8. Evaporators

A. Introduction

8.01 The evaporator is one of the four basic and necessary hardware components of the refrigeration system. (The refrigerant may be considered as a fifth, most important, component.) Depending on the application the design of the evaporator will differ.

In the first part of this chapter different types of evaporators are presented and their applications discussed. The presentation will then focus on methods for calculating heat transfer and pressure drop, both on the refrigerant side and on the heat source side of the evaporator. Knowledge of such methods are necessary when sizing and designing refrigeration systems.

Methods of enhancing heat transfer, on the refrigerant side as well as on the heat source side are then discussed briefly.

Finally, a few words are spent on design optimization of evaporators.

It should be noted that what is presented in this chapter concerning the calculation of heat transfer and pressure drop on the heat source side of the evaporator is generally also applicable to the heat sink side of condensers.

8.02 As an introduction to the treatment of evaporators it is appropriate to recapitulate the physical processes involved. In the evaporator, the refrigerant is evaporated by the heat transferred from the heat source. The heat source may be a gas or a liquid or, e.g. in food freezers, a solid. During evaporation, the temperature of a pure refrigerant is constant, as long as the pressure does not change. The basic temperature profile through an evaporator with liquid or gas phase heat source is therefore as shown in Figure 8.02. As shown, the temperature of the refrigerant must be below that of the heat source. This low refrigerant temperature is attained as a result of the reduction in pressure caused by the compressor: When the compressor is started and the pressure reduced, the equilibrium between liquid and vapor in the evaporator is disturbed. To re-establish equilibrium, more vapor is formed through evaporation of liquid. The heat of vaporization necessary for this is taken from the liquid itself, and therefore the liquid temperature drops. As heat starts to flow from the heat source, a new equilibrium temperature is established.

In the evaporator there is thus a balance between the heat transferred to it due to the temperature difference between the evaporator and the surroundings, and the heat transferred from it in the form of heat of vaporization of the vapor drawn into the compressor.

Figure 8.02: Basic temperature profile in evaporator.

Author: Björn Palm
B. Description of different types of evaporators

Introduction

8.03 A natural way of classifying evaporators is based on the state of the heat source: gas, liquid or solid. Evaporators utilizing gas as heat source are usually referred to as air coolers as air is the dominating gaseous heat source. Evaporators for liquid heat sources are called liquid coolers. Note that it is quite common to use indirect systems, both for refrigeration/freezing and for heat pumps. In these systems the primary heat source may be air, but the heat is transferred to the evaporator by a brine or secondary refrigerant which is a liquid. The evaporator is thus a liquid cooler even though the primary heat source is air. Indirect systems are used for two prime reasons: First, this allows the system to be charged with refrigerant and tested before leaving the manufacturer, which facilitates installation, and second, the amount of refrigerant in an indirect system is much smaller than in a direct system. This is advantageous as there is less risk of damage to the local or global environment in case of a leak. Also, flammable, poisonous or strongly smelling refrigerants which may cause panic (ammonia) can safely be used with indirect systems as the refrigerant is confined to the machine room, away from public areas. Solid heat sources are rare, but are found in freezers in the food industry and in heat pumps with bedrock or soil as heat source.

8.04 Independently of the type of heat source, there are two possible ways of arranging the refrigerant flow through the evaporator:

The first is to force all refrigerant passing through the expansion device to flow through the evaporator and then on to the compressor. This is called dry expansion, or direct expansion, often abbreviated DX (see Figure 8.04a). To ensure that liquid does not reach the compressor, the refrigerant must be superheated a few degrees (usually 5-7°C) at the evaporator exit. This is achieved by using a thermostatic expansion valve. The drawback of dry expansion is that the heat transfer coefficients are low in the superheat section. It is therefore important to keep the superheat as low as possible. However, if the superheat is too low, the evaporation temperature will start to oscillate. This condition is called hunting and should be avoided. The minimum superheat which gives stable operation depends on the type and size of the evaporator and of the cooling load. For a given evaporator and expansion valve, a curve may be drawn for the minimum superheat as a function of the load. This curve is called the MSS-curve (Minimum Stable Signal) (c.f. Chapter 10).

![Figure 8.04.: a. Direct expansion evaporator | b. Flooded evaporator](image-url)
8.05 The second alternative of arranging the refrigerant flow is to collect the refrigerant after the expansion valve in a low pressure receiver. In this receiver the liquid and vapor phases are separated and the liquid fed into the evaporator while vapor is drawn into the suction line by the compressor. The refrigerant leaving the evaporator is returned to the low pressure receiver, usually as a mixture of liquid and vapor. This type of evaporator is referred to as a flooded or recirculation-type evaporator (see Figure 8.04b). The circulation of refrigerant may be arranged by free convection (thermosyphon) or by pump circulation. Several evaporators may be connected in parallel to the same low pressure receiver. One advantage of this type of evaporator is that the (average) heat transfer coefficient can be expected to be higher than for dry expansion evaporators. As the refrigerant vapor drawn from the low pressure receiver is saturated, thermostatic expansion valves cannot be used for this type of evaporator. Instead, a low pressure float valve is used to regulate the refrigerant flow.

A problem which must be considered when designing recirculation-type evaporators is the return of oil to the compressor. Different strategies have to be used with ammonia and (HC)FC refrigerants, as the oil is insoluble in, and has a higher density than, ammonia, whereas the oil is usually soluble in (HC)FC refrigerants and, if not soluble, has a lower density than these refrigerants.

8.06 In most types of evaporators the heat source fluid or the refrigerant runs through tubes. For HCFC, HFC, CFC, FC and hydrocarbons, copper tubes are used. For ammonia, other materials have to be chosen as this refrigerant is corrosive to copper. Steel tubes are most common, but aluminum tubes are also used. For corrosive heat source fluids such as sea water, flue gases and some industrial liquid or gaseous heat sources, different types of alloys (Cu-Ni, stainless steels) or even titanium, are used.

Air coolers

8.07 Air coolers are designed with the refrigerant flowing inside tubes or channels and the air passing on the outside. As the heat transfer coefficients during evaporation of the refrigerant are at least ten times those on the air side, at moderate air velocities, the outside surface area is usually increased by the use of fins. However, bare tubes are still used in older large ammonia systems. Figure 8.07 shows the interior of a cold store with bare tube evaporators. In this case, heat is transferred on the air side by free or natural convection, i.e. no fans are used to increase the air velocity across the tubes.

8.08 In most small systems as well as in modern large systems finned coil evaporators (cooling batteries, plate-fin-tube evaporators) are used. These consist of one coil, or several coils in parallel, attached to plate fins, in a housing. A fan is connected to force the air through the battery (Figure 8.08). This type of evaporators are manufactured in every size from one kW or less to hundreds of kW. The tubes are usually made of copper (aluminum or steel for ammonia), and the fins of aluminum. The fins are attached by expanding the copper tubes after the assembly of the unit. Good contact between the fin and the tube is of great importance for the heat transfer performance. Usually, the fin is shaped into a collar around the tubes. This collar then also acts as a spacer during production to get the right fin distance.
The fin spacing varies considerably depending on the application. For non-frosting conditions in small systems fin spacings down to 2 mm may be found. For large systems below 0°C spacings up to 12 mm are used.

The fin thickness is usually in the range 0.1 to 0.5 mm. The thickness is often chosen to achieve a certain stability of the fin rather than from heat transfer concerns.

The coil depths range from 40 to 500 mm. Coils with depths <200 mm are preferred, except in cases where the fin spacing is very large. With standard tube spacing this corresponds to a maximum of six tube rows.

8.09 In certain applications, e.g. where high air velocities cannot be tolerated, finned evaporator coils with free convection are being used. A typical design is shown in Figure 8.09a. With free convection, the fin spacing has to be larger than in forced convection, typically 10-15 mm.
Another type of free convection evaporator is used in most domestic refrigerators. It is called roll-bond evaporator (Figure 8.09b) and consists of two plates which have been bonded together over almost their entire surface. The area which is not bonded forms the evaporator channel. After the bonding procedure, which is done in a rolling process, the channels are inflated by high pressure. The expansion device (orifice) is often incorporated in the evaporator simply as a narrow section in the channel.

8.10 If the evaporator surface temperature is below the dew point of the air, the humidity will fall out as water or frost on the surface. As the water deposits on the surface, the heat of vaporization (and for frost, the heat of melting) is transferred to the surface together with the sensible heat of the air. This latent heat transport must be considered when calculating the capacity of the evaporator.
At evaporation temperatures below 0°C it is necessary to arrange defrosting at regular intervals. If the room temperature is above 0°C defrosting may be arranged simply by turning off the compressor. More frequently, however, some active way of defrosting is arranged. This can be done either by electric heaters mounted in parallel with the evaporator tubes in the plate fins, or by reversing the refrigerant flow. In the second case, a reversing valve is installed as shown in Figure 8.10. During defrosting, the evaporator acts as condenser and the condenser as evaporator. This is called hot gas defrosting.

As heat is given off to the surrounding during defrost it is important that it is done as quickly as possible. With electric heaters, the power should be at least three times the capacity of the evaporator. With forced convection evaporators, the fan should be turned off during defrost (except at room temperatures >0°C and when defrosting is accomplished by the room air).

The defrosting can be initiated at regular time intervals or "on demand". Several types of sensors can be used to detect the frost and initiate defrosting: In forced convection evaporators, the air pressure drop or the fan power can be used as frost indicators. In both forced and free convection systems it is possible to measure the difference between the evaporation temperature and the air temperature. With frost forming on the surfaces this difference will increase. Photocells, infra-red detectors and electric conductivity-sensors have also been used as frost detectors.

The defrosting is usually aborted by a defrosting thermostat mounted in between the fins of the evaporator, terminating the defrosting when the temperature has increased above 0°C. Alternatively, defrosting may be terminated simply after a pre-set time period.

The water formed during defrost is collected on a defrost pan below the evaporator and led through a tube to a drain outside the freezing room. Both the defrost pan and the tube must have electric heating to keep the water from freezing.

More information about frosting and defrosting of evaporators is given in Chapter 15 (part II)

**Liquid Coolers**

8.11 Liquid coolers are primarily used for cooling of brine in indirect systems, i.e. in systems where the cooling capacity is distributed by water with anti-freeze additives. They are also used for cooling of other liquids in industrial applications.

Figure 8.11 shows two different types of shell and tube evaporators. In the first, the refrigerant flows inside the tubes, while in the second, the brine flows in the tubes.

With refrigerant inside the tubes, the refrigerant may be evaporated completely (dry expansion). Notice the baffle plates which force the brine to flow up and down across the evaporator tubes, increasing the flow velocity, and thereby the heat transfer coefficient. The

![Figure 8.10: System with hot gas defrosting](image-url)
baffles also result in a more ordered temperature distribution in the brine. This is a common type of liquid cooler for both small and large systems. Evaporation inside tubes has some advantages compared to outside evaporation: First, there is no risk of oil accumulating in the evaporator as the flowing refrigerant will transport the oil to the evaporator exit, second, the refrigerant charge will be much smaller.

An advantage of shell-side evaporation on the other hand is that the brine/liquid side is more easily accessible for cleaning. This is important for example in heat pump systems recovering heat from sewage water.

With shell side evaporation, the oil return problem is identical to that in other types of flooded evaporators as discussed above.

8.12 For smaller systems, coaxial evaporators have been widely used. This type of evaporator consists of two coaxial tubes, usually wound up as shown in Figure 8.12. The refrigerant runs in the inner tube. This tube may be smooth or have some type of surface structure to increase the surface area on either of, or both, the inner and outer surface. Coaxial evaporators are intended for dry expansion.
8.13 Since their introduction, brazed plate heat exchangers have steadily gained in popularity as evaporators for small systems (Figure 8.13). This type has to a large extent ousted the coaxial evaporators from the market. The plate heat exchangers are extremely compact, and therefore also have a small internal volume. This is important as it results in low refrigerant charges. This type of evaporator is generally recognized to give high overall heat transfer coefficients. The available sizes of brazed plate heat exchangers are steadily increasing and they are now manufactured for cooling capacities above 400 kW. As copper is normally used for the brazing, brazed plate heat exchangers cannot be used for ammonia. Instead, semi-welded plate heat exchangers are used. In these, pairs of plates are welded together forming the refrigerant flow path. The welded pairs are stacked in a frame, held by bolts, with gaskets in between each pair. This type of evaporators is manufactured for cooling capacities up to 5000 kW and more.

8.14 Another type of plate evaporator is used in large heat pumps when heat is extracted from lake- sea- or sewage water. In this type, the refrigerant is enclosed between two vertical steel
plates, and water is allowed to flow across the outside of the plates (Figure 8.14). With this design, the evaporator is easy to clean and there is no risk of damage due to freezing. Low temperature water can therefore be used as a heat source, and the water can be cooled to temperatures close to 0°C.
Solid heat source evaporators

As already mentioned, solid heat source evaporators are used only in special applications such as freezers in the food industry and heat pumps using the bedrock or soil as heat source. In general, the heat transfer is extremely good between the evaporator and the solid heat source. The heat transfer rate is in this case determined by the thermal conductivity of the solid material.

A contact plate freezer for packaged foodstuff is shown in Figure 8.15. The packages, containing e.g. fish fillets or chopped vegetables, are placed in-between horizontal evaporator plates. The good thermal contact ensures quick freezing of the product.

Ground source heat pump evaporators usually consist of a plain copper tube buried in the ground or inserted into a water-filled hole in the bedrock. When the soil is used as heat source, it is primarily the latent heat released as the water in the soil freezes which is extracted. The water content in the soil thus has a large influence on the necessary tube length.

Figure 8.15.: Contact plate freezer for packaged foodstuff.
C. Heat transfer

Introduction

As an introduction to heat transfer we will return to the Figure of the temperature profile in the evaporator (Figure 8.16). Here, the temperature profile is slightly idealized as the influence of friction pressure drop and superheat of the refrigerant at the evaporator exit are neglected. Also, we are considering a pure fluid, not a refrigerant mixture.

If we study only a small part of the evaporator we may consider the temperature difference between the fluids as constant and the heat flow across the small area \( A_s \) is then calculated as

\[
\dot{Q} = U \cdot A_s \cdot \dot{\theta}
\]

8.16a

When considering the whole evaporator we have to take into account that the temperature difference changes from one end of the evaporator to the other. The cooling capacity of the evaporator is then calculated as

\[
\dot{Q} = U \cdot A \cdot \dot{\theta}_m
\]

8.16b

where

\( \dot{Q} \) = cooling capacity (W)
\( U \) = overall heat transfer coefficient (W/(m\(^2\) K))
\( \dot{\theta}_m \) = mean temperature difference between the fluids (K)

With the idealization mentioned above, and assuming that the overall heat transfer coefficient is constant along the evaporator, it can be shown that the mean temperature difference \( \dot{\theta}_m \) should be calculated as the logarithmic mean of the inlet and outlet temperature differences.
Eq. 8.16b can be used to calculate the capacity of an evaporator with a known surface area, or to calculate the area necessary to achieve a certain capacity. In either case the overall heat transfer coefficient and the mean temperature difference must be known. The overall heat transfer coefficient is primarily a function of the surface heat transfer coefficients on the refrigerant side and on the heat source side, and most of the remainder of this chapter will be devoted to the calculation of these heat transfer coefficients.

The mean temperature difference, however, can be chosen arbitrarily as long as \( Q \) and \( A \) are not both fixed. The choice of the temperature difference is a matter of economic optimization rather than technical: If a small temperature difference is selected, a large (expensive) surface area is necessary - if the temperature difference is large, a small area will be sufficient, but the \( COP \) of the plant will be low, resulting in high running costs. In the last part of the chapter some guidelines for economical optimum temperature differences will be given.

**Example - Mean temperature difference**

A cooling battery is installed in a cold room. The room temperature is +6°C and the air temperature is reduced by 6°C as it passes through the battery. Calculate the logarithmic mean temperature difference if the evaporation temperature is -2°C.

**Solution:**

The temperature differences at the inlet and outlet are, respectively:

\[
\vartheta_i = +6 - (-2) = 8 \text{ K}
\]

\[
\vartheta_o = 0 - (-2) = 2 \text{ K}
\]

According to eq. 8.16b the logarithmic mean temperature difference is then

\[
\vartheta_m = \frac{\vartheta_i - \vartheta_o}{\ln\left(\frac{\vartheta_i}{\vartheta_o}\right)} = \frac{8 - 2}{\ln\left(\frac{8}{2}\right)} = 4.33\text{K}
\]

**8.18** The overall heat transfer coefficient is defined by the equation 8.16a. Note that the surface areas usually are different on the refrigerant side and on the heat source side and that the overall heat transfer coefficient will be different depending on which area it is referred to. Because of this, it is convenient to speak of the evaporator’s \( U \cdot A \)-value instead of the overall heat transfer coefficient.

**8.19** The heat transfer on either side of the heat transfer wall is calculated by Newton’s law of cooling, which is also the defining equation for the (film) heat transfer coefficient \( \alpha \):

\[
\dot{Q} = \alpha \cdot A \cdot \vartheta
\]

where \( \vartheta \) is the (local or average) temperature difference between the fluid and the surface.
The heat transfer by conduction through the heat transfer wall is determined by Fourier’s law, which is the defining equation for the thermal conductivity

\[ \dot{Q} = -\lambda \cdot A \cdot \frac{\delta t}{\delta x} \]  

8.19b

For a plane surface at steady state (\( \dot{Q} \), \( \lambda \) and \( A \) all constant) this simplifies to

\[ \dot{Q} = \frac{\lambda}{\delta} \cdot A \cdot \theta \]  

8.19c

where \( \delta \) is the thickness of the wall and \( \theta \) is the temperature difference between the inside and outside surfaces (the minus sign is now incorporated in \( \theta \)).

8.20 It is often convenient to speak of thermal resistances rather than heat transfer coefficients and thermal conductivities. These resistances are analogous to electric resistances in electric circuits and are defined by the thermal counterpart to Ohm’s law for electric circuits

Ohm’s law: 
Electric resistance = Electric potential / Current

The thermal resistance \( R \) is thus defined as

\[ R = \frac{\theta}{\dot{Q}} \]  

8.20a

Comparing this to equation 8.16a we find that the total thermal resistance \( R_t \) is the inverse of the \( U \cdot A \)-value

\[ R_m = \frac{\theta_m}{\dot{Q}} = 1 / (U \cdot A) \]  

8.20b

Likewise, we may define the thermal resistances between any of the two fluids and the heat transfer wall as

\[ R_f = 1 / (\alpha \cdot A_f) \]  

8.20c

and the resistance in the heat transfer wall as

\[ R_w = \frac{\delta}{(\lambda \cdot A_w)} \]  

8.20d

where \( A_w \) is some mean of the outside and inside surface areas. (For a cylindrical wall \( A_w \) is the logarithmic mean, for a sphere it is the geometric mean).

8.21 As the heat flow is constant in the flow direction, we have

\[ \dot{Q} = U \cdot A \cdot \theta_m = \alpha_f \cdot A_f \cdot \theta_f = \lambda \cdot A_w \cdot \theta_w / \delta = \alpha_2 \cdot A_2 \cdot \theta_2 \]  

8.21a
This relation can be used to express the temperature differences (e.g. $\vartheta_m = \dot{Q} / (U \cdot A)$). But the temperature differences are related by

$$\vartheta_m = \vartheta_1 + \vartheta_w + \vartheta_2$$  \hspace{1cm} 8.21b

By inserting the expressions for the temperature differences we find that

$$\frac{1}{U \cdot A} = \frac{1}{\alpha_1 \cdot A_1} + \frac{\delta}{\lambda \cdot A_w} + \frac{1}{\alpha_2 \cdot A_2}$$  \hspace{1cm} 8.21c

or

$$R_m = R_1 + R_w + R_2$$  \hspace{1cm} 8.21d

The total thermal resistance is thus the sum of the thermal resistances in the two fluids and in the intermediate wall, and the overall heat transfer coefficient is a function of the film heat transfer coefficients in the two fluids, of the surface areas and of the thickness and conductivity of the wall.

Figure 8.21 shows schematically the temperatures and the temperature differences for a section of a smooth tube evaporator.

8.22 For finned surfaces, the surface temperature of the fin will increase towards the tip of the fin as is shown in Figure 8.22. Because of the lower temperature difference between surface and fluid, the heat flux due to convection will be lower on the fin than on the base surface in between fins. The fin surface is thus less efficient than the base surface, and a fin efficiency $\xi_f$ is defined as the ratio between the actual heat transferred and the heat which would be transferred if the whole fin had the temperature of the fin base. Assuming equal heat transfer coefficients on the fin and on the base, this ratio is equal to the ratio between the mean fin temperature difference, $\vartheta_{m, \text{fin}}$, and the base temperature difference, $\vartheta_{\text{base}}$ (see Figure 8.22a).
For a straight fin with constant cross section, it may be shown that the fin efficiency may be calculated as

$$\xi_f = \frac{\alpha \cdot A \cdot \dot{\vartheta}_{m, fin}}{\alpha \cdot A \cdot \dot{\vartheta}_{base}} = \frac{\dot{\vartheta}_{m, fin}}{\dot{\vartheta}_{base}}$$ \hspace{1cm} (8.22a)$$

For other types of fins, the fin efficiency may be estimated from the diagram in Figure 8.22b.

Figure 8.22a.: Local temperature profile for finned surface.

For a straight fin with constant cross section, it may be shown that the fin efficiency may be calculated as

$$\xi_f = \frac{\tanh(m \cdot L)}{m \cdot L}$$ \hspace{1cm} (8.22b)$$

where

$$m = \left( \frac{\alpha \cdot P}{\lambda \cdot A} \right)^{1/2}$$ \hspace{1cm} (8.22c)$$

where

- $\alpha = \text{heat transfer coefficient (W/(m}^2\text{ K))}$
- $P = \text{perimeter of the fin (m)}$
- $A = \text{cross section area of the fin (m}^2\text{)}$
- $\lambda = \text{thermal conductivity of the fin (W/(m K))}$

For other types of fins, the fin efficiency may be estimated from the diagram in Figure 8.22b.

When determining the overall heat transfer coefficient, or the total thermal resistance, for a finned surface, the area for the finned side is calculated as

$$A_{fin} = A_{base} + \xi_f \cdot A_{fin}$$ \hspace{1cm} (8.22d)$$

and the overall heat transfer coefficient from:
Figure 8.22b.: Fin efficiency for different types of fins. (Ekroth/Granryd 1994)
\[ \frac{1}{U \cdot A} = \frac{1}{\alpha_1 \cdot A_1} + \frac{\delta}{\lambda \cdot A_w} + \frac{1}{\alpha_2 \cdot (A_{base} + \xi_f \cdot A_{fin})} \]  

8.22e

where

- \( A_1 \) = inside surface area (m\(^2\))
- \( A_{base} \) = outside surface area in between fins (m\(^2\))
- \( A_w \) = logarithmic mean of in and outside surface areas of tube (m\(^2\))
- \( A_{fin} \) = fin area (m\(^2\))

8.23 Before going into the details of calculating the surface heat transfer coefficients, approximate values of overall heat transfer coefficients for different types of evaporators are presented in Table 8.23a. The values are referred to the logarithmic mean temperature difference and to the total surface area on the heat source side.

In Table 8.23b approximate values for film heat transfer coefficients are presented. For cases where the heat transfer areas are known, this table can be used for estimating the overall heat transfer coefficient. Also, by comparing the two tables, the dominating thermal resistance

<table>
<thead>
<tr>
<th>Type of evaporator</th>
<th>Fig. no</th>
<th>U ((\text{W}/(\text{m}^2 \text{K})))</th>
<th>Notes</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Air coolers</strong></td>
<td></td>
<td></td>
<td></td>
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<tr>
<td>Free convection</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Smooth tubes</td>
<td>8.07</td>
<td>9-13</td>
<td>Depending on temp. difference, tube diameter and position</td>
</tr>
<tr>
<td>Plates</td>
<td>8.09</td>
<td>12-14</td>
<td>Below 0°C, ( \leq 12 \text{ W}/(\text{m}^2 \text{K}) )</td>
</tr>
<tr>
<td>Finned coils</td>
<td>8.09</td>
<td>4-8</td>
<td>Depending on temperature difference, fin distance, number of tube rows, position</td>
</tr>
<tr>
<td>Forced convection</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Smooth tubes</td>
<td></td>
<td>30-60</td>
<td>Air velocity 3-5 m/s, Tube diameter 25-50 mm</td>
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<td>Finned coils</td>
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<td>12-25</td>
<td>Air velocity 2-4 m/s, Heat flux at inner surface 2000-5000 W/(m(^2)), Dry expansion</td>
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<td><strong>Liquid coolers</strong></td>
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<td>Free convection</td>
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<tr>
<td>Smooth tubes in brine tank,</td>
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<td>100-170</td>
<td>Brine temp. -30-0°C, U reduced by 20% in dry expansion</td>
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<td>refrigerant inside tubes</td>
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<tr>
<td>Smooth tubes in brine tank,</td>
<td></td>
<td>120-170</td>
<td>Brine temp. -30-0°C, Brine velocity inside tubes 1.5 m/s</td>
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<tr>
<td>brine inside tubes</td>
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<td></td>
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<tr>
<td>Smooth tubes in water tank,</td>
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<td>180-220</td>
<td>Brine velocity 0.5 - 1.5 m/s, Brine temp. -8 - 0°C</td>
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<tr>
<td>brine inside tubes</td>
<td></td>
<td></td>
<td></td>
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<tr>
<td>Forced convection</td>
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<td></td>
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<tr>
<td>Shell and tube, brine inside tubes,</td>
<td>8.11a</td>
<td>200-500</td>
<td>Brine velocity 0.5 - 1.5 m/s</td>
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<td>NH(_3) as refrigerant</td>
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<tr>
<td>Shell and tube, brine inside tubes,</td>
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<td>150-400</td>
<td>Brine velocity 0.5 - 1.5 m/s</td>
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<td>(HC)FC as refrigerant</td>
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<td>Shell and tube, (HCFC)-</td>
<td>8.11b</td>
<td>250-600</td>
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<td>refrigerant inside tubes</td>
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<td>Brazed plate heat exchanger,</td>
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<td>800-2500</td>
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<td>(HC)FC as refrigerant</td>
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can be identified.
### Table 8.23b: Approximate film heat transfer coefficients
*(Based on Ekroth/Granryd, 1994)*

<table>
<thead>
<tr>
<th>Type of flow</th>
<th>Approximate heat transfer coefficient (W/(m²·°C))</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Turbulent flow in tubes,</strong> <em>(diameter ≈ 50 - 25 mm)</em></td>
<td></td>
</tr>
<tr>
<td>Water (0.5 - 5 m/s)</td>
<td>1500 - 20000</td>
</tr>
<tr>
<td>Air (1 - 10 m/s)</td>
<td>10 - 50</td>
</tr>
<tr>
<td><strong>Laminar flow in tubes,</strong> <em>(diameter ≈ 50 - 10 mm)</em></td>
<td></td>
</tr>
<tr>
<td>Water</td>
<td>50 - 250</td>
</tr>
<tr>
<td>Air</td>
<td>2 - 15</td>
</tr>
<tr>
<td><strong>Air flow past plates</strong> <em>(1 - 10 m/s)</em></td>
<td></td>
</tr>
<tr>
<td>Water</td>
<td>50 - 250</td>
</tr>
<tr>
<td>Air</td>
<td>2 - 15</td>
</tr>
<tr>
<td><strong>Natural convection</strong></td>
<td></td>
</tr>
<tr>
<td>Water</td>
<td>200 - 1000</td>
</tr>
<tr>
<td>Air</td>
<td>2 - 10</td>
</tr>
<tr>
<td><strong>Condensation</strong></td>
<td></td>
</tr>
<tr>
<td>Water</td>
<td>5000 - 15000</td>
</tr>
<tr>
<td>Refrigerants</td>
<td>1000 - 5000</td>
</tr>
<tr>
<td><strong>Boiling</strong></td>
<td></td>
</tr>
<tr>
<td>Water</td>
<td>1000 - 40000</td>
</tr>
<tr>
<td>Refrigerants</td>
<td>200 - 5000</td>
</tr>
</tbody>
</table>
Heat transfer in boiling

8.24 The physical mechanisms of boiling processes are extremely complicated and cannot be modeled in any detail. The correlations used for calculating heat transfer coefficients in boiling are therefore empirical or semi-empirical.

8.25 Boiling of a stagnant pool of liquid (no forced convection of the liquid) is referred to as pool boiling. Depending on the heat flux different types of pool boiling may appear (c.f. Figure 8.25). At low heat fluxes, heat is transferred from the heated surface to the liquid without bubble formation. The liquid close to the heated surface is slightly superheated, and the heat is transferred through the liquid by free convection to the liquid surface (which has saturation temperature) where evaporation takes place. As there is no bubble formation at the heated surface this is usually not referred to as boiling but as free convection evaporation or surface evaporation. Heat transfer in this region can be calculated by correlations for free convection.

At higher heat flux, bubble formation will start (point A in Figure 8.25), resulting in higher heat transfer coefficients. This is the nucleate boiling region and it is by far the most important region for technical applications. The high heat transfer coefficients can be attributed to vigorous convection in the liquid in conjunction to the bubble formation, and to the bubble formation itself. As the heat flux is increased, the amount of bubbles at the surface will increase steadily, and eventually the bubbles will form a vapor layer insulating the surface from the surrounding liquid (point C in Figure 8.25). Increasing the heat flux above this critical value will result in drastically reduced heat transfer coefficients, which in turn will lead to a sudden increase in the temperature difference (to point D). As the temperature at this point may well be above the melting point of the heated surface, point C is referred to as the burnout point and the corresponding heat flux as the critical heat flux. For the boiling in refrigeration

Figure 8.25.: The boiling curve: Heat flux vs. temperature difference for a heated
evaporators, however, the heat fluxes are far below this point.

8.26 Free convection heat transfer can generally be expressed in dimensionless form as

\[
\text{Nu} = C \cdot (\text{Gr} \cdot \text{Pr})^n \quad 8.26a
\]

where

\[
\begin{align*}
\text{Nu} &= \frac{a \times H}{l} \quad \text{Nusselt no} \\
\text{Gr} &= g \beta \cdot \Delta H^2 / v^2 \quad \text{Grashof no} \\
\text{Pr} &= \mu \cdot c_p / \lambda \quad \text{Prandtl no}
\end{align*}
\]

This expression is also valid for the free convection evaporation region. The constant C and the exponent n depend on the geometry and on the product (Gr·Pr) and are given for a few cases in table 8.26. The characteristic length used in the Grashof no is the height for vertical plates or cylinders, and the diameter for horizontal tubes.

**Table 8.26, adapted from Holman, 1992**

<table>
<thead>
<tr>
<th>Geometry</th>
<th>Gr·Pr</th>
<th>C</th>
<th>n</th>
</tr>
</thead>
<tbody>
<tr>
<td>Horizontal cylinders</td>
<td>$10^4$ - $10^9$</td>
<td>0.53</td>
<td>0.25</td>
</tr>
<tr>
<td></td>
<td>$10^9$ - $10^{12}$</td>
<td>0.13</td>
<td>0.33</td>
</tr>
<tr>
<td>Vertical planes and cylinders</td>
<td>$10^4$ - $10^9$</td>
<td>0.59</td>
<td>0.25</td>
</tr>
<tr>
<td></td>
<td>$10^9$ - $10^{13}$</td>
<td>0.10</td>
<td>0.33</td>
</tr>
</tbody>
</table>

The factors (Gr·Pr/(ΔT·H³)) for some pure refrigerants are shown in Figure 8.26 as a function of temperature.

In the turbulent region (i.e. when Gr·Pr > $10^9$) the heat transfer coefficient is independent of the characteristic length and the above equation leads to the expression

\[
\alpha = K_f \cdot \dot{q}^{0.25} \quad 8.26b
\]
where \( \dot{q} = \frac{Q}{A} \) = surface heat flux (W/m\(^2\))

and \( K_f = \text{constant} \)

Through the relation \( \dot{q} = \alpha \cdot \Theta \) we may also express the temperature difference as

\[
\Theta = \frac{\dot{q}^{0.75}}{K_f}
\]

The values of \( K_f \) are different for each refrigerant, but are fairly independent of temperature. Table 8.26b shows the values for horizontal tubes for a few pure refrigerants at -10\(^\circ\)C. For vertical plates or cylinders the values should be multiplied by 0.82.

### Table 8.26b: Values of constant \( K_f \) in equations 8.26 b and c

<table>
<thead>
<tr>
<th>( K_f )</th>
<th>R12</th>
<th>R22</th>
<th>R134a</th>
<th>R32</th>
<th>R290</th>
<th>NH(_3)</th>
</tr>
</thead>
<tbody>
<tr>
<td>(W/m(^2))(^{0.75})/K</td>
<td>38</td>
<td>48</td>
<td>43</td>
<td>66</td>
<td>46</td>
<td>103</td>
</tr>
</tbody>
</table>

8.27 For the nucleate boiling region, several correlations for the heat transfer coefficient are found in the literature. A common characteristic of these are that

\[
\alpha \propto \dot{q}^m \quad \text{where} \quad 0.6 < m < 0.8
\]

This relation is valid also for some types of flow boiling. Another common characteristic is that the heat transfer coefficient increases with increasing pressure.

When different correlations are compared, it is found that the results may differ considerably, often by 30% or more. The reader is therefore advised not to expect too high accuracy in the estimation of pool boiling coefficients.

The following simple correlation which is considered to be fairly accurate was proposed by Cooper (1984).

\[
\alpha_{nb} = C \cdot 55 \cdot p_r^{(0.12 \cdot 0.2 \cdot \log R_p)} \cdot (-\log p_r)^{0.55} \cdot M^{-0.5} \cdot \dot{q}^{0.67}
\]

where

- \( p_r = \text{reduced pressure} (= p / p_{crit}) \)
- \( R_p = \text{surface roughness (\(\mu\)m)} \)
- \( M = \text{molecular weight} \)

The value of the constant \( C \) is 1 for horizontal, plane surfaces and 1.7 for horizontal copper tubes, according to Cooper’s original paper. However, comparisons with experimental data suggests that better agreement is achieved if a value of 1 is used also for horizontal tubes.

Note that the heat transfer coefficient is a fairly weak function of the surface roughness parameter \( R_p \), which is seldom well known. A value of \( R_p = 1 \) is suggested for technically smooth surfaces. The correlation may thus be expressed in the simple form

\[
\alpha_{nb} = K_{nb} \cdot \dot{q}^{0.67}
\]

where \( K_{nb} \) is a function of the reduced pressure and the molecular weight of the refrigerant.
8.28 The above relation is valid for single smooth tubes and smooth horizontal surfaces. For tube bundles the average heat transfer coefficient will be slightly higher as the heat transfer is enhanced by the convection caused by the rising bubbles. This can be taken into account by multiplying the heat transfer coefficient for the single tube by a bundle factor. For plain and low finned tubes this factor typically is about 1.5.

A better estimate may be found by using a method proposed by Gorenflo (1993). According to this method, a mean heat transfer coefficient for the bundle is calculated as

\[
\alpha_{\text{mean}} = (\alpha_{\text{nb,one}} + f \cdot \alpha_f) \left[ 1 + (2 + \dot{q})^{-1} \right]
\]

where
- \(\alpha_{\text{nb,one}}\) = heat transfer coefficient for single tube in nucleate boiling
- \(f\) = factor depending on bundle size and inlet velocity (0.5 < \(f\) < 1)
- \(\alpha_f\) = free convection heat transfer coefficient (according to 8.26)
- \(\dot{q}\) = heat flux in kW

8.29 Pool boiling evaporators are often made of finned tubes. If the surface heat flux and the heat transfer coefficients both are referred to the total surface area, the boiling heat transfer coefficients of the finned tubes are slightly higher than for smooth tubes. The increase can be attributed to good contact between the surface and the bubbles as these rise in between the fins.

8.30 Flow boiling inside tubes or channels is considerably more complex than pool boiling. As part of the liquid is evaporated, the volume of the refrigerant is increased greatly, causing an
acceleration of the fluid. Investigations of flow boiling have shown that the process may be divided into different flow regimes (Figures 8.30a and 8.30b). The mechanisms of heat transfer are different from one regime to another. Two main heat transfer mechanisms may be discerned: nucleate boiling and convective boiling. In convective boiling, nucleation is suppressed and evaporation takes place at the liquid/vapor interface. To calculate the heat transfer at each point along an evaporator tube it would be necessary to know where one flow regime ends and another starts. A number of investigators have studied the flow regimes in two phase flow, and their dependence on different parameters. The results are usually presented as flow-regime maps where the regime is shown as a function of e.g. mass flux and vapor quality. Studies of the flow in vertical and horizontal tubes have shown that the flow regimes are basically independent of the orientation of the tube. Only at low heat- or mass fluxes the flows are considerably different, as, in horizontal tubes, the gravitational forces will stratify the flow.
Stratification has a large influence on the heat transfer as the coefficient of heat transfer is very low between the vapor and the tube wall. The dry part of the tube in this case acts as an extended surface, transferring heat to the submerged portion of the tube wall.

It is generally said that when the vapor quality is less than 5-8% (Rohsenow and Hartnett 1973) the heat transfer mechanism is closely related to that in pool boiling, the main difference being that in forced convection boiling there is a velocity parallel to the heated surface. In evaporators of heat pumps and refrigeration equipment however, the heat flux is often too small to initiate nucleate boiling even in the low quality region. In vertical evaporators, like plate heat exchangers, the heat transfer coefficient at low heat fluxes are found to be reasonably well predicted by pool boiling correlations.

At higher vapor qualities, the acceleration of the vapor-liquid mixture causes an annular flow, with vapor traveling at high speed in the center of the tube, and liquid covering the walls. In some cases the vapor contains a large amount of liquid drops. In annular flow, heat is transferred by convection through the liquid film and the evaporation takes place at the liquid-vapor interface. At low heat- or mass fluxes, annular flow may never be reached, and the flow may be more or less stratified along the whole evaporator.

Finally, at a vapor quality somewhere between 70 and 100%, the liquid film dries out and only liquid droplets, dispersed in the vapor flow, remain to be evaporated. This kind of flow is called mist flow or dispersed flow. Here, heat is transferred partly by convection through the superheated vapor to the liquid drops, but also by the drops hitting the walls and evaporating on contact.
Figures 8.30a and b indicate that other regimes than the three discussed above may be present in the tube, but often three flow regimes are considered enough to describe boiling heat transfer inside a tube.

8.31 As stated above, different heat transfer mechanisms are dominant in the different flow regimes, and for an accurate calculation of the heat transfer coefficients along an evaporator tube it would be necessary, first, to have an accurate correlation for predicting the flow regimes and, second, to have accurate correlations for the heat transfer coefficient in each regime. As of today there are no such correlations available, and the heat transfer is estimated by simplified empirical or semi-empirical correlations.

![Figure 8.31a: Schematic diagram of heat transfer coefficient in flow boiling as a function of heat flux and mass flux.](image-url)
In many cases in traditional refrigeration applications the heat transfer coefficient does not vary very much along the evaporator. It is then possible to use correlations for the mean heat transfer coefficient for the whole evaporator. Such correlations also have the benefit of being very simple to use compared to correlations for the local heat transfer coefficient.

In some more general correlations the local heat transfer coefficient is considered to be the net effect of the two mechanisms, nucleate boiling and convective evaporation. Some general observations can be stated concerning the heat transfer in these two regimes:

When nucleate boiling is dominant, heat transfer is strongly dependent of heat flux ($\alpha \propto \dot{q}^{0.7}$) but only weakly dependent on mass flux and vapor fraction.

In the convective evaporation regime, the heat transfer coefficient is almost independent of heat flux, but increases with increasing mass flux and increasing vapor fraction. These general conclusions are shown schematically in Figures 8.31a and b. In both regimes, the heat transfer coefficient increases with increasing pressure.

8.32 One of the most well known correlations for the mean heat transfer coefficients was proposed by Pierre (1953). His original work was based on the boiling of R12 in horizontal copper tubes, heated by flowing water in a concentric tube. Later, the correlation was confirmed by Pierre (1957 and 1969) for R22 and R502 by further experiments and for R11 and methyl chloride by using experimental results of other investigators. Pierre (1969) gives two correlations, one for complete evaporation ($5 - 7$ K superheat) and one for incomplete evaporation:

- **Complete evaporation:** $\text{Nu}_m = 1.0 \cdot 10^{-2} \cdot (\text{Re}^2 \cdot K_f)^{0.4}$
- **Incomplete evaporation:** $\text{Nu}_m = 1.1 \cdot 10^{-3} \cdot \text{Re} \cdot K_f^{0.5}$

\[ \text{Nu}_m = \alpha_{\text{mean}} \left( \frac{d}{\lambda} \right) \]
\[ \text{Re} = 4 \cdot \frac{m}{(\pi \cdot d \cdot \mu)} \]

where $\text{Nu}_m$ = mean Nusselt number
$\alpha_{\text{mean}}$ = mean heat transfer coefficient
$\lambda$ = thermal conductivity
$d$ = tube diameter
$\mu$ = dynamic viscosity
$\dot{m}$ = mass flow rate
$\text{Re}$ = Reynolds number
Kf = Δh / (L · g)  
πerreboiling number
Δh = change in enthalpy between inlet and outlet (J/kg)
L = tube length (m)
g = acceleration due to gravity

Conditions:  
Re² · Kf < 3.5 · 10¹¹
Nu_m < 420

Since the article was published, more accurate data on the thermal conductivity and viscosity of refrigerants has been presented. The change in property data will directly influence the constants in the two equations and it is therefore suggested that the constants are reduced by 15%, if recent data for conductivity and viscosity is used.

Experimental results have shown that the Pierre equations give accurate results also with modern (HC)FC refrigerants (except R32). However, the predictions are according to some sources less accurate for hydrocarbon refrigerants (Melin, 1996).

8.33 One of several equations for the local heat transfer coefficient in which the heat transfer is assumed to be the sum of a nucleate boiling and a convective contribution was suggested by Gungor and Winterton (1986):

\[ \alpha_{lp} = \alpha_{nuc} + \alpha_{conv} \]  

The nucleate boiling contribution \( \alpha_{nuc} \) is calculated from the pool boiling correlation of Cooper (eq. 8.27a). However, as the nucleate boiling may be suppressed by the convection, they included a suppression factor, \( S \), such that

\[ \alpha_{nuc} = \alpha_{pb} \cdot S \]  

The convective contribution they assumed could be described by a modified form of the Dittus-Boelter equation for one phase liquid flow (c.f. eq. 8.62a), multiplied by an enhancement factor \( E \), as the heat is transferred in this case through a thin liquid film which is disturbed at the surface by the evaporation:

\[ \alpha_{conv} = \alpha_{liq} \cdot E \]  

where
\[ \alpha_{liq} = (k_l/d) \cdot 0.023 \cdot Re_l^{0.8} \cdot Pr_l^{0.4} \]  

and
\[ Re_l = G \cdot d \cdot (1-x)/\mu_l \]  
\[ G = \text{mass flux (kg/(m}^2 \text{s))} \]
\[ \mu_l = \text{liquid dynamic viscosity (Ns/m}^2 \text{)} \]
\[ x = \text{vapor fraction} \]
\[ Pr_l = \text{liquid Prandtl number} \]
The enhancement factor is calculated as 
\[ E = 1 + 24000 \cdot Bo^{1.16} + 1.37 \cdot (1/X_{tt})^{0.86} \] 
where 
\[ Bo = \frac{\dot{q}}{(G \cdot r)} \]
\[ X_{tt} \equiv \left( \frac{1-x}{x} \right)^{0.9} \cdot \left( \frac{\rho_r}{\rho_l} \right)^{0.5} \cdot \left( \frac{\mu_r}{\mu_l} \right)^{0.1} \]

The suppression factor was found to be a function of the enhancement factor 
\[ S = \left( 1 + 1.15 \cdot 10^{-6} \cdot E^2 \cdot Re_l^{1.17} \right)^{-1} \]

For horizontal tubes and Froude numbers \( Fr_L = G^2/(p^2 \cdot g \cdot d) < 0.05 \) the enhancement- and suppression factors should be modified as follows:
\[ E_{mod} = E \cdot Fr_L^{0.12} \cdot Fr_h \]
\[ S_{mod} = S \cdot (Fr_L)^{1/2} \]

The correlation gives local heat transfer coefficients (circumferential averages for each point along the evaporator tube). When using this type of correlation it is necessary to divide the evaporator tube into five to ten sections, calculate a local value in the middle of each section and consider this as an average for the section.

For most types of evaporators the correlations presented above can be used directly. For plate heat exchangers, however, the refrigerant flow channel is neither a liquid pool nor a circular tube. As of today there are no correlations suggested in the open literature for this type of heat exchanger. The designer is therefore more or less in the hands of the manufacturers who usually supply computer programs for choosing the correct size of heat exchanger. Test results published in the literature suggest that the heat transfer coefficients of plate heat exchangers is the sum of a nucleate boiling contribution and a convective contribution just as is the case for boiling in tubes. Some test data has been well correlated just using Cooper’s correlation for pool boiling (eq. 8.27a).

So far, we have assumed that the refrigerant is a pure substance and thus evaporates at constant temperature along the evaporator. For most refrigerant mixtures, however, this is not true and this has a considerable influence on heat transfer. Refrigerant mixtures were discussed in some detail in Chapter 5, and the reader is assumed to be familiar with the terminology introduced there. In the following section we will concentrate on the implications on heat transfer of using mixtures.

When a zeotropic mixture boils the vapor formed has a higher concentration of the most volatile component than the boiling liquid. The composition of the liquid will therefore shift during evaporation towards higher concentration of the less volatile component and, because of this shift, the boiling point will increase. The difference in temperature between that at which boiling starts (the bubble point) and that at which all liquid is evaporated (the dew point) is called the glide of the mixture. The glide is a function of the components, the concentration of the components and of the pressure. For some zeotropic mixtures the glide is small and has therefore little influence on the system. These are referred to as near-azeotropes. There is no strict definition of how large the glide can be of a near-azeotrope, but a glide of a couple of degrees will have a noticeable influence on heat transfer and on the system.
If a zeotropic refrigerant is used in a pool boiling or re-circulation type evaporator, the concentration of the less volatile component will be higher than in the original charge in the liquid in the evaporator and lower in the circulating refrigerant. This leads to an increase in the condensing pressure and a reduction in the evaporation pressure, which in turn leads to a decrease in the capacity of the system.

The heat transfer coefficient in the evaporator will be lower than the weighted average of that of the pure components. The main reason for this is that as the liquid boils, the more volatile component is concentrated in the bubbles and transferred to the liquid surface, while the less volatile component is concentrated close to the heated wall. This leads to an increase in the bubble point at the wall and therefore also to an increase in the temperature difference between the wall and the liquid bulk (Figure 8.36).

If the heat transfer coefficient is defined as for a pure fluid

\[ \alpha = \frac{q}{(t_{\text{wall}} - t_{\text{bubble}})} \]

where \( t_{\text{bubble}} \) is now the bubble point of the liquid bulk it is evident that the extra temperature difference associated with the difference in concentration will seem to decrease the heat transfer coefficient.

A second reason for the decrease in heat transfer is that the thermodynamic properties (conductivity and viscosity) of a mixture are usually less favorable to heat transfer than the weighted average of the properties of the pure components. Also, it has been shown that the nucleation of bubbles is impaired, leading to fewer active nucleation sites.

The increased concentration of the less volatile component in the evaporator and of the most volatile component in the circulating refrigerant will also cause other problems in the system. Because of the low heat transfer and these other problems it is recommended not to use pool boiling or re-circulation-type evaporators with zeotropic mixtures.

In dry-expansion evaporators zeotropic mixtures are more easy to handle, and may even have some benefits. Because of the glide, the refrigerant temperature will increase along the
evaporator with increasing vapor fraction. If this increase is matched with the temperature
decrease of the heat source fluid (in counter-current flow) it can be shown that the
thermodynamic efficiency (COP) can be higher for a mixture than for a pure fluid.
In Figure 8.37 the temperature profiles of a mixture and of a pure fluid are shown. The mass
flow of the heat source fluid is assumed to be the same, and as the temperature change of this
fluid is also the same in both cases, the cooling load $Q$ is equal. Assuming also that the overall
heat transfer coefficients are equal for the mixture and the pure fluid, and remembering that

$$Q = U \cdot A \cdot \theta_m \quad 8.37$$

we find that the logarithmic mean temperature differences must also be equal. However, from the figure it is clear that the evaporation temperature will be higher with the mixture than with the pure fluid. As the COP of a refrigeration cycle is approximately inversely proportional to the temperature difference between the evaporator and the condenser, it is clear that the mixture should be expected to give higher COPs under the assumptions stated above. Note that this gain can only be achieved with counter-current flow!

8.38 In practice, the overall heat transfer coefficient cannot be expected to be the same for the mixture and the pure fluid as the boiling heat transfer coefficient will be lower for the mixture. The reasons for the decrease in the heat transfer coefficient are basically the same as in pool boiling: Concentration of the least volatile component at the heated surface, less favorable thermodynamic properties and impeded bubble nucleation. The physical process is considerably more complex in flow boiling, however, for at least two reasons: First, there are concentration gradients not only perpendicular to the heated surface, but also in the axial direction of the tube and along the circumference. Second, as the vapor travels faster than the
liquid inside the tube, the local conditions at any point will be in neither thermal (temperature) nor thermodynamic (concentration) equilibrium. Several correlations have been suggested in the literature for calculating flow boiling heat transfer coefficients for refrigerant mixtures. Most of these are based on correlations for pure fluids. The correlations are often cumbersome to use and the accuracy is still in question. It is considered outside the scope of this text to suggest a correlation for flow boiling of mixtures. The interested reader is referred to Rohlin (1996), Melin (1996) or Steiner (1993).

8.39 A different problem which may hinder the possible benefits of glide-refrigerants is the necessity of superheat and the setting of the expansion valve. As shown in Figure 8.37 the temperature difference at the evaporator exit is considerably smaller with the mixture. For this reason, a larger part of the evaporator must be used for superheating. To ensure stable operation of the expansion valve a superheat of 5-7 K is normally needed at the evaporator exit. With refrigerant- and heat source temperatures being parallel, this may be difficult to achieve, as the desirable mean temperature difference is usually of about this magnitude.

8.40 The pressure drop of evaporative two-phase flow inside a straight tube is the sum of three components: The wall friction pressure drop, the static pressure difference and the momentum pressure drop.

$$\Delta p_{\text{tot}} = \Delta p_f + \Delta p_s + \Delta p_m$$

8.40

If the evaporator has bends, additional pressure drop is caused by these. The wall friction pressure drop, $\Delta p_f$, is due to the shear forces between the flowing refrigerant and the tube walls, and to some extent between liquid and vapor. The static pressure difference, $\Delta p_s$, is caused by the gravitational force acting on the vapor/liquid column in the tube. The momentum pressure drop, $\Delta p_m$, is the effect of the increase in velocity (or kinetic energy) of the vapor/liquid mixture inside the tube.

8.41 The static pressure drop of a tube with height $H$ is calculated as

$$\Delta p_s = \int_0^H \rho_m dz$$

8.41a

The difficulty in this equation is to determine $\rho_m$, the density of the liquid/vapor mixture. If the homogeneous model, in which the velocities of the two phases are considered to be the same, is applied, $\rho_m$ may be calculated from:

$$1/\rho_m = v_m = (1-x) \cdot v_l + x \cdot v_v$$

8.41b

where $v_l =$ specific volume of liquid

$v_v =$ specific volume of vapor

$x =$ vapor fraction
The integration of equation 8.41b may be avoided by dividing the evaporator into sections and assuming the density to be constant in each section. The vapor fraction can, for an estimate, be assumed to increase linearly along the evaporator and the density can thus be calculated for each section.
Alternatively, the mean density of the mixture can be calculated as proposed by Pierre (see Section 8.44).

8.42 The momentum pressure drop is a result of a velocity increase of the refrigerant. This increase may be due to a decrease in the tube diameter or to an increase in the specific volume. The specific volume will increase as the refrigerant evaporates due to either heat transfer or pressure drop.

The momentum pressure drop can be written in terms of the difference in kinetic energy of the mixtures entering and leaving the evaporator:

\[ \Delta p_m = G^2 \cdot (v_{m2} - v_{m1})/2 \quad 8.42a \]

Where \( G \) is the mass flux in the tube (kg/(m\(^2\) s)) and \( v_{m2} \) and \( v_{m1} \) are the specific volumes of the mixtures leaving and entering the evaporator respectively.

If the liquid and the vapor are considered to be traveling at the same velocity (a condition of no slip), \( v_{m1} \) and \( v_{m2} \) may be calculated by eq 8.41b. In this case, the momentum pressure drop may also be written as

\[ \Delta p_m = (p_{m2} \cdot w_2^2)/2 - (p_{m1} \cdot w_1^2)/2 \quad 8.42b \]

where \( w_1 \) and \( w_2 \) are the velocities of the refrigerant entering and leaving the evaporator respectively. This clearly shows the relation to kinetic energy.

Note that a quick estimate of the upper bounds of the momentum pressure drop of a dry expansion evaporator is found by using the vapor and liquid specific volumes in equation 8.42b.

8.43 Most correlations for frictional pressure drop belong to either of two groups: The homogeneous models, where liquid and vapor are assumed to travel at the same velocity as a homogeneous mixture, and the separated flow models, where the phases are considered as separated flows, which may be flowing at different velocities.

In both type of models, the pressure drop of the two phase flow \( \Delta p_{tp} \) may be related to the pressure drop of one phase flow \( \Delta p_l \) (as calculated by standard correlations) by a two phase multiplier, \( \phi^2 \), defined by:

\[ \Delta p_{tp} / \Delta p_l = \phi^2 \quad 8.43a \]

where \( \phi^2 \) is a function of flow- and heat transfer parameters and of fluid properties. Only homogeneous models will be treated here. It should be noted that the accuracy in the prediction of any of the existing models is quite low.

In the homogeneous models, average properties are assigned to the liquid/vapor flow and existing single phase correlations are used to evaluate the pressure drop. The frictional pressure drop may then be written as

\[ \Delta p_f = \int_0^l f \cdot p_m \cdot w^2 \cdot \frac{1}{d} \, dz \quad 8.43b \]
where $\rho_m$ is calculated from equation 8.41b, and

\[
\nu = G / \rho_m, \text{ flow velocity}
\]

\[
d = \text{diameter of the tube}
\]

\[
f = \text{friction factor}
\]

The simplest, but less accurate method of determining $f$ is to use the one-phase friction factor for turbulent flow from the *Blasius equation*

\[
f = 0.158 \text{ Re}^{-1/4} \tag{8.43c}
\]

or from *Moody's diagram*.

The Reynolds number in the above equations is determined as

\[
\text{Re} = \frac{G \cdot d}{\bar{\mu}} = \frac{4 \cdot \bar{m}}{\pi \cdot d \cdot \bar{\mu}} \tag{8.43d}
\]

where $\bar{m}$ is the mass flow and $\bar{\mu}$ is a mean viscosity which may be calculated as (McAdams, 1942)

\[
\frac{1}{\bar{\mu}} = \frac{x}{\mu_r} + \frac{1-x}{\mu_l} \tag{8.43e}
\]

Instead of integrating equation 8.43b it is suggested (Griffith, 1973) that the evaporator is divided into a number of sections and the vapor quality assumed to be constant within each section.

**8.44 Special friction factors for two phase flow** have also been suggested. Pierre (1957) investigated the evaporation of R12 and R22 in horizontal tubes and concluded that the mean friction factor for oil free refrigerant can be expressed as (c.f. one-phase flow)

\[
\Delta p_f = f_m \cdot G^2 \cdot \nu_m \cdot L/d \tag{8.44a}
\]

if the friction factor is determined as

\[
f_m = 0.0185 \cdot K_f^{1/4} \cdot \text{Re}^{-1/4} \tag{8.44b}
\]

where $K_f$ is the Pierre boiling number, $K_f = h (L \cdot g)$, and the Reynolds number is calculated using the liquid viscosity.

In the presence of oil, (6-12%) a considerably higher friction factor was found

\[
f_m = 0.053 \cdot K_f^{1/4} \cdot \text{Re}^{-1/4} \tag{8.44c}
\]

In the calculation of the pressure drop, Pierre determined the mean density $\rho_m = 1/\nu_m$ from
\[ v_m = x_m \cdot v_{m''} \]

8.44d

where \( x_m \) is the mean vapor fraction, calculated as

\[ x_m = 4.4 \cdot d^{1/4} \cdot L^{-1/2} \]

8.44e

and \( v_{m''} \) is the specific volume of the vapor at average evaporation temperature. The correlation for \( x_m \) was based on experimental results.

Note that no integration of equation 8.43b is necessary with this method.

### Methods of enhancing boiling heat transfer

8.45 Heat transfer in pool boiling and in nucleate boiling in general may be enhanced, without increasing the surface area, by facilitating the nucleation of bubbles. The most commonly used method of achieving this is by giving the surface a rough or porous texture.

One of the first studies of the influence of surface finish on heat transfer was done by Jakob and Fritz (1931). They boiled water at atmospheric pressure on several specially prepared horizontal copper plates. Of these, one was sandblasted and one had a square grid of finely machined grooves. They found that sandblasting the surface increased the heat transfer coefficient by 25% at a given heat flux. The grooved surface increased the heat transfer coefficient even more, initially about 300%. Both increases diminished with time and the heat transfer coefficients tended towards those of a smooth surface.

The findings during the 1950’s that reentrant cavities (Figure 8.45a) functioned as stable nucleation sites stimulated the development of new evaporator surfaces. It was not until the 1970’s, however, that such surfaces became commercially available. A number of methods for manufacturing such enhanced surfaces, have been tried. Two groups of surfaces may be distinguished depending on the manufacturing method:

In the first group are found surfaces made by applying material to a smooth surface. They are usually made by flame spraying or sintering of metal onto the substrate to produce a porous layer.

The second group of surfaces are made by machining the surface of a smooth heat transfer wall (usually a low-finned tube) to form the desired reentrant cavity surface. Examples of geometries of commercial enhanced tubes are shown in Figure 8.45b.

The porous surfaces’ enhancement of heat transfer can be attributed to the following:

![Figure 8.45a. Re-entrant cavity. (Griffith and Wallis, 1960)](image-url)
• During boiling in a porous structure, some vapor will be left when a bubble departs from the surface. This vapor will act as a "mother-bubble" for new bubbles.
• In the porous structure the wetted surface area is very large.
• The boiling mechanism is evaporation of very thin liquid films, which can be expected to result in high heat transfer coefficients.

The heat transfer coefficients have been shown to increase by 100 to 1000% with these surfaces, compared to a smooth tube.
Figure 8.45b.: Commercially available enhanced surfaces for pool boiling.
8.46 In flow boiling inside tubes only one of the above surfaces (High Flux) has been used. The prime reason is probably that they are difficult to manufacture on the inside of a tube. It has been demonstrated, however, that porous surfaces do have a considerable effect also on flow boiling.

A more common way of enhancing heat transfer in flow boiling is to use tubes with micro-fins. Such tubes are manufactured by several companies, but the geometries are quite similar. The fin height is usually only about 0.2 mm, the fins have a helix angle of 18-30° and the distance between fins is 1 mm or less (Figure 8.46). The heat transfer coefficients of these tubes are 50-200% higher than those of plain tubes. In spite of this, the pressure drop is reported to increase by only 10-40%.

Several explanations have been proposed for the heat transfer enhancement of micro-fins:
- The fins cause a disturbance of the boundary layer close to the tube wall.
- The wetting of the tube wall is better in stratified flow.
- The surface area is increased by the fins.
- Bubble nucleation may be enhanced in the sheltered regions in between fins.
- At high vapor quality where the liquid is present only as droplets in the vapor flow, the helical shape of the fins will force a rotating flow, whereby the droplets are thrown to the wall of the tube.

All of these probably have influence on the total effect.

As of today, there are no general correlations for predicting heat transfer coefficients neither for the porous surface tubes for pool boiling nor for the micro-fin tubes. The designer is obliged to resort to the data or computer programs supplied by the manufacturers.

**Heat Transfer at the Heat Source Side of Evaporators**

8.47 To be able to predict the overall heat transfer coefficient of an evaporator we need correlations for the heat source side. In the following, such correlations are given for a number of cases which are frequently encountered in refrigeration evaporators. For a deeper discussion on the theoretical background of the correlations, the reader is referred to textbooks on heat transfer.

The presentation is divided into two main sections depending on the phase of the heat source, gas (air) or liquid.
8.48 The dominating gaseous heat source is, of course, air. For this case and most other gaseous heat sources, heat may be transferred to the surface by three mechanisms, convection, diffusion of water vapor and radiation. The total heat flux transferred may then be written as a sum of three contributions

\[ \dot{q}_t = \dot{q}_c + \dot{q}_d + \dot{q}_r \]

where 

\[ \dot{q}_c = \alpha_c \cdot \theta_e, \]
\[ \dot{q}_d = \alpha_d \cdot \theta_r, \]
\[ \dot{q}_r = \alpha_r \cdot \theta_r, \]

and \( \alpha_c, \alpha_d \) and \( \alpha_r \) are the heat transfer coefficients due to convection, diffusion and radiation, respectively. Note that the temperature difference associated with diffusion is the same as that for convection, i.e. the temperature difference between the evaporator surface and the air, \( \theta_r \), however, is the difference between the evaporator surface and the surrounding wall surfaces. In most cases, the walls surrounding the evaporator will have the same temperature as the air and the heat transfer coefficients may then be summed as

\[ \alpha_{tot} = \alpha_c + \alpha_d + \alpha_r \]

8.49 The radiation heat transfer coefficient \( \alpha_r \) between a "black" surface which is totally surrounded by a larger isothermal surface is shown in the diagram in Figure 8.49. As seen, a value of about 4 W/(m\(^2\) K), is applicable for temperatures typical for an evaporator and its surrounding.

For real (non-black) surfaces the values should be multiplied by the emissivity of the evaporator surface. Most surfaces except polished metal have emissivities in the range 0.8 -
0.95. Note that frost also has emissivity in this range and thus is almost "black" to thermal radiation.

More important, however, is the fact that in many evaporator designs most of the surface area does not exchange radiation with the surrounding walls but with other parts of the evaporator itself. This is the case for most finned surfaces. In such cases it is appropriate to use the front area of the finned coil (or the circumscribed area in case of free convection coils) as the heat exchange area and to consider this surface to be "black" as radiation will be trapped in between the fins. As the active surface area is considerably smaller than the total area, and as the radiation heat transfer coefficient is fairly low, radiation heat exchange may often be neglected in the case of finned coils.

8.50 The diffusion heat transfer coefficient can be shown to be related to the convection heat transfer coefficient. According to Bäckström (see Ekroth/Granryd, 1994) this relation can be expressed as

$$\frac{\alpha_d}{\alpha_c} = C \cdot \frac{p_s - p_a}{t_s - t_a}$$  \hspace{1cm} 8.50

where:

- \( C = 1750 \) for frost covered surfaces, 1530 for wet surfaces
- \( t_s \) = surface temperature of evaporator (°C)
- \( t_a \) = air temperature of free air (°C)
- \( p_s \) = saturation pressure of water vapor at surface temperature \( t_s \) (bar)
- \( p_a \) = partial pressure of water vapor in free air (bar)

The equation 8.50 must be applied to the local conditions at the cold surface. In a forced convection air cooler, for example, the partial pressure and the temperature of the air will decrease as it passes through the coil. Most of the frost will therefore be deposited towards the inlet side of the evaporator.

At conditions usually found in freezing rooms the ratio \( \alpha_d/\alpha_c \) is about 0.10 while in cold storage rooms above 0°C the value may be 0.4-0.5

A detailed analysis of the heat transfer, including vapor deposition, of finned coils require the use of computer programs with which the local conditions are calculated.

8.51 The convective heat transfer coefficient \( \alpha_c \) is strongly dependent on the velocity of the air and on the geometry of the heat exchange surface. The air can be propelled either by forced convection, in which case the velocity may be assumed to be known, or by free convection. In this case the air is flowing as a result of the change in density caused by the heat transfer.

As the heat transfer coefficient on the air side is always considerably lower than on the refrigerant side, it is common practice to extend the air side surface by fins. This is the case both in forced and free convection. However, in some older plants plain tubes are still in use.

8.52 In forced convection across a tube, the heat transfer coefficient will be different on the front and the backside. An average Nusselt number for the circumference can be estimated from the following empirical relationship (Holman, 1992):
\[ \text{Nu}_f = C \cdot \text{Re}_f^n \cdot \text{Pr}_f^{0.33} \quad 8.52a \]

where
\[ \text{Nu}_f = \alpha \cdot \frac{d}{\lambda}, \]
\[ \text{Re}_f = \frac{w}{d} \cdot \text{N} \]

The constant \( C \) and the exponent \( n \) are evaluated according to Table 8.52a. The Reynolds number is calculated using the tube diameter as the characteristic length. Fluid properties should be evaluated at the film temperature (average of surface and free air temperature). For computer calculations the following correlations for single tubes may be easier to apply

\[ \text{Nu} = (0.43 + 0.50 \cdot \text{Re}^{0.5} \cdot \text{Pr}^{0.38} \cdot \left(\frac{\text{Pr}_f}{\text{Pr}_w}\right)^{0.25}) \quad \text{for } 1 < \text{Re} < 10^3 \]
\[ \text{Nu} = 0.25 \cdot \text{Re}^{0.6} \cdot \text{Pr}^{0.38} \cdot \left(\frac{\text{Pr}_f}{\text{Pr}_w}\right)^{0.25} \quad \text{for } 10^3 < \text{Re} < 2 \cdot 10^5 \quad 8.52b, c \]

The indexes on the Prandtl numbers indicate that the values refer to the film- and wall temperatures respectively.

In Figure 8.52a the equations 8.52a and b/c are plotted assuming a Prandtl number of 0.7, equal at film- and wall temperature.

The equation 8.52c may be rewritten to give the heat transfer coefficient as a function of the flow velocity, the film temperature and the tube diameter. For air of normal pressure and temperatures in the range -40 - +20 °C the temperature dependence can be omitted, and we get

\[ \alpha = 4.5 \cdot \frac{w^{0.6}}{d^{0.4}} \quad \text{for } 10^5 < w \cdot d < 0.01 \quad 8.52d \]

This equation is plotted for some different tube diameters in its range of validity in Figure 8.52b.

8.53 Equation 8.52a can also be used for calculating average Nusselt numbers for staggered and inline tube banks. In this case, the constant \( C \) and the exponent \( n \) are dependent on the ratios of the c/c tube spacings normal (\( S_n \)) and parallel (\( S_p \)) to flow, and to the tube diameter (see table 8.53a). The Reynolds number should be calculated using the maximum velocity occurring in the tube bank (i.e. in the narrowest section).

The heat transfer coefficients calculated with the values of \( C \) and \( n \) in Table 8.53a are valid for banks with a depth of 10 tube rows or more in the direction of air flow. For thinner banks, the values should be multiplied with the ratios in Table 8.53b.
Figure 8.52a.: Comparison of equations 8.52a and b/c, assuming Pr_f = Pr_w = 0.7.

Figure 8.52b. Heat transfer coefficient of single tube in air at normal pressure.
Table 8.53a.: Constants and exponents in equation 8.52a for heat transfer in tube bundles. (From Grimison, 1937, according to Holman, 1992)

<table>
<thead>
<tr>
<th>$S_n/d$</th>
<th>1.25</th>
<th>1.5</th>
<th>2.0</th>
<th>3.0</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>$C$</td>
<td>$n$</td>
<td>$C$</td>
<td>$n$</td>
</tr>
<tr>
<td>In line</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>1.25</td>
<td>0.386</td>
<td>0.592</td>
<td>0.305</td>
<td>0.608</td>
</tr>
<tr>
<td>1.5</td>
<td>0.407</td>
<td>0.586</td>
<td>0.278</td>
<td>0.620</td>
</tr>
<tr>
<td>2.0</td>
<td>0.464</td>
<td>0.570</td>
<td>0.332</td>
<td>0.602</td>
</tr>
<tr>
<td>3.0</td>
<td>0.322</td>
<td>0.601</td>
<td>0.396</td>
<td>0.584</td>
</tr>
<tr>
<td>Staggered</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>0.6</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td>0.9</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>0.495</td>
</tr>
<tr>
<td>1.0</td>
<td>-</td>
<td>0.552</td>
<td>0.558</td>
<td>-</td>
</tr>
<tr>
<td>1.125</td>
<td>-</td>
<td>-</td>
<td>0.531</td>
<td>0.565</td>
</tr>
<tr>
<td>1.25</td>
<td>0.575</td>
<td>0.556</td>
<td>0.561</td>
<td>0.554</td>
</tr>
<tr>
<td>1.5</td>
<td>0.501</td>
<td>0.568</td>
<td>0.511</td>
<td>0.562</td>
</tr>
<tr>
<td>2.0</td>
<td>0.448</td>
<td>0.572</td>
<td>0.462</td>
<td>0.568</td>
</tr>
<tr>
<td>3.0</td>
<td>0.344</td>
<td>0.592</td>
<td>0.395</td>
<td>0.580</td>
</tr>
</tbody>
</table>

Table 8.53b.: Ratios of the heat transfer coefficient for $N$ rows deep to that of 10 rows (Kays and London, 1952, according to Holman 1992)

<table>
<thead>
<tr>
<th>$N$</th>
<th>1</th>
<th>2</th>
<th>3</th>
<th>4</th>
<th>5</th>
<th>6</th>
<th>7</th>
<th>8</th>
<th>9</th>
<th>10</th>
</tr>
</thead>
<tbody>
<tr>
<td>Staggered tubes</td>
<td>0.68</td>
<td>0.75</td>
<td>0.83</td>
<td>0.89</td>
<td>0.92</td>
<td>0.95</td>
<td>0.97</td>
<td>0.98</td>
<td>0.99</td>
<td>1.0</td>
</tr>
<tr>
<td>In-line tubes</td>
<td>0.64</td>
<td>0.80</td>
<td>0.87</td>
<td>0.90</td>
<td>0.92</td>
<td>0.94</td>
<td>0.96</td>
<td>0.98</td>
<td>0.99</td>
<td>1.0</td>
</tr>
</tbody>
</table>

Figure 8.53.: Definition of $S_n$ and $S_p$ in tables 8.53a and b.
Heat transfer is generally better in staggered than in in-line tube banks. However, the pressure drop is also higher. According to Pierre (1979) a staggered arrangement with $S_n/d = 1.3$ and $S_p/d = 2.7$ is recommended.

8.54 The heat transfer coefficients of finned surface is always lower than for plain tubes at the same air velocities.

Two simple relations for forced convection air-coolers (batteries) with fin spacings 4 and 8 mm, applicable for quick estimates at typical conditions, are suggested by Granryd (1965):

$$\alpha_f = 17 \cdot (w_{fr} \cdot \rho)^{0.75} \quad \text{W/(m}^2\text{K)} \quad \text{fin spacing 8 mm} \quad 8.54a$$

$$\alpha_f = 23.5 \cdot (w_{fr} \cdot \rho)^{0.5} \quad \text{W/(m}^2\text{K)} \quad \text{fin spacing 4 mm} \quad 8.54b$$

where $w_{fr}$ is the front air velocity and $\rho$ is the density of the air. Both equations are valid for front velocities in the range 2-4 m/s and for coil depths of 100 mm. For the 8 mm spacing, the correlation applies for coil depths up to 300 mm. In the case of 4 mm spacing, the heat transfer coefficients are reduced by 10% for 200 mm depths and by 14% for 300 mm depths.

8.55 For forced convection air coolers with smooth fins, Granryd (1965) also developed a more elaborate method for predicting the heat transfer and pressure drop under dry conditions. It has been found reliable in a large number of cases.

The Granryd method was based on a series of experiments with electrically heated copper plates simulating the fins, and cylindrical or oval pieces of brass simulating the tubes. Table 8.55a shows the range of the individual geometric parameters used in the experiments. Both staggered and in-line tube configurations were simulated.
The surface heat transfer coefficient was defined by

\[ \alpha_s = \frac{\dot{Q}}{(A_t \cdot \delta_m)} \]  \hspace{1cm} 8.55a

where

\[ \dot{Q} = \text{heat load (W)} \]

\[ A_t = \text{total surface area of fins and tubes (m}^2\text{)} \]

\[ \delta_m = \text{measured difference between the plates' mean temperature and the mean temperature of the air.} \]

In the first series of experiments, only the plates simulating the fins were used. From the results, equations describing the heat transfer coefficients in narrow slots (\(\alpha_s\)) were derived (see table

Table 8.55a Range of individual parameters of geometries tested by Granryd (1965)

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Range</th>
</tr>
</thead>
<tbody>
<tr>
<td>Fin spacing</td>
<td>(s = 3, 5, 8, 16 \text{ mm})</td>
</tr>
<tr>
<td>Fin thickness</td>
<td>(\delta = 1.2 \text{ mm})</td>
</tr>
<tr>
<td>Length of individual fin in direction of air flow</td>
<td>(L = 100, 200, 300 \text{ mm})</td>
</tr>
<tr>
<td>Number of fins in direction of air flow</td>
<td>(z = 1, 2, 3)</td>
</tr>
<tr>
<td>Tube diameter</td>
<td>(d = 0, 10, 15, 20, 35 \text{ mm})</td>
</tr>
<tr>
<td>Tube spacing transverse to air flow</td>
<td>(S_T = 100, 50, 33.3 \text{ mm})</td>
</tr>
<tr>
<td>Tube spacing in direction of air flow</td>
<td>(S_L = S_T)</td>
</tr>
<tr>
<td>Number of tubes in individual fin in direction of air flow</td>
<td>(z_r = 1, 2, 3)</td>
</tr>
<tr>
<td>Tube arrangement</td>
<td>staggered and in-line</td>
</tr>
</tbody>
</table>

In the first series of experiments, only the plates simulating the fins were used. From the results, equations describing the heat transfer coefficients in narrow slots (\(\alpha_s\)) were derived (see table

Table 8.55b.: Nusselt number for slot without tubes (valid for \(3 \leq L / d_b \leq 20\))

<table>
<thead>
<tr>
<th>Re range</th>
<th>Nusselt number formula</th>
<th>Equation</th>
</tr>
</thead>
<tbody>
<tr>
<td>(500 &lt; Re &lt; 2500):</td>
<td>(Nu = 2.09 \cdot (Re \cdot d_b / L)^{0.35})</td>
<td>Eq. 8.55b</td>
</tr>
<tr>
<td>(2500 &lt; Re &lt; 7000):</td>
<td>(Nu = 0.407 \cdot Re^{0.55} \cdot (d_b / L)^{0.3})</td>
<td>Eq. 8.55c</td>
</tr>
<tr>
<td>(7000 &lt; Re &lt; 20000):</td>
<td>(Nu = 0.0358 Re^{0.8} (d_b / L)^{0.2})</td>
<td>Eq. 8.55d</td>
</tr>
</tbody>
</table>

Figure 8.55.: Heat transfer coefficients and pressure drop according to Granryd (1965) for coil depth 200 mm, tube pitch 50x50 mm, tube diameter 15 mm.
In a second series, the brass cylinders were used as well. In this case, the heat transfer to the cylinders, $\alpha_c$, was calculated as for a bundle of smooth tubes by the method of Grimison (1937) described above (Equation 8.52a and Table 8.53a). The influence of the tubes and of the fin spacing on the heat transfer to the fins was taken into account by introducing a factor $C_h$ to be multiplied by the coefficient of heat transfer of a narrow slot without tubes. The dependence of the coil depth was simulated by the introduction of a second factor $k_z$.

This leads to the following expression for the mean heat transfer coefficient of the total area

$$\alpha_t = k_z \cdot (\alpha_c \cdot A_c / A_t + C_h \cdot \alpha_s \cdot (1 - A_c / A_t))$$  \hspace{1cm} 8.55e

In air batteries where the fin surface is considerably larger than the tube surface, $(A_c / A_t < 0.1)$ this equation may be simplified, without noticeable error, to

$$\alpha_t = k_z \cdot C_h \cdot \alpha_s$$  \hspace{1cm} 8.55f

When calculating an overall heat transfer coefficient $U$, the fin efficiency should be taken into account as usual according to the equation

$$
\frac{1}{U \cdot A_t} = \frac{1}{\alpha \cdot A_t} + \frac{\delta}{\lambda \cdot A_{tot}} + \left[ \alpha_t \cdot \left( \frac{1}{A_t} + \xi_A \cdot A_f \right) \right]^{-1}
$$  \hspace{1cm} 8.55g

In equations 8.55e-g:

- $k_z$ = factor depending on the number of fin rows $(z)$ in the direction of air flow and of the Reynolds no. $k_z$ is taken from Table 8.55c.
- $C_h = 1.05 + k_{Re} \cdot (k_A \cdot k_{zr} - 1.05)$ for $A_c / A_t \geq 0.01$. \hspace{1cm} 8.55h
  For $A_c / A_t = 0$, $C_h = 1$  

- $k_{Re}$ = factor depending on the Reynolds number, see Table 8.55d.
- $w_s$ = air velocity in between plates calculated without considering the tubes. \hspace{1cm} (m/s)
- $d_h$ = hydraulic diameter, calculated without considering the tubes. For a slot with the cross section $s \cdot b$, $d_h = 2 \cdot s \cdot b / (s+b)$ \hspace{1cm} (m)
- $k_A$ = factor depending on the ratio between the surface area of the tubes and the total area $(A_c/A_t)$ according to Table 8.55e.
- $k_{zr}$ = factor depending on the number of tube rows $(z_r)$ in the direction of air flow on a single fin. See Table 8.55f.
- $\alpha_s$ = surface heat transfer coefficient for the slot without tubes, calculated according to equations in Table 8.55b. \hspace{1cm} (W/(m$^2$ °C))
- $\alpha_c$ = surface heat transfer coefficient in tube bundles calculated according to Grimison's data (Equation 8.52a and Table 8.53a). \hspace{1cm} (W/(m$^2$ °C))
- $A_c$ = free outer surface area of the tubes. \hspace{1cm} (m$^2$)
- $A_f$ = surface area of the fins. \hspace{1cm} (m$^2$)
- $A_t$ = $A_c + A_f$ = total outer surface area \hspace{1cm} (m$^2$)

<table>
<thead>
<tr>
<th>Table 8.55c. The factor $k_z$</th>
<th>When $z = 1$, $k_z = 1$</th>
</tr>
</thead>
<tbody>
<tr>
<td>$Re$</td>
<td>$\leq 1000$</td>
</tr>
</tbody>
</table>

8:48
### CHAPTER 8. EVAPORATORS

$k_z$ for in-line tube arrangement:

<table>
<thead>
<tr>
<th>$z_r$ = 1; $z$ = 2</th>
<th>0.89</th>
<th>0.89</th>
<th>0.90</th>
<th>0.95</th>
<th>1.00</th>
<th>1.05</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>0.86</td>
<td>0.86</td>
<td>0.87</td>
<td>0.94</td>
<td>1.01</td>
<td>1.06</td>
</tr>
<tr>
<td></td>
<td>0.83</td>
<td>0.83</td>
<td>0.84</td>
<td>0.93</td>
<td>1.01</td>
<td>1.08</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>$z_r$ = 2; $z$ = 2</th>
<th>0.89</th>
<th>0.90</th>
<th>0.94</th>
<th>1.01</th>
<th>1.05</th>
<th>1.05</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>0.86</td>
<td>0.87</td>
<td>0.92</td>
<td>1.01</td>
<td>1.06</td>
<td>1.06</td>
</tr>
<tr>
<td></td>
<td>0.89</td>
<td>0.95</td>
<td>1.01</td>
<td>1.05</td>
<td>1.05</td>
<td>1.05</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>$z_r$ = 3; $z$ = 2</th>
<th>0.89</th>
<th>0.95</th>
<th>1.01</th>
<th>1.05</th>
<th>1.05</th>
<th>1.05</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>0.86</td>
<td>0.94</td>
<td>1.01</td>
<td>1.06</td>
<td>1.06</td>
<td>1.06</td>
</tr>
</tbody>
</table>

$k_z$ for staggered tube arrangement:

<table>
<thead>
<tr>
<th>$z_r$ = 1; $z$ = 2</th>
<th>0.93</th>
<th>0.94</th>
<th>0.97</th>
<th>1.01</th>
<th>1.05</th>
<th>1.05</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>0.90</td>
<td>0.91</td>
<td>0.95</td>
<td>1.01</td>
<td>1.06</td>
<td>1.06</td>
</tr>
<tr>
<td></td>
<td>0.88</td>
<td>0.89</td>
<td>0.93</td>
<td>1.01</td>
<td>1.08</td>
<td>1.08</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>$z_r$ = 2; $z$ = 2</th>
<th>0.93</th>
<th>0.98</th>
<th>1.02</th>
<th>1.05</th>
<th>1.05</th>
<th>1.05</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>0.90</td>
<td>0.97</td>
<td>1.02</td>
<td>1.06</td>
<td>1.06</td>
<td>1.06</td>
</tr>
</tbody>
</table>
8.56 Granryd also developed a method for predicting the pressure drop in forced convection finned coils. The pressure drop for narrow slots without tubes was first measured, and the derived equations were combined with equations for the pressure drop in tube bundles. The influence of the tubes on the pressure drop of the fins was considered by the introduction of additional parameters. This treatment resulted in the following equation for the pressure drop:

$$\Delta p = z \cdot p_z \cdot \frac{\left(\rho \cdot w_s^2\right)}{2} \left[\zeta_{t,2} + z_r \cdot \zeta_r + \frac{C_f \cdot f_s \cdot L}{2 \cdot s}\right]$$

where:
- $p_z$ = factor depending on the number of fin rows in the direction of air flow. Taken from Table 8.56a.
- $\zeta_{t,2}$ = represents the sum of the pressure drops as the air enters and leaves the finned volume. For $\delta / (\delta + s) \leq 0.2$ this factor is calculated as $\zeta_{t,2} = \delta / (\delta + s)$.
- $\zeta_r$ = $\zeta_{10} \cdot \left[S_i / (S_i - d)\right]^2$ where $\zeta_{10}$ is taken from Figure 8.56a. This factor depends on the number of tube rows in the direction of air flow in a tube bundle.
- $f_s$ = friction factor for the slot between two fins without tubes. Calculated according to equations in Table 8.56b.
- $C_f$ = $\phi \cdot (Re / 10000)^n \cdot 8.4 \cdot d / S_i + 0.05 \cdot [S_i / d - 3]$ (if 700 < Re < 10000 and 2.5 < S_i / d < 4.5) where $\phi = 1$ for in-line tube configurations and $= 1.05$ for staggered configurations.
- $n = 0.05$ for $z_r = 1$
- $= 0.17$ for $z_r = 2$

**Table 8.55d. The factor $k_{Re}$**

<table>
<thead>
<tr>
<th>In-line tube arrangement:</th>
<th>$k_{Re}$</th>
<th>$k_{Re}$</th>
<th>$k_{Re}$</th>
</tr>
</thead>
<tbody>
<tr>
<td>$Re \leq 1000$</td>
<td>0</td>
<td></td>
<td></td>
</tr>
<tr>
<td>$1000 \leq Re \leq 6000$</td>
<td></td>
<td>$(Re / 1000)^{0.39} - 1$</td>
<td></td>
</tr>
<tr>
<td>$6000 \leq Re$</td>
<td></td>
<td>1</td>
<td></td>
</tr>
<tr>
<td>Staggered tube arrangement:</td>
<td>0</td>
<td></td>
<td></td>
</tr>
<tr>
<td>$Re \leq 500$</td>
<td></td>
<td>1</td>
<td></td>
</tr>
<tr>
<td>$500 \leq Re \leq 5000$</td>
<td></td>
<td></td>
<td>$(Re / 500)^{0.3} - 1$</td>
</tr>
<tr>
<td>$5000 \leq Re$</td>
<td></td>
<td>1</td>
<td></td>
</tr>
</tbody>
</table>

**Table 8.55e. The factor $k_A$**

| In-line tube arrangement: | $k_A = 1.92 \cdot (A_c / A_t)^{0.13}$ |
| Staggered tube arrangement: | $k_A = 2.39 \cdot (A_c / A_t)^{0.19}$ |

**Table 8.55f. The factor $k_{zr}$**

<table>
<thead>
<tr>
<th>$z_r$</th>
<th>1</th>
<th>2</th>
<th>3</th>
<th>5</th>
</tr>
</thead>
<tbody>
<tr>
<td>$k_{zr}$</td>
<td>1</td>
<td>1.19</td>
<td>1.30</td>
<td>(1.38)</td>
</tr>
</tbody>
</table>
= 0.25 for \( z_r = 3 \)

and

\[
d = \text{tube diameter} \\
L = \text{length of individual fin in direction of air flow} \quad \text{mm} \\
w_s = \text{air velocity in between plates calculated without considering the tubes} \quad \text{m/s} \\
Re = \frac{w_s \cdot d_i}{v} \\
s = \text{fin spacing} \quad \text{mm} \\
S_t = \text{Tube spacing perpendicular to flow} \quad \text{m} \\
z = \text{number of tube rows in flow direction} \\
z_r = \text{number of tubes in individual fin in direction of air flow} \quad \text{mm} \\
\delta = \text{fin thickness} \\
\rho = \text{air density} \quad \text{kg/m}^3
\]

**Table 8.56a**

The factor \( p_z \)

When \( z = 1, p_z = 1 \)

<table>
<thead>
<tr>
<th>Re ( \leq 1000 )</th>
<th>2000</th>
<th>3000</th>
<th>5000</th>
<th>( \geq 8000 )</th>
</tr>
</thead>
<tbody>
<tr>
<td>( p_z ) for in-line tube configurations ( z_r = 1; z = 2 )</td>
<td>0.78</td>
<td>0.81</td>
<td>0.85</td>
<td>0.91</td>
</tr>
<tr>
<td>3</td>
<td>0.72</td>
<td>0.76</td>
<td>0.81</td>
<td>0.89</td>
</tr>
<tr>
<td>(5)</td>
<td>0.67</td>
<td>0.71</td>
<td>0.77</td>
<td>0.86</td>
</tr>
<tr>
<td>( z_r = 2; 3 )</td>
<td>( z = 2 )</td>
<td>0.85</td>
<td>0.86</td>
<td>0.89</td>
</tr>
<tr>
<td>3</td>
<td>0.81</td>
<td>0.83</td>
<td>0.85</td>
<td>0.89</td>
</tr>
<tr>
<td>( p_z ) for staggered tube configurations ( z_r = 1; 2 )</td>
<td>( z = 2 )</td>
<td>0.88</td>
<td>0.89</td>
<td>0.91</td>
</tr>
<tr>
<td>3</td>
<td>0.85</td>
<td>0.86</td>
<td>0.88</td>
<td>0.90</td>
</tr>
<tr>
<td>(5)</td>
<td>0.82</td>
<td>0.83</td>
<td>0.85</td>
<td>0.87</td>
</tr>
</tbody>
</table>

**Table 8.56b**

Equations for the friction factor in long, narrow slots for which \( 3 \leq \frac{L}{d_h} \leq 20 \)

| \( 500 \leq Re < 2000 \) | \( f_{sl} = 22.5 \left( \frac{L}{d_h} \right)^{0.3} \cdot Re^{0.65} \) | eq. 8.56b |
| \( 2000 < Re < 7000 \) : | \( f_{sl} = 3.35 \left( \frac{L}{d_h} \right)^{0.3} \cdot Re^{0.4} \) | eq. 8.56c |
| \( 7000 < Re \leq 20000 \) : | \( f_{sl} = 0.577 \left( \frac{L}{d_h} \right)^{0.3} \cdot Re^{0.2} \) | eq. 8.56d |
8.57 In some modern air coolers the fin surface has been modified so as to increase the heat transfer coefficient. This can be achieved by wavy fins or by so called louvered fins on which the fins are split and bent to create a geometry which decrease the thickness of the boundary layer (Figure 8.57). Such surfaces may increase the heat transfer coefficient by up to 80% compared to a plain fin.

The louvered fins are usually not used for frosting conditions as the frost will block the narrow passages through the fin.

Wavy fins are often used just as much to increase the stability of the fin as to increase the heat transfer.

8.58 In *free convection* the driving force for the flow is the buoyancy forces induced by density differences between the free air and the cooler air close to the evaporator surface. The flow in free convection is characterized by the Grashof number. As already noted in the section on convective pool boiling, the heat transfer coefficient in free convection may be determined by the equation

\[
Nu = C \cdot (Gr \cdot Pr)^n
\]

Table 8.26, adapted from Holman, 1992

<table>
<thead>
<tr>
<th>Geometry</th>
<th>GrPr</th>
<th>C</th>
<th>n</th>
</tr>
</thead>
<tbody>
<tr>
<td>Horizontal cylinders</td>
<td>(10^4 - 10^9) (laminar)</td>
<td>0.53</td>
<td>0.25</td>
</tr>
<tr>
<td></td>
<td>(10^9 - 10^{12}) (turbulent)</td>
<td>0.13</td>
<td>0.33</td>
</tr>
<tr>
<td>Vertical plates and cylinders</td>
<td>(10^4 - 10^9) (laminar)</td>
<td>0.59</td>
<td>0.25</td>
</tr>
<tr>
<td></td>
<td>(10^9 - 10^{13}) (turbulent)</td>
<td>0.10</td>
<td>0.33</td>
</tr>
</tbody>
</table>
CHAPTER 8. EVAPORATORS

The constant $C$ and the exponent $n$ are given for cylinders and vertical planes in Table 8.26, reprinted here for convenience. The heat transfer coefficient determined by this equation is referred to the temperature difference between the free, undisturbed air and the surface, which is assumed to be isothermal. The air properties should be evaluated at the film temperature. The characteristic length in the Grashof and Nusselt numbers is, for horizontal tubes, the diameter and for vertical plates, the height of the plate.

Note that the factor $Gr \cdot Pr / (\theta \cdot H^3)$ is a factor of fluid properties only and may be tabulated or plotted as a function of temperature. This is done for air in Figure 8.58.

8.59 The equation 8.58 may be rearranged to give the heat transfer coefficients directly. For temperatures in the range -20 - 0 °C the temperature dependence can be neglected and we get

$$\alpha_l = K_l \cdot (\theta/H)^{1/4} \quad 10^4 < (Gr \cdot Pr) < 10^9 \quad \text{Laminar} \quad 8.59a$$

$$\alpha_t = K_t \cdot (\theta/H)^{1/3} \quad 10^9 < (Gr \cdot Pr) < 10^{13} \quad \text{Turbulent} \quad 8.59b$$

where $K_l$ and $K_t$ are taken from Table 8.59. For horizontal tubes, the values from the table should be multiplied by 0.89 for laminar flow and by 0.85 for turbulent flow. The characteristic length is the diameter of the tube.

Table 8.59: Constants $K_l$ and $K_t$ for vertical plates in air

<table>
<thead>
<tr>
<th>$T_{film}$</th>
<th>$(Gr \cdot Pr)/(\theta \cdot H^3)$</th>
<th>$K_l$</th>
<th>$K_t$</th>
</tr>
</thead>
<tbody>
<tr>
<td>-50</td>
<td>$34.8 \cdot 10^3$</td>
<td>1.57</td>
<td>1.88</td>
</tr>
</tbody>
</table>
Note: The values of Table 8.59 are taken from Granryd’s *Collection of Formulas and Tables*. This table was based on a different source than that of Table 8.26. The resulting heat transfer coefficients will therefore differ slightly depending on which table is used.

The calculations often have to be done in an iterative procedure as the temperature difference is usually not known from the start.

The equations 8.58 and 8.59 may be used for the case of single tubes, stacks of closely spaced tubes, or single vertical plates, e.g. roll bond evaporators of domestic refrigerators.

8.60 In *free convection fin coils*, the heat transfer coefficient will decrease if the thermal boundary layers of adjacent fins come in contact, as will happen if the fin spacing is small. This problem was investigated by Elenbaas (1942) who suggested a semi-empirical correlation which is still widely used:

\[
Nu_s = \frac{1}{24} \cdot (Gr_s \cdot Pr \cdot \frac{s}{H}) \cdot \left\{1 - \exp \left[-\frac{35}{(Gr_s \cdot Pr \cdot \frac{s}{H})}\right]\right\}^{3/4}
\]

The Nusselt number and the Grashof number are calculated with the fin distance \(s\) as the characteristic length. The equation gives the mean Nusselt number for a fin of height \(H\) and is valid for symmetrically heated isothermal plates under laminar flow conditions.

As for the previous free convection correlations, properties are determined at film temperature and the heat transfer coefficient is referred to the temperature difference between the fins and the free air (\(\vartheta\) at inlet). Equation 8.60 is plotted in Figure 8.60a.

The implications of this equation is clearly seen if the heat transfer coefficient is plotted vs. \(\vartheta/H\) for different values of the fin distance \(s\). This is done in Figure 8.60b. At small fin distances, an increase of \(s\) has a substantial influence on the heat transfer coefficient. At some stage we reach a point where a further increase in \(s\) does not change \(\alpha\). At this stage, the boundary layers of the two fins are no longer in contact and each fin act as a separate vertical plate.

8.61 Elenbaas also treated the problem of *optimizing the fin spacing* so as to achieve the highest possible heat transfer per meter of finned tube. He showed that this optimum fin spacing could be expressed as

\[
s_{q_{\text{max}}} = 2.71 \left[(Gr_s \cdot Pr)/(s^3 \cdot H)\right]^{3/4}
\]

which can be rearranged to

\[
s_{q_{\text{max}}} = 2.71 \left[\left(\frac{Gr_s \cdot Pr}{\vartheta \cdot s^3}ight) \cdot \frac{\vartheta}{H}\right]^{-1/4}
\]
where the inner parenthesis is only a function of thermodynamic properties which for air can be
taken from Figure 8.58. The fin distance and the fin height are expressed in meters and the
temperature difference in degrees Centigrade. In Figure 8.61 this relation is plotted for air of
different temperatures.
Figure 8.60a.: $Nu_s$ vs. $Gr_s \cdot Pr \cdot s/H$ for free convection in between fins.

Figure 8.60b.: Heat transfer coefficient vs. $\theta / H$ for different fin spacings in air at $0^\circ C$. 
8.62 The liquid flow in liquid coolers may flow inside tubes or perpendicularly across the outside of the tubes. When flowing inside tubes, the flow is usually turbulent and the heat transfer coefficient may be calculated from the Dittus-Boelter equation

$$\text{Nu} = 0.023 \cdot \text{Re}^{0.8} \cdot \text{Pr}^{0.4}$$

valid when $\text{Pr} > 0.7$
$\text{L/d} > 60$
$\text{Re} > 10000$

For fluids with low viscosity, $\mu < 2 \cdot \mu_{\text{H}_2\text{O}}$, this equation may be used down to $\text{Re} \approx 2300$. The heat transfer coefficient is referred to the logarithmic mean temperature difference, and fluid properties should be evaluated at bulk temperature.

An alternative correlation is proposed by Gnielinski (1976).

$$\text{Nu} = \frac{(f / 8) \cdot (\text{Re} - 1000) \cdot \text{Pr}}{1 + 12.7 \cdot (f / 8)^{0.5} \cdot (\text{Pr}^{0.5} - 1)}$$

where

$$f = [0.79 \cdot \ln(\text{Re}) - 1.64]^2$$

valid when $0.5 < \text{Pr} < 2000$
$2300 < \text{Re} < 5 \cdot 10^6$
This equation is considered more accurate in the transition- and low turbulence region. The correlations are compared in Figure 8.63 for the case of Pr = 10 and long tubes (no influence of entrance region). For Prandtl numbers around 1 the agreement is much better.

8.63 **Laminar flow inside tubes** should normally be avoided because of the resulting low heat transfer coefficients\(^1\). Still, it may occur for example with high viscosity secondary refrigerants at low temperatures (c.f. Chapter 6). In laminar tube flow the heat transfer coefficient will decrease in the first part of the tube (the thermal entrance region) and then stabilize at a value corresponding to Nu = 3.7 (for circular tubes). For “long” tubes this value may be used as an average if the heat transfer coefficient is referred to the logarithmic mean temperature difference.

The length of the entrance region can be estimated as

\[
x_i = 0.05 \cdot \text{Re} \cdot \text{Pr} \cdot d \quad 8.63a
\]

An equation for the average heat transfer coefficient for both the entrance region and the later part of the tube in laminar flow is proposed by Hausen (1943).

\[
\text{Nu} = 3.66 + \frac{0.0668 \cdot (d / L) \cdot \text{Re} \cdot \text{Pr}}{1 + 0.04 \cdot [(d / L) \cdot \text{Re} \cdot \text{Pr}]^{2/3}} \quad 8.63b
\]

\(^1\) For very small hydraulic diameters high heat transfer coefficients may be reached also in laminar flow. This is utilized in certain types of compact heat exchangers.
8.64 The friction pressure drop in flow inside tubes is calculated as

\[ \Delta p_f = f_1 \cdot (\rho \cdot w^2) \cdot L / d \]  

where

\[ f_1 = 0.158 \cdot \text{Re}^{-1/4} \]  

for turbulent flow in smooth tubes

and

\[ f_1 = 32 / \text{Re} \]  

for laminar flow.

To the friction pressure drop should be added the pressure drop in bends, valves and fittings. This is calculated as

\[ \Delta p_{bv} = \zeta \cdot (\rho \cdot w^2)/2 \]  

where \( \zeta \) can be considered as a constant, independent of velocity. The value of \( \zeta \) for bends and changes in cross section area are found in most textbooks in fluid mechanics. For valves it should be given by the manufacturer.

8.65 It has become common practice to use tubes with internal enhanced surfaces on the heat source side of liquid coolers. These surfaces may have micro-fins similar to those used for evaporation inside tubes, or they may have little dimples or other small protrusions on the inside tube surface. Tests have shown that these surfaces may enhance heat transfer considerably without having too large influence on the pressure drop. The designer who wishes to use these surfaces is forced to rely on the manufacturer’s information for the calculation of heat transfer coefficients and pressure drop.

8.66 For liquid flow across single tubes as well as tube bundles Equation 8.52a and the accompanying tables can be used. The reader is referred to Sections 8.52 - 8.53. There are also correlations especially adapted for calculating the heat transfer coefficients on the shell side of shell-and-tube heat exchangers. This subject is, however, considered to be outside the scope of the present text.
D. Optimum design criteria

8.67 In designing the evaporator there are several choices the designer has to make which will influence the performance and the cost of the system. Making the right choices is, in the end, a matter of economic optimization rather than technical. The goal should be to invest each penny where it has the largest influence on performance, and to get a payoff of the last invested penny equal to the user’s interest rate.

The subject of optimum design criteria will be treated in more detail in Chapter 16, Part II. For the moment we will give only a few guidelines pertaining to the design of the evaporator. It is important, however, to bear in mind that the evaporator cannot be optimized by itself, only as a part of the system.

8.68 The size of the evaporator (the heat transfer area) will influence the temperature difference between the refrigerant and the heat source, and thereby the evaporation temperature and the COP of the system. A large evaporator will have a high first cost, and the correct balance between first cost and COP is highly dependent on the yearly running time.

The problem of choosing the right size could be restated to that of choosing the optimum temperature difference. This problem was treated by Bäckström (1940). Even though the analysis is 60 years old it has been shown that the conclusions still hold (indicating that the relative prices have not changed), and in Table 8.68 the optimum temperature differences suggested by Bäckström are shown for different types of evaporators.

<table>
<thead>
<tr>
<th>Running time per year (h)</th>
<th>1000</th>
<th>2000</th>
<th>5000</th>
<th>8000</th>
</tr>
</thead>
<tbody>
<tr>
<td>Type of evaporator</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Air coolers</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Free convection</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Log. mean temp. difference</td>
<td>13</td>
<td>10</td>
<td>7</td>
<td>6.5</td>
</tr>
<tr>
<td>Inlet temp. difference</td>
<td>18</td>
<td>14</td>
<td>10</td>
<td>9</td>
</tr>
<tr>
<td>Forced convection</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Mean temp. difference</td>
<td>10</td>
<td>8</td>
<td>5.5</td>
<td>5</td>
</tr>
<tr>
<td>Inlet temperature difference</td>
<td>13</td>
<td>10</td>
<td>7.2</td>
<td>6.5</td>
</tr>
<tr>
<td>Air temperature decrease</td>
<td>5.5</td>
<td>4.5</td>
<td>3.0</td>
<td>2.8</td>
</tr>
<tr>
<td>Liquid coolers</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Secondary refrigerants</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Mean temp. difference</td>
<td>7.2</td>
<td>6.0</td>
<td>4.0</td>
<td>3.5</td>
</tr>
<tr>
<td>Inlet temperature difference</td>
<td>8.8</td>
<td>7.6</td>
<td>5.5</td>
<td>5.0</td>
</tr>
<tr>
<td>Liquid temperature decrease</td>
<td>3</td>
<td>3</td>
<td>3</td>
<td>3</td>
</tr>
</tbody>
</table>

8.69 In dry expansion air coolers, the total length of the refrigerant tubes often has to be divided into a number of parallel loops to limit the pressure drop. A method of optimizing the length of each such loop has been proposed by Granryd (1966, 1992). The method is based on the assumption that the cost of the evaporator is not influenced by the number of parallel loops, only by the total tube length. The length of each parallel loop should thus be chosen to give the
highest possible COP, and this is determined by the saturation temperature at the evaporator outlet. Stated in another way, the temperature difference at the evaporator outlet should be as low as possible.

A change in the tube length influences this temperature difference in two ways: Through the pressure drop and through the heat transfer coefficient. In a short tube the pressure drop will be low and the evaporation temperature will therefore not decrease very much from inlet to outlet. But using short tubes means that a larger number of parallel tubes must be used to have the same total surface area. Thereby the mass flow in each tube will be low and this will result in a low heat transfer coefficient, which in turn will lead to a large temperature difference.

Granryd used the correlations for heat transfer and pressure drop developed by Pierre to deduce the following relation for the optimum length of each parallel loop:

\[ L_{opt} = C_L \cdot \frac{d^{1.29} \cdot \Delta x^{0.83}}{\dot{q}^{0.62} \cdot (y \cdot F)^{3.34}} \]  \hspace{1cm} 8.69a

where

- \( d \) = tube diameter (m)
- \( \dot{q} \) = heat flux (W/m\(^2\))
- \( F \) = combined friction factor, \( \approx 0.02-0.03 \) for systems with oil separator, \( \approx 0.04-0.06 \) for systems without oil separator
- \( y \) = factor dependent on the shape of the pressure drop curve, \( \approx 0.6 \)
- \( \Delta x \) = vapor quality change in evaporator
- \( C_L \) = factor depending on the fluid properties of the refrigerant. Determined as

\[ C_L = 0.56 \left( \frac{k^{0.8} \cdot r^{3.4}}{\lambda \cdot (\nu''-\nu') \cdot \nu'' \cdot T_2} \right)^{1/2.9} \]  \hspace{1cm} 8.69b

The values of \( C_L \) are shown for some refrigerants in Fig. 8.69.

For quick approximations, Granryd gives the following recommendation:

"As a rule of thumb: Design for a pressure drop of about \( \frac{1}{4} \) of the temperature difference on the refrigerant side for smooth tubes."
Figure 8.69.: Diagram for the constant $C_L$ in equation 8.69. (from Granryd, 1992).
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