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A simple and realistic model for the scavenging process in a crankcase-scavenged two-stroke cycle engine

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A semi-empirical model for the scavenging process in a crankcase-scavenged two-stroke cycle engine is proposed. The model is based on the assumption that, typically, the time variation profile of the exhaust gas purity, β, exhibits a sigmoid-type curve while a rounded protuberance usually appears on its back, thus forming a 'crest' profile. The hump has been interpreted as the occurrence of an extensive short-circuiting. An exponential function of the form of

$$\beta = \eta_s^{(1 - c\lambda t)/b}$$

is suggested to fit this curve, from which the scavenging efficiency has been derived. Two parameters, namely the mixing degree, b, and the short-circuiting degree, c, are to be calibrated. However, it was found that the selection of the best fitting values for b and c do not depend on the engine speed or on the engine load. Rather, b and c are inherent properties of the engine design. For modern engine design a value of 0.57 is recommended for b and a value of 0.81 for c/c_{max}. Calculations of the scavenging efficiency were found to be in excellent agreement with predictions of a detailed computer program for three small Schnürle-type different make engines. The present model does not consider a specific mechanism for the gas exchange process, and as such it is believed that it is applicable to other engine types (cross, loop or uniflow) to the same extent.

1 INTRODUCTION

The performance and emissions of an internal combustion engine are dependent on the thermodynamic properties of the mixture trapped inside the cylinder at the commencement of the compression stroke. In two-stroke cycle engines these properties are closely related to the efficiency of the charging process. In this process, the products of combustion are replenished by the fresh charge (fresh air for a direct fuel-injected engine or fresh mixture for carburated engine). The gas exchange process begins at the time when the exhaust port is exposed (in a piston controlled port) or the exhaust valve opens, and is completed at the time when both ports (exhaust and scavenge) close.

The most efficient process will be that whereby the products of combustion are completely replaced by the fresh charge, at charge pressure and temperature, and wasting no fresh charge, through the exhaust. The success of this process is evaluated mainly by the charging efficiency η_c and the scavenging efficiency η_s. Another important parameter is the delivery ratio λ. Following the Society of Automotive Engineers (SAE) recommended terminology (**1**), these are defined as follows:

$$\eta_c = \frac{\text{mass of delivered charge retained}}{\text{displaced volume} \times \text{ambient density}} \quad (1)$$

$$\eta_s = \frac{\text{mass of delivered charge retained}}{\text{mass of trapped cylinder charge}} \quad (2)$$

$$\lambda = \frac{\text{mass of delivered charge}}{\text{displaced volume} \times \text{ambient density}} \quad (3)$$

A reliable description model of the gas exchange process in a two-stroke cycle engine would be a powerful tool to the engine designer. To benefit the most, the designer should select the best model for a particular purpose. Optimizing the geometry of a cylinder and ports assembly would require a very detailed model which considers the effect of all the relevant geometrical parameters on the gas dynamics inside the cylinder. Such a model could involve a complex computer program which might be found to be an impractical tool to engine designers who wish to predict the overall characteristics of a given engine under different operating conditions. For this purpose an appropriate simple correlation between the charging efficiency and the delivery ratio (or other parameters) may be preferable. A comprehensive review of scavenging models may be found elsewhere (**2**).

Recently, a semi-empirical model—the 'S' shape model—has been proposed (**3**) to simulate the scavenging process in cross, loop or uniflow scavenged engines. The model was based on the assumption that the time variation of the mass fraction of fresh charge content in the gas passing through the exhaust port, β, always exhibit a sigmoid ('S'-type) curve (**4–7**). At this stage, it is interesting to note that none of the classic models (perfect displacement, isothermal and non-isothermal perfect mixing) predicts such a typical behaviour. The failure of these models is emphasized at the first half of the scavenge process.

In general, the 'S'-type curve may be represented by an exponential function as follows:

$$\beta = 1 - \exp(-\gamma\lambda t^{\delta}) \quad (4)$$

where δ and γ are the form and shape factors respectively (a value of 2.0 is recommended for δ and 1.7 for γ), and t is defined as

$$t = \frac{\theta - \theta_{so}}{\theta_{sc} - \theta_{so}} \quad (5)$$

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Here, the subscripts so and sc are scavenge port opens and scavenge port closes, respectively. It can be shown (3) that the charging efficiency and the scavenging efficiency may be evaluated by

$$\eta_c = \lambda - \int_0^t \frac{\beta\dot{\lambda}}{\beta + (1-\beta)T_b/T_i}\,dt \tag{6}$$

and

$$\eta_s = \left(\frac{T}{T_0}\right)\eta_c \tag{7}$$

where T_0, T_b, T_i and T are the ambient temperature, the burnt gas temperature, the fresh charge temperature, and the average instantaneous temperature of the cylinder content, respectively.

This model suggests that the scavenging process may be interpreted as a combination of perfect displacement scavenging [the first term in equation (6)] and charging losses (the second term), while the character of the process is determined by the form and the shape factors, δ and γ. Although the 'S' concept has been used successfully by several authors (**5**, **7**, **8**) it seems that it should further be developed for the following two main arguments:

1. In the present form of the 'S' shape model, the instantaneous composition of the exhaust gas is prescribed *a priori* and does not consider the actual instantaneous composition of the cylinder content. However, these are not independent variables. Such an assumption may lead, in some cases, to an absurd situation where the exhaust gases will contain more burnt gas (as prescribed by the model) than available inside the cylinder.
2. A careful examination of the composition of the exhaust gas [reference (**4**) for experimental observation and reference (**6**) for computed results using a detailed three-dimensional computational fluid dynamics simulation] shows that it does not behave as a smooth 'S' shape, but rather it exhibits a 'crest' profile, that is a rounded protuberance usually appears on the 'S' shape back (Fig. 1). The hump may be interpreted as the occurrence of an extensive short-circuiting, as will be discussed later. The basic 'S' shape model does not consider this important deformity.

It is the purpose of the present work to propose a new model for the scavenging process in a two-stroke engine which is a realistic model for modern engine design, on the one hand, and a practical model for engine simulation, on the other.

2 THE 'CREST' SHAPE MODEL

It is a basic assumption of the perfect mixing model that, as it enters, the fresh charge mixes instantaneously with the cylinder contents to form a homogeneous mixture. The instantaneous composition of the gas leaving the cylinder through the exhaust port is therefore a mixture of fresh charge and burnt gas at the same mixing ratio ($\beta = \eta_s$). While this situation is the lower bound for the scavenging process in modern design engines, the perfect displacement ($\beta = 0$ for $\lambda < 1$) is the higher bound. It is the basic assumption of the present proposed model that the instantaneous exhaust gas composition depends mainly on the instantaneous compositon of the cylinder contents. We propose the following relation:

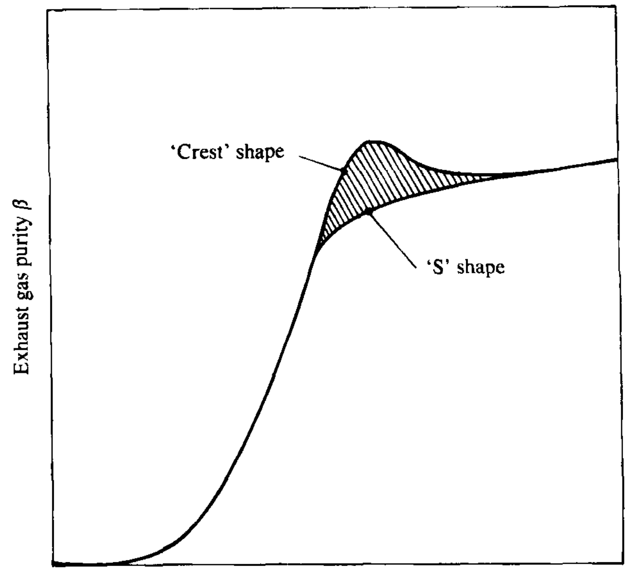

Fig. 1 The 'crest' profile and the 'S' shape models. The hump may be interpreted as the occurrence of an extensive short-circuting

$$\beta = \eta_s^{(1-\varepsilon)/b} \tag{8}$$

where b and ε are parameters that consider the actual mixing and short-circuiting processes. This type of expression assures that not only the exhaust gas purity is closely related to the composition of the cylinder contents, but also that the exhaust gases (first argument) will never contain more burnt gas than available inside the cylinder; thus:

$$\int_{so}^{sc} (1-\beta)\dot{m}_e\,dt \leqslant (m_{c,\,g})_{so} \tag{9}$$

where $\dot{m}_e$ is the exhaust gas mass flowrate and $(m_{c,\,g})_{so}$ is the mass of the burnt gas inside the cylinder at scavenge port opens. Moreover, if a realistic rate of delivery ratio ($\dot{\lambda}$) is considered, that is a bell-type curve on $\dot{\lambda}$–t coordinates, an 'S' shape curve for β will result. The exact character of the curve is determined by the parameters b and ε. While b is a measure of the mixing degree ($0 < b \leqslant 1$), which is assumed to be only a function of the engine type, ε is a measure of the short-circuiting degree ($0 \leqslant \varepsilon$).

Further, we assume that the short-circuiting degree is not only an inverse function of the engine speed (**9**) but also a linear function of the incoming mass flowrate ($\dot{m}_c$). The latter is based on the experimental observations in Schnürle-type engine design (**4**), for which it was clearly shown that the short-circuiting flux is mainly constructed from the backflow stream resulting from the combination of the two inlet jets (Fig. 2). Thus:

$$\varepsilon = c\dot{\lambda}\tau \tag{10}$$

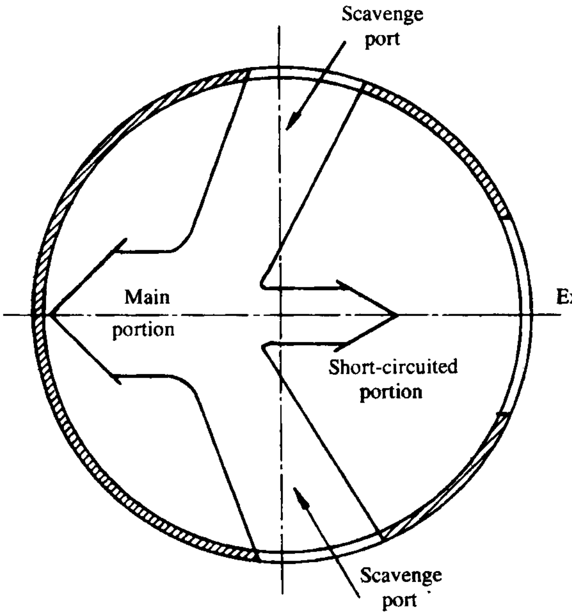

Fig. 2 The short-circuiting flux is mainly constructed from the backflow stream resulting from the combination of the two inlet jets

where

$$\tau = \frac{\theta_{sc} - \theta_{so}}{2\pi N} \tag{11}$$

is the characteristic time for the scavenging process and c is a calibration parameter which also takes into account the path type of the short-circuiting charge.

Substitution of equations (10) and (11) in equation (8) results:

$$\beta = \eta_s\left(1 - c\dot{\lambda}\,\frac{\theta_{sc} - \theta_{so}}{2\pi N}\right)\Big/ b \tag{12}$$

Since the scavenging efficiency is always $\leqslant 1$, a perfect displacement process is represented by $b \to 0$, while a perfect mixing by $c = 0$ and $b = 1$. A perfect short-circuiting is represented by $b \to \infty$, but this case is impractical and is outside the interest of the present paper. For modern engine design, the instantaneous exhaust gas purity β is always smaller than the scavenging efficiency η_s and therefore:

$$0 \leqslant c \leqslant \frac{1 - b}{\dot{\lambda}_{max}\,\tau} \tag{13}$$

for $\dot{m}_i$ and $\dot{m}_e > 0$, and $c = 0$ otherwise.

The proposed model is therefore formulated by the above three simple equations (12), (6) and (7).

3 DISCUSSION

3.1 The constant volume model

Equation (6) suggests that the scavenging process may be interpreted as a combination of a perfect displacement scavenging minus some charging losses. The type of process is determined by the calibration parameters: the mixing degree b and the short-circuiting degree c. It also suggests that for a given function of the delivery ratio (versus time), the charging efficiency depends mainly on the prescribed function of the exhaust gas purity; this is determined by the parameters b and c. In order to evaluate the effect of these parameters, a simplified process, in which the gas exchange was assumed to occur at constant volume, was examined. However, a realistic function for the delivery ratio which takes into account the variable area of the scavenge ports as well as the compression effect of the crankcase volume, has been considered. Two typical cases were studied. In the first the rate of the delivery ratio $\dot{\lambda}$ was represented by a simple symmetric function as follows:

$$\dot{\lambda} = 1 - \cos 2\pi\tau \tag{14}$$

In the second case, a backflow from the cylinder to the crankcase, which typically occurs at low engine speeds (**10**), was allowed to occur at the end of the scavenging period. Then, the rate of the delivery ratio was represented by the following function:

$$\dot{\lambda} = 2(1 - \tau - \cos 2\pi\tau) \tag{15}$$

For both cases, the total delivery ratio (at the end of the scavenging process) is equal to 1. The two types of supplied delivery ratio are shown in Fig. 3. Figures 4 and 5 show the effect of the parameters b and c on the exhaust gas purity β and the resultant scavenging efficiency η_s, as calculated via equations (6) and (7). The upper limit for β, which is also the lower bound for η_s, is represented by the curve $b = 1$ and $c = 0$. As the mixing degree (b) decreases, the purity of the exhaust gas decreases (in both cases) and the scavenging efficiency increases over the entire time interval. For the second case, whereas a backflow occurs at $\tau = 0.785$, the purity of the exhaust gases gradually increases and then decreases towards the end of the scavenging process. It is interesting to note that the exhaust gas purity maximizes at a time that depends on the degree of the short-circuiting, thus

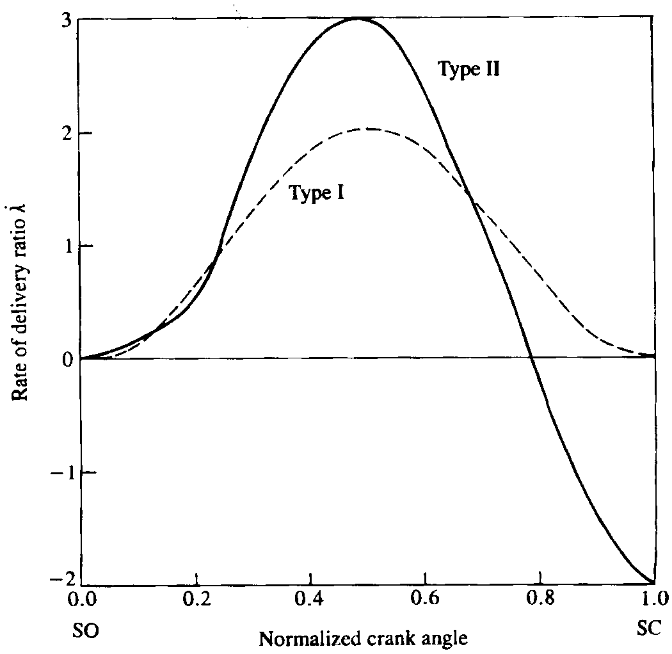

Fig. 3 The two types of the delivery ratio supplied for the constant volume model. For both cases, the total delivery ratio (at the end of the scavenging process) equals 1

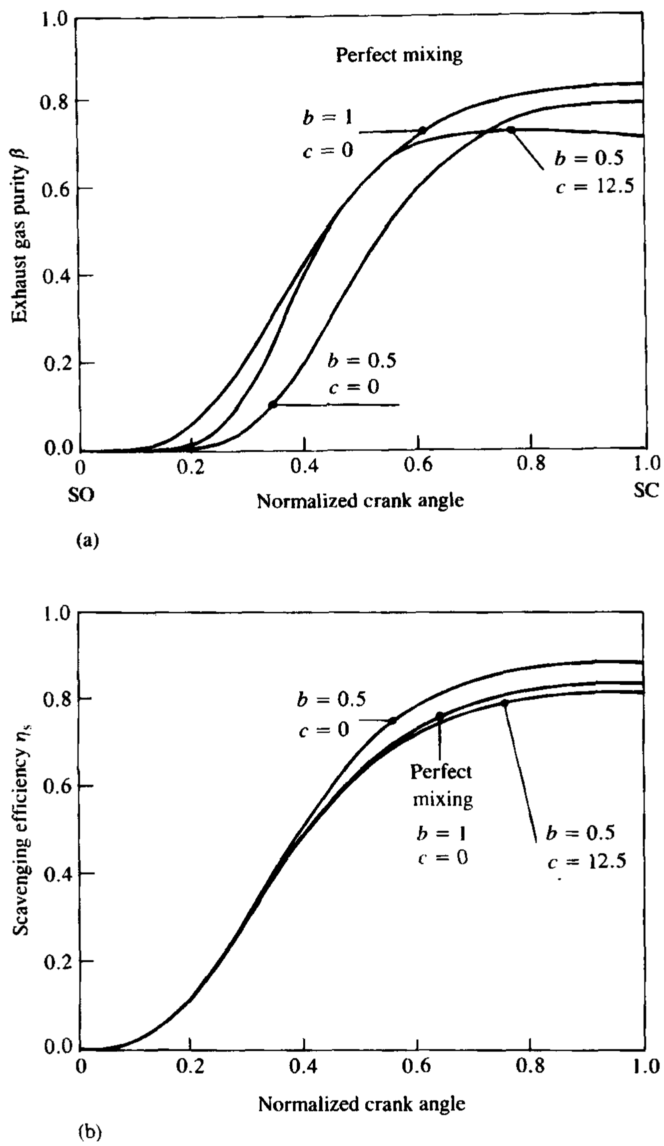

Fig. 4 The effect of the mixing and short-circuiting degree (b and c) on (a) the exhaust gas purity and (b) the scavenging efficiency using the constant volume model with the first type of delivery ratio (Fig. 3)

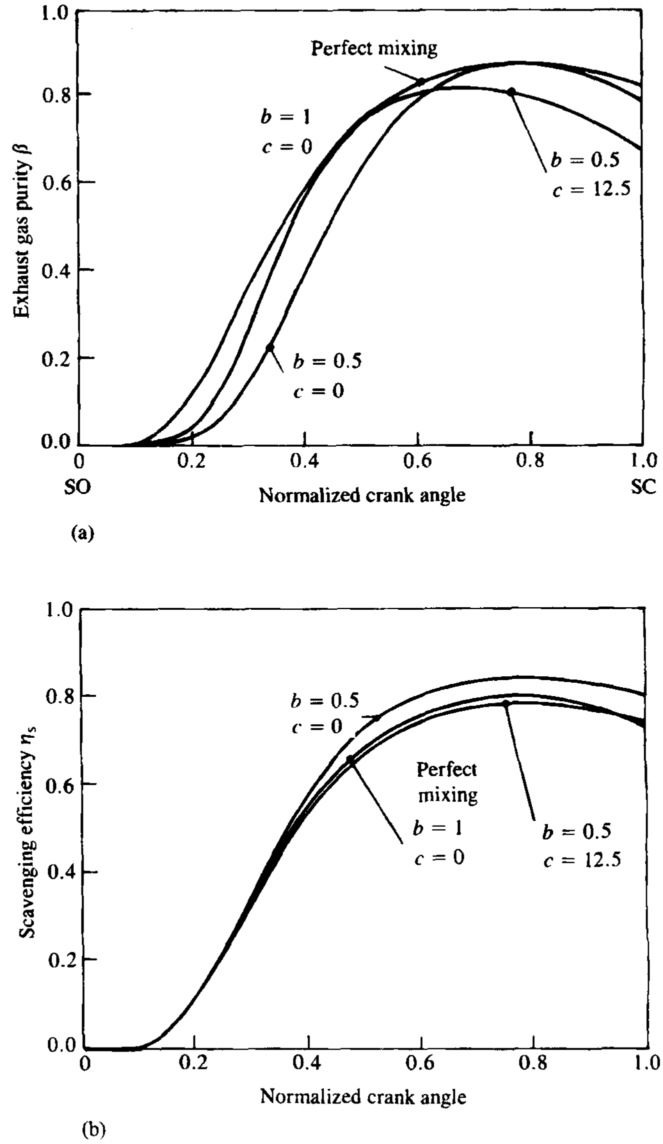

Fig. 5 (a) The effect of the mixing and short-circuiting degree (b and c) on (a) the exhaust gas purity and (b) the scavenging efficiency using the constant volume model with the second type of delivery ratio (Fig. 3)

forming a rounded protuberance, as observed elsewhere (**4**, **6**). For $c = 0$ (no short-circuiting) the purity maximizes, as expected, at $\tau = 0.785$, where the exhaust gases flow inward. In this period, the scavenging efficiency is moderately deteriorated (Figs 4b and 5b).

3.2 The comprehensive model

In the real engine, the mass flowrate of the incoming charge does not necessarily follow a simple cosine curve. Rather, its shape is a complicated function of the operation conditions. The instantaneous mass flowrate is determined by not only the time-dependent geometry of the scavenging and exhaust ports but also by the pressure differences between the crankcase volume, cylinder and ambient, and by the engine speed.

In order to evaluate the proposed model under more realistic conditions as compared with the constant volume restriction, a comprehensive computer program to simulate the complete cycle of a crankcase-scavenged two-stroke engine has been employed. The algorithm for solving the model equations is based on a previous work of the authors (**8**). However, in the present work, a different approach has been adopted as follows:

1. Assuming that the entering mass, which does not short-circuit to the exhaust port, undergoes a perfect mixing process with the cylinder contents, the rate of the pressure change inside the cylinder is calculated by

$$\frac{dP}{dt} = \frac{(\dot{Q} - \dot{m}_e C_p T + \dot{m}_i C_p T_i)(k - 1) - kP\, dv/dt}{v} \tag{16}$$

where $\dot{Q} = \dot{Q}_c + \dot{Q}_w$, $\dot{Q}_c$ is the rate of heat release during combustion, $\dot{Q}_w$ the rate of heat transfer from the cylinder wall to its contents, k the specific heat ratio and $\dot{m}$ the mass flowrate through any port.

2. The instantaneous mass flowrate $\dot{m}$ through any port in any direction (including backflow) was determined by

$$\dot{m} = A_{ef}\left(\frac{2kP_H\rho_H g}{k-1}\right)^{1/2} y^{1/k}(1 - y^{(k-1)/k})^{1/2} \quad (17)$$

where $y = P_L/P_H$ for sonic flow, $A_{ef} = C_D A$, A is the instantaneous port area, C_D the coefficient of discharge which is a function of the opening fraction (**11**) and H and L denote upstream and downstream conditions respectively.

3. The instantaneous heat interaction between the cylinder contents and its confined walls was calculated by using the empirical expression of Annand for a two-stroke engine (**12**) as follows:

$$\frac{\dot{Q}_w}{A} = 0.76\frac{K}{D}\left(\frac{u_p D}{\gamma}\right)^{0.64}(T - T_w) + 0.54\sigma(T^4 - T_w^4) \quad (18)$$

where K is the thermal conductivity of the gas, γ is its kinematic viscosity, u_p the piston velocity, D its diameter, σ the Stefan–Boltzmann constant and T_w the temperature of the inner side of the cylinder wall. For the present calculations, the temperature was taken as 600 K.

4. The mass fraction of the burned mixture at any time was calculated by using a Wiebe function as follows:

$$X = 1 - \exp\left[-n\left\{\frac{(\theta - \theta_{ig})}{\Delta\theta_b}\right\}^m\right] \quad (19)$$

where θ and θ_{ig} are the crack and the start of the combustion angles, $\Delta\theta_b$ is the crank angle interval from start to completion (95 per cent) of combustion, and m and n are calibration parameters which were taken as 3.0 and 5.0 respectively.

5. The crank angle interval from the spark onset to the start of combustion, as well as the interval from start to completion of combustion, were estimated by using the proposed correlations of Hires *et al.* (**13**) (the one-third law):

$$(\theta_{spark} - \theta_{ig}) \sim N^{1/3}u_b^{-2/3} \quad (20)$$

and

$$\Delta\theta_b \sim N^{1/3}u_b^{-2/3} \quad (21)$$

where N is the engine speed and u_b is the laminar burning velocity which was calculated by using the correlation of Metghalchi and Keck (**14**):

$$u_b \sim T^\alpha P^\beta(1 - 2.1f)\phi' \quad (22)$$

where α, β and ϕ' are functions of the equivalence ratio and f is the residual mass fraction ($f = 1 - \eta_s$).

6. The friction power was estimated by using fundamentally based scaling laws for reciprocating spark-ignition engines, as recommended by Patton *et al.* (**15**). Friction from pistons, friction from piston rings, which include the effect of the increasing cylinder pressure, and friction from the connecting rod bearings were taken into consideration. These are represented by the first seven terms of the comprehensive expression (**15**) for the total engine friction mean effective pressure (f.m.e.p.).

7. The composition of the cylinder content during the scavenging period was analysed for CO_2, CO, H_2O, H_2, O_2 and unburnt fuel molecules, by considering two fast reaction steps (**16**). The first is a global reaction between the fuel and the oxygen molecules, which is assumed to proceed to completion, while the second is a fast reaction; a conversion from CO_2 to CO is as follows:

$$CO_2 + H_2 \rightarrow CO + H_2O \quad (23)$$

The equilibrium constant is a strong function of the temperature. The evaluation of the exhaust gas composition is therefore based on the proportions between the cylinder gas and the short-circuited gas that composes the exhaust outflow.

3.3 Results of the comprehensive model

Figures 6 and 7 show the effect of the mixing degree b and the short-circuiting degree c, on the predicted exhaust gas purity and the scavenging efficiency. In general, a lower mixing degree and a lower short-circuiting degree result in lower charging losses and therefore a higher scavenging efficiency. An increase in the short-mixing degree results in the augmentation of the rounded protuberance on the 'S' shape back, thus forming a raised crest—an effect that is prominent at lower engine speeds for which ε in equation (8) is higher. An appropriate selection of b and c can produce almost any curve between the perfect mixing and the perfect displacement processes. A practical procedure for calibrating the model is to set the value of the mixing degree b to fit the measured exhaust gas purity at the end of the scavenging process, and then to set the short-circuiting degree c to fit any other measured value near the bottom centre. This technique has been performed to fit a β curve to a calculated curve as predicted by a detailed computer program (**17**) for three different makes of engines; some results are summarized in Table 1. The engine specifications are given in Table 2. The results of the present model for the scavenging efficiency

Table 1 Comparisons between the present model results and predictions of a detailed computer program (**17**) for three different make two-stroke engines at 4500 r/min and 100 kPa intake pressure. (See Table 2 for the engine specifications.) Here, BC and SC stand for bottom centre and scavenge closes respectively

	Engine		
	1	2	3
Detailed model results:			
β at SC	0.81	0.86	0.77
β at BC	0.67	0.72	0.61
λ at SC	0.61	0.71	0.53
Calibration of the present model:			
b	0.58	0.61	0.57
c/c_{max}	0.81	0.86	0.78
Comparison:			
η_s at BC			
Detailed model	0.76	0.84	0.80
Present model	0.78	0.83	0.78
η_s at SC			
Detailed model	0.87	0.93	0.87
Present model	0.89	0.93	0.86

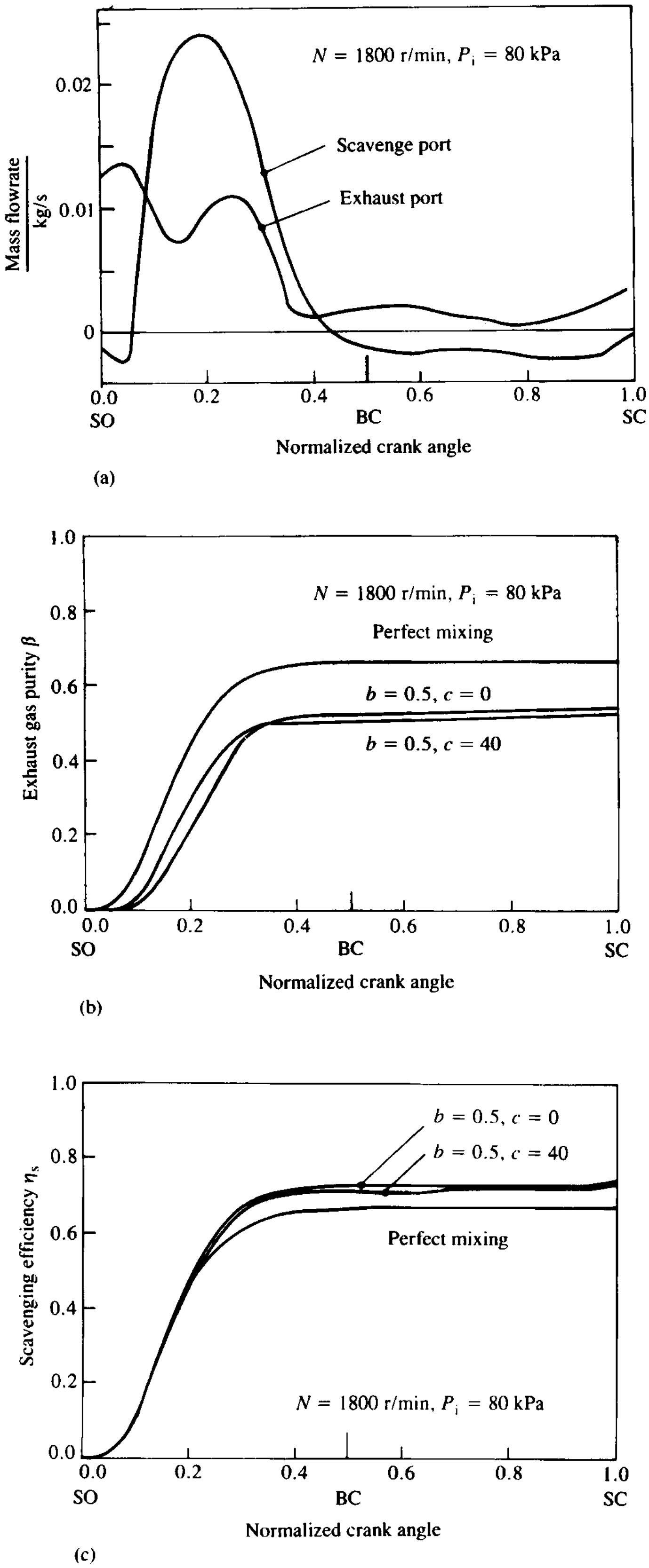

Fig. 6 (a) Mass flowrates through the scavenge and exhaust ports at an engine speed of 1800 s/min and intake pressure of 80 kPa, as calculated by the comprehensive model. The effect of the mixing and short-circuting degree (b and c) on (b) the exhaust gas purity and (c) the scavenging efficiency at an engine speed of 1800 r/min and intake pressure of 80 kPa as calculated by the comprehensive model

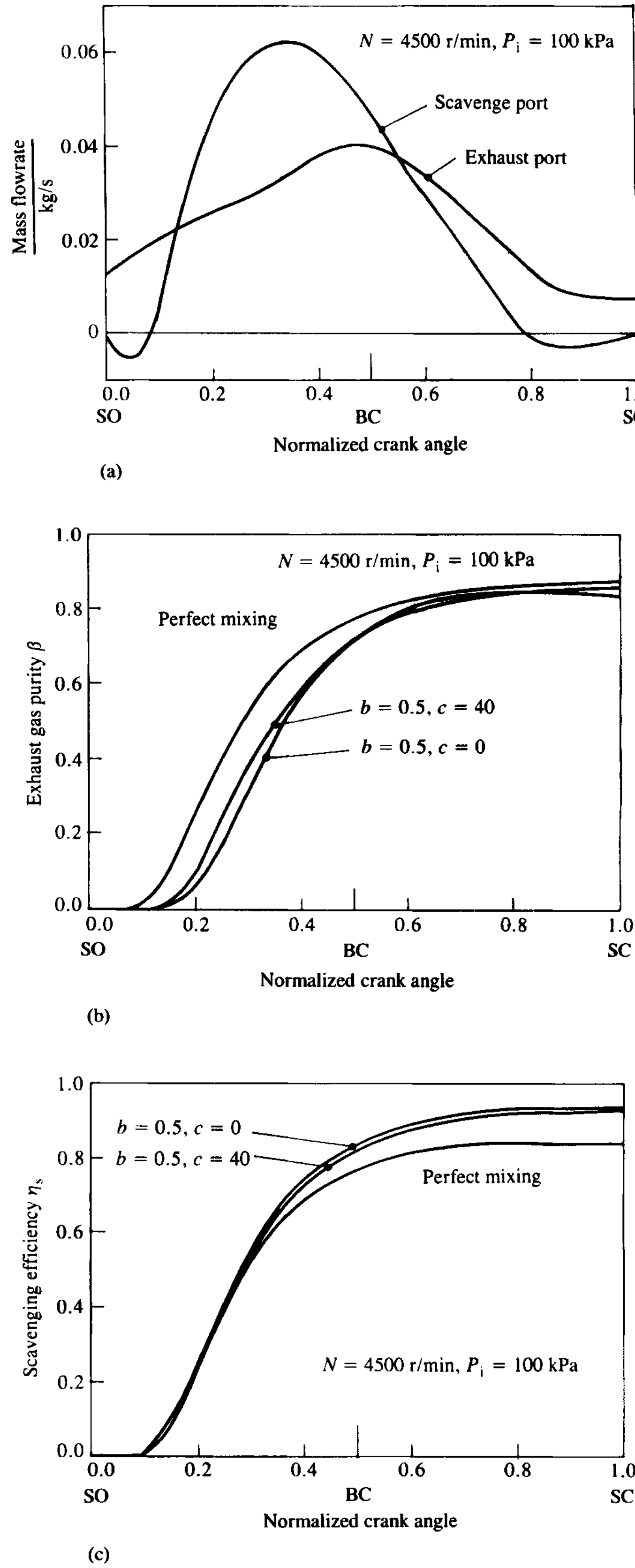

Fig. 7 (a) Mass flowrates through the scavenge and exhaust ports at an engine speed of 4500 r/min and intake pressure of 100 kPa, as calculated by the comprehensive model. The effect of the mixing and short-circuiting degree (b and c) on (b) the exhaust gas purity and (c) the scavenging efficiency at an engine speed of 4500 rpm and intake pressure of 100 kPa as calculated by the comprehensive model

Table 2 Engine specifications. The three engines are crankcase-scavenged two-stroke, spark-ignition engines having a piston controlled port and a loop scavenging of the Schnürle type

	Engine		
	1	2	3
Manufacturer	Karl-Schmidt	USSR	Limbach
Type	STIHL 075	—	L 275 E
Number of cylinders	1	1	2
Swept volume (cm^3)	111.0	124.3	136.8 × 2
Crankcase/swept volume	3.5	5.3	3.2
Compression ratio	8	7.6	8
Cylinder bore (cm)	5.80	5.18	6.60
Stroke of piston (cm)	4.20	5.90	4.00
Port timing:			
Exhaust port open	67	67	66.5
Scavenge port open	56	56	53
Port width (cm):			
Exhaust	3.25	2.65	3.4
Scavenge	2.4	1.55	3.2

at both times (BC and SC) show an excellent agreement with the predictions of the detailed model. This agreement was obtained by setting the mixing degree b to 0.59 ± 0.02 and the short-circuiting degree, c/c_{max} to 0.82 ± 0.04, where c_{max} is determined by equation (13). It should be noted that for modern engine design, that is for the first and third engines, the best fitting was obtained with $b = 0.57$ and 0.58, and $c/c_{max} = 0.81$ and 0.78 respectively. Table 3 shows the effect of the engine load and engine speed on the selection of the best fitting values for b and c/c_{max}. It is clearly seen that the selection does not depend on the engine speed nor on the engine load. Rather it appears that the primary parameter which affects the charging losses at off-design engine speeds is the deterioration of the delivery ratio. It is therefore concluded that, in qualitative terms, the mechanism of scavenging the burnt gas is mainly dependent on the engine design configuration and possibly, but only to a limited extent, on the operation conditions. This conclusion is also supported by another observation (**18**) in which it was found that the characteristics of the gas exchange process are an inherent property of the engine design, while the deterioration of the scavenging efficiency under off-design operation conditions is attributed to the reduction in the delivery ratio.

Table 3 The effect of the engine speed and engine load on the selection of the best fitting values for b, the mixing degree, and $c' = c/c_{max}$, the short-circuiting degree

Intake pressure (kPa)	80		100	
Engine speed (r/min)	1800	4500	1800	4500
Engine 1:				
Total delivery ratio (λ)	0.31	0.75	0.29	0.52
Mixing degree (b)	0.56	0.57	0.57	0.58
Short-circuiting degree (c')	0.83	0.83	0.82	0.81
Engine 2:				
Total delivery ratio (λ)	0.33	0.72	0.37	0.70
Mixing degree (b)	0.62	0.60	0.59	0.61
Short-circuiting degree (c')	0.83	0.84	0.87	0.86
Engine 3:				
Total delivery ratio (λ)	0.31	0.75	0.29	0.53
Mixing degree (b)	0.54	0.58	0.56	0.57
Short-circuiting degree (c')	0.77	0.78	0.76	0.78

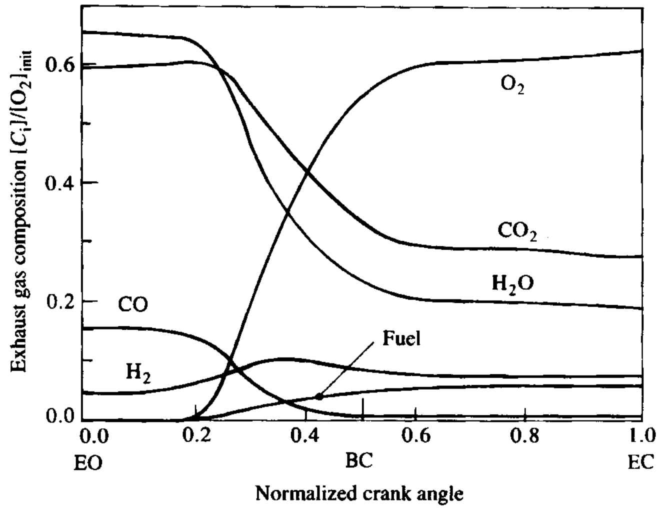

Fig. 8 Exhaust gas composition versus time as calculated by the comprehensive model for engine 2 under the following conditions: engine speed = 4500 r/min, intake pressure = 80 kPa, equivalence ratio = 1.1, mixing degree $b = 2$ and short-circuiting degree $c = 50$

Figure 8 shows some calculated profiles of the exhaust gas composition versus time. At exhaust opens (EO), the exhaust gas is composed of purely burnt gas and it contains CO, CO_2, H_2O and H_2 molecules. As the time advances, the exhaust purity increases and the O_2 molecules become the main component. Unsurprisingly, the O_2 profile resembles the β function profile. In fact, one of the most accurate methods for evaluating the scavenging efficiency in a fired two-stroke engine is to situate a sampling valve in the exhaust pipe just outside the exhaust ports and to trace the oxygen content over time.

The overall performance of one of the examined engines (2), as calculated by the comprehensive model, is presented in Fig. 9. Superimposed are some experimental results. While it shows that the short-circuiting degree has a limited effect, though still significant, the mixing degree seems to dominate the engine performance. However, as the exhaust purity function is much more sensitive to these two parameters, as compared with the overall performance of the engine, the calibration of b and c would be better based on measurements of the exhaust gas purity function.

4 CONCLUSIONS

A simple and realistic model for the scavenging process in a two-stroke cycle engine has been proposed. The model is based on the assumption that typically the time variation profile of the exhaust gas purity exhibits a sigmoid-type curve, while a rounded protuberance usually appears on its back, thus forming a 'crest' profile. The hump has been interpreted as the occurrence of an extensive short-circuiting. An exponential function has been suggested to fit this curve, from which the scavenging efficiency has been derived. The model equation suggests that the scavenging process may be interpreted as a combination of a perfect dis-

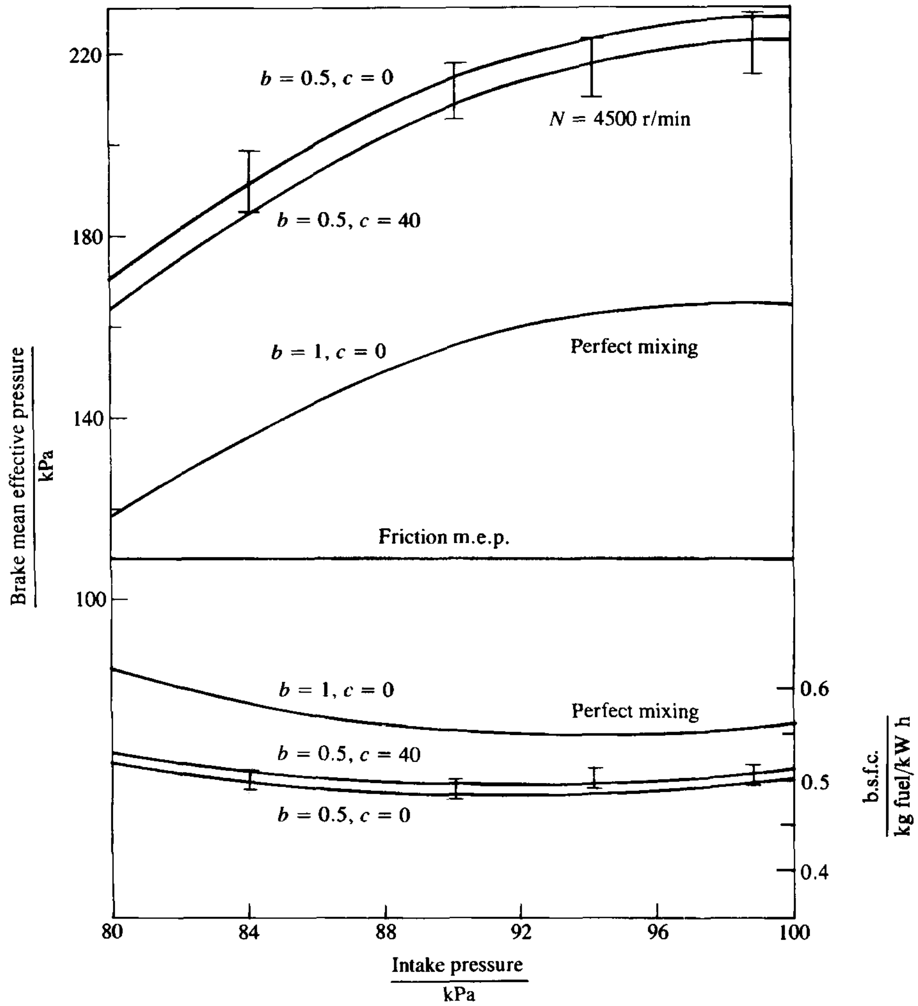

Fig. 9 Brake mean effective pressure, friction mean effective pressure and brake specific fuel consumption versus intake pressure, as calculated by the comprehensive model for the operation conditions of Fig. 8. Superimposed are some experimental results

placement scavenging minus some charging losses. The type of process is determined by the calibration parameters—the mixing degree b and the short-circuiting degree c. In order to evaluate the effect of these parameters and to calibrate the proposed model, two approaches have been adopted: the constant volume and the comprehensive approaches. Based on this evaluation, the following conclusions were drawn:

1. An appropriate selection of b and c can produce almost any curve between the perfect mixing and the perfect displacement processes.
2. An increase in the short-circuiting degree results in the augmentation of the rounded protuberance on the 'S' shape back, thus forming a raised crest—an effect that is prominent at lower engine speeds.
3. A practical procedure to calibrate the model is to set the value of the mixing degree b to fit the measured exhaust gas purity at the end of the scavenging process, and then to set the shirt-circuiting degree c to fit any other measured value near the bottom centre.
4. Calculations of the scavenging efficient were found to be in excellent agreement with predictions of a detailed computer program for three different makes of engines. For modern engine design, a value of 0.57 is recommended for b and 0.81 for c/c_{max}.
5. The selection of b and c does not depend on the engine speed nor on the engine load, but rather b and c are inherent properties of the engine design.

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