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MODELING AIR CHARGE IN VARIABLE VALVE TIMING ENGINES

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ABSTRACT

Variable valve timing is a modern technique that aims to acquire the optimum air capacity at different operating conditions for internal combustion engines. The present work proposed a mathematical model to simulate the thermodynamic processes for motored engine including the air charge flowing through the intake manifold equipped with throttling valve. The pressure waves propagating through the intake manifold is considered. An experimental test rig consists of an engine equipped with measuring devices to record the engine speed, instantaneous cylinder and intake manifold pressure. Volumetric efficiency is estimated based on the experimental data at different engine throttling valve positions. The recorded data were compared with computational output in order to validate the proposed model. The model is used to address the effects of intake valve timing as well as throttling valve position at different engine speeds on the volumetric efficiency and equivalent compression ratio. The results show the importance of delaying the inlet valve closing angle with increasing engine speed to insure the best charge. The results show also remarkable effect of valve timing on the cylinder pressure. When delaying inlet valve closing angle, the effective compression ratio tends to decrease.

يقدم هذا البحث نموذجا رياضياً لتمثيل سريان الهواء داخل محركات الاحتراق الداخلى التى تستخدم اسلوب تغيير زاوية غلق صمام السحب بهدف الحصول على أعلى كفاءة حجمية عند مختلف ظروف التشغيل. يحتوى النموذج الحالى على تمثيل السريان الموجى للهواء خلال مشعب السحب مع تمثيل لصمام الخنق على هيئة فوهة متغيرة المساحة. يحتوى البحث كذلك على دراسة معملية بهدف الحصول على بيانات تجريبية لتقييم نتائج النموذج الرياضى. وقد استخدم النموذج الرياضى بعد ذلك فى دراسة تأثير كل من وضع صمام الخنق وزاوية غلق صمام السحب على الكفاءة الحجمية ونسبة الانضغاط المكافئة. وقد أوضحت الدراسة مدى التأثير الكبير لتوقيت غلق صمام السحب على الضغط داخل الاسطوانة وأنه بتأخير غلق صمام السحب يمكن خفض نسبة الانضغاط المكافئة الأمر الذى يساعد على تطوير محركات تعمل على أنواع مختلفة من الوقود كما يساهم التحكم فى نسبة الانضغاط عند الأحمال الجزئية الى تحسين أداء وكفاءة المحرك.

Keywords: Variable valve timing, intake manifold, air charge, pressure waves, volumetric efficiency.

1. INTRODUCTION

The flow of air charge through the intake manifold of internal combustion engines plays an important role in governing the induction process, volumetric efficiency, engine performance and resultant emission. In naturally aspirated engines, the charge must flow to the engine cylinder with the lowest possible pressure drop along the entire speed and load range of operation. The energy losses in the throttling valve (especially at no-load and part loads) and the fixed timing of the inlet valve motion represent the most important barrier facing the optimization of charge process and consequently the resultant emission. For these reasons, variable valve timing arose as an appropriate solution [1-4]. The technical problems accompanying the variation of valve timing were solved recently. Numerous attempts were made to explain the physical phenomenon associated with the induction process of fresh charge [5]. The complexity of the flow comes from its cyclic nature and pressure waves propagating through the manifold. Several factors were found to be effective in controlling the air charge into the engine cylinders. The manifold geometry (length and cross-section), engine load and speed and finally valve timing were found to be the most important factors governing the instantaneous air charge in internal combustion engines. In variable valve timing engines, the opening and closing angles for each valve can be regulated by using different methods. Recently, phase shifter for camshaft proceeds to be the standard method for this purpose. It works to adjust the relative position of cam with respect to crankshaft by using hydraulic or mechanical devices [6]. Other mechanical systems were developed also to manage the valve timing and lift in order to enhance the charge motion inside cylinders at part load, as well as high load [7-8]. Sophisticated control systems were developed to ensure fast response for dealing with transient operating conditions, not only for optimizing induction process but also to achieve the best fuel metering process[9-10]. The other important method employs a camless technique [11]. It offers full control in the valve timing and its lift, so optimization of inlet charge at different load and speed can be achieved. Beside enhancing the conventional engines, camless technique was found to be useful in optimizing the combustion process in homogenous charge compression ignition (HCCI) engines [12-16], which suffer from the absence of any physical means for controlling the combustion timing and duration. The difficulties in camless engines are not restricted only to valve timing but also to the harmony between the valve motion and piston motion especially near top dead center (*TDC*). The other challenge is in the landing velocity where it must be as low as possible to reduce or prevent the probabilities of valve bounce. Therefore, various sophisticated techniques were introduced to perform these tasks. Some of these methods depend on actuating the valves hydraulically [17-19]. Other method uses electro-mechanical systems in opening and closing the valves [20-21]. In all cases, the electronic control units were employed to ensure fast response and accurate timing in case of idling speed [22-24] and during the entire load and speed range.

In the present work, numerical model is introduced to simulate the induction process in variable valve timing engines. Since the study is concerned about the charging process, combustion is not considered. This generalizes the application of the model to cover all reciprocating internal combustion engines area. The model combines the instantaneous charge motion and mechanical parts motion. Pressure waves traveling through the manifold and across the throttling valve are considered. Basic conservation equations of momentum, energy and continuity are employed. The system of equations are solved together numerically using the control volume method. An experimental test rig is installed in order to measure the important parameters defining the induction process. The test rig consists mainly of a single cylinder, air cooled diesel engine motored by an electric motor. Experimental data are compared with the predicted results of the model for validation. Details of the proposed model and experimental work are introduced as follows.

2. MATHEMATICAL MODEL

The flow of air through the inlet manifold is a cyclic pulsating flow. The sucked air is affected directly by valve timing and engine speed. The manifold size is usually proportional to the engine capacity, so the cylinder size effect is less significant on the suction process. On the contrary, the valve size has strong influence on the suction process. It contributes in reducing the air velocity and consequently the pressure drop through the whole induction system, which, in turn, improves the volumetric efficiency and the overall engine performance. The mathematical simulation for the suction process requires the solution of basic conservation equations of continuity, momentum and energy as well as the equation of state.

1. Continuity equation
   Assuming one dimensional, compressible and time dependent flow, the continuity equation is as follows:
   $$\frac{\partial \rho}{\partial t}+\frac{\partial \rho u}{\partial x}=0 \tag{1}$$
2. Momentum equation
   According to the previous assumptions, the momentum equation is given as:
   $$\frac{\partial \rho u}{\partial t}+u\frac{\partial \rho u}{\partial x}=-\frac{\partial p}{\partial x}+\frac{\partial}{\partial x}\left(\mu\frac{\partial u}{\partial x}\right) \tag{2}$$
3. Energy equation
   In the absence of heat flux from the boundary, the energy equation becomes as follows:
   $$\frac{\partial \rho h}{\partial t}+\frac{\partial \rho u h}{\partial x}=\frac{\partial}{\partial x}\left(\frac{\mu}{Pr}\frac{\partial h}{\partial x}\right) \tag{3}$$
   $$h=\frac{1}{m_w}c_pT+\frac{1}{2}u^2$$
   where, h is the total specific enthalpy, c_p is the specific heat at constant pressure.
4. Equation of state
   For perfect gas, equation of state is as follows:
   $$\rho=\frac{P}{R_{air}T} \tag{4}$$
5. Sonic wave propagation
   The sonic speed at which the pressure waves travel in stagnant medium is calculated from the following relation:
   $$a^2=\frac{dp}{d\rho} \tag{5}$$

Because of the strong effect of throttling valve in controlling the engine load, the valve is simulated as an orifice located inside the induction manifold.

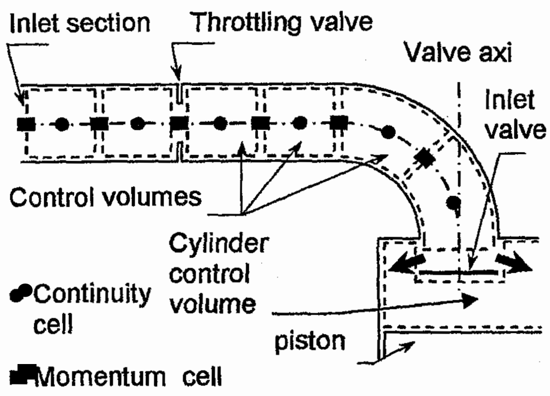

Fig. 1 Grid system used in numerical treatment

2.1 Numerical Solution of Governing Equations

The previously mentioned governing equations in differential form are solved together numerically by using finite volume technique. The domain under consideration is treated as shown in Fig. 1. Two different types of control volume are employed forming overlapped cells. The continuity cell accounts for scalar quantities like pressure, temperature and fluid physical properties. The second cell is the momentum cell, which accounts for the local fluid velocity. The conservation equations for continuity, momentum and energy can be written in the following general form.

$$\frac{\partial}{\partial t}(\rho\phi)+\frac{\partial}{\partial x}\left[\rho u\phi-\Gamma\frac{\partial\phi}{\partial x}\right]=S_{\phi} \tag{6}$$

where, ϕ stands for velocity components or enthalpy in momentum and energy conservation equations, respectively, and Γ is the diffusion coefficient. Its value depends on the meaning of ϕ. For momentum equation, it becomes the effective viscosity and for energy equation, it becomes μ/Pr. For $\phi=1$, and $\Gamma=0$, the equation stands for conservation of mass. In the present work, the grid size for neighboring momentum cells (Δx_{WP} , Δx_{PE} and Δx_{we}) are equal.

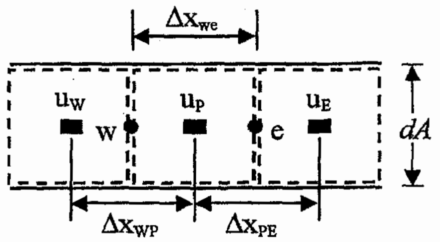

Fig. 2 Construction of grid used in numerical treatment

Integrating the last equation along the momentum cell represented in Fig. 2 gives the following discretization form [25]:

$$A_P\phi_P=\left(a_E\phi_E+a_W\phi_W\right)+S_{\phi} \tag{7}$$

where, a_E and a_W are the coefficients of discretization equation, while S_{ϕ} is the source term of arbitrary variable ϕ. The coefficient A_P is given as follows:

$$A_P=a_E+a_W+F_e-F_w+\left(\frac{\rho_P^n\Delta v}{\Delta t}\right) \tag{8}$$

where, F_E and F_W are the convection/diffusion entering or leaving the considered cell faces. For momentum conservation equation, the source term becomes as follows:

$$S_{\phi}=\left(\frac{\rho_P^o\phi_P^o\Delta v}{\Delta t}\right)-\left(P_e\cdot A_e-P_w\cdot A_w\right) \tag{9}$$

Equation (7) is solved iteratively in implicit finite volume scheme assuming the initial flow field inside the manifold for the first time step. Continuity equation is solved numerically by using *SIMPLE* [25] algorithm. After convergence, the new field is considered as the initial field required for the next time step iteration.

Criterion of stability

The previously mentioned differential equations represent highly time dependent flow so the grid size, time step and speed of sound must be linked together to ensure stable and accurate solution. This can be done when the following condition is satisfied [26]:

$$C=\left(|u|+a\right)\frac{\Delta t}{\Delta x}<1 \tag{10}$$

2.2 Throttling valve

Throttling valve is the main load controller device in SI engines. At part load, the valve reduces the effective cross section area in order to regulate the amount of mixture entering the cylinder as shown in Fig. 3. Therefore, the throttling valve position represents the load ratio. In present model, throttling valve is replaced by an orifice having the same effective cross section area as represented in Fig. 4. The relation between load ratios represented by the angular position of throttling valve and the effective area ratio is as follows [26]:

$$A_r=\left(1-R_{\theta}\right)+\frac{2}{\pi}\left[a_r\sqrt{1-b^2}-R_{\theta}\sin^{-1}b\right.$$
$$\left.-a_r\sqrt{1-a_r^2}+\sin^{-1}a_r\right] \tag{11}$$

where,

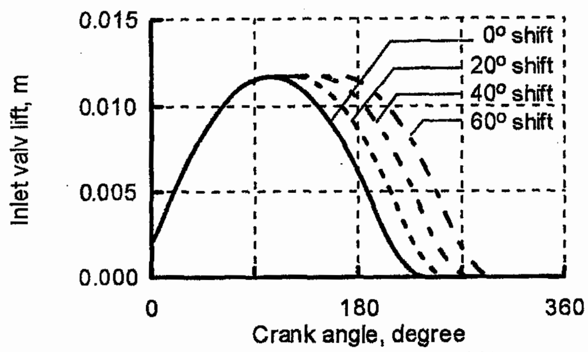

Fig.7 Valve lift at different phase shift

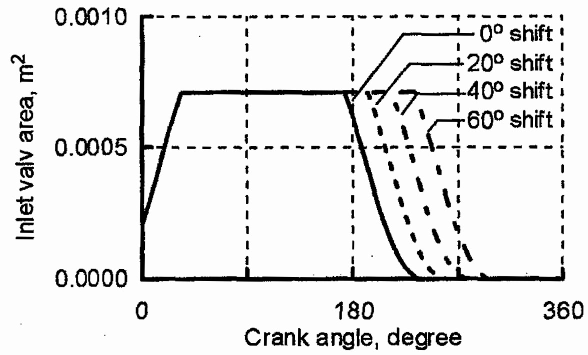

Fig. 8 Area of inlet valve at different phase shift

Figure 8 represents the effective discharge area of inlet valve at different closing angles. The instantaneous flow rate of charge through valves is considered as compressible and adiabatic flow. Therefore, it can be calculated as follows:

$$\frac{dm}{dt}=\frac{C_d A_{ve} P_m}{\sqrt{RT_m}}\sqrt{\frac{2\gamma}{\gamma-1}\left[\left(\frac{P_c}{P_m}\right)^{2/\gamma}-\left(\frac{P_c}{P_m}\right)^{\gamma+1/\gamma}\right]^{1/2}} \tag{16}$$

When flow velocity reaches the sonic speed, the flow is chocked and the maximum amount of mass entering or leaving the valve becomes as follows:

$$\frac{dm}{dt}=C_d A_{ve} P_m\sqrt{\frac{\gamma}{RT_m}\left(\frac{2}{\gamma+1}\right)^{(\gamma+1)/(\gamma-1)}} \tag{17}$$

In the previous equations, P_m and T_m are the manifold pressure and temperature. Their values are considered as in the first control volume in the manifold.

2.4 In-cylinder Process

The processes occuring inside the engine cylinder when running at motored mode is modeled. The gaseous charge is considered as pure air and treated as perfect gas. In the absence of combustion and considering that the experimental engine is air cooled, the heat losses become less significant, so adiabatic process is considered. The flow of air in the inlet or exhaust manifolds is considered as boundary conditions for the cylinder. For complete simulation, energy and continuity equations as well as equation of state are employed as follows.

Energy Equation for in-cylinder contents

$$-P_c\frac{dV_c}{dt}+H_{in}-H_{ex}=\frac{dE_c}{dt} \tag{18}$$

where, H_{in} and H_{ex} are the rate of change of total enthalpy for inlet and exhaust flow, respectively. The presence of both terms in the energy equation depends on valve timing and they appear together during the overlap period only. The instantaneous value of total enthalpy is calculated as follows:

$$H=\frac{dm}{dt}\left(\frac{1}{m_w}c_pT+\frac{1}{2}u^2\right) \tag{19}$$

where, c_pT is the specific enthalpy of air and u is the velocity of the air flowing through the valves. The rate of change in cylinder volume is determined by the following equation:

$$V_c=\frac{\pi}{4}D_c^2\left(\frac{2R}{cr-1}+s\right) \tag{20}$$

where,

$$s=(l_{con}+R)-\left(R\cos\theta+\sqrt{l_{con}^2-(R\sin\theta)^2}\right)$$

Where the cylinder content has constant composition, the rate of change in internal energy depends on the mass of air and its temperature. For closed cycle, temperature becomes the governing factor. This can be expressed mathematically by the following equation:

$$\frac{dE_c}{dt}=\frac{d}{dt}\left(m_c\cdot c_vT_c/m_w\right) \tag{21}$$

Continuity Equation

The cumulative amount of air inside the cylinder depends on the rate of air entering and leaving the cylinder each time step as follows.

$$\left(\frac{dm}{dt}\right)_c=\left(\frac{dm}{dt}\right)_{in}-\left(\frac{dm}{dt}\right)_{ex} \tag{22}$$

Equation of State

As a perfect gas, equation of state for air is as follows:

$$P_cV_c=m_cR_{air}T_c \tag{23}$$

The previously mentioned set of equations are solved together to define the instantaneous cylinder pressure, temperature and volume at each crank angle step. By the end of the suction process specified by closing the inlet valve, the volumetric efficiency is then determined as follows:

$$\eta_v = \frac{P_c}{R_{air} T_c} \frac{(V_c)_{IVC}}{\rho_o (V_c)_{BDC}} \tag{24}$$

The present mathematical model is used to simulate the induction process for both spark ignition engines and compression ignition engines. In case of compression ignition engines, the area ratio of the throttling valve is set to unity.

3. EXPERIMENTAL WORK

Experimental setup consists of *DEUTZ F1L511* single cylinder, air cooled, compression ignition engine. Piezo-electric transducer as well as capacitive transducer is installed in the engine cylinder and intake manifold, respectively, in order to record their instantaneous pressure. Magnetic inductive transducer is mounted opposite to flywheel to pick up instantaneous crank shaft position. The engine is motored via variable speed motor/dynamometer system as shown in Fig. 9. Large air box fitted with inlet manifold is employed to damp the pulsating flow of air charging the cylinder. This technique facilitates the measurement of volumetric efficiency. The original inlet manifold is replaced by another model that has circular cross section with the same area of the original. The new model is equipped with an orifice fitted inside the inlet manifold as shown in Fig. 9. Orifices with different area ratios were used in order to simulate the throttling valve effect. Capacitance pressure transducer is located at 5D upstream the throttling orifice in order to capture the manifold pressure during the cycles. The technical data of measuring instruments are given in Appendix A. The measurements in the present work are concerned with the relation between load ratio and maximum cylinder pressure. This later enables verifying the relation between load ratio and equivalent compression ratio. For this purpose, maximum cylinder pressures at different load ratios are recorded. The other purpose of the experimental work is to validate the proposed mathematical model. Regarding this, the instantaneous cylinder pressure and manifold pressure were recorded. The volumetric efficiency at different loads was evaluated also. In the present study, all experimental work were carried out at motored condition. The speed of the electric motor is kept constant during the experimental work at 1500 rpm.

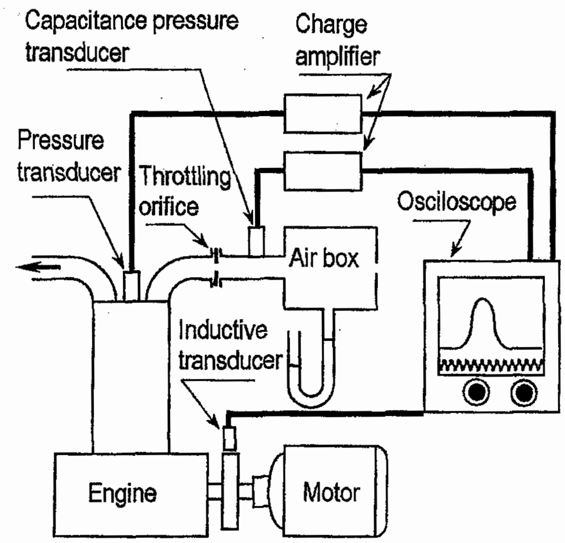

Fig. 9 Experimental layout of present work

4. RESULTS AND DISCUSSION

The present model is employed to study the factors affecting induction process and volumetric efficiency. In order to validate the model, selected results are compared with experimental data. Comparison shows fair agreement in assigning the pressure waves inside the inlet manifold, specifically during the main suction period as shown in Fig. 10. Highly fluctuated pressure waves were recorded after closing the inlet valve, which can be attributed to the interaction between the forward pressure waves with that reflected from the valve side after closing. The cylinder pressure recorded during the compression process is somewhat lower than predicted as shown in Fig 11. The heat losses and blow-by process may be the main reasons. For the same reasons, the maximum pressure recorded at different throttling orifices area ratio is lower than that predicted as shown in Fig. 12. The lower cylinder pressure expresses low contents of air mass, which consequently means lower volumetric efficiency as represented in Fig. 13.

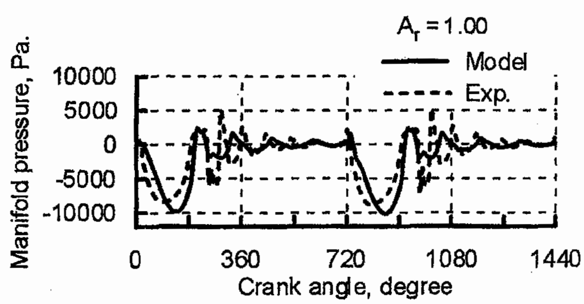

Fig. 10 Manifold pressure upstream the throttling orifice

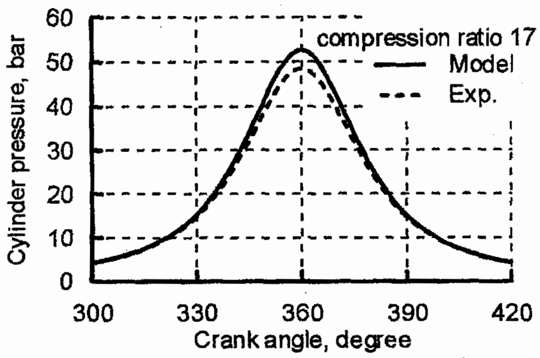

Fig. 11 Cylinder pressure during compression and expansion process

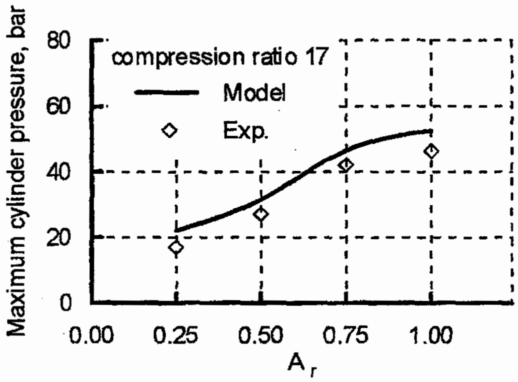

Fig. 12 Variation of maximum cylinder pressure with respect to throttling orifice area ratio

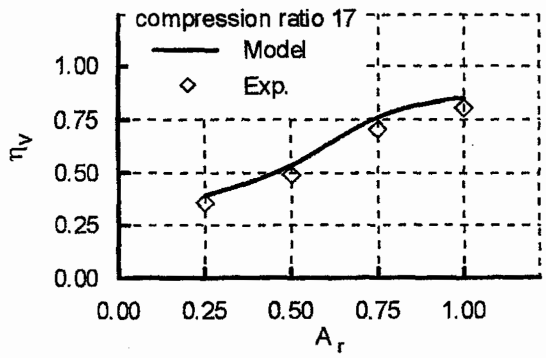

Fig. 13 Variation of volumetric efficiency with respect to throttling orifice area ratio

The effect of the throttling valve appears clearly as shown in the following figures. The lower throttling valve area ratio contributes to damping the pressure waves traveling through the inlet manifold and raising the upstream pressure as shown in Fig. 14. On the other hand, the pressure behind the throttling valve decreased sharply leading to corresponding reduction in cylinder pressure during the suction and compression processes as shown in Figs. 15-16.

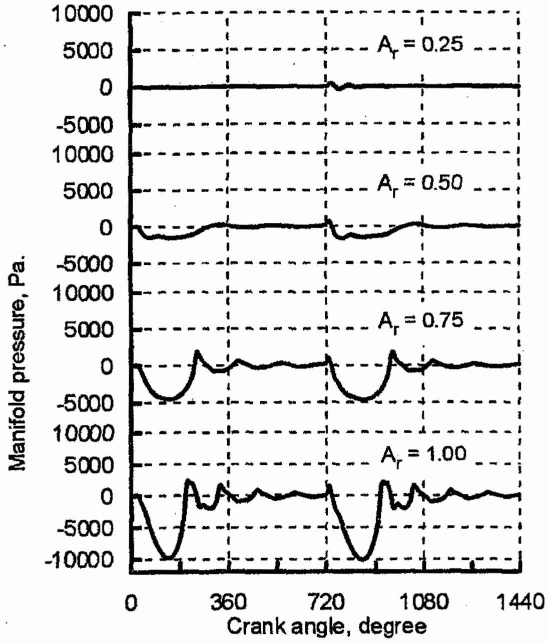

Fig. 14 Manifold pressure during the cycles at different throttling orifice area ratio

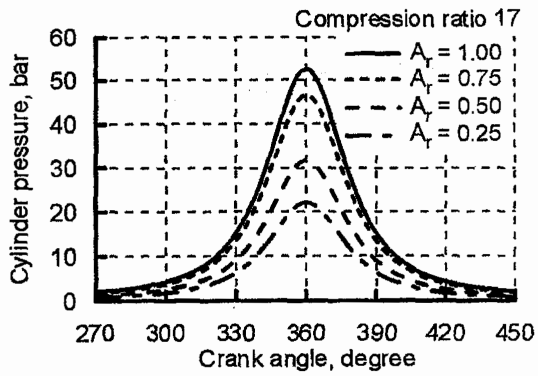

Fig. 15 Cylinder pressure during the compression and expansion strokes at different throttling orifice area ratio

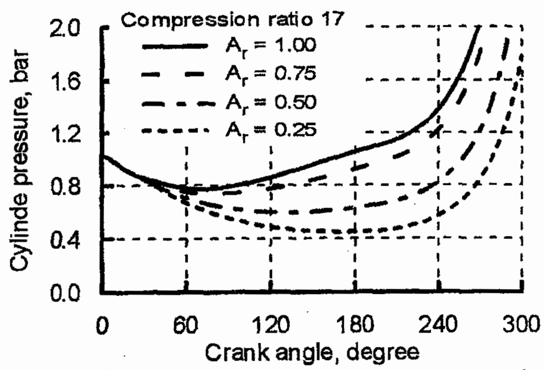

Fig. 16 Effect of throttling orifice area ratio on the cylinder pressure during the suction process

When delaying the inlet valve closing angle, cylinder pressure exceeds the manifold pressure and back flow occurs. This process leads to reducing the amount of air trapped inside the cylinder. The effective compression stroke becomes shorter and consequently the maximum cylinder pressure at *TDC* decreased as shown in Fig. 17. The shortness in compression stroke means that it behaves as variable capacity engine. On the other hand, and for fixed inlet valve closing angle, the increase in the compression ratio leads to corresponding increase in the cylinder pressure as shown in the same figure. This leads to the following conclusion: when delaying inlet valve closing angle, the effective compression ratio reduced as shown in Fig. 18. The effective compression ratio can be defined as the compression ratio that gives the same cylinder pressure at *TDC* as the geometrically defined compression ratio. This result indicates that variable valve timing can play an important role for governing compression ratio at different operating conditions. This technique may be useful in stratified charge engines with in-cylinder injection to prevent the reverse motion of fresh charge into the manifold during the long delaying for closing inlet valve. It may used also in multifuel engines like gasoline-natural gas, which contributes to compromise between the requirements of each fuel with the suitable compression ratio, taking into consideration that in-cylinder injection is essential in all cases. The results show also that variable valve timing technique can be used only to reduce the effective compression ratio below the designed value. So, employing high geometrical compression ratio engines is important to offer wide range of effective compression ratio to fit different operating conditions and fuel types.

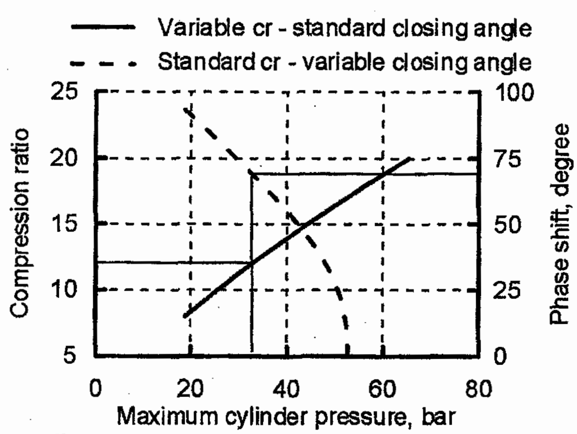

Fig. 17 Relations between maximum cylinder pressure, compression ratio and closing angle phase shift

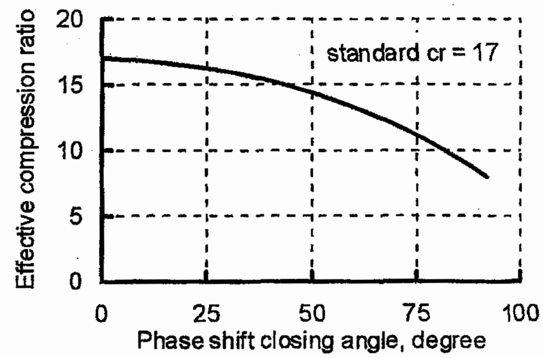

Fig. 18 Variation of effective compression ratio with respect to phase shift closing angle

5. CONCLUSION

The present work proposed a mathematical model to simulate and predict the charging process inside reciprocating engines equipped with a variable valve timing technique. The model incorporates the flow of fresh charge inside the inlet manifold with in-cylinder process taking into consideration the pressure waves propagating inside the manifold. Basic conservation equations of momentum, energy and mass, as well as the kinematics of mechanical parts motion, were employed. The model is used to address the effect of throttling valve and inlet valve closing angle on the charging process and in-cylinder condition. Volumetric efficiency and effective compression ratio are used to evaluate the effect of the previous parameters. Experimental work is carried out to evaluate the proposed model. It is concluded from the present study that:

- Comparison between the results of the proposed model and the experimental data shows fair agreement and indicates that the model provides an important tool for studying and evaluating the charging process for variable valve timing engines.
- The results show the great effect of inlet valve closing angle and throttling valve on the volumetric efficiency of the engine and effective compression ratio. So, variable valve timing can be used as variable compression ratio engines. This technique enables improving the engine performance at part loads, i.e. for low range of power, inlet valve must close early and for higher range, the valve timing must be delayed.
- Delaying inlet valve closing angle leads to reducing the effective compression stroke. This means that, variable valve timing is considered as variable capacity engines.

This type of engines is suitable for stratified charge engines or multifuel engines with in-cylinder injection. This condition is essential to prevent back flow of the fresh charge from the inlet manifold.

NOMENCLATURE

A	Area	m^2
A_p	Area of inlet port	m^2
A_r	Throttling valve area ratio	-
A_v	Instantaneous valve area	m^2
A_{ve}	Effective area of valve	m^2
a	Sonic speed	m/s
a_r	Orifice/manifold diameter ratio	-
C	Courant number	-
C_d	Coefficient of discharge	-
c_p	Specific heat at constant pressure	$J/kmol.K$
c_v	Specific heat at constant volume	$J/kmol.K$
cr	Compression ratio	-
D_c	Cylinder diameter	m
D_p	Diameter of valve port	m
D_v	Diameter of valve seat	m
d_o	Orifice diameter	m
d_s	Diameter of valve stem	m
E_c	Internal energy for cylinder contents	J
h	Total specific enthalpy of air	J/kg
l_{con}	Connecting rod length	m
l_r	Load ratio	-
l_v	Valve lift	m
k	Thermal conductivity	W/mK
m	mass	kg
m_w	Molecular weight of air	$kg/kmol$
P	Pressure	$Pa.$
Pr	Prandtl number $\left(\mu \cdot c_p/k\right)$	
R	Crank radius	m
R_{air}	Gas constant of air	$J/kg.K$
T	Temperature	K
t	Time	s
u	Velocity	m/s
V_c	Cylinder volume	m^3
x	Axial coordinate	m

Subscript

c	Cylinder
ex	Exhaust
in	Inlet port
m	Manifold
o	Ambient condition

Greek letters

Γ	Diffusion coefficient of conservation equations	
γ	Specific heat ratio $\left(c_p/c_v\right)$	
Δ	Difference	
μ	Dynamic viscosity	$Pa.s$
θ	Crank angle	$degree$
θ_t	Throttling valve angular position	$degree$
θ_m	Maximum throttling valve angular position $\left(90^o\right)$	$degree$
θ_o	Throttling valve angle at closing position	$degree$
ρ	density	kg/m^3
ϕ	Arbitrary variable	

ABBRIVIATIONS

$ABDC$	After bottom dead center
$ATDC$	After top dead center
$BBDC$	Before bottom dead center
$BTDC$	Before top dead center
$E.V.C$	Exhaust valve closing angle
$E.V.O$	Exhaust valve opening angle
$I.V.C$	Inlet valve closing angle
$I.V.O$	Inlet valve opening angle

6. REFERENCES

[1] Lancefield, T., Methley, I, Räse U., and Kuhn, T., "The Application of Variable Event Valve Timing to a Modern Diesel Engine" SAE Technical paper, No. 2000-01-1229, pp. 1-13, 2000.

[2] Roepke K. and Fischer M. "Efficient Layout and Calibration of Variable Valve Trains" SAE Technical paper, No. 2001-01-0668, pp. 1-11, 2001

[3] Kaahaaina N. B., Simon A. J., Caton P. A. and Edwards C. F. "Use of Dynamic Valving to Achieve Residual-Affected Combustion" SAE Technical paper, No. 2001-01-0549, pp. 1-12, 2001.

[4] Bohac S. V. and Assanis D.N. "Effect of Exhaust Valve Timing on Gasoline Engine Performance and Hydrocarbon Emission" SAE Technical paper, No. 2004-01-3058, pp. 1-11, 2004.

[5] Hong H., Parvate-Patil G. B. and Gordon B. "Review and Analysis of Variable Valve Timing Strategies—Eight Ways to Approach" Proc. I. Mech. Engrs Vol. 218 Part D: J. Automobile Engineering, pp. 1179-1200, 2004.

[6] Pierik R. J. and Burkhard J. F. "Design and Development of a Mechanical Variable Valve Actuation System" SAE Technical paper, No. 2000-01-1221, pp. 1-8, 2000.

[7] Sellnau M. and Rask E. "Two-Step Variable Valve Actuation for Fuel Economy, Emissions, and Performance" SAE Technical paper, No. 2003-01-0029 , pp. 1- 17 , 2003.

[8] Nagaya K., Kobayashi H. and Koike K. " Valve Timing and Valve Lift Control Mechanism for Engines" Mechatronics, Vol. 16, pp. 121–129, 2006

[9] Gorinevsky D., Cook J. and Vukovich G. " Nonlinear Predictive Control of Transients in Automotive Variable Cam Timing Engine Using Nonlinear Parametric Approximation" ASME, Journal of Dynamic Systems, Measurement, and Control, Vol. 125, pp. 429 – 438, September 2003.

[10] Umut Genç A. and Glover K. "Identification of the Twin Independent Variable Cam Timing Engines for AFR Control" ASME, Journal of Dynamic Systems, Measurement, and Control, Vol. 127, pp. 589-600, December 2005

[11] Puchalsky C., Megli T., Tiller M., Wang N. T. Y. and Curtis E. "Modelica Applications for Camless Engine Valvetrain Development" International *MODELICA* Conference, Germany, March 18-19, pp. 77 – 86, 2002

[12] G M Shaver1 G. M. , Roelle1 M. J., Caton1 P. A., Kaahaaina1 N. B., Ravi1 N., Hathout J-P, Ahmed J., Kojic A., Park S., Edwards1 C. F., and Gerdes J. C. "A physics-Based Approach to the Control of Homogeneous Charge Compression Ignition Engines with Variable Valve Actuation" IMechE, Int. J. Engine Res. Vol. 6, pp. 361-375, 2005.

[13] Zhao L. C. H., Jiang X., and Kalian N. "Understanding the Influence of Valve Timings on Controlled Autoignition Combustion in a Four-Stroke Port Fuel Injection Engine" Proc. I. Mech Engrs Vol. 219 Part D: J. Automobile Engineering, pp. 807-823, 2005.

[14] Koopmans L. and Denbratt I. "A Four Stroke Camless Engine, Operated in Homogeneous Charge Compression Ignition Mode with Commercial Gasoline" SAE Technical paper, No. 2001-01-3610, pp. 1-14, 2001.

[15] Shaver G. M., Gerdes J. C., Roelle M. J., Caton P. A. and Edwards C. F. "Dynamic Modeling of Residual-Affected Homogeneous Charge Compression Ignition Engines with Variable Valve Actuation" ASME, Journal of Dynamic Systems, Measurement, and Control, Vol. 127,pp. 374-381, September 2005

[16] Shaver G. M., Matthew M. J. and Gerdes J. C., "Modeling Cycle-to-Cycle Dynamics and Mode Transition in HCCI Engines with Variable Valve Actuation" Control Engineering Practice, Elsevier Pub. Co., Vol. 14, pp. 213–222, 2006.

[17] Sérgio Barros da Cunha, J. Karl Hedrick and Albert P. Pisano "Variable Valve Timing By Means of a Hydraulic Actuation" SAE Technical paper, No. 2000-01-1220, pp. 1-17, 2000.

[18] Brader J. S. and Rocheleau D. N. "Development of a Piezoelectrically-Controlled Hydraulic Actuator for a Camless Engine. Part 1: System Design" Proc. I. Mech. Engrs Vol. 218 Part D: J. Automobile Engineering, pp.817-822, 2004.

[19] Brader J. S. and Rocheleau D. N. "Development of a Piezoelectrically-Controlled Hydraulic Actuator for a Camless Engine. Part 2: System Modeling" Proc. I. Mech. Engrs Vol. 218 Part D: J. Automobile Engineering, pp.823-830, 2004.

[20] Tai C. and Tsao1 T. "Control of an Electromechanical Camless Valve Actuator" Pro. of the American Control Conference Anchorage, AK May 8-10, pp. 262-267, 2002.

[21] Qiu Y. H., Parlikarz T. A., Changz W. S., Seemanz M. D., Keimy T. A., Perreaulty D. J. and Kassakian J. G."Design and Experimental Evaluation of An Electromechanical Engine Valve Drive" 35th Annual IEEE Power Electronics Specialists Conference, Aachen, Germany, pp. 4838–4843, 2004

[22] Wang Y., Stefanopoulou A. and Smith R. "Inherent Limitations and Control Design for Camless Engine Idle Speed Dynamics" Int. J. Robust Nonlinear Control, Vol. 11, pp. 1023-1042, 2001.

[23] Ashhab M. S. S., Stefanopoulou A. G., Cook A. J. and Levin M. B. "Control Oriented Model for Camless Intake Process-Part I" ASME, Journal of Dynamic Systems, Measurement, and Control, Vol. 122, pp. 122- 130, March 2000.

[24] Ashhab M. S. S., Stefanopoulou A. G., Cook A. J. and Levin M. B. "Control of Camless Intake Process—Part II" ASME, Journal of Dynamic Systems, Measurement, and Control, Vol. 122, pp. 131- 139, March 2000.

[25] Patankar S. V. "Numerical Heat Transfer and Fluid Flow" Hemisphere Pub. Co., 1980.

[26] Heywood John B. "Internal Combustion Engine Fundamentals" McGrawHill, pp 306, 1988.

[27] Heldt P. M. "High Speed Combustion Engines" Oxford & IBH Pub. 2nd , chapter 14, pp. 360-368, 1969.

Appendix A

Technical specification of experiments equipments

Engine specification

Type	*DEUTZ F1L511* Four stroke air-cooled
No. of cylinders	Single cylinder
Bore $\times$ Stroke	100 mm $\times$ 105 mm
Cylinder capacity	825 cm^3
Injection angle	24° *BTDC*
Compression ratio	17
I.V.O	32° *BTDC*
I.V.C	59° *ABDC*
E.V.O	71° *BBDC*
E.V.C	32° *ATDC*

Technical data of piezo-electric transducer

Type	: AVL-piezo-electric transducer
Measuring range	: up to 500 bar
Temperature range	: 240 °C
Sensitivity	: 11.31 pC/bar
Insulation range	: 3×10^{13} ohm
linearity	: <1%
Natural frequency	: 100 kHz