Indian Institute of Technology, Kharagpur
Mechanical Engineering Department

Heat Transfer (ME 30005) – Autumn Semester 2016
<u>Class Test – 2</u>

1. (a) Two large parallel planes with emissivity of 0.8 are at 1000 K and 400 K. A shield with one side treated and having an emissivity of 0.05 while the emissivity on the other untreated side was 0.6 was proposed to be used. The designer wanted the low emissivity side to face the hotter plane. During installation by mistake the side with higher emissivity was placed facing the hot side. Investigate the change in performance in terms of radiative heat transfer. What will the temperature of the radiation shield be in these two cases?

(b) Two long concentric cylinders are exchanging heat by radiation. To reduce the radiative heat transfer a radiation shield in the form of another concentric cylinder is placed between them. Will the radius of the intermediate cylinder affect the heat transfer between the inner and the outer cylinder? Clearly justify your answer.

5 +5 = 10

2. (a) Lubricating oil at a temperature of 60°C enters a 1 cm dia. tube with a velocity of 3 m/s. The tube surface is maintained at 40°C. Assuming that the oil has the following average properties, ρ = 865 kg/m^3; k = 0.140 W/m-K; c_p = 1.78 kJ/kg-K, calculate the tube length required to cool the oil to 45°C assuming fully developed flow. Sketch the variation of wall heat flux along the length of the tube. μ= 0.04 kg/ms

(b) Consider two hot isothermal vertical flat plates of height H. One of them is cooled by forced convection of air and the other by natural convection of air. On the same graph plot h/h_{avg} as a function of H for these two cases, where h = local heat transfer coefficient and h_{avg} = average heat transfer coefficient over the entire plate. H is restricted such that the boundary layer in both the cases is laminar. Your sketch should bring out the true nature of variation.

5+5 = 10

<u>orced Convection – Internal Flow Correlations</u>

Circular pipe – Laminar Flow

$\overline{Nu_D} = 3.66$ -> Isothermal wall

$\overline{Nu_D} = 4.36$ -> Isoflux wall

Circular pipe – Turbulent Flow

<u>Dittus-Boelter correlation</u> for smooth walls ($Re_D > 10{,}000$)

$\overline{Nu_D} = 0.023 Re_L{}^{4/5} Pr^n$ *[n=0.3 for heated wall; n=0.4 for cold wall]*

<u>Gnielinski correlation</u> for $Re_D > 3000$

$$Nu_D = \frac{(f/8)(Re_D - 1000)Pr}{1 + 12.7(f/8)^{1/2}(Pr^{2/3} - 1)}$$, where $f = (0.790 \ln Re_D - 1.64)^{-2}$ for smooth surface and needs to read from Moody chart for rough surface

Non-circular tubes

- Use hydraulic diameter
- Nu_D depends strongly on aspect ratio especially for laminar flow (*table shown in class*)
- For turbulent flow, Nu_D for circular pipe can be used with reasonable accuracy

Circular pipe – Entry length (laminar flow, uniform temp)

- Combined Entry length

$$\overline{Nu}_D = 1.86\left(\frac{Re_D Pr}{L/D}\right)^{1/3}\left(\frac{\mu}{\mu_s}\right)^{0.14} \quad \text{if} \quad \left[Re_D Pr/(L/D)\right]^{1/3}(\mu/\mu_s)^{0.14} > 2:$$

$$\overline{Nu}_D = 3.66 \quad \text{if} \quad \left[Re_D Pr/(L/D)\right]^{1/3}(\mu/\mu_s)^{0.14} < 2:$$

- Thermal Entry length

$$\overline{Nu}_D = 3.66 + \frac{0.0668(D/L)Re_D Pr}{1 + 0.04\left[(D/L)Re_D Pr\right]^{2/3}}$$

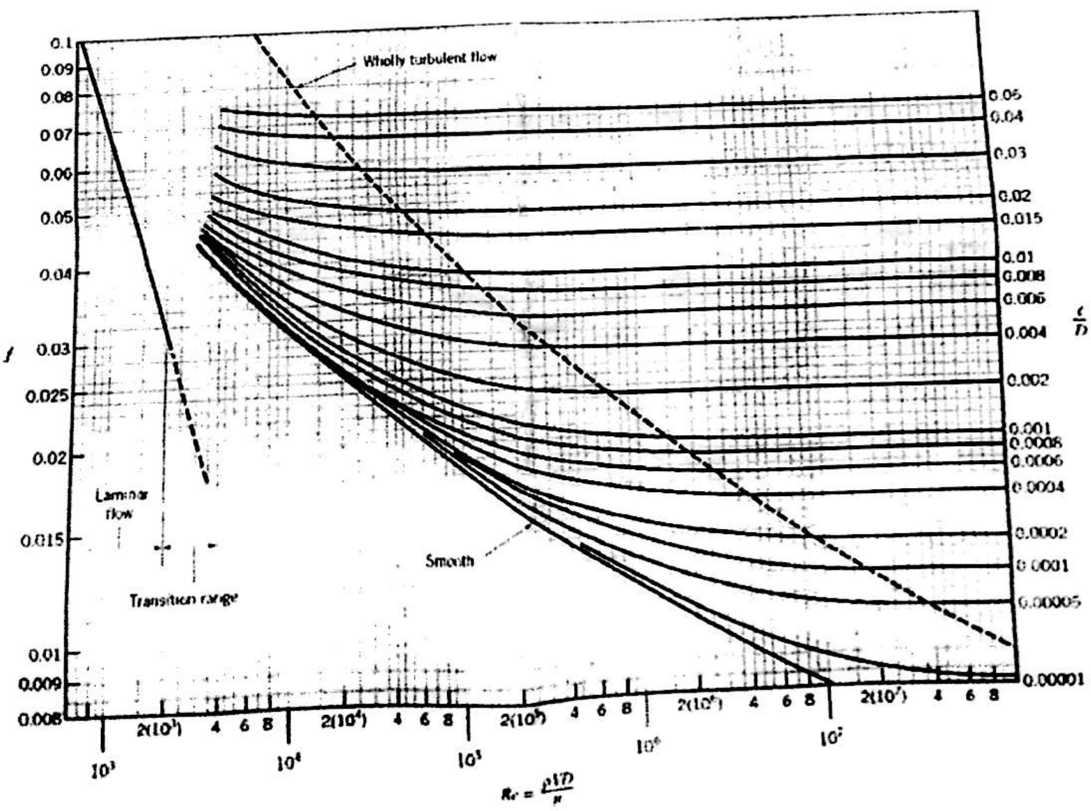
Circular pipe – Entry length (turbulent flow, uniform temp)

- For short tubes (L/D < 60),

$$\frac{\overline{Nu}_D}{Nu_{D,fd}} \approx 1 + \frac{C}{(L/D)^m}$$ where C =1, m ≐ 2/3

- For long tubes (L/D > 60),

$$\overline{Nu}_D \approx Nu_{D,fd}$$



Moody Chart