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# Performance Improvement Of Self-Aspirating Porous Radiant Burner By Controlling Process Parameters

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**ABSTRACT**
This paper presents the heat transfer characteristics of a self-aspirating porous radiant burner (SAPRB) that operates on the basis of an effective energy conversion method between flowing gas enthalpy and thermal radiation. The temperature field at various flame zones was measured experimentally by the help of both FLUKE IR camera and K-type thermocouples. The experimental setup consisted of a two layered domestic cooking burner, a flexible test stand attached with six K-type thermocouples at different positions, IR camera, LPG setup and a hot wire anemometer. The two layered SAPRB consisted of a combustion zone and a preheating zone. Time dependent temperature history from thermocouples at various flame zones were acquired by using a data acquisition system and the temperature profiles were analyzed in the ZAILA application software environments. In the other hand the IR graphs were captured by FLUKE IR camera and the thermographs were analyzed in the SMARTView software environments. The experimental results revealed that the homogeneous porous media, in addition to its convective heat exchange with the gas, might absorb, emit, and scatter thermal radiation. The rate of heat transfer was more at the center of the burner where a combined effect of both convection & radiation might be realized. The maximum thermal efficiency was found to be 64% which was having a good agreement with the previous data in the open literature.
***Keywords*** - IR Thermography, LPG, Self-Aspirating Porous Radiant Burner (SAPRB), Thermal Efficiency, Transient Temperature.

## Nomenclature

| | |
|---|---|
| $A_V$ | Surface Area Density |
| V | Gas Velocity, m/s |
| $\emptyset_f$ | Porosity |
| d | Hydraulic Diameter, m |
| $N_u$ | Local Nusselt Number |
| $R_e$ | Reynolds Number |
| H | Local Heat Transfer Co-efficient, $W/m^2k$ |
| ρ | Mass Density of LPG, $kg/m^3$ |
| $m_w$ | Mass of Water, kg |
| $m_v$ | Mass of Vessel, kg |
| $m_f$ | Mass of Fuel, kg |
| $C_{pw}$ | Specific heat of aluminum, kJ/kg K |
| $C_{pv}$ | Specific heat of aluminum, kJ/kg K |
| CV | Calorific value of the fuel, kJ/kg |
| $T_1$ | Initial Temperature of Water, °C |
| $T_2$ | Final Temperature of Water, °C |

## I. Introduction

The heating systems used for domestic purposes that operate on free flame, leads to relatively low thermal efficiency. A lot of previous works are available on the use of porous media in direction of increasing thermal efficiency. Energy conversion aspects between flowing gas enthalpy and thermal radiation by using porous medium becomes an interesting approach for improving the overall performance of these systems. Proper parametric control may lead to better enhancement of burner performance. In India, LPG (liquefied petroleum gas) is the most commonly used conventional fuel [1].So as it is common in other Asian countries also. As the economy of the country is getting better day by day so as the standard of living of the people. The population of the India is approximately 1.25 billion so as the LPG market is very huge. Government of India is spending huge amount of money for subsidizing the domestic LPG cylinder price [2]. As it is a burden on the Government & ultimately on the people, different ways should be invented to increase the thermal efficiency of the conventional burners. N.K. Mishra *et al.* [3] has tested Medium-Scale (5-10 kW) Porous Radiant Burners for LPG Cooking Applications with a good effect. He found the PRB with 5kw thermal load yielded the maximum thermal efficiency of about 50%, which is 25% higher than the efficiency of the conventional burner also the emission are much lower than the conventional burner. V.K Pantangi *et al.* [4] has studied the porous radiant burner for LPG cooking application. In his study he found that the maximum thermal efficiency of PRB is found to be 68% which is 3% higher than the conventional burner with less CO & $NO_X$ emission. P.Muthukumar & P.I Shyamkumar [5] developed a novel porous radiant burner for LPG cooking applications. They have tested PRB having different porosity with different equivalence ratio &

wattages. The reported maximum thermal efficiency is about 75% which is 10% higher than the conventional burners. S.B Sathe *et al.* [6] has studied both theoretically & experimentally the thermal performance of the PRB. The results indicate that stable combustion at elevated flame speeds can be maintained in two different spatial domains: one spanning the upstream half of the porous region and the other in a narrow region near the exit plane. The heat release and radiant output are also found to increase as the flame is shifted toward the middle of the porous layer. Akbari et al. [7] carried out a study to investigate the lean flammability limits of the burner and the unstable flash-back/blow-out phenomena. Hayashi et al. [8] presented a three-dimensional numerical study of a two-layer porous burner for household applications. They solved the mathematical model using CFD techniques which accounted for radiative heat transport in the solid, convective heat exchange between solid and fluid. Talukdar *et al.* [9] presented the heat transfer analysis of a 2-D rectangular porous radiant burner. Combustion in the porous medium was modeled as a spatially-dependent heat generation zone. A 2-D rectangular porous burner was investigated by Mishra *et al.* [10]. Methane–air combustion with detailed chemical kinetics was used to model the combustion part. S.W Cho *et al.* [11] has experimentally studied the performance optimization of a radiant burner with a surface flame structure. The experiment was conducted in three different firing rates (80.5, 107.4 and 134.2 $kW/m^2$) and different equivalence ratios ranged from 0.6 to 1.3. In his study the firing rate of 107.4 $kW/m^2$ was regarded as an optimal condition and water-boiling efficiency was found to be 40%.Sharma *et al.* [12] investigated the effect of porous radiant inserts in conventional kerosene pressure stove. The found that using porous insert, the efficiency of the conventional stove increases from 55% to 62%. Sharma *et al.* [13] modified conventional kerosene pressure stove. They used ceramic ($ZrO_2$) insert in the combustion zone and a ceramic ($Al_2O_3$) heat shield surrounding the burner. The maximum efficiency was found to be 70%, which was 15% higher than the efficiency of a conventional kerosene pressure stove. S. Wood and A.T Harris [14] have studied on porous burners for lean burn applications. In this paper, we attempted to find out thermal performance of a self-aspirant porous radiant burner by controlling its process parameters.

## II. Experimental Set Up and Procedure

A schematic of the experimental set-up used for testing the performance of PRB is shown in Fig.1. The experimental setup is a flexible one and it consists of a LPG setup, a hot wire anemometer, a burner, 6 k-type thermocouple, a data acquisition system, an IR camera and a computer. The LPG setup supplies the fuel through a Teflon pipe of 0.6 mm diameter. The gas passes through the hot wire anemometer with suitable control valves which measures the flowing gas velocities. The air and fuel mixtures are tested at different velocities. The two layered SAPRB consists of a preheating zone & a combustion zone. Combustion zone was formed with high porosity, highly radiating porous matrix, and the preheating zone consisted of low porosity matrix. Three k-type thermocouples as thermocouple-1, thermocouple-2 and thermocouple-3 are arranged in vertical manner i.e. respectively 28mm, 64mm and 115 mm away from the surface of the burner and the other three thermocouples as thermocouple-4, thermocouple-5 and thermocouple-6 are arranged in horizontal manner i.e. respectively 12mm, 35mm and 25mm away from the center of the burner, which is having a radius of 38mm. All the thermocouples are connected to the data acquisition system which is also connected to a computer for live data analysis. In the meantime all the radiative temperatures of the surface and the flame is being measured by the Fluke IR camera. The temperature profiles of data acquisition system were analyzed in the ZAILA application software environments. In the other hand the thermographs were analyzed in the SMARTView software environments. The whole experimental procedure has been conducted in a very controlled condition. The gas velocity was measured by the hot wire anemometer. It was positioned just at the exit of the nozzle.

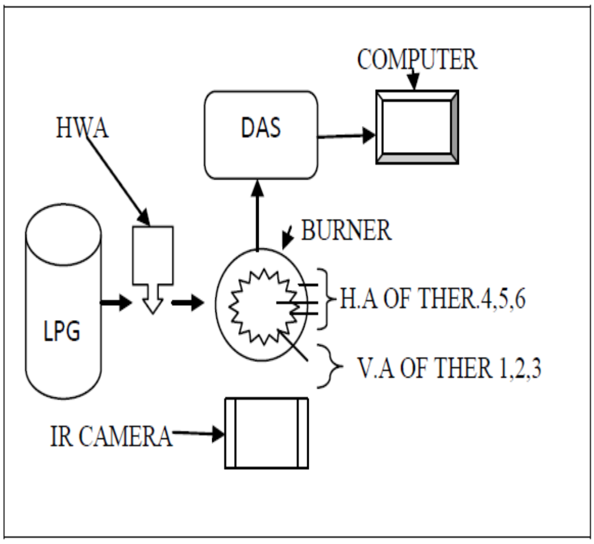


Figure.1. Schematic of the experimental setup

A sample of thermocouple arrangement on the plate is shown in the following.

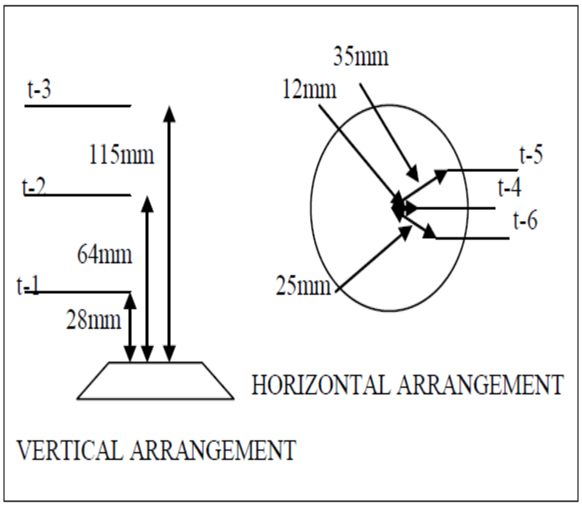


Figure.2: Sketch of thermocouple arrangement

The experimented was conducted in three different velocities viz. 3.6 m/s, 3.0 m/s and 0.4 m/s. Then the temperature data are measured by the thermocouple which has been arranged both in vertically and horizontally on the top surface of the burner. The temperature data are assessed by the data acquisition system. The temperature data are assessed by the data acquisition system. Also the surface and flame temperature are measured by IR camera. Then the heat transfer co-efficient h, was calculated by the following co-relation [15].

$$N_u = 0.95 \mathrm{R_e}^{0.35} \quad (1)$$

$$\mathrm{Re} = \frac{\rho \varphi_f V d}{\mu} \quad (2)$$

## III. Results and Discussion

We have conduct the experiment in three different gas velocities i.e. 3.6 m/s, 3.0 m/s, 0.4 m/s. So the time dependent temperature graphs are mentioned in the Fig.3-14. Also the IR thermograph is shown in Fig. 17.

### 3.1 Effect of Heat Transfer at Maximum Gas Velocity

Fig.3 and Fig.4 shows the time dependent temperature histories of the vertical and horizontal arranged thermocouples respectively at gas velocity 3.6 m/s. From the Fig.3 the maximum temperature attained is 1247°C by thermocouple-3.

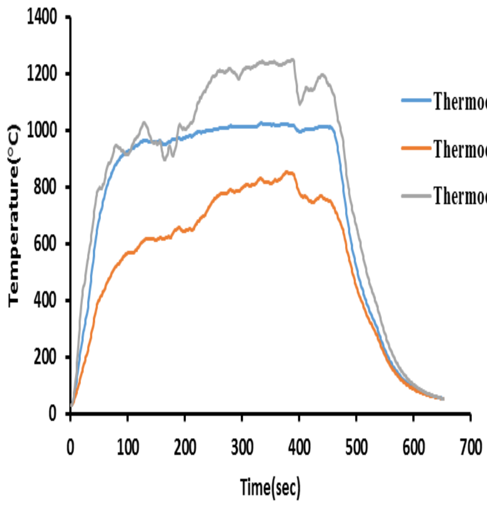


Figure-3. Time dependent temperature distribution of burner (vertical at v= 3.6 m/s)

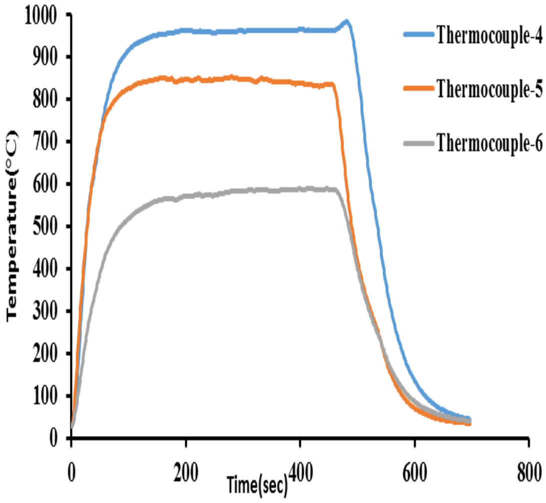


Figure-4. Time dependent temperature distribution of burner (horizontal at v= 3.6 m/s)

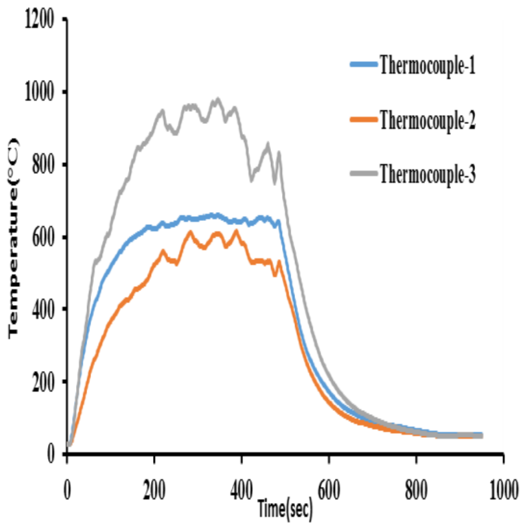


Figure-5. Time dependent temperature distribution of burner (vertical at v = 3.0 m/s).

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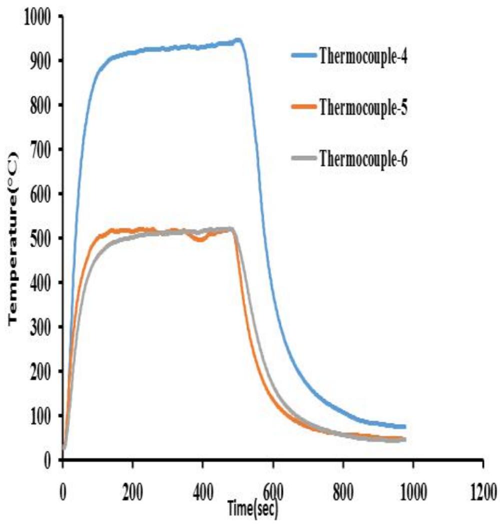


Figure-6. Time dependent temperature distribution of burner (horizontal at v= 3.0 m/s)

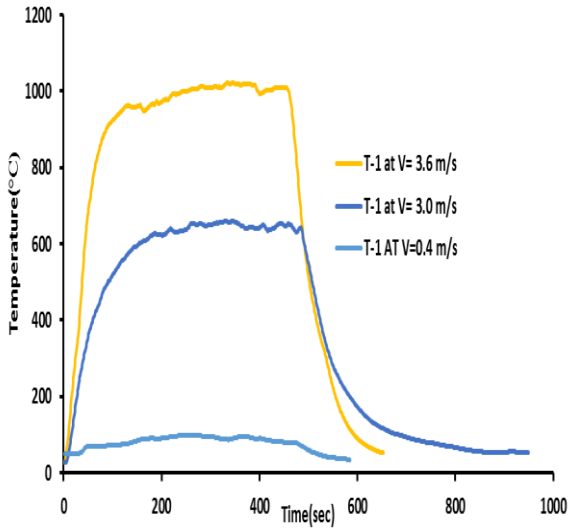


Figure-9. Temperature-time graph of t-1(vertical) at variable gas velocity

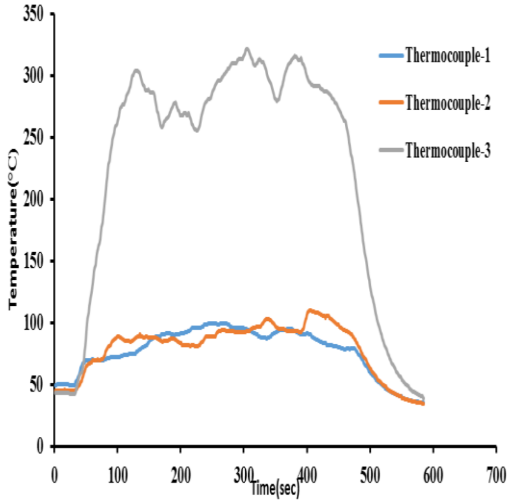


Figure-7. Time dependent temperature distribution of burner (vertical at v= 0.4 m/s)

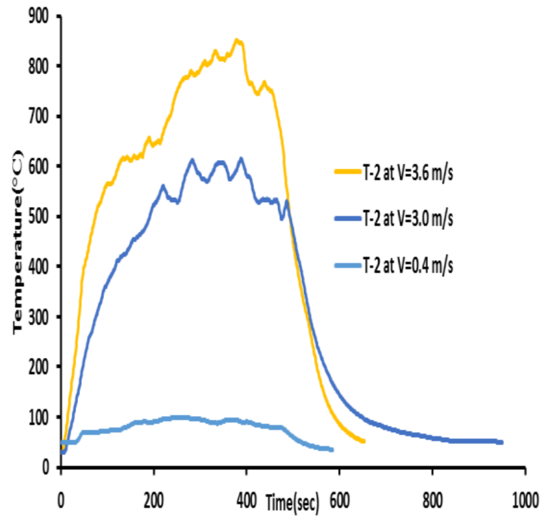


Figure-10. Temperature-time graph of t-2(vertical) at variable gas velocity

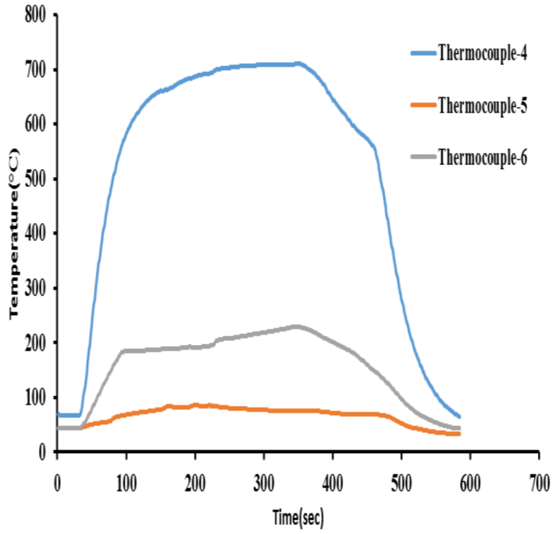


Figure-8. Time dependent temperature distribution of burner (horizontal at v= 0.4 m/s)

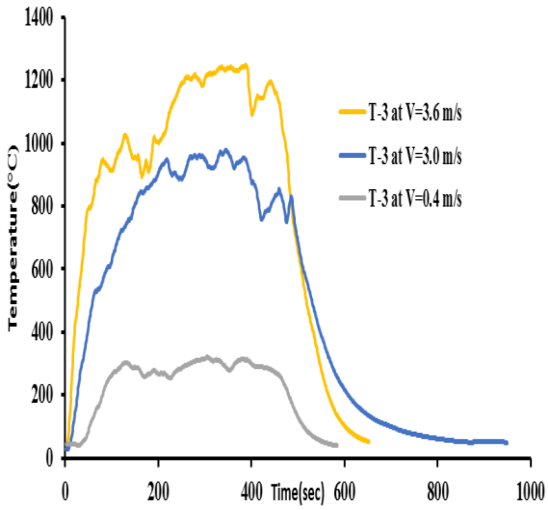


Figure-11. Temperature-time graph of t-3(vertical) at variable gas velocity

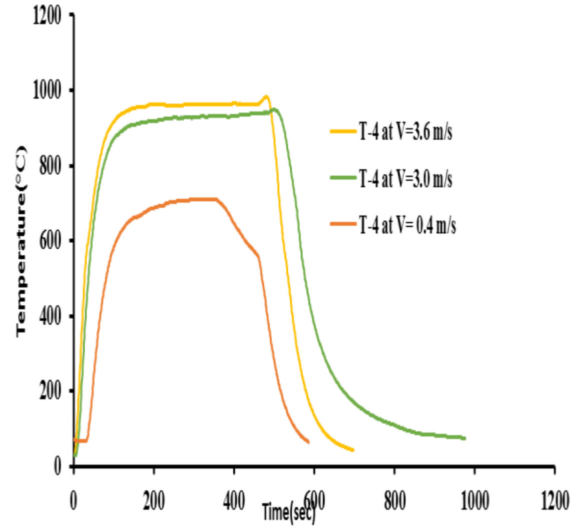

Figure-12. Temperature-time graph of t-4(horizontal) at variable gas velocity

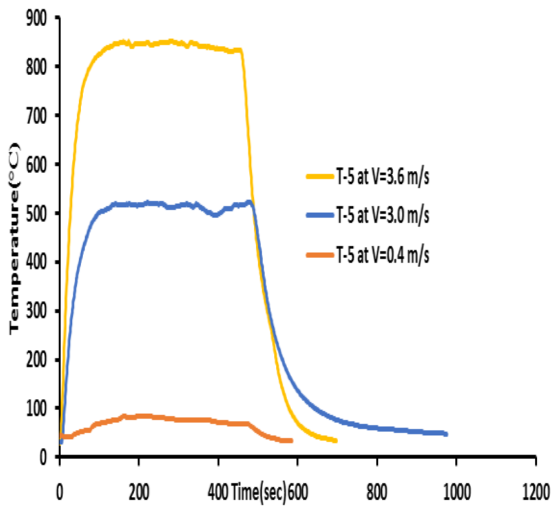

Figure-13. Temperature-time graph of t-5(horizontal) at variable gas velocity

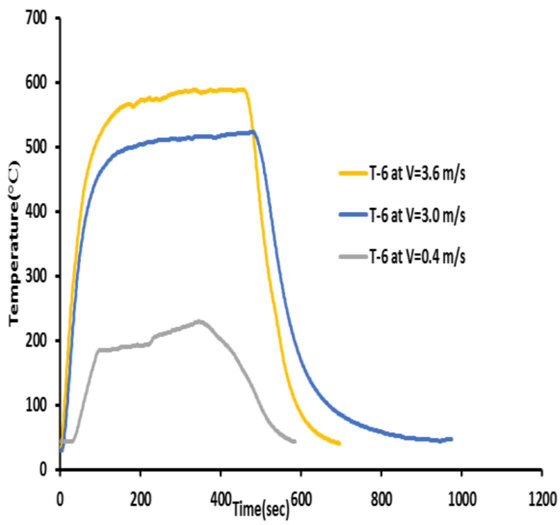

Figure-14. Temperature-time graph of t-6(horizontal) at variable gas velocity

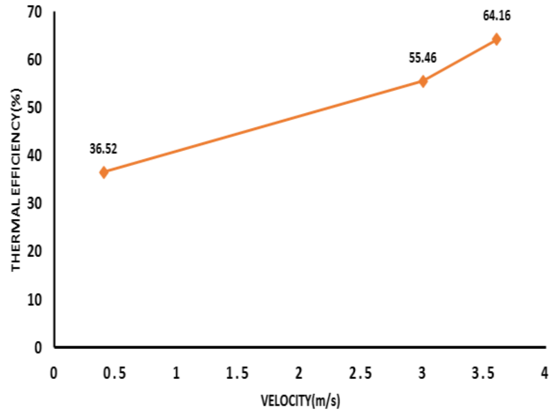

Figure-15. Gas velocity-Thermal efficiency graph

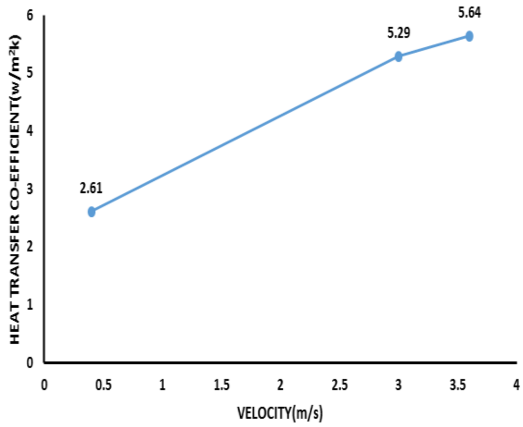

Figure-16. Gas velocity-Heat transfer co-efficient graph.

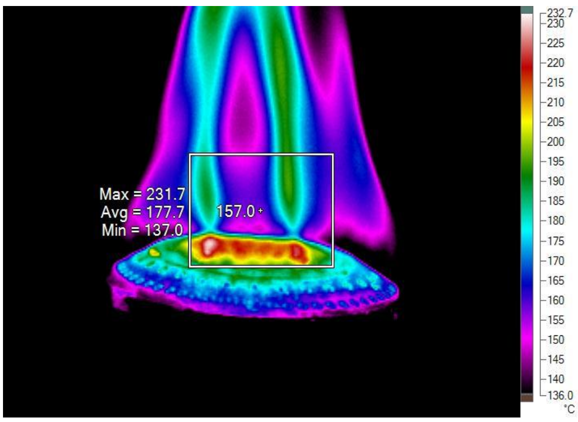

Figure-17. IR thermograph of SAPRB at v=3.6m/s.

It attains the maximum temperature because in this zone both convection and radiation heat transfer might be realized. Similarly from Fig. 4 the maximum temperature attained is 969°C. It attains the maximum temperature in that zone because it was very close to the center of the burner and the convection heat transfer is maximum whereas the radiation heat loss is minimum. By calculating, the local heat transfer coefficient h, at gas velocity 3.6m/s is found to be 5.64 $W/m^2k$.

### 3.2. Effect of Heat Transfer at Average Gas Velocity

Fig.5 and Fig.6 shows the time dependent temperature histories of the vertical and horizontal arranged thermocouples respectively at gas velocity 3.0 m/s. From the Fig. 5 the maximum temperature attained is 970°C by thermocouple-3. It attains the maximum temperature because in this zone both convection and radiation heat transfer occurs. Similarly from Fig.6 the maximum temperature attained is 946°C. It attains the maximum temperature because it was very close to the center of the burner and the convection heat transfer is maximum whereas the radiation heat loss is minimum. By calculating the local heat transfer coefficient h, at gas velocity 3.0m/s is found to be 5.29 $W/m^2k$.

### 3.3. Effect of Heat Transfer at Minimum Possible Gas Velocity

Fig.7 and Fig.8 shows the time dependent temperature histories of the vertical and horizontal arranged thermocouples respectively at gas velocity 0.4 m/s. This is the minimum velocity where the combustion could possible in this experimental apparatus. From the Fig.7 the maximum temperature attained is 315°C by thermocouple-3. It attains the maximum temperature because in this zone both convection and radiation heat transfer takes place. Though the thermocouple 1 and 2 are close to the surface but they could not attain that much of temperature like thermocouple-3.Similarly from Fig.8 the maximum temperature attained is 710°C. It attains the maximum temperature among all the thermocouples in the minimum gas velocity because it was very close to the center of the burner and the convection heat transfer is maximum whereas the radiation heat loss is minimum. By calculating the local heat transfer coefficient h, at gas velocity 0.4m/s is found to be 2.61 $W/m^2k$. From the above facts it is clear that the convective heat transfer is the predominantly mode of heat transfer and the radiation heat transfer lends the required support to it. The radiation heat transfer is more when the gas velocity is more and vice-versa is also true.

### 3.4 Transient Temperature Measurement

Fig.9-14 shows the time dependent temperature graphs of a single thermocouple (horizontal or vertical) with different velocities. From the graph we conclude that, vertically & horizontally the rate of heat transfer increases with increase in gas velocity and vice versa. But the notable factor is the thermocouple-4 attains the temperature about 703°C with the minimum gas velocity 0.4m/s. So it indicates that the rate of heat transfer is more in that zone where both convection and radiation heat transfer may be realized.

### 3.5. Calculation of Heat Transfer Co-efficient (h).

The following "Table" contains the calculated values of $R_e$, $N_u$ and h form the "equation (1)".

| Gas Velocity | 3.6 m/s | 3.0 m/s | 0.4 m/s |
|---|---|---|---|
| $R_e$ | 0.0524 | 0.0437 | 0.0058 |
| $N_u$ | 0.3384 | 0.3176 | 0.1566 |
| h | 5.64 $w/m^2k$ | 5.29 $w/m^2k$ | 2.61 $w/m^2k$ |

### 3.6. Thermal Efficiency Calculation

The percentage of thermal efficiency η of burner is estimated based on following formula [5].

$$\eta = \frac{(m_w c_{pw} + m_v c_{pv})(T_2 - T_1)}{m_f CV} \quad (3)$$

Where, $m_w$ and $m_v$ are the masses of water and vessel and $m_f$ are the mass of the fuel consumed during experiment. The calorific value of the fuel (CV) is 45780 kJ/kg. Specific heats of aluminum and water are $C_{pv}$ = 0.8959 kJ/kg K and $C_{pw}$ = 4.1826 kJ/kg K, respectively. By calculating the thermal efficiency of the burner is found to be 64.16% at gas velocity 3.6 m/s, 55.46% at gas velocity 3.0 m/s and 36.52% at gas velocity 0.4 m/s. So the maximum thermal efficiency obtained in this experiment is 64.16% which is very good considering the main focus of this experiment was process optimization instead of other aspects.

## IV. Conclusion

- The performance of SAPRB was analyzed experimentally to realize the effects of various controlling parameters.
- The thermal efficiency of SAPRB was computed up to 64 % which showed a good agreement with the previous works.
- The controlling parameters such as gas velocity, nozzle exit to the burner distance were critically varied to realize the convection heat transfer from the burner surface and flame.
- K-type thermocouples were used to measure the transient local temperature distribution and FLUKE IR thermography confirmed the radiation dominancy.
- It was confirmed that the overall thermal efficiency of SAPRB can be enhanced by optimizing the controlling process parameters even though the design is not modified.

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