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Steady-State and Transient Simulation of Gas Flow Pressure in Intake Port Engine

[1]Semin, [2]Abdul Rahim Ismail and [2]Rosli Abu Bakar
[1]Institute of Technology Sepuluh Nopember, Surabaya 60111, Indonesia
[2]Faculty of Mechanical Engineering, University Malaysia Pahang, 25000 Kuantan, Malaysia

Abstract: This study presents the gas flow pressure in the intake port of four-stroke direct-injection compression ignition engine using GT-Suite software for steady-state and transient simulation. To investigate and simulate the intake port gas flow pressure profile of compression ignition engine is using GT-Power engine model were developed in this study. GT-Power is sub-system menu from GT-Suite. The engine model is developed from the real compression ignition engine data and input to software library. In this research, the simulation of engine model is running in variations engine speeds. The simulation output data is collected from the GT-Post results plots and cases RLT in post processing. The simulation results of the intake port engine model are shown the characters in intake port pressure profile of engine in variations engine speeds. The detail performance intake port gas flow pressure is shown in graphs in this study.

Key words: Compression ignition engine, computational model, intake port, pressure, simulation

INTRODUCTION

The compression ignition 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 (Challen and Baranescu, 2003). Engine modeling is a fruitful research area and as a result many universities have produced their own engine thermodynamics models, of varying degrees of complexity, scope and ease to use. There are also now available a number of fairly comprehensive models which have a wider, more general purpose use with refined inputs and outputs to facilities their use by engineers other than their developers, most of these models had their origins in university-developed models. They include WAVE from USA, PROMO from Germany, TRANSEG/ICENG/MERLIN from UK, among others. A new code covering completes engine systems have emerged recently, GT-Suite (Challen and Baranescu, 2003).

In this study, the steady-state and transient of gas flow in intake port of engine is simulate using GT-Suite. This research is focuses on single cylinder four stroke direct injection compression ignition engine. The aim is to give an insight into the engine intake port gas flow thermodynamics performance using GT-Suite simulation model, how the engine model developed and how the components interact.

The direct injection compression ignition engines is internal combustion engines, where fuel is injected by the fuel injection system into the engine cylinder toward the end of the compression stroke, just before the desired start of combustion (Bakar and Semin, 2006a, b; Bakar *et al*., 2007a; Kowalewicz, 1984; Stone, 1997; Heywood, 1998; Ganesan, 1999; Challen and Baranescu, 2003). The liquid fuel, usually injected at high velocity as one or more jets through small orifices or nozzles in injector tip, atomizes into small drops and penetrates into the combustion chamber. The fuel vaporizes and mixes with high temperature and high pressure cylinder air. The air is supplied from intake port of engine. Since the air temperature and pressure are above the fuel's ignition point, spontaneous ignition of portions of the already-mixed fuel and after air a delay period of a few crank angle degrees. The cylinder pressure increases as combustion of the fuel-air mixture occurs (Atkinson and Andersson, 1999; Eriksson *et al*., 2002; Klein *et al*., 2002; Piedrahita *et al*., 2003; Sanders *et al*., 2003). The major problem in compression ignition engine combustion chamber design is achieving sufficiently rapid mixing between the injected fuel and the air from intake port in the cylinder to complete combustion in the appropriate crank angle interval close to top-center (Bakar and Semin, 2006a; Baker *et al*., 2007a; Blair, 1999; Challen and Baranescu, 2003; Heywood, 1998; Ganesan, 1999; Klein *et al*., 2002; Semin *et al*.,

Corresponding Author: Semin, Center for Graduate Studies, University Malaysia Pahang, Locked Bag 12, 25000 Kuantan, Pahang, Malaysia

2007a). Horsepower output of an engine can be dramatically improved through good intake port design and manufacture (Jawad *et al*., 2003).

To determine gas flow conditions right through the engine is the essence of modeling at small intervals time. Appropriate summation of these gas conditions over an engine cycle then leads to an estimate engine performance. Gas flow condition through the engine is basically meant pressures, temperatures, gas composition and mass or energy flows. The core of any model is the energy equation for each control volume in the engine (Challen and Baranescu, 2003). The first essential the engine performance model is the energy for a control volume, which is derived from First Law Thermodynamics and the Perfect Gas Law. The first law states that the rate of change of internal energy of the volume from connected gas flows, less any nett heat transfer out through the volume walls and less any work done by the control volume gas against its surroundings is shown in Eq. 1.

$$\frac{d(uM)}{dt} = \sum_{1} h_i \dot{m}_i - \frac{dQ}{dt} - P\left(\frac{dV}{dt}\right) \quad (1)$$

Where u is the specific internal energy per unit mass of gas in the control volume and there are 1 pipes connecting to the control volume, the flow in the ith pipe having specific enthalpy h_i and mass flow rate m_i (negative for outflow). Net heat transfer out of the control volume is dQ/dt, M is the mass of gas in the control volume at pressure P and this gas carrier out nett work on its surroundings of PdV/dt, where dV/dt is the control volume's current rate of change of volume (zero for manifold control volume and not for cylinder control volume).

To determine flow through a pipe and constriction is needed the orifice equation. The stagnation or total temperature T_t at any point in a flow is given by Eq. 2.

$$c_p T_t = c_p T + \frac{v^2}{2} \quad (2)$$

Where, T_t is the temperature that the gas flowing at velocity v with static temperature T would reach if it were brought to rest adiabtically, the equation is simply an energy balance, $c_p T$ being a measure of the static energy, $v^2/2$ the kinetic energy and $c_p T_t$ the total energy (enthalpy). And to determine the mach number Ma is using Eq. 3, where c is velocity of sound, γ is equal with cp/cv and cp-cv = R.

$$M_a = \frac{v}{c} = \frac{v}{\sqrt{(\gamma R T)}} \quad (3)$$

The total to static temperature relationship is obtained using Eq. 4 and 5. Then the total to static pressure equation is shown in Eq. 6.

$$\frac{T_t}{T} = 1 + \frac{v^2}{2c_p} = 1 + \frac{\gamma R T M_a{}^2}{2c_p} \quad (4)$$

$$\frac{T_t}{T} = 1 + (\gamma - 1)\frac{M_a{}^2}{2} \quad (5)$$

$$\frac{P_t}{P} = \left(\frac{T_t}{T}\right)^{\frac{\gamma}{\gamma-1}} = \left[1 + \left(\frac{\gamma - 1}{2}\right) M_a{}^2\right]^{\frac{\gamma}{\gamma-1}} \quad (6)$$

MATERIALS AND METHODS

The real engine data is used in four stroke single cylinder direct injection compression ignition engine model. The model of compression ignition engine has been developed using GT-POWER software based from real engine selected data. According to Bakar and Semin (2006a), Baker *et al.* (2007b, c), Ismail *et al.* (2007), Semin *et al.* (2007a), Baker and Semin (2007b-d) the specification of four stroke direct injection compression ignition engine and it intake port is shown in Table 1 and the physical illustration of intake port is shown in Fig. 1.

According to Bakar *et al.* (2007b, c) and Semin *et al.* (2007a, c) in the GT-POWER engine model development, a typical intake port is modeled using Import, engine is modeled using EngCylinder and EngineCrankTrain component objects, Valve*Conn and EngCylConn connection objects. Import is used to define the basic geometry and characteristics of intake port, EngCylinder and EngineCranktrain are used to define the basic geometry and characteristics of engine. These objects further refer to several reference objects for more detailed modeling information on such attributes as gas flow, combustion and heat transfer. Import must be connected to the engine cylinder with Valve*Conn, Cylinder must be connected to the engine with EngCylConn part made from the predefined object which available in the template library. While Pipe, EngCylConn parts have no user defined attributes, the global cylinder number for cylinder is assigned by the port number where the EngCylConn connection is attached to the engine. Cylinder are connected to intake and exhaust ports with Valve*Conn connections. Many Valve*Conn connection

Table 1: Specification the engine and intake port

Engine and intake parameters	Value
Bore (mm)	86.0
Stroke (mm)	70.0
Displacement (cc)	407.0
Number of cylinder	1
Connecting rod length (mm)	118.1
Exhaust Valve Open (EVO) (°CA)	147
Exhaust Valve Close (EVC) (°CA)	282
Intake Valve Open (IVO) (°CA)	395
Intake Valve Close (IVC) (°CA)	530
Injection Start (°CA)	-22.0
Combustion start (°CA)	-21.0
Number of nozzle injector	1
Number of nozzle injector holes	4
Diameter of nozzle injector holes (mm)	0.1
Intake port diameter inlet end (mm)	40.69
Intake port diameter outlet end (mm)	32.78
Intake port length (mm)	55.2
Intake port discreatization length (mm)	34.4
Intake port wall temperature (°C)	176.85

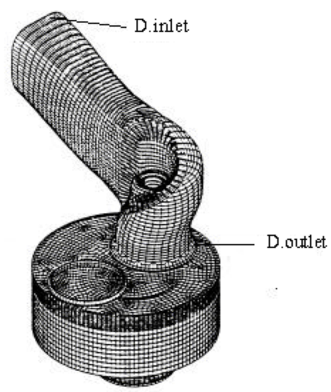

Fig. 1: Intake port of engine

templates are available to define different types of valve, port and their characteristics.

To create the GT-POWER model, open the GT-SUITE software and select the GT-POWER menu, select window and then Tile with Template Library from the menu. This will place the GT-POWER template library on the left hand side of the screen. The template library contains all of the available templates that can be used in GT-POWER. Some of these templates those that will be needed in the project need to be copied into the project before they can be used to create objects and parts. For the purpose of this model, click on the icons listed and drag them from the template library into the project library. Some of these are templates and some are objects that have already been defined and included in the GT-POWER template library. Then, the engine components size data input to the GT-POWER library of the all engine components data. All of the parameters in the model will be listed automatically in the case setup and each one must be defined for first case of the simulation. The engine model is shown in Fig. 2.

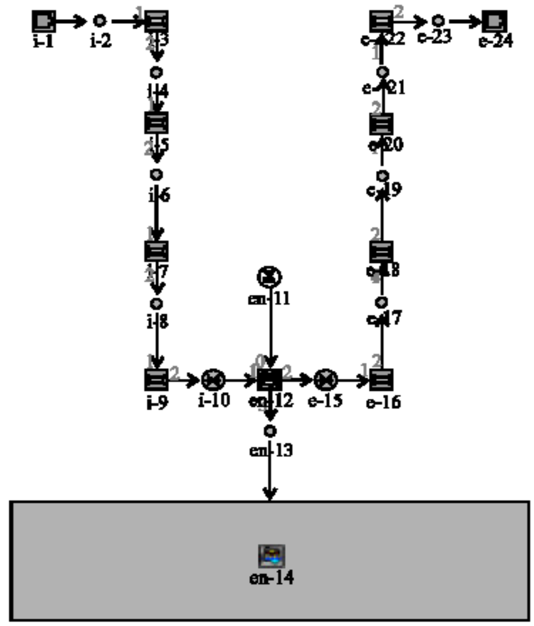

Fig. 2: Single cylinder compression ignition engine model using GT-POWER, i-9 is intake port

In this research, the solver of GT-POWER determines the performance of an engine simulation based on engine speed mode in the EngineCrankTrain object. Speed mode is the most commonly used mode of engine simulation, especially for steady states cases (Gamma Technologies, 2003). In the research imposes the engine speed as either constant or by a dependency reference object. 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.

RESULTS AND DISCUSSION

The simulation of the four stroke direct injection compression ignition engine model is running in 8 cases engine speed. Case 1 is engine model running at 500 rpm, case 2 is engine model running at 1000 rpm, case 3 is engine model running at 1500 rpm, case 4 is engine model running at 2000 rpm, case 5 is engine model running at 2500 rpm, case 6 is engine model running at 3000 rpm, case 7 is engine model running at 3500 rpm and case 8 is engine model running at 4000 rpm. In this research, the result of the model is viewed from GT-Post plots and casesRLT. GT-Post plots result is intake port static pressure versus crank angle degree based on engine speed and GT-Post cases RLT result is intake port pressure versus engine speed based on any cases.

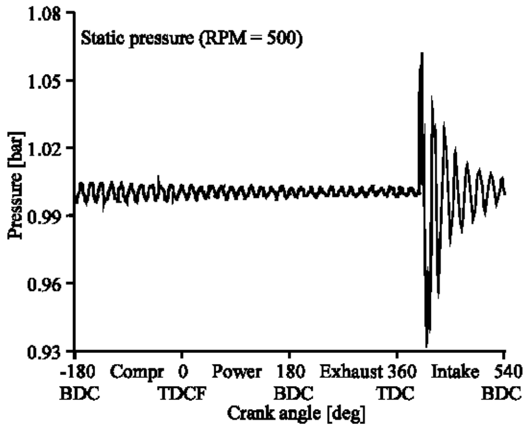

Fig. 3: Intake port static pressure at 500 RPM

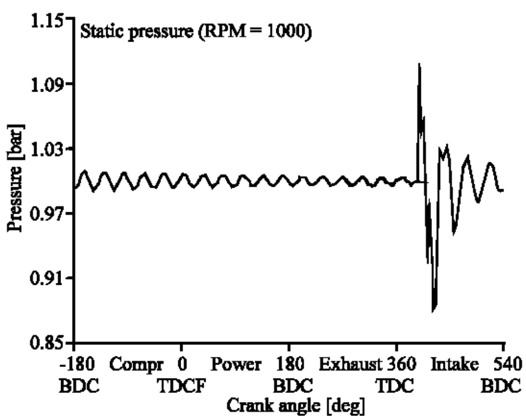

Fig. 4: Intake port static pressure at 1000 RPM

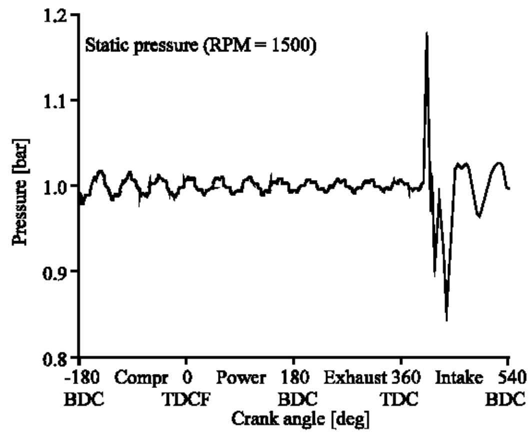

Fig. 5: Intake port static pressure at 1500 RPM

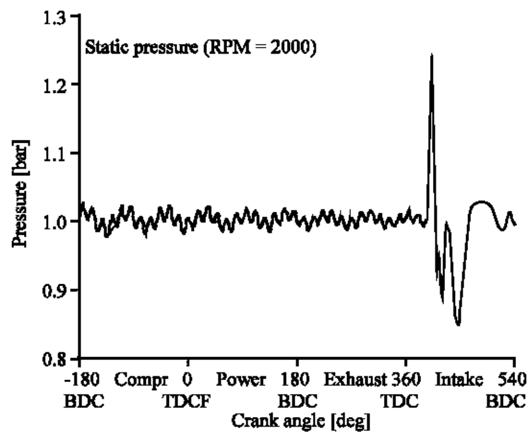

Fig. 6: Intake port static pressure at 2000 RPM

The results of the simulations from GT-Post plots are shown in Fig. 3-10. The results from the GT-Post plots investigation are focuses on inlet port static pressure. Figure 3 shows the intake port static pressure versus crank static pressure versus crank angle profile at 1500 RPM engine speeds, Fig. 6 shows the intake port static pressure versus crank angle profile at 2000 RPM engine speeds, Fig. 7 shows the intake port static pressure versus crank angle profile at 2500 RPM engine speeds, Fig. 8 shows the intake port static pressure versus crank angle profile at 3000 RPM engine speeds, Fig. 9 shows the intake port static pressure versus crank angle profile at 3500 RPM engine speeds and Fig. 10 shows the intake port static pressure versus crank angle profile at 4000 RPM engine speeds.

The static pressures characteristics in intake port profile of four stroke direct injection compression ignition engine is shown in Fig. 3-10. The results have been plotted from simulation result output of GT-Post plots. Figure 3-10 shows the profile of the minimum and the maximum intake port static pressure versus crank angle profile at 500- 4000 RPM engine speeds in compression stroke, power stroke, exhaust stroke and intake stroke of engine. In this engine speed, nominal static pressure in the intake stroke is most extreme then in the compression stroke compare with power stroke and exhaust stroke. In the intake stroke is started from static pressure when the intake valve is just start opened, so the pressure not extreme different compare with another stroke. In the intake valve opened the static pressure is increase extremely because in the stroke the cylinder is needed suction air into the engine cylinder to mixture with fuel for engine combustion to angle profile at 500 RPM engine speeds, Fig. 4 shows the intake port static pressure versus crank angle profile at 1000 R PM engine speeds, Fig. 5 shows the intake port product the power. In the intake valve closed processing until intake valve is closed the pressure static is very low because in this process the gas flow in the intake port is crass with back static pressure from intake valve closed. Then the static pressure is become to lower and lower in

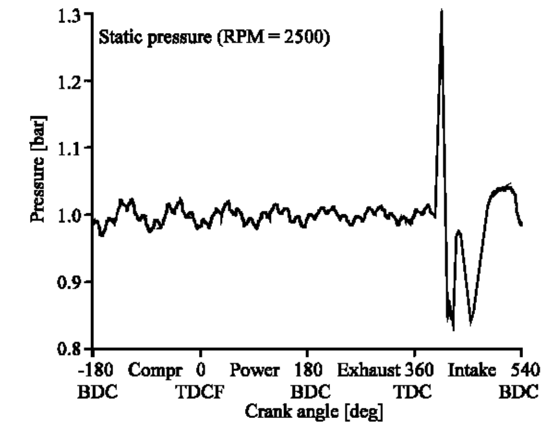

Fig. 7: Intake port static pressure at 2500 RPM

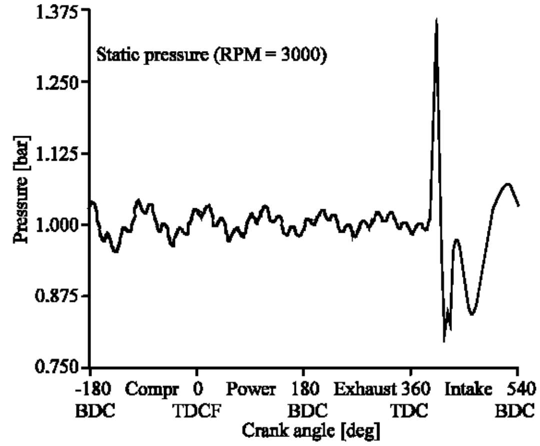

Fig. 8: Intake port static pressure at 3000 RPM

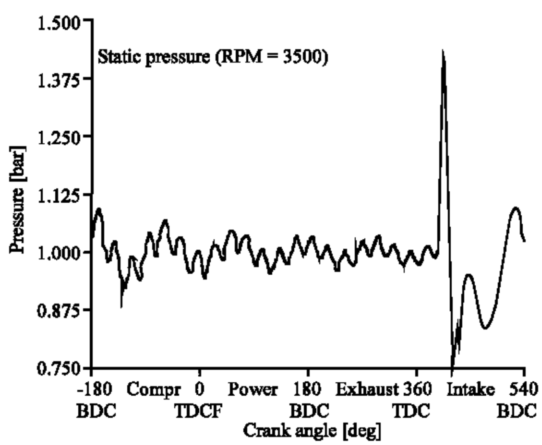

Fig. 9: Intake port static pressure at 3500 RPM

compression stroke, power stroke and in exhaust stroke is the lowest. In the intake stroke, the highest static pressure is declared in 3500RPM engine speed and shown

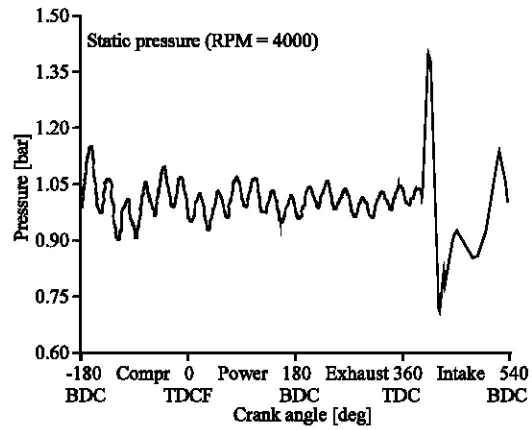

Fig. 10: Intake port static pressure at 4000 RPM

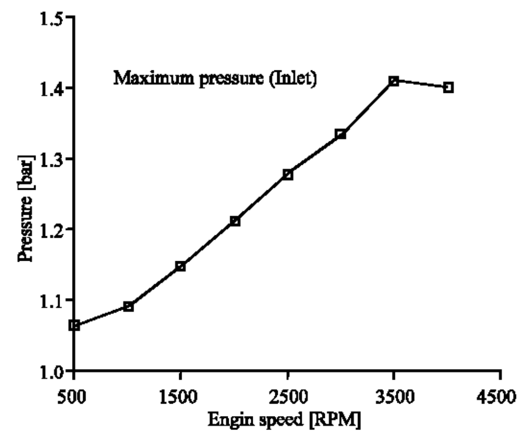

Fig. 11: Maximum pressure inlet at intake port

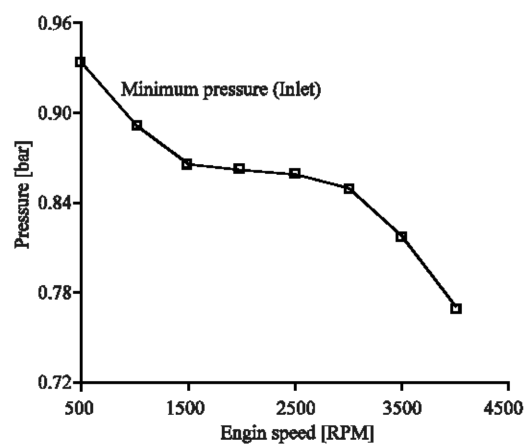

Fig. 12: Minimum pressure inlet at intake port

in Fig. 9, because in this engine speed the combustion is excellent dramatically so need most of air to combustion process and the minimum static pressure is in 4000 RPM engine speed and shown in Fig. 10, because in this engine

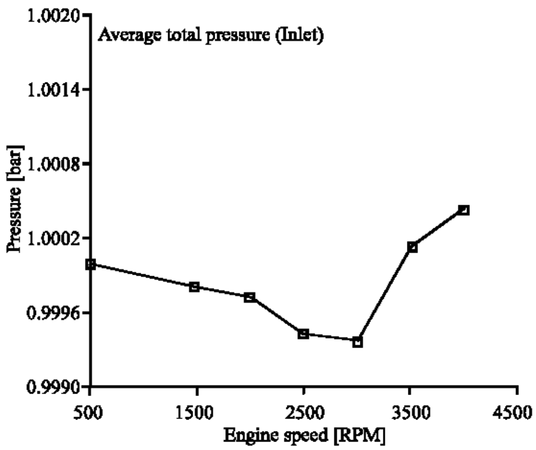

Fig. 13: Average total pressure inlet at intake port

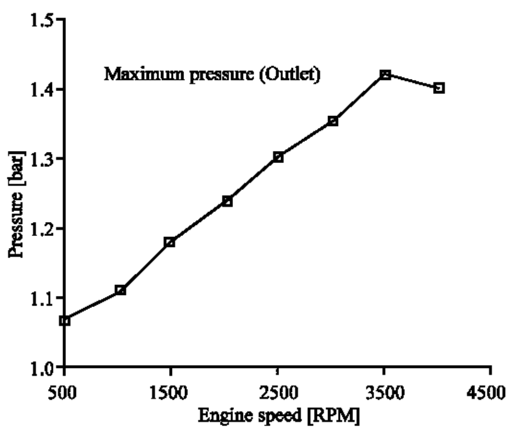

Fig. 14: Maximum pressure outlet at intake port

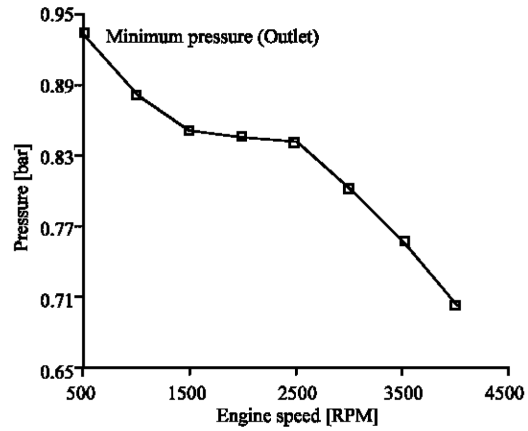

Fig. 15: Minimum pressure outlet at intake port

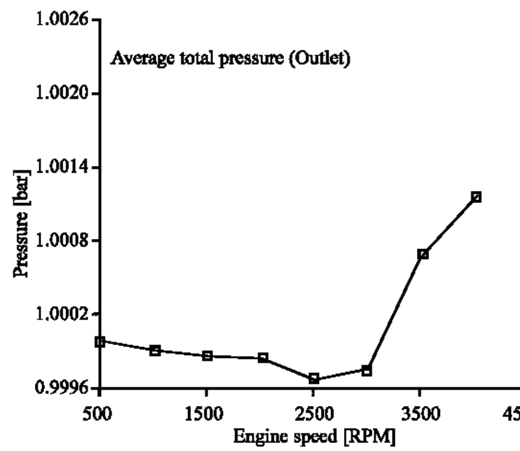

Fig. 16: Average total pressure outlet at intake port

speed investigation the combustion and the intake valve lift and intake valve close moving is most quickly so the air flow back static pressure from the intake valve closing is highest than the other engine speed.

The results of the simulations from GT-Post casesRLT are shown in Fig. 11-18. The results from the GT-Post casesRLT investigation are focuses on any cases in intake port pressure. Figure 11 shows the maximum pressure inlet at intake port on 500-4000 RPM engine speed, Fig. 12 shows the minimum pressure inlet at intake port on 500 - 4000 RPM engine speed, Fig. 13 shows the average total pressure inlet at intake port on 500-4000 RPM engine speed, Fig. 14 shows the maximum pressure outlet at intake port on 500-4000 RPM engine speed, Fig. 15 shows the minimum pressure outlet at intake port on 500-4000 RPM engine speed, Fig. 16 shows the average total pressure outlet at intake port on 500-4000 RPM engine speed, Fig. 17 shows the inlet pressure at end of cycle at intake port on 500-4000 RPM engine speed and Fig. 18 shows the outlet pressure at end of cycle at intake port on 500-4000 RPM engine speed.

The pressure performance characteristics of intake port engine in any cases shown in Fig 11-18. The Fig. 11 shows that the inlet maximum pressure to the intake port from the intake runner the highest is in 3500RPM engine speed, likely the static pressure in 3500RPM the combustion is most excellently so the engine cylinder is needed most air volume for in cylinder combustion process. The Fig. 12 shows that the minimum inlet pressure to intake port the lowest is in 4000 RPM engine speed, because in the engine speed the combustion and the intake valve opened and closed is most quickly. It can be the back flow of air from cylinder and intake valve closed is very high. The air back flow pressure can be reduced the intake pressure inlet to intake port. The outlet maximum pressure from the intake port to the engine cylinder is shown in Fig. 14. The highest maximum outlet pressure is in 3500 RPM engine speed, in 3500 RPM

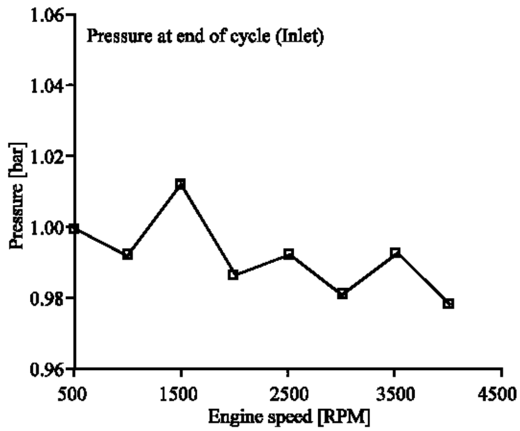

Fig. 17: Inlet pressure at end of cycle at intake port

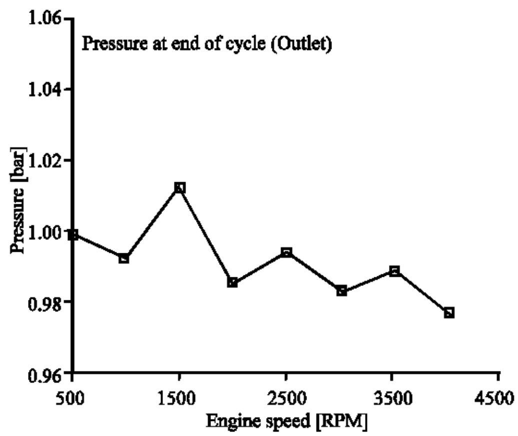

Fig. 18: Outlet pressure at end of cycle at intake port

engine speed the combustion is most excellently so the engine cylinder is needed most air volume for in cylinder combustion process, so the engine need most pressure and volume compare the other speed. The minimum outlet pressure from intake port to engine cylinder the lowest is in 4000RPM engine speed too, because in the engine speed the combustion and the intake valve opened and closed is most quickly. It can be the back flow of air from cylinder and intake valve closed is very high. The air back flow pressure can be reduced the intake pressure outlet from intake port to the engine cylinder. Average total pressure is minimum pressure add maximum pressure divided two. Average of total pressure in inlet process to intake port and outlet process from intake port to engine cylinder is shown in Fig. 14 and 16. The figures shows that the average pressure inlet to intake port the highest is in 4000RPM engine speed and the lowest is in 3000RPM engine speed, the average pressure outlet from intake port to engine cylinder the highest is in 4000 RPM engine speed and the lowest average pressure outlet from intake port to engine cylinder is in 2500RPM engine speed. The pressure at end of cycle at intake port is shown in Fig. 17 and 18. Figure 17 is shows the intake pressure at end of cycle at intake port and Fig. 18 shows the outlet pressure at end of cycle at intake port. The highest inlet and outlet pressure at end of cycle at intake port is in 1500 RPM engine speed and the lowest inlet and outlet pressure at end of cycle at intake port is in 4000 RPM. It mean that in 1500 RPM engine speed the back flow pressure is lowest and in 4000 RPM engine speed the back flow pressure is highest.

CONCLUSION

Intake port pressure versus crank angle and engine speed data can be used to obtain quantitative information to predict the characteristics of the intake port gas flow pressure were needed on the progress of combustion. The simulation results are shown that the highest intake port static pressure is in 3000 RPM, the highest outlet pressure intake pressure to the engine cylinder is in 3500 RPM and the lowest outlet pressure intake port to engine cylinder is in 4000 RPM engine. The simulation result shown the optimum pressure in intake port is if the engine operated in 3000-3500 RPM. If the engine is operated higher than 3500 RPM it can be highest back flow pressure.

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