Heat Exchangers: Theory and Selection

Heat exchangers are devices that transfer heat between two fluids. They can transfer heat between a liquid and a gas (i.e., a liquid-to-air heat exchanger) or two gases (i.e., an air-to-air heat exchanger), or they can perform as liquid-to-liquid heat exchangers. These devices are used in many applications, such as air conditioning, gas turbines, automobiles and electronics cooling. For example, the radiator in a car is a water-to-air heat exchanger that cools the heated water returning from the engine.

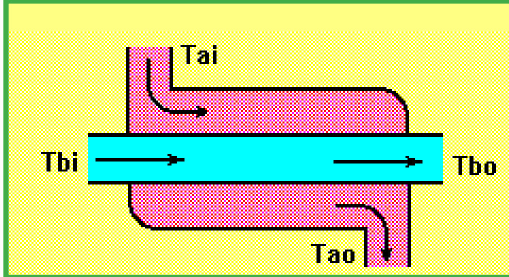

Figure 1. Parallel flow in a heat exchanger.

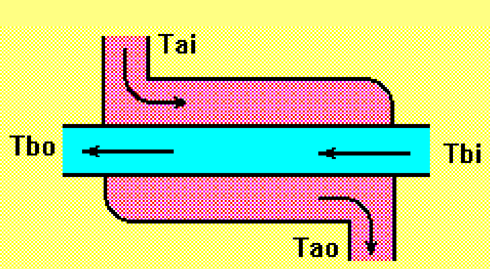

Figure 2. Counter flow in a heat exchanger.

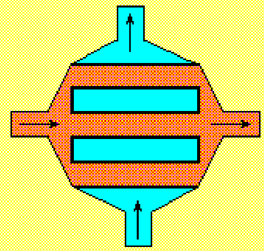

Figure 3. Cross flow heat exchanger.

The many types of heat exchangers can be characterized by flow arrangement and construction. Some common examples are parallel flow heat exchangers, in which both fluids move in the same direction (Figure 1); counter flow heat exchangers, with fluids moving in the opposite directions (Figure 2); and cross flow heat exchangers, where one fluid moves perpendicular to the other (Figure 3).

The cross flow exchanger can be arranged so that one fluid is confined to a tube and the other fluid mixed, or both fluids can be confined to tubes and unmixed. The tubes can have any geometry such as flat or round.

Shell and tube exchangers with different numbers of shells are industrial type exchangers (Figure 4). Compact heat exchangers have very large surface areas, typically on their gas side. Since the gas side has a much smaller heat transfer coefficient compared to the liquid side, increasing the surface area reduces thermal resistance on the gas side. In this type of heat exchanger, the tubes have fins on the outside, or parallel plates are attached to maximize the air side surface area.

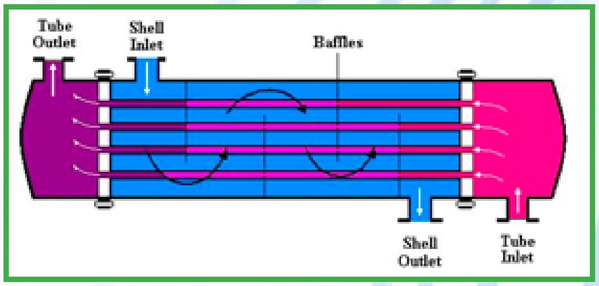

Figure 4. Shell and tube heat exchanger.

The most fundamental variable in heat exchanger design is the definition of the overall effective heat transfer coefficient, U [1]. The formula is:

$$\frac{1}{UA} = \frac{1}{(hA)_h} + R_c + \frac{1}{(hA)_c} + R''_f$$

Where:

U = overall effective heat transfer coefficient

A = area of the hot or cold side of the fluids,

h = heat transfer coefficient for either the cold or the hot side

R_c = conduction resistance of the material separating the hot and cold fluids.

R''_f = fouling factor

The subscripts h and c refer to the hot and cold sides. The term R''_f is the fouling factor thermal resistance. The fouling resistance is caused by impurities of liquids and chemical reactions (e.g., oxidation) that can increase the overall resistance between the two fluids. The above equation calculates the overall heat transfer coefficient based on the hot or the cold side and depending on the selection of the area (A).

Another useful definition is the log mean temperature difference (LMTD). To understand this term, consider a simple counter flow heat exchanger as shown in Figure 5. The temperature distribution of the two fluids is shown in Figure 6. The hot fluid enters the exchanger at temperature $T_{h,i}$ and exits with a temperature of $T_{h,o}$. The cold fluid enters at temperature $T_{c,i}$ and exits with temperature $T_{c,o}$. The temperature difference between the two fluids on the inlet side of the hot fluid is designated as ΔT_1. The temperature difference between the fluids on the inlet side of the cold fluid is designated as ΔT_2 .

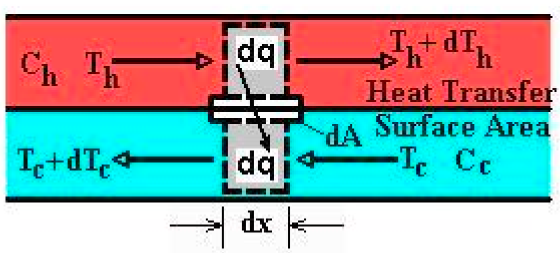

Figure 5. Typical shell and tube heat exchanger.

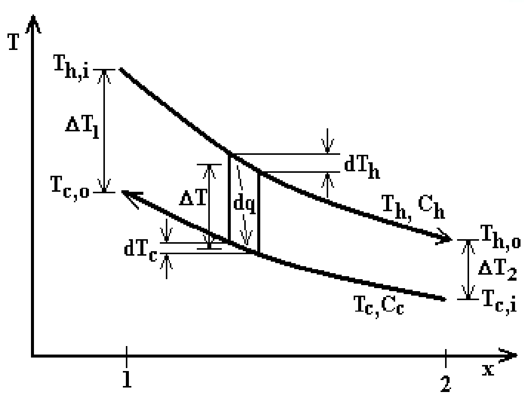

Figure 6. Temperature distribution of hot and cold fluids

By writing the conservation of energy equations between the inlet and outlet of the two fluids, and the differential energy transfer (dq) between the two fluids on a differential distance dx [1], the overall heat transfer between the two fluids can be expressed as:

$$Q = UA \frac{\Delta T2 - \Delta T1}{\ln(\Delta T2 / \Delta T1)}$$

Where U is the overall effective heat transfer coefficient, and A is the contact surface area between the two fluids on either the cold or the hot side.

By defining the LMTD as:

$$\Delta T_{lm} = \frac{\Delta T2 - \Delta T1}{\ln(\Delta T2 / \Delta T1)}$$

The heat transfer can easily be expressed as;

$$Q = UA\Delta T_{lm}$$

In complicated flow paths (such as cross flow and multipass) in heat exchangers with multiple shells and tube passes, there are graphs that correlate a correction factor F to other variables that are functions of inlet and outlet temperatures. The correction factor can be used to modify the LMTD as:

$$\Delta T_{lm} = F\,\Delta T_{lm,CF}$$

Where F is the correction factor and $\Delta T_{lm,CF}$ is the LMTD, if the flow was assumed to be counter flow. These correlations, for a variety of heat exchangers, are described in detail in the classic book of Compact Heat Exchangers by Kays and London [2].

The above procedure becomes tedious, if the outlet temperatures are not known or easily determined. In this case, it becomes an iterative procedure to calculate the temperatures:

1. The outlet temperatures have to be guessed first.
2. The LMTD method is then used to calculate the q.
3. The outlet temperature can then be found from the energy balance.

This procedure must be repeated until convergence.

A more convenient procedure has been devised, called the Effectiveness-NTU method. To explain this method, a few terms need to be defined:

The effectiveness of a heat exchanger is defined as:

$$\varepsilon = \frac{q}{q_{max}}$$

Where q is the real heat transfer, and q_{max} is the maximum possible heat transfer that can be achieved. This can be done with a counter flow and an infinite length heat exchanger.

It can be shown that maximum heat transfer is:

$$q_{max} = \dot{m}_c C_{p,c}(T_{h,i} - T_{c,i}) \quad \text{if} \quad \dot{m}_c C_{p,c} > \dot{m}_h C_{p,h}$$

$$q_{max} = \dot{m}_h C_{p,h}(T_{h,i} - T_{c,i}) \quad \text{if} \quad \dot{m}_h C_{p,h} > \dot{m}\, cC_{p,c}$$

Another variable, the number of transfer units (NTU), is defined as:

$$NTU = \frac{UA}{C_{min}}$$

Where C_{min} is the minimum value of $\dot{m}Cp$ on either the cold or the hot side. It is then shown that the effectiveness can be calculated as:

$$\varepsilon = f\left(NTU, \frac{C_{min}}{C_{max}}\right)$$

The above relationship has been graphed for numerous heat exchanger configurations. Reference 2 provides a comprehensive list of these graphs. For example, if the mass flow rates, the inlet temperatures, the overall heat transfer coefficient and the surface area are all known, the designer can easily calculate C_{min}, C_{max} and NTU. By knowing the type of the heat exchanger, the value of ϵ can be found from the appropriate graphs. By calculating q_{max} and ϵ, q can be calculated. A simple energy balance between the inlet and outlet of the two fluids will then determine the outlet temperatures. A simple example is shown below:

A 1 KW component is attached to a cold plate. A heat sink mounted inside the cold plate draws heat from the component. The heat transferred from the component to the sink is dissipated to the ambient air through a liquid to air heat exchanger. The heat exchanger is of a cross flow type, with both fluids unmixed. It is necessary to find the heat sink base temperature.

The input parameters to the problem are as follows:

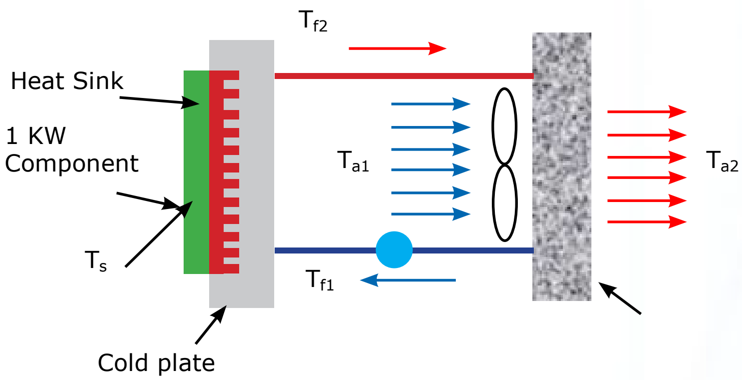

Figure 7. Cooling system using a microchannel [3].

1. q = 1,000 Watts — power dissipation

2. R_{hs} = 0.05 °C/W — Heat sink thermal resistance

3. h_{air} = 50 W/m²K — Air side heat transfer coefficient

4. A_{air} = 3 m² — Air side surface area

5. $\dot{m}_{air}$ = 0.05 (kg/s) — Air mass flow rate

6. $\dot{m}_{water}$ = 0.1(kg/s) — Water mass flow rate

7. T_{a1} = 20 °C — Ambient air temperature

The following steps show how to solve the problem:

1. $C_{air} = \dot{m}_{air} C_{PAir}$

2. $C_{Wat} = \dot{m}_{wat} C_{Pwat}$

3. $C_{min} = \min(C_{Air}, C_{max})$

4. $C_r = \dfrac{C_{min}}{C_{max}}$

5. $NTU = \dfrac{h_{Air} A_{Air}}{C_{min}}$

6. $\varepsilon = 1 - \exp[(1/C_r)(NTU)^{0.22} \{\exp[-C_r(NTU)^{0.78}] - 1\}]$ for cross flow and both fluids unmixed [2]

7. q = 1,000W

8. $q_{max} = \dfrac{q}{\varepsilon}$

9. $q_{max} = C_{min}(T_{f2} - T_{a1})$, find T_{f2} from here

10. $q_{max} = C_{min}(T_{f2} - T_{a1})$, find T_{f1} from here

11. $R_{hs} = \dfrac{T_s - T_{f1}}{q}$ Find heat sink base temperature T_s from here.

T_s = 89.1 °C

The design of a heat exchanger is a complex task and requires attention to many parameters. Of utmost importance for a liquid to air heat exchanger are the power requirements and size of the pump and fan. The design might not be practical if it results in a very low overall thermal resistance, but has very a large pressure drop in the liquid loop or the air passage through the fins. Other factors that must be considered during the design are the reliability of the components, cost, size, fouling factors and manufacturability.

Different applications require different design criteria. For example, in environments such as electronics cooling or aircraft gas turbines, where space is premium, the design should be based on minimizing pressure drops, size and weight. In an air conditioning system for a building, for which size is not an important concern, the design should be based on maximizing performance while minimizing cost.

Since the air side is the usual bottleneck, the designer must decide which type of fins are best for the particular application. Figure 8 shows that the Colburn J factor/friction factor, as a function of Reynolds number, has the highest value for straight fins and lowest value for pin fins [3]. The Colburn J factor is directly related to heat transfer and is defined as:

$$J_H = St\,Pr^{2/3}$$

and the Stanton number is defined as:

$$ST = \frac{NU}{Re\,Pr}$$

Where Nu, Re and Pr are the Nusselt number, Reynolds number, and Prandlte numbers, respectively.

Figure 8 shows that, if the goal is to have a higher value of heat transfer per unit of pressure drop, a straight fin heat sink is a good choice. Figure 9 shows that if the heat transfer per unit of height is important, a pin fin sink is a suitable choice.

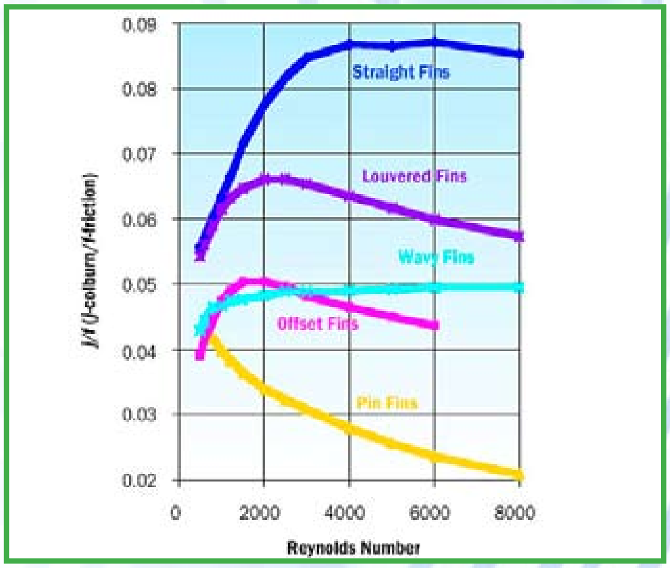

Figure 8. Heat transfer/pressure drop for different fin stocks used in a heat exchanger (ducted flow) [3].

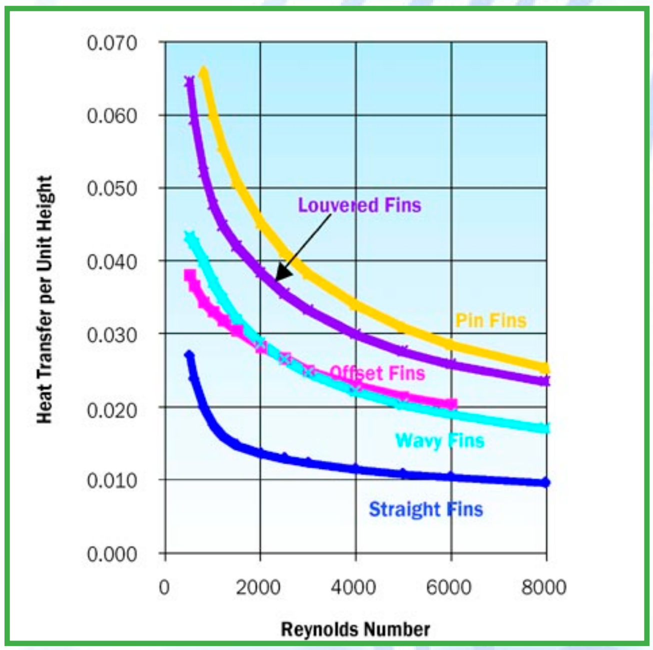

Figure 9. Heat transfer per unit height for different fin stocks used in a heat exchanger (ducted flow) [3].

Nomenclature:

q	Total heat transfer (W)
h	Heat transfer coefficient (W/m2°C)
T_s	Surface temperature (°C)
T_a	Ambient temperature (°C)
Re	Reynolds number
Nu	Nusselt number
Pr	Prandtl number
K	Thermal conductivity (W/m°C)
m°	Mass flow rate (kg/sec)
R	Thermal resistance (°C/W)
A	Surface area (m^2)
C_p	Thermal capacitance (°C/W)
T_h	Hot fluid temperature (°C)
T_c	Cold fluid temperature (°C)
St	Stanton number
j_H	Colburn J factor
T_f	Fluid temperature (°C)
Є	Heat exchanger efficiency
C_{min}	Minimum heat capacity rate (W/°C)
C_{max}	Maximum heat capacity rate (W/°C)
LMTD	Log mean temperature difference
NTU	Number of transfer units

References:

1. Incropera, F. and Dewitt, D., Introduction to Heat Transfer, 1985.

2. Kays, M. and London, A., Compact Heat Exchangers, Third Edition, McGraw-Hill, 1984.

3. Marthinuss, J., Hall, G., Air Cooled Compact Heat Exchanger Design For Electronics Cooling, Electronics Cooling Magazine, Feb 2004, Vol 10, No.1.