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Flow Modulation in Turbulent Vortex Chambers

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ABSTRACT

The modelling of the chamber region of an axial vortex valve by the numerical solution of the equations governing axisymmetric turbulent swirling flow in a right-cylindrical chamber is described.

For this flow it was found that an isotropic turbulence closure model was inappropriate and that it was vital for the anisotropy of the turbulence to be taken into account. The turbulent viscosity for the axial and radial momentum equations was calculated from the two-equation k-ε turbulence model and the turbulent viscosity for the tangential momentum equation was calculated from a mixing length formulation. This model correctly predicts the type of tangential velocity profiles observed in vortex valves; those of the combined vortex type where a forced vortex is embedded in a free vortex. In contrast, the isotropic k-ε model, by itself, only predicts profiles of the forced vortex type and is therefore unsatisfactory.

The results of a number of parametric studies, which focus on the throttling effect observed in vortex chambers, are presented and discussed.

INTRODUCTION

The operation of a vortex valve, as a no-moving-parts valve, is based upon the throttling effect that occurs when a swirling flow passes through a contraction. The effect is most simply explained in terms of the mechanism that is appropriate for an inviscid flow. As the flow passes through the contraction the conservation of angular momentum means that the tangential velocity is magnified and hence the radial pressure gradient is increased. For a given energy source driving the flow the increased pressure drop across the contraction will cause the flow to be modulated. In a vortex valve, swirl is imparted to the main supply flow by a number of tangential control jets after which the total flow passes through the chamber and exits at a smaller radius. When the swirl is sufficient the main supply flow is shut off and only the control flow discharges downstream of the valve.

The paper describes the development of a model to predict the operating characteristics of an axial vortex valve. The work complements earlier experimental studies by Duggins and Frith (1979) and Frith (1984) into the geometry and performance of vortex valves. The complex nature of the flow phenomena observed in vortex valves has delayed the implementation of theoretical models and favoured the use of experimental studies for design development. Such studies, well reviewed by Wormley (1976), have identified the major parameters affecting the valve's performance.

Wormley and Richardson (1970) showed that an inviscid model can only predict the first thirty percent reduction in the total flow and that the effects of dissipation within the valve become more important as the valve approaches shutoff. The model of Bichara and Orner (1969) is typical of existing analyses, the flow is treated as one-dimensional and the dissipation is solely accounted for by a wall shear stress term. Whilst this model is capable of matching the characteristic curve of a valve, over a wide range of valve geometries and operating conditions, there is a need to resort to large variations in the values of the empirical constants to do so and this highlights the limitations of such models.

The model described here is based on the numerical solution of the governing equations of flow. This approach was made feasible by the recent progress in numerical solution techniques and turbulence closure models.

MODEL FORMULATION

The vortex valve was modelled as two separate regions, the inlet region and the vortex chamber region, as illustrated in Figure 1. The inlet region was handled by a simple one-dimensional analysis and the vortex region was handled by solving the equations governing axisymmetric turbulent swirling flow in a right-cylindrical chamber. The separate treatment of the supply and control flow mixing and the assumption of axisymmetric flow within the chamber meant that a two-dimensional rather than a three-dimensional computational scheme could be used. The above assumption is realistic as axisymmetric flow is enhanced by both the geometry of the axial vortex valve and the higher swirl levels within the valve.

The model of the inlet region will not be described here in detail as it is a standard treatment. It assumed that the mixing of the supply and control flows within the entry annulus was lossless and that it was completed prior to the total flow entering the chamber. The inlet conditions for the vortex chamber model, i.e. the inlet axial and tangential velocities, were calculated from the one-dimensional continuity, energy (discharge) and angular momentum equations which formed the inlet model. The experimental results supported this approach as it was found that the effect of a wide range of inlet configurations - number of ports, diameter of ports, pre-swirl and annulus thickness - on the percentage reduction of the supply flow could be simply characterised by the magnitude of the rate of angular momentum entering through the inlet ports.

The major task in implementing the model of the vortex chamber region was the determination of an appropriate turbulence closure model. The review of the literature on swirling flows and turbulence closure models showed that there is now a substantial body of evidence which supports the contentions that (i) both the turbulence level and the rate of dissipation are modified by rotation of the flow (ii) the turbulence is anisotropic in flows with swirl or with other forms of streamline curvature, and (iii) generally the basic two equation k-ε model is not sufficient for predicting such flows. A number of alternative turbulence models were considered for handling the anisotropy in swirling flows and these are discussed in Frith and Duggins (1985).

The chosen model used two different turbulent viscosities to take account of the anisotropy. The axial and radial momentum equations used a viscosity calculated from the two equation k-ε model and the tangential momentum equation used a viscosity calculated from a mixing length formulation. The mixing length formulation is similar to that derived by Rochino and Lavan (1969), from similarity conditions in swirling flow, except that the turbulent

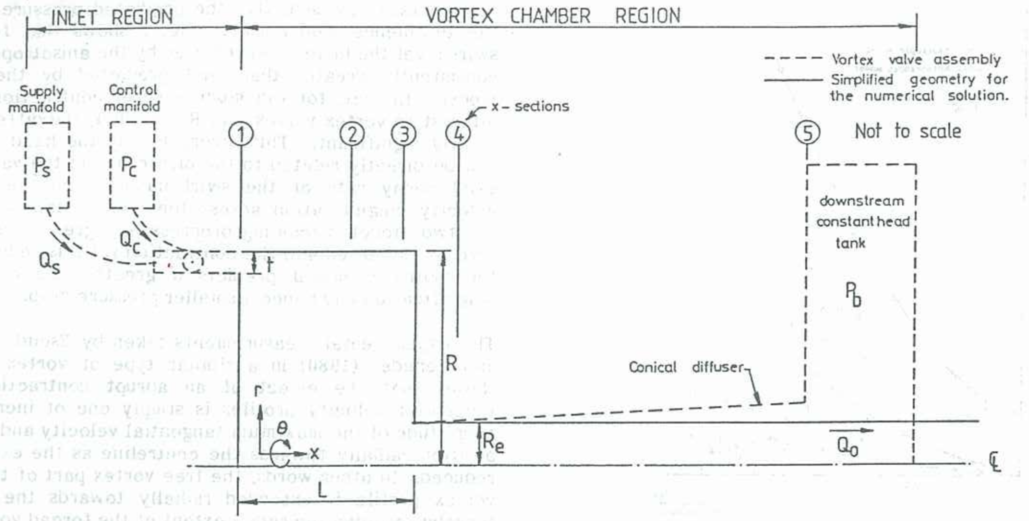

Fig 1 : Simplified valve configuration.

viscosity in the near-wall region ($r/R > 0.94$) is calculated from the two equation model. This model was tested against available experimental data for swirling flow in straight pipes. It was found to correctly predict the type of tangential velocity profiles observed in vortex valves; those of the combined vortex type where a forced vortex is embedded in a free vortex. Whilst providing good quantitative predictions it did slightly underpredict the maximum tangential velocity and its radial position from the centreline. In contrast, the isotropic k-ε model, by itself, provided good predictions of a predominately rotational flow but continued to predict forced vortex profiles for the case of a combined vortex type of swirling flow. Therefore it was not appropriate for vortex valve flow. Conversely, it would be incorrect to use the present anisotropic model for rotational flows as it only predicts profiles of the combined vortex type.

Whilst the particular anisotropic model used here is a somewhat restrictive formulation it has proved adequate primarily because of the nature of the flow patterns observed in vortex chambers. Due to the effect of swirl and the abrupt contraction there are marked changes in the axial and radial velocity profiles throughout the flowfield whereas the tangential velocity profiles remain similar throughout the chamber. This facilitates the use of a simple mixing length formulation for the tangential momentum equation but still requires the use of a more sophisticated model - the k-ε model - for the axial and radial momentum equations.

The numerical solution of the elliptic partial differential equations for the axial u, radial v and tangential w velocities and the turbulent kinetic energy k and its rate of dissipation ε were based on the CHAMPION 2/E/FIX program developed by Pun and Spalding (1977). Most of the features of this program are common to earlier codes developed at Imperial College, London and as a result of widespread use these features are well documented in the literature. The code is primarily intended for handling steady flows, which are plane or axisymmetrical by means of finite-difference equations formulated with respect to a staggered grid of nodes and cells, the location of which are specified at the start. The individual momentum equations for the axial and radial velocities are solved first and then adjusted to satisfy continuity. All variable finite difference equations are solved line by line and the finite difference equations for the nodes on a line are solved simultaneously by the matrix inversion procedure known as the Tri-Diagonal Matrix Algorithm.

The modifications necessary to extend the program to handle axisymmetrical turbulent swirl were simply the addition of the tangential velocity and viscosity terms to (i) the equations of motion, (ii) the turbulent energy production term in the k-ε equations and (iii) the wall functions modelling the near-wall region.

The implementation of the chamber geometry, especially the abrupt contraction, through the wall boundary conditions constituted the bulk of the additions to the code. The staggered grid makes it possible to block off those regions of the rectangular finite grid that are not in the solution domain. This was done by locating those wall boundaries which are not at the edge of the rectangular domain at the centre of the finite difference cell for the velocity normal to the wall, i.e. the right-chamber end-wall at the node of a axial velocity cell and the exit pipewall at the node of a radial velocity cell. The velocity at the wall was then suppressed to zero by an appropriate modification of the coefficients in the finite difference equation, especially through the use of large source terms. The accuracy of this method was checked by computing the head loss coefficients for the case of a straight flow through an abrupt contraction. These agreed with the empirical values published in the textbooks.

Although flows involving variable body forces are known to be prone to numerical instability, the use of lower relaxation factors and the use of realistic initial fields for all variables, usually the previous solution, were sufficient to prevent divergence of the solution for the flow configurations investigated.

RESULTS AND DISCUSSION

The first set of results presented are for swirling flow through an abrupt contraction, i.e. Fig. 1 without the left chamber end-wall. The swirl number S_θ defined as the axial flux of angular momentum divided by the chamber radius R times the axial flux of axial momentum has been used to characterise the swirl level of the flow. The velocity and pressure profiles at each x-section were also integrated to provide averaged values at a x-section, i.e. u, v, w and p, for comparative purposes and for calculation of the head loss.

The contraction was placed thirty chamber radii downstream of the inlet, i.e. L = 30R, to isolate the contraction effect from possible inlet effects and to ensure the swirling flow entering the abrupt contraction was stable and developed. In this case, x-section 2 and 4 of Fig. 1 designated, respectively, the upstream and downstream limit to the influence of the contraction on the flowfield. On average, for these studies, the upstream limit was 6R from the contraction, although it did move further upstream with higher swirl levels and smaller exit radii.

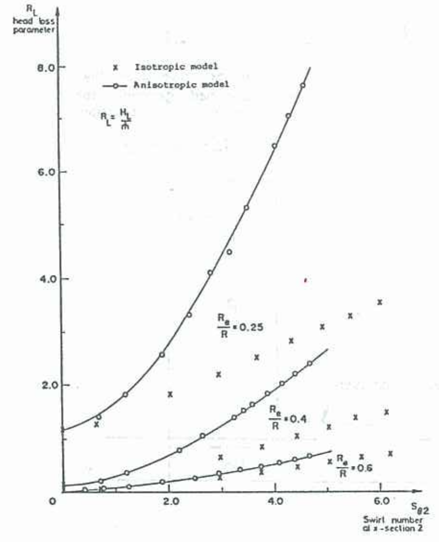

Fig 2 : The effect of the turbulence model and contraction ratio on the predicted head loss parameter.

Fig. 2 shows a plot of the predicted flow resistance R_L, the head loss between x-section 2 and 4 divided by the mass flow, versus the swirl number of the flow entering the contraction $S_{\theta 2}$. The results are for both the isotropic and anisotropic turbulence model and for three chamber contraction ratios, the exit pipe radius R_e divided by the chamber radius R, specifically, R_e/R = 0.6, 0.4 and 0.25. In these predictions, the inlet swirl was varied by increasing the magnitude of the inlet tangential velocity - but same radial distribution - whilst the inlet axial velocity was held constant. The results of Fig. 2 are as expected, the magnitude of the head loss across the contraction increases as the incoming swirl level is increased and the chamber contraction ratio is reduced.

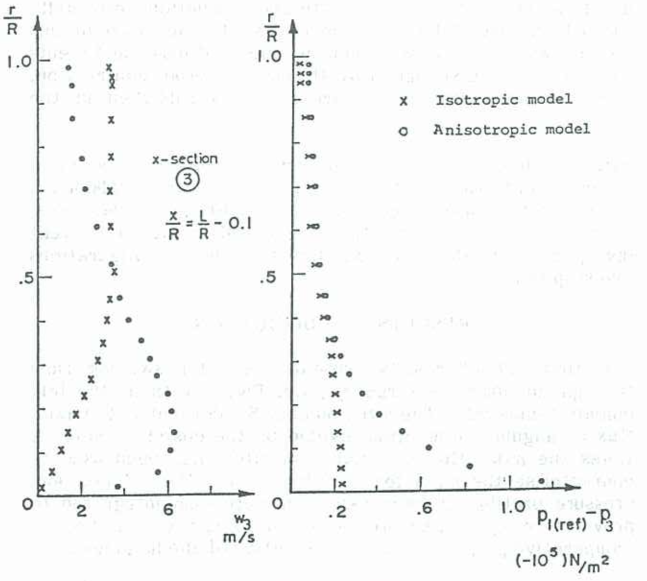

Fig 3 : Typical tangential velocity and pressure profiles for the isotropic and anisotropic turbulence models.

Fig. 3 shows the tangential velocity and pressure profiles typically predicted by the two turbulence models. As previously indicated, the forced vortex profile of the isotropic k-ε model is inappropriate for vortex valve flow, however, this model was included in the computations to demonstrate how sensitive the predicted pressure drop is to the turbulence model used. Fig. 2 shows that for a given swirl level the head loss predicted by the anisotropic model is consistently greater than that predicted by the isotropic model. In fact, for the much severer contraction ratios of interest to vortex valves, i.e. $R_e/R < 0.4$, the differences are clearly significant. This divergence in the head loss values can be directly related to the difference in the values of the axial decay rate of the swirl number and the tangential velocity magnification across the contraction predicted by the two models becoming progressively greater as the swirl level is increased and the contraction ratio is reduced. Here the isotropic model predicts a greater decay, a smaller magnification and hence a smaller pressure drop.

The experimental measurements taken by Escudier, Borstein and Zehnder (1980) in a similar type of vortex tube have shown that the effect of an abrupt contraction on the tangential velocity profiles is simply one of increasing the magnitude of the maximum tangential velocity and moving its position radially towards the centreline as the exit radius is reduced. In other words, the free vortex part of the Rankine vortex profile is extended radially towards the centreline thereby reducing the radial extent of the forced vortex region i.e. the vortex core. The typical tangential velocity profiles predicted by the anisotropic model for varying contraction ratios are shown in Fig. 4 and these demonstrate that the predictions are consistent with the observed behaviour. Furthermore, the predicted variation in the axial velocity profile is consistent with the observed variation between a wavelike core - one where the centreline velocity is retarded - which generally occurs at larger contraction ratios, and a jet like core which generally occurs at smaller contraction ratios. Therefore one is confident that the anisotropic model is realistically accounting for the effect of the abrupt contraction on the flowfield.

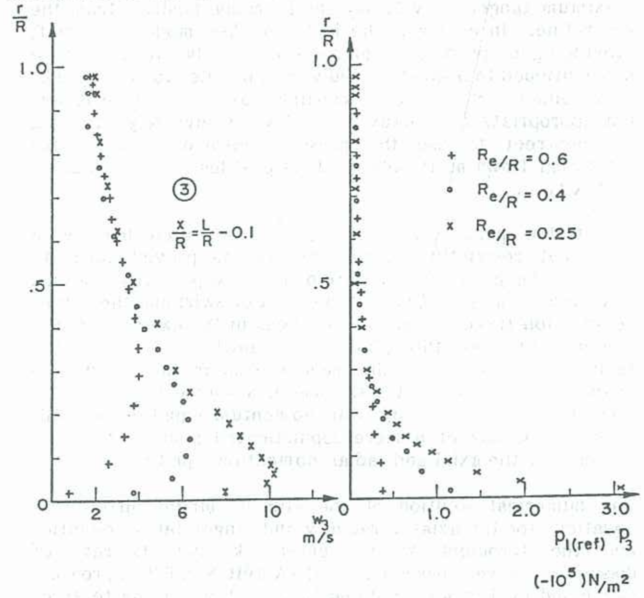

Fig 4 : The effect of the abrupt contraction on the tangential velocity and pressure profiles for the anisotropic model.

The initial set of computations were for an increasing swirl but constant axial throughflow, i.e. a constant Reynolds number. However, the operation of a vortex valve is such that as the swirl level increases the axial throughflow decreases and so predictions were obtained for a range of Reynolds numbers, $10^3 < Re < 10^6$, to determine the effect of the axial throughflow on the predicted pressure drop. These results are not presented here as the variation of the resistance parameter R_L with Reynolds number proved to be a simple translation of the curves presented in the R_L versus $S_{\theta 2}$ plot of Fig. 2. For a given swirl level, R_L varied in direct proportion to the magnitude of the inlet axial velocity u_1. Using a more familiar scale, the results for varying Reynolds numbers can be collapsed onto a single curve by plotting $\Delta p/U_1^2$ against $S_{\theta 2}$.

For the initial swirl levels, $S_{\theta 2} < 0.6$, the predicted magnification of the tangential velocity across the contraction w_4/w_2 was equal to the inverse of the contraction ratio, the value appropriate to an inviscid flow and a free vortex profile. As the swirl level increased the magnification decreased from its inviscid value. A w_4/w_2 versus $S_{\theta 2}$ plot for all the u_1 and w_1 variations considered showed that the data could be fitted by a linear curve in each of two regions of swirl. For $R_e/R = 0.4$, the constant of proportionality was -0.27 for $0.6 < S_{\theta 2} < 3.0$ and -0.12 for $3.0 < S_{\theta 2} < 7.0$. The values for $R_e/R = 0.6$ and 0.25 also collapsed onto the same curve if their w_4/w_2 values were multiplied by $(R_e/R)/0.4$. The results show, for the anisotropic model, that the reduction in the magnification of the tangential velocity from its inviscid value due to dissipation is being well characterised by the swirl number of the flow entering the contraction. This can be related to the fact that the axial decay rate of the swirl - the dissipation of the angular momentum across the contraction - is dependent on the initial swirl level of the flow, with the higher swirls leading to greater decay rates.

The geometry of the vortex chamber was completed by adding the left chamber end-wall to the configuration of the abrupt contraction. Predictions were obtained for three chamber lengths L, specifically L = 3OR, 1OR and 2R, to examine the effect of reducing the chamber length from a longer length typical of swirling pipe flow to a shorter length typical of vortex chamber flow. It was found that the presence of the left chamber end-wall did not alter the relationship between the predicted pressure drop across the contraction and the swirl level entering the contraction. For a given inlet swirl, the influence of the chamber length on the predicted pressure drop is simply one of setting the level of the decay in the swirl by the contraction. A smaller chamber length leads to a lower decay in the swirl by the contraction and hence a greater pressure drop. The additional pressure losses which are associated with the wall friction losses over the larger chamber lengths are negligible compared with those across the contraction.

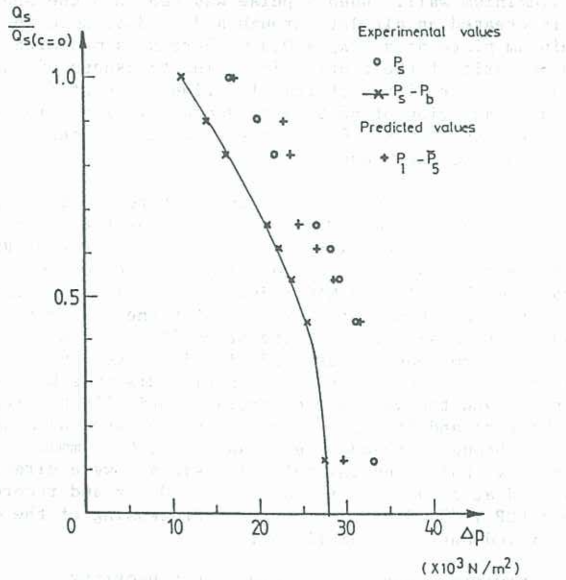

Fig 5 : Comparison of the predicted pressure drop for an axial vortex valve with the experimental values.

The ability of the model to correctly predict the characteristic curve of an axial vortex valve was evaluated by comparing the predicted pressure drop with the corresponding experimental ones. A valve with a good index of performance was chosen. The inlet configuration was as follows, pre swirl of supply flow via tangential entry, six supply ports of 25 mm dia. and six control ports of 4 mm dia. The chamber dimensions, as defined in Fig 1, were as follows, t = 6 mm, L = 102 mm, R = 51 mm and R_e = 12.5 mm. For a given operating point on the experimental characteristic, the inlet conditions for the model of the vortex chamber region u_1 and w_1 were determined by substituting the supply Q_s and control Q_c flow values together with the geometric data into the equations of the inlet model. The program was then run to calculate the pressure drop Δp. The predicted and experimental pressure drops for the axial vortex valve are shown in Fig 5. This figure shows that the model does overpredict the pressure drop. An assessment of the pressure recovery available for swirling flows in conical diffusers indicated that at least half of the overprediction of the pressure drop could be attributed to modelling the exit geometry as a straight pipe rather than the actual conical diffuser.

CONCLUDING REMARKS

The present model has been able to achieve good predictions of the pressure drop across a vortex valve without the need to resort to empirical constants which are device dependent. This can be attributed to the model, firstly, accounting properly for the flow interactions as it solves the three momentum equations at individual nodes throughout the flow domain whereas the existing analyses have generally been one dimensional and, secondly, the dissipation within the flow is well modelled by the turbulence closure model whereas existing models have solely modelled the dissipation through a wall shear loss term.

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