ISSN No: - 2456 – 2165

Analysis for Prediction of the Volumetric Efficiency with Continuous Variable Valve Lift Mechanism in Single Cylinder SI Engine

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Abstract— A small single cylinder four stroke SI engine varying capacity from 100cc to 500 cc generally used for two wheelers has a wide range of engine speeds during operation and may vary from 1000 to 8000 rpm. Such variation in speed of engine demands variation in the valve timing and lift of engine at different speeds for optimum performance.

Volumetric efficiency is the measures of performance of an engine. It is often referred as breathing capacity of engine. The power output of an engine depends directly on the amount of charge that can be inducted in the cylinder. When an engine is throttling, a plate obstructs the air intake flow and causes a pressure drop across the plate. In conventional SI engines, the throttle-based air control wastes about 10% of the input energy in pumping the air. Throttling not only reduces the amount of air induced into the engine, but also it introduces flow losses which reduce an engine's efficiency. One of the methods of increasing the volumetric efficiency of an engine is reducing the mechanical losses associated with throttling through Continuous Variable Valve Lift (CVVL) mechanism which provides throttle free load control. In this lift is controlled by the inlet valve, according to load condition.

The present work aims at analyzing the volumetric efficiency of a four stroke spark-ignition engine with CVVL mechanism. Nature of volumetric efficiency has been obtained and compared for with throttle and without throttle valve condition. The analysis has highlighted a great influence of throttle valve on volumetric efficiency and hence on valve lift. Volumetric efficiency with throttle free operation achieved with CVVL mechanism is always superior to throttled operation in SI engine.

Keywords— Volumetric Efficiency, Throttling, Continuous Variable Valve Lift (CVVL) Mechanism, Throttle Free Load Control

I. INTRODUCTION

Volumetric efficiency is used as an overall measure of the effectiveness of a four stroke cycle and its intake and exhaust systems as an air pumping device.

There are mainly two factors which affects the volumetric efficiency:
a) Throttling losses/ pumping losses b) Pressure losses

A. *Throttling losses/ Pumping losses [1]*

In conventional SI engine, load and hence speed of engine is controlled by throttle valve opening. Actual P-V diagram deviates from the theoretical P-V diagram. In actual P-V diagram corners are rounded off because the inlet and outlet valve do not open and close suddenly but take some time to do so. Because of the resistance of inlet valve of the entering charge, the actual pressure inside the cylinder during suction is slightly less than the atmospheric. Similarly because of the resistance of the outer or the exhaust valve to the exhaust gases leaving the cylinder, the actual pressure inside the cylinder during exhaust stroke is slightly higher than the atmospheric, this gives the shaded loop. This shaded area is known as "*pumping loss*". This area is treated as negative and therefore subtracted from the area of larger loop to give net work done.

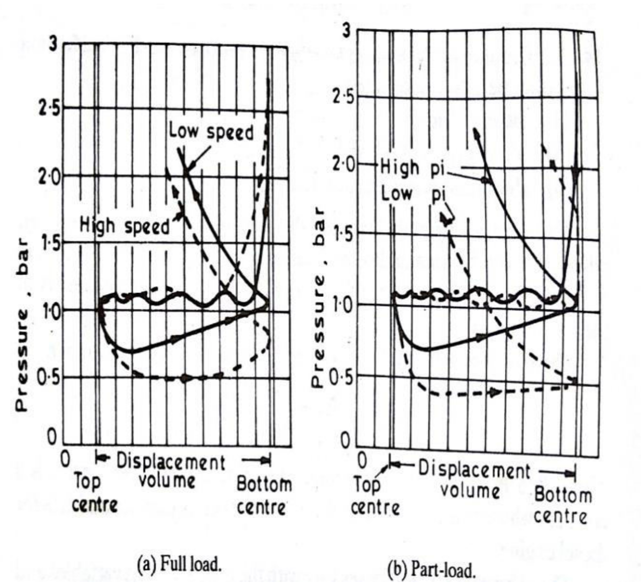


Fig 1.Actual P-V diagrams a) at Full Load Condition And B) At Part Load Condition [1]

Throttle valve is placed before the inlet manifold and before carburettor. The figure 1 shows the actual P-V diagram for Part load condition and full load condition. Actual P-V diagram When the engine is at low load (Part load condition), the throttle valve is almost closed, allowing very less air to enter the inlet manifold. The fuel is supplied accordingly to maintain air-fuel ratio. As the load is increased (full load condition), the throttle valve is opened allowing more air to pass into inlet manifold. During this throttling process, lots of throttling losses takes place, and hence less air is pumped due to this loss. Hence volumetric efficiency is reduced.

B. Pressure Losses

When the throttle valve is partially opened, the pressure in the inlet manifold is less than the atmospheric pressure. But the pressure in the combustion chamber is almost equal to atmospheric pressure at the start of combustion. Hence the charge cannot pass inside the chamber as soon as the inlet valve is opened. Due to this less charge is passed inside the combustion chamber than its actual capacity. Therefore, the volumetric efficiency of engine is reduced [2].

Volumetric efficiency is defines as

$$\eta_v = \frac{2\, m_a}{\rho_{a,0} V_d N}$$

The air density $\rho_{a,0}$ can be evaluated at atmospheric conditions; η_v is the overall volumetric efficiency. Or it can be evaluated at inlet manifold conditions; η_v then measures the pumping performance of the cylinder, inlet port and valve alone [3]. This discussion covers unthrottled (Wide open throttle) engine operations. It is the air flow under these conditions that constrains maximum engine power. Lesser the air flows in SI engines are obtained by restricting the intake system flow area with the throttle valve [3].

II. CONTINUOUS VARIABLE VALVE LIFT (CVVL) MECHANISM

Continuous variable valve actuation mechanism which provides throttle free load control, allows the valve lift to continuously change according to various engine operating conditions. Even though this type of system is typically more complex, costly, and difficult to implement in powertrain, it generally carries greater potential benefits in terms of fuel economy, exhaust emission, and engine performance [4-8].

The CVVL mechanism mainly consists of the following components: electric motor, eccentric shaft, intermediate lever, roller rocker arm, and cam/ camshaft. The servomotor turns the eccentric shaft which moves the intermediate lever back and forth. The intermediate lever has a roller in the middle which is in direct contact with the cam. The upper end of the lever is in contact with the eccentric shaft while its lower end is in contact with the roller rocker arm, which eventually activates the valve motion, as shown in Figure 2.

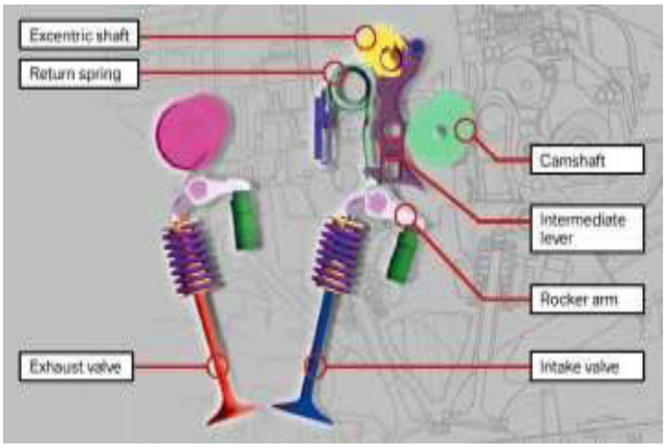
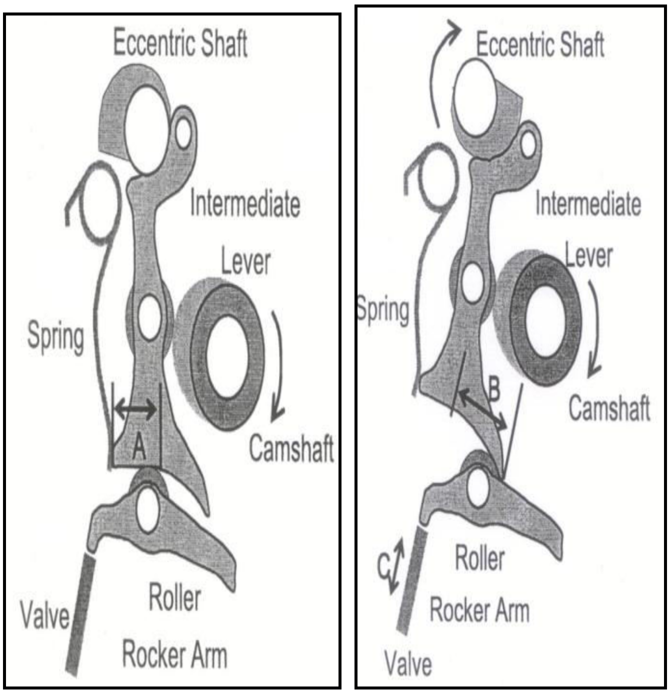


Fig 2.Description of CVVL Mechanism [5]

In the case of no or low lift, the motor turns the eccentric shaft so that the contact surface between the intermediate lever and the roller rocker arm remains almost flat. In this case, the roller rocker arm moves only along the flat surface so that the rotation of cam shaft produces no or very small valve lift as intended.

In the case of high lift, however , the motor turns the eccentric shaft so that the contact surface between the intermediate lever and the roller rocker arm becomes more round. The roller rocker arm then moves along the rounded surface so that the rotation of camshaft now results in a high lift of the intake valve. Based on this operating principle, the system can generate the intake valve lift profile [4].



(a) No or Low Lift (b) High Lift

Figure 3 Operating Principle of CVVL Mechanism [7]

ISSN No: - 2456 – 2165

III. VOLUMETRIC EFFICIENCY

A. *Throttle Valve Analysis*

Except at or close to wide-open throttle, the throttle provides the minimum flow area in the entire intake system. Under typical road-load conditions, more than 90 % of the total pressure loss occurs across the throttle plate. The minimum-to-maximum flow area ratio is large typically of order 100. Throttle plate geometry and parameters are illustrated in fig 4.

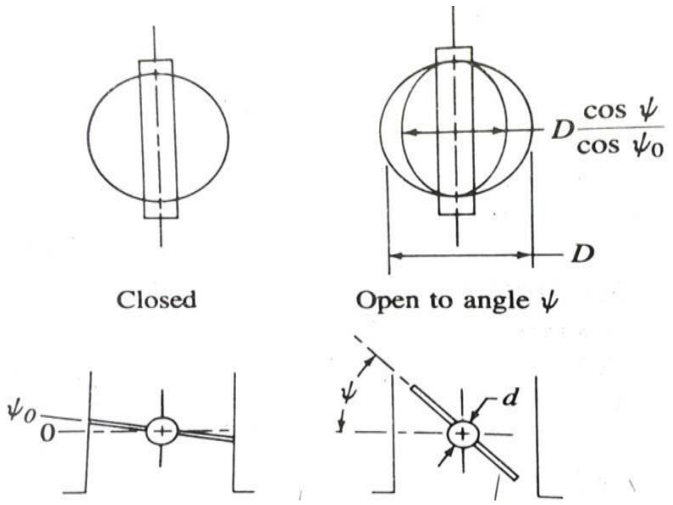


Fig 4.Throttle Plate Geometry

A throttle plate of conventional design such as Fig.1 creates a three-dimensional flow field. At part throttle operating conditions the throttle plate angle is in the *20* to 45◦range and jets issue from the "crescent moon shaped open areas on either side of the throttle plate. The jets are primarily two dimensional immediately below the throttle plate.

In analyzing the flow through the throttle plate, the following factors are considered:

a) The throttle plate shaft is usually of sufficient size to affect the throttle open area.
b) To prevent binding in the throttle bore, the throttle plate is usually completely closed at some nonzero angle (5, 10, or 15◦).
c) The discharge coefficient of the throttle plate is less than that of a smooth.
d) Converging diverging nozzle, and varies with throttle angle, pressure ratio, and throttle plate Reynolds number.
e) Due to the manufacturing tolerances involved, there is usually some minimum leakage area even when the throttle plate is closed against the throttle bore. This leakage area can be significant at small throttle openings.
f) The measured pressure drop across the throttle depends (+ 10 percent) on the circumferential location of the downstream pressure tap.
g) The pressure loss across the throttle plate under the actual flow conditions (which are unsteady even when the engine speed and load are constant) may be less than under steady flow conditions[3]

Consider the flow of an ideal gas with constant specific heats through the duct. For the ideal flow, the stagnation temperature and pressure T_0 and P_0 are related to the conditions at other locations in the duct by the steady flow energy equation

$$T_0 = T + \frac{V^2}{2C_p} \quad \text{.........} \quad (1)$$

Where T is temperature and V is velocity of air at intake manifold side and the isentropic relation

$$\frac{T}{T_0} = \left(\frac{P_t}{P_0}\right)^{\frac{\gamma-1}{\gamma}} \quad \text{.......} \quad (2)$$

By introducing the Mach number M = V/Vs, where Vs is the sound speed $\left(= \sqrt{\gamma RT}\right)$, the following equations are obtained:

$$\frac{T_0}{T} = 1 + \frac{\gamma - 1}{2} M^2 \quad \text{........} \quad (3)$$

$$\frac{P_0}{P_t} = \left(1 + \frac{\gamma - 1}{2} M^2\right)^{\frac{\gamma}{\gamma-1}} \quad \text{.......} \quad (4)$$

The mass flow rate m_{ideal} is

$$m_{ideal} = \rho A_{th} V \quad \text{.....} \quad (5)$$

With the ideal gas law and the above relations for P_t and T, this can be rearranged as

$$\frac{m_{ideal}\sqrt{\gamma RT_0}}{A_{th} P_0} = \gamma M \left(1 + \frac{\gamma - 1}{2} M^2\right)^{\frac{-(\gamma+1)}{2(\gamma-1)}}$$

(or)

$$\frac{m_{ideal}\sqrt{\gamma RT_0}}{A_{th} P_0} = \gamma \left(\frac{P_t}{P_0}\right)^{\frac{1}{\gamma}} \times \left(1 + \frac{\gamma - 1}{2} M^2\right)^{\frac{-(\gamma+1)}{2(\gamma-1)}} \quad \text{....} \quad (6)$$

For given values of Po and To, the maximum mass flow occurs when the velocity at the minimum area or throat equals the velocity of sound. This condition is called choked or critical flow. When the flow is choked the pressure at the throat Pt is related to the stagnation pressure P0 as follows

Critical Value

$$= \left(\frac{2}{\gamma + 1}\right)^{\frac{\gamma}{\gamma-1}} \quad \text{........} \quad (7)$$

For a real gas flow, the discharge coefficient is introduced. The throttle plate discharge coefficient (which varies with A_{th} and minimum leakage area, must be determined experimentally. The mass flow rate through the throttle valve can be calculated from standard orifice equations for compressible fluid flow. For pressure ratios across the throttle more than the critical value (Pt/P0 =0.528; for $\gamma = 1.4$) ,the mass flow rate is given by

$$\text{Mth}=\frac{P0\times Cd\times Ath}{\sqrt{(RT0)}}\times K \qquad \text{....... (8)}$$

Where $K=\sqrt{\frac{2\gamma}{\gamma-1}\left[\left(\frac{Pt}{P0}\right)^{\frac{2}{\gamma}}-\left(\frac{Pt}{P0}\right)^{\frac{\gamma+1}{\gamma}}\right]}$ (for $\left(\frac{Pt}{P0}\right)$> Critical value)

Where A_{th} is the throttle open area, Po and To are the upstream Pressure and temperature, p_t is the pressure downstream of the throttle plate (assumed equal to the pressure at the minimum area (i.e. no pressure recovery occurs), and $C_d$is the discharge coefficient (determined experimentally). For pressure ratios less than the critical ratio, when the flow at the throttle plate is choked

$$\text{Mth}=\frac{P0\times Cd\times Ath}{\sqrt{(RT0)}}\times K \qquad \text{........ (9)}$$

Where $K=\sqrt{\frac{2\gamma}{\gamma-1}\left[\left(\frac{2}{\gamma+1}\right)^{\frac{2}{\gamma-1}}-\left(\frac{2}{\gamma+1}\right)^{\frac{\gamma+1}{\gamma-1}}\right]}$

(for $\left(\frac{Pt}{P0}\right)$< Critical value)

The throttle plate open area A_{th}as a function of angle ψ for the geometry in Fig. 5 is given by

$$\frac{4\times Ath}{\pi\times D^2}=\left(1-\frac{\cos\psi}{\cos\psi 0}\right)+\frac{2}{\pi}\left[\frac{a}{\cos\psi}\left((\cos\psi)^2-a^2(\cos\psi)^2\right)^{1/2}-\frac{\cos\psi}{\cos\psi 0}\sin^{-1}\left(\frac{\cos\psi 0}{\cos\psi}\right)-a(1-a^2)^{1/2}+\sin^{-1}a\right] \qquad \text{..... (10)}$$

Where a = d/D, d is the throttle shaft diameter, D is the throttle bore diameter and $\psi 0$ is the throttle plate angle when tightly closed against the throttle bore When $\psi = \cos^{-1}(a\cos\psi 0)$ the throttle open area reaches its maximum value ($\approx \Pi\, D^2/4 - dD$).

The relation between air flow rates, throttle angle, intake manifold pressure, and engine speed for a two-barrel carburetor and a 4.7-dm^3 (288-in^3) displacement eight-cylinder production engine is shown in Figure 9.4. While the lines are from a quasi-steady computer simulation, the agreement with data is excellent. The figure shows that for an intake manifold pressure below the critical Value (0.528 x P_{atm}= 53.5 kN/m^2 = 40.1 cm of Hg) the air flow rate at a given throttle position is independent of manifold pressure and engine speed because the flow at the throttle plate is choked.

Basic formulae to obtain air flow rate for various intake manifold pressure for different engine speed.
Equating flow through throttle body to engine flow requirement the throttle bore diameter D as a function of engine rpm can be given as

$$D=\sqrt{\frac{4\times V_s\times\frac{N}{2}\times\eta_{vol}}{C_d\Pi\times V_t}} \qquad \text{.... (11)}$$

Where C_d coefficient of discharge through throttle body, η_{vol} is volumetric efficiency of engine, N is engine speed, Vs is displacement volume, and V_t is air velocity through throttle body and is given by

$$V_t=\sqrt{\frac{2P_0}{\rho_0}\left(\frac{\gamma}{\gamma-1}\right)\left(1-\frac{P_t}{P_0}\right)^{\frac{\gamma-1}{\gamma}}} \qquad \text{... (12)}$$

Where Po is atmospheric pressure and P_t is intake manifold pressure.

Air flow rate $\dot{m}$ as function of throttle bore diameter D, can be given as

$$\dot{m}th=\frac{\Pi D}{4}\sqrt{2P_0\rho_0\frac{\gamma}{\gamma-1}\left(\frac{P_t}{P_0}\right)^{\frac{2}{\gamma}}\left(1-\frac{P_t}{P_0}\right)^{\frac{\gamma-1}{\gamma}}} \qquad \text{(13)}$$

Actual mass flow rate of air in combustion chamber

m_{actual}= C_d x m_{th}

C_d = 0.57 (c_d corresponding to maximum lift)

Volumetric Efficiency at N rpm

η_{vol} = 2 x m_{actual} / (ρ x V_dx N)

where V_d= pi x 0.072 x 0.072 x 0.049 /4;
(Engine displacement)

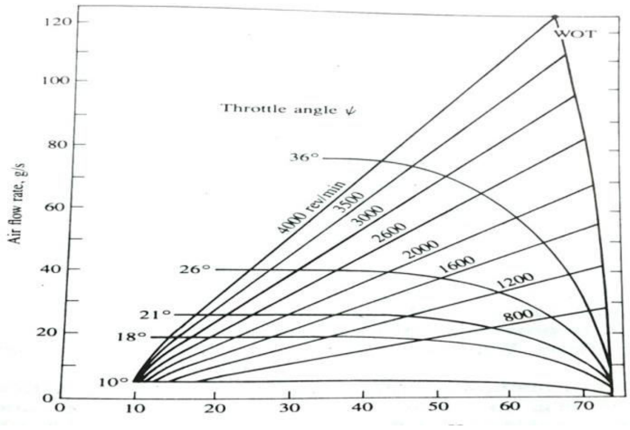


Fig 5.Variation in Air Flow Rate past A Throttle, With Inlet Manifold Pressure Throttle Angle, And Engine Speed For Multi Cylinder SI Engine [3]

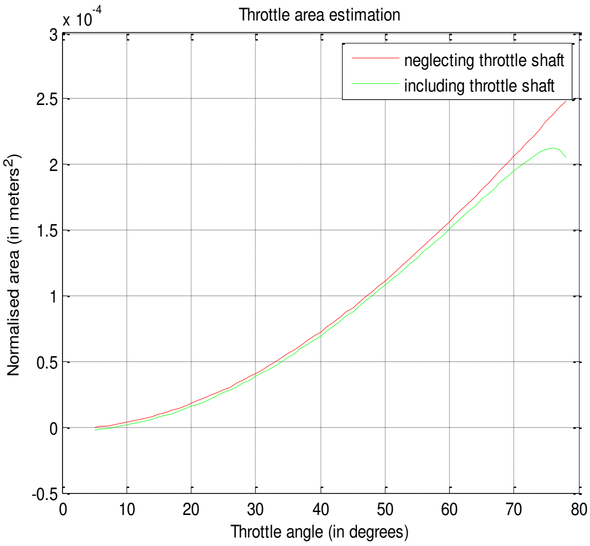


Fig 6.Normalized Throttle Area Estimation

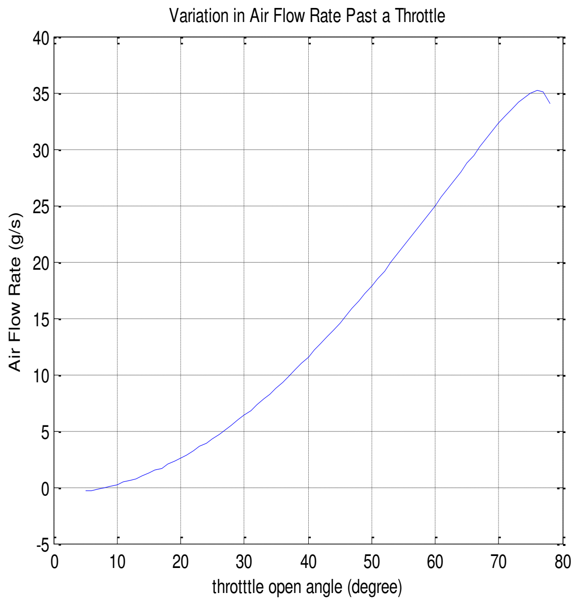


Fig 7. Variation in Air Flow Rate past A Throttle

B. *Throttle Free Analysis [9]*

Mass flow rate through inlet manifold without throttle

$$m_{th\ free} = (A \times P_0) / (\text{sqrt} (R \times T_0) \times K1 \quad (14)$$

C_{d2} coefficient of discharge at inlet valve corresponding to six Lift of valve/Diameter of valve ratio

C_{d2} = (0.0688, 0.2065, 0.293, 0.39, 0.501 and 0.57)

$$m_{th\ free} = C_{d2} \times m_{th\ free} \quad \ldots\ldots (15)$$

$$\eta_{vol} = 2 \times m_{th\ free} / (\rho \times V_d \times N) \quad \ldots(16)$$

L/D	Cd
0.021658148	0.031990475
0.043316296	0.068814826
0.064974443	0.106997006
0.086632591	0.206607417
0.108290739	0.240172975
0.129948887	0.293834418
0.151607035	0.384537464
0.173265182	0.413812279
0.19492333	0.461553583
0.216581478	0.518473818
0.238239626	0.537809318
0.259897774	0.559317345
0.281555921	0.582563395
0.303214069	0.589732738
0.324872217	0.597988344
0.346530365	0.607710407

Table 1: Lift/Diameter versus Coefficient of Discharge for the Suction Valve

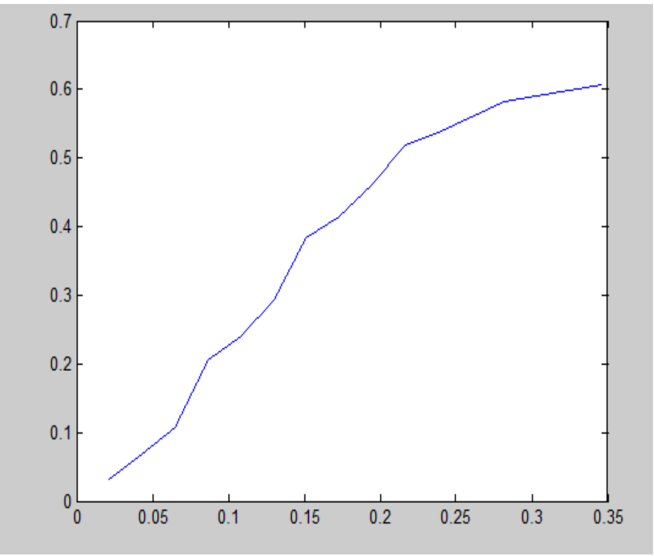


Fig 8. Lift/Diameter versus Coefficient of Discharge for the Suction Valve

IV. RESULTS AND DISCUSSION

Volumetric efficiency for four stroke single cylinder SI engine was evaluated for conventional throttled operation and throttle free operation utilizing the CVVL mechanism for full range of load and compared.

A. Volumetric Efficiency with Throttled Operation

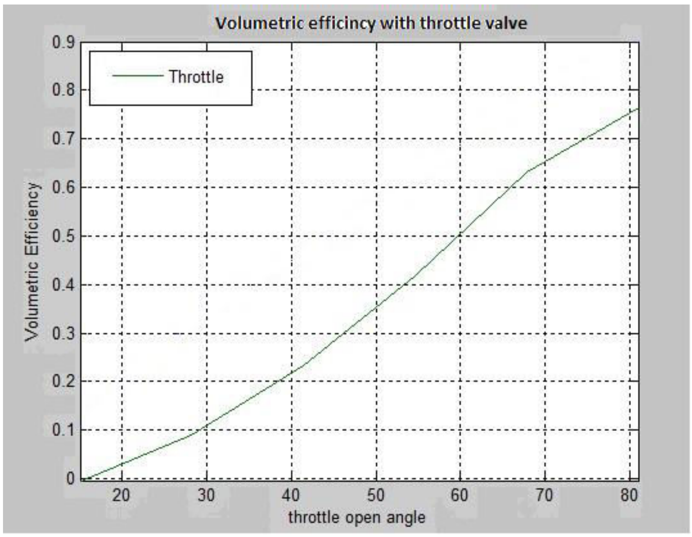


Fig 9.Volumetric Efficiency with Throttled Opearion

B. Volumetric Efficiency with Throttle free operation

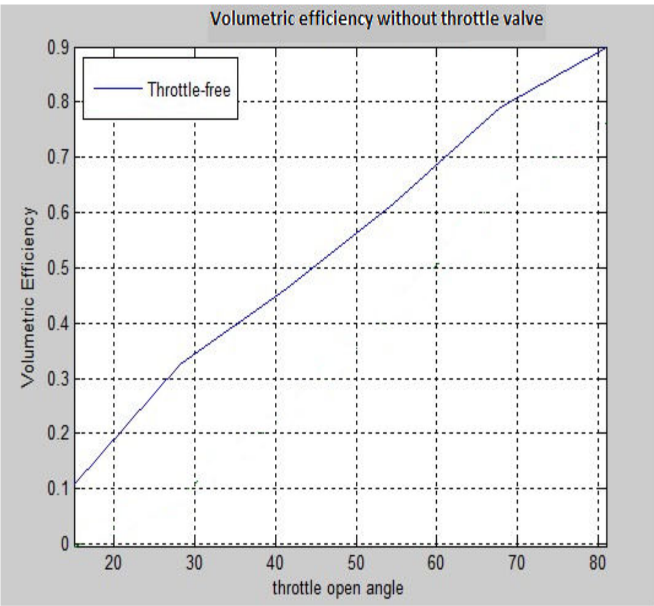


Fig 10.Volumetric Efficiency with Throttle Free Operation

C. Comparison of Volumetric Efficiency with Throttle Free and Throttled Operation

It is seen that the volumetric efficiency with throttle free operation (with CVVL mechanism) is always superior to throttled operation in SI engine.

At part load condition near Idling the improvement in volumetric efficiency is 21% whereas at 50 % load condition the improvement is 19 %. At full load condition the throttle free operation is 14% better than throttle operation.

This proves the effectiveness of CVVL mechanism more prominently at part load than the full load condition to minimize the throttling loss.

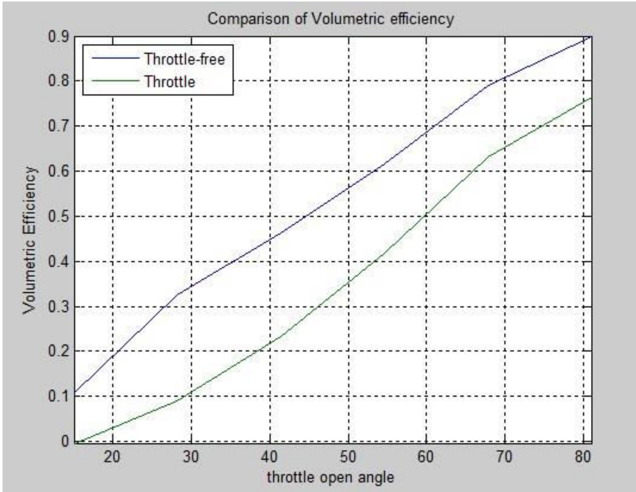


Fig 11.Comparison of Volumetric Efficiency with Throttle free and Throttled Operation

V. CONCLUSIONS

- Volumetric efficiency i.e. breathing capacity of engine measures the performance of an engine.
- Throttling reduces the amount of air induced into the engine and introduces flow losses which reduce an engine's efficiency.
- Throttle free load control is one of the method of increasing the volumetric efficiency of an engine, which reduces the mechanical losses associated with throttling.
- Volumetric efficiency with throttle free operation (with CVVL mechanism) is always superior to throttled operation in SI engine.
- At part load condition near Idling the improvement in volumetric efficiency is 21% whereas at 50 % load condition the improvement is 19 %.
- At full load condition the throttle free operation is 14% better than throttle operation.
- The effectiveness of throttle free load control mechanism is more prominent at part load than the full load condition to minimize the throttling losses.

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