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HEAT TRANSFER BY FREE AND FORCED CONVECTION

1. INTRODUCTION

The transfer of energy in the form of heat occurs in many chemical and other types of processes which combine with other unit operations, such as drying of lumber or foods, alcohol distillation, burning of fuel, and evaporation. When two objects at different temperatures are brought into thermal contact, heat flows from the object at higher temperature to that at the lower temperature. The net flow is always in the direction of the temperature decrease. The mechanisms by which the heat may flow are three: conduction, convection and radiation.

In conduction, heat transfer is present to some extent in all solids, gases or liquids in which a temperature gradient exists. The heat is conducted by the transfer of the energy of motion between adjacent molecules and can also be transferred by free electrons, which is quite important in metallic solids (through walls of exchanger or a refrigerator, freezing of the ground during the winter, etc.).

Heat transfer by ***convection*** implies the transfer of heat by bulk transport and mixing of macroscopic elements of warmer portions with cooler portions of a gas or a liquid. It also often involves the energy exchange between a solid surface and a fluid. A distinction must be made between forced convection heat transfer, where a fluid is forced to flow past a solid surface by a pump, fan, or other mechanical means, and natural or free convection, where warmer or cooler fluid next to the solid surface causes a circulation because of a density difference resulting from the temperature differences in the fluid. (Loss of heat from a car radiator where the air is being circulated by a fan, cooking of foods in a vessel being stirred, cooling of a hot cup of coffee by over the surface, etc.).

Radiation differs from heat transfer by conduction and convection in that no physical medium is needed for its propagation. Radiation is the transfer of energy through space by means of electromagnetic waves in much the same way as electromagnetic light waves transfer light. The same law which governs the transfer of light governs the radiant transfer of heat. Solids and liquids tend to absorb the radiation being transferred through it, so that radiation is important primarily in transfer through space or gases. The most important example of radiation is the transport of heat to the earth from the sun (others; cooking of food when passed below red-hot electric heaters, heating of fluids in coils of tubing inside combustion furnace, etc.).

2. THEORY

2.1. Fourier's Law of Heat Conduction

All three of the molecular transport processes of momentum, heat or thermal energy and mass are characterized in the elementary sense by the same general type of transport equation.

$$\text{Rate of a transfer process} = \frac{\text{driving force}}{\text{resistance}}$$

This equation states that we need a driving force to overcome a resistance in order to transport a property.

The transfer of heat by conduction also follows this general transport equation and is written as Fourier's Law for heat conduction in fluids or solids.

$$\frac{q_x}{A} = -k\frac{dT}{dx} \qquad \text{(Eq.1)}$$

where q_x is the heat transfer rate in the x direction in watts (W), A is the cross-sectional area normal to the direction of flow of heat in m^2, T is temperature in K, x is distance in m, and k is the thermal conductivity in W/m.K in the SI system. The quantity q_x/A is called the heat flux in W/m^2. The quantity dT/dx is the temperature gradient in the x direction.

In the gases, the mechanism of thermal conduction is relatively simple. The molecules are in continuous random motion, colliding with one another and exchanging energy and momentum. If a molecule moves from a high temperature region to a region of lower temperature, it transports kinetic energy to this region and gives up this energy through collisions with lower energy molecules. Since smaller molecules move faster, gases such as hydrogen should have higher thermal conductivities.

The thermal conductivity increases approximately as the square root of the absolute temperature and is independent of pressure up to a few atmospheres. At very low pressures (vacuum), however, the thermal conductivity approaches zero.

The physical mechanism of conduction of energy ***in liquids*** is somewhat similar to that of gases, where higher energy molecules collide with lower energy molecules. However, the molecules are packed so closely together that molecular force fields exert a strong effect on the energy exchange. The thermal conductivity of liquids varies moderately with temperature and often can be expressed as a linear variation,

$$k=a+bT \qquad \text{(Eq.2)}$$

where a and b are empirical constants. Thermal conductivities of liquids are essentially independent of pressure.

Heat is conducted ***through solids*** by two mechanisms. In the first, which applies primarily to metallic solids, heat like electricity, is conducted by free electrons which move through the metal lattice. In the second mechanism, present in all solids, heat is conducted by the transmission of energy of vibration between adjacent atoms.

2.2. Convective Heat Transfer

There are two main classifications of convective heat transfer. The first is free or natural convection, where the motion of the fluid results from the density changes in heat transfer. The buoyant effect produces a natural circulation of the fluid, so it moves past the solid surface. In the second type, forced convection, the fluid is forced to flow by pressure differences, a pump, a fan, and so on.

It is well known that a hot piece of material will cool faster when air is blown or forced by the object. When the fluid outside the solid surface is in forced or natural convective motion, we express the rate of heat transfer from the solid to the fluid, or vice versa, by the following equation:

$$q=hA(Tw-Tf) \qquad \text{(Eq.3)}$$

where q is the heat transfer rate in W, A is the area in m^2, Tw is the temperature of the solid surface in K, Tf, is the average or bulk temperature of the fluid flowing by in K, and h is the convective heat transfer coefficient in W/ m^2K. The coefficient h is a function of the system geometry, fluid properties, flow velocity, and temperature difference.

Heat Transfer by Forced Convection

Heat transfer by convection is due to fluid motion. Cold fluid adjacent to a hot surface receives heat which imparts to the bulk of the cold fluid by mixing it. When the fluid is mechanically agitated, the heat is transferred by forced convection. The mechanical agitation may be supplied by stirring, although in most process applications it is induced by circulating the hot and cold fluids at high rates on the opposite sides of pipes or tubes.

In most situations involving a liquid or a gas in heat transfer, convective heat transfer usually occurs as well as conduction. In most industrial processes where heat transfer is occurring, heat is being transformed from one fluid through a solid wall to a second fluid.

The velocity gradient, when the fluid is in turbulent flow, is very steep next to the wall in the thin viscous sublayer where turbulence is absent. Here, the heat transfer is mainly by conduction with a large temperature difference of T_2-T_3 in the warm fluid.

As we move farther away from the wall, we approach the turbulent region, where rapidly moving eddies tend to equalize the temperature. Hence, the temperature gradient is less and the difference T_1-T_2 is small. The average temperature of fluid A is slightly less than the peak value T_1. A similar explanation can be given for the temperature profile in the cold fluid.

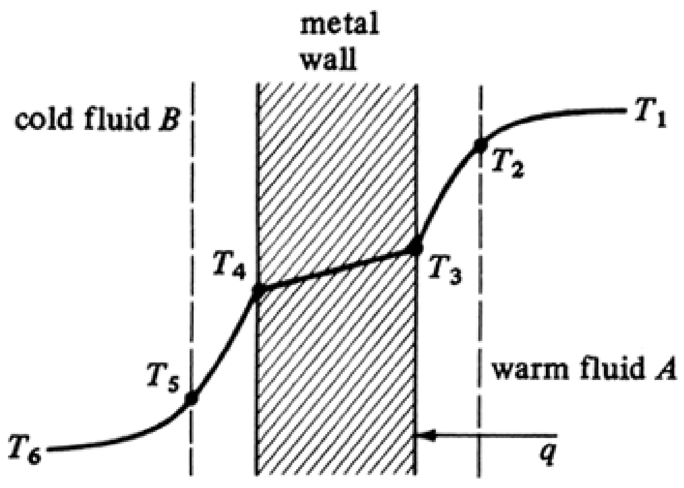


Figure 1. Temperature profile for heat transfer by convection

The convective coefficient for heat transfer through a fluid is given by,

$$q = hA\,(T - T_w) \qquad \text{(Eq.4)}$$

The type of fluid flow, whether laminar or turbulent, of the individual fluid has a great effect on the heat transfer coefficient h, which is often called a film coefficient, since most of the resistance to heat transfer is in a thin film close to the wall. The more turbulent flow, the greater the heat transfer coefficient.

The Prandtl number is the ratio of the shear component of diffusivity for momentum μ/ρ to the diffusivity for heat $k/\rho C_p$ and physically relates the relative thickness of the hydrodynamic layer and thermal boundary layer

$$N_{Pr} = \frac{\mu C_p}{k} \qquad \text{(Eq.5)}$$

The dimensionless Nusselt number, N_{Nu} is used to relate data for the heat transfer coefficient h to the thermal conductivity k of the fluid and a characteristic dimension D.

$$N_{Nu} = \frac{hD}{k} \qquad \text{(Eq.6)}$$

Certainly, the most important convective heat transfer process industrially is that of cooling or heating a fluid flowing inside a closed circular conduit or pipe. Different types of correlations for the convective coefficient are needed for laminar flow (N_{Re} below 2100), for fully turbulent flow (N_{Re} above 6000), and for the transition region (N_{Re} between 2100 and 6000).

For laminar flow of fluids inside horizontal tubes or pipes, the following equation of Sieder and Tate can be used for N_{Re}<2100:

$$(N_{Nu})_a = \frac{h_a D}{k} = 1.86\left(N_{Re} N_{Pr} \frac{D}{L}\right)^{1/3} \left(\frac{\mu_b}{\mu_w}\right)^{0.14} \qquad \text{(Eq.7)}$$

where D is pipe diameter in m, L is pipe length before mixing occurs in the pipe in m, μ_b is fluid viscosity at bulk average temperature in Pa.s. μ_W is the viscosity at the wall temperature, C_p is the heat capacity in J/kg.K, k thermal conductivity in W/m.K, h_a average heat transfer coefficient in W/m^2.K, and N_{Nu} dimensionless Nusselt number.

All the physical properties are evaluated at the bulk fluid temperature except μ_W.

The Reynolds number is

$$N_{Re} = \frac{Dv\rho}{\mu} \qquad \text{(Eq.8)}$$

When the Reynolds number is above 2100, the flow is turbulent. Since the rate of heat transfer is greater in the turbulent region, many industrial heat transfer processes are in the turbulent region.

For turbulent flow inside a pipe **or a tube** the following equation has been found to hold for a $N_{Re}>6000$, a N_{Pr} between 0.7 and 16000, and L/D>60

$$N_{Nu} = h_L D/k = 0.027\, N_{Re}^{0.8} N_{Pr}^{1/3} (\mu b/\mu w)^{0.14} \qquad \text{(Eq.9)}$$

where h_L is the heat-transfer coefficient based on the log mean driving force ΔT_{lm}. The fluid properties except μ_w are evaluated at the mean bulk temperature. If the bulk fluid temperature varies from the inlet to the outlet of the pipe, the mean of the inlet and outlet temperatures is used.

In many cases a fluid is flowing over completely ***immersed bodies at various geometries*** such as spheres, tubes, plates, and so on, and heat transfer is occurring between the fluid and solid only. Many of these shapes are of practical interest in process engineering. The sphere, cylinder, and flat plate are perhaps of greatest importance with heat transfer between these surfaces and a moving fluid frequently encountered. When heat transfer occurs during immersed flow, the flux is dependent on the geometry of the body, the position on the body, the proximity of other bodies, the flow rate and the fluid properties. The heat transfer coefficient varies over the body. In general, the average heat transfer coefficient on immersed bodies is given by

$$N_{Nu} = C N_{Re}^{m} N_{Pr}^{1/3} \qquad \text{(Eq.10)}$$

where C and m are constants that depend on the various configurations. The fluid properties are evaluated at the film temperature $T_f = (T_w+T_b)/2$, where T_w is the surface or wall temperature and T_b the average bulk fluid temperature. The velocity in the N_{Re} is the undisturbed free stream velocity v of the fluid approaching the object.

Table 1. Constants for use in Eq.10 for heat transfer to cylinders with axis perpendicular to flow ($N_{Pr} > 0.6$)

N_{Re}	*m*	*C*
1 – 4	0.330	0.989
4 – 40	0.385	0.911
40 – 4x10^3	0.466	0.683
4x10^3 – 4x10^4	0.618	0.193
4x 10^4 – 2.5x10^5	0.805	0.0266

When the fluid is following ***parallel to a flat plate*** and heat transfer is occuring between the whole plate of lenght L and the fluid, the N_{Nu} is a follows for a N_{Re} below $3x10^5$ in the laminar region and a N_{Pr}>0.7.

$$N_{Nu} = 0.664\, N_{Re,L}^{0.5}\, N_{Pr}^{1/3} \qquad \text{(Eq.11)}$$

where $N_{Re,L} = Lv\rho/\mu$.

For the completely turbulent region at a N_{Re} above $3x10^5$ and N_{Pr} >0.7

$$N_{Nu} = 0.0366 N_{Re,L}^{0.8}\, N_{Pr}^{1/3} \qquad \text{(Eq.12)}$$

Heat Transfer by Natural Convection

Natural convection heat transfer occurs when a solid surface is in contact with a gas or liquid which is at a different temperature from the surface. Density differences in the fluid arising from the heating process provide the buoyancy force required to move the fluid. Free or natural convection is observed as a result of the motion of the fluid. An example of heat transfer by natural convection is a hot radiator used for heating a room. Cold air encountering the radiator is heated and rises in natural convection because of buoyancy forces. The theoretical derivation of equations for natural convection heat transfer coefficients requires the solution of motion and energy equations.

For an isothermal vertical surface or plate with height L less than 1 m, the average natural convection heat transfer coefficient can be expressed by the following general equation:

$$N_{Nu} = hL/k = a\left(\frac{L^3 \rho^2 g \beta \Delta T}{\mu^2}\frac{C_p\,\mu}{k}\right)^m = a(N_{Gr}\,N_{Pr})^m \qquad \text{(Eq.13)}$$

where a and m are constants, N_{Gr} is the Grashof number, ρ is the density in kg/m^3, μ is the viscosity in kg/m.s, ΔT is the positive temperature difference between the wall and bulk fluid or vice versa in K, k is the thermal conductivity in W/m.K, C_p is the heat capacity in J/K.kg, β is the volumetric coefficient of expansion of the fluid in 1/K, and g is 9.80665 m/s^2.

All the physical properties are evaluated at the film temperature $T_f = (T_w+T_b)/2$. In general, for a vertical cylinder with length L m, the same equations can be used as for a vertical plate.

Table 2. Constants for vertical planes and cylinders.

$N_{Gr}N_{Pr}$	a	m
$<10^4$	1.36	1/5
10^4 - 10^9	0.59	1/4
$>10^9$	0.13	1/3

2.3. Extended Surface or Finned Exchangers

The use of fins or extended surfaces on the outside of a heat exchanger pipe wall to give relatively high heat transfer coefficients in the exchanger is quite common. An automobile radiator is such a device, where hot water passes inside through a bank of tubes and loses heat to the air. On the outside of the tubes, extended surfaces receive heat from the tube walls and transmit it to the air by forced convection.

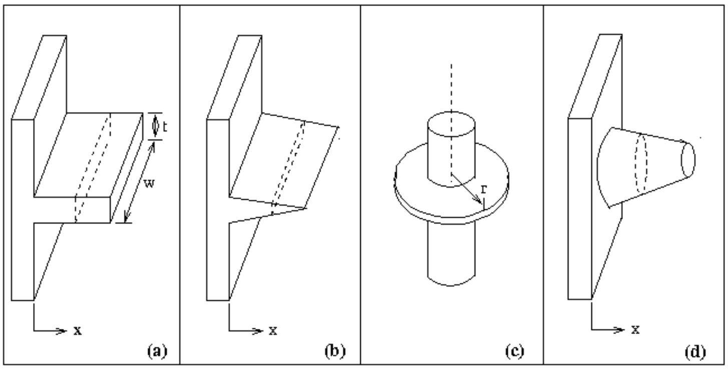


Figure 2. Various extended surfaces

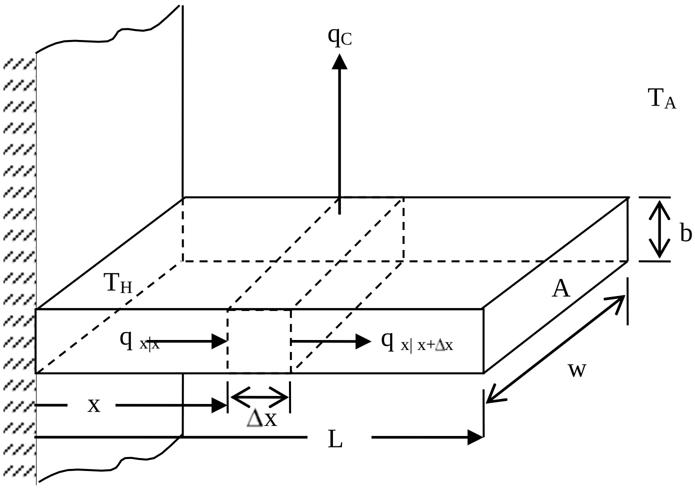


Figure 3. Heat balance for one – dimensional conduction and convection in a rectangular fin with constant cross sectional area.

We will consider a one – dimensional fin exposed to a surrounding fluid at temperature T_A as shown in Figure 3. At the base of the fin the temperature is T_H and at point x it is T. At steady state, the rate of heat conducted in to the element at x is $q_{x\,|\,x}$ and is equal to the rate of heat conducted out plus the rate of heat lost by convection.

$$q_{x\,|\,x} = q_{x\,|\,x+\Delta x} + q_c \qquad \text{(Eq.14)}$$

Substituting Fourier's equation for conduction and the convection equation,

$$-kA\frac{dT}{dx}\bigg|_{x} = -kA\frac{dT}{dx}\bigg|_{x+\Delta x} + h(P\Delta x)(T - T_A) \qquad \text{(Eq.15)}$$

where A is the cross - sectional area of the fin in m^2, P the perimeter of the fin in m, and (P Δx) the area for convection. Rearranging Eq.15, dividing by Δx, and letting Δx approach zero,

$$\frac{d^2T}{dx^2} - \frac{hP}{kA}(T - T_A) = 0 \qquad \text{(Eq.16)}$$

Letting $\theta = T - T_A$,

Eq.16 becomes

$$\frac{d^2\theta}{dx^2} - \frac{hP}{kA}\theta = 0 \qquad \text{(Eq.17)}$$

The first boundary condition is that $\theta_H = T_H - T_A$ at x = 0. For the second boundary condition needed to integrate Eq.17, several cases can be considered, depending upon the physical conditions at x=L. In the first case, the end of the fin is insulated and $d\theta/dx=0$ at x=L. In the second case the fin loses heat by convection from the tip surface so that $-k(dT/dx)_L = h(T_L - T_A)$. The solution using case 2 is quite involved and will not be considered here. Using the first case where the tip is insulated, integration of Eq.17 gives

$$\frac{\theta}{\theta_0} = \frac{\cosh[m(L - x)]}{\cosh(mL)} \qquad \text{(Eq.18)}$$

where $m = (hP/A)^{1/2}$.

The heat lost by the fin is expressed as

$$q = -kA\frac{dT}{dx}\bigg|_{x=0} \qquad \text{(Eq.19)}$$

Differentiating Eq.18 with respect to x and combining it with Eq.19,

$$q = (hPkA)^{1/2}(T_H - T_A)\tanh mL \qquad \text{(Eq.20)}$$

In the actual fin the temperature T in the fin decreases as the tip of the fin is approached. Hence, the rate of heat transfer per unit area decreases as the distance from the tube base is increased. To indicate this effectiveness of the fin to transfer heat, the fin efficiency is defined as the ratio of the actual heat transferred from the fin to the heat transferred if the entire fin were at the base temperature T_H.

$$\eta_f = \frac{(hPkA)^{1/2}(T_H - T_A)\tanh(mL)}{h(PL)(T_W - T_A)} = \frac{\tanh(mL)}{mL} \qquad \text{(Eq.21)}$$

where PL is the entire surface area of fin.

The expression for mL is

$$mL = \left(\frac{hP}{kA}\right)^{1/2} L = \left[\frac{h(2w+2b)}{k(wb)}\right]^{1/2} L \qquad \text{(Eq.22)}$$

For fins which are thin, 2b is small compared to 2w and

$$mL = \left(\frac{2h}{kb}\right)^{1/2} L \qquad \text{(Eq.23)}$$

Eq. 23 holds for a fin with an insulated tip.

3. EXPERIMENTAL SET UP

The apparatus consists of a vertical rectangular duct supported by a bench mounted stand. A flat plate, pinned or finned exchanger may be installed in the duct and secured by a quick- release catch on each side. Each exchanger incorporates an electric heating element with thermostatic protection against overheating. The temperature at the base of each exchanger is monitored by a thermistor sensor with connecting lead. The exchanger in use may be generated viewed through an acrylic window in the wall of the duct.

An upward flow of air may be generated in the duct with a variable speed fan mounted at the top. Air velocity in the duct, whether free or forced, is indicated on a portable anemometer held in a bracket on the duct wall. The anemometer sensor is inserted through the wall of the duct.

A thermistor probe permits measurement of the in-going and out-going air temperatures, together with surface temperatures of exchanger pins and fins. These temperatures are determined by inserting the probe through access holes in the duct wall.

An electric console incorporates a solid state power regulator with a digital read-out to control and indicate power supplied to the exchanger on test. The exchanger is connected to the console via the supply lead. A variable low voltage D.C. supply is provided for the fan via the supply lead. A digital read-out indicates the temperature using a thermistor probe connected to a flexible lead. Power is supplied to the equipment via a supply lead connected to a rear of the console.

4. EXPERIMENTAL PROCEDURE

In the heat transfer unit provided in Chemical Engineering Lab. I; the following will be practiced.

- Study of the relationship between power input and surface temperature in free convection.
- Study of the relationship between power input and surface temperature in forced convection.
- Study of the use of extended surfaces to improve heat transfer from the surface.
- Study of the temperature distribution along an extended surface.

4.1. Free Convection from a Vertical Fin Surface at Various Powers

- Place the finned heat exchanger into the test duct,
- Record the ambient air temperature (T_A),
- Set the heater power control to 20 Watts,
- Allow sufficient time to achieve steady state conditions before noting the heated plate temperature (T_H),
- Repeat this procedure at 40 and 60 Watts.

4.2. Forced Convection from a Vertical Fin Surface at Constant Power

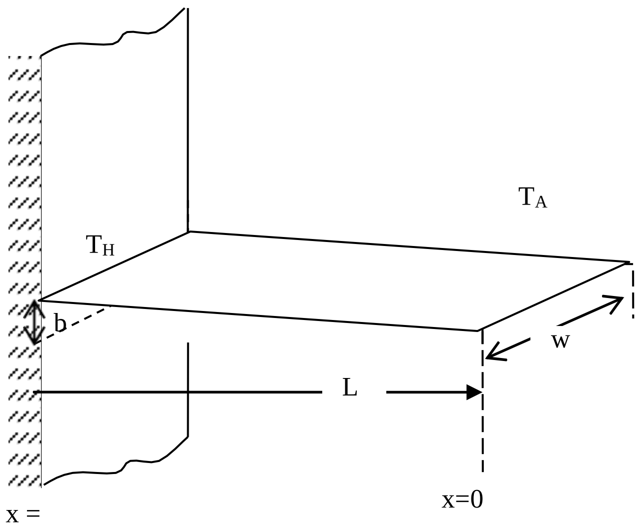
- Place the finned exchanger into the duct,
- Note the ambient air temperature (T_A),
- Set the heater power control 60 Watts,
- Allow sufficient time to achieve steady state conditions before noting the heated plate temperature (T_H),
- Insert the temperature probe into the duct through the hole nearest the heated plate, ensuring that the tip of the probe is in contact with the fin,
- Note this temperature (T_1),
- Record the fin temperatures (T_2 and T_3),

- Repeat this procedure by set the fan speed control to give a reading of 0.5 m/s and 1m/s, allow sufficient time to achieve steady state conditions.

Same procedure will be applied for pin and flat plate.

5. CALCULATIONS

5.1. Calculations for Finned Surface



w= 10 cm

b= 0.5 cm

L= 6.6 cm

Figure 4. Finned surface used in the experiment

As shown in Figure 4, T_H is the heated plate temperature and T_A is the ambient temperature. Thermal conductivity values for Aluminum which is fin material can be calculation by interpolation [1].

Table 3. Thermal conductivity values for Aluminum

T(K)	k (W/m.K)
300	273
400	240

5.1.1.a Calculation of h_0 values of fin <u>for free and forced convection</u>:

Equation 24 shows the heat transfer rate for whole finned surface at x=l.

$$Q|_{X=L} = \frac{kbw}{L^{1/2}}\theta_L\sqrt{B}\frac{I_1(2\sqrt{BL})}{I_0(2\sqrt{BL})} \qquad \text{(Eq.24)}$$

where $I_0(2\sqrt{BL})$ and $I_1(2\sqrt{BL})$ are the Bessel Functions and calculated by using Excel.

Here;

$\theta_L = T_H - T_A$ and

$$B = \frac{2h_0}{bk}\sqrt{(\frac{b}{2})^2 + L^2}$$

For 20, 40, 60 W, by assuming h_0 values, try to equal left hand side of the Equation 24 to the right side.

5.1.2. Calculation of heat transfer rate values at different points (Q_x) on the surface

$$Q_X = -kA_X \frac{dT}{dx} = -kA_X \frac{d\theta}{dx} \qquad \text{(Eq.25)}$$

where A_x is the cross-sectional area at x.

$$A_X = \frac{bw}{L}x \qquad \text{(Eq. 26)}$$

$$\frac{d\theta}{dx} = \frac{\theta_L\sqrt{B}}{\sqrt{x}}\frac{I_1(2\sqrt{Bx})}{I_0(2\sqrt{BL})} \qquad \text{(Eq. 27)}$$

By using Eq. 25, heat transfer rate values (Q_x) are calculated at different values.

5.1.3. Calculation of temperature values at different points (T_x) on the surface for both free and forced convection (Q=20 W)

First of all, heat transfer coefficient values are calculated for different fan speed values as done in section 5.1.1. Then, temperature values are calculated by using Equation 28 at different points on the fin surface.

$$\frac{\theta_X}{\theta_L} = \frac{T_X - T_A}{T_H - T_A} = \frac{I_0(2\sqrt{Bx})}{I_0(2\sqrt{BL})} \qquad \text{(Eq. 28)}$$

5.1.4. Calculation of fin efficiency

$$\eta_f = \frac{I_1(2mw)}{mL \times I_0(2mw)} \qquad \text{(Eq. 29)}$$

where $m = (2\ h_o/kb)^{1/2}$

By using Equation 29, fin efficiency values are calculated for 20, 40 and 60W.

5.2. Calculations for Flat Plate Surface

5.2.1. Calculation of heat transfer coefficient values (h_0) <u>for free convection</u>;

Heat transfer coefficient values (h_0) for free convection are calculated by using Equation 13 and Table 2.

5.2.2. Calculation of values heat transfer coefficient values (h_0) <u>for forced convection</u>;

By using Equation 11, heat transfer coefficient values (h_0) for forced convection are calculated at different fan speed, 0.5 m/s and 1 m/s.

5.3. Calculations for Pin Surface

5.3.1. Calculation of heat transfer coefficient values (h_0) for free convection;

Heat transfer coefficient values (h_0) for free convection are calculated by using Equation 13 and Table 2 as done for plate surface in section 5.2.2. **(D_{pin}=1.2 cm)**

5.3.2. Calculation of values heat transfer coefficient values (h_0) for forced convection;

By using Equation 10 and datum given in Table 1, heat transfer coefficient values (h_0) for forced convection are calculated at fan speed, 0.5 m/s and 1 m/s.

Suggestions for Discussion

1. How does the heat transfer coefficient vary with fan speed?
2. How does the heat transfer coefficient vary with position?
3. How does the fin efficiency vary with power?
4. How do the temperature measurements (including shape of the profile) compare with calculated values?

6. SYMBOLS

A	: Heat transfer area (m^2)
A_i, A_0	: Inside and outside heat transfer area (m^2)
C_p	: Specific heat at constant pressure
D_i, D_0	: Inside and outside diameter (m)
h	: Local individual heat transfer coefficient ($W/m^2.K$)
h_a	: Average heat transfer coefficient ($W/m^2.K$)
h_i, h_o	: Inside and outside heat transfer coefficient ($W/m^2.K$)
I_0, I_1	: Bessel functions
k	: Thermal conductivity (W/m.K)
L	: Lenght of the heat transfer surface (m)
N_{Gr}	: Grashof number (dimensionless)
N_{Nu}	: Nusselt number (dimensionless)

N_{Pr} : Prandtl number (dimensionless)

P : The perimeter of the fin (m)

Q : Heat flow rate (W)

T : Temperature (K)

T_A : Ambient air temperature (K)

T_c, T_h : Cold fluid and hot fluid temperature (K)

T_H : The heated plate temperature (K)

v : Linear velocity of fluid (m/s)

η_f : Fin efficiency

ΔT : Overall temperature difference (K)

ρ : Density (kg/m^3)

ν : Momentum diffusivity (m^2/s)

α : Thermal diffusivity (m^2/s)

μ : Viscosity (Pa.s)

β : Volumetric thermal expansion coefficient (K^{-1})

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8. DATA SHEET

Name Surname:

Group No:

Assistant: **Date:**

8.1. Experimental Results for Free Convection for Fin and Pin

Ambient air temperature T_A = ……………

Q (W)	Fin T_H (°C)	Pin T_H (°C)
20		
40		
60		

8.2. Experimental Results for Free and Forced Convection for Fin

Power input: Q= 20 W Ambient air temperature: T_A = ……………

x (cm)	*6.6*	*5.8*	*3.1*	*0.6*
V (m/s)	**T_H (°C)**	**T_3 (°C)**	**T_2 (°C)**	**T_1 (°C)**
0.0				
0.5				
1.0				

8.3. Experimental Results for Free and Forced Convection for Pin

Power input: Q= 20 W Ambient air temperature: T_A = ……………

x (cm)	*6.6*	*5.8*	*3.1*	*0.6*
V (m/s)	**T_H (°C)**	**T_3 (°C)**	**T_2 (°C)**	**T_1 (°C)**
0.0				
0.5				
1.0				

8.4. Experimental Results for Free and Forced Convection for Pin, Fin and Flat Plate

Power input: Q = 20 W Ambient air temperature: T_A = …………….

x (cm) = 6.6	*Pin*	*Fin*	*Flat Plate*
V (m/s)	**T_H (℃)**	**T_H (℃)**	**T_H (℃)**
0.0			
0.5			
1.0			