CHEG 3143	Free Convection Heat Transfer from	CHEG 3342
Heat Transport	an Upward Facing Plate	Chem. Eng. Lab. II
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INTRODUCTION AND PURPOSE

Free convection heat transfer is encountered in many practical applications. There are correlations for predicting free convection heat transfer coefficients for many different geometries, e.g., refer to page 468, Table 9-1 [Cengel (2003), ref. 1]. One of the important geometries is the upward facing heated surface which is the subject of this investigation.

The purpose of this investigation is to determine the experimental coefficient of free convection heat transfer for the top surface of a horizontal hot plate exposed to air, and to compare these results with literature correlations.

EXPERIMENTAL EQUIPMENT LIST

- Hartman Pro-Tech Model 1600 hair dryer, 1600 watts
- 25.25”x22”x16” cardboard heating box.
- 1.1875” thick Styrofoam insulation lining the cardboard box
- Wooden stand to elevate aluminum plate
- 18”x12”x1.5” aluminum plate that has a black painted finish
- Omega HH12 thermocouple reader
- 1/8” diameter x 12” long sheathed thermocouples
- Stopwatch, graduated in 0.01s
- 1.1875” thick Styrofoam sheet insulation

SCHEMATIC OF EXPERIMENTAL EQUIPMENT

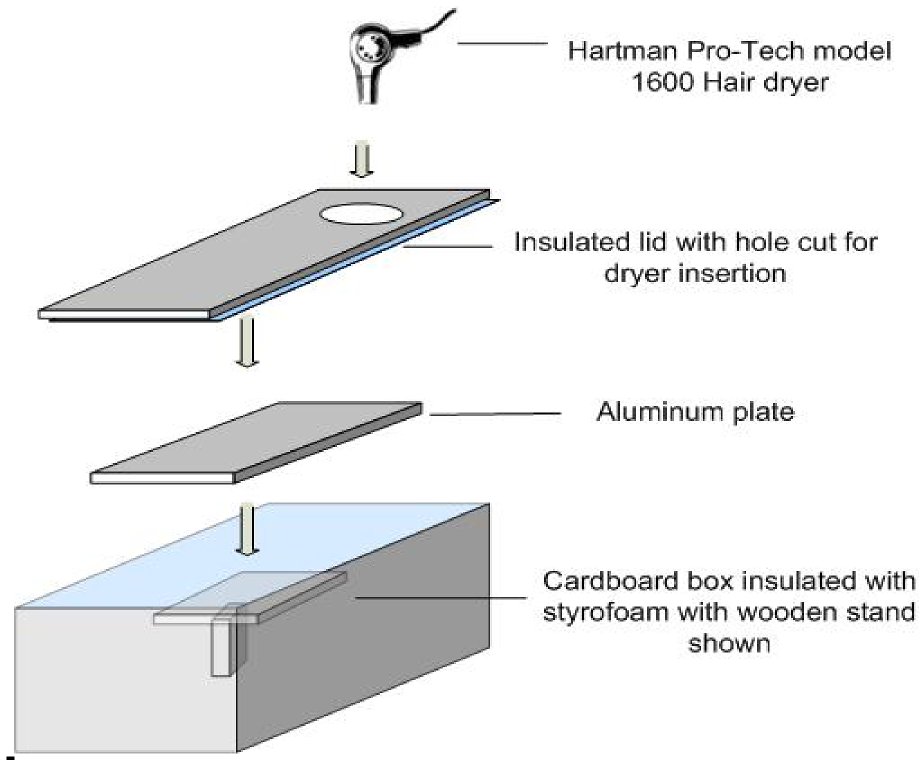

FIGURE 1. ***Schematic for the equipment used in heating the aluminum plate***

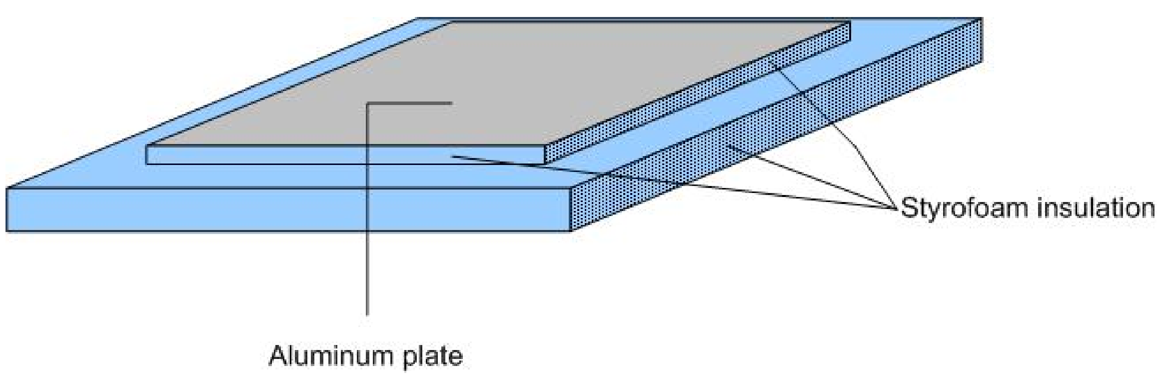

FIGURE 2. ***Schematic for equipment used in cooling of the aluminum plate***

DESCRIPTION OF EQUIPMENT

- **Hartman Pro-Tech Model 1600 hair dryer**
  This is a typical household hair dryer, which can be purchased at Wal-Mart. The aluminum plates are heated by inserting the nozzle of the dryer into the circular opening of the top of the cardboard box lid.
- **Styrofoam insulation**
  The inside of the heated box is insulated with 1.1875" thick Styrofoam insulation. The bottom and sides of the aluminum plate are insulated with the same 1.1875" Styrofoam insulation during an experimental run. This insulation is used in order to heat the aluminum plate efficiently using the hair dryer.
- **Insulated Heating Box (Figures 1 and 3)**
  The cardboard box provides an insulated enclosure for heating the aluminum plate with the hairdryer. A wooden stand is used to hold the plate above the insulated bottom of the box in order to heat the plate thoroughly.
- **Aluminum plate (Figures 2 and 4)**
  18" x 12" x 1.5" aluminum plate with weight of 14.3354 kg. It has a heavily oxidized mill finish. Nathan Weston and Dave Marrs (Group 4) have estimated the emissivity of a black painted surface is 0.98. A 5/32" diameter by 4" deep hole is drilled into the 1.5"x12" face of the plate for insertion of a 1/8" diameter sheathed thermocouple.
- **Omega HH12 thermocouple reader**
  This digital device determines temperature from the electrical signal from the thermocouple.
- **Sheathed thermocouples**
  The type K thermocouple is 1/8" in diameter and 12" long.
- **Stopwatch**
  It is used to determine the time at each °C of cooling.

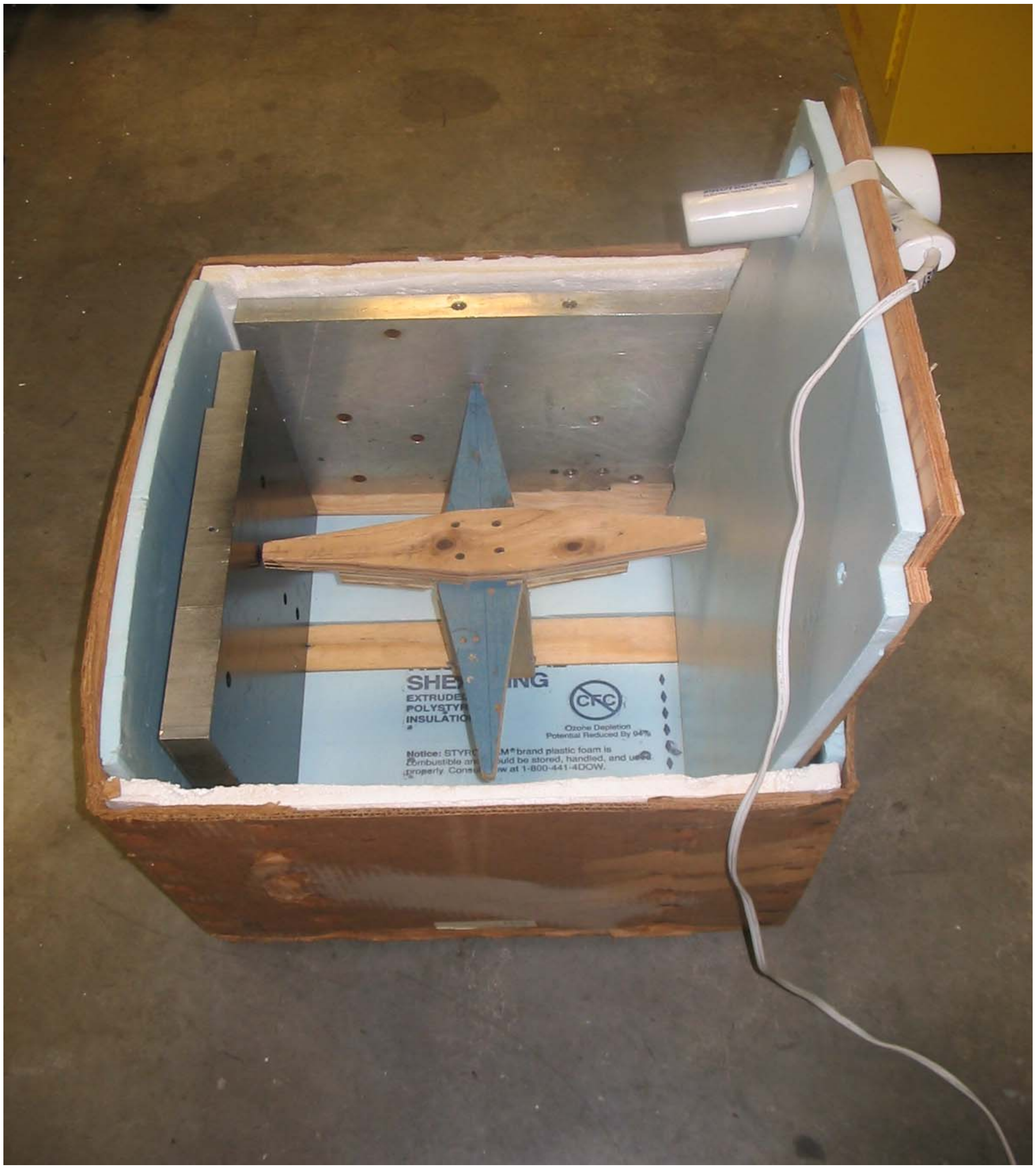

FIGURE 3. Insulated cardboard heating box along with household hairdryer. Wooden stand is shown.

FIGURE 4. Aluminum plate cooling apparatus. Stopwatch and Thermocouple/reader are shown.

EXPERIMENTAL PROCEDURE

Setup

1. Using a scale, weigh the aluminum plate.
2. Determine the surface area of the 12”x18” face of the plate.
3. Make sure air conditioning systems are turned off.
4. Remove the box lid, set the aluminum plate inside the box, atop the stand
5. Place insulated lid on top to seal the box.
6. Place nozzle of the hair dryer into the hole in the lid, plug in, and start.
7. Allow the dryer to heat the plate up to a temperature of approximately 150°F.
8. Turn off the dryer and remove lid.
9. Using insulated gloves, set the aluminum plate on a sheet of styrofoam insulation, and wrap the two 12”x1.5” and two 18”x1.5” faces with the Styrofoam insulation

10. Connect the sheathed thermocouples to the thermocouple reader and insert into the plate.

Testing

1. Start the stopwatch as soon as the temperature changes.
2. Record the time at each successive change in temperature.
3. Repeat as necessary.

SAFETY

- Wear safety glasses at all times.
- Use insulated gloves when handling the heated plate.
- Make sure there is no water present around the electric hair dryer.

DATA REDUCTION PROCEDURE

1. Do a heat balance on the plate.

$$q_{IN} - q_{OUT} + q_{GEN} = q_{ACC} \quad \textbf{(1)}$$

where $q_{IN} = 0$ because the plate is cooling and $q_{GEN} = 0$ because there is no heat generated within the plate.

2. Reduce the equation accordingly.

$$-q_{OUT} = q_{ACC} \quad \textbf{(2)}$$

3. The plate is cooled by free convection and radiation as follows:

$$q_{OUT} = q_{CONV} + q_{RAD} = hA_S(T_{SURFACE} - T_\infty) + \varepsilon\sigma A_S(T_{SURFACE}^4 - T_\infty^4) \quad \textbf{(3)}$$

4. The plate accumulates heat with an inverse relationship to time as it cools back to room temperature. Determine the accumulation of heat.

$$q_{ACC} = m(C_p)\frac{dT}{dt} = \rho V(C_p)\frac{dT}{dt} \quad \textbf{(4)}$$

Therefore, the heat balance reduces to :

$$-\left(hA_S(T_{SURFACE} - T_\infty) + \varepsilon\sigma A_S(T_{SURFACE}^4 - T_\infty^4)\right) = \rho V(C_P)\frac{dT}{dt} \quad \textbf{(5)}$$

5. Input the experimental data of temperature vs. time into a TK data reduction program.

6. Use the data to determine the best fit experimental heat transfer coefficient by integrating **Eq. 5** numerically using a 4th order Runga-Kutta integration with the TK Solver software.

7. Assuming that the surrounding air is an ideal gas, calculate the volumetric expansion coefficient.

$$\beta = \frac{1}{T} \qquad \textbf{(6)}$$

8. Using the length of the plate as the characteristic length in free convection, calculate the Rayleigh number.

$$Ra = \frac{g\beta\left(T_{SURFACE} - T_{\infty}\right)L^3}{\nu^2}\mathbf{Pr} \qquad \textbf{(7)}$$

9. The heat transfer coefficient in **Eq. 5** will vary with the plate temperature; thus the best fit literature coefficient is determined in the integration by using the Churchill/Chu relation from page 468, equation 9-22 (Cengel, 2nd Ed.) to determine the variation of h with plate temperature and air physical properties.

$$Nu = 0.54Ra^{1/4} \qquad 10^4 < Ra < 10^7 \qquad \textbf{(8)}$$

and

$$h_{CORR} = \frac{kNu}{L} \qquad \textbf{(9)}$$

10. The experimental coefficient will not agree with the literature correlation; thus, a fraction of the theoretical coefficient is used to obtain the best fit of **Eq. 5** to the experimental data.

The experimental data are presented in Table 1 and are plotted in Figure 5.

COMPARISON OF EXPERIMENTAL RESULTS WITH CORRELATIONAL CALCULATIONS

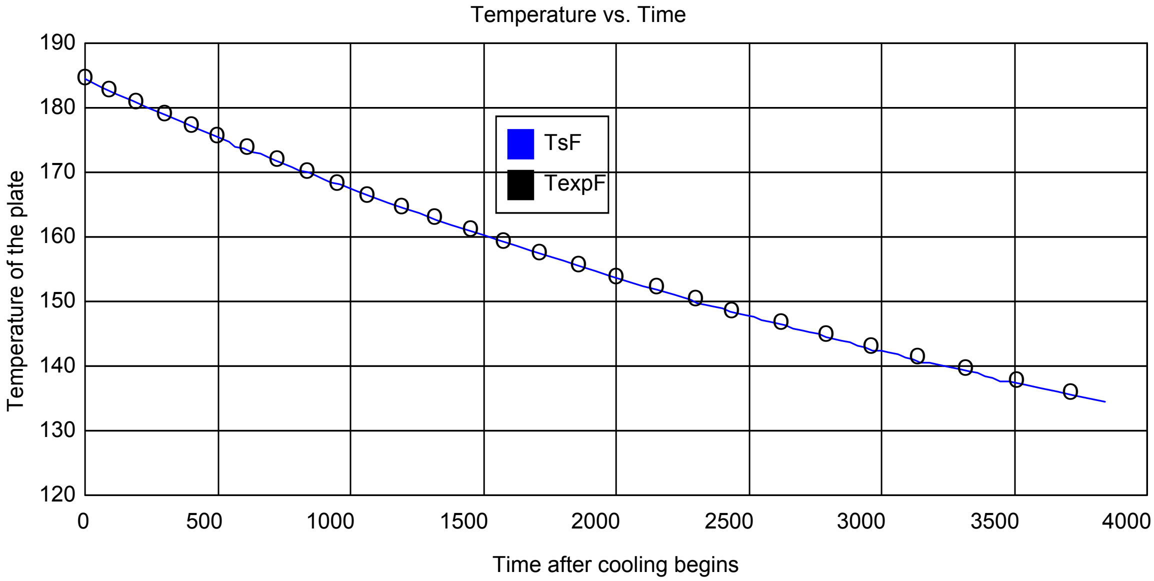

FIGURE 5. ***TK Solver plot for comparison of the experimental and correlated cooling rates for an upward facing aluminum plate***

TABLE 1. ***Experimental data for free convection for the upward facing plate***

Time (s)	T_{exp} (°C)	$h_{calculated}$ (W/m^2K)	h_{used} (W/m^2K)	Time (s)	T_{exp} (°C)	$h_{calculated}$ (W/m^2K)	h_{used} (W/m^2K)
0	85	4.345	8.256	1578.74	71	4.147	7.879
92.58	84	4.338	8.242	1712.23	70	4.132	7.85
194.45	83	4.323	8.213	1860.55	69	4.108	7.806
301.17	82	4.311	8.19	1999.52	68	4.093	7.777
399.97	81	4.299	8.169	2150.84	67	4.078	7.748
498.34	80	4.284	8.14	2298.94	66	4.055	7.705
611.15	79	4.269	8.111	2436.54	65	4.037	7.67
720.4	78	4.254	8.082	2621.47	64	4.005	7.61
835.08	77	4.238	8.053	2789.65	63	3.995	7.59
951.86	76	4.226	8.03	2960.42	62	3.973	7.548
1060.66	75	4.208	7.995	3135.69	61	3.958	7.52
1191.74	74	4.195	7.97	3313.65	60	3.928	7.463
1319.03	73	4.177	7.937	3505.71	59	3.906	7.421
1451.44	72	4.162	7.908	3710.49	58	3.884	7.379

As seen in Figure 5, the cooling of the aluminum plate determined experimentally closely resembles the model produced from the Churchill/Chu relationship used to determine the correlated convection heat transfer coefficient. Though the models share similar trends, the experimental coefficient does not exactly match the correlated coefficient. It was determined that an aforementioned correction ratio was needed, which had a value of 1.9:1 or 1.9, in order to correctly match the experimental data with the correlation. This value was found by trial and error so that the correlated data matched that data found experimentally. The need for this correction arises from the introduced forced convection. It is very difficult to obtain and keep an ideal, free convection atmosphere due to existing air currents. Isolating the apparatus in an enclosed space, turning all air conditioning systems off, and preventing any disturbances caused by movement of any kind kept these air currents to a minimum. Some other general errors were introduced in the experiment such as taking measurements of time and temperature. For instance, as the temperature reached a borderline in changing, the reading from the thermocouple reader oscillated between the two temperatures. The time was recorded as the oscillation began which could introduce an error of the recorded time by ±5 seconds. There were also holes drilled into the upward facing plate. These affect the surface area and volume measured and thus the mass calculated. These measurement errors are insignificant as seen in the reliable data in Figure 5.

CONCLUSIONS

Based on the results of this activity and utilizing a correction coefficient (ratio), it was found that:

1.) The corrected (for forced convection) Churchill/Chu relationship provides a reliable model of how natural convection occurs over a horizontally upward facing plate,

2.) the correction ratio for the experiment conducted is specific to the conditions of the environment at that time and place,

3.) the value for this experiment has the value of 1.9, implying that the experimental heat transfer coefficient is 1.9 times the correlated heat transfer coefficient at a given time, and

4.) in a perfect environment, where no heat transfer is due to forced convection, (i.e. no air currents besides the natural convection current,) the correction ratio would be much closer to 1:1.

RECOMMENDATIONS

1. **BETTER FITTING THERMOCOUPLE**

By using a better fitting thermocouple, more accurate temperatures of the aluminum plate could be measured.

2. BETTER INSULATION ON THE SIDES

By better insulating the sides of the plate, more accurate results for free convection from the *upward* facing plate could be found.

3. ELIMINATE FORCED CONVECTION IN THE ROOM

Finally, by eliminating any other air currents besides the natural convection current, the correction ratio could be decreased, with the experimental convection heat transfer coefficient a better match to the correlation coefficient.

. **NOMENCLATURE

A_S	area for convection, m^2
C_p	specific heat of the aluminum plate, J/kgK
dT	temperature difference, $T_{SURFACE}-T_{SURFACE,0}$, K
dt	time change between readings, $t-t_0$, s
g	gravitational constant, m/s^2
Gr	Grashoff number of the fluid
h	convection heat transfer coefficient, W/m^2K
H	height of the plate, m
hC	correlated heat transfer coefficient, W/m^2K
k	fluid thermal conductivity, W/mK
L	length of the plate, m
m	mass of the plate, kg
NuC	correlated Nusselt number
Pr	Prandtl number of the fluid
q	heat transferred, W
Ra	Rayleigh number of the fluid
$T\infty$	temperature of the surroundings, K
$T_{SURFACE}$	temperature at the surface of the plate, K
V	volume of the plate, m^3
W	width of the plate, m
B	volumetric expansion coefficient, K^{-1}
ε	emissivity of the painted surface
μ	dynamic viscosity of air, Ns/m^2
ν	kinematic viscosity of air, m^2/s
ρ	density of the aluminum plate, kg/m^3
σ	Stefan-Boltzmann constant, W/m^2K^4

REFERENCES

1. Cengel, Yunus A. Heat Transfer: A Practical Approach. 2nd ed. New York: McGraw-Hill, 2003. 459-500.

TABLE 1. ***TK Solver variable sheet for the experimental heat transfer coefficient***

Sta	Input	Name	Output	Unit	Comment
					TRANSIENT COOLING OF A 12"x18"x1.5"
					ALUMINUM PLATE THROUGH UPWARD
					FACING SURFACE.
	2700	ñC			plate Density, kg/m^3
	904	CpC			Specific Heat plate, J/kg K
	.4572	L			Length of plate, m
	.3048	W			Width of plate, m
	.0381	H			Height of plate, m
	299	Ta			Ambient Temperature, K
L	323.15	Ts			Surface Temperature, K
		ì	1.85E-5		Fluid Viscosity, N s/m^2
	1.18	ñ			Fluid Density, kg/m^3
	.000015712	í			Kinematic Viscosity, m^2/s
	.025584	k			Fluid Thermal Conductivity, W/m K
	1007	Cp			Fluid Specific Heat, W/kg K
L		Gr			Grashoff Number (gâ[Ts-Ta]D^3/í^2)
	.72932	Pr			Prandtl Number, ìCp/k
		Ra			Rayleigh Number (GrPr)
		NuC			Correlated Nusselt Number, hD/k
		hC			Correlated Heat Transfer Coefficient, W/m^2 K
		A			Surface Area of plate, m^2
		V			Volume of plate, m^3
		M			Mass of plate, kg
	1	å			Surface Emissivity
	1.95	Ratio			Experimental h/Calculated h
	'G	Fluid			

TABLE 2. ***TK Solver rule sheet for the experimental heat transfer coefficient***

Status	Rule
Comm	; Free Convection & Radiation for a Long Horizontal Cylinder
Sati	Pr = μ*Cp/k
Sati	IF Fluid = "G" THEN β = 1/Tf
Sati	Tf = (Ta + Ts)/2
Cancel	v = μ/ρ
Sati	CALL EulerIntegration(;A,V,M)
Cancel	Gr = ABS(9.8*β*(Ts-Ta)*L^3/v^2)
Cancel	Ra = Gr*Pr
Cancel	hC = NuC*k/D
Cancel	'qConv[i] = hC*A*(Ts[i] - Ta)
Cancel	'qRad[i] = A*ε*σ*(Ts[i]^4 - Ta^4)
Cancel	'qTotal[i] = 'qConv[i] + 'qRad[i]
Cancel	'FractionRad[i] = 'qRad[i]/'qTotal[i]

TABLE 3. ***TK Solver Euler Integration procedure function sheet for the experimental heat transfer coefficient***

Statement
P := 1 ; SYSTEM PRESSURE, ATM
R := 0.082056 ; GAS CONSTANT, M^3-ATM/K KG-MOL
MW := 29 ; MOLECULAR WEIGHT OF AIR, KG/KG-MOL
σ := 5.67e-8; STEFAN-BOLTZMANN CONSTANT
A := L*W ; Surface area of heat transfer
V := L*W*H; Volume of the plate
M := ρC*V; Mass of the plate
dt := 60 ; Step size for time
For i = 53 to 79; This loop converts the experimental temperature from Celcius to Farenheit
'TexpF[i]:=(('TexpC[i]+273)*1.8)-460
next i
For i = 1 to 50; This loop calculates various properties of air as temperature changes
'TsF[i] := ('Ts[i]*1.8 - 460) ; Temperature conversion
'Time[1] := 0 ; Initial time is 0
'Tfilm[i] := ('Ts[i] + Ta)/2 ; Film temperature of the fluid
'β[i] := 1/'Tfilm[i] ; Volumetric expansion coefficient for the fluid
'ρFilm[i] := P*MW/(R*'Tfilm[i]) ; Density of fluid by Ideal Gas Law
'μFilm[i] := 5.36985e-7 + 7.01677e-8*'Tfilm[i] - 3.435e-11*'Tfilm[i]^2 ; dynamic viscosity of the fluid
'vFilm[i] := 'μFilm[i]/'ρFilm[i]; kinematic viscosity of the fluid
'PrFilm[i] := 'μFilm[i]*Cp/k ; Prandtl number for the fluid
'Gr[i] := ABS(9.8*'β[i]*('Ts[i]-Ta)*L^3/'vFilm[i]^2); Grashoff number for the fluid
'Ra[i] := 'Gr[i]*'PrFilm[i]; Rayleigh number for fluid
'NuC[i]:=.54*'Ra[i]^.25 ; Correlated Nussalt number for upward facing flat plate, eq. 9-22 from Table 9-1 in Cengal
'hC[i] := Ratio*'NuC[i]*k/L; Correlated heat transfer coefficient
'qConv[i] := 'hC[i]*A*('Ts[i] - Ta); Heat transfer due to convection
'qRad[i] = A*ε*σ*('Ts[i]^4 - Ta^4); Heat transfer due to radiation
'qTotal[i] := 'qConv[i] + 'qRad[i]; Total heat transfer
'dTdt[i] := - 'qTotal[i]/(M*CpC); Heating rate
'Ts[i+1] := 'Ts[i] + 'dTdt[i]*dt; Temperature at next time step
'qTotal[i] := 'qConv[i] + 'qRad[i]; Total heat transfer
'FractionRad[i] := 'qRad[i]/'qTotal[i]; Fraction of heat transfer due to radiation
'Time[i+1] := 'Time[i] + dt; New time after time step
NEXT i