Computer Aided Design and Analysis of Piston Mechanism of Four Stroke S.I. Engine

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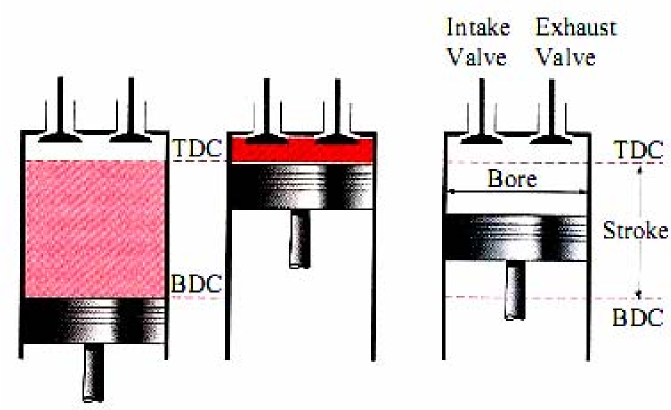
Abstract - **This paper illustrates design procedure, kinematics and dynamic analysis of piston of single cylinder of air-cooling S.I. engine at maximum power and maximum torque condition. It is the study of the geometry of motion. Kinematics analysis involves determination of position, displacement, rotation speed, velocity and acceleration and inertia force and dynamic analysis involves determination of tangential force, radial force, forces on bearing, and torque on the crankshaft. In this paper the complete kinematic and combined static and dynamic force analysis of a single cylinder, four stroke internal combustion engines are discussed. The Complex Algebra analytical approach is used for analysis is less time consuming if it is programmed for the computer solution. The pressure for one cycle at maximum power condition data for the analysis of the engine has been calculated by using Engine Pro V 3.9 engine cycle simulation program software. The computer program is prepared in visual basic language software for Kinematic and dynamic analysis of the engine at the every crank interval.**

Key words—Design, Dynamic, Engine Pro V 3.9, Kinematic, Visual Basic Program

I. INTRODUCTION

The piston is the heart of the internal combustion engine and is subjected to a combination of loads such as thermal and structural stress from a number of sources. The piston reciprocates vertically within the cylinder. The two extremes of this motion are referred to as Top Dead Center (TDC) and Bottom Dead Center (BDC) referred to Fig.1 Top Dead Center is the position of the piston that creates the smallest volume in the cylinder, which is defined as the clearance volume, *Vc*. This is where combustion takes place in the engine and is also known as the combustion chamber. The Bottom Dead Center is when the piston creates the largest volume in the cylinder. The distance between TDC and BDC is referred to as the stroke and the volume, which the piston displaces during this moment, is called the displacement volume, *Vd.*

The piston is connected to the crankshaft via the connecting rod, which allows rotation at both connections. The crankshaft converts the linear motion of the piston into rotational motion. To translate linear motion into rotational motion the crankshaft has an offset rotation.



(a) Displacement (b) Clearance Volume (c) Nomenclature Volume
Fig.1. Cross Section of a Reciprocating Engine

The internal combustion engine employs a very popular mechanism for translating motion is known as slider crank mechanism. In this paper, by using complex algebra method carried out complete kinematics and dynamic analysis of reciprocating group of machines.

Kinematic analysis of the piston-crank mechanism determines the motions of various links of the mechanism viz. displacement, velocity and accelerations of like connecting rod and piston. In the present work, the complete Kinematics analysis of the engine has been carried out by analytical method, as this method is more accurate than graphical method and can give results for all the types of the mechanism. (1)

Dynamic analysis of the engine includes static and inertia force analysis for all the possible phases of the engine, which leads to an important aspect of the engine. Neglecting the effect of friction carries out the complete force analysis of the reciprocating piston mechanism by summation of the individual effects of gas forces and inertia forces. With the help of analytical expressions derived equation, a complete structural static analysis of the single cylinder, four-stroke carried out with help of programming. [4]

II. PISTON DESIGN

The elements of a piston are shown in Fig. 2. The head thickness based on strength can be calculating assuming the head to be a flat of uniform thickness and fixed at the edges. The gas load is considered as a uniform load over entire cross section. The thickness of the piston head (t_H), according to Grashoff's formula.

$$t_H = \sqrt{\frac{3PD^2}{16\sigma_t}} \tag{1}$$

Where t_H is thickness of the piston head, P is maximum gas pressure or explosion pressure, D is cylinder bore and σ_t is allowable stress in bending (tensile) stress for the material of the piston. It may be taken as 50 to 90 MPa for aluminum alloy [5, 6].

$$t_H = \frac{H}{12.56K\left(T_C - T_E\right)} \tag{2}$$

Where H is heat flowing through the piston head and K is heat conductivity factor. The value of heat conductivity factor is 174.75 W/m/°C for aluminum alloys. T_C is the temperature at the center of the piston head, and T_E is the temperature at the edges of the piston head. The temperature difference may be taken as 75°C for aluminum with Tc about 260^0 C [5]

The heat flowing through the piston head (H) may be determined by the following expression

$$H = C\ X\ HCV\ X\ m_1 X\ BP \tag{3}$$

Where C is constant representing that portion of the heat supplied to the engine. Its value is usually taken, as 0.05 HCV is higher calorific value of the fuel. It may be taken as 47 × 10^3 kJ/kg. m_1 is mass of the fuel used and BP is brake power of the engine per cylinder.

The radial thickness (t_1) of the ring may be obtained by considering the radial pressure between the cylinder wall and the ring.

$$t_1 = D\ X \sqrt{\frac{3P_w}{\sigma_{t1}}} \tag{4}$$

Here D is cylinder bore, P_w is pressure of gas on the cylinder wall (0.025 MPa to 0.042 Mpa), and σ_{t1} is allowable bending (tensile) stress (84 MPa to 112 MPa for cast iron rings). The axial thickness (t_2) of the rings may be taken as $0.7\ t_1$ *to* t_1. The minimum axial thickness (t_2) may also be obtained from the following empirical relation is given by [5]

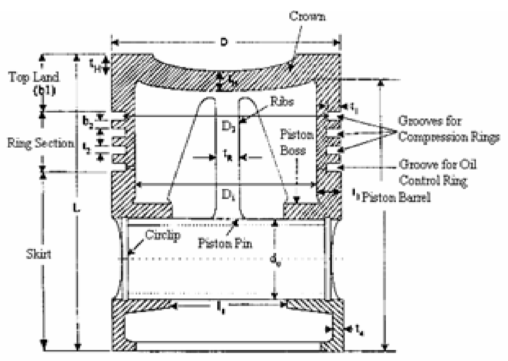


Figure. 2. Elements of a Piston

$$t_2 = \frac{D}{10n_R} \tag{5}$$

Where n_R is number of rings. The width of the top land is made larger than other ring lands to protect the top ring from high temperature conditions existing at the top of the piston.

So width of top ring land is given by

$$b_1 = t_H \text{ to } 1.2\ t_H \tag{6}$$

The width of other ring lands in the piston may be made equal to or slightly less than the axial thickness of the ring (t_2) .The lands between the piston rings are loaded by the forces of the gases, of inertia of the piston rings, and of friction. Ordinarily the land between the first and second rings is higher. The lands are gradually reduced in height away from the crown. The height of the first land, h_l is $0.04\ D$ to $0.055\ D$ for petrol engine.

$$b_2 = 0.75\ t_2\ to\ t_2 \tag{7}$$

The depth of the ring grooves should be more than the depth of the ring so that the ring does not take any piston side thrust. The gap between the free ends of the ring is given by $3.5t_1$ to $4\ t_1$ .The gap when the ring is in the cylinder, should *be* $0.002\ D$ to $0.004\ D$. The maximum thickness (t_3) of the piston barrel may be obtained from the following empirical relation.

$$t_3 = 0.03D + b + 4.5 \tag{8-i}$$

Where b is radial depth of piston ring groove, D is the bore diameter.

$$b = t_1 + 0.4 \tag{8-ii}$$

The piston wall thickness (t_4) towards the open end is decreased and should be taken as $0.25\ t_3$ *to* $0.35\ t_3$ [1]. The portion of the piston below the ring section is known as piston skirt. It acts as a bearing for the side thrust of the connecting rod. The side thrust (R) on the cylinder liner is usually taken as 0.03 to 0.1 times of the maximum gas load on the piston [5].

The maximum gas load on the piston, is given by

$$P_{Max} = P\,x\frac{\pi D^2}{4} \tag{9}$$

Maximum side thrust on the cylinder, is given by

$$R = (0.03\ to\ 0.1)\ P_{Max}\ x\frac{\pi D^2}{4} \tag{10}$$

The side thrust (R) is also given by:

$$R = P_{br}\ x\ D\ x\ L_{ps} \tag{11}$$

Where L_{ps} is length of the piston skirt and P_{br} is the bearing pressure. In actual practice, the length of the piston skirt is taken as 0.65 to 0.8 times the cylinder bore. Now the total length of the piston (Lp) is given by:

$$L_p = L_{ps} + Length\ of\ ring\ section + Top\ ring \qquad (12)$$

The length of the piston usually varies between, *1.16 D* to *1.5 D.* The material used for the piston pin is usually case hardened steel alloy containing nickel, chromium, molybdenum or vanadium having tensile strength from 710 MPa to 910 MPa. The length of the pin in the connecting rod bushing will be about 0.45 of the cylinder bore allowing for the end clearance of the pin etc. The piston pin should be designed for the maximum gas load or the inertia force of the piston, whichever is larger.

The outside diameter of the piston pin (*do*) is obtained

$$P_{Bearing\ Force} = Bearing\ Pressure\ X\ Bearing\ Area$$

$$= P_{b1}\ X\ d_0 X\ L_1 \qquad (13)$$

Where, d_o is outside diameter of the piston pin, L_1 is length of the piston pin in the bush of the small end of the connecting rod *(0.3D* to *0.45D), P_{b1}* is bearing pressure at the small end of the connecting rod bushing. The pin diameter is selected up to an optimum of about 40 percent of piston diameter based on a maximum bearing pressure of 60 MPa in aluminum [2].

The piston pin may be checked in bending by assuming the gas load to be uniformly distributed over the length L_1 with supports at the centre of the bosses at the two ends. Therefore, the maximum bending moment at the centre of the pin,

$$M = \frac{PD}{8} \qquad (14)$$

The hollow piston pin is mostly used if *di* and d_0 is the outside and inside diameters of the piston pin. The section modulus,

$$Z = \frac{\pi}{32} x \left[\frac{(d_0)^4 - (d_i)^4}{(d_0)^4} \right] \qquad (15)$$

The inside diameter of the piston pin has the limited range of *0.75* d_0 to *65.0* d_0

The maximum bending moment is,

$$M = Z\sigma_b \qquad (16)$$

Where, σ_b is allowable bending stress 84 Mpa for case hardened steel and 140.0 Mpa for heat-treated alloy steel.

III. CRANK SLIDER MECHANISM

A. Theoretical Analysis

1) *Kinematics analysis by complex Algebra Method*

The kinematics drawing and vector equation expressing complex rectangular notation of the piston crank planar mechanism is shows in Fig.3.

$$\vec{R}_1 + \vec{R}_2 + \vec{R}_3 = 0 \qquad (17)$$

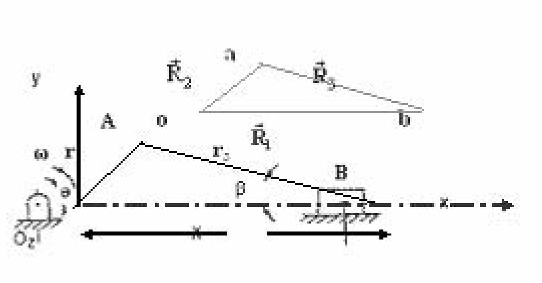


Figure .3 Graphical representations of vector

Expressing the above vectors in complex rectangular

$$(-r_1 + j0) + (r_2 \cos\theta + j\, r_2 \sin\theta) + (r_3 \cos\beta + j\, r_3 \sin\beta) = 0 \qquad (18)$$

$$-r_1 + r_2 \cos\theta + r_1 \cos\beta = 0 \qquad (19)$$

$$r_2 \sin\theta - r_3 \sin\beta = 0 \qquad (20)$$

Where,

r_1 = Linear Displacement of the Slider, m.
r_2 = Radius of the crank, m
r_3 = Length of the connecting rod, m
θ = Angular displacement of the crank, deg.
β = Angular displacement of connecting rod, deg.

As mobility of the mechanism is one and the rotational speed of the crankshaft is constant, input parameters are r_2, r_3, ω_2 and θ. [4]

$$\beta = \sin^{-1}\left(r_2 \sin\theta / r_3\right), \omega_3 = \frac{r_2\ \omega_2\ \cos\theta}{r_3\ \cos\beta} \qquad (21)$$

$$\alpha_3 = \frac{\left(r_2 \cos\theta - r_2\omega_2^2 \sin\theta + r_3\omega_3^2 \cos\beta\right)}{r_3 \cos\beta} \qquad (22)$$

$$V = \dot{X} = \left(r_2\omega_2 \sin\theta + r_3\omega_3 \sin\beta\right) \qquad (23)$$

$$A_p = \ddot{X} = (r_2 \sin\theta + r_2\omega_2^2 \cos\theta + r_3\omega_3^2 \cos\beta + r_3 \sin\beta) \qquad (24)$$

Where,

ω_2 = Angular velocity of the crank, rad/sec.
ω_3 = Angular velocity of the connecting rod, rad/sec.
α_2 = Angular acceleration of the crank, rad/sec^2
α_3 = Angular acceleration of connecting rod, rad/sec^2
V_p =Velocity of the piston, m/sec.
A_p = Acceleration of the piston, m/sec^{-2}
X = Piston displacement, m

With the use of above equations the values of all the above kinematics parameters at the every crank angle are calculated with the help of computer program. [3,4]

B. Static Force Analysis

The gas force acts *P* on the piston due to the combustion of fuel, which is measured by using engine Simulation software for complete one cycle, and this force varies during the cycle of operation. [9]. Fig. 4 shows graphical analysis of the gas force. From the force polygon,

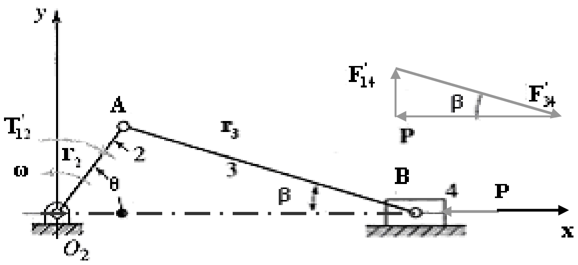


Figure .4 Analysis of Gas

Force acting along connecting rod

$$F'_{14} = P \tan\beta j \;\; , F'_{34} = P / \cos\beta \qquad (24)$$

Crankshaft torque or turning moment delivered to the crank is obtained by taking the product of the gas force and piston coordinate *x*. Therefore crankshaft torque in vector form is

$$T'_{21} = (F'_{14} g x)\, k \qquad (25)$$

C. Inertia Force Analysis

The resultant bearing loads are made up of the following components:

1. The gas force components, designated by a single prime.

2. Inertia force due to the weight of the piston assembly, designated by a double prime.

3. Inertia force of that part of the connecting rod assigned to the piston pin end, triple primed.

4. Connecting rod inertia force at the crankpin end, quadruple primed.

Equations for the gas force components have been determined earlier and reference shall be made to them in finding the total bearing loads. Fig. 3, shows graphical analysis of the forces in the engine mechanism with zero gas force and subjected to an inertia force resulting only from the weight of the piston assembly. From Fig. 5, the analytical expressions for the forces are given as (3)

$$F''_{41} = -m_4 \ddot{x} \tan\beta j \qquad (26)$$

$$F''_{34} = m_4 \ddot{x} i - m_4 \ddot{x} \tan\beta j \qquad (27)$$

$$F''_{32} = -F''_{34} \qquad (28)$$

Where,

m_4 = Mass of the piston assembly, Kg.

$$F'''_{32} = m_{3B} \ddot{x} i - m_{3B} \ddot{x} \tan\beta j \qquad (29)$$

$$F'''_{32} = m_{3A} r_2 \omega_2^2 \; (\cos\theta\, i + \sin\theta\, j) \qquad (30)$$

Where,

m_{3B} = Mass of connecting rod lumped at Wristpin B, Kg

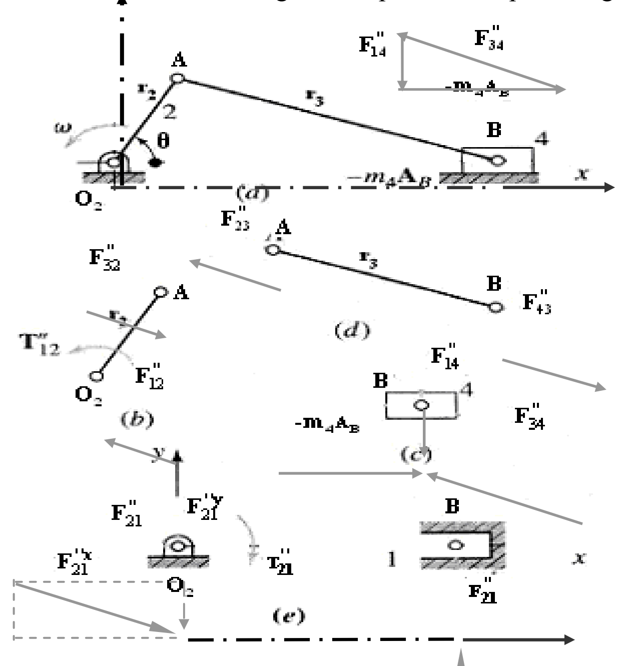


Figure .5 (a), (b,), (c), (d),(e) Analysis of force in the Engine Mechanism when only inertia force due to the weight of Piston assembly

In Fig. 6, the analysis neglecting all forces except those, which result because of the inertia of that part of the mass of the connecting rod, which is assumed to exist at the piston pin center, is given. The analytical expressions for the forces are given as [3, 4]

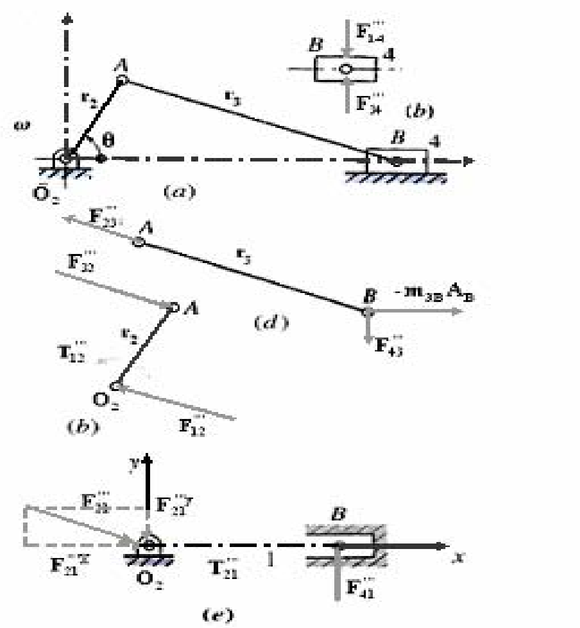


Figure. 6 (a), (b), (c), (d),(e) Graphical Analysis of Forces resulting solely from the mass of the connecting rod, assumed to be concentrated at the wrist pin end.

$$F'''_{41} = -m_{3B}\ddot{x}\tan\beta\, j \qquad (31)$$

$$F'''_{32} = m_{3B}\ddot{x}\, i - m_{3B}\ddot{x}\tan\beta\, j \qquad (32)$$

$$F'''_{34} = F'''_{41} \qquad (33)$$

Fig. 7 shows the forces, which result because of that part of the connecting rod mass, which is concentrated at the crankpin end.[3,4]

The analysis gives,

$$F'''_{32} = m_{3A}\, r_2\, \omega_2^2\, (\cos\theta\, i + \sin\theta\, j) \qquad (34)$$

Where,

m_{3A} = Mass of connecting rod lumped at crankpin A, Kg.

From the above analysis of inertia forces the total force of the piston against the cylinder wall, piston pin and crankpin respectively is given by, [4]

$$F_{41} = F'_{41} + F''_{41} + F'''_{41} = [\left(m_{3B} + m_4\right)\ddot{x} + P]\tan\beta j \qquad (35)$$

$$F_{34} = \left(m_4\ddot{x} + P\right)i - [\left(m_{3B} + m_4\right)\ddot{x} + P]\tan\beta j \qquad (36)$$

$$F_{32} = \left[m_{3A} r_2 \omega_2^2 \cos\theta - \left(m_{3B} + m_4\right)\ddot{x} - P\right] i + \left\{m_{3A} r_2 \omega_2^2 \sin\theta + \left[\left(m_{3A} + m_4\right)\ddot{x} + P\right]\tan\beta\right\} j \qquad (37)$$

Where,

F_{34} = Wristpin load, N, F_{32} = Crankpin load, N.
F_{41} = Cylinder wall force, N.

The crankshaft torque or turning moment delivered by the engine is obtained from the equation. [4]

$$T_{21} = -\left(F_{41}\cdot x\right)k = \left[\left(\left(m_{3B} + m_4\right)\ddot{x} + P\right)\tan\beta\, k\right] \qquad (38)$$

Crank speed (1) 7000 rpm. (2) 4500 rpm

Stroke = 52.5, r_3 / r_2 = 3.444

Mass of piston, piston pin and main bearing, m_4 = 0.112 Kg.

Mass of connecting rod, m_3 = 0.1151 Kg.

Centre of gravity of connecting rod is 21.42 mm from the crankpin centre,

From the above data,

m_{3B} = [7.44 / (16.86+7.44)] x 1.75 = 0.02677 Kg,

m_{3A} = 1.75 – 0.535 = 0.08832 Kg.

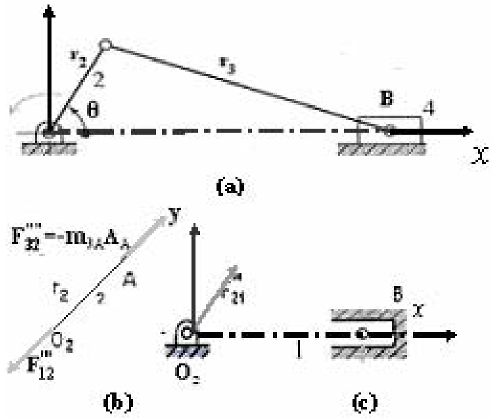


Figure.7 (a), (b,), (c) Graphical analysis of forces resulting solely from the mass of the connecting rod, assumed to be concentrated at the crank pin end.

III. VISUAL BASIC COMPUTER PROGRAM

The computer program has been developed in visual basic language program to compute the kinematic, dynamic and bearing loads for the all possible phases for the single-cylinder, four-stroke petrol engine for the complete working cycle at the every crank interval. Output of the visual basic program shown in table IV and V.

IV ENGINE SIMULATION SOFTWARE

The cycle simulation Engine pro analyzer V 3.9 program requires the basic engine design parameter as input in order to predict engine pressure for maximum pressure and temperature condition. This data can be practically measured but the cost of this is very high hence it is decided that to use simulation technique. Output of Engine simulation software for gas pressure for complete cycle at maximum power and torque condition given in Table I and II

V. RESULTS AND DISCUSSION

In the of design procedure trunk type (flat) of piston of 7.5 HP petrol engine, structural rigidity combined with lightness must always be the first consideration. The fundamental concepts and design methods concerned with single cylinders petrol engine have been studied in this paper. When a piston is designed, care should be taken to have a minimum weight with strength to withstand pressure and inertia forces, bearing are sufficient to prevent wear, gas and oil sealing of the cylinder. Piston designs for this engine by selecting aluminum alloy have also been evaluated, shown in Table III.

The results computed from the visual basic program developed for kinematic analysis of the engine for the complete working cycle are given at the crank interval of 60 degree, shown in Table IV and V

TABLE .I

GAS FORCE AT MAX. POWER (7000 RPM)

Suction		Compression		Power		Exhaust	
θ	*P*	θ	*P*	θ	*P*	θ	*P*
Deg	Bar	Deg	Bar	Deg	Bar	Deg	Bar
0	189.9	360	7733	360	7733	720	189.8
30	158.2	330	1942	390	8433	750	167.7
60	129.1	300	737.4	420	3336	660	226.3
90	132.5	270	258.2	450	1773	630	436.2
120	154.5	240	258.2	480	1172	600	436.2
150	173.6	210	208.7	510	903	570	623.4
180	186.7	180	186.7	540	764	540	764.9

TABLE II

GAS FORCE AT MAX TORQUE (4500 RPM)

Suction		Compression		Power		Exhaust	
θ	*P*	θ	*P*	θ	*P*	θ	*P*
Deg	Bar	Deg	Bar	Deg	Bar	Deg	Bar
0	208.2	360	9195	360	9195	720	207.5
30	161.1	330	2246	390	8030	750	268.0
60	160.0	300	842.7	420	3182	660	197.2
90	193.9	270	286.4	450	1716	630	166.7
120	209.5	240	286.4	480	1120	600	272.5
150	215.1	210	236.5	510	849	570	487.9
180	220.5	180	220.5	540	687	540	687.9

TABLE III

RESULTS OF PISTON DESIGN

Parameters	Results (mm)
Piston length	41
Piston crown thickness	54
Piston diameter	52
Length of piston skirt	14
Piston pin outside diameter	13
Piston pin inside diameter	8
Piston pin length	37
Piston pin thickness	5
Thickness of piston barrel	12
Piston thickness at open end	2.5

TABLE IV

BEARING LOADS AT MAXIMUM POWER CONDITION

θ (Deg)	F_{32} (N)	F_{34} (N)	F_{41} (N)	T_{21} (N-m)
0	2790.0	2799.2	0.00	0.00
60	766.51	658.90	-635.09	-16.91
120	1395.0	1600.9	1141.21	30.67
180	1533.0	1719.7	0.00	0.00
240	1395.0	1174.8	835.27	22.50
300	766.51	1553.5	-1498.9	-39.90
360	2790.0	10523	0.00	0.00
420	766.51	2654.0	2555.53	68.19
480	1395.0	2652.3	1895.74	50.82
540	1533.0	2297.9	0.00	0.00
600	1395.0	990.94	702.60	18.98
660	766.51	332.97	318.04	8.511
720	2790.0	2979.8	0.00	0.00

TABLE V

BEARING LOADS AT MAXIMUM TORQUE CONDITION

θ (Deg)	F_{32} (N)	F_{34} (N)	F_{41} (N)	T_{21} (N-m)
0	1153.1	944.7	0.00	0.00
60	316.77	162.1	-156.3	-4.18
120	576.51	812.0	578.83	15.48
180	633.55	854.0	0.00	0.00
240	576.51	299.9	213.24	5.70
300	316.77	1197	-1155.5	-30.91
360	1153.1	10348.	0.00	0.00
420	316.77	2959.8	2849.9	76.236
480	576.51	1752.9	1252.8	33.514
540	633.55	1321.5	0.00	0.00
600	640.44	314.3	222.8	5.9619
660	316.77	530.9	-512.77	-13.716
720	1153.1	1360.5	0.00	0.00

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APPENDIX

(A) SHORT BLOCK SPECIFICATION:

Title	*Detail*	*mm*	*inch*
Bore Diameter		52	2.063
Stroke		53.5	2.123
No. of Cylinders			1
Piston Rings	No. of Rings +Tension; Exp Required	3 Low Tension	
Connecting rod Length	Center Distance	94.25	3.71
Piston Skirt specs	Skirt/Bore Type-0.45, S-0.32, VS-0.18 I -0.59	Typical Skirt 0.48	
Bearing Size Coefficient	Representation of journal bearing, It-ball bearing	0.35	
Piston Top	Coating	No Coating	
Cylinder Leakage	Relative Leakage by pistons rings	Low Leakage	
Cooling Fan Type	No, Electric, Clutch, Flex, Solid Steel	No Fan	
Water Pump & Drive	None, Electric, Lower Belt ratio	No Pump	
Inertia [lb-ft^2] Crankshaft Design	High Oil drag, Typical Windage	High Oil Drag, Splash Lubrication	

(B) CYLINDER HEAD SPECIFICATIONS

	Intake Port Specs			*Exhaust Port Specs*		
Title	*mm*	*inch*		*mm*	*inch*	
Valves / ports	1 Valve			1 Valve		
Valves Diameter	23	0.905		20	0.79	
Avg. Port Diameter	20.65	0.813		18.6	0.73	
Port Length	66	2.598		61.7	2.04	
Anti Reversion %	0			0		
Single flow Coefficient	Valve Lift (Inch)	L/D	CFM	Valve Lift (inch)	L/D	CFM
Use flow table	0.04	0.05	7.09	0.039	0.05	7.55
	0.91	0.1	14.78	0.078	0.1	14.76
	0.13	0.5	21.98	0.118	0.5	21.78
	181	0.2	25.27	0.157	0.2	27.02
	0.22	0.25	25.88	0.196	0.25	30.91
	0.27	0.3	26.3	0.236	0.3	32.71
	0.317	0.35	26.4	0.157	0.2	27.02
Combustion Chamber						
Comp. Ratio		9.0				
Chamber Design	Hemi, Pent roof, Dual plug etc	Flat				
Material / Coating	Aluminium, Cl, Coated Cl, Coated Aluminium	Aluminium Alloy				
Swirl Rating	No, Some, Good	No				

(C) INTAKE AND EXHAUST SYSTEM SPECIFICATION

Intake system specification			*Exhaust System specification*		
Title	*mm*	*inch*	*Title*	*mm*	*inch*
Runner Dia	22.5	0.88	Runner Length	109	4.32
Runner Length	111.4	4.38	Inside Dia at head	21.4	0.8425
Runner Length Coeff.	2.3		Total length	67.9	2.67
Runner Taper,	0 Deg		Design	18" stepped	
Manifold Type	Ind Runner + Carb		Runner Flow Coefficient	0.591	
Intake Heat	No Heat		Test Pressure	20	
Fuel Delivery Calc	Not required for two wheeler		# Valve / cy		
			Valve lift tested	5	
Total CFM Rating	60		Flow w/o runner (CFM)	30.91	
Air Cleaner Specification			Runner Dia	21.3	
Air Cleaner CFM	112		Flow with runner (CFM)	21.29	
Shape	Square		Inside Dia at Exit	30.4 mm	1.196 inch
Length / Diameter	100		Exhaust / Muffler system	Full Exhaust	
Wight /Height	100		CFM rating	14	
Element Type	Foam		Engine HP	8.5	
Silencing Snorkel	No		Type of Vehicle	Prod Sporty	
Total CFM Rating	60		Collector Specs	Detailed Collector	
			Collector Length	308 mm	12.1 inch
			Collector Diameter	58.45 mm	2.59 inch
			Collector Taper	5 Deg	

(D) CAM/VALVE TRAIN SPECIFICATION:

Intake Cam Profile		*Exhaust Cam Profile*	
Title	*Detail*	*Title*	*Detail*
Use cam file		Use cam file	
Centerline deg a TDC	100.7	Centerline deg a TDC	110.37
Actual Valve Lash	0.1	Actual Valve Lash	0.1
Rocker Arm Ratio	1	Rocker Arm Ratio	1
		Total Cam Advance	10 deg advance 10 retard Automatically Can be changed
		Lifter Profile Type	Mild Solid Roller