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# ANALYSIS OF ENGINE SPEED EFFECT ON THE FOUR - STROKE GDI ENGINE PERFORMANCE

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***Abstract:*** *Application of simulation codes on modeling of thermodynamic cycle of ICE has become more important during last years. The computational simulation model development is use for commercialization of computational fluid dynamics software for development of internal combustion engines performance simulation. Advances in physical models, numerical methods and computational power together have brought large-eddy simulations (LES) to the point where it warrants serious consideration for computing in-cylinder turbulent flows. Recently, a simulation tools, which models the different subsystem in an integrated manner has been developed. Analysis of the results of the model make possible investigation of volumetric efficiency, brake efficiency, burning rate, temperature and cylinder pressure value. The numerical analysis and calculations demonstrates that the numerical computational fluid dynamics (CFD) tools, at their stage of development, can help the engine designers to obtain more complete information on direct injection (DI) gasoline engine. For this investigation, a four-cylinder DI gasoline engine was tested under different engine speeds. The present study focuses on the effect of engine speed on engine performance.*

***Key words:*** *Computational simulation, Engine performance, Gasoline engine, 1D CFD, Spark ignition engine.*

## 1. INTRODUCTION

The gasoline direct injection engine performance theory to link together with computer modeling of the engine thermodynamics in engine simulations are great challenge, as the latter make the most complete use of the former and the use models is becoming widespread. Engine modeling is a very large subject, in part because of the range of engine configurations possible and the variety of alternative analytical techniques or sub-models, which can be applied in overall engine models Engine modeling is a fruitful research area and as a result many research laboratories have produced their own engine thermodynamics models of varying degrees of complexity, scope and ease to use [1 and 2].

In the last decades, the legislation on internal combustion engines (ICEs) has severely reduced the limits for pollutant and noise emissions. These requirements have established the research activity at design phase as a key stage in the engine production process. Therefore, an intensive investigation on ICEs has been carried out, focusing on the optimization of performances and fuel consumption. In particular, an important effort has been done seeking the improvement of the combustion and gas exchange processes, using tools such as Computational Fluid Dynamics (CFD).

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CFD simulations allow researchers to understand flow behavior and quantify important flow parameters such as mass flow rates or pressure drops, provided that the CFD tools had been properly validated against experimental results. For reasons such as the aforementioned, CFD simulations have become a valuable tool in helping both the analysis and design of the intake and exhaust systems of an ICE.

Simulating an intake or exhaust system is just a great exponent of such sort of problems. These systems are mainly composed of ducts, which can be accurately simulated by means of one-dimensional, non-viscous codes. However, there are several components that manifest a complex three-dimensional flow behavior, such as turbomachinery or manifolds, therefore being unable to be simulated properly by 1D codes, and thus requiring viscous, 3D codes.

Hence, it is a right choice to save computational time by simulating the complex components by means of a 3D code and modeling with a 1D code the rest of the system, i.e. the ducts. In this way, a coupling methodology between the 1D and the 3D code in the respective interfaces is required, being the objective of numerous authors [3–6].

## 2. INFORMATION

### 2.1. Fluid dynamics governing equations

The flow model involves the solution of the Navier-Stokes equations, namely the conservation of continuity, momentum and energy equation. These equations are solved in one dimension, which means that all quantities are averages across the flow direction. There are two choices of time integration methods, which affect the solution variables and limits on time steps. The time integration methods include an explicit and an implicit integrator. The primary solution variables in the explicit method are mass flow, pressure and total enthalpy.

In broad terms, a model is created using two types of discritization. Firstly, the complete powertrain system is grouped into general components. These components consist of air cleaners, valves, piping, valves, fuel injectors, mufflers, resonators, catalytic converters, combustion chambers and resonators. The second aspect is separating each component into multiple control volumes. Each control volume is bounded by another control volume or wall. By discritizing the system into sufficiently small volumes, the properties of the fluid in that volume can be assumed to be constant. The scalar variables (pressure, temperature, density, internal energy, enthalpy, species concentration, etc.) are assumed to be uniform over each volume. The vector variables (mass flux, velocity, mass fraction fluxes, etc.) are calculated for each boundary. These types of discretization is referred to as a “started grid".

Figure 1 illustrates the difference between vector and scalar properties.

The conservation equation (1), energy equation (2), enthalpy equation (3) and momentum equation (4) are shown below.

$$\frac{dm}{dt} = \sum_{boundaries} \dot{m} . \quad (1)$$

$$\frac{d(me)}{dt} = -p\frac{dV}{dt} + \sum(\dot{m}H) - hA_s(T_f - T_w) . \quad (2)$$

$$\frac{d(\rho HV)}{dt} = \sum(\dot{m}H) + V\frac{dp}{dt} - hA_s(T_f - T_w) . \quad (3)$$

$$\frac{d\dot{m}}{dt} = \frac{dpA + \sum\limits_{boundaries}(\dot{m}u) - 4C_f\frac{\rho u|u|}{2}\frac{dxA}{D} - C_p\left(\frac{1}{2}\rho u|u|\right)A}{dx} \quad (4)$$

where, $\dot{m}$ – boundary mass flux into volume, $m$ – mass of the volume, $V$ – volume, $p$ –pressure, $\rho$ – density, $A$ – flow area (cross-sectional), $A_s$ heat transfer surface area, $e$ – total internal energy (internal energy plus kinetic energy) per unit mass, $H$ – total enthalpy, $h$ – heat transfer coefficient, $T_f$ – fluid temperature, $T_w$ – wall temperature, $u$ – velocity at the boundary, $C_f$ – skin friction coefficient, $C_p$ – pressure loss coefficient, $D$ – equivalent diameter, $dx$ – length of mass element in the flow direction (discretization length), $dp$ – pressure differential acting across $dx$

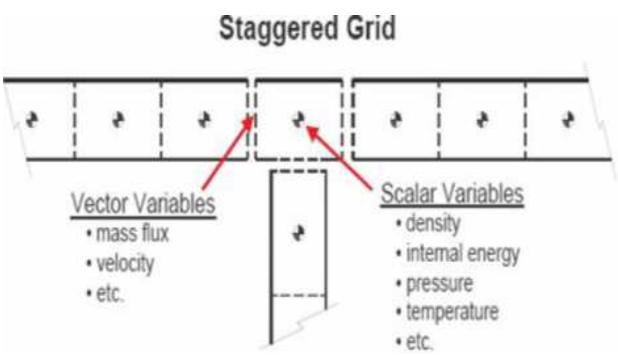


**Fig.1 .** Illustration of Component Discritization [1].

*Table 1*

**Table 1. Specification of the engine**

| Engine Parameters | Value |
|---|---|
| Bore (mm) | 86.0 |
| Stroke (mm) | 86.0 |
| Displacement (cc) | 2000 |
| Number of cylinder | 4 |
| Compression ratio | 9.5 |
| Connecting rod length (mm) | 175 |
| Piston pin offset (mm) | 0 |
| Intake valve open (0CA) | 361 |
| Intake valve close (0CA) | –98 |
| Exhaust valve open (0CA) | 131 |
| Exhaust valve close (0CA) | 384 |
| Brake power (KW) | 63.8 |
| Brake torque (Nm) | 135.3 |
| Model : Four Cylinder, Four-Stroke, Vertical, Air Cooling, Port Injection. | |

### 2.2. Main engine data

The development of the four cylinder modeling and simulation for four-stroke port-injection gasoline engine was presented in this paper. The specification of the selected gasoline engine model was presented in Table 1.

### 2.3. Result and disscusion

Speed mode is the most commonly used mode of engine simulation, especially for steady states cases. This method typically provides steady-state results very quickly because the speed of the engine is imposed from the start of the simulation, thus eliminating the relatively long period of time that a loaded engine requires for the crankshaft speed to reach steady-state.

Overall engine work can be determined by integrating the area under the pressure-volume diagram or *P–V* diagram. So many previous works concerned mainly prediction the pressure inside the combustion chamber [7–9]. But the pressure and volume are influenced by engine geometries during variation of crank angle. So the pressure and displacement volume are needed to convert as functions of crank angle. Kuo [8] and Kirkpatrick [10] proposed the method that can calculate the pressure and volume at any crank angle. The combustion process can be described by Wiebe function [11 and 12].

The running simulation result is all of the engine performance data with the different engine speed (rpm). This model was running on any speed from 1000 to 6000 rpm. In this paper the simulation result of engine performance are volumetric efficiency, brake efficiency, burning rate, indicated efficiency and temperature and pressure in cylinder.

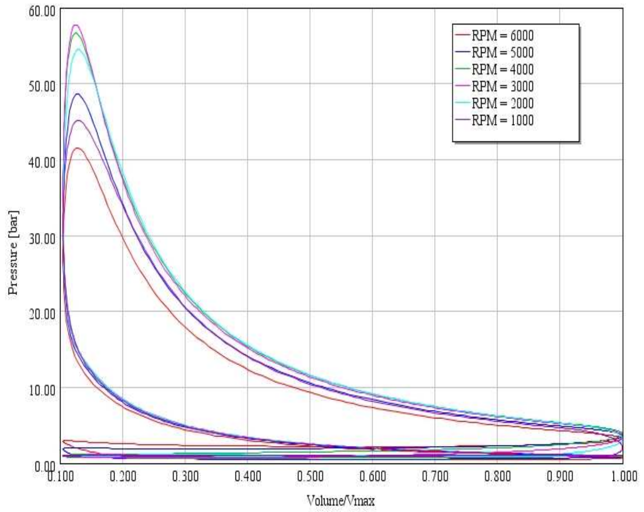


**Fig. 2.** Pressure-volume diagram of firing spark ignition engine.

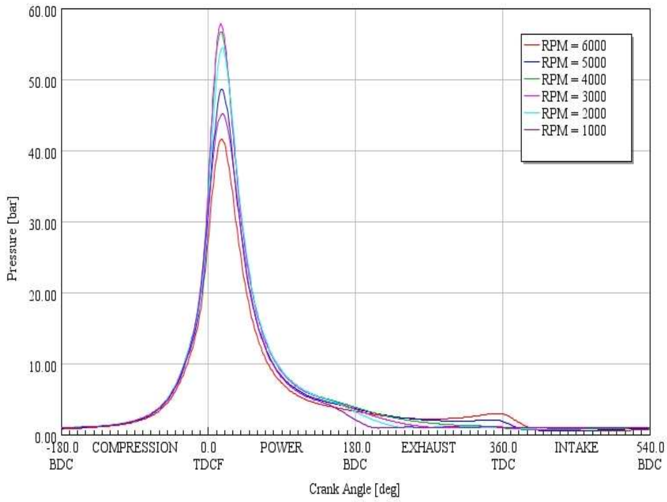


**Fig. 3.** Cylinder pressure as a function of crank angle for a spark ignition engine.

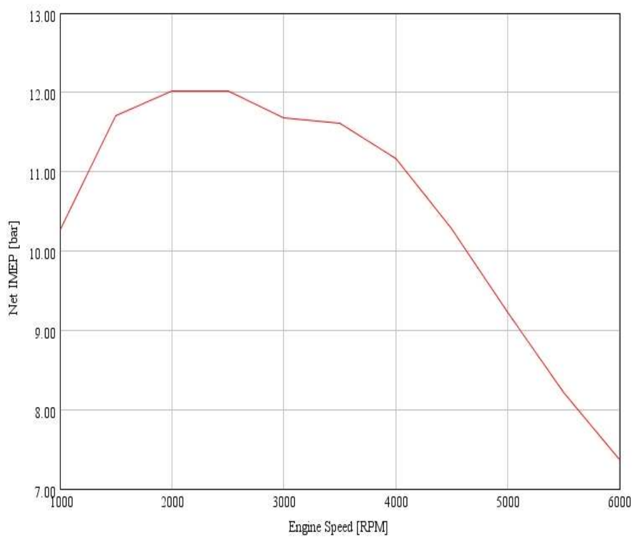


**Fig. 4.** Indicated mean effective pressure of engine model.

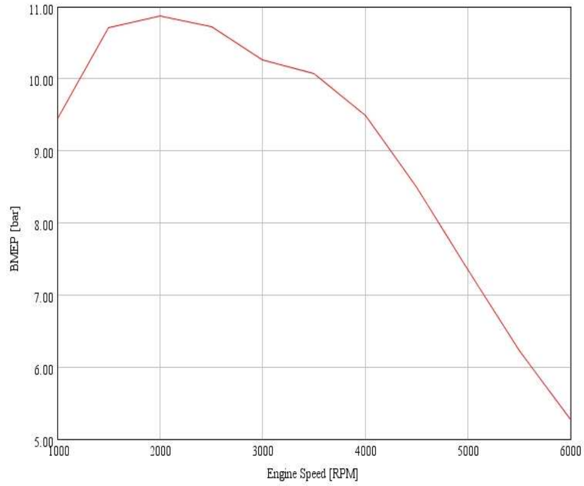


**Fig. 5.** Breake mean effective pressure of engine model.

A computational pressure-volume diagram for a spark-ignition engine operating at full load is shown in Fig. 2. The computational cylinder pressure as a function of crank angle for a spark ignition is shown in Fig. 3. The highest cylinder pressure is 57.76 bar at engine speed 3 000 rpm and the lowest of the cylinder pressure is 45.1 bar at engine speed 1 000 rpm.

Figures 3, 4 and 5 show the pressure in engine cylinder, indicated mean effective pressure and the brake mean effective pressure of the gasoline direct injection engine model simulation. Mean effective pressure is defined as that hypothetical constant pressure acting on the piston during its expansion stroke producing the same work output as that from the actual cycle. The constant depends on the mechanism used to get the indicator diagram and has the units is bar/m. The mean effective pressure is quite often used to calculate the performance of an internal combustion engine. If the work output is indicated output then it is called indicated mean effect pressure. The highest pressure of the engine cylinder pressure is in TDCF, because this step is need the highest pressure to combustion. The highest pressure is 57.76 bar after ignition at 14.316 crank angle degree. The highest indicated mean effective pressure and brake mean effective pressure is 12.0171 bar and 10.872 bar at both 2 000 rpm engine speed. Before the engine speed is 2 000 rpm the imep and bmep is low and increase until on 2 000 rpm. After the engine speed is over than 2 000 rpm the imep and bmep is go to down.

Pumping Mean Effective Pressure (MEP) performance of the gasoline engine is shown in Fig. 6. The highest pumping PMEP for gasoline engine is 0.0357 bar at 1000 rpm engine speed and minimum is 1.7497 bar at 6 000 rpm engine speed.

The computational temperature in cylinder as a function of crank angle for a spark ignition is shown in Fig. 7. The highest maximum computational temperature in cylinder for gasoline engine is 2851.40 K at 4 000 rpm engine speed and the lowest maximum computational temperature in cylinder is 2 323.61 K at engine speed 1 000 rpm.

The computational burned fuel fraction as a function of crank angle for a spark ignition is shown in Fig. 8.

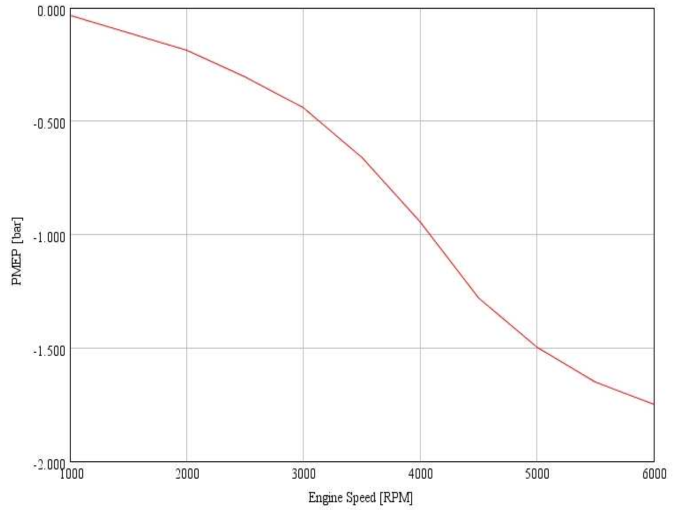


**Fig. 6.** Pumping mean effective pressure of engine model.

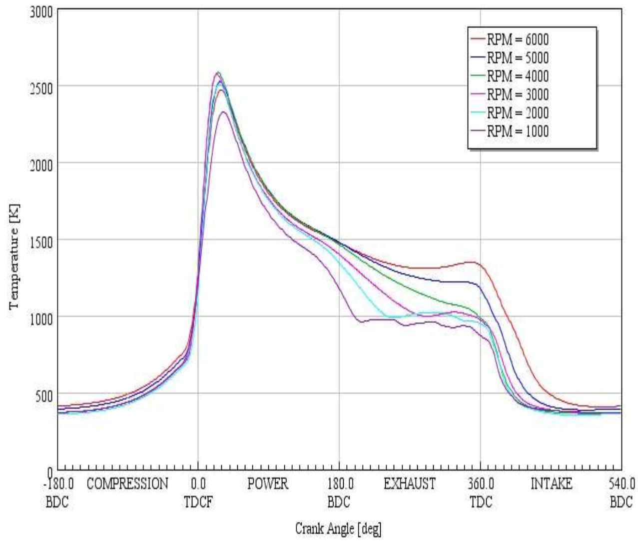


**Fig. 7.** In cylinder temperature of engine model.

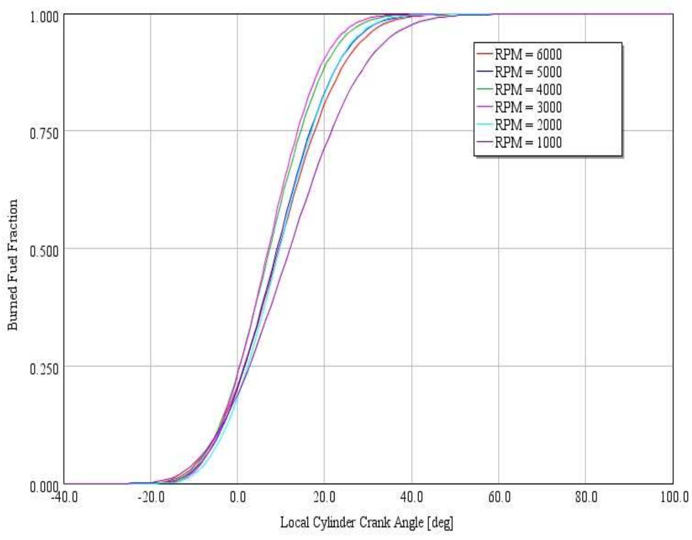


**Fig. 8.** Burned fuel fraction of engine model.

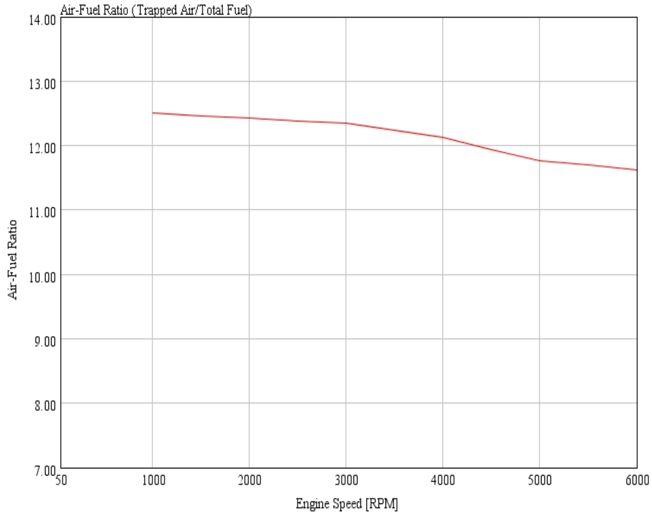


**Fig. 9.** Air-fuel ratio of engine model.

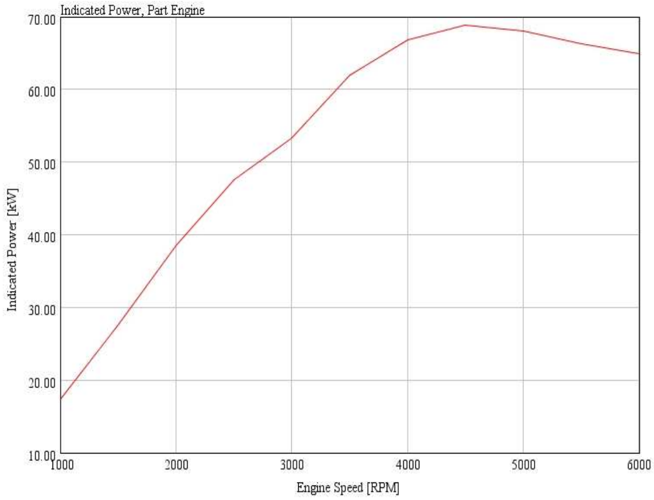


**Fig. 10.** Indicated power of engine model.

The air-fuel ratio of the engine model performance shown in Fig. 9 above, the air-fuel ratio is high in the engine speed is low and the air-fuel ratio is low in the engine speed is high. The highest of the air-fuel ratio is 12.499 on engine speed 1 000 rpm and the lowest of the air-fuel ratio is 11.6279 on engine speed 6 000 rpm. The trend of air-fuel ratio is decreased if the engine speed is increased.

Indicated power of an engine is tells about the health of the engine and also gives an indication regarding the conversion of chemical energy in the fuel into heat energy [11–13]. Indicated power is an important variable because it is the potential output of the cycle. Therefore, to justify the measurement of indicated power, it must be more accurate than motoring and other indirect methods of measuring friction power. For obtaining indicated power the cycle pressure must be determined as a function of cylinder volume. It may be noted that it is of no use to determine pressure accurately unless volume or crank angle can be accurately measured. In this model the engine indicated power performance on variation speed shown in Fig. 10. The performance of indicated power the engine model is with variation on engine speed. At engine speed 1 000 rpm the indicated power of engine is low, and if the engine speed increases over 1 000 rpm the indicated power arises until 4 500 rpm. When the engine speed is over than 4 500 rpm the indicated power decreases. The lowest engine indicated power model is 17.12 kW at minimum engine speed 1 000 rpm and the maximum indicated power of the engine model is 77.09 kW at engine speed 4 500 rpm.

The brake power of the engine model is shown in Fig. 11. Brake power is usually measured by attaching a power absorption device to the drive-shaft of the engine (any type of brake). The brake power of engine lowest is 16 kW at minimum engine speed 1000 rpm and after that if the engine speed is increased the brake power is increased too until engine speed 4 500 rpm. The maximum brake power of the engine model is 63.76 kW at engine speed 4 500 rpm and after that the brake power decreases.

The brake specific fuel consumption is shown in Fig. 12. The model simulation result shown that the minimum brake specific fuel consumption is 288.24 g/kWh at 2 000 rpm.

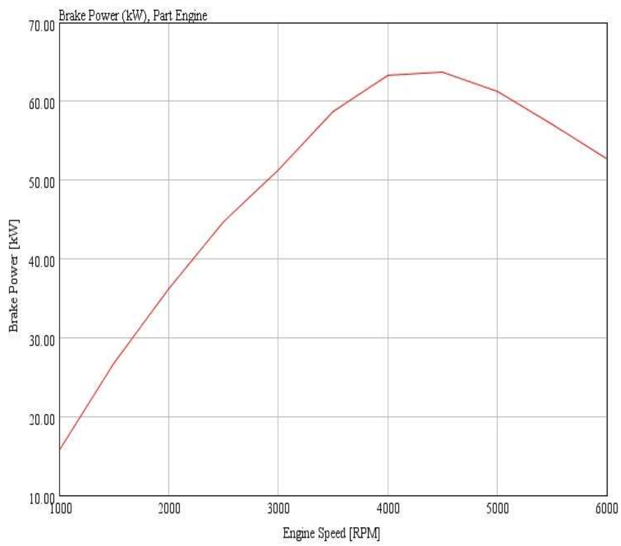


**Fig. 11.** Brake power of engine model.

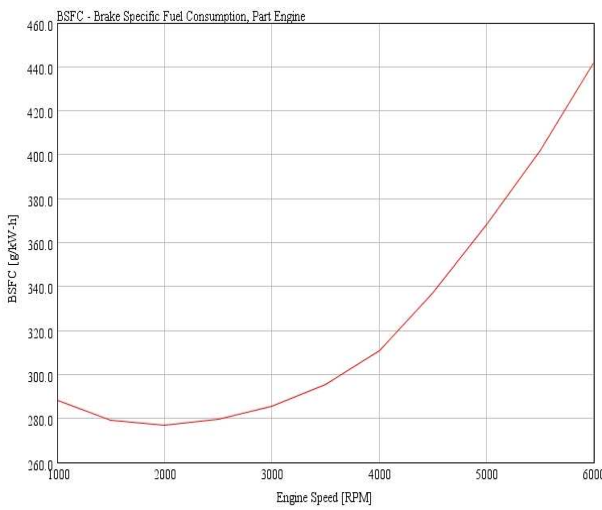


**Fig. 12.** Brake specific fuel consumption of engine model.

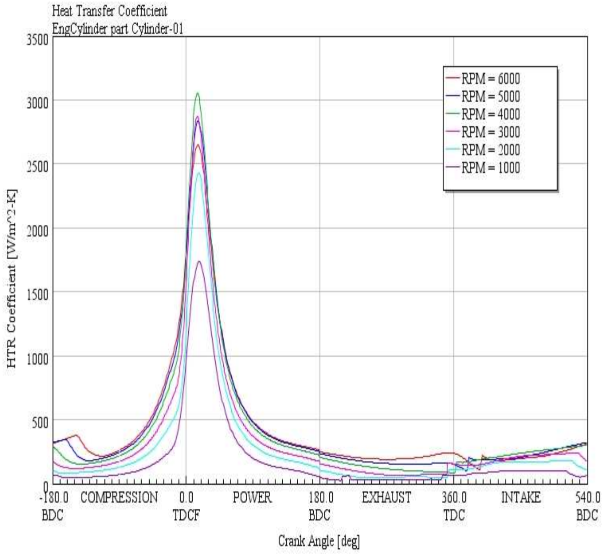


**Fig. 13.** Heat transfer coefficient of engine model.

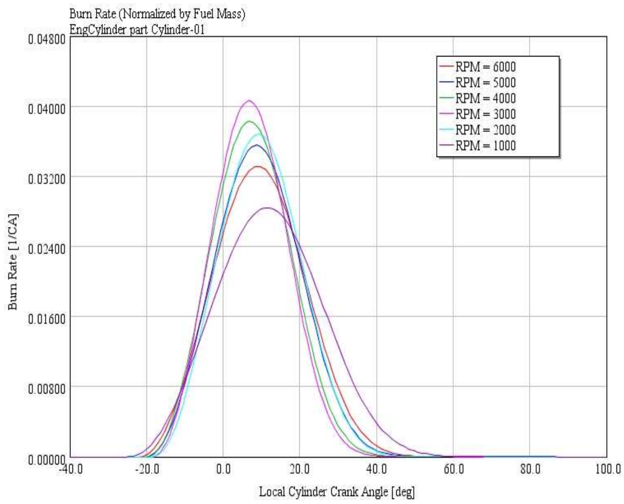


**Fig. 14.** Burn rate normalized by fuel mass.

The heat transfer coefficient is shown in Fig. 13. The model simulation result shown that the maximum heat transfer coefficient is 3 048.39 w/m$^2$K at 4 000 rpm.

Ignition is assumed to occur at a specified crank angle with the instantaneous burn of small specified fraction of the unburned gas. The user specifies the appropriate data such as wall heat transfer area behind the flame and the projected flame area, each one divided by the bore area, as function of the fraction of volume burned. This flame area, together with the evolving turbulence velocity and a specified laminar flame speed are used to determinate the burn rate.

The burn rate normalized by fuel mass is shown in Fig. 14. The model simulation result shown that the maximum burning rate is 0.041 1/CA at 3 000 rpm.

## 3. CONCLUSIONS

The theoretical analysis has been carried out using simulation model.

The simulation investigation results are shown that the highest cylinder pressure is 57.76 bar at engine speed 3 000 rpm and the lowest of the cylinder pressure is 45.1 bar at engine speed 1 000 rpm.

The investigation result is shown that the highest pumping MEP for gasoline engine is −0.0357 bar at 1 000 rpm engine speed and minimum is −1.7497 bar at 6 000 rpm engine speed.

The highest maximum computational temperature in cylinder for gasoline engine is 2851.40 K at 4 000 rpm engine speed and the lowest maximum computational temperature in cylinder is 2 323.61 K at engine speed 1 000 rpm.

At engine speed 2 000 rpm, the engine model showed the minimum brake specific fuel consumption.

On the 2 000 cc four cylinder four-stroke port-injection gasoline engine modeling showed that the highest brake power is 63.76 kW at 4 500 rpm and indicator power is 77.09 kW at 4 500 rpm speed.

## REFERENCES

[1] S. Iliev, H. Stanchev, *Simulation on four-stroke diesel engine and effect of engine performance*, Proceedings of the union of scientist-Ruse, Ruse 2012, pp. 68-73, 2012.

[2] S. Iliev, H. Stanchev, *Computer Simulation of a Model of an Internal Combustion Engine*, Automotive Engineering and Environment CAR2011, Pitesti, Romania, 2011.

[3] A. Onorati, G. Montenegro, G. D'Errico, *Prediction of the attenuation characteristics of IC engine silencers by 1-D and multi-D simulation models*, SAE Technical Paper Series 2006, Tech. Rep. 2006-01-1541, doi:10.4271/2006-01-1541, 2006.

[4] G. Montenegro, A. Onorati, *Modeling of silencers for IC engine intake and exhaust systems by means of an integrated 1D-multiD Approach*, SAE International Journal of Engines 1 (1), 466, doi:10.4271/2008-01-0677, 2009.

[5] G. Montenegro, A. Onorati, F. Piscaglia, G. D'Errico, *Integrated 1D-Multi-D Fluid Dynamic Models for the Simulation of ICE Intake and Exhaust Systems*, SAE Technical Paper Series 2007, Tech. Rep. 2007-01-0495, doi:10.4271/2007-01-0495, 2007.

[6] J. Galindo, A. Tiseira, P. Fajardo, R. Navarro, *Coupling methodology of 1D finite difference and 3D finite volume CFD codes based on the Method of Characteristics*, Elsevier, Mathematical and Computer Modelling, doi:10.1016/2010-11-078, 2010.

[7] L. Eriksson, I. Andersson, *An Analytic Model for Cylinder Pressure in a Four Stroke SI Engine*, SAE Technical Paper No.2002-01-0371, 2002.

[8] P. S. Kuo, *Cylinder Pressure in a Spark Ignition Engine: A Computational Model*, pp.141-145, J. Undergrad. Sci.3, 1996.

[9] P. Zeng, R. G. Prucka, Z. S. Filipi, *Reconstructing Cylinder Pressure of a Spark-Ignition Engine for Heat Transfer and Heat Release Analyses.* ASME Internal Combustion Engine Division Technical Conference, Paper No.ICEF2004-886, 2004.

[10] A. Kirkpatrick, *Internal Combustion Engine Thermodynamic Online*, available at: `http://www.engr.colostate.edu`, accessed: 2005-11-31.

[11] J.B. Heywood,. *Internal Combustion Engine Fundamentals,*. International edition, McGraw-Hill, Singapore, 1998.

[12] V. Ganesan, *Internal Combustion Engines*, Second Edition, Tata McGraw-Hill, New Delhi, 1999.

[13] R. Stone., *Introduction to Internal Combustion Engines* Second Edition, SAE Inc., USA, 1997.