التنبؤ بأعطال رشاشات الوقود في محركات الديزل

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ملخص البحث :

يتلخص هذا البحث بدراسة إمكانية تخمين ضغط الاسطوانة أو ضغط الاحتراق وشكل موجات ضغط غازات العادم الناتجة اثنا عملية تبادل الغازات في محركات الديزل لتنبو بأعطال رشاشات الوقود . بمعرفة الإبعاد الهندسية لبعض أجزاء منظومة طرد غازات العادم وتحليل الدورة الثرموديناميكية للمحرك وذالك بقياس وعمل نموذجية الأعطال الناتجة في منظومة حقن الوقود وتحليل عملية الاحتراق لمعرفة الطاقة الحرارية المنتجة والضغط الناتج داخل اسطوانة المحرك. تم دراسة علاقة وتأثير كمية الوقود المحقون للاسطوانة أثناء عملية الاحتراق في كل شوط قدرة على ضغط غازات العادم عند شوط طرد الغازات، كما وقد تم بناء برنامج حاسوبي (MATLAB) لكل نموذج رياضي يجسد عمليات دورة تشغيل المحرك ، و قد اعتمد البحث على النتائج المعملية التي تم قياسها من محرك ديزل نوع فورد رباعي الأشواط ذو أربعة اسطوانات تم تشغيله تحت ظروف مختلفة لمحاكاة بعض الأعطال التي قد تحدث لرشاشات الوقود.

لقد أظهرت النتائج تقاربا جيدا بين قيم الضغط المقاسة عمليا ، وقيم الضغط المستنتجة من هذه الدراسة ، وقد تم استنتاج تأثير أعطال رشات الوقود على شكل موجات الضغط لغازات العادم وإمكانية استعمالها لتنبؤ ببعض أعطال رشاشات الوقود في محركات الديزل.

Diesel Engine Exhaust Gas Flow Modeling for Fuel Injector fault Detection

Abstract

When the engine basic design parameters are known, the performance of the engine can be predicted with the assistance of simulation programs into the less time, cost and near value of actual. However, Mathematical modeling and dynamical simulation of the combustion processes in internal combustion engines have advanced rapidly, and is being increasingly accepted in the industry as a design aid. Most of these models have used the geometric parameters of the engine's systems and their relationship with the cylinder pressure wave as input data to predict engine performance. In this articles, cylinder pressure and an exhaust gas pressure wave based analysis was developed, for monitoring of fuel injector faults. A quasi-dimensional model was developed to simulate a 4- stroke direct injection diesel engine. The model includes the prediction of incidence of various faults such as fuel system or fuel injector damage in a combustion system of a direct injection diesel engine. Obtained results have shown that the simulated exhaust pressure behaviors and measured exhaust pressure waveforms are in good qualitative agreement and hence enable the waveforms to be used for the accurate diagnosis of the different fault sources from the complicated combustion system.

Keywords. Diesel engines, Exhaust pressure, Fuel injector.

1. Introdictions

The future designs of diesel engines are centering on the combustion system, valve timings and the fuel injector characteristics so that they can meet the ever stringent emission regulations. Such designs include the use of high pressure fuel injectors, micro injector nozzles, variable valve timing and smart control systems. If any of these components or systems were to fail or degrade there would be a negative effect leading to variances in the evaporation rate, bulk spray formation, spray penetration, engine gas exchange and fuel injector delivery rate variances. This would then result in an imperfect combustion processes and therefore a reduction in the engine performance.
As the requirements for emission reduction and fuel economy improvements have increased significantly the development of more complex engine control systems and combustion techniques become essential, for reducing engine emissions while maintaining the engine efficiency [1-3].

However, presently it is not possible to achieve such a monitoring scheme without combustion monitoring sensors such as cylinder pressure sensors, optical sensors or ionization sensors. These are however expensive, difficult to install and therefore are unsuitable for use in diesel engines.
The use of exhaust gas pressure measurements as a combustion monitoring tool for internal combustion engines has been investigated for misfire detection, valve leakage and exhaust manifold leakage [4-9]. The results from this work are promising for further development in diesel engines.

Recently the application of exhaust pressure measurements in diesel engine combustion monitoring was introduced. This method was used for valve leakage fault detection [8] and fuel injector fault detection [9-10] by using both the time and frequency domain analysis of the exhaust pressure wave measured at various locations along the exhaust system. The results from this research show that the accuracy of this method strongly depends on the measurement locations and engine speed. Measurement locations have a strong effect due to the variation of the exhaust system components geometry and their distance from the combustion events.

The effect of fuel injector faults on the engine exhaust pressure waveform were investigated using different exhaust systems and by varying the sensor locations. This clearly shows its capability for detecting minor combustion deviations between engine cylinders.

However, it is difficult to identify the exact causes of the deviations because there are many factors affecting the engine combustion process. Therefore, this study concentrates on engine modeling and simulation to find the features of exhaust pressure variations with different system configurations including various abnormalities. It attempts to bridge the gap between highly detailed chemical kinetics, computational fluid dynamic simulations, and the more simple dynamic models used in flow analysis. The features developed in this way can thus be relied on to diagnose the root of the faults which are detected from exhaust pressure measurements.

2. Engine Cycle Modeling

The engine cycle modeling has been proven to be one of the ways to understand the engine thermodynamics and working cycle operation. During the last three decades the mathematical modeling and dynamical simulations of the processes in internal combustion engines have advanced rapidly. Their use is being increasingly accepted in industry as a design aid. There are generally three types of models, gas dynamics, filling and emptying and quasi-steady models for calculating the intake and exhaust gas flow. The gas dynamics models describe the spatial variation in the flow and pressure in the manifolds. The filling and emptying models take into account the finite volume of critical manifold components and the quasi-steady models model the flow through restricted areas [11]. As the exhaust and intake manifold pressure ratio and engine volumetric efficiency play a key role in the parameters of engine performance, most of these models have used the geometric parameters of the engine's systems and their relationship with the pressure wave as input data to predict engine performance.

An analysis of the engine gas exchange process determines that the air mass to the cylinder depends on the pressure ratio between the exhaust manifold pressure and the intake manifold pressure [11] and [12]. Any change in exhaust manifold pressure will therefore require a change in the amount of injected fuel and mass flow rate of the inlet air which is governed by the engines valve timings. Several pieces of research on exhaust pressure estimation are reported in [9] and [7]. Each of these studies shows a particular aspect of the subject and uses a different approach from the other.

3. Gas Exchange Modeling

The model developed in this work is based on the first law of thermodynamics for engine cycle simulation and consideration of the exhaust valve opening period and gas flow from the engine as a thermodynamic open system. The flow calculation take into consideration the changing cylinder volume with movement of the piston to predict the mass flow and pressure variation in the exhaust system. The model includes a one dimensional analysis of the gas flow in the engine process by solving the mass and energy balance on each side of the engine system.

Figure 1 shows the gas exchange processes through the neutral breathing diesel engine components. When the inlet valve is open during the intake stroke the air enters the cylinder where it is compressed, the fuel is injected and then burned in the combustion process.

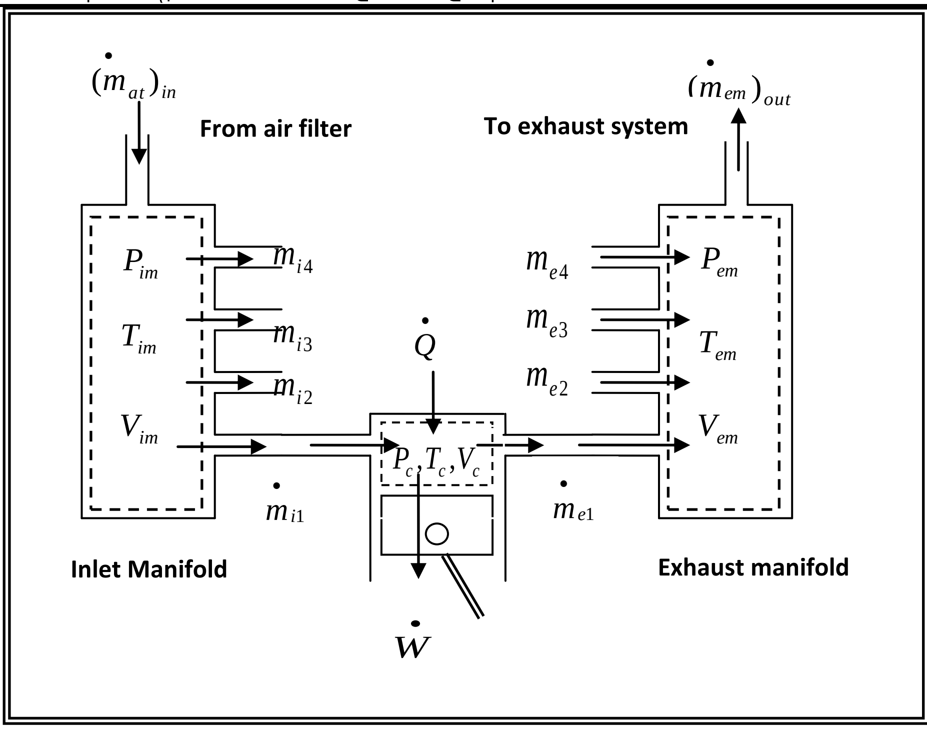

Figure 1 Schematic diagram of a 4 cylinder diesel engine, (subscript 1 refers to cylinder 1, etc)

The combustion products leave the combustion chamber when the exhaust valve opens. This process is repeated with every cycle of the engine. To describe this process accurately, it is necessary to develop the mathematical models of the dynamic cylinder pressure, heat release, exhaust gas pressure, valve movement and fuel injection.

In the models used for this study, the system is treated as an open system with the transfer of mass, enthalpy and energy in the forms of work and heat and with the engine cylinder as a variable control volume. Ignition commences when there is positive heat release and the resulting combustion process is simulated as a uniform heat release with the rate of heat release proportional to the rate of fuel injection to the engine cylinder during the entire combustion process. The fraction of the total fuel injected is burned in both period and depending on combustion period and engine load and speed, without any heat transfer to the cylinder wall or other engine system.

3.1 Flow Through Valves

Lets assumed that there is no leakage or any reverse flow from the engine cylinders to the inlet manifold or from the exhaust manifold to engine cylinders. The mass flow rate through the engine valves will include a mass of fresh air flowing from the inlet manifold to the engine cylinder $\dot{m}_i$, and mass flow rate of exhaust gases $\dot{m}_e$ from the cylinder to the exhaust manifold as shown in Figure 1. Considering the variation of the valve open area with valve lift, the equation for calculating the flow rate can be simplified by using the assumption of compressible, steady state and one-dimensional isotropic flow through the inlet valve [11].

$$\frac{dm_e}{dt} = \frac{C_d A_c P_{em}}{\sqrt{RT_{ec}}} \left(\frac{P_{ec}}{P_{em}} \right)^{\frac{1}{\gamma}} \left[\frac{2\gamma}{\gamma - 1} \left(1 - \left(\frac{P_{ec}}{P_{em}}\right)^{\frac{(\gamma-1)}{\gamma}} \right) \right]^{\frac{1}{2}} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots(1)$$

and for unchecked flow as

$$\frac{dm_e}{dt} = \frac{C_d A_c P_{ec}}{\sqrt{RT_{em}}} \sqrt{\gamma} \left(\frac{2}{\gamma + 1} \right)^{\frac{(\gamma+1)}{2(\gamma_{-1})}} \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots(2)$$

Where A_c is the valve open area,P_{im} is the manifold pressure, P_{ic} is the cylinder pressure during intake event, $T_{im}$is the inlet manifold temperature, R is the gas constant and γ specific heat ratio. In the case of the exhaust valve the exhaust gas flow rate $\dot{m}_e$ is calculated by the same equations by replacing P_{im} and P_{ic} in equation 1 and 2 with P_{em} and P_{ec}. Where P_{em} is the exhaust manifold pressure, P_{ec} is the cylinder pressure during the exhaust process and C_d is the coefficient of discharge for a given valve area which is assumed as constant and equal 0.6 [14]. To compute the instantaneous valve curtain area, A_c the valve lift and crankshaft rotation were experimentally determined from the engine by measuring the valve lift displacement as a function of the crank angle. Approximating the measured points using a polynomial function, the equation for instantaneous lift was determined by the given equation:

$L_V(\theta) = a_1(\theta)^3 + a_2(\theta)^2 + a_3(\theta) + a_4$ ....(3)

Where a_1, a_2, a_3 and a_4 are polynomial constant.

3.2 Cylinder Pressure Calculation

By applying the concept of a control volume to the engine cylinder, as shown in Figure 1 and neglecting the heat transfer from the system, the general equation for a cylinder pressure calculation [11] can be derived as:

$$\frac{dP_c}{dt} = \frac{\gamma - 1}{A_p z}\frac{dQ_{in}}{dt} - \frac{\gamma}{z}P_c\frac{dz}{dt} + \frac{\gamma}{A_p z}P_i v_i\frac{dm_i}{dt} - \frac{\gamma}{A_p z}P_e v_e\frac{dm_e}{dt} \quad \text{....................(4)}$$

where A_p is the piston cross-sectional area and z is displacement of the piston referring to TDC.

For the induction stroke, Equation (4) is simplified as

$$\frac{dP_c}{dt} = -\frac{\gamma}{z}P_c\frac{dZ}{dt} + \frac{\gamma}{A_p z}P_i v_i\frac{dm_i}{dt} \quad \text{.....................................(5)}$$

For combustion stroke equation 4 is simplified as

$$\frac{du_c}{dt} = \frac{dQ}{dt} - p\frac{dV}{dt} + \dot{m}_f h_f \quad \text{..(6)}$$

Where $\dot{m}_f$ is the total mass flow rate of the injected fuel during the combustion phases and h_f is the enthalpy of the mixture within the cylinder. Considering the gas as ideal and neglecting the enthalpy component ($\dot{m}_f h_f$) and assuming c_p is constant over the range of combustion temperature, from equation (4) and (6):

$$\frac{dQ}{dt}=\frac{\gamma}{\gamma-1}P_c\frac{dV}{dt}+\frac{1}{\gamma-1}V\frac{dP_c}{dt}$$ ....(7)

The heat release rate during the combustion phase is presented by:

$$Q_{ch}=Q_{LHV}.m_{f,b}$$ ....(8)

Where $m_{f.b}$ is the total mass of fuel burning during premixed x_b and mixed controlled combustion x_m and both can be calculated by the Wiebe function [13]:

$$\dot{x}_b=\frac{a\omega(m+1)}{\Delta\theta}\left(\frac{\theta-\theta_0}{\Delta\theta}\right)^m\exp\left[-a\left(\frac{\theta-\theta_0}{\Delta\theta}\right)^{m+1}\right]$$ ....(9)

Where θ is the crank angle, θ_0 is the crank angle at the start of combustion, $\Delta\theta$ is the change in crank angle that occurs during combustion and a and m are adjustable parameter obtained from Stone [13] where typically a = 5 and m = 4 . When there is no mass flow in or out of the valve the cylinder pressure during the combustion period can be calculated by:

$$\frac{dP_c}{dt}=\frac{\gamma-1}{V}\frac{dQ}{dt}-\gamma\frac{P_c}{V}\frac{dV}{dt}$$ ....(10)

In the exhaust period, the exhaust valve opens at the end of the expansion. During this period the inlet valve is closed and the exhaust gases are forced out of cylinder by the piston into the exhaust manifold. Assuming there is no back flow into the engine cylinder, equation (4) becomes

$$\frac{dP_{ec}}{dt}=-\frac{\gamma}{z}P_{ec}\frac{dZ}{dt}-\frac{\gamma}{A_pz}P_{ex}V_{ex}\frac{dm_e}{dt}$$ ....(11)

Where p_{ec} is the cylinder pressure at the exhaust period, p_{ex} is the exhaust gas pressure and V_{ex} is the volume of exhaust gas from one cylinder. The exhaust gas mass flow rate from the cylinder can be determined by equations (1) and (2).

3.3 Exhaust Gas Pressure Calculation

The total mass flow rate from all cylinders to the exhaust manifold ($\dot{m}_{em}$) is simply calculated from conservation of mass law.

$$\frac{d}{dt} m_{em,} = \sum_{n=1}^{n} \frac{dm_{ex}}{dt} \quad (12)$$

Where n is the number of cylinders. Assuming there is no heat transfer from and to the cylinder, from the ideal gas law the exhaust manifold can be modeled as a controlled chamber with a flow-in and flow-out. The pressure in this chamber can therefore be obtained by

$$\frac{P_{em}}{dt} = \frac{RT_{em}}{V_{em}} [(m_{em})_{in} - (m_{em})_{out}] \quad (13)$$

Where T_{em} and V_{em} are the manifold temperature and volume respectively.

4. The Experimental Work

All the experiments and data used in this work are from the Ford FSD 425 four cylinder 2.5 litter direct injection diesel engine. The major engine geometry and specifications used in this model are shown in Table 1.

Table 1. Physical dimensions of engine and valves

Number of cylinder	4	Exhaust valve out head diameter	36.4 mm
Firing order	1, 2, 4, 3	Exhaust valve inner head diameter	32.0 mm
Bore	93.67mm	Exhaust valve stem diameter	8.92 mm
Stroke 90.54	90.54mm	Inlet valve stem diameter	8.94 mm

Connecting road length	153.89mm	Exhaust valve lift	10.50mm
Exhaust manifold volume	$0.000062m^3$	Inlet valve head outer diameter	41.90 mm
Inlet valve open	13° before TDC	Inlet valve head inner diameter	37.90 mm
Inlet valve closed	39° after BDC	Exhaust valve seat angle	45°
Exhaust valve open	51° before BDC	Inlet valve seat angle	45°
Exhaust valve close	13° after TDC	Inlet and valve left	10.50 mm

To establish the validity of the model and to investigate the ability to detect the fuel injector faults at different operating conditions, the measurements of cylinder pressure, exhaust manifold pressure, exhaust temperature and engine load were collected as shown in figure 2.

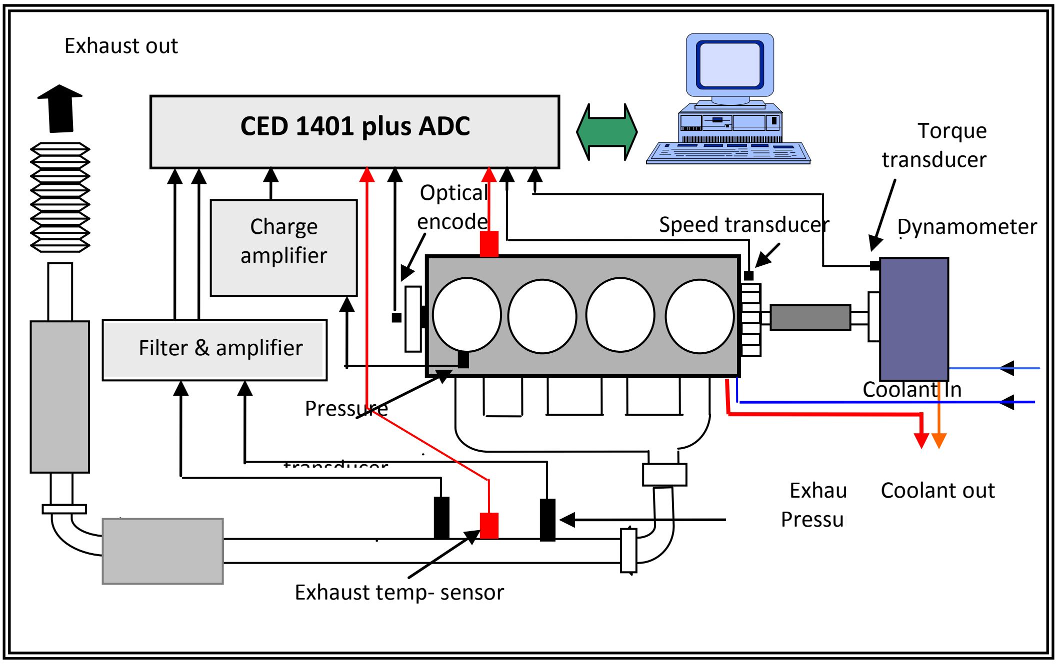

Figure 2. Schematic diagram of engine and measurement equipments

5. The Model Evaluation

To validate the model and determine how well it supported the objectives of this work it was tested with different input data to simulate engine operation at different loads and speeds. Figure 3a shows the instantaneous measured and predicted cylinder pressure at different engine loads. The measured pressures are very close to those of the predicted ones. An accurate prediction of the cylinder pressure when the exhaust valve is open is essential for this work. This is because it strongly affects the exhaust pressure wave characteristics. Looking closer at the pressure peak it can be seen that there is a slight difference between measured and predicted cylinder pressures in the case of high load (63 Nm). This variation in pressure is due to an increase in the amount of fuel injected and the heat released as shown in Figure 3b.

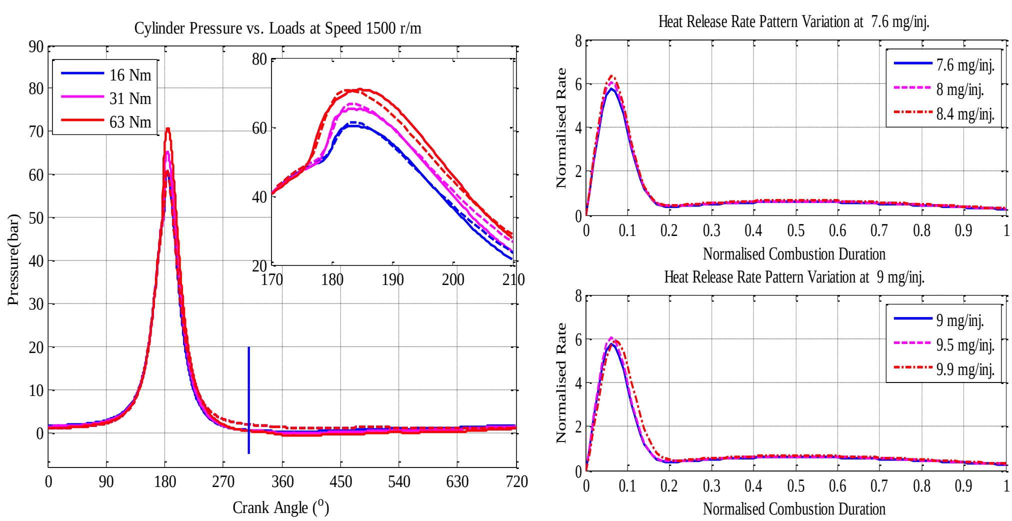

Figure 3. (a) measured and predicted cylinder pressure at different load (b) predicted heat release at different amount of fuel injected

The valve lift and valve opening area are the two main factors that control the out flow of the engine cylinder. The rate of mass through the valve and the exhaust pressure wave fluctuations are determined from the model and show that the flow rate follows the path of the piston motion. This would agree with Assanis and Heywood [15]. This shows the exhaust flow has two peaks, the first due to a rapid blow-down event when the exhaust valve is open and the second is consistent with the piston motion. Figure 4 and 5 shows the measured and predicted exhaust pressure for three different fuel injection rates (engine loads) for an engine speed of 1500 rpm. Comparing them, it can be seen that both the measured and the predicted pressure have similar variation behaviors with the engine load. The main peak of the pressure waveform for individual cylinders is just after the end of the exhaust stroke because of the effects of the advanced opening and the retarded closing of the valve motion. There are two distinct types of pressure feature can be observed. The first type, around the major peak, is due to the exhaust stroke from the upward piston motion. The second is between the smaller dip and the larger dip. This is mainly due to the residual pressure inside the cylinder at the moment of valve opening. With an increase in combustion pressure, the amplitudes of the main peak increase whilst the secondary peaks decrease. Moreover, the shape of the waveforms is uniform over the four cylinders. It is these features upon which condition monitoring can be performed for a multiple cylinder engine without the extra requirement of baseline signatures.

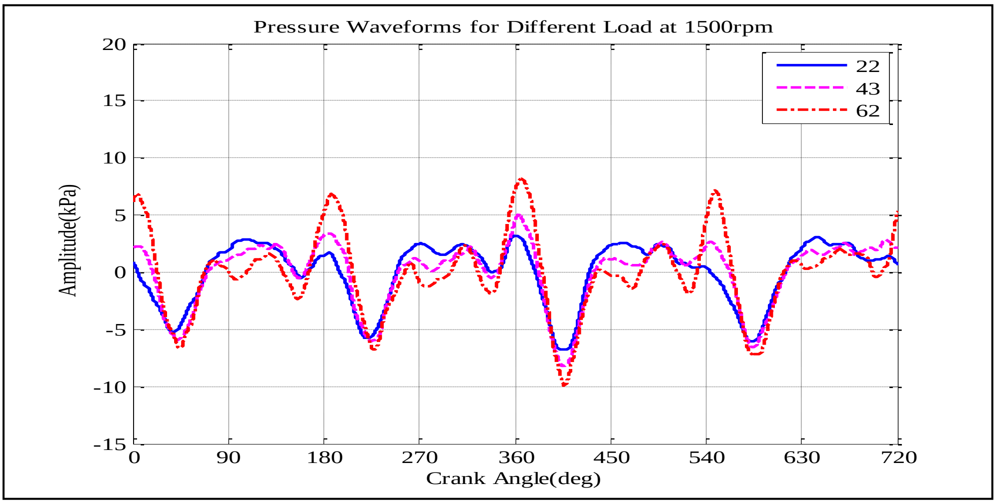

Figure 4. Measured exhaust pressure at different load

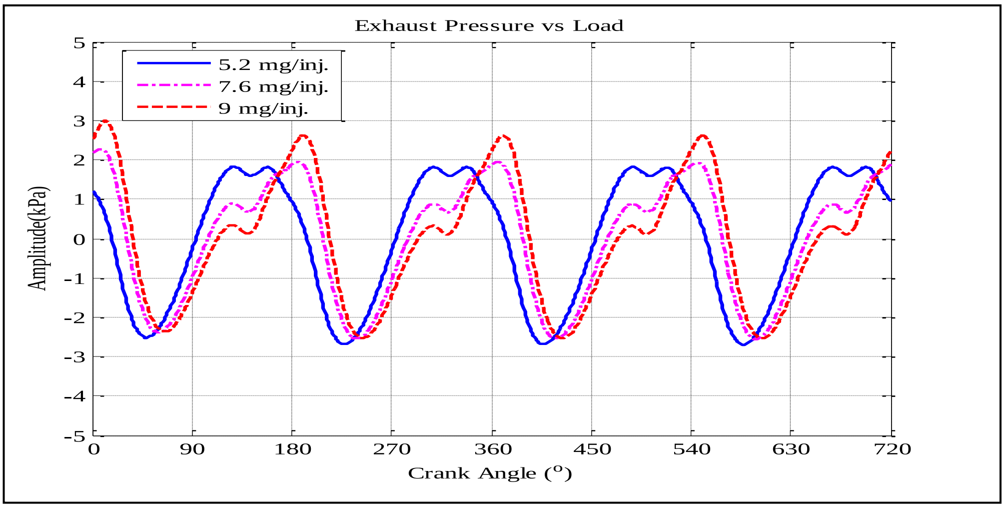

Figure 5. Predicted exhaust pressure as function of injected fuel (load)

6. Fuel Injector Fault Detection

The fault diagnosis was conducted on abnormal fuel injector faults which is one of the most common combustion faults in diesel engines. By comparing the predicted exhaust pressure behaviors with those of measured at when the engines running with faulty injector. By varying the amount of fuel injected into cylinder 1, the severity of an injector fault was simulated. The variation of the fuel quantity is from 7.6 mg/injection (for a normal injector) to 8.0 and 8.4 mg/injection in low load conditions and from 9.0 to 9.5 and 9.9 mg/injection for high load conditions. However, the amount of fuel injected into other cylinders remains the same.

Figure 6 shows the measured waveforms from a faulty injector installed in cylinder1. The fault was seeded by setting the injection break pressure from 250bar to 190bar. Because of the low injection break pressure, more fuel is injected into the cylinder. The three waveforms corresponds three different load conditions for all the cylinders. By comparing the waveforms between different cylinders it is seen that the main peak from cylinder 1 is clearly high than that of others. Because of this non-uniformity, it can be concluded easily that the fault is in cylinder 1.

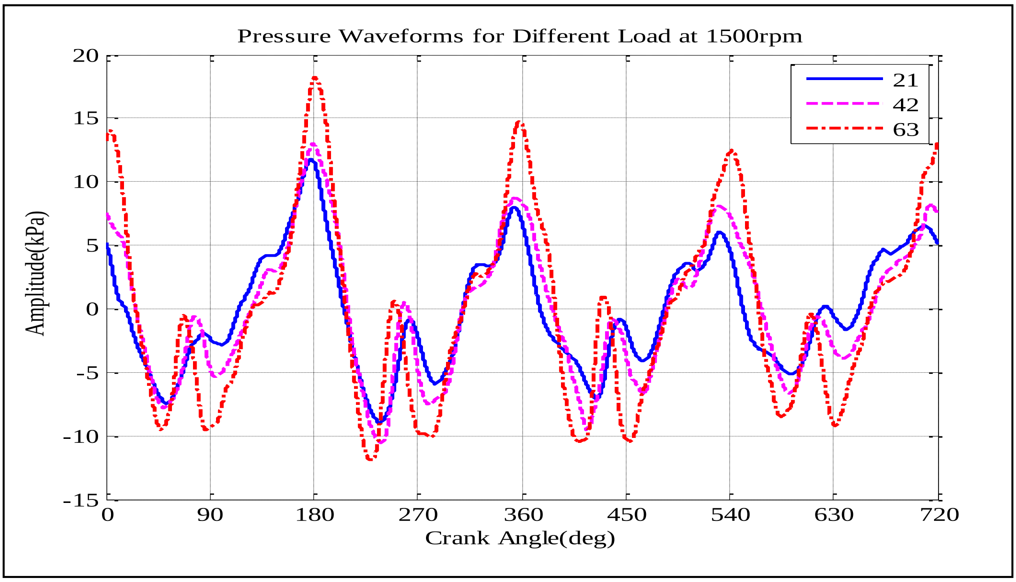

Figure 6. Measured exhaust pressure with injection faults at cylinder 1

Figure 7 shows the predicted exhaust pressure waveforms with cylinder 1 faulty injector. Compared with the normal fuel injection, the two higher fuel quantities result in the exhaust pressure amplitude to increase at the major peak while the amplitude at the secondary peak remains the same. Based on the change in the amplitudes it is easy to identify the fault is from the cylinder 1 because its wave shape is clearly different from the others.

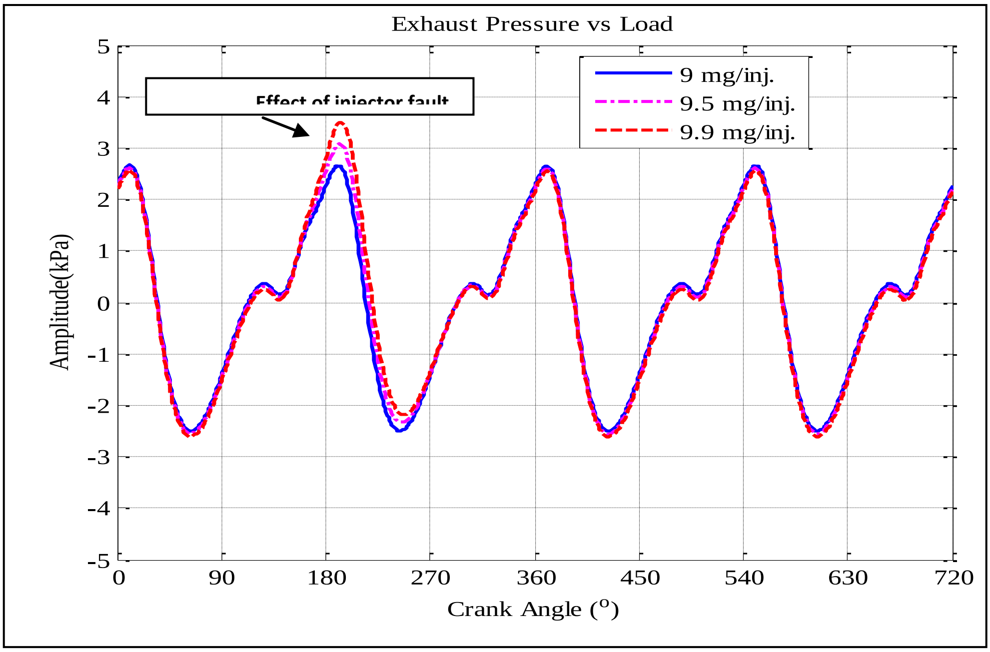

Figure 7. Predicted exhaust pressure with injector fault at cylinder 1 at high load

Figure 8 shows the effect of reducing the amount of fuel from 9.0 to 7.6 mg/injector, on the predicted exhaust pressure waveforms with cylinder 1 faulty injector. Compared with measured pressure waveform, there are some changes around the secondary peak of the waveforms. This may be caused by the variation of speed and load settings during tests and the influences of exhaust pipeline.

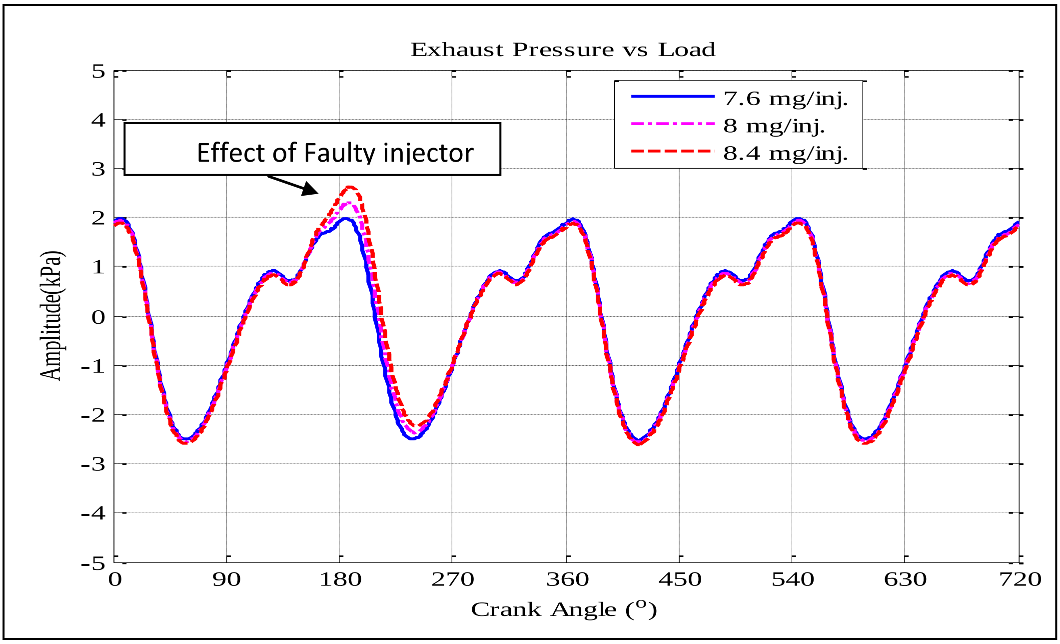

Figure 8. Predicted exhaust pressure with injector fault at cylinder 1 at low load

7. Conclusion

An attempt has been made to develop a simple mathematical engine model based on the first law of thermodynamics, capable of simulating the four stages of the diesel engine cycle and to predict the exhaust pressure for injector faults. The model has been tested against the experimental measurements for a number of engine operating conditions. From the model results there is a strong correlation between the amounts of fuel injected to cylinder and the both heat release and predicted cylinder pressure. Furthermore, the exhaust pressure waveform shows close agreement between modeled and measured pressure waves The variation of the exhaust pressure peak with deviation of the amount of fuel injected into the cylinder from the healthy condition proves confirms the possible application of the model for fuel injector fault detection. Injector faults such as low opening pressure, wear or clogged injector nozzle which led to excess fuel can be clearly detected and diagnosed.

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