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Modelling and Simulation of the Turbocharged Diesel Engine with Intercooler

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Abstract: There are many approaches towards diesel engine modelling. Their accuracy and complexity varies at a great extent. In this paper we developed zero dimensional model that will accurately describe the processes taking place in the multi-cylinder of turbo charged diesel engine. Application of programming software Matlab in given model will be used for computer simulation and calculation of engine parameters which are based in the concept of technology of engine downsizing. In paper are presented simulation results for the temperatures and pressures in the cylinder and manifolds, mass flow rates of gas exchange, heat transfer rate for engine speeds 1200 rpm and 1800 rpm as functions of crank angle for Cummins engine type N-14.

Keywords: Downsizing engine, turbo charging, intercooler, mass flow rate, valve, manifolds, simulation.

1. INTRODUCTION

Driven by a demand for better fuel economy and increasingly stringent emission regulations over a wide range of customers and applications, engine manufacturers have turned towards engine downsizing as the most potent enabler to meet these requirements. Key elements in this concept are fuel system and turbo charging system for engine.

The turbocharger is an inherent downsizing technology regardless of the type that is used in practice, as it increases boost levels above ambient that forces more power out of a given size of engine. The increased specific maximum power due to engine downsizing is made possible by the appropriate redesigning of the air system (turbocharger, intercooler), combustion chamber (piston and cylinder head) and fuel - injection system.

Development of mathematical models and computer simulation is made fot recognition of the working processes that are developed in the engine. By using mathematical modelling of the thermodynamic processes it can achieve large advantages compared with experimental ones, such as:

- Prediction of work processes during the large and small constructive changes (valve timing diagram, valve-lift profiles, effective flow area as a function of intake and exhaust valve lift, cross-section and length of the intake and exhaust manifold, turbine and compressor characteristics, etc.) without building the expensive real engine by a large number of variants,
- Determining engine parameters that cannot be acquired by experimental way, e.g. instantaneous temperature of the gas in the cylinder,
- Prediction possibility of the engine process over very small change which is impossible to be carried out experimentally due to measurement errors,
- Possibility of using the optimization method in order to obtain optimal constructive and process parameters.

According to author Jankovic (1984), the calculation of working cycle of a real engine was first implemented by author Kuhler, to continue after by authors Justi and Lüder. With the development of engines there is a need for continuous calculation of working cycle and investigation of the impact of more parameters.

Working processes that are developed in the turbocharged diesel engine with intercooler are very complex. For that reason there are three basic methods developed for calculation of engine cycle.

The first method of calculation is stationary method which includes the processes that takes place in the valves. The flow is defined by their geometry and discharge coefficient, usually by determining empirically the valve effective flow area. As a representative of this method are: Hasselgruber (1961), Orlin & Kruglov (1968), etc.

Second method is quasi steady-state and is developed by Jankovic (1984), Heywood, J.B. (1988), Dobovišek et al (1986). This method takes in consideration processes that occur in the reciprocator cylinder, valves, and intake & exhaust manifold. The condition of gases (pressure and temperature) inside the manifold in the specific time is constant. The third method of calculation of gas exchange processes takes into account dynamic phenomena in the intake & exhaust manifold. This method is developed by Song Zhu (1991) and Wong Hai at all (2006).

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In this paper is developed mathematical model based in second method - quasi steady-state, Lajqi (2006). The application of programming software Matlab in given model will be adapted for computer simulation and calculation of the engine parameters during redesigning of the air system (turbocharger, intercooler, volume intake manifold), volume exhaust manifold, fuel injection rate, etc.

Contributions of this work in selected topics of Conference are: *Models & Simulation* as well as *Cost Effective Operation and Maintenance*.

2. MODELLING TURBOCHARGED DIESEL ENGINE WITH INTERCOOLER

For modelling simplification, used engine is divided into: reciprocator engine and its components compressor, intercooler, intake manifold, turbine, exhaust manifold (Fig. 1).

Fig. 1 shows schematic diagram of a turbocharged diesel engine with intercooler. Turbine powered by exhausted air drives the compressor to compress fresh air. The compressed air cooled by intercooled is fed into intake manifold which can maintain a certain pressure. Fuel and air burns in the cylinder to push piston and produces torque. The exhaust gas is cleared out of the cylinder after combustion is completed which makes the temperature of the exhausted manifold to rise.

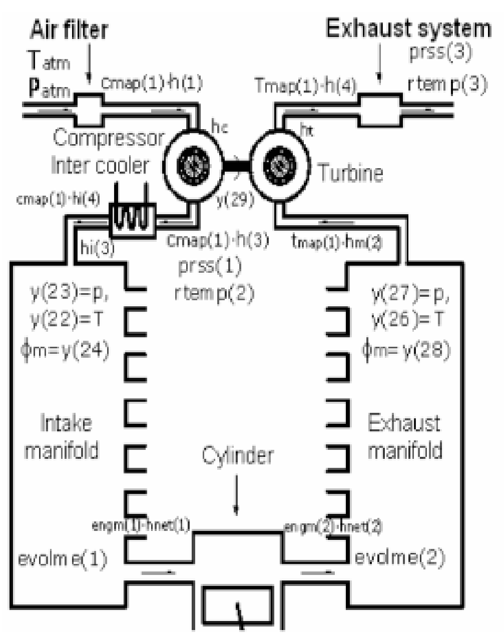

Fig. 1. Overall model structure turbocharged diesel engine with intercooler.

In general, this engine is an open thermodynamic system. It is also known as system with variable mass and contents of gases. With the help of the model of reciprocator, engines parameters can be calculated in one master cylinder of a multi-cylinder engine, while the manifold and the other components of the model are not separated during the suitable model of the multi cylinder engine. Interaction between the master cylinder and other components is calculated within manifolds. Effects of other cylinders depend on the appropriate phase angle.

This system is open to the transfer of mass, enthalpy, and energy in form of work and heat transfer. Law for conservation of mass is used to develop differential equations for the rate of change of the total mass flow and fuel, Heywood (1986):

$$\dot{m} = \sum_j \dot{m}_j ; \qquad \dot{m}_f = \sum_j \dot{m}_{f,j} \tag{1}$$

The rate of change of energy based on law for conservation of energy is:

$$\dot{E} = \sum_j \dot{m}_j \cdot h_j - \dot{Q}_W - \dot{W} \tag{2}$$

$$\dot{E} = \frac{d}{dt}(m \cdot h) - \frac{d}{dt}(pV)$$

Expression for the enthalpy (h) and density (ρ) for the mixture of air and combustion products, are:

$$h = h(T, p, \Phi) \tag{3}$$

$$\rho = \rho(T, p, \Phi) \tag{4}$$

Hence, the rate of change of enthalpy and density with respect to crank-angle are:

$$\frac{dh}{d\varphi} = (\frac{\partial h}{\partial T})_{p,\Phi} \cdot \frac{dT}{d\varphi} + (\frac{\partial h}{\partial p})_{T,\Phi} \cdot \frac{dp}{d\varphi} + (\frac{\partial h}{\partial \Phi})_{T,p} \cdot \frac{d\Phi}{d\varphi} \tag{5}$$

$$\frac{d\rho}{d\varphi} = \left(\frac{\partial \rho}{\partial T}\right)_{p,\Phi} \cdot \frac{dT}{d\varphi} + \left(\frac{\partial \rho}{\partial p}\right)_{T,\Phi} \cdot \frac{dp}{d\varphi} + \left(\frac{\partial \rho}{\partial \Phi}\right)_{T,p} \cdot \frac{d\Phi}{d\varphi} \tag{6}$$

The equation of state for ideal gases:

$$p = R \cdot \rho \cdot T \tag{7}$$

From the differential form of the equation of state (7) can be expressed the rate of change pressure with respect to crank-angle:

$$\frac{dp}{d\varphi} = \frac{\rho}{\partial \rho / \partial p} \cdot \left(-\frac{\dot{V}}{V} - \frac{1}{\rho}\frac{\partial \rho}{\partial T} \cdot - \frac{1}{\rho} \cdot \frac{\partial \rho}{\partial \Phi} \cdot \dot{\Phi} + \frac{\dot{m}}{m} \right) \tag{8}$$

Equation for the rate of change of temperature is:

$$\frac{dT}{d\varphi} = \frac{\frac{1}{\partial \rho / \partial P} \cdot (1 - \rho \cdot C_T)}{C_P + \frac{\partial \rho / \partial T}{\partial \rho / \partial P}(\frac{1}{\rho} - C_T)} \cdot \left(\begin{array}{l} \frac{\dot{m}}{m}\left(1 - \frac{h}{\frac{1}{\partial \rho / \partial P} \cdot (1 - \rho \cdot C_T)}\right) - \frac{\dot{V}}{V} - \frac{C_\Phi + \frac{\partial \rho / \partial \Phi}{\partial \rho / \partial P}(\frac{1}{\rho} - C_T)}{\frac{1}{\partial \rho / \partial P} \cdot (1 - \rho \cdot C_T)} \cdot \dot{\Phi} + \\ \frac{1}{\frac{1}{\partial \rho / \partial P} \cdot (1 - \rho \cdot C_T) \cdot m}(\sum_j \dot{m}_j \cdot h_j - \dot{Q}_W) \end{array} \right) \tag{9}$$