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Design of Variable Valve Timing and Electronic Control for Honda R18A

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Abstract: This paper explores the Variable Valve Timing and Lift Electronic Control (VTEC) system developed by Honda, focusing on its mechanisms, design methodology, and impact on engine performance. VTEC optimizes engine efficiency and power by switching between cam profiles suited for low-RPM stability and high-RPM power output, controlled by the engine's computer based on operational conditions. We examine the intricate workings of the VTEC system, detailing its hydraulic actuation and the resulting enhancements in torque and horsepower. Our study centers on the Honda R18A engine, targeting increased maximum RPM and power output. Using theoretical analysis validated through Ricardo Wave software and MATLAB simulations, we determine optimal valve lift and timing to achieve the desired performance. Additionally, the research addresses the geometric parameters influencing airflow through the valves, providing a comprehensive understanding of the VTEC system's contribution to engine efficiency and driving experience. This investigation not only underscores the technological advancements in variable valve timing but also presents practical insights for automotive engineering applications.
Index Terms: Variable Valve Timing, VTEC, Honda R18A engine, cam profiles, hydraulic actuation, engine performance, valve lift, Ricardo Wave software, MATLAB simulations, torque, horsepower, fuel efficiency.

I. INTRODUCTION

VTEC is a type of variable valve-timing system developed and used by Honda. It stands for Variable Valve Timing & Lift Electronic Control. Like most other variable-valve timing systems, VTEC varies oil pressure to shift between different cam profiles. At higher engine speeds, the cam profile allows greater valve lift, which allows more air into the cylinder. This helps generate more horsepower. Since its introduction in the late 1980s, VTEC has been used in many of Honda's best performance cars including the NSX, Integra Type R, S2000, and Civic Type R.

II. HOW IT WORKS

The original VTEC system replaced a single cam lobe and rocker with a locking multi-part rocker arm and two cam profiles: one optimized for low-RPM stability and fuel efficiency and the other designed to maximize higher-RPM power output. The VTEC system essentially combines low-RPM fuel efficiency and stability with high-RPM performance. And the transition occurs seamlessly, allowing for smooth performance across the entire powerband. The switching operation between the two cam lobes is controlled by the engine computer. Based on speed, load, and engine RPM, the computer switches between the efficient cam and the high-performance cam. A solenoid is actuated that engages the rocker arms on the high-performance cam. At that point the valves open and close according to the high-lift profile, opening the valves further and for a longer time. This allows more air and fuel to enter and burn, creating stronger torque and horsepower. Honda cars equipped with VTEC technology tend to be more efficient across a wider rpm range than many comparable vehicles, and they're a lot of fun to drive in the right conditions, but most motorists won't notice their VTEC kicking in. It's active when the engine is operating relatively high in the rev range, and you rarely get there in normal driving conditions, especially if your car has an automatic transmission. But, if you're the shift-your-own-gears type and you like twisty roads, VTEC makes a noticeable difference. [1][2]

III. MECHANISM

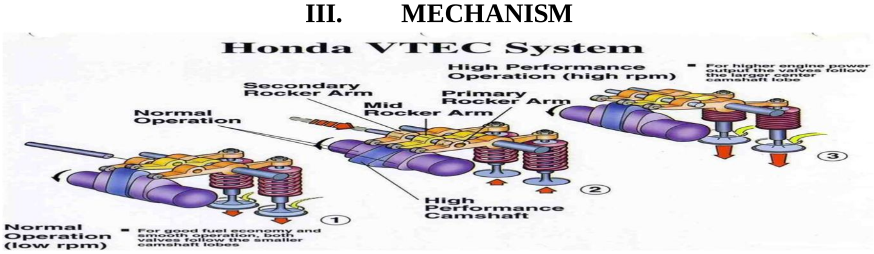

Figure 1

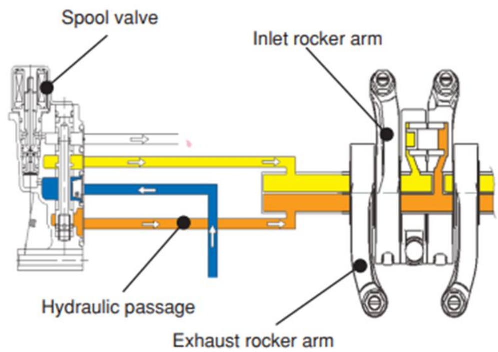

Figure 2

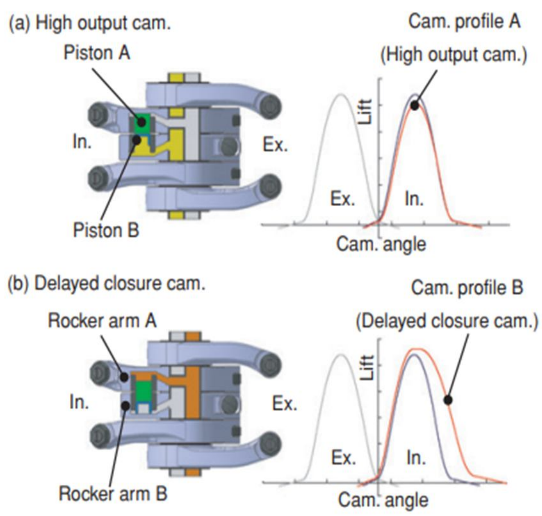

Figure 3

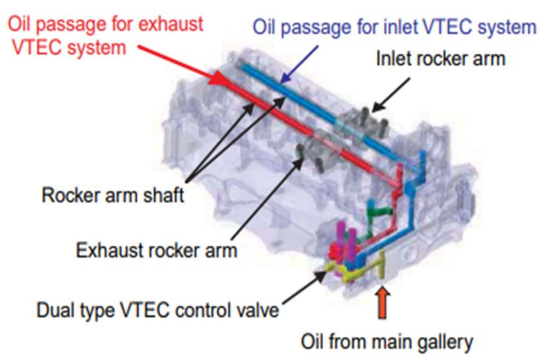

Figure 4

IV. LITERATURE REVIEW

The figures 1, 2, 3 & 4 show the working mechanism of the VTEC system. The three cam lobes present there will have three separate rocker arms mounted on an independent rocker arm shaft. The three cams consist of a small cam lobe for lower rpm and another for higher. The third cam lobe is constantly operating on mean conditions. A hydraulic fluid is passed through the rocker shaft and arm through which a hydraulic cylinder actuator is actuated. The actuator cylinder works like a pin and couples the first two rocker arms together. The actuator is retracted back by releasing the fluid pressure. [1][2].

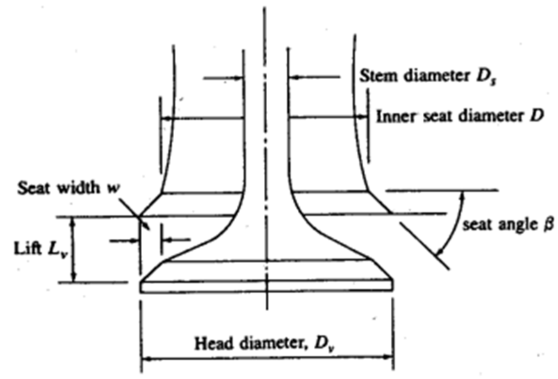

Figure 5

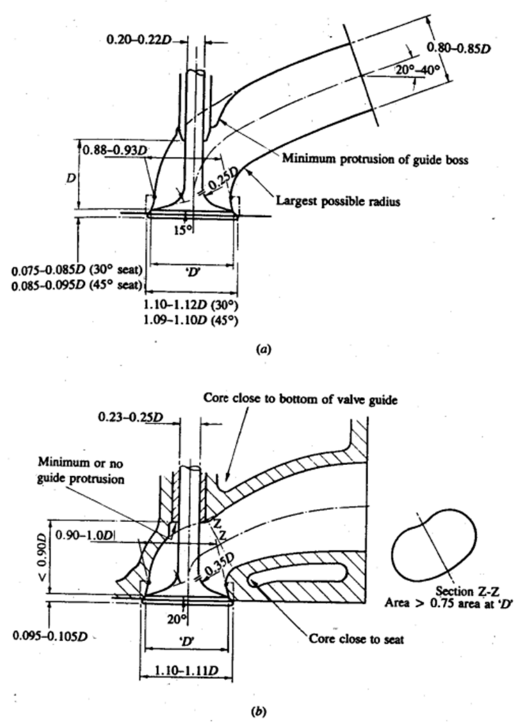

Figure 6

Figure 5 shows the main geometric parameters of a poppet valve head and seat. Figure 6 shows the proportions of typical inlet and exhaust valves and ports, relative to the valve inner seat diameter D. For low valve lifts, the minimum flow area corresponds to a frustum of a right circular cone where the conical face between the valve and the seat, which is perpendicular to the seat, defines the flow area.

For this stage: $\frac{w}{sin\beta cos\beta} > Lv > 0$

And the minimum area is :

$$Am = \pi L_v cos\beta(Dv - 2w + \frac{L_v}{2} sin2\beta) \quad \text{………..(1)}$$

where β is the valve seat angle, Lv is the valve lift, Dv is the valve head diameter (the outer diameter of the seat) and w is the seat width (difference between the inner and outer seat radii).

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For the second stage, the minimum area is still the slant surface of a frustum of a right circular cone, but this surface is no longer perpendicular to the valve seat. The base angle of the cone increases from $(90-\beta)^0$ toward that of a cylinder, 90'.

For this stage:

$$\left[\left(\frac{D_p^2-D_s^2}{4D_m}\right)^2-w^2\right]^{\frac{1}{2}}+wtan\beta \geq L_v > \frac{w}{sin\beta cos\beta}$$

And

$A_m = \pi D_m[(L_v - wtan\beta)^2 + w^2]^{\frac{1}{2}}$…………(2)

where D_p is the port diameter, D_s is the valve stem diameter and D_m is the mean seat diameter $(D_v - w)$.
Finally, when the valve lift is sufficiently large, the minimum flow area is no longer between the valve head and seat; it is the port flow area minus the sectional area of the valve stem. Thus, for

$$L_v > \left[\left(\frac{D_p^2-D_s^2}{4D_m}\right)^2-w^2\right]^{\frac{1}{2}}+wtan\beta$$

then $A_m = \frac{\pi}{4}(D_p^2 - D_s^2)$…………(3)

Intake and exhaust valve open areas corresponding to a typical valve-lift profile are plotted versus camshaft angle in figure 4c. These three different flow regimes are indicated. The maximum valve lift is normally about 12 percent of the cylinder bore.

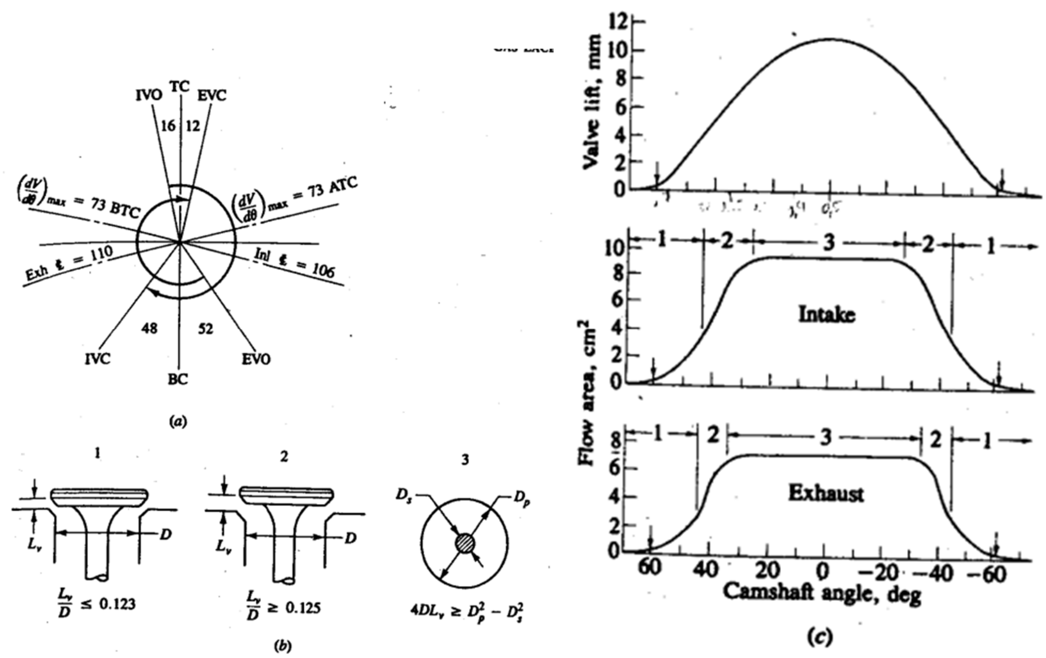

Figure 7

Note from the timing diagram (figure 4a) that the points of maximum valve lift and maximum piston velocity do not coincide.
The effect of valve geometry and timing on air flow can be illustrated conceptually by dividing the rate of change of cylinder volume by the instantaneous minimum valve flow area to obtain a pseudo flow velocity for each valve:

$v_{ps} = \frac{1}{A_m}\frac{dV}{d\theta} = \frac{\pi B^2}{4A_m}\frac{ds}{d\theta}$…………(4)

where V is the cylinder volume, B is the cylinder bore, s is the distance between the wrist pin and crank axis and Am is the valve area given by eqs (1), (2) or (3).

V. FLOW RATE & DISCHARGE COEFFICIENTS

The mass flow rate through a poppet valve is usually described by the equation for compressible flow through a flow restriction. This equation is derived from a one-dimensional isentropic flow analysis and real gas flow effects are included by means of an experimentally determined discharge coefficient Cd.

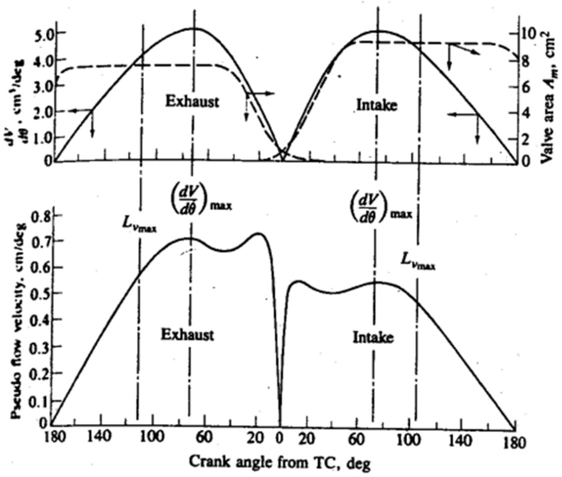

Figure 8

The air flow rate is related to the upstream stagnation pressure po and stagnation temperature To, static pressure just downstream of the flow restriction (assumed equal to the pressure at the restriction, pt) and a reference area Ar characteristic of the valve design:

$$m_a = \frac{C_D A_R p_O}{(RT_O)^{\frac{1}{2}}}\left(\frac{p_t}{p_o}\right)^{\frac{1}{2}}\left\{\frac{2\gamma}{\gamma-1}\left[1-\left(\frac{p_t}{p_o}\right)^{\frac{(\gamma-1)}{\gamma}}\right]\right\}^{\frac{1}{2}} \text{.......(5)}$$

When the flow is choked, i.e., $p_t/p_o \leq [2/(\gamma+1)]^{\frac{\gamma}{(\gamma-1)}}$, the appropriate equation is

$$m_a = \frac{C_D A_R p_o}{(RT_o)^{\frac{1}{2}}}\gamma^{\frac{1}{2}}\left(\frac{2}{\gamma+1}\right)^{\frac{(\gamma+1)}{2(\gamma-1)}} \text{.............(6)}$$

For flow into the cylinder through an intake valve, po is the intake system pressure pi and pt is the cylinder pressure. For flow out of the cylinder through an exhaust valve, po is the cylinder pressure and pt is the exhaust pressure.

The value of Cd and the choice of reference area are linked together: their product, CdAr, is the effective flow area of the valve assembly Ae. Several different reference areas have been used. These include the valve head area $\pi D_v^2/4$, the port area at the valve seat $\pi D_p^2/4$, the geometric minimum flow area [eqs (1), (2) and (3)] , and the curtain area $\pi D_v L_v$, where Lv is the valve lift. The choice is arbitrary, though some of these choices allow easier interpretation than others. As has been shown above, the geometric minimum flow area is a complex function of valve and valve seat dimensions. The most convenient reference area in practice is the so-called valve curtain area:

$$A_C = \pi D_v L_v \text{ ....(7)}$$

since it varies linearly with valve lift and is simple to determine. [4]

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VI. INLET VALVES

Figure 6 shows the results of steady flow tests on a typical inlet valve configuration with a sharp-cornered valve seat. The discharge coefficient based on valve curtain area is a discontinuous function of the valve-lift/diameter ratio. Typical maximum values of L_v/D_v are 0.25.

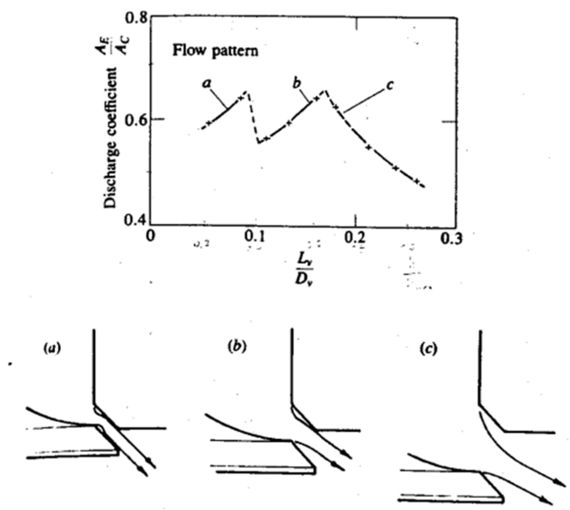

Figure 9

At high engine speeds, unless the inlet valve is of sufficient size, the inlet flow during part of the induction process can become choked (i.e., reached sonic velocity at the minimum flow area). Choking substantially reduces volumetric efficiency. Various definitions of inlet Mach number have been used to identify the onset of choking. Taylor and coworkers correlated volumetric efficiencies measured on a range of engine and inlet valve designs with an inlet Mach Index Z formed from an average gas velocity through the inlet valve:

$$Z = \frac{A_p \underline{S_p}}{C_i A_i a} \quad \text{………….(8)}$$

where Ai is the nominal inlet valve area $(\pi D_v^2/4)$, Ci is a mean valve discharge coefficient based on the area Ai and a is the sound speed. From the method used to determine Ci, it is apparent that CiAi is the average effective open area of the valve (it is the average value of $C_d \pi D_v L_v$). Z corresponds closely, therefore, to the mean Mach number in the inlet valve throat. Taylor's correlations show that η_v decreases rapidly for Z≥0.5. An alternative equivalent approach to this problem has been developed, based on the average flow velocity through the valve during the period the valve is open. A mean inlet Mach number was defined:

$$\underline{M_i} = \frac{\underline{v_i}}{a} \quad \text{………….(9)}$$

where $\underline{v_i}$ is the mean inlet flow velocity during the valve open period. $\underline{M_i}$ is related to Z via

$$\underline{M_i} = \frac{Z(\eta_v/100)180}{\theta_{IVC}-\theta_{IVO}} \quad \text{………….(10)}$$

This mean inlet Mach number correlates volumetric efficiency characteristics better than the Mach index. For a series of modern small four-cylinder engines, when $\underline{M_i}$ approaches 0.5 the volumetric efficiency decreases rapidly. This is due to the flow becoming choked during part of the intake process. This relationship can be used to size the inlet valve for the desired volumetric efficiency at maximum engine speed. Also, if the inlet valve is closed too early, volumetric efficiency will decrease gradually with increasing $\underline{M_i}$, for $\underline{M_i}$<0.5, even if the valve open area is sufficiently large.

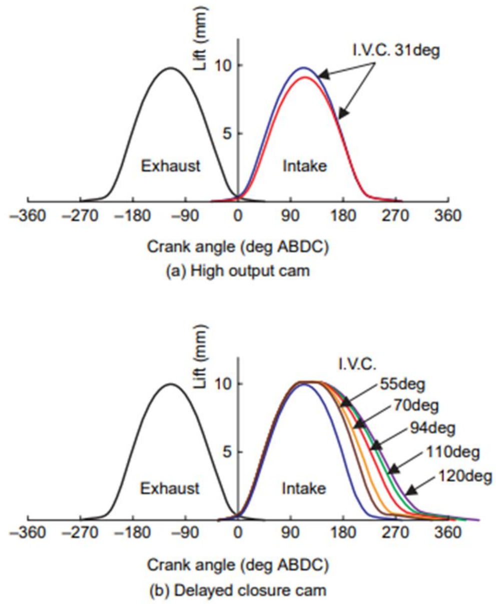

Figure 10

The above diagram shows a reference example of the original valve timing diagram and the valve timing diagrams for two cases, being, change in intake valve lift and opening timing; and change in closure timing of the intake valve. [4]

VII. TARGETED PERFORMANCE PARAMETER

For designing purposes, the target values for properties like Maximum Power, Maximum RPM, Fuel Economy, etc. The factors into consideration are dependent on the type and targeted usage of the engine. For this case a Honda type R, R18a engine is considered. The engine in the study is a performance type engine focused on producing higher torque at lower rpm ranges and also sufficient power at maximum rpm while also increasing the maximum rpm limit. Hence power and rpm are considered targeted and the values are sourced from the engine data:

Maximum Power: 103kW @6300 RPM;

Maximum Torque: 174Nm @4300 RPM;

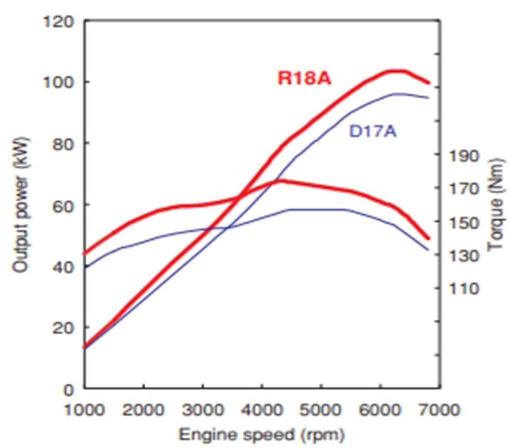

Figure 11

The same can be seen from the graph. From the above figure, it can be seen that the power starts to drop after the 4300 rpm point. Hence the VTEC needs to be activated around that point to shift the system towards maximum RPM. From this the shifting point for VTEC was decided as 4500 rpm. From that point onwards, the engine should ideally follow the second curve in order to produce higher power at higher rpm.

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VIII. TYPES OF CAM PROFILE & CONSTRUCTION

Modern automobile engines employ the following types of cams: convex, tangential, concave and harmonic. The convex profile cam may be used for lifting a flat, convex or roller follower. The tangential profile cam is mainly used for roller followers.

The cam profile is constructed starting with a base circle. Its radius ro is chosen to meet the requirement of providing enough rigidity of the valve gear.the convex profile being formed by two arcs having radii r_1 and r_2 and a tangential cam whose profile is formed by means of two straight lines tangential to the base circle of r_0, at points A and A' and an arc having radius r_2.

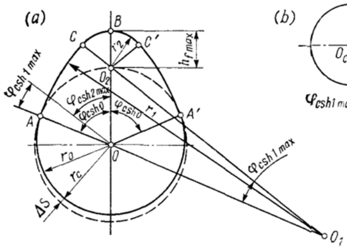

Figure 12

The value of camshaft angle φ_{csho} is determined according to selected valve timing. For four-stroke engines

$$\varphi_{csho} = (\varphi_{ad} + 180^0 + \varphi_{re})/4$$

With a convex cam profile(Figure 12 (a))

$$r_1 = \frac{r_0^2 + a^2 - r_2^2 - 2r_0 acos\varphi_{csho}}{2(r_0 - r_2 - acos\varphi_{csho})}$$

$$r_2 = \frac{r_0\ b - 0.5h_f^2\ max - (r_1 - r_0)(r_0 + h_f^2 max)\ cos\varphi_{csho}}{b - (r_1 - r_0)\ cos\varphi_{csho}}$$

For a tangential cam(Figure 12 (b)) with a roller follower:

$$h_{f1} = (r_0 + r\ \ )(1 - cos\varphi_{csh1})/cos\varphi_{csh1}$$

$$h_{f2} = a(cos\varphi_{csh2} + \frac{1}{a_1}\sqrt{1 - a_1^2 sin^2\ \ \varphi_{csh2}}\ \ ) - (r_0 + r\ \ )$$

$w_{f1}=(r_0 + r\ \ )\omega_c \sin\varphi_{csh1}/cos^2\varphi_{csh1}$

$w_{f2} = \omega_c a[\sin\varphi_{csh2} + (a_1 sin2\varphi_{csh2})/(2\sqrt{1 - a_1^2 sin^2\ \ \varphi_{csh2}}\ \ )]$

$j_{f1} =(r_0 + r\ \ )\omega_c^2(1+sin^2\ \ \varphi_{csh1})/(cos^2\varphi_{csh1})$

$j_{f2}=-\omega_c^2 a[cos\varphi_{csh2}+(a_1\cos2\varphi_{csh2}+a_1^3 sin^4\ \ \varphi_{csh2})/(1 - a_1^2 sin^2\ \ \varphi_{csh1})^{3/2}]$

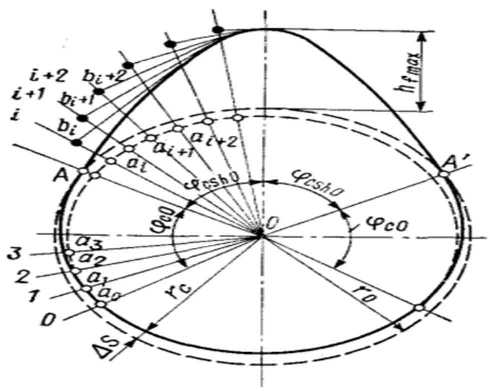

Figure 13

And Harmonic cams as represented in figure 13.

IX. DESIGN METHODOLOGY FOR DESIGNING VTEC

Vtec stands for variable valve timing and lift electronic control . Hence involves designing of cam shaft which include determining cam lift , angle of advance , angle of retard , (i.e. deciding the valve timing diagram).

Design of vtec can be done by targeting performance parameters of engine such as Power , RPM or fuel economy . That is the whole purpose of using vtec system . For example , consider Honda's R18A engine , this engine reaches maximum RPM of 4500 without the vtec system (note that the values are at ideal condition . the condition for which the vehicle is designed .) To get a performance, vtec is being used . after using vtec engine can reach 6300 rpm.

DATA :

Now we know the targeted RPM , N = 6300;

The engine is having Power at that RPM , Power = 103 kw;

Engine has some break thermal efficiency , $\eta bth = 23\,\%$

(here an assumption is made that ηbth does not depend on operating condition . This assumption has to be made because we can not predict effeciency at each RPM . We have to take break thermal efficiency of a non vtec engine (@ 4500 RPM) as we can not proceed without it)

RPM N =

Air fuel ratio , Air/F = 13:1

Molar gas constant R = 8.314 KJ/K

Manifold pressure Po = 1 atm or 101325 pa

Manifold Temperature , T = 27^{0} or 300 k

All the constant terms of ma formula can be replaced by k = 0.6847 ($\gamma = 1.4\ for\ air$)

Valve diameter , Dv = 0.032 m

Calorific value of petrol , CV = 45000

Coefficient of discharge of valve , Cd = 0.45 to 0.65

Type	R18A
Cylinder configuration	In-line 4-cylinder
Bore × Stroke (mm)	81 × 87.3
Displacement (cm³)	1799
Compression ratio	10.5:1
Valve train	SOHC i-VTEC Inlet delayed closure
Number of valves	4 per cylinder
Valve diameter (mm) In./Ex.	32/26
Cylinder offset (mm)	12
Intake manifold	Variable intake system
Gasoline	Regular (RON91)
Max. power (kW/rpm)	103/6300
Max. torque (Nm/rpm)	174/4300

Table 1

AP is piston area

B is bore dia

L is stroke

Sp is mean speed of piston =2LN

Cd is discharge coefficient

Lv is valve lift

Z is the mach index

Mfcycle is the mass flown inside the cyclinder in a suction stroke

Mf is the mass flow rate (Kg/s)

Ar is the curtain area = pi DV LV

Ar1 and Ar2 are the areas for two valves
Wc is speed of crank shaft = 2PiN/60

$Ap = \frac{\Pi}{4} B^2$

$$Z = \frac{Ap\ Sp}{ArCia}$$
$$Z = \frac{Ap\ Sp}{\Pi DvLv}$$
$$Lv = \frac{Ap\ SP}{\Pi DvCda}$$
$$mfcycle = mf \times time\ of\ suction\ stroke$$
$$mf = \frac{mfcycle\ Wc}{\theta}$$
$$mf = mf1 + mf2$$
$$mf = \frac{Ar\ Cd\ Po\ K\ f}{\sqrt{RT}\ Air}$$

Here $K = \gamma^{\frac{1}{2}} \left(\frac{2}{\gamma+1}\right)^{\frac{(\gamma+1)}{2(\gamma-1)}}$

$$m_f = \frac{Ar_1 C_d P_o KF}{(RT_o)^{\frac{1}{2}} Air} + \frac{Ar_2 C_d P_o KF}{(RT_o)^{\frac{1}{2}} Air}$$
$$m_f = \frac{C_d P_o KF}{(RT_o)^{\frac{1}{2}} Air} (Ar_1 + Ar_2)$$
$$m_f = \frac{m_{fcycle}(w_c)}{\theta}$$
$$\theta = \frac{m_{fcycle}(w_c)(RT_o)}{C_d P_o KF(Ar_1 + Ar_2)}$$
$$m_{fcycle} = \frac{w_{fcycle}}{\eta C_v}$$
$$P_{cycle} = \frac{w_{cycle}}{time\ of\ cycle}$$

$$\theta = \frac{Pcycle \times time\ of\ cycle \times Wc \sqrt{RT}\ air}{Cd\ Po\ k\ F\ (Ar1 + Ar2)}$$

We have used the software Ricardo wave and matlab code for validation of the calculated Valve Lift and Opening duration of the valve. Validation was done for checking targeted performance parameters.

X. MATLAB PROGRAM

```
clear;
clc;
%(Lv from mach index formula and Theta from heywood formula and then Power )
N = 6300;
B = 0.081;
S = 0.0873;
Ap = (pi/4)*B^2;
Ci = 0.6;
a = 343;
Dv = 0.032;
```

```
Sp = 2*N*S/60;
Z1 = 0.5; %mach index
Lv1 =(Ap*Sp)/(Ci*a*pi*Dv*Z1);
Lv2 = 0.008535;
Pcycle = (103*10^3)/4;
R = 8.314;
T = 300;
Air = 13;
Cd = 0.6;
CV = 45000;
eff = 0.23;
Po = 101325;
K = 0.6847;
K1 = 0.0148;
F = 1;
wc = 2*pi*N/60;
timeofcycle = 4*pi/wc;
Ar1 = pi*Lv1*Dv;
Ar2 = pi*Lv2*Dv;
ThetaR=(Pcycle*timeofcycle*wc*((R*T)^0.5)K1*Air )/(eff*CV*Cd*Po*K*F(Ar1+Ar2));
Theta = ThetaR*180/pi;

wc = 2*pi*N/60;
time = ThetaR/wc;  %(time of suction stroke )
timeofcycle = 4*pi/wc;
cycleperhr = 18900;

Ma1 = (Cd*Ar1*Po*K)/(Air*(R*T)^0.5);
Mf1 = Ma1/13;
Ma1cycle = Ma1*time;
Mf1cycle = Ma1cycle/13;
L1pers = Mf1/0.769;
L1perc = Mf1cycle/0.769;
L1perhr = L1perc*cycleperhr; %(eka tasat jitkya cycle hotat tyacha)

Ma2 = (Cd*Ar2*Po*K)/(Air*(R*T)^0.5);
Mf2 = Ma2/13;
Ma2cycle = Ma2*time;
Mf2cycle = Ma2cycle/13;
L2pers = Mf2/0.769;
L2perc = Mf2cycle/0.769;
L2perhr = L2perc*cycleperhr; %(eka tasat jitkya cycle hotat tyacha)

Wpers = (Mf1+Mf2)*eff*CV;     %(work per sec means power)
Wperc = (Mf1cycle+Mf2cycle)*eff*CV; %(work per 1 cycle)

Pcalc = Wpers;

P1cycle = Wperc/timeofcycle; %(power of 1 cycle )
```

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Name	Value
a	343
Air	13
Ap	0.0052
Ar1	9.1807e-04
Ar2	8.5803e-04
B	0.0810
Cd	0.6000
Ci	0.6000
CV	45000
cycleperhr	18900
Dv	0.0320
eff	0.2300
F	1
K	0.6847
K1	0.0148
L1perc	3.6264e-05
L1perhr	0.6854
L1pers	0.0059
L2perc	3.3893e-05
L2perhr	0.6406
L2pers	0.0055
Lv1	0.0091
Lv2	0.0085
Ma1	0.0589
Ma1cycle	3.6253e-04
Ma2	0.0550
Ma2cycle	3.3882e-04
Mf1	0.0045
Mf1cycle	2.7887e-05
Mf2	0.0042
Mf2cycle	2.6063e-05
N	6300
P1cycle	29.3154

Name	Value
Pcalc	90.6620
Pcycle	25750
Po	101325
R	8.3140
S	0.0873
Sp	18.3330
T	300
Theta	232.8107
ThetaR	4.0633
time	0.0062
timeofcycle	0.0190
wc	659.7345
Wperc	0.5584
Wpers	90.6620
Z1	0.5000

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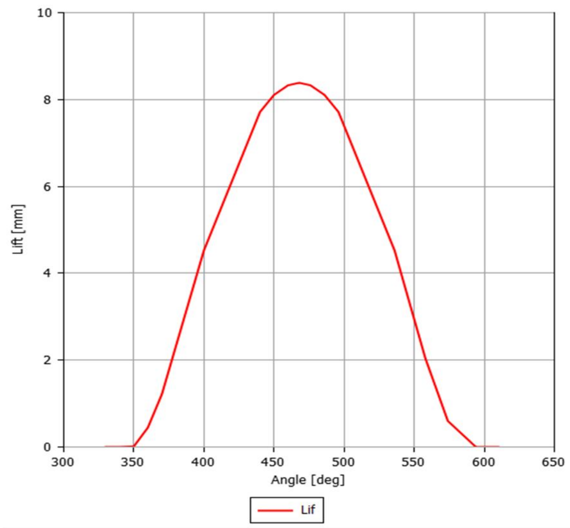
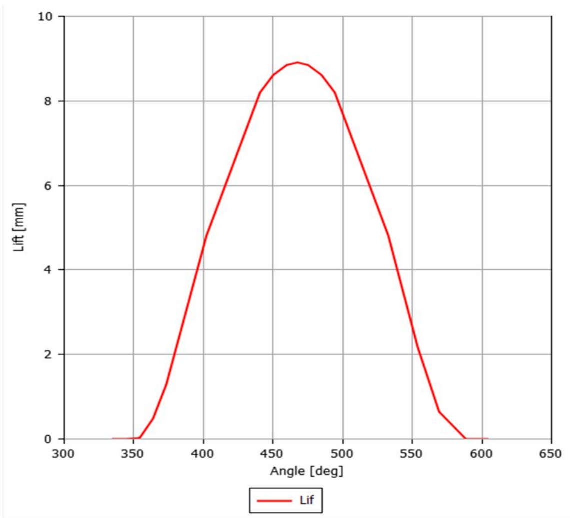
XI. RICARDO WAVE SOFTWARE



Medium Cam Lobe Valve Lift



Vtec Caam Lobe Lift

	Angle [deg]	Lift [mm]
1	0	0
2	10	0.02
3	20	0.17
4	30	0.62
5	40	1.42
6	70	4.87
7	110	8.19
8	120	8.6
9	130	8.83
10	138	8.89
11	146	8.83
12	156	8.6
13	166	8.19
14	206	4.87
15	228	2.28
16	244	0.78
17	264	0.1
18	280	0

Valve Lift Table

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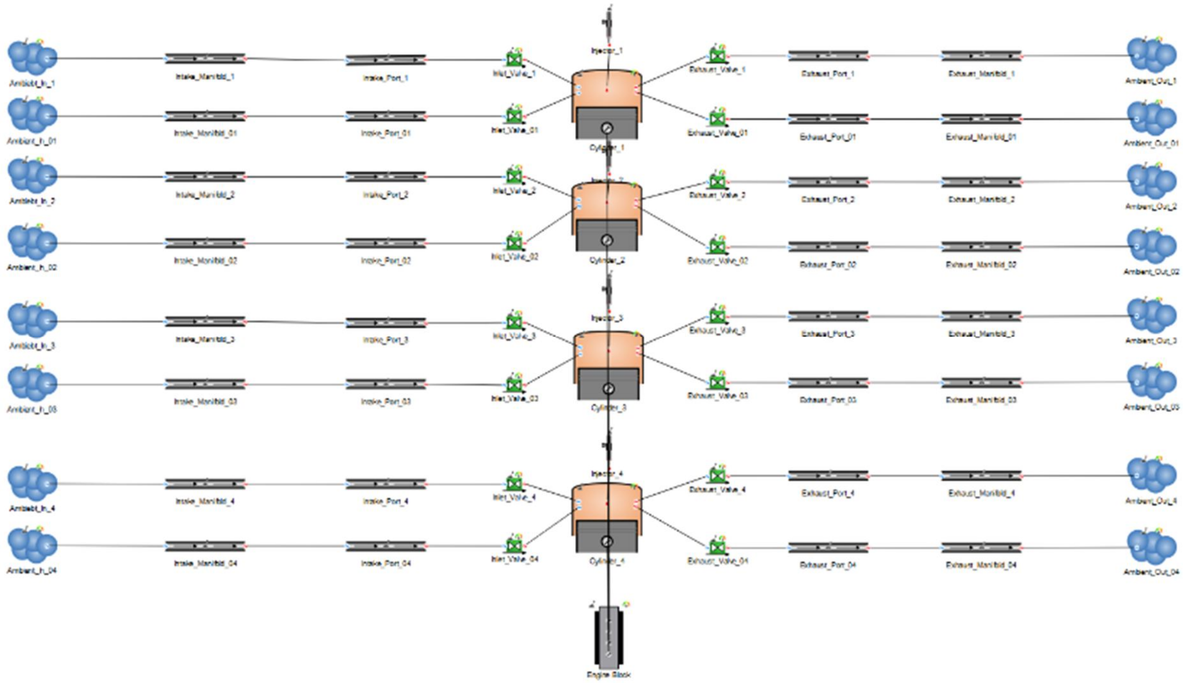
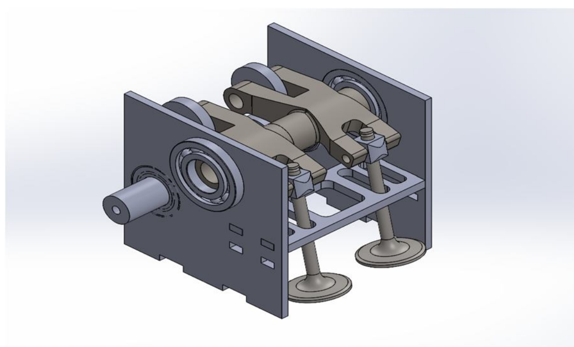
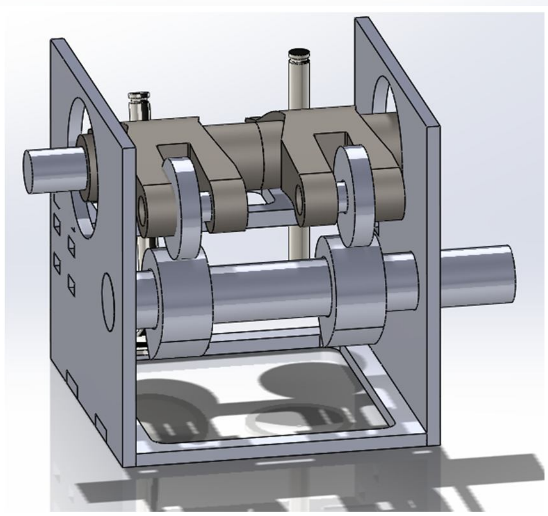


EngineSetup

XII. CAD MODEL WITH PROPOSED MECHANISM





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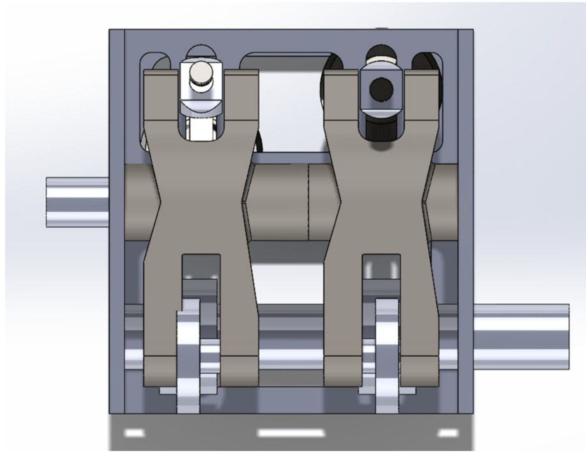

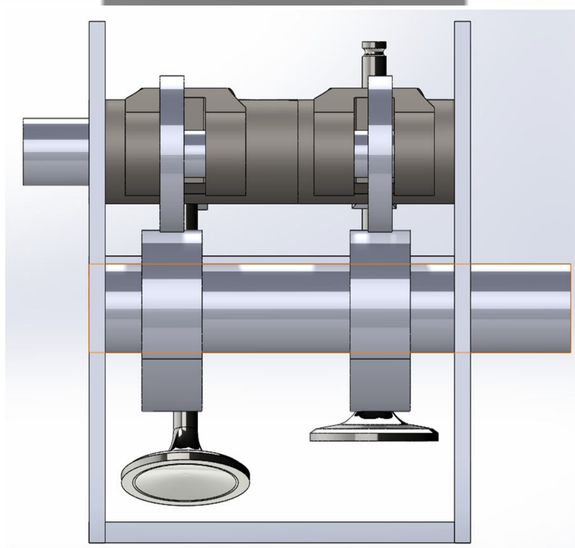

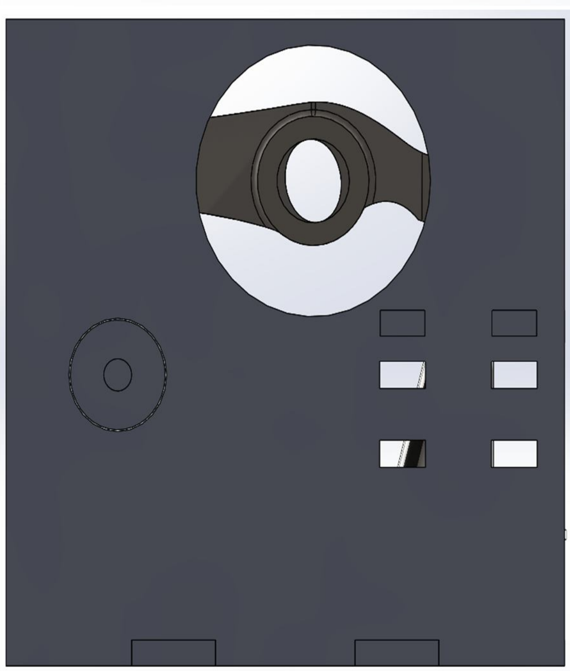

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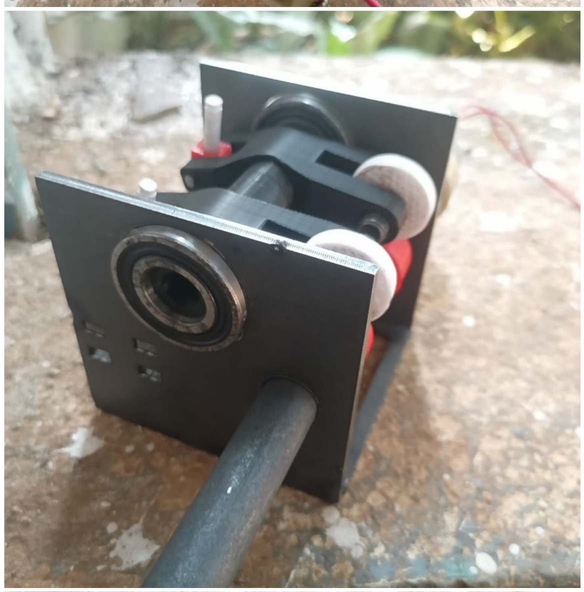
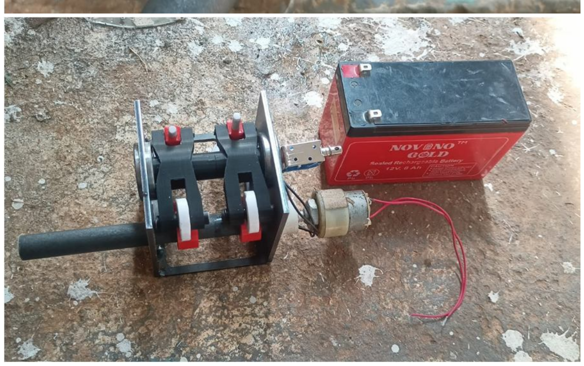
XIII. ACTUAL PROTOTYPE





XIV. ACTUAL DATA

Cam Lobe:

Vtec lobe: Height = 36.1mm
Face Width = 9.7mm
Base Circle Diameter = 30.06mm

Lift = 9.06mm

Medium Lobe: Height = 35.75mm
Face Width = 8.8mm
Base circle Diameter = 30.06mm
Lift = 8.53mm

Low Lobe: Height = 35.3mm
Face Width = 9.56mm
Base Circle Diameter = 30.06mm

Figure 14

Figure 15

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