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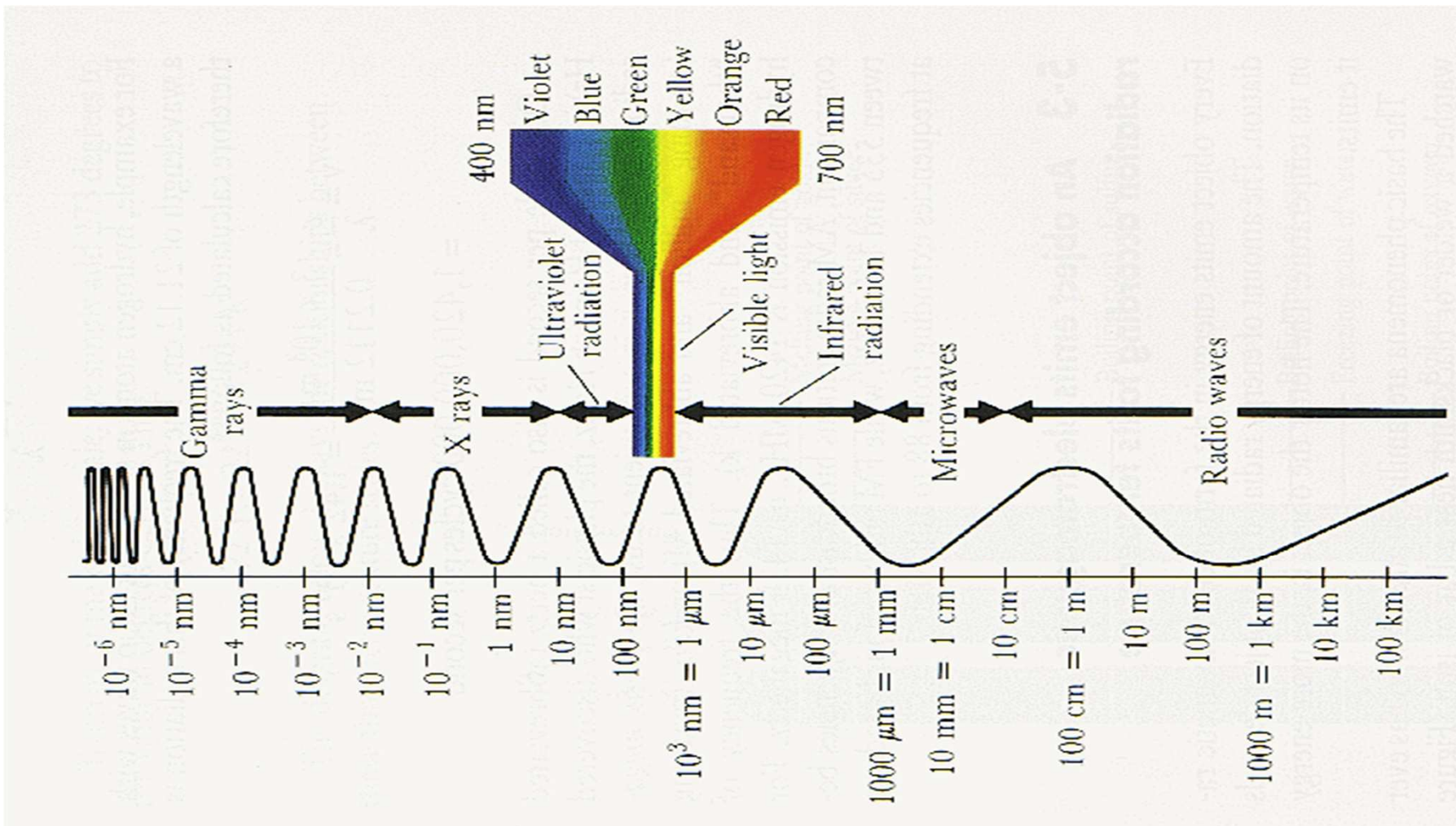
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Radiation



Learning outcomes:

- Explain Weins Law and Planck's law
- Explain emissivity for grey bodies
- Outline transmissivity and opaqueness
- Discuss wavelength dependence of emissivity and effect for greenhouse's



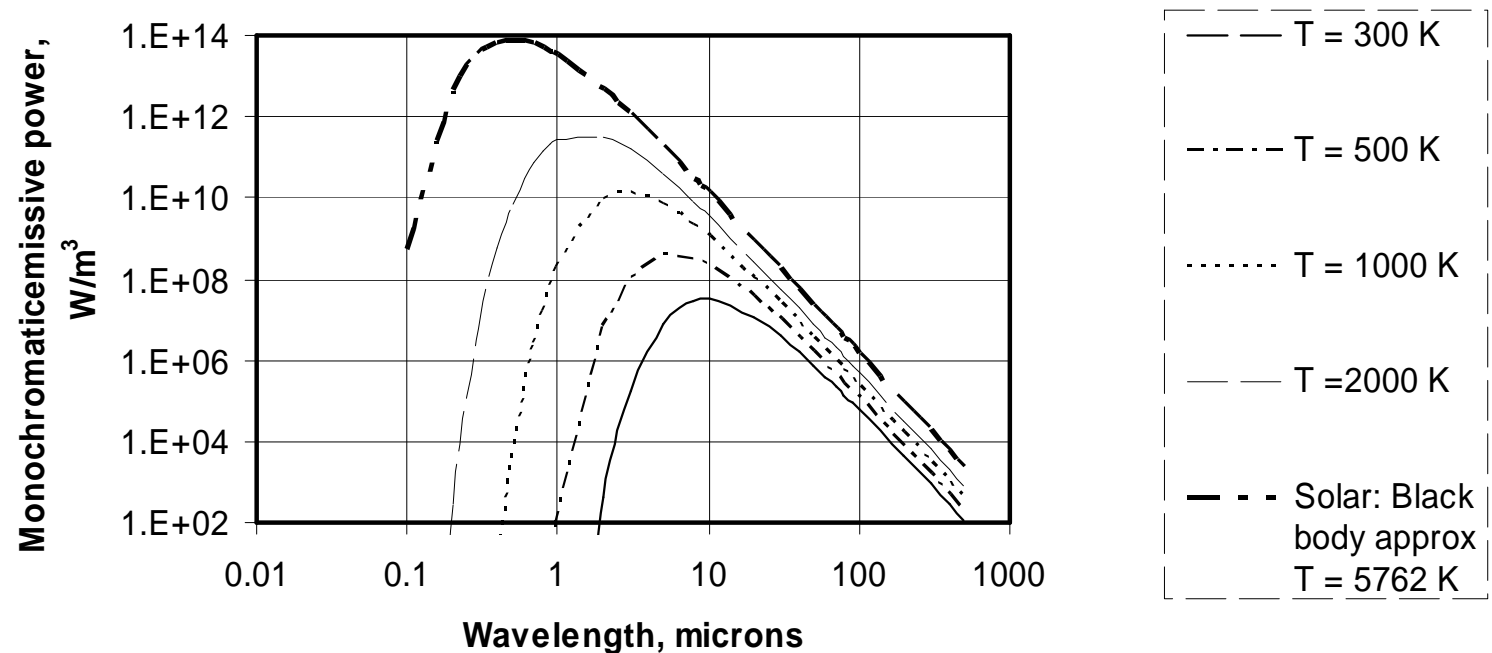
Do not think of visible and IR wavelengths as separate areas. They are linked and the same thing.



- A black body is an ideal emitter
- Energy radiated can be calculated using Stefan-Boltzmann Law

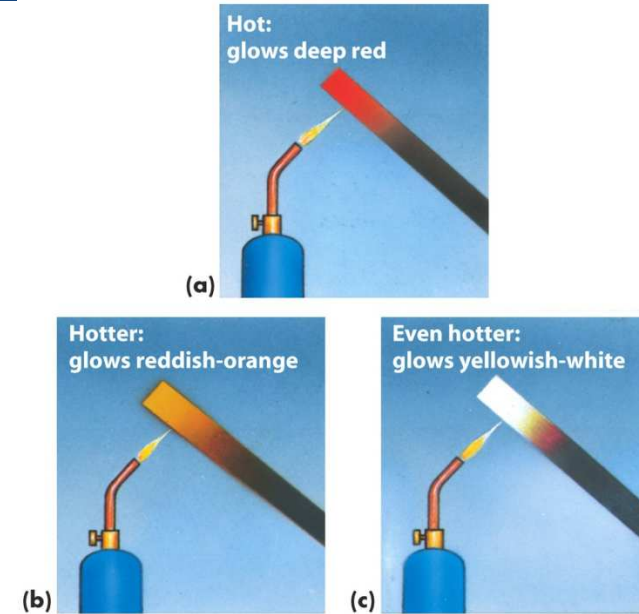
$$\dot{q}'' = \sigma T^4 \quad \sigma = 5.67 \times 10^{-8} \text{ W}/(\text{m}^2 \cdot \text{K}^4)$$

- At any temperature, $E_{b,\lambda}$ has a maximum value at a particular wavelength
- The maximum value increases strongly with temperature: $E_{b,\lambda} |_{\text{max}} \propto T^5$
- As the temperature increases, the maximum occurs at shorter wavelengths
- The visible range extends from about 0.7 microns (red) to 0.4 microns (violet).
- *The peak for solar energy lies within this range.*

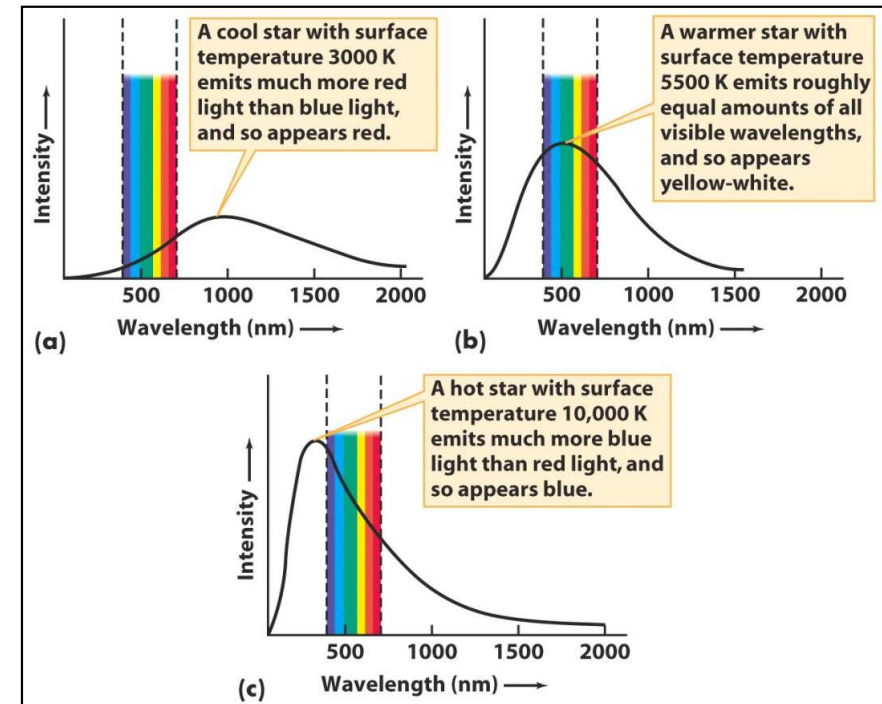




- Increasing temperature decreases wavelength of maximum intensity.
- $\lambda_{\text{max}} T = 2898 \mu\text{m K}$



- Colour of star depends on temperature of surface.
- Cold stars red. Hot stars blue.





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**Emissivity,
absorption
and other
animals.**



Grey bodies, selective emitters and real bodies

- Real surfaces are not black. At any wavelength, the emission from a real body is some fraction, ε_λ , of that of a black body at the same temperature.
- For a *grey body* the monochromatic *emissivity* ε_λ is constant, but most real surfaces have a “favoured” band or bands within which most of their emission occurs -they are, to some extent, *selective emitters*.
- As an engineering approximation, surfaces are often assumed to be grey, and the radiation emitted is calculated using:

$$\dot{q} = \varepsilon \dot{q}_b = A \varepsilon \sigma T^4$$

- ε (< 1) is the emissivity, which properly should be defined for a particular temperature range.
- In general, emissivity varies not only with wavelength and temperature, but also with direction.
- Fortunately, this is usually only important for highly polished surfaces.
- We have seen that increasing temperature causes more energy to be emitted at short wavelengths.
- Since emissivity varies with wavelength, this causes the average emissivity to change.



Transmission, absorption and reflection of radiation

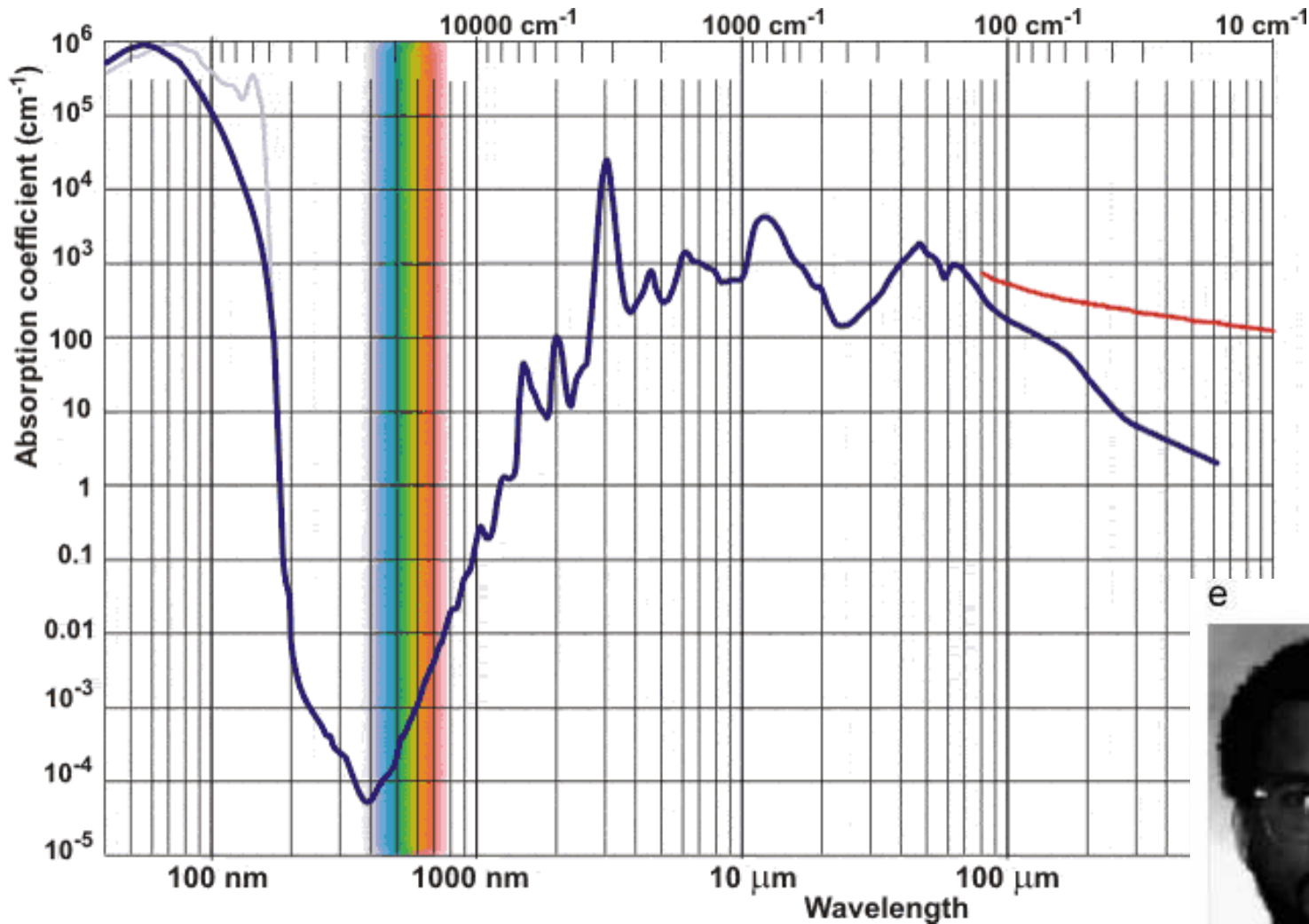
- The fractions of the radiation incident on a surface which are transmitted absorbed and reflected are referred to as the *transmissivity*, τ , the *absorptivity*, α , and the *reflectivity*, ρ .
- Since together they account for all the incident radiation

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

- Opaque surfaces transmit no radiation ($\tau = 0$).
- Reflection may be specular, in a direction determined by the angle of incidence as for mirrors, or diffuse, scattering incident radiation in all directions.
- In general, τ , α and ρ are functions of wavelength, like ϵ , and, again, we approximate with constant values appropriate to particular temperature ranges

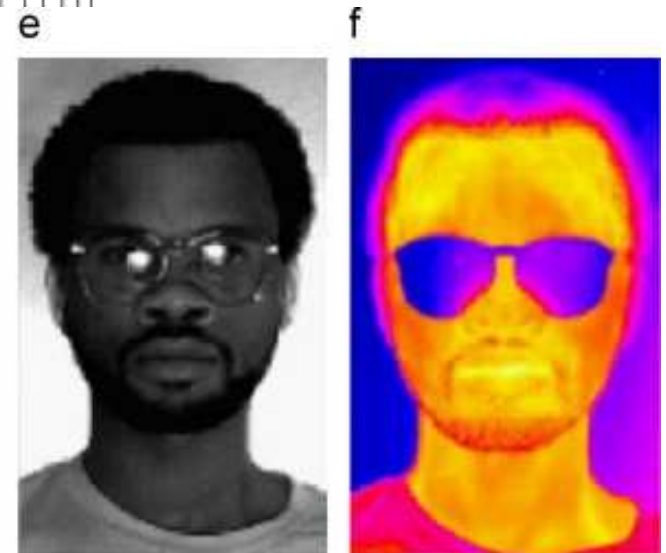


Absorption spectra for water



Water absorbs 10^7 as much EM radiation in the IR than in the blue

As you go lower in the ocean, blue light can penetrate better than red light.



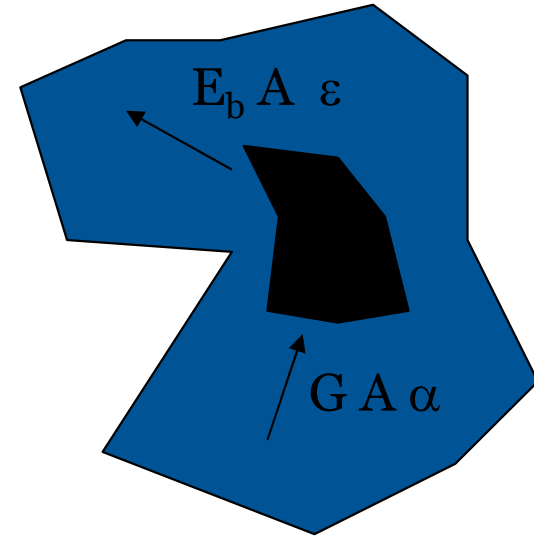
Same for glass. IR is absorbed more by glass and plastic than visible.



- It might be imagined that since both the emission and absorption of radiation are related to the excitation of molecules in and under the surface α_λ and ε_λ should be related.
- By considering the equilibrium between a small grey body and a black surface enclosing it, Kirchoff showed that both had to be at the same temperature and that this was only possible if -

$$\alpha_\lambda = \varepsilon_\lambda$$

- ***Kirchoff's Law.***



G is irradiance
 E_b is emitted energy
A is area



This illustrates a potential Pitfall for the Unwary.

As noted earlier, the temperatures are given in the emissivity table because the wavelength varies with temperature.

If the wavelength of the incident radiation differs greatly from that of the emitted radiation, the mean absorptivity will **not** be equal to the mean emissivity.

The sun can be modelled as a black body at 5762 K (see figure p2) and most of the incident energy lies in the visible part of the spectrum.

Car roofs, as most terrestrial surfaces, don't approach that temperature, and so the appropriate emissivities differ markedly from the absorptivities.

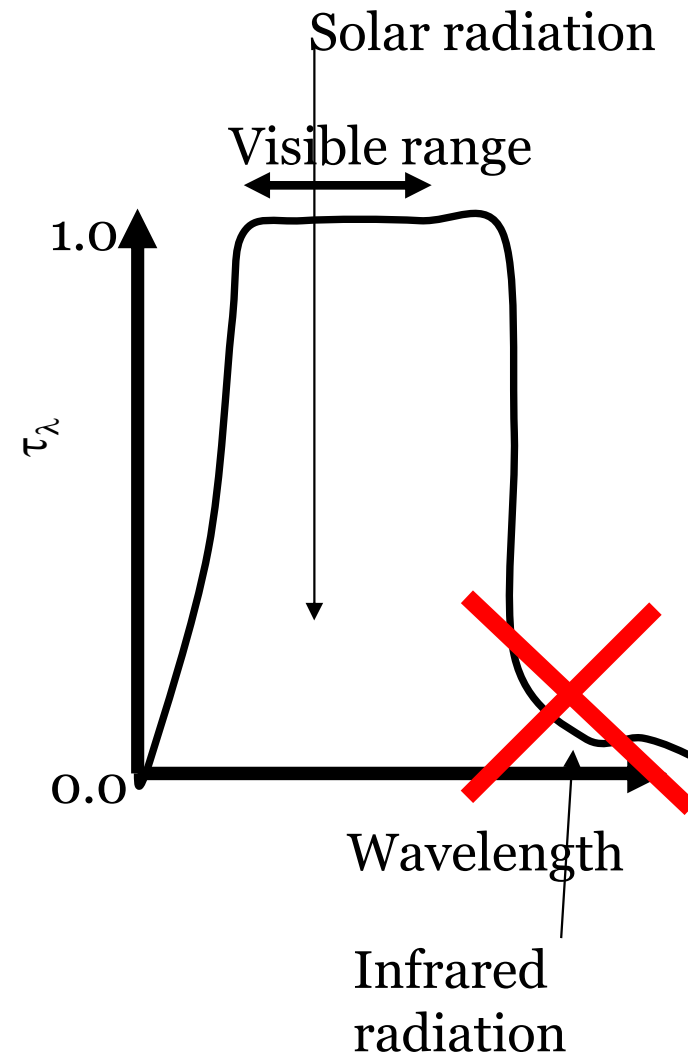
The table below compares absorptivities for solar and low temperature radiation

Surface	"Solar" absorptivity	"Low temp" absorptivity (at about 300 K)
Polished aluminium	0.15	0.04
Red brick	0.75	0.93
Flat black lacquer	0.96	0.95
White paints	0.12-0.16	0.9-0.95
Cast iron	0.94	0.21
Polished stainless steel	0.37	0.60

It is evident that white is a cool colour for cars. Note also that black is good for solar panels.



- Since solar energy is predominantly in the visible range, greenhouses admit this readily.
- The low temperature contents radiate in the infra-red and these wavelengths are reflected (i.e. blocked) by the glass.
- Ozone has a similar absorption spectrum





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View factors



General consideration of the radiation heat transfer process requires definition of the ***intensity*** of radiation emitted by a surface i.e. the rate of emission in a particular direction.

If the surface is a diffuse emitter, the intensity does not vary with direction.

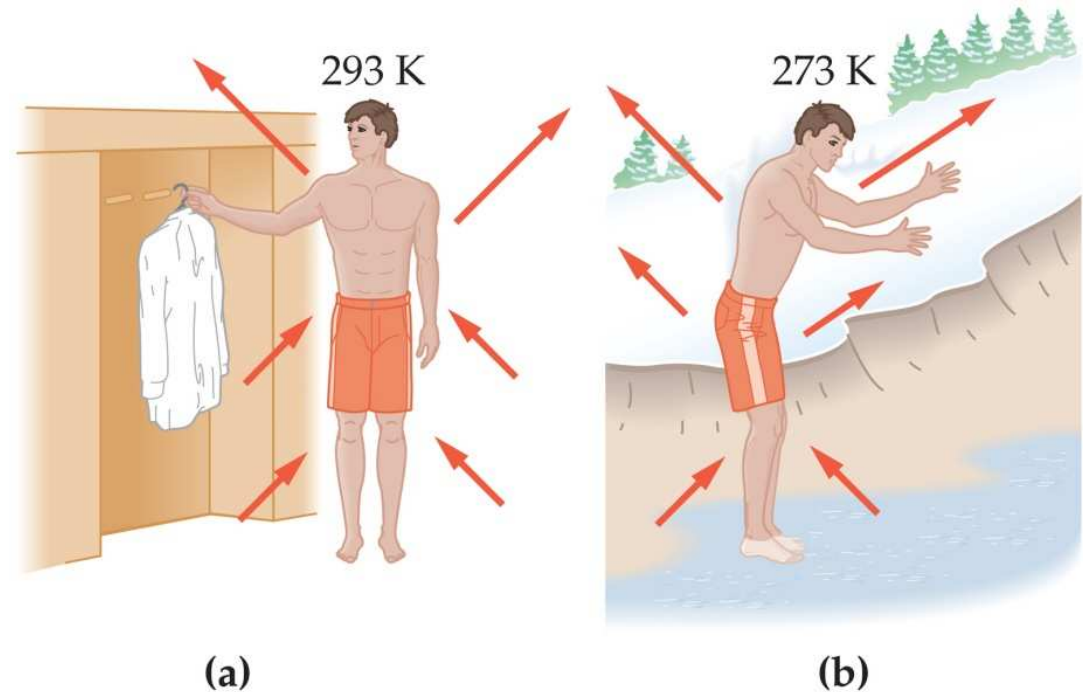
This is the only case we will consider here as it is at least approximately true of most surfaces encountered in engineering practice.

Subject to this restriction, it can be shown that the intensity is $I_b = \frac{\sigma T^4}{\pi}$

Radiation intensity is dependent on surface temperature (and surface properties)

but is independent of distance from the body.

- A person has a surface area of 1.15 m^2 and a surface temperature of 303K . What is the net radiated power from
 - (a) that person in a room where the temperature is 290K and
 - (b) outside where the temperature is 273K ? Assume an emissivity for the person's skin is 0.9 .



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$$\begin{aligned}(a) P_{net} &= eA\sigma(T^4 - T_s^4) \\ &= (0.9)(1.15)(5.67E-8)(303^4 - 293^4) \\ &= 62.1W\end{aligned}$$

$$\begin{aligned}(b) P_{net} &= eA\sigma(T^4 - T_s^4) \\ &= (0.9)(1.15)(5.67E-8)(303^4 - 273^4) \\ &= 168.7W\end{aligned}$$



Q: what is the net heat TO the person from the room

Not trivial.

Much of the radiation from the room walls impact the room walls!!!.

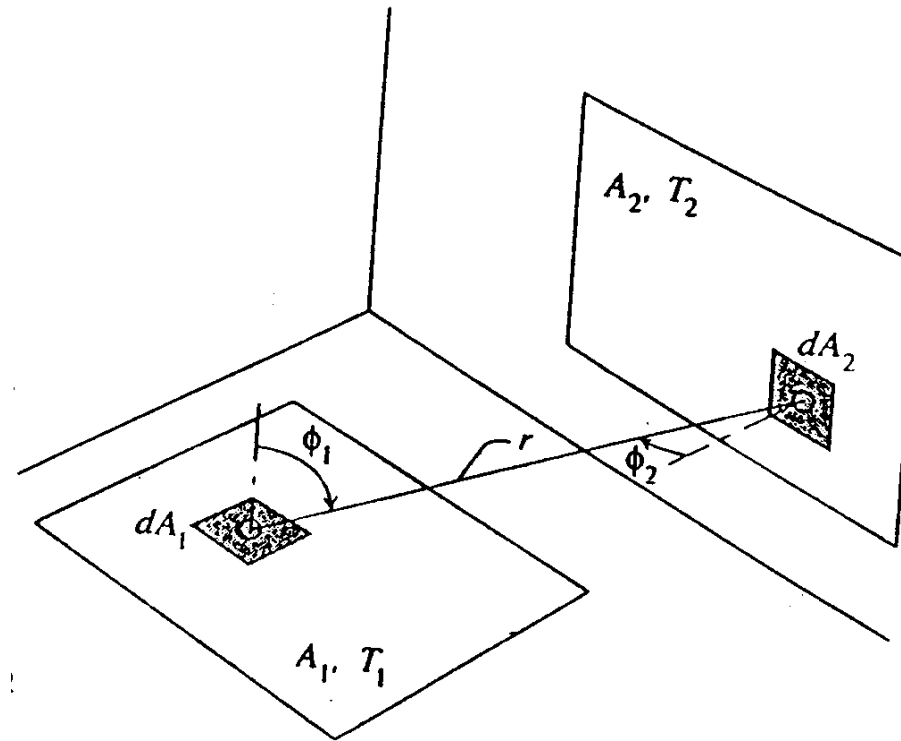
Only a small fraction impacts the person standing in the room.

This fraction is a number from 0-1 and is called the
View Factor



Black body radiation exchange.

- In considering the radiation heat transfer between two bodies we must recognise that each emits radiation to, and receives radiation from, the other body.
- The heat transferred is the difference between the two.
- Clearly, not all energy emitted by one body is received by the other.
- *The proportion of the total energy radiated by body 1 that is received by body 2 is called the **view factor** F_{1-2} . It is evaluated from the geometry.*





Solving this obtains the ***reciprocity relationship*** which says there is a sort of symmetry between how the two areas see each other:

$$A_1 F_{1-2} = A_2 F_{2-1}$$

In fact, this can be extended to any pair of black bodies within a group of such bodies simply writing:

$$A_n F_{n-m} = A_m F_{m-n}$$

Using the reciprocity relationship, the *net* heat transfer between the bodies can be written either as

$$\dot{q}_{b12} = A_1 F_{1-2} \sigma (T_1^4 - T_2^4) \quad \text{or} \quad \dot{q}_{b12} = A_2 F_{2-1} \sigma (T_1^4 - T_2^4)$$



For simple geometries, view factors are easily obtained, and are extensively catalogued in the literature. A few useful cases are noted here.

If A_1 has no re-entrant (concave) bits, and is enclosed within A_2 *all* the energy it radiates falls on A_2 . Thus, by definition $F_{1-2} = 1$, and reciprocity gives $F_{2-1} = A_1 / A_2$.

The sum of all view factors from a surface must equal 1.

$$e.g. F_{1-1} + F_{1-2} + \dots + F_{1-N} = 1$$

View factor of a surface to itself is zero if the surface is not concave

$$F_{1-1} = 0$$

Can set up simultaneous equations to calculate view factors of surfaces.

$$F_{1-2} + F_{1-3} = 1, F_{2-1} + F_{2-3} = 1$$

$$F_{3-1} + F_{3-2} = 1$$

Six equations with six unknowns. Can solve simultaneously if we know areas.

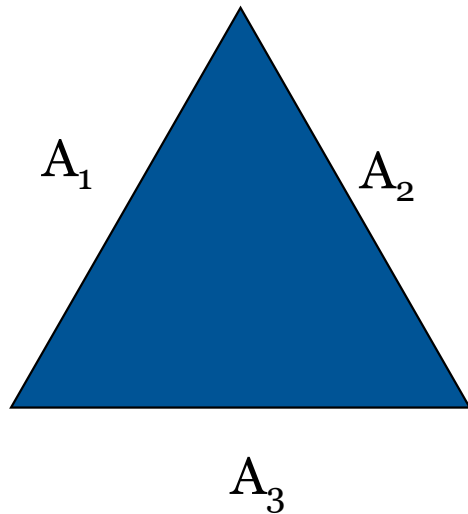
$$A_2 F_{2-1} = A_1 F_{1-2}, A_3 F_{3-1} = A_1 F_{1-3}$$

$$A_3 F_{3-2} = A_2 F_{2-3}$$



View factors: symmetry can help

For a triangular cross section prism of infinite length, show that the view factor F_{12} and F_{13} are:

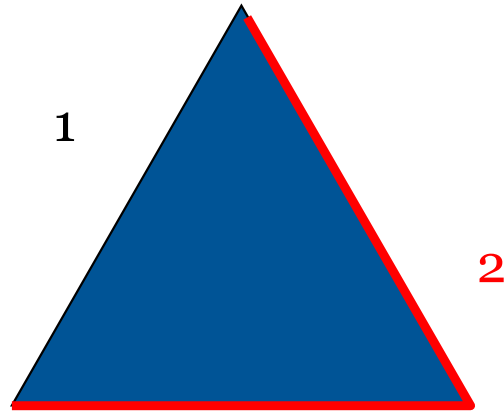


$$F_{12} = \frac{A_1 + A_2 - A_3}{2A_1}$$

$$F_{13} = \frac{A_1 + A_3 - A_2}{2A_1}$$

Symmetry can make it easier to determine view factor.

1. Sketch a cylinder.
2. Using geometrical arguments show that the view factor from either end to the other is the same.
3. Using geometrical arguments show that the view factor from either end to the sidewalls are the same.
4. Consider that the system was a prism with any cross-section. What about points 2 and 3 for this prism.



- What is the view factor from surface 1 to surface 2?
- What is the view factor from surface 2 to 1?

The entire view of surface 1 impinges on only surface 2 so $F_{12} = 1$.

Only some of the surface 2 impinges on surface 1.

So $F_{22} \neq 0$

How much from surface 2 impacts on surface 1.

$$A_1 F_{12} = A_2 F_{21} \quad , \quad A_2 = 2A_1, \quad \text{so } F_{21} = \frac{A_1}{2A_1} F_{12} = 0.5$$

So half of the radiation hits surface 1 and only half hits surface 2.

Consider the figure to the right:
 The view factor is given by:

$$F_{1-2} = \frac{(L_5 + L_6) - (L_3 + L_4)}{2L_1}$$

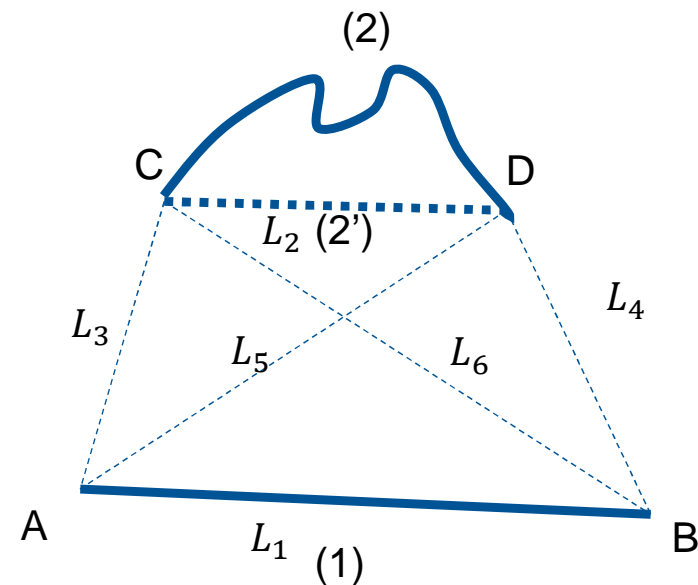
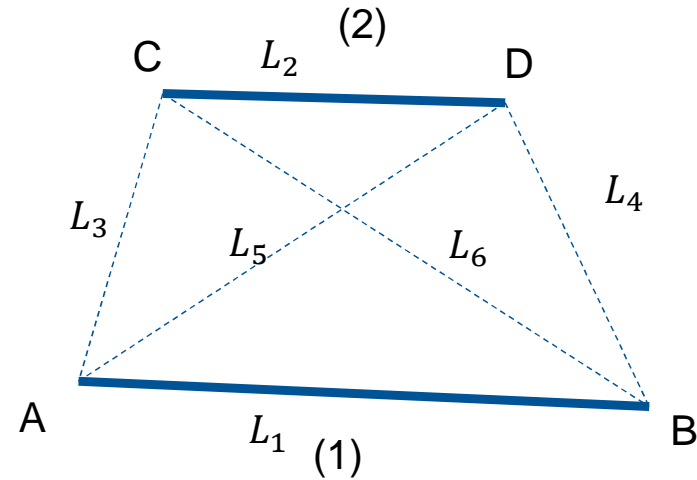
Similarly:

$$F_{2-1} = \frac{(L_5 + L_6) - (L_3 + L_4)}{2L_2}$$

Note that the shape of the surface is irrelevant. We can construct a virtual surface. The view factor between (2) and (2') is 1.

So the view factor from (2') to (1) is the same as the view factor from (2) to (1).

Only valid for shapes that are VERY long in one direction only.





View factor from 1 to 2 is 1.

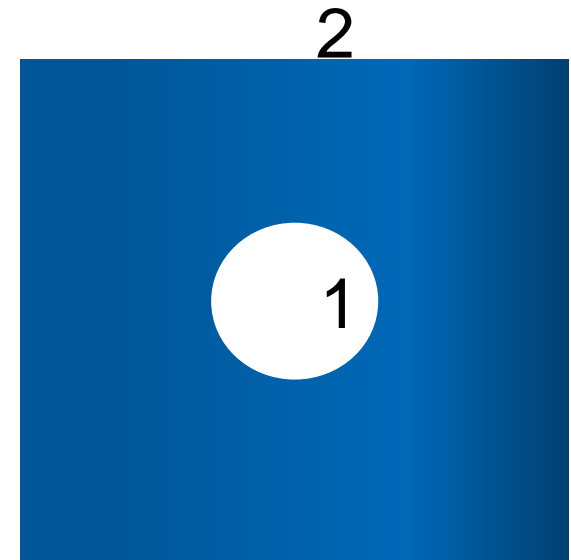
$$F_{12} = 1$$

View factor from 2 to 1 is dependent on the sizes

$$F_{21} \neq 1$$

Need to solve with reciprocity relationship.

$$A_1 F_{12} = A_2 F_{21} \rightarrow F_{21} = \frac{A_1}{A_2} F_{12} = \frac{A_1}{A_2}$$



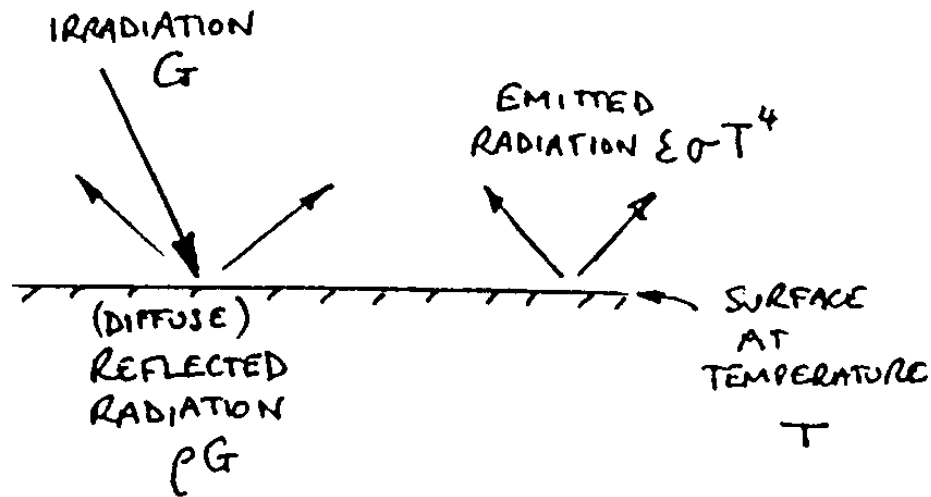
Think convex and concave and remember that if you place an imaginary surface in front of the target area (because it is convoluted), the view factor to that imaginary surface is the same



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**Radiosity:
body is no
longer black**



$$J = \epsilon \sigma T^4 + \rho G$$

- The radiation leaving a real surface is the *sum* of the radiation *emitted* by the surface and the radiation *reflected* from the surface.
- We must therefore distinguish between the radiation flux falling on the surface – the *irradiation*, G – and that leaving the surface – the *radiosity*, J .
- For a grey surface of emissivity ϵ and reflectivity ρ



- The *net* radiation flux leaving per unit area of the surface is $J - G$. Using the above equation this can be written:
$$\mathbf{J} = \varepsilon \sigma T^4 + \rho G$$

$$J - G = J - \frac{J - \varepsilon \sigma T^4}{\rho} = \frac{J(\rho - 1) + \varepsilon \sigma T^4}{\rho} = \frac{\varepsilon(\sigma T^4 - J)}{\rho}$$

(Recall also that *if the surface is opaque* $\rho = 1 - \varepsilon$, since $\alpha = \varepsilon$)

- For grey bodies, it is the radiosity that is exchanged, not just the emitted radiation.
- Clearly, this exchange is governed by the same geometric constraints as the black body situation considered earlier
- i.e. we can use the same view factors but apply them to the radiosity not the emitted radiation.



networks for radiation and radiosity exchange problems

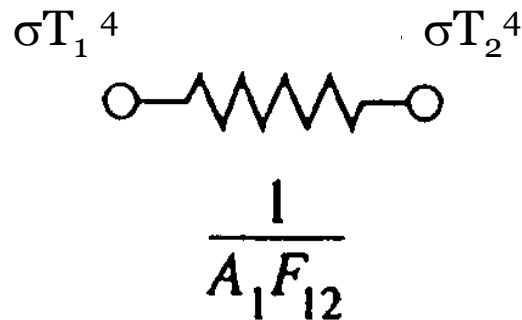
For exchange between two black bodies we have the net heat flow from 1 to 2 as:

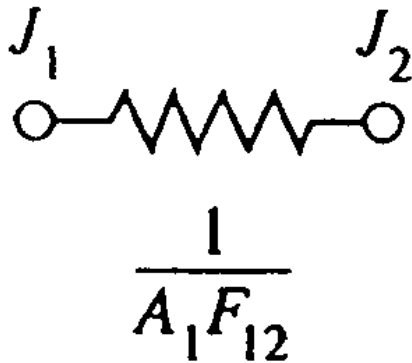
$$\dot{q}_{b12} = A_1 F_{1-2} \sigma (T_1^4 - T_2^4)$$

Treating the heat flow as a current and the black body emission flux densities

$$q''_{b1} = \sigma T_1^4 \quad \text{and} \quad q''_{b2} = \sigma T_2^4$$

as voltages, the equation is completely modelled if we take $1/A_1 F_{1-2}$ as the "spatial resistance" connecting surfaces 1 and 2.





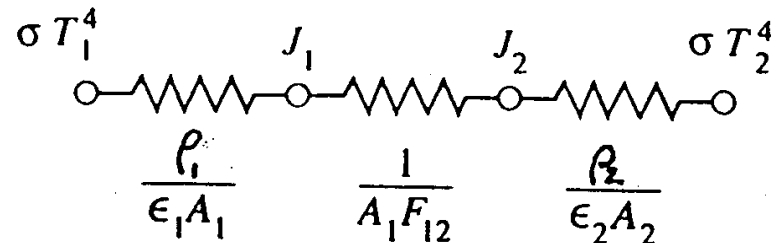
The identical spatial resistance applies for grey body *radiosity* exchange i.e we have two terminals at J_1 and J_2 connected by a resistance $1/ A_1 F_{1-2}$.

To complete the grey body model we must introduce further “*surface resistance*” elements. $J - G = \frac{\varepsilon (\sigma T^4 - J)}{\rho}$
The net heat flow from A_1 is, from previously:

This gives a radiation exchange between the surface and the Radiosity value of:

$$A_1(J_1 - G_1) = \frac{A_1 \varepsilon_1 (\sigma T^4 - J_1)}{\rho_1} = \frac{\sigma T_1^4 - J_1}{R_{b_1-J_1}}, \text{ where } R_{b_1-J_1} = \frac{\rho_1}{A_1 \varepsilon_1}$$

Thus, the *network for the exchange between two grey bodies is:*

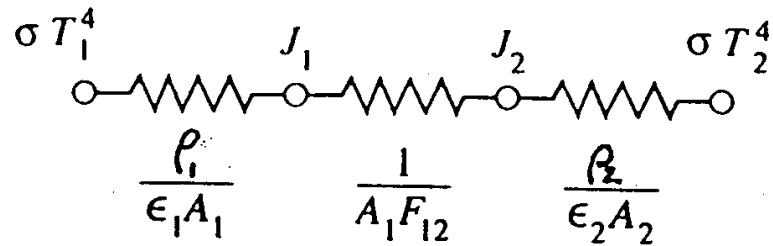


More complex situations can be built up to include any number of bodies.

We will stop at three.



Grey body view factor



This radiation network for the exchange between 2 grey bodies gives the heat transfer as:

$$\dot{q}_{12} = \frac{\sigma(T_1^4 - T_2^4)}{\left\{ \frac{\rho_1}{A_1 \epsilon_1} + \frac{1}{A_1 F_{12}} + \frac{\rho_2}{A_2 \epsilon_2} \right\}} = \sigma A_1 \mathfrak{F}_{12} (T_1^4 - T_2^4)$$

where $\mathfrak{F}_{12} = \frac{1}{\left\{ \frac{\rho_1}{\epsilon_1} + \frac{1}{F_{12}} + \frac{A_1 \rho_2}{A_2 \epsilon_2} \right\}}$ is the "Grey body view factor"

A reminder: For opaque surfaces, $\alpha + \rho = 1$, so, since $\alpha = \epsilon$, $\rho = 1 - \epsilon$

It is more important to remember the resistance network than the formula.



Q: Will you tan faster by sitting close to your (outdoor) pool?

Q: Can you measure the temperature of a welding torch flame by shoving an exposed bead thermocouple into it?



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Examples of radiation



Examples in Radiation heat transfer

Domestic radiator with shielding to reduce loss to the wall.

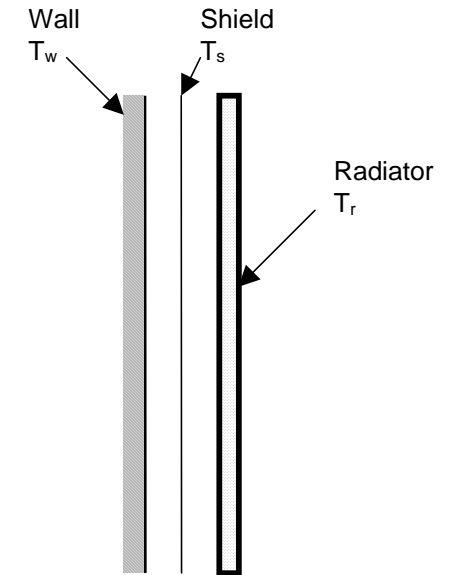
Emissivities of radiator and wall, $\varepsilon_r = \varepsilon_w = 0.9$

Emissivity of shield, $\varepsilon_s = 0.2$

Radiator temperature, $t_r = 77^\circ\text{C}$

Wall temperature, $t_w = 17^\circ\text{C}$

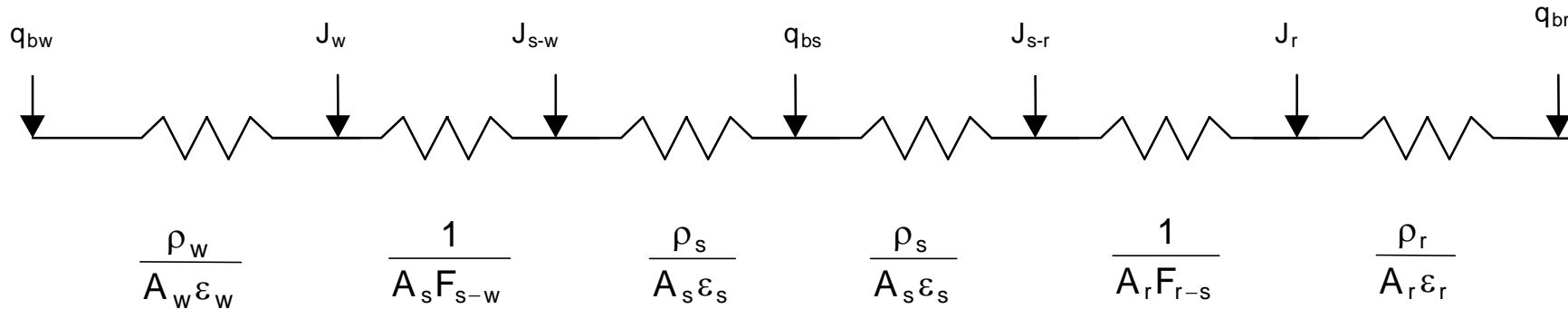
Area = 1 m^2



Diffuse grey body emission is assumed, with the back surface of the radiator exchanging radiation only with the shield, and the shield exchanging radiation only with the radiator and the wall.



The network modelling the radiation exchange is:



Noting that

$$A_s = A_r = A_w,$$

and that if we neglect end losses

$$F_{s-w} = F_{r-s} = 1$$

since $\rho = 1 - \epsilon$.

$$R_{w-r} = \frac{1}{A_r} \left(\frac{\rho_w}{\epsilon_w} + 1 + \frac{\rho_s}{\epsilon_s} + \frac{\rho_s}{\epsilon_s} + 1 + \frac{\rho_r}{\epsilon_r} \right)$$

$$= \frac{1}{A_r} \left(\frac{1 - \epsilon_w}{\epsilon_w} + 2 \frac{1 - \epsilon_s}{\epsilon_s} + \frac{1 - \epsilon_r}{\epsilon_r} + 2 \right)$$

The total network resistance is

Inserting values gives for $A_r = 1\text{m}^2$ $R_{w-r} = 10.22 \text{ K}^4/\text{W}$



Hence, with the shield, the radiation loss to the wall is:

$$\dot{q}_{r-w} = \frac{\sigma (T_r^4 - T_w^4)}{R_{w-r}} = \frac{5.67 \times 10^{-8} (350^4 - 290^4)}{10.22} = \underline{\underline{44.0 \text{ W}}}$$

If the shield is omitted, the resistance becomes

$$R_{w-r} = \frac{1}{A_r} \left(\frac{\rho_w}{\varepsilon_w} + 1 + \frac{\rho_r}{\varepsilon_r} \right) = 1.22 \text{ K}^4 / \text{W}$$

and the loss is 368.6W

The loss is greatly reduced by the shield. What temperature does this reach?

$$\dot{q}_{r-s} = \frac{\sigma (T_r^4 - T_s^4)}{R_{r-s}} = A_r \frac{\sigma (T_r^4 - T_s^4)}{\left(\frac{\rho_r}{\varepsilon_r} + \frac{1}{F_{r-s}} + \frac{\rho_s}{\varepsilon_s} \right)} = \dot{q}_{r-w}$$

which gives
 $T_s = 51.2 \text{ }^\circ\text{C}.$



In reality, free convection heat transfer is also present, so that the shield temperature will be reduced and the radiation loss increased.

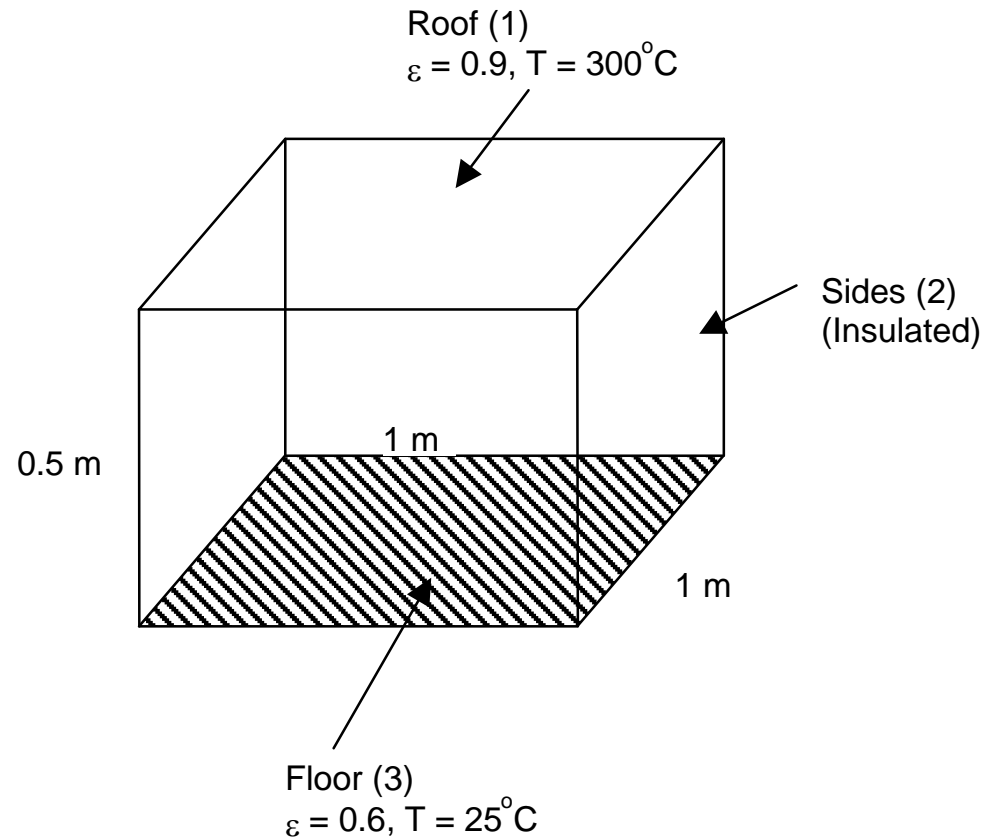
Nevertheless, there is still an advantage gained from the use of a shield.

So far, only two- body exchange has been considered. As promised earlier, the complexity introduced by an additional body is now examined.



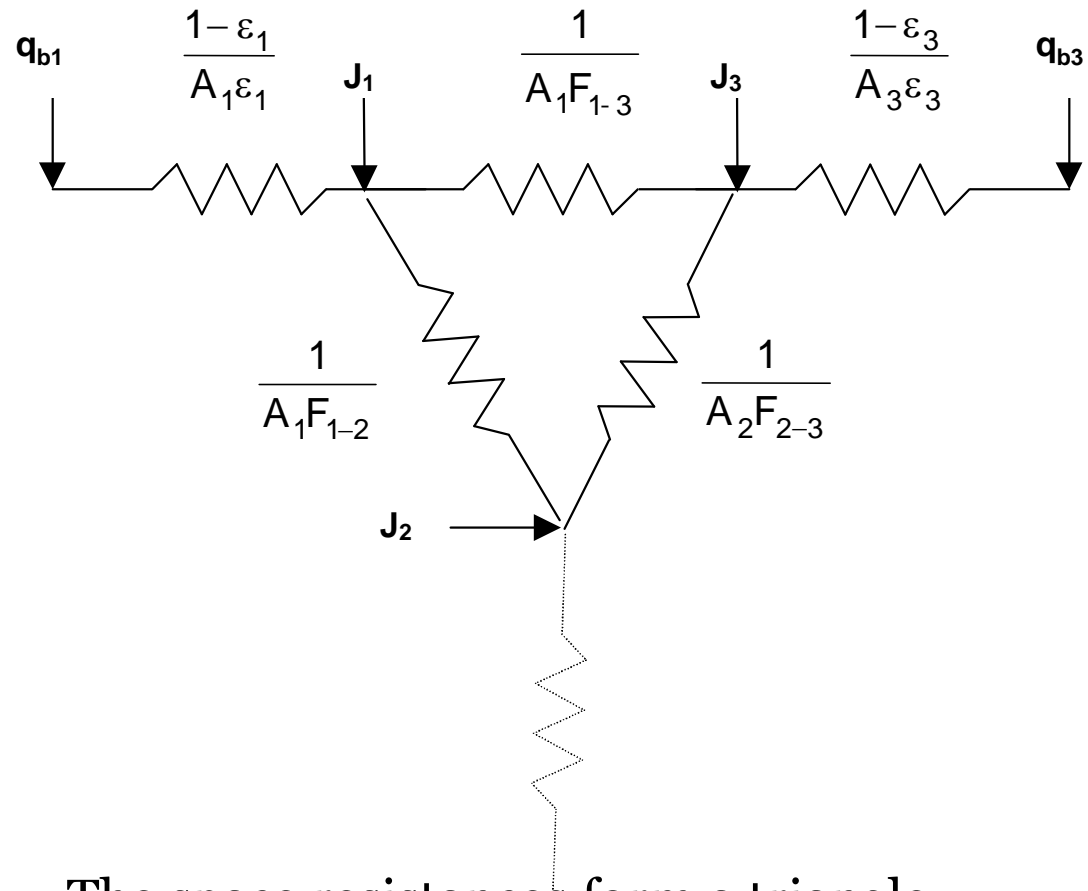
Radiation exchange between 3 grey surfaces.

- The diagram shows an oven with material to be heated covering its floor.
- We want to calculate the rate of radiative heat transfer from the roof to the floor and the equilibrium temperature of the insulated walls for the stated (initial) roof and floor temperatures.
- The view factor from roof to floor F_{1-3} is given as 0.41.



Since all the radiation from the roof is intercepted by the floor and the sides, it follows that the view factor from roof to sides is $F_{1-2} = 1 - 0.41 = 0.59$
Geometric symmetry also means that $F_{3-2} = F_{1-2}$.

- Note that the resistance connected to J_3 has been drawn dotted.
- This is because the surface is insulated and so there can be no net flow of radiation to or from it – the “current” through the surface resistance is zero.
- The reflectivity $\rho = 1 - \varepsilon$, and this has been recognised in the expressions given for surface resistances.



The space resistances form a triangle connecting the three terminals which are at a “voltage” corresponding to the radiosity of each surface, so that each surface exchanges radiosity with the other 2. These radiosity terminals are fed through surface resistances from the black body emissions.



The net radiation heat transfer is: $\dot{q}_{1-3} = \frac{(\dot{q}_{b1}'' - \dot{q}_{b3}'')}{R_{1-3}}$ [1]

where R_{1-3} is the total resistance between 1 and 3

$$R_{1-3} = \frac{1 - \varepsilon_1}{A_1 \varepsilon_1} + \left(\frac{1}{A_1 F_{1-3} + \frac{1}{\left(\frac{1}{A_1 F_{1-2}} + \frac{1}{A_2 F_{2-3}} \right)}} \right) + \frac{(1 - \varepsilon_3)}{A_3 \varepsilon_3}$$

[A practical tip: It is usually easiest to substitute numerical values and work out the resistance between J_1 and J_3 first, and then to add on the other 2]

Inserting the numerical values and noting equalities worked out earlier gives

$$R_{1-3} = 2.196 \text{ K}^4/\text{W}$$

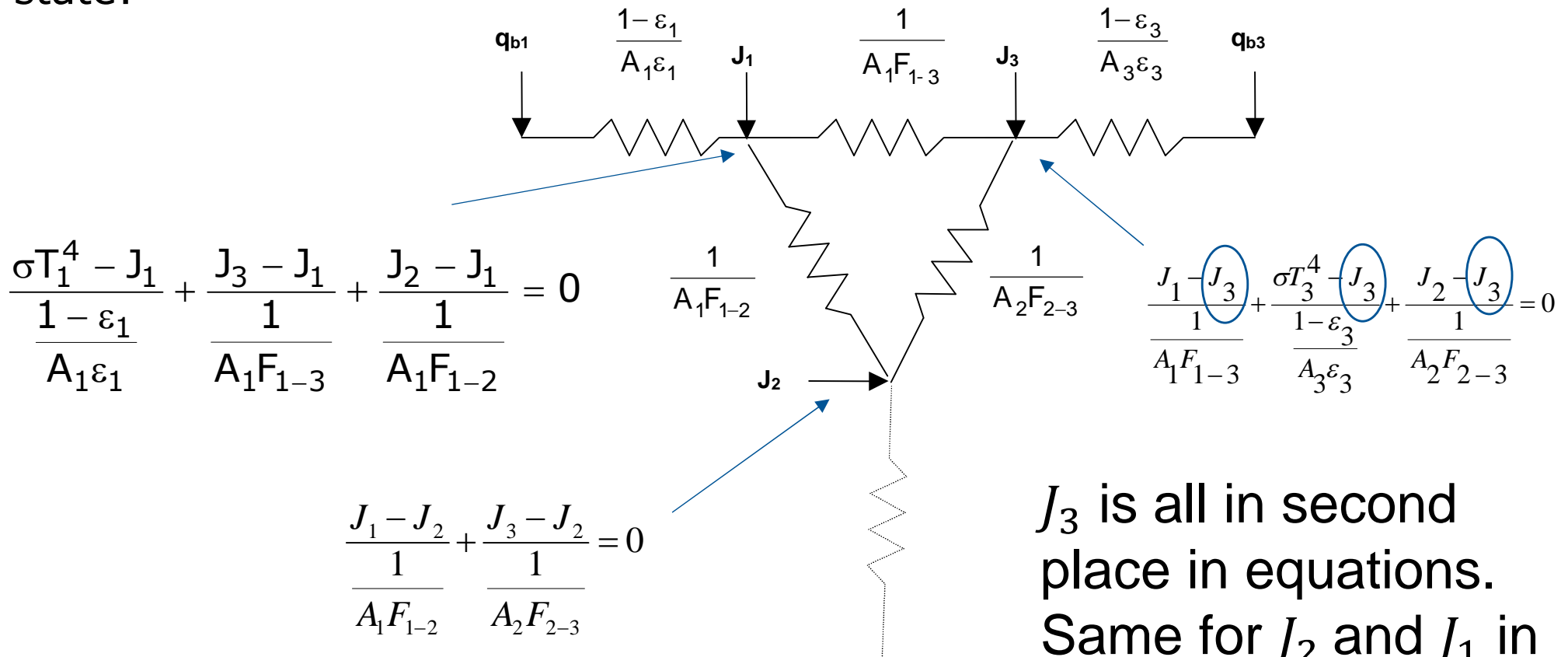
Since $\dot{q}_{b1}'' = \sigma T_1^4$ and $\dot{q}_{b3}'' = \sigma T_3^4$, substituting in [1] then gives

$$\dot{q}_{1-3} = \frac{\sigma(T_1^4 - T_3^4)}{R_{1-3}} = \frac{5.67 \times 10^{-8} (573^4 - 298^4)}{2.196} = \underline{\underline{2.58 \text{ kW}}}$$



General approach: obtaining temperatures

A more general approach to solving networks of any size is to recognise that we can apply an energy balance at each of the nodes –
the net rate of energy inflow to any node must be zero in the steady state:



J_3 is all in second place in equations. Same for J_2 and J_1 in other equations



A flat paving stone open to the sky.

A hollow cylinder with one end heated. The curved walls are insulated.

A hollow cylinder with one end heated. The curved walls are not insulated.

Triangular cross-section prism.

Hexagonal cross section prism with one end heated and the sides insulated.



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Combining radiation, conduction and convection in problems

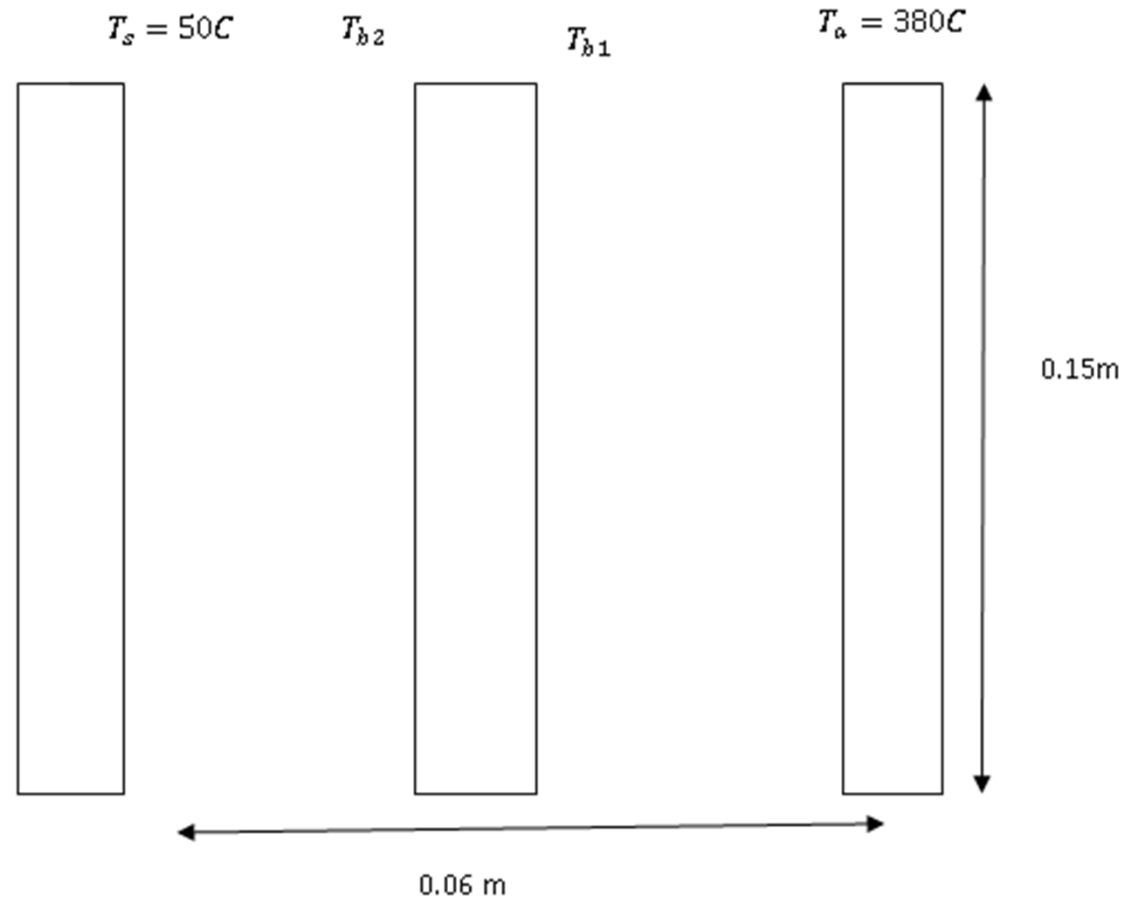


Heat transfer by combined modes

- Most practical heat transfer problems involve more than one mode of heat transfer.
- Although one may be sufficiently dominant to allow the others to be neglected, it will usually be necessary to do at least a back-of-envelope sum to check the validity of such an assumption, and a complete analysis involving all modes simultaneously may be required.
- If the roof of a car on a windy day is hotter than the surrounding air, heat will be convected from the surface cooling the surface.
- Radiation will also be emitted from the surface and cause the surface to lose heat.
- In many circumstances the two rates of heat loss are comparable and both need to be taken into consideration.



- Imagine that you have a hot surface at 380C positioned 6cm from a cold surface at 50C.
- Each plate is square with length of side 0.15 m. Take k of shield = 0.23 W/m²K.
- $A_a = A_{b1} = A_{b2} = A_s = 0.15^2 m^2$.
- $\epsilon_a = 0.3, \epsilon_{b1} = 0.2,$
- $\epsilon_{b2} = 0.2, \epsilon_s = 0.9$.
- Thickness of shield is 1 cm and it is spaced equidistant from hot and cold surface.
- No shield has a heat flow of 63 W.

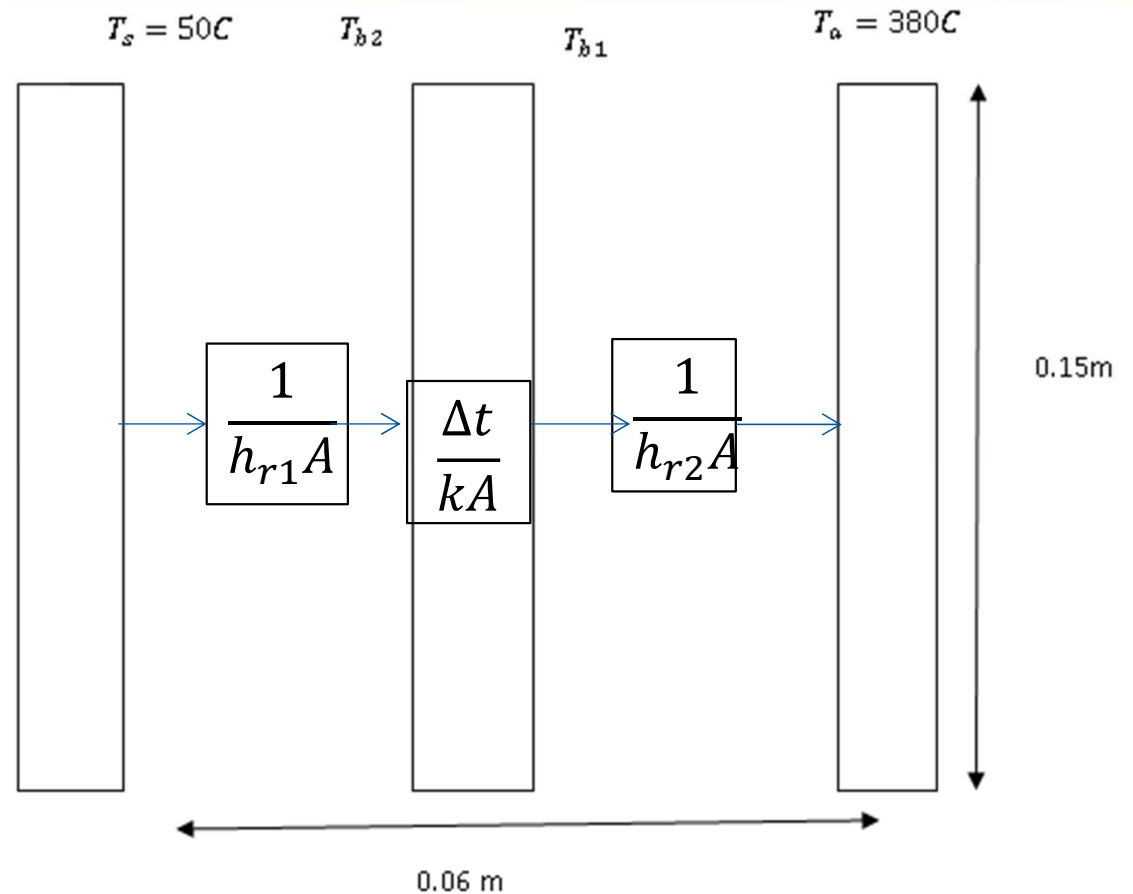


$$q = \frac{\sigma(T_a^4 - T_s^4)}{R_{rad}} = \frac{\sigma(T_a^4 - T_s^4)}{\frac{1 - \epsilon_a}{\epsilon_a A} + \frac{1}{F_{as} A} + \frac{1 - \epsilon_s}{\epsilon_s A}}$$

$$= 63.3\text{W}$$



- Can construct a three resistor model.
- Heat shield has thermal mass, so will react slowly to input of energy.
- Can rewrite radiation as heat transfer coefficient



$$h_{r1} = A_a F_{a-b1} \sigma (T_a^2 + T_{b1}^2) (T_{b2} + T_s) \quad (1)$$

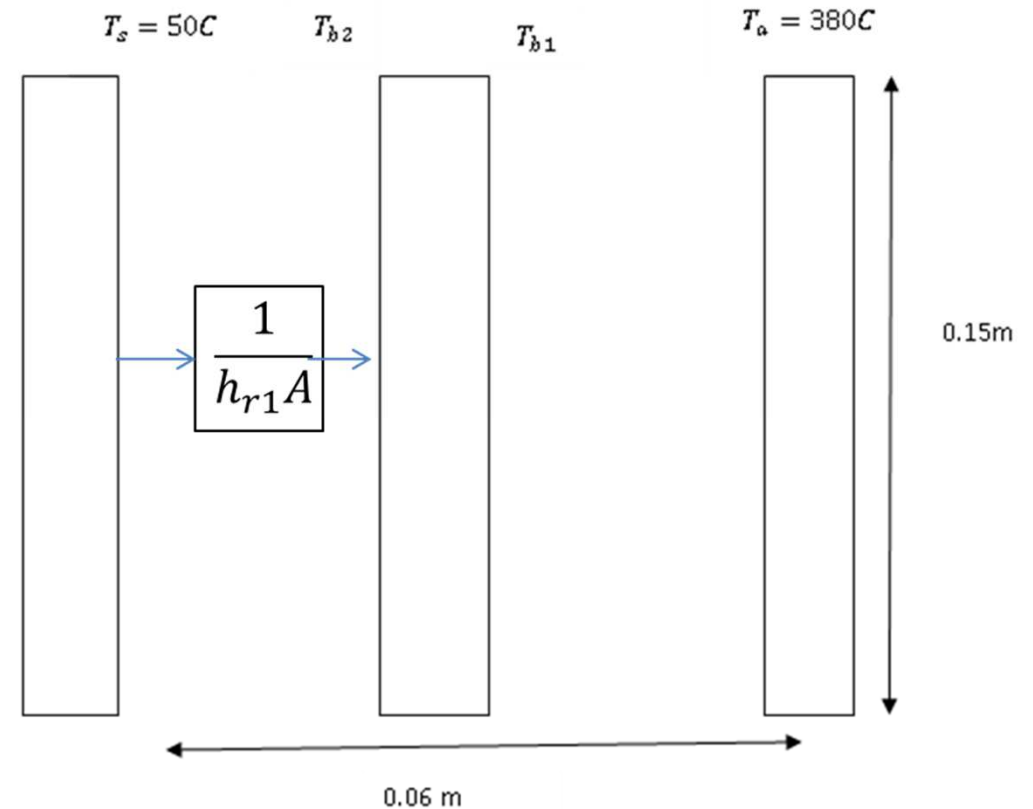
$$h_{r2} = A_b F_{b2-s} \sigma (T_{b2}^2 + T_s^2) (T_{b2} + T_s) \quad (2)$$

$$F_{a-b1} = \frac{1}{\frac{1 - \epsilon_a}{\epsilon_a} + \frac{1}{F_{a-b1}} + \frac{(1 - \epsilon_{b1})}{\epsilon_{b1}}}$$



Derivation of explicit formulation for Surface with radiation:

- Net heat into node = change in internal energy at node



$$h_{r1}A(T_a - T_{b1}^0) + \frac{kA}{\Delta x}(T_{m-1}^0 - T_{b1}^0) = \frac{\rho c_p \Delta x A (T_{b1}^1 - T_{b1}^0)}{2\Delta t}$$

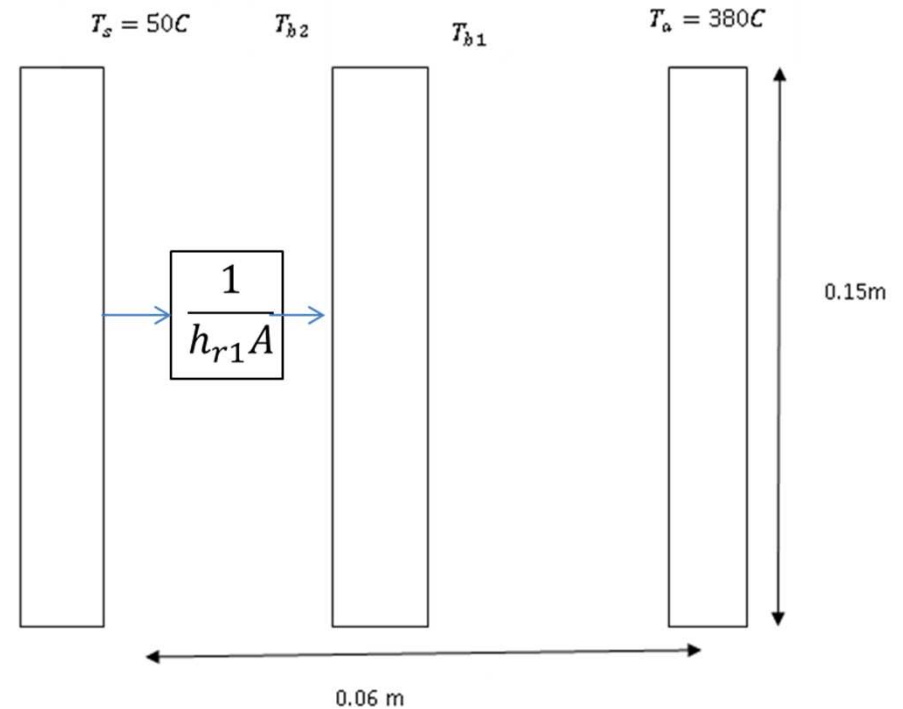
$$\rightarrow 2 \left(\frac{\alpha \Delta t}{\Delta x^2} \right) \left(\frac{h_{r1} \Delta x}{k} \right) (T_a - T_{b1}^0) + 2 \left(\frac{\alpha \Delta t}{\Delta x^2} \right) (T_{m-1}^0 - T_{b1}^0) + T_{b1}^0 = T_{b1}^1$$

$$T_{b1}^1 = T_{b1}^0 (1 - 2FoBi_{r1} - 2Fo) + 2Fo T_{m-1}^0 + 2FoBi_{r1} T_a \quad (3)$$



Derivation of implicit formulation for Surface with radiation:

- Net heat into node = change in internal energy at node



$$h_{r1}A(T_a - T_{b1}^1) + \frac{kA}{\Delta x}(T_{m-1}^1 - T_{b1}^1) = \frac{\rho c_p \Delta x A (T_{b1}^1 - T_{b1}^0)}{2\Delta t}$$

$$T_{b1}^1 \left(1 + \frac{2\alpha\Delta t}{\Delta x^2} + \frac{2h_{r1}\Delta x \alpha\Delta t}{\Delta x^2 k} \right) = T_{b1}^0 + \frac{2\alpha\Delta t}{\Delta x^2} T_{m-1}^1 + \frac{2h_{r1}\Delta x \alpha\Delta t}{\Delta x^2 k} T_a$$

$$T_{b1}^1 = (T_{b1}^0 + 2Fo T_{m-1}^1 + 2FoBi_{r1}T_a) / (1 + 2Fo + 2FoBi_{r1}) \quad (4)$$



- 1) Estimate temperatures of shield.
- 2) Use temperatures and gray body view factor to estimate heat transfer coefficients from equations (1) and (2).
- 3) Use estimated of heat transfer coefficients to estimate total resistance. (three resistances in series)
- 4)
$$R_T = \frac{1}{h_{r1}A} + \frac{t}{kA} + \frac{1}{h_{r2}A}$$
- 5) Use total resistance to estimate heat flow through entire system.
- 6)
$$Q_T = \frac{T_a - T_s}{R_T}$$
- 7) Total heat flow through entire system is equal to heat flow through each resistor. Use this fact for new guess of shield wall temperatures.
- 8)
$$T_{b1}^1 = T_a - \frac{Q_T}{h_{r1}A}, \quad T_{b2}^1 = T_s + \frac{Q_T}{h_{r2}A}$$
- 9) Use new temperatures and return to [3].



Calculations: work left to right in steps

Guess temperatures

Use new temperatures

Use guess to estimate h values

Use h values to estimate total resistance

Heat flow can now be estimated

Heat flow can be used to estimate temperature

iteration	Temperatures				Variables						check	
	Ta	Tb1	Tb2	Ts	hr1	hr2	RT	QT	Tb1a	Tb2a	Q2	Q1
1	653	488	488	323	5.86	3.08	23.94	13.79	548.49	349.64	11.439	21.76
2	653	548.49	349.64	323	6.76	1.69	34.80	9.48	590.61	341.32	1.013	15.88
3	653	590.61	341.32	323	7.45	1.63	35.20	9.37	597.11	341.11	0.671	10.46
4	653	597.11	341.11	323	7.57	1.63	35.14	9.39	597.85	341.15	0.663	9.517
5	653	597.85	341.15	323	7.58	1.63	35.13	9.39	597.92	341.15	0.664	9.407
6	653	597.92	341.15	323	7.58	1.63	35.12	9.40	597.93	341.15	0.664	9.396
implicit	653	550.80	520.62	323	6.79	3.51	21.13	15.62			15.620	0



- 1) Estimate spatial step as $Bi \ll 1$. Say we use $Bi=0.1$, then $\Delta x \leq 0.1 * \frac{k}{h}$. Use largest h value as this will give smallest value of spatial step.
- 2) Estimate time step necessary from Fourier number; $\Delta t \leq \frac{\Delta x^2}{(2(1+Bi)\alpha)}$. Use largest Bi as this will give smaller time step.
- 3) Choose a number of nodes across shield with spatial step smaller than value in step [1]. (In this case we choose $\Delta x = 0.0025\text{m}$ so we can have exactly 4 nodes).
- 4) Use temperatures and gray body view factor to estimate heat transfer coefficients from equations (1) and (2).
- 5) Use h to estimate the Biot number.
- 6) Use equations (4)-(6) above to estimate temperature after time step.
- 7) Repeat from step [4] at new time Δt later until convergence is achieved.



What is size of cells needed?

					Actual Bi and Fo
GFab1	0.136363636			Bir1	0.078677
GFb2s	0.195652174			Bir2	0.017502
alpha	0.000001			Fo	0.4
deltx<	0.003177554			deltx	0.0025
deltt<	2.897067773			deltt	2.5
deltt<	3.071248273				

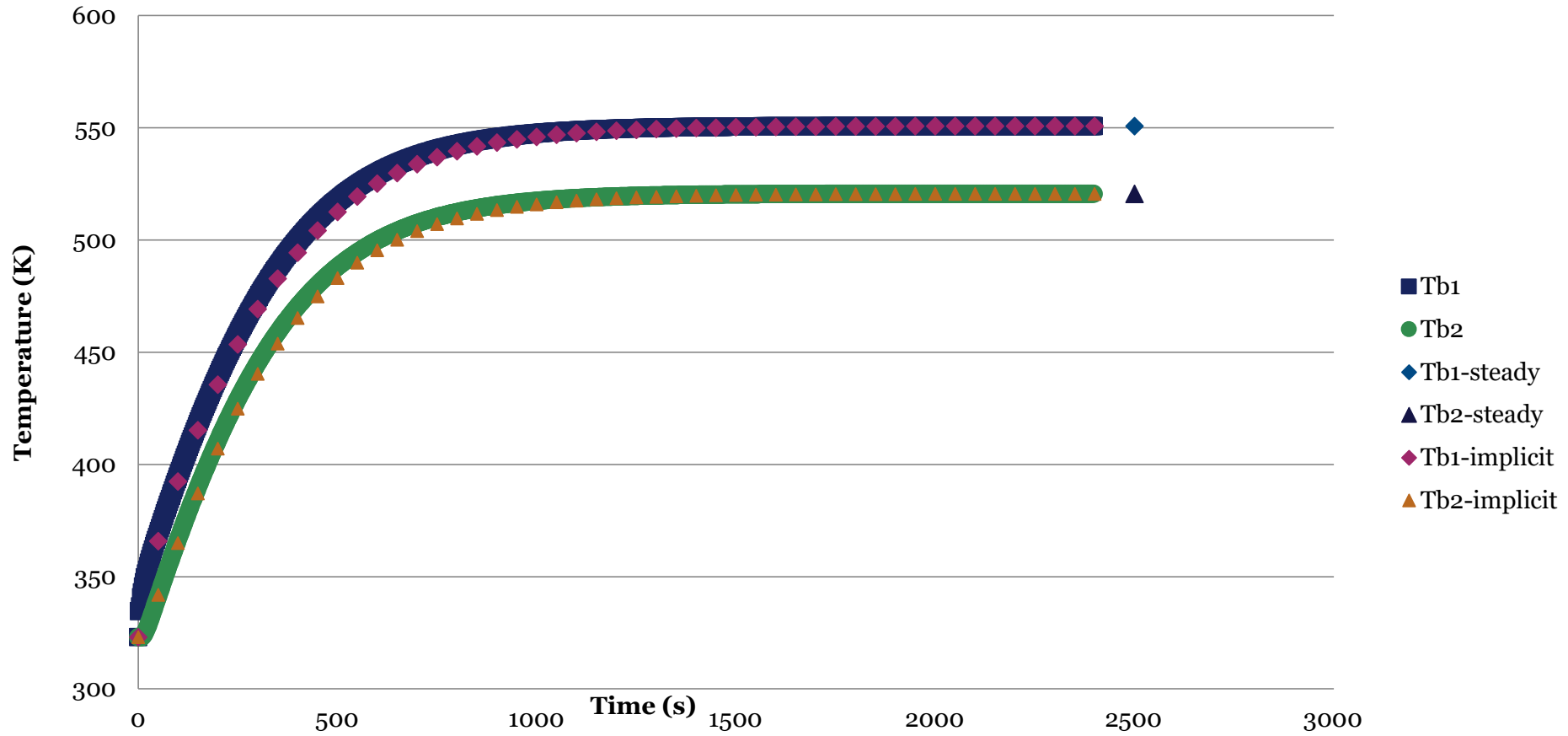
Maximum cell size allowed

Maximum time step allowed

Values used



Transient analysis





- 1) The effect of convection will be to decrease the surface temperatures further. This will have the effect of decreasing the heat flow through the system. Since the temperatures are very low, h_{r1} is of the order of 7 and 3.5. Convection heat transfer coefficients will have to be equivalent to make any impression on the temperature difference.
- 2) The effect of end effects will be that the view factor will be smaller than 1. This will decrease the heat transfer directly, but a secondary effect is that some of the radiation from the surface will be emitted to the ambient. Both of these effects will decrease the heat flow through the system and decrease the surface temperatures.
- 3) The steady state solution is only possible since the temperatures are constant. In reality this is not true. The hot surface will cool unless additional heat is added and the cold surface will heat up unless heat is removed. The transient solution would be better with a constant heat input term rather than a stable temperature. With the shield, the hot surface will in effect become hotter and this can have additional problems.
- 4) The flow through the shield will be essentially 1D. Very little 2D effects will occur in this problem as formulated.



Natural and forced convection example

The plate glass window of a shop measures 3.5m long by 1.8m high and is 6mm thick. The air temperature within the shop is maintained at 22°C. Calculate the heat lost through the window due to natural convection on the inner (shop) side and forced convection on the outside where the wind speed is 10 m/s and the atmospheric temperature is 4°C. Assume the wind is horizontal and tangential to the window.

For forced convection past a plane surface of length L , the correlation:

$$Nu_L = 0.664 Re_L^{1/2} Pr^{1/3}$$

Applies, with the fluid properties determined at the mean film temperature.

For natural convection over a vertical surface of height H , the average heat transfer coefficient is given by:

$$\bar{h} = 1.42 \left(\frac{\Delta T}{H} \right)^{1/4} \quad \text{for } 1E4 < Gr_H Pr \leq 1E9$$

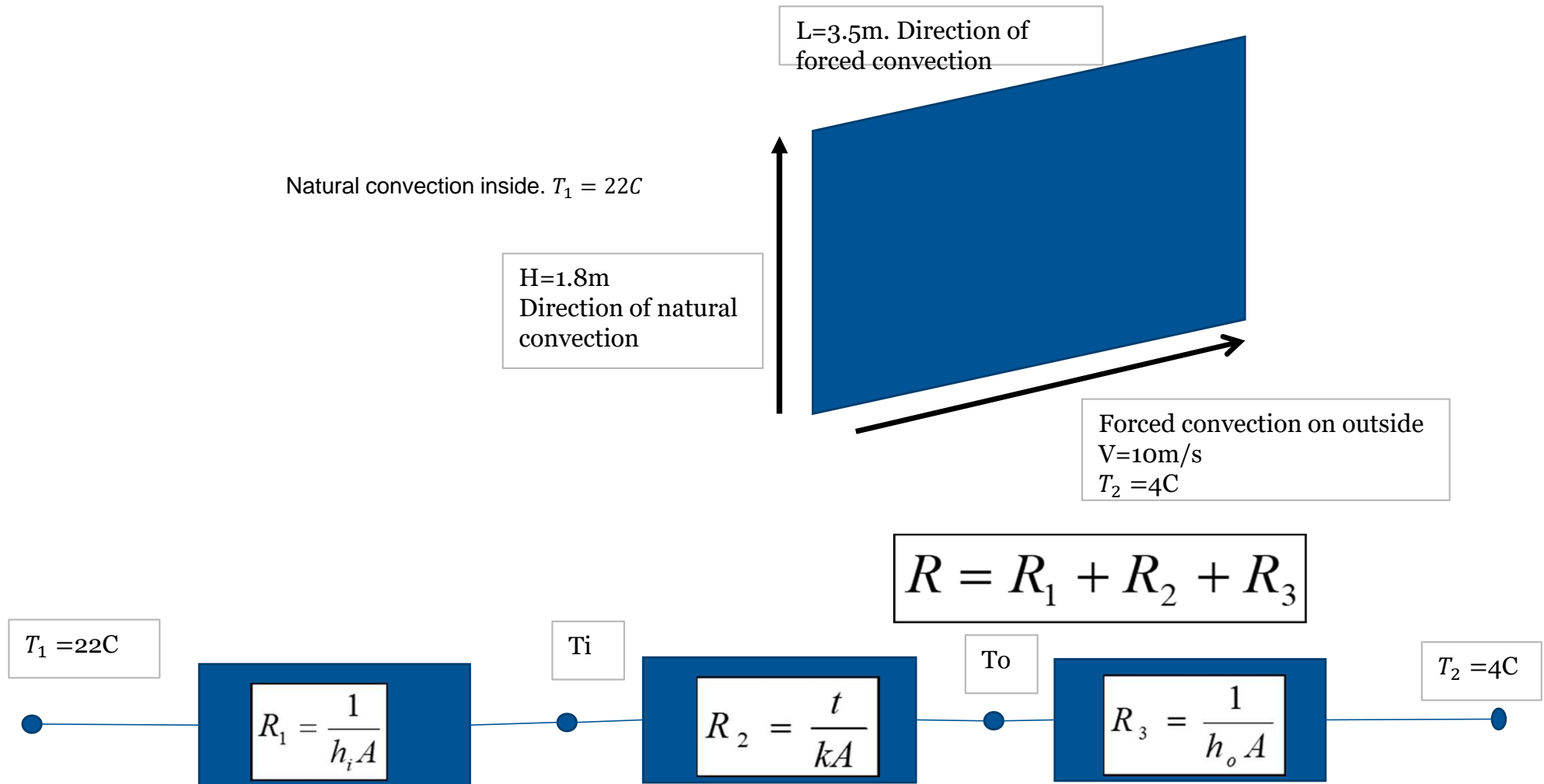
or by

$$\bar{h} = 0.95 (\Delta T)^{1/3} \quad \text{for } Gr_H > 1E9$$

Take the thermal conductivity of glass as 0.78 W/mK and use properties of air from the tables provided. Iteration to a fully converged solution is not expected, but a clear and complete explanation of the necessary procedure should be given. [14]



Visualise problem



Direction of convection is different on inside and outside.

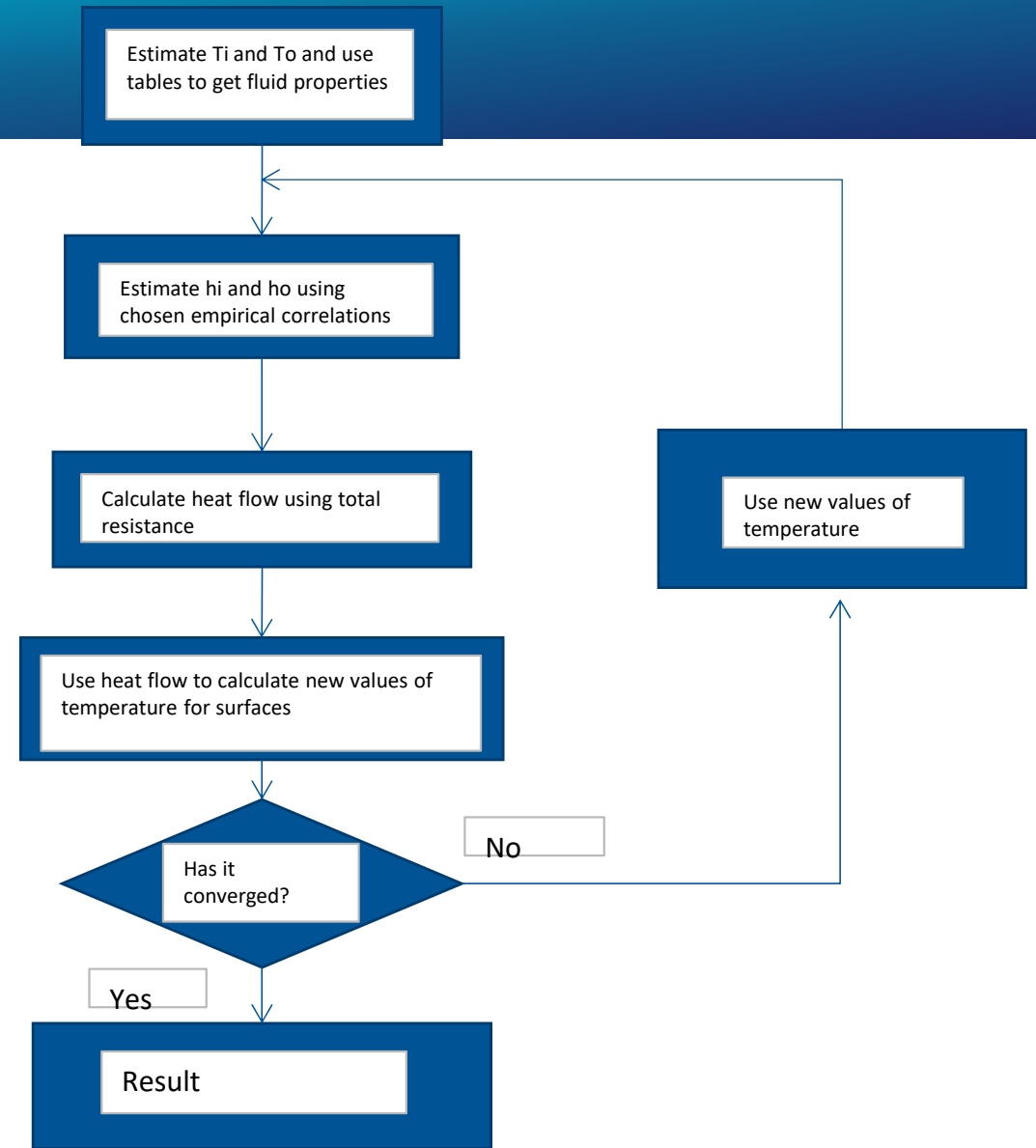
No idea of surface temperatures. So estimate for first pass and then iterate to solution.

h_o does not change, so use as fixed value.



Flow chart of procedure

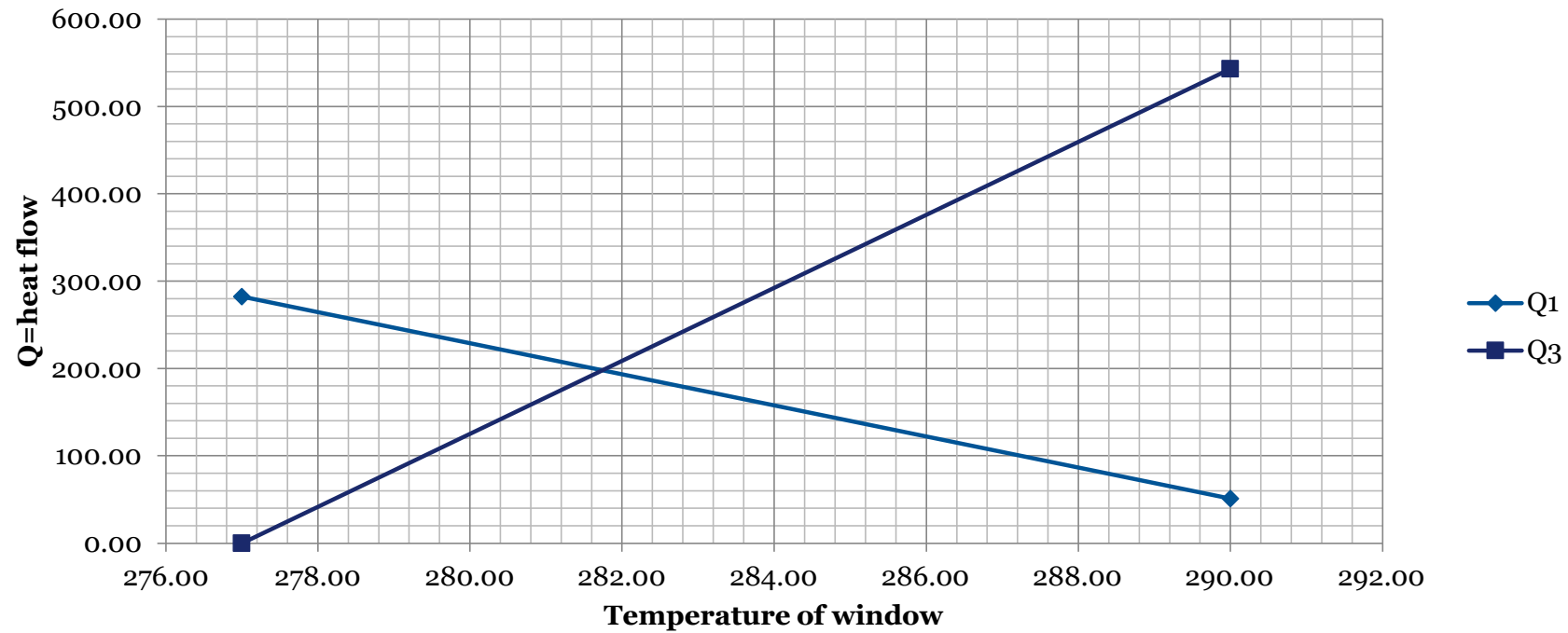
- Assume $T_i = T_o = 13^\circ\text{C}$: half way between v values
- Film temperature
 $T_f = 18.5^\circ\text{C} \sim 291.5\text{K}$
- Assume Film temperature
 $T_o = 7^\circ\text{C} \sim 280\text{K}$
- $\nu_i = 1.456 \times 10^{-5} \text{ m}^2/\text{s}$
- $\text{Pr} = 0.71$
- $k_i = 2.526 \times 10^{-5} \text{ kW/mK}$
- Error in using values for 288K for outer correlation is not significant, so let's do that.



	T_i	h_i	T_o	h_o	R	Q	T_{i+1}	T_{o+1}
Iteration 1	286.00	1.82	286.00	6.63	0.1125	159.98	281.03	280.83
Iteration 2	281.03	2.29	280.83	6.63	0.0945	190.40	281.79	281.56
Iteration 3	281.79	2.25	281.56	6.63	0.0959	187.79	281.73	281.50
Iteration 4	281.73	2.25	281.50	6.63	0.0957	188.02	281.73	281.50
Iteration 5	281.73	2.25	281.50	6.63	0.0957	188.00	281.73	281.50



Alternative graphical technique

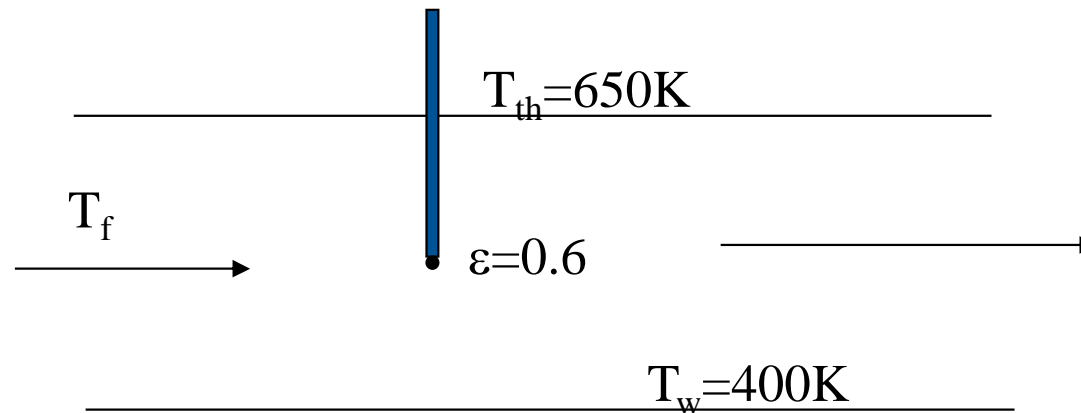


Alternative	Q1	Q3
Try 1	290.00	459.35
Try 2	277.00	459.35
Try 3	288.00	542.86
Try 4	288.00	0.00



Temperature of a thermocouple: is it right?

- A thermocouple used to measure the flow of hot air flowing in a duct whose walls are maintained at $T_w=400$ K shows a temperature reading of $T_{th}=650$ K. Assuming the emissivity of the thermocouple junction to be $e=0.6$ and the convection heat transfer coefficient to be $h=80$ W/m²K, determine the actual temperature of the air.



- The walls of the duct are at a considerably lower temperature than the air in it. Thus we expect the thermocouple to show a reading lower than the actual air temperature as a result of the radiation effect. The actual air temperature is determined as follows.

Consider the heat flow: At steady state:

heat flow into the thermocouple = heat flow from the thermocouple

OR

Heat flow IN due to convection = heat flow OUT due to radiation.



- Heat flow IN due to convection = heat flow OUT due to radiation can be written mathematically as:

$$\dot{q}_{conv} = \dot{q}_{rad}$$

$$h(T_f - T_{th}) = \varepsilon_{th} \sigma (T_{th}^4 - T_w^4)$$

$$\Rightarrow T_f = T_{th} + \frac{\varepsilon_{th} \sigma (T_{th}^4 - T_w^4)}{h}$$

$$\rightarrow 650 + \frac{0.6(5.67 E - 8)[(650^4 - 400^4)]}{80}$$

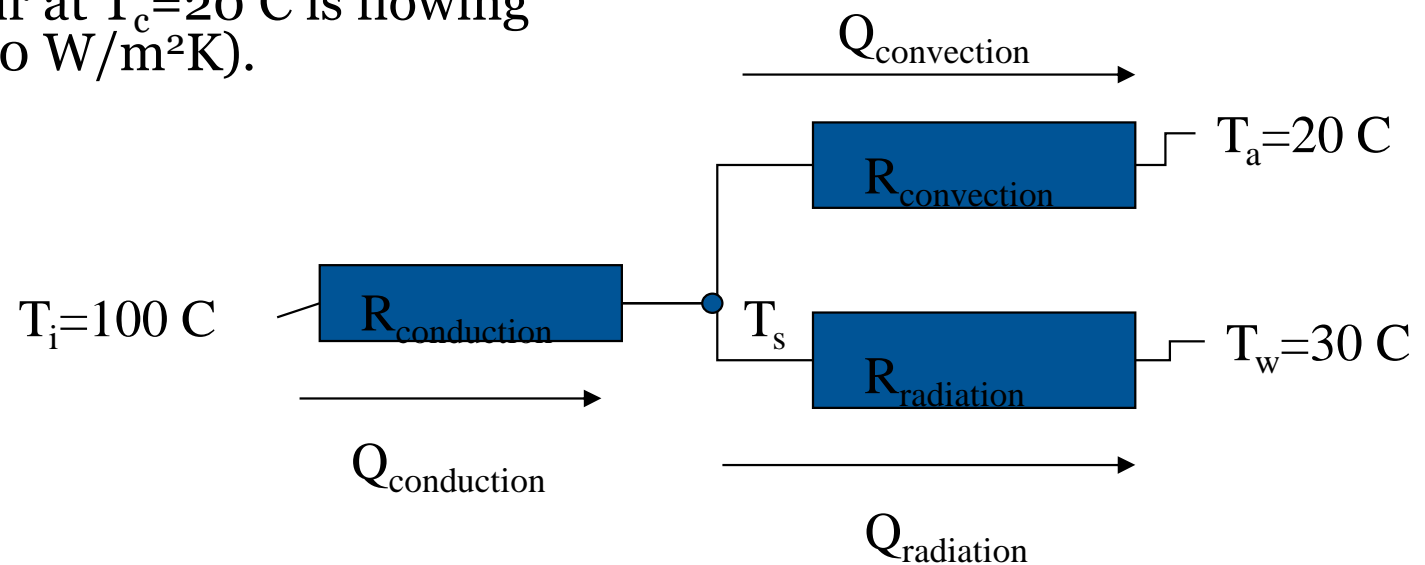
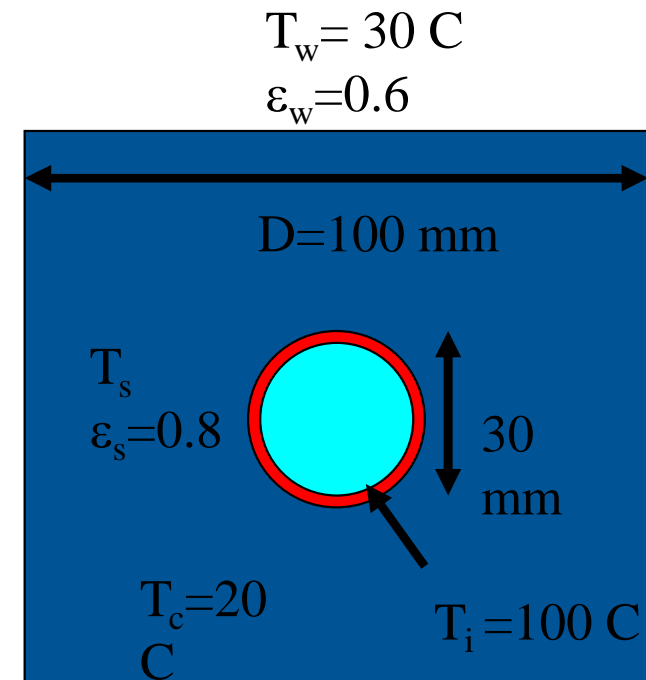
$$T_f = 715 K$$

So actual temperature of the air is 65 K larger than temperature of the thermocouple due to radiation effects. BEWARE!!!.



Cooling due to radiation

- A steam pipe filled with steam at 100 C runs through a square cross section enclosure as shown in the diagram to the right.
- The steam pipe has an emissivity of $\epsilon=0.8$, and the walls have an emissivity of $\epsilon=0.6$.
- Assume that the temperature of the inside wall of the cylinder is $T_i=100$ C, the temperature of the walls of the enclosure are kept at a constant temperature of $T_w=30$ C
- The inner diameter of the steam pipe is 25 mm and the pipe is made from copper ($k=401$ W/mK).
- What is the temperature of the surface of the steam pipe (T_s) if air at $T_c=20$ C is flowing along the pipe ($h=60$ W/m²K).





Heat gain to outside of pipe flow through wall due to conduction *use cylindrical version)

$$\dot{Q}_{conduction} = \frac{(T_i - T_s)}{\frac{\ln(r_2 / r_1)}{2\pi Lk}}$$

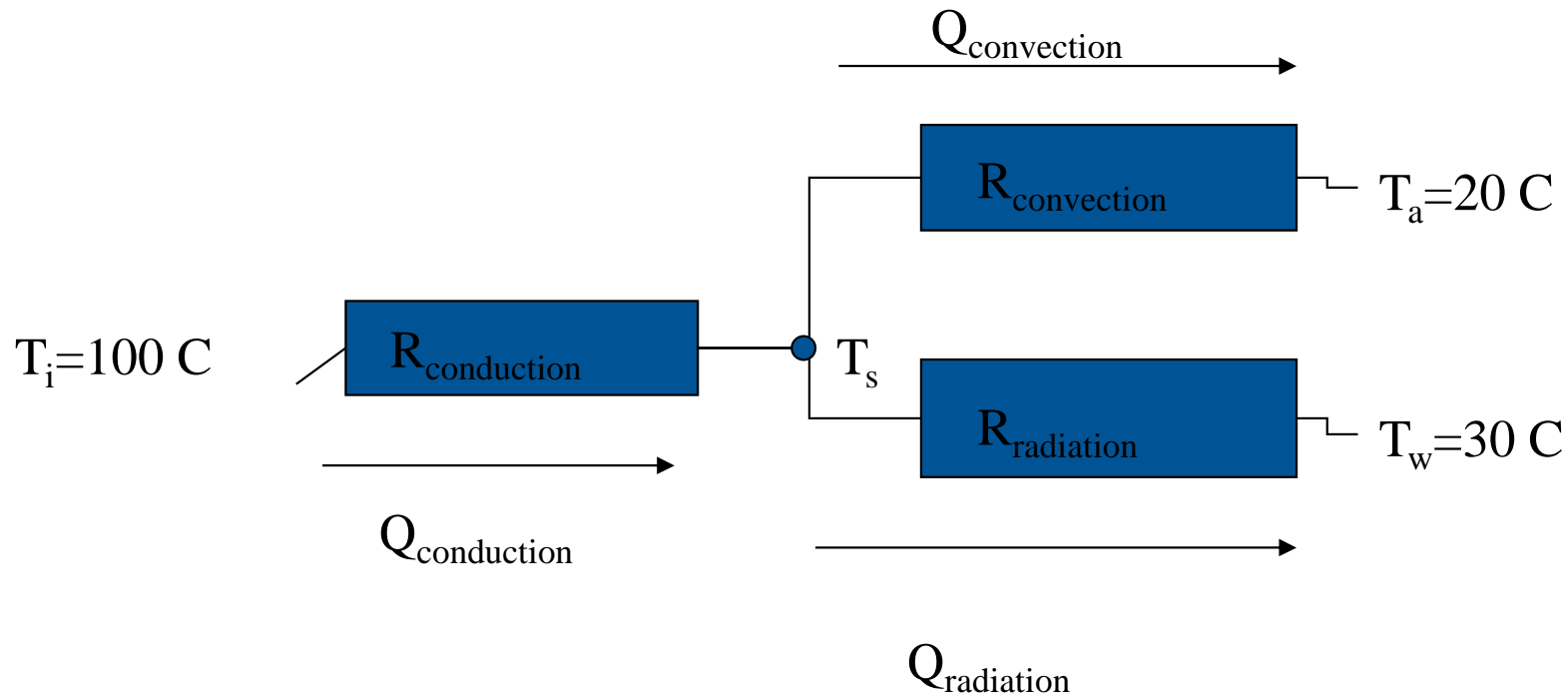
Heat loss from outside of pipe due to convection

$$\dot{Q}_{convection} = h(2\pi r_2 L)(T_s - T_c)$$

Heat loss from outside of pipe due to radiation losses.

$$\dot{Q}_{radiation} = \frac{\sigma(T_s^4 - T_w^4)}{\frac{1 - \epsilon_s}{A_s \epsilon_s} + \frac{1}{A_s F_{sw}} + \frac{1 - \epsilon_w}{A_w \epsilon_w}}$$

$A_s = 2\pi r_2 L$, $A_w = 4 * D * L$, $F_{sw} = 1$, Assume unit length of pipe, so $L = 1m$.



$$\dot{Q}_{\text{conduction}} = \dot{Q}_{\text{radiation}} + \dot{Q}_{\text{convection}}$$

$$\dot{Q}_p = \frac{(T_i - T_s)}{R_p} = \frac{(T_i - T_s)}{\frac{\ln(r_o / r_i)}{2\pi Lk}} \quad \dot{Q}_r = \frac{\sigma(T_s^4 - T_w^4)}{GF_{sw}} = \frac{\sigma(T_s^4 - T_w^4)}{\frac{1 - \epsilon_s}{A_s \epsilon_s} + \frac{1}{A_s F_{sw}} + \frac{1 - \epsilon_w}{A_w \epsilon_w}} \quad \dot{Q}_c = \frac{(T_s - T_a)}{R_c} = \frac{(T_s - T_a)}{1 / hA_w}$$



Thermal resistances	value	unit
R _p	7.23625E-05	K/W
R _c	0.176838826	K/W
GF _{sw}	14.92957859	m ²

Table : Thermal resistance values as defined in the three equations as shown.

Now the problem has been defined, we can generate a few equations. The problem is that in order to determine the heat flows, we need the temperature at the outer surface of the pipe, which is not known. We can however use the energy balance equation to estimate it using an iterative of graphical technique.

$$Q_p + Q_c + Q_r = 0 \rightarrow \frac{(T_i - T_s)}{R_p} + \frac{(T_a - T_s)}{R_c} + \frac{\sigma(T_w^4 - T_s^4)}{GF_{sw}} = 0 \rightarrow T_s = \left(\frac{R_p R_c}{R_p + R_c} \right) \left(\frac{T_a}{R_c} + \frac{T_i}{R_p} + \frac{\sigma(T_w^4 - T_s^4)}{GF_{sw}} \right)$$



- We now have the temperature of the surface of the pipe on both sides of the equation. You can solve this three ways, by analytical calculation, by iteration and by graphical techniques.
- By iteration we first guess a value of T_s and then use it on the RHS to get a new estimate of T_s .

iteration	Guess	new guess	heat flow through pipe
0	273.000	372.968	1381930.6
1	372.968	372.964	441.3
2	372.964	372.964	493.7
3	372.964	372.964	493.7
4	372.964	372.964	493.7
5	372.964	372.964	493.7

Table : results of iteration .

By iteration, the surface temperature is 372.9K and the heat flow through the pipe is about 493W. This technique is robust and will always work.

- Another technique is by graphical means. The high thermal resistances of the radiation and convection compared to the conduction through the pipe suggest that the temperature is close to that of the steam in the inside.
- If we take a selection of temperature value, we can plot the heat through the pipe and the heat from the pipe on a graph and look for where they intersect.

Temperature	heat flow to pipe surface	heat flow from pipe surface
373	0	493
372.5	6909	490

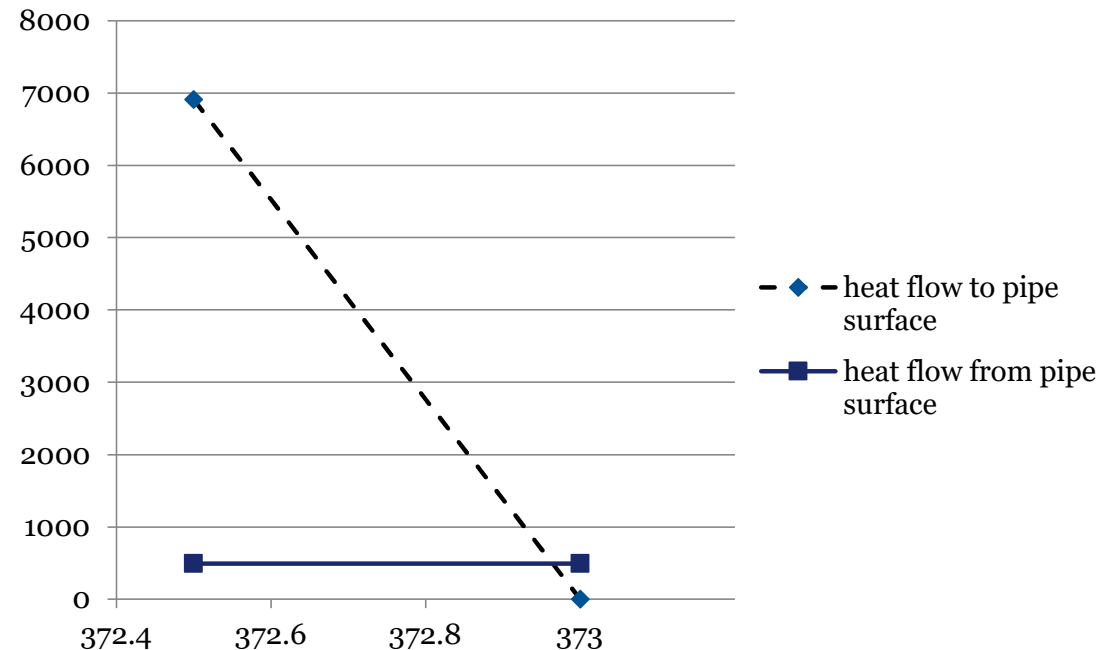


Table 5: results of temperature selection.

The two lines intersect close to 372.9K at Q approximately 492 W. You can add other points to refine this if you wish. This technique has the advantage that you know how many calculations you should attempt, while the iterative technique can take more iterations, but be more accurate.

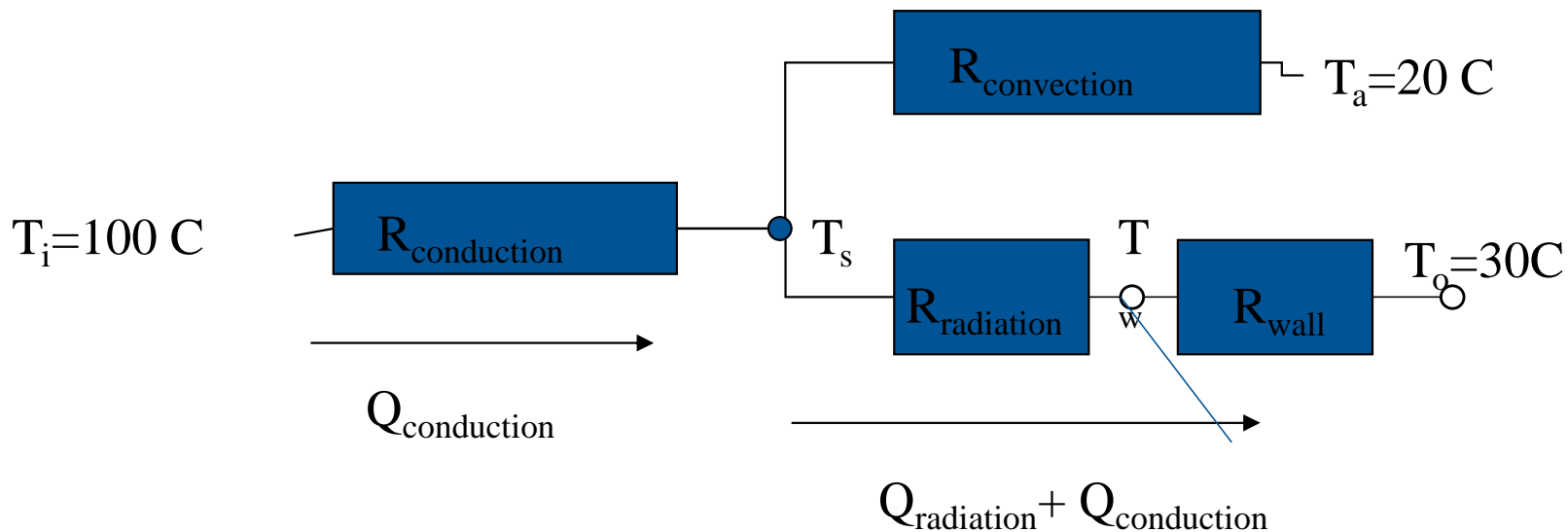
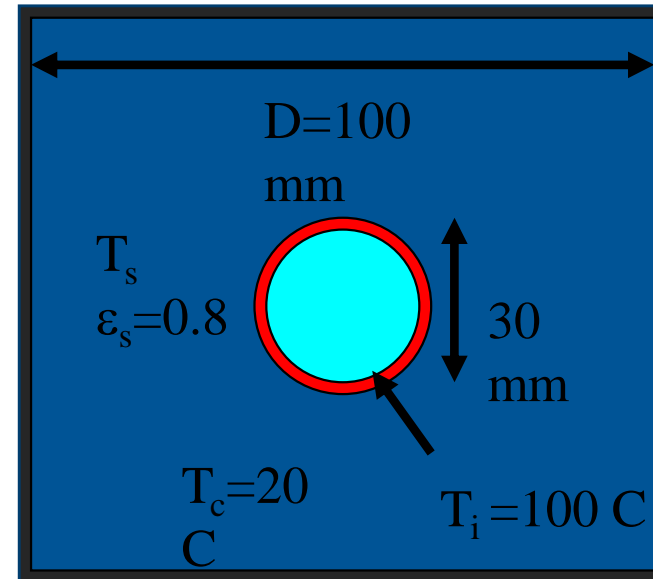


$$T_w = 30 \text{ C } t = 10 \text{ cm}$$

$$\epsilon_w = 0.6 \text{ k} = 1 \text{ W/m}^2\text{K}$$

- In the previous example the wall was at a constant temperature.
- How can we solve this if we add conduction through the wall.
- Heat flow is not the best method of solution now. Must use a new technique.

$$\dot{Q}_p = \dot{Q}_{R+W} + \dot{Q}_c$$





Thermal resistances	value	unit
Rp	7.23625E-05	K/W
Rc	0.176838826	K/W
GFsw	14.92957859	m ²
Rw	0.25	K/W

$$\dot{Q}_r = \frac{(T_s - T_w)}{R_r} = \frac{(T_s - T_w)}{\left(\frac{1 - \epsilon_s}{A_s \epsilon_s} + \frac{1}{A_s F_{sw}} + \frac{1 - \epsilon_w}{A_w \epsilon_w} \right) / \sigma (T_s^2 + T_w^4)(T_s + T_w)}$$

$$\dot{Q}_w = \frac{(T_o - T_w)}{R_w} = \frac{(T_o - T_w)}{t / k_w A_w}$$

Table 3: Thermal resistance values and new equations

$$Q_p + Q_c + Q_r = 0 \rightarrow \frac{(T_i - T_s)}{R_p} + \frac{(T_a - T_s)}{R_c} + \frac{(T_o - T_s)}{R_r + R_w} = 0 \rightarrow T_s = \left(\frac{R_p R_c (R_r + R_w)}{R_p + R_c + R_r + R_w} \right) \left(\frac{T_a}{R_c} + \frac{T_i}{R_p} + \frac{(T_o)}{R_r + R_w} \right)$$

We have to a real problem here, we need to estimate the thermal resistance of the radiation from the temperatures first before working out the temperatures. This can then be used with the above equation to obtain a new guess for the surface temperature of the pipe.

After getting the surface temperature of the pipe, you can obtain the heat flow through radiation and conduction.

Once this is derived, you can use it to obtain the temperature of the walls.



$$R_r = \frac{\left(\frac{1 - \epsilon_s}{A_s \epsilon_s} + \frac{1}{A_s F_{sw}} + \frac{1 - \epsilon_w}{A_w \epsilon_w} \right)}{\sigma (T_s^2 + T_w^4)(T_s + T_w)} \quad \dot{Q}_{rw} = \frac{(T_o - T_s)}{R_w + R_r}$$

iteration	Guess Ts	Guess Tw	Rr (using guesses)	New Ts	Qp (for show)	Qrw: use new Ts and Rr guess	New Tw
0	300.000	373.000	1.708	373.0	487.93	35.741	311.935
1	372.965	311.935	1.626	373.0	489.48	37.290	312.323
2	372.965	312.323	1.624	373.0	489.53	37.342	312.335
3	372.965	312.335	1.624	373.0	489.53	37.343	312.336
4	372.965	312.336	1.624	373.0	489.53	37.343	312.336
5	372.965	312.336	1.624	373.0	489.53	37.343	312.336

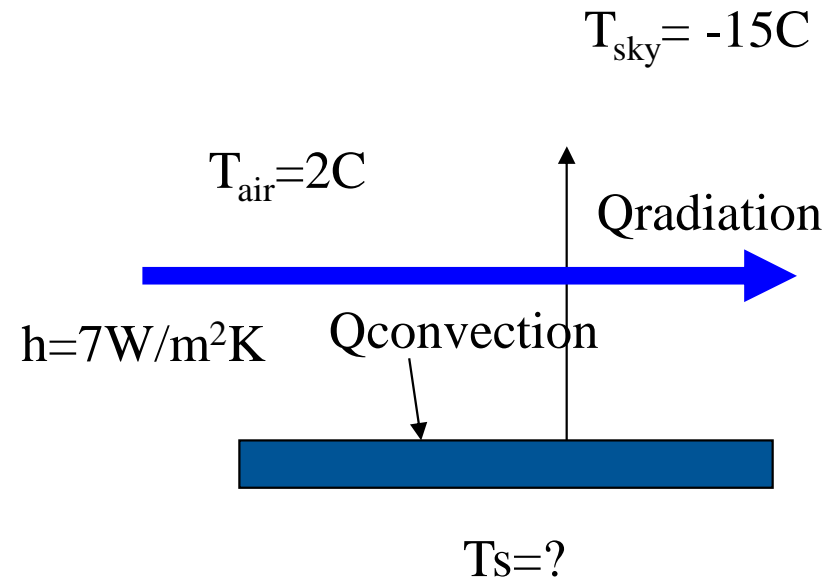
Table : results of iteration .

By iteration, the surface temperature is 372.9K and the heat flow through the pipe is about 490W. Very little has changed although the walls are slightly warmer. We should have added convection on the inside walls as well to make this robust.



Does Frost form differently on a white compared to a black patio paving stone?

- Flow of air over paving stone.
- Imaging two slabs one white ($\epsilon=0.1$) and one black ($\epsilon=0.9$).
- Assume area of slab is $A=1\text{m}^2$. and sky is blackbody.
- Will frost form on the surface of the slab?(is T_s lower than 0C ?)



Energy gain to paving slab due to convection:

$$\dot{q}_{\text{convection}} = hA(T_s - T_{\text{air}})$$

Energy loss from paving slab due to radiation:

$$\dot{q}_{\text{radiation}} = \sigma F_{12} (T_s^4 - T_{\text{sky}}^4)$$

Since Area of sky \gg Area of slab, $F_{12} = \epsilon_1 A$

- Can use Iterative technique and graphical technique.
- See attached sheets



$$Q_c = Q_r = Q = hA(T_a - T_s) = h_{rad}A(T_s - T_{sky})$$

$$T_{s+1} = \frac{hT_a + h_{rad}T_{sky}}{h + h_{rad}}$$

$$h_{rad} = \sigma \mathfrak{F}_{12} (T_s^2 + T_{sky}^2)(T_s + T_{sky})$$

$$\mathfrak{F}_{12} = \frac{1}{\frac{1-\epsilon}{\epsilon A} + \frac{1}{F_{12}A} + \frac{1-\epsilon_2}{\epsilon_2 A_2}}$$

Guess value as half air-sky temperature

White slab
 $\epsilon=0.1$

	ϵ	ϵ_2	F_{12}	\mathfrak{F}_{12}	A (m ²)	Estimated T_s (K)	T_{sky} (K)	T_a (K)	Estimated h_{rad} (W/m ² K)	h (W/m ² K)	New estimate T_{s+1} (K)	Q_c (W)	Q_r (W)
First guess	0.1	1	1	0.1	1	266	258	274	0.408	7	273.119	56.000	3.264
	0.1	1	1	0.1	1	273.12	258	274	0.425	7	273.084	6.168	6.427
	0.1	1	1	0.1	1	273.08	258	274	0.425	7	273.084	6.412	6.411
	0.1	1	1	0.1	1	273.08	258	274	0.425	7	273.084	6.411	6.411
	0.1	1	1	0.1	1	273.08	258	274	0.425	7	273.084	6.411	6.411
	0.1	1	1	0.1	1	273.08	258	274	0.425	7	273.084	6.411	6.411
	0.1	1	1	0.1	1	273.08	258	274	0.425	7	273.084	6.411	6.411



- White slab energy radiated 6.8W Temperature of slab 1 C
- Black slab energy radiated 41.4W Temperature of slab -4 C.
- Frost will form only on black slabs!!!



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Participating media

Why does your beer change colour based on how much
is left in the glass.....

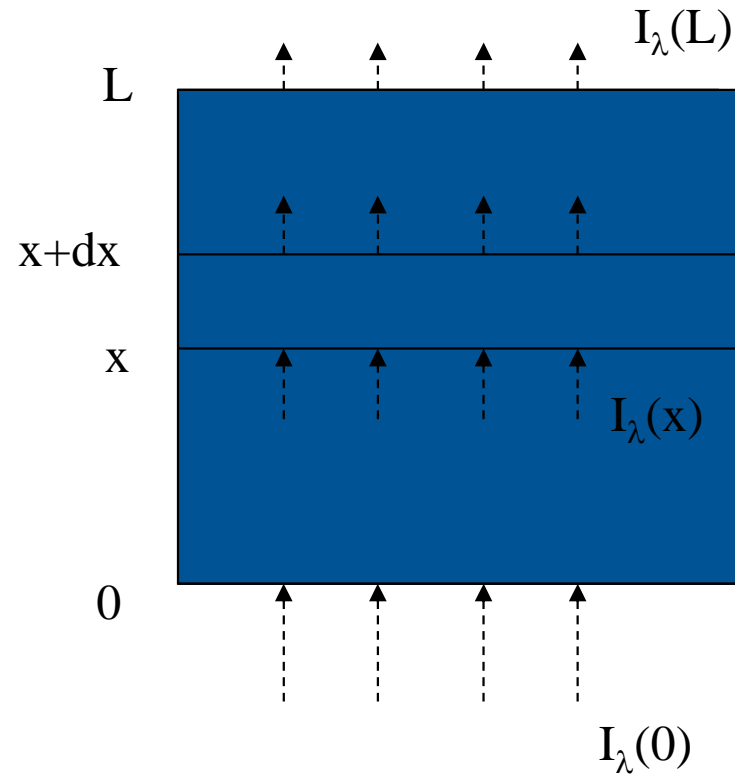
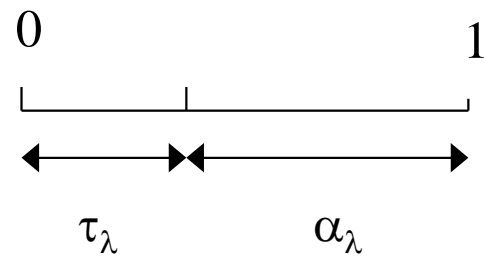
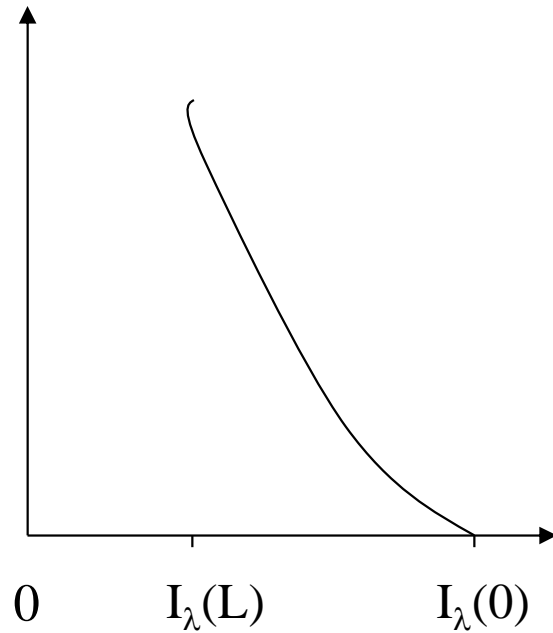


- What happens when the medium absorbs or scatters radiation?
- Ozone for example?
- Water vapour for example (clouds)?
- What is Beer's Law?

- All the work to date were based on the assumption that the medium is transparent.
- This is generally true for a vacuum, monatomic molecules and symmetrical diatomic molecules.
- E.g. $N_2, O_2, H_2, Ne, Ar, Xe$.
- When the medium absorbs and scatters light this model of non-participation is not valid.
- Strongly polar gases which exhibit asymmetry are prime examples of that.
- E.g. $H_2O, CO_2, NH_3, O_3, CO, SO_2, NO$
- These are called **PARTICIPATING MEDIA**.



The attenuation of the monochromatic radiation that penetrates a volumetrically absorbing medium





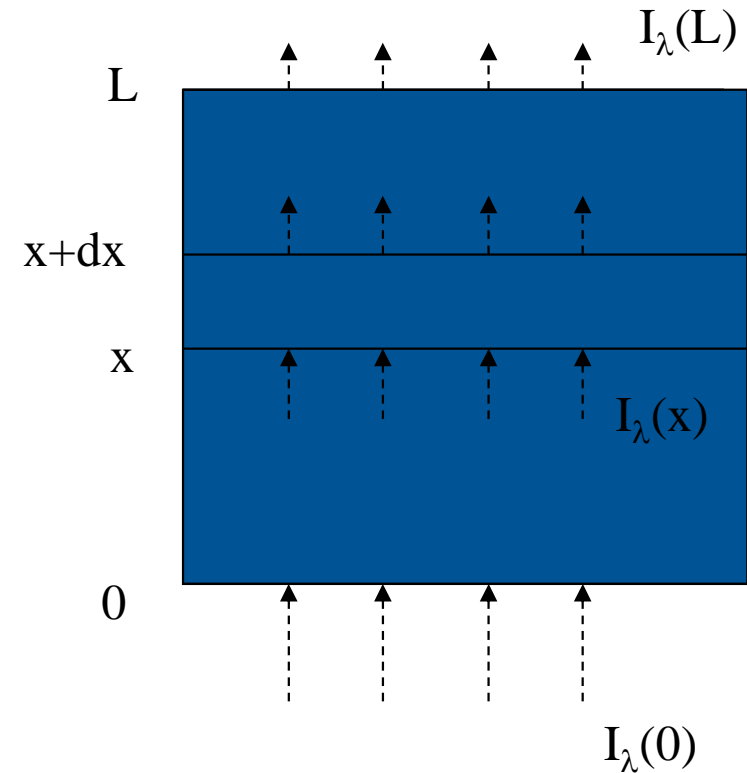
Experiments have shown that the local attenuation dI_λ is proportional to the local intensity of the beam I_λ .

$$dI_\lambda = -\kappa_\lambda I_\lambda dx$$

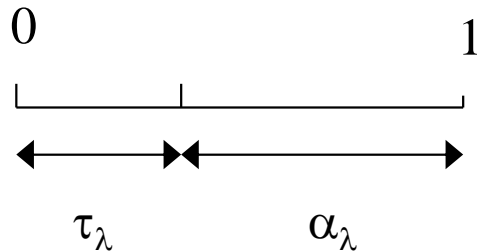
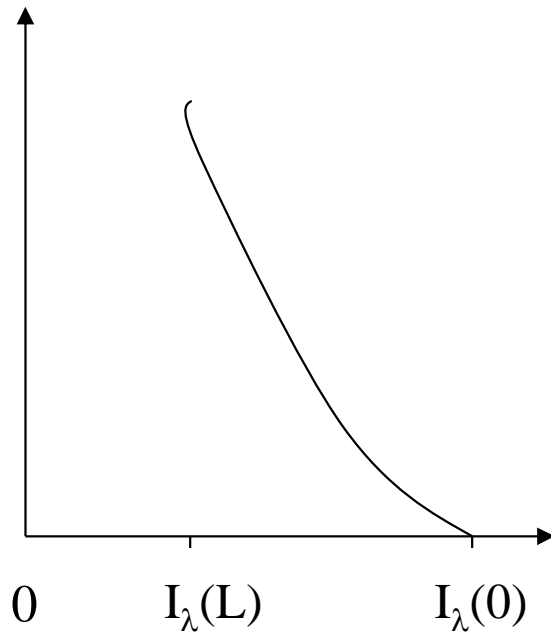
This proportionality defines the **MONOCHROMATIC EXTINCTION COEFFICIENT** κ_λ which has units m^{-1}

Integrating we get:

$$I_\lambda(x) = I_\lambda(0) \exp(-\kappa_\lambda L)$$



This is known as **Beer's Law**.



Starting with Beer's Law

$$I_\lambda(x) = I_\lambda(0) \exp(-\kappa_\lambda L)$$

We note that the fraction of incident radiation that escapes (transmitted) the other side ($x=L$) is

$$\tau_\lambda = \frac{I_\lambda(L)}{I_\lambda(0)} = \exp(-\kappa_\lambda L)$$

This fraction is less than 1 and represents the monochromatic transmissivity of the layer. The remaining fraction that is absorbed in the L -thick layer is:

$$\alpha_\lambda = 1 - \tau_\lambda = 1 - \exp(-\kappa_\lambda L)$$

Gases do not reflect the radiation that passes through them.



If temperature T_g is uniform and does not differ much from the temperature T_s of the surface we may invoke Kirchhoff's law.

$$\varepsilon_\lambda = \alpha_\lambda = 1 - \exp(-\kappa_\lambda L)$$

ε_λ is the monochromatic emissivity of the layer.

In reality we must account for every band of wavelengths that contribute to the process. It is more convenient to work with total quantities. So the gas transmissivity τ_g is given by integrating over all wavelengths.

$$\tau_g = \frac{\int_0^\infty I_\lambda(L) d\lambda}{\int_0^\infty I_\lambda(0) d\lambda}$$

And $\varepsilon_g = \alpha_g$ (if $T_g = T_s$, i.e., if Kirchhoff's law applies)



Gas mixture contains only one participating component (CO_2 for example). Other components are non-participating (transparent).

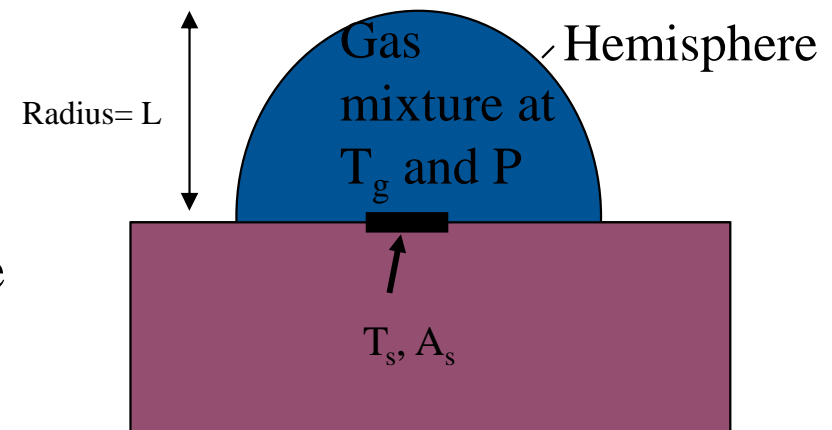
Gas mixture is at uniform temperature T_g and total pressure P . Partial pressure of carbon dioxide is P_c . Radius of Hemisphere is L .

The radiation heat current emitted by gas and arriving at A_s is given by (assume that A_s is a blackbody)

$$q_{g \rightarrow s} = \varepsilon_g A_s \sigma T_g^4$$

$$\varepsilon_g = f_c(T_g, P_c L, P)$$

This is experimentally measured and you use graphs to obtain value.





Charts show that the emissivity increases as the product $P_c L$ increases.

Increasing the partial pressure or increasing the volume of gas will increase the number of CO_2 molecules that emit towards target A_s .

Similarly increasing pressure increases emissivity.



Consider reverse interaction between participating gas and surface A_s

A fraction of the energy q_a is absorbed by the gas is a fraction of the energy emitted by the surface

$$q_a = \alpha_g q_{s \rightarrow g}$$

Absorptivity is then related to:

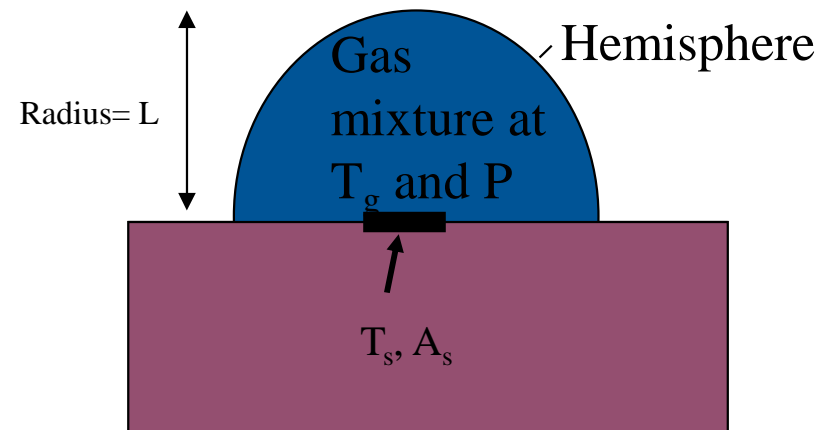
$$\alpha_{CO_2} = f_c \left(T_s, P_c L \frac{T_s}{T_g}, P \right) \times \left(\frac{T_g}{T_s} \right)^{0.65}$$

$$\alpha_{w vapour} = f_c \left(T_s, P_w L \frac{T_s}{T_g}, P + P_w \right) \times \left(\frac{T_g}{T_s} \right)^{0.45}$$

The radiation heat current emitted by A_s is given by (assume that A_s is a blackbody)

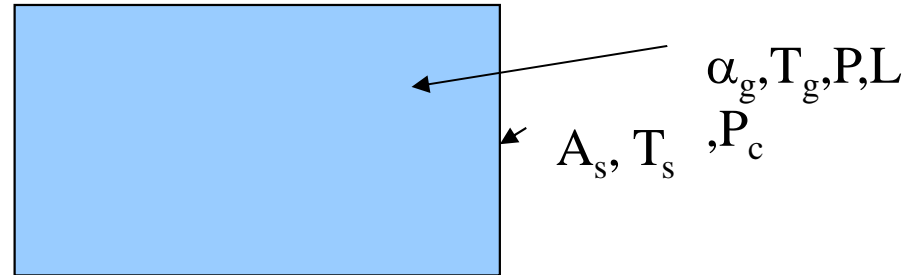
Energy emitted from surface is a fraction of the black body emission.

$$q_{s \rightarrow g} = A_s \sigma T_s^4$$





Gas surrounded by a black surface



If the internal surface A_s is black, it absorbs all the radiation emitted from the gas. ϵ_g is determined from T_g, P, L, P_c

$$q_{g \rightarrow s} = \epsilon_g \sigma T_g^4 A_s$$

The radiation heat current emitted by A_s is : $\sigma T_s^4 A_s$

The portion absorbed by the entire gas volume is:

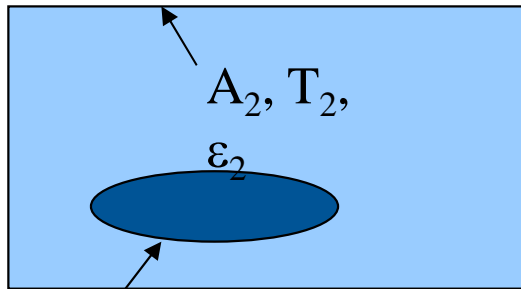
$$q_{s \rightarrow g} = \alpha_g \sigma T_s^4 A_s$$

The instantaneous net rate of heat transfer from the gas to the black enclosure is:

$$\begin{aligned} q_{g-s} &= q_{g \rightarrow s} - q_{s \rightarrow g} \\ &= \sigma A_s (\epsilon_g T_g^4 - \alpha_g T_s^4) \end{aligned}$$



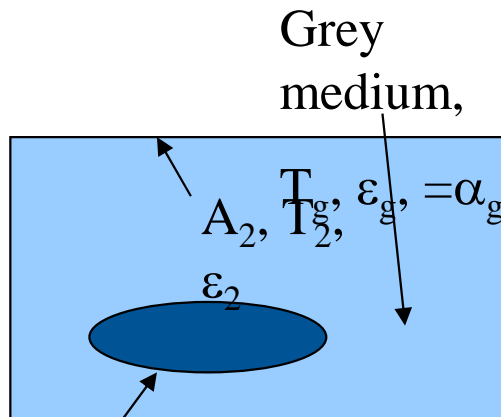
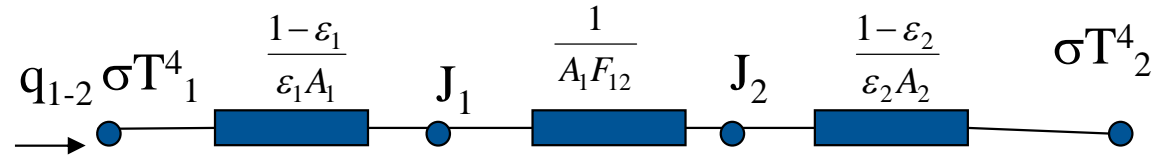
Adding an internal surface and participating gas between



A_1, T_1, ϵ_1

Two surface gray body problem: no participating media

This is a two body problem, so resistance network is given by:

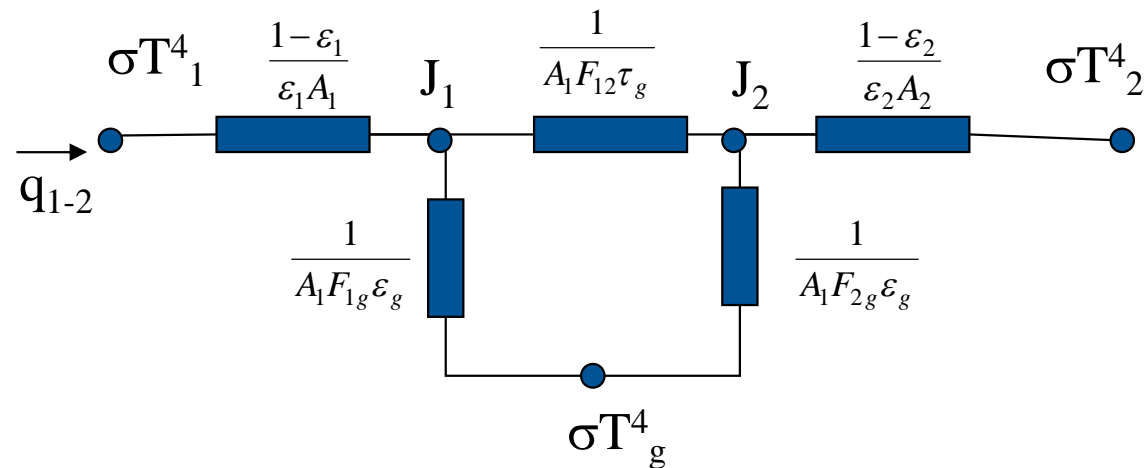


A_1, T_1, ϵ_1

Consider a two surface enclosure as shown in the figure. Each surface is diffuse gray and isothermal. Whereas the medium is isothermal and gray.

An observer on A_1 sees both the medium and the enclosure so this is a three body problem.

Two surface gray body problem with participating medium





Welding! What happens if you have a participating media.

