

CHAPTER 6

HEAT TRANSFER APPLICATIONS

The principles of heat transfer are widely used in food processing in many items of equipment. It seems appropriate to discuss these under the various applications that are commonly encountered in nearly every food factory.

HEAT EXCHANGERS

In a heat exchanger, heat energy is transferred from one body or fluid stream to another. In the design of heat exchange equipment, heat transfer equations are applied to calculate this transfer of energy so as to carry it out efficiently and under controlled conditions. The equipment goes under many names, such as boilers, pasteurizers, jacketed pans, freezers, air heaters, cookers, ovens and so on. The range is too great to list completely. Heat exchangers are found widely scattered throughout the food process industry.

Continuous-flow Heat Exchangers

It is very often convenient to use heat exchangers in which one or both of the materials that are exchanging heat are fluids, flowing continuously through the equipment and acquiring or giving up heat in passing. One of the fluids is usually passed through pipes or tubes, and the other fluid stream is passed round or across these. At any point in the equipment, the local temperature differences and the heat transfer coefficients control the rate of heat exchange.

The fluids can flow in the same direction through the equipment, this is called *parallel flow*; they can flow in opposite directions, called *counter flow*; they can flow at right angles to each other, called *cross flow*. Various combinations of these directions of flow can occur in different parts of the exchanger. Most actual heat exchangers of this type have a mixed flow pattern, but it is often possible to treat them from the point of view of the predominant flow pattern. Examples of these heat exchangers are illustrated in Figure 6.1.

In parallel flow, at the entry to the heat exchanger, there is the maximum temperature difference between the coldest and the hottest stream, but at the exit the two streams can only approach each other's temperature. In a counter flow exchanger, leaving streams can approach the temperatures of the entering stream of the other component and so counter flow exchangers are often preferred.

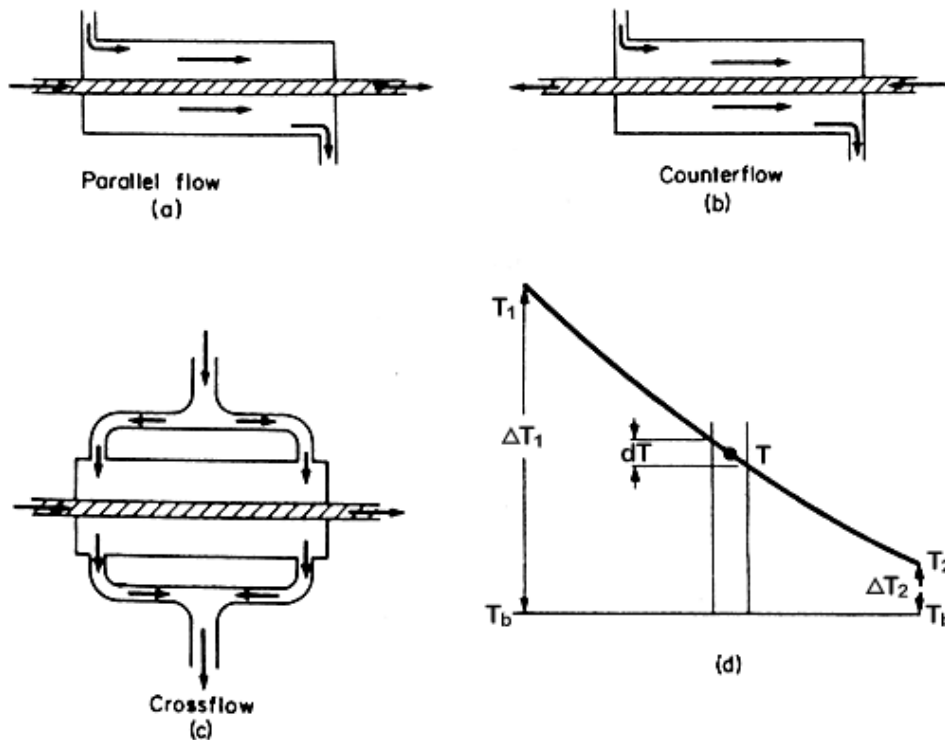


Figure 6.1 Heat exchangers

Applying the basic overall heat transfer equation for the heat transfer in the heat exchanger:

$$q = UA \Delta T$$

uncertainty at once arises as to the value to be chosen for ΔT , even knowing the temperatures in the entering and leaving streams.

Consider a heat exchanger in which one fluid is effectively at a constant temperature, T_b as illustrated in Fig. 6.1(d), where time t is on the x-axis and temperature T on the y-axis. Constant temperature in one component can result either from a very high flow rate of this component compared with the other component, or from the component being a vapour such as steam or ammonia condensing at a high rate, or from a boiling liquid. The heat transfer coefficients are assumed to be independent of temperature.

The rate of mass flow of the fluid that is changing temperature is $G \text{ kg s}^{-1}$, its specific heat is $c_p \text{ J kg}^{-1} \text{ }^\circ\text{C}^{-1}$. Over a small length of path of area dA , the mean temperature of the fluid is T and the temperature drop is dT over time dt . The constant temperature fluid has a temperature T_b . The overall heat transfer coefficient is $U \text{ J m}^{-2} \text{ s}^{-1} \text{ }^\circ\text{C}^{-1}$

Therefore the heat balance over the short length is:

$$c_p G dT = U(T - T_b) dA$$

Therefore $(U / c_p G) dA = dT / (T - T_b)$

If this is integrated over the length of the tube in which the area changes from $A = 0$ to $A = A$, and T changes from T_1 to T_2 , we have:

$$\begin{aligned} (U/c_p G) A &= \ln[(T_1 - T_b)/(T_2 - T_b)] && \text{where } \ln = \log_e \\ &= \ln(\Delta T_1 / \Delta T_2) \end{aligned}$$

in which $\Delta T_1 = (T_1 - T_b)$ and $\Delta T_2 = (T_2 - T_b)$

Therefore
$$c_p G = UA / \ln (\Delta T_1 / \Delta T_2)$$

From the overall equation, the total heat transferred per unit time is given by

$$q = UA\Delta T_m$$

where ΔT_m is the mean temperature difference.

But the total heat transferred per unit is also:

$$q = c_p G (T_1 - T_2)$$

so
$$\begin{aligned} q &= UA\Delta T_m = c_p G (T_1 - T_2) \\ &= [UA / \ln (\Delta T_1 / \Delta T_2)] \times (T_1 - T_2) \end{aligned}$$

but $(T_1 - T_2)$ can be written $(T_1 - T_b) - (T_2 - T_b)$

so
$$(T_1 - T_2) = (\Delta T_1 - \Delta T_2)$$

therefore
$$UA\Delta T_m = UA(\Delta T_1 - \Delta T_2) / \ln (\Delta T_1 / \Delta T_2) \tag{6.1}$$

so that

$$\Delta T_m = (\Delta T_1 - \Delta T_2) / \ln(\Delta T_1 / \Delta T_2) \tag{6.2}$$

where ΔT_m is called the log mean temperature difference.

In other words, the rate of heat transfer can be calculated using the heat transfer coefficient, the total area, and the log mean temperature difference. This same result can be shown to hold for parallel flow and counter flow heat exchangers in which both fluids change their temperatures.

The analysis of cross-flow heat exchangers is not so simple, but for these also the use of the log mean temperature difference gives a good approximation to the actual conditions if one stream does not change very much in temperature.

EXAMPLE 6.1. Cooling of milk in a pipe heat exchanger

Milk is flowing into a pipe cooler and passes through a tube of 2.5cm internal diameter at a rate of 0.4 kgs⁻¹. Its initial temperature is 49°C and it is wished to cool it to 18°C using a stirred bath of constant 10°C water round the pipe. What length of pipe would be required? Assume an overall coefficient of heat transfer from the bath to the milk of 900Jm⁻²s⁻¹ °C⁻¹, and that the specific heat of milk is 3890 Jkg⁻¹°C⁻¹.

Now

$$\begin{aligned}q &= c_p G (T_1 - T_2) \\ &= 3890 \times 0.4 \times (49 - 18) \\ &= 48,240 \text{Js}^{-1}\end{aligned}$$

Also

$$\begin{aligned}q &= UA \Delta T_m \\ \Delta T_m &= [(49 - 10) - (18 - 10)] / \ln[(49 - 10)/(18 - 10)] \\ &= 19.6^\circ\text{C}.\end{aligned}$$

Therefore

$$\begin{aligned}48,240 &= 900 \times A \times 19.6 \\ A &= 2.73 \text{m}^2\end{aligned}$$

but

$$A = \pi DL$$

where L is the length of pipe of diameter D

Now

$$\begin{aligned}D &= 0.025 \text{m} \\ L &= 2.73 / (\pi \times 0.025) \\ &= \underline{34.8 \text{m}}\end{aligned}$$

This can be extended to the situation where there are two fluids flowing, one the cooled fluid and the other the heated fluid. Working from the mass flow rates (kgs^{-1}) and the specific heats of the two fluids, the terminal temperatures can normally be calculated and these can then be used to determine ΔT_m and so, from the heat transfer coefficients, the necessary heat-transfer surface.

EXAMPLE 6.2. Water chilling in a counter flow heat exchanger

In a counter flow heat exchanger, water is being chilled by sodium chloride brine. If the rate of flow of the brine is 1.8 kgs^{-1} and that of the water is 1.05 kgs^{-1} , estimate the temperature to which the water is cooled if the brine enters at -8°C and leaves at 10°C , and if the water enters the exchanger at 32°C .

If the area of the heat-transfer surface of this exchanger is 55 m^2 , what is the overall heat-transfer coefficient? Take the specific heats to be 3.38 and $4.18 \text{ kJkg}^{-1}\text{C}^{-1}$ for the brine and the water respectively.

With heat exchangers a small sketch is often helpful. In this counter flow exchanger, Figure 6.2, the brine flows along the top (temperatures T_1 and T_2), and water along the bottom (temperatures T_1' and T_2').

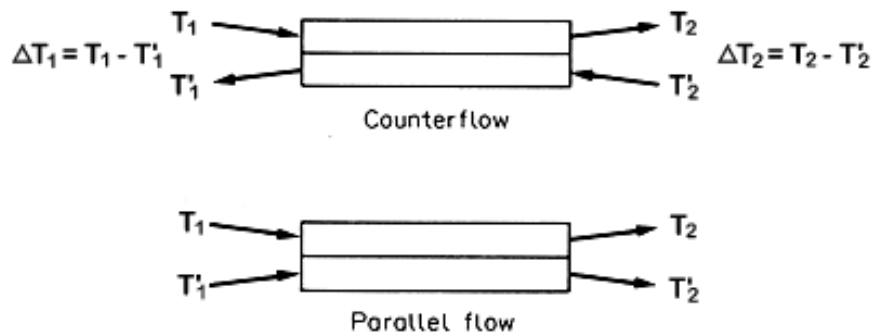


Figure 6.2. Diagrammatic heat exchanger

Three temperatures are known and the fourth T_2' can be found from the heat balance:

Brine: entering $T_1 = -8^\circ\text{C}$, leaving $T_2 = 10^\circ\text{C}$ Water: entering $T_2' = 32^\circ\text{C}$, leaving T_1'

$$\begin{aligned} \text{Heat gain in brine} &= \text{heat loss in water} \\ c_p G (T_1 - T_2) &= c_p G (T_1' - T_2') \\ 3.38 \times 1.8 [10 - (-8)] &= 4.18 \times 1.05 \times (32 - T_2') \\ (32 - T_1') &= 25 \\ \text{Therefore } T_1' &= \underline{7^\circ\text{C}}. \end{aligned}$$

And so for counter flow

$$\begin{aligned} (T_1 - T_1') &= \Delta T_1 \quad (T_2 - T_2') = \Delta T_2 \\ \Delta T_1 &= [-8 - 7] = 15^\circ\text{C} \quad \text{and} \quad \Delta T_2 = [10 - 32] = 22^\circ\text{C}. \end{aligned}$$

$$\begin{aligned} \text{Therefore } \Delta T_m &= (22 - 15) / \ln(22/15) \\ &= 7/0.382 \\ &= 18.3^\circ\text{C}. \end{aligned}$$

For the heat exchanger, from the heat balance, heat loss from brine = heat gain to water

$$\begin{aligned} q &= \text{heat passed across heat transfer surface} \\ &= UA\Delta T_m \end{aligned}$$

Therefore

$$\begin{aligned} 3.38 \times 1.8 \times 18 &= U \times 55 \times 18.3 \\ U &= 0.11 \text{kJm}^{-2}\text{s}^{-1}\text{C}^{-1} \end{aligned}$$

$$\underline{\text{Overall heat transfer coefficient} = 110 \text{Jm}^{-2}\text{s}^{-1}\text{C}^{-1}}$$

Parallel flow can be worked out similarly making appropriate adjustments.

In some cases, heat exchanger problems cannot be solved so easily; for example, if the heat transfer coefficients have to be calculated from the basic equations of heat transfer which depend on flow rates and temperatures of the fluids, and the temperatures themselves depend on the heat transfer coefficients. The easiest way to proceed then is to make sensible estimates and to go through the calculations. If the final results are coherent, then the estimates were reasonable. If not, then make better estimates, on the basis of the results, and go through a new set of calculations; and if necessary repeat again until consistent results are obtained. For those with multiple heat exchangers to design, computer programmes are available.

Jacketed Pans

In a jacketed pan, the liquid to be heated is contained in a vessel, which may also be provided with an agitator to keep the liquid on the move across the heat-transfer surface, as shown in Fig. 6.3(a).

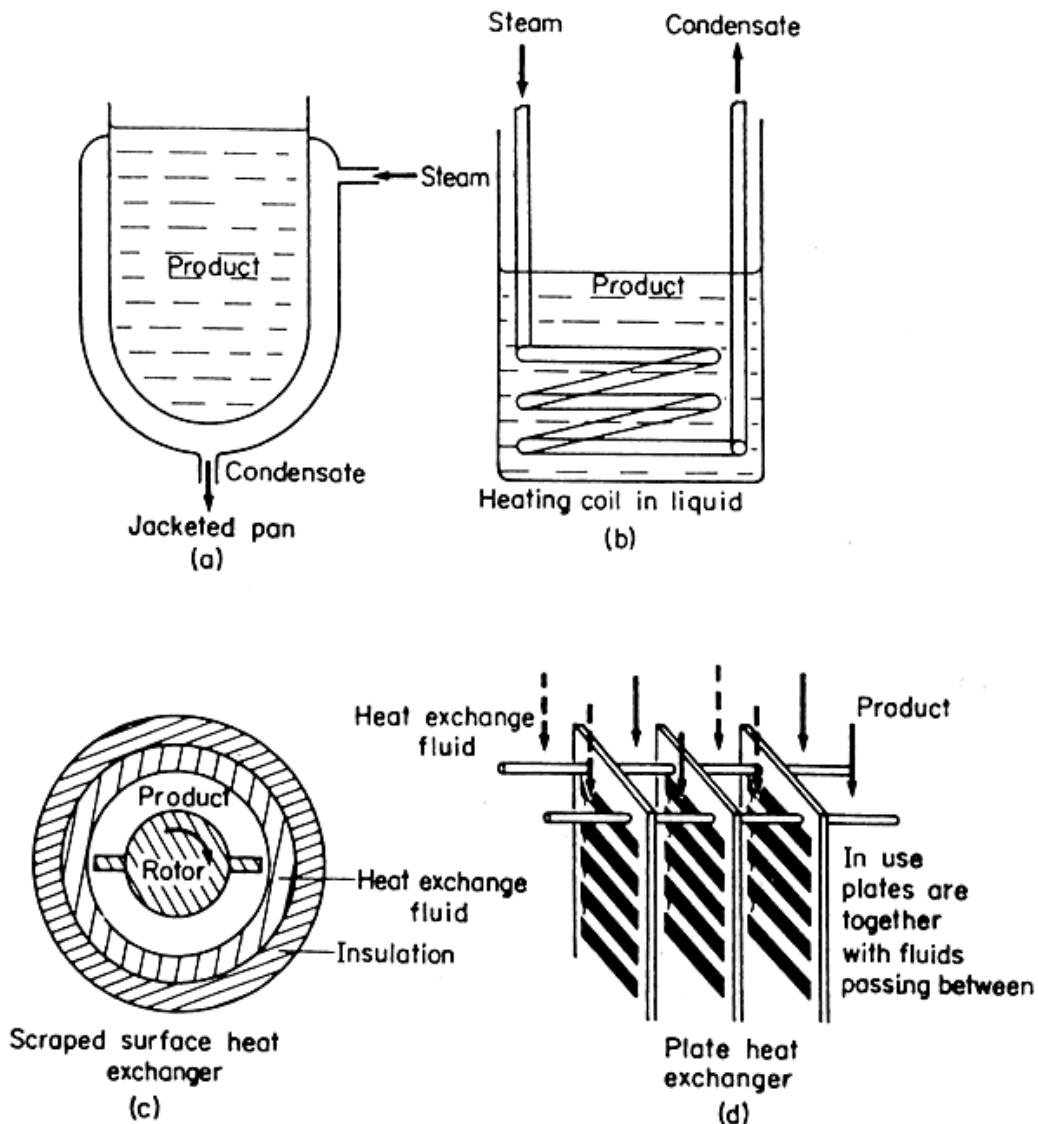


Figure 6.3 Heat exchange equipment

The source of heat is commonly steam condensing in the vessel jacket. Practical considerations are:

1. There is the minimum of air with the steam in the jacket.
2. The steam is not superheated, as part of the surface must then be used as a de-superheater over which low gas heat transfer coefficients apply rather than high condensing coefficients.
3. Steam trapping to remove condensate and air is adequate.

The action of the agitator and its ability to keep the fluid moved across the heat transfer surface are important. Some overall heat transfer coefficients are shown in Table 6.1. Save for boiling water, which agitates itself, mechanical agitation is assumed. Where there is no agitation, coefficients may be halved.

TABLE 6.1
SOME OVERALL HEAT TRANSFER COEFFICIENTS IN JACKETED PANS

Condensing fluid	Heated fluid	Pan material	Heat transfer coefficients $\text{Jm}^{-2}\text{s}^{-1}\text{C}^{-1}$
Steam	Thin liquid	Cast-iron	1800
Steam	Thick liquid	Cast-iron	900
Steam	Paste	Stainless steel	300
Steam	Water, boiling	Copper	1800

EXAMPLE 6.3. Steam required to heat pea soup in jacketed pan

Estimate the steam requirement as you start to heat 50kg of pea soup in a jacketed pan, if the initial temperature of the soup is 18°C and the steam used is at 100kPa gauge. The pan has a heating surface of 1 m² and overall heat transfer coefficient is assumed to be 300 $\text{Jm}^{-2}\text{s}^{-1}\text{C}^{-1}$.

From steam tables (Appendix 8), saturation temperature of steam at 100kPa gauge = 120°C and latent heat = $\lambda = 2202\text{kJkg}^{-1}$.

$$\begin{aligned}
 q &= UA\Delta T \\
 &= 300 \times 1 \times (120 - 18) \\
 &= 3.06 \times 10^4 \text{Js}^{-1}
 \end{aligned}$$

Therefore amount of steam

$$\begin{aligned}
 &= q/\lambda = (3.06 \times 10^4)/(2.202 \times 10^6) \\
 &= 1.4 \times 10^{-2} \text{kgs}^{-1} \\
 &= 1.4 \times 10^{-2} \times 3.6 \times 10^3 \\
 &= \underline{50\text{kg h}^{-1}}.
 \end{aligned}$$

This result applies only to the beginning of heating; as the temperature rises less steam will be consumed as ΔT decreases.

The overall heating process can be considered by using the analysis that led up to eqn. (5.6). A stirred vessel to which heat enters from a heating surface with a surface heat transfer coefficient which controls the heat flow, follows the same heating or cooling path as does a solid body of high internal heat conductivity with a defined surface heating area and surface heat transfer coefficient.

EXAMPLE 6.4. Time to heat pea soup in a jacketed pan

In the heating of the pan in Example 6.3, estimate the time needed to bring the stirred pea soup up to a temperature of 90°C, assuming the specific heat is 3.95 $\text{kJkg}^{-1}\text{C}^{-1}$.

From eqn. (5.6)

$$\begin{aligned}
 (T_2 - T_a)/(T_1 - T_a) &= \exp(-h_s A t / c_p V) \\
 T_a &= 120^\circ\text{C} \text{ (temperature of heating medium)} \\
 T_1 &= 18^\circ\text{C} \text{ (initial soup temperature)} \\
 T_2 &= 90^\circ\text{C} \text{ (soup temperature at end of time } t)
 \end{aligned}$$

$$\begin{aligned}
 h_s &= 300 \text{ Jm}^{-2}\text{s}^{-1}\text{°C}^{-1} \\
 A &= 1\text{m}^2, c = 3.95 \text{ kJkg}^{-1}\text{°C}^{-1}. \\
 \rho V &= 50\text{kg}
 \end{aligned}$$

$$\begin{aligned}
 \text{Therefore } t &= \frac{-3.95 \times 10^3 \times 50}{300 \times 1} \times \ln \frac{90}{18-120} \\
 &= (-658) \times (-1.22) \text{ s} \\
 &= 803\text{s} \\
 &= \underline{13.4\text{min.}}
 \end{aligned}$$

Heating Coils Immersed in Liquids

In some food processes, quick heating is required in the pan, for example, in the boiling of jam. In this case, a helical coil may be fitted inside the pan and steam admitted to the coil as shown in Fig. 6.3(b). This can give greater heat transfer rates than jacketed pans, because there can be a greater heat transfer surface and also the heat transfer coefficients are higher for coils than for the pan walls. Examples of the overall heat transfer coefficient U are quoted as:

- 300-1400 for sugar and molasses solutions heated with steam using a copper coil,
- 1800 for milk in a coil heated with water outside,
- 3600 for a boiling aqueous solution heated with steam in the coil.

with the units in these coefficients being $\text{Jm}^{-2}\text{s}^{-1}\text{°C}^{-1}$.

Scraped Surface Heat Exchangers

One type of heat exchanger, that finds considerable use in the food processing industry particularly for products of higher viscosity, consists of a jacketed cylinder with an internal cylinder concentric to the first and fitted with scraper blades, as illustrated in Fig. 6.3(c). The blades rotate, causing the fluid to flow through the annular space between the cylinders with the outer heat transfer surface constantly scraped. Coefficients of heat transfer vary with speeds of rotation but they are of the order of $900\text{-}4000 \text{ Jm}^{-2}\text{s}^{-1}\text{°C}^{-1}$. These machines are used in the freezing of ice cream and in the cooling of fats during margarine manufacture.

Plate Heat Exchangers

A popular heat exchanger for fluids of low viscosity, such as milk, is the plate heat exchanger, where heating and cooling fluids flow through alternate tortuous passages between vertical plates as illustrated in Fig. 6.3(d). The plates are clamped together, separated by spacing gaskets, and the heating and cooling fluids are arranged so that they flow between alternate plates. Suitable gaskets and channels control the flow and allow parallel or counter current flow in any desired number of passes. A substantial advantage of this type of heat exchanger is that it offers a large transfer surface that is readily accessible for cleaning. The banks of plates are arranged so that they may be taken apart easily. Overall heat transfer coefficients are of the order of $2400\text{-}6000 \text{ Jm}^{-2}\text{s}^{-1}\text{°C}^{-1}$.

THERMAL PROCESSING

Thermal processing implies the controlled use of heat to increase, or reduce depending on circumstances, the rates of reactions in foods.

A common example is the retorting of canned foods to effect sterilization. The object of sterilization is to destroy all microorganisms, that is, bacteria, yeasts and moulds, in the food material to prevent decomposition of the food, which makes it unattractive or inedible. Also, sterilization prevents any pathogenic (disease-producing) organisms from surviving and being eaten with the food. Pathogenic toxins may be produced during storage of the food if certain organisms are still viable. Microorganisms are destroyed by heat, but the amount of heating required for the killing of different organisms varies. Also, many bacteria can exist in two forms, the vegetative or growing form and the spore or dormant form. The spores are much harder to destroy by heat treatment than are the vegetative forms. Studies of the microorganisms that occur in foods, have led to the selection of certain types of bacteria as indicator organisms. These are the most difficult to kill, in their spore forms, of the types of bacteria which are likely to be troublesome in foods.

A frequently used indicator organism is *Clostridium botulinum*. This particular organism is a very important food-poisoning organism as it produces a deadly toxin and also its spores are amongst the most heat resistant. Processes for the heat treatment of foodstuffs are therefore examined with respect to the effect they would have on the spores of *C. botulinum*. If the heat process would not destroy this organism then it is not adequate. As *C. botulinum* is a very dangerous organism, a selected strain of a non-pathogenic organism of similar heat resistance is often used for testing purposes.

Thermal Death Time

It has been found that microorganisms, including *C. botulinum*, are destroyed by heat at rates that depend on the temperature, higher temperatures killing spores more quickly. At any given temperature, the spores are killed at different times, some spores being apparently more resistant to heat than other spores. If a graph is drawn, the number of surviving spores against time of holding at any chosen temperature, it is found experimentally that the number of surviving spores fall asymptotically to zero. Methods of handling process kinetics are well developed and if the standard methods are applied to such results, it is found that thermal death of microorganisms follows, for practical purposes, what is called a first-order process at a constant temperature (see for example Earle and Earle, 2003). This implies that the fractional destruction in any fixed time interval is constant. It is thus not possible, in theory at least, to take the time when all of the organisms are actually destroyed. Instead it is practicable, and very useful, to consider the time needed for a particular fraction of the organisms to be killed.

The rates of destruction can in this way be related to:

- (1) The numbers of viable organisms in the initial container or batch of containers.
- (2) The number of viable organisms that can safely be allowed to survive.

Of course the surviving number must be small indeed, very much less than one, to ensure adequate safety. However, this concept, which includes the admissibility of survival numbers of much less than one per container, has been found to be very useful. From such considerations, the ratio of the initial to the final number of surviving organisms becomes the criterion that determines adequate treatment. A combination of historical reasons and extensive practical experience has led to this number being set, for *C. botulinum*, at $10^{12}:1$. For other organisms, and under other circumstances, it may well be different.

The results of experiments to determine the times needed to reduce actual spore counts from 10^{12} to 1 (the lower, open, circles) or to 0 (the upper, closed, circles) are shown in Fig. 6.4.

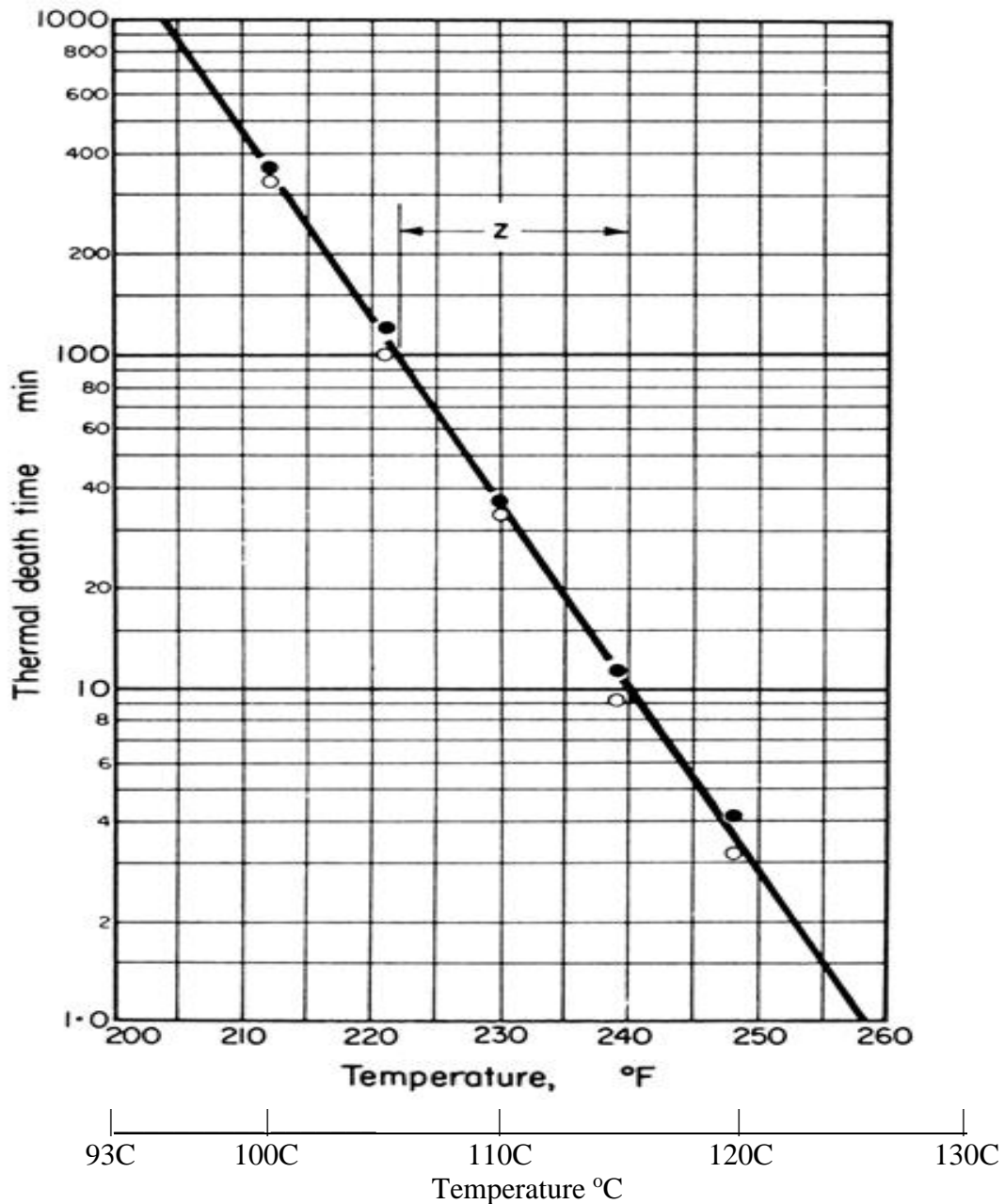


Figure 6.4. Thermal death time curve for *Clostridium botulinum*
Based on research results from the *American Can Company*

In this graph, these times are plotted against the different temperatures and it shows that when the logarithms of these times are plotted against temperatures, the resulting graph is a straight line. The mean times on this graph are called *thermal death times* for the corresponding temperatures. Note that these thermal death times do not represent complete sterilization, but a mathematical concept which can be considered as effective sterilization, which is in fact a survival ratio of $1:10^{12}$, and which has been found adequate for safety

Any canning process must be considered then from the standpoint of effective sterilization. This is done by combining the thermal death time data with the temperature/time relationships at the point in the can that heats slowest. Generally, this point is on the axis of the can and somewhere close to the geometric centre. Using either the unsteady-state heating curves or experimental measurements with a thermocouple at the slowest heating point in a can, the temperature/time graph for the can under the chosen conditions can be plotted. This curve has then to be evaluated in terms of its effectiveness in destroying *C. botulinum* or any other critical organism, such as thermophilic spore formers, which are important in industry. In this way the engineering data, which provides the temperatures within the container as the process is carried out, are combined with kinetic data to evaluate the effect of processing on the product.

Considering Fig. 6.4, the standard reference temperature is generally selected as 121.1°C (250°F), and the relative time (in minutes) required to sterilize, effectively, any selected organism at 121°C is spoken of as the *F* value of that organism. In our example, reading from Fig. 6.4, the *F* value is about 2.8 min. For any process that is different from a steady holding at 121°C , our standard process, the actual attained *F* values can be worked out by stepwise integration. If the total *F* value so found is below 2.8 min, then sterilization is not sufficient; if above 2.8 min, the heat treatment is more drastic than it needs to be.

Equivalent Killing Power at Other Temperatures

The other factor that must be determined, so that the equivalent killing powers at temperatures different from 121°C can be evaluated, is the dependence of thermal death time on temperature. Experimentally, it has been found that if the logarithm of *t*, the thermal death time, is plotted against the temperature, a straight-line relationship is obtained. This is shown in Fig. 6.4 and more explicitly in Fig. 6.5 where the x-axis is $^{\circ}\text{C}$ and y-axis is time.

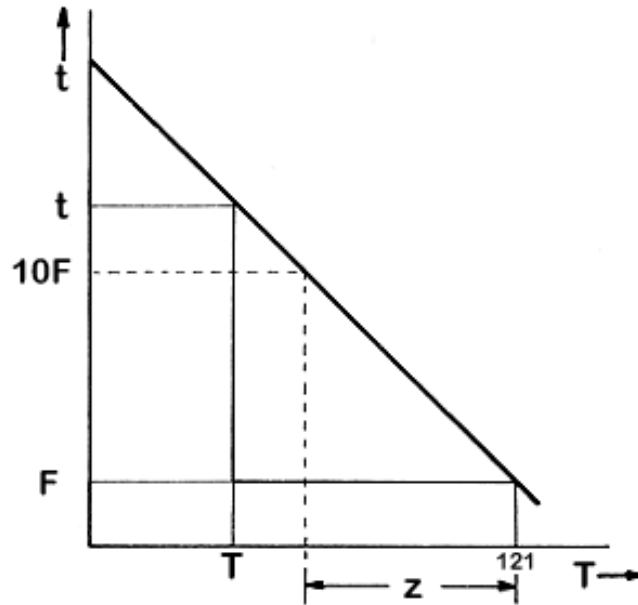


Figure 6.5 Thermal death time/temperature relationships

We can then write from the graph

$$\log t - \log F = m(121 - T) = \log t/F$$

where t is the thermal death time at temperature T , F is the thermal death time at temperature 121°C ($t_{121} = F$) and m is the slope of the graph.

Also, if we define the z value as the number of degrees below 121°C at which t increases by a factor of 10, that is by one cycle on a logarithmic graph,

$$t = 10F \text{ when } T = (121 - z)$$

so that,

$$\begin{aligned} \log 10F - \log F &= \log (10F/F) = 1 \\ 1 &= m[121 - (121 - z)] \end{aligned}$$

and so

$$z = 1/m$$

Therefore

$$\log (t/F) = (121 - T)/z$$

or

$$\begin{aligned} t &= F \times 10^{(121-T)/z} \\ &= t_{121} \times 10^{(121-T)/z} \end{aligned} \tag{6.3}$$

Now, the fraction of the process towards reaching thermal death, dS , accomplished in time dt is given by $(1/t_1)dt$, where t_1 is the thermal death time at temperature T_1 , assuming that the destruction is additive.

That is

$$dS = (1/t_1)dt$$

or
$$= (1/F)10^{-(121-T)/z} dt$$

When the thermal death time has been reached, that is when effective sterilization has been achieved,

$$\int dS = 1$$

that is
$$\int (1/F) 10^{-(121-T)/z} dt = 1$$

or
$$\int 10^{-(121-T)/z} dt = F \tag{6.4}$$

This implies that the sterilization process is complete, that the necessary fraction of the bacteria/spores have been destroyed, when the integral is equal to F . In this way, the factors F and z can be combined with the time-temperature curve and integrated to evaluate a sterilizing process.

The integral can be evaluated graphically or by stepwise numerical integration. In stepwise numerical integration, the contribution towards F of a period of t min at a temperature T is given by $t \times 10^{-(121 - t)/z}$. Breaking up the temperature-time curve into t_1 min at T_1 , t_2 min at T_2 , etc., the total F is given by

$$F(\text{i.e. } t_{121}) = t_1 \times 10^{-(121-T_1)/z} + t_2 \times 10^{-(121-T_2)/z} + \dots$$

This value of F is then compared with the standard value of F for the organism, for example 2.8 min for *C. botulinum* in our example, to decide whether the sterilizing procedure is adequate.

EXAMPLE 6.5. Time/Temperature in a can during sterilisation

In a retort, the temperatures in the slowest heating region of a can of food were measured and were found to be as shown in Fig. 6.6. Is the retorting adequate, if F for the process is 2.8 min and z is 10°C?

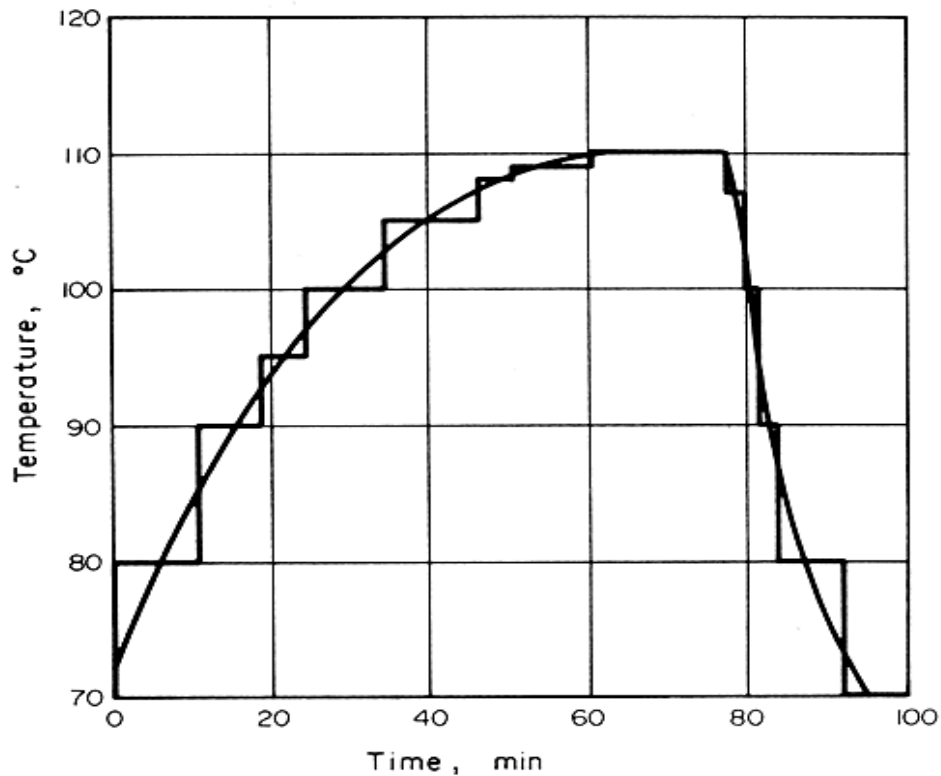


Figure 6.6 Time/Temperature curve for can processing

Approximate stepped temperature increments are drawn on the curve giving the equivalent holding times and temperatures as shown in Table 6.2. The corresponding F values are calculated for each temperature step.

TABLE 6.2
TIMES/TEMPERATURES IN CANNING STERILISATION

Temperature T (°C)	Time t (min)	$(121 - T)$	$10^{-(121-T)/10}$	$t \times 10^{-(121-T)/10}$
80	11	41	7.9×10^{-5}	0.00087
90	8	31	7.9×10^{-4}	0.0063
95	6	26	2.5×10^{-3}	0.015
100	10	21	7.9×10^{-3}	0.079
105	12	16	2.5×10^{-2}	0.30
108	6	13	5.0×10^{-2}	0.30
109	8	12	6.3×10^{-2}	0.50
110	17	11	7.9×10^{-2}	1.34
107	2	14	4.0×10^{-2}	0.08
100	2	21	7.9×10^{-3}	0.016
90	2	31	7.9×10^{-4}	0.0016
80	8	41	7.9×10^{-5}	0.0006
70	6	51	7.9×10^{-6}	0.00005
Total				2.64

The above results show that the F value for the process = 2.64 so that the retorting time is not quite adequate. This could be corrected by a further 2 min at 110°C (and proceeding as above, this would add $2 \times 10^{-(121 - 110)/10} = 0.16$, to 2.64, making 2.8).

From the example, it may be seen that the very sharp decrease of thermal death times with higher temperatures means that holding times at the lower temperatures contribute little to the sterilization. Very long times at temperatures below 90°C would be needed to make any appreciable difference to F , and in fact it can often be the holding time at the highest temperature which virtually determines the F value of the whole process. Calculations can be shortened by neglecting those temperatures that make no significant contribution, although, in each case, both the number of steps taken and also their relative contributions should be checked to ensure accuracy in the overall integration.

It is possible to choose values of F and of z to suit specific requirements and organisms that may be suspected of giving trouble. The choice and specification of these is a whole subject in itself and will not be further discussed. From an engineering viewpoint a specification is set, as indicated above, with an F value and a z value, and then the process conditions are designed to accomplish this.

The discussion on sterilization is designed to show, in an elementary way, how heat-transfer calculations can be applied and not as a detailed treatment of the topic. This can be found in appropriate books such as Stumbo (1973), Earle and Earle (2003).

Pasteurization

Pasteurization is a heat treatment applied to foods, which is less drastic than sterilization, but which is sufficient to inactivate particular disease-producing organisms of importance in a specific foodstuff. Pasteurization inactivates most viable vegetative forms of microorganisms but not heat-resistant spores.

Originally, pasteurization was evolved to inactivate bovine tuberculosis in milk. Numbers of viable organisms are reduced by ratios of the order of $10^{15}:1$. As well as the application to inactivate bacteria, pasteurization may be considered in relation to enzymes present in the food, which can be inactivated by heat. The same general relationships as were discussed under sterilization apply to pasteurization. A combination of temperature and time must be used that is sufficient to inactivate the particular species of bacteria or enzyme under consideration. Fortunately, most of the pathogenic organisms, which can be transmitted from food to the person who eats it, are not very resistant to heat.

The most common application of pasteurization is to liquid milk. In the case of milk, the pathogenic organism that is of classical importance is *Mycobacterium tuberculosis*, and the time/temperature curve for the inactivation of this bacillus is shown in Fig. 6.7.

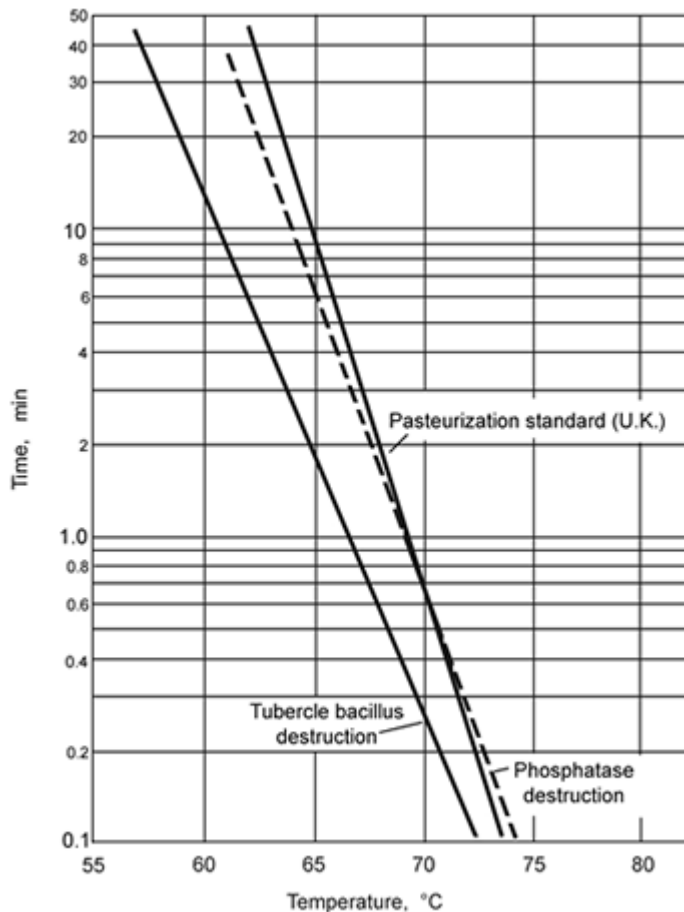


Figure 6.7 Pasteurization curves for milk

This curve can be applied to determine the necessary holding time and temperature in the same way as with the sterilization thermal death curves. However, the times involved are very much shorter, and controlled rapid heating in continuous heat exchangers simplifies the calculations so that only the holding period is really important. For example, 30min at 62.8°C in the older pasteurizing plants and 15sec at 71.7°C in the so-called high temperature/short time (H.T.S.T.) process are sufficient. An even faster process using a temperature of 126.7°C for 4 sec is claimed to be sufficient. The most generally used equipment is the plate heat exchanger and rates of heat transfer to accomplish this pasteurization can be calculated by the methods explained previously.

An enzyme present in milk, phosphatase, is destroyed under somewhat the same temperature/time conditions as the *M. tuberculosis* and, since chemical tests for the enzyme can be carried out simply, its presence is used as an indicator of inadequate heat treatment. In this case, the presence or absence of phosphatase is of no significance so far as the storage properties or suitability for human consumption are concerned.

Enzymes are of importance in deterioration processes of fruit juices, fruits and vegetables. If temperature/time relationships, such as those that are shown in Fig. 6.7 for phosphatase, can be determined for these enzymes, heat processes to destroy them can be designed. Most often this is done by steam heating, indirectly for fruit juices and directly for vegetables when the process is known as blanching.

The processes for sterilization and pasteurization illustrate very well the application of heat transfer as a unit operation in food processing. The temperatures and times required are determined and then the heat transfer equipment is designed using the equations developed for heat-transfer operations.

EXAMPLE 6.6. Pasteurisation of milk

A pasteurization heating process for milk was found, taking measurements and times, to consist essentially of three heating stages being 2min at 64°C, 3min at 65°C and 2min at 66°C. Does this process meet the standard pasteurization requirements for the milk, as indicated in Fig. 6.7, and if not what adjustment needs to be made to the period of holding at 66°C?

From Fig. 6.7, pasteurization times t_T can be read off the UK pasteurisation standard, and from these and the given times, rates and fractional extents of pasteurization can be calculated.

	At 64°C, t_{64}	= 15.7 min
so	2 min is $\frac{2}{15.7}$	= 0.13
	At 65°C, t_{65}	= 9.2 min
so	3 min is $\frac{3}{9.2}$	= 0.33
	At 66°C, t_{66}	= 5.4 min
so	2 min is $\frac{2}{5.4}$	= 0.37

$$\text{Total pasteurization extent} = (0.13 + 0.33 + 0.37) = 0.83.$$

$$\text{Pasteurization remaining to be accomplished} = (1 - 0.83) = 0.17.$$

$$\text{At 66°C this would be obtained from } (0.17 \times 5.4) \text{ min holding} = 0.92 \text{ min.}$$

So an additional 0.92 min (or approximately 1 min) at 66°C would be needed to meet the specification.

REFRIGERATION, CHILLING AND FREEZING

Rates of decay and of deterioration in foodstuffs depend on temperature. At suitable low temperatures, changes in the food can be reduced to economically acceptable levels. The growth and metabolism of microorganisms is slowed down and if the temperature is low enough, growth of all microorganisms virtually ceases. Enzyme activity and chemical reaction rates (of fat oxidation, for example) are also very much reduced at these temperatures. To reach temperatures low enough for deterioration virtually to cease, most of the water in the food must be frozen. The effect of chilling is only to slow down deterioration changes.

In studying chilling and freezing, it is necessary to look first at the methods for obtaining low temperatures, i.e. refrigeration systems, and then at the coupling of these to the food products in chilling and freezing.

Refrigeration Cycle

The basis of mechanical refrigeration is the fact that at different pressures the saturation (or the condensing) temperatures of vapours are different as clearly shown on the appropriate vapour-pressure/temperature curve. As the pressure increases condensing temperatures also increase. This fact is applied in a cyclic process, which is illustrated in Fig. 6.8.

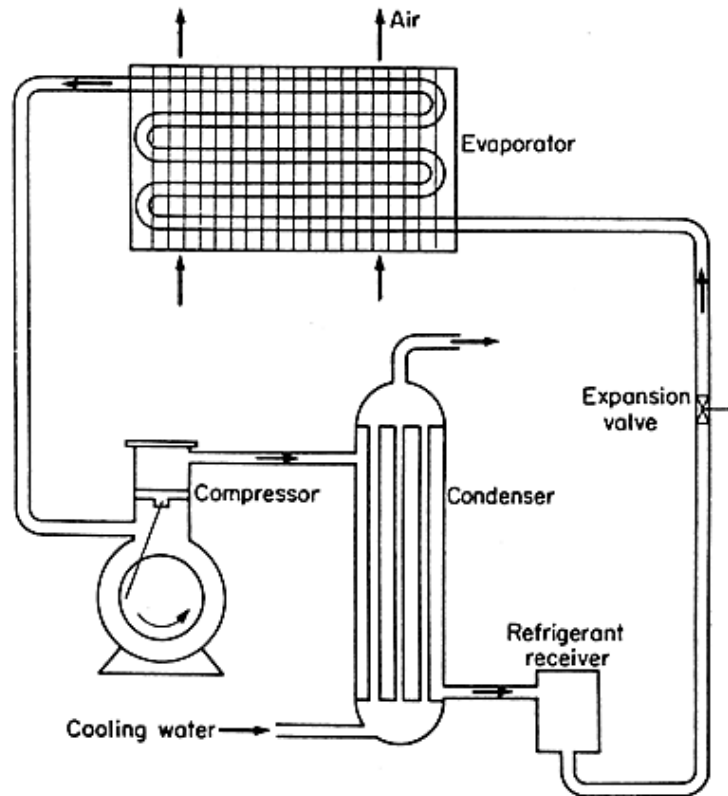


Figure 6.8 Mechanical refrigeration circuit

The process can be followed on the pressure/enthalpy (P/H) chart shown on Fig. 6.9.

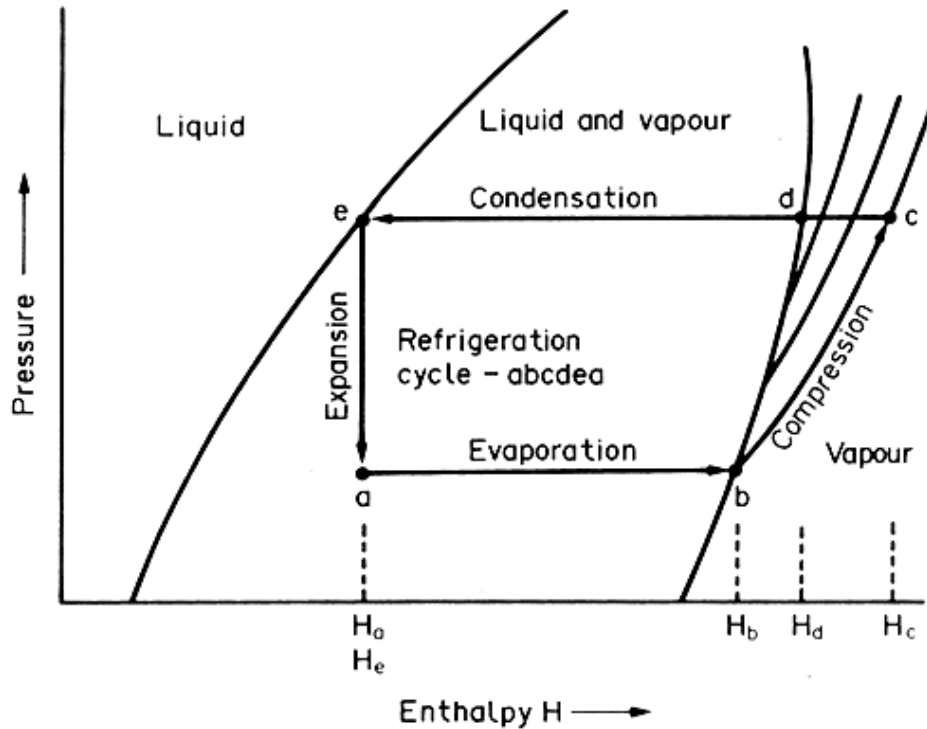


Figure 6.9 Pressure/enthalpy chart

This is a thermodynamic diagram that looks very complicated at first sight but which in fact can make calculations straightforward and simple. For the present purposes the most convenient such chart is the one shown with pressure as the vertical axis (for convenience on a logarithmic scale) and enthalpy on the horizontal axis. On such a diagram, the properties of the particular refrigerant can be plotted, including the interphase equilibrium lines such as the saturated vapour line, which are important as refrigeration depends on evaporation and condensation. Figure 6.9 is a skeleton diagram and Appendix 11 gives charts for two common refrigerants refrigerant 134A, tetrafluoroethane (in Appendix 11a) and ammonia (in 11b); others for common refrigerants can be found in the ASHRAE Guide and Data Books.

To start with the evaporator; in this the pressure above the refrigerant is low enough so that evaporation of the refrigerant liquid to a gas occurs at some suitable low temperature determined by the requirements of the product. This temperature might be, for example -18°C in which case the corresponding pressures would be for ammonia 229kPa absolute and for tetrafluoroethane (also known as refrigerant 134A) 144kPa absolute. Evaporation then occurs and this extracts the latent heat of vaporization for the refrigerant from the surroundings of the evaporator and it does this at the appropriate low temperature. This process of heat extraction at low temperature represents the useful part of the refrigerator. On the pressure/enthalpy chart this is represented by ab at constant pressure (the evaporation pressure) in which 1 kg of refrigerant takes in $(H_b - H_a)$ kJ. The low pressure necessary for the evaporation at the required temperature is maintained by the suction of the compressor.

The remainder of the process cycle is included merely so that the refrigerant may be returned to the evaporator to continue the cycle. First, the vapour is sucked into a compressor which is essentially a gas pump and which increases its pressure to exhaust it at the higher pressure to the condensers. This is represented by the line bc which follows an adiabatic compression

line, a line of constant entropy (the reasons for this must be sought in a book on refrigeration) and work equivalent to $(H_c - H_b)$ kJ kg⁻¹ has to be performed on the refrigerant to effect the compression. The higher pressure might be, for example, 1150 kPa absolute for ammonia, or 800 kPa absolute for refrigerant 134A, and it is determined by the temperature at which cooling water or air is available to cool the condensers.

To complete the cycle, the refrigerant must be condensed, giving up its latent heat of vaporization to some cooling medium. This is carried out in a condenser, which is a heat exchanger cooled generally by water or air. Condensation is shown on Fig. 6.9 along the horizontal line (at the constant condenser pressure), at first cd cooling the gas and then continuing along de until the refrigerant is completely condensed at point e. The total heat given out in this from refrigerant to condenser water is $(H_c - H_e) = (H_c - H_a)$ kJ kg⁻¹. The condensing temperature, corresponding to the above high pressures, is about 30°C and so in this example cooling water at about 20°C, could be used, leaving sufficient temperature difference to accomplish the heat exchange in equipment of economic size.

This process of evaporation at a low pressure and corresponding low temperature, followed by compression, followed by condensation at around atmospheric temperature and corresponding high pressure, is the refrigeration cycle. The high-pressure liquid then passes through a nozzle from the condenser or high-pressure receiver vessel to the evaporator at low pressure, and so the cycle continues. Expansion through the expansion valve nozzle is at constant enthalpy and so it follows the vertical line ea with no enthalpy added to or subtracted from the refrigerant. This line at constant enthalpy from point e explains why the point a is where it is on the pressure line, corresponding to the evaporation (suction) pressure.

By adjusting the high and low pressures, the condensing and evaporating temperatures can be selected as required. The high pressure is determined: by the available cooling-water temperature, by the cost of this cooling water and by the cost of condensing equipment. The evaporating pressure is determined by either the low temperature that is required for the product or by the rate of cooling/freezing that has to be provided. Low evaporating temperatures mean higher power requirements for compression and greater volumes of low-pressure vapours to be handled therefore larger compressors, so that the compression is more expensive. It must also be remembered that, in actual operation, temperature differences must be provided to operate both the evaporator and the condenser. There must be lower pressures than the pressure that corresponds to the evaporating coil temperature in the compressor suction line, and higher pressures in the compressor discharge than the pressures that correspond to the condenser temperature.

Overall, the energy side of the refrigeration cycle can therefore be summed up:

- heat taken in from surroundings at the (low) evaporator temperature and pressure $(H_b - H_a)$,
- heat equivalent to the work done by the compressor $(H_c - H_b)$ and
- heat rejected at the (high) compressor pressure and temperature $(H_c - H_e)$.

A useful measure is the ratio of the heat taken in at the evaporator (the useful refrigeration), $(H_b - H_a)$, to the energy put in by the compressor which must be paid for $(H_c - H_b)$. This ratio is called the coefficient of performance (COP). The unit commonly used to measure refrigerating effect is the ton of refrigeration = 3.52 kW. It arises from the quantity of energy to freeze 2000lb of water in one day (2000lb is called 1 short ton).

EXAMPLE 6.7. Freezing of fish

It is wished to freeze 15 tonnes of fish per day from an initial temperature of 10°C to a final temperature of - 8°C using a stream of cold air. Estimate the maximum capacity of the refrigeration plant required, if it is assumed that the maximum rate of heat extraction from the product is twice the average rate. If the heat transfer coefficient from the air to the evaporator coils, which form the heat exchanger between the air and the boiling refrigerant, is 22 Jm⁻²s⁻¹°C⁻¹, calculate the surface area of evaporator coil required if the logarithmic mean temperature drop across the coil is 12°C.

From the tabulated data (Appendix 7) the specific heat of fish is 3.18 kJkg⁻¹°C⁻¹ above freezing and 1.67 kJkg⁻¹°C⁻¹ below freezing, and the latent heat is 276kJkg⁻¹.

$$\begin{aligned}\text{Enthalpy change in fish} &= \text{heat loss above freezing} + \text{heat loss below freezing} + \text{latent heat} \\ &= (10 \times 3.18) + (8 \times 1.67) + 276 \\ &= 31.8 + 13.4 + 276 \\ &= 321.2\text{kJkg}^{-1}\end{aligned}$$

$$\begin{aligned}\text{Total heat removed in freezing} &= 15 \times 1000 \times 321.2 \\ &= 4.82 \times 10^6\text{kJday}^{-1}\end{aligned}$$

$$\begin{aligned}\text{Average rate of heat removal} &= (4.82 \times 10^6)/(24 \times 60 \times 60) \\ &= 55.8\text{kJ s}^{-1}\end{aligned}$$

$$\begin{aligned}\text{If maximum is twice average} & \\ \text{then maximum} &= 111.6\text{kJ s}^{-1} \\ &= q\end{aligned}$$

$$\text{Coil rate of heat transfer } q = UA\Delta T_m$$

$$\begin{aligned}\text{and so } A &= q/UA\Delta T_m \\ &= (111.6 \times 10^3)/(22 \times 12) \\ &= 0.42 \times 10^3\text{m}^2 \\ &= \underline{420\text{m}^2}.\end{aligned}$$

The great advantage of tracing the cycle on the pressure/enthalpy diagram is that from the numerical co-ordinates of the various cycle points, performance parameters can be read off or calculated readily.

EXAMPLE 6.8. Rate of boiling of refrigerant

Ammonia liquid is boiling in an evaporator under an absolute pressure of 120 kPa. Find the temperature and the volumetric rate of evolution of ammonia gas if the heat extraction rate from the surroundings is 300 watts.

From the chart in Appendix 11(b), the boiling temperature of ammonia at a pressure of 120kPa is - 30°C, so this is the evaporator temperature.

$$\begin{aligned}\text{Also from the chart, the latent heat of evaporation} & \\ &= (\text{enthalpy of saturated vapour} - \text{enthalpy of saturated liquid}) \\ &= 1.72 - 0.36 \\ &= 1.36 \text{ MJ kg}^{-1} \\ &= 1.36 \times 10^3 \text{ kJkg}^{-1}\end{aligned}$$

$$\begin{aligned}\text{For a heat-removal rate of 300 watts} &= 0.3 \text{ kJ s}^{-1} \\ \text{the ammonia evaporation rate} &= (0.3/1360) \\ &= 2.2 \times 10^{-4} \text{ kg s}^{-1}.\end{aligned}$$

The chart shows at the saturated vapour point for the cycle (b on Fig. 6.8) that the specific volume of ammonia vapour is:

$$0.98 \text{ m}^3 \text{ kg}^{-1}$$

and so the volumetric rate of ammonia evolution is:

$$2.2 \times 10^{-4} \times 0.98 = \underline{2.16 \times 10^{-4} \text{ m}^3 \text{ s}^{-1}}.$$

So far we have been talking of the theoretical cycle. Real cycles differ by, for example, pressure drops in piping, superheating of vapour to the compressor and non-adiabatic compression, but these are relatively minor. Approximations quite good enough for our purposes can be based on the theoretical cycle. If necessary, allowances can be included for particular inefficiencies.

The refrigerant vapour has to be compressed so it can continue round the cycle and be condensed. From the refrigeration demand, the weight of refrigerant required to be circulated can be calculated; each kg s^{-1} extracts so many Js^{-1} according to the value of $(H_b - H_a)$. From the volume of this refrigerant, the compressor displacement can be calculated, as the compressor has to handle this volume. Because the compressor piston cannot entirely displace all of the working volume of the cylinder, there must be a clearance volume; and because of inefficiencies in valves and ports, the actual amount of refrigerant vapour taken in is less than the theoretical. The ratio of these, actual amount taken in, to the theoretical compressor displacement, is called the *volumetric efficiency* of the compressor. Both mechanical and volumetric efficiencies can be measured, or taken from manufacturer's data, and they depend on the actual detail of the equipment used.

Consideration of the data from the thermodynamic chart and of the refrigeration cycle enables quite extensive calculations to be made about the operation.

EXAMPLE 6.9. Operation of a compressor in a refrigeration system

To meet the requirements of Example 6.7, calculate the speed at which it would be necessary to run a six-cylinder reciprocating ammonia compressor with each cylinder having a 10cm bore (diameter) and a 12cm stroke (length of piston travel), assuming a volumetric efficiency of 80%. The condensing temperature is 30°C (determined from the available cooling water temperature) and the evaporating temperature needed is -15°C . Calculate also the theoretical coefficient of performance of this refrigeration system.

From the chart Appendix 11(b), the heat extracted by ammonia at the evaporating temperature of -15°C is (1.74-0.63)

$$\begin{aligned} &= 1.11 \text{ MJ kg}^{-1} \\ &= 1.11 \times 10^3 \text{ kJ kg}^{-1} \end{aligned}$$

Maximum rate of refrigeration	= 111.6 kJ s ⁻¹	
Rate of refrigerant circulation	= 111.6/1110	= 0.100 kgs ⁻¹
Specific volume of refrigerant (from chart)		= 0.49 m ³ kg ⁻¹
Theoretical displacement volume	= 0.49 x 0.100	= 0.049 m ³ s ⁻¹
Actual displacement needed	= (0.049 x 100)/80	= 61.3 x 10 ⁻³ m ³ s ⁻¹
Volume of cylinders swept out per rev	= $(\pi/4) \times (0.10)^2 \times (0.12 \times 6) = 5.7 \times 10^{-3}$	

$$\begin{aligned}\text{Speed of compressor} &= (61.3 \times 10^{-3}) / (5.7 \times 10^{-3}) \\ &= 10.75 \text{ revs.s}^{-1} \\ &= 645 \text{ rev.min}^{-1}\end{aligned}$$

$$\begin{aligned}\text{Coefficient of performance} &= \\ (\text{heat energy extracted in evaporator}) / (\text{heat equivalent of theoretical energy input to the compressor}) &= \\ &= (H_b - H_a) / (H_c - H_b)\end{aligned}$$

$$\begin{aligned}\text{Under cycle conditions, from chart, the coefficient of performance} &= \\ &= (1.74 - 0.63) / (1.97 - 1.74) \\ &= 1.11 / 0.23 \\ &= \underline{4.8}.\end{aligned}$$

Performance Characteristics

Variations in load and in the evaporating or condensing temperatures are often encountered when considering refrigeration systems. Their effects can be predicted by relating them to the basic cycle.

If the heat load increases in the cold store, then the temperature tends to rise and this increases the amount of refrigerant boiling off. If the compressor cannot move this, then the pressure on the suction side of the compressor increases and so the evaporating temperature increases tending to reduce the evaporation rate and correct the situation. However, the effect is to lift the temperature in the cold space and if this is to be prevented additional compressor capacity is required.

As the evaporating pressure, and resultant temperature, change, so the volume of vapour per kilogram of refrigerant changes. If the pressure decreases, this volume increases, and therefore the refrigerating effect, which is substantially determined by the rate of circulation of refrigerant, must also decrease. Therefore if a compressor is required to work from a lower suction pressure its capacity is reduced, and conversely. So at high suction pressures giving high circulation rates, the driving motors may become overloaded because of the substantial increase in quantity of refrigerant circulated in unit time.

Changes in the condenser pressure have relatively little effect on the quantity of refrigerant circulated. However, changes in the condenser pressure and also decreases in suction pressure, have quite a substantial effect on the power consumed per ton of refrigeration. Therefore for an economical plant, it is important to keep the suction pressure as high as possible, compatible with the product requirement for low temperature or rapid freezing, and the condenser pressure as low as possible compatible with the available cooling water or air temperature.

Refrigerants

Although in theory a considerable number of fluids might be used in mechanical refrigeration, and historically quite a number including cold air and carbon dioxide have been, those actually in use today are only a very small number. Substantially they include

only ammonia, which is used in many large industrial systems, and approved members of a family of halogenated hydrocarbons containing differing proportions of fluorine and chosen according to the particular refrigeration duty required. The reasons for this very small group of practical refrigerants are many. Important reasons are

- actual vapour pressure/temperature curve for the refrigerant which determines the pressures between which the system must operate for any particular pair of evaporator and condenser temperatures,
- refrigerating effect per cubic metre of refrigerant pumped around the system which in turn governs the size of compressors and piping, and the stability and cost of the refrigerant itself.

Ammonia is in most ways the best refrigerant from the mechanical point of view, but its great problem is its toxicity. The thermodynamic chart for ammonia, refrigerant 717 is given in Appendix 11(b). In working spaces, such as encountered in air conditioning, the halogenated hydrocarbons (often known, after the commercial name, as Freons) are very often used because of safety considerations. They are also used in domestic refrigerators. Environmental problems, in particular due to the effects of any chlorine derivatives and their long lives in the upper atmosphere, have militated heavily, in recent years, against some formerly common halogenated hydrocarbons. Only a very restricted selection has been judged safe. A thermodynamic chart for refrigerant 134A (the reasons for the numbering system are obscure, ammonia being refrigerant 717), the commonest of the safe halogenated hydrocarbon refrigerants, is given in Appendix 11(a). Other charts are available in references such those provided by the refrigerant manufacturers and in books such as the ASHRAE Guide and Data Books.

Mechanical Equipment

Compressors are just basically vapour pumps and much the same types as shown in Fig. 4.3 for liquid pumps are encountered. Their design is highly specialized, particular problems arising from the lower density and viscosity of vapours when compared with liquids. The earliest designs were reciprocating machines with pistons moving horizontally or vertically, at first in large cylinders and at modest speeds, and then increasingly at higher speeds in smaller cylinders.

An important aspect in compressor choice is the compression ratio, being the ratio of the absolute pressure of discharge from the compressor, to the inlet suction pressure. Reciprocating compressors can work effectively at quite high compression ratios (up to 6 or 7 to 1). Higher overall compression ratios are best handled by putting two or more compressors in series and so sharing the overall compression ratio between them.

For smaller compression ratios and for handling the large volumes of vapours encountered at low temperatures and pressures, rotary vane compressors are often used, and for even larger volumes, centrifugal compressors, often with many stages, can be used. A recent popular development is the screw compressor, analogous to a gear pump, which has considerable flexibility. Small systems are often "hermetic", implying that the motor and compressor are sealed into one casing with the refrigerant circulating through both. This avoids rotating seals through which refrigerant can leak. The familiar and very dependable units in household refrigerators are almost universally of this type.

Refrigeration Evaporator

The evaporator is the only part of the refrigeration equipment that enters directly into food processing operations. Heat passes from the food to the heat transfer medium, which may be air or liquid, thence to the evaporator and so to the refrigerant. Thermal coupling is in some cases direct, as in a plate freezer. In this, the food to be frozen is placed directly on or between plates, within which the refrigerant circulates. Another familiar example of direct thermal coupling is the chilled slab in a shop display.

Generally, however, the heat transfer medium is air, which moves either by forced or natural circulation between the heat source, the food and the walls warmed by outside air, and the heat sink which is the evaporator. Sometimes the medium is liquid such as in the case of immersion freezing in propylene glycol or in alcohol/water mixtures. Then there are some cases in which the refrigerant is, in effect, the medium such as immersion or spray with liquid nitrogen. Sometimes there is also a further intermediate heat transfer medium, so as to provide better control, or convenience, or safety. An example is in some milk chillers, where the basic refrigerant is ammonia; this cools glycol, which is pumped through a heat exchanger where it cools the milk. Some types of evaporator systems are in Fig.6.10.

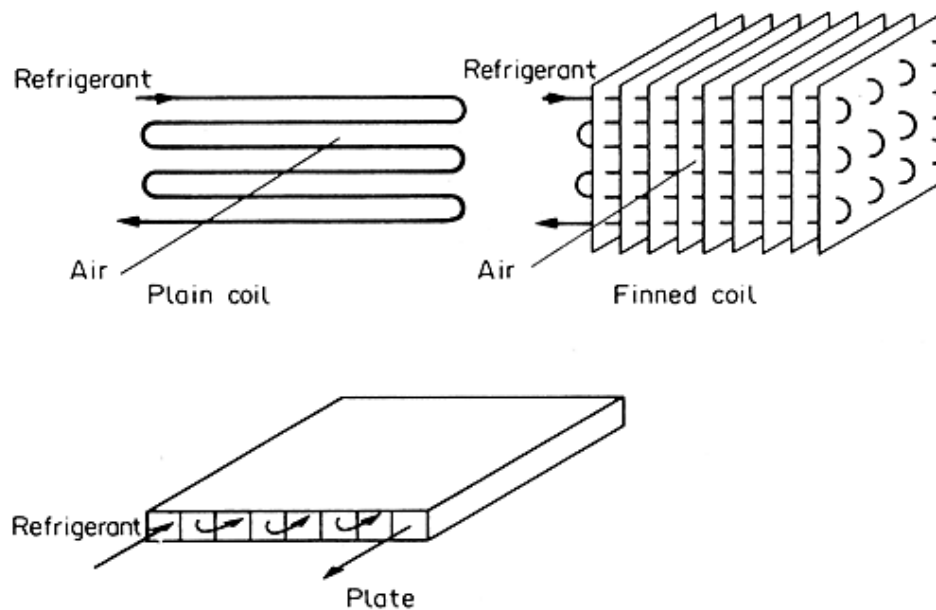


Figure 6.10 Refrigeration evaporators

In the freezer or chiller, the heat transfer rates both from the food and to the evaporator, depend upon fluid or gas velocities and upon temperature differences. The values for the respective heat transfer coefficients can be estimated by use of the standard heat transfer relationships. Typical values that occur in freezing equipment are given in Table 6.3.

TABLE 6.3
EXPECTED SURFACE HEAT TRANSFER COEFFICIENTS (h_s)

	$\text{J m}^{-2} \text{s}^{-1} \text{ } ^\circ\text{C}^{-1}$
Still-air freezing (including radiation to coils)	9
Still-air freezing (no radiation)	6
Air-blast freezing 3 ms^{-1}	18
Air-blast freezing 5 ms^{-1}	30
Liquid-immersion freezing	600
Plate freezing	120

Temperature differences across evaporators are generally of the order $3 - 10^\circ\text{C}$. The calculations for heat transfer can be carried out using the methods discussed in earlier sections, including their relationship to freezing and freezing times, as the evaporators are just refrigerant-to-air heat exchangers.

The evaporator surfaces are often extended by the use of metal fins that are bonded to the evaporator pipe surface. The reason for this construction is that the relatively high metal conductance, compared with the much lower surface conductance from the metal surface to the air, maintains the fin surface substantially at the coil temperature. A slight rise in temperature along the fin can be accounted for by including in calculations a *fin efficiency* factor. The effective evaporator area is then calculated by the relationship

$$A = A_p + \phi A_s \quad (6.5)$$

where A is the equivalent total evaporator surface area, A_p is the coil surface area, called the primary surface, A_s is the fin surface area, called the secondary surface and ϕ (phi) is the fin efficiency.

Values of ϕ lie between 65% and 95% in the usual designs, as shown for example in DKV Arbeitsblatt 2-02, 1950.

Chilling

Chilling of foods is a process by which their temperature is reduced to the desired holding temperature just above the freezing point of food, usually in the region of -2 to 2°C

Many commercial chillers operate at higher temperatures, up to $10-12^\circ\text{C}$. The effect of chilling is only to slow down deterioration changes and the reactions are temperature dependent. So the time and temperature of holding the chilled food determine the storage life of the food.

Rates of chilling are governed by the laws of heat transfer, which have been described in previous sections. It is an example of unsteady-state heat transfer by convection to the surface of the food and by conduction within the food itself. The medium of heat exchange is generally air, which extracts heat from the food and then gives it up to refrigerant in the evaporator. As explained in the heat transfer section, rates of convection heat transfer from

the surface of food and to the evaporator are much greater if the air is in movement, being roughly proportional to $v^{0.8}$

To calculate chilling rates it is therefore necessary to evaluate:

- (a) surface heat transfer coefficient,
- (b) resistance offered to heat flow by any packaging material that may be placed round the food,
- (c) appropriate unsteady-state heat conduction equation.

Although the shapes of most foodstuffs are not regular, they often approximate the shapes of slabs, bricks, spheres and cylinders.

EXAMPLE 6.10. Chilling of fresh apples

Before apples are loaded into a cool store, it is wished to chill them to a central temperature of 5°C so as to avoid problems of putting warm apples with the colder ones in storage. The apples, initially at 25°C, are considered to be spheres of 7cm diameter and the chilling is to be carried out using air at -1°C and at a velocity that provides a surface heat-transfer coefficient of 30Jm⁻²s⁻¹ °C⁻¹.

The physical properties of the apples: $k = 0.5\text{J m}^{-1}\text{s}^{-1} \text{°C}^{-1}$, $\rho = 930\text{kgm}^{-3}$, $c = 3.6 \text{kJ kg}^{-1}\text{°C}^{-1}$. Calculate the time necessary to chill the apples so that their centres reach 5°C.

This is an example of unsteady-state cooling and can be solved by application of Fig. 5.3,

$$\text{Bi} = \frac{h_s r}{k} = \frac{30 \times 0.035}{0.5} = 2.1$$

$$1/\text{Bi} = 0.48$$

$$(T - T_0)/(T_1 - T_0) = [5 - (-1)]/[25 - (-1)] = 0.23$$

and so reading from Fig.5.4, the Fourier Number

$$\text{Fo} = 0.46 = kt/\rho cr^2$$

$$t = \text{Fo} \rho cr^2/k$$

so

$$t = [0.46 \times 3600 \times 930 \times (0.035)^2]/0.5$$

$$= 3773\text{s}$$

$$= \underline{1.05\text{h}}$$

A full analysis of chilling must, in addition to heat transfer, take mass transfer into account if the food surfaces are moist and the air is unsaturated. This is a common situation and complicates chilling analysis.

Freezing

Water makes up a substantial proportion of almost all foodstuffs and so freezing has a marked physical effect on the food. Because of the presence of substances dissolved in the water, food does not freeze at one temperature but rather over a range of temperatures. At temperatures just below the freezing point of water, crystals that are almost pure ice form in

the food and so the remaining solutions become more concentrated. Even at low temperatures some water remains unfrozen, in very concentrated solutions.

In the freezing process, added to chilling is the removal of the latent heat of freezing. This latent heat has to be removed from any water that is present. Since the latent heat of freezing of water is 335 kJkg^{-1} , this represents the most substantial thermal quantity entering into the process. There may be other latent heats, for example the heats of solidification of fats which may be present, and heats of solution of salts, but these are of smaller magnitude than the latent heat of freezing of water. Also the fats themselves are seldom present in foods in as great a proportion as water.

Because of the latent heat removal requirement, the normal unsteady-state equations cannot be applied to the freezing of foodstuffs. The coefficients of heat transfer can be estimated by the following equation:

$$1/h_s = 1/h_c + (x/k) + 1/h_r$$

where h_s is the total surface heat-transfer coefficient, h_c is the convection heat transfer coefficient, x is the thickness of packing material, k is the thermal conductivity of the packing material and h_r is the radiation heat-transfer coefficient.

A full analytical solution of the rate of freezing of food cannot be obtained. However, an approximate solution, due to Plank, is sufficient for many practical purposes. Plank assumed that the freezing process:

- (a) commences with all of the food unfrozen but at its freezing temperature,
- (b) occurs sufficiently slowly for heat transfer in the frozen layer to take place under steady-state conditions.

Making these assumptions, freezing rates for bodies of simple shapes can be calculated. As an example of the method, the time taken to freeze to the centre of a slab whose length and breadth are large compared with the thickness, will be calculated.

Rates of heat transfer are equal from either side of the slab. Assume that at time t a thickness x of the slab of area A has been frozen as shown in Fig. 6.11. The temperature of the freezing medium is T_a . The freezing temperature of the foodstuff is T_f , and the surface temperature of the food is T_s . The thermal conductivity of the frozen food is k , λ is the latent heat of the foodstuff and ρ is its density.

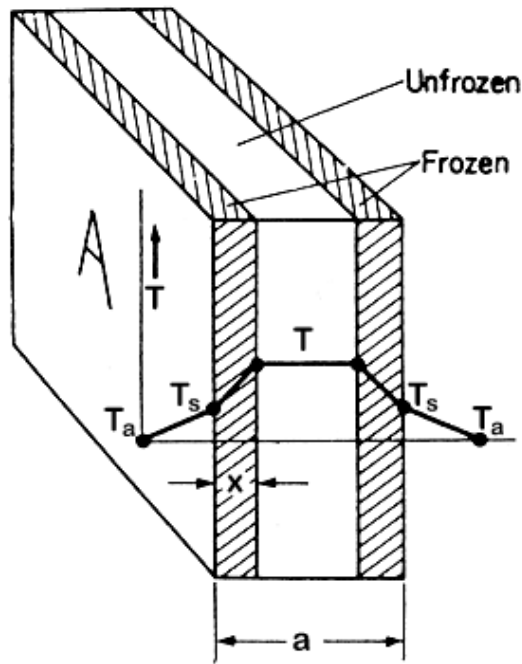


Figure 6.11 Freezing of a slab

The rate of movement of the freezing boundary multiplied by the latent heat equals the rate of heat transfer of heat to the boundary:

$$q = A \lambda \rho \, dx/dt$$

Now, all of the heat removed at the freezing boundary must be transmitted to the surface through the frozen layer; if the frozen layer is in the steady-state condition we have:

$$q = (T - T_s)k/x$$

Similarly, this quantity of heat must be transferred to the cooling medium from the food surface, so:

$$q = Ah_s(T_s - T_a)$$

where h_s is the surface heat transfer coefficient.

Eliminating T_s between the two equations gives:

$$q = (T - T_a)A \times 1/(1/h_s + x/k)$$

Since the same heat flow passing through the surface also passes through the frozen layer and is removed from the water as it freezes in the centre of the block:

Therefore

$$\begin{aligned} A(T - T_a)/(1/h_s + x/k) &= A \lambda \rho \, dx/dt \\ (T - T_a)/(1/h_s + x/k) &= \lambda \rho \, dx/dt \\ dt(T - T_a) &= \lambda \rho (1/h_s + x/k)dx \end{aligned}$$

Now, if the thickness of the slab is a , the time taken for the centre of the slab at $x = a/2$ to freeze can be obtained by integrating from $x = 0$, to $x = a/2$ during which time t goes from t to t_f .

Therefore $t_f(T - T_a) = \lambda\rho(a/2h_s + a^2/8k)$

And $t_f = \frac{\lambda\rho}{(T - T_a)}(Pa/h_s + Ra^2/k)$ (6.6)

In his papers Plank (1913, 1941) derived his equation in more general terms and found that for brick-shaped solids the change is in numerical factors only.

A general equation can thus be written

$t_f = \frac{\lambda\rho}{(T - T_a)}(Pa/h_s + Ra^2/k)$ (6.7)

where $P = 1/2$, $R = 1/8$ for a slab; $P = 1/4$, $R = 1/16$ for an infinitely long cylinder, and $P = 1/6$, $R = 1/24$ for a cube or a sphere. Brick-shaped solids have values of P and R lying between those for slabs and those for cubes. Appropriate values of P and R for a brick-shaped solid can be obtained from the graph in Fig. 6.12. In this figure, β_1 and β_2 are the ratios of the two longest sides to the shortest. It does not matter in what order they are taken.

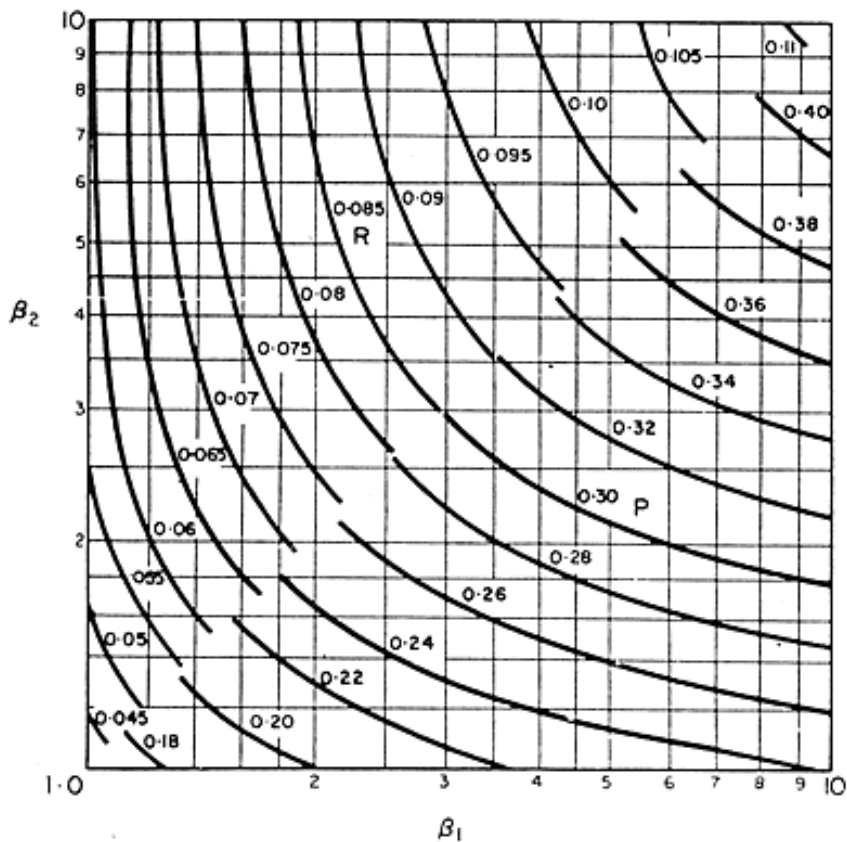


Figure 6.12 Coefficients in Plank's Equation
Adapted from Ede, *Modern Refrigeration*, 1952

Because the assumptions made in the derivation of Plank's equation lead to errors, which tend towards under-estimation of freezing times, more accurate predictions can be made if some allowances are made for this. One step is to use the total enthalpy changes from the initial to the final state of the product being frozen, that is to include the sensible heat changes both above and below the freezing temperature in addition to the latent heat. Even with this addition the prediction will still be about 20% or so lower than equivalent experimental measurements for brick shapes indicate. Adaptations of Plank's equation have been proposed which correspond better with experimental results, such as those in Cleland and Earle (1982), but they are more complicated.

EXAMPLE 6.11. Freezing of a slab of meat

If a slab of meat is to be frozen between refrigerated plates with the plate temperature at -34°C , how long will it take to freeze if the slab is 10cm thick and the meat is wrapped in cardboard 1mm thick on either side of the slab? What would be the freezing time if the cardboard were not present? Assume that for the plate freezer, the surface heat transfer coefficient is $600\text{Jm}^{-2}\text{s}^{-1}\text{C}^{-1}$, the thermal conductivity of cardboard is $0.06\text{Jm}^{-1}\text{s}^{-1}\text{C}^{-1}$ the thermal conductivity of frozen meat is $1.6\text{Jm}^{-1}\text{s}^{-1}\text{C}^{-1}$, its latent heat is $2.56 \times 10^5\text{Jkg}^{-1}$ and density 1090kgm^{-3} . Assume also that meat freezes at -2°C .

Conductance of cardboard packing $= x/k = 0.001/0.06 = 0.017$.

$$1/h_s = x/k + 1/h_c = 0.017 + 1/600 = 0.019.$$

In a plate freezer, the thickness of the slab is the only dimension that is significant. The case can be treated as equivalent to an infinite slab, and therefore the constants in Plank's equation are $1/2$ and $1/8$.

$$\begin{aligned} t_f &= \frac{\lambda\rho}{(T-T_a)} [\text{Pa}/h_s + \text{Ra}^2/k] \\ &= \frac{(2.56 \times 10^5 \times 1090)}{(-2 - \{-34\})} \times [(0.5 \times 0.1 \times 0.019) + (0.125 \times \{0.1^2/1.6\})] \\ &= 1.51 \times 10^4\text{s} \\ &= \underline{4.2\text{h}} \end{aligned}$$

And with no packing $h_s = 600$

so that $1/h_s = 1.7 \times 10^{-3}$ and

$$\begin{aligned} t_f &= (2.56 \times 10^5 \times 1090)/[-2 - \{-34\}] \times [(0.5 \times 0.1 \times 1.7 \times 10^{-3}) + (0.125 \times \{0.1\}^2/1.6)] \\ &= 7.54 \times 10^3\text{s} \\ &= \underline{2.1\text{h}} \end{aligned}$$

This estimate can be improved by adding to the latent heat of freezing, the enthalpy change above the freezing temperature and below the freezing temperature.

Above freezing, assume the meat with a specific heat of $3.22 \times 10^3\text{Jkg}^{-1}\text{C}^{-1}$ starts at $+10^{\circ}\text{C}$ and goes to -2°C , needing $3.9 \times 10^4\text{Jkg}^{-1}$.

Below the freezing temperature assume the meat goes from -2°C to the mean of -2°C and

-34°C, that is -18°C, with a specific heat of $1.67 \times 10^3 \text{ J kg}^{-1} \text{ } ^\circ\text{C}^{-1}$ needing $2.8 \times 10^4 \text{ J kg}^{-1}$. This, with the latent heat of 2.56×10^5 gives a total of $3.23 \times 10^5 \text{ J kg}^{-1}$ and amended freezing times of:

$$\text{and} \quad \begin{array}{l} 4.2 \times (3.23/2.56) = \underline{5.3 \text{ h}} \\ 2.1 \times (3.23/2.56) = \underline{2.6 \text{ h}}. \end{array}$$

If instead of a slab, cylinder, or cube, the food were closer to a brick shape then an additional 20% should be added.

Thus, with a knowledge of the thermal constants of a foodstuff, required freezing times can be estimated by the use of Plank's equation. Appendix 7 gives values for the thermal conductivities, and the latent heats and densities, of some common foods.

The analysis using Plank's equation separates the total freezing time effectively into two terms, one intrinsic to the food material to be frozen, and the other containing the surface heat transfer coefficient which can be influenced by the process equipment. Therefore for a sensitivity analysis, it can be helpful to write Plank's equation in dimensionless form, substituting ΔH for $\lambda\rho$:

$$\frac{t_f h \Delta T}{a \Delta H P} = (1 + R \text{ Bi} / P) \quad \text{where Bi} = \frac{h a}{k} = \frac{h_s r}{k}$$

This leads to defining an efficiency term:

$$\eta = \frac{R \text{ Bi}}{P} / (1 + \frac{R \text{ Bi}}{P}) = \frac{\text{Bi}}{P/R + \text{Bi}} = \frac{\text{Bi}}{4 + \text{Bi}} \quad (6.8)$$

(remembering that $P/R = 4$ for a slab)

in which η (eta) can be regarded as an efficiency of coupling of the freezing medium to the food varying from 1 for $\text{Bi} \rightarrow \infty$ to 0 for $\text{Bi} \rightarrow 0$. Taking an intrinsic freezing time t_f' for the case of unit driving force, $\Delta T = 1$; then for general ΔT :

$$t_f = t_f' / (\eta \Delta T) \quad (6.9)$$

and (6.9) can be used very readily to examine the influence of freezing medium temperature, and of surface heat transfer coefficient through the Biot number, on the actual freezing time. Thus the sensitivity of freezing time to process variations can be taken quickly into account.

EXAMPLE 6.12. Freezing time of a carton of meat: controllable factors

Determine the intrinsic freezing time for the carton of Example 6.11. By putting the equation for the freezing time in the form of eqn. (6.9) evaluate the effect of

- changes in the temperature of the plates to -20°C, -25°C, and -30°C
- effect of doubling the thickness of the cardboard and
- effect of decreased surface coefficients due to poor contact which drops the surface heat transfer coefficient to $100 \text{ J m}^{-2} \text{ s}^{-1} \text{ } ^\circ\text{C}^{-1}$

$$Bi = \frac{h_s a}{k} = \frac{52.6 \times 0.1}{1.6} = 3.3$$

$$P/R = 4, \text{ and so } \eta = 3.3/7.3 = 0.45$$

And from Example 6.11, calculated freezing time is 4.2h with driving force 32 °C

$$\text{Therefore } t_f' = 4.2 \times 0.45 \times 32 = 60.5\text{h}$$

$$\text{for (a) } t_f = \begin{aligned} &60.5 / (0.45 \times 18) = \underline{7.5 \text{ h for } -20^\circ\text{C}} \\ &60.5 / (0.45 \times 23) = \underline{5.8\text{h for } -25^\circ\text{C}} \\ &60.5 / (0.45 \times 28) = \underline{4.8\text{h for } -30^\circ\text{C}} \end{aligned}$$

for (b) x/k becomes 0.034, therefore $1/h_s = 0.034 + 0.0017$

$$h_s = 28 \text{ Jm}^{-2} \text{ s}^{-1} \text{ } ^\circ\text{C}^{-1}$$

$$Bi \text{ becomes } (28 \times 0.1) / 1.6 = 1.75$$

$$\text{Therefore } \eta \text{ becomes } 0.30$$

$$\text{so } t_f = 60.5 \times 1/0.30 + 1/32 = \underline{6.3 \text{ h}}$$

for (c) $1/h_s$ becomes $0.017 + 0.01 = 0.027$,

$$h_s = 37 \text{ Jm}^{-2} \text{ s}^{-1} \text{ } ^\circ\text{C}^{-1}$$

$$Bi \text{ becomes } (37 \times 0.1) / 1.6 = 2.3$$

$$\text{Therefore } \eta \text{ becomes } 0.37$$

$$\text{So } t_f = 60.5 \times 1/32 \times 1/0.37 = 5.1 \text{ h}$$

Cold Storage

For cold storage, the requirement for refrigeration comes from the need to remove the heat:

- coming into the store from the external surroundings through insulation
- from sources within the store such as motors, lights and people (each worker contributing something of the order of 0.5 kW)
- from the foodstuffs.

Heat penetrating the walls can be estimated, knowing the overall heat transfer coefficients including the surface terms and the conductances of the insulation, which may include several different materials. The other heat sources require to be considered and summed. Detailed calculations can be quite complicated but for many purposes simple methods give a reasonable estimate.

SUMMARY

1. For heat exchangers:

$$q = UA \Delta T_m$$

where

$$\Delta T_m = (\Delta T_1 - \Delta T_2) / \ln(\Delta T_1 / \Delta T_2)$$

2. For jacketed pans:

$$q = UA \Delta T$$

and

$$(T_2 - T_a)/(T_1 - T_a) = \exp(-h_s A t / c_p V)$$

3. For sterilization of cans:

- thermal death time is the time taken to reduce bacterial spore counts by a factor of 10^{12} .
- F value is the thermal death time at 121°C ; or *Clostridium botulinum*, it is about 2.8 min.
- z is temperature difference corresponding to a ten-fold change in the thermal death time
-

$$t_{121} = t_T \times 10^{-(121-T)/z} \quad \text{or}$$

$$t_T = t_{121} \times 10^{(121-T)/z}$$

4. The coefficient of performance of refrigeration plant is:

(heat energy extracted in evaporator)/(heat equivalent of energy input to compressor).

4. Freezing times can be calculated from:

$$t_f = \frac{\lambda \rho}{(T - T_a)} (Pa/h_s + Ra^2/k)$$

where for a slab $P = 1/2$ and $R = 1/8$ and for a sphere $P = 1/6$ and $R = 1/24$. An improved approximation is to substitute ΔH over the whole range, for λ . In addition for brick shapes a multiplier of around 1.2 is needed. a is thickness of slab.

PROBLEMS

1. A stream of milk is being cooled by water in a counter flow heat exchanger. If the milk flowing at a rate of 2 kgs^{-1} , is to be cooled from 50°C to 10°C , estimate the rate of flow of the water if it is found to rise 22°C in temperature. Calculate the log mean temperature difference across the heat exchanger, if the water enters the exchanger at 5°C .
(11.8°C)
2. A flow of 9.2 kgs^{-1} of milk is to be heated from 15°C to 65°C in a heat exchanger, using 16.7 kgs^{-1} of water at 95°C . If the overall heat transfer coefficient is $1300 \text{ Jm}^{-2} \text{ s}^{-1}\text{C}^{-1}$,

calculate the area of heat exchanger required if the flows are (a) parallel and (b) counter flow.

((a) 53m^2 (b) 34m^2)

3. In the heat exchanger of worked Example 6.2 it is desired to cool the water by a further 3°C . Estimate the increase in the flow rate of the brine that would be necessary to achieve this. Assume that: the surface heat transfer rate on the brine side is proportional to $v^{0.8}$, the surface coefficients under the conditions of Example 6.2 are equal on both sides of the heat transfer surface and they control the overall heat transfer coefficient.

(Increased by 25% to 2.25kgs^{-1})

4. A counter flow regenerative heat exchanger is to be incorporated into a milk pasteurization plant, with a heat-exchange area of 23 m^2 and an estimated overall heat-transfer coefficient of $950\text{Jm}^{-2}\text{s}^{-1}\text{C}^{-1}$. Regenerative flow implies that the milk passes from the heat exchanger through further heating and processing and then proceeds back through the same heat exchanger so that the outgoing hot stream transfers heat to the incoming cold stream. The flow of the milk is 2kgs^{-1} . Calculate the temperature at which the incoming colder milk leaves the exchanger if it enters at 10°C and if the hot milk enters the exchanger at 72°C .

(hot milk 26.3°C , cold milk 55.7°C)

5. Olive oil is to be heated in a hemispherical steam-jacketed pan, which is 0.85m in diameter. If the pan is filled with oil at room temperature (21°C), and steam at a pressure of 200 kPa above atmospheric is admitted to the jacket, which covers the whole of the surface of the hemisphere, estimate the time required for the oil to heat to 115°C . Assume an overall heat transfer coefficient of $550\text{ Jm}^{-2}\text{s}^{-1}\text{C}^{-1}$ and no heat losses to the surroundings.

(14minutes)

6. The milk pasteurizing plant, using the programme calculated in worked Example 6.6, was found in practice to have a 1°C error in its thermometers so that temperatures thought to be 65°C were in fact 64°C and so on. Under these circumstances what would the holding time at the highest temperature (a true 65°C) need to be?

(6mins.42sec.)

7. The contents of the can of pumpkin, whose heating curve was to be calculated in Problem 9 of Chapter 5, has to be processed to give the equivalent at the centre of the can of a 10^{12} reduction in the spore count of *C. botulinum*. Assuming a z value of 10°C and that a 10^{12} reduction is effected after 2.5min at 121°C , calculate the holding time that would be needed at 115°C . Take the effect of the heating curve previously calculated into consideration but ignore any cooling effects.

(79s)

8. A cold store is to be erected to maintain an internal temperature of -18°C with a surrounding air temperature of 25°C . It is to be constructed of concrete blocks 20cm thick and then 15cm of polystyrene foam. The external surface heat transfer coefficient is $10\text{ Jm}^{-2}\text{s}^{-1}\text{C}^{-1}$ and the internal one is $6\text{ Jm}^{-2}\text{s}^{-1}\text{C}^{-1}$, and the store is $40 \times 20 \times 7\text{ m}$ high. (a) Determine the refrigeration load due to building heat gains from its surrounding air. Assume that ceiling and floor loss rates per m^2 are one-half of those for the walls.

- (b) Determine also the distance from the inside face of the walls of the 0°C plane, assuming that the concrete blocks are on the outside.
 ((a) 14.82kJ s^{-1} (b) 5.7cm)
9. For a refrigeration system with a coefficient of performance of 2.8, if you measure the power of the driving motor and find it to be producing 8.3 horsepower:
 (a) estimate the refrigeration capacity available at the evaporator,
 (b) the tons of refrigeration extracted per kW of electricity consumed, and
 (c) the rate of heat extraction in the condenser.
 Assume the mechanical and electrical efficiency of the drive to be 74%.
 ((a) 12.82kW (b) 0.59tons per kW (c) 17.4kW, kJ s^{-1})
10. A refrigeration plant using ammonia as refrigerant is evaporating at -30°C and condensing at 38°C , and extracting 25 tons of refrigeration at the evaporator. For this plant, assuming a theoretical cycle, calculate the:
 (a) rate of circulation of ammonia, kgs^{-1}
 (b) theoretical power required for compression, kW
 (c) rate of heat rejection to the cooling water, MJ s^{-1}
 (d) COP,
 (e) volume of ammonia entering the compressor per unit time, m^3s^{-1}
 ((a) 0.0842 kgs^{-1} (b) 31.6kW (c) 0.12 MJ s^{-1} (d) 2.8 (e) $0.08\text{ m}^3\text{s}^{-1}$)
11. It is wished to consider the possibility of chilling the apples of worked Example 6.10 in chilled water instead of in air. If water is available at 1°C and is to be pumped past the apples at 0.5ms^{-1} estimate the time needed for the chilling process.
 (31mins 7s)
12. Estimate the time needed to freeze a meat sausage, initially at 15°C , in an air blast whose velocity across the sausage is 3ms^{-1} and temperature is -18°C . The sausage can be described as a finite cylinder 2cm in diameter and 15cm long.
 (37mins 5s)
13. If the velocity of the air blast in the previous example were doubled:
 (a) what would be the new freezing time?
 (b) Management then decided to pack the sausages in individual tight-fitting cardboard wraps. What would be the maximum thickness of the cardboard permissible, if the freezing time using the higher velocity of 6ms^{-1} were to be no more than it had been originally in the 3ms^{-1} air blast.
 ((a) 25mins 52s (b) 0.5mm)
14. If you found by measurements that a roughly spherical thin plastic bag, measuring 30cm in diameter, full of wet fish fillets, froze in a -30°C air blast in 16 h, what would you estimate to be the surface heat transfer coefficient from the air to the surface of the bag? The fish fillets are 80% water, and assume the density is the same as water.
 ($12.8\text{ J m}^{-2}\text{s}^{-1}\text{C}^{-1}$)