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A REVIEW STUDY OF AIR CAPACITY FOR FOUR STROKE DECK'S ENGINE

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ABSTRACT: In this review research, the air capacity of four stroke deck's engine is investigated and studied thoroughly from the considerations of Introduction; effect of residual-gas temperature; effect of operating conditions on volumetric efficiency which constitutes the following: Effect of engine size, engines with similar cylinders, effect of the pressure ratio $\{P_e/P_i\}$, effect of fuel injection on volumetric efficiency, effect of design on volumetric efficiency which in turn includes (i.e. effect of design on $\{\Delta T\}$); effect of valve overlap; inlet system design; important general relations; multi cylinder inlet systems; effect of exhaust pipe length; effect of heat transfer; effect of camshaft on volumetric efficiency which includes the following: Camshaft profile, the effect of lift camshaft, duration, lobe separation, overlap and profile; effect of atmospheric conditions on performance from the following considerations: Effect of changes in atmospheric temperature, and effect of inlet temperature; effect of coolant temperature; intake and exhaust system heat transfer; the study of the influence of valve profile from the following points of view: Types of valves, poppet valves, inlet valve flow capacity and exhaust valve flow capacity; the study of the effect of valve or port timing on engine performance from the viewpoints of the impact of valve events upon engine performance and emission, the parameters or variable quantities; effects of valve overlap; Valve peak lift; industry trends; Exhaust Gas Recirculation (EGR) systems; variable valve timing; phase changing systems; profile switching systems; variable event timing systems; variable lift systems; and electro-magnetic valve actuation systems.

KEYWORDS: Air Capacity; Deck's Engine; Residual Gas; Valve Profile; Coolant Temperature

1 INTRODUCTION

In studying air capacity, it is convenient to set up a figure of merit, which is independent of cylinder size. Such a figure of merit for the four-stroke cycle is the volumetric efficiency. The intake system restricts the amount of air that can be inducted into the cylinder during one cycle. The volumetric efficiency is the parameter describing the effectiveness of the induction process [1] – [16].

Volumetric efficiency is defined as the mass of fresh mixture, which passes into the cylinder in one suction stroke, divided by the mass of this mixture, which would fill the piston displacement at inlet density. It is expressed mathematically as follows: [1].

$$\eta_v = \frac{2M_i}{NV_d\rho_i} \tag{1}$$

Where:
$\eta_v =$ volumetric efficiency.
$M_i =$ mass of fresh mixture per unit time.
$N =$ number of revolution per unit time.
$V_d =$ total displacement volume of the engine.
$\rho_i =$ inlet density.
The factor 2 in this equation arises from the fact that in the four-stroke engine there is one cycle for every two-crank revolution [1].

2 EFFECT OF RESIDUAL GAS TEMPERATURE

Residual gas will require a volume in the cylinder, which otherwise could have been filled with fresh mixture. Thus, less residual gas will increase the volumetric efficiency. The heat transfer from hot residual gas to cool fresh mixture is often presumed to decrease the volumetric efficiency.

Diesel engines generally have much higher compression ratio than gasoline engines. The effect of residual gas will thus be smaller on diesel engines [17]. It has often been supposed that heat transfer between hot residual gases and the fresh mixture reduces volumetric efficiency when the two gases mix during the induction process [1].

3 EFFECT OF OPERATING CONDITIONS ON VOLUMETRIC EFFICIENCY

3.1 Effect of engine size

Similar engines running at the same values of mean piston speed and at the same inlet and exhaust pressures inlet temperature, coolant temperature, and fuel-air ratio will have the same volumetric efficiency within measurable limits [1].

3.2 Engines with similar cylinders

Under certain limitations, the foregoing consideration can be extended to engines having similar cylinders, even though the number and arrangement of the cylinders have their effects.

Engines with similar cylinders, but with different manifold designs will not necessarily have similar curves of volumetric efficiency against piston speed. However, in many cases, the effects due to common inlet and exhaust manifolds are small and the rule that volumetric efficiency among cylinders of similar design is the same at the same mean piston speed is a good first approximation when actual test data are not available [1].

3.3 Effect of the pressure ratio $\{P_e/P_i\}$

Figure 1 below illustrates the effect of exhaust pressure to inlet pressure ratio on ideal-cycle volumetric efficiency, [1].

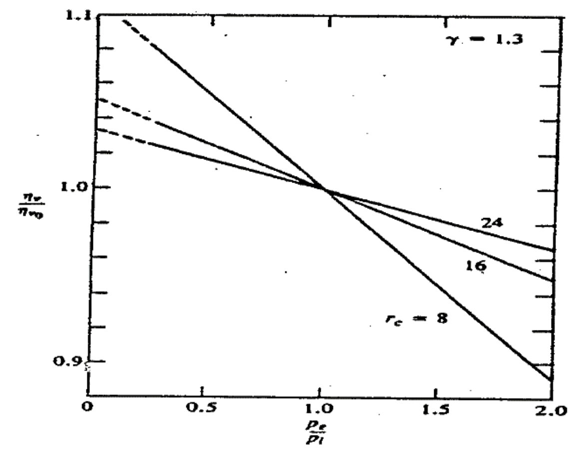

Figure 1. The Effect of exhaust pressure to inlet pressure ratio on ideal-cycle volumetric efficiency, [1].

As the pressure ratio $\{P_e/P_i\}$ and the compression ratio $\{P_c\}$ are varied, the fraction of the cylinder volume occupied by the residual gas at the intake pressure varies. As this volume increases, volumetric efficiency decreases. These effects on ideal – cycle volumetric efficiency is given in the following equation:

$$\eta_v = \left(\frac{M}{M_a}\right)\left(\frac{p_i}{p_{a,0}}\right)\left(\frac{T_{a,0}}{T_i}\right)\frac{1}{[1+(F/A)]}\left\{\frac{r_c}{r_c-1} - \frac{1}{\gamma(r_c-1)}\left[\left(\frac{p_e}{p_i}\right)+(\gamma-1)\right]\right\}$$

For $\gamma = 1.3$, these effects are shown in Figure 1 above, [18].

3.4 Effect of fuel injection on volumetric efficiency

The injection of fuel in diesel engine occurs after induction, and the injection process itself has no direct effect on volumetric efficiency, [1].

3.5 Effect of design on volumetric efficiency

Design here is taken to mean the geometrical shape of the cylinder, valves gear, and manifold system and the materials of which they are made. Thus, assemblies of different size are taken to be of the same design provided all the ratios of corresponding dimensions to the bore, which remain unchanged, and the same materials are used in corresponding parts, [1].

3.5.1 Effect of design on $\{\Delta T\}$

Design affects $\{\Delta T\}$ largely through its effect on the temperature of the surfaces to which the gases are exposed during induction. Thus, designs, which minimize the temperatures of inlet, manifold inlet ports, and inlet valves are desirable from this point of view. Obviously, exhaust heating of the inlet system is undesirable from the reasons of fuel distribution or evaporation.

The inlet valves and valves seats tend to run at temperature far above that of the coolant. Therefore, improvement of heat conductivity between these parts and the coolant is effective in reducing $\{\Delta T\}$. The same can be said for the conductivity between inner cylinder surface and the coolant.

Aside from heat-transfer effects, the design factors having important effects on volumetric efficiency are:

Inlet-valve and design, already considered in connection with the Mach index, Z.
Valve timing.
Exhaust-valve size and design.
Stroke-bore ratio.
Compression ratio.
Design of inlet system.
Design of exhaust system.
Cam contour shape.

The two feature of valve timing which have important effects on volumetric efficiency are valve overlap angle and inlet-closing angle. Within the limits of conventional practice, the other valve events have little effect on volumetric efficiency [1].

4 EFFECT OF VALVE OVERLAP

If $\{\boldsymbol{P_e}\}$ is greater than $\{\boldsymbol{P_i}\}$, gas will flow from the exhaust manifold back into the cylinder and push out gas to the inlet manifold. Therefore, the mass of the exhaust gas will take room from the fresh mixture. Thus, back flow will decrease the volumetric efficiency.

If the $\{\boldsymbol{P_e}\}$is smaller than $\{\boldsymbol{P_i}\}$, gas will flow from the inlet manifold to combustion chamber and push out gas through the exhaust port. Assuming that the gas pushed out contains exhaust gas infers that the back flow in this case will increase the volumetric efficiency. Large overlap is often used in turbo charged diesel engines. Flow from the valve overlap is used to reduce the turbine temperature, [18].

Whenever, valve overlap is considered for supercharged engines it should be remembered that the power required by the compressor would be increased in direct proportion to the increase in mass flow [1].

5 INLET SYSTEM DESIGN

It has long been known that high volumetric efficiencies can be obtained at certain speeds by means of long inlet pipes. The effects noted, are caused by the inertia and elasticity of the gases in the inlet pipe and cylinder [1].

6 IMPORTANT GENERAL RELATIONS

The following general relations are important:

If viscosities are assumed negligible and, if similar engines have similar inlet system, the effects of inlet dynamics on volumetric efficiency will be the same at the same piston speed, if other operating variables being held constant.

This conclusion also applies to engine of different stroke-bore rations, provided the cylinder design is otherwise the same and the rations pipe diameter to bore and pipe-length-to stroke are held the same.

The dynamic pressure at the inlet port at the end of induction is the sum of effects caused by "standing" waves, which have been set up in the inlet pipe by previous inlet stroke, and the effects of the transient wave set up by the induction process.

There are no sudden changes in volumetric efficiency curves. Even at points where the "organ pipe" frequencies of the inlet pipe are multiples of the revolution speed.

Long pipes with small rations of D/B give high volumetric efficiencies at low piston speeds because high kinetic energy is built up in the pipe towards the end of the induction process. At higher piston speeds, the flow restriction offered by small D/B rations becomes dominant and the volumetric efficiency falls.

Long pipe with large ratios of D/B show maximum volumetric efficiencies at intermediate piston speeds due to kinetic energy built up in the pipe. At high piston speeds, the air mass in such pipes is slow to accelerate, and volumetric efficiency falls off.

As pipe become shorter, the maximum gains in volumetric efficiency over that with no inlet pipe grow smaller, but the range of piston speeds over which some gain is made grows wider [1].

7 MULTI CYLINDER INLET SYSTEMS

When several cylinders are connected to one inlet manifold, the problem of manifold design becomes complicated, especially if liquid fuel as well as air must be distributed [1].

8 EFFECT OF EXHAUST PIPE LENGTH

The effect of changing exhaust pipe length on volumetric efficiency is usually small because the effect of changing in $\{\boldsymbol{P_i}\}$ on volumetric efficiency is slight [1].

9 EFFECT OF HEAT TRANSFER

Heat transfer can be described by the total increase of temperature, $\Delta\boldsymbol{T}$ in the gas. A model describing this is the formula of heat transfer for gas flowing through a tube.

$$\Delta T = \eta\left(T_s - T_g\right)$$

Where η is constant, T_s is the surfer temperature of the cylinder wall and T_g is the mean gas temperature. The effect of the heat transfer will decrease the volumetric efficiency. As shown in above equation, metal temperature and gas temperature in the inlet manifold will affect$\Delta\boldsymbol{T}$. When the gas temperature in the cylinder is higher than in the inlet manifold, volumetric efficiency will decrease.

This is because when the density of the gas in the combustion chamber will decrease, the mass of the gas in combustion chamber will decrease. The metal temperature can vary a lot during engine operation. Thus, T_s , is a dynamic effect. A good dynamic model of T_s is probably the best way to achieve a dynamic volumetric efficiency model that compensate for the temperature effects in the transient [17].

10 EFFECT OF CAMSHAFT ON VOLUMETRIC EFFICIENCY

10.1 Camshaft profile

The camshaft is a shaft having some semi-oval protrusions, which are designed to control the open and close intervals of the inlet and exhaust poppet valves in the gasoline and diesel engines. Rotation of the cam, which takes its movement from the crankshaft via a chain or a trigger belt, causes its profile to slide against the smooth flat closed end of a cylindrical member known as follower, [18].

10.2 The Effect of lift camshaft

The intake and exhaust valves need to open to let air/fuel in and exhaust out of the cylinders. Generally, opening the valves quicker and further will increase engine output. Increasing valve lift, without increasing duration, can yield more power without much change in the nature of the power curve.

However, an increase in valve lift almost and always is accompanied by an increase in duration. This is because ramps are limited in their shape, which directly related to the type of lifters being used, such as flat or roller.

10.3 Duration

Duration is the angle in crankshaft degrees that the valve stays off its seat during the lifting cycle of the cam lobe. Increasing duration keeps the valve open longer, and can increase high rpm power. Doing so, increases the RPM range of power production of the engine. Increasing duration without a change in lobe separation angle will result in increased valve overlap.

10.4 Lobe separation

Lobe separation is the angle in camshaft degrees between the maximum lift point of the intake and exhaust valves. It is a result of the placement of the intake and exhaust lobes on the camshaft. Lobe separation affects valve overlap, which affects the nature of the power curve.

10.5 Overlap

Overlap is the angle in crankshaft degrees that both the intake and exhaust valves are open. This occurs at the end of the exhaust stroke and the beginning of the intake stroke. Increasing lift duration and/or decreasing lobe separation increases overlap.

At high engine speeds, overlap allows the rush of exhaust gases out the exhaust valve to help pull the fresh air/fuel mixture into the cylinder through the intake valve. Increased engine speed enhances the effect. Increasing overlap increases top end power and reduces low speed power and idle speed quality.

10.6 Profile

If a cam profile has more area under the curve, it has the potential to make more power. Roller profiles can be more aggressive and accelerate the tappet more than a flat tappet profile. Flat profiles can only be shaped up to the point where the tapped "design to" the profile. Roller tappet profiles are not limited by this condition. Therefore, much that even "inverted radius" profile is possible. This benefits engine performance in two ways a more tappet lift can be achieved without the added duration that would normally be required to "ramp up". A flat tappet to the added lift makes the lift curve more "broader" without increasing lift. Of course, both of these benefits can be combined to create a profile that can easily outperform flat tappet cams [19].

11 EFFECT OF ATMOSPHERIC CONDITIONS ON PERFORMANCE

Under this heading we will show the effects of variation in temperature, pressure and humidity and the combination of changes in these factors which occurs with changing altitude. From Equation (1), [1].

$$\rho_a = \frac{29P_a}{RT_i} = \frac{29}{RT_i}\left\{\frac{1}{1 + F_i\left(\frac{29}{m_f}\right) + 1.6h}\right\} \quad (1)$$

$$\frac{\rho_{a1}}{\rho_{a2}} = \frac{T_{i2}}{T_{i1}}\left\{\frac{P_{i2}\left[1 + F_i\left(\frac{29}{m_f}\right) + 1.6h\right]_1}{P_{i1}\left[1 + F_i\left(\frac{29}{m_f}\right) + 1.6h\right]_2}\right\} \quad (2)$$

Where:

$\rho_a = density\ of\ dry\ air\ in\ inlet\ manifold.$
$T_i = inlet\ temperature.$
$P_i = inlet\ pressure.$
$F_i = inlet\ fuel - vapor\ to\ dry - air\ rate.$
$h =$
$moisture\ content\ mass\ of\ water\ vapor\ to\ mass\ of\ dry\ air\ ratio.$
$m_f = molecular\ weight\ of\ fuel.$

11.1 Effect of changes in atmospheric temperature

Effects of atmospheric temperature changes on measured engine performance are shown in Figure 2 below.

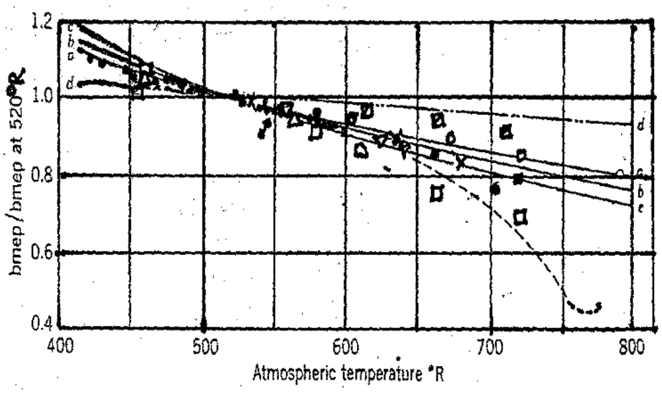

Figure 2. Effect of atmospheric temperature on engine output

Test result from diesel engines:
■ Multi cylinder four-stroke engine.
☐ One-cylinder four-stroke engine, constant F.
☑ One-cylinder four-stroke engine, constant fuel flow.
× Four-stroke sleeve-valve cylinder.
All data at constant speed, constant barometer.
Throttle setting constant except _ _ _.

Fuel-air ratio constant except curve d.
Figure 2 also indicates that diesel engine operated with constant fuel-air ratio shows the same trends as spark ignition engines when allowance is made for their higher friction mep as compared to aircraft engines. In diesel engines, F_i is zero in all cases, [1].

11.2 Effect of inlet temperature

In diesel engines, in practice, the maximum fuel-pump delivery rate is usually set at the factory. When such engines are tested over the usual range in air temperature, at maximum fuel-pump setting, the quantity of fuel per cycle will not change. Thus, although volumetric efficiency varies as in spark-ignition engines, engine output will vary only as the indicated efficiency is changed by the resultant changes in fuel-air ratio and in initial temperature of the cycle. In the usual range of diesel-engine operation, the change in power due these causes may be too small to be noticed.

On the other hand, if a diesel engine is rated at its maximum allowable fuel-air ratio, fuel flow should be changed with airflow to hold the fuel air-ratio constant, [1].

12 EFFECT OF COOLANT TEMPERATURE

Figure 3 below shows volumetric efficiency with respect to coolant temperature for two engines.

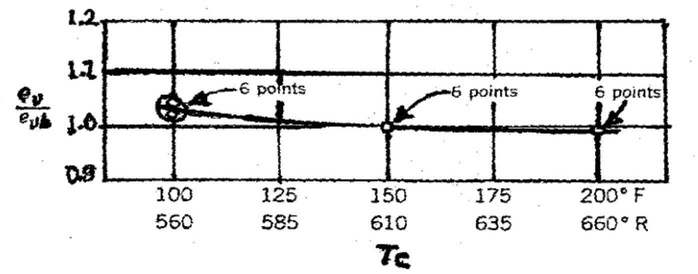

Figure 3. Effect of coolant temperature on volumetric efficiency ρ_{vb} = Volumetric efficiency when T_c = 610 ^oR. Tests on two engines, plymouth 1937 six and ford 1940 V-8: T_i = 460 to 610 ^{o}R;$P_e = P_i = 13\ psia$; s = 2000 ft/min, full throttle. (Markller and Taylor, [1]).

When coolant temperature is changed, the chief effect must be a change in the mean wall temperature to which the gases are exposed during induction, [1].

13 INTAKE AND EXHAUST SYSTEM HEAT TRANSFER

Much higher flow velocities drive convective heat transfer in the intake and exhaust systems. These velocities are greater than that in cylinder heat transfer.

Intake system's heat transfer is usually described by steady, turbulent pipe flow correlations.

With the liquid fuel present in the intake, the heat transfer phenomena become especially complicated. Exhaust flow heat transfer rates are the largest in the entire cycle due to very high gas velocities developed during the exhaust blow down process and the high gas temperature. Exhaust system heat transfer is important since it affects emissions burn up in the exhaust system, catalyst, or particularly to the engine cooling requirements, [1].

14 THE INFLUENCE OF VALVE PROFILE

14.1 Types of valves

The poppet valve is now universally used in- four cycle engines. Sleeves and piston valves have been used in the past but are now obsolete because of their high cost, considerable friction and their adverse effect on oil consumption.

The same disadvantage applies in an even larger degree to the great number of rotary valves. Slide valve, which have been proposed from time to time, have never proved commercially successful.

In two cycle engines, ports controlled by piston motion are used either exclusively or in combination with poppet valves. Some large two-cycle engines use auxiliary automatic or rotary valves in connection with the ports of loop-scavenged cylinder in order to achieve unsymmetrical timing. Some crank case in compression two- cycle engines use rotary or automatic inlet valves for the crank case inlet, [18].

14.2 Poppet valves

The advantages of the poppet valve over other types include the flowing:

1. It can give larger values of valve flow area to piston area than most other types.
2. It has excellent flow coefficients if properly designed.
3. The manufacturing cost of the poppet valve system is lower than that of any other type.
4. It involves very little friction and requires less lubrication than any other type (i.e., no bearing surfaces are under cylinder pressure while the valve is in motion).

14.3 Inlet valve flow capacity

In the importance of adequate valve flow capacity, it was shown that, the inlet valve Mach index is critical. That index is defined as follows, [17]:

$$Z = \left\{\frac{A_p}{A_i}\right\} \frac{S}{C_i a} \qquad (3)$$

Where:

$A_p = piston\ area.$

$A_i = Inlet\ valve\ nominal\ area, \left(\frac{\pi D^2}{4}\right) times\ number\ of\ valves.$

$D = valve\ outside\ diameter.$

$S = mean\ piston\ speed.$

$a = sound\ velocity\ in\ inlet\ gases.$

$C_i = mean\ valve\ flow\ coefficient\ based\ on\ area\ \left(\frac{\pi D^2}{4}\right)$, [17].

From the science of fluid mechanics, we know that the controlling velocity in a compressible flow system is the velocity in the smallest cross section. Considering typical engine designs, it appears that the smallest cross-section exists in the inlet valve opening. For the usual cam contours, it might be expected that the mean flow area through the inlet valve will be proportional to$\left(\frac{\pi D^2}{4}\right)$, where (D) is the valve diameter. In this case, (A_p/A_i) becomes$\left(\frac{b}{D}\right)^2$, where (b) is the cylinder bore.

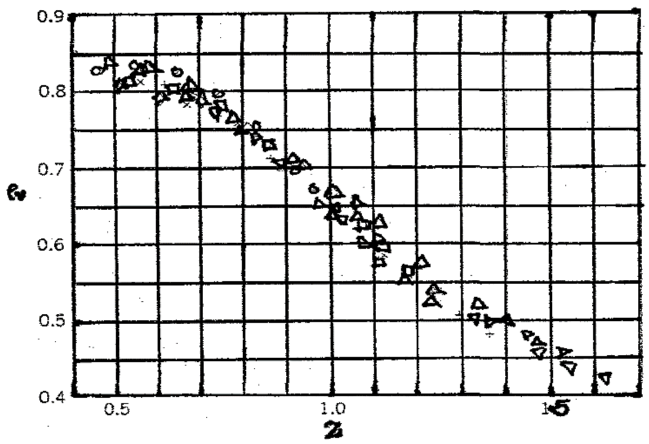

Figure 4. Volumetric efficiency Vs inlet valve mach index [1]

Figure 4 shows η_v plotted against $\left(\frac{b}{D}\right)^2 \frac{S}{C_i a}$ for a given engine equipped with inlet valves of various size shapes, and lifts. Inlet temperature, and therefore (a) was constant. The poor correlation obtained indicates that the velocity $\left(\frac{b}{D}\right)^2 \frac{S}{C_i a}$ was not controlling. Figure 4 shows volumetric efficiency plotted against an inlet valve Mach index defined as:

$$Z = \left\{\frac{b}{D}\right\}^2 \frac{S}{C_i a} \tag{4}$$

The plot shows that $\{\boldsymbol{\eta_v}\}$ is a unique function of Z within limits off measurement of engine operation over a wide range of engine speeds and of inlet valve diameters, lifts, and shapes. Valve timing was held constant. Examination of Figure 4 shows that $\{\eta_v\}$ begins to fall off sharply when Z exceeds about (0.5).

Maximum Z Value:

From the foregoing results it is possible to draw the very important conclusion that engines should be designed, if possible, so that (Z) does not exceed (0.5) at the highest rated speed.

14.4 Exhaust valve flow capacity

The exhaust valve capacity can be less than the inlet valve capacity. A good design compromise is to make the ratio of exhaust valve to inlet valve's flow capacity 0.7 to 0.75. Assuming equal flow coefficients and an equal number of exhaust and inlet valves, this means an exhaust valve diameter 0.83 to 0.87 of the inlet valve diameter.

Results:

We can see that the effect of valve profile outside diameter valve on the index Mach number (Z) and that can be affected on volumetric efficiency directly by this equation [17]:

$$\eta_v = \frac{\overline{M}_i(\theta_{ivc} - \theta_{ivo})}{Z \times 180} \times 100 \tag{5}$$

$\eta_v = Volumetric\ efficiency.$
$\overline{M}_i = Mean\ inlet\ Mach\ number.$
$\theta_{ivc} = Inlet\ valve\ close\ angle.$
$\theta_{ivo} = Inlet\ valve\ open\ angle.$
$Z = Inlet\ Mach\ index.$

Where the diameter is decreased, the Mach index will increase. Hence, when the Mach number exceeds about (0.5), volumetric efficiency decreases and fall sharply.

15. THE EFFECT OF VALVE OR PORT TIMING ON ENGINE PERFORMANCE

High performance engines normally include either valve or port timing that has considerable overlap to achieve high power output. Although such overlapping valve or port timing is very effective to improve the high-performance output of an internal combustion engine, such an arrangement for increasing the power output significantly reduces the performance at mid-range conditions particularly when several cylinders of the engine discharge into a common exhaust device such as an expansion chamber. The reason for this is that there will exist at the exhaust port of the engine higher pressure during a stage of the engine operation when the intake valve is also opened, [19].

15.1 The impact of valve events upon engine performance and emission

The effect of the basic parameters used in the specification of valve timing on engine performance and emissions will be discussed below. Figure 5 below illustrates the conventional 4-stroke cycle of an internal combustion engine.

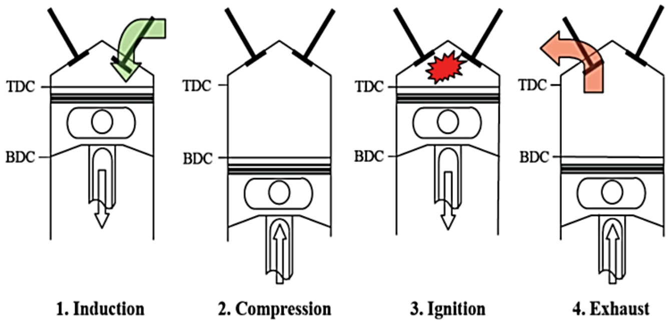

Figure 5. The conventional 4-stroke cycle of an internal combustion engine

The diagrams above illustrate the conventional 4-stroke cycle of an internal combustion engine. It can be seen, that both the intake and exhaust valves remain closed during the compression and ignition phases of the cycle. It is therefore usual for any discussion of valve timing to focus on parts 1 and 4 of the cycle that is the valve motion in the periods either side of piston Top Dead Centre (TDC) on the non-firing stroke.

A camshaft that rotates at half the speed of the crankshaft controls the valve motion. During the four-stroke cycle the crankshaft rotates twice, causing two piston cycles, whilst the camshaft rotates once, causing one cycle of each valve. The different speeds of the crankshaft and camshaft can be the cause of some confusion when describing the timing of valve opening and closing with angles, as 360° of crankshaft rotation is equivalent to 180° of camshaft rotation.

It is normal to discuss the parameters of valve timing with reference position of the piston using crankshaft angle measured from piston TDC (Top Dead Centre) or BDC (Bottom Dead Centre). This will hold to this convention but it should be noted that the duration of valve lift in crankshaft degrees is twice the duration of the profile actually ground onto the camshaft. Duration is defined as the angle of crankshaft rotation between the opening and the closing of the intake or the exhaust valve.

Figure 6 below shows the intake and exhaust valve events, as they would typically appear around the end of the exhaust stroke (TDC).

The term “valve event” refers to the opening or closing of either the intake or the exhaust valve(s) with reference to piston TDC or BDC. The graph shown above represents the intake and exhaust valve events, as they would typically appear around the end of the exhaust stroke (TDC).

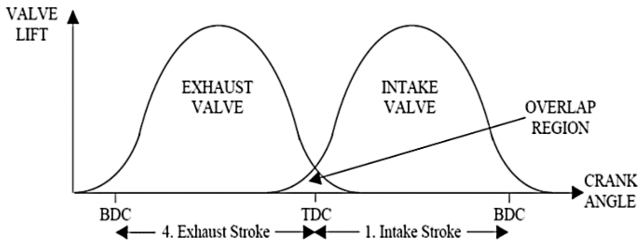

Figure 6. The intake and exhaust valve events

15.2 The parameters

The main parameters to be discussed are:
1. Exhaust Valve Opening Timing – EVO.
2. Exhaust Valve Closing Timing – EVC.
3. Intake Valve Opening Timing – IVO.
4. Intake Valve Closing Timing – IVC.
5. Peak Valve Lift.

The overlap region (if present) is the difference between IVO and EVC (if positive) and is thus affected by changes in either IVO or EVC.

It can be seen from the above graph shown in Figure 6, that the valve events do not coincide with TDC and BDC as depicted in the "theoretical" four-stroke cycle. The reasons for this and the compromises inherent in the selection of valve event timings will form the basis for discussion below, [19].

15.2.1 Effects of changes to Exhaust Valve Opening timing - EVO

As the exhaust valve opens, the pressure inside the cylinder, which results 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 backpressure, 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 work 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 as shown in Figure 7 below.

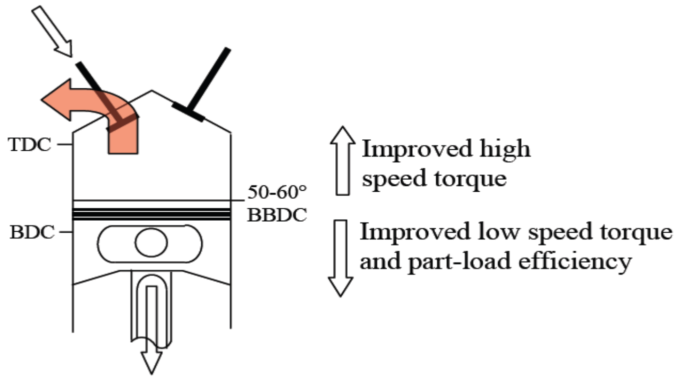

Figure 7. A typical EVO timing is in the region of 50-60° before BDC

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 backpressure 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 backpressure, [19].

15.2.2 Effects of changes to Exhaust Valve Closing timing – EVC

The timing of EVC has a very significant effect 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 effect 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 and 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 (i.e., Pressure waves are generated by exhaust gas flow from the different cylinders), 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 and fuel. Moving EVC Timing further after TDC increases the level of internal EGR (Exhaust Gas Recirculation) with a corresponding reduction in exhaust emissions.

There is a limit to how much EGR the cylinder can tolerate before combustion becomes unstable, 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 as is illustrated in Figure 8 below. This timing largely eliminates internal EGR so as not to detrimentally affect full load performance, [4].

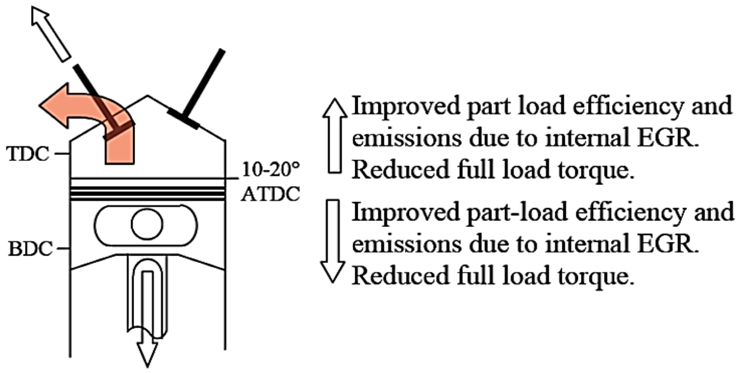

Figure 8. Typical EVC timings are in the range of 5-15° after TDC

15.2.3 Effect of changes to Intake Valve Opening timing – IVO

Typical IVO timing is around 0-10° before TDC, which results in the valve overlap being symmetrical around TDC. This timing is generally set by full load optimization and, as such, is intended to avoid internal EGR. Refer to Figure 9 below.

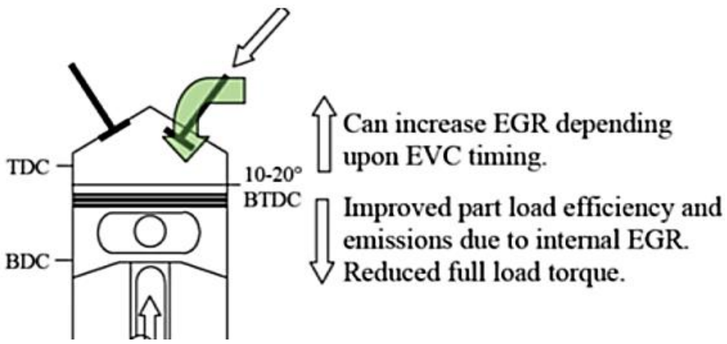

Figure 9. Typical IVO timing is around 0-10° before TDC

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. 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.

15.2.4 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.

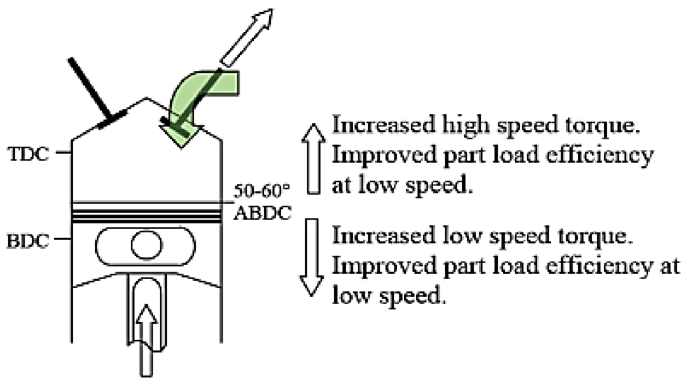

Figure 10. A typical timing for IVC is in the range of 50-60° after BDC

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° (refer to Figure 10) 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 appositive airflow into the cylinder as the intake valve closes.

16 EFFECTS OF VALVE OVERLAP

As discussed previously, valve overlap is the time when both intake and exhaust valves are open. In simple terms, this provides an opportunity for the exhaust gas flow and intake flow to influence each other. Overlap can only be meaningfully assessed in conjunction with the pressure waves present in the intake and exhaust systems at any particular engine speed and load.

In an ideal situation, the valve overlap should allow the departing exhaust gas to draw the fresh intake charge into the cylinder without any of the intake gas-passing straight into the exhaust system. This allows the exhaust gas in the combustion chamber at TDC to be replaced and therefore the amount of intake charge to exceed that, which could be drawn into the cylinder by the swept volume of the piston alone.

A given amount of overlap unfortunately tends to be ideal for only a portion of engine speed and load conditions. Generally, the torque at higher engine speeds and loads can benefit from increased overlap due to pressure waves in the exhaust manifold aiding the intake of fresh charge. Large amounts of overlap tend to result in poor emissions at lower speeds as fuel from the intake charge can flow directly into the exhaust. High overlap can also result in EGR, which, although beneficial to part load economy, reduces full load torque, and can cause poor combustion stability especially under low load conditions such as idle. Poor idle quality can therefore result from too much overlap.

The valve overlap tends to be symmetrical about TDC on most engines. The further away from TDC that valve overlap is present, the more effect the piston motion will have on the airflow. Early overlap may result in exhaust gasses being expelled into the intake manifold and late overlap may result in exhaust gasses being drawn back into the cylinder. Both of these situations result in internal EGR that can be beneficial to part load emissions and efficiency. As discussed earlier, internal EGR tends to be avoided due to the detrimental effect it has on full load torque, [19].

17 VALVE PEAK LIFT

Valve peak lift directly affects the ability for air to flow into the cylinder and exhaust to leave the cylinder and as a result, it significantly influences engine performance. There are some practical limitations of peak lift in most engines.

These limitations are dependent on the design of the particular engine:

Normally the piston crown is profiled to maximize the clearance adjacent to the valves but there are limitations as to how much valve lift can be accommodated without unduly compromising the piston design. The duration of the valve lift imposes a restriction on valve lift due to the acceleration required to achieve a high lift with a short duration. The higher the running speed of the engine, the more the peak lift is restricted for a given valve lift duration.

Low values of valve peak lift will clearly restrict the ability of gas to flow into and out of the cylinder. Maximum engine power output will generally benefit from as much valve lift as possible up to the point where airflow becomes restricted by other features such as the manifold system or cylinder head porting. It does not follow that engines should have the maximum possible valve lift as this can adversely affect low speed performance. Lower intake valve lifts result in higher gas velocity past the valve and this improves fuel mixing and combustion. For maximum torque at a given speed, lift should be kept as low as possible up to the point where the intake of fresh charge becomes restricted. The chosen value of peak valve lift is therefore a compromise between low speed and high-speed full load requirements. A typical value of peak valve lift for a production engine is in the range of 8-10 mm, [19].

18 INDUSTRY TRENDS

There is a constant need for internal combustion engines to become more powerful and efficient with reduced levels of emissions driven by both legislation and increasing customer expectations. For an engine with fixed valve timing, there are inherently compromises made between emissions, high/low speed torque and full/part load efficiency. Ways of avoiding compromise between different engine requirements are constantly being incorporated into new engines and investigated for application to future engines, [19].

19 EXHAUST GAS RECIRCULATION (EGR) SYSTEMS

The presence of exhaust gas in the combustion chamber can be beneficial for emissions, yielding reductions in NOx, HC and CO. This is known as “internal” EGR and is generally avoided due to its impact upon full load torque.

“External” EGR systems are now becoming more common where gas from the exhaust system is pumped back into the intake manifold at part load conditions. This provides benefits in part load emissions and improved efficiency due to a reduction in intake pumping losses. As the quantity of EGR can be changed to suit engine speed and load conditions, there need not be any detriment to full load torque.

Internal EGR however, does have two significant benefits over external EGR: External systems are expensive and are prone to durability problems due to their continual exposure to hot, dirty gasses. The intricate components within

EGR control systems are susceptible to the buildup of deposits causing leakage or blockage. The re -circulated gas in the case of internal EGR is the last portion to have left the cylinder. This portion generally contains the gasses from any crevice volumes in the cylinder and therefore contains a significant portion of the unburned hydrocarbons from the combustion process. External EGR takes a portion of all the exhaust gasses once they are mixed and so has much less ability to reduce hydrocarbon emissions, [19].

20 VARIABLE VALVE TIMING

An increasing number of engines are using variable valve timing systems to avoid some of the compromises of fixed valve timing. A fully flexible system, which could vary valve lift and intake and exhaust valve event timings independently for different engine speed and load conditions, could in principle overcome all of the compromises inherent in a conventional valve train system. In practice however, the more flexible a variable valve timing system becomes, the more complex and hence expensive it tends to become.

There are a number of different variable valve timing systems, currently available and under development, to control different valve timing parameters. Although there are many different designs for achieving such variations, these systems can be grouped in terms of their operation, [19].

21 PHASE CHANGING SYSTEMS

These systems change the timing of the camshaft in relation to the crankshaft in order to advance or retard the timing of the engine valve events. If applied to an engine with a single camshaft, all of the valve events are shifted by the same amount i.e., if IVC is to be retarded by 10° IVO, EVO and EVC will be retarded by 10°. On engines with separate Intake and Exhaust camshafts, a phase change system can be used to change the timing of the intake valve events or the exhaust valve events. The use of two phasing devices can permit independent control over Intake and Exhaust timing changes. Phase change systems have no effect on peak valve lift and cannot change the duration of the valve events i.e., IVO and IVC cannot be moved independently.

Phase changing systems have been available on production engines for a number of years but have tended to be applied only to the highest specification engine in a particular range. Phasing of the intake camshaft to gain increased performance with a mechanism that can be moved between two-fixed camshaft timings is the most common application with the change in timing normally occurring at a particular engine speed.

Recently, there has been a move towards more flexible control systems that allow the camshaft phasing to be maintained at any point between two fixed limits. This has facilitated camshaft phase optimization for different engine speed and load conditions and has allowed exhaust camshaft phasing to be used for internal EGR control. Engines with both intake and exhaust camshaft phase control are now being introduced, [19].

22 PROFILE SWITCHING SYSTEMS

This type of Variable Valve Timing system is capable of independently changing valve event timing and valve peak lift. The system switches between two different camshaft profiles on either or both of the camshafts and is normally designed to change at a particular engine speed. Due to these systems having an inherently two position operation, they are not suitable for optimizing valve timing parameters under different load conditions e.g., EGR control. The ability to change valve event timing, lift and duration ensures that these systems are capable of providing very high-power output from a given engine whilst still complying with emissions legislation, [19].

23 VARIABLE EVENT TIMING SYSTEMS

Variable event timing systems are probably the most flexible type of variable valve timing system to be available on a production engine. Whilst they do not change peak valve lift, they are able to change both the phasing and the duration of valve events. These systems can be controlled to any setting between two extremes and are most effective when optimized for different engine speed and load conditions. Considerable increases in full load torque can be achieved with variable event timing, generally most significant at the extremes of the engine speed range where fixed valve timing is most compromised. Reductions in part load emissions and fuel economy are also achievable through full optimization of the system, [19].

24 VARIABLE LIFT SYSTEMS

There are two main types of variable lift system currently under investigation, the first “scales” the valve lift such that its opening duration is unchanged whilst the second “truncates” the lift profile such that the valve opening duration reduces as lift reduces. Both of these types of system can only be used effectively when combined with a phase changing device. One major benefit of variable lift systems is the potential of throttling the cylinder by reducing the intake valve lift thus saving the pumping losses associated with the conventional throttle. This requires very close control of lift to match changes in engine speed and load conditions and has yet to be fully proven, [19].

25 ELECTRO-MAGNETIC VALVE ACTUATION SYSTEMS

In many ways, this system could provide the greatest potential for optimizing the engine valve events. In principal EVO, EVC, IVO, IVC and possibly valve lift could all be directly controlled by an engine management system. EVA systems are still in the early stages of development and have yet to demonstrate whether they can match the performance of a mechanical valve train in terms of durability, power consumption and refinement whilst maintaining an acceptable system price, [19].

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