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SIMULATION OF PRECESSION IN AXISYMMETRIC SUDDEN EXPANSION FLOWS

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ABSTRACT

In this paper, time-dependent calculations are carried out using CFX4 and the standard k-ε model to simulate the turbulent flow in axisymmetric sudden expansions for Reynolds numbers of the order of 10^5. These flows are unstable, and a large-scale coherent structure has been simulated. The visualisation of instantaneous flowfields indicate that a large recirculation zone exists and a significant azimuthal flow is found behind the expansion face. The effect of expansion ratio on the precession frequency is presented in the range of 3.95 to 6.0. The sensitivity to the inlet profiles for velocity, turbulence intensity and dissipation length scale is investigated. Low turbulence intensity at the inlet is necessary for a sustained regular oscillation.

NOMENCLATURE

D,d	diameter of downstream and upstream pipes
d^*	dimensionless turbulence dissipation length scale
E	expansion ratio of diameter D/d
f	precession frequency
I	turbulent intensity
k	turbulent kinetic energy
L,l	length of the pipes
M	jet momentum flux
r	radius variable
r_o	pipe radius
Re	Reynolds number at the inlet
U^*	dimensionless velocity profile
U,V,W	Reynolds-averaged velocity components
U_I,U_o	bulk velocity at the inlet and exit
ε	turbulence dissipation rate
ρ	density
μ	laminar viscosity
θ	angle of flapping relative to radial direction

INTRODUCTION

Sudden expansion flows are fundamental to many engineering applications, such as burners and spray driers. Coherent large-scale structures, such as a precessing jet (PJ) and precessing vortex core (PVC), have been frequently observed (Gupta *et al*, 1984). A fundamental study of the effect of jet precession on combustion has shown that it can be used to increase flames luminosity in and reduce global NO_x emissions from open natural-gas flames (Nathan *et al*, 1996). On the other hand, the oscillations arising from such precession within a large expansion tends to cause particle deposition on walls in spray driers, and thus is undesirable. The PVC in swirling flows can also cause many other problems, such as vibrations and noises in process equipment. A good understanding of these complex turbulent flows is necessary to improve methods of prediction and to allow better control.

Dellenback *et al*. (1988) undertook measurements in a turbulent swirling flow through an abrupt axisymmetric expansion with an expansion ratio of 1.94, and examined the influence of the swirl number. They noted that the PVC occurs for low swirl levels, and one of the two regions that were especially difficult to resolve is the precession frequency for very low swirl numbers (S<0.1). It was shown by the work of Hill *et al*. (1995) that upstream swirl is not a prerequisite for precession provided that the expansion ratio is sufficiently large, when the precession of a reattachment was identified in a large expansion into a long axisymmetric pipe.

Nathan *et al*. (1998) experimentally investigated a continuously unstable precessing flow within a short cylindrical chamber following a large sudden expansion. They discovered that with certain expansion ratios of the nozzle inlet orifice and for sufficiently large flow rates, an asymmetric precessing jet flow developed, which has proved beneficial in combustion applications. Two flow modes, i.e., an instantaneously highly asymmetric 'precessing jet' (PJ) and a quasi-symmetric 'axial jet' (AJ), were identified, with the temporal probability (proportion of time) of the PJ mode increasing with the Reynolds number.

In order to observe the nature of a precessing jet and to determine the transition between the two flow regimes identified by Schneider *et al*. (1992) in a mechanically precessing jet, Manohar *et al*. (1996) solved steady, laminar Navier-Stokes equations at low Reynolds numbers in a rotating frame of reference. Since a jet entry angle of 30 degrees with the precession axis was assumed, their technique was not appropriate for resolving the inherent unsteadiness of the flow with axisymmetric boundary conditions.

In our previous paper (Guo *et al*, 1998), an instability was predicted in a three-dimensional axisymmetric sudden expansion flow calculated using CFX4. A sustained precessing instability, as well as a coexisting swinging oscillation, was identified with higher Reynolds numbers of the order of 10^5 at the inlet. The frequency of precession (normalised as a Strouhal number based on the smaller diameter and average downstream velocity) presented for the expansion ratio of five agreed well with a range of experimental data reported by Hill *et al*. (1995). The good agreement between frequencies measured and predicted by the numerical simulation gives confidence in the procedures used in the simulation and the possibility to investigate the fluidic precessing jet directly rather than dissecting it into its component features (Mi *et al*., 1998). The objective of the present paper is to further draw information on the characteristics of the flow field. The effect of the expansion ratio and the sensitivity to the boundary condition are examined. Our motivation to investigate this problem is to further validate the capability of the CFD approach to resolve and to gain more understanding of the precessing instability.

GEOMETRY AND BOUNDARY CONDITIONS

The geometry shown in Figure 1 consists of two coaxial pipes of diameter d and D with a smooth surface. The expansion ratio E refers to the ratio of the larger diameter D and the inlet tube diameter d. The x-axis of the Cartesian coordinates is coincident with the axis of the pipes and the origin is at the centre of the expansion face. The simulation regime has been taken as l=5~l0d and L=20D.

A fully three-dimensional structured grid (as shown in Figure 2) has been used and constructed to have small orthogonality deviations. The cell density is approximately uniform in a cross-sectional plane, but it is non-uniform along the x-axis, and increases progressively as the expansion plane is approached in order to resolve the large gradient of flow variables, with the neighbouring cells varying in size by no more than 10%. The final grids have as many as 150,000 cells in total.

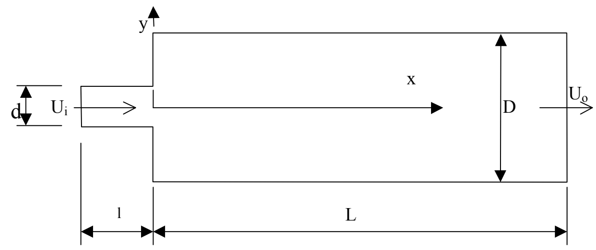

Figure 1: Schematic diagram of the geometry

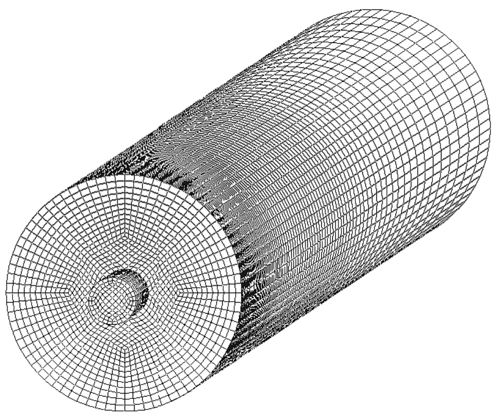

Figure 2: The grid used in the calculations

The working fluid is chosen as incompressible, Newtonian and its physical properties have been set in the range of 1.0-1.295 kg/m^3 for the density (ρ) and 1.0-1.812$\times$10^{-5} kg/m.s for the laminar viscosity (μ), respectively. These values are representative of those for air.

The velocity at the inlet is parallel to the inlet tube axis without cross-stream components. Neumann boundary conditions are imposed for both the inlet and the outlet, which means that the velocity components and turbulence quantities are assumed to be fully developed for the inflow, where the mass flux is specified, but to have a zero normal gradient for the outflow. The sensitivity to inlet profiles has been checked. Wall functions are used for all walls, so that a fine grid is not required to resolve the boundary layer.

NUMERICAL CONSIDERATIONS

The CFD package CFX4 (CFX, 1997) has been employed, which provides a finite-volume based flow solver. The Reynolds-averaged Navier-Stokes equations have been solved, with the standard k-ε turbulence model being used. The higher order differencing scheme QUICK has been used for velocities and Van Leer has been used for k and ε. PISO is employed for pressure correction. The equation solvers are BLOCK STONE for velocities and AMG for pressure. A steady calculation has been used as an initial estimate for the transient calculations. The sum of the absolute values of the mass fluxes into or out of every cell in the flow domain has been tested as a convergence criterion for mass conservation. The final mass source residual for each time step has been less than 0.1% of the total mass flow rate. Second-order, fully implicit Backward Euler differencing in time have been used. A typical time step is 0.02s.

TIME VARIATION OF FLOW QUANTITIES

In the previous paper, a time series at a monitoring point on the centre-line was presented, through which a precession and a quasi-flapping motion (as we define it) can be easily identified, as shown in Figure 3. The flapping jet, relative to a rotating frame of reference, may in fact move along a complex path other than a straight line and may be at an angle, θ, to the radial direction. It was also shown that the precession and flapping have no bias about the centre-axis, in spite of the instantaneous asymmetry. Figure 4 shows a time series of variables at an off centre-line monitoring point. The variation between the neighbouring cycles results from the flapping oscillation. The axial velocity (U) is constantly reversing direction and the positive peaks correspond to the moments when the jet sweeps past the point.

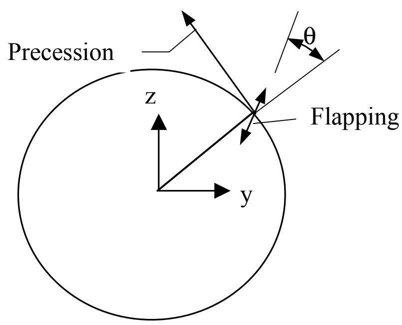

Figure 3: Schematic diagram showing jet precessing and flapping

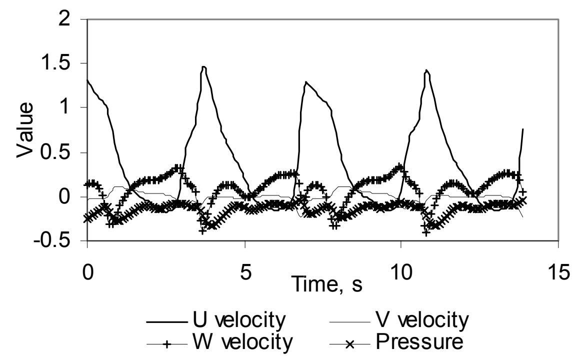

Figure 4: Flow variables at the monitoring point 0.4 D from centre-line and 4D downstream from the expansion (E=5.0, Re=100,000)

VISUALISATION OF FLOWFIELD

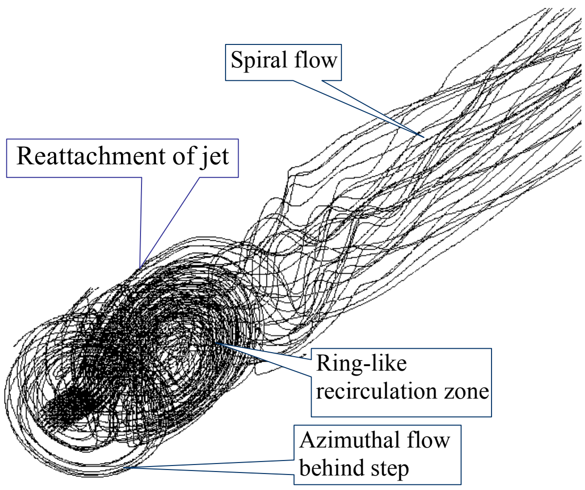

Figure 5: Instantaneous streamlines calculated for precessing flow in an axisymmetric sudden expansion (E=5.0 Re=100,000)

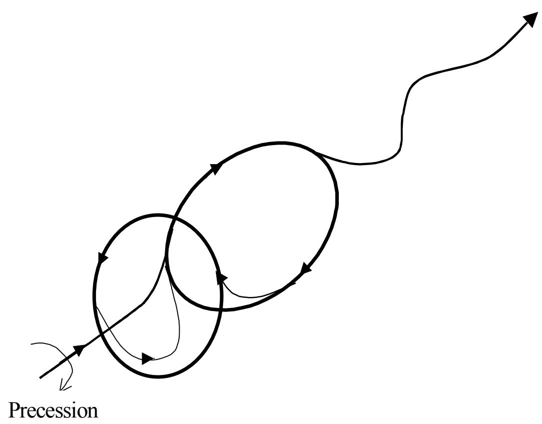

Figure 6: A schematic interpretation of the instantaneous streamlines

Although the instantaneous flowfield changes with time, the fundamental features remain the same. Figure 5 shows a picture of the instantaneous streamlines for the case of E=5.0, produced using CFX-Visualise, and Figure 6 is the interpretation of the corresponding streamlines. The fluid emanating from the inflow tube separates at the expansion and moves towards the large cylindrical wall. It reattaches to the wall at around 1.5D. This point may vary with time due to the flapping oscillation. The reattachment occurs over a length of about 3D. Then the jet separates from the wall and bends inward towards the other side of the wall. While a smaller proportion of the jet fluid moves down, most of the fluid reattaches at the opposite side of the wall and then flows upstream along the wall, thus creating a significant reverse flow. In this way a large recirculation zone is formed, which looks like an elliptical ring. The length scale of this ring-like recirculation is 1D in the cross-stream direction and about 4D in the streamwise direction. The jet is gradually dispersed and becomes very weak after the recirculation zone. Due to the mass inertia, the downstream flow has a specific time lag so that the precessing jet passes a downstream point at a later time azimuthally, thus the jet spirals in the opposite direction to the precession, as can be seen in an iso-surface of the velocity in Figure 7. This spiral nature of the jet is important for the jet precession as it produces a non-zero azimuthal momentum to the jet, which is balanced by the remaining fluid motion.

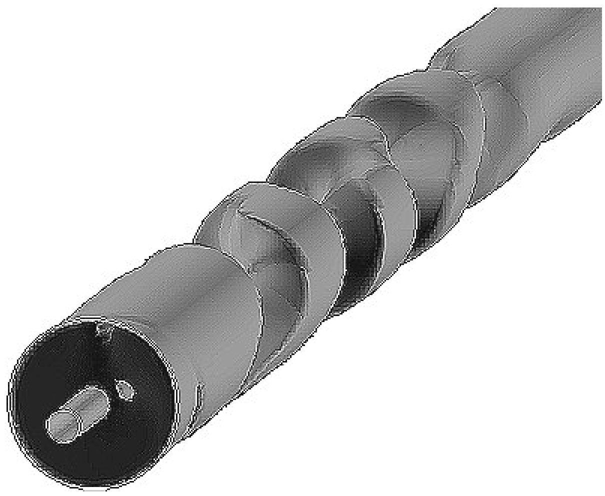

Figure 7: An iso-surface for velocity (E=5.0 Re=100,000)

Just behind the expansion face, an azimuthal flow moves in the opposite direction to the precession. This azimuthal flow is fed by the central recirculation flow, and is partly entrained into the main jet. Figure 8 shows the velocity vectors in a cross-stream plane near the expansion, which indicates that the swirling flow is not uniform in the circumferential direction. In spite of the azimuthal flow and precessing motion being induced, the net angular momentum over any cross section remains small (two orders of magnitude smaller than the axial momentum). However significant mixing between upstream and downstream flows, as well as azimuthal mixing, would be expected.

A "bouncing around" pattern of jet movement has been predicted in Guo *et al.* (1998). Figure 9 shows an instantaneous pressure contour map. At this moment, a low-pressure core has departed from the centre-line and created a pressure difference across the jet which drives the jet away from the centre-line towards the wall.

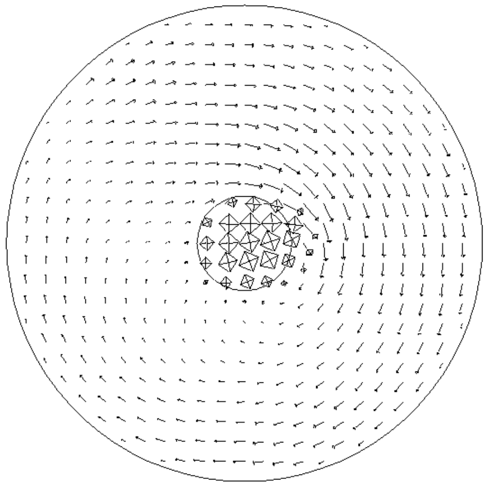

Figure 8: Instantaneous velocity vecters in a cross-stream plane 0.2D downstream from the expansion (E=5.0, Re=100,000)

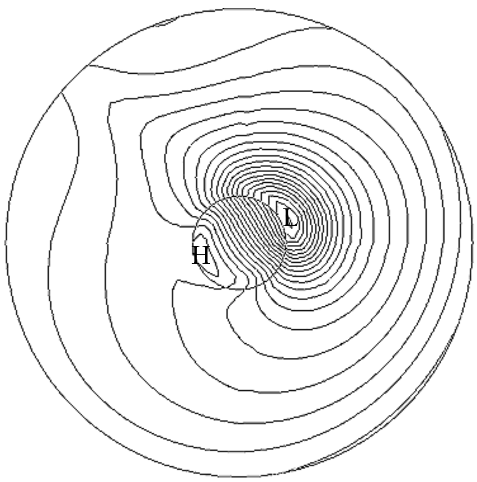

Figure 9: Instantaneous pressure contours in a cross-stream plane 0.2D downstream from the expansion (E=5.0, Re=100,000)

THE EFFECT OF EXPANSION RATIO AND COMPARISON WITH EXPERIMENTS

The effect of expansion ratio has been examined and the results for E=3.95-6.0 are presented in this paper. Within this range both the precession and the flapping oscillation occur. A further decrease in the expansion ratio produced a quasi-flapping oscillation while a further increase resulted in a complex pattern. For the cases of E=6.0, there appears to be a critical Reynolds number above which the oscillations are highly regular and below which the complex pattern occurs.

The data shown in Figure 10 are only for the cases of regular oscillations in the current simulations. The precession frequencies are normalised by a Strouhal number $St = \frac{f\sqrt{\rho}D^2}{\sqrt{M}}$. The Reynolds number is based on the bulk velocity and the diameter at the inlet. Some experimental data on precession are also quoted from the literature. The geometrical configurations used are variations of the sudden expansion. For example, Hill *et al*. (1995) used a smooth contraction inlet to a sudden expansion (E=3.75-14.0), whereas the nozzle used by Nathan *et al*. (1998) had a sharp-edged inlet orifice and a contracted exit (E=6.43). In spite of the geometric variations, it can be seen that the calculated frequencies of the precession are of the same order as those in the experiments, suggesting that they are closely related. However, the Strouhal number in the current simulations clearly shows an increasing trend with the expansion ratio, while this trend is not apparent in the experiments. The Reynolds number has no apparent effect on the Strouhal number for a specific expansion, but the Strouhal numbers measured by Hill *et al*. (1995) and Nathan *et al*. (1998) seem to decrease as the Reynolds number increases. They suggested that compressibility effects are the probable cause of the Reynolds number dependence. However this was not examined here since the present simulations assume incompressible flow.

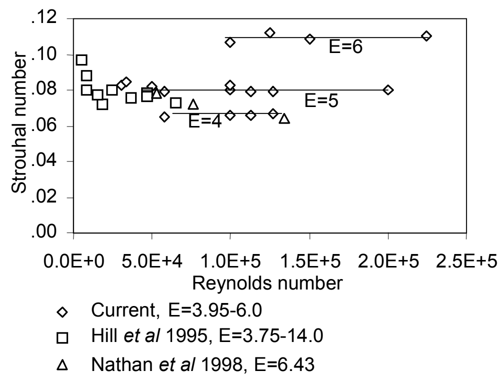

Figure 10: Strouhal number against Reynolds number

Regarding the flapping oscillation, the observation of video images by Nathan *et al*. (1998) that the exit angle varies from cycle to cycle relative to the nozzle axis appears to be consistent with the prediction here, but they did not provide quantitative information on the frequency.

SENSITIVITY TO INLET CONDITIONS

Usually the initial flow fields are not critical. However, this is not true for the boundary conditions. In the previous calculations, a mass flow boundary for inflow has been implemented, so that all velocities and turbulence quantities have zero gradient, and it is not necessary to set any profiles. Therefore the Neumann boundary (ie., mass flow boundary and pressure boundary) conditions are not applicable for checking the sensitivity to the variation of the inflow quantities, and Dirichlet boundary conditions have to be chosen. There are several parameters that might influence the downstream conditions. One parameter was adjusted and the others were kept fixed to realistic profiles. To determine these profiles a fully-developed pipe flow has thus been simulated, and the results are shown in Figure 11 in a dimensionless form.

$U^* = \dfrac{U}{U_i}$ is the velocity profile, $I = \dfrac{1}{U_i}\sqrt{\dfrac{2k}{3}}$ is the turbulence intensity, and $d^* = \dfrac{k^{1.5}}{0.3\varepsilon d}$ is the dissipation length scale normalised by the inlet diameter.

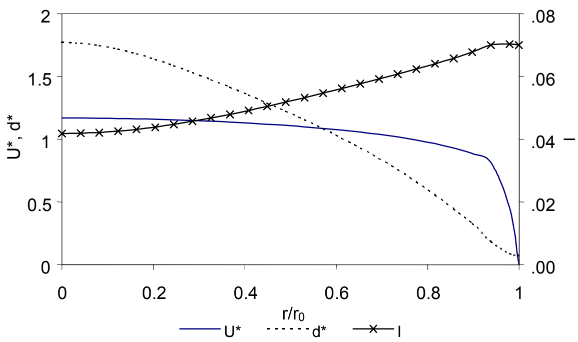

Figure 11: Profiles for fully developed pipe flow (Re=100,000)

Together with the profiles calculated for the fully-developed pipe flow, an expansion ratio of 5.0 is used as a case study for the axisymmetric sudden expansion. When a length of inflow pipe is included within the simulation domain, if a non-equilibrium value is set at the inlet, it may be dissipated in the inflow pipe. Any disturbance may also decay away before reaching the expansion. Therefore, the simulation result is less sensitive to the inlet condition with an inlet pipe than without it. To make the influence apparent, the inlet condition is imposed just at the expansion plane (no inflow pipe included) so that the variation of the inlet condition is absorbed and evolves only in the larger cylinder.

Velocity profile

When a velocity profile from a fully-developed pipe flow is replaced by a uniform velocity profile, the periods (the inverse of the frequencies) of both the flapping and the precession increase by about 5-10%.

Dissipation length scale

When the dissipation length scale is increased to 10d, the oscillation appears to be damped and the amplitude of the flapping oscillation becomes weaker. However, both the flapping and precession frequencies are reduced by only about 3%.

Turbulence intensity

When the turbulence intensity is decreased from 0.037 to 0.01, the oscillation at the monitoring point is slightly enhanced in strength whilst the frequencies are not much affected. However when the turbulence intensity is increased to 0.1, the stable flow oscillation becomes unsustainable and is replaced by a more chaotic pattern. The main feature is that switching between a symmetrical jet and an asymmetric jet precessing in alternate directions occurs. In this state, the precession predominates most of the time, whilst the breakdown of the precession occurs over short time intervals.

Therefore, the turbulence intensity at the inlet is more important than the dissipation length scale. The present calculations are based on a perfectly circular and smooth pipe. In practice, due to the inevitable nature of the wall surface roughness and shape imperfections, the inlet turbulence intensity is expected to be higher than assumed. Moreover, disturbances from upstream flow or vibrations and noises from the surrounding equipment may also strengthen the turbulence level. Consequently these tend to bring about a chaotic oscillation instead of a regular one, as has been simulated using fully-developed inlet conditions with low turbulence.

Upstream oscillation

When a length of inflow pipe is included within the simulation domain, and a mass flow boundary is set at the inlet, the velocity vectors immediately upstream from the expansion deflect from the centre-line due to the influence of the downstream flow asymmetry. For the case of an expansion ratio E=4.87 and a Reynolds number Re=10^5, this deflection angle at the expansion is about one degree, and the influence ranges about one diameter upstream from the expansion plane.

To identify the effect of this upstream oscillation, the inlet is set at 1.0d upstream of the expansion, where non-uniform profiles are imposed for U^*, I and d^*. The subsequent oscillations become regular compared with the case of no inflow pipe, due to the release of the restriction. Therefore the existence of the upstream oscillation tends to make the downstream oscillation regular.

CONCLUSIONS

Fully three-dimensional, transient Reynolds-averaged Navier-Stokes equations are calculated using CFX4 and the standard k-ε model to simulate the turbulent flow in axisymmetric sudden expansions. Large-scale coherent structures and self-sustained oscillations are predicted. The visualisation of an instantaneous flowfield indicates that a large ring-like recirculation zone exists which creates a large-scale entrainment of downstream fluid back into the upstream flow. A significant azimuthal flow is found behind the expansion face, which enhances the cross-sectional mixing. Further downstream, the jet, while spreading gradually, exhibits a spiral nature with a direction opposite to the precession.

The Strouhal number (based on the inflow momentum flux and downstream diameter) is basically independent of the Reynolds number, but increases as the expansion ratio increases in the range of 3.95 to 6.0 examined. The calculated values of the Strouhal number are of the same order as the reported data.

The inlet conditions have an influence on the regularity and frequency of the oscillations. Compared with the case of a fully-developed velocity profile, the oscillation frequencies are about 10% higher when using a uniform velocity profile. Low turbulence intensity is necessary for sustaining a regular oscillation. Otherwise an intermittent oscillation may result.

ACKNOWLEDGMENT

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