Simulation of exhaust and intake processes in a four-stroke direct-injection diesel engine by control volume approach

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A rigorous mathematical treatment for the determination of the air admission rate by the simulation of the intake and exhaust systems of a four-stroke direct-injection (DI) diesel engine solving the continuity and energy conservation equations in a control volume is presented in this paper. The treatment allows the variation of valve timing, cross-sectional area of the valves, speed of the engine and intake and exhaust manifold pressures and temperatures. The heat transfer effect from the cylinder walls to the gas and vice versa is considered. The methodology adopted for the validation of the data is by comparing the pressure-time histories during suction and exhaust and finally comparing the volumetric efficiencies between the numerical and experimental investigations. From the present study it is concluded that the scheme described in this paper is capable of handling both intake and exhaust process quite satisfactorily.

The problem formulation for intake and exhaust processes is a highly involved work as it deals with an open system and comprises of unsteady flow across valves and the heat transfer. The literature available deals with the problem at times over-simplifying the physical processes or not dealing with the aspects which a design or a simulation engineer is looking for. Borman has formulated a model and has proposed an optimisation technique for the reduction in parameters that affect the volumetric efficiency change. Goyal et al. answered the problem by treating the flow as one-dimensional unsteady and presented the cylinder pressure histories with crank angle at different speeds. The advantages of the early opening of the exhaust valve resulting in better exhaust of the gases in naturally aspirated engines are shown by Charlton & Pappas. The Ram effect in inlet manifolds on the mass flow rate into the cylinder and volumetric efficiency have been studied by Ohata and Ishida. The effect of valve geometry, valve timing on air flow and evaluation of instant flow velocity through valves are given by Asmus. The detailed analysis in the variation of coefficient of discharge, $C_d$, by Heywood and Bicen and Whitelaw for different valve lifts and valve geometries helps the modeller in the selection of proper valve of $C_d$.

The state-of-the-art technology in this field is in the design and operation of variable valve timing and valve lift to suit the various operating conditions. In this category of operation, variable valve timing effects are studied and compared with the calculated values of Leonard and Stone. They have reported that at part load operation, reduction in valve overlap results in better performance of the engine. Titolo of Fiat group, has reported that, an overhead four-cam shaft, four-valves per cylinder, produced record level performance from medium to high speed operation. Hosaka and Hamazaki of Honda group, have not only varied the valve timing but also the valve lift in an operating racing car engine. The effect of resonance created on the intake manifold, in the operational range of 5000 rpm to 7500 rpm and its use, found to increase the volumetric efficiency considerably in comparison to the conventional operation. To date, this valve operating mechanism is quoted to be the latest in the field. By the principle of conservation of energy in the control volume, Campbell has developed the mass flow equations to evaluate pressure variations with time. For solving the energy equations, the necessary equations for choked and unchoked flows across the valves are formulated and the solution methodology is described.

A methodology has been developed in this study to plot cylinder pressure histories with crank angle at different speeds, different valve areas and compression ratios. At each computation the volumetric efficiency and loop work are also calculated.
Basic Equations

The solution of mass and energy conservation equations in the control volume for exhaust and intake processes are given in Appendix 1 and Appendix 3.

The solutions of Eqs (A1.8) and A1.9, as formulated in Appendix 1, results in the plot of pressure histories with crank angle. From now on the Eqs (A1.8) and (A1.9) are numbered as 1 and 2, respectively. For solving Eqs (1) and (2), the term dM/dθ, i.e., the rate of mass entering or leaving the cylinder is to be calculated. The rate of mass flow through effective area, A, of the valve depends on the pressure gradient and the ratio of specific heats, K and the critical pressure ratio across the valves. When the critical pressure ratio

\[ \frac{P}{P_0} < \left( \frac{K+1}{2} \right)^{K/(K-1)} \]  

... (3)

the flow is subsonic or unchoked.

and when

\[ \frac{P}{P_0} > \left( \frac{K+1}{2} \right)^{K/(K-1)} \]  

... (4)

the flow is supersonic or choked.

\( (P/P_0) \) is checked at each time step and the appropriate flow equation as given below is used.

For subsonic flow,

\[ \frac{dM}{dt} = \frac{AP_0}{RT(K-1)} \left( \frac{P}{P_0} \right)^{K-1}/K \times \left[ \left( \frac{P}{P_0} \right)^{(K-1)/K} - 1 \right]^{0.5} \]  

... (5)

For supersonic flow,

\[ \frac{dM}{dt} = \frac{AP}{RT(K+1)} \left[ \frac{2}{K+1} \right]^{(K+1)/(K-1)} \]  

... (6)

Instant valve areas of opening, to account for the valve timing of the engine are calculated by the sine function as explained in Appendix 2. The geometrical modelling of the area opening as given in Appendix 2, is arrived at after carefully studying the cam profile of the experimental engine. This is given as the input to the program. A computer program in FORTRAN is formulated and the SIEMENS Main frame computer is used for running the program. A flow chart explaining the basic features of the program are shown in Fig. 1. When there is a valve overlap \( (dM/dt) \) is calculated taking into account both the inlet and exit valve opening areas (Fig. A2) and the corresponding pressure ratios.

Experimental Procedure

The experimental set-up used for the comparison of the numerical calculations is described in

![Flow chart for intake and exhaust processes](image-url)
Appendix 4. An AVL pressure transducer is used to obtain pressure signals and an Iwatsu (SM2100B) signal analyser is used to store the pressure signals. Mass rate of air consumption is measured for different operating conditions for comparison with the simulated values. A part of the experimental data obtained for the comparison of the results of a two zone combustion model of a DI diesel engine are used for comparing the calculated and experimental results.

Results and Discussions
The gas exchange process simulation during the intake–exhaust process is part of the overall simulation. The main aim of the present work is to study the intake–exhaust processes more closely. Since, most of the studies in the literature is on combustion process, it was decided to investigate the parameters which have a direct bearing on the volumetric efficiency which ultimately affect the performance of the engine. For this purpose a control volume approach has been adopted which is already described. The parameters which have a direct bearing on the volumetric efficiency which ultimately, affect the performance of the engine are presented and discussed.

Effect of compression ratio (CR)—Figs 2 and 3 show the variation of cylinder pressure during the open period, viz., from EVO to IVC. When the exhaust valve opens, because, the pressure ratio, $P/P_0$, is more than the critical value, the flow takes place at supersonic condition with choked situation. It is evident from the figures that compression ratio is an important parameter and higher the compression ratio, the choked flow duration is more. It can be seen from the figures that there is a discrepancy between the predicted and experimental values during the choking conditions. This is to be expected because $C_d$, during the choked flow condition cannot be exactly determined. It is satisfying to note that the predicted values match quite well with the measured one in the subsonic range.

Valve area—Reduction of valve area increases the duration of choking and also the loop work. This phenomenon is clearly seen in Fig. 6 where valve area is reduced to 2.5 sq. cm. from 3 sq. cm. The increase in duration of choking at this particular condition of rated load, rated speed and rated compression ratio is about 3° CA. Different valve areas for exhaust and intake can also be considered. Reduction in valve areas increases...
accuracy in the prediction of choked flow condition. At this stage, it is important to note that in DI diesel engines, there is a constraint on the area availability in the cylinder head for providing the valves. Given an option, increase in inlet valve area, or providing more than one inlet valve, results in better performance at high speed operations, to make this category of engines a good competitor to the other high-speed prime movers.

**Conclusions**

From the present study the following conclusion are drawn: (i) compression ratio has significant effect on the duration of choking flow which affects volumetric efficiency, (ii) suitable modifications are required for better predictions during choked flow conditions, (iii) the present set of equations are able to handle the subsonic flow regions quite satisfactorily, (iv) both inlet valve and exhaust valve area has effect on the cylinder pressure variation during the intake and exhaust process, and (v) engine speed has significant effect on volumetric efficiency.

**Nomenclature**

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>A</td>
<td>area, $m^2$</td>
</tr>
<tr>
<td>AIV</td>
<td>area of inlet valve, at wide open position, $m^2$</td>
</tr>
<tr>
<td>AEV</td>
<td>area of exhaust valve, at wide open position, $m^2$</td>
</tr>
<tr>
<td>BDC</td>
<td>bottom dead centre</td>
</tr>
<tr>
<td>CA, $\theta$</td>
<td>crank angle</td>
</tr>
<tr>
<td>$C_d$</td>
<td>coefficient of discharge</td>
</tr>
<tr>
<td>$C_p$</td>
<td>specific heat of gas at constant pressure, kJ/kg K</td>
</tr>
<tr>
<td>$C_v$</td>
<td>specific heat of gas at constant volume, kJ/kg K</td>
</tr>
<tr>
<td>$dN_a$</td>
<td>mass of air entering the control volume from crank case end, kg</td>
</tr>
<tr>
<td>$dM$, $dp$, $dT$, $dV$</td>
<td>small increments in respective parameters</td>
</tr>
<tr>
<td>$K$</td>
<td>heat capacity ratio</td>
</tr>
<tr>
<td>$M$</td>
<td>mass of gas, kg</td>
</tr>
<tr>
<td>$P$</td>
<td>pressure, N/m$^2$</td>
</tr>
<tr>
<td>$P_0$</td>
<td>atmospheric pressure N/m$^2$</td>
</tr>
<tr>
<td>$R$</td>
<td>gas constant, kJ/kg K</td>
</tr>
<tr>
<td>$T$</td>
<td>temperature of gas, K</td>
</tr>
<tr>
<td>TDC</td>
<td>top dead centre</td>
</tr>
<tr>
<td>$T_m$</td>
<td>temperature of gas in intake manifold, K</td>
</tr>
<tr>
<td>$U_i$</td>
<td>internal energy of atmospheric air, kJ/kg</td>
</tr>
<tr>
<td>$h$</td>
<td>enthalpy, kJ/kg</td>
</tr>
<tr>
<td>$t$</td>
<td>time, s</td>
</tr>
<tr>
<td>$\Delta t$</td>
<td>time step, s</td>
</tr>
<tr>
<td>$V$</td>
<td>volume of gas, $m^3$</td>
</tr>
</tbody>
</table>
Subscripts

E = exhaust
I = intake
P = piston
a = air, after

References
11. Takefum Hosaka & Minoru Hamazaki, Development of the variable valve timing and timing (VTEQ) engine for the Honda NSX, SAE Paper 910009.

Appendix 1

Rate of pressure change formulation in intake and exhaust strokes

With reference to Fig. A1, the thermodynamic properties of working gas in the control volume (CV) at time \( t \) are \( P, V, T \) and \( M \) and at time \( (t+dt) \) are \( P+dP, V+dV, T+dT \) and \( M+dM \). Formulation of energy balance equation.

At time \( t \), energy in CV is \( MC_v T \) \( \ldots \) (A1.1)
During time interval \( dt \) energy leaving CV is \( (-dMC_v T) \) \( \ldots \) (A1.2)

Work done by force, \( F \), entering the CV is \( F \, dx \)

\[ F \, dx = (P - P_0) \, A_r \, dx = -(P - P_0) \, dV \] \( \ldots \) (A1.3)
(Change in volume between \( t \) and \( t + dt \) is negative). During time interval \( dt \), at mass \( dN_a \) enters the CV bringing in energy equal to

\[ h_a \, dN_a = (U_a + P_0 \, v_a) \, dN_a = (U_a \, dN_a - P_0 \, dV) \] \( \ldots \) (A1.4)

Energy in CV at

\[ t + dt = (M + dM) \, C_v (T + dT) + U_a \, dN_a \] \( \ldots \) (A1.5)

Now energy conservation equation is formed by equating \( (A1.1+A1.2+A1.3+A1.4) \) to \( (A1.5) \), dropping second-order terms and rearranging

\[ - P \, dV + (C_p - C_v) \, T \, dM - M \, C_v \, T \, dT = 0 \] \( \ldots \) (A1.6)

From the gas law \( PV = MRT \)

\[ dP + \frac{dV}{V} = \frac{dM}{M} + \frac{dT}{T} \] \( \ldots \) (A1.7)

By eliminating \( dT \) from Eqs (A1.6) and (A1.7)

\[ dP = K_p \, P \left( \frac{1}{M} \, dM - \frac{1}{V} \, dV \right) \]

This equation can be written as piston moves in time step \( dt \). Then,

\[ \frac{dP}{dt} = K_p \, P \left( \frac{1}{M} \, \frac{dM}{dt} - \frac{1}{V} \, \frac{dV}{dt} \right) \] \( \ldots \) (A1.8)

Similarly the energy balance in CV for intake process results in

\[ \frac{dP}{dt} = K_r \left( \frac{RT_a}{V} \, \frac{dM}{dt} - \frac{P \, dV}{V \, dt} \right) \] \( \ldots \) (A1.9)

Appendix 2

Instant valve areas

The ratio of instant valve areas to area at wide open position taking the variations in valve closing and opening periods are modelled as shown below and shown schematically in Fig. A2.

Exhaust valve—Instant area of the valve \( A \) during opening

Fig. A1—Diagram for the analysis of the exhaust stroke for a piston moving at finite velocity

\[ h_a \, dN_a = (U_a + P_0 \, v_a) \, dN_a = (U_a \, dN_a - P_0 \, dV) \] \( \ldots \) (A1.4)

Energy in CV at

\[ t + dt = (M + dM) \, C_v (T + dT) + U_a \, dN_a \] \( \ldots \) (A1.5)

Now energy conservation equation is formed by equating \( (A1.1+A1.2+A1.3+A1.4) \) to \( (A1.5) \), dropping second-order terms and rearranging

\[ - P \, dV + (C_p - C_v) \, T \, dM - M \, C_v \, T \, dT = 0 \] \( \ldots \) (A1.6)

From the gas law \( PV = MRT \)

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Similarly the energy balance in CV for intake process results in

\[ \frac{dP}{dt} = K_r \left( \frac{RT_a}{V} \, \frac{dM}{dt} - \frac{P \, dV}{V \, dt} \right) \] \( \ldots \) (A1.9)
Fig. A2—Variation of valve area ratio with crank angle

$$A = AEV C_d [\sin(\theta_1 - EVO)]^{0.33} \quad \ldots \quad (A2.1)$$

Limits of $\theta_1$ are $EVO < \theta_1 < (EVO + D1)$

During closing:

$$A = AEV C_d [\sin(EVC - \theta_2)]^{0.33} \quad \ldots \quad (A2.2)$$

Limits of $\theta_2$ are $(EVC - D_2) < \theta_2 < EVC$

Intake valve—Instant area of the valve during opening

$$A = AIV C_d [\sin(\theta_3 - IVO)]^{0.33} \quad \ldots \quad (A2.3)$$

Limits of $\theta_3$ are $IVO < \theta_3 < (IVO + D_3)$

During closing:

$$A = AIV C_d [\sin(IVC - \theta_4)]^{0.33} \quad \ldots \quad (A2.4)$$

Limits of $\theta_4$ are $(IVC - D_4) < \theta_4 < IVC$. In this study $D_1$ to $D_4$ are taken as 90°.

Appendix 3

Heat transfer

During the analysis of the heat transfer, for each crank angle, from the gas to the cylinder walls or vice-versa is considered. The pressure drop or rise due to the heat transfer is considered during the suction and exhaust strokes. Heat transfer by convection is given by

$$q_c = h_c A (T_g - T_w) \Delta t, \quad \text{kJ/CA} \quad \ldots \quad (A3.1)$$

where

- $h_c =$ heat transfer coefficient by convection
- $A =$ instant area of cylinder walls, m²
- $T_g =$ gas temperature, K
- $T_w =$ wall temperature, K

For evaluation of $h_c$, Hohenbergs Correlation is used

$$h_c = 0.13 V_c^{-0.06} P^{0.8} T^{-0.1}(C_m + 1.4)^{0.8} \quad \text{kW/m²K} \quad \ldots \quad (A3.2)$$

where

- $V_c =$ cylinder volume, m³
- $P =$ cylinder pressure, bar
- $T =$ mean gas temperature, K
- $C_m =$ mean piston speed, m/s

The method adopted for computation of drop or rise in pressure due to heat transfer is as follows:

$$\Delta P = q_c P(\theta) \Delta t \frac{\Delta t}{m} C, T(\theta), \quad \text{bar/CA} \quad \ldots \quad (A3.3)$$

where

- $\Delta t =$ time, s/CA.
- $P(\theta) =$ pressure at each crank angle, bar
- $m =$ gas mass, kg
- $T(\theta) =$ mean gas temperature at each crank angle, K

Appendix 4

Test set-up details

Name of the Engine
AV1 Kirloskar

General Details
Four stroke, compression-ignition water-cooled, DI diesel engine

Number of cylinders
One

Bore
80 mm

Stroke
110 mm

Swept volume
553 cc

Compression ratio
16.5

Rated output
3.7 kW at 1500 rpm

Speed
1500 rpm

Injection nozzle
MICO-BOSCH

Three hole, 0.2 mm dia.

Injection timing
153° CA aBDC

Fuel injection pressure
200 bar

Combustion chamber
Hemispherical, open combustion chamber, bowl-in-piston 28 cc

Bowl-in-piston to bore ratio
0.7

Length of piston rod
230.1 mm

Dynamometer
Swinging field dynamometer for loading and starting the engine

Pressure transducer
AVL

Signal analyser
SM 2100B, Iwatsu (Japan)