

# Analysis, Design and Fabrication of Forced Convection Apparatus

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**ABSTRACT:** The analysis of Forced Convection is simplified by way of the Newton's Law of cooling which relates the rate of heat transfer and the finite temperature difference between the surface of a solid and the surrounding fluid temperature in which the solid is kept immersed.

This can be done by assigning pre-fixed values based on previous experimentation or theoretically by employing the Non-dimensional numbers or by using suitable numerical techniques with standard software. The challenge lies in adopting a procedure which would yield consistent results. The work was carried out in phases with each phase attempting to bring down the difference between the Experimental and Theoretical Heat transfer coefficients.

**KEYWORDS:** Forced Convection, heat transfer coefficient, thermal boundary layer

## I. INTRODUCTION

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

The flow in boundary layer starts as smooth and streamlined which is called laminar flow. At some distance from the leading edge, the flow turns chaotic, which is called turbulent and it is characterized by velocity fluctuations and highly disordered motion. The transition from laminar to turbulent flow occurs over some region which is called transition region. The velocity profile in the laminar region is approximately parabolic, and becomes flatter in turbulent flow. The turbulent region can be considered of three region laminar sublayer (where viscous effects are dominant), buffer layer (where both laminar and turbulent effects exist), and turbulent layer. The intense mixing of the fluid in turbulent flow enhances heat and momentum transfer between fluid particles, which in turn increases the friction force and the convection heat transfer coefficient.

In convection, it is a common practice to non-dimensionalize the governing equations and combine the variables which group together into dimensionless numbers (groups).

Nusselt number non-dimensional heat transfer coefficient:

$$Nu = \frac{h\delta}{k} = \frac{q_{conv}^{\bullet}}{q_{cond}^{\bullet}}$$

where  $\delta$  is the characteristic length, i.e. D for the tube and L for the flat plate. Nusselt number represents the enhancement of heat transfer through a fluid as a result of convection relative to conduction across the same fluid layer. Reynolds number ratio of inertia forces to viscous forces in the fluid:

$$Re = \frac{\text{inertia forces}}{\text{viscous forces}} = \frac{\rho V \delta}{\mu} = \frac{V \delta}{\nu}$$

At large Re numbers, the inertia forces, which are proportional to the density and the velocity of the fluid, are large relative to the viscous forces; thus the viscous forces cannot prevent the random and rapid fluctuations of the fluid (turbulent regime). The Reynolds number at which the flow becomes turbulent is called the critical Reynolds number. For flat plate the critical Re is experimentally determined to be approximately Re critical = 5 x 10<sup>5</sup>. Prandtl number: is a measure of relative thickness of the velocity and thermal boundary layer

$$Pr = \frac{\text{molecular diffusivity of momentum}}{\text{molecular diffusivity of heat}} = \frac{\nu}{\alpha} = \frac{\mu C_p}{k}$$

Where, fluid properties are:

mass density :  $\rho$ , (kg/m<sup>3</sup>) specific heat capacity :  $C_p$  (J/kg · K)

dynamic viscosity :  $\mu$ , (N · s/m<sup>2</sup>) kinematic viscosity :  $\nu$ ,  $\mu / \rho$  (m<sup>2</sup>/s)

thermal conductivity :  $k$ , (W/m· K) thermal diffusivity :  $\alpha$ ,  $k/(\rho \cdot C_p)$  (m<sup>2</sup>/s)

## II. GENERAL SPECIFICATIONS

- Test Pipe: Stainless steel Pipe,
- ❖ Pipe 1 : Outer Diameter 48.3 mm
- ❖ Pipe 2 : Outer Diameter 60.3 mm
- Electric Heater: Capacity 3000 W, Supply 230 V AC
- Air Blower: Centrifugal Blower, 230 V AC
- Pitot tube : For Air Flow measurement
- Manometer: U tube manometer, 0-400 mm WC, 0-200 mm WC
- Dimmer: Range 0-250 V AC, 2 A
- Digital Voltmeter: Range 0-500 V AC
- Temperature Sensors: RTD, PT type
- Temperature Indicator: 8 Channel Indicator with Selector Switch

## III. EQUATIONS

The following equations have been to arrive at the heat transfer coefficient and shown below is a SAMPLE CALCULATION.

PIPE 1: SET No. 1

### ❑ To find experimental heat transfer coefficient:-

#### ▪ Air temperature (°C),

$$\begin{array}{ll} T_1 = 35 & T_4 = 38 \\ T_2 = 36 & T_5 = 38 \end{array}$$

#### ▪ Pressure loss, $\Delta P$

$$\begin{array}{l} \text{density of water, } \rho_w = 1000 \text{ kg/m}^3 \\ \text{acceleration due to gravity, } g = 9.81 \text{ m/s}^2 \\ \Delta P = \rho_w g h \\ = 735 \text{ N/m}^2 \end{array}$$

#### ▪ Density of air at a temperature = 38 °C, $\rho_a$

$$\rho_a = 1.109 \text{ kg/m}^3$$

#### ▪ Velocity ( $v$ )

$$\Delta P = \frac{1}{2} \rho_a v^2$$

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density of air,  $\rho_a = 1.109 \text{ kg/m}^3$   
 $v = 36 \text{ m/s}$

- **Mass flow rate of air through the pipe ( $\dot{m}$ )**

$$\dot{m} = \rho_a A_c v$$

Hence,  $\dot{m} = 0.056 \text{ kg/s}$

- **Heat gained by air ( $Q_a$ )**

$$Q_a = \dot{m} c_p \Delta T_a$$

Where,

$\dot{m}$  = mass flow rate of air in kg/s

$$c_p = \text{specific heat of air} = 1005 \text{ J/kgK}$$

$$\Delta T_a = \text{change in air temperature} = 3$$

$$\text{Hence, } Q_a = 168.84 \text{ w}$$

- **Heat lost by pipe/Convective heat transfer ( $Q_h$ )**

$$Q_h = h A_s \Delta T_p$$

Where,

$h$  = convective heat transfer coefficient in  $\text{W/m}^2\text{K}$

$$A_s = \text{surface area of test specimen} = 0.113 \text{ m}^2$$

$$\Delta T_p = T_3 - \frac{T_1 + T_5}{2} = 20.2$$

- **Hence Convective heat transfer coefficient ( $h_{exp}$ ),**

$$\dot{m} c_p \Delta T_a = h A_s \Delta T_p$$

$$h_{exp} = 74.58 \text{ W/m}^2\text{K}$$

- **To find Theoretical heat transfer coefficient ( $h_{th}$ )**

- Average fluid film temperature,  $T_f$

$$T_f = \frac{T_s + T_a}{2}$$

$$\text{At } T_f = 46.8 \text{ }^\circ\text{C}$$

- Reynold's number, Re

$$\text{Re} = \frac{vD}{\nu}$$

$$= 87510.2$$

- Nusselt number, Nu

$$\text{Nu} = .023 \text{Re}^{.8} \text{Pr}^{.4}$$

$$= 176.5$$

- Theoretical heat transfer coefficient ( $h_{th}$ )

$$h_{th} = \frac{\text{Nu} K}{D}$$

$$= 116.4 \text{ W/m}^2\text{K}$$

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**IV. FIGURES**

The picture taken during the 1<sup>st</sup> phase of the experimental set-up is as shown below.



**V. TABLES**

THE FOLLOWING ARE THE RESULTS OF THE EXPERIMENT AND ANALYSIS DONE IN TWO PHASES.

Phase 1 – Pipe OD = 60 mm

Sl No	V	Velocity	AIR TEMPERATURE(°C)				T <sub>wall</sub>	REYNOLDS No	CONVECTIVE HEAT TRANSFER COEFFICIENT W/m <sup>2</sup> K		
		m/s	T1	T2	T4	T5	T3	Re	<i>h<sub>exp</sub></i>	<i>h<sub>th</sub></i>	<i>h<sub>ana</sub></i>
1	80	19.6	35	37	38	38	64	59390.63	38.76	67.63	62.85
2	120	19.7	36	41	43	43	101	53754.27	34.56	65.06	62.06
3	160	19.8	37	45	47	47	156	46704.6	31.49	62.65	61.71

Phase 2 : The experiment was repeated with better insulation and tighter packing.

S.No	V	T1 °C	T2 °C	T3 °C	T4 °C	T5 °C	T6 °C	T <sub>avg</sub> °C	<i>h<sub>exp</sub></i> W/m <sup>2</sup> K	Re	Nu	<i>h<sub>th</sub></i> W/m <sup>2</sup> K
1	70	36	39	39	74	42	46	41.5	54.7282	44819.5	105.597	52.701
2	80	35	39	39	73	46	44	42	57.9589	69984.4	139.335	69.539

#### VI. CONCLUSION

(Phase- I) Percentage difference between hth and hexp at 80 V, taking hth as base value = 42.63%

(Phase – II) Percentage difference between hth and hexp at 80 V, taking hth as base value = 16.65%

From the above, the following may be inferred:

1. There is marked improvement in the heat transfer values after using better insulation materials & packing methods.
2. As can be foreseen, the theoretical values are always higher due to several forms of losses by which the entire heat is not being transferred into the flowing air (insulation losses, poor thermal conductivity of pipe material, insufficient pipe length or flow losses)
3. When repeated readings were being taken continuously with blower running for longer periods, the motor gets heated up and thus the air inlet temperature increases which thereby reduces the effective temperature difference and consequently the heat transfer coefficient.
4. At higher wall temperatures ( $> 90^{\circ}\text{C}$ ) the ambient heat losses are more which could be due to the insulation ineffectiveness.

#### FUTURE SCOPE :

The effect of the following design changes may be studied :

1. Changing the pipe material from stainless steel to Aluminium or Brass or Copper
2. Increasing the pipe test length in an attempt to achieve a fully developed thermal boundary layer
3. Incorporating vents in the surface of the existing pipe wall to enhance the convective heat transfer.
4. Cooling the air from the blower before entering the heater section.
5. Using better insulation materials with increased thickness.
6. One or more combinations of the above methods.

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