Master's Thesis

A Numerical Study on Effect of Ignition Timing and Mixing Ratio of Natural Gas Spark Ignition Engine on Engine Performance and Exhaust Emission Characteristics

The Graduate School

of the University of Ulsan

Department of Mechanical Engineering

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ABSTRACT

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Interest in natural gas as an internal combustion engine fuel has been renewed due to its increasing domestic availability and stable price relative to other petroleum fuel sources. Natural gas, comprised mainly of methane, allows for up to a 25% reduction in engine out CO2 emissions due to a more favorable hydrogen-to-carbon ratio, relative to traditional petroleum sources. Traditional methods of injecting natural gas can lead to poor part-load performance, as well as a power density loss at full load due to air displacement in the intake manifold. Natural gas direct injection, which allows the fuel to be injected directly into the cylinder, leads to an improvement in the in-cylinder charge motion due to the momentum of the gaseous injection event. In this study, the potential improvement in thermal efficiency and emission characteristics of SI engines fueled with natural gas running with various ignition timing was investigated. The engine was tested at 1800 rpm, volume percent of propane in the methane propane blends ranged from 0 to 20%. Performance tests were performed via measuring brake thermal efficiency, brake specific fuel consumption, cylinder pressure and exhaust emissions (CO, THC, NOx). The use of natural gas as an alternative fuel is a promising solution. The potential benefits of using natural gas in SI engines are both economic and environmental. Chemical kinetic simulation of natural gas combustion has been carried out based on available literature data. Comparative results are given for various ignition timing and various mixing ratio, revealing the effect of ignition timing, and mixing ratio on engine performance and exhaust emissions.

Keywords: methane, propane, natural gas, SI engine, emissions characteristics, performance, ignition timing, mixing ratio.

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TABLE OF CONTENTS

LIST OF FIGURES

LIST OF TABLES

ABBREVIATION AND NOMENCLATURES

BTDC	Before top dead center
CA	Crank angle
CA50	Combustion phasing (50% the heat release)
CI	Compression ignition
CO	Carbon monoxide
CO_2	Carbon dioxide
CR	Compression ratio, dimensionless
EVO	Exhaust valve open
HC	Hydrocarbons
HR	Heat release
HRR	Heat release rate
HTHR	High temperature heat release
IMEP	Indicated mean effective pressure
IVC	Intake valve closure
IVO	Intake valve opening
kW	Kilowatt
NG	Natural gas
LTHR	low temperature heat release
MPa	Mega pascal
N_2	Nitrogen gas
NOx	Nitrogen oxides

PPRR	Peak pressure rise rate
PRR	Pressure rise rate
SI	Spark ignition
SOHC	Single overhead cam
TDC	Top dead center
THC	Total hydrocarbons
aTDC	after top dead center
bTDC	before top dead center
dP/dt	pressure rise rate, MPa/ms
$HR_{j,Tt}$	heat release rate by the jth elementary reaction at T_t, J/mol·cm^3
N_e	engine speed, rpm
Nu_h	Nusselt number
P	pressure, MPa
P_c	gas pressure [Pa]
P_{in}	intake pressure, MPa
P_m	motored cylinder pressure, MPa
PM	particulate matter
P_{max}	maximum pressure, MPa
P_r	Prandtl number, dimensionless
Q_{in}	input heat calorie, J/cycle
R	ideal gas constant, kJ/mol·K
Re	Reynolds number, dimensionless

RI	ringing intensity, kW/m^2
S_p	mean piston speed, cm/s
T_{in}	intake temperature, K
T_{max}	maximum temperature, K
V_c	gas volume [m^3]
V_d	displacement volume, cm^3
V_{in}	volume, cm^3

1. INTRODUCTION

1.1 Background

Alternative fuels have been getting more attention as concerns escalate over exhaust pollutant emissions produced by internal combustion engines, higher fuel costs, and the depletion of crude oil. Various solutions have been proposed, including utilizing alternative fuels as a dedicated fuel in spark ignited engines, diesel pilot ignition engines, gas turbines, and dual fuel and bi-fuel engines. Among these applications, one of the most promising options is the spark ignition engine with natural gas as the supplement fuel.

Natural gas is a naturally occurring hydrocarbon gas mixture consisting primarily of methane, but commonly including varying amounts of other higher alkanes, and sometimes a small percentage of carbon dioxide, nitrogen, hydrogen sulfide, or helium.[1] It is formed when layers of decomposing plant and animal matter are exposed to intense heat and pressure under the surface of the Earth over millions of years.[2] The energy that the plants originally obtained from the sun is stored in the form of chemical bonds in the gas.

In recent years, roughly 70% of natural gas flows across the globe are transported to market destinations within the country of production, while an additional 20% flows cross international borders through pipelines and nearly 10% is moved to market destinations as liquefied natural gas (LNG). The evolution of global natural gas market is dependent on natural gas resources/reserves and production in conjunction with the ability to meet the demands and supplies.

Natural gas resources are abundant and geographically diverse. Like oil, estimates of recoverable gas have grown over the last decade as the application of horizontal drilling and hydraulic fracturing technology has enabled economic extraction of unconventional gas resources that were previously considered too difficult or too costly to produce. The EIA estimates the world’s remaining recoverable natural gas resources to be about 807 trillion cubic meter (TCM) as of year-end 2013, more than 200 times the natural gas the world currently consumes in a year.

From a global perspective, proved reserves of natural gas have continually grown over the last several decades in all regions, while natural gas production as a percentage of reserves has generally decreased (Figure 2.1). Much of the increase in conventional production comes from non-OECD (Organization for Economic Cooperation and Development) countries, with marked increases in the Middle East, China and Russia. Data from BP Statistical Review of World Energy 2015 shows that the global proved reserves of natural gas by year end 2014 is 187 TCM. Increases in the volume of proved reserves from 2000 to 2014 have been greatest in Qatar, Turkmenistan, Iran and correspondingly the largest reserves are currently found in the Middle East (79.8 TCM) and Eurasia (58 TCM).

As the natural gas production is projected to grow in almost all regions, the significant portion of this growth is likely to come from unconventional natural gas, particularly the shale gas produced in North America. From 2010 to 2013, North American unconventional gas production (primarily in the U.S.) grew by more than 30% to almost 1.6 billion cubic meter per day (BCMD), on par with Asia Pacific's total gas production. Within a few years, North America's unconventional gas output is expected to exceed the Middle East's total gas production. Around 2020, North America is expected to surpass Russia/Caspian as the largest gas-producing region.

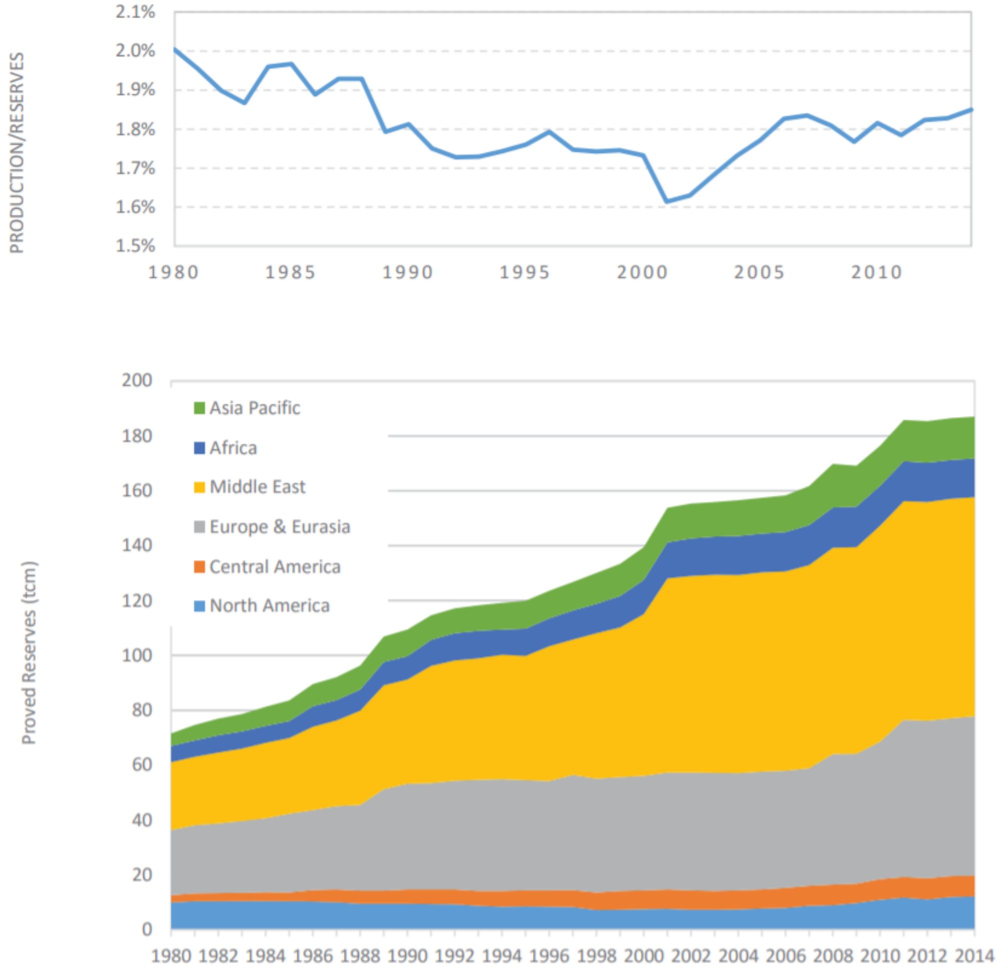


Figure 1-1. 1 Yearly Production as the percentage of Proved Reserves and Proved Reserves of Global Natural Gas by year end 2014 (Source: BP Statistical Review of World Energy 2015)

Technologies to extract unconventional gas are being applied in other regions too, particularly Asia Pacific and Latin America. The growth of unconventional gas development and production in both regions is estimated, although the pace and scale of growth are not expected to match North America's due to differences in geology, governing policies, supporting infrastructure, market maturity and development economics. In Asia Pacific, which will see the fastest rate of growth in natural gas production of any region, unconventional gas is expected to account for 80% of production growth after 2025.

Globally, two-thirds of the increase in natural gas demand through 2040 is forecast to be met by unconventional gas. Exxon forecasted in Energy Outlook 2015 that by 2040, unconventional supplies are expected to account for 35% of global gas production, up from 15% in 2010. Nonetheless, investments to maintain and expand conventional gas production also are critical to meeting the world's demand for natural gas. Conventional gas production should continue to account for the majority of growth in Russia/Caspian, the Middle East and Africa. Conventional production is expected to grow in all regions except North America and Europe.

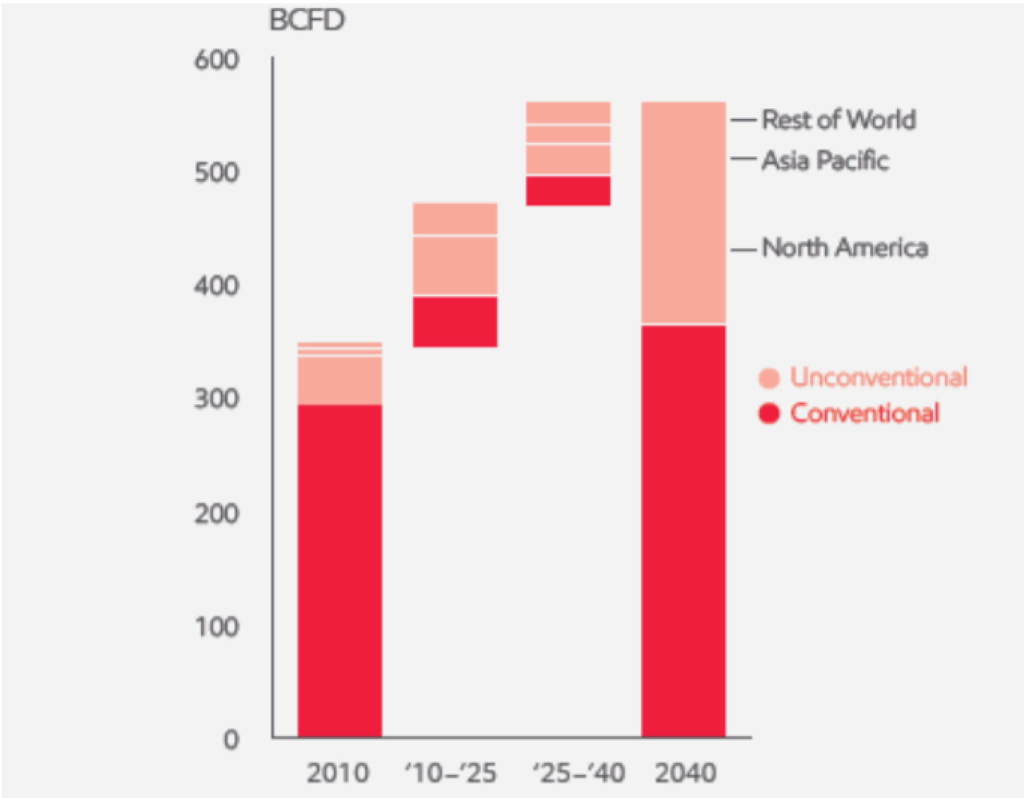


Figure 1-2. Global natural gas production by resource type (Source: Exxon Energy Outlook-2015)

Globally, two-thirds of the increase in natural gas demand through 2040 is forecast to be met by unconventional gas. Exxon forecasted in Energy Outlook 2015 that by 2040, unconventional supplies are expected to account for 35% of global gas production, up from 15% in 2010.

Nonetheless, investments to maintain and expand conventional gas production also are critical to meeting the world's demand for natural gas. Conventional gas production should continue to account for the majority of growth in Russia/Caspian, the Middle East and Africa. Conventional production is expected to grow in all regions except North America and Europe. In Asia, China has the most favorable conditions to establish shale gas production and has already begun to import the relevant technologies. It should be noted that the lack of gas infrastructure and strictly limited water resources will not allow China in short term to have production growth as fast as in the U.S.

To increase shale gas production, one needs a large number of modern drilling rigs. At present the appropriate fleet is available only in North America, where it is fully utilized. Global capacity to manufacture such drilling rigs is estimated at 300 rigs per year (Lukoil, 2014). Lack of qualified personnel, as well as a lack of capacity for the water injection necessary for hydraulic fracturing will also constrain unconventional gas production around the world.

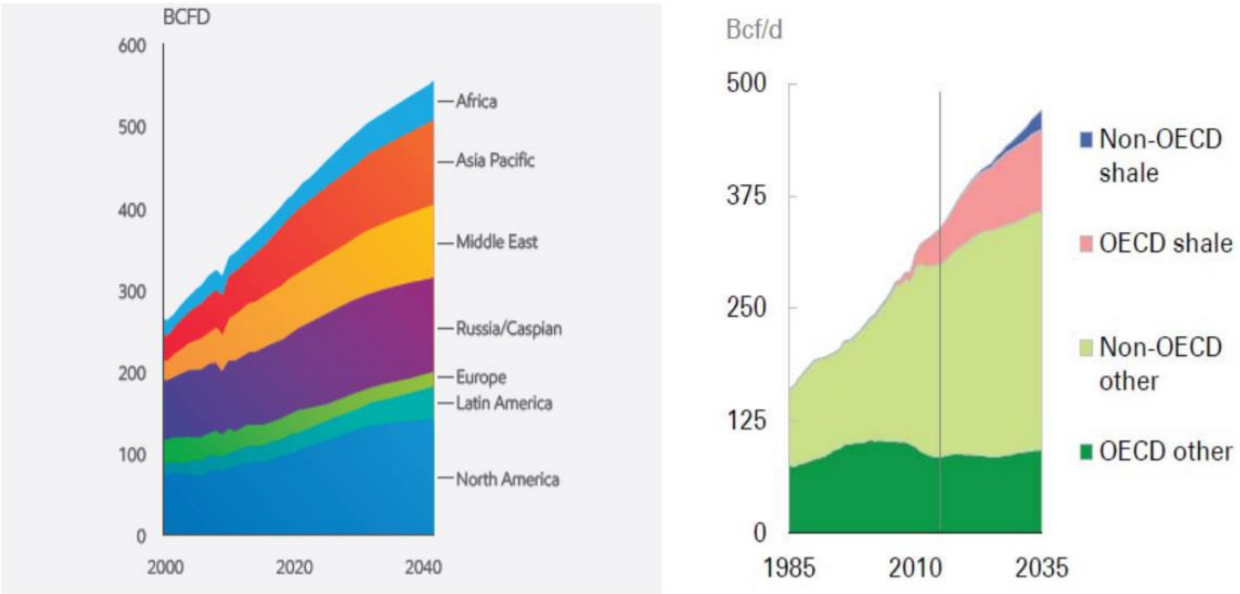


Figure 1-3. Global natural gas production and projection by region (Source: Exxon Energy Outlook-2015 & BP Energy Outlook-2016)

According to data from EIA, as of 2013 all unconventional gas production made up an estimated 18% of global gas production. The majority comes from North America, with around 358 BCM produced in 2013, one-third of which is tight gas and just under half shale gas. As of 2015, shale gas output was still concentrated in the United States.

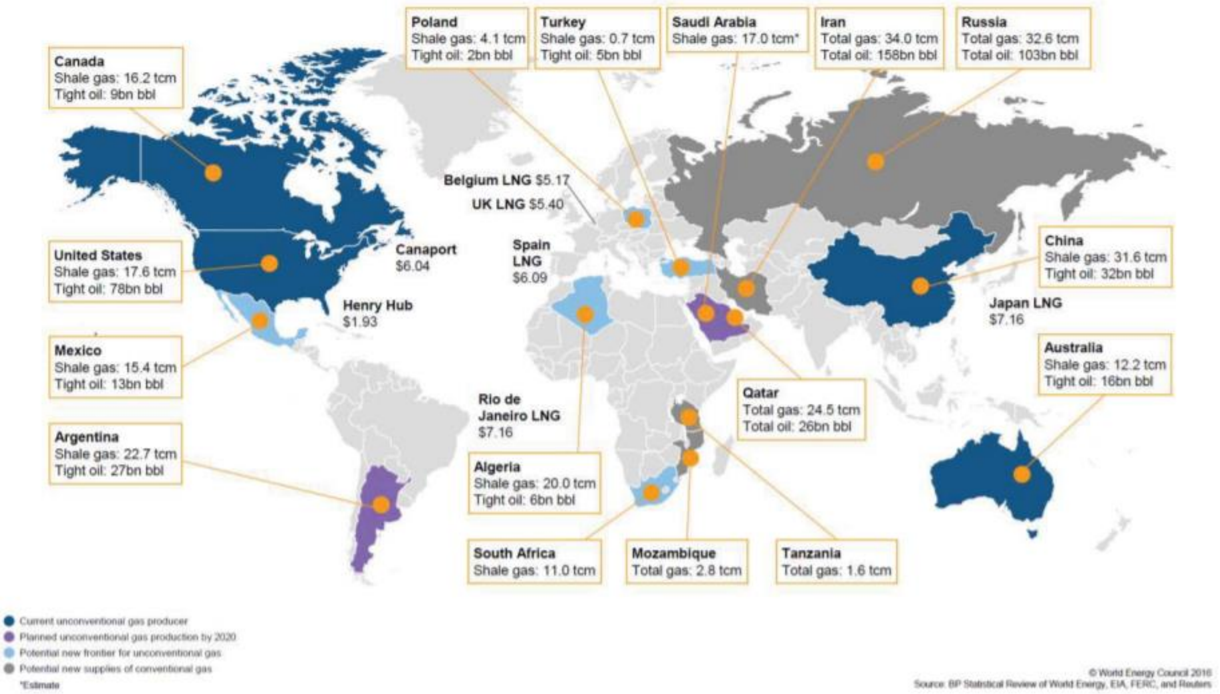


Figure 1-4. World unconventional gas phenomenon (Source: World Energy Council 2016)

Key growth factors in the demand for gas which initially was considered a by-product of oil production, were its environmental credentials (reducing CO2 emissions) and low cost in comparison with other types of fossil fuels. In Asia and Middle East, gas-fired electricity generation will replace coal and oil-powered plant respectively. Gas consumption will also continue to grow in North America. Another growth driver for gas-fired generation is the worldwide concern of safety and reliability of the nuclear power. In addition to power generation, population growth will also contribute the growth in gas consumption in the residential and industrial sectors. Natural gas is expected to supply 135% more electricity in 2040 than in 2010 and overtake coal as the largest source of electricity.

China will be the major region for gas consumption growth and by 2040 will become one of the world's largest consumers and importer of gas. Currently, China already surpassed the U.S. 9 as the world's largest electricity consumer and its demand is projected to grow by more than 140% from 2010 to 2040.

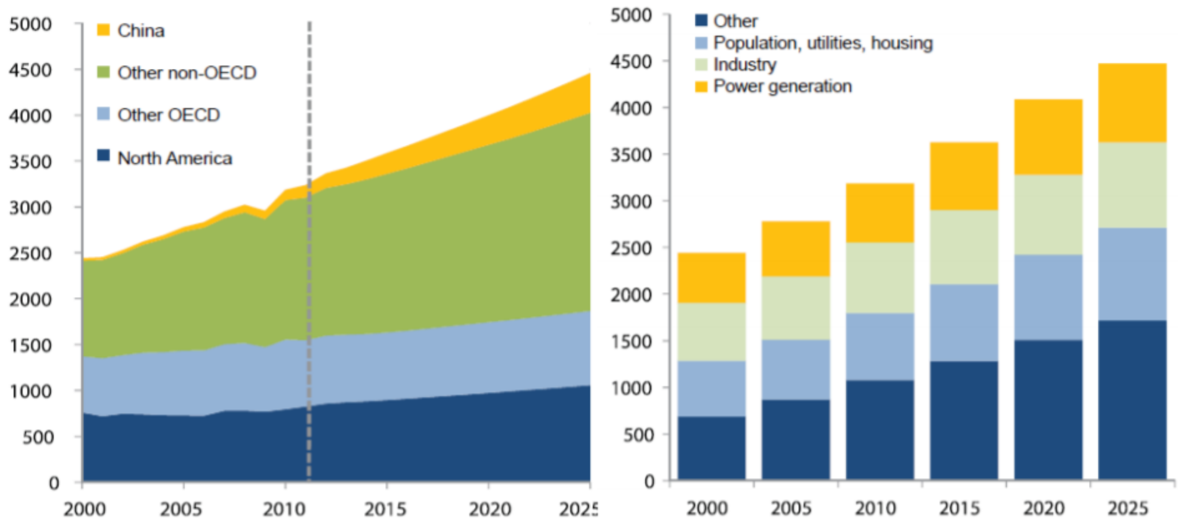


Figure 1-5. Natural gas consumption by region & sector, BCM (Source: IHS CERA, LUKOIL Estimates)

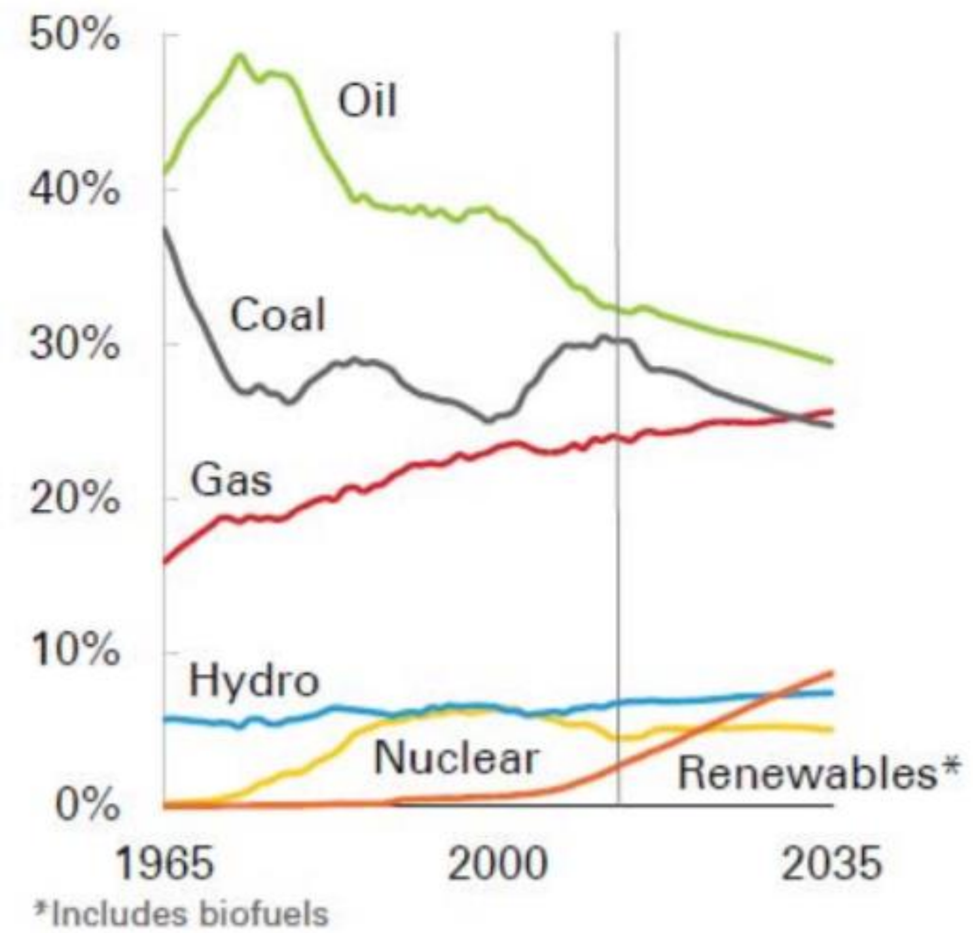


Figure 1-6. Shares in primary energy (Source: BP Energy Outlook-2016)

International trade in gas grows broadly in line with global consumption. But within that, LNG trade grows twice as fast as consumption, with LNG's share of world demand rising from 10%

in 2014 to 15% in 2035. Over 40% of the increase in global LNG supplies is expected to occur over the next five years as a series of in-flight projects are completed.

The largest producer of LNG is Qatar, which held a 31% market share in 2014. Qatar has seen a massive expansion of its capacity, up more than 63 BCM since early 2009 to reach 105 BCM. Indonesia, Malaysia, Australia and Algeria are also significant LNG exporters. Australia, whose gas production is on track to increase by 230% from 2014 to 2020, is set to become the second-largest LNG exporter behind Qatar, overtaking Malaysia. Gorgon LNG, the largest LNG liquefaction plant in Australia with capacity of 21.5 BCM has accomplished its construction and will have the first LNG production by mid-year 2016, while some others were at advanced stages of construction, with all representing more than 80 BCM of new capacity. By 2035, LNG surpasses pipeline imports as the dominant form of traded gas. The growing importance of LNG trade is likely to cause regional gas prices to become increasingly integrated.

The growth of LNG coincides with a significant shift in the regional pattern of trade. The U.S. is likely to become a net exporter of gas later this decade, while the dependence of Europe and China on imported gas is projected to increase further.

Even though the long-term contract is still central to the LNG trade, some significant changes has taken place in recent years. Over the past five years ago or so it has become acceptable in industry practice for even contractually committed LNG with a specified destination to be diverted to another market (Zhuravleva, 2009). This is become the base definition LNG arbitrage which defined as a physical cargo diversion from one market to another, which offers high price. The diversion of the cargo can be regarded as arbitrage if the cargo was initially committed to first market and initial buyer in commercial contract.

One form is seen as LNG arbitrage is reloading LNG reload is a cargo diversion and implies a purchase of the LNG cargo, discharge from vessel into the storage tank and a subsequent reloading of the LNG into another ship. The reloaded LNG is diverted to higher priced markets, thus acting as balancing forces. For example, a buyer in Spain under long term contract with a supplier like Algeria or Trinidad can unload the cargo in Spain and reload it to another or the same vessel and ship it to Japan where prices were much favorable during some years.

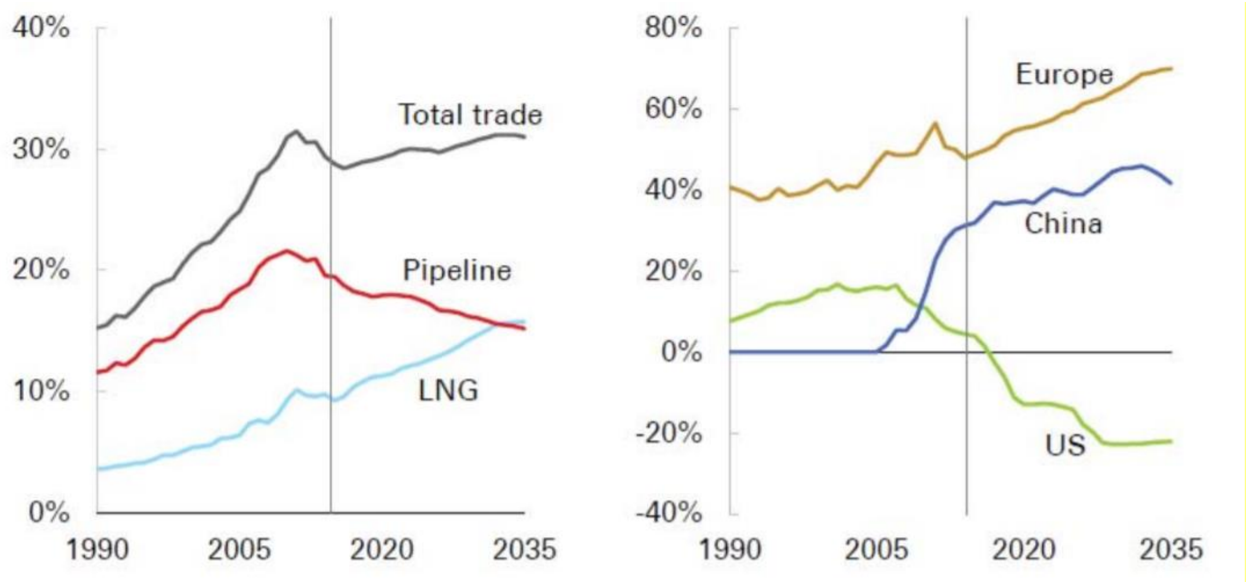


Figure 1-7. Gas trade as share of global consumption and gas imports by the largest consumers (Source: BP Energy Outlook-2016)

1.2 Engine knock

Atypical combustion, or knock, events in spark ignited (SI) engines occur in two forms of auto-ignition, spark knock and surface ignition knock. Auto-ignition is the spontaneous combustion of fuels. Given enough time, this phenomenon occurs at elevated pressures (above 10 bar) and temperatures (above 400C) (Silva, 2005). Auto-ignition causes extreme in cylinder pressure fluctuations as seen in Figure 2 below. Spark knock is the repeatable knock that is3 rhythmic and audible. Surface ignition knock is the result of the air-fuel charge being exposed to a hot surface. Surface ignition knock may occur before the spark ignites the charge of after the spark-originated flame has passed (Jang, 2016). Spark knock is affected by the spark timing. Sustained operation with heavy knock, spark or surface ignition, can result in accelerated engine damage. Figure. shows how the auto-ignition event differs from the controlled combustion event

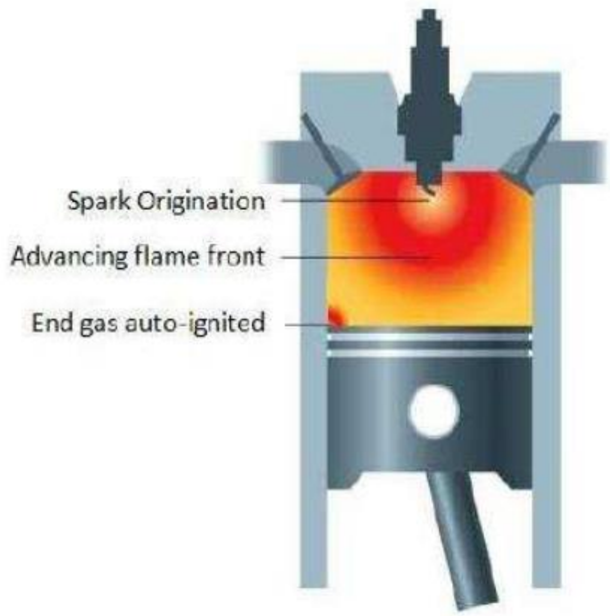


Figure 1-8. Auto-ignition of end gas that results in engine knock

The combustion is controlled when the flame propagating from the spark, the large red area in Figure 1-8, reaches the end gas before auto-ignition occurs. Figure 1-9. shows how the engine cylinder pressure trace is affected by knock (Wise, 2013). The rapid fluctuations cause high pressure rise rates. The pressure rise rate is the time derivative of the shown graphs. High pressure rise rates are exceedingly hard on piston rings in Internal Combustion Engines (ICEs) and lead to shortening the usable lifetime of the machinery (Wise, 2013). The goal of the knock tests will be to determine if these additives can be used immediately without having to modify the engine. The CFR engine provides an ideal test bed as the compression ratio and spark timing can be changed in order to induce knock.

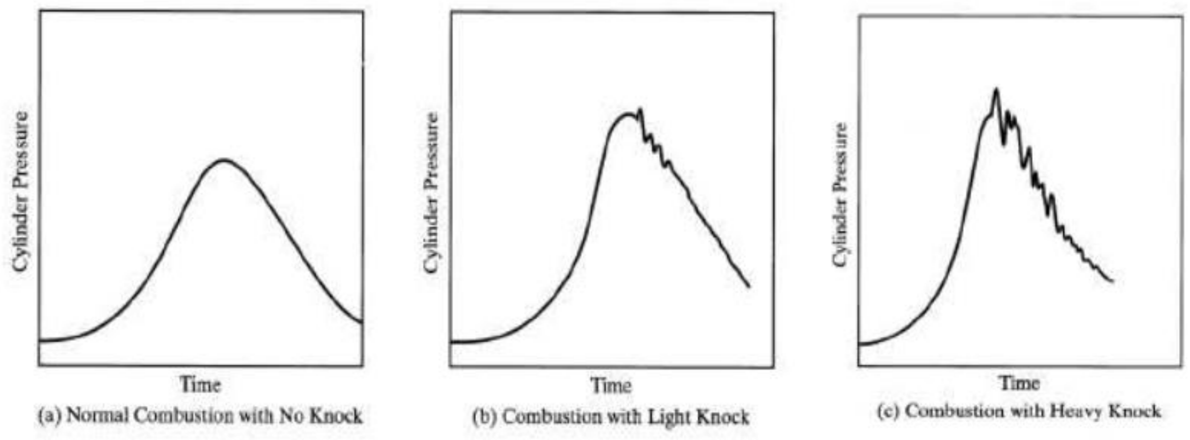


Figure 1-9. Cylinder pressure traces that increases in knock from left to right versus time in a spark ignited engine

1.3 Natural gas as an alternative for engine

The natural gas contains a mixture of methane, ethane, propane, butanes, pentanes and other hydrocarbons. It is formed from decayed animals and plants million years ago, built up in thick layers and trapped beneath the surface of the earth. Over time, intense heat and pressure changed these fossils into black oils, coals and natural gases. Natural gas is extracted from the underground formation through a well with other liquid hydrocarbons and non-hydrocarbons, which later are filtered from these components and sent through pipelines for distribution. Technology advances have enabled the domestic energy production to rapidly grow in Canada, China, Netherlands, Poland, Germany and the USA. The developments of horizontal drilling and hydraulic fracturing technologies have improved the exploration of natural gas from shale reservoirs. Shale gas, which is a natural gas that is trapped within the shale formations, has become the fastest growing source of natural gas in the USA. In the USA, at least 2 million wells of oil and shale gas have been hydraulically fractured since 1860. Today 95% of new wells are hydraulically fractured which accounts for more than 43% of total U.S. oil production and 67% of natural gas production.

A chemical substance called Mercaptan is added to natural gas to add a scent of rotten egg as a safety precaution so that leakage may be detected by the human olfactory sense (Info Comm, 2005). Inhalation of natural gas will not interfere with the body functions or cause detrimental health damage to our body. Barbotti CNG (2002) mentioned that the natural gas does not emit any aldehydes and other air toxins, which may be an issue with other fuel types.

Apart from that, natural gas is also lighter than air due to its low density. According to Clean Air Technologies Information Pool (2005), a natural gas spill would be less dangerous compared to a gasoline or diesel oil spill since the natural gas vapor would dissipate into the air and not accumulate on the ground.

The non-corrosive characteristic of natural gas is favorable to prevent oxidation of storage tanks and hence will reduce the possibility of contamination. Table 2.2 provides a comparison on the physical properties of compressed natural gas (CNG) and conventional diesel fuel. As can be seen, natural gas comprises primarily of CH4 while the hydrocarbon chains in diesel are longer and more complex. CNG also shows a lower molecular weight and specific gravity compared to diesel.

Research has shown that natural gas has a narrow combustion limit between 5 to 15 percent (Questar Gas, undated). This implies that combustion of natural gas will only take place when concentration of natural gas in the air lies in the range mentioned. Combined with its high ignition temperature, natural gas can be safely used without the high risk of accidental explosion.

Table 1-1. Properties of Natural Gas and Diesel (Alternative Fuels Data Centre, 2004)

Property	Compressed Natural Gas	Conventional Diesel
Chemical Formula	CH4	C3 to C25
Molecular Weight	16.04	≈200
Specific Heat, J/kg K	-	1800
Carbon	75	84-87
Hydrogen	25	13-16
Specific Gravity	0.424	0.81-0.89
Density, kg/m 3	128	802-886
Boiling temperature, °C	-31.7	188-343
Freezing point, °C	-182	-40-34.4
Flash point, °C	-184	73
Autoignition temperature, °C	540	316

In 2011, natural gas supply and demand reached record levels, with 23 trillion cubic feet (tcf) of domestic dry gas production and total consumption of 24.4 tcf (EIA Annual Energy Review, 2011). The average wellhead price was $3.95 per thousand cubic feet (mcf), and the natural gas price at citygate locations was the lowest (in inflation-adjusted terms) in a decade (EIA Annual Energy Review, 2011). This low price is mostly attributed to the technological improvements in natural gas recovery from unconventional sources, namely, shale rock; prior to which natural gas was recovered from the same reservoirs as oil, as they were often found together, and was priced based on oil contracts and consequently strongly correlated with oil prices (MIT Natural Gas Study, 2011). With an expanded resource base, the price of natural gas more closely correlates with the fundamentals of its recovery process.

The U.S. natural gas resource base has been estimated to be at about 2,100 tcf, including gas from shale rock ("shale gas") and Alaska natural gas, and at current production rates, this corresponds to about 90 years of natural gas supply (MITEI Natural Gas Study, 2011). The potential supply base of shale gas is very large, and may not yet be fully characterized.14 The MIT Natural Gas Study estimated that a considerable portion of the shale resource base could be produced economically at prices between $4/mcf and $8/mcf. If current oil prices remain high at $100/barrel, these natural gas prices would still remain cheaper.

The current supply outlook suggests that domestic natural gas resources could support a significant alternative fuels infrastructure, either in the form of CNG or through conversion to another fuel. For example, it was estimated that operating 50% of the current light-duty vehicle fleet on CNG would increase current natural gas demand by about one-third (Koonin, 2012). However, this could change if other competing uses develop, including LNG exports. It is also worth noting that the same technological process that enabled the supply of natural gas has also been used to extract oil from shale rock ("shale oil"), which could create a downward pressure on the need for new fuels.

When recovered from conventional reservoirs, raw natural gas is composed primarily of 70-90% methane, 0-20% ethane, and a mixture of other gases and undergoes a simple process to refine it down to specific gases and remove impurities [Table 4]. This often occurs at the natural gas plants before it is delivered through transmission and distribution pipelines. At this stage, it can undergo additional transformations into other fuels or in preparation for industrial feedstock purposes, compressed into CNG, or also be cooled down to -260ºF, where it becomes liquefied natural gas ("LNG") and can be transported or shipped in cryogenic tanks. The fact that natural gas is a gas at room temperature and as a liquid must be kept at extremely cold temperatures, has made it a commodity that is generally more expensive to transport by truck or barge, and is one of the contributing reasons why natural gas is primarily traded in regional markets, and not globally.

Table 1-2. typical composition of natural gas

Compound	Chemical Structure	Percentage
Methane	CH_4	70-90%
Ethane	C_2H_6	0-20%
Propane	C_3H_8	
Butane	C_4H_{10}	
Carbon Dioxide	CO_2	0-8%
Nitrogen	N_2	0-5%
Hydrogen sulphide	H_2S	0-5%
Rare gases	A, He, Ne, Xe	trace

1.4 Problem statement

Hydrocarbon emissions (HC) are the problems of SI engines. These emissions can be contributors to air pollutant, which have serious environmental and health implications. The emission control technology such as after treatment systems are being developed, however, expensive, complicated and reduce the main advantage of SI engines. Moreover, limitations of vehicle emission regulations especially SI engine are getting to be more stringent in all over the world. Therefore, to develop engine combustion systems that offer high efficiency, but low emissions motivate engine researchers to study the utilization of high volatile fuels, such as gasoline and alternative fuels, for SI engines. Natural gas, which is made from various renewable resources, is known to be very suitable as a sustainable alternative fuel for SI engines[3][4].

Engine experiments are very important to analyze and explain the real phenomena of combustion and emission characteristics of an IC engine. However, it is usually costly, need big effort and complicated, therefore only a few of researchers interested to do so in their study. Traced from the extensive research literature, only a few references on the experimental study of SI engines and more rarely for fueled with natural gas. Most of the studies are using simulation and numerical method. Thus, to fill these lack information, by not ruling out the important role of simulation method, sequential and complementary experimental works are conducted to achieve a better understanding on the combustion process and emission characteristics of SI engine fueled with natural gas.

1.5 Objective of the study

The objectives of this study are as follows:

(i) The primary objective of this study is to improve the efficiency and emission characteristics of SI engine fueled with natural gas.

(ii) To investigate the effects of a various ignition timing on performance and emission of SI engine fueled with natural gas.

(iii) To investigate the effects of small propane content of the methane-propane blends on performance and emission of SI engine.

1.6 Scope of the study

This thesis makes efforts to improve the efficiency and emission characteristics of SI engine fueled with natural gas. The scopes of the study include:

(i) Explain the effect of the gas composition, with the content of propane from 5% to 20% by volume.

(ii) Thermal efficiency was evaluated between different percent of propane during the combustion process.

(iii) Measurement and evaluate the performance of engine via IMEP, Brake Torque and Power.

(iv) Measurement and analysis of regular exhaust emissions such as CO, HC, NOx.

(v) Combustion efficiency calculated based on the CO and HC emissions.

(vi) The obtained results may only valid for the simulation model was used in this study.

1.7 Thesis organizing

This thesis was organized into five chapters.

Chapter 1 described about natural gas fuel and advantages/drawbacks after providing the background of research. Then current research trends which are related to those problems were introduced and drawn the objectives of this study.

Chapter 2 described a review of the most important findings of the previous work related to the objectives of the present study is given. This review summarized the application of natural gas in SI engine, the experimental works, and simulation studies. Based on this review, some issues can be inferred as a starting point for the present study.

Chapter 3 provided the calculation methods setting in detail where CHEMKIN & Senkin chemical kinetics rate code with SI engine model. Low temperature reaction, high temperature reaction, combustion duration, knocking and misfire were described stepwise.

Chapter 4 was focused on effects of ignition timing and mixing ratio of methane, ethane and propane blends on brake torque, power, thermal efficiency, ignition delay.

Chapter 5 summarized the major results from this thesis.

2. LITERATURE REVIEW

2.1 Introduction

Due to an increase in pollution from automotive vehicles running on conventional liquid fuels, alternative fuels that could result in similar performance became potential candidates to be used as substitutes in short and long-term plans [5]. In the upcoming years, alternative fuels are expected to substitute conventional fuels concerning environmental and energy security issues. While the future trend in the transportation sector is inclining toward electric vehicles, a leap jump is unlikely due to existing infrastructure and resources. There is a need for an alternative transitional fuel that can lead toward 21st century ambitions of zero-carbon emission. As a result, natural gas has become the leading candidate to fill the gap in recent trends [6].

Natural gas (NG) is a naturally occurring hydrocarbon gas mixture consisting primarily of methane, but commonly including varying amounts of other higher alkanes, and sometimes a small percentage of carbon dioxide, nitrogen, hydrogen sulfide, or helium. The composition of NG fuel varies with location, climate and other factors. NG is a fossil fuel used as a source of energy for heating, cooking, and electricity generation. It is also used as fuel for vehicles and as a chemical feedstock in the manufacture of plastics and other commercially important organic chemicals [7].

The low cost of natural gas relative to diesel and gasoline combined with various emissions related regulatory measures continues to create significant interest in natural gas as an alternative fuel for internal combustion engines. Engine makes have responded by supplying new, purpose built natural gas engines in sizes ranging from small light-duty engines of a few kW to low speed two-stroke marine engines of over 60 MW [8].

Natural gas engines can be categorized based on numerous parameters including mixture preparation (premixed or non-premixed), ignition (spark ignition or diesel pilot) and the dominant engine cycle (Otto or Diesel). One common categorization is, Figure 2-1 [9]:

- Premixed charge, spark ignition, natural gas only
- Premixed charge, diesel pilot ignition, natural gas/diesel dual fuel
- High pressure direct injection of natural gas, diesel pilot ignition, natural gas/diesel dual fuel

While the above grouping adequately covers commercial engine sizes up to about 2.5 L/cylinder, when larger engines are also considered, it creates some challenges in presenting common concepts between some of the different approaches. More specifically, lean burn dual fuel engines ignited by a small (<~5% fuel energy) diesel micro-pilot share more in common with lean burn SI engines than they do with dual fuel engines using a much larger diesel pilot (>~15% fuel energy). It also does not cover some concepts in the development stage. The following categorization is more general and reflects common concepts between the different approaches:

- Stoichiometric Otto cycle engines
- Lean burn, Otto cycle engines
- Dual fuel mixed cycle (combination of Otto and Diesel) engines
- Diesel cycle natural gas engines

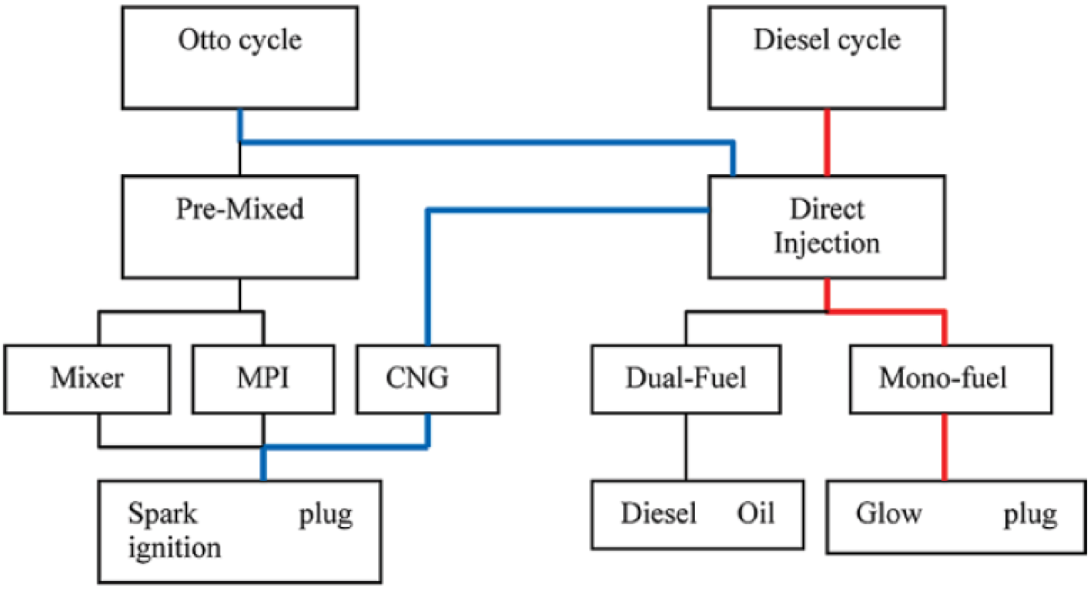


Figure 2-1. Combustion modes for compressed natural gas (CNG) in internal combustion (IC) engines [9].

There are three basic application modes for NG engines or vehicles: pure NG engines/vehicles including CNG and LNG, gasoline/NG bi-fuel mode refitted from spark ignition (SI) gasoline engines, and diesel/NG dual fuel (DF) mode based on compression ignition (CI) diesel engines. NG has high ignition temperature and can be hardly burned by compression. Initially, gasoline vehicles are modified into gasoline/ CNG bi-fuel mode by adding a CNG gas cylinder. The bi-fuel vehicle has two fuel supply systems and only one system can be used at a time. NG has higher octane number than gasoline, which exhibits better antiknock performance. However, when fueled with NG, the bi-fuel vehicles cannot improve the thermal efficiency by increasing the compression ratio (CR), to meet the requirement of antiknock for gasoline. In other words, advantages of NG cannot be exhibited in bi-fuel mode. Accordingly, pure NG vehicles are designed and manufactured with higher CRs (11e13) than those of gasoline ones (9.5e11) and thus the thermal efficiency is improved. Theoretically, CRs of NG vehicles can be further increased when fueled with pure methane, and in practice are generally no more than 13 concerning for the complicated composition and the localized difference of NG. NG can also be applied on diesel vehicles by using dual fuel mode. The dual fuel vehicles also have two fuel supply systems and they must work simultaneously, because NG need to be burned by diesel, which is compressed ignited. As a result, the CRs of dual fuel engines are commonly higher than 16.

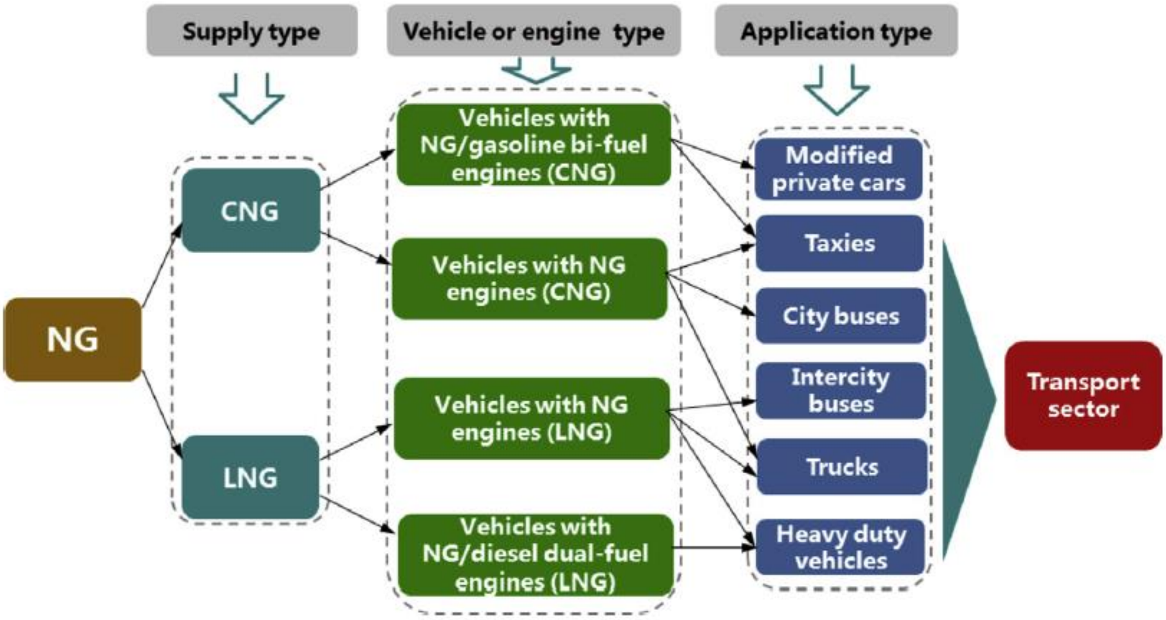


Figure 2-2. Engine type and corresponding application type of NG engines [7]

2.2 Engine classifications for automotive purpose

2.2.1 SI engine

The majority of NGVs use port fuel-injected natural gas engines. However, as natural gas displaces some of the air in the intake manifold upon injection, a reduction in volumetric efficiency occurs, which in turn leads to proportional reduction in power. One way to avoid this is to inject fuel directly into the combustion chamber [10]. Port fuel injectors have operating pressures of the order of 0.5 MPa, which are lower than the pressures of 8 MPa (or higher) used to overcome operating cylinder pressures for direct gas injection. Direct natural-gas injection in SI (and CI) engines requires development of specialty high-pressure gaseous injectors [11,12,13] which currently are not available in the open market.

In addition, direct natural gas injection has been shown to extend the fuel-lean operating limit of normal engine operation compared to port fuel injection [14]. For example, a port-injected natural gas engine running very fuel-lean mixtures (fz0.6) had a COV of IMEP of more than 10%,while the same engine using an in-cylinder injector built into the spark plug was operating at the same equivalence ratio with a COV of IMEP of less than 5% [12]. This is a result of increased mixture turbulence in the cylinder, in addition to locally fuel-rich mixtures that become available close to the spark plug. Flame propagation is also accelerated, resulting in higher rates of energy change of the working fluid and higher thermal efficiencies. However, higher levels of NOx were recorded, caused by high combustion chamber temperatures [12]. Contrary to this, reference [13] reports that NOx is lower for direct injection engines because of increased charge stratification. It is likely that other engine parameters play a significant role, as reference [12] goes on to say that retarding the spark timing (relative to combustion TDC) can reduce NOx emissions while affecting power output. Similar performance effects are also seen with increasing compression ratio in direct injection engines [15], where a compression ratio of 12:1 is reported as the optimum compromise value between performance and emissions for direct injection natural gas SI engines.

When fuel-injection timing is varied care must be taken not to start injecting the fuel into the cylinder too late. The experiments reported in reference [13] indicate early injection timing (during the intake stroke) increases the combustion chamber pressure and rate of energy change of the working fluid, resulting in favorable power output. These trends are shown when 150^o BTDC to 180^o BTDC (TDC here relates to combustion TDC) are the start-of-injection timings during the compression stroke and 180^o BTDC to 210^o BTDC are the start-of-injection timings during the intake stroke. The optimum injection timing in this work is shown to be at 180^o before combustion TDC. Retarded injection timing during the compression stroke does not allow sufficient time for the fuel to mix and oxidize, resulting in poor flame propagation as well as reduced and delayed peak rate of energy change of the working fluid. This can be seen exhaust emissions follow a similar trend to that of engine power output [13]. For example, NOx emissions increase significantly with injection timing earlier in the compression stroke, while there is little effect with further injection timing advance into the intake stroke. HC emissions follow the opposite trend to NOx emissions while CO emissions did not change significantly with fuel-injection timing [13].

2.2.2 CI engine

Most of the dual-fuel engine studies to date are conducted in conventional diesel engines modified to induct natural gas into the combustion chamber via the intake manifold, while maintaining the original in-cylinder injector for pilot fuel injection. Like port-injected SI engines, inducting natural gas via the intake manifold reduces volumetric efficiency, and therefore potential power output (c.f. equations (8)e(11)). In some studies, both the pilot fuel as well as natural gas are injected directly into the cylinder via the same injector [16,17]. Dual-fuel engines with direct natural gas injection maintain power output and thermal efficiency levels compared with conventional non-dual-fuel diesel engines [16]. Comparatively lower emissions of NOx and particulate matter were also recorded [16].

Further improvement in direct natural gas injection dual-fueled CI engines can be obtained by varying the injection pressure of the natural gas jet and the diesel pilot fuel. For normal non-dual-fuel CI engines, increasing the fuel-injection pressure improves fuel atomization upon injection as well as fuel-air mixing rates prior to combustion [17,18]. A similar effect occurs in direct natural gas injection dual-fueled engines [17]. Increasing the injection pressure of both the diesel pilot fuel and the natural gas injection (from 21 MPa to 30 MPa) results in a shortened ignition delay of the pilot fuel [17] because of faster mixing between the pilot fuel and air during the ignition delay period. Higher combustion-progress rates are recorded, resulting in a shorter overall combustion duration [17]. NOx emissions are increased slightly compared with lower injection pressure conditions, in addition to lower HC, CO and significantly lower smoke emissions. These emission trends are caused by better levels of mixing between the pilot fuel, natural gas and air, in addition to the faster combustion-progress rates. Thermal efficiency levels were not significantly affected by varying injection pressure []. These emission trends are shown in Fig. 33 (which are plotted in specific terms of mass per unit gross indicated kilowatt hour, GikWhr, where GikWhr is the energy derived from the indicated rather than the brake power). The data was obtained at a particular intake oxygen mass fraction (YintO2) of 0.19 in the total intake charge, and combustion timing (50% IHR ¼17.5 ATDC). The figure also shows the gross indicated specific fuel consumption (GISFC), which is proportional to the inverse of thermal efficiency.

These trends vary significantly with the operating conditions, especially with engine speed [17]. At low speeds, the higher injection pressure (30 MPa) have more influence on combustion quality than at high speeds. This is because the higher injection pressures increase turbulence in the cylinder at low speeds, while at high speeds cylinder turbulence is inherently high because of the piston motion. In addition, at a particular engine speed, increased turbulence brought on by the higher injection pressures is more significant at higher loads [17]. This can result from a larger pressure difference between the higher fuel-injection pressure and chamber pressure, compared with the lower injection pressures (21 MPa) which are comparatively more effective at low loads. These parameters influence the level and rate of mixing in the cylinder, which in turn influence emission levels. For example, particulate emissions are lowered to a larger extent at higher loads than at lower loads [17].

A developed strategy to improve SI engine efficiency is direct injection (DI) of gasoline fuel as contrary to port fuel injection. In direct-injection SI engines, air is sucked into the cylinder, and the fuel is injected directly into the cylinder during the intake or compression strokes (early enough for evaporation and mixing to occur before the time of spark). By using direct injection, the fuel can be maintained away from the crevices (reducing the emission of CO and UHC) and cylinder walls (reducing heat losses). Furthermore, the cooling effect of evaporation reduces the in-cylinder temperature. Thus a higher CR can be used without suffering engine knock[39].

2.3 Review of previous study on engines with natural gas

The study presents experimental results on engine efficiency and exhaust gas concentration at different ignition and injection timing for compression natural gas (CNG) engines performed by MK Hassan et al [19]. The objective of this experiment is to study the effects of injection and ignition on the brake torque, brake power and exhaust emissions at the maximum brake torque (MBT) interval. CNG DI engine efficiency is measured using a computer-controlled eddy current dynamometer. The emissions components CO, CO_2, HC and flame ionization for THC were measured by Horiba analyzer. The recorded exhaust concentration levels are related to the engine speed, ignition timing and injection timing. The results show a linear relationship between the NOx concentration and the injection timing. In addition, advanced ignition timing can increase the NOx component. For the HC component in the flue gas, it tends to be low at the early injection timing but increases when the late injection process is applied. Low CO concentrations occurred at the time of late injection and the lowest emissions were 0.011% when we used 30^0 bTDC ignition at 360^0 CA injection time. CO composition results show that the most influencing factor on CO development is ignition timing.

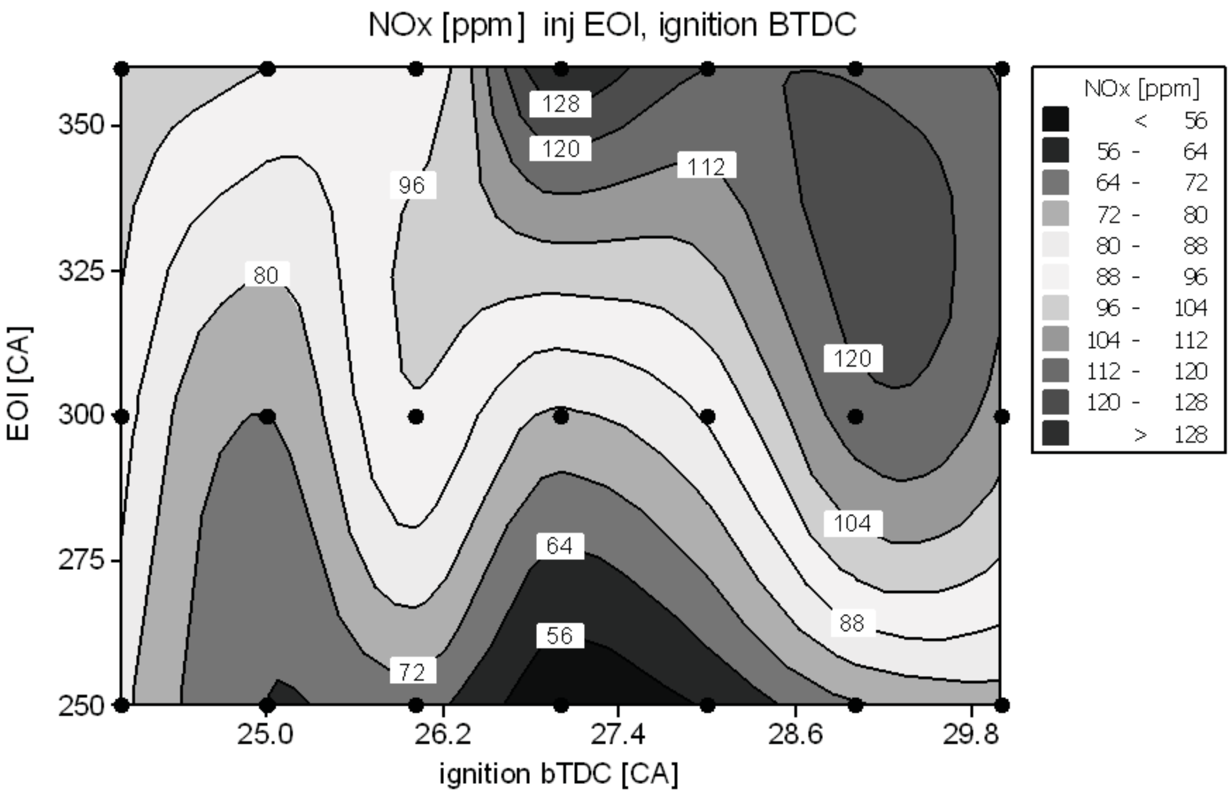


Figure 2-3. NOx concentration [19]

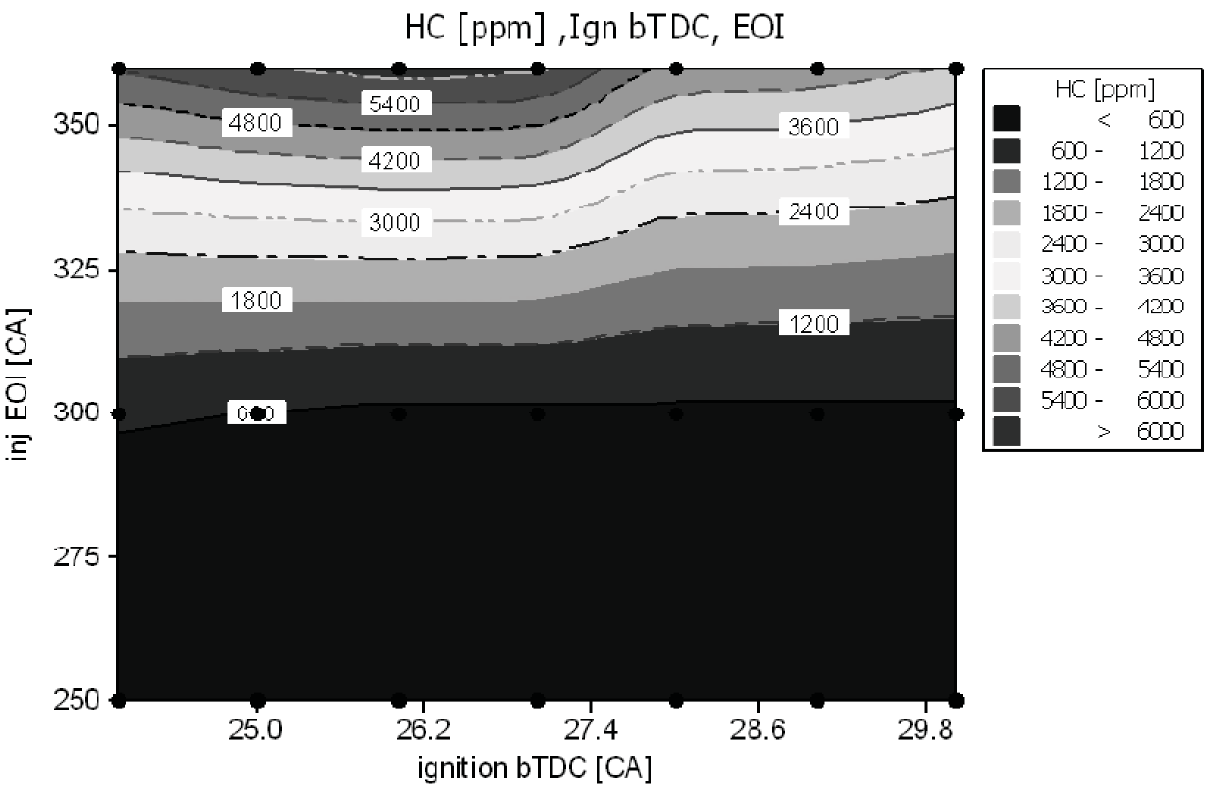


Figure 2-4. HC concentration [19]

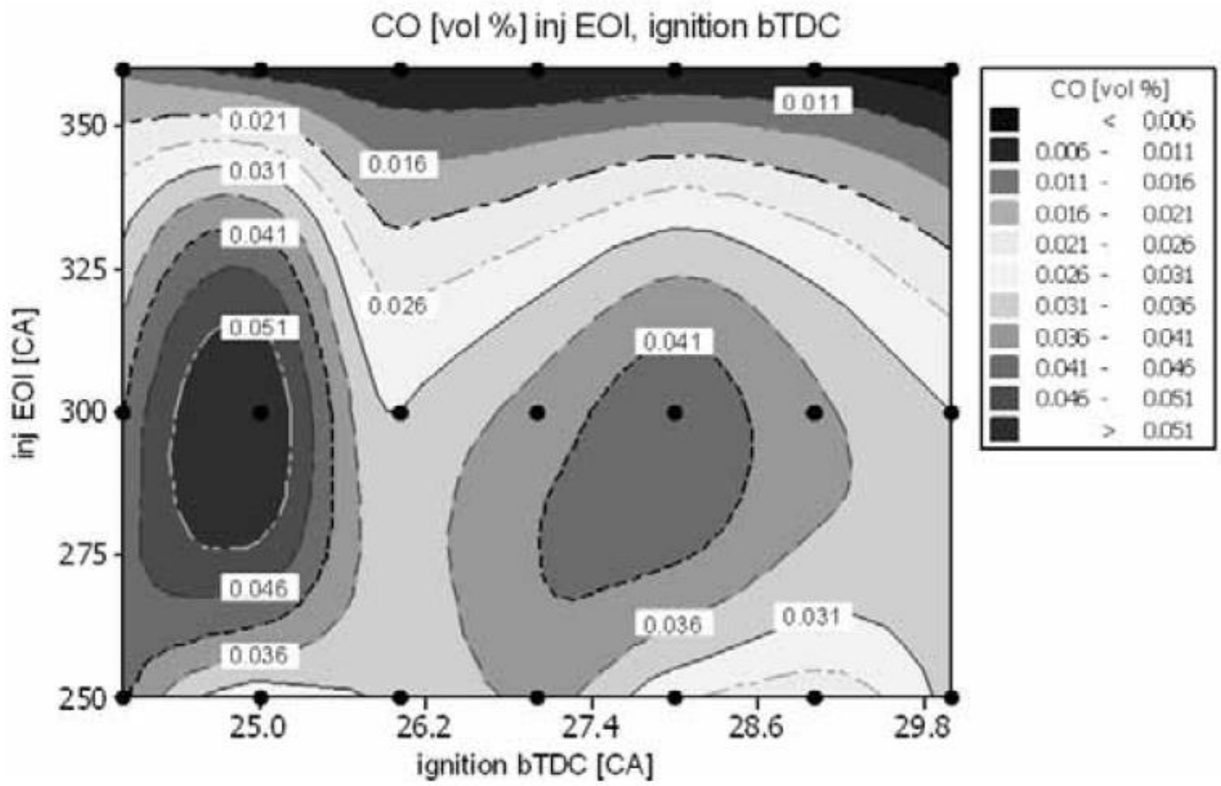


Figure 2-5. CO concentration [19]

Diming et al. conducting an experimental investigation on the effects of exhaust gas recirculation (EGR) and spark timing on the combustion, performance, and emission characteristics of a China-VI heavy-duty, natural gas engine fueled with high-methane content [20]. The results showed that increasing the EGR rate extends the spark timing range and slows the combustion. This then increases ignition delay, prolongs combustion duration, and decreases heat release rate. Peak in-cylinder pressure (PCP) and indicated thermal efficiency (ITE) initially increase because of higher boost pressure with increasing EGR rate. However, as EGR rate increases further, PCP and ITE begin to decrease because of the deviation of combustion phasing. Lower in-cylinder temperature caused by higher EGR rate may cause nitrogen oxide (NOx) emissions to reduce significantly, while total hydrocarbon (THC) and carbon monoxide (CO) emissions increase, and THC emissions could increase exponentially at high EGR rates. In cylinder pressure, temperature, and heat release rate increase with early spark timing, but the rate of increase is reduced at higher engine speeds. Early spark timing causes THC and CO emissions to increase at part-load conditions, whereas there is little change at full-load conditions. NOx emissions also increase with early spark timing because of the higher in-cylinder temperature.

Governmental policies on renewable energy and environmental act are aggressively being enforced to mitigate recent climate change [21]. Natural gas is not renewable, but it is the most abundant and has the lowest Lifecycle CO2 emission among fossil fuel. Realizing such promising alternative, many logistics and transportation companies are converting their existing diesel-fueled vehicle to CNG-fueled. Researchers have shown that CNG engines offer advantages compared to diesel and gasoline engines such as high efficiency and low emissions. Prior to this work, a 4.3L 4- cylinder diesel engine was modified and retrofitted with a CNG mono gas system. However, it was observed that the engine, CNG-fueled combustion is not stable especially at idling speed. The purpose of this study is to optimize the ignition timing best suited for idling both in normal operating mode (700-850 rpm) and in cold start mode (1000-2000 rpm). The ignition timings tested were 20oBTDC and 25oBTDC. The measurements were made at engine speeds from 700 to 2500 rpm. Some irregularities were found in the result, but overall, the ignition timing 25oBTDC is better than 20oBTDC in terms of fuel consumption and exhaust gas emissions. For this particular system, the results recommend that the idling engine speed should be at 700-800 rpm and 1500 rpm during the normal mode and cold start mode respectively. The use of engine speed of 1000 to 1300 rpm should be minimized to reduce overall exhaust gas emissions.

Javad Zareei et al [22] was to investigate the effect of ignition timing and hydrogen fraction on the performance and exhaust emissions of an engine. The engine was converted from port injection system to direct injection with enrichment hydrogen to natural gas at WOT condition and engine operation at 4000 rpm. The results showed that when hydrogen volume fraction was raised from 0 up to 50% at 20 bar injection pressure and at the ignition timing from 19 to 28 °CA BTDC, a good stratified effect was achieved. Also, performance results demonstrated that fuel conversion efficiency, power output, and torque were increased up to 3.7% and the brake thermal efficiency first rose and then fell with an increment of hydrogen enrichment to CNG. The peak cylinder pressure and temperature with the advance of ignition timing increased. In addition, the advance of the ignition timing caused NOx, HC, and CO emissions to decrease but with the increase in hydrogen fraction in the blend of fuel, HC, and CO dropped and NOx emission rose. Additionally, this study concluded that the 30% blends of hydrogen and 21° BTDC ignition timing can optimize engine performance and emission without any engine modification.

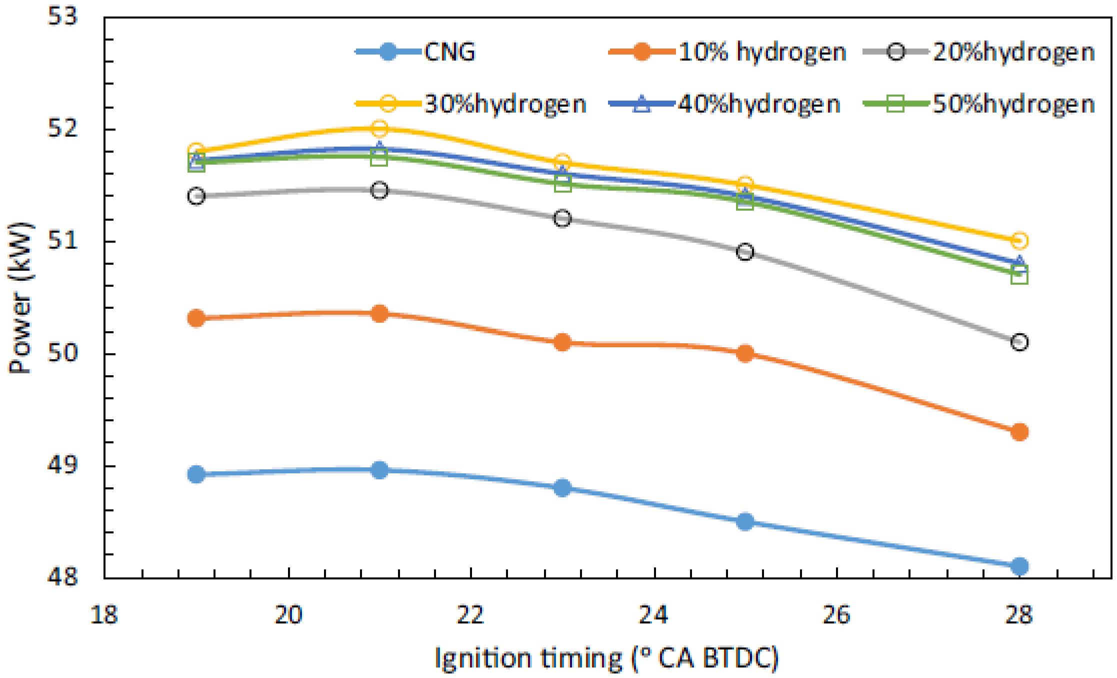


Figure 2-6. Engine output power versus ignition timing for different hydrogen volume fractions [22]

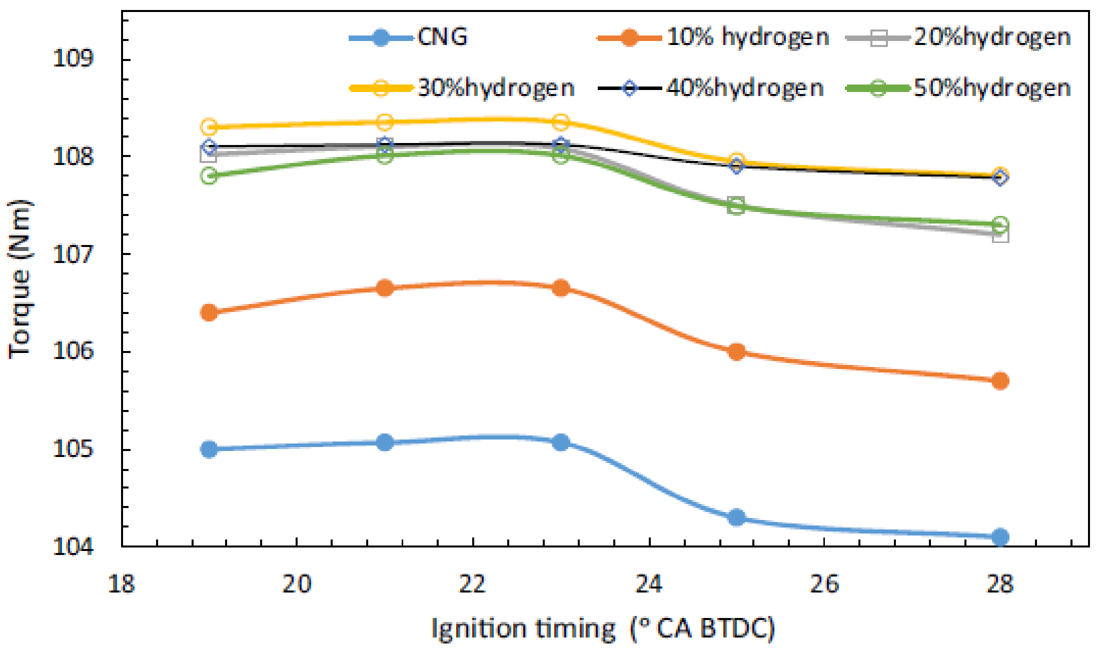


Figure 2-7. Engine torque versus ignition timing for different hydrogen volume fractions [22]

2.4 Summary

Natural gas is a practical fuel for SI engines, and for CI engines in the dual-fuel mode, with varying degrees of success. Natural-gas fueled SI engines can operate at higher compression ratios resulting in similar or slightly higher thermal efficiencies compared to gasoline-fueled engines. Natural gas injection or induction in the intake manifold adversely affects volumetric efficiency hv. The 2.2% lower LHVf and 17.2% higher (F/A)st of natural gas compared to gasoline also affects power. Overall, the product of all three factors affects power output (equations (8)e (11)) resulting in 10e15% reduction in power compared to gasoline- fueled engines. Direct in-cylinder injection of natural gas avoids the volumetric efficiency effect. Injecting natural gas into the cylinder requires high pressure (of the order of 30 MPa) and as a result specialist injector are required. In addition, ideal power levels are obtained only at injection timings where high NOx emissions are produced. The higher hydrogen-to-carbon ratio of natural gas compared to conventional gasoline higher leads to relatively minor reductions of CO2 emissions compared to gasoline engines. These engines use high compression ratios and advanced spark timing (compared to typical gasoline engines), which generally increases NOx emissions. Corresponding reductions in unburned non-methane HC and CO emissions are also reported. Most of HC emissions are methane, so despite reductions in overall HC emissions, the methane emissions of natural-gas fueled engines are higher than those of gasoline engines. EGR can be used to reduce NOx emissions but it also results in increasing HC and CO emissions. The lean-burn strategy generally resolves emissions issues, but unburnt methane emissions remain relatively high. Ultra-lean operation results in misfire and unstable engine operation. Fuel-lean operation of natural gas engines is desirable in order to reduce specific NOx emissions. Natural gas engines running stoichiometric fueleair mixtures produce lower thermal efficiencies compared to lean-burn natural gas engines because of the lower specific heat ratio of the charge []. Comparable NOx levels to lean-burn natural gas engines are reported if spark timing is advanced relative to TDC appropriately (compared to typical gasoline engine spark timing settings).

In CI engines natural gas is ignited by the use of a pilot fuel (that can ignite with typical CI compression ratios) in a special mode of operation known as “dual-fueling”. A small amount of “pilot” high-cetane fuel is injected directly into the cylinder, which provides an ignition source for the premixed natural gas air mixture. There is no significant loss in power in dual-fuel operation compared to conventional CI-engine operation provided a sufficient amount of natural gas air mixture can be admitted in the chamber. In some of the literature the induction method employed prevents a premixed natural gas air mixture to form in the inlet manifold (by keeping the natural gas supply separate from the incoming air until very close to the intake valve). This minimizes the volumetric efficiency penalty of natural gas induction or injection in the intake manifold, but it also results in a loss of power at higher speeds because comparatively lower amounts of natural gas are inducted per cycle. Failure of the pilot fuel to ignite the entire natural gas-air charge at the low and intermediate loads (as a result of low charge temperatures) causes lower thermal efficiencies. Low engine operating temperatures at these loads (caused by pockets of local air-fuel mixtures that are too lean to support combustion as the flame propagates along the chamber) also result in lower NOx emissions compared to normal CI-engine operation. Conversely, HC and CO emissions are significantly increased, while at high loads NOx, HC and CO emissions are comparable to normal CI-engine levels. A variety of alternative high-cetane fuels can be used as pilot fuels, while water-in-fuel pilot fuel emulsions and water injection can be used to reduce emissions at select equivalence ratios.

In order to derive the full benefits of natural-gas fueled SI and CI engines, extensive performance and emissions optimization of both engine types is required. It is likely that a combination of different engine operating modes is needed. SI engines operating at high loads can employ high EGR rates as well as catalytic converters to reduce NOx emissions; while advanced spark timing, high compression ratios and forced induction can be used at all conditions to improve power output. For example, a combination of turbocharging, high compression ratio, catalytic converters and engine control unit (ECU) reprogramming allows increased power and efficiency while at the same time reduces NOx and CO2 emissions [23]. In dual-fueled CI engines, natural gas can be used in smaller proportions compared to the pilot fuel at lower loads to reduce emissions of HC and CO (i.e. the pilot fuel would provide more than 50% of the total fuel energy input). High-pressure pilot fuel injection (of the order of 100 MPa) can provide more ignition points distributed more extensively throughout the natural gas air charge. An increased number of smaller-than-standard diesel injector holes allow better atomization and mixing of the pilot fuel with the natural gas [24,25]. Uncooled EGR at low to intermediate loads speeds up combustion progress and improves combustion efficiency, and therefore reduces unburned HC as well as CO emissions. In addition to these modifications, an additional fuel that can improve the burning characteristics of natural gas can be included in the intake charge. Hydrogen is effective at increasing the flame speed of combustion in both SI and CI natural gas engines, as well as reducing COV and increasing engine stability. Small amounts of inducted hydrogen increase thermal efficiency and improve EGR tolerance.

Storage of natural gas is an issue with NGVs. Natural gas requires modified or new types of fuel storage and supply systems due to its low density (typically stored in compressed gas tanks atz20 MPa). These fuel storage options generally carry less fuel energy per mass or volume than the typical diesel or gasoline fuel tank (Fig. 5). Exhaust gas catalysts are effective at improving NGV exhaust emissions, but their maintenance and disposal costs should be taken into account. NGVs that have secondary fuels on-board also require secondary fueling systems (dual-fuel CI engines, engines with hydrogen addition etc). As a result of limited fuel storage, NGVs face the problem of a limited operating range. Operation over long distances requires more frequent refueling. Short distance travel in inner cities and typical commuter routes helps reduce photochemical smog as well as carcinogenic hydrocarbon emissions (but the increased methane emissions contribute to greenhouse gas buildup). These areas allow easier installation of a natural gas refueling infrastructure.

As natural gas is not a renewable fuel, renewable sources of methane are required in order to ensure its use in the long term. The technology to produce renewable methane from waste biomass at reasonable cost and quality has not yet been developed. Biogas (harvested from landfills, for example) are the only major sources of renewable methane. This biogas can be used in natural gas engines fairly readily; however, this would result in reduced power and efficiency caused by contaminants in the biogas, such as CO2 and Sulphur dioxide (SO2). Biogas requires purification prior to use, increasing production complexity and cost of the fuel in the process. Overall, natural gas engines can supplement the existing engine portfolio and assist in the conservation of the limited supply of crude oil. Incorporation of other engine technologies (e.g. turbocharging, higher compression ratios, control of autoignition) is required to improve operating range and power output. Efforts to improve the current position of renewable fuels should be continued (where well-to-wheel life cycle analyses of the entire production, supply and use is considered) in order to further reduce dependence on conventional fossil fuels.

3. CALCULATION METHOD

3.1 Calculation methods

3.1.1 CHEMKIN overview

The original CHEMKIN was published in 1980 at Sandia National Laboratory in USA [28-29]. CHEMKIN II is a revised, generalized version of CHEMKIN in 1990. CHEMKIN II expanded this capability, with inclusion of an accurate and efficient means of describing pressure-dependent reactions. Furthermore, CHEMKIN III was revised in 1996 with capability of the multi fluid generalization and currently it's developed until Chemkin4.1. In this study, CHEMKIN II was used since it is provided as an open code. The CHEMKIN package is composed of two blocks of FORTRAN code and two files:

- Interpreter (code)
- Gas-Phase Subroutine Library (code)
- Thermodynamic Database (file)
- Linking File (file)

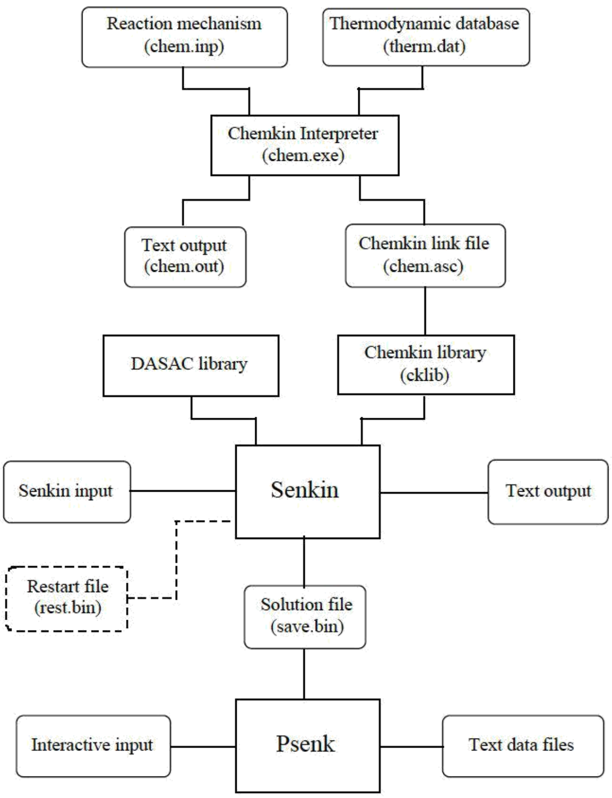


Figure 3-1. Relationship of Senkin to the Chemkin preprocessor and the associated input and output files

3.1.2 The governing equations and assumptions

In this study, there are four kinds of governing equations and assumptions for calculating the elementary reactions.

(1) All the chemical species of the gas is ideal gas.

$P_cV_c = nRT_c$

P_c: gas pressure [Pa]

V_c: gas volume [m^3]

n: mole [mol]

R: gas constant of air [J/mol/K]

T_c: gas temperature [K]

(2) The law of mass conservation

$$\frac{dm}{dt} = 0$$

m: total mass of the mixture [kg]

t: time [s]

(3) No heat transfer, energy is conserved (first law of thermodynamics)

$$dq = du + P.dv$$

q: specific heat [J/kg]

u: internal energy [J/kg]

P: mixture pressure [Pa]

(4) Adiabatic change

$$dq = du + P.dv = 0$$

q: specific heat [J/kg]

u: internal energy [J/kg]

P: mixture pressure [Pa]

3.1.3 The process of the SI engine model calculation

Numerical calculations of elementary reaction in this study are performed in a range between just after intake valve close and just before exhaust valve open. After inputting the initial pressure, temperature and volume, the pressure increased gradually, and the volume was altered by the piston compression right after intake valve closed. The amount of volume change was once again converted to pressure which was equilibrated. The equilibrated pressure was substituted by the next pressure value.

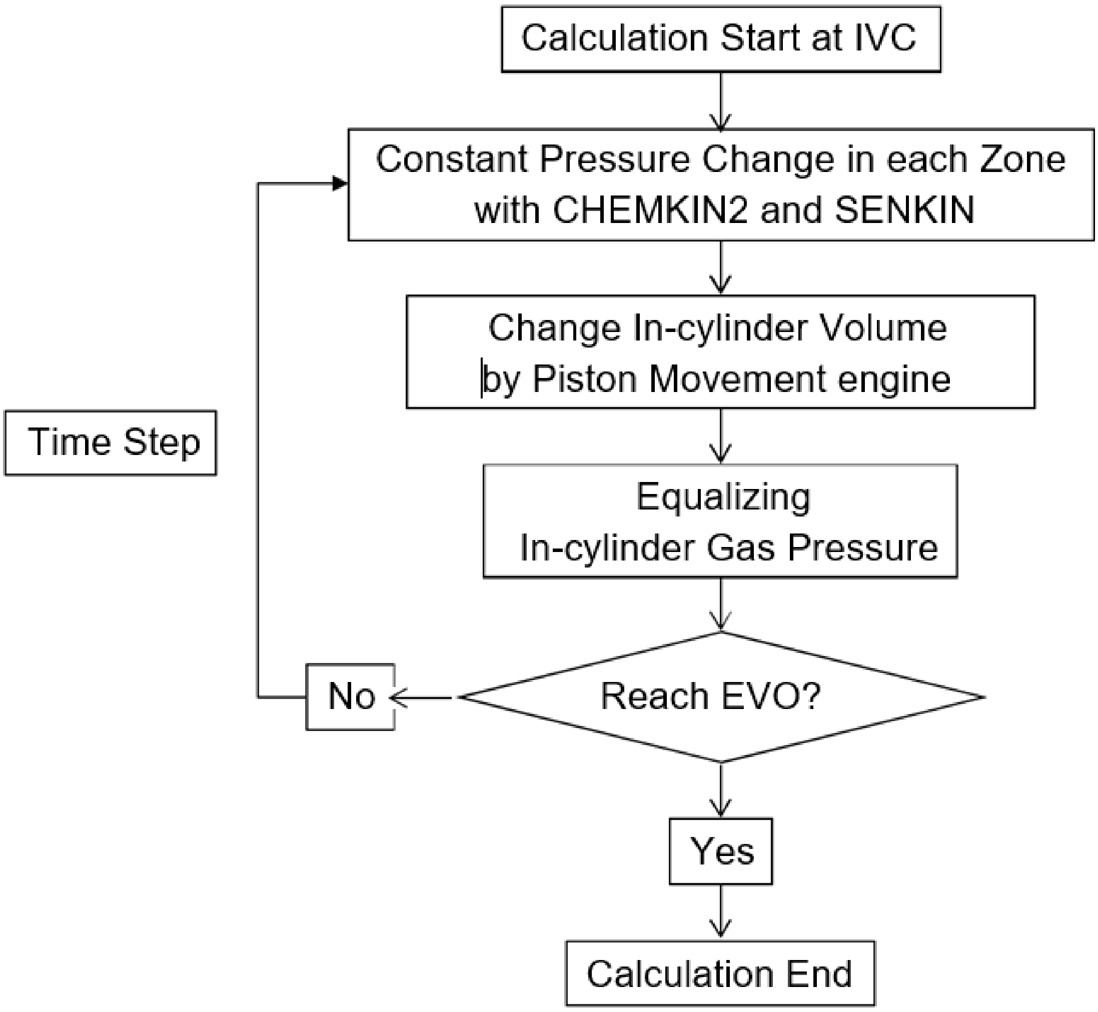


Figure 3-2. Algorithm flow diagram for SI engine

3.1.4 Engine specifications

In order to not consider the residual gases, it is assumed that cylinder is closed and calculation is initiated from just after intake valve close (IVO) at -90 aTDC to just before exhaust valve open (EVO) at 120 aTDC where compression/expansion stroke was allowed in the model (Figure). The engine specifications in calculations are listed in Table.

Table 3-1. Engine specifications

Process	Only 1 Compression and Expansion
Bore x Stroke (mm)	123x155
Displacement (cc)	1842
Compression ratio	11.5
Intake Valve Close, IVC	ATDC -90^0
Exhaust Valve Close, EVO	ATDC 120^0

3.2 Determination of low temperature reaction (LTR), high temperature reaction (HTR), CA50 and ignition delay

In Figure, starting and ending time of both low temperature reaction (LTR) and high temperature reaction (HTR) and 50% burn point of total heat release were shown. It might be increasing accordingly the heat release rate [J/ms] if input calorie is increased. In order to eliminate such effect, the value as (dQ/dt)/Qin [(J/ms)/J] where heat release rate (dQ/dt [J/ms]) is divided by input calorie (dQ/dt [J/ms]), was used. The starting time and ending time of both LTR and HTR were fixed basically as 0.05 (J/ms)/J. 50% burned point of total heat release was defined as CA50.

3.3 Determination of knocking and misfire

To prescribe the SI combustion without occurrence of knocking in the operational region, the simplest basic criterion is to set the limits of the maximum pressure-rise rate. Although the limits of the maximum pressure-rise rate are a useful measure to set up, but not always appropriate standards []. For this study, the ringing correlation developed by Eng [] is used to quantify the knock level:

$$Ringing\ Intensity\ \approx \frac{1}{2\gamma}\cdot\frac{(0.05\cdot\frac{dP}{dt,max})^2}{P_{max}}\cdot\sqrt{\gamma R T_{max}} \qquad 3\text{-}1$$

$dP/dt,_{max}$: maximum pressure-rise rate

P_{max}: maximum pressure

T_{max}: maximum temperature

γ: Cp/Cv

R: gas constant of air

Eq.(3-1) shows that the ringing intensity is proportional to the square of the absolute maximum pressure-rise rate (in kPa/ms). (P_{max} should be given as kPa, whereby Eq.(3-1) gives the ringing intensity in kW/m2. The speed of sound comes in through the square-root expression, and should be in m/s.) The use of a time-based pressure-rise for computing ringing/knock is justified by the fact that the acoustic timescales are independent of engine speed. 5MW/m2 was chosen as the limit for allowable ringing.

3.4 Combustion analysis

In-cylinder data such as pressure and HRR provide important insights about combustion behavior in the cylinder. This data can be used to obtain information about the combustion process such as the HRR and burn durations. The method of analysis is adapted from Gatowski et al., 1984 []. In a diesel engine, the combustion chamber is treated as closed system where only flow of fuel is considered and the crevice flow is neglected. The overall heat release using the energy balance equation is derived as:

$$\frac{dQ}{d\theta} = \left(\frac{C_p P}{R}\right)\frac{dV}{d\theta} + \left(\frac{C_V P}{R}\right)\frac{dP}{d\theta} - \frac{dQ_w}{d\theta}$$

where $dQ/d\theta$ is the heat transfer rate across the system with respect to crank angle degree, C_p and C_v are the gas specific heats at constant pressure and volume, R is the specific ideal gas constant, $dV/d\theta$ is the displaced volume across the system and $dP/d\theta$ is the differential pressure as piston travels.

The indicated mean effective pressure (IMEP) is used to quantify indicated work per piston displacement (work density), while the combustion stability is determined by the coefficient of variance (COV). The COV represents the standard deviation of the data as a percentage of its mean value. The COV of IMEP is used to indicate the cyclic variability in indicated work per cycle. Crank angle locations of peak pressure and 10, 50, and 90% mass fraction burned are analyzed to indicate combustion speed. Other parameters involved are thermal efficiency (η_t) and BSFC which represent the engine's performance.

3.5 Fuel properties

Methane is lighter than air, having a specific gravity of 0.554. It is only slightly soluble in water. It burns readily in air, forming carbon dioxide and water vapor; the flame is pale, slightly luminous, and very hot. The boiling point of methane is −162 °C (−259.6 °F) and the melting point is −182.5 °C (−296.5 °F). Methane in general is very stable, but mixtures of methane and air, with the methane content between 5 and 14 percent by volume, are explosive. Explosions of such mixtures have been frequent in coal mines and collieries and have been the cause of many mine disasters.

Propane is the third alkane in the saturated hydrocarbon (paraffin) series starting with methane. Propane is a gas at atmospheric conditions and can be compressed into a liquid for transportation. As a commodity in this state, it is often termed liquefied petroleum gas (LPG) and contains small amounts of propylene, butane, and butylene. Propane is a flammable hydrocarbon gas that is liquefied through pressurization and commonly used for fuel in heating, cooking, hot water and vehicles. Propane can also be used for refrigerants, aerosol propellants and petrochemical feedstock. Propane gas can be compressed into liquid at relatively low pressures. Propane is generally stored, as a liquid, in steel vessels ranging from small BBQ gas bottles to larger gas cylinders and LPG storage tanks.

Table 3-2. Fuel properties

Name of product	Methane	Propane
Formula	CH_4	C_3H_8
Boiling point (^oC)	-161.5	-42
Liquid density (g/cm3(@20^oC)	-	0.49
Gas specific gravity	0.55	1.52
Saturated steam pressure (atm, 25^oC)	-	9.3
Ignition temperature (^oC)	632	504
Explosion limit (%)	5~15	2.1~9.4
Cetane rating	0	5
Low heat value (kcal/kg)	12,000	11,100

3.6 Validation of simulation model

The confidence of the simulated model was evaluated based on the comparison between the experimental results and simulated results. The black curves describe the experimental results while the red curves describe simulated results. Before using the simulated model to estimate the effect of ignition timing, this model was calibrated and validated. The values of engine's specification and experimental data (bore, stroke, connecting length, intake and exhaust tube length and diameter, experimental conditions) were used as input data for the simulated model. In figure 3-3, the simulated values were very close to the experimental values.

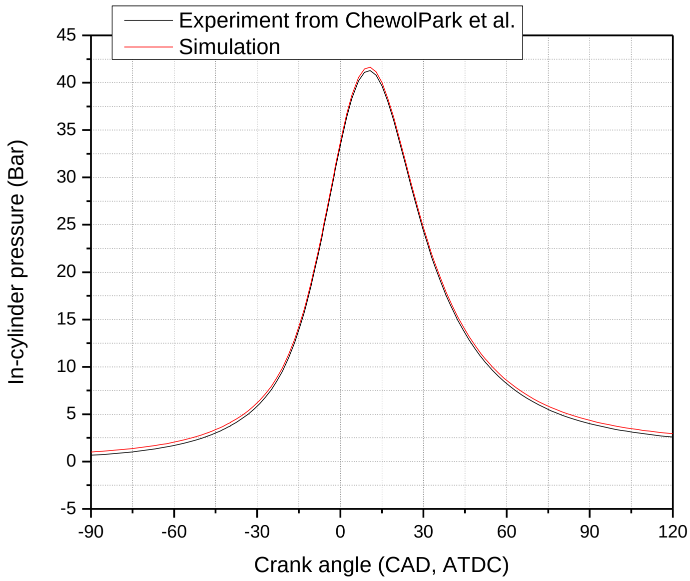


Figure 3-3. Validate of simulation model

4. RESULT AND DISCUSSION

4.1 Effect of ignition timing and mixing ratio on engine performance of engine fueled with natural gas

4.1.1 Effect of ignition timing and mixing ratio on IMEP

Figure 4-1 shows the effect of spark ignition timing on the peak temperature for each percent of propane when mixing methane with propane. It is shown that as the percentage of propane in the mixture increases and advanced the ignition timing, the peak temperature increases gradually. Due to propane has more stored energy per unit volume than methane and thus releases more heat when the same amount of propane is burned. On the other hand, when increasing the spark advanced ignition timing, it gives the air fuel mixture have more combustion duration, which helps the combustion take place completely leading to increase in peak temperature.

Figure 4-2 shows the effect of spark ignition timing on the peak pressure for each percent of propane when mixing methane with propane. It is indicated that as the percentage of propane in the mixture increases and advanced the ignition timing, the peak pressure increases gradually. The maximum peak pressure is 42.4 bar when the percentage of propane is 20% and at 48 degree of spark ignition timing. This is because an increase in the percentage of propane in the mixture and an increase in spark advanced ignition timing leads to an increase in the peak temperature in the engine resulting in increased peak pressure.

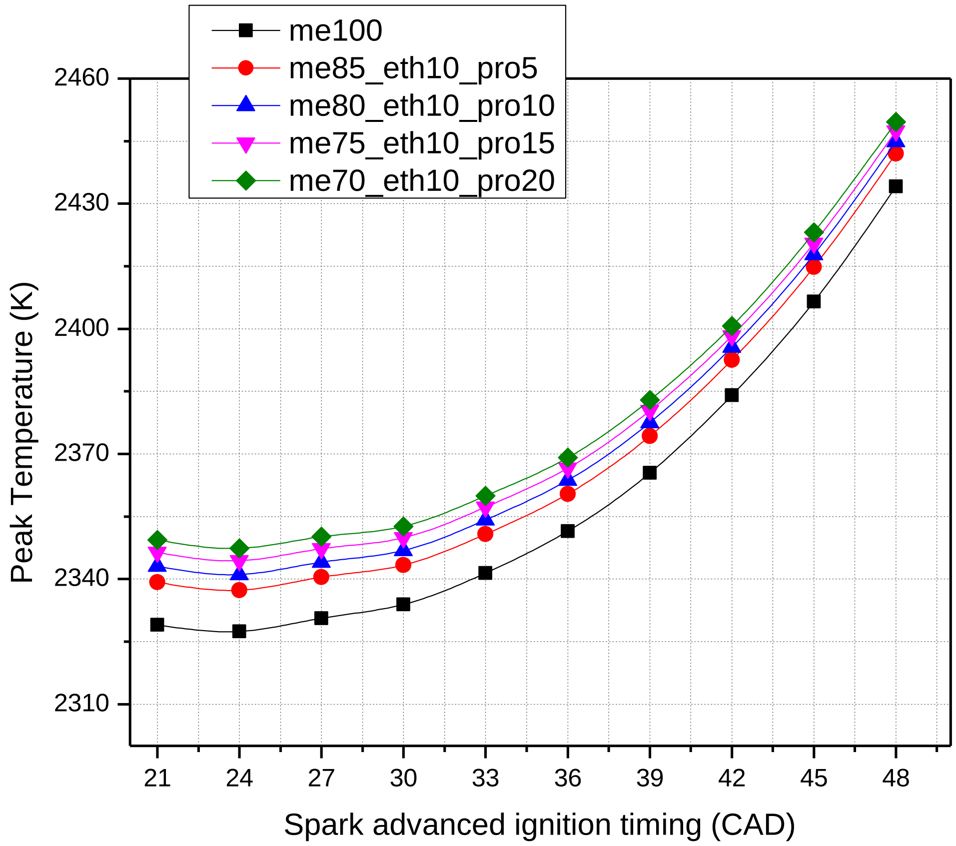


Figure 4-1. Effect of ignition timing and mixing ratio on peak temperature

Figure 4-3 shows the effect of spark ignition timing on the IMEP for each percent of propane when mixing methane with propane. The value of IMEP increases as the percentage of propane in the mixture increases. Because propane has more stored energy per unit volume than methane and thus releases more heat when the same amount of propane is burned, so an increase in the percentage of propane in the mixture results in an increase in the temperature and pressure in the cylinder thereby increasing the IMEP of the engine. In addition, the IMEP values tends to increase with increasing spark advanced ignition timing from 21 to 39 degrees but tends to decrease from 42 to 48 degrees

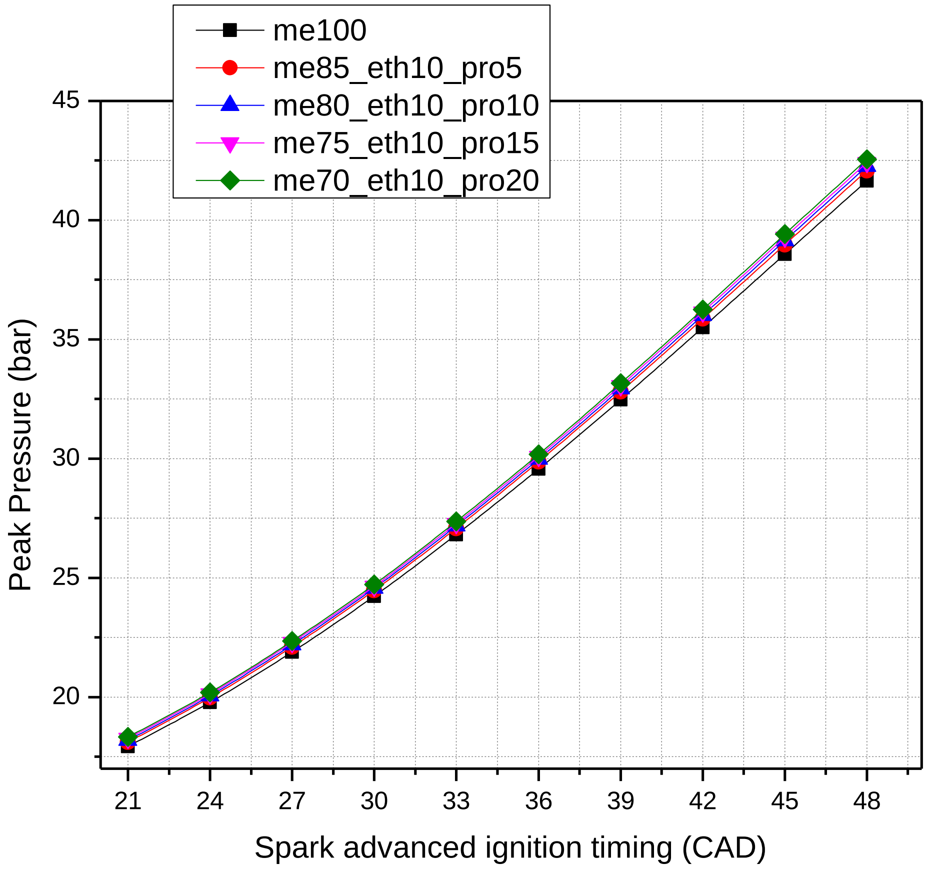


Figure 4-2. Effect of ignition timing and mixing ratio on peak pressure

The maximum IMEP of engine was 6.897 bar with 20 percent of propane and at 39 degree of spark advanced ignition timing. If combustion starts too early in the cycle, the work transfer from the piston to the gases at the end of the compression stroke is too large. If the combustion starts too late, the peak cylinder pressure is reduced and the expansion stroke work transfer from the gas to the piston decreases. The ignition delay may cause the peak pressure to reduce which results in IMEP decrease since the work done through the control volume is less.

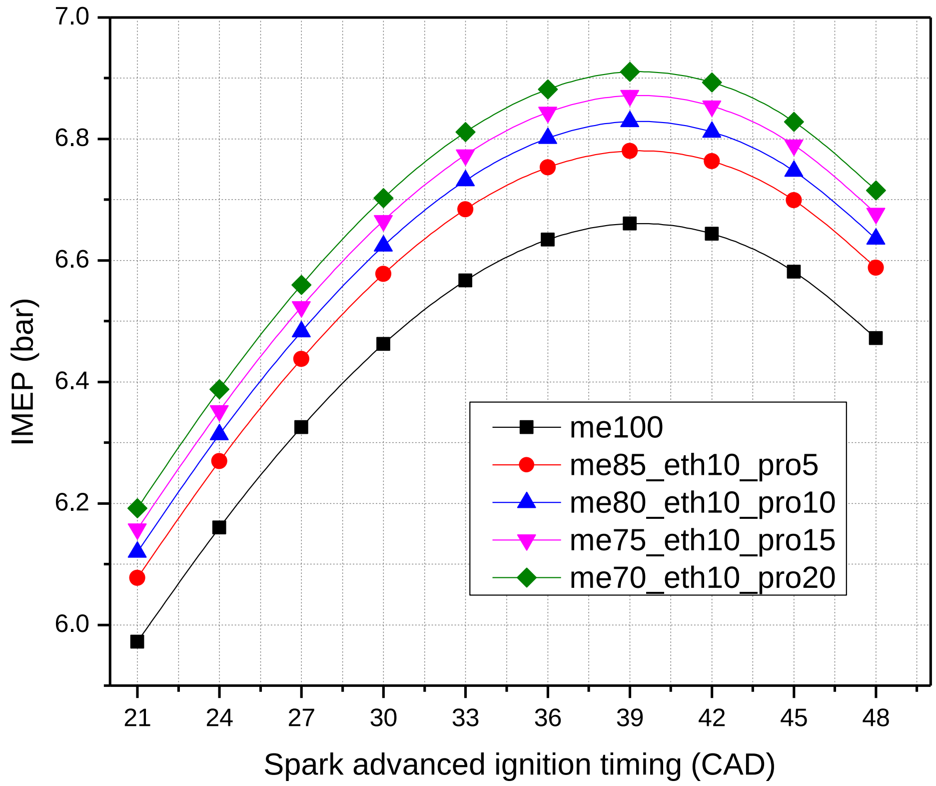


Figure 4-3. Effect of ignition timing and mixing ratio on IMEP

4.1.2 Effect of ignition timing and mixing ratio on brake torque

The effect of spark ignition timing on the brake torque for each percent of propane when mixing methane with propane was shown in figure 4-4. The value of brake torque increases as the percentage of propane in the mixture increases. In addition, the brake torque tends to increase with increasing spark advanced ignition timing from 21 to 39 degrees but tends to decrease from 42 to 48 degrees. The maximum brake torque of engine was 101.0898 Nm with 20 percent of propane and at 39 degree of spark advanced ignition timing. The minimum brake torque of engine was 87.526 Nm with pure methane and at 21 degree of spark advanced ignition timing.

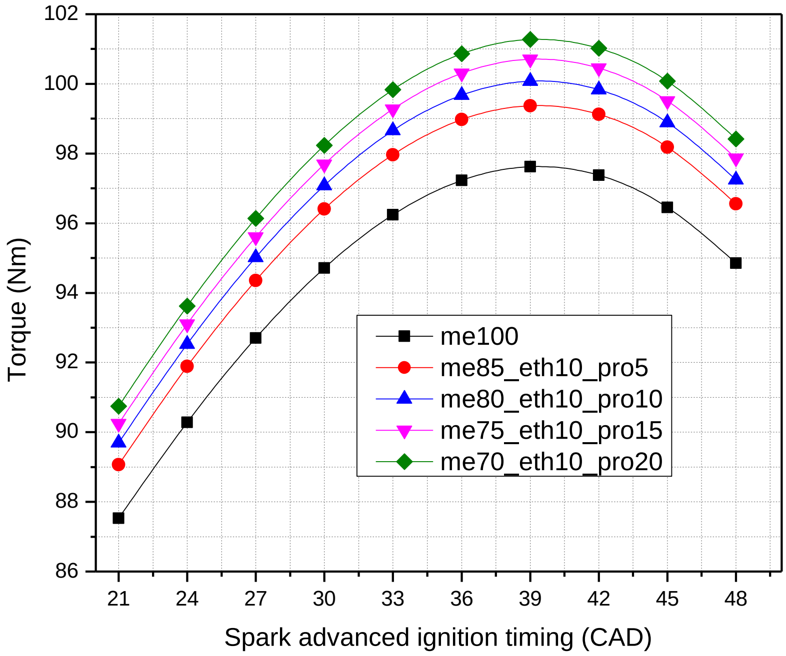


Figure 4-4. Effect of ignition timing and mixing ratio on brake torque

4.1.3 Effect of ignition timing and mixing ratio on power

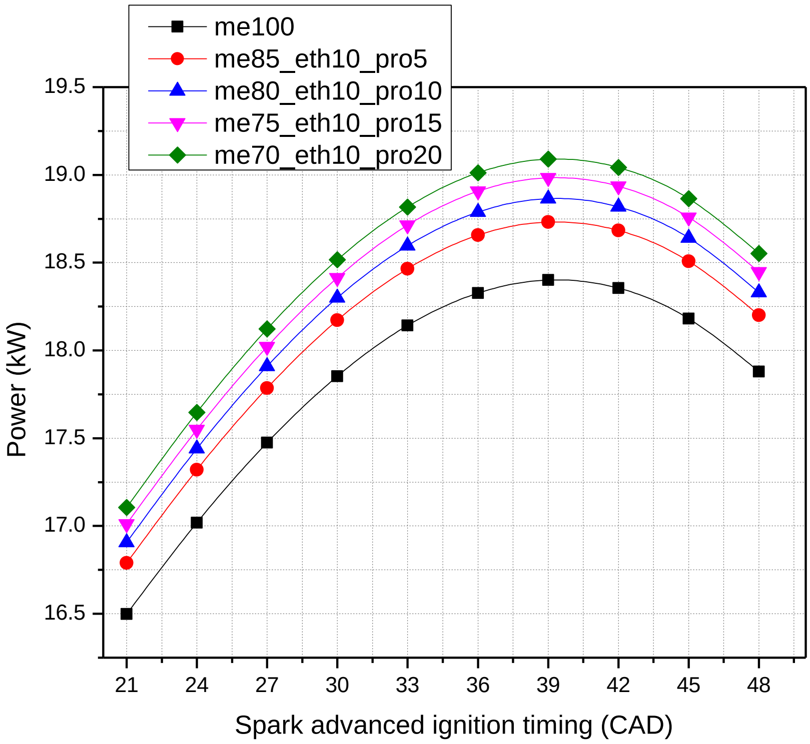


Figure 4-5. Effect of ignition timing and mixing ratio on power

The effect of spark ignition timing on the power for each percent of propane when mixing methane with propane was shown in figure 4-5. The value of power increases as the percentage of propane in the mixture increases. In addition, the power tends to increase with increasing spark advanced ignition timing from 21 to 39 degrees but tends to decrease from 42 to 48 degrees. The maximum power of engine was 19.055 kW with 20 percent of propane and at 39 degree of spark advanced ignition timing. The minimum brake torque of engine was 16.498 kW with pure methane and at 21 degree of spark advanced ignition timing.

4.1.4 Effect of ignition timing and mixing ratio on ignition delay

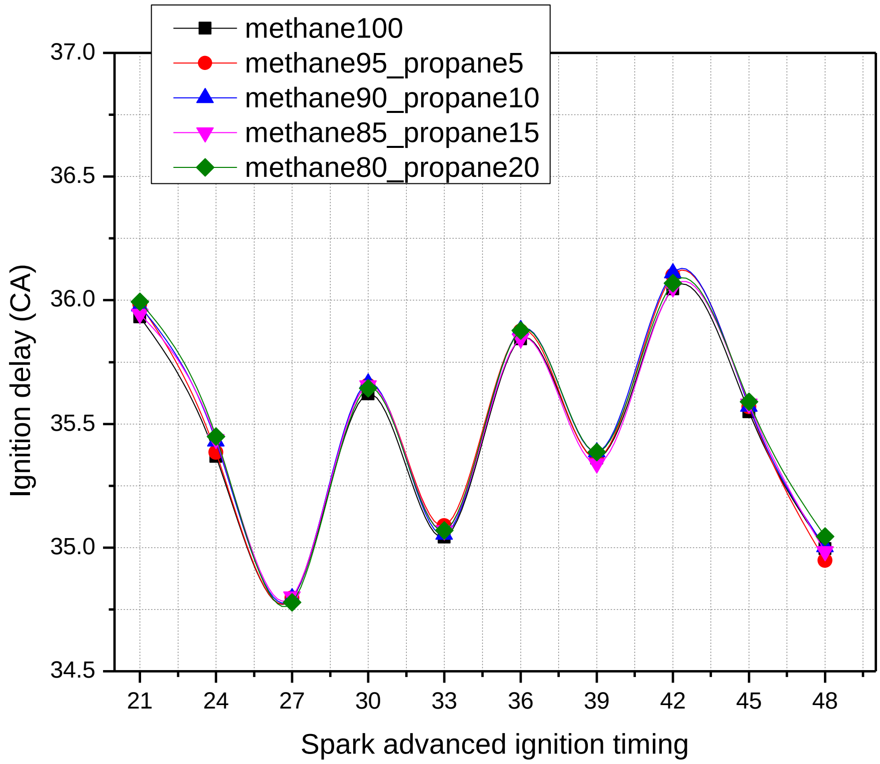


Figure 4-6. Effect of ignition timing and mixing ratio on ignition delay

Figure 4-6 describes the effect of spark ignition timing on the ignition delay for each percent of propane when mixing methane with propane. It was clearly seen that the ignition delay of the blends seemed to be the same, which proved that propane had no great effect on ignition delay of the engine. Furthermore, as the increase of spark advanced ignition timing, the ignition delay fluctuates and the range about 1 degree of the crankshaft. The maximum ignition delay was 10 percent of propane in the blends and at 42 degree of spark advanced ignition timing.

The minimum ignition delay with pure methane and at 27 degree of spark advanced ignition timing.

4.1.5 Effect of ignition timing and mixing ratio on thermal efficiency

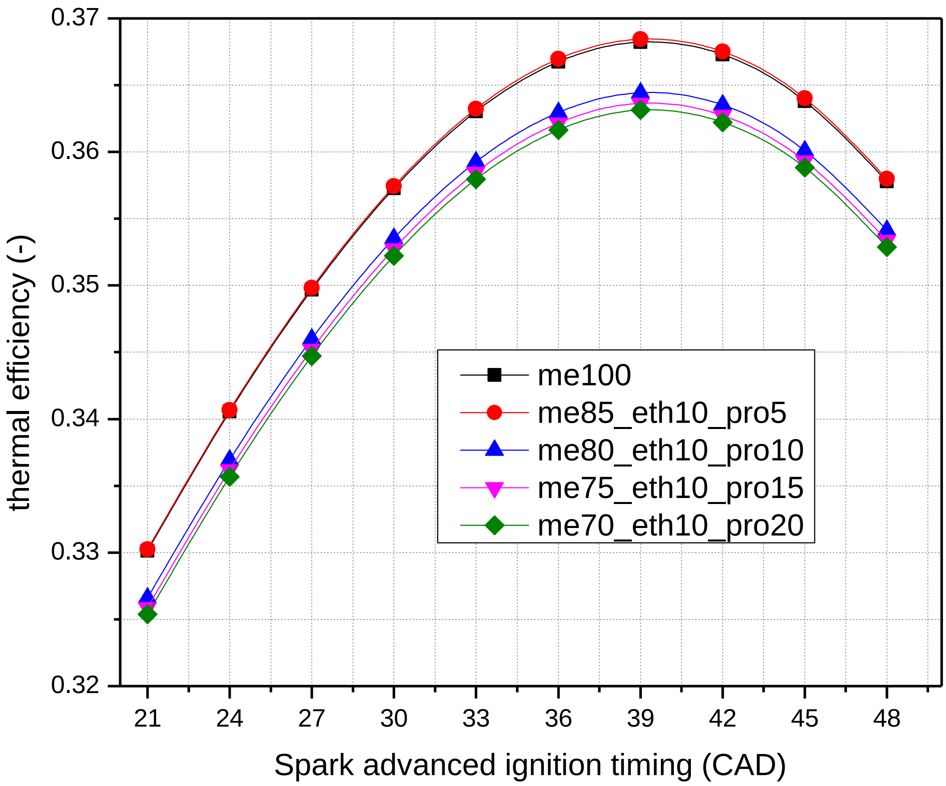


Figure 4-7. Effect of ignition timing and mixing ratio on thermal efficiency

Figure 4-7 describes the effect of spark ignition timing on the thermal efficiency for each percent of propane when mixing methane with propane. It is shown that thermal efficiency decreases as the percentage of propane in the mixture increases. Because propane combustion increases the temperature of engine but also requires more energy to break the bonds in the chemical composition, leading to decrease in the thermal efficiency. In addition, the thermal efficiency tends to increase with increasing spark advanced ignition timing from 21 to 39 degrees but tends to decrease from 42 to 48 degrees. The maximum thermal efficiency was 36.84% with 5 percent of propane and at 39 degree of spark advanced ignition timing. The minimum thermal efficiency was 32.53% with 20 percent of propane and at 21 degree of spark advanced ignition timing.

4.1.6 Effect of ignition timing and mixing ratio on brake specific fuel consumption

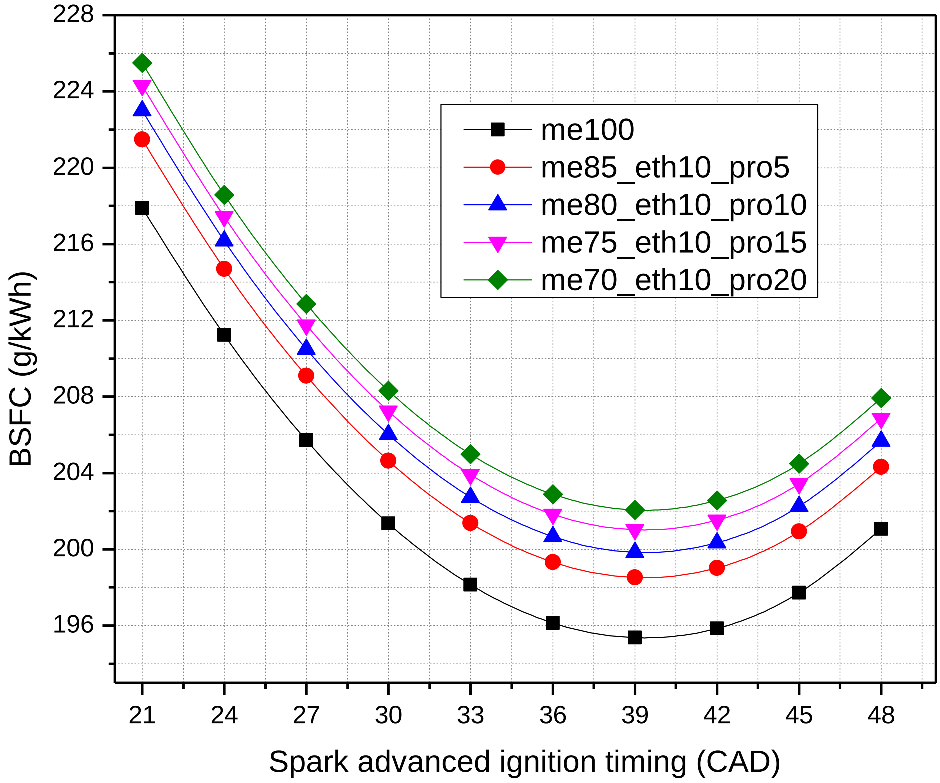


Figure 4-8. Effect of ignition timing and mixing ratio on brake fuel consumption

The effect of spark ignition timing on the thermal efficiency for each percent of propane when mixing methane with propane was shown in figure 4-8. The value of BSFC increases as the percentage of propane in the mixture increases. It should be noted that when IMEP increase, BSFC follows inversely. Moreover, propane combustion increases the temperature of engine but also requires more energy to break the bonds in the chemical composition, leading to need more fuel to completely combustion take place. In addition, the BSFC tends to decrease with increasing spark advanced ignition timing from 21 to 39 degrees but tends to increase from 42 to 48 degrees.

4.2 Effect of ignition timing and mixing ratio on emissions characteristics of engine fueled with natural gas

4.2.1 Effect of ignition timing and mixing ratio on CO

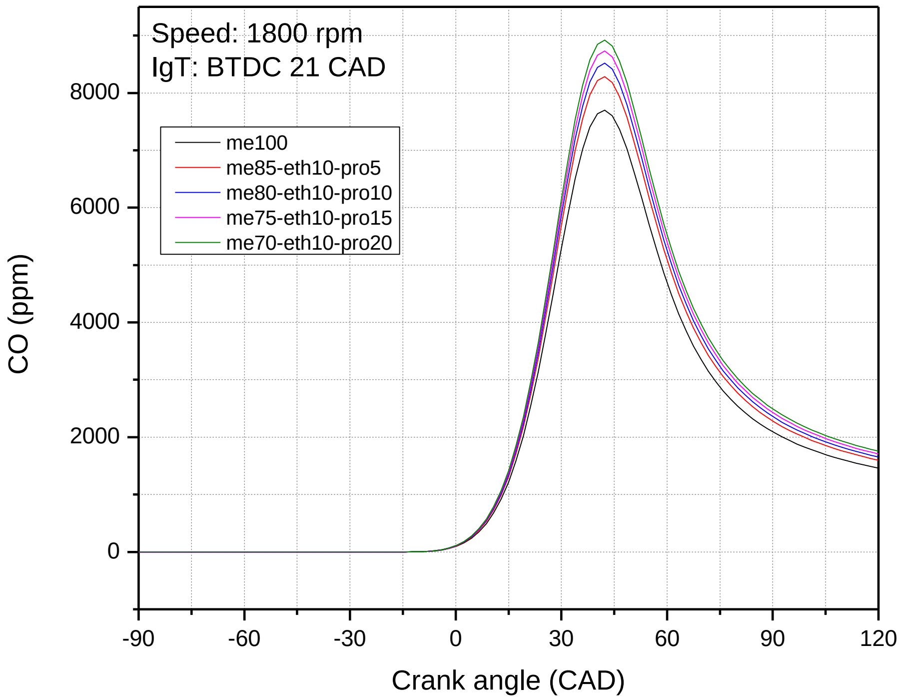


Figure 4-9. CO emissions with ignition timing 21 deg

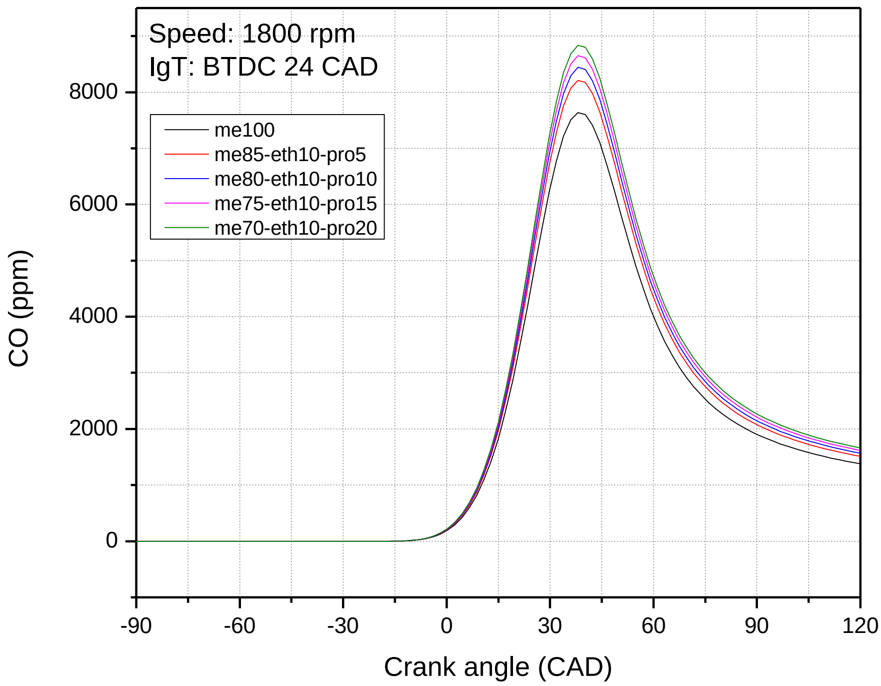


Figure 4-10. CO emissions with ignition timing 24 deg

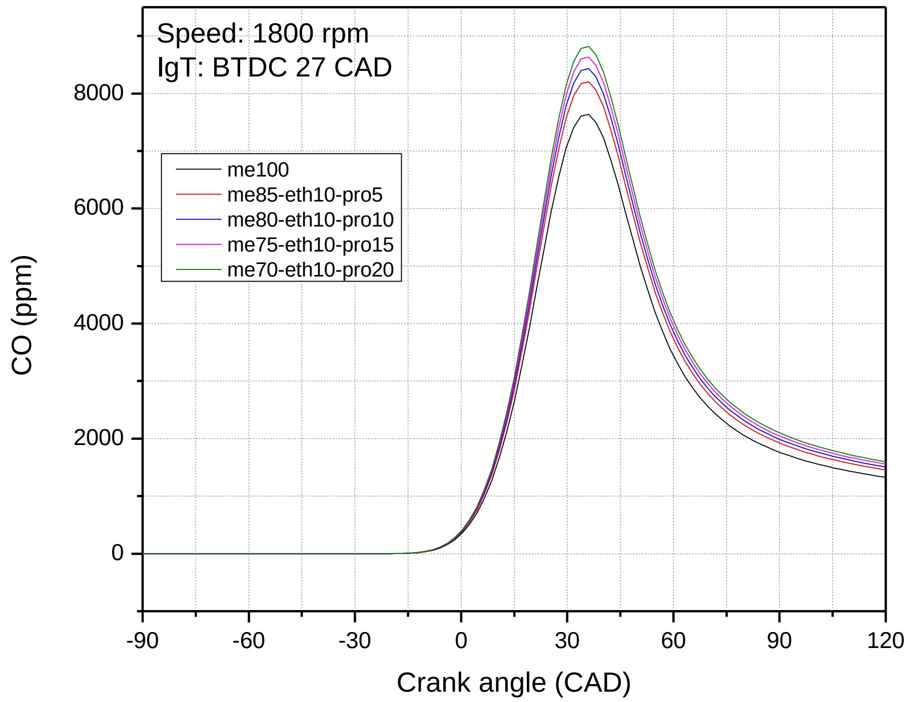


Figure 4-11. CO emissions with ignition timing 27 deg

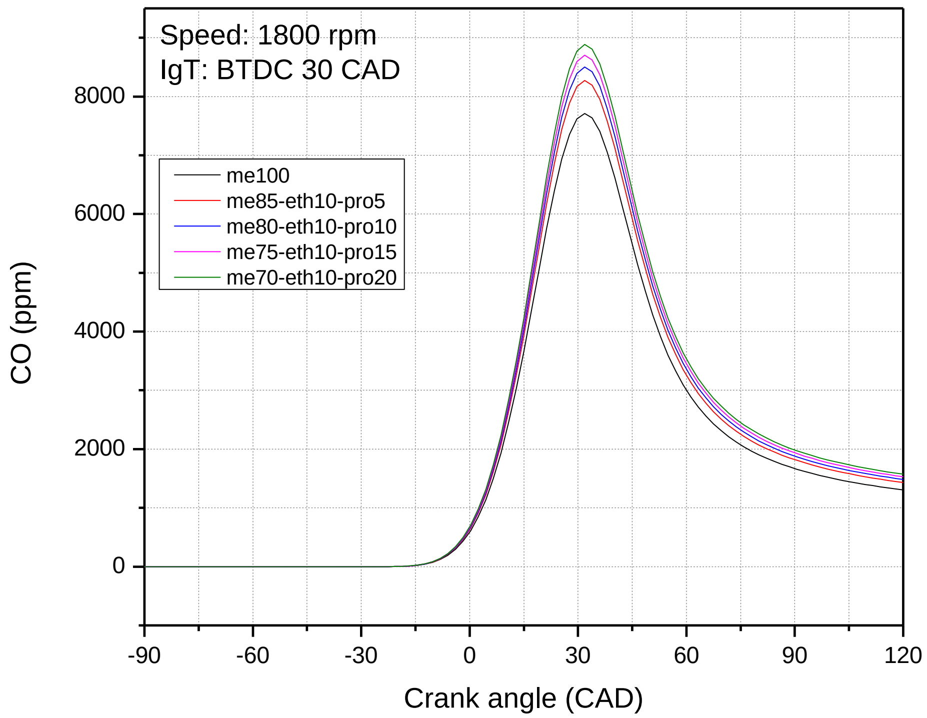


Figure 4-12. CO emissions with ignition timing 30 deg

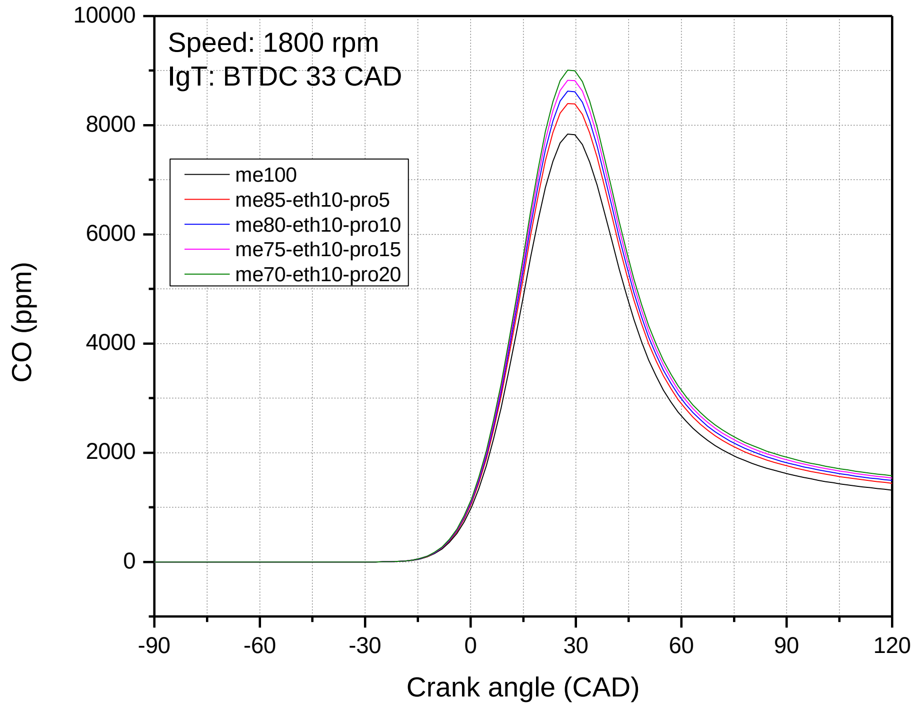


Figure 4-13. CO emissions with ignition timing 33 deg

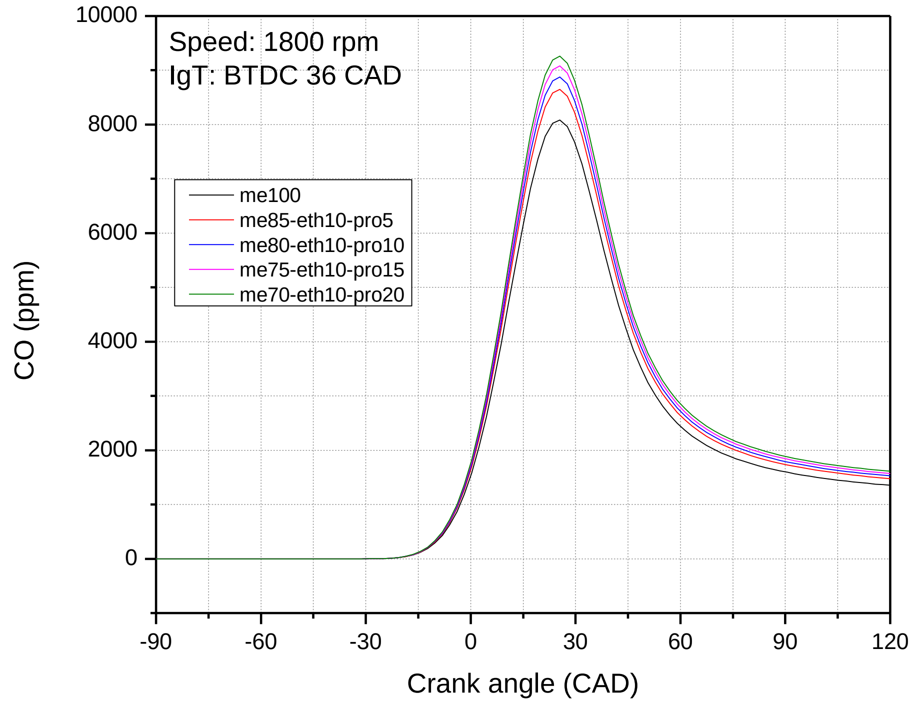


Figure 4-14. CO emissions with ignition timing 36 deg

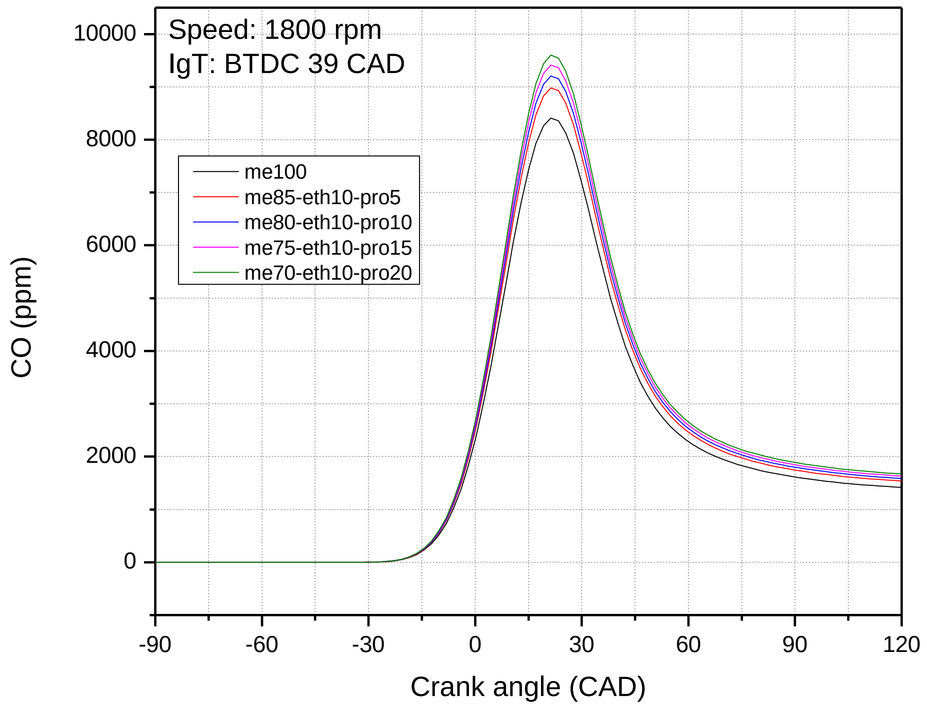


Figure 4-15. CO emissions with ignition timing 39 deg

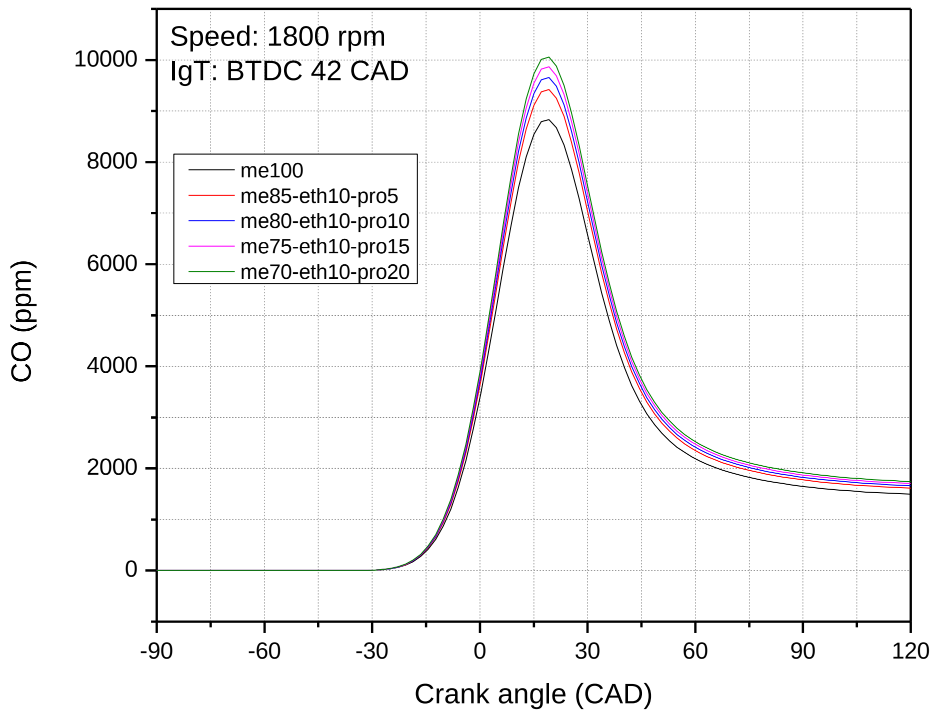


Figure 4-16. CO emissions with ignition timing 42 deg

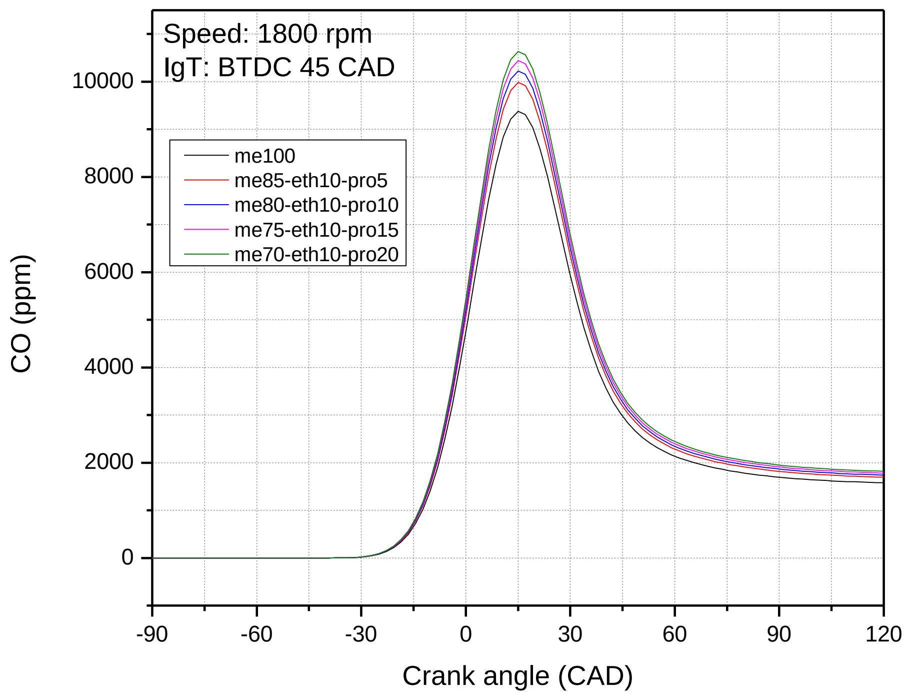


Figure 4-17. CO emissions with ignition timing 45 deg

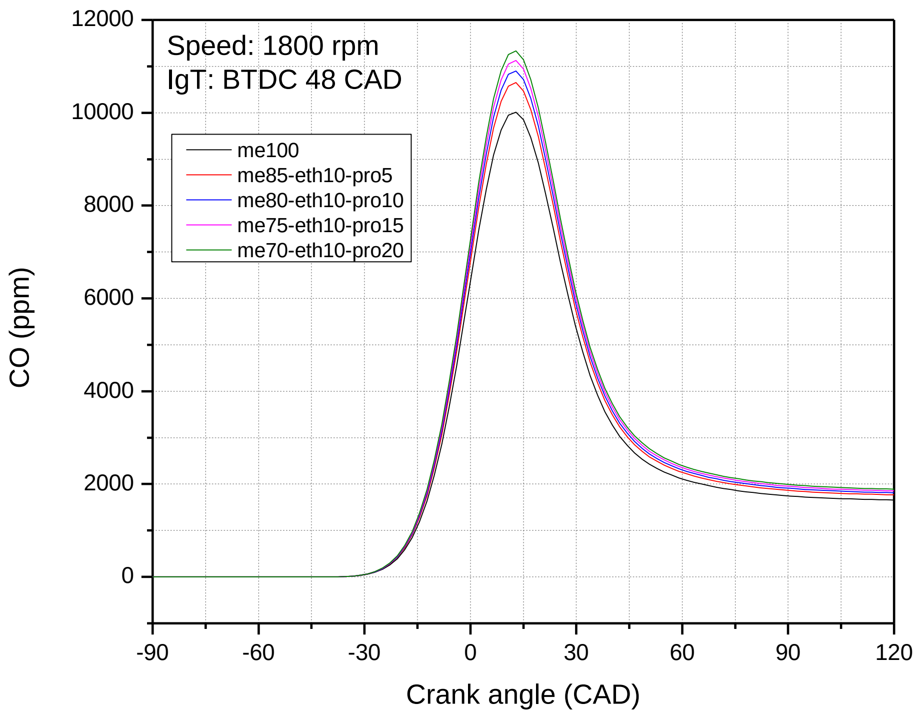


Figure 4-18. CO emissions with ignition timing 48 deg

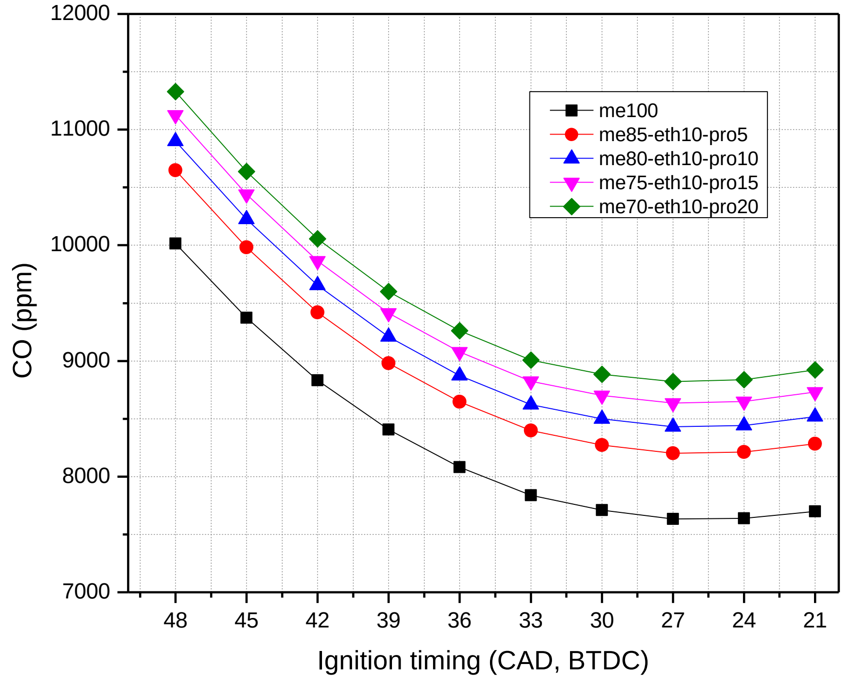


Figure 4-19. Effect of ignition timing and mixing ratio on CO emissions

The effect of spark ignition timing on the CO emissions for each percent of propane when mixing methane with propane was shown in figure 4-19. The value of CO emissions increases as the percentage of propane in the mixture increases. It can be explained that increasing the percentage of propane in the mixture will lead to an increase in the BSFC, thereby decreasing the fuel from completely burning with oxygen leading to an increase in CO emissions in the exhaust gas. Furthermore, When the advanced ignition timing decreases, the CO emissions will decrease because the fuel does not have enough time to react with the oxygen.

4.2.2 Effect of ignition timing and mixing ratio on NOx

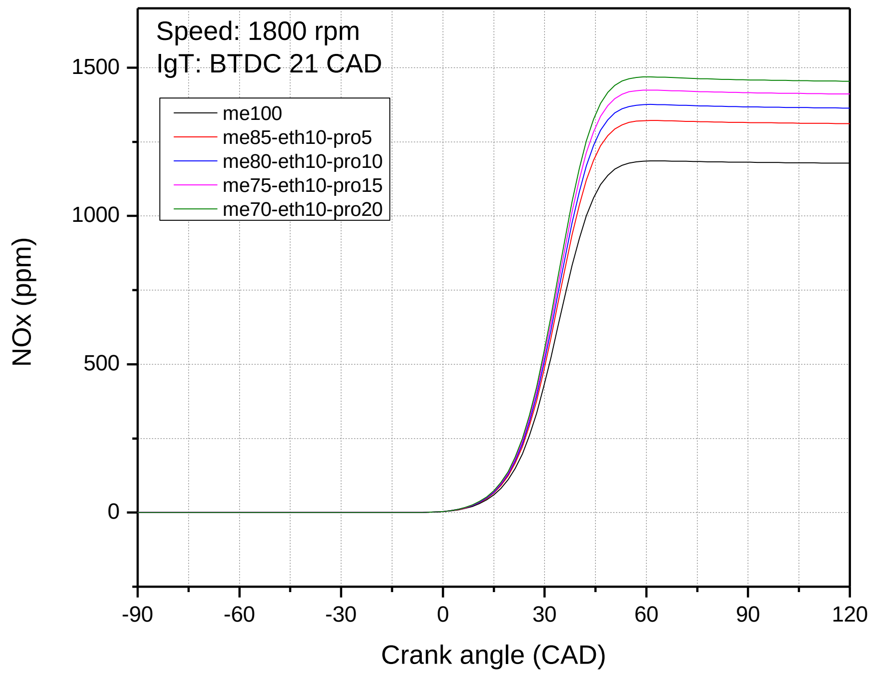


Figure 4-20. NOx emissions with ignition timing 21 deg

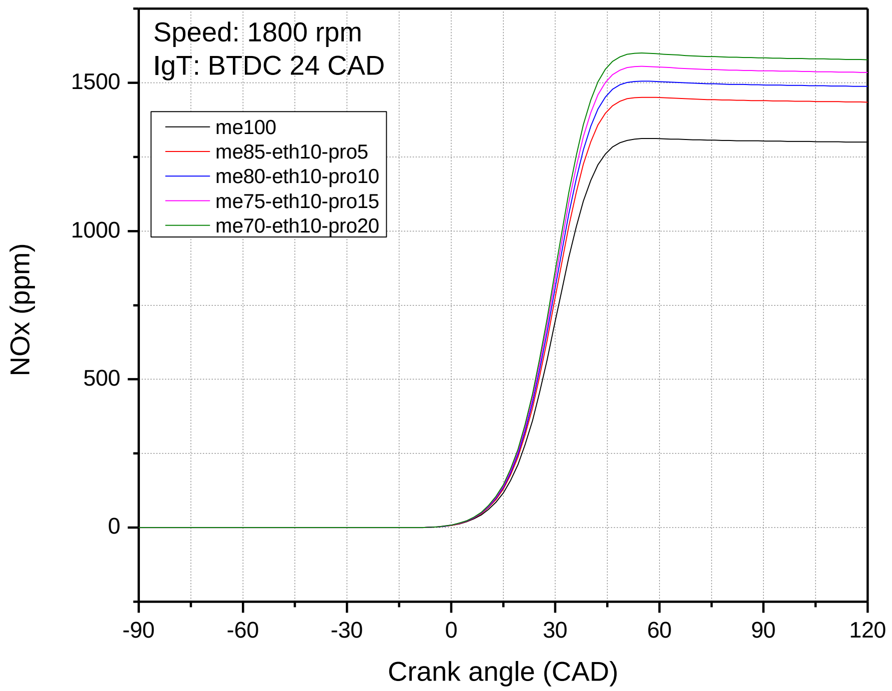


Figure 4-21. NOx emissions with ignition timing 24 deg

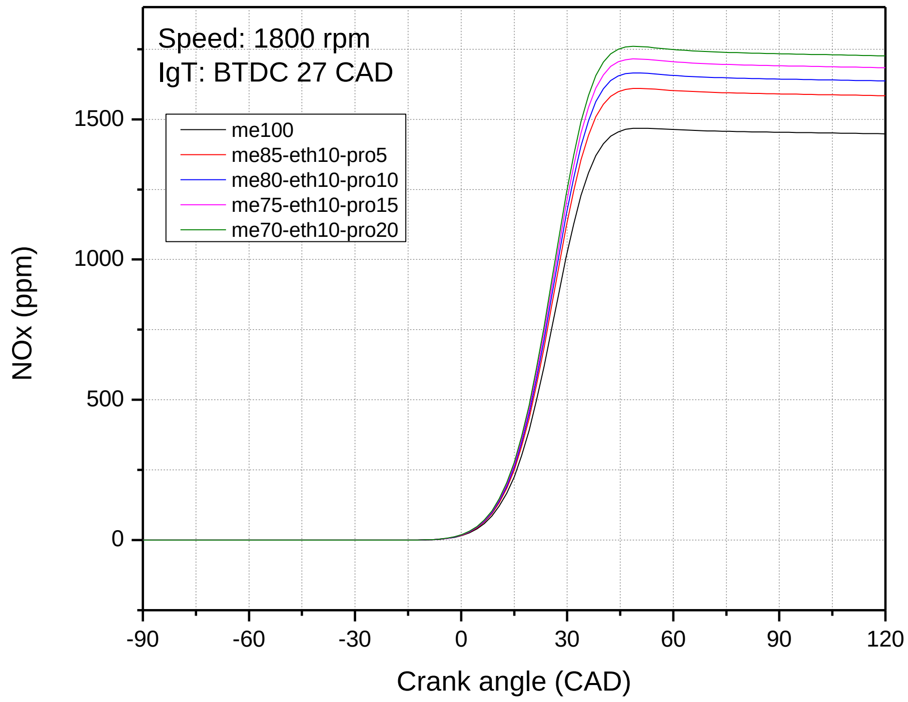


Figure 4-22. NOx emissions with ignition timing 27 deg

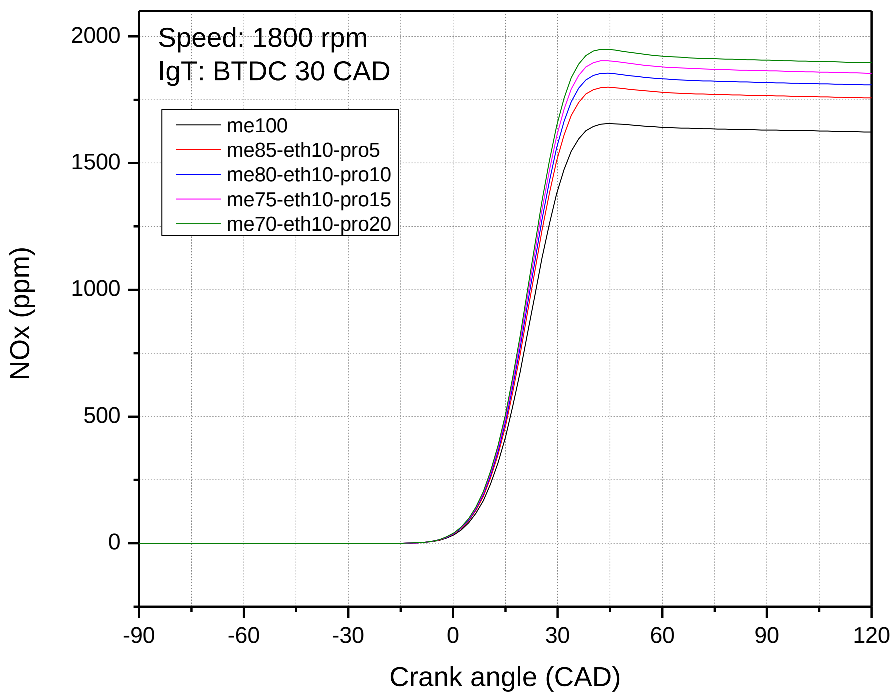


Figure 4-23. NOx emissions with ignition timing 30 deg

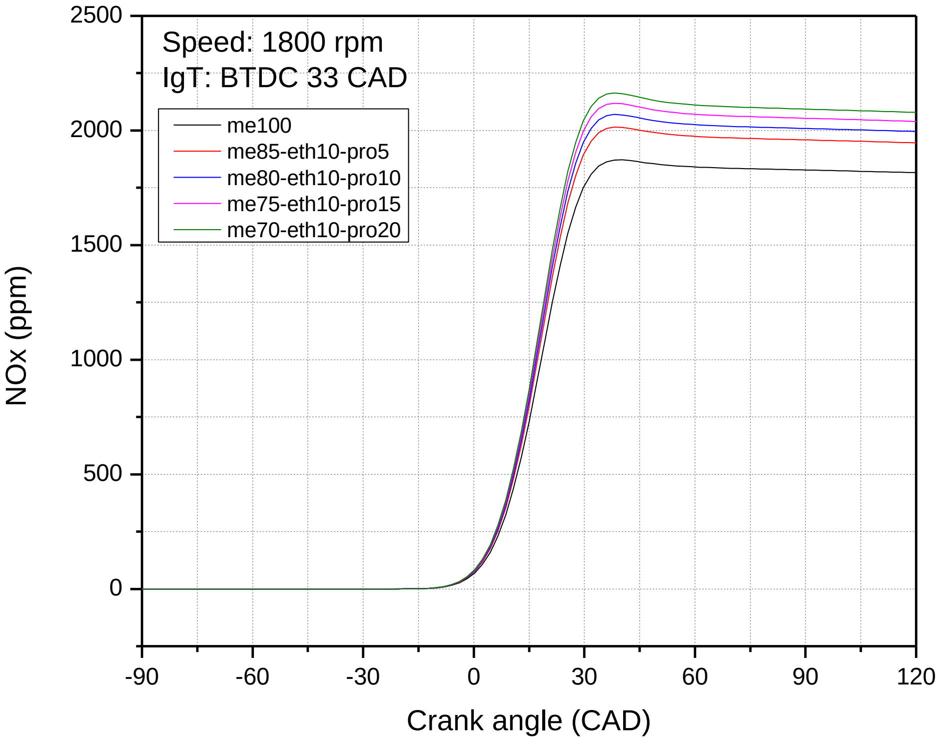


Figure 4-24. NOx emissions with ignition timing 33 deg

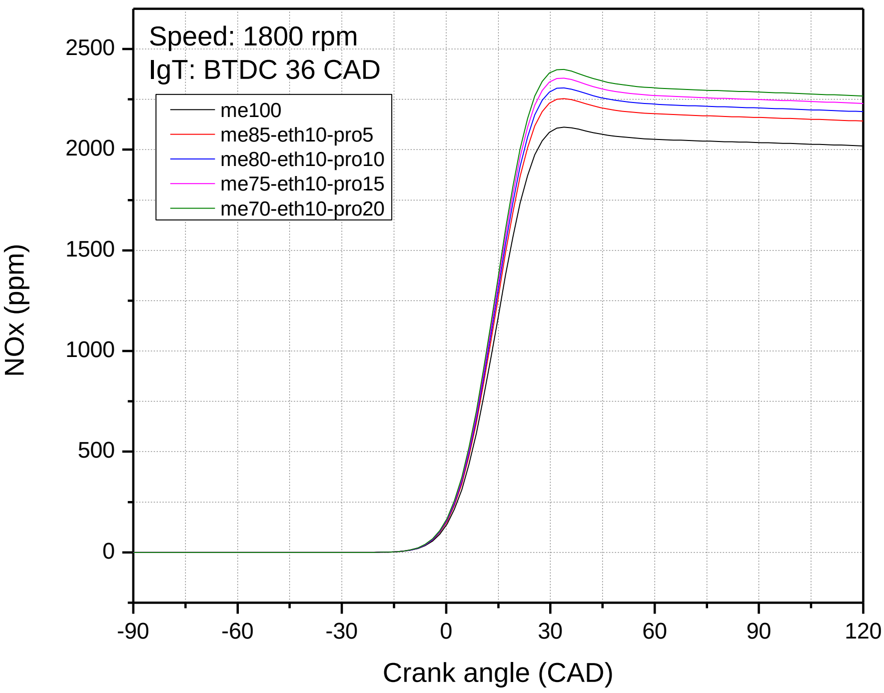


Figure 4-25. NOx emissions with ignition timing 36 deg

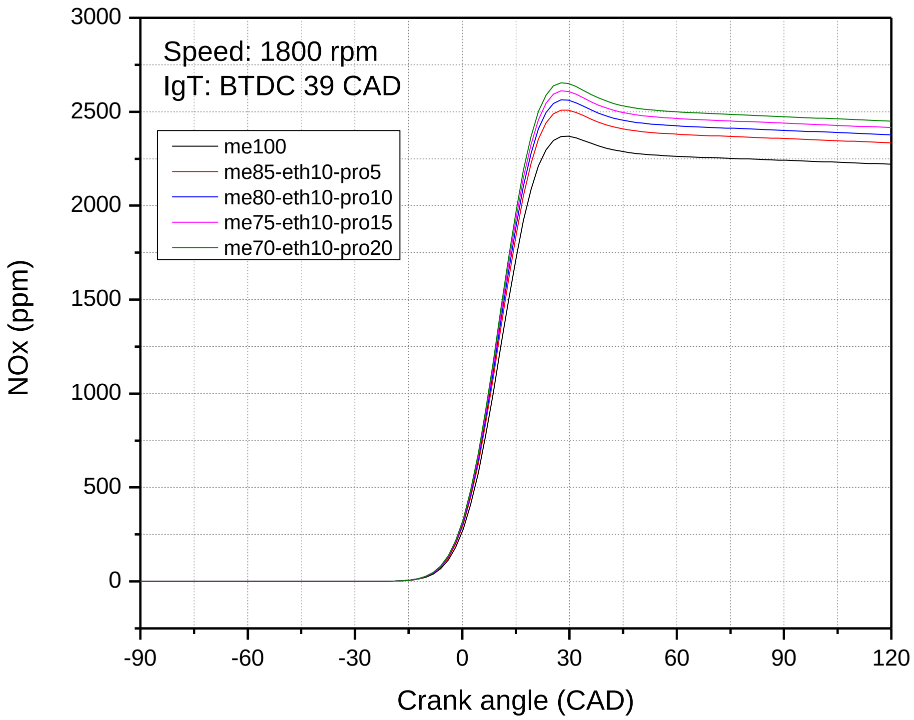


Figure 4-26. NOx emissions with ignition timing 39 deg

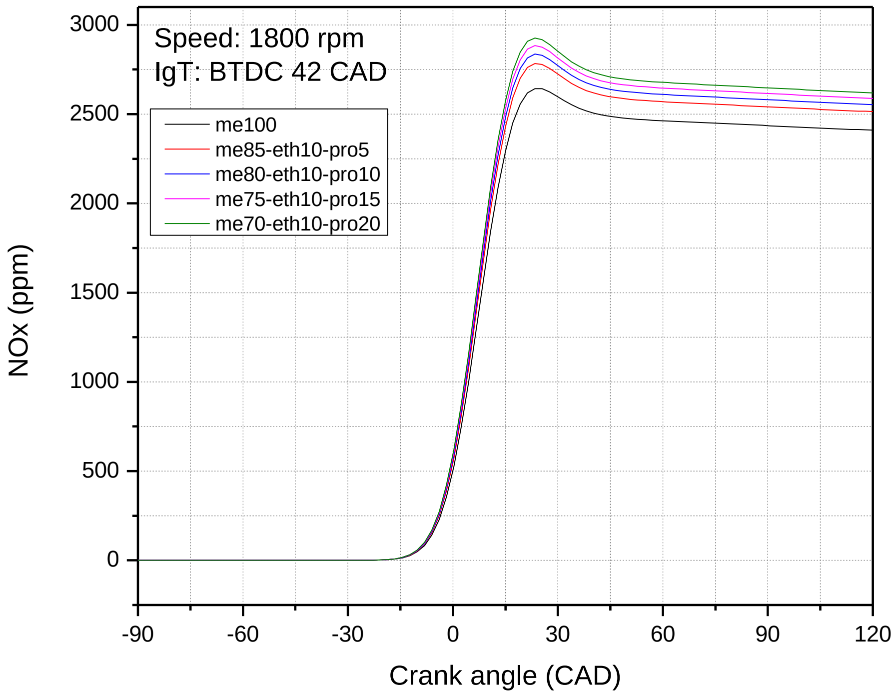


Figure 4-27. NOx emissions with ignition timing 42 deg

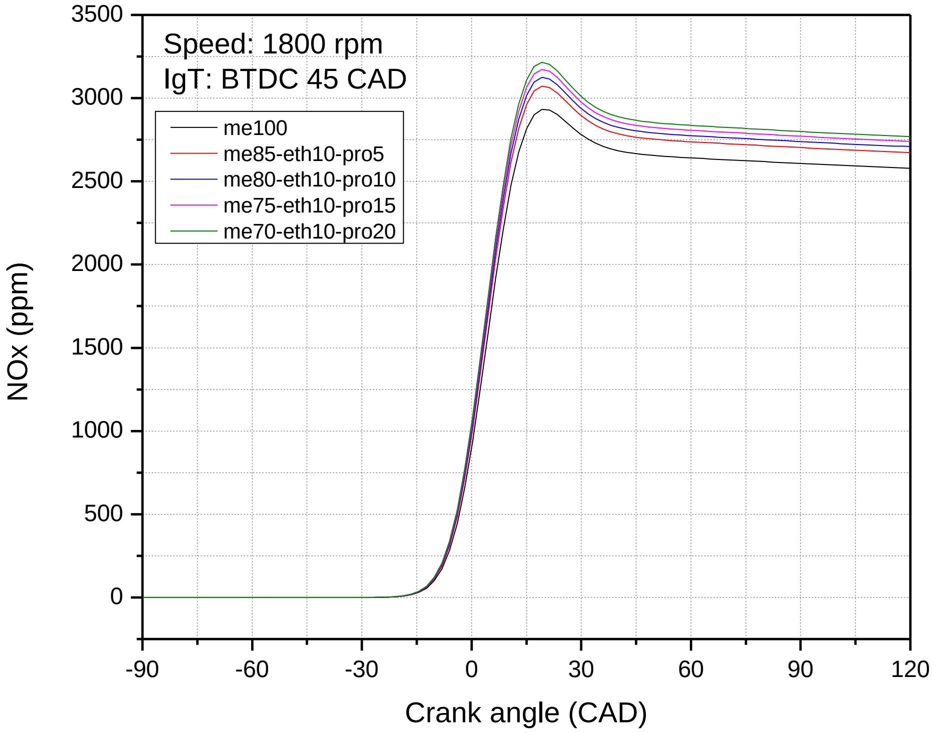


Figure 4-28. NOx emissions with ignition timing 45 deg

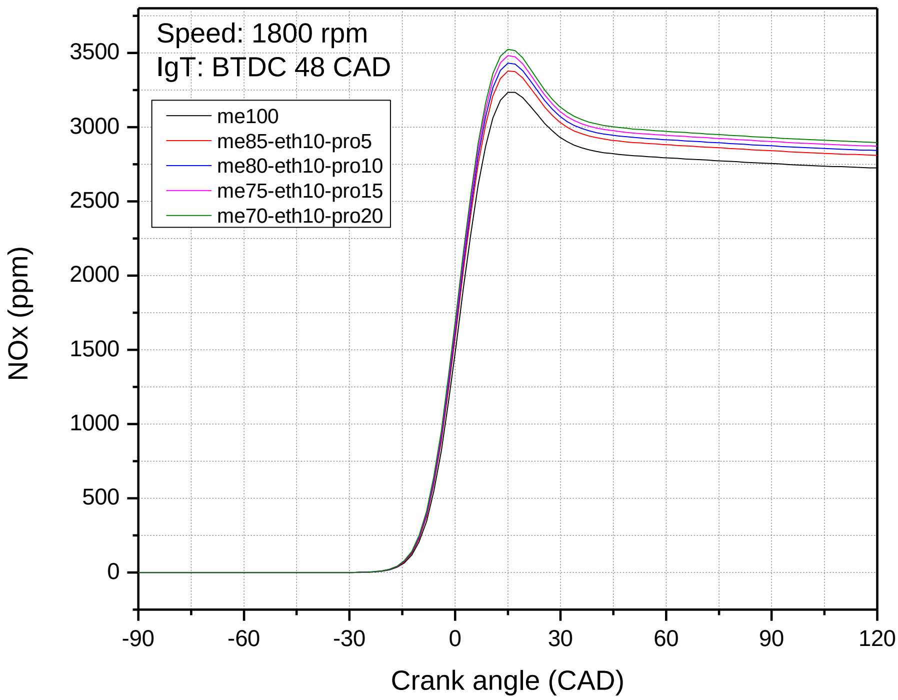
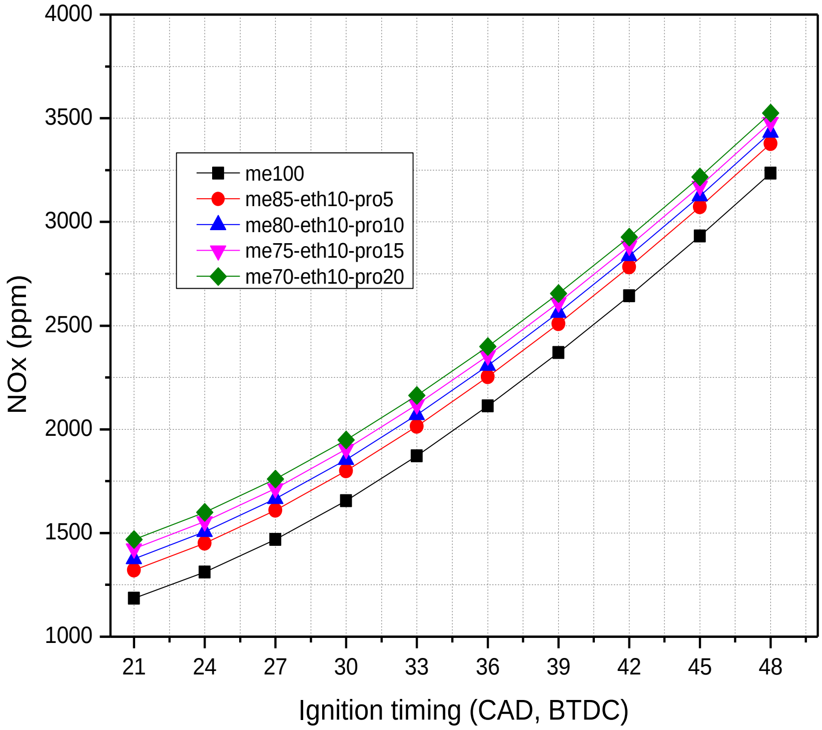


Figure 4-29. NOx emissions with ignition timing 48 deg



4-30. effect of ignition timing and mixing ratio on NOx emissions

The effect of spark ignition timing on the NOx emissions for each percent of propane when mixing methane with propane was shown in figure 4-30. The value of NOx emissions increases as the percentage of propane in the mixture increases. It can be explained that increasing the percentage of propane in the mixture will lead to an increase the temperature in the cylinder, thereby enhancing the reaction between nitrogen and oxygen and as a result the amount of NOx emissions in the combustion chamber increases. Furthermore, When the advanced ignition timing increases, the NOx emissions will increase because increasing advanced ignition timing will lead to an increase in temperature and pressure in the combustion chamber leading to an increase in NOx emissions

5. SUMMARY AND CONCLUSION

A detail investigation of SI engine fueled with natural gas has been performed. The mixing ratio and ignition timing has been significant effect on the performance and emissions of SI engine through simulation model. Following conclusions can be drawn from the overall study:

- Increasing the concentration of propane in the mixture results in an increase in the pressure and temperature in the cylinder, which in turn increases engine power and torque. However, increasing the concentration of propane also reduces thermal efficiency and increases brake specific fuel consumption
- In addition, the ignition delay is the same in all cases, so the mixing ratio of the mixture and the ignition timing do not have much effect on the ignition delay.
- CO and NOx emissions also increase steadily when increasing the concentration of propane in the mixture as well as when increasing the ignition timing of the engine.

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