A SERIES OF HEAT TRANSFER EXPERIMENTS FOR THE MECHANICAL ENGINEERING TECHNOLOGY STUDENT

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Abstract

A series of five heat transfer experiments that are used to teach the laboratory component for a Mechanical Engineering Technology (MET) heat transfer course at Central Washington University (CWU) are presented in this paper. The experiments have been found to be very useful in bridging the gap between theory and hands-on experience. The experiments that will be described in this paper are referred to as: (1) Transient Lumped Mass Heat Transfer, (2) Pin Fin Characterization, (3) Contact Resistance Measurement, (4) Electrically Heated Tube Forced Convection, and (5) Free Convection From a Vertical Heated Surface.

These experiments have all been built by MET students at a relatively low cost and fully tested over the last several years. Design details and approximate costs are presented in the paper so that others may benefit from our experiences. The paper contains a set of test data from each experiment so that the reader may judge the effectiveness of the experiment. All of the experiments utilize state-of-the-art instrumentation and data systems, most of which have been donated by local industry. The students extensively utilize computers for data storage and processing using spread sheets.

The experience gained at CWU in the use of these experiments has been very positive in terms of comments and performance by our students, many of which are non-traditional. Heat transfer at the Engineering Technology level can be difficult to teach, which makes a well thought out set of laboratory experiments crucial to the successful learning of the subject.

Nomenclature

A	cross-sectional area, in^2
A_s	surface area, in^2
Bi	Biot Number
c	specific heat, BTU/lb_m-^oF
D	flow passage diameter, in^2
h	convective heat transfer coefficient, BTU/sec-in^2-^oF
h_c	contact resistance coefficient, BTU/sec-in^2-^oF
k	thermal conductivity, BTU/sec-in-^oF
k_f	thermal conductivity evaluated at the film temperature, BTU/sec-in-^oF
P_n	pressure at location n (n = 0, 1, 2, 3 ...), psia

Pr_f	Prandtl Number evaluated at the film temperature, dimensionless
q_w	heat transfer across a wall, BTU/sec
T_B	fluid bulk temperature, °F
T_f	film temperature ($T_f = 1/2[T_w + T_B]$), °F
T_n	temperature at location n (n = 1,2,3...), °F
$T(t)$	time varying temperature, °F
T_0	initial temperature, °F
T_∞	temperature of an undisturbed fluid, °F
T_w	wall temperature, °F
ΔT_C	temperature change across an interface, °F
$\Delta T/\Delta x$	temperature gradient, °F/in
q/A	heat flux, BTU/sec-in^2
Re_f	Reynolds Number evaluated at the film temperature, dimensionless
S	linear dimension, in
V	volume, in^3
ρ	density, lb_m/in^3
τ	thermal time constant, 1/sec

Introduction

The Mechanical Engineering Technology major at Central Washington University features a laboratory component for almost all courses offered within the major. One such course, Applied Heat Transfer, is taught as a 5-credit hour (quarter system) course which meets for four hours of lecture and two hours of lab per week.

The students normally work in groups of two or three, depending upon the complexity of the experiment. This pooling of energy and knowledge is beneficial to the student. As in industry, the team approach is utilized so the student gets accustomed to collective working and thinking. Each experiment is concluded with a complete lab report which states objectives; lists equipment used; shows a schematic of the experiment, and contains data tables, sample calculations, conclusions, and recommendations. The recommendations are passed on to the next class who then act on the recommendations.

This paper describes a series of five heat transfer laboratory experiments that the student performs, which supports the course syllabus.

Lab 1. Transient Lumped Mass Heat Transfer

The purpose of this experiment is to demonstrate unsteady heat transfer to a lumped mass utilizing simple geometries. Both experimental and analytical approaches are explored. Two masses are used, one an aluminum cube and the other a brass cylinder as shown in Figure 1. Dimensional data are included in Figure 1. Two different thermocouples, K and J types are used so that the student can experience this variation. The thermocouple junction is located at the center of the masses and held in place by using high temperature (500 °F) epoxy.

Aluminum:

$S = 0.750$ in
$V = 0.422$ in^3
$As = 3.357$ in^2

Brass:

$D = 0.750$ in
$V = 0.361$ in^3
A, $= 2.807$ in^2

Figure 1. Test sample geometries, dimensions, and thermophysical data.

The experimental apparatus includes a boiling water and ice water baths, instrumented masses, boiling and ice water thermocouples, and a data logger such as a Fluke Hydra Data Acquisition Unit (Model 2629 A) that we use or alternatively, a strip chart recorder. The mass, a cube or cylinder, is stabilized at room temperature. The room, boiling water bath and ice water bath temperatures are continuously recorded. The mass is then immersed into the boiling water and left there so that the mass's temperature can be recorded as a function of time. After the mass has stabilized in the boiling water, the cube is immersed into the ice water and a measurement of the temperature response of the mass is made. The data is plotted and an analysis is performed to determine the time constant and heat transfer coefficient. Knowledge of the time constant enables an analytical curve of mass temperature versus time to be drawn and compared to the measured temperature response.

Prior to any transient analysis, the Biot Number using equation (1), must be calculated. If Bi < 0.1, then the simple lumped mass analysis applies and equations (2) and (3) are valid.

$$Bi = h(V/A_s)/k \qquad (1)$$

$$[\, T(t) - T_\infty \,] / [\, T_0 - T_\infty \,] = \exp\,(\, t / \tau \,) \qquad (2)$$

$$\tau = \rho\, c\, V / (\, A_S\, h \,) \qquad (3)$$

Equation (2) is solved for τ after setting $T(t) = T^*$ which corresponds to time $t^* = \tau$, the time for the temperature to rise to the 1/e point as shown for heating in figure 2. $T^* = T_0 + 0.632\,(T_o - T_\infty)$. The heat transfer coefficient may be calculated by solving for it from equation (3). The same procedure is followed for cooling except that $T^* = T_0 - 0.632(T_0 - T_\infty)$.

An analytically calculated temperature response curve for the lumped mass may be estimated by using the calculated value for τ and varying t over a specified interval of time. A comparison of experimental and analytical response curves, for heating and cooling of the aluminum cube are shown in Figure 3. Also shown in the figure are the respective computed values for Bi, τ, and h.

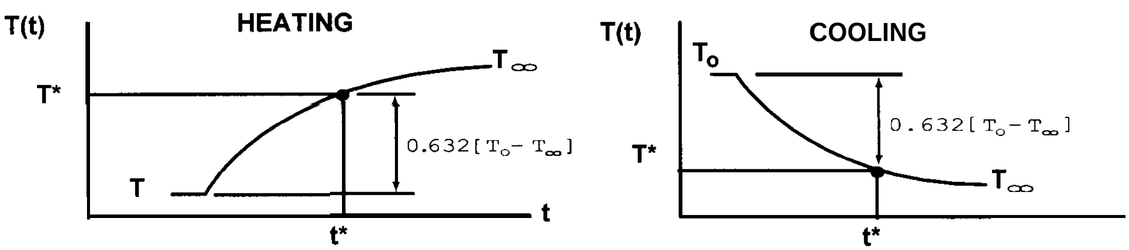
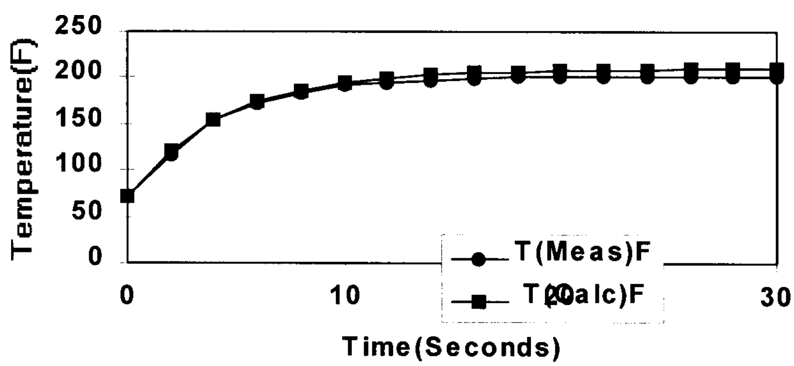


Figure 2. Technique for evaluating the thermal time constant, τ.

The measured and calculated curves are much more in agreement for the case of heating rather than for cooling. The magnitude of the convective heat transfer coefficients is high which suggests very strong convection or possibly moderate boiling. The data for the brass cylinder was qualitatively similar to the data for the aluminum cube.

If a thermocouple readout and strip chart recorder are available, the cost of duplicating this experiment is approximately $50.

Heating of Aluminum Cube

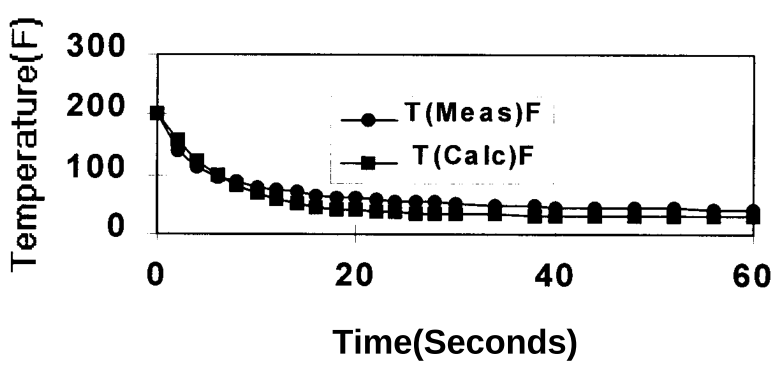


Bi=0.0345

τ = 3.5 sec

h = 4.51×10^{-2} Btu/min-in^2-°F

Cooling of Aluminum Cube



Bi=0.360

τ = 3.5 sec.

h = 4.64×10^{-2}Btu/min-in^2-°F

Figure 3, Comparison of thermal response for heating and cooling of an aluminum cube.

Lab 2. Pin Fin Characterization

Heat transfer from heat exchanger surfaces can be effectively augmented by using fins. Many industrial applications abound. Although the pin fin is not as routinely used as other fin geometries, it does represent a simple configuration to analyze.

The objective of this experiment is to develop an understanding of the thermal performance of a pin tin by performing an experimental and analytical study of several pin fin geometries. Figure 4 shows three fin configurations that were built and tested. Each fin was made from 1050-O aluminum stock and instrumented with 5 thermocouples starting with a base thermocouple and progressing outward toward the tip with the last thermocouple 0.200 in. from the tip.

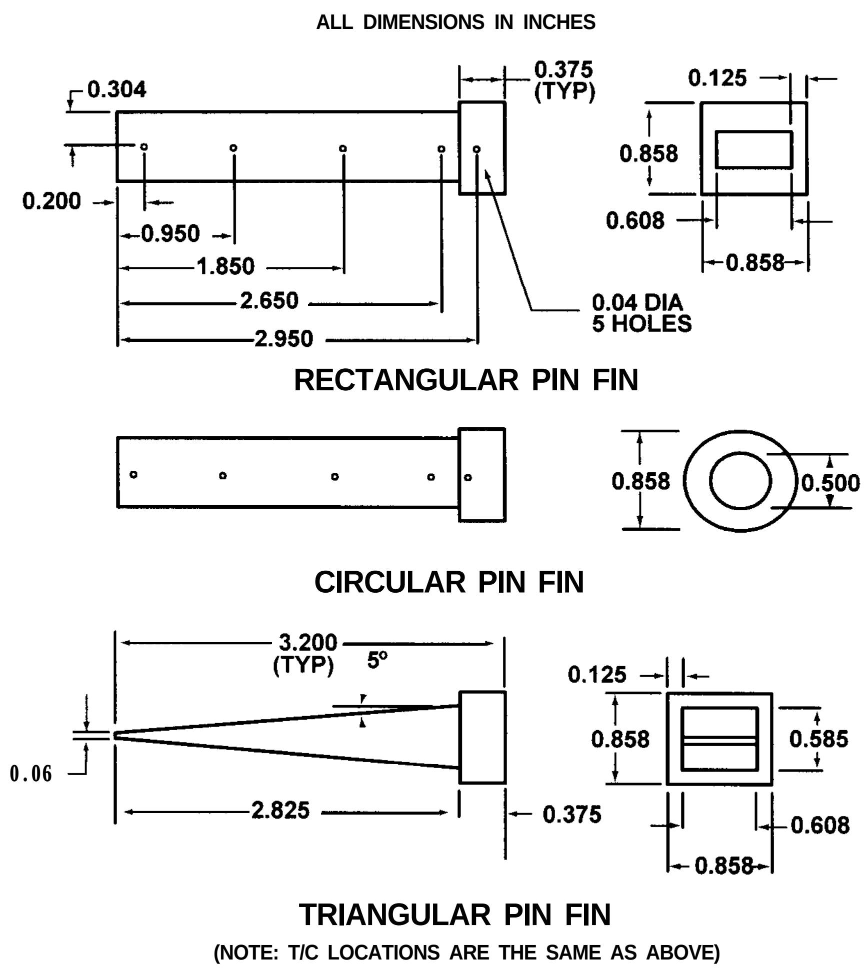


Figure 4. Pin fin configurations.

As shown in Figure 5, the fins were installed just inside of the exit plane of a 3 in. x 4 in. wind tunnel that was constructed of sheet metal by the students. The wind tunnel was driven by a scrapped-out 1/4 hp fan. The flow was straightened by utilizing a rectangular bundle of plastic drinking straws positioned at the midpoint of the duct. Heat was supplied to the fin base by an electric heater which was insulated from the environment to reduce the heat loss. The 5 "K' type thermocouples were

connected to a Fluke Hydra data logger for data acquisition and recording.
The average velocity of the wind tunnel was measured by a Davis Instruments, Turbo Meter, wind speed indicator. These measurements were also verified by traversing a pitot tube across the exit plane of the wind tunnel.

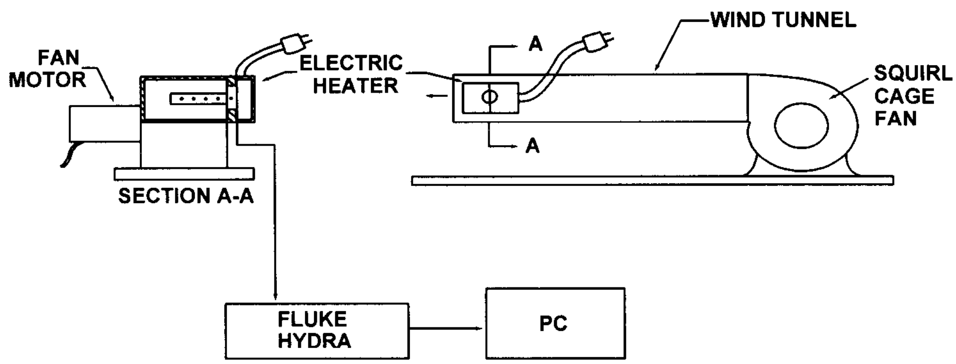


Figure 5. Experimental setup for the pin fin experiment.

The lengthwise variation in fin temperature is shown in figure 6 for the circular, rectangular, and triangular pin fins. Data from the circular and rectangular pin fins indicates only minor differences in temperature profiles. However, the triangular pin fin reaches a much lower temperature at the fin tip. This fact is borne out by data presented in Table 1. The actual and theoretical efficiencies indicate that the triangular shaped fin is much more efficient than the circular or rectangular fins. The least efficient fin is the rectangular fin. Qualitatively, the actual and theoretical fin efficiencies are similar. The actual fin efficiency was calculated by ratioing the actual fin heat transfer to the maximum heat transfer'. The actual fin heat transfer was determined by calculating the temperature gradient between the fin base thermocouple and the first thermocouple just past the base and using the Fourier Law of conduction. The maximum heat transfer was calculated using Newton's equation for cooling by convection. A convective heat transfer coefficient was calculated from measured air velocities using a correlation presented by Holman[2] for circular and non circular cylinders in cross flow. In this analysis it was assumed that the entire fin was operating at the fin base temperature. Theoretical efficiencies were calculated using charts presented by Cengel[3].

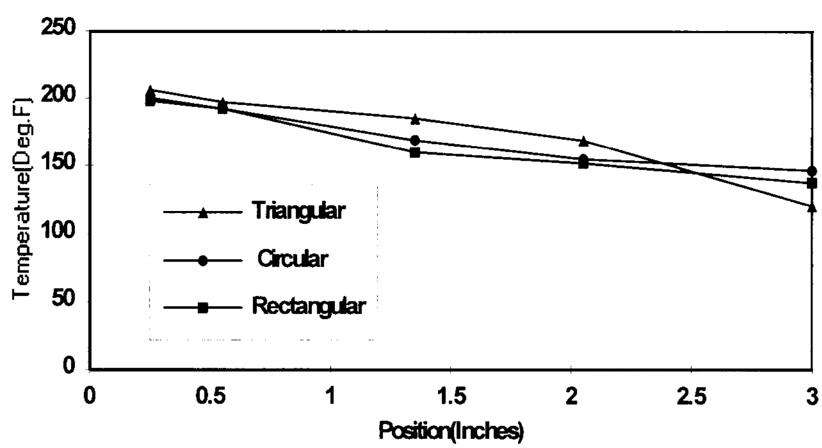


Figure 6. Pin Fin Performance.

Assuming that a thermocouple readout is available as well as access to a machine shop, the experiment can be duplicated for approximately $550.

Table 1. Fin Efficiency Comparison

Fin Type	Actual	Theoretical
Circular	0.61	0.75
Rectangular	0.51	0.73
Triangular	0.94	0.86

Lab 3. Contact Resistance Measurement

If two solids, each having plane flat surfaces, are brought in contact with each other by pressing one surface against the other, the actual direct contact between the two surfaces takes place at only a limited number of spots. Since the surfaces are not perfectly smooth, there are pockets or voids present. The voids are filled with the fluid of the surroundings which is usually air. The heat flowing across the interface takes place by conduction through the fluid in the voids as well as through those areas that have metal-to-metal contact. Since the thermal conductivity through the void fluid is less than through the metal, the void fluid acts as a resistance to the heat flow. The resistance is confined to a very thin layer between the surfaces, thus it is called *thermal contact resistance.* Experiments have shown that there exists a steep drop in temperature across the interface. The magnitude of this thermal resistance depends upon the surface roughness, type of materials, interface pressure, interface temperature level, and type of fluid filling the void.

The objective of this experiment is to demonstrate the principle of thermal contact resistance when two bodies of differing temperature are placed in contact with each other. Figure 7 shows a drawing of the test apparatus. Two aluminum rods, 1.0 in. diameter by 6 in. long, are forced into contact by a screw which applies interfacial pressure to the two mating parts. Heat is applied to the left hand end of the apparatus to produce temperatures in the 220 $^{\circ}$F range using an electric heat gun while cold water cools the right hand end. Three thermocouples are imbedded into the center of the each rod and spaced as shown in Figure 7. The rods are enclosed by a 3.0 in. x 3.0 in. foam insulation block. Two sets of rods have been fabricated, one with smooth surfaces (16G micro-inches RMS), the second with roughened surfaces (250ST micro-inches RMS). The roughened rods were also tested with thermal contact grease to demonstrate the reduction of thermal resistance when the void fluid is replaced with a fluid of much higher thermal conductivity.

Data from the thermocouples shown in Figure 7 was collected by the Fluke Hydra and down loaded into a PC spreadsheet (Microsoft Excel). The measured temperatures are shown graphically in Figure 8. This experiment offers many interesting analyses to be performed. If one uses the temperature gradient between thermocouple 1 and 3 in conjunction with the Fourier Law for conduction (equation 7), then a good estimate of heat flux q/A is obtained.

Equation 8 is then solved for (h_c) the contact resistance coefficient. Values of q/A, ΔT_C, and h_C are presented in Table 1.

$$q/A = k\,(\Delta T / \Delta x) \quad (7)$$
$$h_C = (q/A) / (\Delta T_C) \quad (8)$$

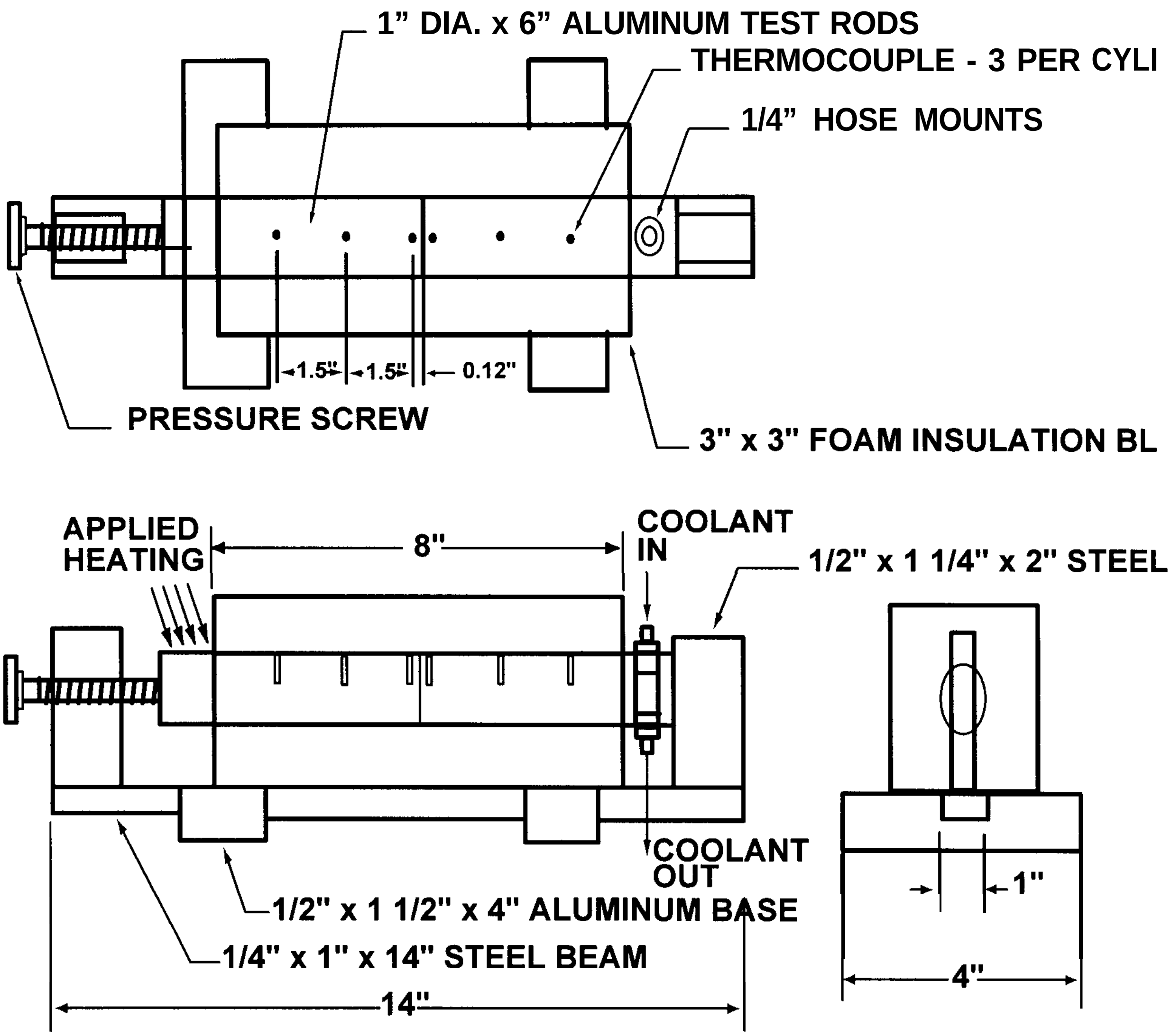


Figure 7. Thermal contact resistance test apparatus.

Results in Figure 8 and Table 2 offer no surprises. There is a significant improvement in heat flux across the interface when thermal grease is used to fill voids in the rough sample. The smooth interface allows a 35% increase in heat transfer over the rough interface.
Comparison of contact resistance coefficient (h_c) data found in heat transfer texts[4] shows reasonably close agreement. For a smooth 100 micro-inch roughened aluminum sample at 12-25 atm pressure, h_c is 3.80×10^{-3} BTU/sec-in^2-°F. This value compares quite well with an h_c of 2.97×10^{-3} BTU/sec-in^2-°F shown in Table 2.

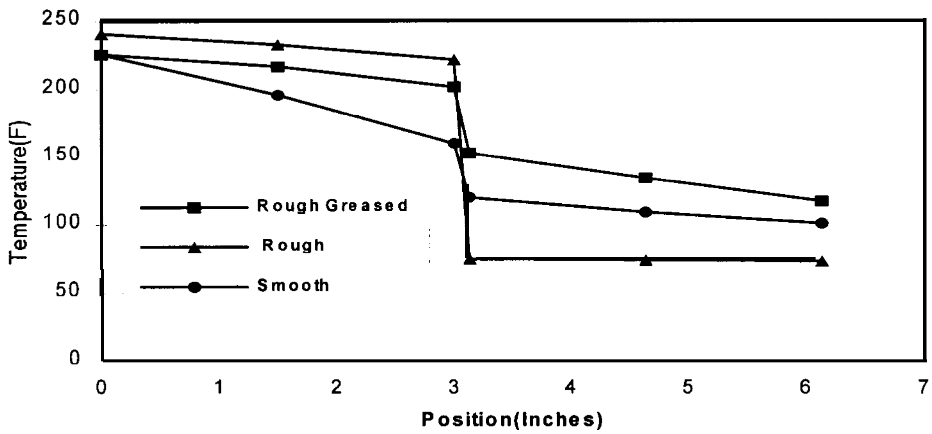


Figure 8. Results from a typical thermal contact resistance experiment.

Table 2. Comparison of Heat Transfer Parameters Across the Contact Area

Interface Type	q/A (BTU/sec-in^2)	(°F)	h_c (BTU/sec-in^2-°F)
Rough	0.0171	66	3.64×10^{-4}
Rough/Grease	0.0210	24	4.29×10^{-4}
Smooth	0.0594	19	2.97×10^{-3}

Lab 4. Electrically Heated Tube Experiment

The heat transfer behavior of a fluid in forced convection flow can be characterized by its ability to transfer heat between itself and a solid surface, across a temperature difference. The conventional relationship used to express heat transfer rate per unit surface area as a function of the convective heat transfer coefficient and the wall to fluid temperature difference is given by equation (9). The magnitude of the convective heat transfer coefficient, as defined in equation (9), is primarily controlled by velocity, properties of the fluid, and geometry of the flow passage.

$$q_w A_s = h\,(T_w - T_F) \quad \text{...} \quad (9)$$

Of the various possible experimental arrangements for the determination of the convective heat transfer coefficient for flow in circular passages, use of electrically heated tubes provides the simplest and most direct approach. In this approach, the fluid being evaluated flows through a thin-walled heated tube; the heat is supplied by resistive heating of the tube wall. The most convenient method of resistive heating uses the tube itself as the resistance element, passing current axially through the tube, thereby causing internal generation of heat within the tube wall. This method is quite satisfactory as long as the electrical conductance of the fluid is small compared to that of the tube material. If the heated tube is thermally insulated over its external surface, nearly all of the heat generated is absorbed by the fluid flowing within the tube. Thus, measuring the power dissipation provides a direct determination of the total heat flow to the fluid, as well as the heat flux distribution over the inner surfaces of the tube. Thermocouples attached to the outer surface of the tube measure temperatures that are readily related to the wall temperature at the inner surface. Measurement of the inlet fluid temperature, pressure, mass flowrate, and electrical power dissipated are used to calculate the fluid bulk temperature along the length of the heated tube and the corresponding convective heat transfer coefficient.

Figure 9 shows a drawing of the electrically heated tube. The tube is made from 316 stainless steel and is 0.250 in. outside diameter with a 0.035 in. wall thickness having a resistance of 28.3×10^{-6} Ohm-inches. Two copper buss bars are clamped to either end of the tube.

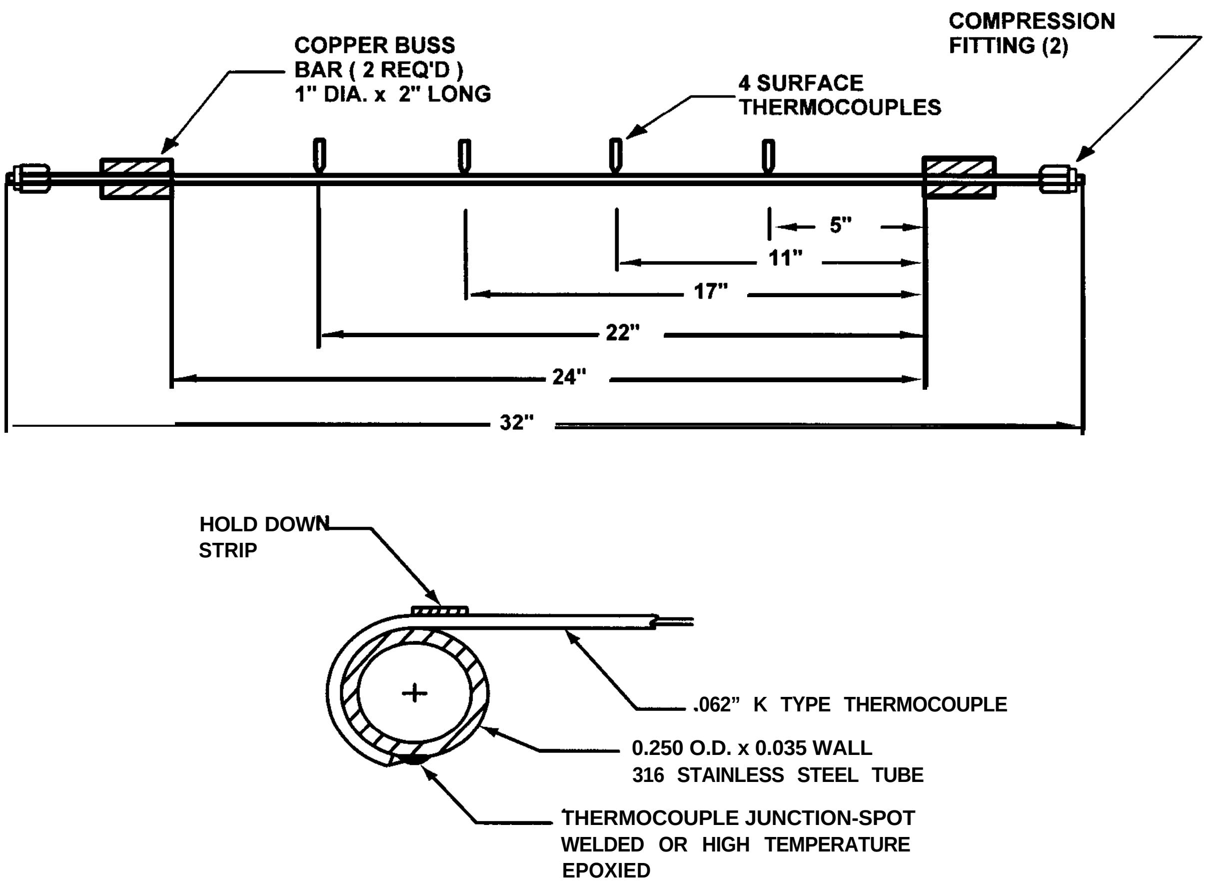


Figure 9. Electrically heated tube instrumented with thermocouples

Four 0.062 in. “K” type thermocouples are attached to the outer surface of the tube, and are spaced at equal distances apart. The tube cross-sectioned view shows how a typical thermocouple is attached to the tube. The thermocouples can either be spot welded or high temperature (500 “F) epoxied to the tube. A thin piece of shim stock provides strain relief for the thermocouple which is wrapped 180° around the tube. The assembly is completed when compression fittings are attached to the ends of the tube and the tube is insulated with fiberglass.

A typical arrangement for the electrically heated tube experiment is shown in figure 10. Standard pressure gages are used to measure P,, P,, and P,. Closed junction thermocouple probes are used to measure inlet and outlet temperatures. Electrical power from an AC arc welder is determined by measuring current and voltage. It should be emphasized that an AC power source is preferred over a DC source primarily because the thermocouples may be directly attached to the tube wall. If a DC source was used then the thermocouples would have to be electrically insulated from the tube surface otherwise each thermocouple would have a DC voltage proportional to the distance from the ground potential buss bar. There are other secondary reasons for this choice.

Typical results of an experiment performed with air flowing through the electrically heat tube apparatus is described here. The inlet conditions were kept constant at 75°F, a

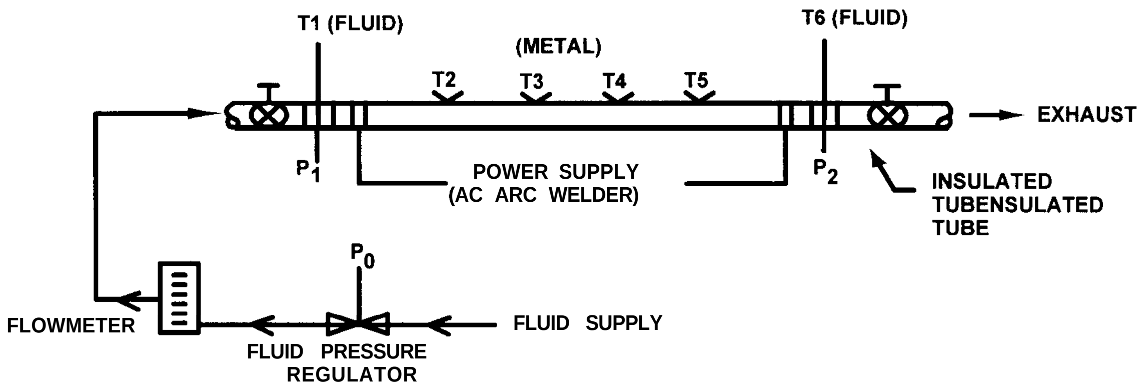


Figure 10. Experimental set-up for an electrically heated tube test.

pressure of 94 psia and an air flow rate of 0.045 lb_m/min. The Reynolds Number was turbulent for this experiment and calculated to range from approximately 4046 to 5128 depending upon power level and location within the tube. The electrical power was varied from 0 to 51.5 Watts with an electrical to thermal conversion efficiency of 81.5%. This efficiency is lower than a desired 90% or better. Figure 11 is a typical plot of the wall and fluid bulk temperature distribution with distance in the flow direction for the 51.5 Watt test point. The plotted points refer to the thermocouple measurements for wall temperature whereas, the fluid temperatures are calculated from know inlet and exit temperature measurements and the fact that electrical heating produces a constant heat flow with length. The fact that the fluid and wall temperatures vary linearly with distance in the flow direction with a constant temperature difference is an indication of constant heat flux heating.

Table 3 shows a comparison of experimental and predicted heat transfer coefficients for the test point of Figure 11. Equation (9) was used to determine the experimental value of (h) whereas the Ditties-Boelter equation (10) was used to determine the predicted values. The agreement is reasonably close despite the fact that the flows were barely in the turbulent regime and the measured outer wall temperatures were taken as the inner wall temperatures. Nevertheless, the experiment is a powerful tool in demonstrating forced convection heat transfer to the student It should be noted that a liquid such as water could also be tested. In

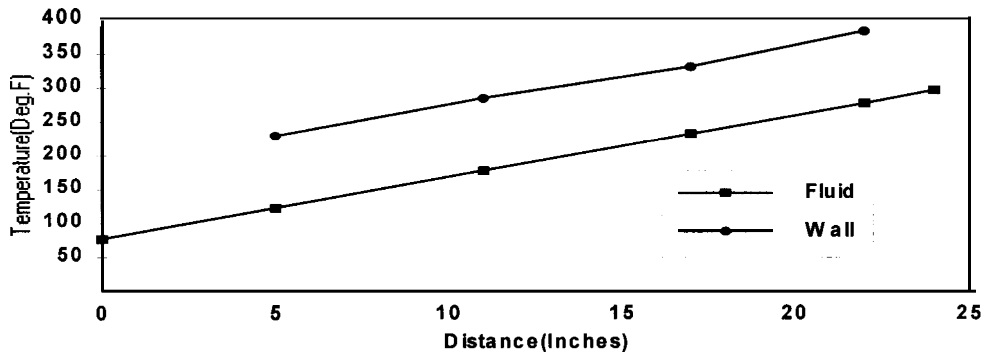


Figure 11. Measured fluid and wall temperatures for heated tube experiment

this case, a short section of non-conducting tubing should be used at the inlet to isolate the electrically heated tube from the remainder of the fluid system.

This experiment could be duplicated for about $800 including the purchase of a small arc welding machine.

$$h=0.023(Re_f)^{0.8}(Pr_f)k_f(l/D) \quad (10)$$

Table 3. Comparison of Experimental and Predicted Forced Convective Heat Transfer Coefficients

Position (inches)	Experimental (h) (Btu/min-ft^2-^{o}F)	Predicted (h) (Btu/min-ft^2-^{o}F)
5.0	0.250	0.296
11.0	0.250	0.306
17.0	0.274	0.330
22.0	0.258	0.282

Lab 5. Free Convection From a Vertical Surface

Natural or free convection occurs whenever a body is placed in a fluid at a higher or lower temperature than that of the body. As a result of the temperature difference, heat flows between the fluid and the body and causes a change in the density of the fluid in the vicinity of the surface. For a vertically heated surface, there is a flow of fluid in the upward direction which is solely due to differences in density resulting from temperature gradients. The mixing motion in free convective currents is not as strong as for forced convection, so the rate of heat transfer is lower than for forced convection, and consequently free convection heat transfer coefficients are also lower. Although free convection heat transfer coefficients are relatively small, many devices depend largely on this mode of heat transfer for cooling such as electrical furnaces, electrical transmission lines, transformers, and some electronic components.

The objective of this experiment is to experimentally determine the thickness of the thermal boundary layer and resulting free convective heat transfer coefficients. As shown in Figure 12, the experimental apparatus used was very basic and simple to setup. A Thermolyne type 1900 hot plate having a 6 in. x 6 in. hot surface is positioned approximately 6 in. above a metal base plate. A 3-axis translation stage holds a 0.060, “K" type, bare junction thermocouple probe. The thermocouple probe is translated vertically upward from the bottom of the plate along three vertical lines, center and 1.5 in. to either side of the center.

Position measurements normal to the hot plate are read accurately using a micrometer. The thermal boundary was determined by measuring the temperature of the air surrounding the hot

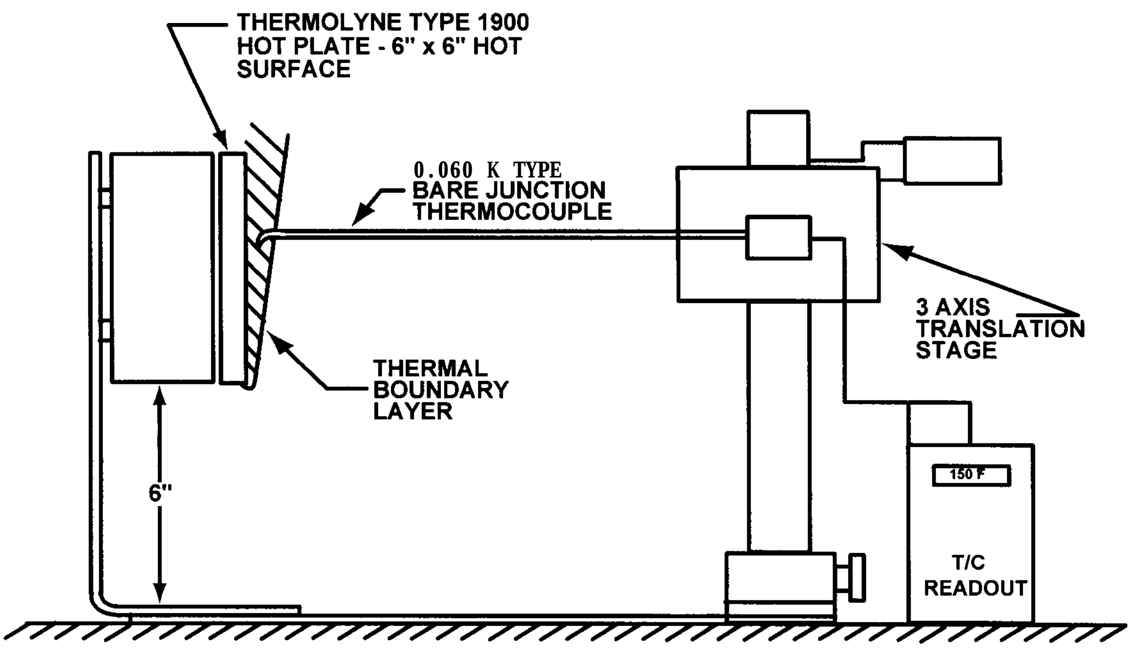


Figure 12. Configuration used for the vertically heatedplate experiments.

plate. The thermocouple probe was translated horizontally, starting at the hot plate and progressively moving outwardly normal to the plate. The edge of the thermal boundary layer was defined as the location where the air temperature was within 2 °F of the undisturbed air. The temperatures were recorded by a Fluke Thermocouple Thermometer.

The metal wall temperature was measured by two independent techniques. Initially, the entire hot surface was thermally mapped by measuring the surface temperature with an infrared thermometer for three levels of hot plate settings. A Kane-May Infratrace 1500 infrared thermometer was used. The second technique involved using the thermocouple probe to touch a desired spot on the hot plate surface. This technique was less accurate than using the infrared thermometer probe due to conduction heat losses to the probe body. Alternatively the Infratrace 1500 integrated temperatures over a spot of approximately 0.5 in. in diameter.

Figure 13 shows how the thermal boundary layer grows in the upward direction along the hot plate. Near the bottom edge of the plate, the thermal boundary layer is 0.175 in. in thickness. It is presumed that the actual start of the boundary layer was around the lower edge of the plate. At the upper edge of the plate, the thermal boundary layer thickness is 0.46 in. Figure 14 shows the temperature profile normal to the plate, on the centerline of the plate, at 3.5 in. from the bottom of the plate. The temperature gradient at the wall was used to estimate the heat flux and a free convective coefficient. The heat flux was 1728 Btu/hr-ft^2 while the heat transfer coefficient was 1.24 Btu/hr-ft^{2}-°F. Using theory developed for natural

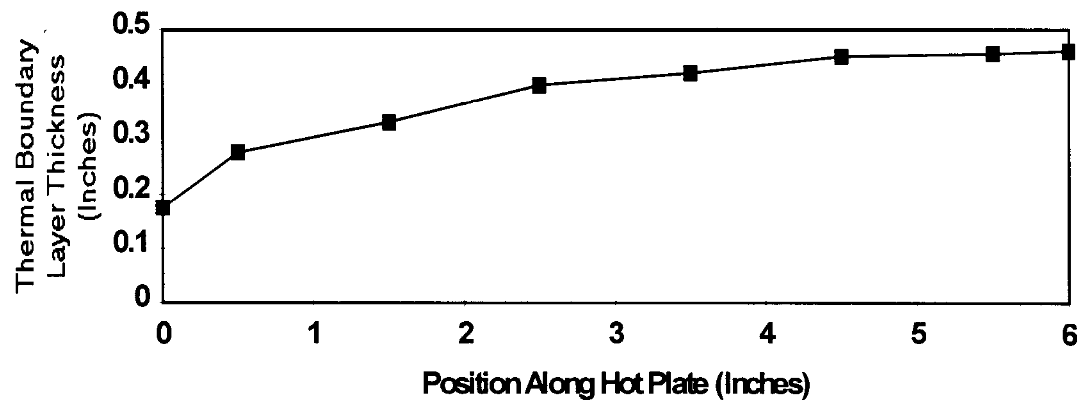


Figure 13. Thermal boundary layer growth along vertical plate.

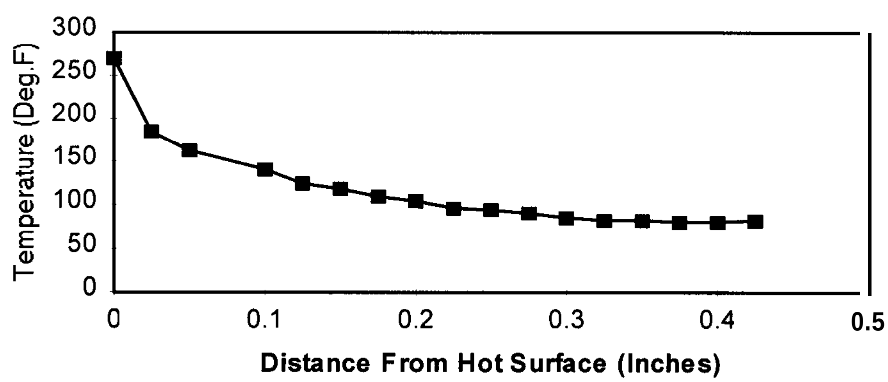


Figure 14. Temperature profile normal to hot plate at centerline, 3.5 inches from bottom edge

convection about a vertical, heated plate', a length Grashof Number of 4.420x10^7 was calculated resulting in length Nusselt Number of 43.89 and a predicted heat transfer coefficient of 1.57 Btu/hr-ft^2-°F. The measured and predicted heat transfer coefficients are close enough to give the student an appreciation for the mechanisms and theory involved in natural convection.

Assuming that a thermocouple readout is available, the experiment can be duplicated for approximately $650.

Conclusions

The experiments described in this paper were found to be both challenging and interesting to the students based on comments and student course reviews. The experiments were found to significantly contribute to the understanding heat transfer and supported the basic framework of the Mechanical Engineering Technology curriculum.

Acknowledgments

The author wishes to thank all of the Mechanical Engineering Technology students, past and present, at Central Washington University for their enthusiastic participation in the heat transfer experiments described in this paper.

References

1. **Cengel, Younis, A.,** Heat Transfer, A Practical Approach, 1st ed., pp 177- 184 WCB/McGraw Hill, Boston, MA, 1998.

2. **Holman, J. P.,** Heat Transfer, 8th ed., pp 302-307, McGraw Hill Companies, Inc., New York, NY, 1997.

3. **Cengel, Younis, A.,** Heat Transfer. A Practical Approach, 1st ed., p 185, WCB/McGraw Hill, Boston, MA, 1998.

4. **Holman, J. P.,** Heat Transfer, 8th ed., pp 56-59, McGraw Hill Companies, Inc., New York, NY, 1997.

5. **Holman, J. P.,** Heat Transfer, 8th ed., pp 336-346, McGraw Hill Companies, Inc., New York, NY, 1997.