Chapter 6
Heat Transfer Equipment

6.1 Introduction

Heat transfer equipment is used in most food processing operations as an important part of the manufacturing or preservation processes. The transfer of energy to or from food materials during processing requires special equipment, which is designed and operated on the basis of the engineering principles of heat transfer and the experience and practice of food process engineering.

This chapter reviews the heat transfer coefficients and correlations of importance to food systems, and it describes briefly some heat exchangers and heaters used in food processing. Basic heat transfer equipment, used in industrial food processing, is described also in the following chapters of this book: 7 (evaporators), 8 (dryers), 9 (coolers, freezers), and 10 (blanchers, pasteurizers, sterilizers). Table 6.1 shows the basic heat transfer equipment used in food processing and food preservation operations.

Table 6.2 shows some special heat transfer equipment, described in this chapter.

Food heating processes may involve chemical, biochemical, and biological changes of the food material that must be considered simultaneously with the physical heating process. The main purpose of cooking, baking, roasting, and frying processes is to improve the eating quality of the food products.

6.2 Heat Transfer Coefficients

The operation of heating equipment is based on the transfer of heat from a heating medium to a food material, usually by forced convection. Natural convection, contact, and radiation heating may also be involved in some applications. The cooling equipment operates on the same principles, but heat is transferred from the food material to a cooling medium, which can also be a cold food stream.

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Table 6.1 Basic heat transfer equipment of food processing

Equipment	Function
Evaporators	Concentration of liquid foods
Dryers	Drying/dehydration of foods
Coolers	Preservation of fresh foods
Freezers	Food preservation
Blanchers	Inactivation of enzymes
Pasteurizers	Inactivation of enzymes and microbes
Sterilizers	Inactivation of microbes and microbial spores

Table 6.2 Special heat transfer equipment used in food processing

Equipment	Function
Ovens	Baking, cooking of foods
Roasters	Thermal treatment of foods
Fryers	Deep-fat or dry frying of foods
Infrared heaters	Heating, drying of foods
Microwave heaters	Heating, thawing, or drying of foods

The heat transport within food materials is expressed by the two basic heat transport properties, thermal conductivity (λ, W/m K) and thermal diffusivity (α, m^2/s), defined by the Fourier equations for one-dimensional (x) transfer:

$$q/A = \lambda(\Delta T/x) \tag{6.1}$$

and

$$\partial T/\partial t = \alpha\left(\partial^2 T/\partial x^2\right) \tag{6.2}$$

where (q/A) is the heat flux (W/m^2) and (ΔT/x) is the temperature gradient (K/m) in the (x) direction.

Data on the thermal transport properties of foods are presented in tables, databanks, and books, such as Rahman (1995) and Saravacos and Maroulis (2001).

The thermal diffusivity (α) is usually calculated from the more common thermal conductivity (λ), using the equation:

$$\alpha = \lambda/\rho C_{\mathrm{p}} \tag{6.3}$$

where ρ is the density (kg/m^3) and C_{p} is the specific heat (J/kg K) of the material.

Densities, specific heats, and other physical properties of foods are found in food properties books, e.g., Rahman (1995). Some typical physical properties of foods useful in the design and operation of heat exchangers and other food processing equipment are given in Appendix B of this book. Table 6.3 shows some typical thermal transport properties of foods and engineering materials, useful for heat

Table 6.3 Typical thermal conductivities (λ) and thermal diffusivities (α) of foods and engineering materials

Material	λ (W/m K)	α ($\times 10^{-7}$ m^2/s)
Air	0.025	190.0
Water	0.62	1.0
Porous food	0.04–0.20	1.1
Wet food	0.50	1.3
Frozen food	1.5	5.0

transfer equipment (Rahman 1995; Saravacos and Maroulis 2001; Perry and Green 1997).

Heat exchangers and heat transfer equipment are characterized by the (film) heat transfer coefficient (h, W/m^2 K) and the overall heat transfer coefficient (U, W/m^2 K), defined by the equations:

$$q/A = h\Delta T \tag{6.4}$$

and

$$q/A = U\Delta T \tag{6.5}$$

In (6.4), (ΔT) is the temperature difference between the bulk of the fluid and the heat transfer surface. Equation (6.5) is applied to heat transfer through the wall of a heat exchanger, and (ΔT) refers to the overall temperature difference between the two fluids, e.g., heating medium and food material. The units of (ΔT) are degrees Kelvin (K), which are identical with degrees Celsius (°C).

The overall resistance to heat transfer ($1/U$) through a wall of thickness (x) and thermal conductivity (λ) is given by the equation:

$$1/U = 1/h_1 + x/\lambda + 1/h_2 \tag{6.6}$$

where h_1 and h_2 are the (film) heat transfer coefficients of the two sides (1, 2) of the heat transfer wall.

Equation (6.6) applies to clean flat heat transfer surfaces and approximately to tubular surfaces of diameter larger than 25 mm (1 in.). For tubes of smaller diameter, the inner and outer diameters (d_i, d_o) must be considered, and (6.6) becomes

$$1/U_\mathrm{i} = 1/h_\mathrm{i} + (d_\mathrm{i}/d_\mathrm{L})(x/\lambda) + (d_\mathrm{i}/d_\mathrm{o})(1/h_\mathrm{o}) \tag{6.7}$$

where (d_L) is the log mean diameter, defined as

$$d_\mathrm{L} = (d_\mathrm{o} - d_\mathrm{i})/\ln(d_\mathrm{o}/d_\mathrm{i}) \tag{6.8}$$

In (6.7), the tube wall thickness (x) is calculated as $x = (d_\mathrm{o} - d_\mathrm{i})/2$.

In practice, the heat transfer surfaces of the heat exchangers are not clean, due to fouling, i.e., deposits which increase significantly the thermal resistance. Thus, a

Table 6.4 Typical film heat transfer coefficients (h)

Heat transfer system	h (W/m^2 K)
Air-drying	20–100
Baking oven	30–200
Air cooling	10–60
Air freezing	20–100
Water in pipe flow	5000
Falling film of water	4000
Condensing steam	10,000

fouling resistance term should be added to the overall resistance (6.6) and (6.7), as discussed in Sect. 6.4.

Table 6.4 shows some typical heat transfer coefficients (h) in food heat exchange systems (Rahman 1995; Saravacos and Maroulis 2001).

6.3 Empirical Correlations of (*h*)

6.3.1 General Correlations

Correlations of heat transfer data are useful for estimating the (film) heat transfer coefficient (h) in various processing equipment and operating conditions. These correlations contain, in general, dimensionless members, characteristic of the heat transfer mechanism, the flow conditions, and the thermophysical and transport properties of the fluids. Table 6.5 lists the most important dimensionless numbers used in heat transfer operations (Saravacos and Maroulis 2001).

The Reynolds number ($Re = uL/\nu$) is used widely in almost all correlations. In this number, the velocity (u) is in (m/s), the length (L) is in (m), and the kinematic viscosity or momentum diffusivity ($\nu = \eta/\rho$) is in (m^2/s). The length (L) can be the internal diameter of the tube, the equivalent diameter of the non-circular duct, the diameter of a spherical particle or droplet, or the thickness of a falling film. In noncircular geometries, the characteristic length (L) is taken as twice the hydraulic diameter ($L = 2d_H$) of the flow channel, which is twice the hydraulic radius of the duct ($d_H = 2r_H$), defined as $r_H = (A/\mathrm{WP})$, where (A) is the cross-sectional area and (WP) is the “wetted” perimeter.

Some dimensionless numbers are used also in mass transfer correlations, i.e., Bi, St, and j_M. In these numbers, the heat transfer coefficient (h) is replaced by the mass transfer coefficient (k_c).

Table 6.6 shows some heat transfer correlations of general application. For natural convection, the parameters (a) and (m) characterize the various shapes of the equipment and the conditions of the fluid (McAdams 1954; Perry and Green 1997; Geankoplis 1993; Saravacos and Maroulis 2001).

Table 6.5 Dimensionless numbers in heat and mass transfer calculations

Number	Applications
Reynolds $Re = uL/\nu$	Flow processes
Nusselt $Nu = hL/\lambda$	Heat transfer
Prandtl $Pr = \nu/\alpha$	Heat transfer
Grashof $Gr = L^3 g(\Delta\rho/\rho)/\nu^2$	Free convection
Graetz $Gz = GAC_p/\lambda L$	Heat transfer
Biot $Bi = hL/\lambda$	Heat transfer
Stanton $St = h/GC_p$	Heat transfer
Schmidt $Sc = \nu/D$	Mass transfer
Heat transfer factor $j_H = StPr^{2/3}$	Heat transfer
Mass transfer factor $j_H = StSc^{2/3}$	Mass transfer

A interfacial area (m^2), L length (m), α thermal diffusivity (m^2/s), C_p specific heat (J/kg K), g acceleration of gravity (8.91 m/s^2), $G = u\rho$ mass flow rate (kg/m^2 s), h heat transfer coefficient (W/m^2 K), η viscosity (Pa s), ρ density (kg/m^3), D mass diffusivity (m^2/s), $\nu = \eta/\rho$ kinematic viscosity (m^2/s), u velocity (m/s)

Table 6.6 General heat transfer correlations

Heat transfer system	Correlation
Natural convection	$Nu = a(GrPr)^m$
Laminar inside tubes	$Nu = 1.86[RePr(d/L)]^{1/3}(\eta/\eta_w)^{0.14}$
Turbulent inside long tubes	$Nu = 0.023Re^{0.8}Pr^{1/3}(\eta/\eta_w)^{0.14}$
Parallel to flat plate (laminar)	$Nu = 0.664Re^{0.5}Pr^{1/3}$
Parallel to flat plate (turbulent)	$Nu = 0.0366Re^{0.8}Pr^{1/3}$
Flow past single sphere	$Nu = 2.0 + 0.60Re^{0.5}Pr^{1/3}$

Dimensionless numbers defined in Table 6.5. a and m, parameters of natural convection characteristic of the system (Perry and Green 1997); L, d length and diameter of tube. Long tubes $(L/d) > 60$.

The ratio of tube diameter to tube length (d/L) is important in the laminar flow ($Re < 2100$), but it becomes negligible in the turbulent flow in long tubes ($L/d > 60$). For shorter tubes, the ratio (d/L) should be included in the correlation.

The viscosity ratio (η/η_w) refers to the different viscosity in the bulk of the fluid (η) and at the tube wall (η_w). This ratio becomes important in highly viscous fluids, like oils, in which the viscosity drops sharply at the high wall temperatures, increasing the heat transfer coefficient.

Several other correlations have been proposed in the literature for different heat transfer in fluid systems, like flow outside tubes, flow in packed beds, etc.

The heat transfer coefficients of condensing vapors have been correlated to the geometry of the tubes and the properties of the liquid film or droplets. Very high heat transfer coefficients are obtained by dropwise condensation. Most liquid foods are non-Newtonian fluids, and their apparent viscosity (η_a) is related to the rheological constants (K, n) of the power-law model (see Chap. 3):

$$\eta_a = K\gamma^{n-1} \tag{6.9}$$

The shear rate γ (s^{-1}) is defined by the equation (Rao 1999):

$$\gamma = 8u/d \tag{6.10}$$

where u (m/s) is the fluid velocity and d (m) is the internal tube diameter.

Most non-Newtonian liquid foods are pseudoplastic, i.e., $n < 1$, and their apparent viscosity decreases as the shear rate (velocity, agitation) is increased (shear-thinning fluids). Therefore, the heat transfer coefficient will increase at higher flow velocities or speeds of agitation.

The empirical correlation for laminar flow of power-law fluids is similar to the correlation given in Table 6.6:

$$Nu = 2.0[(3n+1)/4n]^{1/3} Gz^{1/3} (K/K_w)^{0.14} \tag{6.11}$$

The Graetz (Gz) number is defined as $Gz = (GAC_p)/\lambda L$ (Table 6.5). The viscosity ratio (η/η_w) is replaced by the ratio of the flow consistency index at the bulk/wall (K/K_w), and the flow behavior index (n) is included in the correlation.

6.3.2 Simplified Equations for Air and Water

The heat transfer coefficient of air and water in some important operations can be estimated from simplified dimensional equations, applicable to specific equipment geometries and system conditions (Perry and Green 1997; Geankoplis 1993):

(a) Natural convection of air:

$$\text{Horizontal tubes,} \quad h = 1.42(\Delta T/d_o)^{1/4} \tag{6.12}$$

$$\text{Vertical tubes,} \quad h = 1.42(\Delta T/L)^{1/4} \tag{6.13}$$

(b) Air in drying (constant rate):

$$\text{Parallel flow,} \quad h = 0.0204G^{0.8} \tag{6.14}$$

$$\text{Perpendicular flow,} \quad h = 1.17G^{0.37} \tag{6.15}$$

(c) Falling films of water:

$$h = 9150\Gamma^{1/3} \tag{6.16}$$

(d) Condensing water vapors:

$$\text{Horizontal tubes,}\quad h = 10,800/\left[(Nd_o)^{1/4}(\Delta T/d_o)^{1/3}\right] \tag{6.17}$$

$$\text{Vertical tubes,}\quad h = 13,900/\left[L^{1/4}(\Delta T)^{1/4}\right] \tag{6.18}$$

where (ΔT) is the temperature difference (K), d_o is the outside diameter (m), L is the length (m), G is the mass flow rate (kg/m^2 s), Γ is the “irrigation” flow rate of the film (kg/m s), and N is the number of horizontal tubes in a vertical plane.

6.3.3 Heat Transfer Factor

The heat transfer factor of various heating processes (j_H) is related to the Reynolds number of the system (*Re*) by the empirical correlation (Saravacos and Maroulis 2001):

$$j_H = mRe^n \tag{6.19}$$

where (m) and (n) are characteristic constants of the equipment/material system.

Regression analysis of compiled literature data on various heating operations of foods has yielded the following generalized correlation:

$$j_H = 0.344Re^{-0.423} \tag{6.20}$$

It is thus possible to estimate an approximate value of the heat transfer coefficient (h) for a given flow condition (*Re*), using the equation:

$$j_H = StPr^{2/3} = \left[h/\left(u\rho C_p\right)\right]Pr^{2/3} \tag{6.21}$$

As an illustration, the heat transfer coefficient (h) of water, flowing in a circular tube of internal diameter 25 mm at 2 m/s and 20 °C, is calculated as follows: The thermophysical and transport properties of water at 20 °C are $\rho = 1000$ kg/m^3, $\eta = 0.001$ Pa s, $C_p = 4180$ J/kg, and $\lambda = 0.62$ W/m K. Thus, $Re = (0/025 \times 2 \times 1000/0.001) = 50{,}000$, i.e., the flow will be turbulent. The heat transfer factor will be $j_H = 0.344 \times (50{,}000)^{-0.423} = 0.0032$. The Prandtl number will be $Pr = C_p\eta/\lambda = 4180 \times 0.001/0.62 = 5.74$, and $Pr^{2/3} = 3.22$. From (6.21), the heat transfer coefficient will be approximately $h = 0.0032 \times 1000 \times 2 \times 4180/3.22 = 8308$ W/m^2 K.

It should be noted that the classical correlation for (h) in turbulent pipe flow (Table 6.6) yields $Nu = 0.023 \times (50{,}000)^{0.8} \times (5.54)^{0.33} = 235.1$ and $h = 235.1 \times 0.62/0.025 = 5828$ W/m^2 K.

For a viscous food product of apparent viscosity $\eta_a = 1$ Pa s, flowing in the same tube at the same velocity of the previous example, $Re = 50{,}000 \times 0.001 = 50$, i.e., laminar flow.

The Prandtl number will be about $Pr = 5.74 \times (1/0.001) = 5740$ and $Pr^{2/3} = 330$. The heat transfer factor will be $j_H = 0.0032 \times (1/1000)^{-0.423} = 0.06$ and the heat transfer coefficient $h = 0.06 \times 1000 \times 2 \times 4180/330 = 1520$ W/m^2 K.

6.4 Heat Exchangers

The design of heat exchangers in the process industries is described in general chemical engineering books, such as Perry and Green (1997) and Walas (1988). The design procedures for heating gases and liquids have been standardized and computer packages are available for computer-aided design (CAD).

Several types of heat exchangers are used in food processing, suited to the diverse requirements of heating, cooling, sterilizing, baking, etc., of various food products. Although each type of the major equipment is described briefly in the specific applications of this book, the general principles of heat exchangers are outlined in this section.

Heat exchangers used in refrigeration and freezing equipment, e.g., evaporators and condensers of refrigerants, are discussed in detail in Chap. 9.

The heat load of a heat exchanger (q, kW) is the power required to heat the product (m, kg/s) by a temperature difference (ΔT, K), according to the equation:

$$Q = mC_p\Delta T \tag{6.22}$$

where C_p (kJ/kg K) is the specific heat of the product.

6.4.1 *Overall Heat Transfer Coefficients*

The design and operation of heat exchangers is based on the overall heat transfer coefficient (U), which is defined by the generalized form of (6.5):

$$q/A = U\Delta T_L \tag{6.23}$$

where U is the overall heat transfer coefficient, A is the heat transfer area, and ΔT_L is the logarithmic mean temperature difference, defined in (6.26).

The overall heat transfer coefficient (U) is calculated from the overall thermal resistance of the heat exchanger ((6.6) for flat surfaces), including the fouling resistance (FR):

$$1/U = 1/h_1 + x/\lambda + 1/h_2 + (\mathrm{FR}) \tag{6.24a}$$

For tubes of diameter smaller than 25 mm, (6.7) becomes:

$$1/U_\mathrm{i} = 1/h_\mathrm{i} + (d_\mathrm{i}/d_\mathrm{L})(x/\lambda) + (d_\mathrm{i}/d_\mathrm{o})(1/h_\mathrm{o}) + (\mathrm{FR}) \tag{6.24b}$$

The overall heat transfer coefficient (U_i) refers to the internal surface of the tube, which represents the major thermal resistance, since the food material flows usually inside the tubes and it creates most of the fouling. The heating (or cooling) medium flows outside the tubes, and it is characterized by higher heat transfer coefficients and negligible fouling. When the major thermal resistance is in the outside surface, the coefficient (U_i) is replaced by the (U_o) and the diameters (d_i, d_o) in (6.24a) and (6.24b) are switched.

The fouling factor (FR) or the corresponding "fouling coefficient" $h_\mathrm{f} = 1/(\mathrm{FR})$ is difficult to predict accurately, and for this reason, empirical overall heat transfer coefficients (U) are used in practice. Reliable (U) values are obtained in pilot plant or industrial-scale measurements of heat transfer rates on similar equipment, processing similar products under similar operating conditions. Calculation of the (film) heat transfer coefficients of the product side (h_i) from experimental values of (U), using (6.23) or (6.24a) and (6.24b) and assuming a reasonable fouling resistance (FR), is useful for comparison with empirical correlations of (h).

In most food heat exchangers, stainless steel is used, which has a comparatively low thermal conductivity ($\lambda = 15$ W/m K), and the thermal resistance of the wall (x/λ) can become significant, particularly in thick-walled tubes or other heat transfer equipment. As an example, the wall resistance for a wall of 3-mm thickness will be $(x/\lambda) = 0.003/15 = 0.0002$ $\mathrm{m^2}$ K/W. The heating medium is normally saturated steam with a typical heat transfer coefficient $h_\mathrm{o} = 10{,}000$ W/$\mathrm{m^2}$ K and a heat resistance of $h_\mathrm{o} = 0.0001$ $\mathrm{m^2}$ K/W. For a moderate fouling resistance $\mathrm{FR} = 0.0002$ $\mathrm{m^2}$ K/W, and an experimentally measured overall heat transfer coefficient $U = 2000$ W/$\mathrm{m^2}$ K, the product side (h_i) is calculated as follows from (6.24a) and (6.24b): Overall thermal resistance $(1/U) = 0.0005 = 1/h_\mathrm{i} + 0.0002 + 0.0001$, and therefore, $(1/h_\mathrm{i}) = 0.0002$ $\mathrm{m^2}$ K/W or $h_\mathrm{i} = 5000$ W/$\mathrm{m^2}$ K.

Table 6.7 shows some typical overall heat transfer coefficients of various heat exchangers used in food processing, under normal operating conditions, i.e., not excessive fouling.

Table 6.7 Typical overall heat transfer coefficients (U) in heat exchangers

Heat exchange system	U (W/$\mathrm{m^2}$ K)
Steam/water	2500
Water/water	2000
Water/liquid food	1500
Water or steam/viscous food	1000
Water or steam/vegetable oil	800
Water or steam/air (gas)	100
Air (gas)/air (gas)	30

6.4.2 Fouling of Heat Exchangers

Most of the food liquids tend to foul the surfaces of heat exchangers, used in the food industry, reducing significantly the heat transfer rate and damaging the food quality. Fouling is caused by precipitation of partially soluble salts (scaling), physicochemical reactions of food components (proteins, carbohydrates), precipitation of suspended particles, crystallization (freezing) processes, biological growth, and corrosion reactions (Hallstrom et al. 1988; Fryer 1997; Botts 1995).

Fouling forms an insulating solid layer on heat transfer surfaces, like food heaters and evaporators; it decreases the flow rates, increasing the pressure drop; and it provides a substrate for potential microbial growth. Most of the experimental work on fouling has been done on milk, which contains heat-sensitive components that are deposited on heated surfaces, e.g., milk proteins (lactoglobulin) and calcium/phosphate salts (milkstone).

Fouling is a two-step process, i.e., induction and fouling (Fryer 1997). Heat transfer equipment should be designed to reduce adhesion, control high temperature, and minimize residence time. Adhesion of particles to the heat transfer surface is prevented by proper hydrodynamic design (smooth surface) and high shear rates (high velocities or agitation speeds). High wall temperatures should be avoided, because they increase supersaturation of the salts; they also increase protein precipitation.

The major process variables affecting fouling are velocity and temperature. Additional fouling factors include concentration and characteristics of the foulant and equipment design, materials, and surface properties (Bott 2001).

The fouling resistance (FR) of (6.24a) and (6.24b) is difficult to determine accurately, and only approximate values have appeared in the literature. Table 6.8 shows some approximate values of (FR) for industrial fluids (Perry and Green 1997). As mentioned earlier, experimental data on the overall heat transfer coefficient (U) are preferable, if available.

The fouling rate of a heat exchanger may be determined by estimating the overall heat transfer coefficient (U) and the pressure drop (Δp) as a function of time. The operation of the equipment must be interrupted and the heat exchange surface cleaned, when fouling exceeds an acceptable level. Empirical cleaning techniques have been developed for the various types of equipment. The operating and cleaning times must be optimized to reduce the operating cost. Recent advances in simulation, computational fluid dynamics (CFD), and surface science and engineering may reveal the mechanism of fouling and improve the cleaning operations.

The scheduling of heat exchanger operation and cleaning to mitigate (reduce) fouling can be optimized (O'Donnell 2001). Modeling and simulation of fouling of plate heat exchangers (PHE), related mostly to milk processing, can be used to optimize the operating and cleaning cost (Georgiadis et al. 1998a, b).

Table 6.8 Approximate fouling resistances (FR) of industrial fluids

Fluid	FR (m^2 K/W)
Clean steam	0.00005
Distilled water	0.0001
Seawater	0.0002
Vegetable oils	0.0005
Polymer solutions	0.0010
Inorganic gases	0.0015

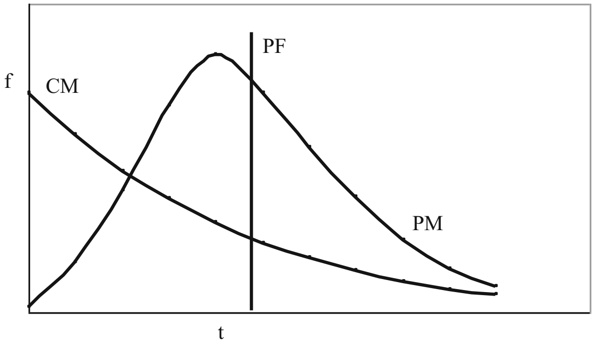


Fig. 6.1 Residence time distribution in continuous flow equipment: *f* frequency of distribution, *t* time, *PF* plug flow, *CM* complete mixing, *PM* partial mixing

6.4.3 Residence Time Distribution

The residence time distribution in heat exchangers (heaters, coolers, evaporators, dryers, etc.) is important for the quality of the food product. The product should remain in the heat transfer equipment only for the minimum time required to accomplish the desired effect, e.g., sterilization, concentration, etc. Any additional residence time is undesirable, because the product may suffer heat damage to the quality and the nutritive value.

Three types of distribution are possible in continuous flow heat exchangers: (a) plug flow, where the product flows through and exits the equipment without mixing; (b) complete mixing, where the product is mixed thoroughly in the equipment and in exits gradually; and (c) intermediate mixing, where the product is partially mixed and it exits gradually from the equipment (Fig. 6.1).

The mean residence time (t_m, s) in a continuous flow equipment is related to the holdup volume (V, m^3), the product flow rate (m, kg/s), and the density of the product (ρ, kg/m^3) by the equation:

$$t_m = V\rho/m \tag{6.25}$$

It is obvious that, for a given product flow rate, short residence time is obtained with small holdup volume. Small holdup volumes are also desirable for the effective cleaning of the equipment.

6.4.4 Tubular Heat Exchangers

The simplest heat exchanger is the double-pipe type, which consists of two concentric pipes with the cold stream flowing usually in the inner tube and the heating medium in the annulus. Two basic operations are possible, cocurrent and countercurrent flow (Fig. 6.2).

The temperature differences, which characterize the operation of the heat exchangers (6.5), are defined at the two ends of the HE as follows (Fig. 6.2):

$$\text{Cocurrent HE}: \Delta T_1 = (T_{h1} - T_{c1}) \quad \text{and} \quad \Delta T_2 = (T_{h2} - T_{c2})$$
$$\text{Countercurrent HE}: \Delta T_1 = (T_{h2} - T_{c1}) \quad \text{and} \quad \Delta T_2 = (T_{h1} - T_{c2})$$

When the two temperature differences (ΔT_1 and ΔT_2) are not the same or not very close to each other, the overall temperature difference (ΔT) of (6.5) is taken as the log mean temperature difference (ΔT_L) is used, defined as follows:

$$\Delta T_L = (\Delta T_1 - \Delta T_2)/\ln(\Delta T_1/\Delta T_2) \tag{6.26}$$

When the two temperature differences (ΔT_1, ΔT_2) are equal or close to each other, the arithmetic temperature difference $\Delta T_M = (\Delta T_1 + \Delta T_2)/2$ should be used, instead of the (ΔT_L).

Tubular heat exchangers are suitable for heating pulp-containing fruit juices, which cannot be handled easily by the popular PHE. A preassembled double-pipe

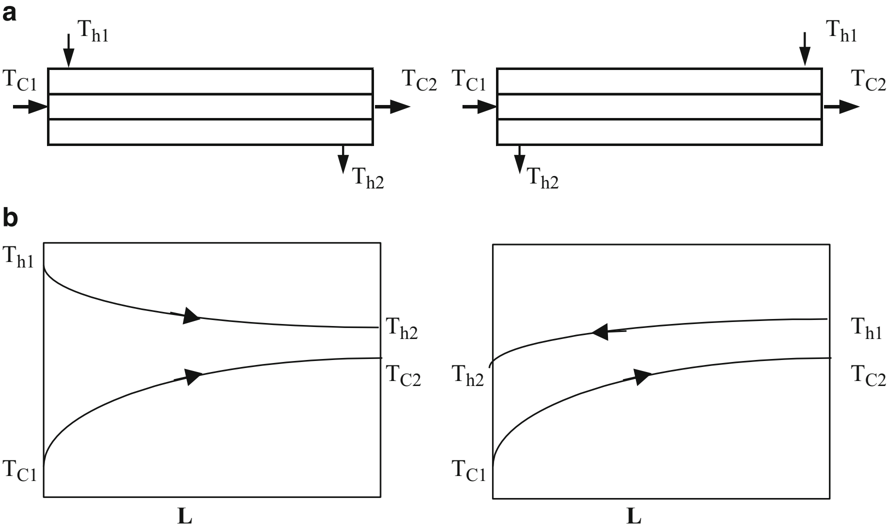


Fig. 6.2 Double-pipe heat exchangers. (**a**) Cocurrent and (**b**) countercurrent. *T* temperature, *c* cold stream, *h* hot stream, *1* entrance, *2* exit, *L* length

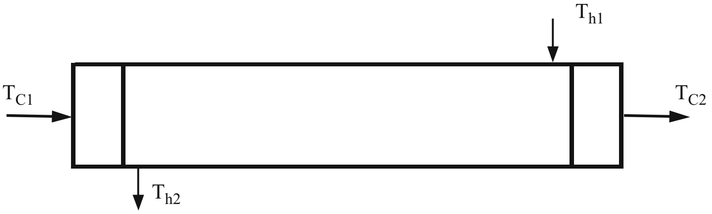


Fig. 6.3 Schematic diagram of a countercurrent shell and tube heat exchanger. *h* heating medium, *c* cold stream, *1* entrance, *2* exit

unit, with diameters 1.5 and 2.5 in. (38 and 65 mm), pasteurizing about 7.5 t/h of pulpy orange juice, was described by Youngworth and Swientek (1987).

The double-pipe heat exchangers have the disadvantage of limited heat transfer surface, which led to the shell and tube heat exchangers, used widely in the chemical process industry. Figure 6.3 shows a diagram of a countercurrent shell and tube heat exchanger, which consists of a bundle of parallel tubes enclosed in a shell. In heating applications, the cold stream flows through the tubes with the heating medium (e.g., saturated steam) in the shell. The heat exchanger can be operated in cocurrent, countercurrent, or cross flow. Temperature differences are calculated following the notation of the double-pipe heat exchangers (Fig. 6.2).

Large heat transfer surfaces can be created by assembling many tubes of small diameter in a shell. The tubes are fixed on two tube sheets, creating two heads at the ends of the heat exchanger. Two or more tube passes may be used to increase the residence time of the tube fluid in the heat exchanger.

The shell and tube heat exchangers are less expensive than the other types of food heat exchangers, especially when high heat loads are transferred. They can be operated at higher temperatures and pressures than the PHE. However, they cannot be cleaned and sanitized easily, like the PHE.

The design and applications of the shell and tube heat exchangers are found in the Chemical Engineering literature, as in Perry and Green (1997), Walas (1988), and Bhatia and Cheremisinoff (1980). Detailed specifications on the dimensions of tubes and shells and on the construction of heat exchangers can be found in technical publications of suppliers of heat exchange equipment, such as the Tubular Exchangers Manufacturers Association (TEMA 1978).

Shell and tube heat exchangers are the basic heat transfer units in evaporation (Chap. 7) and refrigeration (Chap. 9).

The design of tubular and other heat exchangers considers not only the heat transfer rate but also the pressure drop in the system. The heat transfer coefficients tend to increase as the tube diameter is reduced, while the opposite effect is observed with the pressure drop. The heat transfer coefficient (h), according to the empirical correlations, is proportional to the fluid velocity $u^{0.33}$ in the laminar flow or to $u^{0.8}$ in the turbulent flow. The pressure drop (Δp) is proportional to the velocity (u) in the laminar flow or to about the velocity $(u)^2$ in the turbulent flow. The optimum operation of a heat exchanger should combine the highest heat transfer coefficient with a minimum pressure drop. The pressure drop is translated into power requirement (kW) for pumping the fluid through the heat exchanger.

The active length (L) of a heat exchanger can be estimated from the following generalized equation, used for chemical engineering equipment, such as distillation and absorption columns:

$$L = (\mathrm{LTU})(\mathrm{NTU}) \tag{6.27}$$

where length of a transfer unit (LTU) = $(mC_p)/(\pi dU)$, and number of transfer units (NTU) = $(UA)/(mC_p)$. The pressure drop in a heat exchanger can be expressed on the basis of the number of transfer units, i.e., Δp/(NTU) in units of bar/(NTU).

Spiral-tube heat exchangers, with the product inside the tube and the heating medium (e.g., condensing steam) in the outside jacket, are used for the fast heating of liquids. High heat transfer coefficients can be achieved, since turbulence is obtained at lower (Re) numbers. Spiral tubes are suitable for fast heating of liquids in UHT aseptic sterilizers (Chap. 10).

The helical (coiled) heat exchangers have provided some benefits such as compactness, high heat transfer rate, and improved mixing within the tube. Experimental heat transfer coefficients in the range of 3–9 kW/m^2 K I were obtained at temperatures 90–150 °C, using water and pseudoplastic solutions (Coronel et al. 2000).

In a variation of coil heat exchanger, grain products are forwarded upward by vibration. The equipment may be used in a wide range of heating applications, from simple heating up to drying. It requires limited ground space and its capacity is about 1.5 t/h, which, depending on the product and process, lasts 5–25 min. The operational temperature may come up to 250 °C (Fig. 6.4).

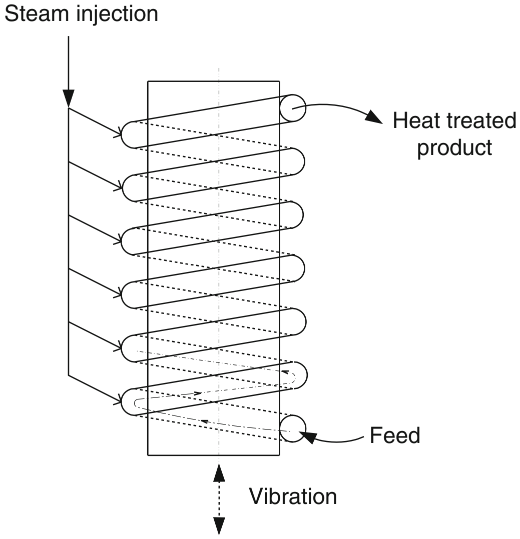
Concentric tube heat exchangers consist of double, triple, or more concentric tubes with associated entry and exit ports for the product and the heating/cooling medium (Lewis and Heppell 2000). Corrugated, instead of straight, tubes may be used to increase turbulence and heat transfer rate. In three or more concentric tubes, the heating/cooling medium and the product flow in alternate annuli and give the largest heat transfer area. They may be used in the final heating stage of UHT sterilizers.

6.4.5 *Plate Heat Exchangers*

The PHE are used widely in the food industry because of their high thermal efficiency, the compactness, and the hygienic and cleaning design. Higher heat transfer coefficients (h) are obtained, due to the geometry of PHE, which promotes turbulent flow at low Reynolds numbers (Alfa-Laval 1969, 1971).

They consist of closely spaced thin plates, clamped together in a frame. A thin gasket seals the plates round their edges. The liquid product and the heating/cooling medium flow alternately in cocurrent or countercurrent fashion (Figs. 6.5 and 6.6).

Fig. 6.4 Vibrating coil heat exchanger



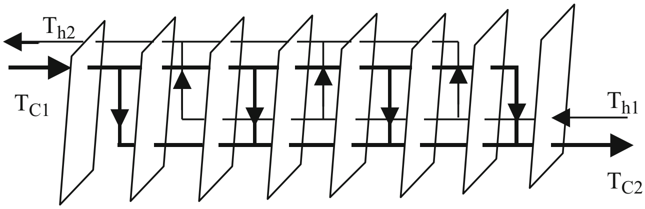
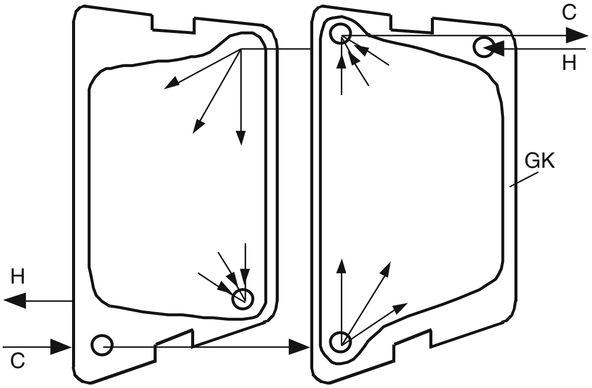


Fig. 6.5 Schematic diagram of a plate heat exchanger assembly

Fig. 6.6 Diagram of flow through two adjacent plates in a plate heat exchanger. *C*, *H* cold, hot streams, *GK* gasket



The plates are corrugated (patterns of ridges) for increasing the rigidity of the plate and improving heat transfer by promoting turbulence.

Gaskets used for PHE include the elastomers nitrile butadiene (NBR) and isobutylene isoprene (IIR) for temperatures up to 135 °C and fluorocarbon polymers for higher temperatures (Carlson 1992).

Dimensions and capacities of PHE (Muller-Steinhagen 1997): standard plates of effective heat transfer areas 0.02–4.75 m^2, plate thickness 0.6–0.9 mm, channel spacing (gaps) 2–6 mm, flow rates 0.5–5000 m^3/h, connection diameters 25–400 mm, up to 1000 plates per pack, operating temperature −35 to 200 °C, and pressure up to 25 bar.

The heat transfer coefficient (h) in PHE is correlated as Nusselt (Nu) number to the Reynolds (Re) and Prandtl (Pr) numbers by empirical equations of the form (Fryer 1997):

$$Nu = aRe^m Pr^{1/3} \tag{6.28}$$

Typical values for the parameters of this correlation for $Re > 5$ are $a = 0.352$ and $m = 0.539$.

The (Re) number is based on the equivalent diameter of the flow channel, which is four times the hydraulic radius, defined as the ratio of the cross-sectional area over the wetted perimeter.

In general, the heat transfer coefficients in PHE are higher than those predicted by the classical correlations (Table 6.6). This is also shown by a comparison of the correlation of the heat transfer factor (j_H) for PHE (Alfa-Laval 1971) to the generalized correlation of heat transfer in food systems (6.20), as shown in Fig. 6.7. The (j_H) for the PHE is significantly higher than the generalized correlations of the heat transfer factor (j_H) at $Re > 10$, reflecting the occurrence of turbulence at low Re numbers, due to the unique flow geometry of the plate system.

Figure 6.8 indicates a combination of PHE. The basic aim is to achieve as far as possible countercurrent heat flow and stimulation of quick heating, holding this heat for a certain short time, and successive cooling up of the liquid product.

The design (sizing) of a plate heat exchanger to handle a heat load of (q, W) is based on the calculation of the total heat transfer surface area (A_t) from the basic

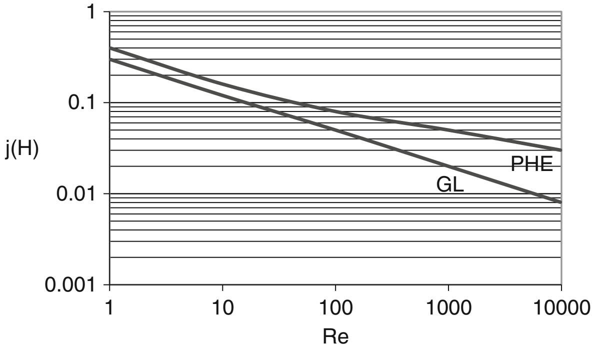


Fig. 6.7 Correlations of heat transfer factor (j_H) versus Re number. *PHE* plate heat exchangers, *GL* generalized correlation

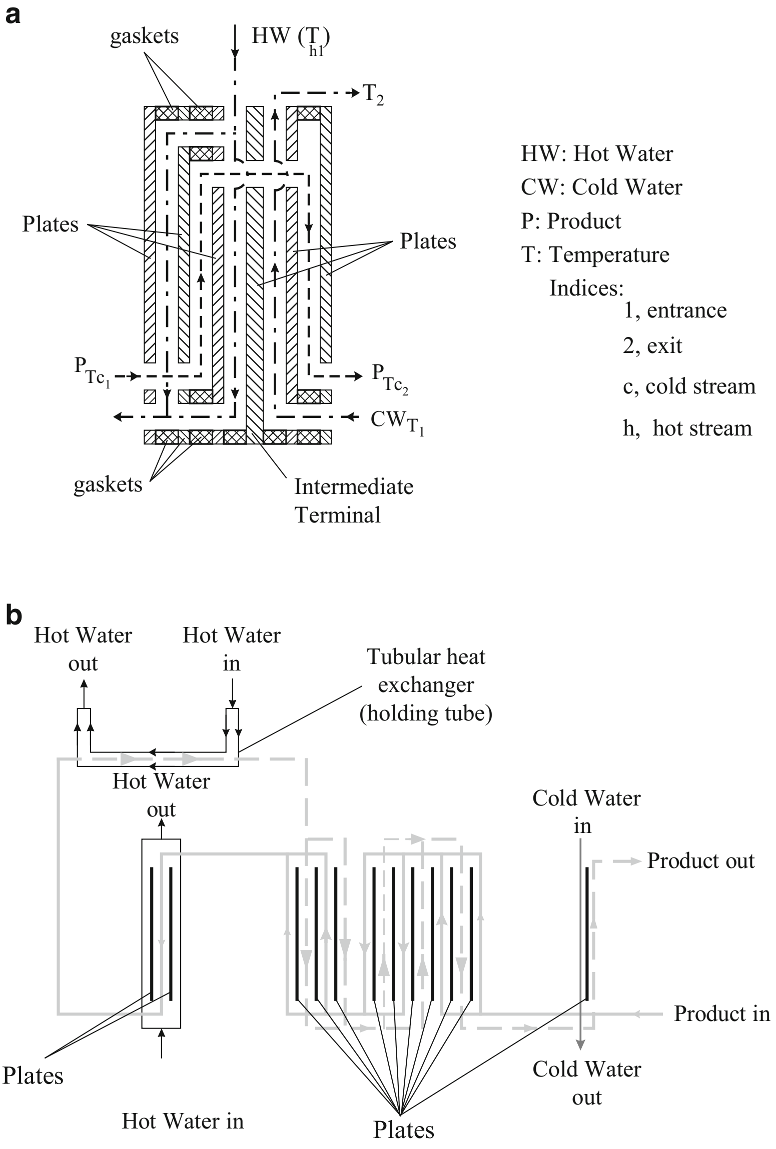


Fig. 6.8 Plate heat exchanger (PHE). (**a**) Liquid flows in a PHE; (**b**) flow diagram for typical heating/cooling application of liquid food with hot/cold water

equation (6.23). Typical overall heat transfer coefficients in PHE are 3000 W/m^2 K (aqueous liquid foods of apparent viscosities up to 1 Pa s) and 800 W/m^2 K (vegetable oils). In PHE, the log mean temperature difference is approximately equal to the arithmetic mean temperature difference between the heating medium and the product.

The pressure drop (Δp) in the PHE is given by an equation analogous to the known Fanning equation for pressure drop in straight tubes (Sinnot 1996):

$$\Delta p = 8 j_{\mathrm{p}} \left(L_{\mathrm{p}}/d_{\mathrm{e}}\right)\left(\rho u^2/2\right) \tag{6.29}$$

where (L_{p}) is the total flow path (length) of the PHE and (j_{p}) is the pressure drop factor, calculated from the empirical equation,

$$j_{\mathrm{p}} = 1.25 Re^{-0.3} \tag{6.30}$$

The total number of the required plates (N_{t}) is calculated from the equation (Alfa-Laval 1971):

$$N_{\mathrm{t}} = A_{\mathrm{t}}/A_{\mathrm{o}} = 2N_{\mathrm{p}}N_{\mathrm{s}} \tag{6.31}$$

where A_{o} is the surface area of one plate and (N_{p}, N_{s}) are the numbers of plates in parallel and in series.

The design pressure of PHE decreases from about 25 to 5 bar, as the operating temperature is increased from about 0 to 200 °C (limitation due to rubber gasket materials). A typical industrial PHE may have plates of heat transfer surface 0.4 m^2/plate and total surface up to 100 m^2 (250 plates).

6.4.6 Agitated Kettles

Agitated kettles and agitated vessels are used for heating (or cooling) viscous and highly non-Newtonian fluid foods, which cannot be handled in normal (plate) heat exchangers. Several types of agitators are used, such as propeller, paddle, blade, helical ribbon, and anchor (see Fig. 4.42). Figure 6.9 shows the diagram of two jacketed kettles with scraping anchor and propeller agitators, used for heating viscous and pulpy foods.

When propeller agitators are used, the propeller axis is inclined (about 15°) against the vertical central axis of the kettle, for reducing circulatory flow and swirling (Perry and Green 1997).

The heat transfer coefficient (h) in the product side is calculated by the empirical correlation:

$$Nu = aRe^{0.67} Pr^{1/3} \left(\eta/\eta_{\mathrm{w}}\right)^{0.14} \tag{6.32}$$

where the parameter (a) depends on the type of agitator and the nature of the fluid. In agitated vessels, the (Re) number is defined as $Re = \left(d_{\mathrm{i}}^2 N\rho\right)/\eta$, where ($d_{\mathrm{i}}$) is the

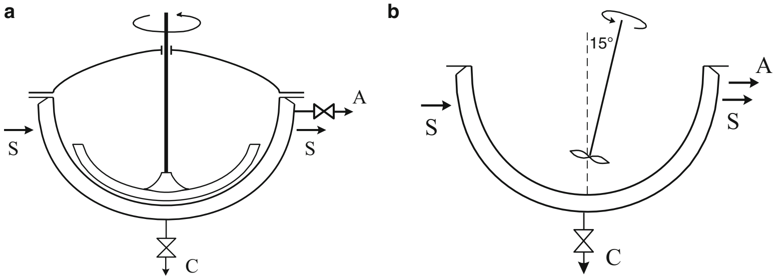


Fig. 6.9 Agitated kettles. (**a**) Anchor; (**b**) propeller agitator; *S* steam, *C* condensate, *A* air

Table 6.9 Parameters (a) of empirical equation (6.32) for agitated vessels

Agitator type/fluid product	a
Paddle, $Re < 4000$	0.415
Paddle, $Re > 4000$	0.36
Blade—flat, $Re < 400$	0.54
Blade—flat, $Re > 400$	0.74
[a]Helical ribbon, $Re > 130$	0.238
Scraping anchor/Newtonian	0.55
Scraping anchor/pseudoplastic	1.474

[a] The second part of (6.32) should be multiplied by $(i/d)^{-0.25}$ where (i) is the pitch and (d) is the diameter of the agitator

diameter of the impeller and (N) is the speed of rotation, $N = (\text{RPM})/60$, s^{-1}. The Nusselt number is defined as $Nu = (hd_T)/\lambda$, where (d_T) is the tank (vessel) diameter.

Table 6.9 shows some (a) values for various types of agitators (Saravacos and Moyer 1967; Hallstrom et al. 1988; Singh 1992). The parameter (a) depends on the type of flow (Re), the geometry and location of the agitator (e.g., flat blade, curved blade, or retreating blade), and the presence of baffles in the vessel. Baffles have no significant effect on heat transfer in viscous pseudoplastic fluids.

Experimental data, obtained in heating food liquids and fruit purees in a 10 gal (40 L) agitated kettle, showed that the estimated heat transfer coefficient (h) correlated well with the empirical (6.32). Figure 6.10 shows the increase of the overall heat transfer coefficient (U) with increasing speed of agitation (Saravacos and Moyer 1967). It is evident that the (U) values of pseudoplastic fluids (e.g., fruit purees) increase at a faster rate with agitation than the (U) of Newtonian fluids (sucrose solution), due to the lower apparent viscosity at higher shear rates (shear-thinning effect).

A water-jacketed kettle may be used, when steam is not available as a heating medium in the food plant. The water-filled jacket is heated directly by combustion of a fuel (e.g., natural gas), producing steam and preventing any burn-on problem on the inside (product) surface of the kettle (Kennedy et al. 1987).

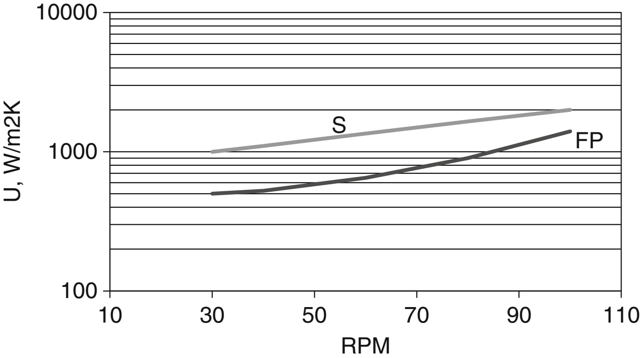


Fig. 6.10 Overall heat transfer coefficients (U) in an agitated kettle. *S* 40 % sucrose solution, *FP* fruit puree. Data from Saravacos and Moyer (1967) and Singh (1992)

The batch (unsteady-state) heating in a jacketed and agitated vessel is expressed by the following integrated form of the heat transferred and heat absorbed by the product (Singh 1992):

$$\ln[(T - T_1)/(T - T_2)] = \left[(UA)/\left(\rho V C_\mathrm{p}\right)\right]t \tag{6.33}$$

where (T) is the temperature of the heating medium (constant, e.g., steam), (T_1, T_2) are the initial and final product temperatures, (U) is the overall heat transfer coefficient, (A) is the heated surface area, (ρ) is the density, (V) is the volume of the product, and (t) is the time of heating (consistent SI units).

Agitated kettles are heated normally by saturated steam or hot water in the jacket, which give high heat transfer coefficients. In small food processing plants, where steam may not be available, heating by direct gas burning may create burn-on problems of the viscous products. In this case, the kettle can be heated with hot water or steam in the jacket, produced by gas burning (Kennedy et al. 1987). Better mixing and heating of viscous/particulate food products can be achieved with inclined (15–30°) agitators.

6.4.7 Scraped Surface Heat Exchangers

The scraped surface heat exchangers (SSHEs) are used for heating or cooling highly viscous, fouling, and crystallizing liquid foods, such as margarine and ice cream. They are double-pipe heat exchangers with agitated scrapers in the inner (product) pipe and the heating or cooling medium in the outer pipe (Fig. 6.11). The scrapers often consist of several knives, positioned in such way that, besides scraping, a screw pumping effect is achieved. The SSHEs can be operated either horizontally or in vertical position.

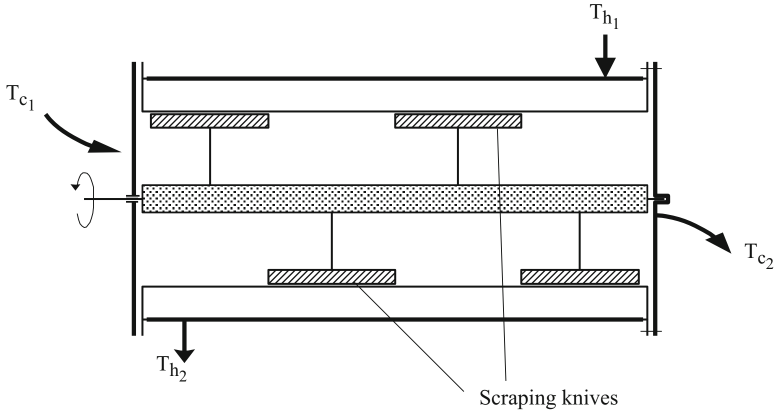


Fig. 6.11 Diagram of a scraped surface heat exchanger

The product is pumped through the SSHEs with a positive displacement pump, and high-pressure drops (Δp) up to 10 bar may develop during operation. Pipe diameters commonly used are 15 cm (inside) and 20 cm (outside). The speed of agitated scraper is in the range of 500–700 RPM. The SSHEs are often used in series in double stacks (Perry and Green 1997). Due to the mechanical construction and the moving parts, SSHEs are more expensive than any other heat exchanger used in food processing. They can be justified only for heating/cooling fluid foods that cannot be handled in normal tube or PHE.

Heat transfer in SSHEs is accomplished by heating (or cooling) the product layer adjacent to the heat transfer wall and scraping and mixing it with the bulk of the fluid food. Back-mixing of the fluid in the agitated pipe may reduce the heat transfer rate. There are no reliable empirical correlations to predict the (film) heat transfer coefficient in SSHEs. The complex hydrodynamics of fluid food flow in SSHEs is discussed by Hallstrom et al. (1988). The flow patterns, mixing effects, residence time distribution, heat transfer, and power requirements of SSHEs are discussed by Harrod (1987).

The thermal design of SSHEs is based on empirical (experimental) overall heat transfer coefficients (U), with typical values in the range of 500–1000 W/m^2 K. Due to the relatively thicker walls of the heat transfer pipe, temperature differences (ΔT) between the heating medium and the product of about 25 °C are used, which are much higher than the low (ΔT) used in PHE (about 5 °C).

SSHEs are relatively expensive equipment, and they can be justified only for very viscous fluids (viscosities higher than 10 Pa s) and particulate suspensions. Less expensive tubular heat exchangers should be used, whenever the food product can be pumped through a piping system, e.g., pseudoplastic tomato paste (Demetrakakes 1999).

6.4.8 Direct Heat Exchangers

Direct heat exchangers are based on mixing of heating steam with the liquid food product, resulting in very fast heat transfer rate. The steam used for direct heating, e.g., of milk, must be clean, tasteless, and free of any dissolved gases and toxic components, coming from the steam boiler. The feed water to the boiler should be of culinary (potable) quality and contain no chemical additives. Mixing of steam with the liquid food can be accomplished by two methods (Hallstrom et al. 1988): (a) steam injection, by injecting steam into the fluid food through small holes, and (b) steam infusion, by mixing the steam with films and droplets of the liquid food, sprayed in a special vessel (Fig. 6.12).

Figure 6.13 indicates an uperization milk equipment. The milk plus the injected water come in the expansion vacuum vessel, where the added water of the steam injection is removed through condensation in the following condenser. It is essential to control the whole flow process, so that the milk that finally leaves the expansion vacuum vessel has almost its initial water content.

Direct heating of foods minimizes fouling, by eliminating the heat transfer surface. The condensed water from the heating steam is usually removed by vacuum flashing, which reduces also the temperature and removes off-flavors and odors from the liquid product, as in UHT sterilization (Chap. 10).

Besides UHT sterilization, the direct steam infusion may be also applied in a wide range of application in the food industry. It may be used in plant sanitation (e.g., CIP), in-line cooking operations, precise pasteurization, defrosting of refrigeration heat exchangers, and confectionary and starch cooking, when in high-density products (up to 80 % solid content), and temperatures 140–170 °C are required, resulting a consistently clear final product. Some further advantages of direct steam injection equipment are their quick and accurate temperature control

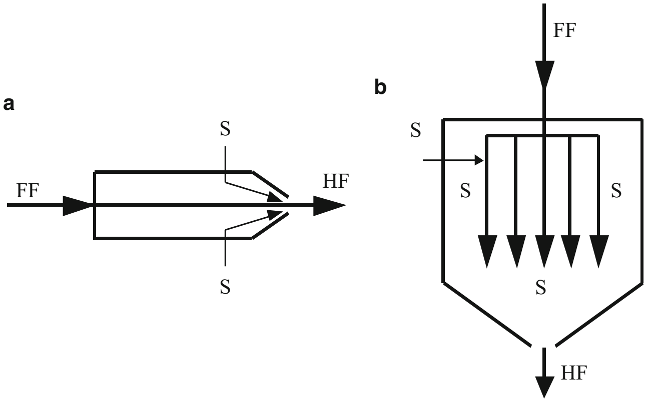


Fig. 6.12 Diagram of direct heating of liquid food: (**a**) steam injection; (**b**) steam infusion. *FF* fluid food, *HF* heated food, *S* steam

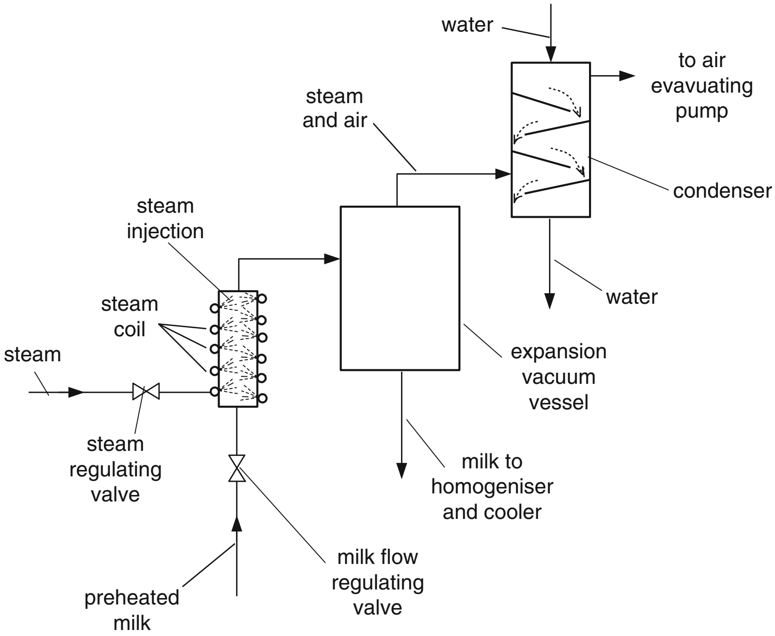


Fig. 6.13 Uperizatiton of milk equipment

(within less than 1 °C), the restricted pressure drop under normal flow rates (<0.14 bar), and their relatively low noise during operation (<80 dB).

Figure 6.14 presents the time–temperature diagram of an uperization process. As indicated also in Chap. 10, the direct contact of steam with milk heats it up to about 150 °C. However, this contact lasts only up to 3 s (Fig. 6.14).

6.4.9 Baking and Roasting Ovens

Baking is used to prepare bread, biscuits, meats, vegetables, etc., for eating, by various heating processes. The term "roasting" has the same meaning, but it usually refers to the thermal treatment of meat, nuts, or coffee. Although ovens are synonymous to baking ovens (Fellows 1990), their use in connection to coffee, cocoa, and nuts is mainly related to the development of color and aroma.

The roasting equipment normally consists of a metallic perforated drum, partially filled with the product. The drum rotates while heated by hot air or flames. A roasting oven of overall dimensions 1.5 × 1.2 m has a capacity of 3 t/h cocoa beans or hazelnuts, requiring about 30 kW electrical power. The roasting temperature can be controlled better by using pressurized hot water of 180 °C.

The ovens consist of either a compartment of several shelves or a tunnel through which the product is baked on a conveyor belt (Hallstrom et al. 1988). The heating

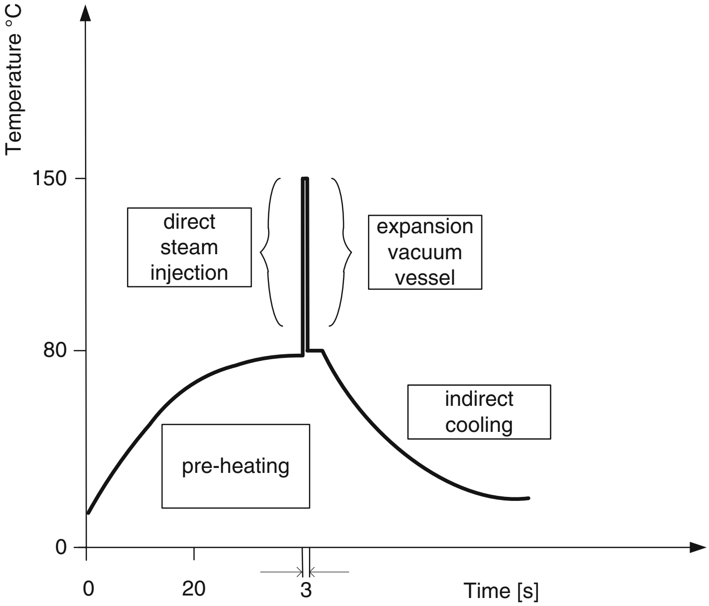


Fig. 6.14 Uperization diagram of milk

medium of the ovens is usually hot air, sometimes mixed with steam, moved by either natural convection or forced circulation. Radiation heating from the oven walls to the product may be also involved.

Ovens operate normally at atmospheric pressure, and the maximum temperature of the wet (high-moisture) product is 100 °C. The wet product is heated by the hot air or hot combustion gases until the surface layer is dried, forming a crust at 100 °C, while its interior remains at a lower temperature. Air temperatures of 150–250 °C are used in baking bread and meat. Heat and mass transfer in industrial cooking was analyzed by Hallstrom (1980).

The heat and mass transfer mechanisms of baking and roasting are similar to the mechanisms of air-drying (Chap. 8). The heat transfer coefficients of baking depend on the air velocity and temperature in the oven, varying in the range of 20–120 $\mathrm{W/m^2\,K}$ (Hallstrom et al. 1988).

The heat transfer coefficient in convection ovens can be estimated from the following empirical correlation of the heat transfer factor (Saravacos and Maroulis 2001), which is based on published experimental data:

$$j_{\mathrm{H}} = 0.801 Re^{-0.39} \tag{6.34}$$

where the (dimensionless) heat transfer factor is defined as $j_{\mathrm{H}} = h/u\rho C_{\mathrm{p}}$ and (h) is the heat transfer coefficient ($\mathrm{W/m^2\,K}$), (ρ) is the density ($\mathrm{kg/m^3}$), (u) is the velocity, and (C_{p}) is the specific heat (J/kg K) of the air (gases).

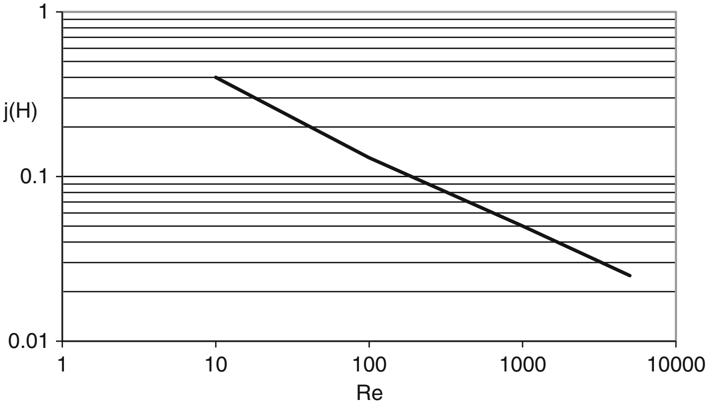


Fig. 6.15 Heat transfer factor (j_H) in a baking oven. *Re* Reynolds number

Figure 6.15 is a graphical representation of the empirical (6.34).

Heat transfer to the baking ovens can be (a) direct heating by combustion gases from a clean gas fuel, such as natural gas or LPG, or from an MW power source, or (b) indirect heating from heated oven walls, from steam tubes, or from electrical resistances (Fellows 1990). Direct heating is preferred, because it is faster and more efficient.

Two types of baking ovens are used in food processing and in catering, i.e., natural convection and forced circulation:

(a) Natural convection
Small compartment ovens are similar to home kitchens. In tunnel ovens, the product is baked while moving slowly on a conveyor belt, going through decreasing temperature regions. Steam may be injected in the first (entrance) region of the oven.

(b) Forced circulation
Small forced circulation ovens are used for baking bread and other products, placed on racks/trolleys or fixed/rotating shelves. Hot air (gas or oil burning) at 50–250 °C is circulated by a fan at velocities 1–10 m/s. In some applications electrical heating may be used, or live steam may be injected in the oven to accelerate the heat transfer process. The baking area of small ovens may be up to 12 m^2, and the baking capacity 1000–1500 small (50 g) breads in each charge.

In large installations, tunnel ovens are used, similar to the natural convection ovens, but with baking areas (higher than 120 m^2; they are equipped with fans at different locations, which recirculate the hot air, heated indirectly in heat exchangers (metallic surfaces) with hot combustion gases (Fig. 6.16). Commercial ovens are normally insulated to reduce heat losses.

Depending on the baked product, forced circulation ovens start with a preheating cycle of about 1 h, before loading with the product; baking (air) temperatures of

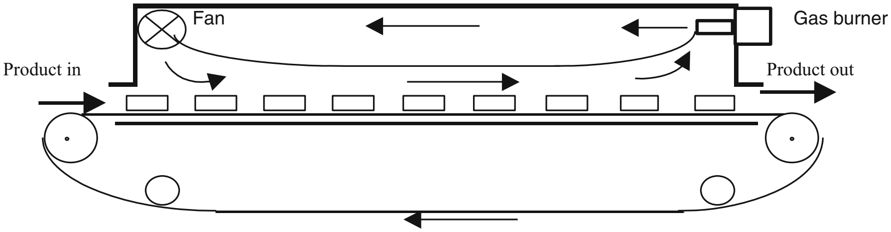


Fig. 6.16 Diagram of a continuous forced circulation baking oven

150–220 °C are used, and the motion of belts or chains is continued after heating is stopped, to avoid mechanical damage due to thermal expansion during cooling. Mechanical and hydraulic systems are used to maintain the right tension of the long belt during operation (see Sect. 3.2.3).

Computer modeling and simulations can be utilized to optimize and control the operation of commercial ovens (Zanoni et al. 1997).

Conveyorized proofing and baking systems, retailer and wholesaler ovens, and other baking equipment are described by Matz (1987).

A traveling band baking oven that utilizes various heating systems is described by Mermelstein (1999). The oven is 0.40 m wide and 15 m long, and it can be heated by gas burning (radiating), air convection, high velocity-impingement jets, and microwaves.

High heat transfer rates in ovens can be achieved by steam injection and steam condensation on the products. Combination of steam injection and air impingement improves heat transfer and increases product yield.

In the baking of biscuits, continuous ovens, with long metallic conveyor belts, are used. Such belts can be 40–45 m long, and they may be divided into three or more compartments, in which the temperature may rise progressively to 210–270 °C, with total residence time of 4–6 min. Multi-deck traveling belts are used for reducing the length of continuous ovens, used in baking products, such as French toast, Swedish rolls, and several cakes.

6.4.10 Fryers

Frying is used primarily to improve the eating quality of foods. At the same time, due to the high-temperature treatment, most of the spoilage microorganisms are inactivated, and a surface crust is formed, improving the preservation and the quality of the food product (Fellows 1990). Crust formation removes the free water from the food material, creating empty capillaries, which are filled with oil. The technology of frying of food products is described by Rossell (2001).

Frying can be accomplished either by shallow frying on a hot surface (pan) or by deep-fat frying in hot fat or vegetable oil. Film heat transfer coefficients in shallow

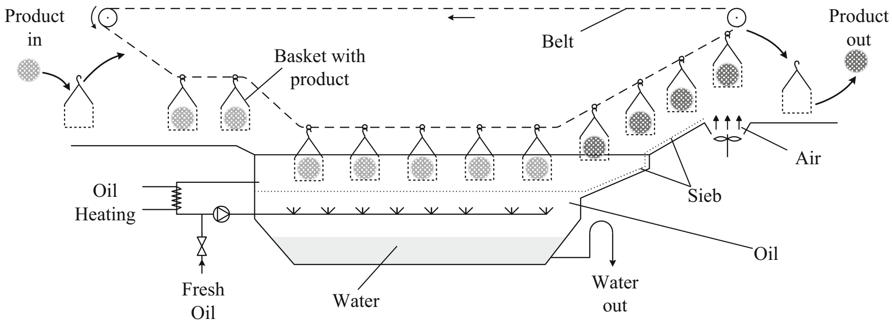


Fig. 6.17 Continuous basket fryer

(contact) frying are higher (250–400 W/m^2 K) than in deep-fat (convection) frying (200–300 W/m^2 K). Oil temperatures 160–180 °C are used in the fryers.

Batch fryers are used in small applications, while continuous units are applied in large installations. The oil is heated by electrical resistances, gas, fuel oil, or steam. Screw or belt conveyors are used to transfer the product through the hot oil in an inclined draining section. Figure 6.17 shows a continuous fryer, using baskets for transferring the food product through the heated oil. Fryers and associated baking equipment are described by Matz (1987).

Depending on the product, a 5–6-m-long continuous fryer has a capacity of 1.5–2.0 t/h. The residence time in a continuous fryer may vary in the range of 3–30 min.

6.4.11 Radiation Heaters

Most of the heat transfer operations in food processing involve heat conduction and/or heat convection, while heat radiation is involved in limited cases. Radiation heating refers mostly to heating by infrared radiation (wavelengths of 1–1000 μm).

Thermal radiation from a body is a strong function of its absolute temperature, $f(T^4)$. The emissivity (e) of a body is defined as $e = W/W_\mathrm{B}$, where W and W_B are the radiated energies of the specific body and the black body ($e = 1$), respectively. Polished metals have low emissivities (e.g., $e = 0.05$), but for normal stainless steel $e = 0.6$. Water and several food materials have emissivities higher than 0.9. Typical food emissivities are dough 0.85, lean beef 0.74, and fat beef 0.78 (Fellows 1990).

The heat exchange ($q_{1,2}$, W) between two “gray” (nonblack) surfaces (1) and (2), which are kept at temperatures (T_1, K) and (T_2, K), is given by the equation (Perry and Green 1997):

$$q_{1,2} = 5.675 A_1 \Phi_{1,2} \left[(T_1/100)^4 - (T_2/100)^4 \right] \tag{6.35}$$

where (A_1) is the radiating surface and ($\Phi_{1,2}$) is the overall radiation exchange factor, defined by the equation:

$$1/\Phi_{1,2} = 1/F_{1,2} + (1/e_1 - 1) + (A_1/A_2)(1/e_2 - 1) \tag{6.36}$$

where ($F_{1,2}$) is the "view factor," i.e., how surface (A_1) "sees" surface (A_2), and (e_1, e_1) are the emissivities of the two surfaces. The view factors are given in Tables of the heat transfer literature (McAdams 1954; Perry and Green 1997).

Equation (6.35) can be written in the classical heat transfer form as

$$q_{1,2} = h_r A_1 \Delta T \tag{6.37}$$

where (h_r) is a radiation "heat transfer coefficient" (W/m^2 K) and ($\Delta T = T_1 – T_2$) is the temperature difference between the two surfaces.

The radiation heat transfer coefficient can be calculated by combining (6.35) and (6.37).

When both heat convection and radiation are significant, the overall heat transfer rate can be calculated from the equation:

$$q_{1,2} = (h_c + h_r) A_1 \Delta T \tag{6.38}$$

where (h_c) is the convection heat transfer coefficient.

6.4.11.1 Infrared Heaters

The temperature of the infrared (IR) radiators determines the spectral distribution and the maximum emitted radiation flux The penetration of IR radiation is limited. Short wavelength radiation ($\lambda < 1.25$ μm) is preferred in food processing, because it has higher penetrating power (Hallstrom et al. 1988). Radiation heating should avoid the overheating and burning of the food surface.

The wavelengths of IR radiation used in food processing are given by Schormueller (1966). Two types of IR radiators are used in food processing: (a) gas heated, which give large wavelengths, and (b) electrically heated, which include the tubular and ceramic heaters (long λ) and the quartz and halogen heaters (short λ). Some high-intensity radiators require water or air cooling to avoid overheating.

Water vapor and carbon dioxide absorb part of the IR radiation, reducing the efficiency of IR heating of foods, e.g., during the end of the baking process, when the product may be sprayed with water.

IR radiation is used for the drying of vegetables, in baking/roasting, and in frying of foods. The main advantages are fast heating, high efficiency, and easy process control.

6.4.12 Heat Generation Processes

The need for faster heating of heat-sensitive food products has led to the development of the heat generation processes, which are based on the conversion of electrical energy into heat within the food material. Electrical heating can be achieved either as a result of friction during molecular rotation of water and other molecules (microwave or dielectric heating) or by the electrical resistance of the material to the passage of electric current (ohmic heating).

6.4.12.1 Microwave and Dielectric Heating

The most common radiations used in electrical heating of foods are the microwaves (MW) at frequencies 915 or 2450 MHz, and dielectric or radio-frequency (RF) at frequencies 3–30 MHz (1 Hz = s^{-1}). The wavelengths (λ) in the air, corresponding to these frequencies, are 915 MHz (32.8 cm), 2450 MHz (12.3 cm), and 3–30 MHz (10–1 m).

The heat generation rate (Q, W) per unit volume of a material is given by the equation:

$$Q = 0.56 \times 10^{-10} E^2 \varepsilon' \omega \tan\delta \tag{6.39}$$

where (E) is the electric field strength (V), (ω) is the frequency (s^{-1}), (ε') is the dielectric constant, and ($\tan\delta$) is the loss tangent of the material, defined as $\tan\delta = \varepsilon''/\varepsilon'$, where ($\varepsilon''$) is the dielectric loss.

The dielectric constant (ε') is a measure of the MW rate of penetration in the food. It indicates the ability of material (food) to store electrical energy. The dielectric loss (ε'') is a measure of how easily is this energy dissipated. In general, $(\varepsilon'/\varepsilon'') > 1$.

The dielectric properties are difficult to predict and they are measured experimentally. They depend strongly on the composition of the food material, and they usually decrease with increasing temperature, with liquid water having the highest values. Typical values of dielectric constants and dielectric losses (ε', ε'') are (Datta et al. 1995): water (80, 20), fruit (55, 15), meat (40, 10), vegetable oils (2.5, 1.5), and ice (3.2, 0.003).

The very low dielectric constants of ice are of importance in the microwave thawing of frozen foods, since the liquid water tends to be heated much faster than the melting of ice, damaging the food quality. For this reason, microwave thawing should be controlled carefully (tempering of frozen foods).

In addition to the dielectric constants, the penetration depth of MW and RF radiation in the materials is of fundamental importance in heating applications. The penetration depth (d), defined as the depth where the radiation intensity decays by 37 % ($1/e$) of its surface value, is given by the equation:

$$d = l_{\mathrm{o}}\left(e^{'}\right)^{0.5}/2pe^{''} = l_{\mathrm{o}}/2p\left(e^{'}\right)^{0.5}\mathrm{tand} \tag{6.40}$$

where (λ_{o}) is the wavelength of the radiation in the air (Schiffmann 1987).

According to (6.40), the penetration of radiation in a material increases with the wavelength. Thus, MW radiation at 915 MHz penetrates more than the 2450 MHz radiation, i.e., 30 versus 10 mm.

Application of MW and RF to the heating of food products requires the knowledge of dielectric properties (ε', ε'') and the thermal and transport properties (C_{p}, λ, α, D) of the food materials.

The electrical properties of foods, related to MW and RF heating, were investigated in the COST bis cooperative project of the European Union (Jowitt et al. 1987). Applications of the electrical properties to food processing are discussed by Mudgett (1990, 1995), Kent (1987), Ohlsson (1987), and Datta et al. (1995).

Equipment used in MW-RF food processing is discussed by Decareau and Peterson (1986), Schiffmann (1987), Mudgett (1990), Reuter (1993), and Datta and Anantheswaran (2001). MW-RF equipment used in food dehydration is reviewed in Chap. 8. Magnetrons, waveguides, and applicators are used in various arrangements to produce and direct the radiation to the food product, which is usually moved on a conveyor belt. For pasteurization and cooking at temperatures below 100 °C, open conveying systems are used with a hood to keep the water vapors (or steam) and prevent surface drying out of the product. For high-temperature sterilization (110–130 °C), closed systems are used. When food in plastic trays of pouches is sterilized, overpressure (air or steam) is required to prevent bursting of the packages. The presence of steam during MW-RF treatment reduces also the corner and edge effects and the cold spots of the food products.

Food applications of MW-RF, in addition to dying (Chap. 8), include tempering of frozen foods, precooking of meat and other foods, and pasteurization and sterilization of foods.

Tempering of frozen foods is one of the most important MW-RF applications (Mermelstein 1999). The frozen product is heated to a temperature just below the freezing point and then allowed to fully thaw at low temperature, reducing sharply the normal thawing time. The process is applied to the packaged frozen food, without taking the packages apart, and it reduces significantly drip losses (more economical than conventional thawing). Thawing of frozen foods is discussed in Chap. 9.

MW-tempered frozen meat and fish (in bulk) can be processed fast by slicing, grinding, and forming into patties, which can be frozen again for storage.

MW heating is used in combination with hot-air and infrared heating (surface browning) in meat cooking.

Precooking of meat, poultry, and other foods by MW-RF is faster than conventional heating. The products are packaged in plastic or tray containers (Mermelstein 1997).

MW and RF pasteurization and sterilization of foods in plastic pouches and trays are in the development stage, and some technical problems must be solved before industrial application. Sterilization of foods in plastic pouches and trays can be accomplished by MW treatment of the containers immersed in water to prevent "cold spots" and edge effects (Hallstrom 1988).

Capacity and cost data of MW food installations were presented by Edgar (1986).

6.4.12.2 Ohmic Heating

Ohmic or electric resistance heating refers to heat generation within the food material by electrical current. It is a function of the electrical resistance of the food product and the applied electrical potential. It is particularly suitable for heating pumpable particulate foods, in which the particles cannot be heated evenly with the food liquid during conventional (heat transfer) heating. The food suspension is heated by passing through special electrodes. The ohmic heating process is under development with the specific aim of UHT sterilization of particulate foods (Chap. 10).

Low-frequency alternating current (50–60 Hz) is supplied to the ohmic heating column, which consists of four or more electrodes (Fig. 6.18). The electrodes are connected using stainless steel spacer tubes, lined with insulating plastics, e.g., polyvinylidene (APV 2000).

The heating column is mounted in a vertical position with the flow of product in the upward direction. Each heating section has the same electrical impedance, and the interconnecting tubes increase in length toward the outlet, to account for the increasing ionic mobility by the increased temperature.

Commercial-scale ohmic heating systems have been designed for power outputs of 75 and 750, corresponding to product capacities of approximately 750 and 3000 kg/h, respectively, for a temperature rise of water of 75 °C.

6.4.12.3 Pulsed Electric Field

The application of pulsed electric field (PEF) in food preservation is a relatively new method. In pasteurization of liquids, these are exposed to high eclectic fields between 20 and 80 KV/cm for a few μ s. Its main advantage is the application

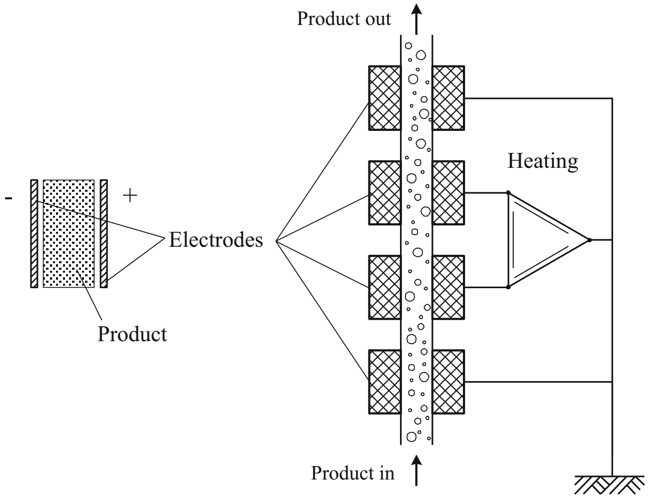


Fig. 6.18 Schematic diagram of an Ohmic hating unit

low temperature (up to 50 °C), which causes little harm to sensory characteristics of the processed food (more about this method in Chap. 12, Novel Food Processes).

6.4.13 Hygienic Considerations

Heat transfer equipment, used in food processing, should conform to the general hygienic requirements, outlined in Chap. 2.

Ovens and roasters usually operate at high temperatures and in the dry state (low moisture), which destroy vegetative cells of contaminating bacteria. However, they should be cleaned on regular basis, applying cleaning schedules, established by experience, and depending on the product, the process, etc.

Heat exchangers should be cleaned by circulating water or water containing detergents. Heat exchangers should be dismantled easily for inspection and cleaning, using, for example, clamping closures. Leaking from hot or cold water tubes should be avoided or corrected. Leaks can be detected by visual inspection or pressure drop measurements (Troller 1993).

Jacketed vessels (kettles or SSHEs) should be provided with drains for immediate and complete drainage of unwanted product residues or cleaning/rinsing fluid.

Example 6.1 Design a plate heat exchanger for the pasteurization of orange (OJ) of the orange processing plant of Example 1.1.

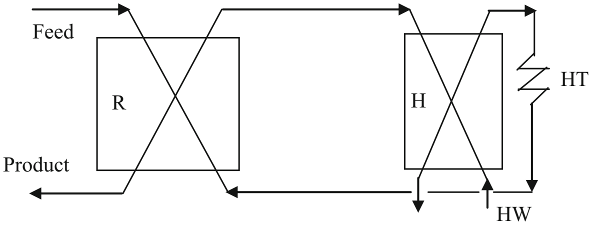


Fig. 6.19 Diagram of the HTST pasteurizer of Example 6.1. *R* regenerator, *H* heater, *HW* hot water, *HT* holding tube

Data and Assumptions

Flow rate of OJ 12 °Brix $m = 20{,}000 \times 0.45 = 9000$ kg/h $= 9000/3600 = 2.5$ kg/s.

The OJ is preheated in the regenerator from 20 to 70 °C and subsequently is heated in the heater/pasteurizer from 70 to 90 °C. The pasteurized OJ flows through the holding tube for 10 s and is then cooled from 90 to 40 °C in the regenerator by the incoming OJ. The pasteurizer is heated with hot water, which is heated by steam injection at 110 °C and exits at 90 °C.

Orange juice of 12 °Brix is considered as a Newtonian fluid with viscosity (η) about 1.5 Pa s at 20 °C, 0.6 mPa s at 55 °C, and 0.4 mPa s at 80 °C. The viscosity of water at 100 °C is taken as 0.0025 mPa s (Saravacos and Maroulis 2001).

The specific heat of water and OJ is assumed to be independent of temperature, C_p(water) = 4.18 kJ/kg and C_p (12 °C) = 3.86 kJ/kg.

Thermal conductivity of water and OJ is taken approximately as $\lambda = 0.65$ W/m K.

The density of OJ is taken as 1000 kg/m^3 and of water 958 kg/m^3 at 100 °C.

Figure 6.19 shows a simplified diagram of the proposed HTST pasteurizer (see also Fig. 6.7).

Assume a stainless steel plate heat exchanger for both regeneration and heater sections. Plate dimensions 1.6 m × 0.6 m with an effective heat transfer area 0.9 m^2/plate.

Plate thickness 0.6 mm and channel spacing 3 mm.

Flow Data

Mean velocity of OJ and water in the heat exchanger channel, $u = (2.5)/(1000)(0.6 \times 0.003) = 1.4$ m/s.

The effective flow diameter (L) of the channel is defined as $L = 4 \times$ (cross-sectional flow area)/(wetted perimeter) or $L = 4 \times (0.6 \times 0.003)/2(0.6 + 0.003) = 0.006$ m.

Temperatures: mean temperature of cold OJ in regenerator, $T = (20 + 70)/2 = 45$ °C; hot OJ in the regenerator, $T = (90 + 40)/2 = 65$ °C; hot OJ in heater, $T = (70 + 90)/2 = 80$ °C; and hot water in heater, $T = (110 + 90)/2 = 100$ °C.

Reynolds numbers: cold and hot OJ in regenerator, $Re = Lu\rho/\eta = (0.006 \times 1.4 \times 1000)/0.0006 = 14{,}000$.

Hot OJ in the heater, $Re = (0.006 \times 1.4 \times 1000)/0.0004 = 21{,}000$.

Hot water in the heater, $Re = (0.006 \times 1.4 \times 958)/0.00025 = 32189$.

Heat transfer factors: use the generalized correlation for the heat transfer factor j_H (Saravacos and Maroulis 2001), (6.20). For the OJ in the regenerator, $j_H = 0.344\ (14{,}000)^{-0.423} = 0.0061$. For the OJ in the heater, $j_H = 0.344\ (21{,}000)^{-0.423} = 0.0051$. For the hot water in the heater, $j_H = 0.344\ (32189)^{-0.423} = 0.0043$.

Prandtl numbers, $Pr = (C_p\eta/\lambda)$: for the OJ in the regenerator, $Pr = (3860 \times 0.0006)/0.65 = 3.56$, $Pr^{2/3} = 2.31$.

For the OJ in the heater, $Pr = (3680 \times 0.0004)/0.65 = 2.37$, $Pr^{2/3} = 1.77$.

For the hot water at 100 °C, $Pr = (4180 \times 0.00025)/0.65 = 1.61$, $Pr^{2/3} = 1.37$.

Heat transfer coefficients: from the heat transfer factors (j_H), using (6.21), $h = (j_H u \rho C_p) Pr^{2/3}$. For the OJ in the regenerator, $h = (0.0061 \times 1.4 \times 1000 \times 3860)/2.31 = 14{,}270\ \text{W/m}^2\ \text{K}$

For the OJ in the heater, $h = (0.0051 \times 1.4 \times 1000 \times 3860)/1.77 = 15{,}570\ \text{W/m}^2\ \text{K}$.

For the hot water in the heater, $h = (0.0043 \times 1.4 \times 958 \times 4180)/1.37 = 17{,}596\ \text{W/m}^2\ \text{K}$.

Overall heat transfer coefficients (U), (6.6): for the regenerator, assuming no fouling (clean heat transfer surfaces), $1/U = 1/14{,}270 + x/\lambda + 1/14{,}270$. The plate (wall) thickness is taken as $x = 0.6$ mm and the thermal conductivity of stainless steel, $\lambda = 16$ W/m K. Therefore, $1/U = 2/14{,}270 + (0.0006/16) = 1.78 \times 10^{-4}$, and $U = 5618\ \text{W/m}^2\ \text{K}$.

For the heater, $1/U = 1/15{,}570 + (0.0006/16) + 1/17596 = 1.6 \times 10^{-4}$ and $U = 6250\ \text{W/m}^2\ \text{K}$.

Plate Heat Exchangers

The design of heat exchangers must take into consideration the fouling of the heat transfer surfaces. In this example, the cloudy orange juice contains suspended particles and colloids and dissolved biopolymers, which may cause fouling during operation. In the absence of actual heat transfer data from similar operating systems, the design overall heat transfer coefficient are taken as equal to about 50 % of the calculated "clean" values of U, i.e., $U(\text{regenerator}) = 3000\ \text{W/m}^2\ \text{K}$ and $U(\text{heater}) = 3500\ \text{W m}^2\ \text{K}$. Lower values must be taken with more viscous and severely fouling fluids.

The heat transfer duty of the regenerator will be $q = 2.5 \times 3860 \times (70 - 20) = 482{,}500$ W and that of the heater $q = 2.5 \times 3680 \times (90 - 70) = 193{,}000$ W. The temperature difference in the regenerator will be $\Delta T = 65 - 45 = 20$ °C and in the heater $\Delta T = 100 - 80 = 20$ °C (the log mean ΔT is not needed in this example). Therefore, heat transfer area in the regenerator, $A = (482{,}500)/(3000)(20) = 8.04\ \text{m}^2$, and in the heater $A = (193{,}000)/(3500)(20) = 2.8\ \text{m}^2$.

Number of plates of (1.60 m × 0.60 m) dimensions (0.8 m^2 effective surface area) in the regenerator. $N = 8/0.8 = 10$, and in the heater, $N = 2.8/0.8 = 3.5$ or 4 plates.

Example 6.2 Design a shell and tube heat exchanger to heat 5 t/h of "hot break" tomato paste of 32 % TS from 50 to 100 °C. The heated paste is held at 100 °C for 15 s (sterilization), cooled to 30 °C and packed aseptically in plastic-lined drums.

Data and Assumptions

Rheological properties of tomato paste 32 % TS, $K = 100$ Pa s^n, $n = 0.3$ (20 °C), density $\rho = 1130$ kg/m^3, specific heat $C_p = 3500$ J/kg K, thermal conductivity $\lambda = 0.55$ W/m K.

The rheological constant (K) is related to the temperature with the Arrhenius equation and $E_a = 20$ kJ/mol. The flow behavior index (n) is independent of temperature (Saravacos and Maroulis 2001). Heating with saturated steam is at 120 °C, while the steam condensate leaves the HE as saturated liquid (120 °C).

Reynolds number Flow rate of paste $m = 5000/3600 = 1.39$ kg/s.

Assume stainless steel tubes of 10 mm internal diameter and mean velocity of the paste $u = 0.5$ m/s.

Shear rate of the paste in the tube $\gamma = 8u/d = 8 \times 0.5/0.01 = 400$ s^{-1}.

Rheological constant (K) at the mean temperature $(50 + 100)/2 = 75$ °C:

$$K(75) = K(20)\exp\{E/R(1/348 - 1/293)\}, \quad (E/R) = 20{,}000/8.31 = 2406,$$
$$\ln[K(75)/K(20)] = -1.3, \quad K(75) = 0.272, \quad K(20) = 27.2 Pa s^n.$$

Apparent viscosity of the paste at $\gamma = 400$ s^{-1} and 75 °C, $\eta = 27.2 \times (400)^{-0.7} = 0.41$ Pa s.

Reynolds number in the tubes $Re = (0.5 \times 0.01 \times 1130)/0.41$, $Re = 13.8$ (laminar flow).

Heat transfer coefficient Apply (**6.11**) for heat transfer in tubes in laminar flow: $Nu = 2.0GZ^{1/3}(K/K_w)^{0.14}$ where the Graetz number (Gz) is $Gz = mC_p/\lambda L = (1.39 \times 3500)/(0.55)(10) = 884$. The length of the pipe is assumed $L = 10$ m (for a more detailed analysis a trial-and-error calculation will be needed). $Gz^{1/3} = 884^{1/3} = 9.4$. The ratio of the rheological constants at the bulk and wall temperatures is $(K/K_w) = 100/27.2 = 3.7$, and $(K/K_w)^{0.14} = 1.2$, $Nu = 2 \times 9.4 \times 1.2 = 22.6$, $Nu = (hd)/\lambda = 0.01$ h/0.55 = 0.0182 h = 22.6, and $h = 1241$ W/m^2 K.

The heat transfer coefficient (h) can also be estimated from the generalized correlation of the heat transfer factor (j_H) for food materials, (6.20), Saravacos and Maroulis (2001): $j_H = 0.344(13.8)^{-0.423} = 0.113$. The Prandtl number will be $Pr = (C_p\eta)/\lambda = (3500 \times 0.41)/0.55 = 2609$ and $Pr^{2/3} = 179$. From (6.21) it follows that $h = (0.113 \times 0.5 \times 1130 \times 3500)/179 = 1248$ W/m^2 K.

The overall heat transfer coefficient (U) in this heat exchanger includes the resistance of the heating (steam) side and the tube wall, which are considered very small, compared to the thermal resistance of the product side. Therefore, for design purposes, an approximate value $U = 1000$ W/m^2 K can be assumed, considering some fouling effect.

Heat transfer surface:
The log mean temperature difference in the heat exchanger will be $\Delta T_M = (\Delta T_1 - \Delta T_2)/\ln(\Delta T_1/\Delta T_2)$, where $\Delta T_1 = 120 - 50 = 70$ °C and $\Delta T_2 = 120 - 100 = 20$ °C. Therefore, $\Delta T_M = (70 - 20)/\ln(70/20) = 40$ °C.

The heat flux in the HE will be $q = 1.39 \times 50 \times 3500 = 243{,}250$ W and the surface area, $A = 243{,}250/(40 \times 1000) = 6\ \text{m}^2$.

The number of tubes of the HE will be $N = 5000 \times 4/[0.5 \times 3.14 \times (0.0001) \times 3600 \times 1130] = 32$.

The length of each tube in the HE will be $L = 6/32 \times 3.14 \times 0.01 = 6$ m.

A double-pass heat exchanger with tubes 3 m long will be more practical.

Example 6.3 Show that the heat transfer area of a tubular heat exchanger is smaller when operated in countercurrent than in cocurrent flow, under the same flow and heating conditions.

The heat transferred in cocurrent (1) and countercurrent (2) flow will be

$$q_1 = U_1 A_1 (\Delta T_{L1}) \quad \text{and} \quad q_2 = U_2 A_2 (\Delta T_{L2})$$

Since the heat transferred and the flow conditions are the same, $q_1 = q_2$ and $U_1 = U_2$.

Therefore, $A_1\ (\Delta T_{L1}) = A_2\ (\Delta T_{L2})$. According to the definition of the log mean temperature difference, (6.26) and Fig. 6.2: $\Delta T_L = (\Delta T_1 - \Delta T_2)/\ln(\Delta T_1/\Delta T_2)$. For cocurrent flow $\Delta T_1 = (T_{h1} - T_{c1})$ and $\Delta T_2 = (T_{h2} - T_{c2})$, and for countercurrent flow $\Delta T_1 = (T_{h1} - T_{c2})$ and $\Delta T_2 = (T_{h2} - T_{c1})$.

Thus, the ratio of the cocurrent to countercurrent log mean temperatures will be: $\Delta T_{L1}/\Delta T_{L2} = \ln[(T_{h2} - T_{c1})/(T_{h1} - T_{c1})]_2/\ln[(T_{h1} - T_{c1})/(T_{h2} - T_{c2}]_1$. Referring to Fig. 6.2, it is shown that $[(T_{h2} - T_{c1})/(T_{h1} - T_{c2})]_2 < [(T_{h1} - T_{c2})/(T_{h2} - T_{c2})]_1$, and therefore $\Delta T_{L1}/\Delta T_{L2} < 1$.

Since it is $A_1(\Delta T_{L1}) = A_2(\Delta T_{L2})$, it is concluded that A_2 (countercurrent) $< A_1$ (cocurrent).

As an illustration, consider the heating of a food liquid from 35 to 70 °C in a tubular heat exchanger with heating medium water, which enters at 85 °C and leaves at 75 °C. Assuming similar heating conditions (q, U), the log mean temperature difference in cocurrent (1) and countercurrent (2) flow will be $\Delta T_{L1} = (50 - 5)/\ln(50/5) = 19.5$ °C and $\Delta T_{L2} = (40 - 15)/\ln(40/15) = 25.5$ °C and $\Delta T_{L1} < \Delta T_{L2}$.

Therefore, $A_2/A_1 = (\Delta T_{L1})/(\Delta T_{L2}) = (19.5)/(25.5) = 0.76$, i.e., the heat transfer area for a countercurrent operation is 24 % smaller than for the cocurrent system.

It should be noted that when (ΔT_1) approaches (ΔT_2), the calculation of the log mean temperature (ΔT_L) becomes difficult, and the arithmetic mean temperature difference should be used $[\Delta T_M = (\Delta T_1 + \Delta T_2)/2]$.

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