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Partial Differential Equation Surface Generation and Functional Shape Optimization of a Swirl Port

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In this paper we demonstrate how a novel approach to surface design may be used for the representation of a generic swirl port geometry. The advantage of this methodology, referred to as the partial differential equation (PDE) method, lies in the rapid generation and speed of alteration of geometries, coupled with the fact that a small parameter set controls the geometry. In the first part of the paper, we illustrate how a generic swirl port may be generated, which, when analyzed using computational fluid dynamics (CFD) techniques, is shown to give performance characteristics similar to the geometries of existing designs. In the second part of the paper we discuss performance characteristics, in particular we deal with the swirl generated in the flow. We discuss how steady-state CFD calculations may be used to calculate established rule-of-thumb performance measures and illustrate how the PDE geometry may be easily manipulated to show the impact the geometry has on the performance measures. Finally, we demonstrate the benefits of using the PDE method for functional design, specifically how the control of the swirl port geometry with a small number of parameters enables us to use numerical optimization to obtain an improved performance measure by altering the surface geometry.

Introduction

WITH the ever-increasing demand for greater efficiency in internal combustion engines, there is a need to improve the engine's operating characteristics. The flow structure and its development in the cylinder during the intake stroke is strongly influenced by the geometry of the engine head, the inlet valves, and the inlet port. In particular, in a compression ignition engine it is important to control the flow of the air motion and fuel injection from the swirl port into the chamber, so that rapid combustion can occur near the end of the compression stroke. In many engines this is achieved with an ordered rotation of the flow about the chamber axis, a process known as swirl, which is one of the most important factors in the design of engine configurations since it helps to produce mixing between the entrained air and fuel, thus enabling rapid combustion.[1]

In practice, the starting point in the development of a new engine head is an established design for which there is available performance data. A blowing box is constructed, through which air is blown to produce a steady flow and from which measurements such as nondimensional swirl ratios and pressure drops across the valve seat are extracted.[2] If results prove unsatisfactory, then a new geometry must be created and the test rig used again. This iterative process can take a great deal of time and effort. However, there is a growing trend for computational fluid dynamics (CFD) to play an ever-increasing role in the development of new designs.[3] For instance, with the increase in computational power, it is frequently of great benefit to use CFD analysis as a preliminary design tool to assess the potential of new (and existing) designs. If the analysis of a proposed design indicates an improved performance, then the manufacturer may consider constructing a prototype for validation on a test rig as before.

The prerequisite of a CFD analysis is, however, to generate a CAD model of the geometry. This may often take weeks or months depending on the complexity of the engine head. Often, on producing the final CAD surface model using polynomial-based systems, holes are found to be present between adjacent surface patches, which prevent a reliable computational mesh being generated automatically. Software frequently includes tools to stitch such models, but this takes additional time and such procedures are not usually automatic. What is needed is a CAD system that can build geometric models that remain closed as the design variables are altered. Furthermore, it is preferable that the number of design variables is small and that they have an intuitively obvious effect upon the geometry. The bulk of the time required to produce a new design will then primarily depend upon the time required for the CFD analysis, which can be minimized with the acquisition of suitably powerful hardware.

In this paper we illustrate how the partial differential equation (PDE) method may be used to produce such a system by generating the surface model of a swirl port. The PDE method regards surface generation as a boundary-value problem, in which control of the surface is achieved by a set of parameters introduced through boundary conditions imposed around the edges of a surface and through the choice of PDE. This implies a small number of parameters (relative to other surface generation techniques) controlling the geometry. A complicated surface is generated by joining together a collection of individual surface patches that meet at a network of curves that forms the boundaries between patches. The surface geometry is completely closed because of boundary conditions, ensuring positional and higher-order geometric continuity between adjacent patches, which thus removes any need for stitching of surface models prior to analysis, so that, as the geometry is changed, associativity between adjacent patches is automatically maintained. Finally, it should be noted that the resulting surfaces are of a form compatible with STL format (basically a triangulation of the surface), which is widely used for the importing and exporting of surface data to/from software packages for analysis and manufacture. This fact, coupled with the closed geometry ensures that volume meshing may be

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achieved automatically by the software package used in this work.

In experimental work, steady-state analysis of the engine's performance is often performed to obtain design criteria that are less difficult to determine than results from a full unsteady analysis.[1] However, the criteria are chosen so that they give results that are consistent with actual experience. We therefore use steady-state fluid flow analysis to achieve performance measures similar to those obtained on experimental test rigs. Analyses are performed on several models and the results compared to see the performance variation afforded by changes in the design parameters.

Having only a small parameter set to define and manipulate the geometry enables us to carry out automatic design optimization. However, a full optimization over all parameters is still, given the complexity of the physical problem, computationally expensive, and so we choose to vary only those design parameters that we feel have the strongest influence on the flow through the swirl port. A suitable measure to quantify engine performance is chosen based on the swirl number and manufacturing constraints imposed on the system. These may be required, for example, to limit the volume occupied by the swirl port, since cooling jackets need to be fitted around the engine head. In this case we use penalty functions to control the space over which the PDE parameters are allowed to vary. An improved design is then sought by numerical optimization over the parameter set. In the second case, physical constraints are incorporated to maintain a feasible pressure drop and the optimization rerun to investigate the difference this has on the resultant geometry.

We commence by explaining the PDE method.

PDE Method

The PDE method was first developed as a method of generating blend surfaces,[4] from which it progressed to free-form surface design by relaxing the usual constraints placed on a blending surface (positional and tangent plane continuity at the edge of the blend) to obtain more scope for surface manipulation.[5] To establish its usefulness for free-form surface modeling, work has been carried out on the design of functional surfaces, such as aircraft,[6] propellers,[7] and ship design.[8] In addition, data transfers between the PDE surface representation and other types (e.g., B-splines) have been investigated.[9]

The PDE method regards the generation of a surface patch $\underline{X}(u, v) = [x(u, v), y(u, v), z(u, v)]$ as the solution of an elliptic PDE

$$\underline{D}^{m}_{u,v}\underline{X} = \underline{F}(u, v) \tag{1}$$

typically over a rectangular region of a (u, v) parameter space in R^2. The partial differential operator $D^{m}_{u,v}$ of order m is chosen to be elliptic since this ensures that $\underline{X}(u, v)$ is smooth and can be found by posing conditions around the boundary of the domain. The elliptic operator acts as a global-averaging process between the posed boundary conditions.[10] The boundary conditions typically take the form of $\underline{X}$ and its derivatives with respect to u or v, expressed as parametric functions of u and v. The order of the equation is chosen to accommodate the number of boundary conditions required. For example, if we only specify positional boundary conditions, i.e., $\underline{X}$, then a second-order PDE is solved, whereas for G^1 continuity, we require tangent plane continuity that implies specifying positional and derivative boundary conditions and thus a fourth-order PDE is needed. The derivative boundary conditions can be used to control the direction and rate of propagation of the surface away from the boundaries.

In much of the previous work, and indeed in this paper, we have concentrated upon solutions of the PDE

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

which is solved independently for x, y, and z. In general, this equation must be solved numerically; however, a method has been developed which, even for general boundary conditions, allows the rapid solution of this equation.[11]

There are two main classes of swirl port: 1) the helical and 2) directed ports. In the helical port, swirl is generated before entering the combustion chamber and is thus termed prevalve swirl. The swirl is achieved by the spiral shape of the swirl port with the pitch controlling the magnitude of the angular momentum imparted to the flow. In a directed port, the swirl is generated by channeling the airflow tangentially around the inlet valve with the subsequent impingement on the cylinder walls inducing the spiral motion. This is often referred to as postvalve swirl, the magnitude of which depends on the slope of the port relative to the cylinder head.[12] In general, the directed port gives larger swirl numbers; however, it also produces lower discharge coefficients because of the unequal air distribution at the valve.[2]

A geometric model that is capable of representing the features of both the helical and directed port is therefore necessary, since if we cannot represent a wide range of geometries then we are restricting the range of possible designs when we perform shape optimization.

The basics of the swirl port and inlet duct generation have previously been described by Bloor and Wilson.[13] However, following discussions with engine designers, certain improvements were made to the geometry of the swirl port and it is on these improvements that we concentrate.

The swirl port, valve guide, and valve are described by five surface patches joined to form a closed surface. Each surface patch was generated by solving Eq. (2) over the region $0 \le u \le 1$, $0 \le v \le 2\pi$ subject to appropriate boundary conditions.

The boundary conditions that produce the main surface of the swirl port are of the form

$$\underline{X}(0, v) = \begin{bmatrix} xd \\ yd + y1 \sin v + y3 \sin 3v \\ zd + z1 \cos v + z3 \cos 3v \end{bmatrix}$$

$$\underline{X}(1, v) = \begin{bmatrix} r \cos v \\ r \sin v \\ 0 \end{bmatrix}$$

where the boundary curve at $u = 0$ corresponds to the swirl port/combustion chamber interface, and $u = 1$ corresponds to the inlet duct swirl port interface, as illustrated in Fig. 1. The variables r, xd, yd, zd, $y1$, $y3$, etc., control the location and shape of the trimlines. The derivative boundary conditions are specified in terms of the partial derivatives $\underline{X}_u$ by

$$\underline{X}_u(0, v) = \begin{bmatrix} 0 \\ 0 \\ sz0(1 + szv0 \cos v) \end{bmatrix}$$

$$\underline{X}_u(1, v) = \begin{bmatrix} sx1(1 + sxv1 \cos v) \\ 0 \\ sz1(1 + szv1 \cos v) \end{bmatrix}$$

in which

$$sx1 = \tfrac{1}{2}(sx11 + sx10), \qquad sz0 = \tfrac{1}{2}(sz01 + sz00)$$

$$sxv1 = \frac{(sz01 - sz00)}{(sz01 + sz00)}, \qquad szv0 = \frac{(sz01 - sz00)}{(sz01 + sz00)}$$

and similarly for $sz1$, $szv1$. Note that a hole has been cut in the surface of the swirl port that will form a boundary to which the valve guide is blended.

In the present work the parameters $szv0$, $sxv1$, and $szv1$ have been added, and $sz0$, $sz1$, and $sx1$ are now determined by the maximum and minimum values that each derivative takes across the boundaries. The notation is such that $sz00$ refers to the minimum value of the z derivative on the boundary curve $u = 0$, and $sz01$ refers to the maximum value of the z derivative on the boundary curve $u = 0$, etc. For example, in Fig. 2, $sz00$ has a small value (denoted by the magnitude of its vector) and $sz01$ has a larger value. Whereas in previous work $sz0$ uniformly stretched or contracted the swirl port from the cylinder head, now the z derivative is stronger on the right-hand side than on the left and pushes the outer wall of the swirl port further from the boundary. A similar parametrization affects the duct/port interface with $sx1$ and $sz1$, now varying according to the magnitudes of the vectors shown in Fig. 2. In particular, we are now able to produce surfaces similar to helical and directed ports, in which the internal flowfield can be made to generate pre- or postvalve swirl. Results will later demonstrate this.

Figure 1 illustrates that a hole is cut in the surface in the neighborhood of the z axis, and a valve guide is blended to the edge of the hole. The valve guide is then joined to the cylindrical valve stem, as shown in Fig. 2. The parameters sg and zg in Fig. 2 are derivative parameters introduced at the boundary where the guide meets the valve stem that control the radial expansion and axial stretching of the valve guide as it leaves the valve stem. The geometry of this region strongly influences the internal volume available for the buildup of swirl inside the port. It should also be noted from the two diagrams of Fig. 2 that the surface in the neighborhood of the hole can be made to fold into or protrude out of the swirl port, depending on the parameter zg, as can be found in existing designs.

The generation of the valve is a simple problem: basically it is a surface blend between two coaxial circles aligned with the valve stem axis. The shape of the valve remains fixed in this study, although there is plenty of scope for an investigation based solely on the geometries of the valve and valve seat to improve performance.

The parameters just described are the most influential in terms of manipulating the swirl port geometry. In all there are approximately 50 parameters defining the geometry, i.e., including cylinder and inlet manifold; however many, such as xd, yd, zd, $y1$, $y3$, etc., remain fixed throughout the lifetime of this study.

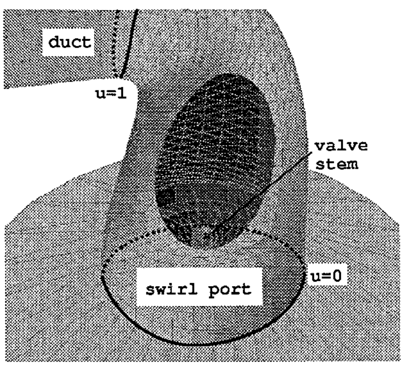

Fig. 1 Defining boundaries of the PDE port.

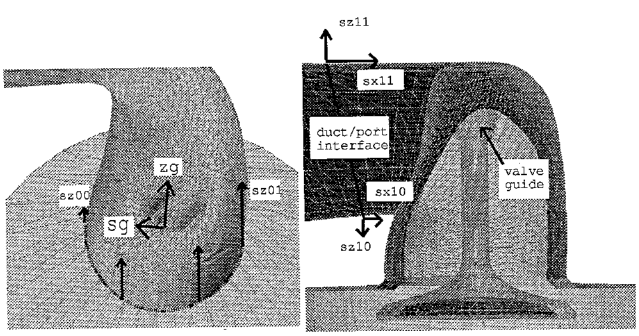

Fig. 2 Parametrization of the PDE port.

Finally, we reemphasize that the generation and manipulation of the geometries are almost instantaneous on an SG workstation, and thus allow a user at a workstation to obtain a feel for the parameters' control by interactively altering the design parameters to create a whole range of geometries. Also, the surface geometry remains completely closed during this process and may be converted to an STL triangulation with little effort for volume meshing prior to CFD analysis or for rapid prototyping of a physical model.

CFD Analysis of Swirl Ports

In the design of the induction system, the two main parameters for engine performance are the flow capacity, which is expressed in terms of a discharge coefficient at the valve exit, and the velocity distribution in the opening between the inlet valve head and seat.[2] In a good design we seek a high value of the discharge coefficient for a specific mass flow rate. Importance is also attached to the production of swirl to improve a diesel engine's performance. However, inducing more swirl in an engine can have an adverse effect on the volumetric efficiency (and, hence, the discharge coefficient) of the engine.

It proves difficult to obtain appropriate measurements in a running engine because of the time-varying pressure differences present. However, a great deal of information can be deduced about the running engine by using a test rig in which the relationship between flow and pressure drop is studied for fixed configurations, such as constant valve lift and piston position for a steady flow.[14]

By running a steady-state analysis we may obtain performance measures akin to those achieved experimentally. For this study we use a geometry that has a fixed valve lift and fixed piston. To maximize flow through this, care is needed to avoid flow separation between the valve head and the valve seat. In general, for a low valve lift, the jet of air from the swirl port will fill the gap between the head and seat, yet for high lift the emerging jet will separate from one of the surfaces. Separation may be avoided by appropriate design of the geometry of the valve seat (rounding of edges) and the valve itself, but this is neglected in the present study.[2] It is possible to predict the best valve lift for induction and exhaust processes using optimization techniques,[15] however, in this study we simply take an intermediate valve lift and concentrate on the geometry of the swirl port.

Choosing appropriate values for the design parameters produces the geometry illustrated in Fig. 3. This was generated to resemble the geometry of a helical port. By altering the parameters associated with the swirl port we are able to produce a larger region for prevalve swirl to be generated. For example, increasing $sx11$, sg, and $sz01$ generates a larger region around the valve guide, with a less steep slope than that of Fig. 2, from the inlet duct to the base of the swirl port.

The code used is a commercially available package, known as VECTIS.[16] It is based on a finite volume approach with a k-ε model for the turbulence. Boundary conditions are needed on walls and inlets/outlets. For the case considered here we

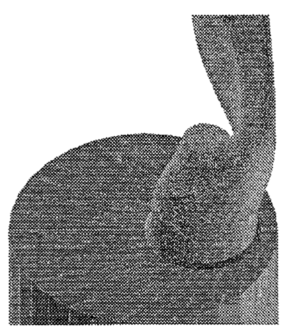

Fig. 3 Initial geometry.

must specify mass flow rate, temperature, and stagnation pressure. The PDE surface geometry is fed straight into the package in STL format from which a volume mesh is automatically generated since the PDE geometry is completely closed. In broad terms the meshing algorithm used by VECTIS is automatic once given a valid geometry, and works by adaptively decomposing the flow volume into an array of rectangular cells of varying mesh sizes, with the level of adaptation chosen by the user with regard to the desired accuracy and the time taken for calculations. For example, it is known that the region between the valve seat and valve guide has a strong impact on the flow in the cylinder, and so we generate a more refined mesh within this region to account for this. Furthermore, studies undertaken by the producers of VECTIS have shown the code to be robust in comparison with experiments,[16] and to produce consistent results between one calculation and another. Because of this the output may be considered reliable in this class of application.

Qualitative information about the swirl port can be obtained by visualizing the flowfield inside the geometry, however, to perform numerical optimization we require a performance measure. As already stated, the discharge coefficient and swirl number are important considerations in the design of a swirl port. Annand and Roe[2] state that for small pressure drops, the discharge coefficient may be defined as the ratio of the effective area to a reference geometric area

$$C_D = A_E/A_r \tag{3}$$

where the effective area A_E is given by

$$A_E = \dot{m}/\sqrt{2\rho_{01}(p_{01} - p_2)} \tag{4}$$

and ρ_{01}, p_{01}, and $\dot{m}$ are air density, pressure, and mass flow rate at upstream stagnation conditions, respectively, and p_2 the downstream pressure.

The reference area used by Uzkan et al.[14] is

$$A_r = \pi d L[1 + (L/d)\sin\theta\cos\theta]\cos\theta \tag{5}$$

where θ is the valve seat angle, L the valve lift, and d the valve diameter. Using VECTIS we may extract information about the pressure distribution inside the combustion chamber, and hence, obtain a value for the discharge coefficient.

To measure the swirl generated on a test rig, the swirling flow is straightened by passing it through a honeycomb arrangement located on a spindle downstream of the combustion chamber entrance. The rotation rate of the honeycomb is taken as a measure of the swirl produced. Of course, this is an idealized situation, since the actual swirl does not behave as a solid body rotation and in different regions the flow may swirl in opposite senses, which would tend to cancel when measured by such an apparatus. However, this measurement arrangement is still widely employed since it gives a good indication of overall engine performance.[1] The option to automatically calculate a swirl number is present in VECTIS.

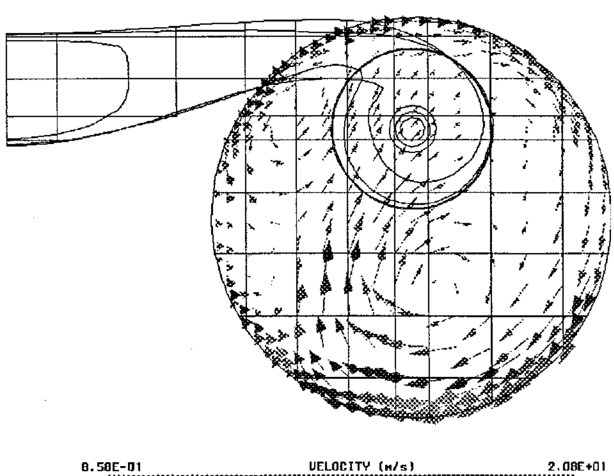

Fig. 4 Swirling flow in-cylinder.

Table 1 Performance of several PDE geometries

Port	$sz01$	$sz00$	$sx11$	$sx10$	$sz11$
1	3.0	0.1	−5.5	−1.5	1.5
2	8.0	1.5	−7.5	−1.5	0.0
3	4.0	0.5	−4.5	−1.5	0.0
4	5.0	1.2	−9.0	0.0	2.0

Port	Pressure drop, Pa	C_D	Swirl no.
1	1800	0.755	125.2
2	1370	0.865	75.20
3	2120	0.696	146.4
4	2470	0.645	183.5

Figure 4 shows results of a typical calculation in the form of the velocity distribution across just one plane of the cylinder for the steady flow. The cut plane is taken at the same location as a swirl meter would be located on a test rig. As can be seen, an organized swirling flow is generated in this generic geometry. However, although the flow appears to be swirling, we notice a region of counterclockwise flow swirling in the opposite sense to the bulk of the flow. To reduce the size of this region is one way to improve the flowfield by optimization of the geometry. Above the valve, the flow is also moving with a swirling motion, and so this geometry can be thought of as a helical port. What may be required is not so much rotation about the valve, but more of a directed flow toward the combustion chamber wall opposite the inlet duct.

Many calculations were performed for different values of the geometry parameters. The choice of parameters to vary was made intuitively on the basis of those expected to produce the largest noticeable changes in the generated flowfield and would therefore significantly influence the production of swirl and pressure drop.

Table 1 illustrates the results obtained for a range of geometries that because of space limitations are not displayed. The magnitudes of the calculated discharge coefficients are of the same magnitude as those found by others and reported in textbooks,[1] however, because of the many different measures used for the swirl number, a comparison of the swirl number produced here with other measures is not possible.[2] Importance is attached to the consistency of the numbers extrapolated from the analysis. Finally, it should be noted that Port 1, referred to in Table 1, corresponds to the geometry shown in Fig. 3.

Thus, for suitable running conditions, we may conclude that the PDE method is capable of producing port geometries that give similar flowfields to those produced elsewhere[14] and comparable performance measures.[2]

Optimization

As it stands the present work may be used as a design tool on which the performance may be improved on a trial-and-error basis as if it were a numerical test rig. This may still provide savings in terms of time and money over experimental facilities. However, by performing computer-aided optimization we may automate the process of design improvements.

As Uzkan et al.[14] state, current swirl ports often combine some of the features of both directional and helical ports and thus produce a high degree of swirl in the flow. For the optimization we therefore select design parameters whose variation will generate geometries ranging from those of directed ports to those typical of helical ports. The parameters chosen for this purpose are $sz01$, $sx11$, $sz10$, zg, and sg, as illustrated in Fig. 2.

The initial design for the optimization is Port 1 in Table 1 and Fig. 3. Next we consider two cases in which constraints have been placed upon the optimization. In the first case geometric constraints are imposed, derived from possible manufacturing considerations, whereas in the second case, physical constraints derived from performance considerations have been used. For the sake of example, we seek to optimize the production of swirl that plays a significant role in the combustion process. Both physical and geometric constraints are imposed in terms of penalty functions. The optimization procedure, along with the penalty functions, are based on the Fletcher and Powell method[17] and is described in earlier work.[7]

Case 1: Geometric Constraints

In this example we describe an optimization in which constraints are placed to limit the range over which the PDE parameters may vary. In particular, they are imposed such that the swirl port may not protrude beyond the circumference of the valve seat. The objective function to be optimized thus comprises the swirl number plus a penalty function that penalizes regions of the parameter space corresponding to undesirable geometries:

$$f_1(x_1, \ldots, x_n) = f_0(x_1, \ldots, x_n) + p_1 + \cdots + p_n \tag{6}$$

where f_1 is the objective function, $x_1, \ldots, x_n$ are the PDE parameters, f_0 is the value of the swirl number output by VECTIS, and p_n is the penalty function of the form

$$p_n = -c_n \sinh(h_n) \tag{7}$$

where c_n is termed the cost function and h_n a difference function (between the maximum value of the parameter and its current value). This form is used since it gives a rapid change in value for p_n for slight changes in h_n. The function c_n is determined so there is no bias toward any one constraint, for example, caused by the relative size of each constraint. If a design alters so that any constraint is broken then the penalty function becomes highly negative; all penalty functions are set to zero if their constraints remain unviolated. Figure 5 illustrates the optimized geometry with the corresponding parameter values displayed in Table 2.

In the optimization the derivative parameters $sx11$ and $sz10$ have taken extreme values beyond which an allowable geometry cannot be achieved. The effect of the parameter $sz10$ has been to deflect the flow toward the outer wall and reduce the cross section at the duct/swirl port interface to increase the speed of the flow. Unlike the previous geometry, there is little volume around the valve guide and little swirl is generated above the valve. However, to generate greater swirl, the geometry is trying to mimic a directed port in that the flow follows the outside wall of the port and is directed through the valve gap onto the far side of the cylinder from the inlet duct, generating a powerful swirling motion about the cylinder axis. Figure 6 shows the velocity distribution across the same cut plane as that shown in Fig. 4.

However, from Table 2 we can see a noticeable drop in pressure, and hence, the discharge coefficient, which is consistent with experiment in that with a directed port we can gain a dramatic increase in swirl at the expense of volumetric efficiency.[13]

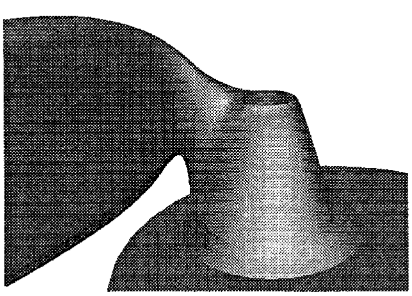

Fig. 5 Directed port geometry.

Table 2 Optimized PDE geometries

Port	$sz01$	$sx11$	$sz10$	zg	sg
Fig. 3	3.0	−5.5	0.0	1.877	−0.523
Fig. 5	0.1	−3.1	−1.8	1.5	−0.3
Fig. 7	3.56	−5.125	−0.12	1.818	−0.488

Port	Pressure drop, Pa	C_D	Swirl no.
Fig. 3	1800	0.755	125.20
Fig. 5	4710	0.467	267.15
Fig. 7	1800	0.755	142.35

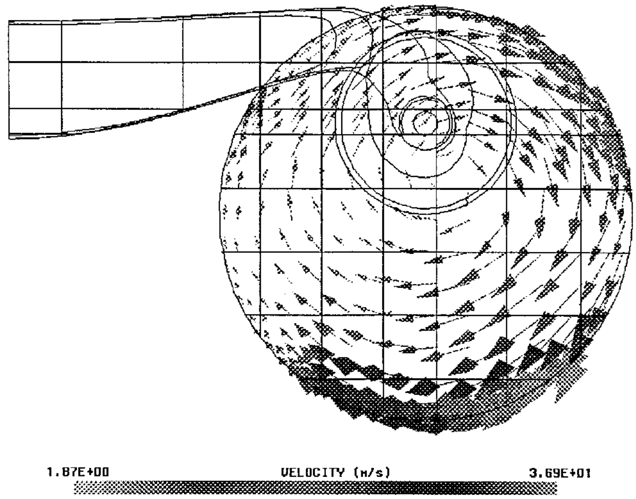

Fig. 6 Directed port velocity profile.

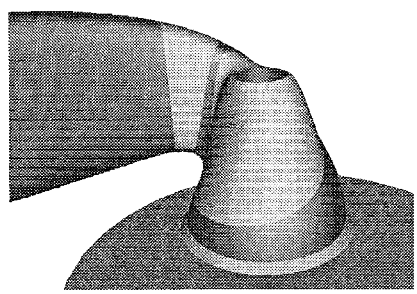

Fig. 7 Improved swirl port geometry with acceptable discharge coefficient.

Case 2: Discharge Coefficient Constraints

In the second optimization we attempt to increase the swirl intensity and yet maintain the discharge coefficient of the inlet system. To do this we rerun the optimization procedure and impose additional physical constraints on the system.

For this calculation we use a new objective function. The function to be optimized is composed of the swirl number plus a penalty function that strongly penalizes regions of the parameter space corresponding to values of the discharge coefficient lower than a specified critical value.

A suitable critical value for the discharge coefficient is 0.755, which corresponds to a pressure drop of 1800 Pa, which is often the specified operational pressure drop on a test rig. For the initial geometry in Fig. 3 this produces a swirl number of 125.2. The optimization is rerun and the results displayed in Table 2 and Fig. 7.

Figure 8 illustrates the velocities through the same cut plane as before. The increase in swirl number over the initial (helical) geometry has been achieved by combining the features of

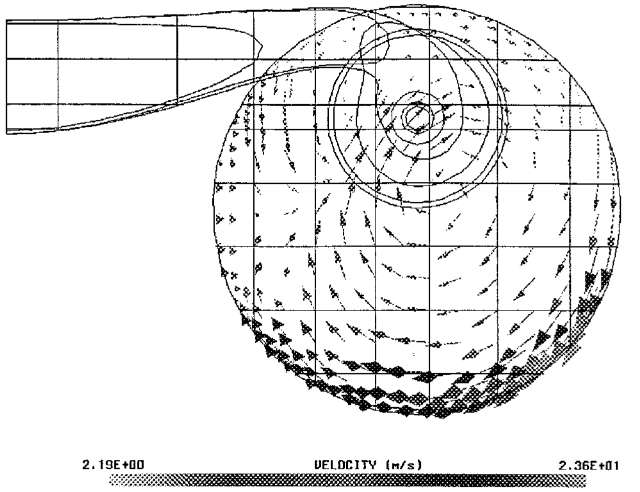

Fig. 8 Improved velocity profile.

a helical port with those of a directed port, i.e., a prevalve swirling flow combined with the slope into the valve region. In fact, in terms of the performance, we have a 14% increase in terms of generated swirl for the same discharge coefficient.

Conclusions

In this work we have illustrated how the PDE method may be used as the basis for a system of modeling, analysis, and the functional design of a swirl port. The ability to achieve this lies in the rapid generation and manipulation of surface geometries by a small design parameter set. Furthermore, by using a surface generation method that controls the geometry in terms of a small parameter set, we have shown that the method enables design optimization and yet still can produce a wide variety of geometries. In addition, the PDE method automatically generates closed surfaces, compatible with STL format, which are often required for volume meshing before CFD analysis.

Thus, the PDE method may be used as an interactive tool for concept design in which a surface geometry may be produced by manipulation of the design parameters which, when coupled with CFD analysis, enables us to determine performance characteristics before physical prototyping.

We have also illustrated that functional design through optimization of the geometry may be accomplished. Depending on which aspect of the port's performance we are seeking to improve we can choose appropriate performance measures to be optimized; for instance, we could choose to optimize the discharge coefficient for a fixed swirl number. In addition we can impose different constraints on the system with great ease to generate different geometries.

Finally, because of a convenient mathematical expression of the geometry, a physical prototype could be manufactured using layered manufacturing technology. The prototype may then be tested on a flow rig to validate the results obtained using CFD. Within the context of computer-aided rapid prototyping this would prove much quicker and cheaper than the conventional approach of altering physical models to find a satisfactory design.

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