First Semester Examination
2019/2020 Academic Session

December 2019 / January 2020

EMH 441 – Heat Transfer
[Pemindahan Haba]

Duration : 3 hours
[Masa : 3 jam]

Please check that this paper contains **EIGHTEEN [18]** printed pages including appendix before you begin the examination.

*[Sila pastikan bahawa kertas soalan ini mengandungi **LAPAN BELAS [18]** mukasurat bercetak beserta lampiran sebelum anda memulakan peperiksaan.]*

INSTRUCTIONS : Answer **ALL FIVE [5]** questions.
*[**ARAHAN** : Jawab **SEMUA LIMA [5]** soalan.]*

Answer Questions In **English** OR **Bahasa Malaysia**.
*[Jawab soalan dalam **Bahasa Inggeris** ATAU **Bahasa Malaysia**.]*

Answer to each question must begin from a new page.
[Jawapan bagi setiap soalan mestilah dimulakan pada mukasurat yang baru.]

In the event of any discrepancies, the English version shall be used.
[Sekiranya terdapat sebarang percanggahan pada soalan peperiksaan, versi Bahasa Inggeris hendaklah diguna pakai.]

Note: Formulations and Tables are given in the Appendix.
Formula dan Jadual diberikan dalam lampiran.

1. [a] **The inner surface of a plane brick wall is at 45^{o}C and the outer surface is at 15^{o}C. The wall thickness is 250mm and the thermal conductivity of the brick is 0.52W/mk. Calculate the rate of heat transfer per unit area of wall surface.**

Permukaan dalaman bagi sebuah satah dinding bata adalah pada 45^{o}C dan permukaan luar adalah pada 15^{o}C. Ketebalan dinding adalah 250mm dan kekonduksian terma bagi bata adalah 0.52W/mK. Kirakan kadar pemindahan haba per unit luas bagi permukaan dinding.

(20 marks/*markah***)**

[b] **A wall has two layers of insulation on it. The layers are of equal thickness but the thermal conductivity of layer 1 is twice that of the layer 2. Calculate:**

Sebuah dinding mempunyai dua lapisan penebat. Lapisan mempunyai ketebalan yang sama tetapi kekonduksian terma bagi lapisan 1 adalah dua kali ganda daripada lapisan 2. Kirakan:

(i) **Thickness should layer 2 be if the heat flow is reduced by 20%, the total thickness and the driving force are remaining unchanged.**
Ketebalan yang sepatutnya bagi lapisan 2 jika aliran haba adalah dikurangkan sebanyak 20%, jumlah ketebalan dan daya panduan adalah tidak berubah.

(20 marks/*markah***)**

(ii) **Percentage reduction in heat flow if the material of layer 1 is replaced by the material of layer 2.**
Peratus pengurangan dalam aliran haba jika bahan lapisan 1 diganti dengan bahan lapisan 2.

(20 marks/*markah***)**

[c] **Aluminum fins of rectangular profile are attached on a plane wall. The fins have thickness t=1mm, length L= 10mm and thermal conductivity k= 200W/mK. The wall is maintained at a temperature T_o= 200^{o}C and the fins dissipate heat by convection into ambient air at $T_{ambient}$= 40^{o}C with a heat transfer coefficient $h_{ambient}$= 50W/m^{2}K. Assume negligible heat loss from fin tip, calculate the fin efficiency**

Sirip aluminum bersusuk segiempat tepat dilekatkan ke atas sebuah satah dinding. Sirip mempunyai ketebalan t=1mm, panjang L= 10mm dan kekonduksian terma k= 200W/mK. Dinding mempunyai suhu tetap T_o= 200^{o}C dan sirip membebaskan haba dengan perolakan ke udara dengan suhu bilik pada $T_{persekitaran}$= 40^{o}C dan pekali pemindahan haba $h_{persekitaran}$= 50W/m^{2}K. Andaikan haba terbebas daripada hujung sirip adalah kecil, kirakan kecekapan sirip.

(40 marks/*markah***)**

2. **Air is forced at $\mathbf{V = 15 m/s}$ and $\mathbf{T_{ambient} = 20°C}$ to cool electronic chips on a circuit board, as depicted in Figure 2. The center point of Chip A and Chip B, are located 6cm and 44cm from the leading edge, respectively. The total length of the board is 50cm. The thermal conductivity of air is assumed to be $\mathbf{0.026\,W/m.K}$. Assume that the kinematic viscosity and the thermal diffusivity of air are $\mathbf{1.2 \times 10^{-5}\ m^2/s}$ and $\mathbf{1.9 \times 10^{-5} m^2/s}$, respectively. The critical Reynolds number is $\mathbf{5 \times 10^5}$. The height of the chip is negligible.**

Udara dialirkan dengan kelajuan $V = 15m/s$ pada $T_{persekitaran} = 20°C$ untuk menyejukkan cip elektronik pada papan litar, seperti yang digambarkan dalam Rajah 2. Titik pusat Cip A dan Cip B, masing-masing terletak pada 6cm dan 44cm dari pinggir depan. Panjang keseluruhan papan litar adalah 50cm. Kekonduksian haba udara diandaikan sebagai $0.026\,W/m.K$. Anggapkan bahawa kelikatan kinematik dan kemeresapan haba udara adalah $1.2 \times 10^{-5}\ m^2/s$ dan $1.9 \times 10^{-5} m^2/s$. Nombor Reynolds kritikal adalah 5×10^5. Ketinggian cip boleh diabaikan.

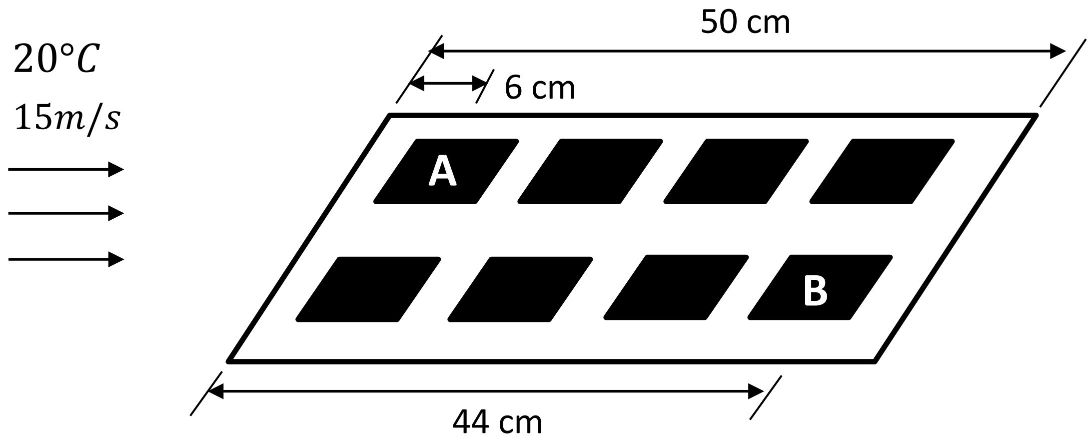

Figure 2
Rajah 2

(iii) **Calculate the velocity boundary layer thickness $(\boldsymbol{\delta})$ at the center of Chip A, $\mathbf{6}\boldsymbol{cm}$ from the leading edge.**
Kirakan ketebalan lapisan sempadan halaju (δ) di pusat Cip A, $6cm$ dari pinggir depan.

(20 marks/*markah***)**

(iv) **Calculate the thermal boundary layer thickness $(\boldsymbol{\delta_t})$ at the center of Chip A, $\mathbf{6}\boldsymbol{cm}$ from the leading edge.**
Kirakan ketebalan lapisan sempadan terma (δ_t) di pusat Cip A, $6cm$ dari pinggir depan.

(20 marks/*markah***)**

(v) **Calculate the surface temperature of the Chip B if it is dissipating heat at 300W/m^2.**
Kirakan suhu permukaan Cip B jika ia memindah haba pada 300W/m^2.

(50 marks/*markah*)

(vi) **If Chip A is also dissipating heat at 300W/m^2, will the surface temperature of Chip A be lower than that of Chip B? Justify your answer? Assume the change in the ambient air over the chip is negligible.**
Jika Cip A juga memindah haba pada 300W/m^2, adakah suhu permukaan Cip A lebih rendah daripada Cip B? Wajarkan jawapan anda? Andaikan perubahan dalam udara sekitar di atas cip boleh diabaikan.

(10 marks/*markah*)

3. **Hot air enters at $20^0\mathrm{C}$ in $10\mathrm{m}$ long $20\mathrm{cm} \times 10\mathrm{cm}$ rectangular shape channel at a mean velocity of $0.1\mathrm{m}/s$. Constant temperature at $200^0\mathrm{C}$ is assumed along the surface of the channel. Assume the flow in the channel to be laminar flow. The properties of air are given as follows:**

Udara panas memasuki saluran sepanjang $10m$ berbentuk segi empat tepat $20cm \times 10cm$ dengan kelajuan $0.1m/s$. Suhu tetap di 200^0C dianggapkan pada permukaan saluran. Andaikan aliran dalam saluran adalah aliran lamina. Ciri-ciri udara diberikan seperti berikut:

$$\rho = 0.9458\, kg/m^3$$

$$k = 0.03095\, W/m.K$$

$$\mu = 2.181 \times 10^{-5}\, kg/m.s$$

$$c_p = 1.009\, kJ/kg.K$$

$$Pr = 0.7111$$

(i) **Neglecting the entrance effect, assume the air flow through the channel is fully-developed. Calculate the air exit temperature.**
Abaikan kesan aliran masuk, andaikan aliran melalui saluran berada dalam terbentuk penuh. Kirakan suhu keluar udara.

(40 marks/*markah*)

(ii) **Assume that the average heat transfer coefficient for flow in entrance region to be twice of the fully developed value. Calculate the air exit temperature.**
Andaikan bahawa purata pekali pemindahan haba untuk aliran di kawasan masuk adalah dua kali ganda dari nilai pada aliran terbentuk penuh. Kirakan suhu keluar udara.

(60 marks/*markah*)

4. **[a]** **A $1.5\mathrm{m} \times 1.5\mathrm{m}$ square plate is exposed to ambient condition, as depicted in Figure 4[a]. The temperature of the hot plate is maintained at $120^0\mathrm{C}$ and the lower surface of the plate is insulated. Ambient air is at $20^0\mathrm{C}$. Assume that the thermal expansion coefficient is 0.003. Calculate the average heat transfer coefficient for natural convection that occurred above the hot plate.**

Plat persegi $1.5m \times 1.5m$ terdedah kepada keadaan sekitar, seperti yang ditunjukkan pada Rajah 4[a]. Suhu plat panas dikekalkan pada 120^0C dan permukaan bawah adalah ditebat. Udara sekitar adalah pada suhu 20^0C. Andaikan bahawa pekali pengembangan haba ialah 0.003. Kirakan pekali pemindahan haba purata untuk perolakan semula jadi yang berlaku di atas plat panas.

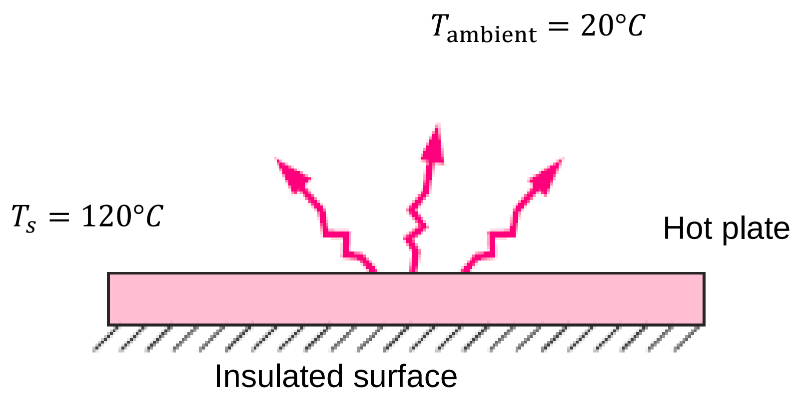

Figure 4 [a]
Rajah 4 [a]

(50 marks/*markah*)

[b] **A thin aluminum plate with an emissivity ε_3 =0.25 on both sides between two very large plate, as shown in Figure 4[b]. Calculate net radiation heat transfer q_{1-2} between two plates per unit surface area. Given Stefan–Boltzmann constant σ=5.67x10^{-8} W/m^{2}K^4.**

Sekeping plat aluminium nipis dengan pekali kepancaran ε_3 =0.25 pada kedua-dua belah permukaan di antara dua plat yang sangat besar, seperti yang ditunjukkan dala Rajah 4b. Kirakan radiasi pemindahan haba bersih q_{1-2} diantara dua plat perunit luas permukaan. Diberi pekali Stefan–Boltzmann σ=5.67x10^{-8} W/m^2.K^4.

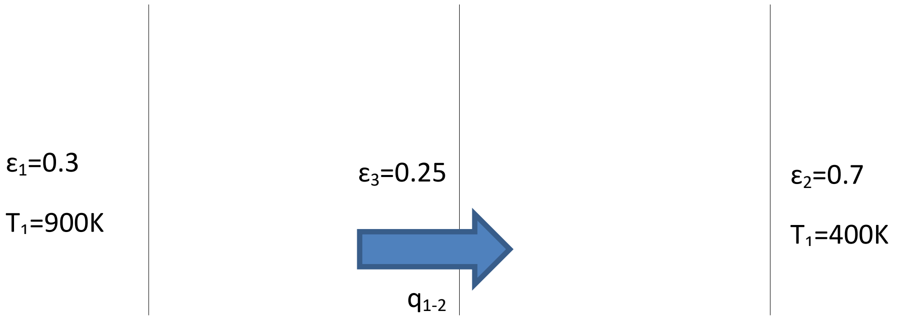

Figure 4 [b]
Rajah 4 [b]

(15 marks/*markah*)

[c] **Five infinitely long surface of width L1, and L2=1.5m, and L3, L4, and L5=2m, as shown in Figure 4 [c]. The temperature of surfaces L1 and L2 are 1000K; and surfaces L3, L4 and L5 are 700K. Given emissivity coefficient ε=1 for all surfaces and Stefan–Boltzmann constant σ=5.67x10^{-8} W/m^{2}K^4. Calculate:**

Lima permukaan panjang tak terhingga berketebalan L1 dan L2 = 1.5m, dan L3, L4 dan L5 = 2m, seperti yang ditunjukkan dala Rajah 4[c]. Suhu permukaan L1 dan L2 adalah 1000K dan permukaan L3, L4 dan L5 adalah 700K. Diberi pekali kepancaran ε=1, untuk semua permukaan dan pekali Stefan–Boltzmann σ=5.67x10^{-8} W/m^2.K^4. Kirakan:

(i) **View factor from Surface L1 to Surface L4, (F_{L1-L4}), as shown in Figure 4 [c].**
Faktor penglihatan dari Permukaan L1 ke Permukaan L4 (F_{L1-L4}) seperti yang ditunjukkan pada Rajah 4 [c].

(25 marks/*markah*)

(ii) **Radiation heat transfer from Surface L2 to Surface L4.**
Radiasi pemindahan haba dari Permukaan L2 ke Permukaan L4.

(10 marks/*markah*)

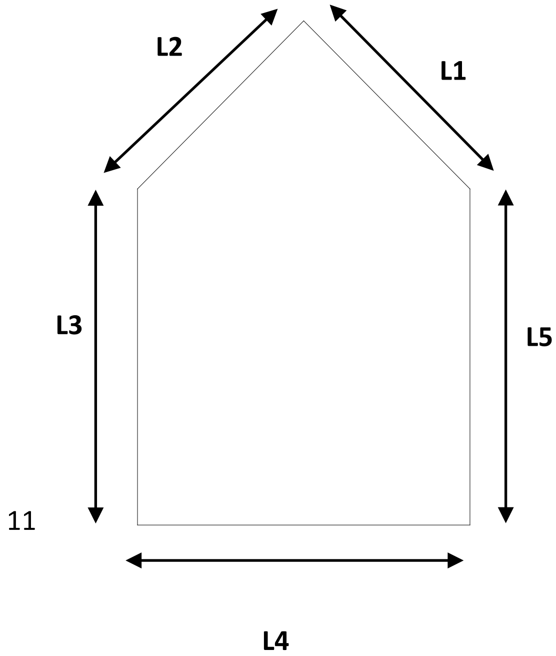

Figure 4 [c]
Rajah 4 [c]

5. [a] Water at rate of 10kg/s is heated from 35˚C to 60˚C by an oil having a specific heat capacity, c_P oil= 2000J/kg.K. The fluids flow in counterflow double pipe heat exchanger. The oil enters 110˚C and leaves at 65˚C. The total surface area is 110m^2. Calculate the overall heat transfer coefficient, U using LMTD method. Given specific heat capacity of water, c_P water= 4200J/kg.K

Air pada kadar 10kg/s dipanaskan dari 35˚C kepada 60˚C mengunakan sejenis minyak yang mempunyai muatan haba tentu c_P minyak=2000J/kg.K. Bendalir-bendalir ini mengalir dalam penukar haba dwipaip lawan arus. Minyak masuk 110˚C dan keluar 65˚C. Kirakan pekali pemindahan haba keseluruhan, U, mengunakan kaedah LMTD. Diberi muatan haba tentu air c_P air= 4200J/kg.K

(35 marks/*markah***)**

[b] Two counterflow double pipe heat exchangers; HE1 and HE2, connected in series on the water side and in parallel on oil side, as shown in Figure 5[b]. HE1 and HE2 are used to heat 0.5kg/s of water from water inlet temperature TWi-1=45˚C to water outlet temperature TWo-2=90˚C by cooling an oil from oil inlet

temperature TOi-1=160˚C to overall oil outlet temperature TOo-3=95˚C. Oil is split by 1.64kg/s for HE1 and 2.52kg/s for HE2. Each heat exchanger have surface area of 2.5m^2, and same heat exchanger effectiveness, ε. Assume water outlet temperature of HE1 = water inlet temperature of HE2, TWo,1=70˚C. Given specific heat capacity of water, c_P= 4200J/kg.K and specific heat capacity of oil, c_P= 2000J/kg.K. Calculate:

Dua penukar haba dwipaip lawan arus; HE1 dan HE2, disambung dalam keadaan siri pada air dam dalam keadaan selari pada minyak, seperti yang ditunjukkan dalam Rajah 5[b]. HE1 amd HE2 digunakan untuk memanaskan 0.5kg/s air dari suhu masukan air TWi-1 45˚C kepada suhu keluaran air TWo-2=90˚C dengan menyejukkan minyak dari suhu masukan minyak TOi-1=160˚C kepada suhu keluaran minyak TOo-3=95˚C. Minyak dibahagikan kepada 1.64kg/s untuk HE1 dan 2.52kg/s untuk HE2, dan mempunyai keberkesanan penukar haba ε yang sama. Andaikan suhu keluaran air HE1 = suhu masukan air HE2 TWo-1= 70˚C. Diberi muatan haba tentu air, c_P air =4200J/kg.K dan muatan haba tentu minyak, c_P oil = 2000J/kg.K. Kirakan:

(i) **HE1 outlet temperature, TOo-1.**
Suhu keluaran HE1 TOo-1. **(30 marks/*markah*)**

(ii) **HE2 outlet temperature, TOo-2.**
Suhu keluaran HE2 TOo-2. **(15 marks/*markah*)**

(iii) **Overall heat transfer coefficient, U.**
Pekali pemindahan haba keseluruhan. **(20 marks/*markah*)**

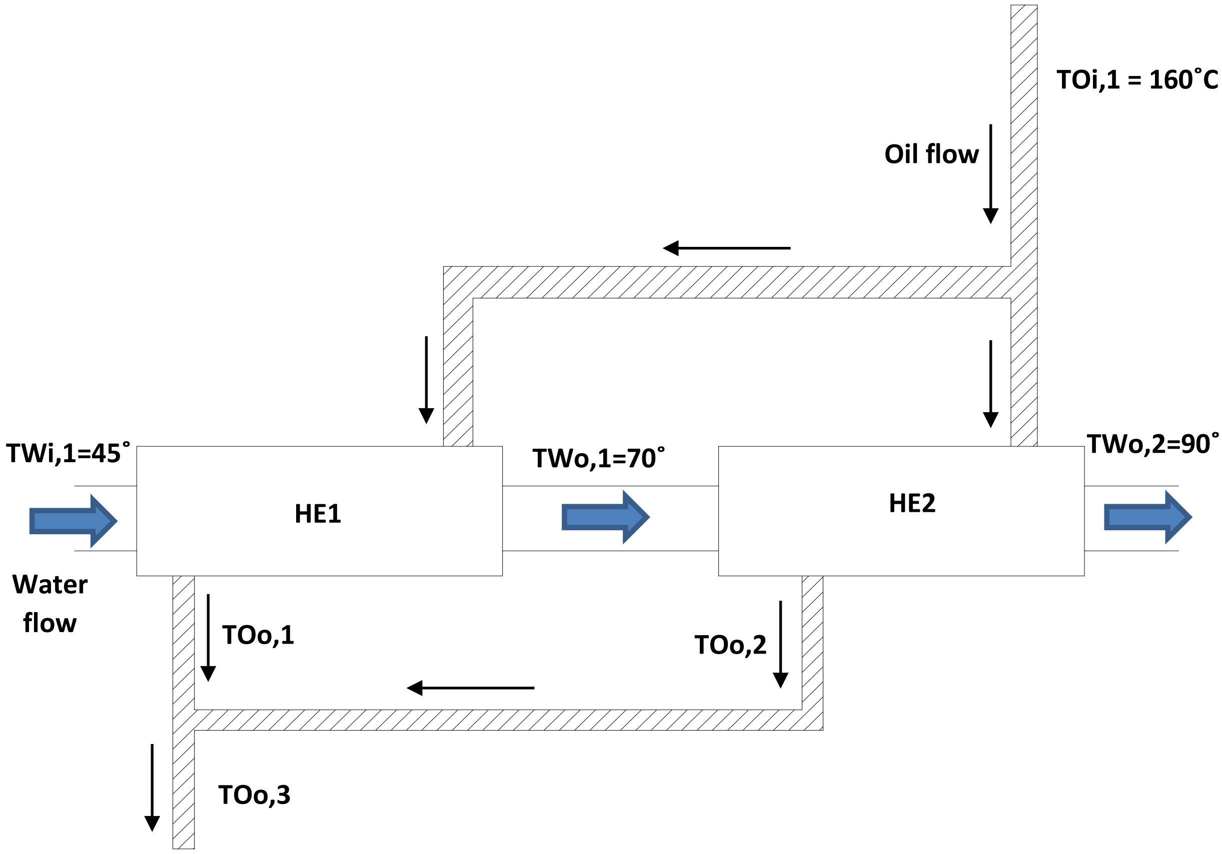

Figure 5 [b]
Rajah 5 [b]

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Apendix A
Lampiran A

FORMULA FOR HEAT CONDUCTION

Cartesian Coordinates:

$$\frac{\partial}{\partial x}\left(k\frac{\partial T}{\partial x}\right)+\frac{\partial}{\partial y}\left(k\frac{\partial T}{\partial y}\right)+\frac{\partial}{\partial z}\left(k\frac{\partial T}{\partial z}\right)+\dot{e}\ \text{gen}=\rho c\frac{\partial T}{\partial t}$$

Cylindrical Coordinates:

$$\frac{1}{r}\frac{\partial}{\partial r}\left(kr\frac{\partial T}{\partial r}\right)+\frac{1}{r^2}\frac{\partial}{\partial \phi}\left(k\frac{\partial T}{\partial \phi}\right)+\frac{\partial}{\partial z}\left(k\frac{\partial T}{\partial z}\right)+\dot{e}\ \text{gen}=\rho c\frac{\partial T}{\partial t}$$

Spherical Coordinates:

$$\frac{1}{r^2}\frac{\partial}{\partial r}\left(kr^2\frac{\partial T}{\partial r}\right)+\frac{1}{r^2\sin^2\theta}\frac{\partial}{\partial \phi}\left(k\frac{\partial T}{\partial \phi}\right)+\frac{1}{r^2\sin\theta}\frac{\partial}{\partial \theta}\left(k\sin\theta\ \frac{\partial T}{\partial \theta}\right)+\dot{e}\ \text{gen}=\rho c\frac{\partial T}{\partial t}$$

Thermal Resistance:

Plane

$$R_{interface}=\frac{1}{h_c A}=\frac{\Delta T_{\text{interface}}}{\dot{Q}}$$

Cylindrical

$$R_{cyl}=\frac{\ln(r_2/r_1)}{2\pi Lk}$$

Fin

$$\dot{Q}_{\text{long fin}}=\sqrt{hpkA_c}(T_b-T_\infty)$$

$$\dot{Q}_{\text{insulated tip}}=\sqrt{hpkA_c}(T_b-T_\infty)\tanh mL$$

Lump System Analysis

Biot Number:

$$Bi=\frac{R_{cond}}{R_{conv}}=\frac{L/kA}{1/hA}=\frac{hL}{k}$$

$$\frac{T(t)-T_\infty}{T_i-T_\infty}=e^{-bt}$$ Where $$b=\frac{hA_s}{\rho c_p \mathcal{V}}$$

Formulation

1. General properties

Table: Properties of air at 1atm pressure

Properties of air at 1 atm pressure

Temp. T, °C	Density ρ, kg/m³	Specific Heat c_p, J/kg·K	Thermal Conductivity k, W/m·K	Thermal Diffusivity α, m²/s	Dynamic Viscosity μ, kg/m·s	Kinematic Viscosity ν, m²/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
90	0.9718	1008	0.03024	3.086×10^{-5}	2.139×10^{-5}	2.201×10^{-5}	0.7132
100	0.9458	1009	0.03095	3.243×10^{-5}	2.181×10^{-5}	2.306×10^{-5}	0.7111

2. External forced convection heat transfer formulations:

Flow over flat plate

Velocity boundary layer thickness, $\delta(x) = \dfrac{5.0x}{Re_x^{1/2}}$

Thermal boundary layer thickness, $\delta_t(x) = \dfrac{\delta}{Pr^{1/3}} = \dfrac{5.0x}{Pr^{1/3}Re_x^{1/2}}$

Forced convection heat transfer over flat plate

i) Isothermal along the plate

a) Local Nusselt number

Laminar:	$Nu_x = 0.332Re_x^{0.5}Pr^{1/3}$	For $Re_x < 5 \times 10^5$, $Pr > 0.6$
Turbulent:	$Nu_x = 0.0296Re_x^{0.8}Pr^{1/3}$	For $5 \times 10^5 \leq Re_x \leq 10^7$, $0.6 \leq Pr \leq 60$

b) Average Nusselt number

Laminar:	$Nu = 0.664Re_L^{0.5}Pr^{1/3}$	for $Re_L < 5 \times 10^5$, $Pr > 0.6$

Turbulent:	$Nu = 0.037Re_L^{0.8}Pr^{1/3}$	for $5 \times 10^5 \leq Re_L \leq 10^7$, $0.6 \leq Pr \leq 60$

ii) Uniform heat flux along the plate

a) Local Nusselt number

Laminar:	$Nu_x = 0.453Re_x^{0.5}Pr^{1/3}$	for $Re_x < 5 \times 10^5$, $Pr > 0.6$
Turbulent:	$Nu_x = 0.0308Re_x^{0.8}Pr^{1/3}$	for $5 \times 10^5 \leq Re_x \leq 10^7$, $0.6 \leq Pr \leq 60$

3. Internal forced convection heat transfer formulations:

A. <u>Entrance region</u>

Circular tube with constant surface temperature,

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

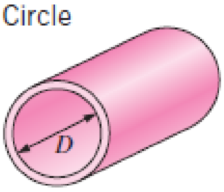
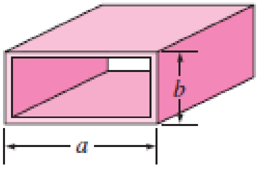
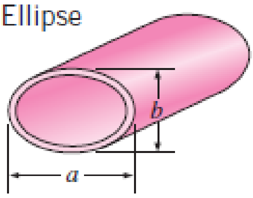
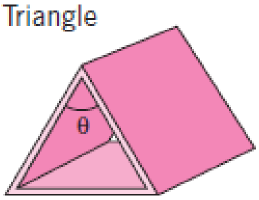
Parallel plate with constant surface temperature,

$$Nu = 7.54 + \frac{0.03(D_h/L)RePr}{1 + 0.016[(D_h/L)RePr]^{2/3}}$$

for $Re \leq 2800$

B. Fully developed region

Nusselt number and friction factor for fully developed laminar flow in tubes and various cross sections.

Tube Geometry	a/b or $\theta°$	Nusselt Number T_s = Const.	Nusselt Number $\dot{q}_s$ = Const.	Friction Factor f
Circle 	—	3.66	4.36	64.00/Re
Rectangle 	a/b			
	1	2.98	3.61	56.92/Re
	2	3.39	4.12	62.20/Re
	3	3.96	4.79	68.36/Re
	4	4.44	5.33	72.92/Re
	6	5.14	6.05	78.80/Re
	8	5.60	6.49	82.32/Re
	∞	7.54	8.24	96.00/Re
Ellipse 	a/b			
	1	3.66	4.36	64.00/Re
	2	3.74	4.56	67.28/Re
	4	3.79	4.88	72.96/Re
	8	3.72	5.09	76.60/Re
	16	3.65	5.18	78.16/Re
Triangle 	θ			
	10°	1.61	2.45	50.80/Re
	30°	2.26	2.91	52.28/Re
	60°	2.47	3.11	53.32/Re
	90°	2.34	2.98	52.60/Re
	120°	2.00	2.68	50.96/Re

C. Thermal condition

i) Constant surface heat flux along circular tube:

$$\frac{\partial T_m}{\partial x} = \frac{\partial T_s}{\partial x} = \frac{\partial T}{\partial x} = \frac{\dot{q}_s p}{\dot{m} c_p}$$

ii) Constant surface temperature along circular tube:

Temperature at exit of tube, $T_e = T_s - (T_s - T_i)\exp(-hA_s/\dot{m}c_p)$

Heat transfer rate, $\dot{Q} = hA_s \Delta T_{ln}$

Logarithmic mean temperature difference, $\Delta T_{ln} = \dfrac{T_i - T_e}{\ln\frac{T_s - T_e}{T_s - T_i}}$

**Formulation can be applied to non-circular tube.*

4. Natural convection heat transfer formulations:

Rayleigh number:

$$Ra_L = \frac{g\beta(T_s - T_\infty)L_c^3}{\nu^2} Pr$$

Empirical correlations for natural convection over surfaces

i) Vertical plates with characteristic length L,

$Nu = 0.59Ra_L^{1/4}$, for $10^4 < Ra < 10^9$

$Nu = 0.1Ra_L^{1/3}$, for $10^{10} < Ra < 10^{13}$

ii) Horizontal plates with characteristic length, $L = A_s/p$

(air above hot plate)

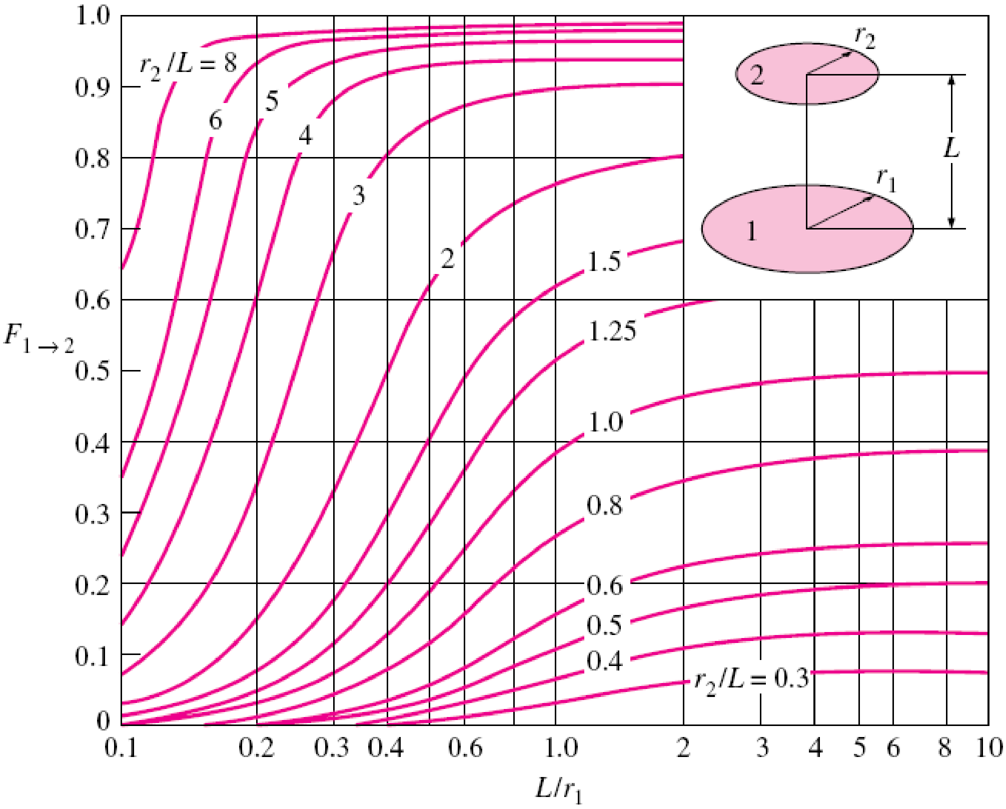
$Nu = 0.54Ra_L^{1/4}$, for $10^4 < Ra < 10^7$

$Nu = 0.15Ra_L^{1/3}$, for $10^7 < Ra < 10^{11}$

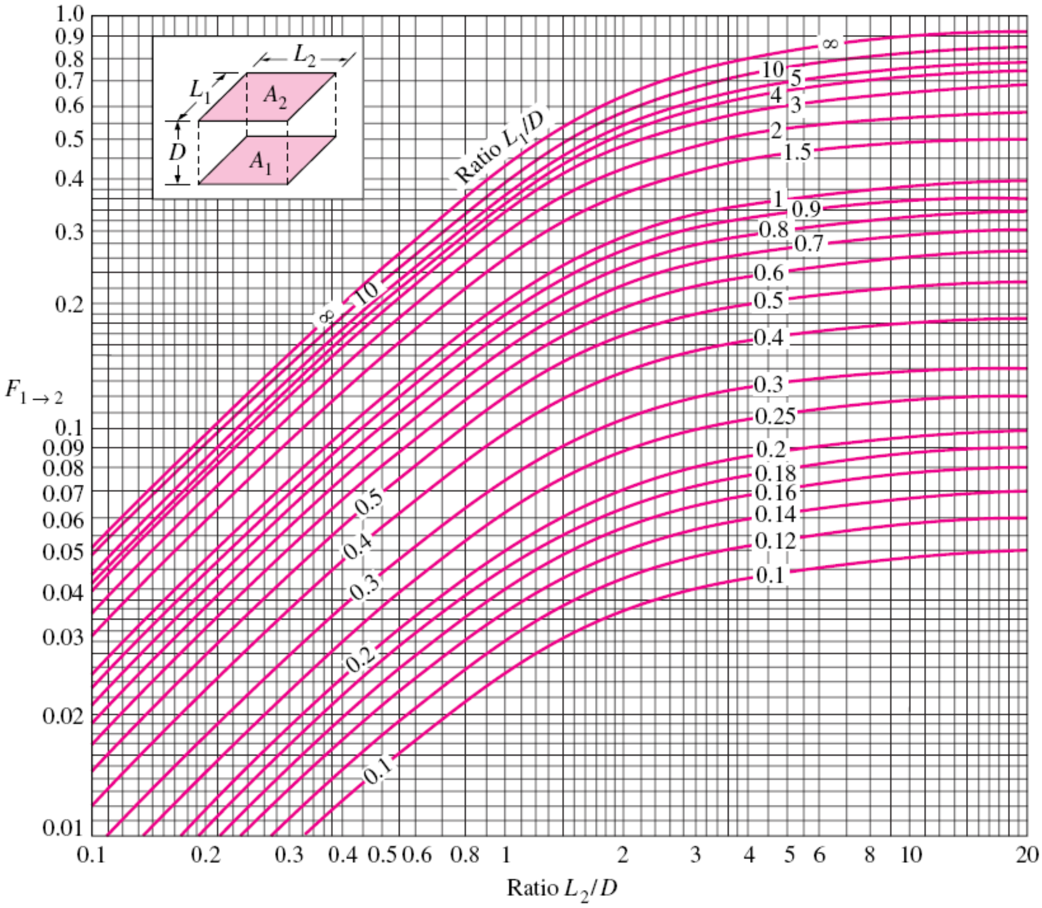
iii) Horizontal plates with characteristic length, $L = A_s/p$

(air below hot plate)

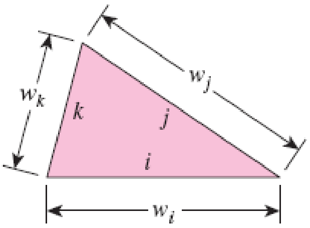
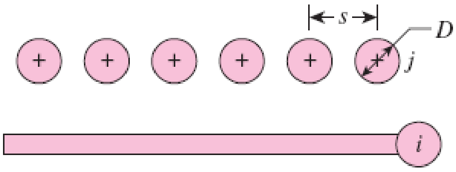
$Nu = 0.27Ra_L^{1/4}$ for $10^5 < Ra < 10^{11}$

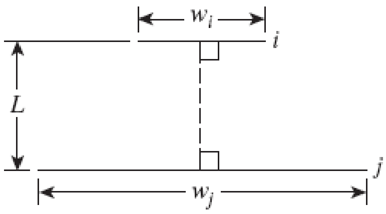
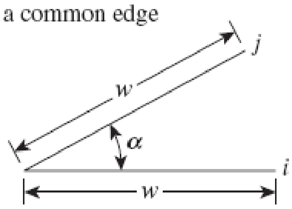
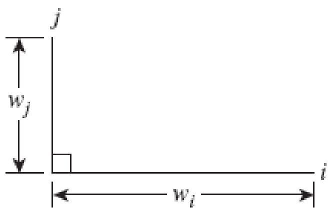


View factor between two coaxial parallel disks

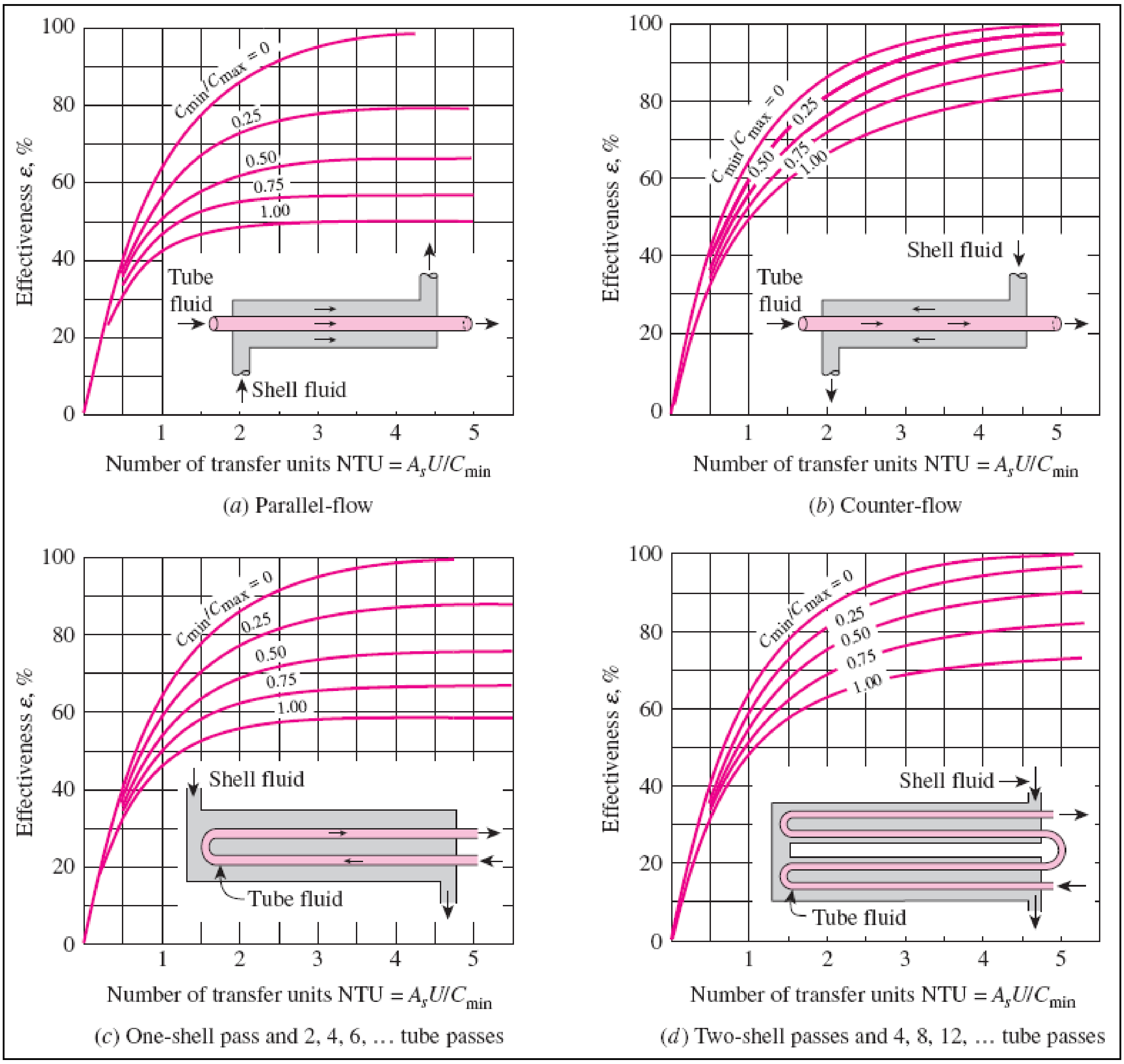


View factor between two aligned parallel rectangles of equal size

Three-sided enclosure 	$F_{i \to j} = \dfrac{w_i + w_j - w_k}{2w_i}$
Infinite plane and row of cylinders 	$F_{i \to j} = 1 - \left[1 - \left(\dfrac{D}{s}\right)^2\right]^{1/2} + \dfrac{D}{s}\tan^{-1}\left(\dfrac{s^2 - D^2}{D^2}\right)^{1/2}$

Geometry	**Relation**
Parallel plates with midlines connected by perpendicular line 	$W_i = w_i/L$ and $W_j = w_j/L$ $F_{i \to j} = \dfrac{[(W_i + W_j)^2 + 4]^{1/2} - (W_j - W_i)^2 + 4]^{1/2}}{2W_i}$
Inclined plates of equal width and with a common edge 	$F_{i \to j} = 1 - \sin\frac{1}{2}\alpha$
Perpendicular plates with a common edge 	$F_{i \to j} = \dfrac{1}{2}\left\{1 + \dfrac{w_j}{w_i} - \left[1 + \left(\dfrac{w_j}{w_i}\right)^2\right]^{1/2}\right\}$

View factor between two coaxial parallel disks



Effectiveness for heat exchanger, NTU method