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Engine Manifold Wave Action under Variable Stroke Length

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Abstract

A theoretical investigation on the pressure wave action of the manifolds of a four-stroke, direct injection (hereinafter referred to as DI), water-cooled, 4-stroke, diesel engine with variable stroke length was carried out. The study was conducted over wide range of speeds (1000 - 3000 RPM at an increment of 500 RPM) and stroke lengths (130 mm to 210 mm at an increment of 20mm). The compression ratio was kept constant by adjusting the piston bowl volume. The study showed that shorter stroke lengths created favorable pressure waves in both inlet and exhaust manifolds at lower speeds, which resulted in improved engine volumetric and thermal efficiencies. At higher speeds, the larger strokes were favorable, however, due to less time available for the suction and exhaust strokes to be executed, the efficiencies were low. Advancing valve timing was one factor that improved the engine performance with larger stroke lengths.

Keywords: diesel engine simulation, variable stroke length, variable compression ratio, manifold wave action, manifold pressure waves

1. Introduction

The main objective of the engine manifolds is to help fill the cylinder with fresh mixture at the beginning of the suction stroke as well as remove the products of combustion at the end of the exhaust stroke. Removing as much as possible of products of combustion is important from combustion point of view. If the amount of products remaining in the cylinder after scavenging process increases, the effect on the combustion process will be damaging.

The power and torque developed by an internal combustion engine can be rewritten in the following form (Heywood, 1989, pp. 56-57):

$$Power = \frac{m_a \eta_f Q_{HV} N\left(\frac{F}{A}\right)}{2} \tag{1}$$

Or

$$Torque = \frac{\eta_v \eta_f \rho_{ai} V_s Q_{HV} N\left(\frac{F}{A}\right)}{4\pi} \tag{2}$$

Referring to the above figures, one can clearly notice the dependent of both parameters on the amount of air admitted to the cylinder at a given speed (or the density of the air admitted to the cylinder). This is primarily the function of the manifold. Thus, it is clear that the main objective of any designer is to design a manifold that will allow maximum amount of air to be inducted and retained within the cylinder. Further, the other objective is to expel maximum amount of products out of cylinder to allow for more fresh mixture to be admitted.

It can also be shown that the specific fuel consumption of the engine can also be improved (reduced) by good engine manifold design. Since the more air drawn into the cylinder and exhaust been expelled out of the cylinder, the volumetric and thermal efficiencies would also improve, hence improving the specific fuel consumption.

This task of the manifold, alone, cannot do the needful job properly. Another factor that will help achieve the required goal is the proper selection of the valve timing.

Mercedes 300SL in 1954 (http://www.autozine.org) tuned intake manifold for improved engine performance. This was done by elongating the intake manifold utilizing what is known as "supercharging effect”. They attributed the improvement in the engine performance to the pressure waves generated in the intake manifold which helps compresses air into the cylinder during the intake stroke. This requires proper intake valve timing to open the intake valve just when the pressure inside the manifold is higher than the cylinder.

As for the exhaust manifold, the reverse action takes place in what is called "reverse-supercharging" or 'scavenging" where the required exhaust valve timing is needed when the pressure inside the manifold becomes lower, hence, helps in discharging more of the exhaust from the cylinder.

This short introduction shows the importance of studying the effect of any engine modifications on the wave action inside the engine's manifold to shed some light on how the new design will be affected by the engine modifications.

Reviewing the literature reveals that extensive research was conducted worldwide in the field of heavy duty diesel engines which resulted in significant improvement of engine performance and emissions. Research related to improvement of CI engine performance can be broadly classified into four main areas i.e. before combustion (e.g. air and fuel mixing and quality), during combustion (e.g. injection, distribution and chamber design), after combustion (e.g. exhaust aftertreatment and EGR) and change in engine design and operational concept (e.g. variable compression ratio, stroke length). As for the first three areas of research, great amount of work was conducted with the aim of either finding a replacement to diesel fuel or enhancing the qualities of the present diesel fuel.

This study is related to the last category i.e. modifying the engine design. This area of research concentrated on trying to modify the existing design in order to improve the engine performance and emission characteristics. The use of the common-rail technique (Tanaka and Jagata, 2004), changing the fuel injection rate (Dasantes et.al. , 2004) and pressure (Hountalas et.al., 2003), intake-port geometry (Corgard and Rietz, 2001), variable injection timing (Kouremenos et.al., 2001), pre-and post-injection timing (Benajes et.al., 2001), variable geometry turbocharging (Hawley et.al, 1999), and many other techniques have all been investigated. Again, each one has resulted in adding to the improvement of engine performance and emission characteristics, but with some penalty. One such techniques which was extensively investigated is the use of the variable stroke length or variable compression ratio technique in the spark ignition (SI) engines (Yamin and Dado, 2004, and Siewert, 1978) and the results have shown good improvement in the engine performance and emission characteristics within the range of speeds and bore-to-stroke ratios covered.

As for the use of variable stroke length on CI engines, (Yamin and Eyad, 2007, Amjad et.al. 2007, Ebrahimi, 2010 and Martyn, 2002) all discussed the use of variable stroke length on the diesel engine behavior.

This study investigates the effect of manifold pressure waves on the engine performance for variable stroke length design. The main objective is to study how varying the stroke length affects the gas pressure variation inside the cylinder for a given valve timing. It will be conducted on compression ignition engine using the Diesel-RK software (Kuleshov, 2005)

2. The study

In this study, the performance parameters were studied over speed range of (1000 rpm to 3000 rpm) and stroke lengths (130mm to 210mm) at constant compression ratio of 16.74:1 and injection timing at 20^{o} btdc. The values of the stroke length, bore-to-stroke ratio and adjusted piston bowl volume are shown in Table 1. The original cylinder diameter was 150 mm and that for the stroke length was 180mm. Constant compression ratio for all tests was achieved by adjusting the piston bowl volume and hence the clearance volume.

The shape of the piston used is shown below in Figure 1 with the injector is centrally located and the clearance height between the tip of the flat piston surface and the cylinder head (hclr) is 3mm. The piston of the left is the final shape while the raw design is shown on the right hand side with top view showing the shape and distribution of the injected fuel.

The piston bowl has volume of 149068.852mm^3, while the flat clearance volume (Vclr) is 53014.376mm^3. This makes the total minimum volume of the engine equals to 202083.228mm^3.

3. Results and Discussion

The study was divided into two parts:

1) Effect of displacement on engine performance, and

2) Effect of displacement on pressure waves in the manifold.

The results of the study are presented in Figures 2 to 11. Before starting the discussion of results, let us relate the performance formulae for 4-Stroke engines with one another as follows:

$$Power = \rho_{ai} S\, A_p \eta_f \eta_v Q_{HV} \left(\frac{F}{A}\right)\left(\frac{N}{60}\right)\left(\frac{n_c}{n_r}\right) \quad (3)$$

$$= MEP\, S\, \eta_f \eta_v K \quad (4)$$

Where

$$K = \rho_{ai}\, Q_{HV} \left(\frac{F}{A}\right)\left(\frac{N}{60}\right)\left(\frac{n_c}{n_r}\right)$$

$$MEP = \rho_{ai} \eta_f \eta_v Q_{HV} \left(\frac{F}{A}\right) \quad (5)$$

$$\eta_v = \frac{\dot{m}_a}{\rho_{ai} S A_p \left(\frac{N}{60}\right)\left(\frac{n_c}{n_r}\right)} \quad (6)$$

where, “ρai” is the air density at inlet conditions, "Ap" is the piston area, "S" is the stroke length, “ηf” is the thermal efficiency, “ηv” is the volumetric efficiency, “QHV” is the fuel’s heating value, "(F/A)" is the fuel air ratio, “N” is the engine speed (rev/min), "nc" is the number of cylinders, "nr" is the number of stroke rotations needed to complete one cycle, "MEP" is the mean effective pressure and "$\dot{m}_a$" is the actual mass flow rate into the engine.

With reference to the above equations, engine power can be improved by increasing the engine displacement, mean effective pressure and/or engine speed. Further, for a given engine speed and dimensions, more power and MEP (or Torque) can be achieved by increasing the mass or density of air (or volumetric efficiency) induced into the cylinder.

Increased displacement increases the engine displacement volume, if this is accompanied by proper manifold design, more air can be drawn into the cylinder and hence greater power could be achieved.

3.1 Effect on the Engine's Power and Torque Parameters

The effect of stroke length on the engine’s power and torque parameters is shown in Figures (2-a, 2-b, and 2-c). It is noticed from figure (2-a) that within the range of stroke length studied, the engine brake power increases as the stroke length increases. This trend is shown only at low engine speeds of 1500rpm. Beyond this speed, the trend is reversed and the engine power decreases as the stroke length increases.

With reference to equation (1) and figures (2-c, 3 and 4) it is clear that the main factors affecting the engine power are mean cycle pressure, stroke length, volumetric and thermal efficiencies as well as the density of air at inlet conditions.

At low engine speeds, the engine's filling ability increases due to relative availability of time for the suction process to be completed. This leads to increased cylinder contents of fresh mixture and its energy content. In addition to that, the thermal efficiency and mean effective pressure are not much lower compared with low stroke length.

Hence, the engine's power will greatly improve. This improvement in the thermal and volumetric efficiencies also causes the engine brake torque to be improved at low engine speeds. This is shown in Figure (2-b).

This trend is completely reversed at higher speeds. As seen from the figures, the great loss in volumetric efficiency, increased friction losses and deterioration in the engine's ability to convert energy to useful power (thermal efficiency), all lead to the rapid destruction in all power and torque parameters as the stroke length is increased. In this situation, shorter stroke lengths are preferred since the losses are low and the gain in volumetric and thermal efficiencies is high. The figures show that there is proportional relation between the changes in engine stroke length with the change in engine brake power and torque parameters.

As shown above, the engine power parameters (power, torque or MEP) changed with stroke length. One main factor for that was the change in volumetric efficiency (as well as friction loss).

In the coming sections, we will try to understand how the variation of stroke length (or bore-to-stroke ratio) affected the air dynamics inside the manifold, and hence, the volumetric and scavenging efficiencies.

3.2 Inlet and Exhaust Manifold Flow Condition

The final item to be explored in this study is the effect of the stroke length on the flow of both fresh charge as well as the exhaust products in the manifold. This is shown in Figures 5 to 10.

Mass flow through the intake valve into a cylinder is shown in Figure (5-a) and (5-b) for low and high engine speeds respectively.

It show that actual flow rate of fresh mixture into the cylinder starts at some degrees after the inlet valve opens (around 10^o to 15^o or 3-5% of the suction process). This delay in the inflow of the fresh mixture causes loss in the engine's volumetric efficiency for all stroke lengths. This negative effect is more dominant at higher stroke lengths and least at lower stroke lengths. During this period, two factors affect the flow rate of the fresh mixture. First, is the valve overlap period where both the valves are opened with relatively higher exhaust pressure than the inlet or fresh mixture one, second is the higher cylinder pressure at the time of the inlet valve open than the fresh mixture pressure. This is clearly shown in Figures (6-a) and (6-b).

These figures clearly show that the valves open at the time when the cylinder pressure is much higher than the manifold pressure. This causes the flow to be outside the cylinder instead of inside the cylinder. This period has adverse effect on the engine's breathing ability and causes the aforementioned loss in volumetric efficiency. This is also shown in Figures (7-a) and (7-b) by the negative velocity values (which means outflow of the cylinder).

The same process is repeated towards the end of the suction stroke. The cumulative effect of the loss in fresh mixture flow at both ends of the process lead to the loss in the amount of oxygen retained inside the cylinder, hence, adversely affecting the thermal efficiency of the engine.

Further noticed, is the lower values of flow rate of fresh mixtures for shorter stroke lengths compared with larger stroke lengths. This can be explained by the lower difference between the manifold and cylinder pressure which causes the flow rate (and speed) to be lower for lower stroke lengths.

Now, since the size of the volume needed to be filled at lower stroke lengths is smaller compared with larger stroke lengths, and since the amount of outflow during the initial and final stages of the process is low, the effect on the engine's breathing efficiency is lower than with higher stroke lengths, hence, improved thermal and volumetric efficiencies. However, since the cylinder volume is low at shorter stroke lengths, the work produced will be low and hence the engine's power and torque will be low.

Regarding the exhaust manifold behavior, Figures (8-10) show how the exhaust manifold is affected with the stroke length for both low and high speeds. It can be seen that at the beginning of exhaust valve opening, due to the high cylinder pressure (relative to the manifold) the mass flow rate is largely accelerated till choked flow will occurs. This condition limits the maximum flow rate till the piston reaches BDC and the cylinder pressure drops down below the critical pressure ratio value. After that the flow out of the exhaust valve is controlled by the piston movement during the exhaust stroke. Now, during the exhaust stroke, while the piston is moving towards the TDS, the maximum piston speed is reached nearly halfway the stroke, this affects the rate of exhaust flow.

It is noticed that at low engine speed, the amount of exhaust gasses discharged during the blowdown period is greater for larger stroke lengths compared with shorter one. This may be attributed to the greater cylinder contents as a result of larger piston volume and the time available for the exhaust to be discharged at low engine speeds.

This is reversed for the blowdown period at higher engine speeds. Possible reason for that may be due to the time effect, the piston reached half way in the exhaust stroke (with maximum velocity) faster for shorter stroke lengths compared with larger stroke lengths. This helps discharging more exhaust during the initial (blowdown) period. Further, as the scavenging process is improved, the effect of such improvement on the volumetric and thermal efficiencies will be positive. This is because, the move exhaust products are discharged, the cooler the engine will be at the end of the exhaust stroke and the less will be the products of combust that , if present, will adversely affect the combustion process.

This improvement of the scavenging process at higher engine speed can be explained with the help of both Figures 9 and 10.

Referring to both figures, it is noticed that at lower engine speed, there not much pressure difference between the engine cylinder and exhaust manifold. This makes the difference in the exhaust flow velocity for all stroke lengths to be low. Hence, the controlling parameters for the exhaust process will be the piston movement and time available and the amount of charge entrained during the suction stroke. This is clearly shown in Figures (9-a) and (10-a).

At higher engine speeds, as noticed in Figures (9-b) and (10-b), the higher pressure difference between the cylinder and manifold lead to greater difference in the flow velocity, hence, more of exhaust products to be removed from the cylinder.

Referring to the above discussion, one notices that larger stroke length has greater advantage over shorter one at lower engine speed. The opposite is true for shorter stroke lengths. Further, referring to the pressure waves in the manifold, it can be noticed that the opening and closing time of the inlet valve does not match well with the cylinder pressure. This might be one of the reasons for the lower engine performance at higher engine speed.

To try to solve this problem, next step of this study is to find if the engine performance can be improved by changing the valve timing. For this study, stroke length S=210 mm and engine speed N=3000 rpm was taken for example. The valve timing was varied as follows: inlet valve opening was varied from 5º-60º degrees bTDC and the same values for the exhaust valve bBDC. The result is shown in Figures 11 and 12.

Figure (11) shows that there is potential improvement if the inlet valve opens at around 40º bTDC (the original design was 16º bTDC) while the exhaust valve opening is at 65º bBDC (the original design was 60º bBDC).

Figure (12) clearly shows the improvement in the wave action after modification. Whereas in the original design, the manifold pressure starts to exceed the cylinder pressure at about 730º (i.e. 10º aTDC), the modified valve timing behavior changes at around 705º (i.e. 15º bTDC). Further, the magnitude of the pressures is less for the cylinder pressure and higher for the manifold pressure compared with the original design. This change in the pressure wave behavior changed the performance of the engine as shown in the table below.

Hence, it is clear that the valve timing has major impact on the engine behavior in terms of performance and manifold wave action behavior which must be explored further.

4. Conclusions

Within the range of stroke lengths studied, it can be conclude that:

1. Engine's performance was better for taller stroke lengths at low speed and shorter stroke lengths at higher engine speeds.
2. Shorter stroke lengths created favorable manifold conditions at low engine speeds.
3. No noticeable change in the manifold behavior for all stroke lengths at higher engine speeds.
4. Noticeable change in the engine performance was observed at taller stroke lengths and higher engine speeds when the inlet valve timing was advanced.
5. More studies should be conducted on the effect of this modification on the stress and strain induced into the engine parts.

Acknowledgment

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References

Benajes, J., Molina, S., & García, J. M. (2001). Influence of pre-and post-injection on the performance and pollutant emissions in a HD diesel engine (No. 2001-01-0526). SAE Technical Paper. https://doi.org/10.4271/2001-01-0526

Corgard, D. D., & Reitz, R. D. (2001). Effects of alternative fuels and intake port geometry on HSDI diesel engine performance and emissions (No. 2001-01-0647). SAE Technical Paper. https://doi.org/10.4271/2001-01-0647

Desantes, J. M., Benajes, J., Molina, S., & Gonzalez, C. A. (2004). The modification of the fuel injection rate in heavy-duty diesel engines. Part 1: Effects on engine performance and emissions. *Applied Thermal Engineering, 24*(17), 2701-2714. https://doi.org/10.1016/j.applthermaleng.2004.05.003

Ebrahimi, R. (2010). Performance of an irreversible Diesel cycle under variable stroke length and compression ratio. *Journal of American Science, 6*(1), 58-64.

Hawley, J. G., Wallace, F. J., & Cox, A. (1999). The Demonstration of Variable Geometry Turbocharging for Improved Engine Performance. In 5th Asia-Pacific International Symposium on Combustion and Energy Untilization (pp. 314-321). University of Bath.

Heywood, J. B. (1989). Internal Combustion Engine Fundamentals, McGraw Hill, New York.

Hountalas, D. T., Kouremenos, D. A., Binder, K. B., Schwarz, V., & Mavropoulos, G. C. (2003). Effect of injection pressure on the performance and exhaust emissions of a heavy duty DI diesel engine (No. 2003-01-0340). SAE Technical Paper. https://doi.org/10.4271/2003-01-0340

Kouremenos, D. A., Hountalas, D. T., Binder, K. B., Raab, A., & Schnabel, M. H. (2001). Using advanced injection timing and EGR to improve DI diesel engine efficiency at acceptable NO and soot levels (No. 2001-01-0199). SAE Technical Paper. https://doi.org/10.4271/2001-01-0199

Kuleshov, A. S. (2005). Model for predicting air-fuel mixing, combustion and emissions in DI diesel engines over whole operating range (No. 2005-01-2119). SAE Technical Paper. https://doi.org/10.4271/2005-01-2119

Roberts, M. (2003). Benefits and challenges of variable compression ratio (VCR) (No. 2003-01-0398). SAE Technical Paper. https://doi.org/10.4271/2003-01-0398

Shaik, A., Moorthi, N. S. V., & Rudramoorthy, R. (2007). Variable compression ratio engine: A future power plant for automobiles-an overview. Proceedings of the Institution of Mechanical Engineers, Part D. *Journal of Automobile Engineering, 221*(9), 1159-1168.

Siewert, R. M. (1978). Engine combustion at large bore-to-stroke ratios (No. 780968). SAE Technical Paper. https://doi.org/10.4271/780968

Tanaka, Y., & Nagata, K. (2004). 1800 bar Common Rail System for Diesel Engine. *Journal-society of Automotive Engineers of Japan, 58*(4), 19-24.

Yamin, J. A., & Abu-Nameh, E. S. (2007). Environmental assessment of a diesel engine under variable stroke length and constant compression ratio. *American Journal of Applied Sciences, 4*(5), 257-263.

Yamin, J. A., & Dado, M. H. (2004). Performance simulation of a four-stroke engine with variable stroke-length and compression ratio. *Applied energy, 77*(4), 447-463. https://doi.org/10.1016/S0306-2619(03)00004-7

Table 1. Engine dimensions used in the study

Stroke Length (mm)	Bore/Stroke Ratio (ND)	Piston Bowl Volume (cm^3)
130	1.15	92.97
150	1.00	115.37
170	0.88	137.89
180	0.83	149.10
190	0.79	160.26
210	0.71	182.68

Table 2. Relative change in engine performance at 3000 rpm

	Brake Power (kW)	Brake Torque (N-m)	Brake SFC (kg/kW-hr)	Brake Thermal Eff (%)	Volumetric Efficiency (%)
Before	67.6	215.08	2.88	11.6	52.7
After	78.1	248.6	0.654	13.0	54.35

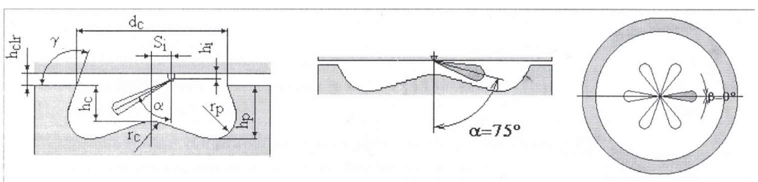

Figure 1. Piston and cylinder configuration used in this study.

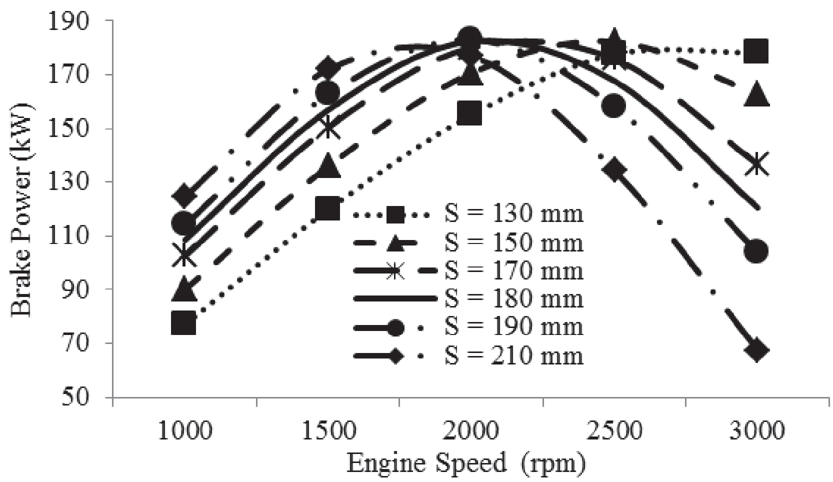

Figure (2-a). Variation of engine power with stroke length at different engine speeds

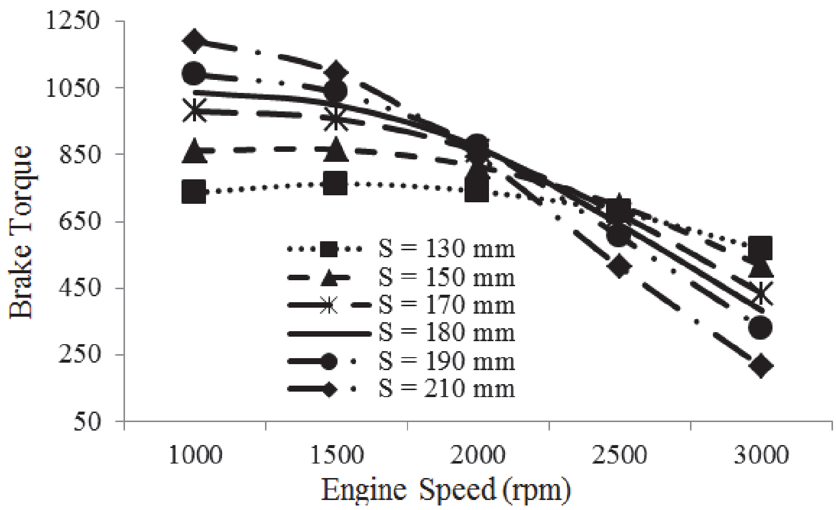

Figure (2-b). Variation of engine torque with stroke length at different engine speeds.

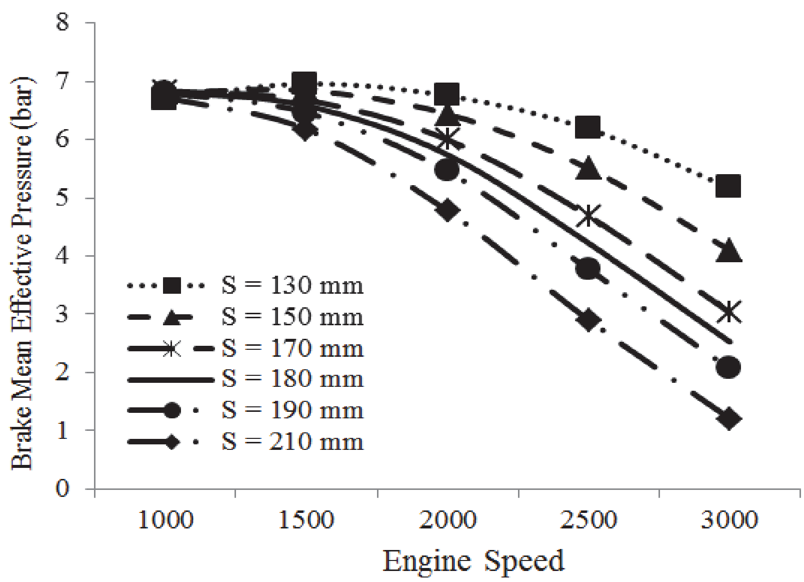

Figure (2-c). Variation of engine mean pressure with stroke length at different engine speeds

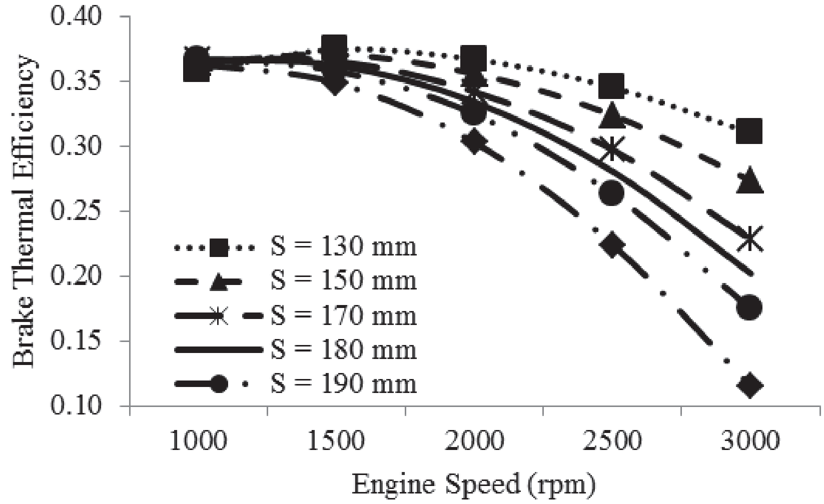

Figure 3. Variation of thermal efficiency with stroke length at different engine speeds

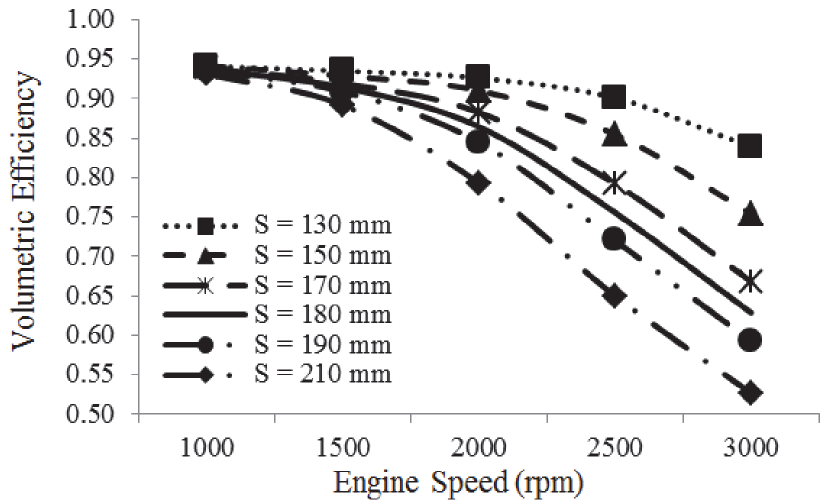

Figure 4. Variation of volumetric efficiency with stroke length at different engine speeds

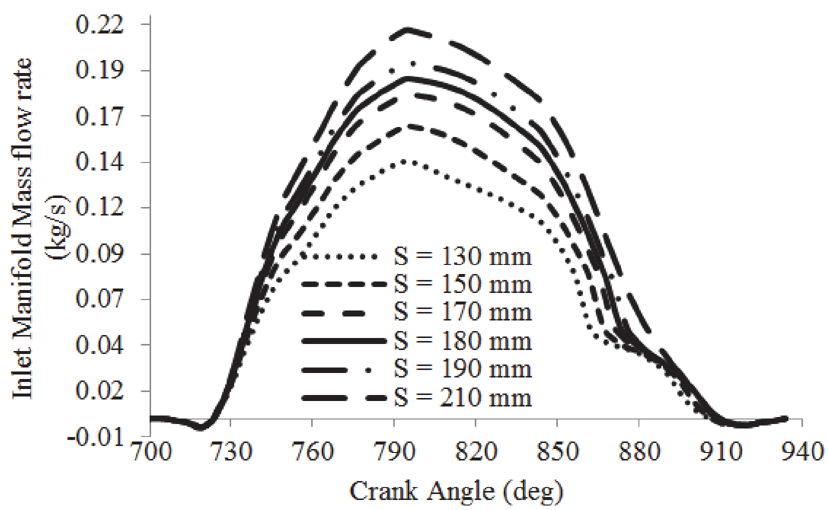

Figure (5-a). Mass flow rate of fresh charge in the inlet manifold at 1000 rpm

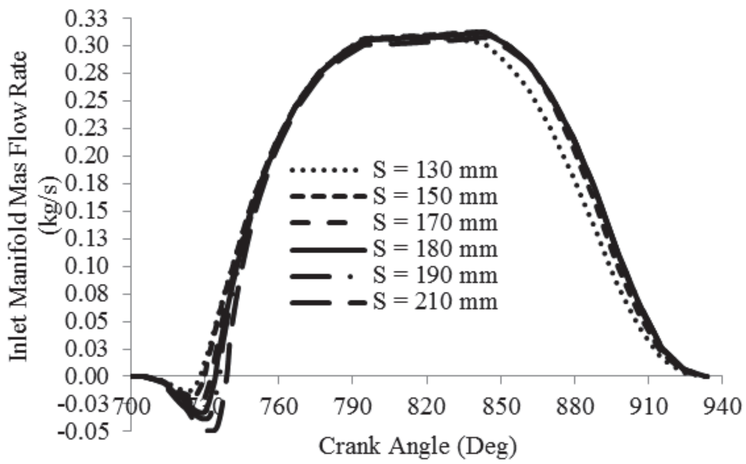

Figure (5-b). Mass flow rate of fresh charge in the inlet manifold at 3000 rpm.

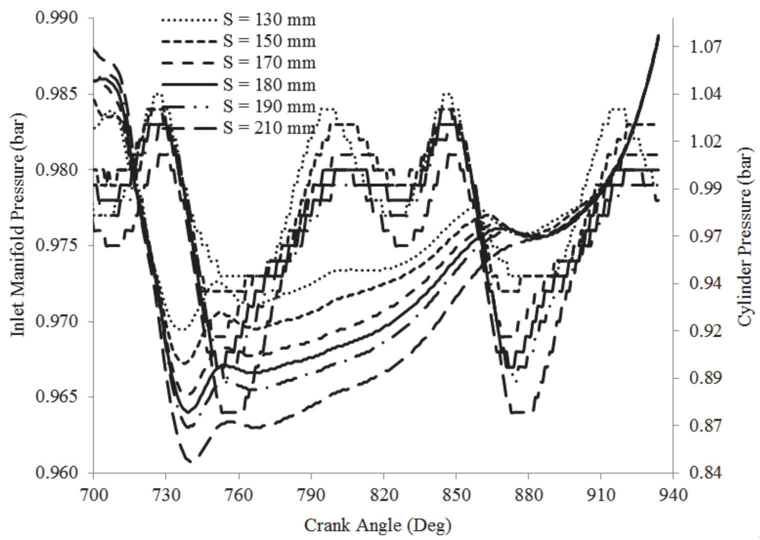

Figure (6-a). Inlet manifold pressure at 1000 rpm

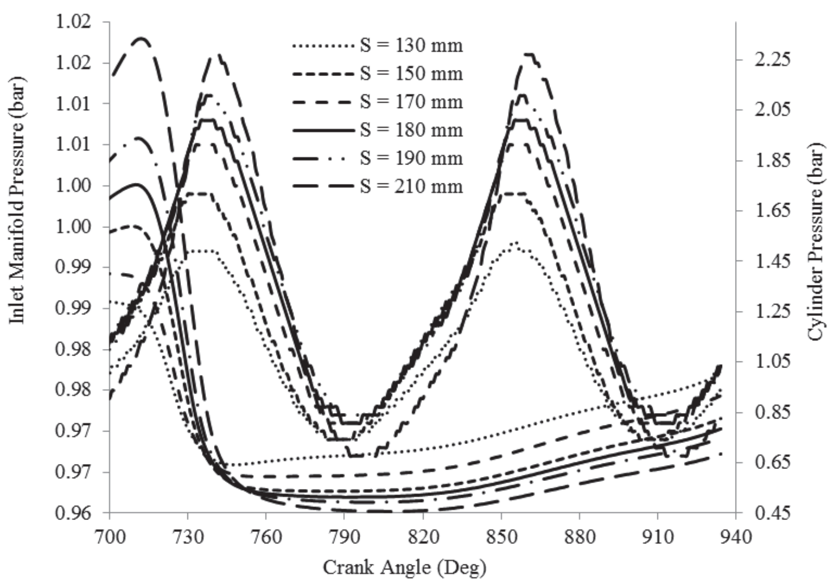

Figure (6-b). Inlet manifold pressure at 3000 rpm

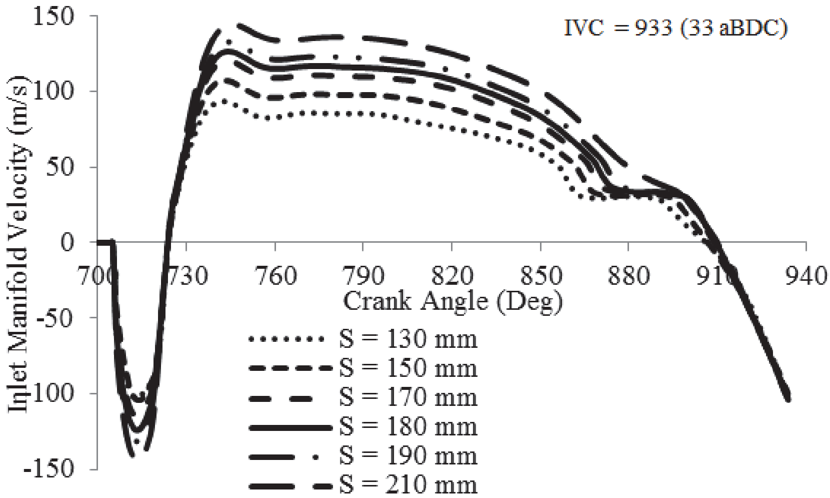

Figure (7-a). Inlet manifold air velocity at 1000 rpm

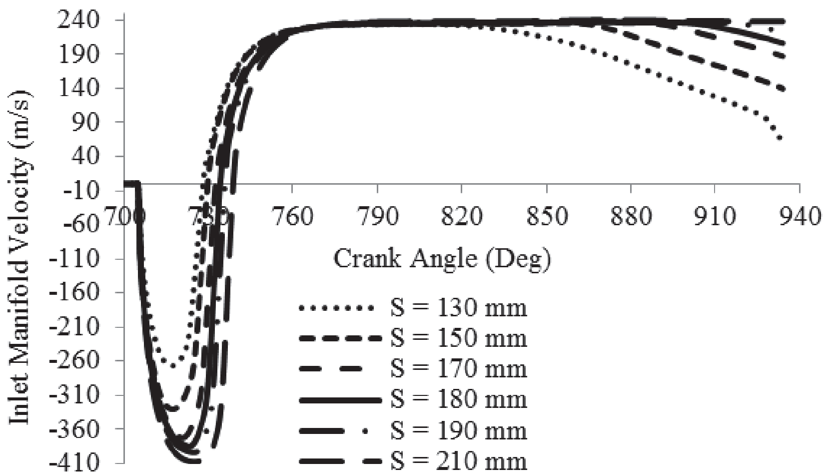

Figure (7-b). Inlet manifold air velocity at 3000 rpm

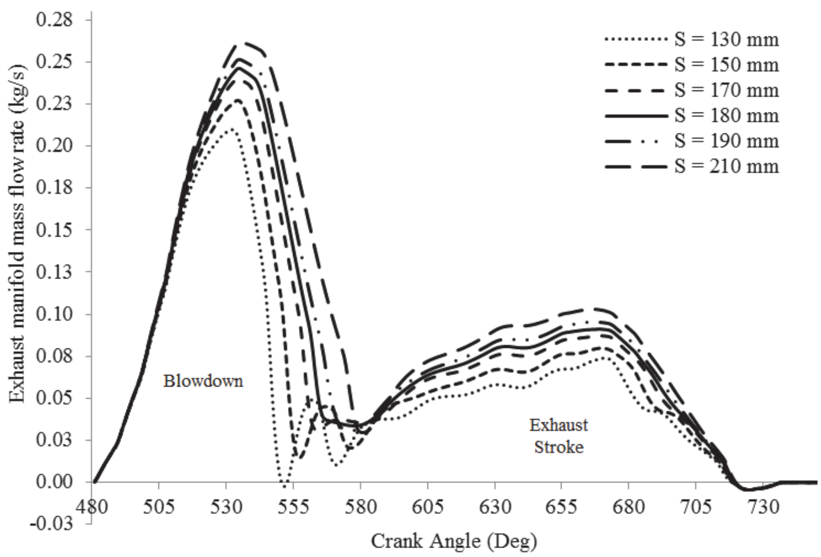

Figure (8-a). Mass flow rate of exhaust products in the exhaust manifold at 1000 rpm

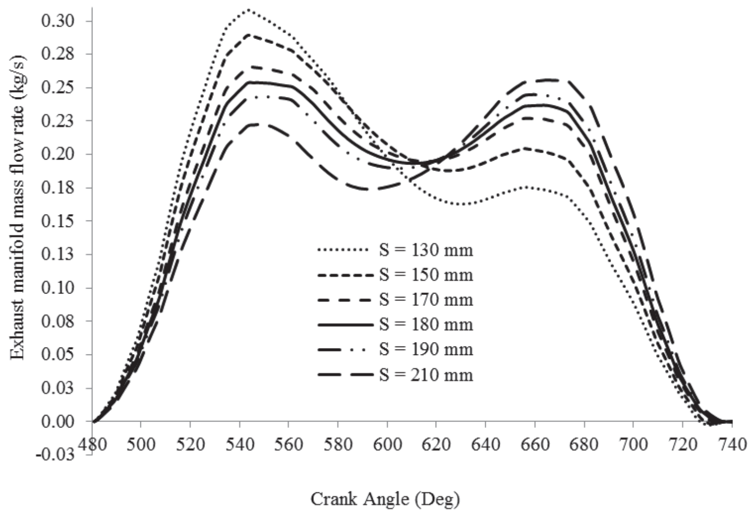

Figure (8-b). Mass flow rate of exhaust products in the exhaust manifold at 3000 rpm

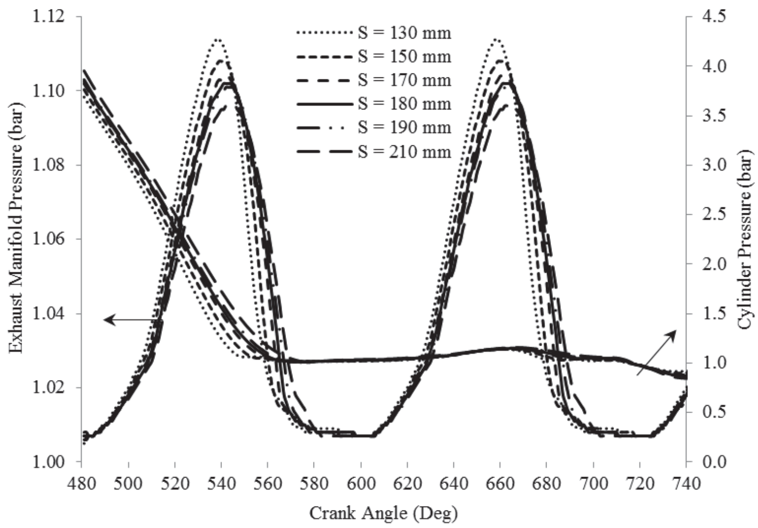

Figure (9-a). Exhaust manifold pressure variation at 1000 rpm

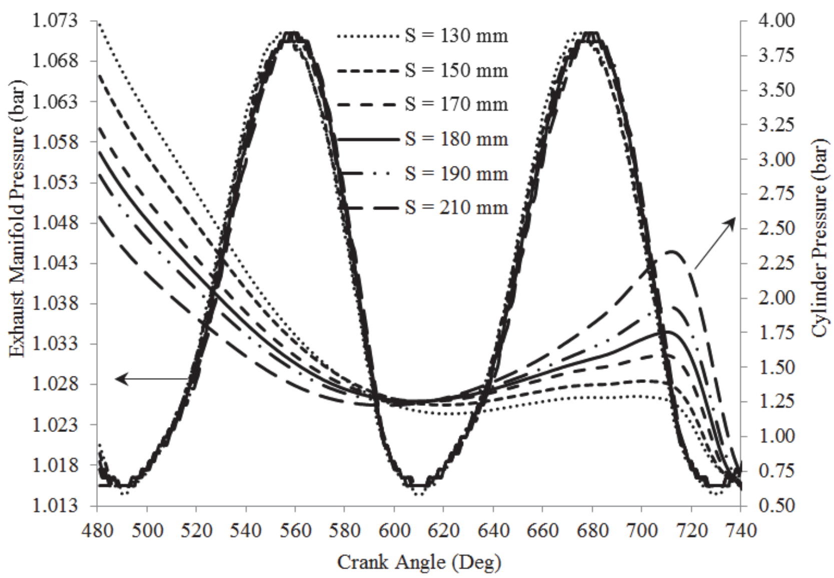

Figure (9-b). Exhaust manifold pressure variation at 3000 rpm

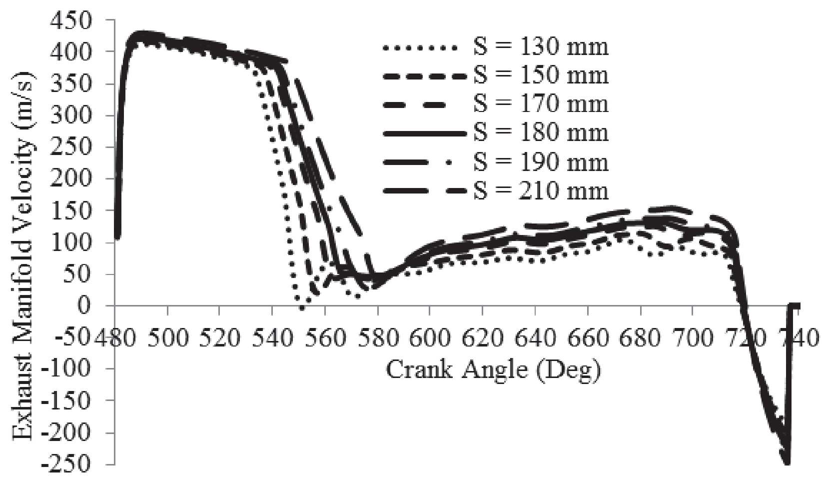

Figure (10-a). Exhaust products velocity at 1000 rpm

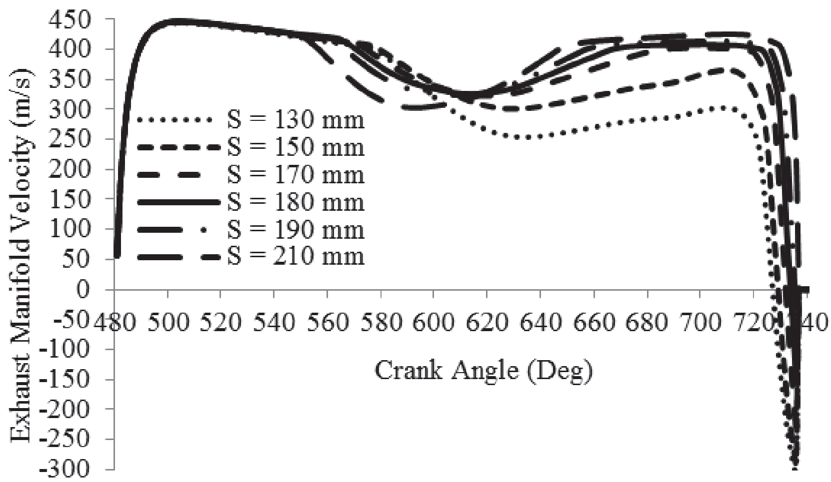

Figure (10-b). Exhaust products velocity at 3000 rpm

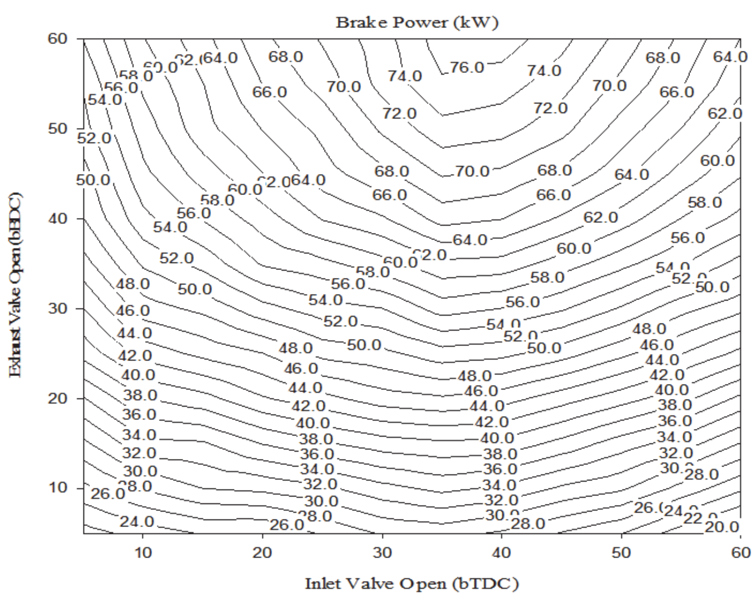

Figure 11. Variation of brake power with inlet/exhaust valve opening

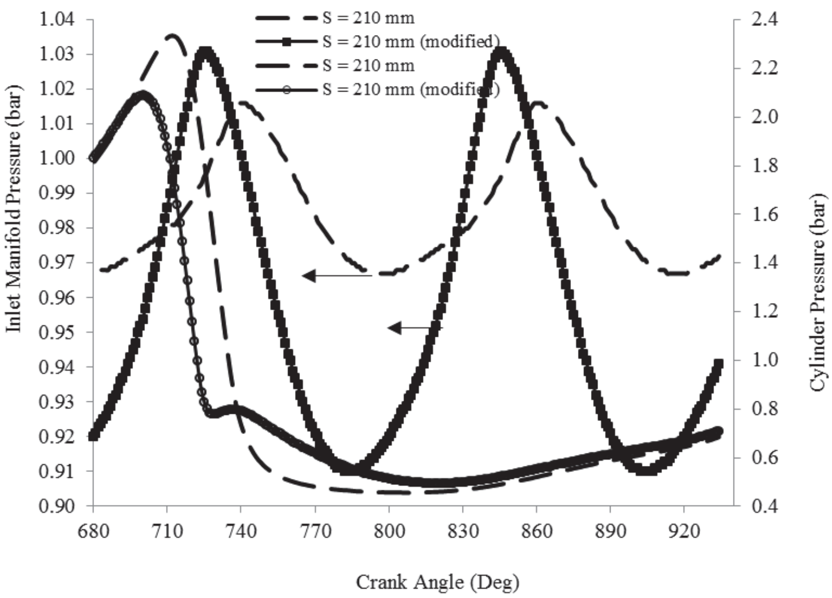

Figure 12. Manifold and cylinder pressures' comparison at 3000 rpm