DESIGN AND PERFORMANCE ANALYSIS OF A SMALL TWO-STROKE DIRECT INJECTION DIESEL ENGINE

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DESIGN AND PERFORMANCE ANALYSIS OF A SMALL TWO-STROKE DIRECT INJECTION DIESEL ENGINE

BY

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DECLARATION

I hereby declare that the work reported in this thesis is the result of my own investigation and that no part of the thesis has been plagiarized from external sources. Materials taken from other sources are duly acknowledged by giving explicit references.

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REKABENTUK DAN ANALISIS PRESTASI ENJIN DIESEL DUA LEJANG KECIL DENGAN PEMANCITAN LANGSUNG

ABSTRAK

Memandangkan sumber minyak mentah mengalami kesusutan dengan kadar yang cepat, kecekapan penggunaan bahan api menjadi semakin penting bagi pengguna. Banyak negara membangun mempunyai bilangan motosikal yang banyak, jadi perbaikan penggunaan bahan api pada motosikal akan mengurangkan permintaan bahan api dengan kadar yang banyak di negara tersebut. Enjin diesel memberi kebaikan seperti perbaikan kecekapan terma, kehilangan dalam pengepaman yang lebih rendah, operasi dengan kandungan bahan api yang kurang dan geseran rendah yang disebabkan oleh operasi kelajuan rendah, tetapi mempunyai kelemahan besar iaitu berat tertentu yang tinggi dan ketumpatan kuasa yang rendah. Enjin dua-lejang mempunyai kelebihan seperti kuasa tinggi, ringan dan geseran rendah tetapi enjin dua-lejang pracampur mempunyai kelemahan utama seperti penggunaan bahan api dan bahan cemar yang tinggi yang disebabkan oleh kepintasan bahan api. Apabila menggabungkan kitar diesel dan kitar dua-lejang, kepintasan bahan api yang bermasalah akan dihapuskan melalui pemancitan bahan api secara langsung sementaranya ketumpatan kuasa diesel yang rendah telah ditingkatkan. Dalam penyelidikan ini, prestasi dari segi pengeluaran kuasa dan penggunaan bahan api sebuah enjin diesel dua-lejang yang kecil berkapasiti 118cc dengan pemancitan langsung telah dinilaikan. Pengujian dinamometer dan simulasi prototaip enjin telah dijalankan untuk tujuan ini. Prototaip enjin itu dibina dengan pengubahsuaian daripada sebuah enjin gasolin dua-lejang sedia-ada sementara simulasi telah dijalankan dengan menggunakan perisian pemodelan enjin satu dimensi. Pengaruh masa pemancitan, penalaan sistem ekzos, nisbah mampatan dan geometri kebuk

letupan ke atas prestasi enjin telah disiasat. Keputusan uji kaji menunjukkan enjin itu berkeupayaan untuk menggerakkan sebuah motosikal kecil kepada halaju 60km sejam dengan perbatuan 65km seliter sementara prestasi enjin sempurna yang disimulasikan oleh model enjin memberi halaju tertinggi sebanyak 70km sejam dengan perbatuan 77km seliter.

DESIGN AND PERFORMANCE ANALYSIS OF A SMALL TWO-STROKE DIRECT INJECTION DIESEL ENGINE

ABSTRACT

As oil resources are rapidly being depleted, vehicle fuel efficiency is becoming more important for consumers. Due to the large number of motorcycles, improving fuel consumption of motorcycles would greatly reduce fuel demand in many developing countries. Diesel engines offer the advantages of improved thermal efficiency, lower pumping losses, lean operation and potentially low friction from low speed operation with the major disadvantages being a high specific weight and low power density. Two-stroke engines have the advantage of high power, lightweight and low friction while premixed two-strokes have the major disadvantages of high fuel consumption and emissions from fuel short-circuiting. When combining the diesel cycle with two-stroke cycle, the problematic fuel short-circuiting is eliminated through direct fuel injection while the low power density of diesel cycle is improved. In this research, the performance in terms of power output and fuel consumption of a 118cc two-stroke direct injection diesel engine was evaluated. Dynamometer testing and simulation of a prototype engine were conducted for that purpose. The prototype engine was built by converting an existing two-stroke gasoline engine while simulation was done using a one dimensional engine modeling software. The influence of injection timing, exhaust system tuning, compression ratio and combustion chamber geometry on engine performance was investigated. Experimental results show that the engine is capable of propelling a small motorcycle to a cruising speed of 60km/h with fuel consumption of 65km/l, while the ideal performance of the engine which was simulated by the engine model gives maximum cruising speed of 70km/h with mileage of 77km/l.

CHAPTER 1 INTRODUCTION

1.0 Research background

It's very true that, nowadays the world is powered by carbon-based energy, especially crude oil. Despite recent high prices and supply sources which are plagued by political issues from time to time, the reliance on crude oil is still inevitable. Looking at the transportation sector specifically, the middle-term solution would be the hybrid vehicles and the long-term solution looking for the full electrical vehicles. However, the process of development and popularization of those technologies can take longer than desired. This challenge is even tougher for developing countries. Therefore, vehicles powered by internal combustion engines will stay around for the foreseeable future. Improving internal combustion engine vehicle fuel efficiency has become increasingly important to ease the demand of crude oil and reduce environmental impact.

Improving fuel economy can be achieved through a number of approach, which include reducing traffic congestion, alteration of driver's behavior toward eco-driving and advancement of public transportation. Refining the efficiency of the vehicle itself will have a more straight-forward effect and the efforts aforementioned will benefit from it as well. For developing countries, motorcycles occupy a major fraction of the personal transportation mix because they are less expensive on ownership and maintenance. As shown in Figure 1.1, the majority of vehicle in those Association of South East Asian (ASEAN) countries are motorcycle. The number of motorcycle in ASEAN countries is predicted to grow through the year of 2035. Any amount of fuel consumption reduction from that category will translate into a very large reduction in fuel demand. Therefore, this category of vehicle should be paid on serious attention.

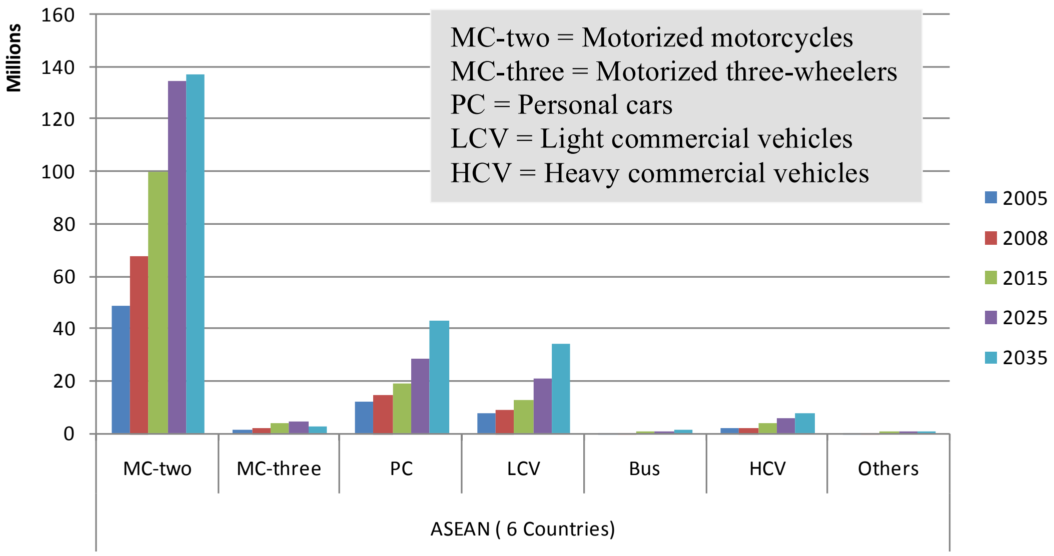


Figure 1.1: Growth of Motor Vehicles per Mode for 6 ASEAN countries: Indonesia, Malaysia, Philippines, Singapore, Thailand, Vietnam (Clean Air Initiative for Asian Cities 2010)

Malaysia is one of the countries that relies heavily on the usage of motorcycles. As shown in Figure 1.2, there is a lot of traffic infrastructure built specifically for motorcycles, which underlines the importance of the two-wheelers in both rural and urban areas. The majority of those motorcycles use four-stroke carbureted gasoline engines, with an engine capacity in the range of 100cc-125cc. Apart from that, there are still a significant number of two-stroke carbureted gasoline engine on the road, although they are slowly being phased out (J. H. Lee, Chong, et al. 2010).

Figure 1.2: Traffic infrastructure in Malaysia built specifically for motorcycles

The nature of a motorcycle limits the fuel economy improvement options compared to a car or heavy-duty vehicle. First, motorcycle's weight is usually already minimum, i.e. no significant weight reduction can be made. Second, its air drag coefficient can not be improved significantly, unless dramatic changes such as adding shell covering the whole motorcycle and riders, which doesn't seem acceptable in the short run. The most obvious option would be improving the motorcycle engine. There are many possibilities for this approach. For four-stroke engines, enforcement of more stringent emission standard has been pushing the adoption of electronic fuel injection systems to replace the carbureted fueling systems. While for two-stroke engine, usually engine conversion needs to be done in order to eliminate short circuiting of fuel. The two-stroke engine can be converted into gasoline direct-injection or gaseous fuel direct-injection. Those techniques do improve the engine efficiency substantially (Teoh 2010).

In this research, a completely different and bold move has been planned and investigated, i.e. adopting a small two-stroke diesel engine for motorcycles. For internal combustion engines, the compression ratio is one of the key parameter that influences its efficiency. Potentially the diesel engine, running at higher compression ratios than gasoline engines, can achieve a better cycle efficiency than the gasoline engine. The combustion efficiency of diesel engines are also higher than the gasoline engine. Being direct fuel injection engine, diesel engine has no unburned air-fuel mixture that trapped inside crevice volume of combustion chamber (the same applied to gasoline or gaseous direct injection engine), which could not be burned during combustion (Kim et al. 1999). In addition, the lean operation of diesel engine gives excess air in fuel-air mixture, thus increases the probability of complete fuel burn. Apart from that, the diesel engine also operates without intake air throttle, which significantly reduces the pumping loss at part load condition compared to carbureted or port fuel injection gasoline engine. Another key difference between diesel and gasoline engine is their operating speeds. Being an engine that runs with homogenous combustion, gasoline engine speed is only constrained by the flame speed or the engine parts' physical limitation at high speed, such as piston stresses, valve size that choke airflow or performance of valve train. That means gasoline engine can be operated at very high engine speeds, at which the mechanical friction gets higher. For a diesel engine, its engine speed is limited by the fuel-air mixing rate, which causes the maximum operating speed to be much lower than gasoline engines. This can also be a disadvantage, as this will lower the power output from a diesel engine. From the point of view of fuel economy, the lower operating engine speed of diesel engine means lower mechanical friction, thus better engine efficiency.

Although the diesel engine has some distinct advantages in term of fuel efficiency, it barely exists in the motorcycle arena, especially in the commuter motorcycle category. One of the major drawbacks preventing the adoption of diesel engine in motorcycle is its power density. Taking comparison of a Yanmar L40AE four-stroke diesel engine with 199cc engine displacement and a Honda Wave 100 with 97.1cc engine displacement, the maximum power for those engines are 3.2kW and 6.6kW respectively. That means the diesel engine power density is a mere quarter of the gasoline engine. In addition, the diesel engine is also generally heavier than its gasoline counterpart because of its tougher parts required and larger flywheel. For large machinery or transport, the low power density of diesel engine is compensated by using larger engine and engine boosting technology (turbocharging or supercharging). The engine weight is a relatively smaller portion of the overall weight of those heavy machines, thus the added weight of larger engine doesn't become a deal breaker. The same doesn't apply to a small motorcycle.

Therefore, the power capacity is the first barrier to overcome for adoption of small displacement diesel engine in a motorcycle. When thinking of fundamentally increasing power output from an engine, the two-stroke engine is an obvious contender. A Honda Icon 108cc four-stroke engine produces its maximum power of 6.1kW at 8000rpm. For comparison, a Modenas Dinamik 118cc two-stroke engine produces 11.8kW at the same speed. Even though the Dinamik has slightly larger engine displacement, the higher power output is mostly contributed from its one power stroke in every crank revolution, as opposed to the Icon's one power stroke every two revolutions. In addition, the lack of valve train in the Dinamik also reduces the mechanical friction. In order to estimate the power output from a small displacement two-stroke diesel engine, an unpublished parametric study of power output of different type of engine was used (Ismail 2010). Specific power of 50 four-stroke gasoline engines, 21 four-stroke diesel engines and 21 two-stroke gasoline engines was gathered. The data is summarized in Table 1.1.

Table 1.1: Specific power of different type of engine (Ismail 2010)

	Gasoline		Diesel
Two-stroke	94.6kW/l	x0.29 →	≈27kW/l
	x0.58 ↓		↑ ÷0.58
Four-stroke	55.2kW/l	← ÷0.29	15.8kW/l

→ Conversion factor of specific power between different types of engines

Clearly, a small naturally aspirated diesel engine will have much lower power output compared to its gasoline counterpart. By using the estimated specific power of 27kW/l for two-stroke diesel engine, a 118cc engine displacement will give power output around 3.2kW. In practice, typical usage of a middle range commuter motorcycle excluding high speed and aggressive operation, a large engine power overhead might not be needed. With fuel efficiency in mind, the diesel engine can be a feasible solution with worthy compromise in power performance. Therefore, this research has been initiated in order to evaluate the potential and performance of a small two-stroke direct injection diesel engine for motorcycles.

1.1 Problems statement

Power density of a diesel engine is generally lower than a gasoline engine if both engines are naturally aspirated. This factor causes the power output of a small displacement diesel engines to be rather limited. Therefore, its potential to be adopted as a motorcycle engine is a challenge and needs further investigation. As a newly developed engine, most of the design and operating parameters have to be optimized through simulations or experiments in order to achieve good engine performance.

1.2 Objectives

The objectives of this research are as follow:

i. To design and convert a small two-stroke carbureted gasoline engine into two-stroke direct injection diesel engine.
ii. To study the performance of a small two-stroke direct injection diesel engine at different operating parameters.
iii. To develop an engine model to predict and improve the performance of a small two-stroke direct injection diesel engine.

1.3 Scope of research

The scope of this research is as follow:

i. Engine test cell design, fabrication and setup. This includes the engine dynamometer and load bank, data acquisition system and data post processing.
ii. Engine model development for gasoline engine and diesel engine that were used in this research using Ricardo WAVE one dimensional engine and gas dynamics modeling software. Parts of the necessary parameters were measured experimentally, while some of the parameters such as flow coefficient, temperature and combustion parameters were estimated or using default values provided by the software.

iii. Convert a two-stroke carbureted gasoline engine into two-stroke direct injection diesel engine. The conversion includes design and fabricate the flexible fuel system, engine head for direct injection, high compression pistons with different piston bowl shapes and an additional flywheel for engine operation smoothing.

iv. Engine performance testing for both gasoline and diesel engine. The performance criteria were limited to engine power and fuel consumption. Emissions were excluded from the evaluation at this early stage of development, but it is definitely worth in-depth study for the next stage of development.

v. Engine model results validation by comparison to initial engine testing result. Without actual measurement of every engine detail as aforementioned, the minimum correlation of 10% was targeted.

vi. Diesel engine performance improvement and characterization at various operating parameters. The engine performance was evaluated with the original and a tuned exhaust system, advanced and retarded fuel injection timing with constant amount of fuel injected per stroke, higher and lower compression ratio and with different piston bowl shape.

vii. Assessment of small two-stroke direct injection diesel engine as a motorcycle engine, specifically in terms of brake power and fuel consumption. While others criteria such as emission, drivability, noise and cost are necessary considerations, ensuring that the engine power capacity is sufficient for driving a motorcycle would be the first priority. As for fuel consumption, it is the sole motivation to initiate this research.

CHAPTER 2 LITERATURE REVIEW

2.0 Overview

The internal combustion engine works upon very simple principles. However, there are a lot of aspects that require sophisticated controls and implementation in order to achieve the highest engine efficiency. Those aspects will be reviewed from the point of view of a compression ignited diesel engine and a two-stroke engine, as well as a small two-stroke diesel engine.

2.1 Internal combustion engine

The internal combustion engine is an energy conversion device, which extracts mechanical power from the heat energy of fuel oxidation. Generally, the working principle of an internal combustion engine is relatively simple. Air will be drawn into the engine combustion chamber and used as the working fluid of the system. Then the air inlet of the combustion chamber will be closed to form an enclosed volume inside the engine. Fuel will be combusted inside that enclosed combustion chamber for heat addition to the air. With the added heat energy, pressure of the air in the combustion chamber will be elevated, thus exerting force directly on the moving component of the engine to produce mechanical work. Finally, the burned products will be expelled from the combustion chamber to leave room for induction of fresh air and ready for the next cycle. In reality, those processes of course are more complicated and implemented in many different ways, but they hold true for every internal combustion engine. In this report, the term internal combustion engine will be specifically referring to a reciprocating piston engine. Non-conventional internal combustion engine such as rotary engine and jet engine are excluded from any discussion and review, even though they are technically in the same category in a more general scope.

Fundamentally, internal combustion engines can be categorized based on the method of fuel ignition, namely the spark ignition (SI) and compression ignition (CI) engine. The spark ignition engine was developed by Otto in 1876 and the compression ignition engine was invented by Diesel in 1892. The engine can be further categorized based on the operating cycle, which are commonly the two-stroke or four-stroke cycle engine.

2.2 Compression ignition engine

Aside from the differences in engine structure, the way of controlling engine load, engine operating speed or the type of fuel being used, the most substantial factor that separate spark ignition engines and compression ignition engines is the combustion process. While the energy is stored in the fuel as chemical energy, the engine is extracting energy from the heat energy that is released from the fuel. This clearly shows the importance of the combustion process for a heat engine. Compression ignition engines heat addition process is achieved through auto-ignition of fuel that is injected into the hot compressed air inside combustion chamber. In order to attain effective heat release process, the fuel must be mixed well with the air and provided suitable condition to form combustible fuel-air mixture, before it can be oxidized. Four major factors are involved in that process:

i. In-cylinder air condition
ii. Fuel injection
iii. Combustion chamber design
iv. In-cylinder air motion

2.2.1 In-cylinder air condition

The first in-cylinder air condition that is needed to initiate compression ignition is the high air temperature. This usually is not a problem after the engine has been started and running since the air temperature will be easily exceed the fuel auto-ignition temperature. However, starting a compression ignition engine could become a problem. Therefore, the compression ratio of the engine must be high enough to attain auto-ignition temperature of the fuel. The auto-ignition temperature of petroleum diesel is about 235ºC (Xingcai 2004). In low temperatures, a cold starting assist system such as glow plug is required to heat the air and raise the final compression temperature. Recent development has enable fast heating of the air, improving the preheat time from about 20 seconds to below 5 seconds, as shown in Figure 2.1 (Mollenhauer & Tschoeke 2010).

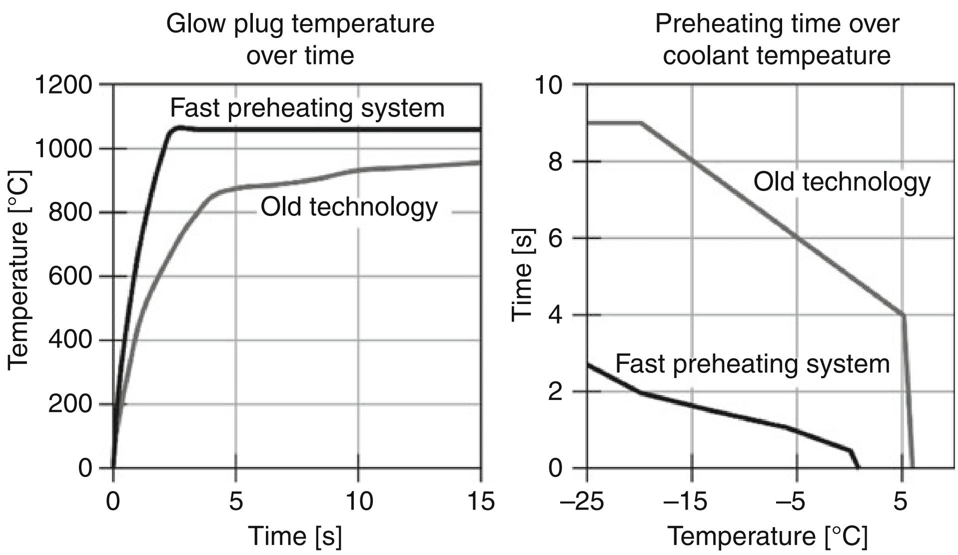


Figure 2.1: Improved performance of modern glow plug (Mollenhauer & Tschoeke 2010)

The in-cylinder air also has to be in high pressure, which is important for the fuel dispersing. The air density inside the combustion chamber of a diesel engine during compression when the fuel is being injected is typically in the range of 20–60 kg/m^3 (Benajes et al. 2005). Higher air density from higher air pressure will give larger fuel spray cone angle but shorter spray penetration (Shao et al. 2003; C. Bae & Kang 2000; Desantes et al. 2006; Shao et al. 2008). As shown in Figure 2.2, the higher air density produce larger fuel spray cone angle, which disperses the fuel into large volume of air and improves the air utilization (Taylor 1985a).

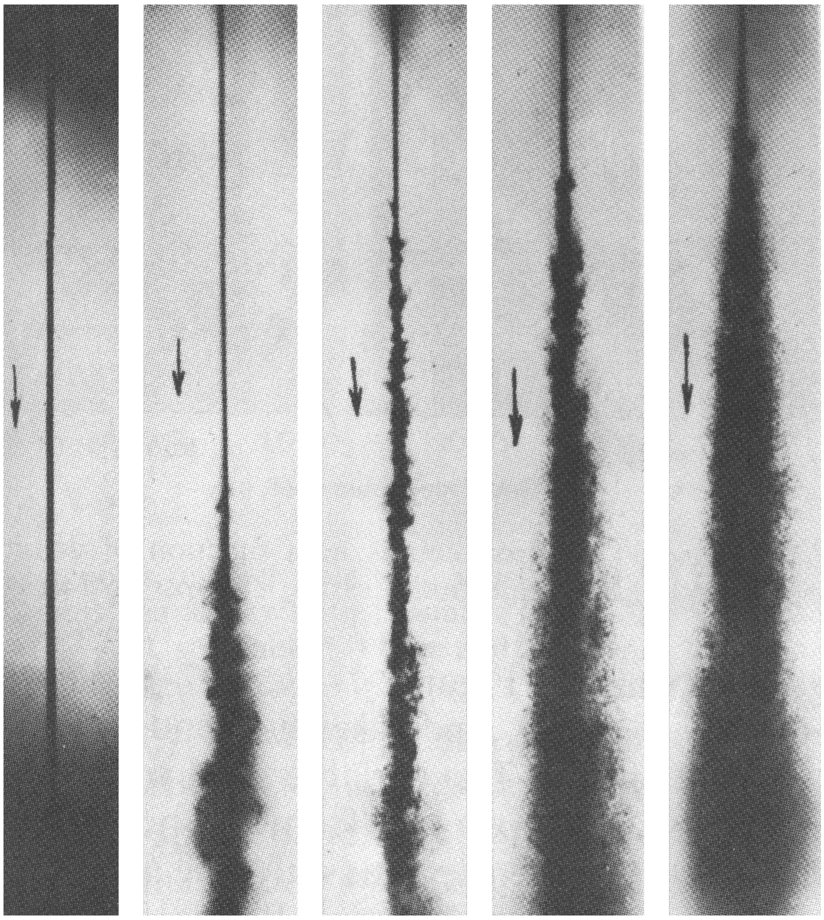


Figure 2.2: Fuel sprays at various air densities, from left to right 0.0013atm, 1.0atm, 4.4atm, 7.8atm and 14.5atm (Taylor 1985a)

The compression ratio is the ratio of maximum cylinder volume to minimum cylinder volume, where maximum cylinder volume is total of the engine clearance volume and swept volume while the minimum volume is the engine clearance volume. The following equation defines the relationship (Heywood 1988):

$$r_c = \frac{V_d + V_c}{V_c} \tag{2.1}$$

where r_c is the compression ratio, V_d is engine displacement volume and V_c is engine clearance volume.

For two-stroke engines, sometimes the term trapped compression ratio is used (Blair 1996). For trapped compression ratio, the trapped volume, i.e. engine displacement volume after exhaust port is closed, is used in the equation 2.1 instead of the engine displacement.

A high compression ratio is not only needed for fuel auto-ignition, but it also contributes to the higher thermal efficiency of compression ignition engine over spark ignition engine. The spark ignition engine is usually described as an ideal gas constant-volume combustion while the Diesel cycle is described as a limited-pressure combustion. The cycle efficiency of constant-volume cycle and limited-pressure cycle are shown in equation 2.2 and equation 2.3 respectively (Taylor 1985b).

$$\eta = 1 - \frac{1}{r_c^{k-1}} \tag{2.2}$$

$$\eta = 1 - \frac{1}{r_c^{k-1}} \left[\frac{\alpha\beta^k - 1}{(\alpha - 1) + k\alpha(\beta - 1)} \right] \tag{2.3}$$

where η is the cycle efficiency, k is the heat capacity ratio, r_c is the compression ratio, β is the cutoff ratio and α is the pressure ratio during constant-volume heat addition.

The theoretical cycle efficiency for both cycles are plotted in Figure 2.3. Even though the Otto cycle is more efficient than the Diesel cycle for a given compression ratio, in practice the compression ignition engines are running at much higher compression ratios. Since there is only air during the compression stroke of compression ignition engines, there is no knock limit as in a spark ignition engine. Assuming the compression ignition engine has compression ratio 18 and the spark ignition engine compression ratio is 9, the theoretical cycle efficiency would be 62% and 54% respectively as shown in Figure 2.3. Therefore, the diesel cycle can achieved higher efficiency than gasoline engines.

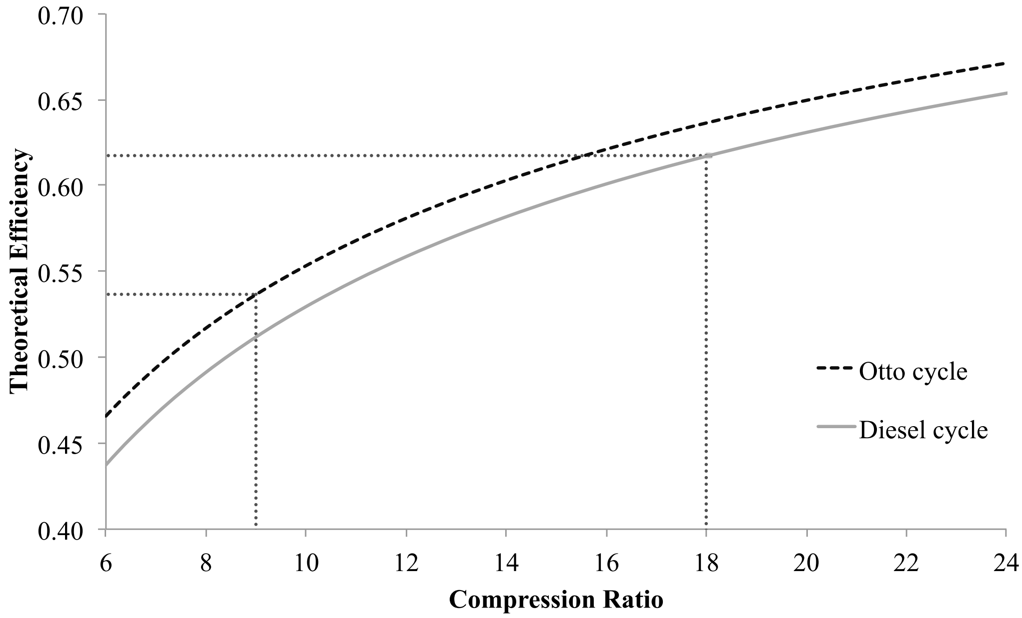


Figure 2.3: Theoretical cycle efficiency of Otto cycle and Diesel cycle as a function of compression ratio, assuming k=1.35 for both case and β=1.5, α=1.5 for diesel cycle

In addition, with higher compression ratio, which gives higher compression pressure, the ignition delay can be decreased significantly (Heywood 1988). However, the compression ignition engine still has its limit on increasing compression ratio. By increasing compression ratio, the peak cylinder pressure will increase as well. Therefore, the physical strength of engine components is one of the major constraints for higher compression ratio. Besides that, compression work, blow-by and heat loss will increase as well. All of that will compensate the benefit of increased compression ratio.

2.2.2 Fuel injection

The fuel injection system is the most significant factor influencing combustion characteristics of a high speed direct injection diesel engine, apart from the engine structural design and combustion chamber properties (Raffelsberger et al. 1995). The in-line fuel injection pump is the most widely used diesel fuel injection technology (Robert Bosch 2005). The fuel injection system is commonly known as pump-line-nozzle (PLN) injection systems, which the name itself describes the most basic components of the system. There is more recent and advanced development of others technologies, such as unit injector, which has shown the ability to reach higher injection pressures (Ichihashi et al. 1992). That study also showed that the newer technologies requires smaller drive system and are more efficient in comparison with pump-line-nozzle systems. However, the fully mechanical system is simpler, having rugged durability and easy of maintenance keeping it relevant nowadays, especially for small utility engine (Robert Bosch 2005).

For a pump-line-nozzle injection system, the fuel is pressurized by a fuel injection pump consisting of a barrel and plunger assembly. The plunger is driven by a cam, which the profile has to be designed to pressurize the fuel synchronous with the crankshaft rotation. The plunger can be rotated so that the time the helical groove on outer surface of the plunger, which uncovers the inlet, can be altered either earlier or later (see Figure 2.4). With that, the effective stroke (from start of fuel pressurization to the inlet uncovering) can be manipulated, so that the amount of fuel injected can be directly controlled according to engine load. With this system, the start of injection will be the same regardless of the engine load. Only the end of injection is affected, therefore a longer effective stroke will give longer injection duration.

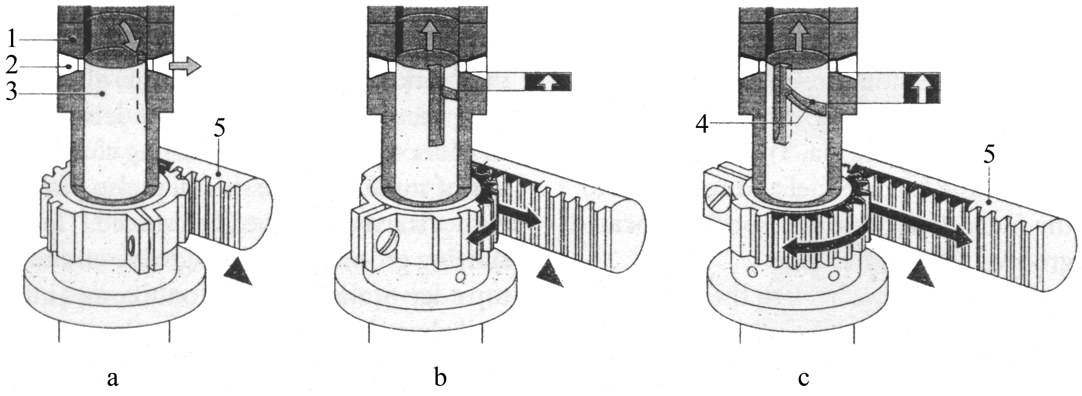


Figure 2.4: Fuel-delivery control of PLN injection system: a) zero delivery, b) partial delivery and c) maximum delivery (Robert Bosch 2005)

The pressurized fuel is then delivered to the fuel injector via high-pressure fuel line. The fuel line must be rigid and kept as short as possible, so that the fuel pressure loss is minimized. With the same fuel pressure from the injection pump, the start of injection depends on the injector needle opening force. As the pump stops pressurizing the fuel and fuel pressure drops below the injector needle closing force, the fuel injection will end. This of course is an idealized description of the fuel injection. In reality, hazardous phenomena such as cavitation, unintentional secondary injection and fuel dribble occur which will affect the combustion process (Benavides et al. 2000).

The fuel injection system has a direct influence on the fuel spray characteristics, which will subsequently affect the combustion process. Some important characteristics of the fuel spray include spray tip penetration, spray cone angle and overall spray Sauter Mean Diameter (SMD) (C. Chang & Farrell 1997). Another spray parameter that is also considered significant is the fuel momentum flux (F. Payri et al. 2004; Desantes et al. 2003). The momentum flux describes the fuel velocity at the injector nozzle's outlet and is related to fuel density and effective diameter of the injector nozzle. Study has shown that when using an alternative fuel such as biodiesel or blend of biodiesel with petroleum diesel, the difference in engine performance is mainly caused by the deviation in fuel injection characteristic, which is a result of different physical fuel properties such as bulk modulus and viscosity (C. S. Lee et al. 2005). Therefore, the fuel injection parameters have to be altered accordingly to optimize the engine performance.

The fuel injection system is responsible to provide the required fuel pressure, timing control and injection rate metering. Higher injection pressure gives finer fuel atomization, which will yield better fuel vaporization and mixing with air subsequently. In addition, the high fuel pressure is also needed for fuel penetration into the highly compressed air in order to penetrate the fuel across the combustion chamber to make the fuel air mixture as homogeneous as possible. However, the fuel penetration should be optimized for the specific combustion chamber and swirl level, since over penetration of fuel will cause wall wetting that reduces the fuel vaporization, while too little fuel penetration will cause the fuel concentrated around the injector giving poor fuel distribution and low air utilization. The injection timing control is essentially determining the start of combustion, thus the phasing of the combustion pressure. Therefore, the start of injection timing must be accurately controlled to within 1^{o} of crankshaft rotation (Robert Bosch 2005). While the fuel injection continues, the injection rate and end of injection, thus the amount of fuel injected, will influence the subsequent combustion process.

With the fuel momentum and entrainment of high turbulence inside the combustion chamber the fuel droplets will atomize into finer sizes, then vaporize by the heated air. The vaporized fuel is easier to mix with air and consequently forms combustible fuel-air mixture. After the ignition delay, the combustible fuel-air mixture will ignite and further elevate the air pressure and temperature. As the fuel continues to be injected and mixed with the remaining fresh air, combustion persists until the end of fuel injection or lack of fresh air for further fuel burning. In practice, the injection duration at full load operation is best in the range of 20º to 40º crank angle (Taylor 1985a).

Aside from granting better combustion which is beneficial to engine efficiency, the fuel injection, especially the fuel spray characteristic also have a strong relation to pollutant formation as shown in Figure 2.5. The major pollutants from a diesel engine are nitrogen oxide (NO_x), soot and unburned hydrocarbon (HC). Nitrogen (with 75% of mass composition of air) will be oxidized at very high temperatures (about 2200K) and lean fuel-air mixtures (Akihama et al. 2001; Poonawala 2007). Therefore, its formation area is around the boundary of fuel-air mixing, where the combustion is happening at the highest local temperature. For soot, it is produced in the region with high temperature but lacking oxygen for oxidation, i.e. at the liquid phase of the fuel spray core. The emission of hydrocarbon originates in regions with low temperature where the combustion does not commence. Those regions usually located far from the fuel spray or being some crevice volumes, which have been cooled by the nearby engine components such as piston, cylinder or engine head.

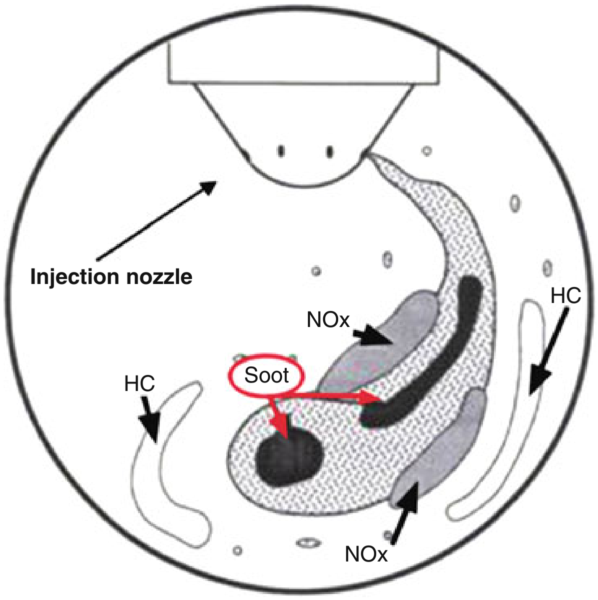


Figure 2.5: Regions of pollutant production in a combustion chamber with a heterogeneous mixture (Mollenhauer & Tschoeke 2010)

In addition, the fuel injection timing is also well known to have strong influence on the formation of soot and nitrogen oxide. The typical emission trend of soot and nitrogen oxide as a function of injection timing is shown in Figure 2.6. Advanced injection timing will give higher maximum in-cylinder temperature, which will cause higher nitrogen oxidation rate. But higher temperature is beneficial to fuel vaporization, which will decrease the emission of soot. The opposite will happen when the injection timing is retarded. The understanding of this phenomena led to the development of better fuel injection strategy such as split injection which can reduce both emission of nitrogen oxide and soot simultaneously (Yehliu et al. 2010; Su et al. 1995). In addition, the split injection is also capable of reduce the combustion noise of a diesel engine by reducing the maximum rate of pressure rise (Mollenhauer & Tschoeke 2010).

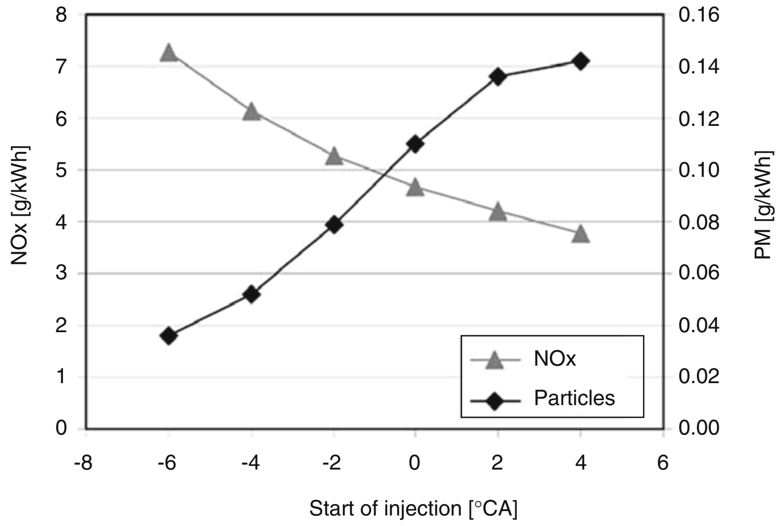


Figure 2.6: Influence of the start of injection on particle matter and nitrogen oxide emissions of a commercial vehicle engine at 1425 rpm and mean load (Mollenhauer & Tschoeke 2010)

2.2.3 In-cylinder air motion

Complex air motion occurs along the engine's air path during the induction of fresh air into engine cylinder and ejection of burned product from the cylinder. The in-cylinder air motion is strongly related to the induced air motion by the intake system or the combustion chamber design. There are two major bulk air motions inside the engine cylinder, which are swirl and squish as shown in Figure 2.7 (Pulkrabek 2004).

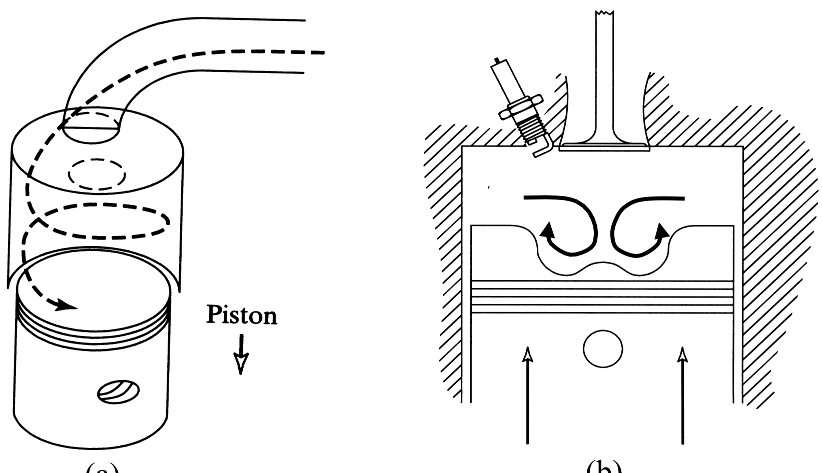


Figure 2.7: Schematic diagram of swirl (a) and squish air motion inside engine cylinder (b) (Pulkrabek 2004)

The swirl can be generated through several methods, such as guiding the intake air to enter the engine cylinder from tangential direction or contoured intake valve (Figure 2.8), resulting in rotation of the air while the piston is moving downward during intake stroke. While the intensity of the swirl motion will decay as the piston moving upward, the angular velocity of swirl usually will be increased when approaching top dead center, on the order of up to five times (Pulkrabek 2004). This is achieved with bowl-in-piston combustion chamber, which reduces the rotating radius of the flow, from the size of engine bore to the diameter of piston bowl (Heywood 1988). Due to conservation of angular momentum, the angular velocity of the swirl will be greatly increased. On the other hand, the squish is generated when the piston is approaching top dead center during compression stroke, forcing the air at the squish area rushing into the piston bowl.

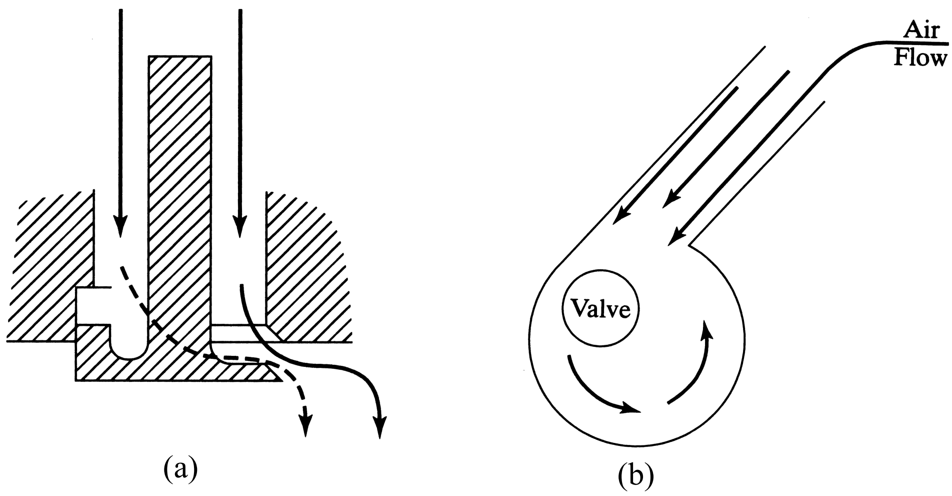
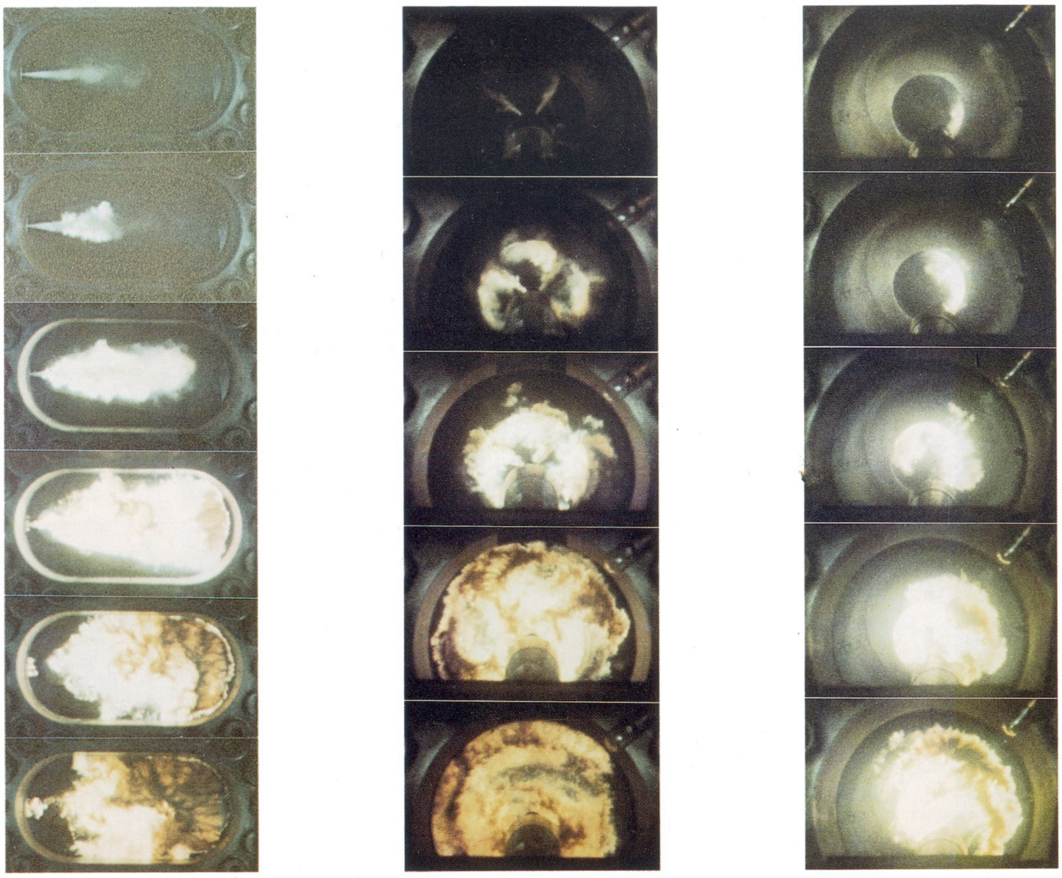


Figure 2.8: (a) Generating swirl using contoured valve or (b) intake path that guiding air entering cylinder tangentially

The swirl motion has great influence on the combustion process, which will affect engine performance significantly, including the engine emissions and fuel consumption. The effect of swirl on the diesel engine fuel distribution, mixing, as well as the combustion flame propagation is shown in Figure 2.9. It can be noticed that the fuel spray and the flame propagation have been deflected and spreading around the piston bowl by the counterclockwise swirl.



(a) (b) (c)

Figure 2.9: (a) Combustion of single spray burning under large direct injection engine; (b) combustion of four sprays in direct injection engine with counterclockwise swirl; (c) combustion of single spray in M.A.N. “M” direct injection diesel (Heywood 1988)

As shown in Figure 2.10 (Pulkrabek 2004), the increase in swirl ratio has reduced the smoke level of the engine, which is due to more homogenous fuel-air mixing. Swirl motion of air can distribute the fuel spray across the engine cylinder, thus improving the air utilization. From the better fuel-air mixing, faster combustion is expected, which gives a higher combustion temperature. This increase of combustion temperature causes higher emission level of nitrogen oxide. However, the formation of nitrogen oxide can be reduced using higher exhaust gas recirculation (EGR) together with higher fuel injection pressure to reduce the added soot emission caused by high EGR (Henein et al. 2001). For the specific fuel consumption, there is an optimum swirl ratio that gives the best result. This condition is caused by higher swirl ratio will increase the heat loss during the combustion and also exerts pumping loss when generating the swirl (Bergstrand & Denbratt 2002). Therefore, while having beneficial effect on the combustion process through improving the fuel-air mixing and combustion rate, a higher swirl level doesn't always give better engine performance. The swirl level has to be optimized in conjunction with other engine parameters such as combustion chamber design and fuel injection.

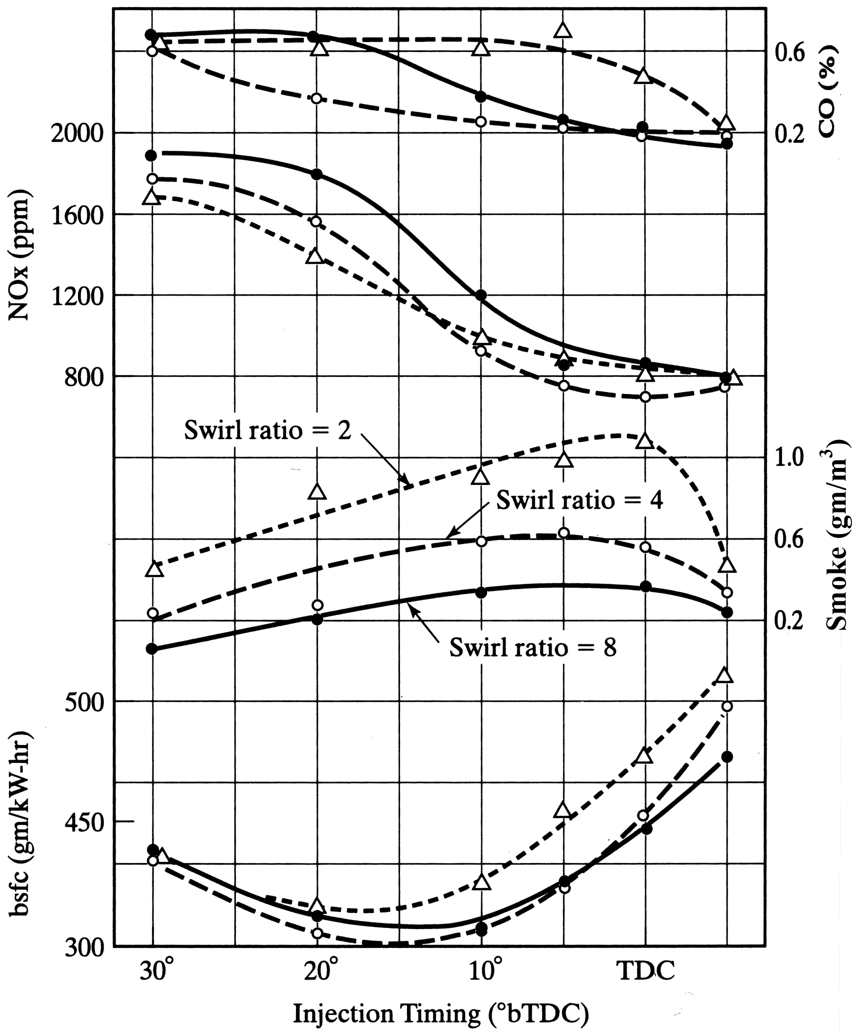


Figure 2.10: Brake specific fuel consumption and emissions level as a function of swirl ratio and injection timing of a single cylinder compression ignition engine (Pulkrabek 2004)

The aid of swirl in the combustion process can be reflected in the combustion pressure trace. As shown in Figure 2.11, the rate of pressure rise is increasing with the swirl ratio, which consequently gives higher maximum combustion pressure. The higher pressure rise is the results of faster combustion. By having heat release shorter and nearer to top dead center, the cycle efficiency will be improved, i.e. greater effective expansion and more work extraction from the heat.

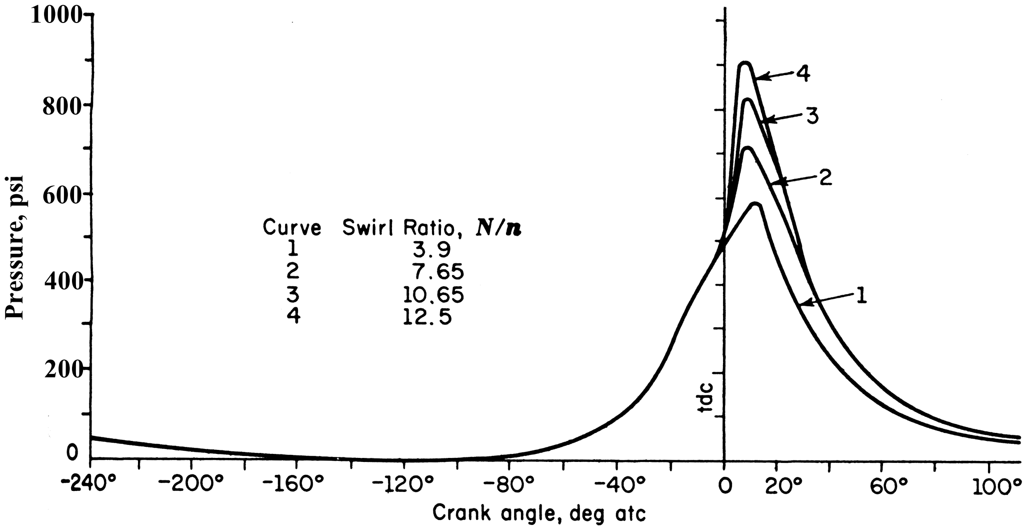


Figure 2.11: Influence of swirl on combustion pressure, N=air swirl, n=engine speed (Taylor 1985a)

Just as with swirl, squish motion is also found to contribute in improving the combustion process. In the effort to reduce the emission of nitrogen oxide and soot simultaneously, two-stage combustion was investigated (Kidoguchi et al. 1999). In that combustion concept, the first stage of combustion is a fuel rich combustion to reduce the emission of nitrogen oxide, while the later stage of combustion requires high turbulence combustion to reduce soot emission. The high turbulence combustion, which results from a high squish combustion chamber, enables flame entrainment into the rich region of the fuel-air mixture, therefore the fuel air mixing of that soot formation area can be improved and subsequently achieves low emission