VALVE TIMING STUDY OF A SINGLE CYLINDER MOTORCYCLE ENGINE

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ABSTRACT

In an internal combustion engine, valve timing is an important design parameter which affects many engine performance parameters. In this study, the effect of intake timing of an engine was investigated. The engine used in this study combines a 4-stroke engine bottom end with an opposed piston in the cylinder head working at half the cyclical rate of the bottom piston. Functionally, the second piston replaces the valve mechanism of the original engine that use poppet valve to control intake and exhaust port opening and closing. For the analysis, Computational Fluid Dynamic (CFD) software has been used to analyze in-cylinder air flow motion during intake stroke process with engine speed of 4000 rpm. The intake port of the engine was modified to vary the intake timing. The modification of intake port was done by using Computer Aided Design (CAD) software, Solidwork. From the CFD analysis, the in-cylinder air flow pattern and flow distribution before and after intake port modification was clarified. Simulation result shows that as the diameter of the port is decreased, the pressure drop and velocity of air flow into the engine cylinder are increased. Modification of the intake port shape from curved port to straight port was result in more symmetrical in-cylinder air flow distribution along the cylinder axis. For further study, it is strongly recommended to verify the simulation result with the experiment result as soon as the engine was successfully fabricated.

ABSTRAK

Dalam sebuah enjin pembakaran dalam, pemasaan injap adalah parameter penting yang mempengaruhi banyak parameter prestasi engine. Dalam kajian ini, kesan daripada masa kemasukan sebuah enjin telah disiasat. Enjin yang digunakan dalam kajian ini menggabungkan enjin 4 lejang di bawah dengan piston yang bertentangan di kepala silinder yag beroperasi separuh daripada kitaran piston di bawah. Secara praktikalnya, piston kedua itu menggantikan sistem injap enjin asal yang menggunakan injap popet untuk mengawal pembukaan dan penutupan liang kemasukan dan liang pembuangan. Untuk penganalisaan, perisian perkomputeran bendalir dinamik (CFD) telah digunakan untuk menganalisis aliran udara ke dalam silinder ketika proses lejang kemasukan dengan kelajuan enjin adalah 4000 rpm. Liang kemasukan enjin ini telah diperbaiki untuk mempelbagaikan masa kemasukan. Pembaikkan liang kemasukan telah dilakukan dengan menggunakan perisian rekaan dibantu komputer (CAD), Solidwork. Daripada analisa CFD, corak aliran udara dan penyebaran udara ke dalam silinder enjin sebelum dan selepas pembaikkan liang kemasukan telah diperhatikan. Keputusan simulasi menunjukkan bahawa apabila diameter liang kemasukan dikurangkan, perbezaan tekanan dan halaju aliran dalam udara di dalam silinder enjin bertambah. Pembaikkan bentuk liang kemasukan daripada liang melengkung kepada liang lurus menyebabkan penyebaran udara ke dalam silinder yang lebih sekata sepanjang paksi silinder. Untuk kajian selanjutnya, amatlah disarankan untuk mengesahkan keputusan simulasi dengan keputusan eksperimen sebaik sahaja enjin ini berjaya dihasilkan.

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LIST OF SYMBOLS

D	Diameter
K	Kelvin
kg/s	Kilogram per second
kPa	Kilopascal
mm	Millimetre
m/s	Meter per second
N	Engine speed
P	Pressure
Pr	Prandtl number
R	Standard universal gas constant
T	Temperature
ρ	Density

LIST OF ABBREVIATIONS

ABDC	After bottom dead centre
ATDC	After top dead centre
BBDC	Before bottom dead centre
BDC	Bottom dead centre
BTDC	Before top dead centre
CA	Crank angle
CAD	Computer aided design
CFD	Computational fluid dynamic
EGR	Exhaust gas recirculation
IVC	Intake valve closing
IVO	Intake valve opening
E_xPC	Exhaust port closing
E_xPO	Exhaust port open
EVC	Exhaust valve closing
EVO	Exhaust valve opening
IPC	Intake port closing
IPO	Intake port opening
IVC	Intake valve closing
IVO	Intake valve opening
rpm	Revolution per minute
TDC	Top dead centre

CHAPTER 1

INTRODUCTION

1.1 PROJECT BACKGROUND

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. 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 six 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.

1.2 PROBLEM STATEMENT

Valve timing is an important design parameter which affects many engine performance parameters like specific fuel consumption, engine emission and others. The intake and exhaust valves must open and close at the right time. If the engine is operating at conditions other than the rated condition, 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 6-stroke engine.

1.3 OBJECTIVE

The objective of this project is to investigate the effect of intake timing of the newly developed piston-controlled intake port single cylinder 6-stroke engine to its in-cylinder air flow.

1.4 SCOPE OF STUDY

The scopes of this project are:

i. Literature review on the valve timing of the 2-stroke and 4-stroke engine.
ii. Study on the six stroke engine operation.
iii. Intake port modification.
iv. Simulation of air flow inside the engine during intake stroke process by using COSMOSFloWorks.

1.5 FLOW CHART

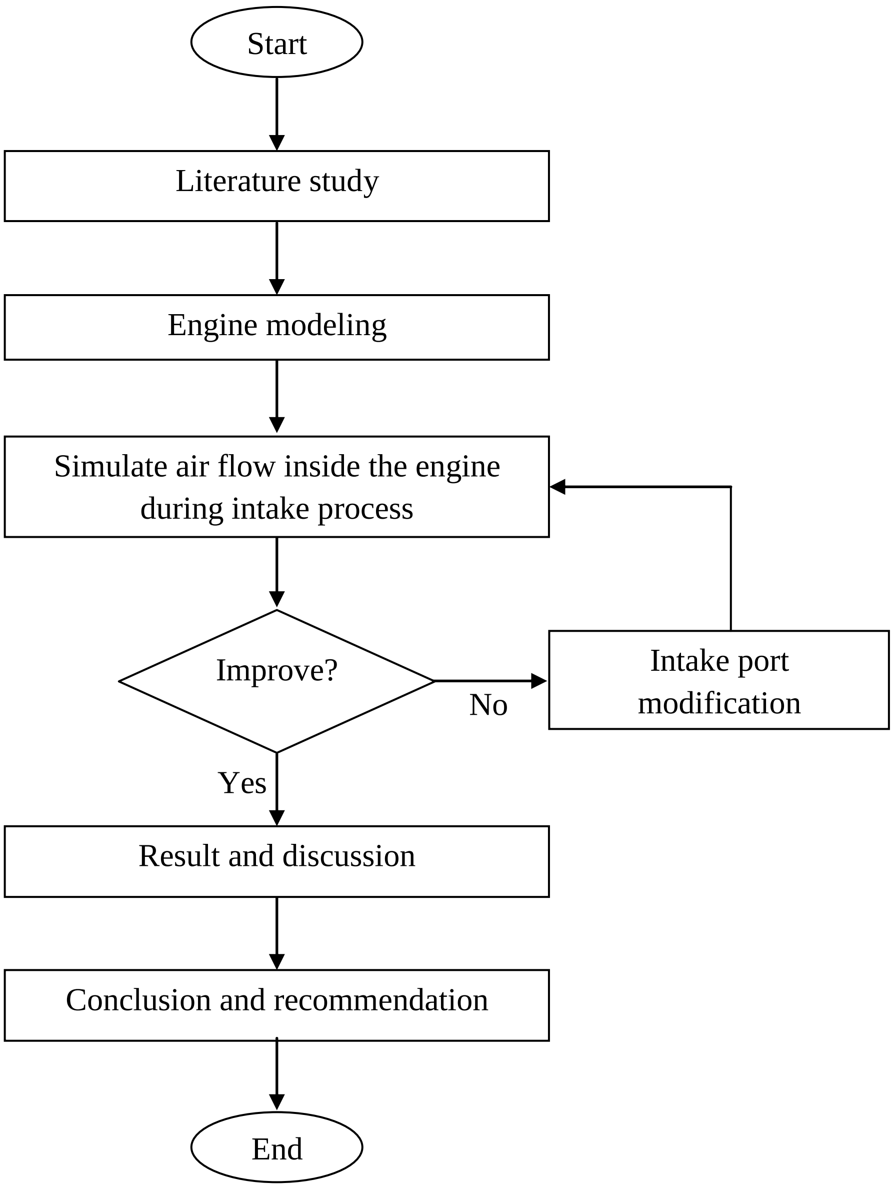

Figure 1.1: Project flow chart

CHAPTER 2

LITERATURE REVIEW

2.1 SIX-STROKE ENGINE

The term six stroke engine describes two different approaches in the internal combustion engine, developed since the 1990s, to improve its efficiency and reduce emissions. In the first approach, the engine captures the usually wasted heat from the four stroke Otto cycle or Diesel cycle and uses it to power an additional power and exhaust stroke of the piston in the same cylinder. Designs of such an engine either use steam or air as the working fluid for the additional power stroke. This not only helps increase the power extracted but also cools the engine and eliminates the need for using a cooling system making the engine lighter and hence increasing the efficiency over the normal Otto or diesel cycle. The piston in such a six stroke engine undergoes reciprocating motion six times for every injection of fuel. These six stroke engines have 2 power strokes: one due to the fuel and one due to the steam or air. The currently notable six stroke engine designs in this class are the Crower's six stroke engine, invented by Bruce Crower of the U.S.; the Bajulaz engine by the Bajulaz S A Company, of Switzerland; and the Velozeta Six-stroke engine built by the College of Engineering, at Trivandrum in India. The second approach to the six-stroke engine uses a second opposed piston in each cylinder that moves at half the cyclical rate of the main piston, thus giving six piston movements per cycle. Functionally, the second piston replaces the valve mechanism of a conventional engine but also increases the compression ratio. The currently notable designs in this class include two designs developed independently are the Beare Head engine, invented by Australian Malcolm Beare, and the German Charge pump, invented by Helmut Kottmann.

2.1.1 Types of Six Stroke Engine

Griffin six stroke engine

The key principle of the "Griffin Simplex" was a heated exhaust jacketed external vaporizer, into which the fuel was sprayed. The temperature was held around 550 ^{o}F (288 ^{o}C), sufficient to physically vaporize the oil but not to break it down chemically. This fractional distillation supported the use of heavy oil fuels, the unusable tars and asphalts separating out in the vaporizer. Hot bulb ignition was used, which Griffin termed the 'Catathermic Igniter', a small isolated cavity connected to the combustion chamber. The spray injector had an adjustable inner nozzle for the air supply, surrounded by an annular casing for the oil, both oil and air entering at 20 lbs sq in pressure, and being regulated by a governor (Knight, 1999).

Bajulaz six stroke engine

The Bajulaz six stroke engine is similar to a regular combustion engine in design. There is however modifications to the cylinder head, with two supplementary fixed capacity chambers: a combustion chamber and an air preheating chamber above each cylinder. The combustion chamber receives a charge of heated air from the cylinder; the injection of fuel begins an isochoric burn which increases the thermal efficiency compared to a burn in the cylinder.

The high pressure achieved is then released into the cylinder to work the power or expansion stroke. Meanwhile a second chamber which blankets the combustion chamber has its air content heated to a high degree by heat passing through the cylinder wall. This heated and pressurized air is then used to power an additional stroke of the piston. The advantages of the engine include reduction in fuel consumption by at least 40%, two expansion strokes in six strokes, multi-fuel usage capability, and a dramatic reduction in pollution (Yuen, 1986).

Velozeta six-stroke engine

In a Velozeta engine, during the exhaust stroke, fresh air is injected into the cylinder, which expands by heat and therefore forces the piston down for an additional stroke. The valve overlaps have been removed and the two additional strokes using air injection provide for better gas scavenging. The engine seems to show 40% reduction in fuel consumption and dramatic reduction in air pollution. Its specific power is not less than that of a four-stroke petrol engine. An altered engine shows a 65% reduction in carbon monoxide pollution when compared with the four stroke engine from which it was developed.

Crower six stroke engine

In a six-stroke engine developed in the U.S. by Bruce Crower, fresh water is injected into the cylinder after the exhaust stroke, and is quickly turned to superheated steam, which causes the water to expand to 1600 times its volume and forces the piston down for an additional stroke (Avinash, 2007). This design also claims to reduce fuel consumption by 40%.

Beare Head

The term “Six Stroke” was coined by the inventor of the Beare Head, Malcolm Beare. The technology combines a four stroke engine bottom end with an opposed piston in the cylinder head working at half the cyclical rate of the bottom piston. Functionally, the second piston replaces the valve mechanism of a conventional engine (Beare, 1998).

Charge pump engine

In this engine, similar in design to the Beare head, a “piston charger” replaces the valve system. The piston charger charges the main cylinder and simultaneously regulates the inlet and the outlet aperture leading to no loss of air and fuel in the exhaust. In the main cylinder, combustion takes place every turn as in a two-stroke engine and lubrication as in a four-stroke engine. Fuel injection can take place in the piston charger, in the gas transfer channel or in the combustion chamber. It is also possible to charge two working cylinders with one piston charger. The combination of compact design for the combustion chamber together with no loss of air and fuel is claimed to give the engine more torque, more power and better fuel consumption (Paswan, 2008).

2.2 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 (SweetHaven, 1985).

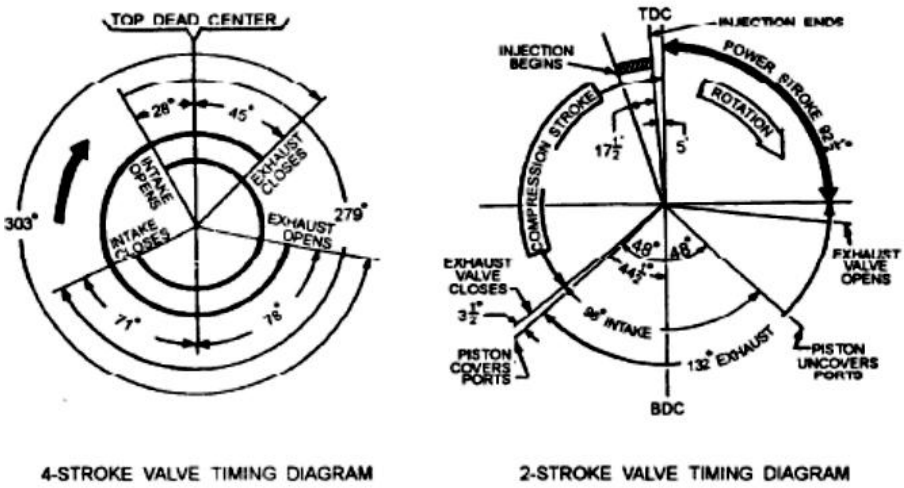

Figure 2.1: Typical valve timing diagram

Source: SweetHaven (1985)

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.

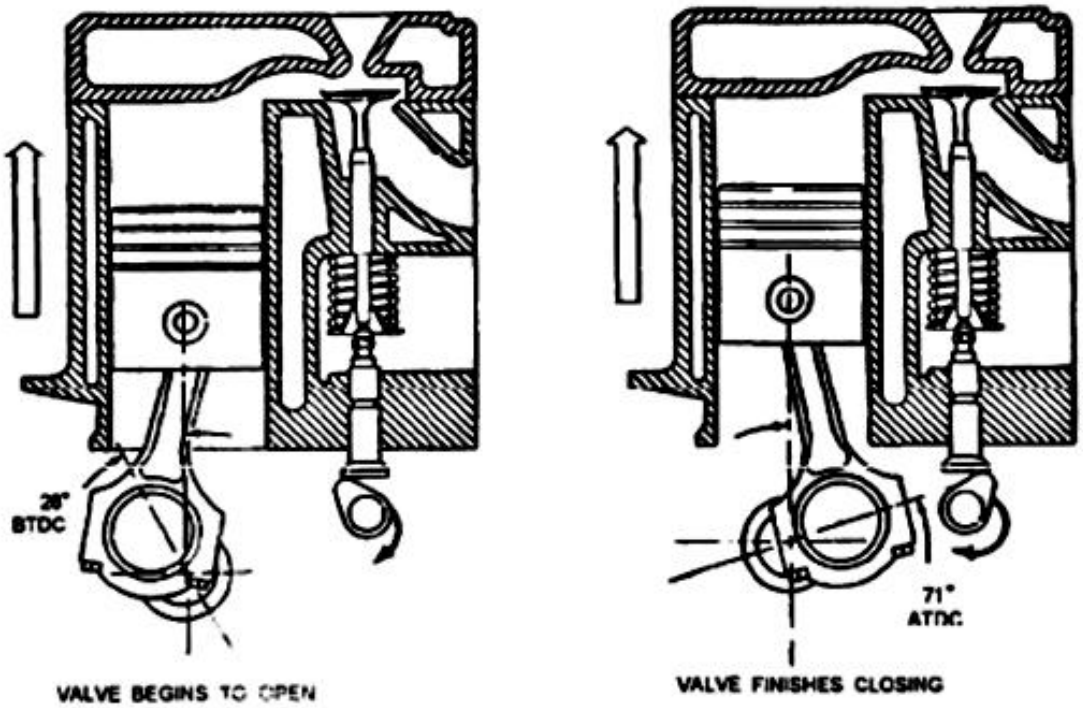

Figure 2.2: Opening and closing point of the valve

Source: SweetHaven (1985)

2. Duration as shown in Figure 2.3 is the amount of crankshaft rotation (in degrees) that a given valve remains open.

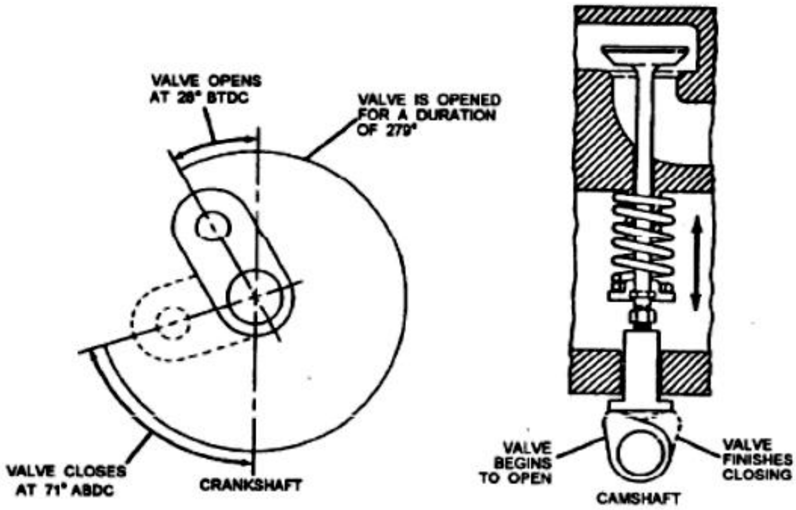

Figure 2.3: Valve opening duration

Source: SweetHaven (1985)

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.

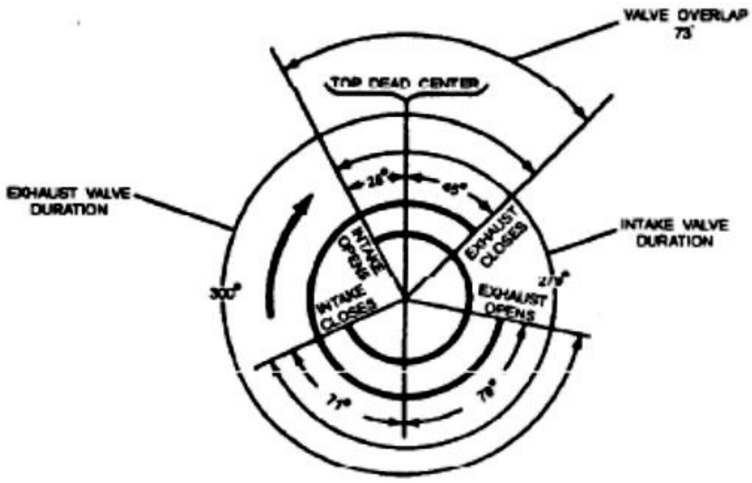

Figure 2.4: Valve timing diagram showing valve overlap in 4-stroke engine

Source: SweetHaven (1985)

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.

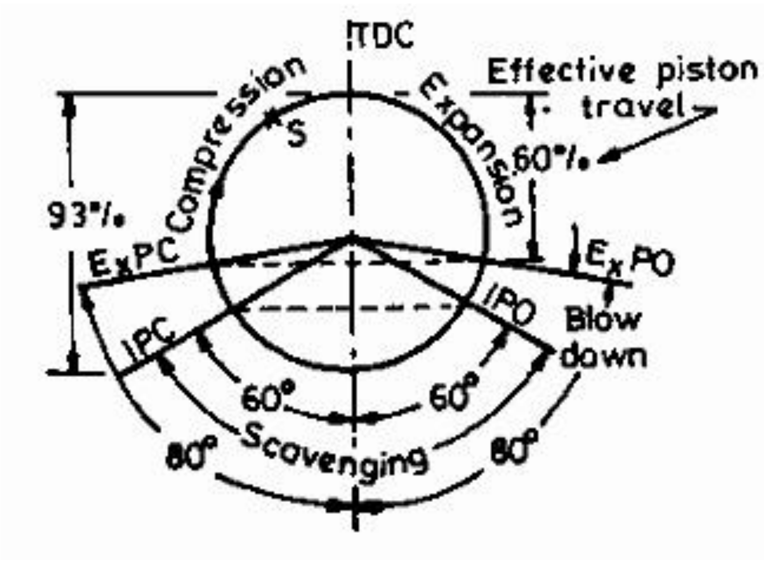

Figure 2.5: Valve timing diagram showing scavenging period in 2-stroke engine

Source: Mathur and Sharma (2000)

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.

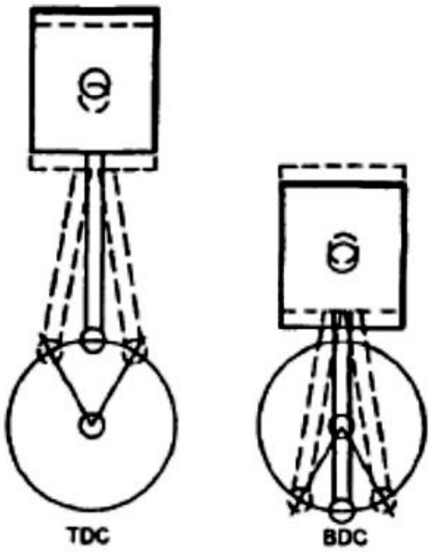

Figure 2.6: Rock position

Source: SweetHaven (1985)

2.3 THE IMPACT OF VALVE EVENTS UPON ENGINE PERFORMANCE AND EMISSIONS.

2.3.1 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.

2.3.2 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.

2.3.3 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.

2.3.4 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.

2.4 IN CYLINDER FLOW ANALYSIS USING CFD

Several studies have been done for the intake region flow using computation and experimental methods. Previous investigations on in-cylinder flow studies in the engine cylinder have reported that high swirl and tumble flows are known to produce very large air velocities and high dissipation rates (Reuss et al., 1995). Dent and Chen (1994) investigated the computational study of flow through a curved inlet port. The authors simulated the three-dimensional flow within the port and cylinder for the intake process and predicted the flow structure affected by the valve lift and port shape. The numerical prediction shows an appreciable pressure recovery for a favorable flow passage between valve seat and the valve head.

Cui, et al (1998) studied the physical mechanisms responsible for cylinder-to-cylinder variation of flows between different cylinders. A validated comprehensive computational methodology was used to generate grid independent and fully convergent results. Very large scale, three-dimension, viscous, turbulent flow simulations, involving finite volume cells and the complete form of the time-averaged Navier-Strokes equations, were conducted to study the mechanisms responsible for total pressure losses in the entire intake system.

Taylor, et al (1998) predicted that the elimination of the valve recess leads to large reductions in total pressure loss at low lifts, with little effect on flow losses at higher lifts. Also redesign of upstream region and overall domain, especially at high valve lifts with the development of complete computational methodologies, the authors are of the view such CFD-based tools can be quite valuable in the geometry specific problems of an industrial design setting.

Bicen, et al (1985) indicated that the flow pattern in the intake region is insensitive to flow unsteadiness and valve operation, and thus could be predicted through steady flow tests and computational simulations with reasonable accuracy. The validity of the quasi-steady assumption has been examined here by comparing LDA measurements of the velocity field at the intake valve exit plane under both steady and unsteady conditions. The results obtained provide detailed information on the flow through the intake valve to minimize the uncertainties with the lack of boundary conditions for calculation methods and quantify earlier findings about the influence of the geometric and flow parameters on the valve performance.

Peters and Gosman (1993) present a numerical simulation method for the calculation of an unsteady, one-dimensional flow and heat transfer in the branch intake manifolds of multi-cylinder engines. The method operates on the one-dimensional differential conservation equation. The equations are solved by a time-marching finite volume method, on a computational mesh in which velocities are located between the pressures, which drive them. The method is assessed by application to flow calculations in the intake manifold for which time-varying velocity data and overall efficiency for a range of speeds. Winterbone and Pearson (2000) explain in detail about the manifold design. Heywood (1998) has explained the working principle and design consideration of port and valve region.

2.4.1 Basic Governing Equation for CFD Analysis

The Navier-Stokes equations are the basic governing equations for a viscous, heat conducting fluid. It is a vector equation obtained by applying Newton's Law of Motion to a fluid element and is also called the momentum equation. It is supplemented by the mass conservation equation, also called continuity equation and the energy equation. Usually, the term a Navier-Stokes equation is used to refer to all of these equations (Navier, 1822). The instantaneous continuity equation (2.1), momentum equation (2.2) and energy equation (2.3) for a compressible fluid can be written as:

$$\frac{\partial \rho}{\partial t} + \frac{\partial}{\partial x_j}\left[\rho u_j\right] = 0 \tag{2.1}$$

$$\frac{\partial}{\partial t}\left(\rho u_i\right) + \frac{\partial}{\partial x_j}\left[\rho u_i u_j + p\delta_{ij} - \tau_{ji}\right] = 0, \quad i = 1, 2, 3 \tag{2.2}$$