CHAPTER – 6

CALIBRATION OF HEAT SOURCE

6.1 INTRODUCTION

The calibration of oxy-acetylene torch is essential because it is used as the heat source for the present sweat-cooling experiment. It is also required to calibrate the heat source inorder to estimate the heat input value over the flat vertical surface, which is cooled by gaseous nitrogen. The following are the known parameters of the system:

(i) The distance between the flat surface of the specimen and the oxy-acetylene flame.
(ii) The rate of flow of the fuel gas and the oxidiser (acetylene and oxygen)

6.2 DESCRIPTION

A water cooled thin foil heat-flux sensor [92] is used to calibrate the heat source. The sensor can be used for long duration and for high heat-flux measurement application. A detailed cross-sectional view of the sensor is illustrated in Fig.6.1. The central part of the sensor is directly exposed to the thermal environment similar to that of an un-cooled sensor. It consists of a thin constantan foil, which is bonded at its periphery to the copper body. The constantan foil is coated with a high absorptivity paint. A water cooling jacket is provided in the body through which water is circulated so that the body is always kept at room temperature. Copper wires are brazed at the centre and edges of the foil thus forming a copper constantan differential thermocouple. Photo-plate 6.1 illustrates heat-flux sensor described above.

6.3 PRINCIPLE OF OPERATION

When sensing part of the sensor is exposed to a convective and radiative thermal environment, heat energy absorbed by the foil is radially transferred to the heat sink
Photo-plate 6.2 Heat-flux sensor used for the calibration of heat source (Oxy-Acetylene flame)

Photo-plate 6.1 Calibration of oxy-acetylene flame (Heat source)
CALIBRATION CURVES FOR COOLED HEAT FLUX SENSORS

b) RESPONSE CURVE (TYPICAL COOLED SENSOR)

c) A TYPICAL COOLED HEAT FLUX SENSOR

\[ C_1 = 16.4 \, \text{W/cm}^2/\text{mV} \]
\[ C_2 = 19.1 \, \text{W/cm}^2/\text{mV} \]
consisting of the copper body and the cooling water in the jacket. Thus the sensor body as well as the foil edge is always kept at or near the ambient temperature, where as the temperature at the centre of the foil rises. The copper constantan differential thermocouple measures the temperature difference between the centre and edge of the foil [89]. For an uncooled sensor it can be shown that the rate of heat-transfer to the foil is given by

\[ q = CE \]

Where

\[ C = \text{A constant and } E \text{ is thermo EMF.} \]

Specifications:

- Rated heat-flux: 150 Wcm\(^{-2}\)
- Accuracy: 5%
- Maximum flow rate of coolant: 3 litres per minute

6.4 CALIBRATION PROCEDURE

The heat-flux sensor is mounted on the test bench. The Chrome-Alumal thermocouple leads are connected to a 12 channel IWATZU, SC 750 high speed electronic datalogger, which gives the output in digital printout. The cooling water is supplied from an overhead storage tank to impart better pressure in the supply of coolant. A control valve and a regulator are incorporated in the water flow pipe to control the flow of cooling water. The maximum cooling water flow requirement is 3 liters per minute. The experiment was carried out by varying the distance between the heat-flux sensor and the oxy-acetylene flame. For all these runs, the rate of flow of oxygen and acetylene through the nozzle of the torch is measured by a calibrated orifice flow meter. Throughout the experiment the total gas flow rate amounted to approximately 2.7 x 10\(^{-3}\) kg s\(^{-1}\) at most efficient oxidiser fuel (O/F) mixture ratio with equal pressures. The experiment was repeated several runs to check the consistency. The average heat flux for each standard distance is calculated and the results are presented in table 6.1. Photo-plate 6.2 shows the experimental procedure for the calibration of heat source using heat-flux sensor.
Table 6.1
HEAT FLUX VALUES

<table>
<thead>
<tr>
<th>Distance in centimeters</th>
<th>Voltage generated in milli volts</th>
<th>Heat - Flux W cm⁻²</th>
</tr>
</thead>
<tbody>
<tr>
<td>05</td>
<td>1.26</td>
<td>79.23</td>
</tr>
<tr>
<td>06</td>
<td>1.13</td>
<td>67.85</td>
</tr>
<tr>
<td>10</td>
<td>0.70</td>
<td>42.47</td>
</tr>
<tr>
<td>12</td>
<td>0.48</td>
<td>28.31</td>
</tr>
</tbody>
</table>

6.5 EVALUATION OF HEAT - TRANSFER COEFFICIENT

For the present analysis the heat-flux has been taken as 28.31 x 10⁴ Wm⁻² which is one of the heat-flux sensor calibrated value (Table 6.1). The first information required by a designer, is the distribution of surface heat-flux, in the absence of any transpiration cooling. This data provides the base line for the thermal design [93]. A common heat-transfer problem involving both conduction and convection is the transfer of heat flow from a flowing fluid (Liquid/Gas) to a flat porous wall, and through the wall to the atmosphere.

A flow system designed with water as coolant flowing through a conduit was permitted to impinge at one face of the porous specimen of 40%, 32% & 28% porosities. The total thickness of the specimen is 1.2 mm as shown in Fig. 6.2. It is made out of 0.6 mm diameter copper wire meshes of two layers brazed in vacuum. The copper is selected because of its higher thermal conductivity.

Two numbers of chrome - alumel thermocouples are installed at each surfaces, that is at hot flame side and coolant side to monitor the surface temperatures when heating and cooling operations are in a steady state condition. One thermocouple on flame side and another one on coolant side is installed to monitor the free stream
All dimensions are in millimeter

Fig.6.2 SPECIMEN FOR HEAT TRANSFER COEFFICIENT EVALUATION
temperature of the flowing coolant. This temperature instrumentation with test setup is illustrated in Fig. 6.3 which is convective and radiative environment similar to the combustion chamber environment in a cryogenic rocket engine.

Calibrated oxy-acetylene torch is used as heat source for the experiment. When cooling water comes in contact with the hot porous stationary specimen wall, a thin boundary layer is developed adjacent to the wall, and in this layer there is no relative velocity with respect to the wall. The porous solid surface is warmer than the coolant, the heat is being transferred from solid surface to the coolant. Heat flow across the stagnant layer is caused by conduction and convection [93]. Since thermal conductivity of water is much lower, the heat flow from the fluid to the wall is mainly due to convection mechanism.

As presented by Sir. Isaac Newton, the convective heat-transfer coefficient, 'h' is a unit conductance used for calculation of convection heat transfer. To describe quantitatively the cooling of objects in air, Newton suggested a law of cooling, a definition for 'h'[93]. Heat transfer coefficient is some function of the following properties : \( h_f \) \((k, C_p, \rho, \mu, V, d)\). The rate of heat flow across solid/liquid interface would be expected to depend on the area of interface and the temperature drop between the liquid and the solid or the heat flux as the products of heat transfer coefficient and the driving temperature, usually the difference between the wall and the free stream temperature of the cooling fluid [94].

\[
q = hA \left( T_{\text{surface}} - T_{\text{fluid}} \right)
\]

(6.1)

Where

\begin{align*}
q &= \text{Local rate of heat-transfer per unit area in watts} \\
A &= \text{Area of cross section of specimen in m}^2 \\
T_{\text{surface}} &= \text{Specimen surface temperature in K,} \\
T_{\text{fluid}} &= \text{The temperature of the flowing fluid in K} \\
h &= \text{The factor of proportionality i.e., the local heat-transfer coefficient}
\end{align*}
Therefore the transfer of heat by convection is a function of the specific surface coefficient, $h$ [52, 95, 96]. This in turn depends on the physical properties of the liquid. In the present study, aerodynamic heating is neglected.

6.5.1 EXPERIMENTAL EVALUATION OF HEAT-TRANSFER COEFFICIENT

Heat transfer coefficient, 
\[
q/A = \frac{h}{T_{\text{surface}} - T_{\text{fluid}}}
\]

Engineering units are Wm$^{-2}$ K$^{-1}$. All of the temperatures are to be measured in absolute units.

The specimen is mounted in the Test bench as shown in Fig.6.3. The cooling water flow through the orifice is regulated at different rates by adjusting the pressure through a pressure regulator. The calibrated oxy-acetylene flame is directed to the front face of the specimen. The heating of the front face and cooling in the opposite face through water pipes is continued for a long time till a steady state heat flow through the specimen is attained. The temperature on both flame side and the coolant side is recorded by chrome-alumel thermocouple with a precision datalogger and as shown in Table-6.2. The temperature of the cooling water leaving the specimen surface immediately near the laminar sub-layer is also recorded by precision industrial thermocouples. By varying the heat flux and the mass flow rate, the experiment is repeated several times and respective heat-transfer coefficient is evaluated.

<table>
<thead>
<tr>
<th>Sl. No.</th>
<th>Specification</th>
<th>Temperature</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Flame Side Wall - $T_{fw}$</td>
<td>793</td>
</tr>
<tr>
<td>2</td>
<td>Free Stream Fluid - $T_{fm}$</td>
<td>639</td>
</tr>
<tr>
<td>3</td>
<td>Coolant Side Wall - $T_{cw}$</td>
<td>636</td>
</tr>
<tr>
<td>4</td>
<td>Free Stream fluid - $T_{cw}$</td>
<td>323</td>
</tr>
</tbody>
</table>

(For corresponding heat-flux values please refer Table 6.1)

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Fig. 6.3 EXPERIMENTAL SET UP TO EVALUATE HEAT-TRANSFER COEFFICIENT.
6.5.2. CALCULATION OF HEAT-TRANSFER COEFFICIENT IN FLAME SIDE, $H_F$

Flame side heat-transfer coefficient, 
$$h_f = \frac{q}{A} \frac{T_{fw} - T_{f\infty}}{T_{lw} - T_{1'o}}$$ (6.3)

Where

$T_{fw}$ - is the flame side temperature in the porous wall surface

$T_{f\infty}$ - is the temperature of the free stream cooling fluid in the flame side

Substituting

$$h_f = \frac{28.31 \times 10^4}{793 - 639} = 1838.31 \text{ Wm}^{-2}\text{K}^{-1}$$

6.5.3 HEAT-TRANSFER COEFFICIENT IN COOLANT SIDE, $h_c$

Coolant side heat-transfer coefficient,

$$h_c = \frac{q}{A} \frac{T_{cw} - T_{f\infty}}{T_{cw} - T_{f\infty}}$$ (6.4)

Where $T_{cw}$ - is the temperature in the coolant side wall

$T_{f\infty}$ - is the temperature of the Free Stream cooling fluid in the coolant side.

$$h_c = \frac{28.31 \times 10^4}{336 - 323} = 904.47 \text{ Wm}^{-2}\text{K}^{-1}$$