ABSTRACT

BANERJI, AYAAN. Numerical Study of a Spark Assisted Compression Ignition (SACI) Engine Using the One-Dimensional Turbulence (ODT) Model. (Under the direction of Dr. Tarek Echekki).

Advanced combustion engine concepts like Spark Assisted Compression Ignition (SACI) are being investigated to provide a solution to the ever-increasing need for novel methods of power generation with high efficiency and low emissions. This study is conducted with the intent to understand the operating conditions for the effective functioning of this combustion concept using the one-dimensional turbulence (ODT) model. The stand-alone ODT model is formulated for engine simulations and can compute chemical interactions in unsteady fluid conditions with a high resolution, while offering computational economy. While spark ignition and compression ignition engines are widely used, SACI combustion is a relatively new concept, which still faces challenges to being accepted commercially. It offers better operating efficiency as well as greater control over combustion over a wide load range but involves numerous operational complexities for effective combustion control. This study investigates the effect major engine operational parameters have on SACI combustion of iso-octane fuel.

The study is divided into two parts. First, the port fuel injection (PFI) method is investigated, by varying operating parameters like compression ratio, equivalence ratio and spark timing. It was observed that spark ignition timing plays a significant role in combustion phasing. Equivalence ratio is observed to impact maximum in-cylinder pressure and combustion phasing. Higher equivalence ratio leads to greater in-cylinder pressure generation, as well as greater heat release during autoignition. Compression ratio increase leads to higher in-cylinder pressures as well as control over combustion phasing, leading to the conclusion that compression ratio is an important parameter while operating in SACI combustion mode.
For the second part of the study, a combined strategy using a combination of PFI and DI methods of fuel injection is employed. The aim is to investigate the effect of fuel-air stratification in the combustion chamber. The multipoint fuel injection (MPFI) strategy is found to control combustion phasing to a certain extent through charge stratification. In this part of the study, the parameters varied were injection timing into the cylinder and the mass fraction of total fuel directly injected into the cylinder. It is found that combustion phasing in SACI combustion is not very sensitive to changes in injection timing, however, a late injection timing causes unstable combustion due to lack of homogeneity. As the fraction of fuel injected directly increases, there is direct effect on autoignition observed. It is delayed with increasing direct injection mass, and becomes more erratic, reducing the in-cylinder pressure and overall heat release rate.
Numerical Study of a Spark Assisted Compression Ignition (SACI) Engine Using the One-Dimensional Turbulence (ODT) Model

by
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DEDICATION

To

Mother, Anindita Banerji, father, Bhargeshwar Banerji

Brother, Amartya Banerji

Family and friends.
BIOGRAPHY

Ayaan Banerji was born on 2nd January 1994, in Patna. He then moved to Mumbai and spent a major part of his life there. He attended St. Mary’s School, for his high school education and completed his Indian Certificate of Secondary education exam in May 2010. He then joined Manipal Institute of Technology, Manipal, Karnataka, India to pursue his undergraduate degree in Mechanical Engineering.

He soon found his passion with automobiles, when he joined Formula Manipal, the official FSAE university team. He worked with the engine designing and fabrication team, learning as much as he could about them. His interest in engine research grew while he was working on this race car prototype for the first two years of his college life. He then participated in the FSAE competitions held in Germany and Czech Republic, where he was exposed to different technical and cultural experiences. This experience determined him to prepare for and pursue his further education in an engineering institution possessing a reputation of varied, contemporary research at an international level.

In Fall 2016, he joined the Department of Mechanical and Aerospace Engineering, at North Carolina State University, Raleigh, NC, to pursue his Master’s degree in Mechanical Engineering, with special interest in Thermal and Fluid Sciences. His dream for internal combustion engine research led him to Dr. Tarek Echekki, who has been his advisor since Spring 2017.
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CHAPTER 1

INTRODUCTION

The requirement for better fuel economy, lower emissions and greater engine efficiency, has led to the development of numerous innovations for the development of spark assisted (SI) and compression ignition (CI) engines. Recent advances in internal combustion engines have increased performance characteristics like volumetric, thermal and combustion efficiencies, harnessing technologies such as turbochargers, superchargers, ECUs, common rail injection, sensors, piezoelectric injectors, and many others. However, as we progress into the 21st century, the restrictions and challenges for vehicles are increasing. In the past two decades there has been rapid development in the field of advanced combustion engines, one of them being Spark Assisted Compression Ignition (SACI) engine. This type of combustion engine design has displayed greater production of in-cylinder pressure, temperature and heat release, compared to SI or CI engines, due to its integration of higher compression ratios along with spark ignition. Research over the years has made it possible to control gasoline combustion under increasingly adverse conditions created by high compression ratios. This has been used to develop homogenous charge compression ignition (HCCI) engines. This engine concept has been used to successfully increase thermal efficiency [1,3] and reduce harmful emission like NOx [4]. However, it was also found that the operating limits of HCCI are restricted to low to medium load conditions [5].

The SACI concept was first viewed as a bridging mechanism while transitioning from low load HCCI combustion modes to high load SI mode in contemporary research [14-23] to propose an engine concept suitable for different operating conditions. It was then developed as a stand-alone engine concept and research was conducted to expand its operating limits from medium to high load conditions. Utilizing exhaust gas recirculation techniques to dilute the fuel-air mixture
in the cylinder, stable combustion along with reduced emissions can be obtained. This type of engine was first proposed and patented by Tanahashi et al. [6] and was later improved by Yang [7]. The controlled combustion phasing combined with stable operation at medium to high load capacity make SACI combustion concept a versatile method of power production, with higher efficiency and reduced emissions. The SACI combustion mode is compatible with alternative fuels such as ethanol [35], indolene and their combinations with conventional fuel sources.

Mazda is the first automotive manufacturer to take a step in this direction. The latest engine designed by the company, called SKYACTIV-X, uses a combustion concept similar to the one discussed in this study, called Spark Controlled Compression Ignition (SPCCI). It combines the principles of SI and CI combustion, using a high compression ratio to pressurize the fuel-air mixture and then using spark ignition to control the combustion phasing, in order to stabilize the autoignition process. This concept is very similar to that of SACI.

1.1 Spark Assisted Compression Ignition (SACI) combustion

The SACI combustion concept was developed as an improvement of the HCCI combustion mode. It can be considered as a combination of the working principles of SI and CI engines, since it employs a high compression ratio to pressurize the combustion charge while providing an energy input via spark ignition to initiate a flame propagation phase, which stabilizes the autoignition of the mixture. This allows more control on the heat released by combustion, compared to combustion dependent solely on spontaneous autoignition of the air-fuel mixture within the chamber. The spark timing can be varied to exercise control on flame propagation and autoignition phases of combustion. This way heat released due to flame propagation and autoignition can be monitored by varying the timing of spark ignition. Fuel injection methods differ with operating conditions of
an engine. Port fuel injection allows greater homogeneity in the charge mixture, which can result in stable combustion, by controlling the equivalence ratio of fuel. Direct injection may also be used, for a more stratified fuel mixture, to promote combustion. To tackle the issue of instability due to excessive input of heat energy at high pressure, charge dilution is generally preferred to control the heat release from combustion reactions. This is achieved using either internal or external exhaust gas recirculation (EGR) or a combination of both. Intake air temperature can be controlled to induct preheated air into the combustion chamber, to promote the combustion (autoignition) of the mixture.

The combustion process occurs in four stages: 1.) spark discharge; 2.) early kernel growth (EKG); 3.) flame propagation; 4.) autoignition. A characteristic SACI combustion will exhibit these stages, which can be analyzed through different methods like in-cylinder pressure plots, heat release rate diagram and flame area growth. The different stages release heat energy, which can be controlled with engine parameter variation. As the heat released during flame propagation is higher, it increases the combustion chamber temperature. This leads to formation of NOx, which can be controlled with charge dilution to prevent excessive heat release and control combustion phasing. SACI allows greater distribution of heat release and lower overall peak heat release rate, which results in a lower peak pressure rise rate (PPRR). The lower PPRR and the resulting lower ringing intensity permits the addition of more fuel resulting in a higher load before ringing limits become constraining.
1.2 HCCI Engines

A major concern is the emission of greenhouse gases like carbon monoxide, carbon dioxide, nitrogen oxides and particulate matter. Also, ensuring complete combustion and extracting maximum power from the fuel charge is one of the primary motives of engine manufacturers. In the past decades, various technologies have been researched to improve the combustion efficiency of engines. Homogenous charge compression ignition (HCCI) is one of them, and currently extensive research is being done to commercialize this technology. The major reason this technology is attractive is its ability to provide a higher thermal efficiency compared to SI engines. These improvements are made possible partly through unthrottled operation at low load, which minimizes pumping loss. Thermal efficiency is also enhanced using a higher compression ratio to promote auto-ignition. Combustion occurs at nearly constant volume (due to the fast burn rate of the mixture), with greater expansion stroke work for a fraction of the fuel charge, provided by a low-temperature lean mixture [1]. In HCCI engines, combustion phasing is thermally controlled with intake temperature heating [2] or variable valve actuation to retain hot residual gases in the combustion chamber [3]. The peak combustion temperatures are relatively low due to high charge dilution with air or residual gas. These lower temperatures elevate the ratio of specific heat of the mixture, which results in greater thermal efficiency. These low combustion temperatures greatly reduce the NOx emissions from the engine, allowing it to meet emissions regulations without NOx after-treatment [4]. It is possible to combust lean air-fuel mixtures by auto-ignition, which is not possible in traditional SI engines. However, HCCI engines are not capable of operating at high load conditions when naturally aspirated, achieving a maximum of about 4.0-5.0 bar IMEP [5]. Beyond this load condition, the operation becomes unstable due to excessive pressure rise rates, leading to knocking. To better harness the benefits of this type of engine type, it is required to
control the combustion stability and ignition of the fuel, in order to control the pressure-rise rate and temperature of combustion.

1.3 Literature Review

An improvement on the HCCI technology, a method of spark assisted compression ignition (SACI) was proposed, where a spark is used to control the combustion. A patent by Tanahashi et al. [6], proposes a spark-assist type self-ignition engine. An engine with a spark plug and an injector in a combustion chamber, along with a sensor, which senses the gas temperature if it is at a temperature close to auto-ignition. Depending on this a valve timing device controls the opening time of the intake valve and maintains the gas temperature close to auto-ignition temperature, which once achieved triggers the ignition of the spark plug. The objective of this combustion system is to control the autoignition of the fuel to prevent knocking and prevention of NOx formation. A subsequent patent proposed by Yang [7], identified disadvantages with the previous proposal. At low engine load, SACI combustion generates higher levels of nitrous oxides and results in lower fuel economy than what can be obtained from HCCI combustion. Also, at high loads, without a degree of spark assist, the compression ignition combustion becomes ineffective due to loss of accurate temperature control in the combustion chamber. Thus, an engine with dual modes was proposed, wherein the first mode involves autoigniting the fuel-air mixture, and the second mode involves assisting the combustion of the mixture with a spark. This was done to add flexibility to the operating conditions of the engine at different load levels.

Four stages of SACI combustion were identified by Reuss et al. [8], namely, spark discharge, early kernel growth, flame propagation and compression ignition. It was found that variations in combustion phasing in SACI mode was dominated by the variations in the early
kernel growth (EKG) stage. However, due to limitations in diagnostic methods, the effect of the spark plasma in this stage could not be measured. This limitation was overcome in the study conducted by Natarajan and Reuss, [9], and the first conclusion of the previous study was confirmed. However, it was found that the spark ignition system did not affect the EKG cyclic variations, but the charge composition distribution did. Pastor et al. [10], combined direct visual diagnostic methods and spectroscopic analysis of natural radiation, with analysis of Rate of Heat Release, to validate the works of Reuss [8,9]. The spectral analysis of the combustion reaction radicals was used to study the progress of the combustion process and identify the transition of the SACI stages. A similar research was conducted by Benajes et al. [11], for a gasoline partially premixed spark assisted compression ignition engine at low load to better observe the combustion process. It was found that, apart from spark assistance and ignition timing, the fuel injection timing and duration had an important role in improving combustion stability, cyclic stability and combustion phase duration. The author continued this work in [12] to show the effect on emissions due to single and double direct fuel injection strategies, and by varying the fuel fraction in the double injection case. It was concluded that air/fuel mixture distribution was improved using double injection strategy and increased the fuel energy conversion efficiency.

In order to help understand the spark assisted process, a multi-mode combustion diagram was proposed by Lavoie et al., [13] to delineate the regimes of ignition, flame propagation, quenching and knocking, in HCCI, SACI and SI combustion in terms of unburned and burned gas temperatures near top dead center. It relates constraints due to gas properties, as a function of unburned gas properties, NOx limits and flame propagation. An analysis of existing experimental data was used to suggest that the effectiveness of spark assisted compression ignition is best at higher and middle loads and decreases as load is reduced, due to low flame speeds which are
unable to sustain combustion in this mode. This diagram is useful to identify limiting reduced equivalence ratios for each mode of combustion. The equivalence ratios were calculated as a function of equivalence ratio of the combustion charge as well as the residual gas fraction (RGF). Analysis of available experimental SACI data in terms of the MMCD, indicates that spark assist is most useful for ignition and burn rate control under moderate to high loads where flame speed is adequate.

Initially, sparking was used to make a transition from SI combustion to HCCI mode and to increase the load capacity of the HCCI engine. Persson et al. [14], conducted experiments in this effort. In this experiment, he used a modified gasoline engine, with a compression ratio for SI combustion. Therefore, in this research work, different air intake temperatures, in conjunction with various speed and load combinations, were used to study the response of the engine. It was noticed that changing the intake temperature caused the combustion timing to vary accordingly. A very low intake temperature leads to high emissions. Also, at high load conditions, which require richer mixtures, dilution by EGR can be reduced and can be substituted by intake air heating instead, to stabilize the combustion thereby minimizing misfires. Another research published by the same author, [15], investigated the variations and correlations in the parametric outputs, in consecutive cycles in each of the cylinders of a six-cylinder engine. The correlation coefficient used previously by Christensen [16], was utilized to detect the periodic behavior on HCCI combustion. There was not much correlation found between the cylinder operation, however, different engine load levels and speeds affected the cyclic behavior of the cylinders. Spark assistance was found to increase the operating regime as well as the efficiency of the HCCI combustion mode.

This technology was utilized by Wang et al. [17], to propose a method for transition between SI and HCCI combustion modes in a twin cylinder gasoline direct injection engine. It was
observed that combining two stage direct fuel injection and spark ignition enhanced control on
ignition and combustion rate of HCCI mode while extending its operating range. At HCCI critical
status, without sparking, the combustion becomes highly unstable due to knocking and misfire,
causing fluctuations in engine speed, which leads to retardation in ignition timing, causing
repetitive instabilities. This however was mitigated with spark ignition. There was a reported
increase of 39% in IMEP, and reduction in emissions, notably 74% in NOx emissions. The spark
ignition enhances the autoignition process in the combustion chamber, leading to a more uniform
and rapid heat release rate, although, visualizations from the optical engine showed minimal flame
propagation. This points to the fact that sparking plays an assistive role rather than a primary
combustion role. Hyvonen et al. [18], conducted an experiment on a Saab variable compression
ratio engine, with 5 cylinders. In this experiment, spark assisted HCCI was considered a particular
case and another case was considered for a mode change from HCCI to SI, with sparking for
combustion with flame propagation. It was found that with spark assistance the compression ratio
and inlet air temperatures required for combustion were lower compared to HCCI combustion, for
the same operating points. It should also be noted, that with the introduction of spark ignition,
cyclic variations resulting in combustion fluctuations, typical to SI engines, occur in dependence
with A/F variations.

To better understand the transition region between the SI and HCCI region, that is the SACI
region, Wagner et al. [19], created a series of probability maps to predict the combustion variations
by heat released at various levels of internal EGR. It was identified that cyclic variations occur in
combustion events, which are essentially a feedback effect of the recirculated exhaust gas. As a
part of this research, maps were created for each transition zone from conventional SI to HCCI
combustion as functions of heat release, which were then used to create a plot showing the patterns
of combustion events occurring in the transition modes at different EGR levels. This pattern was then compared to the predicted model of the combustion events and matched closely to the non-linear prediction model. The model was created utilizing works done by Daw et al. [20-22], on observing non-linear dynamics in a combustion engine. As a continuation of this work, Daw and Wagner [23], conducted another investigation into the reaction kinetics of an engine in the SI-HCCI transition combustion region. Through the analysis of data acquired from an engine operating in these combustion modes, the researchers have created a predictive model for the cyclic variations of different parameters. This shows that the kinetic parameters can be obtained from successive cyclic combustion measurements.

The extensive work done by Wagner, Daw, Edwards and others, [19,24-28], on the investigation of cyclic variability during transition from SI to HCCI combustion modes, was further investigated by Havstad et al. [29]. A multi-zone combustion model was proposed to compute combustion events in an economic computational time, as a compromise between high fidelity engine modeling code, KIVA [30] and existing single zone models [31]. Although the model presented was unable to analyze the complex reaction dynamics which result in the instabilities in the combustion mode transitions, it could however predict patterns of instability during the transition process. The author mentions improvements which could be implemented to improve the fidelity of the model, to better predict information for stabilizing the spark assisted HCCI combustion mode.

Persson et al. [32], conducted an experiment focusing on the effect of spark ignition, as well as certain specific parameters, on combustion initiation and progress. The changes on this phenomenon were studied by varying negative valve overlap, spark timing and load. A blend of 40% ethanol and 60% of n-heptane is used as fuel, due to the low temperature reactions occurring
between isoctane and n-heptane. SACI has stable operation at high dilution percentage levels due to the higher temperature of the residual mixture, which balances the heat losses from the spark kernel, and makes combustion in the very lean mixture possible. Also, varying the spark timing affects the onset of autoignition, advancing the sparking, advances the autoignition. Spark timing also affects the initial flame propagation speed near the spark plug, which can advance or retard the resulting autoignition of the remaining fuel mixture. Following this, Persson et al., investigated the effect of induced swirling on SACI combustion [33]. The swirling of the intake charge was induced by deactivating one of the intake valves on the engine in each scenario. This resulted in an increase in turbulence in the combustion chamber, thereby accelerating combustion timing. Using high percentage of EGR dilution in this scenario, however, results in delay in combustion timing. This can be used for extending the HCCI operating region to lower loads and speeds.

A research conducted by Urushihara et al., [34], utilized the concept of spark assist in HCCI combustion to extend its operating range. It was conducted experimentally in a single cylinder engine with multipoint fuel injection. The data was obtained at 1200 and 2400 rpm, with a fraction of the fuel injected directly, to obtain a stratified mixture near the plug, while the rest of it was injected through port injection, to develop a homogeneous mixture in the combustion chamber. It was observed that the SI-CI combustion has higher load capacity compared to HCCI mode, without intake air heating. However, at low loads, as the intake air temperature rises, through spark and injection timing retardation, stable combustion can be maintained. It was also found that introducing internal EGR by Negative Valve Overlap(NVO), can lead to expansion of operating range of the engine, as well as improve thermal efficiency. The earlier the NVO occurs, the more stable the combustion is at lower loads. Persson et al., [35], carried out a similar research, using ethanol as the fuel and PLIF imaging. In this experiment, two PFI ports were used in each intake
valve, along with a DI in the cylinder, and cases were considered where one port was closed, to induce turbulence in the combustion chamber [33]. Using only PFI, moderate stratification was obtained, which increased as the percentage of fuel injected through DI was increased. However, increased DI percentage lead to slight delay in combustion along with slightly reduced heat release rates.

As per a research carried out by Laura et al., [36], the goal was to extend the high load limit of an HCCI engine using SACI technology, while maintaining a stoichiometric equivalence ratio. In-cylinder pressure rise rate and combustion stability were controlled using cooled external EGR, spark assist and negative valve overlap. Using this strategy, a maximum engine load of around 7.5 bar NMEP was achieved while maintaining efficiency and complying with emission regulations. This study showed that stable dilute, and efficient combustion could be achieved with SACI at loads above the allowable limits for naturally-aspirated HCCI. Spark advance affects the combustion time and heat release rates, increasing the efficiency and expansion work as the combustion becomes more constant volume for a given fueling rate. Laura et al., [37], conducted another research to show that burn duration and combustion phasing for SACI mode can be controlled by varying spark timing and unburned temperature, while holding composition of the cylinder charge constant.

Yun et al., [38], conducted a comprehensive investigation to extend the high load-operating regime of HCCI combustion through spark ignition, to increase fuel economy and avoid knocking and transitions in combustion modes. The method of negative valve overlap (NVO) was used to keep the combustion stable at low load conditions [34,39-42]. It was observed that injection timing, spark timing and higher EGR dilution were essential for efficient spark assisted HCCI combustion at high load operation, achieving a high operating load limit of 10 bar gross IMEP. The authors
found that with increase in engine load, NVO had to be decreased, which led to rise in ringing. This was tackled by retarding spark timing, however, an increase in pumping losses was observed. The authors conducted another study, [43], using positive valve overlap (PVO). Through this method, ringing was relatively lower at high loads, due to lesser hot residuals in the cylinder and more EGR dilution than NVO, and higher efficiency was achieved due to reduction in pumping losses. The study by Xie et al., [44], is focused on the SACI combustion using positive valve overlap (PVO) strategy to optimize the gasoline engine performance at medium–high load. Based on the demand of iEGR and eEGR, an optimized PVO strategy is proposed, in which the PVO is formed mainly by advancing intake valve timing and subordinately by retarding exhaust valve timing. A stable SACI combustion is achieved in the range of 5–9 bar, with significant improvements in fuel economy, pumping loss and NOx emission.

The study by J.B. Martz et al., [45], investigates the correlation of premixed isoctane-air laminar reaction front data under the highly dilute, high preheat temperatures of SACI combustion. HCT, a premixed transient laminar flame simulation, was used to produce reaction front data under conditions pertinent to both SI and SACI combustion modes, extending on the previous works of Metghalchi and Keck [46], and Müller et al., [47], which were based on conditions relevant to SI combustion. The investigation focused on variation of equivalence ratio, unburned temperature and pressure. This analysis suggested that SACI combustion is most useful at medium and high engine loads. Data from the steady reaction front and constant pressure adiabatic flame temperature calculations were used to produce correlations of laminar burning velocity, thickness and adiabatic flame temperature. Another study conducted by Dahms et al. [48], focuses on similar objectives but combines different simulation methods to demonstrate a mixed mode combustion model which can compute and visualize the SA-HCCI combustion mode.
A DNS study was conducted by Yoo et al., [49], to demonstrate the combustion characteristics of iso-octane/air mixture under HCCI and SACI combustion, under various extents of temperature and turbulence fluctuations, and spark ignition timings. A 99-species reduced mechanism for the iso-octane/air chemical kinetics was developed and validated against detailed and skeletal models, using methods established by Mehl et.al, [50], Lu et al., [51-53]. The effects of parameters such as thermal stratification of mixture, spark timing variation and turbulence intensity were simulated. A two-dimensional (temporal and spatial) DNS solver [54] utilizing detailed chemical kinetics mechanism of hydrogen fuel was also developed and studied. Bhagatwala et al., [55], developed a two and three-dimensional DNS model for investigating HCCI and SACI combustion. This model used compression heating through mass source/sink terms to simulate the piston motion. The combustion characteristics were studied, with focus on simulating the flame motion and combustion phasing process.

Ziegler et al., [56], conducted an experimental research on an optical engine, using multi-axis imaging to better understand the combustion process in a SACI engine. Although, not many parametric effects were considered in this paper, it was important in exhibiting the improvement in visualization of the combustion phenomenon. The detailed images show the flame development and propagation over the cycle and are helpful to realise the spark assist results in enhancement of the autoignition of the fuel and heat release, due to an initial flame propagation, while accelerating ignition time. Ziegler et al., [57], found the effects of intake air temperature variations on spark-assisted HCCI combustion in a single cylinder research engine. It was found that the equivalence ratio affected the performance of the combustion mode, however, thermal stratification is a crucial factor in this mode. It was also concluded through imaging techniques that heat release rate data are not good indicators of presence or extent of reaction fronts in a combustion chamber.
Liu et al., [58], proposed a ‘controlled Assisted Spark Stratified Compression ignition(AASSCI) with moderate autoignition’ combustion mode, to minimize knocking while maintaining high thermal efficiency. After running different test conditions, the conditions best identified for the setup was dual stage injection combined with 20% cooled external EGR. Using a fixed high compression ratio and stoichiometric fuel conditions in a GDI engine, it was found that combustion involved a two-stage heat release process. Also, with minimum spark advance (knock-limit), using a diluted stoichiometric stratified fuel-air mixture can minimize knocking.

1.4 One Dimensional Turbulence (ODT)

Turbulent fluid flow is characterized by randomness and unsteadiness coupled with rapid fluid mixing and simultaneous reactions. Such flows possess varied ranges of length and time scales. Contemporary turbulence models separate large scale turbulent motion and small scale molecular transport and chemical kinetics in length and time domains. It was found by Peters [60], that for an extensive range of turbulent combustion cases, scale separation is not a valid aspect for their modelling.

Kerstein developed the Linear Eddy Model (LEM) [61], a turbulence model to effectively predict the mixing-reaction couplings at all scales. It spatially and temporally resolves, in a one-dimensional domain, the processes of turbulent advection, scalar and momentum transport and chemistry. The model simultaneously implements deterministic processes for the diffusive and reactive terms of the governing equations and a stochastic method for the turbulent advection term. This stochastic method applies ‘triplet map’ stirring events, which emulate the compressive strain and rotational folding characteristics of turbulent eddies and arbitrarily select eddies of varying lengths in the 1-D domain [62].
The One-Dimensional Turbulence (ODT) model is an extension of the LEM model [63]. This model categorizes turbulent advection and molecular transport processes as distinct events on a one-dimensional domain. The solution of the velocity vector in the ODT model, provides the necessary details about the shear field, hence a way to drive the turbulence. Thus, it can be concluded that this model is a self-sufficient turbulence model and can be used independently for simple flow problems. This model has two main advantages. Firstly, this model can realistically describe the physics while maintaining fine resolution like DNS. Secondly, this model provides computational economy. The ODT model has been applied to various turbulence problems due to its ability to resolve all temporal and length scales on a one-dimensional domain. Various ODT model formulations have been used as stand-alone models [64,65,66] or as a sub-grid closure model in hybrid multidimensional formulations like ODT/RANS [67] and ODT/LES [68,69].

As the ODT model is an independent turbulence model, it has been used to study the in-cylinder turbulent interactions in a Spark Assisted Compression Ignition (SACI) engine, using iso-octane as the fuel. As the chemical reaction mechanism for iso-octane and air is complex, a primary reference fuel with reduced chemistry is utilized, since it offers computational economy while providing sufficient insight into the combustion chemistry characteristics under turbulent conditions.
1.5 Objective

This study conducts high resolution one-dimensional turbulence simulations of spark assisted compression ignition (SACI) combustion to investigate this concept, using iso-octane as a fuel. Numerical studies have been made using this ODT model, since it provides realistic information regarding chemical interactions in unsteady flow conditions, while providing the flexibility to study different engine parameters over a wide range. The aim is to investigate suitable ranges of operating conditions for characteristic SACI combustion. The engine conditions, like spark timing, equivalence ratio, compression ratio and fuel injection specifications, have been varied over a wide range, for this purpose. The study was conducted to investigate SACI behavior under two different fueling conditions: Port Fuel Injection (PFI) and a combination of PFI and direct injection (DI), termed as Multipoint Fuel Injection (MPFI), to observe the effect of fuel stratification on combustion.

1.6 Overview

This study first introduces the concept of SACI combustion, and reviews the previous works done on this concept as well as the ODT model. Then the model formulation is discussed, including domain specification, the governing equations used and their boundary conditions for this model, followed by a brief discussion of their numerical implementation in this model. After that the observations made for varying different parameters in PFI mode of fuel injection are discussed. This is followed by discussion of results of MPFI strategy of fuel injection. Finally, the major conclusions from this study are presented, along with possible areas which can be further explored in the future.
CHAPTER 2  
NUMERICAL SETUP

This section discusses the formulation of the model used in this study of turbulent combustion of iso-octane air mixture in a spark assisted compression ignition (SACI), single cylinder engine. The model formulation is based on previous ODT models designed by Gowda and Echekki [70], and Echekki et al. [71]. The major difference between the previous studies and the current one is that an additional source term is added in the energy equation, which emulates spark ignition, activating at the time specified for the various simulation cases, in order to implement spark assistance for the HCCI model developed by Echekki and Gowda.

To realistically model the numerous interactions occurring in the combustion chamber during the compression and power strokes in a SACI engine, the fluid is assumed to possess variable density incompressible flow. This means that the flow always has a sufficiently low Mach number to function in incompressible conditions, but the density is allowed to vary to simulate the compression and expansion of the gases in the chamber due to piston movement and heat release. This model does not consider axial variations in velocity, concentration and temperature as discussed in the next section. The governing equations of fluid mechanics are used to calculate thermo-physical properties like density and velocity, which are dependent on the changes in domain length, temperature, pressure and molar concentration changes.

The momentum balance equation is used to calculate the radial variations of fluid velocity, which is dependent on changes in temperature and value of moles due to the various reactions. The species conservation equation is used to calculate molar species concentration change. The energy conservation equation is also used in the model to compute the temperature variations in the system. While the continuity equation has not been included in the model, it has been used to
simplify the other governing equations. Additional equations, like the ideal gas equation of state and overall system mass conservation equation, are applied to compute the density and the in-cylinder pressure at every iteration.

This chapter discusses the domain selection, the assumptions made, the derivation of the governing reaction-diffusion equation, and the details of its implementation as a stand-alone ODT model for SACI mode combustion simulations.

2.1 Domain Specification

In this model the domain for the one-dimensional space is chosen along the lateral direction (x). This domain is assumed to be located on the piston head and is perpendicular to piston movement. The domain is divided into a number of differential elements, each of uniform length (dx), dependent on the domain size specified and the resolution required. These elements are modeled to possess uniform properties in the axial direction, while maintaining equal cross-sectional area. The area of cross section (Acs) and length of differential elements (dx) are kept constant for the duration of the simulation. To account for volumetric change of the combustion chamber in the axial direction, for the compression and expansion strokes, the variation is modeled by implementing and updating the stroke length, ℓ, with every iteration, which is defined in the axial direction as well. This method of domain specification of the combustion chamber, represented in the Fig. 2.1 simplifies the modeling of the in-cylinder fluid flow as well as the implementation of fuel injection and moving boundary, while enabling the realistic implementation of the same. Fuel injection is realized by updating the fuel composition of the differential element (dx) which is located at the position where fuel injection is specified. This method makes the differential elements of equal length as injector nozzle diameter, which is a
reasonable approximation, since the length scales of individual elements (dx) and actual nozzle diameters are similar in order of magnitude. The injection occurs over the duration specified, such that at the end of the injection event the fuel-air mass ratio in the cylinder is equal to the required value of equivalence ratio.

**Figure 2.1:** Schematic representation of the 1-D spatial domain specification along the cylinder bore.
2.2 Assumptions

- The total mass of the system remains constant through the compression and expansion strokes of the engine. It is assumed that IVC takes place before the compression stroke begins, i.e. start of simulation, while EVO occurs after the expansion stroke is complete, i.e. end of simulation. This prevents mass change in the combustion chamber during the simulation.
- The in-cylinder mixture contains no particulate matter (PM) and is composed of gases at all times.
- The in-cylinder gaseous mixture is an ideal gas. It is assumed that for the pressure and volume ranges encountered in an engine, the deviation from ideal gas behavior is negligible for the mixture.
- The cylinder has a uniform bore and a flat piston geometry.
- Variable-density incompressible flow is assumed, and the in-cylinder pressure is assumed to be uniform.
- The gas mixture is modeled as a Newtonian fluid.
- Heat flux due to Dufour effect is neglected.
- It is assumed that there is no heat transfer in the axial direction, in the combustion chamber. The cylinder head and piston surfaces are assumed to be adiabatic, while the wall is modeled as heat sink with a fixed temperature.
- Mixing is dominated by spray injection due to which reaction and diffusion components of the fluid flow are considered for the model.
2.3 Momentum Conservation

The momentum conservation equation is applied over the differential length (dx), to find
the velocity of the fluid flow.

\[
\frac{\partial u}{\partial t} = \frac{1}{\rho} \frac{\partial}{\partial x} \left( \mu \frac{\partial u}{\partial x} \right) + \Omega_u
\]

The above equation is used as the governing equation for momentum conservation for this model.

The boundary conditions: \( u(t, -\frac{L}{2}) = u_{left}(t) \) & \( u(t, \frac{L}{2}) = u_{right}(t) \), for all \( t > 0 \)

2.4 Species Mass Conservation

The species mass conservation equation used in the model is as shown below,

\[
\frac{\partial Y_k}{\partial t} = -\frac{1}{\rho} \frac{\partial}{\partial x} (\rho Y_k V_k) + \frac{1}{\rho} \omega_k + \Omega_k
\]

\( Y_k \) is the species mass fraction, defined by \( \rho_k = \rho Y_k \).

The boundary conditions: \( \frac{\partial Y_k}{\partial x} |_{x=-L/2} = 0 \) & \( \frac{\partial Y_k}{\partial x} |_{x=L/2} = 0 \) for all \( t > 0 \)
2.5 Energy Conservation

The equation for conservation of energy is given below,

\[
\frac{\partial T}{\partial t} = \frac{1}{\rho C_p} \frac{\partial}{\partial x} \left( \lambda \frac{\partial T}{\partial x} \right) - \frac{1}{\rho C_p} \sum_{k=1}^{N} \frac{\partial}{\partial x} \left( \rho_k V_k C_{p,k} T \right) - \frac{1}{\rho C_p} \sum_{k=1}^{N} (h_k \dot{\omega}_k) - \frac{\overline{P_t}}{\rho C_p} \left[ \frac{dl}{dt} \right]_{bdry} + \Omega_T
\]

The walls operate under constant temperature conditions, as required for the simulation case. For constant temperature conditions (emulate external cooling), Dirichlet boundary conditions are applied which are defined as follows,

Dirichlet boundary conditions:

\[ T(t, -\frac{L}{2}) = T_{wall} \quad \& \quad T(t, \frac{L}{2}) = T_{wall}, \text{ for all } t > 0 \]

2.6 Constitutive Relations

2.6.1 Thermodynamic Pressure

The ideal gas law expresses the pressure of a gaseous mixture as a function of its density and temperature.

The volume of the combustion chamber changes with time due to piston movement. The ideal gas law, however needs to be satisfied at every instance of time. This is achieved by applying a volume mean. Another aspect is that the overall mixture density used in the relation is calculated by mass conservation in the total volume of the chamber, as it is assumed that the valves are closed during the simulation, therefore there is no mass exchange throughout the cycle. The density is calculated in the following manner,
\[ \rho = \frac{m_{total}}{V_{total}} \]

where \( m_{total} \) is calculated using initial values of density and volume, \( m_{total} = m_0 = \rho_0 V_0 \).

The in-cylinder thermodynamic pressure is formulated as follows,

\[ P_T = \left\langle \frac{\rho R_u T}{M} \right\rangle = \frac{\int A_{cs} \left( \frac{\rho R_u T}{M} \right) dx}{V_{total}} \]

### 2.6.2 Piston Speed

In this model, the origin is considered to be located at the cylinder head and the top dead center (TDC) as the reference point. If \( V_c \) is the clearance volume, \( A_{cs} \) the bore, \( l_{rod} \) the length of the connecting rod, \( 'a' \) the crank radius and \( l_{bdry} \) the instantaneous distance between the crank axis and piston axis, the instantaneous total volume of the combustion chamber can be written as,

\[ V_{total} = V_c + A_{cs} (l_{rod} + a - l_{bdry}) \]

The instantaneous distance between piston pin and crank axes, can be calculated by considering the system as a slider-crank mechanism [72].

\[ l_{bdry} = a \cos (\omega t) + \left( l_{rod}^2 - a^2 \sin(\omega t)^2 \right)^{1/2} \]

where \( \theta \) is the angular speed of the crank given by \( \omega = 2\pi N \) (\( N \)=Engine RPM in rev/sec).
Also, the following have been defined to establish engine geometry,

\[
H_{stroke} = 2a
\]
\[
\overline{S_p} = 2NH_{stroke}
\]
\[
R = \frac{l_{rod}}{a}
\]

By differentiating the instantaneous boundary length with respect to time, and applying the above relations, an expression for piston speed as a function of time is obtained.

\[
\left[\frac{dl}{dt}\right]_{bdry} = -\frac{\pi}{2} \overline{S_p} sin(\omega t) \left[ 1 + \frac{cos(\omega t)}{(R^2 - (sin(\omega t))^2)^{1/2}} \right]
\]

Here the negative sign on the RHS is because of the simulation beginning from bottom dead center (BDC), i.e. start of compression stroke.
2.7 Summary of the ODT Model Equations

The ODT model equations discussed above have been summarized in Fig. 2.2. To account for turbulent advection, a stochastic term (Ω) is added to each of the deterministic equations. The numerical implementation of this stochastic term has been discussed in Section 2.8.

**Momentum Conservation**

\[
\frac{\partial u}{\partial t} = \frac{1}{\rho} \frac{\partial}{\partial x} \left( \mu \frac{\partial u}{\partial x} \right) + \Omega_u
\]

Boundary conditions

\[u(t, -\frac{L}{2}) = u_{left}(t) \quad \& \quad u(t, \frac{L}{2}) = u_{right}(t), \quad \text{for all} \quad t > 0\]

**Species Conservation**

\[
\frac{\partial Y_k}{\partial t} = -\frac{1}{\rho} \frac{\partial}{\partial x} (\rho Y_k V_k) + \frac{1}{\rho} \dot{\omega}_k + \Omega_k
\]

Boundary conditions

\[\left[ \frac{\partial Y_k}{\partial x} \right]_{x=-L/2} = 0 \quad \& \quad \left[ \frac{\partial Y_k}{\partial x} \right]_{x=L/2} = 0 \quad \text{for all} \quad t > 0\]

**Energy**

\[
\frac{\partial T}{\partial t} = \frac{1}{\rho C_p} \frac{\partial}{\partial x} \left( \lambda \frac{\partial T}{\partial x} \right) - \frac{1}{\rho C_p} \sum_{k=1}^{N} \frac{\partial}{\partial x} (\rho_k V_k C_{p,k} T) - \frac{1}{\rho C_p} \sum_{k=1}^{N} (h_k \dot{\omega}_k) - \frac{P_T}{\rho C_p} \frac{[dl]}{[dt]}_{b dry} + \Omega_T
\]

Boundary conditions

\[T(t, -\frac{L}{2}) = T_{wall} \quad \& \quad T(t, \frac{L}{2}) = T_{wall} \quad \text{for all} \quad t > 0 \quad (\text{Isothermal wall})\]

**Additional Equations**

\[\left[ \frac{dl}{dt} \right]_{b dry} = -\frac{\pi}{2} S_p \sin(\omega t) \left[ 1 + \frac{\cos(\omega t)}{(R^2 - (\sin(\omega t))^2)^{1/2}} \right] \]

\[\rho = \frac{\rho_0 (V_c + A_{cs} H)}{\int A_{cs} \, dx} \]

**Figure 2.2:** Summary of equations used for the One-Dimensional SACI Engine model.
2.8 Numerical Implementation

The one-dimensional turbulence (ODT) model implements the unsteady governing equations, summarized in the previous Section 2.7, numerically. It does so by dividing the terms into two categories. The diffusive terms are computed deterministically using a central difference scheme to update the 1-D spatial profile. Temporal discretization of the governing equations is attained by splitting the diffusion and reaction terms. Diffusion progress is achieved using first-order forward Euler method, while the source term is calculated using DVODE, a stiff-integrator [73]. Mixture-averaged transport properties for heat and mass transfer are computed using transport libraries [74] that are provided within the CHEMKIN II suite [75].

Turbulent advection is implemented stochastically, achieved through the application of a ‘triplet map’ for each stirring event [62,71]. These maps replicate the compressive strain and rotational folding effects that are characteristic of eddy events. Eddy segments are randomly selected with a boundary located at \( \vec{x} \), and size \( \hat{l} \), so that an eddy of length span \( [\vec{x}, \vec{x} + \hat{l}] \) is selected for application of the triplet map in the one-dimensional scalar field. A conservative rearrangement is performed by replacing the selected profile of segment length, \( \hat{l} \), with three duplicate profiles compressed to a third of their original length. The middle copy is then inverted to maintain continuity in values obtained in the new profile. The derivatives at the interfaces of the three duplicate profiles, however, are not continuous.

The velocity component of the scalar field in the domain space is used to calculate the rate of shear, which in turn provides a method of modulating the location and frequency of the stirring events. The frequency is governed by an ‘eddy rate distribution’ which is calculated using a select time scale with every transport event occurring at a given instant. This method of selection of turbulent eddies in the ODT model and their implementation is discussed in detail in [63, 71, 76].
This model also has two parameters $A$ and $\beta$, which are of order unity [71], and have magnitudes similar to those used in an earlier work, which studied ODT simulation of auto-ignition in turbulent jets [77]. The parameter, $A$, is a function of eddy characteristic time and the inverse of the rate of shear applied at the selected eddy. The parameter, $\beta$, is a relation of eddy characteristic time and elapsed temporal evolution of the domain. The purpose of this parameter is to prevent the selection of triplet maps which have longer characteristic times than elapsed simulation time, thus allowing stirring events to have progressively growing eddy sizes.

This study is conducted using a primary reference fuel (PRF), with a 171-species skeletal kinetic mechanism, for Iso-octane ($C_8H_{18}$)-Air chemistry [78], based on the detailed mechanism developed by Lawrence Livermore National Laboratory (LLNL) [79,80], which consists of 874 species and 3796 elementary reactions. The skeletal mechanism consists of 171 species and 861 reactions. The model has the flexibility to vary different parameters, in order to study the effect of their variations, independently or in combinations, on the SACI combustion mode. Table 2.1 lists the common parameters considered in this study along with their values or range of values. The specific setup and run conditions for different problem sets have been discussed in the subsequent chapters along with the results of their variations. The simulation runs were performed on DELL workstations, utilizing a 64bit Windows operating system, equipped with Intel Core i7 2.4 GHz processor and 16GB RAM. The simulations are made over 360 crank angle degrees (CAD), from beginning of compression stroke to the end of expansion stroke, with a computation time of 16 to 18 hours, for each realization.
**Table 2.1:** List of parameters used in the study of SACI combustion and their values or ranges

<table>
<thead>
<tr>
<th>Fixed Parameters</th>
<th>Value/ Range</th>
<th>Units</th>
</tr>
</thead>
<tbody>
<tr>
<td>Stroke length (H)</td>
<td>8.2</td>
<td>cm</td>
</tr>
<tr>
<td>Cylinder Bore (L)</td>
<td>8.0</td>
<td>cm</td>
</tr>
<tr>
<td>Compression ratio</td>
<td>12.5-18</td>
<td>-</td>
</tr>
<tr>
<td>Rod fraction (R)</td>
<td>2.0</td>
<td>-</td>
</tr>
<tr>
<td>Engine speed</td>
<td>2000</td>
<td>Rev/min</td>
</tr>
<tr>
<td>Port Injection velocity</td>
<td>1000</td>
<td>cm/s</td>
</tr>
<tr>
<td>Direct Injection velocity</td>
<td>1000</td>
<td>cm/s</td>
</tr>
<tr>
<td>Wall temperature</td>
<td>380 (107)</td>
<td>K (°C), isothermal</td>
</tr>
<tr>
<td>Intake Pressure</td>
<td>1.0</td>
<td>bar</td>
</tr>
<tr>
<td>Intake charge temperature</td>
<td>500</td>
<td>K</td>
</tr>
<tr>
<td>Equivalence ratio</td>
<td>0.75-1.0</td>
<td>-</td>
</tr>
<tr>
<td>Spark timing (dBTDC)</td>
<td>50-20</td>
<td>-</td>
</tr>
<tr>
<td>Oxidizer temperature</td>
<td>500 (227)</td>
<td>K (°C)</td>
</tr>
<tr>
<td>Fuel Temperature (at injection)</td>
<td>350 (77)</td>
<td>K (°C)</td>
</tr>
</tbody>
</table>
CHAPTER 3
PORT FUEL INJECTION

3.1 Motivation

The aim is to study the behavior of SACI operation using isooctane as a fuel, without the utilization of EGR. Experimental investigations conducted previously, utilize engines modified to operate in SACI mode, with port fuel injection (PFI), direct injection or a multipoint fuel injection strategy with direct injection and port fuel injection. This chapter investigates the range of SACI combustion, the effect on combustion phasing and the characteristic results observed due to changes in equivalence ratio through port fuel injection, spark timing and compression ratio. The multipoint fuel injection strategy is studied in the following chapter. These two methods were chosen to observe the differences due to charge homogenization and fuel stratification.

3.2 Run Conditions

The simulations were done under two different modes of fuel injection. First, port fuel injection was investigated and then a combination of direct and port fuel injections. To compare these two modes of injection, other engine parameters such as the compression ratio, the cylinder dimensions, the intake velocity of fuel/air charge, and the engine speed were kept constant. To study the chemical behavior of the fuel, equivalence ratio, spark timing and compression ratio were varied. The combustion cylinder wall is an isothermal boundary with a fixed temperature during the combustion cycle. An elevated intake charge temperature is used to substitute the effect of exhaust gas recirculation and to ensure that the fuels are injected in gas phase. It is assumed that for a port fuel injection the fuel/air charge forms a homogenous mixture by the end of the intake stroke.
The simulation takes place from the beginning of the compression stroke (180 dBTD'C) and ends at the end of the expansion stroke (180 dATDC). The domain size (bore) is from -4 to +4 cm, i.e. 8 cm bore size. It is divided into 201 discretized grid points, giving a resolution of 0.4 mm. Compression and expansion of the domain length is performed by applying the function for piston speed. The change in length (stroke) is applied only to the work term for cylinder boundary in the governing equations. The other transport and source terms are executed with a constant domain with a resolution as specified above. The extent of change in the length is governed by the compression ratio and the engine speed. Temperature changes of the in-cylinder mixture occurs due to the compression and expansion of the domain.

To simulate PFI, some assumptions are made. The fuel/air charge is injected well before the intake valve closes (IVC), and compression begins, such that, a homogenous mixture is assumed to be formed at the beginning of compression stroke. The spark, modelled as a high energy input at the centre of the 1D domain for a fraction of the domain width (bore), is activated at various timings.
### Table 3.1: Fixed parameters of the engine for the different run conditions

<table>
<thead>
<tr>
<th>Fixed Parameters</th>
<th>Value</th>
<th>Units</th>
</tr>
</thead>
<tbody>
<tr>
<td>Stroke length</td>
<td>8.2</td>
<td>cm</td>
</tr>
<tr>
<td>Cylinder Bore</td>
<td>8.0</td>
<td>cm</td>
</tr>
<tr>
<td>Compression ratio</td>
<td>12.5</td>
<td>-</td>
</tr>
<tr>
<td>Rod fraction (R)</td>
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<td>-</td>
</tr>
<tr>
<td>Engine speed</td>
<td>2000</td>
<td>Rev/min</td>
</tr>
<tr>
<td>Injection velocity</td>
<td>1000</td>
<td>cm/s</td>
</tr>
<tr>
<td>Wall temperature</td>
<td>380</td>
<td>K, isothermal</td>
</tr>
<tr>
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<td>bar</td>
</tr>
<tr>
<td>Intake charge temperature</td>
<td>500</td>
<td>K</td>
</tr>
</tbody>
</table>

**Fuel and Oxidizer composition**

<table>
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</tr>
<tr>
<td>Oxidizer</td>
<td>O₂</td>
</tr>
<tr>
<td></td>
<td>N₂</td>
</tr>
</tbody>
</table>

Table 3.2 below summarizes the cases, which are considered for the PFI parametric study. The equivalence ratio is varied over four values, for different spark timings. Then, the corresponding cases are analysed to obtain suitable ranges of equivalence ratios and spark timings.

### Table 3.2: PFI run conditions

<table>
<thead>
<tr>
<th>PFI run conditions</th>
</tr>
</thead>
<tbody>
<tr>
<td>Equivalence ratio</td>
</tr>
<tr>
<td>Spark timing (dBTD)</td>
</tr>
</tbody>
</table>
3.3 Results and Discussions

3.3.1 Spark Timing Sweep

**Figure 3.1**: Plot showing the pressure curves for isoctane under SACI combustion at stoichiometric condition (FA=0.067) with different ignition timings.

Figure 3.1 shows the variation in the in-cylinder pressure due to changing ignition timing. The legends denote the different spark timings for which the curves have been plotted. Also shown with dashed lines is the pressure curve corresponding to motoring (i.e. no chemistry). A trend is observed as the spark timing is advanced. At 35dBTDC spark timing, the maximum pressure is around 46 bar, which is lower than the other curves. With spark advance, the maximum pressure rises and then falls, with the curve for 40dBTDC spark timing exhibiting the largest in-cylinder pressure of about 65 bar. The ignition of the charge in all cases is almost immediate, as seen in
the figure where the curves depart from the motoring curve at approximately the time of sparking. It is observed that the pressure curves rise gradually, almost smoothly, until there is a sharper jump after TDC. This can be attributed to the flame propagation phase of the SACI combustion mode. The spark ignition of the local charge gives rise to kernel formation. This leads to flame propagation, SI type combustion, which increases the temperature and pressure inside the cylinder. As a result, the unburnt mixture is elevated to a temperature sufficient for self-ignition of the mixture. This causes autoignition of the charge in the cylinder, represented by the rapid increase in the pressure, right after TDC.

Another observation that can be made is that spark advance causes the peaks to advance. An earlier spark timing results in earlier flame-front formation and propagation through the cylinder. The mixture starts burning earlier, causing an earlier pressure rise, and an earlier deviation from the motoring curve. This leads to a quicker rise in the in-cylinder pressure and temperature, thus achieving autoignition conditions earlier. This mechanism is consistent with the observed trends. This trend has been observed by Manofsky [36] as well.

Figure 3.2 shows the heat release rate plot for the stoichiometric charge, sparked at 40dBTDC. This figure exhibits the 4 stages of combustion characteristic to SACI combustion, first observed by Reuss [8]. They are: (1) spark discharge, (2) early kernel growth (EKG), (3) flame propagation and (4) autoignition. The vertical line represents the onset of autoignition just after TDC. This is a beneficial scenario, as the high pressure rise rate due to autoignition will enhance the expansion work done, which can be observed from the heat released during autoignition stage. It is also interesting to note that the pressure plot mirrors the heat release diagram, and the sharp pressure rise after TDC is in fact caused by autoignition. Figures 3.3 and 3.4 show the heat release evolutions for the spark timing cases of 45 and 35 dBTC.
Figure 3.2: Heat release rate diagram for spark time 40dBTDC at stoichiometric condition, to show the various stages of SACI combustion.

Figure 3.3: Heat release rate diagram for spark time 45dBTDC at stoichiometric condition.
The pressure and heat release plots for 35dBTDC, stoichiometric condition show a delayed autoignition event. Such retardation of combustion phasing with retardation in spark timings has been observed by Yun [38] as well. Till around 18dATDC, combustion is dominated by the flame propagation phase, after which autoignition begins. The pressure rise due to autoignition occurs later and is of a lower rate compared to earlier spark timings. It is possible that the equivalence ratio of 1 could be too high to support efficient SACI combustion. The heat release and pressure rise due to flame propagation, before TDC is reached, may not be sufficient to initiate autoignition. Since the flame-front is formed later, the fuel consumption begins later as well. At TDC, the amount of unburnt fuel remaining is more compared to in-cylinder charge that is ignited by sparking earlier. This excess charge requires a greater temperature to be heated to self-ignition limit and absorbs heat from the surroundings to reach that stage. This could explain the drop in heat release rate after TDC. Well after TDC, the jump in heat release rate represents autoignition.

**Figure 3.4:** Heat release rate diagram for spark time 35dBTDC at stoichiometric condition.
stage. However, as expansion stroke is underway, the dropping pressure due to increase in volume, does not allow a significant heat release, and the result is a series of autoignition events peaking at about the same heat release rate value as shown in Figure 3.4. Also, these peaks are not much higher, that is the maximum heat release rates during autoignition are similar to the flame propagation phase. This is a sign of low efficiency combustion cycle and possible instability inside the cylinder.

**Figure 3.5:** Plot of isoctane mole fraction vs domain (normalized), for stoichiometric condition, at TDC, for spark timings shown in legend. Autoignition point for 40dBTDC sparking case.

**Figure 3.6:** Plot of isoctane mole fraction vs domain (normalized), for stoichiometric condition, at 18 dATDC, for spark timings shown in legend. Autoignition point for 35dBTDC sparking case.
The above Figs. 3.5 & 3.6 represent the mole fraction of isooctane over the cylinder domain, which is normalized to represent the total domain size. It is observed that, at TDC, the in-cylinder fuel composition of isooctane is slightly lower for 40dBTDC case than 35dBTDC case. This is when the autoignition phase for the prior case begins. At 18dATDC, when the autoignition for the 35dBTDC case begins, unburnt fuel remains in the cylinder. However, for the 40dBTDC sparking case, the fuel has been completely consumed, and at this stage probably different species reactions are taking place in the process of releasing the final products of combustion. This effect on combustion phasing for SACI mode of combustion due to spark timing is an important observation. The spikes observed in the mass fraction plots are caused due to autoignition occurring at the location of the troughs. A majority of the fuel around the center is burnt due to flame propagation phase of the combustion, followed by autoignition spots in different locations of the cylinder. These phenomena are better observed through the 1-D domain temperature profiles, which are shown as follows.

The temperature plots shown below in Fig. 3.7, are taken from before sparking at 80dBTDC to just before TDC. The spark ignition takes place at 40dBTDC, which represents stage 1 in Fig. 3.2, followed by kernel growth around time 30dBTDC. Then flame propagation is observed till TDC.
Figure 3.7: Temperature plots showing flame propagation in chamber, for stoichiometric case at 40dBTDC sparking. Times for the instances are shown in the legends.

The plots shown below in Fig. 3.8 depict the autoignition phase of SACI combustion for the specified case. At TDC, there is a temperature spike observed on the left side of the chamber. This is the first region of autoignition flame development. As the cycle progresses, more autoignition regions appear in the unburnt mixture region, and these eventually merge with the flame front as it propagates and the volume expands with piston movement. Thus, autoignition is
assisted by flame propagation, to harness maximum energy of the fuel, without releasing excess energy, resulting in a stable combustion.

Figure 3.8: Temperature plots showing autoignition stage in chamber, for stoichiometric case at 40dBTDC sparking. Times for the instances are shown in the legends.

Figure 3.9 shows the temperature over the domain for the later ignition case of 35dBTDC.

As it is observed here, the late initiation of autoignition phase seen in Fig. 3.4, is reflected here for
the image at 18dATDC. Till then, flame front is propagating in the combustion chamber. This may point towards instability in combustion and lower thermal efficiency of the combustion cycle. As the autoignition starts late during the expansion stroke, a lot of the heat produced is more easily dissipated in the expanding volume.

Figure 3.9: Temperature plots for the case of spark timing 35dBTDC, stoichiometric mixture.

Shown below in Fig. 3.10 are the temperature plots over the cylinder domain from the spark discharge time, till autoignition phase ends. At 45dBTDC is the ignition, and at 38dBTDC is the early kernel growth stage. Till TDC flame propagation phase is dominant. At 3dATDC,
autoignition phase begins and a majority of the heat is released by about 18dATDC, similar to the case of 40dBTDC spark timing, show in Fig. 3.8. However, a fraction of the charge still remains unburnt at the same crank angle, towards the right of the chamber. This self-ignites later at around 35dATDC and completes by 40dATDC. This is reflected in Fig. 3.10, where heat release spikes are seen towards the latter part of the plot. Thus, the spatial temperature plots are useful in determining the location of observed discrepancies in the time dependent plots, showing that the model is able to predict the time and location of unstable behavior in the operation of an engine.

Figure 3.10: Temperature plots to demonstrate major combustion events for spark timing 45dBTDC, stoichiometric case.
The plots shown thus far reflect the range of spark timings which are effective in showing SACI like combustion phasing. The stability of flame propagation followed by significant energy production due to autoignition can be predicted by the model. Now, the early and late ignition timings are presented and briefly analyzed, to check their suitability for SACI combustion.

![Pressure curves for isoctane under SACI combustion at stoichiometric condition (FA=0.067) with early and late ignition timings.](image)

**Figure 3.11:** Pressure curves for isoctane under SACI combustion at stoichiometric condition (FA=0.067) with early and late ignition timings.

In Fig. 3.11, we observe the salient differences in the late and early combustion timings for an equivalence ratio of 1. The plot for 40dBTDC sparking case is also included to show the difference between these cases and a suitable case for SACI. For early spark timing of 50dBTDC, the maximum pressure reached is comparatively lower. Also, autoignition phase starts later than TDC. The autoignition phase shows an unstable rise in pressure, the rate reduces to a constant
slope, and a small peak is seen, followed by an immediate reduction. While this case exhibits SACI-like combustion, it is not an ideal case, since autoignition should exhibit a significant rise in pressure. In this case, it is likely that a lot of the chemical energy of the fuel is released in the flame propagation stage, hence autoignition contribution is significantly reduced. This is different from observations made by Wang, [17], where spark ignition plays an assistive role, to stabilize and enhance the autoignition process.

For late spark timings of 30 and 25 dBTD C, varied differences are observed. For the 30dBTDC case, autoignition begins before TDC, resulting in a significant pressure rise. Such combustion before TDC is not conducive to stable engine operation, creating opposing pressure forces in the combustion chamber due to piston movement in the opposing direction. This could lead to knocking, as seen in the curve where there is an uncharacteristic curve in the pressure trace for the 30dBTDC case just around TDC. As the volume expansion begins, there is autoignition which takes place under more permissible conditions, which is seen at the peak of the curve. For the 25dBTDC spark ignition case, the in-cylinder pressure reached is much lower, with minimal autoignition effects. Also, the autoignition phase is reached much later, near 10dATDC. Here, flame propagation phase is seen to extend for a longer duration, failing to elevate the conditions in the cylinder to support self-ignition at TDC. This could be because of the late ignition of the mixture, which cannot result in characteristic autoignition of the unburnt fuel charge. Since the flame front starts propagating late, it cannot raise the pressure and temperature conditions in the cylinder enough to create sites of autoignition. A look at the heat release curves below give a better insight into the combustion process.
Figure 3.12: Heat release rate diagram for spark time 50dBTD at stoichiometric condition.
Figure 3.13: Heat release rate diagram for spark time 30dBTDC at stoichiometric condition

Figure 3.14: Heat release rate diagram for spark time 25dBTDC at stoichiometric condition
The above figures reflect the pressure variations observed for each of the corresponding cases. As observed in the pressure trace of the 50dBTDC sparking case, the autoignition starts later than TDC, represented by the heat release peaks around 7dATDC. Secondly, the heat released is not very high, compared to other cases, especially during autoignition. This could be attributed to the extended flame propagation phase, which begins around 40dBTDC, earlier than the other cases. This extended burning due to flame front propagation could result in a large part of the fuel being consumed before TDC is reached. This phenomenon was also observed and explained by Yun [38]. Also, the conditions favorable for autoignition of the limited lean fuel mixture remaining in the cylinder are reached after TDC, resulting in a late autoignition. In the case of 30dBTDC sparking, the pressure in the cylinder during the combustion cycle is very high, so is the heat releases as seen in Fig. 3.13. The autoignition phase begins very early, around 8 degrees before TDC. This early autoignition during compression stroke, causes a significant increment in heat release rate, which is reflected in the pressure curve as well. Although this high expulsion of heat energy may lead to higher pressures, this case however is not a very favorable one. The multiple spikes, which occur after autoignition begins, signifies instability in the ignition process. The sharp spike of heat release following this section signifies a very high energy release in short time-span, which is possible grounds for knocking. A reason for such an immense energy release for this spark timing could be that the kernel formation occurs at elevated temperature and pressure conditions, which might accelerate the reaction process during the flame propagation stage, causing autoignition to occur earlier.

This contrasts with the 35dBTDC case, where a later ignition timing leads to a much later autoignition timing. The 25dBTDC sparking case leads to lower overall pressure and heat release rate. This could be attributed to the late spark ignition timing. As seen in Fig. 3.14, the heat release
rate is very high during the early kernel growth stage, shown in Fig. 3.2. This is because the area around the spark plug is at a higher temperature at 25dBTC. Therefore, the local mixture combusts more quickly, releasing a greater amount of heat energy compared to other cases. As the flame propagates, there is not enough time to create self-ignition conditions in the chamber, leading to a later autoignition. While this is observed at 35dBTC as well, the reason for that case could be that the fuel remaining in the chamber before autoignition was at the extremities and took longer to burn due to their lower volume, as it required more time to achieve self-ignition conditions. This is in contrast to the current case where, the amount of fuel needed to be burnt by flame propagation to elevate the conditions in the chamber to support autoignition, requires more time, pushing the autoignition beyond TDC.

The temperature plots for the 30 and 50 degrees BTDC cases are shown below, during autoignition, in order to get a better understanding of the late and early autoignition respectively. For the early spark timing case, the flame propagation phase is dominant till 6dATDC, as shown in Fig. 3.15, when autoignition begins. Till that point, flame propagation has consumed a lot of the fuel charge, as can be observed by the width of the temperature peak over the domain. Secondly, the piston has already commenced its downward motion for the expansion stroke. Hence, the autoignition phase does not increment the in-cylinder pressure significantly.
Figure 3.15: Temperature plots for 50dBTDC sparking case (stoichiometric), showing autoignition phase

In Fig. 3.16, for the case of 30dBTDC spark timing the autoignition begins early at 8dBTDC. As can be seen from the figure, the flame front has not spread too far over the domain, compared to the prior case. As a result, there is a significant volume of fuel charge left unburnt, which is self-ignited. The early autoignition could be due to the formation of ignition hot spots in the cylinder volume, due to high heat input from late spark ignition, at an elevated pressure. As
observed, the autoignition sites form in the right side of the domain first and after exhausting the fuel in that region, autoignition on the left begins.

![Temperature plots for 30dBTDC sparking case (stoichiometric), showing autoignition phase](image)

**Figure 3.16**: Temperature plots for 30dBTDC sparking case (stoichiometric), showing autoignition phase

The temperature plots for 25 and 20dBTDC case are not shown as they were predominantly flame propagation with minimal autoignition phasing. This shows that very late spark timing is not favorable for SACI combustion mode. Although the model was designed for homogenous fuel
air mixture for port fuel injection, at the beginning of the compression stroke, it is observed that
spark timing plays an important role in determining combustion phasing in SACI combustion, in
accordance with investigation conducted by Manofsky [37].

Figure 3.17: Comparison of maximum pressures for different spark ignition timing cases
discussed above, at stoichiometric conditions. The cases in red are the unfavorable conditions for
the SACI combustion mode.

Figure 3.17 compares the maximum pressures attained for the different spark timings cases,
at stoichiometric conditions, is presented. The readings marked in red were the early and late spark
timing cases, which were observed to have uncharacteristic combustion compared to SACI mode
or were unstable. While there was complete combustion of fuel in all cases, combustion phasing
was a deciding factor in further analysis conducted in this report. The spark timings of 35dBTDC,
40dBTDC and 45dBTDC are chosen for further analyses, as this ignition timing range was
observed to show consistent characteristic SACI mode results.

A set of results were obtained by simulating an equivalence ratio sweep for the varied
spark timings, which have been analyzed till now. However, they followed a similar trend, with
35, 40 and 45 degree BTDC spark timing cases showing prominent SACI like results. Hence, the equivalence ratio sweep analysis discussed further, correspond to the aforementioned spark timing cases.

### 3.3.2 Equivalence Ratio Sweep

![Pressure traces for spark timing 45dBTDC cases at the different equivalence ratios.](image)

**Figure 3.18:** Pressure traces for spark timing 45dBTDC cases at the different equivalence ratios.

Figures 3.18 to 3.20 display the pressure traces for the 4 equivalence ratios chosen at the beginning of the chapter. ‘ϕ’ corresponds to equivalence ratio. In Fig. 3.18, a trend is observed as the equivalence ratio is decreased, the maximum pressure decreases. As the equivalence ratio decreases, the flame propagation phase emits approximately the same energy, however, the autoignition combustion pressure decreases significantly. This is because the lesser fuel volume is ignited early, causing a large part of it to burn before self-ignition conditions are attained in the
combustion chamber. Lower equivalence ratio results in lesser fuel mass in the combustion charge, for the same injection conditions for all cases. Therefore, after TDC, as self-ignition conditions are reached for the different cases shown above, there is lesser fuel available in the chamber for the successive leaner cases. As a result, the autoignition peaks reduce as the charge becomes leaner. A similar trend is observed in Figs. 3.19 and 3.20. The maximum pressure reduces with reduction in equivalence ratio. However, in Fig. 3.20, for the 35dBTC case, it is observed that the pressure rises and falls as the charge becomes leaner. This could be attributed to the later ignition timing, which seems to play an important role in combustion phasing as established earlier. As seen earlier, later ignition timing delayed the beginning of the autoignition phase, affecting the in-cylinder pressure rise. The flame propagation phase lasted longer, before autoignition conditions for the charge were attained, which could be due to the requirement of a leaner fuel mixture for complete self-ignition. This condition is satisfied in the leaner cases, where self-ignition conditions are attained earlier, as the fuel density is lower, and the thermodynamic properties in the cylinder are suitable for autoignition. For the leanest case ($\phi=0.75$), the fuel charge is too lean near TDC, when self-ignition conditions are attained, for significant autoignition heat release to occur. As a result, the autoignition phase of this case has marginal pressure rise.
**Figure 3.19:** Pressure traces for spark timing 40dBTDC cases at the different equivalence ratios.

**Figure 3.20:** Pressure traces for spark timing 35dBTDC cases at the different equivalence ratios.
Figure 3.21: Comparison of maximum in-cylinder pressure for different equivalence ratios at each case of spark timing

Figure 3.21 shows a comparison of the maximum pressure reached at different cases of equivalence ratio as spark ignition timing is increased. For the case of 45dBTDC, it is observed that the max. pressure decreases as a leaner fuel air mixture is used. The same is observed for 40dBTDC, but the change is relatively greater in magnitude. For the last case however, as a leaner mixture is used, the max. pressure first starts increasing then decreases. These variations have been discussed earlier to some degree. This bar chart also displays the trends for the spark timing cases at different equivalence ratios. These cases showed characteristic SACI like combustion, with flame propagation phase assisting the autoignition phase of combustion. The earlier spark timings for the richer cases exhibited HCCI like combustion characteristic with burning of the fuel charge for a short duration followed by spontaneous combustion of the unburnt mixture. For the later
spark ignition cases, an SI type combustion was observed, with a majority of the heat release due to flame propagation accompanied by weak autoignition events in the combustion chamber. The compression ratio of 12.5, used for these cases was not enough to support SACI mode of combustion for the leaner fuel air mixtures.

**Figure 3.22:** Plot of autoignition start timings vs. spark ignition timing for different equivalence ratio cases.

In Fig. 3.22, the autoignition initiation timings are shown for the above cases. The observation is that autoignition advances with spark advance. Also, the sensitivity of combustion phasing increases with equivalence ratio, indicated by the significant change in slope of the trendline as $\phi$ is increased. This shows that varying equivalence ratios may result in change in the thermodynamic characteristics, key to SACI combustion mode, but a good degree of control on combustion phasing can be obtained by maintaining spark ignition within a certain range.
3.3.3 Compression Ratio Sweep

Compression Ratio is an important parameter affecting combustion in a reciprocating engine. This section investigates the effect of varying compression ratio on SACI combustion mode. Here, the effect of four different compression ratios were investigated through numerical simulation, while keeping other engine parameters constant, as in the previous section (Tables 3.1 & 3.2). In the past, experimental investigations were conducted with the same objective, by Hyvonen [18], to study the effect on the mixed spark assisted HCCI combustion mode. The effects of compression ratio change are investigated in the article, to identify a suitable range of operating conditions, for controlling the mixed combustion mode. While there is limited literature studying this case, an effort was made in this direction, to understand the extent of changes in the combustion characteristics.

![Figure 3.23: Variation in max. pressure with compression ratio for the spark timing cases at phi=0.75](image)

**Figure 3.23:** Variation in max. pressure with compression ratio for the spark timing cases at phi=0.75
Figure 3.24: Variation in max. pressure with compression ratio for the spark timing cases at phi=0.82

Figure 3.25: Variation in max. pressure with compression ratio for the spark timing cases at phi=0.89
Figure 3.26: Variation in max. pressure with compression ratio for the spark timing cases at phi=1

The above figures represent the maximum pressures attained inside the cylinder. The sequential bars are in order of increasing compression ratio (C.R.), as shown in the legends. There is a general trend observed as compression ratio is increased. The maximum pressures increase as the C.R. is increased. This follows the trend observed in most engines. As the compression ratio is increased, the pressure at which the charge mixture is compressed increases as a function of the CR, and heat energy released is much higher, leading to a greater rise in maximum pressure. A trend can be observed as the spark timing is varied, for corresponding CR cases. The pressure attained for the 45dBTDC cases are higher as CR is increased, while that of the 35dBTDC case is lower. It was observed that for the earlier case of sparking, the autoignition event was dominant, resulting in higher pressures. As for the later spark timing case, flame propagation was observed to dominate the combustion process. This could be due to the burn duration of the charge, which
begins earlier, for earlier spark timing cases, and rapidly elevates the in-cylinder conditions to support self-ignition. This, however, is not possible for the later spark timings, as a result elongating the burn duration.

The figures shown below, compare the maximum in-cylinder pressures attained, with change in equivalence ratio, for fixed spark timings. While the effect of the CR sweep on combustion can be observed, through rising pressure, trends can be observed by varying equivalence ratios as well, at different compression ratios. There is a trend of gradual increase in pressure as equivalence ratio is increased, for each CR case. However, there are cases which show a different trend. Taking the case of spark timing 45dBTDC, in Fig. 3.27, the maximum pressure decreases for equivalence ratio of 0.82 and then rises, rather than showing gradual increase. This is because there is considerable autoignition for the case of equivalence ratio of 0.75, beginning very early in the cycle, generating a very high pressure. This becomes more stable for the richer case, showing more stable combustion phasing, resulting in a relatively lower maximum pressure. As the equivalence ratio is increased, the pressure increases. This pattern is observed for CR cases of 14 and 16.

For the spark timing case of 40dBTDC, such an anomaly is observed for an equivalence ratio of 0.89 for CR cases of 14 & 16. There is an observed proclivity for the combustion to shift toward SI dominated combustion. A longer flame propagation stage, burns the fuel charge for a longer duration, delaying and minimizing the autoignition heat release.
Figure 3.27: Variation in max. pressure with compression ratio for changing equivalence ratio cases at a spark timing of 45dBTDC

Figure 3.28: Variation in max. pressure with compression ratio for changing equivalence ratio cases at a spark timing of 40dBTDC
Following the maximum pressure charts, the pressure curves are shown. The following curves show the pressure traces for the different CR cases, for equivalence ratios of 0.75 and 1, and spark timing at 40dBTDC. Firstly, it is noticed that the pressure rise rates are larger as CR is increased, resulting in higher in-cylinder pressures. Secondly, as the CR is increased, the autoignition phase generates a larger pressure. This is possible due to the autoignition time advancing as the CR is increased. An explanation for this could be that the higher CR leads to greater pressure within the cylinder, which is observed in the figures below. Coupled with the high pressure rise rate due to the propagating flame-front, the thermodynamic state required for the autoignition of the unburnt mixture is reached earlier. This causes the unburnt mixture to self-

**Figure 3.29:** Variation in max. pressure with compression ratio for changing equivalence ratio cases at a spark timing of 35dBTDC
ignite earlier, resulting in a greater energy being released during the autoignition phase. As seen earlier, it is observed that as the equivalence ratio is increased, the peak pressure increases as well.

The pressure plots for the 45 and 35dBTDC cases are presented, in the subsequent figures, comparing the behavior under different compression ratios, at $\phi$ of 0.75 and 1. When the lean mixture ($\phi=0.75$) is ignited at 45dBTDC, there is considerable advancement observed in the autoignition phase. This however, can be controlled by increasing the equivalence ratio, as seen in the case of stoichiometric ratio ($\phi=1$). For the 35dBTDC case, it was observed that as CR was increased, the dominant effects of flame propagation, due to late sparking, were controlled and autoignition was more prominent, resulting in sharper pressure rise.

![Pressure plots comparing CR cases at $\phi$ 0.75 and 1 for sparking at 40dBTDC.](image)

**Figure 3.30:** Pressure plots comparing CR cases at $\phi$ 0.75 and 1 for sparking at 40dBTDC.
Figure 3.31: Pressure plots comparing CR cases at $\phi$ 0.75 and 1 for sparking at 45dBTDC.

Figure 3.32: Pressure plots comparing CR cases at $\phi$ 0.75 and 1 for sparking at 35dBTDC.
The following plots show the heat release rates for the corresponding cases shown above. For the 40dBTDC sparking cases, it is observed that as CR is increased, the autoignition stage advances. Thus, CR is seen to impact combustion phasing. For a compression ratio of 18, it is observed that there is a consistent advance of autoignition before TDC. A leaner fuel charge shows a relatively stable and longer flame propagation phase, followed by autoignition. As the fuel charge becomes richer, the proclivity towards autoignition increases, due to more fuel rich unburnt zones exposed to higher temperature and pressure.

For the case of sparking at 45dBTDC, it is seen that the autoignition occurs very early for the lean mixture. However, as the CR is increased, the flame propagation phase is observed to last for a longer duration before autoignition can begin. This control on combustion phasing prevents misfiring in the combustion chamber due to early autoignition. An advance in autoignition timing was observed in the cases of richer charge mixtures. These observations show that CR can control the combustion phasing of the mixture.

For the 35dBTDC sparking time case, it was observed that an increase in CR for the leaner mixture advanced the autoignition of the SACI combustion. However, for the richer mixture, a peculiar observation was made. The wider autoignition phase for a lower CR was controlled to a relatively narrower time duration, where a majority of the heat energy was released. These observations show that CR has a significant impact on the combustion phasing, controlling the flame front propagation, as well as the autoignition duration. Although there is an increase in peak pressure, it is observed that the plots show successive peaks as the richness of the charge mixture is increased. Also, increasing CR also shows similar effects on combustion. This could indicate that increasing compression ratio impacts the stability of the SACI combustion mode and has potential for higher ringing and possible knocking.
Figure 3.33: Heat release rate diagrams for sparking at 40dBTDC, and $\phi$ of 0.75 and 1, for the different CR cases.
Figure 3.34: Heat release rate diagrams for sparking at 45dBTDC, and $\phi$ of 0.75 and 1, for the different CR cases.
Figure 3.35: Heat release rate diagrams for sparking at 35dBTDC, and $\phi$ of 0.75 and 1, for the different CR cases.
Figure 3.36: Autoignition points for cases of sparking at 45dBTDC

Figure 3.37: Autoignition points for cases of sparking at 40dBTDC
Figures 3.36 to 3.38 represent the points at which the autoignition phase begins for each of the cases considered. The plots shown are at successive values of compression ratio for each value of equivalence ratio. The plots show that different values of equivalence ratio $\phi$, behave differently as CR is increased. However, for most cases, a general trend is observed which shows a tendency for combustion phasing to advance with increasing compression ratio. This is most noticeable in the case of sparking at 40dBTDC. As mentioned in earlier sections, as spark ignition timing is advanced, the autoignition occurs very early. Also, later ignition timings lead to longer durations of flame propagation. It is observed that changes in spark timing increases sensitivity of combustion phasing to compression ratio.

**Figure 3.38:** Autoignition points for cases of sparking at 35dBTDC
3.4 Conclusion

From the above studies, the following conclusions can be made:

1. It was observed that SACI combustion mode operates in 4 stages, i.e., spark discharge, early kernel growth, flame propagation and autoignition. Varying the operating parameters of the engine affect the combustion phasing for each stage.

2. Spark ignition timing plays a significant role in combustion phasing. Characteristic SACI combustion was observed to occur within the range of spark ignition timings of 35 to 45dBTDC. Consistent combustion phasing was observed near TDC with prominent flame propagation events followed by autoignition.

3. A very early spark ignition timing results in possible advance in the autoignition phase, resulting in uncontrolled autoignition and misfiring. Such events have proclivity to high ringing in the combustion chamber and potential for knocking.

4. Late spark ignition times cause delayed combustion phasing, by elongating the duration of flame propagation. This results in autoignition of the end gas later in the cycle, when the combustion volume is expanding due to piston movement, hampering the in-cylinder pressure rise.

5. Equivalence ratio is observed to impact maximum in-cylinder pressure and combustion phasing. For stoichiometric cases (ϕ=1), in-cylinder pressure reaches greater values compared to leaner mixtures. It is also noticed that as the equivalence ratio increases, the heat release during the autoignition event increases.

6. It is seen in this study that, while there is no optimum value of equivalence ratio suitable for every spark ignition time case, an engine can be operated at a certain equivalence ratio for certain spark timings. As the spark timing is delayed, for instance the case of 35dBTDC, leaner
charge mixture is more beneficial for SACI combustion. However, earlier ignition timing ranges exhibit better operation characteristics under richer conditions.

7. As compression ratio is increased, the maximum in-cylinder pressure attained increases. However, it was observed that the heat release became more unstable as the compression ratio was increased. A leaner mixture showed better combustion stability compared to richer mixtures.

8. Compression ratio also controls combustion phasing. As it is increased, the autoignition phase advances. It is seen that for a very high compression ratio (CR=18), the autoignition occurs relatively early, just before TDC. Also, it is noticeable that higher compression ratio restricts the autoignition phase to a limited duration. For the case of sparking at 35dBTDC, it was noticed that autoignition was occurring much before TDC, especially at leaner conditions. This was controlled by increasing compression ratio.
CHAPTER 4
MULTIPLE POINT FUEL INJECTION STRATEGY

4.1 Motivation

In this chapter, we study the behavior of SACI operation using iso-octane as a fuel, introduced in the combustion chamber utilizing a multi-point fuel injection strategy, a combination of direct injection and port fuel injection. This chapter investigates the effect of charge stratification and partial homogenous mixtures on SACI combustion. Observations pertaining to combustion phasing and thermodynamic results are discussed, as a result of direct injection at different times during the engine cycle and by creating variations in charge homogeneity, as percentage of total fuel injected by the two methods is varied. The trends which arise with increasing percentage of direct fuel injection, are examined and show that charge homogeneity is essential for stable engine operation. It is also observed that fuel stratification to a certain extent improves the phasing of combustion. The results of this multiple point fuel injection (MPFI) strategy are then compared with those of port fuel injection (PFI), studied in Chapter 2. While these results are not indicative of optimal engine operation under SACI mode, they investigate a general range of operation where characteristic SACI combustion can be sustained in an engine.

In the past, research has been conducted in this direction. Urushihara et al., [34], investigated the effect of fuel stratification near the spark plug, with a very lean mixture injected directly into the engine, along with port fuel injection. It was found that the lean pocket of stratified charge near the spark plug assisted in the formation of flame kernels, which resulted in stabilized flame propagation stages followed by autoignition. This was found to extend the operating load of the engine compared to running in HCCI mode. A study conducted by Persson et al., [35], investigates the effect of PFI combined with direct injection (DI), and ethanol was used as the fuel.
It was found that increased stratification led to longer durations of burning of the charge mixture. Also, greater stratification resulted in the occurrence of more autoignition sites in the combustion chamber.

This chapter first details the run conditions used in the study of this strategy. This is followed by discussion of the effect of changing injection timing and percentage of total fuel injected directly into the cylinder. Then a brief discussion compares the PFI strategy to MPFI strategy.

4.2 Run Conditions

The simulations utilize two different modes of fuel injection. To be able to compare the two modes of injection, other engine parameters such as compression ratio, cylinder dimensions, intake velocity of fuel/air charge, and engine speed were kept constant. Parameters such as spark timing, equivalence ratio and intake charge temperatures were also kept constant to study the effect of fuel injection strategy on SACI combustion. The combustion cylinder wall is an isothermal boundary with a fixed temperature during the combustion cycle. An elevated intake charge temperature is used to substitute the effect of exhaust gas recirculation. It is assumed that for a port fuel injection the fuel/air charge forms a homogenous mixture by the end of the intake stroke.

The simulation takes place from the beginning of the compression stroke (180 dBTDC) and ends at the end of the expansion stroke (180 dATDC). The domain size (bore) is from -4 to +4 cm, i.e. 8 cm bore size. It is divided into 201 discretized grid points, giving a resolution of 0.4 mm. Compression and expansion of the domain length is performed by applying the equation for piston speed (Eq. 2.24). The change in length (stroke) is applied only to the work term for cylinder boundary in the governing equations. The other transport and source terms are executed with a
constant domain with a resolution as specified above. The extent of change in the length is governed by the compression ratio and the engine speed. Temperature changes of the in-cylinder mixture occurs due to the compression and expansion of the domain.

To simulate PFI, some assumptions are made. The fuel/air charge is injected well before the intake valve closes (IVC), and compression begins, such that, a homogenous mixture is assumed to be formed at the beginning of compression stroke. The spark, modelled as a high energy input at the centre of the 1D domain for a fraction of the domain width (bore), is activated at various timings. Direct injection in this code is simulated by updating the mass composition of the domain elements, which lie within the width of injection specified, with the fuel composition specified. This occurs over a duration which is specified in terms of degrees of crank angles (CAD), such that at the end of that duration the amount of fuel in the chamber, due to both injection strategies, is equal to the equivalence ratio specified. The percentage of total fuel mass which is injected into the cylinder through PFI and DI is also specified. The compression ratio (CR) is kept constant at 16 for the purpose of this study, to provide sufficient thermodynamic conditions to support autoignition at all conditions, while ensuring long enough duration of flame propagation burning.

The tables below show the engine parameters selected for the MPFI cases, which remain constant for all cases. Also, the parameters which are varied for the cases, considered for the MPFI parametric study, are also tabulated in the following table. The injection timing is varied over a range of four values. Also, 5 different direct injection percentage values are also considered for each injection timing.
Table 4.3: Fixed parameters of the engine for the different run conditions.

<table>
<thead>
<tr>
<th>Fixed Parameters</th>
<th>Value</th>
<th>Units</th>
</tr>
</thead>
<tbody>
<tr>
<td>Stroke length</td>
<td>8.2</td>
<td>Cm</td>
</tr>
<tr>
<td>Cylinder Bore</td>
<td>8.0</td>
<td>Cm</td>
</tr>
<tr>
<td>Compression ratio</td>
<td>16</td>
<td>-</td>
</tr>
<tr>
<td>Rod fraction (R)</td>
<td>2.0</td>
<td>-</td>
</tr>
<tr>
<td>Engine speed</td>
<td>2000</td>
<td>Rev/min</td>
</tr>
<tr>
<td>Injection velocity</td>
<td>1000</td>
<td>cm/s</td>
</tr>
<tr>
<td>Spark timing</td>
<td>140</td>
<td>dBTC</td>
</tr>
<tr>
<td>Equivalence Ratio</td>
<td>1</td>
<td>-</td>
</tr>
<tr>
<td>Wall temperature</td>
<td>380</td>
<td>K, isothermal</td>
</tr>
<tr>
<td>Intake Pressure</td>
<td>1.0</td>
<td>bar</td>
</tr>
<tr>
<td>Intake charge temperature</td>
<td>500</td>
<td>K</td>
</tr>
<tr>
<td>Injection location</td>
<td>-</td>
<td>Center of the domain</td>
</tr>
<tr>
<td>Injection width</td>
<td>16</td>
<td>mm</td>
</tr>
<tr>
<td>Temperature of fuel injected</td>
<td>350</td>
<td>K</td>
</tr>
<tr>
<td>Duration of injection</td>
<td>1</td>
<td>CAD</td>
</tr>
</tbody>
</table>

Fuel and Oxidizer composition

<table>
<thead>
<tr>
<th>Stream</th>
<th>Composition (mole%)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Fuel</td>
<td>Isooctane (C₈H₁₈) 100%</td>
</tr>
<tr>
<td>Oxidizer</td>
<td>O₂ 21%</td>
</tr>
<tr>
<td></td>
<td>N₂ 79%</td>
</tr>
</tbody>
</table>

Table 4.4: MPFI run conditions

<table>
<thead>
<tr>
<th>MPFI run conditions</th>
</tr>
</thead>
<tbody>
<tr>
<td>DI percentage (%)</td>
</tr>
<tr>
<td>Injection timing (dBTDc)</td>
</tr>
</tbody>
</table>
4.3 Results and Discussions

This chapter investigates the effect of charge stratification and partial homogenous mixtures on SACI combustion. Observations pertaining to combustion phasing and thermodynamic results are discussed, as a result of direct injection at different times during the engine cycle and by creating variations in charge homogeneity, as percentage of total fuel injected by the two methods is varied. First, there is a comparison made between different direct injection (DI) timings, keeping the percentage of fuel injected through PFI constant. Following that, DI percentage (DI%) sweeps are made for each DI timing, to show the effect of charge stratification. The spark ignition timing is kept constant at 40dBTDC, as it showed consistently stable combustion phasing, for a majority of the cases considered in Chapter 2. Equivalence ratio of 1 is considered for all cases, to ensure sufficient fuel homogenization in all cases. The fuel is injected within 1 degree of crank angle, as longer injection durations created excessive turbulent perturbations within the chamber, affecting the homogeneity of the charge in the chamber, resulting in extremely unstable combustion due to advanced autoignition.

4.3.1 Injection Timing Sweep

Figure 4.1 shows the plots of pressure for the injection timing sweep cases, with 30% of the fuel mass directly injected at the times specified, while the rest of the fuel charge is introduced at the beginning through PFI. Sparking occurs at 40dBTDC, as stated earlier. It can be observed that the pressure traces are similar, with a stable flame propagation phase, followed by autoignition. It is noticed that the earlier injection timing shows a prominent combustion phasing. When injection occurs at 120dBTDC, it is noticed that the curve follows a smooth trajectory beyond TDC, indicative of a longer flame propagation stage, followed by late autoignition. The
late injection at 100dBTDC is relatively very unstable with a less prominent autoignition peak, although there is pressure rise near TDC. The injection timing of 110dBTDC exhibits the highest in-cylinder pressure of all four cases, with a more prominent autoignition peak. These curves show the effects of charge stratification on SACI combustion. An earlier injection timing allows more time for the fuel to mix with the charge already present in the chamber. Due to increased homogeneity, there is lesser stratified mixture directly exposed to the energy of the spark. This allows the formation of flame kernels which are enough to initiate flame propagation within the chamber. A later injection timing implies lesser time available for mixing, leaving a greater amount of fuel charge near the spark plug which leads to higher heat energy being expelled at kernel formation. The local charge temperature rises rapidly, which leads to early initiation of autoignition spots, making the combustion stable. The injection at 110dBTDC shows a suitable balance of combustion phasing, with a relatively greater amount of energy being expelled during autoignition.
Figure 4.1: Pressure curves for the injection timing sweep cases, with a DI% of 30%

All the cases exhibit slight perturbations around 20dATDC, which signify that autoignition occurs with excess fuel charge remaining in the end gas, which represents that the fuel injected creates locations of relatively richer mixture, which complete combustion later than the majority of the in-cylinder charge.
Figure 4.2: Heat release rate plot for injection at 130dBTDC, DI% 30%

Figure 4.3: Heat release rate plot (magnified) for injection at 120dBTDC, DI% 30%
Figure 4.4: Heat release rate plot for injection at 110dBTDC, DI% 30%

Figure 4.5: Heat release rate plot for injection at 100dBTDC, DI% 30%
The heat release rate plots shown above, in Figs. 4.2 to 4.5, correspond to the pressure plots discussed above. It is observed that the plots are similar to those discussed in Chapter 2. There is, however, significant autoignition at later crank angles. This observation is absent in the case of injection at 110dBTDC, where the combustion is characteristic SACI combustion. There is a prominent autoignition peak, which follows a relatively smooth flame propagation phase. This shows that an injection timing of 110dBTDC, provides a good balance between charge stratification and homogenization, compared to the other timings. For the delayed injection timing of 100dBTDC, a rapid rise in heat release rate is seen early during the combustion cycle. This is due to lesser time being available for mixing of the stratified fuel mixture, and a greater concentration of fuel mixture remains at the time of sparking. Therefore, after the formation of the kernel the local richer mixture ignites, causing a jump in energy release. This phenomenon is followed by flame propagation burning of the gas mixture, which eventually causes the end gas to self-ignite. The case of 120dBTDC injection shows that while there is autoignition occurring near TDC, it does not cause a significant rise in pressure, with a less prominent autoignition peak.
The comparison shown in Fig. 4.6 displays the maximum pressure which is attained at each injection timing. It is observed that an injection timing of 110°BTDC has the highest maximum pressure, however it does exhibit a rapid rise in heat release rate before TDC during autoignition. This could indicate a tendency for more ringing in the chamber. The temperature plots in the following pages, for the cases of 110 and 100°BTDC injection timing, provide a better understanding of the events taking place within the combustion chamber.

**Figure 4.6**: Comparison of maximum pressure for each of the injection timing cases
Figure 4.7: Temperature plots over the normalized spatial domain, for the injection case of 110dBTDC, during the combustion events.
Figure 4.8: Temperature plots over the normalized spatial domain, for the injection case of 100dTDC, during the combustion events.
The plots shown in Fig. 4.7 represent the temperature variations over the normalized one-dimensional spatial domain, for the injection timing case of 110dBTDC. Sparking occurs at 40dBTDC, which leads to kernel growth, followed by flame propagation, which is indicated by the spread of the central temperature peak where sparking takes place. Autoignition occurs at 2dBTDC, when the in-cylinder thermodynamic conditions support the self-ignition of the end gas mixture. As cycle time progresses, it is noticed that autoignition is predicted to occur first on the right-hand side of the chamber, followed by the opposite side. This is followed by various other autoignition spots in close succession throughout the cylinder, so that the majority of the energy has been released by 10dATDC. There are occurrences of minor trailing autoignition spots, which lead to minimal rates of heat release as can be seen in Fig. 4.4. By 23dATDC, a majority of the fuel is consumed due to combustion, and the autoignition processes extinguish gradually as volume expansion occurs.

Figure 4.8 shows the spatial temperature profiles for the injection timing case of 100dBTDC. Referring to Fig. 4.5, a jump in the heat release rate has been observed. This can be explained by the plot shown at 15dBTDC, where there is creation of additional temperature peaks in the vicinity of the central temperature peak. This indicates that excessive fuel mixture stratification takes place near the region of early kernel growth (EKG, discussed in Chapter 2) which ignites soon after the formation of the kernel. As a result of late injection of fuel, the time available for mixture homogenization is not enough causing excessive heat release very early in the cycle. However, this does not lead to cascading autoignition sites within the chamber, instead evolving into a flame front which burns the mixture as it propagates through the chamber, eventually causing autoignition of the end gas at 1dATDC. It is seen that the maximum pressure for this case is comparatively lower and it can be said that the lack of homogenization is a cause.
The temperature at the center of the domain is greater than the surrounding regions. Also, it is observed that significant autoignition sites are formed till later, at 17dATDC compared to the earlier case of 10dATDC, which explains the late heat release peak in Fig. 4.5.

**Figure 4.9**: Plot depicting autoignition timing vs. injection timing, for DI%=30%

Figure 4.9 shows the relation between the autoignition timing and injection timing for the cases considered above. While the difference is not significant, it is observed that as injection timing is delayed, autoignition shifts beyond TDC. For the case of 30% direct injection of fuel, it can be said that combustion phasing is less sensitive to injection timing. The delay in autoignition with injection timing delay, as seen earlier in this section, could be due to reduced homogenization of the mixture charge. The richer mixture stratified near the spark plug delays the transition from kernel growth to flame propagation phase, thus causing delay in combustion phasing. The next section discusses the effect of varying percentage of directly injected fuel at every case of injection timing discussed thus far. The intent is to study the effect on combustion phasing due to variation in the concentration of stratified mixture near the spark plug and the effect due to varying the time at which this mixture is injected into the chamber.
4.3.2 Direct Injection Percentage Sweep

![Pressure plots for injection timing fixed at 130dBTDC, while DI% is varied](image)

**Figure 4.10**: Pressure plots for injection timing fixed at 130dBTDC, while DI% is varied

For this set of cases, the injection timing is kept constant at 130dBTDC, and the DI% is varied from 20% to 60% in increments of 10%, to study the variation over a range of fuel mixture stratification. In Figure 4.10, the pressure plots obtained from the simulations are presented. The in-cylinder pressure reduces as charge stratification is increased. This is because of presence of excess fuel near the spark plug which cannot be homogenized in time. These rich zones absorb more heat due to the energy required to combust the fuel, as the fuel is being injected at a temperature much lower than in-cylinder temperature. This causes a delay in combustion phasing, leading to erratic heat release. As a result, the stability of transition to flame propagation from
kernel formation is affected. This can be observed in the heat release curves in Fig. 4.11. As DI% increases, there is a delay between kernel formation and flame propagation where the excess fuel near spark plug combusts, indicated by the early jump in the plots shown in Fig. 4.11. This has been observed in the earlier section for later injection timings as well. The heat release curves also show a delay in combustion phasing as DI% is increased, observed as shifting of the autoignition peaks to a later time beyond TDC.
Figure 4.11: Heat release rate curves for the cases considered. Injection timing 130dBTDC, DI%=20%-60%
Figure 4.12: Pressure plots for injection timing fixed at 120dBTDC, while DI% is varied

Figure 4.12 shows the pressure curves for MPFI simulations with injection timing fixed at 120dBTDC, are presented. Figure 4.13 shows the heat release rate plots for the same set of cases. It is observed that increase in DI% decreases in-cylinder pressure. However, this decrease is not significant except for the case with DI%=60%. In this case, it is observed that flame propagation exists for a majority of the combustion duration, with very late and significantly low autoignition. This is because the stratified mixture requires a longer time to burn, for the end gas to reach self-ignition conditions. This case significantly affects combustion phasing. As fuel stratification increases, the stability of the combustion cycle deteriorates with multiple autoignition sites forming early in the cycle. It is also observed that increase in fuel stratification and injection delay instigates the early spontaneous combustion of the stratified fuel mixture located in the vicinity of the spark plug. This is an effect of lack of homogeneity in the charge mixture.
Figure 4.13: Heat release rate curves for the cases considered. Injection timing 120dBTDC, DI%=20%-60%
For the cases shown in Figs. 4.14 and 4.15, the injection timing is fixed at 110dBTDC and the DI% is varied. These cases shows more instability compared to earlier injection timings, because of lesser homogenization of the mixture before sparking occurs. A low DI% shows more stable flame propagation but as DI% is increased, charge stratification near the spark plug increases, which in turn increases proclivity for autoignition. Compared to earlier cases of injection, a 40% direct injection causes a very early spontaneous combustion seen in injection timing cases of 130 and 120 dBTDC as well, but for DI% of 50% and 60%. This signifies that the later injection offers lesser time for a mixture with sufficient homogeneity to form. In this injection timing case it is observed that the 20% DI fraction of fuel has a lower in-cylinder pressure compared to 30% DI of fuel. Since injection timing is delayed, more stratification supports flame propagation while creating self-ignition conditions earlier in the cycle, as can be seen for the respective cases in Figure 4.15.
Figure 4.15: Heat release rate curves for the cases considered. Injection timing 110dBTDC, DI%=20%-60%
Figure 4.16: Pressure plots for injection timing fixed at 100dBTDC, while DI% is varied

A later injection time considered here, 100dBTDC, for different DI% conditions have two major hindering factors. Firstly, a later injection timing leaves lesser time for charge mixing and homogenization. This causes greater amount of fuel to accumulate near the spark plug, resulting in delayed combustion phasing due to the barrier created by the concentrated fuel mixture. Once the local fuel molecules have combusted to form flame kernels, flame propagation is initiated. This kernel formation acts as a sink for heat energy created by the EKG, due to its low temperature when injected, resulting in unstable spontaneous combustion zones near the center, as seen in the earlier section, which lead to low pressure rise due to combustion cycle, and unstable autoignition spots in the combustion chamber. These observations are depicted in Figs.16 and 17, with pressure plots and heat release rate plots respectively. A trend of decreasing in-cylinder pressure is observed as fuel stratification is increased.
Figure 4.17: Heat release rate curves for the cases considered. Injection timing 100dBTDC, DI%=20%-60%
Figure 4.18: Comparison of maximum in-cylinder pressure for different DI% values as injection timing is delayed

Figure 4.18 compares the maximum in-cylinder pressure generated due to each of the cases considered in this section. A general trend of pressure decrease with increase in fuel stratification is observed. As explained earlier, increased fuel stratification does not allow the in-cylinder charge to mix completely preventing the formation of homogenous mixture. As the injection timing is delayed, the mixture homogeneity is adversely affected, causing the combustion to become erratic, with spontaneous combustions and autoignition spots arising at different points simultaneously. 20% DI of fuel shows relatively stable combustion phasing as injection timing progresses. However, this is not the case as fuel stratification is increased. As injection timing is delayed, the combustion phasing loses characteristic SACI properties, at higher levels of injected fuel.
The plot in Fig. 4.19 shows the autoignition points for each of the cases considered above, against the injection timing. Comparison is also shown with variation of DI%. It is seen that the sensitivity of combustion phasing to injection timing decreases as percentage fuel injected (DI%) increases. The combustion phasing is, however, sensitive to DI% change, as the trends for autoignition timing can be seen to delay with increase in DI%, for all injection timings.

**Figure 4.19**: Plot showing autoignition points for different injection timing cases, comparing variation in DI%
4.3.3 PFI vs. MPFI

This section presents a brief comparison between PFI case and MPFI cases, with an injection timing of 130dBTC, with DI% variation. As discussed in the earlier section, MPFI cases displayed a decrease in in-cylinder pressure with increasing DI%, that is increasing stratification. PFI case shows the highest pressure rise, which is justified due to the presence of a homogenous charge in the combustion chamber. The increase in stratification was observed to control combustion phasing to a certain extent, reducing the heat released during autoignition. This could be due to the combustion of the local fuel mixture which accumulated near the spark plug. Figures 4.20 and 4.21 show the comparison of the in-cylinder pressure conditions. There is a very clear trend observed.

![Figure 4.20: Comparison between PFI and MPFI cases with injection timing 130dBTC](image)

**Figure 4.20**: Comparison between PFI and MPFI cases with injection timing 130dBTC
Figure 4.21: Comparison of maximum pressure between PFI and MPFI cases

Figure 4.22: Autoignition timing vs. DI% for injection timing 130dBTDC

Figure 4.22 shows the change in autoignition timing with DI%. For the same simulation case, PFI mode predicted autoignition at TDC. As the DI% for fuel is increased, there is a general trend of advance in combustion phasing observed. This shows that combustion phasing is sensitive to DI%, and can be controlled by varying the mass fraction of total fuel injected into the cylinder.
4.4 Conclusion

In this section, engine operating conditions were kept constant while direct injection parameters, that is injection timing and mass fraction of directly injected fuel, were varied. The following are the conclusions.

1. Multipoint Fuel Injection Strategy (MPFI) creates charge stratification, which is useful in controlling combustion phasing. It is observed that SACI combustion is affected by injection timing. A later injection timing increases the instability in the combustion chamber, due to lower homogeneity in the fuel-air mixture.

2. Autoignition timing is observed to advance with injection timing advance, however combustion phasing was not found to be very sensitive to injection timing.

3. As the percentage of fuel mass directly injected (DI%) into the combustion chamber increases, pressure is observed to reduce as a general trend. It is found that greater DI% leads to accumulation of fuel near the spark plug, which delays the flame propagation affecting the in-cylinder pressure rise.

4. Combustion phasing is found to be sensitive to DI%, and is delayed with increase in DI%. Sensitivity to injection timing is higher at lower DI%, advancing with injection timing advance. However, it decreases as DI% is increased.

5. In-cylinder pressure attained is lower for MPFI case compared to PFI case under same engine operating conditions.
CHAPTER 5
SUMMARY AND FUTURE WORK

5.1 Summary

The Spark Assisted Compression Ignition (SACI) mode of combustion has been investigated numerically using a One-Dimensional Turbulence (ODT) model to investigate the effects of different parametrical changes on the combustion characteristics of this method of engine operation. The ODT model provides a realistic representation of the unsteady turbulent combustion process due to its high spatial and temporal resolution and serves as an essential tool in the study of SACI combustion. The study is conducted in two parts. Firstly, the port fuel injection method is used and parameters like equivalence ratio, spark timing and compression ratio are varied, to study the effect on SACI combustion. A number of cases are simulated by varying the parameters under homogenous fuel-air mixture conditions to identify a suitable range for characteristic SACI combustion. The spark ignition timings of 45, 40 and 35dBTDC, are found to exhibit characteristic SACI combustion, for a stoichiometric mixture. Simulations were then conducted, using the three spark timings mentioned above, by varying the equivalence ratio and compression ratio, to investigate the behavior of a SACI engine over a range of different conditions. Equivalence ratios of 0.5 to 1 were used and compression ratio values were varied over the values of 12.5, 14, 16, and 18. Thermodynamic parameters like in-cylinder pressure, heat release rate and spatial temperature variations are presented to study the effects on SACI combustion phasing. For the second part, a combination of PFI and direct injection (DI), termed as multipoint fuel injection (MPFI) strategy, is used to study the effect of fuel stratification on SACI combustion. Effects due to injection timing variation are investigated over four injection timings, followed by changing the amount of fuel injected via direct injection (DI%) for each of the injection timing cases. The individual
conclusions for the two major parts of the study are detailed in Chapters 3 & 4. The following are
the major findings.

It was observed that SACI combustion mode operates in 4 stages, i.e., spark discharge, early kernel growth, flame propagation and autoignition. Spark ignition timing plays a significant role in combustion phasing. Characteristic SACI combustion was observed to occur within the range of spark ignition timings of 35 to 45dBTDC. A very early spark ignition timing results in possible advance in the autoignition phase, resulting in uncontrolled autoignition and misfiring. Such events have proclivity to high ringing in the combustion chamber and potential for knocking. Late spark ignition times cause delayed combustion phasing, by elongating the duration of flame propagation. This results in autoignition of the end gas later in the cycle, when the combustion volume is expanding due to piston movement, hampering the in-cylinder pressure rise.

Equivalence ratio is observed to impact maximum in-cylinder pressure and combustion phasing. For stoichiometric cases (ϕ=1), in-cylinder pressure reaches greater values compared to leaner mixtures. It is also noticed that as the equivalence ratio increases, the heat release during the autoignition event increases. It is seen in this study that, while there is no optimum value of equivalence ratio suitable for every spark ignition time case, an engine can be operated at a certain equivalence ratio for certain spark timings.

As compression ratio is increased, the maximum in-cylinder pressure attained increases. However, it was observed that the heat release became more unstable as the compression ratio was increased. A leaner mixture showed better combustion stability compared to richer mixtures. Compression ratio also controls combustion phasing. As it is increased, the autoignition phase advances. It is seen that for a very high compression ratio (CR=18), the autoignition occurs
relatively early, just before TDC. Also, it is noticeable that higher compression ratio restricts the autoignition phase to a limited duration.

Multipoint Fuel Injection Strategy (MPFI) creates charge stratification, which is useful in controlling combustion phasing. It is observed that SACI combustion is affected by injection timing. A later injection timing increases the instability in the combustion chamber, due to lower homogeneity in the fuel-air mixture. Autoignition timing is observed to advance with injection timing advance, however combustion phasing was not found to be very sensitive to injection timing.

As the percentage of fuel mass directly injected (DI%) into the combustion chamber increases, pressure is observed to reduce as a general trend. It is found that greater DI% leads to accumulation of fuel near the spark plug, which delays the flame propagation affecting the in-cylinder pressure rise. Combustion phasing is found to be sensitive to DI%, and is delayed with increase in DI%. Sensitivity to injection timing is higher at lower DI%, advancing with injection timing advance. However, it decreases as DI% is increased. In-cylinder pressure attained is lower for MPFI case compared to PFI case under same engine operating conditions.

5.2 Future Work

The following are possible avenues which could be investigated beyond this study.

1. A study on cyclic variability of the combustion characteristics could be investigated, to get a better idea of change in combustion phasing over multiple cycles.

2. While a heated intake gas mixture is used in this study, the cases can be studied using external and internal EGR, to study the effects on combustion phasing and stability to extend operating limits of characteristic SACI combustion.
3. A study of major emission species could be conducted, like NOx, CO, etc., over various cycles, to investigate the emission characteristics of SACI combustion.

4. Other fuels such as ethanol and dimethyl ether (DME), with similar reactivity as iso-octane, could be used to investigate the SACI mode of combustion.

5. Injections at multiple locations could be investigated with this model, to study the effect of charge stratification over a greater domain space.

6. This model assumes the combustion charge to be an ideal gas mixture. The study could be expanded to consider soot and droplet models to simulate particulate matter and fluid-vapor interactions in the cylinder.

7. This ODT SACI engine model could be used as a sub-grid model for a 3-D CFD solver, to obtain high resolution simulation results for the entire cylinder volume.
REFERENCES


