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Analysis of Flow Characteristics In Inlet And Exhaust Manifolds of Experimental Gasoline Combustion In A VCR Engine

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Abstract

In the present work, an approach to estimate of flow characteristic in inlet and exhaust manifolds of internal combustion engines is performed using a four-stroke variable compression ratio single cylinder gasoline engine

In the theoretical part a computer simulations of the flow field in the intake and exhaust systems as well as the cylinder cavity for the experimental data obtained in the gas exchange cycle program using the method of characteristics for the engine dimensions and timings used in the experimental study as well as the data obtained from the gas exchange cycle program for the sake of comparison and presentation of flow characteristic.

In the experimental work, the compression ratio was varied from 7 to 11 at variable speed with constant throttle opening, where engine performance was obtained.

Results of engine performance as well as pressure, temperature and velocity fields in the intake and exhaust systems obtained by the gas exchange cycle program using the method of characteristics are presented.

Keywords: Internal combustion engine, flow field in pipes, method of characteristic

تحليل خصائص الحريان لمحرك ذو نسبة انضغاط متغيرة

الخلاصة

يتضمن البحث الحالي حساب خواص الجريان في قنوات السحب والعادم لمحركات الاحتراق الداخلي نفذ تجريبيا باستخدام محرك رباعي الأشواط ذو أسطوانة واحدة بنسبة انضغاط متغيرة يعمل بوقود البنزين.

في الجانب النظري تم عمل محاكاة بالحاسبة للجريان لنظام السحب والعادم وكذلك تجويف الاسطوانة بواسطة برنامج دورة التبادل الغازي باستخدام طريقة الخصائص بالاعتماد على البيانات المستحصلة من جانب العملي وفقآ لإبعاد وتوقيتات المحرك المستعملة في الدراسة التجريبية لأغراض المقارنة.

في الجانب العملي تضمن العمل على تغيير نسبة الانضغاط من 7 إلى 11 في اختبارات السرعة المتغيرة ,تم الحصول على أداء المحرك. وفي الجانب النظري تم عرض نتائج أداء محرك وكذلك مجالات الضغط , درجة الحرارة والسرع في قنوات السحب والعادم المستخلصة من برنامج التبادل الغازي بواسطة طريقة الخصائص.

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Nomenclature:

a_0 - Stagnation speed of sound
a_c - Cylinder stagnation speed of sound
C_v - Specific heat at constant volume ($\frac{kJ}{kg \cdot K}$)
D -The hydraulic diameter (mm)
E - Internal energy (kJ)
E_c - The instantaneous surface area inside the cylinder
F - Area of cross-section
f - Friction factor
G - Gibbs function= $f\frac{u^2}{2}\frac{u}{|u|}\frac{4}{D}$
k - Specific heat ratio
m_{frac} - The mass fraction burned
M, A - Coefficients in Wiebe equation
p_c - Cylinder Pressure (bar)
Re - Reynolds number
Rmol - Universal gas constant
T - Temperature (K)
t - Time
u - Gas velocity (m/sec)
V_c - Cylinder volume (m^3)
VCR-Variable compression ratio
x - Distance
ρ - Density (kg/m^3)

Abbreviations:

IVC - Intake valve close
IVO - Intake valve open
EVC - Exhaust valve close
EVO - Exhaust valve open
BDC - Bottom dead center
TDC - Top dead center

Subscripts:

c - cylinder
e - exhaust
i - inlet
w - wall

1. Introduction

The effects of the gas dynamic behavior within an engine strongly influence engine performance, and through the obligatory installation of a flow restriction into the intake system, the gas dynamics and consequently engine performance are adversely altered. It is important therefore to design an intake and exhaust systems geometry that will reduce the impact of this flow restriction.

Wave action techniques involve the solution of the compressible gas flow equations and allow heterogeneous pressure levels to exist throughout the intake and exhaust manifolds.

The method of characteristics has been extensively used to study the unsteady non - homentropic flow in ducts. This method calculates pressure wave and consider one-dimensional flow from the experimental data obtained. This method was used to estimate different flow parameters of the apparatus used in this experimental work.

Benson, R, S., Garg, R, D. And Woolatt, D. [1] appear to be the first to use computers to solve the characteristics equations and apply such techniques to actual engine pipe work systems,

Blair and Arbuckle [2] studied the induction system They have tested a motored, crank – case compression, piston ported, loop scavenged, two stroke cycle engine over a range of engine speeds (*2000-7000 r.p.m.*), for several intake pipe lengths and different inlet port timing. They made a comparison between the test results and the theoretical results using the method of characteristics. They record Pressure–time variations in the intake pipe and crank case. In addition, the volumetric efficiency was recorded. They observed that the overall correlation between theoretical and experimental pressure–time and volumetric efficiency trends is satisfactory.

Newlyn and Few [3] made a comparison between the method of Characteristics and Mach index method (Resonance equation) in calculating the volumetric efficiency.

Yousif [4] made a simulation model of a four–stroke compression ignition engine and computer program was written to simulate the engine. He studied the change in pressure in intake pipe of a single cylinder four strokes engine; he obtained the changes in pressure in the exhaust pipe and the cylinder pressure and temperature.

Mahmoud [5] studied pressure waves formed as a result of the unsteady flow in the induction system, and its effect on the mass flow into the cylinder in reciprocating internal combustion engines of one or more cylinders, for this purpose, a computer program, based on method of characteristics was written and arranged so that it could deal with different configurations of induction and exhaust pipes.

A pressure-loss-junction model, developed by Pearson and Bassett [6], has been validated for the propagation of pressure waves through both simple Y-junctions and a five-into-one junction of the type used in *V10* Formula *1* engines. Simple empirical approaches to designing exhaust systems neglect the large difference in propagation speed of the forward- and reverse-travelling component pressure waves that can lead to large errors in the calculation of tuned lengths.

The present study is an attempt to investigate engine performance and flow characteristics in inlet and exhaust manifolds of internal combustion engines using a four-stroke variable compression ratio single cylinder gasoline engine.

2. Theoretical Approach

One of the classifications of internal combustion engines is based according to the cycle of operations. The cycle of operations can be subdivided into the gas exchange cycle, where the products of combustion are exhausted and replaced by afresh charge, and the power cycle, in which the charge is compressed, ignited, and the hot gas expanded to produce useful work.

Gas exchange cycle calculation starts when *EVO*, a depression in the cylinder pressure occurs and a mass of gases will leave the cylinder to the exhaust pipe. This will initiate a pressure wave, which will propagate along the pipe. The gas exchange cycle calculation is subdivided into two parts:

1. Pipe calculations.
2. Cylinder calculations.

2.1 Pipe calculation

The air in the intake pipe and gases flow in exhaust pipe is unsteady non - homentropic flow of a perfect gas in a constant area pipe, and the flow is compressible. The method of characteristics has been extensively used to study the non-steady flow in ducts [7].

The basic equations for a flow with gradual area changes, wall friction, heat transfer and entropy changes are:

Continuity equation:

$$\frac{\partial r}{\partial t}+r\frac{\partial u}{\partial x}+u\frac{\partial r}{\partial x}+\frac{ru}{F}\frac{dF}{dx}=0 \quad \ldots (1)$$

Momentum equation:

$$\frac{\partial u}{\partial t}+u\frac{\partial u}{\partial x}+\frac{1}{r}\frac{\partial p}{\partial x}+G=0 \quad \ldots(2)$$

where:

$$G=f\frac{u^2}{2}\frac{u}{|u|}\frac{4}{D} \quad \ldots(3)$$

$$F=\frac{p}{4}D^2 \quad \ldots.(4)$$

And

$$f=\frac{t_w}{\frac{1}{2}r\,u^2} \quad \ldots (5)$$

with t_w = wall shear stress

First law of thermodynamics is:

$$qrF\,dx=\frac{\partial}{\partial t}\left\{(rF\,dx)\left(C_vT+\frac{u^2}{2}\right)\right\}+\frac{\partial}{\partial x}\left\{(r\,u\,F)\left(C_vT+\frac{u^2}{2}+\frac{p}{r}\right)\right\}dx \quad ..(6)$$

Where: q is the rate of heat transfer per unit time per unit mass.

The above hyperbolic partial differential equations can be replaced by total differential equations along certain characteristic line. For this set of equations, numerical scheme was proposed by Benson [8,9] to solve the equations by the method of characteristics using Riemann variables.

In calculation of the characteristic, Benson [8,9] divided-pipe into meshes. The calculation proceeded from one time to the other and the mesh points formed a lattice in the time distance diagram. The Riemann variables were calculated at the mesh points. Continuous gas path lines were considered and their corresponding – entropy change values were calculated at the end of each time step and were stored at distance with reference to one end of the pipe.

The interface between the intra-pipe gas dynamic calculation and the boundary condition is dealt with by using the method of characteristics.

2.2 Boundary Conditions

In the program, the following boundary conditions are used:

1. Boundary conditions at the cylinder;
   (a) Boundary conditions at exhaust valves;
   (b) Boundary conditions at inlet valves;
2. Boundary conditions at junctions;
3. Boundary conditions at exists;
4. Boundary conditions at inlets;

Their details are described by Benson [8,9].

2.3 Cylinder calculation

A- Power cycle:

The power cycle, i.e. when the valves are closed, is divided into three parts:

1- Compression stroke.
2- Heat added.
3- Expansion stroke.

The pressure and temperature can be estimated in every step from the following relation:

$$T_2=T_1\left[\frac{V_1}{V_2}\right]^{k-1}=T_1\left[\frac{V_1}{V_2}\right]^{R_{mol}/C_v(T_1)} \quad ..(7)$$

The specific heats have been considered as a function of temperature .The first estimation of new temperature (Eq. 7) has been calculated using the specific heat at constant volume of gases at previous temperature, then a check is done using first law of thermodynamics. P_2 is to be calculated from:

$$P_2 = \left[\frac{V_1}{V_2}\right]\left[\frac{T_2}{T_1}\right]P_1 \qquad \ldots(8)$$

At each time step of a cycle the change of a working fluid parameters in the cylinder can be defined by using the first law of thermodynamics:

$$dQ - dW = dE = E(T_2) - E(T_1) \quad \ldots(9)$$

dQ is heat transfer through cylinder walls, and has been calculated from Annand equation [10].

Heat transfer to the three major surfaces comprising the combustion chamber (cylinder head, piston and liner) is evaluated from the difference between the instantaneous gas temperature value and the metal surface temperature. The convection heat transfer coefficients used in these calculations are derived from the well know semi-empirical relationship of Annand. Heat transfer between the gases and the wall is determined by using Annand equation as quoted in Benson and Whitehouse [10].

$$\dot{Q} = \frac{a\,E_c\,g}{B}(\mathrm{Re})^b(T_c - T_w) + E_c\left(T_c^4 - T_w^4\right) \quad \ldots.(10)$$

Where:

g - is the thermal conductivity of the contents of the cylinder;

B - is the bore of the cylinder;

E_c-is the instantaneous surface area inside the cylinder; and

a , b and c are constants which depend on the type of the engine.

Once the trapped mass of air, fuel, and exhaust residual gas in the engine cylinder has been calculated, the next task is to simulate the energy release rate during the combustion process. For this purpose a heat release model is used, in which the mass fraction of burnt fuel at any instant is evaluated using a Wiebe function.

The Wiebe function defines the mass fraction burned as Wiebe [11].

$$m_{frac} = 1.0 - \exp^{-A\left[\frac{q}{q_b}\right]^{M+1}} \quad \ldots(11)$$

where:

A - A coefficient in Wiebe equation = *10* for gasoline.

M - A coefficient in Wiebe equation = *2* for gasoline.

θ - actual burn angle (after start of combustion)

θ_b - total burn angle (*0-100%* burn duration)

Simulation of the energy release mechanism from the fuel and air mixture and the heat transfer processes enables one to predict the cylinder pressure and temperature during the combustion event and thus a complete cycle simulation is performed.

B- Gas exchange cycle:

The cylinder pressure at any time during the gas exchange process can be calculated depending on the first law of thermodynamics for open system with variable volume:

$$\frac{dP_c}{dt} = \frac{1}{V_c}\left\{\begin{matrix} a_{oi}^2\left(\frac{dm}{dt}\right)_i - a_c^2\left(\frac{dm}{dt}\right)_e - \\ kP_c\frac{dV_c}{dt}\end{matrix}\right\} (12)$$

Where:

(dm/dt)$_i$ - mass flow rate into cylinder .

$(dm/dt)_e$ - mass flow rate out of cylinder.

(dV_c/dt) - change in cylinder volume.

The cylinder volume is calculated from Benson [8]:

$$V_c = F_c\, r \left\{ \begin{array}{l} 1 + n - \sqrt{n^2 - \sin^2 q} - \\ \cos q + \dfrac{2}{CR - 1} \end{array} \right\} \quad ...(13)$$

where F_c - cylinder cross-sectional area; r - crank radius = stroke / 2 ; n = (connecting rod length) / (crank radius) ; θ - angle from top or inner dead center ; CR- nominal compression ratio .

Moreover, the rate of change of cylinder volume (dVc/dt) is given by:

$$\frac{dV_c}{dt} = F_c\, r \left[\begin{array}{l} \sin q + \\ \dfrac{1}{2}\left\{ \dfrac{\sin 2q}{\sqrt{\left(n^2 - \sin^2 q\right)}} \right\} \end{array} \right] 2p\, N \quad ..(14)$$

where N - crank speed in (rev/sec).

The prediction is obtained by the application of the method of characteristics to intake and exhaust systems of a four-stroke single cylinder engine.

The pipe arrangement of the engine is shown in Fig.1, The cylinder calculations were carried out for complete open period from exhaust valve opening to intake valve closure (EVO to IVC).

3. Experimental work

The experimental work is performed in the University of Technology – equipment and machine department-Baghdad. The engine used in the experimental program is a single cylinder four strokes, variable compression ratio engine with Dynamometric test unit "type ($GR0306/000/037A$) a schematic diagram of which is shown in Fig.2. This test bed was supplied with data acquisition system and computer software from Prodit Company of (Italy) [12] that reads, calculates and saves the specification and data through the tests. Results of the following perimeters could be obtained on a computer with a built in program supplied for this purpose:

- Engine Speed.
- Engine Shaft Torque.
- Brake Power.
- Specific Fuel Consumption.
- Stoichiometric Air / Fuel Ratio.
- Cooling water in–and–out temperature and flow rate for heat Balance determination.
- Pressure–crank angle relation and pressure–displacement diagram.

4. Discussion of the results

Figures (3), (4), (5), (6), (7) and (8) show the effect of engine speed on the brake power, brake specific fuel consumption, brake thermal efficiency and indicated mean effective pressure, mechanical efficiency and volumetric efficiency respectively. Fig. (3) show an increase in brake power with the increase in engine speed and a decrease as the speed increases .The effect of compression ratio shows an increase in brake power with CR.

Fig. (4) shows the decreases in brake specific fuel consumption as the compression ratio increases but brake specific fuel consumption increases when engine speed increases. Figs. (5), (6), (7) and (8) show the increase in brake thermal efficiency and indicated mean effective pressure, mechanical efficiency and volumetric efficiency as the compression ratio increases but they decrease when engine speed increases.

The variation of compression ratio at variable speed, by increase the speed from *1100 – 1600 r.p.m*., show an increase in brake specific fuel consumption of *1.521 %* and show a decrease in thermal efficiency of*0.11089%*.

Pressure and temperature variations in the cylinder with respect to crank angle during gas exchange cycle for different engine speeds are shown in Figs (9) and (10).The cylinder calculations were carried out for complete open period from exhaust valve opening to intake valve closure (*EVO* to *IVC*) .

Fig. (9) reveals that a significant difference in the timing of the wave is observed as engine speed changes. However, it is noticed that the transient pressure is sharply reduced during *EVO* in the "blow down" period and before *IVO* the wave pattern is basically made up of pressure pulses; these pulses combine to give a single pulse as engine speed increases, this is because the number of pulses for a given engine is a function of piston movement (piston position),valve opening and engine speed. The pressure at *IVO* is generally higher than the atmospheric pressure and its value is a function of engine speed ,this is because the air valve is opened before (*TDC*). By the time the exhaust valve closes, the pressure has dropped to a value less than atmospheric pressure, which is again a function of engine speed. The pressure increases again after reaching its minimum and appositive pressure in the cylinder gradually builds up. The wave action plus and the piston motion both help to draw in more fresh charge and improve the efficiency of the induction process.

Fig. (10) shows also the variation in the temperature in cylinder during the gas exchange period. It is observed that the sequence of events for the temperature variation in the cylinder coincides with that of the pressure variations .It also shows that after *EVO* the released temperature decreases rapidly and because of the wave action, the temperature will be sustained or slightly raised until *IVO*, when fresh charge flows in and heat transfer takes place between the fresh charge and the residual gas. This will make the temperature in the cylinder decrease even more rapidly during the blow down period until after *EVC*, after this the temperature will gradually increase until *IVC*.

Fig. (11) show the variation in the pressure at pipe adjacent to the inlet valve (at pipe end no.3) with respect to crank angle, for different engine speeds .It is observed that most graphs exhibit almost similar characteristics in different magnitudes as engine speed changes. This clearly illustrates the unsteady nature of the flow in the intake pipes of the engine. It is also observed that there is a sudden increase in the pressure at the first just after *IVO* ;this is because the pressure inside the cylinder is slightly higher than the pipe pressure and also the air valve opens before (*TDC*) causing a compression wave to propagate towards the valve and results in a pressure rise at that point .After piston movement changes direction ,a depression will promptly develop in the cylinder .This results in a flow of gas from the intake pipe into the cylinder.

Figs. (12) show the pressure variation with respect to crank angle at the exhaust pipe (at pipe end no.1) which is adjacent to the exhaust valve .Pressure variations are for different engine speeds.

Figs. (13) and (14) shows the instantaneous temperature through the inlet and exhaust pipes, with respect to crank angle, for different engine speeds. It has a similar trend as the pressure since temperature is pressure dependent.

Figs. (15) and (16) shows the instantaneous velocity through the inlet and exhaust pipes, with respect to crank angle, for different engine speeds. The sinusoidal nature of the velocity profile at each pipe end is again due to the wave action. Some back flow is indicated on the curves. The velocity profile in the inlet pipe can be divided into three parts: when the inlet valve is closed near *BTC*, overlap near *TDC* (low velocity region) and when inlet valve is open (high velocity region).

5. Conclusions

Based on the results presented and discussed the following conclusions can be drawn:

1- The effect of compression ratio in the variable speed test shows an increase in brake power with compression ratio, when the compression ratio increased for all tests the brake specific fuel consumption decreases.

2- The volumetric efficiency and the mechanical efficiency increase as the compression ratio increases but they decrease when engine speed increases for each compression ratio.

3- The results obtained using method of characteristic on pressure and temperature has a good agreement with other literature.

4- The calculation of the instantaneous velocity profile in inlet and exhaust pipes is in agreement with literature.

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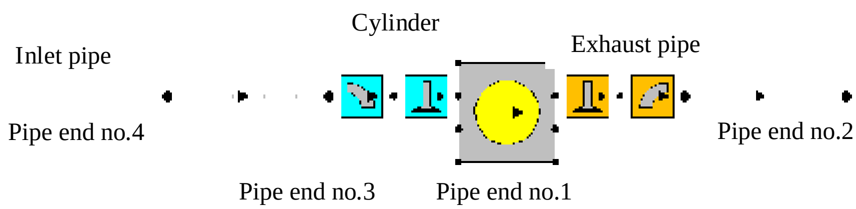

Figure (1) The system to be simulated

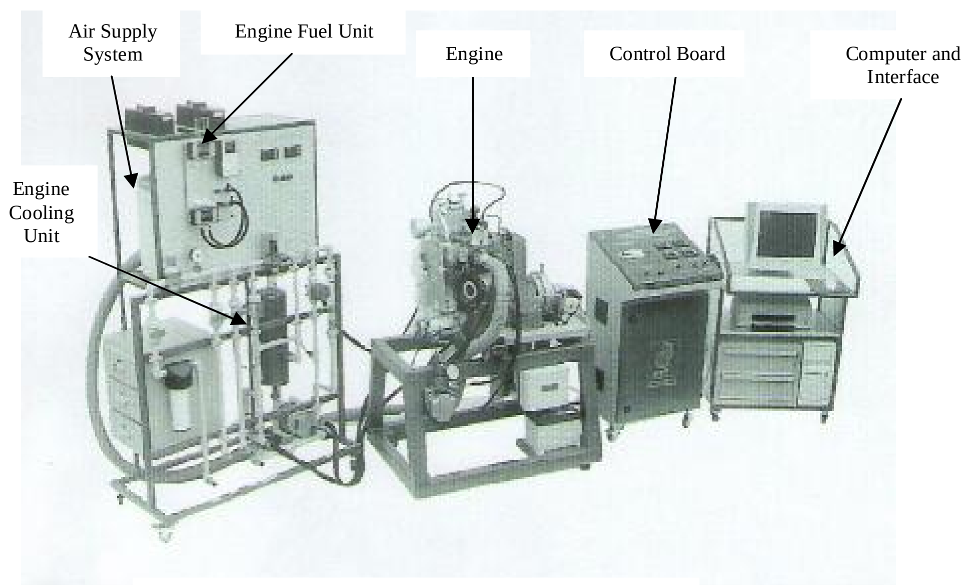

Figure (2)-A Variable Compression

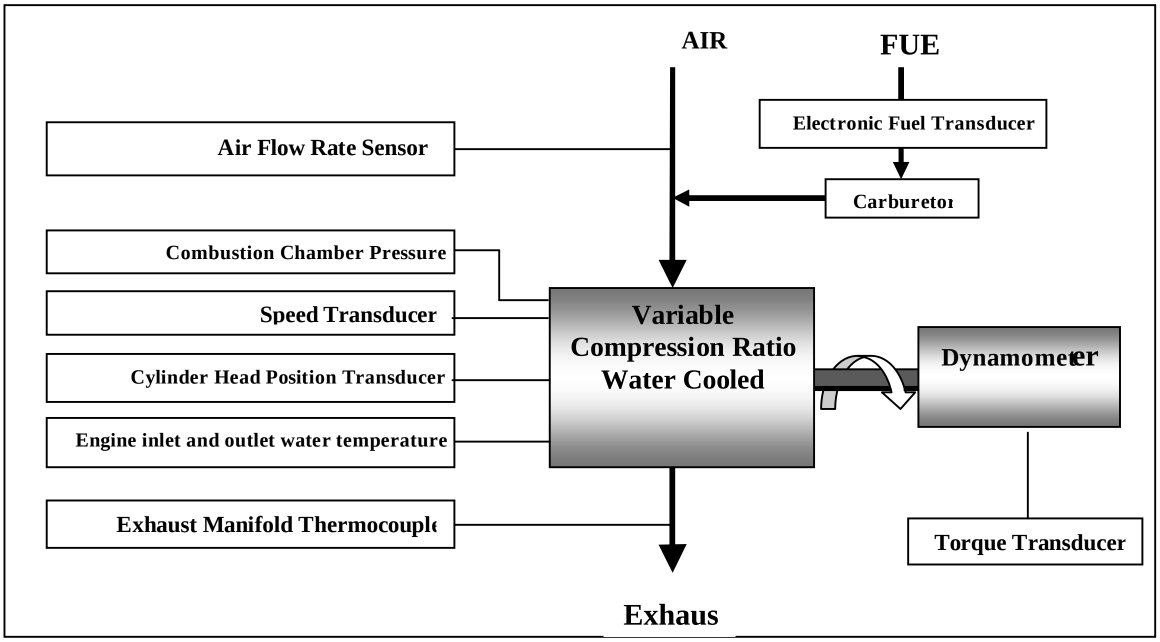

Figure (2)-B Schematic diagram for the variable compression engine

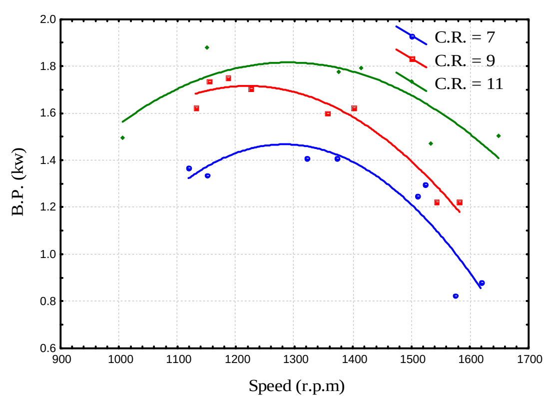

Figure (3) Effect Variable Compression ratio on Brake Power as a Function of Engine Speed.

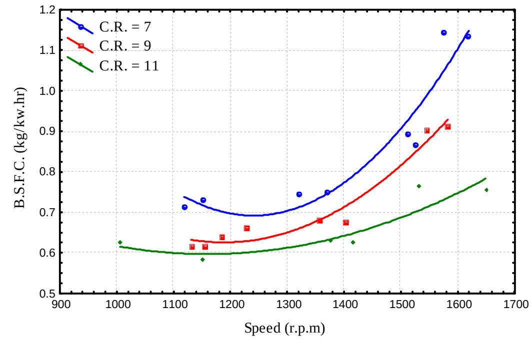

Figure (4) Effect Variable Compression ratio on Brake Specific Fuel Consumption as a Function of Engine

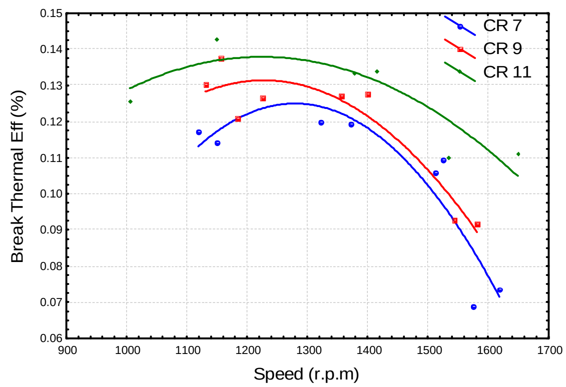

Figure (5)Effect Variable Compression ratio on Brake Thermal Efficiency as a Function of Engine

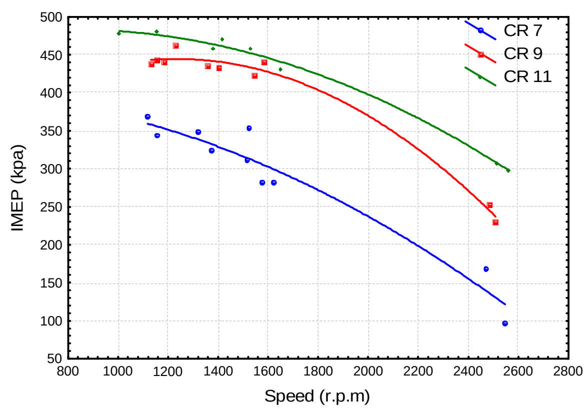

Figure (6) Effect Variable Compression ratio on Indicated mean Effective Pressure as a Function of Engine Speed.

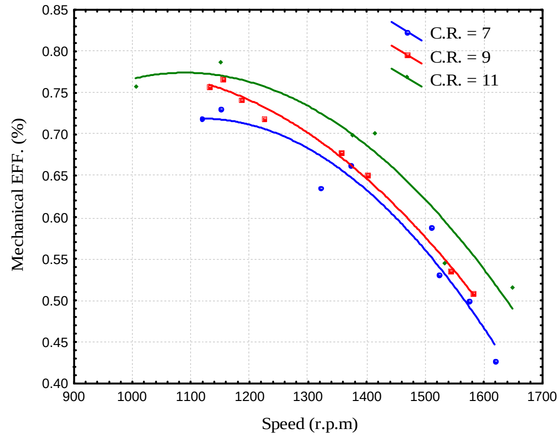

Figure (7) Effect Variable Compression ratio on Mechanical Efficiency as a Function of Engine Speed.

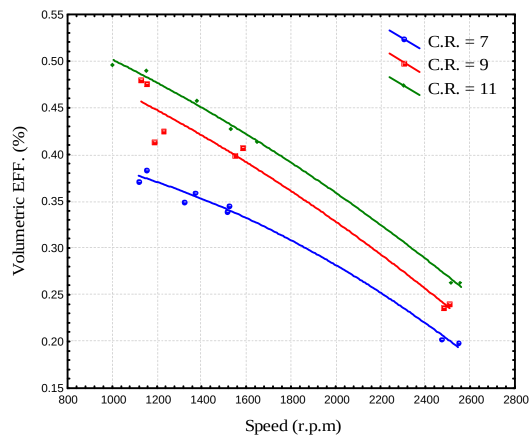

Figure (8) Effect Variable Compression ratio on Volumetric Efficiency as a Function of Engine Speed.

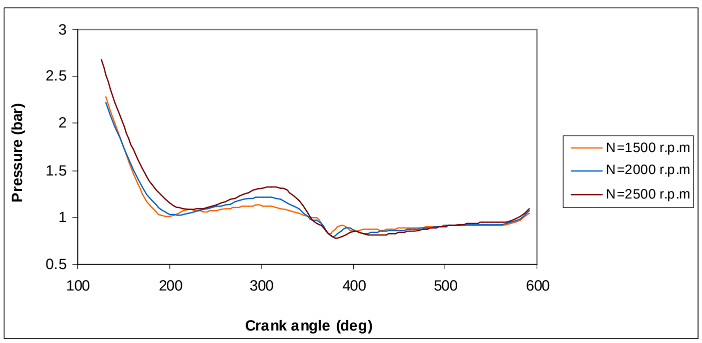

Figure (9) Pressure variations inside the cylinder during gas exchange period at different speeds with CR =9.

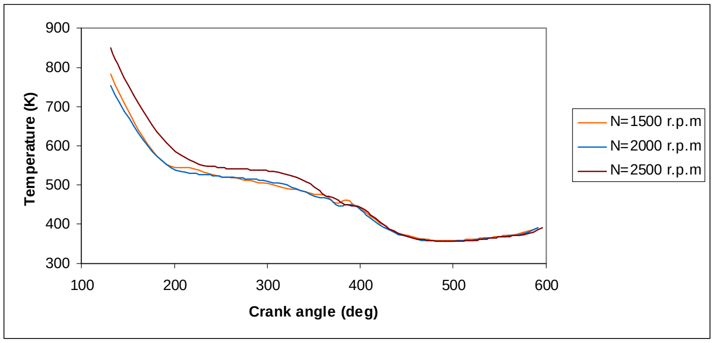

Figure (10) Temperature variations inside the cylinder during gas exchange period at different speeds with CR =9.

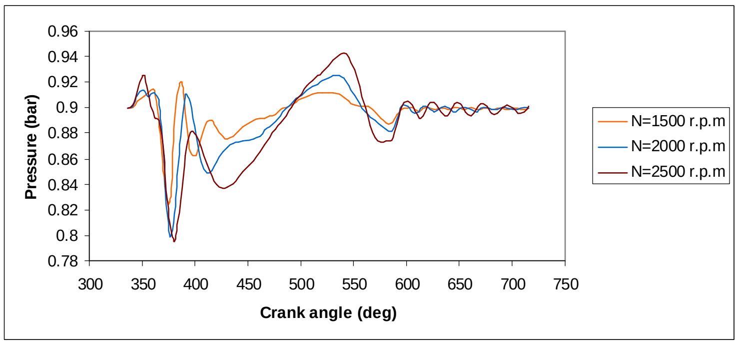

Figure (11) Pressure variations in the air pipe during gas exchange period at pipe end no.3.

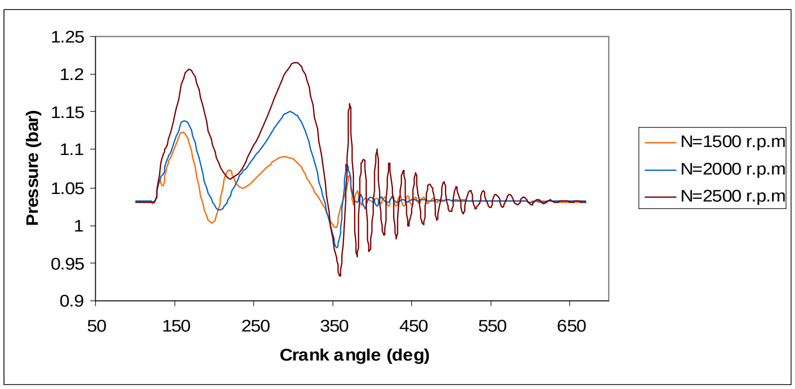

Figure (12) Pressure variations in the exhaust pipe during gas exchange period at pipe end no. 1.

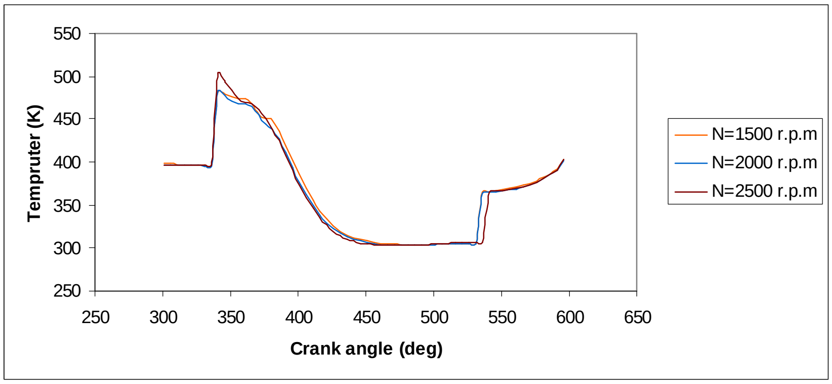

Figure (13) Temperature variations in the air pipe during gas exchange period at pipe end no. 3.

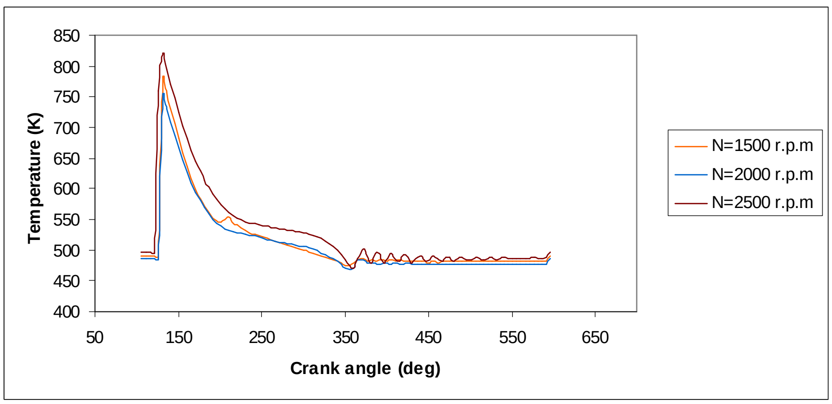

Figure (14)Temperature variations in the exhaust pipe during gas exchange period at pipe end no.1.

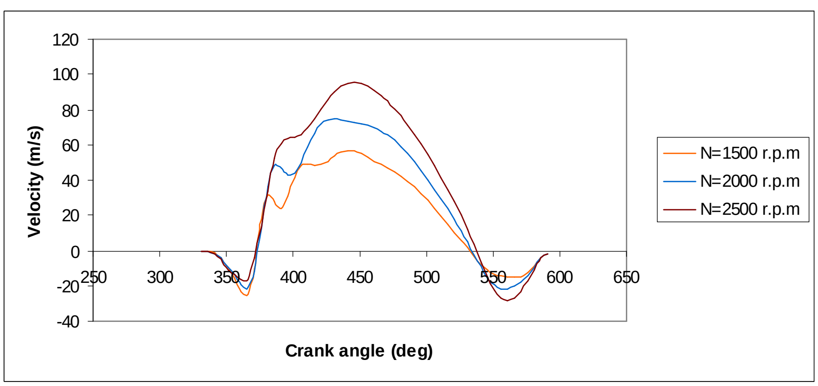

Figure (15) Velocity variations in the air pipe during gas exchange period at pipe end no.3.

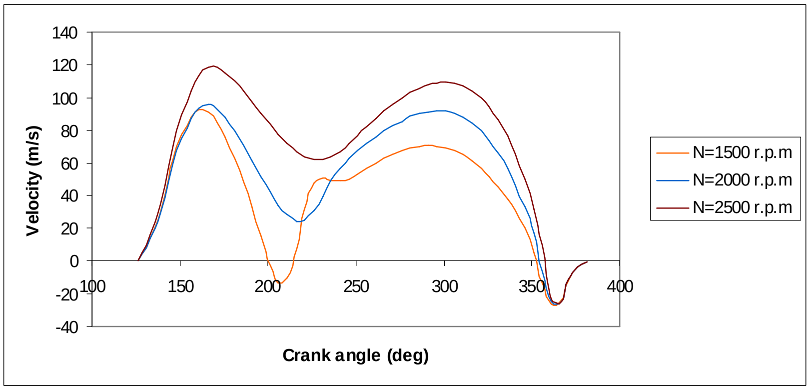

Figure (16) Velocity variations in the exhaust pipe during gas exchange period at pipe end no.1.