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Theoretical, Numerical, and Experimental Investigation of Heat Transfer through the Piston of a Single-Cylinder Petrol Engine

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Abstract: *This paper presents the validation of theoretical and numerical (experimental) investigations of heat transfer in the piston under maximum power conditions of a single-cylinder, four-stroke, spark-ignition (S.I.), air-cooled engine. The pressure and temperature inside the combustion chamber are measured using Engine Pro Analyzer software (V3.9) for one complete engine cycle. The heat transfer coefficient is calculated using Woschni's heat transfer correlation, which is more accurate and less time-consuming compared to other correlations when implemented computationally. The theoretical heat transfer through the piston is determined using the heat balance sheet method. Dry gas analysis of exhaust gases is performed using a smoke meter. A computer program is developed in Visual Basic to compute the heat transfer coefficient at the piston crown for every crank angle interval. Thermo-mechanical analysis of the piston is carried out using ANSYS 10, and modeling is performed using CATIA V9.*

Keywords: Engine pro Analyzer, Heat Transfer, Visual Basic program, Piston, Smoke meter.

I. INTRODUCTION

A piston is a vital component of an internal combustion engine (IC engine) and is used to convert chemical energy into mechanical work. It is one of the most highly stressed components in a reciprocating IC engine. The type most familiar to people, due to its widespread use as the prime mover in motor vehicles, is the reciprocating engine, which operates on either gasoline or diesel fuel. The operating principle of a reciprocating engine involves a series of processes that occur sequentially during one or two revolutions, forming the "working cycle" [8]. Heat transfer between the piston crown and combustion gases in an internal combustion engine occurs predominantly by convection and only to a small extent by radiation. This heat transfer is a critical parameter for simulation and analysis. Heat transfer through the piston significantly influences cylinder pressure, temperature levels, engine efficiency, friction, and exhaust emissions. During the combustion process, the peak gas temperature is limited to 2500 K as metal components of the combustion chamber can only withstand 600 K for cast iron and 500 K for aluminum alloys.

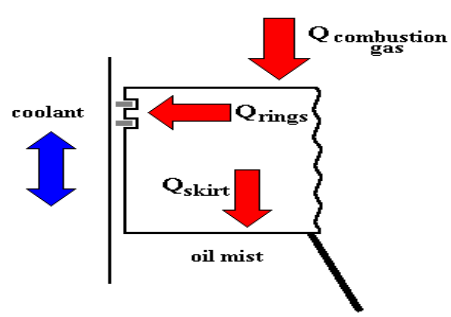

Fig.1. Piston Heat Transfer Model [3]

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Therefore, proper cooling of the cylinder head, engine block, and piston is essential to ensure optimal engine performance. The engine lubricant may oxidize if the temperature exceeds 450 K. Hence, investigating heat transfer through the piston within the cylinder is of great importance. This paper presents a detailed analysis of heat transfer through the piston using both numerical and theoretical methods under maximum power conditions.

II. ENGINE SPECIFICATION

Table. I.: The specification of the Petrol engine

PARAMETER	SPECIFICATION
Type	Four stroke Petrol inline
Bore	52.4 mm
Stroke	57.8 mm
No. Of Cylinders	1
Compression Ratio	9 .5: 1
Max. Power	6.6 KW@ 7000 RPM
Max. Torque	10.35 Kg-Mach @ 4000 RPM
Oil Sump Capacity	900 ml
Cooling type	Air cooling
Reciprocating Mass	0.150378 kg

III. PISTON MATERIAL SPECIFICATION

Table ii: Chemical Combination

Cu	Pb	Sn	Si	Mg	Zn	Fe	Mn	Ni	Ti	Al
2.99	0.23	0.42	14.98	1.49	0.46	0.76	0.046	0.91	0.032	Rest

Physical Property:

a) Density :2720 Kg/m^3

Mechanical Property:

a) Modulus of elasticity: :69 GPa

Thermal Property:

a) Coefficient of Thermal Expansion: 21.5 x 10^{-6} m/m.ºC

b) Thermal Conductivity : 137 W/m-K

IV. MODELING AND STEADY STATE THERMAL ANALYSIS OF PISTON

The piston of an engine includes irregular parts and shapes (such as pin holes, ribs, skirt, and concave land slots on the piston top surface, etc.), so it is very difficult to measure the exact 2-D dimensions of the piston along with geometrical and dimensional tolerances. A few experimental methods are in practice, such as digitization using a coordinate measuring machine for measuring the dimensions of complicated parts of an automobile engine. However, these methods are very expensive and time-consuming Hence, the 2-D dimensions of the piston are measured manually with the help of traditional measuring devices like a vernier caliper, height gauge, etc., using the reverse engineering method [52]. The values of geometrical and dimensional tolerances are taken from manufacturer catalogs and after discussion with R&D personnel involved in the piston design area. Surface and solid modeling of the piston are done in CATIA with the help of the manually measured dimensions, and then the solid model is imported into ANSYS 20. Due to the large number of elements (i.e., 102,577) and the resulting high number of degrees of freedom, it is difficult to perform transient coupled field analysis. Hence, steady-state coupled field structural thermal analysis is carried out to calculate the centre temperature of the piston. To determine the thermal boundary conditions for the thermal analysis, different thermal boundary conditions are required to be established [2].

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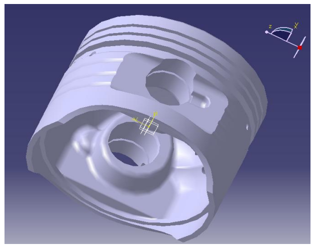

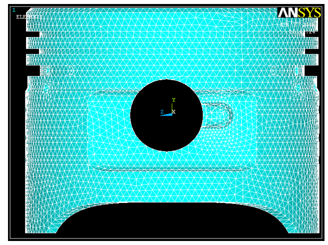

Fig. 2 Modeling in CATIA V9 3. Thermal Analysis in ANSYS10 software

The following (iv-ix) thermal boundary conditions are calculated with help of reference [9, 10]

Maximum Combustion Gas Pressure = 39.77 bar
HTC Average gas temperature = 1393.26 K
HTC Average on Top surface = 300.21W/m^2-K
The ring land boundary-condition = 659.56 W/m^2-K
Underside boundary-condition =1320.45 W/m^2-K
Piston pin boundary-condition =1000 W/m^2-K
Crevice heat transfer boundary-condition =70.075 W/m^2-K
Convective coefficient between region = 115 W/m^2-K
Between the rings
Piston skirt boundary Condition. = 375 W/m^2-K

V. NUMERICALLY ANALYSIS

Engine Simulation Program

Measurement of these values through experimentation at maximum power conditions is a very costly process; hence, engine simulation software is used. The cycle simulation program (Engine Analyzer Pro) requires basic engine design parameters as input in order to predict engine pressure and temperature under maximum power conditions. The main feature of this program is that it can calculate all parameters, including dynamic valve lift, valve flow area, etc., at every 4 degrees of crank rotation. Some parameters, such as cylinder pressure, port pressure, net valve flow, and heat losses, are calculated at every 0.1 degrees of crank rotation. The required input data for the engine cycle simulation software (seven main specifications) are as follows.

- Cylinder head
- Inlet system specs, Exhaust system specs
- Cam/Valve train dynamics
- Turbocharger/supercharger specifications

TABLE III: PRESSURE AT MAXIMUM-POWER-CONDITION

Suction		Compression		Power		Exhaust	
θ	*P*	θ	*P*	θ	*P*	θ	*P*
Deg	Bar	Deg	Bar	Deg	Bar	Deg	Bar
0	0.894	360	36.4	360	36.4	720	0.893
15	0.858	345	14.8	375	56.9	705	0.819
30	0.737	330	9.14	390	39.7	690	0.789
45	0.655	315	5.45	405	24.1	675	0.901
60	0.607	300	3.47	420	15.7	660	1.065
75	0.598	285	2.39	435	11.0	645	1.255
90	0.624	270	1.79	450	8.35	630	1.461
105	0.671	255	1.43	465	6.62	615	1.707

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120	0.727	240	1.21	480	5.51	600	2.054
135	0.778	225	1.07	495	4.75	585	2.476
150	0.817	210	0.98	510	4.25	570	2.935
165	0.848	195	0.92	525	3.89	555	3.289
180	0.879	180	0.87	540	3.601	540	3.601

TABLE IV: TEMPERATURE AT MAXIMUM-POWER-CONDITION

Suction		Compression		Power		Exhaust	
θ	*T*	θ	*T*	θ	*T*	θ	*T*
Deg	(K)	Deg	(K)	Deg	(K)	Deg	(K)
0	560.4	360	1389	360	1389	720	574.8
15	488.4	345	678.4	375	2562.	705	707.4
30	437.5	330	607.5	390	2487	690	830.1
45	389.0	315	537.1	405	2249	675	911.0
60	353.2	300	481.3	420	2053	660	970.2
75	331.9	285	438.8	435	1898.	645	1021
90	323.3	270	407.9	450	1780	630	1071
105	321.5	255	384.5	465	1679	615	1121
120	323.0	240	367.7	480	1597	600	1183
135	325.8	225	355.4	495	1530	585	1246
150	323.0	210	346.4	510	1478	570	1308
165	331.8	195	339.8	525	1435	555	1354
180	335.3	180	335.3	540	1396.	540	1396

Calculations of Heat-Transfer-Coefficients between the hot Gases and Piston-Crown Surface

The piston receives heat from the hot gases formed by the combustion of an air–fuel mixture. The boundary conditions around the piston body vary from region to region. In this work, the heat transfer coefficient is calculated at the piston crown. The mathematical description of forced fluid flow over a cylinder surface is very complex. Similarly, in internal combustion engine components, especially the piston, the effect of hot gases is highly complex. Therefore, to calculate the heat transfer coefficient at the piston crown surface, heat transfer is modeled as forced convection inside a cylinder.The heat transfer from the combustion gases is assumed to be similar to the turbulent heat transfer of gases in a cylinder, as given

$$N_u = C\,Re^m P_r^n \tag{1}$$

Here, Nu is the Nusselt number, Re is the Reynolds number, and Pr is the Prandtl number. The exponent *m* is typically taken as 0.8 for fully developed turbulent flow, while *n* is 0.3 for heating and 0.4 for cooling conditions [6, 8]. The heat transfer coefficient depends on engine geometric parameters such as the exposed cylinder area and bore, as well as piston speed. It also varies with location on the piston crown and with piston position during the engine cycle. Notably, different empirical correlations for the heat transfer coefficient can result in significantly different predictions of both instantaneous and mean heat transfer coefficients [6, 8].

a) Woschni

$$h_g = 3.26\, b^{-0.2} (2.28 V_p)^{0.8} T_g^{-0.53} P^{\,0.8} \tag{2}$$

b) Hoenberg

$$h_g = 130 V^{-0.06} (V_p + 1.4)^{0.8} T_g^{-0.4} P^{0.8} \tag{3}$$

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c) Eichelberg

$$h_g = 2.43 V_p^{0.33} (P T_g)^{0.5} \tag{4}$$

Where,

Where h_g is the heat-transfer-coefficient, is the Gas pressure T_g is the temperature of combustion gas, b is tbore diameter and V_P is the mean speed of piston Three different heat transfer models have been considered for calculating heat transfer coefficient. However, the variation of heat transfer coefficient for three correlations is not too important, since errors in predicting the heat transfer coefficient only have a small effect on the engine performance prediction. In this paper, to model the heat exchange between gas and cylinder wall, the modified Woschni correlation equation has been used, which is more suitable for spark ignition engine and required less time if programmed in Visual basic language. This is given by, [6]

$$h_g = 129.8 b^{-0.2} u^{0.8} T_g^{-0.53} P^{0.8} \tag{5}$$

Woschni assumed and subsequently justified the value of *b* as 0.8 and took the characteristics length (*x*) to be the piston bore (*b*) is 52 mm. Woschni correlation argued that the characteristic velocity would comprise two parts
(a) A contribution due to piston motion.
(b) A contribution due to combustion.
And given by,

$$u = C_1 V_P + C_2 (V_S T_r / P_r V_r)(P - P_m) \tag{6}$$

Woschini correlation expressed the contribution due to combustion as function of pressure rise due to combustion, which $(P - P_m)$. Where P_m is the pressure (motoring pressure) that would occur without combustion. Vs is the swept volume, Vp is the mean piston speed Vr, Pr, Tr evaluated at any reference conditions, such as inlet valve closure. Reference pressure (P_r), temperature (T_r) and volume (V_r) at inlet valve closure at 240^0 crank angle,

Value suggested by Woschni for C_1 & C_2 are

For gas exchange process	$C_1 = 6.18$	$C_2 = 0$
For compression	$C_1 = 2.28$	$C_2 = 0$
For combustion and expansion	$C_1 = 2.28$	$C_2 = 3.24\times10^{-3}$
For IDI Engine	$C_1 = 0$	$C_2 = 6.28\times10^{-3}$

Watson and Janota suggest that the motoring pressure is evaluated by assuming the compression and expansion to be modeled by a polytrophic process

$$P_m = P_r (V_r / V)^k \tag{7}$$

Where $K = 0.3$, P_r is the reference pressure, V_r is the reference volume at inlet valve closure at 240̊ crank angle.Instantaneous engine cylinder volume without considering the effect of combustion (V) calculate as follows,

V = (Instantaneous piston displacement) x (Piston area) $= (Xp.Ap)$ (8)

$$(X_p) = r \cos\theta + \cos(2\theta) / n \tag{9}$$

Where,

r = Crank radius, θ = Crank Angle

n = Ratio of Connecting Rod Length to Crank radius

For gas side of combustion chamber components, application engine pro analysis results for the calculation of the instantaneous values for heat transfer coefficient and mean gas temperature as a function of a crank angle. The time average equivalent one $\overline{h}_g$ and $\overline{T}_g$ are calculated as follows for four-stroke cycle engine with help of numerical method least curve fitting technique. [5]

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$$\overline{h_g} = (1/4\pi)\int_0^{4\pi} h_g d\theta \ , \quad \overline{T_g} = \left(1/4\pi h_g\right)\int_0^{4\pi} h_g T_g d\theta \tag{10}$$

Notice that the mean heat transfer coefficient increase with engine speed. The former increase because there is less time for the gases to lose heat as engine speed increases because of increased gas motion at higher speed. The later increases, as the heat transfer coefficient is directly proportional to engine speed.

Heat transfer from Piston by numerically (experimentally) is given by,

$$Q = \bar{h}_g A_P \left(\overline{T_g} - T_P\right) \tag{12}$$

Where is the $\overline{T_g}$ is the mean temperature, A_p is the Area of piston, T_p is the center temperature of piston calculated from thermal analysis and $\overline{h_g}$ is the coefficient-of-heat-transfer

V. THEORETICAL ANALYSIS

There are three modes of heat transfer and these modes are conduction, convection and radiation. Heat transfer by conduction and convection requires the presence of an intervening medium while heat transfer by radiation does not require. Heat is transferred to the gases during intake and the first part of the compression stroke, and the heat transfer takes place from the gases to the walls and piston during the combustion and expansion.

Total Heat Input = Heat equivalent to effective work + Heat carried away by exhaust gases + Heat transferred to ambient air + Heat transferred to oil + Heat loss due to chemically incomplete combustion of the fuel + unaccounted). (13)

A. Heat equivalent to effective work

The first law of thermodynamics for engine cylinder systems states that [7]

$$dUs = dQ + dW \tag{14}$$

Where,

$dW = PdV,$

$dUs = mCv\, dT,$

$Dt = d\,(PV)/mR,$

$R/Cv = k - 1$

Where dUs is the change in internal energy, dQ is the heat added to the system, dW is the mechanical work done by the system, m is the working charge mass Cv is the constant volume specific heat, P is the pressure, V is volume, T is temperature, k is adiabatic constant, and R is the gas constant.

B. Heat carried away by exhaust gases (Q exhaust):

A dry gas analysis of engine exhaust gives CO, CO_2, HC, and O_2 in volumetric percentages such as α, β, γ, ψ respectively by carried out by smookemeter testing. The four components identified sum up $(\alpha + \beta + \gamma + \psi) = P$ % of the total, which means that the remaining gas (nitrogen) accounts for Q % of the total. Volume percent equals molar percents, so if an unknown amount of fuel burned with an unknown amount of air, the resulting reaction is, [6, 8]

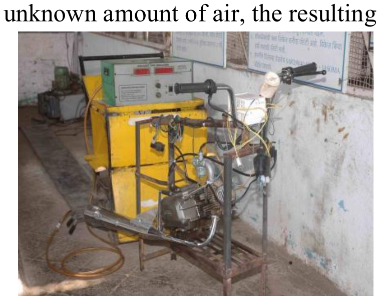

Fig. .5 Smookemeter Testing

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$$XC_8H_{18} + YO_2 + Y(3.76)N_2 \Rightarrow \alpha CO + \beta CO_2 + \gamma C_8H_{18} + \psi O_2 + ZH_2O + QN_2 \quad (15)$$

Where Z is the number of mole of water vapory removed before any analysis.
Conversion of nitrogen during gives,

$$Y(3.76) = Q \quad \text{(i-16)}$$

Conversion of carbon,

$$8X = \alpha + \beta + (8 + \gamma) \quad \text{(ii-16)}$$

Conversion of hydrogen,

$$18X = (18\gamma + 2Z) \quad \text{(iii-16)}$$

A percentage of dry gas analysis components CO, CO_2, C_8H_{18}, O_2 has been calculating for 1kg mole as well as percentage of mass.
Heat carried by exhaust gases is given by

$$Q_{Exhausts} = (m_{exhaust}/100)\,[CO\,(mC_p) + CO_2\,(mCp) + C_8H_{18}\,(mCp) + O_2\,(mCp) + H_2O\,(mCp) + N_2\,(mC_p)]\,(T_{Exhaust} - T_{Ambient}) \quad (17)$$

Where

$Q_{Exhausts}$ = Overall heat transfer (W/m^2)
$T_{Exhuast}$ = Effective gas temperature, typically 800 °C
$T_{ambient}$ = Ambient temperature, typically 80 °C

C. *Heat Transfer to the oil: (Qoil)*

$$Q_{oil} = (m\,C_p\,\Delta T) \quad (18)$$

Where,

m = is the mass of oil = 900 ml
C_p= Specific Heat of oil
ΔT= Temperature difference of oil after testing

D. Heat Loss due to unburned fuel

$$Q_{uf} = m_1 \; X \; HCV \quad (19)$$

Where, m_1 is the mass of fuel, HCV is calorific value of the fuel

E. Heat Transferred ambient air (Q_{air}) [6,8]

Heat transfer to the ambient air = (Heat transfer through engine cylinder block Q_b) + (Heat transfer through engine cylinder head Q_h) + (Heat transfer through piston Q_p)

$$Q_{air} = h_g A_b \left(T_g - T_w\right) + h_g A_h \left(T_g - T_w\right) + h_g A_p \left(T_g - T_w\right) = h_g \left(T_g - T_w\right)\left(A_b + A_h + A_p\right) \quad (20)$$

F. Heat carried by piston

Heat carried out by the piston crown, produced due the combustion of air fuel mixture at maximum power condition

$$Q_P = \frac{A_P}{A_P + A_b + A_h} Q_{Air} \quad (21$$

Where,

A_p= Area of piston

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A_b = Area of block
A_h = Area of head

VI. VISUAL BASIC PROGRAM

The computer program is developed in visual basic language to compute coefficient of heat transfer of the piston crown on the gas side for the single-cylinder four-stroke petrol engine for the complete working cycle. Output of the software can be shown numerically for one working cycle at every crank angle.

VII. RESULTS AND DISCUSSION

A dry gas analysis of an engine exhaust gives the four contents i.e. CO, CO_2, HC, O_2 in volumetric percentages by smoke-meter-testing at the maximum power condition as given in the Table V

The table VI shows the result which is computed from the visual basic computer program developed for coefficient of heat transfer of gas side piston crown for the complete working cycle at every the crank interval. It is calculated by using Woschni-heat-transfer-coefficient

Result of heat transfer by piston of engine theoretically method, is given in Table VII

TABLE V: RESULTS OF SMOOKMETER TEST

CO	CO_2	HC	O_2
6.47	9.4	0.0811	1.16

TABLE VI: COEFFICIENT OF HEAT TRANSFER OF GAS TOP OF PISTON AT MAX. POWER (7000 RPM)

Suction		Compression		Power		Exhaust	
θ	*hg*	θ	*Hg*	θ	*hg*	θ	*hg*
Deg	W/m^2-K	Deg	W/m^2-K	Deg	W/m^2-K	Deg	W/m^2-K
0	94.18	360	1138	360	1138	720	93.61
30	91.39	330	588.0	390	966.4	690	827.9
60	87.37	300	306.6	420	525.09	660	80.47
90	94.01	270	197.0	450	341.87	630	99.79
120	107.1	240	152.5	480	258.13	600	127.3
150	116.1	210	123.3	510	216.40	570	165.9
180	123.3	180	123.3	540	193.03	540	193.0

TABLE VII: HEAT BALANCE SHEET CALCULATED BY THEORETICALLY METHOD

Parameter	Heat Transfer (KW)
Heat equivalent to effective work (*Qa*)	6.25
Heat carried away by exhaust gases (Q*Exhaust*)	12.57
Heat transfer to oil (*Qoil*)	2.064
Heat Loss due to unburned fuel (*Quf*)	0.9535
Unaccounted Heat loss (*Qu*)	4.0
Heat Transferred to ambient air ($Qair_r$)	4.25

By using the numerical method least square curve fitting technique the time average equivalent $\overline{h}_g$ = 300.21 W/m^2-K and $\overline{T}_g$ = 1393.26 K are calculated for four-stroke cycle engine. Steady state thermal analysis of piston is carried out at Max. Power condition it gives 550 K temperature of crown piston.

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VIII. CONCLUSION

The heat transfer through the piston is validated by comparing the results by numerical analysis i.e. 0.537 KW and by theoretically analysis (heat balance sheet) i.e. 0.5152 KW. The error between theoretical and numerical heat transfer is found to be 4.05 %. The deviation is observed because some constants are assumed in theoretically analysis. But deviation is so less that it is applicable for our piston model for further analysis. Methodology of this paper can be used for all types of pistons of internal combustion engine for calculating coefficient-of- heat-transfer on the piston crown.

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Appendix

(A) Short block specification:

Title	Detail		
		mm	Inch
Bore Diameter		52.4	2.062
Stroke		57.8	2.275
No. of Cylinders		1	
Piston Rings	No. of Rings +Tension; Exp Required	3 Low Tension	
Connecting rod Length	Center Distance	92.21	3.63
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, Electrical, 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	27	0.162		22.7	0.89	
Avg. Port Diameter	22.67	0.892		20.8	0.81	
Port Length	71.5	2.814		65.8	2.59	
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	14.7	0.078	0.1	14.76
	0.13	0.5	21.9	0.118	0.5	21.78
	181	0.2	25.2	0.157	0.2	27.02
	0.22	0.25	25.8	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		9.5				

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Chamber Design	Hemi, Pent roof, Dual plug etc	Flat			
Material / Coating	Aluminium, Cl, Coated Cl, Coated Aluminum	Aluminium			
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	23.8	0.93	Runner Length	111	4.37
Runner Length	116.7	4.59	Inside Diameter at head	22.8	0.976
Runner Length Coeff.	2.3		Total length	68.9	2.71
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	1.1969 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	12.126 Inch
			Collector Dia	58.45 mm	2.598 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