University of New Mexico
Mechanical Engineering
2024 PhD qualifying examination

Heat Transfer

Closed book. Formula sheet and calculator are allowed, but not cell phones, computers or any other wireless device.
Time allowed: 150 minutes.

Part 1: General knowledge questions – 24 points total.
Select (circle) and briefly explain the correct answer.

1. Nusselt number. . .
(a) is a measure of heat transfer enhancement via convection (as compared with conduction).
(b) is the ratio of thermal to mechanical dissipation.
(c) equals the ratio of convective heat transfer coefficient h to conductivity k.

2. Objects that radiate relatively well
(a) always absorb radiation relatively well.
(b) reflect radiation relatively well.
(c) have a high emissivity ε.

3. Why do ceramic tiles in a kitchen or bathroom feel cooler than a floor mat?
(a) ceramic has a lower thermal conductivity value.
(b) ceramic has a higher contact area with your feet.
(c) ceramic has a higher thermal conductivity value.

4. The cooking instructions for turkeys always always tell us to measure temperature deep inside the turkey. From this, we can infer that, for the typical baking conditions:
(a) the Biot number is low.
(b) the Biot number is high.
(c) the thermometer will fall off unless we stick it in deep enough.

5. A composite wall consists of two layers with thermal conductivities k_1 and k_2 and thicknesses L_1 and L_2, respectively. If T_1 and T_2 are the temperatures at the surfaces, which expression gives the heat flux through the composite wall in steady state?

(a) $q=\dfrac{T_1-T_2}{\dfrac{L_1}{k_1}+\dfrac{L_2}{k_2}}$

(b) $q=\dfrac{T_1-T_2}{L_1+L_2}$

(c) $q=\dfrac{T_1-T_2}{k_1+k_2}$

6. The emissivity ϵ of a surface determines its effectiveness in emitting thermal radiation. Which of the following statements is true for a surface with emissivity ϵ?
(a) A surface with $\epsilon = 1$ is a perfect reflector.
(b) A surface with $\epsilon = 0$ emits no radiation.
(c) A surface with $\epsilon = 0.5$ emits half the radiation of a blackbody at the same temperature.

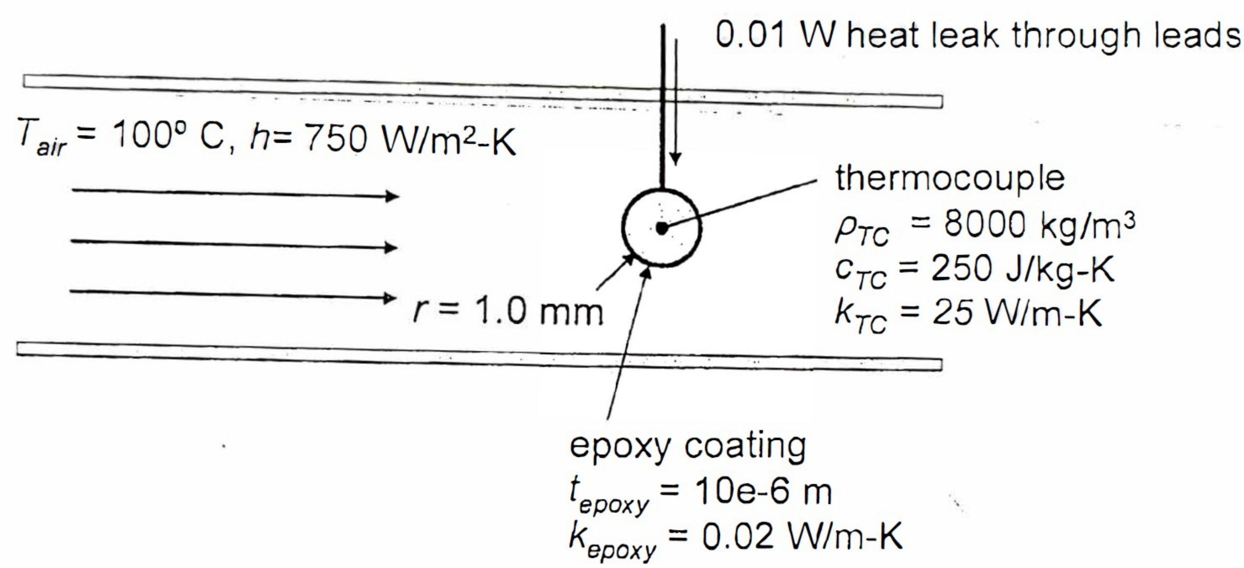
7. In a counterflow heat exchanger, the effectiveness ϵ is defined as the ratio of the actual heat transfer to the maximum possible heat transfer. The heat exchanger has a heat capacity ratio $C_r = C_{max}/C_{min}$. The number of transfer units (NTU) is defined as NTU = UA/C_{min}, where A is the total heat transfer area and U is the overall heat transfer coefficient. Which of the following statements is true for large NTU and a heat capacity ratio C_r close to 1?
(a) The effectiveness ϵ approaches $(1+C_r)^{-1}$
(b) The effectiveness ϵ approaches $1/C_r$
(c) The effectiveness ϵ approaches 1

8. There exists an optimum fin spacing for arrays of naturally convecting fins. This is because:
(a) placing fins closer together enhances natural convection
(b) placing fins further apart enhances natural convection
(c) placing fins closer together degrades natural convection, but increases the available area for heat transfer

Part 2: Problems.
Problem 1 (26 points)
Given: A mild carbon steel plate (AISI 1042) of thickness 1 cm at a temperature 600°C is rapidly immersed in an oil bath at 30°C. The average convective heat transfer coefficient between the plate and the oil is known to be about 400 W/m^2K.
Find: The time it will take for the plate to cool to 90°C.
Note: For steel, assume $\rho = 7800$ kg/m^3, $c = 450$ J/(kg K). Also use the attached tables if necessary. Justify your assumptions regarding the method of solution.

Problem 2 (25 points)

Given: A spherical thermocouple with radius 1.0 mm is suspended in a stream of air. The air temperature is 100°C and the heat transfer coefficient between the air and the thermocouple is 750 W/m^{2}K. The thermocouple is made of a material with very high conductivity, density 8000 kg/m^3, and specific heat capacity 250 J/(kg K). There is a 10.0 mW heat leak into the thermocouple due to conduction through the leads. The thermocouple is coated with a thin, 10 μm, layer of non-conductive epoxy (0.02 W/(m K)).



Find:

a) Under steady state conditions (*i.e.*, the temperature of the flow and the thermocouple are not changing) and neglecting thermal radiation, estimate the error associated with using this thermocouple.

b) If the thermocouple is used under transient conditions, estimate the response time of the thermocouple, *i.e.*, its time constant.

Problem 3 (25 points)

Given: Two very large parallel plates as sketched, with emissivities ε_1 and ε_2.
Find: Heat transfer per unit area between the plates (in W/m^2).
Note: State your assumptions.

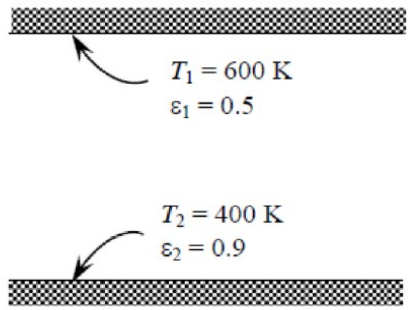


Table A.1*b* Solid metals: Temperature dependence of thermal conductivity k [W/m K] (see Table A.1*a* for metal compositions)

Metal	Temperature, K								
	200	300	400	500	600	800	1000	1200	1500
Aluminum									
Pure	237	237	240	236	231	218			
Duralumin	138	174	187	188					
Alloy 195, cast		168	174	180	185				
Copper									
Pure	413	401	393	386	379	366	352	339	
Commercial bronze	42	52	52	55					
Brass	74	111	134	143	146	150			
German silver		116	135	145	147				
Gold	323	317	311	304	298	284	270	255	
Iron									
Armco	81	73	66	59	53	42	32	29	31
Cast		51	44	39	36	27	23		
Carbon steels									
AISI 1010		64	59	54	49	39	31		
AISI 1042		52	50	48	45	37	29	26	30
AISI 4130		43	42	41	40	37	31	27	31
Stainless steels									
AISI 302		15	17	19	20	23	25		
AISI 304	13	15	17	18	20	23	25		
AISI 316		13	15	17	18	21	24		
AISI 410	25	25	26	27	27	29			
Lead	37	35	34	33	31				
Magnesium									
Pure	199	156	153	151	149	146			
Alloy A8			84						
Nickel									
Pure	105	91	80	72	66	68	72	76	83
Inconel-X-750	10.3	11.7	13.5	15.1	17.0	20.5	24.0	27.6	30.0
Nichrome		13	14	16	17	21			
Platinum	73	72	72	72	73	76	79	83	90
Silver	420	429	425	419	412	396	379	361	
Tantalum	58	58	58	59	59	59	60	61	62
Tin	73	67	62	60					
Titanium									
Pure	25	22	20	20	19	19	21	22	25
Ti-6Al-4V		5.8							
Tungsten	185	174	159	146	137	125	118	112	106
Zirconium									
Pure	25	23	22	21	21	21	23	26	29
Zircaloy-4	13.3	14.2	15.2	16.2	17.2	19.2	21.2	23.2	

Table A.3 *(Concluded)*

	T K	ρ kg/m^3	c J/kg K	k W/m K	α m^2/s$\times 10^6$
Loose fill					
Cellulose, wood or paper pulp	290			0.038	
	300	45		0.039	
	310			0.042	
Vermiculite, expanded	240			0.058	
	260			0.061	
	280			0.064	
	300	122		0.069	
	320			0.074	
	240			0.052	
	260			0.056	
	280			0.059	
	300	80		0.063	
	320			0.068	
Magnesia	300	270		0.062	
(85 %)	350			0.068	
	400			0.073	
	450			0.078	
	500			0.082	
Paper	300	930	2500	0.13	0.056
Polystyrene, rigid	240			0.023	
	260			0.024	
	280			0.026	
	300	30–60	1210	0.028	0.4–0.8
	320			0.030	
Polyurethane, rigid foam	300	70		0.026	
Rubber					
Hard	270	1200	2010	0.15	0.062
Neoprene	300	1250	1930	0.19	0.079
Rigid foamed	260			0.028	
	280			0.030	
	300	70		0.032	
	320			0.034	
Snow	273	110		0.049	
		500		0.190	
Soil					
Dry	300	1500	1900	1.0	0.35
Wet	300	1900	2200	2.0	0.5
Woods					
Oak, parallel to grain	300	820	2400	0.35	0.18
perpendicular to grain	300	820	2400	0.21	0.11
White Pine, parallel to grain	300	500	2800	0.24	0.17
perpendicular to grain	300	500	2800	0.10	0.071
Wool, sheep	300	145		0.05	

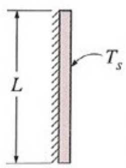
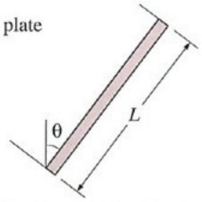
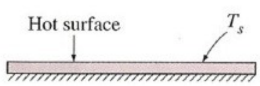
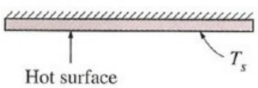
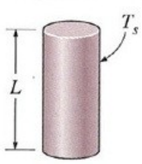
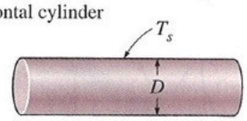
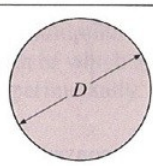
Properties of air at 1 atm pressure

Temp., T, °F	Density, ρ, lbm/ft³	Specific Heat, C_p, Btu/lbm · °F	Thermal Conductivity, k, Btu/h · ft · °F	Thermal Diffusivity, α, ft²/h	Dynamic Viscosity, μ, lbm/ft · h	Kinematic Viscosity, ν, ft²/h	Prandtl Number, Pr
−300	0.24844	0.5072	0.00508	0.0403	0.01454	0.0585	1.4501
−200	0.15276	0.2247	0.00778	0.2266	0.02438	0.1596	0.7042
−100	0.11029	0.2360	0.01037	0.3985	0.03255	0.2951	0.7404
−50	0.09683	0.2389	0.01164	0.5029	0.03623	0.3741	0.7439
0	0.08630	0.2401	0.01288	0.6215	0.03970	0.4601	0.7403
10	0.08446	0.2402	0.01312	0.6468	0.04038	0.4781	0.7391
20	0.08270	0.2403	0.01336	0.6726	0.04104	0.4963	0.7378
30	0.08101	0.2403	0.01361	0.6990	0.04170	0.5148	0.7365
40	0.07939	0.2404	0.01385	0.7259	0.04236	0.5335	0.7350
50	0.07783	0.2404	0.01409	0.7532	0.04300	0.5525	0.7336
60	0.07633	0.2404	0.01433	0.7810	0.04365	0.5718	0.7321
70	0.07489	0.2404	0.01457	0.8093	0.04428	0.5913	0.7306
80	0.07350	0.2404	0.01481	0.8381	0.04491	0.6110	0.7290
90	0.07217	0.2404	0.01505	0.8673	0.04554	0.6310	0.7275
100	0.07088	0.2405	0.01529	0.8969	0.04615	0.6512	0.7260
110	0.06963	0.2405	0.01552	0.9270	0.04677	0.6716	0.7245
120	0.06843	0.2405	0.01576	0.9575	0.04738	0.6923	0.7230
130	0.06727	0.2405	0.01599	0.9884	0.04798	0.7132	0.7216
140	0.06615	0.2406	0.01623	1.0198	0.04858	0.7344	0.7202
150	0.06507	0.2406	0.01646	1.0515	0.04917	0.7558	0.7188
160	0.06402	0.2406	0.01669	1.0836	0.04976	0.7774	0.7174
170	0.06300	0.2407	0.01692	1.1160	0.05035	0.7992	0.7161
180	0.06201	0.2408	0.01715	1.1489	0.05093	0.8213	0.7148
190	0.06106	0.2408	0.01738	1.1821	0.05151	0.8435	0.7136
200	0.06013	0.2409	0.01761	1.2156	0.05208	0.8660	0.7124
250	0.05590	0.2415	0.01874	1.3884	0.05488	0.9818	0.7071
300	0.05222	0.2423	0.01985	1.5690	0.05758	1.1027	0.7028
350	0.04899	0.2433	0.02094	1.7566	0.06020	1.2288	0.6995
400	0.04614	0.2445	0.02200	1.9507	0.06274	1.3598	0.6971
450	0.04361	0.2458	0.02305	2.1508	0.06522	1.4955	0.6953
500	0.04134	0.2472	0.02408	2.3565	0.06762	1.6359	0.6942
600	0.03743	0.2503	0.02608	2.7834	0.07225	1.9300	0.6934
700	0.03421	0.2535	0.02800	3.2292	0.07666	2.2411	0.6940
800	0.03149	0.2568	0.02986	3.6925	0.08088	2.5684	0.6956
900	0.02917	0.2599	0.03164	4.1721	0.08494	2.9112	0.6978
1000	0.02718	0.2630	0.03336	4.6671	0.08883	3.2688	0.7004
1500	0.02024	0.2761	0.04106	7.3465	0.10644	5.2584	0.7158
2000	0.01613	0.2855	0.04752	10.3200	0.12163	7.5418	0.7308
2500	0.01340	0.2922	0.05309	13.5532	0.13501	10.0733	0.7432
3000	0.01147	0.2972	0.05811	17.0526	0.14696	12.8170	0.7516
3500	0.01002	0.3010	0.06293	20.8709	0.15771	15.7428	0.7543
4000	0.00889	0.3040	0.06789	25.1094	0.16745	18.8252	0.7497

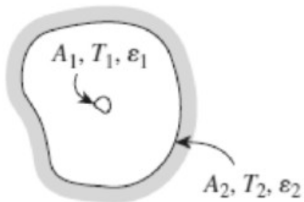
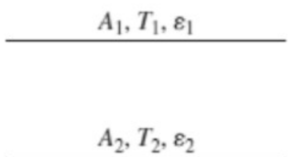
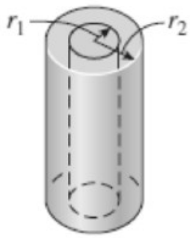
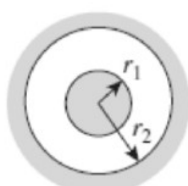
Note: For ideal gases, the properties C_p, k, μ, and Pr are independent of pressure. The properties ρ, ν, and α at a pressure P (in atm) other than 1 atm are determined by multiplying the values of ρ at the given temperature by P and by dividing ν and α by P.

Source: Data generated from the EES software developed by S. A. Klein and F. L. Alvarado. Original sources: Keenan, Chao, Keyes, Gas Tables, Wiley, 198; and *Thermophysical Properties of Matter,* Vol. 3: *Thermal Conductivity,* Y. S. Touloukian, P. E. Liley, S. C. Saxena, Vol. 11: *Viscosity,* Y. S. Touloukian, S. C. Saxena, and P. Hestermans, IFI/Plenun, NY, 1970, ISBN 0-306067020-8.

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

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

Radiation heat transfer relations for some two-surface geometries

Small object in a large cavity 	$\frac{A_1}{A_2} \approx 0$ $F_{12} = 1$	$\dot{Q}_{12} = A_1 \sigma \varepsilon_1 (T_1^4 - T_2^4)$	**(13–37)**
Infinitely large parallel plates 	$A_1 = A_2 = A$ $F_{12} = 1$	$\dot{Q}_{12} = \dfrac{A\sigma (T_1^4 - T_2^4)}{\frac{1}{\varepsilon_1} + \frac{1}{\varepsilon_2} - 1}$	**(13–38)**
Infinitely long concentric cylinders 	$\frac{A_1}{A_2} = \frac{r_1}{r_2}$ $F_{12} = 1$	$\dot{Q}_{12} = \dfrac{A_1 \sigma (T_1^4 - T_2^4)}{\frac{1}{\varepsilon_1} + \frac{1 - \varepsilon_2}{\varepsilon_2} \left(\frac{r_1}{r_2}\right)}$	**(13–39)**
Concentric spheres 	$\frac{A_1}{A_2} = \left(\frac{r_1}{r_2}\right)^2$ $F_{12} = 1$	$\dot{Q}_{12} = \dfrac{A_1 \sigma (T_1^4 - T_2^4)}{\frac{1}{\varepsilon_1} + \frac{1 - \varepsilon_2}{\varepsilon_2} \left(\frac{r_1}{r_2}\right)^2}$	**(13–40)**