

# Design of Cylinder Block of an 80cc Spark Ignition (SI) Aluminum Engine.

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## ABSTRACT

The design of the cylinder block is critical to the production of internal combustion (IC) engines as it houses many other components that make up the IC engine system. Cylinder blocks are designed to withstand the high pressure and temperature conditions that the engine would be subjected to during operation and also designed to transfer the unused heat effectively in order to avoid the metal temperature from reaching the critical limit.

In designing the cylinder block of this study, parameters such as: diameter of cylinder bore; length of stroke; brake power; indicated power; engine torque; compression ratio; mechanical efficiency of the engine; frictional power; brake mean effective pressure; maximum engine speed; number of crank revolution; clearance volume; circumferential stress; longitudinal stress; wall thickness; and bolt or stud diameter, were designed. The designed parameter values were found to be within the recommended range of values for the engine capacity found in standard texts.

(Keywords: internal combustion, IC, cylinder block, swept volume)

## INTRODUCTION

The cylinder blocks are critical components of an engine which performs number of functions hence its design must satisfy a number of requirements like containment and guide of the trunk piston. The internal design of the cylinder block must be extremely precise, because all parts must fit and be able to function properly once the entire engine is assembled (Nguyen, 2005). It is the strongest and relatively largest component of the engine as

it constitutes 20 -25% of the total weight of the IC engine (Kaey, 2002). The block serves as the structural framework of the engine and also carries the mounting pad that supports the engine in the chassis (Kalpakjian, 1997). The overall external dimensional design of the engine also has to fit properly in the car frame (Feng, Ferrick and Campbell, 2002).

In designing IC engines, three parts of the cylinder block which include the cylinder, water jacket (the spaces between the cylinder bores and the outer shell of the block), and the frame are considered. Whereas in medium engines these three parts may be manufactured as different pieces, in smaller engines however, the cylinder and the jacket can be cast as one piece. But in larger engines and most high speed engines separate cylinder liners or sleeves are used (Sharma and Aggarwal, 2012). Separate liners can easily be replaced in the event of wear and tear and therefore more economical.

The cylinder liners are made in two types: the wet and dry liners. The wet liner has the water jacket in direct contact with the outer wall of the liner, while the dry liner is pressed into the cylinder. Engines that have bore diameter of over 13cm are mostly made using the wet type of liner (Sharma and Aggarwal, 2012). A good grade grey cast iron with homogeneous and closed grain structure, nickel cast iron, nickel chromium cast iron, nickel chromium cast steel with molybdenum in some cases are suitable materials for cylinder liners.

Thermal stresses and pressure stresses are the two types of stresses that are induced in the cylinder liner. It has been determined that about 60 per cent of the heat loss to cooling water flows through the cylinder liner proper. This includes

the heat flowing from the piston head to the liner through the piston rings. Wear and distortion are the two main reasons why liners fail (Sharma and Aggarwal, 2012).

These parameters are either standard assumptions that are employed in designs or are parameters being designed.

## METHODOLOGY

The actual calculations involved in the design of the single cylinder aluminum block using formula from standard tests were done and the designed values were compared with available ranges of similar block type.



Figure 1: Single Cylinder Block.

## Specifications

The single cylinder block as shown in Figure1, consists of parameters that are listed in Table 1.

Table 1: Typical Design and Operating Data for IC Engines (Heywood, 1988).

Engine Type	Operating Cycle	Compression Ratio	Bore (m)	Rated Maximum					
				Stroke/bore	Speed, (rpm)	Bmep, (atm)	Power/unit volume, kW/dm <sup>3</sup>	Weight/Power ratio, kg/kW	Approx. Best bsfc, g/kW.h
<b>Spark-ignition Engines:</b>									
Small (e.g., Motorcycles)	2S, 4S	6 - 11	0.05-0.085	1.2- 0.9	4500-7500	4 - 10	20 - 60	5.5- 2.5	350
Passenger Car	4S	8 - 10	0.07 - 0.1	1.1- 0.9	4500-6500	7 - 10	20 - 50	4- 2	270
Trucks	4S	7 - 9	0.09 - 0.13	1.2- 0.7	3600-5000	6.5 - 7	25 - 30	6.5- 2.5	300
Large Gas Engines	2S, 4S	8 - 12	0.22 - 0.45	1.1- 1.4	300-900	6.8 - 12	3 - 7	23- 35	200
Wankel Engines	4S	≈ 9	0.57 dm <sup>3</sup> /chamber		6000-8000	9.5 - 10.5	35 - 45	1.6- 0.9	300
<b>Diesel Engines:</b>									
Passenger Cars	4S	17 - 23	0.075 - 0.1	1.2- 0.9	4000-5000	5 - 7.5	18 - 22	5- 2.5	250
Trucks (NA)	4S	16 - 22	0.1 - 0.15	1.3- 0.8	2100-4000	6 - 9	15 - 22	7- 4	210
Trucks (TC)	4S	14 - 20	0.1 - 0.15	1.3- 0.8	2100-4000	12 - 18	18 - 26	7- 3.5	200
Locomotive, Industrial, Marine	4S, 2S	12 - 18	0.15 - 0.4	1.1- 1.3	425-1800	7 - 23	5 - 20	6- 18	190
Large Engines, Marine and Stationary	2S	10 - 12	0.4 - 1	1.2- 3	110-400	9 - 17	2 - 8	12- 50	180

Other standard range of values for SI engines (Heywood 1988), are:

- (i) Maximum Mean Piston Speed,  $S_p$  should be between 8 to 15m/s.
- (ii) Compression Ratio,  $r_c$  should be from 8 to 12.
- (iii) Bore - stroke Ratio is 0.8 to 1.2.
- (i) Number of Crank Revolutions,  $n_g$  is 2 for a 4-stroke cycle and 1 for a 2-stroke cycle.
- (ii) Mechanical Efficiency,  $\eta_m$  is 90% for speeds between 1800 to 2400 rpm and 75% at maximum rated speed  $N$  of about 3000 rpm.
- (iii) Brake Mean Effective Pressure,  $b_{mep}$  is between 850 to 1050 kPa at maximum engine speed of 3000 rpm and decreases with about 10 - 15 % at maximum rated power.

## Assumptions

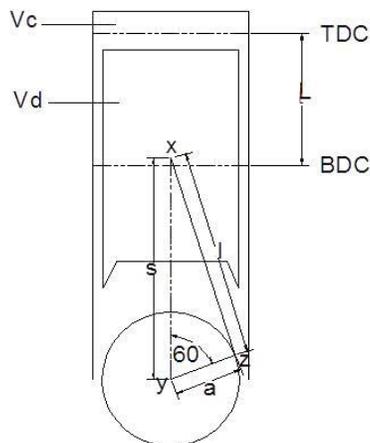
For this study, parameters of cylinder block with volumetric displacement of 80cc were designed with respect to the following basic assumptions from the design values stated above.

- (i) Bore to Stroke ratio:  
Bore diameter, B < Length of stroke, L
- (ii) Operating Engine Cycle: 4-stroke Cycle
- (iii) Maximum Rated speed: 3000 rpm  $\cong$  50 rps.
- (iv) Mechanical Efficiency,  $\eta_m$ : 75%  $\cong$  0.75
- (v) Compression Ratio,  $r_c$ : 10

## ANALYTICAL DESIGN

### Swept Volume of the IC Engine Cylinder

The swept volume ( $V_d$ ) also known as the displaced volume determines the capacity of the engine. It is the volume swept by all the pistons inside the cylinders of a reciprocating engine in a single movement from top dead center (TDC) to bottom dead center (BDC) as shown in Figure 2. It is commonly specified in cubic centimeters (cc or  $\text{cm}^3$ ), liters (l). The swept volume is different from the maximum volume of the combustion chamber because it is the combination of the swept volume and the clearance volume that make up the maximum volume of the combustion chamber (Heywood, 1988).



**Figure 2:** Schematic Diagram of Reciprocating IC Engine.

The swept volume  $V_d$  can be mathematically determined as:

$$V_d = \frac{N \pi B^2 L}{4} \quad (1)$$

Where, N = number of cylinder  
B = diameter of bore  
L = length of stroke

The swept volume for this study is 80cc as this is the capacity of the engine to be developed.

### Clearance Volume of the IC Engine Cylinder

The clearance volume  $V_c$  occurs when the piston is in its uppermost position and at this point, the volume of the combustion chamber is at its minimum. According to Heywood (1988), the clearance volume ( $V_c$ ) can be determined from the compression ratio  $r_c$  which is defined as the ratio of the maximum volume to the minimum volume in the system. The compression ratio is determined as thus:

$$r_c = \frac{\text{maximum cylinder volume}}{\text{minimum cylinder volume}} = \frac{V_d + V_c}{V_c} \quad (2)$$

Typical values of  $r_c$  is between 8 to 12 for small spark ignition engines.

For this design,  $r_c$  of 10 is selected. Therefore, after appropriate substitution,

$$V_c = 8.89\text{cc}$$

### Design of the Cylinder Bore Diameter, Stroke Length and Length of Cylinder

The diameter of the cylinder bore B can be determined from the swept volume using Equation (1).

$$V_d = \frac{N \pi B^2 L}{4}$$

But ratio of cylinder bore to stroke length according to Equation (3) is:

$$R_{bs} = \frac{B}{L} \quad (3)$$

$R_{bs}$  values for small and medium-size engines is from 0.8 to 1.2 (Heywood, 1988). However, for this design,  $R_{bs}$  value of 0.9 is assumed. Therefore:

$$0.9 = \frac{B}{L}$$

$$B = 0.9L$$

Substituting the value of B,  $V_d$  and N into Equation (1),

Where N = 1 because it is a one cylinder engine. Length of cylinder = stroke length + clearance on both sides = stroke length + (10 to 15% of the stroke length)  
The cylinder bore diameter is **45.10mm**, stroke length is **50.10 mm** and the cylinder length is **57.62mm**.

### Design of the Engine Power

The power developed by the pressure of the gas acting on the piston in the engine is the engine power or the indicated power of the engine. It has an indirect proportionality function with the mechanical efficiency  $\eta_m$  of the engine and the engine brake power as shown in Equation (4) Heywood (1988).

$$\text{Mechanical efficiency, } \eta_m = \frac{\text{Brake power, } P_b}{\text{Engine power, } P_e} \quad (4)$$

Standard values for a modern automotive engine at wide-open or full throttle are 90% at speeds less than 30 to 40 rev/sec (i.e. 1800 to 2400 rev/min), decreasing to 75% at maximum rated speed of about 3000 rev/min. The engine power can be determined as thus:

$$P_b = 2\pi TN \quad (5)$$

where, N is in rev/sec.

According to Heywood (1988), the mean effective pressure can be expressed in terms of torque as follows:

$$mep(Kpa) = \frac{6.28 n_g T(N.m)}{V_d(dm^3)} \quad (6)$$

Equation (6) can be rewritten in terms of brake mean effective pressure  $b_{mep}$  as:

$$b_{mep}(kPa) = \frac{6.28 n_g T_b(N.m)}{V_d(dm^3)} \quad (7)$$

For naturally aspirated spark ignition engines, maximum values of  $b_{mep}$  are in the range of 850 to 1050  $kPa$  ( $\sim 125$  to  $150$   $lb/in^2$ ) at the engine speed where maximum torque is obtained (about 3000rpm or 50rev/sec). Therefore, for this design,  $b_{mep} = 1000 kPa$  is selected. For four stroke engine  $n_g = 2$ .

The maximum brake torque is:

$$T_b = 6.37Nm.$$

Substituting the value of the maximum brake torque  $T_b$  into Equation (5), the brake power is calculated as thus,

$$P_b = 2.0kW.$$

Heywood (1988), also stated that the standard range of values for engine power of motorcycle and scooters engine is from 0.75 to 70kW. The engine power or indicated engine power is calculated from Equation (4) to be:

$$P_e = 2.67kW.$$

The design of the power of the engine in this study which is 2.67kW falls within the acceptable standard range of values for small engine capacity as validated by Pulkrabek (2003), which stated that engine power ranges from 1.5 to 5kW (2 to 7hp) for lawn mowers, chain saws, and snow-blower engines.

### Design of the Mean Effective Pressure of the Engine

The mean effective pressure is the ratio of the net work done to the displacement volume of the piston. It is a valuable measure of an engine's capacity to do work that is non-dependent on the engine displacement. Mathematically, the mean effective pressure is defined as:

$$p_{mep} = \frac{T n_g}{V_d} 2\pi \quad (8)$$

where, T = engine torque

$n_g$  = number of revolution per cycle ( for a 4-stroke engine  $n_c = 2$ , for a 2-stroke engine  $n_c = 1$ )

$V_d$  = displacement volume,  $m^3$

The mean effective pressure of the engine can also be determined as thus:

$$P = \frac{\pi B^2 L n_g N p_{max}}{4 \times 60} \quad (9)$$

where, B = diameter of cylinder bore  
 L = stroke length  
 $n_g$  = number of revolution per cycle ( for a 4-stroke engine  $n_g = 2$ , for a 2-stroke engine  $n_g = 1$ )  
 N = speed of rotation, rev/sec

$p_{max}$  = mean effective pressure

$$p_{max} = 3.33 \text{ kN/m}^2.$$

### Design of the Cylinder Wall Thickness

According to Sharma and Aggarwal (2012), cylinder wall thickness,  $t$  can be determined with the relationship:

$$t = \frac{P_{max} \times B}{2f_c} + k \quad (10)$$

where,  $P_{max}$  = maximum gas pressure, N/mm<sup>2</sup>  
 B = cylinder bore, mm  
 $f_c$  = Maximum hoop stress, N/mm<sup>2</sup>  
 k = reboring factor.

$P_{max}$  is between 3.1 to 3.5 N/mm<sup>2</sup>. For this design,  $P_{max}$  value of 3.5 N/mm<sup>2</sup> is selected. From the reboring factor  $k$  values in mm for the different bore sizes,  $k$  value of 1.5 is selected for this design because the bore diameter is 45.10 mm. Maximum hoop stress  $f_c$  is equal to 35 to 105 N/mm<sup>2</sup> depending on the size of the bore, with larger values used for smaller bores.  $f_c$  value of 51 N/mm<sup>2</sup> is selected for this design because of the small size of its bore.

$$t = 3.05 \text{ mm.}$$

According to empirical relation, for liners of oil engines (Sharma and Aggarwal, 2012),

$$t \geq \frac{B}{15} \quad (11)$$

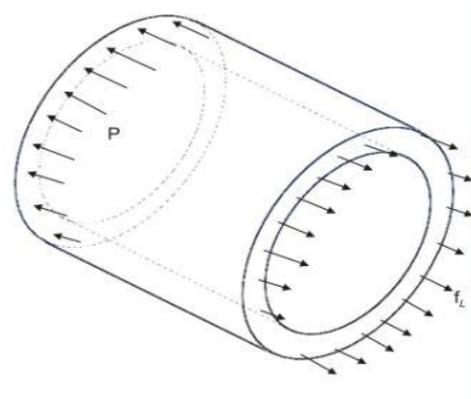
$$t = 3.01 \text{ mm.}$$

Therefore, the designed wall thickness of the cylinder block which is 3.05 mm is greater than the empirical value of 3.01 mm. Thus the designed wall thickness is within the accepted range.

### Determination of Longitudinal and Circumferential Stresses in the Cylinder Wall

The cylinder wall is subjected to gas pressure and piston side thrust. The piston side thrust tends to bend the wall however the stress in the wall due to side thrust is very small and negligible. Longitudinal and circumferential or hoop stresses are two types of stresses that are set up by gas pressure and they act at right angle to each other as shown in Figures 3 and 4, respectively. The longitudinal stress is usually small and negligible. If the cylinder walls are thin and the ratio of the thickness to the internal diameter is less than 0.05, then it can be assumed that the hoop and longitudinal stresses are constant across the thickness of the cylinder. It can also be assumed that the radial stress is small and negligible (Sharma and Aggarwal, 2012). Equation (12) is used to determine the action of longitudinal stress on the cylinder wall.

$$f_l = f_t - \frac{f_c}{m} \quad (12)$$



**Figure 3:** Direction of Longitudinal Stresses acting on the Walls of a Cylinder.

The apparent longitudinal stress  $f_{al}$  can be determined as follows:

$$f_{al} = \frac{\text{force}}{\text{area}} = \frac{[(\pi B^2 / 4) \times P_{max}]}{[\pi (B_o^2 - B^2)] / 4} = \frac{P_{max} \times B^2}{(B_o^2 - B^2)} \quad (13)$$

Where,  $B_o$  = cylinder outside diameter,  
 $B_o = B + 2t$   
 $B_o = 51.2 \text{ mm.}$

Also substituting,

$$f_{ai} = 12.12\text{N/mm}^2$$

Also the apparent circumferential stress is,

$$f_{ac} = \frac{\text{force}}{\text{area}} = \frac{p_{\max} \times B}{2t} \quad (14)$$

$$f_{ac} = 25.88\text{N/mm}^2$$

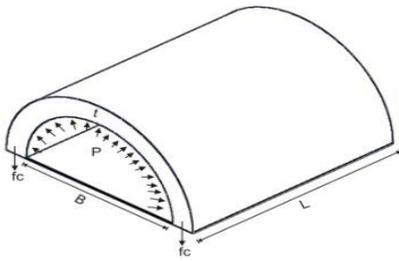
Substituting the apparent longitudinal stress into Equation (15):

$$f_i = f_{ai} - \frac{f_{ac}}{m}, \text{ where } \frac{1}{m} = \text{Poisson's ratio} \quad (15)$$

$$f_i = 5.65\text{N/mm}^2.$$

The circumferential stresses on the wall of the cylinder as shown in Figure 4 can be determined using Equation (16):

$$f_c = f_{ac} - \frac{f_{ai}}{m} \quad (16)$$



**Figure 4:** Direction of Circumferential or Hoop Stresses acting on the Wall of a Cylinder.

Substituting the apparent circumferential stress into Equation (16):

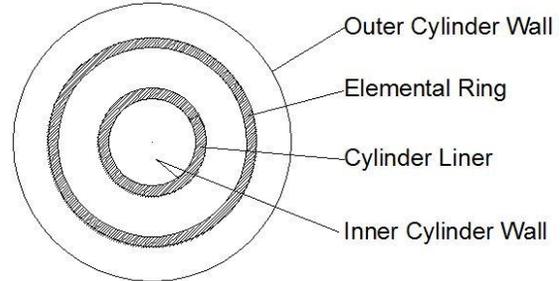
$$f_c = f_{ac} - \frac{f_{ai}}{m}, \text{ where } \frac{1}{m} = \text{Poisson's ratio}$$

$$f_c = 22.85\text{N/mm}^2.$$

### Expansion of the Cylinder

To effectively carry out proper analysis on the possible expansion of the cylinder, the cylinder must be considered to consist of elemental rings as shown in Figure 5. The expansion of the inner ring is restricted due to the negative temperature gradient from the inner walls to the outer walls. This means that the colder outside wall will restrict the expansion of the inner wall.

It is assumed that the expansion of the cylinder cannot exceed that of the outer wall of the cylinder. The expansion of the cylinder  $\Delta B$  can be determined by Equation (17).



**Figure 5:** Schematic Diagram of the Cylinder as Elemental Ring.

$$\Delta B = \alpha B_o \Delta T \quad (17)$$

where,  $\alpha$  = coefficient of linear expansion of the material (this parameter depends only on the material the object is made from, and it is  $22.2 \times 10^{-6}$  mm/mmK for aluminum.

$B_o$  = cylinder block outer diameter (determined as 51.12mm.

$\Delta T$  = change in temperature

$$\text{but, } \Delta T = T_f - T_i \quad (18)$$

where,  $T_f$  = final temperature (block outer diameter)

$T_i$  = initial temperature (room temperature)

Room temperature of 30°C is assumed for this study.

The block outer wall temperature,  $T_f = 80^\circ\text{C}$

Substituting these values into Equation (17):

$$\Delta B = 0.057\text{mm}.$$

### Design of the Diameter of Cylinder Bolt or Stud

The cylinder bolt or studs are used to fasten top engine cylinders to cylinder blocks. The cylinder

bolt diameter  $d_c$  can be determined using Eqn. (19), that is:

$$d_c = B \sqrt{\frac{P_{max}}{z f_t}} \quad (19)$$

but,  $z$  = the number of stud may be taken as  $\left(\frac{B}{100} + 4\right)$  to  $\left(\frac{B}{50} + 4\right)$ .

$P_{max}$  = maximum gas pressure (a value of  $3.5\text{N/m}^2$ ) has been selected for previous design.

$$d_c = 5.93\text{mm} \sim 6\text{mm}.$$

The diameter of the cylinder bolt is calculated to be 6mm which according to Sharma and Aggarwal (2012) should not be more than 16mm. Therefore, the designed value of  $d$  is within the recommended range of values.

Table 2 shows a summary of the designed values and standard values for similar cylinder block size. Also the flanges thickness should not be less than 1.1 to 1.25 $t$  that is **3.36mm to 3.81mm**. However, For this study the range of thickness of 1.2 to 1.4 $t$  was selected which gave **3.66mm to 4.27mm**.

The distance of the end of the flange from the centre of the stud or bolt should not be more than  $d + 6\text{mm}$  which is **12mm**, and not less than 1.5 $d$  which is **9mm**. For this study a distance of 1.67 $d$  which is **10mm** was selected. The use of studs decreases the bending stress at the flange root since the moment arm can be made very small.

## CONCLUSION

Today almost all major components that make up the internal combustion engines (ICEs) available in spare-parts markets are imported into the country. This is so because the technology and capacity involved in the design and manufacture of internal combustion engine parts are not readily available in Nigeria.

Over time, local researchers (Ebhojiaye and Ibadode, 2013; Abdul, Ebhojiaye and Ibadode, 2015; Oladeinde M.H., Ebhojiaye, R.S. and Ajagbofu, A.K., 2016; and Amalu and Ibadode, 2007) have however been made in reverse engineering of these ICE components rather than the design.

**Table 2:** Designed Values and Recommended Values.

Cylinder Block Parameter	Designed Value	Recommended Value
Swept Volume	80cc	-
Clearance Volume	8.89cc	-
Bore Diameter	45.10mm	50mm
Stroke Length	50.10mm	45 – 76.5mm
Length of Cylinder	57.63mm	49.5 – 69mm
Maximum Brake Torque	6.37Nm	-
Brake Power	2.0kW	-
Engine Power	2.67kW	1.5 – 5kW
Engine Mean Effective Pressure	3.33kN/m <sup>2</sup>	-
Cylinder Wall Thickness	3.05mm	> 3.01mm
Longitudinal Stress on Cylinder Wall	5.65N/mm <sup>2</sup>	Negligible effect
Circumferential Stress on Cylinder Wall	22.85N/mm <sup>2</sup>	≤ 35N/mm <sup>2</sup>
Cylinder Wall Expansion	0.057mm	Expansion is negligible
Cylinder Bolt Diameter	6mm	< 16mm
Flange Thickness	3.66 – 4.27mm	>3.36mm to 3.81mm
Flange Distance from Bolt Centre	10mm	9mm – 12mm

This study was conscientiously carried out to bridge this gap. An 80cc cylinder block which has wide application in lawn mowers engines, motorcycle engines, chain saws, snow-blower engines, etc. has been successfully designed. The designed parameter values were found to be within the recommended range of values for the engine capacity found in standard texts.

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