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Injection Parameters Effect on the Performance of Compressed Natural Gas Direct Injection (CNG-DI) Engine Under Lean Stratified Conditions

submitted by

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for the fulfillment of the requirements for the degree of

Masters of Science in Mechanical Engineering

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UNIVERSITI TEKNOLOGI PETRONAS

Injection Parameters Effect on the Performance of Compressed Natural Gas Direct Injection (CNG-DI) Engine Under Lean Stratified Conditions

By

Raja Shahzad Hassan

A THESIS

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AS A REQUIREMENT FOR THE

DEGREE OF MASTERS OF SCIENCE IN MECHANICAL

ENGINEERING

BANDAR SERI ISKANDAR,

PERAK

June, 2008

DECLARATION

I hereby declare that the thesis is based on my original work except for quotations and citations which have been duly acknowledged. I also declare that it has not been previously or concurrently submitted for any other degree at UTP or other institutions.

Signature: ________________

Name : Raja Shahzad Hassan

Date : ________________

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ABSTRACT

Natural gas has been identified as an alternative to crude oil fuels such as gasoline and diesel. Natural gas utilization as an automotive fuel is yet to be fully used to its optimum because most of the vehicles converted to natural gas are still using port injection or carburetors. Natural gas can be used either in compressed (CNG) or liquefied (LNG) form. Compressed natural gas (CNG) has huge potential for improving the thermal efficiency of spark ignited (SI) engines due to combustion-specific properties such as high knock resistance and extreme stratification capabilities for lean air/fuel ratio. The main drawback or disadvantage of using natural gas in the engine is that its performance drops compared to gasoline or diesel engine. For current passenger car standard applications a power drop of approximately 10% is noticed by use of CNG. The prime source of performance drop is because of lower volumetric efficiency, lower energy density and longer combustion duration of natural gas. Reduced volumetric efficiency of induction system is being widely studied to optimize for the losses causing this diminution. This drawback can be compensated by direct injection of CNG straight into the combustion chamber, and therefore giving way to utilize the maximum benefits from using CNG as automotive fuel. Combustion of natural gas is cleaner i.e. lower exhaust emissions, also because of its higher octane number, the engines can be designed with higher compression ratios, hence increasing the thermal efficiency. Direct injection systems for natural gas engine are expected to solve the problem of lower volumetric efficiency. Optimization of injection parameters is required for the optimum outcome of natural gas engine. Besides favorable engine out emissions, the engine concept operates without power loss and with absolute low fuel consumption. The "Direct CNG Injection" may be a highly attractive solution for automotive propulsion systems.

The following research is carried out on a dedicated 4-stroke natural gas spark ignition engine with a compression ratio of 14. A centre direct injection system is used, where the injector is placed at the centre of cylinder head with spark plug offset by 6mm. Engine is being tested for idle and partload conditions using homogeneous and stratified pistons. Injection parameters such as injection timing and injector spray angle are investigated while keeping the injection pressure constant, to find out the effect of injection parameters on

CNG-DI engine. Ignition timing is adjusted to obtain the maximum brake torque (MBT). Results for both stoichiometric and stratified charges at idle and partload conditions are compared to examine the features of both operations under these conditions.

The experimental results are categorized based on each injection parameter. Firstly, injection timing from early injection (300 degree BTDC) to late injection (80 degree BTDC) is investigated for stoichiometric conditions. For stratified operation the injection starts after the closing of air intake valve at 132 BTDC. The injection is delayed further till the limit for each RPM as set by the ECU (Engine Control Unit). Injection timing effect for idle and partloads is compared for stoichiometric and lean stratified operations, for injection timings starting after the closing of intake valve. Engine speed is limited from 2000 to 5000 RPM.

Injection pressure is kept constant at 18 bars for both stoichiometric and stratified operations. Injection pressure is affecting the fuel delivery rate. Lower injection pressure needs longer injection duration to deliver the required fuel to the engine. Two injectors with different injection angle are investigated, 30 deg (NAI) and 70 deg (WAI). Both injectors have their distinctive characteristics which can be applied on certain engine operational conditions.

Lean stratified operation proved to be better for lower engine speeds while having overall lower brake specific fuel consumption over a wide range. Lower performance at higher engine speeds is due to less time for mixture formation, lower fuel content and excessive stratification. Wide angle injector (WAI) proves to be giving better performance than narrow angle injector (NAI) at lean conditions. Faster mixing rate of WAI might be responsible for such behavior.

Nitrogen oxides (NOx) emission for lean stratified operation is higher at lower engine speeds which indicate higher temperatures of combustion, for all injection timings except the lean limit where it is lesser. Unburned hydrocarbons are slightly higher than stoichiometric and tend to increase with the engine speed. Higher cycle to cycle variation, mixture formation, excessive stratification and bulk quenching are the reasons for such behaviors. Carbon monoxide (CO) emissions for lean stratified operation are quite lower compared to stoichiometric for all injection timings.

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ABBREVIATIONS

ABDC	After Bottom Dead Center
AFR	Air Fuel Ratio
ATDC	After Top Dead Center
BMEP	Brake Mean Effective Pressure
BSFC	Brake Specific Fuel Consumption
BTDC	Before Top Dead Center
CH_4	Methane
CNG	Compressed Natural Gas
CO	Carbon Monoxide
COV	Coefficient of Variation
CVCC	Compound Vortex Controlled Combustion
DI	Direct Injection
ECU	Engine Control Unit
EGR	Exhaust Gas Recirculation
ERI	Engine Remote Interface
FMEP	Friction Mean Effective Pressure
FTIR	Fourier Transform Infrared
HC	Hydrocarbon
IANGV	International Association for Natural Gas Vehicle
IC	Internal Combustion
IMEP	Indicated Mean Effective Pressure
IVC	Intake Valve Closed
LNG	Liquefied Natural Gas
LPG	Liquefied Petroleum Gas
MBT	Maximum Brake Torque
NAI	Narrow Angle Injector
N_2	Nitrogen
NO_x	Nitrogen Oxides

NO	Nitric Oxide
N_2O	Nitrous Oxide
NO_2	Nitrogen Dioxide
O_2	Oxygen
OBD	On Board Diagnosis
RCM	Rapid Compression Machine
RPM	Revolutions Per Minute
SI	Spark Ignition
UHC	Unburned Hydrocarbon
VRA	Vehicle Refueling Appliances
VVT	Variable Valve Timing
WAI	Wide Angle Injector
WOT	Wide Open Throttle

CHAPTER 1

INTRODUCTION

1.1 Background

Most automobiles in use today are propelled by gasoline (also known as petrol) or diesel internal combustion engines. Diesel and gasoline are produced from the fractional distillation of the crude oil. Fuels made from crude oil are often called as hydrocarbon fuels. Hydrocarbon fuels have been the main energy source since the development of the automotive industry, and are extensively being used as fuel for transportation since more than a century, but hydrocarbon fuels as a non-renewable source of energy are expected to be depleted, as it would have limited reserves. According to one study the current known reserves are enough just for the next 30 to 35 years [1]. This has forced the automotive industry to find the alternative fuels to reduce the dependency on the hydrocarbon fuels. Furthermore, automotives are known to be the main cause of air pollution and are also blamed for contributing to climate change and global warming. Increasing costs of hydrocarbon fuels and tightening environmental law and restrictions on greenhouse gases emissions are propelling work on alternative power systems for automobiles. Efforts to improve or replace these technologies include hybrid vehicles, electric vehicles and hydrogen vehicles. With increasing concern of energy security and environmental safety, much effort has been focused on the development of alternatives for hydrocarbon fuels. One of the most promising is natural gas utilization as automotive fuel. Natural gas has methane as the main composition (70-98%) along with ethane, butane, propane and small amounts of water vapors. Natural gas is considered as the cleanest fuel after hydrogen and is considered as a good alternative to gasoline and diesel, since it produces considerably less pollutants.

1.2 Natural Gas as Automotive Fuel

The oil shortages of the late 1960s and early 1970s brought renewed interest in natural gas as a fuel source, especially for automobiles. During the last three decades the following four fuels have emerged as the potential contenders to be the alternative fuel for the future automotives, which are methanol, ethanol, natural gas and hydrogen. Natural gas is considered cleanest after hydrogen as an alternative to gasoline and diesel. Since the implementation of strict environmental regulations and energy security, the use of CNG (Compressed Natural Gas) in automotives has been developed rapidly. Technological developments such as direct injection (DI), variable valve timing (VVT), exhaust gas recirculation (EGR) and on-board diagnosis system (OBD) is proven to exhibit the ability to reduce fuel consumption and the emission levels to a very low limit.

Since the drastic increase in fuel prices the conversion of vehicles to CNG in many countries had been increasing steadily. In Germany, CNG-operated vehicles are expected to increase to two million units of motor-transport by the year 2020. Argentina and Brazil, in the South America, are the two countries with the largest fleets of CNG vehicles. Conversion has been facilitated by a substantial price differential with liquid fuels, locally-produced conversion equipment and a growing CNG-delivery infrastructure. CNG has grown into one of the major fuel sources used in car engines in Pakistan, Bangladesh and India. As of July 2007 Pakistan is the largest user of CNG in Asia, and second largest user in the world [2]. Statistical data provided by IANGV (International Association for Natural Gas Vehicle) shows that Malaysia is in the 20th position among all the countries that has been using natural gas vehicle in the transportation sector.

The natural gas products consist of CNG (compressed natural gas), LNG (liquefied natural gas) and LPG (liquid petroleum gas). Propane (also known as Liquefied Petroleum Gas, LP-Gas or LPG), and natural gas are both gaseous fuels. Natural gas can be used as compressed natural gas (CNG) or Liquefied Natural Gas (LNG). About 60% of the propane produced comes from natural gas wells. The rest is a byproduct of crude oil refining. LPG combustion is much cleaner than gasoline, though not as clean as natural gas. Moreover studies showed that LPG has higher thermal efficiency and lower fuel consumption compared to CNG, but it has safety issues and high demand for household applications. The prime safety concern with LPG is that it is heavier than air. If a leak in an LPG fuel system occurs, the gas will have a tendency to sink into any enclosed area and thus poses a risk of explosion and fire. Compressed Natural Gas (CNG) compared to LPG is lighter than air and thus less risky [3].

Table 1-1 Natural Gas Vehicle Statistics [2]

Position	Country	Vehicles*	Refuelling Stations	VRA**	Last Updated
1	Argentina	1,459,236	1,400	32	5-Dec
2	Brazil	1,357,239	1410		7-Mar
3	Pakistan	1,300,000	1230		7-Apr
4	Italy	410,000	558		6-Dec
5	India	334,658	321		Apr 06
20	Malaysia	19,000	46	1	6-Dec
	TOTALS	**6,080,582**	**10,068**	**9,176**	

In today's applications, CNG has been mainly applied on combination with other fuels in the bi-fuel or tri-fuel vehicle system. Engines are generally optimized for a specific fuel to be burned. Hence using other fuels as blends, there is a considerable reduction in the performance. Researchers are focused on how to reduce the performance tradeoffs. Dedicated CNG engine is expected to utilize the advantages of using CNG as fuel. Natural gas as a potential alternative to gasoline and diesel emits low exhaust emissions because of simplicity of the chemical bonding and has low Hydrogen-Carbon ratio.

Most CNG fuelled vehicles today are utilizing the conventional engine technology to create air and fuel mixture, where a conversion kit is added to a vehicle designed for gasoline so that it can operate on an alternative fuel, either on gasoline or on CNG.

Therefore a reduction in the performance is generally experienced. To improve the engine performance, port fuel injection and direct injection systems are introduced by the auto manufacturers. These new systems are claimed to perform better compared to conventional carburettor mixer systems. Further development of CNG engine technology has focused on increasing the engine power that can be achieved by applying current spark ignition (SI) engine technology to the CNG fuelled vehicle [4, 5].

Dedicated CNG engines and the development of direct-injection as fuelling system in CNG operated engines are expected to solve the problem of reduction of performance. DI systems also increase the volumetric efficiency and could enhance the mixture formation due to high delivery rate [6]. EGR on CNG engine, results in low emissions of NO_x and HC at stoichiometric condition. However, EGR decreased the brake efficiency of the engine [7]. Turbochargers in CNG engines gives higher power due to higher intake charge, but the power increased is not considerable at low engine speeds [8].

1.3 Problem Statement

Stratified charge combustion has been widely studied due to its potential for low fuel consumption and low emissions. In production gasoline DI (Direct Injection) engines charge stratification is achieved by using wall-guided spray with enhanced gas motion. Charge stratification has been realized by using fuel injection in most of the previous studies [1, 2], but spatial mixture distribution is difficult in this. Some authors have implied distributed chambers for two different mixtures [3-5]. The fuel air mixture formation in a direct injection stratified charge engine is influenced by various parameters, such as atomization, evaporation, and in-cylinder gas motion at high temperature and pressure. In gas direct injection, turbulent mixing and diffusion of the fuel greatly influence the in-cylinder motion and combustion and thus, require careful control.

1.4 Objectives

The objectives of this research are:

1. To investigate the combustion characteristics of CNG using a direct injection (DI) system on a spark ignition engine under lean operating conditions.
2. To obtain performance characteristics of the CNG direct-injection engine by optimizing injection parameters at idle and partloads.

1.5 Scope of Work

Research is basically focused on injection parameters on CNG-DI spark ignition engine. Injection parameters include injection timing, injection pressure and injection spray angle on the range of engine speed from 2000 to 5000 RPM. For the thesis we are considering injection timing and injection spray angle only, while injection pressure is kept constant. Experiments are conducted at idle and partload conditions using lean mixture. Ignition timing is set to get maximum brake torque.

1.6 Thesis Organization

This thesis consists of six chapters which include introduction, literature review, theoretical background, methodology, results and analysis, and conclusions and recommendations. Experimental work is carried out and is analyzed to meet the objective as described above.

Chapter 1 describes the background of this research that also covered the current development of engine technology mainly for spark ignition engine. Furthermore, chapter 1 consists of explanation of both advantages and disadvantages of this engine development technology.

Chapter 2 describes the literature that has been consulted in order to analyze the past developments and the current drive for the new technologies. The main focus of the literature review is on the fueling system technologies for spark ignition engines while using CNG as fuel and their development and improvement for each technology, their benefits and drawbacks regarding their respective characteristics.

Chapter 3 briefly describes the theoretical background of engine, its types, working cycles, operating parameters etc. It also elaborates the theory on engine performance along with its efficiency and combustion characteristics. Emissions resulted during the combustion process are also explained briefly.

Chapter 4 specifies the methodology adopted for the following research. It also gives details on the equipment utilized, its characteristics and measurement capabilities. The parameters being researched are also explained along with the equations to analyze those parameters. Data collection procedure is also detailed, with the constraints faced during this procedure.

Chapter 5 includes the experimental results and discussions. Engine performance, combustion and emissions results are illustrated for each parameter within a certain range of limitations which are also explained. Performance parameters such as torque, power, fuel consumption and engine efficiency are discussed. Furthermore, combustion process is discussed and emission resulted during this process is carried out for every parameter in the certain range of engine speed.

Chapter 6 contains the concluding remarks of this research and proposed work for future development.

CHAPTER 2

LITERATURE REVIEW

2.1 Internal Combustion Engine Development

The history of engine dates back to 1st century Roman Egypt where Hero Alexandria demonstrated the 1st steam-powered device. Since then successive works have been done by different scientists and engineers to develop the steam engine. Thomas Savery, an English military engineer and inventor who in 1698, patented the first crude steam engine. Savery used this engine to pump water from coal mines. Thomas Newcomen was an English blacksmith, who invented the atmospheric steam engine, an improvement over Thomas Savery's previous design. The Newcomen steam engine used the force of atmospheric pressure to do the work. James Watt, after whose name the unit of power is named, was a Scottish engineer and inventor, who patented his steam engine in 1769. The industrial development during the 18th and 19th century is much credited to the invention of James Watt's steam engine, as it was being used as a source of power in industries, locomotives, ships etc. Along with the development and utilization of steam engines scientists and inventors also started looking for alternative means for power generation. High cost, low efficiency and complex systems of the steam engines being the main source for such developments. The internal combustion engine was a promising alternative to that of steam engines. Over the years, many scientists and inventors patented their designs and proposals for the new alternatives. Nikolaus Otto and Langen in 1866 introduced the first concept of internal combustion engine. Their engine used gasoline as fuel and it had a very low power-weight ratio. In 1877 Otto received a patent for the "Otto Cycle", which is also known as the four stroke cycle. One of the earliest models of an engine operating on Otto cycle is shown in the Figure 2-1, which is placed at London Science Museum. To date Otto cycle being primarily used in the IC engines. Rudolph Diesel in 1892 developed a diesel engine which is operated on compression ignition of the fuel inside the combustion chamber. Higher compressions ratios are required for the diesel engines to auto ignite the fuel. The compression ratios for diesel engines are above 19:1 compared to Otto engine where compression ratios are 10:1 and is limited by the knock limit. Therefore diesel engines have higher thermal efficiency and lower fuel consumption.

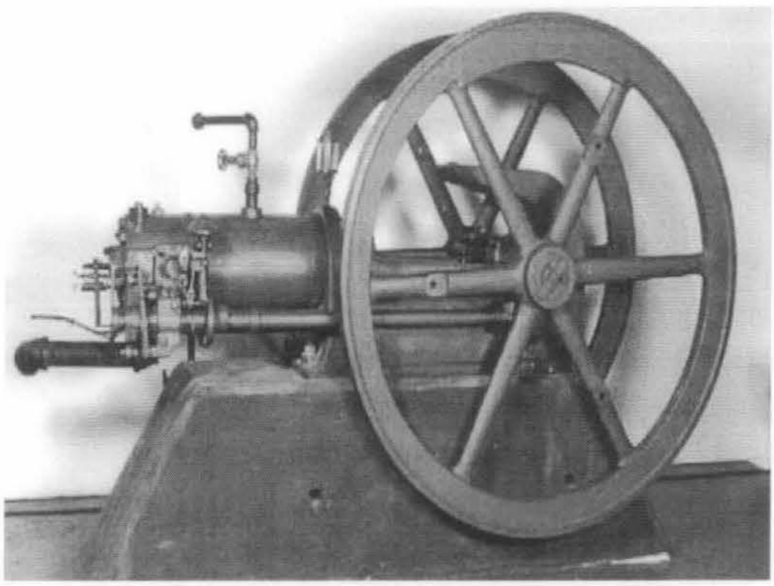


Figure 2-1 Otto Engine 1886
Source: www.enginelogics.com

Since the invention and development of internal combustion engines, its widespread utilization in the industry and automotives had brought up energy and environmental pollution issues. Rapid technological developments have taken place to improve the engine's efficiency and to reduce exhaust emissions. Although gasoline and diesel were the fuel of the choice for the last century, alternatives to these two are also being considered to reduce the dependency on the hydrocarbon fuels. Alternatives like CNG, LNG, hydrogen, methanol are being considered. These alternatives are expected to solve the problems of emissions as well as fuel depletion issues [1].

Technological improvements also occurred in the design and operations of the IC engines. Some of these developments include EFI (Electronic fuel injection), DI (Direct injection), Variable valve timing (VVT), variable compression ratio, catalytic converters, onboard diagnosis etc. Each of these developments contributed in the advancement of engine technology. Spark ignition engines performance has increased considerably with the reduction of NOx up to 80% lesser than compression ignition, but CO and CO_2 emission for SI engine is higher than compression ignition (CI) engine [2,10].

2.2 Compressed Natural Gas as an Automotive Fuel

Natural gas (NG) is a mixture of hydrocarbons, mainly methane (CH4) with minor amounts of ethane, propane, butane and pentane. It is produced either from gas wells or in conjunction with crude oil production. The use of natural gas in engine is not new; Lenoir in 1860 developed and built an engine which used natural gas as fuel. With the discovery for gasoline and diesel as fuels and the related technological developments phased out natural gas as a fuel for Automobiles. Due to the lower energy density of natural gas, it can only be used either in compressed or liquefied form. The actual use of CNG (Compressed Natural Gas) is attributed to the oil shortages and price hikes of 1970's, but fell into decline after petrol prices receded, and there was excessive supply.

Environmental pollution and global warming are some of the main concerns for the developed countries and according to one study automotives being one of the major sources for these problems. Increasing environmental concern and the dependency on the crude oil are the major boosts for the automotive industry to look for the alternative fuels to overcome these problems. Natural gas is seen as one of the potential alternatives, due to its lower cost, clean burning and longer availability compared to crude oil. Exhaust emissions from engines such as NOx, CO, CO_2, HC and particulate matter are the main contributors of environmental pollution. These emissions are responsible for smog forming, acid rain and an increased level of ground level ozone, thus affecting the human life as well as the ecosystem.

Although natural gas is a hydrocarbon fuel, its combustion is considered cleaner compared to gasoline and diesel. It is primarily composed of methane (CH_4) and has a high H/C ratio. The emissions from burning natural gas show reduction in the formation of CO and CO_2 [10, 15] compared to gasoline. The NOx formation from natural gas is expected to be lower [16] but some experimental results showed that it is relatively higher than the conventional gasoline engine [17, 18]. Hydrocarbon emissions are lower due to higher H/C ratio but the emissions in the form of methane are higher compared to gasoline combustion. Table 2-1 shows the combustion related properties for gasoline and CNG.

Table 2-1 Combustion Related Properties of Gasoline & CNG [14]

Properties	**Gasoline**	**CNG**
Motor octane number	80–90	130
Molar mass (kg/mol)	110	16.04
Carbon weight fraction (mass%)	87	75
$(A/F)_s$	14.6	16.79
Stoichiometric mixture density (kg/m^3)	1.38	1.24
Lower heating value (MJ/kg)	43.6	47.377
Lower heating value of stoic. mixture (MJ/kg)	2.83	2.72
Flammability limits (vol% in air)	1.3–7.1	5–15
Spontaneous ignition temperature (°C)	480–550	645

$(A/F)_s$=Stoichiometric air fuel ratio.

CNG burns more completely than gasoline and has a high octane rating of 130 (compared with 87 for regular unleaded gasoline). Hence higher compression ratios can be used to increase the thermal efficiency of the CNG fueled engine. Flammability limit of CNG is wider than gasoline, hence very lean operations can be possible [23, 24] hence improving fuel economy. Along with its clean burning and fuel efficiency CNG also has an advantage of reduced wear and maintenance and reduced oil contamination. Its operation is quite and it has almost zero levels of sulphur, thus, negligible sulphate emissions. Also its performance is better than gasoline-powered vehicles under cold-start conditions.

CNG use in internal combustion engine also possesses some disadvantages. The prime disadvantage being the loss of power, where studies reported about 10% reduction in power compared to gasoline engines [25-27], reason being the gaseous state of CNG that decreases the engine's volumetric efficiency [28]. Secondly, energy density of CNG is lower, and has longer combustion duration [29] hence its driving range is limited. Extra weight of tanks may lead to a higher fuel consumption or loss of pay load. Also its composition varies from region to region hence affecting air/fuel ratios. Conversion of a conventional engine to use natural gas requires some modifications, which includes valve trains, ignition system and fuelling system [26, 30]. On average the advantages of using CNG as fuel overweighs its disadvantages, but still the use of CNG in engine technology is not to its optimum and much of the work is required to see and utilize its full potentials as an alternative to conventional fuels.

2.3 Natural Gas Engine Development

The first use of CNG as automotive fuel was dated back in 1920's where it was used in heavy commercial vehicles with spark ignition engines. Its use as automotive fuel was slowly faced out during 1950's due to the rapid development of both gasoline and diesel technologies, especially the heavy duty vehicles are favoured for the use of compression ignition engines. During the oil shortages of 1970's natural gas once again came in the market due to its easy availability and lower cost. Sufficient oil supplies and regulated prices put natural gas aside as markets favoured the conventional fuels, in the years later. Gasoline and diesel were the fuel of choice for automotives in the last century.

The recent push for the alternatives to oil based fuel is due to increased concerns about environmental pollution and the oil depletion problems. Furthermore, governments and environmental protection authorities are imposing strict regulations on emissions standards. These factors drive the automotive industries to further technological developments along with the alternatives to oil based fuels. CNG is considered one of the potential alternatives in this category. Current problem faced by conventional diesel and gasoline engine are the exhaust emissions, consisting of CO_2, NOx, CO and hydrocarbons. The availability of natural gas and its clean burning are the key factors for the development of natural gas engines. Technological advancements in the form of optimizing power, efficiency and reduced emissions are continuously being looked upon [26-32]. Improvements in the fuel delivery system, combustion strategy, and engine formats are some of the innovative technological developments in CNG fuelled engines.

2.3.1 Fuel Delivery System

Fuel delivery system consists of the necessary equipment required to carry fuel from the tank to the engine. Fuel delivery system ensures the precise metering of fuel and air required by the engine. Along with the fuel supply, mixing control of fuel and air are important factors affected by the fuel delivery system. Hence much of the emphasis is given to the development of efficient fuel delivery systems for precise and accurate control of these parameters. Over the years of research and development, these systems evolved from a conventional carburettor system to a highly sophisticated direct injection systems. Mixture quality and accurate fuel supply plays a vital role in the stable combustion process. Each method has its advantages and drawbacks, which would be discussed in the next session. The delivery systems are as follows;

- Carburetor
- Electronic Fuel Injection
  - Single port injection
  - Multi port injection
  - Direct injection

2.3.1.1 Carburetor

Carburetors are among the earliest fuel delivery devices and are still in use. Carburetors were the usual fuel delivery methods for almost all engines up until the mid-1980s, but with the development of fuel injection and electronic controls for the precise fuel delivery, these new methods are now preferred. The carburetor works on Bernoulli's principle. Differential pressure occurs during the intake process as the high velocity air passes through the venturi. The reduction in the diameter will reduce the pressure causes a vacuum condition at the throttle. This difference in pressures will draw some fuel from the fuel supply, which instantly mixes with the air. A homogeneous mixture is created which then enters the combustion chamber. Carburetors are relatively simpler in design and operation compared to the new techniques. Therefore fuel delivery control is not that precise. More sophisticated carburetors have also been introduced to enhance the fuel supply and mixture formation; however it has its limitations for these parameters. Schematic diagram for the working principle of a carburetor is given in Figure 2-2.

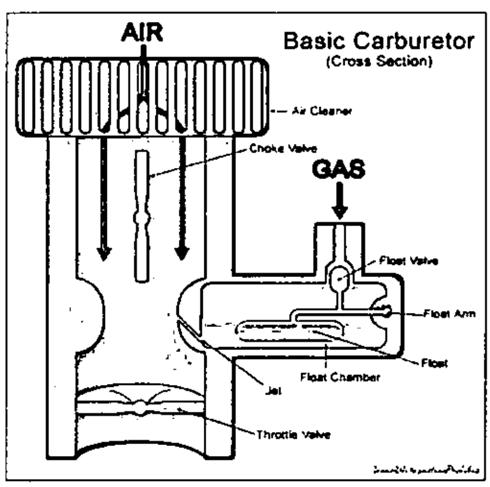


Figure 2-2 Carburetor

Engines using carburetors as fuel delivery system require modification for use of CNG. Studies showed power losses of about 13-30% for carburetted engines using CNG compared to gasoline [33]. The gaseous state of CNG accounts for these losses as it reduces the volumetric efficiency of the engine. AFR controllers and complex systems for carburetors might improve the drops in performance but still it is comparatively lesser than a gasoline engine. The exhaust emissions for CNG are about 90% lesser for CO and 50% for HC emissions respectively [29, 34].

2.3.1.2 Electronic Fuel Injection

Fuel injection system, in an internal-combustion engine delivers fuel or a fuel-air mixture to the cylinders by means of pressure from a pump. It was originally used in diesel engines because of diesel fuel's greater viscosity and the need to overcome the high pressure of the compressed air in the cylinders. For gasoline engines, carburetors were the predominant method to meter fuel before the widespread use of fuel injection. The primary functional difference between carburetors and fuel injection is that fuel injection atomizes the fuel by forcibly pumping it through a small nozzle under high pressure, while a carburetor relies on the vacuum created by intake air rushing through it to add the fuel to the air stream. In gasoline engines the fuel usually is injected into the intake manifold and mixed with air, and the resulting mixture is delivered to the cylinder. Modern fuel injection systems use computers to regulate the process.

Air/fuel ratio control in IC engine is vital for the optimum performance of engine. AFR also effects emissions, driveability and fuel economy. Modern electronic fuel-injection systems meter fuel accurately and precisely, this enables fuel-injected engines to produce less air pollutants than comparable carbureted engines with an increased performance and driveability. Fuel injection systems can react rapidly to changing inputs such as sudden throttle movements, and will control the amount of fuel injected to match the engine's needs across a wide range of operating conditions such as engine load, ambient air temperature, engine temperature, fuel octane level, and altitude (*i.e.*, barometric pressure). Different schemes for fuel injection are briefly explained in the next session.

i. Single point injection / Throttle body fuel injection.

Single point injection (SPI) is an improvement to carburetor systems. Sometimes it is also referred as throttle body injection (TBI). This system uses one or two injector valves mounted in a throttle body assembly. SPI can be considered as a carburetor derivative. The injectors spray fuel into the top of the throttle body where it mixes with air. Unlike carburetors where fuel is sucked due to vaccum pressure, the injector forces the fuel into the air which is taken to the engine through intake manifold. The throttle body injection assembly typically consists of the following: throttle body housing, fuel injectors, fuel pressure regulator, fuel filter, throttle positioner, throttle position sensor, and throttle plates. Power to the system is provided through the power relay. This system controls the amount of fuel more precisely as required by the combustion process. It also helps in decreasing the fuel consumption and emissions reduction.

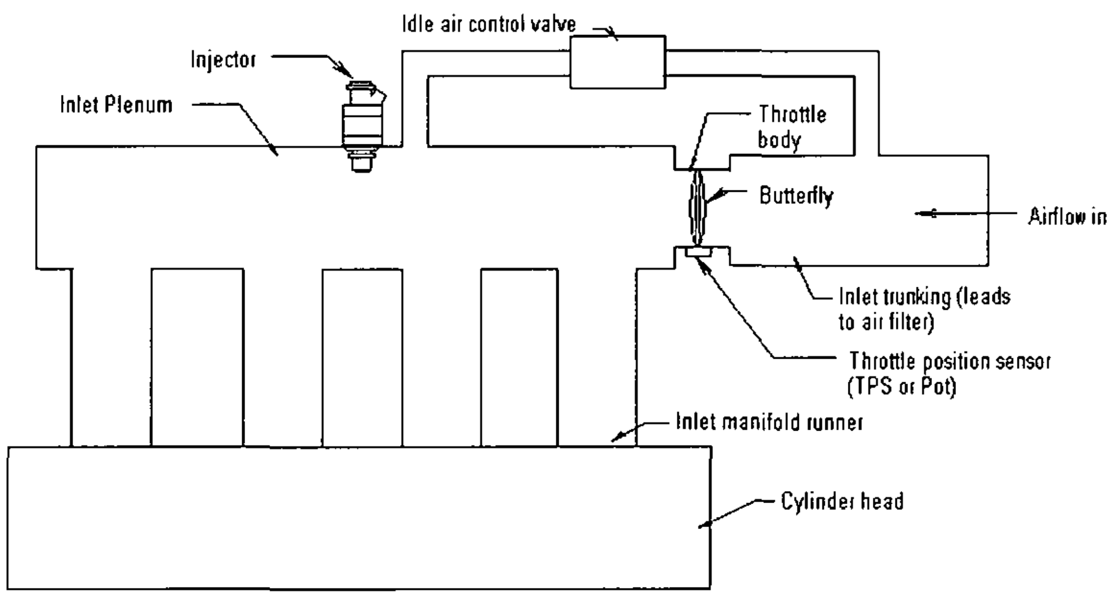


Figure 2-3 Schematic of Single Point Injection with Plenum
Source: http:www.enginelogics.com

SPI system if used for CNG would increase the engine performance compared to carburetor due to precise control over air/fuel ratio [33], though the overall performance in terms of horse power is still less compared to gasoline. Figure 2-3 shows a schematic diagram for single point injection. Although the injector position is shown in the centre of the plenum, this is just for clarity; usually the injector will be mounted on or near the throttle body where air velocity is at its highest.

CNG engines if incorporated with SPI system would have 13.8% lower IMEP at rated speeds in comparison with the gasoline. CNG due to its gaseous state have very small effect on cooling the intake charge, therefore the inlet temperature is higher and thus reduced volumetric efficiency [35].

ii. Multi Port Injection (MPI)

Unlike single point injection, where there is usually one injector, multi point injection employs a dedicated injector for each cylinder. The injectors are either attached to the intake manifold or on the cylinder head, and both aimed at the intake valve. The majority of the MPI systems employ injectors attached to the intake manifold. Mostly, MPI systems in gasoline engines inject fuel while the intake valve is closed to create a fuel/air mixture before the induction of the charge into the cylinder. This time duration between the injection and the consequent induction is known as time lag. This time lag necessitates a homogeneous mixture preparation before it is injected into the combustion chamber [36]. In CNG engines the time lag factor is eliminated, as the fuel is already in gaseous state, thus the injection occurs while the intake valve is open [37, 38].

Figure 2-4 shows a schematic diagram for MPI system. The major parts of MPI system consists of fuel filter, fuel pump, fuel rail, injectors and engine control unit (ECU). The system components are quite similar to that of SPI, but in SPI there is only one injector and no fuel rail. MPI configuration gives much better control of fuelling and better emissions since the fuel can be metered more closely, and there is less opportunity for the fuel spray to condense or drop out of the airflow since it is introduced as a separate small stream for each cylinder rather than one large one in SPI.

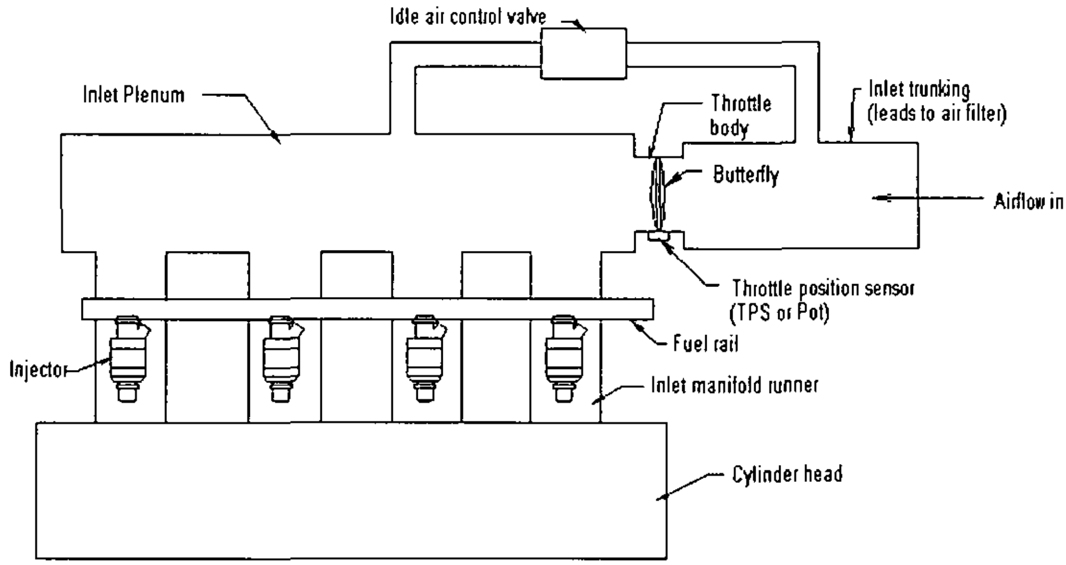


Figure 2-4 Schematic of Multi Port Injection (MPI)
Source: http:www.enginelogics.com

The closer to the inlet valve the fuel injection takes place, the better the economy and transient throttle. MPI systems use more sophisticated system compared to SPI system, since the ECU monitors and controls the injection parameters, such as injection timing, injection duration and the amount of fuel required by each cylinder [40, 41]. Moreover, MPI usage enhances fuel economy and increases performance by controlling the combustion process [37]. It also helps in decreasing the engine out emissions in terms of CO and HC emissions compared to carburetor.

Engines employing MPI for CNG use have a 3% to 5% better performance as compared to SPI-CNG systems, and have 10.4% lower IMEP values for rated speeds compared to MPI-gasoline [35].

iii. Direct Injection (DI) System

The next development to MPI and the more recent one is the direct injection fuel system for spark ignited engines. Direct injection is basically used in compression ignition engines where the fuel is directly injected into the combustion chamber at high pressure. Multi-point injection (MPI), where the fuel is injected through each intake port, is currently one of the most widely used systems. Although MPI provides a drastic improvement in response and combustion quality, it is still limited due to fuel and air mixing prior to entering the cylinder. Researchers were looking into such a development where they can have the features of both gasoline and diesel engines combined i.e. high power output and lower brake specific fuel consumption. Direct injection system for spark ignited engines turned out to be one of the potential solutions. Direct injection systems are engineered to inject the fuel directly into the cylinder in a manner similar to diesel direct injection engines. The load is controlled by varying the amount of fuel being injected, instead of throttling the mixture in the conventional scenario. Direct injection is designed to allow greater control and precision, resulting in better fuel economy. Direct injection is also designed to allow higher compression ratios, delivering higher performance with lower fuel consumption.

Direct injection system for spark ignition engine consists of almost the same components as that of MPI, which are high pressure fuel pump, fuel filter, fuel rail, injectors and ECU. Unlike MPI, the fuel is injected directly into the cylinder at an even higher pressure. ECU controls the amount of fuel injected and injection timing.

Advantages of direct injection compared to MPI system, are higher fuel economy, drive-ability, AFR controllability, combustion stability, better emissions, and about 10-20% increase in engine efficiency. Injector's performance, durability and relatively higher NOx are some of the limitations with direct injection systems [43]. Figure 2-5 represents a direct injection system developed for gasoline direct injection by BMW motors Germany.

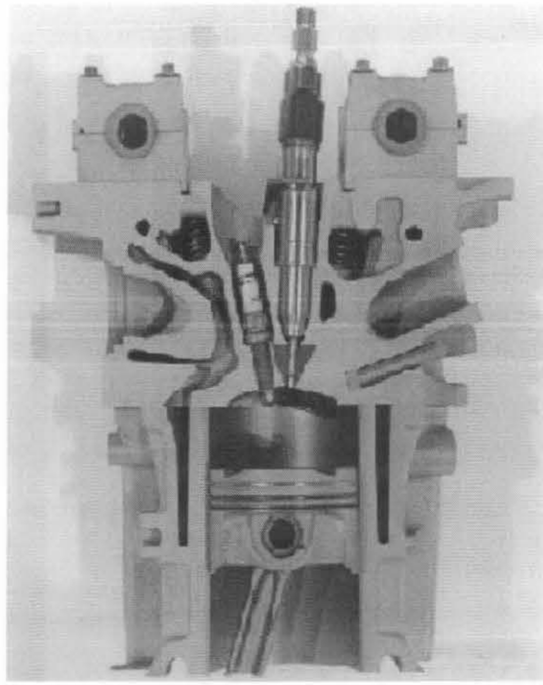


Figure 2-5 Direct Injection Systems
Source: www.bmw.com

CNG fueled engine using direct injection system offers advantages in controlling AFR more precisely, hence improving fuel economy and effecting the exhaust emissions [17]. Lean operations up to 0.1 equivalence ratio are possible [44]. Direct injection generates high turbulence inside the combustion chamber, enabling faster combustion rates and lower cycle to cycle variations [45]. However, CNG-DI engine tend to produce relatively higher levels of NO_x emission due to higher combustion temperature resulted from increased combustion process.

Other researchers compared the direct injection method with mixer system and the results state that the maximum brake power for DI systems is 7.7% higher than that mixer engine, while the thermal efficiency of DI is increased up to 98.8% which resulted decrease in NOx emissions [45,63].

2.3.2 Engine Format

i. Dedicated/Monofuel

An engine that is optimized to use one fuel can be termed as monofuel/dedicated engine. Being optimized to use one fuel this would give maximum performance and lower emissions using that particular fuel. A dedicated engine using CNG has the advantage of being optimized to operate on natural gas thus ensuring maximum efficiency and optimum emission results. The mileage depends upon the CNG volume carried by the vehicle.

ii. Bi-fuel/bivalent

Bi-fuel engine usually has two types of fuels having separated supplies, and the engine is operated on one fuel at a time. An engine which can operate on either natural gas or gasoline is an example of a bi-fueled engine. In case of gasoline-CNG Bi-fuel engine, usually gasoline is used to start the engine, thus a small portion of gasoline is still required. These engines are usually optimized to run on one fuel. The deficiency in running CNG fuel is compensated into the use of a supercharger or turbocharger.

iii. Dual Fuel

A dual-fuel engine utilizes a mixture of natural gas and diesel. Dual-Fuel technology also allows combustion of natural gas at diesel compression levels, so spark plugs and restrictive air throttles are unneeded. The diesel is injected directly into the combustion chamber, while gas is introduced into the air intake by carburetion or by gas injection. Dual-fuel technology results in an engine that provides the low emissions of natural gas, without giving up the performance of a diesel.

The mixture of natural gas and diesel varies according to the load and the duty cycle of the engine, ranging anywhere from 80% gas down to 0% gas. At lower engine loads, diesel use tends to be higher whereas at higher engine loads it is possible to use a higher proportion of gas. Dual-fuel engines are usually the result of a conversion of a diesel engine and have the advantage of not being totally dependent on natural gas for fuel supply, hence can solely operate on diesel as well.

iv. Tri-fuel

Tri-fuel engines combine the features of flex-fuel engine and natural gas. A flexible-fuel engine can use different sources of fuel, either mixed primarily or having separate tanks and fuel systems for each fuel. A common example is an engine which can operate using gasoline mixed with varying levels of ethanol. This is relatively a recent technology and its market is growing. Tri-fuel vehicles first entered the market in 2005 in Brazil, where ethanol and NG are widely used as a transport fuel [46].

2.3.3 Combustion Strategy

Two types of combustion strategies are currently being used with CNG fueled engine, i.e. stoichiometric and lean-burned combustion. Amount of fuel injected to the system are the basis for these strategies. Each of these operations has their advantages and short comings when used with CNG.

2.3.3.1 Stoichiometric Combustion

Stoichiometric or Theoretical Combustion is the ideal combustion process during which there is just exact amounts of oxidizer (usually air) present required to completely oxidize the fuel, thus achieving a complete combustion. Every fuel has its specific air-fuel ratio for stoichiometric combustion. CNG which is primarily composed of methane had a stoichiometric air-fuel ratio around 16.9. The flame propagation for stoichiometric mixture of a typical hydrocarbon fuel with air is slow. Also heat produced by combustion is lower. Since the heat produced is lower and CO and HC emissions are comparatively higher, SI engines employing EGR and catalytic converters give the best performance under stoichiometric conditions. The reason being the efficiency of the catalytic converters is maximum at stoichiometric conditions. Exhaust emissions in terms of NOx and HC are reported to be reduced by 90% using EGR and three-way catalytic converter in stoichiometric gasoline operations [17]. Stoichiometric mixtures in CNG have slower burning rate and longer ignition delay compared to gasoline [47]. Also BSFC (Brake Specific Fuel Consumption) and BMEP (Brake Mean Effective Pressure) for CNG are down to 20% and 15% respectively [18, 48].

Performance of CNG fueled engine can be improved by using feedback control system for regulating air/fuel ratio. This system employs an oxygen sensor in the exhaust manifold to determine the AFR and thus is controlled by the ECU. AFR and combustion stability considerably effects the exhaust emissions [49]. The concentration of all three potential pollutants is high near stoichiometric conditions. The catalytic converter works best when the concentration of these pollutants is high, in other words the engine works best at stoichiometric mixtures.

2.3.3.2 Lean Combustion

When there is more air then stoichiometric maximum (for stoichiometric combustion) in a fuel-air mixture, the mixture is lean and the combustion is termed as Lean-burned combustion. The additional air is known as excess air. In other words there is more oxidizer than is required for the complete combustion. The potential advantage of lean burn combustion is higher thermal efficiency, lower HC and CO emissions and increased fuel economy compared to the engines operating with stoichiometric mixtures. However, NOx emission is slightly higher due to increased combustion temperatures, and the rate of cyclic variation is also high due to the variation of mixture properties in each engine cycle.

Lean burn combustion with CNG offers low fuel consumption with capability to operate under a lean limit approaching 0.02 equivalence ratio due to its wide flammability range, which is much lower than the limit of gasoline direct injection engine i.e. 0.3. In direct injection CNG the combustion rate is faster and shorter with high NOx and lower CO emissions over a wide range of equivalence ratio [50]. Turbo charged lean burn CNG engine happens to produce more power than stoichiometric engine with a higher NOx and HC emissions [51]. Cyclic variation is one of the main problems with lean burn combustion systems, as the mixture properties vary in every cycle. Controlling the mixture formation in such systems influences to reduce cyclic variability hence resulting in a stable combustion process [52].

The application of lean burn engine is in the range of medium to heavy duty application, in which low fuel consumption and reducing emission without after treatment technology are the main concern of its development [39].

2.4 CNG Direct Injection Engine

CNG use in engines with conventional induction systems has a power drop of about 10%. Direct injection technology is adopted with CNG to compensate the power losses that are mainly due to reduction in the volumetric efficiency. Moreover direct injection is expected to increase the fuel efficiency and the power output. This is achieved by the precise control over amount of fuel and injection timings which can be varied according to the load conditions. In addition, there are no throttling losses when compared to a conventional fuel injected or carbureted engine, which greatly improves efficiency. Studies showed that direct injection systems give higher IMEP values than MPI systems, mainly because of the increased volumetric efficiency [45, 52]. Higher knock resistance and extreme stratification capabilities under leaner conditions are the key combustion properties which make it a preferred fuel over the oil based fuels. Thus higher compression ratios can be used to increase the thermal efficiency, and leaner operations can increase the fuel economy.

Emission reduction along with the performance is yet another motive for the automotive manufactures to develop new technologies with having the advantage of both high performance and lower emissions. Energy sources having advantage in both these aspects are of high interest. CNG is considered one of the cleanest fuels due to its clean burning. Where CNG-DI is having the advantage of increased power and efficiency, NOx and HC emissions are comparatively higher than MPI system near stoichiometric conditions [54]. Controlled combustion at different conditions is needed. Mixture formation plays an important role in the combustion control. Injection parameters play an important role in mixture formation. Injection parameters include injection timing, injection duration, injection pressure and injection spray angle. Precise control of these parameters can improve the engine performance and reduced exhaust emissions. Two types of mixture formation methods are generally utilized, homogeneous and stratified [36, 43, 55]. A brief discussion on these two methods is given in the next section.

2.4.1 Homogeneous Charge Operation

Homogeneous charge combustion is a strategy in which fuel and oxidizer are well mixed before the start of combustion. In direct injection systems this configuration is achieved by early injection of the fuel, so that there is significantly enough time available for the fuel and the air to mix extensively creating a homogeneous mixture.

Homogeneous charge operations on gasoline are reported to produce lower emissions at stoichiometric conditions, reason being the efficient three way catalysts operation at stoichiometric conditions. With direct injection homogeneous systems offer more advantages in terms of emissions and performance. Fuel consumption is considered as a disadvantage with homogeneous operations which can be addressed with the implementation of direct injection systems. Stratified operation is considered efficient compared to homogeneous operation in terms of fuel consumption [70].

Combustion efficiency of homogeneous charge is higher for CNG direct injection at stoichiometric conditions. HC and CO emissions are lower but NOx emissions are comparatively higher than stratified charge operation. Higher combustion temperatures at stoichiometric conditions of homogeneous charge are reported to be the main reason [44, 50]. Homogeneous charge operations are investigated on CNG-DI using rapid compression machine (RCM) [24]. The experiments showed that at stoichiometric conditions the combustion efficiency of homogeneous charge is higher than stratified charge. At leaner operations stratified charge operation showed higher combustion efficiencies.

Direct injection systems on CNG prove to be beneficial in the enhancement of combustion process. Longer combustion duration and higher fuel consumption of homogeneous charge are the obstructions that need to be addressed. These are the main reasons of rather fewer studies could be found on homogeneous charge CNG-DI operation [24, 50]. Catalytic converters tend to favour homogeneous systems by reducing the engine emissions compared to stratified system. Yet NOx and methane emissions still needed to be reduced. Direct injection systems produce higher momentum and higher turbulence since they are using higher pressures. Pressure difference between the ambient and the injection are the determining parameters for momentum generation. Higher the pressure difference, higher the momentum generation and the turbulence [36]. Turbulence is considered to be a key factor for homogeneous charge formation, hence improving the combustion speed. Injection parameters influence on the turbulence intensity is confirmed experimentally on gasoline direct injection engine. It is reported that the mixing process is mainly influenced by injection parameters at lower engine speeds [71]. Homogeneous charge operating method is designed to meet the requirement of medium-high engine loads.

2.4.2 Stratified Charge Operation

Stratified charge is the name given to the layering of fuel/air mixture inside the combustion chamber, where rich mixture is near the spark plug and a leaner layer in the rest of the chamber. The spark ignites the richer mixture within its periphery, developing a flame which propagates in the rest of the chamber where mixture is considerably leaner.

The primary drive towards the development of stratified technology was fuel economy and engine emissions. With stratified technology ultra lean operations are possible. Honda's CVCC (Compound Vortex Controlled Combustion) in 1970's is a form of stratified engine operating on gasoline. This engine met the United States emissions standards without a catalytic converter and with an increased fuel economy. The CVCC system had conventional inlet and exhaust valves and a third, supplementary, inlet valve that charged an area around the spark plug. The spark plug and CVCC inlet was isolated from the main cylinder by a perforated metal plate. At ignition a series of flame fronts shot into the very lean main charge, through the perforations, ensuring complete ignition. CVCC technology used carburetor for the mixture preparation. With the development of sophisticated injection systems, MPI and DI for gasoline engine, the mixture formation can be controlled more precisely. The performance and fuel consumption is much improved with the implementation of these technologies. Stratified charge operation was favoured for idle-medium engine load and engine speeds in gasoline direct injection engine in order to achieve maximum fuel economy.

Direct Injection Technology for CNG engine is being studied for the evaluation and optimization of its performance. Mixture formation is an important parameter for the quality of combustion, as mentioned before. Mixture formation is influenced by the time available for mixing. It is well known that the rate of combustion in spark ignition engines is influenced by turbulence and charge motion [56, 57]. The combustion and emission parameters are not significantly influenced by the modes of fuel injection such as the number of injectors, position of the injectors, and the arrangement of the injectors with regard to the spark location. Single-injection gives higher level of mixture stratification than the twin injection modes. Moreover pressure rise due to combustion for direct injection is higher than homogeneous/port injection [44].

The operating range of lean-burn SI engines is limited by the level of cyclic variability in the early development of the flame, typically corresponding to 0–5% of the mass having burned [58, 59]. Charge stratification in lean burn engines is an important parameter for combustion control. Injection parameters have a significant influence on charge stratification. Controllable charge stratification can make the combustion process better. Study on the possibility of controlling the charge stratification revealed that CNG-DI stratified combustions systems can realize overall shorter combustion compared to homogeneous system. And the combustion efficiency can be maintained more than 0.92 in the range of equivalence ratio from 0.1 to 0.9 [24].

Combustion characteristics of CNG have been investigated on various injection timings. Fuel injection timing had a large influence on the engine performance, combustion and emissions and these influences became largely in the case of late injection cases. Advancement of injection timing decreases the volumetric efficiency and an overall increase in equivalence ratio. Wide operational range of equivalence ratio and high-energy conversion efficiency are the basic thoughts for CNG-DI implementation on spark ignition engine. Combustion duration for stratified system is shorter than homogeneous system which is due to the decrease in the time interval between injection timing and ignition timing. Very late injection timing would have insufficient time for the fuel–air mixing of the late part of the injected fuel, bringing poor quality of mixture formation and subsequently resulting in the slow combustion rate, the long combustion duration and high HC concentration. However, early injection gave a slight influence on both engine combustion and emissions [61]. Stratified combustion strategy has been investigated using rapid compression machine with a compression ratio of 10. Improved results for the emissions of CO and CO_2 are reported, while higher heat release rate and high peak pressures lead to high levels of NOx, compared to homogeneous charge operation. Heat release pattern of late injection showed a faster burn in the initial stage and a slower burn in the late stage [60].

CNG-DI engine gives high combustion stability with less cycle-by-cycle variation over a wide range of equivalence ratios. Early injection leads to a longer duration of the initial combustion, whereas late injection leads to a longer duration of the late combustion. NOx emissions and CO emissions show interdependence with the variation of rapid combustion. Combustion efficiency was maintained at a high value over a wide range of equivalence ratio [62].

Injection parameters such as injection timing [54, 61-67], injection pressure [36, 43, 55, 63, 68] and type of injectors [36, 68, 69] had proven to be important parameters that affect the combustion, emissions and performance of direct injection engine. Therefore, the research will be focused to analyze the effect of these parameters on CNG-DI engine while maintaining lean burn stratified charge conditions.

CHAPTER 3

THEORETICAL BACKGROUND

3.1 Thermodynamic Cycles

To understand the operation of an internal combustion engine it is necessary to understand the thermodynamic cyclic process undergone during an engine operation. Engine operations are modeled as ideal air standard cycles based on ideal gases undergoing ideal processes in thermodynamics. However, the actual cycle that an internal combustion engine follows is mechanical cycle. But it is very convenient to consider internal combustion engine with ideal air standard cycle for comparison. Despite the fact that in actual cycle the air and fuel mixture does not behave as an ideal gas, air standard cycles are yet a good approximation to understand the basic concepts.

Commonly three types of air standard cycles which represent internal combustion engine procedure, namely

- Otto cycle or constant-volume cycle
- Diesel cycle or constant-pressure cycle
- Dual cycle or limited pressure cycle

i. Diesel Cycle

The diesel cycle or compression ignition cycle is a thermodynamic cycle that approximates the cycle followed by a diesel engine. Diesel engine differs from the gasoline powered Otto cycle by using a higher compression of the air to ignite the fuel rather than using a spark plug ("compression ignition" rather than "spark ignition"). Figure 3-1 shows a standard air diesel cycle. Following are the processes followed by a four stroke diesel cycle

1-2 Adiabatic compression

2-3 Constant pressure heat addition

3-4 Adiabatic expansion

4-1 Constant volume heat rejection

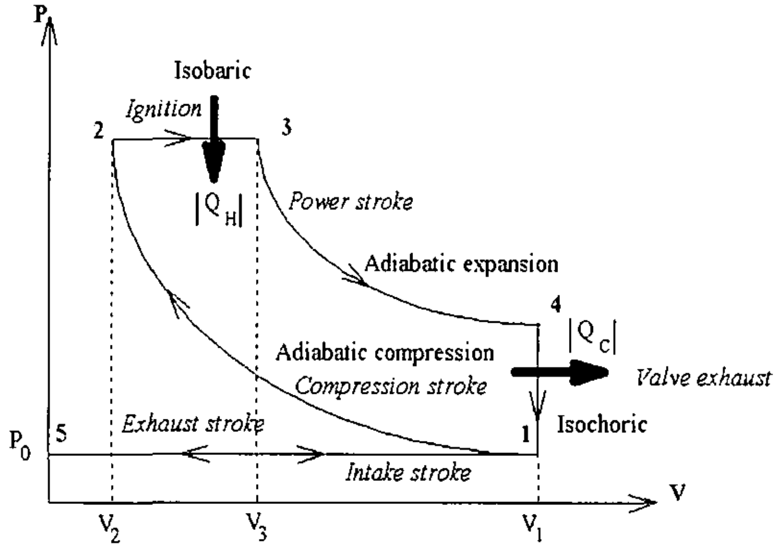


Figure 3-1 PV-Diagram of Air Standard Diesel Cycle

ii. Otto Cycle

The air standard Otto cycle is the theoretical cycle commonly used to represent the processes in the spark ignition (SI) internal combustion engine. It is assumed that a fixed mass of working fluid is confined in the cylinder by a piston that moves from BDC to TDC and back, as shown in Figure 3-2. The cycle consists of isentropic compression of an air-fuel mixture from state 1 to state 2, constant-volume combustion to state 3, isentropic expansion of the combustion gases to state 4, and a constant-volume heat rejection back to state 1, to complete the cycle.

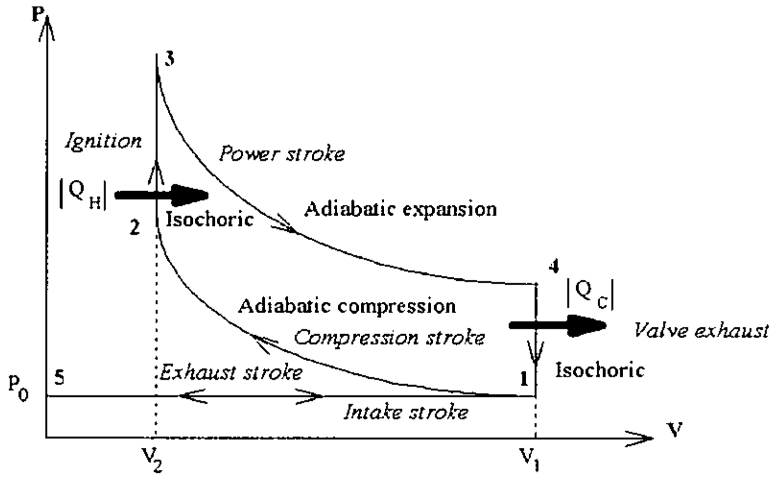


Figure 3-2 PV-Diagram of Air Standard Otto Cycle

iii. Dual Cycle

In dual cycle all the other processes are same as diesel and Otto cycles, except heat addition which partly occurs at constant volume and partly at constant pressure as shown in Figure 3-3.

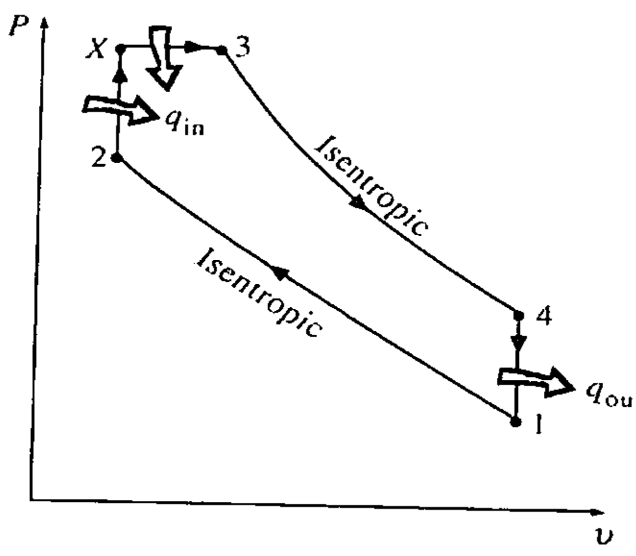


Figure 3-3 PV-Diagram of Dual Cycle

3.2 Actual Cycle in IC Engines

The actual cycle on which internal combustion engines operate is called a mechanical cycle, not a thermodynamic air standard cycle. But still it is quite convenient to analyze the engine on these hypothetical cycles. The actual cycle in internal combustion engine is shown in the pressure volume diagram in Figure 3-4.

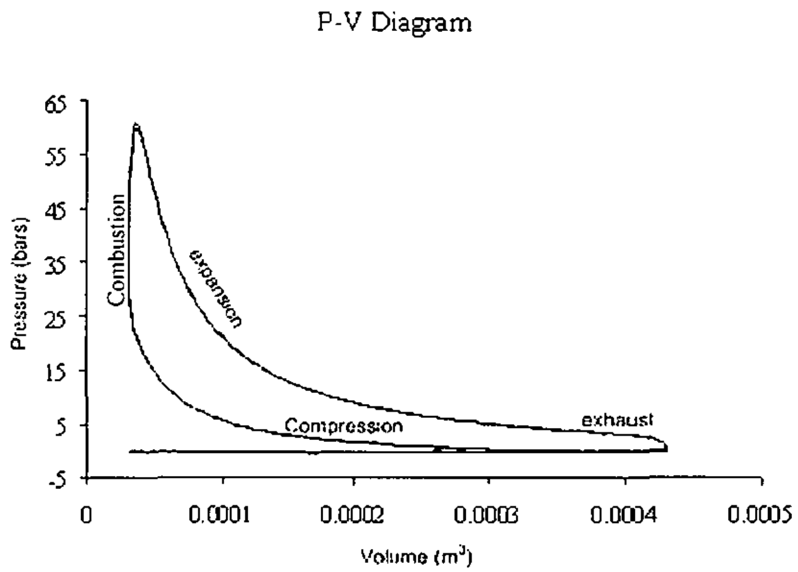


Figure 3-4 Mechanical Cycle of Internal Combustion Engine

Figure 3-4 shows that unlike the standard air cycles the combustion process occurs neither at constant volume nor at constant pressure. Thermodynamic cycles do not account for the heat lost to piston, cylinder walls and to cylinder head during combustion. Also it assumes as combustion is completed during the constant volume heat addition, but actually combustion continues into the expansion stroke. Heat transfer takes place and energy is lost to cylinder walls, head and piston. The mechanical cycle accounts for all these aspects and hence the engine performance is dependent on parameters such as valve opening/closing, air/fuel ratio, ignition, and injection timing etc.

3.2.1 Comparison between Otto Cycle and Mechanical Cycle

For simplicity, the mechanical cycle of a spark ignition engine can be idealized as constant volume combustion cycle or Otto cycle. But in reality, the constant-volume heat transfer process at TDC in the Otto cycle is an artifice to avoid the difficulties of modeling the complex processes that take place in the combustion chamber of the SI engine. These idealized conditions are widely used as a basis for simplest computer model. In comparison, Otto cycle assumes that:

Air behaves as an ideal gas with constant specific heats and the processes are reversible.
It doesn't account for induction or exhaust process, but a fixed quantity of air in the cylinder with no leakage i.e.

- Heat addition is from external sources at constant volume.
- Heat rejected to the environment to complete the cycle.

In an actual cycle normally, the mixture in the combustion chamber must have an air-fuel ratio in the neighborhood of the stoichiometric value for satisfactory combustion. Also the peak pressures as predicted by the Otto cycle are higher than in the actual cycle. Otto cycle assumes fixed air quantity inside the cylinder whereas piston ring blow-by occurs, as well as blowback and blowdown begins with the opening of the exhaust valve. Expansion process is not isentropic as heat is lost to the piston, cylinder walls, cylinder head and coolant. Due to entropy degradation some useful work is lost which is called as "lost work". The area difference between theoretical and actual cycle in Figure 3-5 shows the lost work. The lost work is mainly attributed by the followings;

- heat loss
- mass loss
- finite burn rate and
- finite blow-down rate

These losses result in the actual efficiency being less than that of the equivalent fuel-air cycle by a factor ranging from 0.8 – 0.9 [75].

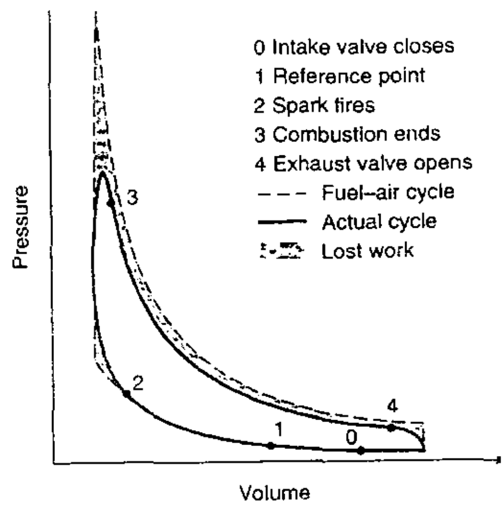


Figure 3-5 Comparison of an Actual Cycle with its Equivalent Air Cycle [75]

3.3 Basic Calculations of Performance Parameters for Internal Combustion Engines

The engine performance parameters of interest are power, torque, mean effective pressure and specific fuel consumption. The engine indicator diagrams provide important information for the engine performance. Further details of these parameters are presented in the next section.

(a) Torque (Nm)

Torque (or often called a moment) can be thought of as "rotational force" or "angular force" which causes a change in rotational motion. When torque is applied it changes an object's rate of rotational motion. The magnitude of the torque is measured by multiplying the perpendicular component of the force applied by the distance between the object's axis of rotation and the point where the force is applied.

Torque = Force x Distance

Torque is the measure of an engine's ability to do work. Dynamometer has been used to measure torque output of the engine tested. Schematic diagram of a dynamometer is shown in the Figure 3-6 The engine is clamped to a test bed and the shaft is connected to the dynamometer. The rotor is coupled to a stator. The torque exerted on the stator with the rotor turning is measured by balancing the stator with weights, spring or pneumatic means.

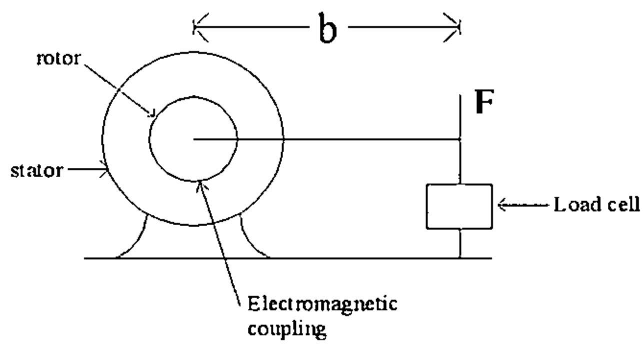


Figure 3-6 Schematic of Dynamometer Operation

Hence torque (T) exerted by the engine is given as:

$$T = b \bullet F \qquad 3\text{-}1$$

The SI units of torque is Newton meter (Nm)

(b) Power (W)

Power can be defined as the amount of work done per unit time.

$$Power(W) = \frac{Work(J)}{Time(s)}$$

The SI unit for power is watt (W).

Power in internal combustion engine can be defined as rate of work done by the engine and it can be derived from torque by using equation:

$$P = \frac{2\pi NT}{60} \qquad 3\text{-}2$$

where: P = power output (W)

N = Engine speed (RPM)

T = Engine Torque (Nm)

(c) Brake Specific Fuel Consumption (BSFC)

Brake specific fuel consumption (BSFC) is a measure of an engine's efficiency. It is the rate of fuel consumption divided by the rate of power production. BSFC has an inverse relation with the engine efficiency; therefore lower values of BSFC are preferred. The new engine research and development is mainly focused towards fuel economy and emissions reduction. BSFC shows how efficiently engine is consuming fuel. The relationship between fuel consumption and power is:

$$BSFC = \frac{FC \cdot 3600}{P} \qquad (\text{ g/kW.h }) \qquad 3\text{-}3$$

where: FC = fuel consumption (g/s)

P = Engine power output (kWh)

(d) Brake Mean Effective Pressure (BMEP)

Brake mean effective pressure (BMEP) is yet another parameter to analyze the engine performance. It is the ratio of the work per cycle to the volume displaced per cycle.

Though the units of BMEP are same as that of pressure, yet it is the measure work output from an engine.

$$BMEP = \frac{Pn}{V_s N} \qquad 3\text{-}4$$

BMEP can also be expressed in terms of engine torque

$$BMEP = \frac{2\pi.2T}{V_s} \qquad 3\text{-}5$$

where: T = engine torque (Nm)

V_s = cylinder Swept Volume (m^3)

n = number of crank revolutions for each power stroke

P = Power (kW)

N = Revolutions per minute (RPM)

(e) Air-Fuel Ratio (AFR)

Air-fuel ratio (AFR) is the mass ratio of air to fuel present during combustion. Each fuel has a specific air-fuel ratio for complete combustion. When there is enough free oxygen in the air to combine completely with the fuel in the combustion chamber, the mixture is chemically balanced and this AFR is called as stoichiometric mixture. Air fuel ratio is an important factor in the combustion process of the engine. Mixtures having AFR less than that of stoichiometric are called rich mixtures. And if there is excess air, means AFR is more than stoichiometric the mixture is called as lean. Both rich and lean mixtures have distinctive effects on the performance of engine. Stoichiometric air fuel ratios can be calculated by using chemical equations which employ the law of definite proportions, for example, the balanced equation for CH_4 with air is:

$$CH_4 + 2(O_2 + 3.76N_2) \Leftrightarrow CO_2 + 2H_2O + 2(3.76N_2) \qquad 3\text{-}6$$

Hence, AFR stoichiometric is:

$$AFR_{stoich} = 2*(1+3.76)*28.9 \Big/ (1*12)+(4*1) = 17.19 \qquad 3\text{-}7$$

(f) Volumetric Efficiency

The effectiveness of the engine's induction system is the measure of its volumetric efficiency. The intake system comprises of air filters, intake runner, intake port and intake valves. The amount of air that enters the combustion chamber depends on the efficiency of the whole intake systems. To get into the engine cylinder air had to pass through these components. During the induction process there are losses that can be observed by calculating the volumetric efficiency of the engine. Volumetric efficiency can be defined as the comparison of actual air mass that enters the chamber to the total displacement volume of specific engine.

$$\eta_v = \frac{2(\dot{m}_a + \dot{m}_f)}{\rho_o V_d N} \qquad 3\text{-}8$$

where: η_v = Volumetric Efficiency

$\dot{m}_a$ = Actual air mass that enters the combustion chamber (g)

$\dot{m}_f$ = Actual fuel mass (g)

V_d = Displacement volume (m^3)

ρ_o = Air intake density (kg/m^3)

In the equation, $\dot{m}_f$ is the mass of fuel inducted. For a direct injection system with injection after intake valve closed (IVC), $\dot{m}_f$ = 0. There are many factors which determine the torque an engine can produce and the RPM at which the maximum torque occurs, but the fundamental determinant is the mass of air the engine can ingest into the cylinders, and there is a nearly-linear relationship between volumetric efficiency and maximum torque. High values of volumetric efficiency of the engine are desired.

Alternatives to conventional induction system have been proposed to increase the volumetric efficiency of the engine. Turbo/super charger, and variable valve timing (VVT) are the most popular CNG engine technologies that have been implemented to new vehicle systems.

(f) Injector Flow Rate

Injector flow rate was estimated by the estimation of fuel velocity at the exit of the injector orifice based on pressure difference between the fuel supplied pressure and the ambient pressure in the combustion chamber. The equation to estimate the velocities is as follows [64]:

$$V_{CNG} = \sqrt{2\frac{k-1}{k}\frac{P_{CNG}}{\rho_{CNG}}\left(1-\frac{P_{ambient}}{P_{CNG}}\right)^{(k-1)/k}} \qquad 3\text{-}9$$

where: V_{CNG} = Injection velocity of CNG (m/s)

P_{CNG} = Injection pressure of CNG fuel (Pa)

ρ_{CNG} = CNG density (kg/m^3)

$P_{ambient}$ = Atmospheric pressure (Pa)

k = Polytropic index (k = 1.33)

3.4 Combustion in Internal Combustion Engine

Combustion can be defined as an exothermic chemical reaction between a fuel and an oxidant, producing heat or both heat and light in the form of either a glow or flames. In IC engines the combustion of fuel is the source of power to the engine. Engine performance is dependent on the quality of combustion of the charge. Combustion chemistry in internal combustion engine is quite complex and depends upon the type of fuel used. Combustion process in spark ignition and compression ignition engines is very different. Compression ignition engine has separate fuel and air supply and the combustion takes place as they mixed together during the compression stroke. On the other hand, in a conventional spark ignition engine fuel and air mixture are inducted into the cylinder through the intake valve during intake stroke. The mixture is then compressed to raise its pressure and temperature during the compression stroke. During the end of the compression stroke the mixture is ignited by the electric discharge produced by the spark plug to initiate the combustion. This results in a steady rise in the temperature and pressure inside the cylinder, hence burned fuel power is transmitted to the crank shaft through piston, and it is called the power stroke. There are several factors that can affect the combustion in the spark ignition engine. Examples are fuel composition, certain engine design, operating parameters and combustion chamber deposits.

Figure 3-7 represents the combustion process in spark ignition engine that can be divided into 4 distinct phases; (1) spark ignition ; (2) early flame development ; (3) flame propagation ; (4) flame termination.

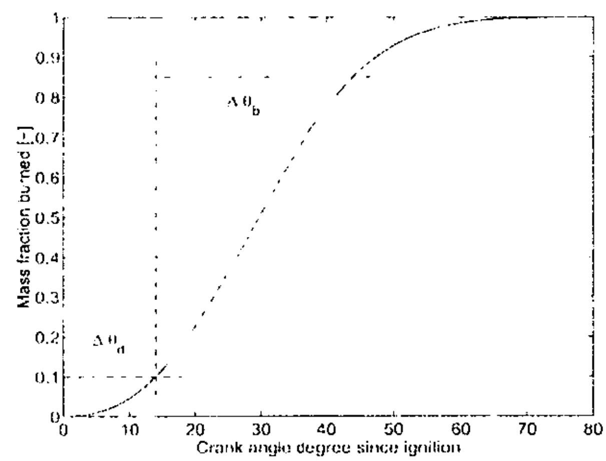


Figure 3-7 Combustion Stages in Spark Ignition Engine [74]

The first stage of combustion corresponds to the initiation of combustion by spark, θ_{ignd}. Combustion ignition vitally depends on the properties of the mixture. The amount of energy required to initiate combustion is called as activation energy. The activation energy in spark ignition engines is mostly provided by the spark plug. Pressure, volume and temperature of the combustion chamber also serve as the source of energy for the combustion initiation. A large amount of energy is provided by the spark plug in a very short duration of time. The combustion is started by the spark plug some 15–40 crankshaft degrees prior to TDC, the point of maximum compression. This ignition advance allows time for the combustion process to develop peak pressures at the ideal time for maximum recovery of work from the expanding gases. This point is typically 14–18 crankshaft degrees ATDC (after top dead center).The time needed from the start of ignition to the start of combustion is called the ignition delay.

The second stage of combustion in SI engines is the development of flame after the charge is being ignited by the spark plug. $\Delta\theta_d$ in Figure 3-7 represents the flame development stage after the initiation of the spark discharge. Early flame develops as the charge near the spark plug gets enough energy from the discharge of the spark plug to be ignited. This mainly depends on the nature of the fuel and the temperature and pressure of air/fuel mixture. Higher fuel availability near the spark plug is important to increase the combustion rate at the initial stage. Injection timing and ignition advance are the key parameters to be controlled in order to get rich condition around the spark for better initiation of combustion process. Combustion chamber shapes, and in-cylinder flow characteristics also have significant effects on the behavior of initial combustion. The initial combustion duration is defined as 0-10% of total combustion duration as described in the Figure 3-7.

After the early development of flame the next stage is flame propagation or rapid combustion. Rapid combustion is defined as 10-90% of combustion duration. Rapid combustion stage is represented by $\Delta\theta_d$ in Figure 3-7. After the early development of flame, the flame grows much faster expanding rapidly across the combustion chamber. This growth is due to the travel of the flame front through the combustible fuel air mixture itself and due to turbulence in the in-cylinder flow. Due to the growth and propagation of the flame across the combustion chamber, there is a steady rise of the pressure. This pressure rise peaks after TDC but still there is charge left in the cylinder to be burned. The pressure continuously decreases during the rest of the expansion stroke. Improving the combustion rate at this stage would result in higher peak pressures and better combustion quality. But higher pressures and temperatures during combustion could cause an immense increase in the emissions of nitric-oxide, a pollutant. Therefore the combustion pressures and temperatures should be controlled in a way to minimize the toxic emissions with as low as possible trade off of the engine performance.

Flame termination is the stage during which the unburned charge left during the rapid combustion is consumed, the region above $\Delta\theta_d$ in the Figure 3-7. As the pressure already peaks in the previous stage with the maximum amount of mixture being burned, fuel availability to the diminishing flame is important to achieve maximum combustion efficiency. Also unburned hydrocarbons are not desired in the exhaust emissions. The combustion process completes with the flame termination with the opening of the exhaust valve just before BDC. The fuel that still left unburned on the final stage will go through with combustion products to the exhaust. End of combustion is defined as 90-100% combustion duration.

Combustion rate is limited by the time available during the process. Total time available in crank angle degree depends on ignition timing, exhaust valve opening, and engine speed. In the well design engine, the end of stage 2 combustion process is close to the point of maximum cylinder pressure. In practice, for optimum performance and efficiency the maximum pressure should occur on 5-20° ATDC.

Definition of each step of combustion process can be analyzed using pressure reading from the engine data. Detail explanation on determination of combustion process can be seen on next discussion.

3.4.1 Abnormal Combustion

In the above section the combustion process in spark ignition engines is described in detail. Such combustion is called as normal combustion in which the flame moves steadily across the combustion chamber. But there are factors which can prevent this normal combustion process, e.g. fuel composition, engine design, engine operating parameters and combustion chamber deposits [74]. Two types of abnormal combustions can take place: knock and surface ignition.

3.4.1.1 Knocking

Engine knock in SI engine is the major abnormal combustion phenomenon. The name refers to a pinging noise that results from the autoignition of the unburned or end gas ahead of the propagating flame. During the flame propagation across the combustion chamber, the unburned mixture ahead of the flame is compressed, causing an increase in its pressure, temperature and density. Some of the portion of the unburned mixture may autoignite prior to normal combustion, hence releasing a large amount of energy which consequently would cause a sudden rise in the pressure inside the cylinder. This causes high frequency pressure oscillations inside the cylinder. The peak of the combustion process no longer occurs at the optimum moment as is desired. The resulting shock-wave reverberates in the combustion chamber, creating the characteristic metallic "pinging" sound, and pressure increases catastrophically. It can range from hardly noticeable to complete engine destruction.

3.4.1.2 Surface Ignition

Surface ignition occurs due the ignition of the fuel air mixture by overheated valves, spark plug, hot spot or glowing combustion chamber deposit. Surface ignition can be said as ignition by any means other than the spark plug. It can occur before the charge is ignited by the spark plug (preignition) or after its occurrence (postignition) [74]. Surface ignition can occur in more than one location. Upon surface ignition a turbulent flame develops at each location and starts to propagate similar to the normal combustion process.

Both knocking and surface ignition have disastrous consequences for the engine. These can cause a catastrophic wear of the combustion chamber, piston or valves and can also lead to the complete destruction of the engine. These abnormal combustion phenomenons can be prevented by:

- The use of a fuel with higher octane rating.
- Using isooctane or other antiknock agents as a blend with the normal fuel.
- Using richer mixtures (lower AFR)
- Reducing the compression ratio.
- Induction of charge at lower temperatures.
- Retardation of spark plug ignition.
- Improving the combustion chamber design to concentrate mixture near the spark plug.
- Controlling the in cylinder flow of the charge, i.e. high turbulence to promote fast burning.
- Using cold rated spark plugs.

3.5 Combustion Analysis

Engine extracts power by combustion of the fuel inside its cylinders. In a combustion chamber proportionate masses of air and fuel undergo a chemical reaction which produces energy and the reactants transform into products also called as exhaust. This reaction causes an increase in the temperature and pressure inside the cylinder. In cylinder pressure and temperature measurements are a common tool for analyzing combustion process in IC engines. Other observations such as flame propagation, flame visualization, in-cylinder flow behavior, and mixture distribution can also be used but needs complicated measurement systems and devices.

3.5.1 Cylinder Pressure

In cylinder pressure measurement can be used to analyze heat release rate, indicated energy by the combustion process (IMEP), burn rate and combustion efficiency. The most common way of in-cylinder pressure measurements is the by means of a pressure transducer. Pressure transducer enables us to monitor the changes in pressure throughout the cycle. These pressure readings can be analyzed in detail to determine the quantities as mentioned above. In-cylinder pressure (kPa or bars) is usually presented against crank angle (degree) or cylinder volume (m^3). Pressure starts to build up with the compression stroke and increases drastically as combustion takes places, and it eventually drops due to expansion and exhaust valve opening. Pressure difference between the motoring cycle and the working cycle is called as the pressure build up due to combustion process. This pressure build up is used to calculate the heat release rate.

3.5.2 Indicated Mean Effective Pressure (IMEP)

Indicated mean effective pressure can be defined as the work per unit displacement volume done by the gas during the compression and expansion stroke. Pressure data can be used to calculate the work done by the gas on piston. This indicated work is obtained by integrating the curve area enclosed on P-V diagram in Figure 3-8.

$$W_c = \oint p dV \qquad 3\text{-}10$$

in four-stroke engine, the indicated work is defined in two ways,

- Gross indicated work per cycle, W_c, work delivered to the piston by compression and expansion stroke
- Net indicated work per cycle, $W_{c,n}$, is the work delivered to the piston over entire cycle.

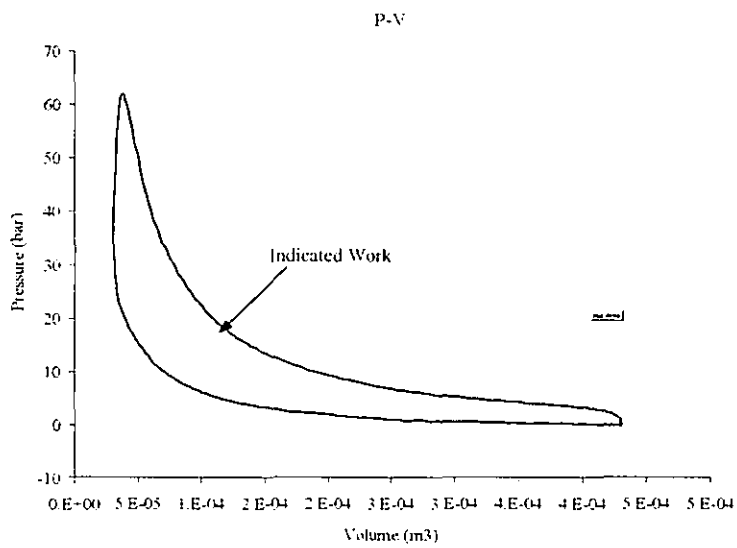


Figure 3-8 Indicated Mean Effective Pressure

The terms of indicated work often refer to the gross indicated work because in the engine analysis, total work at the power cycle is the indicator for engine performance. This indicated work per cycle over displacement volume also called indicated mean effective pressure. IMEP doesn't account for the effects of mechanical friction or the work required pushing the gases in and out of the cylinder, rather it is the reflection of the total useful work available. Volumetric efficiency is related to the variation of IMEP.

3.5.3 Coefficient of Variation (COV)

One of the important parameters to measure the cyclic variability of an engine is the coefficient of variation (COV) in indicated mean effective pressure. It is the standard deviation of IMEP divided by mean IMEP, and is usually expressed in percent:

$$IMEP_{avg} = \frac{1}{n}\sum_{i}^{n} IMEP_i \qquad 3\text{-}11$$

Standard deviation of IMEP defined as:

$$\sigma_{IMEP} = \sqrt{\frac{\sum_{i=1}^{n}\left(IMEP_{avg} - IMEP_i\right)}{n-1}} \qquad 3\text{-}12$$

So, coefficient of variation of IMEP is:

$$COV_{IMEP} = \frac{\sigma_{IMEP}}{IMEP_{avg}} \qquad 3\text{-}13$$

where, n is the number of samples.

COV also indicates the stability of the engine. In order to get a good drivability, COV value should not exceed 10% [74]. Three factors can influence COV:

- The variation in gas motion in the cylinder during the combustion,
- Cycle-by-cycle variation in the amounts of fuel, air, and residual gas to a given cylinder each cycle
- Variations in mixture composition within the cylinder each cycle-especially near the spark plug-due to variations of mixing.

Direct injection systems have higher IMEP compare to manifold due to its higher engine volumetric efficiency. And also, IMEP value in this system also affected by injection timing parameters. Early injection timing has lower IMEP and lower COV (coefficient of variation) of IMEP because of the slower burn rate and higher stability of the mixture. Contrary, late injection timing has higher IMEP and higher COV. The fast burn rate and in-stability of the mixture are the main reason.

3.5.4 Heat Release Analysis

In Otto cycle the combustion is assumed to take place at constant volume conditions. Actual cycles of spark ignition engine do not match with these simple models, and more realistic model is required. Measurement of cylinder pressure can be used to determine the instantaneous heat release, burn fraction and the gas temperature. Single zone and multi zone models are used to analyze the pressure data for the determination of the above mentioned parameters. First law of thermodynamics is used as the basis of these models. In single zone models, the thermodynamic state of the cylinder contents is considered as uniform throughout the cylinder and is represented by average value. In multi-zone or two-zone models the cylinder contents are divided into zones, considering the properties and composition of each zone. Two-zone methods are more accurate compare to single-zone analysis. But, single-zone methods have been widely used in on-line processing to calculate heat release rate from pressure reading. Also it needs less computational time compare to two-zone methods. In the following research a single zone model known as Rassweiler-Withrow model is used to determine the heat release.

When analyzing the internal combustion engine, the in-cylinder pressure has always been an important experimental diagnostic in automotive research and development, due to its direct relation to the combustion and work producing processes [80]. In-cylinder pressure imitate the combustion process, the piston work produced, heat transfer to the chamber walls, as well as mass flow in and out of crevice regions between the piston rings and cylinder liner. Accurate diagnostic of combustion is therefore related to the in-cylinder pressure.

Heat release analysis can be defined as the reduction of the effects of volume changed, heat transfer, and mass loss on cylinder pressure. First law of thermodynamics is used to calculate heat release. The simplest approach is to regard the cylinder contents as a single zone, whose thermodynamic state and properties are modeled as being uniform throughout the cylinder and represented by average values. A more accurate thermodynamic analysis would be to use a multi-zone model, where the cylinder is divided into a number of zones, differing in composition and properties. But single zone models are extensively used in on-line processing to calculate heat release rate from pressure reading. Single zone models are less complex and hence needs less computational time compare to multi-zones methods.

In spark-ignition engines the combustion process varies significantly from one engine cycle to the next one. This cycle-to-cycle variation is an important constraint on engine operation. Cyclic variation is due to varying turbulence within the cylinder from cycle to cycle, inhomogeneous air/fuel mixture and the exhaust residual gas not being fully mixed with the unburned mixture [73].

(a) Single-Zone Model

Single-zone models for analyzing the heat-release rate and simulating cylinder pressure are closely connected; they share the same basic balance equation and can be interpreted as each others inverse. Majority of the models used for heat release use first law as the basis. The energy conservation equation is given as,

$$dQ - dW = dU - \sum_i h_i dm_i \qquad \text{3-14}$$

Where, dU is the internal energy changes of the mass in the system, dQ is the heat transferred to the systems, and dW is work done by the systems, $\sum_i h_i dm_i$ is the enthalpy flux across the system boundary. Following assumptions are made in formulating this model [74];

- in-cylinder contents are uniform throughout the chamber
- combustion is modelled as heat release
- heat released is uniform in the chamber
- the gas mixture is taken as an ideal gas

Consider the combustion chamber to be an open system (single zone), with the cylinder head, cylinder wall and piston crown as boundary. Figure 3-9 shows a schematic of the combustion chamber.

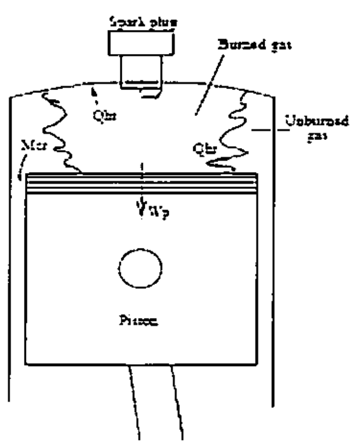


Figure 3-9 Schematic of Combustion Process in the Cylinder

The amount of heat in the cylinder dQ consist of the amount of heat released from chemical energy from the fuel dQ_{ch},which is a heat adding process and the heat transferred from the cylinder to the walls dQ_{ht}, which is a heat removing process. The total heat transfer can therefore be represented as $dQ = dQ_{ch} - dQ_{ht}$. The first law of thermodynamics can be rewritten as:

$$dQ_{ch} - dQ_{ht} - dW = dU - \sum_i h_i dm_i \qquad 3\text{-}15$$

The work done by piston dW can also be written as $dW = pdV$. In the ideal gas law, the change in sensible energy U is a function of temperature, given by $U = mu(T)$ where "m" is the charge mass and T is temperature in the system boundary. Thus

$$dU = mc_v(T)dT + u(T)dm \qquad 3\text{-}16$$

where $c_v = \left(\frac{\partial u}{\partial T}\right)_V$ is the value of specific heat at constant volume.

Crevice effects can usually be modeled adequately by flow into and out of the single volume at the cylinder pressure, with the gas in the crevice at the substantially lower temperature. Leakage to the crank case can be neglected. Then substituting eq. 3-16 into eq. 3-15 and $dm_i = dm_{cr} = -dm$, becomes

$$dQ_{ch} = mc_v dT + (h'-u)dm_{cr} + pdV + dQ_{ht} \qquad 3\text{-}17$$

where

dm_{cr}>0 when flow is from the cylinder into crevice

dm_{cr}<0 when flow is from the crevice to cylinder

h' is evaluated in the cylinder if dm_{cr} > 0 and at crevice if dm_{cr} < 0

Using ideal gas law (neglecting the change in gas constant R) eq. 3-17 gives

$$dQ_{ch} = \left(\frac{c_v}{R}\right)Vdp + \left(\frac{c_v}{R}+1\right)pdV + (h'-u+c_vT)dm_{cr} + dQ_{ht} \qquad 3\text{-}18$$

So the heat release rate with respect to every crank angle change is given as:

$$\frac{dQ_{ch}}{d\theta} = \left(\frac{c_v}{R}\right)V\frac{dp}{d\theta} + \left(\frac{c_v}{R}+1\right)p\frac{dV}{d\theta} + (h'-u+c_vT)\frac{dm_{cr}}{d\theta} + \frac{dQ_{ht}}{d\theta} \qquad 3\text{-}19$$

The specific heat ratio is defined as $\gamma = \frac{c_p}{c_v}$ and assuming ideal gas, the specific gas constant R can be written as $R = c_p - c_v$, so the value of specific heat at constant volume can be written as:

$$c_v = \frac{R}{\gamma - 1} \qquad 3\text{-}20$$

substituting eq.3-20 to eq. 3-19,

$$\frac{dQ_{ch}}{d\theta} = \left(\frac{1}{\gamma-1}\right)V\frac{dp}{d\theta} + \left(\frac{\gamma}{\gamma-1}\right)p\frac{dV}{d\theta} + (h'-u+c_vT)\frac{dm_{cr}}{d\theta} + \frac{dQ_{ht}}{d\theta} \qquad 3\text{-}21$$

The above equation can be used in several ways, if the heat transfer and crevice effects are combined with the heat release term dQ_{ch}, the combination is termed as *net heat release.* A heat loss during the combustion is a small fraction of fuel energy about 10-15 percent [74], the distribution of heat release and heat transfer is different with crank angle. Heat transfer becomes more important at the time when temperature peak near the end of combustion. Integrating the first two terms in the above equation gives the net heat release. If it is normalized to maximum value, it is often interpreted as mass fraction burned.

The mass fraction burned x_b is the normalized integral of the heat release

$$x_b = \frac{\int_{\theta_s}^{\theta} \frac{dQ}{d\theta} d\theta}{\int_{\theta_s}^{\theta_e} \frac{dQ}{d\theta} d\theta} \qquad 3\text{-}22$$

The convective heat-transfer rate to the combustion chamber walls can be calculated from the relation

$$\frac{dQ_{ht}}{dt} = Ah_c(T - T_w) \qquad 3\text{-}23$$

where A is the chamber surface area, T is the mean gas temperature, T_w is the mean wall temperature, and h_c is the heat transfer coefficient (average over the chamber surface area). h_c can be estimated from the heat transfer correlations.

Crevice effect are usually small, a sufficiently accurate model for their overall effect is to consider a single aggregate crevice volume where the gas is at the same pressure as the combustion chamber, but at the different temperature. Since this crevice region are narrow, an appropriate assumption that the crevice is at the wall temperature, by inserting the crevice model into equation (3-21), with $\gamma(T) = a + bT$, gives the chemical energy or gross heat release rate as:

$$\frac{dQ_{ch}}{d\theta} = \left(\frac{1}{\gamma - 1}\right)V\frac{dp}{d\theta} + \left(\frac{\gamma}{\gamma - 1}\right)p\frac{dV}{d\theta} + V_{cr}\left[\frac{T'}{T_w} + \frac{T'}{T_w(\gamma - 1)} + \frac{1}{bT_w}\ln\left(\frac{\gamma - 1}{\gamma' - 1}\right)\right]\frac{dp}{d\theta} + \frac{dQ_{ht}}{d\theta} \qquad 3\text{-}24$$

An example of the use of equation (3-24) to analyze an experimental pressure versus crank angle for a conventional spark ignition engine is shown in Figure 3-10.

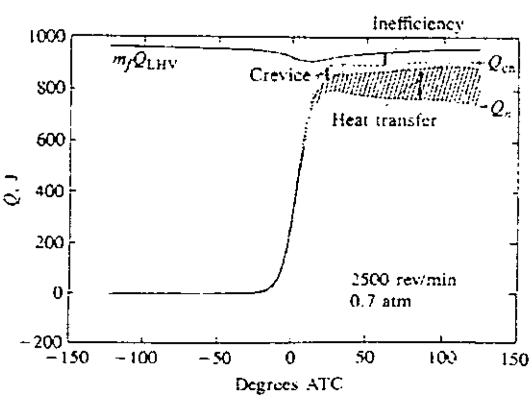


Figure 3-10 Results of Heat Release Analysis Showing the Effect of Heat Transfer, Crevices, and Combustion Inefficiency [74].

The lowest curve shown is the net heat release. Addition of the heat transfer and crevice effect models gives chemical energy release during the combustion process. On the top of the curve is the lower heating value of the fuel multiply by the amount of fuel enters the chamber as the total energy input. The gap between the gross heat release and total chemical energy input is the inefficiency of the combustion process. Inaccuracies of the cylinder pressure reading can also contribute to this different, by this analysis we can also determine the degree of error of the cylinder pressure reading.

From the single zone heat release analysis model, different models with various levels of complexity can be derived. The common single-zone heat release model which is still being used is Rassweiler-Withrow model [78].

(b) Rassweiler-Withrow Model

The Rassweiler-Withrow model also known as polytropic model was originally presented in 1938 and is still being used for determining mass fraction burned, reason being its simplicity and less computational complexity. As this model is also named as polytropic model, so it assumes that the pressure volume relation can be modelled as polytropic relation.

$$pV^{n} = \text{constant} \qquad 3\text{-}25$$

where the constant exponent n value ranges from 1.25 to 1.35. These values of n gives a good fit to experimental data for both compression and expansion process. This exponent n is termed as polytropic index.

Based on eq. 3-24 Rassweiler-Withrow models have some assumption in which there is no crevice effect taking into account $(h' - u + c_v T)\frac{dm_{cr}}{d\theta} = 0$, the heat transfer value is zero $dQ_{ch} = dQ$, the specific heat value can be assumes as constant value and represented as polytropic index, $\gamma = n$, and assuming that there is no release of chemical energy during the compression phase prior to the combustion or during the expansion phase after the combustion, therefore $dQ = 0$. This assumption yields to:

$$dp = -\frac{np}{V}dV \qquad 3\text{-}26$$

from which polytropic index is found by integration and noting that n have constant value.

When considering combustion, equation (3-25) can be rewritten as

$$dp = \frac{n-1}{V}dQ - \frac{np}{V}dV = dp_c + dp_v \qquad 3\text{-}27$$

where dp_c is the pressure change due to combustion and dp_v is the pressure change due to volume changes.

In Rassweiler-Withrow models, the actual pressure change $\Delta p = p_{j+1} - p_j$ during the interval $\Delta\theta = \theta_{j+1} - \theta_j$, is assumed to be made up of a pressure due to combustion Δp_c, and pressure rise due to volume change Δp_V,

$$\Delta p = \Delta p_c + \Delta p_v \qquad 3\text{-}28$$

justified by equation (3-27). The pressure change due to volume change during interval $\Delta\theta$ is given by polytropic relation, which gives

$$\Delta p_v(j) = p_{j+1,v} - p_j = p_j\left(\left(\frac{V_j}{V_{j+1}}\right)^n - 1\right) \qquad 3\text{-}29$$

applying $\Delta\theta = \theta_{j+1} - \theta_j$, equations (3-28) and (3-29) yields to the pressure change due to combustion as

$$\Delta p_c(j) = p_{j+1} - p_j\left(\frac{V_j}{V_{j+1}}\right)^n \qquad 3\text{-}30$$

by assuming that the pressure rise due to combustion occurs in interval $\Delta\theta$ is proportional to the mass of mixture that burns, the mass fraction burned at the end of the process becomes

$$x_b(j) = \frac{m_b(j)}{m_b(total)} = \frac{\sum_{i=0}^{j}\Delta p_c(i)}{\sum_{i=0}^{N}\Delta p_c(i)} \qquad 3\text{-}31$$

where Δp_c is the result of equation (3-30), and N is the total number of crank angle intervals. The result of the mass fraction burned is S-curve. From this mass fraction burned, one can generate each step of the combustion process. Start of combustion,

combustion duration and ignition delay is the properties that can be recognized by analyzing the mass fraction burned curve.

The total heat released by the combustion can be approximate by

$$\Delta Q = \frac{V_{j+1/2}}{n-1}\Delta p_c(j) \tag{3-32}$$

where the volume during the interval j is approximate with $V_{j+1/2}$(the volume at the middle of the intervals).

Rassweiler-Withrow methods have high dependency on the accuracy of the polytropic index value [73]. The polytropic index value for spark ignition engine is commonly in range of 1.25 to 1.35. So the determination of correct polytropic index value is very important before doing the cylinder pressure analysis. Polytropic index can be found using the cylinder pressure reading at the motoring cycle. A common practice is to use the value of 1.3 for the polytropic index in single zone models.

(c) Apparent Heat Release Models

The heat release model that is derived from the first law of thermodynamics was proposed by Krieger and Borman [79]. It was called the computation of net heat release. Apparent heat release model assume there is no heat transfer and crevice effect. Hence, the apparent heat release dQ can be expressed as:

$$dQ = \frac{1}{\gamma - 1}Vdp + \frac{\gamma}{\gamma - 1}pdV \tag{3-33}$$

This apparent heat release analysis has the same expression as Rassweiler and Withrow, based on the assumption of polytropic index value. Results shown by these methods differ slightly from each other. Rassweiler and Withrow shows slightly shorter combustion duration compare to apparent method. Both methods are preferable for combustion analysis because of the least computational time, and less parameters required to analyze the combustion in the engine [81].

3.5.5 Combustion Efficiency

Combustion efficiency is the difference between available energy (fuel) and the final value of heat released by combustion. It is defined as

$$\eta_c = \frac{Q_{out}}{Q_{in}} = \frac{\sum Q_{ch}}{m_f q_{HV}} \qquad 3\text{-}34$$

Where, $\sum Q_{ch}$ is the total heat released by combustion process, m_f is fuel mass, and q_{HV} is the specific heating value of fuel.

3.6 Emissions

During a combustion reaction in an internal combustion engine the air/fuel mixture (reactants) undergo an exothermic reaction producing energy and the products (exhaust) of the chemical reaction. These products are forced out of the engine as exhaust or engine emissions. These emissions are the source of air pollution. Automotives are considered as the largest contributor to air pollution in the world. Therefore strict regulations are being practiced among the world's developed countries to reduce this pollution problem caused by the automotives.

Most of the automotives around the world use gasoline or diesel as fuel. Gasoline and diesel fuels are mixtures of hydrocarbons, compounds which contain hydrogen and carbon atoms. In a "perfect" engine, oxygen in the air would convert all the hydrogen in the fuel to water and all the carbon in the fuel to carbon dioxide. Nitrogen in the air would remain unaffected. In reality, the combustion process cannot be "perfect," and automotive engines emit several types of pollutants. These pollutants mainly contain unburned hydrocarbons (HC), oxides of nitrogen (NOx) and carbon monoxide (CO). Though there are other species formed as the products of the combustion of the fuel but these three are considered as prime contributors to the air pollution.

3.6.1 Hydrocarbon Emission

Hydrocarbon emissions result when fuel molecules in the engine do not burn or burn only partially. Hydrocarbon fuels are composed of 10 to 20 major species and 100 to 200 minor species [75]. Most of these species can be found in the exhaust. The level of unburned hydrocarbon in the exhaust gases is generally specified to the total concentration of hydrocarbons expressed as part per million (ppm). Hydrocarbon emission is a useful measure of inefficiency in combustion process. In general, hydrocarbon emission is categorized into two classes; the methane and non-methane hydrocarbons. All hydrocarbons have potential to react with the atmosphere due to their unstable bonding except methane, which have a very stable bonding and will not react in air. CNG has about 80-90% methane in its composition. Incomplete combustion of CNG would result higher percentage of methane emission [72].

There are four possible HC emission formation mechanisms in spark-ignition engine [74]; (1) flame quenching at the combustion chamber walls, leaves a layer of unburned fuel-air mixture adjacent to the wall; (2) the filling of crevice volumes with unburned mixture. When the flame quenches at the crevice volume, mixture will escape along with the result from primary combustion process; (3) absorption of fuel vapor to oil layers on the cylinder wall during intake and compression and followed by desorption of fuel vapor in to the cylinder during expansion and exhaust stroke; (4) and incomplete combustion in a fraction of the engine operating cycles occurs when the combustion quality is poor due to engine transients operation, non-optimum control of EGR, AFR, and ignition timing. About 9% of the fuel supplied to an engine is not burned during the normal combustion phase of the expansion stroke. Though some 7% of the hydrocarbons are consumed in the other three strokes yet 2% will go out with the exhaust. Table 3-1 shows that the main contributor to the hydrocarbon emissions is the crevice volume.

Table 3-1 Sources of HC [72]

Source	% fuel escaping normal combustion	% HC emission
Crevices	5.2	38
Oil Layers	1.0	16
Deposits	1.0	16
Liquid Fuel	1.2	20
Flame Quench	0.5	5
Exhaust valve leakage	0.1	5
Total	9.0	100

3.6.2 Nitrogen Oxides (NOx)

Under the high pressure and temperature conditions in an engine, nitrogen and oxygen atoms in the air react to form various nitrogen oxides, collectively known as NOx. Nitrogen oxides, like hydrocarbons, are precursors to the formation of ozone. They also contribute to the formation of acid rain. NOx mainly consists of NO (Nitric Oxide) and NO_2 (Nitrogen Dioxide). NO possesses the main percentage in NOx emission. The main source of NOx emission is the oxidation of atmospheric nitrogen during the combustion process, however if there is significant amount of nitrogen in the fuel, its oxidation also causes the NOx emission. NOx forming reactions are very temperature dependent. NOx formation is lower during the engine start and warm-up. In spark ignition engines the nitric oxide (NO) is the dominant component of the NOx, whereas the concentration of nitrogen dioxide (NO_2) is only 1% to 2% [75].

Three reaction mechanisms explain the formation of NO, namely, Zeldovich mechanism, Fennimore or prompt mechanism and the N_2O intermediate mechanism. For internal combustion engines Zeldovich mechanism is of more significance, in which the formation of NO occurs in the high temperature burned gases left behind the flame front.

$$
\begin{aligned}
O + N_2 &\Leftrightarrow NO + N \\
N + O_2 &\Leftrightarrow NO + O \\
N + OH &\Leftrightarrow NO + H
\end{aligned}
$$

The above three chemical reactions are known as Zeldovich mechanism for NOx formation. These reactions contribute significantly to the formation of nitric oxide during combustion process.

Nitric oxide formation in CNG fuelled spark ignition engine has been extensively researched. Zeldovich mechanism for NOx formation is extensively used. It is reported that with the increase in temperature the formation of nitric oxide increases significantly. NO formation becomes significant at temperature above 1200 °C in methane/air combustion [76]. Further, high temperature combustion exceeding 1600 °C, the thermal NO formation became important. For temperature less than 1500 °C, another NO formation route known as N_2O/NO mechanism was suggested.

$$
\begin{aligned}
N_2 + O + M &\leftrightarrow N_2O + M \\
N_2O + O &\leftrightarrow NO + NO \\
N_2O + O &\leftrightarrow N_2 + O_2
\end{aligned}
$$

Another route for NO formation at low temperatures is given as follows [25]:

$$
\begin{aligned}
N_2 + H &\leftrightarrow NNH \\
NNH + O &\leftrightarrow NO + NH
\end{aligned}
$$

Thermal NO mechanism, N_2O/NO and NNH mechanism are the common NO formation mechanism that is used to models and predict the nitric oxide emission. Temperature and equivalence ratio are reported to be the most influential parameters in the formation of nitric oxide. Nitric oxides are maximum for slightly lean mixtures. Other variables that also determine the NOx emission is burned gas fraction of the in-cylinder unburned mixture, and spark timing [6]. Also for direct injection engines (DI), nitric oxide increases with load.

To control nitric oxide emission, two ways can be adopted. First is exhaust gas recirculation (EGR) and the second is using catalytic converter. EGR works by recirculating a portion of an engine's exhaust gas back to the engine cylinders. In spark ignition engines typically 5% to 15% of the exhaust is routed back into the intake as EGR. Intermixing of the incoming air with recirculated exhaust gas dilutes the mixture, hence reducing flame temperature and flame speed. The maximum quantity is limited by the requirement of the mixture to sustain a contiguous flame front during the combustion event; excessive EGR in an SI engine can cause misfires and partial burns. Although EGR does measurably slow combustion, this can largely be compensated for by advancing spark timing. Second is the use of catalytic converter. A catalytic converter is a device used to convert the harmful engine emissions to environmentally acceptable engine out emissions before they leave the exhaust system of the engine. It converts the NOx in the emission to Nitrogen (N_2) and Oxygen (O_2) by reduction reactions.

3.6.3 Carbon Monoxide (CO)

Carbon monoxide is yet another harmful species found in the exhaust of the internal combustion engines. It is formed due to the partial combustion of the carbon containing compounds in the fuel. When there is not enough oxygen in the mixture to convert all the carbon in the fuel to carbon dioxide, CO emission results. The most important parameter influencing carbon monoxide emission is the fuel-air equivalence ratio. Engines running on rich mixtures produce more CO compared to those running lean. Since in high temperature products, even with lean mixtures, dissociation ensures there are significant CO levels, yet these levels are lower than that in rich mixtures.

In premixed hydrocarbon-air flames, CO concentration increases rapidly in the flame zone to maximum value, which is larger than equilibrium value for adiabatic combustion for the fuel air mixture. CO formation principal reaction steps in hydrocarbon combustion mechanism can be summarized by the equation 3-35, as given in [74].

$$RH \rightarrow R \rightarrow RO_2 \rightarrow RCHO \rightarrow RCO \rightarrow CO \qquad 3\text{-}35$$

Where, R stands for radical hydrocarbon such as CH_3, CH_2, C_2H_2 etc that has higher tendency to react according to the reaction process described in reaction chain 3-15. CO can also be formed because of not enough time available during the formation of CO2 which is described in reaction 3-36.

The principal of CO oxidation in hydrocarbon-air flames is

$$CO + OH \Rightarrow CO_2 + H \qquad 3\text{-}36$$

CO formed in the combustion process via this path has a slow rate of oxidation to CO_2. CO emissions can be controlled by closely monitoring the fuel-air equivalence ratio. Since leaner mixtures are less prone to CO emissions, therefore the uniformity of the mixture and leaning-out of the mixture are of importance. The other way is to use catalytic converters, which convert the CO into CO_2 by oxidation.

CHAPTER 4

METHODOLOGY

In the following chapter the equipment being used in the research is explained along with the details of its characteristics and measurement capabilities. It also includes the data gathering, equations being used in the analysis and also the constraints of the parameters being analyzed.

4.1 Engine Test Bed

The engine test bed along with the instrumentation that was being used in the research are explained in this section.

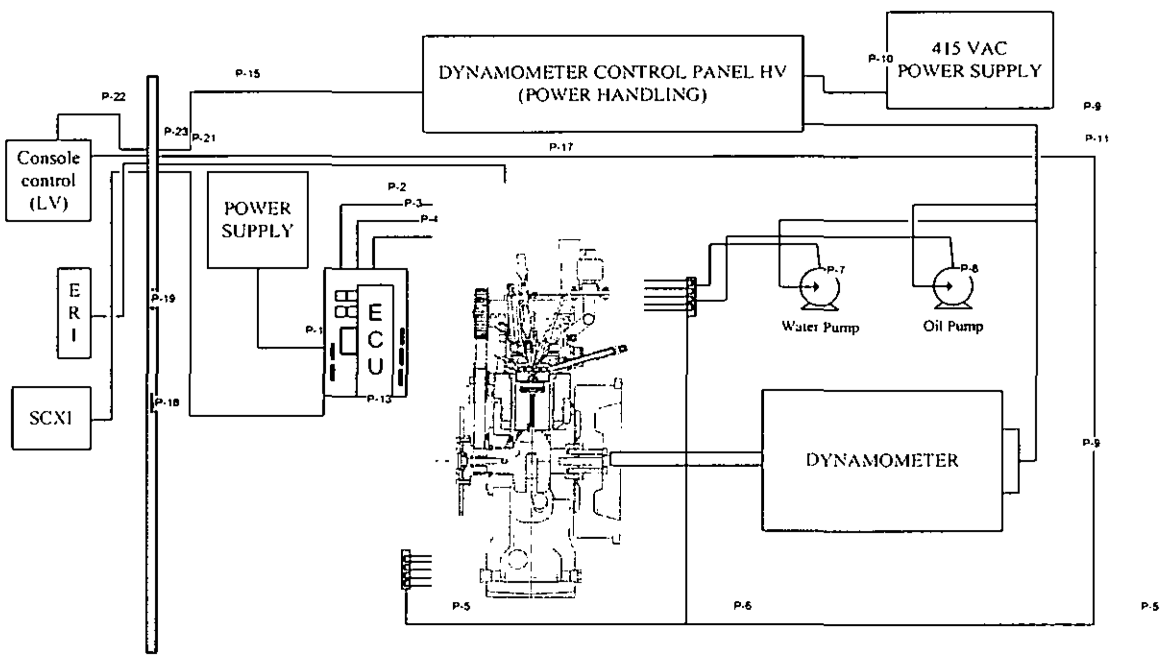


Figure 4-1 Schematic of Engine Test Bed

The engine schematic setup is shown in the Figure 4-1. The engine is a four stroke single cylinder research engine with a displacement volume of 399.25 cc and a compression ratio of 14. This engine is being used to investigate the performance of the engine using homogeneous and stratified operations under idle and part load conditions. Engine parameters such as injection timing, ignition timing and air fuel ratio are controlled by ECU (Engine Control Unit) that is connected to ECU Remote Interface (ERI) installed in a personal computer. The other engine specifications are given in the Table 4-1.

Table 4-1 Engine Specifications and Test Bed

Engine Properties	
Displacement volume	399.25 cm^3
Cylinder Bore	76 mm
Cylinder Stroke	88 mm
Compression Ratio	14
Exhaust Valve Closed	ATDC 10°
Exhaust Valve Open	BBDC 45°
Inlet Valve Open	BTDC 12°
Inlet Valve Closed	ABDC 48°
Dynamometer	Eddy Current with maximum reading of 50Nm
ECU	Orbital Inc

4.1.1 Injector and Spark Plug Position

Figure 4-2 shows the geometry of the combustion chamber with injector and spark plug location. The injector is placed at the centre of the combustion chamber with the spark plug next to it with an offset of 6mm. Spark plug with a longer tip is used compared to the standard one to improve the combustion of CNG.

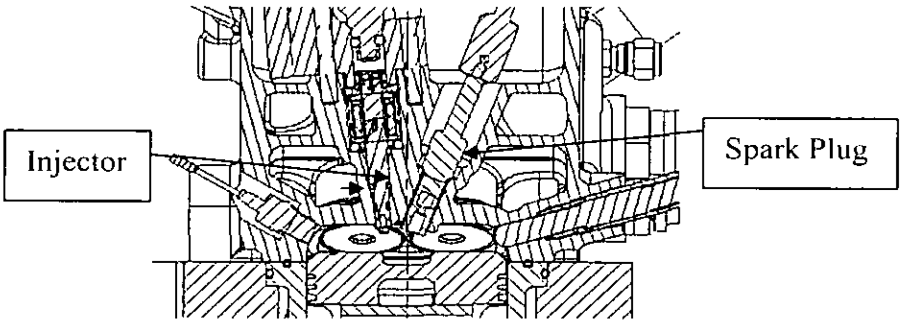


Figure 4-2 Injector and Spark Plug Position

Figure 4-3 illustrates the sectional view of the relative position of injector and the spark plug.

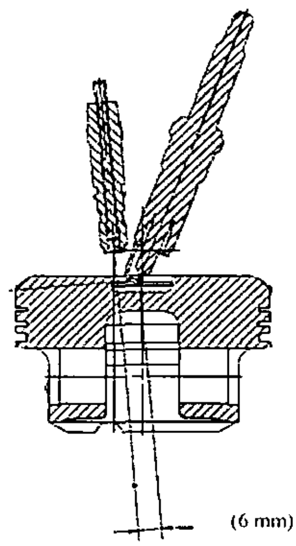


Figure 4-3 Relative Position of Injector and Spark Plug

4.1.2 Pistons

Homogeneous and Stratified pistons are used to carry out homogeneous and stratified conditions respectively. In Figure 4-4 a homogeneous piston is shown. The sectional view is also given for the detailed illustration. This piston has a small cup placed in the centre of the crown.

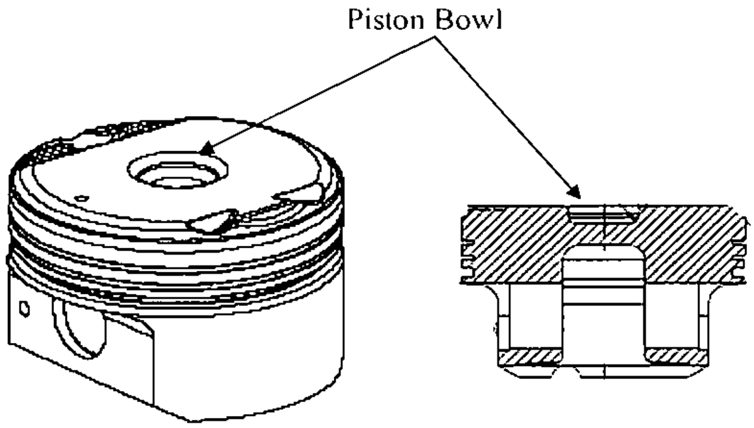


Figure 4-4 Homogeneous Piston

Stratified piston has a bigger cup compared to that of homogeneous piston and is position away from the centre as illustrated in the sectional view. This configuration is specially designed to achieve stratified conditions, in which fuel is deflected back from the piston head to the spark plug, so that a rich mixture is near the spark plug.

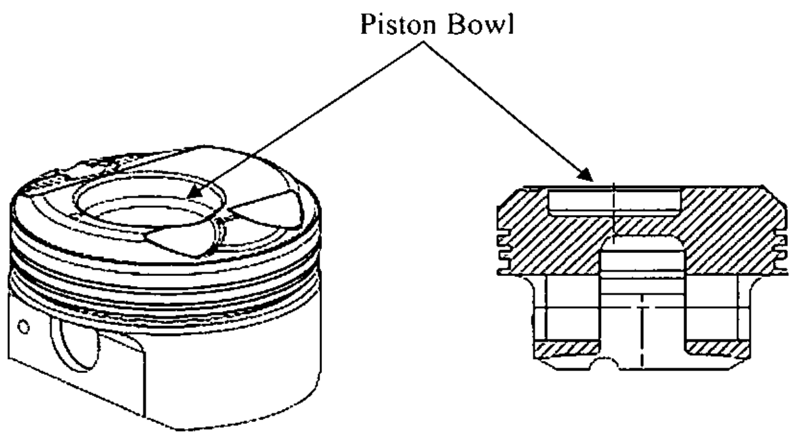


Figure 4-5 Stratified Piston

4.1.3 Dynamometer

A dynamometer or "dyno" for short, is a machine used to measure torque and rotational speed (RPM), from which power produced by an engine, motor or other rotating prime mover can be calculated. A dynamometer can also be used to determine the torque and power required to operate a driven machine such as a pump. There are different types of dynamometers such as;

- Water brake (absorption only)
- Fan brake (absorption only)
- Electric motor/generator (absorb or drive)
- Mechanical friction brake or Prony brake (absorption only)
- Hydraulic brake (absorption only)
- Eddy current or electromagnetic brake (absorption only)

Eddy current dynamometer is used in the experimental work to get the power ratings of the engine. The dynamometer is coupled with the engine to measure the brake torque as the engine output.

4.1.4 Fueling Scheme

The fueling scheme is shown in the Figure 4-6. Pressure regulators are used to reduce the pressure of the CNG from tank which is 200 bars, to 30 bars. Micro-motion mass flow meter is used to measure the flow rate to the engine. It is placed after the pressure regulator and it has a sensitivity of .0001 g/s. Inlet fuel pressure control system is placed after the flow meter. This system is coupled with a compressor to maintain the fuel pressure along the fuel rail.

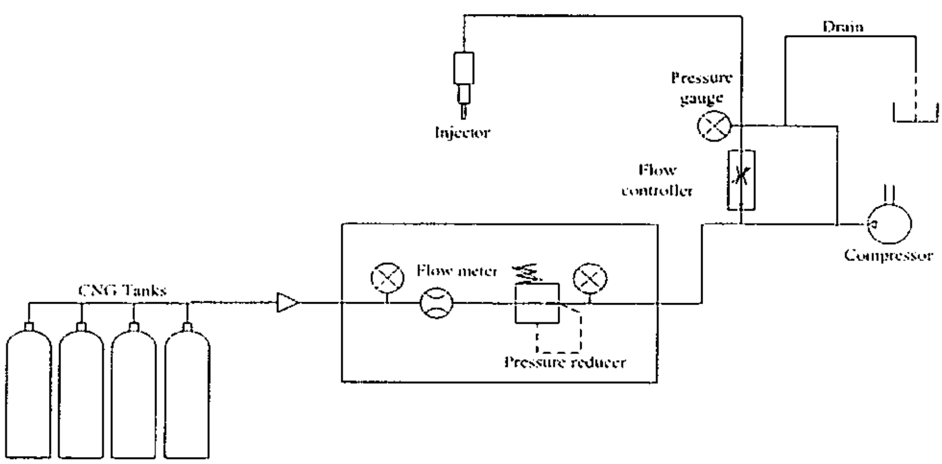


Figure 4-6 CNG Fueling System

4.1.4.1 Micro-motion Mass Flow Meter

A mass flow meter, also known as inertial flow meter or coriolis flow meter is a device that measures how much fluid is flowing through a tube. It does not measure the volume of the fluid passing through the tube; it measures the amount of mass flowing through the device. The specifications of the micro-motion mass flow meter that is being used is given in Table 4-2

Table 4-2 Micro-motion Fuel Flow Meter Specifications

Flow accuracy:	+/-0.05% of flow rate
Gas accuracy:	+/-0.35% of flow rate
Density accuracy:	+/-0.0002 g/cc
Wetted materials:	304L, 316L Stainless Steel or Nickel Alloy
Temperature rating:	-400 to 800°F (-240 to 427°C)
Pressure rating:	1450 - 6000** psi (100 to 413** bar)

4.1.5 Pressure Sensor

A number of methods have been used to measure pressure as a function of cylinder volume. We will restrict our attention to piezoelectric transducers (Figure 4-7), since they are the methods that were used in this experiment.

The piezoelectric effect is the generation of an electric charge to a solid by a change in pressure. There are two primary piezoelectric effects: (1) the transversal effect in which charges on the x-planes of the crystal result from the forces acting upon the y-plane, and (2) the longitudinal effect in which charges on the x-planes of the crystal result from forces acting upon the x-plane. Piezoelectric transducer can be obtained with internal cooling passage and with a temperature compensator. Note that increasing temperature will make the casing expand and uncompressed crystals from load. This cooling is important operational procedures to get correct pressure reading.

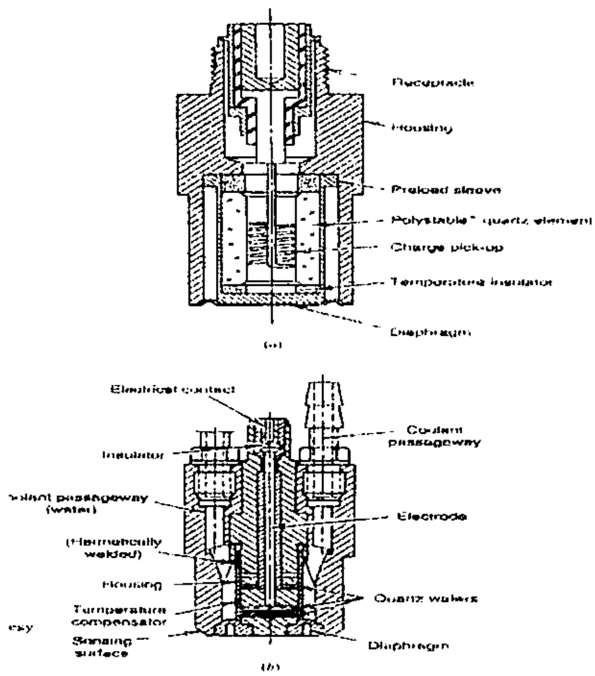


Figure 4-7 Quartz Piezoelectric Pressure Transducers. (a) Courtesy of Kistler Instrument Corp. (b) Courtesy of AVL Corp [75].

A typical system for measuring cylinder pressure as a function of cylinder volume is shown in Figure 4-8. A crank angle encoder is used to establish the top dead centre position and the phasing of cylinder pressure to crank angle.

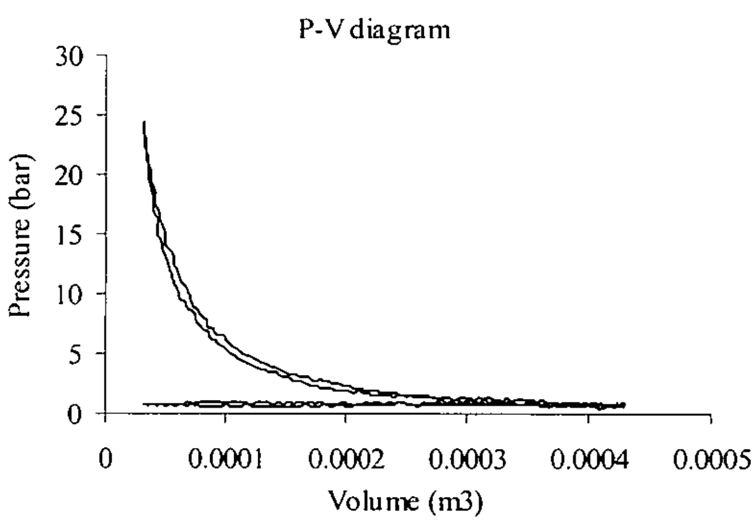


Figure 4-8 Pressure as a Function of Volume in the Engine Cylinder

4.1.6 Exhaust Gas Analyzer

Exhaust gas analyzer is shown in Figure 4-9 (manufactured by GASMET™) which is used to measure the exhaust gases in the engine. The analyzer measures about 50 gas species in the exhaust and give their quantities in parts per million (ppm). General specifications of the gas analyzer are provided in Table 4-3.

Figure 4-9 GASMET™ Stationary FTIR Analyzer

The stationary GASMET™ FTIR analyzer consists of four systems:

1. **Gasmet**™ Cr-2000 gas analyzer with LN_2 detector and fast response sample cell
2. **Gasmet**™ sampling system
3. Heated Filter Unit
4. **Gasmet**™ Industrial Computer

Table 4-3 General Specifications of GASMET™ FTIR Analyzer

General parameters	
Measuring Principle:	Fourier Transform Infrared, FTIR
Performance:	simultaneous analysis of up to 50 gas compounds
Response time:	typically <<25s, depending on the gas flow and measurement
Operating temperature:	15-25°C non condensing
Storage temperature:	-20 - 60°C non condensing
Power supply	100-115 or 230V / 50-60 Hz
Power consumption:	300 W
Measuring parameters	
Zero point calibration:	24 hours, calibration with nitrogen
Zero point drift:	<2 % of measuring range per zero point calibration interval
Sensitivity drift:	none
Linearity deviation:	<2 % of measuring range
Temperature drifts:	<2 % of measuring range per 10 K temperature change
Pressure Influence:	1 % change of measuring value for 1 % sample pressure change. Ambient pressure changes measured and compensated

4.1.7 Injectors

Injector is the most influential component in direct-injection systems. Injector properties have a great influence on the mixing, charge stratification and combustion stability of the engine. Spray angle, penetration length, fuel delivery rate are known to be the critical characteristics of an injector, as they effect the performance of the engine as explained above.

Two types of injectors are used in the experiments, one is narrow angle injector (NAI) which has a spray angle of 30 degree, and other is wide angle injector (WAI) with a 70 degree spray angle. Figure 4-10 shows the injector spray image at atmospheric condition for both injector, WAI and NAI.

Injector Spray Image

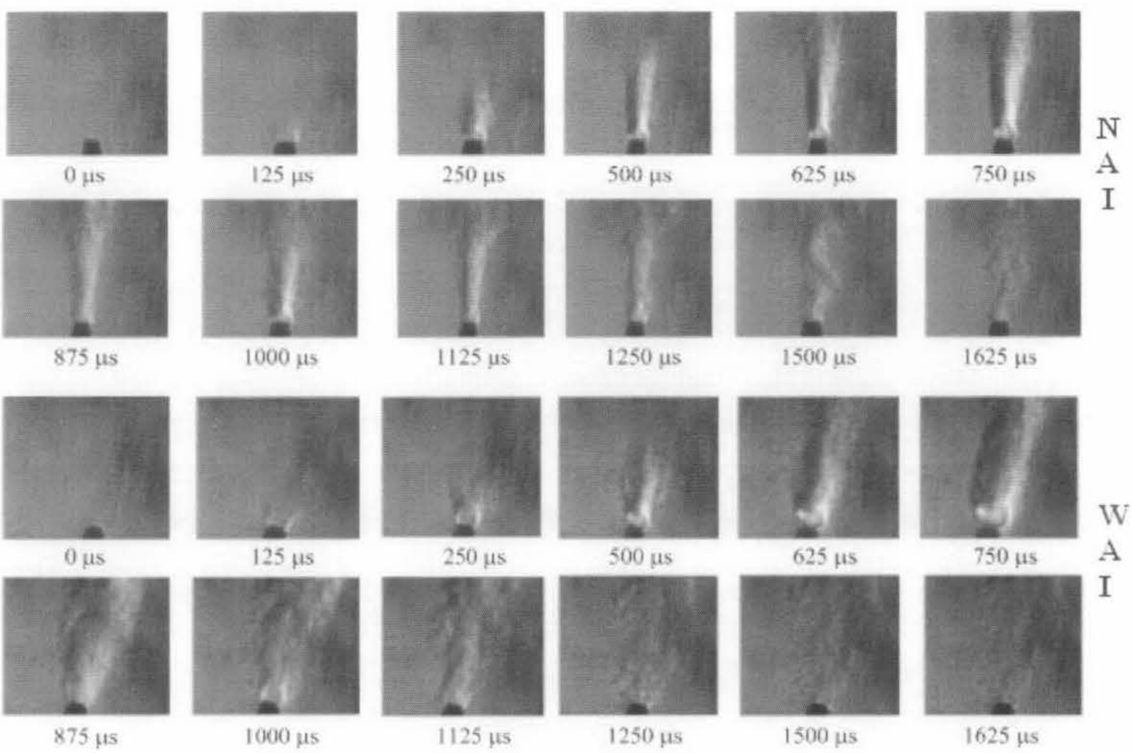


Figure 4-10 Injection Sequence at Atmospheric Condition for WAI and NAI Injector Spray Angle

WAI shows a wider circulation region which may promote better mixing of fuel with air and increasing the turbulence.

4.2 Injection Parameters and Data Collection

Data collecting for these experiments are following SAE standard for Engine performance and Testing. SAE J1995, Engine Power Test Code-Spark ignition and Compression ignition-Gross power rating.

Injection parameters such as injection timing, injection pressure, and type of injectors are investigated to verify the performance of CNG-DI engine. Injection timing, injection pressure and injection spray angle are proven to be significantly affecting direct injection engine performance. Other injection parameters such as injector location, number of injector does not have significant effect to the performance and combustion of CNG direct injection engine [24].

4.2.1 Injection Timing

Injection timing was set from early to late injection timing. At very early injection timing, the fuel was injected when intake valve starts to open, 300 degree BTDC. Very early injection timing at direct injection systems can be considered as port injection. Although there are differences between port injection systems and very early injection timing at direct injection systems, very early injection timing can be considered to be similar to port injection system [62]. The mixing behaviour for both was almost the same.

Injection timing starts from 300 degree BTDC to 80 degree BTDC with increments 20 degrees for the experiment done with homogeneous piston at stoichiometric conditions and at idle and partial loads.

For stratified conditions injection timing was delayed even further. Injection starts at 132 degree BTDC when the inlet valve is closed and is delayed further till the value where the engine is still operating in stable conditions. Again the experiments are done at idle and part load conditions.

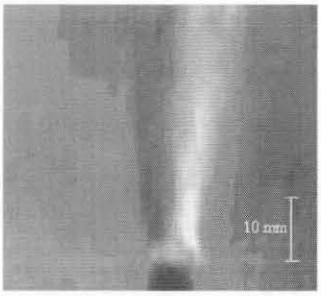
4.2.2 Injection Pressure

Injection pressure is kept constant at 18 bars for stoichiometric and stratified conditions. Pressure changes significantly affect the delivery rate of the injector. Higher injection pressure resulted smaller injection window needed by injector to deliver the fuel required by the systems. As the required window is smaller, the injection timing can be retarded closed to TDC. Injection timing closed to TDC is implemented in stratified operations.

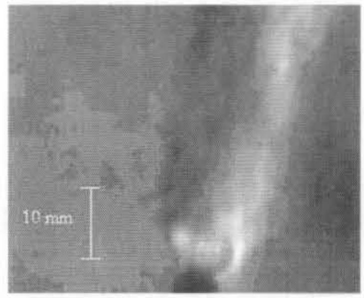
4.2.3 Injector Spray Angle

Two different injectors with different spray angles are used in the experiments. Injector spray angle is expected to affect the engine performance. The two different injectors used in this experiment are, narrow angle injector (NAI) and wide angle injector (WAI). Injector spray angle was affecting the mixing rate of the air fuel mixture. Wider injector angle gives higher percentage of air entrapment during the injection process, so it is expected that wide angle injector have higher mixing rate compared to narrow angle injector.

Figure 4-11 represents the injection sequence for both injector spray angles. The image is taken at 18 bars injection pressure at atmospheric conditions. The fuel disappearance for WAI is faster compared to that of NAI. This may be because of the mixing of the fuel with air. So it can be assumed that WAI gives better mixing rate than NAI.



NAI



WAI

Figure 4-11 Injector Spray Cone Angle

4.3 Engine Performance Parameters

Torque, power, brake specific fuel consumption and volumetric efficiency are considered to be the key parameters for engine performance, which have already been discussed in the previous chapter in section 3.3.

4.3.1 Combustion Analysis

Combustion analysis can be generated from the pressure reading resulted by the pressure sensors. PC-based combustion analysis hardware and software is used to acquire and analyze the pressure data. The scheme of the systems can be seen in Figure 4-12.

The hardware consists of high-speed data acquisition and dedicated signal processor. The software performs statistical and thermodynamics analysis of the pressure data at real-time. Measurements of cylinder pressure can be used to determine not only the location of peak pressure, but also the instantaneous heat release, and burn fraction. Lab-View high speed data acquisition is used to obtain the reading of cylinder pressure, with minimum 100 power cycles is recorded as the result of the experiment. During the cycle of an engine useful work is only done on the power stroke. By measuring the cylinder pressure through a cycle it is possible to calculate the average pressure that is useful. This

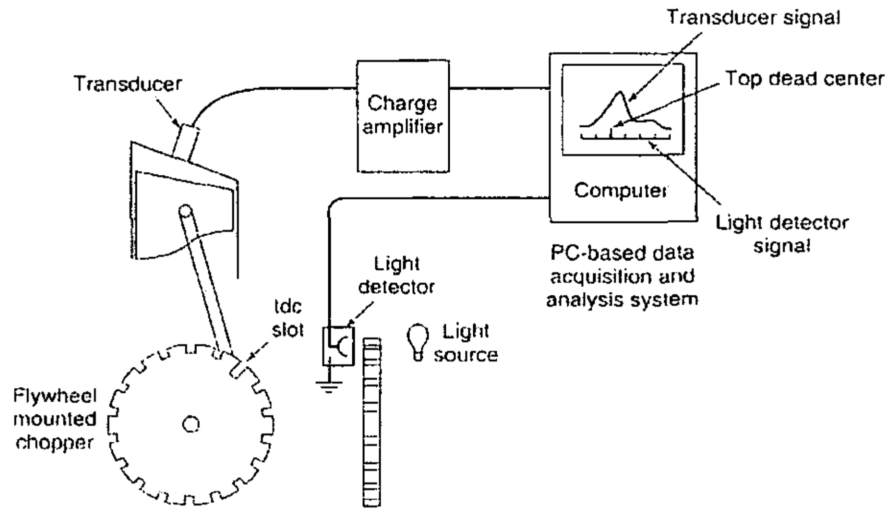


Figure 4-12 Typical Pressure Measurement System [75]

is the Indicated Mean Effective Pressure (IMEP). IMEP can be generally calculated by numerical integration on the PV diagram.

However, this method requires a great number of sampled data per cycle in order to ensure high accuracy in calculation.

There are many alternative equations to get the result of IMEP integration from cylinder pressure data, equation with the least possible error is:

$$IMEP = \frac{\Delta\theta}{V_d} \cdot \sum_{i=n_1}^{n_2} p(i) \frac{dV(i)}{d\theta} \qquad \text{4-1}$$

Where $p(i)$ is cylinder pressure at crank angle i, $V(i)$ is cylinder volume at crank angle i, V_d is cylinder displacement volume, and n_1 and n_2 refers to the BDC crank angle position. Crank angle resolution, pressure transducer accuracy and engine geometry significantly affect the errors in IMEP calculations. These errors can be significantly reduced if the crank angle resolution is less than 10 crank angle degrees.

4.4 Device Calibration

To ensure the accuracy in the data the calibration process is used. In the following experiment, routine calibration was mainly done to dynamometer, pressure data acquisition system, and exhaust gas analyzer.

4.4.1 Dynamometer Calibration Process

Dynamometer was calibrated using calibrated weights as reference. The weights are put on one side of dynamometer and the reading is noted down from the control panel. After every reading the weights are increased in a sequential manner as shown in the Table 4-4.

Table 4-4 Load Sequence for Dynamometer Calibration Process

No	Weight (kg)	Torque (Nm)
1	1	4
2	2	8
3	5	20
4	10	40
5	12	48

When the dynamometer achieves its maximum reading (i.e.; 48 Nm) weights are unloaded. The reading on the control panel should be zero when all the loads are removed. Same procedure is repeated for other arm of dynamometer.

4.4.2 Pressure Data Acquisition Systems Calibration

Pressure data acquisition system is calibrated by using pressure testing device for engine. The procedure for pressure sensor calibration was:

- Turn-on all systems
- Unplug the spark plug from its position

- Install the manual pressure testing device on the spark position
- Motoring the engine in a low rpm condition (<300 rpm)
- Measure the reading by data acquisition systems
- Compare both result from manual pressure device and pressure sensor reading

Maximum pressure was compared using this procedure. The manual pressure reading device was showing pressure gauge reading. If the pressure sensors shows the same reading with pressure gauge, than in order to get absolute pressure reading, the whole pressure sensor reading need to add by 1.

4.4.3 Exhaust Gas Analyzer Calibration

Exhaust gas analyzer is calibrated before starting any experimental works. GASMET exhaust gas analyzer is already equipped with self calibration procedure. This calibration procedure is called *zero calibration.* Zero calibration is necessary to be done before starting the experiment. It will measure the background spectrum for subsequent sample spectrum measurement. Zero calibration can only be valid if the instrument is in steady state condition with certain cell temperature and interferometer temperature.

For zero calibration, the sample must be filled with pure substance such as N_2 to make sure that there is no unwanted sample in the test cell. FTIR system will create background data base of the pure substance analysis as the base line for the measurement process. A typical background spectrum is presented in Figure 4-15. The background spectrum represents the actual absolute intensity of infrared radiation that is transmitted through zero gas filled sample.

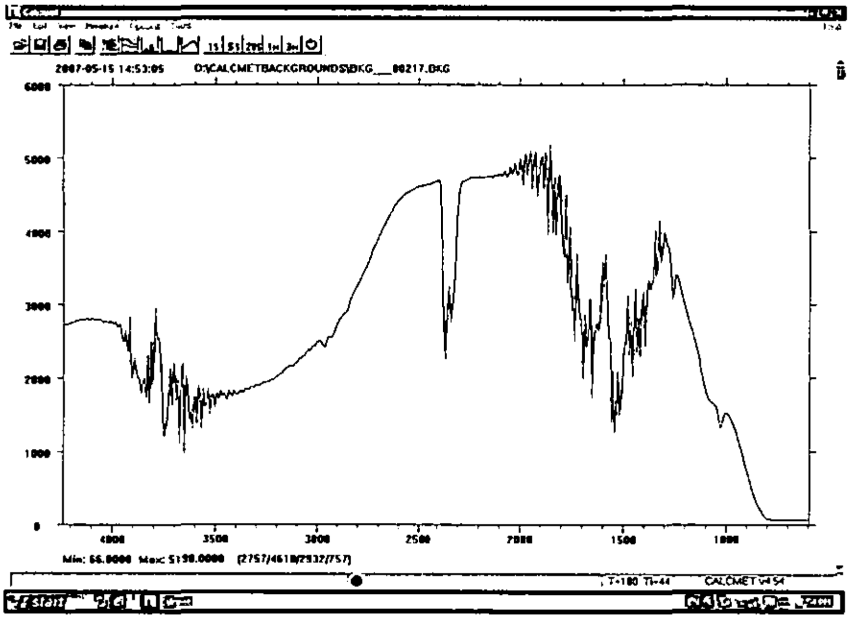


Figure 4-15 Calibration Spectrum on FTIR System for Emission Analysis

CHAPTER 5

RESULTS AND ANALYSIS

5.1 Injection Timing Effect

Injection timing effect on the performance of CNG-DI engine is discussed in this chapter. A single cylinder dedicated research engine is used for the test runs. The engine is having a compression ratio of 14 and is equipped with a central direct injection system. The tests are conducted at idle and partloads conditions. Two different pistons geometries i.e. stoichiometric (Homogeneous) and Stratified piston are used for stoichiometric and lean conditions respectively. Air fuel ratio of CNG was set at stoichiometric (14.7) for homogeneous operation while for the stratified operation it was set at 36.5-37. Ignition timing was set to have maximum brake torque (MBT). A high pressure fuel injector (18bars) was used on the central direct injection system. Injection timing was set to start after the intake valve closes at 132 deg BTDC and is retarded unless is limited by the ignition timing parameters. Minimum limit for retarding the injection timing is different for every speed and for lean operations this limit is further increased. Injection duration for different injection timings is shown in the Figure 5-1. Injection duration increases with the increase in engine speed. However, retarding the injection timing is limited by the ignition parameters.

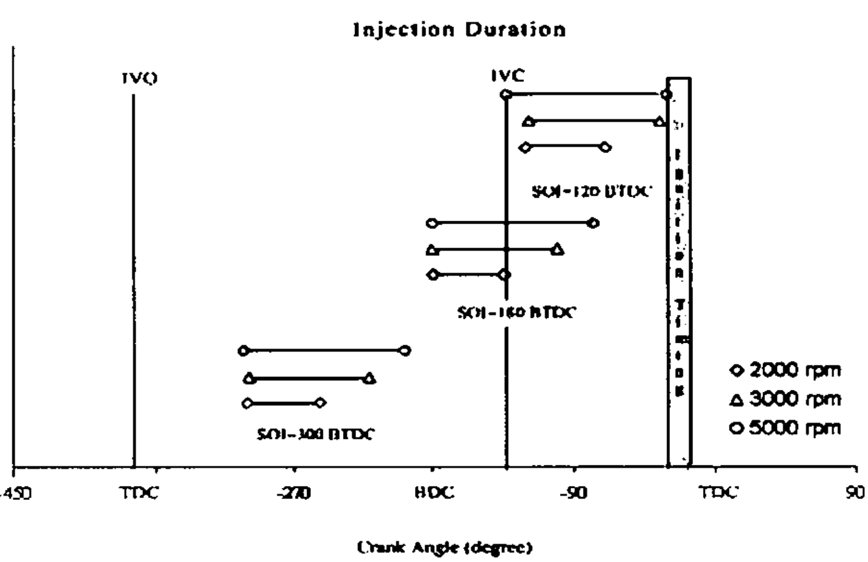
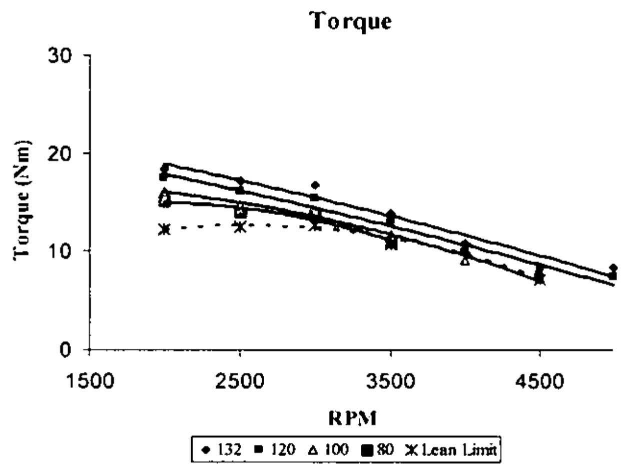


Figure 5-1 Injection Duration for Different Injection Timings.

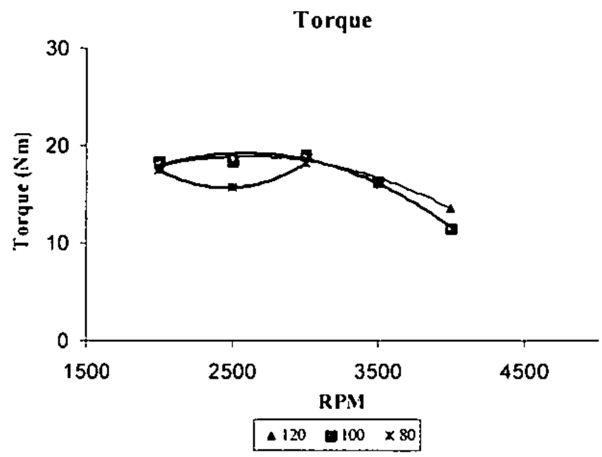
5.1.1 Results of Engine Performance at Idle Loads

Engine performance characteristics at idle loads i.e. when the throttle is fully closed for both lean stratified and stoichiometric conditions are shown in the Figure 5-2. The injection timings for both stoichiometric and lean operations are compared. For lean operations the injection timing right after the closing of intake valve at 132 deg BTDC is also included, whereas for stoichiometric operation the injection starts 12 degrees after the closing of intake valve. The lean limits are also included for the lean conditions in the results. Performance parameters such as torque, power, brake mean effective pressure (BMEP) and brake specific fuel (BSFC) consumption are shown in the Figure 5-2.

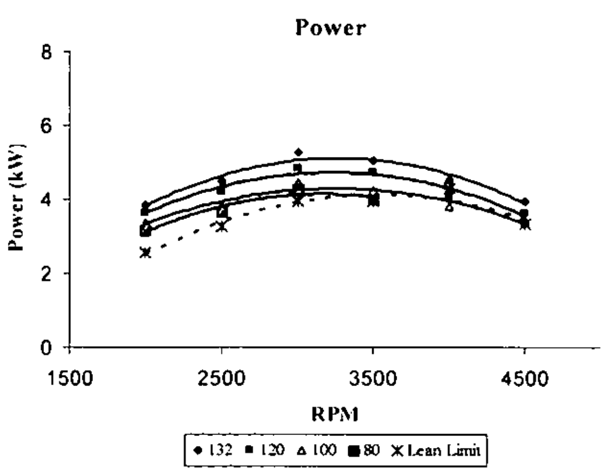
Stratified Lean **Stoichiometric**



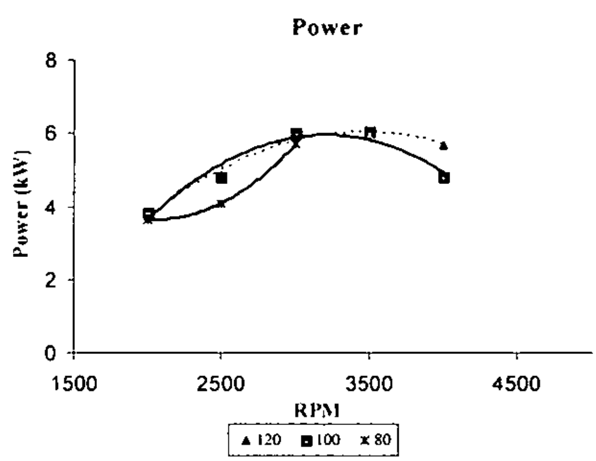
(a1)



(a2)



(b1)



(b2)

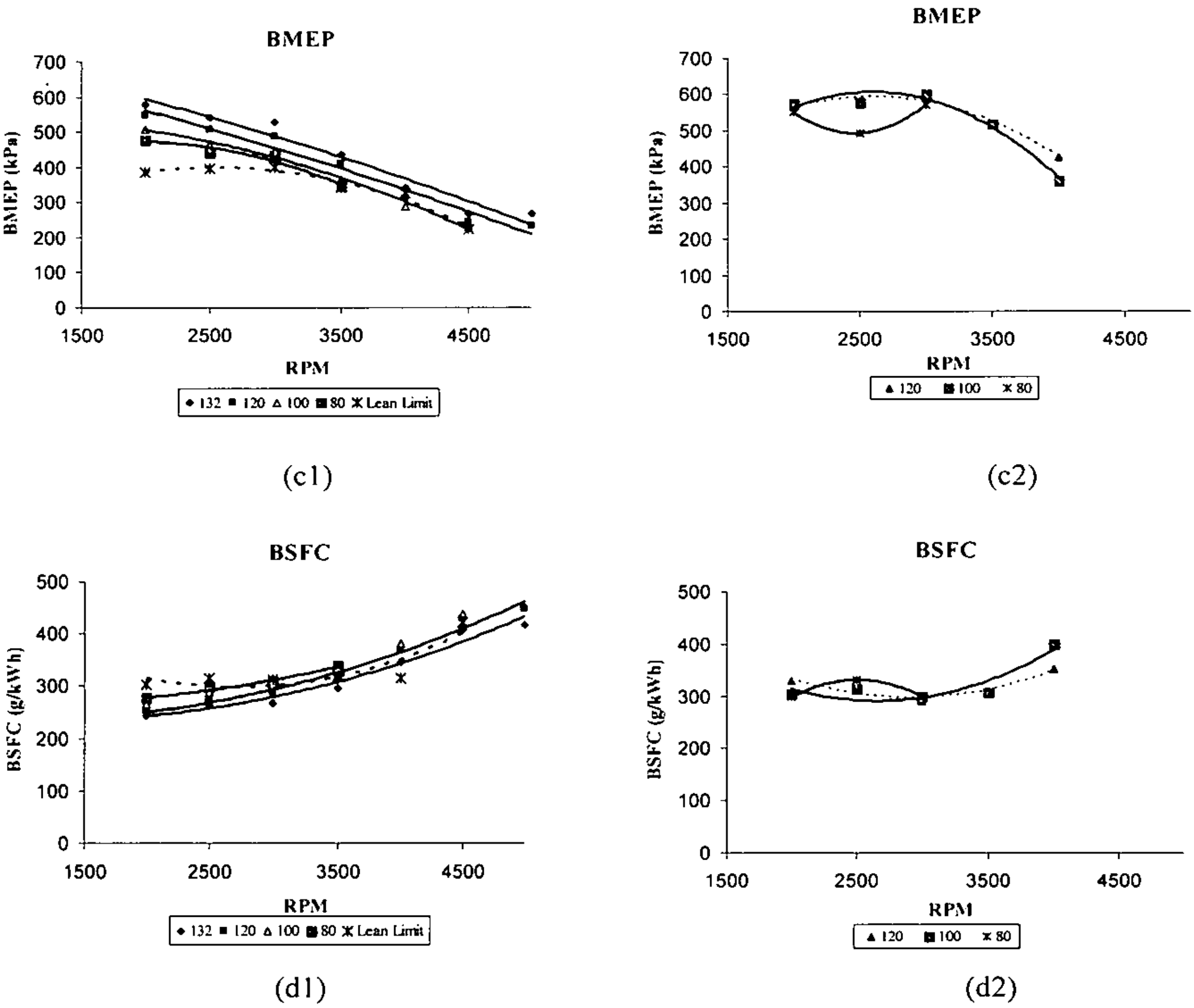


Figure 5-2 Engine Performance Characteristics for Different Injection Timings at Idle Loads for Stoichiometric and Lean Mixtures Respectively.

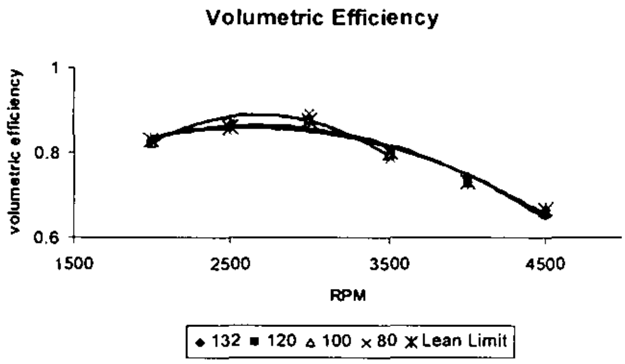
The torque curve are shown in the Figure 5-2 (a1) and (a2) for stratified and stoichiometric operations respectively. For the stoichiometric, torque curve shows an increasing trend till 3000 RPM and then starts to drop. Whereas the stratified operation shows a decreasing trend as the engine speed increases. For stoichiometric conditions the maximum torque is at 120 deg BTDC and 3000 RPM while for the stratified operation the injection timing at 132 deg BTDC i.e. when the intake valve closes gives the highest torque value, which also shows that the mixture formation and combustion are better at this injection timing. Moreover at stoichiometric conditions the operating range is limited as shown in the Figure 5-2 (a1).

Injection timing effect on power is illustrated in Figure 5-2 (b1) and (b2) for both the operations respectively. For stoichiometric case the power increases till 3500 RPM, with injection timing 120 deg BTDC and 100 deg BTDC having the similar trend. For stratified lean operation the power tends to increase till 3000 RPM, for every injection timing shown in the plot, yet the maximum value for each of these injection timings is lesser compared to the stoichiometric operation except at 132 deg BTDC, which shows an improvement of about 2-2.5% at 2000 RPM. The difference becomes quite obvious as the speed increases and the reaches about 20 % lesser performance at 4000 RPM for the stratified case.

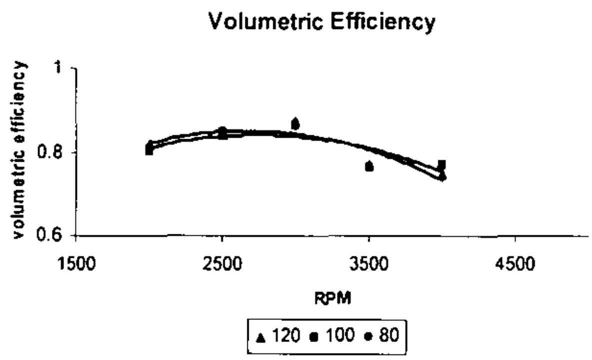
Brake Mean effective pressure for both the cases is shown in the Figure 5-2 (c1) and (c2) respectively. It shows a good relationship with torque curves. It has a maximum value of ~601 kPa and ~580 kPa for stoichiometric and stratified operations at 3000 and 2000 RPM respectively. The difference is only about 3 % higher for stoichiometric. For stratified case maximum value of BMEP at lean limit is ~400 kPa at an AFR of 46.

Figure 5-2 (d1) and (d2) represents the brake specific fuel consumption for stoichiometric and stratified lean operations respectively. Since stratified operation is best known for its fuel economy, it can been seen that even though there is a considerable reduction in power due to leaner operation, the BSFC values for stratified lean operation have lesser values than the stoichiometric ones. At 120 deg BTDC the stratified lean conditions have about 23% less specific fuel consumption compared to stoichiometric at 2000 RPM with the same injection timing. For stoichiometric case the BSFC changes are small and tend to increase after 3500 RPM but for the stratified case it starts to increase as the engine speed is increased. Yet it still is 10% lesser even at higher speeds of 4000 RPM. On average at all injection timings and speeds the brake specific fuel consumption for stratified charge operation gives a 4-10% improvement than its counterpart.

Volumetric efficiency of the engine is shown in the Figure 5-3 for stratified lean and stoichiometric operations respectively. In both cases there is not a considerable difference in the volumetric efficiency. As the throttle valve is fully closed for idle conditions, the volumetric efficiency is less than 90% for both cases. Overall volumetric efficiency is higher at lower speeds, where the maximum value is at about 3000 RPM for both operations. On further increasing the speed there is a reduction in volumetric efficiency which is directly affecting the performance of the engine as shown above. Moreover at higher speeds the time duration for the fuel air mixture preparation is less to have a homogeneous mixture which would enhance the combustion process. Lower volumetric efficiency may be the cause of lower performance at higher speeds.



(a)



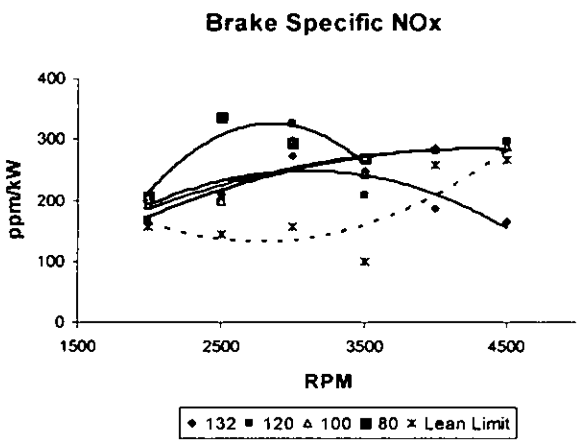
(b)

Figure 5-3 Volumetric Efficiency at Idle Loads with Different Injection Timings

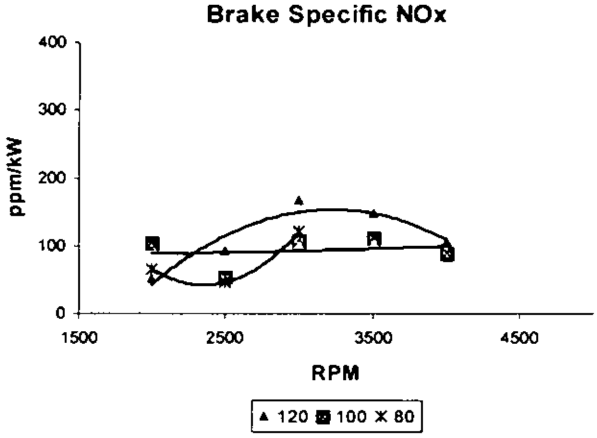
5.1.2 Engine Emissions at Idle Loads

Results of engine emissions with respect to injection timing are shown in the Figure 5-4. Brake specific emissions for stratified lean and stoichiometric operations are illustrated respectively. NOx emissions are illustrated in Figure 5-4 (a1) and (a2) respectively. For stoichiometric case the NOx emission for injection timing 120 deg BTDC and 100 deg BTDC are almost showing the same trends with a minor difference but for 80 deg BTDC the NO_X is comparatively low. For stratified lean operation the NOx tends to increase for every injection timing till 3000 RPM except for the lean limit case where each point is representing different injection timing and air/fuel ratio according to the engine speed. On average the NOx formation for stratified operation is 6-25 % higher than that of stoichiometric operation. The difference is minimal at 120 deg BTDC with only 1%. Higher NOx formation is due to the high combustion temperatures which also indicate the combustion levels. Since the NOx formation is slightly higher in stratified lean operation, it can be concluded that the average combustion efficiency would be higher.

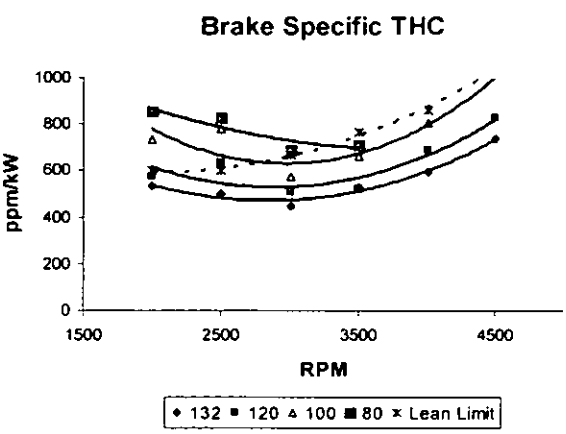
Stratified Lean **Stoichiometric**



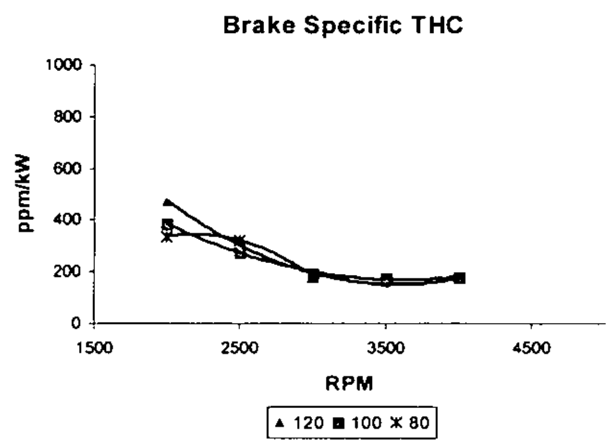
(a1)



(a2)



(b1)



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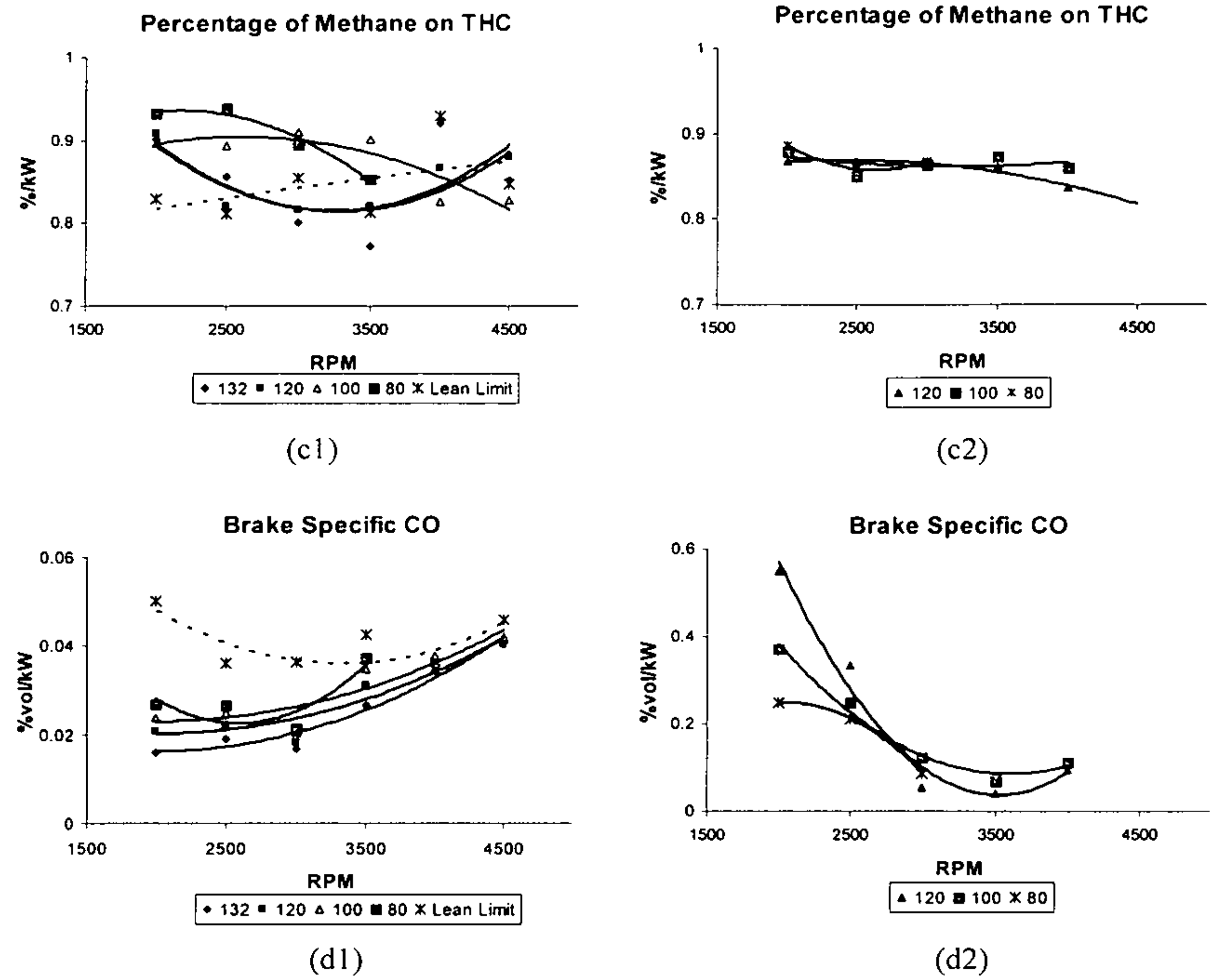


Figure 5-4 Emission Characteristics for Different Injection Timings at Idle Loads

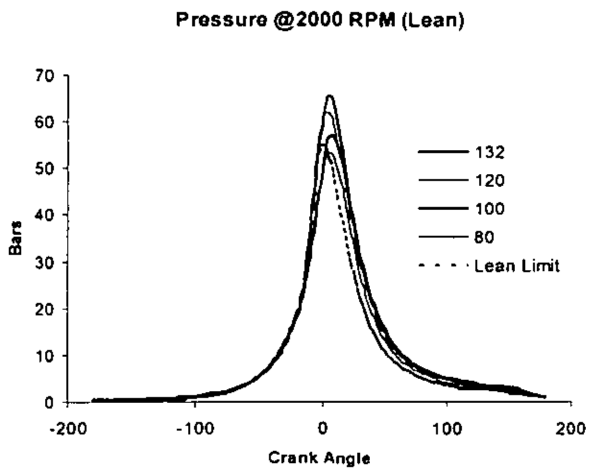
Figure 5-4 (b1) and (b2) illustrates the hydrocarbon emissions for lean and stoichiometric operations respectively. Stoichiometric operation has considerably lesser HC emissions than stratified lean operation at higher speeds. In stoichiometric case the HC emissions are higher at lower speeds and tend to decrease with the increase in the engine speed. Whereas in stratified lean case the HC emissions are lower at lower speeds and tends to increase after 3000 RPM. On average the HC emissions in stratified operation are about 17% more compared to stoichiometric at speeds up to 3000 and increase to about 50 % higher at higher engine speeds. In stoichiometric operation injection timing 120 deg BTDC and 100 BTDC showing almost the same trends, while 80 deg BTDC is having a higher value comparatively. In stratified lean operation the HC emissions tend to increase as the injection timing is further retarded. Injection timing 132 deg BTDC showing the lowest value at 3000 RPM compared to other injection timings. Lean limit curve also shows an increasing trend with the lowest value at 2000 RPM. The reason for higher THC emissions in stratified case might be due to high cycle to cycle variation, which might be due to improper mixing at higher speeds because of less availability of time and leaner mixture whose energy content would be lower than that of stoichiometric mixture. Moreover leaner mixtures are more prone to misfire and incomplete combustion in fraction of engine's operating cycles.

Percentage of methane on total hydrocarbon emissions is given in the Figure 5-4 (c1) and (c2). Methane emissions in lean stratified case are showing almost the similar trend as that of THC. For stoichiometric operation the average percentage of methane on total hydrocarbon emissions is above 80% and having a very minute difference, except at 120 BTDC at 4000 RPM and above. In stratified lean operation injection timing 120 deg BTDC and 132 deg BTDC show the same trends with 100 deg BTDC and 80 deg BTDC having higher methane emissions compared to former. On average the difference of methane emissions on total hydrocarbon emissions is 1-5% higher for lean operations.

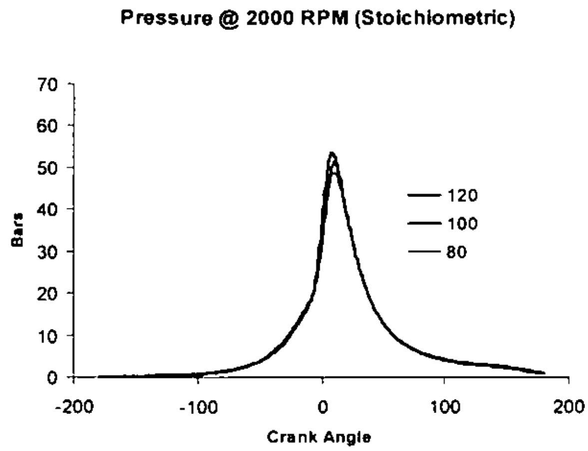
Carbon monoxide (CO) emission is represented in Figure 5-4 (d1) and (d2) for lean and stoichiometric conditions respectively. CO emissions are an order less for stratified lean operation compared to that of stoichiometric. In spark ignition engines CO emissions increase steadily as the equivalence ratio increases, as the amount of excess fuel increases. Whereas for lean mixtures it varies little with the change in equivalence ratio and is usually in the order of 10^{-3} mole fraction [74]. As diesel engines are always operated lean; the CO emissions from the diesel are low enough to be unimportant. Direct injection system with injection after the closing of intake valve closely resembles the diesel operation; therefore CO emissions are considerably lesser in leaner case compared to stoichiometric.

5.1.3 Engine Combustion at Idle Loads

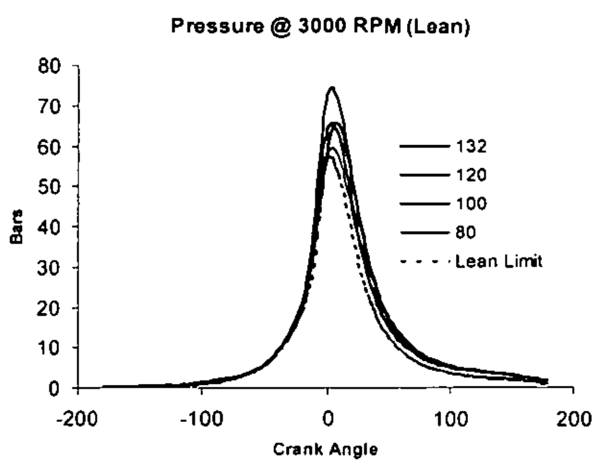
Pressure reading for various injection timings at idle loads and different engine speeds can be seen in Figure 5-5 for lean and stoichiometric operations respectively. Figure 5-5 (a1) and (a2) represents the pressure rise due to combustion at 2000 RPM engine speed for stratified and stoichiometric operations respectively. In both cases the highest pressure is at 100 deg BTDC and having values of ~65.5 and 53.5 bars respectively. Overall the combustion pressures for lean stratified operation are higher than stoichiometric at all injection timing for this engine speed. The reason might be due to higher charge stratification at lean conditions. Small fraction of fuel stratification in the mixture can increase the combustion process [36]. Although injection timings of 120 deg BTDC, 100 deg BTDC and 80 deg BTDC are investigated for both operations. Another reason might be the higher turbulence in the combustion chamber due to specially designed stratified piston which has been used for lean operations. Direct injection systems can generate turbulence inside the combustion chamber, while existence of turbulence in the cylinder flow can improve the combustion process [65]. On average lean stratified operations possess 27% higher peak pressures than stoichiometric. Figure 5-5 (b) and (c) illustrates the pressures at 3000 and 4000 RPM engine speeds for both operations. At 3000 RPM injection timing 120 deg BTDC is having the highest pressure in both scenarios, having ~74 and ~52 bars respectively. The difference is higher at this engine speed while injection timing is same. At 4000 RPM engine speed injection timing 80 deg BTDC is not possible for both lean and stoichiometric operations. At stoichiometric, injection timings 120 deg BTDC and 100 deg BTDC are having similar values i.e. ~43 bars. Pressure at 120 deg BTDC injection timing at lean conditions is about ~45 bars. Injection timings 132 deg BTDC and 100 deg BTDC having slightly higher values i.e. ~ 50 bars. At higher speeds it can be seen that there is a pressure drop. This pressure drop indicates the lesser combustion efficiency at higher speeds. At stratified lean operations this pressure drop is quite obvious compared to stoichiometric, though it drops in both operations. The reason might be due to lesser time available for the proper mixture formation at higher speeds. Cyclic variation is one of the main problems with lean burn combustion systems, as the mixture properties vary in every cycle [52]. Controlling the mixture formation in such systems influences to reduce cyclic variability hence resulting in a stable combustion process.



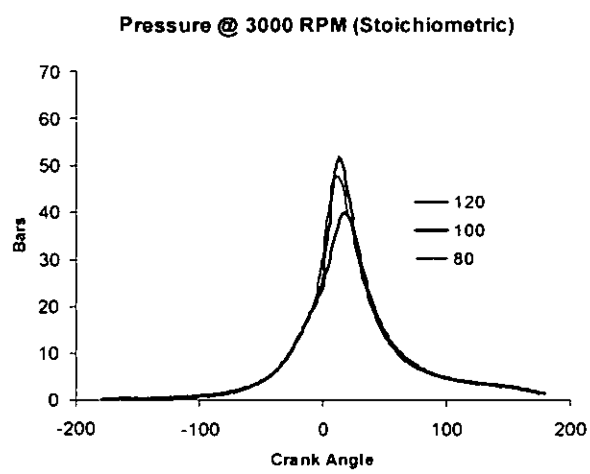
(a1)



(a2)



(b1)



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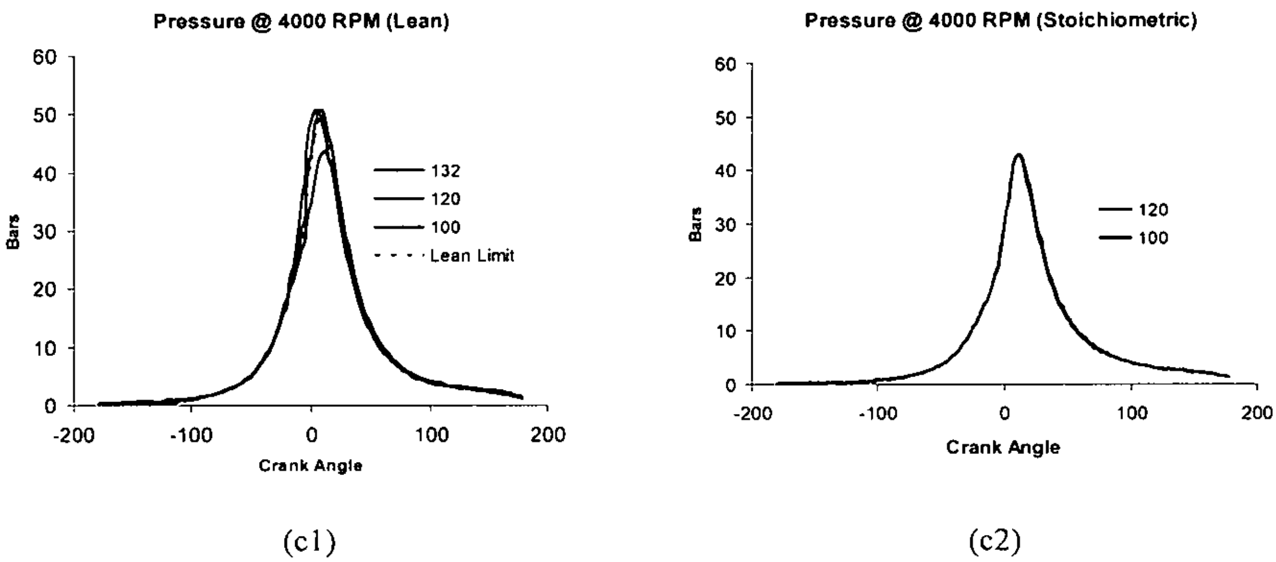
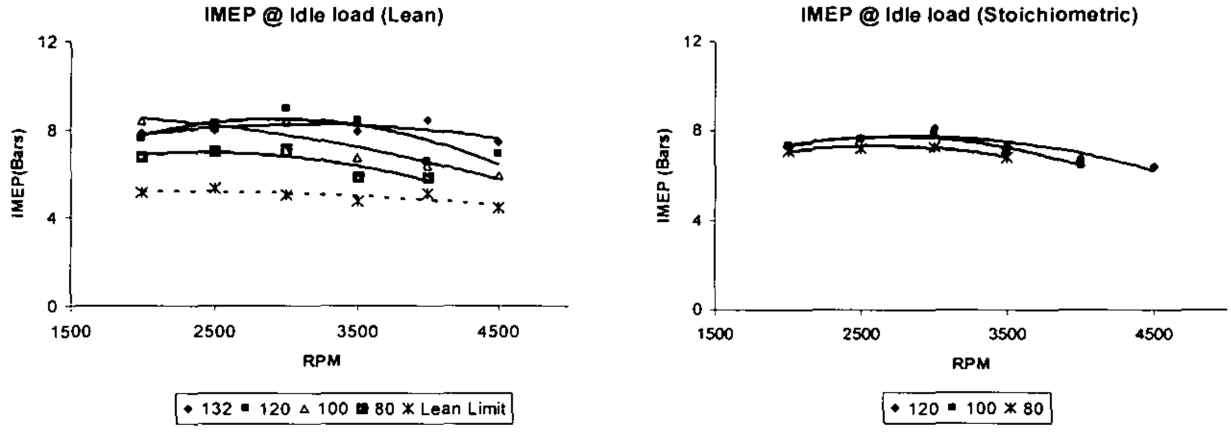


Figure 5-5 Pressure Reading Characteristics for Different Injection Timings at Lean Stratified and Stoichiometric Conditions

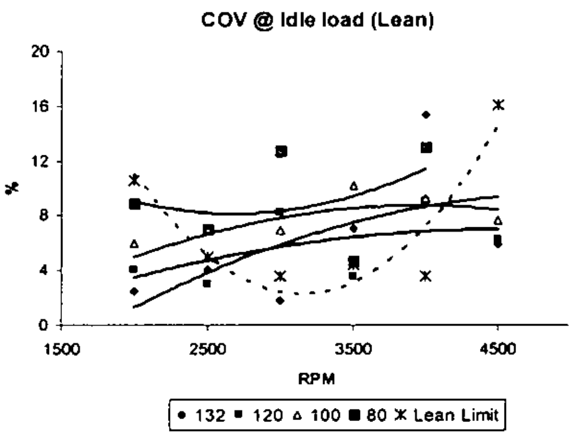
Figure 5-6 (a) and (b) shows the indicated mean effective pressure (IMEP) for stratified lean and stoichiometric operations respectively. IMEP is mainly the work transfer from the combustion to the piston. It is related to the combustion process, i.e. Higher IMEP values indicate good combustion process. Injection timings 120 deg BTDC and 100 deg BTDC at stratified lean conditions show higher IMEP values compared to stoichiometric, with the highest of 8.88 bars at 3000 RPM. Injection timing 120 deg BTDC having 3-14% improvement from 2000-3500 RPM, on the other hand 100 deg BTDC injection timing is having 4-14% improvement till 3000 RPM. At 100 deg BTDC, engine speeds higher than 3000, stoichiometric operation is having 2-4% higher IMEP values. Injection timings later than 100 deg BTDC, stoichiometric operation are having better values. At lean limits IMEP is having the highest value for the injection timings right after the closing of intake valve and at 2000 RPM. Lean limits are showing the lowest IMEP compared to other injection timings.



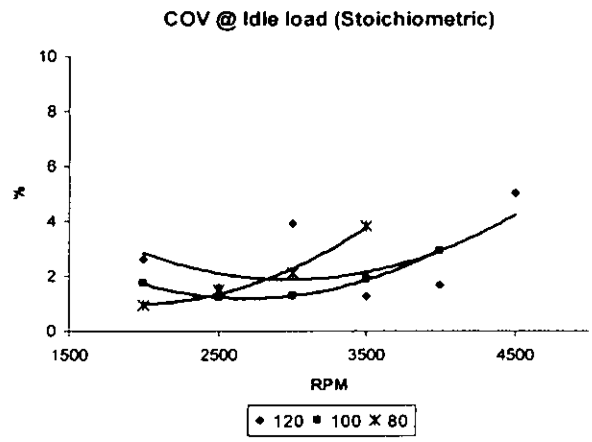
(a) (b)

Figure 5-6 IMEP Characteristics for Different Injection Timings

Coefficient of variation (COV) of IMEP is illustrated in Figure 5-7. Driveability of the vehicle is related to the COV. It is found that the COV values higher than 10% would give poor driveability. COV values for stoichiometric operations are having values lesser than 6% for all injection timings. Cyclic variation is one of the major draw backs in lean operating engines. COV values at lean stratified conditions showing higher values with more variation. Injection timings 132, 120, 100 deg BTDC are having values lesser than 10%, but higher compared to stoichiometric at same injection timings. Injection timings later than 100 deg BTDC showing even higher COV values and the highest is ~ 13% at 4000 RPM. COV at lean limits is having a different trend compared to other injection timings, with highest values at 2000 and 4000 RPM. Engine speeds within 2000-4000 RPM, the COV is well below 10%. Higher cyclic variation at lean conditions limits the engine operating range.



(a)



(b)

Figure 5-7 Cycle by Cycle Variation for Different Injection Timings

Combustion efficiency characteristics for different injection timing at idle loads are shown by Figure 5-8. Stratified and stoichiometric operations are showing different trends for the combustion efficiency. Combustion efficiency for lean stratified operation decreases with the increase in speed, whereas at stoichiometric operations it increases till 3000 RPM engine speed. At engine speeds lesser than 2500 RPM combustion efficiency for lean operation is having about 36% higher values than stoichiometric at injection timings 120 and 100 deg BTDC. Injection timing 120 deg BTDC is showing the highest values in both operations at 2000 and 3000 RPM respectively. At 3000 RPM the difference is least i.e. about 3% higher for stoichiometric case. Higher combustion efficiency at leaner conditions is due to higher amount of air available for combustion. With the increase of speed and higher cycle to cycle variation, less time available for mixture formation might be the causes of lower combustion efficiency at lean conditions. It can be seen that with the increase in engine speed and retarding the injection timing, combustion efficiency becomes lower, having lowest values at 4500 RPM. This result agrees with higher HC emissions at higher speeds and late injection timings at lean conditions.

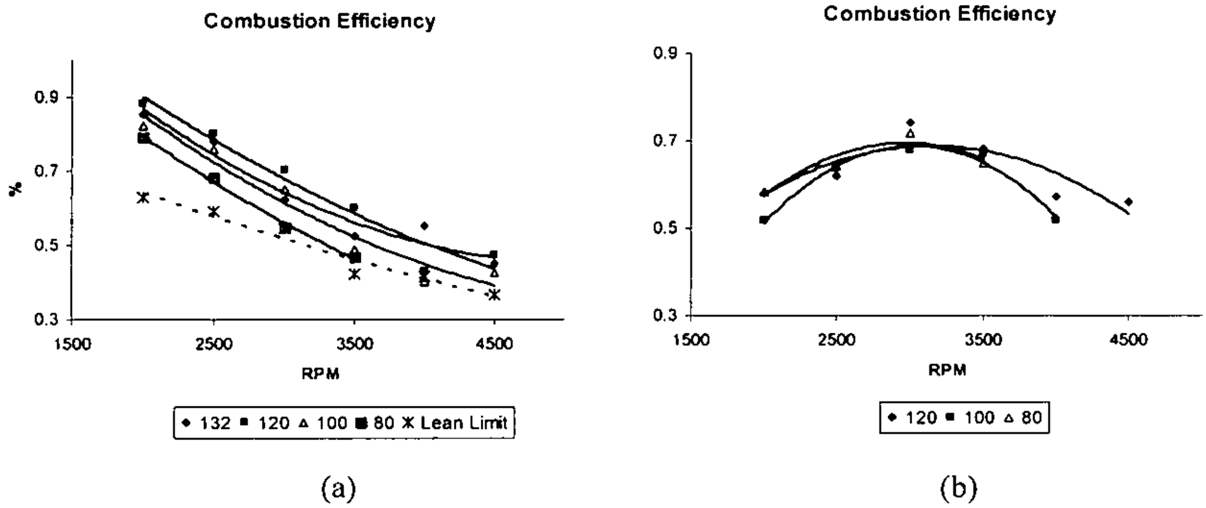
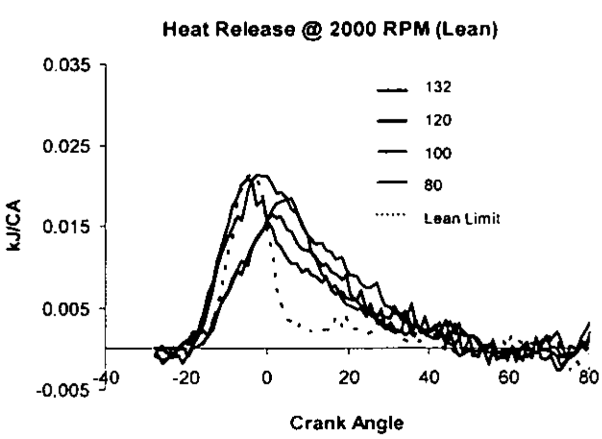
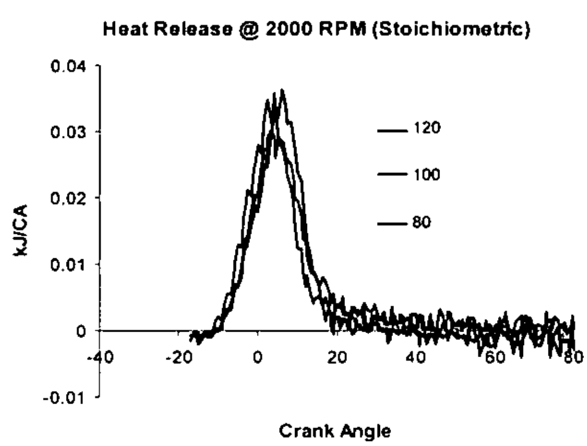


Figure 5-8 Combustion Efficiency for Different Injection Timings at Lean and Stoichiometric Conditions

Heat release characteristics for different injection timings at lean stratified and stoichiometric operations under idle loads are illustrated in Figure 5-9. Heat release for 2000, 3000 and 4000 RPM engine speeds are specified. Maximum heat release for lean stratified operation is obviously lower than stoichiometric due to less available fuel. Both operations are showing different heat release patterns under the given circumstances. In lean stratified operation, there is fast burn rate at the initial stage and slower burn at the later stage. Whereas in stoichiometric conditions there is a slightly slower burn at the initial stage and a moderately faster burn at the later stage. Injection timing 100 deg BTDC is having the highest value of about 0.027 kJ/CA at 2000 RPM for stratified operation, while highest value of 0.044 kJ/CA was experienced by injection timing 120 deg BTDC for stoichiometric. The initial fast burn rate can be observed that at lean stratified operations the highest heat release is reached 6-8 degrees BTDC for all injection timings except 132 deg BTDC, which is having it at about 1 degree BTDC. In stoichiometric conditions at 2000 RPM the highest point is reached during the expansion stroke, about 4-5 degrees ATDC. At 3000 RPM, similar behaviour can be noted for lean stratified operation, where maximum is reached about 4-6 degrees before the piston reaches the top dead centre. Injection timing 100 deg BTDC is having the highest value of about 0.032 kJ/CA, while for stoichiometric the highest is experienced at 120 deg BTDC i.e. 0.044 kJ/CA, and it occurs at about 4 degrees ATDC. 100 deg BTDC is having a relatively longer duration compared to other injection timings. A slightly different behaviour can be seen at 4000 RPM for lean stratified operation, where the maximum heat release occurs very near to top dead centre i.e. about 0.5-1 degrees. Injection timing 132 deg BTDC is having the highest heat release of about 0.022 kJ/CA, while injection timing 100 deg BTDC is having a slower heat release rate compared to others. At stoichiometric conditions, injection timings 120 deg BTDC and 100 deg BTDC are having similar trends; with 120 deg BTDC is having a higher heat release of 0.027 kJ/CA. Heat release patterns can explain combustion process. The faster initial combustion in lean stratified operation might be due to rapid burn of the initial mixture due to higher turbulence, while a slower burn in the later stage due to diffusion. In contrary to that in stoichiometric operations the initial burn is slightly slower, due to moderately strong turbulence and a faster burn due to moderately proceeding mixture. Thus the main effect of fuel injection timing can be explained by the fuel air mixing and the turbulence produced. Furthermore it can be said that fuel injection timing can be an important parameter to control the combustion characteristics in natural gas direct injection systems.



(a1)



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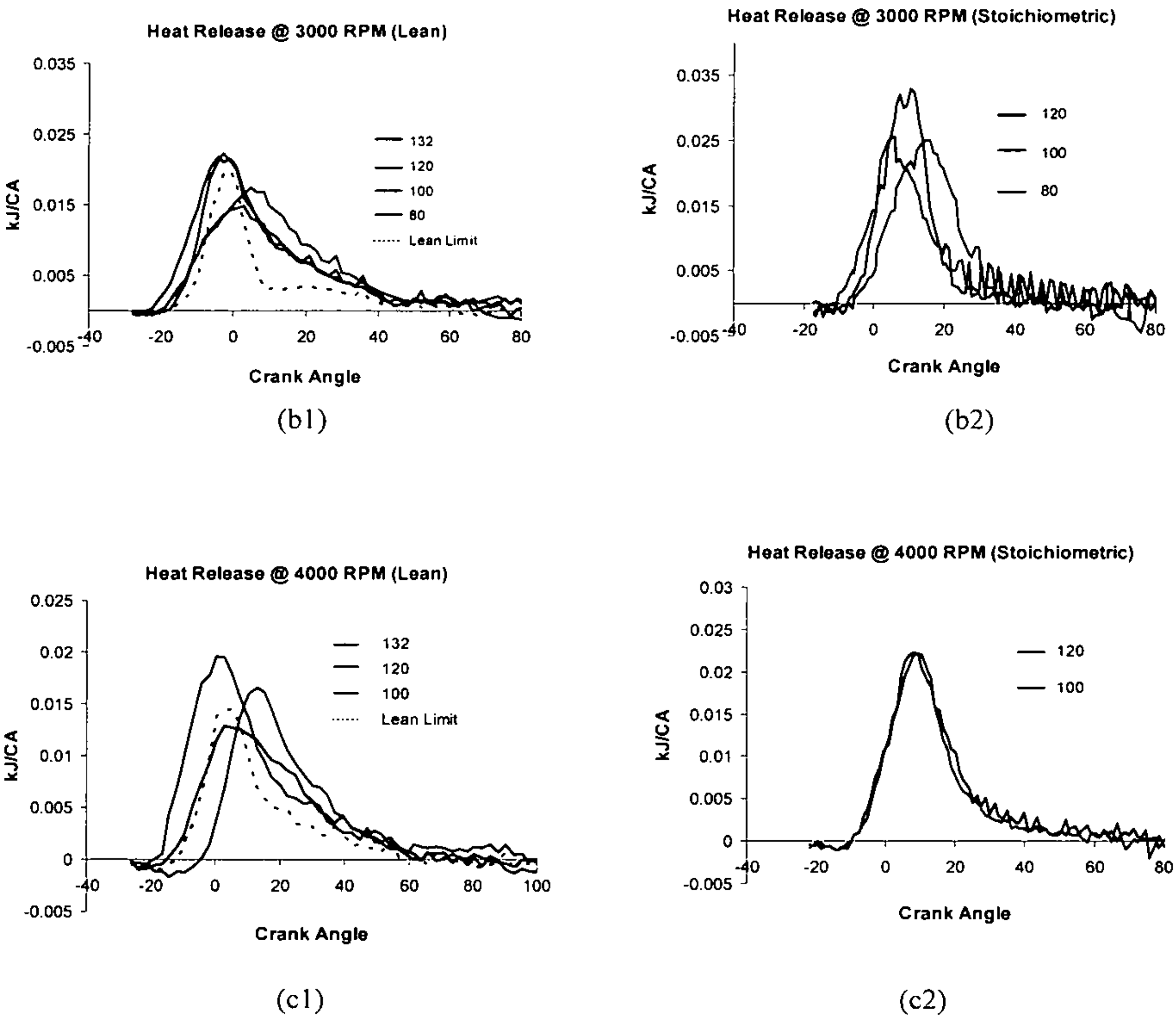
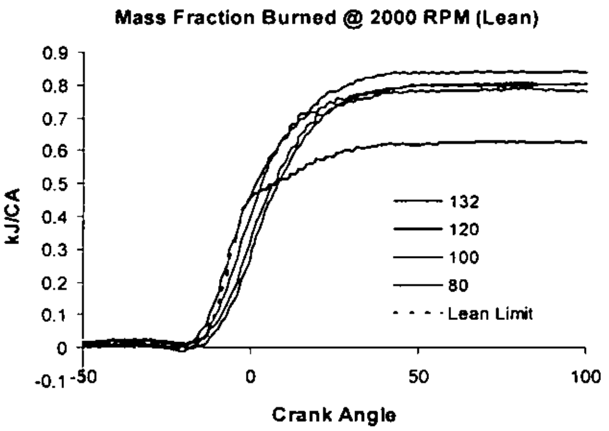
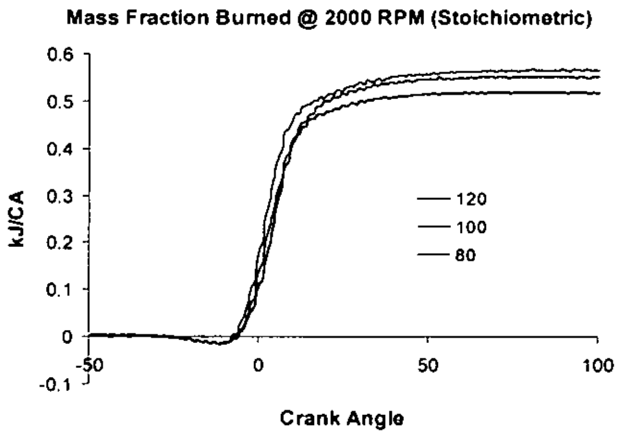


Figure 5-9 Heat Release Characteristics for Different Injection Timings at Lean and Stoichiometric Conditions

Mass fraction burned with respect to injection timings at idle loads for stratified and stoichiometric operations are given in Figure 5-10. As explained earlier that the combustion rate in the initial stage is faster for lean stratified operation, similar behaviour can be seen in the mass fraction burned. The intermediate and later stages of combustion are faster for stoichiometric operation for engine speeds less than 3000 RPM. At 4000 RPM they are having a slightly different behaviour. At 2000 RPM, for lean stratified, injection timing 120 deg BTDC is having the fastest and burn rate, while the rest are having almost similar trends. A slower rate in the initial burn stage is experienced at this engine speed for stoichiometric, while the intermediate and the final stages are faster compared to that of lean stratified. Injection timing 100 deg BTDC is having higher slope and higher mass fraction burned compared to 120 and 80 deg BTDC. At 2000 RPM mass fraction burned for lean operation is higher than stoichiometric. At 3000 RPM engine speed, injection timing effect can be seen in the Figure 5-10 (b). Unlike at 2000 RPM, injection timing effect is quite obvious at this engine speed. Injection timing 132 deg BTDC is having the slowest burn rate compared to others, while injection timing 120 deg BTDC is yet having overall faster and higher mass fraction burned at lean stratified conditions. Lean limits is having the lowest mass fraction burned, but the initial burn rate is similar to that of 120 deg BTDC as shown in the Figure 5-10, for different engine speeds. For stoichiometric, the longest duration is experienced by 100 deg BTDC, and the shortest is at 120 deg BTDC. Mass fraction burned for both operations are having similar values for injection timings 120 and 100 deg BTDC. Injection timings 132, 80 and lean limit are having lower mass fraction burned at this engine speed compared to stoichiometric. At 4000 RPM, injection timings 120 and 100 deg BTDC for stoichiometric are having similar behaviour with 120 deg BTDC having higher rate of mass fraction burned. At lean operating conditions injection timing 132 deg BTDC is having higher rate, and shortest duration. Injection timings 120, 100 and lean limit are experiencing relatively longer duration with lower values of mass fraction burned. Overall the lean stratified operations is having a shorter duration for the initial combustion for all injection timings, and a slightly longer towards the end; whereas for stoichiometric, the initial stage is slower but the duration is shorter towards the end. As explained earlier that this might be due to higher turbulence and rapid combustion at the initial stage for lean operations, where there is relatively a richer mixture present near the spark plug. Due to lesser fuel and much leaner conditions in the rest of the cylinder, the flame velocity is much lower towards the end. This phenomenon is more obvious at higher engine speeds where there is less heat release and slower burning, due to less time available for the proper mixture formation, which is effecting combustion and hence the performance.



(a1)



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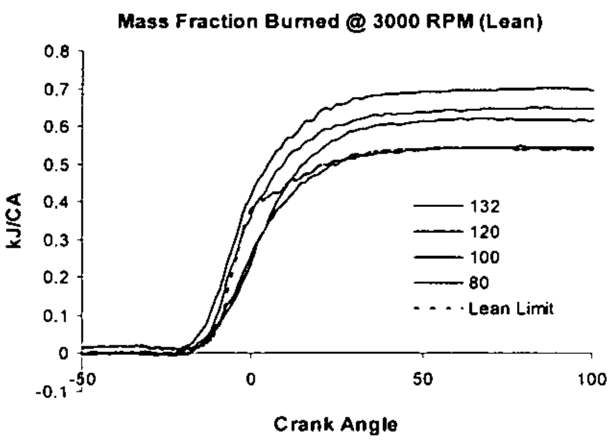

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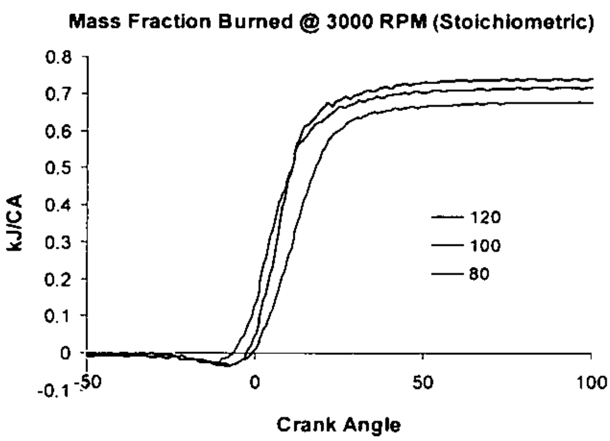

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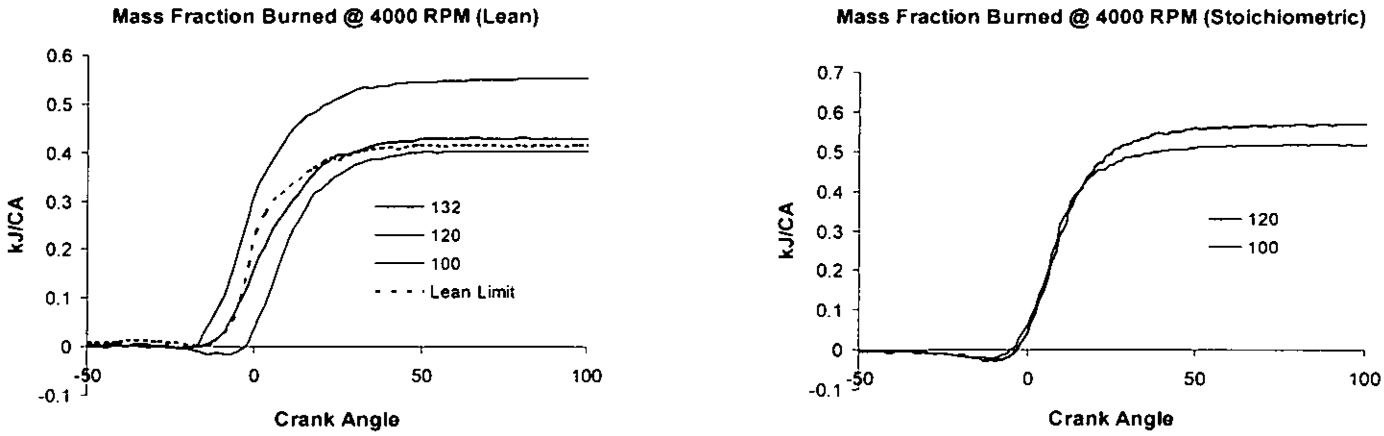


(c1) (c2)

Figure 5-10 Mass Fraction Burned for Various Injection Timings at Idle Loads.

Summary:

Injection timing effect is investigated for lean stratified and stoichiometric homogeneous operations with CNG-DI spark ignition engine at idle and partial loads. At idle loads, with lean stratified operation, torque tends to drop with the increase in engine speed for all injection timings. Whereas at stoichiometric conditions, it increases till 3000 RPM and then drops, moreover at stoichiometric conditions the operating range is limited with the retarding of injection timing. Overall stratified operation is showing an improvement in the performance of about 2-2.5% at 2000 RPM, but with increase in the engine speed it drops to 20% lesser at 4000 RPM. Injection timing 132 deg BTDC is having the highest overall performance for lean stratified operation. Stratified operation is known for the fuel economy, it can be seen that for lower engine speeds i.e. less than 3000 RPM, brake specific fuel consumption for lean stratified operation is lesser than stoichiometric for all injection timings. For instance, at 2000 RPM, injection timing 120 deg BTDC is having 23% lesser BSFC than at stoichiometric. Even at engine speeds of 4000 RPM, the BSFC for lean stratified operation is about 10% lower than at stoichiometric, in spite of having lower power.

On average the NOx formation for stratified operation is 6-25 % higher than that of stoichiometric operation. The difference is minimum at 120 deg BTDC with only 1%. Higher NOx formation is due to the high combustion temperatures which also indicate the combustion levels. In stoichiometric case the HC emissions are higher at lower speeds and tend to decrease with the increase in the engine speed. Whereas in stratified lean case the HC emissions are lower at lower speeds and tends to increase after 3000 RPM. The faster initial combustion and the slower late combustion stages might be responsible for such behaviour. On average the HC emissions in stratified operation are about 17% more compared to stoichiometric at speeds up to 3000 and increase to about 50 % higher at higher engine speeds. CO emissions are an order less for stratified lean operation compared to that of stoichiometric for all injection timings.

Overall the combustion pressures for lean stratified operation are higher than stoichiometric at all injection timings and engine speeds, at idle loads. The reason might be due to higher charge stratification at lean conditions. Another reason might be the higher turbulence in the combustion chamber due to specially designed stratified piston which has been used for lean operations. Maximum heat release for lean stratified operation is obviously lower than stoichiometric due to less available fuel. Both operations are showing different heat release patterns under the given circumstances. In lean stratified operation, there is fast burn rate at the initial stage and slower burn at the later stage. Whereas in stoichiometric conditions there is a slightly slower burn at the initial stage and a moderately faster burn at the later stage. The faster initial combustion in lean stratified operation might be due to rapid burn of the initial mixture due to higher turbulence, while a slower burn in the later stage due to diffusion. In contrary to that in stoichiometric operations the initial burn is slightly slower, due to moderately strong turbulence and a faster burn due to moderately proceeding mixture. Thus the main effect of fuel injection timing can be explained by the fuel air mixing and the turbulence produced.

5.2 Injection Timing Effects on Engine Performance at Partloads

Effect of injection timing on engine performance at partloads when the throttle valve is half opened, is illustrated in the Figure 5-11. Data for stoichiometric and lean operations are plotted together. For stratified lean operation the air/fuel ratio is kept at 36.3- 37 whereas for homogenous case it is kept near stoichiometric. Injection timing is set to start after the closing of intake valve at 132 deg BTDC for stratified lean operation and at 120 deg BTDC for stoichiometric respectively. Injection timing is retarded till it is limited by

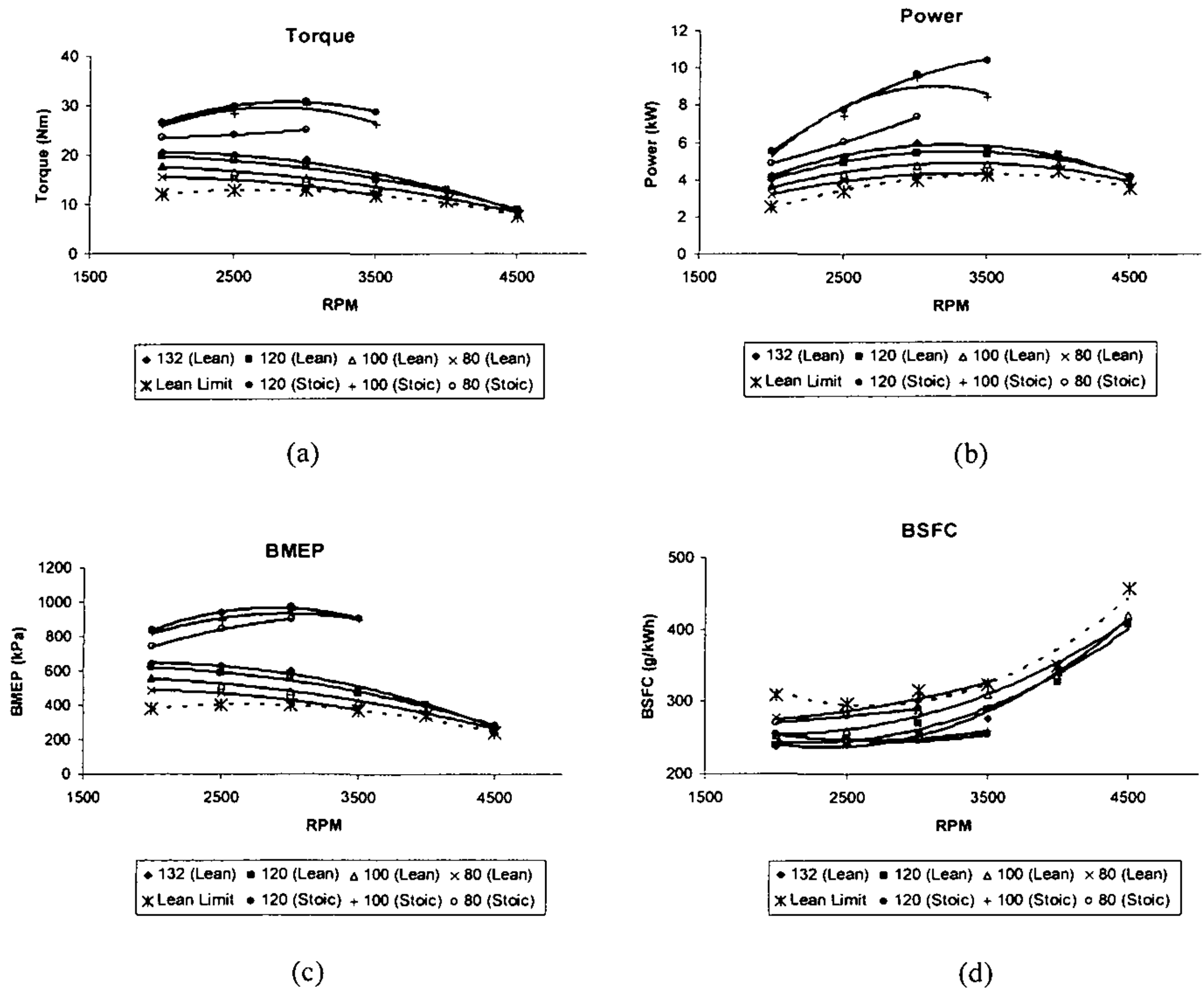


Figure 5-11 Engine Performance Characteristics for Different Injection Timings at Partloads

the ignition timing parameter. Lean limits for the stratified operation are also included in the result for illustration. Performance parameters such as Torque, Power, Brake mean effective pressure (BMEP) and Brake specific fuel (BSFC) consumption are shown in the Figure 5-11.

Unlike idle loads where the torque has an improvement of 2-2.5% for lean conditions at 132 deg BTDC, at partloads the stoichiometric values for torque are much higher comparatively as shown in the Figure 5-11 (a). At 120 deg BTDC and at 2000 RPM the maximum value of torque for stratified is 19.7 Nm compared to stoichiometric which has 26.6 Nm. Moreover for stoichiometric the torque tends to increase with the increase in the engine speed for every injection timing till it is limited by the operating range. Whereas for stratified lean operation the torque for every injection timing has different values and tends to maintain till 3000 RPM and then decreases i.e. with the retarding of injection timing the torque drops.

Injection timing effect on power is shown in Figure 5-11 (b). For stoichiometric operation the power plot shows an increasing trend with the increase in engine speed, but the operation is limited till 3500 RPM due to the ignition parameter limitation. For stratified lean operation the power tends to increase till 3000 RPM and decreases with further increase in the engine speed. Moreover the average power drop is about 25 % at 2000 RPM compared to stoichiometric and keeps on decreasing as the speed increases. As lean mixture is having lesser fuel, hence less energy content and lower power is produced.

Figure 5-11 (c) illustrates the brake mean effective pressure (BMEP). It shows a good relationship to torque for both stoichiometric and lean cases. Brake mean effective pressure for stratified lean operation decreases with the increase of pressure, while for stoichiometric case it increases till 3000 RPM. The maximum values for BMEP is ~969 kPa and ~642 kPa for stoichiometric and lean operations respectively, the difference is about 33%. Brake specific fuel consumption (BSFC) is illustrated in Figure 5-11 (d). Stoichiometric operations is showing similar BSFC values for injection timings 120 deg

BTDC and 100 deg BTDC, but 80 deg BTDC is showing some what higher. For stratified lean operation the BSFC values showing almost similar trend to that of stoichiometric till 3000 RPM and increases after that. The increase is due to lesser power produced due to lean mixture. Even though the power produced is lesser at lower speeds also, BSFC values for stratified operation are still lower compared to stoichiometric. On average at 120 deg BTDC the BSFC values are 1-14% lower from 2000 to 3500 RPM, at 100 deg BTDC it is 5-19 % lower and for 80 deg BTDC it is 1-5% lower. At lean limit the average BSFC is about 348g/kWh. The higher value for BSFC at lean limit is due to very less power production. Apart from lean limit, the stratified lean operation gives 3-10 % improvement on BSFC on average within the compared operating range.

Volumetric efficiency of the engine at partloads when the throttle valve is half open is given in the Figure 5-12, for both stoichiometric and lean conditions. The maximum is at about 3000 RPM for both cases, with lean operation having a bit higher value at speeds lesser than 3000 RPM. At partloads volumetric efficiency is higher compared to idle load conditions. At higher speeds with late injection stoichiometric operation is limited as shown in the Figure. Late injection shows higher performance for both cases because of the higher volumetric efficiency with lean operation having lesser performance due to lesser fuel. Higher performance at lower speeds is may be due to high volumetric efficiency. At higher speeds there is less time available for the proper air fuel mixing and lower volumetric efficiency are affecting the performance.

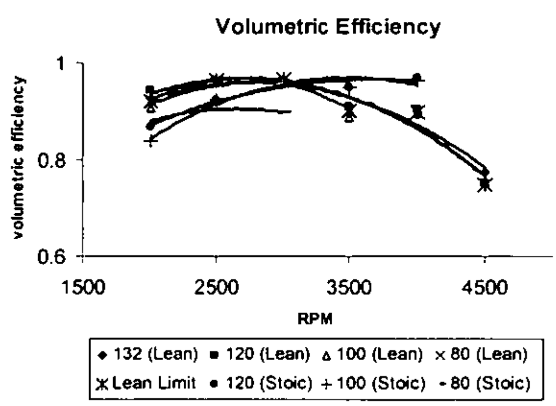


Figure 5-12 Volumetric Efficiency at Partloads

5.2.1 Engine Emissions at Partloads

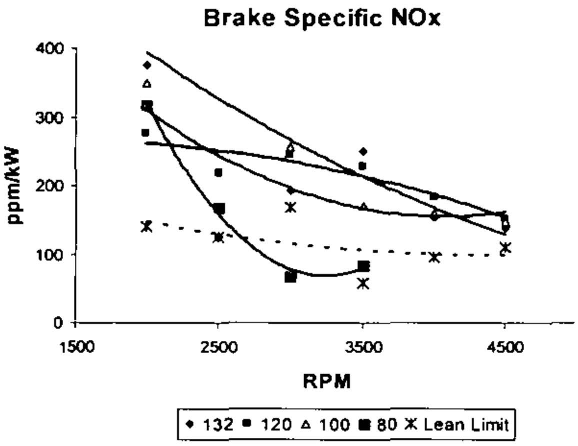
Engine out emissions at partloads for both stoichiometric and lean operations are illustrated in Figure 5-13. Brake specific NOx emissions are shown in Figure 5-13 (a1) and (a2) respectively. Nitrogen oxides (NOx) emission for lean stratified operation is higher at lower engine speeds which indicate higher temperatures of combustion, for all injection timings except the lean limit where it is lesser. As the speed increases it shows a decreasing trend. For stoichiometric operation, at 120deg BTDC NOx emissions show almost a constant trend with the increase in engine speed. At 2000 RPM its having about 15% lower NOx than lean operation, but with the increase of speed this difference becomes 1.3% at 3500 RPM. Other injection timings 100 deg BTDC and 80 deg BTDC are showing a decreasing trend but the operation is limited to lower speeds. At these injection timings the NOx emission is about 23 % and 56% higher at 2000 RPM for lean operations. For engine speeds above 3000 RPM nitrogen oxide emission for both cases are having lesser difference.

Hydrocarbon emissions show a distinctive behaviour as shown in the Figure 5-13 (b1) and (b2). For stoichiometric operation the hydrocarbon emissions decrease with the increase in the engine speed and follow a constant trend after 3000 RPM. Injection timings 120 deg BTDC and 100 deg BTDC have similar trends, but 80 deg BTDC is having higher values comparatively. In case of lean stratified operation the brake specific hydrocarbon emissions is higher, reason being the lesser power at leaner conditions and also at higher speeds the combustion is effected by the mixture formation as well as the amount of the fuel. At lean conditions it is showing an increasing trend with the increase in speed and with retarding the injection timing. Moreover 132 deg BTDC and lean limit is having similar trends with slightly higher values at lean limits. 80 deg BTDC is showing an opposite trend, as hydrocarbon emissions are decreasing with the increase of engine speed. But brake specific value is higher compared to other injection timings, which shows that at lean conditions with the retardation of the injection timings while keeping the air fuel ratio constant, hydrocarbon emissions increase. On average the unburned hydrocarbon emissions for lean stratified operation is higher compared to stoichiometric at same injection timings.

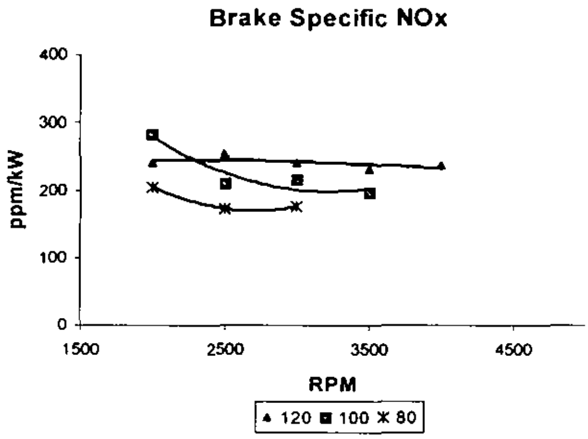
Figure 5-13 (c1) and (c2) shows the percentage of methane in unburned hydrocarbons. For stoichiometric operation the percentage of methane in total unburned hydrocarbons is within 80-90%. In stratified lean operation the methane percentage is showing some what different behaviour. Injection timings 120 deg BTDC and 100 deg BTDC having similar trends and the range is within 80-90%. Injection timings 132 deg BTDC and 80 deg BTDC exhibit more than 90% at engine speeds less than 2500. Moreover at lean limits the percentage of methane is showing same trends as that of 132 deg BTDC. At higher speeds the percentage of methane in total unburned hydrocarbons decreases to less than 80%. On average at lean stratified conditions the percentage of methane is 3-5 % higher for injection timings 120 deg BTDC and 80 deg BTDC. At 100 deg BTDC it is having 2% lesser methane in total hydrocarbons.

Figure 5-13 (d1) and (d2) illustrate the carbon monoxide emissions at partloads for lean and stoichiometric operations respectively. At lean conditions carbon monoxide emissions possess a different behaviour than at idle load conditions. For stoichiometric conditions CO emissions are showing similar trend as that of idle loads, but are less at partload condtions. At lean stratified conditions injection timings 132, 120 and 100 deg BTDC are having similar trends and the amount of CO is less than 0.05%vol. Unlike idle loads injection timing 80 deg BTDC having higher CO but still lesser than 0.1% vol. At lean limits, CO emission is higher than stoichiometric for engine speeds less than 3000 RPM. At engine speeds more than 3500 RPM, CO emissions for stoichiometric and lean stratified operation are having lesser difference, yet stoichiometric operation is limited by the engine speed.

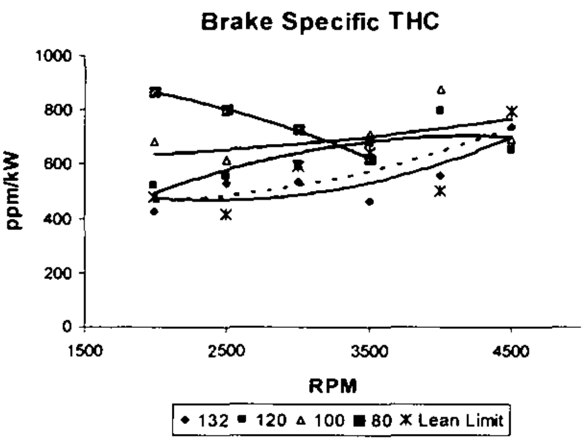
Stratified Lean | **Stoichiometric**



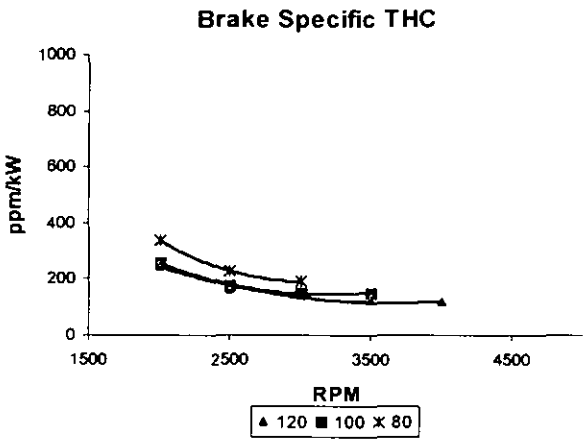
(a1)



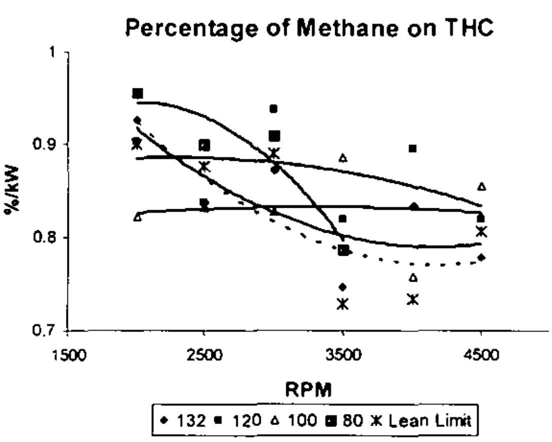
(a2)



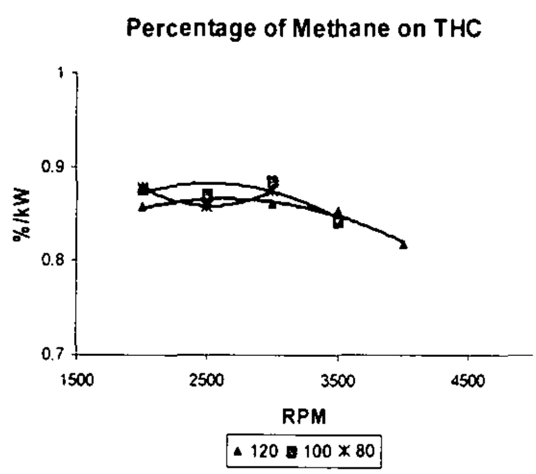
(b1)



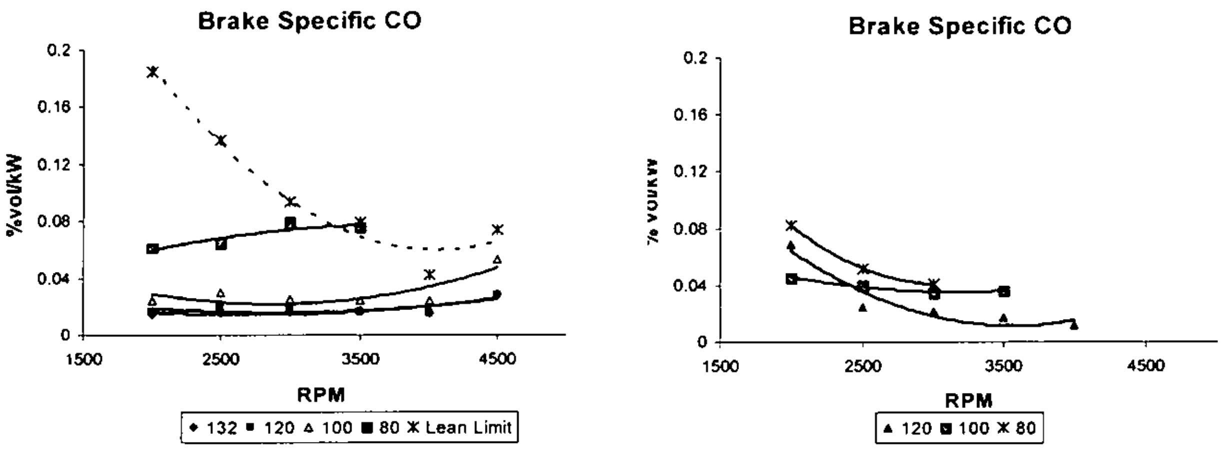
(b2)



(c1)



(c2)



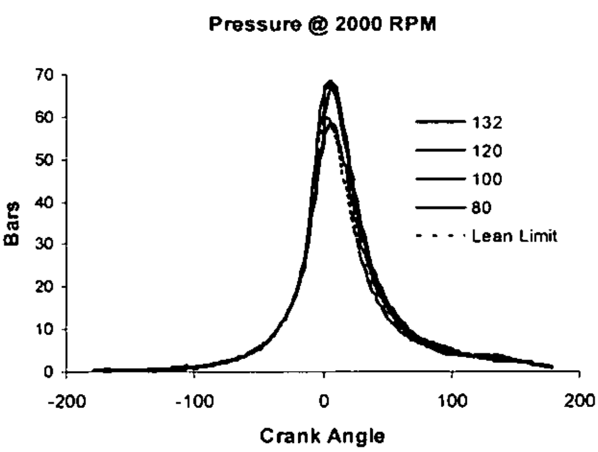
(d1) (d2)

Figure 5-13 Emission Characteristics for Different Injection Timings at Partloads

5.2.2 Engine Combustion at Partloads

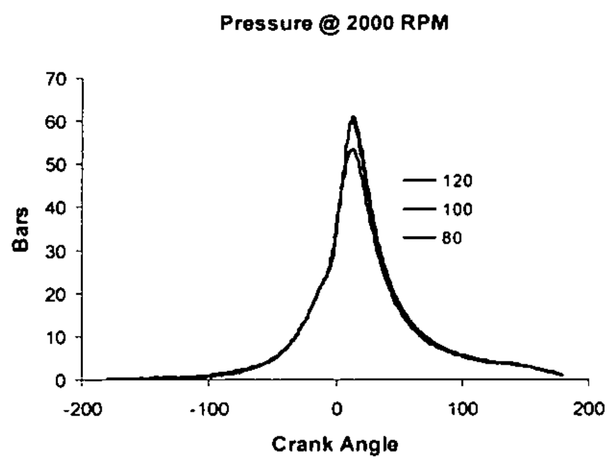
Pressure reading for various injection timings at partloads and different engine speeds can be seen in Figure 5-14 for lean and stoichiometric operations respectively. Figure 5-14 (a1) and (a2) represents the pressure at 2000 RPM for lean and stoichiometric operations respectively. Pressure rise in lean operation is faster compared to stoichiometric as it is having higher gradient comparatively. For stoichiometric operation the maximum pressure is reached at about 10-13 degrees after the top dead centre (ATDC) for all injection timings. Maximum pressure is reached earlier in lean conditions, and is about 3-6 degrees ATDC. Unlike idle loads the pressure difference is lesser, with highest at 120 deg BTDC for both operations having ~ 68 and ~61 bars respectively. Faster pressure rise indicates the faster combustion at lean conditions. Lowest pressure is exhibited by 80 degree BTDC (~ 58 and ~ 53 bars respectively). On average at 2000 RPM engine speed and 50% open throttle, pressure rise at lean operating conditions is about 9-13% higher for respective injection timings. Figure 5-14 (b1) and (b2) illustrates the pressure at 3000 RPM for lean and stoichiometric operations respectively. At 3000 RPM stoichiometric operation is exhibiting higher pressures compared to lean. Pressure rise is again observed as rising at a faster rate compared to stoichiometric, reaching maximum at about 6.5 degrees after top dead centre, whereas the maximum pressure occurs at about 12 degrees ATDC in stoichiometric case. Injection timing 120 degree BTDC at stoichiometric conditions is having about 10.7% higher peak pressure. Whereas in stratified lean operation the maximum (~ 71) is at 132 degree BTDC i.e. after the closing of the intake valve. 100 and 80 deg BTDC is having about 9.4 and 1.1% lower peak pressures compared to stoichiometric. However the operating range for injection timings 100 degree BTDC and 80 degree BTDC are limited to 3500 RPM.

Stratified Lean

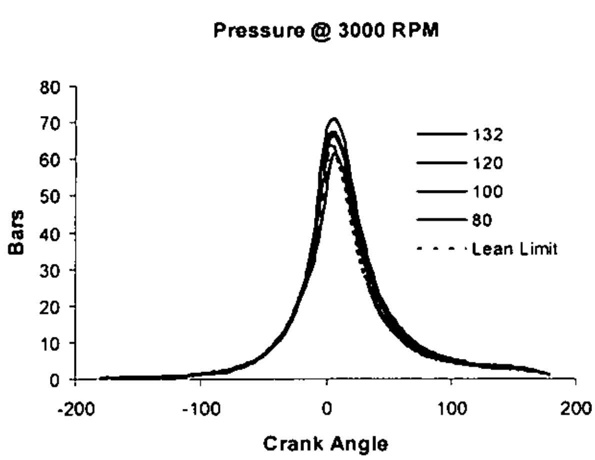


(a1)

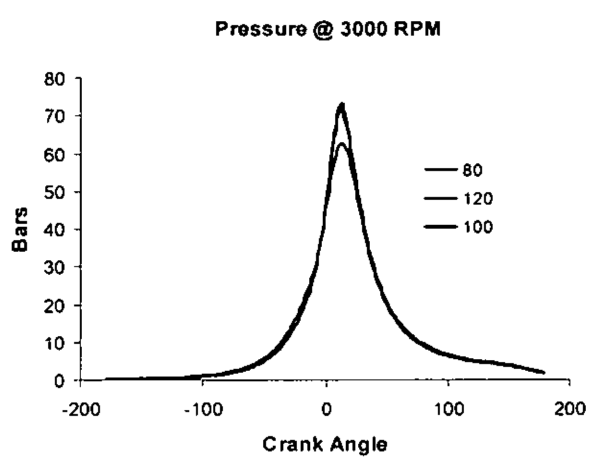
Stoichiometric



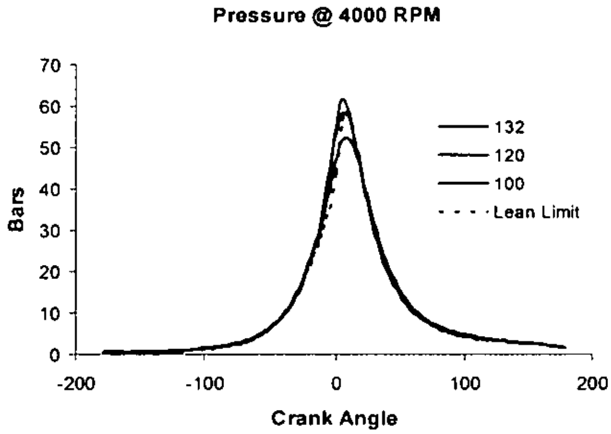
(a2)



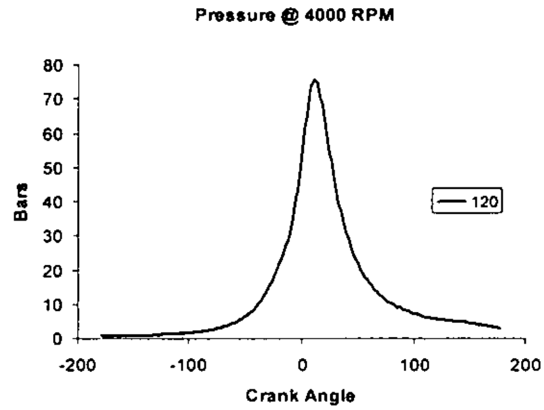
(b1)



(b2)



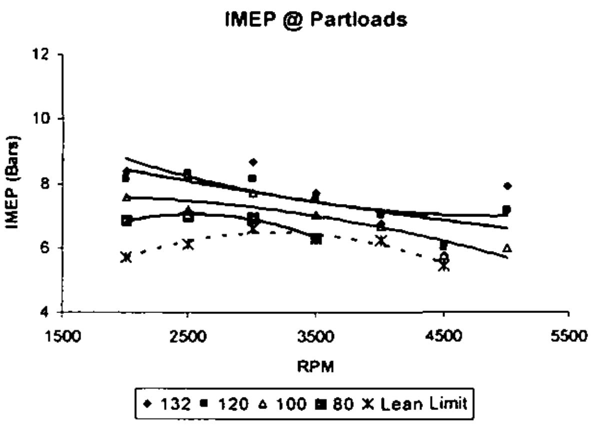
(c1)



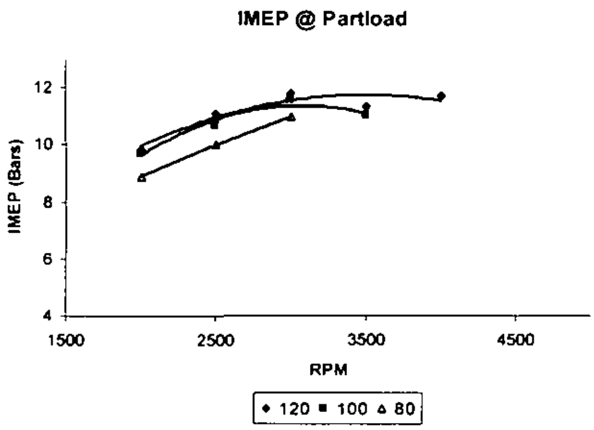
(c2)

Figure 5-14 Pressure Characteristics for Different Injection Timings at Partloads

Figure 5-15 (a1) and (a2) gives the indicated mean effective pressure at partloads i.e. when the throttle is half open, for lean and stoichiometric operations respectively. The Figure explains the effect of injection timing on IMEP. At partloads the IMEP for stoichiometric operations are having higher values than lean. An increasing trend is observed till 3000 RPM engine speed for stoichiometric operation, for all injection timings. At lean conditions the IMEP tends to decrease with the increase in engine speed. However stoichiometric operation is limited for injection timings as shown in the Figure. At lean conditions, IMEP decreases with retarding the injection timings. 132 deg BTDC and 120 deg BTDC are having similar trends and lesser difference compared to late injection timings. Highest value (~ 8.6 bars) is obtained at 132 deg BTDC at 3000 RPM. At stoichiometric conditions the highest value of ~ 11.75 bars is observed at 3000 RPM with 120 deg BTDC injection timing. 120 deg BTDC is having 17- 39% higher IMEP values compared to lean operations. Injection timings 100 and 80 deg BTDC also having an improvement of 21-36% over lean operating conditions. This clearly indicates that at partloads the stoichiometric operation is better than lean operations. On average a decrease of about 30% is observed in IMEP at lean conditions compared to stoichiometric, at respective injection timings.



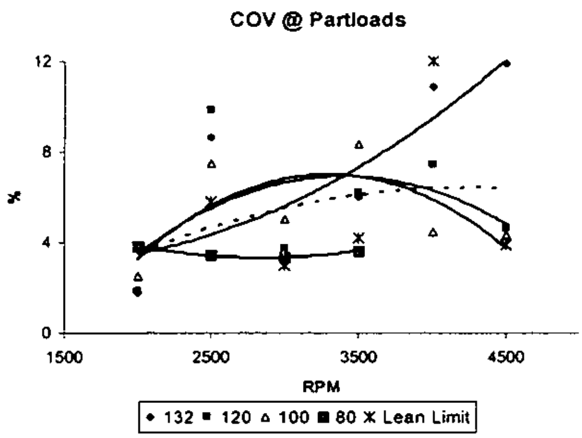
(a1)



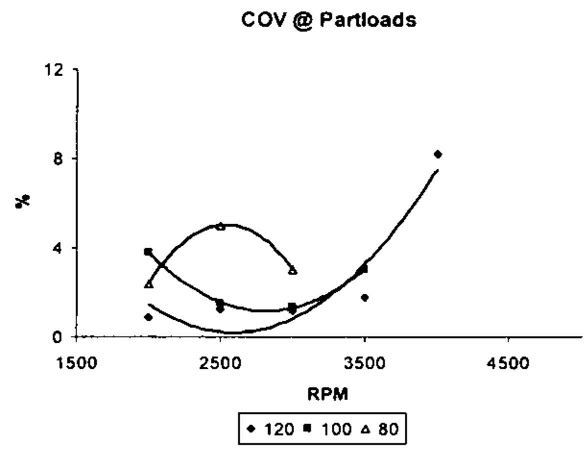
(a2)

Figure 5-15 IMEP Characteristics for Different Injection Timings at Partloads

Coefficient of variation (COV) in indicated mean effective pressure is given in Figure 5-16, for lean stratified and stoichiometric operations respectively. It is already explained that COV is directly related to the driveability, and is expected to be lower than 10%. COV for stoichiometric operations is lesser than 5% for all injection timings and engine speeds, except at 120 deg BTDC and 4000 RPM, yet it is lesser than 10. This shows a better mixture formation and hence better combustion process. COV values for stratified lean operation are shown in Figure 5-16 (a1). Values are much scattered and in some cases is above 10, which indicates the operating limit at lean conditions. Injection timings 120 and 100 deg BTDC is having similar trends. Higher cyclic variation is one of the draw backs under lean operating conditions, as is explained earlier. All injection timings exhibit the lowest values at 2000 RPM, which indicates a good mixture formation at this speed.



(a1)

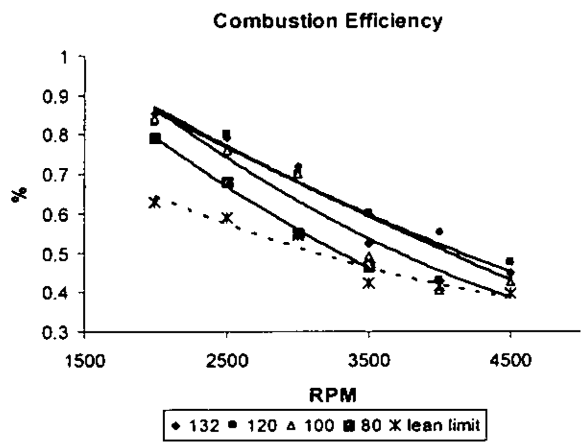


(a2)

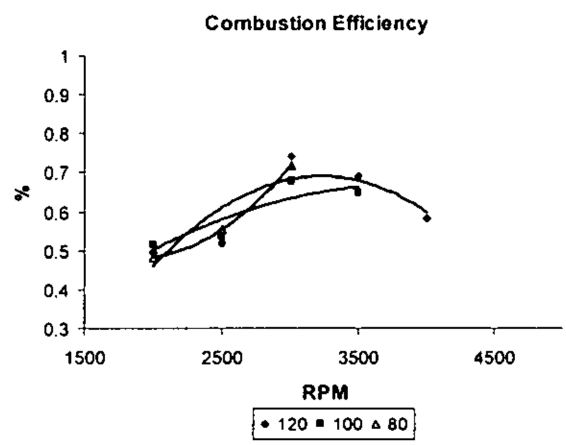
Figure 5-16 Cycle by Cycle Variation for Different Injection Timings at Partloads

Combustion efficiency characteristics for different injection timing under lean and stoichiometric conditions are shown in Figure 5-17. Combustion efficiency for lean stratified and stoichiometric operations are showing similar trends as that in idle load conditions. Lean stratified operation is yet again experiencing higher combustion efficiency at lower engine speeds, and tends to decrease as the engine speed is increased. Injection timings 132, 120 and 100 deg BTDC are having more than 80% for 2000 RPM engine speed, and reaches to about 50% at 3500 RPM. This is in accordance with the hydrocarbon emissions as they tend to increase with the engine speed. The highest efficiency is about 85% at 2000 RPM for injection timing 132 deg BTDC. 120 and 100 deg BTDC are having 83% and 84% respectively at this engine speed. Retarding the injection timings further would decrease the combustion efficiency as shown in the Figure 5-17 (a1). In contrary stoichiometric operations experience lower combustion efficiency at lower speeds and tends to increase with the engine speed, for all injection timings. Injection timing 120 deg BTDC is experiencing the highest (75%) at 3000 RPM, among other injection timings. Higher performance of stoichiometric operations at higher engine speeds might be due to higher combustion efficiency. Moreover the operating range of stoichiometric operation is limited to injection timings as shown in the Figure

5-17 (a2). With the increase of speed and higher cycle to cycle variation, less time available for mixture formation might be the causes of lower combustion efficiency at lean conditions. It can be seen that with the increase in engine speed and retarding the injection timing, combustion efficiency becomes lower, having lowest values at 4500 RPM. This result agrees with higher HC emissions at higher speeds and late injection timings at lean conditions.



(a1)



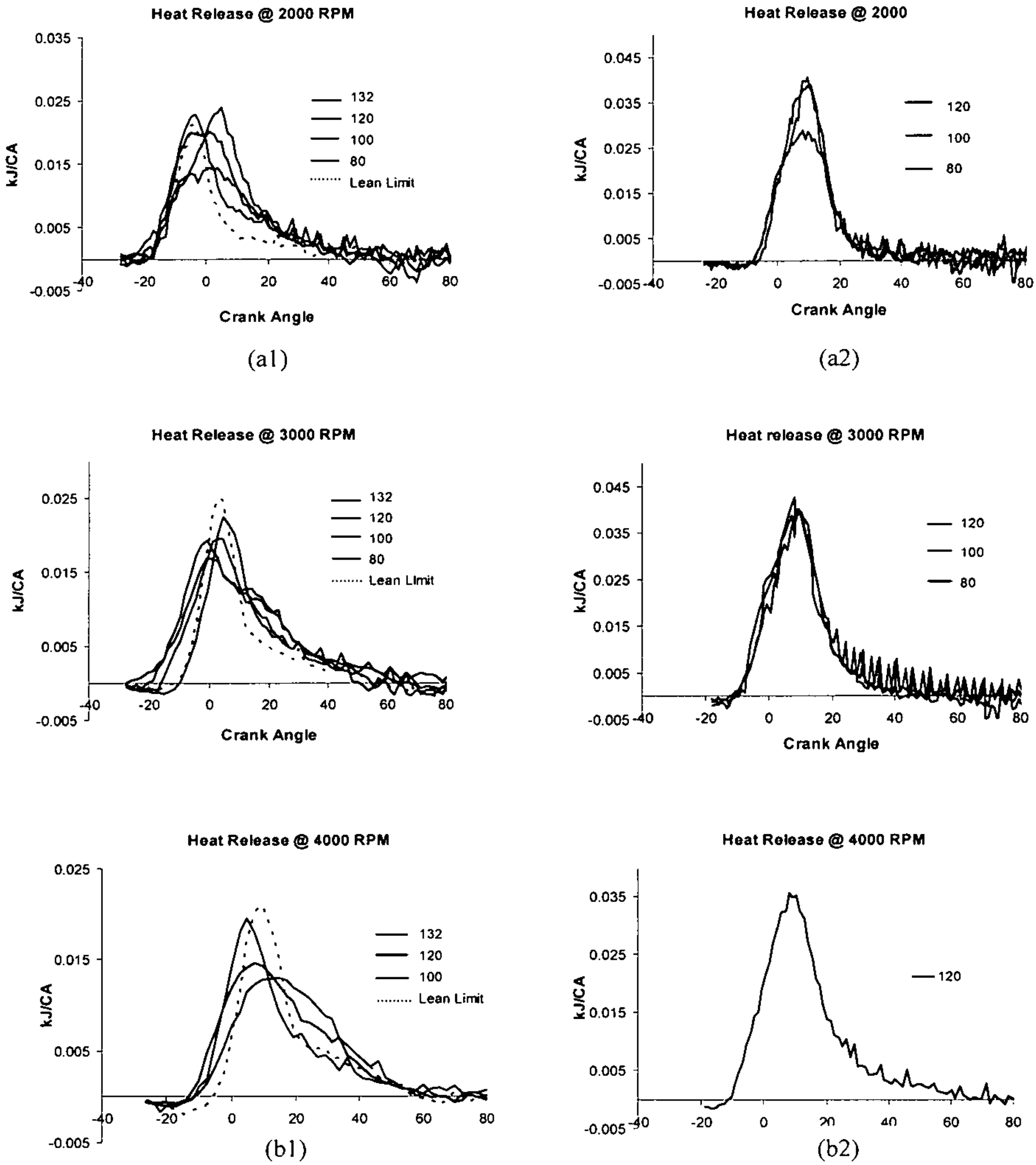
(a2)

Figure 5-17 Combustion Efficiency for Different Injection Timings at Partloads

Heat release characteristics for different injection timings at lean stratified and stoichiometric operations under partial loads are illustrated in Figure 5-18. Heat release for 2000, 3000 and 4000 RPM engine speeds are specified. Initial combustion is much faster in lean stratified operation compared to stoichiometric. For instance at 2000 RPM, the highest heat release for all injection timings is reached about 7-10 degrees before the piston reaches the top dead centre, except for 132 deg BTDC where it happens at 1 degree BTDC, this can also be seen in the pressure characteristics, Figure 5-14 (a1) where maximum pressure rise is much earlier. On the other hand for stoichiometric operations maximum heat release is occurring about 7.7-9.5 degrees ATDC, at 2000 RPM. Injection timing 132 deg BTDC is experiencing the highest heat release of about

0.030 kJ/CA, whereas for stoichiometric operations 100 deg BTDC injection timing is having the highest i.e. 0.049 kJ/CA, at this speed. Moreover it can be noted that at 2000 RPM, combustion duration for stoichiometric is shorter towards the end of combustion. Heat release characteristics at lean limits showing the shortest combustion compared to others. Heat release characteristics at 3000 RPM are also showing the similar trends for both operations, with a slightly slower combustion for stoichiometric towards the end. Maximum heat release at this engine speed for lean stratified operation occurs at about 4-6 degrees BTDC, while for stoichiometric it is occurring at 4-7 degrees ATDC. The highest heat release is experienced at 100 deg BTDC (0.030 kJ/CA) for stratified and 120 deg BTDC (0.064 kJ/CA) for stoichiometric. At 4000 RPM the heat release for both operations is comparatively less. For stoichiometric operation injection timing can not be further retarded from 120 deg BTDC, as engine operation is not stable. Similar to earlier case the initial combustion for stratified is faster and the highest is reached about 4-6 degrees BTDC. Highest heat release for lean stratified is about 0.027 kJ/CA while for stoichiometric is 0.039 kJ/CA. Higher heat release for both operations are experienced at 3000 RPM showing better combustion at this engine speed. Heat release pattern can explain the combustion process in the system. Heat release in stoichiometric is higher due to more available fuel in the cylinder to be burned compared to lean. Lean stratified operation is experiencing a faster burn in the initial stage and relatively slower burn in the later stage which might be due to higher degree of stratification and bulk quenching, whereas at stoichiometric conditions, the initial stage is relatively slower but the later stage is faster compared to that of lean stratified.

Stratified Lean **Stoichiometric**

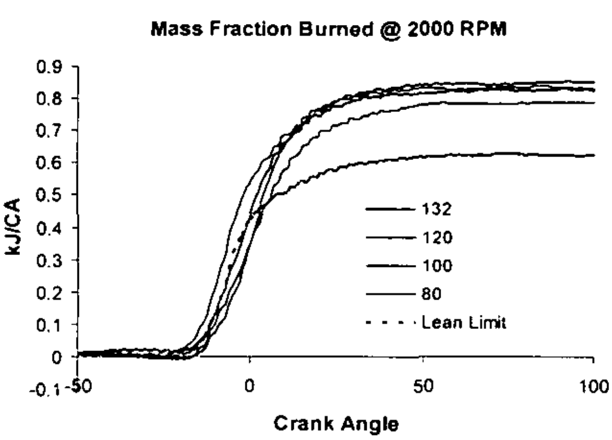


(c1) (c2)

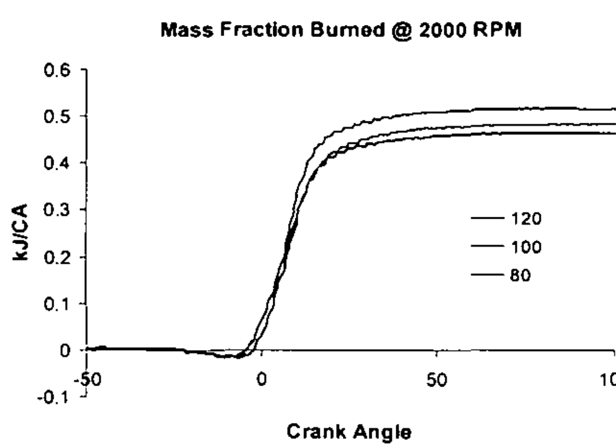
Figure 5-18 Heat Release Characteristics for Different Injection Timings at Partloads

Mass fraction burned with respect to injection timings at part loads for stratified and stoichiometric operations are given in Figure 5-19. For stratified operations, injection timings 132, 120 and 80 deg BTDC are showing similar slopes whereas 100 deg BTDC is having a slightly higher, for the initial and intermediate stages. Combustion duration at lean stratified operation for initial combustion stage is higher for all operating speeds. Mass fraction burned is higher for lean stratified operations at 2000 RPM for all injection timings, compared to that of stoichiometric. For stoichiometric conditions the initial burn is slower but the intermediate and the later stages of combustion are faster compared to lean as indicated by the higher slopes in the Figures. Injection timings 132, 120 and 100 deg BTDC are having higher mass fraction burned at 3000 RPM, for both operations. At 4000 RPM stoichiometric operation is experiencing a slower burning rate compared to lower engine speeds. Moreover injection timing is limited to 120 deg BTDC. For lean stratified operation 120 deg BTDC is having higher mass fraction burned rate compared to other injection timings. A slightly slower combustion is experienced by 132 deg BTDC, while 120 and 100 deg BTDC is having similar slopes. Overall at lean stratified operations the combustion duration is slightly shorter in the initial and intermediate stages compared to that of stoichiometric, although injection timings are same. The shorter duration might be due to charge stratification and faster flame propagation during the earlier combustion stage. Combustion towards the end is relatively longer for stratified operation.

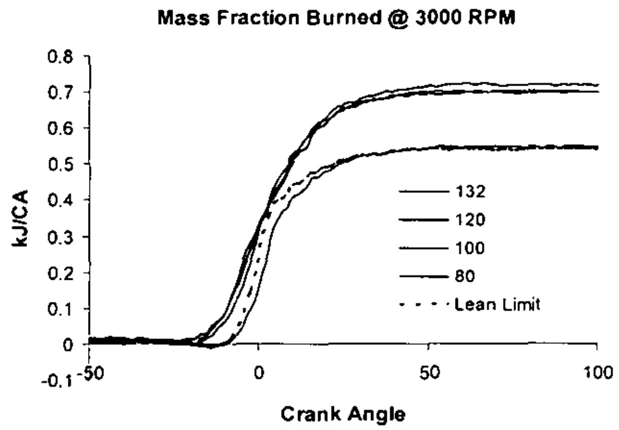
Stratified Lean **Stoichiometric**



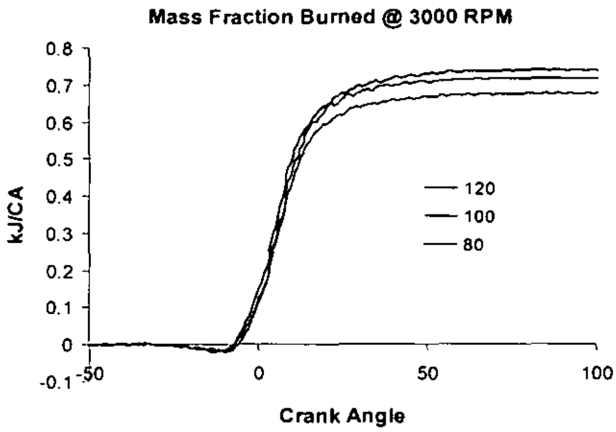
(a1)



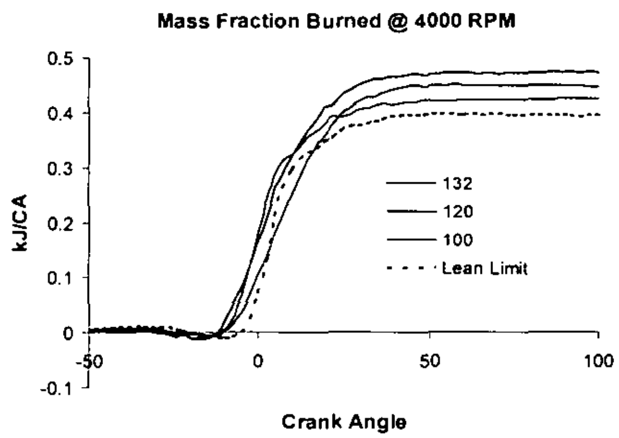
(a2)



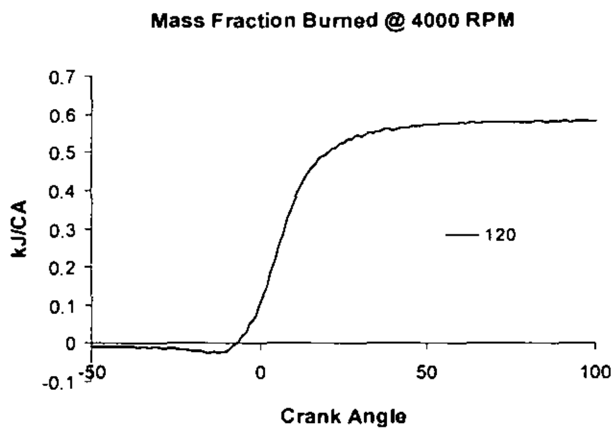
(b1)



(b2)



(c1)



(c2)

Figure 5-19 Mass Fraction Burned for Different Injection Timings at Partloads

Summary:

At partloads the performance of stoichiometric operation is better than that of lean stratified operations for all injection timings. There is a very little overall improvement in stratified conditions compared to idle loads. At partial loads there is about 25% drop in power even at 2000 RPM engine speed for lean stratified case and it decreases with the engine speed but the operation is limited till 3500 RPM due to the ignition parameter limitation. Even though the power produced is lesser at lower speeds, BSFC values for stratified operation are still lower compared to stoichiometric. On average at 120 deg

BTDC the BSFC values are 1-14% lower from 2000 to 3500 RPM, at 100 deg BTDC it is 5-19 % lower and for 80 deg BTDC it is 1-5% lower. The advantage of lower BSFC can be realized on the expense of lower power in lean stratified case.

Nitrogen oxides (NOx) emission for lean stratified operation is higher at lower engine speeds which indicate higher temperatures of combustion, for all injection timings except the lean limit where it is lesser. Injection timings 100 deg BTDC and 80 deg BTDC, the NOx emission is about 23 % and 56% higher at 2000 RPM for lean operations. For engine speeds above 3000 RPM nitrogen oxide emission for both cases are having lesser difference for all injection timings. In case of lean stratified operation the brake specific hydrocarbon emissions is higher, reason being the lesser power at leaner conditions and also at higher speeds the combustion is effected by the mixture formation as well as the amount of the fuel. At lean conditions it is showing an increasing trend with the increase in speed and with retarding the injection timing. For stoichiometric conditions CO emissions are showing almost the similar trend as that of idle loads. At lean stratified conditions injection timings 132, 120 and 100 deg BTDC are having similar trends and the amount of CO is less than 0.05%vol. Unlike idle loads injection timing 80 deg BTDC having higher CO but still lesser than 0.1% vol.

Lean stratified operation experiences faster combustion compared to that of stoichiometric. Pressure rise in lean operation is faster compared to stoichiometric as it is having higher gradient comparatively for engine speeds less than 3000 RPM. For stoichiometric operation the maximum pressure is reached at about 13 degrees after the top dead centre (ATDC) for all injection timings. Maximum pressure is reached earlier in lean conditions, and is about 3-6 degrees ATDC. Heat release characteristics also showing the similar behaviour, the initial combustion stage is much rapid and faster but much slower towards the end, which might one of the reasons of lower performance and higher HC emissions in lean stratified operations.

5.3 Effect of Injector Spray Angle to the Performance of CNG-DI Engine

Two types of injectors have been utilized in the experiments having different injection spray angles. Narrow Angle Injector (NAI) having a cone angle of 30 degree and Wide Angle Injector (WAI) having a cone angle of 70 degrees. Figure 5-20 represents the image of injector angle at atmospheric condition.

(a) NAI Injector

(b) WAI Injector

Figure 5-20 Injector Cone Angle

In direct injection engine, injections properties have a great significance over other engine operations and performance. Turbulence generated in the combustion chamber influence the mixture formation. Cone-angle and penetration length are most common injector properties. Wide cone-angle injector and narrow cone-angle injector used in this experiment could help the understanding of the effect of different injector cone-angle to the natural gas engine performance. Naturally, increasing angle of injection spray could improve the mixing rate of the mixture with compensation lower velocity due to larger distribution area for injected gas. Injection sequence for both injectors is depicted in Figure 5-21 respectively.

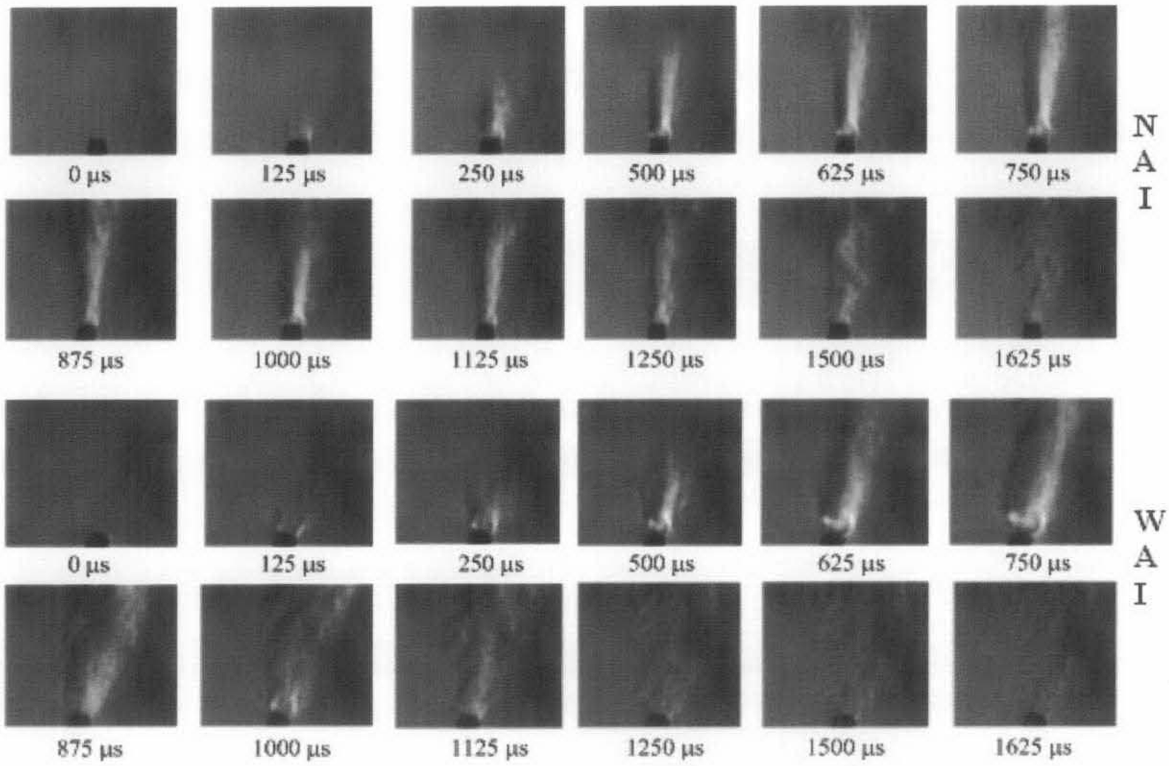


Figure 5-21 Injection Sequence

To acquire the injection image Schilieren method is used which distinguishes the density difference between the injected gas and the surrounding air. The experiment is carried out at atmospheric conditions under controlled environment to prevent the interference of surrounding air to injection process. The image was recorded at the speed of 0.125 ms per frame where injection duration used is 0.750 ms. Injection event stops at 0.75 ms, it can be noticed that the intensity of the injected gas for narrow angle injector (NAI) is higher than wide angle injector (WAI) at 1.25 ms. It can be assumed that for WAI the gas has already been mixed with the surrounding air. At the time of 1.625 ms, the WAI injector shows as if most of the gas has already disappeared while NAI still shows residue from previous injection process. From the following images, we can conclude that the mixing rate in WAI is faster compared to NAI, while the penetration length for NAI is more than WAI.

5.3.1 Engine Performance at Idle Loads

In the following the performance characteristics of CNG-DI are discussed while using wide angle injector (WAI) and narrow angle injector (NAI) under lean stratified conditions. Injection timing is set to 120 deg BTDC and the air fuel ratio is kept within 36.5-37. Throttle valve is fully closed to have idle load conditions. Performance characteristics such as torque, power, brake mean effective pressure (BMEP) and brake specific fuel consumption (BSFC) are illustrated. Figure 5-22 shows the performance characteristics for both the injectors. Figure 5-22 (a) shows the torque characteristics. It can be seen that the torque has a decreasing trend with the increase of the engine speed with both injectors; the gradient is higher at speeds above 3000 RPM. The maximum torque is at 2000 RPM for both NAI and WAI, with wide angle injector giving a 2% improvement. Overall wide angle injector gives better performance over all operating engine speeds, as depicted by the Figure. The improvement tends to increase with the increase of speed, from 2% at 2000 RPM to 16.5% at 4000 RPM. At 4500 RPM the difference is about 38% higher for WAI. The power ratings are illustrated in the Figure 5-22 (b). Power shows an increasing trend with the engine speed uptill about 3000 RPM and then decreases. The maximum power for WAI is about 6.3 kW at 3500 RPM, while for NAI it is 4.9 kW at 3000 RPM, showing about 22% higher for WAI. Faster mixing rate with wide angle injection might be the reason for higher performance of wide angle injector (WAI). Moreover at higher speeds the performance difference becomes considerable. Spray induced flow and better mixing of the fuel with air within short time might be influential for wide angle injector, and contributing to its performance. Brake mean effective pressure (BMEP) is illustrated in Figure 5-22 (c); it shows similar characteristics with torque. The maximum BMEP for WAI is ~ 620 kPa at 2500 RPM, whereas for NAI it is ~ 549 kPa at 2000 RPM, having a difference of about 11%.

Fuel consumption per unit power produced is illustrated in Figure 5-22 (d). Brake specific fuel consumption (BSFC) tends to increase with the increase in the engine speed. Wide angle injector for instance is having better BSFC than narrow angle injector for a wide range, except at 2000 RPM where NAI is having 0.4% better value. BSFC for wide angle injector (WAI) is lower than NAI for rest of the engine speeds. From 2500 RPM to 4500 RPM engine speeds, it is having 8-15% lower BSFC than NAI. As explained earlier lean operations with late injection timings give better fuel consumption at lower engine speeds compared to stoichiometric operations.

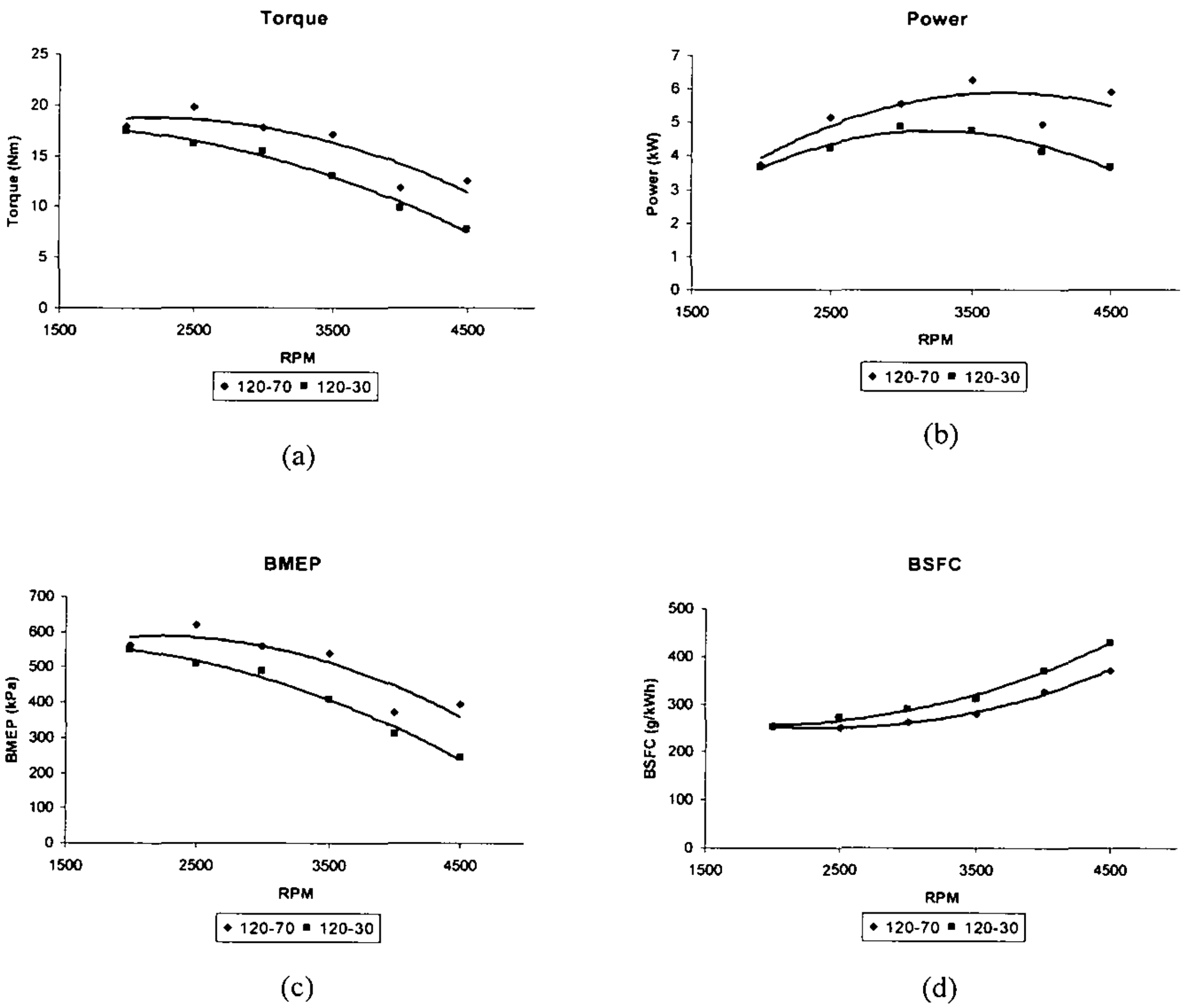


Figure 5-22 Performance Characteristics @120 BTDC Injection Timing for Different Injector Spray Angles

Volumetric efficiency for WAI and NAI at 120 deg BTDC and at idle load conditions is given in the Figure 5-23. Both WAI and NAI are having similar trends and there is not a considerable difference. Volumetric efficiency for both injectors gives maximum value at 3000 RPM, having 86.78 % and 86.90% for WAI and NAI respectively. For engine speeds higher than 3500 RPM WAI is giving 1-4% higher values for volumetric efficiency. Although there is not a noticeable difference in the volumetric efficiency, still wide angle injector is performing better under lean stratified conditions. It can be deduced that the mixture formation under lean conditions is quite prominent for engine performance.

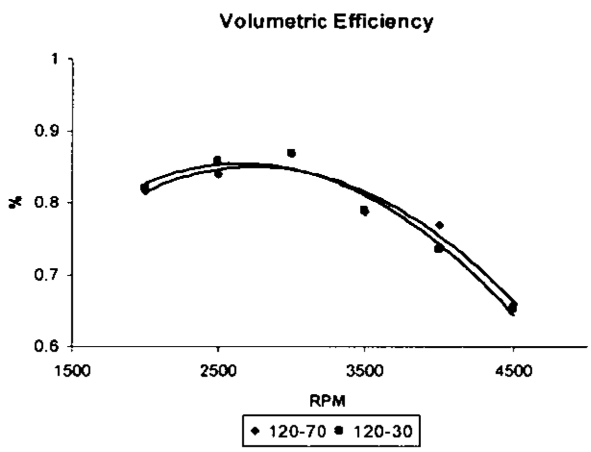


Figure 5-23 Volumetric Efficiency for Different injectors @ 120 deg BTDC Injection Timing

5.3.2 Engine Emissions at Idle Loads

Engine out emissions or combustion products, are important in discussion of engine performance. Three major exhaust emissions will be discussed in the following section, which include nitric oxide (NOx), total unburned hydrocarbons and carbon monoxide (CO). Emission results for narrow angle injector (NAI) and wide angle injector (WAI) are presented in Figure 5-24. Injection timing is set to start at 120 deg BTDC, while engine is operating at idle load conditions.

Nitric oxide (NOx) emissions are given in the Figure 5-24 (a). NOx emission occurs where high temperature condition happens in the combustion chamber. NOx emissions are higher for lean operating engines, thus higher combustion efficiency and higher combustion temperatures. Wide angle injector is showing higher values for NOx than narrow angle injector, for engine speeds less than 3500 RPM, with the highest values of 474 ppm/kW at 2500 RPM. Engines speed greater than 3500; NAI is having about 8-19% higher brake specific NOx emissions. Higher NOx formation at lower speeds indicate that lean stratified operations are having better combustion efficiency at lower engine speeds. Unburned hydrocarbons (THC) in the combustion products are illustrated in Figure 5-24 (b). Hydrocarbon emissions tend to decrease for the engine speeds less than 3500 RPM, for both WAI and NAI. Engine speeds more than 3500 RPM, hydrocarbon emissions start to increase again showing lower performance for lean stratified operation at higher engine speeds. NAI is showing lower values of 574 ppm/kW compared to 690 ppm/kW for WAI, for THC at 2000 RPM engine speed. THC emissions for WAI are considerably lower compared to NAI over rest of the operating range, having the lowest values of about 279 ppm/kW at 3500 RPM. The lowest THC for NAI is 507 ppm/kW at 3000 RPM. The levels of hydrocarbon emissions also indicate the advantage of higher performance of WAI over NAI.

Brake specific methane emissions, Figure 5-24 (c) are showing the similar trends as that of total unburned hydrocarbons (THC). Methane (CH4) is the major constituent, about more than 85%, of compressed natural gas. Methane emissions can also indicate the combustion quality. Lower methane means higher combustion efficiency. WAI shows lower brake specific methane compared to NAI for all engine speeds except at 2000 RPM, where it is having 12% reduced methane emissions. For engine speeds 2500 RPM to 3500 RPM WAI shows a reduction in methane from 17% - 92%. The difference is smallest at 4000 RPM where it is only 5% less for WAI.

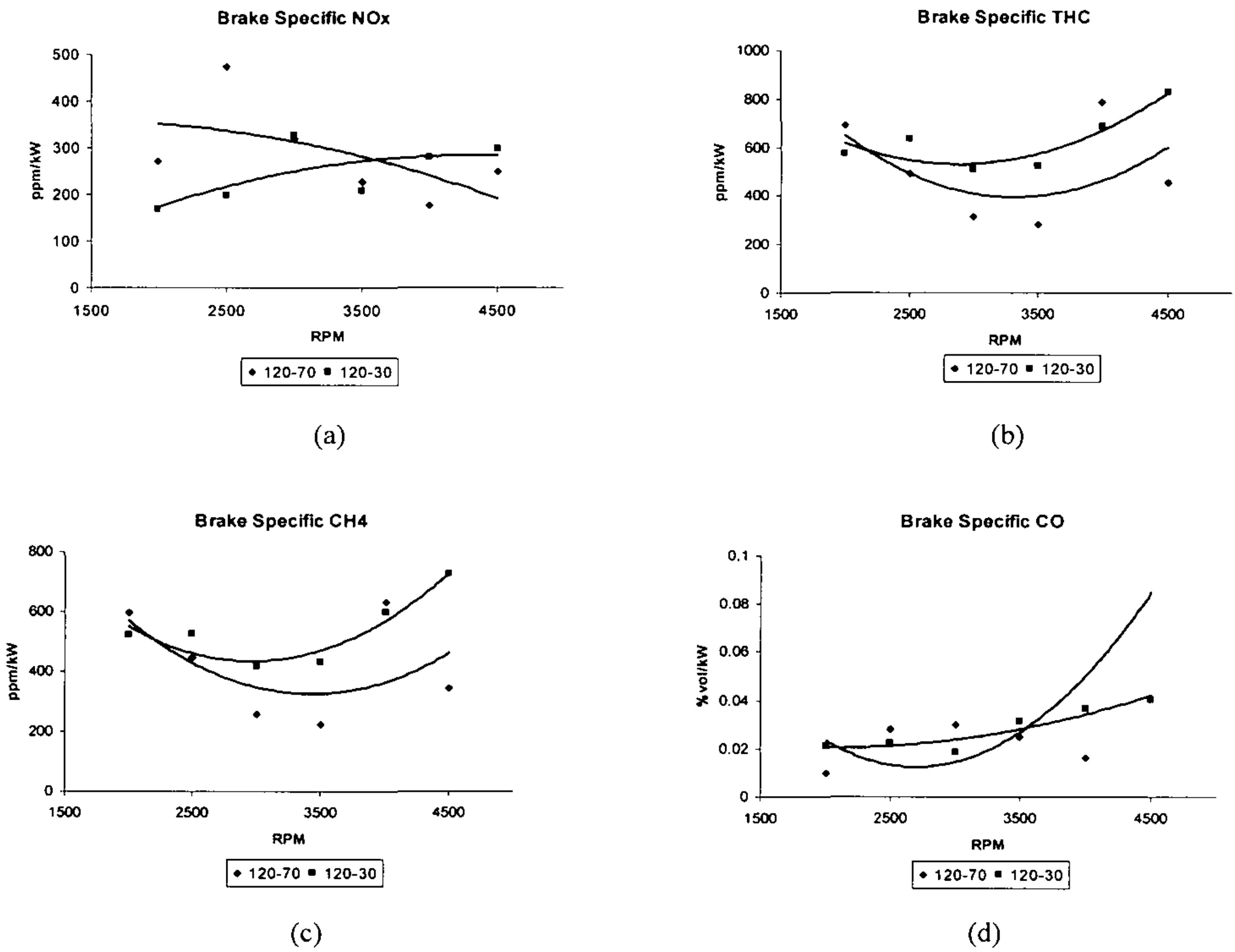


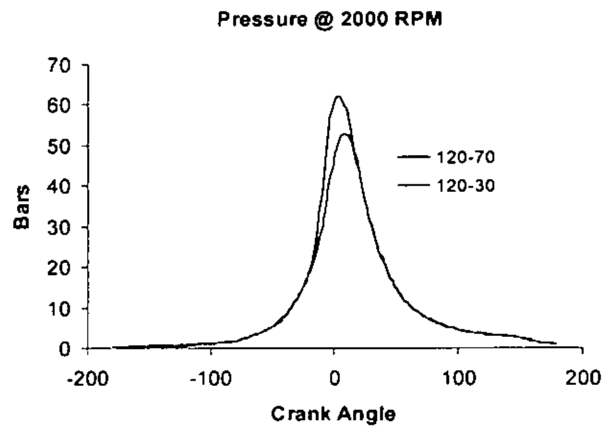
Figure 5-24 Emissions @ 120 BTDC Injection Timing for Different Injectors

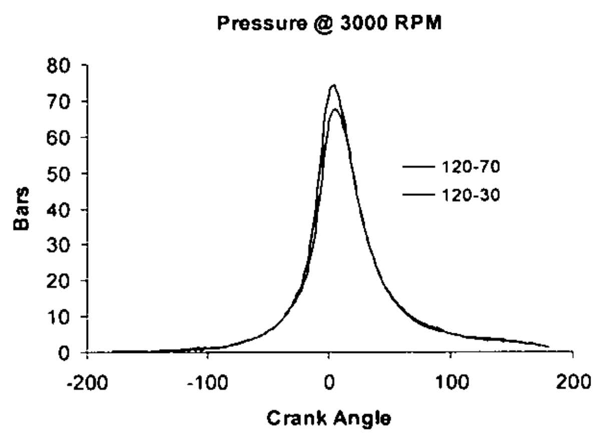
Carbon monoxide (CO) emissions are represented in Figure 5-24 (d). As stated earlier CO emissions are lower for lean operations. In spark ignition engines CO emissions increase steadily as the equivalence ratio increases, as the amount of excess fuel increases, whereas for lean mixtures it varies little with the change in equivalence ratio and is usually in the order of 10^{-3} mole fraction [74]. CO emissions for both WAI and NAI are having less than 0.04 vol%/kW for all engine speeds, except at 4500 RPM, where WAI is having a higher value. For NAI the CO emissions tend to increase but with a smaller gradient. Overall CO emissions at lean conditions are much lesser than that at stoichiometric conditions showing a comparatively higher combustion rates, where most of the CO is converted to CO_2.

5.3.3 Engine Combustion at Idle Loads

The analysis of the combustion process using pressure reading from the engine is presented next. Heat release rate is calculated by using Rassweiler and Withrow method. The combustion analysis is done for both WAI and NAI while injection timing is kept at 120 deg BTDC. Other parameters such as air fuel ratio ignition timings are same as described earlier. Combustion results are presented for engine speeds of 2000, 3000 and 4000 RPM.

Pressure rise due to combustion at 2000 RPM is illustrated in the Figure 25 (a). Pressure rise while using WAI is higher compared to NAI. The maximum is about ~ 62 bars for WAI and ~ 52 bars for NAI. Moreover in WAI the maximum pressure occurs very near to the top dead centre i.e. about 3 degrees ATDC, whereas in NAI it occurs at about 9 degrees ATDC. The difference in pressure rise indicates the better performance of WAI over NAI. At 3000 RPM engine speed pressure rise for WAI is again higher compared to NAI (Figure 5-25 (b)). But the difference in pressure rise is about 6 bars compared to 10 bars at 2000 RPM. The maximum pressure is about ~74 bars and ~68 bars, and occurs at 3 degrees and 5 degrees ATDC for WAI and NAI respectively. At 4000 RPM the pressure curves show lower values compared to lower engine speeds (Figure 5-25 (c)). This indicates that the performance drops at higher speeds for lean stratified operations. Pressure rise for WAI is again higher compared to NAI, with maximum value of ~51bars, compared to ~ 44 bars for NAI. The maximum pressure rise is at 3000 RPM engine speed, which shows that the engine is performing optimum at this speed, increasing the speed further would decrease the performance and is obvious from the performance characteristics and also emissions.





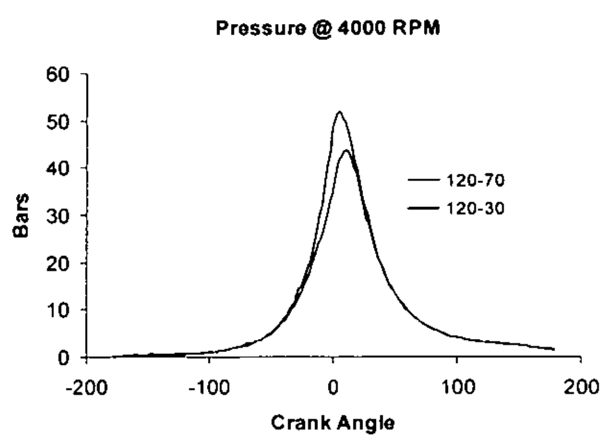


Figure 5-25 Pressure Characteristics @ 120 deg BTDC Injection Timing for Different Injectors

Figure 5-26 illustrates the indicated mean effective pressure for WAI and NAI at idle loads. It can be seen that the WAI is having higher values for IMEP than NAI for almost all engine speeds. IMEP is mainly the work transfer from the combustion to the piston. It is related to the combustion process, i.e. higher IMEP values indicate good combustion process. At 2000 RPM, both WAI and NAI exhibit similar values for IMEP, with WAI having 2% higher. In both cases the maximum IMEP is at 3000 RPM having 8.9 and 7.7 bars respectively. With the increase of speed the difference becomes obvious, as from engine speeds of 2500 to 3500, IMEP for WAI is having 10-17% higher IMEP. Higher IMEP values for WAI show that at lean operating conditions the mixture formation is quite critical, though NAI is having longer penetration length, hence higher turbulence, yet WAI is more effective for lean operations.

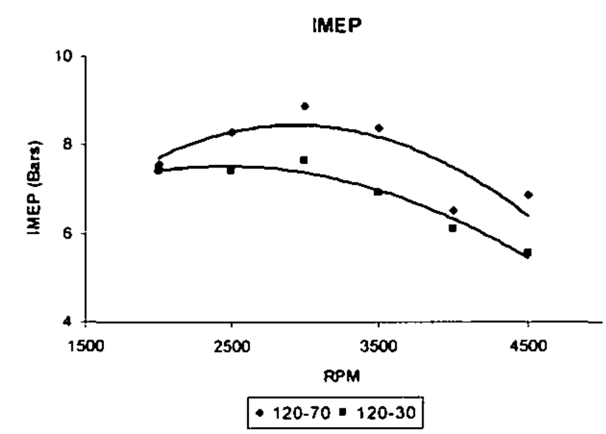


Figure 5-26 IMEP Characteristics for Different Injection Angles

Coefficient of variation (COV) can be seen in Figure 5-27. As already explained that COV implies to the drivability comforts and less than 10% COV is desirable. Lean operating engines have a problem of higher COV than those operating at stoichiometric conditions, which limits the operating range of lean engines. COV shows an increasing trend with the engine speed but at a lower gradient. It can be seen that COV values for both WAI and NAI are having values less than 10% for engine speeds less than 3500

RPM. At 4000 RPM WAI and NAI are having higher values of 9% and 13% respectively. It can be said that at lower engine speeds, the lower values of COV indicate that lean stratified operations are best suited for lower engine speeds. Moreover at higher engine speeds and with injection timing after the closing of intake valve, injection duration would be longer and there would be less time for fuel and air to mix properly.

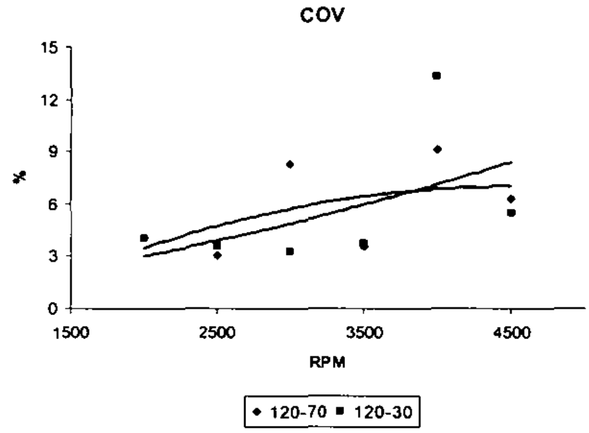


Figure 5-27 COV Values for Different Injection Angles

Combustion efficiency characteristics are presented in Figure 5-28. Combustion efficiency for WAI is higher compared to NAI for all engine speeds. Combustion efficiency tends to decrease with the engine speed, with highest at 2000 RPM. At 2000 RPM both injectors show almost same values for combustion efficiency with a difference of only 1%. Combustion efficiency for lean operations is higher compared to stoichiometric, but lower performance and higher COV are the main drawbacks, at higher engine speeds. Higher combustion efficiency for WAI is in accordance with the performance characteristics and the hydrocarbon emissions.

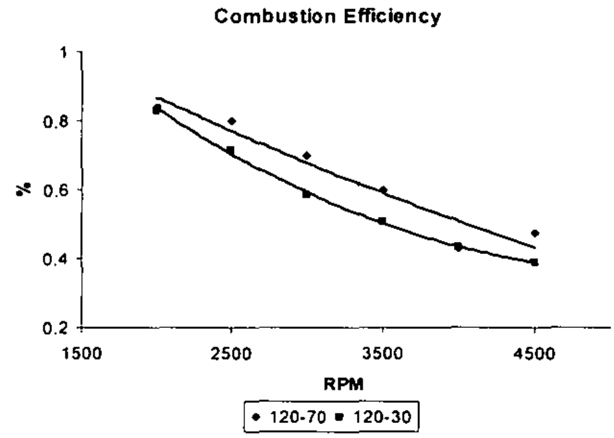


Figure 5-28 Combustion Efficiency for Different Injection Angles

Heat release characteristics for both injectors are represented in the Figure 5-29. Heat release characteristics at 2000, 3000 and 4000 RPM engine speed are illustrated for WAI and NAI. As illustrated earlier WAI angle is having higher pressures than NAI, similarly the heat release for WAI is higher. At 2000 RPM the maximum heat release for WAI is about 0.027kJ/CA at 6 degrees BTDC, whereas for NAI the maximum is about 0.019kJ/CA and it occurs at 3 degrees BTDC. It shows that the peak pressure occurs earlier for WAI due to faster combustion duration. At 3000 RPM engine speed, heat release per degree crank angle is higher compared to other engine speeds. Heat release for WAI is about 0.0299 kJ/CA, and for NAI is 0.0254kJ/CA, and it occurs at about 6 degrees and 2.5 degrees BTDC respectively. However at 4000 RPM heat release rate is lesser compared to lower engine speeds, NAI is having a higher maximum heat release of about 0.017kJ/CA at this engine speed. At 4000 RPM, the maximum heat release for NAI occurs at about 8 degrees ATDC, whereas for WAI it occurs at about 4 degrees BTDC. Overall WAI is showing faster combustion duration and higher pressures than NAI for engine speeds lower than 4000 RPM. At higher speeds higher stratification and lesser time for mixture formation and higher air fuel ratios might be the prime reasons for lower performance and lower heat release rates.

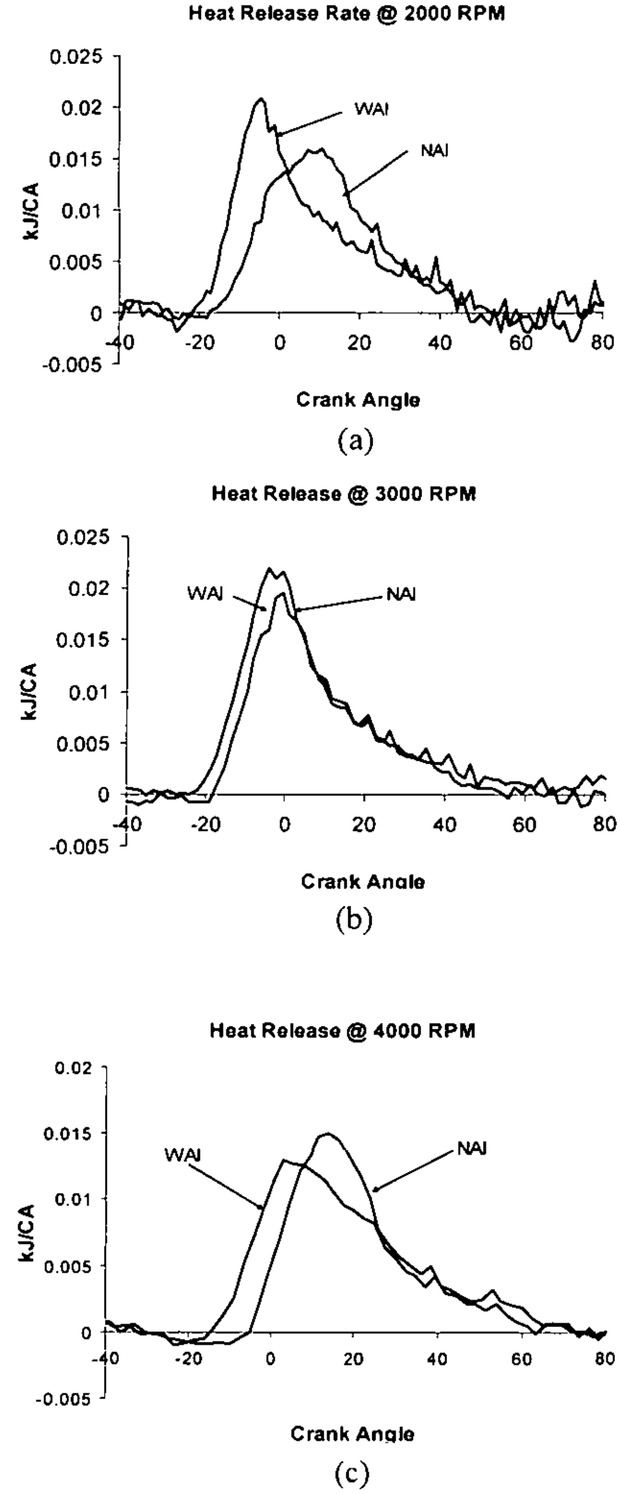
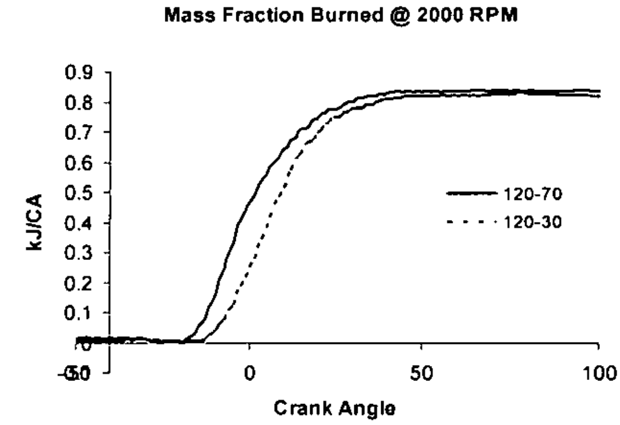
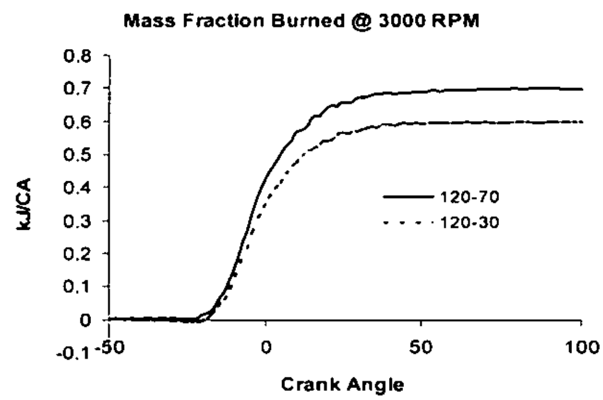


Figure 5-29 Heat Release Characteristics for Different Injection Angles at Various Engine Speeds

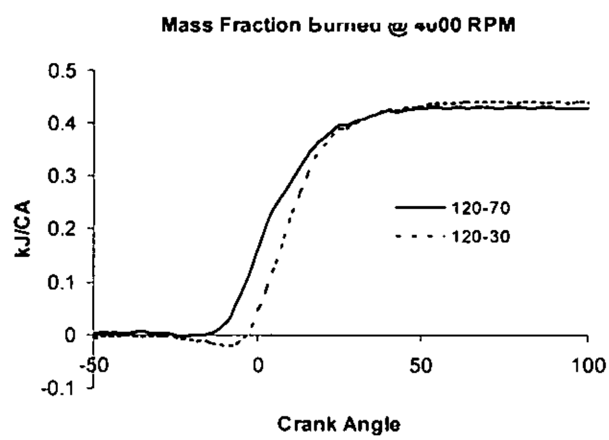
Combustion duration of CNG mixture at idle loads in CNG-DI engine is shown in Figure 5-30. Mass fraction burned for WAI and NAI are presented at different engine speeds. Combustion duration at 2000 RPM is much faster for WAI as shown in the Figure 5-30 (a), though the combustion efficiency at this speed is almost similar, yet WAI experiences higher heat release rates. At 3000 RPM engine speeds, WAI and NAI tend to have the similar slopes for earlier and intermediate combustion stages, yet WAI is having some what faster combustion duration. In contrary at 4000 RPM, NAI is having higher heat release and higher combustion efficiency. The earlier stage of combustion for NAI is much later than WAI, yet NAI is having higher maximum heat release, hence higher percentage of mass fraction burned. At high rpm and lean conditions, CNG engine undergoes a drop in power due to gas form of CNG that leads to lower energy content. Larger injection window is needed to get the same amount of fuel hence proper mixture formation is less likely. Lower combustion efficiency and higher HC emissions also indicate the lower performance with lean stratified operation.



(a)



(b)



(c)

Figure 5-30 Mass Fraction Burned for Different Injection Angles at Various Engine Speeds

Summary:

Effect of injection spray angle is investigated. Two types of high pressure injectors are employed having a cone angle of 30 degree (NAI) and 70 degree (WAI) respectively. With lean stratified operations and injection timing after the closing of intake valve i.e.120 deg BTDC, and idle load conditions, it is observed that the wide angle injector is having overall better performance than that of narrow angle injector for all operating engine speeds. WAI is having an improvement of 2% -16.5% in torque from 2000 to 4000 RPM engine speed. At 4500 RPM WAI is having about 38% higher torque than that of NAI. Faster mixing rate with wide angle injection might be the reason for higher performance of wide angle injector (WAI). Moreover at higher speeds the performance difference becomes considerable. Spray induced flow and better mixing of the fuel with air within short time might be influential for wide angle injector, and contributing to its performance. WAI is also beneficial in terms of brake specific fuel consumption, giving about 0.04% - 15.7% lower BSFC for engine speeds 2000 to 4500 RPM.

NOx emissions for WAI are 2-58% higher for engine speeds less than 3000, whereas for engine speeds higher than 3000, NAI is having 8-19% higher NOx. THC for WAI is lower than NAI for engine speeds more than 2500 RPM. On average WAI exhibits about 40-50% less unburned hydrocarbons than NAI. CO emissions for both injectors are almost having the similar trends and are having lower values.

Overall WAI is showing faster combustion duration and higher pressures than NAI for engine speeds lower than 4000 RPM. At higher speeds higher stratification and lesser time for mixture formation and higher air fuel ratios might be the prime reasons for lower performance and lower heat release rates.

5.4 Effect of Injector Spray Angle on Engine Performance at Partloads

In the following the engine performance characteristics will be discussed. The effects of injection spray angle would be analyzed; while engine is operating at partloads i.e. throttle valve is half opened. The air/fuel ratio is kept within the same levels (36.5-37) and injection timing is fixed to 120 deg BTDC i.e. 12 degrees after the closing of the intake valve. Engine operating speed range is from 2000-5000 RPM. As for the case of idle loads two high pressure injectors are employed in the experiments, having different spray cone angles.

5.4.1 Performance Characteristics at Partloads

Performance characteristics such as torque, power, brake mean effective pressure (BMEP) and brake specific fuel consumption (BSFC) at partloads conditions are illustrated in Figure 5-31, for both wide angle injector, having a cone angle of 70 degrees and narrow angle injector, having a cone angle of 30 degrees. Torque characteristics for WAI and NAI are shown in Figure 5-31(a). Unlike stoichiometric operations where there is a considerable rise in torque with the increase in engine speed at partloads, at lean conditions, torque characteristics are yet having a decreasing trend with the increase in engine speed. Though there is a rise of about 3 units in torque values at lower engine speeds, especially for narrow angle injector. For WAI maximum torque is 19.8 Nm at 2500 RPM, whereas for NAI it is 19.7 at 2000 RPM. Overall higher torque values are experienced with WAI over all operating speeds except at 2000 RPM, where NAI is showing a 0.2% improvement. For engine speeds 2500 to 4500 RPM WAI is showing an improvement of about 6-18%. In contrary to idle loads, NAI performance is better at partloads. Figure 5-31(b) illustrates the power ratings for WAI and NAI. Power tends to increase with the engine speed up to 3500 RPM and then decreases. Maximum power for WAI is about 6.5 kW at 3500 RPM, while for NAI it is about 5.46 kW at 3000 RPM, which is 16.3% higher for WAI. Whereas at idle loads the maximum power for NAI is about 22 % lower than WAI. Higher turbulence generated by NAI might be the reason for the improvement of the performance at partloads, while higher mixing rate for WAI is yet paying for higher performance. Maximum brake mean effective pressure (BMEP) for both WAI and NAI occurs at 2000 RPM, Figure 5-31 (c), having ~ 623 kPa and ~ 620 kPa respectively. BMEP shows similar trends as that of torque.

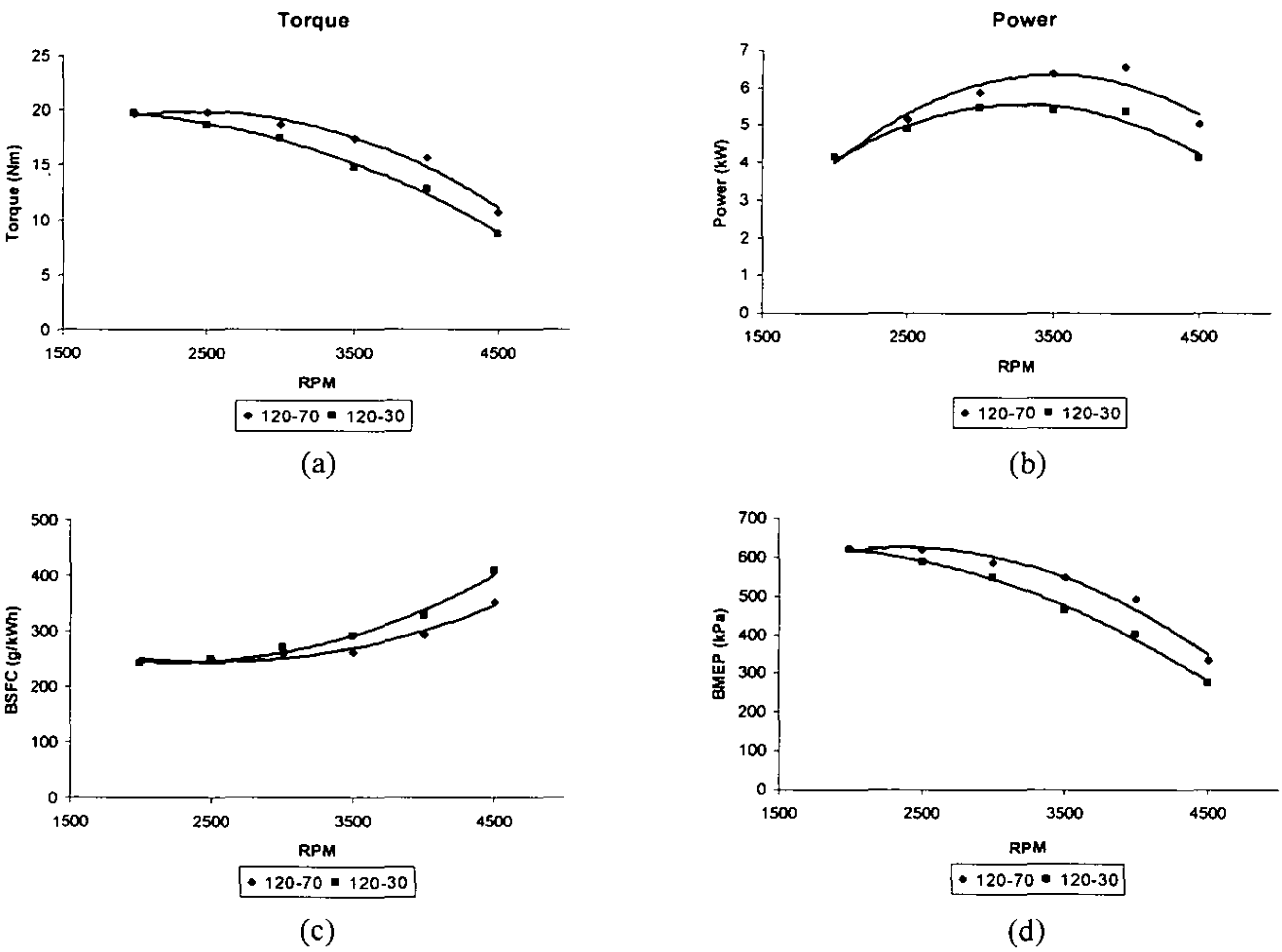


Figure 5-31 Performance Characteristics @120 BTDC Injection Timing for Different Spray Angle Injectors at Partloads

Lean stratified operations are desired for better fuel economy. Brake specific fuel consumption (BSFC) is shown in Figure 5-31 (d). BSFC for WAI and NAI are giving similar values for engine speeds less than 3000 RPM, with WAI having 1-1.7% higher compared to NAI. It means that fuel consumption for NAI is better than WAI for these operating conditions. This is contrary to BSFC values at idle loads, where WAI is giving better BSFC for almost all operating speeds. The lowest BSFC for WAI and NAI are 245 g/kWh and 241 g/kWh respectively at 2000 RPM engine speed. With the increase in the engine speed, BSFC increases, as there is a reduction of performance. It is less than 300 g/kWh for engine speeds less than 3000 RPM for both injectors. WAI is giving better BSFC i.e. 4-16% lower than NAI, for engine speeds higher than 3000 RPM.

Volumetric efficiency of the engine at partloads is illustrated in Figure 5-32. Volumetric efficiency is higher at partloads compared to idle loads, which could be the reason of a slight increase in the performance at partloads. At idle loads it volumetric efficiency is lower (0.65-0.86), compared to partloads (0.75-0.97). WAI and NAI are experiencing similar trends for volumetric efficiency, with maximum of 0.97 for WAI at 2500 RPM and 0.96 for NAI at 3000 RPM. Injection timings after the closing of intake valve are having overall higher volumetric efficiency for both injectors. Higher performance at lower RPM may be due to higher volumetric efficiency of late injection. As the speed is increasing, volumetric efficiency is decreasing. Air-fuel mixing does not have enough time to mix properly, yielding performance drops at high engine speeds. Although volumetric efficiency for NAI is slightly better for engine speeds more than 3000 RPM, yet WAI is having an overall better performance than NAI.

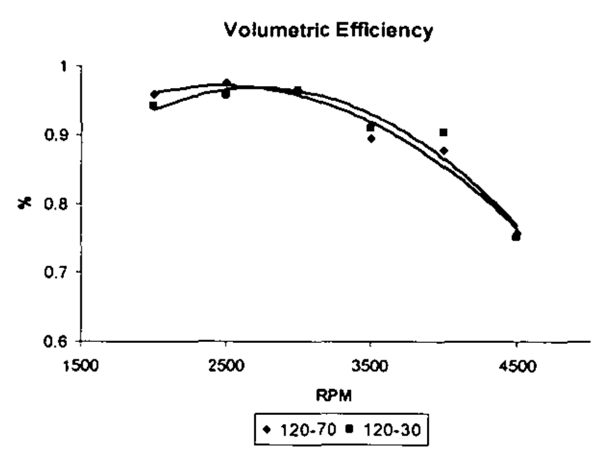


Figure 5-32 Volumetric Efficiency @ 120 deg BTDC Injection Timing for Different Spray Angle Injectors at Partloads

5.4.2 Engine Emissions at Partloads

Engine out emissions at partloads with respect to the injection angle are discussed in the following. Total unburned hydrocarbon (THC), nitric oxide (NOx), and carbon monoxide (CO) are the three main types of emissions which are investigated. Emission results for narrow angle injector (NAI) and wide angle injector (WAI) are presented in Figure 5-33. Injection timing is set to start at 120 deg BTDC, while engine is operating with half open throttle (partload).

NOx emission occurs where high temperature condition happens in the combustion chamber. NOx emissions are higher for lean operating engines, thus higher combustion efficiency and higher combustion temperatures. NOx emissions are shown in Figure 5-33 (a) for WAI and NAI. NOx emissions at partloads are showing different characteristics than at idle loads. For instance, NAI at idle loads is having lower NOx at lower engine speeds and tend to increase with the speed. Whereas at partloads NOx emissions are higher at lower engine speeds for NAI and tends to decrease with the engine speed. At partloads highest brake specific NOx is about 278 ppm/kW at 2000 RPM for NAI, while 252 ppm/kW for WAI at 2500 RPM, which is about 7% higher for NAI. NOx emission for NAI is having higher values except at 2500 and 4000 RPM where it is 13% and 11% higher for WAI respectively. Higher NOx emission for NAI is in accordance with the increase in its performance at partloads. An overall brake specific NOx emission at partloads is slightly lower than that at idle loads due to slight improvement in the performance. Higher NOx formation at lower speeds indicate that lean stratified operations are having better combustion efficiency at lower engine speeds.

Total unburned hydrocarbons (THC) for WAI and NAI are shown in Figure 5-33 (b). Unlike idle loads it exhibit different trends for THC emissions. THC tends to decrease for WAI with the increase in engine speed, having the lowest at 3500 RPM i.e. 391ppm/kW, whereas for NAI the lowest is 519ppm/kW at 2000 RPM engine speed and tends to increase with the speed. Lower THC emissions for WAI also indicated better combustion than NAI. THC emissions for WAI at idle loads and partloads are giving the lowest values at 3500 RPM engine speed. At 2000 RPM both WAI and NAI exhibit similar values for THC, with WAI having 0.6% lower compared to NAI. Methane (CH_4) as the major constituent of natural gas is also found in the exhaust emissions. Brake specific CH_4 emission is presented in Figure 5-33 (c). It is showing almost similar trends as that of THC, as it is the constituent of THC in emissions. Methane emissions for NAI are 2-8% lower for engine speeds less than 3000 RPM. CH4 emissions are 8-81% lower for engine speeds higher than 3000 RPM. On average the THC emissions for WAI is lower compared to NAI, showing better performance of WAI over a wide range.

Carbon monoxide emission for WAI and NAI at partloads is illustrated in Figure 5-33 (d). CO emissions for both WAI and NAI are having less than 0.03 vol%/kW for all engine speeds. A CO emission for lean operated engine is generally lower, as it's been stated in earlier cases as well. At 2000 RPM WAI is having about 14% higher CO emissions than NAI. For the engine speeds less than 3000 RPM, WAI is showing a reduction of about 11- 24% in CO emissions. At still higher speeds CO emissions for NAI are better having 31% lower values than WAI. Overall CO emissions at lean conditions are much lesser than that at stoichiometric conditions showing a comparatively higher combustion rates.

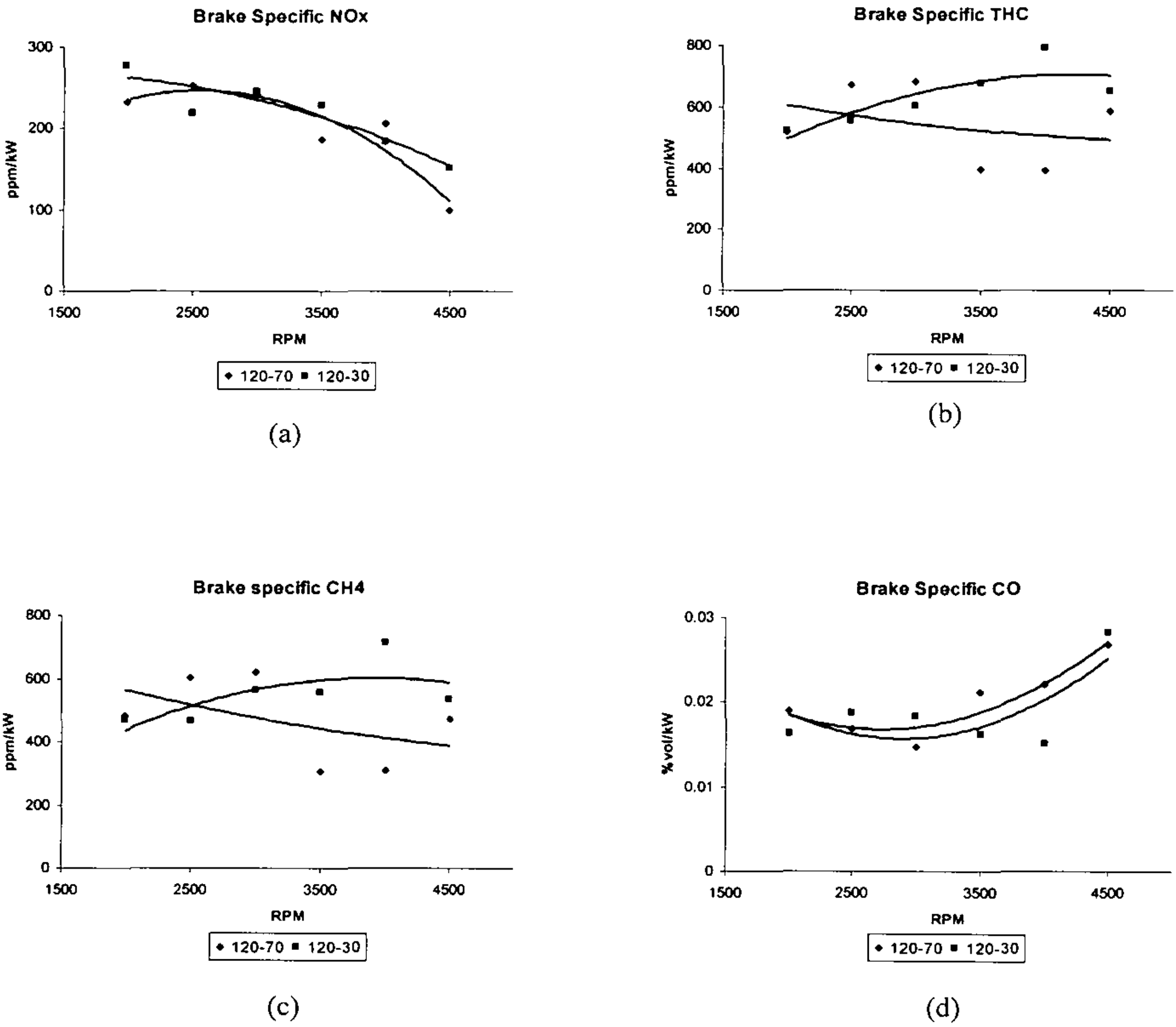
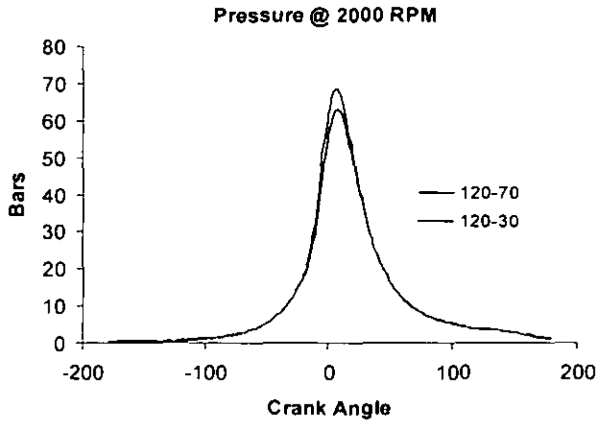


Figure 5-33 Emissions @ 120 BTDC Injection Timing for Different Spray Angle Injectors at Partloads

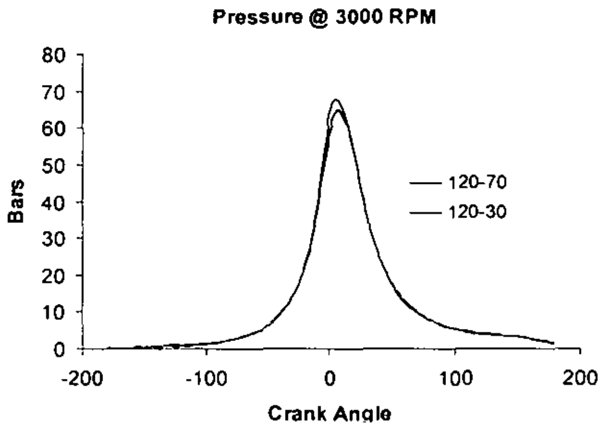
5.4.3 Engine Combustion at Partloads

The analysis of the combustion process using pressure reading from the engine is presented next. Heat release rate is calculated by using Rassweiler and Withrow method. The combustion analysis is done for both WAI and NAI while injection timing is kept at 120 deg BTDC. Air fuel ratio is kept within 36.5-37 and ignition timing is set for MBT (maximum brake torque). Combustion results are presented for engine speeds of 2000, 3000 and 4000 RPM.

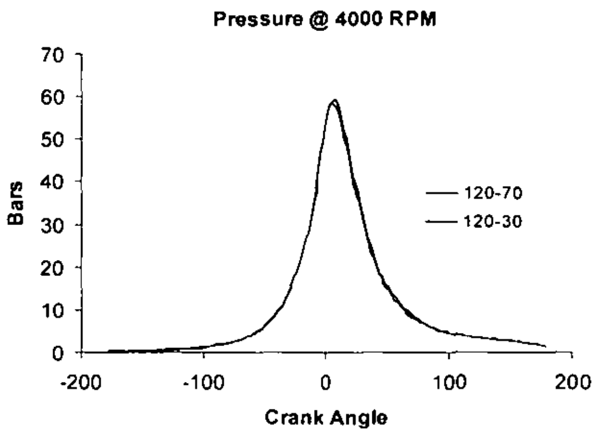
Pressure rise due to combustion is illustrated for different engine speeds for WAI and NAI in Figure 5-34. NAI is having higher pressure rise at 2000 RPM than WAI, Figure 5-34 (a), which is in contrary to idle loads where WAI is having higher pressure than NAI. NAI is showing higher pressures at partloads than at idle loads. The peak pressures at this engine speeds are ~ 63 bars and ~ 68 bars for WAI and NAI respectively. Moreover the gradient is similar and the maximum pressure occurs at 7 degrees and 5 degrees ATDC for WAI and NAI respectively. Higher pressure rise for NAI is an indication for its better performance at partloads due to longer penetration length of the injected fuel.. Moreover at idle loads the peak pressures are reached earlier than at partloads for WAI. Similar results are obtained at 3000 RPM with maximum pressures of ~ 65 bars and ~ 68 bars for WAI and NAI, occurring at 7.5 degrees and 4.5 degrees ATDC, Figure 5-34 (b). At partloads WAI is having lower peak pressures at this engine speeds compared to that at idle loads (~ 74 bars). NAI is experiencing almost the same maximum pressure as that of at idle loads. Unlike at lower engine speeds, WAI is experiencing a 1 bar higher peak pressures at 4000 RPM having similar gradient. The maximum values are ~ 59 bars and ~ 58 bars for WAI and NAI, which occurs at 7.5 degrees and 4.5 degrees ATDC respectively. Pressure is higher at 4000 at partloads compared to idle loads for both injectors. Unlike at idle loads the peak pressures are reached earlier for NAI than for WAI at partloads, showing faster combustion for NAI at partloads.



(a)



(b)



(c)

Figure 5-34 Pressure Characteristics @ 120 deg BTDC Injection Timing for Different Spray Angle Injectors at Partloads

Indicated mean effective pressure (IMEP) for WAI and NAI are illustrated in Figure 5-35. In contrary to idle loads, where WAI is having much higher (2-19%) IMEP than NAI, at partloads the difference is not that considerable. At partloads for engine speeds lower than 3000 RPM, NAI is having a 0.1-0.6% higher IMEP than WAI, whereas for engine speeds 3000 RPM and above WAI is having 3-4% higher IMEP than NAI. IMEP represents the work transfer from combustion to the piston. It can be seen that higher values for NAI at partloads attribute to its better performance than at idle loads. Decreasing trends of the IMEP also indicates the performance drops at higher engine speeds at lean stratified conditions.

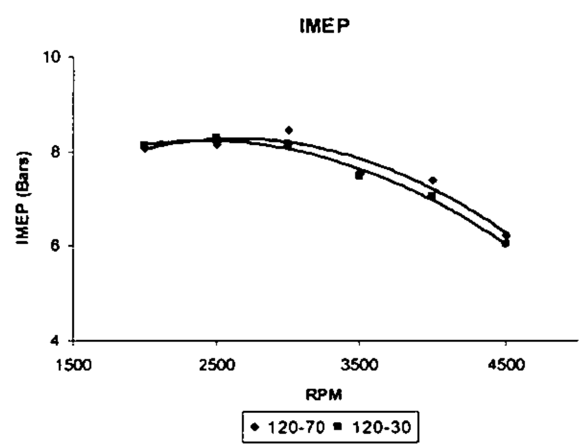


Figure 5-35 IMEP Characteristics for Different Spray Angle Injectors at Partloads

Coefficient of variation (COV) in IMEP at partloads is given in the Figure 5-36. As seen earlier the COV values for lean operations are higher compared to stoichiometric which is one of the limitations for such engines. COV for both WAI and NAI are showing an increasing trend with the increase in the engine speed. However, it can been seen that the COV values with WAI and NAI are having values lower than 10% except at 2500 RPM where COV is 11% and 9.5% for WAI and NAI respectively. Unlike at idle loads the COV data is not that scattered and showing minor difference. WAI is having lower COV values for engine speeds lower than 3500 RPM, except at 2500 RPM where it is having a bit higher values. It can be assumed that at lower engine speeds, the lower values of COV indicate that lean stratified operations are best suited for lower engine speeds. Moreover at higher engine speeds and with injection timing after the closing of intake valve, there would be less time for fuel and air to mix properly, which can be considered as the reason for the performance drop.

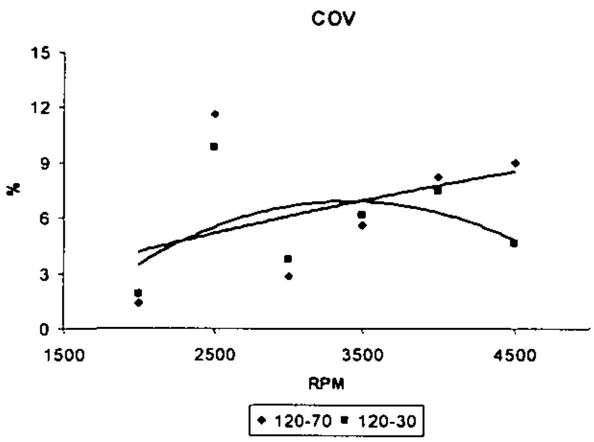


Figure 5-36 COV for Different Spray Angle Injectors at Partloads

Combustion efficiency characteristics for WAI and NAI at partloads with injection timing 120 deg BTDC is shown in Figure 5-37. As for the earlier cases the combustion efficiency at lean stratified operations tends to decrease with the increase in the engine speed. However NAI is showing higher combustion efficiency values at partloads than at idle loads, which is obvious from the performance and pressure characteristics as mentioned earlier. WAI and NAI are exhibiting similar values for combustion efficiency over all wide operating range, except at 2000 and 4000 RPM, where WAI is having a 3% higher. Combustion efficiency for lean operating engines is higher at lower engine speeds compared to stoichiometric; reason might be due to more time available for the mixture formation. The highest value for combustion efficiency for both injectors is above 80 % at 2000 RPM. It drops to 67% and 69% at 2500 RPM, and keeps on decreasing with the increase in the engine speed. Moreover lower volumetric efficiency and higher COV at higher engine speeds also responsible for lower performance.

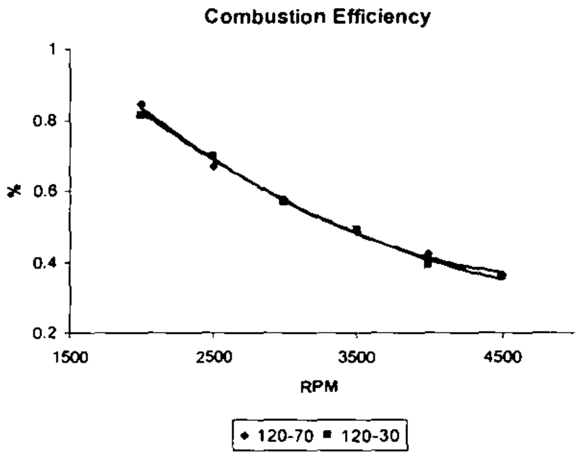
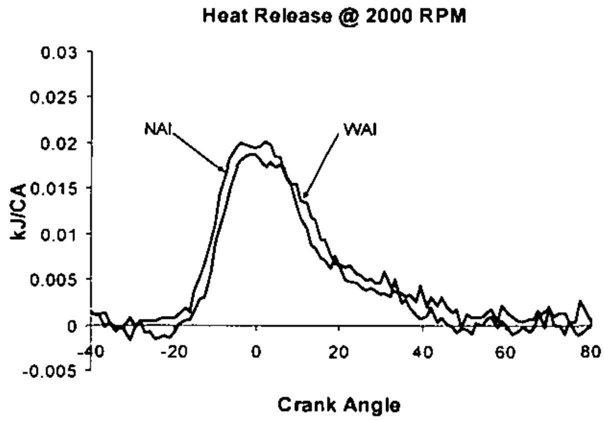
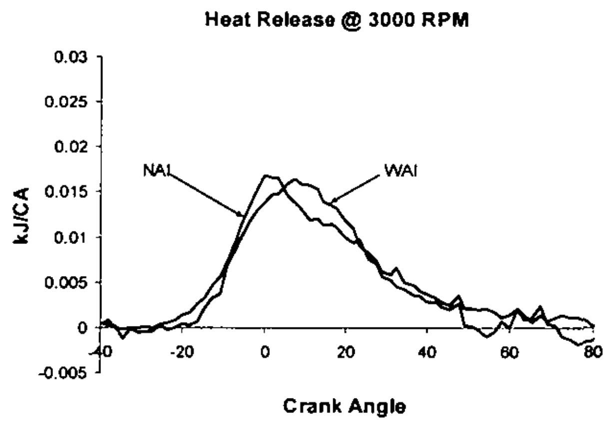


Figure 5-37 Combustion Efficiency for Different Spray Angle Injectors at Partloads

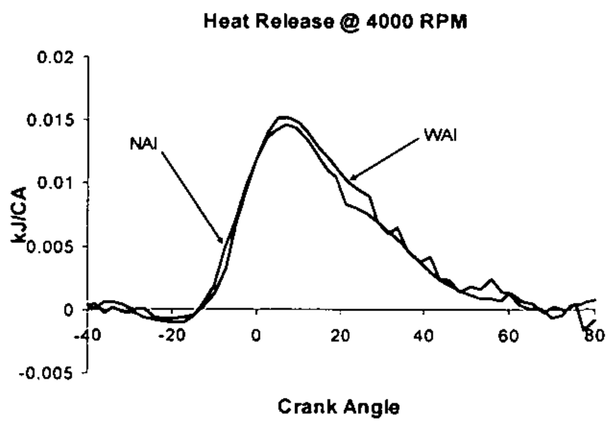
Heat release characteristics for WAI and NAI are shown in Figure 5-38. Heat characteristics at 2000, 3000 and 4000 RPM at partloads for both injectors are presented. Heat release pattern can explain the combustion process that occurs in the system. Maximum heat release for WAI and NAI at 2000 RPM is 0.0252 kJ/CA and 0.0256 kJ/CA respectively. These maximum values occur at 3 degrees and 9 degrees BTDC, with almost similar heat release rates, which is showing that the combustion rate for NAI is faster as compared to WAI at partloads at this engine speed. Moreover the pressure rise for NAI was also higher as explained earlier, Figure 5-34. Whereas at idle loads and at the same speed the maximum heat release and pressure rise was lesser for NAI, which depicts its better performance at partload conditions. Heat release is lesser for WAI at these conditions compared to that at idle loads. Also combustion for NAI is faster compared to WAI. At 3000 RPM the maximum heat release for NAI is 0.0216 kJ/CA which occurs at 4.5 degrees BTDC, while for WAI it is 0.0238 kJ/CA at about 2 degrees ATDC. Although WAI is having higher heat release yet combustion even is faster for NAI. At 3000 RPM engine speeds with idle loads the maximum heat release was higher for both WAI and NAI than at partloads. AT 4000 RPM, WAI and NAI are having almost similar values for heat release i.e. about 0.0158 kJ/CA, which is same as at idle load conditions, yet the pressure rise at partloads is higher. Lower levels of heat release might be due to the higher stratification at higher speeds and slow burning, which reduces the temperature.



(a)



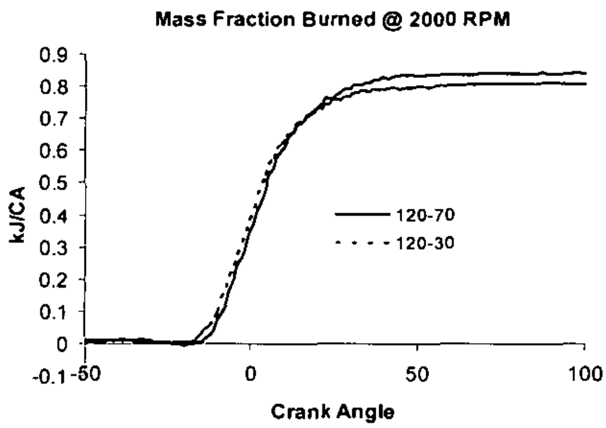
(b)



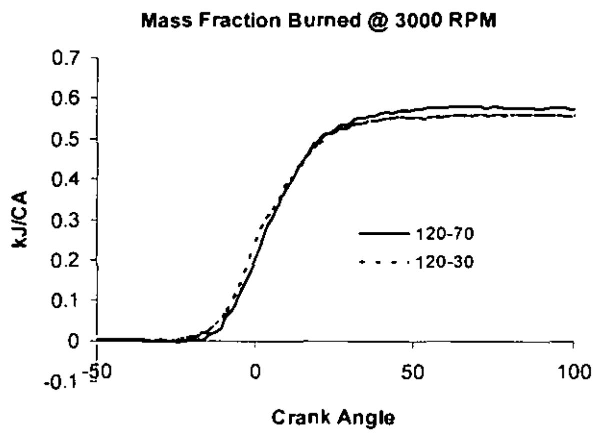
(c)

Figure 5-38 Heat Release Characteristics for Different Spray Angle Injectors at Various Engine Speeds

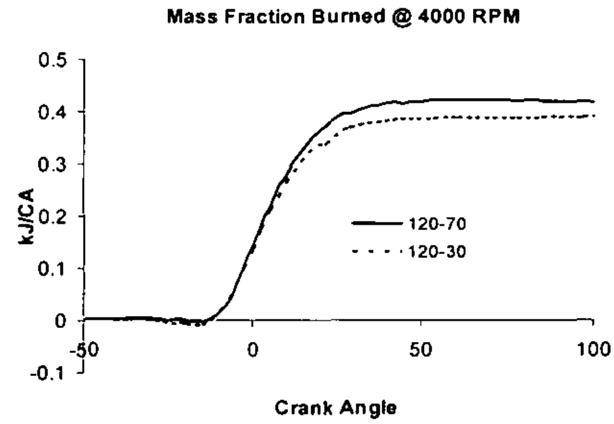
Mass fraction burned at partloads for WAI and NAI at different engine speeds are illustrated in Figure 5-39. Compared to idle loads the combustion duration for WAI and NAI at partloads at 2000 RPM is quite similar, with narrow angle exhibiting a bit faster combustion in the early combustion stage, while WAI having a slightly higher mass fraction burned, Figure 5-39 (a). At 3000 RPM engine speed, earlier and intermediate combustion for NAI is slightly faster compared to WAI with almost similar rate of mass fraction burned. At 4000 RPM, intermediate combustion for WAI is showing faster duration and higher heat release compared to NAI. Yet at higher engine speeds, lower NOx higher THC emissions indicate lower temperatures and lower combustion quality. Lower fuel content in lean stratified operation, with higher stratification at higher engine speeds can also be the reason for lower performance.



(a)



(b)



(c)

Figure 5-39 Mass Fraction Burned for Different Spray Angle Injectors at Various Engine Speeds

Summary:

Unlike at idle loads narrow angle is performing better at partial loads yet it is lower than WAI. For instance, at 2000 RPM NAI is having 0.2% higher torque compared to WAI. For engine speeds 2500 to 4500 RPM, WAI is having 5.8 -18.3% higher values for torque. Moreover for engine speeds less than 3000 NAI gives 0.6-1.7% lower BSFC values. For engine speeds 3000 and above, WAI gives about 4-16% lower BSFC than NAI.

NOx emissions for NAI is about 18% higher at 2000 RPM NOx emission for NAI is having higher values except at 2500 and 4000 RPM where it is 13% and 11% higher for WAI respectively. Higher NOx formation at lower speeds indicate that lean stratified operations are having better combustion efficiency at lower engine speeds. THC tends to decrease for WAI with the increase in engine speed, having the lowest at 3500 RPM i.e. 391 ppm/kW, whereas for NAI the lowest is 519 ppm/kW at 2000 RPM engine speed and tends to increase with the speed. CO emissions for both WAI and NAI are having less than 0.03 vol%/kW for all engine speeds. A CO emission for lean operated engine is generally lower, as it's been stated in earlier cases as well.

In contrary to idle loads the pressure rise for NAI is higher than WAI for engine speeds less than 4000 RPM. At 4000 RPM engine speed WAI is having a 1 bar higher peak pressure than NAI. Heat release rates also showing slightly higher values for NAI for engine speeds 2000 and 3000 RPM. While at 4000 RPM WAI is having higher values. Combustion pattern for both injectors is exhibiting some what similar behaviour.

CHAPTER 6

CONCLUSIONS AND RECOMMENDATIONS

6.1 CONCLUSIONS

The experimental work is done to assess the performance of CNG-DI at idle and partloads while operating at lean stratified conditions. Injection parameters such as injection timing and injection angle are investigated. For injection timing, stoichiometric operation with similar injection timing is compared with lean stratified operation. Lean limits are also found for both idle and partial loads. Following conclusions can be drawn.

1. Injection timing is an influential parameter to the performance of CNG-DI. At lean stratified conditions injection timing effect to the performance can be realized.
2. Injection timing in direct injection systems can effect the charge stratification and mixture strength and thus the performance.
3. Injection timings which start after the closing of intake valve (132 and 120 deg BTDC) gives better performance over a wide range, due to better mixture formation. Retarding the injection timing further would give lower performance with the increase in engine speed which might be due to less time available for the proper mixture formation.
4. At idle loads, and lower engine speeds lean stratified operation proves to be beneficial over stoichiometric. It also gives advantage of lower brake specific fuel consumption over an extended range.
5. At partloads lean stratified operation is experiencing lower torque (about 15-30%) compared to stoichiometric, for engine speeds less than 3000 RPM, but on the other hand is having 5-11% lower brake specific fuel consumption, showing lean stratified operations are beneficial at lower engine speeds.

6. Lean combustion limit for CNG depends upon the injection timing, ignition timing and engine speed. With direct injection systems extremely lean burn capability of CNG can be realized.
7. Lean operations experience higher cycle to cycle variations compared to stoichiometric.
8. NOx emissions are relatively higher showing higher combustion temperatures at lean conditions
9. Unburned hydrocarbons emissions are slightly higher, which might be due to higher cycle to cycle variation, excessive stratification and bulk quenching. Carbon monoxide emissions are lower than stoichiometric.
10. Injector properties, such as injector spray angle also influences the performance of CNG-DI.
11. Wide angle injector (WAI) proves to be giving better performance than narrow angle injector (NAI) at lean conditions. Faster mixing rate of WAI might be responsible for such behaviour.
12. At idle loads, WAI is giving 2-38% higher torque for all operating speeds and 8-15% lower BSFC values for engine speeds higher than 2500 RPM.
13. At partial loads NAI is performing better for lower engine speeds less than 2500 RPM, and 0.6-1.7% lower BSFC for engine speeds less than 3000 RPM.
14. Overall WAI is advantageous over NAI for lean stratified operations.

6.2 RECOMMENDATIONS

CNG stratified operation has a great potential for practical use; hence mixture formation and combustion characteristics of lean stratified operation should be further studied. Mixture formation process inside the combustion chamber is crucial to control combustion in CNG-DI engine, which apparently affects the performance and emissions. Extended examinations on the interaction between injected fuel and in-cylinder flow needs to be carried out to get clear picture of controlling the mixture to control the combustion process. Flow and combustion flame visualization studies are also recommended to fully understand the mixture formation and combustion characteristics at lean stratified conditions. Moreover effect of using turbocharger or supercharger on CNG-DI should be investigated which might increase the power output of the engine.

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APPENDIX A

CNG Composition in Malaysia

Component	Leanest	Reachest	Unit
Methane	96.42	89.04	%
Ethane	2.29	5.85	%
Propane	0.23	1.28	%
Iso-Butane	0.03	0.14	%
N-Butane	0.02	0.10	%
Iso-Pentane	N/A	N/A	%
N-Pentane	N/A	N/A	%
N-Hexane	N/A	N/A	%
Condensate	0.00	0.02	%
Nitrogen	0.44	0.47	%
CO_2	0.57	3.09	%
Gross Heating Value	38130	38960	KJ/Kg

APPENDIX B

General categories of gasoline direct injection injectors are presented as follows:

Classification categories for Gasoline-DI Injectors

Actuation Mechanism	Single solenoid Dual solenoid piezoelectric hydraulic Cam
Fluid State	Single fluid Air-assist
Primary Atomization method	Sheet (swirl-plate) Pressure (hole-type) Pressure (slit-type) Turbulence (compound plate) Pneumatic (air-assist) Cavitations Impingement
Nozzle Configuration	Swirl Slit Multi-hole Cavity
Pintle Opening Direction	Inwardly Opening Outwardly Opening
Spray Configuration	Hollow-cone Solid-cone Fan Offset Multi-plume Shaped

Source: Zhao.F.Gasoline Direct Injection Engines

APPENDIX C

LITERATURE REVIEW on CNG-Direct Injection

Author	Masaru Hayashida et.al	Seiichi Shiga et.al	Deanna E.Wang et.al	Zuohua Huang et.al	Jongwoo Kim	Ke Zeng et.al
Engine Type	Single cylinder, OHV, Water cooled, Four stroke	RCM	Single Cylinder, DOHC, Four stroke, Hydrogen assisted injection	RCM	4 Cylinder, pent-roof cylinder head	Single cylinder, Four stroke
Ignition System	Spark Ignition	Spark Ignition	Spark Ignition	Spark Ignition	Spark Ignition	Spark Ignition
Bore (mm)	94	80	89	80	79.5	100
Stroke (mm)	90	180	66	180	95.5	115
Displacement (cc)	624	904	410	904	1900	903
Compression ratio	11.4	10	11.6	10	10.4	8
Piston Crown	Disk	Flat		Flat	Flat	Bowl-in-shape
Injector Position	Angled, Intake direction	Side Injection		Side Injection	Side injection	Side Injection
Injection Pressure	5 Mpa	9 Mpa	2 Mpa	9 Mpa	20 bar and 80 bar	80 bar
Spark Position	Exhaust valve side	Center		Center	Center	Center
Valve Numbers	2		4		4	2
Max. BMEP	n/a	n/a	n/a	n/a	n/a	0.4 Mpa at 180BTDC
Engine speed	1200				1600	1200
Throttle	WOT	n/a	WOT	n/a	part	WOT
Equivalence ratio	0.83	0.6 and 1	0.56	0.6 and 1	0.6 to 1	0.5

LITERATURE REVIEW on CNG-Engine Fueling Method Development

Year	Author	Research fields			
		CR	Fueling system	Operating condition	Observed parameters
2005	C.C.O. Reynolds et.al	9.26	Mixer	Strafied Condition Part load SI engine	Partially Stratified Charge composition
2005	Chetan S. Mistry	n/a	Carburetor	Variable Load Variable Speed SI engine	Comparison with Gasoline and LPG
2000 / 2005	S Maji et.al	8	Mixer	SI engine WOT	Comparison of CNG and Gasoline as engine fuel Spark timing AFR Compression ratio
1995	P. Corbo et.al	10.8	Carburetor	WOT	Comparison between stoichiometric and lean burn in CNG Heavy-Duty Engine
2003	Jan Czerwinski et.al	11	Multi Port Injection	SI engine Turbocharge engine WOT	Comparison of MPI with Port injection Injection Timing Injection Pressure
2002	Jan-Ola Olsson et.al	Varied	Port Injection	WOT HCCI engine Lean condition	Compression ratio

2000	A. E. Catania et.al	10.35	Multi Point Injection	Injection Pressure 7 bar WOT SI engine	Comparison with Gasoline
1999	Kichiro Kato et.al	11	Port Injection	WOT Injection pressure 764 kPa SI engine	Comparison with Gasoline
1998	Joseph M. Brault	12.5	Multi Port Injection	Starting Condition SI engine	Injection timing