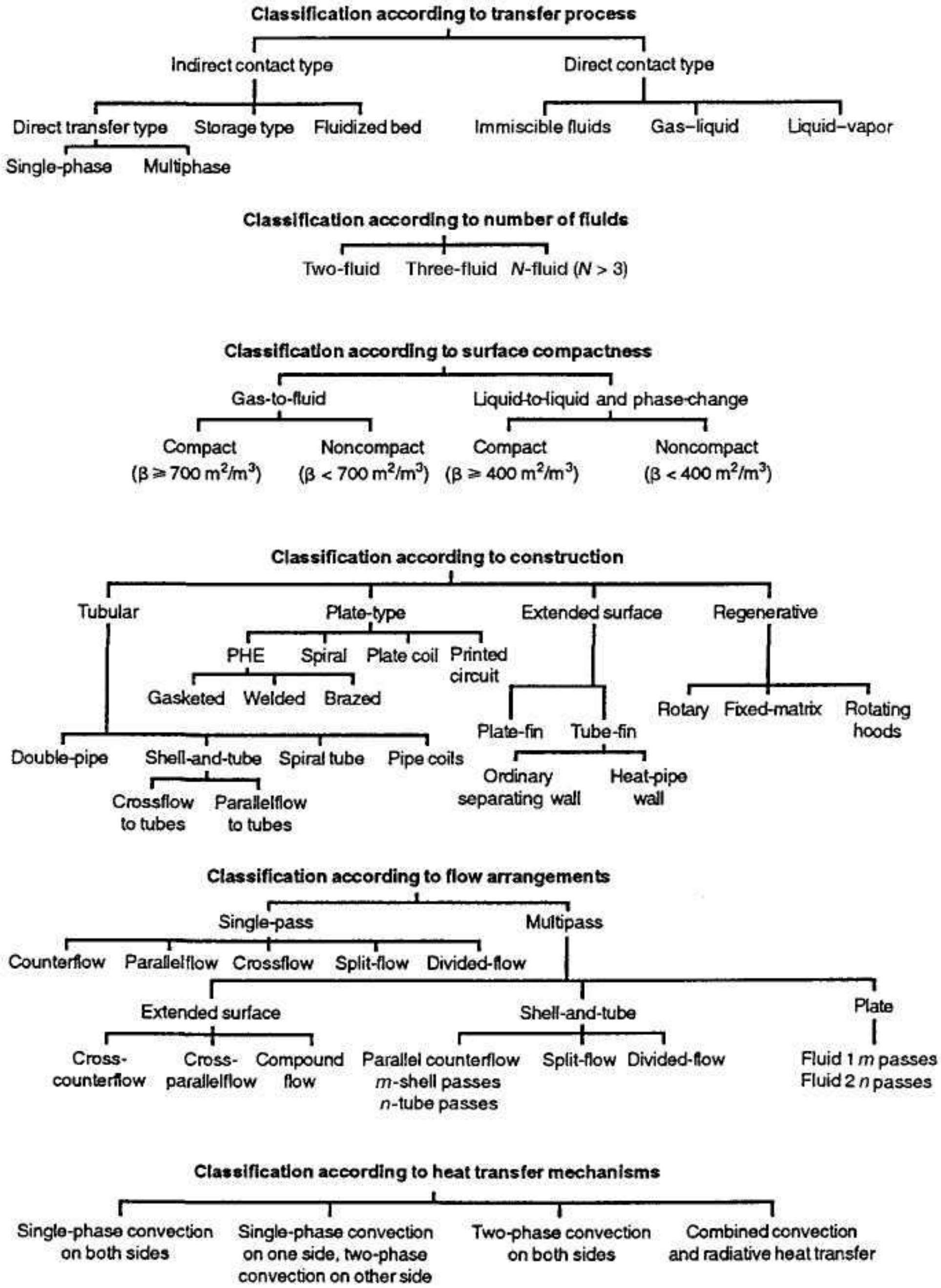
Heat Exchangers

A heat exchanger is a device that is used to transfer thermal energy (enthalpy) between two or more fluids, between a solid surface and a fluid, or between solid particulates and a fluid, at different temperatures and in thermal contact. In heat exchangers, there are usually no external heat and work interactions. Typical applications involve heating or cooling of a fluid stream of concern and evaporation or condensation of single- or multicomponent fluid streams. In other applications, the objective may be to recover or reject heat. In a few heat exchangers, the fluids exchanging heat are in direct contact. In most heat exchangers, heat transfer between fluids takes place through a separating wall or into and out of a wall in a transient manner. In many heat exchangers, the fluids are separated by a heat transfer surface, and ideally they do not mix or leak. Such exchangers are referred to as direct transfer type, or simply **recuperators**. In contrast, exchangers in which there is intermittent heat exchange between the hot and cold fluids - via thermal energy storage and release through the exchanger surface or matrix - are referred to as indirect transfer type, or simply **regenerators**. Such exchangers usually have fluid leakage from one fluid stream to the other, due to pressure differences and matrix rotation/valve switching.

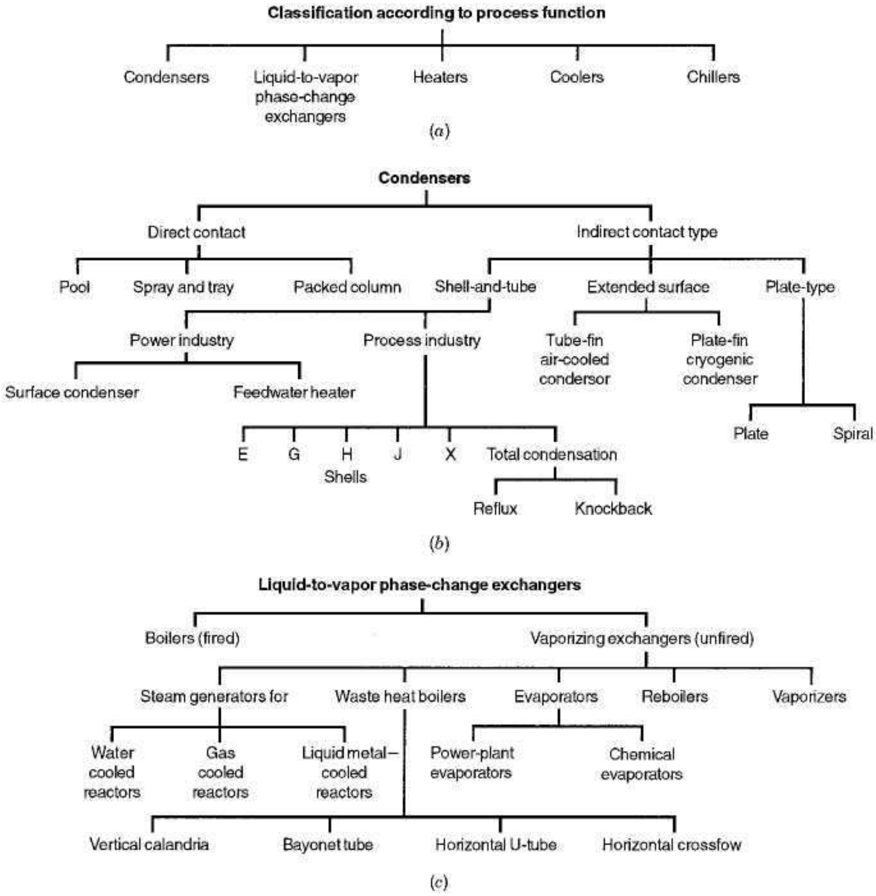


Common examples of heat exchangers are shell-and-tube exchangers, condensers, evaporators, preheaters, dryers and cooling towers. If no phase change occurs in any of the fluids in the exchanger, it is sometimes referred to as a sensible heat exchanger. There could be internal thermal energy sources in the exchangers, such as in electric heaters and nuclear fuel elements. Combustion and chemical reaction may take place within the exchanger, such as in boilers and fired heaters.

Heat transfer in the separating wall of a recuperator generally takes place by conduction. In general, if the fluids are immiscible, the separating wall may be eliminated, and the interface between the fluids replaces a heat transfer surface, as in a **direct-contact heat exchanger**.

A heat exchanger consists of heat transfer elements such as a core or matrix containing the heat transfer surface, and fluid distribution elements such as headers, manifolds, tanks, inlet and outlet nozzles or pipes, or seals. Usually, there are no moving parts in a heat exchanger; however, there are exceptions, such as a rotary regenerative exchanger (in which the matrix is mechanically driven to rotate at some design speed) or a scraped surface heat exchanger.

The heat transfer surface is a surface of the exchanger core that is in direct contact with fluids and through which heat is transferred by conduction. That portion of the surface that is in direct contact with both the hot and cold fluids and transfers heat between them is referred to as the primary or direct surface. To increase the heat transfer area, appendages may be intimately connected to the primary surface to provide an extended, secondary, or indirect surface. These extended surface elements are referred to as fins. Thus, heat is conducted through the fin and convected (and/or radiated) from the fin (through the surface area) to the surrounding fluid, or vice versa, depending on whether the fin is being cooled or heated. As a result, the addition of fins to the primary surface reduces the thermal resistance on that side and thereby increases the total heat transfer from the surface for the same temperature difference. Fins may form flow passages for the individual fluids but do not separate the two (or more) fluids of the exchanger. These secondary surfaces or fins may also be introduced primarily for structural strength purposes or to provide thorough mixing of a highly viscous liquid.
Not only are heat exchangers often used in the process, power, air-conditioning, refrigeration, cryogenic, heat recovery, alternative fuel, and manufacturing industries, they also serve as key components of many industrial products available in the marketplace. These exchangers can be classified in many different ways. We will classify them according to transfer processes, number of fluids, and heat transfer mechanisms. Conventional heat exchangers are further classified according to construction type and flow arrangements. Another arbitrary classification can be made, based on the heat transfer surface area/volume ratio, into compact and noncompact heat exchangers. This classification is made because the type of equipment, fields of applications, and design techniques generally differ. All these classifications are summarized in previous figure and discussed further. Heat exchangers can also be classified according to the process function, as outlined in next figure.



(a) Classification according to process function; (b) classification of condensers; (c) classification of liquid-to-vapor phase-change exchangers.

Additional ways to classify heat exchangers are by fluid type (gas-gas, gas-liquid, liquid-liquid, gas two-phase, liquid two-phase, etc.), industry, and so on.

CLASSIFICATION ACCORDING TO TRANSFER PROCESSES

Heat exchangers are classified according to transfer processes into indirect- and direct-contact types.

Indirect-Contact Heat Exchangers

In an indirect-contact heat exchanger, the fluid streams remain separate and the heat transfers continuously through an impervious dividing wall or into and out of a wall in a transient manner. Thus, ideally, there is no direct contact between thermally interacting fluids. This type of heat exchanger, also referred to as a surface heat exchanger, can be further classified into direct-transfer type and storage type.

Direct-Transfer Type Exchangers. In this type, heat transfers continuously from the hot fluid to the cold fluid through a dividing wall. Although a simultaneous flow of two (or more) fluids is required in the exchanger, there is no direct mixing of the two (or more) fluids because each fluid flows in separate fluid passages. In general, there are no moving parts in most such heat exchangers. This type of exchanger is designated as a recuperative heat exchanger or simply as a recuperator. Some examples of direct-transfer type heat exchangers are tubular, plate-type, and extended surface exchangers. Recuperators are further subclassified as prime surface exchangers and extended-surface exchangers. Prime surface exchangers do not employ fins or extended surfaces on any fluid side. Plain tubular exchangers, shell-and-tube exchangers with plain tubes, and plate exchangers are good examples of prime surface exchangers. Recuperators constitute a vast majority of all heat exchangers.

Storage Type Exchangers. In a storage type exchanger, both fluids flow alternatively through the same flow passages, and hence heat transfer is intermittent. The heat transfer surface (or flow passages) is generally cellular in structure and is referred to as a matrix, or it is a permeable (porous) solid material, referred to as a packed bed. When hot gas flows over the heat transfer surface (through flow passages), the thermal energy from the hot gas is stored in the matrix wall, and thus the hot gas is being cooled during the matrix heating period. As cold gas flows through the same passages later (i.e., during the matrix cooling period), the matrix wall gives up thermal energy, which is absorbed by the cold fluid. Thus, heat is not transferred continuously through the wall as in a direct-transfer type exchanger (recuperator), but the corresponding thermal energy is alternately stored and released by the matrix wall. This storage type heat exchanger is also referred to as a regenerative heat exchanger, or simply as a regenerator. To operate continuously and within a desired temperature range, the gases, headers, or matrices are switched periodically (i.e., rotated), so that the same passage is occupied periodically by hot and cold gases. The actual time that hot gas takes to flow through a cold regenerator matrix is called the hot period or hot blow, and the time that cold gas flows through the hot regenerator matrix is called the cold period or cold blow. For successful operation, it is not necessary to have hot- and cold-gas flow periods of equal duration. There is some unavoidable carryover of a small fraction of the fluid trapped in the passage to the other fluid stream just after switching of the fluids; this is referred to as carryover leakage. In addition, if the hot and cold fluids are at different pressures, there will be leakage from the high-pressure fluid to the low-pressure fluid past the radial, peripheral, and axial seals, or across the valves. This leakage is referred to as pressure leakage. Since these leaks are unavoidable, regenerators are used exclusively in gas-to-gas heat (and mass) transfer applications with sensible heat transfer; in some applications, regenerators may transfer moisture from humid air to dry air up to about 5%.

Direct-Contact Heat Exchangers

In a direct-contact exchanger, two fluid streams come into direct contact, exchange heat, and then are or are not separated. Common applications of a direct-contact exchanger involve mass transfer in addition to heat transfer, such as in evaporative and cooling applications involving only sensible heat transfer are rare. The enthalpy of phase change in such an exchanger generally represents a significant portion of the total energy transfer. The phase change generally enhances the heat transfer rate. Compared to indirect-contact recuperators and regenerators, in direct-contact heat exchangers, (1) very high heat transfer rates are achievable, (2) the exchanger construction is relatively inexpensive, and (3) the fouling problem is generally nonexistent, due to the absence of a heat transfer surface (wall) between the two fluids. However, the applications are limited to those cases where a direct contact of two fluid streams is permissible. The design theory for these exchangers is beyond the scope of this book and is not covered. These exchangers may be further classified as follows.

Immiscible Fluid Exchangers. In this type, two immiscible fluid streams are brought into direct contact. These fluids may be single-phase fluids, or they may involve condensation or vaporization.

Gas-Liquid Exchangers. In this type, one fluid is a gas (more commonly, air) and the other a low-pressure liquid (more commonly, water) and are readily separable after the energy exchange. In either cooling of liquid (cooling towers) or cooling of gas (scrubbers) applications, liquid partially evaporates and the vapor is carried away with the gas. In these exchangers, more than 90% of the energy transfer is by virtue of mass transfer (due to the evaporation of the liquid), and convective heat transfer is a minor mechanism. A "wet" (water) cooling tower with forced- or natural-draft airflow is the most common application. Other applications are the air-conditioning spray chamber, spray drier, flue gas scruber and spray pond.

Liquid-Vapor Exchangers. In this type, typically stem is cooled by injected water, steam is partially or fully condensed using cooling water, or water is heated with waste steam through direct contact in the exchanger. Noncondensables and residual steam and hot water are the outlet streams. Common examples are spray coolers, desuperheaters and open feedwater heaters (also known as deaeraters) in power plants.

CLASSIFICATION ACCORDING TO CONSTRUCTION FEATURES

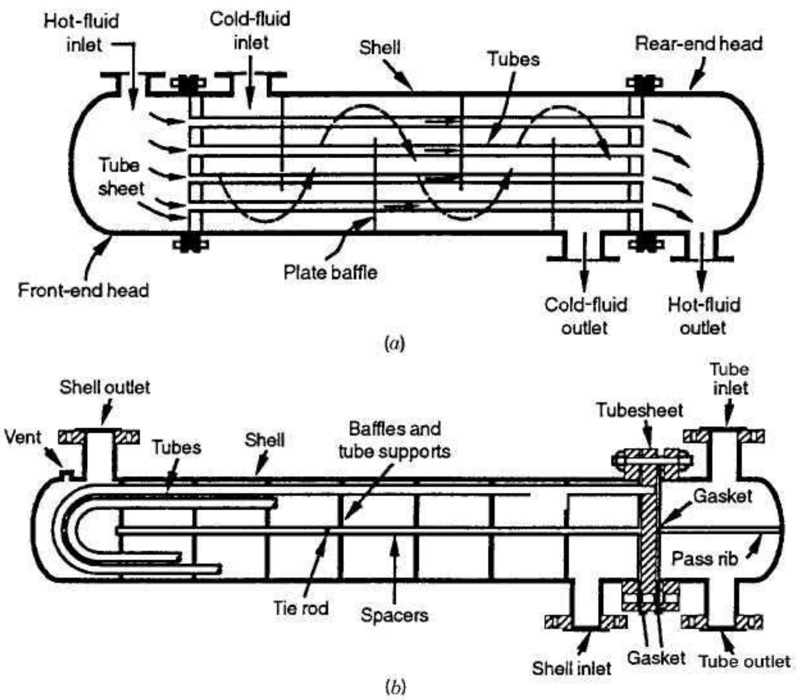
Heat exchangers are frequently characterized by construction features. Four major construction types are tubular, plate-type, extended surface, and regenerative exchangers.

Tubular Heat Exchangers

These exchangers are generally built of circular tubes, although elliptical, rectangular, or round/flat twisted tubes have also been used in some applications. There is considerable flexibility in the design because the core geometry can be varied easily by changing the tube diameter, length, and arrangement. Tubular exchangers can be designed for high pressures relative to the environment and high-pressure differences between the fluids. Tubular exchangers are used primarily for liquid-to-liquid and liquid-to-phase change (condensing or evaporating) heat transfer applications. They are used for gas-to-liquid and gas-to-gas heat transfer applications primarily when the operating temperature and/ or pressure is very high or fouling is a severe problem on at least one fluid side and no other types of exchangers would work. These exchangers may be classified as shell-and-tube, double-pipe, and spiral tube exchangers. They are all prime surface exchangers except for exchangers having fins outside/inside tubes.

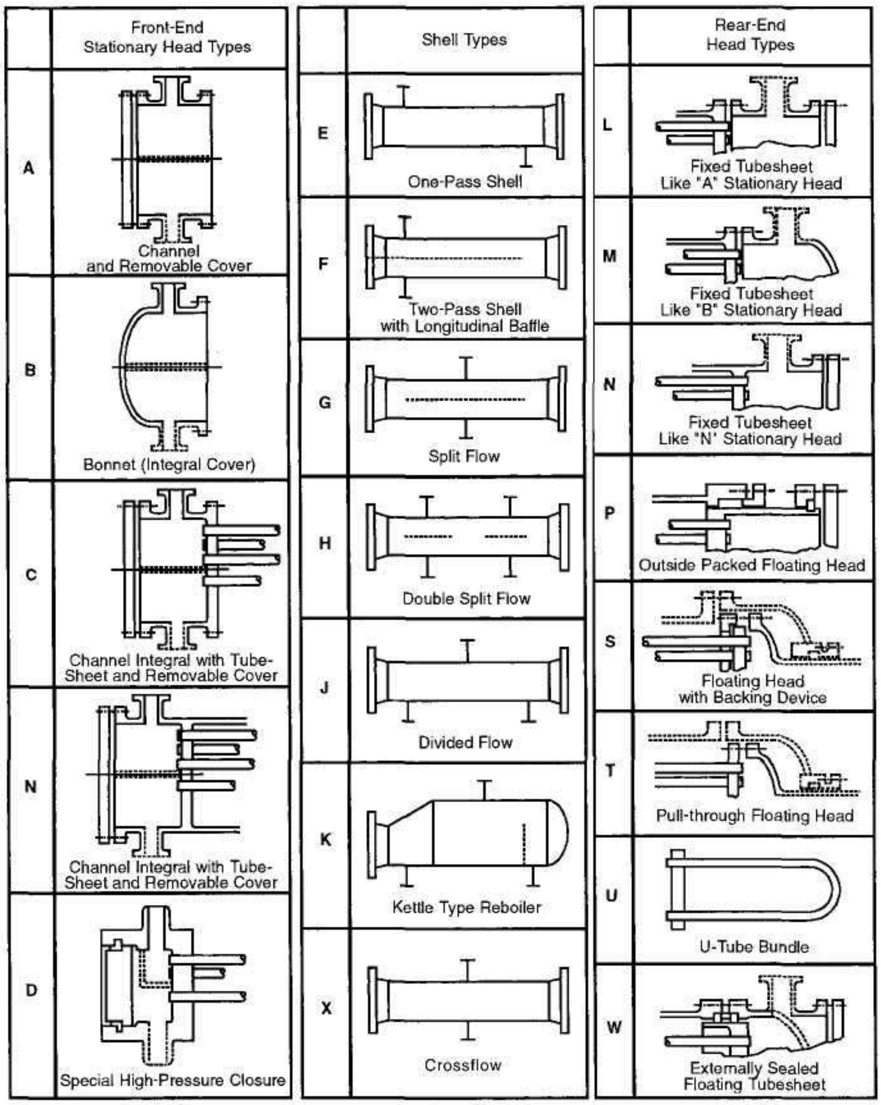
Shell-and-Tube Exchangers.

This exchanger is generally built of a bundle of round tubes mounted in a cylindrical shell with the tube axis parallel to that of the shell. One fluid flows inside the tubes, the other flows across and along the tubes. The major components of this exchanger are tubes (or tube bundle), shell, front-end head, rear-end head, baffles, and tubesheets, and are described briefly later in this subsection.



(a) Shell-and-tube exchanger (BEM) with one shell pass and one tube pass; (b) shell-and-tube exchanger (BEU) with one shell pass and two tube passes.

A variety of different internal constructions are used in shell-and-tube exchangers, depending on the desired heat transfer and pressure drop performance and the methods employed to reduce thermal stresses, to prevent leakages, to provide for ease of cleaning, to contain operating pressures and temperatures, to control corrosion, to accommodate highly asymmetric flows, and so on. Shell-and-tube exchangers are classified and constructed in accordance with the widely used TEMA (Tubular Exchanger Manufacturers Association) standards (TEMA, 1999), DIN and other standards in Europe and elsewhere, and ASME (American Society of Mechanical Engineers) boiler and pressure vessel codes. TEMA has developed a notation system to designate major types of shell-and-tube exchangers. In this system, each exchanger is designated by a three-letter combination, the first letter indicating the front-end head type, the second the shell type, and the third the rear-end head type. These are identified in next figure. Some common shell-and-tube exchangers are AES, BEM, AEP, CFU, AKT, and AJW.

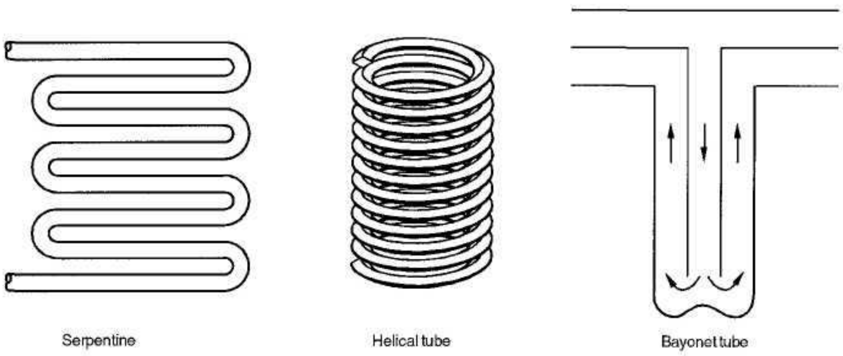


Standard shell types and front- and rear-end head types

The three most common types of shell-and-tube exchangers are (1) fixed tubesheet design, (2) U-tube design, and (3) floating-head type. In all three types, the front-end head is stationary while the rear-end head can be either stationary or floating (see Fig.), depending on the thermal stresses in the shell, tube, or tubesheet, due to temperature differences as a result of heat transfer.

Tubes. Round tubes in various shapes are used in shell-and-tube exchangers. Most common are the tube bundles with straight and U-tubes used in process and power industry exchangers. However, sine-wave bend, J-shape, L-shape or hockey sticks, and inverted hockey sticks are used in advanced nuclear exchangers to accommodate large thermal expansion of the tubes.

Some of the enhanced tube geometries used in shell-and-tube exchangers are shown in figure. Serpentine, helical, and bayonet are other tube shapes that are used in shell-and-tube exchangers.



Shells. The shell is a container for the shell fluid. Usually, it is cylindrical in shape with a circular cross section, although shells of different shapes are used in specific applications and in nuclear heat exchangers to conform to the tube bundle shape. The shell is made from a circular pipe if the shell diameter is less than about 0.6m (2 ft) and is made from a metal plate rolled and welded longitudinally for shell diameters greater than 0.6m (2ft).

Nozzles. The entrance and exit ports for the shell and tube fluids, referred to as nozzles, are pipes of constant cross section welded to the shell and channels. They are used to distribute or collect the fluid uniformly on the shell and tube sides. Note that they differ from the nozzle used as a fluid metering device or in jet engines, which has a variable flow area along the flow length.

Front- and Rear-End Heads. These are used for entrance and exit of the tube fluid; in many rear-end heads, a provision has been made to take care of tube thermal expansion. The front-end head is stationary, while the rear-end head could be either stationary (allowing for no tube thermal expansion) or floating, depending on the thermal stresses between the tubes and shell. The major criteria for selection of the front-end head are cost, maintenance and inspection, hazard due to mixing of shell and tube fluids, and leakage to ambient and operating pressures. The major criteria for selection of the rear-end head are the allowance for thermal stresses, a provision to remove the tube bundle for cleaning the shell side, prevention of mixing of tube and shell fluids, and sealing any leakage path for the shell fluid to ambient.

Baffles. Baffles may be classified as transverse and longitudinal types. The purpose of longitudinal baffles is to control the overall flow direction of the shell fluid such that a desired overall flow arrangement of the two fluid streams is achieved. For example, F, G, and H shells have longitudinal baffles. Transverse baffles may be classified as plate baffles and grid (rod, strip, and other axial-flow) baffles. Plate baffles are used to support the tubes during assembly and operation and to direct the fluid in the tube bundle approximately at right angles to the tubes to achieve higher heat transfer coefficients. Plate baffles increase the turbulence of the shell fluid and minimize tube-to-tube temperature differences and thermal stresses due to the crossflow. Shown in figure are single- and multisegmental baffles and disk and doughnut baffles. Single- and double-segmental baffles are used most frequently due to their ability to assist maximum heat transfer (due to a high-shell-side heat transfer coefficient) for a given pressure drop in a minimum amount of space. Triple and no-tubes-in-window segmental baffles are used for low-pressure-drop applications. The choice of baffle type, spacing, and cut is determined largely by flow rate, desired heat transfer rate, allowable pressure drop, tube support, and flow-induced vibrations. Disk and doughnut baffles/ support plates are used primarily in nuclear heat exchangers. These baffles for nuclear exchangers have small perforations between tube holes to allow a combination of crossflow and longitudinal flow for lower shell-side pressure drop. The combined flow results in a slightly higher heat transfer coefficient than that for pure longitudinal flow and minimizes tube-to-tube temperature differences.

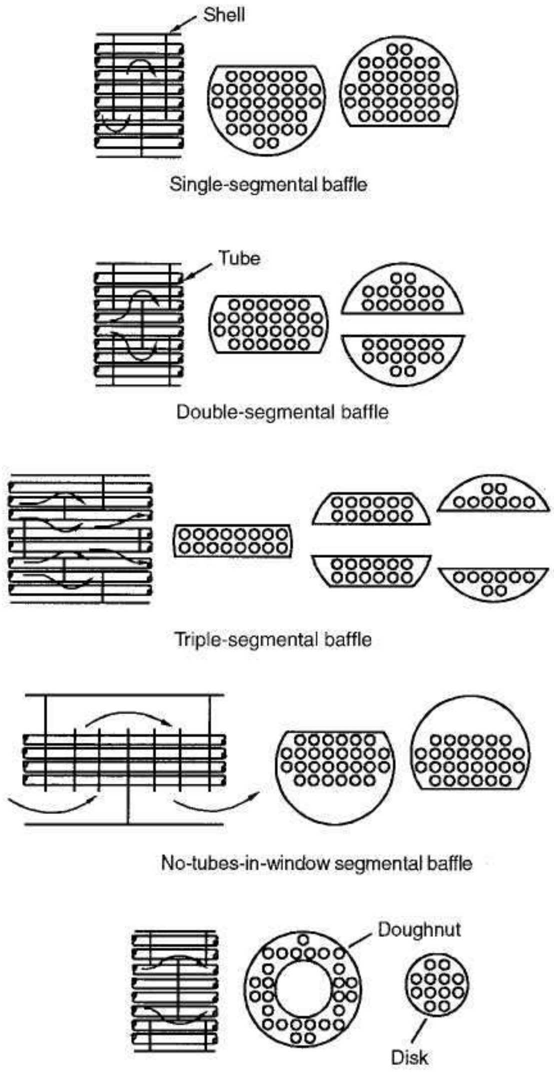
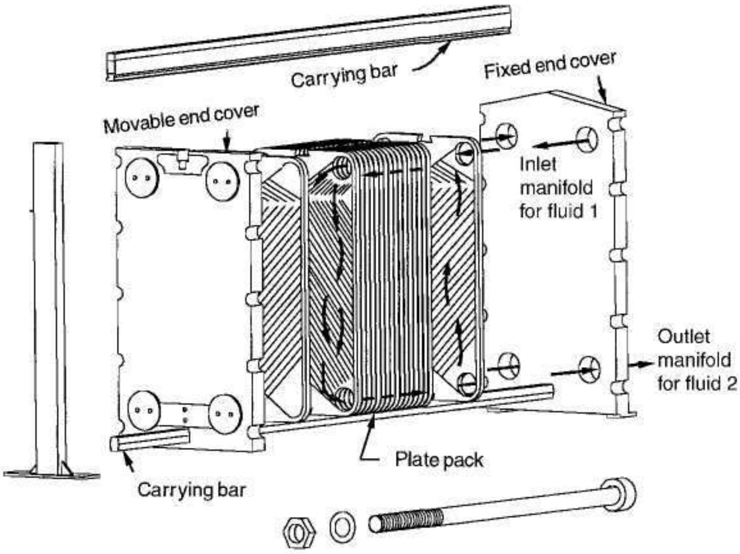


Plate-Type Heat Exchangers

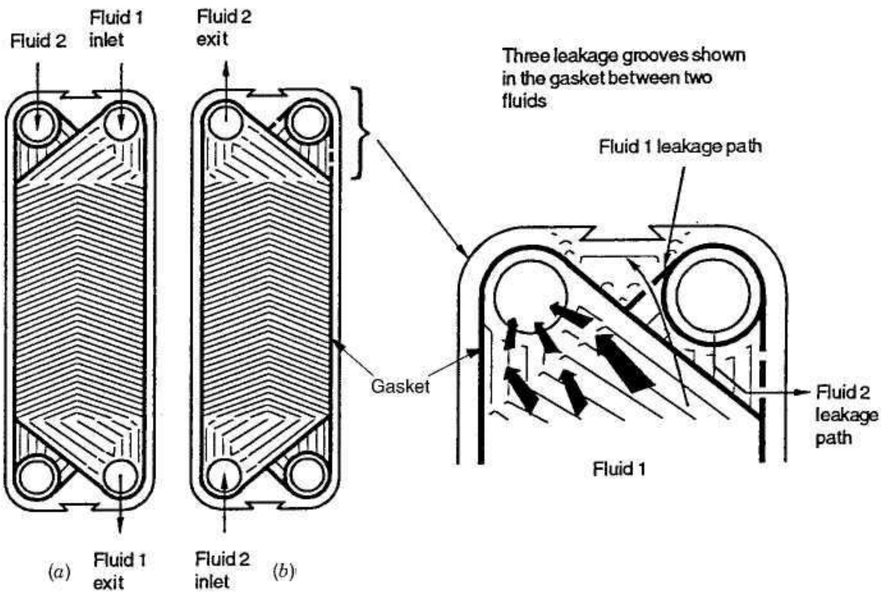
Plate-type heat exchangers are usually built of thin plates (all prime surface). The plates are either smooth or have some form of corrugation, and they are either flat or wound in an exchanger. Generally, these exchangers cannot accommodate very high pressures, temperatures, or pressure and temperature differences. Plate heat exchangers (PHEs) can be classified as gasketed, welded (one or both fluid passages), or brazed, depending on the leak tightness required. Other plate-type exchangers are spiral plate, lamella, and platecoil exchangers. These are described next.

Gasketed Plate Heat Exchangers

Basic Construction. The plate-and-frame or gasketed plate heat exchanger (PHE) consists of a number of thin rectangular metal plates sealed around the edges by gaskets and held together in a frame as shown in figure.



The frame usually has a fixed end cover (headpiece) fitted with connecting ports and a movable end cover (pressure plate, follower, or tailpiece). In the frame, the plates are suspended from an upper carrying bar and guided by a bottom carrying bar to ensure proper alignment. For this purpose, each plate is notched at the center of its top and bottom edges. The plate pack with fixed and movable end covers is clamped together by long bolts, thus compressing the gaskets and forming a seal. The carrying bars are longer than the compressed stack, so that when the movable end cover is removed, plates may be slid along the support bars for inspection and cleaning.
Each plate is made by stamping or embossing a corrugated (or wavy) surface pattern on sheet metal. On one side of each plate, special grooves are provided along the periphery of the plate and around the ports for a gasket, as indicated by the dark lines in next figure.



Typical plate geometries (corrugated patterns) are shown in next figure, and over 60 different patterns have been developed worldwide. Alternate plates are assembled such that the corrugations on successive plates contact or cross each other to provide mechanical support to the plate pack through a large number of contact points. The resulting flow passages are narrow, highly interrupted, and tortuous, and enhance the heat transfer rate and decrease fouling resistance by increasing the shear stress, producing secondary flow, and increasing the level of turbulence. The corrugations also improve the rigidity of the plates and form the desired plate spacing. Plates are designated as hard or soft, depending on whether they generate a high or low intensity of turbulence.

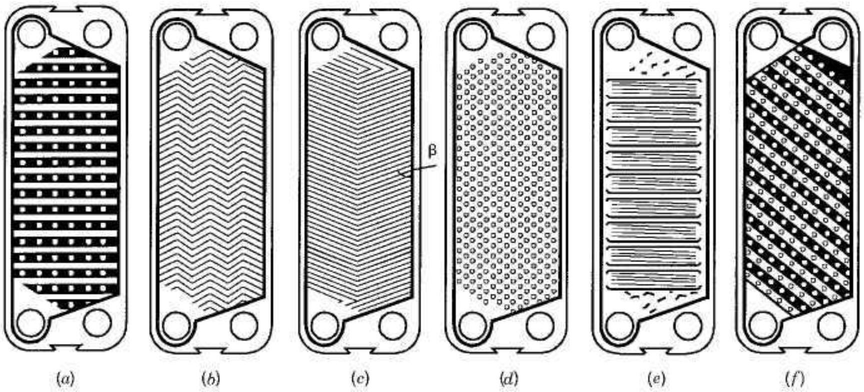


Plate patterns: (a) washboard; (b) zigzag; (c) chevron or herringbone; (d) protrusions and depressions; (e) washboard with secondary corrugations; (f) oblique washboard

Sealing between the two fluids is accomplished by elastomeric molded gaskets (typically, 5 mm thick) that are fitted in peripheral grooves mentioned earlier. Gaskets are designed such that they compress about 25% of thickness in a bolted plate exchanger to provide a leaktight joint without distorting the thin plates. In the past, the gaskets were cemented in the grooves, but now, snap-on gaskets, which do not require cementing, are common. Some manufacturers offer special interlocking types to prevent gasket blowout at high pressure differences. Use of a double seal around the port sections prevents fluid intermixing in the rare event of gasket failure. The interspace between the seals is also vented to the atmosphere to facilitate visual indication of leakage. Typical gasket materials are butyl and nitrile rubber. PTFE (polytetrafluoroethylene) is not used because of its viscoelastic properties.

Each plate has four corner ports. In pairs, they provide access to the flow passages on either side of the plate. When the plates are assembled, the corner ports line up to form distribution headers for the two fluids. Inlet and outlet nozzles for the fluids, provided in the end covers, line up with the ports in the plates (distribution headers) and are connected to external piping carrying the two fluids. A fluid enters at a corner of one end of the compressed stack of plates through the inlet nozzle. It passes through alternate channels in either series or parallel passages. In one set of channels, the gasket does not surround the inlet port between two plates; fluid enters through that port, flows between plates, and exits through a port at the other end. On the same side of the plates, the other two ports are blocked by a gasket with a double seal, so that the other fluid cannot enter the plate on that side. In a 1 pass-1 pass two-fluid counterflow PHE, the next channel has gaskets covering the ports just opposite the preceding plate. Incidentally, each plate has gaskets on only one side, and they sit in grooves on the back side of the neighboring plate. Each fluid makes a single pass through the exchanger because of alternate gasketed and ungasketed ports in each corner opening. The most conventional flow arrangement is 1 pass-1 pass counterflow, with all inlet and outlet connections on the fixed end cover. By blocking flow through some ports with proper gasketing, either one or both fluids could have more than one pass. Also, more than one exchanger can be accommodated in a single frame. In cases with more than two simple 1-pass-1-pass heat exchangers, it is necessary to insert one or more intermediate headers or connector plates in the plate pack at appropriate places.

Any metal that can be cold-worked is suitable for PHE applications. The most common plate materials are stainless steel (AISI 304 or 316) and titanium. Plates made from Incoloy 825, Inconel 625, and Hastelloy C-276 are also available. Nickel, cupro-nickel, and monel are rarely used. Carbon steel is not used, due to low corrosion resistance for thin plates. Graphite and polymer plates are used with corrosive fluids. The heat transfer surface area per unit volume for plate exchangers ranges from 120 to 660m^2/m^3.

Advantages and Limitations. Some advantages of plate heat exchangers are as follows. They can easily be taken apart into their individual components for cleaning, inspection, and maintenance. The heat transfer surface area can readily be changed or rearranged for a different task or for anticipated changing loads, through the flexibility of plate size, corrugation patterns, and pass arrangements. High shear rates and shear stresses, secondary flow, high turbulence, and mixing due to plate corrugation patterns reduce fouling to about 10 to 25% of that of a shell-and-tube exchanger, and enhance heat transfer. Very high heat transfer coefficients are achieved due to the breakup and reattachment of boundary layers, swirl or vortex flow generation, and small hydraulic diameter flow passages. Because of high heat transfer coefficients, reduced fouling, the absence of bypass and leakage streams, and pure counterflow arrangements, the surface area required for a plate exchanger is one-half to one-third that of a shell-and-tube exchanger for a given heat duty, thus reducing the cost, overall volume, and space requirement for the exchanger. Also, the gross weight of a plate exchanger is about one-sixth that of an equivalent shell-and-tube exchanger. Leakage from one fluid to the other cannot take place unless a plate develops a hole. Since the gasket is between the plates, any leakage from the gaskets is to the outside of the exchanger. The residence time (time to travel from the inlet to the outlet of the exchanger) for different fluid particles or flow paths on a given side is approximately the same. This parity is desirable for uniformity of heat treatment in applications such as sterilizing, pasteurizing, and cooking. There are no significant hot or cold spots in the exchanger that could lead to the deterioration of heat-sensitive fluids. The volume of fluid held up in the exchanger is small; this feature is important with expensive fluids, for faster transient response, and for better process control. Finally, high thermal performance can be achieved in plate exchangers. The high degree of counterflow in PHEs makes temperature approaches of up to 18C possible. The high thermal effectiveness (up to about 93%) facilitates economical low-grade heat recovery. The flow-induced vibrations, noise, thermal stresses, and entry impingement problems of shell-and-tube exchangers do not exist for plate heat exchangers.

Some inherent limitations of the plate heat exchangers are caused by plates and gaskets as follows. The plate exchanger is capable of handling up to a maximum pressure of about 3 MPa gauge but is usually operated below 1.0 MPa gauge. The gasket materials (except for the PTFE-coated type) restrict the use of PHEs in highly corrosive applications; they also limit the maximum operating temperature to 260°C but are usually operated below 150°C to avoid the use of expensive gasket materials. Gasket life is sometimes limited. Frequent gasket replacement may be needed in some applications. Pinhole leaks are hard to detect. For equivalent flow velocities, pressure drop in a plate exchanger is very high compared to that of a shell-and-tube exchanger. However, the flow velocities are usually low and plate lengths are "short," so the resulting pressure drops are generally acceptable. The normal symmetry of PHEs may make phase-change applications more difficult, due to large differences in volumetric flows. For some cases, heat exchanger duties with widely different fluid flowrates and depending on the allowed pressure drops of the two fluids, an arrangement of a different number of passes for the two fluids may make a PHE advantageous. However, care must be exercised to take full advantage of available pressure drop while multi-passing one or both fluids.

Because of the long gasket periphery, PHEs are not suited for high-vacuum applications. PHEs are not suitable for erosive duties or for fluids containing fibrous materials. In certain cases, suspensions can be handled; but to avoid clogging, the largest suspended particle should be at most one-third the size of the average channel gap. Viscous fluids can be handled, but extremely viscous fluids lead to flow maldistribution problems, especially on cooling. Plate exchangers should not be used for toxic fluids, due to potential gasket leakage. Some of the limitations of gasketed PHEs have been addressed by the new designs of PHEs.

Extended Surface Heat Exchangers

The tubular and plate-type exchangers described previously are all prime surface heat exchangers, except for a shell-and-tube exchanger with low finned tubing. Their heat exchanger effectiveness is usually 60% or below, and the heat transfer surface area density is usually less than $700 m^2/m^3$. In some applications, much higher exchanger effectiveness is essential, and the box volume and mass are limited so that a much more compact surface is mandated. Also, in a heat exchanger with gases or some liquids, the heat transfer coefficient is quite low on one or both fluid sides. This results in a large heat transfer surface area requirement. One of the most common methods to increase the surface area and exchanger compactness is to add the extended surface (fins) and use fins with the fin density (fin frequency or fins/m) as high as possible on one or both fluid sides, depending on the design requirement. Addition of fins can increase the surface area by 5 to 12 times the primary surface area in general, depending on the design. The resulting exchanger is referred to as an extended surface exchanger. Flow area is increased by the use of thin-gauge material and sizing the core properly. The heat transfer coefficient on extended surfaces may be higher or lower than that on unfinned surfaces. For example, interrupted (strip, louver, etc.) fins provide both an increased area and increased heat transfer coefficient, while internal fins in a tube increase the tube-side surface area but may result in a slight reduction in the heat transfer coefficient, depending on the fin spacing. Generally, increasing the fin density reduces the heat transfer coefficient associated with fins. Flow interruptions (as in offset strip fins, louvered fins, etc.) may increase the heat transfer coefficient two to four times that for the corresponding plain (uncut) fin surface. Plate-fin and tube-fin geometries are the two most common types of extended surface heat exchangers.

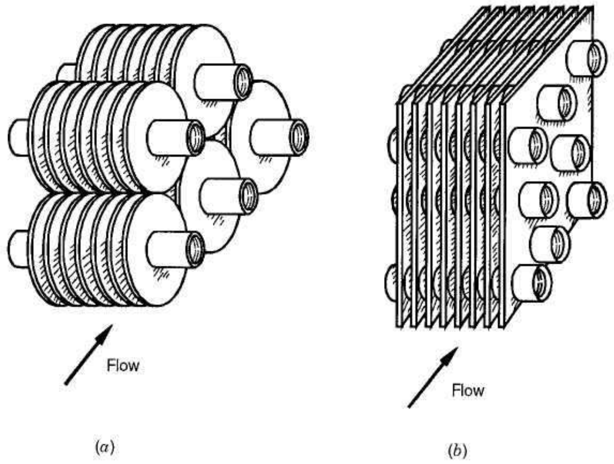
Tube-Fin Heat Exchangers

These exchangers may be classified as conventional and specialized tube-fin exchangers. In a conventional tube-fin exchanger, heat transfer between the two fluids takes place by conduction through the tube wall. However, in a heat pipe exchanger (a specialized type of tube-fin exchanger), tubes with both ends closed act as a separating wall, and heat transfer between the two fluids takes place through this ‘‘separating wall’’ (heat pipe) by conduction, and evaporation and condensation of the heat pipe fluid. Let us first describe conventional tube-fin exchangers and then heat pipe exchangers.

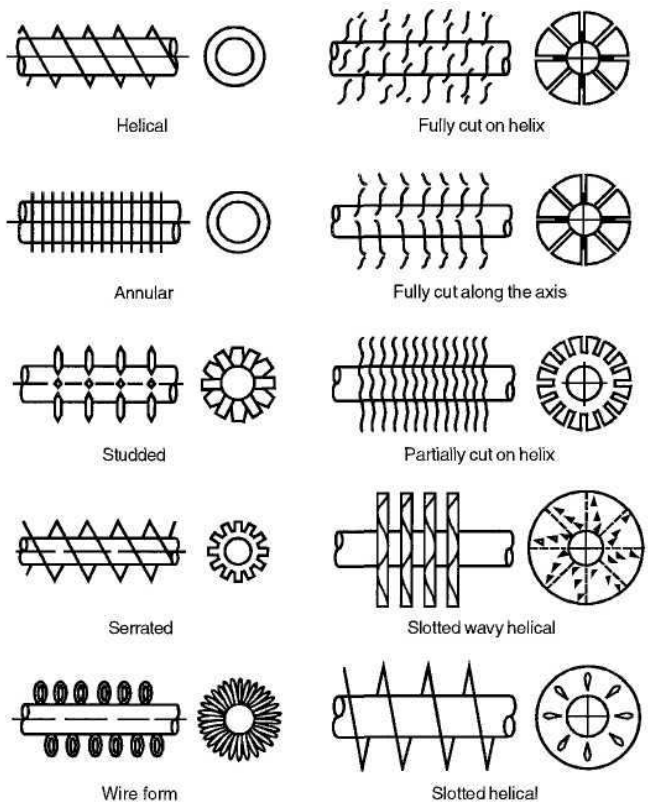
Conventional Tube-Fin Exchangers. In a gas-to-liquid exchanger, the heat transfer coefficient on the liquid side is generally one order of magnitude higher than that on the gas side. Hence, to have balanced thermal conductances (approximately the same hA) on both sides for a minimum-size heat exchanger, fins are used on the gas side to increase surface area A. This is similar to the case of a condensing or evaporating fluid stream on one side and gas on the other. In addition, if the pressure is high for one fluid, it is generally economical to employ tubes.

In a tube-fin exchanger, round and rectangular tubes are most common, although elliptical tubes are also used. Fins

are generally used on the outside, but they may be used on the inside of the tubes in some applications. They are attached to the tubes by a tight mechanical fit, tension winding, adhesive bonding, soldering, brazing, welding, or extrusion.
Depending on the fin type, tube-fin exchangers are categorized as follows: (1) an individually finned tube exchanger or simply a finned tube exchanger, as shown in following figures, having normal fins on individual tubes; (2) a tube-fin exchanger having flat (continuous) fins, the fins can be plain, wavy, or interrupted, and the array of tubes can have tubes of circular, oval, rectangular, or other shapes; and (3) longitudinal fins on individual tubes.



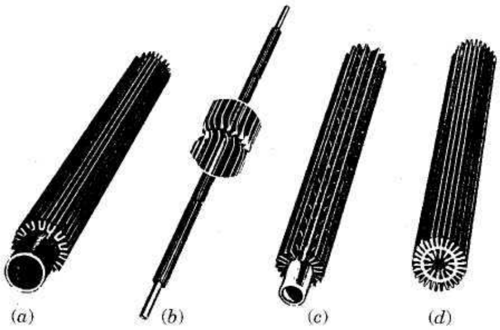
(a) Individually finned tubes; (b) flat (continuous) fins on an array of tubes. The flat fins are shown as plain fins, but they can be wavy, louvered, or interrupted.



Individually finned tubes

A tube-fin exchanger with flat fins has been referred to variously as a plate-fin and tube, plate finned tube, and tube in-plate fin exchanger in the literature.
A tube-fin exchanger of the aforementioned categories 1 and 2 is referred to as a coil in the air-conditioning and refrigeration industries and has air outside and a refrigerant inside the tube. Individually finned tubes are probably more rugged and practical in large tube-fin exchangers. The exchanger with flat fins is usually less expensive on a unit heat transfer surface area basis because of its simple and mass-production construction features. Longitudinal

fins are generally used in condensing applications and for viscous fluids in double-pipe heat exchangers.
Tube-fin exchangers can withstand ultrahigh pressures on the tube side. The highest temperature is again limited by the type of bonding, materials employed, and material thickness. Tube-fin exchangers usually are less compact than plate-fin units. Tube-fin exchangers with an area density of about 3300 m^2/m^3 are available commercially. On the fin side, the surface area desired can be achieved through the proper fin density and fin geometry. Typical fin densities for flat fins vary from 250 to 800 fins/m, fin thicknesses vary from 0.08 to 0.25 mm, and fin flow lengths vary from 25 to 250 mm. A tube-fin exchanger having flat fins with 400 fins/m has a surface area density of about 720m^2/m^3.
Tube-fin exchangers are employed when one fluid stream is at a higher pressure and/ or has a significantly higher heat transfer coefficient than that of the other fluid stream. As a result, these exchangers are used extensively as condensers and evaporators in air-conditioning and refrigeration applications, as condensers in electric power plants, as oil coolers in propulsive power plants, and as air-cooled exchangers (also referred to asfin-fan exchangers) in process and power industries.

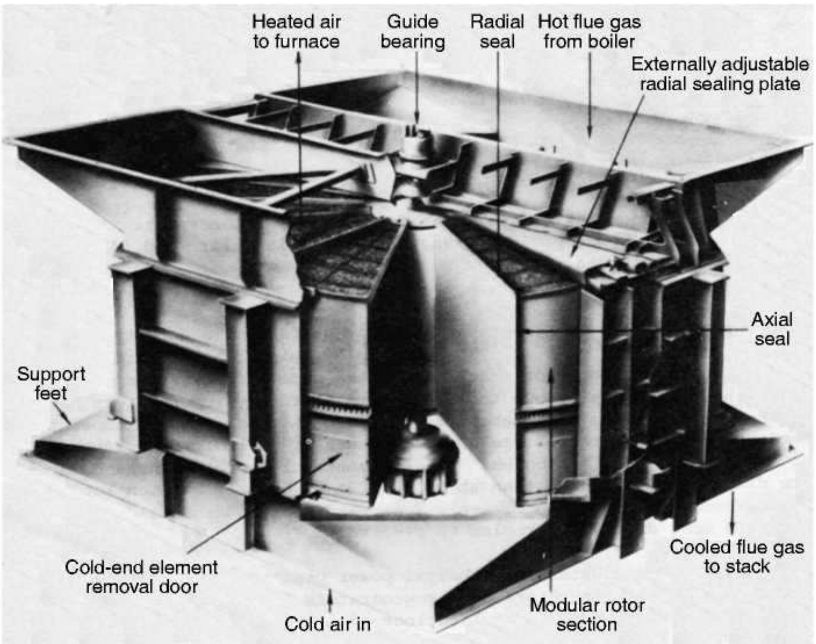


Longitudinal fins on individual tubes: (a) continuous plain; (b) cut and twisted; (c) perforated; (d) internal and external longitudinal fins.

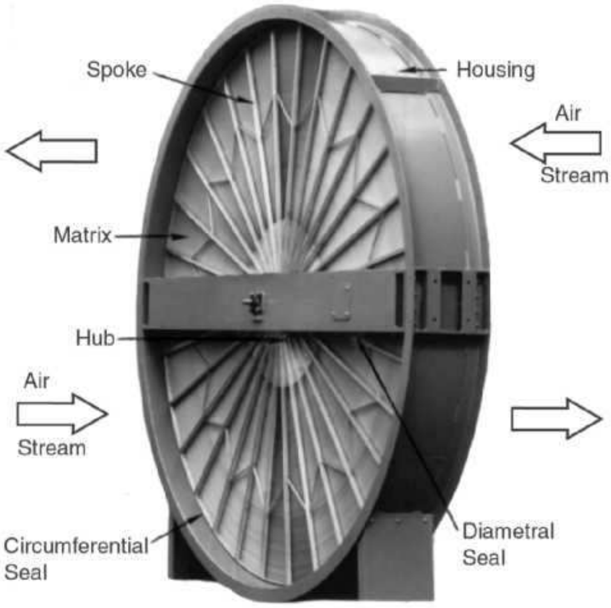
An air-cooled exchanger is a tube-fin exchanger in which hot process fluids (usually liquids or condensing fluids) flow inside the tubes, and atmospheric air is circulated outside by forced or induced draft over the extended surface. If used in a cooling tower with the process fluid as water, it is referred to as a dry cooling tower. Characteristics of this type of exchanger are shallow tube bundles (short airflow length) and large face area, due to the design constraint on the fan power.

Regenerators

The regenerator is a storage-type heat exchanger, as described earlier. The heat transfer surface or elements are usually referred to as a matrix in the regenerator. To have continuous operation, the matrix must be moved periodically into and out of the fixed streams of gases, as in a rotary regenerator.



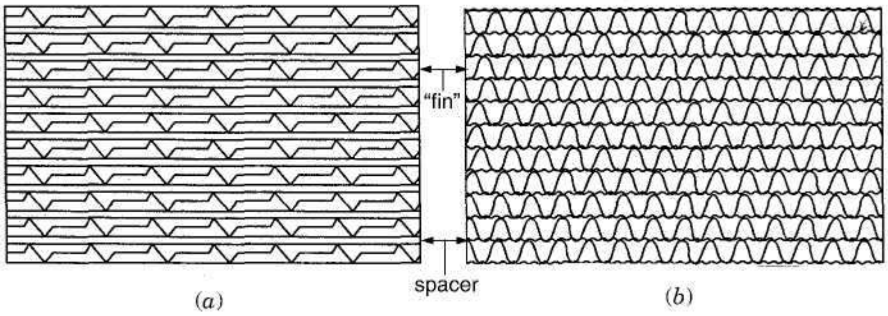
Ljungstrom air preheater



Heat wheel or a rotary regenerator made from a polyester film.

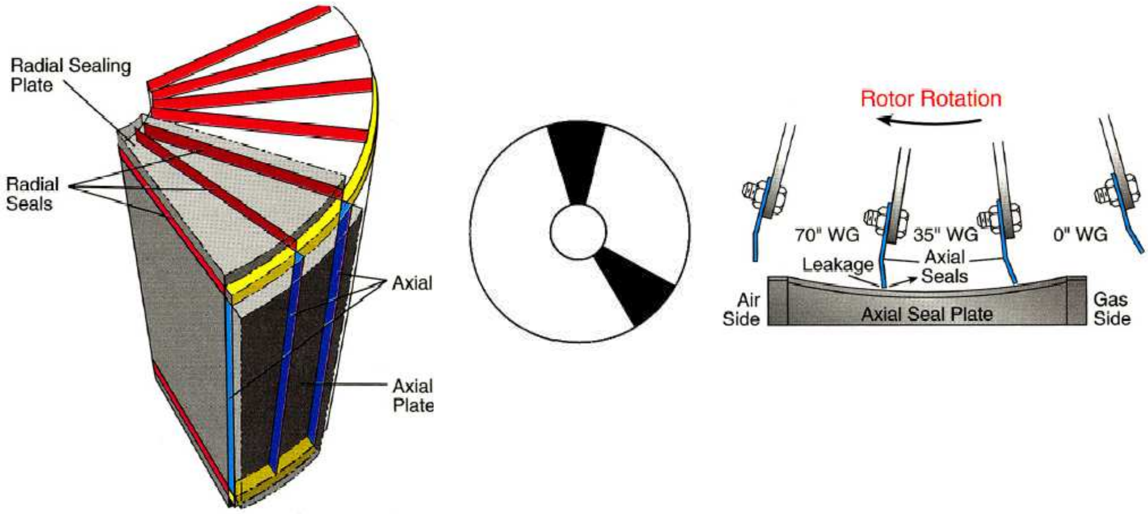
Thus, in a rotary regenerator, the matrix (disk or rotor) rotates continuously with a constant fraction of the core in the hot-fluid stream and the remaining fraction in the cold-fluid stream; the outlet fluid temperatures vary across the flow area and are independent of time. The two fluids generally flow in the opposite directions and are separated by some form of ductwork and rubbing seals on the matrix.
For a rotary regenerator, the design of seals to prevent leakages of hot to cold fluids, and vice versa, becomes a difficult task, especially if the two fluids are at significantly different pressures. Rotating drives also pose a challenging mechanical design problem.
Major advantages of the regenerators are the following. A much more compact surface may be employed than in a recuperator, thus providing a reduced exchanger volume for given exchanger effectiveness and pressure drop and thereby making a regenerator economical compared to an equivalent recuperator. The major reason for having a much more compact surface for a regenerator is that the hot and cold gas streams are separated by radial seals or valves, unlike in a recuperator, where the primary surface is used to separate the fluid streams. The cost of manufacturing such a compact regenerator surface per unit of heat transfer area is usually substantially lower than that for the equivalent recuperator. Similarly, material cost could be lower in a regenerator than in a recuperator. Hence, a compact regenerator usually has a smaller volume and is lower in weight than an equivalent recuperator. However, the leakproof core required in a recuperator is not essential in a regenerator, due to the mode of operation. Another important advantage of a counterflow regenerator over a counterflow recuperator is that the design of inlet and outlet headers used to distribute hot and cold gases in the matrix is simple. This is because both fluids flow in different sections (separated by radial seals) of a rotary regenerator. In contrast, the header design to separate two fluids at the inlet and outlet in a counterflow recuperator is complex and costly. Also, in a rotary regenerator, the flow sectors for the hot and cold gases can be designed to optimize the pressure drop on the hot and cold gases; and the critical pressure drop (usually on the hot side) in a rotary regenerator is lower than that in a comparable recuperator. The matrix surface has self-cleaning characteristics, resulting in low gas-side fouling and associated corrosion, if any, because the hot and cold gases flow alternately in opposite directions in the same fluid passage. Hence, regenerators are used with particulate-laden gases that promote surface fouling in a recuperator. Compact surface area density and the counterflow arrangement make the regenerator ideally suited for gas-to-gas heat exchanger applications requiring high exchanger effectiveness, generally exceeding 85%. Regenerators are used exclusively for gas-to-gas heat and/or energy transfer applications, primarily for combustion air preheat in pulverized coal fired boilers and for waste heat recovery applications.
A major disadvantage of a rotary regenerator is that an unavoidable carryover of a small fraction of one fluid stream trapped in the flow passages under the radial seal is pushed out by the other fluid stream just after the periodic flow switching. Other disadvantages are listed separately in the following part.
In rotary regenerators, any of the plain plate-fin surface geometries could be used in the matrix made up of thin metal sheets. Interrupted passage surfaces (such as strip fins, louver fins) are not used because a transverse (to the main flow direction) flow leakage is present if the two fluids are at different pressures. This leak mixes the two fluids (contaminates the lower pressure fluid) and reduces the exchanger effectiveness. Hence, the matrix generally has continuous (uninterrupted) flow passages. Flat or wavy spacers are used to stack the "fins" (see figure). The fluid is unmixed at any cross section for these surfaces. Two examples of rotary regenerator surfaces

are shown following figure.



Continuous-passage matrices for a rotary regenerator: (a) notched plate; (b) triangular passage.

The herringbone or skewed passage matrix does not require spacers for stacking the ''fins''. The design Reynolds number range with these types of surfaces is 100 to 1000.
The matrix in the regenerator is rotated by a hub shaft or a peripheral ring gear drive. Every matrix element is passed periodically from the hot to the cold stream and back again. The time required for a complete rotation of the matrix is equivalent to the total period of a fixed-matrix regenerator. In a rotary regenerator, the stationary radial seal locations control the desired frontal areas for each fluid and also serve to minimize the primary leakage from the high-pressure fluid to the low-pressure fluid.
Radial and axial seals are used in rotary regenerators - see following figure.

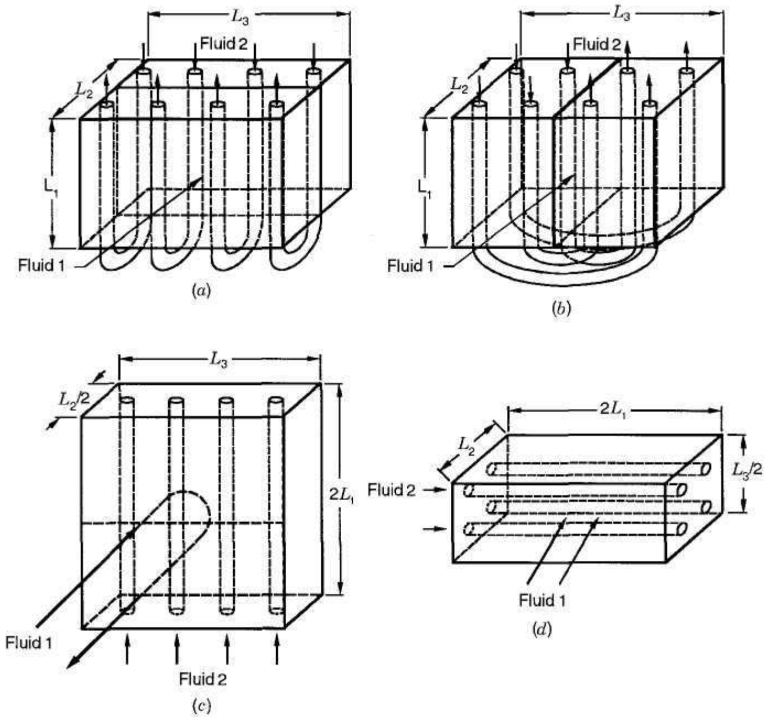


Rotary regenerators have been designed for surface area densities of up to about 6600 m^2/m^3. They can employ thinner stock material, resulting in the lowest amount of material for a given effectiveness and pressure drop of any heat exchanger known today. Metal rotary regenerators have been designed for continuous operating inlet temperatures up to about 790°C. Seal leakage can reduce the regenerator effectiveness significantly. Rotary regenerators also require a power input to rotate the core from one fluid to the other at the rotational speed desired. Ljungstrom air preheaters for thermal power plants and regenerators for the vehicular gas turbine power plants are typical examples of metal rotary regenerators for preheating inlet air. Typical power plant regenerators have a rotor diameter up to 16 m and rotational speeds in the range 0.5 to 3 rpm (rev per min).

CLASSIFICATION ACCORDING TO FLOW ARRANGEMENTS

The choice of a particular flow arrangement is dependent on the required exchanger effectiveness, available pressure drops, minimum and maximum velocities allowed, fluid flow paths, packaging envelope, allowable thermal stresses, temperature levels, piping and plumbing considerations, and other design criteria. Let us first discuss the concept of multipassing, followed by some of the basic ideal flow arrangements for a two-fluid heat exchanger for single- and multipass heat exchangers.
Multipassing. The concept of multipassing applies separately to the fluid and heat exchanger. A fluid is considered to have made one pass if it flows through a section of the heat exchanger through its full length. After flowing through one full length, if the flow direction is reversed and fluid flows through an equal- or different-sized section, it is considered to have made a second pass of equal or different size. A heat exchanger is

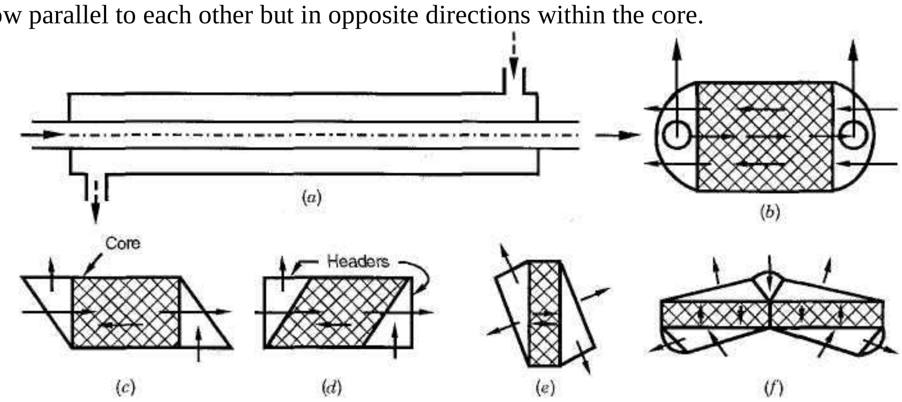
considered as a single-pass unit if both fluids make one pass in the exchanger or if it represents any of the single-pass flow arrangements when the multipass fluid side is unfolded (note that the folding is used to control the envelope size). To illustrate the concept, consider one exchanger with two different designs of inlet headers for fluid 2 as shown in figure a and b; fluid 1 makes a single pass, and fluid 2 makes two passes in both exchangers. If the exchanger b with fluid 2 unmixed in the headers is unfolded to the horizontal direction (the exchanger length for fluid 2 will be $2L_1$), as in figure d, the resulting geometry is a single-pass exchanger having the same inlet temperatures as fluids 1 and 2 of figure b. Hence, the exchanger of figure b is considered a single-pass exchanger from the exchanger analysis point of view. In contrast, the temperature of fluid 1 at the inlet to the first and second pass of fluid 2 is different in figure a. Hence, when it is unfolded vertically as in figure c, the inlet temperature of fluid 1 to each half of the exchanger will be different, due to the presence of two passes, each with one-half of the original flow length L_2. This does not correspond to a single-pass exchanger of the unfolded exchanger height. Therefore, the exchanger of figure a is considered as a two-pass exchanger. An additional degree of freedom is introduced by unfolding. This degree of freedom describes how to lead a fluid between the passes (see, e.g., the case considered in figure c, fluid 1). Depending on how the fluid is routed from the exit of one pass to the inlet of the following pass, several distinct flow arrangements can be identified.



(a) Two-pass cross-counterflow exchanger; (b) single-pass crossflow exchanger; (c, d) unfolded exchangers of (a) and (b), respectively.

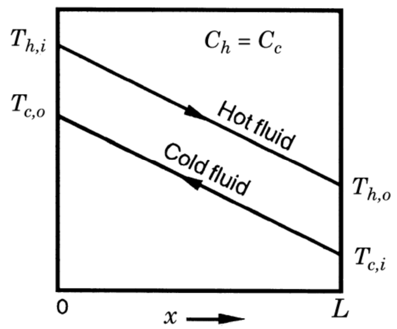
Single-Pass Exchangers

Counterflow Exchanger. In a counterflow or countercurrent exchanger, as shown in following figure a, the two fluids flow parallel to each other but in opposite directions within the core.



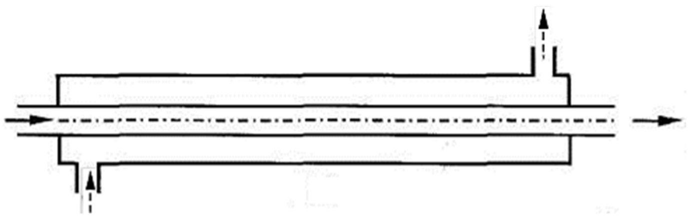
(a) Double-pipe heat exchanger with pure counterflow; (b-f) plate-fin exchangers with counterflow core and crossflow headers

The counterflow arrangement is thermodynamically superior to any other flow arrangement. It is the most efficient flow arrangement, producing the highest temperature change in each fluid compared to any other two-fluid flow arrangements for a given overall thermal conductance, fluid flow rates (actually, fluid heat capacity rates), and fluid inlet temperatures. Moreover, the maximum temperature difference across the exchanger wall thickness (between the wall surfaces exposed on the hot and cold fluid sides) either at the hot- or cold-fluid end is the lowest, and produce minimum thermal stresses in the wall for an equivalent performance compared to any other flow arrangements. However, with plate-fin heat exchanger surfaces, there are manufacturing difficulties associated with the true counterflow arrangement. This is because it is necessary to separate the fluids at each end, and the problem of inlet and outlet header design is complex. Some header arrangements were shown figure b-f. Also, the overriding importance of other design factors causes most commercial heat exchangers to be designed for flow arrangements different from single-pass counterflow if extremely high exchanger effectiveness is not required.discussion since they do not take an active part in heat transfer between two fluids.
The temperature variation of the two fluids in pure counterflow exchanger may be idealized as one-dimensional, as shown in figure.

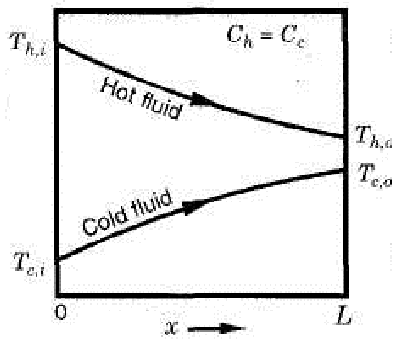


Temperature distributions in a counterflow heat exchanger

Parallelflow Exchanger.

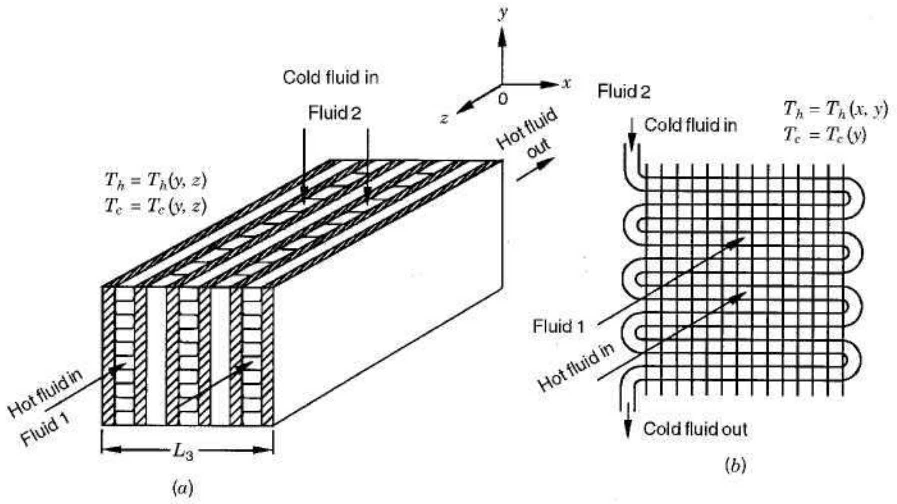


In a parallelflow (also referred to as cocurrent or cocurrent parallel stream) exchanger, the fluid streams enter together at one end, flow parallel to each other in the same direction, and leave together at the other end. Fluid temperature variations, idealized as one-dimensional, are shown in following figure.

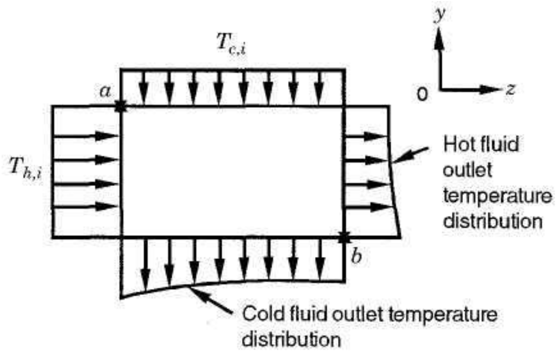


This arrangement has the lowest exchanger effectiveness among single-pass exchangers for given overall thermal conductance and fluid flow rates (actually, fluid heat capacity rates) and fluid inlet temperatures; however, some multipass exchangers may have an even lower effectiveness, as discussed later. However, for low-effectiveness exchangers, the difference in parallelflow and counterflow exchanger effectiveness is small. In a parallelflow exchanger, a large temperature difference between inlet temperatures of hot and cold fluids exists at the inlet side, which may induce high thermal stresses in the exchanger wall at the inlet.

Crossflow Exchanger. In this type of exchanger, as shown in figure, the two fluids flow in directions normal to each other.



Typical fluid temperature variations are idealized as two-dimensional and are shown in following figure for the inlet and outlet sections only.



Thermodynamically, the effectiveness for the crossflow exchanger falls in between that for the counterflow and parallelflow arrangements. The largest structural temperature difference exists at the "corner" of the entering hot and cold fluids, such as point a figure.