22CGA006
Heat Transfer

Semester 2 2022/23 | In-Person Exam paper

This examination is to take place in-person at a central University venue under exam conditions. The standard length of time for this paper is **2 hours**.

You will not be able to leave the exam hall for the first 30 or final 15 minutes of your exam. Your invigilator will collect your exam paper when you have finished.

Help during the exam

Invigilators are not able to answer queries about the content of your exam paper. Instead, please make a note of your query in your answer script to be considered during the marking process.

If you feel unwell, please raise your hand so that an invigilator can assist you.

You may use a calculator for this exam. It must comply with the University's Calculator Policy for In-Person exams, in particular that it must not be able to transmit or receive information (e.g. mobile devices and smart watches are **not** allowed).

Answer **THREE** questions in total. Each question carries 25 marks.

Candidates should show full working for calculations and derivations.

A formula sheet is provided at the end of this exam paper.

1. (a) Hot air enters a rectangular duct (length: 5 m, cross section: 20 cm x 25 cm) at 100 kPa and 60°C with the average velocity of 5 m s^{-1}. The temperature of the hot air in the duct drops to 54°C as a result of heat loss to the surrounding cool air.

(i) Provide the problem statement; sketch a labelled diagram of this problem and list relevant information on it; identify and list the suitable properties from Table Q1 needed for this problem. [4 marks]

(ii) Determine the rate of heat loss from the hot air in the duct to the surrounding cool air under steady conditions. [7 marks]

(iii) State any assumption(s) made. [2 marks]

(b) The temperatures of the inner and the outer surfaces of a wall (height: 3 m, width: 5 m, thickness: 0.3 m) are measured to be 16°C and 2°C respectively.

(i) Provide the problem statement, sketch a labelled diagram of this problem and list relevant information on it. [3 marks]

(ii) Use TWO different methods (i.e. with or without the calculation of conduction thermal resistance) to determine the rate of heat transfer through the wall under steady conditions. [6 marks]

(iii) State any assumption(s) made. [3 marks]

<u>Relevant Data</u>

Thermal conductivity of the wall is $k = 0.9\ \mathrm{W \cdot m^{-1} \cdot K^{-1}}$

Continued/…

Q1 Continued/…

Table Q1

Properties of air at 1 atm pressure

Temp. T, °C	Density ρ, kg/m^3	Specific Heat C_p, J/kg · K	Thermal Conductivity k, W/m · K	Thermal Diffusivity α, m^2/s	Dynamic Viscosity μ, $kg/m \cdot s$	Kinematic Viscosity ν, m^2/s	Prandtl Number Pr
−150	2.866	983	0.01171	4.158×10^{-6}	8.636×10^{-6}	3.013×10^{-6}	0.7246
−100	2.038	966	0.01582	8.036×10^{-6}	1.189×10^{-5}	5.837×10^{-6}	0.7263
−50	1.582	999	0.01979	1.252×10^{-5}	1.474×10^{-5}	9.319×10^{-6}	0.7440
−40	1.514	1002	0.02057	1.356×10^{-5}	1.527×10^{-5}	1.008×10^{-5}	0.7436
−30	1.451	1004	0.02134	1.465×10^{-5}	1.579×10^{-5}	1.087×10^{-5}	0.7425
−20	1.394	1005	0.02211	1.578×10^{-5}	1.630×10^{-5}	1.169×10^{-5}	0.7408
−10	1.341	1006	0.02288	1.696×10^{-5}	1.680×10^{-5}	1.252×10^{-5}	0.7387
0	1.292	1006	0.02364	1.818×10^{-5}	1.729×10^{-5}	1.338×10^{-5}	0.7362
5	1.269	1006	0.02401	1.880×10^{-5}	1.754×10^{-5}	1.382×10^{-5}	0.7350
10	1.246	1006	0.02439	1.944×10^{-5}	1.778×10^{-5}	1.426×10^{-5}	0.7336
15	1.225	1007	0.02476	2.009×10^{-5}	1.802×10^{-5}	1.470×10^{-5}	0.7323
20	1.204	1007	0.02514	2.074×10^{-5}	1.825×10^{-5}	1.516×10^{-5}	0.7309
25	1.184	1007	0.02551	2.141×10^{-5}	1.849×10^{-5}	1.562×10^{-5}	0.7296
30	1.164	1007	0.02588	2.208×10^{-5}	1.872×10^{-5}	1.608×10^{-5}	0.7282
35	1.145	1007	0.02625	2.277×10^{-5}	1.895×10^{-5}	1.655×10^{-5}	0.7268
40	1.127	1007	0.02662	2.346×10^{-5}	1.918×10^{-5}	1.702×10^{-5}	0.7255
45	1.109	1007	0.02699	2.416×10^{-5}	1.941×10^{-5}	1.750×10^{-5}	0.7241
50	1.092	1007	0.02735	2.487×10^{-5}	1.963×10^{-5}	1.798×10^{-5}	0.7228
60	1.059	1007	0.02808	2.632×10^{-5}	2.008×10^{-5}	1.896×10^{-5}	0.7202
70	1.028	1007	0.02881	2.780×10^{-5}	2.052×10^{-5}	1.995×10^{-5}	0.7177
80	0.9994	1008	0.02953	2.931×10^{-5}	2.096×10^{-5}	2.097×10^{-5}	0.7154

2. Figure Q2 shows the flow of oil at 20°C in a pipeline (diameter: 0.3 m, length: 200 m) at an average velocity of 2 m s^{-1}. A section of the pipeline passes through a river at 0°C. Measurements indicate that the surface temperature of the pipe is very nearly 0°C, i.e., the heat transfer resistance on the outside of the pipe is comparatively small and can be neglected.

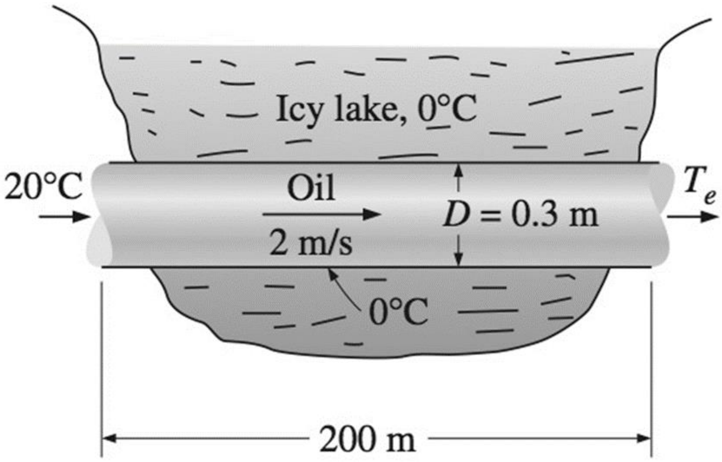

Figure Q2

(a) Provide the problem statement, identify and list the suitable properties from Table Q2 needed for this problem. [3 marks]

(b) Determine the Reynolds number and thermal entry length of this heat transfer problem. [3 marks]

(c) Determine the temperature of the oil at the point in the pipe where it leaves the river. [6 marks]

(d) Determine the rate of heat transfer from the oil. [3 marks]

(e) Determine the pumping power required to overcome the pressure losses and to maintain the flow of the oil in the pipe. [5 marks]

(f) State any assumption(s) made. [5 marks]

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Q2 Continued/…

Relevant Specific Equations

$$L_t \approx 0.05\mathrm{RePr}D$$

$$\mathrm{Nu} = \frac{hD}{k} = 3.66 + \frac{0.065(D/L)\,\mathrm{RePr}}{1 + 0.04[(D/L)\mathrm{RePr}]^{2/3}}$$

$$T_e = T_s - (T_s - T_i)\exp(-hA_s/\dot{m}C_p)$$

where the symbols have their usual meaning.

Relevant Data

Table Q2

Properties of liquids

Temp. T, °C	Density ρ, kg/m^3	Specific Heat C_p, J/kg · K	Thermal Conductivity k, W/m · K	Thermal Diffusivity α, m^2/s	Dynamic Viscosity μ, $kg/m \cdot s$	Kinematic Viscosity ν, m^2/s	Prandtl Number Pr	Volume Expansion Coefficient β, 1/K
Engine Oil (unused)								
0	899.0	1797	0.1469	9.097×10^{-8}	3.814	4.242×10^{-3}	46,636	0.00070
20	888.1	1881	0.1450	8.680×10^{-8}	0.8374	9.429×10^{-4}	10,863	0.00070
40	876.0	1964	0.1444	8.391×10^{-8}	0.2177	2.485×10^{-4}	2,962	0.00070
60	863.9	2048	0.1404	7.934×10^{-8}	0.07399	8.565×10^{-5}	1,080	0.00070
80	852.0	2132	0.1380	7.599×10^{-8}	0.03232	3.794×10^{-5}	499.3	0.00070
100	840.0	2220	0.1367	7.330×10^{-8}	0.01718	2.046×10^{-5}	279.1	0.00070
120	828.9	2308	0.1347	7.042×10^{-8}	0.01029	1.241×10^{-5}	176.3	0.00070
140	816.8	2395	0.1330	6.798×10^{-8}	0.006558	8.029×10^{-6}	118.1	0.00070
150	810.3	2441	0.1327	6.708×10^{-8}	0.005344	6.595×10^{-6}	98.31	0.00070

3. A counter-flow double-pipe arrangement is the design proposed for a heat exchanger to heat water from 30°C to 90°C at a rate of 2.4 kg s^{-1}. The heating is to be accomplished by a hot fluid continuously available at 170°C at a mass flow rate of 4 kg s^{-1}. The inner tube is thin-walled with a diameter of 20 mm, and the overall heat transfer coefficient of the heat exchanger is 800 W m^{-2} K^{-1}. If the specific heats of water and the hot fluid are 4.18 kJ kg^{-1} K^{-1} and 4.31 kJ kg^{-1} K^{-1}, respectively:

(a) Determine the rate of heat transfer between the water and the hot fluid. State all assumptions made. [5 marks]

(b) Determine the temperature at which the hot fluid exits the heat exchanger. Sketch the temperature profiles for both fluid streams. [4 marks]

(c) Determine the log-mean temperature difference for this counter-flow heat exchanger. [6 marks]

(d) Determine the length of the heat exchanger required to achieve the desired heating. [6 marks]

(e) Comment on the proposed design of the heat exchanger and, if necessary, suggest improvements. [4 marks]

4. In manufacturing, a special coating on a curved solar absorber surface of area A_2 = 15 m^2 is cured by exposing it to an infrared heater of width W = 1 m as shown in Figure Q4.1. The absorber and heater are each of length L = 10 m and are separated by a distance H = 1 m. The heater is at T_1 = 1000 K and has an emissivity of ε_1 = 0.9, while the absorber is at T_2 = 600 K and has an emissivity of ε_2 = 0.5. The system is in a large room whose walls are at 300 K. The room can be assumed to be a black body.

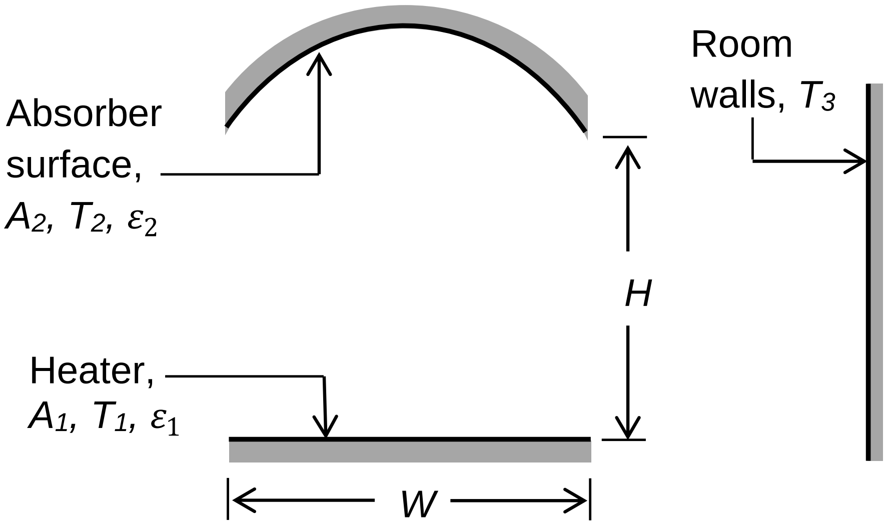

Figure Q4.1. The geometry of the described problem

(a) State all assumption(s) made and sketch the thermal resistance network of the system. [5 marks]

(b) Calculate the view factors for all surfaces in the system. [6 marks]

(c) Calculate the surface and space resistances for the system. [5 marks]

(d) Calculate the surface radiosities of the system. [7 marks]

(e) Determine the net rate of radiation heat transfer to the absorber surface. [2 marks]

<u>Relevant Data</u>

Stefan-Boltzmann constant, $\sigma = 5.67\times10^{-8}$ W m^{-2} K^{-4}

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Q4 Continued/…

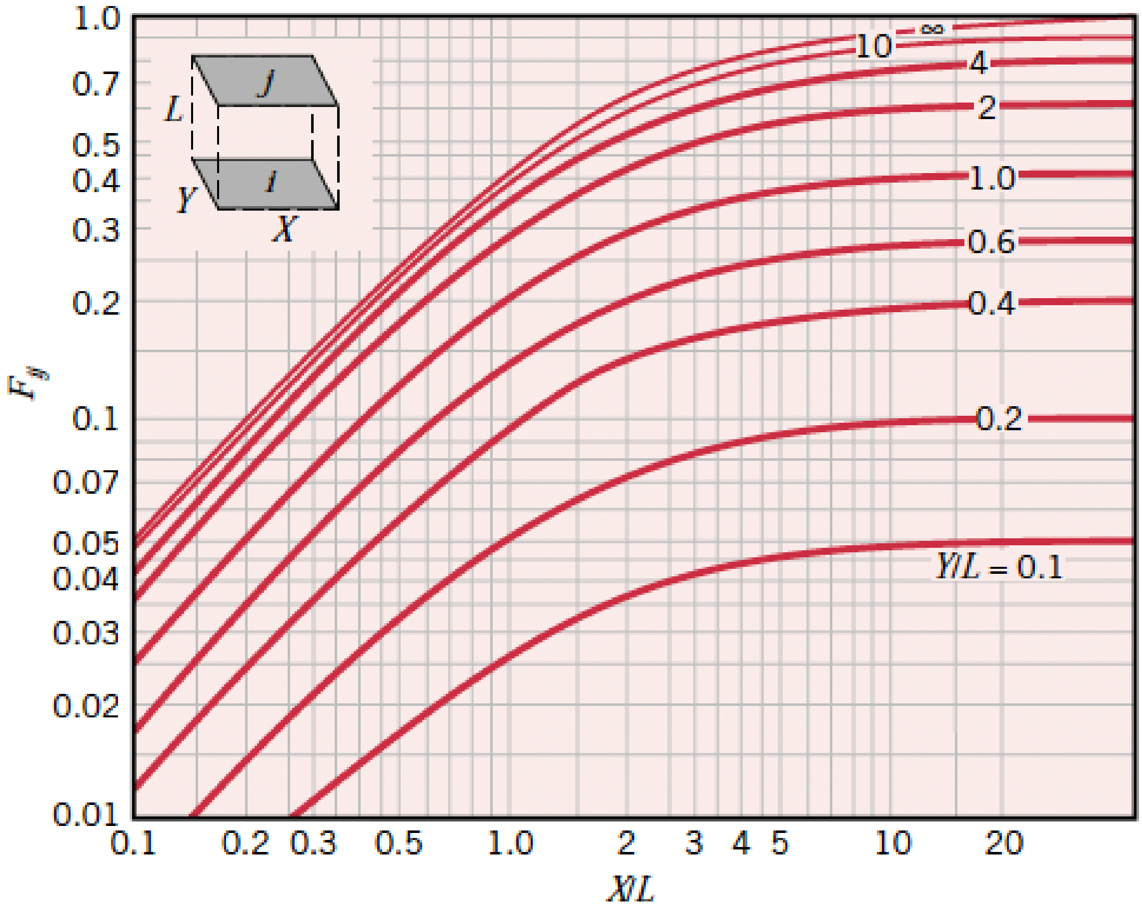

Figure Q4.2. View factor for aligned parallel rectangles.

END OF PAPER

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CGA006 - Heat Transfer
Formulae and Nomenclature

Equation	Definition
General	
$\dot{Q} = \dot{m}c_p\Delta T$	Temperature change of a stream where no phase change occurs
$PV = nRT$	Ideal gas law
$\dot{m} = \rho u_m A_c$	Mass flow rate of fluid in pipe
$\Delta U = mc_v\Delta T$	Change in internal energy
$\Delta H = mc_p\Delta T$	Change in enthalpy
Fluid mechanics relations	
$f = \frac{64}{Re}$	Friction factor (laminar flow)
$\Delta P = f\frac{L}{D}\frac{\rho u_m^2}{2}$	Pressure drop for internal forced flow
$\dot{W}_{pump} = \frac{\dot{m}\Delta P}{\rho}$	Pumping power for internal forced flow
Conduction (slab geometry: e.g., walls, thin pipes)	
$\dot{Q} = -kA\frac{dT}{dx} = kA\frac{T_1 - T_2}{L} = kA\frac{\Delta T}{L} = \frac{\Delta T}{R_{conduct}}$	Conduction heat transfer rate (slab)
$R_{conduct} = \frac{L}{kA}$	Conduction thermal resistance (slab)
$\frac{d^2T}{dx^2} + \frac{\dot{g}}{k} = 0$	Steady heat conduction in slab geometry (with heat generation)
Conduction (radial geometry: e.g., thick pipes, heated wires)	
$\frac{1}{r}\frac{d}{dr}\left(r\frac{dT}{dr}\right) + \frac{\dot{g}}{k} = 0$	Steady heat transfer in radial geometry with heat generation
$\frac{d}{dr}\left(r\frac{dT}{dr}\right) = 0$	Steady heat conduction in radial geometry (no heat generation)
$R_{conduct} = \frac{ln\left(\frac{r_2}{r_1}\right)}{2\pi kL}$	Conduction thermal resistance (cylinder layer)

Equation	Definition
Conduction between two surfaces	
$\dot{Q} = Sk(T_1 - T_2)$	Conduction heat transfer between two surfaces
$S = \dfrac{2\pi L}{\cosh^{-1}\left(\dfrac{4z^2 - D_1^2 - D_2^2}{2D_1 D_2}\right)}$	Conduction shape factor for two parallel pipes separated by distance *L*
Convection	
$\dot{Q} = hA_s(T_{fluid} - T_{wall})$	Convection heat transfer rate from fluid to wall
$\dot{Q} = UA_s(T_{fluid1} - T_{fluid2})$	Convection heat transfer rate between two fluids
$R_{convect} = \dfrac{1}{hA}$	Convection thermal resistance
$R_{overall} = \dfrac{1}{UA} = \sum R_{conduct} + \sum R_{convect}$	Overall thermal resistance
Dimensionless groups (replace L by D if sphere or cylinder)	
$\mathrm{Bi} = \dfrac{hL}{k_{solid}}$	Biot number
$\mathrm{Nu} = \dfrac{hL}{k}$	Nusselt number
$\mathrm{Pr} = \dfrac{\mu c_p}{k}$	Prandtl number
$\mathrm{Ra} = \dfrac{g\beta(T_s - T_\infty)L^3}{\nu^2}\mathrm{Pr}$	Rayleigh number
$\mathrm{Re} = \dfrac{u_m L}{\nu} = \dfrac{\rho u_m L}{\mu}$	Reynolds number
Heat exchangers (no phase change)	
$\dot{Q} = UAF\Delta T_{lm}$	Design equation (no phase change)
$\Delta T_{lm} = \dfrac{\Delta T_1 - \Delta T_2}{ln(\Delta T_1/\Delta T_2)}$	Log mean temperature difference
$\dfrac{1}{U} = \dfrac{1}{h_i} + \dfrac{1}{h_o}$	Overall heat transfer coefficient (neglecting pipe wall resistance)
$\dfrac{1}{U_{Dirty}} = \dfrac{1}{U_{Clean}} + \sum R_f$	Overall heat transfer coefficient with fouling resistances
Radiation	

Equation	Definition
$E_b = \sigma T^4$	Stefan-Boltzmann law
$E_{b\lambda} = \dfrac{C_1}{\lambda^5 \left[exp\left(\frac{C_2}{\lambda T}\right) - 1\right]}$	Planck's law
$\lambda_{max} T = 2898\ \mu\text{m.K}$	Wien's displacement law
$\varepsilon(T) = \alpha(T)$	Kirchhoff's law
$A_i F_{ij} = A_j F_{ji}$	View factor reciprocity rule
$\sum_{j=1}^{N} F_{ij} = 1$	View factor summation rule
$F_{i(j,k)} = F_{ij} + F_{ik}$	View factor superposition rule
$F_{ij} = F_{jk}$	View factor symmetry rule
$\dot{Q}_i = \sum_{j=1}^{N} \dot{Q}_{ij} = \sum_{j=1}^{N} A_i F_{ij} \sigma \left(T_i^4 - T_j^4\right)$	Radiation heat transfer rate between N black surfaces
$\dot{Q}_i = \dfrac{E_{bi} - J_i}{R_i}$	Radiation heat transfer rate from diffuse, grey, opaque surfaces
$R_i = \dfrac{1 - \varepsilon_i}{A_i \varepsilon_i}$	Surface resistance to radiation heat transfer
$\dot{Q}_i = \dfrac{J_i - J_j}{R_{ij}}$	Net radiation heat transfer rate between two diffuse, grey, opaque surfaces
$R_{ij} = \dfrac{1}{A_i F_{ij}}$	Space resistance to radiation heat transfer
$\dot{Q}_{12} = \dfrac{A\sigma(T_1^4 - T_2^4)}{\left(\frac{1}{\varepsilon_1} + \frac{1}{\varepsilon_2} - 1\right) + \left(\frac{1}{\varepsilon_{3,1}} + \frac{1}{\varepsilon_{3,2}} - 1\right)}$	Net radiation heat transfer rate between two large, infinite parallel plates with one radiation shield

Notation

A	Area (m^2)
A_c	Cross-sectional area of flow (m^2)
A_s	Area of surface (m^2)
C_1	First radiation constant = 3.742×10^8 W μm^4 m^{-2}
C_2	Second radiation constant = 1.439×10^4 μm K
c_0	Speed of light in vacuum = 2.998×10^8 m s^{-1}
c_p	Specific heat capacity of fluid at constant pressure (J kg^{-1} K^{-1})
c_v	Specific heat capacity of fluid at constant volume (J kg^{-1} K^{-1})
D	Diameter of pipe (m)

E_b	Total blackbody emissive power (W m^{-2})
$E_{b\lambda}$	Spectral blackbody emissive power (W m^{-2})
f	Darcy friction factor
F	LMTD correction factor
F_{ij}	Surface view factor
g	Gravitational acceleration (m s^{-2})
$\dot{g}$	Rate of internal heat generation per unit volume (W m^{-3})
h	Surface convection heat transfer coefficient (W m^{-2} K^{-1})
h_i	Inside surface convection heat transfer coefficient (W m^{-2} K^{-1})
h_o	Outside surface convection heat transfer coefficient (W m^{-2} K^{-1})
J_i, J_j	Surface radiosities (W m^{-2})
k	Thermal conductivity (W m^{-1} K^{-1})
L	Length or characteristic dimension (m)
m	Mass (kg)
$\dot{m}$	Mass flow rate (kg s^{-1})
n	Number of moles
Nu	Nusselt number
Pr	Prandtl number
Ra	Rayleigh number
Re	Reynolds number
P	Pressure (Pa or bar)
Q	Heat transfer (J)
$\dot{Q}$	Heat transfer rate (W)
R	Resistance to heat transfer (K W^{-1})
	or Ideal gas constant $= 8.314\ \mathrm{J\ mol^{-1}\ K^{-1}}$
R_f	Fouling factor (m^2 K W^{-1})
S	Conduction shape factor
T	Temperature (K or °C)
U	Overall heat transfer coefficient (W m^{-2} K^{-1})
u_m	Mean fluid velocity (m s^{-1})
V	Volume (m^3)
$\dot{W}_p$	Pumping power (W)

Greek symbols:

α	Surface absorptivity (radiation heat transfer)
β	Volumetric thermal expansivity (= 1/T for ideal gas) (K^{-1})
ε	Surface emissivity
ρ	Density of fluid (kg m^{-3})
	or Surface reflectivity (radiation heat transfer)
τ	Surface transmittivity (radiation heat transfer)
μ	Dynamic viscosity of fluid (Pa s)
σ	Stefan-Boltzmann's constant = 5.67 $\times 10^{-8}$ W m^{-2} K^{-4}
λ	Wavelength (m)
ν	Kinematic viscosity of fluid (m^2 s^{-1})