Research Article

Effect of injection angle of primary and secondary holes on the mixing and combustion in a can type combustor using CFD

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Abstract

Combustion is one of the vital process in power generation system and so is the combustor in a Gas Turbine. The performance of combustor has greater impact on the efficiency of the cycle, as a result combustor has always been on the prime focus for researchers to improve the efficiency of gas turbine. The present study is focused on numerical analysis, using ANSYS FLUENT, of combustion performance using slotted swirler with swirl angle of 45°. In this study, considering the importance of primary and dilution injection on combustion performance, the injection angle is varied 60° to − 60° from normal to the liner wall. Furthermore, the curvature of primary and dilution hole is formed in the form of nozzle and diffuser having convergence and divergence angle of 30° and 45°. Because of high swirling effect ($S = 1.32$), RSM (7 equations) turbulence model is used along with Non-Premixed combustion mechanism by generating β-PDF table. The performance is determined by the mass fraction of CO_2 at outlet. Higher the mass fraction of CO_2 at outlet, signifies better combustion. From the study it is reflected that the slotted swirler have better combustion performance than non-slotted swirler whereas variation of injection angle suggests, injection at 30° clockwise is more suitable. Also, primary injection done through nozzle channel (convergence angle of 45°) and dilution injection through diffuser channel (divergence angle of 45°) results in most efficient combustion performance.

Keywords Combustor · Flow characteristics · Combustion · Efficiency · CFD

List of symbols

k	Turbulent kinetic energy (m^2/s^2)
ε	Turbulent dissipation energy (m^2/s^2)
ϕ	Equivalence ratio
θ	Angle of injection (degree)
m_0	Mass of air through swirler (kg/s)
T_0	Temperature of air at inlet (K)
q	Amount of heat released (W)
T_A	Temperature of air in primary zone (K)
C_p	Specific heat of air (kJ/kg-K)
S	Swirl number
ρ	Density of air (Kg/m^3)
u	Velocity component in x-direction (m/s)
v	Velocity component in y-direction (m/s)
w	Velocity component in z-direction (m/s)
Ω	Ratio of inner radius of swirler to flow field
α	Vane angle of swirler
V_r	Velocity components in radial direction (m/s)
V_θ	Velocity components in azimuthal direction (m/s)
V_z	Velocity components in axial direction (m/s)
f	Mixture fraction
P	Pressure (Pa)
Tu	Turbulent intensity
CTRZ	Central toroidal recirculation zone
CIVB	Combustion induced vortex breakdown
PDF	Probability density function
UHC	Unburnt hydro-carbons

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1 Introduction

Combustors are designed for stable combustion against blowout [1], better combustion and minimum pressure drop. Towards the objective different designs and models were proposed with time. High re-circulation zone in the primary zone, where fuel is mixed, atomized and ignited. In this zone the swirled flow through the dome creates low pressure, high turbulence re-circulation zone for flame holding. Secondary zone provides excess air so that fuel molecule like CO and radical H can be fully oxidized. Temperature in this zone is still more than the material limit of turbine blade so it is further diluted in the dilution zone, where the exit temperature is brought within material limit of turbine blade [2].

The basic aerodynamics and geometric features of the combustor were reviewed focusing development it has brought in combustor performance. The first-generation combustors were characterized by the variation in geometry, size and mode of fuel injection. But with time, the design philosophy emerged and by 1950, the basic features of conventional gas turbine combustor were firmly established. With recent developments, combustor performance has improved significantly. The operating pressure have improved from 5 to 50 atm, inlet temp of oxidant has risen to 900 K and the temperature at outlet has risen to 1850 K, resulting in improved efficiency.

Study [3] suggested that mixing and flame stability improves with increasing swirl no. to 1.0, but at higher swirl no. the turbulence level and flame stability is reduced. Also, high swirl, forces the flow to move upstream, closer to wall, resulting in excessive wall heating. An unconfined strongly swirled flow was investigated for different Reynolds numbers using PIV and LES with a Thickened Flame (TF) model [4] and both reacting and non-reacting flow results were presented. Using LES [5] it was observed that an accurate description of the inlet section of the combustor plays a fundamental role in the prediction of the fuel placement and temperature distribution in primary region also flame stabilization does not rely on re-circulation, but on flow divergence [6] and swirl injector creates a central toroidal recirculation zone (CTRZ) which serves as a flame stabilization mechanism [7]. It was also predicted that combustion in a Can-type combustor using producer gas as fuel, aimed to optimize the characteristics of combustor for turbocharger test rig [8]. The annular combustion chamber in a gas turbine engine for a low by-pass turbofan engine with distributed mass flow rate of combustor air were designed and variation of parameter at different stages were numerically evaluated [9]. To further improve the combustion performance, variable geometry dual-mode combustor was examined [10, 11]. A numerical study for performance improvement of a variable geometry dual mode combustor by optimizing deflection suggests that interaction between shock caused by wedge system and heat released by combustion results in improved combustion efficiency and total pressure recovery coefficient with the deflection angle and maximum is obtained at 12° deflection angle [12]. Influence of fuel injection method on performance of upward swirl can-type combustor states reverse fuel injection provides rapid mixing and better fuel penetration. Further the effect of swirl field on fuel concentration distribution and combustion characteristics were examined in a gas combustor with cavity and resulted in high efficiency and low emission combustion [13]. A numerical study [14] on single and double layered wall micro combustor combustion efficiency was compared and it was concluded that high recirculation is enhanced and heat loss is reduced in double layered combustor [15]. Numerical calculation has been performed to analyze the combustion flow in different advanced vortex combustors. For the rear blunt body, when slot size is 2 mm and half open angle θ is 50°, advanced vortex combustor has the best combustion flow characteristics [16]. The effects of fuel–air equivalence ratio Φ and fuel flow rate Q_{CH4} on chemical emissions and thermodynamic properties of a thermoacoustic swirl combustor was also examined. All these studies indicate that the interaction of individual component results in very complex flow field, at the combustor exit. The non-uniformities in pressure and temperature profile is significantly contributed by swirler, film cooling slots, primary holes and dilution holes [17]. The results of the above-mentioned research studies also indicate that in a combustor the thermal and flow field in the primary zone are very similar. The noteworthy difference can be seen in the upstream of primary zone where the combustion is stabilized [18]. Because of limited information on thermal and flow field in secondary and dilution zones and variations of injection inclination from normal plane. The effect on recirculation zone, flame stability and combustion performance can be further analysed for different configuration of primary and dilution air injection.

2 Methodology

Primary holes are located at 105 mm downstream of swirler, as shown in Fig. 1. These holes are distributed along its circumference. Diameter of each primary injection holes are 12.5 mm. Similarly, the dilution holes are also distributed along the circumference at 200 mm from swirler. Total number of dilution holes are 20 whose

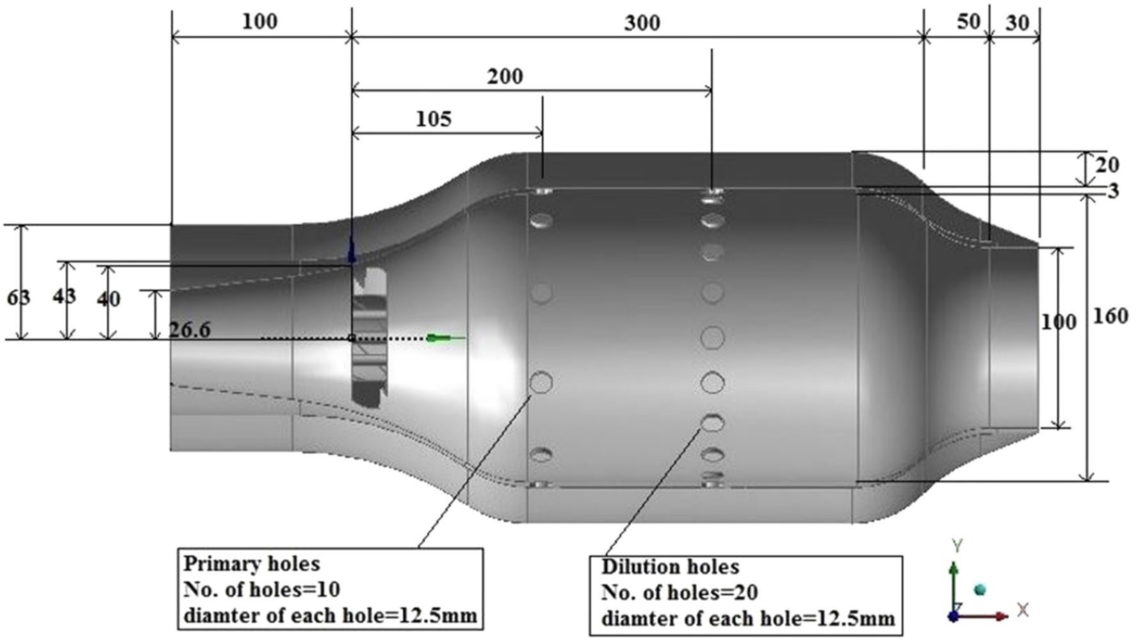


Fig. 1 Dimensions of model Can-type Combustor used for study

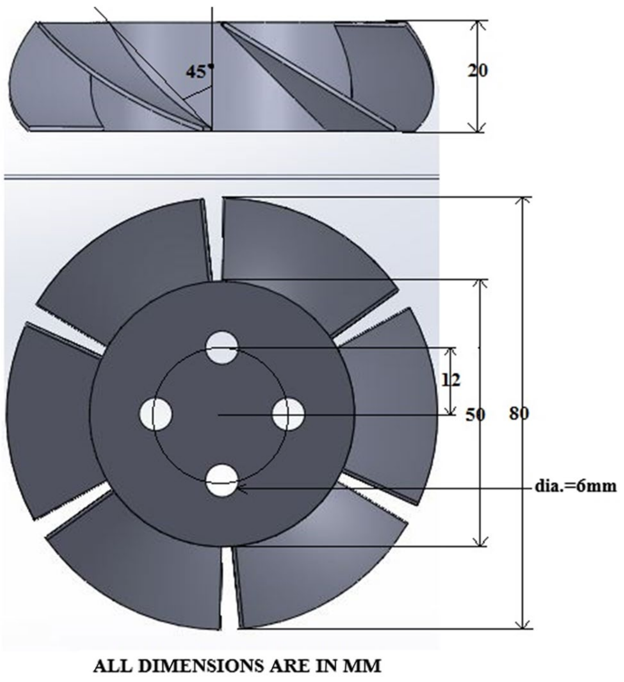


Fig. 2 Swirler with slots

diameter is 12.5 mm. The diameter at outlet of combustor is 100 mm. As per calculation, for the geometry used in the study the mass flow rate of air passing through diffuser is 24.33% and that of through casing is 75.67%, which is in accordance to the literature [19, 20]. 80 mm swirler diameter, with vane angle of 45° is used in the study has slots of 6 mm diameter which are equally distributed at radial distance of 12 mm as shown in Fig. 2.

With this geometry, combustion performance is analysed by modifying primary and secondary injection. These variations include changing injection angle of primary and dilution holes. The injection angles undertaken for study are 30°, 45°, 60°, −30°, −45°, −60° as shown in Fig. 3. Converging and diverging angle of primary and dilution hole are also varied [21]. These combinations are represented in Fig. 4 and tabulated in Table 1.

An unstructured grid with tetrahedron element has been generated using patch conformal method. Skewness of the triangular/tetrahedral mesh in most flow must be kept below 0.85, with an average value that should be less than 0.33 [22]. Aspect ratio is the measure of stretching of the cell. Regarding the stability of energy solution, the maximum ratio should be kept below 35:1 [22]. The element size ranges from 0.139 to 2 mm also the aspect ratio varies from 1.159 to 12.707 with an average of 1.842. The number of elements after grid independency test range from 1.808 to 1.884 million, refer Fig. 5.

2.1 Mathematical formulation

The significance of the central recirculation zone formed after the swirler are longer residence time, which provides longer duration for combustion. Higher mixing rate and elevated temperature are also the features which are observed due shear layer formation between mainstream and back flow. The swirl number of the flow

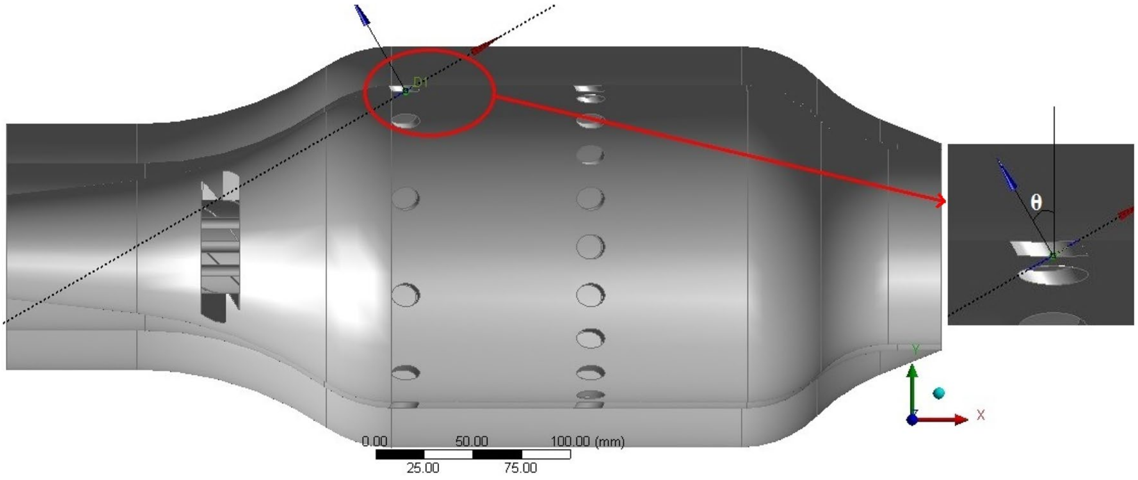


Fig. 3 Geometry of primary and dilution injection at an angle θ, where θ varies as 30°, 45°, 60°, – 30°, – 45°, – 60°

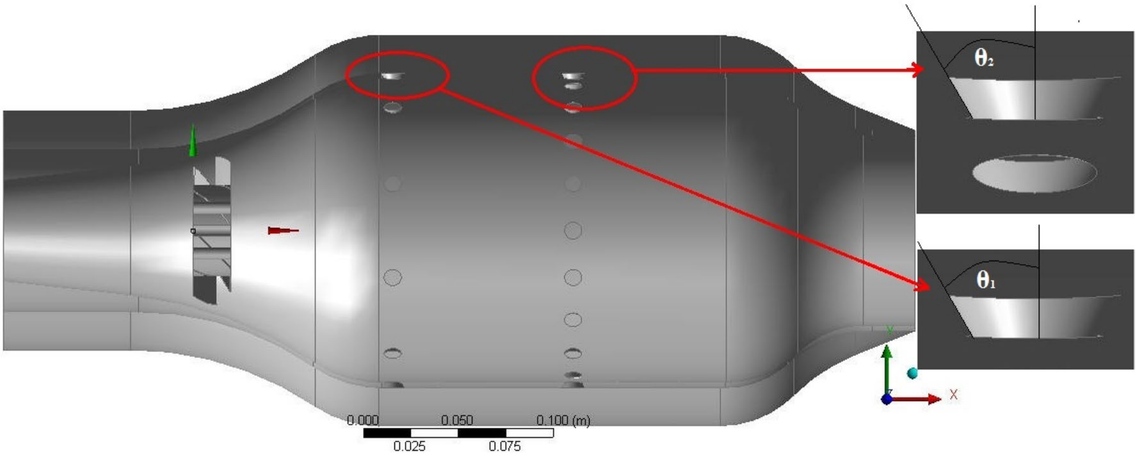


Fig. 4 Geometry for analysis by varying converging and diverging angle of Primary holes as θ1 and Dilution holes as θ2

Table 1 Cases analysed by varying angle θ1 and θ2

Cases	θ_1	θ_2
A	30° converging	30° converging
B	45° converging	45° converging
C	30° diverging	30° diverging
D	45° diverging	45° diverging
E	30° converging	30° diverging
F	45° converging	45° diverging

through swirler with vane angle 45° and 6 blade comes out to be 1.32. This suggests that as the swirl number is high, Reynolds Stress Model (RSM) will capture the vortex more precisely [23]. This has also been validated in comparison with other turbulence models.

2.2 Governing equations

To solve the phenomenon following set of governing equations used. The Axial Navier–Stokes equation expressed in radial—r, azimuthal—θ and axial—z direction is numerically solved to obtain the flow velocity components in r, θ and z direction. Pressure correction technique is implemented assuming pressure distribution to estimate velocity components and then correcting the pressure gradient to satisfy continuity equation.

In the present study because of high swirl Reynolds Stress Model (RSM) (7 equation RSM turbulent model) is used [24]. This is the most elaborate RANS model which do not follow the isotropic eddy-viscosity hypothesis. Since the RSM accounts for the effect of streamline curvature, swirl, rotation, and rapid changes in strain rate in more rigorous manner than one equation or two

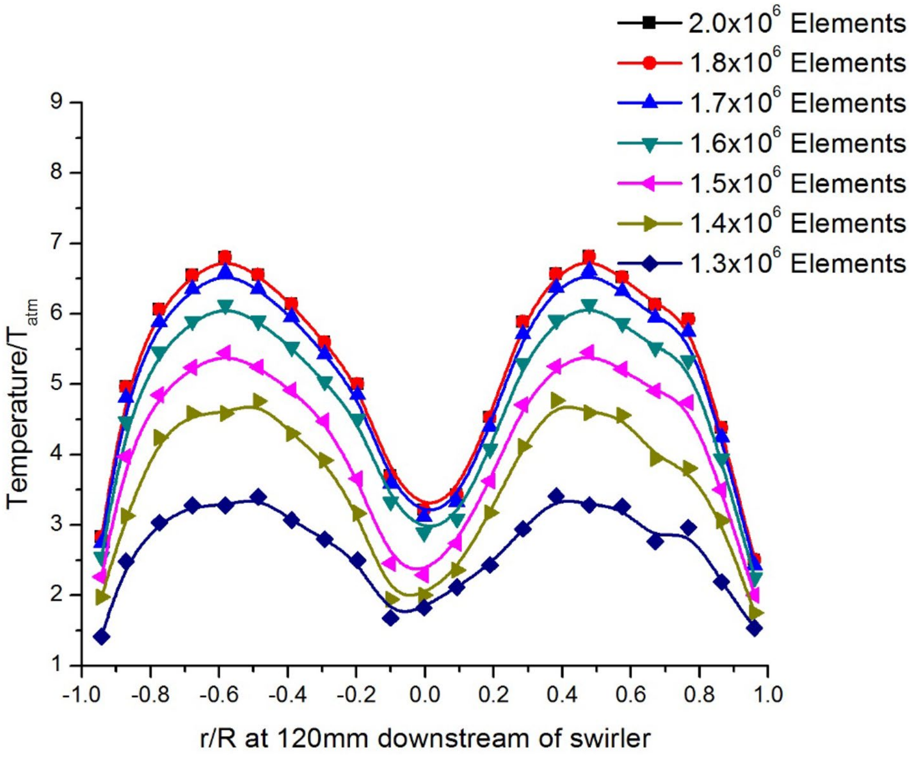


Fig. 5 Grid independency test

equation models, it has greater potential to give accurate predictions for complex flow [25, 26]. However, the fidelity of RSM prediction is still limited by the closure assumptions employed to model various terms. The modelling of pressure strain and dissipation rate term is challenging, and hence responsible for RSM prediction. The Reynolds stress model are non-linearity eddy viscosity models and are usually referred to as second moment closure.

One additional set of transport equation for energy dissipation rate is solved to have a solvable set of equations [27]. Closure coefficients and auxiliary relations are: C1 = 3.4; C1* = 1.8; C2 = 4.2; C3 = 0.8; C3* = 1.3; C4 = 1.25; C5 = 0.4

2.3 Combustion model

Non-premixed modeling involves the solution of transport equation for one or two conserved mixture fractions. Instead of solving individual species, species concentrations are derived from predicted mixture fraction fields. The turbo-chemistry calculation is pre-processed and tabulated for look-up in FLUENT. The interaction of turbulence and chemistry is accounted for with an assumed shape Probability Density Function (PDF) [22].

2.4 Fluid properties and boundary conditions

The fluid passing through combustor inlet is air whose density, ρ is equal to 1.23 kg/m3, and kinetic viscosity, υ is equal to 1.789 e^{-5} m^2/s. The fuel used for combustion is aviation kerosene i.e. C12H23. Various other parameters have been tabulated in Table 2.

Fuel is injected through DPM method where the point of injection is located at 5 mm upstream of swirler. The particle type of the fuel is droplet type, with group injection of 20 number of streams. Number of continuous phase iteration per DPM iteration is 5 [28]. The material type of air–fuel is droplet mixture. For defining turbulence quantity turbulence intensity and hydraulic diameter method is selected. The hydraulic diameter of the given problem is

Table 2 Parameters for present study [22]

Parameters	Values
Mass flow rate of air	0.6 kg/s
Reynolds number of main flow	110,000
Mass flow rate of fuel	0.0015 kg/s
Air inlet temperature	450 K
Fuel temperature	293 K
Atomization Pressure	0.6 MPa
Diameter of liquid fuel droplets	0.04 mm
velocity of fuel	65.63 m/s

0.028 m whereas the turbulent intensity is calculated using following equation [29], Tu % = 0.16 $Re^{(-1/8),}$ which comes out to be 3.75% (Table 3).

2.5 Validation

The turbulence model is validated from the experimental result proposed by Li et al. Keeping the same boundary conditions and solver setting as stated in the previous headings, a computational study was done using K-e, RNG K-e and RSM turbulence model. Figure 6 depict the temperature profile obtained by the above-mentioned models at 120 mm downstream of swirler. It is concluded that RSM model gives result in accordance to the experimental values. The maximum error spotted by this model is 12.8% at one test location. The error among the solver models and the experimental result exists because, the complex swirl zones formed in the combustor could not be captured beyond certain extent. Although the RSM model employs 7 equations to capture high swirl zone but still the error exists because of the Taylor's series truncation error in the discretization scheme. Based on these results for the further study RSM Turbulence Model is used along with non-premixed chemical equilibrium model.

Table 3 Cases obtained by varying converging and diverging angle

Cases	θ_1 (primary injection)	θ_2 (dilution injection)
A	30° converging	30° converging
B	45° converging	45° converging
C	30° diverging	30° diverging
D	45° diverging	45° diverging
E	30° converging	30° diverging
F	45° converging	45° diverging

3 Results and discussion

During the investigation, it is observed that a large recirculation is formed in the primary zone. This recirculation zone enhances the mixing, vaporization and combustion of fuel. At the same time, near wall recirculation zone is observed in the secondary zone. In this zone dissociation of UHC takes place, hence it is of vital importance as it gives ample time for dissociation. In the dilution zone the addition of fresh air from secondary zone enhances the complete combustion, uniformity in temperature profile and assist in cooling of liner material.

3.1 Combustion performance of combustor with slotted and non-slotted swirler

Combustion is carried out in Can-type Combustor model with non-slotted and slotted swirler. This investigation is carried out for the combustors with same boundary condition at input and the temperature profile in primary zone (at 90 mm downstream of swirler), secondary zone (at

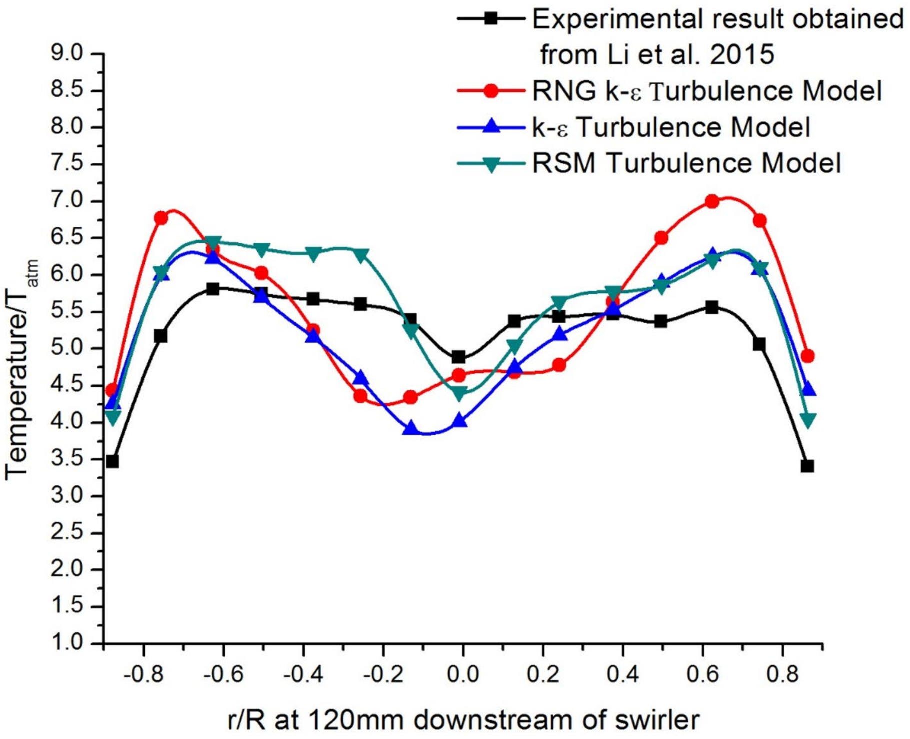


Fig. 6 Temperature profile at 120 mm downstream of swirler using K-e, RNG K-e and RSM Turbulence model

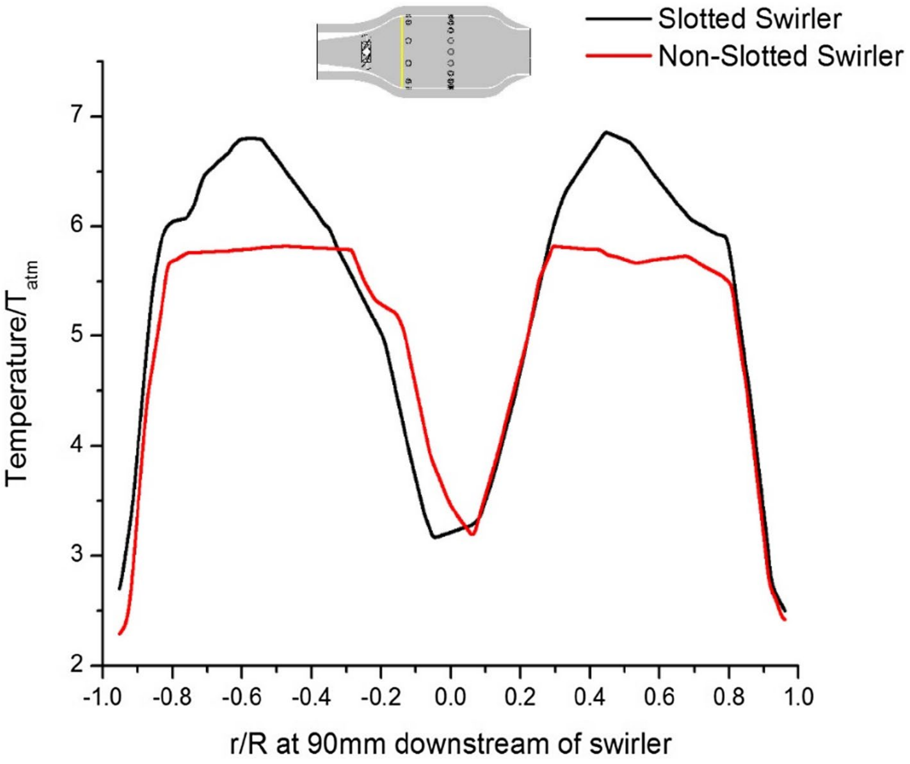


Fig. 7 Temperature profile in primary zone for slotted and non-slotted swirler

150 mm downstream of swirler), dilution zone (at 250 mm from swirler), and at outlet, refer Fig. 7. In the primary zone temperature profile for slotted swirler is higher than that of non-slotted swirler justifying the thermodynamic analysis made under Methodology. In the slotted swirler the shear stress developed between the layers due to jet from the slots of the slotted swirler and CTRZ augments the oxidation process which results in higher temperature profile.

In the secondary zone, the dissociation of unburned hydrocarbons due to mixing of charge between primary and secondary recirculation results in further completion of combustion. The temperature in this zone raises as the high swirling flow mixing increases from centreline towards the liner wall. In the slotted swirler this zone is larger than that of non-slotted swirler which intensifies the rate of dissociation as a result the temperature of slotted swirler combustor is higher than that of non-slotted combustor. Figure 8 identifies these zones of recirculation and temperature contours in these zones, which indicates that recirculation augments the combustion. From the velocity vectors, recirculation zones are distinctly visible in primary zone and the temperature contours shows that centre of

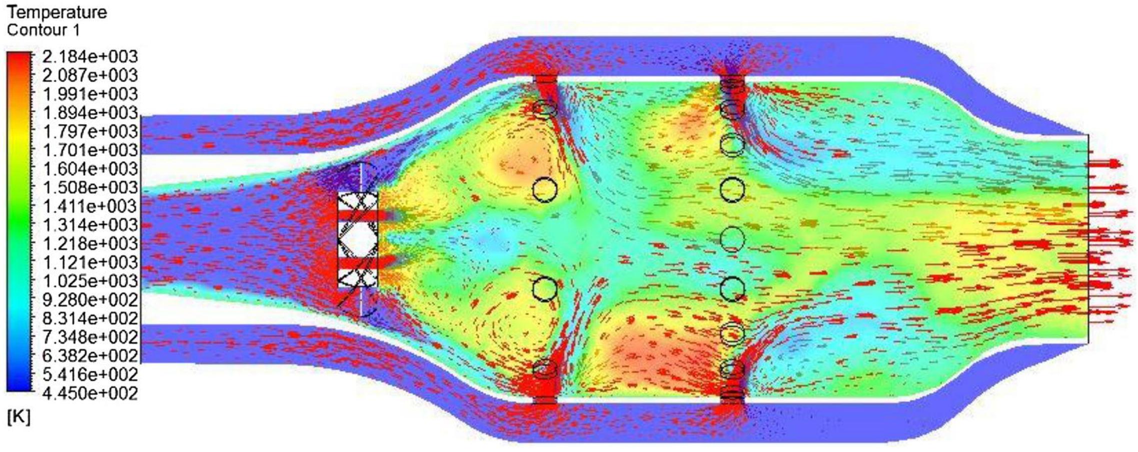


Fig. 8 Temperature contour and velocity vector for slotted swirler combustor

combustion lies in that zone only. In the dilution zone, more fresh air is added to dilute the combustion products and introduces uniformity in temperature profile so that the temperature of the combustion products is well within turbine material specifications. Also, dilution helps in liner wall cooling, these effects can easily be seen from Fig. 8.

In the dilution zone, the temperature distribution for non-slotted swirler temperature is higher than slotted swirler. This is because there is more unburned hydrocarbon in the zone for non-slotted swirler and addition of more air results in combustion of those unburnt hydrocarbons in the dilution zone, which is highly undesirable. At outlet, temperature profile of slotted swirler is more uniform and higher than that of non-slotted swirler.

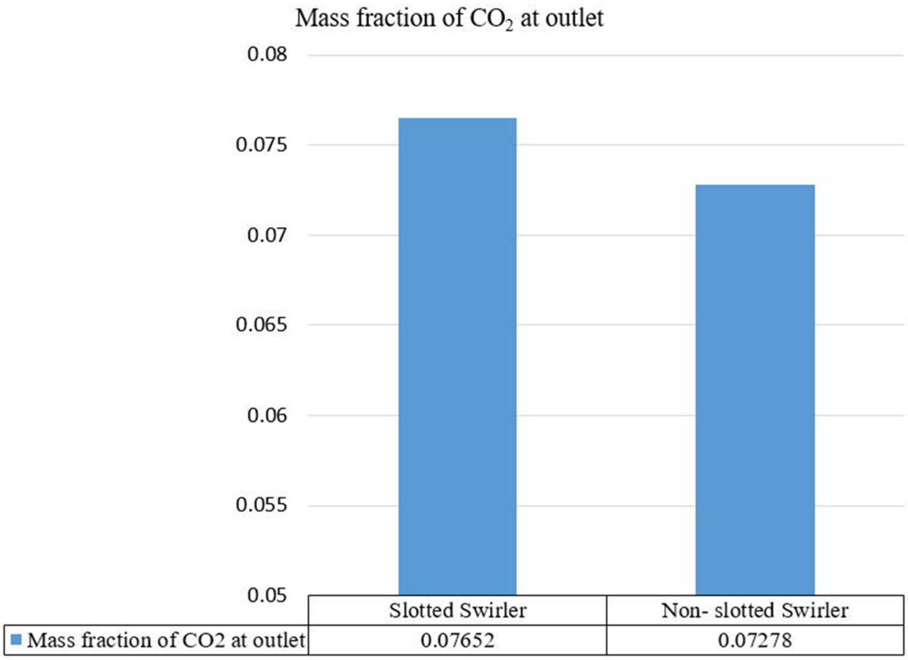
Along with temperature, pressure variation also have its significance in the combustion process. As per standard Brayton cycle the combustion should occur at constant pressure, which is practically not possible. But one can always design for minimum pressure drop. The introduction of swirler at the inlet inherit some drop in pressure [30]. Pressure variation along the streamline shows that there is sudden pressure drop across swirler in both the cases, but the overall % pressure drop for non- slotted swirler is more than that of slotted swirler. For slotted swirler pressure drop is 16.6% whereas for non-slotted swirler it is 19.7%. This result also advocates the applicability of slotted swirler.

Figure 9 indicates mass fraction of CO_2 at the outlet of the combustor, for the quantification for completion of combustion. Higher mass fraction of CO_2 at the outlet can be related as more dissociation of CO and hence more efficient combustion [31, 32]. The figure shows that the mass fraction of CO_2 for slotted swirler is more when compared to that of non-slotted swirler. Hence it can be concluded that combustor with slotted swirler have better combustion performance than that of non-slotted swirler.

3.2 Combustion performance of combustor by varying the primary and dilution injection angle

Other than swirler, the size of recirculation zone is deeply affected by the injection angle for primary and dilution injection. The injection angle of primary and dilution hole is varied in anti-clockwise direction to 30°, 45° and 60° and then in clockwise direction from normal to the liner surface to – 30°, – 45° and – 60°. These variations assisted us in analysing the importance of the recirculation zones. Figure 10 shows temperature contours along with velocity vectors for injection at 30°. As we move in anticlockwise direction from normal to 30°, 45° and 60° the recirculation zone becomes feebler. The effect of which can easily be seen in combustion performance, as mass fraction of CO_2 keeps degrading, Fig. 12.

In similar fashion, as we move in clockwise direction from normal to – 30°, – 45°, – 60°, it is observed that the size of recirculation zone in primary zone for – 30° injection, Fig. 11, is slightly larger than that of normal injection, Fig. 8, but as we move further to – 45° and – 60° the strength subdued significantly. It shows that ignition in the primary zone of the can is better in case of injection at – 30° but keeps tumbling for the other two cases.



	Slotted Swirler	Non- slotted Swirler
■ Mass fraction of CO2 at outlet	0.07652	0.07278

Fig. 9 Mass fraction of CO_2 at outlet of the combustor

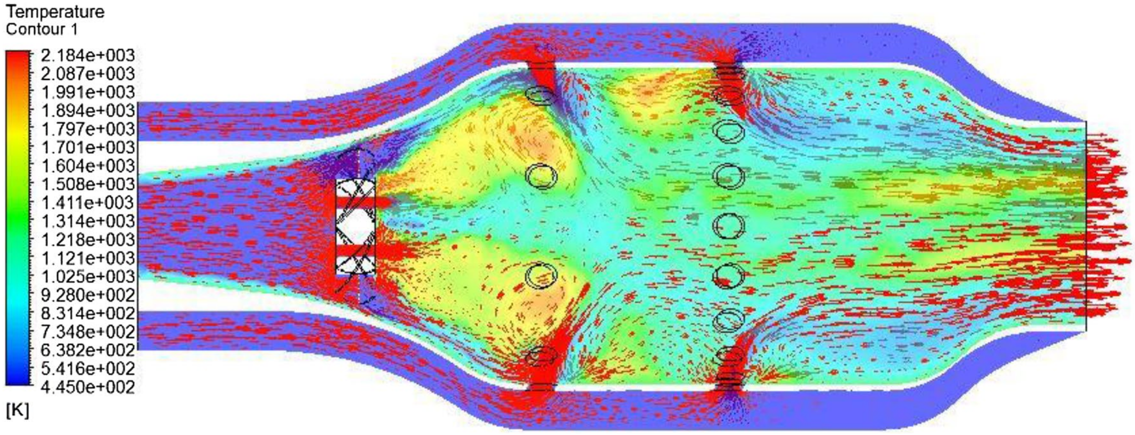


Fig. 10 Temperature contour and velocity vectors for injection at 30° from normal

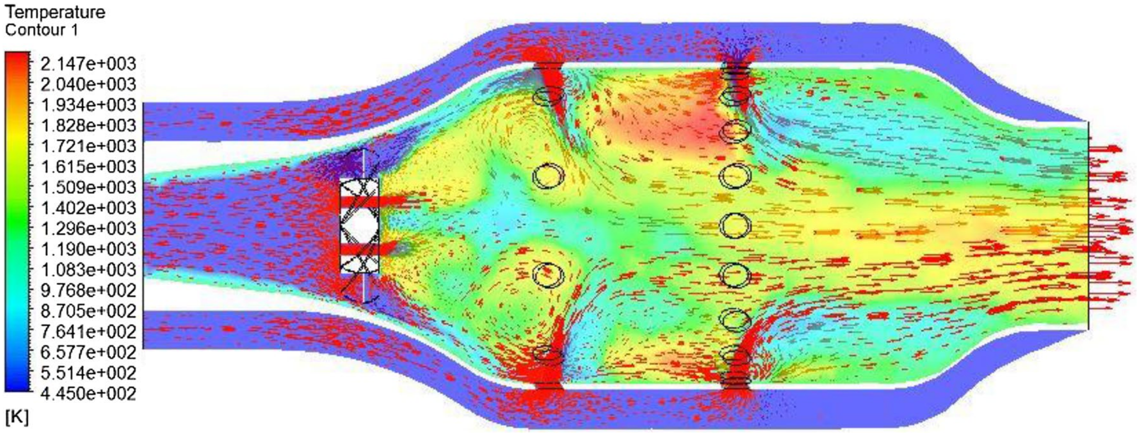
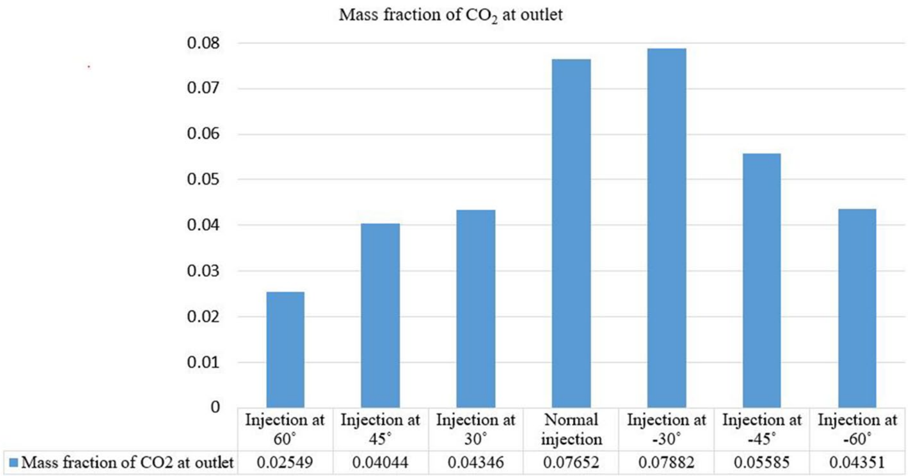


Fig. 11 Temperature contour and velocity vectors for injection at – 30° from normal

Fig. 12 Mass fraction of CO2 at outlet of combustor at different injection angle



The inclination of injection port from normal affects the level of penetration of primary and dilution air towards the centreline of the combustor. The size recirculation zones is reduced because of the reduced penetration. This phenomenon is distinctly visible in below mentioned contours, where penetration from primary holes are shown with the help of velocity vectors for the above-mentioned cases. It is observed that as the injection angle increases (away from normal), the penetration of velocity vector decreases and hence recirculation zone gets weaker.

The combustion performance is determined by the mass fraction of CO_2 at outlet, as it shows completion of combustion. For the above-mentioned cases Fig. 12 shows that combustion performance is more efficient when the primary and dilution injection angle is – 30°.

3.3 Combustion performance by varying the converging and diverging angle of primary and dilution injection in the form of nozzle and diffuser

By varying the converging and diverging angle of primary holes and dilution hole, following cases are taken under consideration.

At first both the primary and dilution injection was kept converging (like nozzle) and angle is varied to 30° and 45°. It is observed that higher the converging angle of primary injection higher is the penetration and larger recirculation zone is formed which boosts the combustion process. On the other hand, in case of dilution injection, higher penetration of flow towards centreline will affect the liner wall cooling and the combustion product is not cooled uniformly. Also, a larger recirculation zone is formed in secondary zone which enhances combustion but simultaneously adversely affects the combustion products and temperature profile at outlet, resulting in low mass fraction of CO_2 at outlet. The combustion performance is not considered to be efficient keeping in mind that the liner material will not sustain that high temperature for long.

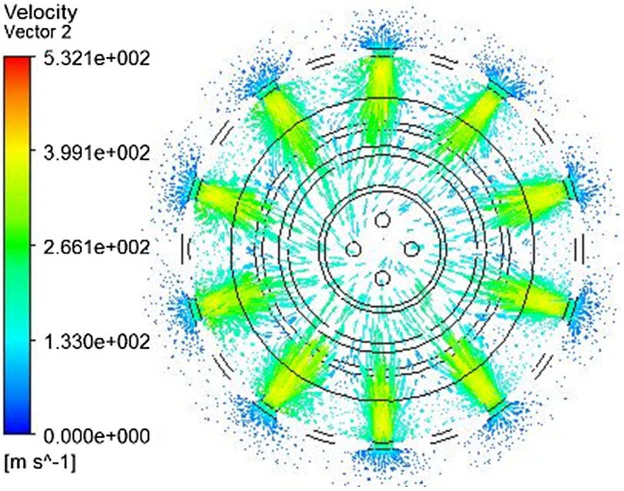


Fig. 13 Primary injection through converging angle of 30°

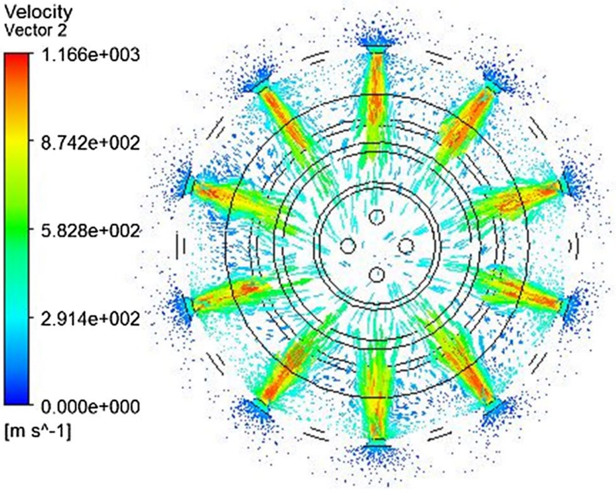


Fig. 14 Primary injection through converging angle of 45°

Figures 13 and 14 shows the depth of penetration of flow through primary injection at 30° converging profile of primary hole (Fig. 15). It can also be easily witnessed that for convergence angle of 45° the depth of penetration is more when compare with 30° convergence. The depth of penetration affects the size of recirculation in primary zone. More is the depth of penetration larger is the size of recirculation, and larger the recirculation higher will be the combustion rate. The combustion performance is measured on the scale of CO_2 mass fraction at outlet. Mass fraction of CO_2 at outlet for both above cases is shown in Fig. 16, stating that the combustion performance is more efficient when the angle of convergence is 45°.

Next for the other two case, the primary and dilution injection is kept diverging (like diffuser) and their angle of divergence is varied to 30° and 45°. In these two cases, it is observed that when the primary injection is through diverging channel, flow penetration is lesser than that of through converging channel. As a result, recirculation zone is smaller but since the dilution hole is also through diverging channel, it helps in uniform mixing of air to the combustion product and enhances the combustion process in both secondary and dilution zone. It also helps in cooling of liner wall.

So far, the study suggests that in primary zone for larger recirculation converging channel is more beneficial and in dilution zone, for proper wall cooling and uniform mixing of combustion products with fresh air through dilution hole divergent hole is helpful. Based on these studies the other two cases undertaken for study is combustor with primary injection through converging hole and dilution through diverging hole, by varying than angle to 30° and 45°.

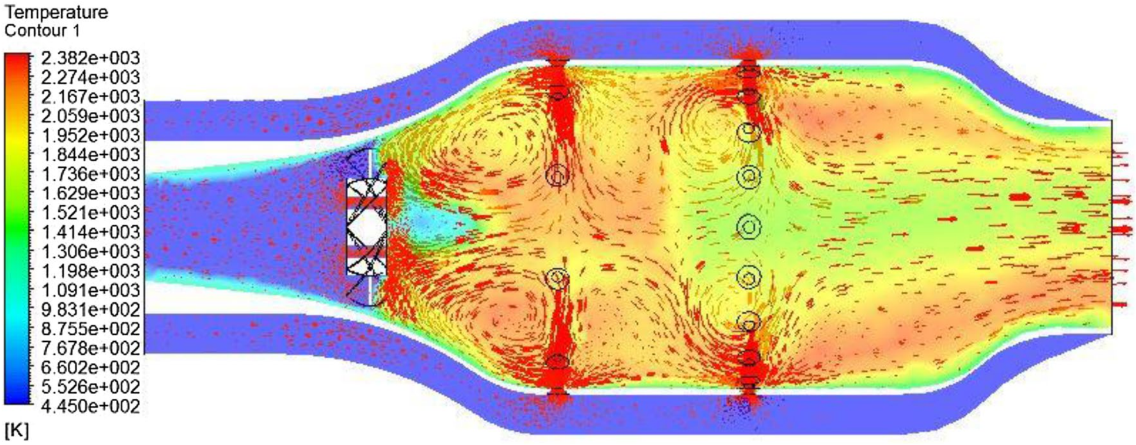
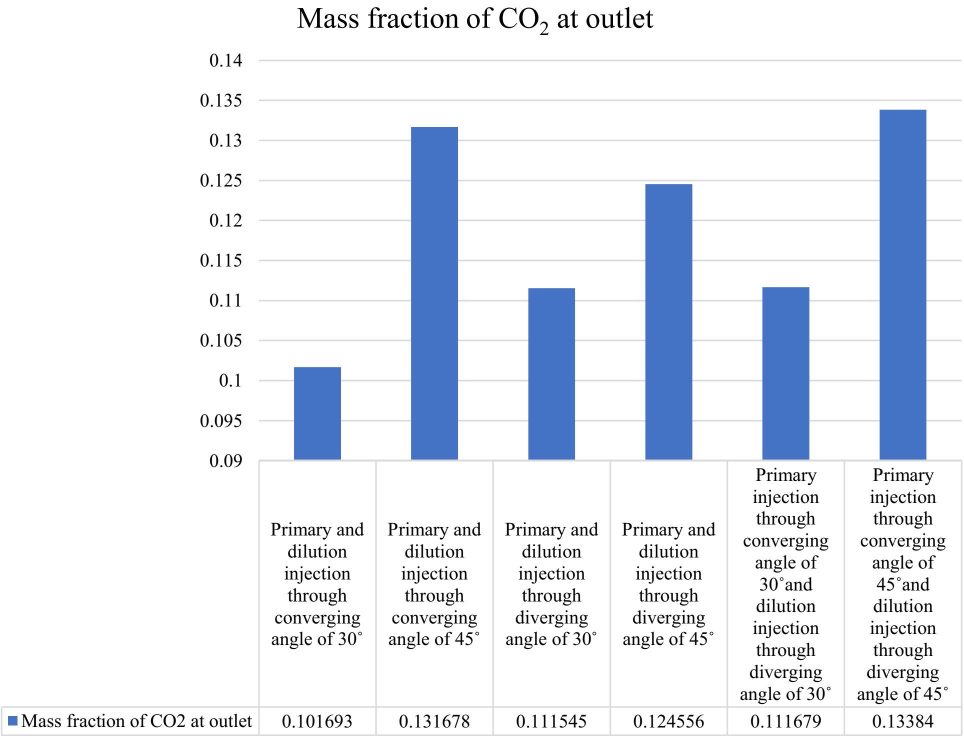


Fig. 15 Temperature contour and velocity vectors when primary injection is through converging hole of 45° and dilution is through diverging hole of 45°

Fig. 16 Mass fraction of CO_2 for case A, B, C, D, E, F



For all the above cases, to investigate the combustion performance, mass fraction of CO_2 at outlet of combustor is a means of measure. The more efficient the combustion is higher will be the mass fraction of CO_2. As shown by Fig. 16, combustor with primary injection through a converging channel of 45° and dilution injection through diverging channel of 45° is the most efficient design.

4 Conclusions

In this study after validating the turbulence and combustion model from [22], the effect of slotted swirler inside a can-type combustor model, in the environment of primary and dilution injection, is analysed and combustion performance is estimated based on flue gas analysis. The analysis suggests that higher mass fraction of CO_2 at outlet reflects more efficient combustion.

From study 1, it is concluded that the recirculation zone formed by the slotted swirler is larger than that of non-slotted swirler and higher shear due to the jets from the slots of swirler augments the mixing of charges. This higher mixing boosts the ignition and flame propagation inside the combustor. The temperature in primary zone after combustion for slotted swirler is (Maximum temperature is 2314 K) higher than non-slotted swirler (Maximum temperature is 1958 K) and hence higher mass fraction of CO_2 i.e. 0.07652, at outlet indicating efficient combustion in can-type combustor with slotted swirler. Pressure drop being the other measure to analyse the performance also suggest the application of slotted swirler results in small drop in pressure, which gets recovered by the outlet.

In the second study, the injection angle of primary and dilution hole is varied in both clockwise and anti-clockwise direction from normal to the liner surface. The injection angle is varied from 30°, 45°, 60°, −30°, −45°, −60°. After investigation it is reported that the combustion performance for −30°, i.e. mass fraction of $CO_2 = 0.07882$, is slightly more than that of normal injection i.e. mass fraction of $CO_2 = 0.07652$. For the remaining cases as the angle of inclination from normal increases the mass fraction of CO_2 decreases at outlet, causing poor performance.

In the third study, the primary and dilution injection is made through converging and diverging channels. After the analysis of number of cases, it is noteworthy that convergence channel of primary injection supports larger recirculation zone and enriches ignition and flame propagation. Whereas for dilution hole, it is necessary that dilution flow helps in liner cooling as well as in reducing the temperature of combustion product under specified limit for turbines, and ensure uniform temperature profile. For these tasks it is necessary that the flow must be disseminated in the dilution zone and near wall recirculation in the secondary zone to dissociate the UHCs and hence the diverging channel for dilution injection helps in the cause. The study also suggests that primary injection through converging channel of 45° and dilution injection through diverging angle of 45° gives most efficient combustion performance i.e. mass fraction of $CO_2 = 0.13384$ among the cases studied.

Compliance with ethical standards

Conflict of interest The authors declare that they have no conflict of interest.

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