

Sudan University of Science & Technology College of Graduate Studies College of Engineering



School of Mechanical Engineering

Effect of intake and exhaust valves timing on gasoline engine performance and emissions

تآثير توقيت صمامى الهواء والعادم على أداء وإنعاثات محرك البنزين

A Thesis Submitted in Partial Fulfillment of the Degree of M.Sc. in Mechanical Engineering (POWER)

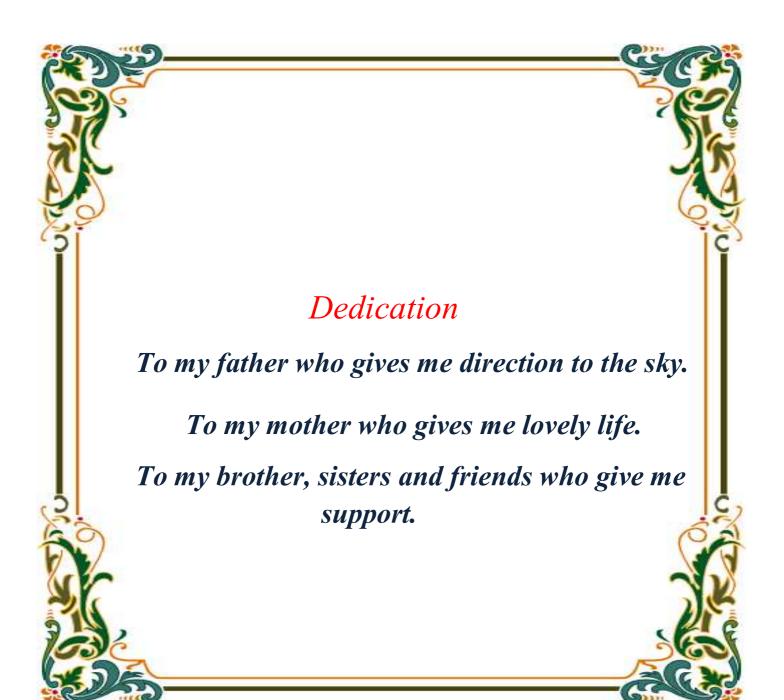
Prepared by: Mohammed Abass Ali Mohammed Salih

Supervisor:

Dr. Eihab Abdelraouf Mustafa

June 2017





Acknowledgement

First of all I would like to thank my scientific supervisor, Dr. Eihab Abdelraouf Mustafa at the Mechanical Engineering Department of Sudan University for his guidance and encouragement throughout the duration of this Thesis. I would also like to thank my colleagues from MSc of Mechanical Engineering, Finally I would also like to thank my family.

Abstract

In an internal combustion engine, valve timing is an important design parameter which affects many engine performance parameters. In this study, engine specifications were applied to the DIESEL-RK program to test engine exhaust performance and emissions and the results were there is a reduction in engine performance by de-creasing the Exhaust valve opening (EVO) angle Before bottom dead centre BBDC at all engine speeds and an increase of Oxides of Nitrogen (NOx) emissions and the effect of Exhaust valve closing (EVC) reduction is beneficial to Power, Brake Mean Effective Pressure (BMEP), and Specific Fuel Consumption (SFC) for different speed and also will reduce engine Oxides of Nitrogen (NO) and Carbon Monoxide (CO) emission down to 20° .

المستخلص

في محركات الاحتراق الداخلي يمثل توقيت الصمامات عامل مهم من العوامل التي تؤثر على أداء المحركات ، في هذه الدراسة ، تم تطبيق مواصفات محرك على برنامج محاكاة على أداء المحركات ، في الدراسة ، تم تطبيق مواصفات محرك على برنامج محاكاة DIESEL-RK لاختبار أداء وانبعاثات غازات عادم المحرك وتمثلت النتائج في انخفاض في أداء المحرك عن طريق نقصان زاوية (IVO) وذلك في جميع سرعات المحرك وزيادة انبعاثات أكاسيد النيتروجين وأيضا عند نقصان زاويه (EVC) عند السرعات المختلفه تقلل نسبة انبعاثات ثاني أكسيد الكربون((CO_2)) واكاسيد النتروجين ((NOx)) الى (CO_2) .

Table of Contents

	Content	Page
	الأيــة	I
	Dedication	Ii
	Acknowledgement	Iii
	Abstract	Iv
	المستخلص	V
	Table of Contents	Vi
	List of Tables	Viii
	List of Figures	Ix
	List of Symbols	X
	List of Abbreviations	Xi
	Chapter One	711
	Introduction	
1.1	Introduction	2
1.2	Problem Statement	2
1.3	Research Objectives	3
1.4	Methodology	3
1.5	Research Schedule	3
	Chapter Two	
	Literature Review	
2.1	4-Stroke And 2-Stroke Valve Timing	5
2.2	The Impact of Valve Events Upon Engine Performance And Emissions	9
2.2.1	Effect of Changes to Intake Valve Opening Timing – IVO	9
2.2.2	Effect of Changes to Intake Valve Closing Timing – IVC	10
2.2.3	Effects of Changes to Exhaust Valve Opening Timing - EVO	11
2.2.4	Effects of Changes to Exhaust Valve Closing Timing - EVC	12
2.3	Literature Review	13
	Chapter Three	
	Research Methodology	
3.1	Methodology	23
3.1.1	General engine parameters	23
3.1.2	Process of Diesel -RK	24
3.1.3	Combustion chamber design fuel enjection equipment	28
3.1.4	Gas exchange system	29
3.1.5	Super-or turbo-charging and EGR system	30
3.1.6	Engine operating points data	31
3.1.7	Fuel ingector design	31
3.1.8	Fuel properties	32
3.2	Engine Specifications	33

Chapter Four Results And Discussions								
4.1	Introduction	35						
4.2	Data Analysis	35						
4.3	Effect of engine speed and specific carbon dioxide emission	41						
4.4	Effect of engine speed and fraction of wet NOx	42						
4.5	Effect of engine speed and Summary emission PM and NOx	44						
4.6	Effect of engine speed and engine brake power	45						
4.7	Effect of engine speed and specific fuel consumption	47						
4.8	Effect of engine speed and efficiency of piston engine	48						
Chapter Five Conclusion & Recommendations								
5.1	Conclusions	51						
5.2	Recommendations for Future Works	52						
	References							
	References 53							

List of Tables

Table	Title	Page
3.1	Engine Specifications	33
4.1	change After Top Dead Center and After Bottom Dead Center	35
4.2	change timing valve and (CO2)emission	42
4.3	change timing valve and (NOx)emission	43
4.4	change timing valve and Summary emission	45
4.5	timing valve and engine brake power	46
4.6	timing valve and specific fuel consumption	48
4.7	change timing valve and efficiency of piston engine	49

List of Figures

Figure	Title	Page
2.1	Typical valve timing diagram	5
2.2	Opening and closing point of the valve	6
2.3	Valve opening duration	7
2.4	Valve timing diaggram showing valve overlap in 4-stroke engine	7
2.5	Valve timing diagram showing scavenging period in 2-stroke engine	8
2.6	Rock position	9
3.1	working cycle, fuel and method of ignition	24
3.2	basic engine design and number of cylinders	25
3.3	cylinder bore, piston stroke, engine spead and compression ratio	25
3.4	Pressure and Temperature	26
3.5	Super-or turbocharged engine	26
3.6	Inlet valve timing	27
3.7	Exhaust valve timing	27
3.8	Injector design parameters	28
3.9	Engine ports esign	29
3.10	General engine scheme	30
3.11	Window and data injection profile simulation at every operating mode	32
4.1	Effect of engine speed and specific carbon dioxide emission	41
4.2	Effect of engine speed and fraction of wet NOx	42
4.3	Effect of engine speed and Summary emission PM and NOx	44
4.4	Effect of engine speed and engine brake power	45
4.5	Effect of engine speed and specific fuel consumption	47
4.6	Effect of engine speed and efficiency of piston engine	48

List of Symbols

Symbol	Details
D	Diameter
kg/s	Kilogram per second
mm	Millimetre
M	Mass
P	Power
T	Temperature
V	Volume
ρ	Density

List of Abbreviations

Abbreviations	Details
ABDC	After bottom dead centre
ATDC	After top dead centre
BBDC	Before bottom dead centre
BDC	Bottom dead centre
BTDC	Before top dead centre
EGR	Exhaust gas recirculation
IVC	Intake valve closing
IVO	Intake valve opening
EVC	Exhaust valve closing
EVO	Exhaust valve opening
IPC	Intake port closing
IPO	Intake port opening
rpm	Revolution per minute
TDC	Top dead centre
CO	Carbon Monoxide
CO2	Carbon Dioxide
NOx	Oxides of Nitrogen
BP	Brake Power
η	Thermal Efficiency
SFC	Specific Fuel Consumption
LIVC	Late Intake Valve Close
SI	Spark-Ignition

Chapter One Introduction

1.1 Introduction

Valve timing is a system used to measure valve operation in relation to crankshaft position (in degrees), specifically the points when the valves open, how long they remain open, and the points when they close. In internal combustion engines, valves behavior (lift and timing) is one of the most important parameters which have a major effect on the engine operation and emission. The intake and exhaust valves must open and close at the right time. Otherwise, the performance of the engine will be poor [1]. The valves in four-stroke cycle engines are almost universally of a poppet type which are spring loaded toward a valve-closed position and opened against that spring bias by cam on rotating camshaft with the cam shaft being synchronized by the engine crankshaft. The valves in two-stroke cycle engines are generally simple apertures or ports in the cylinder sidewall which are uncovered or opened by piston movement. In four-stroke cycle engines, the valve timing is controlled by the camshaft and it can be varied by modifying the camshaft Many two-stroke cycle do not have a camshaft, and the valve timing can only be varied by machining the valve ports. In this study, a four stroke engine is used for the simulation where the valve train on the cylinder head of the engine is replaced with a piston that controls the intake and exhaust port opening and closing. The intake port modification has been carried out to vary the intake timing of the engine to see the effect of intake timing on in cylinder air flow[2].

1.2 Problem Statement

Valve timing is an important design parameter which affects many engine performance parameters as brake power, brake thermal efficiency and specific fuel consumption, on other hand engine emissions NOx , CO and CO2 are considered. The intake and exhaust valves should open and close at the right time. If the engine is operating at conditions other than the design conditions, non-optimized

performance is obtained. Thus, the effect of valve timing configuration should be investigated for optimum performance of the newly developed piston-controlled intake port single cylinder 4-stroke engine.

1.3 Research Objectives

The objective of this research is to investigate the effect of intake and exhaust valve timing of gasoline 4-stroke engine on engine performance (BP, η , and SFC) and exhaust emissions (NOx, CO, and CO2).

1.4 Methodology

The full cycle thermodynamic engine simulation software DIESEL-RK is designed for simulating and optimizing working processes of two- and four-stroke internal combustion engines with all types of boosting. The program can be used for modeling the following types of engines:

- * Diesel engines.
- * Petrol engines.
- * Gas engines.

The DIESEL-RK is a best tool for a new engine concept analysis because one allows creation of a new project very easily and fast. Philosophy of work of preand post-processors is focused on help to engineer.

1.5 Research Schedule

A ativities	First month			second month				third month				
Activities												
Literature review		_					_					
Conceptual design												
Design analysis												
Results discussion												

Chapter Two Literature Review

2.1 4-Stroke And 2-Stroke Valve Timing

Valve timing is a system developed for measuring valve operation in relation to crankshaft position (in degrees), particularly the points when the valves open, how long they remain open, and when they close. Valve timing of 4-stroke and 2-stroke engine can be drawn into valve timing diagram as shown in the Figure (2-1). Valve timing is probably the single most important factor in tailoring an engine for special needs. An engine can be made to produce its maximum power in various speed ranges by altering the valve timing [22].

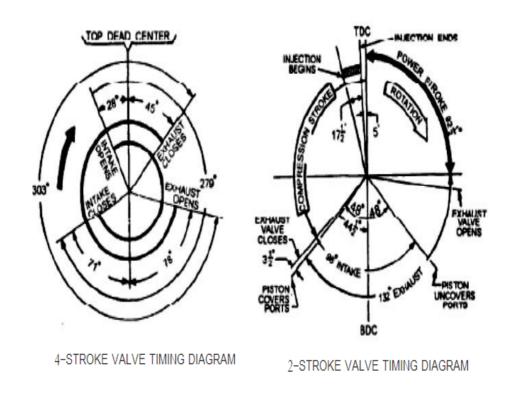


Figure (2-1): Typical valve timing diagram

Source: SweetHaven (1985) [22]

The following factors together make up a valve operating sequence:

1. The opening and closing points as shown in the Figure (2-2) are positions of the crankshaft (in degrees) when the valves just begin to open and just finish closing.

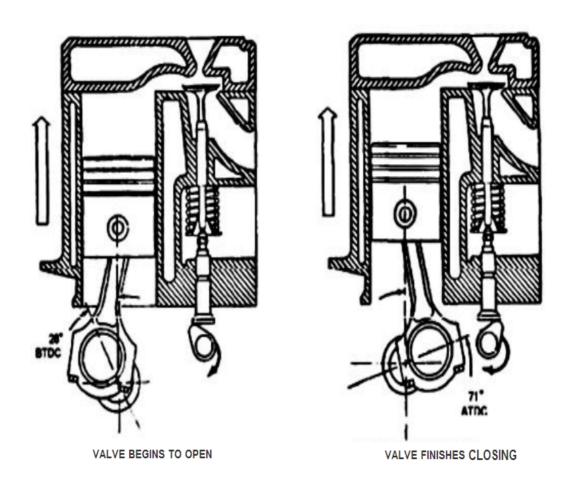


Figure (2-2): Opening and closing point of the valve

Source: SweetHaven (1985) [22]

2. Duration as shown in Figure (2-3) is the amount of crankshaft rotation (in degrees) that a given valve remains open.

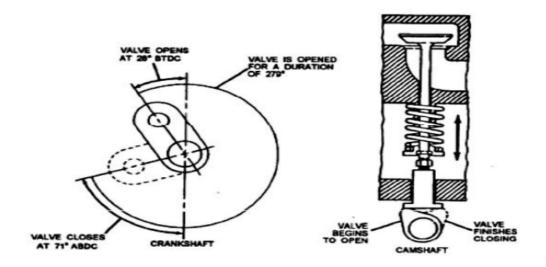


Figure (2-3):Valve opening duration

Source: SweetHaven (1985) [22]

3. Valve overlap as shown in Figure (2.4) is a period in a four-stroke cycle when the intake valve opens before the exhaust valve closes.

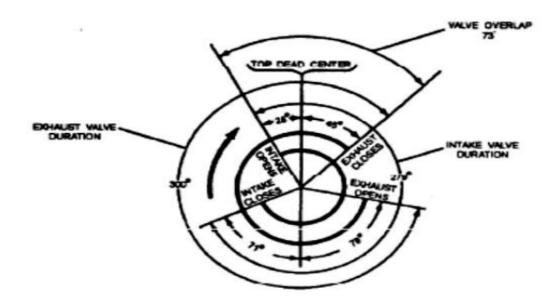


Figure (2-4): Valve timing diaggram showing valve overlap in 4-stroke engine

Source: SweetHaven (1985) [22]

4. Scavenging as shown in Figure 2.5 is a period in a two-stroke cycle when the replacement of the products of combustion in the cylinder from the previous power stroke with fresh-air charge to be burned in the next cycle[22].

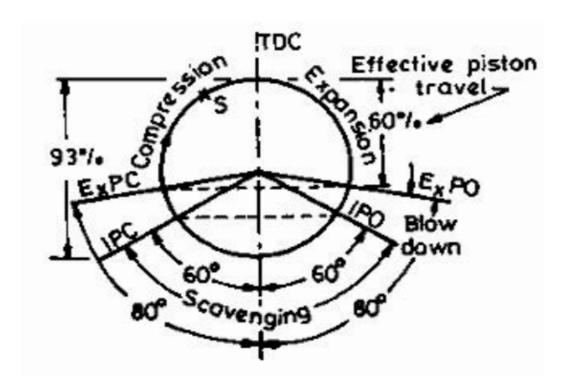


Figure (2-5):Valve timing diagram showing scavenging period in 2-stroke engine

Source: SweetHaven (1985) [22]

5. Valve timing considerations, throughout the crankshaft revolution, the speed of the piston changes. From a stop at the bottom of the stroke, the piston reaches its maximum speed halfway through the stroke and gradually slows to a stop as it reaches the end of the stroke. The piston behaves exactly the same on the down stroke. One of these periods begins at approximately 15 to 20 degrees before top dead center (BTDC) and ends at approximately 15 to 20 degrees after top dead center (ATDC). The other period begins approximately 15 to 20 degrees before

bottom dead center (BBDC) and ends approximately 15 to 20 degrees after bottom dead center (ABDC). These two positions are shown in Figure 2.6. These positions are commonly referred to as rock positions[22].

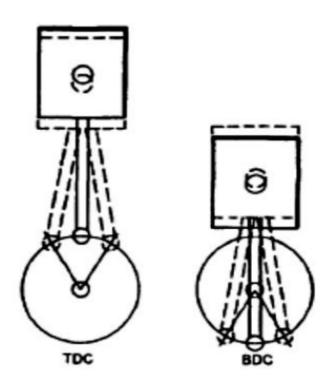


Figure (2-6): Rock position

Source: SweetHaven (1985) [22]

2.2 The impact of valve events upon engine performance and emissions

2.2.1 Effect of Changes to Intake Valve Opening Timing – IVO

The opening of the intake valve allows air/fuel mixture to enter the cylinder from the intake manifold. In the case of direct injection engines, only air enters the cylinder through the intake valve. The timing of IVO is the second parameter that

defines the valve overlap and this is normally the dominant factor when considering which timing is appropriate for a given engine.

Opening the intake valve before TDC can result in exhaust gasses flowing into the intake manifold instead of leaving the cylinder through the exhaust valve. The resulting EGR will be detrimental to full load performance as it takes up space that could otherwise be taken by fresh charge. EGR may be beneficial at part load conditions in terms of efficiency and emissions as discussed above.

Later intake valve opening can restrict the entry of air/fuel from the manifold and cause in-cylinder pressure to drop as the piston starts to descend after TDC. This can result in EGR if the exhaust valve is still open as gasses may be drawn back into the cylinder with the same implications discussed above. If the exhaust valve is closed, the delay of IVO tends not to be particularly significant, as it does not directly influence the amount of fresh charge trapped in the cylinder.

Typical IVO timing is around 0-10° before TDC which results in the valve overlap being fairly symmetrical around TDC. This timing is generally set by full load optimization and, as such, is intended to avoid internal EGR[12].

2.2.2 Effect of Changes to Intake Valve Closing Timing – IVC

The volumetric efficiency of any engine is heavily dependent on the timing of IVC at any given speed. The amount of fresh charge trapped in the cylinder is largely dictated by IVC and this will significantly affect engine performance and economy.

For maximum torque, the intake valve should close at the point where the greatest mass of fresh air/fuel mixture can be trapped in the cylinder. Pressure waves in the intake system normally result in airflow into the cylinder after BDC and consequently, the optimum IVC timing changes considerably with engine speed. As engine speed increases, the optimum IVC timing moves further after

BDC to gain maximum benefit from the intake pressure waves. Closing the intake valve either before or after the optimum timing for maximum torque results in a lower mass of air being trapped in the cylinder. Early intake closing reduces the mass of air able to flow into the cylinder whereas late intake closing allows air inside the cylinder to flow back into the intake manifold. In both cases, the part load efficiency can be improved due to a reduction in intake pumping losses.

A typical timing for IVC is in the range of 50-60° after BDC and results from a compromise between high and low speed requirements. At low engine speeds, there will tend to be some flow back into the intake manifold just prior to IVC whereas at higher speeds, there may still be a positive airflow into the cylinder as the intake valve closes[12].

2.2.3 Effects of Changes to Exhaust Valve Opening Timing - EVO

As the exhaust valve opens the pressure inside the cylinder resulting from combustion is allowed to escape into the exhaust system. In order to extract the maximum amount of work (hence efficiency) from the expansion of the gas in the cylinder, it would be desirable not to open the exhaust valve before the piston reaches Bottom Dead Centre (BDC). Unfortunately, it is also desirable for the pressure in the cylinder to drop to the lowest possible value, i.e. exhaust back pressure, before the piston starts to rise. This minimizes the work done by the piston in expelling the products of combustion (often referred to as blow down pumping work) prior to the intake of a fresh charge. These are two conflicting requirements, the first requiring EVO to be after BDC, the second requiring EVO to be before BDC.

The choice of EVO timing is therefore a trade-off between the works lost by allowing the combusted gas to escape before it is fully expanded, and the work required raising the piston whilst the cylinder pressure is still above the exhaust

back-pressure. With a conventional valve train, the valve lifts from its seat relatively slowly and provides a significant flow restriction for some time after it begins to lift and so valve lift tends to start some time before BDC. A typical EVO timing is in the region of 50-60° before BDC for a production engine.

The ideal timing of EVO to optimize these effects changes with engine speed and load as does the pressure of the gasses inside the cylinder. At part load conditions, it is generally beneficial if EVO moves closer to BDC as the cylinder pressure is much closer to the exhaust back pressure and takes less time to escape through the valve. Conversely, full load operation tends to result in an earlier EVO requirement because of the time taken for the cylinder pressure to drop to the exhaust back-pressure[12].

2.2.4 Effects of Changes to Exhaust Valve Closing Timing - EVC

The timing of EVC has a very significant affect on how much of the Exhaust gas is left in the cylinder at the start of the engine's intake stroke. EVC is also one of the parameters defining the valve overlap, which can also have a considerable affect on the contents of the cylinder at the start of the intake stroke.

For full load operation, it is desirable for the minimum possible quantity of exhaust gas to be retained in the cylinder as this allows the maximum volume of fresh air & fuel to enter during the Intake stroke. This requires EVC to be at, or shortly after TDC. In engines where the exhaust system is fairly active, the timing of EVC influences whether pressure waves in the exhaust are acting to draw gas out of the cylinder or push gas back into the cylinder. The timing of any pressure waves changes with engine speed and so a fixed EVC timing tends to be optimized for one speed and can be a liability at others.

For part load operation, it may be beneficial to retain some of the exhaust gasses, as this will tend to reduce the ability for the cylinder to intake fresh air &

fuel. Retained exhaust gas thus reduces the need for the throttle plate to restrict the intake and results in lower pumping losses in the intake stroke. Moving EVC Timing further after TDC increases the level of internal EGR with a corresponding reduction in exhaust emissions.

There is a limit to how much EGR the cylinder can tolerate before combustion becomes unstable and this limit tends to become lower as engine load and hence charge density reduces. The rate of combustion becomes increasingly slow as the EGR level increases, up to the point where the process is no longer stable. Whilst the ratio of fuel to oxygen may remain constant, EGR reduces the proportion of the cylinder contents as a whole that is made up of these two constituents. It is this reduction in the ratio of combustible to inert cylinder contents which causes combustion instability. Typical EVC timings are in the range of 5-15° after TDC. This timing largely eliminates internal EGR so as not to detrimentally affect full load performance[12].

2.3 Literature Review

There are many previous studies about natural gas engine. In addition, most of them focus on the studying of improving the engine performance and emission. Yusoff et al.[2]worked on finding the effects in different valve timing and ignition issues in compressed natural gas direct injection. The intake and exhaust valves must open and close at the right time. Otherwise, the efficiency, fuel consumption and emission will be poor. Injection time and ignition time also have to be in exactly right time to produce maximum power and minimum pollution. Moreover, they found the better spray characteristics at proper pressure and temperature can accelerate the air-fuel mixing. Kalam et al.[3] tried to improve a natural gas engine. They compared between the gasoline and natural gas in three difference situations based on the same engine. They found that: natural gas gave 15% - 20%

lower power than gasoline but the specific fuel consumption was also 18% less. This was testing at the same throttle position. For the same output power, natural gas also had lower fuel flow rate and better emission except the NOx (Oxides of Nitrogen). Finally, they set up the output Natural gas was little higher but the emission was much better except the NOx.Department of Mechanical Engineering, Federal University of Technology [4] compensated the longer ignition delays and slower burning rates by advanced injection timing. The testing was undergoing a natural gas diesel engine (compressed ignition engine). The standard injection timing was 30° BTDC (Before Top Dead Center). The advanced injection timing was 33.5° BTDC. Result found that advanced injection timing was not recommended for high load condition because of high HC (Hydrocarbon). The test was continued by advancing another 1.5° more. But the engine could not run smoothly. Michael et al. [5] investigated on naturally aspirated Miller Cycle SI engine (Spark-Ignition Engine) with LIVC (Late Intake Valve Close) based on first and second law analyses. Their analytical methodology was on two computermodeling tools. They assumed that the cylinder was divided into two zones, unburned and burned zone. Each zone was uniform. Combustion was modeled as a turbulent flame. Heat transfer, homogeneous mixture, temperature etc. were considered as well. They found that LIVC required less fuel to produce the same output and could achieve up to 6.3% higher indicated thermal efficiency at part load.. Yorihiro et al. [6] applied the Miller cycle to a lean-burn gas engine cogeneration. They tested two types of combustion chamber shape. One was high turbulent type and the other was low unburnt type. They found the low unburnt type less likely to cause knocking. Therefore, advance ignition timing could be applied which improved the exhaust of total hydrocarbon concentration and thermal efficiency. The higher swirl ratio was, the higher temperature and heat loss were. power of natural gas 10% higher. They found the fuel consumption of Bassett et al. [7] simulated a simple and cheap mechanism that allows two-state LIVC control. This device allowed the engine to operate with wider than normal throttle settings at low load, which reduced pumping losses. They located a reed valve in the intake manifold. At full load, reed valve prevented the charge from being rejected out from the cylinder. At low load, the reed valve allowed the charge to return freely. This can reduce BSFC (Brake Specific Fuel Consumption) around 7% and also reduce NOx. Shiga et al. [8] found that the intake capacity chamber installation reduced the pumping loss by applying LC (Late Closing). They varied the valve timing and compression ratio. They found that the pumping loss trend was not really affected by the expansion ratio but it was mainly affected by intake valve timing. And pumping loss could be decreased by LC. They could not clearly conclude the effect of intake valve timing on the BSFC. But BSFC decreased with the increasing of expansion ratio. The experiment results could be explained by calculations that the expansion ratio was ten times as effective as the Therefore, advance ignition timing could be applied which improved the exhaust of total hydrocarbon concentration and thermal efficiency. The higher swirl ratio was, the higher temperature and heat loss were. power of natural gas 10% higher. They found the fuel consumption of Bassett et al. [7] simulated a simple and cheap mechanism that allows two-state LIVC control. This device allowed the engine to operate with wider than normal throttle settings at low load, which reduced pumping losses. They located a reed valve in the intake manifold. At full load, reed valve prevented the charge from being rejected out from the cylinder. At low load, the reed valve allowed the charge to return freely. This can reduce BSFC (Brake Specific Fuel Consumption) around 7% and also reduce NOx. Shiga et al. [8] found that the intake capacity chamber installation reduced the pumping loss by applying LC (Late Closing). They varied the valve timing and compression ratio. They found that the pumping loss trend was not really affected by the expansion ratio but it was mainly affected by intake valve timing. And pumping loss could be decreased by LC. They could not clearly conclude the effect of intake valve timing on the BSFC. But BSFC decreased with the increasing of expansion ratio. The experiment results could be explained by calculations that the expansion ratio was ten times as effective as the compression ratio in increasing the thermal efficiency. Chih Wu et al. [9] used the computer simulation the Miller cycle comparing to Otto cycle based on thermodynamic method. They simulate both Miller cycle with and without supercharger. The Miller cycle without supercharger processed lower mass than Otto cycle without supercharger. The pressure and temperature at the end of compression process were lower. Then they assumed the intake pressure to be 110 kPa for supercharge Miller cycle. They still found that temperature at the end of compression stroke was lower than that of Otto cycle without supercharger. The net work, MEP (Mean Effective Pressure) and mass inside the cylinder output of Miller cycle were also lower than that of Otto cycle without supercharger. Then they simulated the Mazda engine that operated on Miller cycle. The pressure of supercharger was 196.5 kPa which higher than they simulated. The result was that there was more mass in the cylinder, higher MEP and more net work output. They suggested that Miller cycle should operate with supercharger. Gyeung Ho Choi et al. [10] simulated the Miller cycle through the computer simulation according to the EIVC (Early Intake Valve Closure) and LIVC (Late Intake Valve Closure) method by construction the test engine compression ratio in increasing the thermal efficiency. Chih Wu et al. (9) used the computer simulation the Miller cycle comparing to Otto cycle based on thermodynamic method. They simulate both Miller cycle with and without supercharger. The Miller cycle without supercharger processed lower mass than Otto cycle without supercharger. The pressure and temperature at the end of compression process were lower. Then they assumed the intake pressure to be 110 kPa for supercharge Miller cycle. They still found that

temperature at the end of compression stroke was lower than that of Otto cycle without supercharger. The net work, MEP (Mean Effective Pressure) and mass inside the cylinder output of Miller cycle were also lower than that of Otto cycle without supercharger. Then they simulated the Mazda engine that operated on Miller cycle. The pressure of supercharger was 196.5 kPa which higher than they simulated. The result was that there was more mass in the cylinder, higher MEP and more net work output. They suggested that Miller cycle should operate with supercharger. Gyeung Ho Choi et al. [10] simulated the Miller cycle through the computer simulation according to the EIVC (Early Intake Valve Closure) and LIVC (Late Intake Valve Closure) method by construction the test engine using the engine analysis program and by changing the valve close timing. The real engine was also tested. They observed that the error from the simulation was 5 Pferde Starke (PS) Finally, they found that the intake valve closing at 55 degree ABDC increased power, torque and brake thermal efficiency around 2 PS, 1.5 kg·m and 2% respectively. Wang et al. [11] studied the Miller cycle to reduce NOx emission in a diesel engine. They compare the original valve timing with three different Miller cycles. Late intake valve opens and early intake valve closures are used as follow. the intake valve opened 20° late and closed 20° earlier. Miller 2, the intake valve opened 25° late and closed 25° earlier. Miller 3, the intake valve opened 10° late and closed 10° earlier. They found that the different output powers were quite small. Miller cycle 1 was the best for reducing NOx, which can reduce more than 10%. The exhaust gas temperatures of Miller cycle were lower than normal. They compare the original valve timing with three different Miller cycles. Late intake valve opens and early intake valve closures are used as follow. the intake valve opened 20° late and closed 20° earlier. Miller 2, the intake valve opened 25° late and closed 25° earlier. Miller 3, the intake valve opened 10° late and closed 10° earlier. They found that the different output powers were quite small. Miller cycle 1 was the best for reducing NOx, which can reduce more than 10%. The exhaust gas temperatures of Miller cycle were lower than normal. Alla et al. [12]researched on effect of injection timing on the performance of dual fuel engine. They worked on a single cylinder indirect injection diesel engine fueled with gaseous fuel. Diesel fuel was used as the pilot fuel and methane or propane was used as the main fuel, which was inducted in the intake manifold to be mixed with the intake air. Three values of injection timings of 25°, 27.5 ° and 30° BTDC were used in the test. They found that retarding injection timing (at 25° BTDC) delayed combustion. The temperature of mixture is not enough to propagate in the whole mixture. The amount of unburned hydrocarbon and CO (Carbon Monoxide) increase as injection timing retards. While NOx and thermal efficiency increases with the advanced injection timing.

Takagaki and Raine [13] used a single cylinder, spark ignition engine to study effects of the compression ratio on nitric oxide emissions using natural gas. They found that for fixed ignition timing nitric oxide emissions increased with increasing compression ratio. But for Maximum Brake Torque (MBT) timing, nitric oxide emissions first increased and then decreased. Koichi et al. [14] investigated the effect of Miller cycle on MEP for high-pressure supercharged gasoline engine. Intake valve closing timing was set at 75 degrees in the case of the late intake valve closure. They found that the exhaust gas temperature did not increase and the maximum BMEP (Brake Mean Effective Pressure) increased because of knocking limit improvement. Miller-cycle with a supercharger, which is highly efficient at high-pressure ratio and an intercooler, with high efficiency, can increase IMEP (Indicated Mean Effective Pressure). Caton [15] simulated the nitric oxide emissions in spark-ignited automotive engine using a cycle simulation, which employed three zones for the combustion process: unburned gas, adiabatic

core region and boundary layer gas. The effects of engine parameters such as equivalence ratio, ignition timing, inlet manifold pressure and engine speed were examined. He found that maximum nitric oxide was at about 5 equivalent ratio of 0.9. Nitric oxide increased as advanced ignition timing and higher inlet manifold pressures. For an equivalent ratio of 0.9, the decreasing available time as engine speed increases dominates the increase of gas temperature. Caton[16] focused on the effect of compression ratio on nitric oxide emissions for a spark ignition engine. The study completed for a commercial, 5.7 liters spark ignition V-8 engine operating at a part load condition at 1400 rpm with an equivalence ratio of one and MBT (Maximum Brake Torque) ignition timing. He mentioned that there are many researches on this effect, which showed different results. A number of previous studies indicated that the increment of compression ratio increased nitric oxide emissions. However, other studies showed the opposite. He expected that results might be affected by uncontrolled and variable condition (temperature, pressure and humidity) and the ignition timing. Furthermore, the conclusion might be different depending on whether the ignition timing was constant or set to MBT ignition timing and equivalence ratio. For his investigation, he adjusted to provide MBT timing and constant throttle position (constant load). He found that increasing the compression ratio resulted in decreasing brake specific nitric oxide value due to the changes of gas temperature, cylinder pressure and brake specific fuel consumption. However, it could decrease as compression ratio increased at high compression ratio, which might involve with the burn duration. Engineers at Tokyo Gas Co., Ltd., and Yanmar Diesel Engine Co., Ltd.[17] modified a diesel engine to Miller-Cycle natural gas engine. They both designed for EIVC and LIVC for this 23.15-liter engine. LIVC required lower cost and fewer design changes. The engine operated on premixed natural gas with turbocharger. It was also a close loop control. The engine could achieve 36.1% brake thermal efficiency. Moreover,

the cogeneration system produced 300 kW of electric and achieved 83.5% energy efficiency. Sarkhi et al. [18] modeled the efficiency of a Miller engine in term of thermodynamics calculation. They found that the effects of the temperaturedependent specific heat of the working fluid on the cycle performance were significant and should have been considered in design. A slight increase in some parameter would have an impact on the thermal efficiency of the cycle. 6 Akira et al. [19] developed Miller gas engine for the purpose of attaining electrical efficiency equivalent to that of a diesel engine on the basis of the lean burn gas engine for high efficiency and low NOx emission. Miller cycled gas engine cogeneration package improved efficiency to 40% level by the Miller cycle. Mohamed (20)used propane as a fuel. He tested at the speed of 1500 to 3000 rpm with the interval of 500. He varied load of 50%, 75% and 100%. He also tested at different ignition timings and found the relation among BMEP, speed, load, ignition timing, MBT, BSFC and emission. Results showed that the engine could be operated with propane over a wide range of air-fuel ratios with less carbon dioxide, carbon monoxide and hydrocarbon emissions compare to operation with gasoline. The differences in fuel characteristics, the operation of the engine on propane were accompanied with some power loss. Sarkhi et al. [18] modeled the efficiency of a Miller engine in term of thermodynamics calculation. They found that the effects of the temperature-dependent specific heat of the working fluid on the cycle performance were significant and should have been considered in design. A slight increase in some parameter would have an impact on the thermal efficiency of the cycle. 6 Akira et al. [19] developed Miller gas engine for the purpose of attaining electrical efficiency equivalent to that of a diesel engine on the basis of the lean burn gas engine for high efficiency and low NOx emission. Miller cycled gas engine cogeneration package improved efficiency to 40% level by the Miller cycle. Mohamed [20] used propane as a fuel. He tested at the speed of 1500

to 3000 rpm with the interval of 500. He varied load of 50%, 75% and 100%. He also tested at different ignition timings and found the relation among BMEP, speed, load, ignition timing, MBT, BSFC and emission. Results showed that the engine could be operated with propane over a wide range of air-fuel ratios with less carbon dioxide, carbon monoxide and hydrocarbon emissions compare to operation with gasoline. The differences in fuel characteristics, the operation of the engine on propane were accompanied with some power loss. The fuel economy of the engine on propane got poor with increase in speed from 2500 to 3000 rpm. HC and CO of the propane was lower comparing to gasoline. But CO2 (Carbon Dioxide) was higher. Lee Ju Hee[21] researched on the thermal efficiency on an industrial engine with Miller cycle. A diesel engine was retrofitted to natural gas engine for better duration. He changed the closing time of intake valve for adapting Miller cycle. Intake cam lift compensation test was added on the EIVC test and also effective compression pressure compensation test was added on the LIVC test. He found that EIVC had less thermal efficiency than the basic cam experiment. LIVC test at 51 degree-ABDC (After Bottom Dead Center) bettered the fuel consumption ratio around 5-8% and brake thermal efficiency around 2-3%. LIVC test at 77 degree-ABDC bettered the fuel consumption ratio and brake thermal efficiency around 3-7% and 1-2% respectively. The quantity of NOx was reduced about 5-10%.

Chapter Three Research Methodology

3.1 Methodology

DIESEL-RK is A full cycle thermodynamic engine simulation software. One is designed for simulating and optimizing working processes of **two-** and **four- stroke** internal combustion engines with all types of boosting. The program can be used for modeling the following types of engines:

- DI Diesel engines, and engines fueled by bio-fuels.
- SI petrol engines.
- SI gas engines systems, and engines fueled by different gases: Methane, Propane-Buthane, Biogas, Wood gas, Synge's, etc.
- Two-stroke engines with uniflow and loop scavenging, opposed piston engines (OP or Junkers engines).
- The DIESEL-RK is a thermodynamic software: engine cylinders are considered as open thermodynamic systems[23].
- Input data are grouped together according to main systems of an engine:

3.1.1 General engine parameters:

- Bore of cylinder.
- Stroke of piston.
- Nominal engine speed (RPM).
- Mechanism of transformation of a movement of the piston in rotation of the shaft: crank or other (For other mechanism it is necessary to set the function of the piston position versus).
- Length of the connecting-rod. For Junkers engine specify length of the both connecting rods.
- And angular offset.

- Number of cylinders.
- Design: (V or L, etc.).
- Compression ratio in the cylinder.
- Material of the piston or piston crown (iron / steel / aluminum).
- Material of the cylinder head (iron / steel / aluminum).
- Parameters of a blades on the cylinder head at air cooling:
- step of blades;
- Average height of blades.

3.1.2 Process of Diesel -RK

* Choices of working cycle, fuel and method of ignition see figure(3-1)

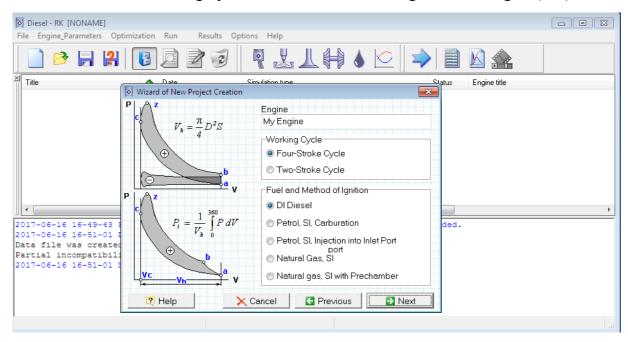


Figure (3-1): working cycle, fuel and method of ignition

* Choices of basic engine design and number of cylinders see figure (3-2)

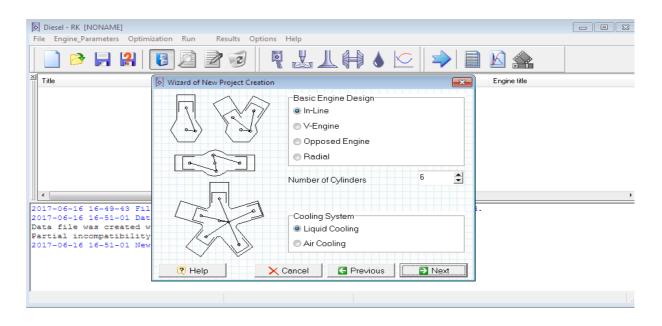


Figure (3-2): basic engine design and number of cylinders

* Input cylinder bore , piston stroke ,engine spead and compression ratio see figure(3-3)

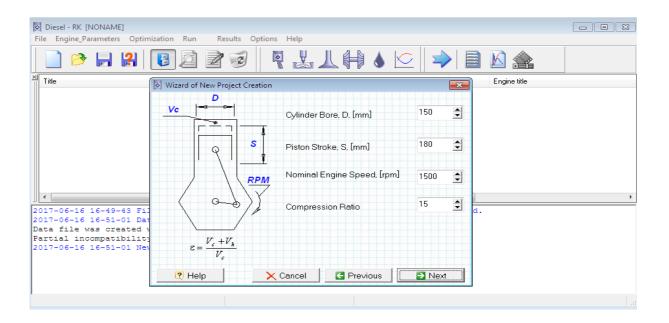


Figure (3-3): cylinder bore, piston stroke, engine spead and compression ratio

* Input Pressure and Temperature see figure(3-4)

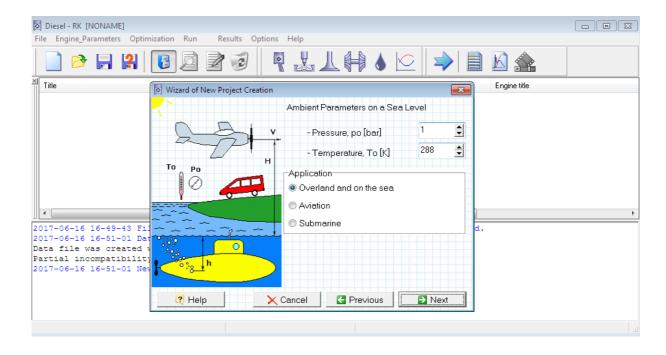


Figure (3-4): Pressure and Temperature

* * Choices of Super-or turbocharged engine see figure (3-5)

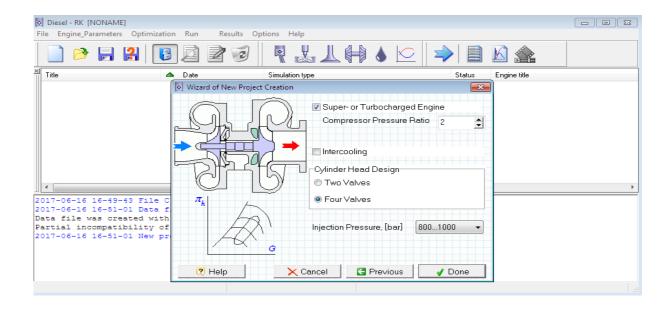


Figure (3-5): Super-or turbocharged engine

* Inlet valve timing, open area diagram of intake port see figure (3-6)

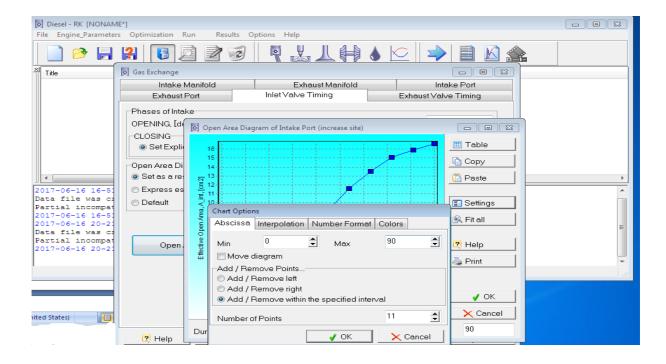


Figure (3-6): Inlet valve timing

*Exhaust valve timing, open area diagram of exhaust port see figure (3-7)

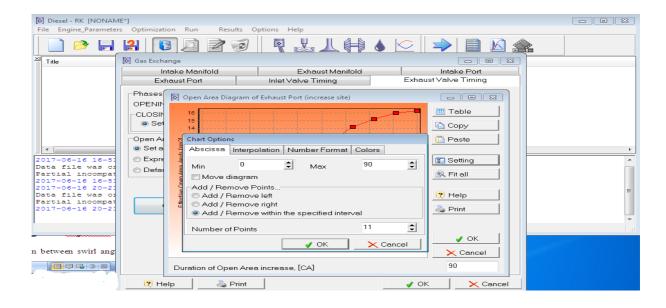


Figure (3-7): Exhaust valve timing

3.1.3 Combustion chamber design fuel enjection equipment

- Swirl Ratio Rs (it is the relation between swirl angular velocity and crank rotation velocity)
- may be specified by one of two ways:
- by the velocity of air in a combustion chamber at the end of compression;
- by the velocity of air in the cylinder at the end of intake.
- Specify Rs and way of its obtaining.
- Number of injectors. (It is actual for side injection system.)
- Number and Diameter of the injector nozzles.
- Nozzle discharge coefficient obtained as a result of the tests in atmospheric conditions.
- Design of injector nozzles: orientation of each nozzle in both planes.

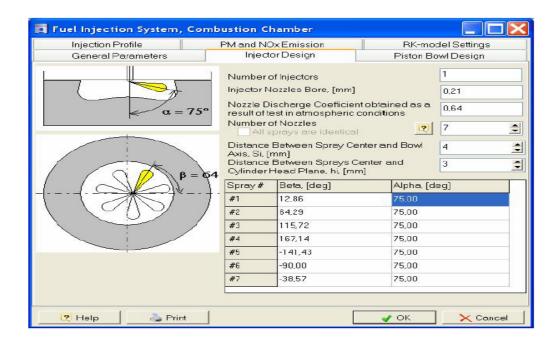


Figure (3-8): Injector design parameters

Source: [23]

3.1.4 Gas exchange system

- There is example for the four stroke engine only. For the two stroke engine see help system and DIESEL-RK preprocessor.
- Exhaust ports .
- · EVO: Angle of the exhaust beginning [deg. before BDC].
- · EVC: Angle of the exhaust beginning [deg. after TDC].

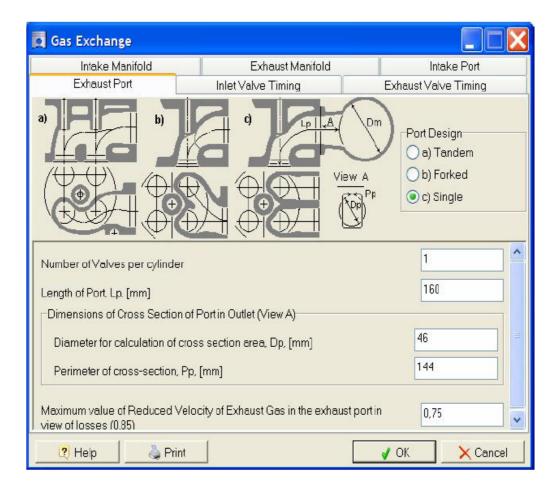


Figure (3-9): Engine ports esign

Source: [23]

3.1.5 Super-or turbo-charging and EGR system

• General engine scheme is presented in the figure(3-3)

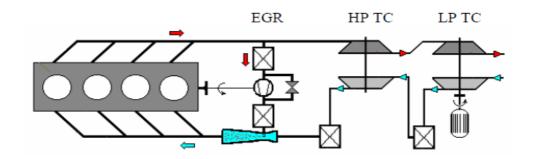


Figure (3-10): General engine scheme

Source: [23]

- Number of High Pressure Turbines on engine.
- High Pressure Turbine Map (in SAE format)*.
- Number of Low Pressure Turbines on engine.
- Low Pressure Turbine Map (in SAE format).
- Type of the turbines: axial or radial.
- Number of Low Pressure Compressors on engine.
- Low Pressure Compressor's Map (in SAE format)*.
- Electrical or mechanical assistance for LP or HP stage presented / not presented.
- Efficiency of Electrical or mechanical assistance for LP or HP stage.
- Intercooler after the Low Pressure Compressor presented / not presented.
- Thermal efficiency of Intercooler at nominal point,
- Pressure losses in Intercooler at nominal point,
- Temperature of coolant in Intercooler at nominal point,
- - Thermal efficiency of Intercooler at nominal point,

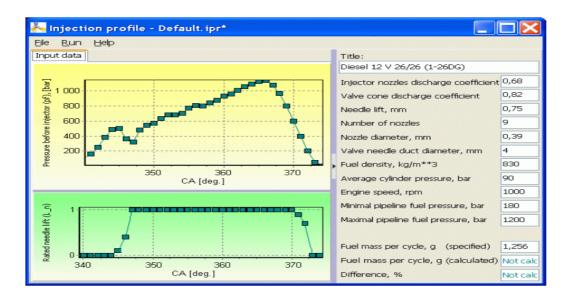
- First EGR Intercooler presented / not presented.
- Thermal efficiency of EGR Intercooler at nominal point,
- Pressure losses in EGR Intercooler at nominal point,
- Temperature of coolant in EGR Intercooler at nominal point.
- EGR pump presented / not presented.
- Efficiency of EGR Intercooler at nominal point.
- Second EGR Intercooler presented / not presented.

3.1.6 Engine operating points data

- The data being presented are necessary for setting of engine operating modes and for
- Calibration of engine models. For correct calibration of engine models it is necessary to specify data for several engine operating points with different engine speed and loads.
- The pre-processor of the code automatically offers only those data which are necessary for a chosen.
- Configuration of engine and configuration of its simulation. The presented document has not functions of the pre-processor therefore the list of input data includes only data for conventional diesel engine working in conditions at sea level.

3.1.7 Fuel ingector design

Fuel pipeline pressure profile and needle lift diagram are used for simulation of fuel flow in lock cone and nozzle to obtain the injection profile. The data being necessary for simulation at every operating mode are presented in figure (3-4)



Figure(3-11)Window and data injection profile simulation at every opeating mode.

Source: [23]

3.1.8 Fuel properties

- · Composition of fuel (0.87 C , 0.126 H , 0.004 O), [%].
- · Low Heating Value, [MJ/kg].
- · Density of fuel at 323 K, [kg/m3].
- · Specific Vaporization Heat, [J/kg].
- · Fuel Thermal Capacity at temperature of injector, [J/kg/K].
- · Molecular Mass (or weight).
- · Sulfur Fraction in the fuel, S, [%], is used for SO2 emission calculation.

- · Apparent Activation Energy for the Autoignition Process is used for autoignition delay
- . Calculation in diesel.

3.2 Engine Specifications

Table (3.1): Engine Specifications

No. of Cyls.	& Arrangeme	nt	4-cylinder, In-line
Valve Mechanism			16-valve , Chain Drive
Combusti	on Chamber		Pentroof Type
Fuel	System		SFI
Displaceme	nt cm3 (cu. in	.)	1798 (109.7)
Bore * Str	oke mm (in).		80.5 * 88.3 (3.17 × 3.48)
Compre	ssion Ratio		10.0 : 1
Max Output	(SAE ·	-NET)	98 kW @ 6000 rpm
			(132 HP @ 6000 rpm)
Max Torque	(SAE	- NET)	174 N.m @ 4400 rpm
			(128 ft.lbf @ 4400 rpm)
		Open	1° - 56° BTDC
	Intake	Close	65° – 10° ABDC
Valve Timing		Open	51° – 11° BBDC
	Exhaust	Close	3° – 43° ATDC
Firin	g Order		1-3-4-2
Engine Service Mass (Reference)		ence)	117 kg (257.9 lb)

Chapter Four Results And Discussions

4.1 Introduction:

This section deals with the analysis, presentation and discussion of results obtained in the investigation. All analyses and presentations were done using DIESEL-RK simulation software.

4.2 Data Analysis

After inserting data to DIESEL-RK simulation software, the results showed at the tabe (4-1), and used to graph program to draw the charts.

Table (4.1) shows the change After Top Dead Center and After Bottom Dead
Center

	50°C ATDC , 45°C ABDC					
SPEAD (RPM)	BP (KW)	SFC (Kg/KWh)	η	CO2	Nox	SE
750	8.5050	0.27395	0.30920	882.73	335.62	0.55929
1000	12.096	0.25704	0.32955	828.23	245.66	0.38483
1500	21.001	0.24707	0.34284	796.11	122.78	2.0846
2000	26.904	0.25859	0.32756	833.24	48.932	3.9358
2500	31.421	0.27800	0.30469	895.79	23.838	6.1764
		50°C ATDC	, 48°C ABD	С		
SPEAD (RPM)	BP (KW)	SFC (Kg/KWh)	η	CO2	Nox	SE
750	8.5043	0.27498	0.30805	886.04	326.49	0.54426
1000	12.075	0.25681	0.32984	827.49	235.28	0.36897
1500	19.078	0.24743	0.34234	797.28	119.15	0.18019
2000	26.960	0.25827	0.32798	832.19	49.127	3.9451
2500	31.393	0.27838	0.30428	897.01	24.248	6.2768
		50°C ATDC	, 51°C ABD	С		
SPEAD (RPM)	BP (KW)	SFC (Kg/KWh)	η	CO2	Nox	SE
750	8.5066	0.27339	0.30984	880.91	336.32	0.55956
1000	12.062	0.25654	0.33018	826.63	225.69	0.35379
1500	21.098	0.24644	0.34372	794.07	125.58	2.1053
2000	27.006	0.25791	0.32844	831.03	49.323	3.9473
2500	31.433	0.27802	0.30468	895.83	24.203	6.2740
		50°C ATDC	, 54°C ABD	С		
SPEAD (RPM)	BP (KW)	SFC (Kg/KWh)	η	CO2	Nox	SE
750	8.4754	0.27341	0.30981	880.99	327.33	0.54516
1000	12.085	0.25621	0.33061	825.56	227.31	0.35582
1500	21.149	0.24605	0.34426	792.83	127.17	2.1058
2000	27.047	0.25758	0.32886	829.97	49.265	3.9444
2500	31.466	0.27779	0.30493	895.10	24.153	6.2821
50°C ATDC , 57°C ABDC						
SPEAD (RPM)	BP (KW)	SFC (Kg/KWh)	η	CO2	Nox	SE
750	8.5730	0.27231	0.31106	877.46	337.39	0.55859
1000	12.085	0.25596	0.33094	824.75	226.60	0.35436
1500	21.189	0.24578	0.34464	791.96	121.68	2.0204
2000	27.038	0.25788	0.32848	830.93	50.316	4.0241
2500	31.505	0.27763	0.30511	894.57	24.158	6.3025

	53°C ATDC , 45°C ABDC					
SPEAD (RPM)	BP (KW)	SFC (Kg/KWh)	η	CO2	Nox	SE
750	9.5027	0.26972	0.31405	869.11	334.70	0.54919
1000	13.405	0.25494	0.33226	821.47	252.32	0.39309
1500	21.034	0.24706	0.34285	796.09	124.13	2.0814
2000	27.013	0.25809	0.32821	831.62	50.258	3.7180
2500	31.529	0.27769	0.30503	894.79	24.308	6.1368
		53°C ATDC	, 48°C ABD	С		
SPEAD (RPM)	BP (KW)	SFC (Kg/KWh)	η	CO2	Nox	SE
750	9.4599	0.27063	0.31299	872.04	332.29	0.54773
1000	13.421	0.25458	0.33273	820.32	252.46	0.39277
1500	21.099	0.24681	0.34320	795.29	124.53	2.1071
2000	27.025	0.25819	0.32808	831.93	49.722	3.9300
2500	31.576	0.27739	0.30537	893.81	24.326	6.1428
		53°C ATDC	, 51°C ABD	С		
SPEAD (RPM)	BP (KW)	SFC (Kg/KWh)	η	CO2	Nox	SE
750	9.4458	0.27039	0.31328	871.24	327.31	0.53949
1000	13.429	0.25425	0.33316	819.24	248.78	0.38674
1500	21.149	0.24639	0.33316	793.91	126.44	2.1002
2000	27.076	0.25779	0.32858	830.67	49.966	3.9260
2500	31.620	0.27700	0.30579	892.56	24.326	6.1382
		53°C ATDC	, 54°C ABD	С		
SPEAD (RPM)	BP (KW)	SFC (Kg/KWh)	η	CO2	Nox	SE
750	9.4779	0.26995	0.31378	869.84	325.76	0.53601
1000	13.405	0.25420	0.33323	819.08	249.35	0.38772
1500	21.200	0.24603	0.39119	792.75	128.42	2.1009
2000	27.115	0.25748	0.32899	829.65	50.094	3.9243
2500	31.674	0.27665	0.30619	891.42	24.418	6.1321
53°C ATDC , 57°C ABDC						
SPEAD (RPM)	BP (KW)	SFC (Kg/KWh)	η	CO2	Nox	SE
750	9.4390	0.27009	0.31362	870.28	326.64	0.53822
1000	13.440	0.25370	0.33388	817.49	249.19	0.38654
1500	21.242	0.24603	0.34430	792.75	128.42	2.1009
2000	27.227	0.25662	0.33008	826.88	51.538	3.7060
2500	31.627	0.27720	0.30558	893.19	24.675	6.2477

	56°C ATDC , 45°C ABDC					
SPEAD (RPM)	BP (KW)	SFC (Kg/KWh)	η	CO2	Nox	SE
750	9.5290	0.26966	0.31413	868.89	334.26	0.54825
1000	13.437	0.25493	0.33227	821.43	254.91	0.39694
1500	21.086	0.24703	0.34290	795.98	125.45	2.0764
2000	27.077	0.25798	0.32835	831.26	50.883	3.6999
2500	31.640	0.27734	0.30542	893.66	24.820	6.0918
		56°C ATDC	, 48°C ABD	С		
SPEAD (RPM)	BP (KW)	SFC (Kg/KWh)	η	CO2	Nox	SE
750	9.4721	0.27065	0.31297	872.10	333.12	0.54901
1000	13.460	0.25453	0.33279	820.17	254.08	0.39516
1500	21.147	0.24676	0.34328	795.10	126.21	2.0997
2000	27.086	0.25810	0.32819	831.67	50.227	3.9158
2500	31.686	0.27703	0.30576	892.66	24.819	6.0990
		56°C ATDC	, 51°C ABD	С		
SPEAD (RPM)	BP (KW)	SFC (Kg/KWh)	η	CO2	Nox	SE
750	9.4711	0.27038	0.31328	871.22	329.27	0.54251
1000	13.460	0.25424	0.33317	819.22	250.01	0.38853
1500	21.196	0.24633	0.34388	793.72	126.14	2.0945
2000	27.130	0.25777	0.32861	830.58	50.462	3.9150
2500	31.728	0.27668	0.30616	891.51	24.786	6.0960
		56°C ATDC	, 54°C ABD	С		
SPEAD (RPM)	BP (KW)	SFC (Kg/KWh)	η	CO2	Nox	SE
750	9.4690	0.27024	0.31344	870.78	330.45	0.54440
1000	13.443	0.2549	0.33323	819.07	252.60	0.39262
1500	21.244	0.24601	0.34431	792.71	129.34	2.0955
2000	27.173	0.25742	0.32906	829.47	50.469	3.9136
2500	31.773	0.27638	0.30649	890.55	24.808	6.0982
56°C ATDC , 57°C ABDC						
SPEAD (RPM)	BP (KW)	SFC (Kg/KWh)	η	CO2	Nox	SE
750	9.4802	0.26994	0.3380	869.80	331.23	0.54500
1000	13.496	0.25364	0.33396	817.28	255.16	0.39534
1500	21.283	0.24567	0.34479	791.61	130.59	2.0956
2000	27.220	0.25716	0.32938	828.64	50.536	3.9231
2500	31.810	0.27622	0.30666	890.04	24.780	6.1130

	59°C ATDC , 45°C ABDC					
SPEAD (RPM)	BP (KW)	SFC (Kg/KWh)	η	CO2	Nox	SE
750	9.5352	0.26968	0.31410	868.95	329.21	0.54021
1000	13.474	0.25486	0.33237	821.20	254.98	0.39696
1500	21.136	0.24698	0.34297	795.82	126.87	2.0694
2000	27.125	0.25798	0.32834	831.27	51.257	3.6954
2500	31.827	0.27635	0.30652	890.45	24.504	5.9297
		59°C ATDC	, 48°C ABD	С		
SPEAD (RPM)	BP (KW)	SFC (Kg/KWh)	η	CO2	Nox	SE
750	9.4900	0.27062	0.31300	872.00	334.72	0.55155
1000	13.475	0.25469	0.33258	820.68	256.43	0.39913
1500	21.191	0.24676	0.34327	795.12	127.09	2.0967
2000	27.181	0.25765	0.32876	830.22	51.459	3.7048
2500	31.795	0.27671	0.30612	891.62	25.381	6.0546
		59°C ATDC	, 51°C ABD	С		
SPEAD (RPM)	BP (KW)	SFC (Kg/KWh)	η	CO2	Nox	SE
750	9.4614	0.27062	0.31300	872.00	334.72	0.55155
1000	13.497	0.25421	0.33322	819.11	254.86	0.39590
1500	21.244	0.24629	0.34393	793.59	129.35	2.0888
2000	27.232	0.25728	0.32924	829.01	51.688	3.7052
2500	31.914	0.27569	0.30725	888.34	24.567	5.9371
		59°C ATDC	, 54°C ABD	С		
SPEAD (RPM)	BP (KW)	SFC (Kg/KWh)	η	CO2	Nox	SE
750	9.4810	0.27059	0.31304	871.91	329.80	0.54417
1000	13.477	0.25414	0.33330	818.91	256.00	0.39772
1500	21.292	0.24595	0.34441	792.49	130.54	2.0900
2000	27.271	0.25695	0.32966	827.94	51.628	3.7009
2500	31.875	0.27609	0.30680	889.64	25.349	6.0617
59°C ATDC , 57°C ABDC						
SPEAD (RPM)	BP (KW)	SFC (Kg/KWh)	η	CO2	Nox	SE
750	9.4895	0.26995	0.31378	869.84	330.43	0.54379
1000	13.533	0.25356	0.33406	817.04	259.52	0.40183
1500	21.325	0.24566	0.34481	791.56	131.28	2.0909
2000	27.279	0.25706	0.32952	828.30	51.110	3.9046
2500	31.911	0.27593	0.30698	889.10	25.312	6.0758

62°C ATDC , 45°C ABDC						
SPEAD (RPM)	BP (KW)	SFC (Kg/KWh)	η	CO2	NOx	SE
750	9.5411	0.27104	0.31252	873.34	328.12	0.55655
1000	13.483	0.25505	0.33211	821.84	257.42	0.40114
1500	21.180	0.24697	0.34298	795.80	128.15	2.0658
2000	27.206	0.25767	0.32874	830.26	52.417	3.6508
2500	32.546	0.23798	0.35594	766.81	328.41	0.88791
		62°C ATDC	, 48°C ABD	C		
SPEAD (RPM)	BP (KW)	SFC (Kg/KWh)	η	CO2	NOx	SE
750	9.4944	0.27084	0.31275	872.71	338.81	0.55893
1000	13.504	0.25468	0.33259	820.64	256.71	0.39951
1500	21.251	0.24654	0.34357	794.42	130.37	2.0681
2000	27.265	0.25727	0.32925	828.99	52.883	3.6486
2500	60.408	0.23856	0.35508	768.68	313.52	0.92409
		62°C ATDC	, 51°C ABD	С		
SPEAD (RPM)	BP (KW)	SFC (Kg/KWh)	η	CO2	NOx	SE
750	9.4752	0.27059	0.31304	871.90	330.14	0.54474
1000	13.522	0.25435	0.33303	819.56	259.84	0.40375
1500	21.289	0.24626	0.34397	793.51	130.27	2.0838
2000	27.285	0.25719	032936	828.71	52.215	3.6890
2500	60.443	0.23833	0.35542	767.95	312.69	0.92237
		62°C ATDC	, 54°C ABD	С		
SPEAD (RPM)	BP (KW)	SFC (Kg/KWh)	η	CO2	NOx	SE
750	9.4880	0.27025	0.31344	870.80	330.22	0.54416
1000	13.515	0.25411	0.33334	818.80	258.50	0.40138
1500	21.331	0.24594	0.34442	792.47	131.60	2.0863
2000	27.324	0.25686	0.32977	827.67	52.260	3.6865
2500	60.517	0.23802	0.35587	766.96	312.55	0.92098
62°C ATDC , 57°C ABDC						
SPEAD (RPM)	BP (KW)	SFC (Kg/KWh)	η	CO2	NOx	SE
750	9.4974	0.27000	0.31373	870.00	333.05	0.54818
1000	13.551	0.25378	0.33377	817.75	265.24	0.41097
1500	21.368	0.24563	.34486	791.46	132.40	2.0854
2000	27.333	0.25694	0.32967	827.92	51.711	3.8842
2500	60.573	0.23774	0.35630	766.04	312.19	0.91945

4.3 Effect of engine speed and specific carbon dioxide emission

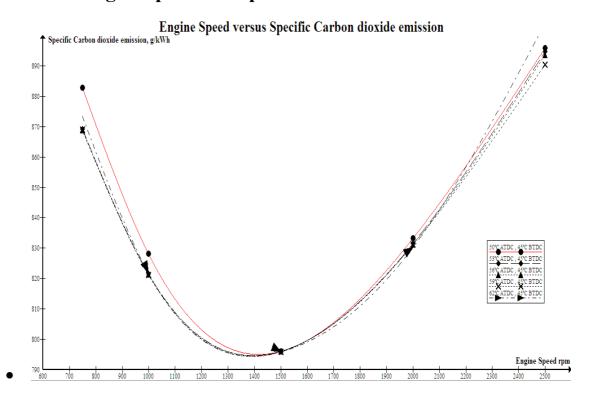


Figure (4-1): Effect of engine speed and specific carbon dioxide emission

As Shown in the figure(4-1) the emissions of (CO2)is decreased as the engine spead increased, until reached a mid speed (1500rpm), and started to increased as the engine speed increased, such as:

At system: (53°C ATDC, 45°C ABDC):

At 750rpm, (CO2)emission is found to be 869.11g\kWh, and at 1000rpm, (CO2)emission is found to be 821.47 g\kWh is in decreased, and at 1500rpm, (CO2)emission is found to be 796.09 g\kWh (reached amid), and at 2000rpm, (CO2)emission is found to be831.62 g\kWh, and at 2500rpm, (CO2)emission is found to be 894.79 g\kWh is in increased.

As Shown in the figure(4-1) the time increased as the emissions of (CO2)is decreased , until reached amid (56° C ATDC), and started to increased time as the (CO2)emission increased, such as:

At speed 750rpm

Table(4.2): shows the change timing valve and (CO2)emission

ATDC	Emission (Co2) (g\Kwh)
50°	882.73
53°	869.11
56°	868.89
59°	868.95
62°	873.34

4.4 Effect of engine speed and fraction of wet NOx

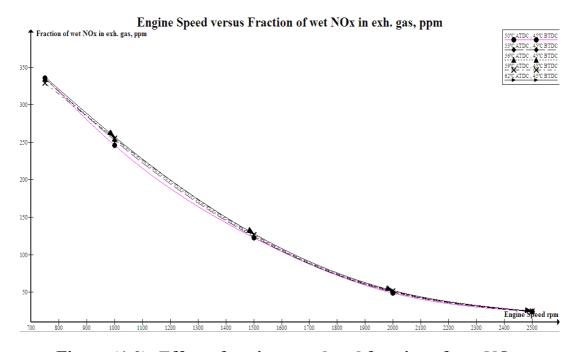


Figure (4-2): Effect of engine speed and fraction of wet NOx

As Shown in the figure(4-2) the emissions of (NOx)is decreased as the engine speed increased, until reached a mid speed (1500rpm), and started to decreased as the engine speed increased, such as:

: At system: (50°C ATDC, 45°C BTDC):

At 750rpm, (NOx)emission is found to be 335.62ppm, and at 1000rpm, (NOx) emission is found to be 245.66ppm is in decreased, and at 1500rpm, (NOx)emission is found to be 122.78ppm(reached amid), and at 2000rpm, (NOx)emission is found to be48.932ppm, and at 2500rpm, (NOx)emission is found to be 23.838ppm is in decreased.

As Shown in the figure (4-2) the time increased as the emissions of (NOx) is decreased, until reached amid (56°C ATDC), and started to increased time as the (NOx)emission decreased, such as:

At speed 750rpm

Table(4.3): shows the change timing valve and (NOx)emission

ATDC	NOx (ppm)
50°	335.62
53°	334.70
56°	334.26
59°	329.21
62°	328.12

4.5 Effect of engine speed and Summary emission PM and NOx

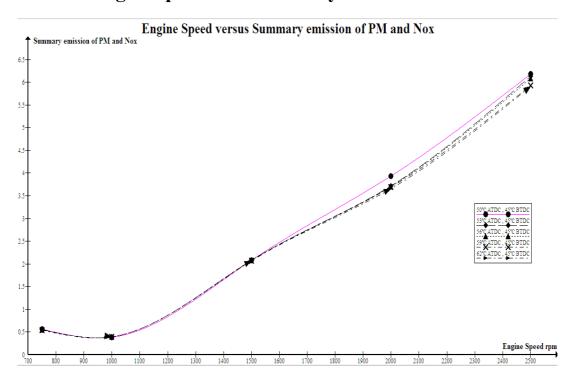


Figure (4-3): Effect of engine speed and Summary emission PM and NOx

As Shown in the figure(4-3) Summary emission of PM and NOx (S E) is increased as the engine speed increased, until reached a mid speed (1500rpm), and started to increased as the engine speed increased, such as:

At system (53°C ATDC, 45°C BTDC):

At 750rpm, Summary emission of PM and NOx (S E) is found to be 0.54919 ,and at 1000rpm, Summary emission of PM and NOx (S E) is found to be 0.39309 is in decreased, and at 1500 rpm, Summary emission of PM and NOx(S E) is found to be 2.0814(reached amid), and at 2000rpm, Summary emission of PM and NOx (S E) is found to be 3.7180, and at 2500rpm, Summary emission of PM and NOx (S E) is found to be 6.1368 Iis in increased.

As Shown in the figure(4-3) the time increased as the Summary emission of PM and NOx (S E) is decreased, until reached amid (56° C ATDC), and started to increased time as the Summary emission of PM and NOx (S E) decreased, such as:

At speed 750rpm

Table(4.4): shows the change timing valve and Summary emission

ATDC	SE
50°	0.55929
53°	0.54919
56°	0.54825
59°	0.54021
62°	0.54000

4.6 Effect of engine speed and engine brake power

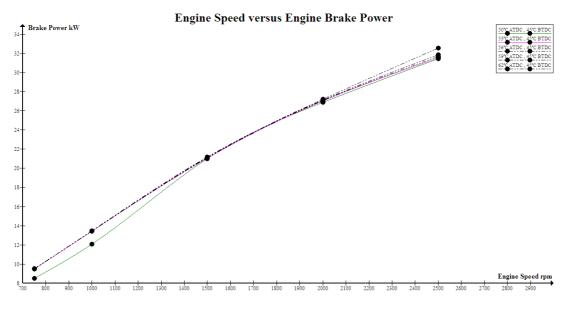


Figure (4-4): Effect of engine speed and engine brake power

As Shown in the figure (4-4) the engine Brake Power(BP)is increased as the engine speed increased, until reached a mid spead (1500rpm), and started to increased as the engine spead increased, such as:

At system (62°C ATDC, 45°C BTDC):

At 750rpm (BP) is found to be 9.4821kW, and at 1000rpm (BP) is found to be 13.483 kW is in increased, and at 1500rpm (BP) is found to be 21.180 kW (reached amid), and at 2000rpm (BP) is found to be27.206 kW, and at 2500rpm (BP) is found to be 32.546 kW is in increased.

As Shown in the figure (4-4) the time increased as the engine brake power is increased, until reached amid (56°C ATDC), and started to increased time as the brake power increased, such as:

At speed 750rpm

Table(4.5): shows the change timing valve and engine brake power

ATDC	BP (kW)
50°	9.4821
53°	9.5029
56°	9.5290
59°	9.5352
62°	9.5411

4.7 Effect of engine speed and specific fuel consumption

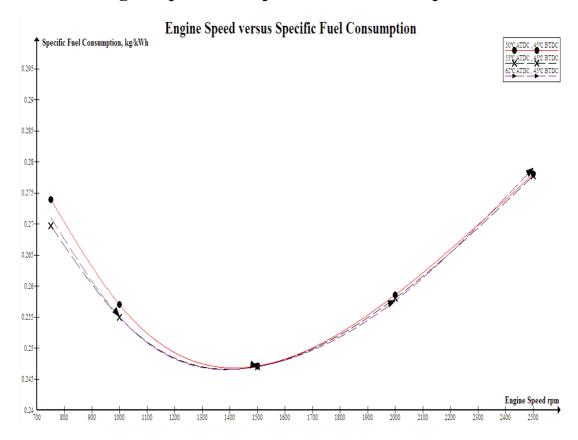


Figure (4-5): Effect of engine speed and specific fuel consumption

As Shown in the figure (4-5) the Specific Fuel Consumption (SFC) is decreased as the engine speed increased, until reached a mid speed (1500rpm), and started to increased as the engine speed increased, such as:

At system:(50°C ATDC, 45°C BTDC):

At 750rpm the Specific Fuel Consumption (SFC)is found to be 0.27395 kg\kWh, and at 1000rpm the Specific Fuel Consumptio(SFC)is found to be 0.25704Kg\kwh (decreased), and at 1500rpm the Specific Fuel Consumption (SFC)is found to be 0.24707kg\kWh(reached amid), and at 2000rpm the Specific Fuel Consumption (SFC) is found to be0.25859 kg\kWh, and at 2500rpm the Specific Fuel Consumption (SFC) is found to be 0.27800 kg\kWh is in increased.

As Shown in the figure(4-5) the time increased as the the Specific Fuel Consuption (SFC)is decreased, until reached amid (56°C ATDC), and started to increased time as the Specific Fuel Consumption (SFC) increased, such as:

At speed 750rpm

Table(4.6): shows the change timing valve and specific fuel consumption

ATDC	SFC
50°	0.27395
53°	0.26972
56°	0.26966
59°	0.26968
62°	0.27104

4.8 Effect of engine speed and efficiency of piston engine

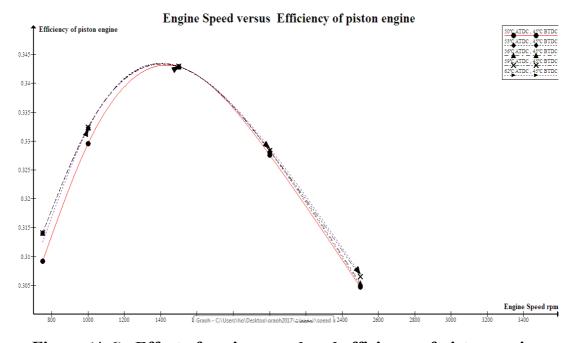


Figure (4-6): Effect of engine speed and efficiency of piston engine

As Shown in the figure(4-6) the Efficiency of piston engine (η) is increased as the engine speed increased , until reached a mid speed (1500rpm) , and started to decreased as the engine speed increased , such as:

At system:(50°C ATDC, 45°C BTDC):

At 750rpm Efficiency of piston engine (η) is found to be 0.30920, and at 1000rpm Efficiency of piston engine (η) is found to be 0.32955 is in increased, and at 1500rpm Efficiency of piston engine (η) is found to be 0.34284(reached amid), and at 2000rpm Efficiency of piston engine (η) is found to be 0.32756, and at 2500rpm Efficiency of piston engine (η) is found to be 0.30469 is in decreased.

As Shown in the figure(4-6) the time increased as the Efficiency of piston engine (η) is increased, until reached amid (56°C ATDC), and started to increased time as the Efficiency of piston engine (η) decreased, such as:

At speed 750rpm

Table(4.7): shows the change timing valve and efficiency of piston engine

ATDC	η
50°	0.30920
53°	0.31405
56°	0.31413
59°	0.31410
62°	0.31232

Chapter Five Conclusion & Recommendations

5.1 Conclusions

This study provides results the of spark ignition natural gas diesel engine (2.2 liters, 4-stroke-4-cylinder Daedong 4A220A-S1 engine).

Even though, the useful data are in the MBT timing. The data were intended to show versus the ignition timing between the range 15°BTDC and 54°BTDC.

This way of presentation can illustrate a primary overview, which leads to better understanding. Then, the MBT ignition-timing region becomes easier to be utilized.

The following conclusions have been reached:

Higher load comes with higher power, torque, volumetric efficiency, SFC · NOx and CO while brake thermal efficiency and CO2 are lower.

Engine speed affects the heat loss, friction loss and volumetric efficiency, which affect output torque. Engine speed limits the combustion time, which raises the exhaust gas temperature around 50°C for every increment of 500 rpm.

- 1. There is a little effect on engine performance when (EVC) angles reduced to less than 25° bTDC at low speeds.
- 2. There is a reduction in engine performance by de- creasing the (EVO) angle B BDC at all engine speeds and an increase of NOx emissions
- 3. The effect of (EVC) reduction is beneficial to Power, BMEP, and BSFC for different speed and also will reduce engine NO and CO emission down to 20° degrees.

5.1 Recommendations for Future Works

This experiment is relatively rough but it shows many effects. It shows quite clear effects of loads. Intake valve timings and speeds relate to each other.

On the other hand, the effects of injection timings seem to have the least effect among the parameters. It does not show the overview trend but it shows that injection timing has a specific effect in each valve timing, speed and/or load.

The future works can focus on camshaft no.1 and compare with other valve timings around 35° ABDC, which can be 25°, 30°, 40° and 45° ABDC. The load 25%, 50% and 100% should be enough for the investigation. Speed can move closer to get more detail. Especially, the injection timing should be very detailed. The experiment should go over the entire period of valve timing. The fuel should be injected from around 40° BTDC (before intake valve opens.) until the intake valve closure, with a small interval. The combustion analyzer should be brought to the data collection.

References

- 1. PTT Public Company Limited. Natural Gas Road Map. [Online] 2005. [cited 15 Dec. 2006].
- 2. Yusoff, A., Zailani, M. and Muthana, I. K. Valve Timing and Ignition Issues in fuel system for Compressed Natural Gas Direct Injection (CNGDI).

Faculty of Engineering, Universiti Kebangsaan Malaysia.

- 3. M. A. Kalam, et al. Power Improvement of a Modified Natural Gas Engine. Department of Mechanical Engineering, University of Malaya.
- 4. Effect of advanced injection timing on the performance of natural gas in diesel engines. India . Sadhana Vol. 25 (Feb. 2000) .
- 5. Michael K. A. et al. "First and Second Law Analyses of a Natural-Aspirated, Miller Cycle, SI Engine with Late Intake Valve Closure." SAE International Paper 980889. (1998).
- 6. Yorihiro F., et al. "Development of High Efficiency Miller Cycle Gas Engine." Mitsubishi Heavy Industry, Ltd. Technical Review. Vol. 38 No.3 (Oct. 2001).
- 7. M. D. Basset, et al. "A simple Two-State Late Intake Valve Closing Mechanism."

Proc. Instn. Mech. Engrs. Vol. 211 (1997).

8. S. Shiga, et al. "Effect of Over-Expansion Cycle in a Spark-Ignition Engine using

Late-Closing of Intake Valve and Its Thermodynamic Consideration of the Mechanism." International Journal of Automotive Technology. Vol. 2 No.1 (2001).

9. Chih Wu, et al. "Performance Analysis and Optimization of a Supercharged Miller Cycle Otto Engine." Applied Thermal Engineering. 23 (2003).

- 10. Gyeung H. C., et al. "An experimental and numerical study of a miller cycle for gas engine converted from a diesel engine.
- 11. Y. Wang. et al. "Experimental investigation of applying Miller cycle to reduce NOx emission from diesel engine." Proc. IMechE. Vol.219 (2005).
- 12. G.H. Abd Alla, et al. "Effects of injection timing on the performance of a dual fuel engine." Energy Conversion&Management. Vol. 43 (2002).
- 13. Takagaki S. "The effects of compression ratio on nitric oxide and hydrocarbon emissions from a spark-ignition natural gas fuelled engine."
- 14. Koichi H. et al. "A study of the improvement effect of Miller-cycle on mean effective pressure limit for high-pressure supercharged gasoline engines.
- 15. Jerald A. C. "The Use of a Three-Zone Combustion Model to Determine Nitric Oxide Emissions from a Homogeneous-Charge, Spark-Ignited Engine." 2003 Spring Technical Conference. (11-14 May 2003).
- 16. J. A. Caton. "Effects of the compression ratio on nitric oxide emissions for a spark ignition engine: results from a thermodynamic cycle simulation."
- Int. J. Engine Res.. Vol. 4 No. 4 (2003).
- 17. Tsukida et al. "Production Miller-Cycle Natural Gas Engine." Inter- Tech Energy Progress, Inc.. (1999) .
- 18. A. Al-Sarki. et al. "Efficiency of a Miller Engine." Applied Energy.
- 19. Willard, W. Engineering Fundamentals of the Internal Combustion Engine. 2nd ed. USA . Prentice Hall, c2004.
- 20. Richard, S. Introduction to Internal Combustion Engines. 3rd ed. Great Britain . SAE, c1999.
- 21. F. O. Edward, Internal Combustion Engine and Air Pollution. 1st ed. USA . HarperCollinsPublishers, c1973.
- 22. (SweetHaven, 1985).
- 23.www.diesel-rk.bmstu.ru