Lecture 7
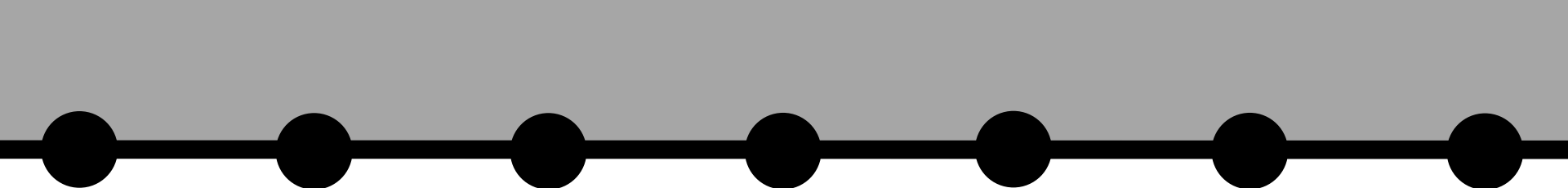
By: Mohammad Mehrali

- Last Lectorial session => Feedback on assignments

- Q&A Session (To be announced)

General Learning Objectives

At the end of the E&HT course the student is expected to:

- **Classify definitions and basic concepts in the field of heat and energy**

- **Explain the physical principles involved in the energy and heat management of products**

- **Recognize the forms of energy (conversion) and heat transfer in a product**

- **Be able to make energy and heat management calculations/estimations for products using the provided equations**

- **Critically judge and compare data and results.**

WORK, ENERGY, POWER

- Work W, energy E, power P

- Units J, W = J/s, kWh, hp, …

- Work *W*, Energy *E* in

- Power   in W = J / s

- Units kWh, kcal, hp, ….

- Comparison / estimating / proportion

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Work, energy, power

Symbol	Definition	Units
W	Work	kJ
w	Specific work = work per unit mass, w = W/m	kJ/kg
	Power = rate of work*	kW (= kJ/s)
Q	Heat transfer	kJ
q	Specific heat transfer = heat transfer per unit mass, q = Q/m	kJ/kg
$\dot{Q}$	Rate of heat transfer*	kW (= kJ/s)

- **NOTE: Rates are denoted by a dot on top of the variable.**

- Energy is **always conserved**!

- First law of Thermodynamics:

The First Law of Thermodynamics
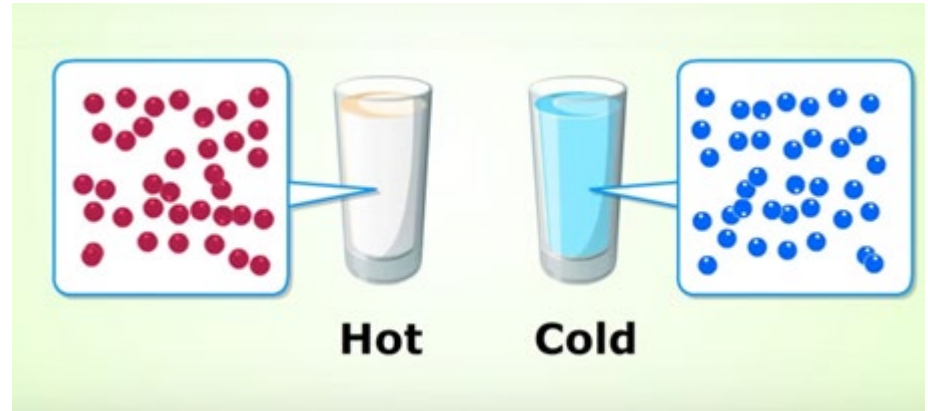Energy transformation
Energy before → Energy after
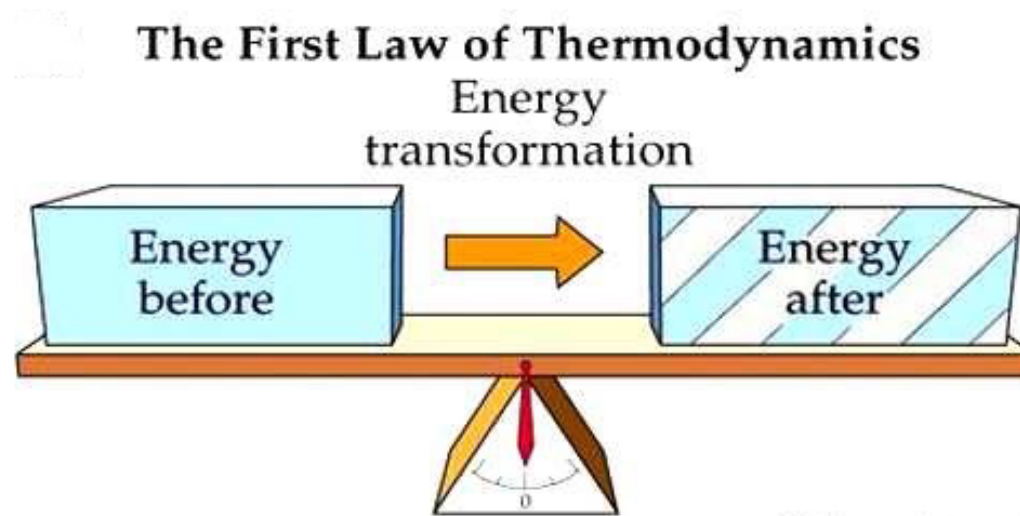
- Efficiency = fraction of "useful" work/power used
  - What is defined as useful?
  - What is the reference?

- Efficiency $\quad \eta = \dfrac{\text{useful work}}{\text{input energy}} = \dfrac{\text{useful power}}{\text{input power}} \quad$ (-)

- Use a Sankey diagram!

- Thermal energy: kinetic energy of molecules and atoms.

Heat Transfer: $\Delta E = Q$ [J]

Hot   Cold

m = mass of "system" (kg)

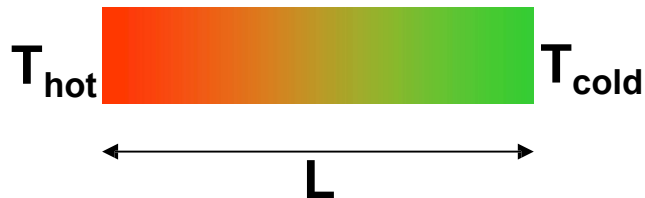ΔT = temperature change of system during process (K)

c = specific heat (J / (kg · K))
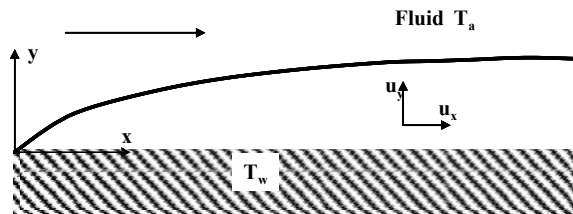
$$Q = m \cdot c \cdot \Delta T \text{ [J]}$$
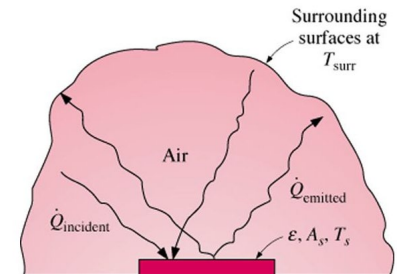
Heat Transfer Modes

Conduction

T_{hot} T_{cold}

L

Convection

Fluid T_a

y

x

u_y u_x

T_w

- **Natural Convection**
- **Forced Convection**

Radiation

Surrounding surfaces at T_{surr}

Air

$\dot{Q}_{emitted}$

$\dot{Q}_{incident}$

ε, A_s, T_s
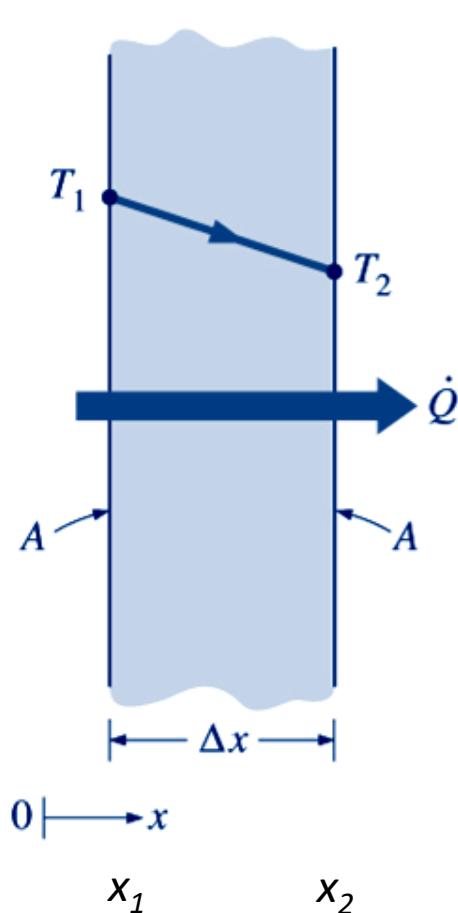
Cross section of part of a wall (almost flat)

Heat transfer rate : $\dot{Q}$

- **Steady state condition-1D**
- Proportional to area *A*
- Proportional to temp. Difference : $T_2 - T_1$
- Inversely proportional to thickness: $\Delta x = x_2 - x_1$
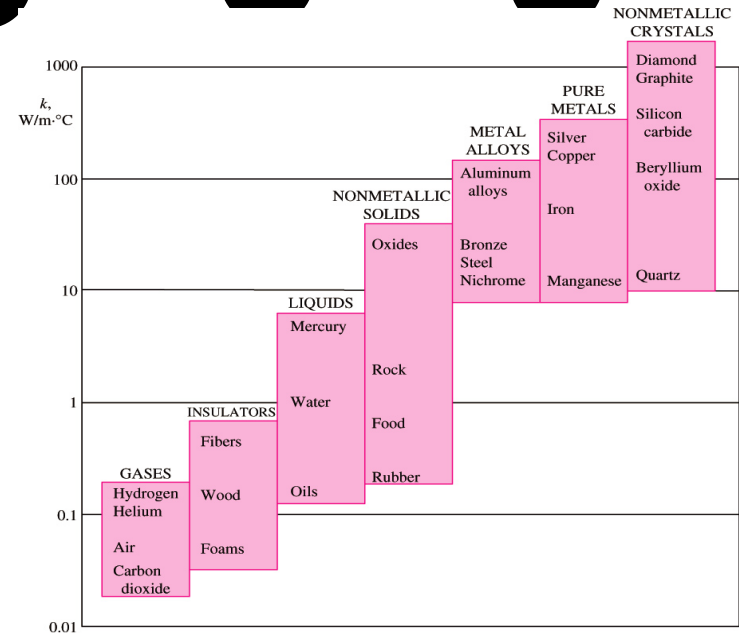- Dependent on material → thermal conductivity: *k*

$$\dot{Q} = -k\,A\,\frac{T_2 - T_1}{x_2 - x_1} = -k\,A\,\frac{\Delta T}{\Delta x} \quad (\text{W})$$

Negative sign since heat flow is positive at a negative temperature gradient.

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Unit: $\dfrac{\mathrm{W}}{\mathrm{m} \cdot \mathrm{K}}$

$\Rightarrow$ The amount of power conducted through 1 m of material at a 1 K temperature difference

Heat transfer rate: $\dot{Q} = -k\, A \dfrac{\Delta T}{\Delta x}$ (W)

Heat flux: $\dot{q} = -k \dfrac{\Delta T}{\Delta x}$ (W / m²)

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Heat conduction through plane wall

Fourier's law:

$$\dot{Q} = -k\,A\frac{T_2 - T_1}{\Delta x}$$

$$= +k\,A\frac{T_1 - T_2}{\Delta x}$$

$$= \frac{T_1 - T_2}{\dfrac{\Delta x}{k\,A}}$$

$$\dot{Q} = \frac{T_1 - T_2}{R_{cond}} \text{ with } R_{cond} = \frac{\Delta x}{kA} \quad (\text{K/W})$$

Heat transfer in "Ohmic way"!

Electrical resistance

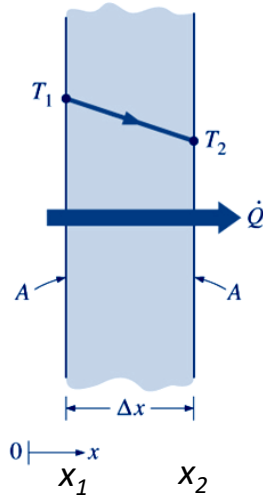Ohm's law:

$$I = \frac{V_1 - V_2}{R}$$

$$V_1 - V_2 \quad \leftrightarrow \quad T_1 - T_2$$

$$I \quad\quad \leftrightarrow \quad \dot{Q}$$
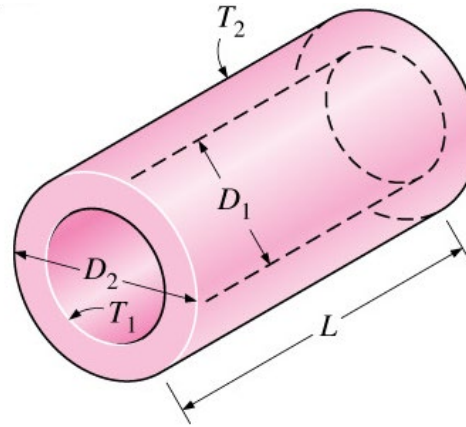
$$R \quad\quad \leftrightarrow \quad R_{cond}$$

$$\Rightarrow \dot{Q} = \frac{T_1 - T_2}{R_{cond}}$$

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Plane wall

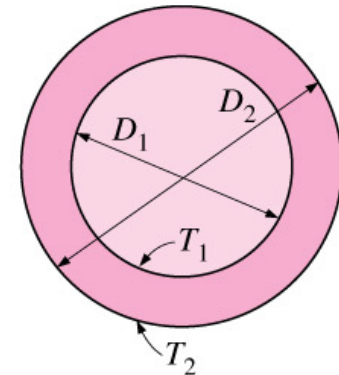Cilindrical pipe

Spherical shell

$$R = \frac{\Delta x}{kA}$$

$$R = \frac{\ln(\frac{D_2}{D_1})}{2\pi L k}$$

$$R = \frac{D_2 - D_1}{2\pi k D_1 D_2}$$
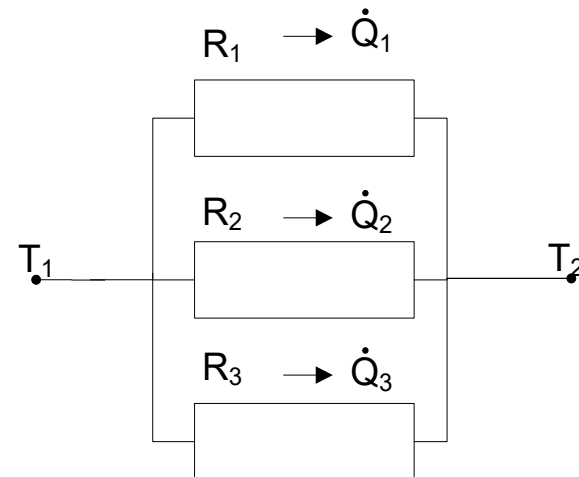
$$\dot{Q} = \frac{T_1 - T_2}{R}$$

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Series Resistors
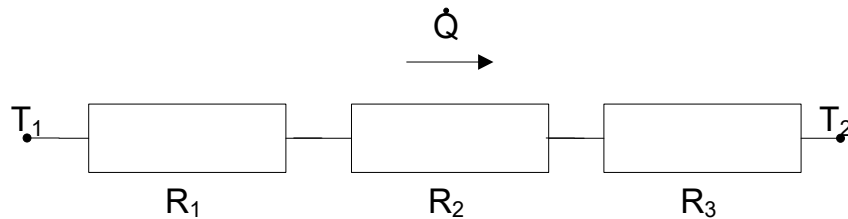
$$R_{tot} = \sum_i R_i$$

(Add Resistors)

Parallel Resistors
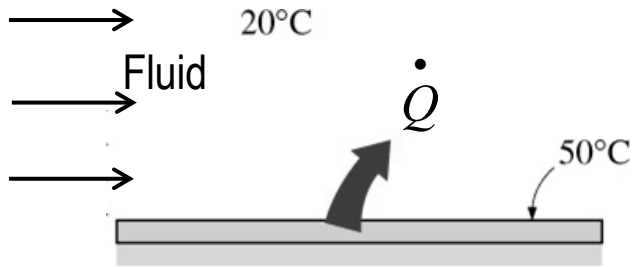

$$\frac{1}{R_{tot}} = \sum_i \frac{1}{R_i}$$

(Add Heat Flows)

20°C

Fluid

$\dot{Q}$

50°C

In the case of Convection

$\dot{Q}$

Depends on :

1) Temperature Difference

2) convection heat transfer coefficient

3) Surface area of the object

Newton's Law:

$$\dot{Q} = h.A.\Delta T (\mathbf{W})$$

h is the **"convection heat transfer coefficient"** which basically takes care of various effects of fluid properties and flow properties

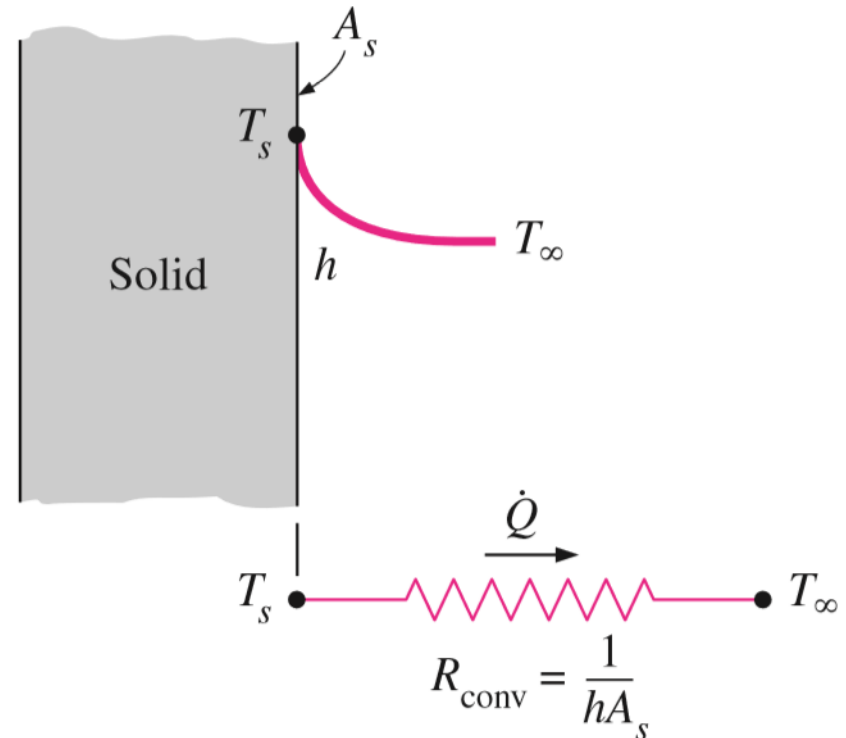*Unit:* $\dfrac{\mathbf{W}}{\mathbf{m^2 \cdot K}}$

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$$\dot{Q} = hA\Delta T = \frac{\Delta T}{\frac{1}{hA}} \text{ with } \Delta T = T_S - T_\infty$$

$$\Rightarrow \dot{Q} = \frac{\Delta T}{R_{conv}}$$

Where **convection resistance**:

$$R_{conv} = \frac{1}{hA} \left(\frac{K}{W}\right)$$

Remember
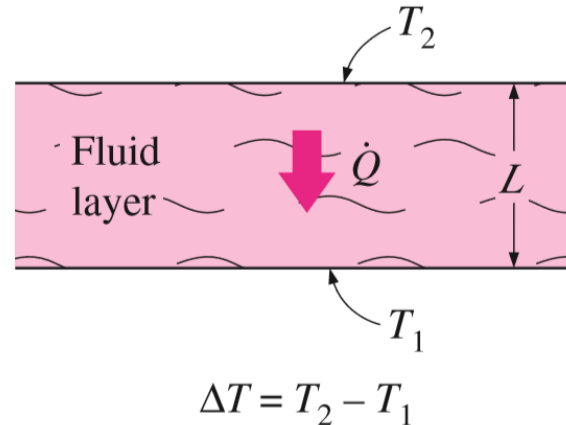$$R_{Cond}, plane = \frac{\Delta x}{kA} \quad \left(\frac{K}{W}\right)$$

$$\dot{q}_{conv} = h\Delta T$$

$$\dot{q}_{cond} = k\frac{\Delta T}{L}$$

$$\Delta T = T_2 - T_1$$

Taking their ratio gives

$$\frac{\dot{q}_{conv}}{\dot{q}_{cond}} = \frac{h\Delta T}{k\Delta T/L} = \frac{hL}{k} = \text{Nu}$$

- **The larger the Nusselt number, the more effective the convection.**
- **A Nusselt number of Nu=1 for a fluid layer represents heat transfer across the layer by pure conduction.**

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General (also natural convection):

$$\dot{Q} = h A \Delta T \quad (\text{W})$$ Newton's cooling law $$\dot{q} = h \Delta T \quad (\text{W/m}^2)$$

"Supporting" equations for *h (Forced Convection)*:

$$\text{Nu} = a \cdot \text{Re}^b \text{Pr}^c$$ *a, b, c* dependent on geometry and flow regime (laminar / turbulent)

Nusselt Number $$\text{Nu}_L = \frac{hL}{k}; \quad \text{Nu}_D = \frac{hD}{k} \quad (-)$$

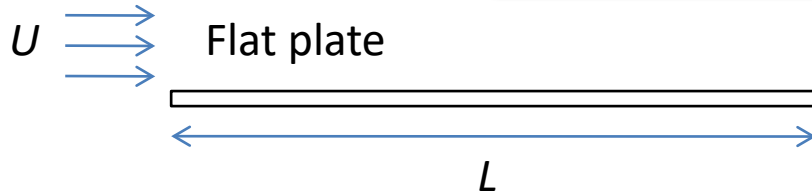Reynolds Number $$\text{Re}_L = \frac{\rho U L}{\mu}; \quad \text{Re}_D = \frac{\rho U D}{\mu} \quad (-)$$

Prandtl Number $$\text{Pr} = \frac{\mu c_p}{k}$$

Dimensionless numbers make similar shaped situations comparable; "universal" parameters

External flow $\quad \mathrm{Nu} = a \cdot \mathrm{Re}^b \mathrm{Pr}^c$

U Flat plate

$a = 0{,}664$; $b = 0{,}5$; $c = 1/3$ (Re $< 5 \cdot 10^5$)
$a = 0{,}037$; $b = 0{,}8$; $c = 1/3$ (Re $> 5 \cdot 10^5$)

$$\mathrm{Re} = \frac{\rho U L}{\mu}$$

L
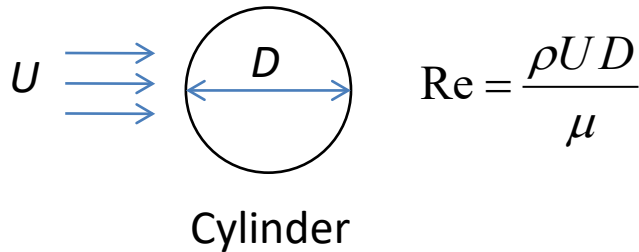
U $\quad D \quad$ $\mathrm{Re} = \dfrac{\rho U D}{\mu}$

$a = 0{,}193$; $b = 0{,}618$; $c = 1/3$ $\quad$ (4000 $<$ Re $<$ 40.000)
$a = 0{,}027$; $b = 0{,}805$; $c = 1/3$ $\quad$ (40.000 $<$ Re $<$ 400.000)

Cylinder

$$\mathrm{Nu_{cyl}} = \frac{hD}{k} = 0.3 + \frac{0.62\,\mathrm{Re}^{1/2}\,\mathrm{Pr}^{1/3}}{[1 + (0.4/\mathrm{Pr})^{2/3}]^{1/4}}\left[1 + \left(\frac{\mathrm{Re}}{282{,}000}\right)^{5/8}\right]^{4/5}$$
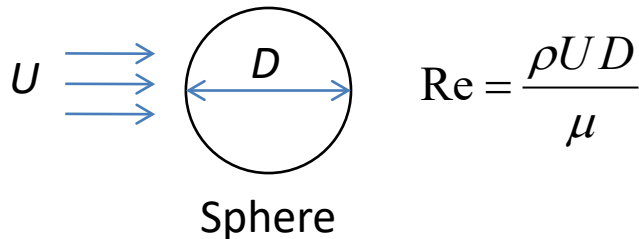
U $\quad D \quad$ $\mathrm{Re} = \dfrac{\rho U D}{\mu}$
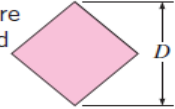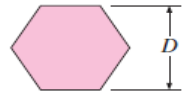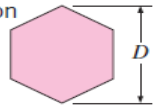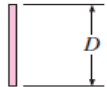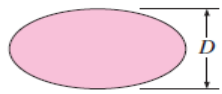
$\mathrm{Nu} \approx 2 + [0{,}4\mathrm{Re}^{1/2} + 0{,}06\mathrm{Re}^{2/3}]\,\mathrm{Pr}^{0{,}4}$
(optimal for Re $<$ 80.000)

Sphere

TABLE 7-1

Empirical correlations for the average Nusselt number for forced convection over circular and noncircular cylinders in cross flow (from Zukauskas, Ref. 14, and Jakob, Ref. 6)

Cross-section of the cylinder	Fluid	Range of Re	Nusselt number
Circle	Gas or liquid	0.4–4 4–40 40–4000 4000–40,000 40,000–400,000	$Nu = 0.989Re^{0.330} Pr^{1/3}$ $Nu = 0.911Re^{0.385} Pr^{1/3}$ $Nu = 0.683Re^{0.466} Pr^{1/3}$ $Nu = 0.193Re^{0.618} Pr^{1/3}$ $Nu = 0.027Re^{0.805} Pr^{1/3}$
Square	Gas	5000–100,000	$Nu = 0.102Re^{0.675} Pr^{1/3}$
Square (tilted 45°)	Gas	5000–100,000	$Nu = 0.246Re^{0.588} Pr^{1/3}$
Hexagon	Gas	5000–100,000	$Nu = 0.153Re^{0.638} Pr^{1/3}$
Hexagon (tilted 45°)	Gas	5000–19,500 19,500–100,000	$Nu = 0.160Re^{0.638} Pr^{1/3}$ $Nu = 0.0385Re^{0.782} Pr^{1/3}$
Vertical plate	Gas	4000–15,000	$Nu = 0.228Re^{0.731} Pr^{1/3}$
Ellipse	Gas	2500–15,000	$Nu = 0.248Re^{0.612} Pr^{1/3}$

If $\dot{Q}$ must be found:

- Calculate at film temperature : $T_f = \dfrac{T_s + T_\infty}{2}$
- Pull out ingredients like μ, ρ, k, Pr from tables – like assignment bundle: air or given fluid) at $T_f = \dfrac{T_s + T_\infty}{2}$

- Calculate Re and choose approiprate correlation based on geometry and Re
- Calculate Nu
- Derive h from it
- Fill out Newton's cooling

law: $\dot{Q} = hA\Delta T$

Natural convection

Forced convection: velocity U imposed

Natural convection: velocity follows from temp.difference T_s - T_∞ *

$\Rightarrow$ Alternative for Reynolds number, with temp.diff. instead U:

Grashof number $\mathrm{Gr} = \dfrac{g\beta(T_s - T_\infty)L_c^{\,3}}{\nu^2}$ $(-)$

Greek letter "nu"

- Gravitational acceleration g = 9,81 m/s²

- Volume expansion coefficient β (K⁻¹);   most gases: $\beta = \dfrac{2}{T_s + T_\infty}$
  (temperature in **Kelvin**; 0 ˚C = 273,15 K)

- Length L_c characteristic for geometry (length L for
  plate, diameter D for sphere/cylinder)

- Kinematic viscosity $\nu = \dfrac{\mu}{\rho}$ $(\mathrm{m}^2/\mathrm{s})$

- at average temperature $T_f = \dfrac{T_s + T_\infty}{2}$

*) Choose ΔT positive for convenience

$$\text{Rayleigh Number} : \text{Ra} = \text{Gr} \cdot \text{Pr} = \frac{g\beta(T_s - T_\infty)L_c^{\,3}}{v^2}\text{Pr}$$

Rayleigh number, which is the product of the Grashof and Prandtl numbers

Constant
coefficient

Constant
exponent

$$Nu = C\,Ra_L^n$$

Nusselt   Rayleigh
number   number
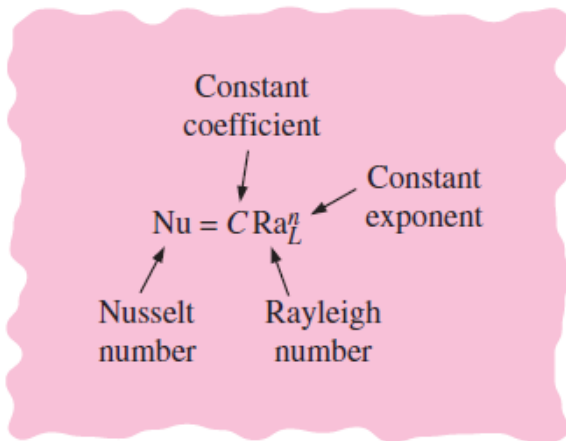
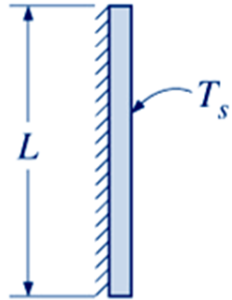**The values of the constants C and n depend on the**
✓ geometry of the surface
✓ the flow regime

All fluid properties are to be evaluated at the film temperature Tf =$(T_s+T_{inv})/2$.

FREE CONVECTION CORRELATIONS
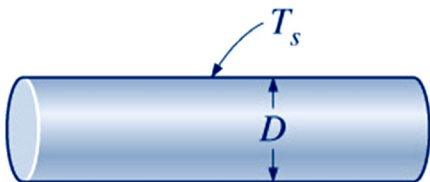
Vertical, flat plate

$Nu = 0{,}59\ Ra_L^{1/4}$ with $L_c = L$ $(10^4 < Ra_L < 10^9)$

$Nu = 0{,}1\ Ra_L^{1/3}$ with $L_c = L$ $(10^{10} < Ra_L < 10^{13})$

Entire range $\Longrightarrow$ $Nu = \left\{ 0.825 + \dfrac{0.387 Ra_L^{1/6}}{[1 + (0.492/Pr)^{9/16}]^{8/27}} \right\}^2$
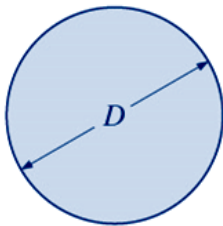
(complex but more accurate)

Horizontal cylinder (e.g. pipe)

$Nu = \left\{ 0.6 + \dfrac{0.387 Ra_D^{1/6}}{[1 + (0.559/Pr)^{9/16}]^{8/27}} \right\}^2$

Sphere (e.g. light bulb)

$Nu = 2 + \dfrac{0.589 Ra_D^{1/4}}{[1 + (0.469/Pr)^{9/16}]^{4/9}}$

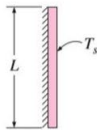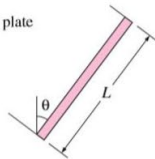**Notation:**
Ra_L is Ra with $L_c = L$;
Ra_D is Ra with $L_c = D$

Table 9-1, Cenegel

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TABLE 9–1

Empirical correlations for the average Nusselt number for natural convection over surfaces

| Geometry	Characteristic length L_c	Range of Ra	Nu	
Vertical plate	L	10^4–10^9	$Nu = 0.59 Ra_L^{1/4}$	(9-19)
		10^9–10^{13}	$Nu = 0.1 Ra_L^{1/3}$	(9-20)
		Entire range	$Nu = \left\{ 0.825 + \dfrac{0.387 Ra_L^{1/6}}{[1 + (0.492/Pr)^{9/16}]^{8/27}} \right\}^2$ (complex but more accurate)	(9-21)
Inclined plate	L		Use vertical plate equations for the upper surface of a cold plate and the lower surface of a hot plate Replace g by $g \cos\theta$ for $Ra < 10^9$	
Horizontal plate (Surface area A and perimeter p) (a) Upper surface of a hot plate (or lower surface of a cold plate)	A_s/p	10^4–10^7	$Nu = 0.54 Ra_L^{1/4}$	(9-22)
		10^7–10^{11}	$Nu = 0.15 Ra_L^{1/3}$	(9-23)
(b) Lower surface of a hot plate (or upper surface of a cold plate)		10^5–10^{11}	$Nu = 0.27 Ra_L^{1/4}$	(9-24)
Vertical cylinder	L		A vertical cylinder can be treated as a vertical plate when $D \geq \dfrac{35L}{Gr_L^{1/4}}$	
Horizontal cylinder	D	$Ra_D \leq 10^{12}$	$Nu = \left\{ 0.6 + \dfrac{0.387 Ra_D^{1/6}}{[1 + (0.559/Pr)^{9/16}]^{8/27}} \right\}^2$	(9-25)
Sphere	D	$Ra_D \leq 10^{11}$ ($Pr \geq 0.7$)	$Nu = 2 + \dfrac{0.589 Ra_D^{1/4}}{[1 + (0.469/Pr)^{9/16}]^{4/9}}$	(9-26)

Step-by-step plan natural convection

If $\dot{Q}$ must be found:

- Determine ingredients necessary for dimensionless no.
  (Pr, *k* and *v* at average temperature $\frac{T_s + T_\infty}{2}$)
- Determine Ra: $\mathrm{Ra} = \mathrm{Gr} \cdot \mathrm{Pr} = \dfrac{g\beta(T_s - T_\infty)L_c^{\,3}}{v^2}\mathrm{Pr}$
- Choose appropriate correlation based on geometry and Ra
- Determine Nu
- Resolve *h* from it
- Fill out Newton's cooling law: $\dot{Q} = hA\Delta T$

In the figure below a cross section of the wall of a house is given. The inside wall of the house has a temperature $T_i = 20\,°C$. The temperature of the air outside is $T_\infty = -5\,°C$. The wall is 2 m wide, and has heights $h_a = 1.50$ m, $h_b = 1$ m, and thicknesses t_a=30 mm and t_b=50 mm. The thermal conductivity of the layers of brick is $k_1 = k_4 = 0.72\,\mathrm{W\,m^{-1}\,K^{-1}}$. The thermal conductivity of the rockwool layer at 2 is $k_2 = 0.0350\,\mathrm{W\,m^{-1}\,K^{-1}}$, and that of the wooden layer at 3 is $k_3 = 0.0550\,\mathrm{W\,m^{-1}\,K^{-1}}$. We will analyse this problem on a day with no wind.

In this exercise, we want to know what the heat loss through the wall is. When writing down your answers, please keep in mind the following:

- Clearly state the assumptions you make.
- Clearly mention the values and their units.
- Clearly mention the relations you use and why you are using them.
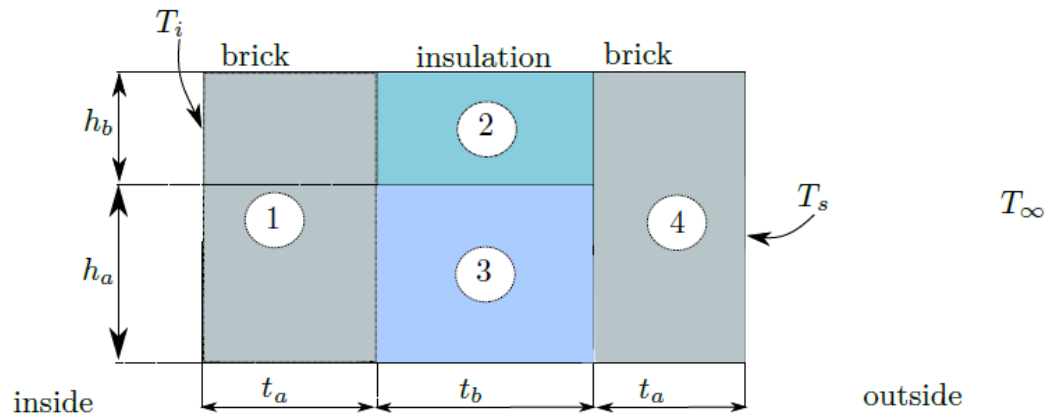
a) Draw the resistance network of the wall and determine the formula for the total resistance R_{tot} inside of the wall.
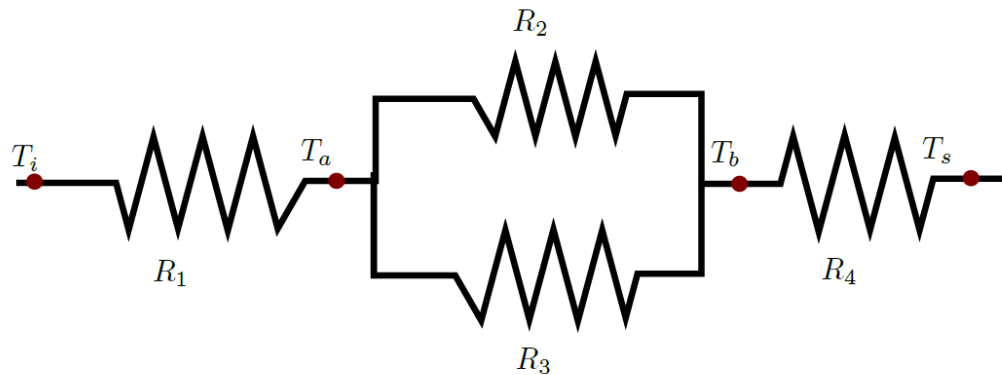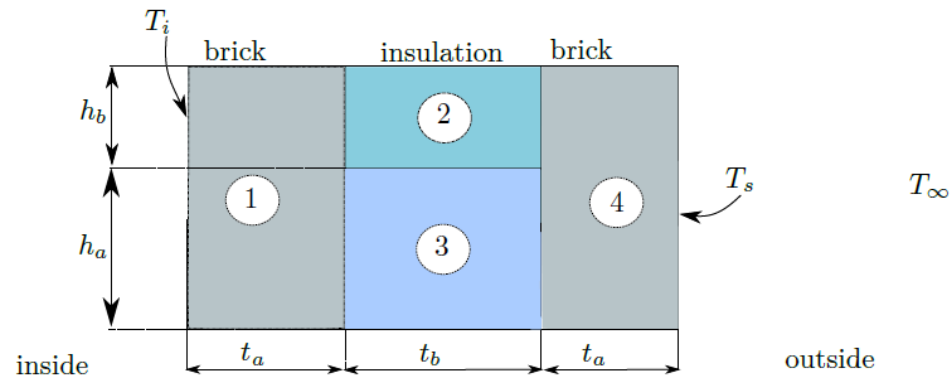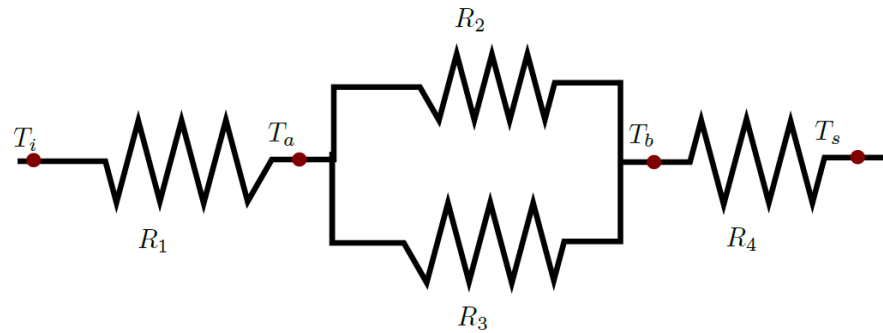
The equations for the resistances are:

$$R_1 = \frac{t_a}{k_1 A_1} \qquad R_2 = \frac{t_b}{k_2 A_2} \qquad R_3 = \frac{t_b}{k_3 A_3} \qquad R_4 = \frac{t_a}{k4 A_4} \tag{1.1}$$

The resistance of the parallel part can be determined using

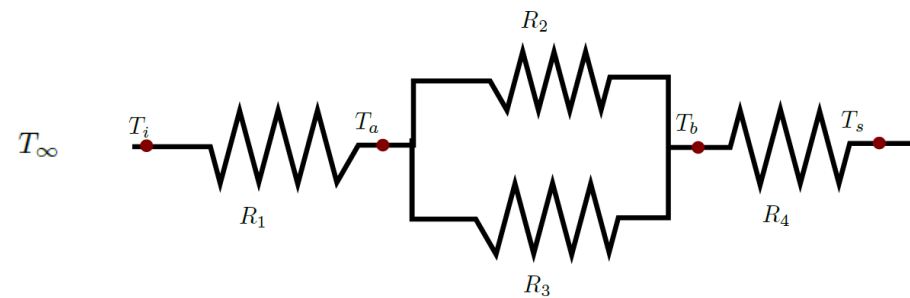$$\frac{1}{R_{23}} = \frac{1}{R_2} + \frac{1}{R_3} \implies R_{23} = \frac{R_2 R_3}{R_2 + R_3} \tag{1.2}$$

The total resistance can then be determined using:

$$R_{tot} = R_1 + R_{23} + R_4 \tag{1.3}$$

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b) Determine the total heat loss from the wall. If you need to iterate, do so until a precision of Q of 5 percent is achieved. What is the surface temperature of the wall?
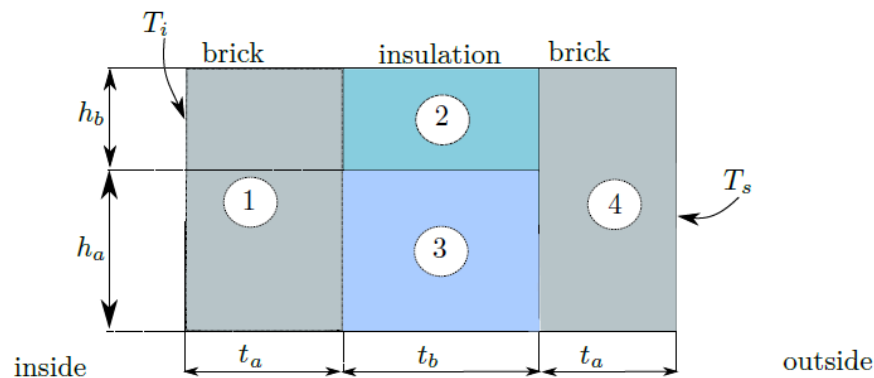
In order to solve this, we need the following energy balance:

$$Q_{conduction} = Q_{naturalconvection} \tag{1.4}$$

Correct energy balance - 2 pt.

Problem: T_s unknown → starting values for step-by-step plan unknown (Pr, k, v, …)

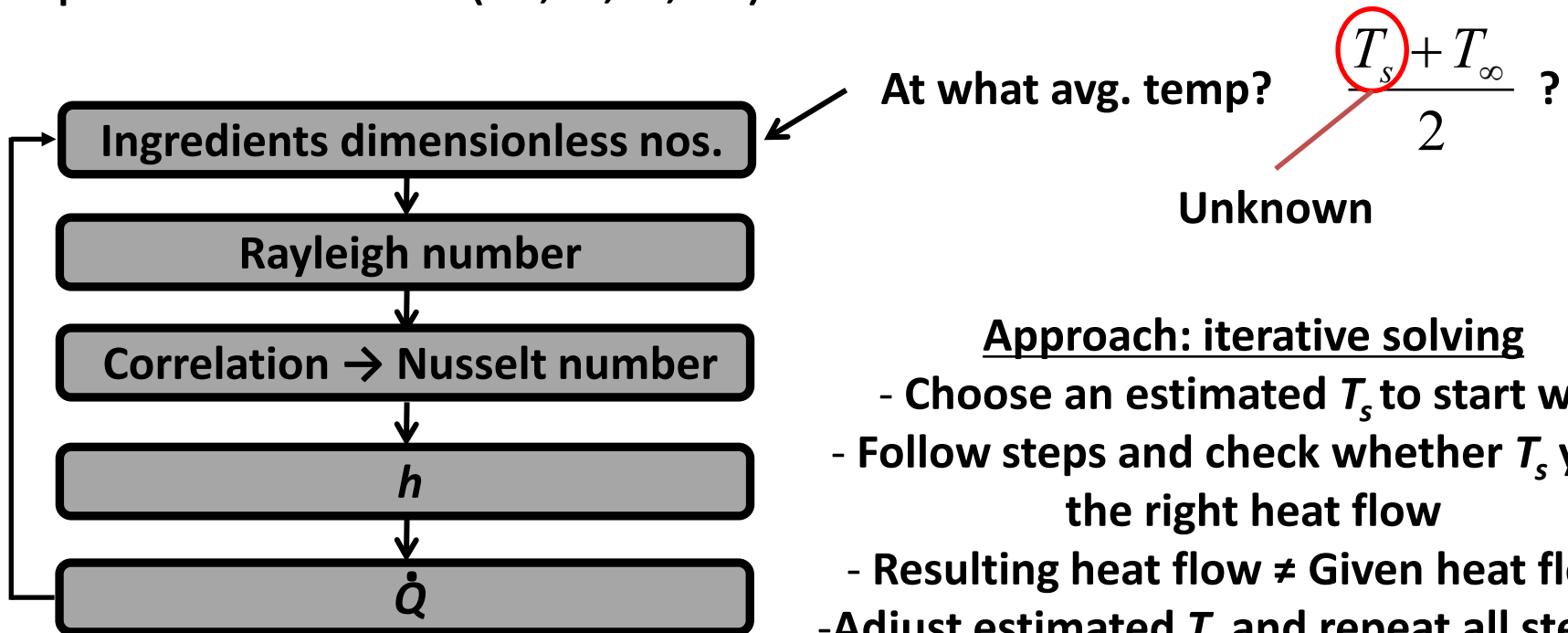At what avg. temp? $\dfrac{T_s + T_\infty}{2}$?

Unknown

Ingredients dimensionless nos.

Rayleigh number

Correlation → Nusselt number

h

$\dot{Q}$

Approach: iterative solving
- Choose an estimated T_s to start with
- Follow steps and check whether T_s yields the right heat flow
- Resulting heat flow ≠ Given heat flow ?
-Adjust estimated T_s and repeat all steps till you find a T_s that (approximately) yields Given heat flow

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c) Determine the temperatures after each layer and draw the temperature profile including the outside.
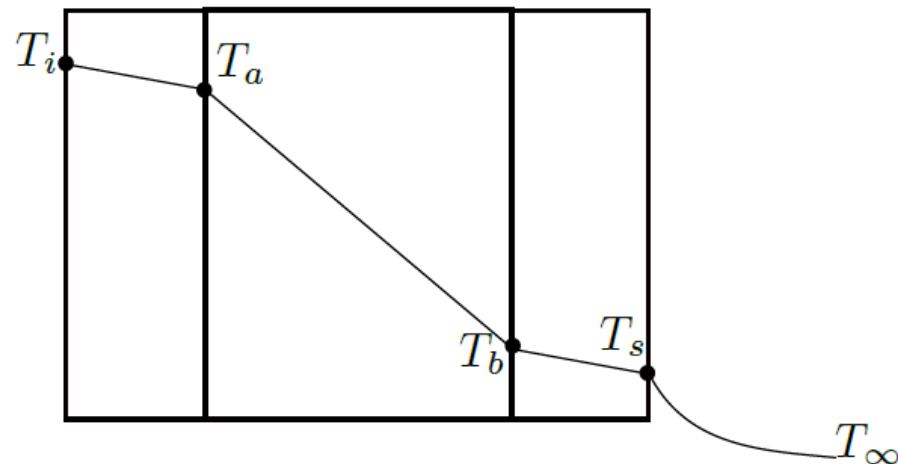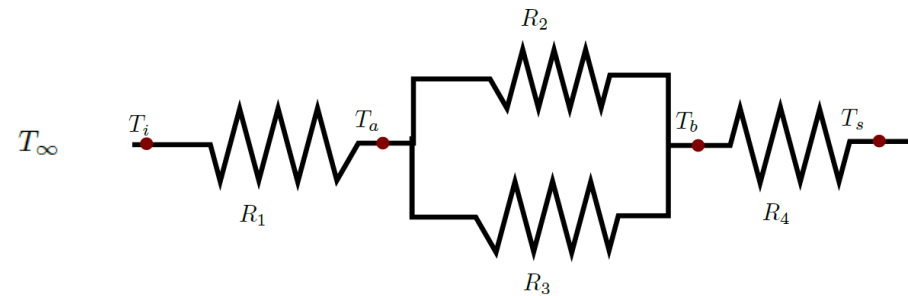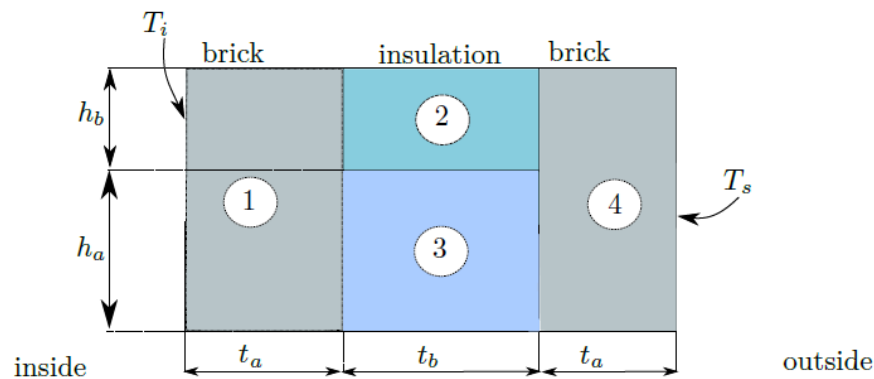
Assume the house has a total circumference of 40 meters, where the walls are made from the material above and has the exact same properties as above. With the current national gas shortage, the price of a cubic meter of gas is €0.79, and consists of 32 MJ m^{-3}. During the entire month of December, the average temperature outside is -5°C.
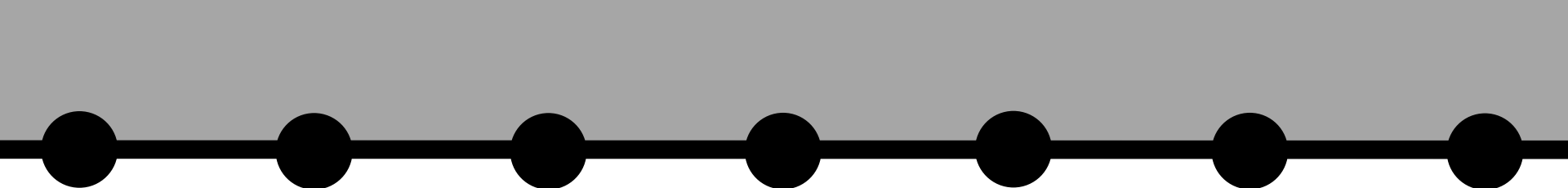
d) How much energy does the house lose in the month December and how much does it cost to compensate this. You may assume an ideal energy conversion.

Deduction of points

| Standard: |
| --- |
| Missing/wrong units: -0.5pt per time, -3 pt. total |
| Missing/wrong assumptions: -0.5pt per time, -3 pt. total |
| Missing/wrong conclusion/discussion: -0.5pt per time, -3 pt. total |
| Missing/wrong description: -0.5pt per time, -3 pt. total |

CONCEPT OF RADIATION

♣ Consider a hot object that is suspended in an evacuated chamber whose walls are at room temperature.

♣ The Hot object will cool down and reach equilibrium with the chamber walls due to change in **internal energy**.

Vacuum chamber

Hot object
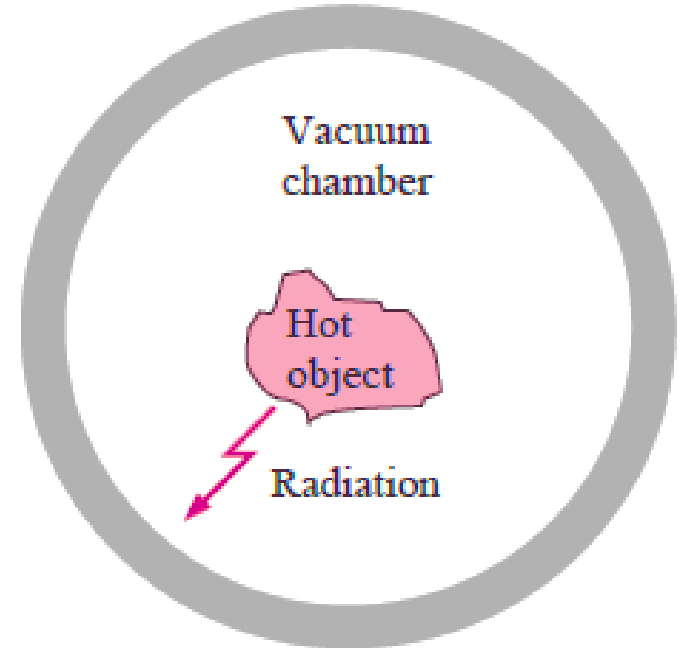
Radiation

But How is this heat (internal energy) being transported without any medium ? ?

Electromagnetic Waves or Electromagnetic Radiation
(no medium required!)

Heat transfer in the form of electromagnetic waves due to change in internal energy.

Stefan-Boltzmann law :

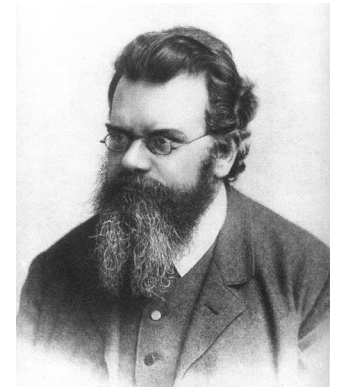❖ **Estimates the Total power emitted by blackbody**

$$\dot{Q}_b = \sigma \cdot A \cdot T^4 \ [W]$$

Josef Stefan

$$\sigma = 5.670 \times 10^{-8} \ \frac{\mathrm{W}}{\mathrm{m}^2 \mathrm{K}^4}$$ **Stefan-Boltzmann constant**

Ludwig Boltzmann

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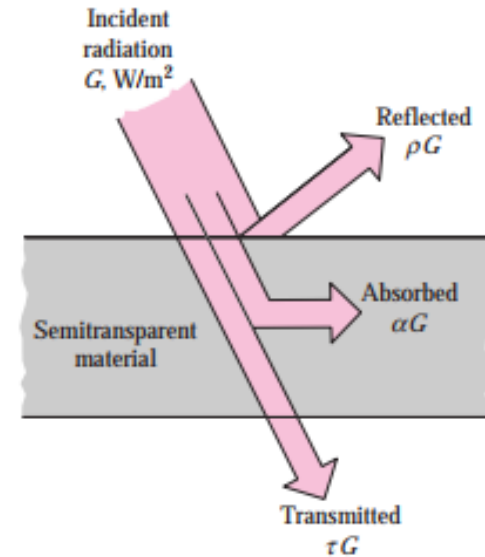
$Absorptivity: \alpha = \dfrac{\text{Absorbed radiation}}{\text{Incident radiation}}$

$Reflectivity: \rho = \dfrac{\text{Reflected radiation}}{\text{Incident radiation}}$

$Transmissivity: \tau = \dfrac{\text{Transmitted radiation}}{\text{Incident radiation}}$

How are these properties related ?

$$\alpha \; + \; \tau \; + \rho = 1$$

EMISSIVITY

- **In reality all materials: non ideal emitter**
- **Less power emitted than blackbody**

Adjusted Stefan-Boltzmann Law for emitted radiation from object:

$$\dot{Q}_b = \sigma \cdot A \cdot T^4 \ (W) \ \textbf{so} \ \ \dot{Q} = \varepsilon \cdot \sigma \cdot A \cdot T^4 (W)$$

Emissivity ε (0 ≤ ε ≤ 1) of a surface (material property):

Measure for emitted radiation ($\dot{Q}$) in comparison to radiation $(\dot{Q}_b)$ emitted by a blackbody (ε = 1) at same temperature.

$$\varepsilon = \frac{\dot{Q}}{\dot{Q}_b}$$

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Radiation Heat Transfer

To calculate the heat transfer rate by radiation, we must include terms for power output and energy received from the surroundings.

$$\varepsilon\sigma A T_s^4$$

$$\sigma\alpha A T_\infty^4$$

Making the usual assumption that $\varepsilon = \alpha$, and multiplying by area yields:

$$\dot{Q} = \varepsilon\sigma A(T_s^4 - T_\infty^4)$$

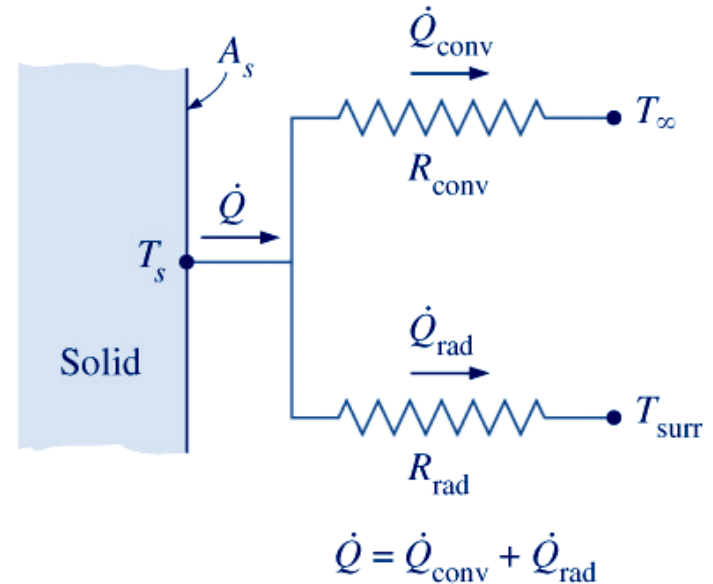**This is the expression for an object totally enclosed by surroundings at T_∞.**

$$\dot{Q} = \dot{Q}_{conv} + \dot{Q}_{rad}$$

$$= h_{conv} A \Delta T + h_{rad} A \Delta T$$

$$= (h_{conv} + h_{rad}) A \Delta T$$

$$\Rightarrow \dot{Q} = h_{tot} A \Delta T \quad \text{with} \quad h_{tot} = h_{conv} + h_{rad}$$
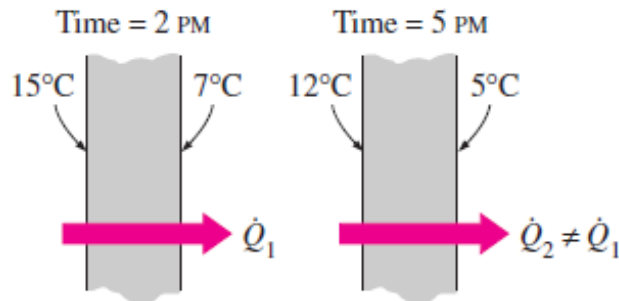

$$\dot{Q} = \dot{Q}_{conv} + \dot{Q}_{rad}$$

$$h_{rad} = \varepsilon \cdot \sigma \cdot (T_s^2 + T_\infty^2) \cdot (T_s + T_\infty)$$

$$\text{with} \quad R_{rad} = \frac{1}{h_{rad} A} \quad \Longrightarrow \quad \frac{1}{R_{tot}} = \frac{1}{R_{conv}} + \frac{1}{R_{rad}}$$

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Time = 2 PM Time = 5 PM

15°C 7°C 15°C 7°C

$\dot{Q}_1$ $\dot{Q}_2 = \dot{Q}_1$

Steady State

Time = 2 PM Time = 5 PM

15°C 7°C 12°C 5°C

$\dot{Q}_1$ $\dot{Q}_2 \neq \dot{Q}_1$
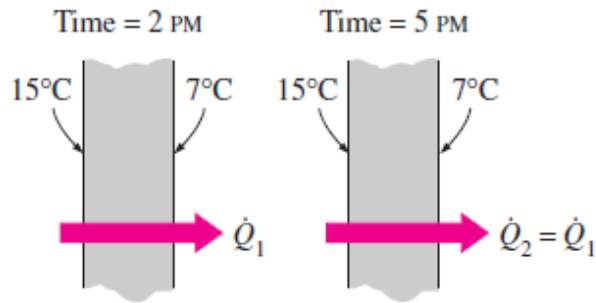
**Unsteady State**

- In **steady state heat transfer**, the temperature at any particular point in the system remains constant after equilibrium is attained.

- The amount of heat entering any section is then equal to the amount of heat exiting the section, because the driving force (temperature difference) is constant.

- In **unsteady state**, the temperature within an object itself keeps changing with time.

- The heat entering a section thus might not be the same as the heat exiting the section, as the temperature difference across the section keeps changing with time.

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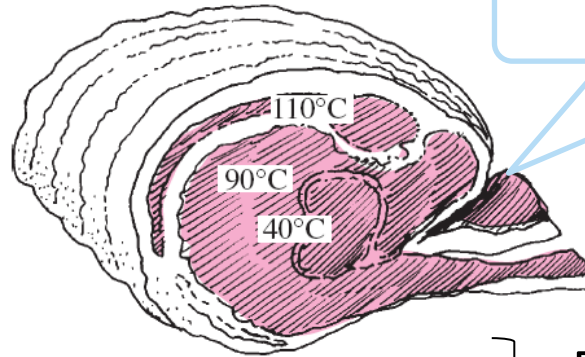
Temperature is a function of time and space

110°C

90°C

40°C

Convection: heat from outer layer
Conduction: heat transferred from outer layer to core

Factors:
$h, k,$
geometry

➤ **Interior temperature** of some bodies remains essentially uniform at all times during a heat transfer process.

➤ The **temperature** of such bodies can be taken to be a function of time only, $T(t)$.

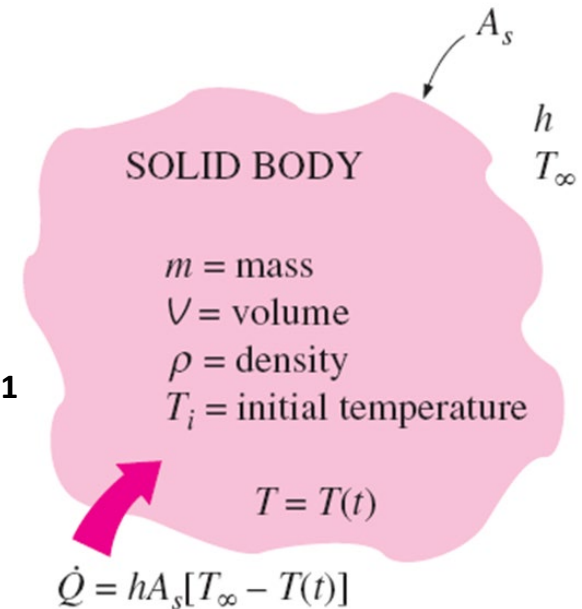➤ Heat transfer analysis that utilizes this idealization is known as **lumped system analysis**.

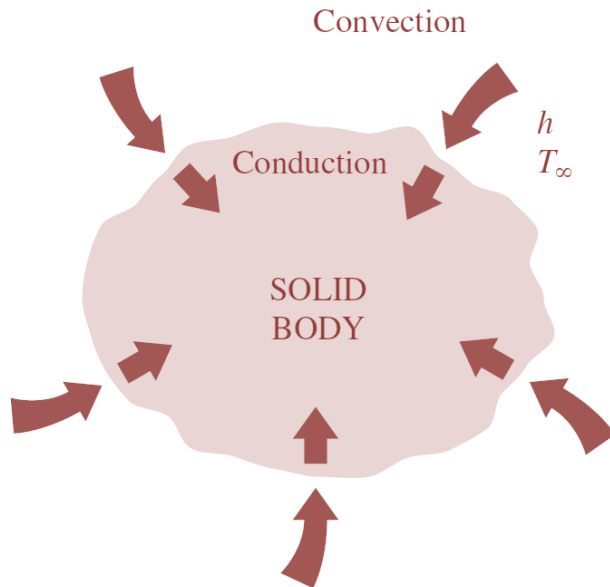Temperature is only function of Time

70°C

70°C   70°C

70°C   70°C

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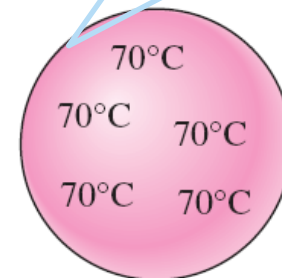
$$\frac{T(t) - T_\infty}{T_i - T_\infty} = e^{-bt}$$

Where $b = \dfrac{hA_s}{\rho \vee c_p}$

- h *:* heat transfer coefficient around object (W · m⁻² · K⁻¹
- A_s: external surface area (m²)
- ρ *:* density of object (kg · m⁻³)
- c_p : specific heat of object (J · kg⁻¹ · K⁻¹)
- V : volume of object, (m³)

SOLID BODY

A_s

h
T_∞

m = mass
V = volume
ρ = density
T_i = initial temperature

$T = T(t)$

$\dot{Q} = hA_s[T_\infty - T(t)]$

Only for lumped system analysis

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Convection

Conduction

h
T_∞

SOLID
BODY

$$Bi = \frac{\text{heat convection}}{\text{heat conduction}}$$

$$Bi = \frac{h}{k/L_c} \frac{\Delta T}{\Delta T} = \frac{\text{Convection at the surface of the body}}{\text{Conduction within the body}}$$

or

$$Bi = \frac{L_c/k}{1/h} = \frac{\text{Conduction resistance within the body}}{\text{Convection resistance at the surface of the body}}$$

$$Bi = \frac{hL_c}{k}$$ **When characteristic length is :** $$L_c = \frac{V}{A_s}$$

almost uniform temperature for Bi ≤ 0,1
"lumped system"

Nusselt number

$$\text{Nu} = \frac{hL_c}{k}$$

$\neq$

Biot number

$$\text{Bi} = \frac{hL_c}{k}$$

Dimensionless measure for convection so increase of heat transfer due to flow
k of fluid!

Dimensionless measure for degree of temperature distribution within body
k of surrounded object!

Definitions seem similar but are substantially different!

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Thank you for your attention