A

Seminar report

on

Cryogenic Heat Exchangers

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Cryogenic Heat Exchangers

One of the critical components of many cryogenic systems, such as liquefiers and cryocoolers, is the heat exchanger. In many conventional systems, such as regenerative gas turbine power plants, the system will operate even if the heat exchanger is not highly effective (e.g., less than 5%). In contrast, a cryogenic liquefier will produce no liquid if the heat exchanger effectiveness is less than approximately 85% (Barron 1985).

Some of the design principles for several types of heat exchangers that are commonly used for cryogenic service are discussed below. These heat exchangers include the Giauque–Hampson exchanger, the plate-fin exchanger, and the perforated plate exchanger. Secondary effects, including longitudinal conduction and variable specific heat, will also be considered because they may become of primary importance in high-performance cryogenic heat exchangers. The storage-type heat exchanger or *regenerator* is extensively used in cryogenic systems because of some of its beneficial characteristics.

Cryogenic Heat Exchanger Types

There are many different configurations for non-storage cryogenic heat exchangers or recuperator, which can be generally classified as tubular heat exchangers, plate-fin heat exchangers, and perforated plate heat exchangers. The characteristics of each of these types will be considered in this section.

1] Tubular Exchangers

The simplest tubular exchanger is the concentric-tube or double-pipe heat exchanger (Figure 1.1).This type of heat exchanger was used by Linde in his original air liquefier (Linde 1885). The heat exchanger consists of a small inner tube, in which the high-pressure stream usually flows, coaxial with a larger tube, with the low-pressure stream flowing in the annular space between the two tubes. The unit is usually wound in a helical spiral to conserve space, and the coil is often placed within an evacuated container to reduce heat transfer from ambient.

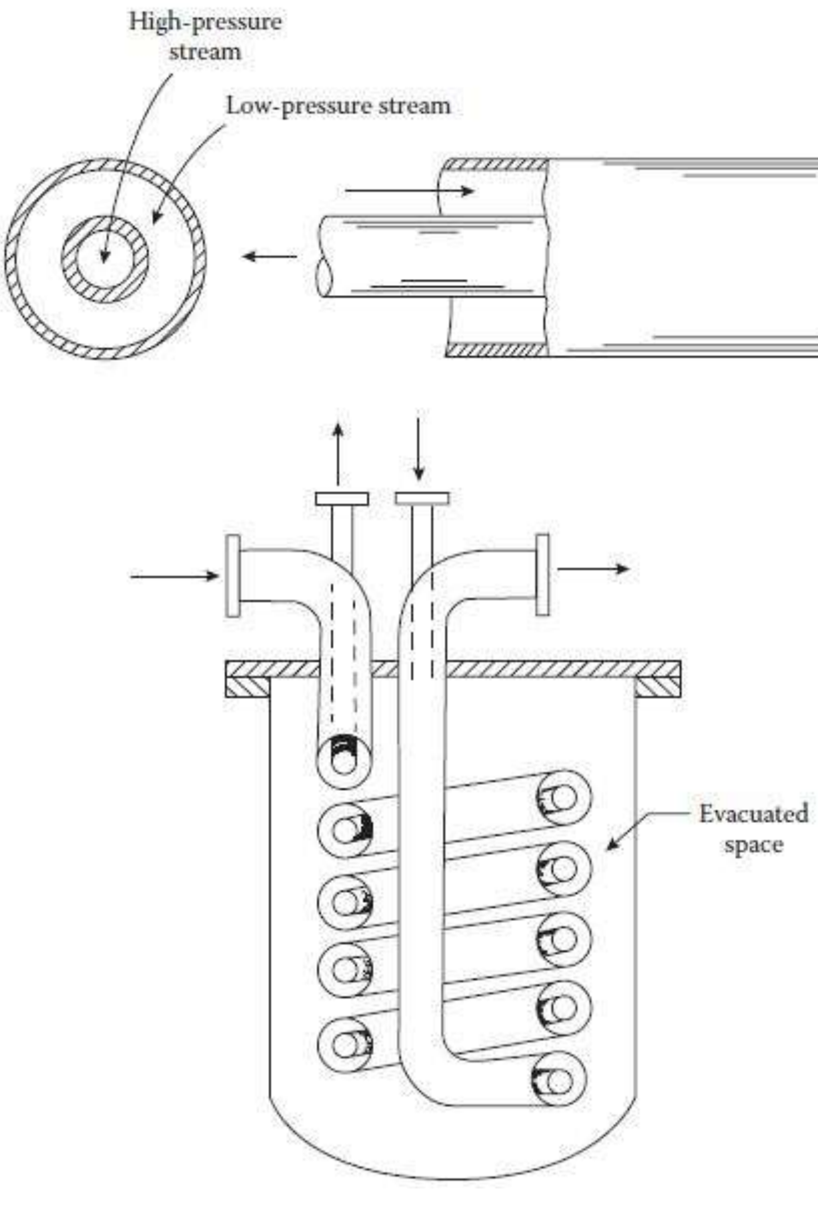


Fig. 1.1: Concentric tube heat exchanger

A wire or plastic spacer may be spirally wound on the outside of the inner tube of the concentric-tube heat exchanger (Figure 1.2). The use of the spacer allows the inner pipe to be fixed within the outer pipe, and also causes the fluid in the annular space to follow a longer helical path. This configuration results in an increase in the fluid velocity and an increase in the heat transfer coefficient, in comparison with the basic concentric tube arrangement. The use of the spiral spacer also results in an increase in the fluid pressure drop. This tube unit is usually wound in a helical spiral to conserve space.

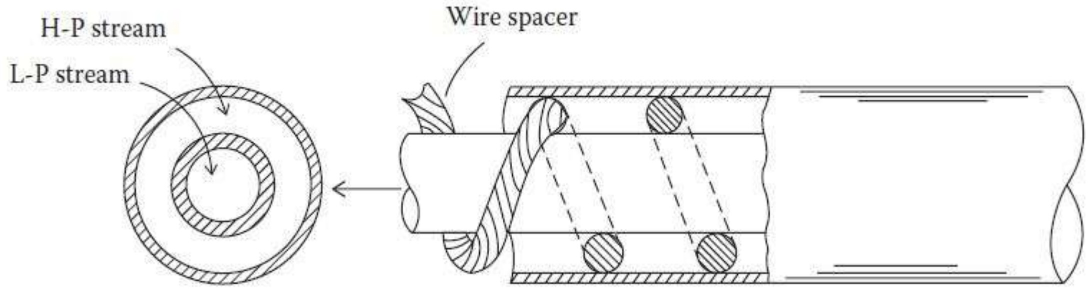


Fig. 1.2: Concentric-tube heat exchanger with wire spacer

In some cryogenic systems, a need exists to exchange heat between three or more streams. One simple heat exchanger for this application is the multiple-tube exchanger (Figure 1.3). The high-pressure stream flows in one of the small tubes, the intermediate-pressure stream flows in the other two small tubes, and the low-pressure stream flows in the space between the small tubes and the large tube. Elements such as spacers, spiral windings, and radial fins have been used to support the small tubes within the large tube.

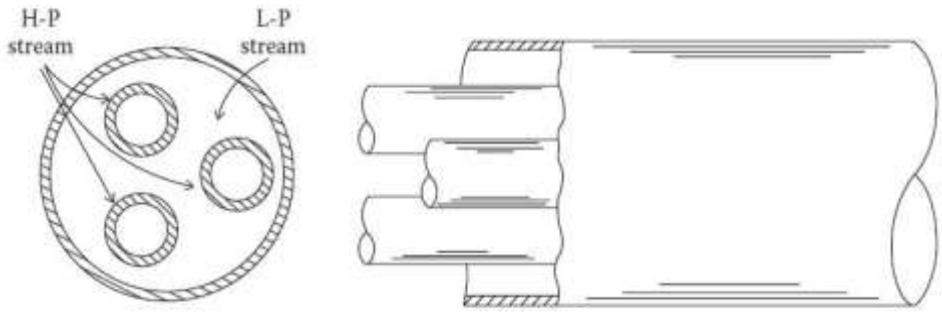


Fig. 1.3: Multiple-tube heat exchanger

A unique extended-surface heat exchanger design involving flow in concentric tubes (Figure 1.4) was developed by Prof. Sam Collin for the Collins helium liquefier (Scott 1959).The exchanger consisted of several concentric copper tubes with an edge-wound copper ribbon helix wrapped in the annular spaces between the tubes. The helix was soft soldered to both sides of the annular space in which it was wound. The ribbon helix acted as a fin to increase the heat transfer surface area of the exchanger.

These heat exchangers have been manufactured with as many as 10 separate flow passages.

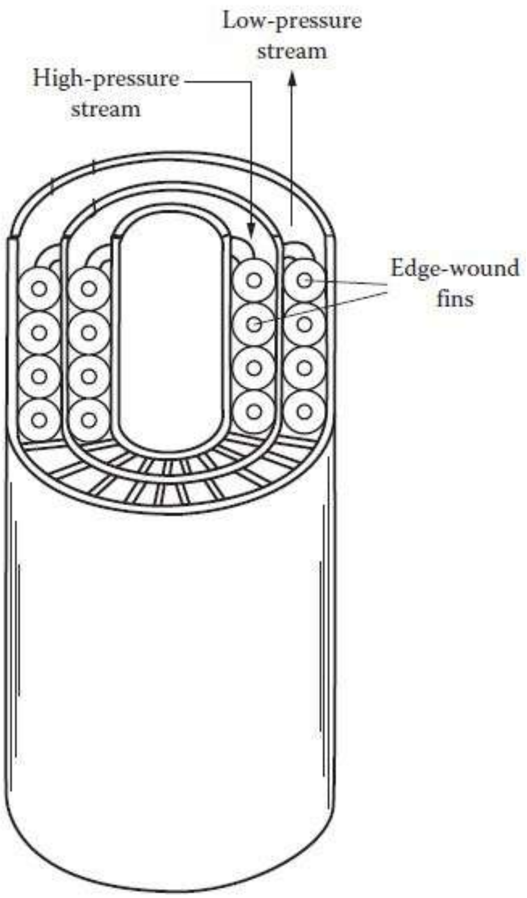
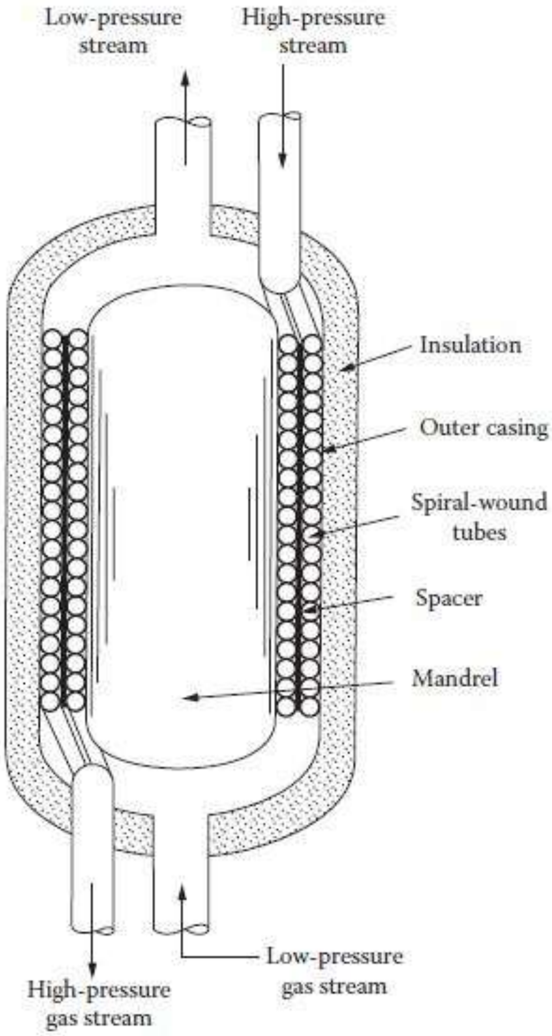


Fig. 1.4: The Collins heat exchanger

Giauque–Hampson Exchanger

The classic heat exchanger used in large-scale air and LNG liquefaction systems is the Giauque–Hampson exchanger (Figure 1.5). The heat exchanger consists of carefully spaced helixes of small-diameter tubing wound in several layers around a core cylinder or mandrel. The mandrel provides mechanical stability and support during manufacture and operation of the heat exchanger. The minimum diameter of the core cylinder is determined by the diameter at which flattening of the smaller tubes would occur during winding around the mandrel. Successive layers of tubes are wound in opposite directions and separated by spacing strips (Abadzic and Scholz 1973). The tubes are attached to headers at both ends of the heat exchanger to allow the stream to be brought into or out of the exchanger in a single pipe. A close-fitting outer sheath is placed around the outside of the coil, and the entire unit is thermally insulated.



The high-pressure stream flows inside the small tubes in the Giauque–Hampson exchanger, and the low-pressure stream flows in cross flow over the small tubes in the annular space between the mandrel and the outer casing. Several layers of tubes may be used to provide multiple parallel paths for the high-pressure stream, thereby reducing the frictional pressure drop. The Giauque–Hampson heat exchanger may be designed for three or more streams, where the high-pressure and intermediate-pressure streams, for example, flow inside the small tubes. To maintain a uniform flow distribution for the streams flowing inside the small tubes, the length of each flow passage is usually the same from layer to layer. This is accomplished by varying the helical pitch of each successive layer.

It is quite important to ensure that the radial spacing of the small tubes is uniform; otherwise, the low-pressure stream will tend to follow the path through the widest spacing or least frictional resistance, and the flow will not be uniformly distributed over the cross section of the heat exchanger. Giauque at the University of California solved the spacing problem by using punched brass spacer strips to maintain uniform tube spacing (Walker 1983). A thin strip of cellulose acetate was placed between the spacer and the tube as the tube was being wound. The acetate was dissolved by acetone after the tubes had been wound, and controlled reproducible tube spacing was achieved.

Plate-Fin Exchangers

Plate-fin heat exchangers consist of stacks of alternate layers of corrugated die-formed metal sheets (the fins) separated by flat metal separation sheets (the plates) (Figure 1.6). The flow is introduced to and collected from the finned flow passages through headers welded or brazed at each end of the core stack. Several types of corrugations have been used as fins, including straight fins, wavy fins, herringbone fins, and serrated fins. Various stacking arrangement may be used to produce flow configurations such as counter flow, cross flow, and multipass heat exchangers. The counter flow configuration is most commonly used in cryogenic systems because of the better thermal performance for a counter flow exchanger for a given heat transfer surface area.

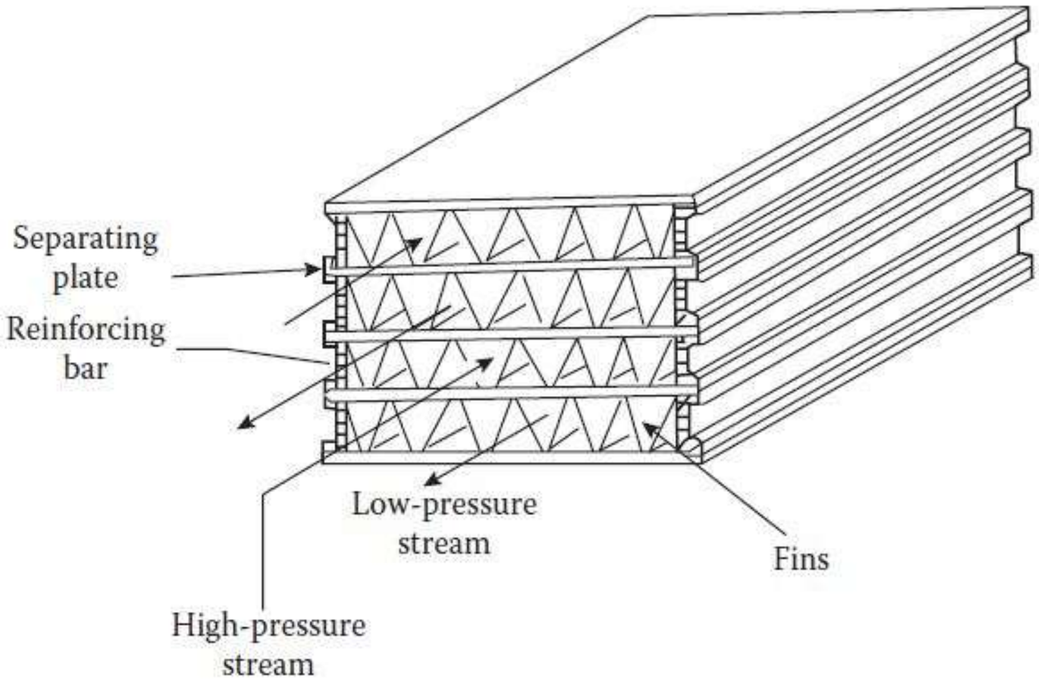


Fig. 1.6: Plate fin heat exchanger

Perforated-Plate Exchangers

A schematic of the perforated-plate heat exchanger is shown in Figure 1.7. The perforated- plate exchanger consists of a series of parallel perforated plates separated by low thermal conductivity spacers or gaskets (Fleming 1969). The plates are constructed of a high-thermal conductivity metal, such as aluminum or copper. The spacers are bonded (sealed) to the plates to prevent leakage between the two streams. Older models of the perforated plate exchangers used a plastic material for the spacer to reduce the problem of longitudinal conduction. More recent models have used stainless steel spacers to achieve better mechanical reliability (Hendricks 1996). The configuration for the perforated plate exchanger is usually counter flow.

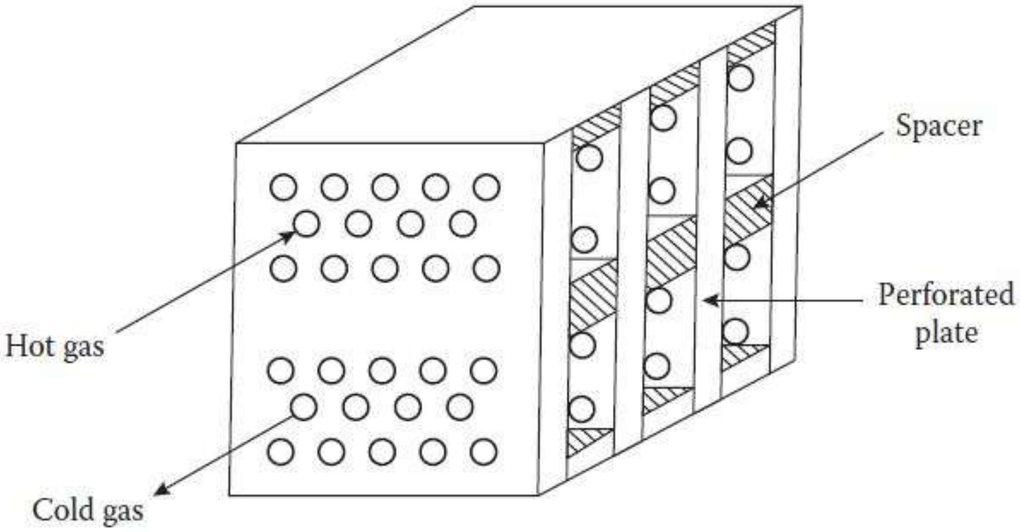


Fig. 1.7: Perforated Plate Heat Exchanger

Small perforations (diameters ranging from 1.5 mm or 1/16 in. to less than 0.4 mm or 1/64 in.) are made into the plates so that a large heat transfer coefficient is achieved. The ratio of the plate thickness (length of the hole in the plate) to the diameter of the hole is on the order of 0.75; therefore, the thermal and hydrodynamic boundary layers do not become fully developed within the perforations, which results in high heat transfer coefficients and correspondingly high friction factors.

Wire screens have been used to replace the perforated plates in some compact heat exchangers used in small cryocoolers (Vonk 1968; Lins and Elkan 1975). The fluid streams flow through the wire layers, and heat is exchanged between the two streams through conduction along the wires. The wire screens are separated by plastic or resin-impregnated paper spacers or stainless steel metallic spacers that are diffusion-bonded or brazed to the screens (perforated plates).

NTU–Effectiveness Design Method

There are two basic approaches used in heat exchanger thermal design: (1) the *NTU*–effectiveness approach and (2) the log mean temperature difference (LMTD) method. The two methods involve the same principles; however, the design variables are arranged differently in the two approaches. In the *sizing problem* or the design situation in which the heat transfer rate and all of the terminal temperatures are known and the heat transfer surface area (the heat exchanger *size*) is to be determined, both methods result in the same level of utility.

On the other hand, for the *rating problem* or design situation in which the heat exchanger surfaces area and the inlet fluid temperatures are known and the heat transfer rate is to be determined, the *NTU*–effectiveness method has some specific advantages. First, the LMTD approach in the rating problem always involves iteration, whereas the *NTU*–effectiveness method is direct—no iteration is needed. Second, the value of the LMTD does not give an explicit indication of how *good* the heat exchanger is or how effective the heat exchanger is in performing its task of energy exchange. The heat exchanger effectiveness is a direct measure of how closely the heat exchanger approaches the best possible performance in transferring energy. In addition, the *NTU* is directly related to the Stanton number for each stream. In contrast, the LMTD is not explicitly related to the dimensionless moduli used in correlating the heat transfer data.

Finally, the heat exchanger effectiveness gives a direct measure of the influence of the heat exchanger performance on the cryogenic system performance. For example, if the regenerator (heat exchanger) effectiveness in a Sterling refrigerator is 99%, more than 10% of the refrigeration effect for the refrigerator may be wasted because the regenerator performance is less than ideal (Barron 1985, p. 265). This relationship between heat exchanger performance and system performance is not explicit for the LMTD.

Heat Exchanger Factor of Safety

In mechanical design, a part is generally sized such that the part does not fail except under desired conditions. For example, an electric fuse must *fail* when a specified electric current is applied, so that the remainder of the electric system may be protected. On the other hand, the size of a crankshaft in an automobile engine is selected such that the crankshaft will not break during operation of the engine even if the stress in the part exceeds normal levels to within some factor of safety.

In thermal and hydraulic design of heat exchangers, a factor of safety is also needed. This quantity is often called the *design factor*. The design factor is intended to compensate for uncertainties in the thermal and hydraulic design that are similar to the uncertainties involved in mechanical design.

Although there is always uncertainty in the numerical values for mass flow rates, inlet temperatures, and thermal properties, such as specific heat, viscosity, and thermal conductivity, the major uncertainty in the heat exchanger design is in the correlations used to evaluate the convective heat transfer coefficient and the friction factor.

The correlations for the heat transfer coefficient and the friction factor reported in the literature are generally obtained from a least-squares curve-fitting technique that results in a *best fit* between the experimental data and the correlating curve (Weisberg 1980). In this case, one-half of the data points lie above the best-fit curve and one-half are below the best fit curve. This fact means that 50% of the time, the heat transfer coefficient experienced in operation of the heat exchanger will be *smaller* than the heat transfer calculated from the correlation. There is a 1-in-2 chance that the heat exchanger will not operate as designed that is generally not satisfactory.

Effect of Longitudinal Heat Conduction

Longitudinal conduction along the separating surfaces of the two streams in conventional heat exchangers is often an insignificant effect. For miniature cryogenic heat exchangers, on the other hand, longitudinal conduction may result in serious performance deterioration.

The reason for this difference in behavior is that miniature cryogenic heat exchangers have small distances (on the order of 100–200 mm or 4–8 in.) between the warm and cold ends (short conduction lengths); whereas, conventional heat exchangers have conduction lengths that are much longer. Because of the inherent requirement of high effectiveness for cryogenic heat exchangers, the *NTU* values are usually large (as high as 500–1000), while the *NTU* for conventional heat exchangers are on the order of 3–10. The effect of longitudinal conduction is most pronounced for heat exchangers having short conduction lengths and large *NTU*. The general result of longitudinal conduction is to reduce the local temperature difference between the two streams, thereby reducing the heat exchange effectiveness and the heat transfer rate.

Effect of Heat Transfer from Ambient

Heat exchangers used in cryogenic applications operate at significantly different temperatures from that of the ambient surroundings and, as a result of this temperature difference, there is always some heat exchange between the heat exchanger and its environment. The effect of external heat transfer is usually negligible in conventional heat exchangers; however, because of the more stringent requirement on effectiveness, environmental heat transfer may result in serious deterioration of the performance of cryogenic heat exchangers.

The physical effect of heat transfer from the surroundings to either or both of the fluid streams is to increase the local temperature of the hot stream. This effect causes the hot stream to exit the heat exchanger at a higher temperature than would be achieved if the ambient heat transfer were zero. From the standpoint of its interaction with a cryogenic system, such as a cryocooler, the performance of the heat exchanger is degraded, because the hot stream is not as cold when it leaves the heat exchanger as it would have been if ambient heat transfer were negligible.

Regenerators

In the other heat exchangers or recuperators presented in this chapter, the two fluids exchanging energy flow in separate flow channels and are separated by a solid surface, such as the tube wall (e.g., the Giauque–Hampson heat exchanger) or a plate (e.g., the plate-fin heat exchanger). In a regenerator, the same space is occupied alternately by the hot fluid and the cold fluid. Energy is first transferred from the hot stream into a solid material, called the regenerator packing material or matrix. When the cold fluid flows through the matrix during the next part of the cycle, the energy is transferred from the solid material to the cold stream.

The temperature of the fluid within the regenerator and the temperature of the regenerator matrix are functions of both position within the regenerator and time of operation. As a result, the performance of a regenerator is a function of the mass of the matrix, the specific heat of the matrix material, and the frequency of switching of the flow through the regenerator, in addition to the other parameters (capacity rate ratio and NTU) that are important for an ordinary heat exchanger (Schmidt and Willmott 1981).

The primary advantages of regenerators over ordinary heat exchangers (recuperators) include the following:

1) A much larger heat transfer surface area per unit volume can be achieved for a regenerator. For example, as much as 4000 m^2 of surface area per cubic meter of volume can be obtained in a regenerator, compared with 1475 m^2/m^3 for a typical compact plate-fin heat exchanger.
2) Regenerators are generally simpler to fabricate than recuperators. In its most simple form, a regenerator could consist of a tube or pipe filled with lead shot. As a result of the simplicity of fabrication, regenerators tend to cost less than a recuperator for the same heat transfer rate.
3) Because of the alternate direction of flow of the hot and cold streams through the same flow passage, the regenerator tends to be self-cleaning. Impurity buildup problems are not nearly as severe in a regenerator as those encountered in ordinary heat exchangers.

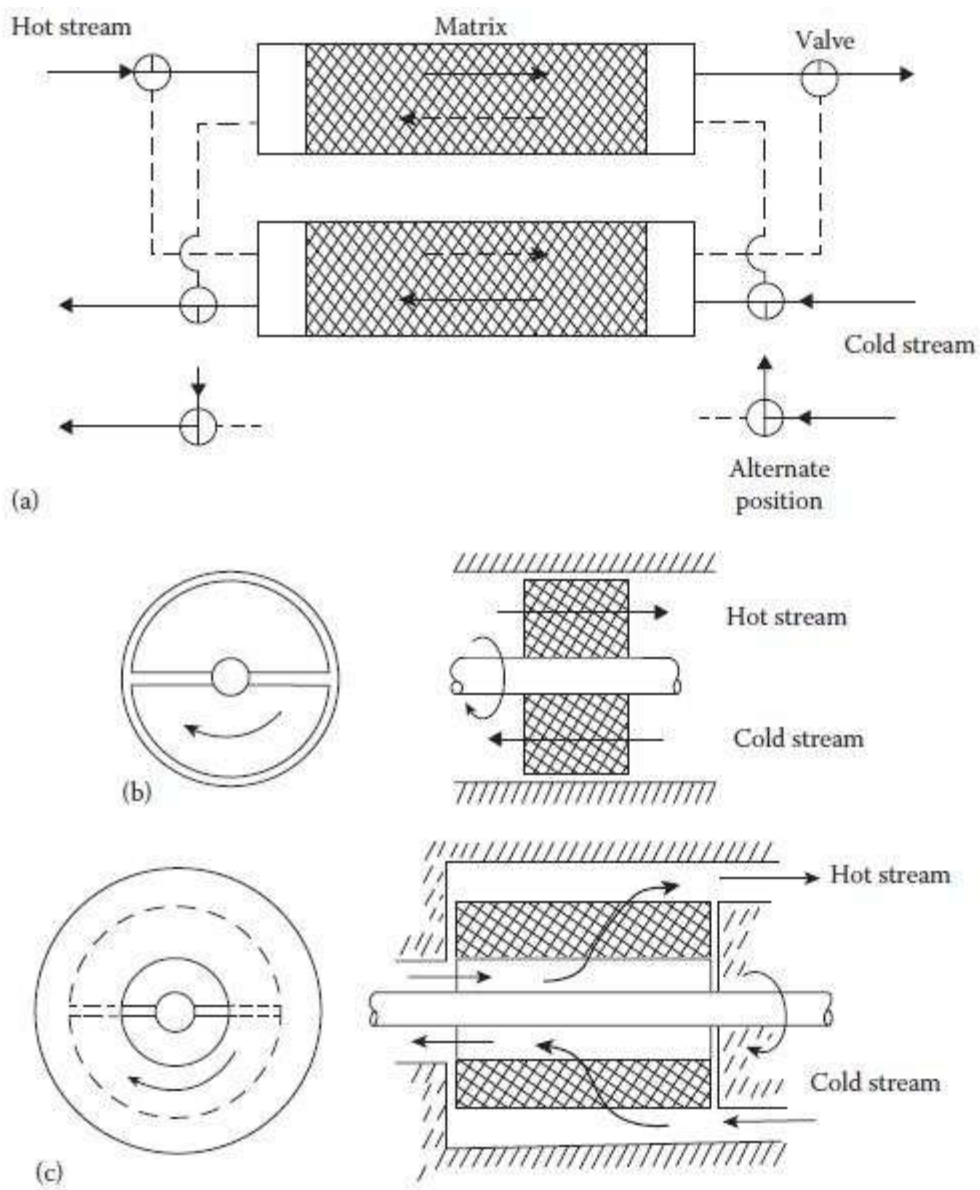


Fig. 1.8: Types of regenerators: (a) fixed-bed or valved type regenerator; (b) rotary regenerator, axial flow type; and (c) rotary regenerator, radial flow type.

The primary disadvantages of regenerators, compared with regular heat exchangers, include the following:

1) There is always some mixing of the warm and cold streams during the switching process in a regenerator. This carry-over may not be important if the two streams are the same fluid, for example; however, excessive carry-over degrades the regenerator thermal performance.
2) For the rotary regenerators, leakage at the dynamic seals may become appreciable. This problem is particularly acute when the two streams are at significantly different pressures.

The regenerator acts as a *thermal sponge* or *thermal flywheel* to store energy during the period of flow of the hot stream and deliver this energy back to the cold stream when it flows through the regenerator. The effectiveness of the storage process is related to the thermal capacity (product of the mass and specific heat per unit mass) of the matrix material.

If the specific heat is excessively small, the thermal capacity of the matrix material is insufficient to produce effective energy storage, and the resulting regenerator thermal performance is poor.

References

1) Randall F. Barron, “Cryogenic System”, Second edition, Oxford university Press, 1985.

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