

**INVESTIGATION OF IMPACT OF SWIRL ON
CHARACTERISTICS OF COMBUSTION AND EMISSION OF
PREMIXED OXY-METHANE FLAMES IN SWIRL-
STABILIZED MODEL GAS TURBINE COMBUSTORS**

BY

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A Thesis Presented to the
DEANSHIP OF GRADUATE STUDIES

KING FAHD UNIVERSITY OF PETROLEUM & MINERALS

DHAHRAN, SAUDI ARABIA

In Partial Fulfillment of the
Requirements for the Degree of

MASTER OF SCIENCE

In

MECHANICAL ENGINEERING

DECEMBER 2022

KING FAHD UNIVERSITY OF PETROLEUM & MINERALS

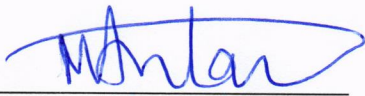
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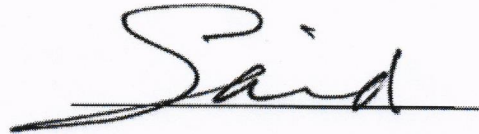
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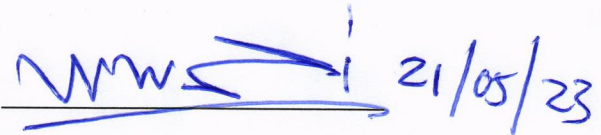
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2022

I would like to dedicate my thesis work to my beloved teachers and parents.

ACKNOWLEDGMENTS

I would like to use this opportunity to thank KFUPM for giving me a scholarship to pursue a fully funded Master of Science in Mechanical Engineering.

I am pleased to express my gratitude to my thesis advisor Dr. Mohammed Abdulaziz Mustafa Habib for his guidance and supervision in my thesis work. He has always been available and guided me with thesis related concerns. His suggestion, assistance and encouragement helped me in completing my thesis.

Dr. Ahmed Abdelhafez, my committee member, deserves special thanks for his unwavering support throughout my thesis, especially for helping me out in experimental study. I would like to express my gratitude towards my committee member Dr. Syed Ahmad Mohammad Said, for adding value to my thesis work. Dr. Medhat Ahmed Nemitallah deserves my especial thanks for his insightful comments and guidance in numerical study. I am grateful to my colleagues, friends, and lab technicians for helping me with my thesis work. I would like to express my special gratitude for my friend Mr. Qazi Talal and Mr. Oajedul Islam for encouraging me in pursuing my masters studies and supporting me in my difficult times. In the end, I would like to thank my parents and my wife for constant encouragement to pursue graduate studies.

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LIST OF ABBREVIATIONS

CFD	:	Computational fluid dynamics
CO	:	Carbon monoxide
CO₂	:	Carbon dioxide
DLE	:	Dry low emissions
DLN	:	Dry low NO _x
EGR	:	Exhaust gas recirculation
EV	:	Environmental vortex burner
FGR	:	Flue gas recirculation
GT	:	Gas turbine
IRZ	:	Inner recirculation zone
LES	:	Large Eddy Simulations
ORZ	:	Outer recirculation zone
RANS	:	Reynolds Averaged Navier-Stokes equations
SEV	:	Sequential environmental vortex burner

Greek Symbols

ρ	:	Density
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ϕ : Equivalence ratio

Subscripts

ad : Adiabatic

L : Laminar

th : Thermal

ABSTRACT

Full Name : Shorab Hossain

Thesis Title : Investigation of impact of swirl on characteristics of combustion and emission of premixed oxy-methane flames in swirl-stabilized model gas turbine combustors.

Major Field : Mechanical Engineering

Date of Degree : December 2022

Despite the invention of new technology of renewable energy, fossil fuel usages continue to rise because of large energy demand. Because of the stricter rules to save the environment, gas turbine manufacturers are to develop new technology so that low NO_x and CO₂ emission is enforced. To reduce emissions and increase combustion efficiencies when using fossil fuels, the development of sustainable cleaner energy production will be implemented. As a result, the focus of this thesis work is on oxy-combustion methods with various burner designs for power generation in gas turbines with CO₂ capture. Under flexible fuel/oxidizer oxy-combustion settings, combustion, and flame stability characteristics for different combustion methods have been explored and compared numerically. Large Eddy Simulation (LES) work was carried out. Different challenges and solutions in combustion, and flame characteristics and stability technique have been reviewed. Flame stabilization mechanisms such as impact of swirl on flame stability is discussed & experimental and simulation work is done. This study looks at the stability and combustion characteristics of premixed oxy-methane flames in a dry low emission (DLE) model combustor for clean fuel/oxidizer-flexible combustion applications in gas turbines. The blowout and flashback limits of the oxy-methane flame are defined by

equivalency ratios (ϕ) ranging from 0.1 to 1.0 and oxygen fractions (OF: volumetric concentration of O_2 in the O_2/CO_2 oxidizer) ranging from 25% to 70%. Flame stabilization mechanisms, for example impact of swirl on flame stability is studied experimentally and simulation work is done for swirler with angles 45° and 55° . The results showed that the lower the angle the wider the stable zone. Flame stability and combustion characteristics have been shown to be mostly determined by the adiabatic flame temperature (AFT). The blowout curve on the stability map similarly follows a constant AFT shape, however the flashback curve does not. There is very little change for blowout with swirler angle but the higher the angle the lower the flashback line. The flame becomes long, and V shaped for 45° Swirler. As the swirl increases (55° Swirler), the shape of the flame becomes wider and smaller. As the equivalence ratio (ϕ) increases, the outer recirculation zone (ORZ) becomes stronger and the eddy in IRZ shifts further downstream this makes the flame shorter and compact.

The temperature profile for different swirler and specific OF and equivalence ratio ϕ is determined. Experimental and simulated results of different swirl stabilized oxy-methane combustion are compared in this study.

ملخص الرسالة

الاسم الكامل: شهاب حسين

عنوان الرسالة: التحقيق في تأثير الدوامية على خصائص الاحتراق وانبعثات لهب أوكسي الميثان المخلوط مسبقا في احتراق التوربينات الغازية النموذجية المستقرة

التخصص: الهندسة الميكانيكية

تاريخ الدرجة العلمية: جمادى الأول ١٤٤٤

على الرغم من اختراع تقنية جديدة للطاقة المتجددة ، تستمر استخدامات الوقود الأحفوري في الارتفاع بسبب الطلب الكبير على الطاقة. بسبب قواعد الحفاظ على البيئة الأكثر صرامة ، يتعين على مصنعي توربينات الغاز تطوير تقنية جديدة بحيث يتم فرض انبعاثات أقل من أكاسيد النيتروجين و ثاني أكسيد الكربون. لتقليل الانبعاثات وزيادة كفاءة الاحتراق عند استخدام الوقود الأحفوري ، سيتم تنفيذ تطوير إنتاج مستدام للطاقة النظيفة. نتيجة لذلك ، ينصب تركيز هذه الأطروحة على طرق الاحتراق بالأكسجين مع تصميمات مختلفة للموقد لتوليد الطاقة في توربينات الغاز مع احتجاز ثاني أكسيد الكربون. في ظل إعدادات احتراق الوقود / المؤكسد المرنة ، تم استكشاف خصائص الاحتراق واستقرار اللهب لطرق الاحتراق المختلفة ومقارنتها عددياً. تم تنفيذ عمل محاكاة إيدي كبيرة (LES). تمت مراجعة التحديات والحلول المختلفة في الاحتراق وخصائص اللهب وتقنية الاستقرار. تمت مناقشة آليات تثبيت اللهب مثل تأثير الدوامية على استقرار اللهب ويتم عمل تجريبي ومحاكاة. تبحث هذه الدراسة في خصائص الاستقرار والاحتراق لهب الأكسجين والميثان الممزوج مسبقاً في جهاز الاحتراق الجاف ذي الانبعاث المنخفض (DLE) لتطبيقات الاحتراق المرن / المؤكسد للوقود النظيف في توربينات الغاز.

يتم تحديد حدود الانفجار والارتجاع لللهب أوكسي ميثان بنسب التكافؤ (ϕ) التي تتراوح من 0.1 إلى 1.0 وكسور الأكسجين (OF: التركيز الحجمي لـ O_2 في مؤكسد O_2 / CO_2) تتراوح من 25 إلى 70٪. آليات تثبيت اللهب ، على سبيل المثال يتم دراسة تأثير الدوامية على استقرار اللهب بشكل تجريبي ويتم عمل

المحاكاة للدوامة ذات الزوايا 45 درجة و 55 درجة. أظهرت النتائج أنه كلما انخفضت الزاوية كلما اتسعت المنطقة المستقرة. لقد ثبت أن خصائص استقرار اللهب والاحتراق يتم تحديدها في الغالب من خلال درجة حرارة اللهب ثابت الحرارة (AFT). يتبع منحنى الانفجار على خريطة الاستقرار بالمثل شكل AFT الثابت ، ومع ذلك ، فإن منحنى الفلاش باك لا يفعل ذلك. هناك تغيير طفيف للغاية في الانفجار بزاوية دوامة ولكن كلما زادت الزاوية انخفض خط الفلاش باك. يصبح اللهب طويلاً وعلى شكل حرف V لـ 45 درجة دوامة. مع زيادة الدوامة (55 درجة دوامة) ، يصبح شكل اللهب أوسع وأصغر. مع زيادة نسبة التكافؤ (ϕ) ، تصبح منطقة إعادة التدوير الخارجية (ORZ) أقوى وتتحرك الدوامة في IRZ بشكل أكبر في اتجاه مجرى النهر ، مما يجعل اللهب أقصر ومضغوطاً. يتم تحديد ملف تعريف درجة الحرارة لدوامة مختلفة ونسبة محددة من OF والتكافؤ ϕ . ستم مقارنة النتائج التجريبية والمحاكاة لاحتراق الأكسجين والميثان المستقر في دوامة مختلفة في هذه الدراسة.

CHAPTER 1

INTRODUCTION

Greenhouse-caused global warming has a significant impact on the change of climate. Changes in climate extreme responsible for less precipitation and drought, heat stress, in some places on the other hand heavy rainfall, flood, storm, land fall, erosion in other places leading to decrease in vegetation, rice and food production and water crisis making human life miserable. [1][2][3][4][5][6][7][8][9][10][11]. Combating global warming and sustaining economic prosperity at the same time is an imperative necessity. The 2015 Paris Agreement aims to reduce temperature increases below 2° C higher than pre-industrial levels (1861-1880) by dropping greenhouse gases, primarily CO₂ [12][13]. The only practical way to decrease CO₂ is by using non-carbon generating technologies or through CO₂ capture and storage CCS. Yet non-carbon generation technologies or green energy sources except biomass account for just 4.4 percent of the use of major energy [14], while the use of fossil fuels was 85 percent in 2015 and the share would be 78 percent by 2035 [15][16][17]. As a consequence, carbon capture is urgently needed, which will affect 15-55 percent of global CO₂ decrease by 2050 [18]. Among the technologies of CO₂ capture, premixed combustion of oxy-fuel is very encouraging which is a combustion with pure O₂ subsequent in flue gas there are CO₂ and H₂O only. H₂O can be condensed to remove easily leaving CO₂. Some CO₂ is recirculated to control temperature of flame so that turbine blade or combustor material is not damaged [14].

1.1 Global climate control initiatives and the critical need to perform carbon capture.

Fighting global warming and maintaining economic development at the same time are desperately needed. In the Paris Agreement, the parties to the United Nations Framework Convention on Climate Change (UNFCCC) set a long-term temperature objective for climate security for 2015. This agreement aims to restrict mean temperature of globe rises to below 2 ° C above pre-industrial stages (1861-1880).[12]. The ambient concentration of carbon dioxide, which is the most important greenhouse gas, has risen from 280 ppm in the pre-industrial era to over 370 ppm today, and is projected to grow above 500 ppm by the conclusion of this century.[19]. Therefore, performing carbon capture is critically important. CCS is expected to potentially lead to 15–55 percent of global CO₂ reduction by 2050.[18]

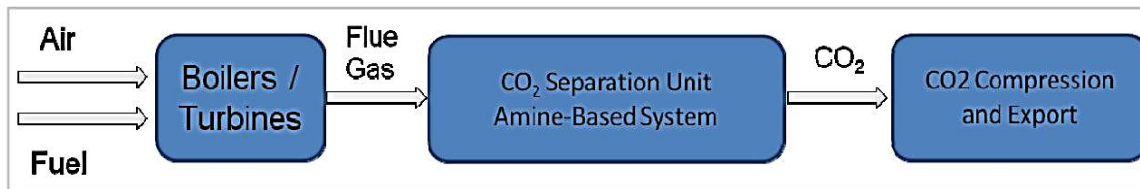


Figure 1 Main sources: Boiler and Gas Turbine based power generation[15].

Around the world, efforts are being made to improve the energy efficiency of the existing amine-based CO₂ collection methods. There is also considerable research in developments including gasification, the use of hydrogen as a gas turbine fuel, oxygen-enriched combustion, recycling of flue gas and removing CO₂ from membranes. Any of these, or a combination of them with improved heat integration with CO₂ sources, may eventually result in a 10–20 percent decrease in CO₂ capture operating costs.[15] There are three

primary types of methods of CO₂ capture: pre-combustion, post-combustion, and oxy-fuel combustion. Amongst these technologies to capture CO₂ in gas turbine applications, oxy-fuel combustion is very promising. Boiler and Gas Turbine based power generation is illustrated in figure 1. Burning of Fuel is done with pure O₂ in this technology to produce carbon dioxide and water as the main combustion ingredients, rendering the separating step of CO₂ simple & economical after condensation of H₂O.[20]. Oxy-fuel combustion, however, poses its own problems in case of combustion stabilization and combustion efficiency.[21]

1.2 Oxy-fuel combustion as a solution that simplifies the carbon-capture process.

The four main categories of carbon capture technologies are as follows. capture post-combustion, capture pre-combustion, oxyfuel combustion, and developing technologies like membrane separation and chemical loop combustion, among others. In pulverized coal-fired conventional power plants, carbon dioxide is separated from the flue gas during post-combustion capture. Typically, separation is accomplished by monoethanol-amine (MEA) or sterically obstructed amine chemical absorption (KS-1) [22] [23]. Absorption of amine is a established technology in the process industry [22][24]. However, the demonstrated operating scale is considerably smaller than the usual volume of power plants and there are major disadvantages for plant performance at the existing development stage [25] [26] [27] [28].

In the pre-combustion capture the process is usually recommended for use in conjunction with Integrated Combined Cycle Gasification (IGCC) power plants where it is known as IGCC-CCS. To achieve a gas (syngas) which contains CO, CO₂, and H₂, coal gasification

is used. The water-gas change reaction converts the CO into CO₂ which can then be isolated from the remaining gas-containing hydrogen until this is combusted in a gas turbine. As An Alternative, hydrogen can be isolated from the combusted CO and syngas in an environment of O₂/CO₂ [29]. Some techno-economic calculations [30] [24] indicate that IGCC has have high potential characteristics of improving process economics and plant performance. Yet, excessive investment costs are correlated with plant building, and typically IGCC plants are very complex [21]. In the case of combustion of oxy-fuel: the flue gas should consist primarily of CO₂ and H₂O by removing molecular N₂ from the combustion medium. The traditionally suggested plant design includes recirculation of the flue gas to the burners to monitor the flaming temperature within reasonable boiler content thresholds.

Compared to the above-mentioned post-combustion absorption processes, introduction of oxy-fuel combustion technology in current power plants fired by pulverized-coal would result in a greater improvement in plant design. This is largely because chemistry of combustion is changed by the exchange of recovered flue gas (partially CO₂ and H₂O) for N₂ in the oxidizer. A number of recent techno-economic evaluation studies show that oxyfuel combustion will be the highly energy-efficient and cost-effective technology for capture of carbon. [31] [27] [32] [33]. The oxyfuel combustion technology, one of the potential CO₂ capture technologies, offers a promising answer that is suitable for power production systems. This method concentrates on burning pure oxygen (O₂) instead of air, producing flue gas that is primarily made up of CO₂ and water (H₂O), which can be readily separated through condensation, so removing some impurities and leaving only pure CO₂ for storage [14]. Once the CO₂ collected for transport to the storage sites is compressed

and liquefied, CCS allows CO_2 to be isolated from all other flue gas components. Using amine absorbers ("scrubbers") and cryogenic coolers, large-scale CO_2 capture are commercially obtainable. However, these existing capture techniques require a lot of energy, making them inefficient for lowering carbon emissions [34].

1.3 Problem Statement

There is scarce understanding of stable combustion zones pertaining to the premixed CO_2 -diluted oxy-methane flames ($\text{CH}_4/\text{O}_2/\text{CO}_2$) in terms of the influence of oxygen fraction (OF) on flame structure and effect of swirler angle. The knowledge of stability and combustion characteristics of these flames will aid gas turbine designers and operators in prospective shift toward oxy-fuel and enhanced- O_2 combustion. The present work endeavors to fill this gap by investigating the stability and flame macrostructures of methane-based flames. Therefore, this study also incorporates large eddy simulations (LES) for investigating the performance of $\text{CH}_4/\text{O}_2/\text{CO}_2$ flames.

1.4 Objectives

The literature review shows superior potential of gas turbine swirl stabilized combustor model. However, in the case of oxy methane combustion, the impact of using different swirler with different angles were not performed. So, this thesis work will experimentally and computationally investigate premixed oxy-methane combustion using swirler of 45° and 55° in a model swirl stabilized combustor of gas turbine. The specific objectives are given below.

- To find the stability maps of blowout and flashback regime.
- To perform flame visualization/imaging for different swirl numbers at specific OF and equivalence ratio ϕ .
- To determine the temperature profile for different swirlers at specific OF and equivalence ratio ϕ .
- To evaluate the effect of ϕ , AFT and swirl angle on the flame structure and combustion stability.

CHAPTER 2

LITERATURE REVIEW

2.1 Air-fuel vs. oxy-fuel combustion: Control of combustion process and flame temperature

In terms of the operability of the combustor and the nature of the flame, the principles of oxy-fuel and air-fuel combustion vary greatly. The equivalence ratio is the key regulating factor in the process of air-fuel combustion in gas turbines. This is used to regulate the temperature of flame and to avoid or reduce combustion instability. Mainly in premixed combustors, the complete equivalence ratio is often held lean to minimize emissions of NO_x and change the exit temperature of combustor to the first stage cycle specifications and the turbine's metallurgical limits.

In all gas-turbine activity modes, the composition of O_2/N_2 oxidizer of air remains stable. Recirculation of flue-gas (FGR) is voluntary and mainly tends to decrease emissions of NO_x and reduce combustion instability. Oxy-fuel engines for combustion, nevertheless, are very special. The process of pure O_2 production for combustion consumes a lot of energy that renders this O_2 a scarce valuable resource as opposed to air. Therefore, oxyfuel combustors are usually used close to stoichiometry to save O_2 . In this case FGR (flue gas recycling) works as the key factor of the power. FGR that means part of the CO_2 effluent recirculation is compulsory in order to dilute the main reactants to control the temperature of flame and adjust the exit temperature of combustor to the first stage process

specifications and the turbine metallurgical limits[35] [36]. Therefore, based on the FGR ratio, the makeup of the O_2/CO_2 oxidizer is changeable. Consequently, air and oxy-combustors vary greatly in architecture and operability.[37]. The combustion of oxy-fuel and air-fuel is again very different from the viewpoint of combustion mechanics and the kinetics of reaction, as the air-based N_2 is replaced by carbon dioxide. The disparity in physical properties between nitrogen and carbon dioxide has a substantial influence on the physical properties[38][39][40] that is the transport properties, volumetric heat capacity and density of the reacting mixture. [41].

Dynamic viscosity, thermal conductivity, and mass diffusivity are included in the transport properties. CO_2 has a density greater than N_2 , impacting the form of the flame, the velocity of the bulk and the pressure drop at the head of the combustor, and the overall mass flowing via the turbine and combustor. In case of volumetric heat capacity, the CO_2/O_2 oxidizer has greater value than air, which in essence lowers the temperature of the flames, lowering flame intensity and stability.[37]. Chemical kinetics rates are inversely correlated with combustion efficiency and laminar flame velocity. The laminar flame velocity and combustion efficiency both decrease when carbon dioxide is present in high amounts during oxy-fuel burning. This is because flames from oxy-fuel burning have delayed chemical kinetic levels [42][29]. Moreover, dissociation of CO_2 is an extra challenge in oxy-fuel flames. Because this is an endothermal reaction which uses energy and influences the kinetics of the reaction [43][44][45][46], especially at high concentrations of CO_2 . So if an air-fuel combustion engine is transformed into an oxy-fuel system without any geometry changes and at the same equivalence ratio, the oxygen fraction OF (by volume) of 21% in the O_2/CO_2 oxidizer is not adequate to maintain a steady flame[47][48], that

means the OF needed is higher than 21% [37]. It is demonstrated by Ditaranto et al [49][50] that the oxy-combustion cycle needs minimum 30 percent OF to attain stable combustion in comparison to air-combustion. To account for the higher volumetric heat capacity of carbon dioxide contrasted to nitrogen the higher oxygen concentration is needed. These problems include the alteration of burner designs so that stable flames can be maintained under conditions of oxy-fuel combustion.[37].

2.2 Combustion and stability characteristics

The switch of the flame from the unmixed (diffusion) form to the premixed one outcomes in higher rates of flame instability, rendering it very difficult to manufacture gas turbines. Swirl typically helps to achieves the flame stabilization. It also improves premixing of fuel and oxidizer. The development of a recirculation by swirl at downstream of the burner is an effective way of sustaining the flame where there is no physical flame keeper. [51]. Reactant premixing raises the uniformity of the temperature and material's concentration fields. But premixing of reactants makes the combustor further susceptible to swirl-produced oscillations.[52][53][54]

It is possible to divide combustion instability between static and dynamic instability. The systems for static instability are Flashback and Flame Blowout. These processes are specifically applicable to conditions of operation. When the speed of the wind exceeds the speed of the flame, the flame bursts out, while the flashback emerges when the speed of the flame approaches the velocity of the fire. Alternatively, as a consequence of combining the heat release with acoustic pressure fluctuations creating unwelcome pressure waves, thermos-acoustics or dynamic instabilities are developed. [55][56]. To some extent, these

waves of pressure can be dangerous & can result in destruction to the hardware of combustor. [57].[58] . Premixed combustion is known to be steady where pressure fluctuation amplitude is less than five percent of the pressure of chamber. Combustion that is oscillatory or unstable exceeds 5%. [57] [58] [59][60] [37]. For two factors, instability in combustion may be produced. The 1st theory is that turbulence can be caused by some unnatural disturbance outside of the combustion chamber. The 2nd theory is that inside the system it will generate uncertainty. For the first cause, chaos is instigated by an external disturbance that is steady to minor disturbances. [61][62].

The instability caused by the oscillations induced by a slight fluctuation resulting from the noise of the device.[63][64][65]. Instabilities of combustion in gas turbines can be categorized as medium, moderate and high frequency instabilities [66][67][68]. The methods and techniques to monitor these instabilities have been developed by numerous researchers. [69][70][71].

Two methods for the damping of gas turbine combustion oscillations, namely passive and active control have been identified by Rashwan et al [70]. In the technique of passive control, several alterations are made to increase the energy loss rate or decrease the heat escape rate by modifying the fuel injection system, fuel / oxidizer composition, the combustion chamber configuration or by causing acoustic dampers .[71] In the process of active control, in contrast, a source of energy is presented into the combustion mechanism to interrupt the relation between the acoustic waves and the release of heat. [69] [37]

2.3 Challenges with oxy-fuel combustion

Engines for combustion of oxy-fuel are very unique. Here, for the equivalence ratio to be replaced as the primary parameter of control, FGR becomes a must. Some CO_2 is recirculated to dilute the major reactants specifically to control the temperature of flame [36] & to adjust the temperature of the combustion exit to the process specifications of the first stage and the metallurgical boundaries of the turbine[35]. The disparity in physical properties between N_2 and CO_2 has a substantial influence on the physical properties of the mixture of reactants [39][38][40]. the physical properties are mainly density, transport properties and volumetric heat capacity [41].

Mass diffusivity, elastic viscosity and thermal conductivity are included in the transport properties. CO_2 has a greater density than N_2 , impacting the shape of the flame, the velocity of the bulk and the reduction in pressure across the combustion core, and the overall mass flowing into the turbine and combustion. Compared to air, the oxidizer of CO_2/O_2 has greater volumetric heat capacity, which lowers the temperature of the flames, decreasing stability and flame velocity. The greater O_2 concentration is needed to account for the greater CO_2 volumetric heat capacity in comparison to nitrogen. Ditaranto et al [49][50] demonstrated that the cycle of oxy-combustion necessitates minimum 30% OF in order to ensure steady combustion in comparison to air-combustion. These require the alteration of burner configurations in order to continue steady flames under oxyfuel combustion conditions [37].

2.4 Existing swirl-based gas-turbine combustion systems

2.4.1 Dry low-NO_x/low-emissions (DLN/DLE) burners

Latest developments in gas turbine technology are primarily aimed at increasing performance and reducing emissions of nitrogen oxides (NO_x). Several methods have been established to reduce the NO_x emissions. Formation of NO_x for the period of combustion is an important feature of temperature of flame .[72] Apparently the best encouraging solution for decreasing emissions of NO_x is to lower the temperature using diluents such as water or steam, especially in stationary systems of power generation. It typically results in low engine performance, potential corrosion owing to impurities of water and CO burnout reduction or quenching, making it detrimental[63]. Solutions that do not include water or steam injection consist of DLN or DLE technologies implemented by LPM combustors [73][74]. These combustors can decrease emissions of NO_x down to a single digit in gas turbine applications [75]. Figure 2 below displays a schematic diagram that compares the idea of DLE to a traditional burner.

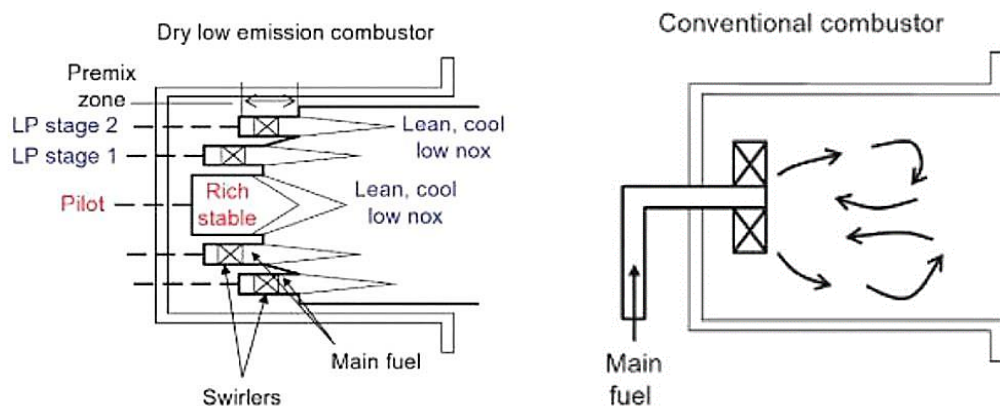


Figure 2 Comparison of a usual DLE combustor with a traditional one in the schematic diagram [75]

Both burners stable in a swirl. In the DLE burner the volume of the fuel injector is bigger in order to compensate for the greater amount of airflow in the region of premixing. The main model requirements for DLE burners' optimum performance are (a) compliance with standards of emission at peak load and (b) emission control across the engine load range. The ability of the system to respond quickly to load changes, the conservation of minimal combustion acoustics, and the smooth shift from one fuel to another, particularly in engines of dual fuel, are other significant parameters for maintaining stability of combustion & a wide choice of operability under all circumstances [76]. For the primary purpose of lowering emissions of NO_x without using injection of water or steam, various producers have various model approaches to the concept of DLE combustor. The diagram illustrated in Figure 3 below illustrates the concept of the two-stage premixed DLN-1 combustor design capable of utilizing both liquid fuels and natural gas [77]. Four key elements compose the burner: venturi, liner, fuel injectors, and middle body/cap assembly. There are three core areas principal, secondary and dilution supplied by these elements.

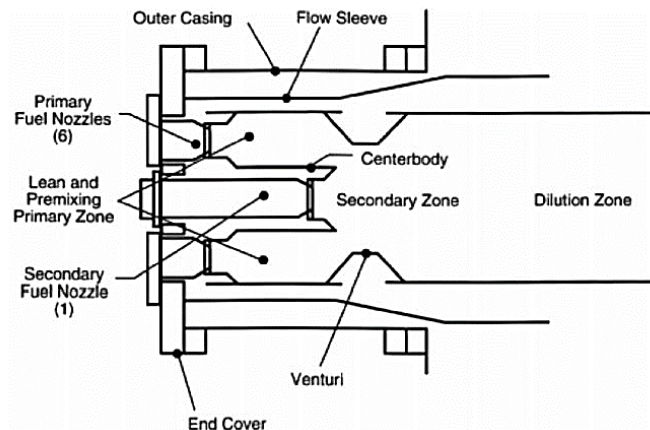


Figure 3 DLN combustor by General Electric [77].

The burner works in 4 different modes, such as Premix, Lean-Lean, Primary, and Secondary. Fuel is only delivered through the main nozzle in primary mode & main air is supplied through the swirlers. Ignition of flame & small loads up to twenty percent are carried out in this mode. For intermediate engine loads the lean-lean mode is chosen, providing fuel to both the prime and secondary regions. The fuel blends with air & forms the secondary combustion region at the center body exit after passing through the swirler. The secondary operating mode is the intermediate mode between the premix modes and lean-lean, with the flame being maintained by only the secondary field. Delivery of fuel to the principal nozzle is progressively lowered whereas delivery for the secondary nozzle rises, which destroys the primary flame. Some fuel is always delivered to the primary nozzle in Premix mode, but only in the secondary flame area the flame is maintained. This mode makes minimal exhaust emissions possible. NO_x and CO levels can be as small as twenty five ppmv and nine ppmv, respectively, when utilizing natural gas at base load. [76]

2.4.2 Environmental (EV/AEV/SEV) burners.

While the LPM gas turbine industry has numerous competing technologies of combustion, a few ranks out depending on their impressive versatility & ultra-low emissions. For example, EV burners, compete powerfully with MMs especially because EV burners provide dramatically efficient stabilization of flame over a large range of operations, increasing the turndown of the engine. Both EV-burners and MMs have never been tested under conditions of oxy-combustion; therefore, the subsequent thorough review is devoted to EV-burners to explore their possibility for application in gas turbines of zero-emission. The EV burner in LPM gas turbines has the capability to reach ultra-low emissions of NO_x[78]. The geometry comprises of two half-cone shells, one of which, as shown in

Figure 4, is slightly offset towards the other in the radial direction while maintaining parallel axes.

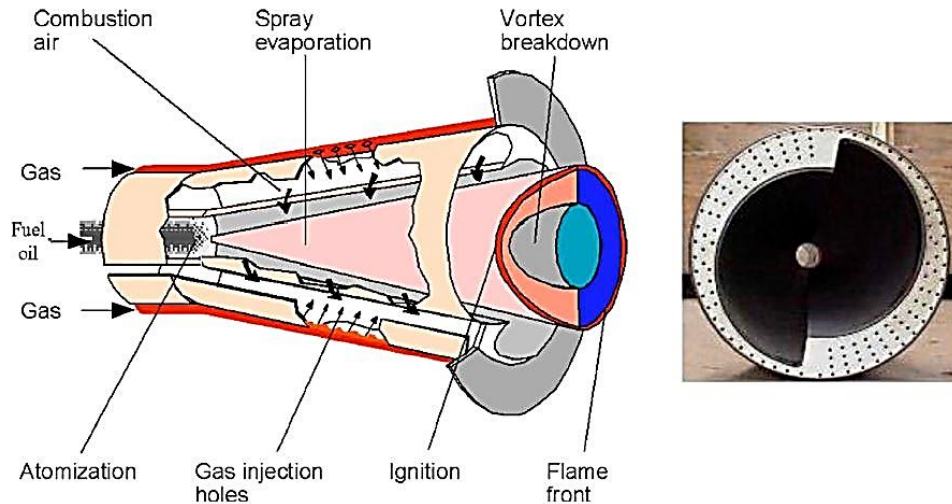


Figure 4 EV-burner principle [79].

Therefore, 2 tangential slots of steady breadth are provided between the shells. The air of combustion reaches the burner through these slots tangentially, causing swirling core flow with breakdown of vortex adjacent to the exit of the burner [80]. The swirl strength rises in the direction of axis due to the rising diameter; the angle of divergence is deliberately constructed to place the point of breakdown of vortex near the exit of burner, as seen in Figure 5. Gaseous fuel premix is pumped into the combustion to mix with air by a row of holes situated upstream of every air slot; this crosswise injection increases the efficiency of mixing and is a crucial requirement for ultra-low emissions of NO_x. EV-burners are capable of dual fuel, which is water injection liquid fuel and gaseous fuel (DLN) [78].

A sudden and abrupt change in the strongly swirling flow mechanism is Vortex breakup, one of the key features of EV-burners. Some research has shown that the onset of breakdown of vortex is autonomous of the Reynolds number (Re) of flows, but depends primarily on the number of swirls[81]. The vortex breakup occurs in two different forms, including the axisymmetric bubble-like and the spiral shaped ways, based on the configuration of the burner. The axial flow slows down along the axis of vortex, and as the flow recirculation within the combustor is spread over a large area, the velocity gradients within the vortex weaken. [82]. This action results in a total blending of the reactants and higher stability of flame, both at start-up & under very-small load situations. Good mixing avoids spots of high temperature in the flame from being formed and can also keep concentrations of NO_x at a low level. The center of the flow accelerates downstream, thereby eliminating the flame flashback. Which facilitates safe operation at ultra-lean conditions, resulting in full combustion at controlled temperatures and lower concentrations of CO and NO_x .

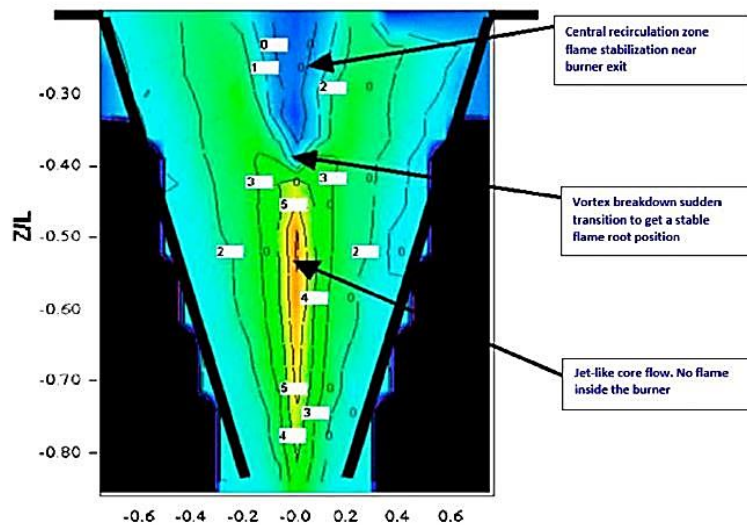


Figure 5 Distribution of axial velocity within an EV burner [79].

The EV-burner design is distinguished by several features [79] from other premix hardware of DLN [83]:

1. In the axial direction, the distribution of air intake dramatically accelerates the core flow, forming a normal flashback barrier.
2. Strong swirls are produced simply through tangential intake of air; sophisticated swirlers are not needed.
3. Without the need for additional stability devices such as dump extensions, bluff heads, and ramps that may possibly be harmed owing to flame impingement, the point of breakdown of vortex is fixed aerodynamically in the free flow.
4. A central or inner recirculation area resides downstream of the point of vortex breakup, which steadies the flame efficiently.
5. To help stabilize pre-mixed flames at medium to high loads, no pilot fuel injection is required; the burner can operate in a 100 percent pre-mixed mode that allows ultra-low emissions of NO_x in the exclusion of diffusion combustion.

A separate pilot circuit injects fuel at low loads into a lance at the centerline of the burner, where the equivalence ratio of the burner is below the extinction point of the premix. The core flow becomes rich in fuel at the vortex breakup point, and the resultant flame is stabilized again. Likewise, if the burner is driven with liquid fuel, it is pumped into the head of the cone through a central plain jet nozzle. The jet breaks up into tiny droplets which scatter within the burner cone over the entire flow area. With the aid of an inner recirculation region the resulting flame is stabilized. The need to improve liquid fuel

evaporation and mixing to further reduce the related emissions of NO_x contributed to the advancement of an Advanced EV (AEV) burner[84][85][86], as seen in Figure 6.

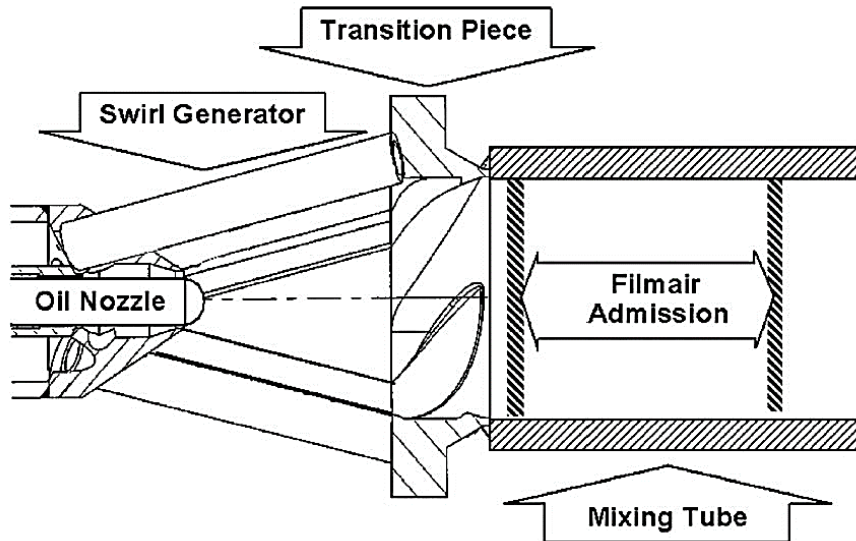


Figure 6 Design features of AEV burner [79]

Four inlet air slots are utilized in place of 2, minimizing the tangential variance of the radial component of velocity & thereby eliminating the impression of fuel droplets on the walls of the cone. To tackle complete liquid fuel evaporation and to facilitate gaseous mixing, an extra tube for mixing is added to the burner. The axial and tangential velocity profiles demonstrate a jet-like core flow with a definite tiny vortex body inside the conical segment (Figure 7).

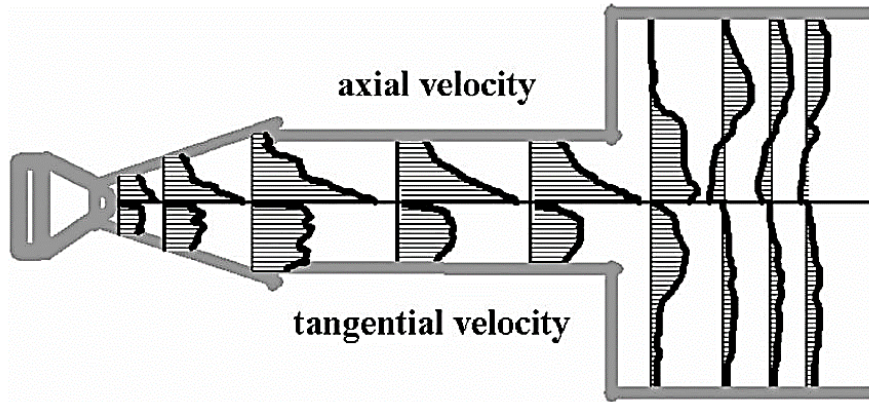


Figure 7 AEV burner velocity profiles Axial (top) and tangential (bottom) [79].

The centerline peak axial speed exceeds its average by twice at the cone exit. [79]. This strong axial core flow rate continues in the mixing channel, thus increasing the flashback range. It is also possible to improve the lowest velocity outside the mixing tube walls by sufficiently admitting an air film less than ten percent of full airflow. This film significantly removes the threat of flashbacks on the wall and dilutes any nearby residues rich in fuel. The former Alstom Company designed and introduced the EV-burner in its gas turbine GT11, which housed a central ignition burner plus 36 burners in a 3-ring hexagonal composition.[87].

Since the initial configuration was not yet fitted with central pilot injection, all burners were only fired in premixed mode. By switching on and off the individual burners, full combustor load and turndown have been achieved. The same hexagonal configuration has also been utilized in the Alstom GT11N22, GT13, GT8, and GT9 gas turbines.[88]. The EV-burner pilot-fuel-injected configuration was first introduced in the Alstom GT10 and GT13E2 gas turbine annular combustors & soon after in the GT26, GT24 and GT8C2 engines. At present, all Alstom gas turbines have EV and AEV-burners-based low-NO_x combustion systems. Paschereit et al [89] explored the impact of bringing the central lance

of fuel closer to the point of vortex break-down. This plain modification in design granted for the fixation of the axial location of the point of breakdown over a wide range of conditions of operation & superior control over instability of combustion.

The extending of central lance also strengthened the aerodynamics of the burner and the flames without central fuel injection. In order to measure the impact of the extension of lance on instabilities of combustion (oscillations of pressure and heat release) & on CO, NO_x, and emissions of unburned hydrocarbon, visualizations of flow were done. The researchers tried to optimize the location of the axial lance and the quantity of central fuel at different output powers and burner equivalence ratios. In order to build a premixed staged burner which achieves much lower emissions of NO_x at lesser component loads in comparison to standard EV-burner service, Zajadatz et al[80] scrutinized the regulated splitting of gaseous fuel between the major circuit of premix & the central pilot lance. There were no big changes in the configuration of the burner or its fuel supply mechanism in this smart fuel staging, which made the concept applicable for most modern EV gas turbines. Cho et al [90] analyzed numerically, in a related review, the impact of staging of fuel and unmixed fuel-air on emissions of NO_x. They suggested the splitting of fuel between the central pilot lance and the premix circuit to eliminate non-mixing and the subsequent NO_x. To increase the mixing of fuel-air, a further design enhancement was also proposed, for example by raising the gap between fuel holes in the circuit of premix to allow extra time to blend fuel with air.

Wind et al [91] tested natural gas fuel for the Alstom GT26 engine containing varying concentrations of H₂ & higher hydrocarbons (C₂ +), seeking greater flexibility of the fuel. They investigated both reheat combustor's primary EV-burners and the Sequential EV-

burners. H_2 & higher hydrocarbons influence the burning speed and delay period of auto-ignition, which is a primary design factor for SEV burners. The typical procedure of operation of the engine had to be altered just a little to adjust some fuel composition variations without any modifications of hardware. EV-burners have supreme longevity, reliability and turndown based on this simplistic review, rendering them an encouraging alternative for oxy-fuel combustion to conquer the reduced performance of carbon dioxide diluted oxy-combustion relative to conventional combustion of air-fuel. As EV-burners have never been tested before at conditions of oxy-combustion, there is a need for a detailed investigation to measure the pertinence of this technology in gas turbines of zero-emission.

2.5 Premixed oxy-fuel combustion

2.5.1 Characteristics of premixed oxy-fuel combustion

The dynamic equilibrium of flames for premixed $CH_4/CO_2/O_2$ in stoichiometry was investigated by Speth et al [92]. The adiabatic flame temperature (AFT) of the mixture of reactant was modified by the OF. Though oxy-flames and air-flame have distinct speeds of flame at the same AFT, correlation between the two flames was found in that the adiabatic flame temperature mainly controls the transition from one mode of stability to another. High-speed photographs that showed identical modes of vortex collapse backed the above result. These modes, which are symptomatic of the associated geometries of turbulent flame, reflect a shift in AFT, as AFT shifts the turbulent flame geometries.

In addition, Kutne et al [93] researched and assessed flames of air and oxy-methane, but under the conditions of premixed partially. They regarded 0.5 to 1.0 equivalence ratios and 20 to 40 percent oxygen fractions. For quantifying the flame structure, Raman spectroscopy

and OH*-chemiluminescence imaging were used. The oxygen fraction has been reported to have a considerable influence on characteristics of flame, while the equivalence ratio has a minor influence. Compared with oxy-flames, the inner recirculation zone in air-flames is situated further upstream, suggesting the latter's inferior stability. The researchers have reported that the twenty one percent OF is unachievable in oxy-flames. Greater volumetric heat capacity of carbon dioxide relative to nitrogen was reason for the lower oxy-flame speeds and combustion temperatures. These observations were also strengthened by numerous other researches [94][44]. According to Song et al, the profiles of temperature were observed to align CH₄/air and CH₄/36 percent O₂/64 percent CO₂ flames.[94].The literature [95][96] recorded similar values.

Rashwan et al [97] stated similar outcomes; for stable oxygen combustion with FGR, an oxygen fraction of 29 to 40 percent is required. In addition, owing to the negative effect of combining carbon dioxide to the kinetics of combustion, oxy-combustion restricts the stability period of combustor. Abdelhafez et al [98] demonstrated that steady oxy-flame could not be achieved in a model of swirl-stabilized premixed gas-turbine combustor at OF beneath 22 percent, even under conditions of stoichiometry. They stated that the impact of the OF is considerably larger than that of the equivalence ratio, which means that steady flames with higher oxygen fractions can be achieved under much leaner conditions. The blow-out mechanisms of premixed flames of CH₄/O₂/N₂ and CH₄/O₂/CO₂ in a swirl combustor is contrasted by Jerzak and Kuznia [45].They indicated that, relative to air-flames, FGR oxy-flames have significantly narrower windows of operability. An experimental study involving partially premixed CH₄/O₂ and CH₄/air flames over a PP burner was performed by Rashwan et al [99]. The data demonstrated that as the ratio of

premixing rises, the NO_x and CO emissions reduces. The color of the flame moved to blue at greater ratios of premixing from reddish color at smaller ratios; higher soot formation readings were measured in diffusion flames. The experiments have shown that, as the OF of the mixture of oxidizer rises, the steadiness of the flames of the oxy-fuel rises because of the rise in speed of flame.

2.5.2 Flame stabilization mechanisms in past studies

An efficient way to stabilize the flames where there is no physical flame keeper is the development of a swirl-generated zone of recirculation in downstream of the burner. [51]. However, reactant premixing enhances the homogeneity of the species' temperature & concentration fields, rendering the combustor more susceptible to oscillations produced by swirls in pressure and/or equivalence ratio. [52][53][54]. Permanent harm can occur by resonance when the oscillation frequencies interact constructively with natural frequencies of the geometry of the combustion system, [52]. Rashwan et al [70] identified 2 techniques for the damping of oscillations of gas turbine combustion, namely active and passive control. Some adjustments are made in the passive control technique to lessen the rate of heat release or rise the energy loss rate by adjusting the composition of the fuel / oxidizer, the fuel injection system, the configuration of the combustion chamber or by inducing acoustic dampers [71]. On the other hand, in the technique of active control, to disrupt the coupling between the waves of acoustic & heat release, an energy source is inserted into the combustion system [69]. Some strategies have recently been proposed to avert situations that lead to oscillations in combustion systems while preserving large operability ranges. These methods incorporate approaches for managing the mixing profile. It is necessary to have two active control strategies, namely pulsing pilot fuel injection at the

instability frequency [100] and fuel staging [101]. The latter technique enables simultaneous reduction of Emissions of NO_x and suppression of instabilities without the complexity of the pulsing-injection technique with high-frequency actuation. Lacarelle et al. [102] researched the impact on residency time delays & mixing performance in a premixed system of various fuel injection techniques. They suggested that for a given fuel delivery, an escalation in mixing efficiency is obtained where the line of fuel injection consists of two rotating jets at 25 ° and 55 ° angles of crossflow in comparison to a 45 ° angle single jet.

Lacarelle and Paschereit [103] devised a fluidic oscillator to avoid drawbacks in mechanically induced flows. In a basic jet-in-crossflow system, the efficiency of mixing of four such injectors was contrasted with that of a standard non-oscillating jet. This was claimed that the fluidics injectors worked better than the standard jets and that the use of fluidics injectors could achieve around 50 percent decrease in mixing duration for a given mixing quality. In a lean premixed environment, Estefanos [104] examined the effects of air-fuel mixing on emissions of NO_x and combustion instabilities. He suggested that axially injecting fuel into an environment of heavy unstable mixing helps to detach the dynamics of combustion by dampening the unstable mixing.

CHAPTER 3

EXPERIMENTAL AND NUMERICAL MODEL

3.1 Experimental Setup

For the purpose of researching premixed oxy-methane-CO₂ flames for clean and effective power generation, the LPM DLE swirl stabilized premixed fuel-flexible model combustor for gas turbine was used. Static stability of the oxy-flames was found with the help of two flame extinction limits i.e. blowout and flashback limits. The figure of experimental setup is shown in Figure 8. The detailed Diagram of the Model gas turbine combustor rig equipped with swirler is shown in Figure 9. There are two radial inlets of 4 mm inner diameter opposite to each other for diluent and oxidizer (CO₂ + O₂) at 35 mm above from the base. While fuel CH₄ enters the plenum from the bottom through a single tube. For premixing, fuel and oxidizer are passed through the premixing chamber of approximate 1 m length before the swirler inlet and approximate diameter 50.8 mm. For achieving controlled composition of premixes 45° and 55° swirler is used. Immediately after the swirler a 30° gradual converging and 45° diverging nozzle insert of 20 mm throat diameter is used for acceleration. To confine the flame a quartz tube of inner diameter 63.5 mm and length 30 cm is placed on top of combustor rig. From high pressure cylinder through computerized flow controller fuel and oxidizers are supplied as shown in figure 8.

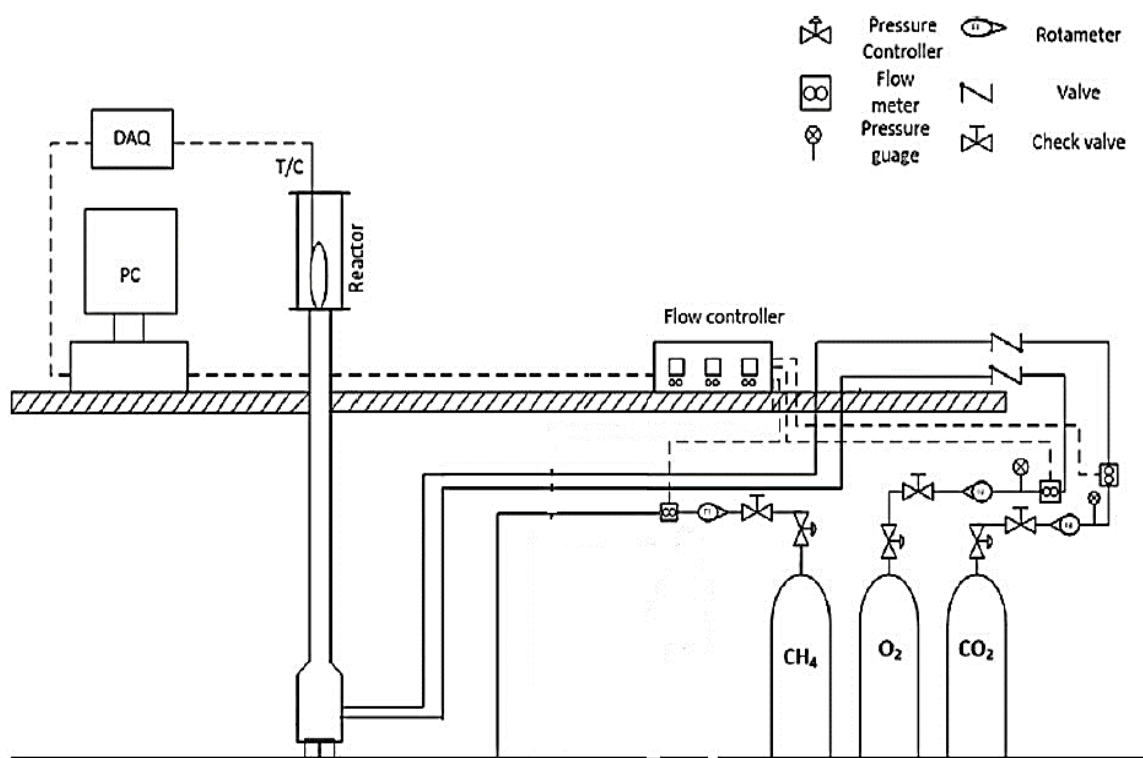


Figure 8 Diagram of Experimental Setup.

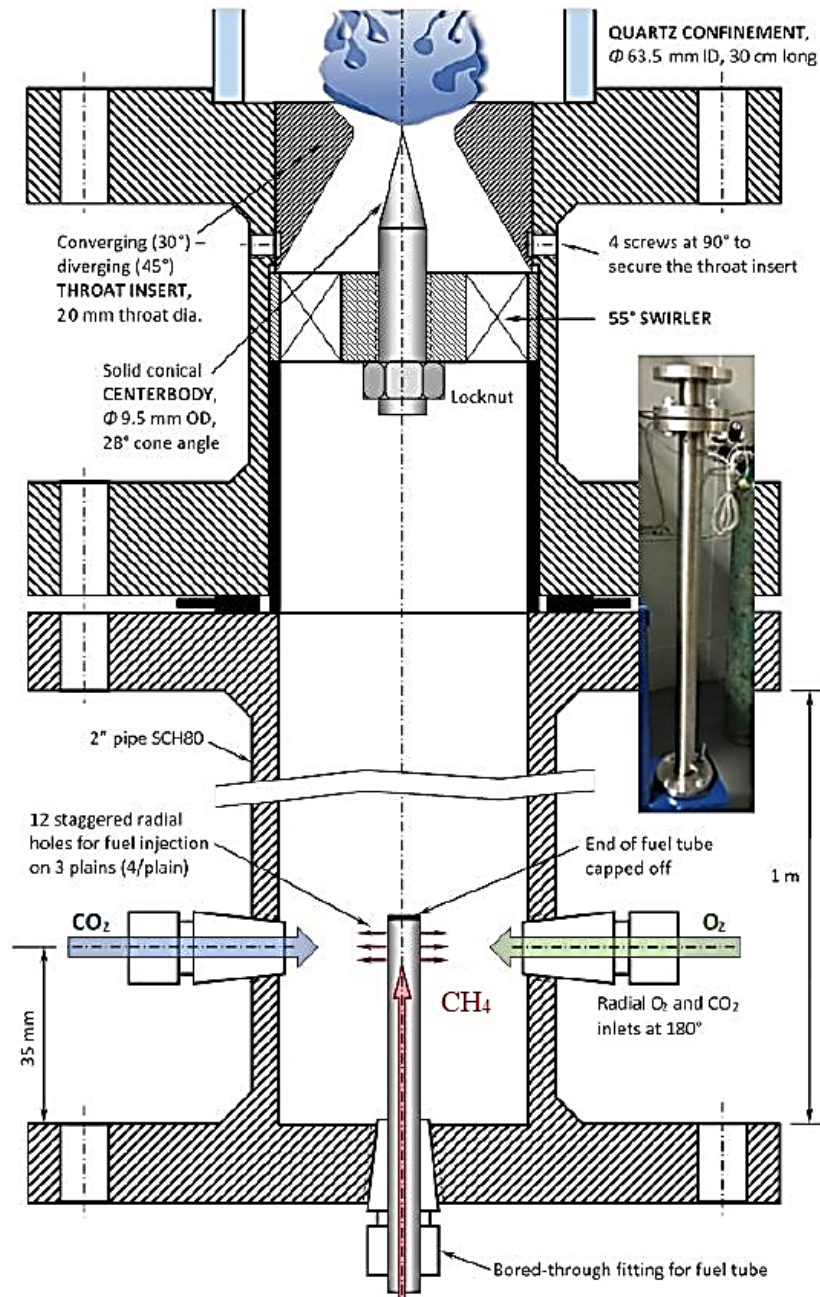


Figure 9 Diagram of the Model gas turbine combustor rig equipped with swirler.

3.2 Test condition

The current study examines premixed CH₄/O₂/CO₂ flames for swirlers with various angles at a fixed flame-inlet bulk velocity. For all of the CH₄/O₂/CO₂ flames that were studied, the method used to fix the bulk throat velocity at 5.2 m/s was thoroughly explained by Abdelhafez et al. [19].

The following species flow rates are found. [20]

$$\dot{m}_{CH_4} = 16 \left(\frac{pAv_a}{R_u T} \right) \left(1 + \frac{2}{\phi OF} \right)^{-1}$$

$$\dot{m}_{O_2} = 64 \left(\frac{pAv_a}{R_u T} \right) \left(\phi + \frac{2}{OF} \right)^{-1}$$

$$\dot{m}_{CO_2} = 88 \left(\frac{pAv_a}{R_u T} \right) \left(\frac{1 - OF}{\phi OF + 2} \right)$$

where OF and ϕ were evaluated across the corresponding ranges of 21–70% and 0.2–1.0.

Utilizing the formula, the bulk dynamic viscosity of the reactant combination was determined [98].

$$\mu_{mix} = \frac{\sum y_i \mu_i \sqrt{M_i}}{\sum y_i \sqrt{M_i}}$$

From the bulk density of the reactant mixture,

$$\rho_{mix} = \frac{p \sum \dot{m}_i}{R_u T \sum \left(\frac{\dot{m}_i}{M_i} \right)}$$

Thus, it was possible to determine the Reynolds number at the burner throat using

$$Re = \frac{vD\rho_{mix}}{\mu_{mix}}$$

3.3 Experimental procedure:

The flow rates of methane, O₂, and CO₂ were adjusted in such a way that it makes a constant bulk velocity of 5.2 m/s at throat and desired OF and ϕ is obtained. As a result, even if both data points had the same OF or ϕ , switching from one data point to the next required adjusting all three flows.

The procedure used to do the experiments is as follows:

1. Firstly, a safe ignition point is determined away from flashback and blowout for premixed oxy-methane combustion.
2. Then gradually flowrates of CO₂, O₂ and fuel CH₄ are adjusted gradually for achieving the desired OF and assumed equivalence ratio ϕ
3. Then, to reach to the necessary regime of flashback or blowout for the given OF gradually flowrates of CO₂, O₂ and fuel CH₄ are adjusted until reach that regime and record the equivalence ratio ϕ . Inlet velocity is 5.2 m/s
4. The flame gets cup shape and outer recirculation zone when its weak, but it gets V shape of inner recirculation zone when stronger.

5. From the stable point equivalence ratio ϕ is reduced to reach the blowout point and equivalence ratio ϕ is increased to reach flashback point for same OF. Before blowout flame gets weak whereas it gets stronger with huge noise before flashback.
6. The steps 1 – 5 are repeated for different swirler of 45° , and 55° .

3.4 Numerical Model Setup of CH₄/O₂/CO₂ flames

To perform simulation work of swirl stabilized premixed oxy-methane combustion Ansys fluent CFD tool is used. ANSYS Design modeler and ANSYS Meshing software are used for preparing geometry and meshing, respectively. Type meshing utilized is tetrahedron. Instead of the transport equations from the Reynolds-averaged solution, Probability density function (PDF) was implemented as any thermochemical moment such as RMS temperature, average reaction rate, etc. can be obtained with lower computational time from the PDF process. To investigate the effect of swirl on premixed oxy-methane combustion, considered different conditions are given in table 1.

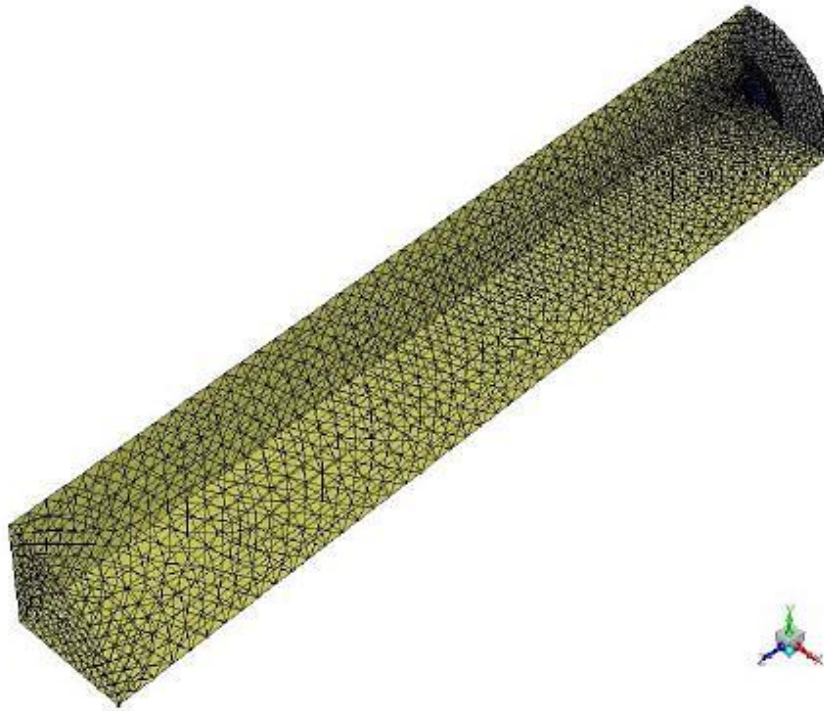


Figure 10 Tetrahedron type meshing used for swirl stabilized combustor setup.

Table 1 Oxy-methane cases to be simulated for various swirl angle.

swirler	Case	Oxygen fraction	Equivalence ratio ϕ
55°	1	35%	0.568
	2	50%	0.374
	3	65%	0.269
45°	4	35%	0.568
	5	50%	0.374
	6	65%	0.269

3.4.1 Governing equations

The Large Eddy Simulation (LES) model for Swirl-stabilized flames in ANSYS Fluent 19R3 was used to solve the conservation equations of momentum, mass, & energy, as well as the scalar variable's transport equation in the three-dimensional (3D) computing environment. the scalar variable's transport equation in the three-dimensional (3D) computing environment. In order to examine eddy-formation and its effect of the combustion, LES model was selected for the swirl-stabilized combustor. Solving the equations for the 3D domain simultaneously predicted heat transport, flow, and emission characteristics. In equation (1), Momentum, Energy & Mass conservation equations is represented where Γ_ϕ is the diffusion component, S_ϕ is the source term, u_j is the velocity component along j direction, while Reynolds averaged, and fluctuating components of the dependent variable are denoted by Φ and ϕ respectively.

$$\frac{\partial}{\partial x_j} (\bar{\rho} \bar{U}_j \Phi + \bar{\rho} \bar{u}_j \phi) = \frac{\partial}{\partial x_j} \left[\Gamma_\phi \frac{\partial \Phi}{\partial x_j} \right] + \rho \bar{S}_\phi \quad (1)$$

3.4.2 Radiation model

Radiant heat transfer was anticipated utilizing the discrete ordinate (DO) modeling strategy, which has been proposed for modeling oxyfuel combustion and is applicable for a wide range of optical thickness.[105] One radiative iteration was run in the DO model for this experiment for every ten energy iterations. The gaseous mixture's absorption coefficient was determined utilizing the WSGGM (weighted sum of gray gas model), and it was confirmed under numerous working conditions. [106] Equation (2) shows total emissivity (ε) in terms of the weighted sum of gray gases.

$$\varepsilon = \sum_{i=0}^{i=I} \alpha_{\varepsilon,i}(T)[1 - e^{-\kappa_i P L}] \quad (2)$$

Where $\alpha_{\varepsilon,i}$ refers to the emissivity weighting factor for the grey gas i at temperature T , P represents the partial pressure of the absorbing gases, κ_i indicates the adsorption coefficient, and L represents the gas layer thickness (or path length).

3.4.3 Modelling for Turbulence

Large eddy Simulation (LES) turbulence model

There are two types of eddies that make up the turbulent flow: small & large eddies. While kinetic energy of turbulence is dispersed by small eddies, the usual length of the mean flow corresponded by large eddies. In this study, the LES modeling approach was used to solve turbulence by sorting out smaller eddies. The low-pass sorting procedure for a quantity can be performed using Equation (3) [107], where G is the function for filter used to estimate

the scale of resolved eddies and the integration takes into account the entire domain. Equations (4) and (5) demonstrate the representation for continuity and the Navier-Stokes equation following the integration of the function of filtering.

$$\bar{\varphi}(x, t) = \int G(r, x) \varphi(x - r, t) dr \quad (3)$$

$$\frac{\partial \rho}{\partial t} + \frac{\partial}{\partial x_i} (\rho \bar{u}_i) = 0 \quad (4)$$

$$\frac{\partial}{\partial t} (\rho \bar{u}_i) + \frac{\partial}{\partial x_j} (\rho \bar{u}_i \bar{u}_j) = \frac{\partial}{\partial x_j} \left(\mu \frac{\partial \sigma_{ij}}{\partial x_j} \right) - \frac{\partial \bar{p}}{\partial x_i} - \frac{\partial \tau_{ij}}{\partial x_j} \quad (5)$$

Here σ_{ij} tensor of stress from molecular viscosity and τ_{ij} , corresponding to subgrid-scale stress, are expressed respectively by Equations (6) and (7).

$$\sigma_{ij} \equiv \left[\mu \left(\frac{\partial \bar{u}_i}{\partial x_j} + \frac{\partial \bar{u}_j}{\partial x_i} \right) \right] - \frac{2}{3} \mu \frac{\partial \bar{u}_i}{\partial x_j} \delta_{ij} \quad (6)$$

$$\tau_{ij} \equiv \rho \overline{u_i u_j} - \rho \bar{u}_i \bar{u}_j \quad (7)$$

In order to replicate the subgrid-scale stresses that resulted from the filtering procedure, filtered static pressure terms & the isotropic portion of the subgrid-scale stresses were combined. Equation (8) depicts the deviator component modeling using turbulent viscosity

$$\tau_{ij} = -2\mu_t \bar{S}_{ij} \quad (8)$$

Where, $\bar{S}_{ij}$ and μ_t represents the turbulent viscosity & the strain tensor rate, respectively. The Smagorinsky-Lilly model was implemented to solve the Turbulent viscosity (μ_t), provided in Equation (9).

$$\mu_t = \rho L_s^2 |\bar{S}| \quad (9)$$

Here, L_s denotes the length of mixing for subgrid-scale as stated in Equation (10), the term $|\bar{S}|$ can be calculated from Equation (11). Here, the terms d , κ , V , & C_s in Equation (10) denote distant to the closest wall, the Kármán constant, volume of the computational cell, and, Smagorinsky constant respectively.

$$L_s = \min(\kappa d, C_s V^{1/3}), \quad (10)$$

$$|\bar{S}| \equiv \sqrt{2\bar{S}_{ij}\bar{S}_{ij}} \quad (11)$$

Reynolds-Averaged Navier-Stokes (RANS) turbulence model

Unlike the swirl-stabilized combustor, the total inlet stream is divided into multiple-smaller streams reducing/eliminating the swirl-effect and make the flow with reduced turbulent and more towards laminar regime. Due to this, (RANS) turbulence model is applied in this section in-stead of computationally heavier LES model. Fluent 19R3 ANSYS, where the following Equations (12) and (13), respectively, are utilized to calculate the dissipation rate (ϵ) & kinetic energy (k). Here, G_k and G_b correspond to generation of turbulence kinetic energy as result of velocity gradients and buoyancy, while Y_M implies to contribution of fluctuating dilatation in compressible turbulence to overall rate of dissipation. The terms $C_{1\epsilon}$, $C_{2\epsilon}$ and $C_{3\epsilon}$ are the constants, where as Prandtl numbers for k and ϵ , respectively relate by σ_k and σ_ϵ . Finally, the turbulent viscosity (μ_t) is calculated by combining the ϵ & k , as presented in Equation (14) [107].

$$\frac{\partial}{\partial t}(\rho k) + \frac{\partial}{\partial x_i}(\rho k u_i) = \frac{\partial}{\partial x_j} \left[\left(\mu + \frac{\mu_t}{\sigma_k} \right) \frac{\partial k}{\partial x_j} \right] + G_k + G_b - \rho \varepsilon - Y_M + S_k \quad (12)$$

$$\begin{aligned} \frac{\partial}{\partial t}(\rho \varepsilon) + \frac{\partial}{\partial x_i}(\rho \varepsilon u_i) \\ = \frac{\partial}{\partial x_j} \left[\left(\mu + \frac{\mu_t}{\sigma_\varepsilon} \right) \frac{\partial \varepsilon}{\partial x_j} \right] + C_{1\varepsilon} \frac{\varepsilon}{k} (G_k + C_{3\varepsilon} G_b) - C_{2\varepsilon} \rho \frac{\varepsilon^2}{k} + S_\varepsilon \end{aligned} \quad (13)$$

$$\mu_t = \rho C_p \frac{k^2}{\varepsilon} \quad (14)$$

3.4.4 Reaction model

The combustion of CO₂-diluted CH₄/O₂ was described utilizing the partially premixed model combustion, which was expanded to support fully premixed combustion by choosing the inflow mean mixture percent. In ANSYS FLUENT, the partially premixed model, which is a arrangement of diffusion & premixed combustion models, has the advantages of both preceding types while also enabling more control over reactions & species than the model for premixed combustion. [107]. Transport equations for mean mixture fraction ($\bar{f}$), mean reaction progress variable ($\bar{c}$), & mixture fraction variance ($\bar{f'^2}$), have been resolved to acquire other scalar variables like density weighted average scalars. The mass fraction of a suitable species, which is zero in the unburned mixture and 1.0 in the completely burned mixture, is the reaction progress variable in this case. The fraction of the mixture can be estimated. from Equation (15), where Y and s are respectively mass fraction & oxygen to fuel ratio. The term Y_{fu} denotes the fuel mass fraction, Y_{ox} is the oxygen mass fraction, the subscripts 0 and 1 relate to the oxidizer and fuel inlet streams, respectively.

$$f = \frac{sY_{fu} - Y_{ox} + Y_{ox,0}}{sY_{fu,1} + Y_{ox,0}} \quad (15)$$

Equation can be used to calculate temperature and species fractions, which are density-weighted average scalars, using the probability density function (PDF) and the progress variable (16). Subscripts, u and b , in this expression indicates the unburnt and burnt mixtures, respectively.

$$\bar{\phi} = \bar{c} \int_0^1 \phi_b(f) p(f) df + (1 - \bar{c}) \int_0^1 \phi_u(f) p(f) df \quad (16)$$

Alternatively, the progress variable can be stated using the mass fraction of the species and calculated using Equation (17), here Y_i and $Y_{i,ad}$ are the mass fraction of the species as a result of complete combustion and the mass fraction of the species after that. The progress variables have values ranging from 0 to 1, with 0 indicating an unburned mixture and 1 indicating a burnt mixture.

$$c = \left(\frac{\sum_{i=1}^n Y_i}{\sum_{i=1}^n Y_{i,ad}} \right) \quad (17)$$

Equation can be used to express the rate of product formation, also known as PFR, which is the rate at which product formation occurs (18).

$$PFR = \frac{\Delta[Products]}{\Delta t} \quad (18)$$

Thickness of flame (δ) can be premeditated from the correlation among unburnt thermal conductivity (λ_u), specific heat capacity (c_p), density (ρ_u), and laminar speed of flame (S_L), as shown in Equation (19).

$$\delta = \frac{\lambda_u}{\rho_u c_p S_L} \quad (19)$$

In CHEMKIN, the laminar speed of flame (S_L) for premixed reactants can be calculated using a method for approximation, utilizing Equation (20), here T^0 , T_u , T_b , and $Y_{f,u}^m$ correspond to layer temperature, unburned temperature, burned temperature, and mass

fraction of the unburned fuel.[108], [109] The work by Göttgens et al. provides more information on the expression. [108]

$$= AT^0 Y_{f,u}^m \frac{T_u}{T^0} \left(\frac{T_b - T^0}{T_b - T_u} \right)^2 \quad (20)$$

The Damköhler number (Da), assessed from equation (21), indicates to the proportion of rate of reaction to rate of diffusion. The terms u' & l_t depict scale root mean square (RMS) of fluctuation of velocity and the integral length.

$$Da = \frac{l_t / u'}{\delta / S_L} \quad (21)$$

It is possible to calculate power density. from Equation (22). Here, $\dot{m}_{fuel}$ and CV_{fuel} are the rate of mass flow & fuel's calorific value, whereas $V_{combustor}$ and $P_{combustor}$ are the volume and pressure of the combustor [110], [111]. Calorific values of the methane, CO, and H₂, are respectively 55.5, 10.1, 141. and 8, MJ/kg, [112].

$$PD = \frac{\dot{m}_{fuel} \times CV_{fuel}}{V_{combustor} \times P_{combustor}} \quad (22)$$

factor of Pattern (PF) as given in Equation (23) resembles to the uniformity of temperature at the outlet of combustor, here $T_{maximum}$ and $T_{average}$ are the maximum and average temperatures at the combustor's output, respectively [76].

$$Pattern\ factor = \frac{T_{maximum} - T_{average}}{T_{average} - T_{inlet}} \quad (23)$$

Thermal power (TP) is the power of the exhaust gases as expressed in Equation (24), where $\dot{m}_{air\ or\ oxidizer}$ and $\dot{m}_{fuel}$ refer to the mass flow rates of oxidizer and fuel, respectively. Here C_p is the mean heat capacity of the oxidizing mixture [113].

Thermal power

$$= (\dot{m}_{air\ or\ oxidizer} + \dot{m}_{fuel}) \times C_p \times (T_{average,outlet} - T_{inlet}) \quad (24)$$

3.5 Validation of the developed model

The experimental data from the authors' previous study on the combustion of oxy-propane served as the basis for the validation of the swirl-stabilized combustion numerical design model utilized in this study [114]. The case used for validation has $\Phi=89\%$ and $OF=0.30$. Using the current numerical model, the experimentally measured temperatures were compared to the computed radial (at $z=10$ cm) and axial (at the combustor centerline) temperature distributions, as shown in Table 11. The distributions reveal that the calculated and recorded data are in good agreement. In the direction of the combustor axis, there is a tiny variation between the experimental and numerical temperatures. The extreme temperature of flame core and inaccurate temperature measurement of thermocouple are the causes of this. This temperature difference can also be significantly influenced by simulation inaccuracies. However, when you go closer to the combustor wall, the temperature difference becomes insignificant. The quartz combustor wall specifications have been thoroughly tested against the physical configuration. Additionally, the LES turbulence model and the PDF model utilized in this study's numerical simulations are extensively utilized & supported in the open literature for premixed turbulent flames [115]–

[117]. The comparisons in the section that follows show a good degree of agreement between experimental data & numerical results in terms of flame macrostructure. Further validation work has also been conducted for the oxy-methane and hydrogen-enriched oxy-methane flames. Numerical analysis of the same combustion chamber using the same combustor for oxy-methane combustion can provide information of the validation procedure for $\text{CH}_4/\text{O}_2/\text{CO}_2$ [118], $\text{CH}_4/\text{H}_2/\text{O}_2/\text{CO}_2$ [118], [119] flames.

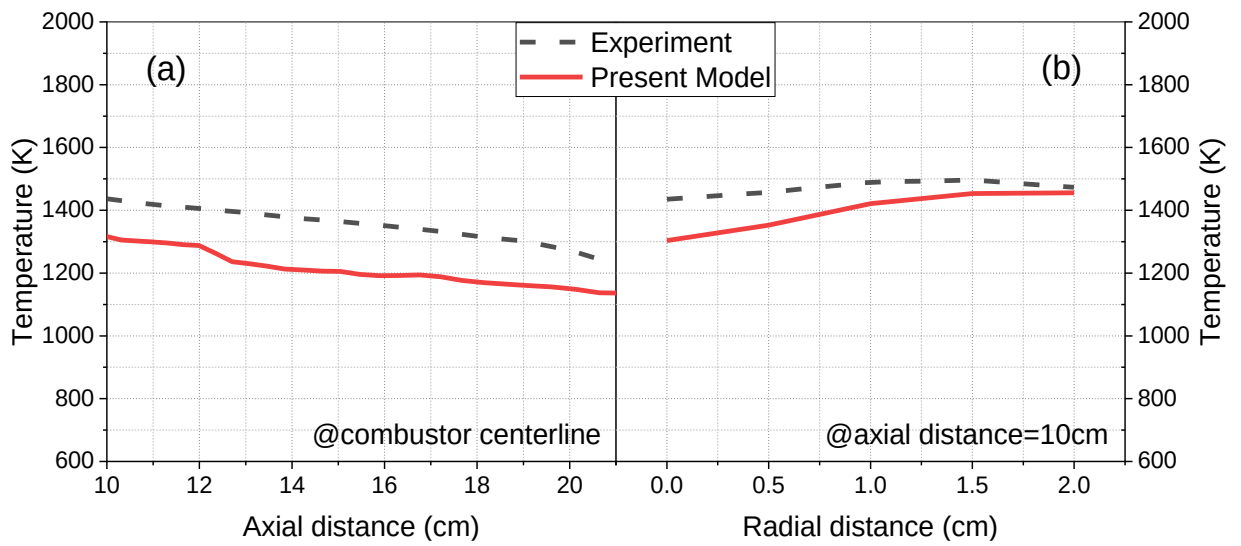


Figure 11 Experimental temperature profiles versus Numerical temperature profiles along (a) axial direction and, (b) radial direction at $\text{OF}=0.30$, $\Phi=89\%$. and axial location of $z=10$ cm, , Experimental data from Ref [115].

CHAPTER 4

RESULTS AND DISCUSSION

4.1 Experimental Results of CO₂-diluted oxy-methane flames

4.1.1 Combustor Stability Maps

The stability of the premixed oxygen-methane (CH₄/O₂/CO₂) flames is quantified for DLE combustor of gas turbine over ranges of oxygen fraction (25%-70%), and equivalence ratio (0.1-1) at fixed inlet velocity of 5.2 m/s. Stability maps (blowout & flashback limits) are presented. OF(Oxygen-Fraction) is supposed to play a major role in oxy-fuel flames, particularly in influencing essential combustion characteristics such as adiabatic flame temperature (AFT), which is critical for premixed flame investigations. Figure 12 illustrates the obtained stability maps for 55 degree swirler stabilized oxy -methane flames. For each *OF*, blowout point is designated by triangle icon, whereas flashback point is portrayed with diamond symbol. The exponential fits were observed for adequately representing blowout and flashback limits. The stable zone of combustion is shown by the area between the fit lines, while the blowout and flashback zones are represented by the areas below and above. Figure 13 illustrates the obtained stability maps for 45 degree swirler stabilized oxy -methane flames. For each *OF*, the round symbol represents the blowout point, while the square symbol represents the flashback point.

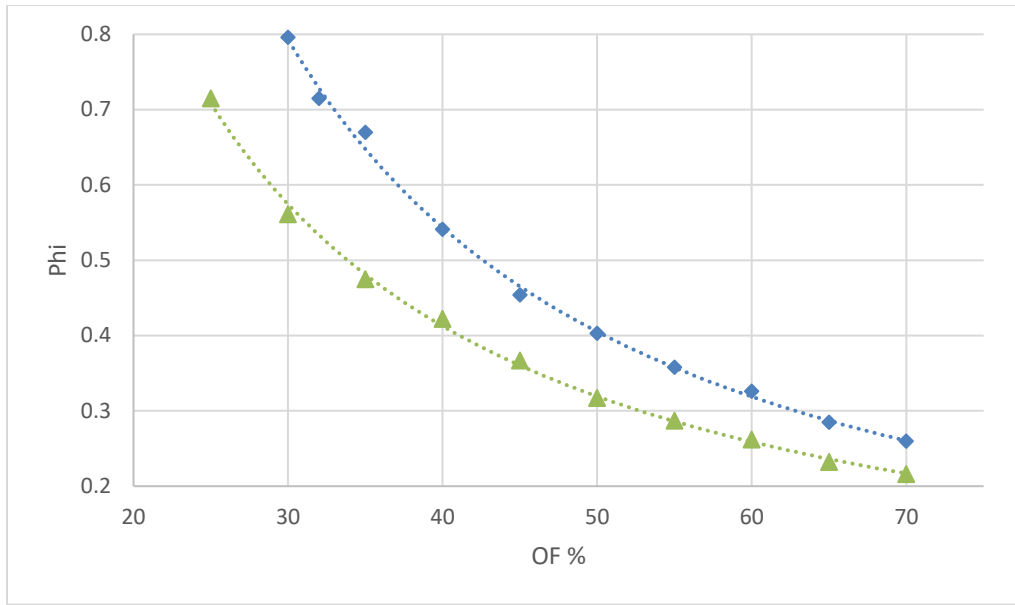


Figure 12 Stability map for 55 degree swirler (diamond icon flashback line and triangle icon blowout line).

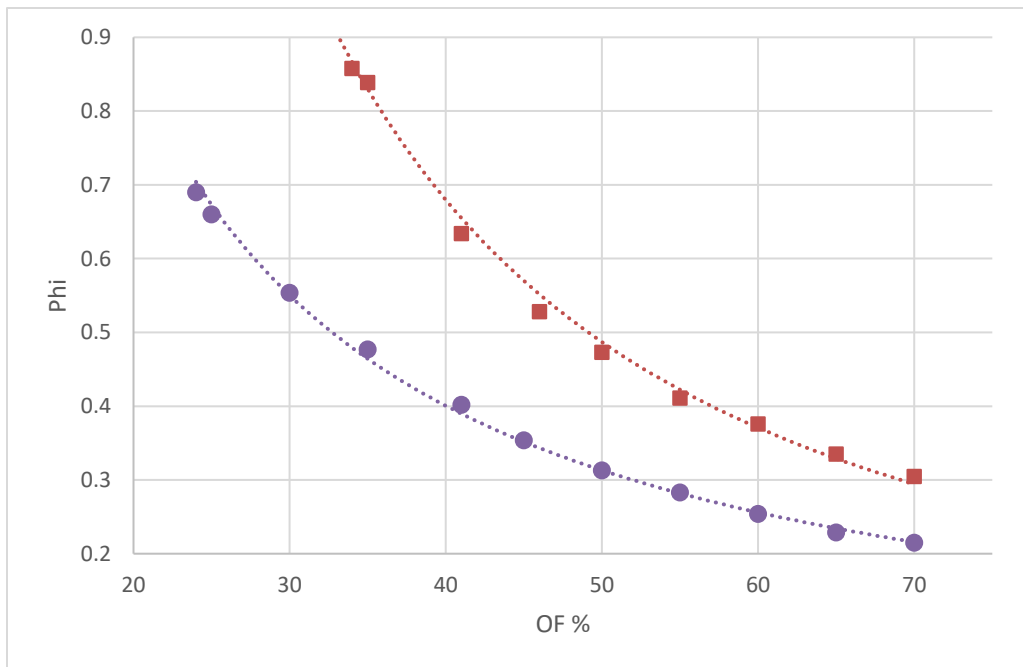


Figure 13 Stability map for 45 degree swirler. (Square icon flashback line and round icon blowout line)

Figure 14 illustrates the comparison of stability map, for flashback and flame blowout limits, for 45 and 55 degree swirler of the model gas turbine combustor for oxy -methane flames. It is observed that the flashback limits of 45 degree swirler flames are reached at higher ϕ in comparison with those of the 55 degree swirler flames for the same OF. The results showed that the lower the angle the wider the stable zone. Therefore 45 degree swirler has wider stable zone in comparison to 55 degree swirler. The stable zones for Both the swirler narrow down toward higher OF and lesser ϕ . The blowout curve on the stability map similarly follows a constant AFT shape, however the flashback curve does not. There are very little change for blowout with swirler angle but the higher the angle the lower the flashback line.

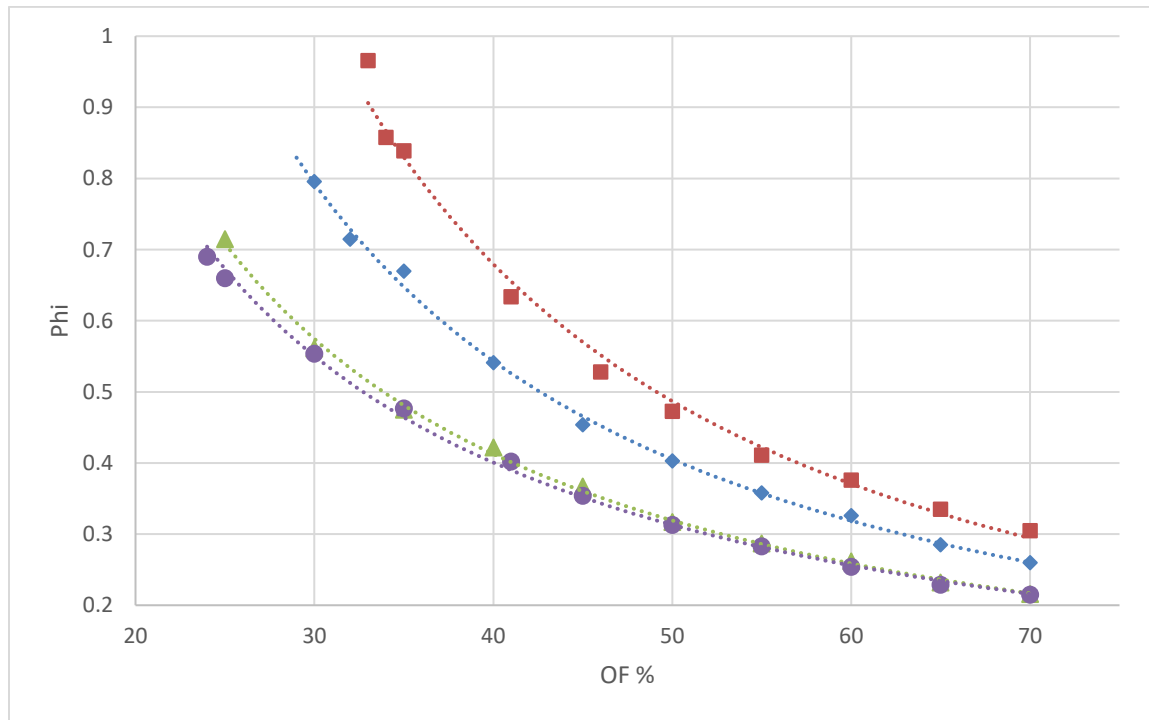


Figure 14 Comparison of Static stability map, in terms of flashback limits & flame blowout, for 45 and 55 degree swirler of the model gas turbine combustor holding CH₄/O₂/CO₂ flames

A wider stability map, indicating a broader range of stable operating conditions, can be achieved with a 45-degree swirler compared to a 55-degree swirler when using oxy-methane combustion. In 55 degree swirler the flame is short and wider and close to upstream. Boundary layer induced flashback occurs early in 55 degree swirler in comparison to 45 degree swirler. As a result, the stable zone is narrower for 55 degree swirler.

Lower flame flashback tendency: The broader swirling flow of the 45-degree swirler reduces the likelihood of flame flashback by distributing the flame over a larger area. This minimizes the chance of the flame propagating back into the oxy-methane supply, improving safety and expanding the stability map. Enhanced flame stabilization: The 45-degree swirler provides better flame stabilization characteristics due to its broader flame shape. The extended flame length increases the likelihood of flame stabilization, enabling the flame to sustain combustion over a wider range of oxy-methane compositions. In summary, the wider stability map achieved with the 45-degree swirler for oxy-methane combustion is a result of reduced flame flashback tendency and enhanced flame stabilization capabilities compared to the 55-degree swirler.

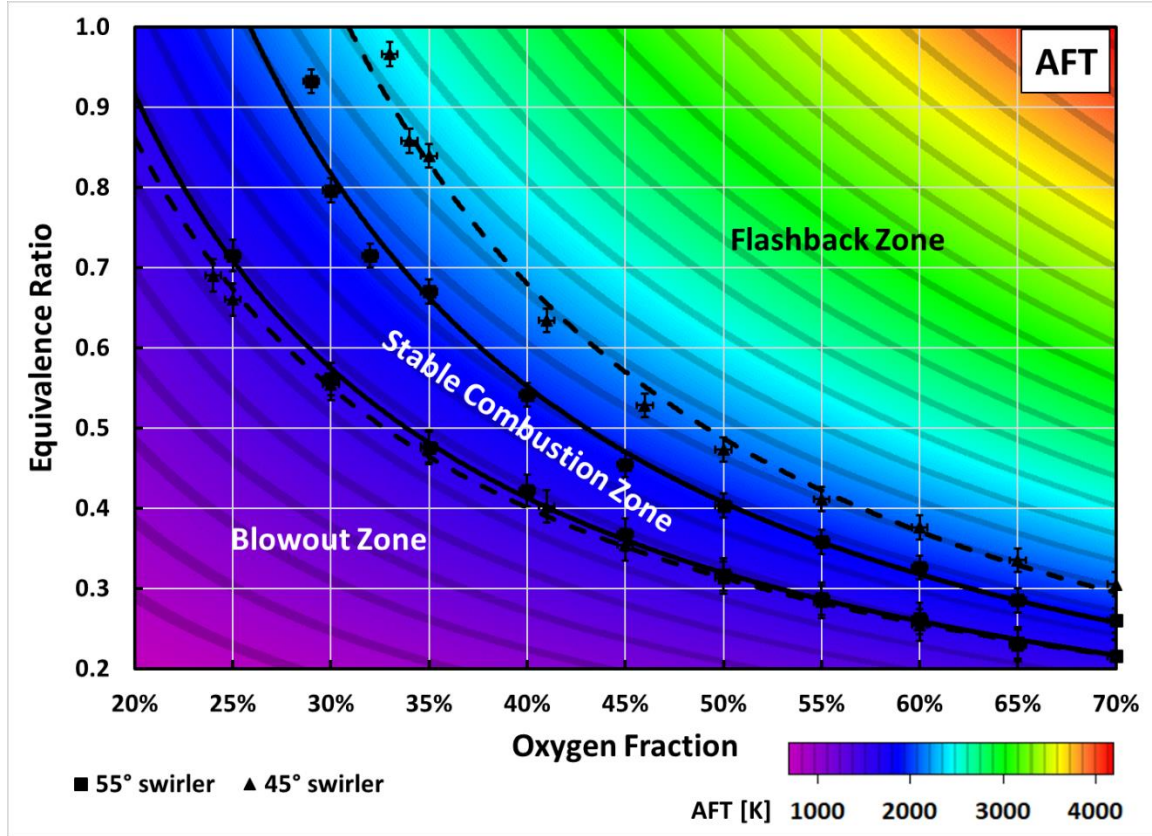


Figure 15 Combustor stability maps of CH_4/O_2 flames along with constant AFT curves, with a throughout fixed inlet velocity of 5.2 m/s.

Figure 15 illustrates that the blowout curve on the stability map follows a constant AFT shape, however the flashback curve does not. There is a sharp increase of equivalence ratios with the decrease of oxygen fraction for flashback line. Figure 16 shows the comparison of stability maps for 55 degree and 45 degree swirler. The blue line is for 45 degree swirler and red line indicates for 55 degree swirler. It is obvious that 45 degree swirler has wider stable zone narrowing down towards the higher oxygen fractions.

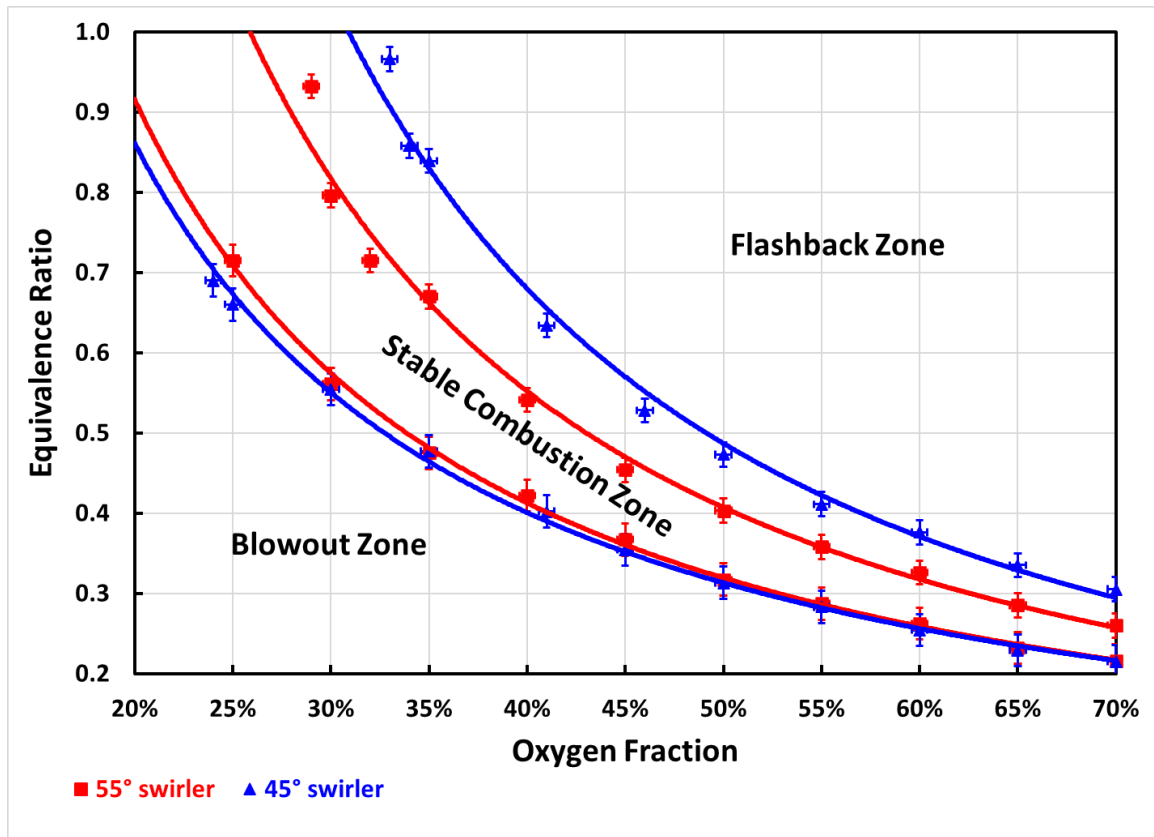


Figure 16 Comparison of the stability maps of 55 degree and 45 degree swirler

4.1.2 Flame Structure Analysis

Test Conditions for Capturing Images of Flames

Table 2 Flame-imaging matrix

Flame	OF	phi	AFT [K]
1	35%	0.505	1670
2		0.640	1940
4	50%	0.333	1670
5		0.420	1940
7	65%	0.240	1670
8		0.302	1940
10	32.4%	0.568	1700
11	37.7%		1900
12	46.4%	0.374	1700
13	53.5%		1900
14	60.7%	0.269	1700
15	69.4%		1900

Table 2 gives the flame imaging matrix for experimental work. The figure 17 gives a comparative image of flame shape and macrostructure for 45° and 55° swirler stabilized flames. Recirculation zones, in particular the ORZ at the combustor wall and the IRZ close to the centerline of combustor, are necessary for the stability of the flame. For 55° swirler the flame is wider and shorter while on the other hand for 45° swirler the flame is narrower and longer because of effect of IRZ and ORZ.

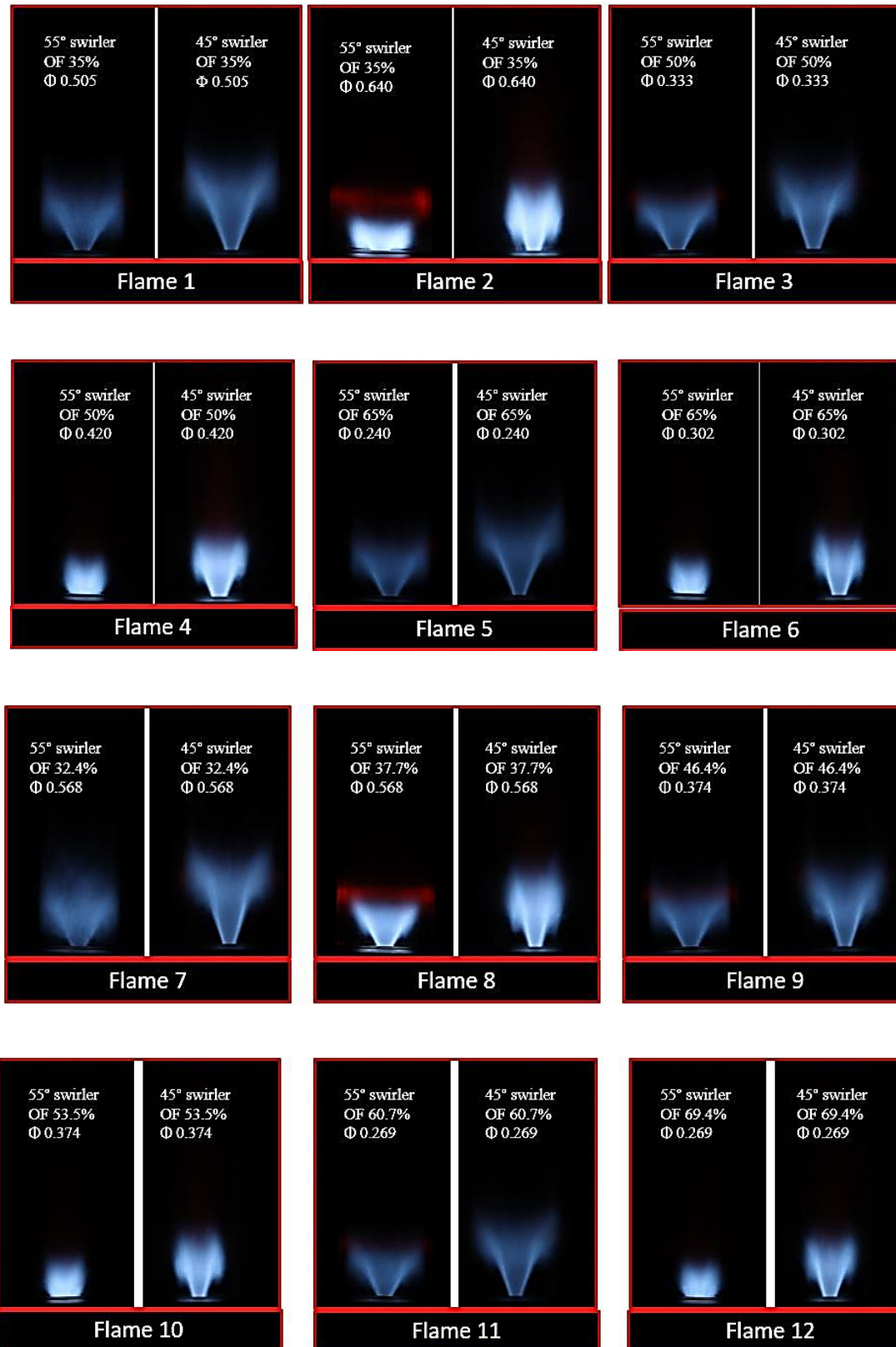


Figure 17 Comparison of flame images for 55 degree and 45 degree swirler

Comparing each flame for both the swirler it is evident that 45° swirler produce taller and more stable flames compared to 55° swirler. Flame 10 and 12 show that for the same OF, phi and AFT as compared to 45° swirler, 55° swirler produce louder noise and stronger flames as it is close proximity of flashback. While 45° swirler flames are stable and less noisy and away from flashback limit.

4.1.3 Dependence of Flame Macrostructure on AFT

Flames 1, 3, 5 have different OF and phi ϕ but have a common AFT of ~ 1670 K. These flames show common shape. Same way Flame 2, 4, 6 have a common shape due to having a common AFT of ~ 1940 K while their OF and phi ϕ is different. This highlights the clear correlation between macrostructure of flame and reaction kinetics & AFT at constant bulk velocity at inlet and is in complete agreement with Shroll et al findings [21]. Figure 18 illustrates the transition from IRZ to ORZ for 55° swirler combustor. We can see the transition happens very quick as the stable zone is narrower.

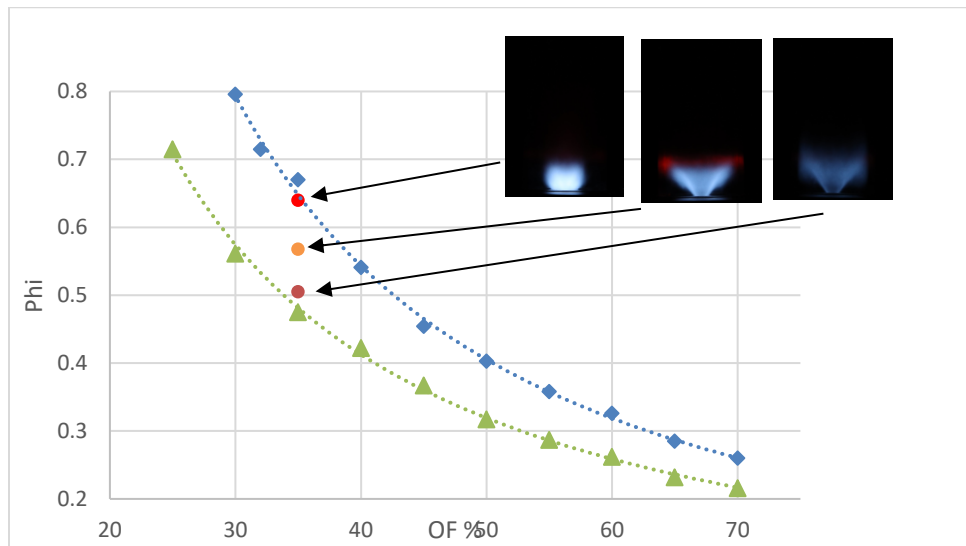


Figure 18 Transition from IRZ to ORZ stabilization for 55 degree swirler combustor

Figure 19 illustrates the transition from IRZ to ORZ for 45 degree swirler combustor. We can see the transition happens comparatively slow as the stable zone is wider.

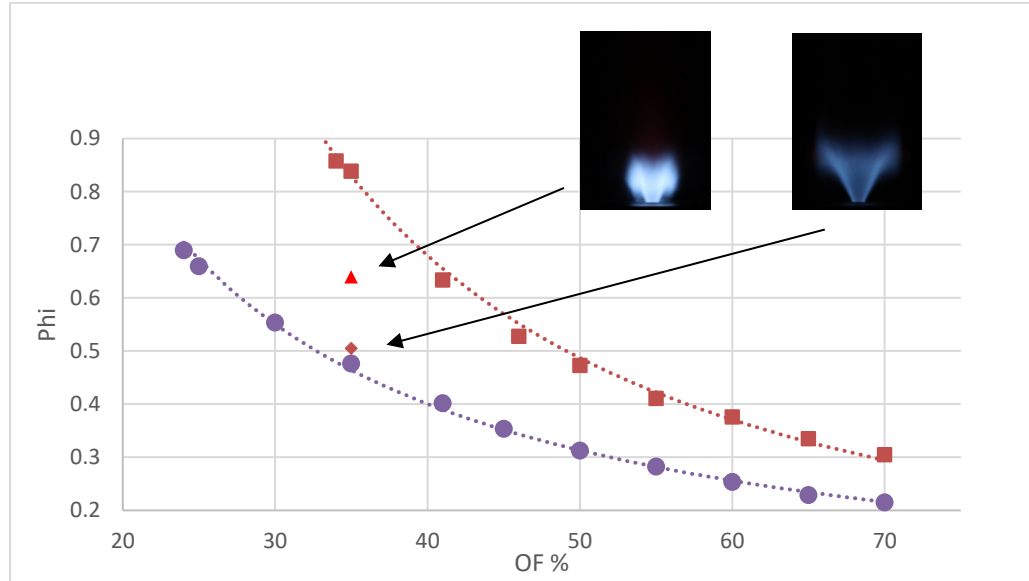


Figure 19 Transition from *IRZ* to *ORZ* stabilization for 45 degree swirler combustor

4.1.4 Characterization of Flame Temperature

To learn more about the connection between AFT and the flame structure, flame temperature was monitored at several points inside the quartz confinement. In some cases, reference values for the numerical simulations of premixed turbulent reacting flows under conditions of oxy-combustion can be obtained from measurements of flame temperature.

Table 3 Test condition of Temperature Measurement

<u>Temperature-measurement matrix for</u> <u>55° swirler</u>			
Flame	OF	phi	AFT [K]
1	35%	0.505	1670
3	50%	0.333	1670
5	65%	0.24	1670

Local mean temperatures for flames 1, 3, and 5's operating conditions are listed in Table 3, and their subsequent flame forms are shown in figure 20. The axial distance, Z , was used to reference the temperature observations (inch). The exit plane of the burner divergent section, or the base of the quartz confinement, was given a bottom reference value of $Z = 0$ inch, and the top value of $Z = 30$ cm/11.8 inch was given to the quartz confinement's departure plane.

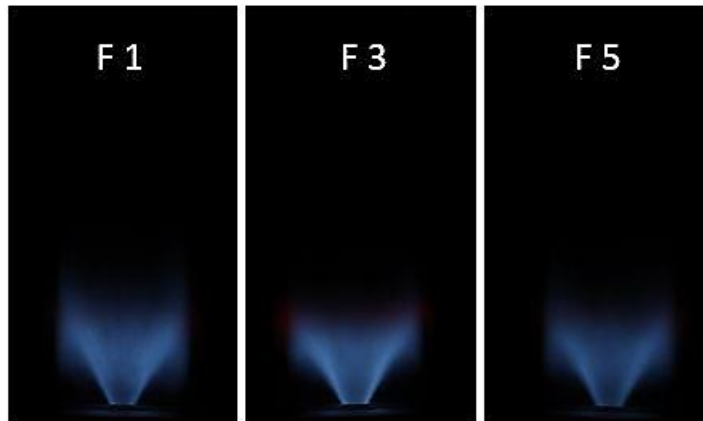


Figure 20 Images of flames 1,3, and 5 listed in Table 3

The centerline axial temperature distribution of the chosen flames is shown in Figure 20 from $Z = 1.311$ inch to $Z = 11.811$ inch above the flame structure. Since the AFT values of the two flames are remarkably comparable, identical tendencies may be seen in their

temperature profiles. Additionally, Table 3 shows that, even though the AFT of the two flames is about similar, the combination of ϕ and OF is very different. Additionally, flame 5 has a slightly higher temperature than the others at practically every axial region, demonstrating that the calculated temperatures are marginally different from the theoretical AFT (see Figure 21).

While experimenting with various flames, it was discovered that as the flame's AFT rises, so does the noise emanate from the quartz tube. Figure 20 demonstrates that AFT has a significant influence on flame shape because flames 1, 3, and 5 all have a similar flame shape. Although the OF & ϕ conditions of the two flames are very diverse, just a similar AFT causes the flame figures to resemble one another. These results support AFT's position as the primary determining factor of flame structure and stability.

In summary, the axial temperature profile exhibits a consistent trend for different oxygen fractions and equivalence ratios in the 55-degree swirler configuration when targeting the same adiabatic flame temperature. This is due to the thermal equilibrium achieved at the desired temperature, similar combustion kinetics, and the efficient heat transfer effects within the flame. These factors result in a uniform temperature distribution along the axial direction, regardless of variations in oxygen fraction and equivalence ratio.

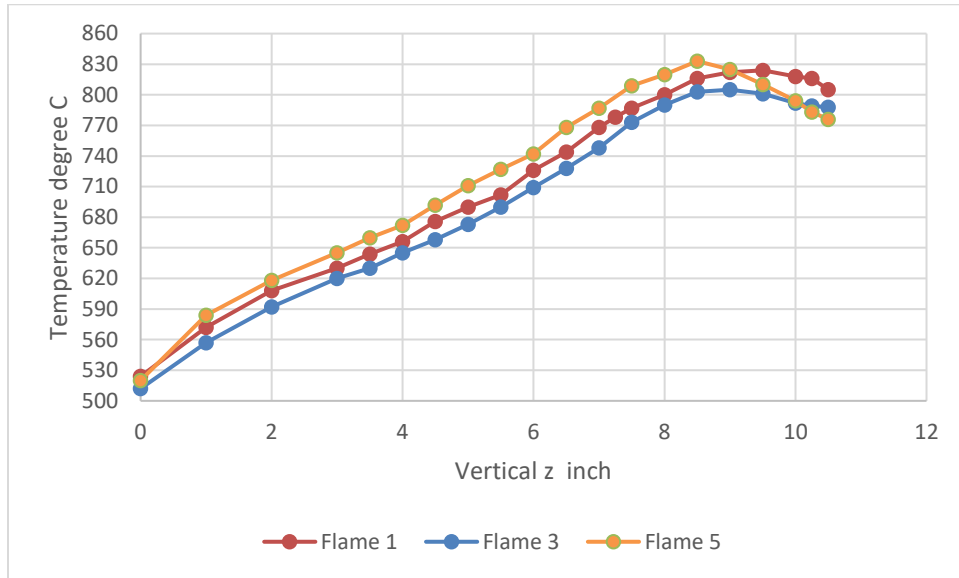


Figure 21 Axial centerline temperature distributions for flames 1, 3 and 5

The kinetics of burning prevent flames from being regarded as slug flow reactors. For a better understanding of the flame structure, the radial temperature profiles of the various flames were also measured at a few axial points, $Z = 9.8$ inch, $Z = 3.3$ inch, $Z = 2.8$ inch, and $Z = 2.3$ inch. The findings are shown in Figures 22 and 23, respectively.

Figure 22 depicts radially outward from the plume and toward the quartz tube wall locations with decreasing temperatures. The temperatures are higher axially downward towards the vicinity of flame core according to the figures 22-25. The two figures' temperature profiles hint at the flow field's two-dimensionality. Because the combustor has an inner recirculation zone (IRZ), the maximum temperatures are not along the centerline of the combustor. With a reduction in axial distance, higher radial distances yield the highest temperatures.

Figures 24 and 25 both depict the highest temperature at radial distances of 0.6 to 0.8 inches and 0.8 to 1 inches, respectively. The discrepancy here suggests that there is a local IRZ of relatively bigger size near the burner's exit point, which subsequently gradually shrinks toward the downstream along the axial direction.

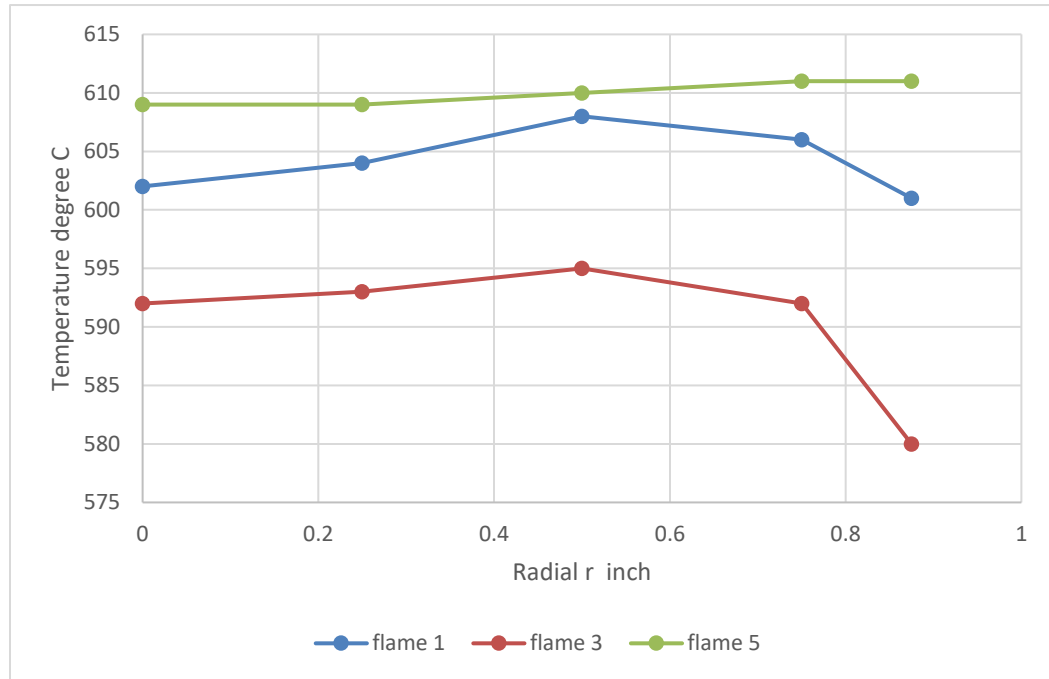


Figure 22 Radial Temperature distribution for flames 1, 3, and 5 at Z=9.8 inch

Figure 22 shows the radial temperature away from the centerline of the flame at a distance $Z=9.8$ inch from the burner base, and is the furthest downstream point measured, and shows significant difference in behavior for each flame, with the radial temperature distribution for flame 5 remaining nearly constant, whereas both flame 1 and flame 3 undergo great variation. The variation in flame 3 is particularly significant, as the temperature falls more than 10 degrees Celsius along the axial line, with a slight increase in temperature being observed at an axial distance of 0.5 inch. Temperature fall along the axial line is also

observed for flame 1, albeit the overall drop as measured from the centerline to the farthest point is much smaller. More significantly, the temperature increase at 0.5 inch is much larger, around 5 degrees Celsius.

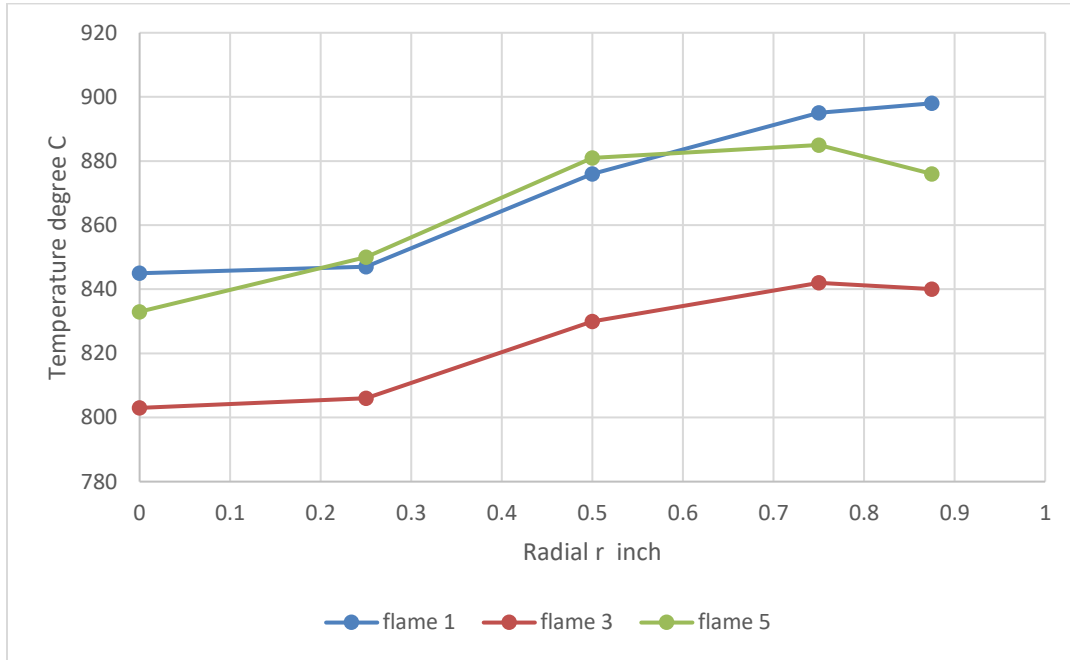


Figure 23 Radial temperature distributions for flames 1 , 4 and 7 at Z = 3.3 inch

Unlike the downstream scenario, radial temperature variation at points nearer to the burner are observed to follow similar trends, albeit to different extents. Figure 23 shows the radial temperature distribution at a distance of 3.3 inch in axial direction from the burner base, and the flames are observed to behave similarly, with temperature increasing in the radial direction away from the centerline. However, the magnitude of the temperatures varies between the flames, with the flame 3 operating at a much lower range compared to flame 1 and 5. Regardless, the overall temperature variation along the axis for each flame is

similar, 53 degree Celsius for flame 1, 37 degree Celsius for flame 3 and 43 degrees Celsius for flame 5.

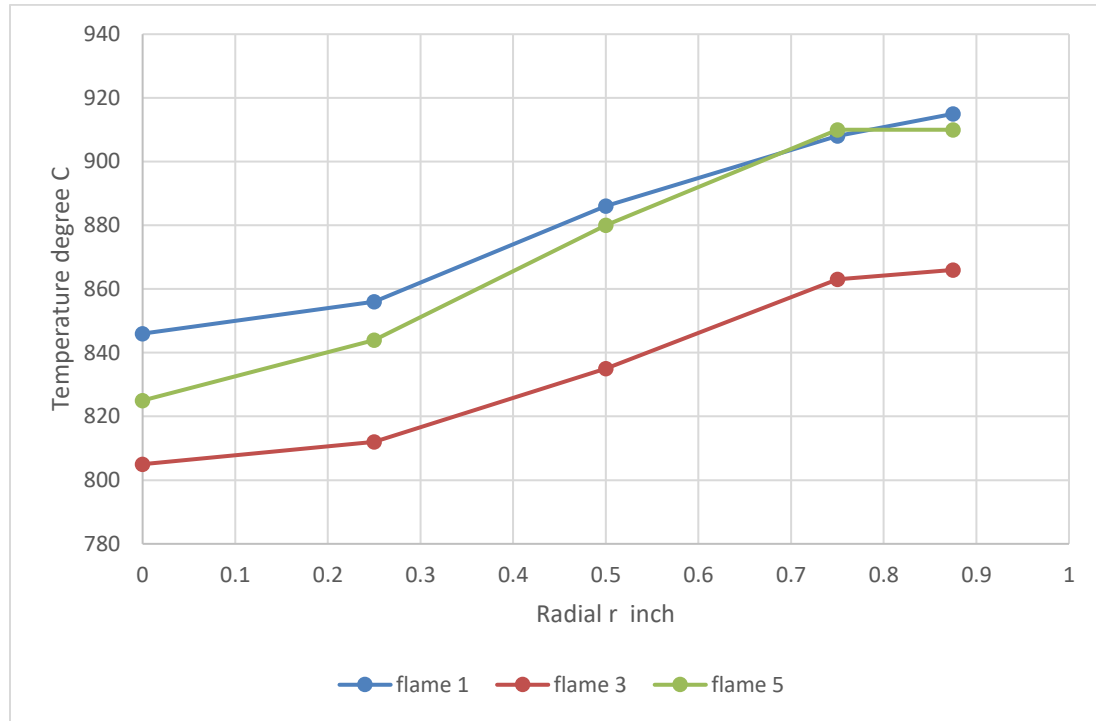


Figure 24 Radial temperature distributions for flames 1 , 3 and 5 at Z = 2.8 inch.

Similar trends as described above are observed at Z=2.8 inches, with flame 4 once again operating at a lower range. The variation in centerline temperature between the flames is of note, with flame 1 being at 846 degree Celsius, flame 3 at 805 degree Celsius, and flame 5 at 825 degree Celsius. Flame 1 and 5 are observed to reach near identical flame temperatures at the point farthest along the axis, around 910 degree Celsius, whereas the highest temperature measured for flame 3 is 866 degree Celsius at the farthest point along the measured axis.

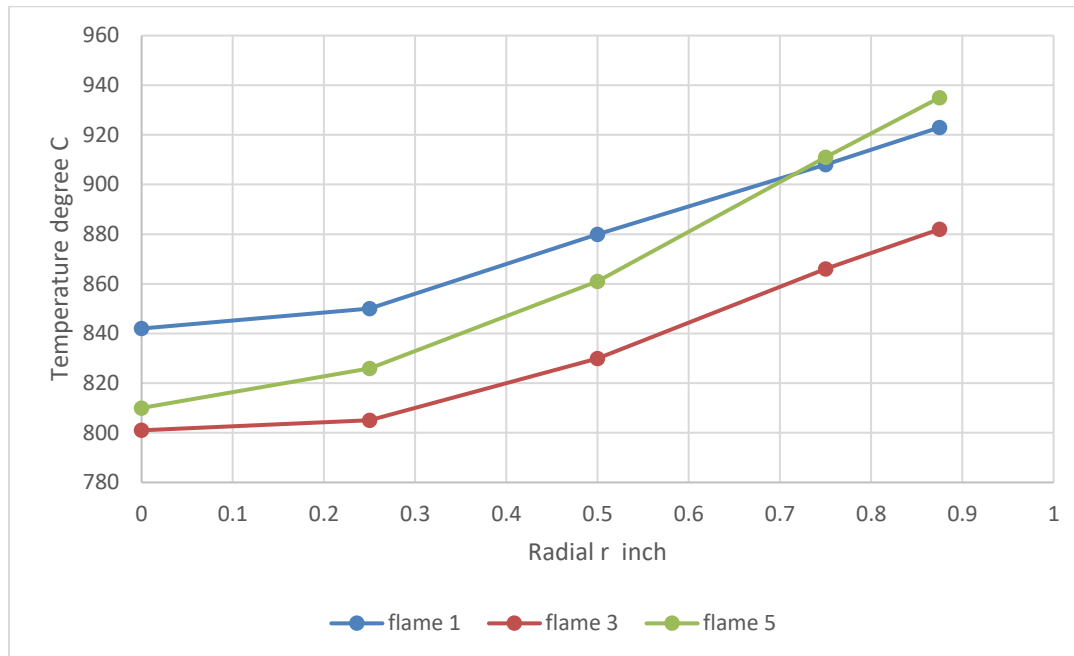


Figure 25 Radial temperature distributions for flames 1 , 3 and 5 at Z = 2.3 inch

For the axial line closest to the burner base, at Z= 2.3 inches, steep increase in temperature with increase in radial distance is observed, with flame 1 once again operating at the highest temperature range, and flame 3 at the lowest. The centerline variation is also similar to that observed at Z=2.8 inches, with flame 1 at 840 degree Celsius, flame 3 at 801 degree Celsius, and flame 5 at 810 degree Celsius. The magnitude of the temperature increase is however much larger at this lower axial distance, with an overall increase of 80 degree Celsius being observed for flame 1 and 3, and 125 degree Celsius for flame 5.

4.2 Simulation Results of CH₄/O₂/CO₂ flames

Table 4 shows the Oxy-methane cases simulated for various swirl angles with different OF and Equivalence ratio ϕ .

Table 4 Oxy-methane cases to be simulated for various swirl angle.

swirler	Case	Oxygen fraction	Equivalence ratio ϕ
55°	1	35%	0.568
	2	50%	0.374
	3	65%	0.269
45°	4	35%	0.568
	5	50%	0.374
	6	65%	0.269

4.2.1 Structure and Stability of CO₂-diluted oxy-methane flames: Effect of Swirl

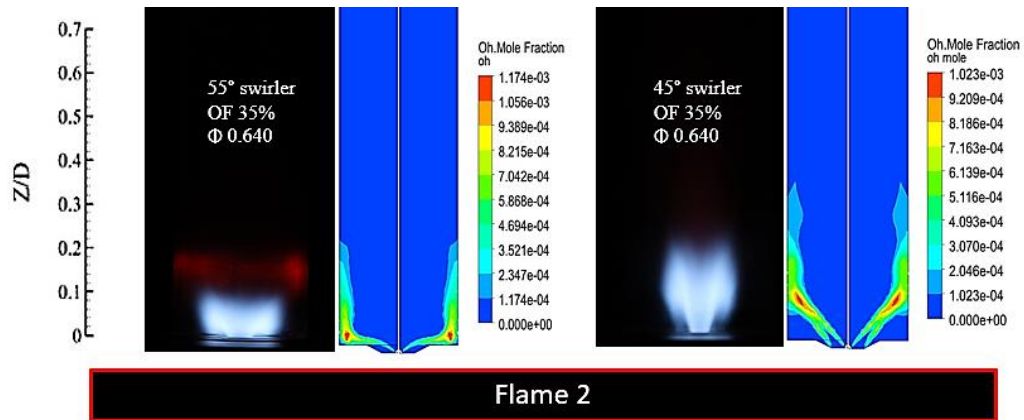


Figure 26 Numerical (OH contour plots) and Experimental (images taken with high-speed camera) flame structure comparison for 55 degree and 45 degree swirler combustor

Figure 26 shows a comparative study for experimental and simulated results for flames in 55 degree and 45 degree swirler combustor. It illustrates symmetry in between experimental and simulation flames. The figure shows the OH mole fraction contours for the cases under investigation. The flame was found to be longer and more characteristically V-shaped at a lower degree of swirl angle (45 degree). The flame length decreases, and the flame widens as the Swirl angle rises (to 55 degrees), as shown in the illustration. The flame characteristics of a burner, such as its length and compactness, can be influenced by the design of the swirler. The angle of the swirler determines the swirling flow pattern and subsequent flame behavior. In the case of a 45-degree swirler, the higher axial velocity and longer swirling flow pattern allows for a longer flame length. The broader flame shape results in a lower flame compactness or density, as the combustion zone is spread out over a larger area. On the other hand, a 55-degree swirler generates a shorter swirling flow pattern. This has lower axial velocity. As a result, the flame length is shorter. Additionally, the restricted swirling flow leads to a higher flame compactness, as the combustion zone is more concentrated within a smaller area.

Therefore, the angle of the swirler plays a significant role in determining the flame length and compactness. When comparing a 45-degree swirler to a 55-degree swirler, the flame sheet thickness is generally larger with the 45-degree configuration. Because of lower power density. This is primarily because the broader swirling flow pattern of the 45-degree swirler promotes a larger and more dispersed flame shape. The wider distribution of the flame leads to a thicker flame sheet, representing the region where combustion reactions occur. In contrast, the narrower swirling flow pattern of the 55-degree swirler results in a more concentrated flame shape. The more confined combustion zone contributes to a

thinner flame sheet. Thus, the swirler angle influences the flame sheet thickness, with the 45-degree swirler typically producing a thicker flame sheet compared to the 55-degree swirler.

Table 5 Cases for Same OF and phi for different swirler:

cases	swirler	OF	phi
1	55 degree	35%	0.640
2	45 degree	35%	0.640
3	55 degree	35%	0.568
4	45 degree	35%	0.568
5	55 degree	50%	0.374
6	45 degree	50%	0.374

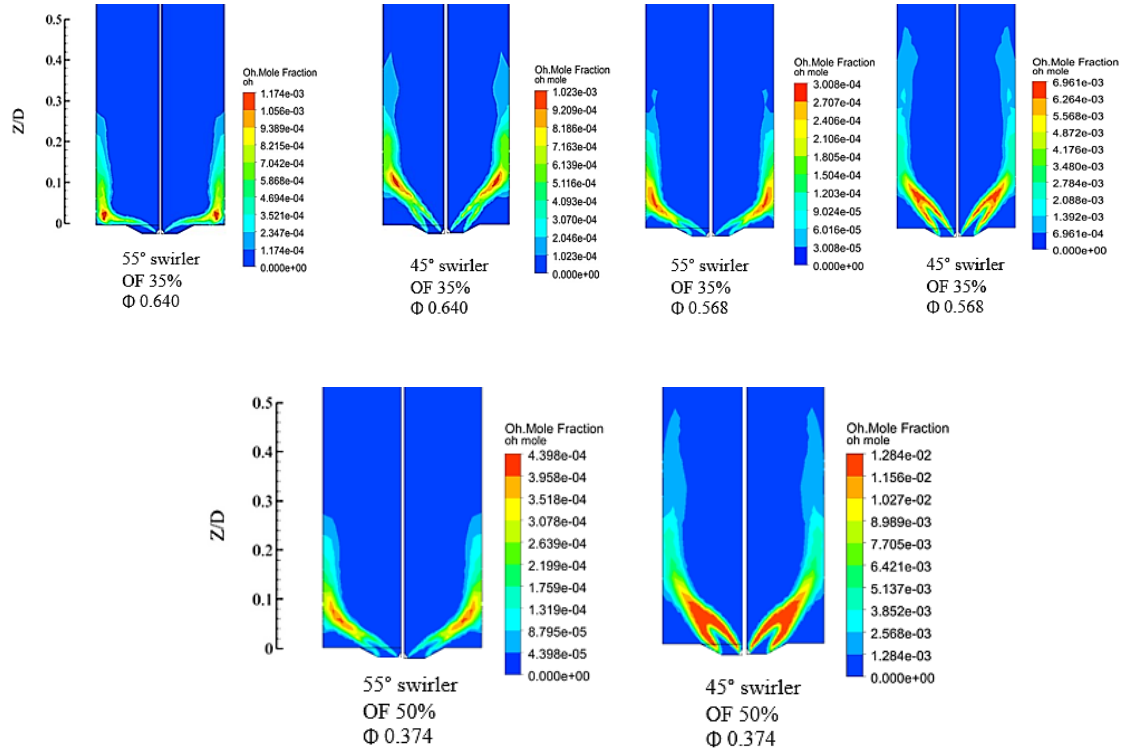


Figure 27 Numerical flame shapes of OH mole fraction at different swirler for same OF and phi.

The contour of OH mole fractions numerically investigated for OF 35% & 50%; $\phi = 0.640, 0.568$ & 0.374 , for 45° and 55° swirler. It is very clear from the figure that the flame becomes long and V shaped for 45° Swirler. As the swirl increases, the shape of the flame becomes wider and smaller. Figure 28 shows the velocity flow field mapped over temperature for OF=35% and 55%, $\phi = 0.640, 0.568$ and 0.374 for the swirl angle 45° and 55° . For higher swirl angle, here 55° , the eddy in IRZ is larger and this squeezed the outer ORZ. Moreover, the eddy in IRZ goes further downstream with decrease in swirl angle. Therefore, it can be said that for lower the swirl angle, the in IRZ is more compact and squeezed.

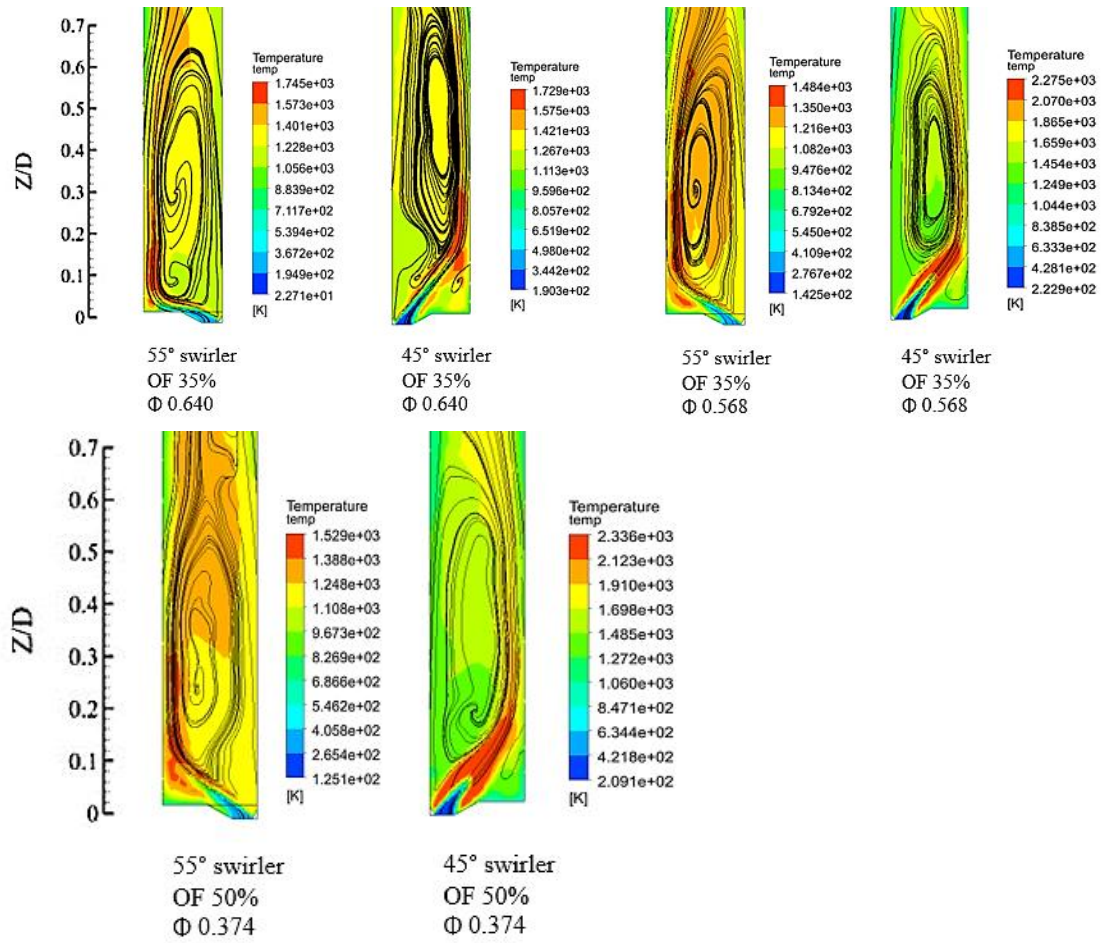


Figure 28 The temperature contour is mapped over by the velocity flow field at different swirler for same OF and phi

4.2.2 Structure and Stability of CH₄/O₂/CO₂ flames: Effect of equivalence ratio

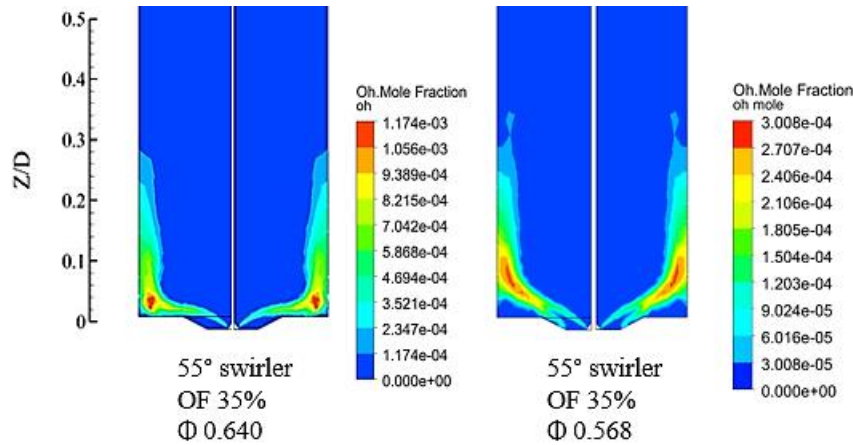


Figure 29 Mole fraction of OH for same swirler at 55° with different equivalence ratio

Figure 29 signifies the behavior of the flame at same swirl angle, 55° and same OF = 35% but different $\phi = 0.640$ and 0.568. While the equivalence ratio, ϕ decreases the flame becomes more V shaped. Furthermore, the length of the flame becomes sharper and more visible. In the other hand, the reverse phenomena happen while increasing the equivalence ratio. In addition, in this case, the flame become shorter and stronger which creates high noise.

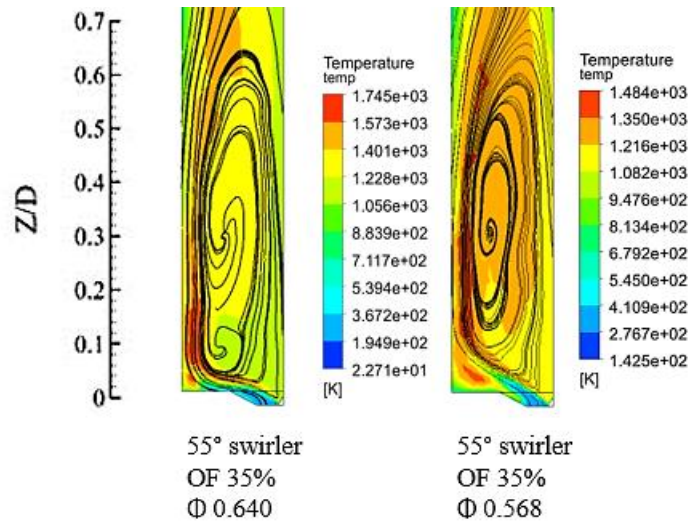


Figure 30 The temperature contour is superimposed over by the velocity streamlines for different equivalence ratio at 35% OF with same swirler angle at 55

Figure 30 illustrates velocity streamline mapped over Temperature at same swirl angle, 55° and same OF = 35% but different $\phi = 0.640$ and 0.568. In this situation, for high equivalence ratio, the flame temperature increases and flame becomes more compact. The stabilization of the flame switches from swirl stabilized to corner stabilized with a reduction in equivalence ratio. Figure 31 signifies the behavior of the flame at same swirl angle, 45° and same OF = 35% but different $\phi = 0.640$ and 0.568.

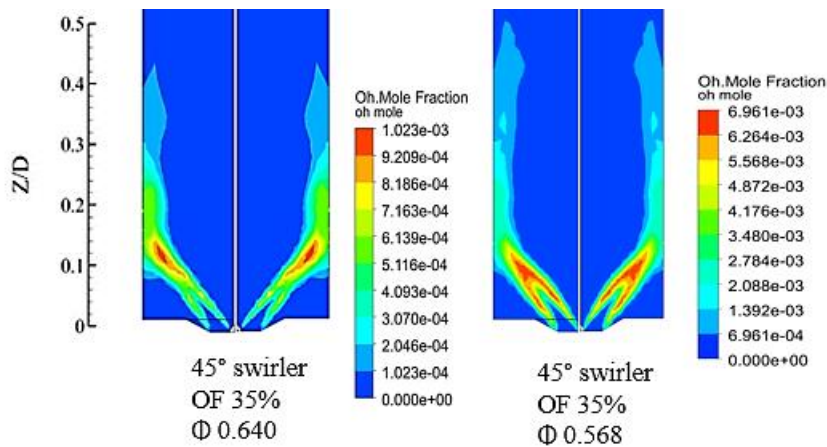


Figure 31 Mole fraction of OH for same swirler at 45 with different equivalence ratio

The flame becomes more V-shaped as the equivalence ratio reduces. Furthermore, the flame's length becomes sharper & more visible. When the equivalence ratio is increased, the opposite phenomenon occurs. Furthermore, in this case, the flame becomes shorter and stronger, producing a lot of noise. Previously, the same type of phenomena was experienced for 55° swirl. Figure 32 illustrates velocity streamline mapped over Temperature at same swirl angle, 55° and same OF =35% but different $\phi = 0.640$ and 0.568. As the equivalence ratio increases, the outer recirculation zone (ORZ) becomes stronger and the eddy in IRZ shifts further downstream this makes the flame shorter and compact.

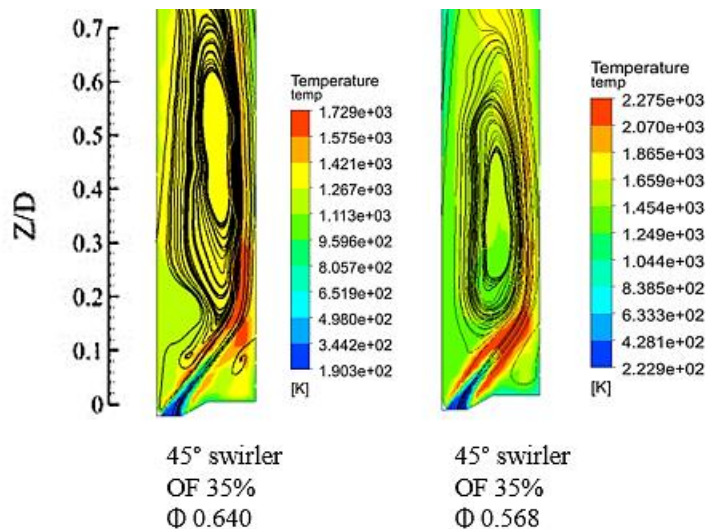


Figure 32 The temperature contour is superimposed over by the velocity streamlines for different equivalence ratio at 35% OF with same swirler angle at 45.

CHAPTER 5

CONCLUSION

- The combustion characteristics and stability map of premixed oxy-methane flames are investigated in this work for clean oxy-methane flexible combustion applications in gas turbines using a dry low emission (DLE) model combustor.
- The blowout and flashback limits of the oxy-methane flame are indicative of its stability over equivalence ratios (ϕ) of 0.1 to 1.0 and oxygen fractions (OF: volume concentration of oxygen in the O_2/CO_2 oxidizer) of 25 to 70%.
- Flame stabilization mechanism for example impact of swirl on flame stability is studied experimentally and simulation work is done for swirler with angles 45° , 55° .
- The results showed that the lower the angle the wider the stable zone. 45° swirler has wider stable zone compared to 55° swirler.
- Additionally, it was discovered that the blowout curve on the stability map follows a constant AFT shape while the flashback curve does not follow constant AFT.
- There is very little change for blowout with swirler angle but the higher the angle the lower the flashback line. 45° swirler has higher value of flashback line compared to 55° swirler because of boundary layer induced early flashback.
- The flame becomes long and V shaped for 45° Swirler. As the swirl increases, the shape of the flame becomes wider and smaller. It is because of the change of axial

velocity hence change in power density. The flame sheet in 45 degree swirler is thicker in comparison to 55 degree swirler because of low power density.

- As the equivalence ratio (ϕ) increases, the outer recirculation zone (ORZ) becomes stronger and the eddy in IRZ shifts further downstream this makes the flame shorter and compact.

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