Use of Hydrogen in Fiat Lancia Petrol engine, Combustion Process and Determination of Optimum Valve Timing

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Abstract: *To our path towards green economy, Hydrogen is often regarded to have a potential growth in the coming future. However, the high cost of operation of fuel cell has often been a setback. If we could make use of hydrogen gas as a fuel directly, the scope of development broadens. Owing to these aspects, this work primarily focuses on the simulation technique of an Internal Combustion Spark Ignition Engine powered by Hydrogen gas. The simulations of various stages have been carried out using the discrete approach, thereby investigating the pressures and temperatures at various instants in the cycle. For the relative performance discussion we have simulated the different cycles as ideal cycle, air fuel cycle and actual cycle. The resultant cyclic graph indicates various discrepancies between ideal, air fuel and actual cycle. This analysis serves as a tool for a better understanding of the variables involved and helps in optimizing engine design and fixing of various parameters, including the determination of valve timings. Besides this, backfire, is the commonly faced problem with the hydrogen engines. To reduce this effect, a fuel injectoris used for adding the gaseous fuel to the combustion chamber.*

Keyword: *Actual cycle, air fuel cycle, backfire, discrete approach, fuel injector, fuel cell, hydrogen, ideal cycle, spark plug, valve timings ,Fiat Lancia,inlet valve, exhaust valve, optimum valve timing .*

I.   Introduction

An intense research effort is on-going in the field of engines to develop new engines with improved efficiency and decreased harmful emission. Further, to overcome the high dependency of automobiles on the limited crude oil, huge focus is laid on the development of engines that are powered by alternate fuels, which can even reduce the rate of pollution in the world that is currently being done by the hydrocarbons. Amongst all, hydrogen, a fuel that powers the sun, is looking forward to run our vehicles too. The merits of using it as a fuel are immense, few of which includes it production from renewable energy sources, a better efficiency over petrol and harmless exhaust gases, mostly water vapour.

To study different engines, computer based thermodynamic models have been developed which can predict the performance, avoiding the difficulties associated with the detailed experimental analysis.Simulation of ideal cycle involves much less complexities since the initial conditions are known. But it is quite clear that an ideal cycle cannot predict the exact conditions that an engine will face during its working period. An actual model incorporates equations,which need to be solved using the method of iterations. Two approaches can be followed for simulation of an IC engine- (i) The Differential Approach (ii) The Discrete Method. The first method involves solving of partial differential equations based on energy, mass and momentum conservation (1). The later model considers the calculations at each degree of crank rotation (or smaller) considering a few assumptions at the initial stage but by iterating, ultimately reaching the exactconditions inside the combustion chamber. This later approach is followed in this work and calculations are made which are discussed later in the section of tables. The graphical charts plotted later, provides a comparison of the actual cycle with the ideal and air fuel cycle.

Throughout this present work, a single cylinder, four stroke spark ignition engine, with a hydrogen fuel injector, is used to illustrate all of the thermodynamic points to be made. The engine has a bore (B) of 73 mm diameter, stroke length(L) of 70 mm, flywheel of radius (r) 35 mm and connecting rod of length (l) 170 mm. Further the values presented here are for engine running at 1000 rpm. The compression ratio taken is 7.96 with a clearance volume of 4.2023 x 10^{-5} m^3. The variations of specific heats, enthalpy and equilibrium constants of different gases with temperature were obtained based on relationships established from JANAF tables.

II.   Mathematical Models

The simulations of any engine is done and compared with the results of ideal cycle to get a clear idea of the cycle.

2.1 The Ideal Cycle & Air-fuel cycle

The ideal cycle serves as a standard for comparison of operations of various cycles. In ideal cycle we have taken air as a working medium and a few standard assumptions for obtaining maximum achievable efficiency.

Furthermore, by considering air-fuel cycle, we move a step closer towards actual analysis, since most variations as a function of temperature are taken into consideration into this calculation, expect for the friction and convective heat losses. Also the process of suction and exhaust is not taken into account in these two cycles. The unknown value of temperature (T) at different crank angles is calculated using below equation obtained by integrating the equation for entropy balance utilized only for air fuel analysis.

$$-R\ln\frac{V_1}{V_0} = a\ln\frac{T}{t_0} + b(T - t_0) \qquad ....(1)$$

$$where\ C_v = a + bT$$

2.2 The Discrete Approach for Actual Cycle

In the present simulations, the governing parameters of the engine- pressure, temperature, volume and mass are calculated at each degree of crank rotation. In an actual cycle, the remains of exhaust gases have to be considered for simulation of suction process. However, the quantity of remaining exhaust gases and its temperature and pressure can be known only after the whole cycle is simulated. Hence here initially these parameters are assumed to simulate the processes of suction, compression, combustion, expansion and exhaust. The parameters obtained after exhaust are taken to iterate the cycle again. The detailed method of simulation of these processes is discussed in the following topics.

Assumptions:

In this work, the gas mixture has been considered as an ideal gas. Also, the losses in inlet and exhaust manifold are not considered. In combustion chamber the gases have been considered to have uniform temperature. Pre-ignition has not been considered and the combustion is considered to be occurring at constant volume. The wall temperature (T_{wall}), convective heat transfer coefficient (H) and friction loss (F_q) is calculated using the following equations.

$$T_{wall} = (0.388 \times \theta + 423)K \qquad ...(2)$$

$$F_q = 2 \times 8.5 \times 10^{-7} \times V_{mean}{}^2 \times \frac{\pi BL}{2} \times \frac{(|T_1 - T_0|)}{v_2} \qquad ...(3)$$

$$H = 0.013 \times B^{-0.2} \times P_1{}^{0.8} \times T_1{}^{-0.55} \times V_{mean}{}^{0.8} \qquad ...(4)$$

2.2.1 Suction

The calculation of the process of suction begins with the assumption that the temperature remains constant during the specific degree of crank rotation. Based on this and assuming the values of mass, temperature and pressure of the exhaust gases we calculate the different parameters using the below mentioned equations.

$$P_1 = \frac{P_0 V_0}{V_1} \qquad ...(5)$$

$$\Delta P = P_{atm} - P_1 \qquad ....(6)$$

$$velocity\ (u_1) = C_d \sqrt{\frac{2|\Delta P|}{\delta}} \qquad ...(7)$$

Where ρ is density calculated taking into consideration mass leaving or fresh air entering depending on the pressure difference.

$$\Delta m = \rho A u_1 \Delta t \qquad ...(8)$$

$$mass\ (m_1) = m_0 + \Delta m \qquad ....(9)$$

The calculations for mass fraction and gas constant are also done at the particular instant. Further, using the values of calculated mass, mass fraction and taking specific heats of each constituent as a function of temperature, obtained from JANAF Tables, We have than calculated the value of temperature by solving quadratic equation obtained by the energy conservation in flow process using the equation below, since its value changes due to exchange of mass.

$$m_1 C_{p_1} T_{new} = \begin{cases} m_o C_{p_0} T_0 + \Delta m \left\{ \dfrac{\left(C_{p_o} T_0 \right) + \left(C_{p_1} T_{new} \right)}{2} \right\} & ; \Delta m \leq 0 \\ m_o C_{p_0} T_0 + \Delta m C_{p_{air}} T_{air} & ; \Delta m > 0 \end{cases} \quad ...(10)$$

Using the momentum and energy conservation law, the modified temperature can be calculated considering the losses occurring in friction and convection (q)which further can be used to find out modified pressure.

These new values are used to iterate the suction process from the start. Generally, 2-3 iterations give a good level of accurate values.

2.2.2Compression
2.2.2.1 Compression prior to injection
After the suction stroke, the calculations of compression begin by assuming adiabatic process and hence calculating the temperature T_1 after each 1^0 rotation throughout the stroke.

$$T_\theta \left(V_\theta \right)^{\gamma-1} = T_{\theta+d\theta} \left(V_{\theta+d\theta} \right)^{\gamma-1} \quad ....(11)$$

The losses in compression follow same approach as in suction, thereby giving variation in temperature. Further,the new value of γ is calculated at an average temperature of corrected and initialones. Then using this value of γ we do good number of iterations to get the exact temperature. Pressure at this corrected temperature can be obtained easily by,

$$\frac{P_\theta \times V_\theta}{T_\theta} = \frac{P_{\theta+d\theta} \times V_{\theta+d\theta}}{T_{\theta+d\theta}} \quad ....(12)$$

2.2.2.2 Fuel injection
Fuel injector is opened at a specific degree of crank rotation during compression and the process follows the one followed during suction. It has been started at $280^0$and kept open till the required particular mass entersconsidering pre-ignition and turbulence factors. Inlet temperature of fuel is 300K.

2.2.2.3 Compression post injection
After the injection of fuel the same process of compression is followed again, once the fuel injector is shut off.

2.2.3 Combustion
The simulation of combustion involves complexities of solving seven equations to obtain moles of six products and the temperature after combustion. To solve these equations, the method of iteration has been used. These equations can be obtained as:

The combustion equation yields three equations by mole comparison of H, O and N, where the reactant parameters-the temperature, pressure, moles and mass can be knownfromcompression stroke.

$$xH_2 + yO_2 + y'N_2 \rightarrow aO_2 + bN_2 + cH_2 +$$
$$dH_2O + eOH + fNO$$
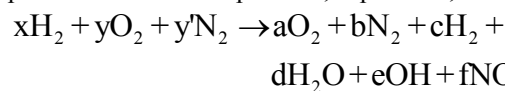
$$2x = 2c + 2d + e \quad ....(13) \qquad 2y' = 2b + f \quad ...(14)$$

$$2y = 2a + d + e + f ....(15)$$

The equilibrium constants can be expressed in terms of partial pressure as:

$$k_1 = \frac{P_{H_2O}{}^2}{P_{H_2}{}^2 \times P_{o_2}} ...(16) \qquad k_2 = \frac{P_{OH}{}^2}{P_{H_2} \times P_{o_2}} \quad ...(17)$$

$$k_1 = \frac{P_{NO}{}^2}{P_{N_2} \times P_{o_2}} ...(18)$$

Finally the energy balance on the reactant and product sides gives

$$\sum H_p - \sum_p n_j R T_j = \sum H_r - \sum_r n_i R T_i \quad ....(19)$$

Thus, to solve these 7 equations, initially we can suppose a temperature and find out the values of the constants in equation (13) - (18). The values of x and y are determined from the mass and moles fractions. Then the obtained values are verified using equation (19). After multiple iterations, we have obtained the moles of the products and the temperature which can be used to find pressure by ideal gas equation.

2.2.4 Expansion and Exhaust

The simulation of expansion, after combustion, follows the same approach as done in the compression stroke 3.2.2. Similarly, using the equations and method of suction 3.2.1, we can simulate the exhaust to obtain the final pressure and temperature.

This final temperature and pressure is used as initial data for suction to iterate the whole cycle repeatedly, until sufficient accuracy in values of these parameters are not obtained.

2.2.5 Area Planar of the inlet-exhaust openings

Since the analysis is based on *Fiat Lancia* engine, in this work, for our simulation, we have taken the area of the openings of the valve, keeping the ascent and descent angle of both the valves as designed. Also the variations in the valve lift and area of opening of the valves has been taken as designed which is presented in the graph below. However due to large area of the opening the exhaust blow-down loss is high. Hence various simulations are run for different areas and finally reduced the area by 75% of the practical valves for appropriate suction and compression.
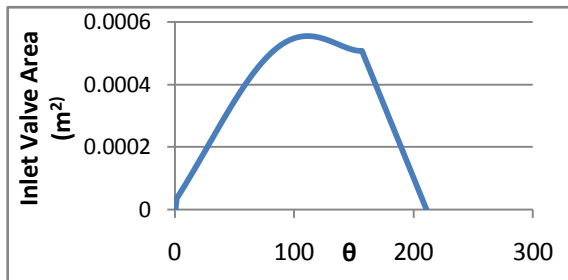
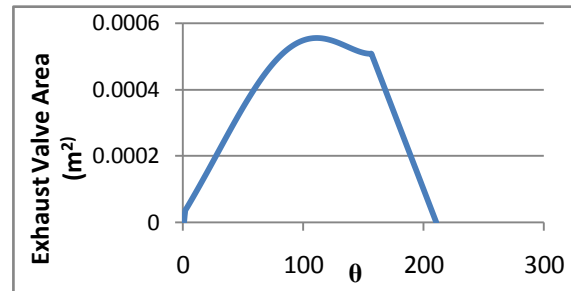

Figure 1: Inlet Valve Area vs θ

Figure 2: Inlet Valve Area vs θ

2.2.6 Determination of valve timings

We analyzed different valve timings to fix an optimum value giving maximum efficiency. But the inlet and exhaust valve advance are mutually dependent. However, the changing in work done by the engine varies quite less with inlet advance, since the pressure during the inlet valve opening is quite less as compared to the exhaust valve opening, as observed from the table (3). We have, hence, initially fixed the inlet valve advance as 10^0 before TDC, as referred from V. Ganesan [3], to thereby determine the accurate valve timings for the exhaust valve. Now, using the previously fixed timings, we can finally obtain the same for the inlet valve considering maximum mass entry of Oxygen. The parameters for different valve timings have been noted in the table (6).

According to the data of the engine valves, the inlet valve is kept open for 210^0 and exhaust valve for 240^0 of crank angle.

III.   Simulation results

Table (a). Variations in Exhaust valve timings

Exhaust Advance (Before BDC)	Work Done (J)	Efficiency (%)
40	483.95	39.506
35	484.379	39.541
30	**499.18**	**40.75**
25	483.915	39.503
20	482.905	39.42

Table (b). Variations in inlet valve timings

Inlet Advance (Before TDC)	Amount of Oxygen entering (kg) (x 10e-4)
22	0.7978
16	0.8146
15	**0.81476**
14	0.8136
10	0.78677

Table 1.Suction

Theta	Volume (m3) (10e-5)	Temperature (K)	Pressure N/m2 (10e5)
1	42.050	510.1035	0.9934
2	42.130	511.1578	1.0264
180	33.4851	325.2976	0.9871
194	33.1433	325.2099	1.0020
195	33.0925	325.2101	1.0034

Table 2.Compression

Theta	Volume (m^3) (10^{-5})	Temperature (K)	Pressure (10e5)
196	33.038	325.2066	1.0034
280	17.815	411.3374	2.32356
Mass before Injection			**3.5622 x 10^{-4}**
281	17.555	408.1219	2.4166
283	17.033	402.215	2.6071
285	16.512	396.9172	2.8089
287	15.992	392.1395	3.0233
289	15.475	387.8091	3.2511
291	14.960	383.8663	3.4937
Mass After Fuel is Injected			**3.6496 x 10^{-4}**
360	4.2024	622.6619	20.504

*** = Fuel Injector is kept open

Table 3. After Combustion Parameters

Compound	Variables	Moles
O_2	a	2.3966 x 10^{-8}
N_2	b	0.0095
H_2	c	4.921 x 10^{-10}
H_2O	d	0.0050
OH	e	3.9899 x 10^{-9}
NO	f	4.32 x 10^{-5}
Pressure		89.54
Temperature		3131

Table 4.Expansion

Theta	Volume (m^3) (10e-5)	Temperature (K)	Pressure N/m^2(10e5)
361	4.2041	3130.674	89.49586
509	31.7668	1865.216	7.066
510	31.8742	1863.46	7.035

Table 5. Exhaust

Theta	Volume (m3) (10e-4)	Temperature (K)	Pressure N/m2 (10e5)
511	31.9780	1863.46	7.035
715	4.2795	713.1197	0.979
716	4.2534	663.1463	0.9828
719	4.2073	550.5722	0.9911
720	4.2026	521.9448	0.9934

IV. Graphs
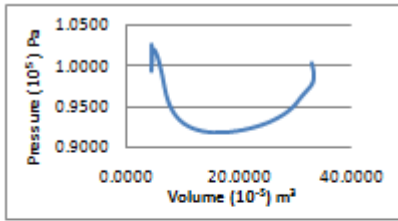
4.1 Suction

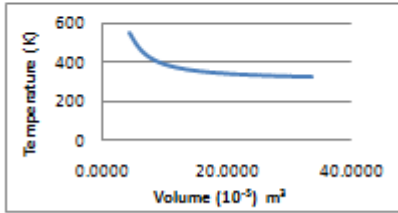Figure 3a: Pressure vs Volume (Suction)

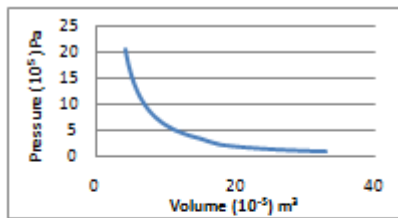Figure 3b: Temperature vs Volume (Suction)

4.2 Compression

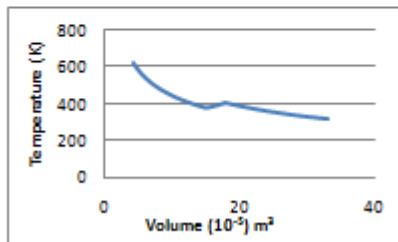Figure 4a: Pressure vs Volume (Compression)

Figure 4b: Temperature v Volume (Compression)
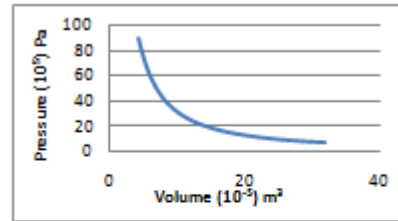
4.3 Expansion
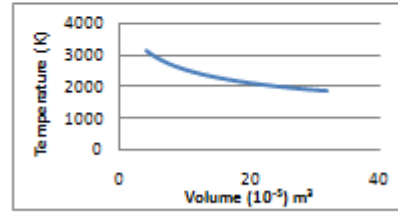

Figure 5a: Pressure vs Volume (Expansion)

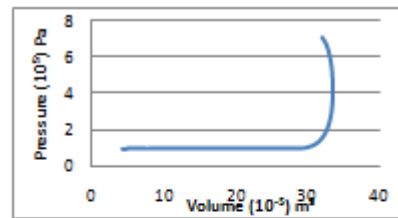Figure 5b: Temperature vs Volume (Expansion)

4.4 Exhaust

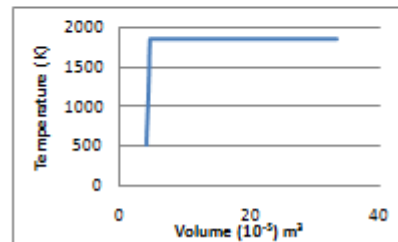Figure 6a: Pressure vs Volume (Exhaust)
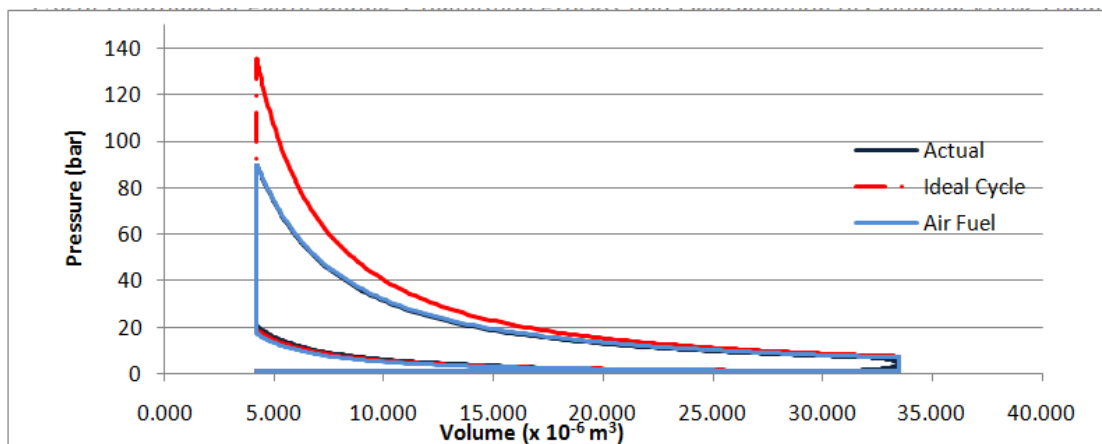

Figure 6b: Temperature vs Volume (Exhaust)
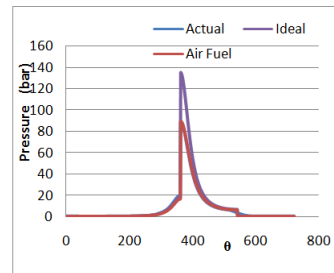

Figure 7: PV Cycle Comparison
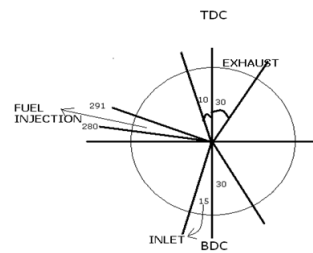
Figure 8: P vs θ cycle comparison

Figure 9 Optimum Valve timing Diagram

V.   Interpretation

The dependency of efficiency is quite low on inlet advance, compared to exhaust advance since the pressure is quite high at exhaust opening in contrast to the instant at inlet advance angle.

By calculating efficiencies for various exhaust angle advance, maximum efficiency is found to be occurring at 510^0, resulting in optimized piston work against high cylinder pressure and loss of expansion stroke and is closed at 10^0 after TDC.

At 1000 rpm and $\Phi=1$, we selected -15^0 and 195^0 of crank rotation for opening and closing of inlet valve to permit the maximum entry of O_2 gas in enginesince inlet valve is designed to be kept open for 210^0.

During the crank rotation angle of 281^0-291^0, there is rapid increase in pressure because of injected hydrogen in combustion chamber. For equivalence ratio equal to 1 the mass entry required is 8.74 x 10^{-6}kg and hence the fuel injector is kept open for 10^0 of crank rotation.

At equivalence ratio 1, combustion of 1g of H_2 produces 0.13334 g of NO and hence the %NO emission is 0.391% which is very less considering NO to be a dangerous emission.

The work done in the actual cycle is 499.18 J/cycle and hence the efficiency of the cycle is 40.75% and the efficiency of ideal cycle is 56.25 %

VI.   Conclusions

From the results of simulation, we obtained the efficiency of actual cycle 40.75% which is 72.39% of the ideal cycle when Exhaust valve advance is 30^0 and closes 10^0 after TDC. Inlet valve advance is 15^0 before TDC and closes at 195^c of crank rotation.We have seen that the emission percentage of NO has been very less and here there is no emission of CO_2 orCO and hence it is environment friendly.

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