Mansoura Engineering Journal (MEJ), Vol.20, No.4, December 1995

Computer Model For Blowby and Piston Top Land Crevice Flow In Reciprocating Internal Combustion Engines

نموذج حسابي للشحنة المتسربة خلال فراغات المكبس والشنابر بمحركات الاحتراق الداخلي الترددية

By

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الخلاصة: يهتم هذا البحث بنمذجة الشحنة المتسربة خلال فراغات المكبس والشنابر بمحركات الاحتراق الداخلي. وكان الهدف الرئيسي للبحث هو استنباط نموذج حسابي دقيق لحساب هذة الكتلة في دورة الاحتراق بالمحرك والتي لم تؤخذ في الاعتبار في النماذج السابقة لدورة الاحتراق. بتطبيق هذا النموذج على محركات مختلفة، وجد أن الشحنة التي تتسرب الى داخل الفراغات المختلفة بين المكبس والشنابر والجدار الداخلي للاسطوانة أثناء شوط الانضغاط تصل الى ٨ ٪ من الشحنة داخل الاسطوانة، ثم يعود جزء كبير منها مرة أخرى الى غرفة الاحتراق خلال شوط التمدد وهذا بدورة لة تأثير واضح على كمية الهيدروكربونات المنبعثة في محركات الاشعال بالشرارة.

ABSTRACT

This paper is concerned with modelling of blowby and piston top land crevice flow in reciprocating internal combustion engines. The ultimate object was the development of satisfactory computer model for mass leakage through the piston ring pack to be used in an engine cycle model.

The well established simple orifice and volume method for predicting the gas pressures between piston rings in reciprocating internal combustion engines has been extended to take account of the mass flow into the piston top land crevice. Based on this method, a computer model has been developed. This model predicts, as a function of crank angle, the mass flow into and out of the top land and the various piston ring crevices.

Application of the computer model to different engines, has shown that as much as eight percent of the in-cylinder mass leaks into the piston top land and the various crevice regions. A significant proportion of this mass re-emerges into the combustion chamber during the expansion stroke; this contributes to hydrocarbon emissions in spark ignition engines.

KEY WORDS

Blowby, piston top land, mass leakage in I.C.Es.

1. INTRODUCTION

Although the total mass loss by blowby over a cycle of internal combustion engines may be small (3% or less), it is important to note that as much as eight percent of the mass contained in the cylinder may be trapped in the piston's top land crevice and in regions behind and between rings. This occurs at critical points in the cycle, close to where peak pressure is developed [1]. This mass is unavailable for normal combustion, being compressed into the crevices during compression and combustion processes; it is hence of great significance in cycle modeling. After a certain point in the expansion stroke, where the in-cylinder gas pressure becomes lower than the inter ring gas pressure, a significant fraction of this mass re-emerges into the combustion chamber; this mass emerges when the cylinder temperature and pressure are low. Consequently it does not readily burn and so may contribute to UHC emissions in both gasoline and diesel engines.

The development of computer models for blowby and piston top land crevice flow are helpful in modelling engine cycle performance as well as in computation of hydrocarbon emissions.

Most of the work concerned with piston ring blowby modelling is tribological in origin; concerned with friction, wear, and lubrication of the cylinder liner. More recently effort has been motivated by concern with hydrocarbon emissions. Two main methods of modelling blowby have generally been favoured: (i) simple orifice and volume model, and (ii) more complex model accounting for piston ring motion. These will be described in turn.

1.1 Simple Orifice and Volume Model

The simple orifice and volume theory was proposed by Eweis [2], in consideration of tribological aspects of ring pack gas flow. This early work became the basis for several later studies concerned with various aspects of ring pack gas flow. This theory was based upon the similarity of a ring pack to a labyrinth type seal. An interpretation of this theory as reported by Ting and Mayer [3] was utilized by Ruddy [4] for prediction of ring pack gas flow.

Despite the possibility of gas leakage through the ring side clearance and through the clearance between the ring face and cylinder liner, Ting and Mayer [3] assumed that the ring gaps were the only gas leakage paths. The ring pack was represented by a series of in-line square-edged orifices and adjacent volumes, as shown in Fig. 1 for a three ring pack. The orifices represented the total effective leakage paths via the ring gaps and the radial clearance between the piston and cylinder liner. The inter-ring space volumes represented the total volumes between adjacent rings, piston land and cylinder liner.

The gas flow through the orifices was assumed to be unsteady, one dimensional and adiabatic and to satisfy the ideal gas law. The gas was considered to be cooled to the adjacent piston surface temperature, as soon as it entered the volume concerned.

In modeling the flow of gas through the piston ring pack, Ting and Mayer [3] used two principal equations: (i) an equation for the flow rate through an orifice given upstream and downstream pressures, and (ii) an equation for the rate of change of the pressure within the volume, given the net flow rate into it; these equations are set out in Appendix A.

Ruddy [4] verified the simple orifice and volume model by matching his computer model predictions of inter-ring pressures for a particular engine with measured values. The engine modelled was an Abingdon B1 two stroke engine, for which the variation in cylinder and inter-ring gas pressures with crank angle had been previously measured by Baker and Jones [5]; relevant details of this engine are given in Appendix A.

Based on his comparisons of model with experiment, Ruddy concluded that the simple orifice and volume model was an acceptable method for predicting inter-ring pressures within a ring pack.

1.2 More Complex Model

In the simple method described before the passage area available for blowby was assumed constant and equals to the geometrical ring gap area. However, in a real engine, this area varies during a cycle because of ring motion in its groove. Therefore, for more accurate representation, the ring pack flow model must be coupled with a model for ring motion. Namazian and Heywood [6] studied the piston-cylinder crevice flow in a real engine and, taking the effect of ring motion into consideration, developed a more comprehensive blowby model. This model, hereafter termed the Namazian model, is described below.

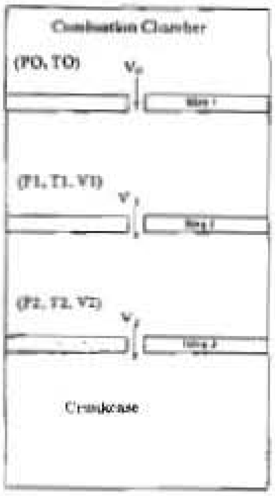

Fig. 1 Representation of three compression-ring pack [4]

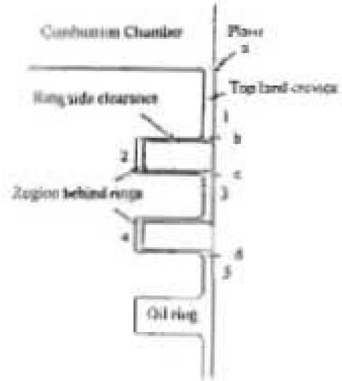

Fig. 2 Schematic of typical production spark ignition engine and ring assembly showing gas flow path [6]

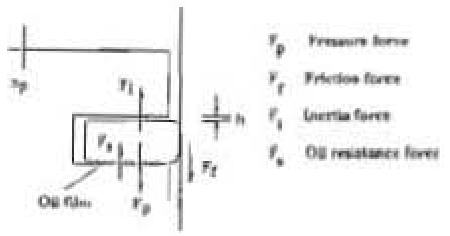

Fig. 3 Schematic of the forces acting on the ring [6]

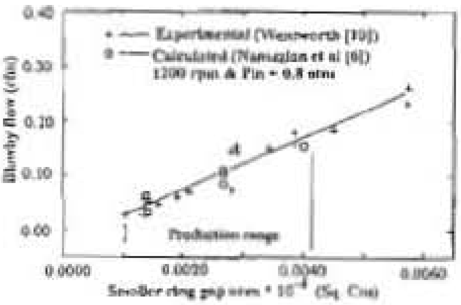

Fig. 4 Comparison between predicted and measured Blowby data for one cylinder of the V-6 GM engine as a function of the smallest ring gap area [6]

The following assumptions were made:

1- The possible gas leakage paths past the piston ring were: (1) ring gap and (2) ring side- clearance areas. Shown in Fig.2 is a schematic diagram of a piston of the type analysed by Namazian and Heywood. This piston was fitted with two compression rings, representative of current spark-ignition engine design.
2- The flow through the ring gap area was considered to be the same as adiabatic, one dimensional orifice flow.
3- The flow through the ring side-clearance was considered as an isothermal compressible flow through a narrow channel.

As discussed previously, there are two possible gas leakage areas in this model: the ring gap area and the ring side-clearance area. The first area was calculated directly from the geometry of the piston-ring-pack and was assumed constant during the cycle. The second is a function of the ring side clearance which changes according to the change in ring position in its groove. The ring can tilt or move axially up and down in the groove . It was proved, experimentally by Furuhama and Hiruma [7] and theoretically by Rungert [8], that ring tilt with an angular-displacement of the order of 5 minutes was not enough to have a marked effect on the flow passage area. The ring must tilt to the extent of 1 degree (60 minutes) to fully block both ring side-clearances. Therefore the authors considered only axial motion of the ring in the groove and assumed that the surfaces were flat. The ring side-clearance was calculated by application of the equation of motion to each ring [6]:

$$M_r \, d^2h/dt^2 = F_p + F_f + F_i - 0.1F_s \qquad (1)$$

where; M_r is the mass of the ring,
h is the ring side-clearance,
F_p is the pressure force,
F_f is the friction force,
F_i is the inertia force and
F_s is resistance of the squeezed oil force

F_s exists only when the ring is sitting on the groove surface and was suggested by Furuhama et al [9] to describe their experimental observations of the floating ring slowing down as it approached the groove surface.

The above forces that act axially on the ring are shown in Fig.3. The equations by which these forces were calculated are given in Appendix B. The rate of change of pressure in each volume crevice was obtained by applying the continuity equation to this volume. The equations by which the mass flow rates through the ring gap and ring side clearance areas and rate of change of pressure were calculated are also given in Appendix B.

In summary, the model was formulated in terms of Eq. 1 (one equation for each of the two compression rings) and the set of these equations listed as equation (13) in Appendix B. The latter set of equations determine the pressures in the regions behind and between the rings. These five differential equations, coupled with those for mass flow rate (Equations (11) and (12) in Appendix B) were solved numerically using the Runge-Kutta integration method.

Namazian and Heywood [6] verified their model predictions by comparing the calculated blowby flow rate values for a V-6 GM engine with those measured earlier by Wentworth [10], for the same engine at the same operating conditions. This comparison is reported in Fig.4. For the model input, two combinations of three ring-gap sizes were used, covering the range of manufacturing tolerance of the actual production engine. There was good agreement between Namazian's model prediction and Wentworth's experimental results for the same engine.

2. MODIFICATION OF THE SIMPLE ORIFICE AND VOLUME MODEL

As has been mentioned before, the simple orifice and volume model was developed to predict tribological behaviours. In the currently reported study, this model has been modified to predict the mass flow rate into and out of the piston top land crevice as well as through the piston ring crevices. This was done by combining a model for piston top land crevice flow with the simple orifice and volume model.

The flow in the piston top land crevice was assumed to be that of a fully developed laminar flow in a channel with constant pressure, equal to the cylinder pressure. It was also assumed that this flow was isothermal, as justified by Namazian et al [6]. They found that the characteristic time for cylinder gases to reach the crevice wall temperature was about 2 crank degrees. This finding has been supported by the measurement of Furuhama et al [11], who established that the gas temperature in the piston top land crevice differed from the piston temperature by only a few degrees.

Using the above assumptions and applying the continuity equation to the control volume comparising the piston top land crevice, the following relation is obtained;

$$\frac{d\overline{W_{ptc}}}{dt} = \frac{d\overline{W_{n-1}}}{dt} + \frac{d\bar{P}}{dt} \qquad (2)$$

where , W_{ptc} is the mass flow past the piston top land crevice,
W_{n-1} is the mass flow through the top ring gap,
P is the pressure in the piston top land volume

This relationship was combined with the simple orifice and volume equations to predict the mass flow rate into and out of the piston top land crevice.

3. CURRENT COMPUTER MODEL

A computer program, based on the modified simple orifice and volume model has been written in Fortran 77. The computational procedure was to:

1- guess the inter-ring gas pressures.
2- calculate the flow rate through each ring gap.
3- calculate the rate of change of pressure within each volume.
4- using the second order Runge-Kutta method, find the inter ring gas pressure.
5- calculate the flow rate through each ring gap.
6- calculate the flow rate past the piston top land crevice.

This model can be applied to both two and four stroke engines, with any number of compression rings. In the two stroke versions, the inter ring pressures are vented as the rings pass the ports in the cylinder liner.

The model predicts, as a function of crank angle, the pressure between the rings, the mass flow through different ring gaps and the mass flow into and out of the piston top land crevice. The inputs required for the calculations are: combustion chamber pressure versus crank angle, details of the engine and piston ring pack geometry and the relevant operating parameters for the cycle under study.

Namazian et al [6] suggested that ignoring piston ring motion could lead to errors of up to 50 % in estimation of that blowby flow rate. However, in the current study, improved accuracy might be expected by the inclusion of the ring groove clearance volume and the volume behind rings into the calculations.

3.1 Examples of the Current Model Prediction

To examine the validity of the current computer model, it was applied to both the Abingdon B1 and V-6 GM engines used previously in similar studies [4 and 6] since well documented experimental measurements for each engine were available in the literature. The specifications and operating parameters for the two engines are given in Appendices A and B.

Shown in Figs.5 and 6 are the results obtained with the current model for the Abingdon engine. The inputs to the program were the cylinder pressure variation and the relevant physical data for this engine, as used earlier by Ruddy [4] and presented in Appendix A. In these calculations, it was assumed that the piston radial clearance remained constant, such that the orifice leakage area changed only with ring gap.

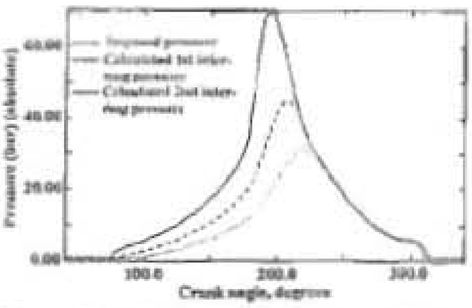

Fig. 5 Current model prediction of inter ring pressure for Abingdton B1 engine

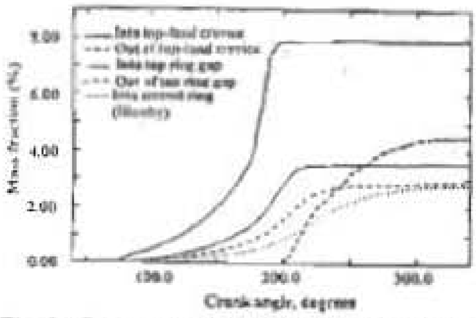

Fig. 6 Current model prediction of % of mass flow into and out of piston-cylinder-ring crevices for Abingdton B1 engine

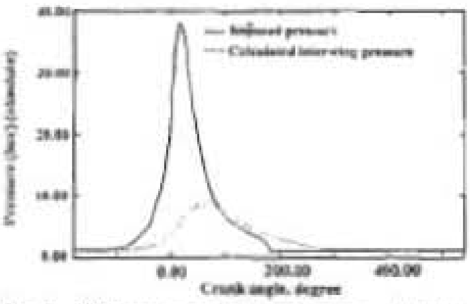

Fig. 7 Current model prediction of inter ring pressure for V6-GM engine

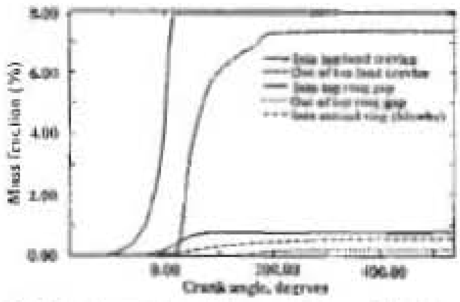

Fig. 8 Current model prediction of % of mass flow into and out of piston-cylinder-ring crevices for V6-GM engine

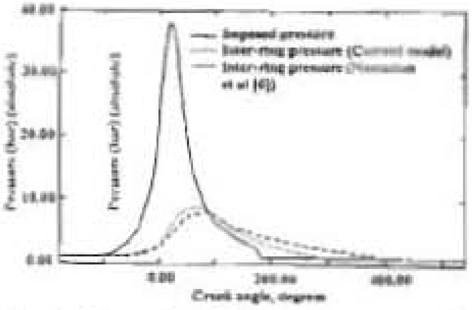

Fig. 9 Comparison between current model and Namazian model predictions of inter ring pressure for V6-GM engine

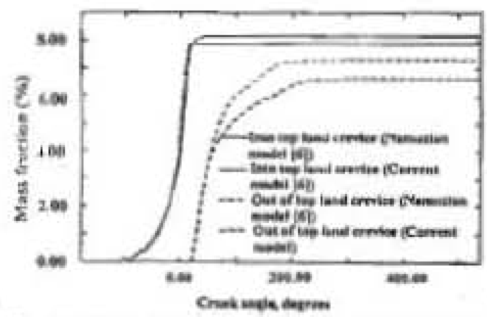

Fig. 10 Comparison between current model and Namazian model predictions of mass flow into and out of piston-cylinder-ring crevices for V6-GM engine

Similarly, changes in the inter-ring volumes were related to variation in axial ring spacing. The effect of thermal expansion was included by assuming a temperature rise for each ring from 20 °C to the value shown for the relevant ring groove temperature. The ring gap sizes were calculated using linear thermal expansion relationships.

Given in Fig.5 are the computed inter ring pressures. Shown in Fig.6 are the corresponding calculated mass flows as a percentage of total cylinder mass, into and out of the piston top land crevice. The mass leakage through the various ring gaps are also shown. For this engine, operating at the conditions described, the total blowby flow rate to the crankcase is about 3 % of the total internal in-cylinder mass. The maximum mass contained in the piston top land crevice reached as much as 8 % of the total in-cylinder mass.

Another example of modelled results is illustrated in Figs.7 and 8. This is the prediction for the inter ring pressure and mass flow rate through the various ring gaps and past the piston crown into the top land for the V-6 GM engine at the indicated operating conditions. The imposed pressure for this cycle is also shown. It may be seen in Fig.7 that, at a certain point in the expansion stroke, the in-cylinder pressure becomes lower than the inter-ring pressure; this causes a reverse flow. It has been suggested by Namazian et al [6] that this reverse flow of air and unburned fuel mixes with the in cylinder combustion products; the latter are relatively cold at this point and do not readily ignite the re-emerging gases. This substantial amount of unburned gases can clearly contribute to emissions of unburned hydrocarbons. For this condition, it can be seen from Fig.8, that the re-emerging gases amount to as much as 7 % of the mass of the in-cylinder charge can clearly contribute to emissions of unburned hydrocarbons. This mass is clearly unavailable for combustion by the normal flame at the optimum temperature in the cycle. This has an adverse effect on engine power and efficiency. This must clearly be taken into account in any engine cycle performance model.

3.2 Current Model Verification

Predicted inter-ring pressures for the V-6 GM engine are set out in Fig.9. There is clearly good agreement between the values determined by the current model and that of Namazian et al. Similarly, good agreement obtained for the mass flow into and out of the piston top land crevice for the V-6 GM engine, as illustrated in Fig. 10

4. CONCLUSIONS

1- The Namazian and Heywood [6] model is likely to yield more accurate predictions of the mass leakage through the various piston ring crevices, over a wide range of engine conditions. However, it requires considerable computational time, extensive physical data for the piston and the ring geometry
2- The simple orifice and volume model provides a reasonably accurate and more rapid method for calculating the inter-ring pressures and mass leakage through the top and second rings.
3- The extension to the simple orifice and volume model incorporated into the current program presents a computationally efficient and reasonably accurate assessment of the mass contained in the top land and other crevices and so unavailable for combustion by the normal flame front at the optimum time in the cycle.

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APPENDIX A

A.1 Simple Orifice and Volume Model

The well established simple orifice and volume model consists mainly of the following equations

(i) flow rate through ring gap equation.

$$\frac{d\overline{W}_{n-1}}{d\theta} = k_i\left(\frac{A_n}{A_i}\right)\left(\frac{P_{n-1}}{\sqrt{T_{n-1}}}\right)\left(\frac{P_n}{P_{n-1}}\right)^{\frac{1}{\gamma}}\left[1-\left(\frac{Pn}{Pn-1}\right)^{\frac{\gamma-1}{\gamma}}\right]^{\frac{1}{2}} \quad (3)$$

(ii) rate of change of pressure within the inter-ring volume equation.

$$\frac{dP_n}{d\theta} = \overline{T}_n\left(\frac{V_i}{V_n}\right)\left(\frac{d\overline{W}_{n-1}}{d\theta} - \frac{d\overline{W}_n}{d\theta}\right) \quad (4)$$

According to the second order Runge-Kutta method and based on an initial value of pressure, the rate of change of pressure within each volume is given by equation (4) and the inter-ring gas pressures are given by

$$(P_n)_{\theta+\delta\theta} = (P_n)_\theta + \frac{\delta\theta}{2}\left[\left(\frac{dP_n}{d\theta}\right)_\theta + \left(\frac{dP_n}{d\theta}\right)_{\theta+d\theta}\right] \quad (5)$$

where $(P_n)_\theta = P_n$ at crank angle θ and $\delta\theta$ = crank angle increment

Through out this work it was assumed that γ =1.3 and the ring gap could be considered as a squared-edged orifice with K_c = 0.65 as suggested by Ting and Mayer [3].

If the pressure ratio across an orifice was less than the critical pressure ratio, then equation (3) simplified to

$$\frac{d\overline{W}_{n-1}}{d\theta} = 0.227K_i\left(\frac{A_n}{A_i}\right)\left(\frac{P_{n-1}}{\sqrt{T_{n-1}}}\right) \quad (6)$$

The new pressure generated from equation (5) were used to calculate new flow rates and the procedure repeated until the pressures (P_{ij}) had converged to within 0.5 percent.

A.2 Abingdon B1 engine data

Cylinder bore, mm	203
Piston stroke, mm	207
Compression ratio	12.5:1
Max. cylinder pressure, MPa	20.6
Operating cycle	Two-stroke
Running speed range, rpm	300-400
Average piston speed, m/s	2.72-3.63
Scavenge system	Constant pressure
Scavenge arrangement	Engine scavenge ported liner, 4 exhaust valves in head
Combustion chamber	Open, without swirl aids
Fuel injection	Central, single fuel valve.

APPENDIX B

B.1 Namazian and Heywood model

B.1.1 Forces acting on piston ring.

Referring to Figs. 2 and 3, the forces acting on the piston ring were calculated as follows.

<u>The pressure force</u>

$$F_p = A_r\left(\frac{P_1 + P_2}{2}\right) - A_r\left(\frac{P_2 + P_3}{2}\right) = A_r\left(\frac{P_1 + P_3}{2}\right) \tag{7}$$

where A_r is the area in the radial direction and P_1, P_2 and P_3 are the pressure at the region 1, 2 and 3.

<u>The friction force</u>

$$F_f = P(\Pi d_r t_r) f \tag{8}$$

where f is the coefficient of friction between the ring face and liner and is given by McGeehan [12] as

$$f = 4.8\left(\mu_{oil}\frac{U_p}{P}\right)^{\frac{1}{2}}$$

where P is the pressure behind the ring.

<u>The inertia force</u>

$$F_i = M_r a_p \tag{9}$$

where M_r is the ring mass and a_p is the piston acceleration.

<u>The resistance of squeezed oil force</u>

$$F_s = \frac{3}{2}\mu_{oil} L_r \frac{dh_s}{dt}\left(\frac{W_r}{h_s}\right)^{3} \tag{10}$$

B.1.2 Gas flow equations

i- the mass flow rate through the ring gap between n^{th} and $(n+1)^{th}$ regions is;

$$\dot{m} = K_t A_g \rho C \tag{11}$$

ii- the mass flow rate through the ring side clearance was considered as an isothermal compressible flow through a narrow channel of width h and length w_r ($h/w_r = 0.01$) and was calculated using the relation given by Shapiro [13] as.

$$m = \frac{Ah^3}{24W_r}\frac{1}{\mu RT}\left(P_u^{\,2} - P^2_d\right) \tag{12}$$

where P_u and P_d are the upstream and downstream pressures.

iii- the rate of change of pressures in regions 2, 3 and 4 are given as.

$$\begin{aligned}
\frac{dP_2}{dt} &= \left[\dot{m}_{12} - \dot{m}_{23}\right]\left(\frac{P_{o2}}{m_{o2}}\right)\\
\frac{dP_3}{dt} &= \left[\dot{m}_{13} + \dot{m}_{23} - \dot{m}_{34} - \dot{m}_{45}\right]\left(\frac{P_{o3}}{m_{o3}}\right)\\
\frac{dP_4}{dt} &= \left[\dot{m}_{34} - \dot{m}_{45}\right]\left(\frac{P_{o4}}{m_{o4}}\right)
\end{aligned} \tag{13}$$

where P_o and m_o are the initial pressure and initial mass in a volume

B.2 Specifications of the V-6 GM engine

Compression ratio	8.1: 1
Displaced volume, c.c	631.9
Bore, mm	96.5
Stroke, mm	86.4
Conn. rod length, mm	15.15
Ring gap area; production range, Cm^2	
small	$1.4 * 10^{-3}$
midrange	$2.4 * 10^{-3}$
large	$4.1 * 10^{-3}$
Dead Volume	
Piston cylinder crevice volume,	
vol.1, cc	0.93
Region behind first ring,	
vol.2, cc	0.467
Region between rings,	
vol.3, cc	0.681
Region behind second ring,	
vol.4, cc	0.467
Total crevice volume	2.55