Experimental Study on Heat Transfer Enhancement for Air Flow through A Corrugated Duct

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Abstract— In this experimental study on flow and local heat/mass transfer characteristics inside corrugated duct, the present study investigates the flow and heat mass transfer characteristics of wavy duct for the primary surface heat exchanger at this application. In this experiment the local heat and mass transfer coefficient on the corrugated duct side walls are measured. The flow observed & analysis using various commercial cycle. Our experiment corrugation angle of duct is 145° & aspect ratio is 7.3. [1] The Reynolds number is based on hydraulic diameter it vary from 100 to 5000. In this experiment we study the various parameters like Reynolds number, Nusselt number & Flow characteristics etc. We found that heat transfer enhancement increases by using corrugated duct.

Keywords—Blower, Corrugated Duct, Heat Exchanger, Plenum Chamber, Rectangular duct, Thermocouple.

I. INTRODUCTION

For obtaining high effectiveness and low pressure losses, minimum volume and weight, and high reliability and low cost, all this parameter are designed in general power plants and micro turbine systems. To reduce size and increase cycle efficiency of gas turbine recuperators, the effective heat exchanger is required. For this application, generally the heat exchangers contain flow channel with various cross-sectional shapes, and corrugated, curved, to enhance the heat transfer rates.

Various numerical studies and approaches to the flow and heat transfer characteristics in corrugated ducts are conducted. Asako and Faghri (1987) conducted the numerical studies to predict heat transfer coefficients, friction factors, and streamlines of the corrugated duct at laminar flow region (Reynolds numbers range from 100 to 1500). Sawyers et al. (1998) investigated steady laminar heat transfer characteristics in corrugated channels using a combination of analytical and numerical techniques. Rokni and Gatski (2001) conducted numerical calculations of convective heat transfer coefficients in trapezoidal ducts. They found that enhancement of heat transfer and increase in pressure losses were obtained by changes of duct shape.[1]

Many experimental studies about heat/mass transfer characteristics of straight ducts have been conducted. Cho et al. (2003) investigate flow and local heat/mass transfer characteristics of a rectangular duct with rib turbulators. As a successive study, O’Brien and Sparrow (1982) carried out heat transfer measurements to determine the convective heat transfer coefficients. They measured friction factors in a corrugated duct with a corrugation angle of 30° and an aspect ratio of 10.[1]

In this experiment the Reynolds numbers, based on the duct hydraulic diameter, vary from 1000 to 5000. The tested corrugation angle (α) and aspect ratio (H/W) of the wavy duct are 145° and 7.3, respectively.

Many numerical studies have been conducted; however, most have not confirmed the validity of assumptions included in their calculations due to the shortage of verified experimental data. For the experimental studies, most researches deal with the flow and average heat transfer characteristics of wavy ducts, but not with the local heat/mass transfer characteristics. The present study focuses on measurements of the detailed local mass transfer coefficients in wavy ducts having a rectangular cross section for application to compact heat exchangers. In this study, the effects of the flow velocity on the local heat/mass transfer in corrugated duct are investigated.

<table>
<thead>
<tr>
<th>Nomenclature</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>$D_h$</td>
<td>hydraulic diameter</td>
</tr>
<tr>
<td>$f$</td>
<td>friction factor</td>
</tr>
<tr>
<td>$H$</td>
<td>duct height</td>
</tr>
<tr>
<td>$H/W$</td>
<td>aspect ratio</td>
</tr>
<tr>
<td>$h_m$</td>
<td>mass transfer coefficient</td>
</tr>
<tr>
<td>$L$</td>
<td>duct length</td>
</tr>
<tr>
<td>$\dot{m}$</td>
<td>mass transfer rate per unit area</td>
</tr>
<tr>
<td>$Nu$</td>
<td>Nusselt number</td>
</tr>
<tr>
<td>$P$</td>
<td>corrugation pitch</td>
</tr>
<tr>
<td>$Pr$</td>
<td>Prandtl number</td>
</tr>
<tr>
<td>$Re$</td>
<td>Reynolds number</td>
</tr>
<tr>
<td>$W$</td>
<td>Duct width</td>
</tr>
<tr>
<td>$\alpha$</td>
<td>corrugation angle</td>
</tr>
</tbody>
</table>
II. EXPERIMENTAL SETUP

Schematic view of the experimental apparatus is shown in Fig. 1 and consists of a blower, an orifice flow meter, a plenum chamber, thermocouples, U-tube manometer, corrugated duct, rectangular duct, cock, test section. The dimensions of the corrugated duct are shown in table 1.

<table>
<thead>
<tr>
<th>Sr. No</th>
<th>Specification</th>
<th>Symbol</th>
<th>Dimensions</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Duct length</td>
<td>L</td>
<td>1200 mm</td>
</tr>
<tr>
<td>2</td>
<td>Duct height</td>
<td>H</td>
<td>120 mm</td>
</tr>
<tr>
<td>3</td>
<td>Duct width</td>
<td>W</td>
<td>150 mm</td>
</tr>
<tr>
<td>4</td>
<td>Aspect ratio</td>
<td>H/W</td>
<td>7.3</td>
</tr>
<tr>
<td>5</td>
<td>Corrugation angle</td>
<td>α</td>
<td>145°</td>
</tr>
<tr>
<td>6</td>
<td>Hydraulic diameter</td>
<td>Dₜ</td>
<td>12.8 mm</td>
</tr>
</tbody>
</table>

In this experiment for both setup Figure 1 & Figure 2 common components are blower, orifice flow meter, U-tube manometer, valve, plenum chamber etc. and only difference in the rectangular duct & corrugated duct.

III. EXPERIMENTAL PROCEDURE

Atmospheric air entrained from the inlet contraction flows through the test duct and the plenum chamber then is discharged out of the room by the blower (3.7 kW). Flow rates are measured using a thin plate orifice flow meter installed between the plenum chamber and the blower. The temperatures of the flow and room air are measured by the thermocouples (J-type). The geometry and coordinate system of the test section are shown in Fig. 3.
The corrugated duct consists of ten pairs of pressure side and suction-side walls. Hence, the total length of the wavy duct is five-corrugation pitches and the test section is located at the fourth pitch from the inlet contraction. The duct has sharp-edged turns and a rectangular cross-section. The aspect ratio and corrugation angle of the corrugated duct are 7.3 and 145°, respectively. The corrugation pitch (P) between the turning edges is 64.0 mm. The detailed dimensions and hydraulic diameter of the wavy ducts are described in Table 1.

IV. THEORY

1. Heat Transfer

The heat transfer in convection is given by Newton’s law of cooling,

\[ q = h \times A \times \Delta T \]

where,

- \( q \) is the heat transfer rate (W)
- \( h \) is the heat transfer coefficient (W/m²k)
- \( \Delta T \) is the temperature difference (K)
- \( A \) is the surface area through which transfer occurs (m²)

To increase the heat transfer rate \( q \), whether increase the surface \( A \) or increase the heat transfer coefficient \( h \). But in some case it is not possible to increase \( h \) because \( h \) is almost constant (5 to 12 W/m²k) whenever the heat is convected to atmosphere and the temperature difference can not be controlled, in that case the only way is increase the surface area, the surface area is increased by attaching some extra material in the form of rod (circular or rectangular) on the surface where higher heat transfer rate is required. That extra material attached to the surface is called the surface or fins. In order to compensate for low convective heat transfer coefficient in case of gases compared to liquids, the surface area on gas side can be extended by providing fins. The fins are called plane surface fins if they are attached to plane surface, circumferential fins if attached to cylindrical surface. The cross-section of fin may be circular, rectangular, triangular or parabolic.

2. Fin Effectiveness

The aim of providing the fins is to increase the heat transfer rate from a surface area. However, the fin itself blocks the base area of the surface and acts as a resistance to heat conduction. Therefore the heat transfer from the base surface decreases. Thus, one is not sure whether the heat transfer rates will increases or not by providing the fins on the base surface.

It is assessed by a term called fin effectiveness (\( \varepsilon \)). Effectiveness of a fin is defined as the ratio of the heat transfer with fin to heat transfer from the surface without fin.

Generally materials used for fins must have high thermal conductivity. E.g. material like copper, aluminium etc. however the weight also taken into consideration while choosing the materials. Effectiveness increases with increases in (P/A), for this reason thin fins are preferred. Effectiveness increases if the convective heat transfer coefficient ‘h’ is low. For this reason the fins are provided on the gas side having natural convection.

3. Fin Efficiency

The efficiency of a fin is defined as the ratio of the actual heat transferred by the fin to the maximum heat transferred to the fin if the entire fin area is at the same temperature. The efficiency of most fins used in practice is above 80 percent.

V. OBSERVATION AND OBSERVATION TABLES

1. OBSERVATIONS

1) Voltmeter= 57 V.
2) Ammeter= 0.25 A.
3) Manometric height= 16.5 cm.

2. OBSERVATION TABLES

Table 2
Temperature reading in straight duct

<table>
<thead>
<tr>
<th>No. Of Thermocouples</th>
<th>T1</th>
<th>T2</th>
<th>T3</th>
<th>T4</th>
<th>T5</th>
<th>T6</th>
</tr>
</thead>
<tbody>
<tr>
<td>Temperatures In C</td>
<td>52</td>
<td>51.3</td>
<td>50.4</td>
<td>49.9</td>
<td>49.9</td>
<td>37.9</td>
</tr>
</tbody>
</table>

Table 3
Temperature reading in corrugated duct

<table>
<thead>
<tr>
<th>No. Of Thermocouples</th>
<th>T1</th>
<th>T2</th>
<th>T3</th>
<th>T4</th>
<th>T5</th>
<th>T6</th>
</tr>
</thead>
<tbody>
<tr>
<td>Temperatures In C</td>
<td>51.1</td>
<td>50.3</td>
<td>49.6</td>
<td>48.7</td>
<td>48.5</td>
<td>38.3</td>
</tr>
</tbody>
</table>

VI. CALCULATION

A. Calculations For Straight Rectangular Duct

1) Average surface temperature of fin is given by,

\[ T_s = \frac{(T_1 + T_2 + T_3 + T_4 + T_5)}{5} \]

\[ T_s = (52 + 51.3 + 50.4 + 49.9 + 49.9)/5 \]

\[ T_s = 50.7 \ ^\circ C \]
Ts = 323.7 K
T6 = ambient temperature
T6 = 37.9°C
T6 = 310.9 K

2) Mean film temperature

\[ T_{mf} = \text{mean temperature} = \frac{T_s + T_6}{2} \]

\[ T_{mf} = \frac{323.7 + 310.9}{2} \]

\[ T_{mf} = 317.3 \text{ K} \]

Table 4.

Properties of air (Rectangular Duct)

<table>
<thead>
<tr>
<th>Temp (k)</th>
<th>Density(ρ) (kg/ m³)</th>
<th>Kinematic viscosity (ν) (m²/s)</th>
<th>Thermal conductivity of air (k) W/mK</th>
</tr>
</thead>
<tbody>
<tr>
<td>300</td>
<td>1.1614</td>
<td>15.89 x 10⁻⁶</td>
<td>26.3 x 10⁻³</td>
</tr>
<tr>
<td>350</td>
<td>0.995</td>
<td>20.92 x 10⁻⁶</td>
<td>30 x 10⁻³</td>
</tr>
<tr>
<td>317.3</td>
<td>1.1038</td>
<td>17.6303 x 10⁻⁶</td>
<td>27.58 x 10⁻³</td>
</tr>
</tbody>
</table>

By NEWTONS interpolation method

3) Density

\[ \rho_{air} = 1.1614 - 3.328 \times 10^{-3} (T_{mf} - 300) \]

\[ \rho_{air} = 1.1614 - 3.328 \times 10^{-3} (317.3-300) \]

\[ \rho_{air} = 1.1038 \text{ kg/ m}^3 \]

4) Kinematic Viscosity

\[ \nu_{air} = 15.89 \times 10^{-6} + 1.006 \times 10^{-2} (T_{mf} - 300) \]

\[ \nu_{air} = 15.89 \times 10^{-6} + 1.006 \times 10^{-7} (T_{mf} - 300) \]

\[ \nu_{air} = 17.6303 \times 10^{-6} \]

5) Thermal Conductivity

\[ k_{air} = 26.3 \times 10^{-3} + 7.4 \times 10^{-5} (T_{mf} - 300) \]

\[ k_{air} = 27.58 \times 10^{-3} \]

6) Volume flow rate of air

\[ Q_{air} = C \times (\pi/4) \times d_o^2 \times \sqrt{2 \times g \times h_w \times (\rho_w/ \rho_a)} \]

\[ Q_{air} = 0.64 \times (\pi/4) \times (0.02)^2 \times \sqrt{2 \times 9.81 \times 0.165 \times \left(\frac{1000}{1.1038}\right)} \]

\[ Q_{air} = 10.88 \times 10^{-3} \text{ m}^3/\text{s} \]

Where β = 0.39

d_o is the diameter of orifice = 0.02m

Coefficient of discharge (C) = 0.64

Density of Water (ρ_w) = 1000 Kg/m³

Density of air (ρ_a) = 1.1038Kg/m³

7) Velocity of air in the duct (V)

\[ V_{air} = Q_{air} / (A_{cs}) \]

\[ V_{air} = (8.81988 \times 10^{-3}) / (0.15 \times 0.1) \]

\[ V_{air} = 0.5879 \text{ m/s} \]

Where,

A_cs = cross sectional area of duct

W = height of the duct = 0.15m

B = width of the duct = 0.1m

8) Velocity of air at T_{mf}

\[ V_{Tmf} = \frac{T_{Tmf}}{T_{air}} \times V_{air} \]

\[ V_{Tmf} = \left[\frac{T_{mf}}{T_{air}}\right] \times V_{air} \]

\[ V_{Tmf} = [317.3/310.9] \times 0.5879 \]

\[ V_{Tmf} = 0.60 \text{ m/s} \]

9) Reynolds number

\[ Re = \frac{(V_{Tmf} \times L_2)}{\nu_{air}} \]

\[ Re = \frac{(0.60 \times 12.7 \times 10^{-3})}{17.6303 \times 10^{-6}} \]
Re = 432.27

10) Nusselt number

\[
(Nu) = 0.615(Re^{0.466})
\]

But, \( Nu = \frac{h \times L_c}{K_{air}} \)

Where,

\[ h \] is heat transfer coefficient
\[ L_c \] = diameter of pin fin
\[ h = \frac{(Nu \times K_{air})}{L_c} \]

\[ h = \frac{(10.4024 \times 5\times 10^{-2})}{12.7 \times 10^{-3}} \]

\[ h = 22.59 \text{ W/m K} \]

\[ m = \frac{hP}{\sqrt{KA_{cs}}} \]

\[ m = \frac{22.59 \times 4}{\sqrt{204 + 12.7 \times 10^{-3}}} \]

\[ m = 5.9767 \]

11) Heat transfer rate

\[ Q_{fin} = \sqrt{hP \times K_{cs} \times (T_o - T_a)} \times \tanh(ml) \]

\[ Q_{fin} = 1.363 \]

12) Fin efficiency

\[ \eta_{fin} = \frac{(\tanh(ml))}{ml} \]

\[ \eta_{fin} = 0.8478 \]

\[ \eta_{fin} = 84.78\% \]

13) Fin effectiveness

\[ \varepsilon = \frac{(\eta \times P \times l)}{(A_{cs})} \]

\[ \varepsilon = \frac{(0.8478 \times 4 \times 0.125)}{(12.7 \times 10^{-3})} \]

\[ \varepsilon = 33.37 \]

\[ B. \text{ Calculations For Corrugated Duct} \]

1) Average surface temperature of fin is given by,

\[ T_s = \frac{(T_1 + T_2 + T_3 + T_4 + T_5)/5}{\text{Temp. Distribution of A Fin Along A Length For Insulated Tip}} \]

i) \( \text{At } x = 0, T = T_1 \)

\[ \frac{T_1 - T_a}{T_0 - T_a} = \frac{\cosh(m(0-x))}{\cosh(ml)} \]

\[ T_1 = 51.1^\circ C \]

ii) \( \text{At } x = 30 \times 10^{-3}, T = T_2 \)

\[ \frac{T_2 - T_a}{T_0 - T_a} = \frac{\cosh(5.9767(0.125 - 0.03))}{\cosh(5.9767 + 0.125)} \]

\[ T_2 = 50.61^\circ C \]

iii) \( \text{At } x = 60 \times 10^{-3}, T = T_3 \)

\[ \frac{T_3 - T_a}{T_0 - T_a} = \frac{\cosh(5.9767(0.125 - 0.06))}{\cosh(5.9767 + 0.125)} \]

\[ T_3 = 49.64^\circ C \]

iv) \( \text{At } x = 90 \times 10^{-3}, T = T_4 \)

\[ \frac{T_4 - T_a}{T_0 - T_a} = \frac{\cosh(5.9767(0.125 - 0.09))}{\cosh(5.9767 + 0.125)} \]

\[ T_4 = 49.05^\circ C \]

v) \( \text{At } x = 120 \times 10^{-3}, T = T_5 \)

\[ \frac{T_5 - T_a}{T_0 - T_a} = \frac{\cosh(5.9767(0.125 - 0.12))}{\cosh(5.9767 + 0.125)} \]

\[ T_5 = 48.81^\circ C \]
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\[ T_s = \left( \frac{51.1+50.3+49.6+48.7+48.5+38.3}{5} \right) \]

\[ T_s = 49.64 \quad ^\circ C \]

\[ T_s = 322.64 \quad K \]

\[ T_6 = \text{ambient temperature} \]

\[ T_6 = 38.3 \quad ^\circ C \]

\[ T_6 = 311.7 \quad K \]

2) **Mean film Temp.**

\[ T_{mf} = \frac{(T_s + T_6)}{2} \]

\[ T_{mf} = \frac{322.64+38.3}{2} \]

\[ T_{mf} = 316.97 \quad K \]

**Table 5. Properties of air (Corrugated Duct)**

<table>
<thead>
<tr>
<th>Temp (k)</th>
<th>Density((\rho)) (kg/ m(^3))</th>
<th>Kinematic viscosity ((v)) (m/s(^2))</th>
<th>Thermal conductivity of air ((k)) (W/mK)</th>
</tr>
</thead>
<tbody>
<tr>
<td>300</td>
<td>1.1614</td>
<td>15.89 x 10(^{-6})</td>
<td>26.3 x 10(^{-3})</td>
</tr>
<tr>
<td>350</td>
<td>0.995</td>
<td>20.92 x 10(^{-6})</td>
<td>30 x 10(^{-3})</td>
</tr>
<tr>
<td>316.97</td>
<td>1.1049</td>
<td>17.59 x 10(^{-6})</td>
<td>27.56 x 10(^{-3})</td>
</tr>
</tbody>
</table>

By NEWTONS interpolation method

3) **Density of air**

\[ \rho_{air} = 1.1614 - 3.328 x 10^{-3} \quad (T_{mf} - 300) \]

\[ \rho_{air} = 1.1614 - 3.328 x 10^{-3} \quad (316.97-300) \]

\[ \rho_{air} = 1.1049 \quad \text{kg/ m}^3 \]

4) **Kinematic Viscosity**

\[ v_{air} = 15.89 x 10^{-6} + 1.006 x 10^{-7} \quad (T_{mf} - 300) \]

\[ v_{air} = 15.89 x 10^{-6} + 1.006 x 10^{-7} \quad (316.97-300) \]

\[ v_{air} = 17.59 x 10^{-6} \]

5) **Thermal Conductivity**

\[ k_{air} = 26.3 x 10^{-3} + 7.4 x 10^{-5} \quad (T_{mf} - 300) \]

\[ k_{air} = 26.3 x 10^{-3} + 7.4 x 10^{-5} \quad (316.97-300) \]

\[ k_{air} = 27.56 x 10^{-3} \]

6) **Volume flow rate of air**

\[ Q_{air} = C \cdot \frac{d}{2} \cdot \sqrt{g \cdot h_{w} \cdot \left( \frac{\rho_{w}}{\rho_{air}} \right)} \]

\[ Q_{air} = 0.64 \pi (0.02) \cdot \sqrt{2 \cdot \frac{3}{2} \cdot \frac{26.3}{27.56} \cdot \frac{1000}{1.1049}} \]

\[ Q_{air} = 10.88 \times 10^{-3} \quad \text{m}^3/\text{s} \]

Where,

\[ \beta = 0.39 \]

\(d\) = diameter of orifice = 0.02m

Coefficient of discharge (\(C\)) = 0.64

Density of water (\(\rho_{w}\)) = 1000 Kg/m\(^3\)

Density of air (\(\rho_{air}\)) = 1.1049 Kg/m\(^3\)

7) **Velocity of air in the duct (V)**

\[ V_{air} = \frac{Q_{air}}{A_{cs}} \]

\[ V_{air} = \frac{10.88 \times 10^{-3}}{0.15 \times 0.1} \]

\[ V_{air} = 0.7255 \quad \text{m/s} \]

Where,

\(A_{cs}\) = cross sectional area of duct

\(W\) = height of the duct = 0.15m

\(B\) = width of the duct = 0.1m

8) **Velocity of air at T_{mf}**

\[ \frac{V_{Tmf}}{V_{air}} = T_{Tmf} / T_{air} \]

\[ V_{Tmf} = \left[ \frac{316.97}{38.3} \right] \times 0.7255 \]

\[ V_{Tmf} = 0.7387 \quad \text{m/s} \]
9) Reynolds number

\[ Re = \frac{(V_T m \cdot L_c)/v_{\text{air}}}{(0.7387 \cdot 12.7 \cdot 10^{-3})/17.59 \cdot 10^{-6}} \]

\[ Re = 533.39 \]

10) Nusselt number

\[ (Nu) = 0.615(Re^{0.466}) \]

\[ (Nu) = 0.615(533.39^{0.466}) \]

\[ (Nu) = 11.4730 \]

But,

\[ Nu = (h \cdot L_c)/(h_{\text{air}}) \]

Where,

- \( h \) is heat transfer coefficient (W/m²k)
- \( L_c \) = diameter of pin fin

\[ h = (Nu \cdot L_c)/(h_{\text{air}}) \]

\[ h = \left[ \frac{(1.4730 \cdot 27.56 \cdot 10^{-3})}{12.7 \cdot 10^{-3}} \right] \]

\[ h = 24.89 \text{ W/m²K} \]

11) Heat transfer rate

\[ Q_{\text{fin}} = \sqrt{(h \cdot \rho \cdot C_v / \theta)} \cdot x \cdot (T_\text{o} - T_\text{a}) \cdot \tanh(ml) \]

\[ = \sqrt{(24.89 \cdot \pi \cdot 12.7 \cdot 10^{-3} \cdot 204 \cdot (\pi/4) \cdot (12.7 \cdot 10^{-3})^2} \]

\[ x \cdot (51.1-38.3) \cdot \tanh(6.199 \cdot 0.125) \]

\[ Q_{\text{fin}} = 1.332 \]

12) Fin efficiency

\[ \eta_{\text{fin}} = \tanh(ml)/ml \]

\[ = \tanh(6.199 \cdot 0.125) \]

\[ (6.199 + 0.125) \]

\[ \eta_{\text{fin}} = 83.85 \% \]

13) Fin effectiveness

\[ \epsilon = \frac{(\eta \cdot p \cdot l) / A_{\text{cs}}}{(0.8478 + 0.125)} \]

\[ \epsilon = (12.7 \cdot 10^{-3}) \]

\[ = 33.37 \]

Temp. Distribution of a Fin Along a Length For Insulated Tip

i) At \( x = 0 \), \( T = T_1 \)

Let,

\[ \frac{T_1 - T_\text{a}}{T_0 - T_\text{a}} = 1 \]

\[ T = T_1 \]

\[ T_1 = 51.1 \text{ °C} \]

ii) At \( x = 30 \times 10^{-3} \), \( T = T_2 \)

\[ \frac{T_2 - T_\text{a}}{T_0 - T_\text{a}} = \frac{\cosh(6.199 \cdot 0.125 \cdot 0.03)}{\cosh(6.199 \cdot 0.125)} \]

\[ = 0.8958 \]

\[ T_2 = 49.76 \text{ °C} \]

iii) At \( x = 60 \times 10^{-3} \), \( T = T_3 \)

\[ \frac{T_3 - T_\text{a}}{T_0 - T_\text{a}} = \frac{\cosh(6.199 \cdot 0.125 \cdot 0.06)}{\cosh(6.199 \cdot 0.125)} \]

\[ = 0.82269 \]

\[ T_3 = 48.83 \]

iv) At \( x = 90 \times 10^{-3} \), \( T = T_4 \)

\[ \frac{T_4 - T_\text{a}}{T_0 - T_\text{a}} = \frac{\cosh(6.199 \cdot 0.125 \cdot 0.09)}{\cosh(6.199 \cdot 0.125)} \]

\[ = 0.7781 \]
2. 

**VII. RESULTS AND GRAPHS**

1. **Result Table**

**Table 6. Properties of air w.r.t temp.**

<table>
<thead>
<tr>
<th></th>
<th>T_s (°C)</th>
<th>T_mf (°C)</th>
<th>ρ_air (kg/m³)</th>
<th>ν_air (m²/s)</th>
<th>k_air (W/m K)</th>
<th>V_Tmf (m/s)</th>
</tr>
</thead>
<tbody>
<tr>
<td>RECTANGULAR DUCT</td>
<td>50.7</td>
<td>44.3</td>
<td>1.103</td>
<td>17.63 x 10⁻⁶</td>
<td>27.58 x 10⁻³</td>
<td>0.589</td>
</tr>
<tr>
<td>CORRUGATED DUCT</td>
<td>49.6</td>
<td>43.9</td>
<td>1.104</td>
<td>17.59 x 10⁻⁶</td>
<td>27.56 x 10⁻³</td>
<td>0.738</td>
</tr>
</tbody>
</table>

**Table 7. Heat transfer parameters**

<table>
<thead>
<tr>
<th></th>
<th>Re</th>
<th>Nu</th>
<th>h (W/m² K)</th>
<th>m</th>
<th>Q (W)</th>
<th>η (%)</th>
<th>ε</th>
</tr>
</thead>
<tbody>
<tr>
<td>RECTANGULAR DUCT</td>
<td>432.27</td>
<td>10.40</td>
<td>22.59</td>
<td>5.97</td>
<td>1.36</td>
<td>84.8</td>
<td>33.4</td>
</tr>
<tr>
<td>CORRUGATED DUCT</td>
<td>533.39</td>
<td>11.47</td>
<td>24.89</td>
<td>6.19</td>
<td>1.33</td>
<td>83.8</td>
<td>33.1</td>
</tr>
</tbody>
</table>

**Table 8. Comparison of temp. Distribution**

<table>
<thead>
<tr>
<th>TEMP.</th>
<th>RECTANGULAR DUCT</th>
<th>CORRUGATED DUCT</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>EXPERIMENTAL</td>
<td>CALCULATED</td>
</tr>
<tr>
<td>T₁</td>
<td>52</td>
<td>52</td>
</tr>
<tr>
<td>T₂</td>
<td>51.3</td>
<td>50.61</td>
</tr>
<tr>
<td>T₃</td>
<td>50.4</td>
<td>49.64</td>
</tr>
<tr>
<td>T₄</td>
<td>49</td>
<td>49.05</td>
</tr>
<tr>
<td>T₅</td>
<td>49.9</td>
<td>48.81</td>
</tr>
</tbody>
</table>

2. **GRAPHS**

From graph number.1 and .2 we have seen that, in rectangular duct there are large changes in experimental and calculated value of temp. along a pin fin but in corrugated duct experimental values deviated from theoretical values by small differences.
VIII. CONCLUSION

Corrugated channels play a great role in heating and cooling systems. In the duct system more scope of heat transfer enhancement. We are determining Heat transfer coefficients are experimentally for different corrugation angles and their results are compared with rectangular duct. For higher corrugation angle, the heat transfer rates are higher due to the generation of higher turbulence. By using pin fin we come to know that, in forced convection rate of heat transfer in corrugated duct is greater than heat transfer rate in straight duct. Our aim for providing corrugation angle is satisfied as coefficient of heat transfer increases as compared to straight duct. We are also observing impact of fin on the coefficient of heat transfer. The efficiency of pin fin is also differs from straight duct. Efficiency is higher in natural convection & less in forced convection, because of values of ‘h’ is higher in forced convection. There is no heat loss at the end of pin because the tip of fin pin is insulated.

REFERENCES


