PART 1B HEAT TRANSFER

Cambridge University Engineering Department

Stuart Scott (sas37@eng.cam.ac.uk)

Recommended Books

(1)*	MORAN, M.J. & SHAPIRO, H.N.		FUNDAMENTALS OF ENGINEERING THERMODYNAMICS Wiley, 5th Edition (SI Units) 2006	VA 192
(2)	ROGERS, G.F.C & MAYHEW, Y.R.		ENGINEERING THERMODYNAMICS: WORK AND HEAT TRANSFER Harlow, 4th Edition. 1992	VA 178
(3)	HOLMAN, J.P.		HEAT TRANSFER McGraw-Hill, 7th Edition. 1990	

Copies are available in the CUED library and college libraries.

All lecture notes and additional material will be added to the moodle site.

- Suitable tripos questions: 2016 Q1, 2015 Q1, 2014 Q3, 2013 Q1, 2012 Q1, 2011 Q3, 2010 Q1.

Introduction

The study of Heat Transfer aims to help us predict the ***rate*** of energy transfer which takes place due to a difference of temperature. Therefore, this is a different subject from Thermodynamics, which is only concerned with equilibrium states. Thermodynamics tells about what will happen after a certain quantity of heat is transferred, but not how long the transfer will take. Despite this, "introductory" Heat Transfer is often taught within a Thermodynamics course, and we follow this custom here.

There are three mechanisms of heat transfer:-

- Conduction – transfer by molecular vibration, Brownian motion and electron migration,
- Convection – transfer by bulk motion
- Radiation - transfer by electro-magnetic (EM) waves.

In many practical problems, two or even all of these mechanisms may need to be considered together.

1. Conductive Heat Transfer

When a temperature gradient exists in a body, we are unsurprised to note that there is an energy transfer from the hotter to the cooler part. This process is called heat conduction, and experience shows that the rate of energy transfer is proportional to the normal temperature gradient, and therefore we can write

$$\dot{q} = heat\ flux = \quad \frac{\dot{Q}}{A} \propto \frac{\partial T}{\partial x}$$

Where $\dot{Q}$ is the heat transfer (W), T is the temperature, A is the area perpendicular to the temperature gradient in the x direction. Generally we will use lower case $\dot{q}$ to indicate heat flow per unit area. The constant of proportionality is called the ***thermal conductivity*** λ (W/mK), (note many texts use k), and since the heat transfer is in the direction of decreasing temperature, (the Second Law insists on this),

$$\frac{\dot{Q}}{A} = -\lambda \frac{\partial T}{\partial x}$$

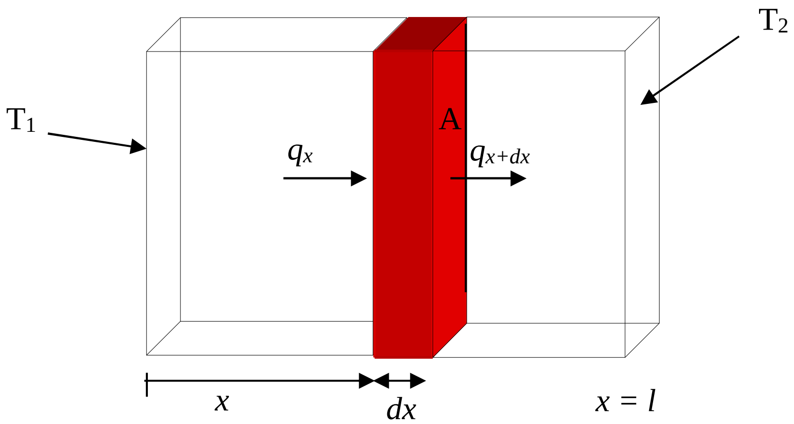
This is called Fourier's Law of heat conduction; though note, (like for example Ohm's Law), it is actually an empirical relationship. Table 1, below, gives some values for thermal conductivity at 20 ºC. There is always some variation of λ with temperature, though often it can be neglected.

Table 1: Thermal conductivity of various materials at 20 °C

Material	Thermal Conductivity W/mK	Material	Thermal Conductivity W/mK
Diamond	2300	Brick	0.72
Silver	429	Water (liquid)	0.613
Copper	401	Human Skin	0.37
Gold	317	Wood (oak)	0.17
Aluminium	237	Glass Fibre	0.043
Mild Steel	55	Air (gas)	0.026
Stainless Steel	16	Urethane (rigid foam)	0.026
Glass	0.78	Argon (gas)	0.018

1.1. *Using Fourier's Law – Conduction in one dimension*

Let us consider first a ***steady state***, ***one dimensional*** heat flow. Consider a slab of material as in the figure below.



For the element of thickness *dx*, the heat transferred into the face at *x*, and the heat conducted out of the face at *x+dx*, are given, per unit area, by:-

$$\dot{q}_x = -\lambda \frac{dT}{dx} \qquad \text{and} \qquad \dot{q}_{x+dx} = \dot{q}_x + \frac{d}{dx}(\dot{q}_x)dx$$

Note we have dropped the partial derivatives as this is a 1D problem (x variation only), in steady state (no time derivatives). **At steady state there can be no accumulation of energy in the element, meaning that**

$$\dot{q}_x = \dot{q}_{x+dx} \qquad \Longrightarrow \qquad \dot{q}_x = \dot{q}_x + \frac{d}{dx}(\dot{q}_x)dx$$

$$\frac{d}{dx}(\dot{q}_x) = 0$$

so $$\frac{d}{dx}\left(-\lambda\frac{dT}{dx}\right) = 0 \quad \Longrightarrow \quad -\lambda\frac{dT}{dx} = constant = \dot{q}_x = \frac{\dot{Q}}{A}$$

If the thermal conductivity is constant, we get the obvious result that the temperature gradient is constant. Integrating again (with constant thermal conductivity), and inserting the boundary conditions, we obtain

$$T = T_1 - \frac{x}{l}(T_1 - T_2) \quad \text{and} \quad \dot{Q} = A\lambda\frac{(T_1 - T_2)}{l}$$

It is often useful to express the last equation as:

"Voltage Drop" ↔ Temperature drop

"Current" ↔ heat flow

$$\dot{Q} = A\lambda \frac{(T_1 - T_2)}{l} = \frac{\overbrace{(T_1 - T_2)}}{R_{th}},$$

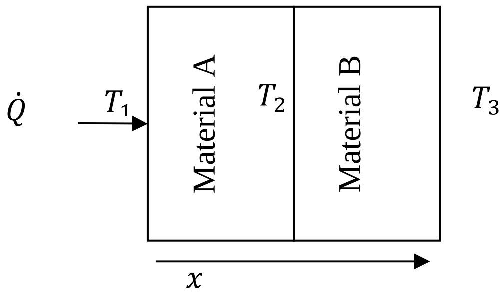
I = V/R

where (by analogy with Ohm's law)

$$\boxed{\boldsymbol{R_{th} = \frac{l}{\lambda A}} \textbf{ is the thermal resistance}}$$

1.2. Heat transfer resistance in series

Consider heat flowing through slabs different materials:



For this ***1D, steady*** conduction problem, the heat flow (and heat flux) is not a function of x.

Material A: $\dot{Q} = \frac{T_1 - T_2}{R_A}$ $\quad R_A\dot{Q} = (T_1 - T_2)$

Material B: $\dot{Q} = \frac{T_2 - T_3}{R_B}$ $\quad R_B\dot{Q} = (T_2 - T_3)$

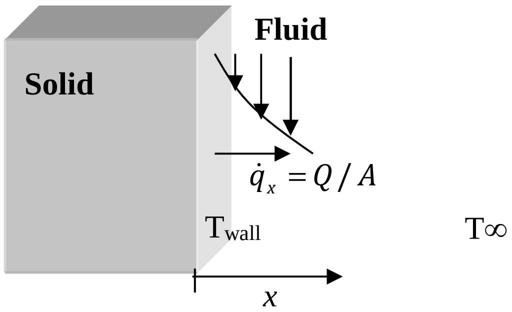
Adding (and noting that $\dot{Q}$ is the same)

$$\dot{Q}(R_A + R_B) = T_1 - T_3 \qquad \Longrightarrow \qquad R_{Total} = R_A + R_B$$

Heat transfer resistances in series add! This should be obvious from the Ohms law analogy.

1.3. Another boundary condition - Convective heat losses at fluid interface

Previously, we assumed the temperatures at the ends were known. Often heat from a surface is transferred to a flowing fluid (i.e. a convective boundary condition).



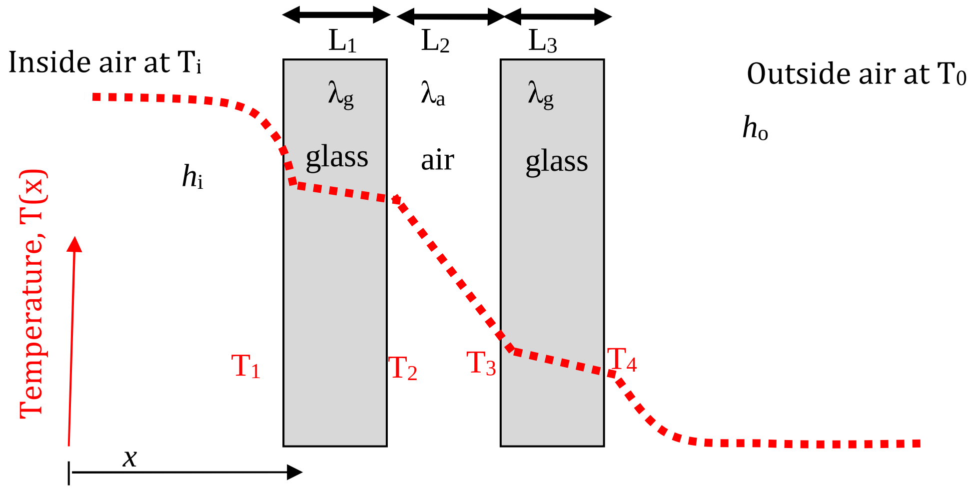
The heat flux out of the solid is proportional to the temperature difference (T_∞ - T_{wall}),

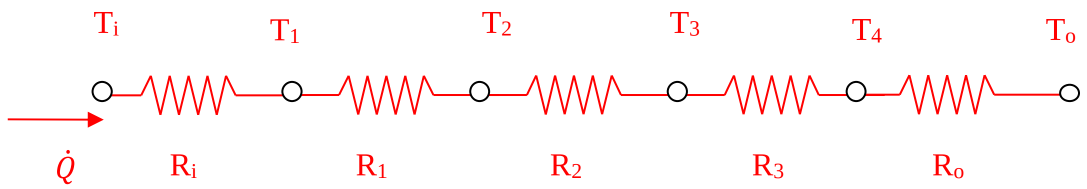
$$\dot{Q} = -hA(T_\infty - T_{wall}) = hA\,(T_{wall} - T_\infty)$$

Where h is the coefficient of heat transfer and $R_{th} = \frac{1}{hA}$

1.4. Example- Double glazed windows.

Determine the heat transfer through a window consisting of two 3 mm thick glass panes, with a gap of 7 mm between them. Take the convective heat transfer coefficient as 10 W/m²K, T_0 = 5 °C, T_i = 25 °C; and assume that the air between the panes acts as a conductive layer. λ_a = 0.026 W/mK and λ_g = 0.78 W/mK.

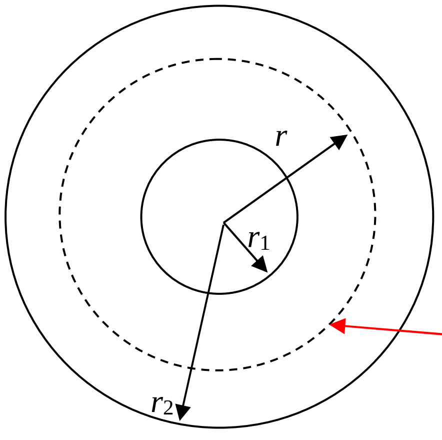




	Thermal resistance $(\frac{m^2k}{W})$	Temperature difference (°C)
Inside	$R_i = \dfrac{1}{h_i A} = \frac{1}{10\times1} = 0.1$	$\Delta T = \dot{Q}R_i = 4.2$
Glass	$R_1 = \dfrac{L_1}{\lambda_g A} = \frac{3\times10^{-3}}{0.78\times1} = 3.85\times10^{-3}$	0.16
Air gap	$R_2 = \dfrac{L_2}{\lambda_a A} = \frac{7\times10^{-3}}{0.026\times1} = 0.269$	11.3
Glass	$R_3 = \dfrac{L_3}{\lambda_g A} = \frac{3\times10^{-3}}{0.78\times1} = 3.85\times10^{-3}$	0.16
Outside	$R_o = \dfrac{1}{h_0 A} = \frac{1}{10\times1} = 0.1$	4.2
Total	0.477	20

Therefore the total heat loss is: $\dot{Q} = \frac{T_o - T_i}{R_T} = 20/.477 = 43$ W

1.5. Radial Heat Transfer – Cylinders, steady state conduction



Consider a circular cylinder as shown, of length *l*. With boundary conditions:- at r = r_1, T = T_1, and at r = r_2, T = T_2.

- $\dot{Q} = -2\pi r l \lambda \frac{dT}{dr}$
- $\dot{Q}$ is a constant
- $\dot{q} = \dot{Q}/A$ varies with r

$A = 2\,\pi r l$

Starting with

$$\dot{Q} = -2\pi r l \lambda \frac{dT}{dr}$$

$$\frac{\dot{Q}}{r} = -2\pi l \lambda \frac{dT}{dr}$$

$$\int_{r_1}^{r_2} \frac{\dot{Q}}{r} dr = -\int_{T_1}^{T_2} 2\pi l \lambda dT$$

$\dot{Q}$ is a constant!

$$\dot{Q} \int_{r_1}^{r_2} \frac{1}{r} dr = -2\pi l \lambda \int_{T_1}^{T_2} dT$$

$$\dot{Q}[\ln(r)]_{r_1}^{r_2} = -\,2\pi l \lambda (T_2 - T_1)$$

$$\dot{Q} \ln\left(\frac{r_2}{r_1}\right) = 2\pi l \lambda (T_1 - T_2)$$

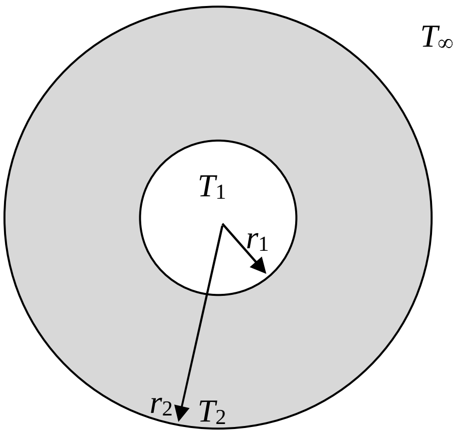
$$\dot{Q} = \frac{2\pi l \lambda (T_1 - T_2)}{\ln\left(\frac{r_2}{r_1}\right)}$$

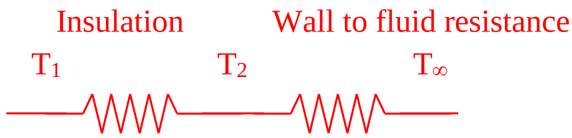
The thermal resistance is then

$$R_{th} = \frac{T_1 - T_2}{\dot{Q}} = \frac{\ln\left(\frac{r_2}{r_1}\right)}{2\pi l \lambda}$$

1.6. *Example- Insulation on a pipe*

Note that as the area for heat transfer increases with radius, it is possible to add material but <u>increase</u> the heat transfer. For example, consider a hot pipe, wrapped n insulating material of thermal conductivity λ. The inner and outer radii of the insulating material are r_1 and r_2, respectively. The temperature at r_1 is T_1. The outer surface of the insulator is in contact with the environment, where the convective thermal resistance is $1/hA$.

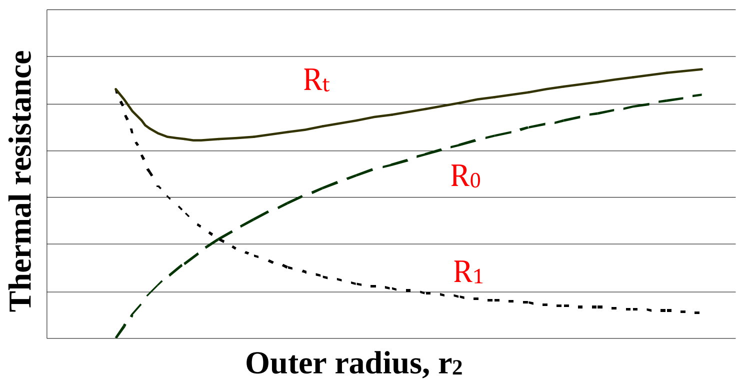




$$\mathrm{R}_0 = \ln\left(\frac{r_2}{r_1}\right)/2\pi\lambda l \qquad R_1 = \frac{1}{2\pi r_2 l h}$$

The overall thermal resistance is thus

$$R_t = \frac{\ln\left(\frac{r_2}{r_1}\right)}{2\pi\lambda l} + \frac{1}{2\pi r_2 l h}$$



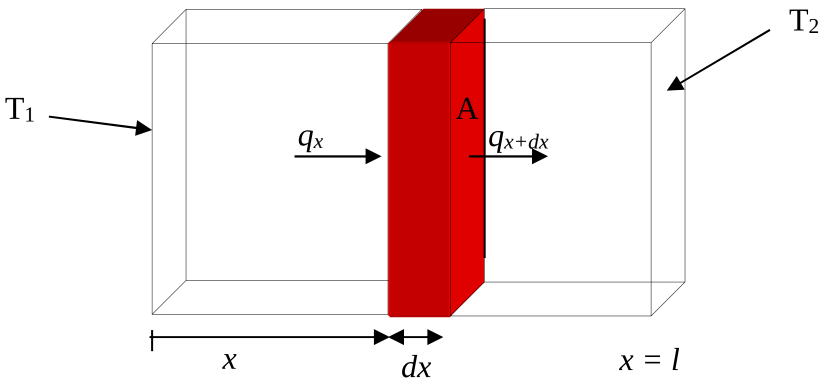
The first term on the RHS increases with insulator thickness, but the second term decreases. Therefore there can be a minimum value of R_t- i.e. there maybe some non-zero thickness of insulation that actually maximises the heat transfer! (There is a question in the exercise sheet that examines this behaviour in more detail.)

1.7. *Key points for 1D steady heat conduction*

- The rate of heat flux is given by Fourier's law, $\frac{\dot{Q}}{A} = -\lambda \frac{\partial T}{\partial x}$.
- In a planar geometry, both the heat flux and the heat flow are constant.
- In a cylindrical geometry, the heat flux is a function of position.
- We can break problems down into a series of thermal resistances, and use the analogy with ohms law to build an equivalent circuit.
- For a planar system, the thermal resistance of a material is given by $R_{th} = \frac{l}{\lambda A}$
- For a cylindrical system, the thermal resistance of a material is given by $R_{th} = \frac{\ln\left(\frac{r_2}{r_1}\right)}{2\pi l \lambda}$.
- Where heat is transferred from a surface to a fluid, the convective thermal resistance is given by $R_{th} = \frac{1}{hA}$.

2. Unsteady Heat Conduction.

Let's return to conduction in a slab of planar material, but now relax the assumption that system is at steady state. We can also consider the case where heat is generated internally (ohmic heating for example).



Let us assume that the internal heat generation is G W/m^3, and that the temperature change of the element in time δt is δT. The net energy input to the element must then equal the change in the internal energy of the element, so,

$$\rho c A dx \frac{\delta T}{\delta t} = q_x A \qquad - \left[q_x + \frac{\partial q_x}{\partial x} dx \right] A \qquad + G A dx$$

Accumulation = Energy inflow – energy outflow + Generation

where c is the material specific heat.

$$\rho c \frac{\partial T}{\partial t} = - \frac{\partial q_x}{\partial x} \quad + G$$

$$\rho c \frac{\partial T}{\partial t} = - \frac{\partial}{\partial x} \left[-\lambda \frac{\partial T}{\partial x} \right] \quad + G$$

If the thermal conductivity is constant, then the above equation reduces to

$$\frac{\partial T}{\partial t} = \left(\frac{\lambda}{\rho c} \right) \frac{\partial^2 T}{\partial x^2} \quad + \frac{G}{\rho c}$$

We have already used the stead

form of this equation, i.e. $\frac{\partial T}{\partial t} = 0$

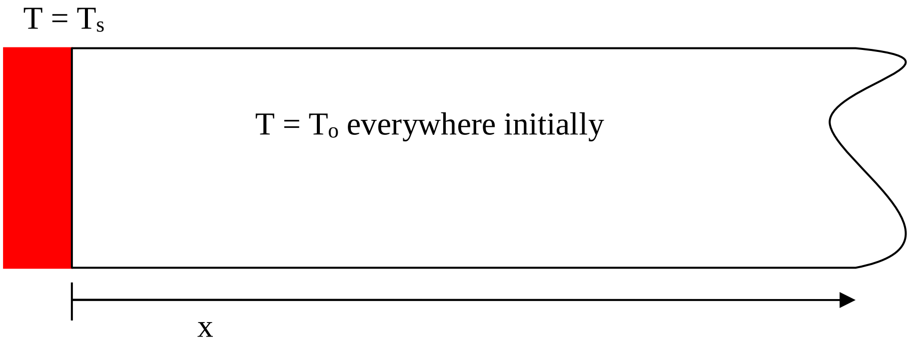
$\lambda / \rho c = \alpha$ is called the thermal diffusivity (m^2/s).

In the absence of internal heat generation, we have the 1D transient heat diffusion equation,

$$\frac{\partial T}{\partial t} = \alpha \frac{\partial^2 T}{\partial x^2}$$

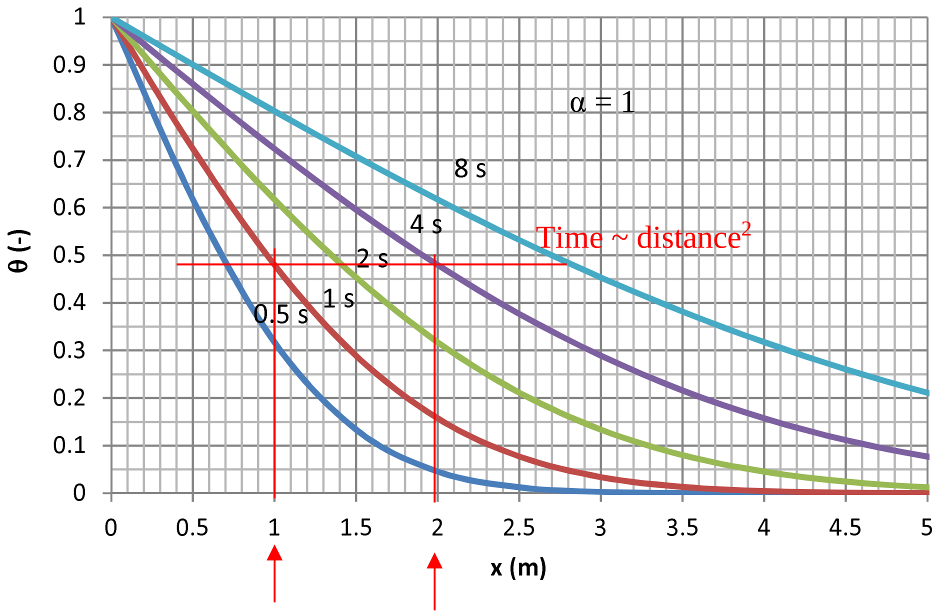
Exact solutions of this equation only exist for a few simple geometries and boundary conditions, so numerical solution is usually required.

2.1. Example- conduction in a semi-infinite slab



This is one case where it is possible to derive an analytical solution (you will not be asked to reproduce this, it is included here for illustrative purposes only)

$$\theta = \frac{T - T_0}{T_s - T_0} = 1 - \mathrm{erf}\left(\frac{x}{2\sqrt{\alpha t}}\right)$$



Characteristic time for heat to penetrate a distance *x* is when,

$$\frac{x}{\sqrt{\alpha t}} = 1\ (say) \quad i.e. \quad t = \frac{x^2}{\alpha}$$

More generally, the "characteristic time" for heat diffusion is s^2/α, where *s* is a characteristic dimension of the body, e.g. (volume)/(surface area).

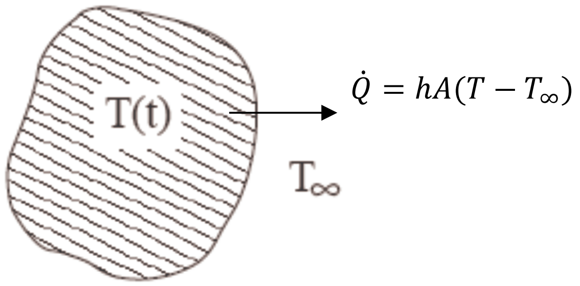
Another non-dimensional group that is useful in transient heat conduction problems is the Fourier number, *Fo*,

$$Fo = \frac{\text{time, } \tau}{\text{characteristic time for heat diffusion through body}} = \frac{\alpha\tau}{s^2}$$

A body can be considered to have heated up when Fo > 1

2.2. *Lumped Heat Capacity Analysis*

In many situations, a transient heat transfer situation involves a solid (with relatively low "internal" thermal resistance) and a solid/fluid interface with a relatively high thermal resistance. In such cases, the assumption can often be made that the solid body is at a uniform temperature. Let us consider a "lump" of material of specific heat c, density ρ, volume V, area A, with convective heat transfer coefficient h.



If the temperature is uniform throughout the body, then we can write

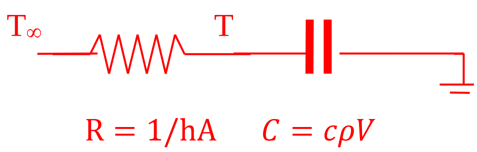
$$c\rho V \frac{dT}{dt} = -\dot{Q} = -[hA(T - T_\infty)]$$

If the initial condition is $T = T_0$ at $t = 0$, the solution is

$$\frac{T - T_\infty}{T_0 - T_\infty} = \exp\left(-\frac{t}{\tau_c}\right),$$

where $\tau_c = c\rho V / hA$.

The analogy with a first order electrical RC network is clear,



2.2.1. *When is the lumped capacity model valid?*

As noted above, the assumption is that

$$\frac{internal\ thermal\ resistance}{surface\ thermal\ resistance} = \frac{\frac{s}{\lambda A}}{\frac{1}{hA}} = \frac{hs}{\lambda} \ll 1$$

[where s = V/A is assumed to represent a characteristic length scale of the body].

The group hs/λ is called the ***Biot number (Bi)***. Though arbitrary, *Bi < 0.1* is often used as an acceptable condition for applying a lumped heat capacity analysis

2.3. Worked Example

A steel ball (specific heat 0.46 kJ/kg.K, λ = 55W/m.K, density = 7800 kg/m^3) 5 cm in diameter, initially at 450 °C, is placed in an environment at 100 °C. The convection heat transfer coefficient is 100 W/m^{2}K. Should a lumped heat capacity approach be used? – If so, what is the cooling time constant?

First we need to calculate the Biot number

$$Bi = \frac{hs}{\lambda} = \frac{hV}{\lambda A}$$

$$= \frac{h\frac{\pi d^3}{6}}{\lambda \, \pi d^2} = \frac{hd}{6\lambda}$$

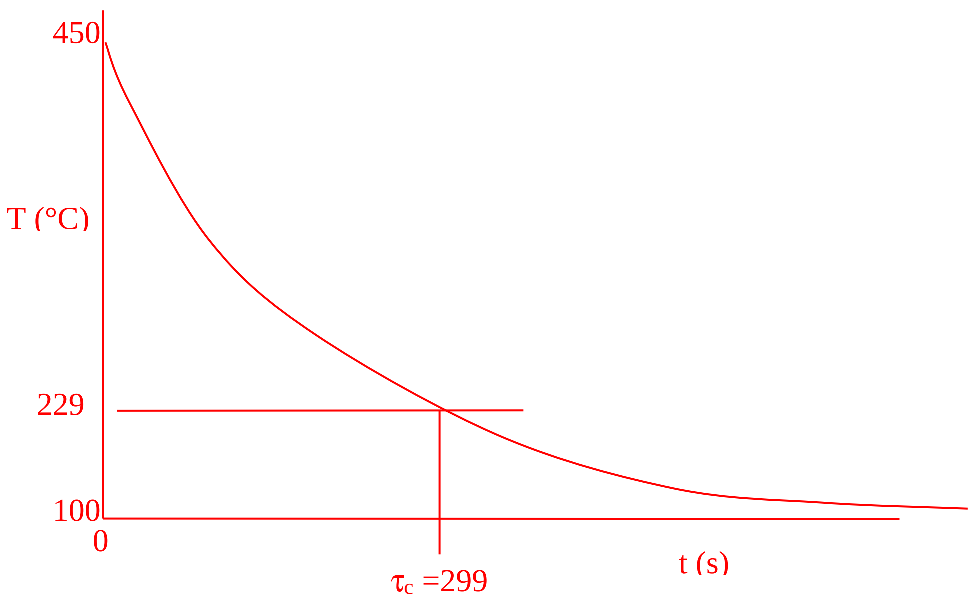
$$= \frac{100 \times 0.05}{6 \times 55} = 0.0126$$

Therefore the lumped heat capacity model is appropriate!

The time constant for heating is

$$\tau_c = \frac{c\rho V}{hA} = \frac{c\rho d}{h6}$$

$$= \frac{460 \times 7800 \times 0.05}{6 \times 100} = 299\, s\ (5\ minutes)$$



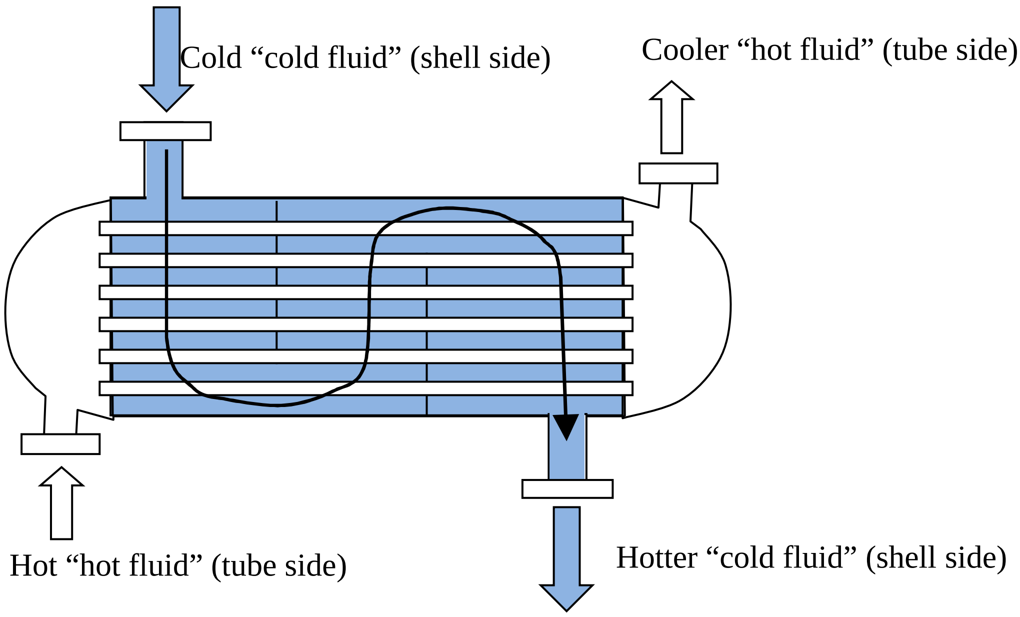
2.4. Key Points for unsteady conduction

- The “characteristic time” for heat diffusion through a body is s^2/α, where s is a characteristic dimension of the body, e.g. (volume)/(surface area).
- $\alpha = \lambda/\rho c$, is the thermal diffusivity.
- The Fourier number is the ratio of time, τ to the characteristic time for heat diffusion, i.e. $Fo = \frac{\alpha\tau}{s^2}$
- The Biot number is given by Bi = hs/λ .
- When Bi < 0.1 conduction within the body is very fast, compared to transport of heat to the body by convection from the environment. The body can be treated as if it has a uniform temperature.

3. Heat Exchangers

There is little need to emphasise the importance of heat exchangers – they are all around us – in vehicles (the "radiator", the passenger compartment heater, the oil cooler), in power stations (the "boiler", feed heaters), etc.

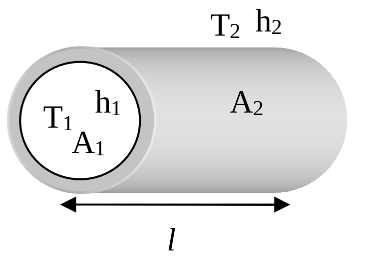
One of the most common forms of heat exchanger consists of banks of tubes (known as a shell and tube heat exchanger).



3.1. *The overall heat transfer coefficient*

Previously, in the double glazed window example, we determined an overall thermal resistance for the combination of convection $(1/hA)$ and conduction $(x/\lambda A)$ resistances, such that the heat transfer is given by $\dot{Q}=\Delta T/R_{th,overall}$. Though this formulation is to be preferred, common usage is to express the same relationship as $\dot{Q}=UA\Delta T$, where ***U* is called the *Overall Heat Transfer Coefficient***, and $UA=1/R_{th,overall}$.

For problems with radial conduction, there is the issue of what area A refers to. Consider the pipe, of length l as shown, with inner and outer radii r_1 and r_2, respectively.



The equivalent circuit for this problem is:

$$R_1=\frac{1}{A_1h_1}\qquad R_2=\ln\left(\frac{r_2}{r_1}\right)/2\pi\lambda l\qquad R_3=\frac{1}{A_2h_2}$$

Giving an overall resistance of

$$R_t=\frac{1}{A_1h_1}+\frac{\ln\left(\frac{r_2}{r_1}\right)}{2\pi\lambda l}+\frac{1}{A_2h_2}$$

and

$$\dot{Q}=U_1A_1(T_1-T_2)=U_2A_2(T_1-T_2)=\frac{T_1-T_2}{R_t}$$

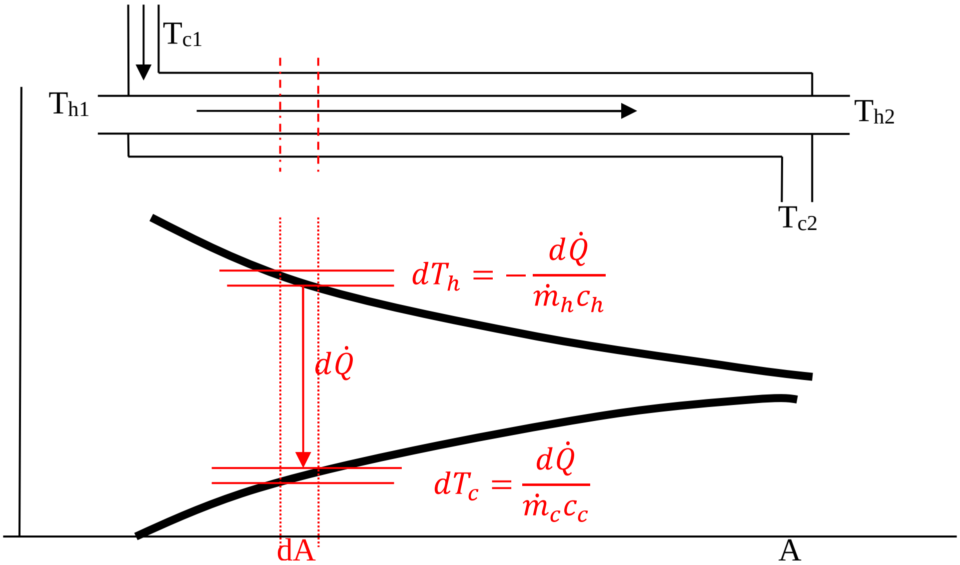
$$U_1A_1=U_2A_2=\frac{1}{R_t}$$

and so $U_1=\dfrac{1}{\dfrac{1}{h_1}+\dfrac{A_1\ln(r_2/r_1)}{2\pi\lambda l}+\dfrac{A_1}{A_2}\dfrac{1}{h_2}}$ $\qquad U_2=\dfrac{1}{\dfrac{A_2}{A_1}\dfrac{1}{h_1}+\dfrac{A_2\ln(r_2/r_1)}{2\pi\lambda l}+\dfrac{1}{h_2}}$

Where U_1 is the "Overall heat transfer coefficient based on the inner diameter" and U_2 is the "Overall heat transfer coefficient based on the outer diameter".

3.2. *Performance of heat exchangers*

One of the simplest heat exchanger geometries is as shown in the figure. This is called a co- or parallel-flow HX (we will use this common abbreviation for Heat eXchanger); if the fluids were flowing in opposite directions, it would be termed a counter-flow HX. Clearly the heat transfer is taking place due to a continuously changing temperature difference, so at first sight it appears a tricky problem to predict the total heat exchanged, but clearly there must be some "mean" temperature ΔT_m, which is defined by $\dot{Q} = UA\Delta T_m$ (where UA is either $U_1 A_1$ or $U_2 A_2$).



Consider a small element of the heat exchanger of area dA, where the heat exchanged is $d\dot{Q}$.

$$d\dot{Q} = -\dot{m}_h c_h dT_h = \dot{m}_c c_c dT_c = UdA(T_h - T_c)$$

We will assume that U, the overall heat transfer coefficient, and the fluid specific heats are constant. We can write $dT_h = -d\dot{Q}/\dot{m}_h c_h$, and $dT_c = d\dot{Q}/\dot{m}_c c_c$

$$\Rightarrow dT_h - dT_c (= d(T_h - T_c)) = -d\dot{Q}\left(\frac{1}{\dot{m}_h c_h} + \frac{1}{\dot{m}_c c_c}\right)$$

$$\therefore \quad \frac{d(T_h - T_c)}{T_h - T_c} = -U\left(\frac{1}{\dot{m}_h c_h} + \frac{1}{\dot{m}_c c_c}\right) dA.$$

Integrating from one end to the other,

$$\ln\left(\frac{T_{h2} - T_{c2}}{T_{h1} - T_{c1}}\right) = -UA\left(\frac{1}{\dot{m}_h c_h} + \frac{1}{\dot{m}_c c_c}\right).$$

Now from the first law, $\dot{Q} = \dot{m}_h c_h (T_{h1} - T_{h2}) = \dot{m}_c c_c (T_{c2} - T_{c1})$, so finally we obtain

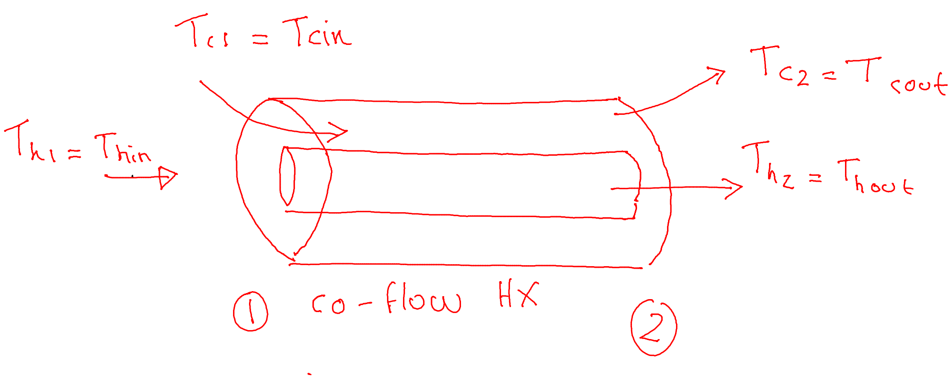
$$\dot{Q} = UA\left(\frac{(T_{h2} - T_{c2}) - (T_{h1} - T_{c1})}{\ln[(T_{h2} - T_{c2})/(T_{h1} - T_{c1})]}\right)$$

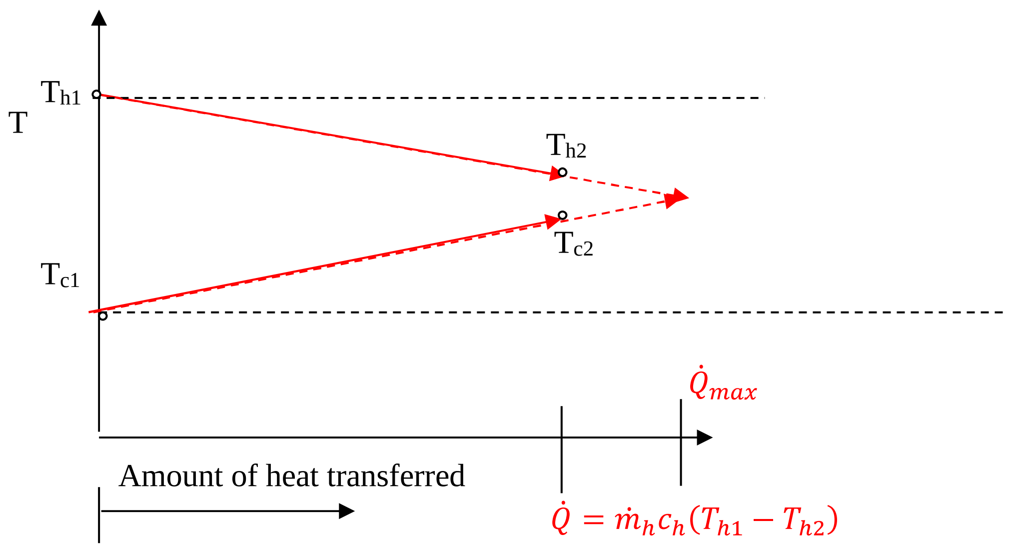
$$\dot{Q} = UA\Delta T_m \text{ and } \Delta T_m = \frac{\Delta T_2 - \Delta T_1}{ln\left(\frac{\Delta T_2}{\Delta T_1}\right)}$$

ΔT_m **is called the *log mean temperature difference*** and is the appropriate "average" temperature driving force for heat transfer. These relationships apply to counter-flow HX's as well[1].

[1] You can also easily show that this also holds when the temperature is constant on one side of the heat exchanger, e.g., when steam is condensing on the shell side of an exchanger.

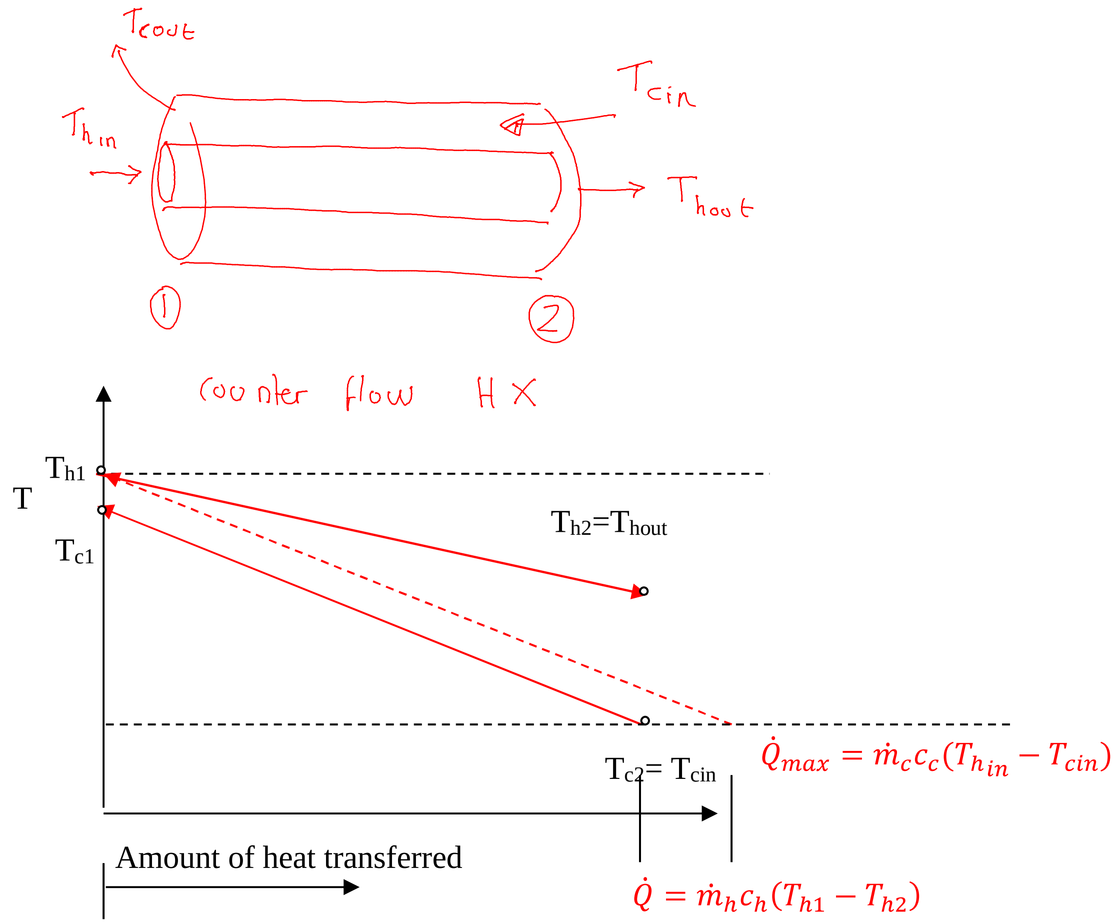
3.2.1. Co-flow heat exchanger T-Q plot





- $\Delta T_m = \frac{\Delta T_2 - \Delta T_1}{ln\left(\frac{\Delta T_2}{\Delta T_1}\right)}$ with $\Delta T_2 = T_{hout} - T_{cout}$, $\Delta T_1 = T_{hin} - T_{cin}$
- The gradient of a T-Q plot is $1/\{\dot{m}c\}$
- For a co – flow heat exchanger, the maximum heat transfer is when the exit temperatures become equal ($T_{h2} = T_{c2}$)
- Maximum heat transfer requires infinite heat transfer area, $\Delta T_{lm} = 0$

3.2.2. Counter flow heat exchanger T-Q plot



- As you move along the heat exchanger, the cold stream can approach the inlet hot fluid temperature OR the hot stream can approach the inlet temperature of the cold fluid. It depends on which stream has the lowest $\dot{m}c$.

- At end (2) the coldest hot fluid transfers heat to the coldest cold fluid. At end (1) the hottest hot fluids transfers heat to the hottest cold fluid.
  - o The lower temperature difference reduces irreversibilities.
  - o More heat can be transferred.

- $\Delta T_m = \frac{\Delta T_2 - \Delta T_1}{ln\left(\frac{\Delta T_2}{\Delta T_1}\right)}$ with $\Delta T_2 = T_{hout} - T_{cin}$, $\Delta T_1 = T_{hin} - T_{cout}$

- As drawn, with the cold flow having the lowest value $\dot{m}c$, the maximum heat possible heat transfer is when $T_{cout} = T_{hin}$.

- Maximum heat transfer requires infinite heat transfer area, $\Delta T_{lm} = 0$

3.2.3. *Summary of T-Q plots and heat exchanger effectiveness*

- For a constant $\dot{m}c$, the lines of T vs Q are straight.
- The maximum possible heat exchange will always be $(\dot{m}c)_{\min}$, times the biggest temperature difference existing in the HX.
- Maximum possible heat exchange requires an infinite area for heat transfer
- The *effectiveness (ε)* of a HX is defined as the actual heat exchange, divided by the maximum possible (**achieved in counter flow mode**).

For the situation above, this will be given by

$$\varepsilon = \frac{\dot{m}_h c_h \left(T_{hin} - T_{hout}\right)}{(\dot{m}c)_{\min}\left(T_{hin} - T_{cin}\right)}.$$

3.3. *Key points for heat exchangers*

- We often use an overall heat transfer coefficient U, whose value depends on the area it is associated with.
- The temperature difference varies along the heat exchanger. The correct appropriate average driving force for heat transfer is the log mean temperature difference.
- Counter flowing heat exchangers are better than co-flowing heat exchangers.
- The *effectiveness (ε)* of a HX is defined as the actual heat exchange, divided by the maximum possible (**achieved in counter flow mode**).