JOURNAL OF PROPULSION AND POWER
Vol. 17, No. 3, May–June 2001

Shape Parameterization and Optimization of a Two-Stroke Engine

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We demonstrate a procedure for designing and optimizing the scavenging properties of a two-stroke engine. A method of surface design, known as the partial differential equation method, is used to represent the internal geometry of the engine. We then run steady-state computational fluid dynamics calculations to assess the scavenging flow inside the engine. To quantify the scavenging, the trapping efficiency is used, and calculated values of this measure are obtained for different geometrical designs. We use a locally weighted regression-smoothing model to fit the data, and then we optimize the resulting response surface. The optimization captured two local minima, one of which corresponds to accepted manufacturing practice and the other one is a new and counter-intuitive way of scavenging the engine, having optimum scavenging characteristics.

Nomenclature

C	=	contour
M	=	mass of air
n	=	number of particles
r_0	=	bore of the cylinder
u, v	=	surface parametric coordinates
V	=	volume of air
$\boldsymbol{X}$	=	vector of Cartesian coordinates of surface points
$\boldsymbol{X}_n$	=	normal derivative along the boundary
x, y, z	=	Cartesian coordinates
xsu	=	derivative parameter of the transfer port, in x direction, on C_0
ysl	=	derivative parameter of the transfer port, in y direction, on C_1
ysu	=	derivative parameter of the transfer port, in y direction, on C_0
zsu	=	derivative parameter of the transfer port, in z direction, on C_0
α	=	smoothing parameter
ρ	=	density

Subscripts

s	=	supplied
t	=	trapped
0, 1	=	upper and lower surface boundaries, respectively

Introduction

THE use of two-stroke engines has been mainly promoted by their simple design and manufacture, as well as their high specific output and compact size. On the other hand, their main drawback is their tendency to mix the incoming charge with the combustion products and the passage of the charge directly into the exhaust system. This results in high hydrocarbon emission levels and poor fuel economy.[1,2] To overcome these shortcomings, the fluid-dynamic interaction between the incoming air/fuel mixture and the combustion residuals, called the scavenging process, needs to be studied and optimized.

There are a number of methods for scavenging, depending on the flow path in the cylinder and the position of the transfer ports. The main methods are cross, uniflow, and loop scavenging. The loop-scavenged engine, on which most interest has concentrated, has various layouts in which the transfer ports (two or more) are located symmetrically on either side of the exhaust port. Furthermore, the objective is to produce a scavenging flow directed toward the side of the cylinder farthest away from the exhaust port.[2]

A first approach to investigate the scavenging of a two-stroke engine can be made with Computational Fluid Dynamics (CFD) for predicting the gas flow in the cylinder. In the past decade CFD has been used extensively in this area of research, and several validations of CFD multidimensional calculations have been provided, the most detailed of which are given by Raghunathan and Kenny,[3] Amsden et al.[4] and Haworth et al.[5]

Raghunathan and Kenny[3] used the commercially available software package STAR-CD to simulate the flow in a motored, two-ported loop scavenged engine by modeling the transfer duct, cylinder, and exhaust duct. The results indicated that the flow within the cylinder is highly complex and that the scavenging is affected by the reverse flow through the exhaust port. The in-cylinder velocity field was accurately predicted after the closure of the transfer port. Also, major differences in the predicted and the measured flows were noted, first within the transfer port because of flow separation and second because the flow attachment in the cylinder wall was not replicated.

Amsden et al.[4] used KIVA-III to calculate the three-dimensional flow of a motored crankcase-scavenged two-stroke engine. Transfer ducts, cylinder, and exhaust port were modeled. Predictions were compared with the measurements of Fansler and French.[6] The in-cylinder velocities and pressures were found to agree with the measured values.

Haworth et al.[5] used a multidimensional calculation of scavenging in a loop-scavenged engine to compare the computed mean and rms velocities with laser Doppler velocimetry (LDV) measurements of the same quantities in a commercially available engine under normal engine speed and flow conditions. The geometry model included six transfer ports, cylinder, exhaust duct, and a simplified crankcase.

In all of these cases described, the multidimensional CFD modeling has been validated by comparison with the experimental measurements. However, even where CFD calculations have been used as a cheaper alternative to experiment, the cost in terms of computational effort has prohibited their extensive use as a design tool.[2]

Another related problem is the need to represent the geometry of the engine efficiently with as small a set of parameters as possible, but at the same time to have sufficient control of its shape to produce an improvement in design. With the introduction of the partial differential equation (PDE) method[7,8] in geometric design, it is feasible to design and alter the geometry of a two-stroke engine significantly by changing only a small set of design parameters. The method has been previously used to demonstrate the ability to generate the critical elements of the internal geometry of a two-stroke engine.[9]

Received 27 November 1999; revision received 11 April 2000; accepted for publication 26 June 2000.

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Also, steady-state CFD calculations have been performed and the crucial effect that the transfer port design has on the scavenging performance has been investigated.

The aim of our work is to expand this earlier research by developing a fully automated process that could be used at the concept design stage to determine a configuration that is near optimal. A much-reduced calculation to expensive experimental validation would then follow. This, in conjunction with the ability to rapidly alter the geometry, gives us the opportunity to consider shape optimization in a small parameter design space. However, to perform this automated procedure, a quantitative measure of the scavenging performance of a two-stroke engine is needed.

Jante in her pioneering work[10] introduced a method of assessing the scavenging efficiency of a two-stroke engine. She proposed to monitor the distribution of the scavenging air velocity above the open cylinder and to represent it on a contour velocity graph. She stated that "the scavenging currents must so interact with the piston crown and the cylinder wall that a stable closed rising current is obtained on the wall opposite to the exhaust ports, having its maximum velocity at the wall and near zero velocities on the line perpendicular to the plane of symmetry and the cylinder axis."[10] With the use of that idea she was able to distinguish "good" from "bad" scavenging but no quantitative measure was determined.

Blair and Kenny[11,12] improved Jante's method by providing an automated way of collecting and processing the data required by the method. They also proposed several quantities for measuring the scavenging ability of an engine. These are the flow-symmetry ratio (FSR), mixing-length ratio (MLR), scavenged-area ratio (SAR), and the dimensionless scavenge strength number (DSSN). However, all of these parameters provide an indirect measure of scavenging. Therefore, in this paper, a more direct measure based on trapping efficiency[2] (TE) is introduced and implemented.

In this paper, first the design of a two-stroke engine by the PDE method is discussed. Then we give the results of steady-flow calculations, explaining the capability of intuitively generating new geometries by altering a small set of parameters. Then the trapping efficiency is defined and the effect it has on the scavenging is explained. Finally, we use optimization techniques to examine the scavenging performance by altering two of the design parameters that may be regarded as the most important.

PDE Method

Here a brief description of the method for parameterizing the engine geometry is given. The PDE method was introduced in the area of computer-aided geometric design as a method for blend generation.[7] It was then extended to generate free-form surfaces,[8] and its applicability to the design of a wide range of surface shapes and objects, such as ship hulls,[13] aircraft,[14] and propeller blades,[15] has been demonstrated.

The PDE method views surface generation as a boundary-value problem, and it creates complex surfaces from a collection of individual surface patches $X(u, v)$. Also, it regards each surface patch

$$X(u, v) = [x(u, v), y(u, v), z(u, v)] \tag{1}$$

as a solution to a PDE for each of the Cartesian coordinates x, y, and z, which are parameterized in terms of the independent variables u and v. The function X can be seen as a mapping from a point (u, v) in the parametric space $\Omega \subseteq R^2$ into a point (x, y, z) in the Euclidean space $E^3 \subseteq R^3$. In general X is a solution to a PDE of the form

$$D^m_{u,v}(X) = F(u, v) \tag{2}$$

where $D^m_{u,v}$ is a partial differential operator of the order of m in the independent variables u and v and F is a vector function of u and v. Because we want X to be generated from boundary curves surrounding the patch and because it is required to be smooth, the differential operator $D^m_{u,v}$ is chosen to be elliptic. In that way $D^m_{u,v}$ acts as a smoothing process, giving X a value that is an average of its neighboring weighted values. The actual weights and the details of the averaging process depend on the operator D.

The PDE used most often in previous work[8,16] is

$$\left(\frac{\partial^2}{\partial u^2} + \alpha^2 \frac{\partial^2}{\partial v^2}\right)^2 X = 0 \tag{3}$$

A solution of Eq. (3) requires boundary conditions on X and its normal derivative X_n along the boundary of the domain $\partial\Omega$. The derivative conditions X_n, which are usually expressed in terms of the coordinate vectors X_u and X_v, can be used either as a design tool, giving us the ability to specify the direction and speed in which the surface propagates away from the boundary, or to impose continuity between adjacent patches. Also, the parameter α controls the relative smoothing between the u and the v directions and is called the smoothing parameter.[7]

A method[16] has been introduced that allows PDE patches to be obtained in closed form, even in the case of general boundary conditions, providing us with a fast surface-generating technique. In particular, the surface patch may be approximated by the expression

$$X(u, v) = A_0(u) + \sum_{i=1}^{N} [A_n(u)\cos(nv) + B_n(u)\sin(nv)] + R(u, v) \tag{4}$$

where N is finite and $R(u, v)$ is a remainder term derived from its form of the boundary conditions. The coefficient functions A_n and B_n have the following form:

$$A_0(u) = a_{00} + a_{01}u + a_{02}u^2 + a_{03}u^3 \tag{5}$$

$$A_n(u) = a_{n1}e^{\alpha n u} + a_{n2}ue^{\alpha n u} + a_{n3}e^{-\alpha n u} + a_{n4}ue^{-\alpha n u} \tag{6}$$

$$B_n(u) = b_{n1}e^{\alpha n u} + b_{n2}ue^{\alpha n u} + b_{n3}e^{-\alpha n u} + b_{n4}ue^{-\alpha n u} \tag{7}$$

where a and b are constant vectors. These are determined by Fourier analysis of the boundary conditions and a comparison of the Fourier coefficients with the values of the functions A_n and B_n.

The remainder function $R(u, v)$ is determined in such a way that the patch satisfies exactly the boundary conditions imposed.[16]

Therefore a complex surface is built from a series of individual patches joined at their mutual boundaries. In our case, the two-stroke engine geometry has been based on that of a small chain-saw engine.

Geometry Definition

In this section a brief description of the geometry used for the two-stroke head is given. The dimensions of the engine are given in Fig. 1, but for a detailed description, the reader is referred to Dekanski et al.[9]

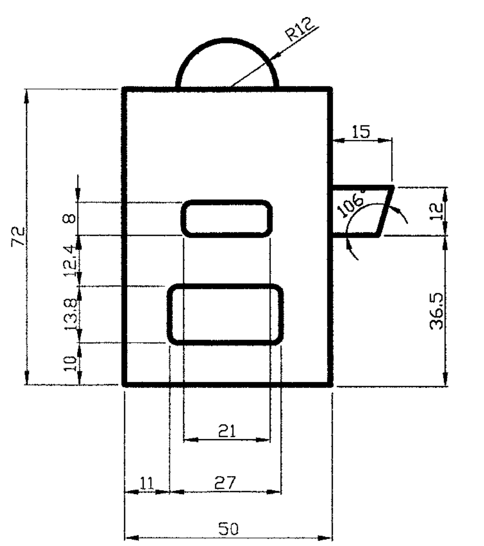

Fig. 1 Main dimensions of the cylinder (in millimeters).

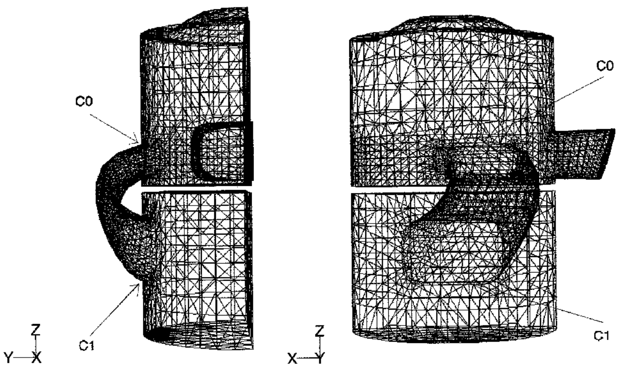

Fig. 2 Geometry of the two-stroke engine.

For the fluid-flow analysis, it is sufficient to model the internal geometry of the two-stroke engine that, as can be seen in Fig. 2, consists of the upper and the lower parts of the cylinder, the exhaust, and the transfer ports. The geometry has been simplified in that the inlet port is assumed to be closed, and so is not modeled. Also the piston is assumed to be flat, and a vertical plane of symmetry has been introduced to cut down on computational time because that does not affect the results.[17] Thus the flow enters the lower part of the crankcase and propagates to the upper part of the cylinder through the transfer ports. Then it scavenges the cylinder by pushing the combustion residuals into the exhaust.

Thus the geometry of the two-stroke engine has been created by adding together the surface of the cylinder (represented conventionally) and the exhaust and the transfer port, both generated as a solution of Eq. (3) by the imposition of appropriate boundary conditions. Furthermore, at the intersection of the ports with the cylinder geometry, the boundary nodes of the port and the cylinder have been connected consistently so as to ensure that no holes appear in the model.

The exhaust port has been generated by specification of a contour on the surface of the cylinder and a similar one on the exit of the exhaust. These contours are appropriately rectangular in shape with radiused corners. With these as boundary conditions, the exhaust patch is produced by the PDE method.

As is well known,[2] the transfer port has a great impact on the general performance of the scavenging. In our model its surface has been generated as a solution of Eq. (3) subject to position $\boldsymbol{X}$ and derivative $\boldsymbol{X}_u$ boundary conditions specified at the two contours C_0 and C_1 (Fig. 2).

For the positional boundary condition, C_0 and C_1 are defined as isoparametric lines $u = 0$ and 1, respectively, and they are given by

$$\boldsymbol{X}(0, v) = \left[0.09r_0 + 0.75xx,\ \left(r_0^2 - x^2 - z^2\right)^{\frac{1}{2}},\ c_0 - zz\right] \tag{8}$$

$$\boldsymbol{X}(1, v) = \left[0.9xx,\ \left(r_0^2 - x^2 - z^2\right)^{\frac{1}{2}},\ -c_1 + 2zz\right] \tag{9}$$

where

$$xx = 1.5\cos(v) - 0.12\cos(3v) \tag{10}$$

$$zz = 1.5\sin(v) + 0.2\sin(3v) \tag{11}$$

r_0 is the bore of the cylinder and c_0 and c_1 are constants. In a similar fashion we impose derivative boundary conditions $\boldsymbol{X}_u(0, v)$ on the contour C_0 and $\boldsymbol{X}_u(0, v)$ on C_1, which are described simply by

$$\boldsymbol{X}_u(0, v) = [xsu, ysu(2 - z), zsu] \tag{12}$$

$$\boldsymbol{X}_u(1, v) = (0, ysl, 0) \tag{13}$$

By changing these parameters, xsu, ysu, zsu, and ysl, we can create a finger or cup-handled transfer port without any need of redesigning the whole engine. Also, this straightforward approach provides us with a wide variation of flows if we just alter these

Table 1 Parameter values

Parameter	Value
xsu	4.0
ysu	3.0
zsu	1.5
ysl	1.2

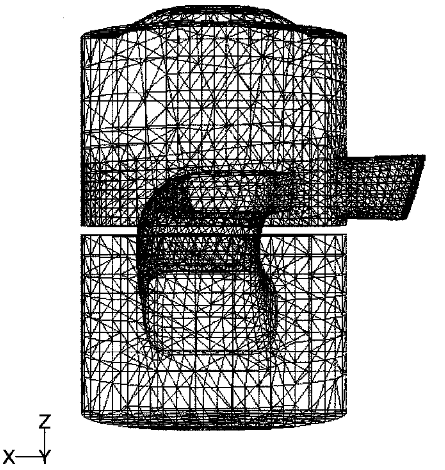

Fig. 3 Geometry of the two-stroke engine for $xsu = -4$ and $zsu = 0$.

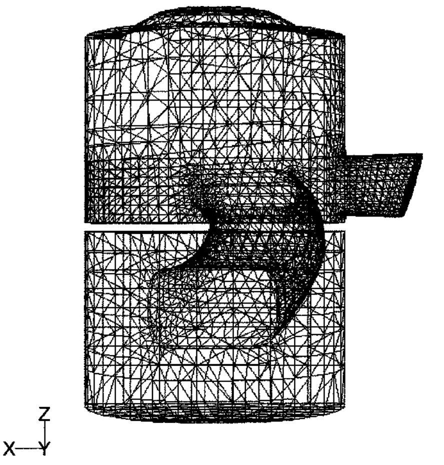

Fig. 4 Geometry of the two-stroke engine for $xsu = 4$ and $zsu = 3$.

parameters. For example, we have generated two geometries by altering only the two parameters xsu and zsu. In the first case we have chosen $xsu = -4$, $zsu = 0$ and in the second one $xsu = 4$, $zsu = 3$. As can be seen in Fig. 3, the transfer port has been directed toward the exhaust port, which in the second case, Fig. 4, is toward the back of the cylinder and up to the dome. This example demonstrates the ease with which we can generate a two-stroke engine with different transfer port angles.

We can make further changes in the geometry by altering the shape of the boundary curves or the position at which the transfer port intersects the cylinder. Both of these will result in an increased number of design parameters, and so are ignored in the interest of reducing computational time. Thus, for the purpose of this paper, we restrict ourselves to changing only the angle of the transfer port, i.e., the four parameters.

In Table 1, the values of the parameters for the geometry specified in Fig. 2 are illustrated.

Steady-Flow Analysis

In recent years, as computational power has increased, papers have appeared in the literature based on multidimensional calculations and comparisons with fired engine tests.[3–5] Although these

Table 2 Boundary conditions

Boundary conditions	Value
Inlet velocity (m/s)	2.13
k (m^2/s^2)	1
ϵ (m^2/s^3)	1
Temperature (K)	300

papers carry out unsteady simulations of complete engine cycles, they are extremely costly in computational time, and thus their approach is prohibitively expensive when used in the context of numerical optimization.

A more convenient approach is to identify the favorite design among various different geometries on the basis of a steady-flow calculation and then to produce a prototype in which to perform firing tests in order to determine the quality of scavenging.

Our approach is to calculate a steady flow through the cylinder in which the piston has been fixed at the bottom dead center, thus keeping the transfer and exhaust ports continuously open. This is based on the tests performed by Jante[10] and the improvements proposed by Blair and Kenny.[12] Also quantitative support for this kind of technique has been provided by the extensive LDV results presented by Fansler and French,[6] in which a two-stroke engine under motored conditions has been tested.

For the fluid flow analysis, the commercial CFD package Fluent v4.1 is used. First, the geometry is created as described in the previous section. Then the preprocessor PreBFC is used to read the geometry and produce an unstructured surface mesh, consisting of triangular faces in three dimensions. The result is used by Tgrid, which is the grid generation facility of Fluent, to produce and refine an unstructured triangular/tetrahedral grid consisting, in our case, of ~17,100 cells and having an overall skewness of 0.39. Finally the solver Fluent/UNS (unstructured) is used to estimate the fluid flow in the domain defined by the grid. UNS is based on a finite-volume technique, and a second-order upwind scheme has been used for the solution of the conservation of mass and momentum. Also, because the flow into the cylinder is highly turbulent, the k–ϵ model has been implemented. To predict the flow into the engine, UNS needs to be provided with boundary conditions on the walls and inlet and outlet surfaces as well as the fluid properties.

In Fluent, all the boundaries through which the fluid enters/leaves the computational domain are considered as inlets/outlets. All the other surfaces, except the symmetries, are considered walls and the no-slip condition is applied.

In our model, the surface cutting the cylinder on the plane $z = -0.047$ (see Fig. 2) is defined as an inlet. Also, the exit of the exhaust is considered an outlet.

A velocity inlet condition has been imposed in the inlet boundary and an outflow in the outlet. In the first case a velocity vector at the inlet plane is required, whereas at the outflow a zero normal gradient for all flow variables except pressure is assumed. The values of the boundary conditions being used can be seen in Table 2 and are in agreement with the previous research carried out in the same geometry by Dekanski et al.,[9] who used values for calculations of actual engine designs. Also in the simulation procedure, air has been used as fresh charge for the engine.

Figure 5 illustrates the flow of a few particles of fuel through the engine designed in the previous section. As can be seen, the scavenging of the engine is poor because most of the fresh charge is directed straight into the exhaust. This increases the amount of short circuiting, and it is one of the things we want to reduce, because it directly affects the hydrocarbon emissions.[18] The time needed for one simulation was ~90 min in Indy R4600 SC Silicon Graphics.

Assessment of Scavenging

Calculating particle flow tracks for a steady flow can be used to give an insight of the scavenging performance of an engine. However, if we use optimization methods to improve scavenging, then a quantitative measure is required.

Blair and Kenny[12] have proposed several quantities to measure the scavenging performance of an engine, such as the FSR, the MLR,

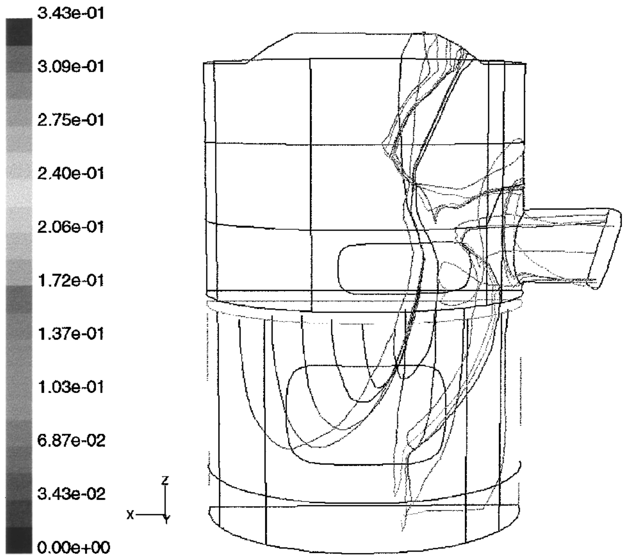

Fig. 5 Particles flow through the engine, colored by particle residence time(s).

the SAR, and, most importantly, the DSSN. All of these are based on the initial suggestion of Jante for ideal scavenging in which she proposed recording the distribution of the scavenging air velocity just above the open cylinder and plotting the data in a contour velocity map.[10] Furthermore Blair and Kenny have shown that if Jante's method is carried out with their quantitative measure, the DSSN, an improvement in scavenging can be achieved. More evidence for the effectiveness of Jante's method has been provided by Sanborn and Roeder[19] and Ishihara et al.[20] Although Jante's method has been extensively used in the design of two-stroke engines during the past 30 years, because it is not an absolute measure of scavenging-efficiency-to-scavenge ratio or trapping-efficiency-to-scavenge ratio, it does not provide a direct comparison among the scavenging flows conducted in the enormous variation of different geometries in existence.[2] So, in order to compare the scavenging of the two-stroke engine, the TE has been chosen as a measure of merit.

The TE is defined as the capture ratio of mass of delivered air that has been trapped M_t to that supplied M_s.[21] Thus, assuming that V_t/V_s is the volume of the trapped/supplied air and n_t/n_s is the number of particles trapped/supplied, each of volume V_p, we have that

$$\mathrm{TE} = \frac{M_t}{M_s} = \frac{\varrho_{\mathrm{air}} V_t}{\varrho_{\mathrm{air}} V_s} = \frac{n_t V_p}{n_s V_p} = \frac{n_t}{n_s} \tag{14}$$

So, in order to measure the TE, a mechanism of estimating the number of trapped particles n_t for each of the steady-flow calculations performed is needed.

In a steady-flow calculation, a number of marker particles can be injected into the flow, giving us details of their coordinates as time progresses. So, assuming that t_0 can be used as an appropriate time scale, a direct comparison between the position of the particles and the position of the exhaust plane can give us the number of the particles trapped n_t in the cylinder after the engine cycle, and so providing us with the TE of the engine.

For the purposes of optimization, the inlet surface has been divided into small increments and 192 particles of anthracite, equally spaced, have been injected into the flow. Further, these particles have a constant diameter of 10^{-6} m and a mass of 8.11×10^{-16} kg. Then the residence time for each of the particles has been calculated and compared with the reference time of $t_0 = 0.02$ s, which is the time needed for a stroke when the engine operates at 1500 rpm.

Thus, by measuring the number of particles that have left the cylinder after some specific time, we may try to optimize the scavenging performance of the two-stroke engine under consideration by maximizing the TE or, in other words, by minimizing the quantity $1 - \mathrm{TE}$.

Optimization

Now, to reduce the computational time of the optimization, we restrict ourselves by taking into consideration only the shape parameters xsu and zsu that have the greatest impact on the geometry of the transfer port. By varying xsu, we can direct the flow toward and away from the exhaust port. On the other hand, by changing zsu, we direct the transfer port upward toward the dome or downward toward the base of the cylinder. Furthermore, we bound these parameters into the region of interest:

$$-4.0 \leq xsu \leq 4.0 \tag{15}$$

$$0.0 \leq zsu \leq 3.0 \tag{16}$$

The first experiments carried out revealed that the objective function, i.e., the TE, varies nonsmoothly with its shape parameters and contains noise that is due to the numerical simulation of the complex flow in the engine. Therefore a naive application of a gradient-based optimization method, such as quasi-Newton BFGS (Broyden-Fletcher-Goldfarb-Sanno)[22] could not be applied in our problem. A different approach to analyzing noisy data is to fit a surface to the data points locally, so that at any point the surface depends on only the observations at that point and a specific number of neighboring points.[23] Then, having a smooth objective function, we can perform optimization by using any standard technique. This is sometimes an attractive solution because it relies on the data to specify the form of the response surface, yet it does not require an advanced optimization method in order to avoid noise.

With this approach, 211 observations for different geometrical designs in the domain of interest were made and the residence time for 192 particles recorded in each case. The locally weighted regression-smoothing model[24] was implemented, and the function produced can be seen in Fig. 6.

It should also be noted that changes in the number of particles have negligible effect on the calculation of the TE. Similarly, when k–ϵ boundary conditions are altered, no significant changes are observed. Further, although an increase in the node density of the mesh does result in a significant change in the TE, the overall behavior of the fitted surface remains the same; in other words, the same trends are observed and the same optimum found.

Results

As can be seen from the graph of Fig. 6, there are two local minima in the domain of interest, both of them lying within the boundaries of the domain. This has been confirmed by optimization applied to the smoothed objective function that provide us with the local minima values, presented in Table 3. Although both of them occur for almost the same value of the shape parameter zsu, the value of xsu in these two cases, is diametrically opposite. In the first case (minima A), the transfer port is directed toward the exhaust port whereas in the second (minima B) the port is directed toward the back of the cylinder.

Comparing these two points, we can see that point A is the minima of the domain of interest, providing us with optimal values for the parameters xsu and zsu. The fluid flows of both of these have been calculated, and the results are presented in the form of particle tracks in Figs. 7–10.

In Figs. 7 and 8, which correspond to minima B in Fig. 6, the transfer port is directed away from the exhaust port and toward the middle of the dome. Therefore the flow goes up to the dome, coming back to the left of the transfer port and going down the cylinder before moving toward the exhaust. This results in a flow that scavenges the whole of the cylinder and intuitively has been

Table 3 Local minima of the objective function

Variable	Minima A	Minima B
xsu	−4.0	2.1
zsu	2.7	3.0
1 − TE (%)	1.12	3.18

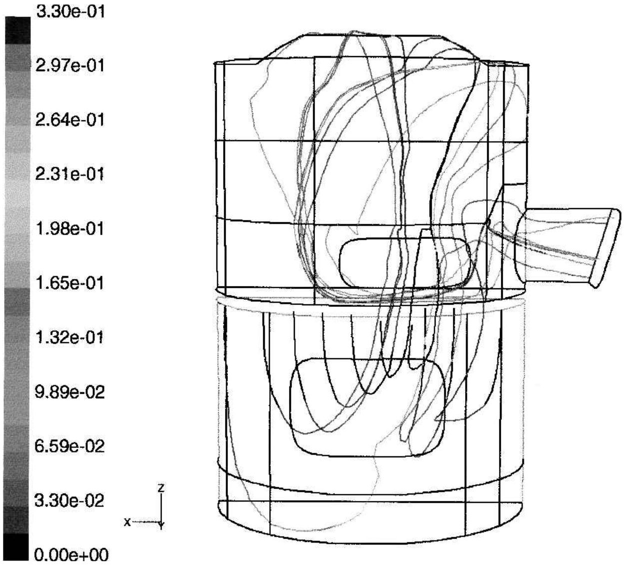

Fig. 7 Fluid flow through a transfer port directed into the back of the cylinder (xsu = 2.1 and zsu = 3.0).

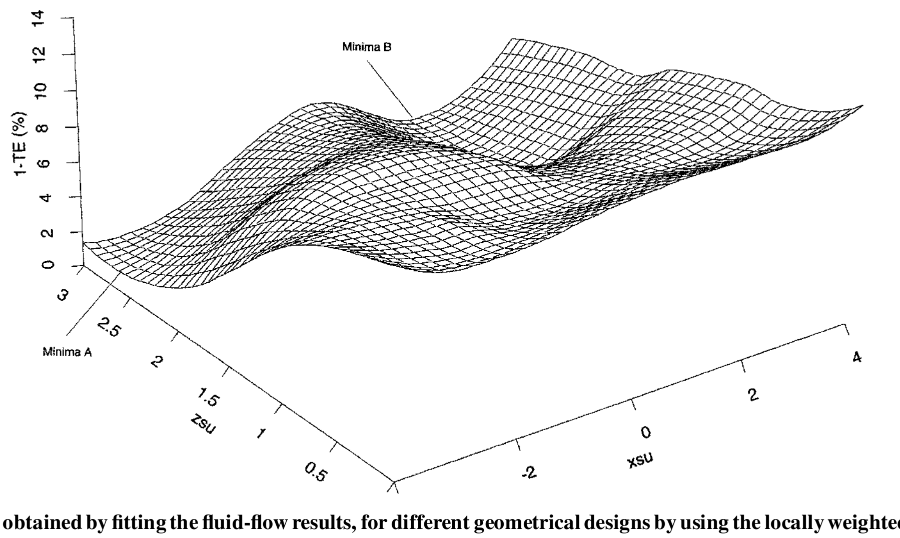

Fig. 6 Surface of TE, we obtained by fitting the fluid-flow results, for different geometrical designs by using the locally weighted regression-smoothing model.

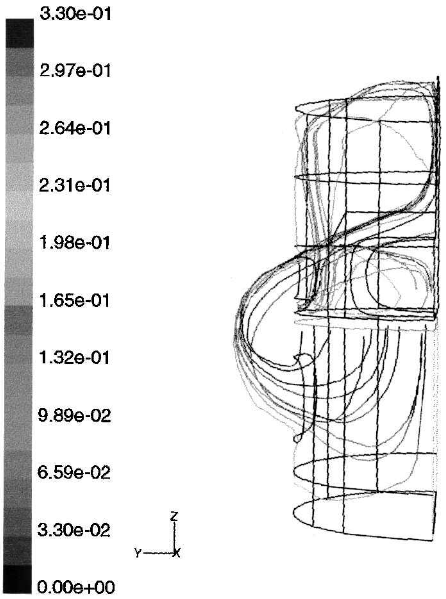

Fig. 8 Fluid flow through a transfer port directed into the back of the cylinder (xsu = 2.1 and zsu = 3.0).

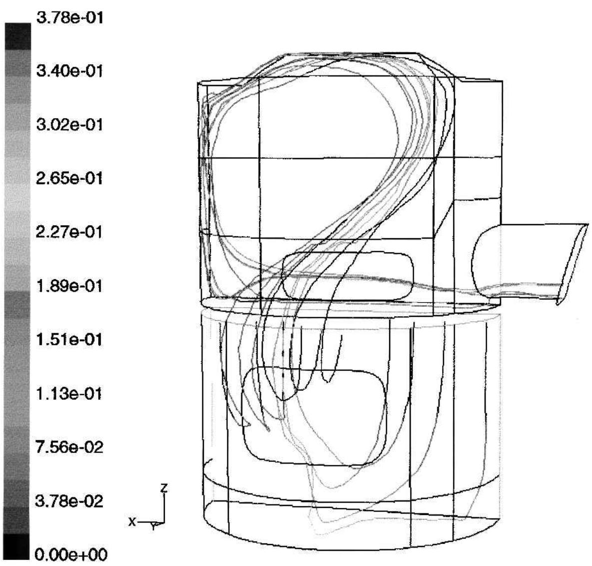

Fig. 9 Fluid flow through a transfer port directed toward but above the exhaust of the cylinder (xsu = − 4.0 and zsu = 2.1).

expected to be the best case in the domain of optimization and corresponds to accepted manufacturing practice.

On the other hand, the case of minima A gives a rather unexpected result. As is illustrated in Figs. 9 and 10, the flow is directed toward but above the exhaust port, then follows the dome to the back part of the cylinder, going down the back wall and then to the exhaust at the end, suggesting a new efficient way of scavenging the two-stroke engine. Following this path, the flow obeys the first principles of scavenging, that is, it does not create short circuiting and it pushes the burnt gases into the exhaust without mixing with them. Therefore a qualitative comparison of the scavenging between the two cases (minima A and B) based on the fluid-flow calculations performed illustrates the way in which the flow behaves and confirms the initially counterintuitive configuration A as optimal.

It is worth emphasizing that this result does not conform to accepted design practice. As has been already mentioned, in the designing of two-stroke engines, the transfer port is directed toward the wall opposite the exhaust, rising up on the back of the cylinder and through the dome, and finally moving downward to the exhaust.

A qualitative difference in the fluid flow between the two optimum

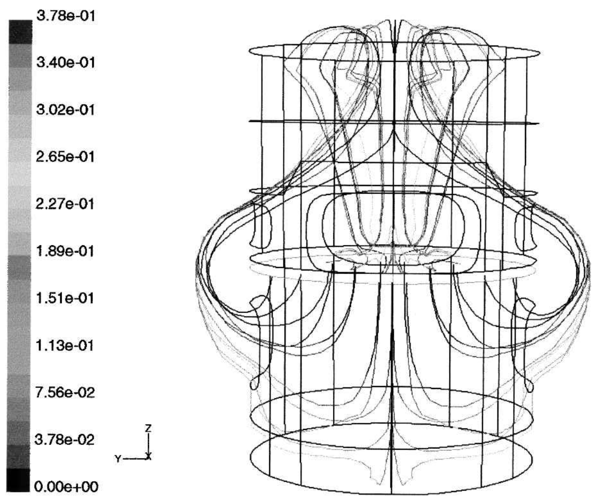

Fig. 10 Fluid flow through a transfer port directed toward but above the exhaust of the cylinder (xsu = − 4.0 and zsu = 2.1).

cases is that, in case A, the particles move in such a way that they tend to displace and not mix with the combustion residual, whereas in case B, there appears to be more likelihood of mixing.

Conclusions

We have presented a method for the automated design of a two-stroke engine that gives us the ability to change several parameters and to produce different geometries with a great variety of fluid flows in the cylinder. The cylinder, transfer ports, and the exhaust have been incorporated into the geometric model. From Dekanski et al., four important parameters affecting the transfer port and directing the flow into different parts of the cylinder have already been considered.[9] We have limited ourselves to the variation of two of these parameters, and, by introducing a measure of merit related to the TE of the engine, we were able to compare and distinguish the performance of different geometries.

By calculating the value of our objective function TE at different points in the domain of interest, we were able to fit a smooth function to the whole domain, which provided us with a overview of the performance of the different engines. Further, an optimization technique was applied to capture numerically the two local minima, which have been observed on the graph of the objective function. One of the minima, referred to as B, corresponded to the usual scavenging mechanism discussed in the literature, namely, to direct the transfer ports away from its exhaust port. However, minima A, in which the transfer port is directed toward the exhaust, actually gave better scavenging characteristics to the engine. This conclusion was supported by the flow diagram produced to examine the validity of the result and confirm the belief that a new efficient way of scavenging the engine has been found. It remains to be seen whether the result can be validated by fired engine tests.

The procedure suggested is fully automated and can be used to examine the effectiveness of different two-stroke engines, especially in cases in which there is lack of experimental results. It provides the engineer with a quick and effective method to model and alter the geometry of the engine as well as a small set of parameters that makes it feasible to perform optimization techniques.

Acknowledgment

The authors wish to acknowledge the support of Engineering and Physical Sciences Research Council (U.K.) Grant GR/L05730 (“the efficient parametrization of mechanical parts”).

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