Two-Phase Heat Exchangers

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Contents

Abstract

This chapter provides the working principles of classical types of two-phase heat exchangers and heat exchangers based on microchannel technology. The applications of these exchangers in the power and process industry as well as in air conditioning and electronic cooling are discussed. The classifications of basic types of liquid-to-vapor phase-change exchangers depending on flow arrangement and construction features are presented. The inclusion of the latent heat of evaporation in thermal energy transfer improves the transport capacity and

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intensifies the heat transfer. The approach for thermal analysis of the condensers and evaporators based on overall heat transfer coefficient and log-mean temperature difference is discussed. The literature on heat transfer coefficient prediction for liquid-to-vapor phase-change exchangers is reviewed and experimentally proved correlations are presented. The general constructions of tubular, plate-type, and extending surface two-phase heat exchangers for various applications are discussed. Also, the working principles of microchannel heat exchangers in flow boiling and condensing modes are defined in association with benefits using these systems for air conditioning and electronic cooling.

1 Introduction

During single-phase heat exchanger operation, all the fluids remain in the same phase (i.e., liquid or gas), and the heat is transferred by convection and by conduction through a wall separating the flows. The limitation on thermal energy transfer for this case is defined by the mass flow rate and heat capacity of the fluids as far as surface area of the heat exchanger. Two-phase heat exchanger is the device that transfers thermal energy between two fluids during the phase-change process, when the latent heat of vaporization is released. Transfer of the thermal energy from hot fluid causes the boiling in cold liquid as shown in Fig. 1a. Otherwise, if hot fluid is saturated vapor, then transfer of the thermal energy to cold fluid causes the condensation as shown in Fig. 1b in both cases, variation of the enthalpy in a flow with the phase change is defined by a change in the vapor quality, rather than a change in the flow temperature. The inclusion of the latent heat of evaporation in thermal energy transfer improves the transport capacity and intensifies the heat transfer. Therefore more than 60% of the heat exchangers in industry operate in two-phase mode (Mayinger 1988). The diversity of heat exchanger constructions and the variety of operating conditions prevent a complete presentation of two-phase heat exchangers design in this chapter. Therefore this chapter will focus mainly on a description of the key points regarding thermo-hydraulic design and general constructions of the most frequently utilized two-phase heat exchangers.

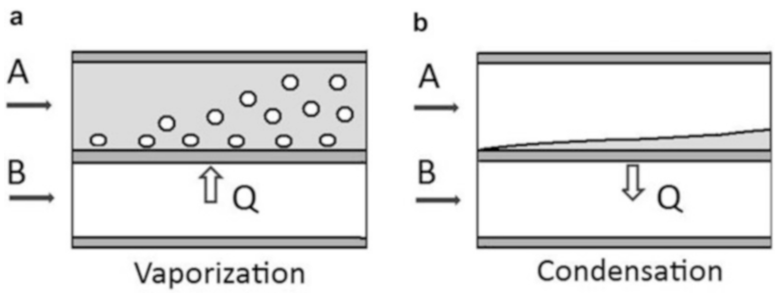


Fig. 1 Scheme of the thermal energy transfer during boiling (**a**) and condensing (**b**)

2 Applications of Two-Phase Heat Exchangers

Two-phase heat exchangers are widely used in the power engineering, petroleum, chemical, and food industries (Mayinger 1988; Pate 1991; Smith 1986; Butterworth 1988). They can be used either as evaporator to produce a vapor or as condensers to produce a liquid, for heating or cooling of the fluid streams, to control the temperature of the process fluid and solid surface, or to recover or reject heat, multiphase distillation, and so on. Two-phase heat exchangers have many different aspects and features depending on application (Mayinger 1988). Some applications of two-phase heat exchangers are presented in Fig. 2. They can be met in the process industry as reboilers, evaporators, condensers, freezers, microstructured heat exchangers, and MEMS devises. Power plants commonly use heat exchangers to prepare the steam for steam-driven turbines (Mayinger 1988). Two-phase heat exchangers for steam production are often called boilers or steam generators. In power engineering these heat exchangers are used as an important element of the nuclear plant as condensers and steam generators (Mayinger 1988). In the pressurized nuclear power plants, special heat exchangers transfer heat from the primary system (water at high pressure) to the secondary system, producing steam from water. The fossil boilers and nuclear power plants have surface condensers to convert the exhaust steam after the turbines into condensate (Kitto and Albrecht 1988). Distillation columns in refineries typically use the combined condensers and evaporators for energy saving (Butterworth 1988). Two-phase heat exchangers have very important applications in air conditioning, refrigeration, and electronic cooling (Carey and Shah 1988; Garimella 2003). The phase change occurs at a constant temperature, which allows us to optimize the cooling strategy for high-power laser diodes. Another area where it can be used is high-power aircraft electronics and computer chips cooling (Mudawar 2001).

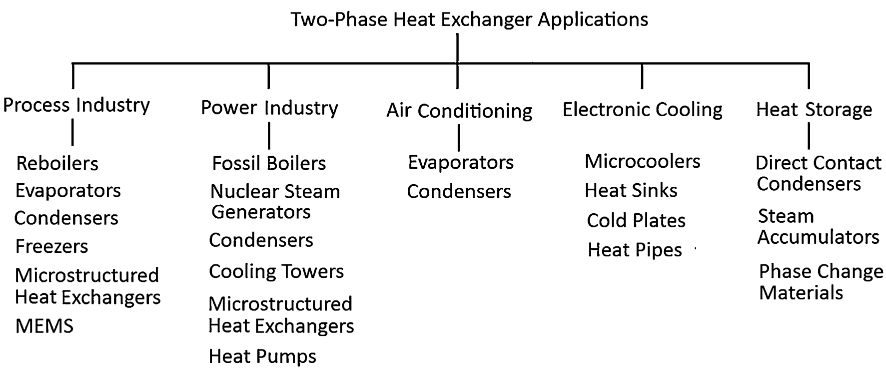


Fig. 2 Applications of two-phase heat exchangers

3 Basic Types of Two-Phase Heat Exchangers

The basic types of two-phase heat exchangers were presented in many books and research papers (Shah 1981, 1998; Mayinger 1988; Shah and Sekulic 2003; Kakaç et al. 2012). There are two approaches that are usually considered in the presentation of the basic types of two-phase heat exchangers. The first considers the flow arrangement within the exchanger, while the second is based on the classification of exchanger type by its construction. In more detail, the diversity of two-phase heat exchanger can be classified according to the transfer processes, degree of surface compactness, construction features, flow arrangement, number of fluids, and heat transfer mechanisms (Shah 1981). Classifying heat exchangers in accordance with the transfer process, it is necessary to select the indirect contact type (recuperators and regenerators) and direct compact type exchangers; see Fig. 3. In indirect contact type exchanger, the wall participates in the transfer of heat energy between fluids that does not allow them to mix (Shah 1981, 1998). The heat transfer wall (primary surface) is used in recuperators to separate the fluids and transfer the thermal energy by conduction.

Recuperative heat exchangers are most common for application in various fields of technology. Some examples of this type of heat exchangers are tubular, plate-type, extended surface, and microchannel heat exchangers. To increase the heat transfer area, fins may be connected to the primary surface to provide an extended, secondary surface (Shah and Webb 1983; Pate 1991). In extended surface exchangers, heat is conducted through the fins and separating wall to the surrounding fluid, or vice versa, depending on whether the flow is being cooled or heated. As a result, the addition of fins to the primary surface reduces the thermal resistance and therefore increases the total heat flux at the same temperature difference. In microchannel heat exchangers the fins may form a flow passage considerably increasing the exchanger specific surface area (Shah 1991, 1998).

In regenerative heat exchangers, the thermal storage matrix firstly accumulates the thermal energy from hot fluid and later transfers it to cold fluid (Shah and Sekulic 2003; Kakaç et al. 2012). The thermal storage matrix is alternately in contact with fluids and can be as switching and rotating. The regenerator matrix occurs in the

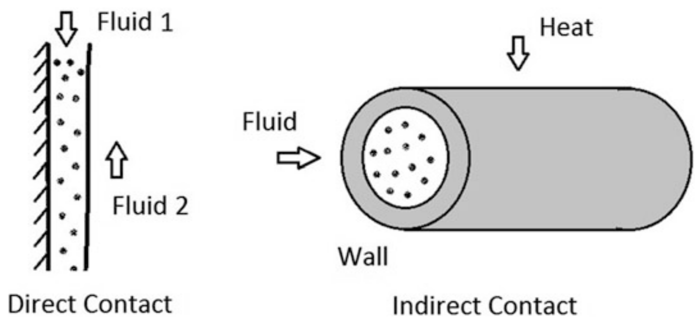


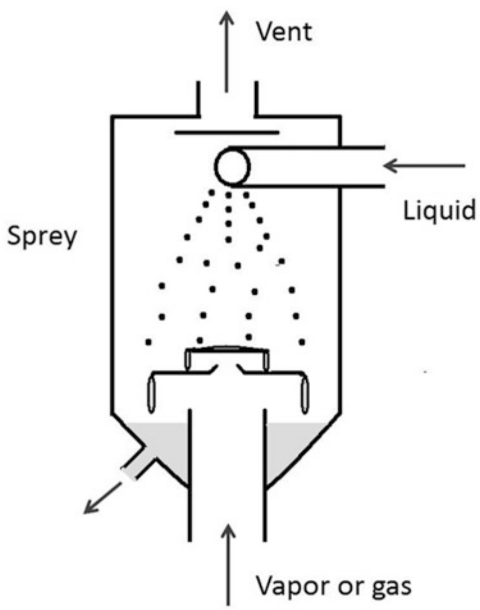
Fig. 3 Classification of two-phase exchangers according to transfer process

form of packed bed, structured packing, or permeable monolith, through which alternating the primary and the secondary flows are led. The regenerators are mainly used in energy-saving equipment.

In direct contact type exchanger, the heat is transferred between hot and cold streams of two phases in the absence of a separating wall (Shah 1981). The interface between the phases forms a heat transfer surface, which can be extremely high as the result of interface instability development and special organization of the heat exchanger internals. Most of these heat exchangers are gas–liquid type, and thermal energy is transferred between a gas (vapor) and liquid in the form of thin film, droplets, or sprays (Butterworth 1988), as shown in Fig. 4. These heat exchangers are predominantly used as direct contact type condensers and cooling tower of power plant, in the cryogenics, air conditioning, humidification, and water condensing plants.

Heat exchangers provide heat transfer between two or more of the coolant flows. Therefore, the important characteristic of a heat exchanger design is the relative direction and mutual geometry of the flows (Shah 1981; Kakaç and Paykoç 1991; Shah and Sekulic 2003). Indirect contact type heat exchangers are classified according to relative direction of the flows into cocurrent flow, counter flow, cross-flow, and multi-pass flow. In cocurrent heat exchanger, two fluids A and B flow parallel to each other in the same direction. This type of flow is shown schematically in Fig. 5a. In counter-current heat exchanger, two fluids flow parallel to each other but in counter direction. This type of flow is shown schematically in Fig. 5b. In practice, large diameter shell (heat exchanger shell) may contain many heat transfer tubes inside. Counter-current heat exchangers are most effective for single-phase

Fig. 4 Direct contact type exchanger (Butterworth 1988)



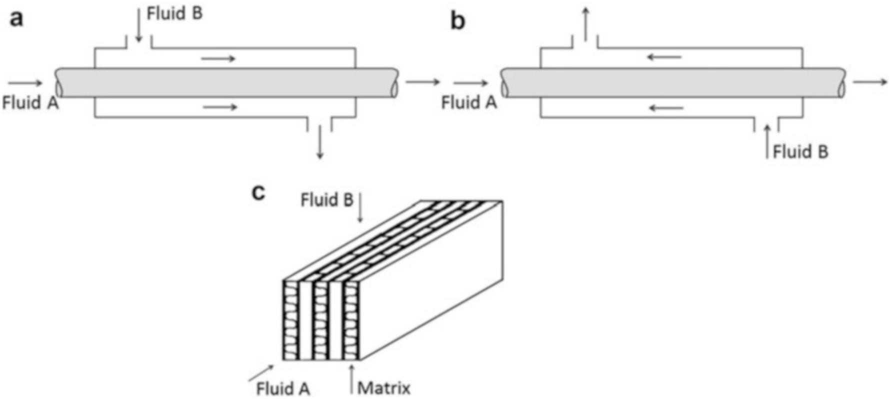


Fig. 5 Cocurrent (**a**), counter-current (**b**), and cross-flow (**c**) heat exchangers

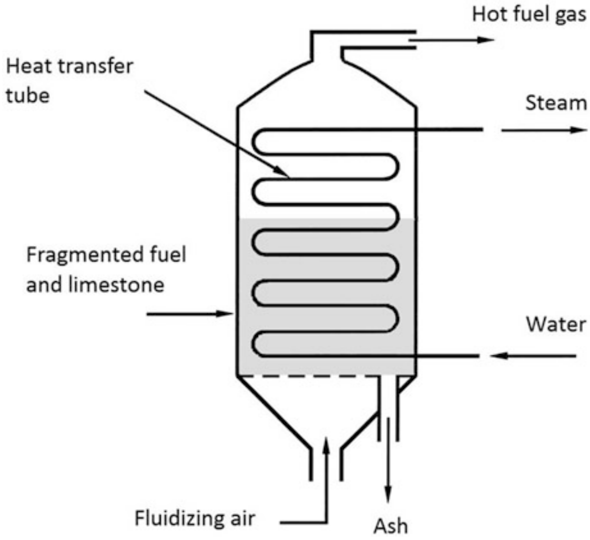
flow, but in the presence of phase transition, their efficiency is leveled. In cross-flow heat exchanger, two fluids move at right angles to each other. For example, the fluid A can flow inside the fin passage (tubes), and a second fluid flows in other fin passage (or shell) as shown in Fig. 5c. This type of heat exchanger allows easier to design the flows. In multi-pass heat exchangers, the combination of cocurrent and counter-current flows inside the shell is organized.

The characterization of the heat exchangers according to the heat transfer mechanisms is based on the selection of the heat transfer due to the phase change and a two-phase convective heat transfer. A typical example of two-phase convective heat transfer without phase change is the fluidized bed where a mixture of gas and solid particles transfers the heat to or from a wall which may be round tube or flat surface (Shah and Sekulic 2003); see Fig. 6. The heat transfer mechanism in fluidized bed is very important in power engineering due to the development of fluidized bed combustors. In this combustor, tube bundles are immersed in the fluidized bed of coal particles, ashes, and hot gas. The water boiling occurs inside these tubes producing steam from the heat of combustion.

Two-phase heat exchangers can be characterized according to the patterns of fluid motion on heat exchangers where the natural convection, forced convection, or flow under gravity occurs. Natural convection is important for evaporators, boilers, and heat pipes where the fluids move due to density difference. Forced convection is important for the recuperators and regenerators.

According to the process function, two-phase heat exchangers can be classified as condensers, boilers, evaporators, coolers, and chillers. In indirect contact type condensers, two fluids (vapor and liquid) are separated by a heat transfer surface. In direct contact condensers, a physical contact between the fluids occurs, allowing for the condensation to be accomplished with the mixing process. The vapor is condensed to a liquid with or without superheating and/or subcooling. Important aspects of condenser operation involve desuperheating, condensation, subcooling, and the presence of noncondensable gas in the condensing vapor. The evaporators

Fig. 6 Fluidized bed exchanger



and condensers have important application in power and process engineering, because of the need for increasing the unit power per station. In the case of nuclear power plants, the desired thermal power is higher than 2000 MWth per station, which lead to a necessity in the development of modern boilers, new steam generators, and condensers (Mayinger 1988).

4 Thermal and Hydraulic Design of Two-Phase Heat Exchangers

4.1 Basic Equations for Heat Exchanger Design

The important decision underlying selection of heat transfer equipment is its thermal and hydraulic design. Although complete design of the heat exchanger requires solution of many problems, the purpose of the thermo-hydraulic analyses is a solution of the two most important problems that are referred to as the sizing and rating. The solution of sizing problem determines a surface area of the heat exchanger for desirable application. The performance calculations (rating problem) are associated with the determination of the heat transfer and pressure drop performance when the surface area of heat exchanger is determined. To perform these calculations, the flow arrangement, fluid flow rate, and inlet conditions should be taken into account as far as suitable dimensionless equations for prediction of the heat transfer coefficient and pressure drop. Then the fluid outlet temperatures, total heat transfer rate, and pressure drop can be determined during solving of the rating problem.

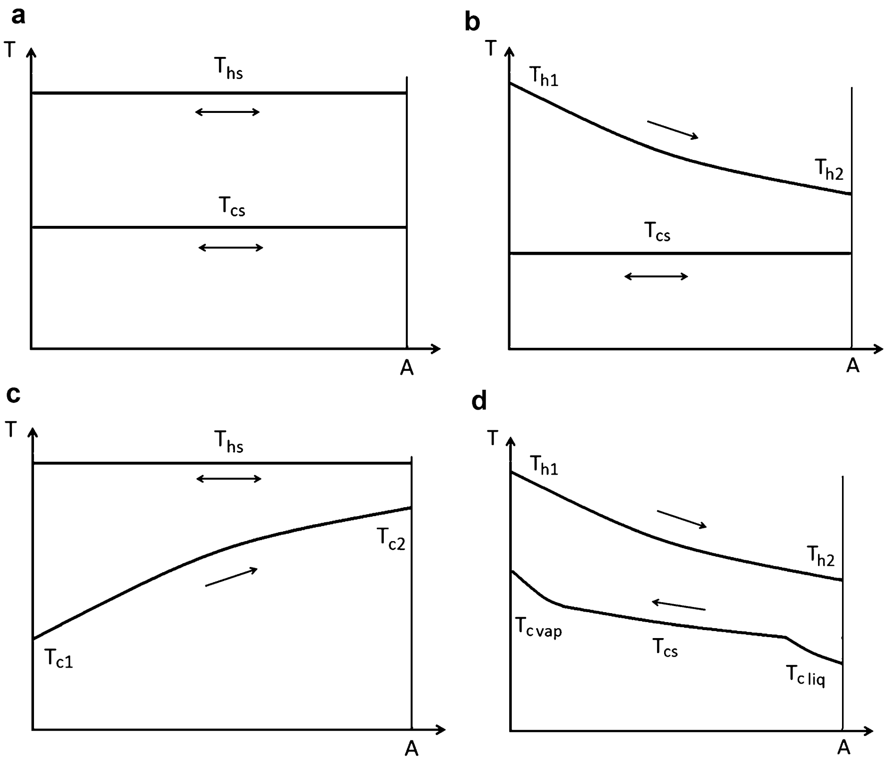


Fig. 7 Variation of temperature difference in dependence on heat transfer area for phase change in both fluids (**a**), for boiling (**b**) and condensing (**c**) in one single-component fluid, and for condensing of multicomponent fluid (**d**)

Most common heat exchangers operating under two-phase and multiphase flow conditions are condensers and vaporizers (Shah and Sekulic 2003; Kakaç et al. 2012). Therefore the objective of this section is to summarize the key points regarding thermo-hydraulic design for these heat exchangers. The design of indirect contact type condensers and vaporizers, where heat transfer between fluids occurs through a separating wall, will be discussed.

Two-phase recuperative heat exchangers can be divided on three groups: heat exchangers with constant temperature of both fluids (evaporator–condenser of single-component fluids without superheating/subcooling), heat exchangers with constant temperature of one fluid (evaporator or condenser of single-component fluid), and heat exchanger with variable temperature of both fluids (evaporator/condenser of multicomponent or superheating/subcooling fluids, high-pressure drop). All these cases are shown in Fig. 7a, b, c, d, where the variation of temperature difference in the fluids is plotted in dependence of heat transfer surface area increasing (distance from heat exchanger inlet). Referring to Fig. 7, the variation of temperature difference is plotted in Fig. 7a with hot condensing fluid and cold evaporating fluid at constant temperatures. In Fig. 7b, c, the variation of temperature difference is plotted for the cold fluid at constant temperature (evaporator) and hot fluid at constant temperature (condenser). The variation of temperature difference in the case of evaporation of the multicomponent fluid is plotted in Fig. 7d. In these figures, the slope of temperature profiles for fluid without phase change depends on its heat capacity ($\dot{m}c_p$) as far as latent heat h_{fg} and pressure drop for fluid with phase change.

From the first law of thermodynamics for open system under steady-state conditions, neglecting potential and kinetic energy changes, the enthalpy balance in the hot fluid for an evaporator (Fig. 7b) can be written as follows:

$$Q_h = \dot{m}_h(h_{h1} - h_{h2}) \tag{1}$$

Hear Q_h is the heat transfer rate, $\dot{m}$is the mass flow rate, h is the enthalpy, and indexes 1 and 2 correspond to the heat exchanger inlet and outlet accordingly. For cold fluid with phase change, it is suggested that its temperature is not varied along the length of the heat exchanger and equals to the saturation temperature T_{cs} at a pressure in corresponding heat exchanger section. In this case, Eq. (1) can be written as follows:

$$Q_h = \left(\dot{m}c_p\right)_h(T_{h1} - T_{h2}) \tag{2}$$

As it can be seen from Fig. 7b, the temperature difference between the hot and cold fluids ($\Delta T = T_h - T_{cs}$) depends on the position in heat exchanger. Therefore, it is convenient to determine an approximate mean value of the temperature difference between hot and cold fluids, which allows us to present the total heat transfer rate Q_t between the fluids in the standard form:

$$Q_t = UA\Delta T_m \tag{3}$$

Here A is the heat transfer area, which should be determined as solution of sizing problem, and U is the overall heat transfer coefficient based on that area. It is clear that the solution of sizing problem comes down to determination of the overall heat transfer coefficient and mean temperature difference.

4.2 Overall Heat Transfer Coefficient

The overall heat transfer coefficient for plain extended surface U is determined by thermal resistance in hot fluid and cold fluid and of the wall. Combining all these resistances in series gives (Kakaç and Paykoç 1991):

$$\frac{1}{UA} = \frac{1}{(\eta_0 \alpha A)_h} + \frac{1}{\Omega k_w} + \frac{1}{(\eta_0 \alpha A)_c} \tag{4}$$

where η_0 is the surface efficiency of inner and outer surfaces, α is the heat transfer coefficients for the inner and outer surfaces, k is wall thermal conductivity, and Ω is a shape factor for the wall separating hot and cold fluids.

In Eq. (4), the surface efficiencies account the fin existence for hot and cold fluids and are related to the fin efficiency as follows:

$$\eta_0 = \left(1 - \left(1 - \eta_f\right)\frac{A_f}{A_i}\right) \tag{5}$$

Here A_f is fin surface area and A_i is total surface area for hot and cold fluids. For straight fins of length L and thickness δ_f, the fin efficiency is as follows:

$$\eta_f = \tanh(mL)/mL, m = \sqrt{2\alpha_i/\delta_f k_f} \tag{6}$$

For another fin configuration, the expression for fin efficiency is available in Kern and Kraus (1972).

The shape factor in Eq. (4) equals to $\Omega = A/\delta_w$ for plain surface and $\Omega = 2\pi L/\ln(r_0/r_i)$ for cylindrical tube. The fouling increases the heat resistance, and the additional terms should be added in Eq. (4) (Kakaç and Paykoç 1991).

The heat transfer coefficient in fluids with phase change depends on the surface conditions, mass and heat fluxes, vapor quality, and so on. The correlations for heat transfer prediction are presented in the next chapters of this book. Some examples of the equations for calculation of heat transfer coefficient in application for two-phase heat exchangers are briefly presented further.

During condensation on cold surface, the condensate is accumulated in a form of liquid film or droplets depending on the wettability conditions. Laminar film condensation of quiescent vapor on a single horizontal tube is characterized by average heat transfer coefficient as follows (Nusselt 1916):

$$Nu_m = \frac{\alpha_m d}{k_{\text{liq}}} = 0.728\left[\frac{\rho_{\text{liq}}\left(\rho_{\text{liq}} - \rho_{\text{gas}}\right) g h_{fg} d^3}{\mu_{\text{liq}}(T_s - T_w) k_{\text{liq}}}\right]^{0.25} \tag{7}$$

At high vapor velocities, the forced convection heat transfer coefficient depends on the interphase shear stress and grows with the vapor velocity increasing (Butterworth 1977):

$$Nu_m = 0.416\text{Re}^{0.5}\left(1 + \sqrt{(1 + 9.47F)}\right)^{0.5},$$
$$\text{Re} = u_{\text{gas}} d/\nu_{\text{liq}}, F = \frac{g d \mu_{\text{liq}} h_{fg}}{u_{\text{gas}}^2 (T_s - T_w) k_{\text{liq}}} \tag{8}$$

During film condensation inside a horizontal tube, the interface shear stress defines a liquid film thickness and heat transfer coefficient. For refrigerant condensing, the correlations for the determination of the heat transfer coefficient were developed by Traviss et al. (1972), Cavallini and Zecchin (1974), and Shah (1979). These correlations can be used for prediction of the heat transfer coefficients during condensation inside a horizontal tube.

Downward laminar condensation inside a vertical tube at low vapor velocity is controlled by gravity, and heat transfer coefficient may be calculated according to Nusselt theory for vertical surface:

$$Nu_m = \frac{\alpha_m L}{k_{\mathrm{liq}}} = 0.943\left[\frac{\rho_{\mathrm{liq}}\left(\rho_{\mathrm{liq}} - \rho_{\mathrm{gas}}\right) g h_{fg} L^3}{\mu_{\mathrm{liq}}(T_s - T_w) k_{\mathrm{liq}}}\right]^{0.25} \tag{9}$$

For turbulent condensate flow, the averaged heat transfer coefficient may be calculated by the methods proposed by Marto (1991).

The flow boiling heat transfer inside of vertical and horizontal tubes may be calculated according to Chen (1966) and Shah (1982) correlations, Liu and Winterton (1991) methods for subcooled and saturated boiling, and Kandlikar (1990) correlation. Detail description of the methods for prediction of the flow boiling heat transfer is given in Kandlikar et al. (1999). These correlations can be applied before heat transfer crises where the rapid deterioration of heat transfer occurs.

4.3 Log Mean Temperature Difference

The local heat transfer rate for heat exchanger element with surface area *dA* is determined through overall heat transfer coefficient as follows:

$$dQ_h = U(T_h - T_{cs})dA \tag{10}$$

where the value of U must correspond to the position in heat exchanger. The solution of Eq. (10) is as follows:

$$A = \int_0^{Q_t} \frac{dQ_h}{U(T_h - T_{cs})} \tag{11}$$

Equation (11) can be solved numerically for a given heat exchanger, utilizing local overall heat transfer coefficient and local fluid temperature difference. To evaluate total heat transfer area from Eq. (11), simple procedure may be applied also for most heat exchangers. The key assumptions that need to be made are as follows (Shah 1998):

1. All fluids in heat exchanger are in contact with heat transfer surface.
2. The heat exchanger is at steady state.
3. The specific heat of fluids is constant.
4. The overall heat transfer coefficient is constant.
5. Heat loss or gains to/from the surroundings are negligible.
6. Longitudinal heat is negligible.
7. The flow arrangement is either purely counter-current or purely cocurrent.

Under these assumptions, the heat transfer rate can be excluded from Eq. (2) for local enthalpy balance in hot fluid

$$dQ_h = -\left(\dot{m}c_p\right)_h dT_h \tag{12}$$

and from Eq. (10) to obtain

$$-\left(\dot{m}c_p\right)_h dT_h = U(T_h - T_{cs})dA \tag{13}$$

After the integration of Eq. (13), the equation for temperature drop variation along the length of the heat exchanger can be obtained as follows:

$$T_h - T_{cs} = (T_{h1} - T_{cs})e^{-UA/\left(\dot{m}c_p\right)_h} \tag{14}$$

This equation can be used for the determination of the log-mean temperature difference:

$$\Delta T_{LM} = \frac{1}{A_t}\int_0^{A_t}(T_h - T_{cs})dA = \frac{T_{h1} - T_{h2}}{\ln((T_{h1} - T_{cs})/(T_{h2} - T_{cs}))} \tag{15}$$

In assumption of constant overall heat transfer coefficient, the integration of Eq. (10) gives:

$$A_t = Q_t/(U\Delta T_{LM}) \tag{16}$$

where ΔT_{LM} is defined from Eq. (15).

Equation (16) was obtained for one pass recuperator. For another type of heat exchanger, the correction factor must be applied because the actual flow arrangement should be accounted. The correction factor F as multiplier to the log-mean temperature difference in Eq. (16) is a ratio of the true mean temperature difference to the log-mean temperature difference (Shah and Sekulic 2003). If condensation takes place for one fluid and evaporation for the other fluid (see Fig. 5a), the good approximation is $F = 1$ for different flow arrangements.

In practical application, the overall heat transfer coefficient U varies along the length of the exchanger because of the dependence of the individual heat transfer coefficients on local heat flux, wall temperature, and vapor quality for fluid with phase change. The temperature in the fluid with phase change also is not constant because of possible fluid superheating or subcooling in the inlet and outlet sections of exchanger; see Fig. 7d. The difficulties in thermal design of the heat exchanger with varied overall heat transfer coefficient can be avoided by dividing the condenser's total heat transfer load in an appropriate number of heat duty zones and subsequently writing the energy balances based on enthalpy differences for each zone (Shah 1998; Kakaç and Paykoç 1991). It is necessary to establish for each zone the corresponding zonal mean overall heat transfer coefficients and mean temperature differences. As a result, the heat transfer surface for each zone may be calculated using Eq. (16). Total heat transfer area needed for design is equal to the sum of the heat transfer areas of all zones: $A_t = \sum_1^N \Delta A_i$. The calculations can be performed also using the numerical calculations. This procedure for shell-and-tube condensers and evaporators was proposed and described in detail by Butterworth (1988).

4.4 Heat Exchanger Pressure Drop

Pressure drop in heat exchangers is an important consideration during the design stage, and its calculations are required for both fluids. Pressure drop is determined by many factors, including the exchanger core geometry and the flow patterns. At exchanger exit, when a fluid enters the exchanger core, it is subjected to an entrance loss due to sudden reduction in flow area, then the heat exchanger core itself contributes a loss due to friction and other internal losses, and as a fluid exits the core, it is subjected to a loss due to sudden expansion. In addition, if the vapor quality changes through the core due to phase change, the acceleration or deceleration pressure drop is experienced. This also contributes to the overall pressure drop.

The total pressure drop during a flow with phase change can be represented as the sum of frictional (ΔP_f), gravitational, and momentum pressure drop (ΔP_m) components as well as the pressure loss (ΔP_c) and recovery (ΔP_e) due to the inlet contraction and outlet expansion. $\Delta P = \Delta P_\mathrm{f} + \Delta P_\mathrm{m} + \Delta P_\mathrm{c} + \Delta P_\mathrm{e}$. The gravitational component can be neglected for horizontal channels. For calculations of the pressure drop across the inlet and exit parts of exchanger, the model of Abdelall et al. (2005) may be used. For calculation of momentum pressure drop, Zivi (1964) equation can be applied for calculation of the void fraction.

The predominant flow pattern in two-phase heat exchanger is the annular flow. Therefore, in calculation of the friction pressure drop, Lockhart and Martinelly (1949) model may be used as follows:

$$\Delta p_f = \Phi^2 (dp/dz)_{\mathrm{liq}} L \tag{17}$$

Two-phase multiplier in this model can be selected according to Chisholm (1983) model:

Table 1 Criteria for selecting value of parameter C

Liquid	Gas	C-value
Laminar	Laminar	5
Laminar	Turbulent	12
Turbulent	Laminar	10
Turbulent	Turbulent	21

$$\Phi^2 = 1 + \frac{C}{X} + \frac{1}{X^2} \tag{18}$$

Here $X^2 = \left(\frac{dP}{dz}\right)_{\text{liq}} / \left(\frac{dP}{dz}\right)_{\text{gas}}$, $\left(\frac{dP}{dz}\right)_{\text{gas}} = \frac{f_{\text{gas}} G^2 x^2}{2\rho_{\text{gas}} D}$, $\left(\frac{dP}{dz}\right)_{\text{liq}} = \frac{f_{\text{liq}} G^2 (1-x)^2}{2\rho_{\text{liq}} D_h}$, $G = \dot{m}/S$, friction factors for liquid flow f_{liq} and gas flow f_{gas} are calculated using $\text{Re}_{\text{liq}} = G(1-x)D/\mu_{\text{liq}}$ and $\text{Re}_{\text{gas}} = GxD/\mu_{\text{gas}}$. Criteria for selecting value of the parameter C are given in Table 1.

5 Construction Features of Basic Types of Heat Exchangers

5.1 Design Concepts of Recuperative Exchangers

The selection of heat exchanger type which can be suitable for the given application is the first stage in the design process. All heat exchangers consist of common elements such as a core containing the heat transfer surface and fluid distribution system including headers, manifolds, inlet, and outlet pipes or nozzles (Walker 1990; Taborek 1991). The heat transfer surface of the exchanger core is in direct contact with fluids and transfers heat by conduction. For efficiency, heat exchangers must be designed to maximize the heat transfer surface area between the two fluids while minimizing resistance to fluid flow through the exchanger.

Despite the presence of common elements in a construction, many design concepts are used for two-phase heat exchangers depending on the application. From pure geometrical aspects, the tubular and plate concepts are by far the most common. The separating wall which transfers the heat energy may have extending and non-extending surface. For extending surface heat exchanger, the fins can be used for the tubular as far as for plate arrangement to enhance the heat transfer. Therefore the four major construction types of the indirect contact type heat exchangers are tubular, plate-type, extended surface, and regenerative exchangers. Figure 8 shows the Shah (1981) classification according to the construction features adopted for classical two-phase heat exchangers. Other constructions of heat exchangers are also available, such as tank heater, cooler cartridge exchanger, and others (Hewitt et al. 1994). To select the most applicable for particular process type of heat exchanger, it is advisable to survey the range of available basic equipment types.

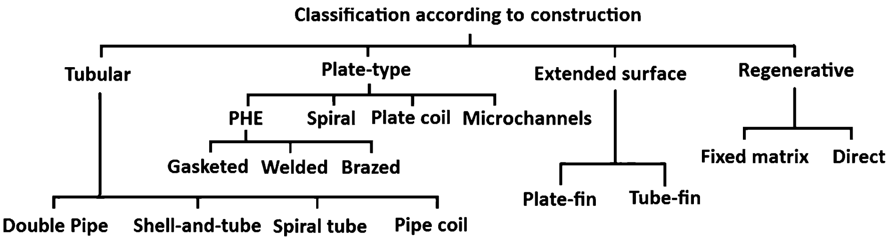


Fig. 8 Classification of heat exchangers according to the construction features

5.2 Tubular Heat Exchangers

Tubular heat exchangers are the most popular regenerative heat exchanger for process industry and power industry because of the reliability and well-proven technology of their manufacturing and cleaning. These exchangers are generally made from the circular, elliptical, or round/flat twisted tubes. They are flexible in design because the core arrangement can be easily varied by changing the tube diameter, tube density, and installation of the baffle for flow arrangement. Tubular exchangers are widely used in industry because they are designed for various operating conditions, namely, from high vacuum to static pressure over 100 MPa and from cryogenics to temperatures up to 1100 °C (Mayinger 1988). They can be used for special operating conditions including heavy fouling, corrosion, radioactivity, and so on.

Two-phase tubular exchangers are widely used for liquid-to-vapor and vapor-to-liquid phase-change applications as nuclear steam generators, condensers, and fossil boilers in power industry, as process heat exchangers in the petrochemical industries, and as condensers and evaporators in air conditioning and refrigeration and in waste heat recovery. These exchangers may be subdivided on double-pipe, shell-and-tube, and spiral tube exchangers. To increase heat transfer coefficient, they can have heat transfer intensifiers and fins on the tube surface.

5.2.1 Double-Pipe Exchanger

Double-pipe exchanger is usually applied for small-capacity applications and may be used as small boilers and condensers. A typical double-pipe heat exchanger is shown in Fig. 5a, b. This exchanger consists of central pipe placed inside another one with an annular gap for fluid flow. The inner pipe may have internal and external microfins to enhance heat transfer for the fluid with low heat transfer coefficient especially for two-phase applications. Their characteristic features are the flexibility of operation and simple installation. The constructions of double-pipe exchangers are discussed in Schlunder (1983) and Kirchner (2010).

5.2.2 Shell-and-Tube Heat Exchanger

Shell-and-tube heat exchangers are one of the most popular types of exchanger due to possibility to operate in wide range of pressures and temperatures (Schlunder

1983; Saunders 1988; Shah 1998). Figures 9 and 10 show the schemes of a condenser and kettle-type reboiler that can be met in a petrochemical plant (Collier 1988). A shell-and-tube exchanger consists of a number of tubes installed inside a cylindrical shell. The exchanger typically consists of four major parts: front-end head, where the vapor or liquid enters the tube side of the exchanger, rear-end where the tube side vapor or condensate leaves the exchanger or where it is returned back for design with multiple tube side passes, tube bundle arrangement, and cylindrical shell (Shah 1998; Shah and Sekulic 2003). Once fluid flows inside the tubes, the other flows across and along the tubes. To enhance heat transfer, the tubes with serpentine, helical, and bayonet tube shapes are used as far as microfin surface; see

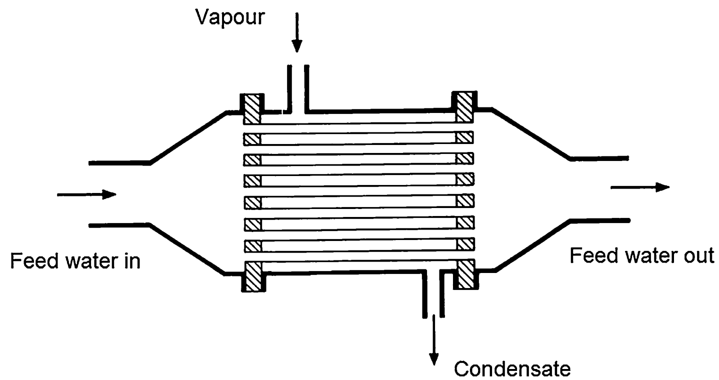


Fig. 9 Scheme of shell-and-tube condenser

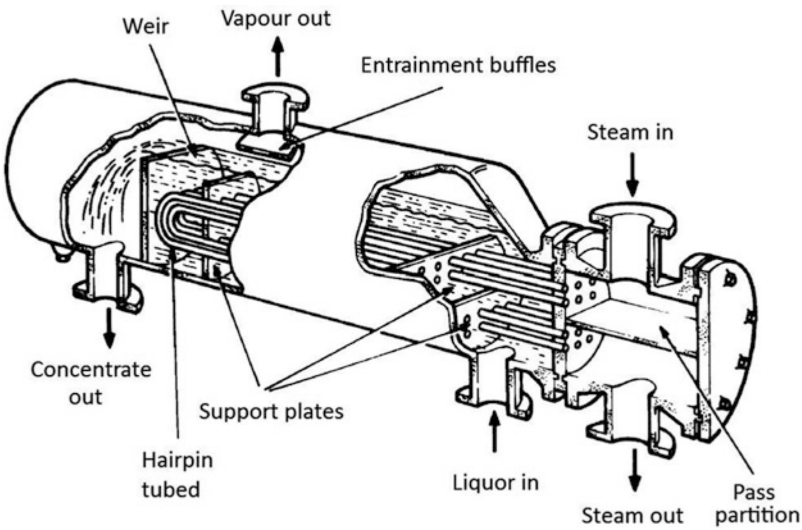


Fig. 10 Scheme of kettle-type reboiler (Collier 1988)

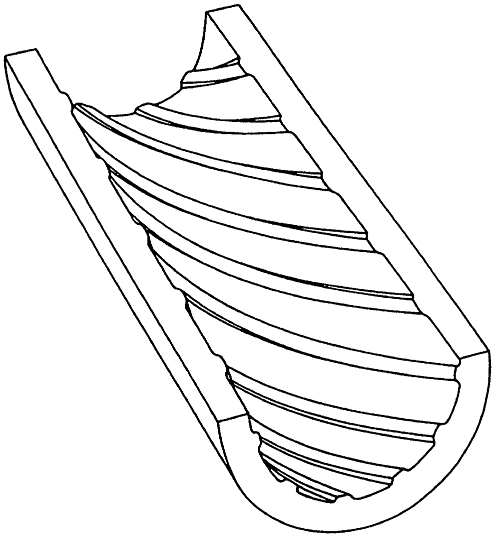


Fig. 11 Low-fin tube for air-conditioning equipment (Kitto and Albrecht 1988)

Fig. 11. For low-fin tubing, the special high-flux surfaces for boiling enhancement are used. For more details, refer to Butterworth (1977, 1988), Saunders (1988), and Shah (1998).

The characteristic features of shell-and-tube heat exchanger are the wide range of temperature and pressure from vacuum, highly defined by material limit. The pressure drop and fluid arrangement can be varied on the shell side within a wide range changing baffle design. Shell-and-tube exchangers tend to be covered by standards from TEMA (1999) and ASME boiler and pressure vessel codes. The three most common types of shell-and-tube exchangers are fixed tube design, U-tube design, and floating head type (Shah 1998; Shah and Sekulic 2003). In all these types, the front-end head is fixed, while the rear-end head can be either fixed or floating, depending on the thermal stresses in the parts of exchanger due to temperature differences.

5.2.3 Spiral Tube Exchanger

Spiral tube heat exchanger consists of spirally wound coils which are fitted in the shell. It allows to install considerable amount of heat transfer surface in an internal space using one or more tubes. Heat transfer enhancement for this case is associated with higher heat transfer coefficient for spiral tube compared with a straight tube. The constructions of spiral tube exchangers are discussed in Shah (1998) and Kirchner (2010). These exchangers can be applied for natural gas liquefaction.

5.3 Plate-Type Heat Exchangers

At the present time, more compact and higher-performance heat exchangers are required for efficient utilization of heat energy and energy saving. In order to satisfy these requirements, plate-type exchangers can be used to increase heat transfer area per unit volume and to improve heat transfer coefficient. A plate-type heat exchanger uses metal or ceramic plates to transfer heat between two fluids. The major advantage of plate-type heat exchanger over tubular heat exchanger where the fluid flows through a large number of channels with complex geometry is the larger ratio of surface area to volume of heat exchanger. Plate-type heat exchangers are usually made from thin plain or corrugated plates providing high heat transfer coefficient for fluid flow in complex shape channels. Some versions of these heat exchangers have surface area from three to four times those of shell-and-tube units (Shah 1982, 1991; Shah and Webb 1983).

5.3.1 Plate Heat Exchanger

Although plate heat exchanger is used primarily for liquid-to-liquid heat transfer, its performance is also good in evaporation and condensation applications. Plate heat exchanger consists of welded or brazing thin corrugated sheets with complex surface texture. The view of a corrugated plate is shown in Fig. 12. Each fluid flows between the sheets alternately, and exchanger has many parallel channels with complex cross section. The resulting flow passages are narrow and highly tortuous and enhance the heat transfer rate in two-phase flow by increasing the shear stress and increasing the level of turbulence. It leads to higher heat transfer coefficient but also increases pressure drop. Specially designed plates are available for condensing and boiling of ammonia and refrigerants, as well as for combined evaporation/condensation applications (Shah 1982, 1991, 1998; Shah and Webb 1983). The disadvantage of these designs is the loss of flexibility due to welding of the plates.

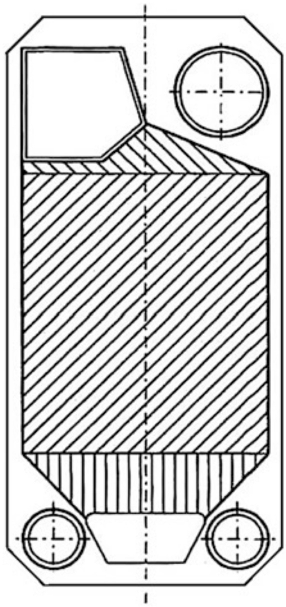


Fig. 12 Scheme of the sheet of plate evaporator (Kirchner 2010)

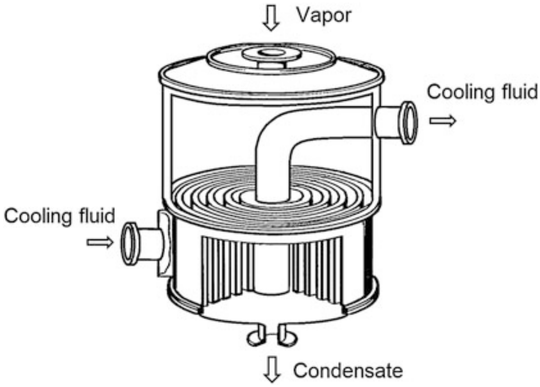


Fig. 13 View of spiral plate condenser

5.3.2 Spiral Plate Exchanger

A spiral plate heat exchanger consists of two sheets of metal wrapped helically around a split mandrel to form a pair of spiral channels for two fluids, as shown in Fig. 13. For sheet spacing, they are provided by welded studs. The main advantage of the spiral plate heat exchanger is its highly efficient use of space. This exchanger is suited as a small condenser or reboiler when it is mounted vertically. It is often used also as a thermosiphon or kettle reboiler; the counter-flow spiral unit is used for condensing applications (Butterworth 1988; Saunders 1988). The scheme of spiral plate condenser is shown in Fig. 13.

5.4 Extended Surface Heat Exchangers

The tubular and plate-type exchangers described previously are the prime surface of heat exchanger, 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 less than $700 m^2/m^3$ (Shah 1991). One of the most common methods to increase the heat transfer surface area and exchanger compactness is extension of the plain surface by fins, depending on the design requirement. Addition of fins can increase the primary surface area by 5 to 12 times, depending on the design. The resulting exchanger is referred to as an extended surface exchanger. Flow interruptions as in the case of offset strip fins and louvered fins may increase the heat transfer coefficient two to four times that for the corresponding plain fin surface. Tube-fin and plate-fin geometries are the two most common types of extended surface heat exchangers (Shah 1998).

5.4.1 Tube-Fin Heat Exchangers

Tube-fin exchangers are employed when one fluid is at a higher pressure and/or has a significantly higher heat transfer coefficient than that of the other fluid. These exchangers are used extensively as condensers and evaporators in air-conditioning and refrigeration applications, as condensers in electric power plants, and as

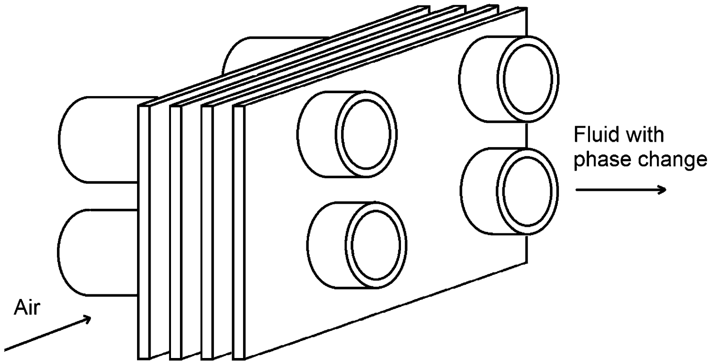


Fig. 14 Tube-fin exchanger for air-conditioning equipment (Pate 1991)

air-cooled exchangers in process and power industries (Pate 1991). The scheme of the segment of tube-fin exchanger for air-conditioning equipment is shown in Fig. 14. In a tube-fin exchanger, round and rectangular tubes typically are 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, adhesive bonding, soldering, brazing, and welding. In a gas-to-fluid exchanger, the heat transfer coefficient on the gas side is generally one order of magnitude lower than that on the fluid with phase-change side. Hence, to have balanced thermal resistance on both sides, fins are used on the gas side to increase surface area and can be plain, wavy, or interrupted (Shah 1998). Tube-fin exchangers usually have less compact than plate-fin units.

5.4.2 Plate-Fin Heat Exchanger

A plate-fin heat exchanger are vacuum brazed and designed for high-temperature and high-pressure applications. They consist of stainless steel or aluminum corrugated plates and two end plates (Butterworth 1988); see Fig. 15. Plate-fin heat exchanger widely used in heat pumps and refrigeration, cryogenic, gas liquefaction, air-conditioning, and waste heat recovery systems. This type of exchanger has corrugated fins between parallel plates referred to as parting sheets.

The plates separate the two phase-change fluids, and the fins form the flow passages for every fluid. Fins are attached to the plates by brazing, soldering, or welding; therefore high fin density can be archived. A cryogenic plate-fin exchanger has about 10% of the volume of an equivalent shell-and-tube exchanger (Reay 1999). Plate fins can be plain, perforated, or interrupted, such as offset strip fins which produce higher heat transfer coefficient than do plain fins. Plate-fin exchangers can be manufactured in different shapes and sizes and can be made from a variety of materials (Shah 1982b, 1991; Shah and Webb 1983; Shah and Focke 1988). The scheme of plate-fin heat exchanger for air-conditioning equipment (Carey and Shah 1988) is shown in Fig. 16.

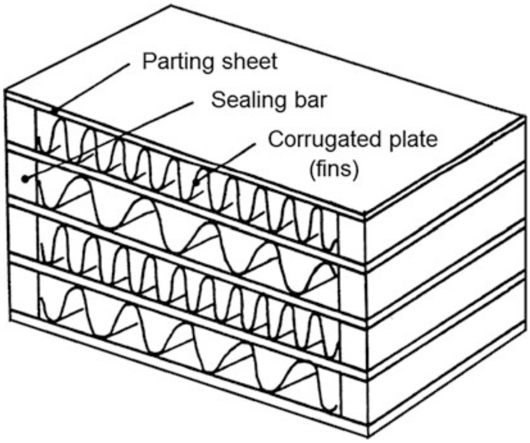


Fig. 15 Plate-fin heat exchanger of the cryogenic plant (Butterworth 1988)

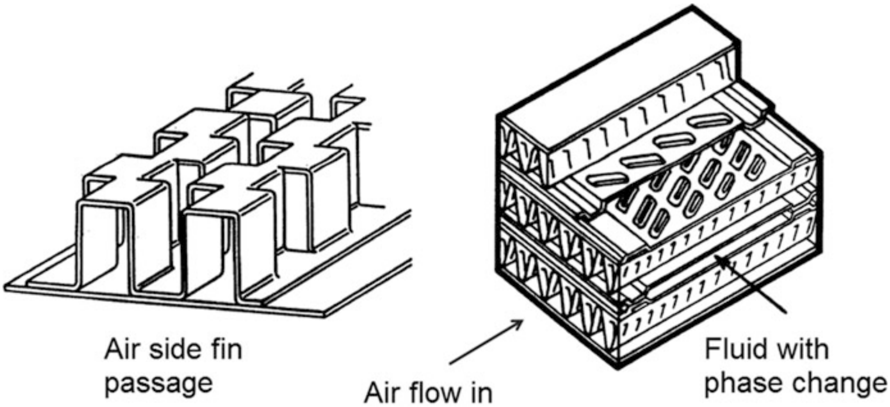


Fig. 16 Plate-fin heat exchanger for air-conditioning equipment (Carey and Shah 1988)

5.5 Microchannel Heat Exchangers

5.5.1 Microchannel Heat Exchanger Applications

Two-phase microchannel heat exchanger refers to the microstructuring heat exchanger with transverse dimension of the channels in the range of hundred microns, which ensures the excellent heat and mass transfer properties with respect to conventional channels with diameter larger than 6 mm. The channels' cross section may have different shapes, but typically it is rectangle. To dissipate heat energy from planner surfaces, flow boiling can be achieved inside multiple parallel channels that are formed in a high-conductivity substrate, named as microchannel heat sink (Tuckerman and Pease 1981). The view of microchannel heat sink is shown in Fig. 17. Flow boiling in this heat exchanger has the potential to meet the large heat dissipation demands of high-power electronics and computing systems at low flow

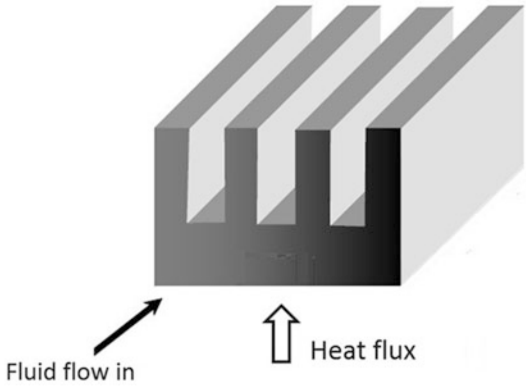


Fig. 17 Two-phase cooled microchannel heat sink for electronic packages

rate of the cooling fluid. High heat flux cooling is required also in many modern applications such as plasma-facing components, high heat-load optical components, laser diode arrays, and X-ray medical devices. However, they face practical challenges which are concerned with large pressure drop and formation of significant flow instability.

One of the important applications of two-phase microchannel heat exchangers is effective thermal management for the development of next-generation computational systems. Nowadays, the advanced electronic components generate heat fluxes higher than 100 W/cm^2, while some future power electronic components, such as high-power laser and electronic radar systems, have been projected to generate heat fluxes over 1000 W/cm^2 (Mudawar 2001; Kandlikar and Bapat 2007). The International Technology Roadmap for Semiconductors 2007 predicts total power dissipation requirements of over 200 W for high-performance packages, junction temperature less than 90 °C, and thermal resistances less than 0,1 °C/W (David et al. 2011). The difficulty achieving both of these requirements increases when the complexities in providing the required cooling in perspective 3D integrated circuits is considered, where logic, communication, and storage are combined together in a single-stacked structure with the heat-generating component deep within the stack. The application of single-phase cooling for this system faced with high-pressure head at high liquid flow rate. One promising solution to reduce the pressure head while providing large heat dissipation is two-phase convective cooling. Phase change also provides small thermal resistance and may improve temperature uniformity in zones with high heat load by maintaining the two-phase fluid at the saturation temperature. Figure 18 shows 3D integrated circuit based on two-phase microchannel evaporator proposed by Koo et al. (2005).

The two-phase microchannel heat exchangers also can be used for many small head load applications including condensers of heat pumps and air-conditioning systems in automobiles (Garimella 2003). These condensers consist of cold plate with multiple parallel microchannels cooled during boiling of a refrigerant, and the plate cools the air flowing across multilouver fins. Garimella and Wicht (1995) demonstrated that microchannel condenser allows us to archive the capacity of

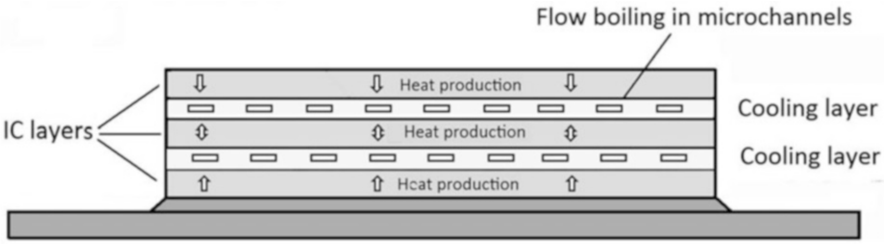


Fig. 18 3D integrated circuit based on microchannel evaporators according to Koo et al. (2005)

18 kW for ammonia–water absorption conditioning system for residential application instead of conventional round tube, flat-fin design.

The important characteristic of microchannel heat exchangers is extremely high heat transfer surface area density, which can be more 10^4 m^2/m^3 (Shah 1991). Therefore microchannel heat exchangers are characterized by two important parameters which are compactness and effectiveness. The compactness is provided by high heat transfer surface area density. It reduces essentially the size of the heat exchanger needed for a given thermal load. Microchannel heat exchangers can be made from diverse materials including metals, silicon, silica, polycarbonate/polyimide, or low-cost plastics. It may be fabricated by variety of processes depending on the channel dimensions and materials used. The methods of the channels fabrication can be divided into two groups (Ohadi et al. 2013). Conventional fabrication technologies include micro-deformation, micro-sawing and micro-milling, and dicing (Alting et al. 2003). Modern microchannel fabrication techniques include MEMS (microelectromechanical system) methods, laser micro-machining, electro-discharge machining, and micro-molding. The application of microfabrication techniques such as LIGA (Ehrfeld 1990) and stereolithography allows us to produce very effective configurations of the flow distribution sections of heat exchanger. The scheme of the fabrication of high aspect ratio resist relief using deep-etch X-ray lithography (LIGA) is shown in Fig. 19. All these methods allow us to fabricate microchannels with near-rectangular cross section but with different surface roughness, which can change considerably the heat transfer regularities.

5.5.2 Thermal Design of Microchannel Heat Exchangers

The key points regarding thermo-hydraulic design for microchannel heat exchangers are the same as described in Sect. 3. However, the development of two-phase microchannel technology requires understanding of thermal aspects of the phase change in microchannels under substantial contribution of capillary forces. The published models for flow boiling heat transfer prediction in microchannels are based either on modification of the models obtained for conventional tube, for example, Kandlikar and Balasubramanian (2004) model, or on the special-purpose models for calculation of heat transfer (Bertsch et al. 2009). The three-zone heat transfer model for elongated bubble flow, when the transient evaporation of thin liquid film around the elongated bubble is essential, was proposed by Thome et al.

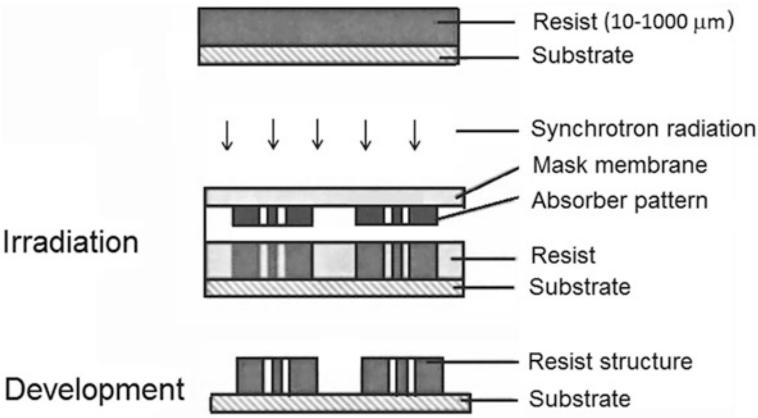


Fig. 19 Fabrication of a high aspect ratio resist relief using deep-etch X-ray lithography (Ehrfeld 1990)

(2004). The Liu and Winterton (1991) model was extended by Kuznetsov (2010) and Kuznetsov and Shamirzaev (2016) to predict better the data for small-size channels. In Liu and Winterton (1991) model, the heat transfer coefficient is calculated as superposition of boiling and convection terms. It was proposed that nucleate boiling is suppressed for extremely thin liquid films, when the diameter of active nucleus becomes comparable with the thermal boundary thickness (Kuznetsov 2010). When nucleate boiling is suppressed, evaporation on liquid film surface becomes the significant mechanism of heat transfer.

For this case, the heat transfer model can be presented as follows:

$$h^2 = (h_{\text{con}}F)^2 + \left(h_{\text{boil}}\Psi_{\text{sup}}S\right)^2 + (h_{ev}E)^2 \tag{19}$$

Here h_{con}, h_{boil}, and h_{ev} are forced convection, nucleate boiling, and liquid film evaporation heat transfer coefficients, accordingly; F and S are the factors of forced convection enhancement and nucleate boiling suppression proposed by Liu and Winterton (1991) as follows:

$$\begin{aligned} F &= \left(1 + x\mathrm{Pr}\left(\rho_{\text{liq}}/\rho_{\text{gas}} - 1\right)\right)^{0.35}, \\ S &= \left(1 + 0.055F^{0.1}\mathrm{Re}_L{}^{0.16}\right)^{-1}, \end{aligned} \tag{20}$$

and Ψ_{sup} is boiling suppression factor introduced by Kuznetsov (2010). Evaporation factor E for elongated bubble flow and transition flow equals to the volume vapor fraction $E = (1 + \rho_{\text{gas}}(1 - x)/\rho_{\text{liq}}x)^{-1}$ and for annular flow $E = 1$ (Kuznetsov and Shamirzaev 2016).

For microchannels with all-liquid flow in the laminar region, corresponding convective heat transfer coefficient h_{con} should be calculated accounting to channel geometry. The correct selection of nucleate boiling model h_{boil} in Eq. (19) is very important for accurate heat transfer prediction, especially for the case of dominant nucleate boiling. Nucleate boiling suppression for thin liquid film was accounted by Kuznetsov (2010) using boiling suppression factor Ψ_{sup} as the multiplier for nucleate boiling suppression term S in Eq. (19). For the rectangular microchannel, it is necessary to account the absence of nucleate boiling suppression near the channel corners. The size of this area can be approximated as a half of the width of the channel's short side and boiling suppression factor Ψ_{sup} is obtained as follows:

$$\psi_{\text{sup}} = \left(\tanh^2\left(2.5 \cdot 10^{-3}\Theta_{\text{sup}}^2\right)(a-b) + 2b\right)/(a+b) \tag{21}$$

Here, $\theta_{\text{sup}} = \left(\frac{y_{vis}}{d_{\tan}}\right)^{0.6} / \left(Bo_x^{0.4} We_{\text{all\_liq}}{}^{-0.08} \Pr^{1/3}\right)$, $y_{\text{vis}} = 5\nu_{\text{liq}} / \sqrt{\tau_w / \rho_{\text{liq}}}$ is thickness of the viscous sublayer in a liquid film, Bo_x is the Boiling number defined via local liquid flow rate, $d_{\tan}$ is diameter of active nucleus based on the Hsu's tangential criteria for nucleation. For calculation of shear stress τ_w, the model of Asali et al. (1985) with entrainment may be used.

Thus, the heat conduction through the thin film on the channel wall could be the possible mechanism of heat transfer in the case of nucleate boiling suppression. To determine heat transfer coefficient of an evaporating liquid film, the turbulent liquid film model in the annular flow with calculation of film thickness using shear stress for wavy film (Asali et al. 1985) may be used with account for limitation of heat transfer due to the film rupture.

High heat transfer surface area density provides thin film thickness and low thermal resistance in microchannel condenser. The models for flow condensing heat transfer prediction were presented and discussed in Kandlikar et al. (2006). These models presented in Kandlikar et al. (2006) together with flow pattern map may be used for prediction of the overall heat transfer coefficient during condensation in microchannel condensers.

6 Cross-References

- ▶ Boiling and Two-Phase Flow in Narrow Channels
- ▶ Boiling on Enhanced Surfaces
- ▶ Compact Heat Exchangers
- ▶ Energy Efficiency and Advanced Heat Recovery Technologies
- ▶ Evaporative Heat Exchangers
- ▶ Film and Dropwise Condensation
- ▶ Flow Boiling in Tubes
- ▶ Fundamental Equations for Two-Phase Flow in Tubes
- ▶ Heat Exchanger Fundamentals: Analysis and Theory of Design
- ▶ Heat Exchangers Fouling, Cleaning, and Maintenance
- ▶ Heat Pipes and Thermosyphons
- ▶ Heat Transfer Media and Their Properties

► Internal Annular Flow Condensation and Flow Boiling: Context, Results, and Recommendations
► Introduction and Classification of Heat Transfer Equipment
► Mixture Boiling
► Nucleate Pool Boiling
► Process Intensification
► Single- and Multiphase Flow for Electronic Cooling
► Transition and Film Boiling

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