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Modeling and simulation of the thermodynamic cycle of the Diesel Engine using Neural Networks

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Abstract: In this paper, a unique single zone combustion model is proposed to predict Diesel engine's performance, pressure, and temperature based on the conservation of mass and energy. In order to simulate all phases of combustion, the proposed model takes in consideration the dynamics of the intake and exhaust gas through the valves, the ignition delay, the instantaneous change in gas properties, the properties of the burned fuel, and the heat losses by the walls. Validation of this model has been realized by experimental data. Important issue has been recognized that the physical model takes too much time in calculation. For this purpose, a Feed-Forward Neural Network (FFNN) model is developed and validated experimentally to predict the pressure and temperature in the cylinder in nominal and faulty operations. Finally, the influence of some possible faults that may be produced on the diesel engine cycle during the operation has been analyzed.

Keywords: Diesel Engine, Thermodynamic cycle, Artificial Neural Network Modeling, Faulty operation mode.

1. INTRODUCTION

Diesel engine is a very complicated mechanical system. It is characterized by the ruggedness in construction, simplicity in operation and ease of maintenance. It has become quite popular in transportation and agriculture sector, because of higher efficiency and longer operational time. In this paper, we present a physical model of thermodynamic cycle of diesel engine. During the engine cycle, gas temperature, pressure and mass can be found by applying the conservation of energy and mass (Xin, 2013) and by the application of the ideal gas equations in the cylinder. Initial condition of pressure, temperature and mass can be estimated from the intake manifold parameters, and the computation is conducted on a crank angle basis. A zero-dimensional model combined with a single zone model is used in this work, in order to have a fast and accurate analysis of the engine performance (Basbous et al., 2012). The model is then implemented in Matlab/Simulink in a complete diesel engine simulator proposed by (Nahim et al., 2015a), and then the global model is validated experimentally. But due to the fact that the simulation time of one engine cycle is too long, we transformed the physical model of thermodynamic cycle into FFNN model and we validated it experimentally.

The neural networks are used in the field of diesel engine. Several studies on the control of turbocharged diesel engine using neural networks are presented by (Omran et al., 2008) and (Dovifaaz, 2002). (Michael, 2013) presents a study on the estimation of NOx emissions Using Artificial Neural Networks. (Ghobadian et al., 2009)presents a work on the diesel engine performance and the exhaust emission analysis using an artificial neural network. In this work we are interested to create a neural network model that simulates the combustion cycle operation in normal and faulty conditions. Thus, the FFNN are trained using the experimental data. Network architecture and learning rate parameters are optimized using Back-propagation algorithm and Levenberg-Marquardt algorithm to optimize the performance function (Hagan et al., 2014). The main contribution of this paper can be resumed by the development of neural network model that estimates the variation of temperature and pressure in the cylinder in normal and faulty operation mode. In our previous works (Nahim et al., 2015a), we have developed a diesel engine simulator that simulates the engine operation in presence of failure (Nahim et al., 2015b). The developed model can be used to simulate the influence of the fault on the global system, and it can be used in real time simulation to detect any failure in the engine.

2. COMBUSTION MODEL

The purpose of a zero-dimensional, single zone combustion model developed in this work is to simulate the thermodynamic cycle from the beginning of the intake phase to the end of the exhaust phase. The model is derived based on the conservation of mass and energy (Awad et al., 2013)(Verhelst and Sheppard, 2009).

2.1 Modelling assumptions

This section gives a description of the main hypotheses on which the thermodynamic model is based, such as the energy and mass balances and the elements required to calculate these balances (Xin, 2013) such as:

1. Single zone model.
2. Zero-dimensional state parameters.

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3. State equations of Ideal gases

$$P.V = m.r.T \qquad (1)$$

4. The quasi-steady-state process for the gas flowing into and out of the cylinder.
5. Ignoring the kinetic energy of the intake and exhaust gas.
6. The fuel is injected in the combustion chamber at a constant temperature and it is immediately burned following Wiebe law (Stone, 2012).
7. The heat transfer occurs through the five boundary limits (cylinder head, piston, cylinder wall, exhaust valve and admission valve) that are at a constant and uniform temperature.

2.2 Model equations

The equation of the conservation of energy applied in the cylinder can be expressed as (Stone, 2012)(Xin, 2013):

$$\frac{dU}{d\phi} = \frac{dW}{d\phi} + \sum_i \frac{dQ_i}{d\phi} + \sum_j h_j \frac{dm_j}{d\phi} \qquad (2)$$

Where ϕ is the crank angle (degrees), U is the internal energy of the gas in the cylinder, W is the mechanical work acting on the piston, Q_i is the heat exchanged through the system boundary and fuel combustion, and $h_j.m_j$ is the energy brought into and out of the cylinder by the intake and exhaust gas flow. Each term of the equation (2) is further expressed as:

$$\frac{dU}{d\phi} = \frac{d(m.u)}{d\phi} = u.\frac{dm}{d\phi} + m.\frac{du}{d\phi} \qquad (3)$$

$$\frac{dW}{d\phi} = -P.\frac{dV}{d\phi} \qquad (4)$$

$$\sum_i \frac{dQ_i}{d\phi} = \frac{dQ_{fuel}}{d\phi} + \frac{dQ_{wall}}{d\phi} \qquad (5)$$

Where V is the cylinder instantaneous volume, m is the mass of the gas in the cylinder, Q_{fuel} is the heat energy released from fuel combustion, and Q_{wall} is the heat transfer through the walls of the cylinder head, the piston and the liner.
The energy of the intake and exhaust gas exchange mass flow is given by:

$$\sum_j h_j.\frac{dm_j}{d\phi} = h_{in}.\frac{dm_{in}}{d\phi} + h_{ex}.\frac{dm_{ex}}{d\phi} \qquad (6)$$

Where m_{in} is the intake gas mass flowing into the cylinder, m_{ex} is the exhaust gas flowing out of the cylinder, h_{in} and h_{ex} are the specific enthalpies of the gases at the intake and exhaust valves.
Since the specific internal energy for ideal gases can be expressed as: $u = u(T, \alpha)$ where α is the excess air- fuel ratio. The following relationship is obtained:

$$\frac{du}{d\phi} = \frac{\partial u}{\partial T}.\frac{dT}{d\phi} + \frac{\partial u}{\partial \alpha}.\frac{d\alpha}{d\phi} = C_v.\frac{dT}{d\phi} + \frac{\partial u}{\partial \alpha}.\frac{d\alpha}{d\phi} \qquad (7)$$

Substituting these relationships into equation (2), the energy conservation equation becomes:

$$\frac{dT}{d\phi} = \frac{1}{m.C_v}.\left(\begin{array}{l} \frac{dQ_{fuel}}{d\phi} + \frac{dQ_{wall}}{d\phi} - P.\frac{dV}{d\phi} + h_{in}.\frac{dm_{in}}{d\phi} \\ + h_{ex}.\frac{dm_{ex}}{d\phi} - u.\frac{dm}{d\phi} - m.\frac{\partial u}{\partial \alpha}.\frac{d\alpha}{d\phi} \end{array} \right) \qquad (8)$$

The mass conservation equation can be expressed as:

$$\frac{dm}{d\phi} = \frac{dm_{in}}{d\phi} + \frac{dm_{ex}}{d\phi} + \frac{dm_{fuelB}}{d\phi} \qquad (9)$$

Where m_{fuelB} is the fuel mass injected into the cylinder.

2.3 Sub-models
2.3.1 Heat transfer through cylinder walls
The heat transfer terms in the conservation of energy can be derived as follows (Woschni, 1967):

$$\frac{dQ_{wall}}{d\phi} = \sum_i \frac{dQ_{wall,i}}{d\phi} = \frac{-1}{6.N_E}.\sum_i \alpha_g.A_{wall,i}.(T - T_{wall,i}) \quad (10)$$

Where α_g is the instantaneous spatial-average heat transfer coefficient from the cylinder gas to the inner cylinder wall, N_E is the engine speed (rpm), A_{wall} is the heat transfer area, T_{wall} is the spatial-average temperature of the cylinder wall surface.
α_g is critical for heat transfer calculation. In 1967 Woschni has developed a formula that describe heat transfer coefficient.

2.3.2 Mass transfer model

The intake mass flow rate entering the cylinder is given by the Baré Saint-Venant model (Basbous et al., 2012):

$$\frac{dm_{in}}{d\phi} = \frac{C_{f,in}.A_{in}.P_{in}}{6.N_E.\sqrt{R_{in}.T_{in}}}.\sqrt{\frac{2.\gamma}{\gamma - 1}\left[\left(\frac{P}{P_{in}}\right)^{\frac{2}{\gamma}} - \left(\frac{P}{P_{in}}\right)^{\frac{\gamma+1}{\gamma}}\right]} \qquad (11)$$

Where N_E is the engine speed (rpm), $C_{f,in}$ is the intake valve flow coefficient, P_{in} and T_{in} are the pressure and temperature in the intake port just before the intake valve, γ is the ratio of specific heat capacities of the intake gas flow.
The exhaust gas mass flow rate out of the cylinder is given below, when:

$$\frac{P_{ex}}{P} > \left(\frac{2}{\gamma + 1}\right)^{\frac{\gamma}{\gamma - 1}} \qquad (12)$$

The valve flow is subsonic and can be described as: