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Cylinder Pressure Variation Modeling inside a Diesel Engine

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Abstract

The in-cylinder pressure is the key of an internal combustion engine. Since direct measurement is not feasible, effort has been taken to find it find theoretical approach. Engine parameters studying by experimentation time consumes and efforts. Simulation the engine to find the performance output leads to resource saving. In this study a four-stroke diesel engine has been modeled to calculate the pressure inside the combustion chamber, gas and inertia forces instead of experimental data taken from real engine. The paper investigates the pressure variation inside a diesel engine during a complete cycle with respect to the crank angle at a constant speed, without considering properties change with temperature by proximate the real processes. In this work a simple method is proposed to estimate the in-cylinder pressure. The proposed method incorporates a combustion model. The model results have been compared to real data measured from a 6-cylinder compression engine. The results demonstrate the method's capability to update the model parameters, such that accurate in-cylinder pressure estimations are possible even under the influence of unknown disturbances.

Key Words: in- Cylinder pressure, Engine Model, diesel engine.

INTRODUCTION

In-cylinder pressure always has been a significant experimental diagnostic due to its direct relation to the combustion and work producing processes during internal combustion engine analysis. The in-cylinder pressure reflects the combustion process involving piston work produced on the gas due to changes in cylinder volume. Thus, for accurate knowledge of how the combustion process propagates through combustion chamber is required and each of these processes must be related to cylinder pressure Richard Stone [1999], and [1] [1998], Usually a table is created according to the engine data from experiment test data.

There is no measurement available to validate the ability of the engine model to describe a diesel engine. J. Scarpati et al. in 2007, however used the same model to describe the cylinder pressure in straight 6-cylinder diesel engine, [5]. the comparison between the model output and measured cylinder pressure is done for three different engine speeds. There are different approaches and ways of modeling ranging from easy to compound method and may vary in both structure and correctness. Several approaches of recording engine data are possible. In conventional applications data is logged with a fixed acquisition rate, whereby the time interval between two following recordings is fixed. However, because an engine runs in a cycle dictated by a set of mechanical mechanisms slider-crank, poppet valves, etc. – and because these mechanisms have fundamental consequences to how combustion takes place, it is necessary to record data at known crank angle intervals.

Physical equations hypothetically describing the system is the common method since it makes a general model working for many working areas. Its weaknesses it is difficult to describe reality appropriately in theory. Also, it is often resource consuming.

Another approach is to base the model entirely on measurements. The measured data is kept as a table of two or more dimensions in a so-called black box and then fetched when needed depending on one or more input signals. This method often delivers an accurate outcome since it is based directly on empiricism, however it is only defined for a restricted region. A combination of both methods, where the main basis of the model rests on physical equations and black boxes are used to model certain complexities, is also common.

An engine model therefore is either single-zone or multi-zone. In a single-zone model the gas blend inside the cylinder is considered to be consistent for each sample. It is also supposed to be made of perfect gases. In a multi-zone model, the gases are also ideal. However, the homogenous method 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 homogenous and no heat transfer arises between the two sectors. The simplicity of the single-zone model makes it fast and applicable in real time systems. The multi-zone model is more complicated and more accurate and frequently wanted for combustion chamber design. a single-zone model is good enough, for most using.There are two main approaches firstly to model all cylinders as one, describing the entire engine torque as a mean value over one or more engine cycles. Method is called Mean Value Engine Model. An alternative method is a model describing the in-cycle variation torque. It describes each cylinder individually and generates a torque signal with each combustion pulse present. The objective of this study is to model in cylinder-pressure diesel engine. The model should focus on the torque generation and, in order to keep the model complexity as well as the calculation time should be kept to a minimum. Article structure

Reciprocating internal combustion engine working principle

With the intake valve open, the piston makes an intake stroke to draw a fresh charge into the cylinder. Next, with both valves closed, the piston undergoes a compression stroke raising the temperature and pressure of the charge. A combustion stroke is then initiated, resulting in a high-pressure, high-temperature gas mixture. A power stroke follows the compression stroke, during which the gas mixture expands and work is done on the piston. The piston then executes an exhaust stroke in which the burned gases are purged from the cylinder through the open exhaust valve. The processes in the air-standard Diesel cycle are

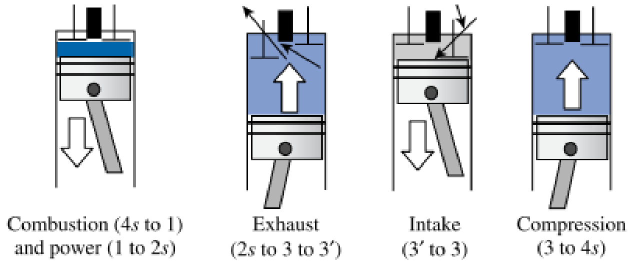

Fig.1. the operation of a four stroke Diesel cycle engine[4]

Air-Standard Dual Cycle

Pressure–volume diagrams of actual internal combustion engines are designated by dual cycle as shown in Fig.2 Process 1–2 is an isentropic compression starting with the piston at bottom dead center. The heat addition occurs in two steps Process 2–3 and 3–4 are a constant-volume+ pressure heat addition; Process 3–4 also makes up the first part of the power stroke. Expansion occurs is entropically from state 4 to state 5 is the remainder of the power stroke. The cycle is completed by a constant-volume heat rejection process, Process 5–1.

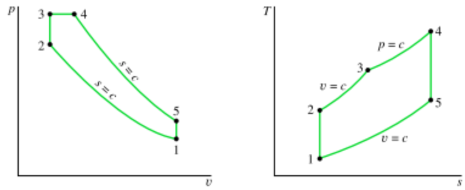

Fig. 2. p–v & T–s diagrams of the air-standard dual cycle

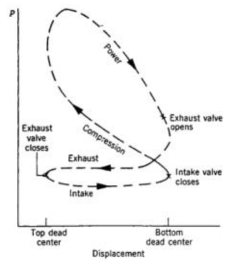

Fig.3. p−V indicator diagram (actual cycle)

Pressure calculation

Crank Mechanism Kinematics

The pressure generated in the cylinder is depend on angular movement of piston

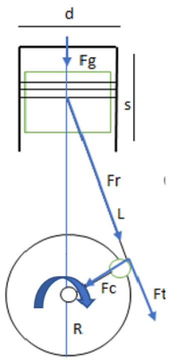

Fig. 4 dynamics of the crank mechanism

The above schematic drawing shows the crank mechanism. It consists of the piston, connecting rod and the crankshaft.

TDC = top dead center,

DC = bottom dead center

d = bore diameter,

L = connecting rod length

R = crank radius

$\boldsymbol{R} = \frac{S}{2}$

X = piston displacement, relative to TDC

θ = crank angle, clockwise relative to TDC

ω =the rotational speed of the crankshaft.

$n = \frac{R}{L}$ crank radius connecting rod ratio

$V_c = 2.R\,.\frac{\pi d^2}{4}$ displacement volume

V_c= clearance volume

$CR = \frac{V_c+V_d}{V_c}$=compression ratio

Using these geometry parameters, the displacement of the piston (relative to TDC) can be estimated. From this expression the piston speed and acceleration can be derived [MOOG00]:

$$V(\theta) = V_c + \frac{\pi d^2}{4} * X(\theta)$$

$$V(\theta) = \frac{\pi.R.d^2}{4} * \left[\frac{2}{CR-1} + (1-\cos(\theta)) + \frac{1}{n} * \left(1 - \sqrt{1 - n^2\, sin^2\theta}\right)\right]$$

This expression for the volume contains only cylinder geometry parameters, which are known.

Where the angle dependent cylinder volume and angular velocity and acceleration can be found as shown below [PISC89]:

The in-cylinder pressure was calculated based on the thermodynamics properties (Heywood, 1988) as given in the equations (5) and (6).

In the suction stroke the pressure is equivalent to atmospheric pressure so it can be approximated by

$$V_{\mathrm{t1}} = V_{\mathrm{d}} + V_{\mathrm{c}}$$

The angular velocity

$$\omega = \frac{2.N.\pi}{60}$$

The crank angle range for one cycle

$$\Theta = 1{:}720$$

$$\Theta = \omega.t$$

1. Intake stroke: θ =0-180

$$\theta \leq 180$$

$$p_{in} = p_{atm}$$

2. Compression stroke: θ =180– 360

$$\theta \leq 360$$

$$p_{comp} = p_{int} \left(\frac{V_{t1}}{V_{\theta}}\right)^{K}$$

3. Heat addition at constant pressure

$\theta \leq 410$

$$p_{comp} = p_{360}$$

4. The expansion stroke: θ =360– 540

$$\theta \leq 540$$

$$p_{comp} = p_{360} \left(\frac{V_{410}}{V_{\theta}}\right)^{K}$$

5. The exhaust stroke: θ =540– 720

$$\theta \leq 720$$

$$p_{EX} = p_{E}$$

RESULTS AND DISCUSSION

The graphs obtained by the ideal system and actual systems have been given below in figure 2 The graphics showed more clearly the difference between two systems.

Serious differences are seen in compression region of the both graph the theoretical one and actual cycle this is due to we don't add the effect of combustion in cycle analysis

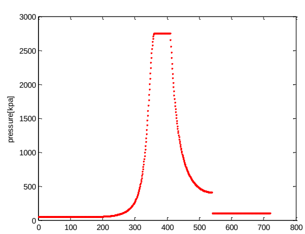

Fig. 2 Crank Angle vs Pressure [kpa]

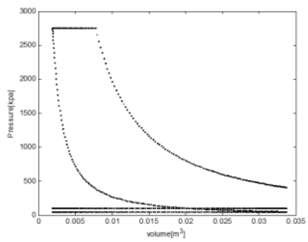

Fig. 3 Volume vs Pressure [kpa]

CONCLUSION

In-cylinder diesel engine model describing the engine torque has been derived and validated. The engine torque produced is the summation of all individual cylinder torques. Each cylinder torque is composed of gas pressure torque and inertia torque working on the crankshaft. The model constructed is neither computationally complex nor does it require any special efforts in order to be parameterized.

The model is general in the sense that it can describe both diesel and petrol engines. It could also be concluded that the algorithm presented in the paper could be expected is proven to be useful for an arbitrary number of cylinders.

The model is tested and evaluated for different scenarios, and the results are then compared to measurements made with a test vehicle in order to determine the accuracy

The gas torque during expansion stroke is more difficult to predict due to the unknown gas composition and initial gas condition at end of the combustion process. Therefore, the expansion period (mostly the crank angle after TDC) was avoided in the final implementation of this method.

The adiabatic process model Eq.12 is used to predict the gas pressure during the compression stroke.

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