Experimental Analysis of Natural and Forced Convective Heat Transfer through Cylindrical Pin Fin

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ABSTRACT

This research work deals with the experimental analysis of convective heat transfer through pin fin in both natural and forced mode. The cylindrical pin fin with dimensions as mentioned in 5.2 was used for experiment on pin fin apparatus. The variations in different parameters were noticed during both modes of convective heat transfer as mentioned in 8 and 9. This work helped to comparative study between the natural and forced convective heat transfer through pin fin.

Key Words: Convective heat transfer, Natural mode, Forced mode, Circular, Pin fin.

Nomenclature:

ρ	=	Density of air, Kg / m^3
D	=	Diameter of pin-fin, m
μ	=	Dynamic viscosity, N.sec/
Cp	=	Specific heat, KJ/Kg ^{0}k
ע	=	Kinematic viscosity, m^2/Sec
K	=	Thermal conductivity of air, W/m ^{0}C
h	=	Heat Transfer coefficient, W/ m^{20}C
g	=	Acceleration due to gravity, 9.81m/sec^2
A	=	Cross section area of the fin.
C	=	Circumference of the fin
L	=	Length of the fin.
T_1	=	Temperature of the fin at the beginning
Tf	=	Duct fluid temperatures
Tm	=	Average fin temperature
Φ	=	$(T - T_{AMB})$ = Rise in temperature
β	=	Coefficient of thermal expansion, per ^{0}C
H	=	Difference of levels in manometer
C_d	=	Coefficient of discharge = 0.64
d	=	Diameter of the orifice = 14mm.

Introduction

1.1 Convective Heat Transfer

The process of heat transfer between the surface and surrounding fluid is known as Convective heattransfer.In this process, the flow of energy is primarily due to the movement of fluid molecules.

There are two types of convective heat transfer.

1.Forced convection 2.Natural convection

Forced convection: In this type, the molecules of fluid are forced to move over the surfacewith the help of some external force. We can get higher rate of heat transfer.

I.D. and F.D. fans are used for to obtain the flow of flue gases in the boiler.

Natural convection: In this type, the fluid motion is caused by buoyancy forces that result from the density variations due to variations of temperature in the fluid.

For example, in the absence of an external source when the mass of the fluid is in contact with the hot surface, its molecules separate and scatter causing the mass of fluid to become less dense. When this happens, the fluid is displaced vertically or horizontally while the cooler fluid gets denser and the fluid sinks. Thus the hotter volume transfers heat towards the cooler volume of that fluid.[2]

1.2 Pin Fin

Extended surfaces (fins) are frequently used in heat exchanging devices for the purpose of increasing the heat transfer between a primary surface and the surrounding fluid. Various

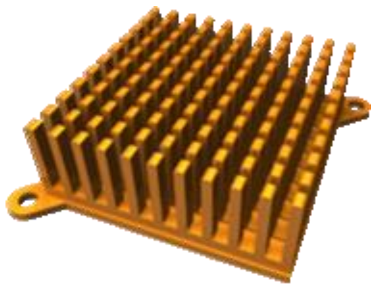


Fig.1 Cylindrical Pin fin array

types of heat exchanger fins, ranging from relatively simple shapes, such as rectangular, square, cylindrical, annular, tapered or pin fins, to a combination of different geometries,have been used. These fins may protrude from either a rectangular or cylindrical base. One of the commonly used heat exchanger fins is the pin fin.[1]

A pin fin is a cylinder or other shaped element attached perpendicular to a wall with the transfer fluid passing in crossflow over the element. Pin fins having a height-to- diameter ratio, H/d, between 0.5 and 4 are accepted as short fins, whereas long pin fins have a pin height-to-diameter ratio, H/d, exceeding 4 . The large height-to- diameter ratio is of particular interest in heat -exchanger applications in which the attainment of a very high heat-transfer coefficient is of major concern.[1]

1. Industrial applications of pin fin: [1]

1. Short pin fins are widely used in the trailing edges of gas-turbineblades.
2. Pin fins are used in the electronic cooling i.e. in cooling of electronic circuitboards.
3. In aero industries, pin fins are widelyused.
4. Pin fins are used in airradiators.

3. Related Research

There have been many investigation regarding heat transfer and pressure drop of channels with pin fins, which are restricted to pin fins with circular or few different cross-sections. Sparrow and co-workers were among the first to investigate the heat transfer performance of cylindrical fins. Tahat et al. determined the optimal spacing of the fins in the span wise and stream wise direction for inline and staggered arranged pin fins. Bilen et al. investigated the heat transfer characteristics of both inline and staggered arranged cylindrical fins. Meinders and Hanjalic studied the turbulent flow structure and distribution of the local surface heat transfer coefficient of a cube placed in a spatially periodic in-line matrix of cubes mounted on one of the walls of a plane channel. Şara investigated the enhancement of the heat transfer from a flat surface in a rectangular channel flow by the attachment of staggered square cross-sectional pin fins. [1]

4. Some dimensionlessnumbers:

4.1 Prandtlnumber:

Prandtl number is the ratio of kinematic viscosity to thermal diffusivity.

$$Pr = (\mu\, Cp) / K$$

$$= (\mu / \rho) / (K / \rho.Cp)$$

$$= \nu / \alpha$$

The physical interpretation of Prandtl number follows its definition as a ratio of momentum diffusivity to thermal diffusivity. Itprovides a measure of relative effectiveness of momentum and energy transport by diffusion in velocity and thermal boundary layers, respectively. [2]

4.2 Reynold'snumber:

Reynolds number may be physically interpreted as the ratio of inertia force to viscous force in the velocity boundary layer. Large values of Re denotes high viscous forces.

$Re = \rho VD / \nu$ = Inertia force / Viscous force

Reynolds number plays an important role in determining whether the flow is laminar or turbulent. In any flow, there exist small disturbances that can be amplified to produce turbulent conditions. For small Reynolds number, the viscous forces are sufficiently large relative to inertia forces to prevent this amplification. Hence, laminar flow is maintained. As Reynolds number increases, inertia forces start becoming more dominant and the disturbances get amplified to a point when turbulence occurs. Hence, for turbulent flow, the value of Reynolds number is relatively larger. [5][6]

4.3 Grashof's number:

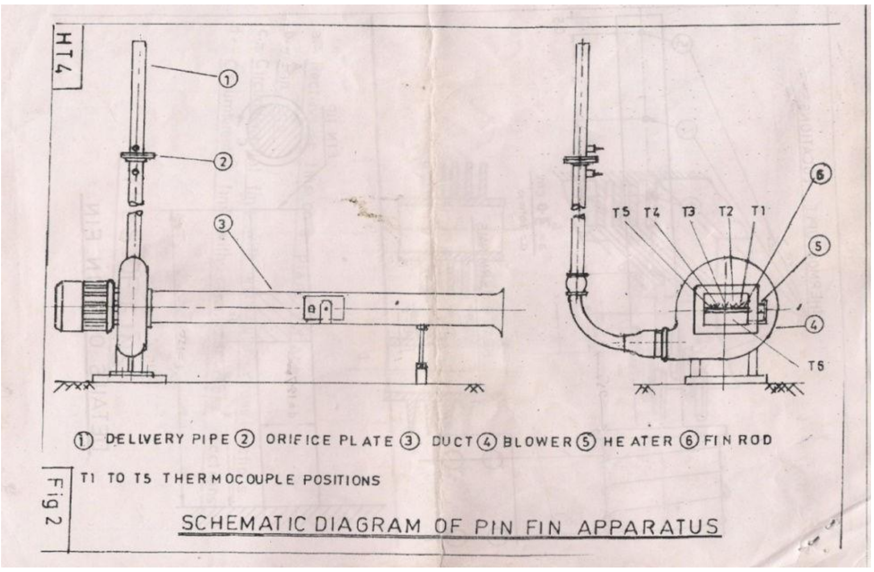
Grashof number (*Gr*) is a dimensionless number used in heat transfer studies involving free or natural convection.

$$Gr = (\beta\, \Delta T D^3)/\nu^2$$

$$= (\, g\, \beta\, d^3\, \Delta T) / \nu^2$$

= (Inertia force x Buoyancy force) / (Viscous force)^2

When fluids are heated or cooled, they expand/shrink causing the density of the fluid to decrease/increase. Thus, if a hot fluid is surrounded by cooler fluid, the hot fluid will rise up due to the upward buoyancy force in presence of gravity. Similarly, cold fluid surrounded by warmer fluid will flow downward. These flows occur naturally or freely, without any external driving force. The induced flow is opposed by viscous drag, usually at the solid walls of the system. The Grashof number compares the buoyancy force to viscous drag.[6]



4. Nusseltnumber:

It is the ratio of convection heat transfer rate to the conduction heat transferrate. [8] Consider an internal flow in a channel of height L and the temperatures at the lower and upper surfaces are T1 and T2 respectively.

The convection heat transfer is, $Q_{conv} = h\ A\ (T1 - T2)$

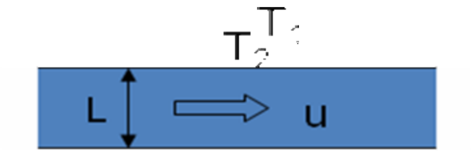
The conduction heat transfer rate is, $Q_{cond} = (k\ A / L)\ (T1 - T2)$

The ratio is,

$Nu_L = Q_{conv} / Q_{cond}$

$= [h\ A\ (T1-T2)] / [(k\ A / L)\ (T1-T2)]$

$= (h\ L) / k$



5. Problemstatement:

“Experimental study of natural and forced convective heat transfer for pin fin”

5.1 Methodology:

A brass fin of circular cross section in fitted across a long rectangular duct. The other end of the duct is connected to the suction side of a blower and the air flows past the fin perpendicular to the axis. One end of the fin projects outside the duct and is heated by a heater. Temperatures at five points along the length of fin are measured by chromelalumelthermocouples connected along the length of the fin. The air flow rate is measured by an orifice meter fitted on the delivery side of theblower.

5.2 Specifications of apparatus:

1. Duct size = 150mm x100mm.
2. Diameter of the fin =12.7mm.
3. Diameter of the orifice =14mm.

4. Diameter of the delivery pipe =42mm.
5. Coefficient of discharge (or orifice meter) C_d =0.64.
6. Centrifugal Blower = Single-phasemotor.
7. No. of thermocouples on fin =5.
8. Thermocouple (6) reads ambient temperature inside of theduct.
9. Thermal conductivity of fin material (Brass) = 110w/m^{0}C.
10. Temperature indicator = 0 – 300 ^{0}C with compensation of ambient temperature up to 50^{0}C.
11. Dimmerstat for heat input control 230V, 2Amps.
12. Heater suitable for mounting at the fin end outside the duct = 400 watts (Bandtype).
13. Voltmeter = 0 – 230V.
14. Ammeter = 0 – 5Amps.
15. Length of fin = 150mm

Fig.4. Experimental setup [9]

6.Theory:

Consider the fin connected at its base to a heated wall and transferring heat to the surroundings.

The heat is conducted along the rod and also lost to the surrounding fluid by convection. Applying the first law of thermodynamics to a controlled volume along the length of the fin at X, the resulting equation of heat balance appearsas:

$(d^2\Phi/dx^2)–(hc/ KA)\Phi=0$ (1)

and the general solution of equation (1) is

$\Phi = C_1.\ e^{mx}+C_2.e^{-mx}$ (2)

Where, $m=\sqrt{\frac{hc}{KA}}$

With the boundary conditions of $\Phi = \Phi1$ at $x = 0$,

Where, $\Phi1= T_1 – T_F$ and assuming the fin tip to be insulated.

$(d\Phi/dx)=0$ at $x= L$ resultsinobtainingequation(2)intheform:

$(T-T_F)/(T_1-T_F)=\cosh\{m(L-x)/\cosh(mL)\}$ (3)

This is the equation for the temperature distribution along the length of the fin. It is seen from the equation that for a fin of given geometry with uniform cross section, the temperature at any point can be calculated by knowing the values of T1, TF and X. Temperature T1 and TF will be known for a given situation and the value of h depends on whether the heat is lost to the surrounding by free convection or forced convection and can be obtained by using the correlation as givenbelow:

1. For free convectioncondition,

$Nu = 1.1\ (Gr.\ Pr)^{1/6}$ …..$10^{-1} <$ Gr. Pr. $< 10^4$ } (4)

$Nu = 0.53\ (Gr.\ Pr)^{1/4}$…...$10^4 <$ Gr. Pr. $< 10^9$ } (4)

$Nu = 0.13\ (Gr.\ Pr)^{1/4}$…..$10^9 <$ Gr. Pr.$< 10^{12}$} (4)

2. For forcedconvection,

$Nu = 0.615\ (Re)^{0.466}$…. $40 < Re < 4000$ (5)

$Nu = 0.174(Re)^{0.618}$ $4000 < Re < 40000$

Where, $Nu = (h.D)/K_{Air}$ =NusseltNumber (a)

$R_e = DV/\nu$ =Reynold's Number (b)

$G_r=(g\beta\ d^3\Delta T)/\nu^2$=GrashofNumber ..(c)

$P_r = \mu\ Cp/K_{Air}$ =PrandtlNumber …(d)

All the properties are to be evaluated at the mean film temperature. The mean film temperature is the arithmetic average of the fin temperature and air temperature.

Tm = Average fin temperature = (T1 + T2 +T3 + T4 + T5) / 5

T_{mF} = Mean film temperature = (T_{AVG}+ T_{AMB}) / 2

β. = Coefficient of thermal expansion = 1/ (T_{mF} +273)

ΔT = TAVG -TAMB

.V = Velocity of air in theduct

The velocity of air can be obtained by calculating the volume flow rate through the duct.

$Q = Cd \times (\pi/4) \times (d_{orifice})^2 \times \sqrt{2gH\left(\frac{\rho water}{\rho air}\right)}$ m^3 / sec Velocity of air at T_{mf} = V= Q / Ductc/sArea m/sec

This velocity is used in the calculation of Re.

The rate of heat transfer from the fin can be calculated as,

$Q = \sqrt{h.c.k.A} \times (T1 - T_f)\tanh(mL)$ (6)

And the effectiveness of the fin can also be calculated as,

$\eta = \tanh(mL)/mL$ (7)

7. Experimental procedure:

For the study of convective heat transfer in naturaland forced mode, the followingprocedure is carriedout.

7.1Natural convection:

i. The heating of pin was started by switching on the heater element and adjusting the dimmerstatvoltage at 50volts.
ii. The thermocouple readings 1 to 5 were noteddown.
iii. After reaching the steady state, final readings 1 to 5 and ambient temperature 6 were noteddown.
iv. The same procedure is repeated for 60V, 70V, 80V &90V and readings were noted down.
v. During all the procedure, the blower was keptoff.

7.2Forced convection:

i. The heating of pin was started by switching on the heater element and adjusting the dimmerstatvoltage at 60volts.
ii. Blower was started and difference of level in the manometer is maintained at 2cm using the gatevalve.
iii. Thethermocouple readings1to5at atimeintervalof5minuteswerenoteddown.
iv. After reaching the steady state, final readings 1 to 5 and ambient temperature 6 were noted down and same procedure is repeated for 70V &80V.

8. Observations:

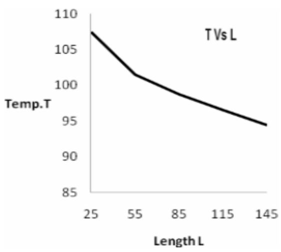
8.1 Table for forcedconvection:

V (volts)	**60**	**70**	**80**
I (amps)	0.318	0.368	0.418
T1	48.4	50.7	53.7
T2	45.9	47.6	49.4
T3	45.1	46.5	48.1
T4	44.6	45.9	47
T5	44	45.1	46.1
Manometer (h cm)	2	2	2
Ambient T6	39	38.9	37.6
T_{avg} ^{0}C	45.6	47.16	48.86
T_{mf} ^{0}C	42.3	43.06	43.23
ΔT	6.6	8.26	11.3
ρa (Kg/m^3)	1.1	1.104	1.11
Kair(W/mk)	0.02692	0.02707	0.02717
νair (m^2 /s)	1.62E-05	1.63E-05	1.62E-05
Re	**209.8233**	**208.5329**	**207.7662**
Nu	**7.4277**	**7.4064**	**7.393**
h (W/m^{20}C)	15.74455	15.78683	15.81799
m	6.7056	6.7146	6.7213

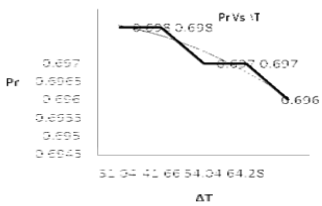
η(%)	75.95999	75.91391	75.87961

8.2 Table for naturalconvection:

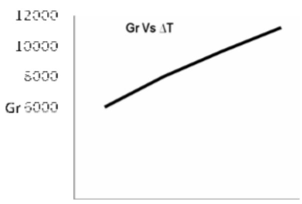
V (volts)	50	60	70	80	90
I (amps)	0.27	0.317	0.368	0.418	0.466
T1	66.3	75.6	88.4	96.9	108
T2	64.1	72.6	84.3	91.2	102
T3	62.9	70.9	82.1	88.7	98.7
T4	62.4	69.9	80.8	87	96.5
T5	61.5	68.8	79.1	85.1	94.4
Ambient T6	32.4	29.9	28.9	25.5	24.4
T_{avg} ^{0}C	63.44	71.56	82.94	89.78	99.72
T_{mf} ^{0}C	47.92	50.73	55.92	57.64	62.06
β	0.003116	0.003089	0.00304	0.003024	0.002985
ΔT	31.04	41.66	54.04	64.28	75.32
Pr	**0.698**	**0.698**	**0.697**	**0.697**	**0.696**
Kair(W/mk)	0.02826	0.02826	0.02856	0.02861	0.02916
ν air (m^2 /s)	0.0000179	0.00001795	0.0000184	0.00001866	0.0000192
Gr	**6032.206**	**8025.786**	**9688.172**	**11219.6**	**12164.79**
Gr*Pr	4210.48	5601.999	6752.656	7820.061	8466.694
Nu	**4.42023**	**4.6356**	**4.78228**	**4.900703**	**4.96602**
h (W/m^{2}^{0}C)	9.835897	10.315314	10.754503	11.040089	11.402309
m	5.300125	5.427756	5.542099	5.615202	5.706575
q (W)	1.521661	2.125107	2.853655	3.468619	4.173587
η(%)	**83.17323**	**82.52405**	**81.94024**	**81.56607**	**81.09755**



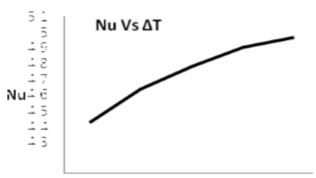
(1)



(2)

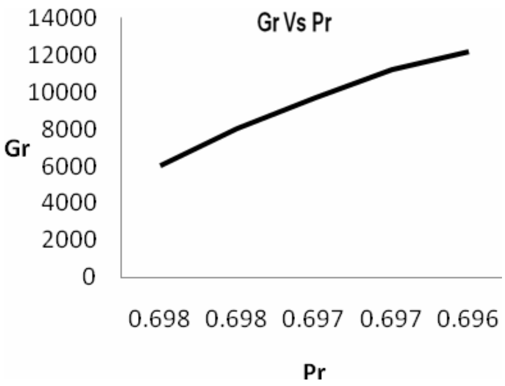


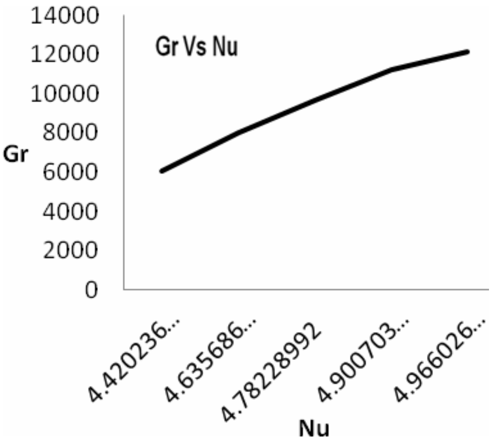
ΔT



ΔT

(3) (4)





(5) (6)

(1) Temperature Vs Distance of thermocouple orL

The equation (3) shows that as the distance of thermocouple mounted on pin fin from base plate goes on increasing in the direction of heat flow, the temperature goes on decreasing.

This happens because temperature is inversely proportional to distance or length of pin fin. The pin fin used for experiment is considered with the insulated end. At the insulated surface, the temperature gradient tends tozero.

(2) Prandtl number Vs∆T

From the graph, it can be observed that as temperature gradient goes on increasing Prandtl number gets decreased. Equation (d) shows that Pr depends on Cp which is dependent on

∆T. Hence, with the increase of ∆T Pr gets decreased.

(3) Grashof's number Vs∆T

The graph shows that with the increasing temperature gradient Grashof's number also increases. It can be cleared from the equation (c) that Gr is directly proportional to ∆T. As temperature gradient increases in natural convection, the density of air decreases resulting in increased kinematic viscosity. Due to this, Gr also increases.

(4) Nusselt number Vs∆T

The graph indicates that with increased temperature gradient Nusselt number also increases. This is because the Nu depends on GrPr product which increases with increased∆T.

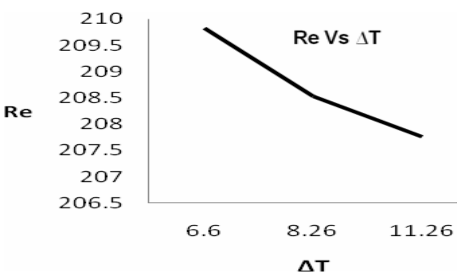
(5) Grashof's number Vs Prandtl number

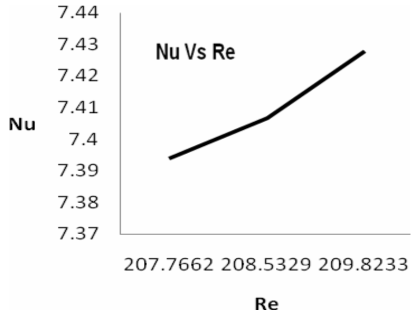
In natural convection, increase in Prandtl number leads to increased Grashof's number. This is observed from the graph(5).

(6) Grashof's number Vs Nusseltnumber

The graph indicates that the Grashof's number increases with the increase in Nusselt number.

9.1 For forcedconvection:





(7) Reynold's number Vs ΔT

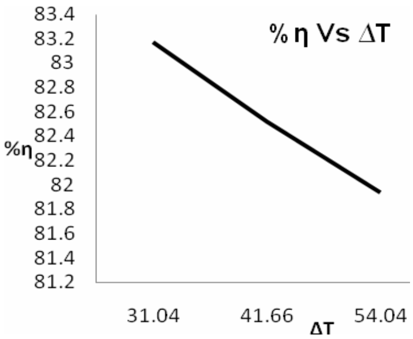
The above graph shows that with increase in ΔT, the Reynold's number decreases. With the increased ΔT, kinematic viscosity of air increases and due to this Re decreases with increased ΔT as from equation (b).

(8) Nusselt number Vs Reynold'snumber

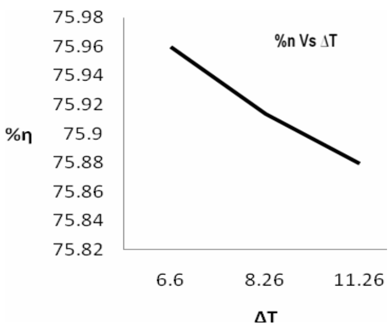
As the Nusselt number is the function of Reynold's number and these are directly proportional to each other. Therefore, from equation (5), it is clear that Reynold's number increases with increase in Nusseltnumber.

(9) % Efficiency of fin VsΔT

% fin efficiency is decreased with the increased temperature gradient. This is because with increase in temperature, thermal conductivity of air increases with increase in heat transfer coefficient. Due to this fin efficiency decreases with increased ΔT in both natural and forcedconvection.



(9.a)NaturalConvection



(9.b) ForcedConvection

9. Conclusions:

1. The temperature gradient decreases with the increase in the distance of point on pin fin from its base wall or plate in both natural and forcedconvection.
2. The value of Prandtl number decreases with increasing temperature gradient in naturalconvection.
3. The values of Greshof's number and Nusselt number increases with increase in temperature gradient in naturalconvection.
4. The value of Reynold's number decreases with increasing temperature gradient in forced convection.

5. Efficiency of pin reduces with increased temperature gradient in both natural and forced convection.

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