.80665 0.00001 (exact)	N = kg m/s ² kg_f lb_f N N	1 N $\approx$ 0.1 kg _f $\approx$ 0.22 lb _f
Amount of substance	lb_m-mol g-mol kg-mol mol	453.6 1.000 1.000 1000	kmol mol kmol kmol	
Mass flow rate	lb_m/h kg/s lb_m/s lb_m/min	0.000 1260 7 936.51 0.4536 0.00756	kg/s lb_m/h kg/s kg/s	10 ³ lb/h = .13 kg/s
Volume flow rate	U.S. gal/min U.S. bbbl/day U.S. bbbl/day ft³/s ft³/min	6.309 x 10 ⁻⁵ 0.15899 1.84 x 10 ⁻⁶ 0.02832 0.000472	m³/s m³/day m³/s m³/s m³/s	
Mass velocity (mass flux)	lb_m/h ft² kg/s m ²	1.356 x 10 ⁻³ 731.5	kg/s m ² lb_m/h ft²	
Energy (work) (heat)	Btu^b Btu Btu kcal ft lb_f W h	1 055.056 0.2520 118.28 4 186.8 1.3558 3 600	J = N m = W s kcal ft lb_f J J J	1 Btu $\approx$ 1 000 J 1 kcal $\approx$ 4 Btu 1 kcal $\approx$ 4 000 J
Power	Btu/h W kcal/h ft lb_f/s hp (metric) Btu/h tons refriger.	0.2931 3.4118 1.163 1.3558 735.5 0.2520 3 516.9	W = J/s Btu/h W W W kcal/h W	10 ⁶ Btu/h $\approx$ 300 kW 1 000 kW $\approx$ 3.5 x 10 ⁶ Btu/h
Heat flux	Btu/h ft² W/m² kcal/cm ² s	3.1546 0.317 41.868	W/m² Btu/h ft² W/m²	1 000 Btu/h ft ² $\approx$ 3.2 kW/m ²
Heat transfer coefficient	Btu/h ft² °F W/m² K kcal/cm ² s °C	5.6184 0.1761 41.868	W/m² K Btu/h ft ² °F W/m ² K	1 000 Btu/h ft ² °F $\approx$ 5 600 W/m ² K
Heat transfer resistance	(Btu/h ft ² °F) ⁻¹ (W/m ² K) ⁻¹	0.1761 5.6784	(W/m ² K) ⁻¹ (Btu/h ft ² °F) ⁻¹	0.001 (Btu/h ft ² °F) ⁻¹ $\approx$ 0.000 18 (W/m ² K) ⁻¹
Pressure	lb_f/in² (psi) kPa bar	6.8948 0.1450 100	kN/m² = kPa psi kPa	1 psi $\approx$ 7 kPa 14.5 psi $\approx$ 100 kPa

(See footnotes on page xxix.)

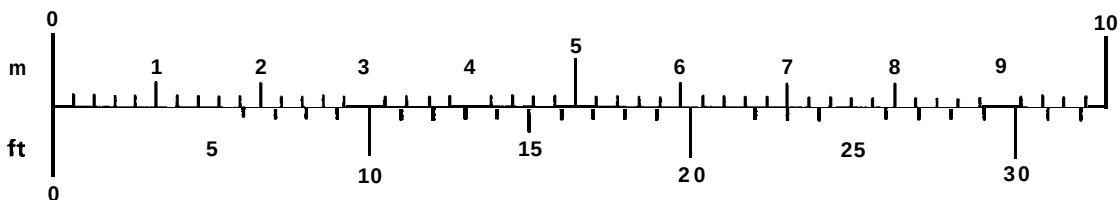
Table 3 Unit conversion (Continued)

Physical quantity	Given in Gives	Multiplied by Divided by	Gives Given in	Approximate or useful relationship
Pressure (Continued)	lb _f /ft ² mm Hg (torr) in Hg mm H ₂ O in H ₂ O at (kg _f /cm ²) atm (normal)	0.0479 0.1333 3.3866 9.8067 249.09 98.0665 101.325	kPa kPa kPa Pa Pa kPa kPa	1 000 kPa = 1 MPa ≈ 150 psi 1 in H ₂ O ≈ .25 kPa atm = 760 mmHg
Mass flux	lb _m /ft ² s lb _m /ft ² h	4.8824 0.001356	kg/m ² s kg/m ² s	
Physical and Transport Properties				
Thermal conductivity	Btu/ft h °F W/m K kcal/m h °C	1.7308 0.5778 1.163	W/m K Btu/ft h °F W/m K	steel ≈ 50 W/m K water (20° C) ≈ 0.6 W/m K air (STP) ≈ 24 mW/m K
Density	lb _m /ft ³ kg/m ³ lb _m /U.S. gal	16.0185 0.06243 119.7	kg/m ³ lb _m /ft ³ kg/m ³	62.4 lb _m /ft ³ ≈ 1 000 kg/m ³
Specific heat capacity	Btu/lb _m °F kcal/kg °C	4 186.8 4 186.8	J/kg K J/kg K	1 Btu/lb _m °F ≈ 4.2 kJ/kg K
Enthalpy	Btu/lb _m kcal/kg _m	2 326 4 186.8	J/kg J/kg	
Dynamic (absolute) viscosity	centipoise (cP) poise (P) cP cP lb _m /ft h lb _m /ft h cP lb _m /ft s	0.001 0.1 1.000 1 000 0.0004134 0.4134 2.4189 1.4482	kg/m s Pa s mPa s μPa s Pa s cP lb _m /ft h Pa s	kg/m s = N s/m ² = Pa s water (100° C), 0.31 cP air (100° C), 0.021 cP
Kinematic viscosity	stoke (St), cm ² s centistoke (cSt) ft ² /s	0.0001 10 ⁻⁶ 0.092903	m ² /s m ² /s m ² /s	
Diffusivity	ft ² /s	0.092903	m ² /s	
Thermal diffusivity	m ² /h ft ² /s ft ² /h	0.0002778 0.092903 25.81 × 10 ⁻⁶	m ² /s m ² /s m ² /s	
Surface tension	dyne/cm dyne/cm lb _f /ft	0.001 6.852 × 10 ⁻⁵ 14.954	N/m lb _f /ft N/m	
Temperature relations:	°C = $\frac{5}{9}$ [°F - 32] °F = $\frac{9}{5}$ (°C) + 32	°C = (°F + 40) $\frac{5}{9}$ - 40 °F = (°C + 40) $\frac{9}{5}$ - 40	ΔT (°C) ≈ $\frac{9}{5}$ ΔT (°F) ΔT (°F) ≈ $\frac{5}{9}$ ΔT (°C)	K = °C + 273.15 R = °F + 459.67
Miscellaneous:	Acceleration of gravity (standard): Gas constant: Stefan-Boltzmann constant:	g = 9.806 65 m/s ² R = 8 314.3 m N/K kmol 5.669 7 × 10 ⁻⁸ W/m ² K ⁴ 1.714 × 10 ⁻⁹ Btu/ft ² h R ⁴		

^aEven though the abbreviations s and h were introduced only with the SI, they are used here throughout for consistency.

^bNote: the calorie and Btu are based on the International Standard Table values. The thermochemical calorie equals 4.184 J (exact) and is used in some older texts.

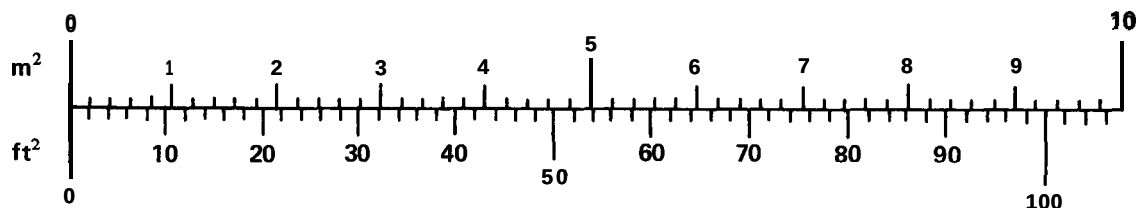
LENGTH: meter, m; foot, ft



$$\text{m} \times 3.2803 = \text{ft}$$

$$\text{ft} \times 0.3048 = \text{m}$$

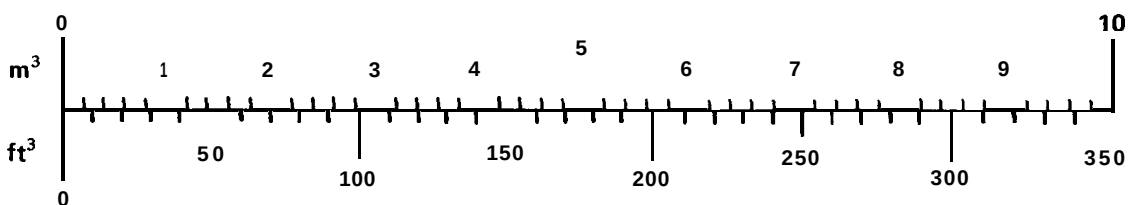
AREA: square meter, m^2 ; square foot, ft^2



$$\text{m}^2 \times 10.764 = \text{ft}^2$$

$$\text{ft}^2 \times 0.0929 = \text{m}^2$$

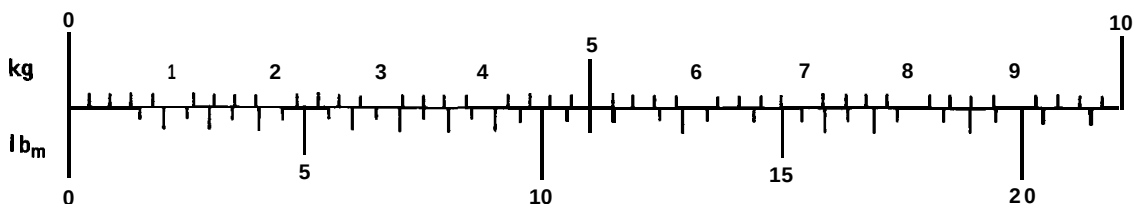
VOLUME: cubic meter, m^3 ; cubic foot, ft^3



$$\text{m}^3 \times 35.314 = \text{ft}^3$$

$$\text{ft}^3 \times 0.083 = \text{m}^3$$

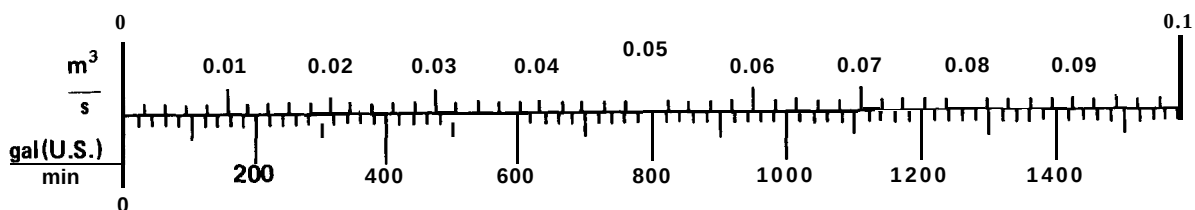
MASS: kilogram, kg; pound-mass, lb_m



$$\text{kg} \times 2.205 = \text{lb}_m$$

$$\text{lb}_m \times 0.454 = \text{kg}$$

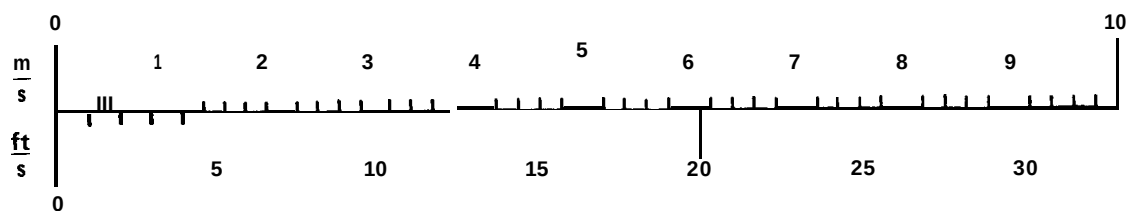
VOLUME FLOW RATE: $\frac{\text{m}^3}{\text{s}}$; $\frac{\text{gal(U.S.)}}{\text{min}}$



$$\frac{\text{m}^3}{\text{s}} \times 1.585(10^4) = \frac{\text{gal(U.S.)}}{\text{min}}$$

$$\frac{\text{gal(U.S.)}}{\text{min}} \times 6.31(10^{-5}) = \frac{\text{m}^3}{\text{s}}$$

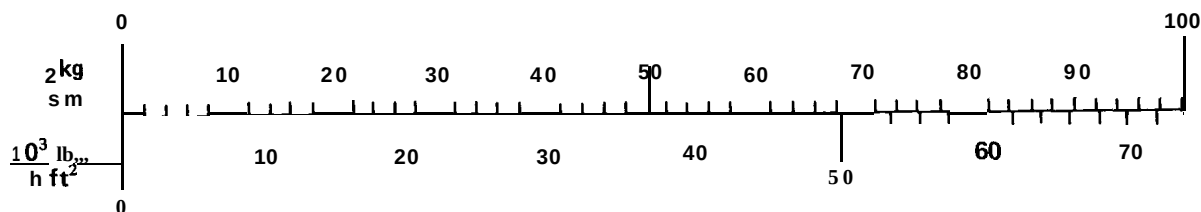
VELOCITY: $\frac{\text{m}}{\text{s}}$; $\frac{\text{ft}}{\text{s}}$



$$\frac{\text{m}}{\text{s}} \times 3.28 = \frac{\text{ft}}{\text{s}}$$

$$\frac{\text{ft}}{\text{s}} \times 0.305 = \frac{\text{m}}{\text{s}}$$

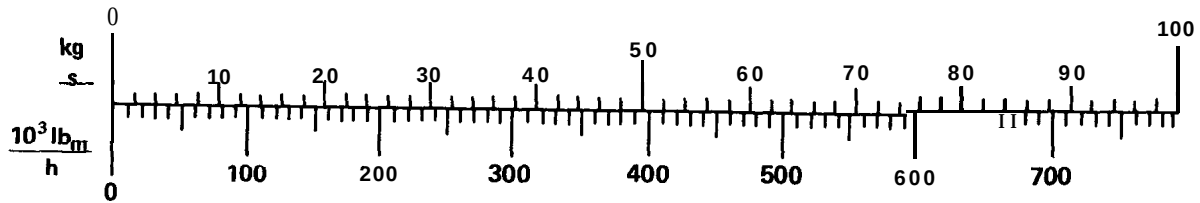
MASS VELOCITY: $\frac{\text{kg}}{\text{s m}^2}$; $\frac{10^3 \text{ lb}_m}{\text{h ft}^2}$



$$\frac{\text{kg}}{\text{s m}^2} \times 0.737 = \frac{10^3 \text{ lb}_m}{\text{h ft}^2}$$

$$\frac{10^3 \text{ lb}_m}{\text{h ft}^2} \times 1.356 = \frac{\text{kg}}{\text{s m}^2}$$

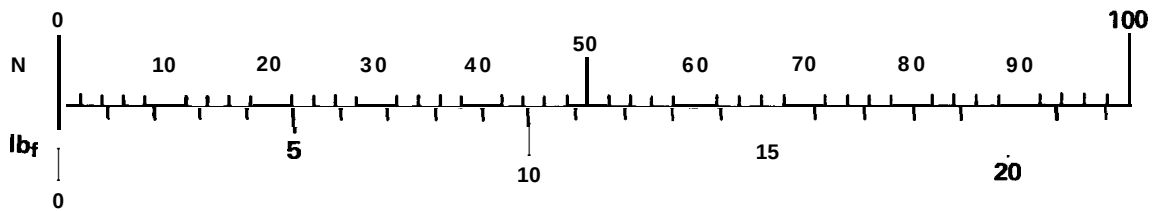
MASS FLOW RATE: $\frac{\text{kg}}{\text{s}}; \frac{10^3 \text{ lb}_m}{\text{h}}$



$$\frac{\text{kg}}{\text{s}} \times 7.937 = \frac{10^3 \text{ lb}_m}{\text{h}}$$

$$\frac{10^3 \text{ lb}_m}{\text{h}} \times 0.126 = \frac{\text{kg}}{\text{s}}$$

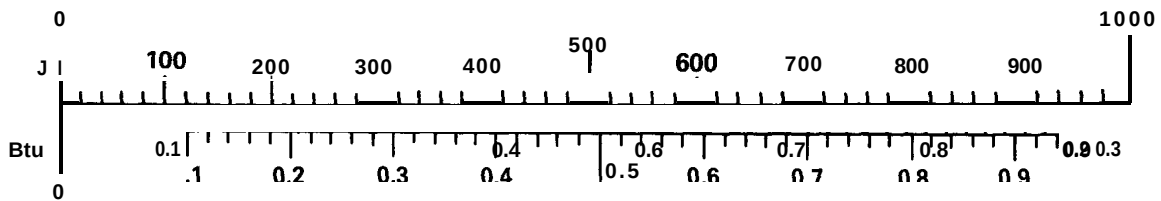
FORCE: newton, $\text{N} = \frac{\text{kg m}}{\text{s}^2}$; pound-force, lb_f



$$\text{N} \times 0.225 = \text{lb}_f$$

$$\text{lb}_f \times 4.448 = \text{N}$$

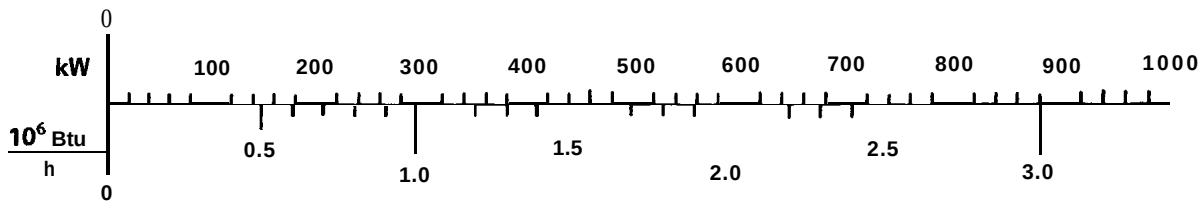
ENERGY (WORK): joule, $\text{J} = \text{N m}$; Btu



$$\text{J} \times 0.948(10^{-3}) = \text{Btu}$$

$$\text{Btu} \times 1055 = \text{J}$$

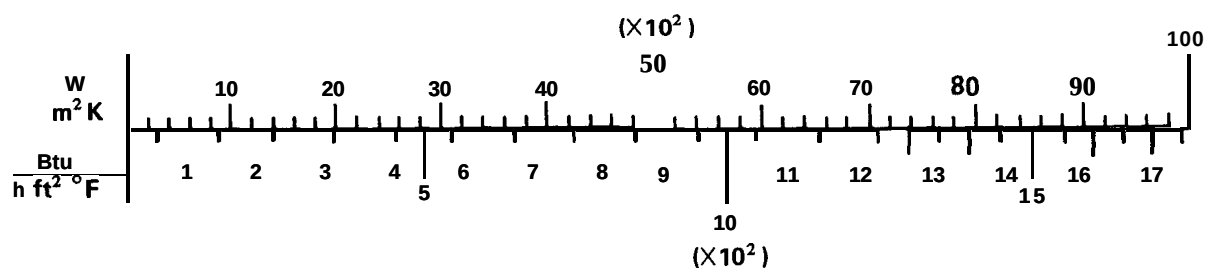
POWER OR ENERGY FLOW RATE: kilowatt, $\text{kW} = \frac{\text{kJ}}{\text{s}}; \frac{10^6 \text{ Btu}}{\text{h}}$



$$\text{kW} \times 3.412(10^{-3}) = \frac{10^6 \text{ Btu}}{\text{h}}$$

$$\frac{10^6 \text{ Btu}}{\text{h}} \times 293 = \text{kW}$$

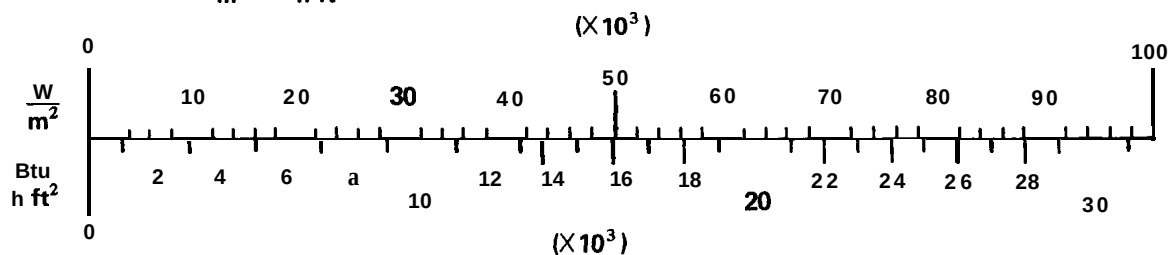
HEAT TRANSFER COEFFICIENT: $\frac{W}{m^2 K}; \frac{Btu}{h ft^2 ^\circ F}$



$$\frac{W}{m^2 K} \times 0.176 = \frac{Btu}{h ft^2 ^\circ F}$$

$$\frac{Btu}{h ft^2 ^\circ F} \times 5.68 = \frac{W}{m^2 K}$$

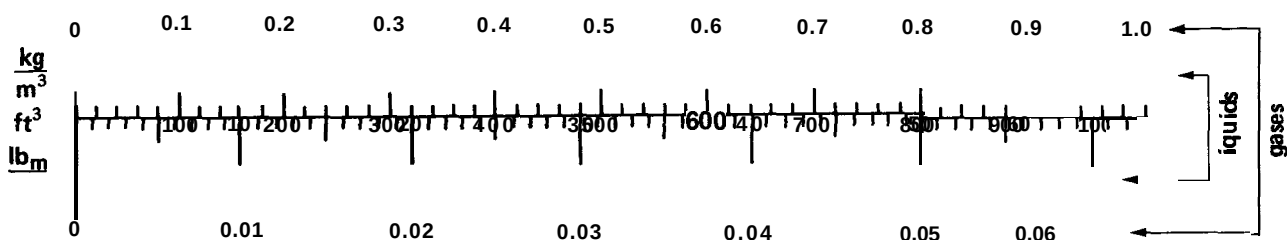
HEAT FLUX: $\frac{W}{m^2}; \frac{Btu}{h ft^2}$



$$\frac{W}{m^2} \times 0.316 = \frac{Btu}{h ft^2}$$

$$\frac{Btu}{h ft^2} \times 3.16 = \frac{W}{m^2}$$

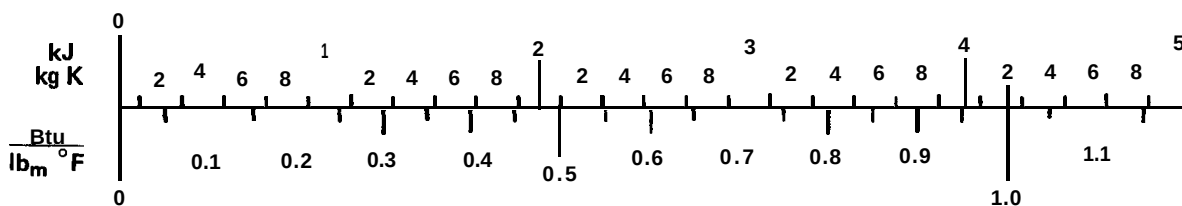
DENSITY $\frac{kg}{m^3}; \frac{lb_m}{ft^3}$



$$\frac{kg}{m^3} \times 0.0624 = \frac{lb_m}{ft^3}$$

$$\frac{lb_m}{ft^3} \times 16.02 = \frac{kg}{m^3}$$

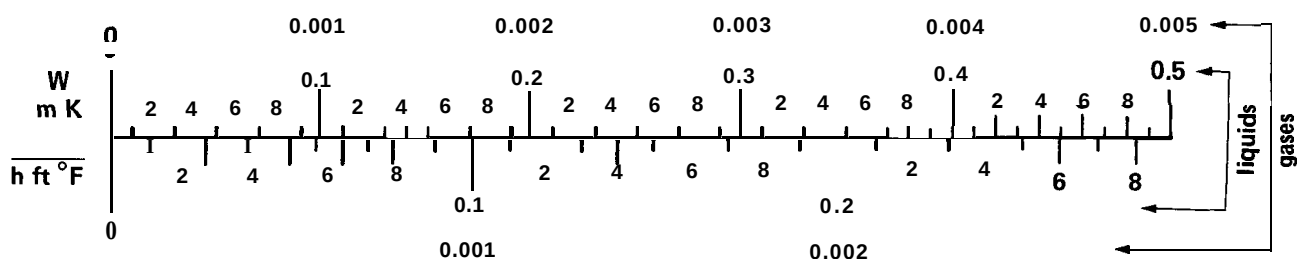
SPECIFIC HEAT CAPACITY: $\frac{\text{kJ}}{\text{kg K}}; \frac{\text{Btu}}{\text{lb}_m \text{ } ^\circ\text{F}}$



$$\frac{\text{kJ}}{\text{kg K}} \times 0.239 = \frac{\text{Btu}}{\text{lb}_m \text{ } ^\circ\text{F}}$$

$$\frac{\text{Btu}}{\text{lb}_m \text{ } ^\circ\text{F}} \times 4.187 = \frac{\text{kJ}}{\text{kg K}}$$

THERMAL CONDUCTIVITY: $\frac{\text{W}}{\text{m K}}; \frac{\text{Btu}}{\text{h ft } ^\circ\text{F}}$

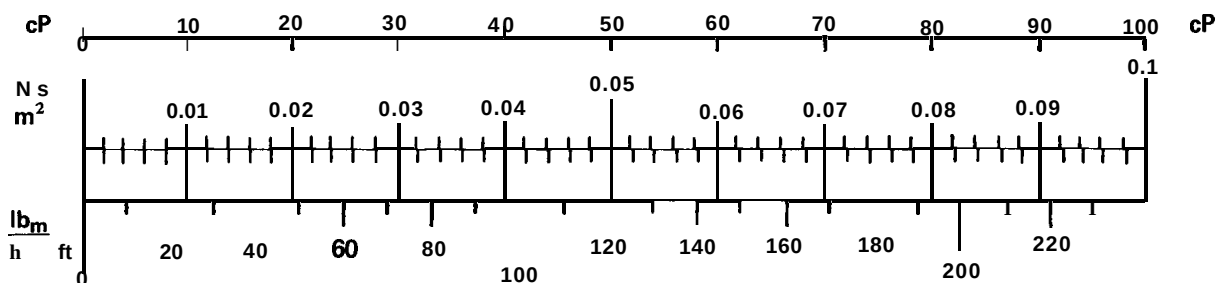


$$\frac{\text{W}}{\text{m K}} \times 0.578 = \frac{\text{Btu}}{\text{h ft } ^\circ\text{F}}$$

$$\frac{\text{Btu}}{\text{h ft } ^\circ\text{F}} \times 1.731 = \frac{\text{W}}{\text{m K}}$$

DYNAMIC (ABSOLUTE) VISCOSITY: $\frac{\text{N s}}{\text{m}^2}; \frac{\text{lb}_m}{\text{h ft}}$

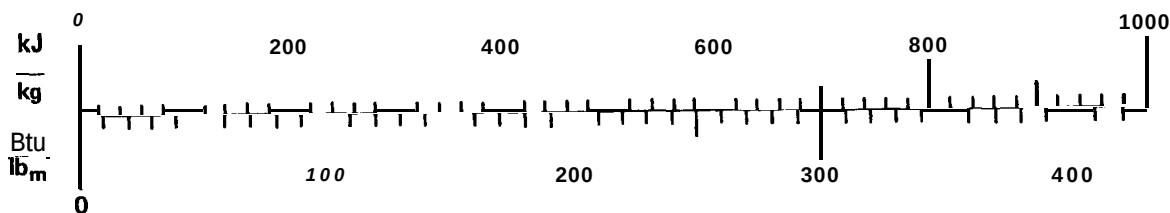
centipoise $\text{cP} = 10^{-3} \frac{\text{N s}}{\text{m}^2} = \text{mPa s}$



$$\text{cP} \times 2.42 = \frac{\text{lb}_m}{\text{h ft}}$$

$$\frac{\text{lb}_m}{\text{h ft}} \times 0.413 = \text{cP}$$

ENTHALPY: $\frac{\text{kJ}}{\text{kg}}$, $\frac{\text{Btu}}{\text{lb}_m}$

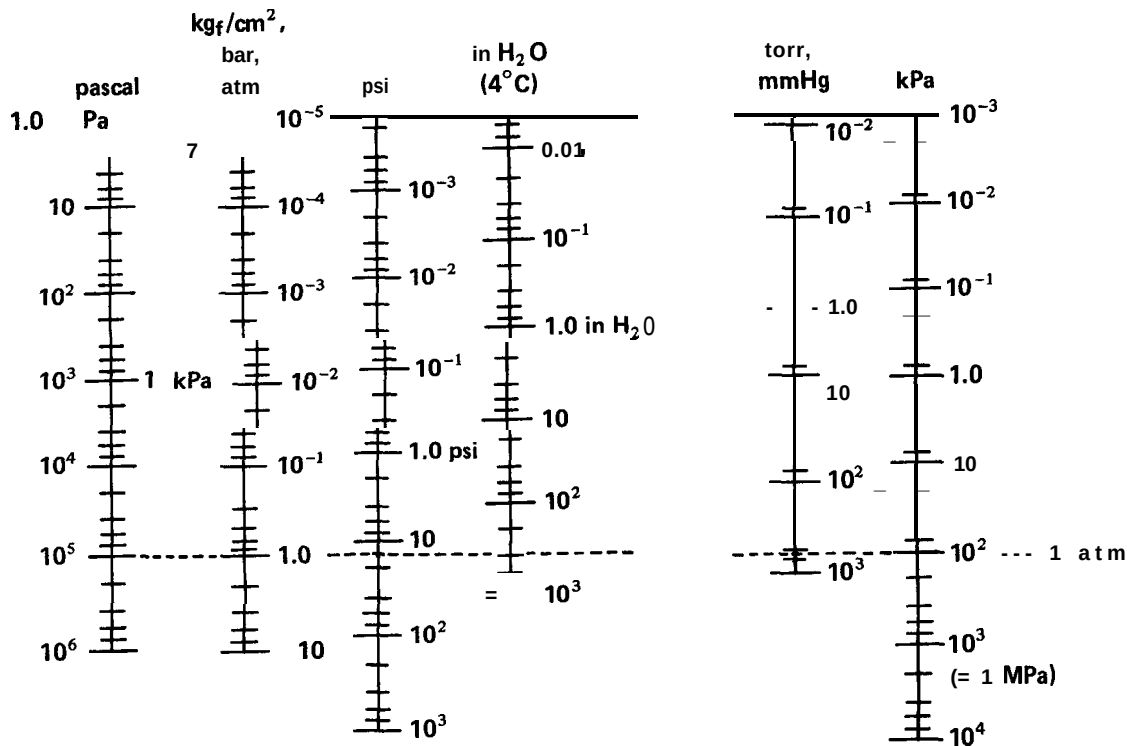


$$\frac{\text{kJ}}{\text{kg}} \times 0.430 = \frac{\text{Btu}}{\text{lb}_m}$$

$$\frac{\text{Btu}}{\text{lb}_m} \times 2.326 = \frac{\text{kJ}}{\text{kg}}$$

PRESSURE CONVERSION CHART

(top line = 1 Pa on all charts)

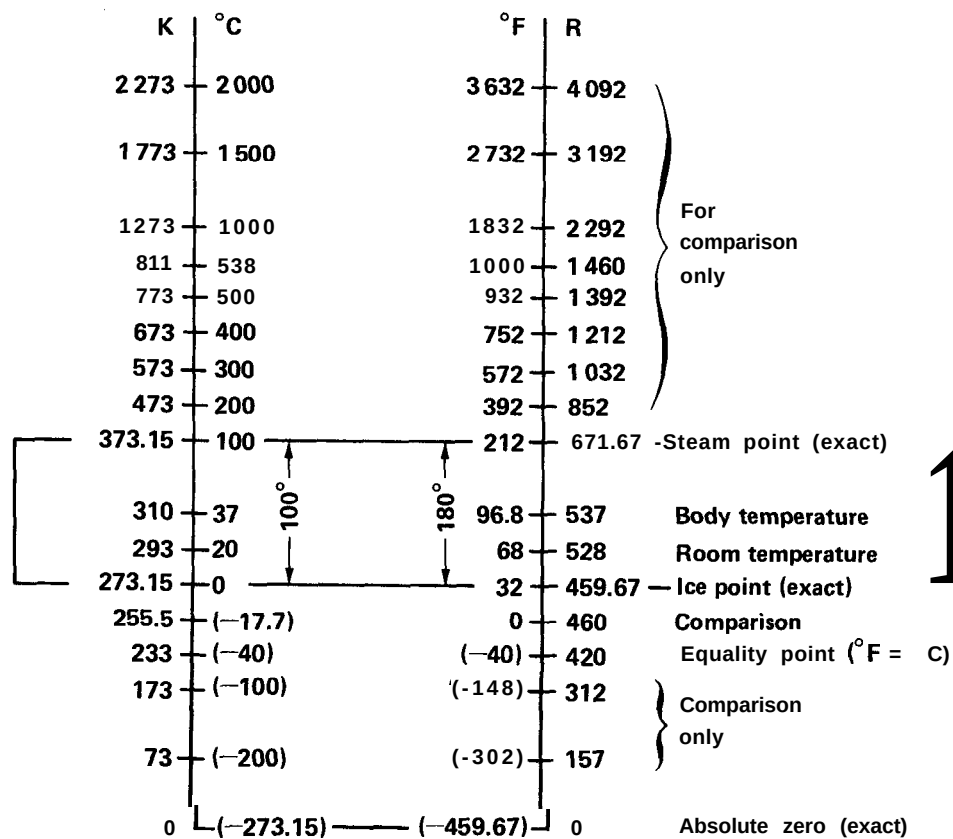


*NOTE: only bar is exact (1 bar = 10⁵ Pa); however, atm and kgf/cm² are within 2% accuracy (see list below).

1 atm = 1.01325 X 10⁵ Pa
 1 psi = 6.89476 X 10³ Pa
 1 in H₂O = 2.49082 X 10² Pa
 1 kgf/cm² = 9.80665 X 10⁴ Pa
 1 bar = 10⁵ Pa
 1 mmHg = 1.33322 X 10² Pa

1 Pa = 9.86923 X 10⁻⁶ atm
 1 Pa = 1.45038 X 10⁻⁴ psi
 1 Pa = 4.01474 X 10⁻³ in H₂O
 1 Pa = 1.01972 X 10⁻⁵ kgf/cm²
 1 Pa = 10⁻⁵ bar
 1 Pa = 7.50064 X 10⁻³ mmHg

TEMPERATURE COMPARISON
(NOT TO SCALE)



$$T(^{\circ}\text{F}) = (^{\circ}\text{C} + 40) \frac{9}{5} - 40$$

$$T(^{\circ}\text{C}) = (^{\circ}\text{F} + 40) \frac{5}{9} - 40$$

$$\Delta T : ^{\circ}\text{C} = \frac{5}{9} (^{\circ}\text{F}) \quad ^{\circ}\text{F} = \frac{9}{5} (^{\circ}\text{C})$$

$$T(\text{R}) = T(^{\circ}\text{F}) + 460$$

$$T(\text{K}) = T(^{\circ}\text{C}) + 273.15$$

4

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HEDH

GENERAL PREFACE

Ernst U. Schlünder

■ The idea to create a Heat Exchanger Design Handbook originated at the seminar Recent Developments in Heat Exchangers, organized by the International Centre for Heat and Mass Transfer (ICHMT) in Trogir, Yugoslavia in 1972.

Even though much heat transfer data and correlations were available, this information was scattered throughout the open literature or buried in the files of proprietary organizations. For manufacturers, vendors, and users of heat transfer equipment it was becoming more and more difficult not only to keep up with the continually growing heat transfer information but also to compare data and correlations from various sources. In addition, these sources were often inconsistent, and it was difficult for users to decide which data and correlations would serve their own purposes best. Moreover, over the years many heat transfer publications had become more analytical and more academic, which delayed their immediate application by practicing heat transfer engineers.

Because of this situation, a group of heat transfer specialists decided to collect heat transfer information available in the existing open literature and also-as far as possible-from proprietary sources in order to create a heat transfer information base. Recommended correlations based on selected experimental material would then be provided for all heat exchanger design data.

Although there was great enthusiasm for this project, not much progress was made until 1974, at the Fifth international Heat Transfer Conference in Tokyo. The idea was formulated then of publishing the information base under the auspices of the ICHMT by Hemisphere Publishing Corporation, the publisher of the

Centre. Eight representatives of universities, industry, and proprietary organizations formed the Editorial Board, which immediately started working on the scope and organization of this Handbook. As a result of these joint efforts the following five parts were planned:

Part 1 presents the theory of heat exchangers defining and explaining all parameters and concepts needed for thermal and hydraulic design and rating.

Part 2 contains recommended correlations for design and rating parameters based on evaluated available information. As far as possible experimental data are presented together with the recommended correlations in order to reveal both the consistency of the data as well as the reliability of the correlations.

Part 3 shows how to apply the material given in Parts 1 and 2 when solving actual heat exchanger design and rating problems. Because of the volume of material, it would be impossible for this part to cover all known kinds of heat exchangers. Nevertheless, most of the commonly used types of equipment are discussed in detail together with related installations such as cooling towers and combustion chambers.

Part 4 is devoted to the mechanical design of heat exchangers, which actually cannot be separated from thermal and hydraulic design. A useful comparison of some major national standards is included.

Part 5 contains physical property data that are needed in design calculations in the other four parts.

For each of the five parts, a member of the Editorial Board agreed to act as Lead Editor. Although each Lead Editor accepted additional administrative responsibility for his particular part, the whole Handbook has been a

cooperative effort with each editor sharing the reviewing and editorial work for all the material.

A large number of authors—well-known experts in their fields—have contributed to this Handbook. Each manuscript went through a review procedure and many of them were discussed during the regular meetings of the Editorial Board, which have been held every six months since 1975.

Nevertheless, this Handbook must be considered as a first step rather than as a final result. Though the concept was clear, the realization has not been easy and there is still room for improvement. Additionally, because new heat transfer data and correlations are published each year, a permanent updating procedure has been established to keep this Handbook current. This supplementary material will be offered regularly to the

subscribers. To ensure that any future material (including personal remarks and comments) can be added easily, a loose-leaf ring binder format was chosen for the Handbook.

The users of this Handbook are urged to communicate with us about what is missing and to make suggestions for improving the contents of the book and its applicability. This kind of interaction and cooperation between users and editors not only should improve the quality of the Handbook but also should stimulate further research work guided by the requirements of heat exchanger design.

The Editorial Board wishes to thank the publisher as well as the contributing authors for their most valuable cooperation.

4

Preface

Duncan Chisholm

■ Successful design of a heat transfer plant involves, during the design phase, an appreciation on the part of the designer of both thermal and mechanical aspects of design. It is the purpose of Part 4 of the Handbook to give the reader a broad appreciation of the latter aspects of the design process.

Part 4 begins with discussion of the basic principles involved in mechanical design, such as stress-strain relations, stress analysis, **thermal** stresses, and stress relieving.

The greater number of sections in this part relate to the mechanical design of shell-and-tube heat exchangers. This is by no means inappropriate: over 90% of heat exchangers in both value and number (excluding the automobile radiator) are of this type. The elements of their construction are discussed in Sec. 4.2.

With few exceptions, shell-and-tube heat exchangers are designed to national codes. Considerable thought was given to the role of the present volume in relation to these codes. It was finally decided to provide an index to the principal international codes, and a guide to the other leading national codes. The objective is to highlight the important clauses of the codes on a heat exchanger element-by-element basis.

Most other forms of heat exchanger tend to be proprietary, and are thus less likely to be designed by

the general heat exchanger designer. For that reason they are given a somewhat less exhaustive treatment than in the case of the shell-and-tube exchanger. Air coolers, plate heat exchangers, and plate fin heat exchangers, among others, are discussed in Sec. 4.4. The problem of material selection both from the viewpoint of strength and corrosion resistance is discussed in Sec. 4.5.

In recent years, the increasing size of heat exchangers, with the associated greater lengths between supports, has resulted in an increased incidence of failures due to vibration. This important topic is discussed in Sec. 4.6. The engineers' responsibility for the heat exchanger does not terminate when the design leaves the design office. It seems appropriate therefore to include a contribution (Sec. 4.7) on the various requirements for testing and inspection required before plant commissioning.

Finally, Part 4 of the Handbook concludes with a number of contributions related to the important topic of the costing of heat exchangers. In conclusion I would like to thank the contributors both for the quality of their contributions and for the expeditious manner in which they were completed.

HEDH NOMENCLATURE

■ The most frequently used symbols for quantities in heat and mass transfer are listed here according to the following principles:

1. Capital Roman letters (not italic) denote absolute quantities expressed in either SI basic units or derived units, e.g., $N = \text{kg m/s}^2$, $F = \text{N m}$, $W = \text{J/s}$.

Exceptions: Time (t), variables, coordinates (x , y , z , r), characteristic lengths in dimensionless numbers (l , d , s).

2. Lowercase Roman letters (not italic) denote the corresponding specific quantities per unit mass, mole, length, area, or volume (but not time). A dot over the symbol denotes an absolute or specific quantity per unit time.

Exception: Velocity (u , v , w).

A function will be indicated by $f(\dots)$.

Molar quantities are denoted by a tilde ($\sim$) over the symbol.

Exceptions: Gas constant ($\tilde{R}$), Loschmidt number ($\tilde{L}$), molecular weight ($\tilde{M}$), chemical potential (μ).

3. Lowercase Greek letters are used for coefficients and physical properties.

Exceptions: Overall heat transfer coefficient (U), heat capacity (c_p , c_v).

Capital Greek letters refer to eddy (turbulent) quantities; e.g., A denotes eddy thermal conductivity.

4. Dimensionless variables should be denoted by a superscript * following the corresponding symbol, e.g., y^* for a dimensionless coordinate.

5. Decimal fractions or multiples of the SI units may be denoted by the corresponding symbol preceding the SI unit, e.g., $MW = 10^6 \text{ W}$.

Basic Quantities

Quantity	Symbol	Unit
Heat rate	$\dot{Q}$	W
Heat quantity	$Q = \int \dot{Q} dt$	J
Heat flux	$\dot{q} = \dot{Q}/A$	W/m ²
Time	t	s
Frequency	f	1/s
Mass	M	kg
Mass flow rate	$\dot{M}$	kg/s
Mass velocity (mass flow per unit area)	$\dot{m} = \dot{M}/S$	kg/m ² s
Mass flux	$\dot{m}_n = \dot{M}_n/A$	kg/m ² s
Number of moles	N	mol
Molar mass flow rate	$\dot{N}$	mol/s
Molar mass velocity	$\dot{n} = \dot{N}/S$	mol/m ² s
Molecular weight	$\tilde{M}$	g/mol
Density	ρ	kg/m ³
Concentration	c	kg/m ³
Molar density	$\tilde{\rho}$	mol/m ³
Molar concentration	$\tilde{c}$	mol/m ³
Mass fraction	x_i, y_i	kg _i /kg
Mass loading	X_i, Y_i	kg _i /kg _j
Molar mass fraction	$\tilde{x}_i, \tilde{y}_i$	mol _i /mol
Molar mass loading	$\tilde{X}_i, \tilde{Y}_i$	mol _i /mol _j
Volume	V	m ³
Volume flow rate	$\dot{V}$	m ³ /s
Velocity	u, v, w	m/s
Surface area	A	m ²
Cross-sectional area	S	m ²
Length or diameter	L, D, l, d, s	m
Coordinate	$\psi, \varphi, r, x, y, z$	
Void fraction	ϵ	—
Fraction of volume flow rate	$\dot{\epsilon}$	—
Quality	x	—
Flow quality	$\dot{x}$	—
Work	W	J

Transport coefficients

Quantity	Symbol	Unit
Heat transfer coefficient	α	W/m ² K
Mass transfer coefficient	β	m/s
Overall heat transfer coefficient	U	W/m ² K
Drag coefficient	ξ	
Friction factor	f	—
Emissivity	ϵ	—
View factor	φ_{12}	
Thermal conductivity	λ	W/m K
Diffusivity	δ	m ² /s
Kinematic viscosity	ν	m ² /s
Dynamic (absolute) viscosity	η	kg/m s
Thermal diffusivity	$\kappa \equiv \lambda/\rho c_p$	m ² /s
Surface tension	σ	N/m
Shear stress	τ	Pa (N/m ²)
Pressure	p	Pa (N/m ²)
Pressure drop	Δp	Pa (N/m ²)

Thermodynamic quantities

Quantity	Symbol*	Unit
Enthalpy (specific)	h	J/kg
Internal energy (specific)	u	J/kg
Gibbs function free enthalpy (specific)	$g = h - T s$	J/kg
Free energy (specific)	$f = u - T s$	J/kg
Entropy (specific)	s	J/kg K
Chemical potential	μ	J/kg
Specific heat capacity	c_p, c_v	J/kg K
Temperature	T	K, °C
Mean logarithmic temperature difference	ΔT_{LM}	K, °C

*For totals, letters are capitalized, H, U .

Chemical reactions

Quantity	Symbol	unit
Reaction enthalpy; phase-change enthalpy; latent heat	Δh	J/kg
Energy of activation	AE	J/kg
Equilibrium constant	K	Various
Rate constant	k	Various
Rate of reaction	$\dot{r}$	mol/m ³ s
Activity coefficient	γ_i	—
Fugacity coefficient	γ_g	—
Stoichiometric factors	ν_i	—

Physical constants

Quantity	Symbol	Unit
Gas constant	$\tilde{R}$	8.3 14 J/mol K
Loschmidt number	$\tilde{L}$	6.0252 · 10
Avogadro number	$\tilde{L}$	1 /mol
Stefan-Boltzmann constant	C_s	5.6697 · 10 ⁻⁸ W/m ² K ⁴
Acceleration of gravity	g	m/s ²
Standard acceleration of gravity	$g_n = 9.80665$	m/s ²

subscripts

Quantity	Symbol
Wall	w
Bulk	b
Mean	m
Solid	s
Liquid	l
Gas	g
Inlet	in
Outlet	out

Dimensionless numbers (most frequently used)

Name	Symbol and definition
Archimedes number	$Ar = g l^3 \Delta \rho / \nu^2 \rho = g l^3 \rho \Delta \rho / \eta^2$
Biot number	$Bi = \alpha_g l / \lambda_s$ or $\alpha_l l / \lambda_s$
Effectiveness	E
Euler number	$Eu = \Delta p / (\rho u^2 / 2)$
Fourier number	$ Fo = \kappa t / l^2 = \lambda t / \rho c_p l^2$
Froude number	$Fr = u^2 / g l$
Galileo number	$Ga = l^3 g / \nu^2 = l^3 g \rho^2 / \eta^2$
Grashof number	$Gr = g l^3 \Delta \rho / \nu^2 \rho = g l^3 \rho \Delta \rho / \eta^2$
Graetz number	$Gz = u d^2 / \kappa l = u d^2 \rho c_p / \lambda l = Re \ Pr \ d / l$
Knudsen number	$Kn = l_0 / l$, where l_0 is the molecular mean free path
Lewis number	$Le = \kappa / \delta = \lambda / \rho c_p \delta$
Mach number	$Ma = u / u_{\text{sound}}$
Number of transfer units	NTU
Nusselt number	$Nu = \alpha l / \lambda$
Péclet number	$Pe = ul / \kappa = ul \rho c_p / \lambda = Re \ Pr$
Phase change number	$Ph = c_p \Delta T / \Delta h_{\text{ph}}$
Prandtl number	$Pr = \nu / \kappa = \eta c_p / \lambda$
Rayleigh number	$Ra = Gr Pr$
Reynolds number	$Re = ul / \nu = ul \rho / \eta = \dot{m} l / \eta$
Schmidt number	$Sc = \nu / \delta = \eta / \rho \delta$
Sherwood number	$Sh = \beta l / \delta$
Stanton number	$St = \alpha / \rho u c_p = Nu / Re \ Pr$
Strouhal number	$Sr = fl / u$
Weber number	$We = u^2 \rho l / \sigma$

Dimensionless mass transfer numbers are distinguished from dimensionless heat transfer numbers by a prime ('), e.g., $Bi' = \beta l / \delta$. Exceptions: $Nu' = \beta l / \delta = Sh$ and $Pr' = \nu / \delta = \eta / \rho \delta = Sc$

HEDH

INTERNATIONAL SYSTEM OF UNITS (SI): RULES, PRACTICES, AND CONVERSION CHARTS

J. Taborek

■ HEDH is written in SI units because this system has been officially adopted by all countries and is replacing the older metric engineering system, the cgs system, as well as the U.S. customary system. In this section we shall review the basic rules for use of the SI. However, recognizing the fact that many references are written in other unit systems, comprehensive unit conversion charts are included.

A. Basic SI units

SI is absolute in the sense that there are only seven fundamental, precisely defined basic units from which all others are derived. The following five units are important to heat transfer and fluid flow.

length (meter, m): defined in terms of krypton-86 wavelength

mass (kilogram, kg): the mass of an international prototype, approximately equal to the weight of 1 liter H_2O at $4^\circ C$

time (second, s): defined in terms of radiation frequency of the cesium-133 atom

temperature (kelvin, K): defined in terms of the triple point of water (ice point = 273.15 K)

amount of substance (mole, mol): amount of substance that contains as many elementary entities as there are atoms in 0.012 kg of carbon-12. The elementary entities must be identified when the mole is used (atoms, molecules, ions).

B. Derived SI units with special names

Because SI is defined by the basic units only, there is a need for establishing definitions for the most frequently used composite or derived units, mainly for the purpose of simplifying unit notation. Thus, for example, newton, N, replaces $kg\ m/s^2$.

There are a number of derived units that have special names, signifying their importance: newton, joule, watt, pascal, and many others. While the unit symbols are capital letters (N, J, W, Pa), the spelled out unit names begin with lowercase letters, that is, newton, joule, watt, and pascal.

The most frequently used derived SI units with special names in the area of heat transfer and fluid flow are shown in Table 1.

Table 1 Most frequently used derived SI units

Quantity	Unit	Symbol	Dimension
Force	newton	N	$kg\ m/s^2$
Energy, work, heat (also power $\times$ time)	joule	J	$N\ m$ or $kg\ m^2/s^2$ $W\ s = J\ s/s$
Power	watt	W	$J/s = kg\ m^2/s^3$
Pressure	pascal	Pa	$N/m^2 = kg/m\ s^2$
Frequency	hertz	Hz	$1/s$
Dynamic viscosity	poise centipoise	P cP	$0.1\ Pa\ s$ $mPa\ s$

C. Prefixes and specification rules

The magnitude of the basic units may often not fit the practice of specific applications without using a large number of decimal digits or power of ten multipliers. SI provides for this by assigning prefixes to the basic units, as shown in Table 2.

Thus, for example, to express very high pressures (stresses) in Pa, the megapascal, **MPa**, is used instead of 10^6 Pa. The preferred graduation is in powers of 10^3 , and there are only a few exceptions to this rule, such as for units that are deeply ingrained in common use, such as centimeter. Some of these are discussed later.

(a) Punctuation and spelling rules

The following general rules should be carefully followed to maintain SI consistency.

1. **Prefixes** are attached directly to the base unit symbol or a derived unit symbol with a special name, such as km for kilometer or MW for megawatt.

2. Prefixed units are spelled out in text as one word, without hyphen, such as meganewton, not **mega-newton**.

3. The multiplication of base units can be expressed by leaving a space between the symbols, such as N s for newton X second. However, for better clarity, a center dot is often used, for example, N·s. This usage becomes especially important when the symbol m is used, which can stand for milli or for meter. Thus **mN** (millinewton) is clearly distinguished from **m·N** (meter X newton).

4. A prefix applied to a base unit constitutes a new unit. When a power is applied to such a unit, the exponent operates on the prefixed unit as an entity, i.e., $\text{mm}^2 = (\text{mm})^2$.

5. The unit of length in SI is correctly spelled **metre** (French spelling). This rule will be hard to enforce in U.S. literature, however, and **meter** will probably be more commonly used.

Table 2 SI prefixes

Factor	Prefix	Symbol	Status
10^{12}	tera	T	Preferred
10^9	giga	G	Preferred
10^6	mega	M	Preferred
10	kilo	k	Preferred
10^2	hecto	h	Accepted
10	deka	da	Accepted
10^{-1}	deci	d	Accepted
10^{-2}	centi	c	Accepted
10^{-3}	milli	m	Preferred
10^{-6}	micro	μ	Preferred
10^{-9}	nano	n	Preferred
10^{-12}	pico	p	Preferred

6. Multiple **prefixes** are prohibited. This presents some problems with the unit kilogram (kg), where the k stands for the **prefix kilo** and is one of the very few inconsistencies in SI (a carryover from the older cgs system). Thus k (for kilo) cannot be used in connection with kg, as this would result in the unacceptable term kkg. Equally incorrect is the use of Mg (megagram), as gram is not a basic SI unit. Thus, the unit used is 10^3 kg or the equivalent unit, metric ton (see below).

Similarly, mkg is not an acceptable way to express 10^{-3} kg, and the more common alternate unit of gram (g) is used.

D. Alternate units and supplementary comments

There are as many derived units as the technology demands, as SI is particularly suitable for this purpose by the clearly defined unit relations. The Nomenclature of this handbook contains practically all such units used in heat transfer and fluid flow, showing the appropriate SI dimensions. Consequently, these are not repeated here.

In some cases, strict SI rules are relaxed to respect generally accepted and deeply established practices. The following list presents most frequently used exceptions, together with some additional comments to alternate units, their definition, and their usage.

length: The unit centimeter (cm) is a very practical measure for everyday usage (person's height, 6 ft $\approx$ 183 cm), and will probably remain in perpetual use.

mass: The unit of kilogram is often too small for expressing quantities of mass experienced in many engineering applications. One of the few weaknesses of SI is the lack of a well-defined alternate unit of mass. As explained above, the prefixed units (kkg, Mg) are not acceptable. Instead, an alternate unit for large quantities of mass is introduced, designated metric ton (symbol t), $t = 1\,000$ kg. While this unit has not yet been accepted as a recognized **SI** unit, it is very much in common use.

In the U.S. customary lb.-based system, several definitions of a unit of mass similar in magnitude to the metric ton exist. In engineering practice, the following two definitions of the **lb_m-based** ton are most frequently used:

short ton, which is equal to 2 000 lb, (exact) and therefore approximately = 907.2 kg

long ton, which is equal to 2 240 lb, (exact), and therefore approximately = 1 016 kg

It is important to be aware of these differences in the inconsistently defined U.S. customary system.

For small quantities of mass, gram (g) is also often used in place of 10^{-3} kg.

time: The unit second (s) should be used whenever possible, as the alternates of minutes, hours, and days are nondecimal entities, and thus complicate the system.

However, to express large quantities of mass per unit of time, the unit hour (h), 3 600 seconds, or the unit day (d), 86 400 seconds, must be used sometimes. The time unit minute (min), often used in the U.S. customary system (**gal/min**), has no real justification, except that it is ingrained in some industrial practices. Otherwise, the differential quantity of a minute, as compared to a second, can be readily expressed by prefixes in **SI** units, to obtain a number with reasonable digits.

force: The newton is a very small unit of force, approximately equal to the weight of water in a small glass. Thus the more practical units of force are often the **kN** or **MN**.

volume: The liter, defined as dm^3 , is a very popular unit of volume for everyday use, such as for water and gasoline. Liter (L) is accepted as a common alternate unit.

amount of substance: The frequently used units of the metric engineering system, gram-mol and kg-mol, are designated in SI as follows: 1 gram-mol = 1 mol (SI), and 1 kg-mol = 1 kmol (SI). Thus 1 mol = 6.023×10^{23} molecules.

pressure: The basic unit of pressure, pascal ($\text{Pa} = \text{N/m}^2$), is a very small quantity, equal to the pressure exerted by about 0.1-mm layer of water over 1 m^2 . Thus in most engineering applications **kPa** or **MPa** is more useful.

Many derived units of pressure are firmly established in various branches of engineering and will probably remain in use indefinitely. The most important metric-based units are:

bar: has exact relationship to SI ($\approx 100 \text{ kPa}$)

torr = 1 mmHg = 0.133 kPa: used for gas pressure measurements

mm water = 9.8067 Pa: used for very small gas pressure measurements

at (technical atmosphere) = $\text{kgf/cm}^2 = 98.066 \text{ kPa}$

atm (normal or barometric atmosphere) = 760 mmHg = 101.3 kPa

temperature: In many engineering applications it is more convenient to use degrees Celsius rather than the absolute unit kelvin. The use of Celsius is permitted as an alternate, with the relationship

$$T(^{\circ}\text{C}) = T(\text{K}) - 273.16 \text{ and } \Delta T(^{\circ}\text{C}) = \Delta T(\text{K})$$

Table 3 Unit conversion

Physical quantity	Given in Gives	Multiplied by Divided by	Gives Given in	Approximate or useful relationship
Length	ft	0.3048	m	$3\frac{1}{4} \text{ ft} \approx 1 \text{ m}$
	in	25.4 (exact)	mm	$1 \text{ in} \approx 25 \text{ mm}$
	mil	0.0254	mm	
	yard	0.9144	m	
	mile (mi)	1609.3	m	$1 \text{ mi} \approx 1.6 \text{ km}$
	km	0.621388	mi	
Area	ft^2	0.092903	m^2	$100 \text{ ft}^2 \approx 9 \text{ m}^2$
	in^2	645.16	mm^2	$1 \text{ in}^2 \approx 650 \text{ mm}^2$
	acre	4 047.0	m^2	
Volume	ft^3	0.028317	m^3	$35 \text{ ft}^3 \approx 1 \text{ m}^3$
	U.S. gal	0.003785	m^3	$260 \text{ gal} \approx 1 \text{ m}^3$
	U.S. gal	3.785	liter (L)	$1 \text{ gal} \approx 3\frac{3}{4} \text{ L}$
	L (liter)	0.2642	U.S. gal	$1 \text{ L} \approx 0.26 \text{ gal}$
	Brit. gal	0.004546	m^3	
	U.S. gal	0.13368	ft^3	
	barrel (U.S. pet.)	0.15898	m^3	
	barrel (U.S. pet.)	42	U.S. gal	
Velocity	ft/s^a	0.3048	m/s	$10 \text{ ft/s} \approx 3 \text{ m/s}$
	m/s	3.2808	ft/s	
	ft/min	0.00508	m/s	$100 \text{ ft/min} \approx 0.5 \text{ m/s}$
	mi/h	1.6093	km/h	$30 \text{ mi/h} \approx 48 \text{ km/h}$
	km/h	0.6214	mi/h	$50 \text{ km/h} \approx 31 \text{ mi/h}$
	knots	1.852	km/h	

(See footnotes on page xxix.)

Table 3 Unit conversion (Continued)

Physical quantity	Given in Gives	Multiplied by Divided by	Gives Given in	Approximate or useful relationship
Mass	lb_m kg metric ton ton (2 000 lb.)	0.45 359 2.2046 2 204.6 907.18	kg lb_m lb_m kg	1 lb, r.45 kg 1 kg $\approx$ 2.2 lb, metric ton = 10^3 kg
Force	lb_f lb_f kg_f kg_f dyne	4.44822 0.45359 2.2046 9.80665 0.00001 (exact)	N = kg m/s ² kg_f lb_f N N	1 N $\approx$ 0.1 kg _f $\approx$ 0.22 lb _f
Amount of substance	lb,-mol g-mol kg-mol mol	453.6 1.000 1.000 1 000	kmol mol kmol kmol	
Mass flow rate	lb_m/h kg/s lb_m/s lb_m/min	0.0001260 7 936.51 0.4536 0.00756	kg/s lb_m/h kg/s kg/s	10^3 lb/h = .13 kg/s
Volume flow rate	U.S. gal/min U.S. bbl/day U.S. bbl/day ft³/s ft³/min	6.309×10^{-5} 0.15899 1.84×10^{-6} 0.02832 0.000472	m³/s m³/day m³/s m³/s m³/s	
Mass velocity (mass flux)	lb_m/h ft² kg/s m ²	1.356×10^{-3} 737.5	kg/s m ² lb_m/h ft²	
Energy (work) (heat)	Btu^b Btu Btu kcal ft lb_f Wh	1 055.056 0.2520 778.28 4 186.8 1.3558 3 600	J = N m = W s kcal ft lb_f J J J	1 Btu $\approx$ 1 000 J 1 kcal $\approx$ 4 Btu 1 kcal $\approx$ 4 000 J
Power	Btu/h W kcal/h ft lb_f/s hp (metric) Btu/h tons refriger.	0.2931 3.4118 1.163 1.3558 735.5 0.2520 3 516.9	w = J/s Btu/h W W W kcal/h W	10^6 Btu/h $\approx$ 300 kW 1 000 kW $\approx$ 3.5×10^6 Btu/h
Heat flux	Btu/h ft² W/m² kcal/cm ² s	3.1546 0.317 41.868	W/m² Btu/h ft² W/m²	1 000 Btu/h ft ² $\approx$ 3.2 kW/m ²
Heat transfer coefficient	Btu/h ft² °F W/m² K kcal/cm ² s °C	5.6784 0.1761 41.868	W/m² K Btu/h ft² °F W/m² K	1 000 Btu/h ft ² °F $\approx$ 5 600 W/m ² K
Heat transfer resistance	(Btu/h ft² °F)⁻¹ (W/m² K)⁻¹	0.1761 5.6784	(W/m² K)⁻¹ (Btu/h ft² °F)⁻¹	0.001 (Btu/h ft ² °F) ⁻¹ $\approx$ 0.000 18 (W/m ² K) ⁻¹
Pressure	lb_f/in² (psi) kPa bar	6.8948 0.1450 100	kN/m² = kPa psi kPa	1 psi = 7 kPa 14.5 psi = 100 kPa

Gee footnotes on page xxix.)

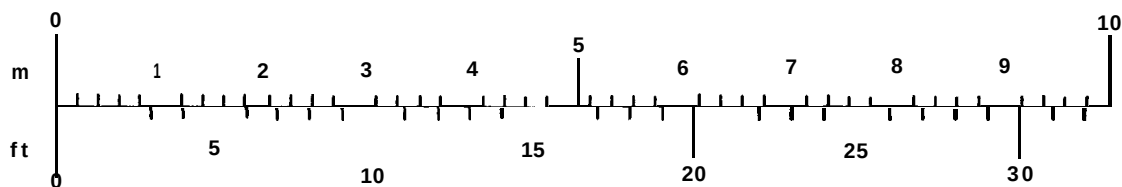
Table 3 Unit conversion (Continued)

Physical quantity	Given in Gives	Multiplied by Divided by	Gives Given in	Approximate or useful relationship
Pressure (Continued)	lb_f/ft² mm Hg (torr) in Hg mm H₂O in H₂O at (kg_f/cm²) atm (normal)	0.0479 0.1333 3.3866 9.8067 249.09 98.0665 101.325	kPa kPa kPa Pa Pa kPa kPa	1000 kPa = 1 MPa ≈ 150 psi 1 in H ₂ O ≈ .25 kPa atm = 760 mmHg
Mass flux	lb_m/ft² s lb_m/ft² h	4.8824 0.001356	kg/m² s kg/m² s	
Physical and Transport Properties				
Thermal conductivity	Btu/ft h °F W/m K kcal/m h °C	1.7308 0.5778 1.163	W/m K Btu/ft h °F W/m K	steel ≈ 50 W/m K water (20°C) ≈ 0.6 W/m K air (STP) ≈ 24 mW/m K
Density	lb_m/ft³ kg/m³ lb_m/U.S. gal	16.0185 0.06243 119.7	kg/m³ lb_m/ft³ kg/m³	62.4 lb _m /ft ³ ≈ 1 000 kg/m ³
Specific heat capacity	Btu/lb_m °F kcal/kg °C	4 186.8 4 186.8	J/kg K J/kg K	1 Btu/lb _m °F ≈ 4.2 kJ/kg K
Enthalpy	Btu/lb_m kcal/kg_m	2 326 4 186.8	J/kg J/kg	
Dynamic (absolute) viscosity	centipoise (cP) poise (P) cP cP lb_m/ft h lb_m/ft h cP lb_m/ft s	0.001 0.1 1.000 1 000 0.0004134 0.4134 2.4189 1.4482	kg/m s Pa s mPa s μPa s Pa s cP lb_m/ft h Pa s	kg/m s = N s/m ² = Pa s water (100°C), 0.31 cP air (100°C), 0.021 cP
Kinematic viscosity	stoke (St), cm² s centistoke (cSt) ft² /s	0.0001 10 ⁻⁶ 0.092903	m² /s m² /s m² /s	
Diffusivity	ft² /s	0.092903	m² /s	
Thermal diffusivity	m² /h ft² /s ft² /h	0.0002778 0.092903 25.81 × 10 ⁻⁶	m² /s m² /s m² /s	
Surface tension	dyne/cm dyne/cm lb_f/ft	0.001 6.852 × 10 ⁻⁵ 14.954	N/m lb_f/ft N/m	
Temperature relations:	°C = $\frac{5}{9} [°F - 32]$ °F = $\frac{9}{5} (°C) + 32$	°C = $(°F + 40) \frac{5}{9} - 40$ °F = $(°C + 40) \frac{9}{5} - 40$	ΔT(°C) = $\frac{5}{9} \Delta T(°F)$ ΔT(°F) = $\frac{9}{5} \Delta T(°C)$	K = °C + 273.15 R = °F + 459.67
Miscellaneous:	Acceleration of gravity (standard): Gas constant: Stefan-Boltzmann constant:	g = 9.806 65 m/s² R = 8 314.3 m N/K kmol 5.669 7 × 10⁻⁸ W/m² K⁴ 1. 7 14 × 10⁻⁹ Btu/ft² h R⁴		

^a Even though the abbreviations s and h were introduced only with the SI, they are used here throughout for consistency.

^b Note: the calorie and Btu are based on the International Standard Table values, The thermochemical calorie equals 4.184 J (exact) and is used in some older texts.

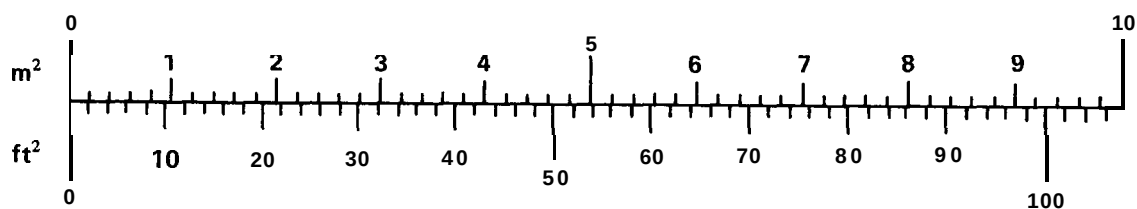
LENGTH: meter, m; foot, ft



$$\text{m} \times 3.2803 = \text{ft}$$

$$\text{ft} \times 0.3048 = \text{m}$$

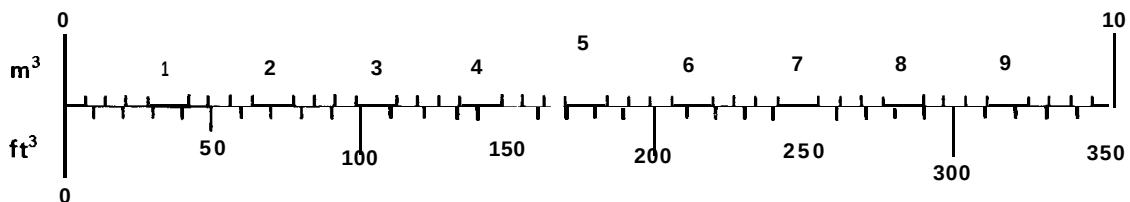
AREA: square meter, m^2 ; square foot, ft^2



$$\text{m}^2 \times 10.764 = \text{ft}^2$$

$$\text{ft}^2 \times 0.0929 = \text{m}^2$$

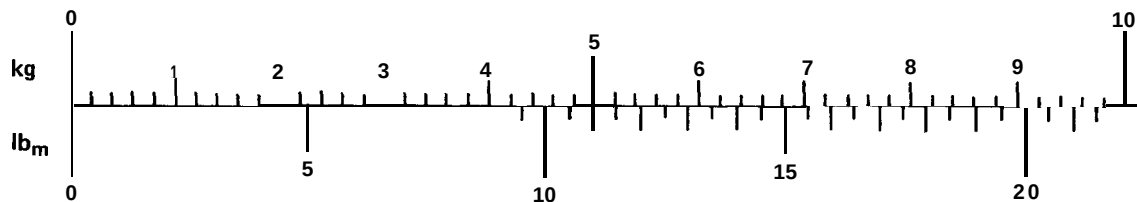
VOLUME: cubic meter, m^3 ; cubic foot, ft^3



$$\text{m}^3 \times 35.314 = \text{ft}^3$$

$$\text{ft}^3 \times 0.083 = \text{m}^3$$

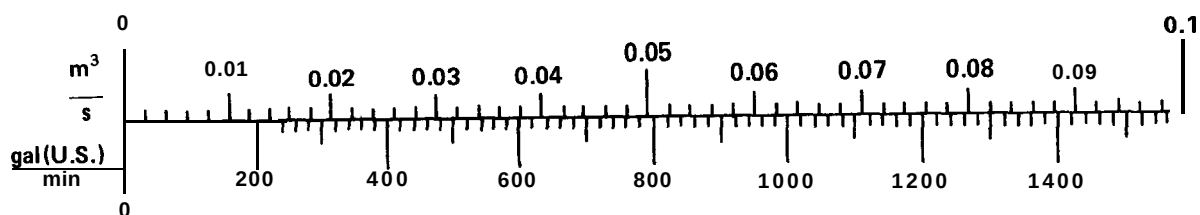
MASS: kilogram, kg; pound-mass, lb_m



$$\text{kg} \times 2.205 = \text{lb}_m$$

$$\text{lb}_m \times 0.454 = \text{kg}$$

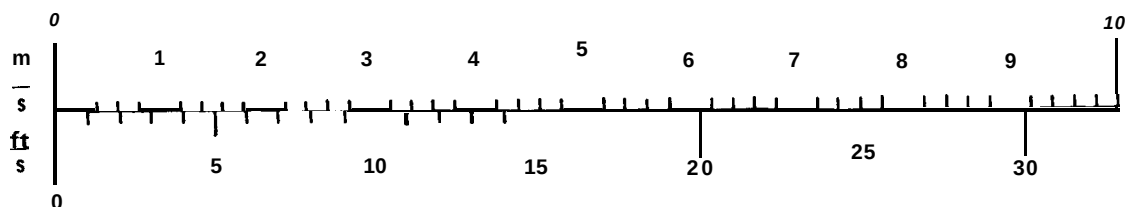
VOLUME FLOW RATE: $\frac{\text{m}^3}{\text{s}}$; $\frac{\text{gal(U.S.)}}{\text{min}}$



$$\frac{\text{m}^3}{\text{s}} \times 1.585(10^4) = \frac{\text{gal(U.S.)}}{\text{min}}$$

$$\frac{\text{gal(U.S.)}}{\text{min}} \times 6.31(10^{-5}) = \frac{\text{m}^3}{\text{s}}$$

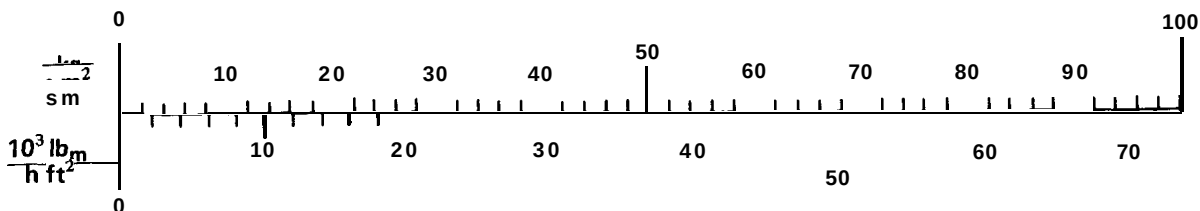
VELOCITY: $\frac{\text{m}}{\text{s}}$; $\frac{\text{ft}}{\text{s}}$



$$\frac{\text{m}}{\text{s}} \times 3.28 = \frac{\text{ft}}{\text{s}}$$

$$\frac{\text{ft}}{\text{s}} \times 0.305 = \frac{\text{m}}{\text{s}}$$

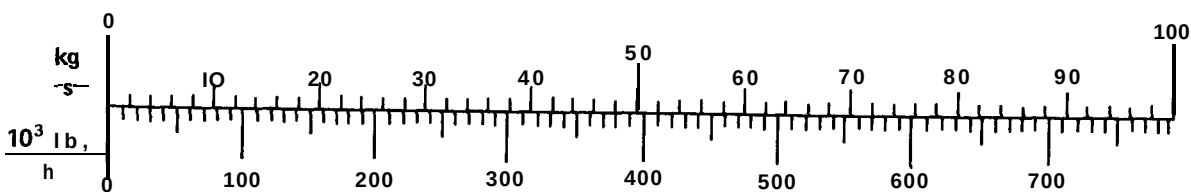
MASS VELOCITY: $\frac{\text{kg}}{\text{s m}^2}$; $\frac{10^3 \text{ lb}_m}{\text{h ft}^2}$



$$\frac{\text{kg}}{\text{s m}^2} \times 0.737 = \frac{10^3 \text{ lb}_m}{\text{h ft}^2}$$

$$\frac{10^3 \text{ lb}_m}{\text{h ft}^2} \times 1.356 = \frac{\text{kg}}{\text{s m}^2}$$

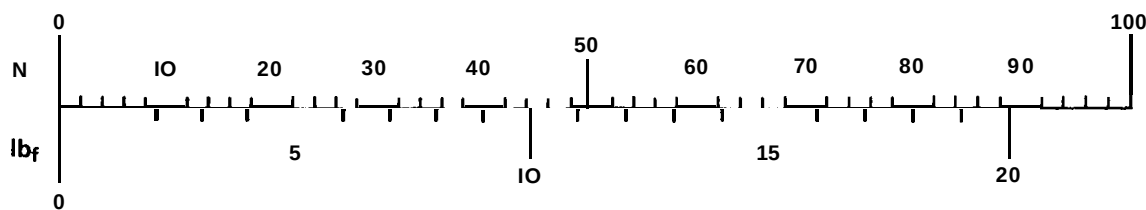
MASS FLOW RATE: $\frac{\text{kg}}{\text{s}}$; $\frac{10^3 \text{ lb}_m}{\text{h}}$



$$\frac{\text{kg}}{\text{s}} \times 7.937 = \frac{10^3 \text{ lb}_m}{\text{h}}$$

$$\frac{10^3 \text{ lb}_m}{\text{h}} \times 0.126 = \frac{\text{kg}}{\text{s}}$$

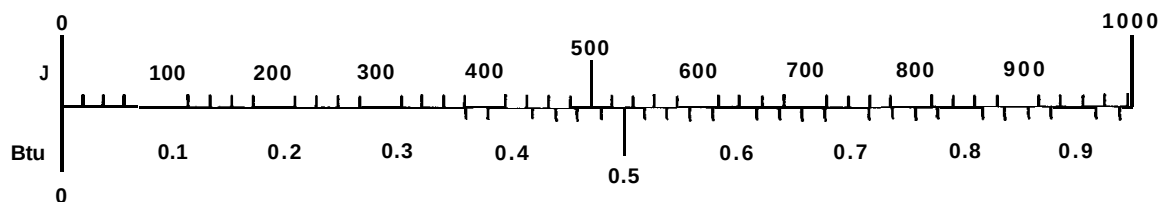
FORCE: newton, $\text{N} = \frac{\text{kg m}}{\text{s}^2}$; pound-force, lb_f



$$\text{N} \times 0.225 = \text{lb}_f$$

$$\text{lb}_f \times 4.448 = \text{N}$$

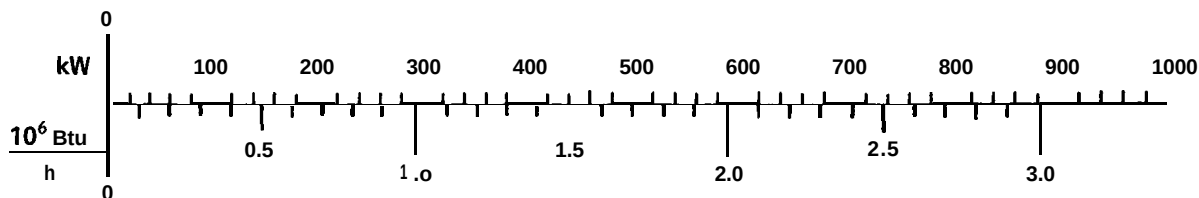
ENERGY (WORK): joule, $\text{J} = \text{N m}$; Btu



$$\text{J} \times 0.948(10^{-3}) = \text{Btu}$$

$$\text{Btu} \times 1055 = \text{J}$$

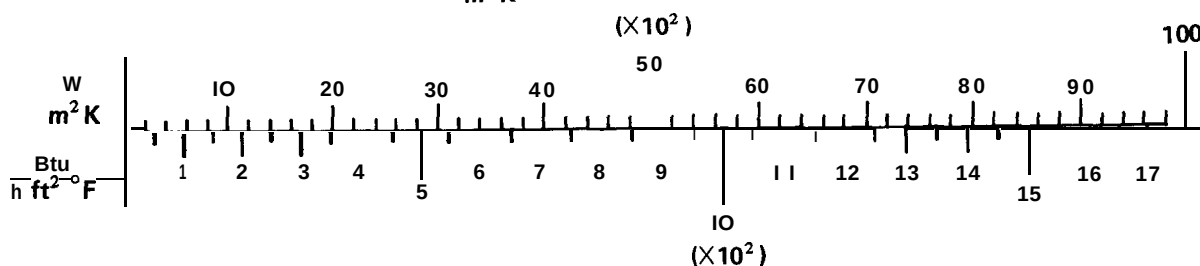
POWER OR ENERGY FLOW RATE: kilowatt, $\text{kW} = \frac{\text{kJ}}{\text{s}}$; $\frac{10^6 \text{ Btu}}{\text{h}}$



$$\text{kW} \times 3.412(10^{-3}) = \frac{10^6 \text{ Btu}}{\text{h}}$$

$$\frac{10^6 \text{ Btu}}{\text{h}} \times 293 = \text{kW}$$

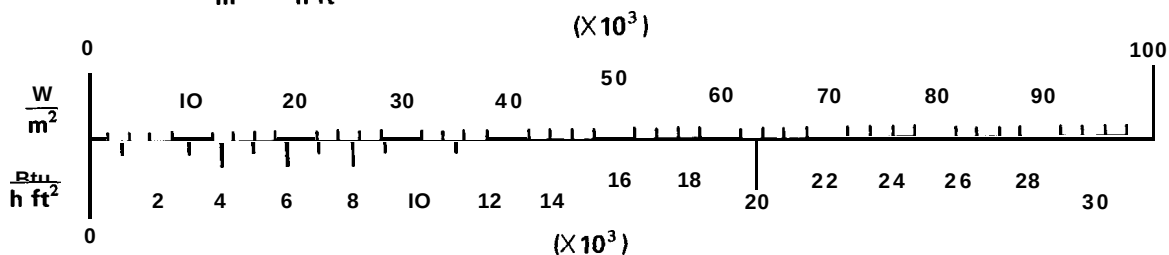
HEAT TRANSFER COEFFICIENT: $\frac{W}{m^2 K}$; $\frac{Btu}{h ft^2 {}^\circ F}$



$$\frac{W}{m^2 K} \times 0.176 = \frac{Btu}{h ft^2 {}^\circ F}$$

$$\frac{Btu}{h ft^2 {}^\circ F} \times 5.68 = \frac{W}{m^2 K}$$

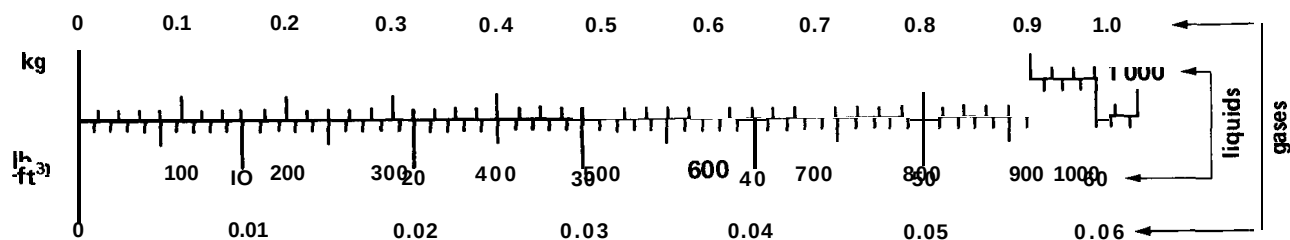
HEAT FLUX: $\frac{W}{m^2}$; $\frac{Btu}{h ft^2}$



$$\frac{W}{m^2} \times 0.316 = \frac{Btu}{h ft^2}$$

$$\frac{Btu}{h ft^2} \times 3.16 = \frac{W}{m^2}$$

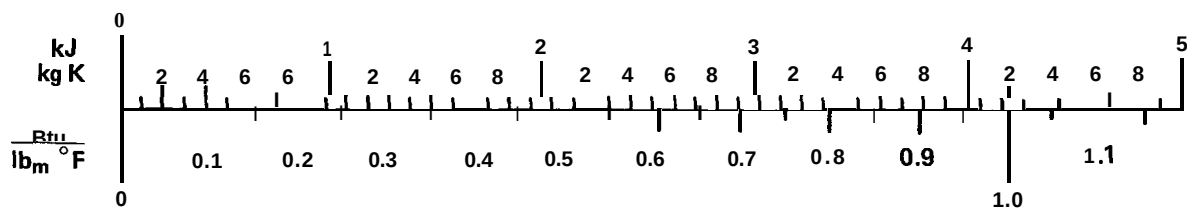
DENSITY: $\frac{kg}{m^3}$; $\frac{lb_m}{ft^3}$



$$\frac{kg}{m^3} \times 0.0624 = \frac{lb_m}{ft^3}$$

$$\frac{lb_m}{ft^3} \times 16.02 = \frac{kg}{m^3}$$

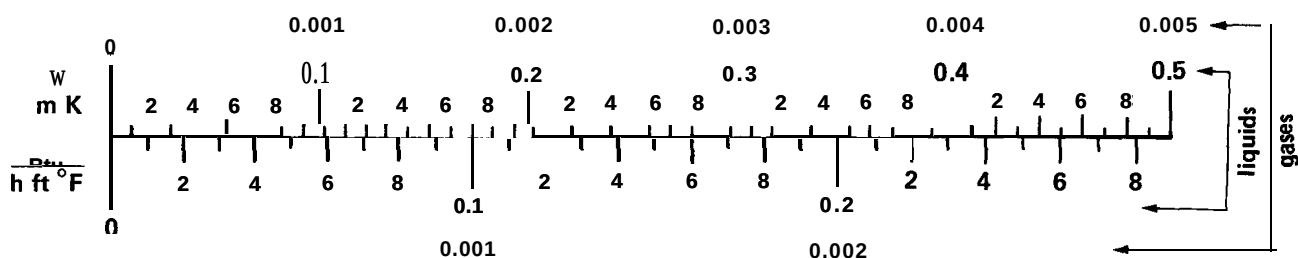
SPECIFIC HEAT CAPACITY: $\frac{\text{kJ}}{\text{kg K}}; \frac{\text{Btu}}{\text{lb}_m \text{ } ^\circ\text{F}}$



$$\frac{\text{kJ}}{\text{kg K}} \times 0.239 = \frac{\text{Btu}}{\text{lb}_m \text{ } ^\circ\text{F}}$$

$$\frac{\text{Btu}}{\text{lb}_m \text{ } ^\circ\text{F}} \times 4.187 = \frac{\text{kJ}}{\text{kg K}}$$

THERMAL CONDUCTIVITY: $\frac{\text{W}}{\text{m K}}; \frac{\text{Btu}}{\text{h ft } ^\circ\text{F}}$

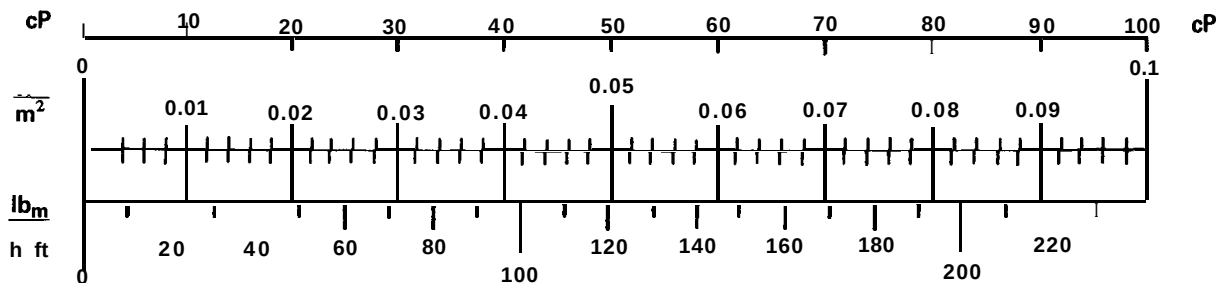


$$\frac{\text{W}}{\text{m K}} \times 0.578 = \frac{\text{Btu}}{\text{h ft } ^\circ\text{F}}$$

$$\frac{\text{Btu}}{\text{h ft } ^\circ\text{F}} \times 1.731 = \frac{\text{W}}{\text{m K}}$$

DYNAMIC (ABSOLUTE) VISCOSITY: $\frac{\text{N s}}{\text{m}^2}; \frac{\text{lb}_m}{\text{h ft}}$

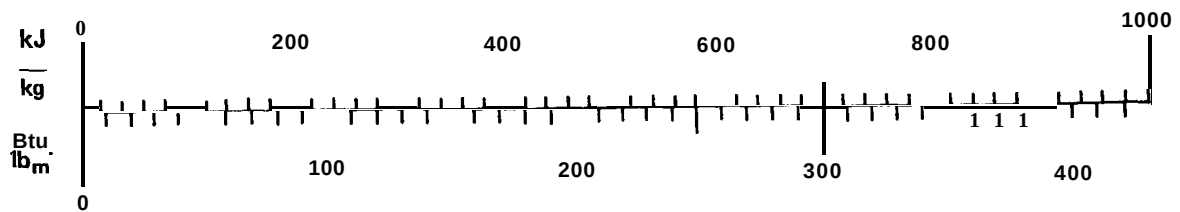
centipoise $\text{cP} = 10^{-3} \frac{\text{N s}}{\text{m}^2} = \text{mPa s}$



$$\text{cP} \times 2.42 = \frac{\text{lb}_m}{\text{h ft}}$$

$$\frac{\text{lb}_m}{\text{h ft}} \times 0.413 = \text{cP}$$

ENTHALPY: $\frac{\text{kJ}}{\text{kg}}$, $\frac{\text{Btu}}{\text{lb}_m}$

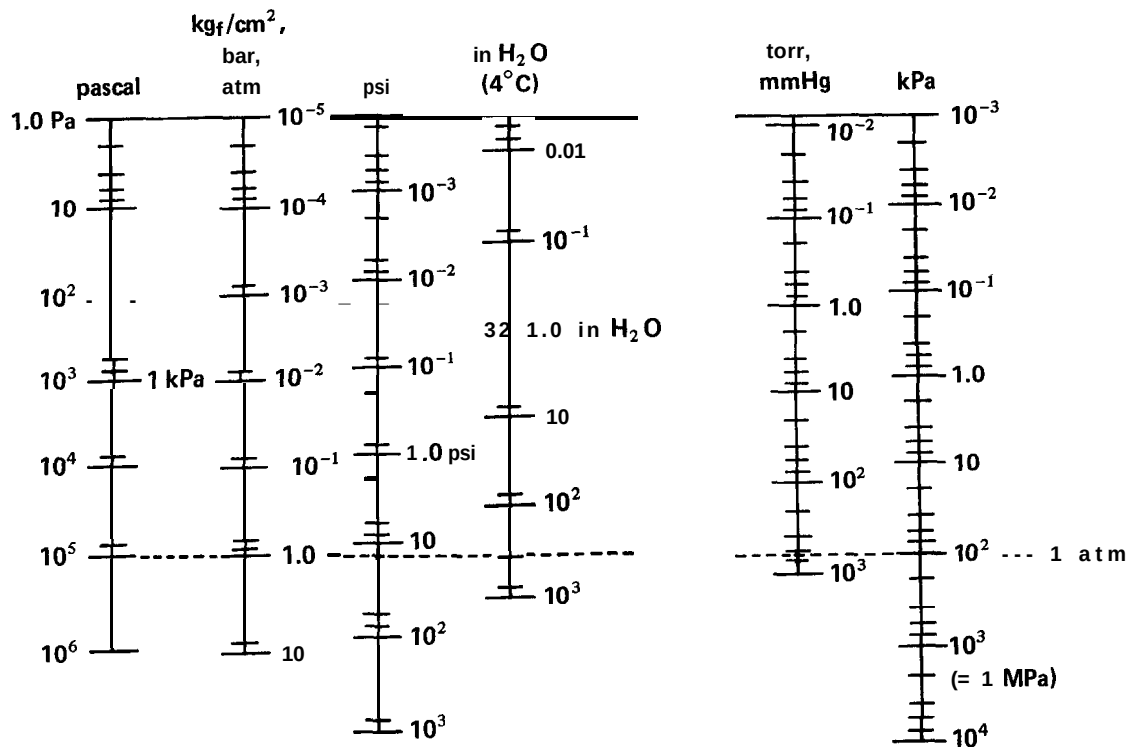


$$\frac{\text{kJ}}{\text{kg}} \times 0.430 = \frac{\text{Btu}}{\text{lb}_m}$$

$$\frac{\text{Btu}}{\text{lb}_m} \times 2.326 = \frac{\text{kJ}}{\text{kg}}$$

PRESSURE CONVERSION CHART

(top line = 1 Pa on all charts)

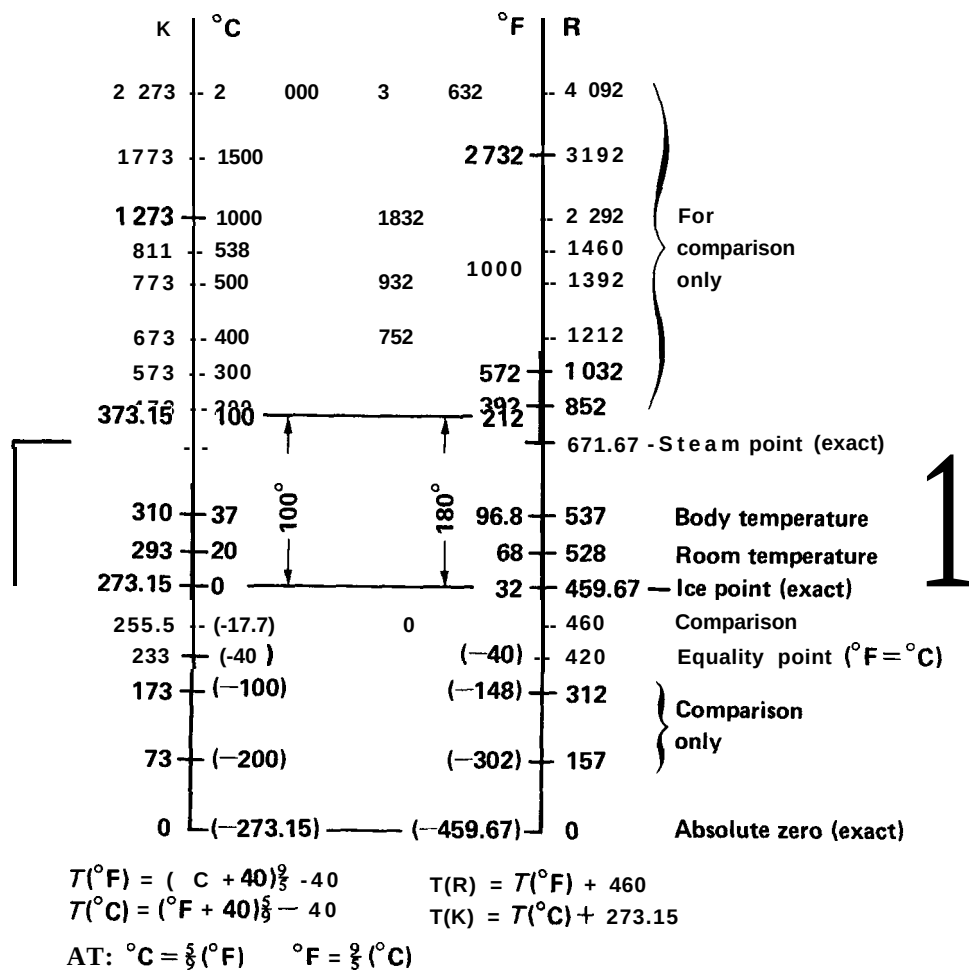


***NOTE:** only bar is exact (1 bar = 10⁵ Pa); however, atm and kgf/cm² are within 2% accuracy (see list below).

1 atm = 1.01325 × 10⁵ Pa
 1 psi = 6.89476 × 10³ Pa
 1 in H₂O = 2.49082 × 10² Pa
 1 kgf/cm² = 9.80665 × 10⁴ Pa
 1 bar = 10⁵ Pa
 1 mmHg = 1.33322 × 10² Pa

1 Pa = 9.86923 × 10⁻⁶ atm
 1 Pa = 1.45038 × 10⁻⁴ psi
 1 Pa = 4.01474 × 10⁻³ in H₂O
 1 Pa = 1.01972 × 10⁻⁵ kgf/cm²
 1 Pa = 10⁻⁵ bar
 1 Pa = 7.50064 × 10⁻³ mmHg

TEMPERATURE COMPARISON
(NOT TO SCALE)



4.1

Basic mechanical principles

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4.1.1

Introduction

C. Ruiz

A. Loads and stresses

The mechanical design of heat exchangers begins with consideration of the service loads and a determination of their values. Loads may be subdivided into two categories, depending on their cause and on their variation with time. In the first category, the following types should be considered:

1. Distributed mechanical load, for example, internal or external pressure.
2. Mechanical load concentrated on a small area, for example, self-weight loading applied at a column or saddle support or load applied at an anchor by a pipe. (Note that when this load is caused by the restricted thermal expansion of a pipe, it is limited by the yielding of the pipe and anchor point.)
3. Thermal loading caused by differential expansion of the shell-and-tube bundle, by the thermal expansion of the heat exchanger on its supports, by temperature gradients through the thickness of a plate or shell, or by differences between thermal expansion coefficients in the junction between two elements.

In the second category, static or quasistatic loads are distinguished from shock loads that may occur in an accident—for example, thermal shock due to direct impingement of cold fluid on a hot surface. Loads may be maintained throughout the whole life of the heat exchanger, change only a few times, or undergo a cyclic variation.

Once the loads normally occurring in service and those anticipated in possible accidents are characterized, the next step is to find the stress distribution,

assuming elastic behavior. As in the case of loads, elastic stresses may be subdivided into several categories, depending on both their origin and the effect they have on the strength of the structure. The ASME Boiler and Pressure Vessel Code [1] categorizes the stresses into a number of groups in accordance with detailed rules that are not always unequivocal. A simple classification is the following:

Type I stress: Stress distributed uniformly through the thickness caused by internal or external pressure—in general, any stress not limited by a displacement and capable of causing widespread yielding and ultimately plastic collapse of the structure. The pressure-induced stress in a cylindrical shell is a typical example.

Type II stress: Bending stress caused by mechanical loading. The bending stress in a tube plate, treated as simply supported, under the effect of the difference in pressure between the tube side and the shell side is an example of this stress, whose value may be permitted to exceed the yield point of the material without producing plastic collapse of the plate. It is important to note that the stresses are calculated under assumed elastic behavior. They are thus used rather as figures of merit than as a correct representation of the true stress conditions. Neither residual stresses nor elastoplastic deformation is considered quantitatively in design; their effects are minimized by following empirical design rules and engineering judgment.

Type III stress: Stresses caused by constraints at junctions or by thermal loading. Limited by displacement, these stresses cannot by themselves bring about ultimate plastic collapse.

Type IV stress: The previous stresses affect a wide area. A type IV stress, on the other hand, is concentrated in the immediate vicinity of a notch, a sharp reentrant corner, a threaded connection, and so on. Such notch-type stress raisers need be considered only when the material used is brittle or when cyclic variations of the load can lead to fatigue failure.

A complete fitness-for-purpose analysis requires the consideration of all possible modes of failure, determination of the strength against each mode, and definition of acceptable safety limits. An elastic stress analysis is insufficient by itself, and more complete treatments including limit analysis, elastic shakedown, and those based on fracture mechanics are needed [2-5]. Elastic analysis, however, is sufficient for selection of the main dimensions. This is achieved by setting acceptable limits to the calculated stresses, based on the mechanical properties of the material. In parallel with most design codes [1, 6, 7], the mechanical properties taken to represent the strength of the material are as follows:

Ultimate tensile stress, σ_u

Yield point or 0.2% proof stress, σ_Y

Stress required to produce failure after 100 000 h at the design temperature, σ_R

Stress required to produce a permanent strain of 1% after 100 000 h at the design temperature, σ_T

The design stress S , depending on the special requirements of the design code, is taken to be the smallest of

$$\frac{\sigma_u}{3} \quad \frac{\sigma_Y}{1.5} \quad \frac{\sigma_R}{1.5} \quad \frac{\sigma_T}{1.1} \quad (1)^\dagger$$

The safety factors that appear as denominators in Eq. (1) are specified by the design code and may differ slightly from those given here. It will be noted that mechanical properties such as the elongation to rupture, the fracture toughness, or the fatigue strength are entirely ignored at this stage. It is only in the selection of materials and in the strength evaluation, when required, that they are used for analysis.

After S has been obtained for the material selected, allowable limits for the four stress types are found as shown in Fig. 1, in which σ_I should be less than S , σ_{II} may be as high as $1.5S$, but the sum of σ_I and σ_{II} must not exceed $1.5S$, and similarly for other combinations. Notch-type stresses, depending less on the overall dimensions and loads than on the severity of the notch, need be limited only by fatigue strength S_D (see Sec. C).

[†] Within a threedigit section the equation, figure, and table numbers do not have the three-digit identifier; for cross-references between sections, the appropriate three-digit number will be given.

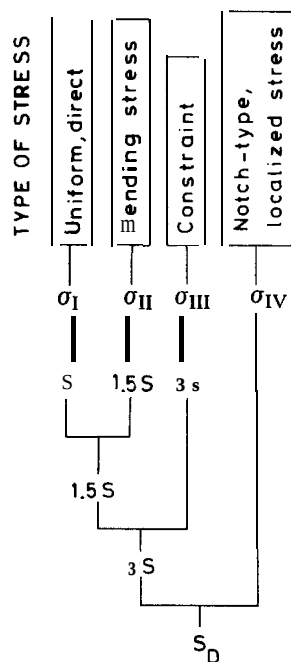


Figure 1 Determination of allowable stress limits.

B. Construction elements

Shells, tube plates, tubes, expansion joints, flanges, and heads are the main elements of construction. In this section their stress analysis will be described briefly; the reader will be referred to specialized publications for further information. Detailed design, following design codes, is considered in other chapters in this part of the handbook. The differences between the behavior of the various elements is illustrated by comparing a cylindrical shell and a plate, as in Fig. 2. In the shell,

$$\sigma_I = \frac{pr}{t} \quad (2)$$

and in the plate, at the center, assuming free support at the periphery,

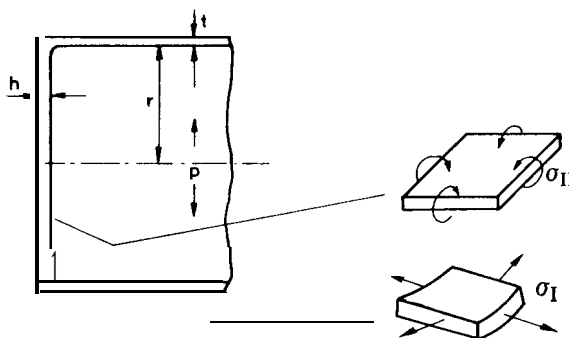


Figure 2 Stresses of types I and II in a vessel consisting of a cylindrical shell and a flat head.

$$\sigma_{II} = \pm \frac{3(1+\nu)pr^2}{8h^2} \quad (3)$$

(see Sec. 4.1.4). Limits to these stresses are set in accordance with Fig. 1. Should there be a combination of both types, as in the beam of Fig. 3, the sum $\sigma_I + \sigma_{II}$ will be maintained below 1.55 S . A better limit for the combined loading case, allowing for the interaction between both types of loading, is given by

$$\frac{\sigma_{II}}{S} < 1.5 \left[1 - \left(\frac{\sigma_I}{S} \right)^2 \right] \quad (4)$$

At the junction between the shell and the flat head of Fig. 2, compatibility of radial displacements and rotations implies the existence of self-equilibrating edge forces and moments that will introduce a system of localized stresses, direct, bending and shear, of type III. Provided that the material is sufficiently ductile, σ_{III} can be allowed to cause plastic deformation since, once yielding has been reached, the load, which is deformation dependent in the elastic field, ceases to grow as the deformation increases to satisfy the compatibility requirements at the junction.

The limitation fixed by Fig. 1 is represented in Fig. 4 by the parallelepiped $OMNPO'M'N'P'$, truncated by the planes

$$\sigma_I + \sigma_{II} \leq 1.5S \quad (5)$$

$$\sigma_I + \sigma_{II} + \sigma_{III} \leq 3S \quad (6)$$

All points within the space bounded by these planes correspond to acceptable stress limits. A much simpler limitation, more conservative than the first, is provided by

$$\sigma_I + \frac{\sigma_{II}}{1.5} + \frac{\sigma_{III}}{3} < S \quad (7)$$

All points within the tetrahedron $OO'MP$ represent acceptable limits.

C. Modes of failure to be considered

The first mode of failure to be considered is **ductile fracture** preceded by large widespread plastic deformation. In most construction elements a biaxial state of

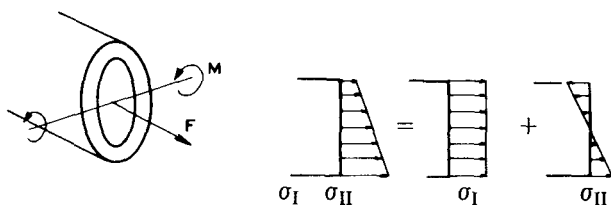


Figure 3 Beam element under a combination of stresses types I and II.

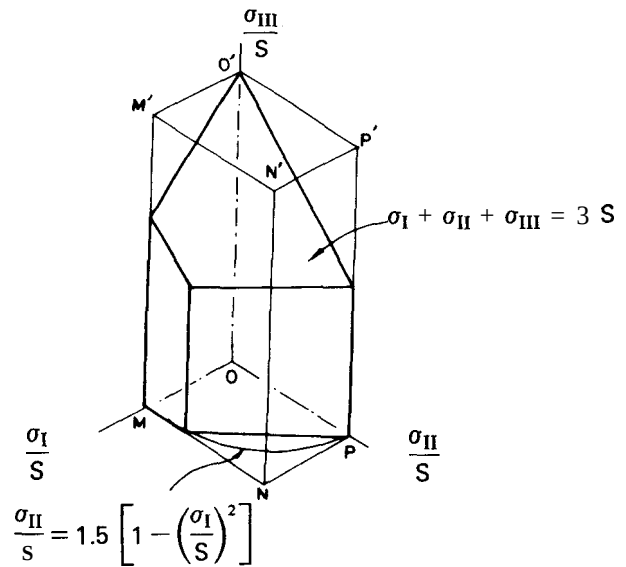


Figure 4 Limitation of combined stresses, considering interaction among types I, II, III. Points within space $OMNPO'$ satisfy criterion of Fig. 1; points within $OMPO'$ satisfy Eq. (7).

stress exists and it is necessary to define a single characterizing parameter. This may be the maximum stress, the maximum shear stress, or a combination of the principal stresses. Refer to the small element of Fig. 5; the stress equivalent to the actual condition is one of the following:

$$\sigma = \sigma_1 \quad (\text{maximum stress criterion}) \quad (8)$$

$$\sigma = \sigma_1 \quad \text{or} \quad (\sigma_1 - \sigma_2)$$

$$(\text{Tresca shear criterion, whichever is larger}) \quad (9)$$

$$\sigma = \sqrt{\sigma_1^2 + \sigma_2^2 - \sigma_1 \sigma_2} \quad (\text{von Mises criterion}) \quad (10)$$

In these equations, σ_1 is larger in absolute value than σ_2 , and both may be positive (tension) or negative (compression). It is very rare that a complete triaxial state is found in the design process. Generally the stress in the third direction, σ_3 , is negligible. When this is not so, it can be included in the equations [2]. Ductile fracture is prevented by limiting the equivalent stress as previously described.

The second mode of failure, closely related to ductile fracture, is **incremental collapse**. This can be illustrated by means of a simple model, consisting of three bars, rigidly fixed to cross members at their ends, as in

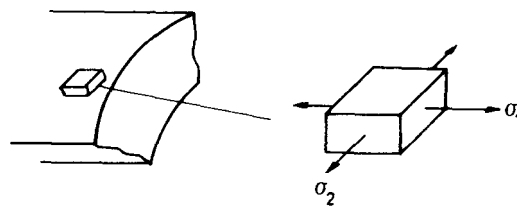


Figure 5 Biaxial state of stress.

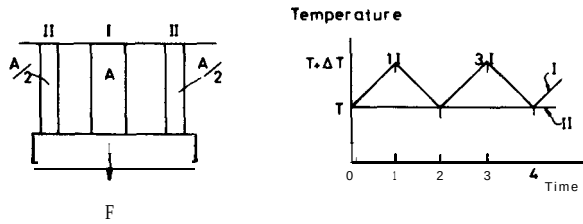


Figure 6 Three-bar system under constant force and cyclic variation of temperature.

Fig. 6. A constant force F is applied to the assembly and the temperature of the central bar is cycled between T and $T + \Delta T$, where T is the initial temperature of the whole assembly. By writing that there is equilibrium of forces and compatibility of displacements, it is easy to distinguish between three possible types of behavior:

1. Elastic cycling, as in Fig. 7, when

$$E\alpha \Delta T < 2 \left(\sigma_Y - \frac{F}{2A} \right) \quad (11)$$

2. Elastic shakedown, after a first half-cycle resulting in a set strain, when

$$2 \left(\sigma_Y - \frac{F}{2A} \right) < E\alpha \Delta T < 4 \left(\sigma_Y - \frac{F}{2A} \right) \quad (12)$$

3. Finally, when

$$E\alpha \Delta T > 4 \left(\sigma_Y - \frac{F}{2A} \right) \quad (13)$$

The central bar deforms along lines 11-21-31, and so on, in Fig. 7c, whereas the two outer bars deform along 1II-2II-3II, and so on, and the assembly as a whole tends to suffer an increasingly large deformation and,

finally, breaks in a ductile manner. This is the process of **incremental collapse**. In the elementary example treated, and accepting the same stress limits as before,

$$\frac{F}{2A} = \sigma_I < S = \frac{\sigma_Y}{1.5}$$

The thermal stress, compressive in the central bar, tensile in the two outer bars, is

$$\sigma_{III} = \frac{E\alpha \Delta T}{2} < 3S = 2\sigma_Y$$

$$\sigma_I + \sigma_{III} = \frac{F}{2A} + \frac{E\alpha \Delta T}{2} < 2\sigma_Y$$

to avoid incremental collapse (Fig. 1), whereas Eq. (13) specifies that incremental collapse will not occur only if

$$2 \frac{F}{2A} + \frac{E\alpha \Delta T}{2} < 2\sigma_Y \quad (14)$$

This is one example of a situation where the design stress limits are not sufficient to prevent failure. It is perhaps exceptional in that the stiffness of the component undergoing thermal cycling is usually much smaller than that of the rest of the structure and it yields, whereas the structure as a whole deforms between two well-defined limits. If, instead of the limits given by Fig. 1, Eq. (7) is applied,

$$\sigma_I + \frac{\sigma_{III}}{3} = \frac{F}{2A} + \frac{E\alpha \Delta T}{6} < S = \frac{\sigma_Y}{1.5}$$

or

$$\frac{3}{2} \frac{F}{2A} + \frac{E\alpha \Delta T}{2} < 2\sigma_Y$$

A comparison with the exact conditions, Eq. (14), shows

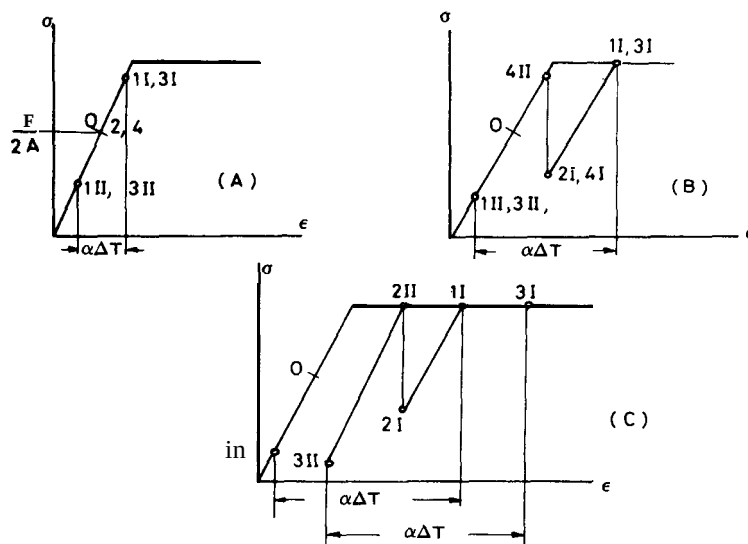


Figure 7 Different possible load-deformation states for the three-bar assembly. (a) Elastic behavior. (b) Elastic shakedown. (c) Incremental collapse.

that this is overconservative. It can be concluded that the criterion for the combination of stresses of Fig. 1 may not be sufficiently conservative in some cases whereas the condition represented by Eq. (7), illustrated by the plane $O'MP$ of Fig. 4, is always safe.

Consider now the same three-bar assembly of Fig. 6 but assume that the cross section of the central bar is much smaller than that of the outer bars, say, $0.1A$. By repeating the same equilibrium and compatibility analysis as before, the behavior under cyclic temperature of the central bar is

1. Elastic cycling, as in Fig. 7, when

$$E\alpha \Delta T < 1.1 \left(\sigma_Y - \frac{F}{2A} \right) \quad (15)$$

2. Elastic shakedown, when

$$1.1 \left(\sigma_Y - \frac{F}{2A} \right) < E\alpha \Delta T < 2.2\sigma_Y \quad (16)$$

3. Finally, when

$$E\alpha \Delta T > 2.2\sigma_Y \quad (17)$$

The central bar cycles between yielding in tension and in compression, as in Fig. 8, following the loop 1I-2I-3I-4I, and so on, whereas the two outer bars cycle elastically between 11I, 21I, and so on. **High-strain fatigue** of the central bar will cause its eventual failure. The same process is found in components in which a notch-type stress raiser increases the stress level above

yield point. Under cyclic loading, the area adjacent to the notch is forced to deform plastically by the rest of the material, which remains elastic. References [8,9] should be consulted.

Heat exchanger tubes are often subjected to axial compression and external pressure. The combination of these loads may cause **elastic buckling**, a mode of failure that should also be considered in shells under partial vacuum or external pressure. See Sec. 4.1.5.

Time-dependent deformation under constant stress at elevated temperature, or **creep**, is not a serious problem in liquid-to-liquid heat exchangers, since the service temperature is never sufficiently high. It may have to be considered in some special gas-to-gas heat exchangers. For a treatment of the subject, see [10].

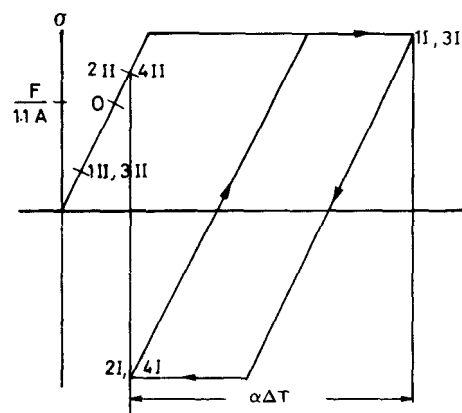


Figure 8 High-strain fatigue.

Nomenclature and References for Section 4.1.1 appear at the end of Section 4.1.8.

4.4.2

Methods of analysis

C. Ruiz

A. Elastic analysis

A heat exchanger consists of an assemblage of elementary shells, plates, and tubes. Each component can be considered as separate from the rest and subjected to pressure, self-weight, and self-induced thermal loading with completely unrestrained ends supported by the necessary reactions. A complete stress, strain, and deformation analysis is possible for most components and loading conditions [11-14], and this will give the value of the displacements at the ends of each component and, in general, at the junctions with others. Usually, the independent, unrestrained displacement of two components with a common junction will not be possible. Self-equilibrating restraining loads will be applied at the junction line or surface to both components in order to achieve compatibility of displacements. This is the essence of the **free-body method** described in detail in [3]. The total stresses, strains, and deformations are the sum of the results of the unrestrained analysis and of the effect of the restraining loads applied at the junctions between components or at rigid anchors.

Although successful in providing a simple, efficient way of solving problems in which there is rotational symmetry, the application of the free-body method becomes much more complex when the load distribution is not symmetric, as in column supports or saddles supporting a horizontal shell. Some standard solutions exist for these cases and will be described later. Shells of revolution under loads that may be described by means of trigonometric series are treated in [12-14] in some detail. A limitation in the method consists in

that, by treating shells as surfaces without thickness, rings as curves, and beams as lines, it cannot give any indication of the stress distribution at junctions, where a three-dimensional stress analysis is needed. In recent years, numerical methods, in particular the finite-element method, have been widely used. The principle of this method consists of subdividing the complete body into a number of matching elements with straight edges and using energy principles to obtain the relationship between loading, assumed to be applied at nodal points, and the displacement of these points. Simplifying assumptions have to be made to make the process of solution practicable with available computing facilities. A critical appraisal of the results is always needed. The method is described in [15, 16], among many other publications, and the reader is referred to the users manuals for specific programs [17, 18]. A combination of numerical integration and finite-element techniques is being used for the solution of problems involving branched shells of revolution [19].

Experimental stress analysis provided, until recently, the only means of determining stress in three-dimensional situations, asymmetric or concentrated loads on shells, and so on. It is now an alternative to numerical analysis. Photoelasticity and strain gauge techniques [20, 21] are firmly established, having the virtue of simplicity and of providing clear, unambiguous values of stresses or strains.

B. Plastic analysis

The value of the load required to cause widespread yielding and ultimate collapse of a structure consist-

ing of shells and rings is estimated assuming that the material is rigid-ideally plastic. The method of analysis is essentially the same as the one used for the limit-analysis linear frames, with the added difficulty of the bidimensional state of stress on the shells as compared with the one-dimensional stresses in the beams forming the framework. Examples of the application of limit analysis to pressure vessel shells and rings are found in [2,3], a general introduction to the subject is contained in [22], and further applications to shell structures are given in [23].

Limit analysis is commonplace in the design of structures such as buildings and bridges. It is not frequently used in the design of heat exchangers because the mode of failure is seldom plastic collapse under static loading. In a flanged joint, elastic behavior must be demanded if leakage is to be prevented and the joint is to be opened periodically for maintenance or inspection. To avoid high-strain fatigue, elastic shakedown of the structure as a whole, under combined constant and cyclic loads, is also a design requirement. In these, as in the design against conventional fatigue, an elastic rather than a plastic analysis constitutes the basis for design.

C. Asymmetric loading

Three types of asymmetric loading conditions are commonly found in practice:

1. Load applied by a pipe or a support column to the heat exchanger shell
2. Saddles supporting a horizontal heat exchanger
3. Variation of temperature across a diameter in the tube plate of a hairpin heat exchanger

The first two types of loading are analyzed by expansion in the form of trigonometric series, a method that is too tedious to be of any practical direct application to design [24]. Bijlaard [25-29] was able to present the results of the stress analysis of cylindrical and spherical shells loaded through a rigid rectangular pad or a radial pipe in a form readily applicable, and his results have been included in [6]. Unfortunately, the stresses predicted have been found to be only in rough agreement with experimental data [30]. It is possible to combine the theoretical and the experimental results to obtain a rough idea of the stress level. Consider the cylindrical shell of Fig. 1. Forces and moments are applied to a radial pipe as shown and the stresses σ_x, σ_θ are calculated for positions A, B, C, D on the outside or the inside of the shell following Table 1, with the geometric parameters from Figs. 2-5. The stresses are combined in accordance with von Mises criterion and the largest value maintained below acceptable limits.

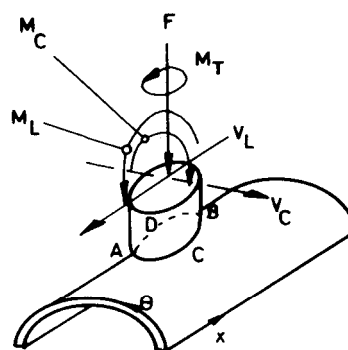


Figure 1 Cylindrical shell under concentrated loading.

When the load is applied through a column of non-circular cross section—an oblique pipe or a rectangular pad—for purposes of analysis the zone in contact is replaced by a circle of equal area.

Referring to Table 1, the designer will fill in the data for applied loads, that is, the value of radial and shear loads (in newtons or pounds force) and of hoop and axial moments and torque (newton-meters or pounds force-inches). For a given geometry, the shell thickness, mean shell radius, and nozzle half-bore (meters or inches) will then be filled in and the load and geometric parameters calculated. From Figs. 2 and 3, as indicated in the first part of Table 1, values are obtained for the dimensionless parameters $(N_x T/F)$, and so on. Multiplying these parameters by (F/T^2) , and so on, gives the components of the stress σ_x at positions A, B, C, D, . . . as shown in Fig. 1. In all cases, the sign is the one given in Table 1. The total axial stress σ_x is the algebraic sum of all terms in a row. The value of the hoop stress is obtained similarly and combined to the shear stress τ_1, τ_2 to give the final values. A consistent system of units should be used.

The stresses can be calculated similarly for spherical shells, Fig. 6 and Table 2, with geometric parameters from Figs. 7-10.

Horizontal heat exchangers are supported on saddles, as in Fig. 11. Design of the saddles and reinforcing pads, when needed, is based on experimental work [31] and is detailed in BS 5500 [6]. The exact analysis is complicated by the ill-defined conditions at the junction between saddle and shell, where the form in which the load is transmitted depends on the friction coefficient, the flexibility of the shell, and the temperature distribution [24,32].

The first step in the design of saddles is to find the bending moment M_C in the middle and M_S in the supports,

$$M_C = \frac{w_1 L}{4} \left[\frac{1 + 2(r^2 - b^2)/L^2}{1 + 4b/3L} - 4 \frac{A}{L} \right] \quad (1)$$

Table 1 Stress calculation for cylindrical shell under concentrated loading

1. Applied loads		2. Shape		3. Parameters			
Radial load	$F =$	Thickness	$T =$	$F/T =$		$\rho = \frac{r}{R} \sqrt{\frac{R}{T}}$	
Hoop moment	$M_C =$	Radius	$R =$	$M_C/rT^2 =$			
Axial moment	$M_L =$	Nozzle $\frac{1}{2}$ bore	$r =$	$M_L/rT^2 =$			
Torsion	$M_T =$						
Shear load	$V_C =$						
Shear load	$V_L =$						
From	Fig.	2	3	2	3	2	3
Read		$N_x T$ F	$\frac{M_x}{F}$	$\frac{N_x T r}{M_C}$ $\frac{M_x r}{M_C}$	$\frac{M_x r}{M_C}$	$\frac{N_x T r}{M_L}$ $\frac{M_x r}{M_L}$	Total
$ \sigma $		$\times \frac{F}{T^2} =$	$\times 6 \frac{F}{T^2} =$	$\times \frac{M_C}{rT^2} =$	$\times 6 \frac{M_C}{rT^2} =$	$\times \frac{M_L}{rT^2} =$	$\times 6 \frac{M_L}{rT^2} =$
σ_x	A_o	—				—	
	A_i	—	+			—	+
	B_o	—	—			+	+
	B_i	—	+			+	—
	C_o	—	—		—		
	C_i	—	+	—	+		
	D_o	—		+	+		
	D_i	—	+	+			
From	Fig.	4	5	4	5	4	5
Read		$N_\theta T$ F	$\frac{M_\theta}{F}$	$\frac{N_\theta T r}{M_C}$ $\frac{M_\theta r}{M_C}$	$\frac{M_\theta r}{M_C}$	$\frac{N_\theta T r}{M_L}$ $\frac{M_\theta r}{M_L}$	Total
$ \sigma $		$\times \frac{F}{T^2} =$	$\times 6 \frac{F}{T^2} =$	$\times \frac{M_C}{rT^2} =$	$\times 6 \frac{M_C}{rT^2} =$	$\times \frac{M_L}{rT^2} =$	$\times 6 \frac{M_L}{rT^2} =$
σ_θ	A_o	—	—			—	—
	A_i	—	+			—	+
	B_o	—	—			+	+
	B_i	—	+			+	—
	C_o	—	—		—		
	C_i	—	+	—	+		
	D_o	—	—	+	+		
	D_i	—	+	+	—		
Shear stress due to M_T		$\tau_1 = M_T/2\pi r^2 T =$					
Shear stress due to V_C		$\tau_2 = V_C/\pi r T =$					
Shear stress due to V_L		$\tau_3 = V_L/\pi r T =$					
Shear stress in A, B		$\tau_1 + \tau_2 =$					
Shear stress in C, D		$\tau_1 + \tau_3 =$					
Combined stress		$\sigma = \sigma_x^2 + \sigma_\theta^2 - \sigma_x \sigma_\theta + 3\tau^2 =$					
A_o	A_i	B_o	B_i	C_o	C_i	D_o	D_i

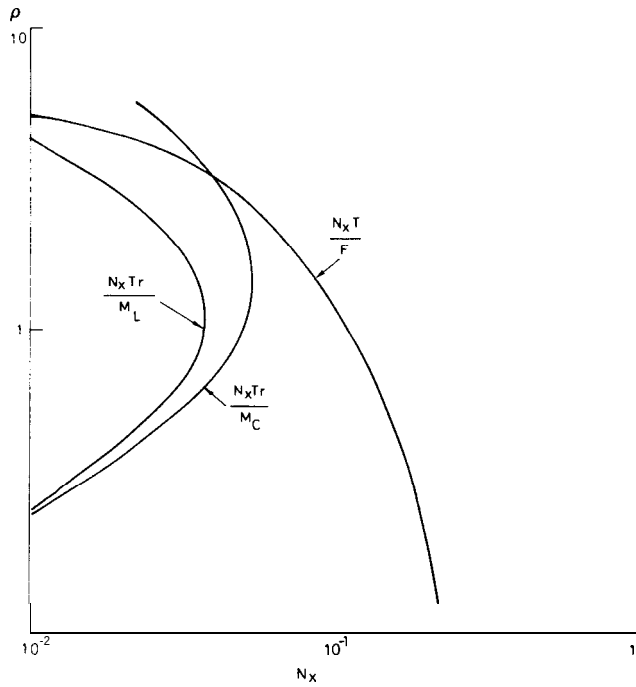


Figure 2 Longitudinal direct stress due to radial force or to bending moment in a cylindrical shell.

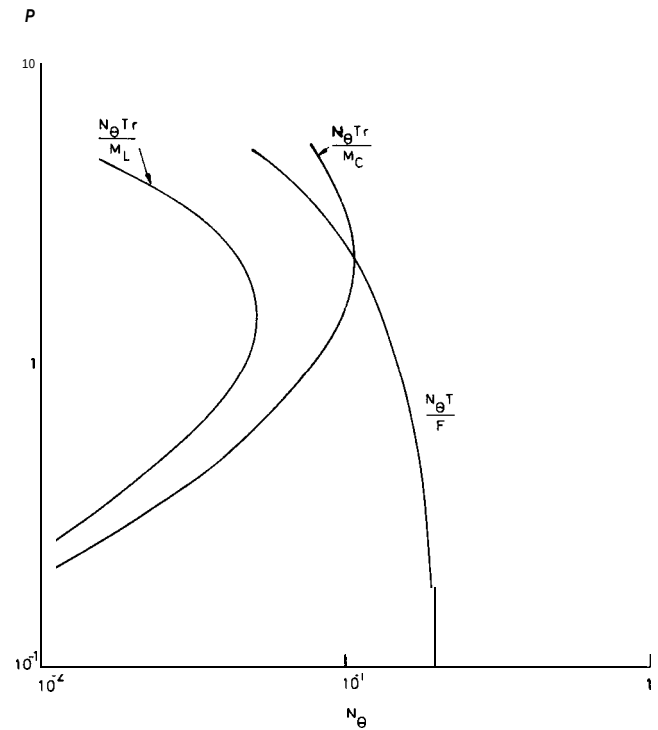


Figure 4 Circumferential direct stress due to radial force or to bending moment in a cylindrical shell.

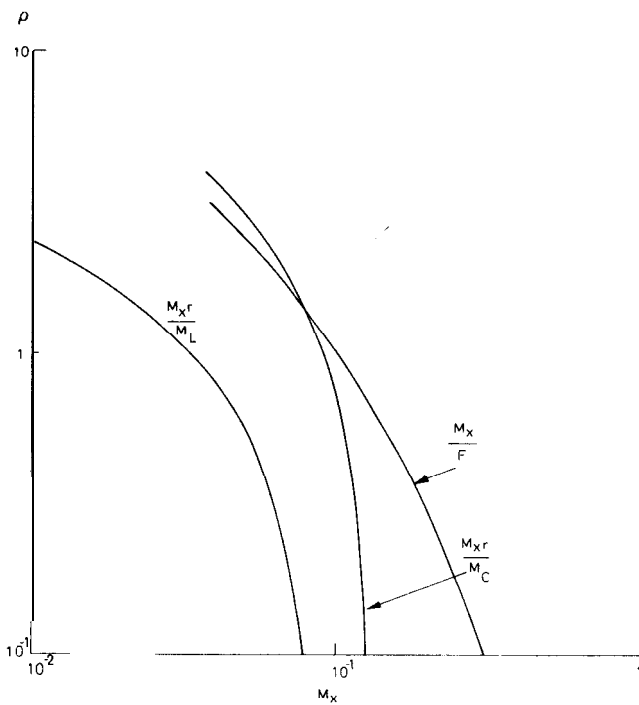


Figure 3 Longitudinal bending stress due to radial force or to bending moment in a cylindrical shell.

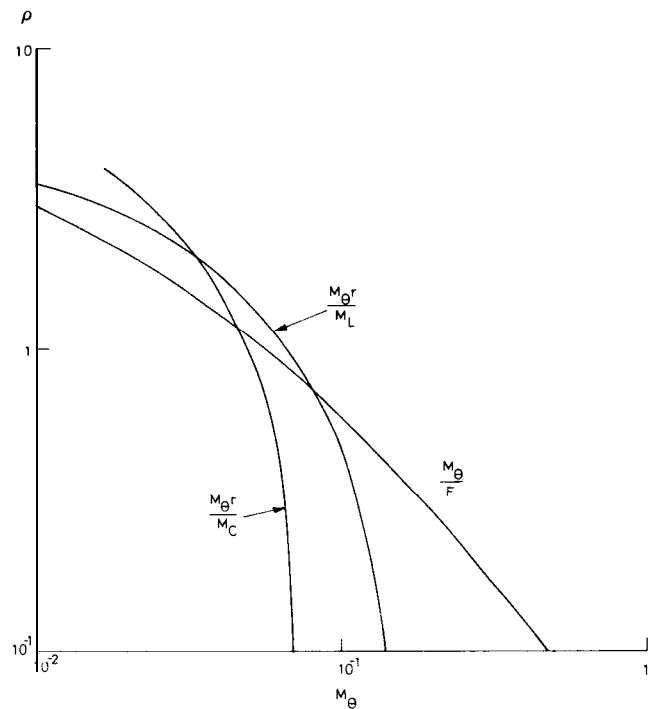


Figure 5 Circumferential bending stress due to radial force or to bending moment in a cylindrical shell.

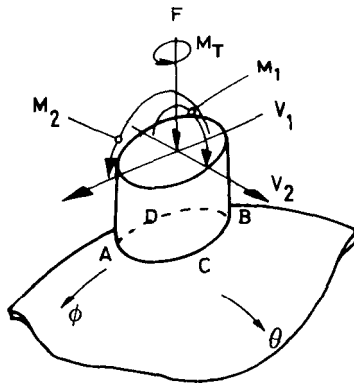


Figure 6 Spherical shell under concentrated loading.

$$M_S = -W_1 A \left[1 - \frac{1 - A/L + (r^2 - b^2)/2AL}{1 + 4b/3L} \right] \quad (2)$$

where all quantities are defined in Fig. 11. The combined longitudinal bending and pressure stress in the middle is

$$(\sigma_x)_{\text{top middle}} = \frac{pr}{2t} - \frac{M_c}{\pi r^2 t} \quad (3)$$

$$(\sigma_x)_{\text{bottom middle}} = \frac{pr}{2t} + \frac{M_c}{\pi r^2 t} \quad (4)$$

and in the plane of the saddles is

$$(\sigma_x)_{\text{top support}} = \frac{pr}{2t} - \frac{M_S}{K_1 \pi r^2 t} \quad (5)$$

$$(\sigma_x)_{\text{bottom support}} = \frac{pr}{2t} + \frac{M_S}{K_2 \pi r^2 t} \quad (6)$$

where K_1 and K_2 are given in Table 3.

Shear stresses appear between the saddle and the shell. Their magnitude is given by

$$q = \frac{K_3 W_1}{rt} \left(\frac{L - 2A - b}{L + b} \right) < 0.8S \quad (7)$$

when the vessel is not reinforced and the saddles are well away from the ends, and

$$q = \frac{K_3 W_1}{rt} < 0.8S \quad (8)$$

in a ring-reinforced vessel. S is the design stress and K_3 is given in Table 4.

Hoop stresses cause an ovalization of the shell. For the unreinforced vessel, at the bottom,

$$(\sigma_\theta) = -\frac{K_5 W_1}{t(b_1 + 10t)} \quad (9)$$

and at the end of the saddle,

$$\sigma_\theta = -\frac{W_1}{4t(b_1 + 10t)} - \frac{3K_6 W_1}{2t^2} \quad \text{for } \frac{L}{r} > 8 \quad (10)$$

$$\sigma_\theta = -\frac{W_1}{4t(b_1 + 10t)} - \frac{12K_6 W_1 r}{Lt^2} \quad \text{for } \frac{L}{r} < 8 \quad (11)$$

K_5 and K_6 are given in Table 5. K_6 varies between 0.01 and 0.05. For ring-reinforced shells, as in Figs. 11 and 12, at the bottom,

$$\sigma_\theta = -\frac{K_5 W_1}{t(b_1 + 10t)} \quad (12)$$

at the junction between ring and shell,

$$\sigma_\theta = \frac{C_4 K_7 W_1 rc}{I} - \frac{K_8 W_1}{a} \quad (13)$$

and the outermost fiber of the ring is subjected to

$$\sigma_\theta = \frac{C_5 K_7 W_1 rd}{I} - \frac{K_8 W_1}{a} \quad (14)$$

C_4 , C_5 , K_7 , and K_8 are also given in Table 5. a is the cross section of the reinforcing ring, defined as in Fig. 12, and I is its second moment of area.

When a saddle cannot take the whole load without inducing excessively high stresses, it is possible to use the full circumferential support of Fig. 13. For this type, the shear stress at the support is

$$q = \frac{0.319 W_1}{rt} \quad (15)$$

with a minimum section modulus

$$Z = \frac{K_{10} W_1 r_2}{S} \quad (16)$$

K_{10} varies between 0.075 for $\phi = 30^\circ$ and 0.015 for $\phi = 90^\circ$.

The preceding section gives only a brief description of the B.S. method. For further details, see [6].

In a hairpin heat exchanger, the two halves of the tube plate are at different temperatures, one side being in contact with the fluid inlet, the other with the outlet. This problem will be treated in Sec. 4.1.4.

Table 2 Stress calculation for spherical shell under concentrated loading

1. Applied loads		2. Shape		3. Parameters			
Radial load	F =	Thickness	T =	$F/T^2 =$		$\rho = \frac{r}{R} \sqrt{\frac{R}{T}}$	
Moment	M ₁ =	Radius	R =	M, /rT ² =			
Moment	M ₂ =	Nozzle $\frac{1}{2}$ bore	r =	M ₂ /rT ² =			
Torsion	M _T =						
Shear load	v ₁ =						
Shear load	v ₂ =						
From Fig.	7	8	9	10	9	10	
Read	$\frac{NT}{F}$	$\frac{M}{F}$	$\frac{N_{\phi}Tr}{M_1}$	$\frac{M_{\phi}r}{M_1}$	$\frac{N_{\phi}Tr}{M_2}$	$\frac{M_{\phi}r}{M_2}$	Total
σ	$\times \frac{F}{T^2} =$	$\times 6 \frac{F}{T^2} =$	$\times \frac{M_1}{rT^2} =$	$\times 6 \frac{M_1}{rT^2} =$	$\times \frac{M_2}{rT^2} =$	$\times 6 \frac{M_2}{rT^2} =$	σ _φ
σ _φ	A _O	—			—	—	
	A _i	—	+		—	+	
	B _O	—	—		+	+	
	B _i	—	+		+	—	
	C _O	—	—	—			
	C _i	—	+	—	+		
	D _O	—	—	+	+		
	D _i	—	+	+	—		
From Fig.	7	8	9	10	9	10	
Read	$\frac{NT}{F}$	$\frac{M}{F}$	$\frac{N_{\theta}Tr}{M_1}$	$\frac{M_{\theta}r}{M_1}$	$\frac{N_{\theta}Tr}{M_2}$	$\frac{M_{\theta}r}{M_2}$	Total
σ	$\times \frac{F}{T^2} =$	$\times 6 \frac{F}{T^2} =$	$\times \frac{M_1}{rT^2} =$	$\times 6 \frac{M_1}{rT^2} =$	$\times \frac{M_2}{rT^2} =$	$\times 6 \frac{M_2}{rT^2} =$	σ _θ
σ _θ	A _O						
	A _i						
	B _O						
	B _i						
	C _O						
	C _i						
	D _O						
	D _i						
Shear stress due to M _T τ ₁ = M _T /2πr ² T = Shear stress due to V ₁ τ ₂ = V ₁ /πrT = Shear stress due to V ₂ τ ₃ = V ₂ /πrT = Shear stress in A, B τ ₁ + τ ₂ = Shear stress in C, D τ ₁ + τ ₃ = Combined stress σ = √(σ _φ ² + σ _θ ² - σ _φ σ _θ + 3τ ²) =							
A _O	A _i	B _O	B _i	C _O	C _i	D _O	D _i

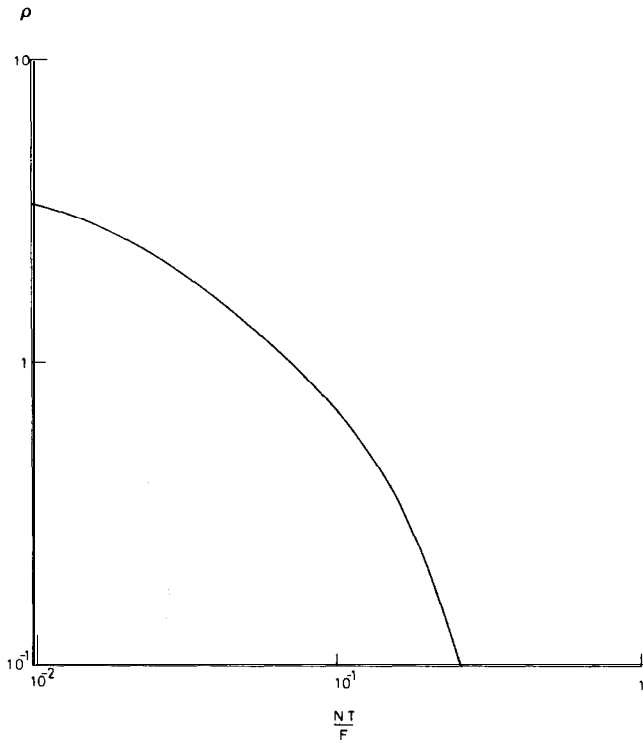


Figure 7 Direct stress due to radial force in a spherical shell.

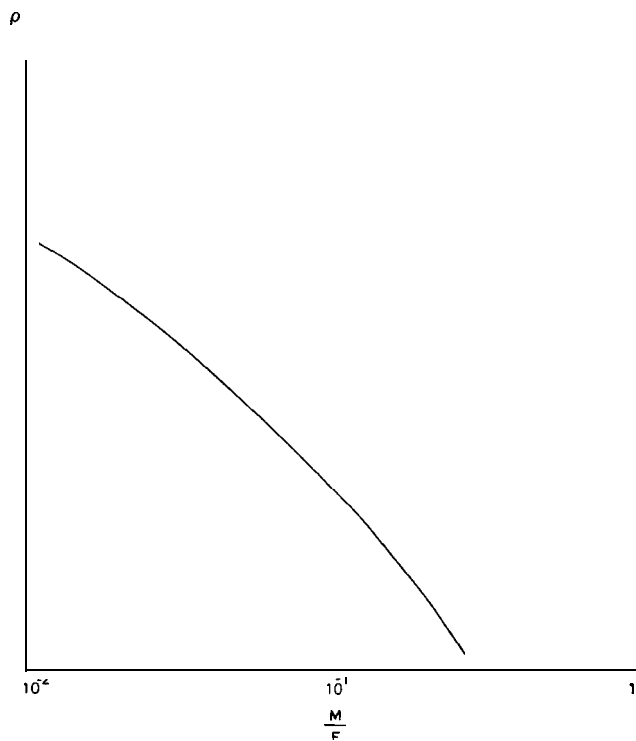


Figure 8 Bending stress due to radial force in a spherical shell.

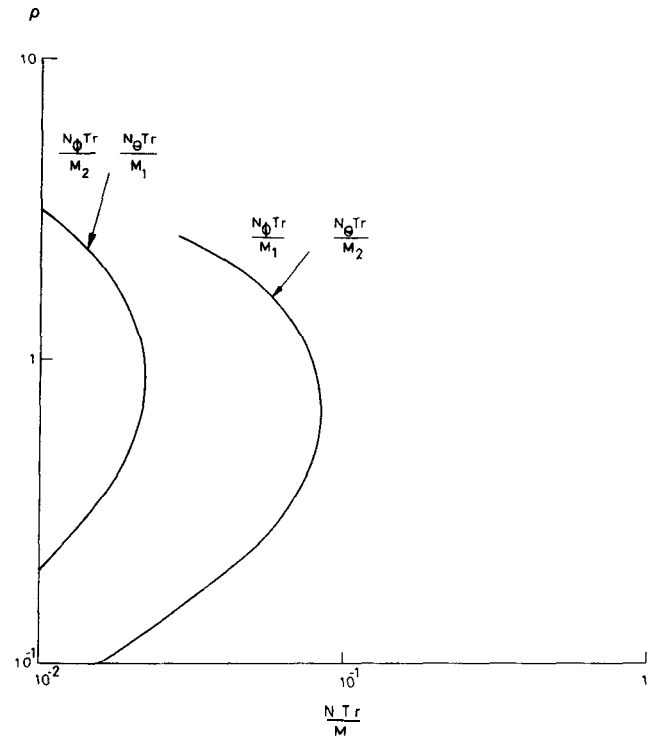


Figure 9 Direct stress due to bending moment in a spherical shell.

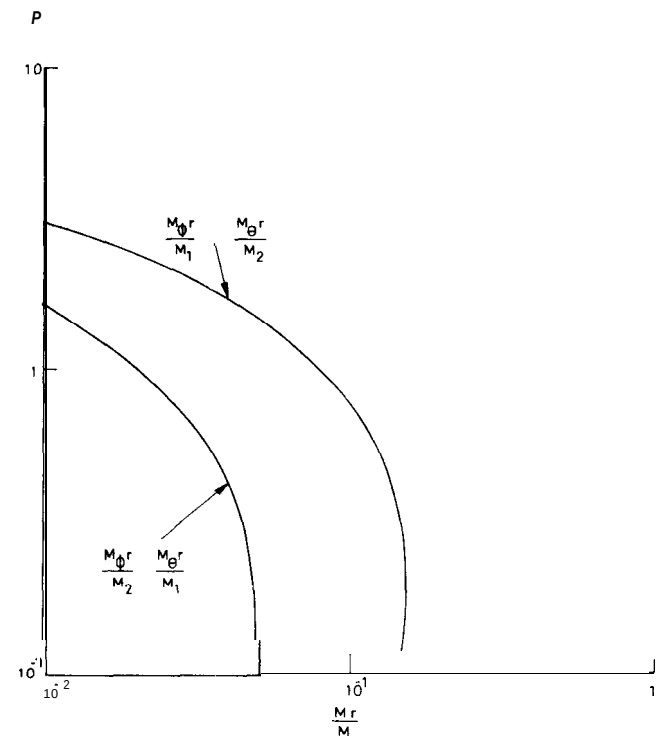


Figure 10 Bending stress due to bending moment in a spherical shell.

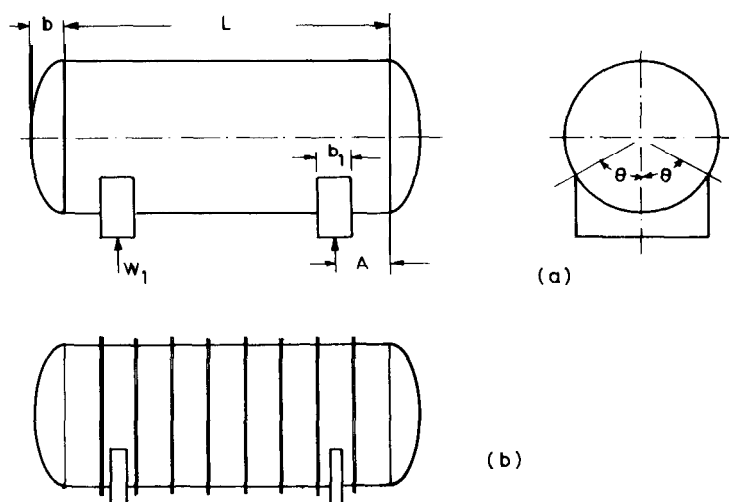


Figure 11 Cylindrical heat exchanger supported on saddles. (a) Without ring stiffeners. (b) With ring stiffeners.

Table 3 Stress calculation for saddle supports: Values of K_1 and K_2

	Angle, degrees	K_1	K_2
Ring-reinforced shells, $A < \frac{r}{2}$	120	1.0	1.0
	150	1.0	1.0
Unreinforced shells	120	0.107	0.192
	150	0.161	0.279

Table 4 Stress calculation for saddle supports: Values of K_3

	Angle, degrees	K_3
Unreinforced shells, $A > \frac{r}{2}$	120	1.171
	150	0.799
Reinforced shells, $A > \frac{r}{2}$	120	0.319
	150	0.319
Unreinforced shells, $b_1 < A < \frac{r}{2}$	120	0.880
	150	0.485
$\frac{b_1}{2} < A < b_1$	120	0.880
	150	0.485

Table 5 Stress calculation for saddle supports: Values of K_5, K_7, K_8, C_4 , and C_5

		120	150°
K_6	$\frac{A}{r} < 0.5$	0.013	0.008
	$0.5 < \frac{A}{r} < 1$	0.030	0.020
	$\frac{A}{r} > 1$	0.053	0.032

	Ring support		Internal rings		External rings	
	120	150	120°	150	120	150
C_4	- 1	- 1	+1	+1	-1	- 1
C_5	+1	+1	- 1	- 1	+1	+1
K_5	—	—	0.76	0.673	0.76	0.673
K_7	0.0528	0.0316	0.0577	0.353	0.577	0.0353
K_8	0.34	0.3	0.263	0.228	0.263	0.228

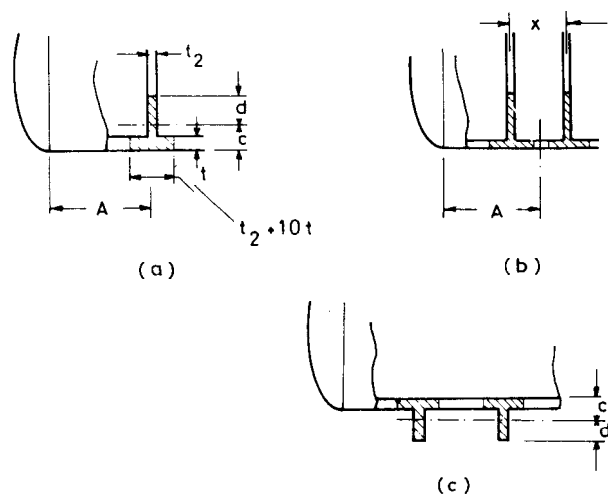


Figure 12 Reinforcing rings in a cylindrical shell.

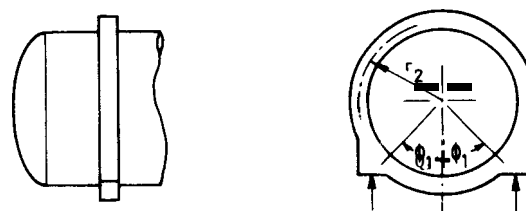


Figure 13 Full circumferential support of a heavy, thin-walled vessel.

Nomenclature and References for Section 4.1.2 appear at the end of Section 4.1.8.

4.1.3

Shells

C. Ruiz

■ Only loads with rotational symmetry will be considered here. For other cases, see [3, 12-15, 33, 34]. Shells are considered thin when the ratio (thickness/diameter) is less than 1:10. Thick-walled vessels are seldom used in heat exchangers [33-35].

A. Cylinders

In the cylindrical shell under internal pressure and edge loading of Fig. 1, the pressure-induced stresses and corresponding displacements and rotations at the edges are

$$\sigma_{\theta} = 2\sigma_x = \frac{pR}{t} \quad u_1 = u_2 = \frac{pR^2(2-\nu)}{2tE} \quad \chi_1 = \chi_2 = 0 \quad (1)$$

and, due to edge loading,

$$(\sigma_{\theta})_1 = \pm \frac{6\nu M_1}{Rt^2} + \frac{2H_1\beta}{Rt} + \frac{2M_1\beta^2}{R^2t} \quad u_1 = \frac{R^2}{2\beta^3D} H_1 + \frac{R}{2\beta^2D} M_1 \quad (2)$$

$$(\sigma_x)_1 = \pm \frac{6M_1}{Rt^2} \quad \chi_1 = \frac{R}{2\beta^2D} H_1 + \frac{1}{\beta D} M_1$$

and the equivalent equations for edge 2. In these equations,

$$\beta^4 = \frac{3(1-\nu^2)R^2}{t^2} \quad D = \frac{Et^3}{12(1-\nu^2)} \quad (3)$$

It is assumed that edges 1 and 2 are remote from each

other, so that there is no interaction between them. This assumption is valid when the shell length exceeds $3R/\beta$, or about $2\sqrt{Rt}$. The stresses induced by the edge loads die out very rapidly away from the edges.

B. Spheres (domes)

In a spherical shell under internal pressure and edge loading as in Fig. 2, under the effect of the pressure

$$\sigma_{\theta} = \sigma_{\phi} = \frac{pR}{t} \quad u = \frac{pR^2(1-\nu)}{2tE} \sin \phi \quad \chi = 0 \quad (4)$$

and, due to edge loading,

$$(\sigma_{\theta})_{\text{edge}} = \pm \frac{6\nu M}{Rt^2 \sin \phi} + \frac{2\beta}{Rt} H + \frac{2\beta^2}{Rt^2 \sin \phi} M \quad (\sigma_{\phi})_{\text{edge}} = \pm \frac{6M}{Rt^2 \sin \phi} + \frac{H}{Rt \tan \phi} \quad (5)$$

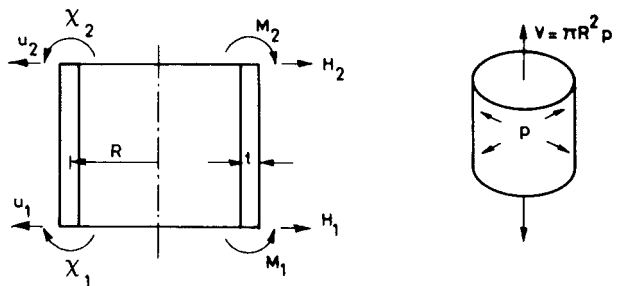


Figure 1 Cylindrical shell under internal pressure and end loading.

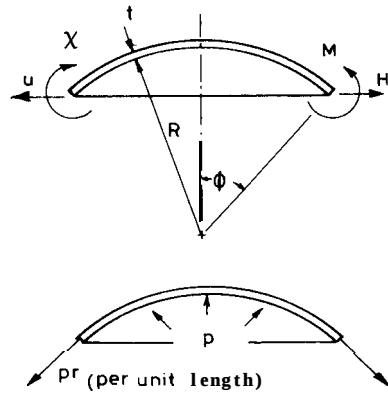


Figure 2 Spherical shell under internal pressure and end loading.

$$u = \frac{R^2 \sin \phi}{2\beta^3 D} H + \frac{R}{2\beta^2 D} M$$

$$\chi = \frac{R}{2\beta^2 D} H + \frac{1}{\beta D \sin \phi} M \quad (5)$$

(Cont.)

where β and D are given by Eq. (3). These equations are valid only when $\beta\phi > \pi$, or $\phi > 2\sqrt{t/R}$ (radians).

C. Cones

For the conical shell of Fig. 3 under internal pressure,

$$\sigma_\theta = \frac{pR}{t \cos \alpha} \quad u = \frac{pR^2(2-\nu)}{2tE \cos \alpha}$$

$$\sigma_\phi = \frac{pR}{2t \cos \alpha} \quad \chi_1 = -\chi_2 = \frac{p(R_1 + R_2)(2-\nu)}{2tE \cos^2 \alpha} \sin \alpha \quad (6)$$

The stresses and edge displacements are given by the

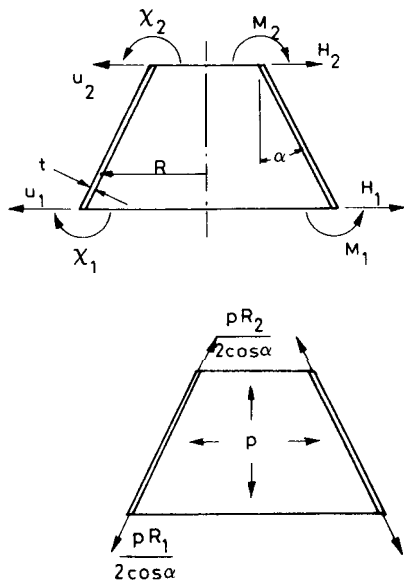


Figure 3 Conical shell under internal pressure and end loading.

same equations as for the sphere, Eq. (5), taking

$$\alpha = \frac{\pi}{2} - \phi$$

D. Toroids

Complete toroidal shells are used as tube bends. Under internal pressure, the stresses are (Fig. 4)

$$\sigma_\theta = \frac{pr}{2t} \quad \sigma_\phi = \frac{pr}{t} \frac{2R + r \cos \phi}{2(R + r \cos \phi)} \quad (7)$$

The effect of edge loads as well as in-plane bending moments applied to a pipe bend has been treated in [3].

E. Thermal stresses

A good treatment of thermal stresses may be found in [35,36], with numerous results summarized in [33]. It is always possible, when the temperature is constant through the thickness, to subdivide the shell into a number of isothermal elements and to calculate the discontinuity stresses at the junction between each pair of elements. For example, consider a cylindrical shell with step variation of temperature or consisting of two materials of different thermal expansion coefficient (see Fig. 5). In either case, the free expansion of one-half relative to the other is

$$u_{\text{free}} = \alpha \Delta T = (\alpha_1 - \alpha_2) \Delta T$$

An edge force and a moment must be imposed to achieve continuity:

$$u_1 + u_{\text{free}} = u_2 \quad \chi_1 = -\chi_2$$

$$H_1 + H_2 = 0 \quad M_1 = M_2$$

From Eq. (2), assuming both halves of the shell to have the same characteristics,

$$H_1 = \frac{\beta^3 D}{R^2} u_{\text{free}} \quad M_1 = 0 \quad \sigma_\theta = \frac{2\beta^4 D}{R^3 t} u_{\text{free}}$$

at the junction.

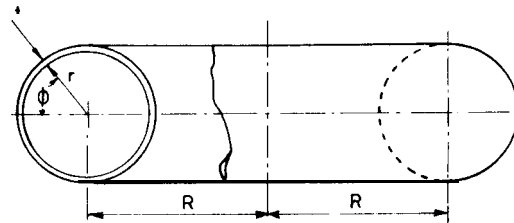


Figure 4 Toroidal shell under internal pressure.

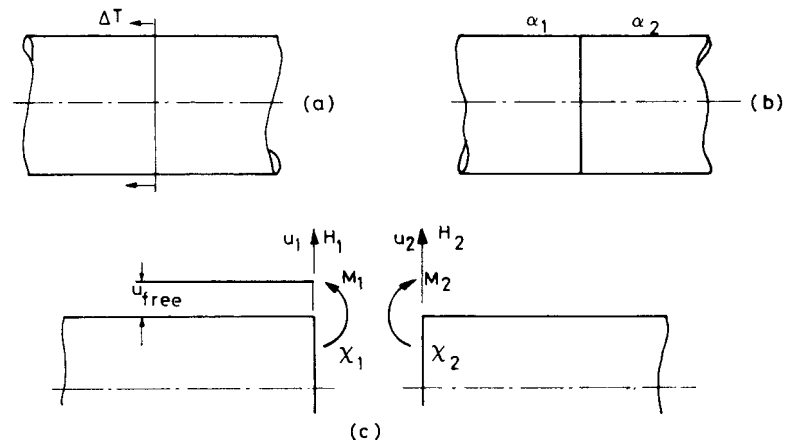


Figure 5 Cylindrical shell (a) with step variation of temperature, or (b) consisting of two halves with different thermal expansion coefficients. (c) Edge conditions.

Nomenclature and References for Section 4.1.3 appear at the end of Section 4.1.8.

4.1.4

Tube plates

C. Ruiz

■ For the purpose of analysis and design, tube plates are replaced by an equivalent, unperforated plate with the same dimensions and bending stiffness. The analysis consists of three parts. First, the loads imposed by the constraints on the equivalent plate are calculated. Then the stresses are obtained also for the equivalent plate. Finally, suitable stress magnification factors are applied to find the stress distribution in the actual plate, considering the effect of the perforations.

A. Equivalent plates

In the equivalent plate, Young's modulus is E^* and Poisson's ratio is ν^* . The ratio E^*/E , where E is Young's modulus for the tube plate material, and ν^* are obtained from Fig. 1, as a function of the relative ligament width [37-40]. The tubes, expanded against the tube plate and, as is common practice at present, welded with or without additional weld metal, have a reinforcing effect that is usually disregarded [41, 42].

B. Stresses and edge rotation of unperforated plates: hairpin heat exchangers

The simplest form of loading and support is shown in Fig. 2a. A more general case, which can serve to build up plates in which the pressure varies with the radial position, is shown in Fig. 2b. Figure 2c is then combined with Fig. 2b to satisfy compatibility at the junctions. The remaining cases are applications of the first three.

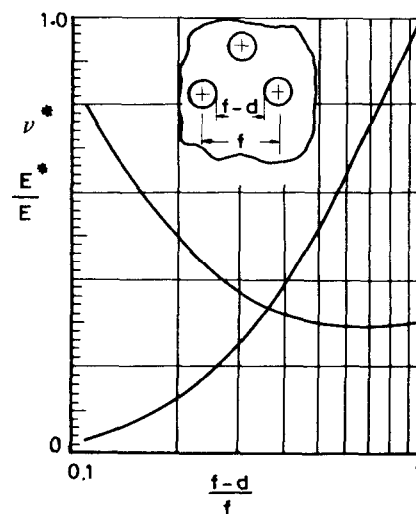


Figure 1 Equivalent elastic constants for a tube plate.

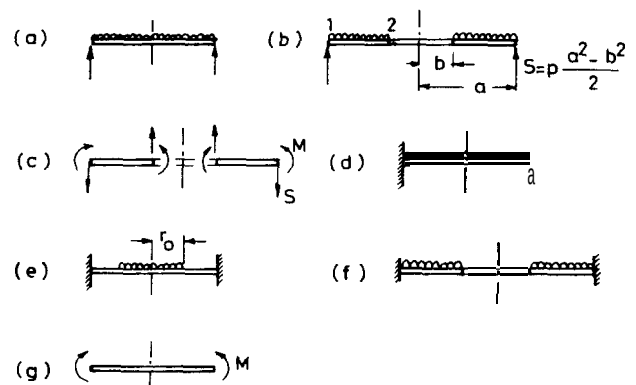


Figure 2 Different types of loading and support of circular plates.

1. Simple support, constant pressure:

$$\chi = \frac{p}{16D} \frac{2}{1+\nu} a^3 \quad (1)$$

$$(\sigma_\theta)_0 = (\sigma_\phi)_0 = \pm \frac{3(3+\nu)}{8t^2} pa^2$$

2. Annular plate, simple support, constant pressure:

$$\chi_1 = \frac{p}{8D} \left[\frac{a^3}{1+\nu} + ab^2 \frac{1+\nu}{1-\nu^2} - \frac{4ab^4}{(1-\nu)(a^2-b^2)} \log \frac{a}{b} \right]$$

$$\chi_2 = \frac{p}{8D} \left[a^2 b \frac{3+\nu}{1-\nu^2} - b^2 \frac{1}{1+\nu} - \frac{4a^2 b^3}{(1-\nu)(a^2-b^2)} \log \frac{a}{b} \right] \quad (2)$$

$$(\sigma_\theta)_1 = \pm \frac{3p}{4t^2} \left[a^2(1-\nu) + b^2(1+3\nu) - \frac{4b^4(1+\nu)}{a^2-b^2} \log \frac{a}{b} \right]$$

$$(\sigma_\theta)_2 = \pm \frac{3p}{4t^2} \left[(3+\nu)a^2 - (1-\nu)b^2 - \frac{4a^2 b^2(1+\nu)}{a^2-b^2} \log \frac{a}{b} \right]$$

3. Annular plate, edge loading:

$$\chi_1 = \frac{1}{D} \left\{ M_1 \frac{(a^2+b^2) - \nu(a^2-b^2)}{(1-\nu^2)(a^2-b^2)} - M_2 \frac{2ab}{(1-\nu^2)(a^2-b^2)} - \frac{S}{4} \left[\frac{2a}{1+\nu} + \frac{4ab^2}{(1-\nu)(a^2-b^2)} \log \frac{a}{b} \right] \right\}$$

$$\chi_2 = \frac{1}{D} \left\{ -M_1 \frac{2ab}{(1-\nu^2)(a^2-b^2)} + M_2 \frac{(a^2+b^2) + \nu(a^2-b^2)}{(1-\nu^2)(a^2-b^2)} + \frac{S}{4} \left[\frac{2b}{1+\nu} + \frac{4a^2 b}{(1-\nu)(a^2+b^2)} \log \frac{a}{b} \right] \right\} \quad (3)$$

$$(\sigma_\theta)_1 = \pm \frac{6}{at^2} \left\{ M_1 \frac{a^2+b^2}{a^2-b^2} - M_2 \frac{2ab}{a^2-b^2} - Sa \left[\frac{(1+\nu)b^2}{a^2-b^2} \log \frac{a}{b} + \frac{1-\nu}{2} \right] \right\}$$

$$(\sigma_\theta)_2 = \pm \frac{6}{bt^2} \left\{ M_1 \frac{2ab}{a^2-b^2} - M_2 \frac{a^2+b^2}{a^2-b^2} - Sb \left[\frac{(1+\nu)a^2}{a^2-b^2} \log \frac{a}{b} + \frac{1-\nu}{2} \right] \right\} \quad (3) \quad (Cont.)$$

4. Built-in edge, uniform pressure:

$$(\sigma_\phi)_1 = \pm \frac{3pa^2}{4t^2} \quad (\sigma_\theta)_1 = \pm \frac{3pa^2}{4t^2} \nu \quad (4)$$

edge moment $M_1 = pa^3/8$ (per radian).

5. Built-in edge, uniform pressure over a central area of radius r_0 :

$$(\sigma_r)_1 = \pm \frac{3pr_0^2}{t^2} \left(1 - \frac{r_0^2}{2a^2} \right) \quad M_1 = apr_0^2 \left(1 - \frac{r_0^2}{2a^2} \right)$$

$$(\sigma_r)_0 = (\sigma_\theta)_0 = \pm \frac{3pr_0^2}{2t^2} \left[(1+\nu) \log \frac{a}{r_0} + (1+\nu) \frac{r_0^2}{4a^2} \right] \quad (5)$$

$$(\sigma_\theta)_1 = \pm \frac{3pr_0^2}{2t^2} \left(1 - \frac{r_0^2}{2a^2} \right)$$

6. Annular plate, built in at the edge and under uniform pressure:

$$(\sigma_r)_1 = \pm \frac{3p}{4t^2} \left[a^2 - 2b^2 + \frac{b^4(1-\nu) - 4b^4(1+\nu) \log(a/b) + a^2 b^2(1+\nu)}{a^2(1-\nu) + b^2(1+\nu)} \right]$$

$$M_1 = \frac{(\sigma_r)_1 t^2 a}{6} \quad (6)$$

$$(\sigma_\theta)_2 = \pm \frac{3p(1-\nu^2)}{4t^2} \left[\frac{a^4 - b^4 - 4a^2 b^2 \log(a/b)}{a^2(1-\nu) + b^2(1+\nu)} \right]$$

7. Annular plate, bending moment applied to edge 1:

$$(\sigma_r)_1 = (\sigma_\theta)_1 = \pm \frac{6M_1}{at^2}$$

$$\chi = \frac{M}{D} \frac{1}{1+\nu} \quad (7)$$

where

$$D = \frac{Et^3}{12(1-\nu^2)}$$

C. Plates connected by tubes: box-type and floating-head heat exchangers

The tube nest acts as a flexible foundation, reducing the stresses of the plate below the value corresponding to the hairpin type. Although it is possible to treat the

problem using a finite-element technique or subdividing the plates into annular elements under uniform pressure and matching edge displacement and edge loading, it is preferable to solve for the two extreme cases of simple and of built-in support. This has been done in [43], which forms the basis of the BS 5500 method of design. In both publications, the peak stresses in the equivalent plate may be obtained from graphs, drawn for the case

$$\nu^* = 0.3 \quad E^* = \frac{f-d}{f} E$$

A more accurate value of the stress may be obtained as follows.

1. Box-type heat exchanger, simply supported plate :

$$\begin{aligned} (\sigma_r)_r &= \pm \frac{(p_1 - p_2)(A - C) - p_2(na + AQ)}{A(QG_1 + G_2)} \frac{R}{ot}^2 \\ (\sigma_\theta)_r &= \pm \frac{(p_1 - p_2)(A - C) - p_2(na + AQ)}{A(QG_1' + G_2')} \frac{R}{ot}^2 \end{aligned} \quad (8)$$

2. Box-type heat exchanger, clamped plate. At the edge :

$$(\sigma_r)_R = \pm \frac{(p_1 - p_2)(A - C) - p_2(na + AQ)}{A(QG_1'' + G_2'')} \frac{R}{ot}^2 \quad (9)$$

$$\sigma_\theta = \nu \sigma_\phi$$

3. Floating-head heat exchanger, simply supported plate :

$$\begin{aligned} (\sigma_r)_r &= \pm \frac{p_1 - p_2}{G_1} \left(\frac{R}{t} \right)^2 \\ (\sigma_\theta)_r &= \pm \frac{p_1 - p_2}{G_1'} \left(\frac{R}{t} \right)^2 \end{aligned} \quad (10)$$

4. Floating-head heat exchanger, clamped plate. At the edge:

$$(\sigma_r)_R = \pm \frac{p_1 - p_2}{G_1''} \frac{R}{ot}^2 \quad \sigma_\theta = \nu \sigma_r \quad (11)$$

In these equations,

$$G_1 = \frac{\gamma R [bei'(\gamma R) f_1(\gamma R) - ber'(\gamma R) f_2(\gamma R)]}{3 [f_1(\gamma R) f_2(\gamma R) - f_1(\gamma R) f_2'(\gamma R)]}$$

$$G_2 = \frac{\gamma^2 R^2 [ber(\gamma R) f_1(\gamma R) + bei(\gamma R) f_2(\gamma R)]}{6 [f_1(\gamma R) f_2(\gamma R) - f_1(\gamma R) f_2'(\gamma R)]}$$

$$G_1' = \frac{\gamma R [bei'(\gamma R) f_1(\gamma R) - ber'(\gamma R) f_2(\gamma R)]}{3 [f_1'(\gamma R) f_2(\gamma R) - f_1(\gamma R) f_2'(\gamma R)]}$$

$$G_2' = \frac{\gamma^2 R^2 [ber(\gamma R) f_1(\gamma R) + bei(\gamma R) f_2(\gamma R)]}{6 [f_1'(\gamma R) f_2(\gamma R) - f_1(\gamma R) f_2'(\gamma R)]}$$

$$\begin{aligned} G_1'' &= \frac{\gamma R \{ [ber'(\gamma R)]^2 + [bei'(\gamma R)]^2 \}}{3 [bei(\gamma R) bei'(\gamma R) + ber(\gamma R) ber'(\gamma R)]} \\ G_2'' &= \frac{\gamma^2 R^2 [ber(\gamma R) bei'(\gamma R) - bei(\gamma R) ber'(\gamma R)]}{6 [bei(\gamma R) bei'(\gamma R) + ber(\gamma R) ber'(\gamma R)]} \end{aligned}$$

where p_1 = shell-side pressure

p_2 = tube-side pressure

n = number of tubes

a = cross-sectional area (metal) per tube

A = cross-sectional area of bore of shell

B = cross-sectional area of shell plate (metal)

C = cross-sectional area of perforated part of tube plate

2ℓ = length of tubes

$E_{t,s}$ = Young's modulus of tubes or shell

$Q = E_t na / E_s B$

$\gamma = (E_t na / LAD^*)^{1/4}$

$f_1 = ber(\gamma r) - (1 - \nu^*) bei'(\gamma r) / \gamma r$

$f_2 = bei(\gamma r) + (1 - \nu^*) ber'(\gamma r) / \gamma r$

$f_1' = \nu^* ber(\gamma r) + (1 - \nu^*) bei'(\gamma r) / \gamma r$

$f_2' = \nu^* bei(\gamma r) - (1 - \nu^*) ber'(\gamma r) / \gamma r$

Tables of the functions ber , bei , and so on, are found in [11, 44, 45]. For a complete analysis, see [3].

D. Stresses in perforated plates

Having calculated the stresses in the equivalent unperforated plate, σ_θ^* , σ_r^* , it is now possible to find the stress distribution in the ligament of the actual tube plate. In Fig. 3,

$$\sigma_n^* = \sigma_r^* \sin^2 \phi + \sigma_\theta^* \cos^2 \phi$$

$$\tau^* = (\sigma_r^* - \sigma_\theta^*) \frac{\sin 2\phi}{2}$$

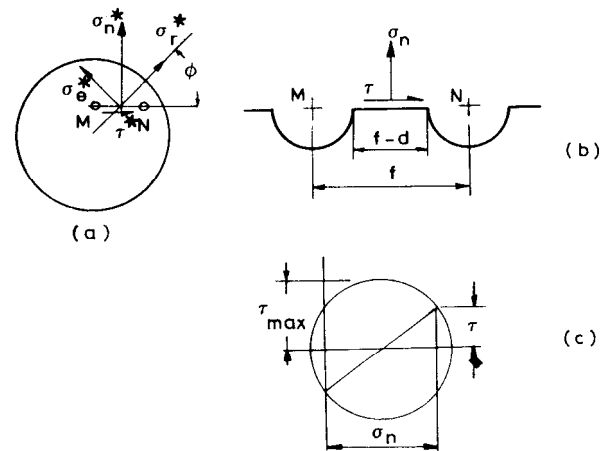


Figure 3 Stress analysis of tube plate. (a) Normal and shear components on a ligament. (b) Average stresses over ligament width. (c) Maximum shear stress.

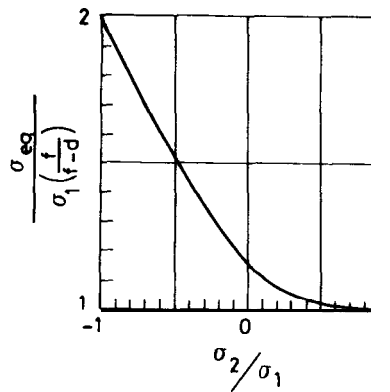


Figure 4 Uniaxial stress equivalent to a biaxial stress system.

The average stresses in the ligament are

$$\sigma_n = \frac{f}{f-d} \sigma_n^* \quad \tau = \frac{f}{f-d} \tau_n^*$$

and the maximum shear stress is

$$\begin{aligned} \tau_{\max} &= \sqrt{\frac{\sigma_n^2}{4} + \tau^2} \\ &= \frac{1}{2} \frac{f}{f-d} [(\sigma_r^* \sin^2 \phi + \sigma_\theta^* \cos^2 \phi)^2 \\ &\quad + (\sigma_r^* - \sigma_\theta^*)^2 \sin^2 \phi]^{1/2} \end{aligned}$$

The equivalent stress, taken to be equal to twice the

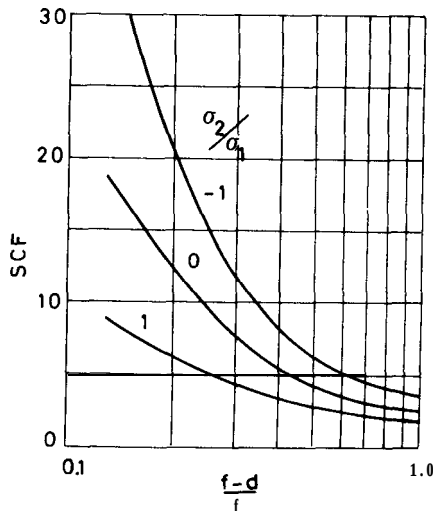


Figure 5 Stress concentration factor.

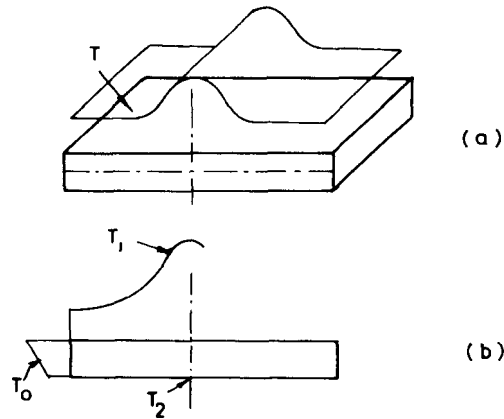


Figure 6 Plate heated along a strip. (a) Temperature fixed through thickness. (b) Linear temperature gradient through thickness.

maximum shear stress, is obtained from Fig. 4, where σ_1 is the largest of σ_r^* and σ_θ^* in absolute value.

In addition to the average stresses, peak stresses exist near the holes that act as notch-type stress raisers. These peak stresses are given by

$$\sigma_{\text{peak}} = \text{SCF } \sigma_1$$

where the stress concentration factor (SCF) is obtained from Fig. 5.

E. Thermal stresses

An upper limit to the thermal stress in a structural element made of a material with Young's modulus E and coefficient of thermal expansion α is $E\alpha \Delta T$, where ΔT is the temperature difference between two points. Several simple cases are considered below.

1. Tube plate with clamped edge, linear thermal gradient between $T + \Delta T$ (top face) and T (bottom face). Bending stress at the edge $\pm \alpha E \Delta T / 2(1 - \nu)$.

2. Plate heated along a strip, temperature distribution as in Fig. 6a. Tensile stress $E\alpha(T - T_0)$.

3. Plate heated as in 2, but with bottom face cooler than top face.

$$\sigma = \frac{1}{2} E\alpha \left[T_1 + T_2 - 2T_0 + \frac{1-\nu}{3+\nu} (T_1 - T_2) \right]$$

4. Cylindrical shell with outside surface at T and inside surface $T + \Delta T$. Linear gradient through the wall.

$$\sigma_\theta = \sigma_x = \pm \frac{\alpha E \Delta T}{2(1 - \nu)}$$

Nomenclature and References for Section 4.1.4 appear at the end of Section 4.1.8.

4.1.5

Tubes

C. Ruiz

■ In tubes under external pressure, the modes of failure can be plastic collapse and buckling. The first mode is avoided simply by maintaining the hoop stress below the yield point of the material. For the second, the critical pressure is given by [3]:

$$p_c = \frac{2E}{1-\nu^2} \frac{t^3}{d^3} \quad (1)$$

where d is the tube bore and t is the thickness. The safety factor usually accepted is about 5.

In box-type and floating-head heat exchangers, the tubes may be subjected to a compressive load and buckle as columns with built-in ends:

$$F_c = \frac{\pi E a^2}{8l^2} \quad (2)$$

where a is the tube cross-sectional area (metal) and l is the length between fixed ends. The baffles partly support the tubes, increasing the buckling strength.

Nomenclature and References for Section 4.1.5 appear at the end of Section 4.1.8.

4.4.6

Expansion joints

C. Ruiz

■ Two main types of convolutions are used in heat exchanger shells: square, as in Fig. 1a, and toroidal, as in Fig. 1b. Both can be analyzed by the free-body method, subdividing them into annular plates and cylinders or into toroidal shells. As a first approach, the convolution may be treated as strips of unit width, ignoring the contribution to the stiffness of the circumferential stresses. In square convolutions, the longitudinal bending stress is found to be

$$\sigma_x = \pm \frac{Et \Delta}{9.33a^2}$$

where Δ is the expansion specified for the joint per convolution. In toroidal convolutions,

$$\sigma_x = \pm \frac{Et \Delta}{4.33R^2}$$

and the stiffness of the joint per convolution is

$$\frac{F}{\Delta} = \frac{E dt^3}{8.91a^3}$$

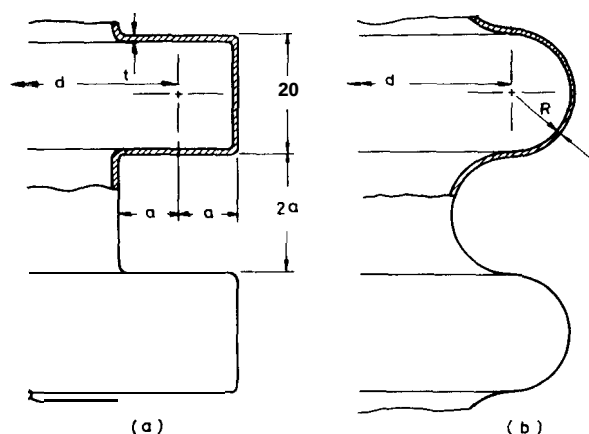


Figure 1 Expansion joints in heat exchangers. (a) Square convolution. (b) Toroidal convolutions.

for the square type and

$$\frac{F}{\Delta} = \frac{E dt^3}{3.34R^3}$$

for toroids.

Nomenclature and References for Section 4.1.6 appear at the end of Section 4.1.8.

4.1.7

Flanges

C. Ruiz

■ One of the major sources of operational failures is the leakage of bolted flanged joints, which often need continuous attention. Methods of design treat flanges as rings and assume elastic behavior of all the constitutive elements, including bolts and gasket [1, 3, 6]. This is very far from the truth. The inadequacy of the elastic analysis to model a complex situation in which the bolts are usually loaded above their yield point and the gasket creeps under the applied contact pressure explains the difficulties in service. Attempts made to improve on the conventional method of design proposed by standard codes [1, 6] have succeeded only in pointing out the fundamental flaws in the elastic model. As a result of plastic deformation, the contact pressure is not uniformly distributed over the gasket, changing with time. In the conventional type of Fig. 1a, the point of application of the gasket reaction is undefined and the com-

bined moment applied to the flange also varies with loading conditions, time, and temperature. The flange rotation cannot be calculated with any accuracy. It is only as a result of the empirical rules developed after many years of practical experience and embodied in the pressure vessel and heat exchanger codes that satisfactory designs can be produced. Joints with self-energized gaskets, like the type illustrated in Fig. 1b, on the other hand, can be analyzed with some accuracy, and a design based entirely on a theoretical analysis is feasible. The same is true of the boltless joint of Fig. 1c. In both cases plastic deformation is either avoided or highly localized—for example, under the ring. The overall behavior is elastic, and the position of the reactions is well defined and does not change with time. The free-body method of analysis, which treats flanges as annular plates, may be used. Alternatively, the finite-element

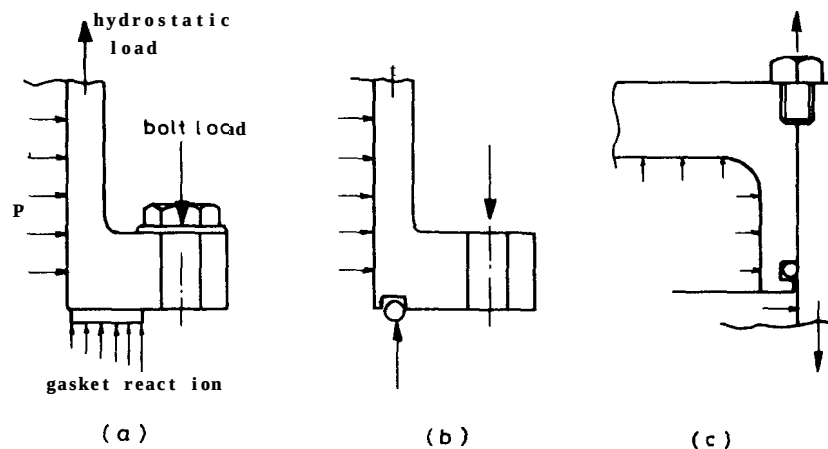


Figure 1 Bolted flanged joint. (a) Conventional joint with a gasket. (b) Self-energized ring gasket. (c) Boltless joint.

method may be preferred as providing a more accurate model of the joint. In either case, bolts are replaced by a cylinder with the PCD and the same metal cross-sectional area as the total bolt area, and the effect of the holes on the stiffness of the flange is included in the analysis, as was done for the perforated tube plates [46]. The problem is thus reduced to one with rotational symmetry. This method has been applied to the design of closure heads for nuclear reactor vessels [47, 48]. The design of boltless closures has been described elsewhere [49]. A survey of this and other types of joints has also been published by the author [SO].

The effect of a bending moment on a ring is to cause a rotation about the centroid, in such a way that the fibers above the neutral axis x - x in Fig. 2 are compressed and the fibers below it are loaded in tension.

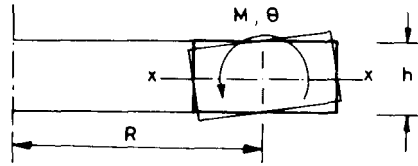


Figure 2 Stresses and rotation of a ring.

It can be shown [12] that the rotation is

$$\theta \cong \frac{MR^2}{EI_x}$$

where I_x is the second moment of area wrt, x - x . The maximum stresses are

$$\sigma = \pm \frac{Mh}{2I}$$

Nomenclature and References for Section 4.1.7 appear at the end of Section 4.1.8.

4.4.8

Heads, openings, and branches

C. Ruiz

A. Heads

The stresses in flat, hemispherical, torispherical, and ellipsoidal heads can be found by means of the general methods of analysis previously described. Some important results are summarized in Figs. 1-3 [3]. Plastic collapse under internal pressure is the mode of failure usually considered to govern the dimensioning of these shells, and limit analysis design has therefore been proposed as an alternative to elastic analysis [51, 52]. In addition, compressive stresses are set up in the circum-

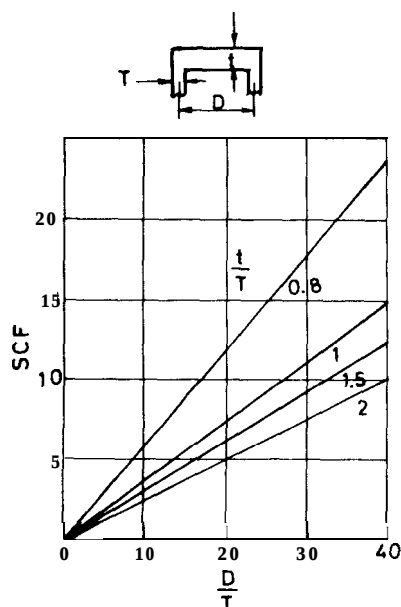


Figure 1 Maximum stress in a flat head.

ferential direction that may cause buckling in torispherical heads with a sharp transition between the cylindrical body and the spherical crown [53 to 55].

B. Openings and branches

The traditional method for the compensation of openings and branches is to add a pad or ring, or to increase the thickness of the branch in such a way that the cross-sectional area of the reinforcement equals the cross-sectional area of the metal removed from the shell. The limits fixed to the compensation depend on the design codes and empirical rules guide the detailed design of welds, transition radii, and so on. Modern

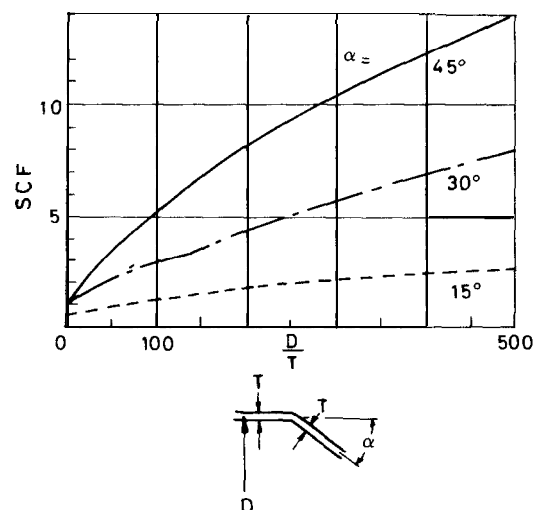


Figure 2 Maximum stress at the junction between a cylindrical shell and a conical transition.

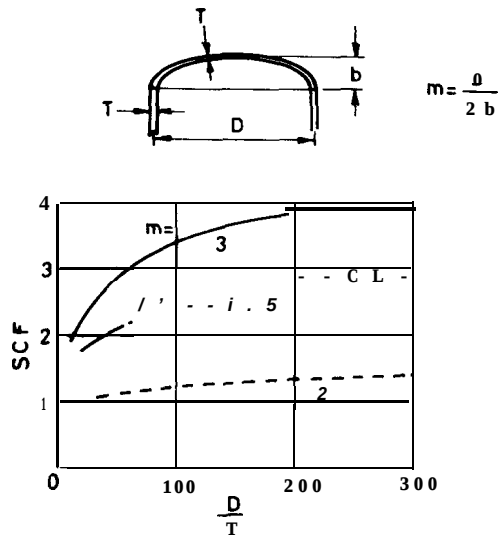


Figure 3 Maximum stress in an ellipsoidal head.

developments of limit analysis techniques have made possible a reassessment of the design philosophy and resulted in a more rational approach based on the prevention of plastic failure under static loading and, in the case of cyclic loads, the elastic shakedown of the assembly [56-60]. The diagram of Fig. 4 summarizes the results of limit analysis and elastic shakedown analysis and compares them with the design curves of the A. D. Merkblatt B9 [60,7]. The dimensionless pressure p^* is defined as the ratio between the bursting pressure of the plain shell and that of the shell with the branch. p' is the maximum cyclic pressure for elastic shakedown and $v_{0.2}$ is the weakening factor defined by the A. D. Merkblatt as

$$v_{0.2} = \frac{\text{pressure to produce 0.2\% set strain in nozzle}}{\text{pressure to produce 0.2\% set strain in plain shell}}$$

From Fig. 4, a branch thickness may be chosen for any specified weakening as characterized by p^* , p' , or $v_{0.2}$.

Elastic stress concentration factors for various types of reinforcements have been obtained by means of the free-body method [61,62], finite elements [63,64], and experimentally [30,65]. The problem is complicated by the pronounced effect on the peak stress of small departures from the theoretical shape due to manufacturing tolerances.

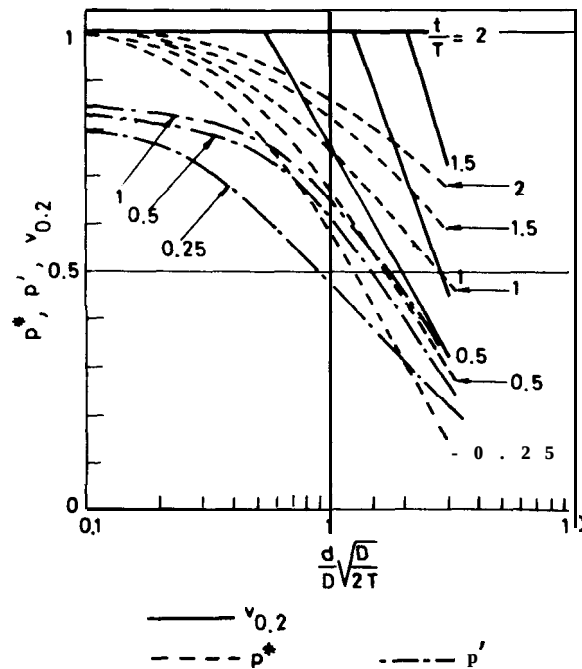


Figure 4 Dimensionless bursting pressure for a branch reinforced opening (nozzle) in a cylindrical shell, where d = nozzle bore, t = nozzle thickness, D = shell bore, and T = shell thickness.

NOMENCLATURE FOR SECTIONS 4.1.1-4.1.8

a	cross-sectional area of tubes or reinforcing rings	C_4, C_5	parameters for stress analysis of saddle supports
A	cross-sectional area of bars or bore of shell; distance between ring reinforcements or supporting saddle and domed end in horizontal vessel	d	tube diameter; bore; distance between centroid of reinforcing ring and end of web
b	depth of domed end; inside radius of annular plate	D	diameter of shell; stiffness defined by Eq. 4.1.3(3)
b_1	width of supporting saddle	D^*	equivalent stiffness of perforated plate
B	cross-sectional area of shell plate	E	Young's modulus
c	distance between centroid of reinforcing ring and inside wall surface	E^*	equivalent Young's modulus of perforated plate
c	cross-sectional area of perforated part of shell plate	f	tube pitch
		F	force
		G_1, G_2	stress functions
		h	thickness of plate or ring
		H	distributed edge force

I	second moment of area	ν^*	equivalent Poisson's ratio for perforated plate
$K_1, K_2, K_3, K_5, K_7, K_8, K_{10}$	parameters for stress analysis of saddle supports	σ	stress
l	length	σ_u	ultimate tensile stress
L	length of cylindrical barrel in vessel with domed ends	σ_Y	yield stress; 0.2% proof stress
M	moment; bending stress resultant	σ_R	stress required to produce failure after 100 000 h at design temperature
n	number of tubes	σ_T	stress required to produce a permanent strain of 1% after 100 000 h at design temperature
N	normal force; membrane stress resultant	τ	shear stress
p	pressure	ϕ	meridional coordinate; angle of saddle with integral reinforcement
p_1, p_2	shell-side and tube-side pressure, respectively	χ	rotation at ends of a shell of revolution
p_c	critical buckling pressure		
q	shear stress		
Q	parameter ($= E_r n a / E_s B$)		
r, R	radius		
s	design stress; distributed shear at the edge of a plate		
SCF	stress concentration factor		
S_D	fatigue strength		
t	thickness		
T	thickness; temperature		
u	displacement; expansion		
v	vertical displacement		
$v_{0.2}$	weakening factor		
W_1	load applied at saddle		
x	distance between ring reinforcements		
Z	minimum section modulus		
α	coefficient of thermal expansion; cone angle		
β	shell parameter defined by Eq. 4.1.3(3)		
γ	rotation of plate at edges		
A	expansion of joint per convolution; increment		
θ	angle of saddle; circumferential coordinate; rotation of ring		
ν	Poisson's ratio		

Subscripts

c	circumferential; central
i	inside
L	longitudinal
o	outside
n	normal
r	radial direction
s	support; shell
t	tube
T	torsional
x	axial direction
θ	hoop direction
ϕ	meridional direction in shells
1,2	principal stresses; ends of a shell of revolution
3	third principal stress
I,II,III,IV	types of stress (defined in text)

Superscripts

*	equivalent magnitude in perforated plate
'	derivative

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4.2

Shell-z-tube heat exchangers: Elements of construction

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4.2.1

Introduction

E. A. D. Saunders

■ A variety of unfired heat transfer equipment is used in industry, but the most common is the shell-and-tube exchanger. Although it is not especially compact, it has the advantage of being robust and versatile. Except for the special-purpose finned-tube air cooler, it is usually the only design that can be considered for large surface areas, pressures greater than 2 MN/m^2 , and temperatures greater than 250°C .

As the name implies, a shell-and-tube heat exchanger consists of a shell (pressure vessel) containing a tube bundle, which is attached to stationary and rear heads. The tubes, which are attached to tubesheets, may be plain or tinned, and run parallel to the longitudinal axis of the shell.

Section 4.2.2 deals with design and construction codes and describes a particular system for designating the various shell configurations, stationary heads, and rear heads. The design of the rear head establishes the type of shell-and-tube exchanger, and the different types, with variations, are described in Sec. 4.2.3. The purpose of the various stationary heads is described in Sec. 4.2.4. Sections 4.2.5 and 4.2.6 deal with each component of a shell-and-tube heat exchanger and describe its thermal or mechanical function.

The nomenclature of the shell-and-tube heat exchanger components is given in Table 4.2.3(1).

4.2.2

Design and construction codes

E. A. D. Saunders

A. TEMA [1]

The pressure parts of a shell-and-tube heat exchanger are designed in accordance with a pressure vessel design code such as ASME [2], BSS 5500 [3], A. D. Merkblatter [4], and so on, but a pressure vessel design code alone cannot be expected to deal with all the special features of shell-and-tube heat exchangers. To give guidance and protection to designers, fabricators, and purchasers alike, a supplementary code is desirable that provides minimum standards for design, materials, thicknesses, corrosion allowances, fabrication, tolerances, testing, inspection, installation, operation, maintenance, and guarantees for shell-and-tube heat exchangers.

One universally accepted code that does this is the **Standards Of Tubular Exchanger Manufacturers Association**, known as TEMA. Although TEMA is designed specifically to supplement the ASME Boiler and Pressure Vessel Code, Section VIII, Division 1, a large portion of it may be used to supplement other pressure vessel codes if required.

TEMA is applicable to shell-and-tube heat exchangers with the following limitations:

- Shell diameter not exceeding 1 524 mm
- Pressure not exceeding 2 1 MN/m²
- Product: shell diameter X pressure not exceeding 10 500 (mm X MN/m²)

The intent of the above limitations is to limit the barrel thickness and stud diameter to approximately 50 mm and 75 mm, respectively. A special section entitled Recommended Good Practice is provided to give the designer guidance for shell diameters up to

2 540 mm where design temperatures and materials of construction result in reasonable thicknesses for pressure containing exchanger parts.

There are three mechanical classes in TEMA:

Class R: generally severe requirements of petroleum and related processing applications

Class C: generally moderate requirements of commercial and general process applications

Class B: chemical process service

B. Type designation

Sections 4.2.3 and 4.2.4 show that there are a variety of exchanger and head types. TEMA provides a simple three-letter system that completely defines almost all shell-and-tube exchangers with respect to exchanger type, stationary end head, rear end head, and shell-side nozzle configuration, the system being shown in Fig. 1. The first letter defines the stationary end head, the middle letter defines the shell type, and the last letter defines the rear end head type.

Figures 4.2.3(3), 4.2.3(5), and 4.2.3(7) show all exchangers having type A stationary heads and type E shells, so that the U-tube, split-backing-ring, and pull-through floating-head exchangers would be designated AEU, AES, and AET, respectively. All could be fitted with bonnet-type stationary heads if required, in which case they would be designated BEU, BES, and BET, respectively. The fixed tubesheet exchanger shown in Fig. 4.2.3(1) has a type A stationary head and a type M rear head, so that it is designated AEM. The bobbin-type stationary head could be replaced by a bonnet-type head

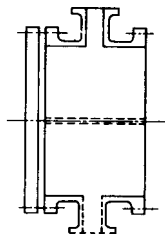
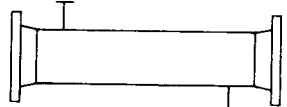
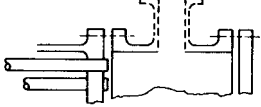
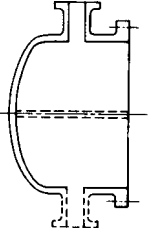
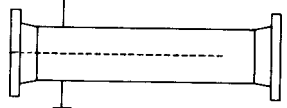
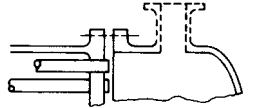
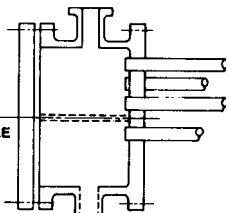
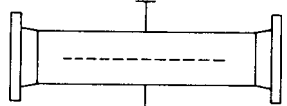
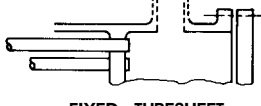
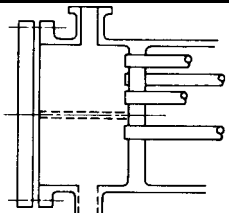
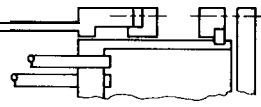
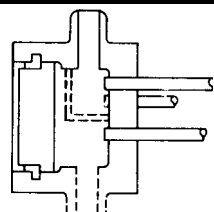
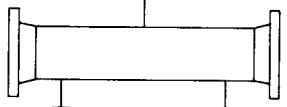
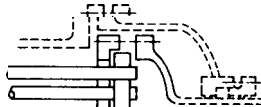
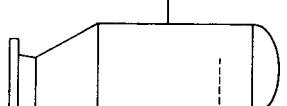
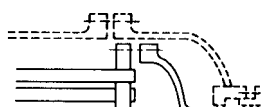
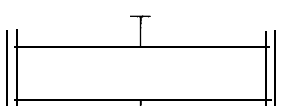
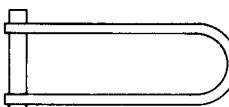
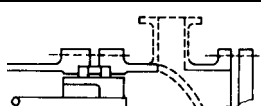
FRONT END STATIONARY HEAD TYPES		SHELL TYPES		REAR END HEAD TYPES	
A		E		L	
	CHANNEL AND REMOVABLE COVER				
B		F		M	
	BONNET (INTEGRAL COVER)				
C		G		N	
	REMOVABLE TUBE BUNDLE ONLY				
N		H		P	
	CHANNEL INTEGRAL WITH TUBE SHEET AND REMOVABLE COVER				
D		J		S	
	SPECIAL HIGH PRESSURE CLOSURE				
		K		T	
		X		U	
				N	
					EXTERNALLY SEALED FLOATING TUBESHEET

Figure 1 TEMA designation system. From Standards of Tubular Exchanger Manufacturers Association, 6th ed., 1978, figure N-1.2.

if required, in which case it would be designated BEM. The kettle shown in Fig. 4.2.3(10) would be designated AKT.

The simple system avoids confusion between fabricator and purchaser regarding the exchanger configuration and enables the latter to ensure that bids are compared on the same basis. For **instance**, a fixed tubesheet exchanger may have nine combinations of stationary and rear heads (omitting type D heads).

C. Size designation

The size of an exchanger is designated by the inside diameter/tube length; that is, an exchanger with 1 500-mm ID having 6 000-mm-long tubes would be designated 1 500/6 000. A kettle having a stationary-head inside diameter of 1 000 mm, a main-shell inside diameter of 1 600 mm, and a tube length of 7 500 mm would be designated 1 000/1600/7 500.

REFERENCES FOR SECTION 4.2.2

1. Tubular Exchanger Manufacturers Association, Inc., 331 Madison Avenue, New York, N.Y. 10017.
2. American Society of Mechanical Engineers, United Engineering Center, 345 East 47th Street, New York, N.Y. 10017.
3. British Standards Specification 5500: 1976, British Standards Institution, 2 Park Street, London W1A 2BS.
4. A. D. Merkblltter, Carl Heymanns Verlag KG, 5 Köln 1, 18-32, F.R. Germany.

4.2.3

Types of shell-and-tube heat exchangers

E. A. D. Saunders

A. Introduction

Heat transfer equipment may be designated by type or by the function it performs, such as chiller, condenser, cooler, heater, reboiler, vaporizer, and so on. The choice of shell-and-tube heat exchanger type is governed chiefly by factors such as provision for differential movement between shell and tubes, design pressure, design temperature, and fouling nature of the fluids, rather than the function. Almost all exchanger types can perform any function.

The various shell-and-tube heat exchanger types, and their variations, are described below, the type letter relating to the particular designation system referred to

earlier in Sec. 4.2.2B. Nomenclature for heat exchanger components is summarized in Table 1. The features of the various types are summarized in Table 2.

B. Fixed tubesheet or fixed head (type L, M, or N)

In this type, shown in Figs. 1 and 2, both tubesheets are welded to the shell, forming a box : for this reason it is sometimes referred to as a box-type exchanger. After removing the head covers, or complete heads, at each end, access to the tube ends is obtained and the inside of the tubes may be cleaned by mechanical means.

Table 1 Nomenclature for heat exchanger components (applicable to Figs. 1, 3, 5, 7, 8, and 9)

Number	Component	Number	Component
1	Shell	16	Stationary-head nozzle
2	Rear-head dished cover	17	Stationary-head cover flange
3	Floating tubesheet	18	Stationary-head/tubesheet flange
4	Floating-head cover	19	Shell flange at stationary head
5	Floating-head backing ring	20	Shell flange at rear head
6	Floating-head cover flange	21	Rear-head flange at shell
7	Stationary-head cover	22	Rear-head barrel
8	Tie rods and spacers	23	Stationary-head barrel
9	Baffle	24	Packing
10	Stationary tubesheet	25	Packing follower ring
11	Pass partition plate	26	Packing box flange
12	Tubes	27	Floating tubesheet skirt
13	Support saddle	28	Slip-on backing flange
14	Expansion bellows	29	Split shear ring
15	Shell nozzle	30	Lantern ring with weep hole

Table 2 Types of shell-and-tube heat exchangers: Summary of features

Characteristic	Fixed tubesheet (type L, M, N)	U tube (type U)	Split-backing-ring floating head (type S)	Pull-through floating head (type T)	Packed-lantern-ring floating head (type W)	Outside-packed floating head (type P)	Bayonet tube
Does it provide for differential movement between shell and tubes?	Yes-with bellows in shell	Yes	Yes	Yes	Yes	Yes	Yes
Is tube bundle removable?	No	Yes	Yes	Yes	Yes	Yes	Yes^a
Is replacement bundle possible?	No	Yes	Yes	Yes	Yes	Yes	Yes^a
Can individual tubes be removed and replaced?	Yes	Outer tubes only	Yes	Yes	Yes	Yes	Yes
Method of cleaning inside	Any	Chemical	Any	Any	Any	Any	Any
Method of cleaning outside^b	Chemical	Any	Any	Any	Any	Any	Any^a
Number of tube-side passes	Any	Any even number	One or any even number ^c	One or any even number	One or two only^d	Any^e	One (special)
Is double tubesheet construction permissible?	Yes	Yes	No	No	No	Yes	Yes
Are there internal gaskets?	No	No	Yes	Yes	No	No	No
Approx. diametral clearance, mm (Shell ID—OTL) ^f	11 to 18	11 to 18	35 to 50	95 to 160	15 to 35	25 to 50	11 to 18

^aAssumes tubesheet for outer tubes not welded to shell.^bExternal mechanical cleaning possible only with square or rotated square pitch, or unusually wide triangular pitch.^cOne-pass construction requires packed gland or bellows at floating head. See Figs. 4.2.5(4) and 4.2.5(5).^dTube-side nozzles must be at stationary end for two passes.^eAxial nozzle required at rear end for odd number of passes.^fSubject to comments given in Secs. 4.2.5D and 4.2.5E.

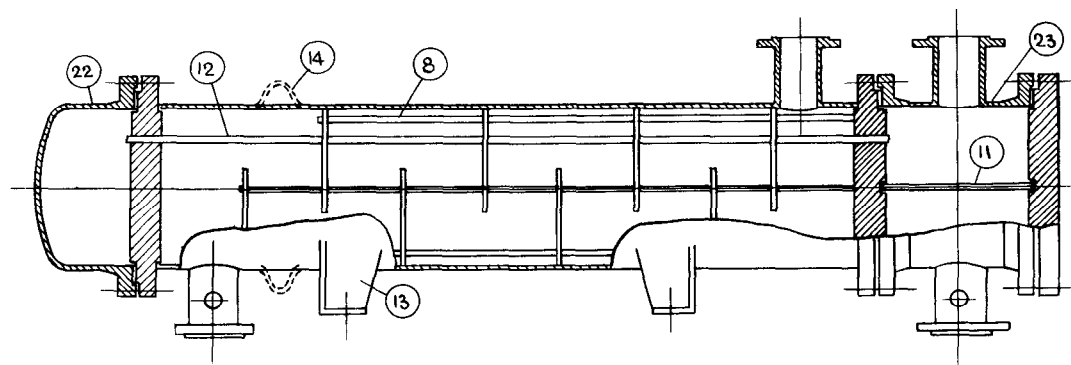


Figure 1 Fixed tubesheet exchanger.

As the tube bundle cannot be removed, cleaning of the outside of the tubes is usually carried out by chemical means. The fixed tubesheet type is therefore limited to applications where the shell-side fluid is nonfouling; any fouling fluids must be routed through the tubes.

It is not possible to replace the tube bundle without cutting the shell, but by using special tube cutters and extractors it is possible to withdraw individual tubes and replace them with new ones. Sometimes during this operation a tube may lodge or break inside the shell, in which case the relevant tube hole in each tubesheet is plugged.

In the construction shown, the shell barrel and tubesheets should be made of metals that can be welded readily to one another, such as steel-steel, but not steel-aluminum or steel-brass. When incompatible shell barrel and tubesheet metals occur, the most common practice is to use bimetal tubesheets, the required tubesheet

metal being placed on the tube side of the sheet and the required shell barrel metal on the shell side. An alternative solution to the problem of incompatible metals is to weld a flange to each end of the shell, in the same metal as the shell, and bolt them to the tubesheets using suitable gaskets. This design is not recommended, however, except for low-pressure, nonlethal service, as the gasket cannot be replaced and the joint has little flexibility for tightening.

The combination of temperatures and thermal expansion coefficients of the shell and tubes during service may cause a differential movement between them. If the differential movement is excessive, the tubes will loosen in the tubesheets and it is therefore important to check the tubesheet design against all differential movements likely to occur, not only during steady operation, but also at startup, shutdown, cleaning out, and so on. A fixed tubesheet exchanger, without a shell

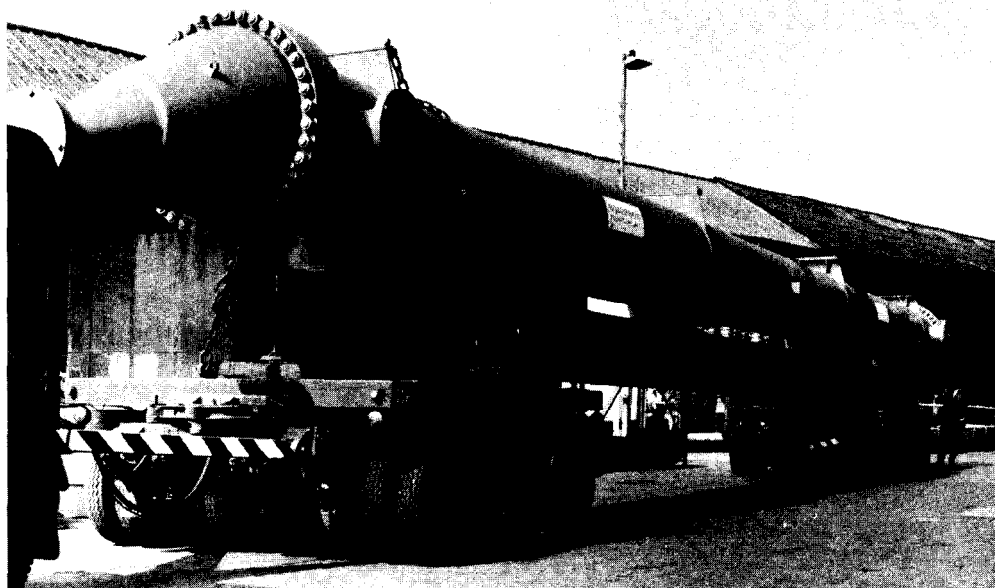


Figure 2 Fixed tubesheet exchanger, 1 067-mm diameter with 21 945-mm-long tubes. Courtesy of Whessoe Heavy Engineering Ltd., Darlington, England.

expansion bellows, can absorb a limited differential movement, but when this becomes excessive a bellows must be used. However, large-diameter, high-pressure bellows, designed for large movements, are expensive, and in such cases it may be prudent to consider an alternative type of shell-and-tube exchanger.

The fixed tubesheet exchanger has the important advantage of having no internal joints, thus eliminating a potential source of leakage of one fluid into the other. In addition, the absence of an internal joint means that peripheral tubes can be placed close to the inside of the shell and more tubes can be accommodated in a given shell diameter for the fixed tubesheet type than for any other type.

C. U tube or hairpin (type U)

The U-tube or hairpin type, shown in Fig. 3, has one tubesheet with each tube free to move relative to the shell, thereby eliminating any problems of differential movement between shell and tubes. As the tube bundle can be withdrawn, the outside of the tubes may be cleaned by mechanical means, but chemical cleaning is usually necessary for the inside of the tubes. The U-tube type is therefore limited to applications where the tube-side fluid is nonfouling; any fouling fluids must be routed through the shell.

Although U-tube exchangers are made in which all U tubes are identical, this may impose thermal design restrictions, and in the majority of cases U tubes having varying-bend radii are used. As a result, once the bundle has been made, only U tubes at the periphery are accessible for renewal, so that split tubes in the inner rows must be plugged.

In common with the fixed tubesheet type, the U-tube type has no internal joints and peripheral tubes can be placed close to the inside of the shell. However, the number of tubes accommodated within a given shell diameter is slightly less than that for the fixed tubesheet type, because there is a practical limit to the bend radius of the innermost U's.

A particular application for the U-tube type is the tank suction heater shown in Fig. 4. The purpose of the tank suction heater is to heat heavy fuel oils, and other viscous liquids, during pumping from a storage tank in order to reduce the viscosity of the liquid and assist pumping. The tank suction heater eliminates the need to maintain the entire tank contents at the required pumping temperature. Steam, which flows through the tubes, is the usual heating medium. The tubes may be plain or longitudinally **finned** externally. The end of the shell inside the tank is cut away to assist entry of the viscous fluid.

D. Split-backing-ring floating head (type S)

In the split-backing-ring floating-head type, shown in Fig. 5, one tubesheet is **fixed** with respect to the shell whereas the other tubesheet floats, thus permitting differential movement between shell and tubes, and also complete tube bundle withdrawal. To separate the **shell**- and tube-side fluids at the floating-head end, the floating tubesheet is fitted with a flanged cover at its periphery, which is held in position by bolting it to a split backing ring on the other side of the tubesheet. The **floating-head** assembly (flange, cover, tubesheet, and backing ring) is located beyond the main shell in a shell cover of larger diameter. A typical floating-head assembly for this type of exchanger is shown in Fig. 6.

Access to the tube ends at the stationary end is obtained by removing the stationary-head cover or complete head; access to the tube ends at the floating-head end is obtained by removing the shell cover, split backing ring, and floating-head cover. The inside of the tubes may then be cleaned in situ, or the complete bundle may be removed for cleaning the outside of the tubes or repair.

The split-backing-ring floating-head type has none of the disadvantages of the fixed tubesheet and U-tube types with regard to cleaning or differential movement between shell and tubes. However, there is an internal joint at the floating head, and careful design is

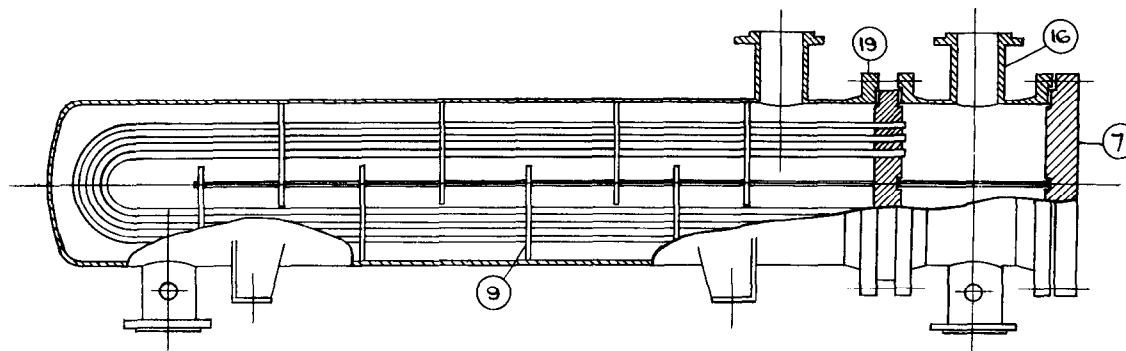


Figure 3 U-tube exchanger.

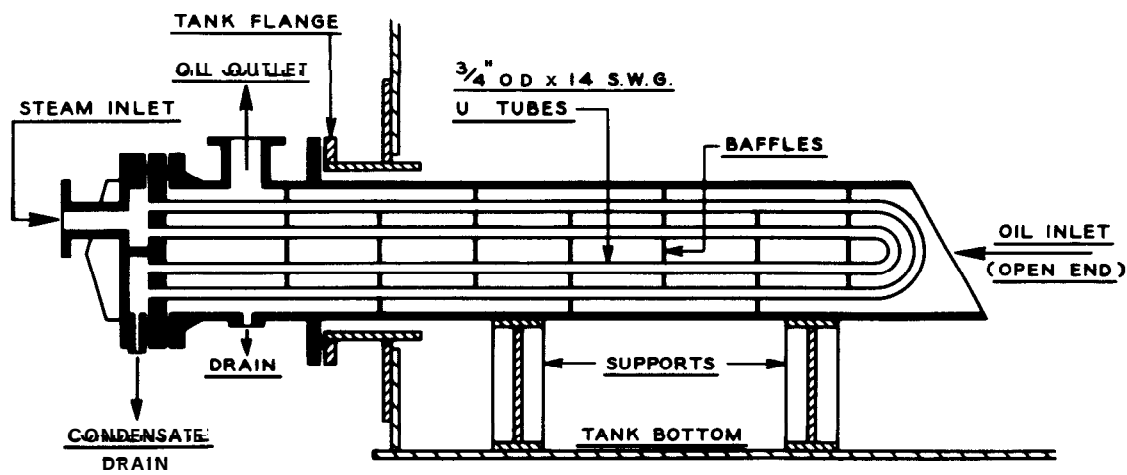


Figure 4 Tank suction heater.

necessary to avoid leakage of one fluid into the other. It will be seen from Fig. 6 that peripheral tubes cannot encroach beyond the floating-head gasket inside diameter, and for this reason the split-backing-ring floating-head type accommodates a smaller number of tubes than fixed tubesheet and U-tube types having the same shell diameter.

E. Pull-through floating head (type T)

Figure 7 shows that the pull-through floating-head type is similar in construction to the split-backing-ring type except that the floating-head cover flange is bolted directly to the floating tubesheet. The latter is increased in diameter to match that of the floating-head flange outside diameter, and a split backing ring is not required. As a result, the shell diameter is approximately the same as the enlarged shell cover diameter of the split-backing-ring type to contain the same number of tubes. The pull-through floating-head type accommodates the smallest number of tubes in a given shell diameter of all types. Figure 6 shows a typical floating-head assembly.

The advantage of the pull-through type, compared

with the split-backing-ring type, is that the bundle may be withdrawn from the shell without removing either the shell or the floating-head cover, thus reducing maintenance time. A significant reduction in cost may be made by omitting the two rear-head flanges, but this means that the tubes cannot be cleaned or inspected in situ and the tube bundle must always be withdrawn and inserted blind. Usually this modification is permitted only if fouling problems are not expected to be serious.

As in the case of the split-backing-ring type, there is an internal joint, and careful floating-head design is required to avoid leakage of one fluid into the other.

F. Packed-lantern-ring floating head (type W)

The feature of the packed-lantern-ring floating-head exchanger, shown in Fig. 8, is that separation of the shell- and tube-side fluids at the floating head is obtained by means of packing rings installed between the outside of the floating tubesheet and recesses in the rear-head flanges. The shell- and tube-side fluids each have their own packing rings, which are separated by a lantern ring

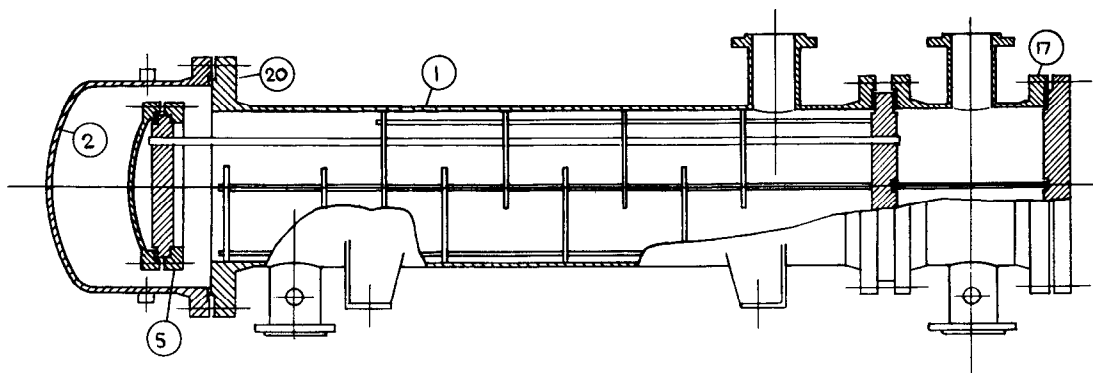


Figure 5 Split-backing-ring floating-head exchanger.

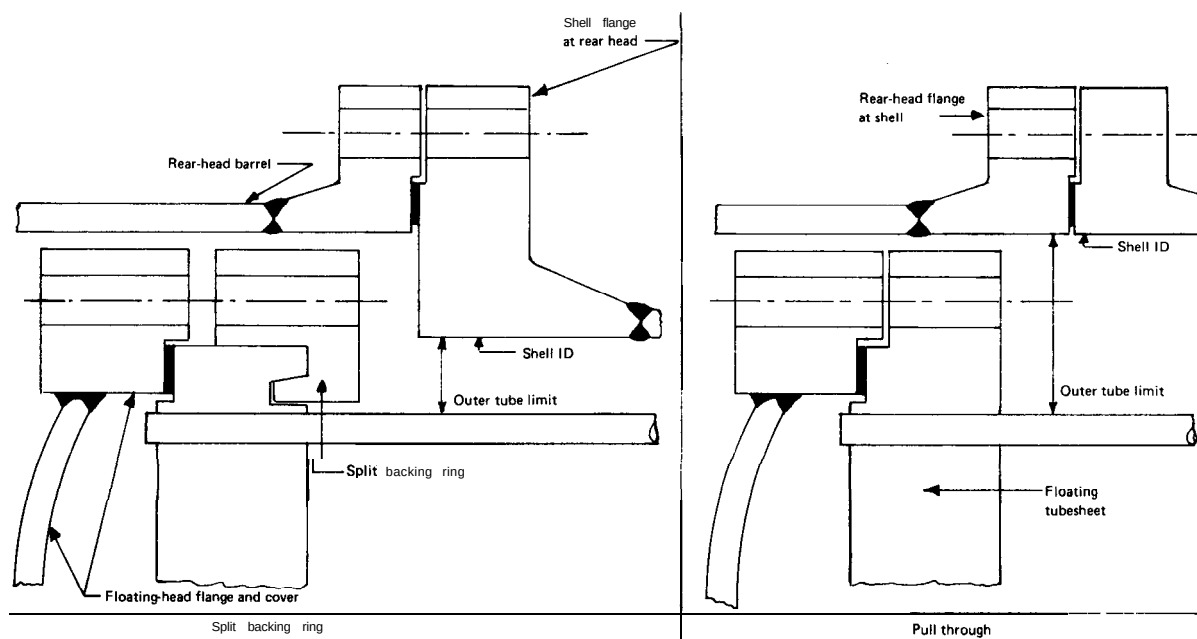


Figure 6 Typical floating-head assembly.

provided with weep holes. Any leakage at the packing will not cause mixing of the shell- and tube-side fluids within the exchanger itself.

The machined bearing surface at the floating tube-sheet must be wide enough to accommodate the two sets of packing, the lantern ring, and any thermal movements. If the pressure thickness of the tubesheet is appreciably less than the required bearing surface, a machined skirt is usually attached to the periphery of the sheet.

This type of exchanger provides all the advantages of floating-head construction in that the inside of the tubes may be cleaned in situ, the whole bundle may be withdrawn for cleaning or complete replacement, and differential movement between shell and tubes can be accommodated; in addition, there are no internal joints. Although this construction is the cheap-

est of the floating-head types, its use is limited to non-lethal service and low pressures. Typical limitations are shown below, the maximum design temperature being about 200°C:

Shell ID, mm	Maximum pressure, MN/m ²
Up to 600	2
Above 600 to 1 060	1
Above 1 060 to 1 525	0.5

If a skirt is provided at the floating head, the clearance between the inside of the shell and peripheral tubes is about the same as the split-backing-ring floating-head type. If a skirt is not required, this clearance may be reduced slightly but adequate allowance must be made

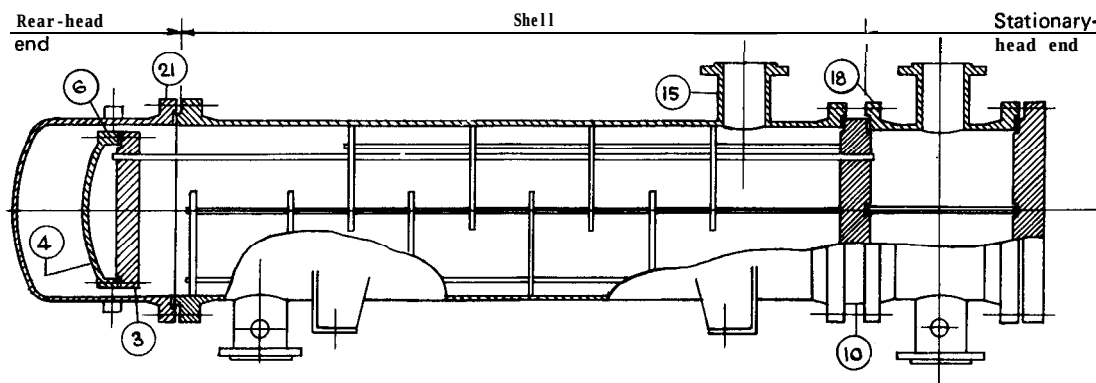


Figure 7 Pull-through floating-head exchanger.

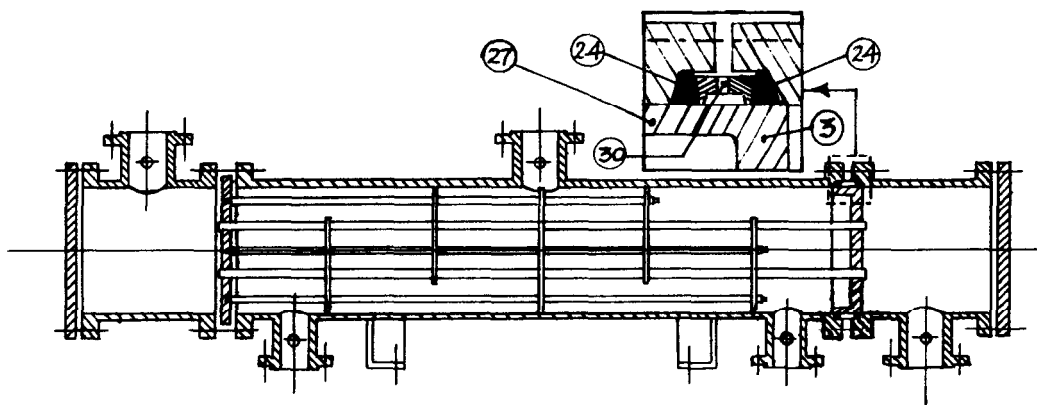


Figure 8 Packed-lantern-ring floating-head exchanger.

for peripheral tube hole distortion arising from expansion or welding of the tube ends.

This type is limited to one or two tube-side passes, and in the latter case the tube-side nozzles must be fitted to the stationary head.

G. Outside-packed floating head (type P)

Another type of floating-head exchanger is shown in Fig. 9. To contain the tube-side fluid at the floating head, one end of a cylindrical barrel (skirt) is welded to the periphery of the floating tubesheet, whereas the other end is fitted with a slip-on backing flange and flat cover. The backing flange is held in place by a split shear ring inserted into a groove in the wall of the skirt. The split shear ring enables the flange and cover to be removed, so that when the bundle is withdrawn, the skirt can slide inside the shell with only a small clearance. The shell-side fluid is contained by packing rings installed between the outside of the skirt, which has a machined finish, and recesses within the rear head flange. The packing follower ring is capable of being

slid off the skirt prior to bundle withdrawal. This type of exchanger offers all the features of the packed-lantern-ring type, with the added advantage that there are no limitations for the tube-side fluid or its design pressure and temperature. The shell-side fluid must be nonlethal and is usually limited to about 300°C and 2 MN/m².

The clearance between the inside of the shell and peripheral tubes is governed by the tube-side design pressure, and for moderate pressures this clearance will be at least of the same order as that of the split-backing-ring type.

H. Kettle-type reboiler (type K)

The kettle-type reboiler is a special application of the U-tube and pull-through type of constructions described earlier, typical designs being shown in Figs. 10 and 11. It is used for vaporization of the shell-side fluid, and the special feature of the design is the space above the tube bundle, which is designed so that disengagement of the vapor and entrained liquid droplets can occur within the

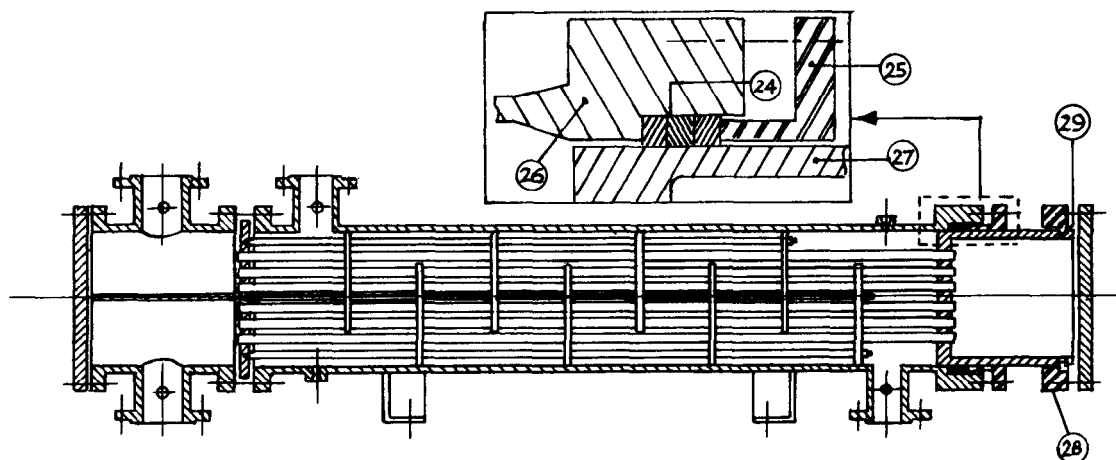


Figure 9 Outside-packed floating-head exchanger.

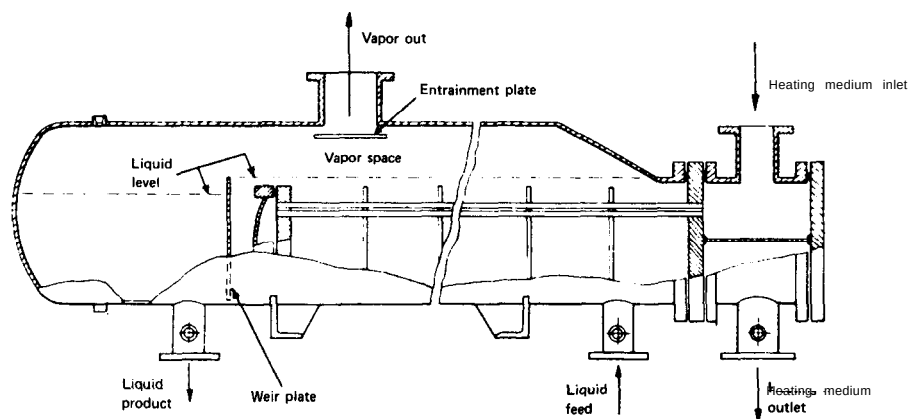


Figure 10 Kettle-type reboiler.

exchanger itself. The calculation of vapor space depth is based on the vapor quantity, density, the number of vapor exit nozzles, and the degree of liquid entrainment permitted, but in typical cases it is about one-third of the shell diameter. It is usual to have a minimum vapor space depth of about 225 mm. In order to reduce liquid entrainment still further, demister pads or dry pipes are sometimes fitted in the vapor exit nozzles. A dry pipe is a horizontal length of pipe, closed at both ends and perforated with small holes along its length. The liquid droplets are knocked out as the vapor passes through the holes.

A weir plate is fitted to ensure that the liquid level in the reboiler is maintained and that the surface

of the upper tubes is not exposed. Vaporization is limited to about 80% of the liquid feed, and the space between the weir and the end of the shell is sized to provide adequate surge capacity.

Although Fig. 10 shows a pull-through floating-head bundle, U tubes may be used if process conditions permit. In most cases, for economy, the diameter of the stationary head is made just large enough to accommodate the tube bundle for the U-tube type, and the complete floating head assembly for the pull-through type, the transition from the enlarged shell to the smaller stationary head being achieved with a cone. However, it is not essential for the shell to be reduced in diameter and if there are no mechanical design problems, the cone

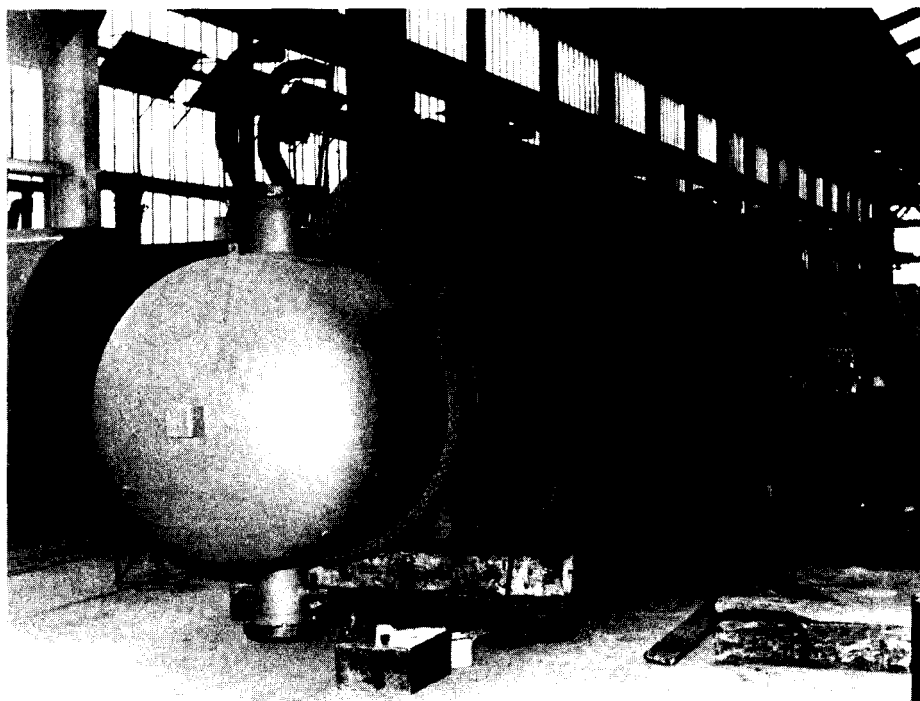


Figure 11 U-tube kettle, 1 778/2 845 mm diameter with 7 925-mm-long straight-leg tubes. Courtesy of Whessoe Heavy Engineering Ltd., Darlington, England.

may be replaced by a flat or domed end having the same diameter as the shell.

In the case of U-tube and floating-head construction, a double-ended kettle, shown in Fig. 12, may be provided, in which a bundle is inserted into each end of the shell, the shell usually being coned at both ends. The single double-ended unit is equivalent to two complete kettles operating in parallel.

I

In a chiller, a fluid is cooled to a temperature lower than that which could be obtained if water were used as the coolant. Except for the omission of the weir plate, a chiller is similar to the kettle-type reboiler. The fluid to be cooled flows inside the tubes and a refrigerant, such as ammonia or Freon, vaporizes on the outside of the tubes. Because of the moderate temperature difference between fluids and the absence of fouling, fixed tube-sheet construction is often used. The transition from the enlarged shell to the smaller stationary and rear heads is usually made with a cone at each end, similar to the double-end unit described earlier. Again, it is not essential for the shell to be reduced in diameter and the cones may be replaced by flat or domed ends, having the same diameter as the shell, if conditions permit.

J. Double-bundle vaporizer

A special non-TEMA type of vaporizer is the double-bundle type illustrated in Fig. 13, which is particularly

suited to the vaporization of liquids at low temperature, with steam as the heating medium. Although a single-bundle unit can be designed for this duty, special care is required to prevent freezing of the condensate. This is likely to occur if the condensate is prevented from flowing away freely due to malfunction or inadequate design.

To overcome this, a buffer fluid is used on the shell side of the double-bundle design. The lower bundle acts as a kettle in which the buffer fluid is vaporized by condensing steam in the tubes. The vaporized buffer fluid condenses on the tubes of the upper bundle to preheat and vaporize the low-temperature liquid in the tubes, the buffer fluid condensate falling back to the boiling pool around the lower bundle. The buffer fluid and its vaporizing/condensing temperature are selected with regard to (1) optimum bundle sizes, (2) moderate shell-side design pressure, and (3) the elimination of freezing in either bundle. In a typical system, steam at 205°C is used to preheat and vaporize liquid ethylene from -104°C to 10°C, methanol being the buffer fluid at 120°C.

K. Bayonet tube

In the bayonet tube, shown in Fig. 14, the inner tube, outer tube, and shell are completely free to move independently of one another. This type is particularly suited to extremely large temperature differences between the two fluids. The free end of each outer tube must be sealed with a cover, thus providing an internal

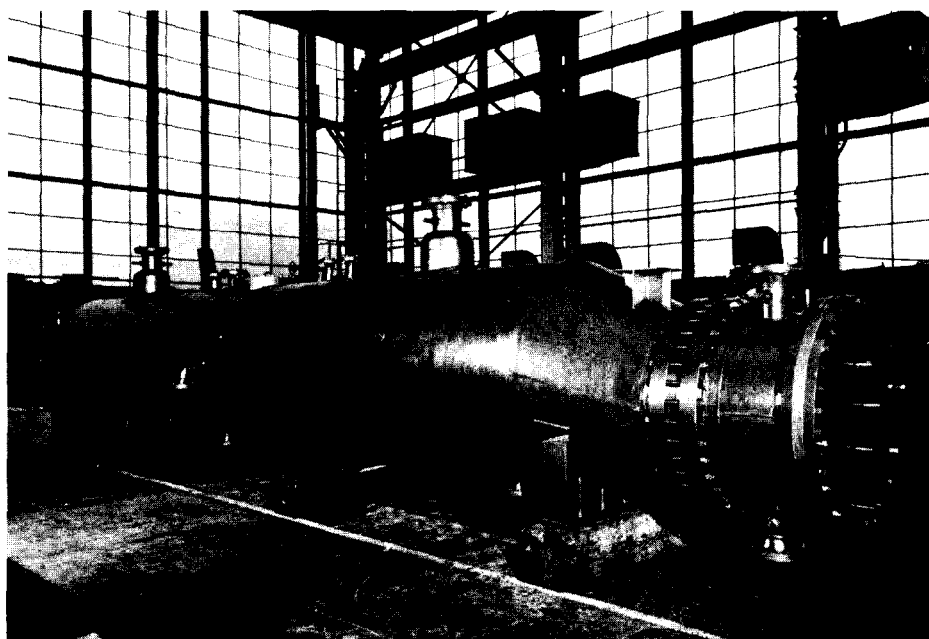


Figure 12 Double-ended U-tube kettle, 1 016/1 371 mm diameter. Courtesy of Whessoe Heavy Engineering Ltd., Darlington, England.

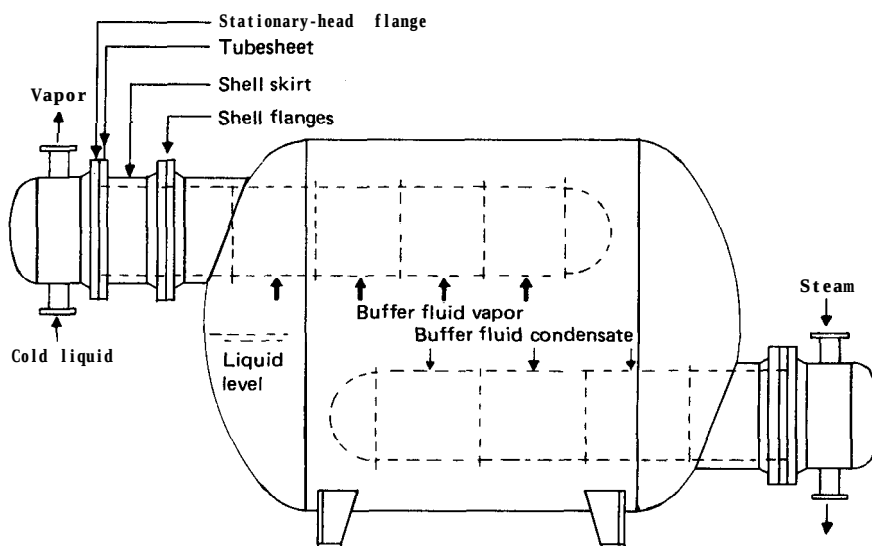


Figure 13 Double-bundle vaporizer.

joint, but because the cover can be welded to the tube it does not present the same risk of leakage as a **gasketed** joint. The heat transfer surface is that of the outer tube.

The shell side may be provided with baffles, similar

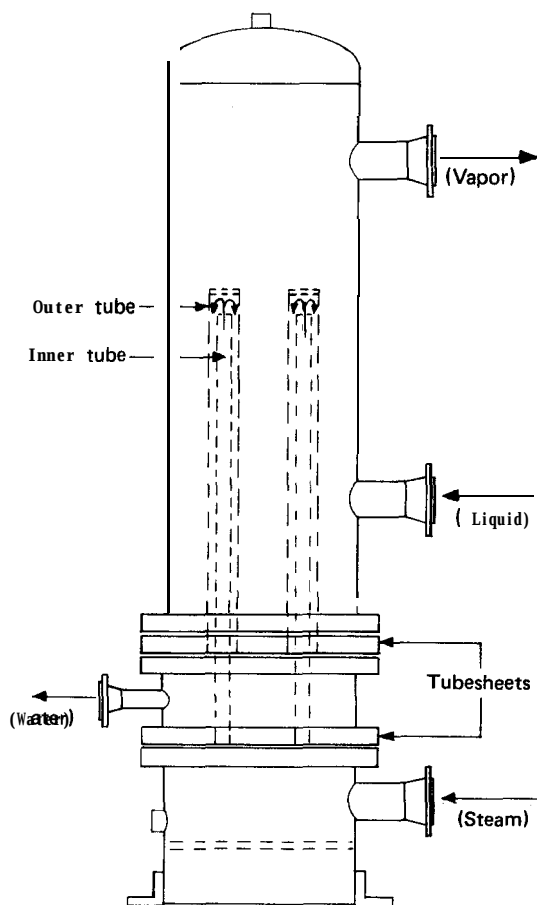


Figure 14 Bayonet tube exchanger.

to the other types above, but short vertical units are sometimes unbaffled and are particularly suited to condensing and vaporizing duties on the shell side where a low pressure loss is essential. The short vertical tube unit shown in Fig. 14 may be used to vaporize liquids at low temperature using steam as the heating medium. The presence of steam in the inlet head and inner tubes prevents condensate freezing.

Both sets of tubes may be welded to their respective tubesheets if required. This may be essential for the outer tubes, as their tubesheets may have to withstand a considerable pressure; it is less important for the inner tubes, as their tubesheets merely have to withstand the pressure loss of the tube-side fluid.

L. Double tubesheets

In some processes the leakage of one fluid into the other, however small, could present a hazard or contaminate the process completely, the most likely source of leakage being the **tube/tubesheet** joints. Double tubesheet construction, as used in the unit shown in Fig. 15, may be used in which fluid leaking through the tube/tubesheet joints in either tubesheet will flow into the space between them, where it may be vented or drained away. No mixing of the fluids within the shell or heads can occur.

Figure 16 shows that double tubesheet construction may be applied to fixed tubesheet, U-tube, **floating-head**, and bayonet-tube types, but in the case of the floating-head type, outside-packed construction is required as shown in Fig. 9. Although the tubes in the outer tubesheet may be attached by expansion or welding, the tubes in the inner tubesheet can be attached only by expansion.

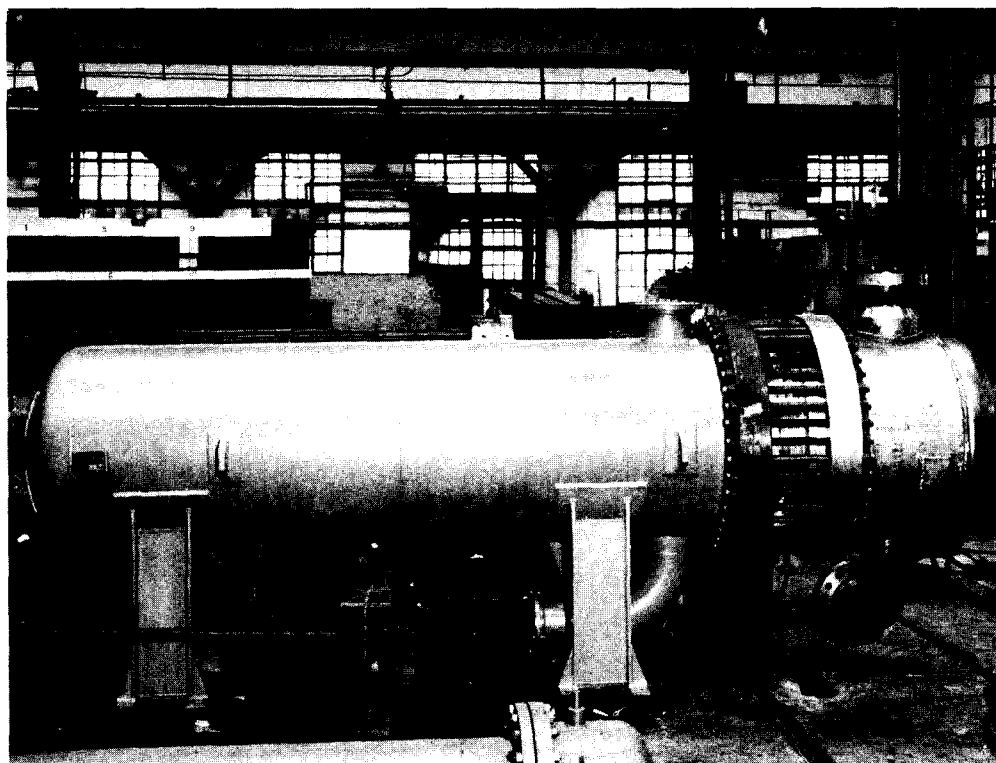


Figure 15 Double-tubesheet U-tube exchanger, 1 066-mm diameter with 3 657-mm-long straight-leg tubes. Courtesy of Whessoe Heavy Engineering Ltd., Darlington, England.

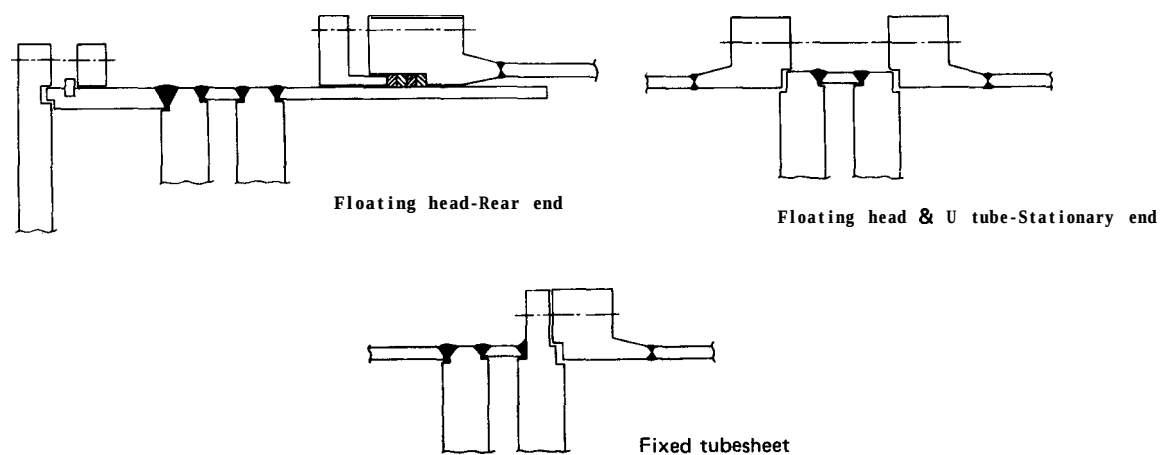


Figure 16 Double tubesheet construction.

4.2.4

Head types

E. A. D. Saunders

A. Introduction

The various exchanger types shown in Figs. 4.2.3(5) and 4.2.3(7)–4.2.3(10) have bolted bobbin-type stationary heads, but this is not an essential requirement and other head types may be used depending on service conditions. In the case of fixed tubesheet exchangers, the head at the rear end need not match that at the stationary end.

The various head types and their applications are described below, the type letter relating to the particular designation system referred to in Sec. 4.2.2B. The first letter refers to the stationary end for all exchanger types and the second letter to the rear end of fixed tubesheet exchangers only.

The channel-type head is commonly known as a bobbin in the United Kingdom and some other countries.

B. Bolted channel (type A or L)

It will be seen that the bolted channel-type stationary head comprises a cylindrical shell or barrel, flanged at both ends, with one flange being bolted to the shell flange or stationary tubesheet and the other flange bolted to a flat cover. Removal of the flat cover provides access to the tube ends without having to break the flanges connecting the piping to the nozzles. If necessary, the complete head may be removed for unrestricted access to the tube ends, or to withdraw the complete bundle. This type of header is desirable when cleaning of the inside of the tubes in situ is expected to be frequent.

The bolted channel-type head may also be fitted at the rear end of the fixed tubesheet type, if required, but this is usually worthwhile only if there is a nozzle in the rear end barrel.

C. Welded channel (type C or N)

For high-pressure and/or lethal service where it is desirable to minimize the number of external joints, the cylindrical barrel is welded, instead of bolted, to the stationary tubesheet. This eliminates one external joint, but access to the tube ends for cleaning or repair must always be carried out inside the head barrel. It is essential to ensure that there is adequate space between peripheral tubes and the inside of the barrel to accommodate the equipment required for tube end attachment, repair, and tube withdrawal.

D. Bonnet (type B or M)

A typical bonnet head is shown at the rear end of the fixed tubesheet exchanger of Fig. 4.2.3(1). Bonnet-type heads provide a cheaper alternative to the bolted or welded channel types and also offer unrestricted access to the tube ends after removal. However, it is necessary to break the flanges connecting the piping to the nozzles prior to removal, and this type of header is generally used at the stationary end when tube cleaning is expected to be infrequent.

Bonnet heads are invariably fitted at the rear end of the fixed tubesheet type if there are no nozzles in the rear end barrel.

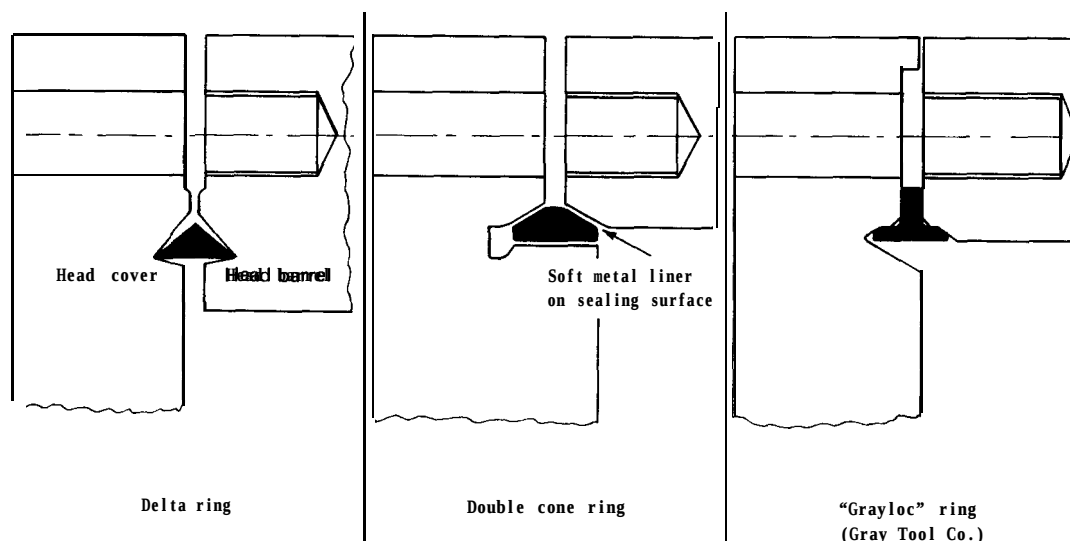


Figure 1 Self-sealing gaskets.

E. Bolted cone

When there is one tube-side pass, an axial-bolted cone-shaped head may be used instead of bolted bobbins or bonnets.

F. High pressure

The A-, C-, L-, and N-type full-diameter closures of conventional design described above are satisfactory over a wide range of diameter/pressure combinations. There are no mandatory rules, but an upper limit of (diameter) X (pressure) of 15 000 (mm X MN/m^2) is a rough guide. However, as head diameter and pressure increase, it becomes increasingly difficult to design a practical, economic closure of conventional type which provides adequate strength, leak-tightness, and service-

ability without huge bolts. For instance, a conventional closure, 600 mm in diameter, is satisfactory for a pressure of 25 MN/m^2 , but even at this modest head size the bolts will have a diameter of about 80 mm, thus necessitating special bolt-tensioning equipment for best results.

When the conventional closure is not practicable, one approach is to replace the conventional gasket with a special type that utilizes the contained pressure itself to force it to follow the dilation of the seating surface. The principle is shown in Fig. 1. Although this design minimizes the risk of leakage, the bolting must be large enough to withstand the hydrostatic end load. When the bolting becomes too large, the nonbolted closure (TEMA type D) is used where the hydrostatic end load is taken by pins or rings. There are numerous designs of this type, and the split-ring design shown in Fig. 2 illustrates the principle only.

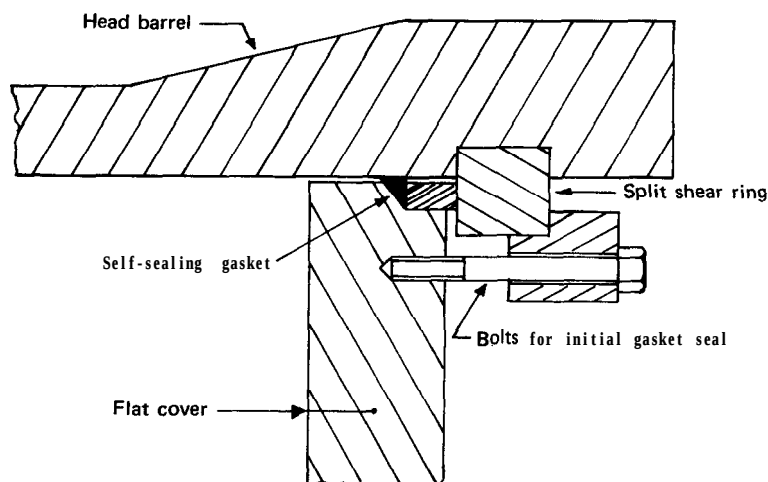


Figure 2 Shear ring closure.

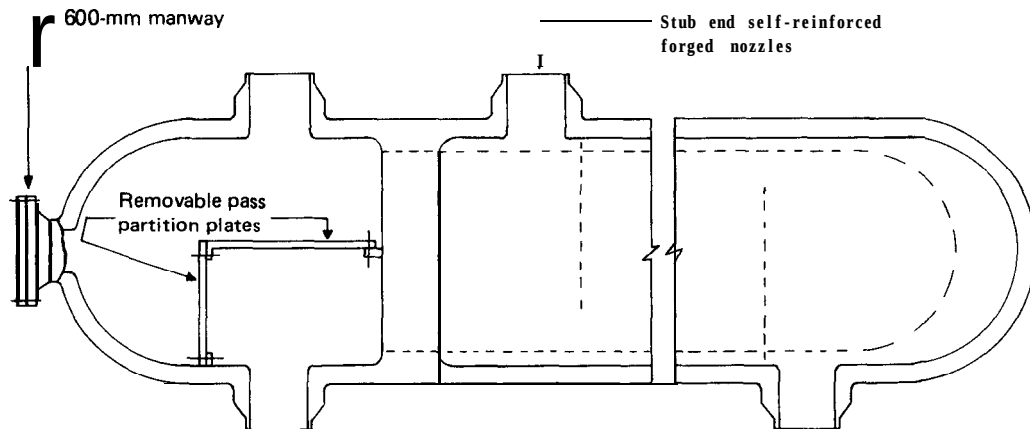


Figure 3 All-welded exchanger.

Large-diameter, high-pressure closures are expensive, and sometimes the all-welded construction shown in Fig. 3 is used instead. All girth and nozzle flanges are

eliminated, and the stub-end nozzles are welded directly to the piping at site. Access to the heads and tubesheets is via **manways**, usually 600 mm in inside diameter.

4.2.5

Features relating to thermal design

E. A. D. Saunders

A .

(a) Diameter and thickness

Small-diameter, closely packed tubes provide compact units, but the tubes may be difficult to clean both inside and outside; large-diameter, widely spaced tubes provide less compact units, but cleaning problems are reduced. The usual outside tube diameter range for petroleum and petrochemical applications is 15 to 32 mm, with 19 and 25 mm being the most common.

Tube thickness must be checked against internal and external pressure, but the diameters and thicknesses of tubes most commonly used can withstand appreciable pressure. For instance, a steel tube 19 mm in outside diameter, 2.1 mm thick, can withstand internal and external pressures of 16 and 7 MN/m², respectively. Hence, in many applications tube thicknesses are selected on the basis of cost, corrosion resistance, and standardization rather than pressure. Typical thicknesses are as follows:

OD	Carbon and low-alloy steels	Stainless steels and copper alloys
15.88 mm ($\frac{5}{8}$ in)	1.65 mm (16 BWG)	1.24 mm (18 BWG)
19.05 mm ($\frac{3}{4}$ in)	2.11 mm (14 BWG)	1.65 mm (16 BWG)
25.40 mm (1 in)	2.77 mm (12 BWG)	2.11 mm (14 BWG)
31.75 mm (1 $\frac{1}{4}$ in)	3.40 mm (10 BWG)	2.77 mm (12 BWG)

Tubes are usually purchased to specifications relating to outside diameter and nominal thickness, rather than nominal bore, because the permitted tolerances for

the latter are greater. Tubes may be purchased to minimum wall or average wall thicknesses. In the case of minimum wall tubes, the wall thickness must not be less than the nominal thickness, but may exceed it up to a certain maximum (typically 18 to 22% greater than nominal thickness). In the case of average wall tubes, the wall thickness may be greater or smaller than the nominal thickness within a certain range (typically ± 8 to 10% of the nominal wall thickness). The bore of minimum wall tubes may be less than the nominal bore and should be allowed for in thermal design, particularly tube-side pressure-loss calculations.

U tubes need special attention because thinning of the wall occurs in the smaller radius bends during bending. If thinning of the wall cannot be tolerated because of pressure and/or corrosion requirements, it is usual to bend the inner two rows from straight tubes having a greater thickness than the remainder (typically 25 to 30% thicker). Where pressure and corrosion requirements are not governing, and the tubes are not susceptible to work hardening, some thinning in the bend may be tolerated. TEMA, for instance, specifies that such thinning should not exceed 17% of the original tube wall thickness. TEMA also limits flattening at the bend to 10% of the tube nominal outside diameter.

(b) Length

For a given surface area the cheapest exchanger is obtained by making it as small in diameter and as long as possible, consistent with the space and handling facilities available at site and in the fabricator's shop. On this basis there is an incentive to make exchangers as long as

possible, limited only by the tube length available from tube suppliers, but practical aspects usually govern. Irrespective of the exchanger type, a clear length of at least twice the tube length must be available for siting the exchanger to permit either tube bundle or tube withdrawal. Long bundles may present difficulties in handling, particularly when withdrawing from, or inserting into, the shell. The most common tube length range is 3 600 to 9 000 mm for removal bundles, with a bundle weight limitation of 20 tons and 3 600 to 15 000 mm for the fixed tubesheet type. Exchangers up to 22 000 mm long have been made to meet the demands of increased process plant capacities.

(c) Pitch

Except for small tubes and shell diameters, TEMA specifies that the tube pitch/outside diameter ratio should not be less than 1.25 and that for external fouling services there should be a minimum gap of 6.35 mm ($\frac{1}{4}$ in) between adjacent tubes to assist external cleaning by mechanical means. Where cleaning lanes are provided, the tubesheet layout should be arranged such that the lanes are continuous throughout the bundle, or if this is not practical, every tube may be reached from one side of the bundle or the other. Typical tube pitches are as follows:

Tube OD	Minimum pitch	
	Clean	Fouling
15.88 mm ($\frac{5}{8}$ in)	19.84 mm ($\frac{35}{32}$ in)	22.22 mm ($\frac{7}{8}$ in)
19.05 mm ($\frac{3}{4}$ in)	23.81 mm ($\frac{15}{16}$ in)	25.40 mm (1 in)
25.40 mm (1 in)	31.75 mm ($1\frac{1}{4}$ in)	31.75 mm ($1\frac{1}{4}$ in)
31.75 mm ($1\frac{1}{4}$ in)	39.69 mm ($1\frac{5}{8}$ in)	39.69 mm ($1\frac{5}{8}$ in)

Figure 1 shows four basic tube pitch orientations. For a given tube pitch/outside diameter ratio, about 1.5% more tubes can be accommodated within a given shell diameter using 30 or 60 pitch compared with 45 or 90 pitch. In addition, heat transfer considerations must be taken into account when selecting tube pitch angle. For single-phase fluids outside the tubes, typical pitch angles are as follows:

Turbulent flow		Laminar flow	
Clean	Fouling	Clean	Fouling
30°	90°	30	45

(d) Bimetal

When it is difficult or expensive to obtain a single metal that will resist the corrosive action of both fluids, bimetal tubes are used. Almost any pair of metals may be drawn together to form a bimetal tube. When the combined wall thickness is not great, the thickness of each tube is made about the same; but as the combined wall thickness increases, it is usual to limit the thickness of the more expensive metal as far as practicable in order to reduce cost.

In most cases the tubesheets will be clad on the tube-side face with the same metal as the inner tube. In order to prevent galvanic action at the tube ends arising from different metals in contact with the tube-side fluid, it is usual to remove about 20 mm of the outer tube at each end and replace it with a ferrule of the same metal as the inner tube, as will be shown in Fig. 4.2.6(18).

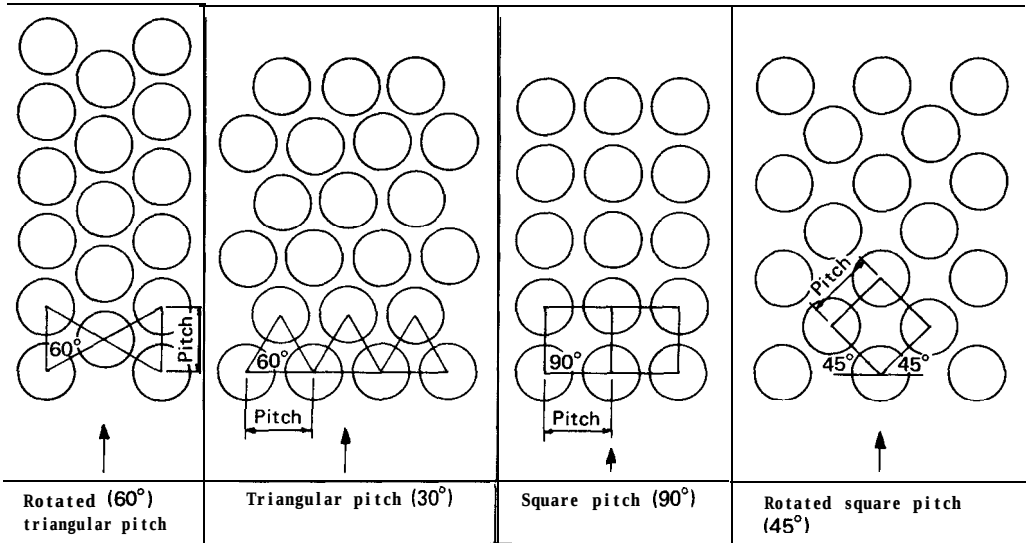


Figure 1 Tube pitch arrangements.



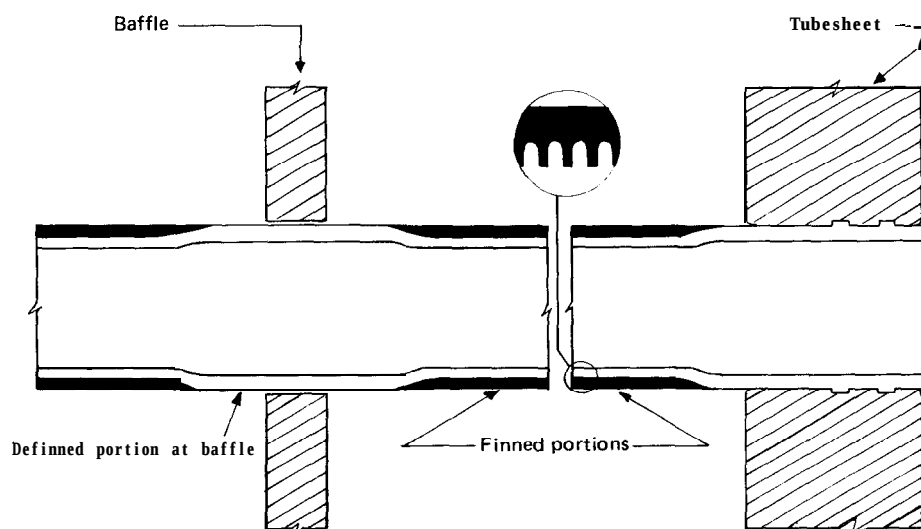


Figure 2 Integrally finned tube.

(e) Integrally finned

Low-finned tubes (19 fins per 25 mm length, 1.6 mm high) and medium-finned tubes (11 fins per 25 mm length, 3.2 mm high) are designed specifically to fit into shell-and-tube heat exchangers in the same manner as plain tubes. The external surface area is respectively 2.5 and 3 times that of a plain tube of the same outside diameter. These tubes may provide a cheaper design than a plain tube one, particularly when an expensive tube material is involved and the external heat transfer coefficient is low in relation to the internal coefficient. Performance of a plain tube exchanger may be boosted by substituting integrally finned tubes.

The special characteristics of low- and medium-finned tubes, as shown in Fig. 2, should be considered in design as follows:

1. The wall thickness under the fins is less than that at the plain end.
2. The bore of the tube in the finned portion is less than that at the plain end. The direct replacement of plain tubes by integrally finned ones may double tube-side pressure loss.
3. The outside diameter of the fins is slightly less than that of the plain end. Unless **defined**, there will be a greater baffle/tube clearance than with plain tubes. **Defining** at each baffle is desirable if vibration problems exist, but the loss of finned surface must be taken into account in thermal design.

B. Tube-side passes

(a) Partition plates

Each traverse of the tube-side fluid from one end of the exchanger to the other is termed a pass. By changing the

number of tube-side passes, the thermal designer is able to change the fluid velocity inside the tubes, and depending on the process conditions, only one tube-side pass, or as many as 16, may be required, although 8 passes is a more common upper limit. The heads are fitted with flat metal plates, known as partition plates, which divide the head into separate compartments.

The thickness of pass partition plates depends on head diameter but is usually 9 to 16 mm for carbon and low-alloy steels and 6 to 13 mm for the more expensive alloys.

Except for D-type and other special high-pressure heads, the partition plates are always welded to the head barrel and also to the adjacent tubesheet or cover, as appropriate, if either is welded to the barrel. If the tubesheet or cover is not welded to the head, the tubesheet or cover is grooved and the edge of the partition plate sealed by a gasket embedded in the grooves.

Typical pass arrangements are shown in Fig. 3, which also shows the shape of the head gaskets and grooving required in the tubesheets and covers.

(b) Number of passes

A fixed tubesheet and outside-packed floating-head exchanger may have any practical number of tube-side passes—one, two, four, six, and eight passes are the most common, with three and five passes, although permissible, less often used. A U-tube exchanger must have an even number of passes, two and four passes being the most common, with six and eight passes, although permissible, less often used. The number of passes in the lantern-ring floating-head exchanger is limited to one or two, and in the latter case the tube-side nozzles must be at the stationary end. The number of passes in split-backing-ring and pull-through floating-head exchangers

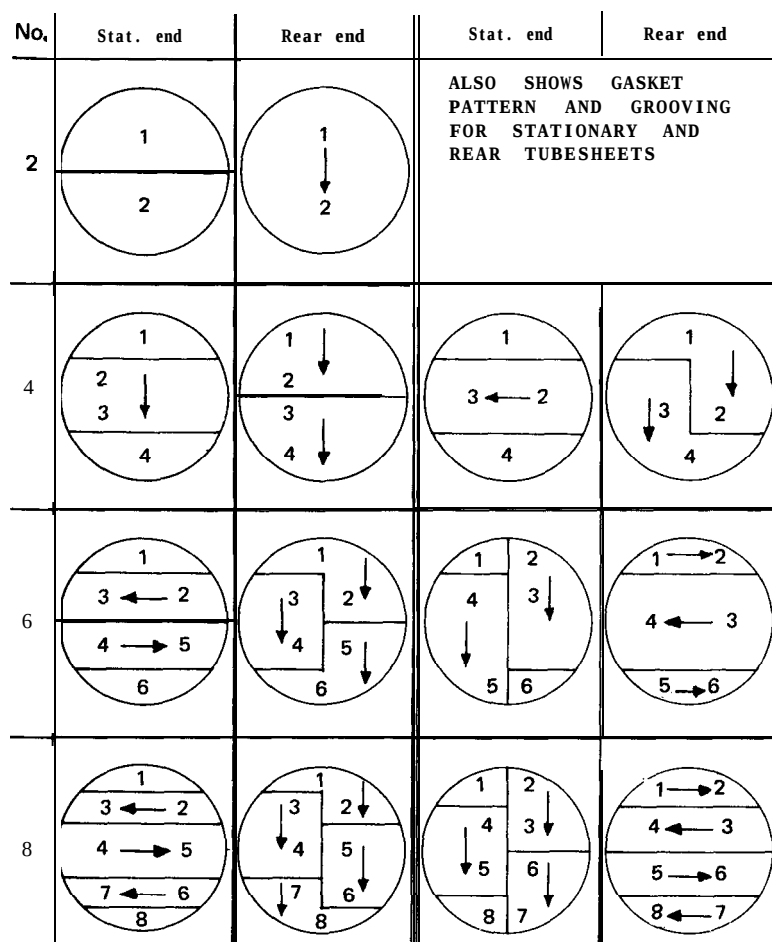


Figure 3 Typical tube-pass layouts.

is usually an even number, for example, two, four, six, or eight. A single-tube-pass floating-head exchanger may be used to provide the thermal advantage of countercurrent flow, but special construction is required as described in Sec. B(c).

(c) Single-tube-pass floating head

The special constructions required at the floating head when a single tube pass is required are shown in Figs. 4 and 5. The packed gland is the simplest, but its application is limited to low-pressure, nonlethal service on the shell side. When the packed gland construction is not permissible, the bellows-type construction must be used. There are various designs for bellows-type construction, and Fig. 5 is typical only. One example is shown in Fig. 6.

(d) "Boxed-in" passes

As mentioned in Sec. B(a), D-type and other high-pressure headers are special, and conventional partition plates are not used. It is necessary to box in the passes

as shown in Fig. 4.2.4(3). The removable pass covers, studs, and gaskets are of relatively light construction, as the cover joint only has to withstand the tube-side pressure loss and not the full tube-side design pressure.

C. Baffles

(a) Cross type

Cross baffles, shown in Fig. 7, have the dual purpose of supporting the tubes adequately at intervals to prevent sag and vibration, and also of enabling the shell-side fluid to flow back and forth across the bundle from one end of the exchanger to the other for heat transfer purposes. By changing the spacing of the baffles, the thermal designer is able to change the shell-side fluid velocity. Figure 8 shows three types of cross baffles used in exchanger design, namely, segmental, double segmental, and triple segmental. Segmental baffles are the most common type, but if the design is governed by shell-side pressure loss, it may be reduced appreciably by resorting to double or triple segmental baffles.

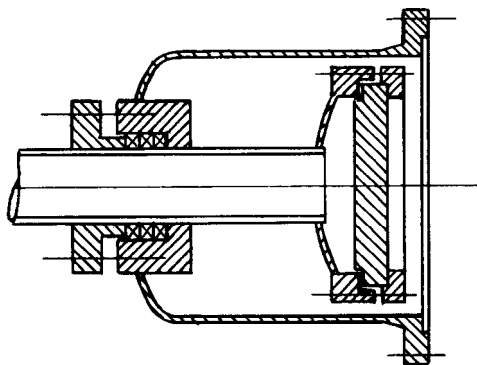


Figure 4 Single-pass floating head with gland.

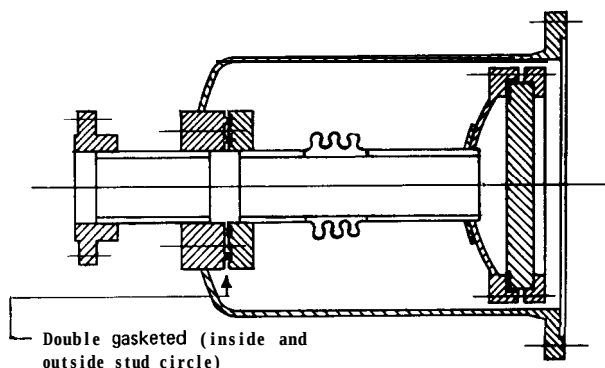


Figure 5 Single-pass floating head with bellows.

The opening in the baffles through which the shell-side fluid flows from one baffle space to the next is termed the baffle window, and the amount cut away is termed the baffle cut. Baffle cut is usually defined as the window opening segment height expressed as a per-

centage of the shell diameter. The usual range of baffle cuts is 15 to 40% for segmental and 20 to 30% for double segmental baffles. There must always be some overlap of the edges of adjacent baffles.

In a horizontal exchanger the baffle edges at the windows may be in the vertical position (vertical-cut baffles) or in the horizontal position (horizontal-cut baffles). The latter is suitable for clean single-phase fluids, but for fouling, condensing, or boiling fluids vertical-cut baffles are used. This prevents dirt settling out in the bottom of the shell for fouling fluids, whereas in the case of condensers it enables condensate to flow freely.

Tubes in the baffle window zones of segmental and double segmental baffles pass through every other baffle and are unsupported for a length of two baffle spaces. Figure 8 also shows a design using segmental baffles in which there are no tubes in the baffle windows and therefore all tubes pass through all baffles. Additional supports, through which all tubes pass, may be installed between the main segmental baffles as required, thus providing extra tube support and an increase in the natural frequency of the tubes, without affecting heat transfer and pressure loss. This design is termed no tubes in window and is used chiefly to eliminate shell-side flow-induced tube vibration. Compared with conventional designs, no-tubes-in-window designs have larger shell diameters to contain the same number of tubes.

(b) Support type

In cases where the performance of the exchanger is not dependent on the shell-side velocity, the cross-type

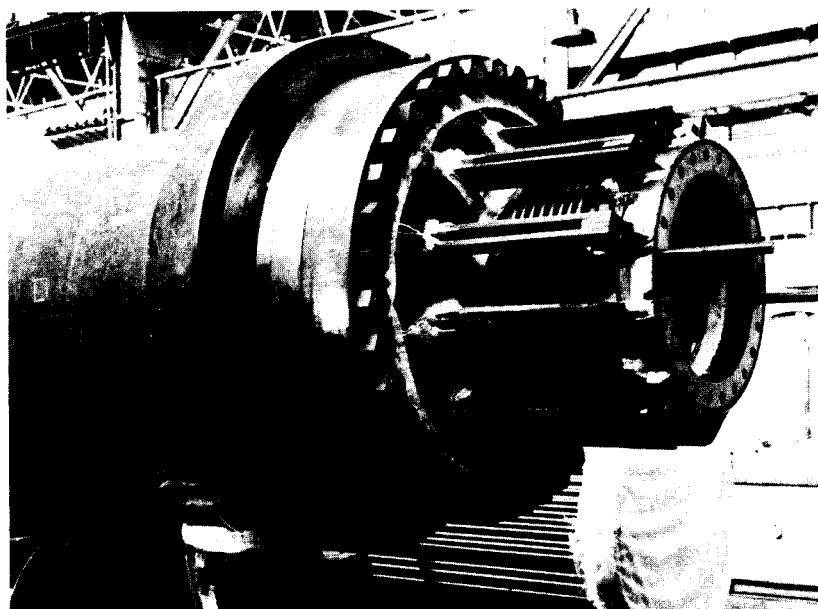


Figure 6 Split-backing-ring floating-head exchanger, with bellows at floating head, showing bellows restraining bars. Courtesy of Whessoe Heavy Engineering Ltd., Darlington, England.

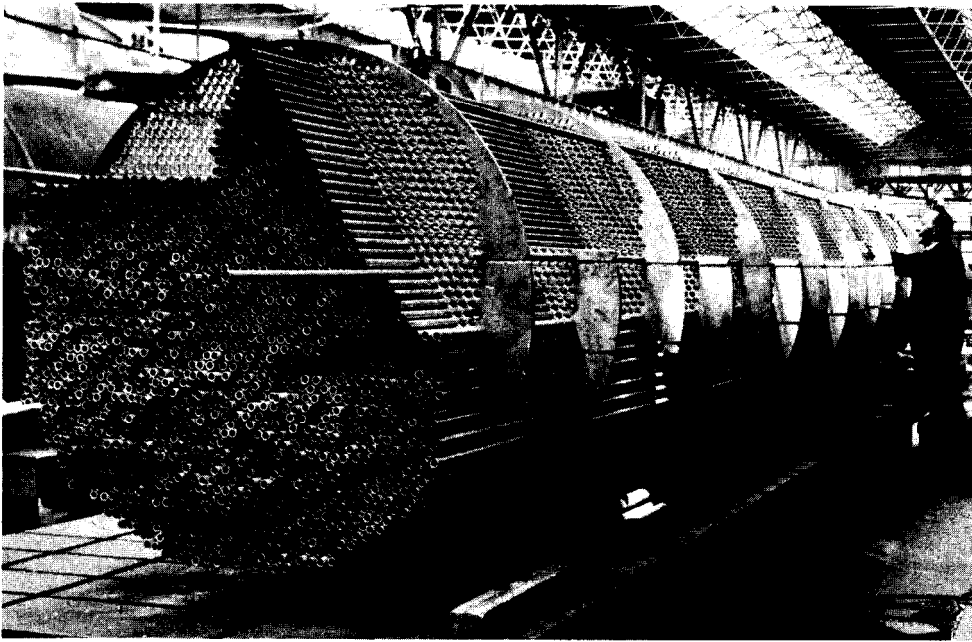


Figure 7 Tubing up to 1 219-mm diameter, 9 144-mm tube length, fixed tubesheet exchanger. Segmental baffles slotted for sealing strips. Courtesy of Whessoe Heavy Engineering Ltd., Darlington, England.

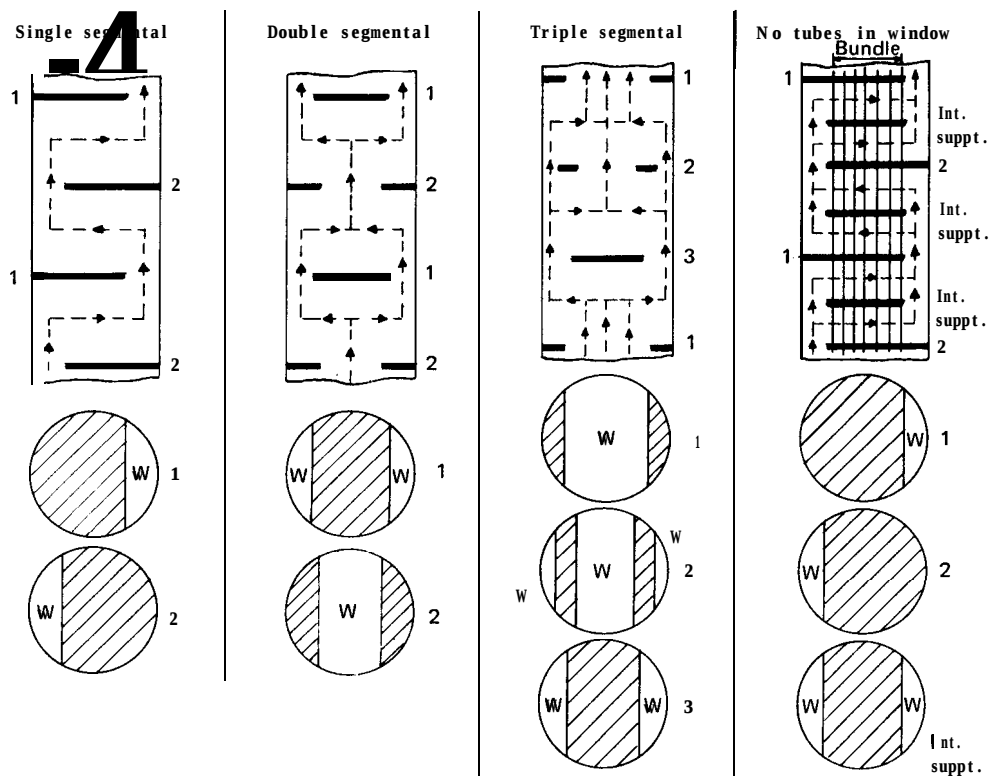


Figure 8 Baffle types. (W denotes baffle window.)

baffle need not be used and certain designs use circular baffles that are not cut at all. These are termed full-circle supports.

Split-backing-ring and pull-through floating-head exchangers have a special support-type baffle adjacent to the floating head to take the weight of the complete floating-head assembly. A full-circle support could be used, but the surface between the baffle and the floating tubesheet would become ineffective. In order to increase the effectiveness of this surface it is usual to cut slots of various shapes in the baffle.

(c) Thickness

TEMA specifies minimum baffle thickness, which is related to diameter and unsupported tube length. Typical examples are as follows:

Shell ID, mm	Baffle thickness, mm, for unsupported tube length, mm	
	Less than 610	Greater than 1524
Less than 355	3.2	9.5
Greater than 1550	9.5	19.1

(d) Maximum pitch

TEMA specifies the following maximum unsupported tube length (MUTL) for carbon, low-alloy, stainless steel, and nickel alloys:

Tube OD	MUTL
15.88 mm ($\frac{5}{8}$ in)	1321 mm (52 in)
19.05 mm ($\frac{3}{4}$ in)	1524 mm (60 in)
25.40 mm (1 in)	1880 mm (74 in)
31.75 mm ($1\frac{1}{4}$ in)	2235 mm (88 in)

For aluminum and alloys, copper and alloys, and titanium the MUTL is reduced to 0.865 of the above values.

In the case of segmental and double segmental baffles, with 19.05 mm OD tubes, for example, adjacent baffles must not be spaced more than 762 mm apart; if triple segmental baffles are used adjacent baffles must not be spaced more than 508 mm apart. The maximum permissible tube length for a completely unbaffled exchanger, having 19.05 mm OD tubes, for example, would be 1524 mm.

Sometimes, in order to accommodate the required nozzle, with possibly a reinforcing pad, the end baffle pitch must be made greater than that in the remainder of the exchanger. In this case the longest unsupported tube length is that which exists in the baffle window

nearest the tubesheets. It is important to check that the distance between the tubesheet and the second baffle from it, at each end, does not exceed the TEMA maximum and that the long tube span does not contribute to flow-induced vibration problems.

(e) Clearances

For construction purposes there must always be a clearance between the tubes and the holes in the baffles, and also between the outside of the baffles and the inside of the shell. Sometimes for heat transfer or antivibration purposes smaller clearances than those shown below are required. This is permissible, but as the clearances are reduced so fabrication costs increase.

If the maximum unsupported tube length is 914 mm (36 in) or less, or the tube outside diameter is greater than 31.75 mm ($1\frac{1}{4}$ in), TEMA specifies that the baffle hole should be drilled 0.794 mm ($\frac{1}{32}$ in) greater than the tube outside diameter. This diametral clearance is halved if the maximum unsupported length is greater than 914 mm (36 in) for tubes 31.75 mm ($1\frac{1}{4}$ in) and smaller.

The diametral clearance between shell inside diameter and baffle diameter depends on the shell diameter, typical values specified by TEMA being 4.45 mm (0.175 in) for diameters between 610 and 991 mm (24 and 39 in) and 11.12 mm (0.438 in) for diameters between 2159 and 2540 mm (85 and 100 in). Where baffle/shell clearance has little effect on shell-side heat transfer coefficient or temperature difference, up to twice the specified clearance may be used (see also Sec. 4.2.6A).

(f) Impingement plates

To protect the tubes below the shell-side inlet nozzle from damage resulting from solid particles or liquid droplets entrained in the shell-side fluid, an impingement plate may be required. TEMA specifies impingement protection when ρV^2 exceeds the following values:

Fluid	ρV^2 ^a
Noncorrosive, nonabrasive single phase	2230 (1500)
Other liquids, including liquids at boiling point	744 (500)
Corrosive vapors and gases, liquid-vapor mixtures	d(0)

^a ρ , kg/m³ (lb/ft³); V, m/s (ft/s).

The impingement plate is usually about 6 mm thick, flat or curved, with dimensions slightly greater than the nozzle bore *d*. If fitted inside the shell, it is usual to position the plate edge at a distance of about *d/4* below

the intersection of the nozzle wall and shell to provide an annular escape area roughly equal to that of the nozzle area. Internally fitted impingement plates mean that the shell cannot be completely filled with tubes and an increasing number of tubes have to be omitted as nozzle/shell diameter ratio increases.

At the outlet nozzle it is usual to position the closest row of tubes at a distance of about $d/6$ below the nozzle in order to provide adequate escape area.

In order to provide as many tubes as possible within a given shell, the impingement plate is sometimes fitted outside the shell in an enlarged nozzle, either conical or domed.

When it is not practical to fit the impingement plate inside or outside the shell, a vapor belt is used as shown in Fig. 9. A vapor belt has the added advantage that any nozzle **sinkage**, as described in Sec. 4.2.6A, occurs in the vapor belt barrel where it cannot influence shell/baffle clearances.

(g) Longitudinal type

Although most exchangers have a single shell-side pass, two shell passes can be provided by installing a flat metal plate, about 6 to 13 mm thick, running axially along the shell at the diameter to divide the tube bundle into two semicircular portions. The plate is in contact with the stationary tubesheet, but does not extend the full tube

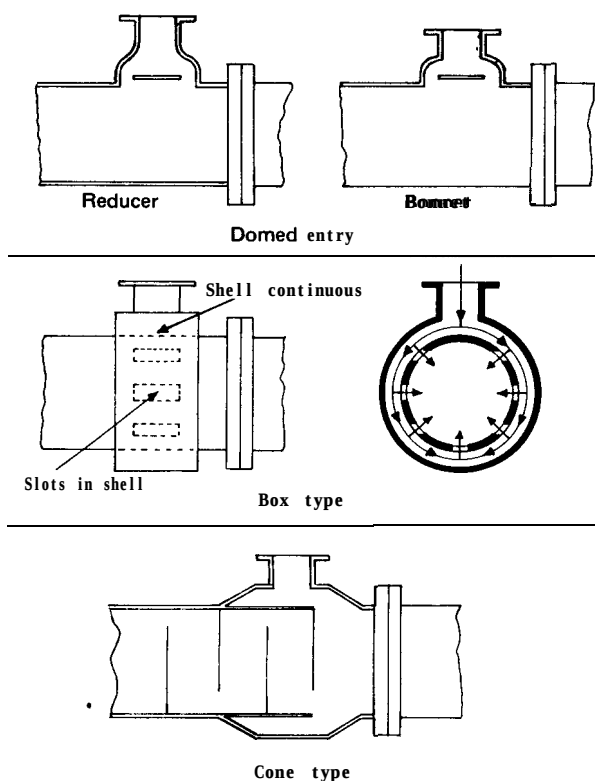


Figure 9 Vapor belts.

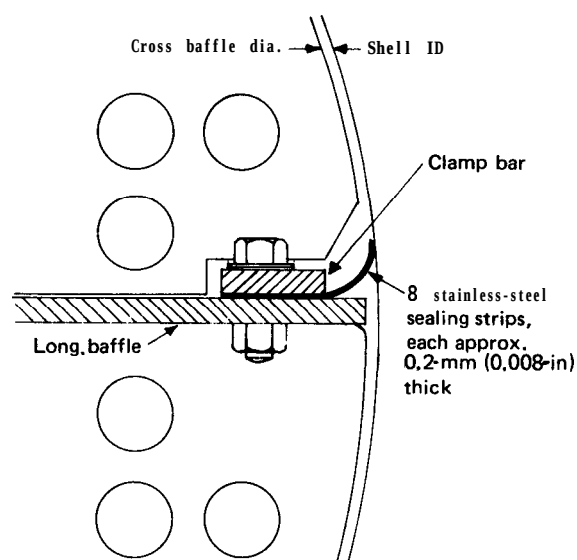


Figure 10 Longitudinal baffle.

length because space is required at the rear end to enable the shell-side fluid to cross over the bundle from the first pass into the second pass. The plate is termed a longitudinal baffle, and an exchanger having two shell-side and two tube-side passes provides the thermal advantage of countercurrent flow. The TEMA shell-type designation is F.

The longitudinal baffle is usually welded to the stationary tubesheet, but means must be provided to prevent leakage of the shell-side fluid from one pass to the other between the longitudinal baffle edges and the shell. The worst leakage point is adjacent to the stationary tubesheet where the inlet and outlet nozzles are diametrically opposed and the pressure (and temperature) differential across the baffle is greatest. In the case of fixed tubesheet exchangers, it is usually possible to weld the longitudinal baffle to the shell, but this is not possible for U-tube and floating-head exchangers because the longitudinal baffle must be removed with the bundle. When removable bundles are involved, the gap between longitudinal baffle and shell is sealed by flexible strips or packing devices, a typical flexible strip design being shown in Fig. 10. The flexible strips are placed on the shell inlet side of the longitudinal baffle so that the higher pressure assists the strips in maintaining contact with the shell.

When flexible strips or packing devices are used, it is usual to design the exchanger such that the pressure difference across the longitudinal baffle is not too great (typically 0.07 MN/m^2).

(h) Tie rods, spacers, and sealing devices

The tube bundle is held together and the baffles located in their correct positions by a number of tie rods and

spacers. The tie rods are circular metal rods screwed into the stationary tubesheet and extending the length of the bundle up to the last baffle, where they are secured by lock nuts. Between every baffle all tie rods have spacers fitted over them, each spacer being tube or pipe, having an inside diameter slightly greater than the tie rod diameter, and a length equal to the required baffle spacing.

The tie rods and spacers may also be used as sealing devices to assist in blocking off undesirable shell-side leakage paths due to pass partition lanes or gaps between the bundle and shell. Dummy tubes (which do not pass through the tubesheets), round oars, and flat longitudinal strips, inserted into slots in the periphery of the baffles, may also be used as sealing devices. Depending on nozzle/shell diameter ratio, the largest bundle/shell gap is often that created by an internally fitted impingement plate. If this creates a leakage path around the bundle, it must be sealed with longitudinal strips or other devices, but these must not be fitted in the inlet baffle space because this would reduce the escape area around the impingement plate considerably. The outlet baffle space must be examined on a similar basis.

Typical details are shown in Fig. 11.

D. Outer tube limit

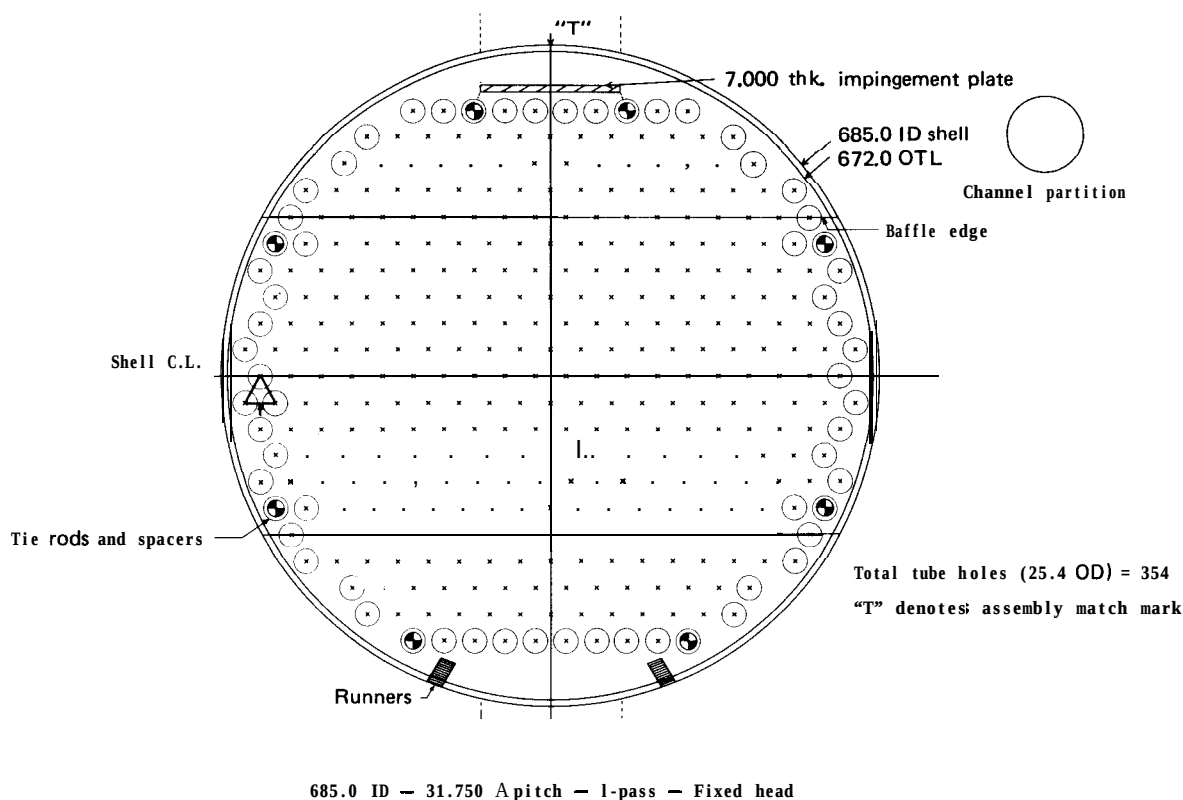
The outer tube limit (OTL) is the diameter of the largest circle, drawn about the tubesheet center, beyond which no tube may encroach. For a given shell inside diameter, the OTL for the split-backing-ring and pull-through floating-head types is dependent on pressure and whether the tubes are expanded or welded to the tubesheets. Welded tube ends provide a slightly smaller OTL than expanded tube ends. The OTL for fixed tubesheet and U-tube exchangers is independent of both pressure and tube end attachment for a given shell inside diameter.

In addition to the above factors, each fabricator has individualized clearances and construction details, so that the OTL data listed in Table 1 are typical only.

E. Tube count

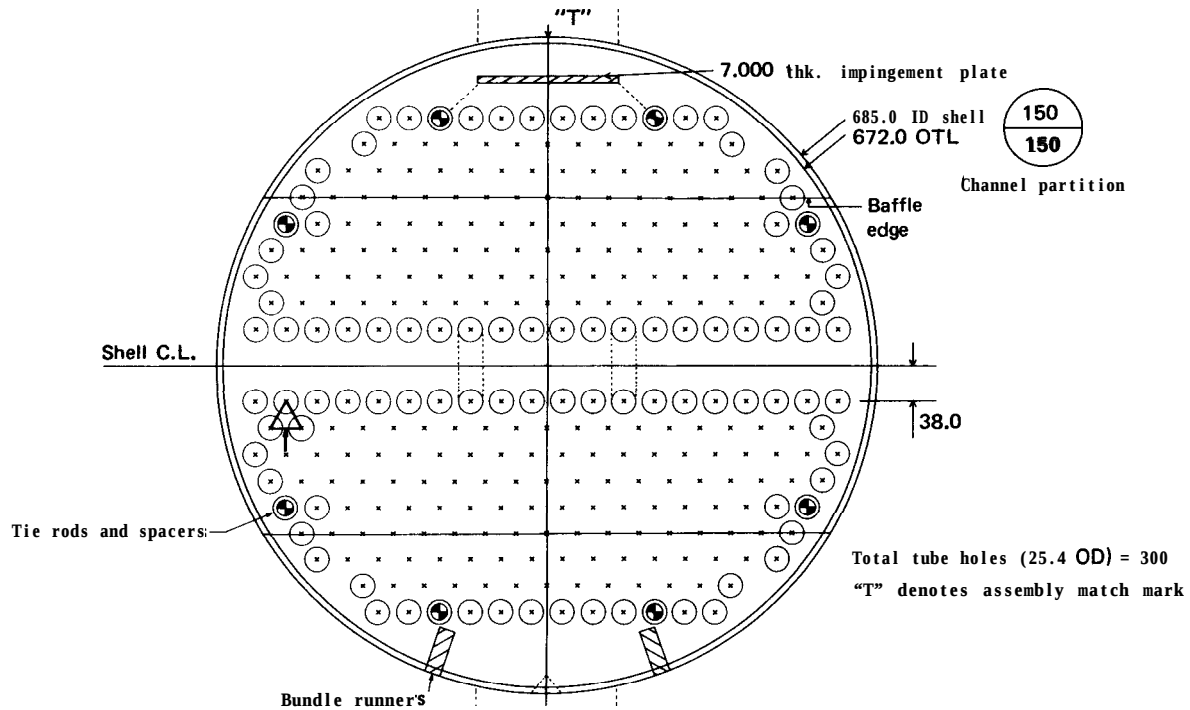
The number of tubes that can be accommodated within a given shell inside diameter, for a given tube outside diameter and pitch, is governed by the following factors:

1. The type of exchanger. This determines the OTL as described in Sec. D.



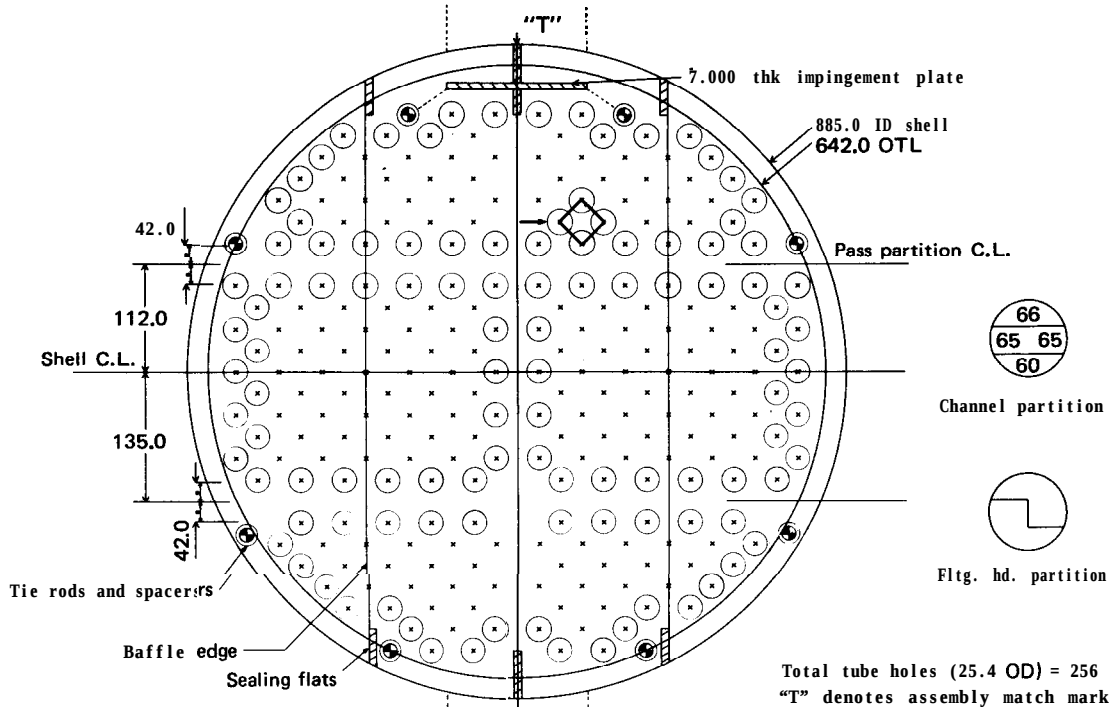
(a)

Figure 11 Typical tube layouts.



685.0 ID — 31.750 A pitch — 2-pass — U-tube bundle

(b)



685.0 ID — 31.750 A pitch — 4-pass — Floating head — Type S

(c)

Figure 11 Typical tube layouts. (Continued)

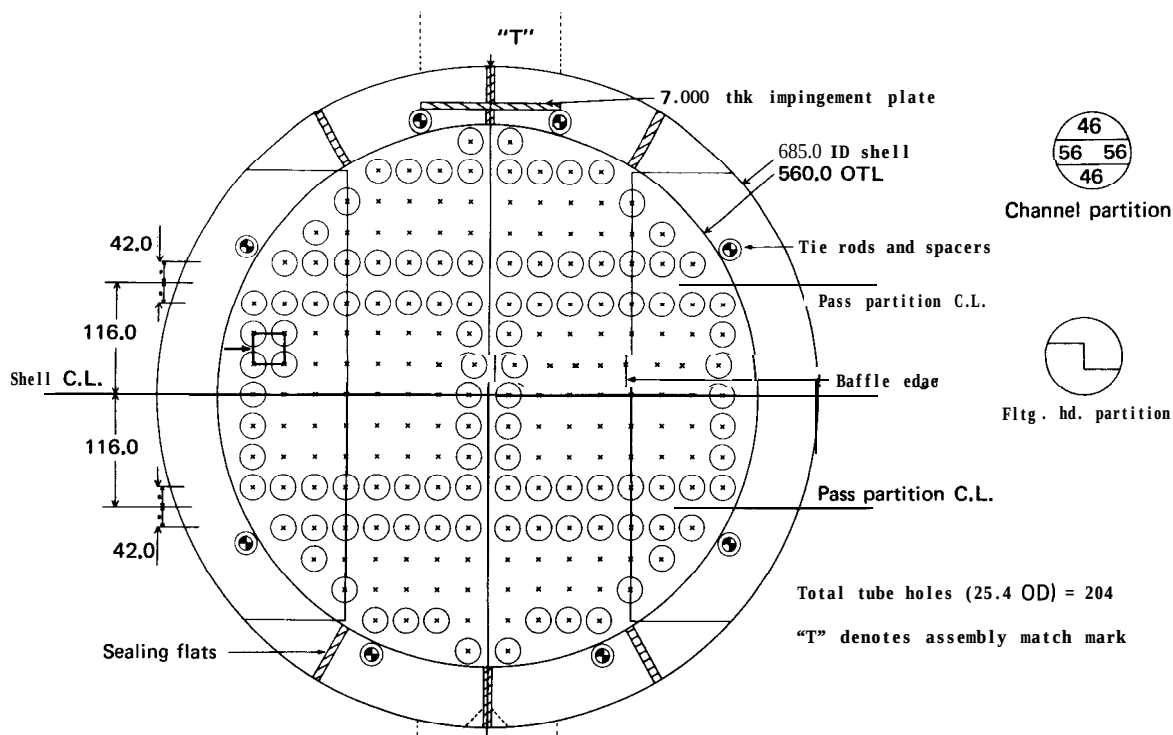


Figure 11 Typical tube layouts. (Continued)

2. The number of tube-side passes. As the number of passes increases, so does the area occupied by pass partitions that cannot be occupied by tubes.

3. The nozzle diameter. If an impingement plate is fitted internally, adequate escape area must be allowed as described in Sec. C(f). The greater the nozzle diameter, the smaller the number of tubes that can be accommodated. The tube count is independent of the nozzle diameter if an external impingement plate or vapor belt is fitted.

4. Tie rods, spacers, sealing devices. Space must be allowed to accommodate these items.

5. Tube end attachment. This may affect the OTL as described in Sec. D. In addition, a greater tube-to-tube center distance may be required at the pass partitions for welded tube ends, thus reducing the space available for tubes still further.

6. Pressure. This may affect the OTL for floating-head exchangers as described in Sec. D.

7. Radiused tubesheets. If the tubesheet has periph-

Table 1 Typical diametral clearance (D_i — OTL), mm^a

Exchanger ID, mm	Fixed tubesheet and U tube ^b	Split-backing-ring floating head ^c	Pull-through floating-head tube-side pressure, MN/m ²		
			1	2	3
up to 500	11	35 (3)	95	95	100
1 000	15	41 (3)	110	116	135
1 500	18	50 (2)	130	136	160
2 500	18	—	—	—	—

^aubes roller expanded. D_i = shell inside diameter.

^bNot affected by pressure for given shell inside diameter.

^cFigures in parentheses denote maximum tube-side pressure.

eral radiused hubs for welding to the shell or head barrels, the OTL may be reduced because tubes must not encroach into the radii.

8. Rotatable bundles. These are rotated through 180° to even out wear at the shell inlet. If a space is left for an internal impingement plate, a similar space must be left on the opposite side of the bundle, resulting in a loss of tubes.

Because of the large number of variables, it is not possible to provide exact tube count tables for all conditions. The tube counts provided in Table 2 are adequate for initial sizing purposes, but for exact counts a tube layout drawing must be made.

F. Shell-side nozzle arrangements

Figure 4.2.2(1) shows various shell-side nozzle arrangements, namely, E, F, G, H, J, K, and X. The E shell, with two nozzles, one at each end, represents the most common arrangement. The purposes of the F and K shells have been described in Secs. C(g) and 4.2.3H, respectively.

When shell-side pressure loss governs a design problem, the divided flow shell, type J, will reduce pressure loss considerably. There is a choice of two inlets/one outlet or one inlet/two outlets, and in order to minimize nozzle diameters it is usual to provide two inlets/one outlet for condensing services and one inlet/two outlets for vaporizing services.

In the cross-flow shell, type X, the shell-side fluid makes one cross-flow pass across the whole bundle, which has full-circle supports except for short units. Additional full-circle supports, through which all tubes pass, may be installed as required, thus providing extra tube support and an increase in the natural frequency of the tubes, without affecting heat transfer and pressure loss. In addition to the elimination of flow-induced vibration, the X shell may solve a design problem governed by shell-side pressure loss. The bundle usually has a flat top and bottom to provide adequate segmental-shaped entrance and exit flow areas. In order to minimize nozzle diameter, a typical design may require up to four nozzles at both the top and bottom of the shell, depending on length. A full-length perforated distributor plate may be required at the top of the bundle to aid shell-side flow distribution.

The G and H shells are used chiefly for certain condensing and boiling applications in which the shell-side heat transfer coefficient is independent of the shell-side velocity. Figure 4.2.2(1) shows that both contain longitudinal baffles, as described in Sec. C(g), but these are usually unsealed at the edges.

Although conventional cross-type baffles may be used, the G shell often has one full-circle support at midpoint on the common center line of the shell nozzles. The H shell is equivalent to two G shells joined end to end and a total of three full-circle supports is required, one in each G shell as before and one at the midpoint of the complete bundle. Inspection of Sec. C(d), dealing with maximum baffle pitch, shows that G and H shells with 19.05-mm OD steel tubes, for example, cannot be longer than 3 048 and 6 096 mm, respectively. By using full-circle support baffles, G and H shells provide longitudinal, rather than cross flow, of the shell-side fluid and the pressure loss is therefore lower.

To improve condensate subcooling, the longitudinal baffles are sometimes perforated to permit condensate formed in the upper half of the bundle to drip on the tubes in the lower half. G and H shells are often used as horizontal thermosyphon reboilers, in which case adequate space is provided at the top of the bundle to permit the vapor-liquid mixture to discharge freely.

G. Routing of fluids

Except for exchangers having specific functions, such as a kettle-type reboiler where the cold fluid must flow on the shell side, it is immaterial to the function of the exchanger whether the hot or cold fluid flows on the shell side. Where problems of fouling, high pressures, and special materials are absent, it is often worthwhile to investigate both possibilities to determine the cheaper exchanger.

When one fluid requires an expensive material for corrosion resistance or strength at high temperature, it is usual to route that fluid to the tube side, this being cheaper than having to make the whole of the shell side in the expensive material. When one fluid is at high pressure it is also usual to route that fluid to the tube side, this being cheaper than having to make a thick shell, with the added possibility that the tubes may have to be thickened to resist the high external pressure.

Guidance regarding the routing of the fluids is given below:

Route on tube side if:	Route on shell side if:
Fouling (except U tube)	Small flow rate
Corrosive	Viscous fluid
High pressure	Low heat transfer coefficient
High temperature	(Consider finned tubes)

Conflicting requirements often arise, however, and it is prudent to establish alternative designs, with the fluids switched, in order to determine the better one.

Table 2a Tube counts for fixed tubesheet exchangers

Shell ID, mm (in)	15.88-mm ($\frac{5}{8}$ -in) OD X 19.84-mm ($\frac{3}{4}$ -in) pitch A								19.05-mm (f-in) OD X 23.81-mm ($\frac{1}{2}$ -in) pitch A							
	1 pass		2 pass		4 pass		6 pass		1 pass		2 pass		4 pass		6 pass	
	0.0	0.33	0.0	0.33	0.0	0.33	0.0	0.33	0.0	0.33	0.0	0.33	0.0	0.33	0.0	0.33
203	77	73	66	62	49	45	39	35	52	50	45	43	32	30	26	24
(8)																
254	125	119	112	106	90	84	78	12	85	81	76	72	60	56	52	48
(10)																
305	186	175	170	159	143	132	129	118	127	120	116	109	98	91	88	81
(12)																
331	230	216	212	198	193	169	161	153	158	148	146	136	125	115	115	105
(13)																
387	310	291	290	271	256	231	238	219	213	200	199	186	175	162	163	150
(15)																
438	401	316	318	353	340	315	319	294	276	259	260	243	234	217	220	203
(17)																
489	505	412	480	447	431	404	414	381	348	325	330	307	301	218	285	262
(19)																
540	621	579	593	551	546	504	521	419	428	400	408	380	316	348	359	331
(21)																
591	148	698	111	661	666	616	638	588	516	481	495	460	459	424	440	405
(23)																
635	869	810	836	777	180	121	151	692	600	559	577	536	539	498	518	477
(25)																
686	1019	948	983	912	923	852	891	820	704	655	619	630	638	589	616	567
(27)																
737	1180	1098	1142	1060	1011	995	1043	961	816	759	189	132	145	688	121	664
(29)																
187	1354	1258	1313	1211	1244	1148	1201	1111	936	810	901	841	860	194	835	169
(31)																
838	1539	1430	1495	1386	1422	1313	1383	1274	1064	989	1034	959	983	908	956	881
(33)																
889	1731	1612	1691	1566	1613	1488	1512	1447	1201	1115	1169	1083	1115	1029	1087	101
(35)																
940	1946	1805	1896	1155	1815	1674	1111	1630	1346	1249	1312	1215	1255	1158	1226	1129
(37)																
991	2161	2009	2115	1957	2029	1871	1983	1825	1499	1391	1463	1355	1403	1295	1372	1264
(39)																
1061	2520	2336	2464	2280	2311	2181	2321	2137	1745	1617	1106	1578	1642	1514	1608	1480
(42)																
1143	2901	2688	2841	2628	2741	2528	2688	2415	2009	1861	1968	1820	1898	1750	1862	1714
(45)																
1219	3308	3063	3244	2999	3138	2893	3081	2836	2291	2121	2241	2017	2113	2003	2134	1964
(48)																
1295	3142	3464	3674	3396	3561	3283	3501	3223	2592	2399	2545	2352	2467	2214	2425	2232
(51)																
1372	4203	3889	4132	3818	4011	3691	3948	3634	2912	2694	2862	2644	2119	2561	2135	2511
(54)																
1448	4690	4338	4615	4263	4488	4136	4420	4068	3250	3006	3198	2954	3110	2866	3064	2820
(57)																
1524	5205	4813	5126	4134	4992	4600	4921	4529	3601	3335	3552	3280	3459	3181	3411	3139
(60)																
1600									3982	3681	3924	3623	3821	3526	3776	3475
(63)																
1616									4316	4044	4315	3983	4214	3882	4160	3828
(66)																
1753									4789	4424	4126	4361	4619	4254	4563	4198
(69)																
1829									5220	4821	5154	4155	5043	4644	4984	4585
(72)																
1981									6138	5668	6066	5596	5946	5476	5882	5412
(78)																
2134									7130	6581	7053	6504	6923	6314	6855	6306
(84)																
2286									8196	7563	8113	1480	7975	7342	1901	7268
(90)																
2438									9331	8614	9249	8526	9101	8378	9022	8299
(96)																
2591									10552	9132	10458	9638	10301	9481	10218	9398
(102)																
2143									11842	10920	11143	10821	11516	10654	11488	10566
(108)																
2896									13205	12176	13100	12071	12925	11896	12831	11802
(114)																
3048									14643	13500	14533	13390	14348	13205	14250	13101
(120)																

^aTube counts given for tube diameters, pitches, number of passes, and nozzle: shell diameter ratios shown. Internal impingement fitted except for 0.0, which refers to vapor belt or external impingement. Refer to Sec. E. Expanded ends.

Table 2a Tube counts for **fixed** tubesheet exchanged (Continued)

Shell ID, mm (in)	19.05-mm ($\frac{3}{4}$ -in) OD X 25.4-mm (1-in) pitch A								25.4-mm (1-in) OD X 31.75-mm (1 $\frac{1}{4}$ -in) pitch A							
	1 pass		2 pass		4 pass		6 pass		1 pass		2 pass		4 pass		6 pass	
	0.0	0.33	0.0	0.33	0.0	0.33	0.0	0.33	0.0	0.33	0.0	0.33	0.0	0.33	0.0	0.33
203 (8)	46	43	40	37	29	26	23	20	28	27	24	23	17	16	17	12
254 (10)	75	71	67	63	53	49	46	42	47	44	42	39	33	30	29	26
305 (12)	112	106	102	96	86	80	77	71	10	66	64	60	54	50	48	44
337 (13 $\frac{1}{2}$)	138	130	127	119	110	102	100	92	81	82	80	IS	69	64	63	58
387 (15 $\frac{1}{2}$)	187	176	175	164	154	143	143	132	115	110	107	102	94	89	87	82
438 (17 $\frac{1}{2}$)	243	221	229	213	206	190	193	177	153	143	144	134	129	119	121	111
489 (19 $\frac{1}{2}$)	306	286	290	210	264	244	250	230	193	180	183	170	167	154	158	145
540 (21 $\frac{1}{2}$)	376	351	359	334	330	305	315	290	238	220	227	209	209	191	199	181
591 (23 $\frac{1}{2}$)	454	423	435	404	404	373	387	356	287	267	275	252	255	235	244	224
641 (25 $\frac{1}{2}$)	527	491	507	471	473	437	455	419	334	311	321	298	300	217	288	265
686 (27)	619	516	597	554	561	518	541	498	392	365	378	351	355	328	342	315
737 (29)	717	667	694	644	654	604	633	583	455	423	440	408	415	383	402	370
787 (31)	823	765	798	740	756	698	734	676	522	485	506	469	419	442	465	428
838 (33)	935	869	908	842	864	198	840	774	594	551	577	534	549	506	533	490
889 (35)	1 056	980	1028	952	980	904	955	879	671	622	653	604	623	514	607	558
940 (37)	1 183	1 098	1153	1 068	1103	1018	1016	991	752	697	734	678	701	646	684	629
991 (39)	1 318	1 222	1 286	1 190	1 234	1 138	1 206	1110	838	176	818	756	784	722	766	704
1 067 (42)	1533	1421	1499	1 387	1442	1 330	1412	1 300	975	903	953	881	917	845	897	825
1 143 (45)	1 765	1 635	1 729	1599	1 668	1538	1 635	1505	1123	1040	1100	1017	1061	978	1040	951
1 219 (48)	2014	1 864	1915	1 825	1911	1 761	1815	1725	1282	1186	1251	1161	1216	1120	1193	1097
1 295 (51)	2278	2 109	2231	2068	2168	1 999	2131	1962	1451	1342	1424	1315	1381	1272	1357	1248
1 372 (54)	2559	2 368	2515	2324	2443	2252	2403	2212	1630	1508	1602	1480	1556	1434	1530	1408
1448 (57)	2856	2641	2810	2595	2133	2518	2692	2477	1820	1680	1790	1652	1741	1603	1715	1577
1524 (60)	3170	2 931	3121	2882	3040	2801	2991	2 758	2020	1867	1989	1836	1931	1784	1909	1756
1600 (63)	3500	3235	3449	3184	3 364	3099	3 318	3053	2231	2061	2198	2028	2144	1974	2115	1945
1616 (66)	3846	3554	3193	3501	3104	3412	3656	3364	2452	2265	2418	2231	2361	2114	2330	2143
1753 (69)	4 209	3 888	4153	3832	4060	3739	4 010	3689	2684	2479	2648	2443	2589	2384	2 557	2351
1829 (72)	4581	4237	4529	4 179	4432	4082	4319	4029	2926	2701	2889	2664	2827	2602	2793	2568
1981 (78)	5 394	4981	5 331	4918	5 226	4813	5 169	4756	3441	3177	3400	3136	3334	3070	3297	3033
2134 (84)	6266	5 783	6 198	5 715	6085	5602	6024	5541	3998	3689	3954	3645	3882	3513	3843	3534
2 286 (90)	7 203	6646	1130	6 573	7 009	6452	6943	6386	4598	4241	4551	4194	4414	4111	4432	4075
2438 (96)	8 206	7511	8128	7493	7999	1364	7929	7294	5238	4832	5188	4782	5106	4700	5061	4655
2591 (102)	9 214	8553	9191	8410	9 054	8 333	8980	8259	5921	5460	5868	5401	5781	5320	5132	5271
2143 (108)	10407	9597	10320	9510	10 174	9 364	10095	9285	6645	6127	6589	6 071	6 496	5 978	6 445	5 927
2896 (114)	11606	10101	11514	10609	11 360	10455	11271	10372	7412	6833	7 353	6174	7255	6676	1201	6622
3048 (120)	12869	11864	12172	11767	12610	11605	12522	11517	8220	7576	8158	7514	8 055	7411	7998	1354

^aTube counts given for tube diameters, pitches, number of passes, and nozzle: **shell** diameter ratios shown. Internal impingement fitted except for 0.0, which refers to vapor belt or external impingement. Refer to Sec. E. Expanded ends.

Table 2a Tube counts for fixed tubesheet exchangers^a
(Continued)

Shell ID, mm (in)	31.75-mm (1 $\frac{1}{4}$ -in) OD X 39.69-mm (1 $\frac{5}{16}$ -in) pitch A							
	1 pass		2 pass		4 pass		6 pass	
	0.0	0.33	0.0	0.33	0.0	0.33	0.0	0.33
203 (8)	11	16	14	13	10	9	—	—
254 (10)	29	27	26	24	20	18	17	15
305 (12)	43	41	39	37	33	31	29	27
337 (13 $\frac{1}{4}$)	54	51	50	41	43	40	39	36
381 (15)	74	69	69	64	61	56	56	51
438 (17 $\frac{1}{2}$)	96	90	90	84	81	75	76	70
489 (19 $\frac{1}{4}$)	122	113	116	101	105	96	100	91
540 (21 $\frac{1}{4}$)	150	140	143	133	132	122	125	115
591 (23 $\frac{1}{4}$)	181	169	113	161	161	149	154	142
646 (25 $\frac{1}{2}$)	211	196	203	188	189	114	182	167
686 (27)	248	231	239	222	224	201	216	199
731 (29)	288	268	278	258	263	243	254	234
787 (31)	331	301	321	291	304	280	295	271
838 (33)	377	350	366	339	348	321	338	311
889 (35)	426	395	414	383	395	364	385	354
940 (37)	478	443	466	431	445	410	435	400
991 (39)	532	493	519	480	498	459	486	441
1 067 (42)	620	574	606	560	583	537	571	525
1 143 (45)	715	661	100	646	675	621	662	608
1 219 (48)	816	755	800	739	714	713	760	699
1 295 (51)	924	854	907	831	819	809	864	794
1 312 (54)	1038	960	1020	942	990	912	915	891
1 448 (57)	1159	1071	1140	1052	1109	1021	1092	1004
1 524 (60)	1281	1189	1267	1169	1234	1136	1216	1118
1 600 (63)	1422	1313	1401	1292	1366	1257	1348	1239
1 676 (66)	1563	1443	1541	1421	1505	1385	1486	1366
1 753 (69)	1111	1580	1688	1551	1650	1519	1630	1499
1 829 (72)	1866	1122	1842	1698	1802	1658	1781	1637
1 981 (78)	2195	2026	2169	2000	2126	1951	2103	1934
2 134 (84)	2551	2353	2523	2325	2471	2219	2452	2254
2 286 (90)	2934	2706	2904	2676	2854	2626	2828	2600
2 438 (96)	3343	3083	3311	3051	3258	2998	3230	2950
2 591 (102)	3779	3484	3745	3450	3689	3394	3659	3364
2 743 (108)	4243	3911	4201	3875	4153	3815	4116	3184
2 896 (114)	4132	4362	4694	4324	4631	4261	4598	4228
3 048 (120)	5249	4837	5209	4797	5145	4733	5101	4695

^aTube counts given for tube diameters, pitches, number of passes, and nozzle: shell diameter ratios shown. Internal impingement fitted except for 0.0, which refers to vapor belt or external impingement. Refer to Sec. E. Expanded ends.

Table 2b Tube counts for U-tube exchanger

Shell ID, mm (in)	15.88-mm (Q-in) OD X 19.84-mm ($\frac{3}{4}$ -in) pitch A				15.88-mm (Q-in) OD X 22.23-mm ($\frac{7}{8}$ -in) pitch □				19.05mm (\$-in) OD X 23.81-mm ($1\frac{1}{8}$ -in) pitch A			
	2 pass		4 pass		2 pass		4 pass		2 pass		4 pass	
	0.0	0.33	0.0	0.33	0.0	0.33	0.0	0.33	0.0	0.33	0.0	0.33
203	58	54	46	42	40	36	32	30	36	34	28	26
(8)												
254	100	94	88	82	70	66	60	56	64	60	56	52
(10)												
305	158	146	140	130	108	102	96	90	104	96	92	84
(12)												
337	200	184	180	166	138	128	124	114	132	122	120	110
(13 $\frac{1}{4}$)												
387	274	254	252	234	190	176	174	162	182	170	168	156
(15 $\frac{1}{4}$)												
438	360	336	336	312	248	230	232	214	242	226	226	208
(17f)												
489	460	426	434	400	318	294	300	276	310	286	292	268
(19 $\frac{1}{4}$)												
540	570	528	542	500	394	366	374	344	386	358	366	338
(21 $\frac{1}{4}$)												
591	694	644	662	612	478	442	456	420	470	434	448	414
(23 $\frac{1}{4}$)												
635	810	750	776	716	560	518	536	494	550	510	528	486
(25)												
686	956	884	918	848	660	610	634	586	650	602	626	576
(27)												
737	1112	1030	1072	900	768	708	740	684	758	702	732	674
(29)												
787	1280	1184	1238	1142	884	818	854	790	874	808	846	780
(31)												
838	1460	1352	1416	1308	1008	932	978	902	1000	924	968	892
(33)												
889	1654	1530	1606	1482	1142	1056	1108	1022	1132	1046	1098	1012
(35)												
940	1858	1718	1808	1668	1282	1186	1248	1150	1272	1176	1238	1142
(37)												
991	2074	1916	2022	1864	1432	1324	1394	1286	1422	1314	1384	1278
(39)												
1067	2420	2236	2366	2180	1672	1544	1632	1504	1662	1534	1622	1494
(42)												
1143	2794	2582	2734	2522	1930	1782	1888	1740	1922	1774	1878	1730
(45)												
1219	3194	2952	3130	2884	2206	2036	2160	1992	2206	2026	2152	1982
(48)												
1295	3622	3344	3554	3276	2500	2308	2452	2260	2492	2300	2444	2250
(51)												
1372	4076	3762	4002	3688	2814	2596	2764	2546	2806	2588	2756	2538
(54)												
1448	4556	4204	4478	4126	3144	2902	3092	2848	3138	2894	3086	2842
(57)												
1524	5064	4672	4982	4590	3494	3224	3440	3170	3488	3216	3434	3162
(60)												

^aTube hole counts given for tube diameters, pitches, number of passes, and nozzle: shell diameter ratios shown. Internal impingement fitted except for 0.0, which refers to vapor belt or external impingement. Refer to Sec. E. Expanded ends.

Table 2b Tube counts for U-tube exchanger (Continued)

Shell ID, mm (in)	19.05-mm ($\frac{3}{4}$ -in) OD X 25.4-mm (1-in) pitch □				19.05-mm ($\frac{3}{4}$ -in) OD X 25.4-mm (1-in) pitch A				25.4-mm (1-in) OD X 31.75-mm ($1\frac{1}{4}$ -in) pitch □			
	2 pass		4 pass		2 pass		4 pass		2 pass		4 pass	
	0.0	0.33	0.0	0.33	0.0	0.33	0.0	0.33	0.0	0.33	0.0	0.33
203 (8)	28	26	22	20	32	28	26	22	14	12	10	8
254 (10)	50	48	44	40	58	54	50	46	28	24	22	20
305 (12)	78	74	70	64	90	84	80	74	46	42	38	36
337 (13 $\frac{1}{4}$)	100	94	90	84	114	106	104	96	58	54	52	48
387 (15-1)	140	130	128	118	160	150	148	136	82	76	74	68
438 (17 $\frac{1}{4}$)	184	170	172	160	212	196	198	182	110	102	102	94
489 (19 $\frac{1}{4}$)	236	220	222	206	272	252	256	236	142	132	134	122
540 (21 $\frac{1}{4}$)	294	272	280	258	338	314	322	296	178	164	170	156
591 (23 $\frac{1}{4}$)	358	330	342	314	414	382	394	364	218	200	208	190
635 (25)	420	388	402	370	484	448	462	426	256	236	246	226
686 (27)	496	458	476	440	572	530	550	506	304	282	292	270
737 (29)	578	534	558	514	666	616	642	592	356	328	344	316
787 (31)	666	616	644	594	770	712	744	686	412	380	398	366
838 (33)	760	702	738	680	878	812	850	784	472	434	456	420
889 (35)	862	796	838	772	996	920	966	890	536	494	520	478
940 (37)	970	896	944	870	1118	1034	1088	1004	604	558	586	540
991 (39)	1084	1000	1056	972	1250	1154	1218	1122	676	622	656	604
1067 (42)	1264	1168	1236	1140	1460	1348	1426	1314	792	730	770	708
1143 (45)	1462	1348	1422	1318	1688	1556	1650	1520	916	844	896	822
1219 (48)	1672	1542	1638	1508	1932	1780	1892	1742	1050	966	1028	944
1295 (51)	1898	1750	1860	1714	2190	2020	2148	1978	1192	1098	1168	1072
1372 (54)	2136	1970	2098	1932	2466	2274	2420	2230	1342	1236	1318	1210
1448 (57)	2390	2202	2348	2162	2758	2542	2710	2496	1504	1384	1478	1358
1524 (60)	2656	2448	2614	2406	3066	2828	3018	2778	1674	1542	1646	1514

^aTube hole counts given for tube diameters, pitches, number of passes, and nozzle: shell diameter ratios shown. Internal impingement fitted except for 0.0, which refers to vapor belt or external impingement. Refer to Sec. E. Expanded ends.

Table 2b Tube counts for U-tube exchanger (Continued)

Shell ID, mm (in)	25.4-mm (1-in) OD X 31.75-mm (1 $\frac{1}{4}$ -in) pitch A				31.75-mm (1 $\frac{1}{4}$ -in) OD X 39.69-mm (1 $\frac{5}{8}$ -in) pitch B				31.75-mm (1 $\frac{1}{4}$ -in) OD X 39.69-mm (1 $\frac{5}{8}$ -in) pitch C			
	2 pass		4	pass	2	pass	4	pass	2	pass	4	pass
	0.0	0.33	0.0	0.33	0.0	0.33	0.0	0.33	0.0	0.33	0.0	0.22
203 (8)	16	14	12		10		8		6	—	8	—
254 (10)	32	28	26	24	16	14	12	12	18	16	14	12
305 (12)	52	48	46	42	28	24	24	20	30	28	26	24
337 (13 $\frac{1}{4}$)	68	62	60	54	36	32	32	28	40	38	36	32
387 (15 $\frac{1}{4}$)	92	88	84	80	50	46	46	42	58	54	52	48
438 (17 $\frac{1}{4}$)	128	118	118	108	68	62	62	58	78	72	72	66
489 (19 $\frac{1}{4}$)	164	152	154	140	88	82	82	74	102	94	96	86
540 (21 $\frac{1}{4}$)	206	188	194	176	112	102	104	96	128	118	120	110
591 (23 $\frac{1}{4}$)	252	232	240	220	136	126	130	118	158	146	148	136
635 (25)	296	274	284	260	162	150	154	140	186	170	176	162
686 (27)	352	324	338	310	192	176	182	168	220	204	210	194
737 (29)	412	380	396	364	224	206	214	198	260	238	248	228
787 (31)	476	438	460	422	260	240	250	230	300	276	288	264
838 (33)	544	502	528	494	298	274	286	264	344	316	332	306
889 (35)	618	570	600	552	338	312	328	300	390	360	378	348
940 (37)	696	642	678	624	382	350	370	340	440	406	428	392
991 (39)	780	718	760	698	426	392	414	380	490	454	478	440
1 067 (42)	912	840	890	818	500	460	488	448	578	532	562	516
1 143 (45)	1 058	974	1 032	950	580	534	566	520	670	616	654	600
1 219 (48)	1 212	1 116	1 186	1 090	664	612	650	596	768	706	750	690
1 295 (51)	1 376	1 268	1 348	1 240	756	696	740	680	874	802	854	784
1 372 (54)	1 550	1 428	1 522	1 400	852	784	836	768	984	906	964	888
1 448 (57)	1 736	1 598	1 706	1 568	954	878	938	862	1 102	1 014	1 082	994
1 524 (60)	1 932	1 780	1 900	1 746	1 064	978	1 044	960	1 226	1 128	1 206	1 108

Tube hole counts given for tube diameters, pitches, number of passes, and nozzle: shell diameter ratios shown. Internal impingement fitted except for 0.0, which refers to vapor belt or external impingement. Refer to Sec. E. Expanded ends.

Table 2c Tube counts for split-backing-ring floating-head exchangers^a

Shell ID, mm (in)	15.88-mm (Q-in) OD × 19.84-mm ($\frac{25}{32}$ -in) pitch A								15.88-mm (%-in) OD × 22.23-mm ($\frac{7}{8}$ -in) pitch □							
	1 pass		2 pass		4 pass		6 pass		1 pass		2 pass		4 pass		6 pass	
	0.0	0.33	0.0	0.33	0.0	0.33	0.0	0.33	0.0	0.33	0.0	0.33	0.0	0.33	0.0	0.33
203 (8)	59	58	48	47	31	30	21	20	40	40	33	33	20	20	14	14
254 (10)	102	100	89	87	67	65	55	53	70	69	61	60	46	45	37	36
305 (12)	158	154	142	138	115	111	101	97	109	106	98	95	80	77	70	67
337 (13 $\frac{1}{4}$)	198	193	180	175	151	146	135	130	137	133	125	121	105	101	94	90
387 (15 $\frac{1}{4}$)	273	265	253	245	219	211	201	193	188	182	174	168	151	145	138	132
438 (17 $\frac{1}{4}$)	359	347	336	324	298	286	277	263	248	239	232	223	206	197	192	183
489 (19 $\frac{1}{4}$)	458	440	433	415	390	372	367	349	316	303	298	285	269	256	253	240
540 (21 $\frac{1}{4}$)	568	544	540	516	493	469	468	444	392	375	372	355	340	323	323	306
591 (23 $\frac{1}{4}$)	690	659	659	628	608	577	580	549	477	455	456	434	420	398	401	379
635 (25)	790	756	757	723	701	667	672	638	545	522	522	499	484	461	463	440
686 (27)	933	890	897	854	837	794	805	762	644	614	619	589	578	548	556	526
737 (29)	1088	1035	1050	997	985	932	951	888	751	715	724	688	680	644	656	620
787 (31)	1255	1192	1214	1151	1145	1082	1108	1045	866	823	837	794	790	747	765	722
838 (33)	1433	1359	1389	1315	1316	1242	1277	1203	989	938	959	908	908	857	881	830
889 (35)	1624	1537	1578	1491	1500	1413	1459	1372	1121	1061	1089	1029	1035	975	1007	947
940 (37)	1826	1727	1776	1677	1695	1596	1651	1552	1261	1192	1227	1158	1170	1101	1141	1072
991 (39)	2034	1921	1982	1869	1896	1783	1850	1737	1404	1326	1368	1290	1308	1230	1277	1199
1067 (42)	2377	2241	2321	2185	2228	2092	2178	2042	1641	1547	1602	1508	1538	1444	1504	1410
1143 (45)	2747	2587	2687	2527	2587	2427	2534	2374	1896	1786	1855	1745	1786	1670	1751	1639
1219 (48)	3143	2955	3079	2891	2973	2785	2916	2728	2170	2040	2126	1996	2052	1922	2013	1883
1295 (51)	3567	3349	3499	3281	3386	3168	3326	3108	2463	2312	2416	2265	2338	2187	2296	2145
1372 (54)	4017	3768	3946	3697	3825	3576	3762	3513	2773	2602	2723	2552	2641	2470	2596	2425
1448 (57)	4494	4210	4419	4135	4292	4008	4224	3940	3103	2907	3051	2855	2963	2767	2917	2721
1524 (60)	4998	4678	4919	4599	4785	4465	4714	4394	3451	3230	3396	3175	3304	3083	3255	3034

^aTube counts given for tube diameters, pitches, number of passes, and nozzle; shell diameters shown. Internal impingement fitted except for 0.0, which refers to vapor belt or external impingement. Refer to Sec. E. Expanded ends.

Table 2c Tube counts for split-backingring floating-head exchange& (Continued)

Shell ID, mm (in)	19.05-mm (f-in) OD X 23.81-mm ($\frac{15}{16}$ -in) pitch A								19.05-mm ($\frac{3}{4}$ -in) OD X 25.4-mm (1-in) pitch □							
	1	pass	2	pass	4	pass	6	pass	1	pass	2	pass	4	pass	6	pass
	0.0	0.33	0.0	0.33	0.0	0.33	0.0	0.33	0.0	0.33	0.0	0.33	0.0	0.33	0.0	0.33
203 (8)	40	39	33	32	20	19	14	13	30	30	24	24	15	15	—	—
254 (10)	70	69	61	60	45	44	37	36	53	52	46	45	34	33	28	27
305 (12)	108	106	97	95	79	77	69	67	82	80	74	72	60	58	52	50
337 (13 $\frac{1}{4}$)	136	133	124	121	103	100	93	90	103	101	94	92	78	76	70	68
387 (15 $\frac{1}{4}$)	188	182	174	168	150	144	138	132	143	138	132	127	114	109	105	100
438 (17 $\frac{1}{4}$)	247	239	231	223	205	197	191	183	188	182	176	170	156	150	145	139
489 (19 $\frac{1}{4}$)	315	303	297	285	268	256	252	240	240	231	227	218	204	195	192	183
540 (21 $\frac{1}{4}$)	392	375	372	355	340	323	323	306	298	285	283	270	258	245	245	228
591 (23 $\frac{1}{4}$)	477	454	456	433	420	397	401	378	363	346	347	330	319	302	305	288
635 (25)	545	522	522	499	484	461	463	440	415	397	398	380	368	350	353	337
686 (27)	644	615	619	590	578	549	556	527	490	468	471	449	439	417	423	401
737 (29)	752	716	725	689	681	645	657	621	572	545	552	525	518	491	500	473
787 (31)	867	824	838	795	791	748	766	723	660	627	638	605	602	569	583	550
838 (33)	991	940	961	910	910	859	883	832	754	715	731	692	692	653	672	633
889 (35)	1123	1063	1091	1031	1037	977	1009	949	855	809	831	785	789	743	768	722
940 (37)	1264	1194	1230	1160	1173	1103	1144	1074	962	909	936	883	893	840	870	817
991 (39)	1407	1330	1371	1294	1311	1234	1280	1203	1071	1012	1044	985	998	939	974	915
1067 (42)	1645	1551	1606	1512	1542	1448	1508	1414	1252	1181	1223	1152	1173	1102	1147	1076
1143 (45)	1902	1791	1861	1750	1791	1680	1755	1644	1448	1363	1417	1332	1364	1279	1346	1251
1219 (48)	2177	2046	2133	2002	2059	1928	2020	1889	1657	1557	1624	1524	1567	1467	1537	1437
1295 (51)	2471	2320	2424	2273	2346	2195	2304	2153	1880	1766	1845	1731	1784	1670	1753	1639
1372 (54)	2783	2610	2733	2560	2650	2477	2606	2433	2118	1987	2080	1949	2017	1886	1983	1852
1448 (57)	3114	2917	3062	2865	2974	2777	2928	2731	2370	2220	2330	2180	2263	2113	2226	2078
1524 (60)	3463	3241	3408	3186	3315	3093	3267	3045	2636	2467	2594	2425	2523	2354	2486	2317

^aTube counts given for tube diameters, pitches, number of passes, and nozzle; shell diameters shown. Internal impingement fitted except for 0.0, which refers to vapor belt or external impingement. Refer to Sec. E. Expanded ends.

Table 2c Tube counts for **split-backing-ring** floating-head **exchangers^a** (Continued)

Shell ID, mm (in)	19.05-mm ($\frac{3}{4}$ -in) OD X 25.4-mm (1-in) pitch Δ								25.4-mm (1-in) OD X 31.75-mm (1 $\frac{1}{4}$ -in) pitch $\square$							
	1 pass		2 pass		4 pass		6 pass		1 pass		2 pass		4 pass		6 pass	
	0.0	0.33	0.0	0.33	0.0	0.33	0.0	0.33	0.0	0.33	0.0	0.33	0.0	0.33	0.0	0.33
203 (8)	35	35	29	29	18	18	—	—	18	18	14	14	8	8	—	—
254 (10)	61	60	53	52	39	38	32	31	33	32	28	27	21	20	17	16
305 (12)	95	93	85	83	69	67	60	58	51	50	46	45	37	36	32	31
337 (13 $\frac{1}{4}$)	119	116	108	105	91	88	81	78	65	63	59	57	49	41	44	42
387 (15 $\frac{1}{4}$)	165	160	153	148	132	127	121	116	89	87	82	80	71	69	65	63
438 (17 $\frac{1}{4}$)	217	210	203	196	180	173	167	160	118	114	110	106	98	94	91	87
489 (19 $\frac{1}{4}$)	277	266	261	250	235	224	221	210	151	145	142	136	128	122	120	114
540 (21 $\frac{1}{4}$)	344	330	327	313	298	284	283	269	188	180	178	170	163	155	154	146
591 (23 $\frac{1}{4}$)	419	399	400	380	369	349	352	332	229	218	218	207	201	190	192	181
635 (25)	479	459	459	439	425	405	407	387	263	251	252	240	233	221	223	211
686 (27)	566	540	544	518	508	482	488	462	311	296	299	284	279	264	268	253
737 (29)	661	629	638	606	598	566	577	545	363	345	350	332	328	310	317	299
787 (31)	762	724	737	699	695	657	673	635	419	397	405	383	382	360	369	347
838 (33)	871	826	844	799	800	755	776	731	479	454	464	439	437	414	426	401
889 (35)	987	934	959	906	911	858	886	833	543	513	527	497	501	471	487	457
940 (37)	1110	1050	1080	1020	1030	970	1003	943	611	577	594	560	567	533	552	518
991 (39)	1237	1168	1205	1136	1153	1084	1125	1056	681	643	663	625	634	596	619	581
1067 (42)	1446	1363	1412	1329	1355	1272	1325	1242	796	750	777	731	746	700	729	683
1143 (45)	1672	1574	1636	1538	1575	1477	1542	1444	921	867	901	846	867	813	849	795
1219 (48)	1913	1798	1874	1759	1810	1695	1774	1659	1054	991	1032	959	996	933	977	914
1295 (51)	2171	2039	2130	1998	2061	1929	2024	1892	1197	1124	1174	1101	1136	1063	1115	1042
1372 (54)	2446	2294	2402	2250	2330	2178	2290	2138	1349	1265	1325	1241	1284	1200	1263	1179
1448 (57)	2737	2563	2691	2517	2614	2440	2573	2399	1510	1414	1484	1388	1442	1346	1419	1323
1524 (60)	3044	2848	2995	2799	2914	2718	2871	2675	1680	1571	1653	1544	1608	1499	1584	1475

^aTube counts given for tube diameters, pitches, number of passes, and nozzle: **shell** diameters shown. Internal impingement fitted except for 0.0, which refers to vapor belt or external impingement. Refer to Sec. E. Expanded ends.

Table 2c Tube counts for split-back&ring floating-head exchangers^a (Continued)

Shell ID, mm (in)	25.4-mm (1-in) OD X 31.75-mm (1½-in) pitch A								31.75-mm (1¼-in) OD X 39.69-mm (1½-in) pitch □							
	1 pass		2 pass		4 pass		6 pass		1 pass		2 pass		4 pass		6 pass	
	0.0	0.33	0.0	0.33	0.0	0.33	0.0	0.33	0.0	0.33	0.0	0.33	0.0	0.33	0.0	0.33
203 (8)	21	21	17	17	10	10	—	—	11	11	9	9	—	—	—	—
254 (10)	38	37	33	32	24	23	20	19	20	20	17	17	13	13	—	—
305 (12)	59	58	53	52	43	42	37	36	32	31	29	28	23	22	20	19
337 (13¼)	75	73	68	66	57	55	51	49	40	39	36	35	30	29	27	26
387 (15¼)	103	100	95	92	82	79	75	72	56	54	52	50	44	42	41	39
438 (17¼)	137	132	128	123	113	108	105	100	74	72	69	67	61	59	57	55
489 (19¼)	175	168	165	158	149	142	140	133	95	91	89	85	81	77	76	72
540 (21¼)	217	208	206	197	188	179	178	169	119	113	113	107	103	97	98	92
591 (23¼)	265	252	253	240	233	220	222	209	145	138	138	131	127	120	122	115
635 (25)	303	290	290	277	269	256	257	244	166	159	159	152	147	140	141	134
686 (27)	359	342	345	328	322	305	309	292	197	187	189	179	176	166	170	160
737 (29)	419	398	404	383	379	358	366	345	230	218	222	210	208	196	201	189
787 (31)	483	459	467	443	440	416	426	402	265	252	256	243	241	228	234	221
838 (33)	553	524	536	507	508	479	492	463	304	288	295	279	279	263	271	255
889 (35)	627	593	609	575	579	545	563	529	345	326	335	316	318	299	320	291
940 (37)	706	667	687	648	655	616	638	599	388	366	377	355	360	338	351	329
991 (39)	786	742	766	722	732	688	714	670	432	408	421	397	402	378	393	369
1067 (42)	920	867	898	845	862	809	842	789	506	477	494	465	474	445	464	435
1143 (45)	1064	1001	1041	978	1002	939	981	918	586	551	573	538	552	517	541	506
1219 (48)	1218	1144	1193	1119	1152	1078	1129	1055	671	630	657	616	634	593	623	582
1295 (51)	1383	1298	1356	1271	1313	1228	1289	1204	762	715	747	700	723	676	711	664
1372 (54)	1556	1461	1530	1433	1484	1387	1458	1361	859	805	844	790	818	764	805	751
1448 (57)	1744	1633	1714	1603	1665	1554	1639	1528	962	900	946	884	918	856	905	843
1524 (60)	1940	1815	1909	1784	1857	1732	1829	1704	1070	1001	1053	984	1024	955	1010	941

^aTube counts given for tube diameters, pitches, number of passes, and nozzle: shell diameters shown. Internal impingement fitted except for 0.0, which refers to vapor belt or **external** impingement. Refer to Sec. E. Expanded ends.

Table 2c Tube counts for split-backing-ring floating-head exchangers (Continued)

Shell ID, mm (in)	31.75-mm (1¼-in) OD X 39.69-mm (1½-in) pitch A							
	1 pass		2 pass		4 pass		6 pass	
	0.0	0.33	0.0	0.33	0.0	0.33	0.0	0.33
203 (8)	13	13	10	10	—	—	—	—
254 (10)	27	23	20	20	14	14	—	—
305 (12)	37	36	33	32	27	26	23	22
337 (139)	47	45	43	41	36	34	32	30
387 (15¼)	65	63	60	58	52	50	47	45
438 (174)	86	83	80	77	71	68	66	63
489 (19¼)	110	106	104	100	93	89	88	84
540 (21¼)	137	131	130	124	119	113	112	106
591 (23¼)	167	159	159	151	147	139	140	132
635 (25)	192	183	184	175	170	161	163	154
686 (27)	227	216	218	207	203	192	195	184
737 (29)	265	252	255	242	240	227	231	218
787 (31)	307	291	297	281	280	264	271	255
838 (33)	351	332	340	321	322	303	312	293
889 (35)	398	376	386	364	367	345	357	335
940 (37)	448	423	436	411	415	390	405	380
991 (39)	499	471	486	458	465	437	453	425
1 067 (42)	585	551	571	537	548	514	536	502
1 143 (45)	676	636	661	621	636	596	623	583
1 219 (48)	775	728	759	712	733	686	719	672
1 295 (51)	880	826	863	809	835	781	820	766
1 372 (54)	992	930	974	912	944	882	929	867
1 448 (57)	1110	1040	1091	1021	1060	990	1043	973
1 524 (60)	1236	1156	1216	1136	1183	1103	1165	1085

^aTube counts given for tube diameters, pitches, number of passes, and nozzle: shell diameters shown. Internal impingement fitted except for 0.0, which refers to vapor belt or external impingement. Refer to Sec. E. Expanded ends.

4.2.6

Features relating to mechanical design and fabrication

E. A. D. Saunders

A. Barrels

The shell barrel must be straight and have no **out-of-roundness**, as a tightly fitting tube bundle must be inserted into it. Most shell and head barrels greater than about 450 mm in inside diameter are rolled from plate, as shown in Fig. 1, and a complete shell barrel may comprise several smaller barrels, or strakes, welded together end to end. If there is any out-of-roundness, individual strakes are rerolled after welding the longitudinal seams. The longitudinal seams of adjoining strakes are always staggered. The inside diameter of a rolled shell should not exceed the design inside diameter by more than 3.2 mm ($\frac{1}{8}$ in) as determined by circumferential measurement. All internal welds must be made flush.

When welding large nozzles to the shell, sinkage may occur at the nozzle/shell junction and effective measures, such as the use of temporary stiffening, must be taken to avoid it. **Sinkage** reduces the shell diameter at the nozzle/shell junction so that the baffle diameter must be reduced accordingly. The increased clearance between baffle and shell may result in reduced thermal performance.

Standard pipe less than 450 mm in diameter is usually available, and this will be used for the shell and head barrels instead of rolled plate. Depending on the fabricator's roll capacity, at thicknesses of the order of 80 mm and greater, or large thickness/diameter ratios, it may be necessary to use forged instead of rolled barrels.

When an expensive barrel metal is required for corrosion resistance purposes only, the barrel is formed

from the selected metal if its thickness is less than about 15 mm. Above this the use of clad metal should be investigated, as it may provide a cost saving. The clad metal will usually comprise a steel plate, having a thickness suitable for the pressure and temperature conditions, with a layer of the required corrosion-resistant metal, about 3 mm thick, bonded to it. The cladding may be applied by explosive, roll bonding, or weld deposition methods.

TEMA specifies minimum shell and head barrel thicknesses, which depend on barrel diameter, metal, and TEMA class. Class R minimum thicknesses are as follows:

Shell diameter	Carbon and low alloy	Alloy
330-737 mm (13 to 29 in)	9.5 mm ($\frac{3}{8}$ in)	4.8 mm ($\frac{3}{16}$ in)
762-991 mm (30 to 39 in)	11.1 mm ($\frac{7}{16}$ in)	6.4 mm ($\frac{1}{4}$ in)
1 016-2 032 mm (40 to 80 in)	12.7 mm ($\frac{1}{2}$ in)	7.9 mm ($\frac{5}{16}$ in)
2 057-2 540 mm (81 to 100 in)	12.7 mm ($\frac{1}{2}$ in)	9.5 mm ($\frac{3}{8}$ in)

B. Heads

Bonnet-type heads are made from dished ends of **semi-ellipsoidal**, **torispherical**, and **hemispherical** forms, the latter being used chiefly for high pressures where its shape provides a smaller wall thickness as the other ends. The minimum thickness of bonnets is usually the same as that for barrels.

Floating-head covers are dished to spherical shape and must be positioned such that the crossover area

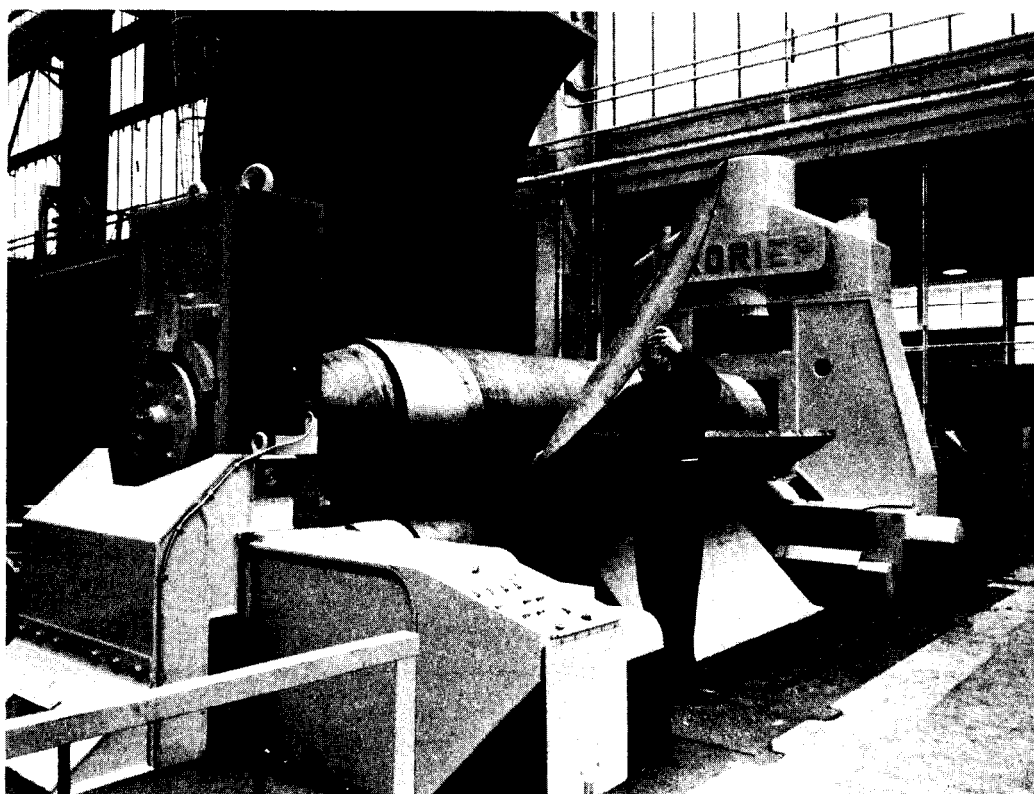


Figure 1 Rolling a shell barrel. Courtesy of Whessoe Heavy Engineering Ltd., Darlington, England.

between successive passes for a multitube pass **TEMA** class R exchanger is at least 1.3 times the flow area through the tubes of one pass. For the other classes this factor is reduced to 1.0. For a single tube pass the depth at the nozzle center line must not be less than one-third of the nozzle diameter.

Although most bobbin, bonnet, and floating heads are fabricated from plate, cast heads are sometimes used, these being restricted to low-pressure, nonlethal service, particularly cooling water. Castings in iron, steel, and aluminum bronze are typical choices.

Where an expensive metal is required for corrosion resistance, the use of clad metal should be investigated as described in Sec. A. In addition, the use of carbon steel heads having nonmetallic linings, such as synthetic rubber, epoxy resin, or phenolic resin, should be investigated in order to reduce cost.

C. Nozzles

Most nozzles comprise a length of standard pipe, welded to the shell or head barrel at one end, and to a slip-on nozzle flange, as shown in Fig. 2, or a weld neck nozzle flange, as shown in Fig. 3, at the other. For moderate service, reinforcement of the nozzle opening, if required, is usually provided by a reinforcing pad as shown in Fig. 3. For high-integrity services, where it is desirable

to minimize the number of welds and eliminate reinforcing pads, forged self-reinforced long weld neck nozzles may be used, as shown in Fig. 4, most of the required reinforcement being in the nozzle barrel itself. As a further refinement, swept forged nozzles, as shown in Fig. 5, may be used. Reinforcement is contained within the wall of the fitting, which is contoured at the base to butt weld directly into the barrel, thus allowing the nozzle/shell weld to be inspected.

In the case of clad barrels, the nozzle barrel and flange may be supplied in carbon or alloy steel with the flange gasket face and inside of the nozzle barrel weld deposited with the corrosion-resistant metal. **Alterna-**

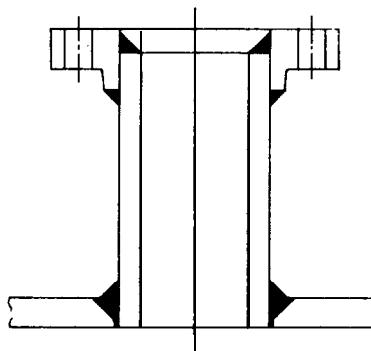


Figure 2 Slip-on nozzle flange.

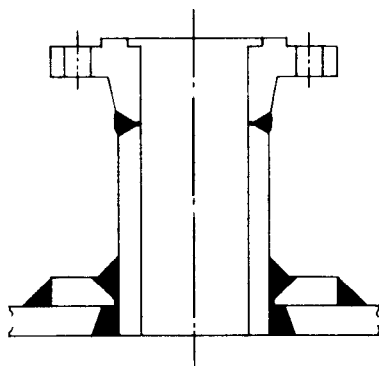


Figure 3 Weld neck nozzle flange.

tively, a thin machined liner in the corrosion-resistant metal may be inserted inside the nozzle barrel as shown in Fig. 6.

At high pressures the nozzle flange is often omitted altogether. A stub end, as shown in Fig. 4.2.4(3), is provided for direct welding to the piping.

D. Flanges

(a) Girth flanges

Ring-type flanges, as shown in Fig. 7, are used for moderate services and may be cut from plate or purchased as rolled or forged rings. As diameter, pressure, and temperature increase, the weld neck flange, as shown in Fig. 8, is used because this type requires less welding, which can be readily carried out by automatic means, and the fit-up is simpler. Weld neck flanges are purchased as rolled or forged rings, and some fabricators may elect to use this type at all times.

When clad barrels are involved, the gasket face and bore of the flange are weld deposited with the corrosion-resistant cladding metal as shown in Fig. 9.

When using an expensive metal that cannot be weld deposited to carbon, or low-alloy steel, lap-type flanges as shown in Fig. 10 are used. The barrel and lap are made in the expensive metal, but the loose backing

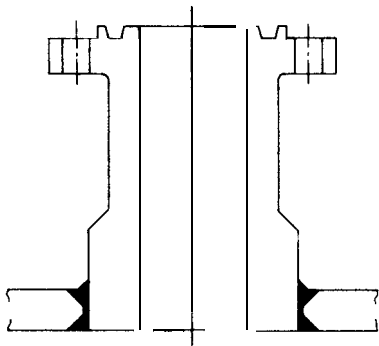


Figure 4 Self-reinforced long weld neck nozzle.

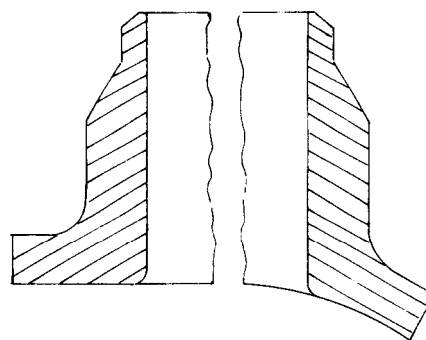


Figure 5 Swept forged nozzle.

flange is made in carbon or low-alloy steel, which avoids having to provide a complete flange in expensive metal.

(b) Floating head

Floating-head flanges and backing rings are also purchased as rolled or forged rings. At diameters below about 450 mm, a hoggedout floating head is often provided as a cheaper alternative to the conventional flange and welded-on dished floating-head cover. Instead of a separate flange and dished cover, an integral flange and flat cover are machined from a plate or forged disk.

(c) Gaskets

In all TEMA classes, metal-jacketed asbestos or solid-metal gaskets are mandatory for all floating-head joints, irrespective of pressure, and all external joints for pressures greater than 2 MN/m^2 . For pressures below 2 MN/m^2 , gaskets for class R exchangers must be metal-jacketed asbestos or solid metal if the contained fluid is a hydrocarbon; otherwise gaskets in other materials such as asbestos composition or rubber are permissible in all classes. Spiral-wound gaskets may be used under all conditions. Gaskets made in more than one

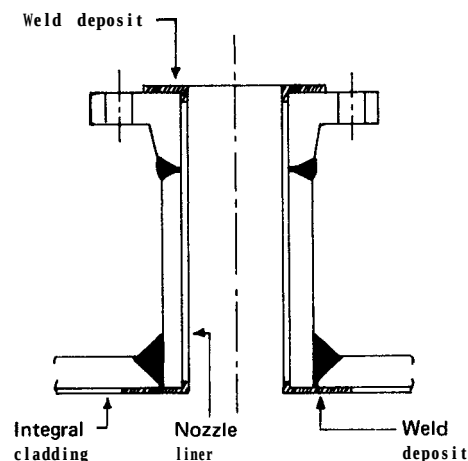


Figure 6 Nozzle with liner.

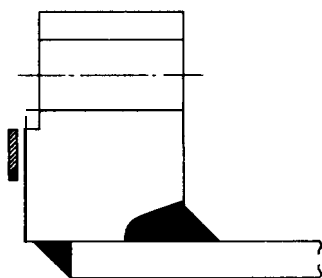


Figure 7 Rig-type flange.

piece should be avoided if possible. If solid-metal gaskets are obtained in more than one piece, the welded joints in the gasket must be smooth and heat treated, and the hardness checked before use.

The minimum width of the gasket peripheral ring is 9.5 mm for diameters less than 580 mm and 12.5 mm for larger diameters. The width of the pass partition bars may be 3 mm less in both cases.

(d) Bolting

Care must be exercised in tightening bolts of small diameter or having low yield values in order to avoid overstressing both bolts and flange. Sometimes when low-yield bolts, such as stainless steel, are involved, temporary low-alloy bolts are used for intermediate hydrostatic tests and the low-yield service bolts are not used until the final joint is made. When bolts larger than about 50 mm are involved, it becomes difficult to obtain the required bolt load by hand wrenching and special hydraulic bolt tensioners or electrical heating methods are required.

A bolt tensioner comprises a hydraulic cylinder attached to a frame that straddles the bolt and grips it above the nut. The bolts must therefore be extended by a length of about one diameter and the bolt spacing must be wide enough to accommodate the frame. The bolt is stretched when pressure is applied to the cylinder and the nut is run down to its seating by hand. The equipment is calibrated so that the bolt can be stretched by any predetermined amount. Additional equipment can be installed so that a number of bolts can be tensioned simultaneously.

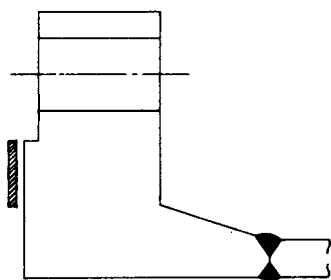


Figure 8 Weld neck flange.

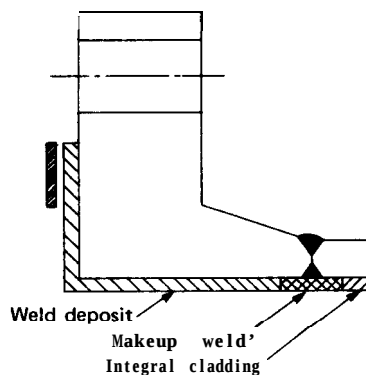


Figure 9 Clad flange.

In electrical heating, a central hole is drilled down the entire length of each bolt to accommodate an electrical heating element. Heating expands the bolt and upon reaching the predetermined amount the nut is run down to its seating by hand. To compensate for the loss of metal due to the central hole, the bolts must be made larger in diameter to provide the required bolt cross-sectional area. The whole operation is not fast, because of the heating-up and cooling-down periods required.

The combination of temperatures and thermal expansion coefficients of the flanges and bolts during service may provide a leaking joint as a result of the bolts loosening. Conditions at startup, shutdown, **cleaning-out**, and so on, in addition to steady operation, should be checked, and if leakage is likely to occur bolt extension sleeves, as shown in Figs. 11 and 13, or spring washers, as shown in Fig. 12, may be necessary.

Special attention should be given to stationary tubesheet joints of A- and B-type headers where the tubesheet is held between through-bolted shell and head flanges [Fig. 4.2.3(3)]. If the flanges, tubesheet, and bolting have different expansion coefficients and/or operate at different temperatures, both peripheral joint and interpass leakage may occur. One solution is to weld the tubesheet to both shell and head barrels, but to enable the bundle to be removed, flanges are provided in the shell about 300 mm from the tubesheet, as shown in the upper bundle of Fig. 4.2.3(13). The problem of achieving a tight joint is greatly reduced, as the shell

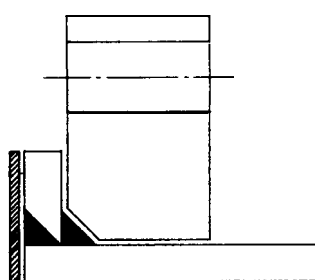


Figure 10 Lap-type flange.

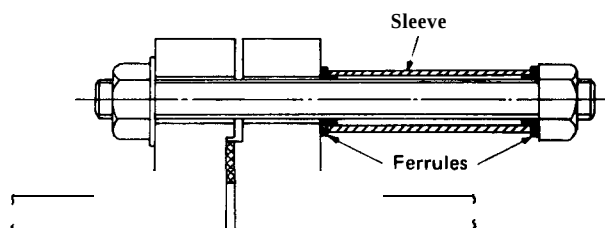


Figure 11 Bolt extension sleeve.

flanges have similar metals and temperatures so no external leakage can occur at the tubesheet.

The flanged shell skirt, tubesheet, and stationary head form an integral unit that is removed with the bundle after breaking the shell flanges.

If there is more than one tube-side pass, it may be necessary to box in the passes, as shown in Fig. 4.2.4(3) and described in Sec. 4.2.5B(d), to eliminate interpass leakage.

E. Tubesheets

Depending on the service, and metal required, tubesheets less than about 100 mm thick are made from plate, but at greater thicknesses or for high-integrity services, they are made from forged disks. When tubesheets in other than carbon or low-alloy steels are required, the use of clad plate should be considered. This may provide

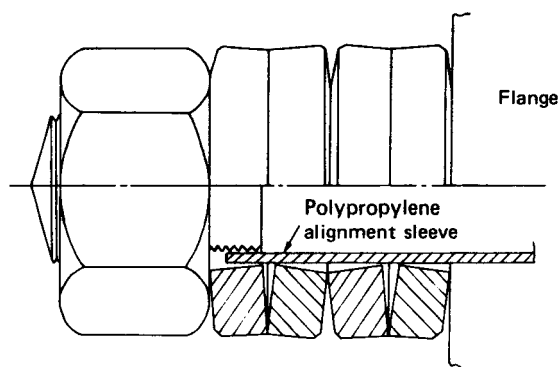


Figure 12 Bolt with spring washers.

a cost saving, particularly at the larger diameters and thicknesses.

A clad tubesheet will consist of a carbon or low-alloy steel backing plate, having a thickness suitable for the design temperature and pressure, with a layer of the required tubesheet metal bonded to it. The cladding will be about 9.5 and 3 mm thick for multi- and single tube-side passes, respectively. Although the clad metal may be applied by weld deposition, explosion-clad tubesheets are widely used because the process provides a high-integrity bond with no contamination of the cladding from the backing metal. In addition, the explosion-clad process provides a much greater combi-

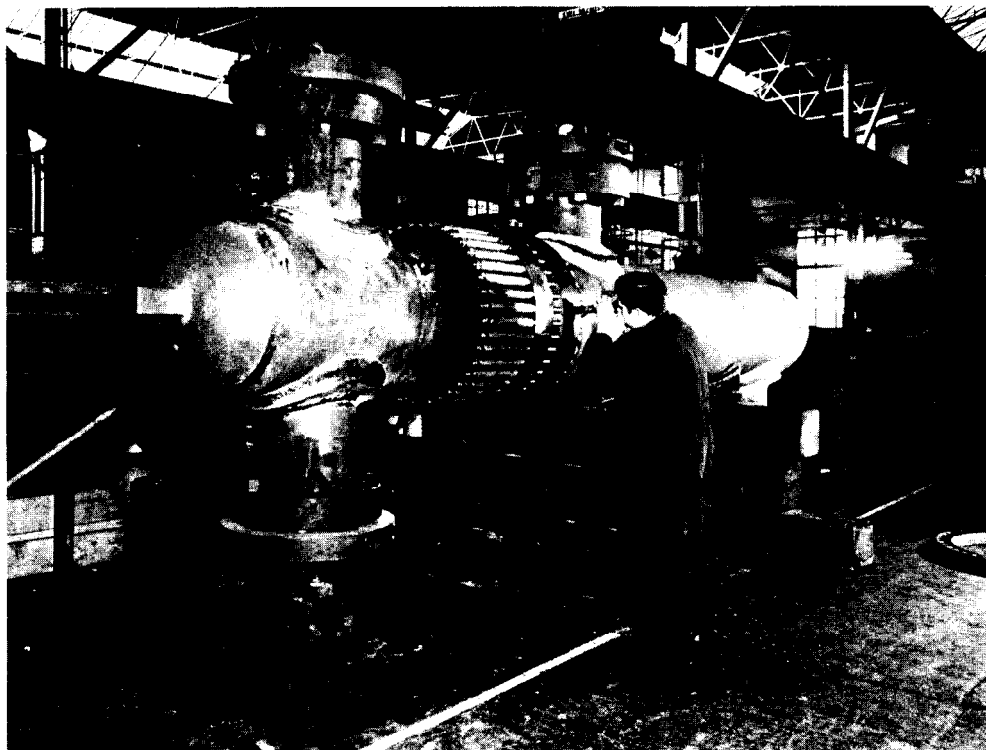


Figure 13 U-tube exchanger with bolt compensating sleeves at stationary tubesheet joints. Courtesy of Whessell Heavy Engineering Ltd., Darlington, England.

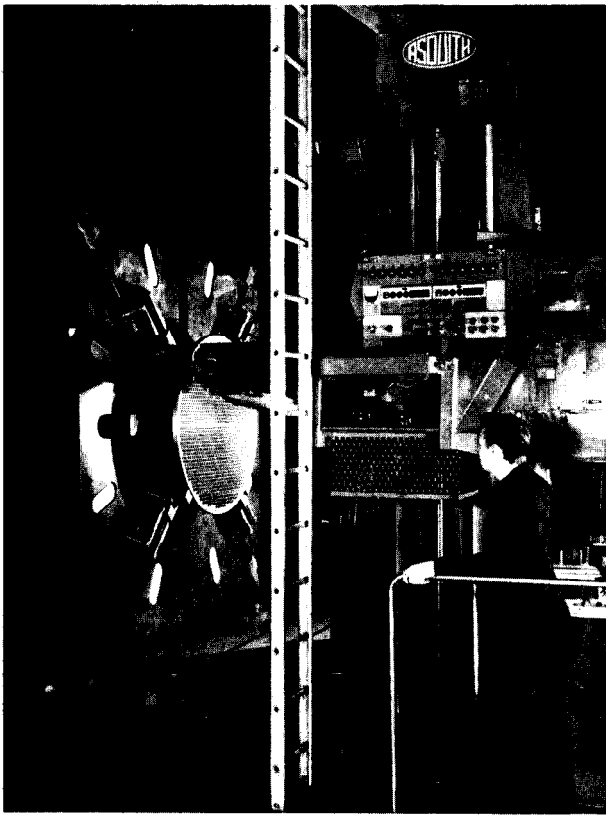


Figure 14 Drilling tubesheet on tape-controlled drilling machine. Courtesy of Whessoe Heavy Engineering Ltd., Darlington, England.

nation of cladding/backing metals than the weld deposition process.

The use of clad tubesheets becomes essential when a single metal that is corrosion resistant to both **shell**- and tube-side fluids is not available or is too costly.

TEMA specifies tolerances for tube hole diameter, ligament width, and drill drift. Hole diameter tolerances provide for standard fit and special close fit of the tubes, the latter fit being recommended for tubes that are susceptible to work hardening. Holes are drilled, reamed, and the edges chamfered slightly to prevent damage during bundle assembly. Each pair of tubesheets of double tubesheet exchangers should be drilled together, or one jig drilled from the other, and held together as a matching pair throughout all stages of manufacture.

Minimum tubesheet thicknesses are provided by TEMA, and for class R the total thickness less corrosion allowance should not be less than the tube outside diameter. Some fabricators may standardize on greater minimum thicknesses in order to reduce distortion during tube end attachment (see Fig. 14).

Typical **shell/tubesheet** joints are shown in Fig. 15.

F. Tube end attachment

(a) Roller expansion

The most common form of tube end attachment is roller expansion, where the force produced by an expanding tool deforms the tube radially outward to make a mechanical seal between tube and hole (see Fig. 16). The reamed finish provides about the optimum surface for this type of joint. At pressures greater than 2 MN/m^2 and/or temperatures greater than 180°C , it is mandatory for all TEMA classes to provide at least two expansion grooves $3.2 \text{ mm wide} \times 0.4 \text{ mm deep}$ ($\frac{1}{8} \text{ in} \times \frac{1}{16} \text{ in}$). At lower temperatures and pressures, grooves are not mandatory for class C, but some fabricators may elect to provide them for all roller expanded ends. The grooves

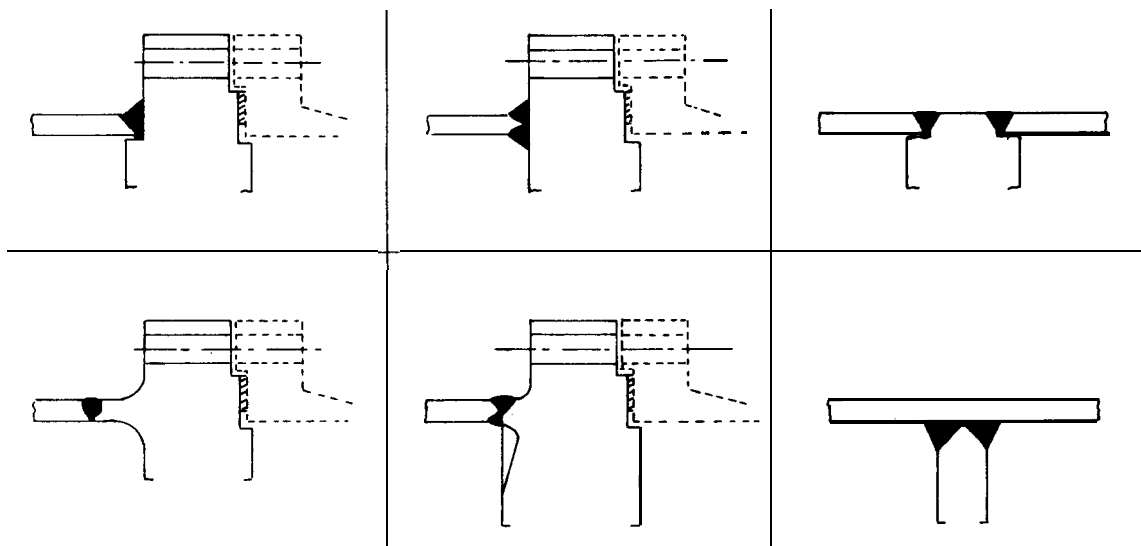


Figure 15 Typical shell/tubesheet joints.

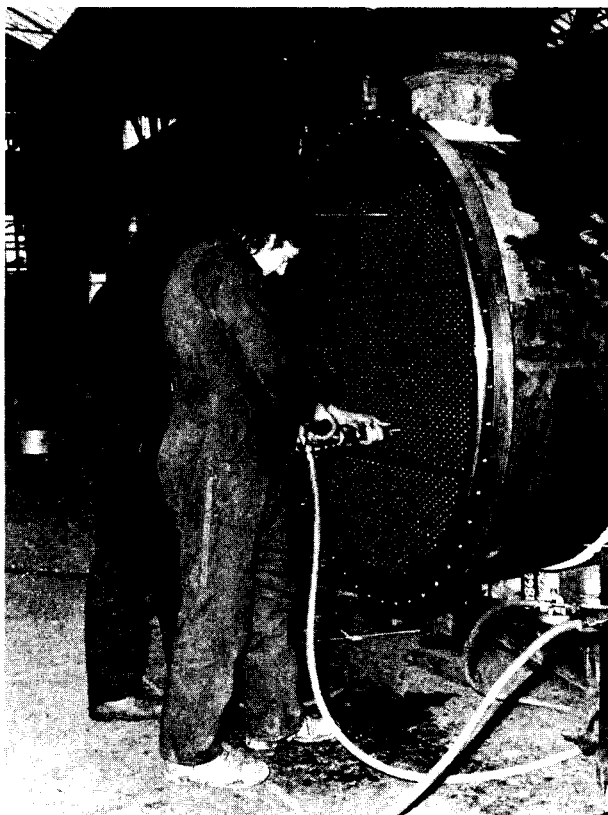


Figure 16 Roller expanding tube ends. Courtesy of Whessoe Heavy Engineering Ltd., Darlington, England.

provide axial strength and seal off any continuous irregularities on the tube surface.

Typical details for single and bimetal tubes are shown in Figs. 17 and 18, respectively.

The expanding operation is not covered by design and construction codes, but is left to the fabricator. Although each fabricator has individualized techniques and criteria, it is considered that an expansion that will thin

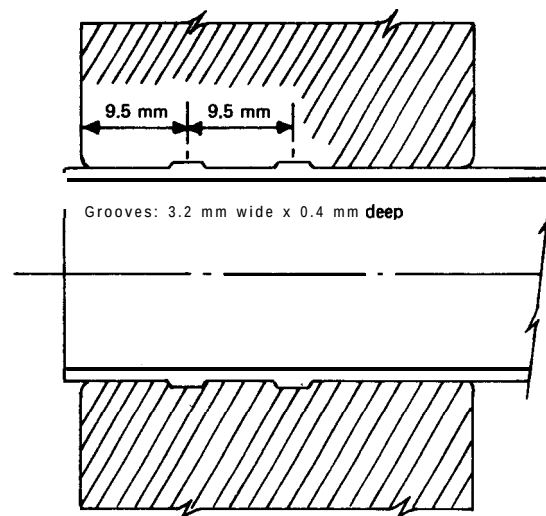


Figure 17 Standard roller expansion.

the tube wall by 3 to 7% produces a sound joint. However, special **tube/tubesheet** combinations may require a test block to determine optimum expansion before production expansion begins. Underexpansion produces a joint that will leak on test, and the only remedy is to reexpand it lightly until it holds the pressure. Overexpansion is more serious in that it cold works the tube excessively, leaving it prone to corrosion, and may distort the tube hole and ligament.

It is essential that expansion not extend further than 3 mm from the inner face of the tubesheet in order to avoid bulging the tube.

(b) Explosive expansion

In explosive expansion an explosive charge is placed inside the tube within the tubesheet thickness. Upon detonation the force produced deforms the tube radially

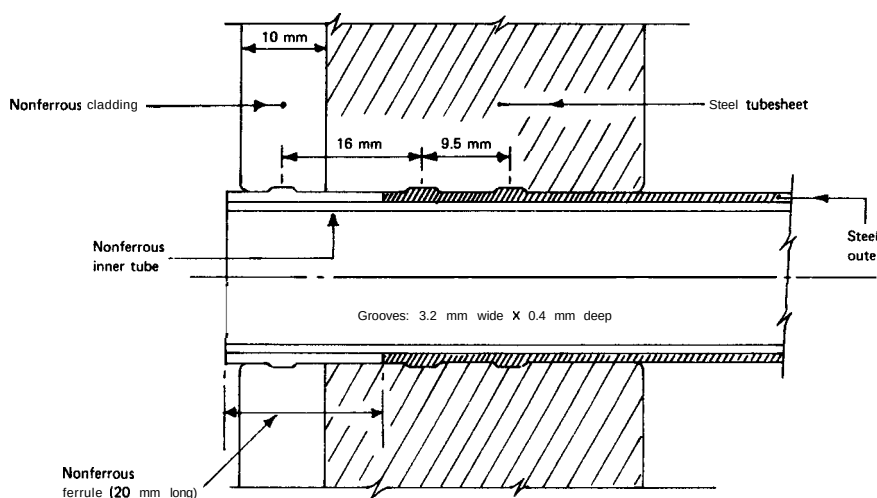


Figure 18 Roller expansion for bimetal.

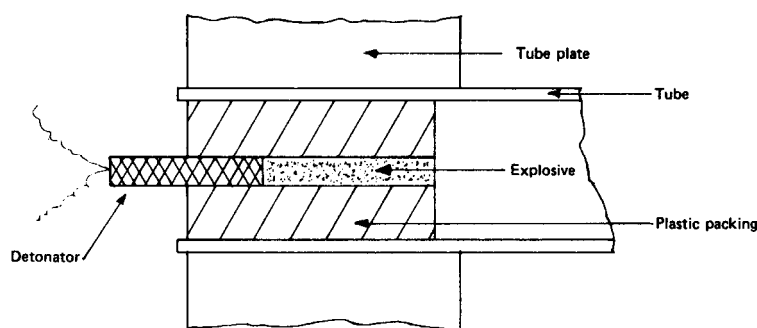


Figure 19 Explosive expansion.

until it fits tightly against the hole. Sound joints may be made with **tube/tubesheet** combinations, or thick-wall tubes, which would be difficult to achieve with roller expansion. The process is not sensitive to the condition of the tube and hole and can cater for misshapen, over-size, or dirty holes. For this reason existing exchangers having roller expanded or welded tube ends, which continually leak, may be repaired by explosive means without disturbing the tubes, and in some cases, without removing the exchanger from the plant.

The principle of explosive expansion is shown in Fig. 19. No alteration to the tube pitches specified in Sec. 4.2.5A(c) is required, but the tube bore should not be less than about 10 mm. Except for the reasons above, explosive expansion is not justified where roller expansion achieves satisfactory results, because of its higher cost.

(c) Arc welding- "outer face "

Depending on tube diameter, thickness, and metal, a roller-expanded joint may withstand pressures in the region of 25 MN/m^2 under hydrostatic test in the shops. However, the condition in the shops bears no relation to service where relaxation of the rolled joint occurs as a result of pressure/temperature fluctuations and **vibra-**

tion. Tube end welding should be considered where leakage must be avoided, particularly if the tube ends are subject to thermal shock, thermal cycling, vibration or a high-pressure difference, say, 6 MN/m^2 . Welding may also provide the best solution for thick-wall or **work-hardening** tubes.

Typical tube end weld preparations for the outer face of the **tubesheet** are shown in Fig. 20 (see also Fig. 21). Current practice demands that no tube end expansion should be carried out prior to welding, as otherwise the welds are likely to be porous. In addition, heavy expansion after welding should be avoided, in which case expansion grooves are not required. However, light expansion, away from the weld, sufficient to bring the tube into intimate contact with the hole, is recommended where crevice corrosion or vibration is likely to occur.

(d) Explosive welding

This is similar to explosive expansion described in Sec. F(b), and the principle is shown in Fig. 22. Various combinations of tube and tubesheet metals, which cannot be welded by conventional means, may be welded by explosive means. The integrity of a joint welded by explosive means is superior to conventionally welded joints

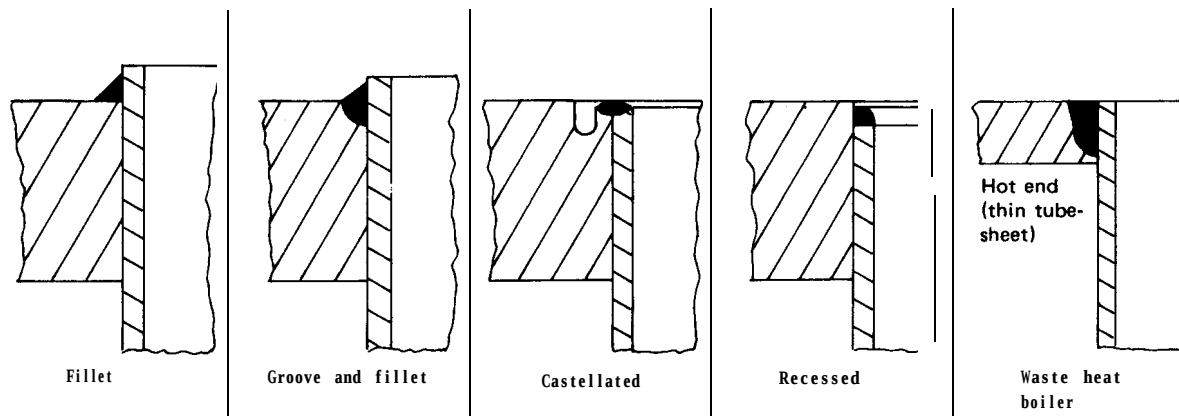


Figure 20 Typical tube end welds.

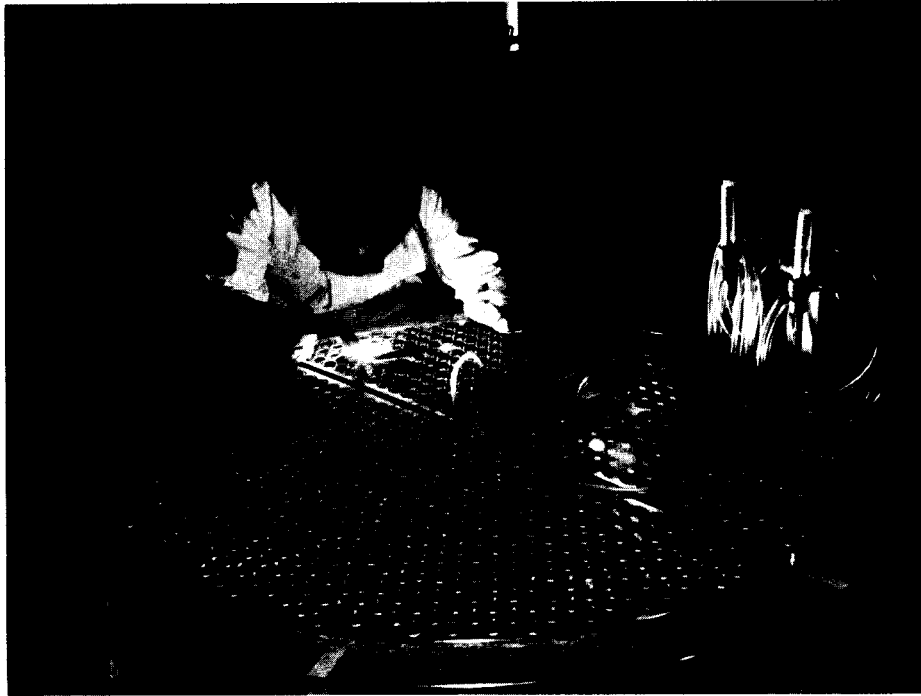


Figure 21 Tungsten inert gas welding of stainless steel tubes. Courtesy of Whessoe Heavy Engineering Ltd., ^{ton,} Darlington, England.

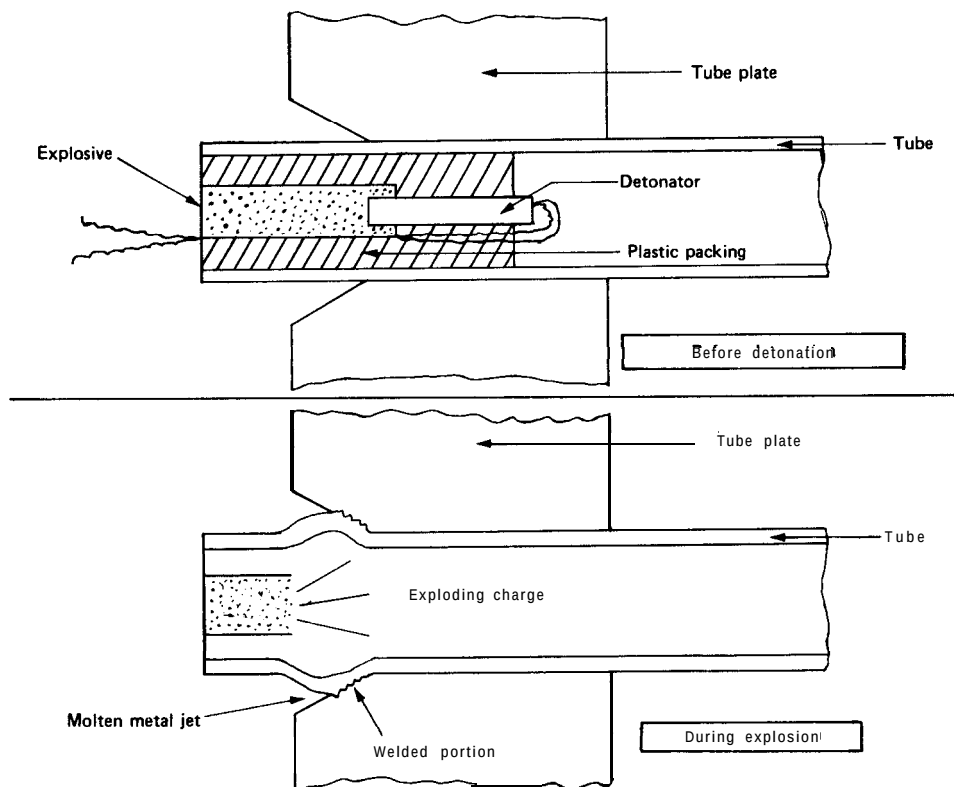
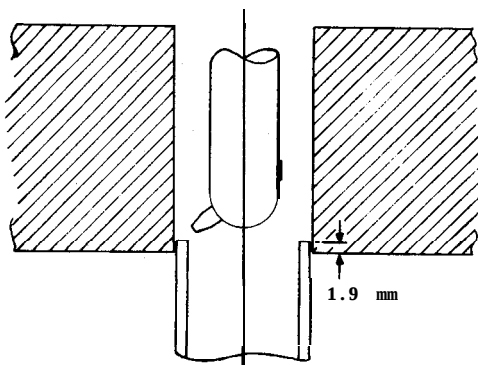


Figure 22 Explosive tube end welding.



(Approx. dimensions for 19-mm OD tube)

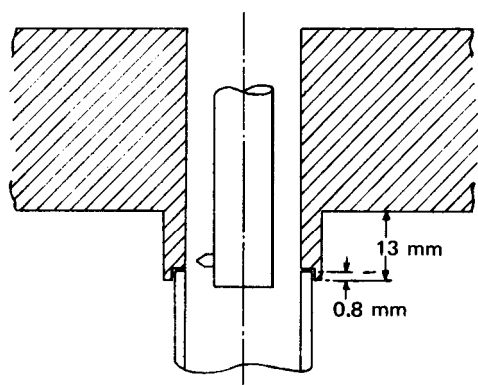


Figure 23 Back-bore tube end welding.

on all counts, namely, strength, fatigue, and pressure. If the front portion of the tube is explosively welded in accordance with Fig. 22 and the remainder of the tube explosively expanded in accordance with Fig. 19, the combination represents a joint of exceptional integrity.

Explosive welding costs more than other attachments, and greater ligament widths are required. These

are a function of tube wall thickness being approximately 9 mm at 1.24 mm (18 BWG) thick, 11 mm at 1.64 mm (16 BWG), and 13 mm at 2.11 mm (14 BWG). Ligaments may be reduced by thinning and tapering the tube wall at the welded portion.

(e) Arc welding- "inner face"

In all the tube end attachments described above there is always a tube length of 3 to 5 mm that is not in contact with the tube hole at the inner face of the tubesheet. As explained in Sec. F(a), no attempt must be made to close this gap by expansion, in which case crevice corrosion may occur.

Crevice corrosion may be eliminated if the tubes are welded to the inner face of the tubesheet, the welding being carried out down the tube hole from the front face. This is known as back-bore welding and the principles of two methods are shown in Fig. 23. The welds may be fully inspected and in the case of thick tubesheets there is a significant saving in tube cost.

G. Expansion bellows

Expansion bellows may be required in the shell of fixed tubesheet exchangers or at the floating end of single-tube-pass floating-head exchangers of the S and T types. There are numerous designs of expansion devices, but three common types made by specialist manufacturers are the thick wall, thin wall, and hot-formed medium wall.

The thick-wall bellows are invariably used in the shell of fixed tubesheet exchangers and are made from plate having the same metal and thickness as the shell barrel. The bellows form is shown in Fig. 24. The complete bellows convolution is assembled from a number of segments welded together longitudinally without any

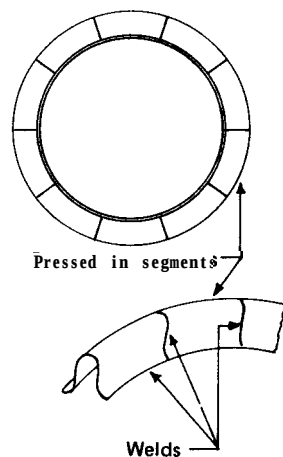
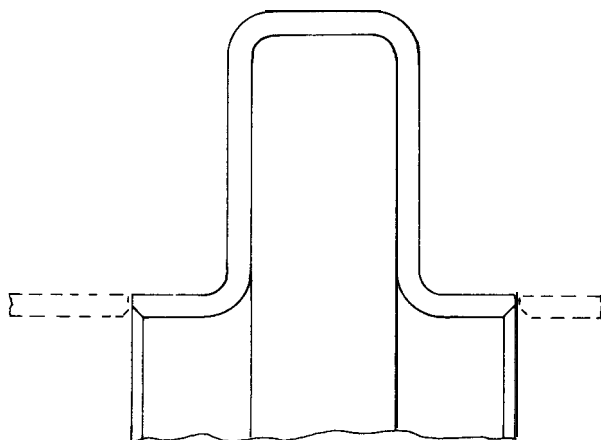


Figure 24 Thick-wall bellows.

circumferential welding in the convolution itself. Thickness is usually in the range 4 to 13 mm, with a convolution depth of 100 to 150 mm, but because of its stiffness the movement of one convolution is limited to about 2 to 5 mm. Larger movements can be accommodated by adding more convolutions, but the cost will then be higher than the thin-wall type. Its chief advantage is that the rugged construction allows the complete exchanger to be handled easily without the need for bellows protection or restraining bars.

The thin-wall bellows, shown in Fig. 25, has convolutions in stainless steel or Incoloy, formed cold, either by rolling or hydraulic forming. Convolution depth is about 24 to 75 mm, with a single-ply thickness of 0.5 to 2 mm, although multi-ply construction is used for higher pressures. This design of bellows offers greater scope than the thick-wall type with regard to movement and pressure. It has the disadvantage that it requires careful handling at all times, and it is usual to fit protective liners both inside and outside. It is also essential to provide external restraining bars to keep the complete unit rigid during handling. Once installed, the restraining bars must be released to permit axial thermal movement.

The hot-formed medium-wall bellows offers a compromise between the thick and thin wall types, being more flexible than the thick wall, yet offering a more rugged construction than the thin wall. Material is usually a chrome-molybdenum alloy or stainless steel, 2.0 to 4.5 mm thick, with a convolution height of 50 to 63 mm.

H. Hydrostatic testing

If tube end welding is involved, the welds are given a low-pressure air or halogen test on the shell side at about 0.07 MN/m^2 before the final hydrostatic tests. It is

easier to make sound tube end weld repairs if the ends have not been in contact with water. If a multipass technique has been used for the tube end welding, then each pass must be given a low-pressure test before proceeding to the next.

Upon completion the exchanger must be hydrostatically tested with water to check the soundness of all welded seams, tube end, and gasketed joints. The shell side and tube side of the exchanger must be tested independently to their specified code test pressures, unless the exchanger has been designed for a differential pressure, in which case both sides must be pressurized together, taking care that the differential test pressure is not exceeded.

The fabricator prefers to test the tube ends with pressure on the outside of the tubes, as this allows every tube end to be checked for tightness at the outer tubesheet face and leakers can be positively identified. This is acceptable if the shell-side test pressure is equal to or greater than the tube-side test pressure. When the reverse is the case, the customer may insist that the tube ends be tested with pressure on the tube side, which means that removable bundles must be tested outside the shell. In this event leakage is detected as drips from the bottom of the bundle at the inner tubesheet face, but positive identification of the actual leakers is more difficult, particularly with the tube configurations commonly used. The fabricator endeavors to minimize unnecessary rerolling of tube ends, but some may be inevitable.

Split straight tubes and outer U tubes are removed and replaced if practicable, but otherwise they are plugged. Inner U tubes must be plugged. Explosive plugging is available for high-integrity services.

Testing of fixed tubesheet exchangers is straightforward in that there are only two tests to carry out. The shell side is pressurized first, with the heads or head covers removed, to check the shell welding, tube ends,

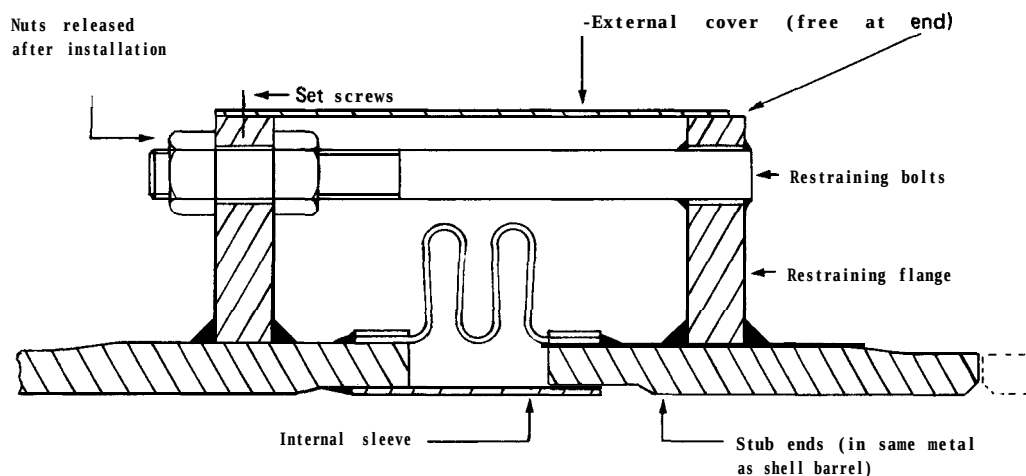


Figure 25 Thin-wall bellows.

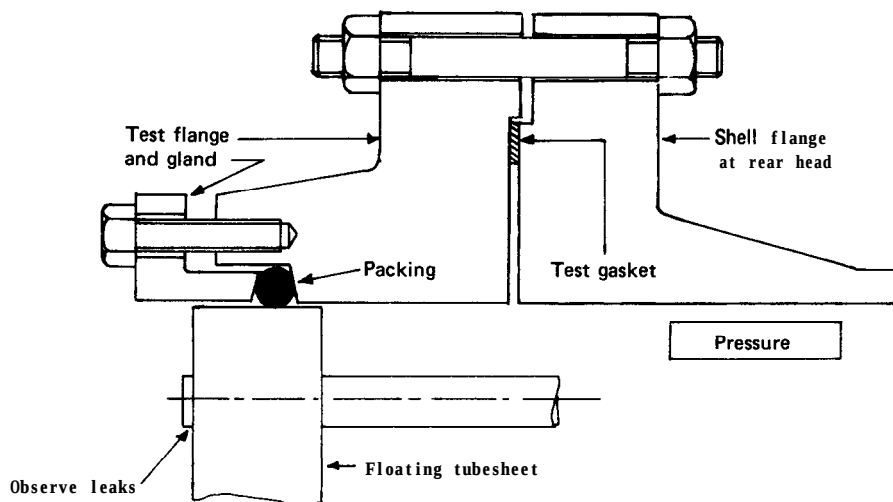


Figure 26 Test flange and gland for floating-head exchanger.

and for split tubes. Unless the exchanger has been designed for a differential pressure, the shell side is drained, the heads or head covers fitted, and the tube side pressurized to check the head welding and all **gasketed** head joints. Although this again tests the tube ends, leaks can be revealed only as drips from bottom **shell** nozzles.

In order to test **S**- and **T**-type floating-head bundles inside the shell, with the test pressure outside the tubes, a special test flange and gland is required at the floating head as shown in Fig. 26. This allows tube ends to be inspected at the outer face of the floating tubesheet. The construction of **W**- and **P**-type floating-head ex-

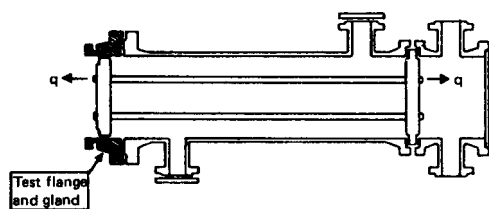
changers is such that they have built-in test flange and gland facilities. It should be noted that to test **T**-type floating-head bundles inside the shell, using the test flange and gland, the bolt holes in the floating tubesheet must be plugged temporarily. If the **T** type has no rear head flanges, the test flange and gland cannot be used, neither can it be used for **T**-type kettles. Sometimes a separate test shell is made to suit these cases.

When testing any removable bundle inside the shell, with the test pressure outside the tubes, a joint must be made between the shell and stationary tubesheet to contain the shell-side pressure. There must also be access to the outer face of the stationary tubesheet to inspect the

Test 1a: Pressure in shell—Test gaskets fitted

Purpose

- (1) To check tube expansions and split tubes (q)

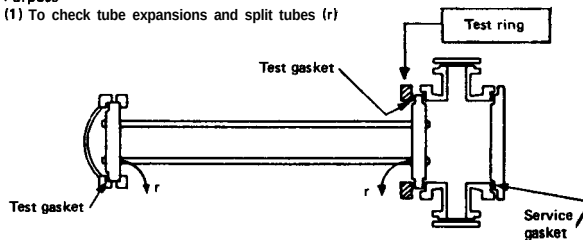


Test when shell-side test pressure $\geq$ tube-side test pressure

Test 1b: Pressure in tubes

Purpose

- (1) To check tube expansions and split tubes (r)

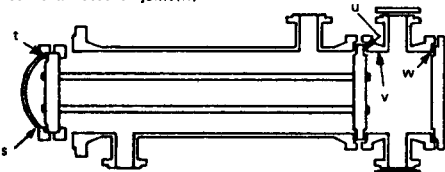


Test when tube-side test pressure $>$ shell-side test pressure

Test 2: Pressure in tubes—Service gaskets fitted

Purpose

- (1) To check floating-head cover welding (s)
 (2) To check floating-head joint (x)
 (3) To check channel/tube plate joint (u)
 (4) To check channel welding (v)
 (5) To check channel/cover joint (w)



Test 3: Pressure in shell—Service gaskets fitted

Purpose

- (1) To check shell end shell cover welding (x)
 (2) To check shell/shell cover joint (y)
 (3) To check shell/tube plate joint (z)

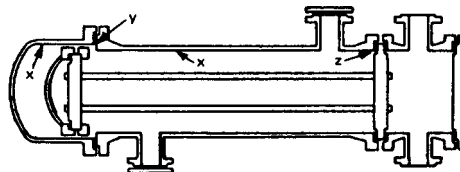


Figure 27 Hydrostatic testing of split-backing-ring floating-head exchanger.

tube ends. Bobbin-type heads satisfy these requirements, but bonnet-type heads do not and they must be omitted for the test. Unless the tubesheet is flanged, it is necessary to replace bonnet-type heads by a special test ring or backing flange on the tube side of the stationary **tubesheet** to hold the **shell/tubesheet** joint.

For testing the bundle outside the shell, with the pressure inside the tubes, a joint must be made between the stationary head and stationary tubesheet to contain the tube-side pressure. Unless the tubesheet is flanged,

or integral with the head, a similar test ring will be required on the shell side of the stationary tubesheet to hold the **tubesheet/head** joint.

The test procedure for an S-type floating-head exchanger is shown in Fig. 27 from which the special test ring will be seen. A similar testing procedure is used for other removable bundle exchangers. If any **gasketed** joints have to be broken during testing, test gaskets are used instead of the service gaskets, these being stored until the service joints are finally made.

4.3

Shell-and-tube heat exchangers: Review of mechanical design codes

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4.3.1

Mechanical design codes

M. Morris

A. Introduction

Pressure vessel codes, which also cover the mechanical design of shell-and-tube exchangers, fulfill various functions; in several countries a national code is legally enforced, and compliance with the code is mandatory for items supplied to that country, whether **built** there or imported. Table 1 shows the status of the principal codes covering exchanger design. Where no code is shown for a country, the table shows the codes generally accepted. A more complete **list** covering **65** countries is given in [1]. In Sec. 4.3.2, the requirements of U.S., U.K., and German codes, as they affect **shell-and-tube** exchangers, are listed. Section 4.3.5 gives a brief summary of major national codes. The codes aim to achieve safe construction and give rules for design and fabrication, which are based on experience with conventional plant. However, when the components differ from conventional design or are outside the range of the code rules, or when more certain assurance of safety is required, it may be necessary to justify the design using alternative codes or a stress analysis as discussed in Sec. 4.1, or component testing. These alternatives must be agreeable to the purchaser and the inspecting authority.

Codes also provide a useful tool for the education of junior engineers or those entering the vessel and exchanger field, and this aspect should not be overlooked when new codes or rules are being formulated. As well as giving rules for design and fabrication, most codes are specific as to acceptable materials, but usually the range of materials is wide enough to allow the designer a

choice. The choice can be made taking into account the following requirements.

Requirements	Source of information
Processes:	
Material compatibility	Operating experience
Corrosion resistance	Suppliers and operating experience
Physical properties	Suppliers and test data
Materials :	
Yield stress and UTS	Code data
Elongation and reduction of area at fracture	Code data
Toughness	Code data
Aging and embrittlement	Code data
Creep strength	Code and test data
Fatigue strength	Code and test data
Commercial requirements:	
Availability	
cost	
Fabrication techniques	

Once the designer has made a choice of material, the design code gives an allowable design stress that is used to dimension the pressurized components, and the code specifies any special fabrication requirements for that material, such as heat treatment, for example. Although codes do list acceptable materials, they also permit other materials.

(a) ASME VIII

The listed materials are taken from specifications of the ASTM. In order to use other materials for **ASME-coded**

Table 1 Principal pressure vessel codes

Country	National code	Approving organization	Regulations	Other acceptable codes
Belgium	—	Ministere de L'Emploi et du Travail	Police Regulations for Dangerous Establishments	Any that satisfy legal requirements
Czechoslovakia	CSN Standards	Cesky Urad Bezpecnosti Prace		Subject to negotiation
Denmark	—	Arbejdstilsynet Direktoratet	Regulations for land-based PVs 22/12/71	BS, ASME , Merkblatter, Swedish
France	SNCT	Ministry of Scientific and Industrial Development	Government Publication 1332	—
F. R. Germany	A. D. Merkblatter	TUV	State insurance codes and VBG17	—
Italy	ANCC	ANCC	Presidential Decree 547 27/4/55	—
Japan	JIS B 8243	Ministry of Labor Safety	Ordinance 33 30/9/72	ASME
Hungary	—	Energiagazdalkadasi Intezet	PV Safety Regulations	BS, Merkblatter
Netherlands	Grondslagen and Rules for PVs	Dienst voor het Stoomwezen	—	—
Norway	General rules for PVs	Statens Arbeidstilsyn	Act of Protection of Workers	
Poland	Technical Supervision Code for Stationary PVs	Urzad Dozoru Technicznego	Technical Supervision Code for Stationary PVs	BS
Spain	—	Ministry of Industry	PV Regulations 1969, amended 1972	—
Sweden	Swedish PV Code	Ångpanneforeningen	Industrial Safety Act	—
United Kingdom	BS standards	Factory Inspectorate	Factories Act 196 1	Any that meet legal requirements
United States	ASME	Varies for each state	Varies for each state	—
U.S.S.R.	GOST standards	Gosgortekhnadzor	Gosgortekhnadzor	By negotiation

vessels, special application must be made to, and approval secured from, the **ASME** Code Committee.

(b) BS 5500

Materials other than those listed in the code may be used by agreement between purchaser and manufacturer provided that they are covered by a written specification as comprehensive as the BS specification for the equivalent material and that the design stresses are determined in a manner consistent to BS.

(c) A. D. Merkblatter

The A. D. Merkblatter W series of specifications lists acceptable materials, but others may be authorized with the agreement of the inspecting authority. In the latter case the W specifications give requirements that must be satisfied.

B. ASME Boiler and Pressure Vessel Code Section VIII, Div. 1

This code [2] gives minimum requirements for the design, fabrication, inspection, and certification of vessels with design pressures between 1.03 bar g (15 psig) and

206 bar g (3 000 psig). Where the design needs to be justified by a full stress analysis, Div. 2 of this code should be used. New editions of the code are usually issued every 3 yr, but interim revisions are made twice yearly in the form of addenda. The **ASME** code references used in Sec. 4.3 refer to the 1977 edition.

C. TEMA: Standards of Tubular Exchanger Manufacturers Association

These standards [3] serve to supplement and define the **ASME** Pressure Vessel Code for all shell-and-tube exchanger applications (double-pipe exchangers are not included). Although TEMA is linked specifically to **ASME** VIII, it is a useful standard that can be used to supplement other national codes. Recommendations for construction are given in three classes, the class being specified by the purchaser. The design rules for each class are the same, the only difference being in dimensions and details of construction. Class R is for the generally severe requirements of petroleum and related processing applications. Class C is for the generally moderate requirements of commercial and general process service. Class B is for chemical process service.

The numbering system is common to all classes, and the TEMA references used in Sec. 4.3 refer to class

R, C, or B of the 1978 edition. This edition carries for the first time a section entitled **Recommended Good Practice relating to aspects not covered in the** main sections of the standard, particularly the requirements for exchangers with shell diameters from 1 524 to 2 540 mm (60 to 100 in). The numbering system used is the same as in the three main sections of TEMA, and an * is used throughout Sec. 4.3 to denote that there is an additional recommended good practice.

D. BS 5500: The British Standard Specification for Unfired Fusion-Welded Pressure vessels

This recently introduced code [4] replaces BS 1500 and BS 1515 and is intended to unify the U.K. requirements for **all** pressure vessels. A major departure for this code is that the purchaser now specifies the construction category (BS 5500 3.4), which then defines the amount of nondestructive testing and restricts the permitted materials. The degree of testing is not now reflected by the use of design stress-reduction factors. The code references used in Sec. 4.3 refer to the third issue of BS 5500.

E. A. D. Merkblatter: German Pressure Vessel Regulations

These regulations [5] are in the form of data sheets covering different aspects of vessel design and construction, and are produced by a group of associations. Revisions are made from time to time to keep up with

advances in technical knowledge. Some aspects of vessel and exchanger design are not covered, and the method is agreed upon by the purchaser, inspecting authority, and designer. The code references used in Sec. 4.3 refer to the 1977 edition of A. D. Merkblatter.

F. International codes

The International Standards Organisation (ISO) has been endeavoring for some time to write an international code for pressure vessels. Various working groups have written draft sections of the code, and these were submitted in late 1973 to form a proposed draft standard DIS 2694. This draft was considered by the subscribing countries to ISO, but when votes were cast in 1974 many major countries voted against its acceptance. The draft was then sent back to the organizing secretariat and working groups with comments, but there has been no action to date.

In Europe, the EEC has published a Council Directive of July 27, 1976, giving a general framework to common provisions for vessels and methods of inspecting them. The intention is that separate directives for different categories of vessels will lay down the technical requirements for design and methods of inspection, and thus will achieve free movement of vessels within the EEC and avoid multiple inspections.

European standards will be organized by CEN (Comité Européen de Normalisation), and a draft for simple pressure vessels has been written. Unfired pressure vessels are being considered by CEN working group 54, but the EEC Directive is awaited.

Bibliography for Section 4.3.1 appears at the end of Section 4.3.5.

4.3.2

Index to U.S., U.K.,
and F.R.G. codes

M. Morris

■ Figure 1 gives the following identification letters for the components to be discussed in detail here.

- A Cylindrical shell and channel
- B Tubes
- C Conical shell
- D Dished head
- E Flat head
- F Floating-head components
- G Girth flange
- H Supports

- I Tubesheet
- J Nozzle
- K Shell bellows

In the following, reference numbers are those in the relevant codes of the following editions:

- | | |
|-------------------|--------------|
| ASME VIII Div. 1 | 1977 Edition |
| BS 5.500 | Third issue |
| TEMA | 1978 Edition |
| A. D. Merkblatter | 1977 Edition |



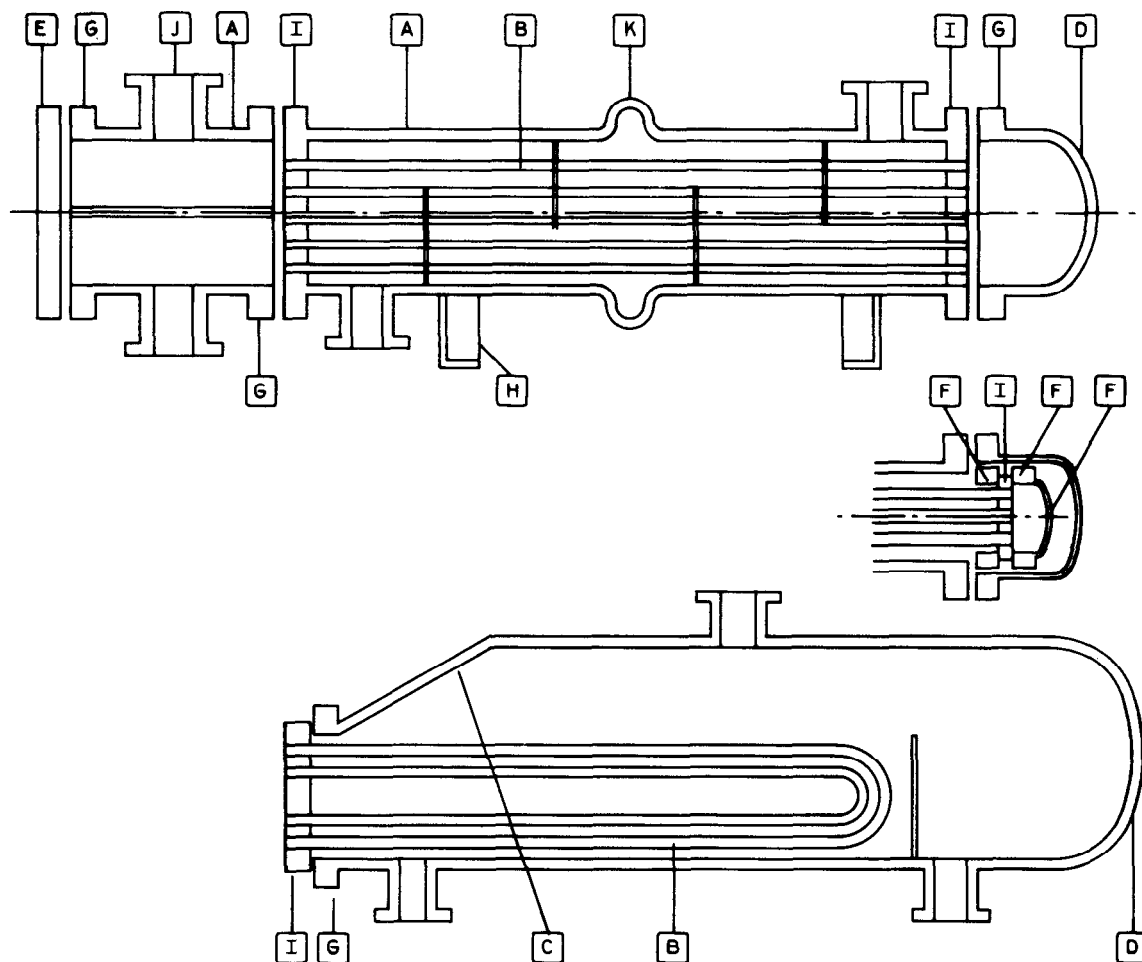
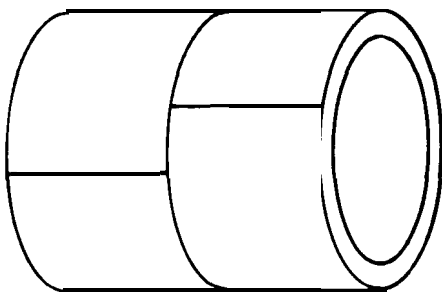


Figure 1 Identification letters for exchanger components.

A. Cylindrical shell and channel



GENERAL

Internal pressure: Usually the hoop stress governs, although for **fixed** tubesheet exchangers the longitudinal stress is also checked.

External pressure: Vacuum conditions are generally the only external pressure applied to exchanger shells.

TEMA

Minimum fabricated thickness 3.13*
Longitudinal stress (**fixed** tubesheet

exchanger), calculated from tube-sheet conditions and takes the place of pressure longitudinal shell stress

7.2

Postweld heat treatment of CS channel 8.14

BS 5500

Internal pressure thickness

$$e = pD_i / (2f - p) \quad 3.5.1.2$$

Vacuum check :

3.6

Calculate $L/2R$ and $2R/e$

Find ϵ

Fig. 3.6(2)

Calculate $\alpha = 1.28 \sqrt{Re}$

Find G

Table 3.6(2)

Calculate γ

3.6.1.1

$S = 1.1$ for austenitic steels

$= 1.4$ for other steels

Hence calculate K

Eq. (11)

Find A

Fig. 3.6(3)

Hence allowable external pressure

Eq. (12)

Longitudinal stress check for **fixed** tubesheet design (see Sec. I)

4.3.3

Analytic basis of code rules

M. Morris

■ A study of Sec. 4.3.2, together with one of the pressure vessel codes, will show that the rules given to size a particular component have an analytic basis. These rules have generally been adapted as a result of experience over the years, and the analytic background is sometimes hidden. The following paragraphs aim to give more information on this background and where possible relate to Sec. 4.1 of this handbook.

A. Cylindrical shell

For internal pressure, the shell thickness is calculated from the hoop stress as given in Eq. 4.1.3(1). The equation is modified so that either the internal or external cylindrical radius can be used. For external pressure (or vacuum) the traditional analysis is referenced in Sec. 4.1.5. A more recent method is used by BS 5500 and is discussed in [6].

B. Tubes

Tubes in fixed tubesheet exchangers are subjected to end loads as well as internal and external pressure. Longitudinal tensile stresses are treated the same way as pressure tensile stresses, but longitudinal compressive stresses may cause the tube to buckle as a column (see Sec. 4.1.5).

C. Conical shell

For small cone angles, the thickness formula is the same as cylindrical shells, but with a $\cos \alpha$ term incorporated. For larger angles, toroidal transitions are needed at the

ends of the cone to absorb the discontinuity bending stresses and a full analysis may be needed. See [11,12] and Sec. 4.1.3C.

D. Dished head

The analysis of hemispherical heads is straightforward, and all code rules are based on Eq. 4.1.3(4). For ellipsoidal and torispherical heads, the analysis is more complex, and in recent years, experimental and theoretical studies have examined the local stresses existing throughout these heads. Although the presentation of rules differs for the various codes, the methods are basically the same and the theoretical background is given in Sec. 4.1.8A.

E. Flat head

With simple edge conditions, a flat plate under pressure is a straightforward bending component. The junction with the cylindrical shell is either welded or flanged, and this junction disturbs the simple stress distribution. The effect of a welded junction is shown in Sec. 4.1.8A, and the resultant stress-concentration factor is accommodated in the code rules by a factor modifying the basic flat-plate formula. With bolted heads an edge moment is added by the adjacent flange bolting and a modifying factor is again used.

F. Floating-head components

The code rules referenced in Sec. 4.3.2F give formulas for the thickness of the dished head, the flange ring, and

the backing ring. However, the formulas are approximate as they do not take into account the continuity between the flange ring and the dished head. A more exact analysis is generally permitted, and this is provided by [19] for both shell-side and tube-side pressure.

G. Girth flange

As discussed in Sec. 4.1.7, the analysis of flanges is complex and depends on the interaction of flange, adjacent shell, bolts, and gasket. The classical paper on flange design [20] is based on elastic analysis, and the method has become known as the Taylor Forge method and is used in ASME and BS 5500.

The shortcomings of an elastic analysis are given in Sec. 4.1.7, and the point is made that flange design satisfactory to code rules is dependent on the input of practical experience. A flange can be said to have failed if it allows leakage from the joint, which can be caused by plastic conditions in the flange, yielding of the bolts, or incorrect gasket specification. With small-diameter flanges there is usually enough margin in the gasket to absorb the resulting rotation without leakage. As the diameter increases, however, this margin is reduced and the correct choice and design of flange components becomes more critical. It is suggested in Sec. 4.3.2G that below 20 bar pressure and 1.5 m diameter, the classical analysis should give acceptable results. Above these criteria, further analysis may be needed. Reference [22] discusses such an analysis, [23] gives the results of gasket testing, and [24] evaluates the interaction of all the flange components.

H. Supports

A full discussion of the analytic basis of the BS 5500 design rules is given in Sec. 4.1.2C.

I. Tubesheets

Tubesheets are the most complex exchanger component, and 15 variables can be listed that affect the loading. TEMA in 1941 first gave rules for the design of U and floating-head tubesheets based on modified flat-plate formulas. Although empirical, these formulas gave satisfactory results, and a form of them is still used in current TEMA rules. An analytic approach for U and floating-head tubesheets was provided by Gardner [30] by treating the tubesheet as a modified solid plate as discussed in Sec. 4.1.4. This method was developed simultaneously for fixed tubesheet exchangers by Gardner [31] and Miller [33] and results in equations given in Sec. 4.1.4C. BS 1500 adopted this approach in

1958, and the 1968 edition of TEMA included rules for fixed tubesheets broadly based on Gardner and Miller but with some improvements. Thinner fixed tubesheets for shell boilers had been obtained for several years, as the semiempirical method split the tubesheet into smaller areas bounded by supports, i.e., tie bars, flue pipe, etc. This method is used by BS 2790 and ASME I and gives satisfactory results for flue gas boilers when the difference in metal temperatures between tubes and shell is small. A similar method is used in current A. D. Merkblatter regulations for fixed tubesheets, and the effect of this is discussed in Sec. 4.3.4. In a later paper Gardner [35] derived a fuller analysis for U-tube and floating-head tubesheets, and this approach is now used by BS 5500. The basic formulas are unchanged but no longer based on arbitrary constants. The parameters encompassed are discussed by Morris [37].

J. Nozzles

For nozzle design, the principal codes illustrate three approaches. The traditional method of nozzle reinforcement by area replacement is used by ASME VIII Div. 1, in which the cross-sectional area of the reinforcement equals the cross-sectional area of the metal removed from the shell. This reinforcement is added as a pad, a thickened branch, or a thickened shell.

The design curves of BS 5500 are based on the need to avoid incremental plastic strain during repeated pressure loading of the vessel. A limited amount of plastic deformation in local areas is accepted during initial operating cycles, but a residual stress distribution is established and the subsequent behavior is completely elastic. This shakedown behavior is achieved by limiting the maximum stress in the nozzle. For branches in spherical shells this maximum is set at 2.25 times the allowable stress in the unreinforced shell. For branches in cylindrical shells the allowable stress concentration factor is calculated from the estimated shakedown factor for the nozzle geometry.

Limit analysis for nozzles is discussed in Sec. 4.1.8B, where the design curves of A. D. Merkblatter are compared.

K. Shell bellows

As seen from Sec. 4.3.2K, the most comprehensive rules are from the Italian regulations ANCC. These rules are for square convolutions but with toroidal corners and are thus a combination of the formulas given separately in Sec. 4.1.6 for square convolutions and toroidal convolutions.

Bibliography for Section 4.3.3 appears at the end of Section 4.3.5.

Cladding 3.3.3
Tolerances 4.2.4.2

ASME

Minimum thickness of shell UG16(b)
Internal pressure thickness
 $t = PR/(SE - 0.6P)$, UG27(c)
where E = weld joint efficiency Uw12
Vacuum check UG28
Tolerance on out-of-roundness UG80
Good practice regarding linings App. F
Clad-shell design thickness UCL23

A. D. MERKBLATTER

Internal pressure: Sec. B1

$$\text{Thickness} = \frac{D_a p}{20K/Sv + p} + C_1 + C_2, \quad 5.$$

where D_a = OD K = yield stress B0.6. S = safety factor B0.7. v = joint efficiency B0.8. C_1 = undertolerance allowance B0.9.1 C_2 = corrosion allowance B0.9.2

Minimum wall thickness 6.

Vacuum: Sec. B6

Check against elastic distortion 7.2

Check against plastic distortion 7.3

B. Tubes**GENERAL**

As well as both internal and external pressure loading, tubes are subjected to end loads (particularly in the case of fixed tubesheet exchangers), which affect the tube end fixing. In thickness calculations account should be taken of the minimum tolerances.

TEMA

Standard length 2.1

Standard diameter and thickness 2.2

Minimum thickness before forming

U bend 2.3.1

Longitudinal stress (fixed tubesheet) 7.23/4

Tube joint loads 7.25

Tube maintenance E.4

BS 5500

Internal pressure (as cylinder) 3.5.1.2

External pressure (as cylinder) 3.6.2.1
For fixed tubesheet exchangers check tube end load from tubesheet design (see Sec. I)

ASME

Thickness for internal and external pressure (as cylinder, Sec. A) UG31
Tube end fixing App. A
Maximum permissible end load depends on code stress, c.s.a., and type and testing of expanded or welded joint.

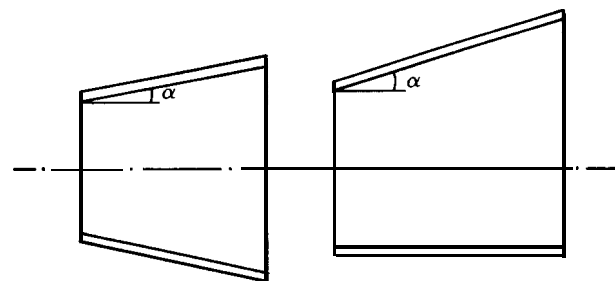
A. D. MERKBLATTER: Sec. B1.1

Internal pressure thickness

$$= \frac{D_a p}{20K/Sv + p} + C_1 + C_2 \quad 5.$$

where D_a = OD K = yield stress B0.6. S = safety factor B0.7. v = welded tube factor B0.8. C_1 = undertolerance allowance B0.9.1 C_2 = corrosion allowance B0.9.2

External pressure minimum thickness 6.1

C. Conical shell**GENERAL**

Commonly used for channels where nozzle diameters are large or pressure drop is critical. Cones also used in kettle shells. Half cone angle α usually limited to 30°.

TEMA

Minimum fabricated thickness 3.13*

Postweld heat treatment of C.S. channels 8.14

BS 5500

Internal pressure thickness $\alpha \leq 70$ 3.5.3.1.2.1Internal pressure thickness $\alpha > 70$ 3.5.3.1.2.2
(N.B.: α is usually 30°)

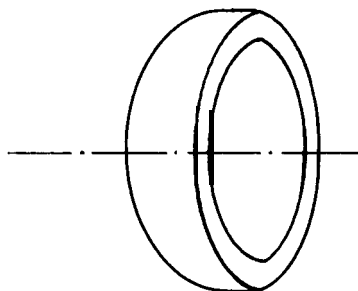
Check for large end reinforcement; if required, calculate	3.5.3.1.3 3.5.3.1.3.1/2
Check for small end reinforcement; if required, calculate	3.5.3.1.4 3.5.3.1.4.1/2
Vacuum check	3.6.2.6

ASME

General requirements	UG36(e)
Minimum thickness	UG16(b)
Internal pressure thickness, $\alpha \geq 30$	UG32(g)
Reinforcement required at conical junctions	UA5
Cones with $\alpha > 30$	UA5(e)
Vacuum check	UG33(f)

A. D. MERKBLATTER: Sec. B2

Knuckle thickness	8.2.1.1
Distance from knuckle at knuckle thickness	8.2.1.3
Thickness away from knuckle	8.3.1
Cone without knuckle ($\alpha \geq 30$)	8.2.2
Check thickness if $\alpha > 70$	8.4
Minimum thickness	9.
Vacuum check	8.5

D. Dished head**GENERAL**

Dished head channels are cheaper than those with bolted flat heads. However, channel nozzle piping must be detached to permit removal of channel for tubesheet inspection.

TEMA

Minimum fabricated thickness	3.13*
Bonnet inside depth for multipass channels	8.12
Postweld heat treatment of C.S. channels	8.14

BS 5500

Shape limitations	3.5.2.2
Unpierced end	3.5.2.3
Calculate overall depth h_e ; from p/f and h_e/D , calculate e	Fig. 3.5.2.3

Pierced end	3.5.2.4
Vacuum check	3.5.2.3/4/5
Tolerances	4.2.4.1

Note for torispherical heads

$$\frac{h_e}{D} = \frac{R_o}{D} - \sqrt{\left(\frac{R_o}{D} - 0.5\right)\left(\frac{R_o}{D} + 0.5 - \frac{2r_o}{D}\right)}$$

D = head OD

R_o = crown outer radius

r_o = knuckle outer radius

ASME

Minimum thickness	UG16(b)
Internal pressure thickness of half- spherical, 2: 1 ellipsoidal, and 6% torispherical	UG32
Internal pressure thickness of other ellipsoidal and torispherical heads	UA4
Tolerance on shape	UG8 1
Attachment welds to cylinder	UW13
Vacuum check	UG33

A. D. MERKBLATTER: Sec. B3

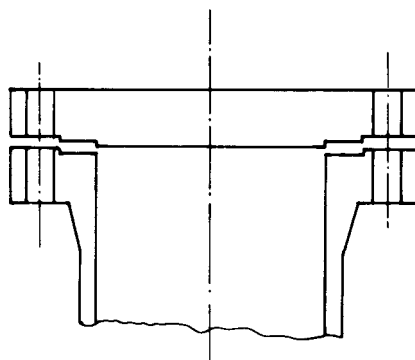
Shape requirements for hemispherical
and the following torispherical
heads (D , = OD)

1. & 2.

Klopper Korbogen

Crown internal radius = D_a = 0.80,
Knuckle internal radius = $0.1D_a$ = 0.1540,

Internal pressure thickness of crown	8.1
Internal pressure thickness of knuckle	8.2
Thickness head with opening	6.
Minimum thickness	9.
Vacuum check	4.3

E. Flat head**GENERAL**

Flat heads are commonly bolted to give access to tube-sheets without disturbing piping. Welded flat heads are sometimes used for low-pressure channels as an alternative to dished heads.

TEMA

Partition groove considered as corrosion allowance 1.514

Bolted channel cover thickness 8.21

This formula is different from the code formula, and the greater thickness should be used.

Depth of partition groove 8.22

BS 5500

Typical types of flat head Fig. 3.5.5

Thickness $e = CD\sqrt{p/f}$ 3.5.5.2.1

Bolted head, C from Fig. 3.5.5(2)(g)

Welded head, C from Fig. 3.5.5(1)

Compensation of opening 3.5.5.2.1

If opening is greater than one-half of shell diameter, calculate as flange (see Sec. G)

ASME

Acceptable types of flat heads Fig. UG34

Flat head thickness UG34(c)(2)

Calculated for both operating and gasket seating conditions. Dependent on edge fixing, pressure, and adjacent flange details

Thickness at edge gasket groove UG34(d)

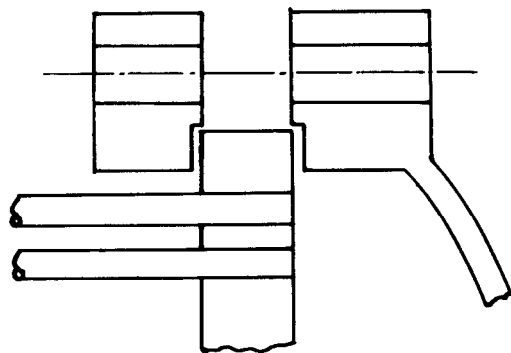
Reinforcement of opening UG39

A. D. MERKBLATTER: Sec. B5

Flat heads with openings 4.1

Thickness welded flat heads 6.1.1

Thickness flanged flat heads 6.3

F. Floating head

GENERAL

The floating head is composed of three components:

1. The floating head cover—a dished or flat head
2. The floating head flange—attached to the cover

3. The backing ring—usually split to allow withdrawal of the tube bundle

In a kettle exchanger the floating head withdraws with the tube bundle and the tubesheet is extended to bolt onto the floating-head flange.

TEMA

Minimum inside depth of floating-head covers 5.11

Postweld heat treatment of CS floating-head covers 5.12

Materials and corrosion allowance 5.13

Tube bundle support plate at floating end 5.14

BS 5500

Dished head thickness, internal pressure 3.5.6.2.1

Dished head thickness, external pressure 3.5.6.3

Flange ring bolting and gasket 3.5.6.2.2

Flange ring thickness 3.5.6.2.3

Formula basis as ASME

ASME

Dished head thickness, internal pressure UA6 and UG32

Dished head thickness, external pressure UG33

Flange ring thickness UA6

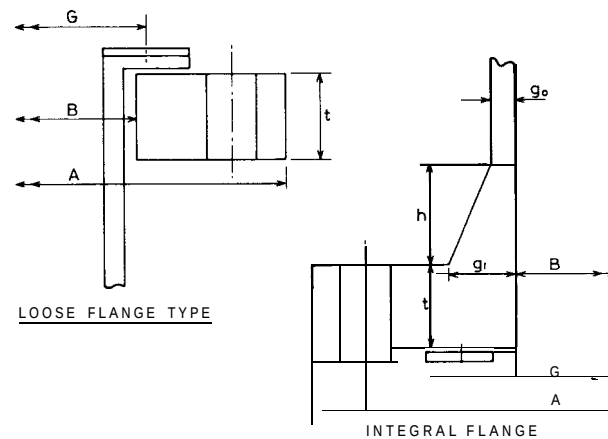
Split-backing-ring thickness UA53

The ASME formulas for dished head and flange ring are only approximate and do not take into account the interaction between head and ring.

A. D. MERKBLATTER

Dished head (in preparation) B4

Split-backing-ring thickness B8.6.11

G. Flange

GENERAL

The successful operation of flanges depends on the correct choice of three elements: the gasket, the bolting, and the flange shape.

Flanges are of two main types, those with full face gaskets, or more commonly those with narrow face gaskets within the bolt circle. The below details are limited to the latter type. Weld neck integral flanges are used for high-integrity flanges, whereas loose or welded ring flanges are used for lower pressures and temperatures.

TEMA

Type and materials of gaskets	6.
Recommendations for two gaskets with common bolting.	6.2
Gasket facing tolerance	6.32
Minimum bolt size	10.1
Maximum and minimum bolt spacing	10.2
Nut/wrench clearances	10.3
Large diameter, low-pressure flanges	10.5*
Bolting, assembly, and maintenance	10.6*
Recommended bolt-tightening procedure	E2.5

BS 5500

Method as ASME	3.8
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ASME

Selection of type of flange	UA48
Select gasket type, dimensions, and thus m, y, and G	Table UA49.1
Calculate gasket operating bolt load w_{m1}	UA49(b)(1)
Calculate gasket seating bolt load w_{m2}	UA49(b)(2)
Calculate bolt area required	UA49(c)
Select number, size, and pitch circle of bolting	TEMA10
Calculate operating flange moment $M_o = M_D + M_T + M_G$	UA47
Calculate gasket seating moment M_o	UA50
For integral flanges select hub shape	
Select g_i (initially, say, $1.5g_o$)	
Check nut clearances (on tubesheet as well)	TEMA10
Select h to give factor f (= 1.25, say)	Fig. UA5 1.6
Find flange shape factors	Figs. UA5 1.1 to .5
Calculate flange stresses for both operating and gasket seating conditions	UA5 1
Check allowable flange stresses	UA52
General requirements for studs	UG12
For large-diameter, high-pressure, high-temperature, etc., flanges read	App. S

Generally the ASME method gives acceptable results when pressure < 20 bar and diameter < 1.5 m

For flanges at trapped tubesheets with common bolting, gaskets with similar compression characteristics should be used and the bolting loads should be transferred

For $P_1 > P_2$

1. Design flange 1 first
2. Design flange 2 using w from flange 1
3. For consistency, on flange 2 take $H_G = W_{m1} (\text{flange 1}) - H(\text{flange 2})$

A. D. MERKBLATTER: Sec. B8

These rules are a simplification of DIN 2505, Calculation of Flanged Connections, and as difficulty can be found in selecting a hub shape to B8, it is recommended that design be carried out to DIN 2505.

Calculate pipe force PR	4.1
Calculate annular force PF	4.2
Hence pressure force PI	4.3
Select gasket	Table 1
Calculate gasket seating force PDV	4.4.1
Calculate gasket pressure force PDB	4.4.2
Thus bolt force PSO	4.5
Check bolt area to A. D. Merkblatter, Sec. B7	

However, the safety factors are low and overstressed bolts can result. Consideration can be given to using ASME/BS 5500 bolt stresses.

Calculate bolting moment M_o	5.1
Calculate operating moment M_1	5.1
Calculate flange resistance w for weld-neck flanges at Secs. AA, BB, CC	5.2.1.1
For loose flanges	5.2.1.2
For slip-on flanges	5.2.1.4
Check strength conditions satisfied	5.3
Note for weld-neck flanges at Sec. BB	

$$e = \frac{F_1 e_1 + F_2 e_2 + F_3 e_3}{F_1 + F_2 + F_3}$$

$$e_1 = 0.5h_F$$

$$e_2 = 0.333(2h_F + h_A)$$

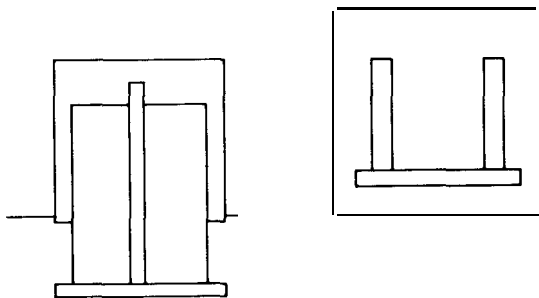
$$e_3 = 0.5(h_F + h_A)$$

$$F_1 = 0.5(d_A - d_i - 2d_L)$$

$$F_2 = 0.5(h_A - h_F)(S_F - S_R)$$

$$F_3 = S_R(h_A - h_F)$$

H. Supports



GENERAL

Supports for heat exchangers are usually of two types:

1. Saddle supports for horizontal units, one **fixed** and one sliding, with support angle usually ≥ 120
2. Support feet for vertical units

BS 5500 provides a comprehensive design check.

BS 5500

Saddle Supports G.3.3.2

Calculate longitudinal bending moment
at midspan G.3.3.2.1

Check longitudinal bending stress at
midspan G.3.3.2.2

Calculate longitudinal bending moment
at saddle G.3.3.2.1

Check longitudinal bending stress at
saddle G.3.3.2.3

Check shear stress G.3.3.2.4

Check circumferential stress G.3.3.2.5

Design saddle, width $\geq$ ID X shell
thickness G.3.3.2.8

(Wear plate, thickness $\geq$ shell thickness-unless
thick shell
overlap ≥ 6 above saddle
width ≥ 15 X shell thickness)

Bracket Support

1. Longitudinal moment G.2.3.2

Bracket size = $C_z (=6C_x)$ by $2C_\phi$ Fig. G.2(2 1)

Check if shape limitations are
satisfied G.2.3

Calculate opposing loads W G.2.3.2

Find $\frac{64r}{L} \frac{C_x}{\phi r}^2$ Fig. G.2(4)

Calculate stresses G.2.3.3(a)

2. Circumferential moment G.2.3.1

Bracket size = $C_\theta (=6C_\phi)$ by $2C_x$ Fig. G.2(20)

Check if shape limitations are
satisfied G.2.3

Calculate opposing loads W G.2.3.1

Find $\frac{64r}{L} \frac{C_x}{\phi r}^2$ Fig. G.2(4)

Calculate stresses G.2.3.3(b)

3. Radial load G.2.2

Bracket size = $2C_\phi \times 2C_x$ Fig. G.2(1)

Check if shape limitations are
satisfied G.2.2

Find $\frac{64r}{L} \frac{C_x}{\phi r}^2$ Fig. G.2(4)

Calculate stresses G.2.2.1.2

ASME

Good practice regarding supports App. G

Material for supports UG6(b)

Loads to be considered UG22

Fitment of supports UG82

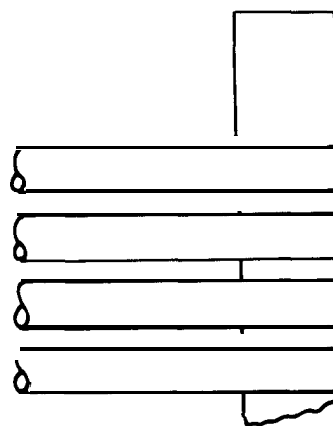
TEMA

No particular requirements

A. D. MERKBLATTER

No particular requirements

I. Tubesheets



TEMA-GENERAL REQUIREMENTS

Effective thickness of tubesheet 7.12

Clad tubesheets-7.122 and 7.9

Tubehole clearances 7.4

Expanded tube joints 7.5

Welded tube joints 7.6

TEMA-U TUBESHEETS

$$T = \frac{FG}{2} \sqrt{\frac{P}{S}} + \text{corrosion} + \text{groove} \quad 7.132$$

$F = 1.25$ for **gasketed** tubesheet; 7.142
for integral tubesheet see 7.151

G = gasket diameter or shell ID

P = maximum differential pressure

S = code design stress (**ASME**)

If $P >$ approximately $0.1S$, check shear 7.133

TEMA-FLOATING-HEAD TUBESHEETS

$$T = \frac{FG}{2} \sqrt{\frac{P}{S}} + \text{corrosion} + \text{groove} \quad 7.132$$

$F = 1.0$ for **gasketed** tubesheets 7.141

G = gasket diameter

P = maximum differential pressure

S = code design stress (**ASME**)

If $P >$ approximately $0.07S$, check shear 7.133

Make both tubesheets same thickness.

GENERAL

Tubesheets have the greatest number of design variables and thus the most complex design rules.

ASME

For tubesheet extended for edge bolting, the thickness at the periphery can be checked as for the outer edge of a flat plate, Fig. (j) **UG34(d)**

Attachment weld between shell and tubesheet **UW13(e)**

Tube end **fixing**, the maximum end load is dependent on code stress,

c.s.a., and type and testing of expanded or welded joint **App. A**

BS 5500—GENERAL REQUIREMENTS

Basis of rules 3.9

Tubesheet-to-shell weld Figs. **E2(39)** to (44)

Tube-to-tubesheet weld detail **E2.10**

BS 5500-U TUBESHEET

$$e = CD_o \sqrt{\frac{P}{\Omega \mu f}} + \text{corrosion} + \text{groove} \quad 3.9.3.1$$

C = design factor = $C_o + AC$ Table 3.9

D_o = diameter of outer tube limit

p = maximum differential pressure **3.9.2.3**

Ω = stress factor = 2 **3.9.2.1**

μ = ligament efficiency

f = code design stress

If $p >$ approximately $0.05f$, check shear **3.9.3.1**

BS 5500—FLOATING-HEAD TUBESHEETS

$$e = CD_o \sqrt{\frac{P}{\Omega \mu f}} + \text{corrosion} + \text{groove} \quad 3.9.3.1$$

C = design factor = $C_o + C$ Fig. **3.9(2)**

D_o = diameter of outer tube limit

p = maximum differential pressure

Ω = stress factor = 2

μ = ligament efficiency

f = code design stress

If $p >$ approximately $0.05f$, check shear **3.9.3.1**

TEMA

FIXED TUBESHEETS

BS 5500

Both TEMA and BS 5500 are basically the same.

Without bellows: $J = 1.0$

7.193		Calculate shell-side pressure factor f_s	3.9.4.3.1
7.194		Calculate tube-side pressure factor f_t	3.9.4.3.2
7.141	}	{ Shell-side support factor F Tube-side support factor F }	{ 3.9.4.2 Fig. 3.9(1) }
Fig. 7.151			
		$F = 0.5 (F_{\text{tube-side}} + F_{\text{shell-side}})$	3.9.4.2
7.191		Calculate strain factor K	3.9.4.2
7.192	P_{Bt}	Calculate effective bolting pressure	P_{B1} 3.9.4.3.5
7.192	P_{Bs}	Calculate effective bolting pressure	P_{B2} 3.9.4.3.5
	T	Assume thickness	e
7.191		Calculate F_q Hence $(1 + JK F_q)$	3.9.4.2
7.193	P'_s	Calculate effective shell pressure	P'_s 3.9.4.3.1
7.194	P'_t	Calculate effective tube-side pressure	P'_t 3.9.4.3.2
7.191	P_d	Calculate differential expansion pressure	P_e 3.9.4.3.3
7.193	P	Hence effective shell-side pressure	P'_1 3.9.4.3.1
7.194	P	and effective tube-side pressure	P'_2 3.9.4.3.2
7.132	T	Check bending thickness against assumed (to within $\pm 1.5\%$)	e 3.9.4.2
7.133		If pressure is high, check shear	3.9.4.2

TEMA	FIXED TUBESHEETS	BS 5500
	<i>With bellows:</i> If spring rate is known, check if light-gauge bellows, i.e., $J = 0$ (if rate not known, assume $J = 0$) and use simplified	
7.194	P pressure formula	p'_2 3.9.4.3
7.132	T Hence thickness	e 3.9.4.2
7.191	If heavy-gauge bellows, calculate J and proceed as above for case without bellows, but using actual value of J instead of $J = 1.0$.	3.9.1
7.22*	Shell longitudinal stress	3.9.4.4
	If greater than design stress, then bellows are required	
7.23*	Tube longitudinal stress	3.9.4.4
	Check maximum allowable buckling stress	3.9.5
	Check maximum allowable tube joint load to ASME VIII App. A.	
	Hanged extension: See ASME UG.34(d).	

A. D. MERKBLATTER: Sec. B5

Basic thickness formula $s = CD \sqrt{\frac{pS}{10Kv}}$

where C = loading constant

D = diameter

p = highest design stress

K = tubesheet yield stress

S = safety factor

v = joint (ligament) efficiency

B0.6.

B0.7.

Eq. (17), (18)

6.7.1

Fixed tubesheets

c = 0.40

D = largest inscribed diameter (d_z)

Fig. 10

Note $v = 1.0$

This formula is for pressure loading only, differential thermal loading check is by agreement

Expanded tube end load check

6.7.1.2

Welded tube joint check

6.7.1.3

Tube buckling check

6.7.1.5

U tubesheet

6.7.2

1. Fully tubed C from

Table 1

or Fig. 5

D shell diameter from

Table 1

2. Partially tubed C from

Fig. 14

D shell diameter from

Table 1

Floating-head tubesheets

6.7.4

C from

Fig. 16

D shell or floating-head diameter

Fig. 15

Fixed-tubesheet exchanger with shell**bellows**

6.7.5

C from

Fig. 16

D shell diameter

p from

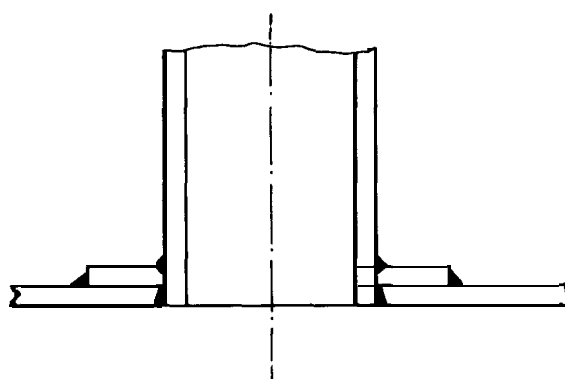
Eq. (26)

Tube end load and joint check

6.7.5.2/3

Flanged tubesheet extension

6.7.7

J. Nozzles**GENERAL**

Code pressure rules for nozzles are of two types. Either the area replacement method, such as ASME, where the area removed from the nozzle cut out is replaced as an external pad, etc., or the method where design curves enable thickened shell or nozzle to be calculated to limit the S.C.F. BS 5500 is the only code giving design rules for external loads on nozzles.

TEMA

General requirements	9.
Vent and drain connections	9.31
Pressure and temperature connections	9.3213
Split flange design	9.5
Nozzle loadings	9.6*

A. D. MERKBLATTER: Sec. B9

Shape limitations	1.1
Area replacement method	4.1

Shell weakening method	
Assume total shell thickness s_A	
Assume total nozzle thickness s_S	
Calculate weakening factor v_A	Fig. 7. or 8.
For increased shell thickness	4.2
For pad reinforcement	4.3
For increased nozzle thickness	4.4
Multiply material strength value	
K by v_A and calculate shell thickness to B_1, B_2 , or B_3 , etc.	
If different from assumed value, iterate to solution	
Multiple openings	4.6

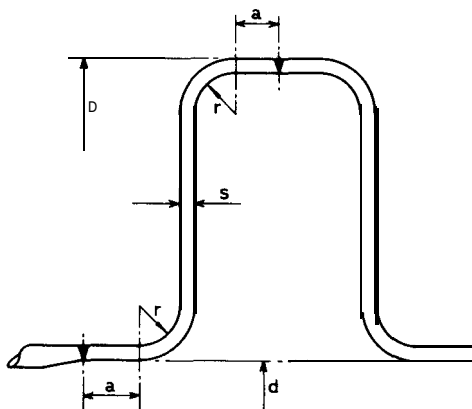
ASME

General requirements	UG36
Maximum nozzle diameter in cylinder using compensation rules is one-half of shell diameter up to 1 524 mm (nozzle not to exceed 508 mm) and one-third of shell diameter over 1 524 mm (nozzle not to exceed 1 016 mm).	UG36(b)(1)
Maximum nozzle diameter in heads is one-half of shell diameter	UG36(b)(2)
Recommendations for large nozzles	UA7
Large nozzles require proof test	UG101
Reinforcement area required	UG37(b)
Limits of reinforcement	UG40
Reinforcement of multiple openings (i.e., pitch < 2 X average diameter)	UG42
Minimum nozzle neck thickness	UG45
Requirements for inspection openings	UG46
Openings and or near welds	UW14
Nozzle attachment welds	UW16
Fitment of reinforcement pads	UG82
Worked example	UA280
Good practice regarding piping reactions	App. G

BS 5500

Calculate branch pressure thickness (see Sec. 4.3.2A)	
Check minimum thickness	3.5.4.7(b)
Choose branch reinforced thick = t_r	
Basic shell thickness = T	
Check value of C	3.5.4.3.1
Choose shell reinforced thick = T_r	
Calculate $\sqrt{\frac{D}{2T_r}}$	
Find t_r/T_r from appropriate	Fig. 3.5.4
Check against assumed t_r	
Iterate to acceptable t_r	
Width of reinforced sections	3.5.4.3.4

Permissible conditions for reinforcing pad	3.5.4.5
Area replacement method, which is acceptable up to $d/D = \frac{1}{3}$	App. F
See code case 5500/2	
Weld details	E2.6/7/8
External loads	
1. Cylindrical shells	G2.3
Use procedure given in Sec. 4.3.2H for long, circumferential moments and radial loads with $C_x = C_\phi$ = nozzle outer radius. Note method is applicable for $d/D \leq \frac{1}{4}$ if shell radius/thickness ≤ 50 . For thinner shell check	Fig. G2(0)
Calculated total stresses are categorized as primary, local, or secondary stresses depending on external load, see	App. A
2. Spherical shells	
Two methods are given G2.5 and G2.6. The latter is based on shake-down criteria and is more accurate.	
If the former method is used, assess the calculated stresses to	A3.3(f)

K. Shell Bellows**GENERAL**

For fixed tubesheet exchangers when the difference between shell and tube metal temperatures becomes large (Δ approximately 50° C for carbon steel), the tubesheet thickness and tube end loads become excessive. Shell bellows absorb the expansion and are of two types.

1. Thin-wall bellows of stainless steel or high-alloy material (see Sec. I for definition).
2. Thick-wall bellows of the same material as the shell and approximately the same thickness.

If thin-wall bellows are used, they should be protected against damage during fabrication, erection, and operation, with a shroud.

If more than two convolutions are used, consideration should be given to methods of equalizing the axial deflection.

Thick-wall bellows are often of welded construction; it is recommended that the straight portion $a \geq 2s$.

A. D. Merkblatter and the Italian Code ANCC give rules that are summarized below. ANCC is generally more conservative.

ANCC VSR1P

Maximum of three convolutions 1.P.2.1

For carbon and low-alloy steels: 1.P.2.1

Maximum thickness = shell thickness, minimum thickness = 4 mm, and $r \geq 3.5s$.

For austenitic steels: 2.P.2

Minimum thickness = 2mm and $r \geq 4.5s$

Thickness is greater of: 1.P.3.2

$$s = \frac{D - d - 3r}{40} \sqrt{\frac{p}{f}}$$

$$s = \frac{p}{100f} \frac{r(D + d)}{(2.3r + D - d)}$$

Permitted expansion per convolution 1.P.3.3

$$w = K \frac{f}{E} \frac{D^2}{s}$$

E = Young's modulus of bellows

$$K = 1.25 \left\{ \frac{x + 1}{(x - y)^2} - \frac{1}{x + 1} \right. \\ \left. \times \left[\frac{2(1 + y)}{-2y - 1} \ln \frac{x - y}{1 + y} \right]^2 \right\}$$

$$x = \frac{D}{d} \quad \text{and} \quad y = \frac{r}{d}$$

Vents and drains are permitted within the cylindrical portion of the bellows 1.P.4.3

A. D. MERKBLATTER: Sec. B13

General requirements 2.

Thickness is greater of

Thickness for bending 4.1.2

Thickness for circumferential load 4.1.3

Bibliography for Section 4.3.2 appears at the end of Section 4.3.5.

4.3.4

Comparison of principal codes

M. Morris

A. Stresses

For historical reasons, design stresses in the various codes are different for similar materials. The logic of the safety factors applied to the mechanical properties is similar and is discussed in Sec. 4.1.1, but the values of the safety factors are different.

Table 1 shows the current safety factors that are used to obtain the basic design stress for carbon and low-alloy steels in the principal codes. The use of different design stresses will thus result in different component thicknesses even if the design rules are similar. In some cases, the rules are different and these themselves will result in different component thicknesses. As seen in Sec. 4.3.1A, the designer very rarely has a choice in the design code to be used, but it is nevertheless instructive to compare the different design rules. The major differences are found in the components discussed below.

B. Cylinders: External pressure

The most widely used expression for predicting buckling pressure in cylinders was derived by von Mises and later

corrected by Windenburg and Trilling. Forms of this expression are used by the major vessel codes, and any differences in the calculated allowable external pressure for simple cylinders will be due to the different safety factors and the different permitted tolerances. The deviation from a true circular shape can have a major effect: As a rough guide, if the deviation is equal to the cylinder thickness, then the critical buckling pressure will be about half that for the true cylinder; and if the deviation is one-tenth the thickness, the reduction will be about a quarter. The permitted deviations vary with the codes:

ASME: Depending on the ratios of length/diameter and diameter/thickness, the deviation can be between 30 and 100% of the thickness.

BS: The shell must be circular to within 0.5% on the radius, measured on the true circle.

Merkblatter: The rules permit a shell circular to 0.75% on the radius measured on the true circle. A formula is given to enable the calculation of external pressure if the deviation is different from 0.75%.

Table 1

	UTS at room temperature	Yield stress at design temperature	Creep rupture stress at design temperature in 100 000 h ^a	Creep strain 0.01% in 1000 h
A. D. Merkblatter	—	1.5	1.5	—
ASME	4	1.6	1.5 (on average stress)	1.0
BS 5500	2.35	1.5	1.3	—

^aNote: BS 5500 considers other design lifetimes.

The most comprehensive rules for stiffeners are given in BS 5500. Two types of stiffeners are considered—heavy stiffeners capable of retaining shell circularity and light stiffeners that cover all other types. The ASME rules for stiffeners are based on the assumption that the rings should be strong enough to resist collapse even after the shell between the stiffeners has buckled. ASME assumes that the shell contributes to the stiffness of the ring and includes it in the effective moment of inertia, whereas Merkblatter considers only the inertia of the stiffener.

C. Tubesheets

As discussed in Sec. 4.3.31, tubesheets may have up to 15 parameters affecting design, which makes a comparison between cases difficult. These parameters and their inclusion in the various codes are discussed by Morris [37]. The easiest comparison can be made by using TEMA, with the more classical approach, as a common basis.

(a) U tubesheets

Both Merkblatter and TEMA equations are similar and are based on the modified flat-plate formula as discussed in Sec. 4.3.31. TEMA has a single modifying factor for ligament efficiency and edge fixing, whereas Merkblatter includes ligament efficiency in the formula and has a factor for edge fixing. For the same pressure, diameter, and design stress, this results in a Merkblatter tubesheet 30% thicker than TEMA for 19-mm tubes on 25.4-mm pitch and 42% thicker for 19-mm tubes on 23.8-mm pitch. BS 5500 also includes ligament efficiency in the formula, and the modifying factor C is dependent on R , the ratio of the shell diameter to the outer tube limit. For the first time a design stress factor, Ω , is introduced. This has a value of 2 and multiplies the basic design stress. It is thus recognized that the calculated average surface bending stress across the ligament can have a higher allowable value. The comparison of thicknesses calculated to TEMA and BS 5500 is shown in Fig. 1 for an example of 19-mm tubes on 25.4-mm pitch.

(b) Floating tubesheets

The Merkblatter and TEMA equations are again similar with the basis as discussed for U tubesheets above. For the same pressure, diameter, and design stress, the equations give the same thicknesses for 19-mm tubes on 25.4-mm pitch, whereas Merkblatter is 12% thicker for 19-mm tubes on 23.8-mm pitch. BS 5500 includes more parameters in the equation and, as well as the factors mentioned above for U tubesheets, tube length, Young's moduli, and flexural efficiency are now con-

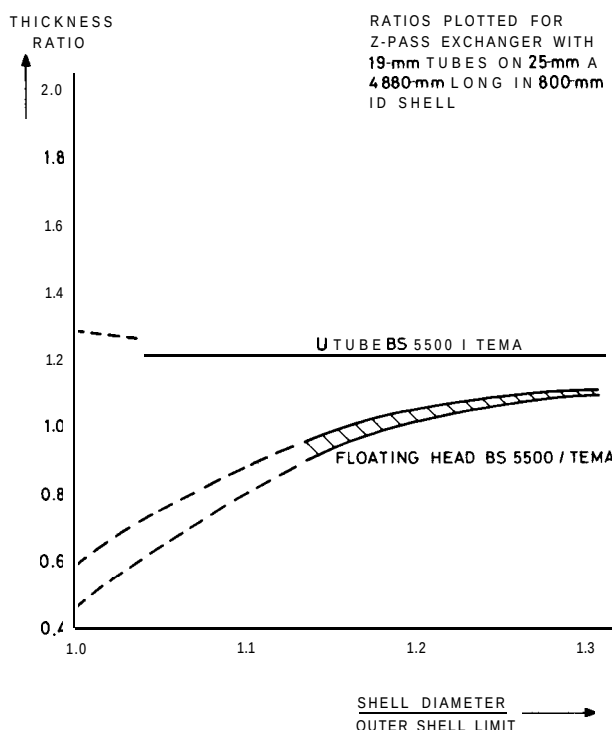


Figure 1 Tubesheet thickness ratios.

sidered. This makes a comparison between BS 5500 and TEMA difficult, but Fig. 1 illustrates an example of a typical exchanger.

(c) Fixed tubesheets

The rules for TEMA and BS 5500 are now basically the same and incorporate loading factors from both pressure and thermal differences between shell and tubes. Merkblatter adopts a different philosophy and generally obtains thinner tubesheets. The method calculates the flat-plate thickness for the largest circular area that can be drawn in the unpierced part of the tubesheet. Codes dealing with flue gas boilers such as BS 2790 and ASME I have used this method for some years and have given satisfactory results. In these boilers, however, the difference in metal temperatures between shell and tubes is usually less than 10°C, and the method should be used with caution for larger temperature differences.

D. Flanges

As stated in Sec. 4.3.2G, A. D. Merkblatter recommends that flanges should be calculated according to DIN 2505 and so a comparison is made between this standard and the ASME method (i.e., also BS 5500). The methods of calculating the applied moments are similar, but a difference is found in the gasket loading

factors. The difference in the operating condition load factor is a secondary effect, as this load is generally smaller than the pressure-end loads to which it is added. However, the bolting-up condition loads result solely from gasket loads and hence the difference can be large. DIN 2505 gives two cases, one for liquids and a higher gasket loading for gases. The DIN loadings are much higher for compressed asbestos fiber and for jacketed asbestos gaskets, whereas for spiral-wound gaskets the DIN loadings are much lower for liquids but higher for gases.

As mentioned in Sec. 4.3.2G, the bolt stresses are different for the codes, and this will result in different bolt areas. A. D. Merkblatter has much higher design stresses, and there is the danger of overstressing the bolts due to overtightening or thermal effects.

The method of analyzing the flange shape under the action of the applied moments is different for ASME and Merkblatter. The former has graphs for the flange shape factors based on the original Taylor Forge work and from these, stresses are calculated for the following sections.

Section	Stress	Allowable value
Hub (both ends)	Longitudinal bending	1.5 X design stress
Flange ring	Radial	Design stress
Flange ring	Hoop	Design stress

The higher allowable stress for the hub recognizes that the stress is a surface bending stress. DIN 2505 uses a moment-of-area method to calculate longitudinal bending stresses in the hub, and these are compared to normal design stresses.

E. Nozzles

The shape limitations of the codes have already been listed in Sec. 4.3.2J and will not be discussed further. A comparison of the reinforcement required by the codes is illustrated in Fig. 2, which shows the nozzle thicknesses, where the reinforcement is wholly in the nozzle, plotted against the nondimensional parameter $d/\sqrt{DT}$:

ASME: The area replacement method is easily calculated and is simple in approach. However, the peak stress level varies with geometry, and no consistent margin is provided for shakedown or fatigue. No compensation is required for 75 NB or 50 NB nozzles (depending on shell thickness), but proportionally large reinforcement is required for small openings as seen in Fig. 2, and this overstiffening could cause failure if associated with abnormalities. It can be seen from the

graph that at large values of $d/\sqrt{DT}$ the compensation is similar to other codes.

BS 5500: These rules are based on stress concentration factors and give calculations for all diameter ratios. Design charts are drawn so that the maximum SCF is limited to allow the structure to shakedown to elastic action. It is seen from Fig. 2 that two lines are drawn for BS 5500 and that extra reinforcement has to be provided when the d/D ratio is greater than 0.2.

A. D. Merkblatter: In this code, weakening factors are calculated and limited to allow the structure to shake down to elastic action. The weakening factors are calculated from the geometry and are made compatible with the membrane stress in the cylinder. The intention is to limit the permanent strain at the nozzle junction to 0.2%. In the last edition of the code the reinforcement requirements have been increased and are now consistent with other codes for large $d/\sqrt{DT}$ ratios.

The designer is given the choice of thickening either the nozzle or the shell for reinforcement, and at small d/D ratios the choice can be an economic one. At large d/D ratios, experimental data (admittedly scarce) would recommend thickening both to give approximately the same nozzle and shell thicknesses.

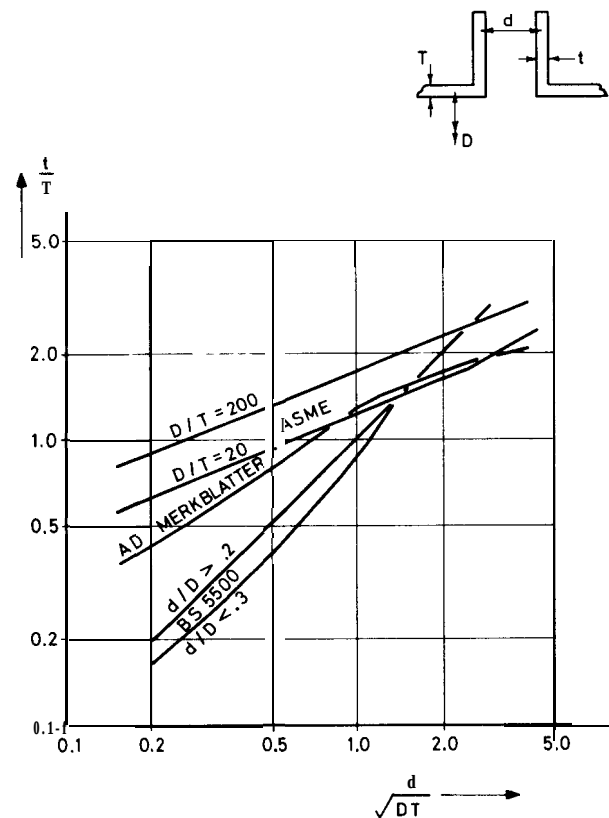


Figure 2 Nozzle reinforcement.

F. Flat heads

Where flanged flat heads are fitted to multipass channels the position plates are generally **gasketed** against the flat head. Distortion caused by pressure loads can cause

leakage between the passes. The TEMA formula for bolted flat head thickness should be considered, as the formula limits the displacement of the flat head rather than the resulting stresses.

Bibliography for Section 4.3.4 appears at the end of Section 4.3.5.

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I. Tubesheets

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J. Nozzles

See references in Sec. 4.1.

K. Shell bellows

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4.3.5

Guides to national practice

M. Morris

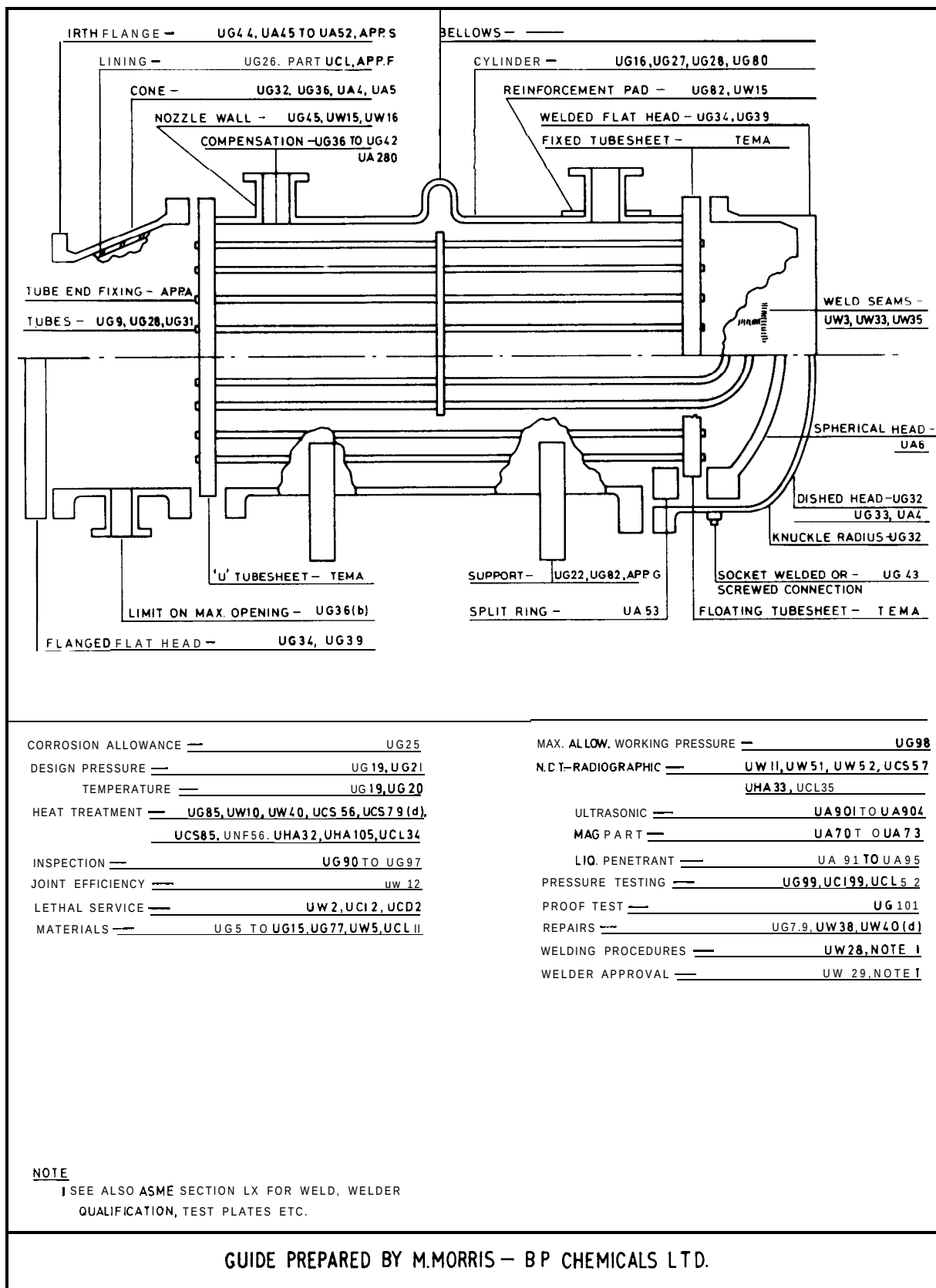
■ In this section guides to national practice are given for the following countries:

A	U.S.	F	Italian
B	British	G	Japanese
C	German (F.R.G.)		
D	Dutch		
E	French		

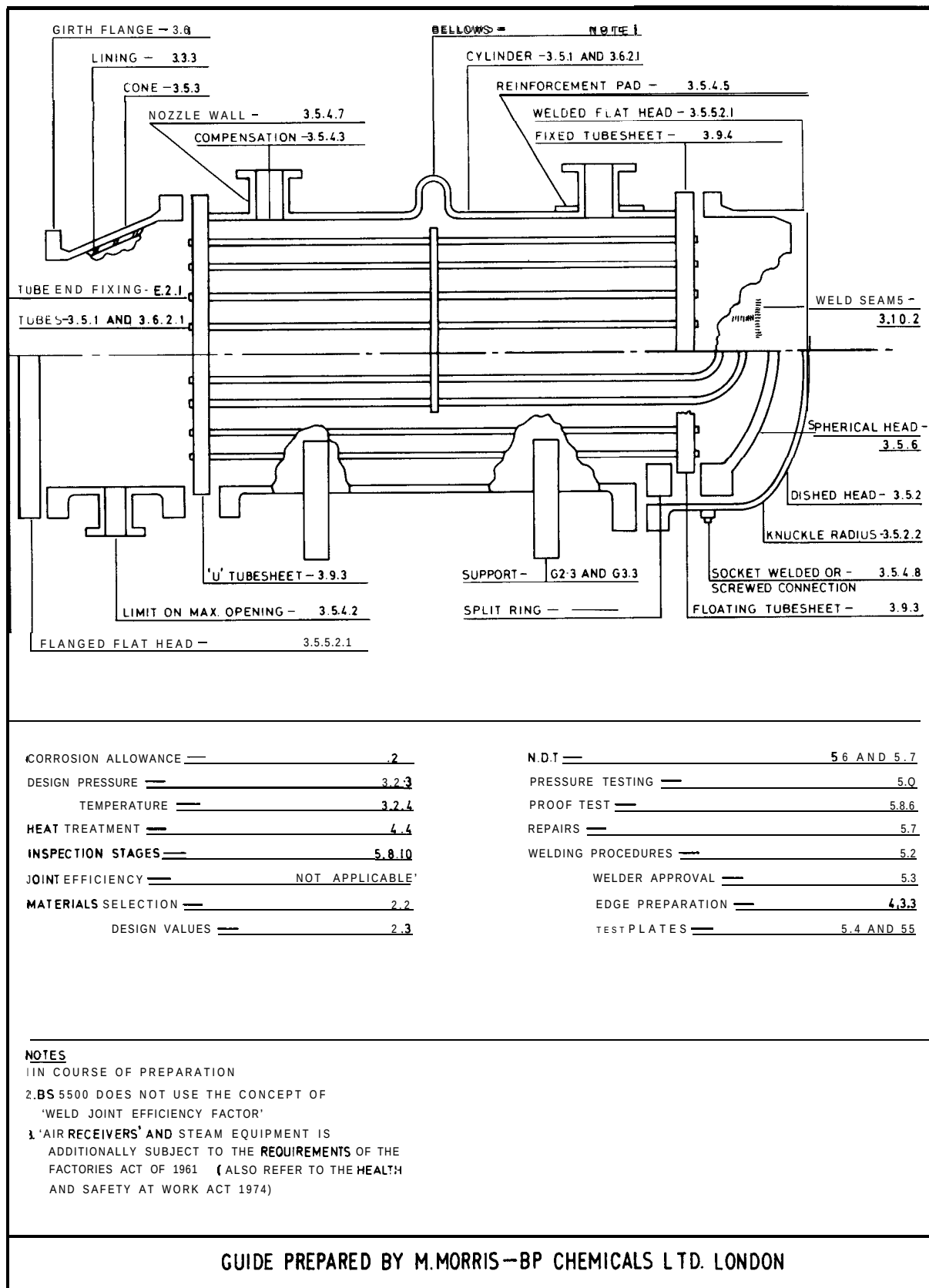
The guides are in the form of index diagrams to codes and standards. Reference numbers throughout this section refer to clauses in those codes or standards.

Guides are included for those countries already discussed in detail in Sec. 4.3.2. Some readers may find the guides sufficiently detailed for their purposes, and more convenient to use.

A. Guide to American vessel code—ASME VIII Div I

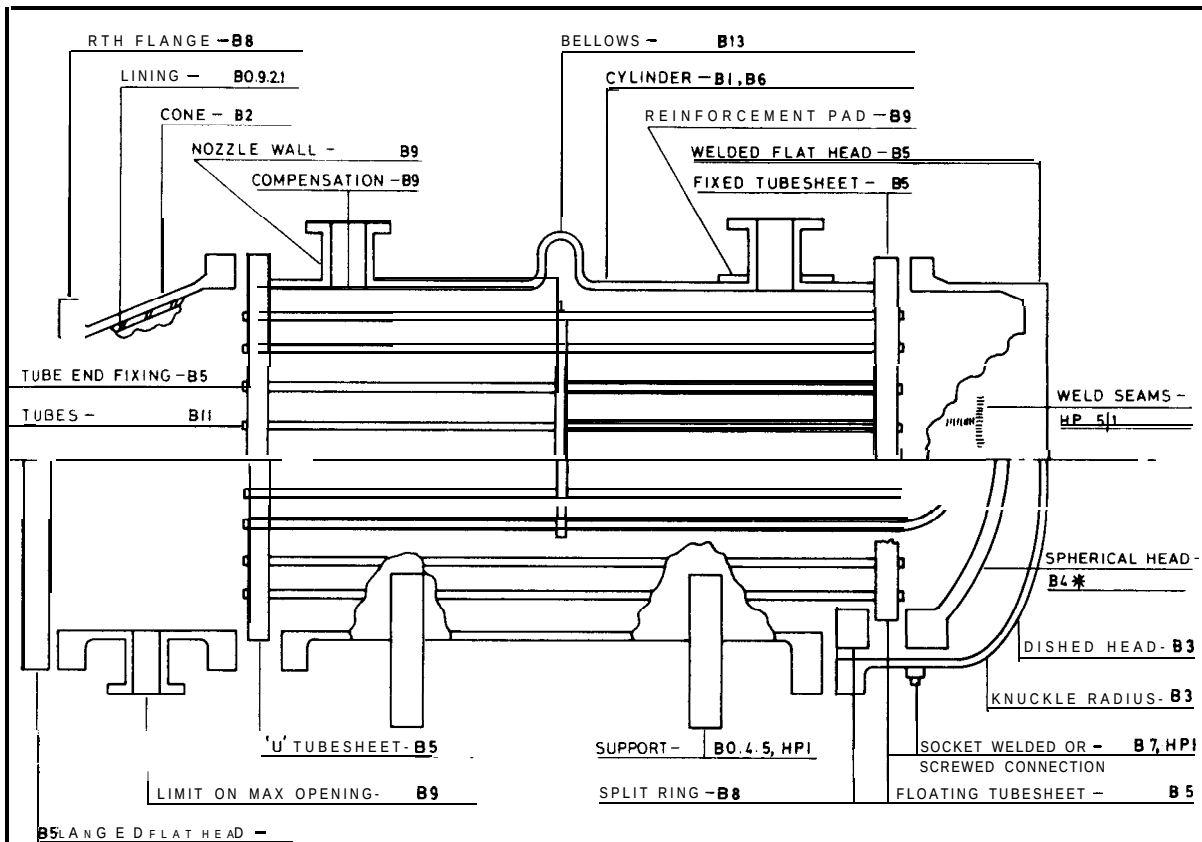


ASME PRESSURE VESSEL CODE IS PUBLISHED BY ASME, UNITED ENGINEERING CENTRE 345E, 47TH ST. NEW YORK NY 10017 USA

B. Guide to British vessel code-BS 5500

BS 5500 IS PUBLISHED BY THE BRITISH STANDARDS INSTITUTION 2 PARK ST., LONDON W1A 2BS

C. Guide to German vessel code-A. D. Merkblatter



CORROSION ALLOWANCE	60.9
DESIGN PRESSURE	B0.4
TEMPERATURE	B0.5
MATERIAL TREATMENT	HP 7/1, 7/2, 7/3
INSPECTION	BPO, BPI, BP2*
JOINT EFFICIENCY	B0.8
MATERIALS	W0 TO W13

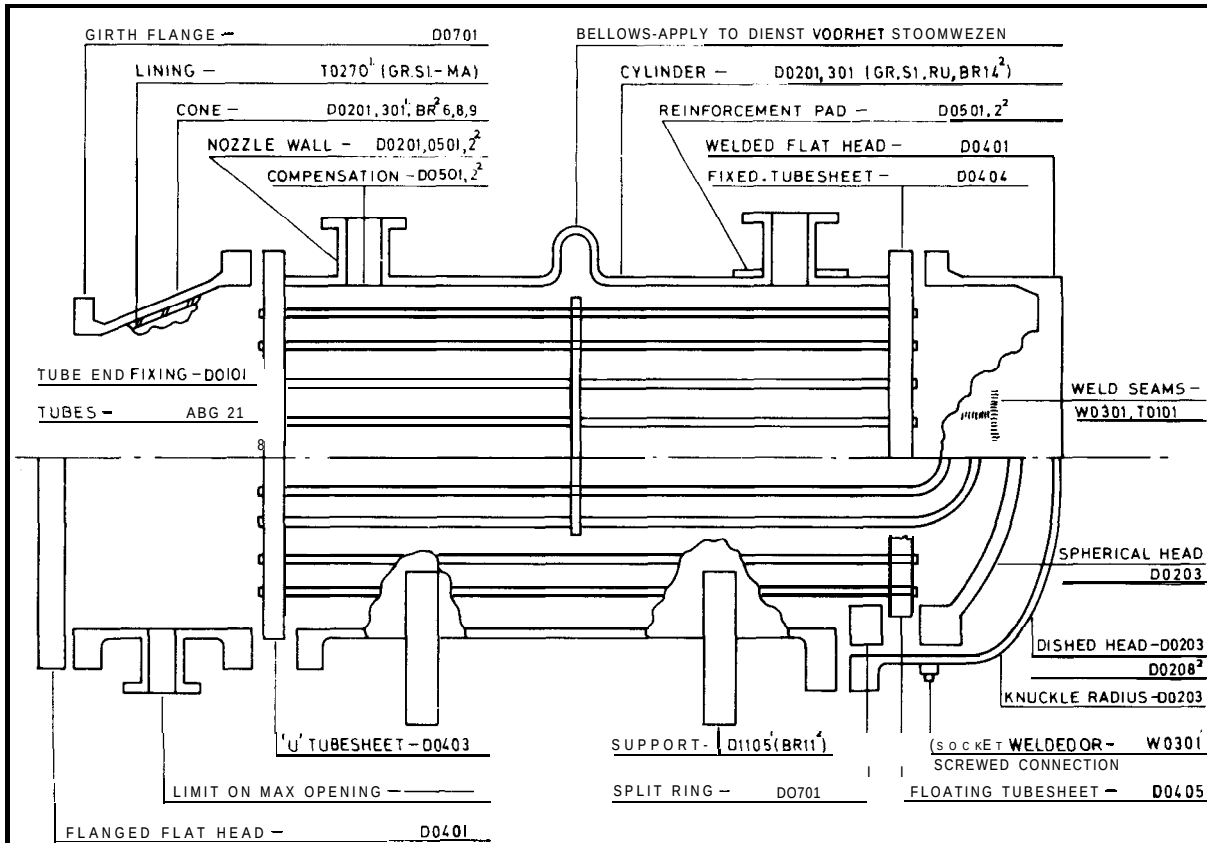
N.D.T.	HP 5/3, HPI
PRESSURE TESTING	HP 20
PROOF TEST	HP 20
REPAIRS	HP 0.6
WELDING - PROCEDURES	HP 2/1
WELDER APPROVAL	HP 3
EDGE PREPARATION	HP 5/1
TEST PLATES	HP 5/2

* IN COURSE OF PREPARATION

GUIDE PREPARED BY H. STEFFEN-VEREINIGUNG DER TECHISCHEN UBERWACHUNGS-
VEREINE e. V.

A. D. MERKBLATTER CODE IS PUBLISHED BY CARL HEYMANNS VERLAG KG, GEREON STRASSE 18-32, D-5000 5 KOLN,
FEDERAL REPUBLIC OF GERMANY.

D. Guide to Dutch vessel code-Stoomwezen



CLASSIFICATION — GO102

N.D.T. — T0101

CORROSION ALLOWANCE — D0101

N.D.T. RADIOGRAPHY — T0110,111,201

DANGER CATEGORIES — G0701

N.D.T. ULTRASONICS — 70110,202

DESIGN PRESSURE — G0203 (1), D0103

N.D.T. MAG PARTICLE, DYE DEN. — T0101

TEMPERATURE — G0203 (1), D0103

PRESSURE TESTING — G0203 (12,13,14)

GENERAL PROCEDURE — G0301,401

PROOF TEST — 10101,240,AKV 23

HEAT TREATMENT — M0110,201,301,601,701,W0701

REPAIRS — G0203 (59),T0110

INSPECTION — G0203 (60),G0402,T0101

WELDING PROCEDURES — T0210

JOINT EFFICIENCY — D0201,35,401,2 D0103,L.5

APPROVAL — T0101,W0301

MATERIALS — M0102,110,201,301,601,701,801,802

EDGE PREPARATION — W0301

T0270¹ (GR SI-MA), ABG 25, AKV 44, BR16²

TEST PLATES — T0101,120,T0220,221

WELDER APPROVAL — 10215

NOTES

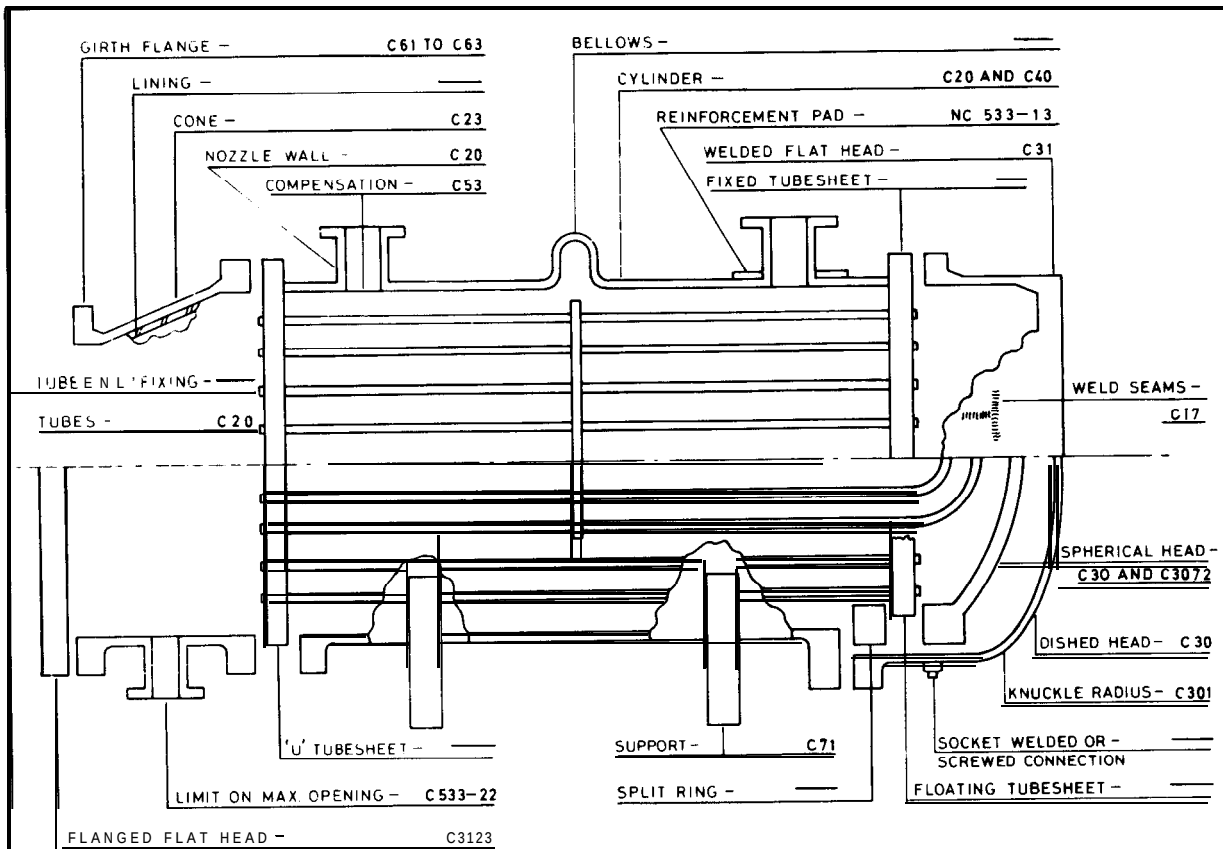
1 IN COURSE OF PREPARATION

2 TEMPORARY

GUIDE PREPARED BY M. VOOIJS, DIENST VOOR HET STOOMWEZEN

DIENST VOOR HET STOOMWEZEN IS PUBLISHED BY GOVERNMENT PUBLISHING OFFICE, CHRISTOFFELPLANTJINSTRAAT, THE HAGUE NETHERLANDS

E. Guide to French vessel code-SNC I



CORROSION ALLOWANCE — C18 AND NC 133-14
 DESIGN PRESSURE — C15
 TEMPERATURE — C15
 JOINT EFFICIENCY — C17 AND NC17

MATERIALS — C133-1 AND C161
 NDT — NC133-11
 PRESSURE TESTING — C164

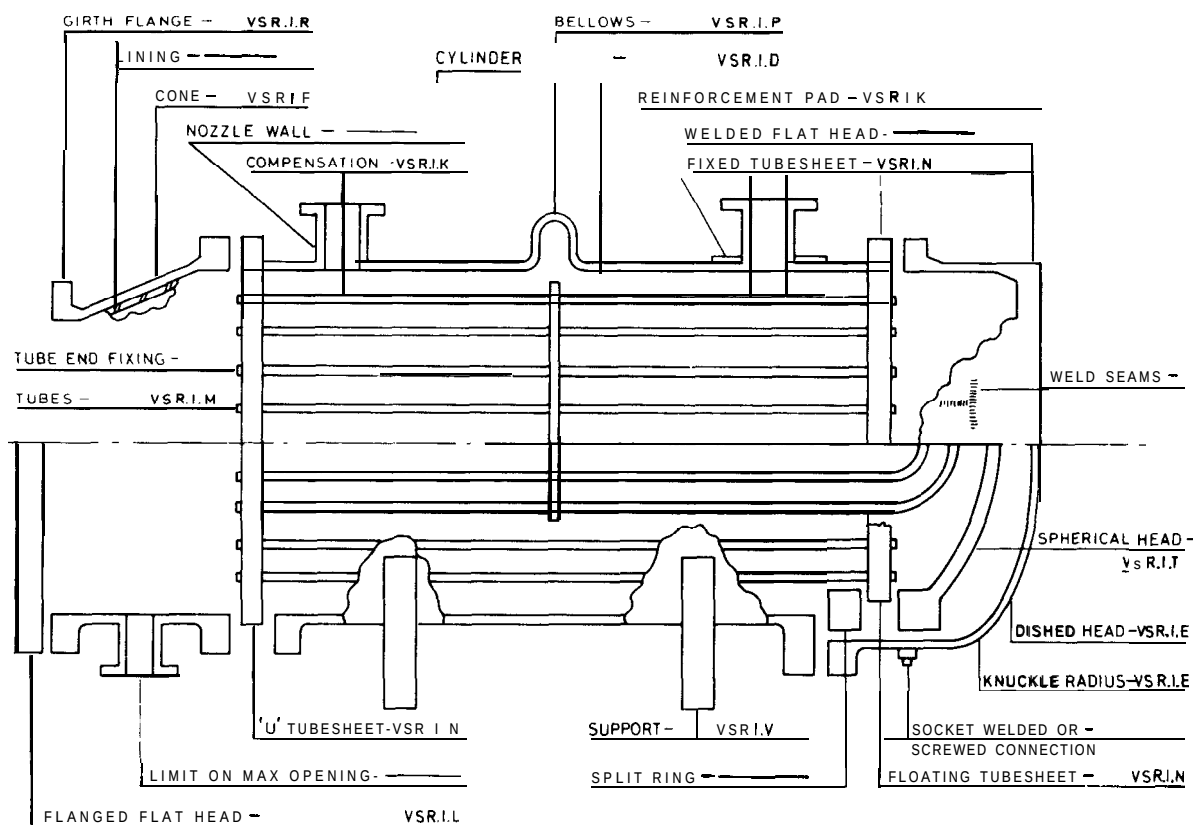
NOTE

THE FRENCH CODE SNC I IS ONLY A CODE FOR CALCULATION PURPOSES

GUIDE PREPARED BY R. MATTERA-DELAS WEIR.

SNC I IS ISSUED BY THE FRENCH MINISTRY OF SCIENTIFIC AND INDUSTRIAL DEVELOPMENT

F. Guide to Italian vessel code-ANCC



CORROSION ALLOWANCE — VSR.I.A
 DESIGN PRESSURE — VSR.I.A
 TEMPERATURE — VSR.I.A
 HEAT TREATMENT — S 8
 JOINT EFFICIENCY — VSR.I.D

MATERIALS — M 2 TO M 11
 N.D.T — S 9
 PRESSURE TESTING — E 0

SECTIONS

VSR-PRESSURE VESSEL STABILITY CONTROL

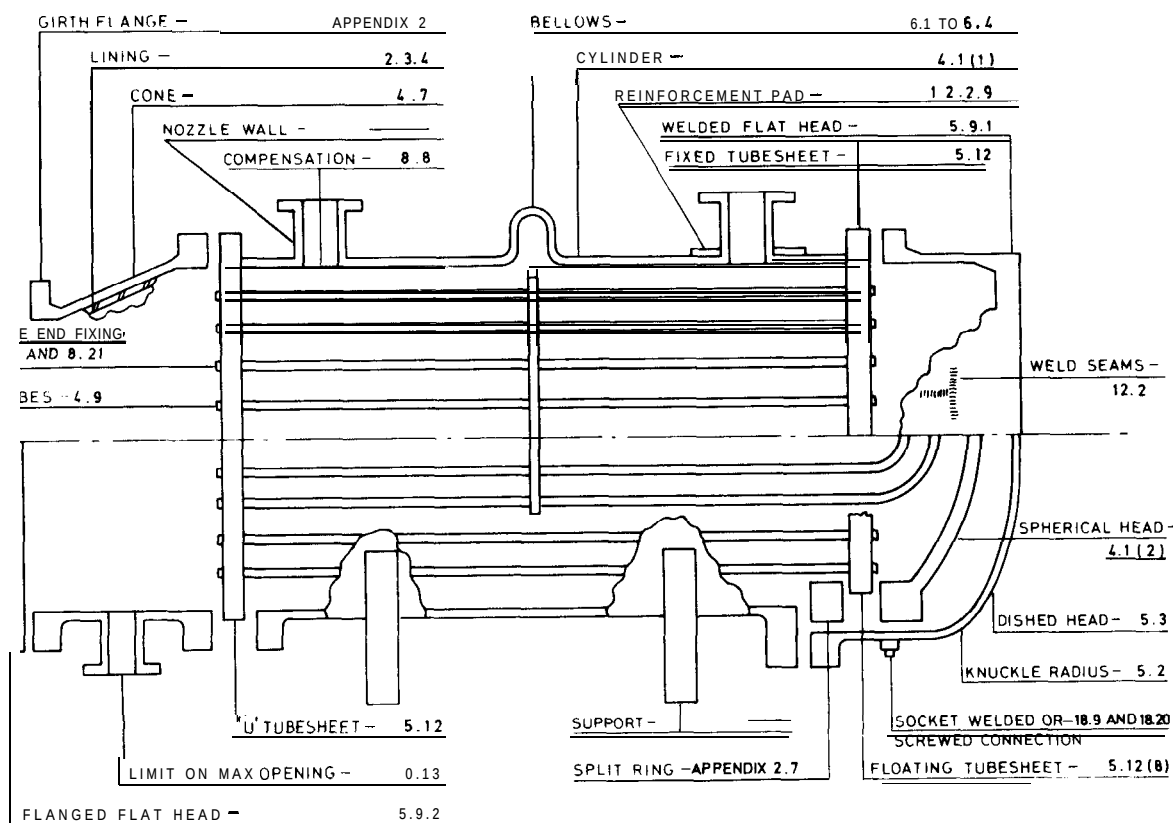
M-USE OF MATERIALS IN MANUFACTURING PRESSURE VESSELS

S -USE OF WELDING, MANUFACTURING AND REPAIRING PRESSURE VESSELS

E-PRESSURE VESSEL OPERATION

GUIDE PREPARED BY L.SANGALLI - INGECO ALTECH GROUP

ANCC. IS PUBLISHED BY CASA EDITRICE LUIGI DLG. PIROLA, VIA COMELICO (PO BOX 3680), MILANO, ITALY (24688)

G1. Guide to Japanese standard vessel construction-JIS-B8243

CORROSION ALLOWANCE	3.3
DESIGN PRESSURE	1.3(3) AND 1.3(4)
TEMPERATURE	1.3.15.1
HEAT TREATMENT	12.6
JOINT EFFICIENCY	12.2.3
MATERIALS	2.1 TO 2.3

NDT	12.8
PRESSURE TESTING	14.1 TO 14.4
PROOF TEST	11.5
REPAIRS	12.7.7 AND 12.7.8
WELDING PROCEDURES	12.3(1)
APPROVAL	12.3(1)
EDGE PREP	12.2.4.3 AND 12.5.2
TEST PLATES	12.7

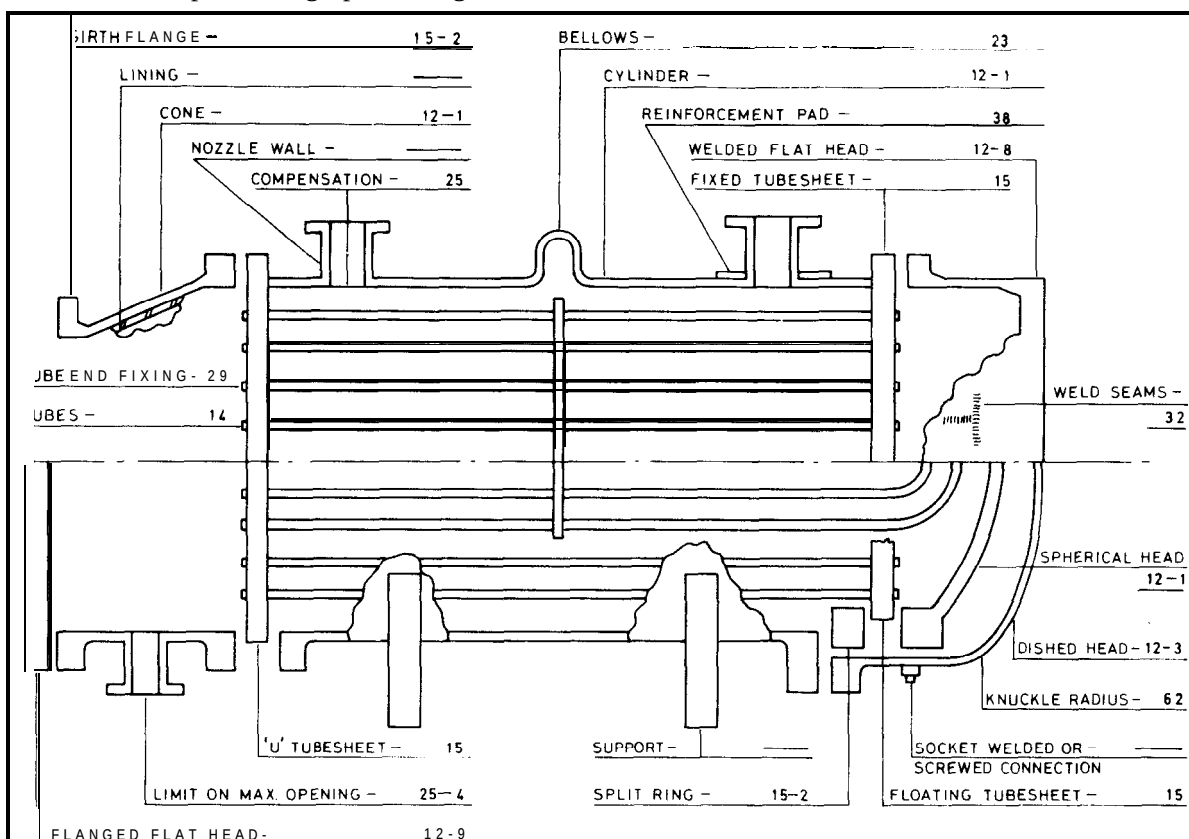
NOTE

JIS-B8243 IS APPLICABLE FOR UNFIRED EQUIPMENT UNDER PRESSURE

GUIDE PREPARED BY TENDO (KOBE STEEL JAPAN)

JIS-B8243 IS PUBLISHED BY JAPAN STANDARDS ASSOCIATION, 4-1-24 AKASAKA, MINATO-KU, TOKYO

G2. Guide to Japanese high-pressure gas control law



CORROSION ALLOWANCE	12
DESIGN PRESSURE	2
TEMPERATURE	2
HEAT TREATMENT	13
INSPECTION	5 AND 69
JOINT EFFICIENCY	20
MATERIALS	9 AND 15

NDT	10 AND 51 TO 57
PRESSURE TESTING	64
REPAIRS	58
WELDING PROCEDURES	
APPROVAL	
EDGE PREPARATION	34
TEST PLATES	44

NOTE

1. JIS-B8243 IS A REFERENCE CODE
2. HIGH PRESSURE GAS CONTROL LAW IS APPLICABLE FOR GASES AND LIQUIDS ABOVE 10 KG/CM².G

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32 - NISHIKUBOTOMOE - CHO, MINATO-KU, TOKYO

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4.3.6

Design example: Floating-head heat exchanger, TEMA type AJS

D. Harris

A. Introduction

This design example is for the mechanical design of a shell-and-tube heat exchanger having a floating tube-sheet. The design conforms with the *Standards of Tubular Exchanger Manufacturers Association*, 6th ed., 1978, and the ASME Boiler and Pressure Vessel Code, Section VIII, Division 1, 1977 edition. The calculations are in SI units.

(a) Specifications

Exchanger type	TEMA AJS
TEMA class	R
Design pressure shell side	2 000 kPa
Design pressure tube side	500 kPa
Design temperature shell side	100° C
Design temperature tube side	15° C
Corrosion allowance shell side	3 mm
Corrosion allowance tube side	3 mm
Shell inside diameter	635 mm
Channel inside diameter	635 mm
Shell inlet nozzle nominal pipe size	203 mm
Shell outlet nozzles (2 off) nominal pipe size	152 mm
Tube-side inlet nozzle nominal pipe size	305 mm
Tube-side outlet nozzle nominal pipe size	305 mm
Number of tubes	468
Tube outside diameter	19.05 mm
Tube wall thickness (14 BWG)	2.11 mm
Tube length	4 060 mm
Tube pitch	23.812 mm

Tube pattern (as defined by TEMA R-2.4)	30
Number of tube passes	4
Number of baffles-segmental	9
Baffle spacing	380 mm
Baffle cut	25%
Impingement protection	None
Weld examination	spot

(b) Material specifications

Carbon steel is used throughout, except for the bolts, which are of low-alloy steel.

Component	Form	Specification
Shell	Plate	SA-5 15-70
Channel	Plate	SA-5 15-70
Channel cover	Plate	SA-515-70
Shell cover cylinder	Plate	SA-515-70
Shell cover formed end	Plate	SA-515-70
Shell flanges	Forging	SA-105
Channel flanges	Plate	SA-515-70
Shell cover flange	Plate	SA-5 15-70
Floating-head cover flange and formed end	Plate	SA-5 15-70
Pass partition plates	Plate	SA-5 15-70
Tubesheets	Plate	SA-515-70
Tubes	Seamless tube	SA-2 1 O-Al
Shell-side nozzles	Pipe	SA-106-B
Tube-side nozzles	Pipe	SA-106-B
Bolts	Bar	SA-193-B7

(c) Order of calculation

For a heat exchanger with a floating tubesheet, assuming that the tube layout is known, it is advisable to design the floating-head cover first. This will confirm if there is sufficient space within the shell and outer tube limit circle diameter to fit the required gasket. The shell, channel, and shell cover cylinder thicknesses may then be calculated. The remaining components may then be designed at random. For this design, the selected tube wall thickness is checked, followed by the design of the flanges and end enclosures. The tubesheets and nozzles calculations complete the calculations for components subjected to pressure. Finally, the dimensions of the nonpressure components are determined.

The units used for pressure are **kPa**, and for material stresses are **MPa** as specified in the **ASME** nomenclature (1 **pascal** = 1 newton per square meter).

B. Floating-head cover**(a) Flange and dish**

The floating-head cover is a spherically dished cover designed to **ASME** UA-6 (4)(b), **ASME** Fig. UA-6(d); see also Sec. 4.3.2F. The minimum dish thickness is the greater of the tube-side or shell-side requirements.

C_{at} = tube-side corrosion allowance = 3 mm

C_{as} = shell-side corrosion allowance = 3 mm

For internal pressure (tube side),

$$t_{fhd} = \frac{5PL}{6S} 10^{-3} \quad (1)$$

where $P = P_t = 500$ kPa, L is the dish corroded inside radius (mm), and S is the maximum allowable stress = 121 MPa.

Assuming that the shell-to-floating tubesheet, radial clearance is 5 mm and assuming a gasket width of 13 mm (TEMA minimum = 12.7 mm), then the flange corroded inside diameter

$$B = 635 - 2(5 + 13) = 599 \text{ mm}$$

The flange uncorroded inside diameter is

$$B - 2C_{at} = 599 - 6 = 593 \text{ mm}$$

Assume that the dish uncorroded inside radius is $0.75 \times 593 = 445$ mm; then

$$L = 445 + C_{at} = 445 + 3 = 448 \text{ mm}$$

Therefore

$$t_{fhd} = \frac{5 \times 500 \times 448}{6 \times 121} 10^{-3} = 1.54 \text{ mm}$$

For external pressure (shell side), the rules in **ASME** UG-33 apply. A thickness is assumed, and the procedure appropriate to the shape of the formed end is followed. From TEMA R-3.13, the minimum allowable corroded plate thickness is 6.35 mm. Assume that $t_{fhd} = 7$ mm and confirm this, using the procedure in **ASME** UG-33(c).

$$\text{factor } A_{ch} = \frac{0.125}{L/t_{fhd}} = \frac{0.125}{448/7} = 1.95 \times 10^{-3} \quad (2)$$

P_a = maximum allowable design pressure

$$= \frac{B_{ch}}{L/t_{fhd}} \quad (3)$$

From **ASME** materials chart, Fig. UCS-28.2,

$$\begin{aligned} \text{factor } B_{ch} &= 14\,500 \text{ lbf/in}^2 = 14\,500 \times 6.895 \\ &= 99\,978 \text{ kPa} \end{aligned}$$

Therefore

$$P_a = \frac{99\,978}{448/7} = 1\,562 \text{ kPa}$$

This is less than the shell-side design pressure (2 000 kPa), therefore assume that $t_{fhd} = 9$ mm and repeat the calculation.

$$\text{factor } A_{ch} = \frac{0.125}{448/9} = 2.5 \times 10^{-3}$$

$$\begin{aligned} \text{factor } B_{ch} &= 15\,200 \text{ lbf/in}^2 = 15\,200 \times 6.895 \\ &= 104\,804 \text{ kPa} \end{aligned}$$

Therefore

$$P_a = \frac{104\,804}{448/9} = 2\,105 \text{ kPa}$$

This is greater than the shell-side design pressure; therefore a corroded dish thickness of 9 mm is satisfactory.

T_{fhd} = dish uncorroded thickness

$$= t_{fhd} + C_{at} + C_{as} = 9 + 3 + 3 = 15 \text{ mm}$$

L_d = dish uncorroded inside radius

$$= L - C_{at} = 448 - 3 = 445 \text{ mm}$$

(b) Flange design

The flange design thickness is the greatest thickness of that required for gasket seating, tube-side pressure, and shell-side pressure, or from geometric considerations to allow for sufficient crossover flow area. The positioning of the dish relative to the flange centroid is an important part of the design calculations; there are numerous methods of approach. For this example the procedure used is as follows.

1. Calculate the flange thickness required for gasket seating.
2. Calculate the flange thickness required for the crossover flow area.
3. Use the greater thickness of (1) or (2) and position the outer edge of the dish at the back of the flange.
4. Calculate the loads and moments exerted on the flange by the tube-side and shell-side pressures.
5. If the flange is overstressed, hold the dish position (thereby keeping the moments constant) and add equal increments to both sides of the flange until an acceptable thickness has been obtained.

The general equation for the flange thickness is given in ASME UA-6, where

$$t = F + \sqrt{F^2 + J} \quad (4)$$

$$F = \frac{(P \times 10^{-3})B\sqrt{4L^2 - B^2}}{8S(A - B)} \quad (5)$$

$$J = \frac{M_o \times 10^3}{SB} \frac{A + B}{A - B} \quad (6)$$

and

$$M_o = M_D + M_G + M_T - M_r \quad (\text{for internal pressure})$$

$$M_o = M_D + M_T - M_r \quad (\text{for external pressure})$$

$$M_o = M_a = Wh_G \quad (\text{for gasket seating})$$

The flange bolt loads and moments are given in Table 1. Note: M_r is the product of H_r and h_r . The dish should be positioned such that M_r is always negative; that is, the dish and tubesheet should always be on the opposite sides of the flange centroid; the absolute value of M_o is used to determine the flange thickness.

For gasket seating,

$$P = 0$$

Therefore

$$F = 0 \text{ and } t = \sqrt{J}$$

Table 1 Floating-head cover bolting and flange moments calculations

Design conditions			
Tube-side design pressure $P_t = 500 \text{ kPa}$	Allowable stresses, MPa		
Shell-side design pressure $P_s = 2\,000 \text{ kPa}$			
Design temperature = 100°C	Flange	Design temperature S_{fo}	121
Flange and dish material = SA-5 15-70		Atmospheric temperature S_{fa}	121
Bolting material = SA-193-B7	Bolting	Design temperature S_b	172
Tube-side corrosion allowance = 3.0 mm		Atmospheric temperature S_a	172
Shell-side corrosion allowance = 3.0 mm			
Gasket and bolting calculations			
Gasket-soft steel jacket asbestos filled			
625 mm OD x 599 ID x 3, with pass partition webs 10 mm wide			
$N = 13 \text{ mm}$ $b_o = 6.5 \text{ mm}$ $b = 6.425 \text{ mm}$ $m = 3.75$ $y = 52.4 \text{ MPa}$			
$G = \text{gasket OD} - 2b = 625 - 2 \times 6.425 = 612.15 \text{ mm}$			
$W_{m2} = b\pi Gy = 647\,459 \text{ N}$	$A_{,,,} = \text{greater of } W_{m2}/S_a \text{ or } W_{m1}/S_b = 3\,764 \text{ mm}^2$		
$H_p = 2\pi b G m P_t \times 10^{-3} = 46\,335 \text{ N}$	Use 24 M20 bolts		
$H = 0.785 G^2 P_t \times 10^{-3} = 147\,081 \text{ N}$	$A_b = 24 \times 214.89 = 5\,157 \text{ mm}^2$		
$W_{m1} = H_p + H = 193\,416 \text{ N}$	$W = 0.5(A_m + A_b)S_a = 767\,206 \text{ N}$		
$H_e = 0.785 G^2 P_s = 588\,322 \text{ N}$	$\beta_1 = \arcsin[B/(2L + t_{fhd})] = 0.661\,9 = 4\,1.44$		
Gasket width check: $N \geq \text{TEMA minimum} = 12.7 \text{ mm}$, or $N_{\min 1}$ or $N_{\min 2}$, where $N_{\min 1} = A_b S_a / 2\pi y G = 4.4 \text{ mm}$			
and $N_{\min 2} = H_e / 2\pi y G = 2.9 \text{ mm}$			
$B = 599 \text{ mm}$ $BCD = 660 \text{ mm}$ $A = 702 \text{ mm}$			
Condition	Load, N	Lever arm, mm	Moment, N m
Tube-side operating	$H_D = 0.785 B^2 P_t \times 10^{-3} = 140\,829$	$h_D = 0.5(BCD - B) = 30.5$	$M_D = 4\,295$
	$H_G = W_{m1} - H = H_p = 46\,335$	$h_G = 0.5(BCD - G) = 23.95$	$M_G = 1\,110$
	$H_T = H - H_D = 6\,252$	$h_T = 0.5(h_D + h_G) = 27.225$	$M_T = 170$
	$H_r = H_D \cot \beta_1 = 159\,515$	$h_r = 20.0$	$M_r = -3\,190$
			$M_o = 2\,385$
Shell-side operating	$H_D = 0.785 B^2 P_s \times 10^{-3} = 563\,317$	$h_D - h_G = 6.55$	$M_D = 3\,690$
	$H_T = H_e - H_D = 25\,005$	$h_T - h_G = 3.275$	$M_T = 82$
	$H_r = H_D \cot \beta_1 = 638\,059$	$h_r = 20.00$	$M_r = -12\,761$
			$M_o = -8\,989$
			$M_a = 18\,374$
Gasket seating	$H_G = W = 767\,206$	$h_G = 23.95$	

$$M_o = 18374 \text{ Nm} \quad \text{and} \quad S = S_{fa} = 121 \text{ MPa}$$

$$t = \sqrt{\frac{18\,374 \times 10^3 \times 702 + 599}{121 \times 599 \times 702 - 599}} = 56.6 \text{ mm}$$

For crossover flow area, from TEMA R-8.12 the actual crossover flow area is to be at least 1.3 times the flow area through one tube pass. There are 468 tubes, 19.05 mm OD X 2.11 mm wall thickness, arranged in four tube passes.

$$\begin{aligned} \text{Flow area/tube pass} &= \frac{468}{4} \frac{\pi}{4} (19.05 - 2 \times 2.11)^2 \\ &= 20\,210 \text{ mm}^2 \end{aligned}$$

The total required crossover area is

$$1.3 \times 20\,210 = 26\,273 \text{ mm}^2$$

The segmental area of the dish = 46 012 mm².

If the effect of the pass partition plates is neglected, the area available under the dish for crossover flow per pass is

$$46\,012 \times 0.5 = 23\,006 \text{ mm}^2$$

If the dish is excluded, the depth of flange required for crossover flow area per pass is

$$\frac{26\,273 - 23\,006}{593 \times 0.5} = 11.02 \text{ mm}$$

Obviously, even allowing for the dish, the gasket seating will require a greater thickness than that for crossover flow area.

Assume a corroded flange thickness of 57 mm and position the dish at the back of the flange; check the thickness for internal and external pressures.

For internal pressure (tube side):

$$P = P_t = 500 \text{ kPa} \quad M_o = 2\,385 \text{ N m}$$

$$S = S_{fo} = 121 \text{ MPa} \quad t = F + \sqrt{F^2 + J}$$

where

$$F = \frac{500 \times 10^{-3} \times 599 \sqrt{4 \times 448^2 - 599^2}}{8 \times 121(702 - 599)} = 2.00 \text{ mm}$$

$$J = \frac{2\,385 \times 10^3 \times 702 + 599}{121 \times 599 \times 702 - 599} = 415.64 \text{ mm}^2$$

$$t = 2.00 + \sqrt{2.00^2 + 415.64} = 22.48 \text{ mm}$$

For external pressure (shell side):

$$P = P_s = 2\,000 \text{ kPa} \quad M_o = 8\,989 \text{ N m}$$

$$S = S_{fo} = 121 \text{ MPa} \quad t = F + \sqrt{F^2 + J}$$

where

$$F = \frac{2\,000 \times 10^{-3} \times 599 \sqrt{4 \times 448^2 - 599^2}}{8 \times 121(702 - 599)} = 8.00 \text{ mm}$$

$$J = \frac{8\,989 \times 10^3 \times 702 + 599}{121 \times 599 \times 702 - 599} = 1\,566.62 \text{ mm}^2$$

$$t = 8.00 + \sqrt{8.00^2 + 1\,566.62} = 48.38 \text{ mm}$$

Therefore, with the dish positioned at the back of the flange, a corroded flange thickness of 57 mm is satisfactory. The cover will have a pass partition plate 13 mm thick, tapered at one end to 10 mm to suit the gasket web; see TEMA R-8.13. The flange will be recessed to confine the gasket. The cover dimensions are given in Fig. 1, and a summary of the flange dimensions is given in Table 11 (flange no. 6).

(c) Floating-head backing device

The backing device clamps the floating-head cover to the tubesheet. There are various types; for this design a single split ring is used, designed to ASME UG-53(a). The split ring is designed as if it were a solid flange (without splits) using 200% of the greater of M_a or M_o calculated in the mating flange design. From Table 1, M_a is greater than M_o for either the tube-side or shell-side operating conditions.

The effective thickness,

$$t_{fhd} = \sqrt{\frac{Y(M_a \times 2)}{S(B \times 10^{-3})}} \quad (7)$$

Shape constant,

$$K = \frac{A}{B} = \frac{702}{599} = 1.172$$

Hence

$$Y = 12.31 \quad \text{from ASME Fig. UA-5 1.1}$$

$$M_a = 18\,374 \text{ Nm} \quad \text{from Table 1}$$

$$S = S_{fa} = 121 \text{ MPa} \quad B = 599 \text{ mm}$$

$$t_{fhd} = \sqrt{\frac{12.31 \times 18\,374 \times 2}{121 \times 599 \times 10^{-3}}} = 79.0 \text{ mm}$$

There will be a recess (5 mm) deep to locate the split ring on the tubesheet. The dimensions of the backing ring are given in Table 11 (flange no. 7), and in Fig. 1.

C. Cylindrical shells

The minimum allowable wall thickness for cylindrical shells is the greater of the ASME or TEMA requirements; see also Sec. 4.3.2A.

The minimum thickness, t , exclusive of corrosion allowance will be the greater thickness from the formulas in ASME UG-27.

For circumferential stress (longitudinal joints):

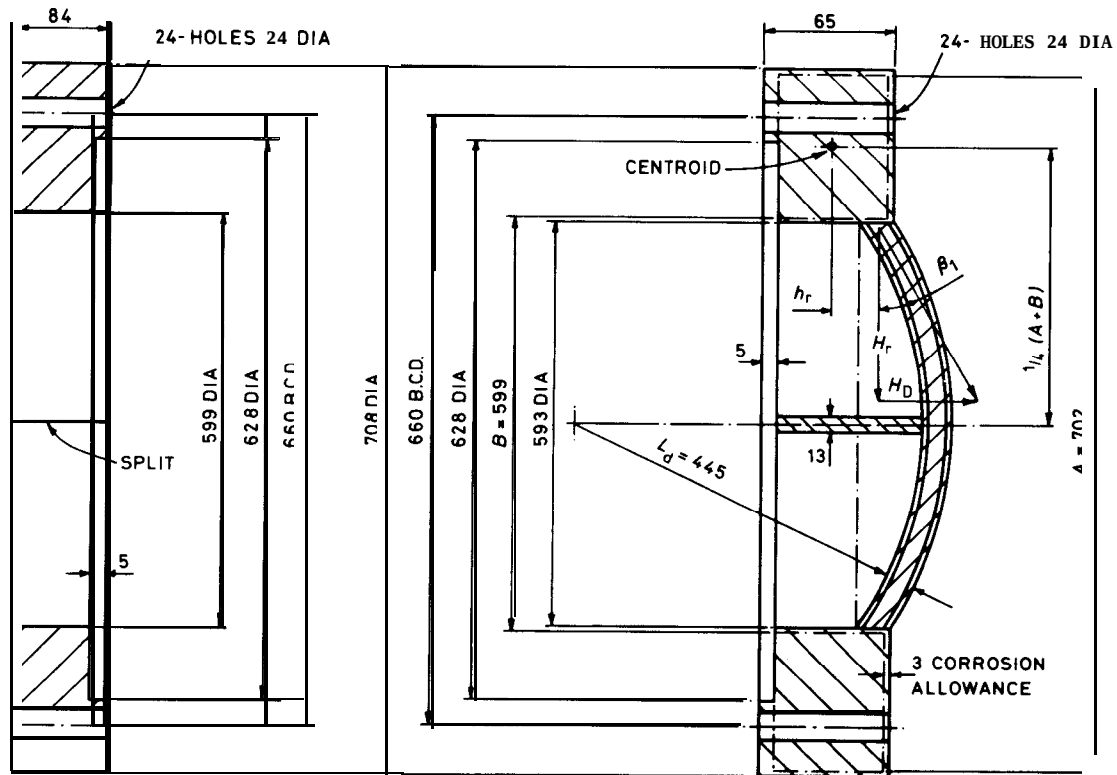


Figure 1 Floating-head cover and backing device.

$$t = P \times 10^{-3} \frac{R_V}{SE - 0.6P \times 10^{-3}} \quad (8)$$

For longitudinal stress (circumferential joints):

$$t = P \times 10^{-3} \frac{R_V}{2SE + 0.4P \times 10^{-3}} \quad (9)$$

For this example the weld examination is spot throughout, thus the joint efficiency $E = 0.85$ for all welded joints (see ASME UW-12); Eq. (9) can therefore be neglected. The calculations of cylinder thickness are summarized in Table 2.

In practice, the selected plate thickness is not necessarily the minimum allowable thickness rounded up to the nearest millimeter. The availability of material may be the deciding factor, or the designer may select a thicker wall to increase the area available for reinforcement at the nozzle locations. The channel will have pass partition plates, of the same material, 13 mm thick, tapered at both ends to 10 mm thick to suit the webs of the gaskets.

D. Tubes

The minimum allowable wall thickness is determined from ASME UG-31; see also Sec. 4.3.2B. The tube dimensions are specified by the thermal designer. The

wall thickness is usually the preferred gauge from TEMA Table R, C, or B, -2.21. The tube length plays an important part in the thermal performance of the exchanger, so the length should not be altered without the agreement of the thermal designer.

d_o = tube outside diameter = 19.05 mm

tube wall thickness = 2.11 mm

P_t = tube-side (internal) pressure = 500 kPa

P_s = shell-side (external) pressure = 2 000 kPa

S = maximum allowable stress = 121 MPa

E = joint efficiency = 1.00

R_V = tube inside radius = $0.5(19.05 - 2 \times 2.11)$
= 7.415 mm

For internal pressure the minimum allowable thickness is calculated using Eq. (8):

$$t = P_t \times 10^{-3} \frac{R_V}{SE - 0.6P_t \times 10^{-3}} = 500 \times 10^{-3}$$

$$\times \frac{7.415}{121 \times 1.00 - 0.6 \times 500 \times 10^{-3}} = 0.031 \text{ mm}$$

For external pressure the rules in ASME UG-28 are

Table 2 Summary of cylinder thickness calculations

	Channel	Shell	Shell cover
P = design pressure, kPa	500	2 000	2 000
R _v = corroded inside radius, mm	320.5	320.5	363
S = maximum allowable stress, MPa	121	121	121
E = joint efficiency	0.85	0.85	0.85
t = minimum design thickness, mm	1.56	6.31	7.12
Corrosion allowance, mm	3	3	3
t + corrosion allowance, mm	4.56	9.31	10.12
TEMA minimum thickness, mm	9.52	9.52	9.52
Selected plate thickness, mm	10	10	11

applied; for this design the procedure when $d_o/t < 10$ is followed.

Step 1. L_t = approximate length of tubes between the tubesheets = 3 960 mm (assuming that the thickness of both tubesheets = 50 mm). Assume that t = tube wall thickness = 2.11 mm, $L_t/d_o = 3\,960/19.05 = 208.4$, and $D_o/t = 19.05/2.11 = 9.03$.

Step 2. Enter **ASME** Fig. UGO-28 in App. V, at the calculated value of L_t/d_o , or 50, whichever is smaller.

Step 3. Move horizontally to the line for the value of d_o/t . From this point of intersection determine the value of factor A_{ch} .

Step 4. Enter the applicable **ASME** materials chart (for this example, Fig. UCS-28.2) using the value of A_{ch} and move vertically to the intersection with the materials/temperature line.

Step 5. From this intersection determine the value of factor B_{ch} . Using the above procedure,

$$A_{ch} = 0.014 \quad \text{and} \quad B_{ch} = 17\,200 \text{ lbf/in}^2$$

Step 6. Calculate the values of pressures P_{a1} and P_{a2} , the lower of which is the maximum allowable external working pressure for a tube.

$$\begin{aligned}
 P_{a1} &= \left(d_o/t - \frac{2.167}{9.03 - 0.0833} \right) B_{ch} \times 6.895 \\
 &= \left(\frac{2.167}{9.03 - 0.0833} \right) 17\,200 \times 6.895 \\
 &= 18\,610 \text{ kPa}
 \end{aligned} \tag{10}$$

$$P_{a2} = \frac{2(S \times 10^3)}{d_o/t} \left(1 - \frac{1}{d_o/t} \right) \tag{11}$$

where S is the lesser of two times the maximum allowable stress value (103 **MPa**) or 0.9 times the material

yield strength (255 **MPa**):

$$S = \text{the lesser of } 2 \times 103 = 206 \text{ MPa}$$

or

$$0.9 \times 255 = 229.5 \text{ MPa}$$

therefore $S = 206 \text{ MPa}$.

$$P_{a2} = \frac{2 \times 206 \times 10^3}{9.03} \left(1 - \frac{1}{9.03} \right) = 40\,573 \text{ kPa}$$

The maximum allowable external pressure for the tube wall is therefore 18 610 **kPa**. Hence a tube wall thickness of 2.11 mm will be satisfactory for both the tube-side and shell-side design pressures.

The tubes will project 3 mm beyond both tubesheets. A maximum projection of one-half tube diameter is permissible for horizontal exchangers when the tubes are expanded to the tubesheets (TEMA R-7.53).

E. Flanges

The flanges are designed to **ASME** App. II, Part A, and TEMA requirements; see also Sec. 4.3.2G. There are three types of flanges, namely, loose, integral, or optional. The type of flanges to be used may depend on the availability of material, design conditions, the process fluids, or manufacturing costs. For example, a carbon steel loose-type flange having a lap joint could be used with a stainless steel shell. Various flange types have been selected at random for this design example; in practice all the flanges may be of the same type. The bolting stress areas were calculated from the minor diameters in British Standard BS 3643: Part 2: 1966, Table 4, medium fit.

The calculations for the flanges are given in the following tables in typical format.

Table 3-Channel cover mating flange-lap type

Table 4-Channel flange-loose type

Table 5-Shell flange-stationary end-integral hub type

Table 6-Shell flange-rear end-integral hub type

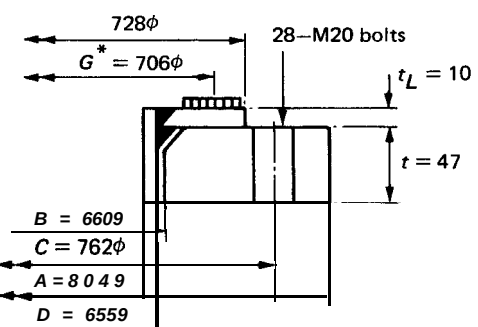
Table 7-Shell cover flange-integral welded type.

The basic gasket seating width, b_o , was calculated using **ASME** Table UA49.2, Column II. For all the flanges of this example, $b_o = N/2$. The hub stress correction factor f should be between 1.00 and 1.25. If it is found to be less than 1, substitute 1 arbitrarily.

From **ASME** UA-52, for materials other than cast iron, the following rule applies to integral-type flanges with the hub welded to the vessel wall, and to optional-type flanges designed as integral type.

The longitudinal hub stress may not be greater than the smaller of 1.5 times the maximum allowable flange

Table 3 Channel cover mating flange calculations

Lap-type flange design					
Design pressure P	500 kPa	Gasket details:	$\left\{ \begin{array}{l} \text{If } b_o \leq 6.35 \text{ mm} \\ b = b_o \end{array} \right.$	N	22 mm
Design temperature	15° C	compressed asbestos 728 OD X 684 ID X 3		b_o	11 mm
Flange material	SA-515-70	Facing details:	$\left\{ \begin{array}{l} \text{If } b_o > 6.35 \text{ mm} \\ b = 2.52 \sqrt{b_o} \end{array} \right.$	b	8.36 mm
Bolting material	SA-193-B7	Type: flat lap 728 OD X 655 ID X 10		y	11 MPa
Corrosion allowance	3 mm	$W_{m2} = 3.14bGy = 203\ 861 \text{ N}$		m	2.0
Allowable bolt stress at	$\left\{ \begin{array}{l} \text{Design temp. } S_b \\ \text{Atm. temp. } S_a \end{array} \right.$	172 MPa 172 MPa	$H_p = 6.28bGmP \times 10^{-3} = 37\ 065 \text{ N}$ $H = 0.785G^2P \times 10^{-3} = 195\ 636 \text{ N}$	$H_p + H = 232\ 701 \text{ N}$	
Allowable flange stress at	$\left\{ \begin{array}{l} \text{Design temp. } S_{fo} \\ \text{Atm. temp. } S_{fa} \end{array} \right.$	121 MPa 121 MPa	$A_{m1} = \text{greater of } \frac{W_{m2}}{S_a} \text{ or } \frac{H_p + H}{S_b} = 1353 \text{ mm}^2$ $A_b = 28 \times 214.89 = 6\ 017 \text{ mm}$ $W = 0.5(A_m + A_b)S_a = 633\ 804 \text{ N}$ $W_{m1} = H_p + H = 232\ 701 \text{ N}$		
All dimensions are shown in mm			Gasket width check	$N_{\min} = \frac{A_b \times S_a}{6.28yG} = 21.2 \text{ mm}$	
Operating condition					
Load, N	Lever arm, mm		Moment, N m		
$H_D = 0.785B^2P \times 10^{-3} = 170\ 973$ $H_G = W_{m1} - H = 37\ 065$ $H_T = H - H_D = 24\ 663$	$h_D = 0.5(C - B) = 51.0$ $h_G = 0.5(C - G) = 28.0$ $h_T = 0.5(C - G) = 28.0$		$H_D h_D \times 10^{-3} = 8\ 720$ $H_G h_G \times 10^{-3} = 1\ 038$ $H_T h_T \times 10^{-3} = 690$ $M_o = 10\ 448$		
Bolting-up condition					
$H_G = W = 633\ 804$	$h_G = 0.5(C - G) = 28.0$		$M_a = H_G h_G \times 10^{-3} = 17\ 746$		
Apply factor C_F when			$C_F = \sqrt{\frac{\text{bolt spacing}}{2(\text{bolt diam}) + 1}} > 1$		
			$K = \frac{A}{B} = \frac{804}{660} = 1.218$, therefore $Y = 9.96$ (ASME Fig. UA-5 1.1)		
			$t = \text{the greater of } \sqrt{\frac{YM_o C_F}{S_{fo} B \times 10^{-3}}} \text{ or } \sqrt{\frac{YM_a C_F}{S_{fa} B \times 10^{-3}}}$		
			but $M_a > M_o$, and $S_{fa} = S_{fo}$		
			therefore $t = \sqrt{\frac{9.96 \times 17\ 746 \times 1.0}{121 \times 660 \times 10^{-3}}} = 47.0$		

$G^* = \text{midpoint of contact between flange and lap}$

Lap material SA-5 15-70
Allowable shear stress for lap = $0.8 \times 121 = 96.8 \text{ MPa}$

Actual shear stress = $\frac{\text{bolt load}}{\pi D(t_L - C_{at})}$
 $= \frac{633\ 804}{\pi 655(10 - 3)} = 44.0 \text{ MPa}$

where the bolt load is the greater of W or W_{m1}

The selected number of bolts was based on a practice commonly used in the heat exchanger industry of one bolt per 25 mm of shell diameter, rounded up to a multiple of 4; hence the reason for the large difference between A_b and A_{m1} .

G* = midpoint of contact between flange and lap

Lap material SA-5 15-70

Allowable shear stress for lap = $0.8 \times 121 = 96.8 \text{ MPa}$

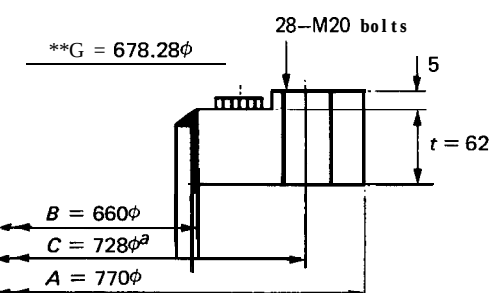
$$\begin{aligned} \text{Actual shear stress} &= \frac{\text{bolt load}}{\pi D(t_L - C_{at})} \\ &= \frac{633\ 804}{\pi 655(10 - 3)} = 44.0 \text{ MPa} \end{aligned}$$

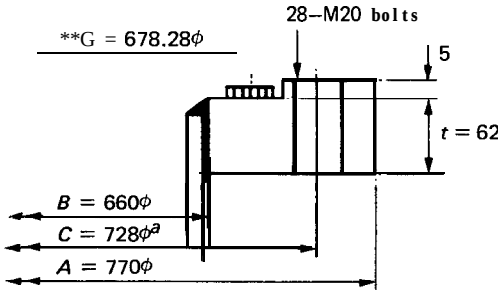
where the bolt load is the greater of W or W_{m1}

The selected number of bolts was based on a practice commonly

used in the heat exchanger industry of one bolt per 25 mm of shell diameter, rounded up to a multiple of 4; hence the reason for the large difference between A_b and A_m .

Table 4 Channel flange (shell mating) calculations

Loose-type flange design					
Design pressure <i>P</i>	500 kPa	Gasket details:	{ If <i>b_o</i> ≤ 6.35 mm <i>b</i> = <i>b_o</i>	N	22 mm
Design temperature	15° C	compressed asbestos 695 OD X 651 ID X 3		<i>b_o</i>	11 mm
Flange material	SA-5 15-70	Facing details:	{ If <i>b_o</i> > 6.35 mm <i>b</i> = 2.52 √ <i>b_o</i>	b	8.36 mm
Bolting material	SA-193-B7	Type: recess 698 diameter X 5		y	11 MPa
Corrosion allowance	3 mm	<i>W_{m2}</i> = 3.14 <i>bGy</i> = 195 856 N		<i>m</i>	2.0
Allowable bolt stress at	{ Design temp. <i>S_b</i> Atm. temp. <i>S_a</i>	172 MPa 172 MPa	<i>H_p</i> = 6.28 <i>bGmP</i> × 10 ⁻³ = 35 610 N <i>H</i> = 0.785 <i>G</i> ² <i>P</i> × 10 ⁻³ = 180 575 N <i>H_p</i> + <i>H</i> = 216 185 N		
Allowable flange stress at	{ Design temp. <i>S_{fo}</i> Atm. temp. <i>S_{fa}</i>	121 MPa 121 MPa	<i>A_m</i> , = greater of $\frac{W_{m2}}{S_a}$ or $\frac{H_p + H}{S_b} = 1257 \text{ mm}^2$ <i>A_b</i> = 28 X 214.89 = 6 017 mm ² <i>W</i> = 0.5(<i>A_m</i> + <i>A_b</i>) <i>S_a</i> = 949 870 N ^a <i>W_{m1}</i> = 864 741 N ^a		
All dimensions are shown in mm		Gasket width check	<i>N_{min}</i> = $\frac{A_b \times S_a}{6.28yG}$ = 22.0 mm		
Operating condition					
Load, N	Lever arm, mm		Moment, N m		
<i>H_D</i> = 0.785 <i>B</i> ² <i>P</i> × 10 ⁻³ = 170 973	<i>h_D</i> = 0.5(<i>C</i> − <i>B</i>) = 34.00		<i>H_D</i> <i>h_D</i> × 10 ⁻³ = 5 813		
<i>H_G</i> = <i>W_{m1}</i> ^a − <i>H</i> = 684 166	<i>h_G</i> = 0.5(<i>C</i> − <i>G</i>) = 24.86		<i>H_G</i> <i>h_G</i> × 10 ⁻³ = 17 008		
<i>H_T</i> = <i>H</i> − <i>H_D</i> = 9 602	<i>h_T</i> = 0.5(<i>h_D</i> + <i>h_G</i>) = 29.43		<i>H_T</i> <i>h_T</i> × 10 ⁻³ = 283		
			<i>M_o</i> = 23 104		
Bolting-up condition					
<i>H_G</i> = <i>W</i> = 949 870	<i>h_G</i> = 0.5(<i>C</i> − <i>G</i>) = 24.86		<i>M_a</i> = <i>H_G</i> <i>h_G</i> × 10 ⁻³ = 23 614		
Apply factor <i>C_F</i> when		<i>C_F</i> = $\sqrt{\frac{\text{bolt spacing}}{2(\text{bolt diam.}) + 1}}$ > 1			
** <i>G</i> = 678.28φ		<i>K</i> = $\frac{A}{B}$ = 1.166, therefore			
		<i>Y</i> = 12.71 (ASME Fig. UA-51.1)			
		<i>t</i> = the greater of $\sqrt{\frac{Y M_o C_F}{S_{fo} B \times 10^{-3}}}$ or $\sqrt{\frac{Y M_a C_F}{S_{fa} B \times 10^{-3}}}$			
		but <i>M_a</i> > <i>M_o</i> , and <i>S_{fa}</i> = <i>S_{fo}</i>			
		therefore <i>t</i> = $\sqrt{\frac{12.71 \times 23\ 614}{121 \times 660 \times 10^{-3}}}$ = 61.3 mm			
** <i>G</i> = gasket OD − 2 <i>b</i>					



**G = gasket OD-2b

Controlled by mating flange.

material stress, or 1.5 times the maximum allowable vessel wall material stress.

F. Shell cover: Dished head

The minimum allowable wall thickness is the greater of the ASME or TEMA requirements; see also Sec. 4.3.2D. The shell cover formed head for this design is a seamless 2:1 ellipsoidal head. From ASME UG-32, the minimum required thickness at the thinnest point after forming shall be the greater of the thicknesses calculated by (1) the formula for ellipsoidal heads, or (2) the formula for hemispherical heads divided by the joint efficiency of the head-to-shell joint.

1. Ellipsoidal head formula:

$$t = P \times 10^{-3} \frac{D}{2SE - 0.2P \times 10^{-3}} \quad (12)$$

where $P = P_s = 2\,000$ kPa

D = shell cover cylinder diameter + 2 X corrosion allowance = $7\,18 + 2 \times 3.0 = 724$ mm

$S = 121 \times 0.85 = 102.85$ MPa, [see ASME UW-12(b)]

$E = 1.0$ (seamless head)

$t = 2\,000 \times 10^{-3} \times 724 / (2 \times 102.85 \times 1.0 - 0.2 \times 2\,000 \times 10^{-3}) = 7.05$ mm

2. Hemispherical head formula:

$$t = P \times 10^{-3} \frac{L_e}{2SE - 0.2P \times 10^{-3}} \quad (13)$$

where P and S are as above,

$$L_e = \frac{D}{2} = \frac{724}{2} = 362 \text{ mm}$$

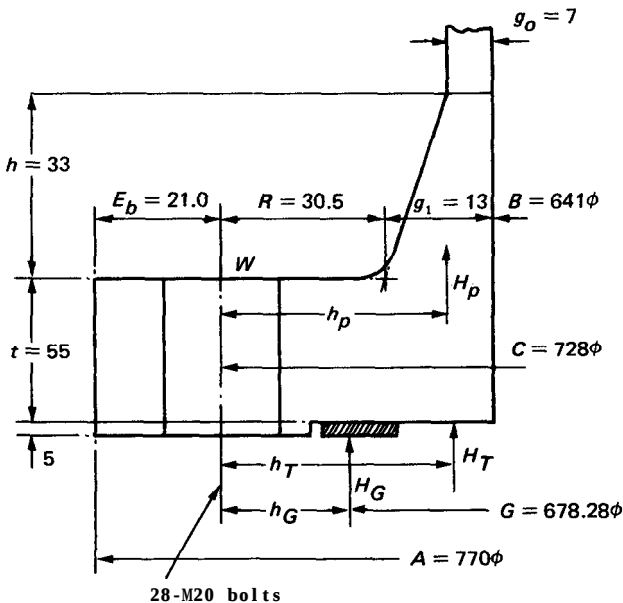
$E = 0.85$ (head to shell welded joint)

Table 5 Shell flange (stationary end) calculations

Integral-type flange design					
Design pressure P		2 000 kPa	Gasket details: compressed asbestos	If $b_o \leq 6.35$ mm	N 22 mm
Design temperature		100° C	695 OD X 651 ID X 3	$b = b_o$	b_o 11 mm
Flange material		SA-105	Facing details: Type: recess	If $b_o > 6.35$ mm	b 8.36 mm
Bolting material		SA-193-B7	698 diameter x 5	$b = 2.52 \sqrt{b_o}$	y 11 MPa
Corrosion allowance		3 mm	$W_{m2} = 3.14bGy = 195\,856$ N		m 2.0
Allowable bolt stress at		Design temp. S_b Atm. temp. S_a	172 MPa 172 MPa	$H_p = 6.28bGmP \times 10^{-3} = 142\,441$ N $H = 0.785G^2P \times 10^{-3} = 722\,300$ N $H_p + H = 864\,741$ N	
Allowable flange stress at		Design temp. S_{fo} Atm. temp. S_{fa}	121 MPa 121 MPa	A, = greater of $\frac{W_{m2}}{S_a}$ or $\frac{H_p + H}{S_b} = 5\,027$ mm ² $A_b = 28 \times 214.89 = 6\,017$ mm ^{2a} $W = 0.5(A_m + A_b)S_a = 949\,870$ N ^a $W_{m1} = H_p + H = 864\,741$ N ^a	
All dimensions are shown in the corroded condition, mm				$N_{min} = \frac{A_b \times S_a}{6.28yG} = 22.0$ mm	
Operating condition					
Load, N		Lever arm, mm		Moment, N m	
$H_D = 0.785B^2P \times 10^{-3} = 645\,083$		$h_D = R + 0.5g_1 = 37.00$		$H_Dh_D \times 10^{-3} = 23\,868$	
$H_G = W_{m1} - H = 142\,441$		$h_G = 0.5(C - G) = 24.86$		$H_Gh_G \times 10^{-3} = 3\,541$	
$H_T = H - H_D = 77\,217$		$h_T = 0.5(R + g_1 + h_G) = 34.18$		$H_T h_T \times 10^{-3} = 2\,639$	
				$M_o = 30\,048$	
Bolting-up condition					
$H_G = W = 949\,870$		$h_G = 0.5(C - G) = 24.86$		$M_a = H_Gh_G \times 10^{-3} = 23\,614$	
Transferred to mating flange.					

Table 5 Shell flange (stationary end) calculations (Continued)

Allowable stress	Stress calculation-operating conditions (use M_1)		Shape constants ASME UA-51.1
$1.5S_{fo}$	Long. hub, $S_H = fM_1/\lambda g_1^2$	159.0	$K = A/B = 1.201$
S_{fo}	Radial flg., $S_R = \beta M_1/\lambda t^2$	15.0	$T = 1.84$
S_{fo}	Tang. flg., $S_T = (M_1 Y/t) - ZS_R$	83.0	$Z = 5.52$
S_{fo}	Greater of $0.5(S_H + S_R)$ or $0.5(S_H + S_T)$	121.0	$Y = 10.70$
			$U = 11.76$
Allowable stress	Stress calculation-gasket seating (use M_2)		$g_1/g_o = 1.857$
$1.5S_{fa}$	Long. hub, $S_H = fM_2/\lambda g_1^2$	125.0	$h_o = \sqrt{Bg_o} = 66.985$
S_{fa}	Radial flg., $S_R = \beta M_2/\lambda t^2$	11.0	$d = \frac{U}{V} h_o g_o^2 = 160831$
S_{fa}	Tang. flg., $S_T = (M_2 Y/t^2) - ZS_R$	65.2	
S_{fa}	Greater of $0.5(S_H + S_R)$ or $0.5(S_H + S_T)$	95.1	
Other stress formula factors			
t (assumed)	55		
$\alpha = t_e + 1$	1.66		
$\beta = 4/3t_e + 1$	1.88		
$\gamma = \alpha/T$	0.902		
$\delta = t^3/d$	1.034		
$\lambda = \gamma + \delta$	1.936		
$M_1 = (M_o/B)10^3$	46 877		
$M_2 = (M_a/B)10^3$	36839		
If bolt spacing exceeds $2d_b + t$, multiply M_a and M_o in above equations by			
$\sqrt{\frac{\text{bolt spacing}}{2d_b + 1}}$			
$G = \text{gasket OD} - 26$			



Transferred to mating flange.

$$t = 2000 \times 10^{-3} \times \frac{362}{2 \times 102.85 \times 0.85 - 0.2 \times 2000 \times 10^{-3}} = 4.15 \text{ mm}$$

Divide this thickness by E to comply with ASME rule UG-32(b):

$$t = \frac{4.15}{0.85} = 4.88 \text{ mm}$$

Therefore, assuming a thinning allowance of 12.5%, the minimum required thickness (including corrosion allowance) to ASME rules is

$$7.05 + 0.125 \times 7.05 + 3.00 = 10.93 \text{ mm}$$

From TEMA R-3.2, the minimum required thickness = 9.53 mm. A plate thickness of 11 mm will be used;

this is the same thickness as the mating shell cover cylinder.

The head dimensions in millimeters are

Thickness = 11

Inside diameter = 718

Outside diameter = 740

Inside depth (excluding skirt) = 179.5

Skirt length = 49.5

Overall length = 240

G. Channel cover

The channel cover is a flat plate, bolted to the channel. The effective thickness is the greater of the ASME or TEMA requirements; see also Sec. 4.3.2E.

t is the effective channel cover thickness, mm

C_c is a factor, dimensionless = 0.3

P is the design pressure = 500 kPa

S is the material maximum allowable stress = 121 MPa

E is the joint efficiency = 1.00

W is the bolt load, gasket seating = 633 804 N

W_{m1} is the bolt load, operating conditions = 232 701 N

h_G is the radial distance from the gasket load reaction
to the bolt circle = 28.0 mm

d_b is the nominal bolt diameter = 20.0 mm

G is the diameter at location of the gasket load reaction
= 706 mm

A_b is the actual total cross-sectional area of the bolts
= 6 017 mm²

where W, W_{m1}, h_G, d_b, G , and A_b have been previously determined from the calculations for the flange that mates with the channel cover; see Table 3.

From ASME UG-34, Fig. UG-34(j), Eq. (2): For operating conditions:

$$t = G \sqrt{\frac{C_c(P \times 10^{-3}) + 1.9 W_{m1} h_G}{S E}} \quad (14)$$

$$= 706 \left(\frac{0.3 \times 500 \times 10^{-3}}{121 \times 1.00} + \frac{1.9 \times 232\,701 \times 28.0}{121 \times 1.00 \times 706^3} \right) = 27.6 \text{ mm}$$

For gasket seating, $P = 0$, therefore

$$t = \sqrt{\frac{1.9 W h_G}{S E G^3}}$$

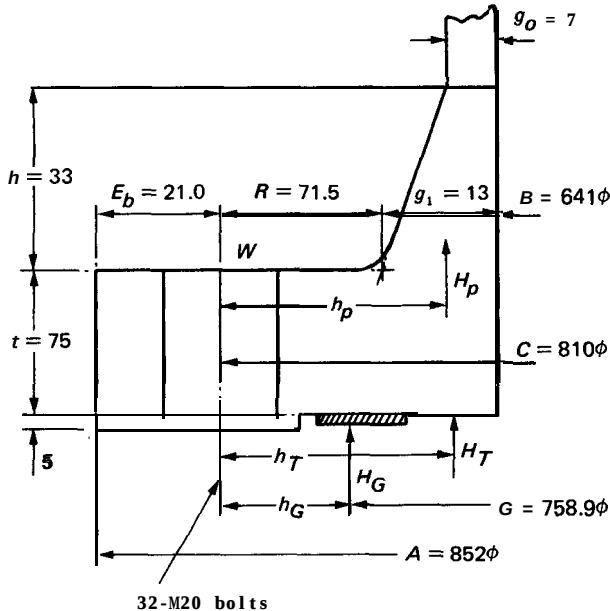
$$= 706 \sqrt{\frac{1.9 \times 633\,804 \times 28.0}{121 \times 1.00 \times 706^3}} = 19.9 \text{ mm (15)}$$

Table 6 Shell flange (rear end) calculations

Integral-type flange design					
Design pressure P	2 000 kPa	Gasket details: compressed asbestos	{ If $b_o \leq 6.35$ mm $b = b_o$	N	23 mm
Design temperature	100° C	776 OD X 730 ID x 3		b_o	11.5 mm
Flange material	SA-105	Facing details: Type: recess	{ If $b_o > 6.35$ mm $b = 2.52 \sqrt{b_o}$	b	8.55 mm
Bolting material	SA-193-B7	779 diameter X 5		y	11 MPa
Corrosion allowance	3 mm	$W_{m2} = 3.14bGy = 224\ 116$ N		m	2.0
Allowable bolt stress at	{ Design temp. S_b Atm. temp. S_a	172 MPa 172 MPa	$H_p = 6.28bGmP \times 10^{-3} = 162\ 993$ N $H = 0.785G^2P \times 10^{-3} = 904\ 209$ N $H_p + H = 1\ 067\ 202$ N		
Allowable flange stress at	{ Design temp. S_{fo} Atm. temp. S_{fa}	121 MPa 121 MPa	A, = greater of $\frac{W_{m2}}{S_a}$ or $\frac{H_p + H}{S_b} = 6\ 205$ mm ² $A_b = 32 \times 214.89 = 6\ 877$ mm ² $W = 0.5(A_m + A_b)S_a = 1\ 125\ 052$ N $W_{m1} = H_p + H = 1\ 067\ 202$ N		
All dimensions are shown in the corroded condition (mm)			Gasket width check $N_{min} = \frac{A_b \times S_a}{6.28yG} = 22.5$ mm		
Operating condition					
Load, N	Lever arm, mm		Moment, N m		
$H_D = 0.785B^2P \times 10^{-3} = 645\ 083$	$h_D = R + 0.5g_1 = 78.00$		$H_Dh_D \times 10^{-3} = 50\ 316$		
$H_G = W_{m1} - H = 162\ 993$	$h_G = 0.5(C - G) = 25.55$		$H_Gh_G \times 10^{-3} = 4\ 164$		
$H_T = H - H_D = 259\ 126$	$h_T = 0.5(R + g_1 + h_G) = 55.02$		$H_T h_T \times 10^{-3} = 14\ 257$		
			$M_o = 68\ 737$		
Bolting-up condition					
$H_G = W = 1\ 125\ 052$	$h_G = 0.5(C - G) = 25.55$		$M_a = H_Gh_G \times 10^{-3} = 28\ 745$		

Table 6 Shell flange (rear end) calculations (Continued)

Allowable stress	Stress calculation-operating conditions (use M_1)	Shape constants ASME UA-5 1.1
$1.5S_{fo}$	Long. hub, $S_H = fM_1/\lambda g_1^2$	$K = A/B = 1.329$
S_{fo}	Radial flg., $S_R = \beta M_1/\lambda t^2$	$T = 1.78$
S_{fo}	Tang. flg., $S_T = (M_1/t^2) - ZS_R$	$Z = 3.61$
S_{fo}	Greater of $0.5(S_H + S_R)$ or $0.5(S_H + S_T)$	$Y = 6.98$
		$U = 7.67$
Allowable stress	Stress calculation-gasket seating (use M_2)	$g_1/g_o = 1.857$
		$h_o = \sqrt{Bg_o} = 66.985$
		$d = \frac{U}{V} h_o g_o^2 = 104\ 896$
		Other stress formula factors
$1.5S_{fa}$	Long. hub, $S_H = fM_2/\lambda g_1^2$	t (assumed) 75
S_{fa}	Radial flg., $S_R = \beta M_2/\lambda t^2$	$\alpha = t_e + 1$ 1.900
S_{fa}	Tang. flg., $S_T = (M_2 Y/t^2) - ZS_R$	$\beta = 4/3t_e + 1$ 2.200
S_{fa}	Greater of $0.5(S_H + S_R)$ or $0.5(S_H + S_T)$	$\gamma = \alpha/T$ 1.067
		$\delta = t^3/d$ 4.022
		$\lambda = \gamma + \delta$ 5.089
		$M_1 = (M_o/B)10^3$ 107 234
		$M_2 = (M_a/B)10^3$ 44 844
		If bolt spacing exceeds $2d_b + t$, multiply M_a and M_o in above equations by
		$\sqrt{\frac{\text{bolt spacing}}{2d_b + t}}$
		$G = \text{gasket OD} - 2b$



From TEMA R-8.2.1

$$t = \left[5.7P \left(\frac{G}{100} \right)^4 + 2 \frac{h_G A_b}{\sqrt{d_b}} \left(\frac{G}{100} \right) \right]^{1/3} \quad (16)$$

In the above TEMA formula, the constants 5.7 and 2 have units of in/lbf and $1/\sqrt{\text{in}}$, respectively. For calculations in SI units, appropriate conversion factors must be used. The conversion factor for constant 5.7 depends on the units being used for the design pressure, for example, bar, kPa, N/m²; the conversion factor for constant 2 is $1/\sqrt{25.4} = 0.198 \text{ mm}^{-1/2}$.

$$t = \left[5.7 \times 5.706 \times 10^{-3} \times 500 \left(\frac{706}{100} \right)^4 + \frac{2 \times 0.198 \times 28.0 \times 6\ 017 \cdot \left(\frac{706}{100} \right)}{\sqrt{(20)}} \right]^{1/3} = 52.6 \text{ mm}$$

The gasket has a compression factor, $m = 2.0$. The thickness obtained by the TEMA formula may be reduced by 20% because m is less than 3.0. Therefore,

$$t = 0.8 \times 52.6 = 42.1 \text{ mm}$$

The thickness calculated by the TEMA formula is greater than the thicknesses calculated by the ASME formula, so the effective thickness is 42.1 mm. The gasket will be confined by a recess 5 mm deep. The cover overall thickness = $42.1 + 5 = 47.1 \text{ mm}$, rounded up to the nearest mm = 48 mm.

The cover dimensions in millimeters are

Outside diameter = 804

Overall thickness = 48

Recess = 732 OD X 632 ID X 5

Pass partition grooves = 13 mm wide X 5 mm deep

Bolt holes = 28 holes, 24 mm diameter
 Bolt circle diameter = 762
 Details of the cover are given in Fig. 2.

H. Tubesheets

For this example both tubesheets are gasketed without edge bolting. The effective thickness is determined from TEMA R-7; see also Sec. 4.3.2I.

(a) Stationary tubesheet

The effective thickness t is the greater of that required for bending or shear.

For bending (TEMA R-7.132):

$$t = \frac{FG}{2} \sqrt{\frac{P \times 10^{-3}}{S}} \quad (17)$$

For shear (TEMA R-7.133):

$$t = \frac{0.31D_L}{1 - d_o/p} \frac{P \times 10^{-3}}{S} \quad (18)$$

where P = the greater of the tube-side or shell-side design pressures = 2 000 kPa

G = the mean diameter of the gasket at the stationary tubesheet = 678.28 mm (from Table 4)

F = a constant = 1.0

S = 121 MPa

d_o = 19.05 mm

p = 23.81 mm

Shear will not be the controlling factor when

$$\frac{P \times 10^{-3}}{S} < 1.6 \left(1 - \frac{d_o}{p}\right)^2$$

$$\frac{P \times 10^{-3}}{S} = \frac{2\,000 \times 10^{-3}}{121} = 0.0165$$

and

Table 7 Shell cover flange calculations

Integral-type flange design					
Design pressure P	2 000 kPa	Gasket details:	$\left\{ \begin{array}{l} \text{If } b_o \leq 6.35 \text{ mm} \\ b = b_o \end{array} \right.$	N	23 mm
Design temperature	100° C	compressed asbestos 776 OD X 730 ID X 3		b_o	11.5 mm
Flange material	SA-5 15-70	Facing details:	$\left\{ \begin{array}{l} \text{If } b_o > 6.35 \text{ mm} \\ b = 2.52 \sqrt{b_o} \end{array} \right.$	b	8.55 mm
Bolting material	SA-193-B7	Type: raised face 727 diameter X 5		y	11 MPa
Corrosion allowance	3 mm	$W_{m2} = 3.14bGy = 224\,116 \text{ N}$		m	2.0
Allowable bolt stress at	$\left\{ \begin{array}{l} \text{Design temp. } S_b \\ \text{Atm. temp. } S_a \end{array} \right.$	172 MPa 172 MPa	$H_p = 6.28bGmP \times 10^{-3} = 162\,993 \text{ N}$ $H = 0.785G^2P \times 10^{-3} = 904\,209 \text{ N}$	$H_p + H = 1\,067\,202 \text{ N}$	
Allowable flange stress at	$\left\{ \begin{array}{l} \text{Design temp. } S_{fo} \\ \text{Atm. temp. } S_{fa} \end{array} \right.$	121 MPa 121 MPa	$A_{m1} = \text{greater of } \frac{W_{m2}}{S_a} \text{ or } \frac{H_p + H}{S_b} = 6\,205 \text{ mm}^2$ $A_b = 32 \times 214.89 = 6\,877 \text{ mm}$ $W = 0.5(A_m + A_b)S_a = 1\,125\,052 \text{ N}$ $W_{m1} = H_p + H = 1\,067\,202 \text{ N}$		
All dimensions are shown in the corroded condition (mm)			Gasket width check	$N_{\min} = \frac{A_b \times S_a}{6.28yG} = 22.5 \text{ mm}$	
Operating condition					
Load, N	Lever arm, mm		Moment, N m		
$H_D = 0.785B^2P \times 10^{-3} = 822\,956$	$h_D = R + 0.5g_1 = 39.00$		$H_D h_D \times 10^{-3} = 32\,095$		
$H_G = W_{m1} - H = 162\,993$	$h_G = 0.5(C - G) = 25.55$		$H_G h_G \times 10^{-3} = 4\,164$		
$H_T = H - H_D = 81\,253$	$h_T = 0.5(R + g_1 + h_G) = 34.28$		$H_T h_T \times 10^{-3} = 2\,785$		
			$M_o = 39\,044$		
Bolting-up condition					
$H_G = W = 1\,125\,052$	$h_G = 0.5(C - G) = 25.55$		$M_a = H_G h_G \times 10^{-3} = 28\,745$		

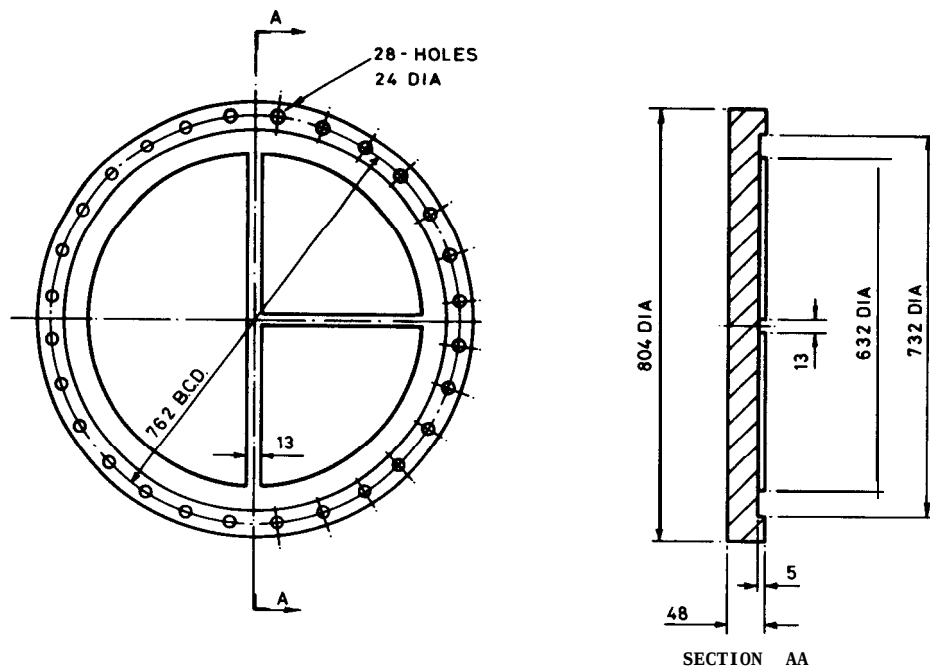


Figure 2 Channel cover.

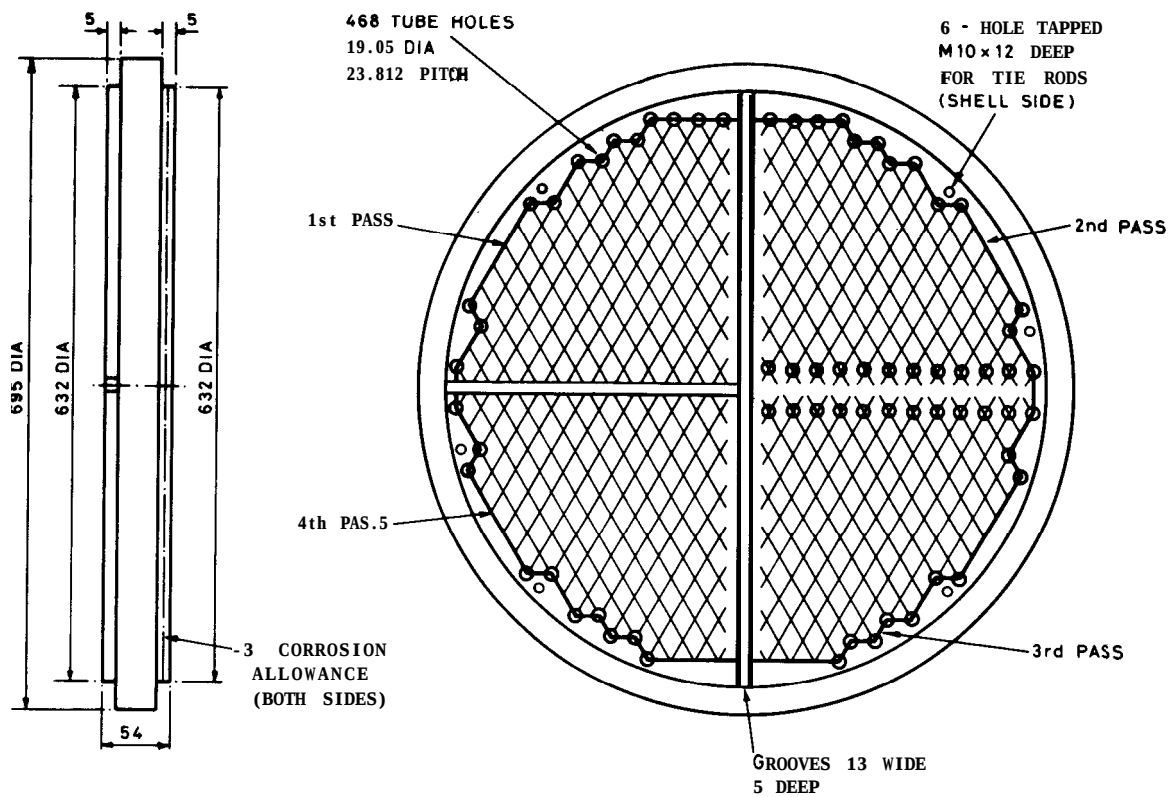


Figure 3 Stationary tubesheet.

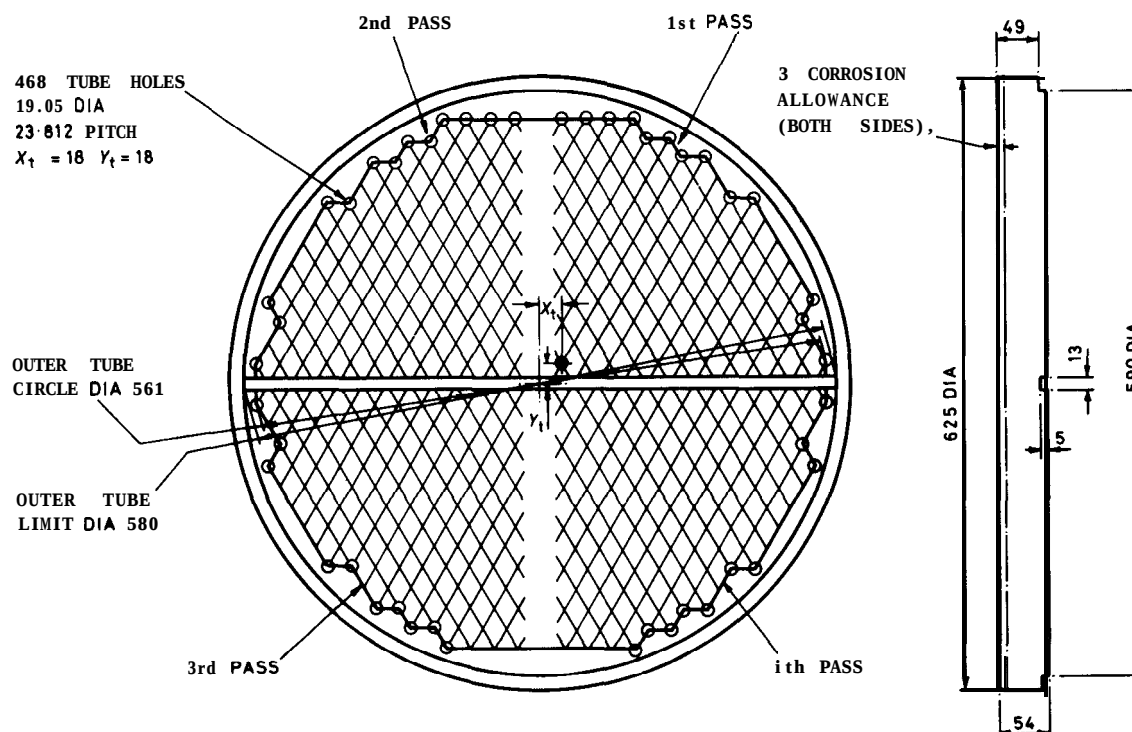


Figure 4 Floating tubesheet.

I. Nozzles and nozzle reinforcements

(a) Nozzles

The nozzle wall thicknesses are calculated to ASME requirements; see also Sec. 4.3.2J and TEMA R-9.

The nozzles will be of seamless pipe. Assuming that there are no external loads, from ASME UG-45, the wall thicknesses of the nozzles may not be less than the smaller of the following.

1. The required thickness t (assuming that $E = 1$) of the shell or head to which the nozzle is attached, plus corrosion allowance provided in the shell (or head) adjacent to the nozzle, but in no case less than the minimum thickness specified for the material in the

applicable part of Subsection C of ASME VIII, Division 1, where

$$t = P \times 10^{-3} \frac{R}{SE - 0.6P \times 10^{-3}} \quad (19)$$

2. The minimum thickness of standard wall pipe plus corrosion allowance on the nozzle. The minimum thickness for all materials is the nominal wall thickness listed in Table 2 of ANSI B36.10-1970, less 12.5%. The length of the nozzles should be kept to a minimum to reduce the effect of external loads. Sufficient length should be specified to allow for the welding of the nozzle, the flange and reinforcement (if any). Adequate length should also be provided for the removal of the stud bolts, taking into account the thickness of the thermal insulation, if fitted.

Table 8 Summary of tubesheet dimensions^a

Tubesheet	Stationary	Floating
Outside diameter	695	625
Tube raised face	632 diameter X 5	590 diameter X 5
Shell-side raised face	632 diameter X 5	—
Pass partition (tube side)	13 wide X 5 deep	13 wide X 5 deep
Tube-side corrosion allowance	3	3
Shell-side corrosion allowance	3	3
Overall thickness	54	54

^aAll dimensions are in millimeters.

Flanges for the nozzles may be of any type in the ANSI Standard, suitable for the design conditions. The flange bores are machined to suit the mating nozzle pipe schedule. A summary of the wall thickness calculations is given in Table 9.

(b) Nozzle reinforcements

When a hole has been cut in a shell or head to accommodate a nozzle, reinforcement may be required to compensate for the metal that has been removed. The reinforcement may be in the form of a forged neck nozzle [see ASME Fig. UG40, (d)-(i)] or a reinforcing plate. This design will deal only with the latter type. The limits of reinforcement are given in Fig. 5, where

$$t_r = P \times 10^{-3} \frac{R}{SE - 0.6P \times 10^{-3}} \quad (20)$$

and

$$t_{rn} = P \times 10^{-3} \frac{R_n}{S_n E - 0.6P \times 10^{-3}} \quad (21)$$

Note: If $S_n < S$, multiply A_2 , A_3 , and A_4 (at nozzle) by S_n/S .

A summary of the reinforcement calculations is given in Table 10. For these calculations $t_{\min}$ is the smaller of 19 mm or the thickness less corrosion allowance of the thinner of the parts joined by a fillet, single-bevel, or single-J weld, mm [ASME UW-16(b)], and t_c is not less than the smaller of 6.35 mm or $0.7t_{\min}$.

TEMA R-9 does not permit shell nozzles to protrude beyond the inside contour of the shell. It is permissible for channel nozzles to protrude beyond the inside contour of the channel, provided that vent and drain connections are flush with the inside contour of

the channel. For this design, all nozzles will be flush with the inside contour of the mating vessel, so in all cases $A_3 = 0$.

J. Nonpressure parts

(a) Transverse baffles

The type, number, spacing, and segmental cut of baffles is specified by the thermal designer. The baffle cut is defined in TEMA R-4.1 as the segmental opening height expressed as a percentage of the shell inside diameter. The maximum baffle-to-shell clearances are defined in TEMA R-4.3. The baffle-to-shell radial clearance is 2.225 mm. The minimum baffle diameter = $635 - 2 \times 2.225 = 630.55$ mm. The minimum baffle thickness = 4.76 mm (TEMA R-4.41). Therefore there will be nine baffles, 631 mm diameter $\times$ 5 mm thick, with a 25% baffle cut.

(b) Longitudinal baffles

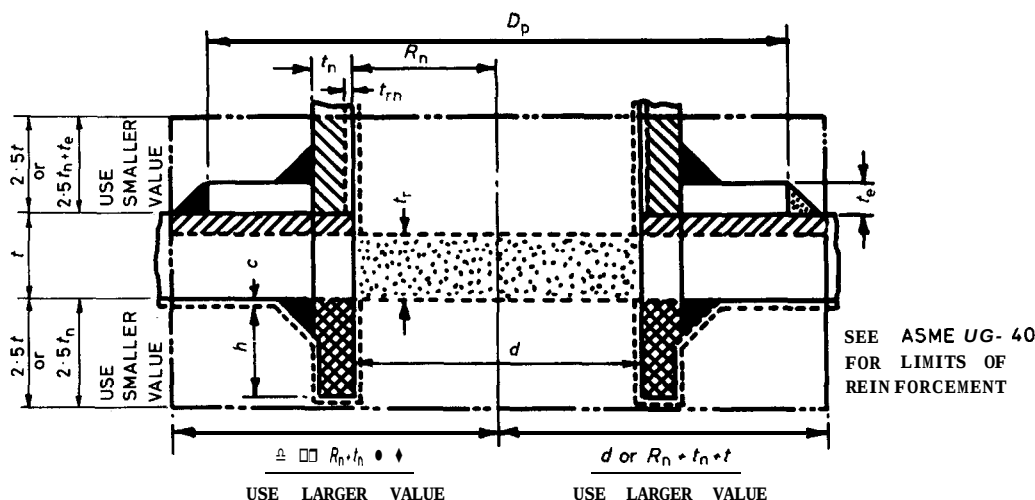
Longitudinal baffles are required when there are two or more shell-side passes. TEMA R-4.42 specifies the minimum thickness as 6.35 mm.

(c) Support plates

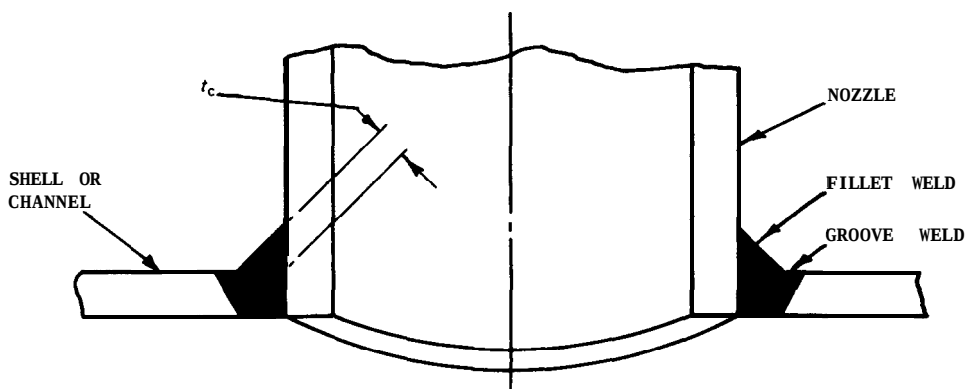
Support plates are used to reduce the effect of the combined weights of the floating-head cover and tubesheet, on the tubes in horizontally mounted exchangers. The plate is normally located inside the shell, close to the floating-head end. The support plates have the same diameter and thickness as the transverse baffles (TEMA R-4.3 and R-4.4). The designer can select any suitable

Table 9 Summary of nozzle wall thickness calculations

	Tube side		Shell side	
Number of nozzles	1	1	1	2
Nozzle label	T1	T2	S1	S2 and S3
Inlet/outlet	Inlet	Outlet	Inlet	Outlet
Nozzle nominal size, mm	305	305	203	152
P = design pressure, kPa	500	500	2 000	2 000
R = corroded inside radius of mating shell or head, mm	320.5	320.5	320.5	320.5
S = maximum allowable stress, MPa	121	121	121	121
E = joint efficiency (assumed)	1.0	1.0	1.0	1.0
c = corrosion allowance, mm	3	3	3	3
t + corrosion allowance, mm	4.33	4.33	8.31	8.31
Standard wall thickness, mm	9.53	9.53	8.18	7.11
Standard wall thickness $\times$ 0.875 + corrosion allowance, mm	11.34	11.34	10.15	9.22
Minimum allowable thickness, mm	4.33	4.33	8.31	8.31
t_{an} = pipe nominal thickness, mm	6.35	6.35	10.31	10.97
Pipe schedule	20	20	60	80
Nozzle outside diameter, mm	323.85	323.85	219.07	168.27



(a)



(b)

Figure 5 Nozzles-boundary limits of reinforcement. (a) With reinforcing element. A , A_1 , A_2 , A_3 , same as without reinforcing element. With a reinforcing element, $2.5t_n$ is measured from the top surface of the reinforcing element. A , becomes the smaller of $(t_n - t_{rn})5t$ or $(t_n - t_{rn})(2.5t_n + t_e)2$. Area of reinforcing element = $(D_p - d - 2t_n)t_e = A_4$. If $A_1 + A_2 + A_3 + A_4 \geq A$, opening is adequately reinforced. If $S_n < S$, multiply A_1 , A_2 , and A_3 (at nozzle) by S_n/S .

Without reinforcing element:



$$= A = dt_r F$$

Area of reinforcement required.



$$= A_1 \left\{ \begin{aligned} &= (E, t - Ft_r)(d - R_n)2 = (E, t - Ft_r)d \\ &\text{or} \\ &= (E, t - Ft_r)(R_n + t_n + t - R_n)2 = (E, t - Ft_r)(t_n + t)2 \end{aligned} \right\}$$

Larger value is area of shell available for reinforcement.



$$= A_2 \left\{ \begin{aligned} &= (t_n - t_{rn})2.5t \times 2 = (t_n - t_{rn})5t \\ &\text{or} \\ &= (t_n - t_{rn})2.5t_n \times 2 = (t_n - t_{rn})5t_n \end{aligned} \right\}$$

Smaller value is area of nozzle wall available for reinforcement.



$$= A_3 = (t_n - c)h \times 2 = (t_n - c)2h \quad [\text{The nozzle projection will not corrode back of any attaching fillet, hence the term } (t_n - c) \text{ is slightly conservative.}]$$



$$= A_4 = \text{area of welds}$$

If $A_1 + A_2 + A_3 + A_4 \geq A$, opening is adequately reinforced. If $A_1 + A_2 + A_3 + A_4 < A$, the difference must be supplied by reinforcing element or otherwise. (From ASME Boiler and Pressure Vessel Code.)

(b) View of acceptable type of welded nozzle without reinforcing element.

Table 10 Summary of nozzle reinforcement calculations

	Tube side		Shell side	
Number of nozzles	1	1	1	2
Nozzle label	T1	T2	S1	S2 and S3
Inlet/outlet	Inlet	Outlet	Inlet	Outlet
P = design pressure, kPa	500	500	2 000	2 000
R = mating vessel inside radius, mm	320.5	320.5	320.5	320.5
S = vessel allowable stress, MPa	121	121	121	121
E, E ₁ , F (dimensionless)	1.0	1.0	1.0	1.0
t _r = thickness of a seamless vessel, mm	1.33	1.33	5.36	5.36
t = nominal thickness of vessel wall, mm	7	7	7	7
R _n = nozzle inside radius, mm	159.36	159.36	103.51	77.53
S _n = nozzle allowable stress, MPa	103	103	103	103
t _{rn} = thickness of a seamless nozzle, mm	0.78	0.78	2.03	1.52
t _n = nominal thickness of nozzle wall, mm	2.56	2.56	6.02	6.6
Nozzle to vessel, weld:				
t _{min} = weld dimension, mm	2.56	2.56	—	—
t _c = fillet throat, mm	1.79	1.79	—	—
Fillet leg = t _c /0.707 1, mm	2.53	2.53	—	—
Nozzle to reinforcing element, weld:				
t _w = fillet throat = 0.7t _{min} , mm	—	—	t _{min} = 6.02 4.21	t _{min} = 6.6 4.62
Fillet leg = t _w /0.707 1, mm	—	—	6	6.53
Reinforcing element to vessel, weld:				
Fillet throat = 0.5t _{min} , mm	—	—	t _{min} = 7 3.5	t _{min} = 7 3.5
Fillet leg = fillet throat/0.707 1, mm	—	—	5	5
d = nozzle inside diameter, mm	318.73	318.73	207.03	155.07
A = required reinforcement area, mm²	423.91	423.91	1109.68	831.19
A _s = available shell reinforcement, mm²	1 807.20	1 807.20	339.53	254.3 1
A _n = available nozzle reinforcement, mm²	19.39	19.39	118.88	151.35
A _{pn} = available nozzle protrusion reinforcement, mm²	0.0	0.0	0.0	0.0
A _w = area of welds, mm²	5.45	5.45	55.64	61.29
Reinforcing element required?	No	No	Yes	Yes
D _p = element outside diameter, mm	—	—	320	270
Element inside diameter, mm	—	—	222	171
t _e = element thickness, mm	—	—	10	10
A _e = element area, mm	—	—	1 009.3	1017.3
Total reinforcement area (A _s + A _n + A _{pn} + A _w), mm²	1 832	1 832	1 523	1484

contour. The maximum unsupported tube length is defined in TEMA R4.52. There will be one support plate 631 mm diameter X 5 mm thick for this example.

(d) Tie rods and spacers

Tie rods and spacers are used to retain all the transverse baffles and support plates in position. One end of the rods screws into the tubesheet and a nut is tightened against the last baffle or support plate at the other end. Spacers are dummy tubes that fit over the tie rods between the baffles to retain the required baffle spacing. TEMA R-4.71 gives suggested tie rod count and diameter; the minimum is four tie rods X 9.5 mm diameter. For this example, six tie rods, screwed M10 X 3 840 mm long, will be used.

(e) Sealing strips

The thermal designer may specify sealing strips to prevent excessive fluid bypassing around the outside of the tube bundle. There are no design constraints placed on these by TEMA, but the material should be compatible with the shell-side fluid.

(f) Impingement baffles and erosion protection

Impingement plates are sometimes necessary to avoid erosion of the tubes at the shell inlet. The impingement plates may be tack-welded in position; they should have a minimum diameter equal to the inlet nozzle outside diameter. The rules for impingement plates are given in TEMA R-4.6.

Table 11 Summary of flange dimension@

Flange no. (see Fig. 6)	1	2	3	4	5	6	7
Outside diameter	804	770	770	852	852	708	708
Inside diameter	660	660	635	635	718	593	599
Hub length	None	None	33	33	None	None	None
Effective thickness	47	62	55	15	80	57	19
Recess or raised face diameter	728	698	698	779	727	628	628
Recess or raised face depth	10	5	5	5	5	5	5
Flange length	5-l	67	93	113	85	65	84
Hub maximum thickness	None	None	16	16	None	None	None
Gasket OD	728	695	695	776	776	625	None
Gasket ID	684	651	651	730	730	599	None
Bolt BCD	762	728	728	810	810	660	660
Number of bolt holes	28	28	28	32	32	24	24
Bolt hole diameter	24	24	24	24	24	24	24

^aAll dimensions are in millimeters. Poisson's ratio = 0.30 for all flanges.

(g) Saddle supports

Neither **ASME** or **TEMA** gives a design method for saddle supports. See Sec. 4.3.2H.

K. Engineering drawings

Before the manufacture of a heat exchanger may commence, the purchaser is given a preliminary arrangement

drawing for approval. When agreement has been reached between the purchaser and supplier, a set of detail drawings (for both fabrication and machining) are completed. A final general arrangement drawing is also required, giving the position of the nozzles and the overall dimensions.

The main dimensions required for the drawings are given in Tables 11 and 12. A sectional view of the heat exchanger is given in Fig. 6.

Table 12 Summary of main dimensions

Component	Inside diameter	Outside diameter	Thickness	Length
Front head cover		804.0	48.0	
Front head barrel	635.0	655.0	10.0	640
Shell barrel	635.0	655.0	10.0	3625
Shell cover skirt	718.0	740.0	11.0	235
Shell cover dish	718.0	740.0	11.0	240
Tubes, 468 off	16.9	19.0	2.1	4060
Shell nozzle S 1	198.5	219.1	10.3	180
Shell nozzle S2	146.3	168.3	11.0	180
Shell nozzle S3	146.3	168.3	11.0	180
Channel nozzle T 1	311.1	323.8	6.4	180
Channel nozzle T2	311.1	323.8	6.4	180
Stationary tubesheet		695	54.0	
Floating tubesheet		625	54.0	
Baffles, 9 off		631	5.0	
Support plate, 1 off		631	5.0	
Tie rods, 6 off	M10			3840
Cover bolts				
Front, 28 off	M20			150
Shell bolts				
Front, 28 off	M20			175
Rear, 32 off	M20			210
Floating-head bolts, 24 off	M20			240

^aAll dimensions are in millimeters. All bolts are metric thread.

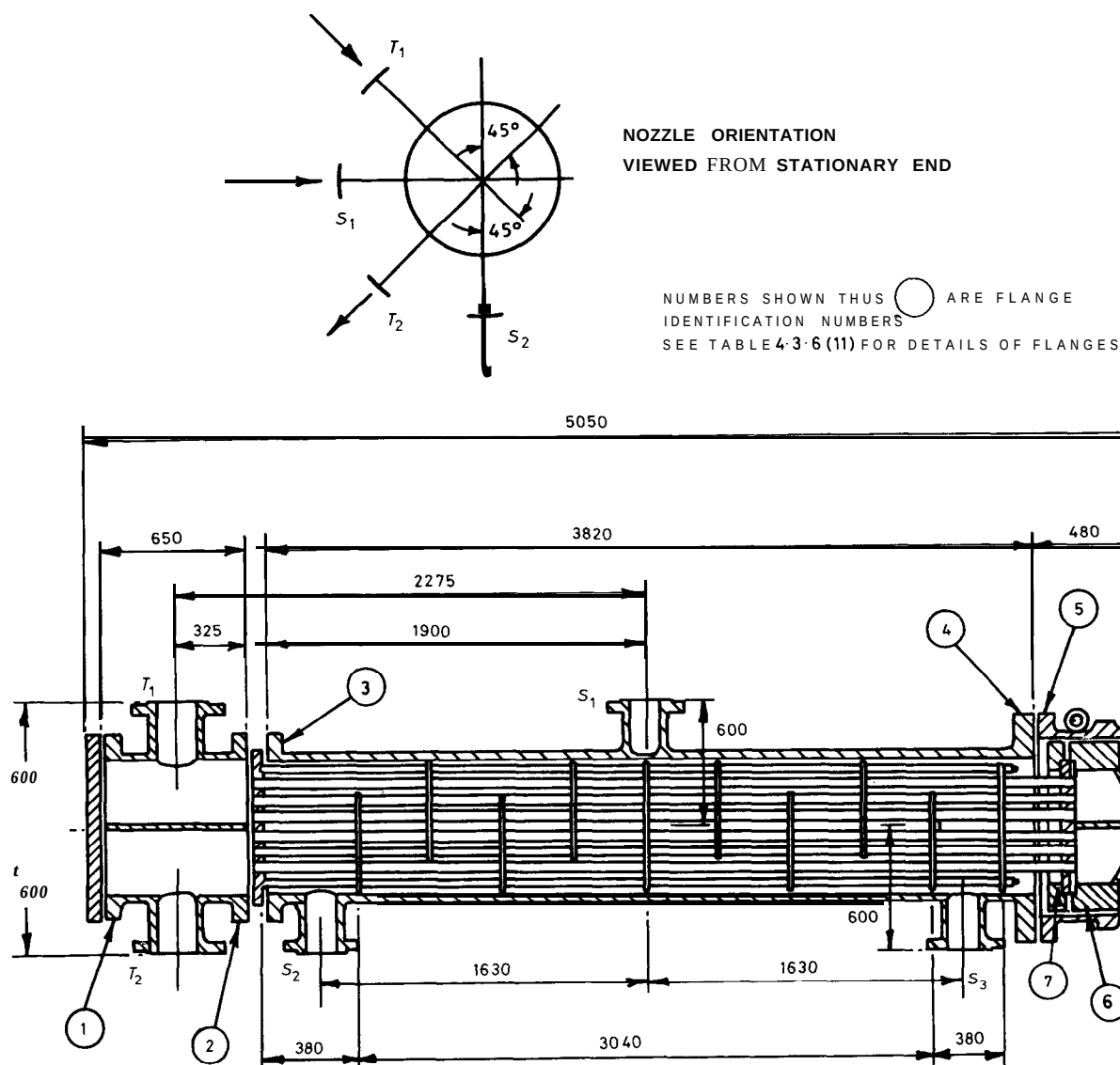


Figure 6 Sectional view of heat exchanger.

NOMENCLATURE FOR SECTION 4.3.6

The nomenclature is in two parts. Part 1 is for all sections except for nozzles (thickness) and reinforcements and Part 2 is for nozzles (thickness) and reinforcements only.

Part 1

- A** outside diameter of flange. For the floating-head cover flange, the corroded outside diameter, mm
- A_b** actual total cross-sectional area of bolts at root of thread, mm²
- A_{ch}** factor **A** from ASME materials charts for components subjected to external pressure

- A_m** total required cross-sectional area of bolts, taken as the greater of **A_{m1}** and **A_{m2}** , mm²
- A_{m1}** total cross-sectional area of bolts at root of thread, required for operating conditions ($= W_{m1}/S_b$), mm²
- A_{m2}** total cross-sectional area of bolts at root of thread, required for gasket seating ($= W_{m2}/S_a$), mm²
- B** inside diameter of flange, corroded, mm
- B_{ch}** factor **B** from ASME materials charts for components subjected to external pressure, kPa
- b** effective gasket seating width, mm
- b_o** basic gasket seating width, mm

C	bolt-circle diameter, mm	M_a	flange moment for gasket seating, N m
C_{as}	corrosion allowance, shell side, mm	M_D	flange moment ($= H_D h_D$), N m
c_{at}	corrosion allowance, tube side, mm	M_G	flange moment ($= H_G h_G$), N m
C_c	a factor used in the calculation of the channel cover	M_o	flange moment for operating conditions, N m
C_F	Taylor-Forge bolt spacing correction factor, applied only when $C_F > 1$	M_r	flange moment ($= H_r h_r$), N m
D	inside diameter, mm	M_T	flange moment ($= H_T h_T$), N m
D_L	equivalent diameter of outer tube circle, mm	m	gasket compression factor
d	a flange shape constant, mm^3	N	gasket width, used to determine b_o , mm
d_b	bolt diameter, mm	$N_{\min}$	minimum gasket width (calculated), mm
d_o	tube outside diameter, mm	P	design pressure, kPa
E	weld joint efficiency	P_a	allowable pressure for vessels subjected to external pressure, kPa
E_b	radial distance from bolt axis to flange outer edge, mm	P_s	shell-side design pressure, kPa
e	a flange shape constant, mm^{-1}	P_t	tube-side design pressure, kPa
F	a flange factor, or dimension required for calculation of floating-head cover flange, mm; or factor required for calculation of tubesheet thickness, as defined in the text	p	tube pitch, mm
f	hub stress correction factor	R	radial distance from bolt circle to point of intersection of hub and back of flange, mm
G	diameter at location of gasket reaction, mm	R_V	inside radius of vessel (corroded) or tubes uncorroded, mm
g_o	thickness of hub at small end (corroded), mm	S	allowable material stress at design temperature, MPa
g_1	thickness of hub at back of flange (corroded), mm	S_a	allowable bolt stress at atmospheric temperature, MPa
H	total hydrostatic end force, N	S_b	allowable bolt stress at operating temperature, MPa
H_D	hydrostatic end force on area inside of flange, N	S_{fa}	allowable stress in the flange material at atmospheric temperature, MPa
H_e	total hydrostatic end force due to external pressure, N	S_{fo}	allowable stress in the flange material at operating temperature, MPa
H_G	gasket load, N	S_H	calculated longitudinal stress in the flange, MPa
H_p	total joint-contact compression load, N	S_R	calculated radial stress in the flange, MPa
H_r	radial component of the load at the intersection of the flange ring with the center line of the dished cover thickness, N	S_T	calculated tangential stress in the flange, MPa
H_T	difference between H and H_D , N	T	a flange factor
h	hub length, mm	T_{fhd}	thickness of floating-head cover dish (uncorroded), mm
h_D	radial distance from the bolt circle to the circle on which H_D acts, mm	t	effective thickness, mm
h_G	radial distance from gasket load reaction to the bolt circle, mm	t_{fhd}	effective thickness of floating-head backing device, mm
h_o	a factor ($= \sqrt{B g_o}$), mm	t_{fhd}	thickness of floating-head cover dish (corroded), mm
h_r	lever arm in floating-head dished end, mm	t_L	lap thickness, mm
h_T	radial distance from the bolt circle to the circle on which H_T acts, mm	U	a flange factor
J	component required for the calculation of floating-head cover flange thickness, mm^2	V	a flange factor
K	flange shape constant ($= A/B$)	W	flange design bolt load for gasket seating, N
L	floating-head internal radius (corroded), mm	w_{m1}	minimum required bolt load for operating conditions, N
L_d	floating-head internal radius (uncorroded), mm	w_{m2}	minimum required bolt load for gasket seating, N
L_e	one-half of the ellipsoidal inside diameter (corroded), mm	x_t	horizontal coordinate of innermost tube, mm
L_t	approximate length of tubes between tubesheets, mm	Y	a flange factor
M	$(M_a/B) \times 10^3$ or $(M_o/B) \times 10^3$ in flange calculations, N	Y	gasket minimum design seating stress, MPa
		y_t	vertical coordinate of innermost tube, mm
		Z	a flange factor
		α	function of flange factors
		β	function of flange factors
		γ	function of flange factors

δ	function of flange factors
λ	function of flange factors
β_1	angle formed by the tangent at the intersection of the floating-head cover flange and dish, deg

Part 2

A	total cross-sectional area of reinforcement required, mm^2
A_1	area of reinforcement available in the shell or channel wall, mm^2
A_2	area of reinforcement available in the nozzle wall, mm^2
A_3	area available for reinforcement when the nozzle wall extends inside the vessel wall, mm^2
A_4	cross-sectional area of welds available for reinforcement, mm^2
A_5	cross-sectional area of material added as reinforcement, mm^2
c	corrosion allowance, mm
D_p	outside diameter of reinforcing element, mm
d	diameter of the opening (in the shell or channel) in its corroded condition, mm
E	weld joint efficiency
E_1	efficiency dependent on type of weld joint (if any) through which the nozzle passes

F	a correction factor that compensates for the variations in pressure stresses on different planes with respect to the axis of a vessel
h	distance nozzle projects beyond the inner or outer surface of the vessel wall, before corrosion allowance is added, mm
P	design pressure, kPa
R	inside radius of shell or channel (corroded condition), mm
R_n	inside radius of nozzle (corroded condition), mm
S	allowable stress value of vessel under consideration, MPa
S_n	allowable stress value of nozzle under consideration, MPa
t	nominal thickness of vessel wall (corroded condition), mm
t_{an}	nominal thickness of nozzle wall uncorroded, mm
t_c	weld dimension (see Fig. 5), mm
t_e	thickness or height of reinforcing element, mm
$t_{\min}$	weld dimension, mm
t_n	nominal thickness of nozzle wall (corroded condition), mm
t_r	required thickness of a seamless shell, mm
t_{rn}	required thickness of a seamless nozzle wall, mm
t_w	fillet throat, mm

4.4

Mechanical design: General forms of heat exchangers

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4.4.1

Mechanical design of air-cooled heat exchangers

K. V. Shipes

A. Introduction

Air-cooled heat exchangers represent a growing segment of equipment being applied in the heat transfer field. It is quite common, these days, for air-cooled heat exchangers to handle more than half the heat dissipation load in a refinery or petrochemical plant. This rapid growth has occurred because of certain advantages inherent in air-cooled heat exchangers as compared to shell-and-tube or other types of heat exchangers. Some of the advantages are the following:

1. The heat exchangers can be located at any point within a refinery, thus eliminating long pipe runs for either the process fluid or the cooling water.
2. Scale buildup on the cooling water side of a shell-and-tube heat exchanger is eliminated, as well as the related shutdowns for cleaning.
3. Cooling tower water treatment and disposal of resulting waste water is eliminated.
4. The overall cost of dissipating heat by this means is generally less than by other methods for fluid streams above 66° C (150°F).

B. Types of air-cooled heat exchangers

The primary distinction in classifying this type of equipment is based on whether the fan is upstream from the finned-tube bundle (forced draft) or downstream (induced draft).

One could also classify air coolers regarding the position of the finned-tube bundles. Although most bundles are in the horizontal plane, or nearly so, there

are some large installations in which the bundles are in the vertical plane. Also, there are some large installations in which the bundles are sloped 30–45° from vertical. These are usually called A frame or V frame, depending on their orientation.

A further distinction may be made regarding the means by which air is caused to flow through the tube bundles. Although axial flow fans are used in the majority of cases, other means used are centrifugal blowers and natural draft towers. One must have a fairly large amount of heat to be dissipated to justify the large and tall towers used in this latter case. Therefore, one rarely sees such a tower except at an electrical power station, where they are applied to both exposed water cooling as well as condensing steam within finned tubes or cooling water within tinned tubes.

Another distinction may be made regarding the extent of accessories for purposes of freeze protection or temperature control. Following are accessories or related equipment that are commonly used:

1. Variable-pitch fans. These may be used not only to vary the airflow, but also to reverse the direction of air flow if required.
2. Air recirculation ducts. Enclosing walls and louvers are used to cause any desired fraction of the air flowing through the bundle to return to its original path and pass through the bundle again. Thus, with proper mixing, one can control the air temperature to the bundle, regardless of how cold the inlet air temperature may be. This is especially important on highly viscous fluids as well as easily frozen fluids.
3. Louvers. These may be used to control the air

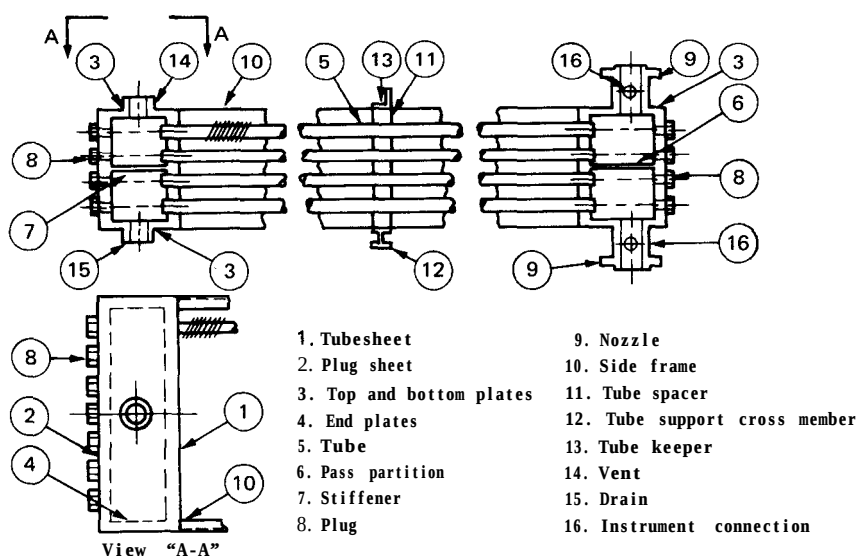


Figure 2 Typical construction of a tube bundle with plug headers.

the finned tube. The majority of finned tubes are composed of tubes ranging from 19 mm (0.75 in) in diameter to 51 mm (2.0 in) in diameter. The fins usually range from 300 to 450 fins/m (7 to 11 fins/in). The fin heights range from 6.0 to 19 mm (0.25 to 0.75 in). Approximately 80% of the finned tubes produced are 25.4 mm (1 in) in diameter and have 350-450 fins/m of 16-mm fin height (9-11 fins/in of 0.625-in fin height). There are a variety of factors that have led to this result. One important factor is the ratio of external surface to bare tube surface for a given tube.

Consider the following table, which shows the ratio of external surface to bare tube surface for tube diameters from 19 to 51 mm (0.75 to 2.00 in). These figures

are based on L-base fins, 400 fins/m, 16-mm fin height, and 0.46-mm fin stock (10 fins/in, 0.625-in fin height, and 0.018-in fin stock). One can readily see that the smaller tubes have higher fin ratios.

Tube diameter, mm (in)	A_o/A_b
19.0 (0.75)	23.591
25.4 (1.00)	20.976
31.8 (1.25)	19.408
38.1 (1.50)	18.363
50.8 (2.00)	17.058

One could, of course, simply increase fin height or fins

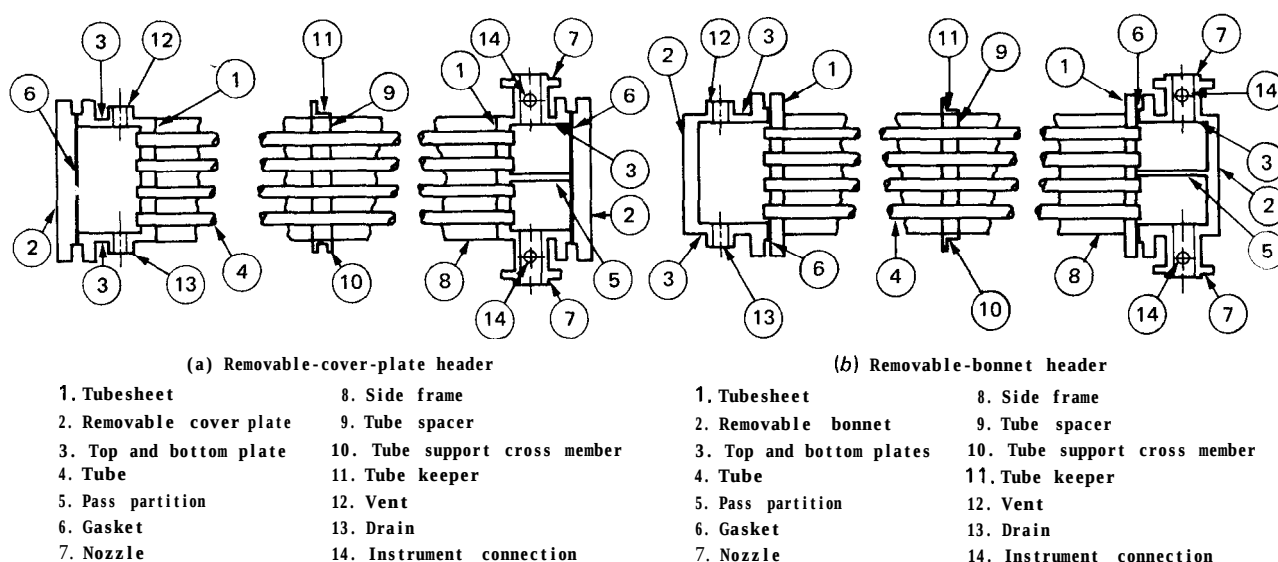


Figure 3 (a) Typical construction of tube bundles with removable-cover-plate headers. (b) Typical construction of tube bundles with removable-bonnet headers.

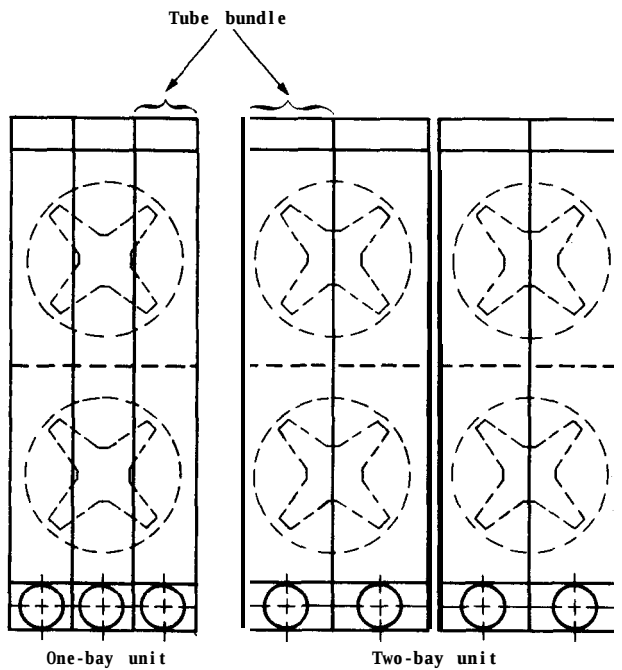


Figure 4 Typical bay arrangements.

per meter to achieve any desired fin ratio. However, such changes are not necessarily cost-effective because of decreased fin efficiency and susceptibility to being clogged with dirt.

Fins may be made from any of several materials. Desirable materials should have high thermal conductivity, should be readily available at reasonable cost, and must be able to withstand the stress and strain involved in applying the material to a tube as a fin. The most common material used is aluminum, which accounts for more than 90% of the total used. Other materials used are copper, steel, stainless steel, and galvanized steel.

Finned tubes are manufactured in several ways. One of the most widely used methods involves winding a strip of fin stock spirally around a tube so that a fin about 16 mm (0.625 in) high results. When such a fin has an L shape to create a foot that encloses the tube OD, it is called an L-base fin. A similar fin having two layers of aluminum in the foot is called an overlapped L-based fin. Another fin having no foot, but being gripped in a groove in the tube OD, is called an embedded fin. Fins are also made by making a multitude of flat plates having one or more holes in them and

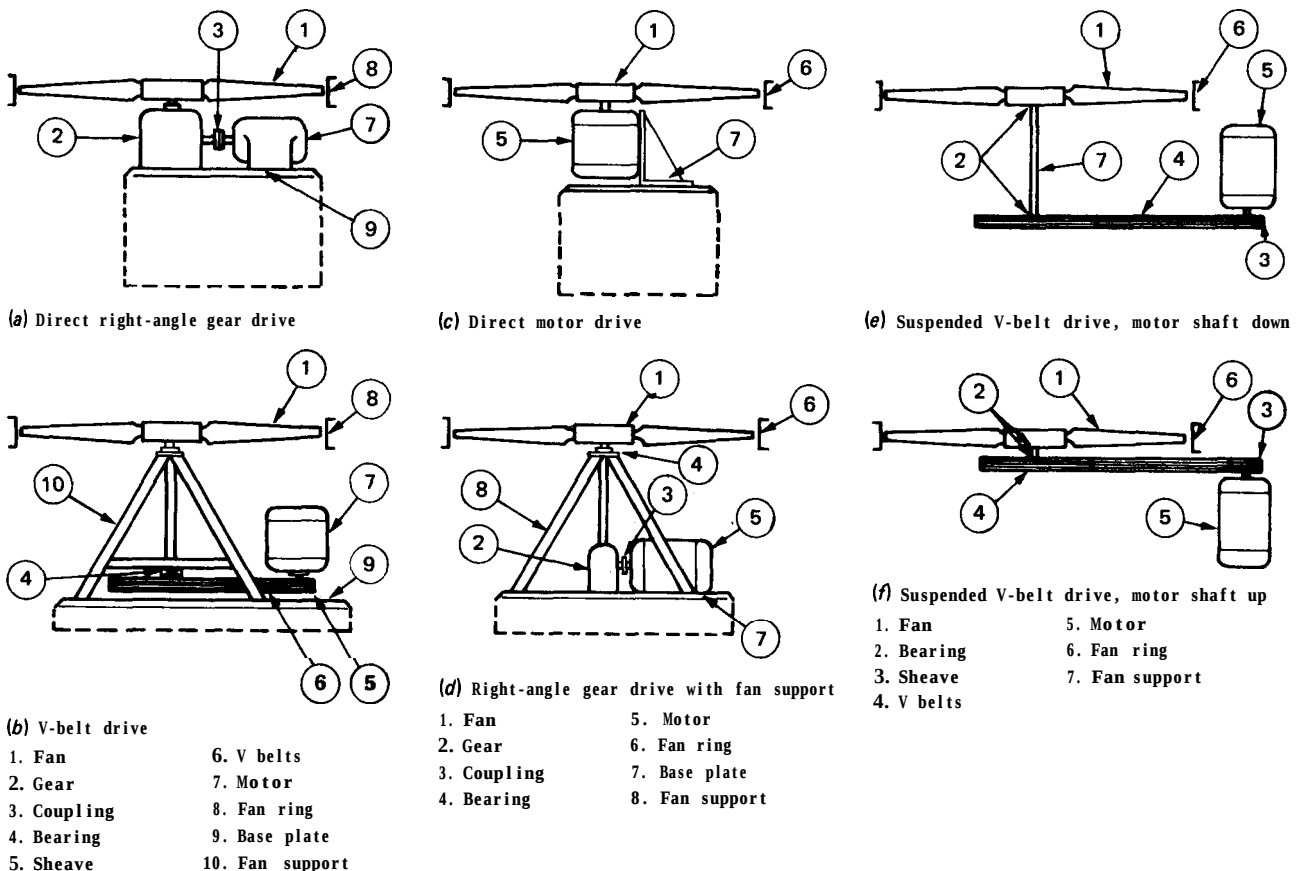


Figure 5 Typical drive arrangements.

sliding them on the tubes. The tube may then be expanded to cause tube-to-fin contact pressure. In one case, such a fin is made of steel, slid on a tube, and is then attached by zinc coating both fin and tube. Fins are also made by sliding an aluminum sleeve over the tube to be finned, and then forming fins from the sleeve by passing the tube with sleeve through a series of forming disks. Such a finned tube has a continuous sheath of aluminum surrounding the tube to prevent entry of moisture between fins.

Fins made of steel or stainless steel have a much lower thermal conductivity than aluminum, but can withstand much higher temperatures. Therefore, fins made of steel or stainless steel are normally not used unless tube wall temperatures in excess of 260° C (500° F) are expected. Such fins are usually made of thicker material (0.635-1.00 mm), and fewer fins per meter are used. Also, they are usually attached to the tube by resistance welding.

(b) Tube bundles

A tube bundle normally consists of a quantity of finned (or bare) tubes attached to headers at each end. Several types of headers are shown in Figs. 2 and 3. The tubes are normally located on a triangular pitch that is equilateral or *isosceles*. An in-line pitch gives a much lower heat transfer rate, and is rarely used. The tubes are supported by means of pitch spacers at intervals of 1.83 m (6 ft) or less along the tube length. Both headers and tubes are held within a closely **fixed** position by the tube-bundle side frames, tube supports, header supports, tube keepers, and tube-pitch spacers.

The headers serve to distribute the flow evenly, and provide for the proper number of passes required of the fluid flow. Also, thermal expansion may be accommodated by placing multiple headers on one or both ends. The variations of construction methods for building headers are almost infinite. There are low-pressure cases where the supply duct serves as a header and the tubesheet is part of the duct wall. Also, in high-pressure cases, forged billets have been drilled longitudinally and then cross-drilled at each tube and plug, to serve as a header.

The number of rows of tubes in the direction of airflow usually ranges from 3 to 6 rows for finned tubes, but may range from 6 to 20 rows for bare tubes.

The optimum number of rows for a given case is a function of a large number of variables, such as overall heat transfer rate, power costs (to move air), installed cost of air-cooled heat exchanger surface, plot plan limitations, and log mean temperature difference (LMTD) for that case.

Several methods are used to attach tubes to headers.

The most common method is to roll the tube into holes in the tubesheet. These holes usually have two grooves **3.18 mm (0.125 in) wide** and 0.4 mm (0.016 in) deep, which aid in resisting axial forces on the tube. Thermal expansion and contraction are the primary cause of these axial forces, although pressure and piping loads may also be contributors. The severity of thermally induced forces are commonly underestimated. Let us assume that one tube in a row becomes plugged or frozen up, while the adjacent tubes remain hot [at 93° C (200° F) tube wall temperature]. Assuming a tube of carbon steel of **2.1-mm (0.083-in) wall, 25.4-mm (1.00-in) diameter**, and an air temperature of 4° C (39° F) to the row in question, this tube experiences a tensile stress of **215 MPa (31 180 psi)** due to thermal forces. The tube end joints experience an axial force of **3 384 kg (7 460 lb)**. Such a condition would probably cause this tube to begin leaking if it had rolled joints. Tubes are also attached to the tubesheet by welding. This welding may or may not be in conjunction with the tube rolling mentioned previously.

The **finned** tubes are held in position with respect to each other by means of a variety of methods. The supporting pieces or methods used are called pitch spacers or pitch-spacing methods. The tubes normally are supported along their length at intervals not exceeding 1.83 m (6 ft), and the pitch spacers are located at these support points. One method used is to attach an aluminum band around the fin OD so that the fin tips are held apart by two thicknesses of the band used. These bands range from 12.7 to 19.0 mm wide (0.5 to 0.75 in). A second method is to attach two halves of a square box shape around the finned tube so that the sides of the box go between the fins and rest on (and fit the curve of) the tube OD. An advantage of this method is that the sides of the box carry the load, instead of the **finns**. Another method is to cast a metal ring around the tube so that this ring encloses both tube and fin, and has an OD larger than the fin OD, and is 12.7-19.0 mm wide (0.05-0.75 in). Thus, adjacent cast rings rest on each other, and all loads are transmitted through these cast rings to the tube wall. Zinc is most commonly used for these rings.

Another method is to form a strip of metal into an **S** shape so that the top of the fins on a given row of tubes fits into the concave surfaces on the lower side of the metal strip, and the bottom of the fins on the next higher row of tubes fits into the concave surfaces on the upper side of the metal strip.

(c) Air-handling system

The vast majority of air-cooled heat exchangers rely on axial-flow fans for air movement. There are some

centrifugal blowers used on **small** units or where high pressure drop across the finned tubes is required, but these are rare exceptions. Also, natural draft towers are used in a few large installations. However, the remarks that follow pertain primarily to axial-flow fans and related equipment. It is desirable to have at least two fans serving each bay, so that if one fan fails, there is still appreciable heat removal from the fluid stream involved.

Fans normally range from 1.8 to 4.1 m in diameter (7 to 16 ft) and may have from 3 to 10 blades. However, four to six blades are more common. Fan blades are made of cast aluminum, extruded aluminum, aluminum sheet, fiberglass-reinforced plastic, and mild steel. The range of pressure drops normally encountered are from 7 to 15 mm (0.28 to 0.60 in) of water. Fan tip speeds of 61 m/s (200 ft/s) have been accepted as a safe upper limit, and the majority of fans were operated at this speed until noise limits caused tip speed decreases. Thus, over the past 8-10 yr, an increasing number of units have been sold with slower fan speeds. At this writing, the average speed probably falls between 46 and 51 m/s (151 and 167 ft/s), with occasional cases as low as 25 m/s (82 ft/s). The emphasis on noise has also led to fan blades of wider design, in order to achieve a reasonable level of air flow at these low fan speeds. Fan drive systems are primarily V belts, with gear reducers accounting for less than 5% of the total. Electric motors account for more than 95% of the drivers used, with the remainder composed of hydraulic motors and steam turbines. It is desirable to have ready access to the motor and V belts as well as lubrication points for fan shaft bearings and motor bearings. V-belt tension should be checked at least four times per year.

Fan efficiency is influenced not only by the fan blade shape, but by tip clearance, fan ring entry, and exit contours (curves or angles), a sealed disk over the center area of the fan, and proper application of the fan to the particular task. API 661 [1] requires a tip clearance no greater than 0.5% of the fan diameter, with upper and lower limits of 19 mm (0.75 in) and 9.5 mm (0.375 in).

When vibration problems occur on air-cooled heat exchangers, one should not automatically assume that the problem is due to fan imbalance, and can be corrected by balancing the fan. Only if the fan is 1.83 m (6 ft) in diameter or less is this likely. Usually, one is dealing with fans from 3 m (10 ft) in diameter to 4.9 m (16 ft) in diameter and fan speeds from 220 to 380 rpm. More than 95% of these cases will be vibrating not at fan rpm but at blade-pass frequency (fan rpm X number of blades). Usually, vibration problems on these large fans require some additional structural rigidity (supporting the fan shaft bearings) or changing the frequency of input forces (increasing rpm or number of blades).

(d) Structure

Air-cooled heat exchangers depend on an adequate supply of unused (unheated) ambient air in order to operate most effectively. Further, the usual installation consists not of a single bay, but of anywhere from 5 to 50 bays, in a variety of sizes and located at several different areas. Therefore, any given bay may be subject to recirculating some fraction of air that has already been heated by itself or by adjacent units. Therefore, it is desirable to plan the location of the units to minimize this risk.

The most important factor in accomplishing this is to elevate all units so that natural air movement can occur beneath the units. Many plants make use of pipe racks as a base on which to place the air-cooled heat exchangers, thus achieving 9-12 m (30-40 ft) of additional elevation of the equipment.

The structural components must be adequate to support the tube bundle, plenum, fan ring, and air-handling package under a variety of conditions. These conditions will be discussed more fully in a later section.

(e) Accessories

There are many items that can be added to an air-cooled heat exchanger to modify its operation or increase its ability to perform under extreme conditions. Louvers are a very common item, and are often used to control the air quantity passing through the unit. Variable-pitch fans are used for the same purpose, and also are used in some cases to reverse the direction of airflow. Recirculation ducts, composed of added panels and louvers, are often used to permit units to operate in colder weather than could otherwise be permitted. Temperature sensors are used to measure fluid temperature, air temperature, and even tube wall temperature. Controllers and various logic components are used to control the function of all the items named above to achieve specific goals. Steam coils may be used to ensure that equipment is heated above a certain temperature while the equipment is shut down, or they may be used to elevate the temperature of equipment to a safe level before introducing a fluid stream.

Normally, the goals are to enable heat removal to continue over a wide range of ambient temperature, and to do so without interrupting the fluid flow or freezing up the equipment. Further, it is often necessary to cool fluids with high pour points, which requires that the inlet air temperature be maintained within narrow limits.

E. Structural design factors

The governing specification for the design, fabrication, and erection of this type of equipment is the AISC

Code of Standard Practice for Steel Buildings and Bridges [2]. The forces that must be considered are as follows:

1. Dead loads. This includes the basic unit plus fluid plus fireproofing.
2. Live loads. This includes both people and equipment. Floor plates must be capable of supporting 4.79 kPa (100 lb/ft²). Ladders and treads must be capable of supporting a moving concentrated load of 227 kg (500 lb).
3. Thermal forces. These include thermal expansion and contraction of the tube bundle parts as well as all connecting piping.
4. Wind loads.
5. Earthquake forces. This normally requires that the equipment be designed according to the Uniform Building Code [3], or ANSI A58.1.1972, Building Code Requirements for Minimum Design Loads in Buildings and Other Structures [4].
6. Dynamic forces, such as are caused by mounting equipment on a moving base, such as a ship.
7. Loading combinations. There are several areas where minimum metal thicknesses have been specified by API 661 [1]. For instance, plenum panels must be at least 1.9 mm (0.075 in) (flat) and 1.5 mm (0.060 in) (ribbed). Also, fan decks must be at least 2.7 mm (0.106 in). Floor plates must have a raised pattern and a minimum thickness of 6.4 mm (0.250 in).

F. Designs for cold climates and viscous fluids

Air-cooled heat exchangers are designed to perform at ambient temperatures ranging from 54 to -51° C (129

to -60° F). There are several areas where the ambient temperature ranges from a high of 32°C (90°F) to a low of -51°C (-60° F). The negative aspect of high ambient temperature is an increase in cost to cool a given fluid stream. However, low ambient temperatures cause freezing of some fluids, which leads to ruptures, shutdowns, and similar equipment failures.

There are several methods used to prevent these problems from happening. One method used quite extensively is to provide a return path for approximately 50% of the exit (heated) air and to mix this heated air with incoming cold air to achieve a safe and controlled inlet air temperature. It is impractical to apply this method to induced draft units, since the unit must have plenums both above and below the coil, plus louver control of both exit air and bypass air.

Another method is to cause one fan of each bay to pump air in an opposite direction from normal, and cause the intake air on at least one side of the tube bundle to be heated to some degree. A third method, applied only to multipass units, is to cause all fans to pump air in an opposite direction from normal, which gives **cocurrent** operation. There are also cases where simply controlling the air flow is adequate.

The condensation of steam at pressures below atmosphere is done on a large scale at many power plants using air-cooled steam condensers. Depending on the particular conditions of a given case, several of the previously mentioned methods are used to prevent freezing. Also, there are several proprietary methods, which reduce or avoid the extensive recirculation of hot air.

REFERENCES FOR SECTION 4.4.1

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2. Code of Standard Practice for Steel Buildings and Bridges, adopted effective October 1, 1972, American Institute of Steel Construction, Inc., Chicago, IL.
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4.4.2

Mechanical design of plate heat exchangers

J. Dennis Usher

A. General description

A plate heat exchanger (PHE) consists of a pack of rectangular pressed plates, suspended vertically, and clamped together in a frame by tie bars or screws. The plates are corrugated to increase strength and to improve heat transfer, and are fitted with peripheral sealing gaskets. Four corner ports, which communicate with appropriate connections mounted on the frame, are arranged so that the two liquids flow through alternate passages between adjacent plates, usually in countercurrent flow, with heat transfer taking place across the plate surfaces.

Because the pressings are exceedingly complex, expensive tools are required for their production. Unlike shell-and-tube units, plates are produced in a limited number of standard sizes and are never custom made to individual requirements. However, the flexibility of their arrangement enables any particularly duty to be easily handled by the correct selection of standard components.

B. Plate construction features

Plates that are normally pressed in light-gauge corrosion-resistant materials (varying in thickness from 0.6 to 1.0 mm) generally embody the following features, as shown in Fig. 1:

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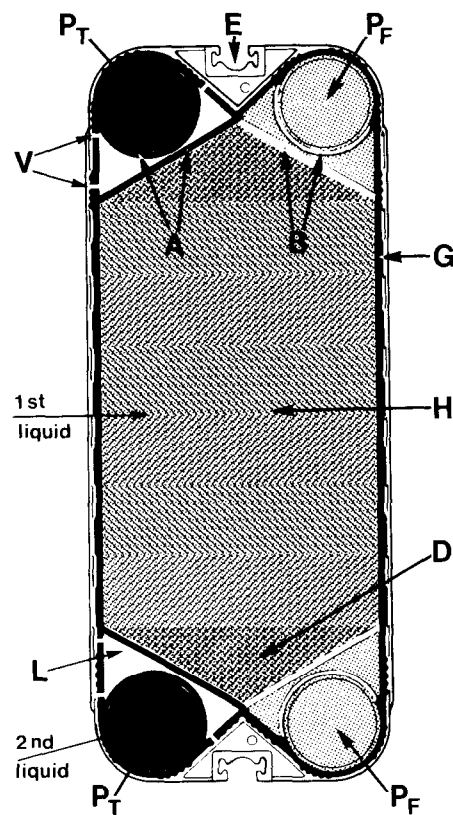


Figure 1 Diagram of heat exchanger plate.

1. Two flow ports P_F , which provide the inlet and outlet for one process liquid to one interplate passage, and two transfer ports P_T , which introduce the other process liquid to the adjacent passage

2. A rectangular corrugated area H, over which the main heat transfer takes place

3. End heat transfer zones D, approximately triangular in shape, which distribute the liquid from the ports P_F to the area H

4. Port leakage areas L, between the transfer port P_T and the area H, which are vented to atmosphere through grooves V so that direct leakage cannot take place between the two process liquids

5. End guides E, which suspend and locate the plates in the frame

6. A gasket G, normally elastomeric, which is cemented into a suitably shaped groove, and which seals both peripherally and around the transfer ports

The ports may be arranged in two ways. Vertical flow takes place between ports located on the same side of the plate (Fig. 2). This means that only one form of plate is required, as the inversion of alternate plates reverses the flow and transfer ports and gives the required flow pattern, thus simplifying manufacturing and stocking problems. Diagonal flow between opposite corners (Fig. 3) gives a different flow distribution pattern down the heat transfer surface, but when inverted the plate has exactly the same port arrangement. To avoid this, two different forms of plate are required, a left hand and a right hand, in which one is the

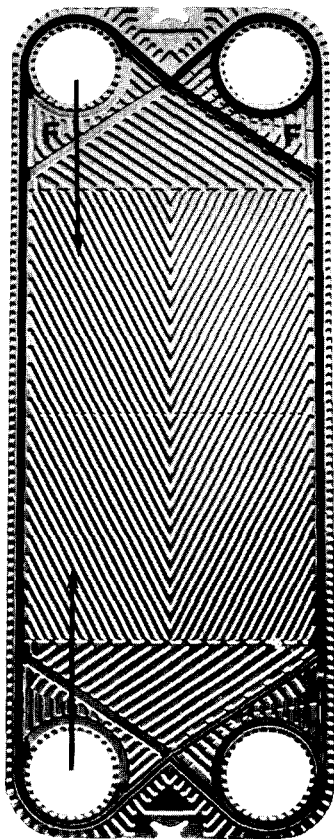


Figure 2 Vertical flow plate with chevron troughs.

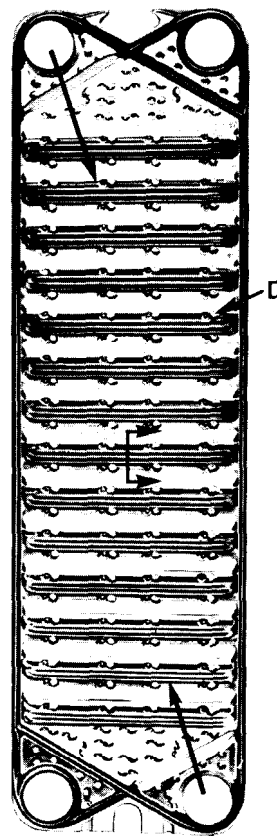


Figure 3 Diagonal flow plate with intermingling troughs.

mirror image of the other, and this involves additional tooling. In the interests of cost and standardization, most modern plates will be found to be of the vertical flow type.

C. Plate arrangements

Plates are assembled in the appropriate number and arrangement to conform to the required duty. Figure 4 shows the flow pattern of a two-pass system with two passages for each liquid in each pass. This indicates how the pass size and change of flow direction at the end of each pass is governed by leaving the appropriate plate port Q unblanked. If there are N active plates, there will be $N+1$ total flow passages, with an additional plate at each end to contain the outer flow passages.

When multiple duties are required in a single machine, a number of individual heating or cooling sections can be provided by the use of connector plates, as shown in Fig. 5. These consist of a grid fitted with four corner bosses B, by which service fluids can be connected to the plate pack at intermediate points. By this means complex duties such as those involving recuperation, heating, and cooling can be incorporated in one frame.

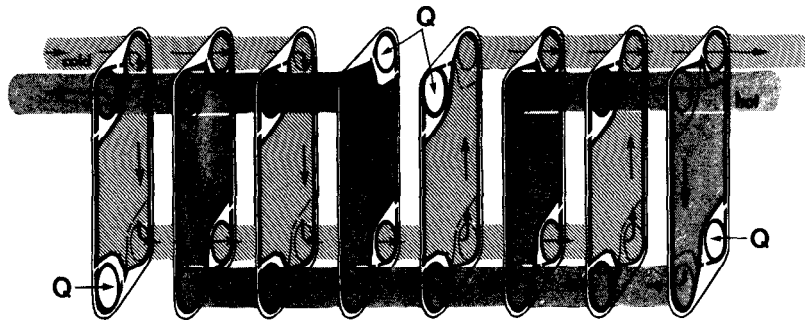


Figure 4 Two-pass/two-pass plate arrangement.

D. Plate design features

(a) Corrugations

Apart from the problems of thermal performance, PHEs may have to withstand pressures of 2 000 kN/m² or more which, besides involving gasket sealing problems, can result in a considerable differential load across the plate if applied to one liquid only. Therefore the corrugations, whose section is determined largely by the required thermal characteristics, must also be designed so that the correct flow gap is maintained between the plates under conditions of maximum differential pressure. The strength of the corrugations depends on the following:

The load at the contact points between the corrugations of two adjacent plates

The stress and deflection in the corrugations themselves

The properties of the materials from which the plate is to be pressed, including the effect of temperature on strength, where relevant

The thickness of the plate blanks, which is an important factor in cost

The two basic types of corrugations are as follows:

1. Chevron or herringbone, which are pressed obliquely across the plate to the same depth as the plate spacing as shown in Fig. 2. The angle of obliquity β is reversed on adjacent plates so that when assembled, the corrugations cross and provide a large number of contact points X, which maintain the plate spacing (see Fig. 6a) and give both rigid support and mechanical strength. This enables high operating pressures to be obtained in relatively light gauges so that this type is particularly suitable for production in expensive materials. The comparatively shallow depth facilitates pressing, and these points together with its adaptability to thermal mixing (see Sec. 3.6.11) makes the chevron trough the most common type in current use.

2. The interlocking or washboard type shown in Fig. 3 has transverse corrugations deeper than the plate spacing. These corrugations align with those on the adjacent plate, the plate spacing being maintained by dimples D pressed into the crests and troughs of the

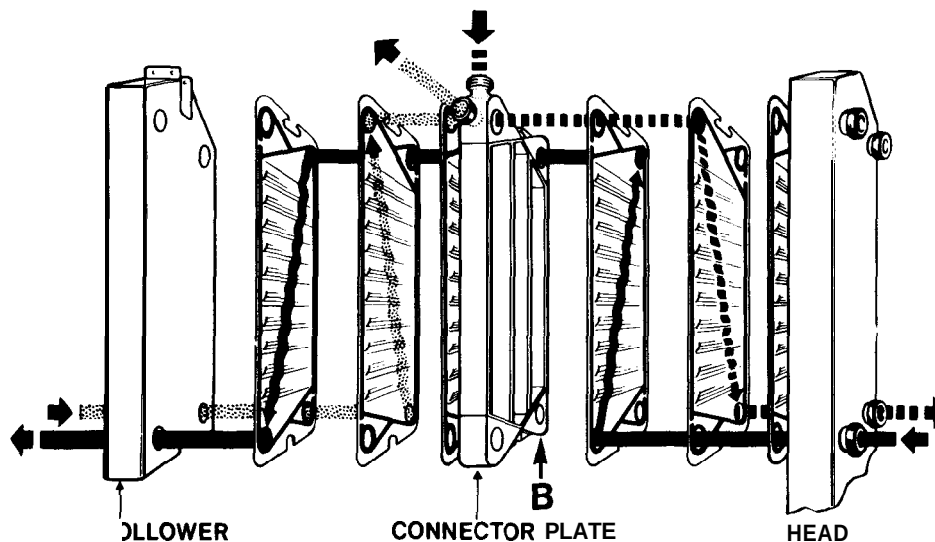


Figure 5 Use of a connector plate for multisection duties.

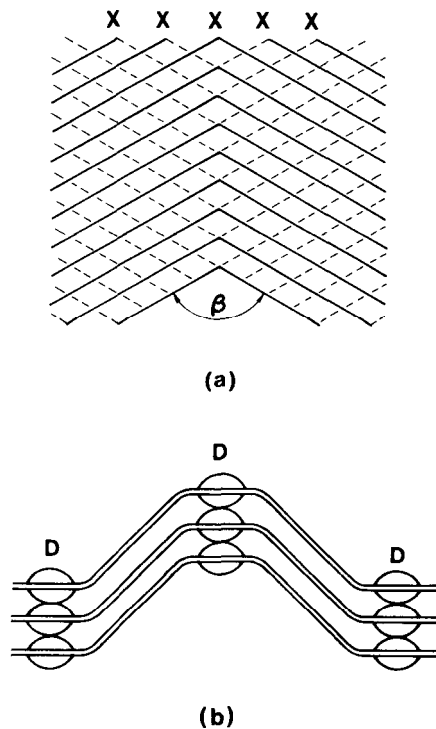


Figure 6 Chevron and intermingling troughs.

corrugations to provide contact with those on adjacent plates (see Fig. 6b). With fewer contact points and greater depth than the chevron design, the intermingling plate calls for heavier gauges and has lower operating pressures, although it is slightly less prone to blockage by large solid particles.

The thermal and hydraulic characteristics of these corrugations are discussed in Part 3 of this handbook.

(b) Pressure sealing and plate support

Apart from the corrugations, it is important that no unsupported areas in other parts of the plate be large enough to suffer serious deflection as a result of differential pressure. Particular attention must be paid to gasket support, where any local deflection of the plate will reduce the gasket load and therefore its sealing properties. The most sensitive areas, as shown in Fig. 1, are the two lengths of gasket A in the port leakage area, as they are completely unsupported by the port entry grooves B of the adjacent plate. Sometimes support is provided by welding reinforcing castellations C in the grooves themselves, as shown in the transverse section illustrated in Fig. 7a, but it is more usual to press convolutions F in the adjacent areas and around the port periphery as shown in Fig. 2. These are staggered on adjacent plates so that when the plates are compressed together they contact one another and thereby withstand the gasket load. Figure 7b shows a

typical section through three plates across the gasket groove in this area, and it indicates how the convolutions F support the flow space B and at the same time maintain the sealing load on the gasket A. Staggered convolutions are also normally pressed along the outside edge of the plate to support the peripheral gasket groove (see Fig. 2).

Frames and connector plates that withstand the plate load must likewise give adequate support in any areas of mutual contact.

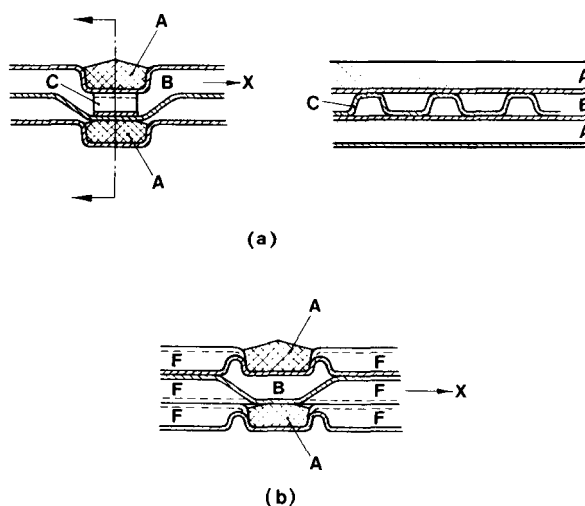
(c) Unblanked ports

In a plate pack, the majority of plates have all four corner ports removed by a blanking operation, thus providing open ports for through flow. However, the end of a pass is marked by not removing the corner port, and such unblanked ports are thus subject to a differential pressure of twice the pressure drop down the pass. Above a certain port diameter some form of strengthening is required, which can take the form of pressed ribs or on larger plates separate disks of increased thickness welded to the port blank.

E. Gasket design and properties

(a) Gasket sections

The gasket section must give a good fit in the gasket groove and normally possesses a peaked profile. It is designed to give approximately 20–30% deformation of its uncompressed thickness, and this is sufficient to ensure a unit load high enough to overcome the internal leakage pressure without leading to local plate distortion.



X indicates direction of flow

Figure 7 Methods of plate support.

Table 1 Compression set for typical rubbers at 142° C

Medium nitrile :	10% after 24 h 40% after 15 d
Fluorocarbon:	3-5% after 24 h 8-10% after 15 d

(b) Mechanical properties

Mechanical properties are adversely affected by temperature, and in all cases the magnitude of the change attributable to thermal deterioration, as a function of time, must be minimal. The important considerations are the following:

1. **Compression set**, defined as the permanent deformation assumed by the gasket, expressed as a percentage of the applied compressive deflection under which it has been maintained for a **fixed** period of time at an elevated temperature. Typical values are shown in Table 1.

2. **Thermal aging**, which must produce minimal changes in the mechanical properties, such as tensile strength and hardness (typically specified at 75-80 IRHD).

(c) Gasket formulations

The polymer types commonly used for gaskets are given in Table 2, but it is the actual formulation of these polymers into rubber compounds that governs the attainment of optimum mechanical and physical properties.

F. Pressability and materials

Any plate must be designed to be pressed in the entire range of materials and thicknesses considered necessary for its market potential, and **all** profiles and sections (e.g., corrugations, support convolutions, gasket grooves) must avoid undue strain. Corrugations in the heat transfer area are normally based on ratios of developed/projected areas varying from 1.3 in the thicker gauges (i.e., 1.0 mm) down to 1.15 in the lighter gauges (i.e., 0.6 mm), whereas the pitch/depth ratios of chevron troughs normally lie between 3 and 4.

Because of the virtual absence of welding and the low cost involved in pressing, **PHEs** are particularly economic in the more expensive corrosion-resistant materials. Austenitic stainless steel presents few pressing problems, as it has good work-hardening properties that minimize localized thinning as it is stretched. This does not apply to titanium, which is much less ductile and sometimes requires special lubricants for satisfactory pressing. Recent work on cold forming is providing data that are making pressability more predictable [1-3], and a number of commonly pressed alloys together with their typical mechanical properties are given in Table 3. Carbon steel is not included, as it is virtually unused because of practical and economic reasons.

G. Frame construction

A plate heat exchanger frame is based on the following units, as shown in Figs. 8 and 9:

Table 2 Gasket material applications

Rubber type and max. temp.	Suitable for	Unsuitable for
Styrene-butadiene (SBR) 87° C	General-purpose rubber for a wide range of aqueous solutions, dilute acids, etc.	Strong acids and many organic liquids
Acrylonitrile-butadiene (nitrile) 137° C	Aqueous liquids, dilute acids, fats, and mineral oils	Strong oxidizing solutions, strong acids, polar organic solvents
Isobutylene-isoprene copolymer (butyl) 152° C	Organic liquids, such as aldehydes, ketones, inorganic acids, and organic acids	Fats, oils, aromatic hydrocarbons, and concentrated mineral acids
Ethylene-propylene rubber (EPDM) 152° C	Similar chemical resistance to butyl but frequently superior	Fats, oils, aromatic hydrocarbons, and concentrated mineral acids
Fluorocarbon (Viton ^b) 177° C	A wide range of organic liquids; fats and oils (at high temperatures) and some mineral acids	Esters, ketones
Silicone	Specific chemical duties and low -temperature applications	Many organic liquids

^aIn addition, gaskets of compressed asbestos fiber are available that are suitable for organic solvents that attack all available grades of rubber (e.g., esters, benzene, and low-molecular-weight chlorinated hydrocarbons). Maximum temperatures up to 250° C but stronger frames are required because of higher compressive loads.

^bViton: registered trademark of duPont Co. Ltd.

Table 3 Typical material properties for plate pressing

Material	0.1% Proof, MN/m ²	Total elongation, %	UTS, MN/m ²
Stainless steel 3 16	275	55	600
Commercially pure titanium	200	40	350
Incoloy 825 ^a	250	40	650
Hastelloy C276 ^b	390	55	820

^aRegistered trademark of the International Nickel Co., Inc.

^bRegistered trademark of The Cabot Corp.

1. Two vertical members, an end support E and a fixed member (or head) H, which carries four liquid connections or nozzles C

2. Top and bottom bars BB, carried between these vertical members, which support and guide the plate pack P

3. A moving member (or follower) F, which compresses the plates against the fixed member H, and which can also carry additional liquid connections

4. A means of clamping the plates between the fixed and moving members H and F

This clamping is normally done one of two ways:

1. The fixed and moving members are compressed together by a number of tie bars T, which thereby leave the end support and top and bottom bars un-

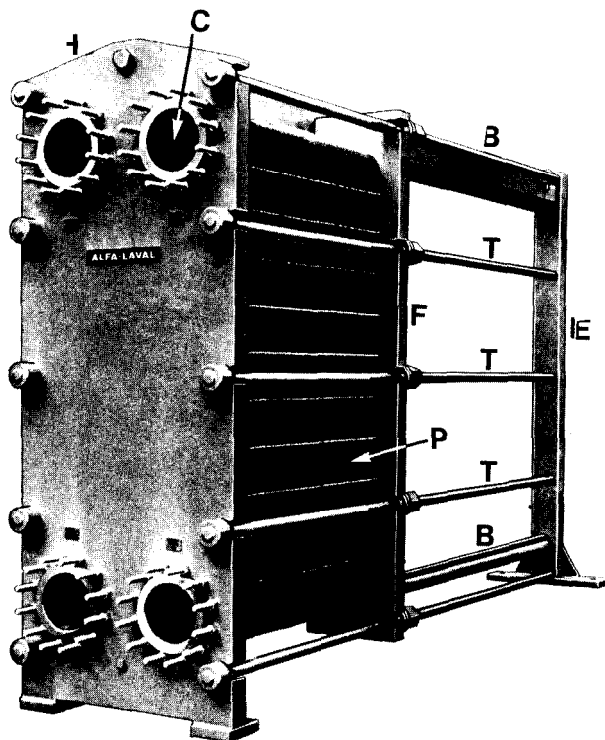


Figure 8 Industrial frame with tie bar tightening.

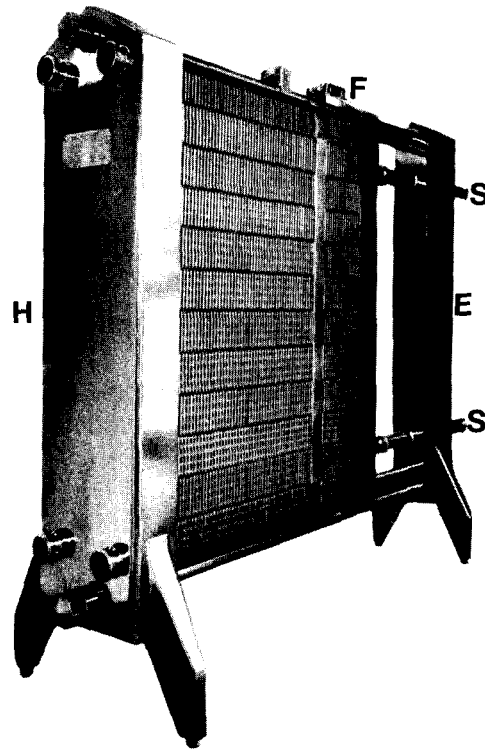


Figure 9 Stainless steel-clad hygienic frame with screw closure.

stressed, except for the effect of the weight of the plates and their contents (see Fig. 8).

2. The compressive load is applied to the moving member by one or two tightening screws S carried by the end support, which is therefore itself stressed together with the top and bottom bars (see Fig. 9). This is a more expensive and less stable construction because the tightening screws are in compression, but it makes opening and closing of the frame easier and this can be further facilitated by some means of power tightening, electric or hydraulic.

In the food industry the use of chemical in-place cleaning methods has reduced the need for frequent opening, and most modern frames use the construction based on tie bars operating in tension. Liquid connections can be cast or fabricated and are fitted with appropriate flanges or screwed fittings; rubber-lined bushes are also available. Stainless steel-clad frames can be provided for hygienic duties, as shown in Fig. 9.

H. Frame loading analysis

Strain as well as stress is important in frame design because excessive deflection will relieve the compressive load on the gaskets and allow leakage to occur. This will become more significant as the size of the plate pack is reduced and the effect on individual gaskets increased.

The main load considerations in frame design are therefore as follows:

1. Stress and deflection in the head, follower, and end support due to the hydraulic and tightening load
2. The resulting tensile stress in the tie bars or compressive stress in the tightening screws
3. Stiffness of the top carrying bar, which has to take the weight of the plates and their liquid contents to keep them clear of the bottom guide bar
4. Lateral strength of the top and bottom bar to ensure stability of the plate pack in a sideways direction.

Special conditions are laid down when frames have to conform to recognized codes such as ASME Section III, Subsection ND, or when they are subject to marine installation or earthquake loading.

The disruptive load on the frame consists of three components:

1. The hydraulic load P_H , which is a function of the operating pressure and the projected plate area within the gasket
2. The gasket load R , which acts as a low-rate spring
3. The metal to metal contact load M between the plates, which acts as a high-rate spring during the last stages of tightening

The application of the hydraulic load P_H causes some deflection of the frame members and tightening mechanism and thus relieves the loads R and H , so that the total load under pressure is less than $P_H + R + M$. A simplified force system is shown in Fig. 10, based on assumed linear moduli of compression for gaskets and plates.

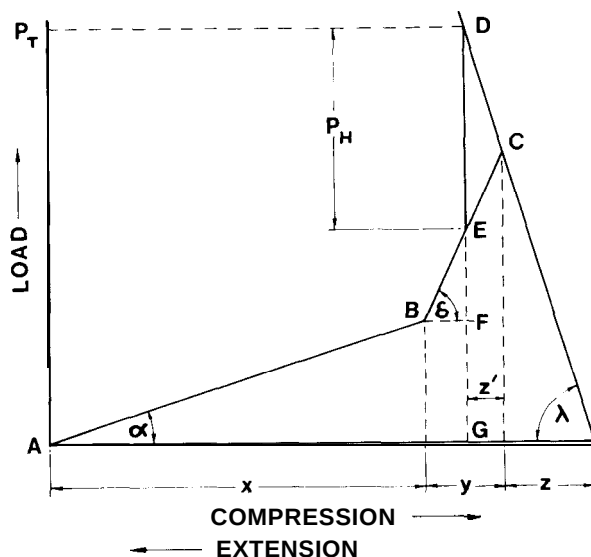


Figure 10 Frame loading diagram.

1. Initial gasket compression x with modulus α is represented by line AB .

2. After compression x , the combined gasket and metal to metal compression starts at B where metal contact first takes place, so that the line BC is followed with modulus δ .

3. Equilibrium is reached at point C after further compression y to the correct plate pitch, where the frame and tightening system with modulus λ have extended by z .

4. The application of the hydraulic load P_H extends the tightening system and reduces the plate compression by z' giving a point of equilibrium at D , so that the total load is given by

$$P_T = DE + EF + FG = P_H + \delta(y - z') + \alpha x \quad (1)$$

The maximum frame loading occurs under fully balanced hydraulic pressure, which implies that both process streams are pressurized equally to the maximum value, thereby resulting in zero differential pressure across the plates. Fully unbalanced pressure with maximum pressure on one side of the plates and zero on the other gives a slightly lower frame loading than the balanced arrangement. The plates, however, are subject to full differential loading, resulting in maximum stress and deflection and less favorable sealing conditions.

Pressure differential can also affect multisection machines; if one section is at a much higher pressure than another, the low-pressure plates will tend to compress under the unbalanced load and reduce the sealing ability of the high-pressure section. The larger the low-pressure section compared to the high-pressure section, the more pronounced the effect can be, but well-designed, adequately supported plates will normally avoid trouble of this nature.

PHEs are normally designed for test pressures up to 50% higher than the maximum working pressure. Some manufacturers make a point of designing their plates to withstand the quoted pressure under fully unbalanced conditions, whereas others qualify their specification by giving a maximum pressure that applies to the frame only and a lower figure for an unbalanced pressure applied to the plates.

I. General summary

Table 4 gives the basic data relating to the mechanical details of PHEs, although it must be appreciated that these figures are only representative of PHEs in general rather than of any particular specification, and not all the factors quoted can be combined in a single design. The main features of PHEs can be summarized as follows:

Table 4 Typical PHE data

Individual plate heat transfer area	0.03–2.5 m ²
Maximum heat transfer area per frame	1 500 m ²
Maximum working pressure	2 000 kN/m ²
Maximum working temperature	
Synthetic rubber gaskets	177° C
Compressed asbestos fiber	250° C
Maximum port size	400 mm
Maximum flow rate ^b	0.7 m ³ /s
Plate thickness	0.6–1 .0 mm
Plate flow gap ^a	1.5–5 mm
Compressed plate pitch	2–6 mm

^aThe flow gap represents the theoretical size of the largest particle that will flow down the plate passage.

^bBy feeding through ports at the opposite ends of the machine, the maximum flow rate can be doubled.

Accessibility
 Compactness and low volumetric capacity
 Flexibility and ease of modification
 High thermal performance giving over 90% recuperative efficiency
 Multiple duties in a single machine
 Low cost in expensive corrosion-resistant materials

NOMENCLATURE FOR SECTION 4.4.2

M	metal-to-metal plate load on frame	z	tie bar extension
P_H	hydraulic load on frame	z'	tie bar extension
P_T	total load on frame	α	modulus of compression
R	gasket load on frame	β	angle of obliquity of chevron troughs
x	platage compression	δ	modulus of compression
y	platage compression	λ	modulus of extension

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2. Painter, M. J., and Pearce, R., Formability of High Strength Low Alloy Steels, *Met. Technol.*, vol. 2, pp. 62-65, 1975.
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4.4.3

Plate fin heat exchangers

R. L. Webb

■ The plate fin heat exchanger is similar to the plate-type exchanger (Sec. 4.4.2), except that extended surface is used between the two separating sheets as shown in Fig. 1. A unique feature of plate fin heat exchangers is their high heat transfer surface area per unit volume. Heat transfer surface area of $1\,300\text{ m}^2/\text{m}^3$ of volume is not uncommon. For this reason they are commonly called compact heat exchangers, although sometimes in the United Kingdom the term matrix heat exchanger is used.

There are two important subclasses of plate fin heat exchangers, the plate fin-plate fin construction and the plate fin-tube construction. The plate fin-plate fin construction, which consists of a number of laminations of the elements in Fig. 1, is normally used for gas-to-gas applications. The plate fin-tube construction, which has tubes normal to the plate structure, is used for gas-to-liquid or gas-change of phase applications; the liquid

or two-phase mixture flows through the tubes. The passenger car radiator is an example of this construction.

In this section we describe first the construction of plate fin-plate fin exchangers, and then, in Sec. D, the construction of plate fin-tube exchangers. A detailed description of the extended surface and additional descriptive information is provided in Part 3 of this handbook.

A. Plate fin-plate fin construction

The most common construction materials are aluminum and steel. The aluminum design employs an all-brazed construction and the steel type may use a combination of brazing and welding. Figure 1 shows the basic components of a brazed aluminum plate fin exchanger. This consists of the corrugated plate fin (1) bonded to

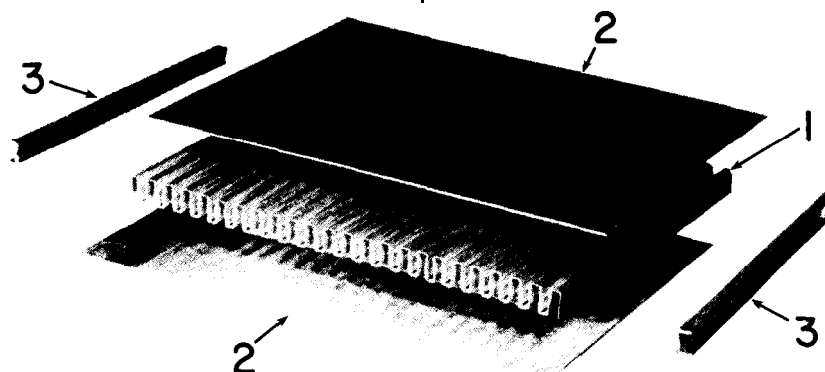


Figure 1 Basic components of plate fin stack. Courtesy of the Trane Co.

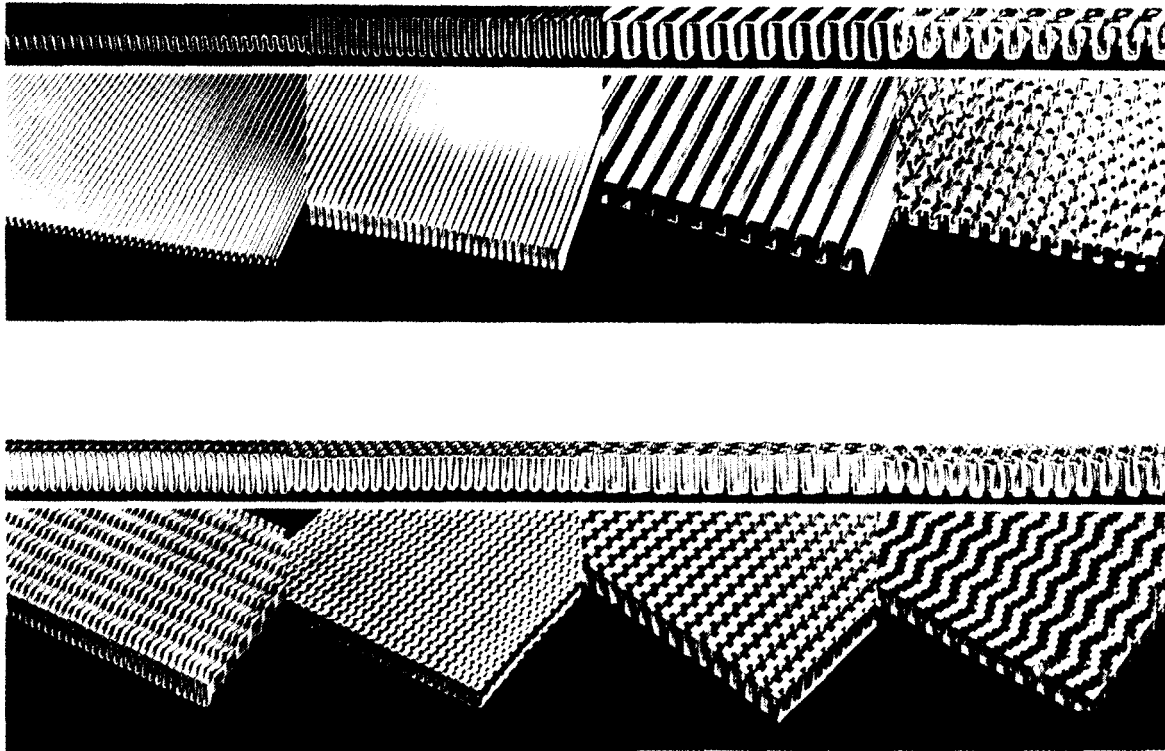


Figure 2 Typical plate fin surface geometries used in brazed aluminum heat exchangers. Courtesy of the Trane Co.

the separating sheet (2) and sealed with extruded side channels (3). Typical plate fin geometries are illustrated in Fig. 2. The heat exchanger core assembly is formed by stacking multiple layers of the plate fin and separating sheets. Figure 3 shows a cross-flow gas-to-gas design, and Fig. 4 shows a core designed for cryogenic service with four fluids. Figure 5 shows the final heat exchanger assembly of the core illustrated in Fig. 4. The fluid inlet and exit manifolds are welded to the core after the brazing operation. Such a heat exchanger assembly may be as large as 1 X 1 X 13 m in size. By connecting a multiple of basic core assemblies in a parallel and series network, a large heat exchanger battery may be assembled. Figure 6 illustrates part of a multicore battery.

Figure 7 shows a counter-flow core assembly in a stainless steel exchanger used as a gas turbine recuperator [1]. The core assembly differs substantially from the previously described brazed aluminum design. In the recuperator, a combination of brazing and seam welding is employed, whereas no welding is used in the aluminum core assembly of Figs. 3 and 4.

The internal design of the recuperator core is shown in Fig. 8. The high-pressure inlet air enters the circular manifolds and passes through the small core passages. The turbine exhaust gases pass in counter flow through the larger passages of the core matrix. High-performance offset strip fins are used for both the air and gas flows. The subassembly formed in the brazing operation is

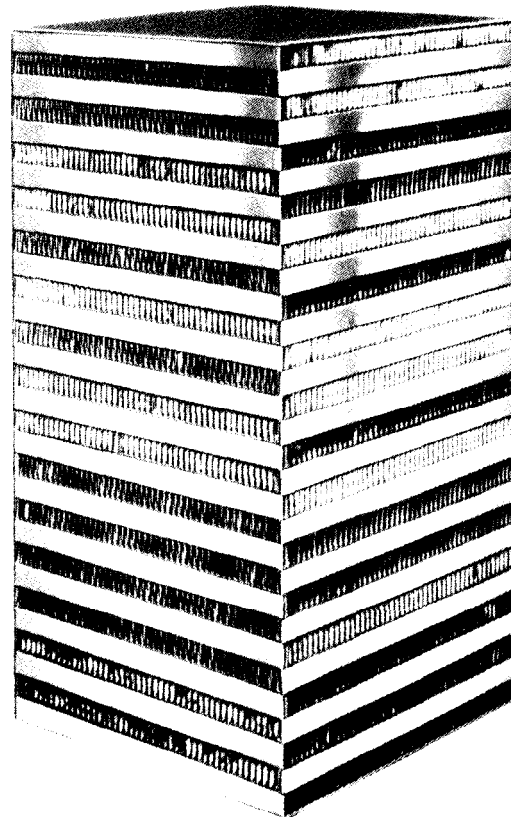


Figure 3 Brazed aluminum core for cross-flow gas-to-gas service. Courtesy of Modine Manufacturing Co.

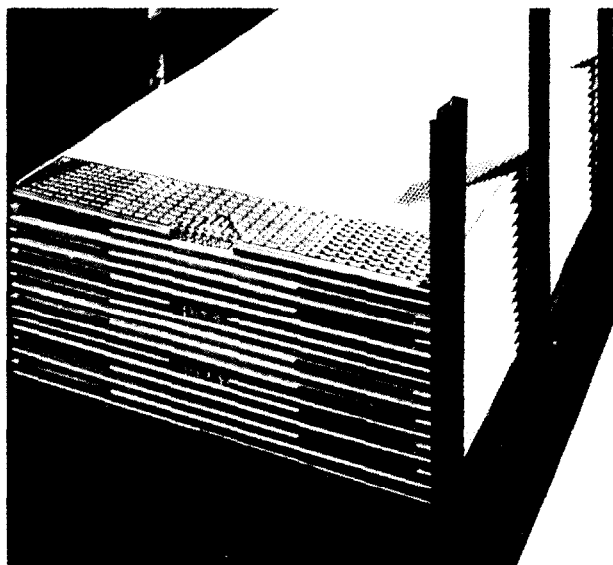


Figure 4 Core assembly of four-fluid brazed aluminum design for cryogenic service.. Courtesy of the Trane Co.

denoted in Fig. 8 and shown in Fig. 9. Two U-shaped rings are edge welded to alternate tubesheets to form the gas closure around the gas manifolds, as shown in Fig. 9. The gas and air closures are made by welding, after brazing of the subassembly. The core assembly of Fig. 7 is made by stacking the completed subassemblies of Fig. 9. Additional edge welding, around the circumference of the air manifold holes, is performed during the subassembly stacking operation. A complete recuperator, consisting of two core assemblies, is shown in Fig. 10. A recuperator designed for 45.4 kg/s and 83% effectiveness is 3.0 m wide, 3.8 m high, and 5.5 m long.

Plate fin exchangers may be made in a variety of sizes and shapes. The designs discussed here are intended to illustrate basic construction methods only.

B. Flow distributors

Special flow distributors are not required for cross-flow designs as illustrated by Fig. 3. However, when fluids enter the exchanger via a pipe or manifold, specially oriented fins are required to distribute the flow evenly across the width of the finned passages. Figure 11 illustrates a popular flow distributor concept used in the brazed aluminum design. A similar type of flow distributor fin is used for the manifold recuperator inlet air, Fig. 7. Here, plain fins distribute the air from the air tubes across the core width. Part of the fins in the distributor section are not for flow distribution, but provide structural integrity. A different type of flow distributor is evident at the end of the core stack in Fig. 4. Here, perforated fins are oriented 90° to the inlet fluid. The high pressure drop, in the flow direction, forces the fluid to distribute itself across the width of the fin passage.

C. Brazing

Two basic brazing methods are employed in construction: salt bath and furnace brazing. Several methods may be used to introduce the brazing alloy.

In the brazed aluminum design, the separating sheets are clad with a brazing alloy. This specially clad separating sheet may consist of an aluminum base metal, with a very thin clad layer containing the braze alloy. A typical clad alloy may be 92.5% aluminum and 7.5% silicone [2]. The silicon reduces the melting point of the cladding approximately 50° C below the melting point of the base aluminum alloy. When the assembled core is submerged in the molten flux bath, the brazing alloy melts and bonds the separate components into

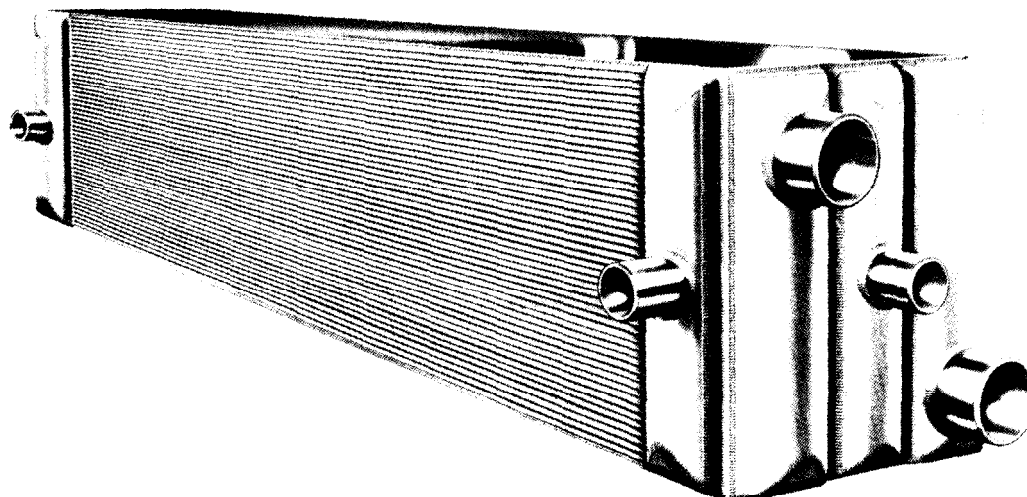


Figure 5 Final heat exchanger assembly of Fig. 4. Courtesy of the Trane Co.

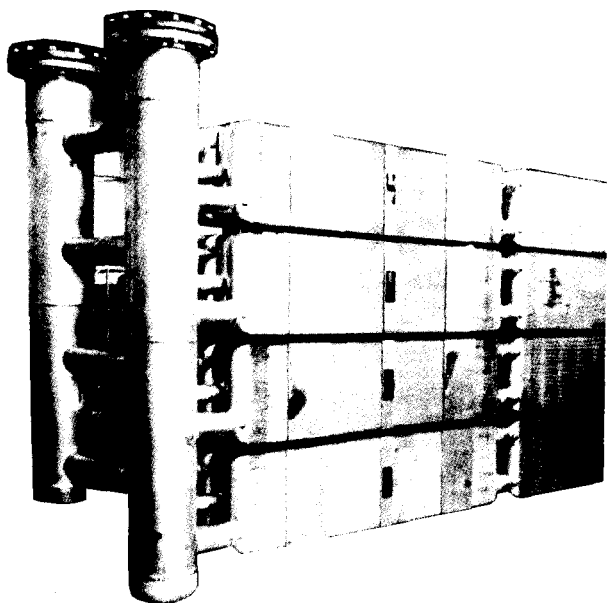


Figure 6 A typical brazed aluminum multicore battery. Courtesy of the Trane Co.

an integral structure. Strong fillet-brazed joints are formed between adjoining components.

The salt bath brazing method requires a number of process steps for successful completion. Prior to core assembly, all parts must be cleaned to remove surface oxides and grease films. After preparatory cleaning, all parts are run through a series of liquid immersions for degreasing and finally flushed with clean water. Once brazing is complete, the cores are steam cleaned,

repeatedly immersed in a **dichromate** solution to remove the brazing flux, and then flushed with clean water. The maximum core size is limited by the physical dimensions of the salt brazing bath.

The stainless steel recuperator of Figs. 7 through 11 does not use brazed clad separating sheets. Rather, the braze alloy is added as a powder or as a film tape. The high-pressure air-side **fins**, which are under tension, are brazed with a nickel alloy. The low-pressure compression-loaded gas fins are copper brazed.

Some manufacturers are turning to furnace vacuum brazing as an alternative to the salt bath brazing method. Vacuum brazing requires fewer processing steps, does not use a flux, and the vacuum environment provides excellent control over surface oxide removal. Problems associated with the postbrazing flux removal do not exist with a vacuum brazing method. This is an important advantage, since flux retained in the core may cause future corrosion problems as well as allow blockage of the fin passages.

Ultrasonic soldering is employed on some all aluminum designs used in the air conditioning industry. The ultrasonic soldering process is performed in an immersion bath and does not require the use of flux for oxide removal.

D. Plate fin-tube construction

This exchanger type transfers heat between a gas and a liquid or a two-phase fluid. Typical applications are automotive radiators, air conditioning evaporators,

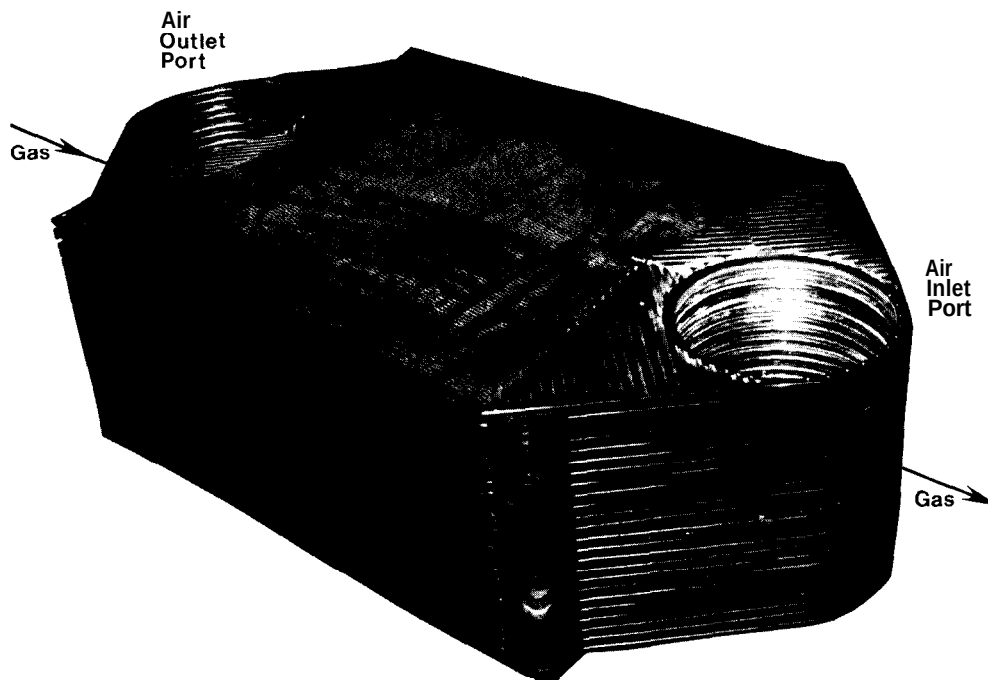


Figure 7 Basic core assembly for a stainless steel gas turbine recuperator. Courtesy of the Harrison Radiator Division.

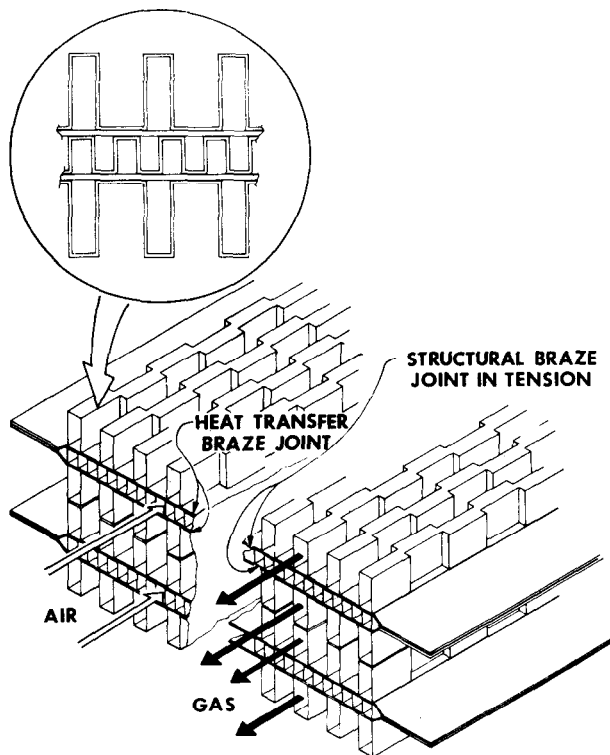


Figure 8 Internal details of recuperator core assembly. Courtesy of the Harrison Radiator Division.

and condensers and liquid or oil coolers. All aluminum designs use assembly and brazing methods similar to those of the previously discussed gas-to-gas designs. The distinguishing feature of the plate fin-tube design is the construction of the liquid or two-phase heat transfer flow passages.

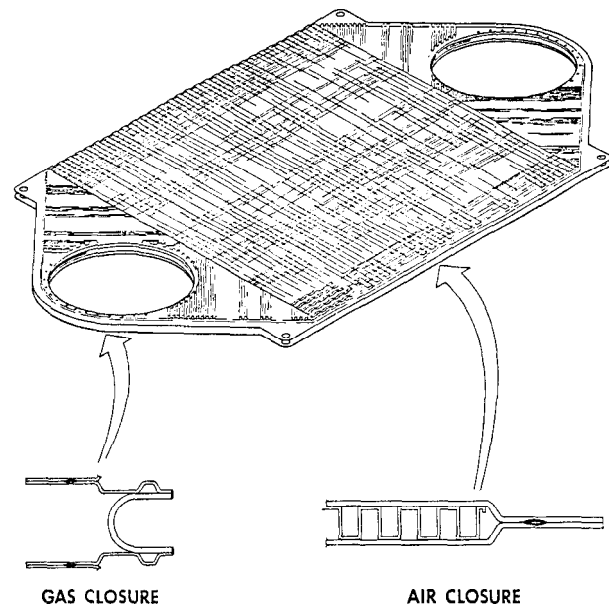


Figure 9 Recuperator subassembly formed in brazing operation. Courtesy of the Harrison Radiator Division.

Figures 12 and 13 show two all-aluminum design types used for air coolers. In Fig. 12, the tube-side passages consist of flattened aluminum tube extrusions [3]. "Turbulator" fins are installed and brazed to the inner surface of the flattened tube. Figure 13 shows the drawn cup oil cooler construction [4]. The oil side contains turbulator strips to provide high heat transfer coefficients on the oil side. The turbulator strips consist of offset strip fins whose fin passages are oriented 90° to the flow direction. The surfaces of the

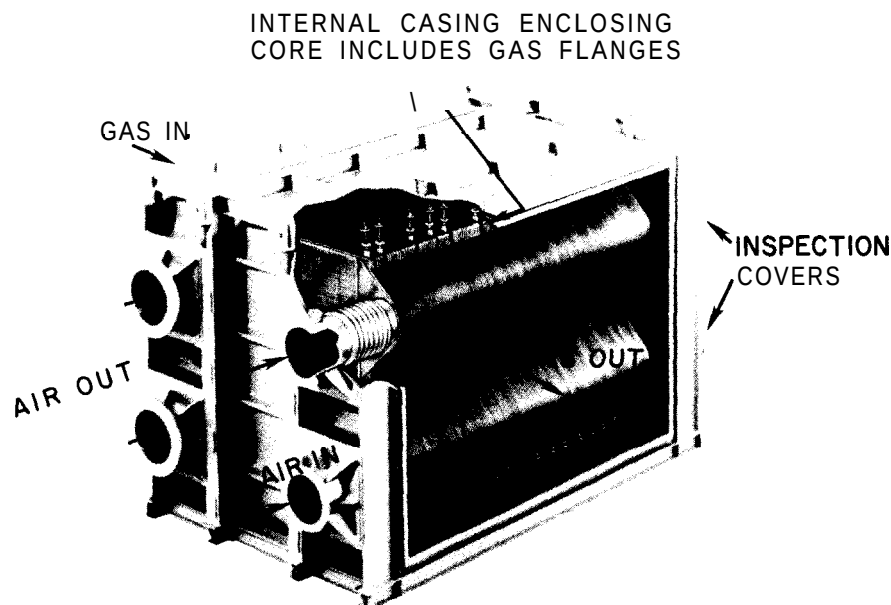


Figure 10 Completed recuperator, consisting of two core assemblies. Courtesy of the Harrison Radiator Division.

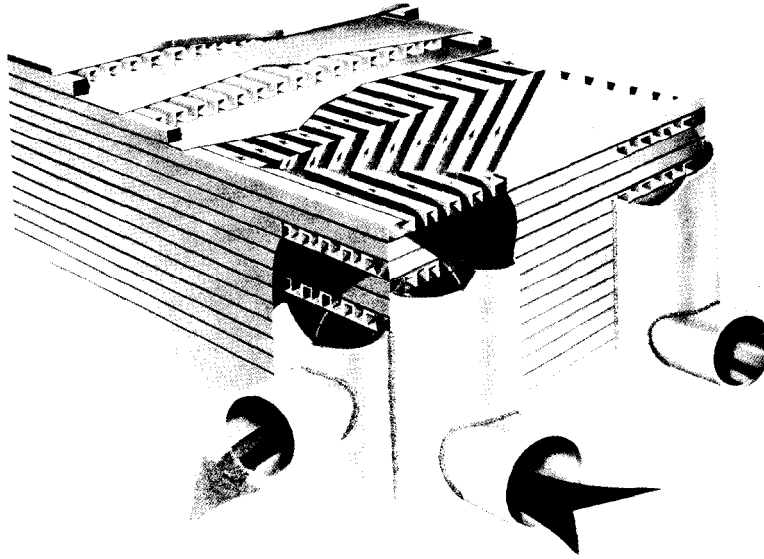


Figure 11 Core assembly showing distributor fins. Courtesy of the Trane Co.

drawn cups are clad with braze alloy, and the entire assembly is either salt bath or furnace brazed.

Figure 14 shows a radiator construction typical of that used in passenger cars. In this design, 0.09-mm brass tubes are soldered to 0.06-mm copper fins.

Figures 15 and 16 show brazed aluminum designs used in air conditioning equipment. Figure 15 is an evaporator used in an automotive air conditioner. It uses the drawn cup construction of Fig. 13. Figure 16

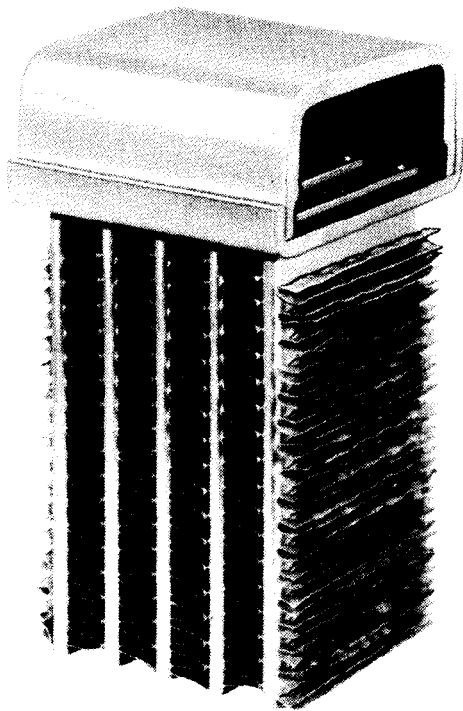


Figure 12 Section of brazed aluminum oil cooler. Courtesy of Modine Manufacturing Co.

is a partial section of the condenser used in automotive or residential air conditioning systems. The flattened tubes are aluminum extrusions, which provide extended area internal surface. Note the webbed full-height fins, which are used for strength requirements.

Figure 17 shows a quite different type of plate fin-tube construction. This figure, taken from a U.S. patent by O Connor and Pasternak[5] shows a flattened tube having integral aluminum fins. The finned tube is made from an all-aluminum extrusion and the fins are machined into the extrusion. Fin formation is accomplished using a modified high-speed punch press, which slits the extrusion at a small angle and lifts a chip to form the fin without creating any scrap metal. The

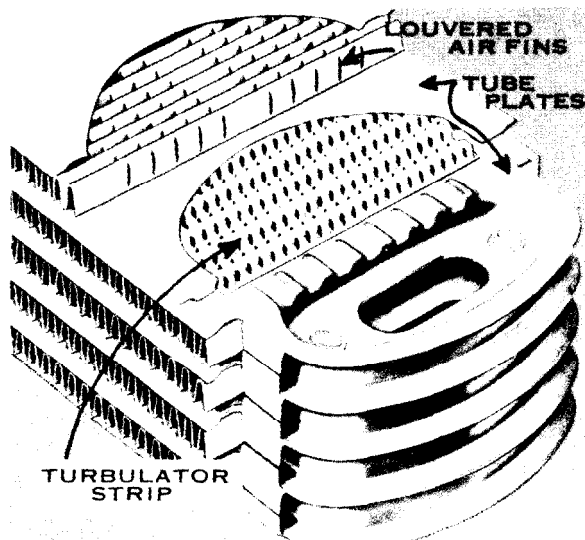


Figure 13 Drawn cup oil cooler. Courtesy of the Harrison Radiator Division.

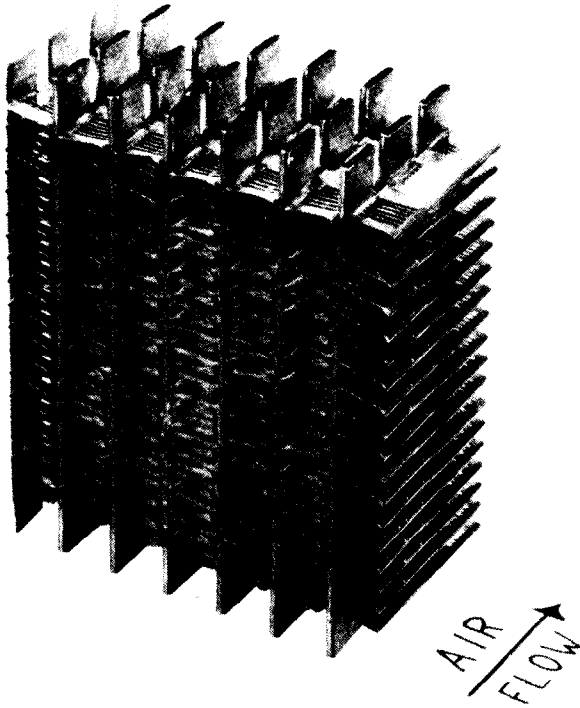


Figure 14 Section of radiator used in automotive applications. Courtesy of Modine Manufacturing Co.

extruded tube is formed with a number of internal webs, which provide structural strength and permit relatively high tube-side design pressures, e.g., 2 000 kN/m². A wide variety of preformed extrusions are possible, which permit various numbers and shapes of internal passages, and various shapes and sizes of internal

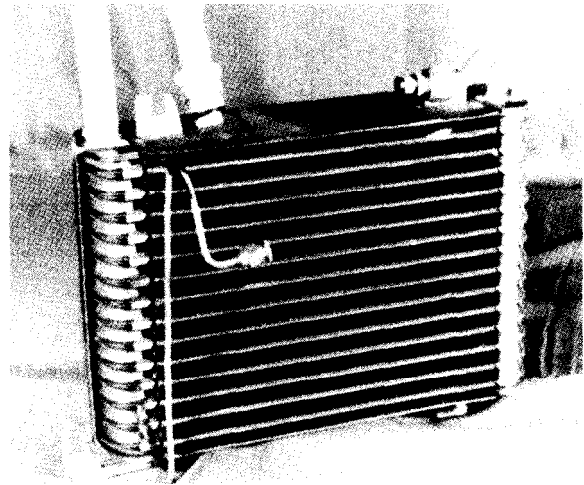


Figure 15 Drawn cup automotive air conditioning evaporator. Courtesy of the Harrison Radiator Division.

and external fin geometry. The extrusion can be made in any length, which permits direct formation of a heat exchanger geometry similar to that shown in Fig. 16. Note that brazing would not be required using this method.

E. Materials

Primary construction materials are aluminum, carbon steel, and stainless steel. The choice of construction

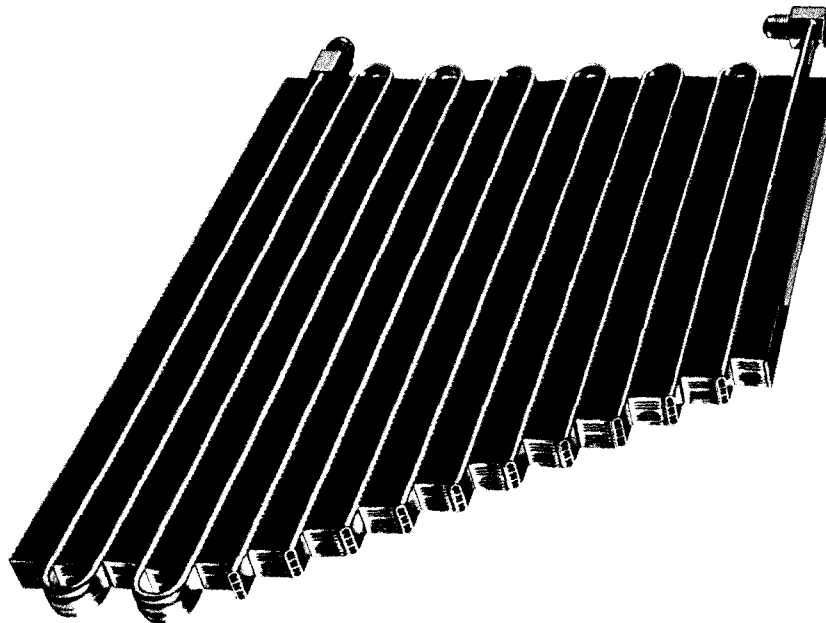


Figure 16 Partial section of refrigerant condenser showing use of extruded aluminum tubes. Courtesy of Modine Manufacturing Co.

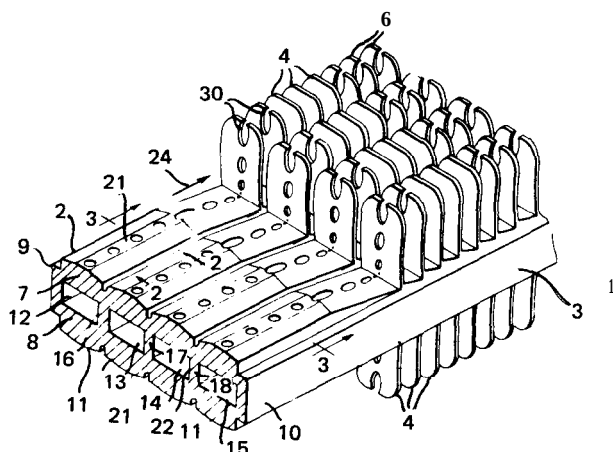


Figure 17 Multiport extruded plate fin-tube. Courtesy of Curtis-Wright Corp.

material is dictated by the pressure and temperature design limits, and the corrosion potential. In the absence of corrosive fluids, the high thermal conductivity of aluminum will provide the lowest-cost heat exchanger. Aluminum is well suited for temperature ranges from cryogenic levels up to 250°C. Carbon steel and stainless steels are used from 250°C up to 650°C. Carbon steel is limited to 480°C. High-temperature and corrosion-free operation favor stainless steels at high temperatures. Copper is suitable for brazed construction and offers ideal thermal properties. However, its use is limited to corrosive environments, where aluminum is unsuitable. Many automotive radiators use copper or copper alloys.

For very high temperatures, above 650°C, compact heat exchangers may be made from ceramic or carbon materials. These are made by special casting techniques and will not be discussed further in this section.

Brazed aluminum plate fin-plate fin designs are normally limited to design pressures below 2 000 kN/m². However, the use of special fin designs may allow moderately higher design pressures. Tube-side design pressures for plate fin-tube type construction are also normally limited to about 2 000 kN/m². Special multiport extruded tube designs may allow moderately higher design pressures. Higher design pressures are possible with stainless steel construction. As a general rule, brazed plate fin exchangers are limited to lower design than is possible with circular tubes and circular pressure vessels typical of shell-and-tube heat exchanger designs.

F. Design for mechanical strength

The basic approach to mechanical design will be illustrated for brazed aluminum designed to meet ASME code requirements. Critical areas of the design are

strength of the brazed core stack, and where noncircular manifolds are welded to the core assembly. Normal design calculations are semiempirical, and justified by burst test (at five times design pressure).

After brazing, the aluminum is fully annealed, so allowable stress must be based on the yield strength of the alloy in its fully annealed state.

The separating sheets are under tension, and the required thickness depends on their separation spacing and the design pressure. For tension-loaded fins, the required fin thickness depends on the fin spacing and design pressure. For fixed fin spacing, thicker fins are required to accommodate increasing design pressure.

The range of fin thickness commonly used is 0.25–0.66 mm. The use of thicker fins will cause increased gas-side pressure loss for the commonly used offset strip tin. In reality, the required fin thickness will be greater than that based on a tension-load calculation. This is because the fin will fail at its foot, where the material is thinned during formation of the corrugated fin array.

High differential pressures must recognize the possibility of failure of the low-pressure side fins due to compression loading.

The second critical design condition occurs where a semicircular manifold pipe is welded to the core structure. It may be necessary to weld on reinforcing plates to meet the design pressure.

After completing the above design calculations, small sample sections of the core assembly and welded manifolds are built and their burst pressure determined. The design is then modified, based on the observed failure. Equipment designers have developed proprietary factors to account for inaccuracies of the simplified procedure described here.

G. Service limitations

Service limitations imposed by design pressure, temperature, and corrosion potential were discussed in the previous section. Normally brazed aluminum is limited to the use of clean, noncorrosive fluids. Fouling consideration impose an important service limitation. Large brazed aluminum exchangers, which use high-performance fin geometries and distributor fins, are virtually uncleanable. Cross-flow designs of shallow depth may be cleaned by any of several washing methods, including steam cleaning, detergent washing, and solvent soaking. Chemical cleaning may be viewed as a possibility, most probably limited to steel designs.

H. State of the art

Two recently published survey articles provide an excellent description of brazed plate fin design tech-

nology. Butt [6] discusses mechanical design of brazed aluminum plate fin exchangers. Parker and Coombs [7] described a variety of special plate fin heat ex-

changer designs, thus illustrating the breadth of the technology. The designs include stainless steel, aluminum, and ceramic plate fin materials.

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4.4.4

Other types of heat exchangers

I. Murray

A. Introduction

Although the shell-and-tube, the air cooler, and the plate-type heat exchangers are the three most commonly used exchangers in the chemical and process industries, there is still considerable scope for other types of exchangers. Once the thermal duty of a heat exchange process has been determined, the designer must decide which type of exchanger should be used to achieve the required duty. The selection, which must obviously be capable of the required thermal performance, will be influenced by factors such as the capital investment available for the equipment, the space available to install the equipment, the compatibility of construction materials with the process fluid, the quality and quantity of service fluid available, requirements and facilities for cleaning and servicing the equipment, and whether or not the process fluid is temperature sensitive or requires particular treatment. Further consideration must also be given to the environmental effects of the plant both

in the vicinity of the equipment and in the surrounding districts, where the presence of discharge emissions and high noise levels could be unacceptable.

In this section a number of alternative heat exchanger types are described, some of which are limited in their applications and others of which are extremely flexible and suitable for a variety of applications. Most can be designed mechanically using appropriate national standards such as these quoted in [1] and [2].

B. Double pipe

As the name suggests, the double-pipe type of heat exchanger (Fig. 1) consists principally of two concentric pipes, with one fluid flowing along the inner pipe and the other fluid flowing in the annulus formed by the two pipes. Other parts that make up the exchanger are return bends, return head, and fluid inlet connections. These exchangers are normally constructed in standard sections made from standard pipe sizes. They

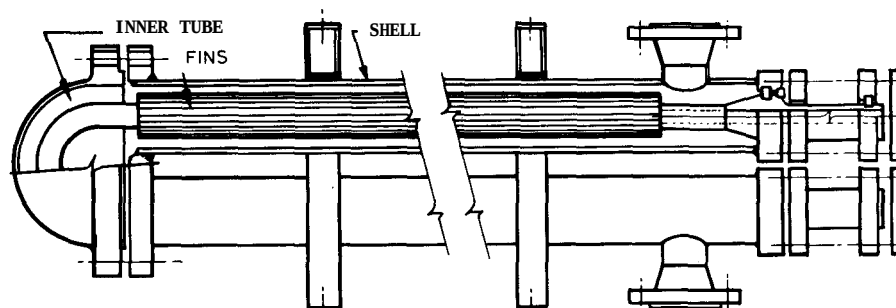


Figure 1 Double-pipe heat exchanger. Courtesy of R. M. Armstrong.

can be constructed in carbon steel, most high- and low-chrome alloys, austenitic stainless steel, Monel, and in fact nearly all other weldable materials. Mechanical design methods for pressure vessels laid down in national standards apply to **this** type of heat exchanger.

In certain applications, corrosion or heat resistance requirements may vary considerably from one end of the exchanger to the other; in such cases, it is possible to construct one section of the exchanger from an expensive resistant material and other sections from less expensive materials. In cases where the heat transfer coefficient of the annular flowing fluid is low, the inner tube can be finned on the outer surface as shown in Fig. 1.

For some applications double-pipe exchangers with multiple inner tubes, with or without finning, can be used. Where necessary, the inner tube, or tubes, are supported by packing glands to prevent sagging, which would result in uneven flow distribution in the **annulus**, thus reducing the performance of the exchanger.

The principal advantages of these exchangers are that they are easily constructed from standard pipe sizes and the duty for which they are used can readily be increased by arranging multiple units in banks connected either in series or in parallel. These are a relatively inexpensive means of providing heat transfer surface [3] where one of the fluids is at high pressure, thus requiring a more expensive unit in the conventional **shell-and-tube** exchanger construction. When bolted connections are used, the exchanger can be easily dismantled for cleaning purposes.

C. Tubular and panel immersion

Tubular immersion exchangers are generally used for heating or cooling process fluids in tanks or reaction vessels, although there is no reason why the primary or process fluid should not be passed through the tube coil if pressure drop and fouling conditions permit. The simplest form of this type of exchanger is a length of pipe bent into a coil or serpentine pattern. In some cases the pipe may be finned, and the coil designed in the form of a bank of tubes with inlet and outlet manifolds. This form of construction is ideally suited for application in tanks with central agitators. Equally, however, the tube coil could be in the form of a horizontally mounted bank of tubes making use of natural convection circulation (Fig. 2). Immersion coil heating is used extensively in chemical reaction vessels where a large amount of heat transfer surface is required to ensure a high degree of temperature uniformity during the reaction. For this application, the tube coil pattern can be quite complex (Fig. 3).

The coils can be manufactured from any suitable

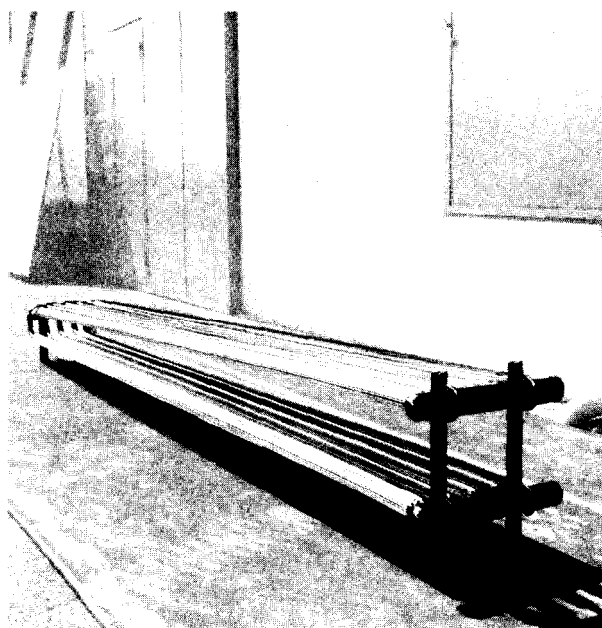


Figure 2 Finned-tube immersion coil. Courtesy of Siirtrec.

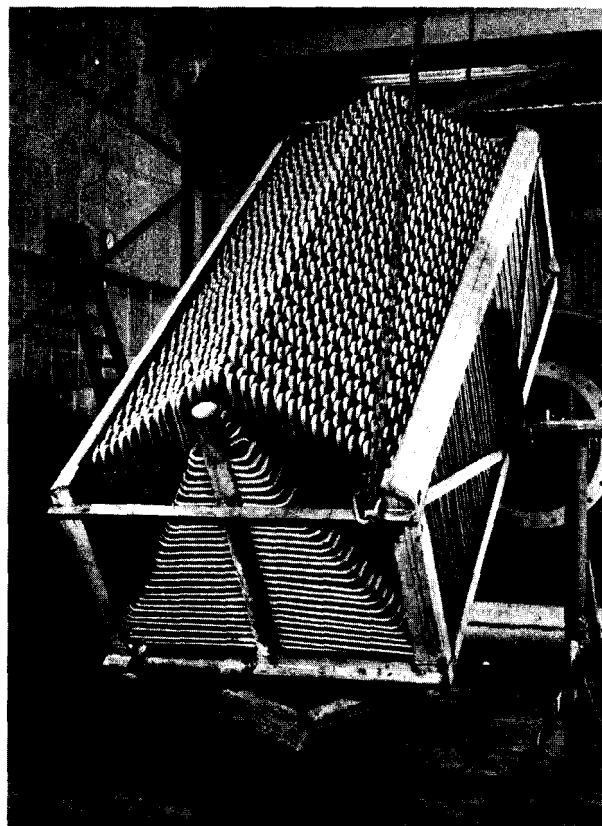


Figure 3 Immersion coil for feedwater preheater. Courtesy of ASME Vessels Ltd.

tube material that can be formed into the required pattern. In recent years a multiple-tube immersion coil made from Teflon tubing [4] has been successfully used in applications where a high degree of corrosion resistance is required (Fig. 4).

Not all immersion heaters are of the coiled tube construction, and in many cases of tank heating, immersion panels are used (Fig. 5). The panels are constructed by combining two embossed panels so that the embossed pattern forms the channels through which the heating or cooling fluid flows. Quite complex flow patterns can be obtained in this manner, provided that effective seals can be achieved between flow channels.

Materials of construction with immersion panels are limited to relatively thin plates which can be pressed to the required pattern and then combined by welding or brazing. In some applications graphite panel immersion heat exchangers have been used where corrosion resistance has been important and process temperatures are within the limits imposed by the impregnant used in manufacturing the exchanger.

The principal advantages of immersion heaters are in applications such as tank or vat heaters, in chemical reaction vessels, and as waste heat recuperators. The shape of the immersion coil is dependent on the physical characteristics of the process tank or vessel rather than on the thermal duty required. Although relatively simple to construct and install, these make cleaning of both the

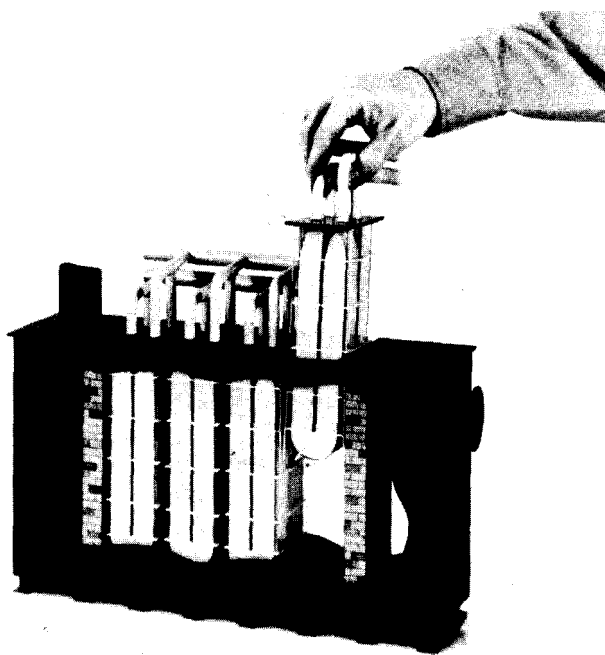


Figure 4 Model of Teflon tube immersion coils. Courtesy of Du Pont (U.K.) Ltd.

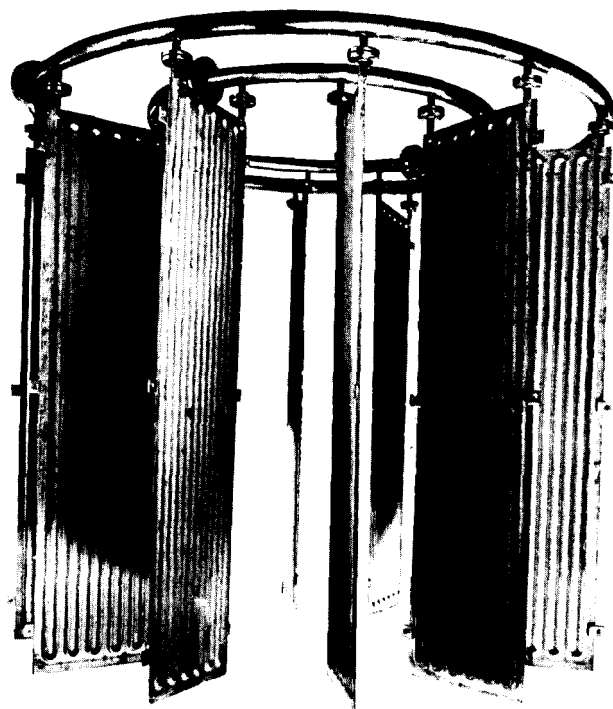


Figure 5 Panel immersion heaters. Courtesy of Dean Products Ltd.

vessel and the exchanger difficult and are best employed in low-fouling applications, or specially designed to simplify the cleaning operation.

Limitations of operating pressures and temperatures are generally specified by the manufacturers. However, the normal design principles used for pipes can be applied to the tubular immersion coils, and flat panels are also usually covered by national standards.

D. Jacketed

Jacket-type heat exchangers can be in two forms, either attached to the outside of an existing vessel or tank to provide heating or cooling, or integral with the wall of the vessel. Some typical applications are shown in Fig. 6. The externally fitted type can be in the form of embossed panels or embossed strips that can be clamped onto the vessel. A principal advantage is that this type need not be made in expensive corrosion-resistant materials. Furthermore, they can be designed to suit specific requirements and also located in particular spots or areas where supplementary heating is required to prevent settling or separation, or for cooling around furnace or oven doors to protect operators.

The jacket exchanger that is integral with the vessel wall is not quite so flexible in application as the clamp-on type, as it must be sized more accurately for the required duty. There are also limitations in the choice of materials of construction, because one

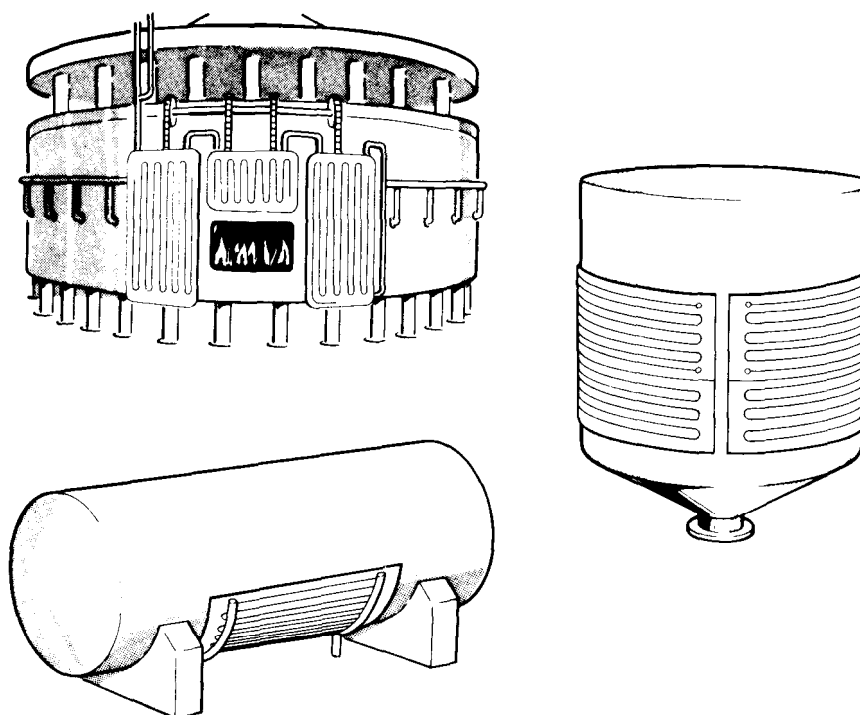


Figure 6 Clamp-on jacket heat exchangers. Courtesy of Elliott Turbomachinery Ltd.

panel of the jacket will be in contact with both process and service fluids and could be subject to corrosion-resistance requirements. The principal advantage of this type, however, is that the thermal resistance between the two fluids is reduced by eliminating the contact resistance between the jacket and the vessel wall. A variety of shapes and applications of this type of heat exchanger is possible, as shown in Fig. 7. Design procedures are detailed in national standards.

E. Block type

Specially treated graphite blocks have been used very successfully in the inorganic chemical field where the principal uses are for hydrochloric, sulfuric, hydrofluoric, phosphoric, and sulfonic acids. Other uses include preheating fresh liquor for electrolytic cells and cooling sodium hypochlorite [5]. These units are also found in the organic chemical, pharmaceutical, and biochemical industries.

A typical graphite block heat exchanger is shown in Fig. 8. Basically the exchanger comprises a number of graphite blocks with flow channels drilled through them to form the service and process fluid paths. The graphite blocks are made of polycrystalline graphite, which is formed by heating blocks of amorphous carbon to temperatures in the range 2 500-2 700° C. A mixture of coke and plastic hydrocarbon is shaped to the required form and then heated to 1 100° C to remove any

noncarbonaceous products and form the amorphous carbon. The finished graphite block, which is neither ductile nor malleable and cannot be cast or welded, has high compressive strength but low tensile and flexural strengths. The principal advantages of graphite as a material for heat exchangers are high thermal conductivity, good chemical and thermal stability, and the fact that it can be easily machined to fine limits.

Figure 8 shows that the process and service fluids flow in cross flow and the peripheral service fluid chambers ensure mixing and help to eliminate any temperature gradient across the flow stream. The modular construction of the exchanger readily permits extension. The blocks are assembled with polytetrafluoroethylene (PTFE) gaskets because the use of cement has inherent disadvantages as its thermal expansion characteristics are different from those of graphite and the effectiveness of cemented joints is limited to the period during which they retain their elasticity.

The design methods used with this construction are very much company oriented and dependent on the composition of the block and distribution of the flow channels through the block.

F. Spiral plate

The spiral plate heat exchanger (Fig. 9) is essentially an extension of the basic plate heat exchanger [6]. It comprises two flat plates welded together along

their long edges and then bent into a spiral coil, thus forming two separate channels through which the service and process fluids flow in true counter flow. Connections are made at the center of the coil to each channel to act as inlet in one case and outlet in the other. Similar connections are made at the outer end of each channel. The spiral exchanger can be enclosed in a pressure vessel, or the outer panel can be incorporated to form the outside of the unit. The exchanger is closed top and bottom with covers bolted to the outer shell of the exchanger. A modification of the basic design can be made so that the process and service fluids can be in mutually perpendicular flow. This flow arrangement is aimed specifically at condensing duties.

The basic plate heat exchanger uses elastomer-type gaskets, which restrict operating temperatures and pressures. In the spiral plate exchanger, compressed asbestos fiber gaskets can be used, thus permitting much higher operating temperatures, typically up to 400°C. Because of the construction, pressure limitations are also higher, and most exchangers can be operated safely at pressures up to 10 atm. Higher pressures yet can be achieved, but these conditions are generally limited to smaller units. The thermal performance of the exchanger is enhanced by the curved flow channels, which assist the promotion of turbulence, but the smooth heat transfer surface results in lower heat transfer coefficients than would be achieved with patterned plates. By virtue of the removable top and bottom covers, this exchanger is

easily cleaned and is therefore ideal for applications involving a high degree of fouling. Indeed, it is widely used for the heating and cooling of slurries. A wide variety of materials can be used for construction, ranging from mild steel to titanium, or in fact any weldable material that can be cold formed. Here again the manufacturer normally specifies the operating limits of temperature and pressure, but the rules for cylindrical vessels subject to external pressure would appear applicable.

G. Scraped surface and screw type

During the processing of waste wash liquors and in processing certain foodstuffs, the process fluid is subject to a concentrating effect, with the result that a deposit can form on the heat transfer surface. This situation also arises in the pharmaceutical industry. The scraped surface heat exchanger has been developed for these applications. Basically it consists of a jacketed vessel through which the service fluid is circulated, and rotating paddles or scrapers that continually remove product from the exchanger's inner surface. The basic construction details and flow paths are shown in Fig. 10. The heat transfer process is aided by the continuous agitation of the scrapers, which also reduce the film thickness to a minimum.

Because of the inherent rigidity of the inner cylin-

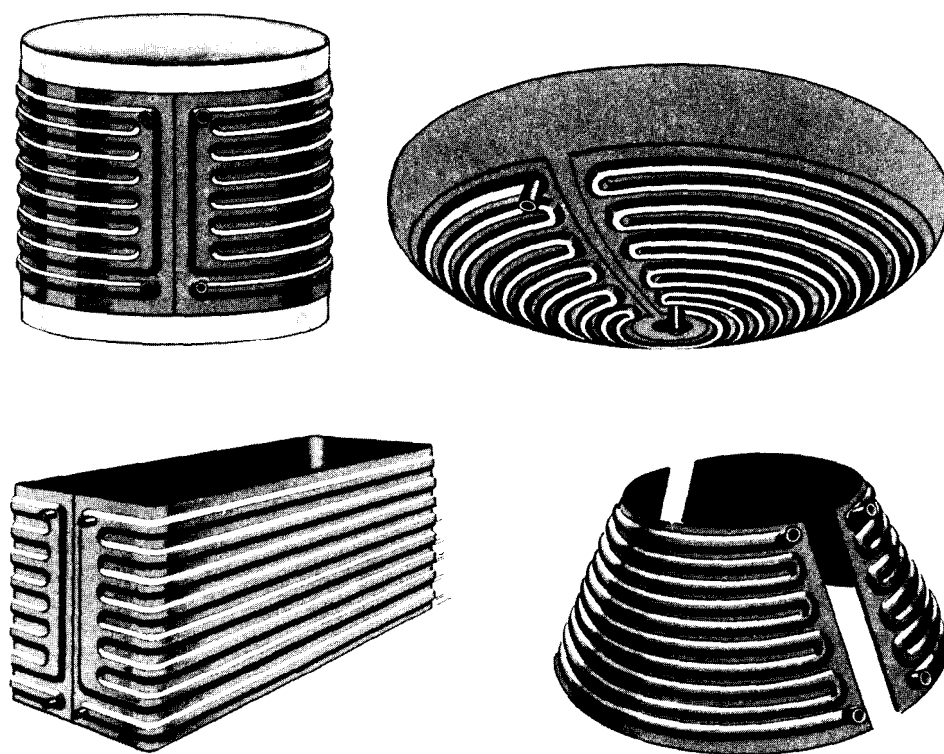


Figure 7 Integral jacket heat exchangers. Courtesy of Dean Products Inc.

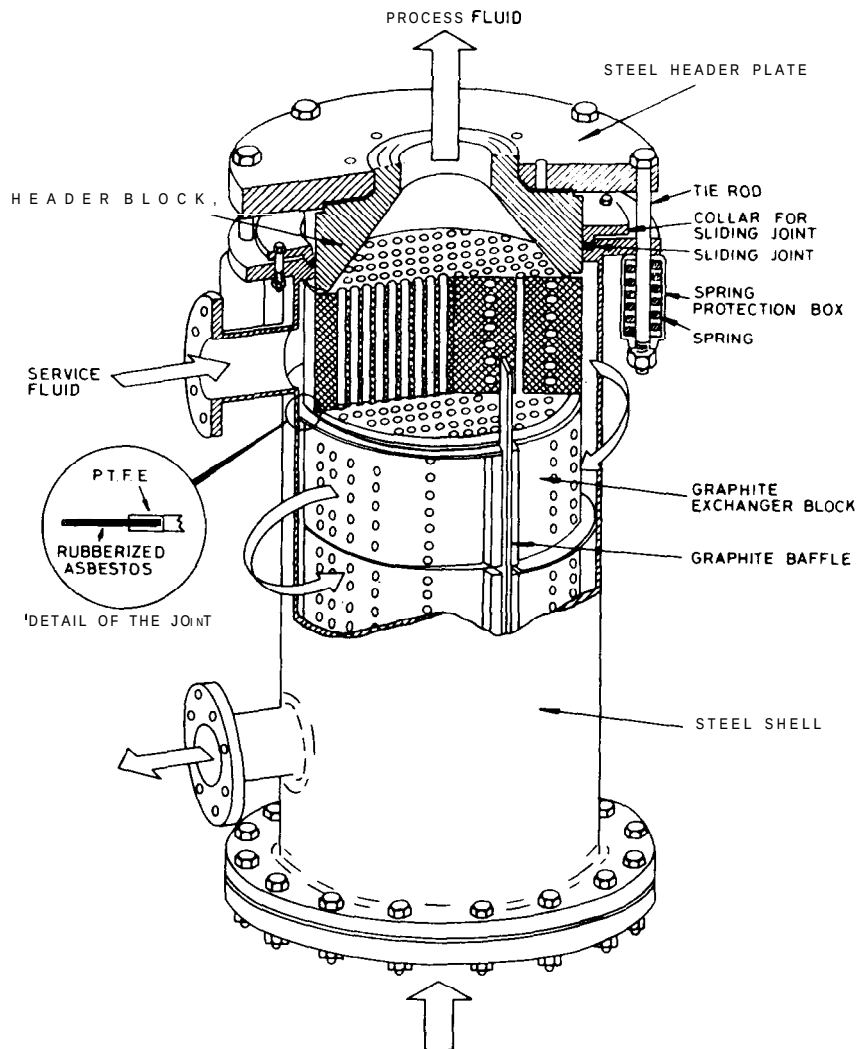


Figure 8 Graphite block heat exchanger. Courtesy of Robert Jenkins and Co. Ltd.

der, the wall can be quite thin, thus reducing thermal resistance. The rotor can be driven by an electric motor or by hydraulic drive, thereby allowing the speed of rotation to be varied according to product compatibility; however, the inner surface must have a high degree of polish because of the action of the scrapers.

Another type of exchanger, which can be likened to the scraped surface exchanger, is the screw type. This consists of a cylindrical shell in which two **contrarotating** hollow screws are mounted. The service fluid is pumped through the hollow screws and the process fluid, which is in counter flow, enters at one end of the cylinder and is passed to the other end by the action of the intermeshing screws. The screws are driven by a variable-speed electric motor. This design of exchanger is particularly suitable for pasty, viscous, or sticky products. Materials of construction can be mild steel or stainless and acid-resistant steel. The design rules for jacketed exchangers apply in this case.

H. Direct contact

Direct contact heat exchangers reduce the resistance to heat flow by eliminating the dividing wall that keeps the process and service fluids separate [7]. Application of this principle is suitable only when the two fluids are compatible.

Examples of successful application of the direct contact principle are in condensers, tank heating, evaporating, and drying, although the most common application is in cooling towers. A further example is in gas-to-gas heat transfer when an intermediate medium is used to carry the heat from one stream to the other.

(a) Condensation

Direct contact condensers [8], shown in Fig. 11, are referred to as spray or jet condensers and are widely

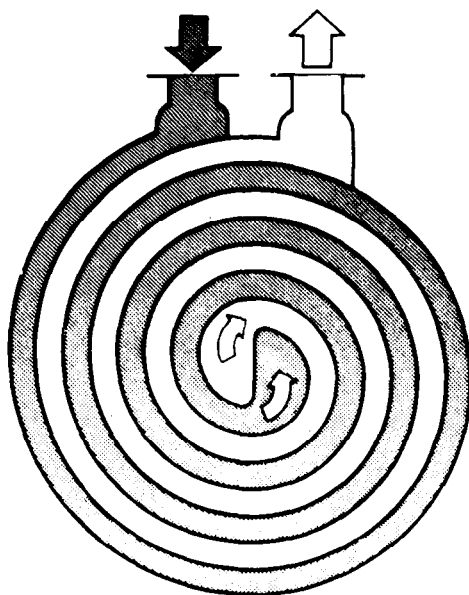


Figure 9 Spiral plate flow principle. Courtesy of Alfa-Laval Co. Ltd.

used with vacuum equipment. The unit comprises a condensing chamber which is connected to a vacuum pump. Coolant is injected into the chamber through spray nozzles, thereby providing a large amount of surface area on which the vapor condenses. The condensing chamber is sealed at the bottom by a well of condensate and coolant, and the vapor enters the top of the chamber. The unit is constructed to withstand vacuum operation, but construction detail depends greatly on the specific installation and the fluids used.

When suitable, a very effective method of tank heating is to inject steam into the tank at the base. This has the added advantage of creating turbulence and promoting good mixing of the tank fluid. This can also be applied to pipelines, in which case specialized injectors are used. Generally with this type of heating, there is no recovery of condensate, the cost of which has two components: heat content and treated water content. The construction of these units is relatively simple, with the principal considerations being given to the design of the nozzles so that maximum dispersion is obtained throughout the bulk fluid.

(b) Evaporation

Evaporators and driers [9] frequently use direct contact heat transfer. The technique of submerged combustion [10], shown in Fig. 12, is used primarily for the concentration and crystallization of corrosive and scaling salt solutions. Fuel and air are supplied under pressure to the combustion chamber and the combustion gases bubble through the process fluid before being dis-

charged. Here again the detailed construction depends largely on the application. During operation, intense turbulence occurs at the end of the dip tube where the combustion gases enter the process fluid and form a large number of small bubbles. Heat and mass transfer rates are high because of the continuous, rapid surface renewal and the high degree of turbulence existing in the annular space between the dip tube and the shroud.

Hot air is widely used as a heat transfer fluid, particularly in drying solids. The wet solid is distributed in the hot air stream for a controlled time until dry and then conveyed to the next stage of the process. The product is protected from overheating, because under atmospheric pressure the temperature cannot rise above 100°C as long as there is water present on the surface.

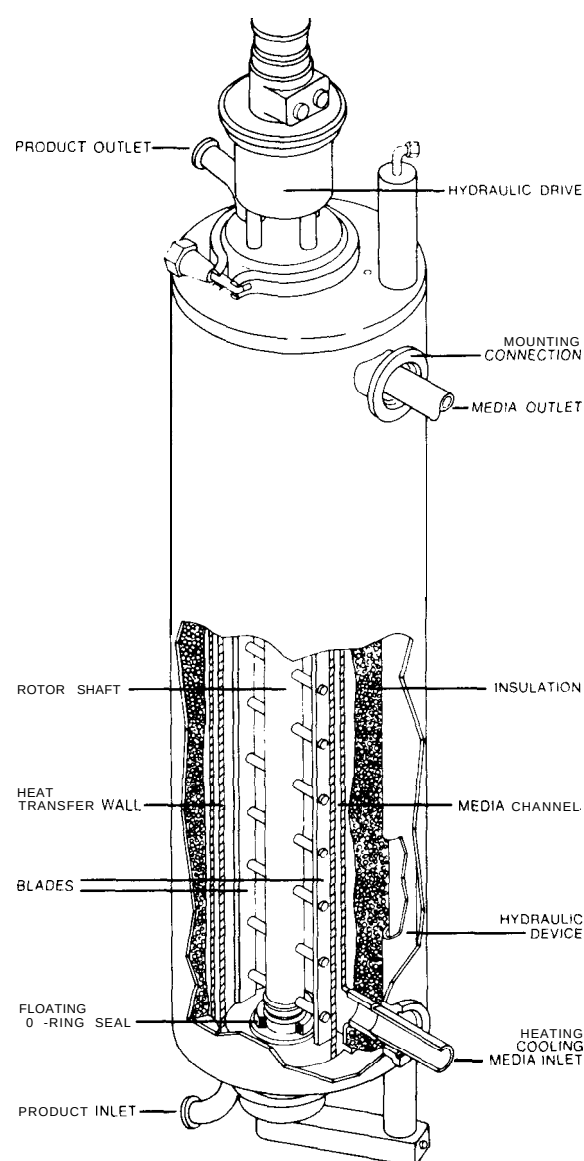


Figure 10 Scraped surface heat exchanger. Courtesy of Alfa-Laval Co. Ltd.

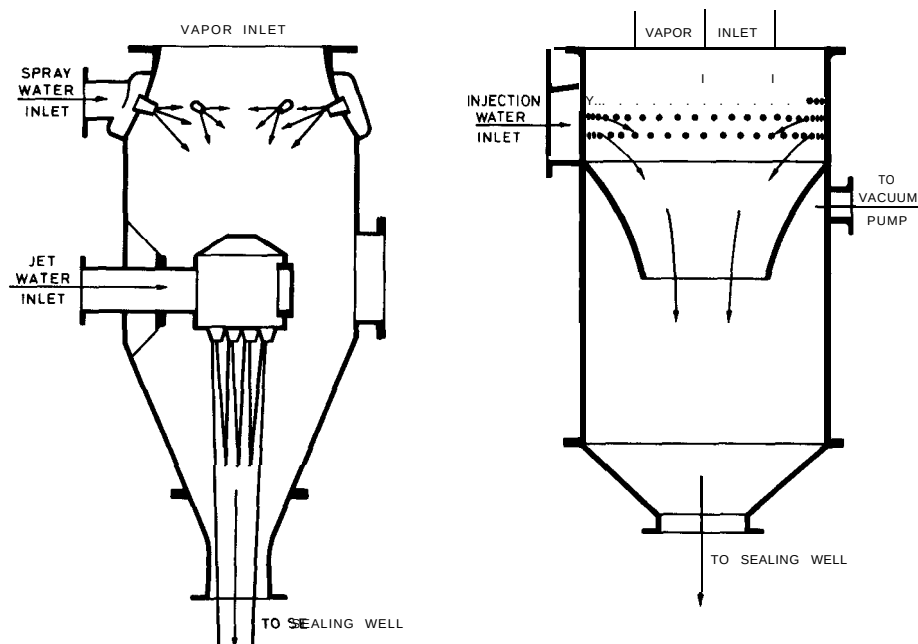


Figure 11 Direct contact condenser.

(c) Cooling

The construction of cooling towers is very much company oriented, particularly with regard to the water distribution system and the packing, both of which have considerable effect on the performance. Materials of construction vary from mild steel with protective coating to plastic panels and molded plastic packing. The packing is designed to provide a frequently changing surface for cooling. This equipment is normally de-

signed as a complete package unit, including fans to produce the airflow and circulating pumps for the water side.

I. Hampson coils

Large-scale refrigeration exchangers, particularly for the Liquefaction of gases, make use of a design first introduced by W. Hampson [11] about the beginning

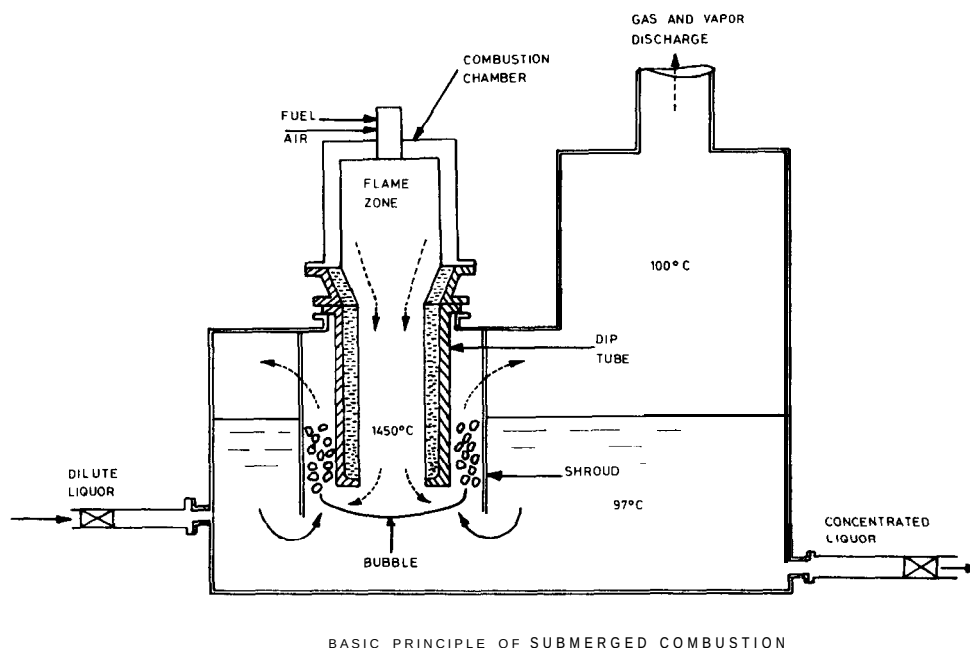


Figure 12 Basic principle of submerged combustion.

of this century. Hampson's design basically comprises a number of tubes wound around a central core so that the tubes follow a helical path from their starting point in one tubesheet inward, or outward, and back again to the other tubesheet (Fig. 13).

In the modern exchanger [12], this principle has been developed so that the tube windings form layers, which are wound in opposite directions and separated by spacers. In general, tubes of the same length are used for each successive layer, which makes the windings less compact as the layers are built up on the central core. This construction is ideal for single pass on the tube side in order to produce uniform flow distribution. For multipass units, it may be desirable to increase the tube length in successive layers for thermal performance reasons. Alternatively, as the diameter of the exchanger coil increases, the number of tubes in successive layers can be increased to maintain a uniform grid pattern throughout the tube bundle.

During manufacture, the central core is rotated and the layers of tubes wound into position and secured in the tubesheets. Interlayer spacers are then placed on top of the windings and the next layer is wound into position in the opposite direction. This process is repeated until the required heat transfer surface area has been obtained. The coil is then placed in a shell and suitable connections provided for the process and

service fluids. Any stressing of the tubes due to differential expansion caused by temperature differences between the two fluids is absorbed by the natural spring of the coiled tubes. There are few practical limitations to the materials of construction, so that most materials compatible with the process fluids and conditions can be used. The design requirements for shell-and-tube exchangers also apply to those exchangers and are described in Sec. 4.4.3.

J. Heat pipe

The heat pipe concept was first described by Gaugler [13] in a patent application in 1942. However, the first use of the name heat pipe appears in Grover's patent application [14] in 1963. Basically the heat pipe [15,16] comprises the container, the wick, and the working fluid (Fig. 14). In operation, heat is absorbed at one end of the container, the evaporator. Some of the working fluid is vaporized and the vapor passes along the central core of the wick to the other end of the container, the condenser, where it condenses releasing its latent heat. The liquid then returns to the evaporating section through the wick by capillary action. The vapor-phase driving force results from the difference in vapor pressure between the evaporator and



Figure 13 Hampson-type coil for refrigeration plant. Courtesy of Air Products Ltd.

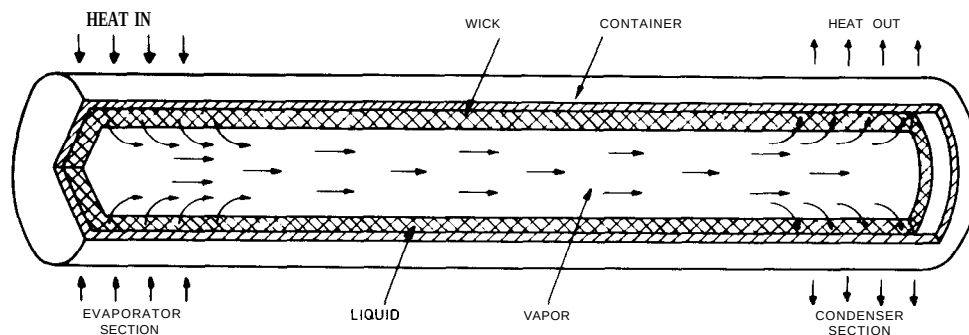


Figure 14 Basic heat pipe.

condenser. On the liquid side, the capillary driving force is caused by the difference in radius of curvature of the menisci in the evaporator and condenser sections.

There are three basic shapes of heat pipe (Fig. 15). The first, and probably the most common, is the closed cylinder. This is made from pipe, or longitudinally

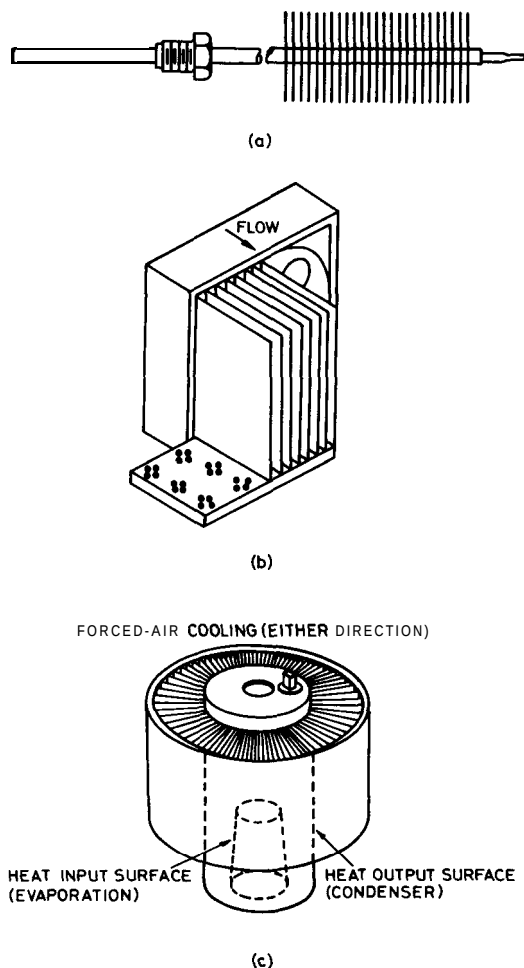


Figure 15 Typical heat pipe configurations. (a) Closed cylinder type. Courtesy of Dynatherm Corp. (b) Flat plate type. Courtesy of Miran Products Ltd. (c) Annular type. Courtesy of RCA Ltd.

drilled rod, into which the cylindrical wick is inserted and held in close contact with the pipe wall. The second shape is a flat plate heat pipe, which can be either circular or rectangular. This is made by welding two flat plates a fixed distance apart, inserting a wick of suitable shape in the space between, and sealing the construction with side walls. The third construction is in the form of an annulus in which two concentric cylinders are joined. A wick is placed in the gap between, and the assembly is sealed.

A wide range of wick geometries can be used in the heat pipe. The principal design constraint is that the wick structure should be capable of supporting the capillary driving force to return the liquid from the condenser to the evaporator. A good contact between the wick and the container inner wall is also essential. This can be achieved by making the wick with a natural radial spring or by holding it against the wall using an inner spring or mandrel. A variety of wick designs have been developed, which fall broadly into homogeneous structures and arterial wicks. The homogeneous wicks are made from meshes, foams, felts, fibers, and sinters. These are normally metallic, but they have also been made from glass fiber and ceramic materials. Arterial wicks are wicks containing channels that provide low-resistance paths for the liquid returning to the evaporator, and are particularly useful if boiling is likely to occur within the wick during the return of the liquid to the evaporator.

The choice of fluid is governed by the heat pipe operating temperature and the fluid's compatibility with the container and wick. Refer to Part 3 of this handbook for more detailed information.

The principal factors to be considered during design and manufacture of a heat pipe are the compatibility of each component, corrosion, gas generation, and operating conditions. Some fairly sophisticated designs have been constructed, notably the constant-temperature heat pipe with integral inert gas reservoir and also the thermal switch. In the first case the reservoir is located close to the condenser and as the tempera-

ture rises, the gas expands, blanketing the condenser and switching the device off. The thermal switch operates by adjusting the vapor flow rate according to an

external sensor. When the device is to be used in gas flows, extended surfaces can be fitted to enhance heat transfer rates.

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4.5

Materials of construction and corrosion

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4.5.1

Introduction

J. F. Lancaster

■ The selection of materials of construction for heat exchangers is in many instances influenced by the design of the equipment. Less often the properties of the required material dictate the type of design that can be used. The need for economy in material on the one hand, and for efficient heat transfer on the other, requires that when metals are used the heat exchange takes place across relatively thin sections, and this in turn means that the selected material must have sufficient corrosion resistance to operate for a reasonable time without perforation.

The fact that in almost every case heat interchange implies fluid flow means that the metal may also be exposed to erosion-corrosion or impingement, which increases the severity of any corrosive effect that may be present. On the other hand, there are cases where the corrosivity is too severe or the temperature too high for metals to be suitable. For highly corrosive

conditions it may be necessary to use a brittle material such as glass or carbon, and the need to minimize exposure to stress is an important factor in determining the design. For high-temperature applications, refractory material may be required and the brittleness of refractories is a factor in the design of, for example, recuperators.

In the space available it will not be practicable to deal with materials for all the various types of interchangers. Moreover, there is some mention of materials in the chapters on special exchangers, and the elastic properties of heat exchanger materials are discussed in detail in Secs. 5.4.5 and 5.4.8. Therefore, the present survey will be concerned mainly with shell-and-tube exchangers and air coolers. The available materials will be considered first, after which the various modes of deterioration that heat exchangers may suffer in service will be reviewed.

4.5.2

Materials of construction

J. F. Lancaster

■ The design codes and standards discussed in Sec. 4.3 list materials that may be used in heat exchangers. Other materials may be used subject to agreement between purchaser, inspecting authority, and the manufacturer; in general, design codes and standards specify minimum qualities of materials.

Materials and the corresponding design stresses to be used in conjunction with the TEMA heat exchanger standards are given in the ASME Boiler and Pressure Vessel Code, Section 8, Division 1. (Codes are mandatory; standards are recommended practice.)

In the case of the United Kingdom, the design stress values of recommended materials for heat exchangers are given in BS 5500.

In the F.R. Germany, the TUV (Technische Überwachungs Verein) is the general approving body for heat exchangers (and many other products). Design methods for pressure vessels (including heat exchangers) are given in the A. D. Merkblätter, while the yield stress values for insertion in the A. D. Merkblatt formulas are obtained from the relevant DIN (Deutsche Normenanschluss) standard.

Before discussing the different metallic and non-metallic materials for heat exchangers, it will be useful to consider the product forms used for various shell-and-tube exchanger parts.

A. Product forms

(a) Tubes

The tubes themselves may be solid drawn, electric resistance welded (ERW), or fusion welded and re-

drawn. It is common practice to prefer solid-drawn tube to ERW tube for high-pressure duties. However, it is not uncommon for solid-drawn tube to contain longitudinal cracklike discontinuities, which can propagate when the tube is pressurized, and it would be difficult to say whether the risk of failure is greater with seamless or ERW. Fusion-welded and redrawn tube is an economic choice for austenitic chromium-nickel tube. In the as-welded condition the weld may have a lower corrosion resistance than the parent material, but this effect is removed by redrawing followed by solution treatment at 1 000-1 050° C.

(b) Tubesheets

Tubesheets are normally forged, and preferably upset forged, that is, forged from a relatively tall cylinder rather than from a flat slab. The use of plate for tubesheets is not normally acceptable because of the risk of lamination or of laminar weakness where the tubesheet is welded to the shell. Tubesheets may be clad with austenitic chromium-nickel steel either by fusion welding or by an explosive technique [1]. Where explosive cladding is used, it may be advisable to take precautions against brittle failure by, for example, using impact-tested plate, ensuring that machined grooves are well radiused, or preheating before welding.

(c) Shells, channels, covers, and bonnets

Shells, channels, channel covers, and bonnets are normally fabricated from plate, although in the case of high-pressure exchangers they may be forgings. Cast-

ings are used to a limited extent: Cast iron has been used for the channels of small condensers, and cast bronze or brass for floating-head covers. The plates of plate heat exchangers are made of sheet metal, which may be formed in various ways for mutual support and to improve heat transfer. Formability is therefore an important property for the material used in certain types of plate heat exchanger. Weldability is also, in most types of exchanger, a necessary requirement. This is an important property for tube-to-tubesheet welds. In spite of advances in welding technology, this type of joint is still prone to failure in service.

B. Materials for noncorrosive service

Typical materials of construction for various temperature ranges in noncorrosive service are listed in Table 1. Austenitic chromium-nickel steel is the most generally useful metal for shell-and-tube exchangers at the lowest operating temperatures. However, a great many cold exchangers (for example, in ethylene or liquid oxygen plants) are of the core type, where alternate layers of flat and corrugated aluminum sheet are brazed together. Such exchangers give the maximum surface for any given volume, and are advantageous in saving space where the process equipment is all enclosed in a cold box [2].

For less severe low-temperature conditions a conventional design may be more economic, and $3\frac{1}{2}\text{Ni}$ or impact-tested carbon steel may be employed in shell-and-tube exchangers. The temperature breakpoint between impact tested and nonimpact-tested steel is indicated in the table to be 0°C , but in some of the codes applicable to heat exchangers this temperature varies with the thickness of the material. In practice, it is usually better to establish a single impact-testing temperature for all parts of the exchanger that are exposed to the cold fluid.

It may also be desirable to apply impact testing to plate or forgings of thick section when the operating temperature is above 0°C . Apblett and Brundige recorded the brittle failure of a high-pressure feedwater

heater, which was ascribed in part to the poor notch ductility of a forging [3]. Karinen [4] described a similar failure of a synthesis gas exchanger in an ammonia plant. The failure was initiated by a preexisting crack, but once again an important factor was the poor notch ductility of the forged channel, which had a transition temperature of about $+100^\circ\text{C}$. Heavy forgings may be prone to brittle failure because of the relatively small amount of mechanical working of the original cast ingot, resulting in a coarse grain size, and because it may be difficult to achieve adequate cooling rates during heat treatment. Karinen recommended the following for heavy steel sections:

1. Carbon content less than 0.15% for weldability.
2. Vanadium content should be 0.10%.
3. Heat treatment should ensure adequate toughness in thick sections. The **postweld** heat treatment temperature for carbon-manganese steel should be over 600°C and for vanadium-bearing steel 650°C or higher.
4. The strength of the material should not be more than 550 MN/m².

At the higher end of the temperature range it may be necessary to use carbon or low-alloy steel protected by a refractory lining. Typical of such applications are the inlets to waste heat boilers immediately downstream of ethylene pyrolysis furnaces and steam-methane reformer furnaces. The object of the lining is to act as a thermal barrier, so enabling the carbon or low-alloy steel to operate at an acceptable temperature. It is therefore necessary to prevent the hot gas penetrating through cracks, and this is done by providing a liner or shroud of austenitic 18Cr-10Ni or 25Cr-20Ni steel. The refractory itself may be attached to the shell or bonnet by means of studs or by hexagonal steel mesh [5, 6].

C. Materials for corrosive service

Materials of construction for corrosive service are shown in Table 2. This list is not intended to be exhaustive, but rather to indicate the application of some common materials in heat exchangers.

(a) Carbon steel

Carbon steel is used for a number of mildly corrosive duties. For example, it is common practice in the hydrocarbon processing industry to employ carbon steel tubes for contact with cooling water in coolers and condensers. There are other more corrosive media in which carbon steel will perform surprisingly well. One example is the overhead condenser for a crude atmospheric distillation tower. The condensate is a mixture of water and hydrocarbon, whereas the gaseous phase may con-

Table 1 Materials of construction for noncorrosive service

Temperature range, $^\circ\text{C}$	Type of heat exchanger	Material
Below -100	Any	Austenitic chromium-nickel steel, aluminum
-100 to -45	Any	$3\frac{1}{2}\text{Ni}$ steel
-45 to 0	Any	Impact-tested carbon steel
0 to 500	Any	Carbon steel
Above 500	Shell and tube	Refractory-lined steel

Table 2 Materials of construction for corrosive service

Material	Typical service use
Carbon steel	Mildly corrosive fluids; tempered cooling water
Ferritic carbon-molybdenum and chromium-molybdenum alloys	Elevated-temperature hydrogen service; sulfur-bearing oils above 300° C
Ferritic chromium steel	Tubes for moderately corrosive service; cladding for shells or channels in contact with corrosive sulfur-bearing oil
Austenitic chromium-nickel steel	General corrosion-resistant duties
Aluminum	Infrequently used for mildly corrosive service
Copper alloys: admiralty, aluminum brass, cupro-nickel	Fresh-water cooling in surface condensers; brackish and sea-water cooling generally
High nickel-chromium-molybdenum alloys	Resistance to mineral acids and Cl-containing acids
Titanium	Seawater coolers and condensers (including plate heat exchangers)
Glass	Air preheaters for large furnaces
Carbon	Severely corrosive duties
Linings:	
Lead and rubber	Channels for seawater coolers
Austenitic chromium-nickel steel	General corrosion resistance
Coatings:	
Aluminum, epoxy resin	Exposure to sea and brackish water

tain H_2S , ammonia, or chlorides. For shell-and-tube condensers the standard materials are brass (admiralty or aluminum brass) for the tube bundle and carbon steel for the shell.

Where air coolers are used for crude overhead condensers, carbon steel tubes and headers will normally give good service, in spite of the fact that the droplets that condense first are aqueous and contain dissolved H_2S and ammonia. Likewise, carbon steel tubes have been used in overhead condensers of stripper towers in CO_2 removal units, where the condensate is carbonic acid. Normal practice is now to employ austenitic chromium-nickel tubes for this duty, but carbon steel may give a reasonable life. In such applications it is necessary to weigh the higher capital cost of alloy material against the cost of unscheduled shutdowns, and with the increasing scale of process plant there is an increased incentive to use alloy instead of carbon steel.

Erosion-corrosion effects may play an important part in determining the life of carbon steel exchangers, and for this reason condensers, where the tubes may suffer droplet impingement, and reboilers, where velocities may be locally increased by two-phase flow, are particularly at risk.

For elevated temperature service, the use of carbon steel may be limited by, for example, hydrogen attack [7] or by corrosion due to mixtures of hydrogen and H_2S [8]. The permissible temperature limits are obtained from charts [7,8], and it is normally acceptable to use the operating fluid temperature on the hot side as the governing figure to apply to the relevant chart. The use of a design rather than an operating temperature is conservative, but may generate an uneconomic and sometimes a less suitable material selection.

In general, it is better to avoid the use of alloy material where carbon steel is a viable alternative: The choice of low-alloy steel may increase fabrication problems, whereas austenitic chromium-nickel steel is subject to the risk of stress corrosion cracking due to accidental contamination, either of the equipment or of the process stream, by aqueous chloride solutions.

(b) Low-alloy steels

Carbon-molybdenum and chromium-molybdenum steels are used in shell-and-tube heat exchangers for corrosion resistance, for example, against attack by mercaptan sulfur in hydrocarbons or for strength at elevated temperature. There have been cases where exchangers made from low-alloy steel have cracked during service. The cracking has been mainly at nozzle connections, and particularly with heavy sections and closely spaced joints. Such problems may be avoided by not using low-alloy steel except where necessary, by spacing nozzles as widely as may be practicable, and by using integrally reinforced nozzles rather than compensating plates.

(c) Ferritic chromium stainless steels

Ferritic chromium stainless steels of the 12Cr and 17Cr types are used for resistance to mercaptan sulfur, to elevated temperature hydrogen/ H_2S attack, and to mildly acidic conditions such as those due to solutions of CO_2 in water. 12Cr-clad steel may be required for shells or channels exposed to hot sulfur-bearing hydrocarbons as an alternative to 5Cr- $\frac{1}{2}$ Mo steel. 12Cr or (more frequently) 17Cr tubes may be specified for coolers or waste heat boilers with water on the shell side and gas that produces a corrosive condensate on the tube side. With this configuration there is a risk of stress corrosion cracking if austenitic chromium-nickel steel tubes are used: Water is evaporated in the space between the tube and tubesheet and even high-purity boiler feedwater may produce a deposit sufficiently high in chloride to cause stress corrosion. Figure 1 illustrates this and other forms of tube-end attack.

The ferritic chromium steels are not affected in this way, although if deposits form on the water side

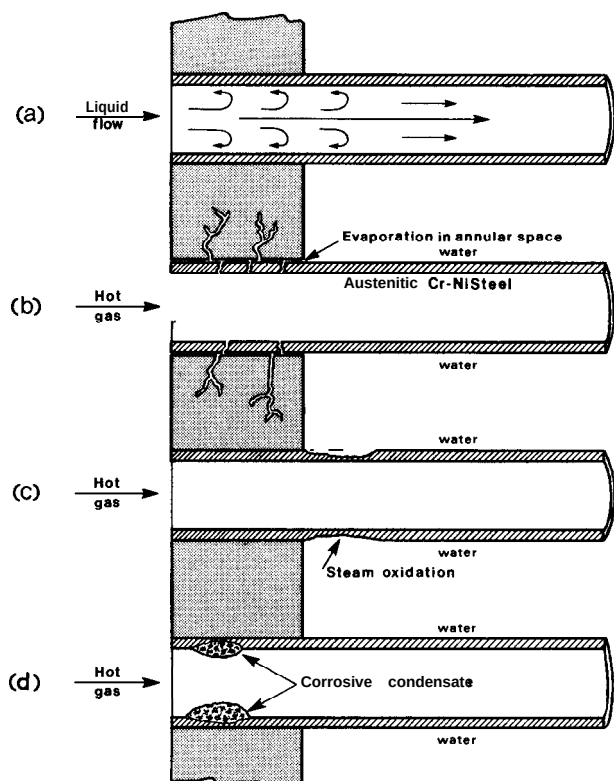


Figure 1 Tube end problems. (a) Inlet turbulence causing pitting or general attack. Remedy: control velocity, inhibitor, ferrules. (b) Excessive inlet gas temperature causes evaporation of water between tube and tubesheet and hence stress corrosion cracking of austenitic chromium-nickel steel. Remedy: reduce inlet gas temperature, use ferritic or ferritic/austenitic material or Monel. (c) Steam oxidation due to excessive heat transfer and film boiling. Remedy: improve configuration of joint (see Sec. 4.5.3, Fig. 2), ferrules, Inconel tube ends. (d) Corrosive condensate forms at inlet (or other part of tube). Remedy: use more corrosion-resistant tube material.

they are likely to be pitted. Where corrosive conditions on the tube side are too severe for 17Cr steel, it may be necessary to utilize one of the proprietary ferritic or ferritic/austenitic stainless steels of the 18Cr-1 $\frac{1}{2}$ Mo or 26Cr-6Ni-1 $\frac{1}{2}$ Mo types. The ferritic and ferritic/austenitic stainless steels are subject to temper embrittlement and must not be operated at temperatures above 400°C. Welds in this type of steel are also brittle, and should be avoided for pressure parts.

(d) Austenitic chromium-nickel steels

Except where there is a known risk of stress corrosion cracking (as in the case mentioned above), austenitic chromium-nickel steels are used for a wide range of corrosive conditions, including organic and inorganic acids, phenol, hydrogen/H₂S mixtures at elevated temperature, and plants where it is required to maintain a high standard of product purity [9,10]. Austenitic

stainless steel is used for coolers with corrosive conditions on the shell side and water on the tube side. In such applications it is necessary to maintain control of the chloride content of the cooling water: A maximum value of 100 ppm chloride has been specified for coolers in the carbamate circuit of urea plants, but 350 ppm chloride would be considered reasonable for most plants, and coolers with stainless steel tubes have been known to operate satisfactorily with up to 1 000 ppm chloride. The grade of stainless steel commonly used for mildly corrosive conditions is AISI type 304, whereas type 316 is specified for more severe environments. Solution treatment of U bends at 1 050° C after forming is sometimes specified, and where this is the case a stabilized (types 321 and 347) or extra-low-carbon grade (304L and 316L) must be used. There is a limited number of applications where nickel-base or cobalt-base tubes are necessary, for example, hydrochloric acid, formic and acetic acids, ferric and cupric chloride, and hot phosphoric or sulfuric acid [11].

Where the shell-side fluid is noncorrosive and provided that there are no problems due to differential thermal expansion or galvanic corrosion, it is not unusual to make all parts other than tubes, including baffles and tie rods, from carbon steel. The tubesheet in such cases will be carbon steel clad with stainless steel on the tube side only.

(e) Copper-base alloys

In U.S. practice, inhibited admiralty brass (approximately 70Cu-30Zn with small additions of arsenic or antimony) is a standard material for condensers cooled by sea or brackish water, with naval rolled brass (60Cu-40Zn) tubesheets. In the United Kingdom and Europe, aluminum brass is often preferred because of a better resistance to velocity effects. Brass has a useful resistance to the CO₂-containing condensate in steam turbine vacuum condensers and (as seen earlier) to H₂S-containing condensate. It may, however, be attacked by solutions of ammonia, and where either the condensate or cooling water may contain ammonia, brass is not normally acceptable.

The other copper-base material that finds substantial use for heat exchanger tubes is cupronickel. Typical compositions are 90Cu-10Ni, 80Cu-20Ni, and 70Cu-30Ni, all of which may also contain additions of iron for increased resistance to erosion-corrosion. The 70:30 grade is resistant to seawater under most circumstances but may be attacked by polluted seawater. It has also been used for overhead condensers and stripper reboilers in CO₂ removal systems. In stripper reboilers, cupronickel will survive provided that the environment is free from oxygen and sulfur. Aluminum bronze (95Cu-5Al) has also been specified for seawater coolers and

condensers, particularly when the cooling water temperature is **higher** than normal. In all applications where copper-base alloys are used, care must be taken to avoid galvanic couples. For example, where a carbon steel channel is in contact with a brass tubesheet in seawater service, the carbon steel should be clad with austenitic chromium-nickel steel weld deposit over the area of contact, or should be electrically insulated in some way, as illustrated in Fig. 2.

(f) Nickel-base alloys

The use of nickel-base alloys for severely corrosive service has been mentioned above. In addition, the Ni-30Cu alloy Monel has been used for a number of duties such as seawater and brackish water coolers and boiler feedwater heaters where the water is on the shell side, and where advantage is taken of its resistance to stress corrosion cracking and other forms of corrosion due to chlorides. Monel metal has a useful degree of resistance to corrosion by fluorides, and has, for example, been used for reboilers and condensers in alkylation units that employ HF as a catalyst [12]. However, in modern plants where it is possible to maintain **water-free** conditions throughout, carbon steel is adequate for all heat exchange equipment [13]. Monel is also used for exposure to hot caustic soda and to hot potassium carbonate solutions.

(g) Titanium

Titanium is used for shell-and-tube heat exchangers that are cooled by seawater, particularly where the process-side fluid is incompatible with brass or other alternative metals. To reduce costs, tubes are 20 or 22 BWG, and with such a thin material it may be necessary to increase the number of tube supports to avoid fatigue failure. The tubesheet may be solid titanium or carbon steel explosion clad with titanium, and it is usual practice to seal weld the tubes to the tubesheet. Titanium is a standard material of construction for plate heat exchangers where the cooling medium is salt, brackish, or corrosive water.

(h) Coatings

Organic coatings may be used to protect any part of shell-and-tube or double-pipe exchangers, mainly against aggressive water, but to a lesser extent against corrosive process streams. Solvent-free, high-build epoxide resin paint is applied to channels and sometimes tubesheets of coolers and condensers operating with sea or brackish water. For the same application, channels may be rubber lined. Tubes may be coated internally or externally by a baked phenolic or epoxyphenolic resin. The resin is applied in a number of coats and baked between each coat; when the required thickness and continuity has

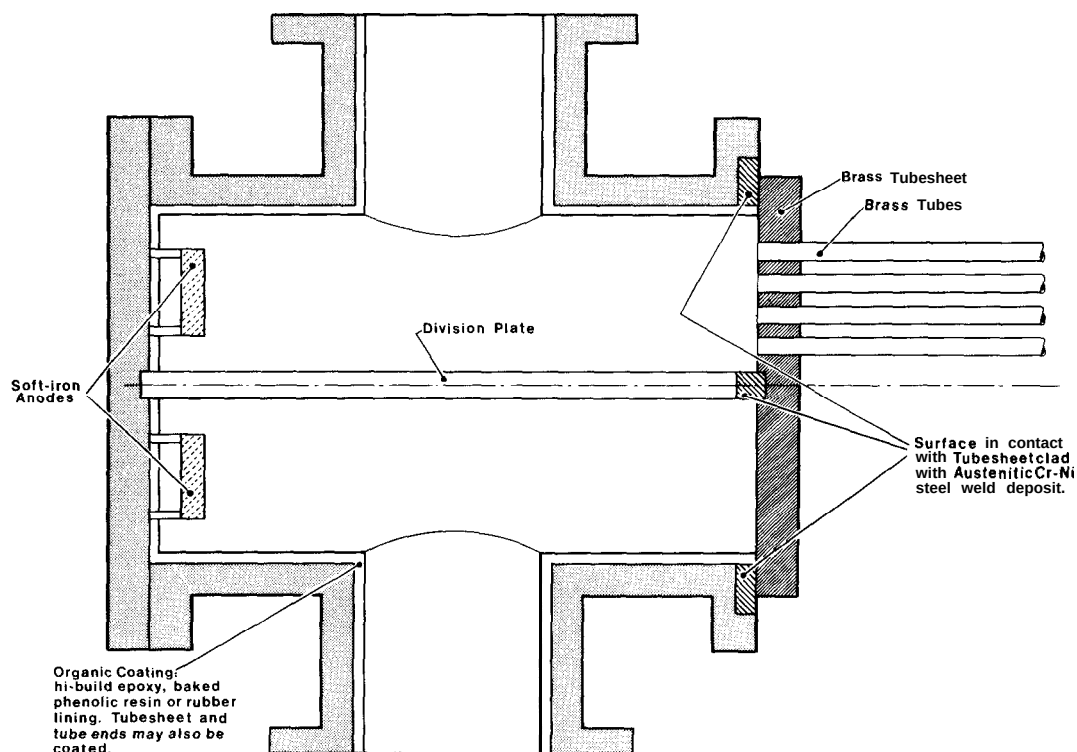


Figure 2 Typical scheme for protection of channel and tube ends of a cooler or condenser with a brass tube bundle. Sea or brackish water service.

been achieved, it is given a final bake at a higher temperature (about 200°C). The most suitable application for such a coating is the inside of a tube bundle. The outside of U-tube bundles may also be coated, but the inherent discontinuities between tubes and baffles are such that the performance would be difficult to guarantee. There are similar uncertainties about coating the shell of a U-tube exchanger, although these may be overcome to some extent by providing stainless steel rails on which the bundle can slide. Coated parts must be designed so as to avoid sharp comers and reentrants. Guidance as to suitable design details is contained in a directive of Verein Deutscher Ingenieure, VDI 2532.

Chemically and thermally cured organic linings have a wide use in industry generally and in heat exchangers are especially useful for seawater condensers, where the lack of fouling more than compensates for any increased thermal resistance due to the film. Air-cured, high-build epoxides may be used up to 70°C, and baked phenolics up to 80°C wet and 120°C dry. Adequate quality control of surface preparation, application, and stoving is

essential, and final acceptance tests must include a test for continuity [14].

(i) Nonmetallic materials

Impervious graphite is employed in special designs of heat exchanger for especially severe conditions such as contact with aqueous hydrofluoric acid or wet chlorine. Glass is used for tubes in air preheaters for large furnaces and power boilers. In this type of unit heat is exchanged between incoming cold air and out-flowing flue gas. Concentrated sulfuric acid condenses on the tube surface due to the presence of SO₃ in the flue gas, and metals are rapidly corroded. The glass tubes are contained in a cast iron duct lined with polytetrafluoroethylene (PTFE) sheet. There is a mechanical seal between the tubes and the cast iron, so arranged that the PTFE membrane is continuous. Shell-and-tube exchangers with a glass shell and either stainless steel or tantalum tubes may also be obtained, tantalum being used for boiling sulfuric acid [15].

References for Section 4.5.2 appear at the end of Section 4.5.3.

4.5.3

Corrosion and other types of damage

J. F. Lancaster

■ Heat exchangers may be subject to various modes of deterioration associated with flow such as erosion-corrosion, fretting, and fatigue, and they may suffer stress corrosion cracking (SCC) due to preexisting or imposed stress. These mechanically induced problems will be discussed first, and then some of the fluids that give rise to corrosion in exchangers will be considered.

A. Mechanically induced damage

There are three main types of wastage directly associated with liquid flow, as illustrated in Fig. 1: cavitation, impingement, and erosion-corrosion.

(a) Cavitation

Cavitation occurs when the combination of flow and static head is such as to generate tensile forces within the liquid. Bubbles are formed that later collapse at or near the metal surface, setting up alternating tensile and compressive stresses in the metal. This cyclic stress may result in fatigue damage, which in turn causes jagged pits to appear even when the liquid is noncorrosive [16]. In heat exchangers this purely mechanical form of damage is exceptional, but a low effective head may develop in the upper tubes of air coolers, resulting in bubble formation, damage to the protective film on the metal surface, and pitting corrosion.

(b) Impingement damage

Impingement of liquid drops on a metal surface may cause damage by the same mechanism as that described

for cavitation. Again, the fluid velocity in a heat exchanger is not high enough for purely mechanical damage, but if the drops are corrosive, rapid wastage can occur due to the removal of protective films. The most severe effect is at the first stage of condensation when the aqueous phase is dispersed as small droplets. A typical case is recorded by Hines and Neufeld [17], in which water drops condensing in a CO₂-containing gas impinged on the tubesheet of a carbon steel exchanger and corroded it at the rate of about 10 mm/yr.

(c) Erosion-corrosion and tube end attack

Erosion-corrosion may occur at any point where the linear fluid velocity is high or where there is turbulence. On the shell side there may be a combination of impingement and erosion-corrosion due to a condensing phase, and it is normal practice to install a baffle plate across the inlet nozzle to deflect the flow laterally. Additionally, or alternatively, the outer one or two rows of tubes may be replaced with solid bar of the same diameter.

Erosion-corrosion may be found inside the tubes if the velocity is too high, for example, because some of the tubes have been blocked by debris. This problem most frequently affects coolers and condensers, particularly with once-through salt or brackish water cooling. Design changes in the cooling water circuit or the circulation of polluted water may also cause failures [18]. Because the flow is turbulent at the tube inlet, corrosion-erosion is most likely to occur at this point (tube end attack). The corrosion usually takes the form of pitting, but there may be areas of more general

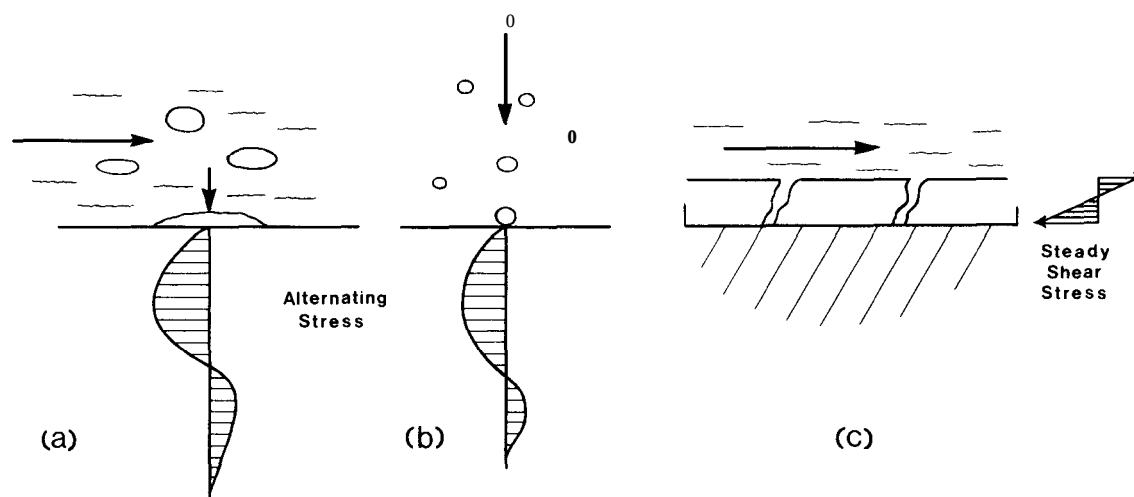


Figure 1 Wastage due to stress. (a) Cavitation: collapse of bubbles at metal surface sets up a stress oscillation transverse to the surface, causing localized fatigue failure. (b) Impingement: a similar effect produced by liquid drops impinging on a surface. (c) Erosion-corrosion: transverse stress due to fluid flow causes rupture or thinning of protective film.

wastage, Tube ends may be vulnerable in other ways [Fig. 4.5.2(1)]. For example, in a waste heat boiler with a high inlet gas temperature, it is possible for film boiling to occur on the outside of the tubes near the tubesheet, resulting in wastage by steam oxidation. Figure 2 illustrates designs to avoid overheating the tube ends. In nitric acid condensers a concentrated acid solution forms at the tube inlet, and this corrodes the 17Cr material that is often specified for this service.

(d) Avoidance of erosion-corrosion

Corrosion-erosion in tubes is avoided in the first place by placing an upper limit on the fluid temperature and velocity; Table 1 shows typical velocity and temperature limits for various materials. A minimum velocity limit and a maximum water temperature are also imposed in order to prevent the formation of deposits, under which pitting corrosion (shielded attack) would take place. Even with a correct design, corrosion may occur due to the type of circumstance discussed earlier, and if the attack is confined to the tube ends, it may be possible to overcome this by inserting ferrules. These are short lengths of tube that are flanged at one end and tapered at the other to provide a smooth transition to the tube proper. If the inlet temperature is low (e.g., in coolers), a plastic ferrule may be used, but there are also corrosion-resistant metallic ferrules and thermally insulated metallic ferrules to minimize heat transfer at the tube inlet. Ferrules are not an entirely satisfactory solution to tube end attack for various reasons: They may become displaced, they may interfere with cleaning, and they may generate some turbulence resulting in corrosion at the end of the ferrules. An alternative solution is to retube with a more corrosion-resistant

material, for example, replacing aluminum brass by 70:30 cupronickel. Tube end attack and corrosion-erosion of brass tubes is also inhibited to a substantial degree by the presence of dissolved iron in the seawater. For this reason, soft iron anodes may be mounted on the channel cover or on the tubesheet itself. Injection of ferrous sulfate is another way of providing the dissolved iron and has proved effective in arresting corrosion where it caused troublesome failures [18,19]. Zinc anodes have been used to protect channels and tube ends in seawater coolers, but they polarize rapidly and there are doubts about their effectiveness [18].

(e) Fretting corrosion

Fretting corrosion has been observed in bayonet boilers. These are waste heat boilers with double tubes; the water passes down the central tube and up the annular space between the inner and outer tubes. Stud-welded rod is used to maintain this annular space, and there may be sufficient relative movement between the rod and outer tube to cause fretting and eventual perforation. Fretting corrosion may also occur at the points of contact in adjacent plates of plate heat exchangers, and between tubes and baffles in shell-and-tube exchangers (Fig. 3).

(f) Fatigue failure

Heat exchangers are subject to cyclic loading in the same way as other pressure vessels by periodic start-up and shutdown of the plant or by periodic loading and unloading of the exchanger itself. They are also subject to pressure fluctuations if there is a reciprocating compressor in the unit. In either case, design techniques to

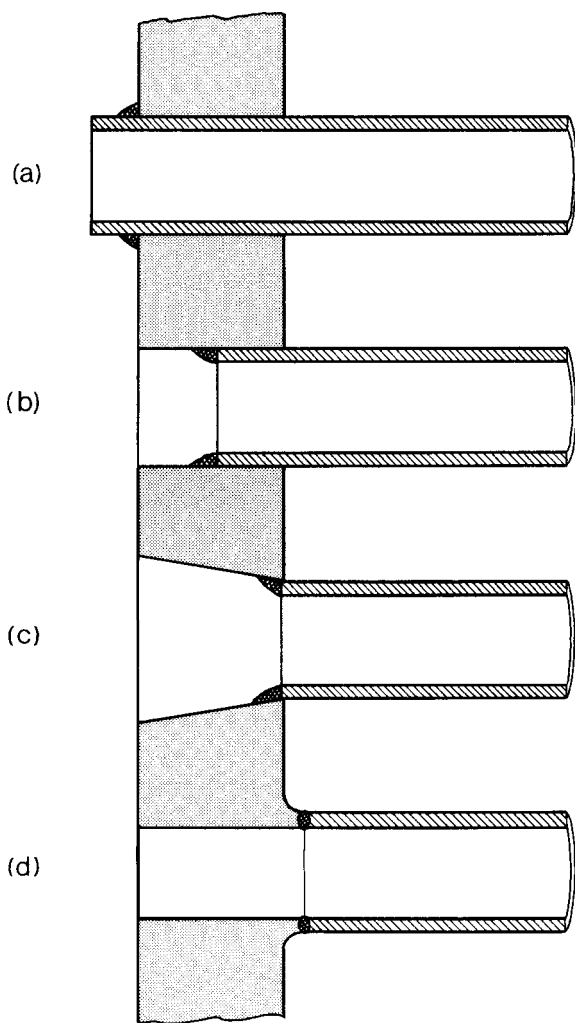


Figure 2 Design of tube/tubesheet joints to avoid overheating tube ends. (a) Joint typically susceptible to overheating. (b) Improvement of design at minimum cost. (c) Changes on tube hole improves gas flow and provides access for welding. (d) Needs special welding equipment and machining of back side of tubesheet. Costly but effective when welded under controlled conditions.

avoid fatigue failure are readily available [20]. Fatigue failure of tubing due to transverse flow of the fluid on the shell side is not uncommon. Tube vibration may be induced if the distance between baffles is relatively large and the fluid velocity too high. The critical velocity that induces vibration is inversely proportional to the square of the unsupported tube length, and there is a formula enabling critical velocities to be calculated [21]. Failures occur at baffle plates or tubesheets and affect the outer rows of tubes preferentially. Vibration may be eliminated by reducing the unsupported length of tube in some way [22], but is best avoided by correct initial design.

Baffle plates across the shell-side inlet nozzle, as well as minimizing the risk of impingement corrosion

Table 1 Temperature and velocity limits for shell-and-tube heat exchangers

Material	Maximum allowable temperature on process side, °C	Permissible velocity range in tubes, m/s
Baked phenolic, resin-coated steel	140	1.0-2.5
Admiralty metal	200	1.0-2.0
Aluminum brass	200	1.0-2.5
Aluminum bronze	250	1.0-3.0
70:30 cupronickel	425	1.0-3.0
Monel	475	1.0-3.5
Carbon steel	525	1.0-2.5
17Cr steel	400	2.0-3.5
18Cr-10Ni steel	800	2.0-3.5

^aMaximum return temperature of water: (1) seawater: 45° C (above this temperature, corrosion and salt deposition may occur); (2) cooling tower water: 50° C (above this temperature, calcium sulfate deposits may form).

(as discussed earlier), also serve to distribute the flow and make the velocity of the fluid as uniform as possible, thereby reducing the risk of local high velocity.

B . Specific failure modes

(a) Tube baffle damage

Localized damage of tubes where they pass through the baffles may also be due to fretting, to corrosion (possibly galvanic corrosion if the tube and baffle materials are different), or to thermal distortion. Such distortion can occur in U-tube exchangers having a longitudinal baffle, where one leg of the U is hotter than the other.

It is possible for a standing wave to be set up due to gas flow through the channel section, and any parts of the associated pipework that resonate to the resulting vibration may fail rapidly by fatigue. The remedy in such a case is to redesign the channel so as to remove features that generate noise.

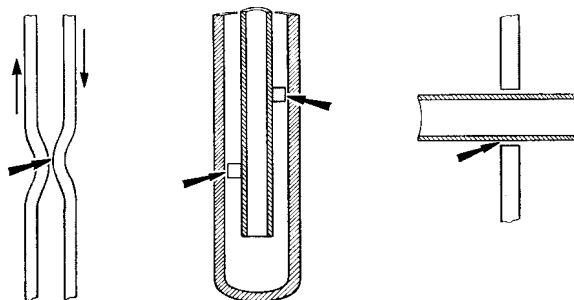


Figure 3 Typical locations that may be subject to fretting corrosion under vibratory conditions.

(b) Vapor blanketing

Vertical shell-and-tube exchangers in which hot gas enters the tubes at the top may be subject to gas blanketing under the top tubesheet if provision is not made for proper venting (Fig. 4). Blanketing may in turn cause the underside of the tubesheet to be subject to alternately wet and dry conditions, resulting in accelerated corrosion [10]. It may also generate a cyclic thermal stress and cause thermal fatigue and cracking of the tubesheet. Apart from venting, this type of problem may be avoided by insulation of the tubesheet, by introducing the hot gas at the bottom instead of the top, or (where film boiling causes damage of the tubes at the bottom tubesheet) by welding the tubes to the back side of the tubesheet. Steam blanketing is also possible in horizontal waste heat boilers, and this may result in differential thermal expansion and bending of the shell.

(c) Waste heat boilers

The problem of stress corrosion cracking of austenitic chromium-nickel steel has been mentioned briefly in Sec. 4.5.2. In waste heat boilers and in certain types of cooler, it is preferable to have water on the shell side, whereas if the hot fluid is corrosive, stainless steel will be required for the tubes and tubesheet. If the inlet

fluid temperature is high enough to evaporate the water in the space between the tube and tubesheet, then austenitic chromium-nickel steel is likely to crack (Fig. 1). The temperature of evaporation is about equal to the saturation temperature of steam at the operating pressure, and therefore austenitic stainless steel may be used provided that the hot gas inlet temperature is below the saturation temperature by a reasonable safety margin, say, 30° C. If this is not the case, a ferritic or ferritic/austenitic bundle may be specified. However, the use of a ferritic or ferritic/austenitic steel may give rise to problems if the tubesheet is welded to the shell because of brittleness in the weld heat-affected zone. There remains the more costly alternative of selecting a higher nickel alloy that is compatible with the process-side conditions. Welding the tubes to the back of the tubesheet is a possible solution to the problem provided that the water is of good quality; if, on the other hand, the water treatment is unreliable, deposits may form on the outside of the tubes, causing pitting and possibly stress corrosion cracking.

Waste heat boilers are subject to many of the corrosion problems associated with large water-tube boilers, particularly with high-pressure (1 500 psi and above) steam systems. Corrosion may result from excess caustic soda that is concentrated during the boiling process to a level where it dissolves the protective magnetite film on the metal surface, resulting in rapid failure of the

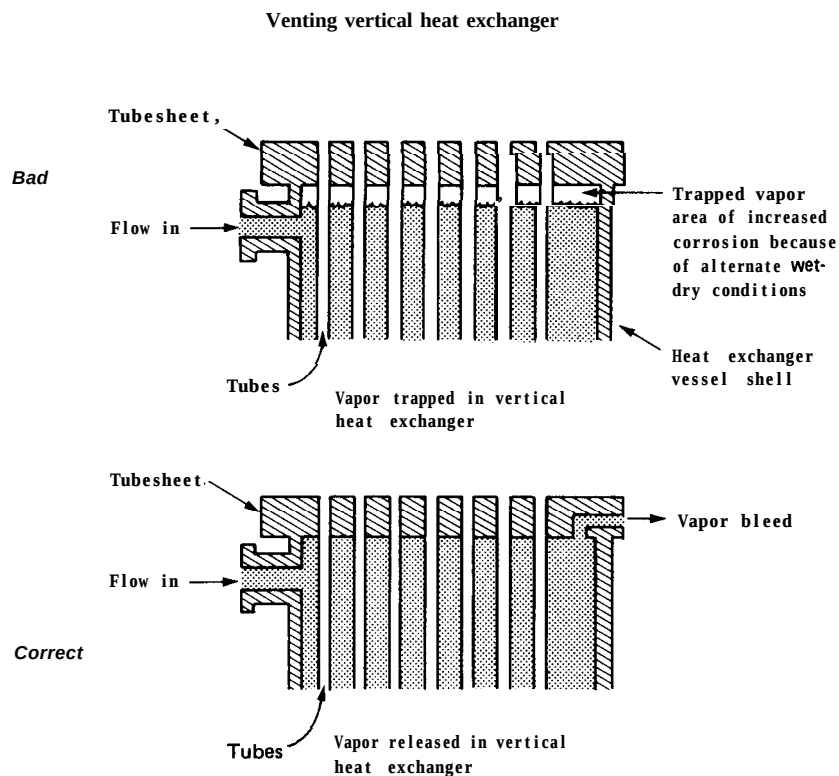


Figure 4 Avoiding vapor blanketing of the top tubesheet of a vertical heat exchanger by venting [9].

tube [23]. Likewise, if there is excess acid in the boiler feed water resulting from cleaning operations, the tubes may fail by hydrogen attack [23]. If the water treatment is ineffective, deposits will form, and if they collect at the inlet-gas end of the boiler, may cause failure by overheating. An unusual case of scaling and overheating failure due to deposits formed by colloidal clay in the makeup water has been described by Ford and Watson [24]. Another somewhat unusual mechanism for deposits forming on the gas side of waste heat boiler tubes has been observed in steam-hydrocarbon reforming plants. At the high operating temperature of the reformer, silica, which may be present in refractories, dissolves in the process steam. When the silica-bearing gas contacts the relatively cool surface of waste heat boiler tubes, the silica is precipitated and deposits build up. Such deposits are avoided by specifying low-silica refractories for contact with high-temperature steam [25].

(d) *Reboilers*

Reboilers are potentially more vulnerable than the average shell-and-tube exchanger for two reasons: First, the formation of a gaseous phase may increase velocities and promote erosion-corrosion; and second, the liquid may decompose upon heating and form a corrosive compound. For example, when mercaptans are heated in contact with a carbon steel surface, they partially decompose, forming H_2S . This may happen in refinery distillation units handling a light sulfur-bearing hydrocarbon. The light mercaptans break down relatively easily, and reboiler tubes corrode more rapidly than, say, piping exposed to fluid at the same temperature. There are other, often quite complex, mechanisms that result in more rapid corrosion of a surface through which there is a heat flux.

(e) *Problems due to phase separation*

Heat exchangers may be exposed to any of the corrosives found in process or power plants. There may, however, be special problems due to phase separations during cooling or heating. One case has already been mentioned: the formation and impingement of water droplets in a CO_2 -containing gas. A similar problem may arise where the gas contains a substantial proportion of H_2S , which is the case in the overhead system of towers in a number of petroleum refining processes, and in such cases it is necessary to consider the use of austenitic stainless steel for the tubes [10]. In certain processes, small amounts of organic acids such as formic, acetic, and butyric acid may form in synthesis reactors, and these may condense preferentially in downstream coolers and again in the overhead system of

distillation towers. Downstream of the point of initial condensation the acids become progressively more dilute and less corrosive. In addition to the bulk components of process streams, it often happens that small amounts of aggressive compounds are formed in reactors, and these may increase corrosion rates. One example is hydrogen cyanide, which forms in fluid catalytic cracking reactors. On the other hand, contamination due to carryover from distillation towers may be beneficial. It was recorded earlier that carbon steel can give good service in the overhead condenser of a CO_2 stripper, in spite of the fact that there is a high concentration of CO_2 in the gaseous phase. This is partly because there is carryover of the potassium carbonate or amine solution from which the CO_2 is stripped, which significantly reduces the acidity of the condensate. Oxygen has the effect of promoting a number of corrosion processes, notably sulfidation by H_2S and corrosion by CO_2 , and accidental contamination by oxygen (for example, by leakage in vacuum systems) may have an important effect on the rate of attack.

(f) *Fouling*

Fouling, that is, the formation of nonmetallic deposits on the heat exchange surfaces, is a problem in many applications. Such deposits may increase the thermal resistance of the surface, cause shielded attack, and even block tubes. Deposits may come from foreign matter such as sand or clay present in cooling water, may be scale generated by a corrosion mechanism or other chemical reaction, or may be high-molecular-weight process stream components.

Cooling water is the most important source of fouling, which may result from either deposition of carbonates from a hard water or corrosion due to the presence of contaminants. The object of water treatment is to minimize these risks, and such treatment, in the case of circulating systems, could include the injection of a corrosion inhibitor plus a water conditioner. The conditioner has the effect of causing carbonate or sulfate to form as a soft nonadherent precipitate rather than as a hard scale. There are cases where such treatment is unsuccessful or uneconomical, and where it is necessary to consider the use of alternative materials, for example, replacing admiralty brass by 70:30 cupronickel [18], or replacing plain carbon steel with baked phenolic-coated material [26].

(g) *Welds*

Where welds in contact with high-temperature boiler feed water are cracked, or contain cracklike defects, the defects are likely to propagate in service. One example of this type of failure was a 1 500-psi waste

heat boiler, which was a **fixed** tubesheet exchanger with tubes welded to the tubesheets. Because of a defective weld procedure, subsurface hydrogen cracks formed in the tube-to-tubesheet welds. After the boilers were placed in service, these cracks extended to the face of the tubesheet in a few days.

(h) Gas/vapor-phase corrosion

Corrosion by the gaseous phase sometimes occurs in heat exchangers. The possibility of steam oxidation on the water side of boilers has already been mentioned. This may be found at the inlet tubesheet if the gas temperature is high and there is steam blanketing or film boiling. Corrosion by hydrogen is possible, due either to the formation of hydrogen at the metal surface by an aqueous corrosion mechanism, or by the presence of high-temperature hydrogen in the process stream (as mentioned in Sec. 4.5.1). In both cases hydrogen diffuses into the steel and combines with carbon to form methane. The methane is precipitated at grain boundaries, causing intergranular fissures and eventually lamination and blistering. Hydrogen corrosion by process gas is avoided by specifying alloy material where this is indicated by the Nelson chart [7]. Mixtures of H_2 and H_2S will also cause rapid corrosion of carbon and low-alloy steel when there is a significant amount of H_2S present and when the temperature exceeds about $300^\circ C$. These conditions are found in catalytic desulfurization plants, and corrosion is avoided by specifying austenitic chromium-nickel for the process side of exchangers where this is indicated by the appropriate charts (e.g., those given in [S]). 12Cr steel is

also used for high-temperature H_2/H_2S attack, but its resistance is only marginally better than that of carbon steel [27].

Metal dusting is a relatively rare form of deterioration that occurs in gas mixtures containing CO and (usually) hydrogen at temperatures between 425 and $1\ 000^\circ C$. The metal is penetrated rapidly in the form of pits, the corrosion product being a powder consisting of a mixture of graphite, metal carbides, and oxides. Corrosion is due to rapid carburization, which starts at isolated points and progresses outward over a hemispherical front, eventually converting the metal completely into a carbide dust. Metal dusting is inhibited by traces of sulfur or by steam [28]. Usually (but not in every case) the resistance to metal dusting decreases as the nickel content of the alloy increases, so that, for example, an 18Cr-10Ni steel may be resistant when a 25Cr-20Ni alloy is not. The alloys most frequently subject to metal dusting are the austenitic chromium-nickel steels, but low-alloy steels may also be affected [27].

It will be evident from the material presented here that the selection of materials for heat exchangers is a complex task that requires skilled and experienced personnel if it is to be done effectively. So far as the more general question of assuring reliability is concerned, this is as much a design problem as it is a metallurgical problem, and close liaison between the disciplines concerned is essential to success. In any organization that designs heat exchangers, it is most desirable to build up a multidisciplinary team capable of anticipating materials problems and that has the knowledge and ability to specify materials and design so as to assure optimum reliability in service.

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4.6

Flow-induced vibration

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4.6.1

Introduction

J. M. Chenoweth

■ In recent years tube vibration has joined heat transfer and pressure drop as primary factors controlling the design of shell-and-tube heat exchangers. Tubes vibrate at their natural frequencies as a result of the shell-side fluid flowing past them. When the flow velocity becomes sufficiently large that the vibrating tubes impact with baffles, with the exchanger shell, or with one another, leaks can occur in the tubes or tube joints. The result is that the exchanger must be taken out of service for repair and modification. Section 4.6 presents an overview of the problems, the methods for predicting their occurrence, comparisons of predictions with field experience, and some procedures for preventing them.

There has been a trend toward larger exchangers with increased shell-side flow velocities to improve heat transfer and to reduce fouling. Scale-ups of exchanger designs have often been made without any consideration of the effect that geometry and flow conditions have in flow-induced vibration.

Although there have been many heat exchangers with reported vibration problems, the understanding of the problems leaves much to be desired. Few investigators have considered the specific problems associated with industrial shell-and-tube heat exchangers. Experimental vibration data have usually been obtained under controlled conditions using single tubes or ideal tube banks exposed to either uniform parallel flow or uniform cross flow. Application of the results of the idealized tests to predict conditions in an actual exchanger may be seriously questioned due to differences in geometry, in the way the flow is coupled to the motion of the tubes, and in the nonuniformity of flow direc-

tion. Consequently, the prediction of flow-induced vibration and the probability of vibration damage must be considered somewhat uncertain.

Tubes are usually the most flexible part of an exchanger and therefore the most susceptible to vibration. The vibration is a natural consequence of fluids flowing in and past the tubes. In most exchangers the intensity of vibration is so low that it does not create a problem. Vibration becomes a problem when the intensity increases to the point that it causes some part of an exchanger to fail mechanically, it upsets the process conditions, or it creates a condition that endangers those who must work in the area.

Tubes vibrate at discrete frequencies depending primarily on their geometry, means of support, and material of construction. The lowest frequency at which tubes vibrate is called their fundamental natural frequency. The intensity of this vibration is evidenced by the amount of periodic movement of the tube, with the largest movement usually at **midspan** between supports. The extent of this peak-to-peak movement about the at-rest centerline is termed the amplitude of vibration. Energy must be fed to the tubes to excite them into vibration. This energy is provided by **flow** parallel to or across the tubes. Internal and external damping dissipate the energy. Prolonged tube vibration with large amplitudes leads to the mechanical failure of tubes, which then permits leakage between the **shell-side** and tube-side fluids. Mechanical failure usually occurs by one or more of the following:

Collision damage. If the amplitude of vibration is sufficiently large, adjacent tubes impact with one

another or against the shell and emit a loud noise. The tube walls can be worn **thin** with time and eventually split open. Tube impact produces characteristic diamond-shaped wear patterns at the **midspans** of the tubes,

Baffle damage. Baffle holes are made larger than tubes to facilitate exchanger bundle assembly, and thus the tubes are free to move at the baffles. The walls of vibrating tubes can be cut by the baffle, especially when the baffles are thin and made from a harder material than the tubes. The cutting action reduces the tube wall until leakage occurs.

Fatigue. Repeated bending of the tubes can lead to a breakdown of the tube material if the stresses are high enough and the vibration continues for a sufficiently long time. Fatigue failure can be accelerated by

corrosion and erosion. This type of **failure** results in cracks or in pieces of the tube wall actually breaking off.

Tube joint leakage. The joint between the tube and tubesheet can fail due to vibration whether welded or expanded.

In addition, vibration failure results from the cutting action of the edge of the tube holes against the tube wall where the tubes emerge from the tubesheet.

Although tube vibration failures have been reported in different parts of an exchanger, regions of long spans and high local velocities are particularly susceptible. The largest number of reported tube failures occurs in tubes that are within two or three tube rows of the tips of baffles, where the flow velocity is usually very high.

4.6.2

Tube bundle vibration characteristics

J. M. Chenoweth

■ Every component of a shell-and-tube heat exchanger vibrates at its own unique natural frequency. Tubes are the most flexible and thus the most easily excited. Tubes can and do vibrate at different frequencies. The lowest natural frequency is called the fundamental or first mode. The higher natural frequencies are referred to as the second mode, third mode, and so on. For conservative design purposes the fundamental frequency will be emphasized and will be called simply the natural frequency of the tubes.

The natural frequency of tubes like that of a simple beam depends on the way the ends are **fixed** (clamped or simply supported), the type of intermediate supports (simply supported, pinned, or clamped), the tube **cross-sectional** geometry, the number of spans, the materials of construction, and the length of spans. Although the natural frequency of the tubes can be measured experimentally, predictive methods are used to give approximate values of the natural frequency.

Tubes are rigidly fastened to the tubesheets and supported at intermediate points along their length by baffles or support plates. Some tubes **in** the center of a bundle may be supported by every baffle, whereas tubes that pass through baffle windows may be supported only by every second or third baffle. Furthermore, end baffle spacings are often longer than central baffle spacings to accommodate the nozzles and the shell flanges. Thus, the lengths of the spans are seldom uniform along the length of a tube, and not **all** tubes in a bundle are supported by the same number of baffles. This results in tubes with different natural frequencies in the same exchanger.

A. Straight-tube natural frequencies

Several different approaches can be used to predict the natural frequency of straight tubes. Most start with the assumption of a uniform beam clamped at least at one end with intermediate supports along its length. The rigorous approach is fairly complicated, with a good presentation of the method given by Gorman [1]. This method considers spans of unequal length between supports by writing the governing differential equation of motion for each span separately. The solution is obtained through the use of the boundary conditions at the ends of the tube and by relating the deflections and slopes at each intermediate support. This yields a set of linear homogeneous equations in **the** constants. The determinant of the coefficients is set equal to zero to yield the characteristic or frequency equation from which the natural frequency can be derived. Moretti and Lowery [2] have presented coefficients and graphic solutions for determining the natural frequency of tubes with various combinations of spans with two different lengths. They also give computer-plotted graphs for the shape of the various span configurations of maximum deflection. Computer programs such as NASTRAN [3], using the finite element approach, can provide accurate determination of the natural frequencies and mode shapes for tubes with unequal spans and varying degrees of intermediate restraint. Figure 1 is an example of the results from such a detailed analysis.

A somewhat less accurate, but normally adequate approach, by MacDuff and Felgar [4], assumes that all

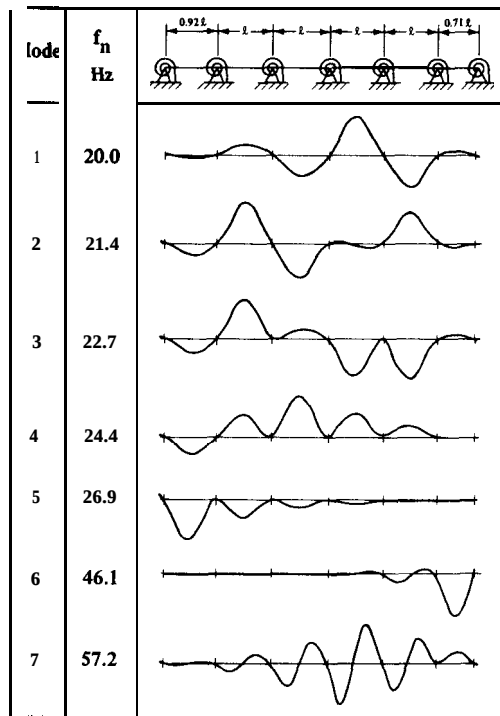


Figure 1 Natural frequencies and mode shapes for a tube with uneven span length and different rotational stiffness at supports.

spans are of equal length L . The natural frequency f_n is given by the equation

$$f_n = 73.03 \frac{C_n}{L^2} \left(\frac{EI}{M_e} \right)^{0.5} \quad (1)$$

where E is the modulus of elasticity of tube material, I is the area moment of inertia, and M_e is the effective mass per unit length. M_e is made up of three parts—the mass per unit length of the tube, the mass of the fluid inside the tube, and the virtual mass per unit length of tube for the shell-side fluid displaced by the tube. Figure 2 shows that the virtual mass is greater than that of the displaced shell-side fluid.

The frequency constant C_n depends on the way the ends of the tube are fastened, the number of spans, and the type of intermediate supports. Values of C_n for tubes clamped at the ends with simple intermediate supports are listed in Table 1. In addition to the values for the fundamental natural frequency (mode 1), the values for the higher natural frequencies (modes 2, 3, 4, etc.) are shown. Notice that with more than four spans the difference in C_n between successively higher modes becomes small. Further, the separation between the fundamental and the several higher-mode natural frequencies becomes small once the number of spans exceeds eight. Thus, for the design of most exchangers,

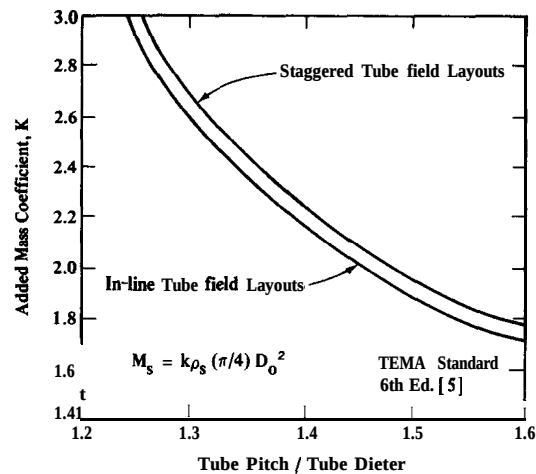


Figure 2 Added mass coefficient.

only the lowest natural frequency of the tubes needs to be considered.

The number of spans depends on the baffle arrangements and location of the tubes in the bundle. Most heat exchangers have their longest unsupported spans passing through the baffle windows and/or in the end zones. Segmental and double-segmental baffle arrangements result in window spans supported by every other baffle, whereas triple-segmental baffle arrangements are supported by every third baffle. Cross-flow exchangers and no-tubes-in-window exchangers provide support at each baffle and support plate. In the same bundle the number of spans may differ by one or two and the associated longest span lengths will vary. Since the natural frequency depends on both the length of span and the number of spans (see Table 1), all combinations in the bundle should be checked to determine the one that produces the lowest natural frequency.

Table 1 Frequency constant C_n in Eq. (1) for uniform beams of equal span length simply supported with extreme ends clamped

Number of spans	Mode				
	1	2	3	4	5
1	12.36	198.34	388.75	642.63	959.98
2	49.59	12.36	160.66	198.34	335.20
3	40.52	59.56	72.36	143.98	178.25
4	37.02	49.59	63.99	72.36	137.30
5	34.99	44.19	55.29	66.72	72.36
6	34.32	40.52	49.59	59.56	67.65
7	33.67	38.40	45.70	53.63	62.20
8	33.02	37.02	42.70	49.59	56.98
9	33.02	35.66	40.52	46.46	52.81
10	33.02	34.99	39.10	44.19	49.59
11	32.37	34.32	37.70	41.97	47.23
12	32.37	34.32	37.02	40.52	44.94

(a) Tube-to-baffle hole clearance effect on natural frequencies

Normal tube-to-baffle hole manufacturing clearances result in the baffles acting like simple supports. Kissel [6] found experimentally that the clearance had to be reduced to nearly a pressed fit before there was any appreciable change in the natural frequency. At about one-tenth the TEMA allowable clearance, the frequency was about 7% higher than that predicted for simple supports. For most exchangers, the tube-to-baffle hole clearance is not a significant parameter for controlling the natural frequency, but it may be important in determining damping and tube wear.

(b) Axial stress effect on natural frequencies

The tubes in a heat exchanger bundle may be under a tension or compression axial loading due to manufacturing procedures and/or to operating temperatures and pressures. Assuming that the axial force P_a is known, the following equation can be used to adjust the natural frequency:

$$(f_n)_s = f_n \left(1 + \frac{P_a L^2}{EI \pi^2} \right)^{0.5} \quad (2)$$

The sign for axial force P_a is minus when it is compressive and plus when it is tensile. Thus, tension increases the natural frequency, whereas compression decreases it. Kissel [6] found a variation in natural frequency due to axial loading in a typical heat exchanger of as much as $\pm 40\%$. Not knowing or considering the axial loading can lead to serious inaccuracy of natural frequency predictions.

(c) Finned-tube natural frequencies

To calculate the natural frequency of finned tubes, the area moment of inertia must be calculated using an effective diameter for the outside diameter. Limited tests [7] on low-finned tubes typically used in shell-and-tube exchangers suggest that a value of about 8% thicker wall for the finned section should be used along with the actual inside diameter under the finned section to calculate the area moment of inertia. The mass of the tube per unit length term in the effective mass M_e should be the actual mass for the finned section. The fins on tubes should not alter the assumed simple support constraint at the baffles.

(d) Span lengths

Of the variables that affect the natural frequencies, none is as influential as the length of the unsupported

spans. The natural frequency varies as the reciprocal of the span length squared. Early in the 1940s the TEMA *Standards* somewhat arbitrarily placed maximum allowable unsupported spans for different tube sizes and tube materials. Figure 3 is a plot of these maximum lengths. By coincidence, these lengths have prevented many serious tube vibration problems. However, the current trends toward higher velocities to improve heat transfer and minimize fouling make the TEMA maximum spans marginal for many exchangers. The current trend is to limit maximum spans to 80% of the TEMA values, which increases the natural frequency by more than 50%.

B. Natural frequencies of U-bend tubes

The vibration characteristics of a U tube are more difficult to predict than those of a straight tube. A detailed method is described by Lee [8] that requires a large number of inputs not generally available. As a result, a simplified approach is often used. This approach considers the straight portions and the bend separately. The straight portion is handled the same as straight tubes. The bend portion uses a method proposed by Kissel [9] and included in the TEMA *Standards* [5]. Assuming simple supports for the first-mode U-tube out-of-plane, natural frequency can be calculated by the following:

$$f_{nu} = 59.55 \frac{C_u}{R^2} \left(\frac{EI}{M_e} \right)^{0.5} \quad (3)$$

where C_u is the first-mode U-tube constant as shown in Fig. 4, and R is the tube bend radius. This simplified approach does not consider intermediate supports around the bend for U tubes that materially increase the natural frequencies by stiffening the overhung sec-

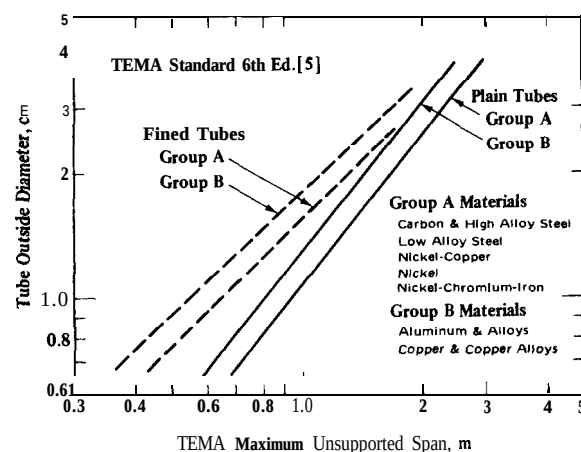


Figure 3 TEMA maximum unsupported span lengths for plain tubes and finned tubes (finned tube outside diameter is for plain end).

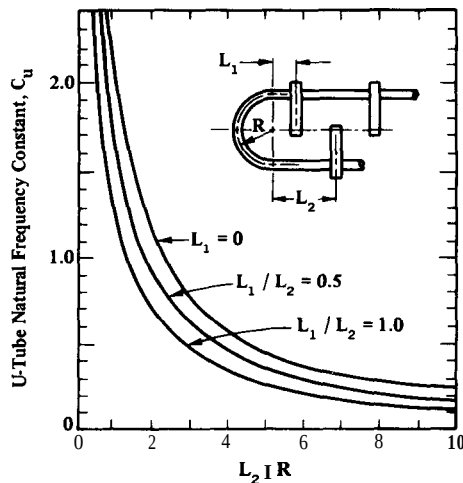


Figure 4 U-tube natural frequency constant for first-mode out-of-plane vibration adapted from TEMA Standards [5].

tion of the bundle. However, this complicates the prediction of the natural frequencies as the extent of their effectiveness is uncertain.

C. Amplitude of tube vibration

The natural frequencies are characteristics of the tubes themselves and are independent of the way they are excited or the amplitude of vibration. On the other hand, the amplitude depends on the natural frequency of the tubes, the mode shape, the frequency and the strength of the exciting mechanism, and the internal and external system damping.

The tubes in many shell-and-tube heat exchangers vibrate, but the amplitude is so small that the vibration is not a problem. For the amplitude to become appreciable the system damping must be small and the energy input large. For forced vibration of lightly damped structures when the ratio of the forcing frequency to the

natural frequency approaches unity, resonance occurs with an attendant amplification of the amplitude. See [10] for further discussion. Thus, if the forcing frequency can be kept appreciably below the natural frequency, the amplitude of vibration will be small.

Amplitudes as large as half the gap distance between adjacent tubes lead to impact collisions. Lesser amplitudes can cause tube cutting by the baffles or failure of the tube joints.

D. System damping

System damping has a strong influence on the amplitude of vibration. In any real system, once the energy exciting the vibration ceases, the amplitude of vibration decays with time. The rate at which the vibration dampens out is often exponential. The logarithm of the difference in successive amplitude peaks is called the log decrement δ_0 and is an indication of the damping. The higher the value of the log decrement, the more heavily damped the system is. Most tubes in heat exchangers are very lightly damped structures with low values of the log decrement.

Damping depends on the mechanical properties of the tube material, the geometry of the intermediate supports, and the physical properties of the shell-side fluid. Tight tube-to-baffle clearances and thick baffles increase damping, as does a very viscous shell-side fluid. Coit et al. [11] measured log decrements for copper-nickel finned tubes of 0.032 in still air. No method is available for predicting the log decrement, although the range of values most probably lies between 0.01 and 0.17 for tubes in heat exchanger bundles.

In the flow-induced vibration situation, if the energy input cannot be dissipated through damping, the amplitude of vibration will increase with time leading to a runaway condition. If there is a balance, the amplitude remains constant. If the damping exceeds the energy input, the amplitude diminishes with time.

Nomenclature and References for Section 4.6.2 appear at the end of Section 4.6.6.

4.6.3

Shell-side velocities in shell-and-tube exchangers

J. M. Chenoweth

● All predictive methods for flow-induced vibration contain a velocity term. The stream analysis method [12] provides a procedure for determining the fraction of the total that flows in each of five shell-side streams. These streams are the main cross flow, the bundle-to-shell leakage, the baffle-to-shell leakage, the tube-to-baffle leakage, and the bypass stream in any lane due to tube-pass partitions parallel to the cross-flow velocity. In most industrial heat exchangers the main cross flow is considerably less than the total flow. Further, as the shell-side fluid flows through a bundle of tubes, the velocity is constantly changing in magnitude and direction.

A. Cross-flow velocities

The definition for cross-flow velocity used by most investigators of flow-induced vibration is based on the minimum flow area through a tube row perpendicular to the primary direction of flow. For an ideal tube bank the selected velocity is well defined. However, for a shell-and-tube exchanger it is somewhat ambiguous, as the number of tubes in each row varies from baffle tip to baffle tip. In order to be consistent, the cross-flow velocity for shell-and-tube exchanger vibration prediction will be based on an integrated average area between the maximum and minimum number of tubes in the rows between baffle tips, on the gaps between adjacent tubes in a tube row, and on the cross-flow fraction of the total flow.

B. Parallel-flow velocities

The flow components parallel to the tubes include baffle-to-shell leakage streams, tube-to-baffle hole leakage streams, and most important, the flow through the baffle windows. The parallel-flow velocity is usually assumed to be the total cross flow plus the bundle-to-shell leakage stream flowing through the net window area. The parallel-flow velocities for multisegmental baffle exchangers are particularly difficult to define.

C. Leakage and bypass stream velocities

The leakage and bypass streams in a shell-and-tube heat exchanger are often disregarded, as they can be predicted only by a technique such as a stream analysis method. These streams can have velocities as much as 10 times higher than the main cross-flow velocity. Many of the tubes that have failed were in the proximity of one of these secondary flow streams. Heavy fouling can block the leakage through the tube-to-baffle clearances.

D. Entrance and escape velocities

The fluid entering the shell through the inlet nozzle impinges directly on the first row of tubes under the nozzle unless some type of impingement device is present. This device can be a row of dummy tubes, a solid plate, a perforated plate, or some type of bar

grid. With an impingement device the bundle entrance velocity should be limited. This may require the removal of several rows of tubes to provide sufficient area between the bottom of the nozzle and the impingement device. The TEMA *Standards* [5] specifies that the bundle entrance ρu^2 should not exceed 6 000 kg/m s². With impingement plates a region of high velocities may be found near the first rows of tubes below the impingement plate in the bundle-to-shell clearance areas. The flow in the end zone between the inlet and the first baffle window is very difficult to analyze because of the large number of possible flow paths. There can be areas where the flow is anything from highly turbulent to nearly stagnant.

At the exit of the exchanger a sufficient escape area must be provided to prevent velocities from

being high enough to initiate vibration in this region. These potentially troublesome areas are not easily analyzed and are often overlooked in a check for vibration problems.

E. Turnaround velocities

It is important to understand the flows in the baffle window turnaround regions, as most of the tubes that fail pass through them. Here the flow makes a reversal of direction between each cross pass. Velocity is continually changing in both magnitude and direction. Even though the flow at any point can be separated into its cross-flow and parallel-flow components, it is uncertain how to use these in the various predictive methods.

Nomenclature and References for Section 4.6.3 appear at the end of Section 4.6.6.

4.6.4

Flow-induced vibration phenomena

J. M. Chenoweth

■ There are several recognized phenomena that are associated with flow-induced vibration. These include vortex shedding, fluid elastic instability, turbulent buffeting, parallel-flow eddy formation, and acoustic vibration. Since any one of these can produce a flow-induced vibration problem, each must be considered in any comprehensive vibration analysis of a shell-and-tube heat exchanger.

A. Vortex shedding

Flow across a tube produces a series of vortices in the downstream wake formed as the flow separates alternately from the opposite sides of the tube as shown in Fig. 1. This shedding of vortices produces alternating forces, which occur more frequently as the velocity of flow increases. For a single cylinder the tube diameter, the flow velocity, and the frequency of vortex shedding f_{vs} can be described by the dimensionless Strouhal number Sr .

$$f_{vs} = \frac{Sr u_c}{D_o} \quad (1)$$

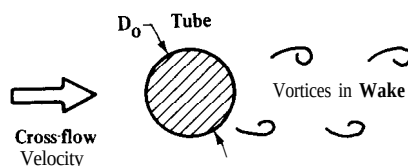


Figure 1 Sketch of vortex shedding resulting from flow across a tube.

For single cylinders the vortex shedding Strouhal number is a constant with a value of about 0.2. Vortex shedding occurs for the range of Reynolds number $100 < Re < 5 \times 10^5$ and $> 2 \times 10^6$, whereas it dies out in between. The gap is due to a shift of the flow-separation point in vortices in the intermediate trans-critical Reynolds number range.

Vortex shedding has also been observed for flow across ideal tube banks. The Strouhal number is no longer a constant, but varies with the arrangement and spacing of the tubes. Vortex shedding is fluid mechanical in nature and does not depend on any movement of the tubes. For a given arrangement and tube size, the frequency of the vortex shedding for nonvibrating tubes increases as the velocity increases. The vortex shedding can excite tube vibration when it matches the natural frequency of the tubes. The vortex shedding frequency can become locked in to the natural frequency of a vibrating **tube** even when the flow velocity is increased. The movement of the tube organizes the separation of the vortices leaving the vibrating tube.

The values of the Strouhal number for ideal tube banks are nearly constant for a wide range of Reynolds numbers, but vary considerably with longitudinal and transverse tube spacing. Some investigators [13,14] question if vortex shedding can exist deep within the bundle of a typical shell-and-tube exchanger. The argument is that there are so many changes of direction that an established cross-flow velocity is never realized.

Chen [15] organized experimental data into plots that relate the dimensionless Strouhal number to the geometry of staggered and in-line tube banks. It should be noted that there were few data points in range of the

industrially significant pitch ratios, typically from 1.25 to 1.75.

Usually the shell-and-tube exchanger tube fields are laid out with equilateral pitches in either square or triangular arrangements. It is therefore convenient to transform the plots by Chen into the one shown in Fig. 2. The Strouhal number is plotted versus the pitch ratio (equilateral tube pitch divided by tube diameter) for lines of constant tube field layout angle.

B. Turbulent buffeting

Extremely turbulent flow of the shell-side fluid contains a wide spectrum of frequencies distributed around a central dominant frequency, which increases as the cross-flow velocity increases. This turbulence buffets the tubes, which selectively extract energy from the turbulence at their natural frequency from the spectrum of frequencies present. This is an extremely complex form of excitation. As the result of investigations with gas flowing across an ideal tube bank, Owen [17] proposed an empirical equation to predict the dominant frequency for turbulent buffeting f_{tb} :

$$f_{tb} = \left[\frac{u_c D_o}{l_t} 3.05 \left(1 - \frac{D_o}{l_t} \right)^2 + 0.28 \right] \quad (2)$$

About the central dominant frequency for turbulent buffeting there is a band of frequencies of lesser energy content. Brothman et al. [18] suggest that the turbulent energy spectrum can be represented by a square band bounded by frequencies ± 9 Hz about the central frequency. When the dominant frequency for turbulent buffeting nearly matches the natural frequency, a considerable transfer of energy is possible leading to significant vibration amplitudes.

Equation (2) is not recommended when the shell-side fluid is a liquid. This is not to say that turbulent buffeting cannot occur for liquids, but rather that the

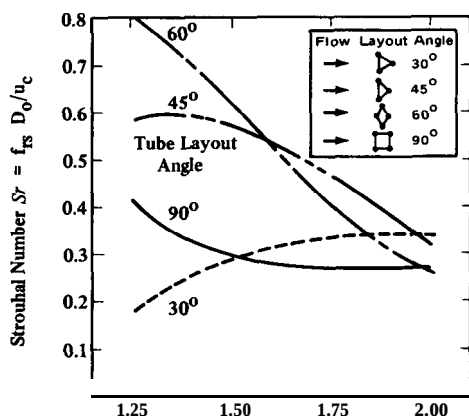


Figure 2 Strouhal numbers for equilateral tube layouts [16].

empirical predictions equation is based only on tests using gases as the shell-side fluid.

C. Fluid elastic whirling

Fluid elastic whirling is evidenced by tubes vibrating in an orbital motion. This motion is produced by flow across the tubes causing a combination of lift and drag displacements of the tubes at their natural frequencies. Typically, once fluid elastic whirling commences, it can lead to a runaway condition if the energy fed to the tubes exceeds that which can be dissipated by damping. Connors [19] developed a method for predicting the critical cross-flow velocity U_{crit} above which fluid elastic whirling can develop:

$$u_{crit} = \beta f_n D_o \left(\frac{M_e \delta_o}{\rho_s D_o^2} \right)^{0.5} \quad (3)$$

where β is the fluid elastic instability threshold constant and δ_o is the log decrement for the tube bundle in the shell-side fluid under static (no-flow) conditions.

Connors used air as the cross-flow fluid and controlled the amount of damping by mounting his tubes on different sets of springs. In his experimental setup he could easily determine the log decrement by striking a tube in still air and observing the decay of the vibration amplitude with time. At this time there is no method for predicting the log decrement. Reported measured values of log decrement vary widely, from 0.01 to 0.17. One value often quoted is 0.036, suggested by Coit et al. [20] for copper-nickel tubes in a large steam surface condenser. The log decrement depends on the mechanical properties of the tube material, the geometry of the bundle, and the viscosity of the shell-side fluid.

Shin and Wambsganss [21] assembled experimental data for the instability threshold constant β . Figure 3 indicates that β is a function of the tube field layout. Hartlen [22] determined the effect of both tube field layout angle and tube pitch-to-tube diameter ratio on β for banks of tubes shown in Fig. 4. These preliminary results also suggest that as the pitch-to-diameter ratio decreases, the value of β decreases.

For the Connors critical velocity method to be fully effective, it is necessary to predict the log decrement and the variation of β with tube type, layout angle, and tube pitch.

D. Parallel-flow eddy formation

Vibration due to axial or parallel flow results from the development of eddies along the tube. Nuclear reactors, and occasionally their associated heat exchangers, have experienced this type of vibration. Typically, they had long, unsupported tube spans, relatively narrow shell-

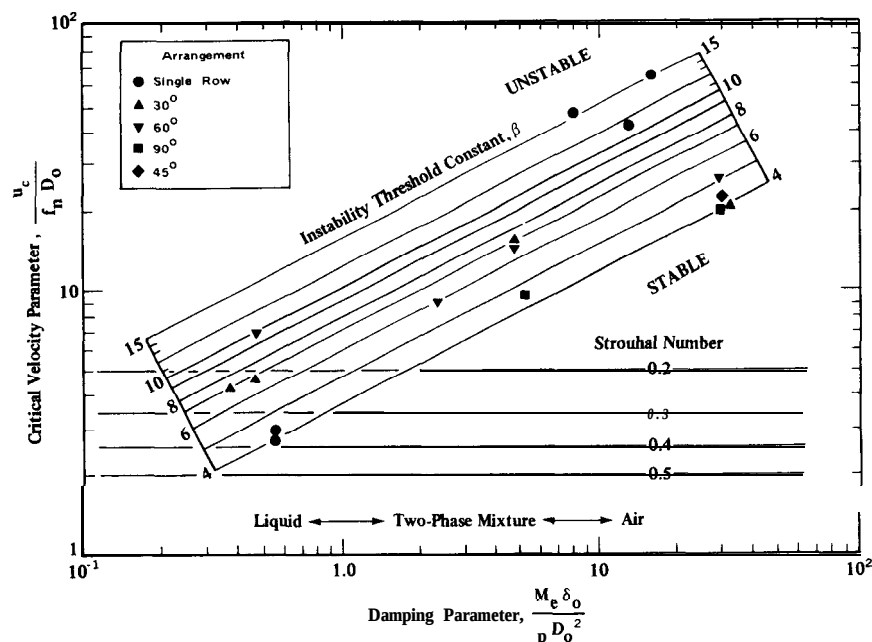


Figure 3 Stability diagram for tube arrays. Adapted from Shin and Wambsganss [21].

side flow passages, and very high axial velocities. The method of Chen and Weber [23] predicts the turbulent eddy frequency initiated by flow parallel to the tubes, which can excite the tubes into vibration at their natural frequency. The method uses an experimentally determined critical velocity to predict the midspan amplitude of vibration. In most industrial shell-and-tube heat exchangers, axial-flow-induced vibration is not a problem, as the axial flow in baffled exchangers is considerably below the critical velocity.

E. Acoustic vibration

Acoustic vibration occurs only when the shell-side fluid is a vapor or a gas. This type of vibration is related to

sound generated in an organ pipe. The characteristic frequency of the acoustic vibration in a heat exchanger depends on some length, usually the shell diameter d , and the velocity of sound in the shell-side fluid u_{sound} . The acoustic frequency f_a can be predicted by the following equation:

$$f_a = \frac{mu_{\text{sound}}}{2d} \quad (4)$$

where m is the mode number, which is a dimensionless integer. The lowest acoustic frequency is achieved when $m = 1$ and the characteristic length is the shell diameter. This is called the fundamental tone, and the higher overtones vibrate at acoustic frequencies two, three, or four times the fundamental ($m = 2, 3, \text{ or } 4$). The first two overtones are illustrated in Fig. 5. Reports

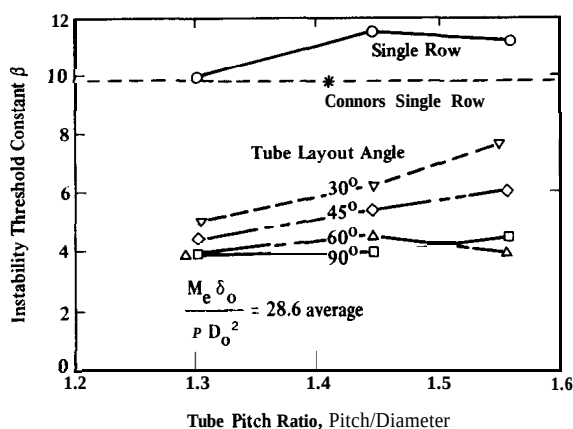


Figure 4 Instability threshold constants [22].

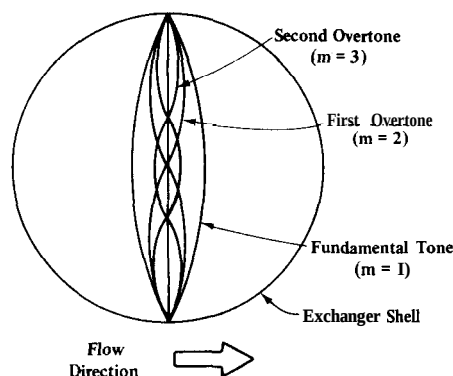


Figure 5 Most probable location of standing acoustic waves for fundamental and first two higher overtones.

of third, fourth, or higher overtones in heat exchangers are rare.

The acoustic frequencies of an exchanger can be excited by either vortex shedding or turbulent buffeting. **Barrington [24]** indicated that so long as the exciting frequencies are within 20% of an acoustic frequency, a loud noise may be produced. This acoustic vibration becomes destructive when it is in resonance with some component of the exchanger. Good designs ensure that the natural frequencies of the tubes differ from the acoustic frequencies of the exchanger shell. The acoustic frequencies of the shell can be changed by inserting a detuning plate parallel to the direction of cross flow to alter the characteristic length. This does not materially alter either the heat transfer or the pressure drop.

F . Combined phenomena

The question arises as to which of the phenomena are significant **in** predicting a vibration problem given the geometry and the process conditions. Once they are known, do they act independently or together to amplify or dampen the vibration? If they operate in phase, then amplification should be expected. If they operate out of phase, a canceling or damping should be expected. Answers to these questions are unknown at this time. To complicate the picture, most of the flow in an exchanger is flowing neither parallel to nor perpendicularly across the tubes. So, although parallel and cross velocities that can excite vibration in ideal tube banks are known, appropriate values to use for predictions of the same phenomena in a shell-and-tube heat exchanger remain uncertain.

Nomenclature and References for Section 4.6.4 appear at the end of Section 4.6.6.

4.6.5

Procedure for vibration predictions

J. M. Chenoweth

A. Primary check

A four-step primary check is suggested for shell-and-tube heat exchanger designs to minimize the probability of flow-induced vibration problems. The steps can be taken in any order and should be repeated for the inlet, central, and outlet regions. For a high probability of not having vibration, vibration must not be predicted at any step. The following are the four steps in the primary check procedure:

Step 1. For a gas or vapor as the shell-side fluid, compare the vortex shedding or turbulent buffeting frequency based on the cross-flow velocity to the acoustic frequency. If within 20%, acoustic vibration is probable.

Step 2. For either gas or liquid as the shell-side fluid, compare the cross-flow velocity u_c to the critical velocity to initiate fluid elastic instability u_{crit} . Vibration and damage are probable when u_c is greater than u_{crit} .

Step 3. For either gas or liquid as the shell-side fluid, compare the vortex shedding frequency to the lowest natural frequency of the tubes. Because of the number of uncertainties, vibration (not necessarily damage) is possible when the ratio f_{vs}/f_n is greater than some value less than 1, often 0.5.

Step 4. For gas or vapor as the shell-side fluid, compare the turbulent buffeting frequency to the lowest natural frequency of the tubes. Because of the number of uncertainties, vibration (not necessarily damage) is possible when the ratio of f_{tb}/f_n is greater than some value less than 1, often 0.5.

Notice that the primary check procedure predicts the probability of vibration and not damage. Certainly, preventing vibration from occurring precludes damage from vibration. On the other hand, the presence of vibration does not automatically infer damage.

B. Secondary checks

The primary check procedure examines only the most obvious causes of tube vibration. There are additional secondary checks that can be made. These include the velocities in the region of the inlet and outlet nozzles, high values of the cross flow ρu^2 , and the parallel-flow and cross-flow amplitudes. The leakage and bypass streams can be checked to see if they are high and may lead to local vibration problems. A check can be made to ensure that the acoustic frequency and tube natural frequencies are not in resonance.

C. Vibration damage relationship

Vibration cannot be equated to tube damage. Many exchangers vibrate but do not experience tube failure. Damage is known to result from fatigue, tube-to-tube collision, and baffle-to-tube cutting. Fatigue is better understood than the other two mechanisms. Tests that simulate the motion, forces, tube materials, and support materials, etc., found in real exchangers are needed to answer questions about low-amplitude (low-stress) vibration for a very large number of cycles.

D. Comparison of predictions with experimental data

Reliable experimental data on flow-induced vibration and tube damage from operating full-scale shell-and-tube exchangers are very scarce. This is not because there are not exchangers that vibrate and sustain damage; rather, it is because obtaining the data to make a meaningful vibration analysis is extremely difficult. Understandably, plant operating personnel are more interested in eliminating vibration problems than obtaining experimental data.

Several vibration tube banks [25–27] have been assembled containing cases where exchangers had experienced flow-induced vibration problems and cases where similar exchangers had no reported vibration

problems. There are more cases **with** reported vibration than cases with no reported vibration, so the data banks are biased. There are cases where the **shell**-side fluid was liquid, gas, or vapor. Of all the cases having vibration, more than half reported damage at the baffles. Other cases indicated collision damage, **tube**-to-tubesheet joint failure, noise, and failure in the U-bend region. As is typical of field data, there are cases with incomplete information. In many cases the missing data are essential to make a meaningful analysis.

The primary check procedure was followed for the cases in two vibration data banks [27] for the inlet, central, and outlet baffle spacing regions of the exchanger. Correct predictions were made for 48 of the 67 cases, or 72% correct. Of the cases with no vibration reported, only about 5% were predicted correctly. The evaluation is thought to be reasonably typical.

Nomenclature and References for Section 4.6.5 appear at the end of Section 4.6.6.

4.6.6

Design considerations

J. M. Chenoweth

● At the design phase there are a number of changes that can be made to reduce the vibration problem. Most require a compromise to satisfy all of the requirements with added cost a likely result. When an exchanger already in service has a serious problem, there are still some improvements that can be made in the field.

A. Possible design changes to correct anticipated vibration problems

If, as the result of a vibration analysis it is found that vibration is probable, design elements can be reconsidered from the standpoint of vibration. Assuming that the process conditions cannot be changed, the following changes should be considered:

Reduce the shell-side velocities. If the flow rate of the shell-side fluid is **fixed**, velocities can be reduced by increasing the tube pitch or using TEMA X or J shell styles. This is particularly attractive when the design is pressure drop limiting, but may result in a larger shell.

Increase the tube natural frequency. The most effective way to increase the tube natural frequency is to reduce the longest unsupported span length. Reducing the span length by 8% increases the natural frequency by more than 5%. Secondary effects can be produced by changing tube materials and increasing tube wall thickness, but neither of these will greatly increase the natural frequency within industrially acceptable limits. The natural frequency can be increased by lacing or by driving wedges between tubes to prevent movement. This technique has been particularly effective in

controlling vibration in the bend region of U-tube bundles.

Reduce nozzle velocities. The nozzle sizes can be increased to reduce nozzle velocities. If an impingement plate is provided to prevent erosion, be sure that the velocities leaving the edges of the impingement plate are not excessively high. A support plate at the centerline of the nozzles gives added tube support near the **midspan** at the end zone. Annular distributors and vapor belts are effective ways of reducing the velocity of the shell-side fluid entering the bundle.

Change baffle type. The no-tubes-in-window type bundle can have its baffles widely spaced, as every tube is supported by every baffle. Further, the **no-tubes-in-window** construction permits intermediate support plates between each baffle without materially affecting the heat transfer or pressure drop performance. A change to a multisegmental baffle will reduce the velocity. Although not common practice, the rod **baffle**-type bundle [29] has been used in low-pressure-drop exchangers.

Add detuning baffles. Acoustic vibration problems can easily be corrected in the design phase by the use of detuning baffles to reduce the characteristic length (see Barrington [22]).

Reduce wear at baffles. Although the use of tighter tube-to-baffle clearance and thicker baffles do not materially change the tube natural frequency, they do reduce damage due to cutting and increase the system damping. A change of baffle material can sometimes reduce tube damage if originally the material was considerably harder than the tube material.

B. Possible solutions for observed vibration problems

Once an operating heat exchanger is found to have a vibration problem, it should be determined if the vibration is transmitted to the exchanger from some external source. If it is ascertained that the vibration is flow induced, there are a number of possible solutions to be considered:

Plug leaking tubes. A temporary fix for leaking tubes, whatever the cause, is to drive plugs into the tube ends to seal off the leaking tube. Replacing a tube in the field is generally not practical.

Reduce shell-side flow rate. Flow-induced vibration often can be stopped by lowering the shell-side flow rate. This will be acceptable only if it can be tolerated within the operating requirements of the plant.

Insert bars, lacing, or wedges to stiffen the bundle.

The natural frequencies of existing exchangers can be increased by inserting lacing or driving wedges between the tubes to restrict movement. This can result in altered thermal and pressure drop performance. One region where lacing and wedging is often used is in the U bend.

Remove tubes to create bypass lanes. This procedure can reduce the cross-flow velocities enough so that vibration can be controlled. However, this must be done with care, as it can significantly reduce the overall heat transfer and pressure drop performance.

Replace the tube bundle. As a last resort, redesign and install a new bundle that circumvents the vibration problem.

Vibration problems are best prevented rather than corrected. A careful analysis at the design stage can greatly reduce future vibration problems.

NOMENCLATURE FOR SECTIONS 4.6.1-4.6.6

C_n	frequency constant, dimensionless [see Table 4.6.2(1)]	M_m	mass of tube per unit length, kg/m
C_u	first-mode U-tube frequency constant, dimensionless [see Fig. 4.6.2(4)]	M_s	virtual mass per unit length of tube for the shell-side fluid displaced by the tube $[= k\rho_s(\pi/4)D_o^2]$, kg/m
d	characteristic length for acoustic vibration; usually shell diameter for heat exchangers, m	M_t	mass per unit length of tube for the tube-side fluid, kg/m
D_i	tube inside diameter, m	P_a	axial load, — compressive or + tensile, N
D_o	tube outside diameter, m	R	U-tube bend radius, m
E	modulus of elasticity of tube material, N/m ²	$\tilde{R}$	gas constant, 8.314 kg m ² /s ² (mol) K
f_a	acoustic frequency, Hz	Sr	Strouhal number, dimensionless
f_n	natural frequency, Hz	t	transverse tube pitch in a tube row, m
f_{ns}	stressed frequency, Hz	T	absolute temperature, K
f_{nu}	natural frequency, out-of-plane U-bend, Hz	u_c	cross-flow velocity in the gaps in a tube row, m/s
f_{tb}	dominant turbulent buffeting frequency, Hz	u_{crit}	critical velocity for fluid elastic instability, m/s
f_{us}	vortex shedding frequency, Hz	u_{sound}	velocity of sound in shell-side fluid $[= (Z\gamma\tilde{R}T/M)^{0.5}]$, m/s
I	area moment of inertia $[= \pi(D_o^4 - D_i^4)/64]$, m ⁴	Z	compressibility factor, dimensionless
k	added mass coefficient, dimensionless [see Fig. 4.6.2(2)]	β	instability threshold constant, dimensionless
l	longitudinal tube pitch between successive tube rows, m	γ	specific heat ratio C_p/C_v , dimensionless ($\gamma \approx 1.4$ for many fluids)
L	unsupported span length, m	δ_0	log decrement of the tube bundle in the shell-side fluid under no-flow conditions, dimensionless
m	mode number, dimensionless integer	ρ_s	shell-side fluid density, kg/m ³
M	molecular mass, g/mol		
M_e	effective mass per unit length $(= M_m + M_t + M_s)$, kg/m		

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4.7 Testing and inspection

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4.7.1

Objectives of inspection, testing, and drawing approval

A. Illingworth

■ Heat exchangers are complex, highly sophisticated pressure vessels, and never more so than when designed with the precision and thoroughness of computer-aided programs. In these circumstances, with the final design approaching nearer and nearer to the theoretical ideal, it is essential that they be fabricated to the required tolerances and to a high standard of workmanship. Thus inspection can no longer be regarded as a desirable appendage but must be an essential, built-in feature of the fabrication process. There are two inspection functions, the manufacturer's own in-house quality control, and third-party inspection by approved authorities who are empowered to issue certificates acceptable to most international inspection authorities. The functions of the two inspection organizations are separate and independent, but complementary in requiring the same end result—a soundly constructed and safe heat exchanger.

Third-party inspection must, by virtue of its responsibilities to government legislation, be more comprehensive than in-house inspection, and commences with design assessment of the pressure-containing parts. Practice varies from country to country; in the United Kingdom normally triplicate copies of the construction drawings should be forwarded to the relevant inspection organization who, when satisfied with the design, will confirm their approval in writing or, if necessary, by stamping the drawings—returning one copy to the manufacturer, forwarding a second to their inspector, and retaining the third copy. Manufacturers should not proceed with construction be-

fore drawings have been approved, and a similar arrangement should be followed with drawing revisions. It is generally necessary in the case of orders for export contracts that the drawings for approval and any revisions be forwarded in ample time to the appropriate inspection or certification authority of the country where the plant will eventually operate. The drawings should be translated into the language of the importing country, dimensions should be in the correct units, and the design should be strictly in accordance with the requirements and regulations applicable within the importing country. The recipient authorities will often delegate inspection to a recognized inspection authority in the manufacturing country or confirm the particular inspection arrangements. Once the drawings have been approved, the third-party inspection organization will carry out the required stage inspections to ensure that the completed vessel is constructed in accordance with the specification and the approved drawings. An important aspect of inspection is the scrutiny of radiograph films and ultrasonic reports and witnessing dye penetrant, magnetic particle, hydraulic, pneumatic test, etc., as required. Upon satisfactory completion of manufacture and testing, the vessel nameplate is branded with the inspection authority's mark, and a certificate is issued.

It is normally necessary and indeed good practice for the manufacturer to produce a dossier containing copies of the approved drawings, material certificates, nondestructive testing results, welding certificates, and the inspection authority's certificate. For certain countries details may have to be recorded

in prescribed forms and, of course, there are usually additional requirements to be met; for example, some countries require by law that the copper rivet heads attaching the vessel nameplate be marked with the inspection authority's stamp.

Close liaison between the manufacturer and the inspection authority is essential to ensure that the construction documentation is in order and acceptable to the overseas authority.

4.7.2

Materials

A. Illingworth

■ A wide variety and combination of materials are used in the construction of heat exchangers, as discussed in Sec. 4.5. Generally the material of construction should comply with the requirements of the design code, but other materials may be used subject to agreement between purchaser, inspecting authority, and manufacturer.

In the case of heat exchangers that are exported, the design, testing of materials, inspection, and certification must comply with the requirements of the regu-

lating authority of the recipient country. It is therefore essential that the manufacturer be fully aware of these requirements at the outset, and such items as valves and fittings should not be excluded when considering these requirements. Nonpressure parts, such as supports, lugs, and baffles, should be of weldable quality and suitable in all respects for the intended purposes.

All material should have identification marks and be supported by certificates giving the physical and chemical properties.

4.7.3

Welding

A. Illingworth

■ The manufacturer must have well-maintained welding equipment and experienced welding supervisors. No production welding should be carried out until the welding procedures have been approved and the welding operators qualified. The standards generally used for weld procedure and welder qualification in various countries are summarized in the Guides to National Practice in Sec. 4.3. The manufacturer is responsible for the documentation and records of weld procedures and

welder approvals, which must be available to the inspection authorities. Weld certificates accepted by a recognized inspection authority should be accepted by other inspecting authorities subject to the certificates being current, correct, and suitable for the welding under consideration. This acceptance should also be applicable to ultrasonic testing procedures and ultrasonic operators certificates.

4.7.4

Stages of inspection

A. Illingworth

■ Inspection by an outside body in no way relieves the manufacturer of responsibility to the purchaser. The inspection stages should be varied to suit the design and materials of the heat exchanger, but the following are the main inspection stages (see also particular design and construction code for specific and mandatory requirements, which **will** of course have to be complied with):

1. Identification of material against material certificates.

2. Checking that the weld procedures and welder approvals have been authenticated by an acceptable inspection body or qualify as necessary. This is also applicable to nondestructive testing procedures and approvals.

3. Examination of part when cut to size and shaped with edges prepared for welding. At this examination attention should be given to the plate edges to ensure that they are free from flaws, cracks, laminations, and slag inclusions.

4. Examination during welding to ensure that the correct procedures are being used and that the parts are within the tolerance specified in the design code.

5. Check to ensure that heat treatments have been carried out as required.

6. Examination of completed welding and witnessing the nondestructive testing where applicable.

7. Examination of tube plates with tube holes drilled. At this stage consideration should be given to the nondestructive testing of the tube plates to ensure that they are free from any unacceptable laminations.

8. Examination of tube bundle assembled in tube plate(s).

9. Witnessing pressure tests and leak tests where applicable.

10. Final examination and checking of exchanger dimensionally against the drawing.

11. Checking marking and stamping of the exchanger and examination of **final** documentation.

4.7.5

Construction

A. Illingworth

■ In the United Kingdom the Health and Safety at Work Act 1974 has brought to the attention of both users and manufacturers of shell-and-tube heat exchangers, and indeed of all pressure plant, the vital need for efficient lines of communication.

The act firmly defines the duty of any person who designs, manufactures, imports, or supplies any item of plant and invests in that person the need to design and construct in such a way that plant is safe and without risk to health when it is properly used.

The act states that where published standards exist, designers and users of plant should pay heed to them and warns that, although standards may have no legal status in the United Kingdom, industry should realize that it may have to explain its reasons for diverging from them should an incident jeopardizing health and safety occur. Some other countries also have quite specific legislation and requirements with respect to imported plant, and it is essential that such information be obtained and considered. The foregoing principles either are or are likely to be introduced in many

countries and form a sound basic principle for future designs.

In Sec. 4.3, established codes of practice are discussed that provide in the majority of cases regulations governing the design of acceptable vessels. Many branches of industry are users of shell-and-tube heat exchangers for heating, cooling, evaporating, and chemical reaction duties, and the purpose of the standards is to specify the essential requirements and to give constructional guidance. Some standards do not deal with any thermal design considerations and rely on fullest operating information being passed between purchaser and manufacturer.

If steam is used on one side of a heat exchanger in the United Kingdom, suitable fittings to comply with the Factories Acts must be provided. Items in this category together with their fittings are by statute required to be inspected periodically at prescribed intervals by a competent person, and a report must be issued. Other in-service regulations exist in most other countries with respect to these unfired steam vessels, and it is prudent for these to be considered, **preferably** at the design stage.

4.7.6

Nondestructive testing

A. Illingworth

A. Introduction

Depending on the severity of duty required of a heat exchanger when in service, various methods of non-destructive testing are available to provide that the parent material and that the welds deposited are sufficiently free from defect to satisfy the known service conditions. During fabrication radiographic, ultrasonic, magnetic particle, and dye penetrant tests are available inspection techniques. The various design codes specify nondestructive testing in varying degree for different vessel categories, and the choice of technique for a particular application requires careful consideration.

There are **internationally** recognized certification schemes to qualify personnel for radiographic, ultrasonic, magnetic particle, and dye penetrant testing, and in the case of ultrasonic inspection it is particularly important that personnel are suitably qualified to undertake work of this nature.

If there is any danger of heat treatment cracking, nondestructive testing should be carried out after heat treatment and the design code requirements followed in this respect.

After acceptance of the nondestructive tests, the integrity of the finished vessel is further established by a pressure test, usually hydraulically applied, although in certain circumstances with adequate safety precautions a pneumatic test may be carried out. The stages at which various forms of nondestructive testing are employed are discussed below.

B. Plate material

Plates are tested ultrasonically for detection of laminar imperfections in accordance with the relevant codes. It may be necessary to make a more comprehensive ultrasonic examination prior to welding of plate edges adjacent to welded seams and regions around branch openings.

C. Weld tests

Welding procedure, welder approval, and production control test plates are usually required to be assessed radiographically or ultrasonically.

D. Main seams

The choice of radiographic or ultrasonic testing or a combination of both is a matter for agreement among all parties involved. Each method has advantages and disadvantages with regard to the detection, identification, sizing, and disposition of flaws. Radiography is most suited for the detection of inclusions, blowholes, piping, porosity, and incomplete root penetration, except in cases where the root faces are butted tightly together. On the other hand, ultrasonic testing is particularly suited for detection of planar-type defects such as cracks and lack of fusion and is to be preferred for testing thick sections. The relevant codes describe the methods and requirements for radiographic tech-

niques, or ultrasonic examination of fusion-welded butt joints in ferritic steels.

In some instances the choice between the two methods is governed by the plate thickness, but often the application of each technique is complementary and for maximum inspection coverage in critical situations consideration should be **given** to specifying both radiographic and ultrasonic examination.

Magnetic particle or dye penetrant tests are usually specified for inspection of the back gouging of the first side welding to ensure that the second side weld is deposited on sound metal. These tests may also be **specified** for checking the weld surface and adjacent parent plate, particularly after heat treatment where **circumstances** make this necessary.

E. Branch welds

If the application of ultrasonic testing is necessary in the case of branch welds where the geometric shape does not allow the use of radiographic techniques, methods for the ultrasonic examination of branch welds are given in the relevant codes. Additional checks for surface flaws, particularly cracking from toes of fillet welds, should be detected by magnetic particle or dye penetrant examination.

F. Tube-to-tube plate welds

Apart from certain sophisticated attachment techniques specifically designed for high-integrity applications that can be examined radiographically, tube-to-tube plate weld attachments are usually examined by magnetic particle or dye penetrant techniques.

G. Pressure testing (overpressure test)

The overpressure test is carried out at a pressure exceeding the design pressure and is applied to the shell side

and tube side of heat exchanger vessels independently in order to prove the mechanical strength and integrity of fabrication and design. It is usual to carry out the test hydraulically, and care must be taken to ensure that the vessel is properly vented to exclude the possibility of air pockets.

Pneumatic testing should be avoided wherever practicable, but where it becomes necessary adequate safety precautions should be taken in view of the possibility and consequence of explosive failure.

It is imperative yet frequently forgotten that when draining a vessel after testing, a vent valve must be opened before commencing the drainage operation.

H. Proof test

Where the strength of a vessel (or part) cannot be assessed by calculation, a proof hydraulic test is required in conjunction with a strain gauge or strain coating technique in order to establish a design pressure. The rules of the appropriate design codes should be followed together with the detail procedures adopted in the codes.

I. Leak testing

After completion of the overpressure test, a gas tightness test may be carried out. There are techniques using air and soapy water or search gas detection, and the test is particularly useful in checking tube/tube plate attachment welds.

In all cases the applicable design code rules for pressure testing should be observed together with any national requirements.

Upon completion of construction, the vessel should be stamped on a prominent part such as flange edge or nameplate to show the pressure tests, the date, whether the vessel was stress relieved, the code of construction, and the manufacturer's name together with the inspection authority's stamp and other relevant data.

4.7.7

Certification

A. Illingworth

■ To ensure that a manufacturing record can be kept throughout the life of the vessel, the following certificates and records should be documented where applicable :

1. Material certificates of chemical and physical properties for each vessel item
2. Welding procedure certificates
3. Welder qualification certificates
4. Records of welding consumables used and their properties

5. Certificate of radiography
6. Certificate of ultrasonic testing or other non-destructive test operations
7. Record of heat treatments applied
8. Certificate of hydraulic tests
9. Reproduction of vessel markings and stampings
10. The vessel detail drawing together with a materials list

4.7.8

Fittings

A. Illingworth

■ It is essential that wrought and cast fittings and valves that are normally purchased outside are subjected to the prescribed inspection and testing requirements, for on occasion the documentation of these components and the relevant inspection has been overlooked with consequent delivery difficulties.

It is of considerable assistance to the inspection

authority if the manufacturer sends copies of suborders to facilitate such inspections. Irrespective of the arrangements made for inspection, valves and fittings should be subjected to the necessary checks before fabricating or fitting into the heat exchanger. The degree of inspection is normally defined in the applicable construction standard.

4.7.9

Cleanliness and storage

A. Illingworth

■ Extensive damage and corrosion can be created in heat exchangers and other plant by debris being left within, and it is normal practice for experienced purchasers to lay down specific requirements with respect to the degree of cleanliness, protection, and preservation and also storing and shipping requirements to prevent such damage. In the absence of such requirements, manufacturers should, in their own interest, take meticulous care in this important function to ensure that all compartments are in the correct condition and inspected before closing. It is recommended that steps be taken to ensure that

1. All surfaces are free from loose scale and debris and cleaned to a degree suitable for the process or con-

nected plant. Note: Care should be taken to ensure that cleaning or testing fluids do not adversely affect or damage any of the materials, e.g., stress corrosion.

2. All fluids used for testing and cleaning are drained off and surfaces dried.

3. The surfaces (where appropriate) are suitably coated with a removable rust preventative or alternative means that meets the purchasers requirements and where necessary gives adequate protection in any possible adverse environmental conditions such as in sea transport.

4. Flanges, screwed threads, and other similar details are protected from mechanical damage by suitable covers or caps, which should also be capable of preventing the ingress of dust and debris, etc.

4.7.10

Preparation and dispatch

A. Illingworth

■ Before a heat exchanger is dispatched, it is wise to ensure that all inspection requirements by all parties concerned have been completed and that all documentation with respect to materials, fabrication, heat treatments, and testing (including those for valves and fitting) are procured and made available for the purchaser and the inspection authority.

In transit, special precautions should be taken to ensure that no abnormal vibration, excessive loading, or mechanical damage results-the latter usually being caused by loading and unloading operations or by storage in unsuitable places. It is recommended that one should consider the provisions made for insurance coverage against such damage.

4.7.11

Quality control and inspection disciplines

R. S. M. Moffatt

■ The previous sections deal with inspection and testing during the fabrication of pressure vessels (including heat exchangers), embracing dimensional control, radiography, ultrasonics, magnetic and dye penetrant crack detection, and final testing. It is therefore necessary for the fabricator to have in operation an overall quality program, run by qualified and experienced personnel, which has been approved by an independent organization.

In the United States the organization concerned with the approval of companies for particular fabrications is the American Society of Mechanical Engineers (ASME) through its boiler and pressure code. This organization will approve fabricating companies outside the United States as well as American companies.

The system involves the submission of a quality control manual by the fabricator, which is reviewed, together with an audit of the company's facility. Registered fabricators are authorized to construct fabrications to the appropriate code and stamp their vessels with the code stamp.

Similar schemes are in existence in other countries; for example, in the United Kingdom the approved scheme is operated by the Pressure Vessel Quality Assurance Board (PVQAB). In all cases the pressure vessel fabricator is overseen by an authorized inspector, who must be suitably qualified. Certain authorities will only allow pressure vessels to be installed and operated in their country if they have conformed with the above systems.

Total quality systems are all basically similar and in general must conform to the following requirements, which are summarized in the company's quality manual.

A. Organizational structure and responsibilities

The fabricator must maintain a documented quality system (as defined in the quality manual). The manual must contain an organizational structure that clearly defines the responsibility of all key personnel, their authority, and their relationship to one another. The organizational charts should show this relationship between engineering (design and drawing), purchasing, actual fabrication, inspection, and quality control.

An overriding requirement is that quality control personnel should report to a level of management such that the pressures of production and costs **do** not affect their authority and independence to make unbiased decisions.

It is also necessary for the head of the organization, normally the managing director, to sign a statement committing the company to quality objectives and reinforcing the responsibility and authority of the person assigned to the position of quality manager.

B. Design and drafting control

It is, of course, of paramount importance that the item is designed correctly and meets the relevant design code and the client's requirements. These requirements must be correctly and clearly translated onto the drawings from which the item will be fabricated, inspected, and tested.

Control procedures must therefore be developed

and documented to ensure that these design requirements are met and are adequately checked by qualified personnel.

In determining detailed design, account must be taken of **both** fabrication and nondestructive examination (NDE) techniques, and it must be ensured that adequate access is present for the welder or welding machine to work satisfactorily.

C. Control of production drawings, specifications, and procedures

It is essential that when all documents have been produced and approved they are circulated to those personnel who are involved in performing the particular activity. The system must also be such that when any document is revised the latest revision is formally circulated and all previous issues withdrawn at the same time. This activity should be documented to demonstrate that only documents of current issue are in the hands of interested parties.

D. Control of purchased items

Controls must be available to ensure that purchased items are ordered correctly, so that they will be produced by the supplier in accordance with the purchase order and referenced specifications and tested to approved procedures.

Controls shall include, as necessary, actual witness inspections by the purchaser's inspection representative and the authorized inspectors.

Test certificates shall be produced by the supplier, suitably authenticated by inspectorates, and these certificates should be sent to the purchaser with a minimum of delay.

Items should be purchased only from suppliers who are on the company's approved supplier list, compiled by the company on the basis of either (1) a quality audit of the company or (2) a history of satisfactory supply of similar items in the past.

E. Material control

On receipt of purchased items into the company, a suitable system shall be applied which ensures that they are correctly identified and meet the requirements of the purchase requisition. This can be done by recording the identity of the item and checking it against the test certificate. In addition, it should be inspected to ensure that there is no damage. Additional tests may be carried out,

including, for instance, checking dimensions and finish. Test certificates should also be checked against **the** specification and purchase order for conformity and stamped as correct. Items should not be released for manufacture until they have cleared this particular activity. Items should be released for fabrication only on receipt of a requisition from the fabrication department ordering the specific item. It is essential that **the** system ensures that only the intended material is used.

During manufacture the identity of all items should be maintained at all times. Before cutting pieces from larger items, a system of transfer of the identity of each smaller piece, prior to cutting, shall be maintained. This degree of traceability may be achieved by marking the pieces clearly with an indelible marking, such as by stamping or vibroetching a unique number or marking that enables **the** piece to be linked to its test certificate. The marking should be in such a location **that** it is visible on the finished component, whenever this is practicable.

F. Inspection, including nondestructive examination

The quality system shall ensure that all the mandatory inspections and tests are carried out at **the** appropriate times and to the approved procedures. This can be accomplished by adherence to a quality plan that lists in sequential order the operations to be carried out by **both** the fabricator's inspector and the authorized inspector. The plan will also specify the hold points at which work must not proceed until specific inspection has been carried out and accepted.

A typical quality plan format that could be used is shown in Fig. 1. This is compiled prior to commencement of the contract and lists operations in a sequential manner from receipt of order to **despatch** of **the** fabrication, together **with the** degree of inspection required by the fabricator and by other interested parties such as the authorized inspector and the client.

It is desirable to conduct a **preinspection** meeting prior to commencement of work with all interested parties to agree on all the inspection activities and their degree of actual inspection.

The system should be such that all the fabrication and NDE procedures required for the contract are identified, written by **the** relevant departments, and approved by the inspection bodies, all in a timely manner to ensure that they are available when required. These procedures and their updating as necessary should be issued to the relevant personnel in accordance with the requirements outlined in Sec. C.

To ensure **that all the** tests and inspections are **car-**

Quality Plan										REVISION No. 0		SHEET 1 OF 1	
DESCRIPTION ASSEMBLY OF SHELL HEAT EXCHANGER ITEM H 101			ORDER No. & CONTRACT 623890			QUALITY PLAN APPROVALS							
INSPECTION AUTHORITY			CLIENT										
OP No.	OPERATION	CONTROL DOCUMENTS / PROCEDURES	RECD BY CLIENT	RECORDS (C.E.S. No.)	INSPECTION / TEST CODE & COMPLETION				INSPECTION IDENTIFICATION AND ADDITIONAL DATA				
					Q.C.	Q.A.	CODE	CLIENT	DATE	AUTH INSPECTOR			
1.	Assemble dislised end to strake C1.										36 to 20		
2.	Assemble core to strake C2.	SPECIMEN COPY									19 to 20		
3.	Assemble flange to cone C3.										22 to 19		
4.	Inspect circ. seam set ups C1	Weld Proc. No. 403A		3408	AI INSPECTOR					A.I. DATE	HOLD		
	C2	Weld Proc No. 403A		3408	AI DATE					AI			
	C3	Weld Proc No. 680		3408	AI					AI			
5.	Weld 1st side and back chip C1			3408	AI INSPECTOR					A.I. DATE	HOLD		
	C2				AI DATE					AI			
	C3				AI					AI			
6.	Weld 2nd sides and flush inside.			3408	S								
7.	Spot X-ray circ. seams	623890/NDE/PT1 ASME VIII - NDE2			AI INSPECTOR					AI			
					DATE								
8.	Deposit weld cladding	Weld Proc No. 944			S					A.I. DATE	HOLD		
9.	Dye penetrant test cladding	623890/NDE/PT1 ASME VIII - NDE5			AI INSPECTOR					AI			
					DATE								
10.	Re-spot X-ray welds	623890/NDE/PT1 ASME VIII - NDE2											
DRAWING Nos. 164603 - 164605			REFER TO CONTROL DOCUMENT STATUS SCHEDULE No.			DESIGN CODE ASME VIII Div. 1 and Clients Specs.			ITEM / ASSEMBLY No.				
PREPARED BY P. Maddison	DATE 1.10.80	INSPECTION/TEST/SURVEILLANCE CODE 100% ACTUAL INSPECTION/TEST 100% ACTUAL INSPECTION/TEST 100% WITNESS INSPECTION/TEST 100% WITNESS INSPECTION/TEST OPERATION SURVEILLANCE MANDATORY HOLD POINT											
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Figure 1 Sample quality plan format.

ried out, a system should be used whereby each activity is signed off by the appropriate inspectors to indicate that the activity has been completed satisfactorily and the appropriate reports produced. This can take the form of an inspection checklist or similar follower **that** accompanies the item concerned. This checklist also serves as an instant reference to the inspection status of the item. A reproduction of the quality plan can be utilized for this purpose.

G. Correction of nonconformities

If, during fabrication, items are found not to conform to the specification or drawing requirements, the quality system shall be capable of ensuring that work is stopped pending resolution of the nonconformity. This may be done by using a hold note **that** explains why the item is held, and also a label attached to the part clearly indicating that the piece is on hold. Small items should be segregated to ensure that no further work is done.

Certain nonconformities that do not infringe the actual code requirements can, in fact, sometimes be accepted from a process point of view. A system must therefore be available to formally submit the concession to the client for approval. In cases of acceptance of the concession the hold can be released from **the** job.

Where rectification (by welding, for instance) is feasible, this, together with the repair procedure, must be agreed on with the authorized inspectorate prior to the repair by use of the same concession route.

H. Welding

The system must ensure that the electrodes are purchased to the correct specification, stored under the correct conditions on delivery, given the correct drying treatment prior to use, and issued to the welder in a controlled manner.

Weld procedures should be drawn up and test samples welded and tested to ensure that they meet the specification and contract requirements prior to any welding on **the** contract.

The welder should be given written instructions on the welding consumables to use. Records should be maintained on details of the welds made and the weld consumables used.

The inspector should check that the controls laid down are being worked to and that the welder is correctly qualified for the weld concerned. The inspector should check periodically that the specified preheat and **other** welding parameters are being maintained during the period of welding.

I. Forming and heat treatment

Forming, and in particular hot forming and any subsequent heat treatment, including the **postweld** heat treatment of completed vessels, must be properly controlled to ensure that the material and weld metal properties are maintained in the finished product.

This entails **the** drawing up of heat treatment procedures for all significant operations where specific aspects such as heating and cooling rates, soaking times, and appropriate temperature ranges are specified. In the case of hot forming, temperature ranges during the operation are specified. These procedures should be approved by the inspectorate before the operations take place.

Temperature recording instrumentation producing chart recordings of temperature against time is required for the records and proof **that** the specified heat treatments have been adhered to.

J. Control of measuring and test equipment

The system should formally specify the requirement that all measuring and test equipment be checked to ensure that it is within the limits of calibration and that this calibration is checked at agreed intervals. The system should be such that all items for inspection and test are controlled satisfactorily.

The system should list comprehensively all the equipment involved, which will range from **fine** measuring items such as micrometers to thread gauges, pressure test gauges, NDE equipment such as ultrasonic and crack detection equipment, and welding **gauges**—the list sometimes appears endless.

Any instrument that is found to be out of calibration must be withdrawn and adjusted or repaired before being issued for use again, or scrapped if not capable of being brought up to standard.

K. Records

Quality assurance records must be available to demonstrate effective management of the quality system. These records should cover all activities and include results of all inspections and tests, signed-off inspection checklists, audits (internal and external), and reviews, as well as records of welding and NDE procedures and welder and NDE operator qualifications, etc.

The above records should be filed and stored in a controlled manner so as to be readily retrievable. Stored conditions should be arranged to safeguard **them** from damage due to fire, storm, etc.

These records **should** be stored for the minimum time specified in the specification, or in **the** particular contract if this time is longer.

L. Audits

The quality system should provide for a planned audit of both internal and external systems to verify compliance with all aspects. All audits should be formally documented, and any nonconformance detected during audits must be subjected to corrective action by the departments concerned. Follow-up audits should be made to ensure that the specified corrective actions have been successfully concluded.

Personnel carrying out quality audits should be suitably trained and qualified for this activity.

M. Authorized inspectorates

In order for vessels to be operated in certain countries the **ASME** authorization of fabricators requires the fabricator to employ, under contract, an authorized inspectorate whose duty is to carry out certain mandatory inspections on contracts, thus allowing the vessel to be formally stamped with the **ASME** stamp. This inspection organization is also responsible for monitoring the fabricator's quality system and is involved in the three-yearly reassessment of the **ASME** authorization.

Similar schemes involving authorized inspectorates are in operation in other countries, for example, by the PVQAB in the United Kingdom.

The authorized inspectorate agency must also be approved by the authorizing body before it can operate as an authorized inspectorate.

N. Nondestructive examination

Nondestructive testing has become a crucial aspect in the fabrication of pressure vessels and its application must be taken into account from the initial design stage in order to inspect weldment in **the** best possible manner. It also follows that the test equipment must be adequate, in good condition, with NDE personnel properly trained and qualified in their particular activity.

Qualification schemes such as ASNT TC, for personnel carrying out radiography, ultrasonic examination, and crack detection on vessels manufactured to the **ASME** code, or CSWIP (Certification Scheme for Weldment Inspection Personnel), managed by the Welding Institute in the United Kingdom, are used in approving inspection personnel for NDE activities.

The acceptance criteria for welds in vessels and the

basic techniques that must be followed are laid down in the appropriate fabrication codes and it is required that NDE personnel work to these standards without deviation. Inspection reporting must be concise, and these reports are signed by **the** person performing **the** test, whose qualifications are listed. A brief summary of the testing techniques available is as follows.

(a) Visual examination

Visual examination is always the first NDE examination an inspector should carry out. Visual inspection standards are now available and a visual inspection report is often required.

(b) Radiography

Radiography consists of transmitting X rays or gamma rays through the item to be examined and exposing a film that is positioned on the opposite side of the item. This method highlights areas of the weld or material under examination where the density varies from that of the base material, and hence discontinuities such as cracks, inclusions, voids, and the like are shown on the radiographic film as normally darker densities. The **film** is viewed on a viewing screen and appropriate interpretations made by **the** qualified radiographer.

Because of the use of X- or gamma rays the process is hazardous and requires strict control at all times. Normally **the** operation is carried out in a vault where the walls are sufficiently thick to reduce the harmful rays to acceptable levels.

Radiography has its limitations in **that** only defects oriented in line with the beam path are shown most effectively and the smaller the defect the less sensitive is the test in showing its presence. The test is also only possible when the joint **geometry** is favorable.

Figure 2 shows the X-ray examination of a circumferential weld seam.

(c) Ultrasonic testing-materials and welds

Ultrasonic techniques consist of transmitting sound waves of a particular frequency through the material to be tested by means of a vibrating transducer. These sound waves become reflected from discontinuities in the material and are gathered by the transducer and presented on a cathode-ray tube in the ultrasonic test equipment (see Fig. 3).

The time taken by the sound waves to travel through the material and return from the discontinuities is capable of being measured accurately, thus enabling the position of the discontinuity to be calculated. The strength of the response is also measured, and by manipulation of the transducer along the material **the**

length and size of the discontinuity can be determined. An experienced operator can also deduce the type of discontinuity and thus report cracks, inclusions, etc.

This method of test is complementary to radiography and has the advantage of being able to test **configurations** that are not possible by radiography. An added advantage of ultrasonics is that it is not hazardous like radiography and can be used in shop areas where other personnel are working.

Ultrasonic testing is ideal for ferritic welds, but there are limitations when testing austenitic welds due to very severe attenuation of the sound waves.

(d) Magnetic crack detection

As the title indicates, this testing method relies on magnetizing the material under test, and it is therefore applicable only to materials capable of being magnetized.

The test is essentially applied to testing surfaces of materials for discontinuities that disturb the uniform

magnetic lines of force. The system relies on coating the item under test with a suspension of fine magnetic material while it is magnetized. Discontinuities are highlighted where the lines of force are concentrated at that area, and the type and size of the defect are observed. The system is ideally suited to the detection of surface cracks.

(e) Dye penetrant testing

This system is based on the application of a colored liquid dye (usually red) to the material under test. After allowing time for the dye to soak into any discontinuity, the surplus is cleaned off. A developer is applied on to which dye in any discontinuity bleeds, thus outlining the shape of the defect.

This process has the advantage over magnetic crack detection of detecting faults in any type of material, whether magnetizable or not.

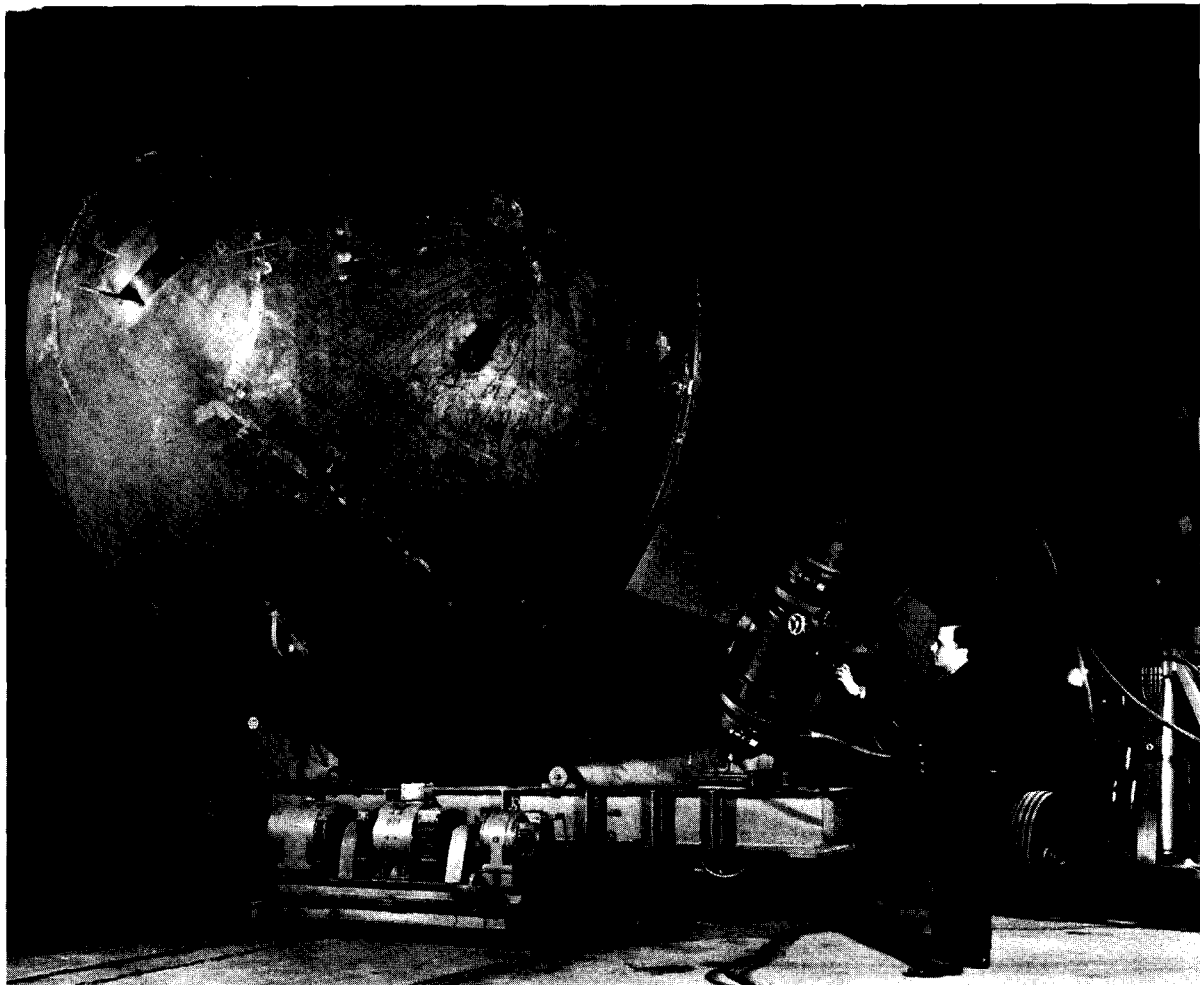


Figure 2 X-ray examination of a circumferential weld seam. (Courtesy, Whessoe Heavy Engineering Ltd.)



Figure 3 Ultrasonic testing. (Courtesy, Whessoe Heavy Engineering Ltd.)

(f) Pressure testing

The final nondestructive examination is normally the proving of pressure tightness. The pressure vessel is prepared for the test by adequately blanking off all branches and openings, filling with water (taking care to exclude any air), and pressure testing to the required test pressure. This pressure is held for a predetermined time and the vessel inspected for leaks.

To ensure that the test is carried out correctly, a checklist is usually made available, highlighting the activities and test criteria, which can be signed off progressively.

It is essential that pressure gauges used are calibrated prior to the test and that at least two are used in any test.

If a pressure vessel is to be pneumatically tested it should be hydraulically tested initially, due to the inherent danger of the pneumatic test. Any subsequent pneumatic tests should be carried out at a lower pressure.

If hydrostatic testing is not possible, then prior to a pneumatic test strict safety precautions must be taken, including the use of blast barriers and the exclusion of personnel from the area during the actual test.

4.8

Costing of heat exchangers

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4.8.4

Introduction

D. Chisholm

■ The objective of this chapter is to provide approximate methods for estimating the cost of heat exchangers. The greater part of the chapter is concerned with the costing of shell-and-tube heat exchangers. This emphasis on the shell-and-tube exchanger is appropriate for two reasons: First, the majority of heat exchangers, if one excludes the automobile radiator, are of this type; second, other heat exchangers tend to be proprietary, and thus it is only infrequently that the general practicing engineer is required to estimate their cost. Nevertheless, there are situations in which approximate

cost estimates for these proprietary units can be useful, particularly in comparative studies.

Thus, in addition to a section concerned with approximate costing of shell-and-tube heat exchangers, further sections deal with air coolers, plate heat exchangers, and compact heat exchangers.

The basic cost quoted in this chapter, unless otherwise stated, is the cost ex works, including insurance inspection at stages during manufacture, but excluding any additional inspection required by the customer, and excluding transportation.

4.8.2

Costing of shell-and-tube heat exchangers

R. C. Cowie

A. Introduction

One of the basic problems in arriving at the cost of a particular project is to arrive at a reasonably accurate price for the heat exchangers involved. Quite often the information available is sketchy and limited to the surface area and tube length in the case of a shell-and-tube heat exchanger. With these parameters it is possible to arrive at a reasonable first estimate of the cost, either on the basis of cost per ton (usually used for vessel and structural steel work) or on the basis of cost per unit of surface area. There are other parameters that can be used to arrive at a more accurate cost estimate, as discussed below.

B. Determination of shell size

If only the surface area is given, the next step is to arrive at the shell diameter that can be used. Table 1 lists shell diameter against the surface area per unit length of shell (m^2/m) for 19-mm ($\frac{3}{4}$ -in) OD tubes with a 25.4-mm (1-in) square pitch. Surface area per unit length is given for four of the basic TEMA configurations (see Sec. G). Above a 508-mm (20-in) shell OD there is a breakpoint between pipe shells and shell rolled from plate. This breakpoint arises from manufacturing considerations and varies from country to country. Table 1 also lists correction factors to the surface area for other pitch configurations and tube diameters.

C. Basic cost

Having determined the shell diameter as discussed above, one can obtain the cost of the heat exchanger by using Figs. 1 and 2, the latter figure allowing for the influence of heat exchanger length on the cost. The basic cost is the ex-works price, including manufacturer insurance-stage inspection. The basis of the figures is a carbon steel exchanger of AES shell type, TEMA R, using 19-mm ($\frac{3}{4}$ -in) OD and 14 BWG (mm wall) tubes of A214 steel on a square pitch. Corrections to obtain the cost of other configurations and materials are discussed below.

D. Design temperature and pressure

The design temperature influences the choice of material and hence the cost of the exchanger. With carbon steel materials the maximum design temperature to be used without change in stress values is 343°C , and the limit for nonferrous materials is from 180 to 200°C for the purpose of the curves.

The basic cost obtained from Figs. 1 and 2 corresponds to a design of $1 \text{ MN}/\text{m}^2$ for both shell side and tube side. Correction factors for the influence of design pressure can be obtained from Fig. 3; the correction factor is the sum of the factors obtained for the shell side and tube side.

E. Construction and baffle arrangement

Factors to allow for the influence of construction arrangement (U tube, etc.) can be obtained from Table 2,

Table 1 Cost of shell of exchanger

Area/meter run of exchanger, 19.05-mm OD tube, 25.4-mm square pitch, two-pass					Factors on area for other pitches and tube diameters, mm			
MM	AES	AET	BEM	BEU	23.8~ 19.95	25.4□ 19.05	31.750 25.4	31.756 25.4
OD								
203	1.67		2.03	1.43	1.3	1.0	0.75	1.14
254	3.1	—	3.58	2.63	1.2	1.0	0.82	1.13
305	4.54	1.67	5.26	4.06	1.3	1.0	0.84	1.09
356	5.6	3.1	6.7	5.38	1.28	1.0	0.85	1.14
406	7.89	4.54	8.84	7.65	1.3	1.0	0.85	1.14
457	10.64	5.6	11.7	10.5	1.3	1.0	0.84	1.13
508	1.5	9.56	10.5	14.6	1.28	1.0	0.84	1.13
ID								
560	18.2	12.4	20.2	18.2	1.3	1.0	0.83	1.13
610	22.2	16.6	24.0	22.0	1.3	1.0	0.83	1.13
635	24.26	18.2	25.8	24.0	1.3	1.0	0.84	1.13
688	28.0	22.22	30.42	28.2	1.3	1.0	0.845	1.14
737	33.1	26.0	35.6	33.1	1.3	1.0	0.83	1.16
787	38.23	28.0	40.34	38.25	1.3	1.0	0.83	1.16
838	43.5	33.1	45.9	43.27	1.3	1.0	0.84	1.16
890	49.5	38.24	51.84	50	1.3	1.0	0.84	1.16
940	55.34	43.51	58.45	56.2	1.3	1.0	0.84	1.16
991	62.16	49.5	65.5	62.63	1.3	1.0	0.83	1.16
1 067	72.67	62.16	76.8	73.0	1.3	1.0	0.84	1.16
1 143	84	69.22	88	84.86	1.3	1.0	0.85	1.16
1 219	96.2	80.32	101.0	97.0	1.3	1.0	0.85	1.16
1 295	107.6	92.0	113.8	110.7	1.3	1.0	0.85	1.16
1 372	120.72	98.8	127.8	124.56	1.3	1.0	0.85	1.16
1524	149.4	113.4	158.4	154.4	1.3	1.0	0.85	1.16

whereas Table 3 permits the influence of the number of baffles to be allowed for in an approximate manner.

F. Materials of construction

The basic cost obtained in Sec. C is for a carbon steel exchanger. Table 4 gives correction factors for other materials. This is used in conjunction with Table 5 which gives the percentage cost of the principal components of an exchanger. Table 5 is based on an AES shell type 6.1 m long of ah-steel construction, but can

be used as an approximate guide for other conditions. The use of these tables is illustrated in the numerical examples in Sec. H. By far the greatest effect on price is the cost of the tubes, and as more exotic materials are increasingly used their proportion of cost becomes greater. Table 6 shows the price differences of tubes for various commonly used tube materials. The channel itself has less effect but, where the tube plates are of material similar to the tubes, this can affect the overall price, and in arriving at the factors in Table 4 an attempt has been made to allow for this effect.

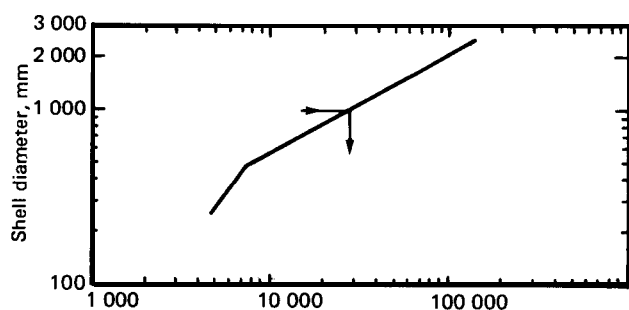


Figure 1 Cost of shell diameter per meter length in dollars. Type AES TEMA R all-steel A214 19.05-mm tubes, 25.4-mm square pitch, two-pass.

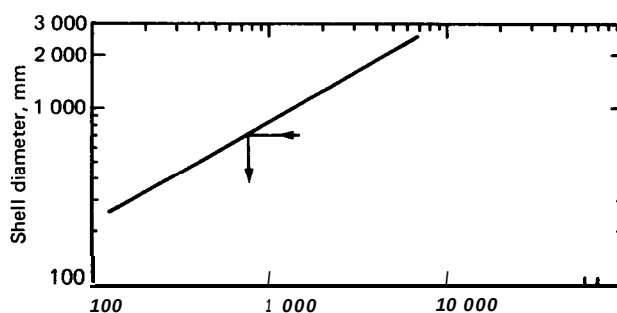


Figure 2 Extra cost per meter of exchanger, in dollars, for all-steel 19.05-mm tubes A214.

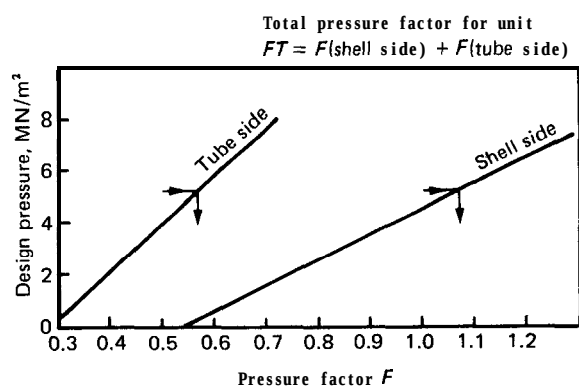


Figure 3 Design pressure factor.

G. Influence of shell type

Before proceeding to numerical examples, some general comments on the influence of shell type on cost are appropriate.

(a) Split-ring floating head (TEMA type AES)

This type of unit is the most expensive to build because of the complexity of its construction, but it does have the great advantage of allowing the bundle to be removed and replaced if necessary, and the floating tube plate takes care of any expansion problems. It has the greatest number of flanges (five), which all have to be sealed, and therefore the labor cost is high. Basically this type of construction has a high labor and material content, making it the most expensive to manufacture. Figure 1 is based on this construction.

Table 2 Factors for type of construction (all steel)

Type	Factor	Comment
Floating-head AES	1.0	
Pull-through floating-head AET	1.045	
Fixed tubesheet BEM	0.9	No bellows included
U-tube BEU	0.85	

Table 3 Factors for baffle pitch

Baffle ratio = $\frac{\text{baffle pitch}}{\text{shell diam.}}$ (consistent units)	Factor
0.25 or greater of shell diam.	1.00
0.20 of shell diam.	1.01
0.167 of shell diam.	1.02
0.143 of shell diam.	1.03
0.125 of shell diam.	1.04
0.111 of shell diam.	1.045
0.10 of shell diam.	1.05

Table 4 Cost factors for materials

Material	Shell ID		
	560 mm	762 mm	More than 762 mm
Carbon steel	1.0	1.0	1.0
18/8 shell and bundle	3.4	4.0	4.5
18/8 bundle CS shell	3.0	3.5	4.0
Aluminum bundle	1.2	1.3	1.35
Killed STL bundle	1.04	1.05	1.1
All killed STL	1.25	1.4	1.5
ADM tubes RNB T/plates	1.25	1.35	1.45
Copper tubes	1.2	1.3	1.4
70/30 copper-nickel tubes	1.4	1.55	1.7
90/10 copper-nickel tubes	1.4	1.5	1.65
304 T/S CS shell	2.2	2.2	2.4
321 T/S CS shell	2.3	2.3	2.5
316 T/S CS shell	2.4	2.5	2.6
304 tubes CS remainder	1.65	1.7	1.7
321 tubes CS remainder	1.7	1.75	1.75
316 tubes CS remainder	1.75	1.8	1.8
5 CR T/S CS shell	1.21	1.25	1.31
C- $\frac{1}{2}$ Mo T/S CS shell	1.2	1.25	1.3
5CR S/S CS shell	1.4	1.5	1.8
C- $\frac{1}{2}$ Mo S/S CS shell	1.4	1.5	1.8

^aBased on 6.096-m-long shell.

The floating-head design is more commonly used in the oil refinery industries than any other type of unit.

(b) Pull-through floating head (TEMA type AET)

This is similar in construction to the AES type except that the floating-head cover bolts directly onto the tube plate. The bundle may be withdrawn from the

Table 5 Percent cost of materials to total material cost of shell, channel, and tube bundle (support plates only—no baffles)

Shell diam., mm	Shell	Channel	Tube bundle
254	0.24	0.232	0.31
356	0.22	0.23	0.35
457	0.205	0.225	0.38
508	0.20	0.225	0.395
560	0.20	0.22	0.41
635	0.19	0.22	0.43
737	0.172	0.219	0.45
787	0.171	0.219	0.46
864	0.168	0.215	0.475
940	0.165	0.215	0.48
991	0.147	0.214	0.50
1067	0.145	0.212	0.51
1219	0.13	0.211	0.54

^aBased on AES, 6-m-long, all-steel shell.

Table 6 Tube prices, 19.05-mm OD

Tube	Price, \$/m	Tube gauge, BWG
A214 steel	1.12	14
A 179 steel	1.43	14
Aluminum brass	3.2	16
Admiralty	3.18	16
A213/304 S/steel	2.92	16
A213/321 S/steel	3.87	16
A2 13/ 3 16 S/steel	4.76	16
70/30 Cu-Ni	4.38	16
90/10 Cu-Ni	3.93	16

shell, without removing the shell cover and **floating-head** cover. This type of unit can reduce maintenance costs on site but has the disadvantage that it requires a larger shell for the same amount of surface as an AES type, but has fewer flanges. From a cost viewpoint the two are very similar on smaller diameters, but this type can be more expensive over a shell diameter of about 610 mm (24 in).

(c) Fixed tubesheet unit (TEMA type BEM)

This is cheaper to construct, as the floating head has been discarded. It is possible to design and construct the unit to take up certain amounts of thermal movements, but should these prove excessive a bellows expansion joint may be required. If so, the price of bellows should be added to the basic cost of the exchanger, as the factors discussed do not take this into account due to the many types of expansion joints that can be used today.

(d) U-tube TEMA type BEU

This is the cheapest heat exchanger and consists of a U-tube bundle that can be removed. It also has the advantage that thermal expansion can be taken up, but the amount of surface area per shell diameter is less than the FTS type because of the mechanical limitation of bending the tubes into a U. The bundle can be readily replaced and is suitable for use with high pressure on the tube side but has the disadvantage that the tubes cannot be cleaned other than by chemical cleaning.

H . Numerical examples

Following this brief description of what is involved in the costing of an exchanger are two worked examples with step-by-step explanations.

(a) Example 1

Find the cost of a **230-m²** exchanger type AES 4.87 m long, **1.0 MN/m²** design pressure on the shell side, **2.0 MN/m²** design pressure on the tube side, and 19.05 mm OD **tubes on 25.4-mm** square pitch.

1. Divide the surface area by the tube length:
 $230/4.87 = 47.22$.

2. Look down Table 1 and select the nearest larger shell diameter corresponding to this figure: AES 19.05 mm tube, **25.4-mm** square pitch = 890.

3. Use Fig. 1 to **find** the cost of the shell size selected: \$22 500.

4. From Fig. 2 find the cost per meter length of exchanger of the shell size selected, multiplied by the (tube length — 1): $\$1\ 150 \times (4.87 - 1) = \$4\ 450$.

5. Add steps 3 and 4 to give the total (basic) cost of the exchanger: \$26 950.

6. Use Fig. 3 to select the shell-side and tube-side design factors:

$$\text{Shell-side : } 1.0 \text{ MN/m}^2 = 0.65$$

$$\text{Tube-side : } 2.0 \text{ MN/m}^2 = 0.402$$

Therefore total design factor = 1.052

7. Use Table 2 to check if a construction factor should be used. As we are using an AES floating head, the factor is 1.0.

8. Use Table 3 to obtain the factor for the baffle pitch. Quite often the baffle pitch is not known, in which case the factor 1.0 should be used (this includes tube supports as per code).

9. Use Table 4 to ascertain the cost factor for materials. In this case, as an all-carbon steel unit is used, the factor is 1.0.

10. The basic cost (step 5) should now be multiplied by the factors obtained in steps 6, 7, 8, and 9; **that is, $\$26\ 950 \times 1.052 \times 1.0 \times 1.0 \times 1.0 = \$28\ 351$** X market factor (1.1). This represents the cost of an **all-steel** exchanger at today's prices (i.e., a market factor of 1.1); all that is required is to check price increases periodically and adjust the market factor accordingly, as the factors involved tend to remain constant relative to each other.

(b) Example 2

Find the cost of a **230-m²** type BEM exchanger 6.096 m long, **1.0 MN/m²** design pressure tube side and shell side, **19.95-mm OD** copper tubes on **23.8-mm ($\frac{15}{16}$ -in)** triangular pitch, baffle pitch 157 mm.

1. $230/6.096 = 37.72$ divided by factor 1.3 for triangular pitch = $37.72/1.3 = 29$.

2. Select the nearest shell diameter under type BEM: 686.
 3. \$14 200 from Fig. 1.
 4. \$740 (6.096 - 1) = \$3 771 from Fig. 2.
 5. Add steps 3 and 4: \$14 200 + \$3 771 = \$17 971.
 6. As the shell-side and tube-side design pressures are 1.0 MN/m², the factor from Fig. 3 is 0.65 + 0.35 = 1.0.

7. The construction factor is 0.9 for a fixed tube-sheet from Table 2.

8. Baffle factor (Table 3) = baffle pitch/shell diam. = 157/686 = 0.22. Hence factor = 1.01.

9. From Table 4, copper tubes gives a factor of 1.3, from the 560-762 mm column.

10. \$17971 x 1.0 x 0.9 x 1.01 x 1.3 = \$21 236 X market factor (1.1).

4.8.3

Costing of air coolers

C. North

■ An air cooler is a combination of pressure vessel, structure, mechanical equipment, and electric drive. As such, cost levels vary with such diverse parameters as the following:

1. Fin tube type
2. Tube length
3. Header type
4. Structure finish
5. Materials of construction
6. Noise limits
7. Drive type

Approximate air cooler costs are most often quoted on the basis of dollars per square meter of outside bare tube surface. In order to detail variations in cost at 1979 levels, let us consider a typical unit:

Size	2 500 m ² bare surface
Core tube	Steel-14 BWG, 25.4-mm OD
Tube length	9 m
Fin pitch	2.3 mm
Fin material	Aluminum
Fans	4-bladed 60 m/s tip speed

Drive	22 KW max; V belt
Draft	Forced
Header	Plug
Pressure	700 kPa
Structure finish	Red lead
Walkways	Header and motor access
Ladders	To grade
Tube rows	6
Fin height	16mm

The cost per unit area of outside bare surface (with no fins) for this unit is given in Table 1. The premium for seamless tube is higher in North America than in Europe; Table 1 is based on the European price. Table 1 is also based on a 25.4-mm (1-in) core tube; for 38.1-mm (1.5-in) core tubes use the rates in Table 1, but treat as one row more in Table 2.

As any number of tube rows can be fitted onto any particular structure/fan arrangement, the cost per unit area of surface varies with the number of tube rows. This can be represented as a multiplier on the cost for a six-row unit as given in Table 2.

Similarly, the cost per unit area varies with tube

Table 1 Cost (U.S. dollars) per square meter bare surface

Core tube	16 BWG	14 BWG	12 BWG	10 BWG
Welded CS tube ASTM A214	135	137	141	158
Seamless CS tube ASTM A 179	147	148	154	161
Welded SS tube TP304	231	241	—	—
Seamless SS tube TP304	241	254	—	—

^aEx-works to ASME code standards but excluding independent inspection.

Table 2 Multiplier for tube rows

	Multiplier	
	Carbon steel	Stainless steel
3 rows tube	1.54	1.34
4 rows tube	1.27	1.18
5 rows tube	1.11	1.08
6 rows tube	1.00	1.00
7 rows tube	0.92	0.92
8 rows tube	0.86	0.90

length as indicated in Table 3; the longer the tube length, the cheaper the air cooler per unit of surface area.

Costs also vary with fin pitch, decreasing with increasing fin pitch as shown in Table 4.

The influence of the type of extended surface can be allowed for using the multipliers in Table 5.

Table 6 provides cost-correction multipliers for whether the cooler configuration is forced draft, induced-draft top drive, or induced-draft bottom drive. Cost variation with draft type does not take into account higher power **consumptions** associated with induced draft. It should be noted that top drive includes handrails around the top deck but no motor access walkway.

The influence of design pressure and header type is obtained from Table 7. The removable-cover and bonnet-header types are limited to low-pressure applications. No account has been taken here of the bundle slope or of split headers.

Noise limits require variations in tip speed of fans,

Table 3 Multiplier for tube length

Length, m	Multiplier
6	1.18
9	1.00
12	0.89
15	0.85

Table 4 Multiplier for fin pitch

Pitch, mm	Multiplier
2.3	1.000
2.5	0.987
2.8	0.979
3.1	0.955
3.6	0.944

Table 5 Multiplier for extended surface type

	Multiplier	
	Carbon steel	Stainless steel
Tension-wound L fin	1.00	1.00
Embedded G fin	1.04	1.06
Extruded aluminum	1.19	1.11
Galvanized steel tin	1.31	1.22

an affect on cost that can be allowed for using the multipliers in Table 8. Noise varies considerably with the fan manufacturer and the specific installation, but the levels in Table 8 are a general indication of the maximum levels that can be expected 1 m from the fan.

Finally, cost multipliers for the influence of the size of the unit and the **finish** of the supporting structure and plenum can be obtained from Tables 9 and 10, respectively.

These 10 tables can be used to estimate the cost of a particular configuration as follows.

Configuration	Multiplier
1. Tube, 12 BWG, A214, 25.4-mm OD	\$141
2. Rows, 4	1.27
3. Length, 12 m	0.89
4. Fin pitch, 2.5 mm	0.987
5. Surface, extruded aluminum c.s.	1.19
6. Cooler configuration, forced draft	1.00
7. Pressure, 4 000 kPa , plug header	1.04
8. Fans, 60 m/s	1.00
9. Surface, 3 000 m²	1.00
10. Structure finish, type a	1.00

$$\text{Cost} = 141 \times 1.27 \times 0.89 \times 0.987 \times 1.19 \times 1.04 \times 3\,000 \\ = \$584\,025$$

The relevance of the above costing system will vary with market conditions. However, the effect of such parameters as maximum tube length is not likely to vary, and the guide gives a good indication of the likely savings of increased maximum tube lengths within the practical limits of transportation.

Table 6 Multiplier for cooler configuration

Configuration	Multiplier
Forced draft	1.00
Induced-draft top drive	0.98
Induced-draft bottom drive	1.03

Table 7 Multiplier for header type and design pressure

Pressure and header		Multiplier						
Pressure, kPa	750	1500	2 000	3 000	4 000	6 000	8 000	10 000
Plug	1.00	1.00	1.01	1.02	1.04	1.05	1.08	1.11
Removable cover	1.04	1.06	1.08	1.10				
Removable bonnet	1.05	1.07						
Welded bonnet	0.98	0.99	1.00	1.02	1.03	1.04	1.05	

Table 8 Multiplier for fan type

Fan type	Noise at 1 m, dBA	Multiplier
60 m/s 4 blades std.	85	1.00
45 m/s 6 blades std.	81	1.021
35 m/s 6-8 blades low noise	77	1.100
30 m/s 8-12 blades low noise	75	1.139

Table 9 Multiplier for small units

Unit size, m ²	Multiplier
2 500+	1.00
1 200	1.02
400	1.10
200	1.30
100	1.75

Table 10 Multiplier for structure finish

Finish	Multiplier
(a) Wire brush + primer (galvanized walkway floor)	1.00
(b) Hot dip galvanized	1.04
(c) Shot blasted + 2 coats red lead	1.03

4.8.4

Costing of plate heat exchangers

J. Dennis Usher

A. Introduction

Plate heat exchangers are different from many other items of chemical plant, where costs can be approximately related to size or capacity by a logarithmic law. Published data [1], for instance, indicate that for a fixed head shell-and-tube heat exchanger the variation in cost is of the order of 0.8 power of the surface area. A plate heat exchanger cost, however, is more of a linear function and can be divided into four main categories:

1. Basic frame
2. Liquid connections
3. Plates
4. Gaskets

Of the four, the basic frame cost is affected mainly by working pressure, as this will influence the thickness of the main structural members and the tie-bar diameter. There will also be a slight increase in cost with platage capacity, owing to the increased length of the top and bottom bars and tie bars, although this is a relatively minor factor. Apart from this, the basic frame cost is independent of any factors depending on process conditions.

Plate costs are also influenced by working pressure, which can affect the thickness; unlike the basic frame, however, the liquid connections and the plates must be compatible with the process liquid, and they are therefore both affected by material cost, although the cost of the liquid connections can be minimized by the adoption of rubber liners. The plates are characterized by the fact that they are pressings, so that unlike some types of welded heat exchangers, the labor cost is only mar-

ginally affected by the metal used, and is therefore virtually constant. Thus manufacture in the more expensive alloys will only increase the material content of cost. Gasket costs also depend on the process requirements, as they may be molded in a variety of elastomers, from medium nitrile to the more expensive fluorelastomers, or may even be made of compressed asbestos fiber. Hence the cost of a plate heat exchanger can be expressed by the following linear function:

$$\begin{aligned}\text{Cost} &= F + Nb + n(p + g) \\ &= F + Nb + \gamma A\end{aligned}$$

where F = basic frame cost

b = cost of liquid connection

p = cost of plate

g = cost of gasket

N = number of liquid connections

n = number of plates

A = total heat transfer surface

γ = cost per unit area of heat transfer surface

This cost may be affected by special connector plates, which are sometimes included in the platage to provide service connections along the plate pack in multisection machines such as sterilizers and pasteurizers.

B. Comparison with shell-and-tube prices

To try to quantify the above relationship is an unproductive exercise, because of the variations in both material and labor costs throughout the world. Apart from this, and because it is selling price rather than cost price that is of interest to the potential customer,

these prices are affected by the policy of the individual plate heat exchanger (PHE) manufacturers. Hence any statement on representative selling costs must be purely general, and most investigations carried out have included shell-and-tube prices on a comparative basis.

One such study [2], made in 1979, compared the cost per unit area of PHEs with stainless steel plates with shell-and-tube units of the following construction:

All stainless steel shell and tube
Stainless steel tubes and carbon steel shells
Carbon steel tubes and shell

The results are presented in Fig. 1, which shows that on an area basis stainless steel PHEs are cheaper than tubular units except those of carbon steel construction. However, the fact that PHEs have considerably higher heat transfer coefficients means that, considered on a basis of similar thermal duties, the PHE is frequently cheaper than all three types of tubular exchanger. It goes without saying that for the more expensive materials such as titanium, where tubular fabrication costs are higher, the cost advantage is even greater.

Another exercise [3], carried out in 1975, compared the capital costs of the two basic types of unit for 54 duties covering the following conditions:

NTU value	0.5-3.0
Process flow rate	9-900 m ³ /h
Flow rate ratio	0.5-5.0

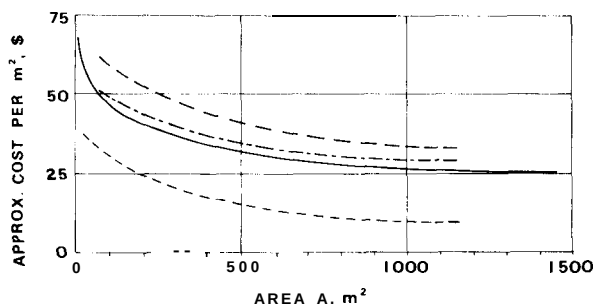


Figure 1 Relative costs of PHEs and tubular heat exchangers. (—) S and T: all stainless (tubes and shell). (---) S and T: stainless tubes, mild steel shell. (- - -) S and T: mild steel tubes and shell. (——) PHE: all stainless contact.

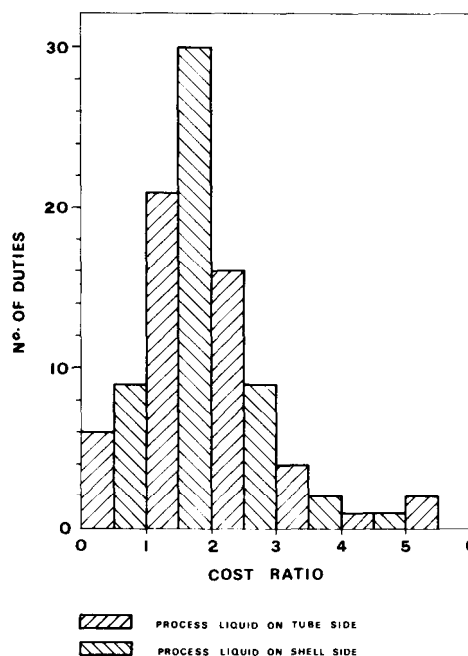


Figure 2 Analysis of tubular heat exchanger costs.

Viscosity of process liquid	0.000 2-1 kg/m s
Process-side pressure drop	14-140 kN/m ²

The PHE costs were based on stainless plates and rubber gaskets and were obtained from a U.K. manufacturer, whereas the tubular units were all carbon steel construction and the costs were obtained from Heat Transfer and Fluid Flow Service (United Kingdom).

The results are shown in the histogram given in Fig. 2, which defines the number of cases lying in different ranges of cost ratio, this ratio being

$\frac{\text{Shell-and-tube cost}}{\text{PHE cost}}$

The study also states that a PHE with titanium plates would be about twice and one with Hastelloy plates about eight times the cost of one with stainless steel plates, but of course, as already stated, the cost differential for a tubular unit would be even more marked.

REFERENCES FOR SECTION 4.8.4

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4.9

Specifications and tenders

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4.9.1

Introduction

David Edge

■ What is a specification and why does an engineer need to write one before purchasing heat transfer equipment? A specification details the technical requirements for the equipment and, together with data sheets and drawings, forms an inquiry package to be sent to suitable vendors. Most equipment is covered by code and standard requirements; so, what should the specification contain over and above these requirements?

Engineers are increasingly concerned with not only more detailed client requirements and quality control but also package plant and skid-mounted units. Documentation and certification considerations now take up a large percentage of engineering time, especially for hazardous plant.

This section gives a brief introduction to some of the problems a contractor's engineer dealing with heat transfer equipment may face and suggests how the process of writing job specifications and assessing vendors

quotations should be handled. Typical checklists, by no means all-inclusive, are included.

Presentation of vendor bid analyses will differ from company to company. Obviously, it is useful to tabulate information on standard forms in order to compare bids. This not only assists in checking at the purchase order stage that the scope of the job is fully covered but also provides a record of exactly what was bought. Two typical bid analyses are given to illustrate this.

The terms contractor, client, and vendor are used throughout the section. A *contractor* will normally act on behalf of the client to design, engineer, and construct a plant. The *client* will normally be an end producer of (say) petrochemicals or power from that plant. The *vendor* will normally design and fabricate particular equipment, such as heat exchange equipment, for the plant. The term engineer is used in the text to denote the person responsible for the contractor's engineering.

4.9.2

Specifications

David Edge

■ What problems would occur if an inquiry were issued without a job specification and simply called up the applicable design code? After the purchase order was placed, the contractor would have to negotiate, typically, the price and delivery of spares, number of copies of drawings, guarantees, extent of supply, special tools and test equipment, and so on.

The code covers the detailed mechanical design methods to be used and refers to a large number of allowable materials of construction with their associated design stresses. It also details fabrication and inspection methods. The code requirements are legally mandatory in many countries.

A specification will add the contractor's and client's technical requirements and clarify the code (or standard) when choices are available. It should contain a design and fabrication section based on experience in fabrication and problems in the field.

The specification, when accepted by the vendor, will be contractually binding. It will not deal with commercial considerations. Table 1 gives a list of items that should be included in most specifications. Table 2 lists typical requirements for shell-and-tube heat exchangers. Table 3 lists typical requirements for air-cooled heat exchangers. Table 4 lists typical requirements for plate and frame type heat exchangers. Table 5 considers a package unit (a cold box), which has much broader scope and requires liaison with other engineering disciplines. Table 6 lists typical requirements for electrical process heaters.

Most contractors (and clients) have standard specifications, and these should be used as a basis for the job specification. Irrelevant detail should be omitted and clients' requirements added, taking care not to duplicate or contradict. Take time over the job specification—it saves many hours if the job is right the first time.

Vendors are often faced with a mass of documentation. Try to keep the job specification to a minimum, avoid duplicating the code, and weed out unnecessary ancillary documents.

A. Shell-and-tube heat exchanger

Table 2 lists some typical requirements that could be included in a shell-and-tube specification. Individual clients and contractors will have specific requirements based on their particular standards and experiments. However, it is a good idea to build up a checklist as shown in Table 2 so that when a job specification is written it will take only a few minutes to check through for omissions.

B. Air-cooled heat exchanger

An air-cooled heat exchanger specification can be written on the basis of options in American Petroleum Institute Standard 661 (API 661). The optional requirement can be referenced as follows at the front of the specification:

Table 1 Typical checklist common to most specifications

Topic	Comment
Scope of the specification	Summary of the specification content (design, materials, fabrication, inspection, testing, painting, preparation for shipment, and documentation). Define the type of equipment.
Codes, standards, etc.	Comprehensive list of all relevant codes and standards with <i>dates</i> .
Internal specifications	
Vendor responsibilities	For thermal design, mechanical design, fabrication, testing, painting, shipment as applicable.
Guarantees	
Extent of supply	Important on package and skid-mounted units. Use diagrams to show what is supplied by vendor and what is supplied by others.
Instruments	
Instrument connections	
Electrical and instrument cables	
Cooling water supply lines on package units, etc.	
Documentation	Specify which documents are required at tender (quotation) stage and which are needed after the order.
Design data	
Drawings	Vendor should supply a schedule of all drawings for the order with a full description of each drawing and latest issue date. This is very important on package units and should be maintained throughout the job.
Data report	
Certified material record report	
Weld procedures	
Certified (insurance) inspection	
Certificates	
Design	See relevant equipment type.
Selection of materials	Do not put onus on the supplier. Generally the supplier will put in the minimum-cost suitable material. Client/contractor should select materials based on experience and required life of the plant.
Generic materials by contractor/client	
Allowable substitutes in various countries	
Clad or solid	
Fabrication	Do not duplicate the code.
Low-temperature requirements	Clarify when alternatives are offered.
Hydrogen service requirements	
Clad plate	
Postweld heat treatment	
Inspection	Extent of nondestructive testing (NDT) should be defined at the tender (quotation) stage. After the order, vendor should supply an NDT map showing locations of all inspection on a general arrangement drawing.
Radiography	
Ultrasonic testing of plates	
Magnetic particle	
Dye penetrant inspection	
Impact testing	
Testing	
Pressure tests, hydraulic/pneumatic	
Special leak tests	
Proof test	Brittle lacquer/strain gauge.
Burst to failure tests	Generally for items not in the scope. of the code.
Quality control	At the tender stage the vendor should submit a quality assurance manual and an inspection/test plan that includes a schedule of manufacture. After receipt of order the vendor should prepare a detailed quality plan.

Specify if after heat treatment; vendor may wish to do this before heat treatment only.

API 661 paragraph no.	Subject	Reference section of standard specification
2.2.8	Noise data sheet required	7.7.1
5.1.6.7	Cover plate bolting type	7.2.2
5.2.7.1	Driver supplied	7.1.9

These examples apply to the second edition (1978) of API 661, and the specification must be rewritten every time there is a new edition or major amendment.

The engineer must make decisions on the use of forced or induced draft, the type of fin allowed, the type of header (plug, cover plate, manifold), and the control requirement (louvers, autovariables) and must issue specifications to cover electric motors, ladders, and handrailing.

Table 2 Shell-and-tube heat exchanger checklist

Topic	comment
1. Tubes	
Seamless or welded (and whether cold worked after welding)	Engineering/metallurgical judgment based on severity of service and corrosion characteristics.
Whether stress relieved	When stress corrosion cracking is a problem.
U bends (ANSI B31.3, paragraph 329)	U bends of small radius, hydrogen service, low-temperature service, etc.
Hydrotest and/or electrical test	When stress corrosion a problem do both (E426). Hydrogen service do both (E570). Hydrotest or both on welded tube.
ASTM A450	Generally for Incoloy, Inconel, Hastelloy severe service application.
Intergranular corrosion test, ASTM G28	Test to be carried out on material in final heat-treated condition.
Nitric acid duty, ASTM A262 Practice C	Test to be carried out on material in final heat-treated condition.
Oxidizing media, ASTM A262 Practice E	
2. Tube bundles	
a. Removable	
Bundle runners	Avoids scoring the shell on removing bundle from U-tube units.
Bundle pulling	Clad plate or low-temperature service should have special nonthreaded arrangement. Check on how bundles are removed on site. Bundle puller? Site access?
Tapped holes in tube plate	Avoids banana if unit is inverted during handling/shipping.
Kettle and pull-through units should have a bundle -retaining device	
Standard assembly for floating head	Ensure that all floating heads disassemble in the same way to avoid difficulty for site maintenance.
F shells	Does client allow longitudinal baffle? Leakage across the baffle.
Internal bellows	For one-tube-side-pass units.
b. Fixed tubesheet	
Allow F-type shell (two-pass shell side)	No leakage because longitudinal baffle welded to shell.
Tubes welded to tubesheet	Generally preferred but especially so for fixed tubesheets-saves time and money on site, fewer leakers than rolled-in joints.
Metal temperatures for fixed tubesheet design check on mechanical data sheet/setting plan supplied by contractor	Stresses at normal operation, start-up, shutdown, hydrostatic test, steaming-out, etc., to be checked in shell, tubes, and tubesheets.
Is a bellows required?	Is a bellows acceptable for the service?
Material?	Stress corrosion cracking of stainless steel?
Cleaning	Chemical cleaning nozzles. Check client requirement.
Inspection holes	Check site requirements.
3. Gaskets	
Supply material standard	Use spiral wound for stacked units. Solid metal or spiral wound for hydrogen service.
	Check seating requirements for spiral-wound gaskets. Calculate stress on bolts (< 90% yield) and in flanges on initial seating.
	Care in selection for floating-head and through-bolted shell/channel connections.
4. Tubesheets	Use forgings unless otherwise agreed.
5. Bolts	For large-diameter bolts use hydraulic pretension equipment.
	Check requirement for internal and external bolts in sour service (NACE MR-01-75).
6. Lifting devices	
Vertical units	Attention to means of lifting minimizes accidents on site.
Horizontal units	
Removable bundles	Lifting lugs on floating-head covers and channel.
Floating-head covers/channel covers	Vertical units must have lugs to lift the whole unit into the vertical position.
7. Vents and drains	
Avoid screwed connections on low-temperature service	Take care with venting condensers, especially at low pressure.
	Vertical units vent and drain through tube plate?
	Continuous vent or piped-up vent or blank-flanged?
8. Hydrogen service	
Seamless tubes	
No screwed connections	
Weld neck flanges	
No reinforcing rings	
Gaskets solid or spiral wound	
Postweld heat treatment	For carbon steel and low alloy.
Materials to API G41	
Ultrasonic testing	To ASTM A388.

Table 2 Shell-and-tube heat exchanger checklist (Continued)

Topic	Comment
9. Postweld heat treatment Fabricated floating-head covers Fabricated channels and bonnets	Specify postweld heat treatment when in severe service and/or multiple pass (with much welding).
10. Maximum weight and dimensions Bundles Floating-head units Fixed tubesheet	20 tons 1500 mm diameter $\times$ 7300 mm tube length 2000 mm diameter $\times$ 9000 mm tube length The values above are possible ones. Consult with client and site.
11. Spare parts	Tubes, gaskets, bolts.

The choice of forced or induced draft may depend on the client's or site engineer's preference. Although induced draft may be attractive because of better air distribution, less air recirculation, and protection of the top of the bundle during construction and maintenance, the client or site engineer may insist on forced draft because of lower power requirements and easier accessibility for maintenance.

The aspect ratio (ratio of bundle to fan area), the distance between fan and bundle, and the relationship between air velocity and static pressure drop across the bundle are all important in ensuring a good air distribution across the bundle.

Fan performance curves are obtained with the fan performing in ideal test situations. The fan will obviously perform differently when placed under a rectangular bundle. The specification must contain guidelines so that all vendors quote on a similar basis.

The margin on motor power must take into account the transmission losses between motor and fan and also the higher winter power if the fan is running at design setting (a fan is a constant-volume machine).

The specification should lay down guidelines for the use of the different types of finned tubes. Differential expansion between carbon steel and aluminum will cause poor contact between fin and tube at high temper-

Table 3 Air-cooled heat exchanger checklist

Topic	Comment
1. Fans Forced draft	Lower power. Motor and drive accessible. Easier maintenance.
Induced draft	Better air distribution over the bundle. Chimney effect lessens air recirculation. Protection for the fin s on the top bundle row. Long drive shaft if motor is located below bundle. Better natural convection performance.
2. Aspect ratio (AR) $AR \leq 2.5$ and $AP_{static}/\Delta P_{dynamic} \geq 3.0$	A maximum aspect ratio (bundle area to fan area) of 2.5 to 1 is often stated. AP dynamic is the maximum dynamic pressure in the stream approaching a resistance and is based on the velocity in the fan area.
3. Margin on motor power	Power is usually stated at the summer design temperature. Allow for transmission losses and the density of winter air. Request fan performance curve.
4. Limitation of fan speed and noise specification Estimated and guaranteed sound pressure and sound power levels	Sound power levels calculated in accordance with Oil Companies Materials Association (OCMA NEWG-I-Noise Procedure Specification).
5. Type of finned tube L foot G (grooved and peened) Extruded Galvanized carbon steel	Check suitability of fin for environmental condition. Check suitability of fin for metal temperature.
6. Minimum tube wall temperature	Tube wall temperature should be at least 8 °C above the process fluid pour point when operating at the minimum (winter) temperature.

Table 3 Air-cooled heat exchanger checklist (Continued)

Topic	comment
7. Tube-to-tubesheet joint-rolled or welded	For high-pressure operation (say > 70 bars) or for toxic or dangerous service, weld tubes to tubesheet.
8. Type of header	
Removable cover plate	Preferred for fouling fluids. Limited by pressure to about 30 bars.
Plug type	Most commonly used.
Manifold	High-pressure, clean fluids.
9. Control	
Louvers (manual or automatic)	Also acts as protection for the tubes on top of the bundle.
Autovariable fan	Saves power. Should not be shared by two different units.
Note: For both of the above, the fail position should be stated (i.e., autolouvers fail to open or shut or as is).	
10. Spare parts	Typical requirements are:
Tube access plugs	10%
Gaskets	300%
Fan and driver spares	Consult client/site.
V-belt drives	100%
Bolting	10%
	Obtain price list of spares at the quotation stage.
11. Layout/structural information sent to vendor	
If pipe rack mounted:	
Dimensioned sketch showing elevations, width of pipe rack and dimensions between stanchions	
If grade mounted:	
Dimensioned sketch showing proposed layout and plot area available	
Instructions on whether units can be grouped together and combined over common fans	Do not allow autovariable fans to serve more than one unit.
12. Structure information required from vendor	
Location and elevation of service walkways, access ladders, gates, and stairs	Supply standards and standard drawings. Check client/site requirement for ladders.
Location and underside clearance of fans and fan drivers	
Clearance around equipment required for maintenance, dismantling, and repairs	
Loadings at each stanchion	Send out drawing showing pipe rack dimensions to vendor. Coordinate with piping and structural.
Details of base plates	
13. Payout period	
Cost of power (\$/kWh)	Vendor should balance saving in capital cost of the exchanger against high power. Typical power cost is 0.4 \$/kWh over (say) 2 years.

ature, so a metal temperature limit should be stated for L fins and for extruded tins. Also, environmental conditions such as a salt-laden atmosphere or the presence of other chemicals may dictate the protection of the inner tube (and fin) and the material of the fin.

The vendor should be supplied with the power cost and the required payoff period so that power used can be optimized against the capital cost of the air-cooled equipment.

Table 3 summarizes typical factors that the engineer must consider.

C. Plate heat exchanger (plate and frame type)

The plate heat exchanger (PHE) is highly efficient and is particularly attractive when expensive material is needed. The engineer's first job is to assess the suitability of the PHE for the service, considering the following questions:

1. Can the PHE accept the design pressure and design temperature?

2. Are suitable gaskets available for the fluids used?

3. Is the PHE safe for the fluids used?

4. Has it been used successfully in this type of service?

The specification must contain the detailed information given in Table 4.

D. Packaged units and skid-mounted units

A skid-mounted unit is a self-contained unit mounted on a common base plate. Instrumentation and electrical items are provided. All necessary valves, controls, and other equipment are completely piped; all inlet and outlet process connections and utility, vent, and drain connections are flanged and extended to terminate at the skid edge.

The heat transfer engineer handling a skid-mounted or packaged unit must ensure that specifications are issued to cover all aspects of the unit. It is important that a consistent policy be applied to all items on the plant, and the assistance of the contractor's specialist engineers (electrical, instrument, structural, piping, pipe stress) is essential. Do not assume that a specialist vendor has expertise in all these disciplines.

Any heat exchange item can be mounted in a skid, but electric heaters and cold boxes are commonly skid-mounted as shown in Figs. 1 and 2. These are considered in the next two sections.

Vacuum steam condensers of both the steam surface type (shell and tube) and air-cooled type, complete with associated venting equipment, pumps, piping, valves, and drums, are usually bought as a package.

E. Cold box (brazed aluminum plate fin heat exchangers, vessels, and pipework)

Although the shell-and-tube exchanger is the most widely used heat exchange unit, the engineer should always consider other types of units that may be more economical. Their suitability for installation, trouble-free operation, and safety must be assessed for the particular process conditions.

For low-temperature fluids that are clean, the high-efficiency brazed aluminum plate fin heat exchanger may offer a number of advantages—not only in economy but also in weight and space-saving. However, these units are not as robust as shell-and-tube or air-cooled exchangers, and special consideration must be given to their design and installation.

Table 4 Plate heat exchanger checklist

Topic	Comment
1. Codes, standards, documents	For nonsevere services such as seawater/water exchangers, it may be considered unnecessary to call up design codes. Money may be saved by using the vendors' standard equipment and test pressures of 1.25 times design pressure.
2. Materials Stainless steel, titanium, Incoloy, Hastelloy, nickel, aluminum brass, etc.	Plate heat exchangers (PHEs) are most economical when expensive materials are needed.
3. Mounting arrangement Floor or pedestal	All nozzles should preferably be at one end so that the plates can be cleaned and/or withdrawn without disturbing the piping.
4. Fouling factors	This is particularly important, and only (say) 20% overcapacity may be needed for an efficient design with optimum velocity.
5. Allowable forces and moments on nozzles Supplied by the vendor for standard units	These will be low compared with typical values on shell-and-tube nozzles.
6. Gasket material	Gasket material should be tested and proved. Take care with hydrocarbons with aromatics content. Gaskets arranged to give a double seal between the two streams.
7. Special tools Power-operated spanner	Large units require this for complete closure of the plate pack.
8. Upstream filters In the line upstream of the PHE or in the inlet part of the PHE and supplied by the vendor.	PHE vendor should specify the maximum particle size.
9. Protective screens To protect personnel from effects of a ruptured gasket.	Should be provided for all units with high pressure and/or hazardous fluids.

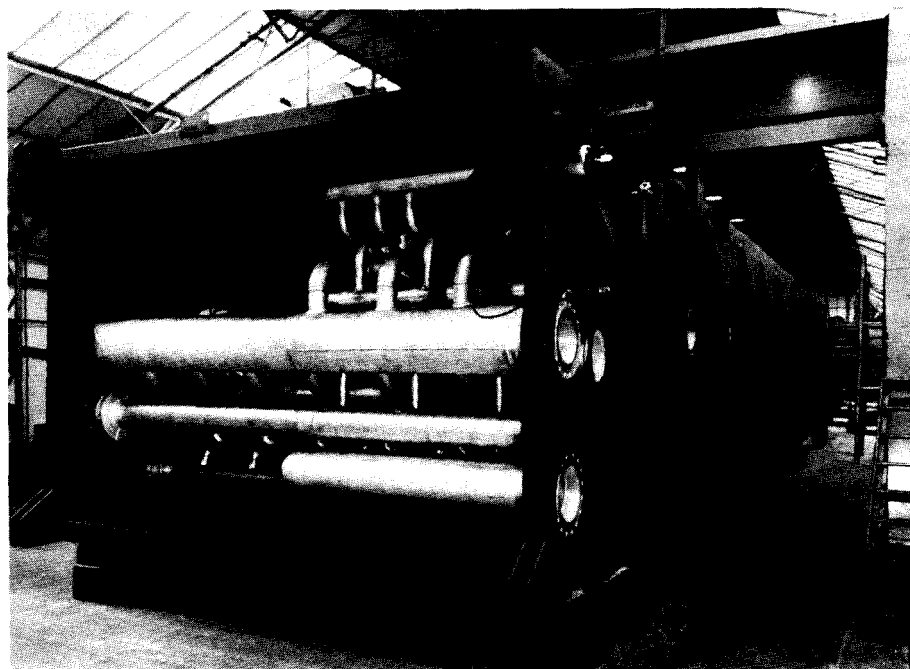


Figure 1 Cold box containing plate fin heat exchangers for use in an ethylene plant. (Photo courtesy of Marston Palmer Ltd.)

If the plate fin unit is shipped to site in a transport frame, it will have to be lifted from the frame and then into place with little or no protection. It must then be insulated and fireproofed, and at all times it will be vulnerable to mechanical damage from falling objects, crane jibs, etc.

If a number of plate fin cores are supplied, most of the above problems can be solved by installing the units in a *cold box* complete with interconnecting piping, manifolds, vessels, and instrument lines (Fig. 1). The units are supported by structural steel within the box, and normally there are two *heat breaks* between the core and the outside structural members to prevent conduction of heat. The cold box is filled with granular expanded mineral material that prevents convection within the box. A nitrogen purge prevents the ingress of moisture. The aluminum inlet and exit pipes to and from the plate fin heat exchangers finish normally with aluminum flanges outside the box. It is possible to anchor most of the smaller piping on the cold box wall, thus solving the problem of transmitting forces and moments through the piping onto the vulnerable aluminum headers. Installation procedures must be written and strictly adhered to during the construction phase.

During operation and especially at start-up it is essential not to block up the plate fin core. Filters should be installed which remove all particles that would not pass easily through the fins. However, the unit may be blocked up by a mixture of fine particles and oil. This mixture may be powder from driers mixed with seal oil

from compressors or, in the case of offshore applications, sand mixed with black oil.

It must be ensured that the unit is not operated, dried, or cleaned at temperatures above **that** recommended by the plate fin fabricator.

The job specification must therefore carefully detail the extent of supply and engineering responsibility of the vendor; it must refer to other specifications, data sheets, and drawings that detail all **the** ancillary equipment. Table 5 is a checklist summarizing some of the considerations **the** engineer will have to deal with for this type of equipment.

F. Electrical process heaters

The engineer will find frequent use for resistance heaters and their associated power controllers on offshore and other installations where no high-pressure steam is available, or when high-temperature heating is required with close temperature control. Typical applications are for (1) start-up heaters, (2) heat transfer fluid heaters, (3) fluidized beds, (4) resin reactors, (5) gases for drier regeneration, molecular sieves, and **desiccant**, (6) preheating fluids for reactors and hydro-treaters, (7) heating gases-nitrogen, argon, natural gas, hydrogen, etc., and (8) fuel gas superheating (to flare).

When specifying the equipment, the collaboration of qualified electrical and instrument engineers is

Table 5 Cold box checklist: brazed aluminum plate fin heat exchangers, vessels, and pipework

Topic	Comment
1. Codes, standards, documents ASME VIII-pressure vessels Division 1 (1980) ASME V-nondestructive examination ANSI B16.5 (flanges) ANSI/ASME B3 1.3 (piping) Structural steel standard Bolting standard Surface preparation and painting Design of structural steel Fabrication of structural steel Handrailing standard^b Standard platform details^b Standard ladder details^b Thermowell details Package instrument specification	This is a package unit and requires standards and specifications to cover all of the equipment supplied by the vendor.
2. Thermal design Type of distribution system for fluids into and out of cores Design data for tins used Pressure drop allowable through the circuit Type of fins used, both core and distribution section	Supply phase separation system for two-phase systems? Friction and heat transfer factors f and j. To include all cores, headers, piping, manifolds, distribution sections. Check suitability for process fluid. Oil contamination? Entrained particles (size)? Box shipped without insulation, inspected and tested at site, then filled shortly before start-up. Write procedure.
3. Insulation	
4. Safety systems Check for small leakage by continuous nitrogen purge. Massive release: use trapdoor system or weak panel to give chimney effect. Earthing Relief system	Continuous purge also helps to keep insulation dry. Continuous earth(s) required throughout the box. In piping external to box.
5. Filters/cleaning Mesh size of filters Injection system for methanol wash	Dictated by size of particle passing through between tins. Prevention of freezing.
6. Pipe stressing Anchors Responsibility for stressing from outside piping through the box onto core headers (tanks)	Small piping may be anchored on the wall of the box. The cold box vendor must accept forces and moments imposed by piping outside the box and (if piping not anchored to the box) check that the core headers are not overstressed.
7. Vents and drains	The system should be designed to be self-venting and self-draining. Avoid pockets in lines, manifolds, and headers.
8. External flanges Aluminum pipe flange bolts to (usually) stainless steel pipe flange with different coefficient of expansion. Lag of bolt behind flange temperature (say 35 °C shock load, 10 °C normal operation).	Calculation to show that joints are leak-free at start-up, shutdown, and normal operation in spite of differential expansion.
9. Testing a. On cores: Hydraulic test to code Leak test (1.1 × b design pressure with air) b. On completed box: Nitrogen pressure tested to code (1.25 × DP) soapy water test at maximum 1.5 bars gauge c. Airflow tests: Specify how many streams to be tested	Specify degree of dryness of drying air when removing water and drying. Interstream leak test. Underwater leak test. Normally on completed blocks-it is very difficult to airflow test a completed box. The airflow test checks the geometry of the unit and for blockage. It does not check the process pressure drop calculation.

Check with platefin heat exchanger fabricators before using other codes.

^bIf applicable-depends on extent of supply in the package.

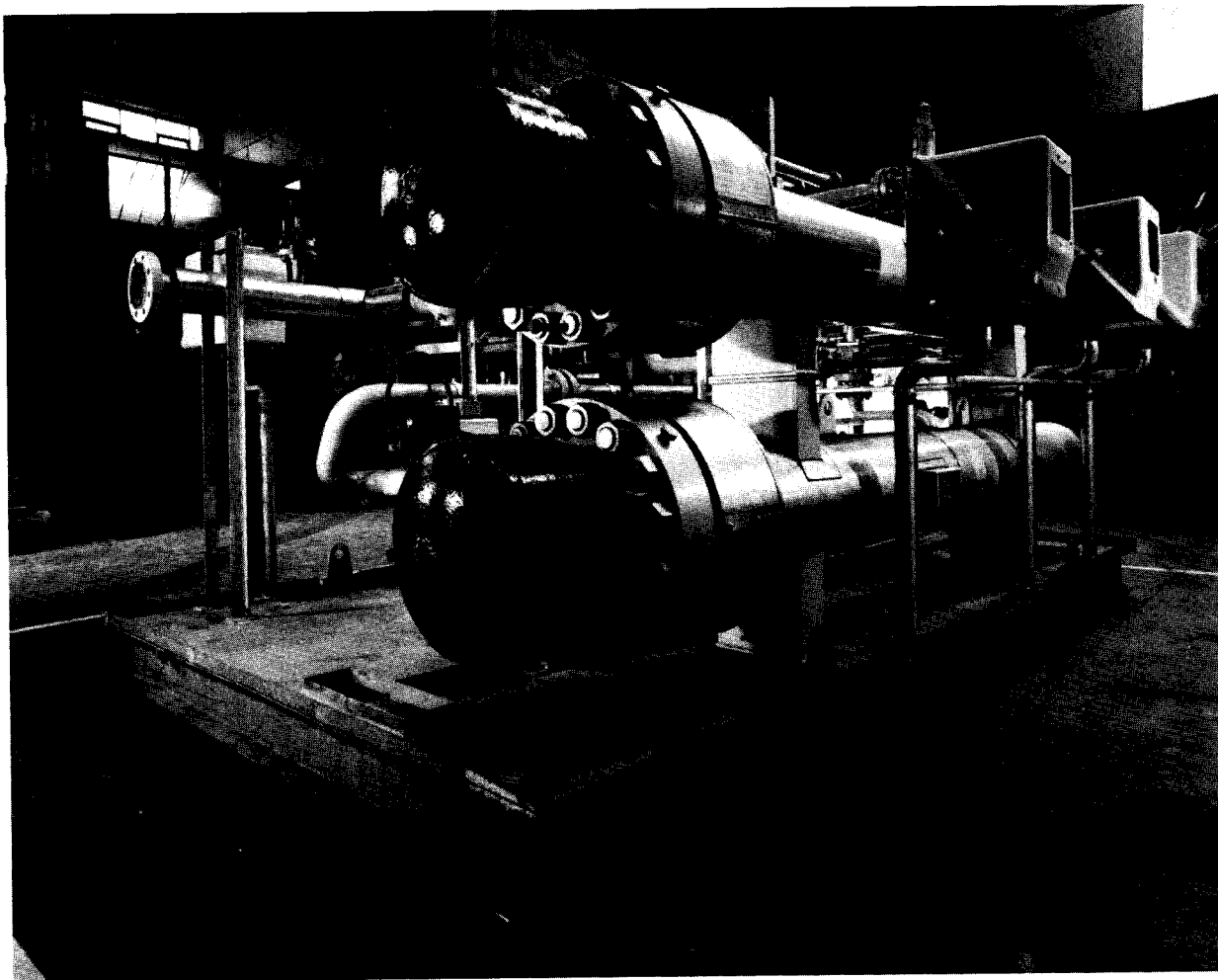


Figure 2 Skid-mounted electrical process heaters. (Photo courtesy of Eltron Ltd.)

needed to ensure adequate reference to electrical standards, local electrical safety requirements, hazard area classification, and surface temperature limitations.

The heater assembly is normally located outdoors on the plant and designed suitably for such a location; the power controller and associated process controls are positioned in a building classified as safe and dust/damp protected.

The shell-side process and mechanical design is similar to **that** for a shell and tube, but allowance must

be made for radiation when specifying the shell design temperature and a careful vibration check must be made to ensure **that** the electrical elements are not damaged.

The electric process heater is a constant-energy, variable-temperature source of heat. Thus the unit, if underdesigned, will tend to overheat.

Table 6 summarizes some general considerations that should be taken into account when specifying this type of equipment. Figure 2 shows a skid-mounted electrical process heater.

Table 6 Electrical process heaters checklist

Topic	Comment
1. Type of electric heater	
a. Shell-and-tube type	Baffling should avoid stagnant areas.
b. Radiant heater (similar to furnace, fired heater type)	Used for very high pressure and/or very low flows. Uniform flux.
c. Water bath type	Good control of heat transfer fluid temperature. Use when process fluid affected by high heater surface temperature.
d. Others: block heaters, intermediate fluid vaporizers, etc.	
2. Electrical supply variations	Variation in rated voltage is generally limited to $\pm 5\%$.
a. Static	Variation in rated frequency is limited to $\pm 3\%$. Sum of variation in voltage and frequency is limited to 5% at any one time.
b. Dynamic	Voltage dips of up to 20% may occur for several seconds. Brief high-voltage spikes due to HRC fuse blowing or deenergizing of unloaded transformers.
3. Control requirements	
Flow 0–100% (if required)	Temperature control.
Process temperature control	Typically $\pm 10^\circ\text{C}$. If this fails there should be a cutout for overtemperature of the process fluid.
Surface temperature	Cutout if the surface temperature of the unit exceeds the area limit.
Unit must be capable of delivering the specified wattage at the minimum voltage. (See 2 above.)	
4. Type of control	
Thyristor	For in-line heat exchangers on-off control is not usually adequate. Control by silicon-controlled rectifier provides stepless proportioning energy input from zero to 100% and has a rapid response.
5. Heater elements	10% spare heater elements should be installed but not connected. (Note: Elements cannot usually be removed and replaced.)
6. Overtemperature protection	At least two thermocouple elements to be clamped (or welded) to the sheath of different elements.
7. Special tests	
a. Electrical insulation of the elements	Normally checked by applying twice the operating applied voltage. Specify the percentage tested.
b. leak-tightness of couplings proved by halogen leak testing	All sheaths, both element and thermocouple, are sealed by mechanical bushing.
8. Temperature limits	
Maximum heat flux and maximum sheath temperature of the heater elements selected to avoid coking or polymerization of the heated material.	

4.9.3

Tenders

David Edge

A. Vendor selection

The engineer may not always have complete control of the list of vendors who are bidding. The client may provide a list, or purchase may be from a particular country that is providing finance. The engineer must insist on a minimum standard for all vendors bidding, or risk being faced with the difficult task of justifying the choice of a more expensive quotation. The time and expense for corrections are usually far greater than the potential savings in accepting vendors who offer low prices but poor quality.

The following should be taken into account before choosing the list of vendors:

1. Technical competence (experience with similar units)
2. Fabrication capability (size, weight, limitations; subvendors)
3. Quality control on fabrication
4. Capacity to deliver on time
5. Provision of drawings on time
6. Qualification requirements for personnel and procedures
7. Whether they have the necessary tools, machinery, and test equipment
8. Facilities to supply the required documentation
9. Previous experience with this type of equipment and record on previous jobs
10. Response to site queries, problems, and changes
11. Financial viability
12. Ability to supply and control package items

B. Inquiry package

When the engineer issues an inquiry package it should contain, as a minimum, the following information:

1. Vendors' document requirements:
 - a. At the proposal stage
 - b. At the purchase order stage
2. Data sheet(s) detailing the process and design conditions for equipment
3. A sketch or drawing of the equipment showing extent of supply
4. A specification detailing codes, standards, vendor responsibilities, design requirements, testing, and inspection
5. Packing and shipping requirements
6. Insurance requirements
7. Liabilities
8. Schedule for:
 - a. Drawings
 - b. Fabrication
9. Type of bid: either sealed tender or open to negotiation
10. Closeout date

Time can be saved and problems avoided if a standard checklist is sent out with the inquiry. Vendors should be informed that their bid may not be considered if they do not complete the form. A typical checklist is shown in Table 1, but ideally a list should be developed for each type of equipment.

Table 1 Typical checklist to be sent with the inquiry package

The vendor must answer yes or no to all the following questions and, where necessary, give details on a separate sheet,

1. Do you accept client conditions of purchase?
2. Are prices subject to escalation?
3. Have you quoted in the currency specified in **the** inquiry?
4. Have you shown country of manufacture?
5. Can you deliver the equipment within the time specified in the inquiry?
6. Is _____ packing included in your price (in accordance with specification _____)?
7. Have you included for the documentation requirements of specification _____?
8. Is your quotation:

Ex works	_____
Delivered jobsite	_____
FOB port of exit	_____
C&F port of destination	_____
10. Does your quotation take into account all the documents in the inquiry?
11. Have you shown weights, dimensions, extent of supply of all items?
Is a weight certificate supplied?
12. Have you included a priced spare parts list?
13. Have you included for commissioning spare parts? (Priced separately?)
14. Have you included for test equipment (test rings, flange and gland, etc.)?
15. Have you detailed all exceptions from the job specification or otherwise stated completely in accordance with all documents, data sheets and specifications ?
16. Is any major part of the equipment fabrication subcontracted?
If yes, have you given details of fabricator, country of manufacture?
17. Have you supplied a list of similar items fabricated over the last (say) 6 years?
18. Have you studied and taken into account offshore certification requirements?
19. Have you sent a copy of your quotation to _____ ?
20. Does your quotation detail daily rates for field installation supervisors and commissioning engineers?

C. Bid evaluation

What should be done when the vendor's quotations arrive? First, there must be agreement with the client on

1. Extensions to late bidders.
2. How many bids are technically reviewed.
3. Whether the client wishes to comment on all bids.
4. A list of people and allied engineering groups who should review and comment on the quotations. Size and weight, for example, may be of particular concern to plant layout and **structural/civils** groups.
5. Client review (if agreed).
6. Questions to vendors.
7. Technical bid tabulation.
8. Commercial bid tabulation. This should be completed after all questions to vendors have been answered and any cost changes incorporated. Costs of power for air-cooled exchangers should be included.

Before placing the order it is advisable to hold a vendor coordination meeting (VCM). The vendor's understanding of the job specification can be confirmed and any outstanding questions and late changes can be incorporated at this stage. Minutes of the VCM should

always be kept and the minutes signed by contractors, vendor, and client. These minutes together with the technical bid analysis should present an accurate record of what was purchased.

Two examples of technical bid analyses are given in Figs. 1 and 2. It should be possible to see omissions or gross errors by comparing one bid with the others, but a detailed design check of the chosen unit(s) is preferred.

List of Publications for Sections 4.9.1-4.9.3

- ASME** Pressure Vessel Code: American Society of Mechanical Engineers, United Engineering Center, 345 East 47th Street, New York, NY 10017.
- ASTM**: American Society for Testing and Materials, 1916 Race Street, Philadelphia, PA 19103.
- ANSI**: American National Standard (published by **ASME**).
- API**: American Petroleum Institute, 2101 L Street, N.W., Washington, DC 20005.
- OCMA**: Oil Companies Materials Association, 14 Belgrave Square, London **SW1X 8PS**, England.

B 805 78.

TECHNICAL BID ANALYSIS FOR ITEM No.

E 101

SERVICE AIR-COOLED TRIM COOLER

JOB No 1156 - A1

COMPILED BY A.T. SHEPHARD

DATA REQUIRED	CONTRACTORS REQUIREMENTS	VENDOR	"A"		"B"		"C"		REF	SELECTED VENDOR	
			REF A 3451 DATED 9-6-85	REF B 9871 DATED 9-17-85	REF C 5782 DATED 9-15-85	REF	A	B			
PROCESS DATA											ALL BIDS ACCEPTABLE
STREAM	HYDROCARBON										
TOTAL FLOW	kg/h	600,000									
DENSITY	kg/m ³	950									
SP. HEAT	kcal/kg °C	0.54									
THERMAL COND	kcal/m ² °C/m	.090									
VISCOSITY	CENTIPOISE	.760									
OP. PRESS	kg/cm ² GAUGE	10.5		10.15		10.13		10.16			
OP. TEMP	°C	146.5									
HEAT EXCH	kcal/h x 10 ⁶	8.2 x 10 ⁶		8.2 x 10 ⁶		8.2 x 10 ⁶		8.2 x 10 ⁶			
OVERALL COEFF	kcal/hm ² °C	FINNED BARE	16.6	390	14.9	308	17	400			
MTD (°C)	°C		73.6		71.5		75.5				
PRESS DROP	kg/cm ²	0.7 ALL CALC	0.7	0.35	0.7	0.37	0.7	0.34			
VELOCITY	m										
FOULING	(kcal/h m ² °C) ⁻¹	.0006	.0006		.0006		.0006				
TUBE DATA											REMARKS ① BID "C" USES 40-hp MOTORS ON "V" BELTS. API 661 LIMIT IS 30hp - ALTHOUGH WE ACCEPT THIS WE PREFER API LIMIT. ② VIBRATION SWITCHES ARE FLAMEPROOF. ③ MOTORS WILL BE FREE ISSUE TO VENDOR BY CLIENT.
TOTAL AREA PROVIDED (FINNED)			6720		7755		6400				
(BARE)			286		376		272				
TOTAL TUBES	NUMBER x OD x BWG x A214	292 x 1" x 12 BWG A214	264 x 3/8" x 12 BWG A214	328 x 25.4 mm x 13 BWG A214							
PASSES	BY VENDOR	ONE	TWO	ONE							
LENGTH	12.2 m (MAX)	12.2	12.2	10.5							
FIN HEIGHT/NO INCH	TYPE OF FIN	5/8" x 11 G FIN	15.9 mm x 11 G FIN	15.9 mm x 11 G FIN							
ROWS	BY VENDOR	4 AT 60 mm PITCH	4 AT 82.6 mm PITCH	4 AT 63.5 mm PITCH							
NO BUNDLES & WT		2	2	2							
AIR DATA											
DESIGN AIR TEMP	/ AIR OUT °C	35 °C	35	80.3	35	86	35	76			
FACE VELOCITY (30 °C)	/ STATIC AP	3.5 m/s 12 mm w.g.	2.5 m/s 11 mm w.g.	3.4 m/s 12 mm w.g.							
TOTAL AIR FLOW	(STANDARD)	184 m ³ /s	154 m ³ /s	185 m ³ /s							
TOTAL POWER (USED)	AT 35 °C	36.4 kW	32.7 kW	44.7 kW							
INSTALLED POWER		44.0 kW	44.7 kW	60 kW							
MOTORS AND		2 x 22 kW	2 x 30 H.P.	2 x 40 H.P.							
POWER MARGIN		21%	37%	33%							
FANS (NO.) / SIZE / A R	16% + TRANSMISSION ASPECT RATIO (A.R.) < 2.5	2 x 13 FT ; A.R. = 2.2	2 x 14 FT ; A.R. = 2.32	2 x 13 FT ; A.R. = 2.2							
AV. A.P.	ADJUSTABLE PITCH	ADJUSTABLE PITCH	ADJUSTABLE PITCH	ADJUSTABLE PITCH							
VIBR. SWITCH	REQUIRED	YES	YES	YES							
SPEED REDUCER	V BELT	V BELT	V BELT	V BELT							
CONSTRUCTION											
WALKWAYS	SEE DRG WL 123	ACCEPTABLE } SEE DRG	ACCEPTABLE } SEE	ACCEPTABLE } SEE							
LADDERS	- " - - " - - " -	- " - - " - - " -	- " - - " - - " -	- " - - " - - " -							
LOUVERS & HEATING COILS	NONE	NO	NO	NO							
PAINTING	SPECIFICATION P101	YES	YES	YES							
SURFACE PREP	- " - - " - - " -	YES	YES	YES							
EQUIPMENT DELIVERED	ASSEMBLED IN PARTS	IN PARTS	IN PARTS	IN PARTS							
SPARES LISTED & PRICED:	REQUIRED (FORM SP100)	YES	YES	YES							
SOUND SPECTRUM	REQUIRED	ACCEPTABLE	ACCEPTABLE	ACCEPTABLE							
NOZZLES PER BUNDLE		2 x 10" IN 2 x 10" OUT	3 x 6" IN 3 x 6" OUT	3 x 6" IN 3 x 6" OUT							
POWER COST (-YRS & Kw)	2 YRS AT .04 \$/KWh	\$ 12,000	\$ 11,600	\$ 15,800							
TOTAL PLOT & WEIGHT		6 m x 12 m x 31 TONS	6 m x 12 m x 26.4 TONS	7 m x 11 m x N/A							
REMARKS											
*BASED ON RATIO OF DENSITY OF AIR AT WINTER DESIGN TEMPERATURE TO DENSITY OF AIR AT SUMMER DESIGN TEMPERATURE.											

53110-B1		SEAWATER / CIRCULATING WATER										JOB No. 1156-A2		COMPILED BY A.T. SHEPHARD	
TECHNICAL BID ANALYSIS FOR ITEM No. E 1001		SERVICE HEAT EXCHANGER													
DATA REQUIRED		CONTRACTOR'S REQUIREMENTS		VENDOR 1		VENDOR 2		VENDOR 3		VENDOR 4		REASON		REMARKS	
				REF V1/ABQ DATED 10-12-85		REF V2/PLC DATED 10-11-85		REF V3/T DATED 10-11-85		REF					
PROCESS DATA (NOTE 1)															
STREAM		SEAWATER		CIRC. WATER		SEAWATER		CIRC. WATER		SEAWATER		CIRC. WATER			
TOTAL FLOW		3,000,000		2,643,000						3,250,000					
DENSITY		1024		993											
SP. HEAT		1.0		1.0											
THERMAL COND.		.833		.54											
VISCOSITY		.73		.71											
OP. PRESS		3.5		8.0											
OP. TEMP		27 → 37		44 → 32.7						27 - 35		44 → 32.7			
HEAT EXCH.		30.0		30.0						30.0		30.0			
OVERALL COEFF.		CLEAN		SERVICE		4.250		3.612		4.025		3.421		4.400	
STREAM COEFF.						N/A		N/A		N/A		N/A		N/A	
PRESS. DROP		1.0		1.0		0.87		0.88		0.9		0.84		1.0	
VELOCITY															
FOULING		SEE BELOW													
OVER CAPACITY		0.85		CLEANLINESS											
TOTAL AREA PROVIDED		m ²		1312		1385		1114							
AREA PER PLATE		m ²		2.1		2.2		2.6							
No. PLATES PER FRAME		MAXIMUM		316		500		315		550		429		530	
MECHANICAL DATA															
DESIGN CODE & SWEL SPEC		ASME VIII DIV 1		ASME VIII DIV 1		ASME VIII DIV 1		ASME VIII DIV 1							
EARTHQUAKE DESIGN		YES		YES		YES		YES							
DESIGN PRESS		10		15		10		15		10		15			
DESIGN TEMP		70		70		70		70		70		70			
MATERIALS OF CONSTRUCTION															
PLATES		TITANIUM (ASTH SB265)		SB 265 Gr 1		SB 265 Gr 1		SB 265 Gr 1							
GASKETS		BY VENDOR		NITRILE		NITRILE RUBBER		NITRILE RUBBER							
BOLTING		//		24 BOLTS A193 B7		16 BOLTS A193 B7		24 BOLTS A193 B7							
FRAME (PLATE & PRESSURE PLATE)		CS		SA 516-60		SA 516-60		SA 516-70							
SLEEVES / NOZZLES															
CONSTRUCTION															
NOZZLES (No. SIZE, RATE & AC)		ANSI 16.5 150*RF		4x14" 150*RF		4x14" 150*RF		4x16" 150*RF							
NOZZLE LOCATION		ALL AT SAME SIDE		YES		YES		YES							
No. PACKS PER UNIT				2		2		ONE							
ARR. OF PACKS SERIES PARA				PARALLEL		PARALLEL									
MISCELLANEOUS															
BOLT TIGHTENING TOOLS REQ.		TORQUE UNIT + SOCKETS		YES		YES		YES							
BOLT TIGHTENING TOOLS SUPPLIED		NO		INCLUDED		QUOTED AS EXTRA		QUOTED AS EXTRA							
PROTECTIVE SCREENS		NO		NO		NO		NO							
STRAINERS		VENDOR RECOMMENDATION		3 → 4mm MESH		2 1/2 mm MESH		N/A							
PAINTING		SPEC SP 123		YES		YES		YES							
SURFACE PREP		SPEC SP 123		YES		YES		YES							
EQUIPMENT DELIVERED		ASSEMBLED IN PARTS		ASSEMBLED & CRATED		ASSEMBLED & CRATED		ASSEMBLED & CRATED							
VENDOR EXCEPTIONS TO SPECS				NONE		NONE		INCREASED SEAWATER							
SPARES LISTED AND PRICED		REQUIRED FORM SP100		GASKETS ONLY		YES GASKET, PLATES, AND BOLTS		GASKETS ONLY							
REMARKS		DO NOT STORE		LARGE NUMBER OF		GASKETS LONG TERM.									
NOTE 1: ONLY WHEN VENDOR MODIFIES															
PROCESS DATA IS IT NECESSARY TO LIST															

Figure 2 Technical bid analysis for plate exchanger

4.10

Expansion bellows

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4.10.1

Introduction

M. Morris

■ An expansion joint is a specially designed component of compact dimensions that allows for differential movements between two adjacent components and that maintains the pressure envelope. This joint can be of all-metal construction or be nonmetallic for low-pressure application; it can be discontinuous as in a packed gland (e.g., TEMA P-type head). In this section the discussion on expansion joints is limited to metallic convoluted bellows.

Many years of the suppliers' experience are incorpo-

rated in the design and fabrication details of the bellows; however, it is important for the exchanger designer to appreciate the capabilities of the bellows unit so that problems are not unwittingly built into the exchanger. The decision to use bellows should be made at the initial thermal and mechanical design stage of the exchanger. This section gives guidance on the type, application, design, and analysis; it suggests the scope of an exchanger bellows specification and discusses fabrication and operational aspects.

4.40.2

Expansion bellows

M. Morris

A. Types of bellows

Bellows can be classified in many ways, for example, by the shape of corrugation, material, presence or absence of reinforcing rings, or number of walls. The classifications used in this section are

- Thin-wall bellows (single or multi-ply)
- Medium-wall bellows
- Fabricated thick-wall bellows

Table 1 illustrates the range of these classifications.

(a) *Manufacture*

Hydraulic forming is used for bellows up to approximately 1.8 m in diameter. A welded cylinder is placed in the center of a stack of split dies, which are machined to determine the final convolution shape. Internal pressure and controlled axial compression are applied, as shown in Fig. 1a. A high pressure is used in the forming process, which thus imposes a leak test on the final bellows; however, because this pressure is applied against external rings, the structural strength and stability of the bellows are not proven in the forming process.

Mandrel and wheel rolling of bellows is available for a wide range of bellows diameters, for thin-wall or medium-wall bellows, and is applied hot or cold depending on plate thickness. A welded cylinder is placed in the forming machine, as seen in Fig. 1b, and internal and external forming wheels exert pressure simultaneously to the inside and outside surfaces while the cylinder revolves.

Thick-wall bellows are generally made from plate of the same material and similar thickness to the shell barrel. Segments are pressed and welded as shown in Fig. 1c or *d*.

Cold rolling, and hydraulic forming to a slightly lesser degree, produces work hardening of the bellows material. **Postform** annealing is sometimes specified to increase corrosion or creep resistance. This operation reduces the cold work yield strength of the bellows, and the effect of this reduction is discussed in Sec. D(a) following.

(b) *Single and multi-ply thin-wall bellows*

Depending on the wall thickness and convolution size, single-wall thin bellows may be limited by stress or stability to lower pressure application. To overcome this limitation, multi-ply bellows are available, made by telescoping two or more cylinders and forming together.

Multi-ply bellows may also be considered for the following reasons:

1. To reduce the risk of sudden and complete failure
2. For multiple material usage, for example, with the inner ply in high-corrosion-resistance material and the outer ply in less costly high-strength material for load resistance.

Note that the fatigue resistance is limited by the inner ply. The condition of the inner ply can be monitored by pressure sensing, although this would not be practical for internal floating-head bellows.

Table 1 Range of bellows classifications

Bellows	Thin wall	Medium wall	Thick wall
Wall thickness, mm	0.5 to 2	2 to 4.5	4 to 13+
Convolution height, mm	25 to 75	50 to 65	75 to 150
Materials include	Stainless steel Monel Inconel Incoloy	CrMo alloy steel Stainless steel	As shell
Manufacture	Cold rolling Hydraulic forming	Hot rolling	Pressing and welding
Comments	Restraining rings often needed Flexible External protection needed	Compromise between thin and thick wall	Rugged but stiff Usually a maximum of two or three convolutions because of difficulty of equalizing movement

(c) Reinforcing or equalizing rings

Reinforcing or equalizing rings are sometimes supplied with thin-wall bellows. Typical rings are shown in Fig. 2. Reinforcing rings resist any distortion of the convolution root and are easily fitted to bellows that are formed hydraulically. Equalizing rings can be of cast or fabricated construction, generally in two halves bolted together. These rings also prevent convolution root distortion but additionally limit the compressive axial deflection taken by each element. Both types of rings are claimed to improve the fatigue life.

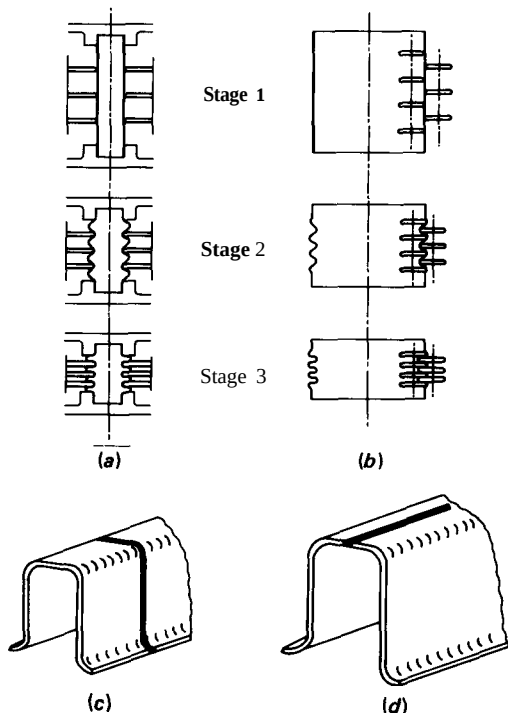


Figure 1 (a) Thin-wall bellows: hydraulic forming process. (b) Thin- and medium-wall bellows: roll forming process. (c) Thick-wall bellows; pressed segments. (d) Thick-wall bellows; pressed halves (flanged and fluted type).

(d) Internal sleeves

Bellows can be sleeved for the following reasons:

1. To reduce turbulence and thus pressure drop
2. To minimize erosion on the walls
3. To restrict entry of foreign material

Consideration should be given to fitting sleeves on steam duty, gas flow rates greater than 7 m/s and liquid flow rates greater than 3 m/s. The *Standards of the Expansion Joint Manufacturers Association (EJMA)* [1] gives detailed recommendations. Sleeves should always be designed with the minimum practical clearance to restrict entry of foreign material. There should be sufficient overlap at the free end to ensure that with all possible movements, especially if lateral movement is involved, there is no chance of the sleeve end fouling the convolutions. A typical sleeve arrangement is shown in Fig. 4.2.6(25).

It is wrong to assume that a sleeve can completely prevent deposition of solid material in the convolutions, since back eddies can easily result in sedimentation behind the sleeve. In fact, a sleeve can frequently help to trap solid material against the bellows, where it might otherwise have been carried away in the turbulent flow. The most practical way to prevent solids from getting

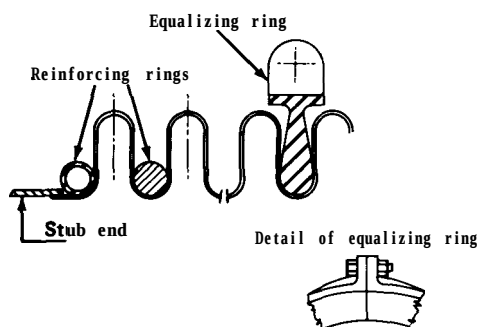


Figure 2 Thin-wall bellows details.

into the bellows/sleeve space is by use of purge medium continuously supplied to this space. The draining of this space of any corrosive products must be considered.

B. Applications

(a) Fixed tubesheet exchangers

Bellows are fitted into the shell of fixed tubesheet exchangers when:

1. The relative axial displacement of tubes and shell due to pressure and/or thermal loads (see Sec. C below) cannot be physically or economically absorbed.
2. Flexibility is required for the installation of shell-side gaskets. This type of exchanger is shown in Fig. 3 and is used when the shell cannot be welded to the tubesheets. Its use is generally limited to low-pressure nonlethal duty as the gasket cannot be replaced and the joint has little flexibility.

All three types of bellows are used for these applications. Internal sleeves are often used, and external protection to guard against accidental damage during construction, erection, and operation is essential for thin-wall, advantageous for medium-wall, but unnecessary for thick-wall bellows.

An alternative bellows type, externally pressurized, is sometimes specified and is shown in Fig. 4. The main advantage claimed [2] is that transferring the pressure to the outside overcomes squirm instability in bellows with high length-to-diameter ratios.

Heavy-wall bellows, with their large convolution volume, may for process reasons require the fitting of drains and vents to the convolutions. These should be specified only where absolutely necessary and only for clean duty. Thin-wall bellows, with their small convolution volume, do not need them and cannot be fitted with them. Medium-wall bellows can be fitted with drains and vents, but it is not recommended.

It is good practice to position exchanger supports remote from thin-wall shell bellows. The resultant local support loads could otherwise lead to bellows distortions.

(b) Floating-head exchangers

For process reasons, single tube-side pass, floating-head exchangers are sometimes required. The relative

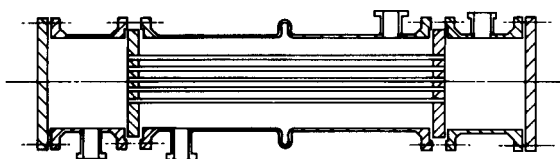


Figure 3 Fixed tubesheet exchanger with shell flanges.

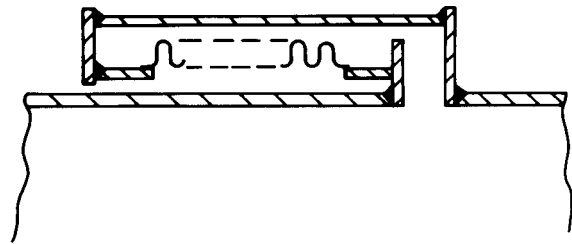


Figure 4 Externally pressurized bellows.

movements between the shell cover and the floating-head cover are absorbed by a bellows unit, which is invariably either a thin- or medium-wall type. Figure 5 shows two configurations of internal and external bellows position. Many variants of flanged or welded installation are possible, depending on ease of manufacture, frequency of maintenance, and so forth.

C. Design loadings

The conflicting design criteria for a bellows require a compromise between ruggedness to resist internal or external pressure and flexibility to absorb movement. This compromise then has to be optimized to give the best fatigue resistance against cyclic variation of pressure and movement. Such design and optimization are carried out by the bellows designer; the applied loads are specified by the exchanger designer.

(a) Pressure

A bellows unit will tend to lengthen under internal pressure (and shorten under external pressure); however, with shell-and-tube exchangers the tube bundle effectively provides total axial restraint to this movement. Thus the stresses in the bellows are the sum of those due to imposed axial movement (see below) plus those due to pressure in the convolution.

Where pressure pulsations are possible, the resonance of the bellows should be considered.

(b) Movements

Relative axial movement will be imposed on both fixed tubesheet and single-pass floating-head exchanger bellows. This movement results from differences in temperature between tubes and shell and also from the Poisson effect of pressure inflation. These movements can be calculated from exchanger code rules and are discussed in Sec. D.

Relative transverse movement in floating-head bellows will be caused by tube bundle distortions and will only be limited by controlled gaps between the floating head and the adjacent shell. Typical spacers are indicated in Fig. 5.

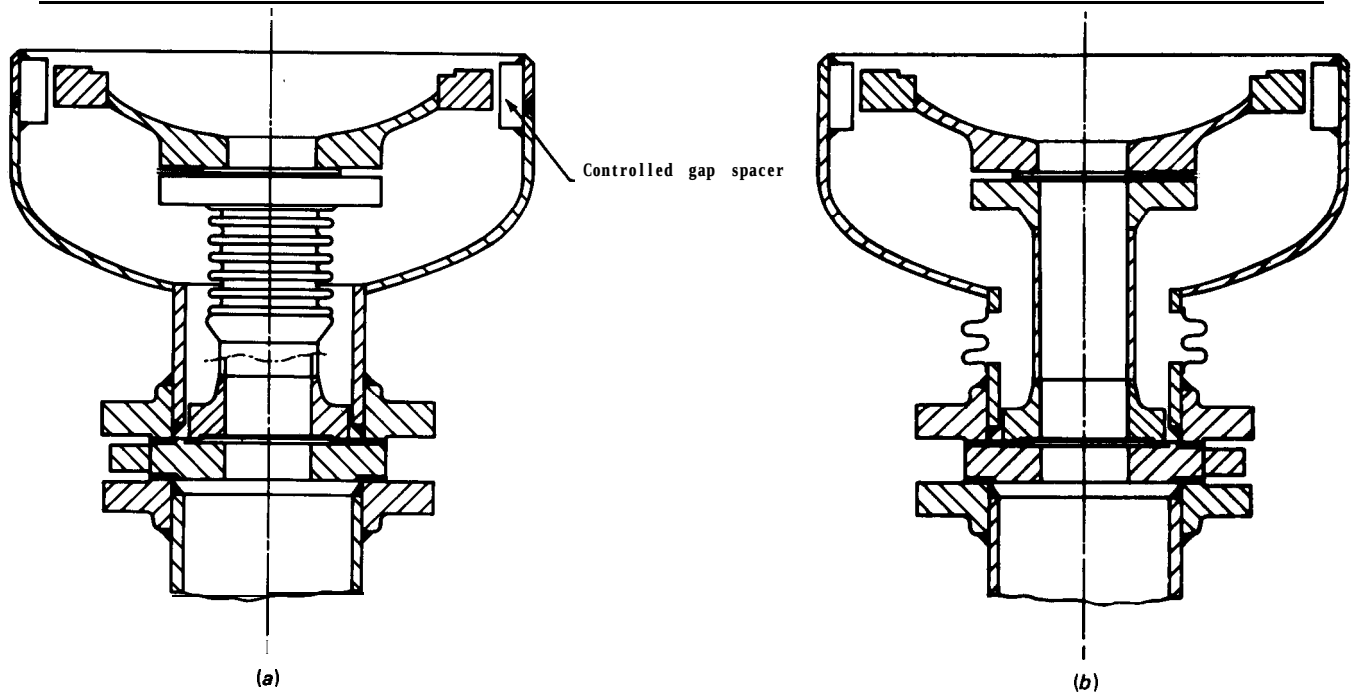


Figure 5 Floating-head bellows: (a) internal type, (b) external type.

The full range of movements of the bellows needs to be specified. Cases to be considered include exchanger testing, start-up, normal and abnormal operation, shut-down, cleaning, and steamout. Fabrication tolerances have to be taken into account, both axial and lateral as well as possibly torsional.

(c) Cyclic conditions

Local stresses in bellows are plastic and thus the cyclic variation of the above loads needs to be realistically assessed and correctly specified. The design life is generally between 10^3 and 10^5 cycles, with 7 500 cycles generally taken as standard unless otherwise specified. The cyclic requirements of individual loads and movements need to be agreed on with the bellows designer. The bellows supplier can justify the fatigue life expectancy by analysis supplemented with test results.

D. Analysis

The purpose of this section is not to present detailed formulas for predicting stress levels, fatigue lives, and so on, but to discuss the background of bellows analysis, to introduce the references available, and to review the important factors in bellows analysis. Bellows are different from other pressurized components as there will be several local areas with stresses close to or at plastic levels. With design lives often between 10^3 and 10^5 cycles it would be normal with other components to try to keep peak stresses to elastic levels and thus remain below the endurance limit.

The early approach to analysis was to reduce the bellows shape to simple elements in order to predict stresses, and to modify the equations by experience and testing.

Currently, numerical computer methods are available to increase the accuracy of analysis and reduce the dependence on verification by testing. However, actual stresses and fatigue resistance are sensitive to the manufacturing method and local geometric deviations, and evidence from testing is likely to remain the best demonstration of design adequacy.

(a) Pressure

All the referenced work appears to have been carried out on internal pressure. Within the limitations of linear elastic analysis, the stresses would be reversible with external pressure.

Early analysis by Kopp and Sayre [3] made simplifying assumptions to allow solution for thick-wall bellows. The toroidal corners were replaced by sharp corners and the annular plates by beam strips. Nevertheless, Kopp and Sayre obtained good agreement with their experimental data obtained from tests on thick-wall bellows, and their design method is still used. It should be noted that the above simplifications, while giving reasonable agreement for predicted bellows flexibility, will overestimate stresses at the corners. Singh and Soler [4] offer an update on the original Kopp and Sayre method.

Anderson [5] based his analysis on elastic shell theory and additionally incorporated toroidal elements. This work forms the basis of the EJMA equations, and

the validation of these equations by a test program is discussed by Thomas [6].

Recently, finite-element analysis of bellows has been carried out. Smith [7] installed strain gauges on thick-walled bellows of two fixed tubesheet exchangers and compared the results with thin-shell and thick-shell finite-element analysis. The bellows were stressed in the elastic range, and reasonable agreement was achieved. Even though the toroidal corner radius-to-thickness ratios were 2.67 and 3.57, there was little difference between the predictions of the thin- and thick-shell finite-element models. Becht et al. [8] compared the results of an elastic plastic test on a thin-wall bellows with an elastic plastic finite-element analysis. They found good agreement with the stiffness predicted by the EJMA equations and some agreement with the EJMA stress equations. They concluded that within the limitations of manufacturing tolerances elastic finite-element analysis can give good agreement with test results and that elastic plastic finite-element analysis is feasible.

As pressure increases above design values, local distortions will increase, leading to bellows instability and failure. The initial distortion is generally bulging of the convolution sidewall, which (within limits) may be acceptable for thin-wall bellows, especially if fitted with restraining rings. Further increases in pressure result in wavelike tilting of convolutions, then bulging out of the root of the convolution. Testing of bellows involving these instabilities is discussed in [6] and [9]. An instability pressure will be essentially proportional to yield stress; thus, annealing, if used, will reduce the resistance to instability. Becht and Skopp [9] found that the EJMA criteria were unconservative for annealed bellows. Bellows are more stable under external pressure, and stability review should not be necessary for this case.

(b) Movement

In both fixed tubesheet and floating-head exchanger applications, the bellows will invariably be more highly stressed by the imposed movements than by the pressure. Fortunately, there appears to be general agreement that the EJMA formulas for stresses resulting from movement and for bellows stiffness are reasonably accurate for elastic conditions.

(c) Stress limits

ASME Code case 1177-7 recognizes that a bellows component can be justified by rational stress analysis but also recognizes that stresses cannot be limited to allowable values used for membrane stresses. In the EJMA formula for bending stress due to pressure, a modifying coefficient is introduced to allow comparison

with allowable membrane stresses. In other pressurized components, similar high local stresses are permitted as secondary stresses, which redistribute into adjacent elastic material. Jetter and Jaquay [10] found that redistribution does not necessarily occur with bellows, and thus the acceptability of high stress levels must be attributed to the benefits of cold working.

(d) Fatigue Analysis

The fatigue life of a bellows will depend on stresses induced by pressure and deflection. While the location of the peak stress due to pressure will not necessarily coincide with the location of peak deflection stress, EJMA sums the two effects for fatigue life evaluation. The EJMA formula has its basis again in the work of Anderson. Fatigue test results and a design formula for the estimation of low cycle fatigue life of bellows are discussed in [11].

Many factors influence the fatigue life of a bellows for a given range of loading conditions, the most important probably being the material condition and the manufactured shape. Thus the manufacturer's fatigue test results for similar bellows are likely to give the best reassurance of bellows acceptability.

E. Code rules

Code requirements are discussed in two parts: (1) codes and standards that deal with the bellows as a component and (2) codes that deal with the application of bellows in exchangers.

(a) Bellows code requirements

I. EJMA

The *Standards of the Expansion Joint Manufacturers Association* [1] is a comprehensive set of recommendations dealing with all aspects of the bellows component. *The Standards* particularly relate to bellows in piping systems, but there are many recommendations that have relevance to exchanger bellows, and in fact these standards refer to the requirements of the ASME [12]. Detailed recommendations in EJMA cover specification, design, manufacture, inspection, testing, documentation, shipping, and installation. In particular, the design section gives formulas for evaluation of stability and stresses due to pressure and movement for reinforced and unreinforced bellows. These formulas are discussed in [6], and it should be noted that EJMA requires the bellows manufacturer to correlate his design with test data.

II. ANCC

The Italian Pressure Vessel Code [13], chapter **VSRI**P, contains rules for single-wall and multi-ply bellows. The thickness formulas are summarized in this volume, Sec. 4.3.2.K, and some requirements are given in the code for manufacture and testing.

III. AD Merkblatter

Section B13 of the German Pressure Vessel Code [14] gives formulas for checking the bending stress and circumferential stress in single-wall bellows.

(b) Exchanger code requirements

The tubesheet formulas of TEMA [15], BS 5500 [16] and AD Merkblatter [14] incorporate the case of bellows in the shell, and these formulas are discussed in this volume, Sec. 4.3.2.1. **ASME** Code case 1177-7 discusses the conditions to be complied with, to allow the incorporation of a bellows into an **ASME** VIII division I vessel. The bellows material and fabrication are required to be inspected at the supplier's, and the exchanger manufacturer is required to obtain supporting data to allow the exchanger to be code stamped. Reference is made to the section of **ASME** VIII division I for the evaluation of membrane stress levels.

F. Specification

The bellows supplier can assume responsibility for his product only if he has been given sufficient engineering data. This must be specified by the exchanger designer and must detail all the requirements relevant to the bellows. The following is a checklist with notes that should be used to draw up a particular bellows specification. These points are discussed in more detail in [17].

(a) Basic data

Acceptable bellows types
Applicable bellows code
Type of ends required:
 Weld preparation for heavy wall
 Weld stub end
 Flange
Internal sleeve
External protection [see Fig. 4.2.6(25)]
Bellows vents and drains

(b) Design data

Design pressure and temperature
Upset pressures and temperatures with frequencies (including start-up)

Movements with frequencies

 Axial (extension and compression)

 Lateral (for floating-head bellows)

Fatigue life requirements

Design justification for bellows (if required)

(c) Material data

Acceptable materials of bellows and ends

Applicable product data

Adjacent shell or pipe materials

Mill test reports

Material certification requirements

Note: The specification of a corrosion allowance is not relevant to bellows; the correct material must be chosen instead. If not, the highly stressed areas will cause selective corrosion and flaking off of any protection oxides.

(d) Dimensional data

Shell or pipe diameter and thickness

Installation position (horizontal or vertical)

Any limitation on maximum length

(e) Manufacturing requirements

Acceptable manufacturing methods

Welder qualification

Inspection required for bellows, ends, and accessories, for example :

 Chemical and mechanical analysis

 Hardness limitations

 X-ray, ultrasonic, dye penetrant, magnetic particle inspection

 Spring rate test

 Leak or hydraulic test

(f) Shipping requirements

Shipping restraints to maintain length during transportation [see Fig. 4.2.6(25)]

Packing requirements-protection for handling and lifting

Documentation and marking

G. Exchanger fabrication aspects

It should be remembered that bellows are relatively fragile items of equipment and in operation will probably be the most highly stressed exchanger component. Bellows should therefore be kept in their shipping crates up to the time they are required for installation. The

shipping restraints should be retained for as long as possible during exchanger fabrication but should be removed before exchanger hydrotesting. For floating-head bellows, the restraint bars will be left in place until after the last tube-side test to avoid any stretching of the bellows. In some cases bellows are supplied in a **pre-loaded condition; in all cases the shell or pipe gap** must be checked before bellows installation.

The following are additional installation reminders:

1. After unpacking, the bellows should be visually checked for dents or defects.
2. If an internal sleeve is fitted, check that the bellows is installed the right way around.
3. Bellows should be protected from weld spatter.
4. Avoid fitting weld leads that would pass welding current through thin-wall bellows.
5. Check that convolutions are free of foreign material.
6. Check any limitation on bellows pressure before applying exchanger pressure test.

H. Operational aspects

(a) Inspection

Care with exchanger bellows must continue after specification, manufacture, and installation to include care during service. Particularly for shell bellows it is recommended that:

1. After commissioning, the exchanger be checked to ensure that there is no unanticipated shell-side vibration.
2. Regular visual examination be done for any surface damage and debris between the convolutions.

(b) Repair

Accidental damage to shell bellows can occur. Experience has shown that denting, arc strikes, tool gouges, and so forth can seriously reduce the fatigue life of the bellows. Unless the exchanger was originally designed with the possibility of a replacement by an oversize bellows in mind, then repair has to be considered. Repair of thick-wall bellows should be possible with weld procedures similar to those used in the original fabrication. If the repair is not made in a highly stressed area of the bellows, the life of the bellows should not be seriously affected. Repair of thin-wall bellows has been evaluated in [18]. The authors intentionally damaged a two-ply 304 stainless bellows by various means: drilling holes, grinding, and grooving. Weld repair was carried out by welding patches over the defects or filling the grooves or contour grinding. Fatigue testing was carried out to check the effect of the various repair techniques. These tests showed contour blending to give the best life, followed by weld filler metal, and then welded patches.

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