DESIGN OF MOTORCYCLE ENGINE COMPONENTS BASED ON FIXED SPECIFICATION

By

Mohd Shamani bin Samin (1526)

Dissertation submitted in partial fulfilment of

The requirements for the

Bachelor of Engineering (Hons)

(Mechanical Engineering)

JUNE 2004

Universiti Teknologi PETRONAS
Bandar Seri Iskandar
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Perak Darul Ridzuan

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FINAL PROJECT REPORT

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CERTIFICATION OF APPROVAL

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Approved:

MSPasricha
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Project Supervisor

UNIVERSITI TEKNOLOGI PETRONAS

TRONOH, PERAK

June 2004

CERTIFICATION OF ORIGINALITY

This is to certify that I am responsible for the work submitted in this project, that the original work is my own except as specified in the references and acknowledgements, and that the original work contained herein have not been undertaken or done by unspecified sources or persons.

Mohammad Shamani Bin Samin

ABSTRACT

This Final Year Project report covers the findings of the author in his final year project which is to design the engine components of a motorcycle based on a specification obtained from existing motorcycle available in the market. The main purpose of this project is to help the author inculcate the skills in applying the knowledge he has acquired in his science, engineering science, social studies and humanity courses. The object is not to teach new engineering courses but to guide the author on how to use purposefully and effectively what is already known to him.

The project concerns primarily the system by which energy is converted into useful mechanical forms, and of mechanisms required to convert the output of the design to the desired forms. In order to do this, a proper step in design is important to avoid any mistakes. This not only related to the design of an engine component, but other design project. The precision of the design is taken into account where all the small and tedious things became important. In doing the project and progressing to the next step, some problem occurs and slowing the process. This is where project planning is important.

The project begins by attaining the specification of the base model to be as a reference. Then the design stage commence whereby the dimensions of the components are calculated. Due to lack of information gathered, some of the parts to be design are revised and redesign according the specification needed. Finally, the drawings of the components were produced. This project is considered a success to the author as it meets the main objective of the project; to expose the author to the design procedure of engine components. It is recommended that for the future, the project be use to design a more powerful, fuel efficiency and cost saving engine.

ACKNOWLEDGEMENT

Thanks to Allah the almighty for giving the author the strength and will to carry out the project throughout the final year. This has been as interesting experience for the author as he discovers and learn new thing. The author has been touched by the help of individual involved during the completion of the project.

The author wishes to gives his heartfelt thanks his supervisor of the project, Dr M.S Pasricha for his guidance and advice in the course of this project. He would also like to thanks all the technician of mechanical department for their help and kindness. For all his family and friends, he would like to thank them for a very supportive encouragement and critics. Completion of this report is all because of them. Thank you very much.

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LIST OF SYMBOLS

Symbol	Quantity	Unit
D	Cylinder diameter	mm
l	Connecting rod length	mm
T	Torque	Nm
N	Engine speed	Min^{-1}
P_b	Brake power	W
P_i	Indicated power	W
P_{eff}	Net horsepower	W
P_{mep}	Mean effective pressure	Pa
b_{mep}	Brake mean effective pressure	Pa
I_{mep}	Indicated mean effective pressure	Pa
P_{max}	Maximum gas pressure	N/m^2
r	Crank radius	mm
L	Stroke	mm
S_k	Piston clearance from TDC	mm
V_c	Cylinder compression volume	dm^3
V_h	Displacement volume of a cylinder	dm^3
V_H	Displacement volume of the engine	dm^3
V_m	Mean piston velocity	m/s
V_{max}	Maximum piston velocity	m/s

z	Number of cylinders	-
ε	Compression ratio	-
θ	Crank angle	O (degree)
$W_{c,i}$	Indicated work per cycle	-
n_v	Number of crank revolutions per stroke	-
F	Maximum gas force	KN
F_i	Inertia force	KN

CHAPTER 1
INTRODUCTION

1.1 Background of Study

The design of a motorcycle engine is quite an interesting subject. It covers many principles learned in the mechanical engineering stream. They are many different designs to suit many different applications. Each of the design has their own merits and drawbacks. But it does not matter how the engine were design, since the basic for principles still apply and basically, all the engines have similar configuration.

In general, the research and study in this project required student to design a motorcycle engine with desired specification. Suitable design software's are used to enable the transfer of mathematical solution into a graphical design. Prior to designing the engine, student should have basic knowledge in material, heat transfer and dynamic analysis. This is important as in the design steps; the numerical steps involved the knowledge mention before.

1.2 Problem Statement

The question of how the shape of an engine component took place has triggered the author curiosity bud to study the design procedure of an engine. And based on the engine specification obtained from an existing motorcycle engine in the market, the design of a motorcycle engine will be carried out along with the generation of the drawings of the components.

1.3 Objective & Scope of Study

1.3.1 Objective of the project

The main point in this project is to study and learn about the design steps and procedure in designing an engine. However, several other objectives have also been identified. To produce a full functioning system of an engine might be a tedious job. Therefore an objective is set to clear the constraint and make it easy. The title of the project gives the hint of what it is all about. Since the project title is to general, another objective had to be set as guidance. In order to make it a success, several objective and task have to be fulfilled. That is:

1. To design the main components of the engine as stated.
2. To find and calculate the dimensions of the components of the engine.
3. To identify the material used to make the component of engine.
4. To expose the student to real engineering work that will be involved in the working environment.
5. To enable the student to analytically analyze data obtain from such projects.

1.3.2 Scope of Study

The project covers the designing of the engine components, mainly the cylinder liner, piston, connecting rod, crankshaft and the bearing used. The design is carried out based on the engine specifications obtained from an existing motorcycle sold in the market. With the results obtained from the analysis, the drawings of the component will be made.

CHAPTER 2
THEORY

An engine is a mechanism that receives energy and transforms it into work to drive other devices. An engine is basically consists of a piston in a cylinder that moves in a linear motion as the engine runs. The cylinder is where the combustion of the fuel takes place and the piston is the component that receives energy from the combustion and transfers that energy to supply power to drive the vehicle. An engine construction can contain as much cylinders but mainly motorcycle engine only uses one cylinder. The number of cylinder does affect the size of the cylinder. The more cylinder used in the engine, it size will be smaller in order for the engine to achieve better performance. [4]

The engine used in the motorcycle is the internal combustion engine. There are basically two principles types of engines; the 2 stroke and the 4 stroke engines. The main parts of an engine are the cylinder, the cylinder head, the piston, the connecting rod and the crankshaft. These are the components in a single cylinder.

2.1 General consideration

The size of the whole engine as a whole is determined on the basis of the mean cylinder pressure and the speed, but the design of the individual parts depends on the maximum cylinder pressure and the acceleration of the moving parts. The stresses in the various parts of the engine come from the following types of load;[1]

1. Due to gas pressure in the cylinder.
2. Due to inertia and centrifugal forces.
3. Due to frictional forces.
4. Due to torsional moment reaction and weight.
5. Due to vibrations, especially of the crankshaft.
6. Thermal loads.

The inertia load is proportional to the square of the speed. The inertia load sue to the reciprocating masses are in opposition to the gas load at Top dead center (TDC) and so it reduces the load on the connecting rod and the bearings. On TDC of the exhaust stroke in four stroke engines, the inertia load will induce a tensile load in the connecting rod. Hence there will be some reversal of stress in the connecting rod.

The size of the engine as a whole is determined on the basis of mean cylinder pressure and the speed, but the design of the individual parts depends on the maximum cylinder pressure and the acceleration of the moving parts. The stress in the variation parts of the engine come from the different loads existed in engine. The gas pressure should be regarded as the shock load while designing the running gear. The inertia load is proportional to the square of the speed.

The centrifugal forces due to the unbalanced rotating masses induce loads on the bearings when the crankshaft rotates. These are in the opposite direction to that of the gas load but due to their unbalanced nature, induce stressed in other parts of the engine.

There will be transverse bending of the connecting rod of the connecting rod due to the centrifugal forces set up by the rotating mass of the connecting rod. Fortunately this load is maximum when the gas load is very much reduced and vice versa, otherwise a complex type of compound fluctuating stresses would occur in the connecting rod. However, this is a reversible load on the connecting rod.

Centrifugal forces due to balanced mass will induced hoop stresses in the material of the parts and this should be kept within the limits. If friction forces are large, failure of some parts may occur due to overheating and seizure. When the crankshaft rotates, the loads due to the turning moment come on the crankshaft coupling and the foundation bolts. The bolts are usually made quite strong to resist the forces due to unbalanced couples however, turning moments should also not be neglected. In some cases, there might be end thrust on the crankshaft and the bearing should be designed accordingly.[1]

The stresses due to vibration are very important, especially, in the case of crankshaft. To reduce their effects, the points below should be observed carefully;[1]

1. The dimensions should be suitably selected to avoid resonance.
2. The parts should be made quite stiff to reduce the amplitude of vibrations.
3. The material and the shape of the parts should be such as to avoid failure by fatigue.
4. Damping in the result should be increased by the use of suitable materials.

Thermal loads on the engine parts exist since a portion of the unused heat leaves through the cooling fins. Temperature gradients on the metals will set a stresses thus lead to unequal expansion. The stresses due to thermal loads will be less if the section is thinner. While designing the various part of the engine, the following temperature effect should be considered:[1]

1. Expansion of metals.
2. Stresses induced in metals due to temperature differences.
3. Reduction in strength of metals at high temperatures.
4. Deterioration of the metal surfaces which are subjected to high and varying temperatures.
5. Growth of cast iron.

2.2 Design consideration of piston

In designing the piston *(refer figure 2.1)*, much consideration had to be taken into account. The profile of the piston head depend on the design of the combustion chamber. The thermal stresses will depend upon the rating and efficiency of the engine. The amount of side thrust on the cylinder wall is the function of the connecting rod. Factor such as weight balancing on the crank will influence the number and type of the piston rings. Engine designer and developer have a trend in developing an internal combustion engines of increased "power capacity". In order to satisfy this, design had been improved in increased compression ratio and engine speed. It is for this reason that piston, which is the fastest moving part of the engine has become vital in engine designing.

To design a piston of an internal combustion engine, the following objective must be achieved;[9]

1. The piston should have enormous strength and heat resistance properties.
2. Minimum weight to minimise the inertia force
3. Good and quick dissipation of heat from the crown to the ring and bearing area and then to cylinder walls
4. Form an effective gas and oil seal.
5. Sufficient rigid construction to withstand thermal and mechanical distortion.
6. Sufficient bearing area to prevent undue wear.
7. Symmetrical design for even expansion under thermal loads, as free as possible from discontinuities.
8. High speed reciprocating with minimal noise.
9. Minimum work of friction
10. Material of the piston must possess good wearing qualities, so that the piston is able to maintain the surface hardness up to operating temperature.
11. Little tendency towards corrosion.

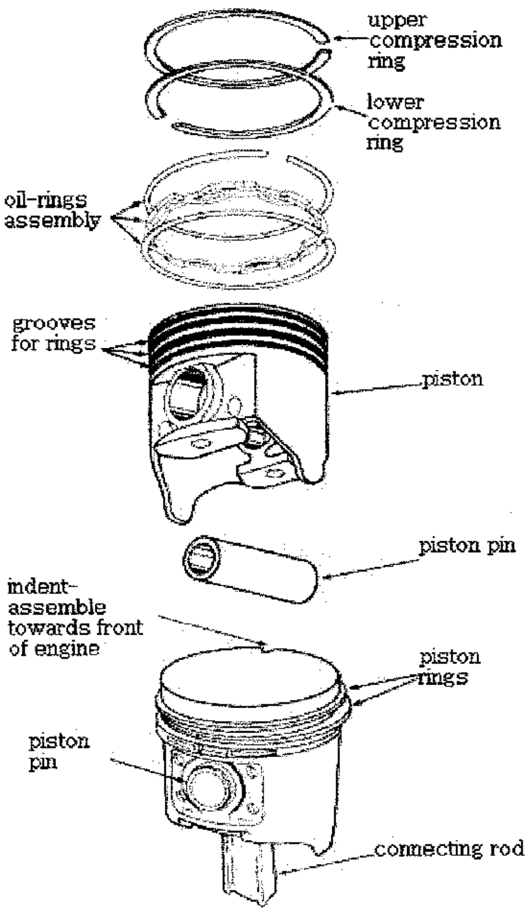


figure 2.1:piston and piston rings.

Tops, ring separated and above the piston. Bottom, piston ring installed in grooves in the piston. Piston attached to the connecting rod by the piston pin. Only upper part of connecting rod is shown.

With the piston-pin at the centre line of piston, the minor thrust face of race skirt remains in the contact with the cylinder wall until the end of the compression stroke. At the top dead centre a sudden shift of the side thrust on the piston takes place from minor to major thrust face. With appreciable clearance, the sudden reversal result in piston blow to the cylinder wall called the piston slap and produces a distinct noise at the top dead centre. Similarly as combustion pressure is applied to the piston head and the connecting rod angle changes from left to right, the side thrust on the piston will cause it to shift abruptly towards the major thrust face, *(figure 2.2)*. However, if the piston pin is offset *(figure 2.3)* the combustion pressure will cause the piston to tilt as the piston will reach near the top dead centre and lower end of the major thrust face of the piston will make contact with the cylinder wall. So when the piston will pass the TDC gradual contact of major thrust face will take place thus reducing the tendency of the piston slap.[3]

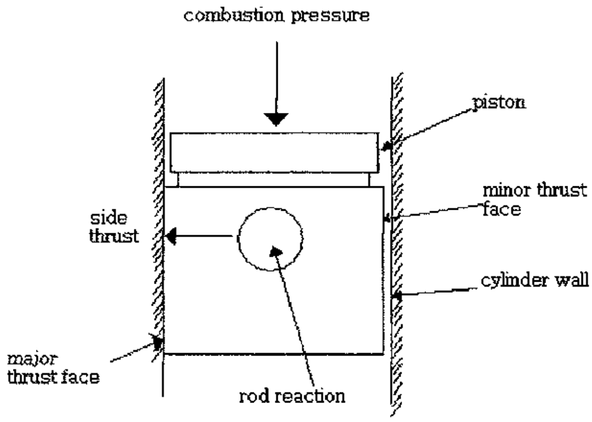


figure 2.2: side thrust cause piston to shift

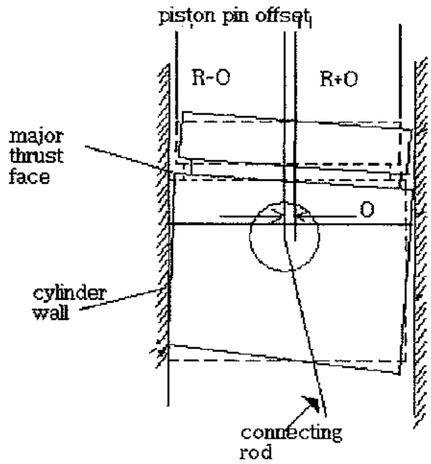


figure 2.3:piston is offset

2.3 Design consideration of connecting rod

The main function of the connecting rod is to transmit the push and pull from the piston to the crank pin. In many cases, its secondary function is to convey the lubrication oil from the bottom end to the top end.[3]

The connecting rods of internal combustion engine are mostly manufactured by drop forged. The connecting rod should have adequate strength and stiffness with minimum weight. The usual shapes of the connecting rod are: rectangular, circular, tubular, I section and H section *(refer figure 2.4)*. [1]

The stresses in the connecting rod are set up by a combination of forces. To design a connecting rod, several factors had to be taken into account. These are;[9]

1. The combined effect of gas pressure on the piston and the inertia of the reciprocating parts.
2. Friction of the piston rings and of the piston.

3. Inertia of the connecting rod.
4. The friction of the two end bearings of the piston pin and bearing and the crank pin bearing.

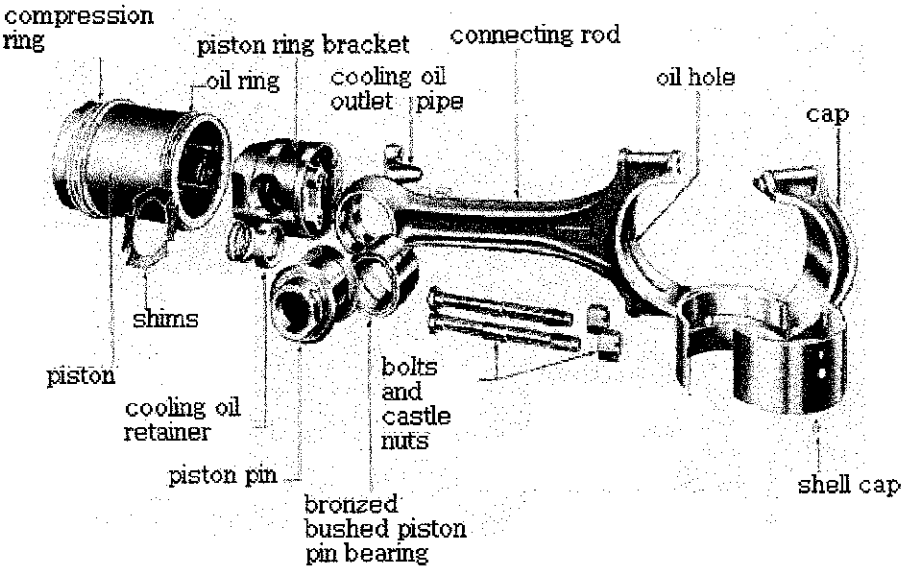


figure 2.4: connecting rod assembly

The load due to the piston inertia [1]

= weight of the reciprocating masses * acceleration

$$\therefore F_i = \frac{F}{g}\omega^2 r\left(\cos\theta + \frac{r\cos 2\theta}{l}\right)$$

$$\begin{aligned}
&where\ F = weight\ of\ reciprocating\ masses, N \\
&\quad = weight\ of\ piston\ including\ that\ of\ rings + weight\ of\ piston\ pin \\
&\quad + weight\ of\ one\ third\ portion\ of\ connecting\ rod(small\ end\ portion). \\
&\omega = angular\ velocity\ of\ crank, rad/s \\
&\theta = crank\ angle\ from\ TDC. \\
&r = crank\ radius, m \\
&l = road\ length, m
\end{aligned}$$

Due to maximum gas load, the rod will be subjected to alternating direct compression and tensile stresses. Since the direct compression stress corresponds to compression and explosion stroke, it will be numerically much larger than the tensile strength stress. Therefore, the connecting rod is designed as a strut [5]. Due to the comparatively large diameter or radius of gyration in the case of I section, with respect to the rod length. Rankine Gordon formula is used;

$$Buckling\ Load = \frac{f_{cU} \times A}{1 + a\left(\frac{l}{k}\right)^2}, N$$

where f_{cU} = ultimate crushing stress, N/m^2

A = section area, m^2

l = equivalent length, m

k = radius of gyration about the axis of buckling, m

a = constant

For the I section of the rod *(refer figure 2.5)*, the ends of the rod are direction free and freely hinged at the piston pin and the crank pin in the plane of motion. Hence for buckling about the neutral axis *xx*, the strut is freely hinged. In the plane, the perpendicular to the plane of motion, for buckling about the axis *yy*, the strut is fixed ended due to the constraining effect of the bearings at piston and crankpins. Therefore, for buckling about axis *yy*, the rod is four times as strong as for buckling about axis *xx*. But the rod should be equally strong in both the planes [1]. That is,

4Iyy = Ixx

$$\therefore K^2yy = \tfrac{1}{4}K^2xx$$

$$A = 11t^2$$

$$Ixx = \tfrac{1}{12}\left(BH^3 - bh^3\right) = \tfrac{1}{12}\left[\left(4t * 5t^3\right) - \left(3t * 3t^3\right)\right] = \frac{419}{12}t^4$$

$$Iyy = \tfrac{1}{12}\left(2tB^3 + ht^3\right) = \tfrac{1}{12}\left[\left(2t * 4t^3\right) - \left(3t * t^3\right)\right] = \frac{131}{12}t^4$$

$$\therefore K^2xx = \frac{I_{xx}}{A} = \frac{419t^4}{12 * 11t^2} = 3.18t^2$$

$$K^2yy = \frac{Iyy}{A} = \frac{131t^4}{12 * 11t^2} = 0.955t^2$$

$$\therefore \frac{K^2yy}{K^2xx} = \frac{1}{3.2}$$

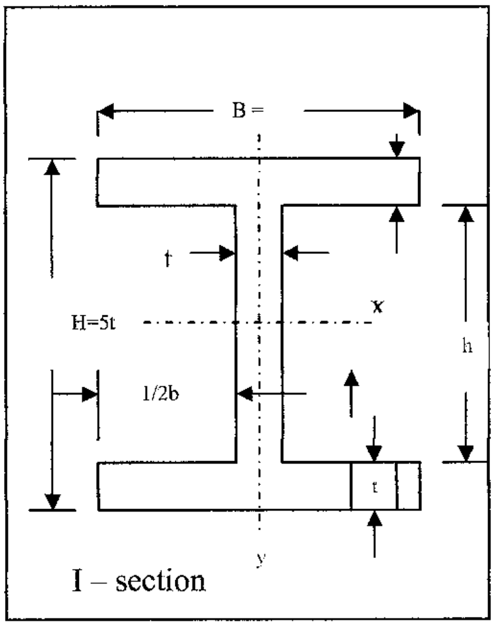


figure 2.5 : I-section

2.4 Design consideration of crankshaft

The function of crankshaft is to transform a reciprocating motion into a rotary or vice-versa [2]. Crankshaft consists of the shaft parts which revolve in the main bearings, the crankpins to which the big ends of the connection rod are connected, the crank webs or cheeks which connect the crank pins and the shaft parts *(refer figure 2.6)*.

To ensure the proper functionality of a crankshaft, the material used to produce a crankshaft should fulfill the following condition [1];

1. Enough to withstand the forces to which it is subjected (the bending moment).
2. Enough rigidity to keep the distortion a minimum.
3. Stiffness to minimise, strength to resist the stress due to torsional vibrations of the shaft.
4. Sufficient mass properly distributed to reduce the effect of vibration.
5. Sufficient projected areas of crankpins and journals to keep the bearing pressure to a value dependent on the lubrication available.
6. Minimum weight.

In manufacturing crankshaft, it is important to consider the process with great care. Usually small crankshafts are drop forged. Larger size are forged and machined to shape [9].

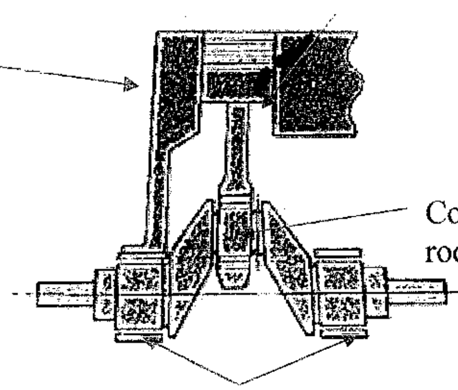


figure 2.6: main or crankshaft bearings fit between main journals and block. Rod bearings fit between rod journals on crank and block

Bearing pressures are very important in the design of crankshafts. The allowable bearing pressure depends upon the journal velocity, change of direction of the bearing pressure, amount and method of lubrication and the maximum gas pressure and space limitation.

The stresses induced in the crankshaft are bending and also shear stresses due to torsional moment on the shaft. Most crankshafts fail due to progressive fracture due to repeated bending or reserved torsional stresses. Thus the type of loading in the crankshaft is fatigue loading, therefore, the design should be based on the endurance limit.

To avoid stress concentration and fatigue, abrupt changes in the section of crankshaft should be avoided. Two different cross sections must be blended with a large fillet '*r*'; if possible, r should not be less than 0.2 *d.* if there is no space for fillet, the crank web should be undercut to obtain the fillet. This will make the web weak and to compensate for it, the width is increased.

Since failure of the crankshaft is serious for the engine, and also because of the inaccuracy in determining all the forces and stresses, a high factor of safety should be used. To be on the safe side, the endurance limits for complete reversal of bending an torsional stresses are taken.

For chrome nickel and other alloy steels, the endurance limit is about 525.0 MPa in bending and about 290.0 MPa in shear. For carbon steel and cast steel, the endurance limit is about 225 MPa in bending and about 124 MPa in shear. For alloy cast iron, the endurance limit is about 140 MPa in bending and in shear.

Therefore, the allowable stress is [1] :

For carbon steel:

Bending = 56.0 to 75.0 MPa

Shear = 31.0 to 42.0 MPa

Combined stress = ½ x elastic limit to pure tension.

For alloy cast iron:

Bending = shear = 35.0 to 47.0 MPa

Combined stress = elastic limit in pure tension.

For chrome nickel and other alloy steels:

Bending = 130.0 to 175.0 MPa

Shear = 72.5 to 97.0 MPa

Combined stress = ½ x elastic limit in pure tension.

There are few guidelines that should be followed when designing a crankshaft. They are[1]:

1. Determine the magnitudes of the various loads acting on the crankshaft.
2. Determine the distances between the supports. The distances will depend upon the lengths of the bearing. The lengths and diameters of the bearing are determined on the basis of maximum permissible bearing pressure, l/d ratios and the acting loads.
3. For the sake of simplicity and safety, the shaft is considered to be supported at the centers of the bearings.
4. The thickness of the crank webs is assumed, about 0.5d to 0.6d, where d is the shaft diameter, or from 0.22D to 0.32D, where D is the cylinder bore.

5. Assume allowable bending and shearing stresses.

It should be noted that all forces and reactions are assumed to be acting at the centers of the bearings.

2.5 Design consideration of cylinder

The cylinder of an internal combustion engine performs a number of duties. Its primary is to contain the working fluid and the secondary function is to guide the trunk piston.

In very small engines, the cylinder, cooling fins and the frame may be built in one piece, while for bigger engine and high speed engine; these component might be built separately. The liners usually manufactured separately as it is more economical and can be replaced after wear and tear. The type of liner used in motorcycle engine basically of dry liner type. Its advantages compared to wet liner are;

1. It is simpler to replace.
2. No danger of leakage into combustion chamber.
3. Due to absence of a heavy flange at the top of the liner, cylinder centre can be reduced
4. Better cooling of the upper part of the liner.

A good cylinder liner should possess the following qualities[1];

1. Strength to resist the gas pressure.
2. Sufficiently hard to resist wear.
3. Strength to resist the thermal stresses due to the heat flow through the liner wall.
4. Corrosion resistant.
5. Capable of taking good bearing surface.
6. It should by symmetrical in shape to avoid unequal deflection due to gas load and unequal expansion due to thermal load.
7. No distortion of the inner surface due to restraining fixings.

Liner should be strong, hard and corrosion resistive and produce a good bearing surface. There are several material recommended for liner[1];

1. A good grade grey cast iron with homogenous and closed grained structure.
2. Nickel cast iron and nickel chromium cast iron
3. Nickel-chromium cast steel with molybdenum.

The importance of lubrication in the cylinder ensures that minimal power loss due to thermal effect and friction load between the liner and the piston rings. Just sufficient lubrication of the liner is needed as excess of oil will do more harm. Liner is well lubricated when it's running, but trouble always happen before starting. The cylinder is oil starved, and possibility of high wear of liner might occur.

CHAPTER 3

METHODOLOGY/PROJECT WORK

In order to proceed with the project, the author has decided to first search and inquire from experience person regarding the matter. Because the basic building of an engine is similar, related books about the topics is important.

With the help from nearby workshop, the author had an opportunity to take a look at the inside view of a motorcycle engine. Author managed to identify the parts and component and its function. With the information obtained, the author then identified the specifications he has chosen based on the project. And the calculations of the importance parameters are done.

3.1 Basic engine parameters

Some of the important basic parameter is used to determine the dimensions of the various engine components.

Swept volume (displacement)

The displacement of a cylinder is given by:

$$V_h = (\pi . D^2 . L)/4$$

the swept volume of the engine;

$$V_H = V_h . z$$

Compression ratio

The compression ratio of the engine:

$$\varepsilon = (V_h + V_c)/V_c$$

Piston movement

Piston clearance from top dead center

$$S_K = r\left[1 + (l/r) - \cos\theta - \sqrt{(l/r)^2 - \sin^2\theta}\right]$$

Mean piston velocity

$$V_m = 2.N.L$$

Engine power

Brake power

$$P_b = 2\pi TN$$

Indicated power:

$$P_i = (z.W_{c.i}.N)/n_r$$

where n_r is the number of crank revolution for each power stroke per cylinder.

N_r=1 for 2 stroke engine

N_r=2 for 4 stroke engine

Net horsepower;

$$P_{eff} = V_H \cdot p_{mep}.N / K$$
$$K = 1\ for\ 2\ stroke\ engine; 2\ for\ 4\ stroke\ engine$$

engine torque

$$T = (V_H \cdot p_{mep})/4\pi$$

mean effective pressure

$$p_{mep} = (n_r.P)/(z.V_h.N) = (2.\pi.n_r.T)/(z.V_h)$$

where p_{mep}, p, and T can be brake or indicated values.

From the basic parameters, the dimension of the engine components may be calculated.

3.2 Design procedure of cylinder

The procedure for the design of the cylinder of the engine is outlined below.

1. The thickness of the cylinder wall is calculated using the formula for a thin cylinder.
2. The length of the cylinder is calculated by incorporating clearance on both ends of the cylinder.

Assumptions are made before calculating the component, which are;

1. steady state condition
2. one dimensional radial construction with fins
3. constant properties
4. negligible radiation exchange with surrounding
5. Uniform convection coefficient over outer surface.

The wall thickness is usually calculated by applying the formula for a thin cylinder.

$$t = \frac{p_{\max} \times D}{2 f_c} + k$$

where f_c = maximum hoop stress and is equal to 35.0 to 105.0 N/mm^2 depending on the size and material, larger values are used for smaller bores.

k = reboring factor. *(table 1)*

P_{mas} = maximum gas pressure, N/mm^2.

Referring to the boring factor table from Sharma and Aggarwal(1997)[1];

Cylinder bore in mm	75	100	150	200	250	300	350	400	450	500
k, in mm	1.5	2.3	4.0	6.0	7.5	9.5	10.5	12.5	12.5	12.5

Table 3.1: boring factor

Length of cylinder = stroke + clearance ob both sides

Cylinder head thickness = $t = D\sqrt{\frac{CP_{\max}}{f_r}}$

Where C = constant, in this case equal to 0.1

F_t = allowable stress, taken to be 35 to 56 N/m

3.3 Design procedure of piston

The procedure for the design of the piston and its components are provided below[9].

1. The mean piston speed is calculated to determine the material for the piston.
2. The thickness for the piston head is calculated.
3. The dimensions of the piston rings are calculated.
4. The dimensions of the piston barrel are calculated.
5. The dimensions of the piston skirt are calculated from the side thrust of the liner.
6. The dimensions of the piston pin are calculated using the maximum gas load.

The formulas below are used to calculate the dimension of the piston.

Piston head thickness

$$t_h = \sqrt{\left[\left(3D^2 p_{max}\right)/16f\right]}$$

Where f_t = 39MPa for closed grained cast iron

= 56.4MPa for semisteel or aluminium alloy

= 83.4MPa for forged steel

Dimensions of piston rings

The radial width or thickness:

$$b_r = D\sqrt{\left(3p_w / f_t\right)}$$

Where f_t = radial width or thickness, mm

p_w = desirable wall pressure, may be taken from 0.0245 to 0.042 N/mm^2

f_t = allowable stress in bending, N/mm^2

the axial width or thickness may be taken as 0.7 b_r to b_r, therefore,

$h = 0.7\ b_r$ to b_r

Where h = axial width or thickness, mm

The thickness between the ring grooves, the land may be taken as equal to or slightly less than the axial thickness of the ring h. Therefore,

Width of top ring land = $0.75h$ to h

The width of the top of the land is made larger than the other ring lands to protect the top ring from the high temperature conditions existing at the top of the piston. Therefore,

Width of top ring land = t_h to $1.2\ t_h$ or 0.2 to 0.3D

The depth of the ring grooves should be more than the ring depth so that the ring does not have to take any piston side thrust. The gap between the free ring ends is given as

$G = 3.5b_r$ to $4b_r$

The gap when the ring is in the cylinder should be 0.002D to 0.004D.

Dimension of the piston barrel

Allowing for the ring clearance,

$D_1 = D - (2b_r + 0.006D + 0.5mm)$ at compression rings

And,

$D_1 = D - (2b_r + 0.006D + 1.50mm)$ at oil grooves.

Where D_1 = diameter of the bottom of the ring grooves.

Empirically, the maximum thickness of the barrel is given as:

$t_1 = b + 0.03D + 4.5mm,$

Where b = depth of ring grooves, about 0.4mm larger than b_r

At the open end, the wall thickness is;

$t_2 = 0.25t_1$ to $0.35t_1$

Piston skirt and piston length

The portion of the barrel below the ring section up to the open end is known as the "skirt". It takes the side thrust of the connecting ring. Its length should be such that the side thrust pressure should not exceed 0.21 to 0.28 N/mm^2 for low speed engines and this may go up to $0.49 N/mm^2$ for pistons of high speed engines[1].

The normal side thrust is calculated as:

Normal side thrust, R = (0.03 to 0.10) * F

Where F = maximum gas load

$$= (\pi/4)D^2 \times p_{max} \; or \; R = l \times D \times p_b$$

where l = skirt length,mm

p_b = side pressure, N/mm^2.

The length of piston, L = l + length of ring section + top land. Empirically,

L = D to 1.5D

Piston pin

The diameter of the pin is determined by equating the gas load and the bearing load.

$$l_p \times d_p \times p_b = (\pi/4)D^2 \times p_{max} = F$$

Where l_p = *length of piston pin in the connecting rod bearing*

d_p = *diameter of the pin*

p_b = *bearing pressure*

Due to the limitation of space, a high value of bearing pressure is taken, usually from about 15.0 N/mm^2 for large piston to about 31.5 N/mm^2 for smaller piston.

The ratio of the piston pin length to the piston pin diameter is given as

$$l_p / l_p = 1.5 \; to \; 2$$

$$M_b = (F \times D)/8 = (\pi/32)d_p^3 \times f_b$$

where f_b = allowable stress in bending

The maximum bending moment will be at the center of the pin, or to reduce the weight of the pin, it is usually made hollow, with outside diameter d_o and inside diameter d_i.

$$\therefore M_b = \pi/32 \times (d_o^4 - d_i^4)/d_o \times f_b$$

where f_b = *allowable stress in bending*

$= 84.0\,N/mm^2$ *for a case hardened carbon steel and* $140.0\,N/mm^2$ *for heat treated allo*

Piston clearance

The recommended piston clearance for aluminium alloy piston is 0.0375 to 0.075 mm for motorcycles

3.4 Design procedure of connecting rod

The procedure for the design of the connecting rod is as below[9];

1. The dimensions for the section of the connecting rod are calculated using the buckling load parameter.
2. The dimension of the parameter of the big end of the rod is calculated using the maximum gas load.
3. The bolt size for the connecting rod cap is calculated using the inertia force.
4. The dimension of the connecting rod is calculated using the bending moment.

The necessary formulas for the calculation of the connecting rod dimension are as below.

Section of rod

The section of the connecting rod can be determined by considering the buckling load about the xx axis for an I section connecting rod.

$$Buckling\ load = f_{cu} \times A / \left[1 + a\left(\frac{l}{k}\right)^2\right], N$$

where f_{cu} = ultimate crushing stress, N/m^2

A = section area, m^2

l = equivalent length, m

k = radius of gyration about the axis of buckling, m

a = constant, $a = 1/7500$ for medium steel

$= 1/9000$ for wrought iron

$= 1/1600$ for cast iron

or

Buckling load = p_{max} x factor of safety

The factor of safety of columns [1]:

Steady load	= 3.375 to 5
Light shock	= 4.2 to 7.5
Heavy shock	= 6.75 to 15

Dimensions of big end

The crank pin dimensions are calculated using the maximum gas load.

$$l_c \times d_c \times p_b = F$$

Where l_c = length of crank pin, mm

d_c = diameter of crank pin,mm

p_b = *allowable bearing pressure,N/mm²*

Dimensions of connecting rod cap

Maximum bending moment,$M_b = (F_i \times S)/6$

Where S = distance between bolts centers

= diameter of bearing + 2x thickness of bearing liner + diameter of bolt + clearance

Thickness of bearing liner = thickness of bearing shell + thickness of bearing metal

Empirically formula for thickness of bearing shell = 0.05D

Also, $M_b = f_b \times Z$

Where $Z = bc^2/6$ where b = width of cap = length of bearing

= thickness of cap,mm

3.5 Design procedure of crankshaft

The procedure to design the crankshaft is provided below [1].

1. First, consider the shaft at the position of top center.
2. The center distance between the bearings is calculated.
3. The thickness and width of the web are calculated.
4. The diameter of the shaft under is calculated.
5. The analysis is considered for the crank at the angle of maximum torsional moment.
6. The forces at the bearings are calculated.
7. The diameter of the crankpin is calculated.
8. The diameter of the right hand bearing is calculated.

9. The stresses are checked to make sure they are within the safe limit.

Formulas below are used to calculate the crankshaft dimensions [1];

Crank at dead center

Bending moment at the center of crankpin;

$M_b = H_1 \times (f/2)$

Thickness of web, $t = 0.65d + 6.35\ mm$

Width of web, $w = (9/8)d + 12.7\ mm$

Bending moment at the left hand crank;

$M_b = H_1 \times [(f/2) - (l_c/2) - (t/2)]$

Bending stress, $f_b = M_b/Z$

Where Z = section modulus = $1/6 \times wt^2$

Lengths of the bearings;

$l_1 = l_2 = l_3 = 2[(f/2) - (l_c/2) - t]$

Where l_c = length of the crankpin

Bending moment due to the belt pull,

$M_b = (F1 + F2) \times g/4$

Total bending moment;

$$M_{to} = \sqrt{M_f^2 + M_b^2} = (\pi/32) \times d_w^3 \times f_b$$

Crank angle at maximum torsional moment

The reaction forces;

$H_{t1} = H_{t2} = Pt/2$

And $H_{r1} = H_{r2} = Pr/2$

Twisting moment at shaft under the flywheel;

$M_t = Pt \times r$

Total twisting moment under the flywheel;

$$M_{te} = \sqrt{M_t^2 + M_{to}^2}$$
$$= (\pi/32) \times d_w^3 \times f_s$$

Bending moment at juncture of right hand crank arm;

$$M_b = H_1 \times [(f/2) + (l_c/2) + (t/2)] - Q(l_c/2) + (t/2)$$

Equivalent twisting moment at juncture of right hand crank arm;

$$M_{te} = \sqrt{M_t^2 + M_b^2} = (\pi/16) \times d_2^3 \times f_s$$

Bending moment due to the radial component at right hand crank web;

$$M_{br} = H_{t2} \times [(f/2) - (l_c/2) - (t/2)]$$
$$= 1/6\, wt^2 f_{br}$$

Bending moment due to tangential component at right hand right crank web;

$$M_{bt} = Pt(r - d_2/2)$$
$$= 1/6\, tw^2 f_{bt}$$

Direct compressive stress at right hand crank web;

$$f_d = \Pr/(2 \times wt)$$

Total compressive stress;

$$f_c = f_{br} + f_{bt} + f_d$$

Twisting moment on arm of right hand crank web;

$$M_t = H_{r2} \times [(f/2) - (l_c/2)]$$
$$= 1/4.5\, wt^2 f_s$$

Combined stress;

$$f_{c\max} = f_c/2 + \sqrt{f_s^2 + (f_c/2)^2}$$

CHAPTER 4
RESULTS AND DISCUSSION

4.1 Result

The project is base on a working model that is available in the market. The specification that the author chooses to base the project is available in the appendix. Based on the specification given, the dimensions of the engine components were calculated.

4.2 Basic parameter

During the power stroke, when the piston is moving towards the bottom dead center, torque is applied by the piston to the crankshaft through the connecting rod. The torque applied produces a turning effect, resulting in the rotary motion of the crankshaft.

Brake power : $P_b = 2\pi .T.N$

Therefore, torque: $T = P_b / (2\pi .N)$

The torque calculated is not the same torque as the maximum torque given in the specification. It is the torque at which the brake power is obtained. The maximum torque is obtained at much lower speed compared to the torque of the power output.

Mean effective pressure

The mean effective pressure, P_{mep} is calculated using the brake power. Therefore, the P_{mep} obtained is the brake mean effective pressure, b_{mep}. The brake mean effective pressure is a measure of specific engine torque, and its maximum value is obtained when the engine torque reaches a maximum.

Mean piston velocity

The mean piston velocity is calculated from the equation:

$v_m = 2.N.L$

The material for the piston is chosen based on the mean piston velocity.

Clearance volume

The clearance volume, V_c is the cylinder volume at top dead center, that is, the volume enclosed by the cylinder head, the cylinder wall, and the piston head surface at top dead center. It is calculated by manipulating the equation for the compression ratio.

Compression ratio:

$$\varepsilon = (V_h + V_c)/V_c$$

4.3 Findings

4.3.1 Dimension of the cylinder

The cylinder is made from aluminum alloy that can prevent the wearing of the cylinder due to the movement of the piston against its wall. The design is calculated as follow;

Cylinder wall thickness

The maximum gas pressure, $P_{max} = 2\text{N/mm}^2$

Reboring factor is taken to be 1.5

$$t = \frac{p_{\max} \times D}{2f_c} + k$$

$$t = \frac{2 \times 50}{2(50)} + 1.5 = 2.5mm$$

The cylinder flange is made thicker than the wall of the cylinder. The common value for flange thickness is 1.2 to 1.4*t*.

Hence, flange thickness = 1.2 (2.5) to 1.4 (2.5) mm

= 3.0 mm to 3.5 mm

Taken to be 3.5 mm.

Length of the cylinder.

The length of the cylinder is taken as the stroke and the clearance on both sides. Th clearance on both sides is taken as 20% of the stroke.

Length = L +0.2L

= 55.5 + (0.2 * 55.5)

= 66.6 mm

Taken to be 70 mm.

Cylinder head thickness

It is assumed to be a flat circular plate. Therefore, the thickness is;

$t = D\sqrt{\frac{CP_{max}}{f_r}}$, where C is 0.1; f_t = 42N/mm^2

Therefore $t = 50\sqrt{\frac{0.1*2}{42}} = 3.45mm$

Summary dimension of cylinder.

Items	Value
Cylinder wall thickness	2.5 mm
Cylinder flange thickness	3.5 mm
Cylinder length	70 mm

Table 4.1: dimension of cylinder head

4.3.2 Dimensions of the Piston

Aluminium alloy will be used for the piston. This is because at such high speed, heavy reciprocating pistons will develop high inertia forces, which are undesirable. Therefore, for high speed engines, a light weight piston is always a choice. Usage of aluminium gives light piston and better heat dissipation due to high thermal conductivity.

Cylinder bore: 50mm = 50 * 10^{-3}m

Stroke: 55.5mm = 55.5 * 10^{-3}m

Maximum gas pressure = 2N/mm^2

Brake mean effective pressure = 0.60 N/mm^2

Fuel consumption = 0.0555 kg/kW/hr

Allowable stress in bending; f_t = 60N/mm^2 (normalized aluminium alloy)

Speed = 3000 rev/min

Piston Head

The thickness of head can be found by either on the basis of strength or the basis of heat dissipation through the piston head.

Thickness by strength;

th = $$0.433D\sqrt{\frac{P_{\max}}{f_t}} = 0.433(50)\sqrt{\frac{2.0}{60}} = 3.95mm \approx 4.0mm$$

Thickness by heat flow;

$$T_h = \frac{H}{12.56k(T_C - T_e)}$$

$$\therefore H = C_W * HCV * BP(heat\ flow\ through\ the\ head, watts)$$

$$C = 0.05$$

$$W = 0.015kg/Kw/hr = 4.17*10^{-6}kg/Kw/s$$

$$BP = \frac{P_m * L * A * n}{60} = \frac{0.60*0.0555*0.785*(50)^2*1500}{60*1000} = 1.63Kw$$

$$HCV = 41870Kj/Kg\ (higher\ heating\ value))$$

$$\therefore T_h = \frac{0.00262}{12.56*174.75*75} = 15.9*10^{-9}mm$$

Since the thickness by heat dissipation is too small,

Therefore the selected head thickness is 4.0 mm

The radial ribs

Thickness of ribs = $\frac{1}{3}T_h \rightarrow \frac{1}{2}T_h = 1.33mm \rightarrow 2.0mm$

Therefore; thickness of ribs = 2.0 mm

Piston ring

The author has decided to use two compression ring and one oil ring. The rings are made of fine grained alloy cast iron containing silicon and manganese. This material resists heat and wear and is elastic enough to allow radial expansion and compression.

$$\textit{The radial thickness of the rings; } b_r = D\sqrt{\frac{3pw}{f_t}}$$

$$\textit{wall pressure; } pw = 0.0245N/mm^2 \rightarrow 0.042N/mm^2$$

$$\therefore pw = 0.040N/mm^2$$

$$\textit{allowable stress in bending; } f_t = 60N/mm^2$$

$$b_r = 50\sqrt{\frac{3*0.040}{60}} = 2.24mm \approx 2.3mm$$

Axial width $= (0.7 \text{ to } 1.0)\ b_r$

$= 1.61$ mm to 2.3 mm

$\approx 2.0mm$

Ring section = 2.3mm*2.0mm

Minimum axial width; h = D/10Z = 50/10(3) = 1.67 mm $\approx 1.7mm$

Width of rind lands = 0.75h ~ h

= 1.2525 mm ~ 1.67 mm

= 1.67 mm $\approx 1.7mm$

Width of top land; t_h to $1.2t_h$ = 4.00 mm ~ 4.8 mm

= 4.8 mm

Gap between free ring = $3.5\,b_r \sim 4.0\,b_r$ = 8.05mm ~ 9.2mm = 8.05 mm

Gap ring in cylinder = 0.002D ~ 0.004D = 0.1 mm ~ 0.2mm = 0.1 mm

Piston barrel

For ideal flow of heat, the thickness of the barrel should be equal to the head thickness at the top, tapering down to zero at the bottom end. Wall strength is never being zero for the reason of strength. The thickness at the ring section must be modified to have the equivalent areas across the ring grooves.

Thickness of barrel, t_1 $= b + 0.03D + 4.5$

$= (0.4+2.3)+0.03(50)+4.5$

$= 8.7$ mm

At open end, wall thickness is;

$t_2 = 0.25t_1$ to $0.35t_1$

$= 2.175$ mm ~ 3.045 mm, say 3.0 mm

Skirt length;

Side thrust pressure = 0.21 N/mm^2 ~ 0.28 N/mm^2 (low speed engine)

= 0.49 N/mm^2 (high speed engine)

Normal side thrust, R = 0.03 F ~ 0.10F

Where F = max gas load

Let R = 0.1F

Normal R = 0.1(2.356) = 0.2356 KN

$$\max imum\ gas\ load;\ F = \frac{\pi}{4} D^2 * P_{\max} = \frac{\pi}{4}(50)^2 * 1.2 = 2.356 KN$$

$$Side\ pressure;\ P_b = 0.45\, N/mm^2$$

$$R = l * D * P_b$$

$$0.2356 * 1000 = l * 50 * 0.21$$

$$\therefore l = \frac{0.2356 * 1000}{50 * 0.21} = 22.44mm \approx 23mm$$

Piston length = skirt length + ring section + top land

$$= 23mm + [(3*2) + (2*1.7)] + 4.8$$

$$= 37.2mm$$

Desirable length of piston; D to 1.5D

= 50mm ~ 75mm, taking 50 mm as length of piston.

Piston pin

Design for maximum gas load;

$$bearing\ pressure;\ P_b \approx 20\,N/mm^2\ (for\,small\,piston)$$

$$F = l * d * P_b$$

$$l = 0.45D = 0.45(50) = 22.5mm \approx 23mm$$

$$2.356*1000 = 23*d*20$$

$$d = \frac{2.356*1000}{23*20} = 5.12mm \approx 6.0mm$$

$$desired\,diameter = 12mm.$$

Material of pin; heat treated alloy steel

Maximum bending moment on the piston pin

$$m_b = \frac{F*D}{8} = \frac{2.356*1000*50}{8*1000} = 14.725Nm = 14.725KNmm$$

$$m_b = \frac{\pi}{32} * d^3 * f_b$$

$$f_b = \frac{m_b * 32}{\pi * d^3} = \frac{14.725*1000*32}{\pi * 10^3} = 149.98\,N/mm^2 \geq 140.0\,N/mm^2$$

To make it lighter; the pin is made hollow.

$$d_o = 10mm$$

$$d_i = 8mm$$

$$m_b = \frac{\pi}{32} * \frac{d_o^4 - d_i^4}{d_o} * f_b$$

$$14.725*1000 = \frac{\pi}{32} * \frac{12^4 - 8^4}{10} * f_b$$

$$\therefore f_b = 108.164\,N/mm^2 \leq 140\,N/mm^2$$

This is safe.

The summary of piston dimensions

Items	Dimension (mm)
Radial thickness	2.0
Axial width	2.0
Gap between free ring ends	8.05
Gap between ring in compression	0.1
Piston head thickness	4.0
Width of ring load	1.7
Width of top ring land	4.8
Thickness of piston barrel	8.7
Piston wall thickness at open end	3.0
Piston skirt length	23.0
Length of piston	37.2 *(desirable 50.0)*
Length of piston pin	6.0 *(desirable 12)*
Outside diameter of piston pin	10.0
Inside diameter of piston pin	8.0

Table 4.2 : dimensions of piston

4.3.3 Dimensions of Connecting rod

Piston diameter = 50mm = $50*10^{-3}$m

Stroke = 55.5mm = $55.5*10^{-3}$m

Length of connecting rod, center to center = 100mm = 0.10m

Compression ratio = 9:1

Weight of reciprocating parts = 120g = 0.12Kg = 1.1772N

Speed = 2000 rev/min with possible over speed at 6000rev/min

Max explosion pressure = 1.45MPa

Material = low carbon steel

The section chosen for the connecting rod is the I-section. This is because it is light; therefore, it will keep the inertia forces small, and can withstand the high gas pressure. Low carbon steel is chosen as the material for the connecting rod.

Buckling load = $\dfrac{f_{CU} * A}{1+a(l/K)^2}$; rankine formula

Safety factor at 5

f_{cu} for low carbon steel = 330.0MPa

Buckling load = max explosion load * safety factor

$$= \frac{\pi}{4} * \left(50*10^{-3}\right)^2 * 1.45*10^6 * 4$$
$$= 11388.3N \approx 11.3883KN$$

Buckling load

$$\frac{f_{CU} * A}{1+a(l/K)^2} = \text{max explosion load *safety factor}$$

$$\frac{330*10^6 * 11t^2}{1+(1/1600)\left(0.1^2/3.18t^2\right)} = \frac{\pi}{4} * \left(50*10^{-3}\right)^2 * 1.45*10^6 * 4$$

Solving for t = 2.12 mm

Depth; 5(t) = 5*2.12 = 10.6 mm

Width; 4(t) = 4*2.12 = 8.48 mm

Depth at crank end = 1.1*10.6 = 11.66mm; take 12 mm

Depth at piston end = 0.75*10.6 = 7.95mm; take 8 mm

Dimension of big end

$$P = l_C * d_C * p_b$$

$$l_c = \textit{length of crank pin, mm}$$

$$d_c = \textit{diameter of crank pin, mm}$$

$$p_b = \textit{allowable bearing pressure}, N/mm^2$$

p = max gas load

$$N = \pi/4 * \left(50*10^{-3}\right)^2 * 1.45*10^6$$
$$= 2.847 KN$$

$$\frac{l_C}{d_C} = 1.4$$

$$p_b = 9 N/mm^2$$

$$1.4 d_C * d_C * 9 = 2.847*1000$$

$$d_C^2 = \frac{2.847*1000}{9*1.4} = 225.952$$

$$\therefore d_C = 15.03 mm$$

$$l_C = 1.4*15.03 = 21.04 mm$$

Dimension of a small end

Bearing pressure range; 15N/mm^2 ~ 31.5N/mm^2

$$l_p * d_p * P_b = P$$

$$\frac{l_p}{d_p} = 1.5 \sim 2 \ \textit{ taking } 1.5$$

$$1.5 d_p * d_p * 15 = 2.847*1000$$

$$d_p^{\ 2} = \frac{2.847*1000}{1.5*15} = 126.53$$

$$d_p = 11.25 mm \approx 12 mm$$

$$l_p = 1.5*12 mm = 18 mm$$

Transverse inertia bending stress

Centripetal force per unit of connecting rod length, acting at the crankpin

$C = \rho A\omega^2 r, \quad newton/metre$

$\rho_{steel} = 7830 Kg/m^3$

$A = 11t^2 = 11*2.12^2 = 4.94384*10^{-5} m^2$

$r = 27.75*10^{-3} m$

$\omega = \frac{2\pi n}{60} = \frac{2\pi(6000)}{60} = 628.32 \ rad/s$

$\therefore C = 7830*4.94384*10^{-5}*624.32^2*27.75*10^{-3} = 4240.83 \ N/m$

Maximum bending moment $= 0.128 F_n l$

$F_n = \frac{1}{2} Cl$

Thus; max bending moment

$\frac{0.128 Cl^2}{2} = \frac{0.128*4240.83*0.10^2}{2} = 2.714 Nm$

Maximum bending stress; $F_b = \frac{M_{max}}{Z} = \frac{2.714 Nm}{1.33*10^{-7} m^3} = 20.41 MPa$

$Z = \frac{Ixx}{depth/2}, Ixx = \frac{419t^4}{12} = 7.05*10^{-10} m^4$

$\therefore Z = \frac{7.05*10^{-10} m^4}{10.6*10^{-3}/2} = 1.33*10^{-7} m^3$

Crank angle, ? which highest value of max bending moment

$\theta = 90 - \frac{3500}{\left(l/r + 7.82\right)^2}$ $\qquad l/r = 100mm/27.75mm = 3.6$

$= 63.16^\circ \ from \ TDC$

The summary of connecting rod dimensions:

Items	Dimensions (mm)
Width of connecting rod	8.48
Depth of middle of connecting rod	10.6
Depth at crank end	12.0
Depth at piston end	8.0
Diameter of big end	15.03
Width of big end	21.04
Diameter of small end	12.0
Width of small end	18.0
Buckling load	11.3883KN

Table 4.3 : dimensions of connecting rod

4.3.4 Dimensions of Crankshaft

The material chosen for the crankshaft is alloy steel having ultimate tensile strength of about 784.0 to 940.0 MPa.

Specification of crankshaft to be design.

Bore and stroke = 50 mm * 55.5 mm

Rpm = 2000 rev/min

Mean Effective Pressure = 0.30 Mpa

Max combustion pressure = 1.45 Mpa

Crank angle = 36 deg

Gas pressure = 0.975 Mpa

Flywheel in used as pulley 19.62KN

Total belt pull = 6.75KN

L/R = 10/2.775 = 3.6

It is assumed that the lengths of the main bearings are equal.

Piston gas load; P $= \frac{\pi}{4} D^2 * P_{max} = \frac{\pi}{4} * (50*10^{-3})^2 * 1.45*10^6 = 2.847KN$

Assume; $f = 5*50*10^{-3} = 0.1$m

$a = a' = f/2 = 50*10^{-3}$m.

$H_1 = H_2 = P/2 = 1.4235$ KN

In such shaft, length of bearing is taken to be equal.

$V_3 = V_2 = F/2 = 19.62/2 = 9.81$KN

$$H_2' = H_3 = \frac{F_1 + F_2}{2} = \frac{6.75}{2} = 3.375KN$$

Crankpin; bending moment at center of pin

$$M_b = H_1 * f/2 = 1.4235*1000*50*10^{-3} = 71.175Nm$$

$$\therefore \pi/32 * d^3 * f_b = 71.175$$

Assume $f_b = 108.164MPa$

$\pi/32 * d^3 * 108.164 = 71.175$

$d^3 = 6.703$

$d = 1.88cm = 18.8mm \approx 20mm$

Allowable bearing pressure ~ 9MPa = P_b

$P = ldp$

$l = P/dp_b = \frac{2.847*1000}{1.88*10^{-2}*9*10^6} = 0.0168m = 01.68cm = 16.8mm \approx 17mm$

Left hand crank;

Thickness of web;

$t = 0.65d + 6.35mm$

$= 0.65(20) + 6.35mm$

$= 19.35mm$

$\approx 20mm$

Width

$w = 9/8\, d + 12.7mm$

$= 9/8\,(20) + 12.7$

$= 35.2mm \approx 36mm$

Bending moment;

$= H_1\left(\frac{f}{2} + \frac{l}{2} - \frac{t}{2}\right) = 1.4235 * (50*10^{-3} - 8.5*10^{-3} - 10*10^{-3}) = 0.045KNm$

Section of modulus of arm;

$z = 1/6\, wt^2$

$= 1/6\,(36*10^{-3})(20*10^{-3})^2$

$= 2.4*10^{-6} m^3$

Bending stress; $f_b = \frac{M_b}{z} = \frac{0.045*1000}{2.4*10^{-6}} = 18.75MPa$

Direct compressive stress;

$$f_d = \frac{H_1}{wt} = \frac{1.4235 * 1000}{(20 * 10^{-3})(36 * 10^{-3})} = 1.99 MPa \approx 2.0 MPa$$

Total stress; $f_b + f_d$ = 18.75+1.99 = 20.74 MPa must not exceed allowable stress in bending.

Right hand crank

Main bearing are assume to be at the same length, $H_1 = H_2$. Analysis of the right hand crank arm will be the same as for left hand crank arm.

Shaft under flywheel. ***(length of the bearing are taken to be equal)***

$$l_1 = l_2 = l_3 = \left(\frac{f}{2} - \frac{l}{2} - t\right) * 2 = 2 * (50 * 10^{-3} - 8.5 * 10^{-3} - 20 * 10^{-3})$$
$$= 0.043m$$
$$= 43.0mm$$

assume width of flywheel $= 30mm.$

$\therefore g = 43mm + 30mm = 73mm$

allowing space for gearing and clearance; $g = 80mm$

bending moment due to flywheel weight

$$M_f = \frac{Fg}{4} = \frac{19.62 * 80 * 10^{-3}}{4} = 0.3924\ KNm$$

Bending moment due to belt pull.

$$M_b = (F_1 + F_2) * \frac{g}{4} = \frac{(6.75)(80 * 10^{-3})}{4} = 0.135 KNm$$

Total bending moment;

$$M_{to} = \sqrt{(M_f^2 + M_b^2)} = 0.4149 \approx 0.415 KNm$$

$$\therefore \frac{\pi}{32} dw^3 * f_b = 0.415 * 1000 \qquad taking\ f_b = 42 MPa$$

$$dw^3 = \frac{0.415 * 1000 * 32}{42 * 10^6 * \pi} = 1.0065 * 10^{-4}$$

$$\therefore dw = 0.0465m = 46.5mm \approx 47.0mm$$

Crank at angle of maximum torsional moment.

$$piston\,gas\,load;\ P' = \frac{\pi}{4} * \left(50*10^{-3}\right)^2 * 0.975*10^6$$
$$= 1914.41N$$
$$= 1.91441KN \approx 2.0KN$$

$$conrod\,thrust;\ Q = \frac{P'}{\cos\phi}$$

$$\sin\phi = \frac{\sin\phi}{L/R} \qquad \theta = 36^\circ \qquad L/R = 3.6$$

$$\sin\phi = \frac{\sin\theta}{L/R} = \frac{\sin 36}{3.6} = 0.1633$$

$$\therefore \phi = 9.4^\circ$$

$$Q = \frac{2}{\cos 79.4} = 2.03KN$$

$$P_r = Q\sin(\theta+\phi);\ P_r = Q\cos(\theta+\phi)$$

$$\therefore P_t = 1.45KN$$
$$P_r = 1.43KN$$

$$Ht_1 = Ht_2 = \frac{P_t}{2} = \frac{1.45}{2} = 0.725KN$$

$$Hr_1 = Hr_2 = \frac{P_r}{2} = \frac{1.43}{2} = 0.715KN$$

Crankpin; bending moment at the center of crankpin

$$d_2^3 = \frac{0.05123*1000*16}{\pi*42*10^6} = 6.212*10^{-6}$$

$$d_2 = 0.01838m$$

$$= 18.38mm \approx 20mm$$

right hand crank web

bending moment due to radial component,

$$M_{br} = H_{r2}\left(\frac{f}{2} - \frac{l}{2} - \frac{t}{2}\right) = 0.715(50*10^{-3} - 8.5*10^{-3} - 10*10^{-3})$$

$$= 0.02252 KNm$$

$$f_{br} = \frac{M_{br}}{\frac{1}{6}wt^2} = \frac{0.02252*1000*6}{(36*10^{-3})*(20*10^{-3})^2} = 9.4 MPa$$

bending moment due to tangential component

$$M_{bt} = P_t\left(R - \frac{d_2}{2}\right) = 1.45\left(27.75*10^{-3} - \frac{20*10^{-3}}{2}\right)$$

$$= 0.02574 KNm$$

$$\therefore f_{bt} = \frac{M_{bt}}{\frac{1}{6}wt^2} = \frac{0.02574*1000*6}{(36*10^{-3})*(20*10^{-3})^2} = 10.729 MPa$$

direct compressive strength;

$$f_d = \frac{P_r}{2wt} = \frac{1.43}{2*(36*10^{-3})*(20*10^{-3})} = 0.993 MPa$$

total compressive strength;

$$f_c = f_{br} + f_{bt} + f_d = (9.4 + 10.729 + 0.993) MPa$$

$$= 21.122 MPa$$

Twisting moment on the arm;

$$M_t = H_{t2}\left(\frac{t}{2}-\frac{l}{2}\right)$$
$$=0.725\left(10*10^{-3}-8.5*10^{-3}\right)$$
$$=1.09*10^{-3}\,KNm$$

$$Z_p = polar\,section\,mod$$
$$=\frac{wt^2}{4.5}$$

$$\therefore f_s = \frac{M_t}{Z_p} = \frac{1.09*10^{-3}*4.5}{\left(36*10^{-3}\right)\left(20*10^{-3}\right)^2} = 0.340\,MPa$$

combined stress;

$$f_{c\max} = \frac{f_c}{2} + \sqrt{\left(f_s^2 + (f_c/2)^2\right)}$$
$$=\frac{21.122}{2}+\sqrt{(0.340)^2+\left(\frac{21.122}{2}\right)^2}$$
$$=21.127\,MPa$$

journals; analysis on bearing 2.

$$Total\ reaction\ at\ this\ bearing \qquad =\frac{P}{2}+\frac{F}{2}+\frac{F_1+F_2}{2}$$
$$=\frac{2.847}{2}+\frac{19.62}{2}+\frac{3.375}{2}$$
$$=12.921\,KN$$

$$\therefore bearing\ pressure \qquad =\frac{12.921*1000}{43*10^{-3}*20*10^{-3}}$$
$$=15\,MPa$$

this is in the range of the allowable bearing pressure for crankshaft.

The summary of crankshaft dimensions;

Items	Dimensions (mm)
Length of crankpin	17.0 *(desirable 20.0)*
Diameter of crankpin	20.0 *(desirable 27.5)*
Length between bearing centers	43.0
Thickness of crank web	20.0
Width of crank web	73.0
Length of bearings	43.0
Diameter of bearings	20.0 *(desirable 55.5)*

Table 4.4 : dimensions of crankshaft

Figure 4.1 shows the forces acting on the crankshaft at the dead center.

Figure 4.2 shows the resulting forces acting on the crankshaft at angle of maximum.

Figure 4.3 shows the resulting forces acting on the piston and connecting rod.

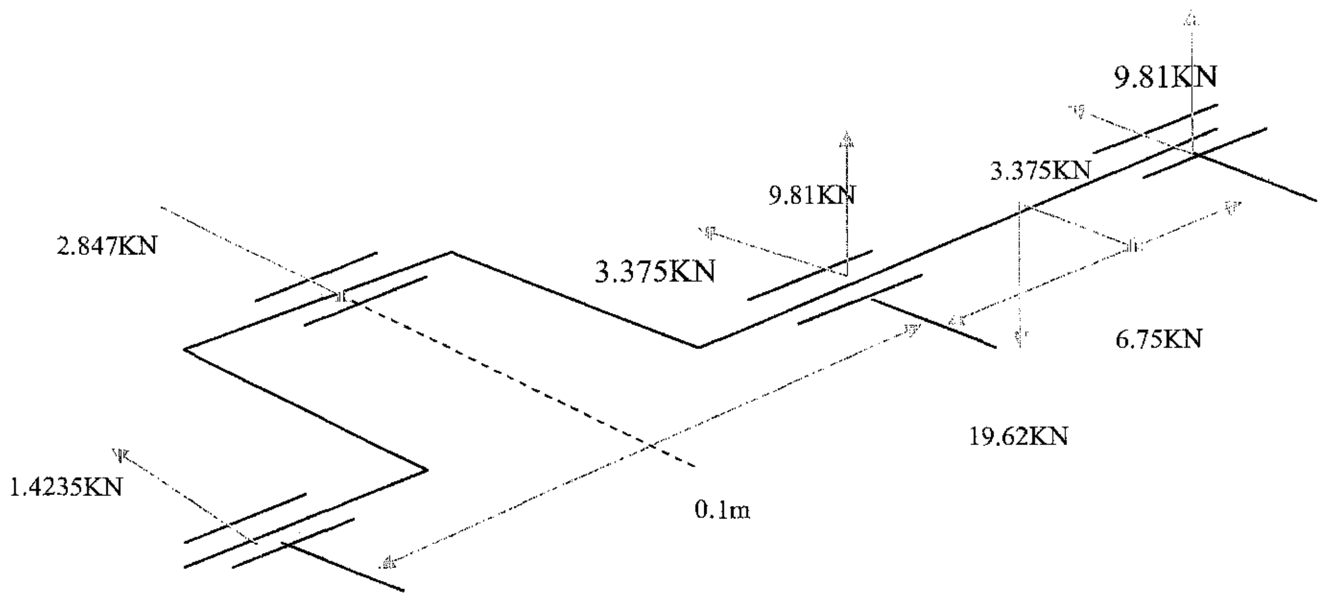


Figure 4.1:Crank at dead center

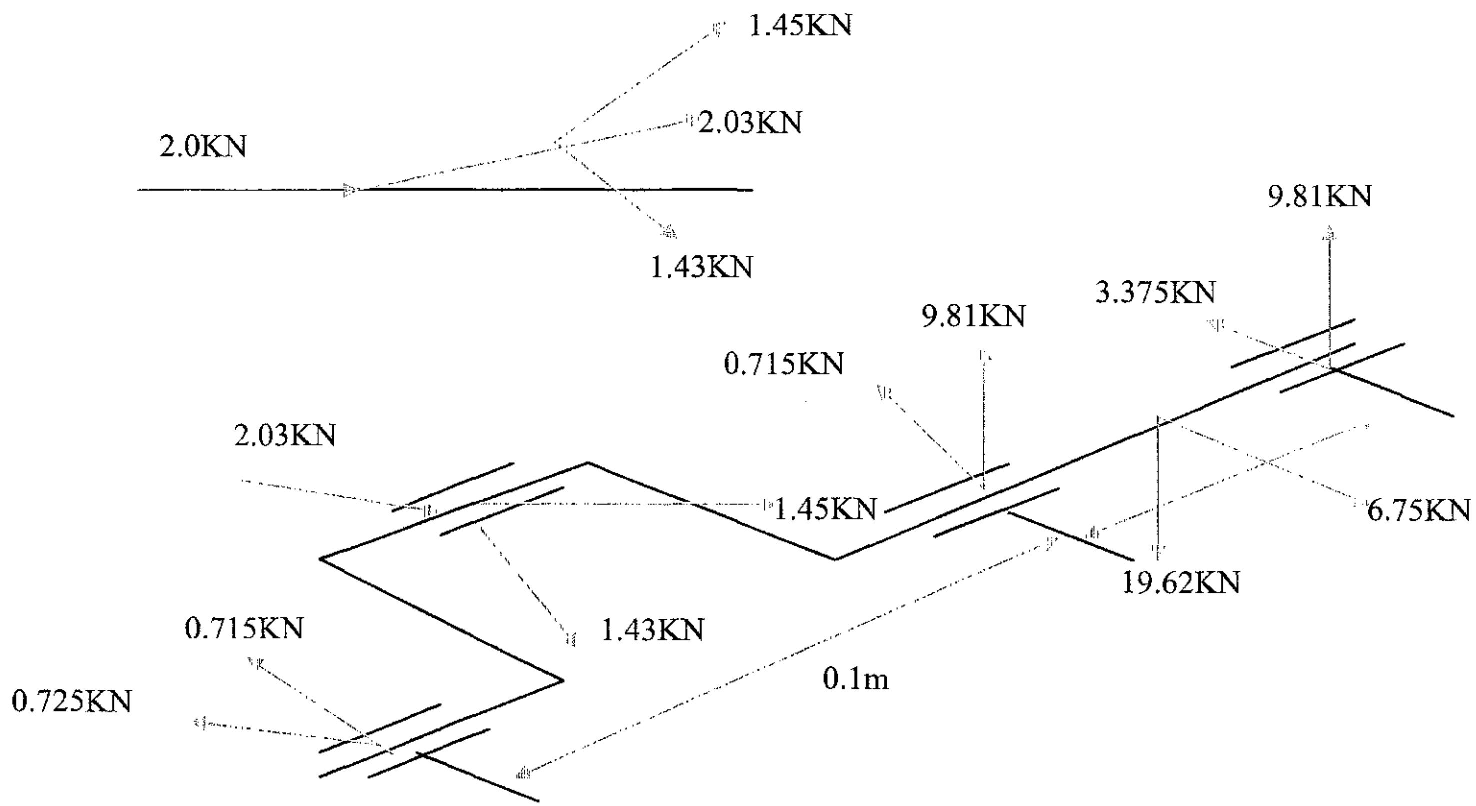


Figure 4.2: Crank at angel of maximum

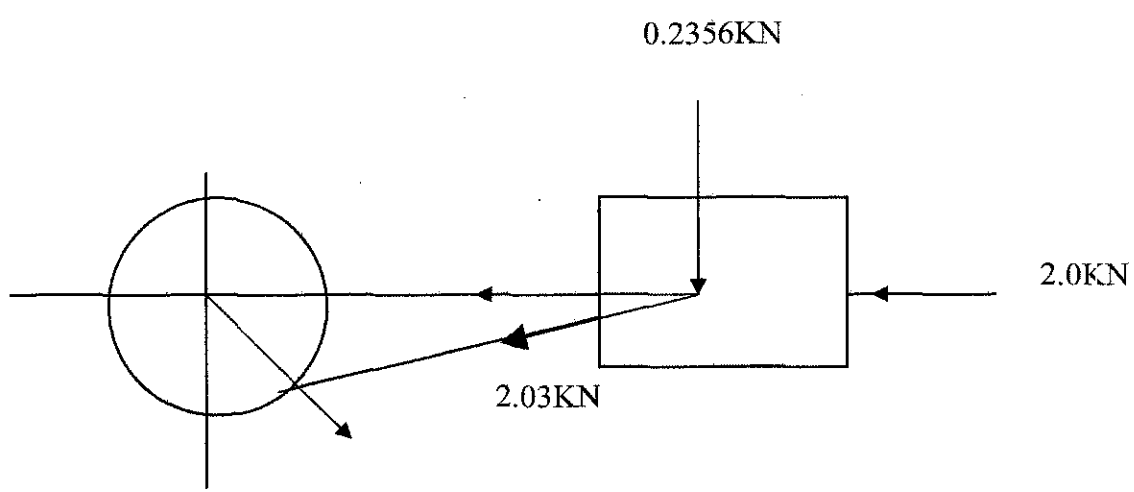


Figure 4.3 : Forces acting on piston
and connecting rod

CHAPTER 5
CONCLUSION AND RECOMMENDATION

5.2 Conclusion

This project gives the author good understanding about the steps taken to design an engine. Moreover this is an advantage for the author since he is taking an automotive course as specialization. This gives him the opportunity to learnt, integrate and put into practices the learning experience for the past four year of study period in University Technology of PETRONAS.

For this project, the author had decided to study the design procedure thus design the selected component. These components are piston, connecting rod, crankshaft, cylinder, and the ring for piston. In the first part of the Final Year Project (FYP), the author had successfully completed of the data required to start the project. And some of the components have been design in the first part. The project continued to the second part, where the rest of the unfinished components are completed. As overall, the author has learnt much from the project and it exposed him to the excitement and enjoyment being an engineer.

5.2 Recommendation

The main objective of the project is for the author to understand the design processes and steps it takes to design an engine. The build up of an engine is not only consisting of the parts that are mention in this project. Many other things are related in these things. Student who wishes to pursue this project in the future could compare the calculated values with the actual values and verify either it agrees or not. This project could be used to make a refinement to existing engine designs to make a better and efficient engine. As this project only emphasize on the design of cylinder head, piston, connecting rod, crankshaft and bearing; continuation of this project in the future can pursue in the design of gears, valve, camshaft and other related things.

REFERENCES

1. Sharma, P.C and Aggarwal, D.K. 1997, *a text book of machine design (mechanical engineering design),* S.S Mubaruk and Brothers Publishing Division.
2. Heywood, J.B, 1988, *internal combustion engine fundamentals,* McGraw-Hill,Inc.
3. Willard W. Pulkrabek , 2002, Engineering fundamentals of the internal combustion engine; Prentice Hall
4. William H. Crouse, Donald L. Anglin , 1998, Small Engine Mechanics; McGraw-Hill
5. S Srinivasan, 1998, automotives engine; Tata McGraw Hill
6. Gordon P Blair , 1997, Design and simulation of four stroke engines; SAE international
7. R. L Norton; 2001, Design of machine; McGraw Hill
8. William A. Schuster; 2001, small engine technology; ITP publication
9. Dr Rajendra Karwa; 1999, A text book of machine design; Laxmi Publication.
10. www.howstuffworks.com
11. www.askjeeves.com

APPENDICES I : REFERENCE ENGINE SPECIFICATION

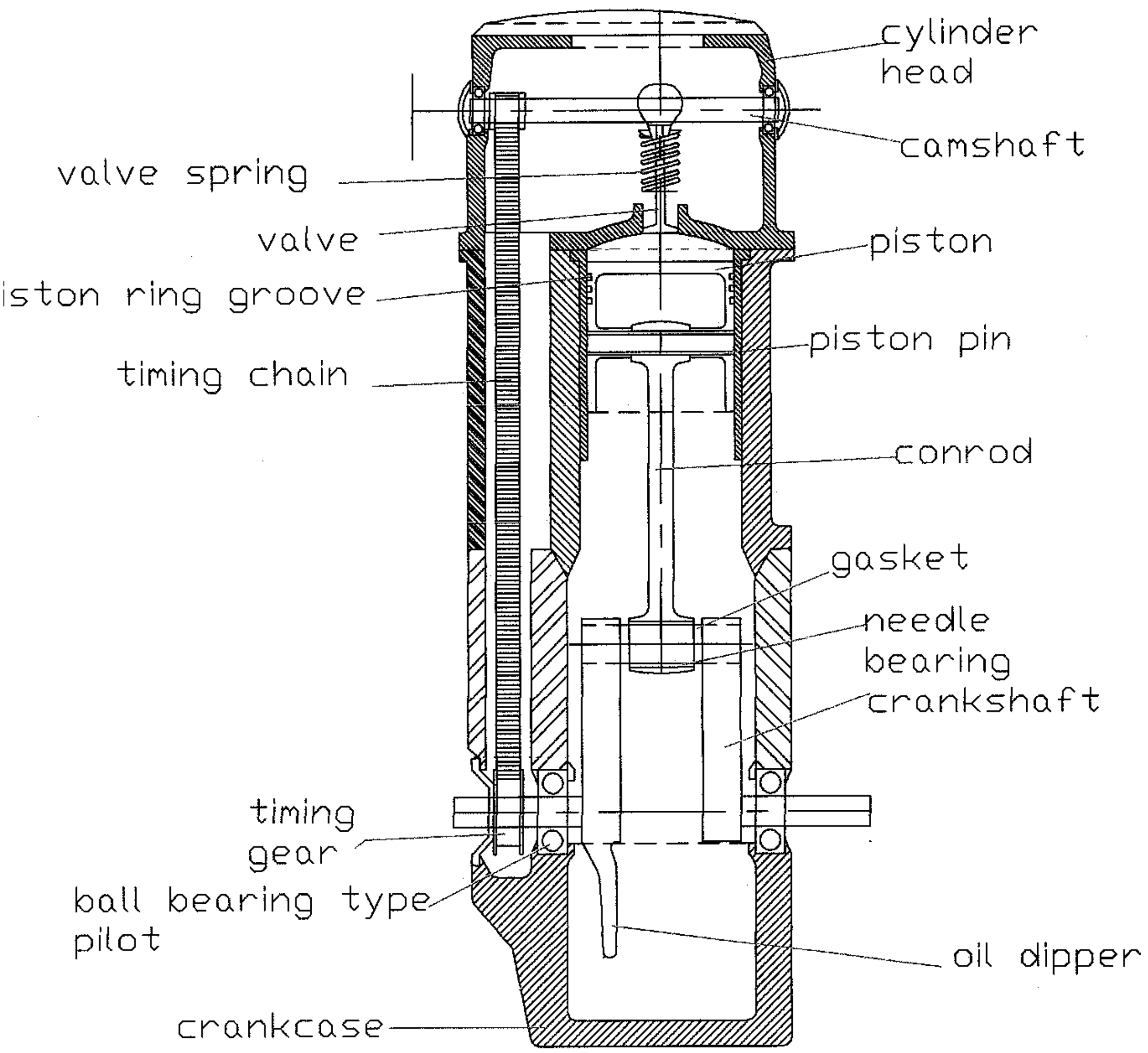
SPECIFICATIONS

Engine	4 stroke overhead cycle, air cooled, spark ignition.
Bore and stroke	50 * 55.5 mm
Displacement	110 cc
Compression ratio	9.0 : 1
Lubrication oil volume	0.9 litre
Starter system	1. Kick starter 2. Electric starter
Ignition system	Compressed direct injection (CDI)
Battery	12V-4AH electric starter
Dimension	
Overall length	1875 mm
Overall width	710 mm
Overall height	1050 mm
Wheel base	1207 mm
Chassis	
Shape	Backbone
Front suspension	Telescope
Back suspension	swingarm
Fuel tank capacity	3.7 litre
Front wheel dimension	60/100-17M/C 33P
Rear wheel dimension	70/90-17M M/C 43P
Performance	
Engine output KW (PS)	8.9 PS/8000 rpm
Engine torque (kg-m/rpm)	0.93kg.m / 6000 rpm

APPENDICES II : DRAWING

1. Engine layout.
2. Cylinder liner.
3. Piston.
4. Connecting rod.
5. Crankshaft.

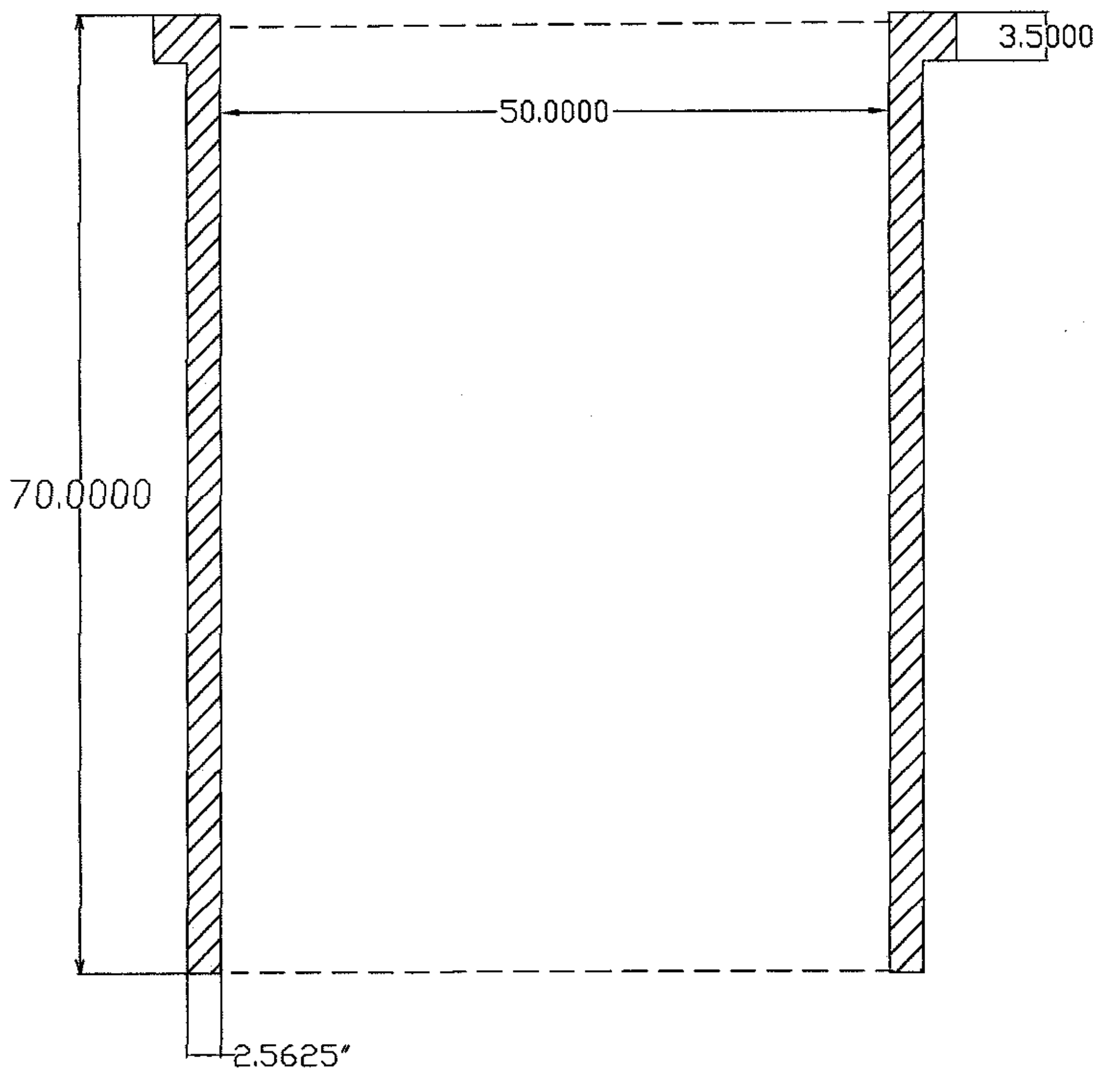
REVISIONS				
ZONE	REV	DESCRIPTION	DATE	APPROVED



component design; a,cylinder liner b,piston and ring c,connecting rod d, crankshaft	Mohd Shamani (1526) design of motorcycle engine component			
	SIZE	FSCM NO.	DWG NO.	REV
	SCALE		SHEET	

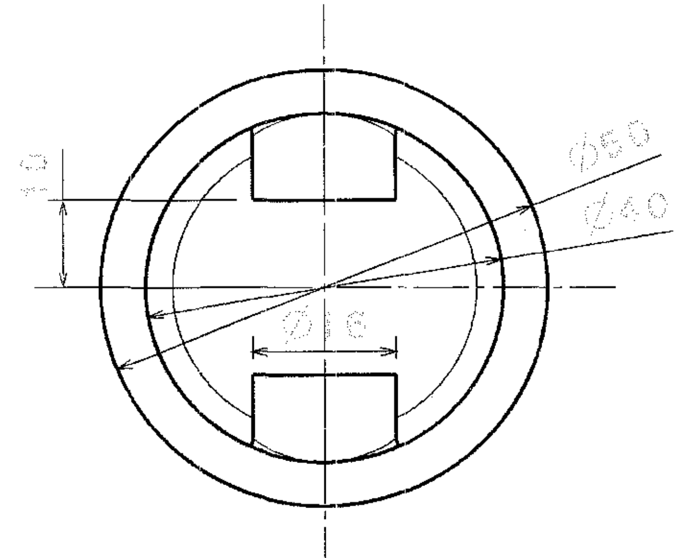
21032004 C-LINER

REVISIONS				
ZONE	REV	cylinder liner dimension	21/3/2004	APPROVED

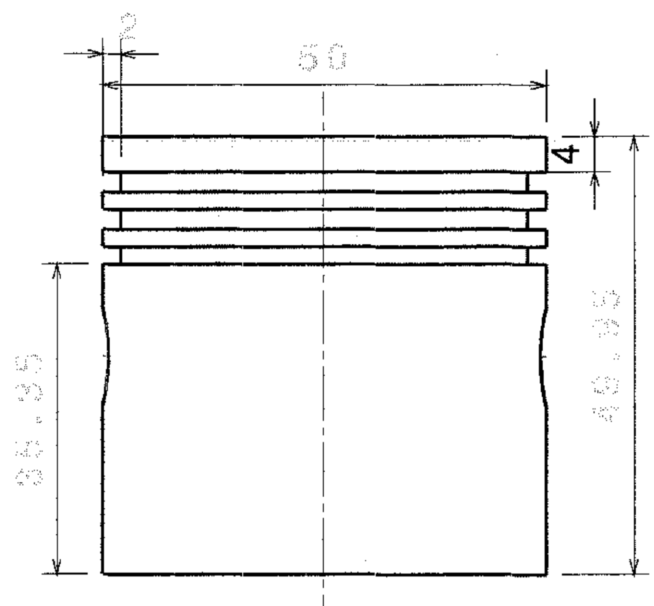


material: iron steel type: dry liner	cylinder liner			
	SIZE	FSCM NO.	DWG NO.	REV
	SCALE		SHEET	

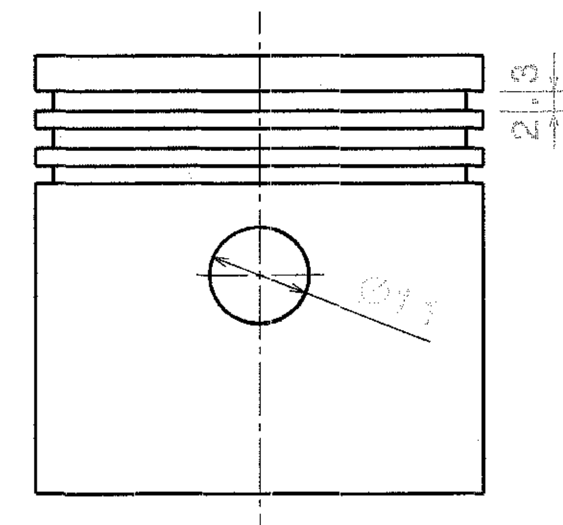
piston head
crown thickness: 4mm
skirt thickness: 3mm
piston inner ⌀: 36.2mm



Bottom view
Scale: 23:20

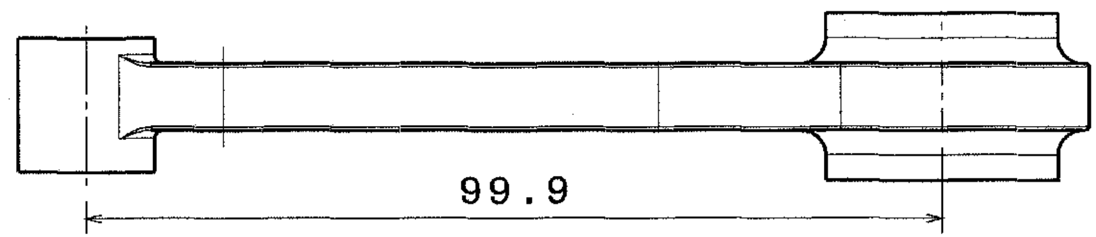


Right view

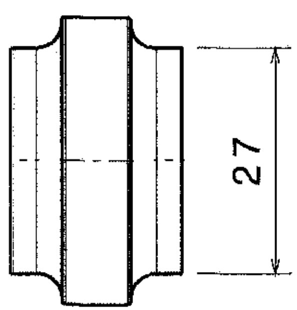


Front view

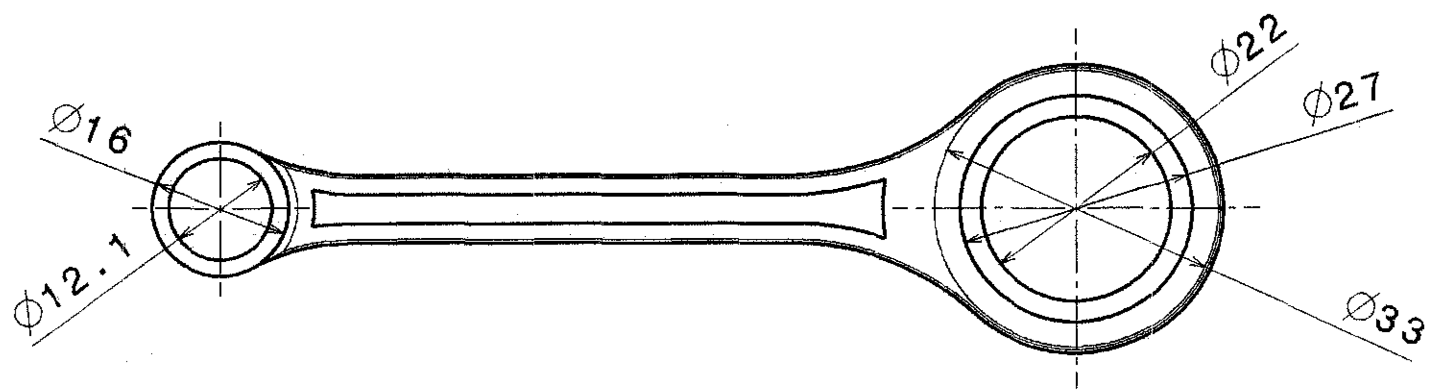
connecting rod
length: 99.9mm
small end ϕ: 12mm
small end length : 18mm
big end ϕ: 16mm
big end length : 22



Bottom view
Scale: 11:10

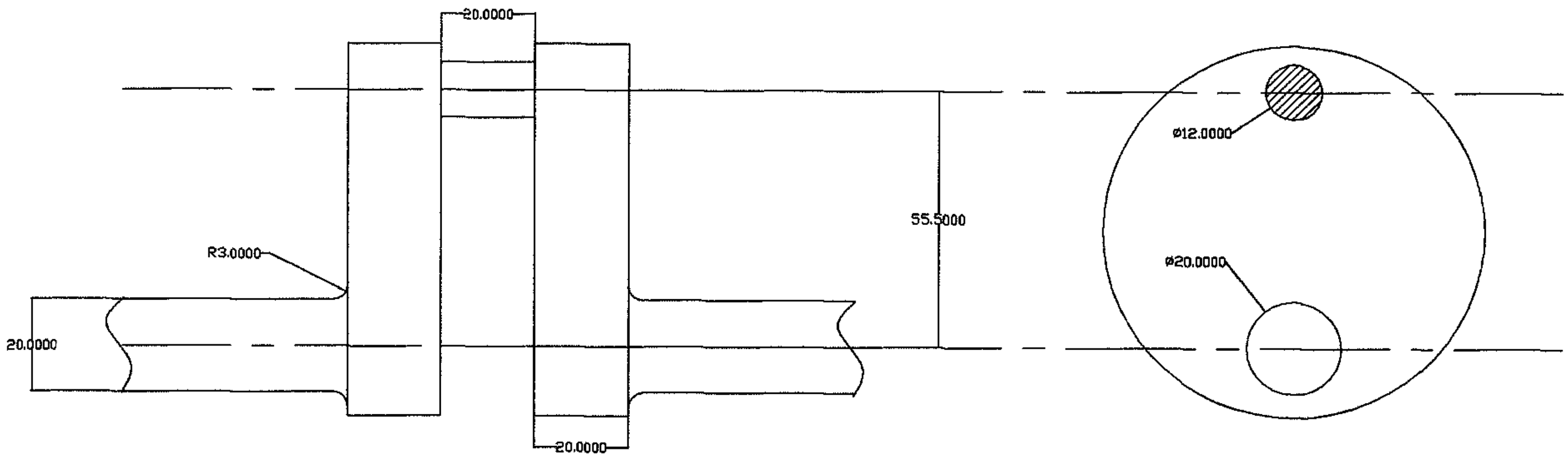


Right view
Scale: 11:10



Front view
Scale: 11:10

ZONE	REV	DESCRIPTION	DATE	APPROVED



Crankshaft			
SIZE	FSCM NO.	DWG NO.	REV
SCALE		SHEET	