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A Cycle Simulation Model for Bio-Fueled Conventional Engine

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ABSTRACT: In recent years, much research has been carried out to find suitable alternative fuel to petroleum products. The use of renewable fuels like ethanol, biogas and biodiesel in diesel engines is significant in this context. Diesel enables then to be engines used in transport, industrial and agricultural applications. The high efficiency and reliability, however they suffer from high smoke and nitrogen oxide (NO_x) emissions. The more strict government regulations on exhaust emission and the fast consumption of world-wide petroleum reserves provide a strong encouragement for the analysis of alternate fuels. Biodiesel is chosen a an alternate fuel in the research work since it is oxygenated, biodegradable, non-toxic and environmentally friendly but experimental analysis of the engine with various biodiesel and its blends requires much effort and time so to analyze the performance characteristics of the compression ignition engine fueled by biodiesel and its blends a theoretical model is developed.

I. INTRODUCTION

The majority of the world's energy needs are supplied through petrochemical sources, such as coal and natural gases. With the exception of hydroelectricity and nuclear energy, all of these sources are finite and at current usage rates will be consumed shortly. The high energy demand in the industrialized world as well as in the domestic sector and pollution problems caused due to the widespread use of fossil fuels make it increasingly necessary to develop renewable energy sources of limitless duration and smaller environmental impact than our traditional sources. This has stimulated recent interest in alternative sources for petroleum based fuels (Meher et al., 2006). Petroleum based fuels become the primary source of energy for transportation needs in the 20th century. This has continued in the beginning of the 21st century with almost all vehicles running on gasoline, diesel or natural gas (Loppacher and Kerr, 2005). Several experimental studies showed that vegetable oils can be used as alternate fuel for diesel engines. Some of these vegetable oils are follows: sunflower, rapeseed, cottonseed, jojoba and Jatropha curcas. As researchers have experimentally investigated the performance characteristics of conventional diesel engines fuels by using bio-fuel its blends but it requires enormous effort, money and time. In contains to that a numerical simulation model reduces the effort and time as well as finance involved , hence a numerical simulation based on mathematical modeling of diesel engine processes have long been used by design engineers to develop new design concepts. However, with bio-fuel, very few works have been done and it is still a newer area of research.

The process of diesel combustion is very complex and heterogeneous in nature. Diesel engine combustion models are mainly described as thermodynamics and fluid dynamic models. Models based on thermodynamics can be further classified as:

(i). Single zone heat release model
(ii). Phenomenological jet based model
(iii). Quasi dimensional multi-zone model
(iv). Fluid dynamic based model.

II. DESIGN METHODOLOGY

The combustion process can be described with varying complexity and accuracy. Normally the degree of complexity is decided by the number of zones in which the cylinder has been divided. An engine model therefore is either single-zone or multi-zone. In a single-zone model the gas mixture within the cylinder is considered to be homogeneous for each sample. It is also assumed to be made up strictly of ideal gases. In a multi-zone model, for example the two-zone model, the gases are still considered ideal. However, the homogeneous approach has been replaced by a heterogeneous one. Here the cylinder is also divided into two zones, one containing injected fuel and the other surrounding air. Each zone itself is homogeneous and no heat transfer occurs between the two-zones. The simplicity of the single-zones model is its biggest advantage. This makes it fast and therefore applicable in real time systems. The multi-zone model is more complex and more accurate compared to the single-zone model. A multi-zone model is often needed for combustion chamber design, but for most aspects of control design a two-zone model is good enough. The brief overall structure of methodology is as shown below in Fig.1. The model due to the complexity of the diesel engine combustion and the turbulent fuel air-mixing it is difficult to develop an effective model that does not have too long computational time. There are different approaches to implement a diesel combustion model i.e., single-zone or multi-zone (Gogoi and Barauch , 2010; Rakopoulos *et al.,* 2007).

To develop model that is accurate enough, has a acceptable simulation time and has a complexity level that reflects the timeframe of this paper, a two-zone model has been selected. The model includes only those processes occurring during closed cycle, when all valves are closed. The compression phase begins at the point of closing of the inlet valve (IVC) and continues up to crank angle at which combustion begins and the period from the end of combustion to the exhaust valve opening (EVO) is the expansion phase, where as the compression and expansion phases are assumed to be as polytropic.

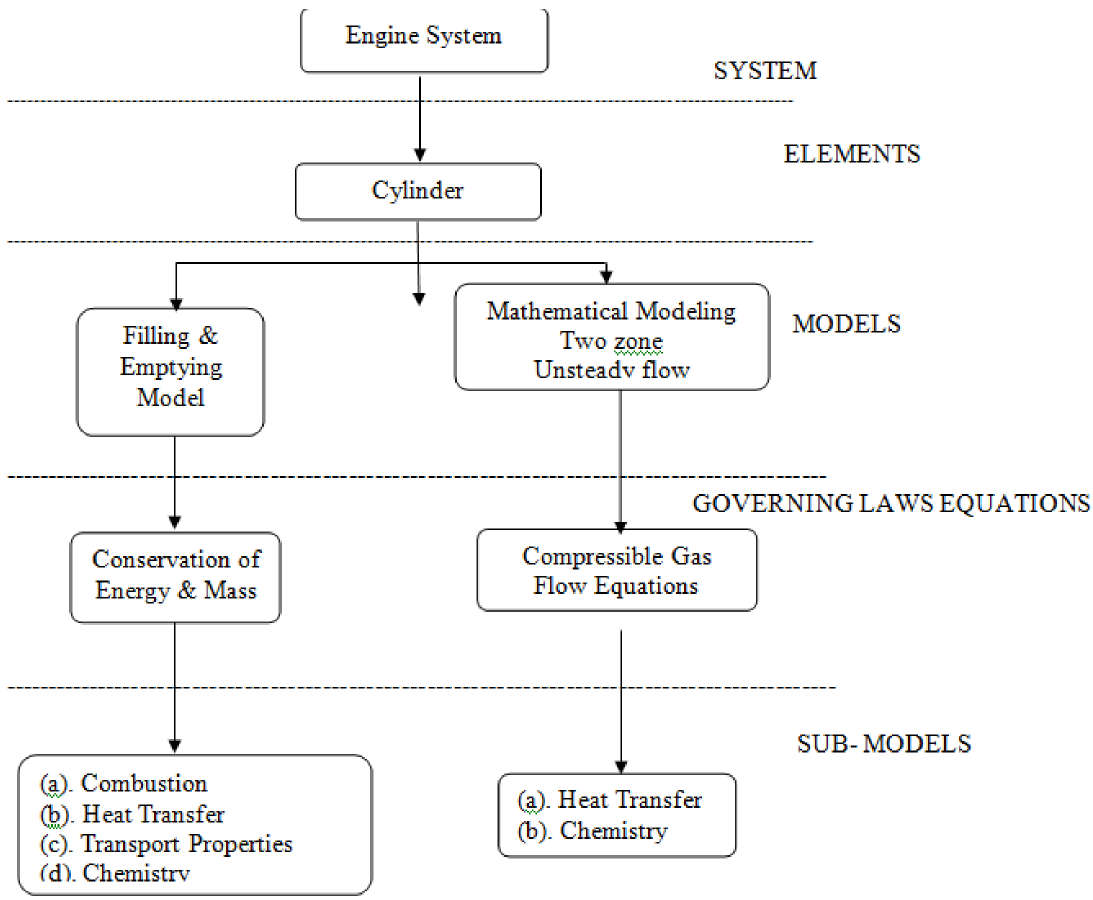

Fig. 1. Overall structure of engine systems model.

III. RESULTS & DISCUSSION

A. Validation of the model and testing of the software

In the combustion modeling the molecular formula of diesel fuel is taken as C_{10} H_{22} . From the fatty acid composition, Jatropha classifies it as linoleic or oleic acid types which are unsaturated fatty acids (Pramanik, 2003). So based on these properties, the molecular formula of Jatropha oil is aapproximated as $C_{18}H_{32}O_2$ (Balat and Balat, 2010; Barnwal and Sharma, 2005). In general, this combustion model, developed for C.I. engine analysis is suitable for any hydrocarbon fuel. This includes diesel, biodiesel or their blends as well as vegetables oil. The engine model is analyzed for the variation of different input parameters with performance and emission parameters for the diesel as well as Jatropha oil. Detailed program is shown in the annexure.

B. Effect of density of air on heat transfer per crank angle

The variations of heat transfer per crank angle with respect to density of air as density of air increases in the combustion chamber, heat transfer is also increases. So such types of variations of heat transfer and air density verify the programming in the software.

C. Net work done output with crank angle

The net work done with crank angle variations is shown in Fig.2. The negative work done in the figure shows that the work is done on the system in the compression phase, which is from 225^0 to 345^0. After the fuel injection in the combustion chamber, burning take place. At this stage pressure rise in cylinder takes place. At this stage pressure rise in the cylinder takes place and expansion phase starts which shows the positive work done.

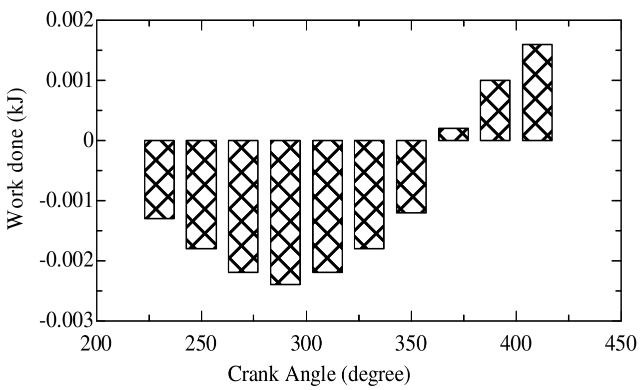

Fig. 2. Variation of Work done v/s Crank Angle.

D. Effect of cylinder bore on heat transfer

As increase the cylinder bore, total heat transfer is increasing. This is due to the total surface area exposed to atmosphere is increased. Variations shows in the Fig.3 verify the programming results.

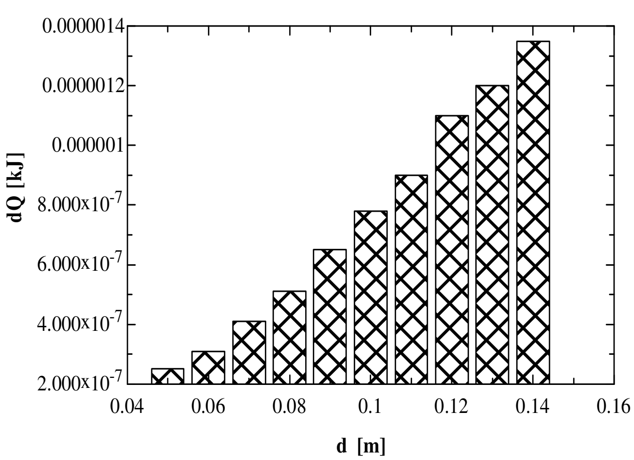

Fig. 3. Variation of Heat transfer v/s Cylinder bore.

E. Effect of thermal conductivity of air on heat transfer

Effect of thermal conductivity of air on heat transfer is explained in bi-quadratic polynomial in logarithmic scale. As thermal conductivity of air is goes on increasing the total heat transfer also increases from the combustion chamber.

F. Correction factor for various performances and emission parameters

Correction factor for different performance and emission parameters can be finding out by different governing equation as explained below. With the help of these equation correction factor can be calculated for any variable condition which are given in the form of x axis variable. After putting these variables in the correction factor equation, value of correction factor can be calculated for that performance and emission parameters. Calculated value for those parameters is added or subtracted in the calculated value of the modified model and get the proposed value. So we can calculate the approximated value of required parameters.

G. NO_x emission

NO_x correction factor variations with fuel injection pressure shows in Fig.4. In both the conditions the value of correction factor are negative as well as major differ from the trends line. These are due to not correctly calculating the numbers of moles of each constituent which are taking part in combustions, which effects the gas properties calculations.

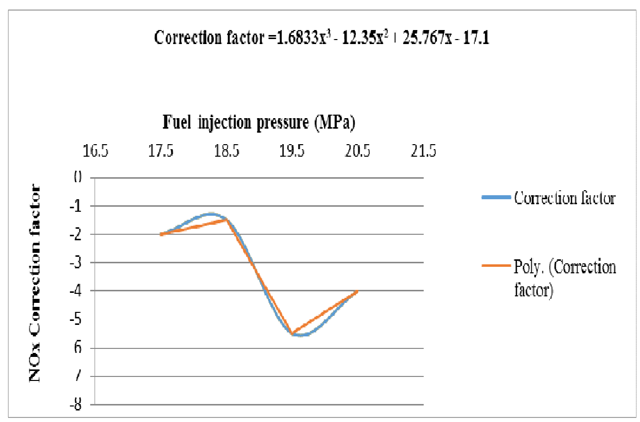

Fig. 4. NO_x correction factor with fuel injection pressure.

H. Comparison between estimated bte and corresponding experimental data v/s crank angle for jatropha oil

Fig. 5 represents the variation of BTE as a function of crank angle at speed 100 rpm and 80% of full load (735 kW) for modeling and experimental conditions .

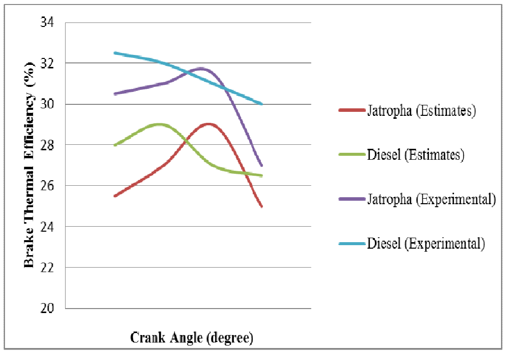

Fig. 5. Comparison between estimated BTE and corresponding experimental data v/s crank angle for Jatropha oil and Diesel at 1000 rpm and 80% of full load and static 17.5 MPa injection pressure

IV. CONCLUSION AND FUTURE SCOPE

A two zone thermodynamic model is modified for analyzing the performance characteristics of the compression ignition engine. The model is modified in such way that it can be used for characterizing any hydrocarbon- fueled engine, as diesel , biodiesel or its blends. This predictive model, which requires relatively small empirical input, was shown to be capable of generating important performance and emission parameters such as brake thermal efficiency, brake specific fuel consumption and NO_x emission. The predictive engine performance and emission parameter values showed a good agreement with the corresponding experimental data.

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