DESIGN OF NOVEL RADIANT BURNER WITH HELICAL COMBUSTION CHANNELS

A PROJECT REPORT

submitted by

ABIRAJ R

Reg. No: TKM21MEIR02

to

the APJ Abdul Kalam Technological University
in partial fulfilment of the requirements for the award of the Degree
of
Master of Technology

in

Mechanical Engineering

Specialization*: Industrial Refrigeration and Cryogenic Engineering*

Department of Mechanical Engineering

THANGAL KUNJU MUSALIAR COLLEGE OF ENGINEERING

KOLLAM

MAY 2023

DEPARTMENT OF MECHANICAL ENGINEERING
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CERTIFICATE

This is to certify that the report entitled **"DESIGN OF NOVEL RADIANT BURNER WITH HELICAL COMBUSTION CHANNELS"** submitted by **ABIRAJ R, Reg No: TKM21MEIR02** during **2022-23** to the APJ Abdul Kalam Technological University in partial fulfillment of the requirements for the award of the Degree of Master of Technology in Industrial Refrigeration and Cryogenic Engineering is a bonafide record of the project work carried out by him under my guidance and supervision. This report in any form has not beensubmitted to any other University or Institute for any purpose.

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DECLARATION

I **Abiraj R**, hereby declare that the project report "Design of novel radiant burner with helical combustion channels", submitted for partial fulfillment of the requirements for the award of degree of Master of Technology of the APJ Abdul Kalam Technological University, Kerala is a bonafide work done by me under supervision of **Dr.Reby Roy K E.** This submission represents my ideas in my own words and where ideas or words of others have been included, I have adequately and accurately cited and referenced the original sources. I also declare that I have adhered to ethics of academic honesty and integrity and have not misrepresented or fabricated any data or idea or fact or source in my submission. I understand that any violation of the above will be a cause for disciplinary action by the institute and/or the University and can also evoke penal action from the sources which have thus not been properly cited or from whom proper permission has not been obtained. This report has not been previously formed the basis for the award of any degree, diploma, or similar title of any other University.

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Finally, I take this opportunity to extend my deep appreciation to family and friends, for all that they meant to me during the crucial times of the completion of this project.

ABIRAJ R
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ABSTRACT

Nowadays, Liquefied Petroleum Gas (LPG) burners are widely used in households and industries. The conventional LPG burners work on Free Flame Combustion and have lower thermal efficiency, lower turn-down ratio and emit higher emissions compared to the burners based on newer technologies such as Porous Media Combustion (PMC). Porous media combustion is characterized by higher flame speed, wider flammability limit and high radiant output. Innovative 3D metal and ceramic additive printing techniques allow for the manufacturing of porous media with helical combustion channels instead of the typical sponge-like matrices used in porous media burners. The presence of helical combustion channels induces secondary flows, which increase the heat transfer rate of combustion to the nearby solid matrix. In this work, CFD analysis of a radiant burner with helical combustion channel is done with the help of ANSYS fluent.

Keywords: Radiant burner; Porous media combustion; Helical combustion channels; ANSYS Fluent.

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NOMENCLATURE

A/F	Air-fuel ratio
CV	Calorific value
c_p	Specific heat
d_m	Mean pore diameter
h_v	Volumetric heat transfer coefficient
k	Thermal conductivity
m_f	Mass flow rate of fuel
Pe	Peclet number
Q_{th}	Thermal power input
SL	Flame velocity
T	Temperature
u	Velocity
w	Molecular weight
Y	Species mass fraction
ε	Emissivity
Φ	Equivalence ratio
$\boldsymbol{\varphi}$	Porosity
ρ	Density
ω	Scattering Albedo

ABBREVIATIONS

CB	Conventional Burner
CZ	Combustion zone
LPG	Liquified Petroleum Gas
PM	Porous matrix
PMC	Porous Medium Combustion
PRB	Porous Radiant Burner
RBWHCC	Radiant burner with helical combustion channels

CHAPTER 1
INTRODUCTION

Given the current energy crisis, environmental concerns, and increasingly stringent emission regulations, there is a rising interest in enhancing thermal efficiency and reducing the emissions of pollutants. Despite this, the majority of energy needs are still fulfilled by traditional combustion devices that rely on fossil fuels. In developing countries, Liquefied Petroleum Gas (LPG) is widely used as a cooking fuel. However, it contributes significantly to indoor air pollution, which can lead to respiratory and cardiovascular diseases. The current LPG cooking stoves available in the Indian market have a thermal efficiency of 60-65%. Unfortunately, these stoves also emit carbon monoxide (CO) and nitrous oxide (NOx) at levels of 50 to 225 ppm and 2 to 7 ppm, respectively, which exceed the standards set by the World Health Organization (WHO) (Pantangi et al., 2011) [28].

The principle behind the conventional burners (CB) used in LPG cooking stoves is based on the Bunsen burner. These burners rely on free-flame combustion that is combustion takes place in a gaseous environment, with convection being the primary mode of heat transfer. The reaction zone in free-flame combustion is extremely thin, leading to a high-temperature gradient across the flame. As gases have low emissivity and thermal conductivity, the radiation and conduction modes of heat transfer from the post-flame to pre-flame zone are negligible. Consequently, conventional burners have low energy efficiency and exhibit undesirable characteristics like low flammability limits, low power density, and high levels of pollutant emissions. The mixture of air and LPG has an equivalence ratio of 1 or greater than 1, and combustion with the conventional burner (CB) is always fuel-rich. This leads to incomplete combustion and emission of carbon monoxide. The thermal efficiency of burner is calculated by the water-boiling test published in IS: 4246:2002.

The Porous Radiant Burner, which works based on porous medium combustion (PMC), is employed in domestic cooking stoves to overcome these issues. In a Porous Radiant Burner (PRB), the combustion occurs inside the cavities of a highly conducting and radiating porous medium. Due to the large surface area of the porous matrix and high heat transfer coefficient, convective heat transfer is also better than free flame combustion. The better heat transport (through the combined modes of conduction, convection, and radiation) results in a homogeneous temperature distribution in the combustion zone.

Novel 3D printing methods for metal and ceramic materials enable the production of porous media with helical combustion channels instead of the traditional sponge-like structures used in porous media burners. As fluid flows through the curved path of these channels, centrifugal force acts upon the different elements of fluid moving at various velocities, leading to secondary circulation. This secondary flow enhances heat transfer characteristics, resulting in a significant increase in the transfer of heat from the combustion product to the solid matrix.

The existence of all three modes of heat transfer leads to better heat transfer and uniform temperature distribution. The PMC is also characterized by a high burning velocity, reduced temperature drop across the reaction zone, high radiant output, high peak flame temperature, and lower emission. In addition to these, low calorific fuels can also be combusted in the porous burners.

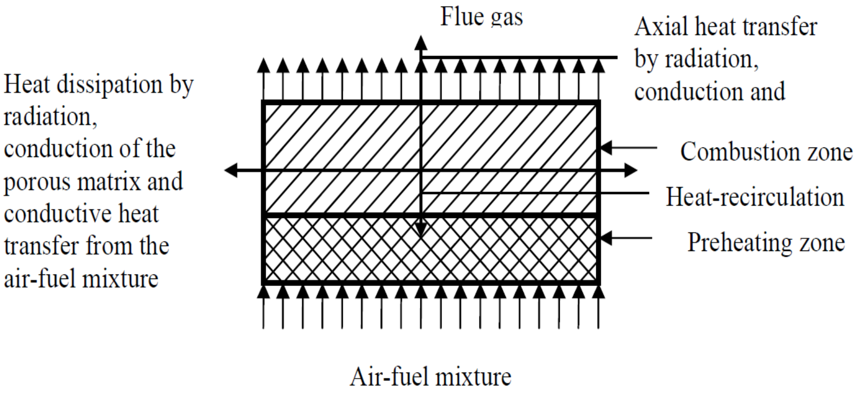

Fig.1.1 – Illustration of combustion occurring in porous radiant burner
(P. Muthukumar and Piyush Anand 2011)

Fig.1.2 – The photographic view of PRB taken at IIT Guwahati (Mishra, 2015)

It is not practical to provide compressed air externally for domestic cooking purposes. Therefore, a self-aspirated LPG stove with a PRB has been developed to operate using natural draft for domestic cooking applications. This project aims to carry out numerical analysis and parametric study of the self-aspirating Radiant burner with helical combustion channels.

A brief history of various literature and the porous radiant burner technology development is discussed in the second chapter. The third chapter presents the objectives of the project. The methodology used is explained in the fourth chapter. The fifth chapter deals with the results of the project work. Conclusions and future work are presented in the sixth chapter.

CHAPTER 2

LITERATURE REVIEW

2.1 EXPERIMENTAL WORK

The distinction between combustion in a porous medium (PM) and combustion in a conventional system is due to improved and more efficient heat transmission from burned gases to unburned mixture.

Sanitjai and Jugjai (1996) [31] developed a revolutionary design for a standard open-flame gas burner employing porous media technology to enhance the thermal efficiency of a regular burner by up to 10%. Porous radiant recirculated burner (PRRB) is a modified design that uses porous medium to convey some of the hot gas enthalpies to a premixed mixture. The PRRB does not work by stabilizing the flame within the porous substance. Instead, its flame is a free flame and the porous media just enhances the heat recirculation from the exhaust gas to the mixture of fuel and air. The porous media that are used in this study are made of several layers of stainless steel wires with 40 meshes per inch.

Jugjai and Rungsimuntuchart (2002) [10]. A novel semi-confined porous radiant recirculated burner (PRRB) concept based on heat-recirculating combustion using the porous medium technology was developed for energy saving in domestic use and in a small-scale food processing industry. The proposed PRRB (SB) provide the maximum thermal efficiency of 60% with improvement in energy saving as well as environmentally compatible emissions. It can be applied in the small-scale food processing industry. The use of porous media to promote the recirculation of heat from the hot combustion products to the combustion air resulted in an efficiency increase of 12% further efficiency improvements are obtained when the burner is combined with a swirling central flame technique. In this case, slightly higher CO and NO_X emissions are observed.

Khan (2010) [14] showed the effect of different porous medium on the thermal efficiency of LPG cooking stoves. The design explains that the thermal efficiency of a conventional LPG burner is 68% but its practical value was found to be 51.15%. With the use of different porous mediums such as brass chips, mild steel chips and ball bearing it was observed that the thermal efficiency was different from each other and in all cases efficiency was found to be more than conventional burner.

All of the above papers explore the heat-recirculating capacity of porous media but do not consider combustion inside porous media. Later, Yoksenakul and Jugjai (2011) [35] developed a self-aspirating porous medium burner (SPMB), where combustion takes place inside the porous matrix, which is formed by a packed bed of 15mm alumina sphere. The burner developed can only be applicable to small and medium-scale enterprises (SMES) and firing rates from 26 to 61 KW were tested.

Patangi et al. (2011) [29] deals with the performance test of PRB (porous radiant burner) used for LPG domestic cooking stove unlike the conventional LPG stoves for which the CO and NO_X emissions were found in the range of 400-1050 mg/m^3 and 162-116 mg/m^3 respectively for the burn with PRB, the same burn in the range of 25-350 mg/m^3 and 12-25 mg/m^3. The axial temperature distribution in the burner shows that the reaction zone was close to the interface of the two zones and at a higher thermal load, it shifted towards the downstream. The surface temperature of the PRB was found to be uniform.

Patanji et al. (2007) [28] concluded that, in general, when porous media combustion was used, the performances of the burner were better than the ones of the conventional burner analyzed. The best results were obtained when metal chips were used in the combustion chamber and the mixing chamber was insulated. In this case and when compared to the best conventional burner, the thermal efficiency of the porous media burner was improved by 4%, the CO emissions were reduced by 52% and the fuel consumption was reduced by 10%.

Keramiotis et al. (2011) [11] An experimental investigation on a two-layer porous burner with an Al_2O_3 flame trap and a 10 ppi (pores per inch) SiC foam. The burner was operated with methane and LPG. The results revealed a homogeneous temperature distribution, low NO_X and CO emissions and wide flexibility with respect to fuels and thermal loads. The effects of fuel interchange on efficiency and emissions were also analyzed.

Pantanji et al. (2007) [28] developed and proposed a porous radiant burner they built a two-layer where the preheating zone was made up of 5mm diameter alumina balls and the combustion zone by SiC form having 90% porosity. They analyzed the influence of the burner diameter, burner casing wall thickness and length of the porous matrices. For the best burner configuration and operating condition analyzed a 68% thermal efficiency was obtained.

Muthukumar et al. (2011) [25] tested a similar LPG porous radiant burner. The diameter of the burner was equal to the one that presented better efficiency in Pantangi et al studies (2011). The combustion zone has the same characteristics; however, the preheating zone is now composed of a ceramic block of 10 mm thickness and 40% porosity. Also, the operating conditions tested were different, since the equivalent ratio tested was significantly higher and the ambient temperature was different. A maximum thermal efficiency of 71% was obtained which is above the efficiency of conventional LPG burners. Efficiency was obtained for an equivalence ratio of 0.68, 1.24 KW power intensity at 310˚C ambient temperature. From a comparison of Pantangi et al. (2011) result with those of Muthukumar et al. (2011), one can conclude that it is better to use a ceramic block of 10 mm thickness and 40% porosity as a preheating layer than 5 mm diameter alumina balls forming a porous media with 12- 15 mm thickness. In their work, Muthukumar et al. (2011) [25] investigated the influence of the ambient temperature on the thermal efficiency of the porous burner and concluded that for the same operating conditions and efficiency improvement of 10% was achieved for varying the ambient temperature from 18.5°C to 350°C therefore when comparing the thermal efficiency of the burners the ambient temperature is the parameter that has to be taken into account.

Muthukumar and sham Kumar (2013) [26] extended pantangi et al.’s (2011) and Muthukumar et al.’s (2011) works. They tested the performance of a two-layer porous burner. In this case, the diameter of the porous matrices was chosen to be 120 mm. Like in Muthukumar et al. (2011), the preheating zone consists of a ceramic matrix with 40% porosity and a thickness of 10 mm and a combustion zone a Sic zone with a thickness of 20 mm. However, in this study the porosity of the SiC was varied from 80 to 90 %. The thermal efficiency and CO and NOX emissions were experimentally obtained for equivalent ratios and power ranging from 0.542 to 0.7 and 1.3 to 1.7 KW, respectively. The highest thermal efficiency obtained was around 75% with the SiC that has the highest porosity. This efficiency is higher than the LPG conventional burners available in the Indian market. The porous burner also performs better than conventional burners as far as emissions are concerned. For the 90% porosity SiC foam, NO_X emissions ranged from 0 to 0.75 mg/m^3 and CO emissions from 12 to 124 mg/m^3. Conventional LPG cooking stoves emit 4 to 7 mg/m^3 of NO_X and 250 to 650 mg/m^3 of CO.

B Herrera, K Cacua, L.Olmos-Villalba. (2015) [2]. A porous burner made of a bed of Al_2O_3 particles coming from grinding residues and combined with ceramic foam of SiSiC has been evaluated with respect to Liquefied Petroleum Gas combustion stability and thermal efficiency for cooking in the food industry. The results showed that for specific heat input rates lower than 154 kW/m^2, the upper and lower equivalence ratio on the stability limit follow approximately a linear trend, as well as the wide range of stability remains constant.

Mishra et al. (2015) [23] developed an LPG cooking stove with PRB for a medium-scale power range of 5–10 kW and evaluated the performance in terms of thermal efficiency and emissions of CO and NO$_x$. The results were compared with the conventional burner of the same power range. They reported that the burner produces flameless combustion in the equivalence ratio range of 0.54–0.72. The maximum reported thermal efficiency was 28% higher than the conventional burner of the same power input. The CO and NO$_x$ emissions in the PRB were much lower than those with the CB.

Mishra and Muthukumar (2018) [22] designed and developed a self-aspirating LPG cooking stove with a two-layer porous radiant burner. They obtained a maximum thermal efficiency of 75.1% for the PRB in the power range of 1–3 kW, while the respective value of the conventional burner (CB) is 65%. The newly designed PRB is self-aspirated and works on the natural draft without any safety issues. The measured CO and NO$_x$ emissions of the newly developed PRB stove are in the ranges of 30–140 ppm and 0.2–3.2 ppm, respectively, compared to 220–550 ppm and 5–25 ppm for conventional domestic LPG cooking stoves (1–3 kW).

2.2 NUMERICAL WORK

Yoshizawa et al. [1] (1988) [34] showed the importance of absorption coefficient and total optical thickness in calculating temperature profiles and flame position within PB. With a decrease in absorption coefficient, the maximum temperature increased, the size of the reaction zone decreased, and its position shifted upstream.

Sathe et al. (1990) [32] analysed a porous radiant burner by considering one-dimensional conduction, convection, radiation, and a premixed flame model where methane-air combustion is modelled using a single-step irreversible reaction. They concluded that the stable flame could be maintained in two different regions. The flame speed near the edge of the porous matrix is controlled by conduction, and radiation controls the flame propagation through the interior of the porous matrix. It was also found that radiative properties such as optical thickness and scattering albedo influence the radiant output. It was revealed that the flame should be stabilized near the centre of the porous medium for maximizing the radiant output.

Hsu and Matthews (1993) [9] developed models using both single-step and multi-step chemical kinetics. It was concluded that multi-step kinetics is crucial to predict temperature distributions, energy release rates, and emissions. Single-step kinetics was enough to predict all the flame characteristics except the emissions for the very lean conditions under which equilibrium favours the complete combustion process dictated by global chemistry.

Hsu et al. (1993) [9] numerically analysed the premixed methane-air combination in a double-layered porous burner. The simulations were done under the equivalence ratio ranging from 0.1 to 0.43. It was found that the lean limits were lower than the limit for free laminar flame and maximum flame speed occurred near the exit plane.

Mohammad et al. (1994) [19] analysed a 2-D model for a matrix-stabilised PMC with single-step kinetics and found the values in ±15% variation with available experimental values. Bouma et al. (1995) concluded that the emissions of CO and NOx were considerably low operating in radiant mode after studying the combustion of a lean premixed methane-air mixture stabilised in a ceramic foam burner using 15 chemical species with added NOx formulation chemistry.

Kulkarni and Peck (1996) [16] did a numerical investigation on a 5 cm-long double-layered PRB and studied the effects of porosity, length, extinction coefficient, and scattering albedo on the radiant output. They concluded that the upstream section of PRB should have low porosity, higher scattering albedo, shorter length, and higher optical thickness than the downstream section to maximize radiant output.

Malico and Pereira (2001) [17] studied the influence of radiative properties on flame speed and temperature distribution. They found that an increase in extinction coefficient resulted in a higher post-flame temperature gradient, improved heat transfer to the pre-flame region, and a decrease in maximum temperature.

Barra and Elzzy (2004) [1] studied a one-dimensional transient combustion process within a porous burner. They numerically quantified heat recirculation, radiation efficiency, solid conduction, and solid-solid radiation and flame speeds at different stable conditions. They concluded that increasing the equivalence ratio results in decreasing heat recirculation efficiency.

Hayashi et al. (2004) [5] presented a three-dimensional numerical study of a two-layer porous burner. They considered a single-step mechanism of heptane to model the combustion process. The burner was to operate in the range of 5–20 kW, fuelled by a mixture of air and vapour of a blend of fuel oil and vegetable oils for domestic heating, and the excess air ratio ranged from 1 to 1.8.

Mishra et al. (2006) [21] analysed a two-dimensional rectangular porous radiant burner using methane-air combustion with detailed chemical kinetics. They studied the effects of the power density, equivalence ratio, extinction coefficient, and volumetric heat transfer coefficient on temperature and concentration profiles.

Khosravy El-hossaini et al. (2008) [15] compared methane combustion in numerical modelling of porous radiant burners using full and reduced kinetics mechanisms. They provided comparable profiles for temperature distribution. The CO mole fraction profiles of full mechanisms show a slight variation due to differences in kinetic rate constants, and the NO concentration is overpredicted compared with experimental data.

Panigrahy et al. (2016) [30] analysed both computationally and experimentally the combustion characteristics of LPG in a two-layer PRB cooking stove. Panigrahy and Mishra (2016) numerically modelled the combustion of LPG in a planar silicon carbide PRB. The stability range of combustion of LPG and CH_4 in PM as well as the combustion of LPG in the free-flame mode were compared, and LPG combustion in PM was found to have a higher operating range than that of CH_4. The radiative heat flux with LPG in the PRB was more than that with CH_4. The CO emission with LPG combustion in the PRB was lower than in the free-flame mode.

Keshtkar et al. (2009) [13] numerically solved the coupled energy equations for the gas and porous medium to analyse the thermal characteristics of porous burners. They used the discrete ordinates method for radiation analysis.

Hashemi and Hashemi (2017) [4] investigated flame stabilization within a two-layer porous burner. They simulated the methane-air combustion process in a two-dimensional PRB. Their results showed that the equivalence ratio of the incoming mixture could control the stability limit and flame temperature within the porous burner.

S.N. Hoda et al. (2019) [6] investigated the thermal characteristics of a 3-D rectangular porous radiant burner. They found that by increasing the excess air ratio, the maximum temperature and amount of pollutants decrease.

Using an external compressor to supply air is not a feasible option for domestic cooking applications. Therefore, self-aspirated PRB is preferred. A study of the literature shows that no researchers have numerically investigated the combustion of LPG domestic cooking stoves with PRB working on a natural draft for the power range of 3-5 kW. In the present work, the numerical analysis of the combustion of LPG in a self-aspirated cooking stove with radiant burner having helical combustion channels is done for an equivalence ratio of 0.77.

CHAPTER 3
PROBLEM STATEMENT & OBJECTIVES

The aim of the work is CFD analysis and performance analysis of an improvised self-aspirating LPG radiant burner with helical combustion channels.

The objectives are as follows:

1. Develop a Radiant burner with helical combustion channels.
2. Examine the flue gas for pollutants and contrast it with that of the porous radiant burner.
3. Compare the heat transfer of the optimized model with that of a porous radiant burner.

CHAPTER 4
METHODOLOGY

A computational model of the self-aspirating radiant burner with helical combustion channels operating on LPG is developed to carry out numerical analysis and to compare the results with a porous radiant burner. Analysis of the radiant burner is done to find maximum temperature at combustion zone, concentration of carbon monoxide in flue gas and to find maximum amount of radiative heat transfer.

4.1 FORMULATION

The three-dimensional structure of the radiant burner (PRB) was designed based on the specifications provided by Mishra and Muthukumar in 2018 [22], while the dimensions of the helical combustion channel were taken from a patent for a radiant burner with an alveolar radiating face (PCT/FR83/00205) [35]. The PRB consists of several components, including a combustion zone, a mixing chamber, a burner pipe, an air slot, and a fuel inlet orifice. The combustion zone is made of silicon carbide (SiC) with a diameter of 70 mm and a thickness of 10 mm. The burner pipe has a diameter of 21 mm, while the orifice for fuel inlet has a diameter of 0.35 mm. The air slot is 30 mm in length and 10 mm in width. The helical combustion channel has a diameter of 1 mm, 5 turns, and a height of 20 mm.

Air is drawn into the PRB through two slots by the venturi effect, which is created by the high-speed LPG jet, causing a low-pressure area near the burner pipe, and allowing the air and fuel to mix and enter the porous material through its bottom surface. The spark ignites the combustion at the top surface of the helical combustion channels, and it spreads throughout the volume of the channels, transferring heat to the SiC matrix. The combustion of a mixture of LPG gases consisting of 60% propane (C3H8) and 40% butane (C4H10) generates heat and produces two product substances: carbon dioxide (CO2) and water (H2O). The Porous Medium (PM) conducts and radiates heat, reflecting conduction and radiation heat transfer throughout the medium. The temperature of the PM before the reaction zone increases due to conduction and radiation, and the incoming air-fuel mixture is preheated through convective heat transfer. Advection contributes to energy transfer following the reaction zone, resulting in a higher gas temperature compared to the PM. Therefore, convective heat transfer occurs from the gas to the solid

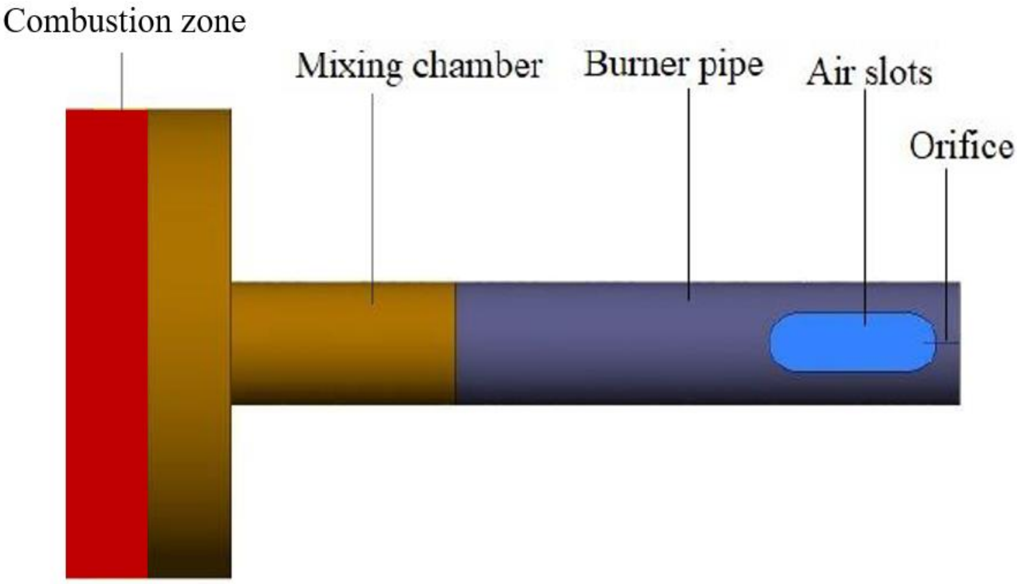

Fig 4.1 Burner model

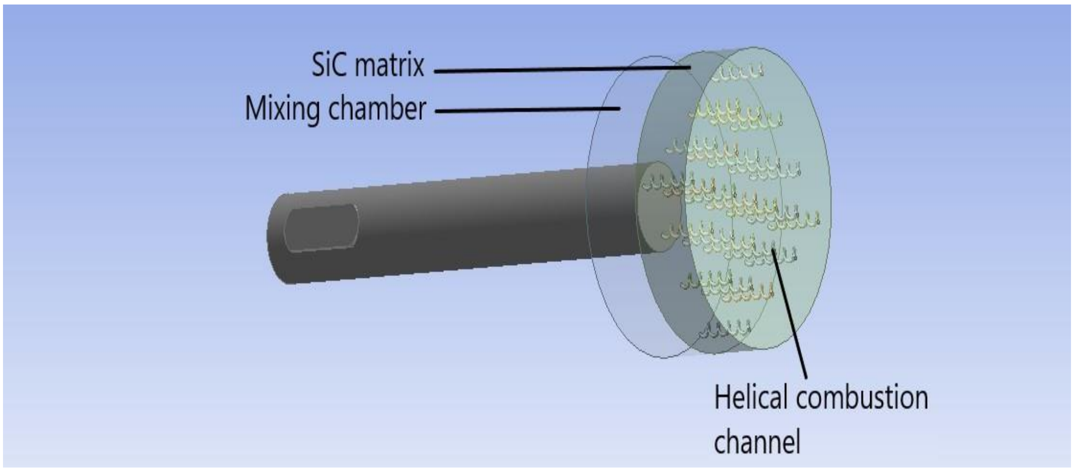

Fig 4.2 Burner with 25 helical combustion channels

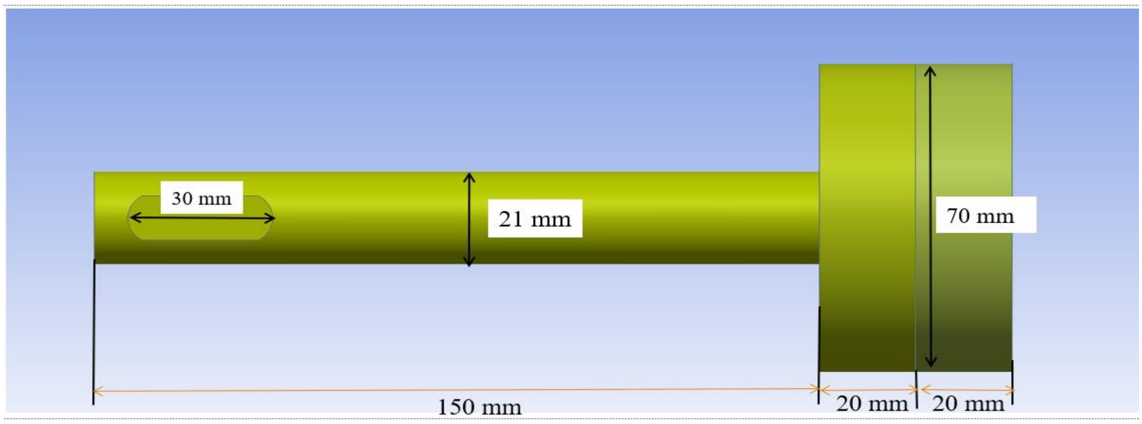

Fig 4.3 Dimensions of the burner

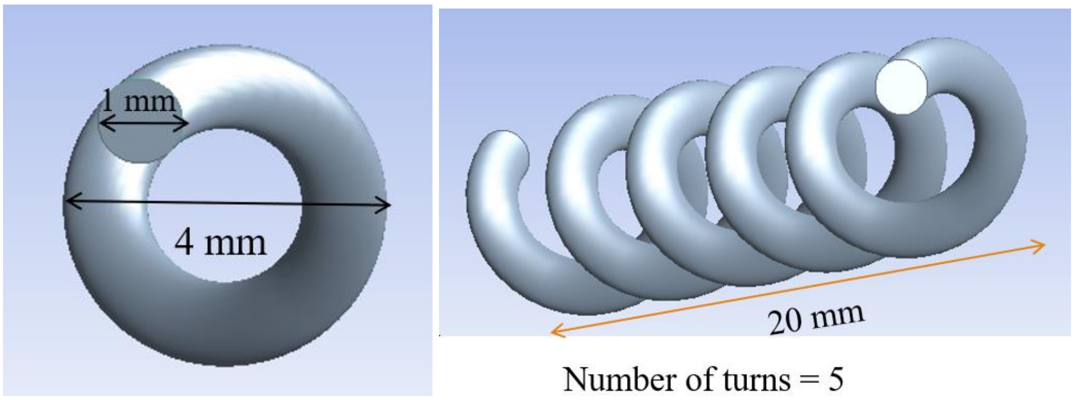

Fig 4.4 Dimensions of a single helical combustion channel

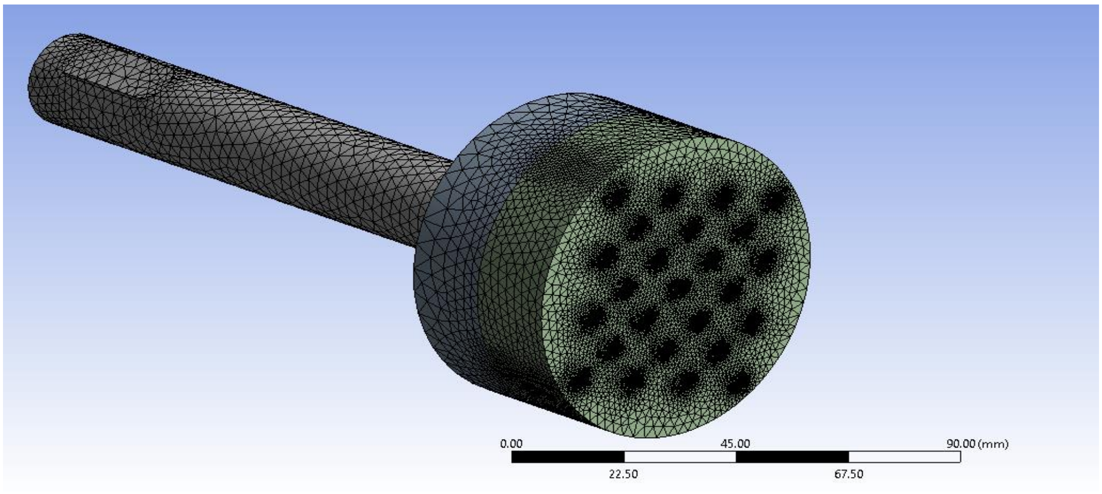

Fig 4.5 Meshed model

Number of elements of the mesh model = 3566850

Analysis of the present problem requires simultaneous solutions of the continuity equation, species conservation equation, the ideal gas equation and radiative transfer equation.

The porous matrix can absorb, emit and scatter the radiation as a grey homogeneous radiating medium, and the gaseous radiation is considered negligible. The Discrete Ordinate Radiation Model is used for radiative heat transfer from a body. It solves the radiative heat transfer equation for a finite number of solid angles that are associated with vector directions fixed in the global Cartesian system. The k-ε model is utilized for turbulence criteria. It belongs to the class of Reynolds Averaged Navier-Stokes Equation (RANS) turbulence models, where the time-averaged Navier-Stokes equation is considered. For the combustion of LPG species transport model with eddy dissipation is utilized, which solves the conservation equation for each component, describing convection, diffusion, and reaction sources.

The governing equations are

Continuity Equation:

The conservation of mass can be represented by the continuity equation as

$$\nabla.(\rho_g \vec{u} \varphi) = 0$$

where φ is the porosity of the medium.

Momentum Equation:

The conservation of momentum can be represented by the momentum equation as

a) For the mixing tube and the mixing chamber:

$$\nabla.(\rho_g \vec{u}\vec{u} \varphi) = -\nabla p + \nabla.(\mu \nabla \vec{u})$$

b) For the porous domains, additional source term Si is added to the right-hand side of the momentum equation:

$$S_i = -\left(\frac{\mu}{K_1}\vec{u} + C_2 \frac{1}{2}\rho_g |\vec{u}|\vec{u}\right)$$

The permeability of the porous media is represented by K_1 and the inertial resistance factor is represented by C_2.

Species Conservation Equation:

$$\nabla.\left(\rho_g \vec{u} Y_i\right) = -\nabla.\left(\rho_g\left(D_{m,i} + D^{d}_{m||}\right)\right) + \dot{w}_i W_i$$

Energy Equation:

$$\nabla(c_g \rho_g T_g \vec{u}) = \varphi \nabla(k_g + c_g \rho_g D^{d}_{||})\nabla T_g - \varphi \Sigma w_i h_i W_i - h_v (T_g - T_s)$$

Ideal gas equation:

$$\rho_g = \frac{P\overline{W}}{RT_g}$$

Radiative transfer equation:

$$\frac{dI(\vec{r},\vec{s})}{ds}+(a+\sigma_s)I(\vec{r},\vec{s})=an^2\frac{\sigma T^4}{\pi}+\frac{\sigma_s}{4\pi}\int_0^{4\pi}I(\vec{r},\vec{s}')\Phi(\vec{s}\cdot\vec{s}')d\Omega'$$

T, W, w, Y, h and k represent the temperature, molecular mass, molar rate of production, mass fraction, molar enthalpy and thermal conductivity, respectively. The subscripts 'v' and 'i' stand for volumetric and i^th^ species, respectively. $D_{||}^{d}$ represents the thermal dispersion coefficient, Dm,i is the diffusion coefficient of species and $D_{m||}^{d}$ is the species dispersion coefficient. The volumetric heat transfer coefficient h_v is employed to model the convective heat transfer term that couples the energy equations for the solid and the gas phase.

The relation for the surface area per unit volume and convective heat transfer coefficient is

$$A = 6(1-\varphi) / dp$$

$$\frac{h_V d_p^2}{\lambda_g}=\left(0.0426+\frac{1.236}{L/d_p}\right)\mathrm{Re}$$

$$d_p=\frac{\sqrt{4\varphi/\pi}}{\mathrm{PPC}}(\mathrm{cm})$$

where PPC is the pores per cm of the ceramic foam. Rep is the particle-based Reynolds number which is defined as $\mathrm{Re} = \rho_g u_g d_p / \mu_g$

Combustion of LPG was modelled using a one-step mechanism and the rate of reaction was calculated by the eddy-dissipation model.

$$C_3H_8 + 5O_2 \rightarrow 3CO_2 + 4H_2O$$

$$C_4H_{10} + 6.5O_2 \rightarrow 4CO_2 + 5H_2O$$

4.2BOUNDARY CONDITIONS

CALCULATION :

Equivalence ratio (Φ) is defined as the ratio of air-fuel ratio (AFR) at the stochiometric condition to actual AFR.

$\phi = \frac{(A/F)_{stoich}}{(A/F)_{actual}},$

$\boldsymbol{\phi}$ < 1-Lean mixture
$\boldsymbol{\phi}$ >1-Rich mixture

- The stoichiometric ratio for burning LPG in air $(A/F)_{stoich}$ = 15.6:1
- Flow rate of air inlet = 0.00137 kg/s (Self aspirating type radiant burner, Minimum mass flow rate required for proper mixing)

$$\phi = 0.77 = \frac{(A/F)_{stoich}}{(A/F)_{actual}}$$

$$(A/F)_{actual} = \frac{15.6}{0.77} = 20.25$$

$$\frac{Mass\ flow\ rate\ of\ air}{mass\ flow\ rate\ of\ fuel} = 20.25$$

$$mass\ flow\ rate\ of\ fuel = \frac{0.00137}{20.25} = 6.765 \times 10^{-5}\ \text{kg/s}$$

In the case of a lean mixture, fuel gets sufficient air to combust, however, as the mixture becomes richer, the combustion by-products increase due to lack of sufficient air, hence increasing the CO and NOx emissions. By the consideration of the emission standard equivalence ratio is taken as 0.77 referring to the data provided in the journal by Mishra and Muthukumar (2018) [22].

Inlet

- Fuel (LPG)

  Mass flow inlet ($\dot{m}_f$) = 6.765×10^{-5} kg/s
  Inlet temperature (T_0) = 300 K
  Mass fraction Y (C_3H_8) = 0.6, Y (C_4H_{10}) = 0.4, Y (O_2) = 0, Y (N_2) = 0
- Air

  Inlet vent
  Inlet temperature (T_0) = 300 K,
  Mass fraction Y (C_3H_8) = 0, Y (C_4H_{10}) = 0, Y (O_2) = 0.21 and Y (N_2) = 0.79

Burner outlet is considered as pressure outlet with gauge pressure zero Pascal (0 Pa). At the burner outlet, zero gradient boundary conditions are set to ensure that complete equilibrium is established.

The property parameters used for numerical analysis are given in Table 4.1 and Table 4.2.

Table 4.1 – Material properties data of solids used for computation

Property	**SiC**
Porosity (φ)	0.9
Density (ρ_s)	3210 kg/m^3
Specific heat (c_p)	824 J/kg K
Thermal conductivity at 200C (k_s)	80 W/m K
Thermal conductivity at 10000C (k_s)	20 W/m K
Scattering Albedo (ω)	0.8
Emissivity (ε)	0.9

The thermal conductivity of SiC is defined as a piecewise-linear function of temperature. For SiC,

$$k_s(T) = 80 + \frac{20 - 80}{1273 - 293}(T - 293)$$

Table 4.2 – Material properties data of fluids used for computation

Property	**C_4H_{10}**	**C_3H_8**
Density (ρ_f)	2.46 kg/m^3	1.91 kg/m^3
Thermal conductivity (k_f)	0.0159 W/m K	0.0177 W/m K
Viscosity	7×10^{-6} kg/m s	7.95×10^{-6} kg/m s

The specific heat capacity of C_4H_{10} and C_3H_8 is defined as a piecewise-polynomial function of temperature.

For C4H10,

for $300 \leq T < 1500$,

$c_p(T) = -322.796+8.413472T-0.006473865T^2+2.913973e\text{-}06T^3-5.835425e\text{-}10T^4$

for $1500 \leq T < 4000$,

$c_p(T) = 2859.143+1.483769T-0.0001374771T^2-6.612954e\text{-}08T^3+1.173366e\text{-}11T^4$

4.3 SOLUTION METHOD

The governing equations, including the gas and solid energy equations, species conservation equations, and radiative transfer equations, must be solved to determine the thermal behavior of porous radiant burners. These equations should be solved simultaneously and numerically to obtain the values of gas and solid temperatures, the concentration of species, and radiative heat fluxes at each nodal point in the 3-D computational domain. The Coupled algorithm is used to solve the pressure-velocity coupling momentum equation and the pseudo transient solution method is enabled. All equations are solved until a relative convergence of 10^{-6} is obtained. The Discrete ordinates method is used to solve the radiative heat transfer equation. This method is based on the discrete representation of the directional behaviour of the radiative intensity. The computational domain was discretized by a polyhedral mesh. The solver ANSYS Fluent is used for simulation of flow, evaluation of species production rate, transport and thermodynamic properties and combustion analysis.

CHAPTER 5

RESULTS AND DISCUSSION

The numerical analysis of combustion of LPG in a self-aspirated radiant burner with helical combustion channels is carried out. The heat flux at SiC matrix, outlet temperature and concentration of pollutants (Carbon monoxide) in the flue gas is studied. In this chapter, the results of the analysis are discussed.

5.1 CONTOURS

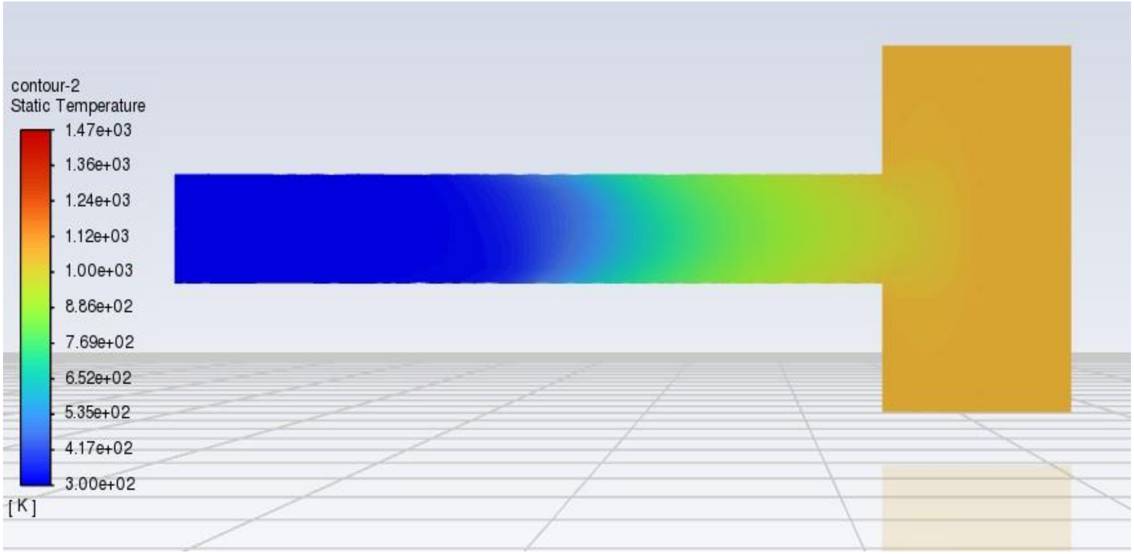

Fig 5.1 Contour of static temperature

The temperature distribution indicates that the combustion zone experiences a higher temperature, with a maximum temperature of 1066 K observed at the outlet. As SiC matrix has the ability to radiate heat in all directions, the temperature in the mixing chamber is elevated due to this radiative heat transfer. Hence, the temperature at the mixing chamber is higher than that at the inlet, which is typically 300 K. Overall, this explains that the temperature in the radiant burner varies across its different regions, with the combustion zone experiencing the highest temperature due to the exothermic combustion process. The SiC matrix in the burner also contributes to the temperature distribution by means of radiative heat transfer.

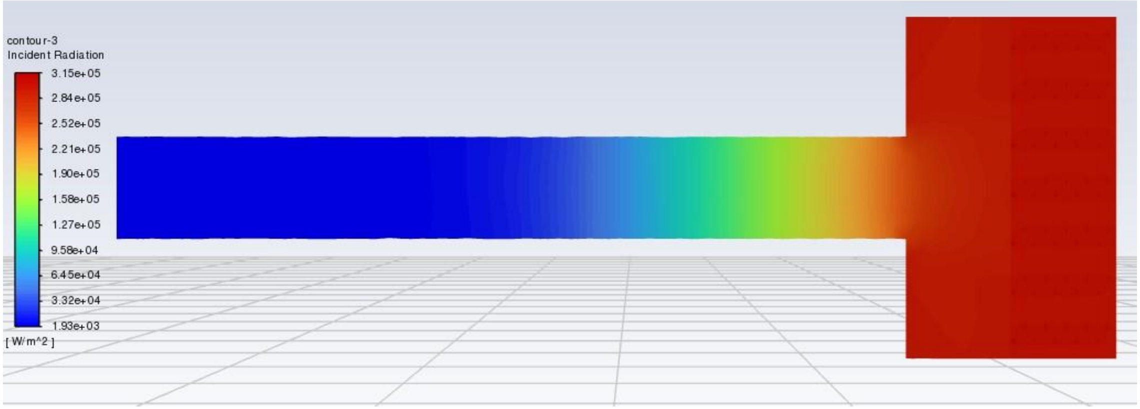

Fig 5.2 Contour of radiation heat flux

The radiation heat flux Contour of a radiant burner indicates the distribution of radiant heat energy emitted from the burner's surface. The contours provide information on the intensity and distribution of heat radiation in the surrounding environment. The radiation heat flux contour shows that the combustion zone has a higher heat flux, and a maximum heat flux of 30499 W/m^2 was obtained.

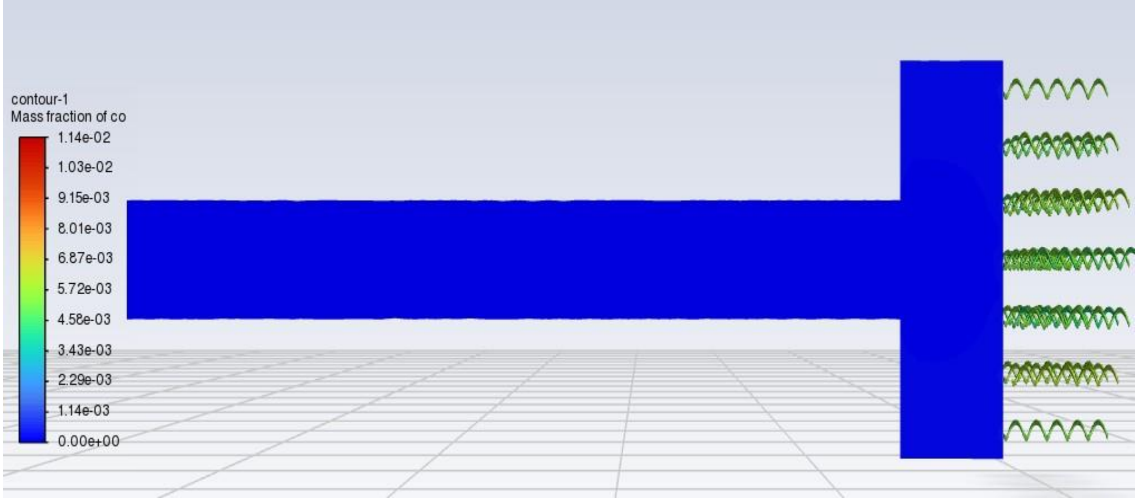

Fig 5.3 Contour of mass fraction of carbon monoxide

The carbon monoxide (CO) mass fraction contour shows a value of 0.0053 at the helical combustion channels, indicating its production. Even though it is a small amount, it is significant because CO emission from a burner signifies incomplete fuel combustion. This means that not all of the carbon in the fuel is converted to carbon dioxide (CO2), which is a less harmful greenhouse gas compared to CO. CO is a toxic gas that can pose health hazards if inhaled in large amounts. Moreover, high levels of CO emissions from a burner can suggest a faulty or inefficient combustion process, which can result in wastage of fuel and reduced burner performance.

5.2 TEMPERATURE AND SPECIES MASS FRACTION PROFILES

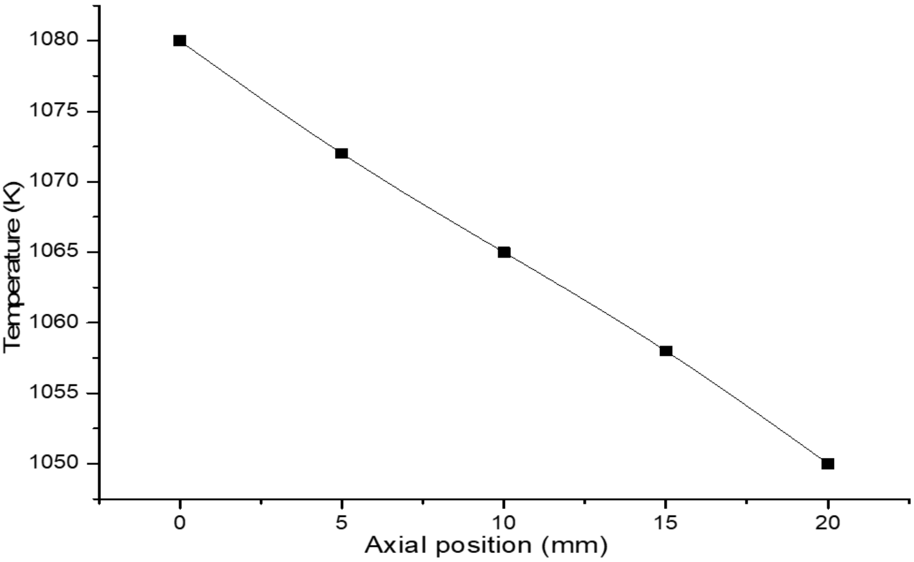

Fig 5.4 Temperature distribution along combustion zone

The radiant burner with helical combustion channels initiates combustion at the starting axial position of the combustion zone, where a maximum temperature of 1080 K is observed. This high temperature is due to the mixing of fuel and air in this region, resulting in a highly exothermic reaction. As the combustion products travel along the combustion zone, they transfer heat to the SiC matrix, which gradually reduces the temperature. This decrease in temperature is expected since heat is being transferred to the surrounding environment through conduction and radiation. By the time the combustion products reach the outlet, the temperature has decreased to 1050 K. This explains burner with helical combustion channels and SiC matrix is able to effectively initiate and maintain combustion while also controlling temperature to prevent damage to the burner and ensure high combustion efficiency.

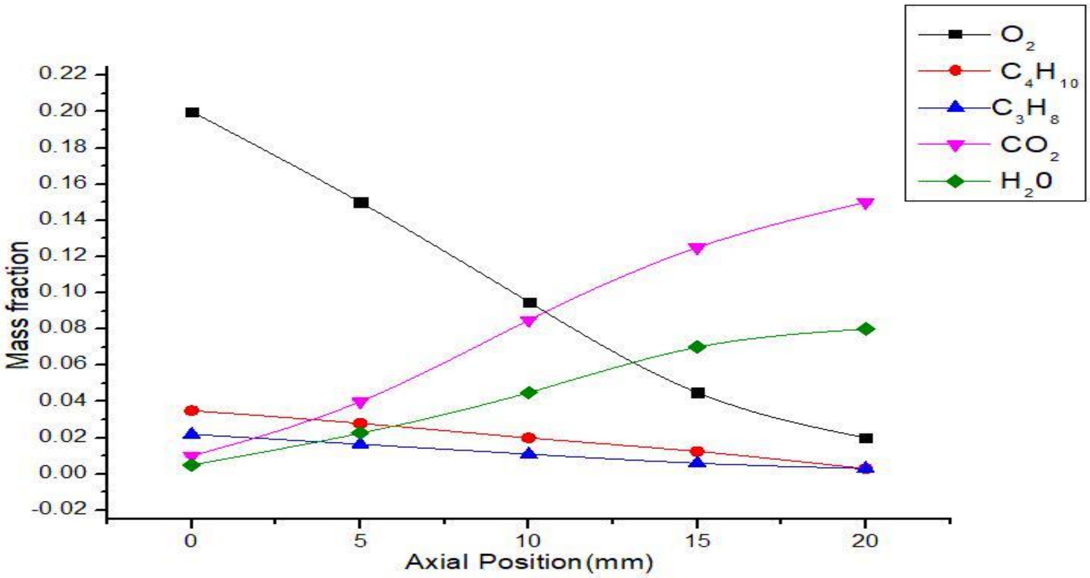

Fig 5.5 Axial distribution of the mass fraction of species

When LPG, which consists of 60% propane and 40% butane, undergoes combustion, the mass fraction of different species will change due to chemical reactions taking place. Propane (C3H8) and butane (C4H10) will react with oxygen (O2) to produce carbon dioxide (CO2), water (H2O), and heat. Consequently, during combustion, the mass fraction of propane and butane will decrease as they are consumed in the reaction. The mass fraction of oxygen will also decrease as it is consumed in the reaction. On the other hand, the mass fraction of carbon dioxide and water will increase as they are produced in the reaction.

5.3 TEMPERATURE DIFFERENCE EXIST BETWEEN SOLID AND GAS

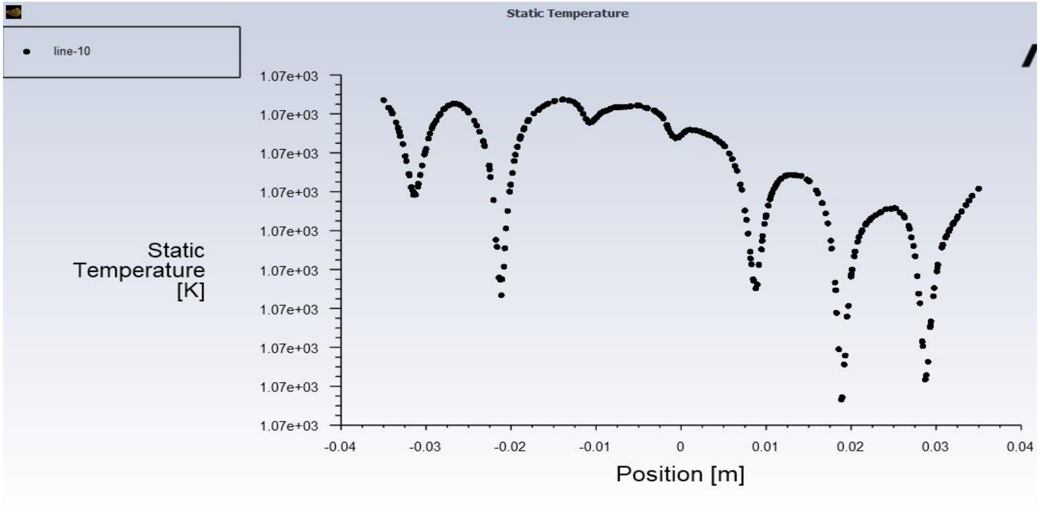

Fig 5.6 Temperature distribution along top surface of the burner

In a radiant burner with a porous material, combustion occurs within the solid material, and the heat produced by the combustion process is transferred to the gas phase by conduction, convection, and radiation. However, the transfer of heat between the gas phase and solid phase may not occur instantaneously, resulting in a thermal non-equilibrium state. This state can significantly affect the burner's performance, so it's essential to model it accurately to understand and optimize the combustion process. Due to the thermal non-equilibrium, there exists a slight temperature difference of 4 K between the SiC matrix (1066 K) and the combustion gas (1070 K). However, this difference is negligible and does not affect the burner's performance.

5.4 COMPARISON OF POROUS RADIANT BURNER AND RADIANT BURNER WITH HELICAL COMBUSTION CHANNELS

The data for porous radiant burner is taken from the journal by Mishra, N.K. and Muthukumar, P. (2018), “Development and testing of energy efficient and environment friendly porous radiant burner operating on liquefied petroleum gas.”[22]

5.4.1 COMPARISON OF OUTLET TEMPERATURE

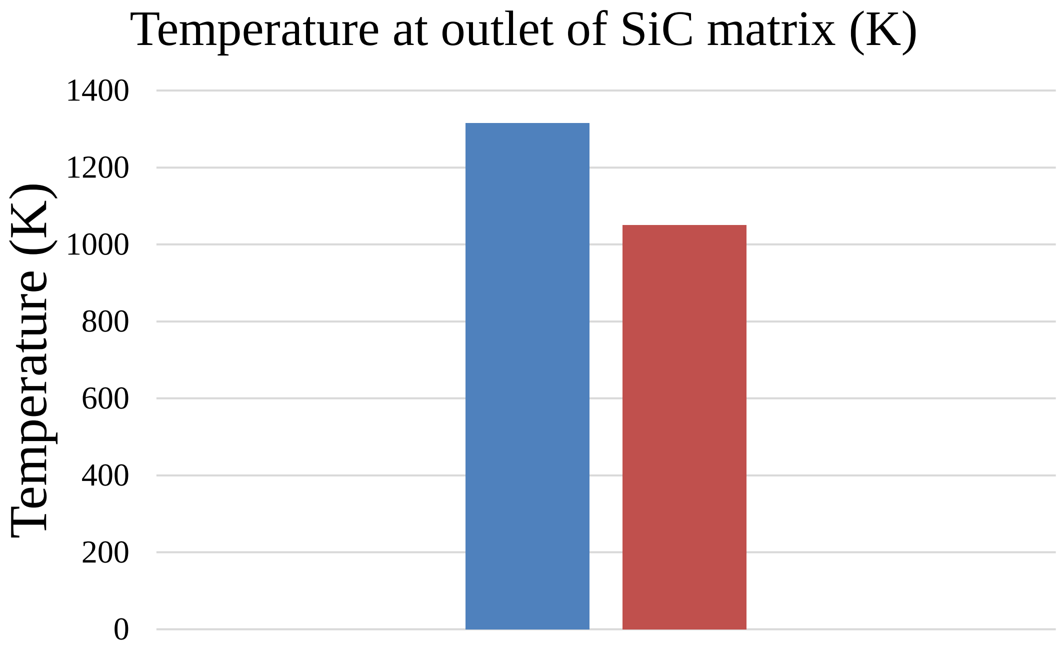

Fig 5.7 Comparison of temperature at outlet of SiC matrix

The maximum temperature at the outlet of the SiC matrix is 1380 K for the porous radiant burner (PRB), while for the radiant burner with helical combustion channels (RBWHCC), the temperature is 1050 K. The decrease in temperature is mainly due to a lower mass flow rate through the helical combustion channels. The analysis only considered 25 helical channels, which accounts for only 0.5% porosity.

5.4.2 COMPARISON OF RADIATION HEAT FLUX AT OUTLET

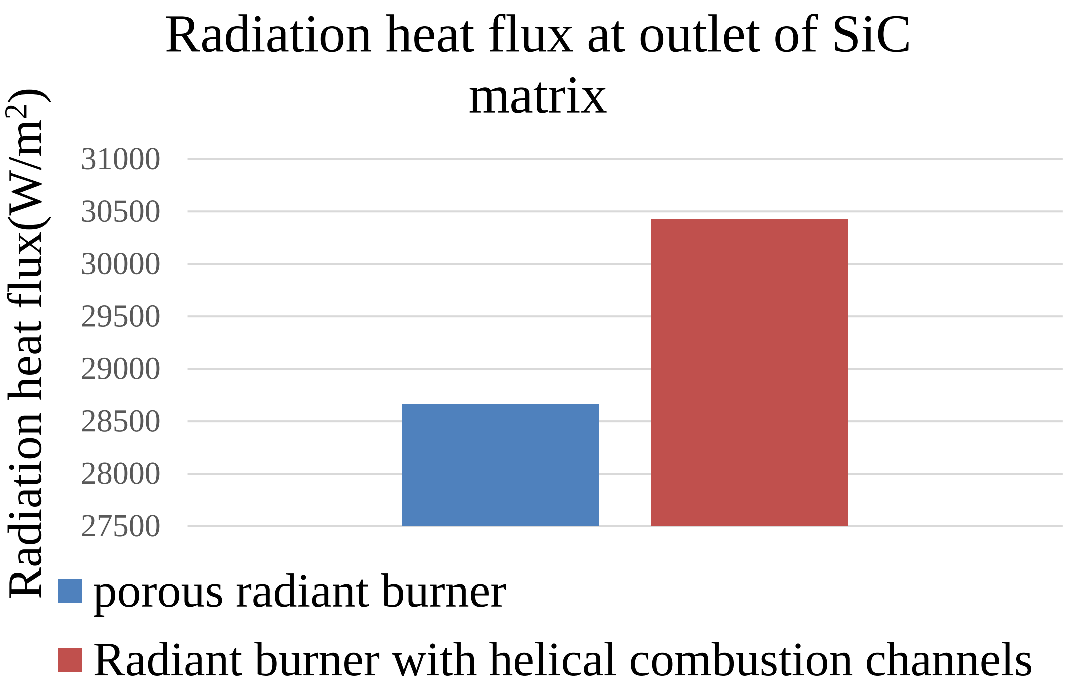

Fig 5.8 Comparison of radiation heat flux at SiC matrix

The porous radiant burner (PRB) produces a radiation heat flux of 28676 W/m^2 at the SiC matrix, whereas the radiant burner with helical combustion channels produces 30499 W/m^2. Despite the fact that the temperature obtained for the radiant burner with helical combustion channels (RBWHCC) is lower than that for the porous radiant burner, the radiation heat flux is higher. This is mainly due to several factors, including differences in burner design, fuel flow rate, combustion efficiency, and the radiation properties of the materials used. When a fluid flows through a curved channel, the centrifugal force acting on the different fluid elements, which are moving with different velocities, creates a secondary circulation. This secondary flow can increase heat transfer from combustion products to the SiC matrix, this effect also resulting in an increase in radiation heat flux.

5.4.3 COMPARISON OF MASS FRACTION OF CARBON MONOXIDE

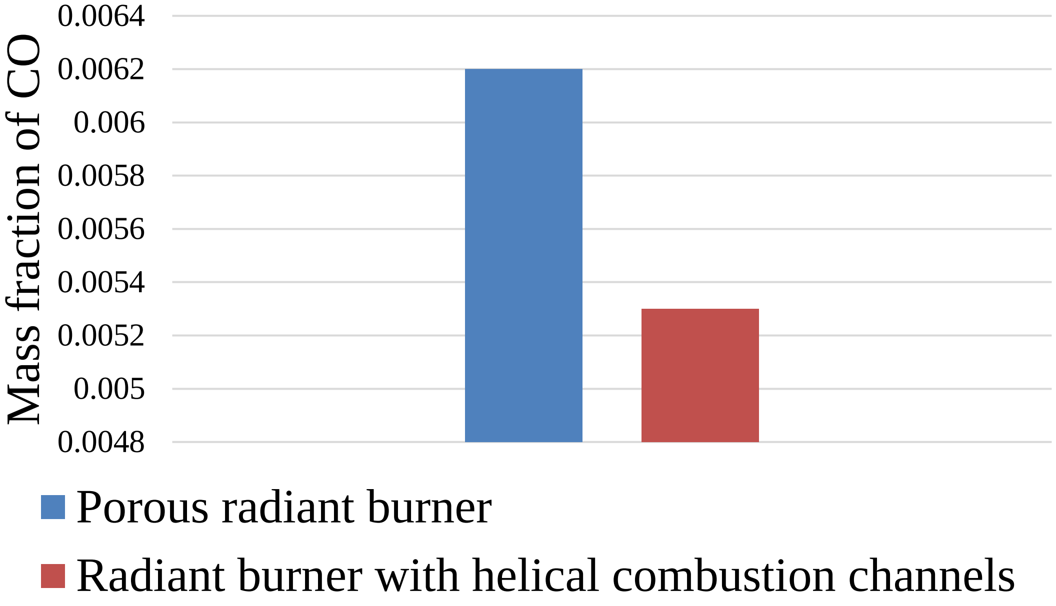

Fig 5.9 Comparison of mass fraction of CO at outlet

The mass fraction of carbon monoxide at the outlet of a porous radiant burner (PRB) is 0.0062, while in the radiant burner with helical combustion channels (RBWHCC), it is lower at 0.0053. The helical combustion channel design produces lower carbon monoxide emissions compared to the sponge-like matrix radiant burner due to various reasons. Firstly, the helical combustion channels facilitate a more efficient combustion process, leading to less incomplete combustion and consequently lower carbon monoxide emissions. Furthermore, the secondary flow generated by the curved channels in the helical combustion design enhances heat transfer from the combustion products to the SiC matrix, contributing to a more complete combustion process and lower carbon monoxide emissions. Finally, differences in burner design and flow rate may also contribute to the reduced carbon monoxide emissions in the helical combustion channel design.

5.4.4 COMPARISON OF TEMPERATURE PROFILE AT COMBUSTION ZONE

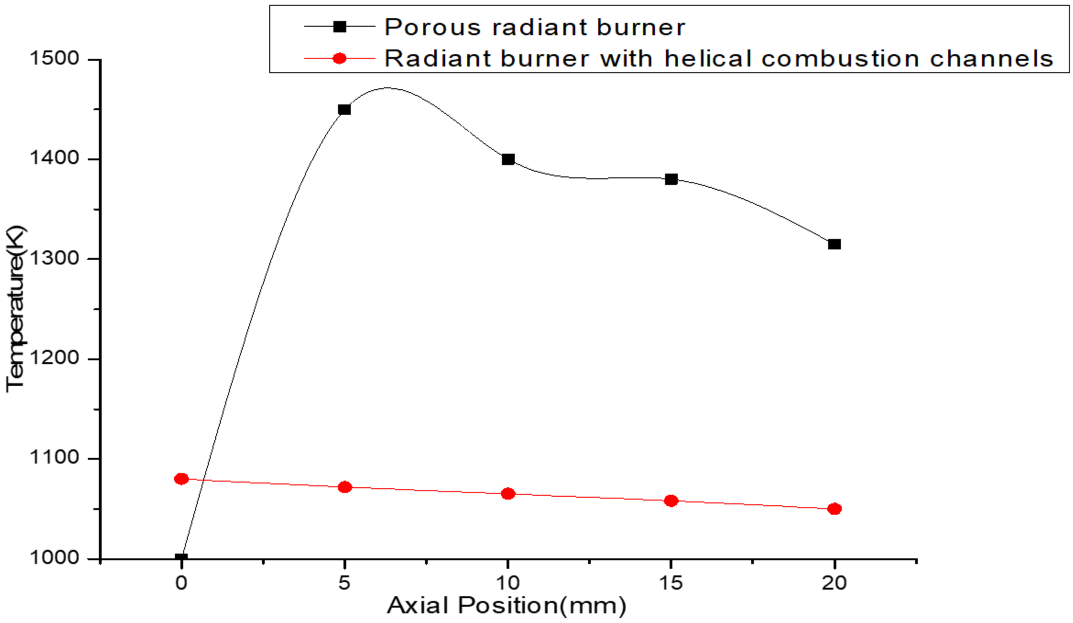

Fig 5.10 Comparison of temperature distribution at combustion zone

The provided table displays temperature values (in Kelvin) at various axial positions for two distinct types of radiant burners: a radiant burner with helical combustion channels (RBWHCC) and a porous radiant burner (PRB). At an axial position of 0 mm, the RBWHCC shows a maximum temperature of 1080 K. On the other hand, the PRB exhibits a maximum temperature of 1450 K at an axial position of 4.2 mm. This indicates that the combustion process in the PRB takes place after the start of the combustion zone, while in the case of the RBWHCC, combustion commences at the start of the combustion zone. The temperature decreases along the axial position for both burners; however, the temperature reduction is only 30 K for the RBWHCC, whereas it is 135 K for the PRB. The reason for the smaller temperature decrease along the combustion zone in the RBWHCC compared to the porous radiant burner is due to the more efficient combustion process and higher heat transfer in the former design. The helical channels in the design induce secondary flow, which results in higher heat transfer from the combustion products to the SiC matrix, contributing to a more complete combustion process. This leads to less incomplete combustion and a smaller temperature decrease along the combustion zone. In contrast, the porous matrix design in the other burner leads to a less efficient combustion process and lower heat transfer, resulting in more incomplete combustion and a greater temperature decrease along the combustion zone.

CHAPTER 6

CONCLUSION AND SCOPE OF FUTURE WORK

6.1 CONCLUSIONS

A numerical model is developed to predict the combustion of LPG in a radiant burner with helical combustion channels and compared the result with porous radiant burner. The radiative transfer equation is used to find the radiative term in the solid energy equation.

- A temperature of 1066 K was achieved in the SiC matrix by providing 25 helical combustion channels.
- An increased heat transfer rate is obtained by the presence of secondary flows and effective mixing of fuel and air in helical combustion channels.
- The sponge-like matrix porous radiant burner produces lower radiation heat flux when compared to the radiant burner that features helical combustion channels.
- Compared to the porous radiant burner, the radiant burner with helical combustion channels emits a lower amount of carbon monoxide because of increased heat transfer rate.

6.2 SCOPE OF FUTURE WORK

The following are the scope for the future research work

- Geometry optimization of the helical combustion channel can be done to increase the heat transfer rate.
- The development of 3D metal and ceramic additive printing technologies enables us to create more complex combustion channels, which could enhance the thermal efficiency of radiant burners.
- The use of materials other than SiC as burner materials for the radiant burner with helical combustion channels can be explored.

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