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Mathematical Modelling and Simulation of Spark Ignition Engine for Predicting Engine Performance Characteristics

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Abstract: Depletion of fossil fuels and environmental pollution concern has led researchers to develop alternate fuels. Alcohols are promising alternative. Ethanol can be blended in gasoline without modification in engine to improve performance characteristics. Since, field testing is expensive and time consuming and moreover it is difficult to maintain favourable condition throughout the experiment. Hence it is easy and convenient to carry out the simulation study in MATLAB environment. This paper put emphasis on the study of engine performance characteristics with ethanol-gasoline blend, a cylinder by cylinder model is designed in MATLAB/Simulink. Whole model is built in the form of blocks which are differential and empirical formulas of engine parameters. These formulas are describing the engine behaviour with respect to crank angle. The pressure inside the cylinder is predicted with another line of information of heat input. Heat input needs data from the mass flow and burn characteristics of combustible fuel which is described by Weibe function. Pressure obtained will be converted to mean effective pressure by subtracting frictional losses. Finally the brake mean effective pressure is calculated. The parameters calculated are brake power, brake specific fuel consumption, fuel consumed, brake thermal efficiency, burn duration and exhaust gas temperature. In this study it is found that ethanol blend with gasoline increases brake power and brake thermal efficiency by lowering the exhaust gas temperature.

Key words: spark ignition engine, MATLAB, Simulink, simulation, performance characteristics, blending.

I. Introduction

In last three decades, there is much progress in the field of technology. Recent development in electronics tends researchers and scholars to enhance engine performance characteristics by developing new models. These development is due to scholar's ability to model engines, examine and test possible innovations. Modelling of an internal combustion engine is an intricate process and includes air gas dynamics, fuel dynamics, thermodynamic and chemical phenomena of combustion. Even in a steady-state condition, in which an engine is hot running at a constant load and speed, the pressure inside each cylinder changes rapidly in each revolution and the heat released by ignited fuel varies during the combustion period. [1]

The main idea behind engine modelling is to elucidate an engine phenomenon by establishing cause and effect dynamics relation between its main inputs and outputs. The dynamic relations used in modelling are differential equations obtained from conservation of mass and energy laws. The input variables in engine modelling are usually throttle angle, spark advance angle, exhaust gas recirculation and air-fuel ratio. The output variables are engine speed, torque, fuel consumption, exhaust emissions, and drivability. The challenge in engine modelling is to establish the relations between the engine input and output variables which best describe the model and predict the output variables in different working conditions of the engine. [2]

Various researchers have done rigorous work in the field to develop the relations and used them in Simulink for analytical analysis of engine performance characteristics. These relations can be used to investigate the behaviour of ethanol blending with gasoline in spark ignition engine.

Ethanol is being used by various scholars to investigate the engine performance characteristics since many decades. Ethanol is being a promising substitute for gasoline. It have been used for blending with gasoline/diesel. Ethanol shows upright behaviour to increasing engine characteristics. Ethanol blended with gasoline in various proportion by various investigators to find out the best optimal ratio of blend to maintain a balance between efficiency and fuel economy.

Wu weibin et al. [3] proposed the design and simulation of CNG/gasoline engine speed controller based on MATLAB. On the basis of reasonable CNG engine model, the digital PID speed controller and adaptive fuzzy PID speed controller model were built respectively in Simulink platform. It is found that fuzzy PID controller improved the CNG/gasoline dual fuel engine control performance, which has higher value of practice and application. Namitha sona et al. [4] designed a legitimate engine control unit so that, it may be introduced to an electricity generator and there by stabilize the system output to generate the expected amount of power. Paolo Casolia et.al. [5] Introduced an improvements in the original library set up in Simulink for control oriented simulation of Internal Combustion Engines (ICE) and powertrains. With a very low computational time, the model showed interesting capabilities in the simulation of the behaviour of automotive engines with crank angle resolution and therefore has been used in an original HiL application.

II. Methodology

A cylinder by cylinder model is developed for predicting the engine performance characteristics. In this model each cylinder are described individually and generate an output signal for example mean effective pressure with each individual combustion pulse present. This model is derived from engine geometries which is virtuous for improving and optimizing the engine in future.

As practically engine has many parameters which cannot be modelled hence assumptions are made to develop the model. The engine to be modelled is a gasoline engine which has no exhaust gas recirculation (EGR) and turbo system. The objective of present model is to predict pressure inside the cylinder and intends to develop a model which can describe the effects of each parameter on the engine performance. Physical and empirical formulas are used to investigate the behaviour of engine due to no perfect equations present which can describe phenomena in the engine. Model assumes multi-port injection system for injecting charge into the cylinder. Evaporation of fuel in gaseous phase occurs completely. Fuel and air mixes with each other perfectly. Mixture of air and fuel throughout the model is an ideal gas. There is no effect of throttle body or there is no pressure drop at throttle body due to wide open throttle condition. There is no effect of combustion chamber design and its temperature is maintained at 400k. There is no effect of exhaust stroke and exhaust system.

Table 2.1 Simulation Conditions

4 stroke, 4cylinder water cooled petrol engine, Hindustan motor, 84 mm bore, 82 mm stroke				
Ambient Pressure	1 atm			
Ambient Temperature	298 K			
Mean cylinder wall temperature	400 K			
Specific heat ratio (k)	1.3			
Air molecular weight	28.97 g/mol			
Air density	1.2 kg/m3			
Molecular weight (g/mol)	gasoline	E10	E20	Ethanol
	114	109.2	88.03	46.06
Heating value (kJ/kg)	44000	41900	40000	27000
Air/fuel ratio	14.6	14	13.5	9

The model is developed having five fundamental blocks which further contains twenty three sub systems describing the behavior of engine. The physical information like displacement volume, exposed area and volume variation with respect to crank angle etc. are calculated by engine geometry information such as bore, stroke, compression ratio etc. The equations of cylinder volume and exposed combustion chamber surface area are shown below [6]:

$$V(\theta) = \frac{V_d}{\varepsilon - 1} + \frac{V_d}{2}\left\{\frac{1}{a} + 1 - \cos\theta - \left(\left(\frac{1}{a}\right)^2 - \sin^2\theta\right)^{\frac{1}{2}}\right\} \quad 2.1$$

$$A(\theta) = \frac{\pi}{2}b^2 + \pi b\frac{s}{2}\left\{\frac{1}{a} + 1 - \cos\theta - \left(\left(\frac{1}{a}\right)^2 - \sin^2\theta\right)^{\frac{1}{2}}\right\} \quad 2.2$$

By having this data pressure inside the cylinder predicted with another line of information of heat input. Heat energy needs data from amount of flow in mass and burn characteristic which is described by Wiebe function.

Overall heat input [7] can be determined by using heating value of fuel and amount of mass which is drawn into cylinder.

$$Q_{in} = \frac{HV\int_{IVO}^{IVC} m(\theta)d(\theta)}{1 + m_{air,stioch}} \quad 2.3$$

A functional form used to represent the mass fraction burned versus crank angle curve is the Wiebe function [8]:

$$f(\theta) = 1 - \exp\left[-5\left(\frac{\theta - \theta_0}{\Delta\theta}\right)^3\right] \quad 2.4$$

The heat release (dQ) over the crank angle change ($\Delta\theta$) is:

$$\frac{\partial Q}{\partial\theta} = Q_{in}\frac{df}{d\theta} \quad 2.5$$

Then take the derivative of the heat release function (f (θ)), with respect to crank angle, being $\frac{df}{d\theta}$.

In spark ignition engines primarily heat transfer mechanism from the cylinder gases to the wall is due to convection and only 5% from radiation. Using a Newtonian model, the heat loss to the wall is given by [9]:

$$Q_{loss} = hA(T_g - T_w) \quad 2.6$$

The Hohenberg correlation for calculating co-efficient of heat transfer which is used in calculating heat loss trough cylinder [10].

$$h = 130V^{-0.06}P^{0.8}T^{-0.4}(C_m + 1.4)^{0.8} \quad 2.7$$

Pressure equation for engine cylinder is derived from the first law of thermodynamics. The pressure is derived as a function of crank angle also.

$$\frac{dP}{d\theta} = \frac{k-1}{V}\left(\frac{\partial Q}{\partial\theta} - Q_{loss}\right) - k\frac{P}{V}\frac{dV}{d\theta} \quad 2.8$$

$\frac{dV}{d\theta}$ can be determined from Eq.2.1 by taking derivative with respect to the crank angle (θ). The Wiebe function for the burn fraction is used for $\frac{dQ}{d\theta}$.

When determining the heat release term ($\frac{dQ}{d\theta}$) the heat loss to the walls has to be taken into account. Substitute Eq.2.5 and Eq.2.6 in Eq.2.8, the pressure over crank angle now becomes:

$$\frac{dP}{d\theta} = \frac{k-1}{V}\left[Q_{in}\frac{df}{d\theta} - \frac{hA}{6N}(T_g - T_w)\right] - k\frac{P}{V}\frac{dV}{d\theta} \quad 2.9$$

The intake system (air filter, carburettor, and throttle plate, intake manifold, intake port, intake valve) controls the amount of air which an engine of given displacement can breathe. The parameter used to measure the effectiveness of an engine's breathing is the volumetric efficiency (η_v).Volumetric efficiency is only used with four-stroke engines which have distinct induction process. It is defined as the volume of air which is drawn into the intake system divided by the volume which is displaced by the piston [11].

$$\eta_v = \frac{m_a}{\rho_a V_d} \quad (2.10)$$

There are no physical model present which can predict the trend of the volumetric efficiency exactly because it is affected by the type of fuel, engine design and engine operating variables.

The fundamental principle of the valve lift design is to satisfy an engine breathing requirement at the design speeds. Valve lift design uses polynomial function as described by Hermann, McCartan and Blair (HMB) technique [12] uses up to 11th order polynomial functions. The alternative approach is the G. P. Blair (GPB) method which considers jerk characteristic of valve motion. The method is employed depends on how smooth the lift and acceleration diagrams are otherwise the forces and impacts on the cam follower mechanism will be considered.

$$L_v(\theta) = \frac{L_{iv,max}(1 + \cos\phi)}{2} \qquad (2.11)$$

$$\phi = \frac{\pi(IVO - IVC + 2\theta + 540)}{IVO + IVC + 180} \qquad (2.12)$$

The mass flow rate through a poppet valve is usually described by the equation for compressible flow through a flow restriction. The 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 (C_d).

$$\dot{m} = \frac{C_D A_R p_0}{RT_0}\left(\frac{P_T}{P_0}\right)^{\frac{1}{k}}\left\{\frac{2k}{k-1}\left[1 - \left(\frac{P_T}{P_0}\right)^{\frac{k-1}{k}}\right]\right\}^{1/2} \qquad (2.13)$$

When the flow is choked, the pressure ratio $\left(\frac{P_T}{P_0}\right)$ will not lower than the following value so called critical pressure ratio.

$$\left(\frac{P_T}{P_0}\right) = \left(\frac{2}{k+1}\right)^{\frac{k}{k-1}} \qquad (2.14)$$

For the mass flow of the mixture into the cylinder through the intake valve, P_0 is ambient pressure, and P_T is the cylinder pressure. T_0 is ambient temperature. For A_R, the most convenient reference area in practice is the so called valve curtain area since it varies linearly with valve lift and is simple to determine [13].

$$A_R = \pi D_v L_v \qquad (2.15)$$

Eq.2.13 should be converted into a function of crank angle also by dividing with 6N.

$$\frac{dm}{d\theta} = \frac{C_D A_R P_0}{6N(RT_0)^{\frac{1}{2}}}\left(\frac{P_T}{P_0}\right)^{\frac{1}{k}}\left\{\frac{2k}{k-1}\left[1 - \left(\frac{P_T}{P_0}\right)^{\frac{k-1}{k}}\right]\right\}^{1/2} \qquad (2.16)$$

Here, although the flow through valve is dynamic behaviour, it has been shown that over the normal engine speed range, steady flow discharge coefficient results can be used to predict dynamic performance with reasonable precision [14].

$$C_D = 190\left(\frac{L_v}{D_{iv}}\right)^4 - 143.13\left(\frac{L_v}{D_{iv}}\right)^3 + 31.248\left(\frac{L_v}{D_{iv}}\right)^2 - 2.5999\left(\frac{L_v}{D_{iv}}\right) + 0.6913 \qquad (2.17)$$

Predicted pressure will be used to determine temperature inside cylinder and also heat transfer from cylinder to wall chamber. Rate of heat loss will be fed back to the pressure prediction function. Resulted pressure will be converted to indicated mean effective pressure subtracted by mean friction, then work and power will be known finally [15].

$$P_{mi} = \frac{\oint PdV}{V_d} \qquad (2.18)$$

Where, P is predicted pressure and V_d is volume displaced which is calculated in engine geometry block.

$$P_{bme} = P_{mi} - P_f \qquad (2.19)$$

Here, P_f is frictional losses [16].

$$P_f = 0.05\left(\frac{N}{1000}\right)^2 + 0.15\left(\frac{N}{1000}\right) + 0.97 \qquad (2.20)$$

For $1000 \leq N \leq 6000$

III. Result and discussion

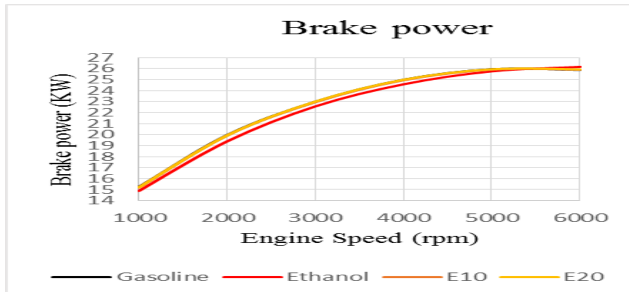

Figure 3.1 comparison of gasoline, E10, E20 and ethanol for brake power vs. engine speed

From Fig.3.1 it can be seen that brake power increases with increasing engine speed and maximum brake power is obtained for gasoline up to engine speed 5250 rpm, for engine speed 5250 to 6000 rpm ethanol results in maximum brake power. Brake power is directly proportional to brake mean effective pressure which completely signifies the behaviour of increasing trend of brake power on increasing engine speed. Blending ethanol in the ratio of 10% and 20% by volume basis respectively as E10 and E20, which did not show much variation in brake power as engine speed increases. As marked in above graph ethanol shows slight increase in brake power due to high brake specific fuel consumption and cooling effect caused by the ethanol during combustion.

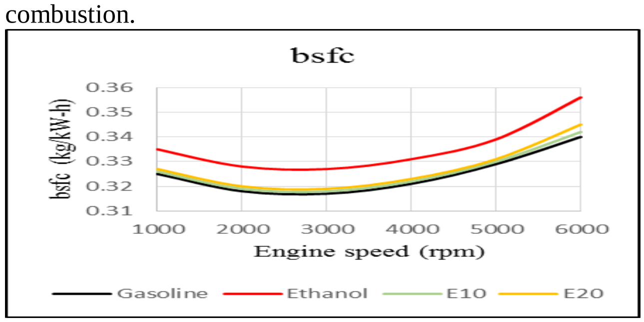

Figure 3.2 comparison of gasoline, E10, E20 and ethanol for bsfc vs. engine speeed

Fig.3.2 shows the behaviour of brake specific fuel consumption with increasing engine speed. It can be observed from the graph that bsfc first decreases than increases at higher rate and for ethanol bsfc is more than that of gasoline and other blends. This is due to the fact that up to engine speed 3000 rpm mass flow increases but brake power increases more rapidly but after engine speed 3000 rpm mass flow has more dominance over brake power. By virtue of this at higher engine speed there is more increase in bsfc.

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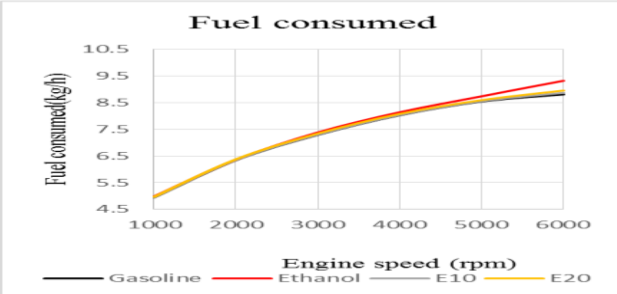

Figure 3.3 comparison of gasoline, E10, E20 and ethanol for fuel consumed vs. engine speed

Above Fig.3.3 implies the trend between fuel consumption and engine speed. As engine speed increases fuel consumption increases accordingly. There look like similar trend up to engine speed 4000 rpm after that there is increase in fuel consumption in case of ethanol with respect to gasoline, E10 and E20 respectively. This trend follow due to lower calorific value of ethanol which increases volumetric efficiency respectively as engine speed increases. Hence, as ethanol in petrol increases fuel consumption increases.

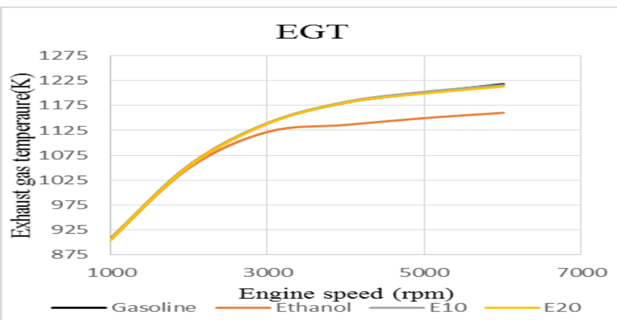

Figure 3.4 comparison of gasoline, E10, E20 and ethanol for EGT vs. engine speed

Above Fig.3.4 describes the exhaust gas temperature (EGT) behaviour with increasing engine speed. It appears from above graph that up to engine speed 2200 rpm exhaust gas temperature follow same tendency after that there is decrease in EGT for ethanol as compared to gasoline which has high EGT. As mentioned earlier ethanol causes cooling effect on combustion which leads to lower exhaust gas temperature. As it can be clearly seen that ethanol mixing with gasoline points toward lower exhaust gas temperature.

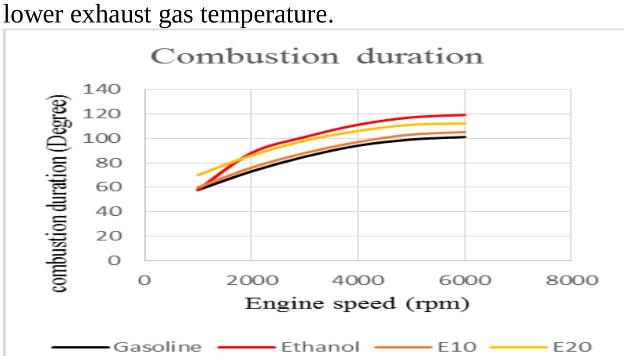

Figure 3.5 comparison of gasoline, E10, E20 and ethanol for combustion duration vs. engine speed

Above Fig.3.5 shows the combustion behaviour of ethanol and different blends with gasoline. At low engine speed ethanol shows less crank shaft angle for complete combustion as engine speed increases combustion duration increases for ethanol. In case of gasoline at low rpm combustion duration is similar to ethanol but at higher rpm it takes less crank shaft angle with respect to ethanol. E10 and E20 follows the trend similar to gasoline but as ethanol increases in gasoline combustion duration increases. As engine speed increases volumetric efficiency increases and due to low heating value of ethanol combustion duration increases. As ethanol added to gasoline heating value affected hence combustion duration increases.

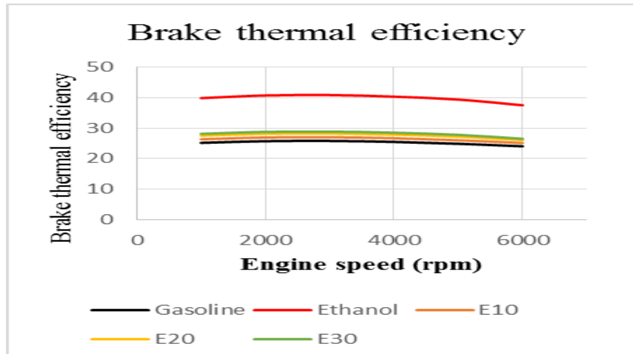

Figure 3.6 comparison of gasoline, E10, E20 and ethanol for brake thermal efficiency vs. engine speed

Above Fig.3.6 shows trend of brake thermal efficiency on comparing Gasoline, Ethanol, E10 and E20. Gasoline has low brake thermal efficiency, ethanol has high brake thermal efficiency comparing to E10 and E20. This is due to increases in volumetric efficiency and minimum exhaust gas temperature of ethanol. As ethanol is blend in gasoline the exhaust gas temperature decrease with increasing percentage of ethanol in gasoline. Hence, for ethanol there is high brake thermal efficiency with respect to gasoline and other blends.

IV. Conclusions

Engine dynamics, pressure and temperature, heat function, wiebe function and burn duration modules etc. are implemented in MATLAB/Simulink environment to analyze the engine performance characteristic analysis. Brake power increases as engine speed increases but in case of ethanol blends (E10 and E20) it follows the trend of gasoline and there is slight increase in brake power with respect to ethanol. Ethanol shows high brake power at high rpm due to increase in mass flow. bsfc is more for ethanol due to low calorific value of ethanol. As ethanol percentage increases in blends bsfc increases and follows the same trend of gasoline. As engine speed increases fuel consumption increases but in case of ethanol it increases more due to low heating value of ethanol. Exhaust gas temperature decrease on increasing ethanol percentage in gasoline which is due to the oxygen molecule present in ethanol. This causes cooling effect in the cylinder and proper combustion of fuel. Brake thermal efficiency is more for ethanol due to less exhaust gas temperature and more fuel consumption on increasing engine speed. In case of ethanol blends brake thermal efficiency increases as ethanol percentage increases. The simulations in order to predict the burning duration of the alternative fuels express many interesting information. Ethanol and ethanol blends shows greater combustion durations as compared to gasoline. As ethanol percentage increases in gasoline there is faster combustion of fuel in the cylinder.

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