NEAR EAST UNIVERSITY

Faculty of Engineering

Department of Mechanical Engineering

HEAT TRANSFER FROM EXTENDED SURFACE

Graduation Project
ME - 400

Student: Eissa Anan

Supervisor: Assist. Prof. Dr. Guner Ozmen

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SUMMARY

The aim of this project is to increase the rate of heat transfer from extended surface area and to study the effect of fin design on the heat transfer efficiency of the heat exchanger.

In first chapter introduction of heat transfer is given. The classification of heat transfer and the definition of each mode of heat transfer are explained. The rate of heat transfer by each mode of heat transfer is discussed and some numerical example are given.

In second chapter the definition of heat transfer by conduction is summarized. The definition of the thermal conductivity and the effective of thermal conductivity on heat flow are explained. The steady heat transfer by conduction through the plane walls, cylinders, and spheres are calculated.

In third chapter the classification of convection heat transfer and some important dimensionless numbers are given. The amount of those numbers is calculated. The rate of heat transfer by forced convection over flat plate, cylinder, and sphere are discussed. The rate of heat transfer by natural convection over vertical plane is calculated. The combine of forced convection and natural convection are given.

In fourth and fifth chapters the classification fins are given. The description of long fin, pin fin, and radial fin are discussed. The rate of heat transfer from long fin, pin fin, and radial fin are calculated. The maximum rates of heat transfer from the fins surface are calculated. The fin efficiency is calculated and the fin effectiveness is calculated. Proper length of a fin is given. The proper length of the fin is calculated.

CHAPTER 1

INTRODUCTION

1.1 HISTORICAL BACKGROUND

Heat was always perceived to be something that produces in us a sensation of warmth, and one would think that the nature of heat is one of the first things understood by human. But it was only in the middle of the nineteenth century that we had a true physical understanding of the nature of heat. Then heat is defined as the form of energy that is transferred between two systems by effect of a temperature difference. The direction of heat transfer is from low temperature to high temperature.

1.2 HEAT TRANSFER MECHANISMS

Heat transfer can be transferred in three difference ways: conduction, convection and radiation. Brief descriptions of each mode of heat transfer are given on the following.

1.2.1 Heat Transfer by Conduction

Conduction is the transfer of the energy from the more energetic particles of substance to the next less energetic ones as result of difference temperature between the particles as in Figure (1-1). Conduction can occur in solid, liquid and gases. The rate of heat conduction through a medium depends on the geometry of the medium, its thickness, and the material of the medium, as well as temperature difference a cross the medium. The rate of heat conduction proportional to the temperature difference a cross the area surface heat transfers.

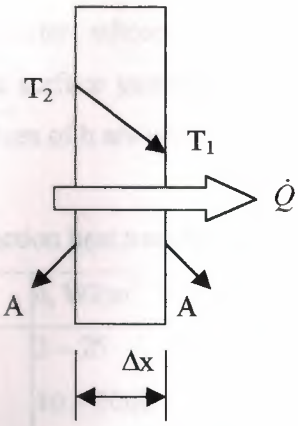


Figure (1-1) Heat conduction through a large plane.

$$\dot{Q}_{cond}=k\ A\ \Delta T/\Delta x$$

Where;

k: is the thermal conductivity of the material, (W/m. °C).

A: is the surface area of heat transfer, (m^2).

$\Delta T/\Delta x$: is the temperature gradient, (°C/m).

1.2.2 Heat Transfer by Convection

Convection is the mode of energy transfer between a solid surface and the near liquid or gas that is in motion, and it involves the combined effects of conduction and fluid motion. The faster the fluid motion, the greater the convection heat transfers. The rate of convection heat transfer is observed to be proportional to the temperature difference, and is conveniently expressed by Newton's law of cooling as;

$$\dot{Q}_{conv}=h.A.(T_s-T_\infty)$$

Where;

h: is the convection heat transfer coefficient in (w/m^2 °C)

A: is the surface area through which convection heat transfer takes place, (m^2)

Ts: is the surface temperature, (°C)

T_∞: is the temperature of the fluid sufficiently far from surface, (°C)

The convection heat transfer coefficient is not property of the fluid. It is an experimentally determined parameter whose value depends on the variables that influence convection, such as the surface geometry, the nature of the fluid motion and the bulk fluid velocity. Typical values of h are given in the Table (1.1).

Table 1-1 Typical values of convection heat transfer coefficient

Type of convection	h, W/(m^2. °C)
Free convection of gases	2 – 25
Free convection of liquids	10 – 1000
Forced convection of gases	25 – 250
Forced convection of liquids	50 - 20,000
Boiling and condensation	2500 - 100,000

1.2.3 Heat Transfer by Radiation

Radiation is the energy emitted by the matter in the form of electromagnetic waves as result of the changes in the electronic configurations of the atoms or molecules. Unlike conduction and convection, the transfer of energy by radiation does not require the presence of an intervening medium. In fact, energy transfer by radiation is fastest, and it suffers no attenuation in vacuum.

In heat transfer studies we are interested in thermal radiation, which is the form of radiation emitted by bodies because of their temperature. All bodies at a temperature above absolute zero emit thermal radiation. The maximum rate of radiation that can be emitted from a surface at an absolute temperature is given by Stefen-Boltzmann law as;

$$\dot{q}_{\max} = \sigma T_s$$

When a surface of emissivity and surface area at an absolute temperature is completely enclosed by a much larger (or black) surface at absolute temperature separated by a gas that does not intervene with radiation in Figure (1-2), the net rate of radiation heat transfer between these surfaces is given by;

$$\dot{Q}_{rad} = \varepsilon.\sigma.A.(T_s^{\ 4} - T_{surr}^{\ 4})$$

Where;

ε: is the emissivity of the surface.

σ: is the Stefen-Boltzmann constant, $(W/m^2. K^4)$.

T_s: is the absolute temperature of surface, (K^4).

T_{surr}: is the absolute temperature of surrounding, (K^4).

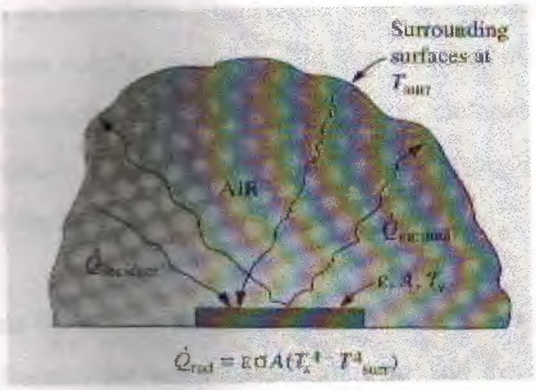


Figure (1-2) Radiation heat transfer between a surface and the surface surrounding it

1.3 SIMULTANEOUS HEAT TRANSFER MECHANISMS

We mentioned that there are three mechanisms of heat transfer, but not all three can exist simultaneous in a medium. For example, heat transfer is only by conduction in opaque solids but by conduction and radiation in semitransparent solids. Thus, a solid may involve conduction and radiation but not convection. However, a solid may involve heat transfer by convection and / or radiation on its surfaces exposed to a fluid or other surfaces. Heat transfer is by conduction and possibly by radiation in a still fluid and by convection and radiation in a flowing fluid. In the absence of radiation, heat transfer through a fluid is either by conduction or convection, depending on the presence of any bulk fluid motion. Convection can be viewed as combined conduction and fluid motion, and conduction can be viewed as a special case of convection in the absence of any fluid motion. Thus, when we deal with heat transfer through a fluid, we have either conduction or convection, but not both. Also, gases are practically transparent to radiation, except that some gases are known to absorb radiation strongly at certain wavelengths. Finally, heat transfer through a vacuum is by radiation only since conduction or convection requires the presence of a material medium

1.4 APPLICATION AREAS FOR HEAT TRANSFER

Heat transfer is commonly encountered in engineering systems and other aspects of life, and one does need to go very far to see some application areas of heat transfer. In fact, one does not need to go anywhere. The human body is constantly rejecting heat to its surroundings, and human comfort is closely tied to the rate of this heat rejection. We try to control this heat transfer rate by adjusting our clothing to the environmental conditions. Many ordinary household appliances are design, in whole or in part, by using the principles of the heat transfer.

Some example include the electric or gas range, the heating and air conditioning system, the refrigerator and freezer, the water heater, the iron, and even the computer, and the TV. Of course, energy efficient homes are designed on the basis of minimizing heat loss in winter and heat again in summer. Heat transfer plays a major role in the design of many other devices, such as car radiators solar collectors, various components of power plants, and even spacecraft. The optimal insulation thickness in the walls and roofs of the house, on hot water or steam pipes, or on water heaters is again determined on the basis of a heat transfer analysis with economic consideration.

EXAMPLE: Heat Transfer between Two Isothermal Plates

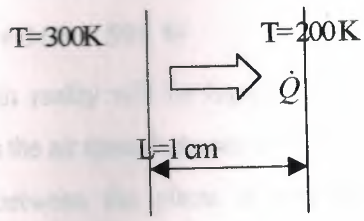
Consider steady heat transfer between two large parallel plates at constant temperature of T_1 = 300 K and T_2 = 200 K that are L =1 cm apart, as shown in figure below. Assuming the surfaces to be black (emissivity ε = 1), determine the rate of heat transfer between the plates per unit surface area assuming the gap between the plates is;

Filled with atmospheric air,

Evacuated,

Filled with urethane insulation, and,

Filled with super insulation that has an apparent thermal conductivity k=0.00002 W/m °C



Schematic for example

SOLUTION:

The total rate of heat transfer between two large parallel plates at specified temperatures is to be determined for four deferent cases

ASSUMPTION:

Steady operating condition exists, There are no natural convection currents in the air between the plates, the surfaces are black and thus

PROPERTIES:

The thermal conductivity at the average temperature of 250 K is;

The air k = 0.0223 W/m °C,

The urethane insulation k = 0.026 W/m °C,

The super insulation k = 0.00002 W/m °C.

The rates of conduction and radiation heat transfer between the plates through the air layer can be calculated as follows;

$$\dot{Q}_{cond} = k\,A\,(T_1 - T_2)/L$$

$$\dot{Q}_{cond} = (0.0223\ W/m°C)(1m^2)\,(300\text{-}200)°C/0.1m = 223W$$

and

$$\dot{Q}_{rad} = \varepsilon.\sigma.\,A.\,(T_1^4 - T_2^4)$$

$$\dot{Q}_{rad} = (1)\ (5.67\times10^{-8}W/m^2K^4)(1m^2)[(300K)^4 - (200\ K)^4] = 368\ W$$

Therefore,

$$\dot{Q}_{total} = \dot{Q}_{cond} + \dot{Q}_{rad}$$

$$\dot{Q}_{total} = 223 + 368 = 591\ W$$

The heat transfer rate in reality will be higher because of the natural convection currents that are likely to occur in the air space between the plates.

When the air space between the plates is evacuated, there will be no conduction or convection, and the only heat transfer between the plates will be by radiation. Therefore;

$$\dot{Q}_{total} = \dot{Q}_{rad} = 368\ W$$

An opaque solid material placed between two plates blocks direct radiation heat transfer between the plates. Also, the thermal conductivity of an insulating material accounts for the radiation heat transfer that may be occurring through the voids in the insulating material. The rate at heat transfer through the urethane insulation is;

$$\dot{Q}_{cond} = k\,A\,(T_1 - T_2)/L$$

$$\dot{Q}_{cond} = (0.026\ W/m\ °C)\,(1m^2)\,(300\text{-}200)°C = 260W$$

The layers of the super insulation prevent any direct radiation heat transfer between the plates. However, radiation heat transfer between the sheets of super insulation does occur, and the apparent thermal conductivity of the super-insulation accounts for this effect. Therefore;

$$\dot{Q}_{cond} = k\,A\,(T_1 - T_2)/L$$

$$\dot{Q}_{cond} = (0.00002\ W/m\ °C)\,(1m^2)\,(300\text{-}200)°C/0.1m = 0.2W$$

DISCUSSION:

This example demonstrates the effectiveness of super insulation and explains why they are the insulation of choice in critical applications despite their high cost.

CONCLUSION

In this chapter firstly the definition of heat transfer is given as energy transfer from higher energy to lower energy as result of temperature difference. Secondly the classification of heat transfer is given as conduction, convection, and radiation. The rate of heat transfer by conduction, convection, and radiation are calculated. Finally some application areas for heat transfer are given.

CHAPTER 2

HEAT TRANSFER BY CONDUCTION

2.1 INTRODUCTION

Heat conduction is defined as the transfer of thermal from the more energetic particles of a medium to the adjacent less energetic ones as a result of difference temperature between the particles. Temperature and heat transfer are closely related but they different nature. Heat transfer has direction also magnitude.

Heat transfer problem are often classified as being steady or transient. The term steady implies no change with time at any point within the medium, while transient implies variation with time or time dependence. In the special case of variation with time but not with position, the temperature of the medium changes uniformly with time. Such heat transfer systems are called lumped system.

A medium through which heat is conducted may involve the conversion of electrical, nuclear, or chemical energy into heat energy. In heat conduction analysis, such conversion processes are characterized as heat generation.

2.2 THERMAL CONDUCTIVITY

Thermal conductivity of material is a measure of ability of a material to conduct heat. Thermal conductivity of material can be defined as the rate of heat transfer through a unit thickness of material per unit area per unit temperature difference. The thermal conductivity of material is measure of how fast heat will flow in that material.

A large value of a thermal conductivity indicates that the material is good heat conduction and low value of a thermal conductivity indicates that the material is poor heat conduction or insulator. In Table 2.1 thermal conductivities of some materials.

Table (2-1) Thermal conductivities of some materials

Material	k, W/(m. °C)
Diamond	2300
Silver	429
Copper	401
Gold	317
Aluminum	237
Iron	80.2
Mercury	8.54
Glass	0.78
Brick	0.72
Water	0.613
Human skin	0.37

The thermal conductivities of materials vary over a wide range. The thermal conductivities of gases such as air vary by factor of 10^4 from those of pure metals as copper. The metals have the highest thermal conductivities, and gases and insulating materials the lowest.

The thermal conductivities of materials vary with temperature. The variation of thermal conductivity over certain temperature range is negligible for some materials. The thermal conductivities of certain solids exhibit dramatic increases at temperatures near absolute zero, when these solids become superconductors.

The temperature dependence of thermal conductivity causes considerable complexity in conduction analysis. Therefore, it is common practice to evaluate the thermal conductivity k at the average temperature and treat it as a constant in calculations.

Thermal insulations are materials used primarily to provide resistance to heat flow. Most insulation is heterogeneous materials made of materials of low thermal conductivity, and they involve air pockets. This is not surprising, since air has one of the lowest thermal conductivities, and it is freely available.

2.3 STEADY HEAT CONDUCTION IN PLANE WALLS

Consider steady heat conduction through the walls of a house in a winter day. We know that heat is continuously lost to the out doors through the wall. Recall that heat transfer in a certain direction is driven by the temperature gradient in that direction. Temperature measurements at several locations on the inner or outer wall surface will confirm that a wall surface is nearly isothermal. That is, the temperatures at the top and bottom of a wall surface as well as at the right or left ends are almost the same. But there will be considerable temperature difference between the inner and the outer surface of the wall.

The small thickness of the wall causes the temperature gradient in that direction to be large. Further, if the air temperatures in and outside the house remain constant then the heat transfer through the wall of the house can be modeled as steady and one-dimensional. The temperature of the wall in this case will depend on one direction only Noting that heat transfer is the only energy interaction involved in this case and there is no heat generation, the energy balance for the wall can be expressed as;

(rate of heat transfer into the wall) – (rate of heat transfer out of the wall) = (rate of change of the energy content of the wall). It can be expressed as;

$$\dot{Q}_{in} - \dot{Q}_{out} = \Delta E_{wall}/\Delta t$$

But $\Delta E_{wall}/\Delta t = 0$ for steady operation, since there is no change in the temperature of the wall with time at any point. Therefore, the rate of heat transfer into the wall must be equal to the rate of heat transfer out of it. In other word, the rate of heat transfer through the wall must be constant,

$$\dot{Q}_{cond,\ wall} = \text{constant.}$$

$$\dot{Q}_{cond,\ wall} = -k.\ A.\ \Delta T/\Delta x$$

2.4 HEAT CONDUCTION IN CYLINDERS AND SPHERES

Consider steady heat conduct through a hot water pipe. Heat is continuously lost to the outdoors through the wall of the pipe as shown in Figure (2-1). The wall of the pipe, whose thickness is rather small, separates two fluids at different and thus the temperature gradient in the radial direction will be relatively large. Further, if the fluid temperatures in an outside the pipe remains constant, then heat transfer through the pipe can be modeled as steady and one-dimensional. In steady operation, there is no change in the temperature of the pipe with time at any point. Therefore, the rate of heat transfer into the pipe must be equal to the rate of heat transfer out of it. In other words, the rate of heat transfer through the pipe must be constant, $\dot{Q}_{cond,\, cyl}$ =constant. Conduction heat transfer for cylinder can be written as;

$$\dot{Q}_{cond,\, cyl} = -k.\ A\ \Delta T/\Delta r$$

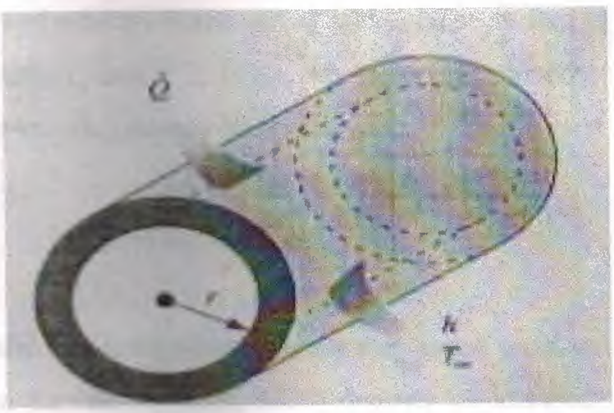


Figure (2-1) Heat transfer through a hot pipe to the air outside

CONCLUSION

In this chapter firstly the definition of heat conduction is given as the transfer of thermal from more energy to lower one of a medium as result of temperature difference. Secondly the classification of conduction heat transfer problem is given as steady or transient. Finally the definition of thermal conductivity is given as the ability of material to conduct heat. Thermal conductivities of some materials are given. The calculation of steady heat conduction in plane walls, cylinders, and spheres are given in detail.

CHAPTER 3

HEAT TRANSFER BY CONVECTION

Convection is the mechanism of heat transfer through a fluid in a presence of bulk fluid motion. Convection is classified as natural or forced convection, depending on how the fluid motion is initiated. In forced convection, the fluid is forced to flow over a surface or in a tube by external means such as a pump or a fan. In natural convection, any fluid motion is caused by natural means such as the boundary effect, which manifests itself as the rise of warmer fluid and the fall of cooler fluid.

3.1 THE CONVECTION TRANSFER PROBLEM

Consider the flow condition of Figure 3.1. A fluid of velocity V and temperature T_∞ flows over a surface of arbitrary shape and of area A_s. The surface is presumed to be at a uniform temperature, T_s and if $T_s \neq T_\infty$, we know that convection heat transfer will occur. The total heat transfer rate may be expressed as;

$$\dot{Q} = h A_s (T_s - T_\infty)$$

The total heat transfer rate is importance in any convection problem. This quantity may be determined from the rate equation, which depends on knowledge of the local and average convection coefficients. It is for this reason that determination of these coefficients is viewed as the problem of convection. However, the problem is not simple one, for addition to depending on numerous fluid properties such as density, viscosity, thermal conductivity, and specific heat, the coefficients depend on the surface geometry and flow conditions. This multiplicity of independent variables results from the fact that convection transfer is determined by the boundary layers that develops on the surface.

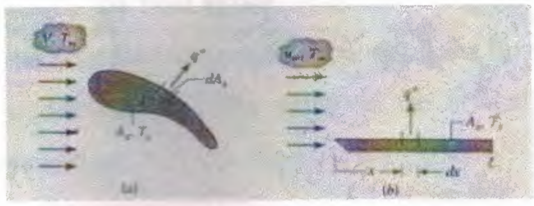


Figure (3-1) Convection transfer through
(a) arbitrary (b) flat plat

3.2 DISCUSSION THE DIMENSIONLESS NUMBERS PARAMETER

The Reynolds number, Nusselt number, and Groshof number are important number for the heat transfer calculation. Those numbers depend on the flow condition (laminar flow or turbulent flow) or/and on the geometries of the body. Those numbers are explained on the following.

The Reynolds Number

The dimensionless Reynolds number governs the flow regime in forced convection. The transition from laminar to turbulent flow depends on the surface geometry, surface roughness, the free-stream velocity, the surface temperature, and type of fluid, among other thing. The flow regime depends mainly on the ratio of the inertia forces to viscous forces in the fluid. This ratio is called the Reynolds number, and is expressed for external flow as;

$$\mathrm{Re} = \frac{v_\infty \delta}{\nu}$$

Where;

v_∞: is free-stream velocity, (m/s).

δ: is characteristic length of the geometry, m

ν: is cinematic viscosity of the fluid, m^2/s

At large Reynolds number, the inertia forces, which are proportional to the density and the velocity of the fluid, are large relative to the viscous forces, and thus the viscous forces cannot prevent the random and rapid fluctuations of the fluid.

At small Reynolds number, however, the viscous forces are large enough to overcome the inertia forces and to keep the fluid in line. Thus the flow is turbulent in the first case, and laminar in the second. The Reynolds number at which the flow becomes turbulent is called the critical Reynolds number. The value of the critical Reynolds number is different for different geometries.

The Grashof Number

Dimensionless number Grashof number governs the flow regime in natural convection. The Grashof number is the ratio of the buoyancy force to the viscous force acting on the fluid. It is expressed as;

$$Gr = \frac{g\beta(T_s - T_\infty)\delta^3}{\nu^2}$$

where;

g: is the gravitational acceleration, (m/s^2).

β: is the coefficient of volume expansion, (1/K).

T_s: is the temperature of the surface, (°C).

T_∞: is the temperature of the fluid sufficiently far from the surface, (°C).

δ: is the characteristic length of the geometry, (m).

ν: is the kinematic viscosity of the fluid, (m^2/s).

The Grashof number plays the role-played by the Reynolds number in forced convection in natural convection. The Grashof number is provides the main criteria in determining whether the fluid flow is laminar or turbulent in natural convection. For vertical plates, the criteria Grashof number is observed to be about 10^9. Therefore, the flow regime on a vertical plate becomes turbulent at Grashof numbers greater than 10^9.

The Nusselt Number

The Nusselt number is defined as a ratio of heat transfer coefficient and the characteristic length over the thermal conductivity. The Nusselt number depends on value of Reynolds and Grashof numbers and also depends on the geometry of the body. Therefore, the general equation of the Nusselt number is expressed as;

$$Nu = \frac{h\delta}{k}$$

3.3 FORCE CONVECTION OVER SURFACE

3.3.1 Flows Over Flat Plate

We will discuss the determination of the heat transfer rate to or from a flat plate by the fluid for both laminar and turbulent flow cases. Surfaces that are slightly contoured such as turbine blades can also be approximated as flat plates with reasonable accuracy.

The friction and the heat transfer coefficients for a flat plate can be determined theoretically by solving the conservation of mass, momentum, and energy equations approximately or numerically. They can also be determined experimentally, and expressed by empirical correlations. In either approach, it is found that the average Nusselt number can be expressed in terms of the Reynolds and Prandtl numbers in the form;

$$Nu = \frac{hL}{k} = C\,\mathrm{Re}_L^m\,\mathrm{Pr}^n$$

where;

C, m and n are constants

L: is the length of the plate in the flow direction, (m)

Pr: is the Prandtl number

Next we discuss the local and average friction and heat transfer coefficient over a flat plate for laminar, turbulent, and combined laminar and turbulent flow conditions.

Laminar Flow

The location friction coefficient and the Nusselt number at location x for laminar flow over a flat plate are given as;

$$C_{f,x} = \frac{0.664}{\mathrm{Re}_x^{1/2}}$$

and

$$Nu_x = \frac{h_x x}{k} = 0.332\,\mathrm{Re}_x^{1/2}\,\mathrm{Pr}^{1/3} \qquad (\mathrm{Pr} \geq 0.6)$$

The average friction coefficient and the Nusselt number over the entire plate are determined as;

$$C_f = \frac{1.328}{\mathrm{Re}_L^{1/2}}$$

and

$$Nu = \frac{hL}{k} = 0.664\,\mathrm{Re}_L^{1/2}\,\mathrm{Pr}^{1/3} \qquad (\mathrm{Pr} \geq 0.6)$$

The relations above give the average friction and heat transfer coefficients for the enter plate when the flow is laminar over the entire plate.

Turbulent Flow

The local friction coefficient and the Nusselt number at location x for turbulent flow over a flat plate are given as;

$$C_{f,x} = \frac{0.0592}{\mathrm{Re}_x^{1/5}} \qquad (5 \times 10^5 \leq \mathrm{Re}_x \leq 10^7)$$

and

$$Nu_x = \frac{h_x x}{k} = 0.0296\,\mathrm{Re}_x^{4/5}\,\mathrm{Pr}^{1/3} \quad (0.6 \leq \mathrm{Pr} \leq 60)$$

The average friction coefficient and the Nusselt number over the entire plate in turbulent flow are determined as;

$$C_f = \frac{0.074}{\mathrm{Re}_L^{1/5}}$$

and

$$Nu = \frac{hL}{k} = 0.037\,\mathrm{Re}_L^{4/5}\,\mathrm{Pr}^{1/3}$$

The two relations above give the average friction and heat transfer coefficient for the entire plate only when the flow is turbulent over the entire plate.

3.3.2 Flows Across Cylinders and Spheres

In this section, we consider flow over cylinders and spheres, which are frequently encountered in practice. For example, the tubes in a tube-and-shell heat exchanger involve both internal flow through the tubes, and external flow over the tubes. Below we consider external flow only.

The characteristic length for a circular cylinder or sphere is taken to be external diameter. The Reynolds number is defined as;

$$\mathrm{Re} = \frac{V_\infty D}{\nu}$$

The critical Reynolds number for flow cross a circular cylinder or sphere is $\mathrm{Re}_{cr} \approx 2 \times 10^5$. That is, the boundary layer remains laminar for $\mathrm{Re} < 2 \times 10^5$, and becomes turbulent for $\mathrm{Re} > 2 \times 10^5$.

The Heat Transfer Coefficient

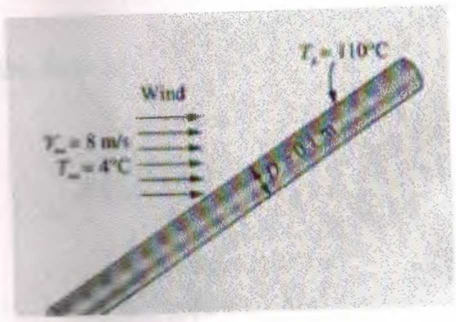
Flows across cylinders and spheres, in general, involve flow separation, which is difficult to handle analytically. Therefore, such flows must be studied experimentally. Several empirical correlations are developed for heat transfer coefficient.

Several relations available in the literature for the average Nusselt number for across-flow over a cylinder. The one proposed by Churchill and Bernstein as follow;

$$Nu_{cyl} = \frac{hD}{k} = 0.3 + \frac{0.62\,\mathrm{Re}^{1/2}\,\mathrm{Pr}^{1/3}}{\left[1 + (0.4/\mathrm{Pr})^{2/3}\right]^{1/4}}\left[1 + \left(\frac{\mathrm{Re}}{28{,}200}\right)^{5/8}\right]^{4/5}$$

EXAMPLE: Forced Convection Over Pipe

A long 10-cm-diameter steam pipe whose external surface temperature is 110°C passes through some open area that is not protected against the winds (Figure 3-7). Determine the rate of heat loss from the per unit of its length when the air is at 1-atm pressure and 4°C and the wind is blowing across the pipe at a velocity of 8 m/s.



Schematic for example

SOLUTION:

This is an external flow problem, since we are interested in the heat transfer from pipe to the air that is following outside the pipe.

The properties of air at 1-atm pressure and the film temperature;

$$T_f = \tfrac{1}{2}(T_\infty + T_s)$$

$$T_f = \frac{1}{2}(4+110) = 57°C = 330K$$

are

k = 0.0283 W/(m. °C)

v = 1.86 × 10^{-5} m^2/s

Pr = 0.708

The Reynolds number of the flow is;

$$\mathrm{Re} = \frac{V_\infty D}{v}$$

$$\mathrm{Re} = \frac{(8m/s)(0.1m)}{1.86\times10^{-5}\,m^2/s} = 43011$$

the flow is laminar because Re is less than 2×10^5

Then the Nusselt number in this case can be determined as;

$$Nu = \frac{hD}{k} = 0.3 + \frac{0.62\,\mathrm{Re}^{1/2}\,\mathrm{Pr}^{1/3}}{\left[1+(0.4/\mathrm{Pr})^{2/3}\right]^{1/4}}\left[1+\left(\frac{\mathrm{Re}}{28{,}800}\right)^{5/8}\right]^{4/5}$$

$$Nu = 0.3 + \frac{0.62(43{,}011)^{1/2}(0.708)^{1/3}}{\left[1+(0.4/0.708)^{2/3}\right]^{1/4}}\left[1+\left(\frac{43{,}011}{28{,}200}\right)^{5/8}\right]^{4/5} = 196.3$$

$$h = \frac{k}{D}Nu = \frac{0.0283W/m.^\circ C}{0.1m}(196.3) = 55.6W/m^2.^\circ C$$

Then the rate of heat transfer from the pipe per unit of its length becomes;

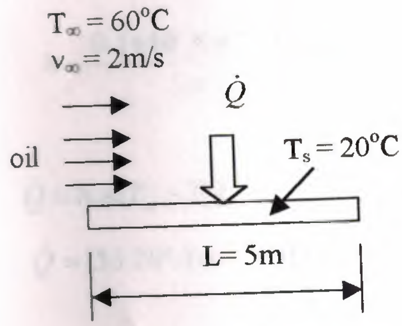
$$A = \rho L = \pi . D. L$$

$$A = \pi\,(0.1\text{m})\,(1\text{m}) = 0.314\ \text{m}^2$$

$$\dot{Q} = hA(T_s - T_\infty) = [55.6W/(m^2.C)](0.314m^2)(110-4)^\circ C = 1851W$$

EXAMPLE: Forced Convection Over Flat Plate.

Engine oil at 60°C flows over a 5-m-long flat plate whose temperature is 20°C with a velocity of 2m/s. determine the rate of heat transfer per unit width of the entire plate.



Schematic for Example

SOLUTION:

The properties of the engine oil at the average temperature of

$$T = \frac{T_s + T_\infty}{2}$$

$$T = \frac{20+60}{2} = 40^\circ C$$

are;

$$\rho = 876\text{kg/m}^3$$

$$k = 0.144\text{W/(m.}^\circ\text{C)}$$

$$\text{Pr} = 2870$$

$$\nu = 242 \times 10^{-6}\ \text{m}^2/\text{s}$$

The Reynolds number at the end of the plate becomes;

$$\text{Re} = \frac{\upsilon_\infty L}{\nu}$$

$$\text{Re} = \frac{(2m/s)(5m)}{242 \times 10^{-6} m^2/s} = 4.13 \times 10^4$$

The flow is laminar over the plate. The Nusselt number is determined using the laminar flow relations for a flat plate.

$$Nu = \frac{hL}{k} = 0.664\,\mathrm{Re}_L^{0.5}\,\mathrm{Pr}^{1/3}$$

$$Nu = 0.664.(4.13\times10^4)^{0.5}.2870^{1/3} = 1918$$

then;

$$h = \frac{k}{L}Nu$$

$$h = \frac{0.144W/(m.^{o}C)}{5m}(1918) = 55.2W/(m^2.^{o}C)$$

and

$$\dot{Q} = h.A(T_\infty - T_s)$$

$$\dot{Q} = [55.2W/(m^2.^{o}C)].(5\times1m^2).(60-20)^{o}C = 11040W$$

3.4 NATURAL CONVECTION OVER SURFACES

Natural convection heat transfer on a surface depends on the geometry of the surface as well as its orientation. It also depends on the variation of temperature on the surface and the thermo physical properties of the fluid involved.

The velocity and temperature profiles for natural convection over a vertical hot plate immersed in a quiescent fluid body are given in Figure (3-2). The fluid velocity is zero at the outer edge of the surface of the plate. The velocity increases with distance from the surface, reach a maximum, and gradually decreases to zero at a distance sufficiently far from the surface. The temperature of the fluid will equal to the plate temperature at the surface, and gradually decrease to the temperature of the surrounding fluid at a distance sufficiently far from the surface, as shown in Figure.

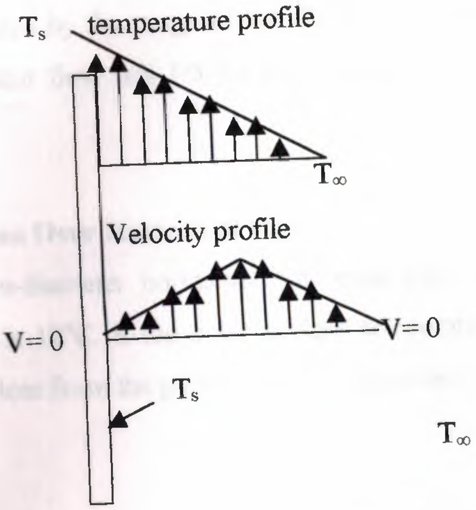


Figure (3-2) Natural convection over vertical surface

The heat transfer rate in natural convection from surface to surrounding is expressed as;

$$\dot{Q} = h A_s (T_s - T_\infty)$$

In order to solve natural convection heat transfer problem, the Grashof number or the Rayleigh number have to be considered on the problem. The Rayleigh number can be calculated as;

$$Ra = Gr.\ Pr$$

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

The average Nusselt number in natural convection is;

$$Nu = \frac{h\delta}{k} C(Gr.\mathrm{Pr})^n = CRa^n$$

Where C and n are constant and their values depend on the geometry of the surface and flow regime, which is characterized by the range of the Rayleigh number. The value of n is usually equal to ¼ for laminar flow and 1/3 for turbulent flow. The value of C is less than one.

EXAMPLE: Natural Convection Over Horizontal Pipe.

A 6-m-long section of an 8-cm-diameter horizontal hot water pipe passes through a large room whose temperature is 18°C. if the outer surface temperature of the pipe is 70°C, determine the rate of heat loss from the pipe by natural convection.

SOLUTION:

We assume the air pressure in the room to be 1atm. The properties of the air are to be evaluated at the average temperature T_v;

$$T_v = \frac{T_s + T_\infty}{2}$$

$$T_v = \frac{70+18}{2} = 44^\circ C = 317K$$

At this temperature, we read

$$k = 0.0273 (W/m.\,^\circ C)$$

$$v = 1.74 \times 10^{-5} m^2/s$$

$$Pr = 0.71$$

$$\beta = 1/T_v = 0.00315 K^{-1}$$

The characteristic length in this case is the outer diameter of the pipe, δ=D=0.08m. Then the Rayleigh number becomes;

$$Ra = \frac{g\beta(T_s - T_\infty)\delta^3}{v^2} \Pr$$

$$Ra = \frac{(9.8 m/s^2)(0.00315 K^{-1})[(70-18)K](0.08m)^3}{(1.74 \times 10^{-5} m^2/s)^2}(0.710) = 1.930 \times 10^6$$

Then the natural convection Nusselt number in this case can be calculated as;

$$Nu = \left\{ 0.6 + \frac{0.387(1.930 \times 10^6)^{1/6}}{\left[1 + (0.559/0.710)^{9/16}\right]^{8/27}} \right\}^2 = 17.2$$

then

$$h = \frac{k}{D} Nu = \frac{0.0273 W/(m.^\circ C)}{0.08m}(17.2) = 5.9 W/(m^2.^\circ C)$$

$$A = \pi.\ L.\ D = \pi.\ (6m).\ (0.08m) = 1.51m^2$$

and

$$\dot{Q} = hA(T_s - T_\infty)$$

$$\dot{Q} = \left[5.9 W/(m^2.^\circ C)\right](1.51 m^2)(70 - 18)^\circ C = 463$$

3.5 COMBINED NATURAL AND FORCED CONVECTION

The presence of a temperature gradient in a fluid in a gravity field always gives rise to the natural convection currents, and thus heat transfer by natural convection. Therefore, forced convection is always accompanied by natural convection.

Heat transfer coefficients encountered in forced convection are typically much higher than those encountered in natural convection because of the higher fluid velocities associated with forced convection. For given fluid, it is observed that parameter Gr/Re^2 represents the importance of natural convection relative to forced convection. Both natural and forced convection must be considered in heat transfer calculations when the Gr and Re^2 are of the same order of magnitude.

When determining heat transfer under combined natural and forced convection conditions, it is tempting to add the contributions of the natural and forced convection in assisting flow, and to subtract the in opposing flows. A review of experimental data suggests a correlation of the form;

$$Nu_{combined} = \left(Nu^n_{forced} \pm Nu^n_{natural}\right)^{1/n}$$

NEAR EAST UNIVERSITY

Faculty of Engineering

Department of Mechanical Engineering

HEAT TRANSFER FROM EXTENDED SURFACE

Graduation Project
ME - 400

Student: Eissa Anan

Supervisor: Assist. Prof. Dr. Guner Ozmen

Nicosia - 2002

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TABLE OF CONTENT

SUMMARY

The aim of this project is to increase the rate of heat transfer from extended surface area and to study the effect of fin design on the heat transfer efficiency of the heat exchanger.

In first chapter introduction of heat transfer is given. The classification of heat transfer and the definition of each mode of heat transfer are explained. The rate of heat transfer by each mode of heat transfer is discussed and some numerical example are given.

In second chapter the definition of heat transfer by conduction is summarized. The definition of the thermal conductivity and the effective of thermal conductivity on heat flow are explained. The steady heat transfer by conduction through the plane walls, cylinders, and spheres are calculated.

In third chapter the classification of convection heat transfer and some important dimensionless numbers are given. The amount of those numbers is calculated. The rate of heat transfer by forced convection over flat plate, cylinder, and sphere are discussed. The rate of heat transfer by natural convection over vertical plane is calculated. The combine of forced convection and natural convection are given.

In fourth and fifth chapters the classification fins are given. The description of long fin, pin fin, and radial fin are discussed. The rate of heat transfer from long fin, pin fin, and radial fin are calculated. The maximum rates of heat transfer from the fins surface are calculated. The fin efficiency is calculated and the fin effectiveness is calculated. Proper length of a fin is given. The proper length of the fin is calculated.

CHAPTER 1

INTRODUCTION

1.1 HISTORICAL BACKGROUND

Heat was always perceived to be something that produces in us a sensation of warmth, and one would think that the nature of heat is one of the first things understood by human. But it was only in the middle of the nineteenth century that we had a true physical understanding of the nature of heat. Then heat is defined as the form of energy that is transferred between two systems by effect of a temperature difference. The direction of heat transfer is from low temperature to high temperature.

1.2 HEAT TRANSFER MECHANISMS

Heat transfer can be transferred in three difference ways: conduction, convection and radiation. Brief descriptions of each mode of heat transfer are given on the following.

1.2.1 Heat Transfer by Conduction

Conduction is the transfer of the energy from the more energetic particles of substance to the next less energetic ones as result of difference temperature between the particles as in Figure (1-1). Conduction can occur in solid, liquid and gases. The rate of heat conduction through a medium depends on the geometry of the medium, its thickness, and the material of the medium, as well as temperature difference a cross the medium. The rate of heat conduction proportional to the temperature difference a cross the area surface heat transfers.

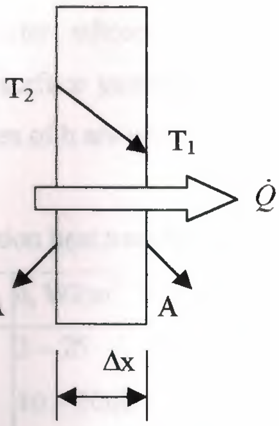


Figure (1-1) Heat conduction through a large plane.

$$\dot{Q}_{cond} = k\ A\ \Delta T/\Delta x$$

Where;

k: is the thermal conductivity of the material, (W/m. °C).

A: is the surface area of heat transfer, (m^2).

$\Delta T/\Delta x$: is the temperature gradient, (°C/m).

1.2.2 Heat Transfer by Convection

Convection is the mode of energy transfer between a solid surface and the near liquid or gas that is in motion, and it involves the combined effects of conduction and fluid motion. The faster the fluid motion, the greater the convection heat transfers. The rate of convection heat transfer is observed to be proportional to the temperature difference, and is conveniently expressed by Newton's law of cooling as;

$$\dot{Q}_{conv} = h.A.(T_s - T_\infty)$$

Where;

h: is the convection heat transfer coefficient in ($w/m^2\,°C$)

A: is the surface area through which convection heat transfer takes place, (m^2)

Ts: is the surface temperature, (°C)

T_∞: is the temperature of the fluid sufficiently far from surface, (°C)

The convection heat transfer coefficient is not property of the fluid. It is an experimentally determined parameter whose value depends on the variables that influence convection, such as the surface geometry, the nature of the fluid motion and the bulk fluid velocity. Typical values of h are given in the Table (1.1).

Table 1-1 Typical values of convection heat transfer coefficient

Type of convection	h, $W/(m^2 . {}^{o}C)$
Free convection of gases	2 – 25
Free convection of liquids	10 – 1000
Forced convection of gases	25 – 250
Forced convection of liquids	50 - 20,000
Boiling and condensation	2500 - 100,000

1.2.3 Heat Transfer by Radiation

Radiation is the energy emitted by the matter in the form of electromagnetic waves as result of the changes in the electronic configurations of the atoms or molecules. Unlike conduction and convection, the transfer of energy by radiation does not require the presence of an intervening medium. In fact, energy transfer by radiation is fastest, and it suffers no attenuation in vacuum.

In heat transfer studies we are interested in thermal radiation, which is the form of radiation emitted by bodies because of their temperature. All bodies at a temperature above absolute zero emit thermal radiation. The maximum rate of radiation that can be emitted from a surface at an absolute temperature is given by Stefen-Boltzmann law as;

$$\dot{q}_{\max} = \sigma T_s$$

When a surface of emissivity and surface area at an absolute temperature is completely enclosed by a much larger (or black) surface at absolute temperature separated by a gas that does not intervene with radiation in Figure (1-2), the net rate of radiation heat transfer between these surfaces is given by;

$$\dot{Q}_{rad} = \varepsilon.\sigma.A.(T_s^4 - T_{surr}^4)$$

Where;

ε: is the emissivity of the surface.

σ: is the Stefen-Boltzmann constant, ($W/m^2.K^4$).

T_s: is the absolute temperature of surface, (K^4).

T_{surr}: is the absolute temperature of surrounding, (K^4).

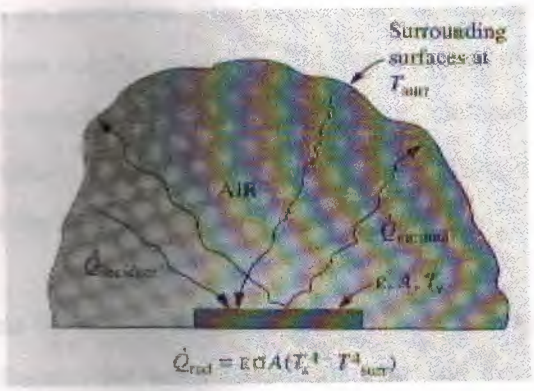


Figure (1-2) Radiation heat transfer between a surface and the surface surrounding it

1.3 SIMULTANEOUS HEAT TRANSFER MECHANISMS

We mentioned that there are three mechanisms of heat transfer, but not all three can exist simultaneous in a medium. For example, heat transfer is only by conduction in opaque solids but by conduction and radiation in semitransparent solids. Thus, a solid may involve conduction and radiation but not convection. However, a solid may involve heat transfer by convection and / or radiation on its surfaces exposed to a fluid or other surfaces. Heat transfer is by conduction and possibly by radiation in a still fluid and by convection and radiation in a flowing fluid. In the absence of radiation, heat transfer through a fluid is either by conduction or convection, depending on the presence of any bulk fluid motion. Convection can be viewed as combined conduction and fluid motion, and conduction can be viewed as a special case of convection in the absence of any fluid motion. Thus, when we deal with heat transfer through a fluid, we have either conduction or convection, but not both. Also, gases are practically transparent to radiation, except that some gases are known to absorb radiation strongly at certain wavelengths. Finally, heat transfer through a vacuum is by radiation only since conduction or convection requires the presence of a material medium

1.4 APPLICATION AREAS FOR HEAT TRANSFER

Heat transfer is commonly encountered in engineering systems and other aspects of life, and one does need to go very far to see some application areas of heat transfer. In fact, one does not need to go anywhere. The human body is constantly rejecting heat to its surroundings, and human comfort is closely tied to the rate of this heat rejection. We try to control this heat transfer rate by adjusting our clothing to the environmental conditions. Many ordinary household appliances are design, in whole or in part, by using the principles of the heat transfer.

Some example include the electric or gas range, the heating and air conditioning system, the refrigerator and freezer, the water heater, the iron, and even the computer, and the TV. Of course, energy efficient homes are designed on the basis of minimizing heat loss in winter and heat again in summer. Heat transfer plays a major role in the design of many other devices, such as car radiators solar collectors, various components of power plants, and even spacecraft. The optimal insulation thickness in the walls and roofs of the house, on hot water or steam pipes, or on water heaters is again determined on the basis of a heat transfer analysis with economic consideration.

EXAMPLE: Heat Transfer between Two Isothermal Plates

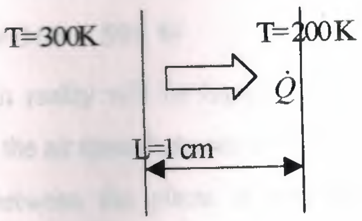
Consider steady heat transfer between two large parallel plates at constant temperature of T_1 = 300 K and T_2 = 200 K that are L =1 cm apart, as shown in figure below. Assuming the surfaces to be black (emissivity ε = 1), determine the rate of heat transfer between the plates per unit surface area assuming the gap between the plates is;

Filled with atmospheric air,

Evacuated,

Filled with urethane insulation, and,

Filled with super insulation that has an apparent thermal conductivity k=0.00002 W/m °C



Schematic for example

SOLUTION:

The total rate of heat transfer between two large parallel plates at specified temperatures is to be determined for four deferent cases

ASSUMPTION:

Steady operating condition exists, There are no natural convection currents in the air between the plates, the surfaces are black and thus

PROPERTIES:

The thermal conductivity at the average temperature of 250 K is;

The air k = 0.0223 W/m °C,

The urethane insulation k = 0.026 W/m °C,

The super insulation k = 0.00002 W/m °C.

The rates of conduction and radiation heat transfer between the plates through the air layer can be calculated as follows;

$$\dot{Q}_{cond} = k\,A\,(T_1 - T_2)/L$$

$$\dot{Q}_{cond} = (0.0223\ W/m°C)(1m^2)\ (300\text{-}200)°C/0.1m = 223W$$

and

$$\dot{Q}_{rad} = \varepsilon.\sigma.\ A.\ (T_1^4 - T_2^4)$$

$$\dot{Q}_{rad} = (1)\ (5.67x10^{-8} W/m^2K^4)(1m^2)[(300K)^4 - (200\ K)^4] = 368\ W$$

Therefore,

$$\dot{Q}_{total} = \dot{Q}_{cond} + \dot{Q}_{rad}$$

$$\dot{Q}_{total} = 223 + 368 = 591\ W$$

The heat transfer rate in reality will be higher because of the natural convection currents that are likely to occur in the air space between the plates.

When the air space between the plates is evacuated, there will be no conduction or convection, and the only heat transfer between the plates will be by radiation. Therefore;

$$\dot{Q}_{total} = \dot{Q}_{rad} = 368\ W$$

An opaque solid material placed between two plates blocks direct radiation heat transfer between the plates. Also, the thermal conductivity of an insulating material accounts for the radiation heat transfer that may be occurring through the voids in the insulating material. The rate at heat transfer through the urethane insulation is;

$$\dot{Q}_{cond} = k\,A\,(T_1 - T_2)/L$$

$$\dot{Q}_{cond} = (0.026\ W/m\ °C)\ (1m^2)\ (300\text{-}200)°C = 260W$$

The layers of the super insulation prevent any direct radiation heat transfer between the plates. However, radiation heat transfer between the sheets of super insulation does occur, and the apparent thermal conductivity of the super-insulation accounts for this effect. Therefore;

$$\dot{Q}_{cond} = k\,A\,(T_1 - T_2)/L$$

$$\dot{Q}_{cond} = (0.00002\ W/m\ °C)\ (1m^2)\ (300\text{-}200)°C/0.1m = 0.2W$$

DISCUSSION:

This example demonstrates the effectiveness of super insulation and explains why they are the insulation of choice in critical applications despite their high cost.

CONCLUSION

In this chapter firstly the definition of heat transfer is given as energy transfer from higher energy to lower energy as result of temperature difference. Secondly the classification of heat transfer is given as conduction, convection, and radiation. The rate of heat transfer by conduction, convection, and radiation are calculated. Finally some application areas for heat transfer are given.

CHAPTER 2

HEAT TRANSFER BY CONDUCTION

2.1 INTRODUCTION

Heat conduction is defined as the transfer of thermal from the more energetic particles of a medium to the adjacent less energetic ones as a result of difference temperature between the particles. Temperature and heat transfer are closely related but they different nature. Heat transfer has direction also magnitude.

Heat transfer problem are often classified as being steady or transient. The term steady implies no change with time at any point within the medium, while transient implies variation with time or time dependence. In the special case of variation with time but not with position, the temperature of the medium changes uniformly with time. Such heat transfer systems are called lumped system.

A medium through which heat is conducted may involve the conversion of electrical, nuclear, or chemical energy into heat energy. In heat conduction analysis, such conversion processes are characterized as heat generation.

2.2 THERMAL CONDUCTIVITY

Thermal conductivity of material is a measure of ability of a material to conduct heat. Thermal conductivity of material can be defined as the rate of heat transfer through a unit thickness of material per unit area per unit temperature difference. The thermal conductivity of material is measure of how fast heat will flow in that material.

A large value of a thermal conductivity indicates that the material is good heat conduction and low value of a thermal conductivity indicates that the material is poor heat conduction or insulator. In Table 2.1 thermal conductivities of some materials.

Table (2-1) Thermal conductivities of some materials

Material	k, W/(m. °C)
Diamond	2300
Silver	429
Copper	401
Gold	317
Aluminum	237
Iron	80.2
Mercury	8.54
Glass	0.78
Brick	0.72
Water	0.613
Human skin	0.37

The thermal conductivities of materials vary over a wide range. The thermal conductivities of gases such as air vary by factor of 10^4 from those of pure metals as copper. The metals have the highest thermal conductivities, and gases and insulating materials the lowest.

The thermal conductivities of materials vary with temperature. The variation of thermal conductivity over certain temperature range is negligible for some materials. The thermal conductivities of certain solids exhibit dramatic increases at temperatures near absolute zero, when these solids become superconductors.

The temperature dependence of thermal conductivity causes considerable complexity in conduction analysis. Therefore, it is common practice to evaluate the thermal conductivity k at the average temperature and treat it as a constant in calculations.

Thermal insulations are materials used primarily to provide resistance to heat flow. Most insulation is heterogeneous materials made of materials of low thermal conductivity, and they involve air pockets. This is not surprising, since air has one of the lowest thermal conductivities, and it is freely available.

2.3 STEADY HEAT CONDUCTION IN PLANE WALLS

Consider steady heat conduction through the walls of a house in a winter day. We know that heat is continuously lost to the out doors through the wall. Recall that heat transfer in a certain direction is driven by the temperature gradient in that direction. Temperature measurements at several locations on the inner or outer wall surface will confirm that a wall surface is nearly isothermal. That is, the temperatures at the top and bottom of a wall surface as well as at the right or left ends are almost the same. But there will be considerable temperature difference between the inner and the outer surface of the wall.

The small thickness of the wall causes the temperature gradient in that direction to be large. Further, if the air temperatures in and outside the house remain constant then the heat transfer through the wall of the house can be modeled as steady and one-dimensional. The temperature of the wall in this case will depend on one direction only Noting that heat transfer is the only energy interaction involved in this case and there is no heat generation, the energy balance for the wall can be expressed as;

(rate of heat transfer into the wall) – (rate of heat transfer out of the wall) = (rate of change of the energy content of the wall). It can be expressed as;

$$\dot{Q}_{in} - \dot{Q}_{out} = \Delta E_{wall}/\Delta t$$

But $\Delta E_{wall}/\Delta t = 0$ for steady operation, since there is no change in the temperature of the wall with time at any point. Therefore, the rate of heat transfer into the wall must be equal to the rate of heat transfer out of it. In other word, the rate of heat transfer through the wall must be constant,

$$\dot{Q}_{cond,\ wall} = \text{constant}.$$

$$\dot{Q}_{cond,\ wall} = -k.\ A.\ \Delta T/\Delta x$$

2.4 HEAT CONDUCTION IN CYLINDERS AND SPHERES

Consider steady heat conduct through a hot water pipe. Heat is continuously lost to the outdoors through the wall of the pipe as shown in Figure (2-1). The wall of the pipe, whose thickness is rather small, separates two fluids at different and thus the temperature gradient in the radial direction will be relatively large. Further, if the fluid temperatures in an outside the pipe remains constant, then heat transfer through the pipe can be modeled as steady and one-dimensional. In steady operation, there is no change in the temperature of the pipe with time at any point. Therefore, the rate of heat transfer into the pipe must be equal to the rate of heat transfer out of it. In other words, the rate of heat transfer through the pipe must be constant, $\dot{Q}_{cond, cyl}$ =constant. Conduction heat transfer for cylinder can be written as;

$$\dot{Q}_{cond, cyl} = -k.\ A\ \Delta T/\Delta r$$

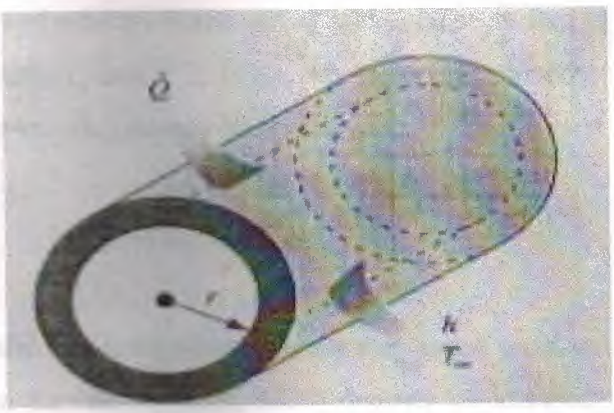


Figure (2-1) Heat transfer through a hot pipe to the air outside

CONCLUSION

In this chapter firstly the definition of heat conduction is given as the transfer of thermal from more energy to lower one of a medium as result of temperature difference. Secondly the classification of conduction heat transfer problem is given as steady or transient. Finally the definition of thermal conductivity is given as the ability of material to conduct heat. Thermal conductivities of some materials are given. The calculation of steady heat conduction in plane walls, cylinders, and spheres are given in detail.

CHAPTER 3

HEAT TRANSFER BY CONVECTION

Convection is the mechanism of heat transfer through a fluid in a presence of bulk fluid motion. Convection is classified as natural or forced convection, depending on how the fluid motion is initiated. In forced convection, the fluid is forced to flow over a surface or in a tube by external means such as a pump or a fan. In natural convection, any fluid motion is caused by natural means such as the boundary effect, which manifests itself as the rise of warmer fluid and the fall of cooler fluid.

3.1 THE CONVECTION TRANSFER PROBLEM

Consider the flow condition of Figure 3.1. A fluid of velocity V and temperature T_∞ flows over a surface of arbitrary shape and of area A_s. The surface is presumed to be at a uniform temperature, T_s and if $T_s \neq T_\infty$, we know that convection heat transfer will occur. The total heat transfer rate may be expressed as;

$$\dot{Q} = h A_s (T_s - T_\infty)$$

The total heat transfer rate is importance in any convection problem. This quantity may be determined from the rate equation, which depends on knowledge of the local and average convection coefficients. It is for this reason that determination of these coefficients is viewed as the problem of convection. However, the problem is not simple one, for addition to depending on numerous fluid properties such as density, viscosity, thermal conductivity, and specific heat, the coefficients depend on the surface geometry and flow conditions. This multiplicity of independent variables results from the fact that convection transfer is determined by the boundary layers that develops on the surface.

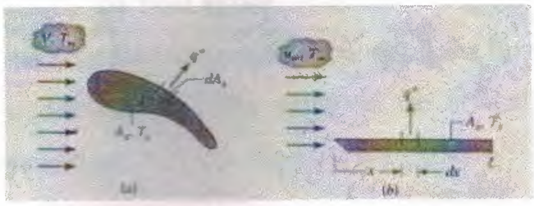


Figure (3-1) Convection transfer through
(a) arbitrary (b) flat plat

3.2 DISCUSSION THE DIMENSIONLESS NUMBERS PARAMETER

The Reynolds number, Nusselt number, and Groshof number are important number for the heat transfer calculation. Those numbers depend on the flow condition (laminar flow or turbulent flow) or/and on the geometries of the body. Those numbers are explained on the following.

The Reynolds Number

The dimensionless Reynolds number governs the flow regime in forced convection. The transition from laminar to turbulent flow depends on the surface geometry, surface roughness, the free-stream velocity, the surface temperature, and type of fluid, among other thing. The flow regime depends mainly on the ratio of the inertia forces to viscous forces in the fluid. This ratio is called the Reynolds number, and is expressed for external flow as;

$$\mathrm{Re} = \frac{V_\infty \delta}{\nu}$$

Where;

v_∞: is free-stream velocity, (m/s).

δ: is characteristic length of the geometry, m

ν: is cinematic viscosity of the fluid, m^2/s

At large Reynolds number, the inertia forces, which are proportional to the density and the velocity of the fluid, are large relative to the viscous forces, and thus the viscous forces cannot prevent the random and rapid fluctuations of the fluid.

At small Reynolds number, however, the viscous forces are large enough to overcome the inertia forces and to keep the fluid in line. Thus the flow is turbulent in the first case, and laminar in the second. The Reynolds number at which the flow becomes turbulent is called the critical Reynolds number. The value of the critical Reynolds number is different for different geometries.

The Grashof Number

Dimensionless number Grashof number governs the flow regime in natural convection. The Grashof number is the ratio of the buoyancy force to the viscous force acting on the fluid. It is expressed as;

$$Gr = \frac{g\beta(T_s - T_\infty)\delta^3}{\nu^2}$$

where;

g: is the gravitational acceleration, (m/s^2).

β: is the coefficient of volume expansion, (1/K).

T_s: is the temperature of the surface, (°C).

T_∞: is the temperature of the fluid sufficiently far from the surface, (°C).

δ: is the characteristic length of the geometry, (m).

ν: is the kinematic viscosity of the fluid, (m^2/s).

The Grashof number plays the role-played by the Reynolds number in forced convection in natural convection. The Grashof number is provides the main criteria in determining whether the fluid flow is laminar or turbulent in natural convection. For vertical plates, the criteria Grashof number is observed to be about 10^9. Therefore, the flow regime on a vertical plate becomes turbulent at Grashof numbers greater than 10^9.

The Nusselt Number

The Nusselt number is defined as a ratio of heat transfer coefficient and the characteristic length over the thermal conductivity. The Nusselt number depends on value of Reynolds and Grashof numbers and also depends on the geometry of the body. Therefore, the general equation of the Nusselt number is expressed as;

$$Nu = \frac{h\delta}{k}$$

3.3 FORCE CONVECTION OVER SURFACE

3.3.1 Flows Over Flat Plate

We will discuss the determination of the heat transfer rate to or from a flat plate by the fluid for both laminar and turbulent flow cases. Surfaces that are slightly contoured such as turbine blades can also be approximated as flat plates with reasonable accuracy.

The friction and the heat transfer coefficients for a flat plate can be determined theoretically by solving the conservation of mass, momentum, and energy equations approximately or numerically. They can also be determined experimentally, and expressed by empirical correlations. In either approach, it is found that the average Nusselt number can be expressed in terms of the Reynolds and Prandtl numbers in the form;

$$Nu = \frac{hL}{k} = C\,\mathrm{Re}_L^m\,\mathrm{Pr}^n$$

where;

C, m and n are constants

L: is the length of the plate in the flow direction, (m)

Pr: is the Prandtl number

Next we discuss the local and average friction and heat transfer coefficient over a flat plate for laminar, turbulent, and combined laminar and turbulent flow conditions.

Laminar Flow

The location friction coefficient and the Nusselt number at location x for laminar flow over a flat plate are given as;

$$C_{f,x} = \frac{0.664}{\mathrm{Re}_x^{1/2}}$$

and

$$Nu_x = \frac{h_x x}{k} = 0.332\,\mathrm{Re}_x^{1/2}\,\mathrm{Pr}^{1/3} \qquad (\mathrm{Pr} \geq 0.6)$$

The average friction coefficient and the Nusselt number over the entire plate are determined as;

$$C_f = \frac{1.328}{\mathrm{Re}_L^{1/2}}$$

and

$$Nu = \frac{hL}{k} = 0.664\,\mathrm{Re}_L^{1/2}\,\mathrm{Pr}^{1/3} \qquad (\mathrm{Pr} \geq 0.6)$$

The relations above give the average friction and heat transfer coefficients for the enter plate when the flow is laminar over the entire plate.

Turbulent Flow

The local friction coefficient and the Nusselt number at location x for turbulent flow over a flat plate are given as;

$$C_{f,x} = \frac{0.0592}{\mathrm{Re}_x^{1/5}} \qquad (5 \times 10^5 \leq \mathrm{Re}_x \leq 10^7)$$

and

$$Nu_x = \frac{h_x x}{k} = 0.0296\,\mathrm{Re}_x^{4/5}\,\mathrm{Pr}^{1/3} \quad (0.6 \leq \mathrm{Pr} \leq 60)$$

The average friction coefficient and the Nusselt number over the entire plate in turbulent flow are determined as;

$$C_f = \frac{0.074}{\mathrm{Re}_L^{1/5}}$$

and

$$Nu = \frac{hL}{k} = 0.037\,\mathrm{Re}_L^{4/5}\,\mathrm{Pr}^{1/3}$$

The two relations above give the average friction and heat transfer coefficient for the entire plate only when the flow is turbulent over the entire plate.

3.3.2 Flows Across Cylinders and Spheres

In this section, we consider flow over cylinders and spheres, which are frequently encountered in practice. For example, the tubes in a tube-and-shell heat exchanger involve both internal flow through the tubes, and external flow over the tubes. Below we consider external flow only.

The characteristic length for a circular cylinder or sphere is taken to be external diameter. The Reynolds number is defined as;

$$\mathrm{Re} = \frac{V_\infty D}{\nu}$$

The critical Reynolds number for flow cross a circular cylinder or sphere is $\mathrm{Re}_{cr} \approx 2 \times 10^5$. That is, the boundary layer remains laminar for $\mathrm{Re} < 2 \times 10^5$, and becomes turbulent for $\mathrm{Re} > 2 \times 10^5$.

The Heat Transfer Coefficient

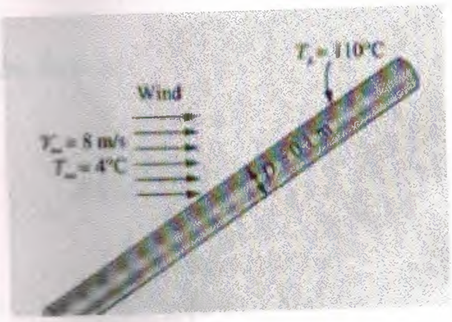
Flows across cylinders and spheres, in general, involve flow separation, which is difficult to handle analytically. Therefore, such flows must be studied experimentally. Several empirical correlations are developed for heat transfer coefficient.

Several relations available in the literature for the average Nusselt number for across-flow over a cylinder. The one proposed by Churchill and Bernstein as follow;

$$Nu_{cyl} = \frac{hD}{k} = 0.3 + \frac{0.62\,\mathrm{Re}^{1/2}\,\mathrm{Pr}^{1/3}}{\left[1+(0.4/\mathrm{Pr})^{2/3}\right]^{1/4}}\left[1+\left(\frac{\mathrm{Re}}{28{,}200}\right)^{5/8}\right]^{4/5}$$

EXAMPLE: Forced Convection Over Pipe

A long 10-cm-diameter steam pipe whose external surface temperature is 110°C passes through some open area that is not protected against the winds (Figure 3-7). Determine the rate of heat loss from the per unit of its length when the air is at 1-atm pressure and 4°C and the wind is blowing across the pipe at a velocity of 8 m/s.



Schematic for example

SOLUTION:

This is an external flow problem, since we are interested in the heat transfer from pipe to the air that is following outside the pipe.

The properties of air at 1-atm pressure and the film temperature;

$$T_f = \tfrac{1}{2}(T_\infty + T_s)$$

$$T_f = \frac{1}{2}(4+110) = 57°C = 330K$$

are

$$k = 0.0283 \text{ W/(m. °C)}$$

$$\nu = 1.86 \times 10^{-5} \text{ m}^2\text{/s}$$

$$\text{Pr} = 0.708$$

The Reynolds number of the flow is;

$$\text{Re} = \frac{V_\infty D}{\nu}$$

$$\text{Re} = \frac{(8m/s)(0.1m)}{1.86\times10^{-5} m^2/s} = 43011$$

the flow is laminar because Re is less than 2×10^5

Then the Nusselt number in this case can be determined as;

$$Nu = \frac{hD}{k} = 0.3 + \frac{0.62\,\text{Re}^{1/2}\,\text{Pr}^{1/3}}{\left[1+(0.4/\text{Pr})^{2/3}\right]^{1/4}}\left[1+\left(\frac{\text{Re}}{28,800}\right)^{5/8}\right]^{4/5}$$

$$Nu = 0.3 + \frac{0.62(43,011)^{1/2}(0.708)^{1/3}}{\left[1+(0.4/0.708)^{2/3}\right]^{1/4}}\left[1+\left(\frac{43,011}{28,200}\right)^{5/8}\right]^{4/5} = 196.3$$

$$h = \frac{k}{D}Nu = \frac{0.0283W/m.^\circ C}{0.1m}(196.3) = 55.6W/m^2.^\circ C$$

Then the rate of heat transfer from the pipe per unit of its length becomes;

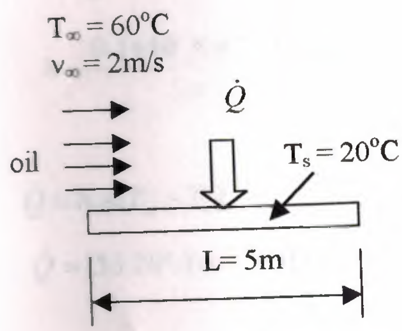
$$A = \rho L = \pi . D . L$$

$$A = \pi (0.1m)(1m) = 0.314 \ m^2$$

$$\dot{Q} = hA(T_s - T_\infty) = [55.6W/(m^2.C)](0.314m^2)(110-4)^\circ C = 1851W$$

EXAMPLE: Forced Convection Over Flat Plate.

Engine oil at 60°C flows over a 5-m-long flat plate whose temperature is 20°C with a velocity of 2m/s. determine the rate of heat transfer per unit width of the entire plate.



Schematic for Example

SOLUTION:

The properties of the engine oil at the average temperature of

$$T = \frac{T_s + T_\infty}{2}$$

$$T = \frac{20+60}{2} = 40^\circ C$$

are;

$$\rho = 876 kg/m^3$$

$$k = 0.144 W/(m.^\circ C)$$

$$Pr = 2870$$

$$\nu = 242 \times 10^{-6} \ m^2/s$$

The Reynolds number at the end of the plate becomes;

$$Re = \frac{\upsilon_\infty L}{\nu}$$

$$Re = \frac{(2m/s)(5m)}{242 \times 10^{-6} m^2/s} = 4.13 \times 10^4$$

The flow is laminar over the plate. The Nusselt number is determined using the laminar flow relations for a flat plate.

$$Nu = \frac{hL}{k} = 0.664\,\mathrm{Re}_L^{0.5}\,\mathrm{Pr}^{1/3}$$

$$Nu = 0.664.(4.13\times10^4)^{0.5}.2870^{1/3} = 1918$$

then;

$$h = \frac{k}{L}Nu$$

$$h = \frac{0.144W/(m.^{o}C)}{5m}(1918) = 55.2W/(m^2.^{o}C)$$

and

$$\dot{Q} = h.A(T_\infty - T_s)$$

$$\dot{Q} = [55.2W/(m^2.^{o}C)].(5\times1m^2).(60-20)^{o}C = 11040W$$

3.4 NATURAL CONVECTION OVER SURFACES

Natural convection heat transfer on a surface depends on the geometry of the surface as well as its orientation. It also depends on the variation of temperature on the surface and the thermo physical properties of the fluid involved.

The velocity and temperature profiles for natural convection over a vertical hot plate immersed in a quiescent fluid body are given in Figure (3-2). The fluid velocity is zero at the outer edge of the surface of the plate. The velocity increases with distance from the surface, reach a maximum, and gradually decreases to zero at a distance sufficiently far from the surface. The temperature of the fluid will equal to the plate temperature at the surface, and gradually decrease to the temperature of the surrounding fluid at a distance sufficiently far from the surface, as shown in Figure.

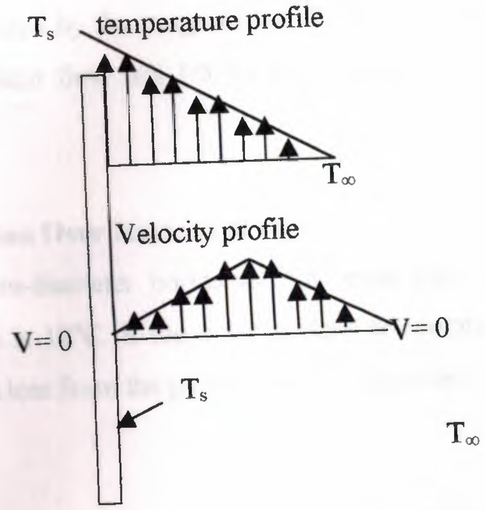


Figure (3-2) Natural convection over vertical surface

The heat transfer rate in natural convection from surface to surrounding is expressed as;

$$\dot{Q} = h A_s (T_s - T_\infty)$$

In order to solve natural convection heat transfer problem, the Grashof number or the Rayleigh number have to be considered on the problem. The Rayleigh number can be calculated as;

$$Ra = Gr.\ Pr$$

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

The average Nusselt number in natural convection is;

$$Nu = \frac{h\delta}{k} C(Gr.\Pr)^n = CRa^n$$

Where C and n are constant and their values depend on the geometry of the surface and flow regime, which is characterized by the range of the Rayleigh number. The value of n is usually equal to ¼ for laminar flow and 1/3 for turbulent flow. The value of C is less than one.

EXAMPLE: Natural Convection Over Horizontal Pipe.

A 6-m-long section of an 8-cm-diameter horizontal hot water pipe passes through a large room whose temperature is 18°C. if the outer surface temperature of the pipe is 70°C, determine the rate of heat loss from the pipe by natural convection.

SOLUTION:

We assume the air pressure in the room to be 1atm. The properties of the air are to be evaluated at the average temperature T_v;

$$T_v = \frac{T_s + T_\infty}{2}$$

$$T_v = \frac{70+18}{2} = 44^\circ C = 317K$$

At this temperature, we read

$k = 0.0273(W/m.\ ^\circ C)$

$v = 1.74\times 10^{-5} m^2/s$

$Pr = 0.71$

$\beta = 1/T_v = 0.00315K^{-1}$

The characteristic length in this case is the outer diameter of the pipe, $\delta=D=0.08m$. Then the Rayleigh number becomes;

$$Ra = \frac{g\beta(T_s - T_\infty)\delta^3}{v^2}\Pr$$

$$Ra = \frac{(9.8m/s^2)(0.00315K^{-1})[(70-18)K](0.08m)^3}{(1.74\times 10^{-5} m^2/s)^2}(0.710) = 1.930\times 10^6$$

Then the natural convection Nusselt number in this case can be calculated as;

$$Nu = \left\{ 0.6 + \frac{0.387(1.930 \times 10^6)^{1/6}}{\left[1 + (0.559/0.710)^{9/16}\right]^{8/27}} \right\}^2 = 17.2$$

then

$$h = \frac{k}{D} Nu = \frac{0.0273 W/(m.^\circ C)}{0.08m}(17.2) = 5.9 W/(m^2.^\circ C)$$

$$A = \pi.\ L.\ D = \pi.\ (6m).\ (0.08m) = 1.51m^2$$

and

$$\dot{Q} = hA(T_s - T_\infty)$$

$$\dot{Q} = \left[5.9W/(m^2.^\circ C)\right](1.51m^2)(70-18)^\circ C = 463$$

3.5 COMBINED NATURAL AND FORCED CONVECTION

The presence of a temperature gradient in a fluid in a gravity field always gives rise to the natural convection currents, and thus heat transfer by natural convection. Therefore, forced convection is always accompanied by natural convection.

Heat transfer coefficients encountered in forced convection are typically much higher than those encountered in natural convection because of the higher fluid velocities associated with forced convection. For given fluid, it is observed that parameter Gr/Re^2 represents the importance of natural convection relative to forced convection. Both natural and forced convection must be considered in heat transfer calculations when the Gr and Re^2 are of the same order of magnitude.

When determining heat transfer under combined natural and forced convection conditions, it is tempting to add the contributions of the natural and forced convection in assisting flow, and to subtract the in opposing flows. A review of experimental data suggests a correlation of the form;

$$Nu_{combined} = \left(Nu^n_{forced} \pm Nu^n_{natural}\right)^{1/n}$$

Where Nu_{forced} and $Nu_{natural}$ are determined from the correlations for pure forced and pure natural convection, respectively. The plus sign is for assisting and transverse flows, and the minus sign is for opposing flow. The value of the exponent n varies between 3 and 4, depending on the geometry involved.

CONCLUSION

In this chapter firstly the classification of convection heat transfer are given as forced and natural convection. The rate of convection heat transfer is calculated. Secondly the important dimensionless numbers are given as Nusselt, Reynolds, and Grashof numbers. Thirdly the rate of forced convection over flat plate, cylinder, and sphere are calculated. The rate of natural convection over vertical plate is calculated. Finally the combined of natural and forced convection are explained.

CHAPTER 4

HEAT TRANSFER FROM EXTENDED SURFACE

4-1 INTRODUCTION

The rate of heat transfer from a surface at a temperature T_s to the surrounding medium at T_∞ is given by Newton's law of cooling as;

$$\dot{Q} = hA(T_s - T_\infty)$$

An inspection of this relation reveals that the only way to increase the heat transfer, for a fixed surface-to-fluid temperature difference is to increase the heat transfer coefficient, the surface area, or both. We can increase the heat transfer coefficient by using a fluid with better heat transfer properties, but our choice of fluids may be limited by system requirements. We could also increase the heat transfer coefficient by the velocity of the fluid. Unfortunately, increasing the fluid velocity requires larger and powerful fans and pumps, which, once again, increase system weight and cost. The most viable approach for increasing heat transfer is to increase the surface area by attaching to the heat transfer surface a solid protrusion called a fin. The surface of the fin exposed to the surrounding fluid is referred to as an extended surface.

Advantages of extending a heat transfer surface may be viewed in another way. In many heat transfer application, the heat transfer and the surrounding fluid temperature are fixed, these parameters being based on the characteristics of the thermal system and the environment in which it operates.

4-2 FIN CLASSIFICATION

Fins are classified according to their geometry, and the most common types are shown in Figure (4-1). A long fin is a flat plate that protrudes from a plane or cylindrical surface. A pin fin, sometimes called a spine, is a rod of circular, square, or rectangular cross section whose end is attached to a surface. A radial fin is an annular disk whose inside edge is attached to a cylindrical object such as a pipe or rod. The surface to which

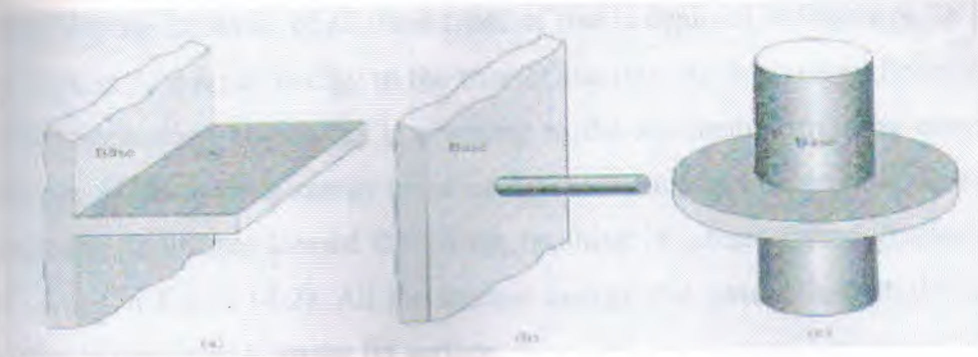


Figure (4-1) The three most common fins are
(a) long fin, (b) pin fin,(c) radial fin.

a fin is attached is called the base, or prime, surface, whereas the end opposite the base is called the fin tip. Fin and the prime surface to which they are attached are normally an integral part constructed of the same material. The most commonly used fin materials are aluminum, copper, and stainless steel. In addition to being used for cooling electronic devices, fins are used on the cylinders of small air-cooled engines and heat exchanger. An automobile radiator, which is actually a heat exchanger, makes use of fins, giving it a very large surface area. Fins are even found in biological systems. The large sail-like structure protruding from the back of the Dimetrodon dinosaur is thought as a large fin for regulating the reptile's body temperature.

Before describing the heat transfer characteristics of longitudinal, pin, and radial fins in detail, we state the following assumptions and simplifications underlying the analysis of all three-fin types.

- The heat transfer is steady and one-dimensional, the direction of heat flow being from the fin base to the fin tip.
- The fin tip is perfectly insulated. This assumption is a valid one, because the surface area of the fin tip is typically much smaller than that of the broad surface.
- The thermal conductivity of the fin is constant. The temperature of the surrounding fluid is constant. The heat transfer coefficient is uniform over the entire fin surface. There is no contact resistance between the fin and the base. There is no heat generation within the fin itself. Radiation from the fin is neglected.

The basic thermal behavior of all three types of fins is depicted in Figure (4-2). A heat source imparts its thermal energy to the base of the fins. As this energy flows through the fin by conduction, the energy is given up to the surrounding fluid by convection. The amount of the thermal energy given up by convection is greatest near the fin base and gradually diminishes toward the fin tip, resulting in a temperature gradient in the fin, as shown in Figure (4-2). All the thermal energy that passes through the base by conduction is convicted from the fin surface.

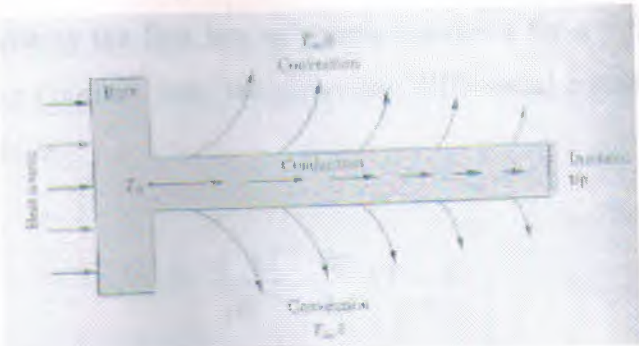


Figure 4-2 Thermal energy conducted along a fin is convicted from the fin surface resulting in a temperature distribution the fin

4-3 THE LONG FINS

In order to analyze the heat transfer for a longitudinal fin, we must first define the pertinent dimensions and parameters. Figure (4-3) describes a typical longitudinal fin, showing the dimensions height, b, length, L, and thickness, t. In the majority of

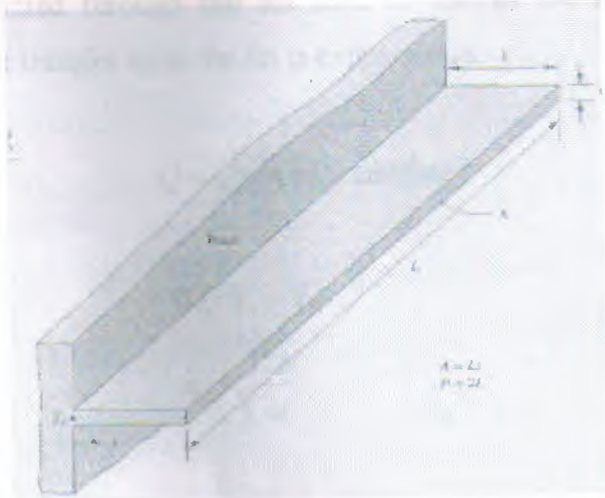


Figure 4-3 A long fin of height b, length L, and thickness t.

Long fins, the length is much greater than the height and thickness is much smaller than the height. The fin's cross sectional area is the area through which conduction occurs along the fin and is given by A= Lt. As heat is conducted along the fin from the base to the tip, heat is lost from both surfaces of the fin to the surroundings by convection. Convection from the two end of the fin is neglected, because their surface areas are small compared to those of the broad surfaces.

We begin the analysis of the long fin by defining an origin at the fin's base, as shown in Figure (4-3). By writing the first law of thermodynamics for a differential element of the fin and applying Fourier's law, the governing differential equation for heat transfer in the fin is obtained as;

$$\frac{d^2\Delta T}{dx^2} - \frac{hp}{kA}\Delta T = 0$$

Where ΔT is defined as the temperature difference between any point along the fin and the surrounding fluid as;

$$\Delta T = T_{(x)} - T_\infty$$

The quantity p is the perimeter of the fin, given by the relation

$$p = 2(L+t) \approx 2L$$

All the heat conducted through the fin base is lost to the surrounding fluid by convection. The heat transfer from the fin is expressed as;

$$\dot{Q} = \sqrt{hpkA}\Delta T \tanh(mb)$$

Where;

$$\Delta T = T_O - T_\infty$$

$$m = \sqrt{\frac{2h}{kt}}$$

The maximum heat transfer is the heat that would be transferred if the temperature of the entire fin were equal to the base temperature. The maximum heat transfer is expressed as;

$$\dot{Q}_{max} = hA_f (T_0 - T_\infty)$$

where A_f is the surface area of the fin.

EXAMPLE: Heat Transfer from Long Fin

A long fin of 6061 aluminum has height, length, and thickness of 3 cm, 20 cm, and 2.5 mm, respectively. Surrounding the fin is the air at 25 °C, where the heat transfer coefficient is 36 W/ m^2. K. If the fin's base temperature is 60 °C, find;

a. The heat transfer from the fin

b. The temperature at the tip of the fin.

SOLUTION:

First, we calculate the dimensions needed to solve the problem. The fin perimeter is

$$P = 2L = 2(0.20m) = 0.40m$$

And the cross sectional area is

$$A = L\,t = (0.2m)\,(0.0025m) = 5.0 \times 10^{-4} m^2$$

The fin base to fluid temperature difference is

$$\Delta T = T_o - T_\infty$$

$$\Delta T = (60 - 25)\,°C = 35°C = 35\,K$$

The thermal conductivity of 6061 aluminum at 300 K is 180 W/m. K. the parameter m is found as;

$$m = \sqrt{\frac{2h}{kt}} = \left[\frac{2(36W/m^2.k)}{(180W/m.k)(0.0025m)}\right]^{1/2} = 12.6m^{-1}$$

Because we will encounter the product mb several times in our calculations, we find its value now;

$$mb = (12.6m^{-1})(0.030m) = 0.378$$

The heat transfer from the fin;

$$\dot{Q} = \sqrt{hpkA}\Delta T \tanh(mb)$$

$$\dot{Q} = \left[36W/(m^2.K)(0.4m)(180W/m.K)(5\times10^{-4}m^2)\right]^{1/2}(35K)\tanh(0.378 = 14.4W$$

The temperature at the fin's tip is found

$$T = \frac{\Delta T}{\cosh(mb)} + T_\infty$$

$$\mathrm{T} = \frac{35^0C}{\cosh 0.378} + 25^0C = 57.6^0C$$

The long fin is probably the most widely used fin, because its simple shape makes it easy to manufacture. An extrusion process generally makes large heat sinks incorporating long fins. Casting, bending, or machining processes usually makes small heat sinks with long fins.

4-4 THE PIN FINS

Most pin fins have a circular or square cross section, as shown in Figure (4-4). The analysis of a pin fin is nearly identical to that of a long fin. The only differences between a pin fin and a longitudinal fin are the formulas for the fin perimeter and cross sectional area. Unlike the long fin, the pin fin has no edges to neglect in the perimeter calculation.

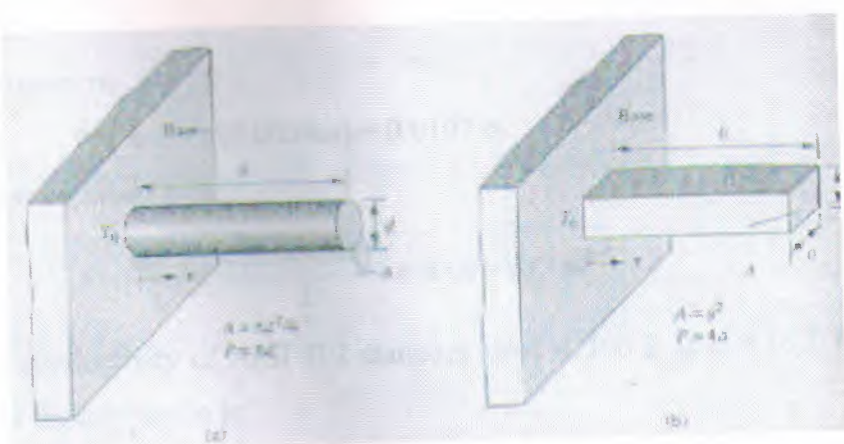


Figure (4-4) Pin fin (a) circular pin, (b) square pin.

The parameter m for a pin fin of circular cross section becomes

$$m = \sqrt{\frac{4h}{kd}}$$

For a pin fin of square cross section P = 4a, and A= a^2 and m becomes

$$m = \sqrt{\frac{4h}{ka}}$$

Sometimes, pin fins have a rectangular cross section, m in which case

$$m = \sqrt{\frac{2h(a+b)}{kab}}$$

where a and b are the dimensions of the edges of the pin fin.

EXAMPLE: Heat Transfer From Pin Fin

A pin fin of AISI 302 stainless steel is attached to a surface whose temperature is 160 °C the diameter of the pin fin is 3.4 mm, and the surrounding the fin is 30°C air, where the heat transfer coefficient is 20W/m^2. K. Find the required fin height if the fin is to transfer 0.7 W

SOLUTION:

The fin perimeter is;

$$P = \pi d = \pi(0.0034m) = 0.0107 \text{ m}$$

And the cross sectional area;

$$A = \frac{\pi d^2}{4} = \frac{\pi(0.0043m)^2}{4} = 9.08 \times 10^{-6} m^2$$

the thermal conductivity of AISI 302 stainless steel at 300 K is k = 15.1 W/m. K. the parameter m for our pin fin is,

$$m = \sqrt{\frac{4h}{kd}} = \left[\frac{4(20W/m^2.K)}{(15.1W/m.K)(0.0034m)}\right]^{1/2} = 39.5m^{-1}$$

The fin base to fluid temperature difference is,

$$\Delta T = T_0 - T_\infty = (160 - 30)\ °C = 130°C$$

The fin height b;

$$b = \frac{1}{m}\tanh^{-1}\left(\frac{\dot{Q}}{\theta_0\sqrt{hpkA}}\right)$$

$$b = \frac{1}{39.5}\tanh^{-1}\left[\frac{0.7}{(130)\left[(20)(0.0107)(15.1)(9.08\times10^{-6})\right]^{1/2}}\right] = 0.073m = 7.3cm$$

In the possible absence of a scientific calculator with the inverse hyperbolic functions, we may utilize the identity;

$$\tanh^{-1}(z) = \frac{1}{2}\ln\left(\frac{1+z}{1-z}\right) \qquad (-1< z< 1)$$

which, as indicated, is defined only for values of z that lie in the range (-1< z< 1). For this pin fin to transfer 0.7 W, the fin must be at least 7.3 cm in height. If the fin height is less than this value, it will not transfer the required amount of heat for the specified conditions.

Suppose that we wanted the fin in the Example 4-2 to transfer 1.0W instead of 0.7 W. The revised calculations would then be;

$$b = \frac{1}{39.5}\tanh^{-1}\left[\frac{1}{(130)\left[(20)(0.0107)(15.1)(9.08\times10^{-6})\right]^{1/2}}\right] = \frac{\tanh^{-1}(1.42)}{39.5}$$

The $\tanh^{-1}$ function is undefined for arguments that lie outside the range indicated in the equation of z function. Hence, if we attempt to find $\tanh^{-1}$ (1.42), an error results. By increasing the heat transfer from 0.7W to 1W, we have attempted to define a situation that is physically impossible. It is impossible for our particular pin fin of any height to transfer 1.0W.

Assuming an infinite fin height b. as z may readily prove this$\rightarrow \infty$, the heat transfer is;

$$\dot{Q} = \sqrt{hpkA}\,\Delta T = \left[(20)(0.0107)(15.1)(9.08\times10^{-6})\right]^{1/2}(130) = 0.703W$$

In order for this pin fin to transfer 1.0 W, one parameter or a combination of parameters must be changed. A heat transfer of 1.0 W could be achieved by increasing the diameter, thermal conductivity, heat transfer coefficient, or the base to fluid temperature difference.

4-5 THE RADIAL FINS

A radial fin is an annular disk whose inside is attached to a cylindrical shaped heat source, such as a hot pipe or rod. A typical radial fin is shown in figure (4-5). The inside and outside radii are R_i and R_o, respectively, and the fin has a uniform thickness t. the outside edge of the fin at $r = R_o$ is defined as the fin tip. The origin is located at the center of the cylinder at r = 0 even thought the fin itself begins at $r = R_i$. Unlike the longitudinal fin and the pin fin, the cross sectional area of a radial fin is not constant but increases with radius r. the governing differential equation for heat transfer in a radial fin is:

$$r^2 \frac{d^2 \Delta T}{dr^2} + r\frac{d\Delta T}{dr} - m^2 r^2 \Delta T = 0$$

Where;

$$\Delta T = T_{(r)} - T_\infty$$

The maximum heat transfer for a radial fin is given as;

$$\dot{Q}_{max} = 2\pi\left(R_o^2 - R_i^2\right)h\Delta T$$

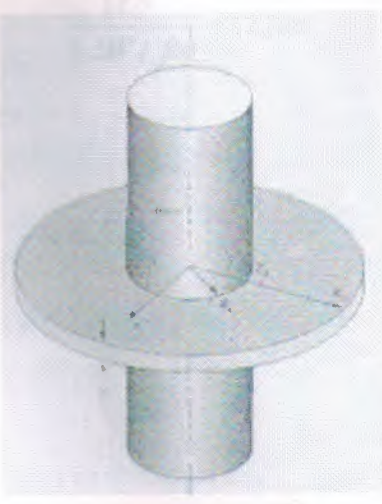


Figure (4-5) Radial fin

4-6 HEAT SINKS

Up to this point we have considered the heat transfer characteristics of single fin only. Single fins are rarely used to extend heat surfaces; because a single fin does not normally have a large enough surface area to achieve the required heat transfer. An array of identical fins that share a common base is normally attached to the heat transfer surface. Such an array of fins having a common base is called a heat sink. Some common types of heat sinks are shown in Figure (4-6).

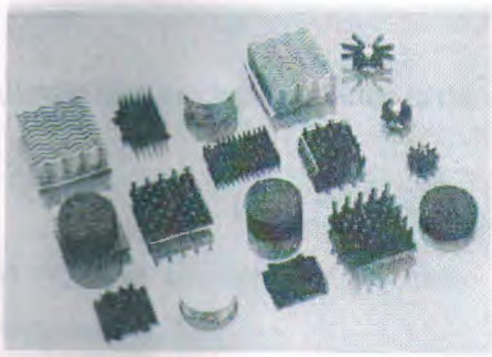


Figure (4-6) Heat sinks incorporating the main fin types

A simple way to extend the analysis of a single fin to a heat sink is to characterize a single fin and then multiply the heat transfer for a single fin by the number of the fins in the heat sink. For example, if we want to design a heat sink or select an off the shelf heat sink to cool an electronic power supply that dissipates 18 W, we might find that a single long fin will transfer 1.5 W, so our heat sink must have;

$$\frac{18W}{1.5W/fin} = 12 fins$$

This simplified approach ignores the heat transfer from the inter fin area, the exposed prime surface between the fins. Even though the simplified method may yield an acceptable heat sink design, a more optimum heat sink may be obtained by accounting for heat transfer from the inter fin areas. Consider the heat sink with longitudinal fins shown in Figure (4-7). And designating the product pb as A_f, the heat transfer from the inter fin areas of the heat sink may be written as;

$$\dot{Q}_{\text{int}erfin} = h(A_t - A_f)\Delta T$$

Where A_t is the total heat sink surface area (the sum of the fin and inter fin surface areas)

The total heat transfer from the heat sink can be written in the compact form as;

$$\dot{Q} = khA_t\Delta T$$

In the foregoing analysis we have assumed that the entire base of the heat sink is maintained at a uniform temperature, T_o

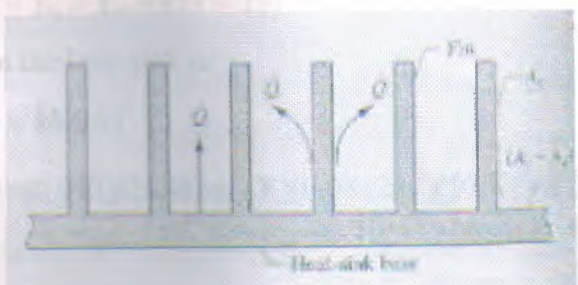
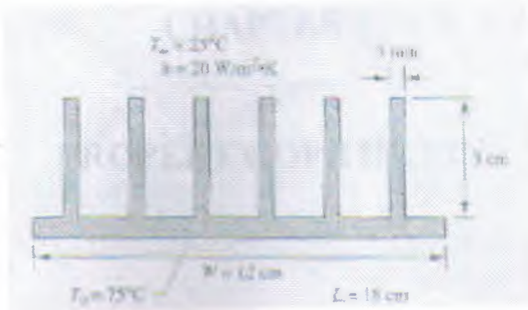


Figure (4-7) Heat transfer from a heat sink occurs from the fins and the areas between the fin.

EXAMPLE: Heat Transfer from Heat Sink.

For the heat sink shown in Figure below, find the heat transfer from the heat sink. The heat sink is constructed of 6061 aluminum, and the fin base temperature is 75 °C. The temperature of the fluid surrounding the fin is 25 °C, and the corresponding heat transfer coefficient is 20 W/m^2. K and the heat sink efficiency is 0.983. The fins span the entire length of the heat sink base.



Heat sinks for example

SOLUTION:

The fin dimensions are b = 0.03m, L = 0.18m, t = 0.003m

The surface area of a single fin is 2Lb, so the surface area for all six fin is;

$$A_f = 6\ (2Lb) = 6(2)(0.18m)(0.03m) = 0.0648\ m^2$$

The total heat sink surface area can be found by adding to the total fin surface area the heat sink base area minus the area occupied by the base of each fin. Thus;

$$A_t = A_f + LW - 6Lt$$

$$A_t = 0.0648 + (0.18)(0.12) - 6\ (0.18)(0.003) = 0.0832\ m_2$$

The fin base to fluid temperature difference is;

$$\Delta T = T_0 - T_\infty = (75 - 25)\ ^\circ C = 50\ ^\circ C = 50\ K$$

The heat transfer from the heat sink is:

$$\dot{Q} = khA_t\Delta T$$

$$\dot{Q} = (0.983)(20\ W/m^2.\ K)(0.0832 m^2)(50\ K) = 81.8\ W$$

CONCLUSION

In this chapter firstly the classification of fins are given as long, pin, and radial fins. The description of those fins is given. Secondly the rates of heat transfer from long, pin, and radial fins are calculated. Finally the heat sink description is given. The numbers of fins in heat sink is calculated. The rate of heat transfer from heat sink is calculate

CHAPTER 5

PROPERTY OF THE FIN

5.1 FIN EFFICIENCY

Consider the surface of a plane wall at temperature T_b exposed to a medium at temperature T_∞. Heat is lost from the surface to the surrounding medium by convection with a heat transfer coefficient of h. disregarding radiation or accounting for its contribution in the convection coefficient h, heat transfer from a surface area A is expressed by Newton's cooling low.

Consider as fin of a constant cross-sectional area and length that is attached to the surface with a perfect contact. This time heat will flow from the surface to the fin by conduction and from the fin to the surrounding medium by convection with the same heat transfer coefficient. The temperature of the fin will be T_b at the fin base and gradually decrease toward the fin tip. Convection from the fin surface causes the temperature at any cross-section to drop somewhat from the midsection toward the outer surface. However, the cross-sectional area of the fin is usually very small, and thus the temperature at any cross-section can be considered to be uniform. Also, the fin tip can be assumed for convenience and simplicity to be insulated by using corrected length for the fin instead of the actual length.

In the limiting case of zero thermal resistance or infinite thermal conductivity, the temperature of the fin will be uniform at the base value of T_b. the heat transfer from the fin will be maximum in this case and can be expressed as;

$$\dot{Q}_{fin,\max} = h.A_{fin}(tT_b - T_\infty)$$

In reality, however, the temperature of the fin will drop along the fin, and thus the heat transfer from the fin will be less because of the decreasing temperature difference toward the fin tip. To account for the effect of this decrease in temperature on heat transfer, we define fin efficiency as;

$$\eta_{fin} = \frac{\dot{Q}_{fin}}{\dot{Q}_{fin,\max}} = \frac{Actual \;\; heat \;\; transfer \;\; rate \;\; from \;\; the \;\; fin}{Ideal \;\; heat \;\; transfer \;\; rate \;\; from \;\; the \;\; fin}$$

or

$$\dot{Q}_{fin} = \eta_{fin}\dot{Q}_{fin,\max} = \eta_{fin}.h.A_{fin}(T_b - T_\infty)$$

This relation enables us to determine the heat transfer from a fin when itrs efficiency is known. For the case of constant cross-section of very long fins and fins with insulated tip, the fin efficiency can be expressed as;

$$\eta_{long,fin} = \frac{\dot{Q}_{fin}}{\dot{Q}_{fin,\max}} = \frac{\sqrt{h.p.k.A}(T_b - T_\infty)}{h.A_{fin}(T_b - T_\infty)} = \frac{1}{L}\sqrt{\frac{k.p}{h.A}} = \frac{1}{a.b}$$

and

$$\eta_{insulated,tip} = \frac{\tanh ab}{ab}$$

An important consideration in the design of finned surfaces is the selection of the proper fin length. Normally the longer the fin, the larger the heat transfer area and thus the higher the rate of heat transfers from the fin. But also the larger the fin, the bigger the mass, the higher the price, and the larger the fluid friction. Therefore, increasing the length of the fin beyond a certain value cannot be justified unless the added benefits outweigh the added cost. Also, the fin efficiency decreases with increasing fin length because of the decrease in fin temperature with fin. Fin lengths that cause the fin efficiency to drop below 60 percent usually cannot be justified economically and should be avoided. The efficiency of most fins used in practice is above 90 percent.

5.2 FIN EFFECTIVENESS

Fins are used to enhance heat transfer, and the use of fins on a surface cannot be recommended unless the enhancement in heat transfer justifies the added cost and complexity associated with the fins. in fact, there is no assurance that adding fins on a surface will enhance heat transfer. The performance of the fins is judged on the basis of the enhancement in heat transfer relative to the no fin case. The performance of fins expressed in terms of the fin effectiveness defined as;

$$\varepsilon_{fin} = \frac{\dot{Q}_{fin}}{\dot{Q}_{no,fin}} = \frac{\dot{Q}_{fin}}{h.A.(T_b - T_\infty)}$$

An effectiveness of ε_{fin} equal to one indicates that is, heat conducted to the fin through the base area is equal to the heat transferred from the same area to the surrounding medium. An effectiveness of fin smaller than one indicates that the fins actually act as insulation, slowing down the heat transfer from the surface. This situation can occur when fins made of low thermal conductivity materials are used. An effectiveness of fin larger than one indicates that fins are enhancing heat transfer from the surface. However, the use of fins cannot be justified unless the effectiveness of fin is sufficiently larger than one. Finned surfaces are designed on the basis of maximizing effectiveness for a specified cost or minimizing cost for a desired effectiveness.

Note that both fin efficiency and fin effectiveness are related to the performance of the fin, but they are different quantities. However, they are related to each other by;

$$\varepsilon_{fin} = \frac{\dot{Q}_{fin}}{\dot{Q}_{no,fin}} = \frac{\dot{Q}_{fin}}{h.A.(T_b - T_\infty)} = \frac{.A_{fin}}{A_b}\eta_{fin}$$

Therefore, the fin effectiveness can be determined easily when the fin efficiency is known, or vice verse. The effectiveness of such a long fin is determined to be;

$$\varepsilon_{long,fin} = \frac{\dot{Q}_{fin}}{\dot{Q}_{no,fin}} = \sqrt{\frac{k.p}{h.A_b}}$$

We can draw several important conclusions from the fin effectiveness relation above for consideration in the design and selection of the fins.

- The thermal conductivity of the fin material should be as high as possible. Thus it is no coincidence that fins are made from metals, with copper, aluminum, and iron being the most common ones. Perhaps the most widely used fins are made of aluminum because of its low cost and weight and its resistance to corrosion.
- The ratio of the perimeter to the cross sectional area of the fin should be as high as possible. Thin plate fins satisfy this criterion or slender pin fins.
- The use of fins is most effective in applications involving convection heat transfer coefficient. Thus, the use of fins is more easily justified when the medium is a gas instead of a liquid and the heat transfer is by natural convection instead of by forced convection. Therefore, it is no coincidence that in liquid to gas heat exchangers such as the car radiator, fins are placed on the gas side.

When determining the rate of heat transfer from a finned surface, we must consider the unfinned portion of the surface as well as the fins. Therefore, the rate of heat transfer for a surface containing n fins can be expressed as;

$$\dot{Q}_{total,fin} = \dot{Q}_{unfin} + \dot{Q}_{fin} = h(A_{unfin} + \eta_{fin}.A_{fin})(T_b - T_\infty)$$

We can also define an overall effectiveness for a finned surface as the ratio of the total heat transfer from the finned surface to the heat transfer from the same surface if there were no fins;

$$\varepsilon_{fin,total} = \frac{\dot{Q}_{total,fin}}{\dot{Q}_{total,nofin}} = \frac{h.(A_{unfin} + \eta_{fin}.A_{fin})(T_b - T_\infty)}{h.A_{nofin}(T_b - T_\infty)}$$

Note that the overall fin effectiveness depends on the fin density as well as the effectiveness of the individual fins. The overall effectiveness is a better measure of performance of a finned surface than the effectiveness of the individual fins.

5.3 PROPER LENGTH OF A FIN

An important step in the design of a fin is the determination of the appropriate length of the fin once the fin material and the fin cross-section are specified. You may be tempted to think that the longer the fin, the larger the surface area and thus the higher the rate of heat transfers. Therefore, for maximum heat transfer, the fin should be infinitely long. However, the temperature drops along the fin exponentially and reaches the environment temperature at some length. The part of the fin beyond this length does not contribute to heat transfer since it is at the temperature of the environment. Therefore, designing such an extra long fin is out of question since it results in material waste, excessive weight, and increased size and thus increased cost with no benefit in return. Fins are so long that the temperature approaches the environment temperature cannot be recommended either since the little increase in heat transfer at the tip region cannot justify the large increase in the weight and cost.

To get a sense of the proper length of a fin, we compare heat transfer from a fin of finite length to heat transfer from an infinitely long fin under the same conditions. The ratio of these two heat transfers is;

$$\frac{\dot{Q}_{fin}}{\dot{Q}_{long,fin}} = \tanh ab$$

A common approximation used in the analysis of fins is to assume the fin temperature varies in one direction only and the temperature variation along other direction is negligible. Perhaps you are wondering if this one-dimensional approximation is a reasonable one. This is certainly the case for fins made of thin metal sheets such as the fins on a car radiator, but we would not be so sure for fins made of thick materials. Studies have shown that error involved in one-dimensional fin analysis is negligible when;

$$\frac{h.\delta}{k} \prec 0.2$$

5-4 FIN OPTIMIZATION

Many modern heat transfer systems that employ extended surfaces demand that the fins transfer the maximum thermal energy using the least amount of the fin. A fin that satisfies this requirement is called an optimum fin. Optimum fins have obvious advantages in the aerospace industry, for example, where weight savings translate directly increased performance and range. Fin optimization is also important in the automotive and computer industries, where stringent weight and space requirements are often imposed.

Because the longitudinal fin is the most commonly used fin, we outline the optimization procedure for the longitudinal fin. We wish to maximize the heat transfer from the fin while minimizing the size of the fin. The fin size may be defined by profile area, A_p, given by the relation A_p/bt.

By replacing the fin thickness with A_p/b, we obtain the heat transfer per unit length as

$$\frac{\dot{Q}}{L} = \sqrt{\frac{2hkA_p}{b}}\theta_o \tanh\sqrt{\frac{2hb^3}{kA_p}}$$

Conclusion

In this chapter firstly the properties of the fin are given as fin efficiency, and fin effectiveness. The definition of the fin efficiency is given. The definition of the fin effectiveness is given. The fin efficiency is calculated. The fin effectiveness is calculated. Secondly the proper length of fin is given. The proper length of the fin is calculated. Finally the optimization fin is given

CONCLUSION

Through this project introduction of heat transfer, heat transfer by conduction, heat transfer by convection, heat transfer from extended surface, and the property of the fin are given.

In the first chapter the heat transfer are classified in three-difference way conduction, convection, and radiation. The three modes of the heat transfer cannot exist in one medium. It must be temperature difference to occur heat transfer.

In the second chapter heat transfer by conduction problem is classified as steady or transient. The rate of heat transfer does not change with time is known as steady heat transfer. The rate of heat transfer changes with time is known as transient heat transfer. The materials have high thermal conductivity are good conduct to heat flow. The materials have low thermal conductivity are poor conduct to heat flow.

In the third chapter heat transfer by convection is classified as forced convection and natural convection. Laminar flow and turbulent flow are important for determination the rate convection heat transfer. In order to know the flow condition laminar or turbulent, the Reynolds number and Grashof number must be calculated.

In the fourth chapter and fifth chapter the fin surface and the properties of the fins are explained. The most common fins used are long fin, pin fin, and radial fin. The fin efficiency and fin effectiveness are important for heat transfer calculation.

Since the increasing surface area is important for increasing heat transfer so I wish that scientists developed new method for increasing heat transfer and to be reasonable cost.

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