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Short note

Generalized equation for the design of a baffle to generate arbitrary flow velocity profiles

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ABSTRACT

Flow velocity profiles can be transformed using non-uniform resistance elements, and the design equation for the distribution of pores in a single baffle plate of uniform thickness was formulated to transform any flow velocity profile into another arbitrary velocity profile. The proposed equation was validated first through numerical analysis for a couple of velocity profiles with satisfactory results, and then verified by an experiment with an excellent agreement of maximum difference less than 4.3%. Also found was an empirical relation between the number of holes in the baffle and the accuracy of the transformed velocity profile.

1. Introduction

Flow velocity control has been required for performance improvement or design optimization in a variety of engineering applications (Ma et al., 2013; Antiohos and Thorpe, 2015; McNally et al., 2015; Parkin et al., 2015). A common means of controlling flow velocity profile is to use a non-uniform distribution of flow resistance (Solovitz and Mainka, 2011; Tong et al., 2007), and a typical example is the perforated plate with non-uniform distribution of pores used in electrostatic precipitators for generating uniform flows Şahin and Ward-Smith (1991). Though the uniformizing baffle is an old technique, most of the design effort to date has been case specific, and the size and number density distribution of pores has been sought through numerical simulation or experimental means (Fan et al., 2008; Liu et al., 2010). Only very recently, Choi et al. (2014) developed a simple formula to determine the non-uniform distribution of porosity or holes for uniformizing an arbitrary non-uniform velocity profile with just a single baffle plate of uniform thickness.

The aim of this study is to extend the previous work on flow uniformization to a more general case of transforming an arbitrary velocity profile to another arbitrary profile, and that using a single baffle plate with non-uniform porosity or holes. A simple theoretical model for the non-uniform porosity distribution needed for the flow transformation is formulated, and validated first through numerical simulations for a couple of velocity profiles and then verified again by an experiment.

2. Transforming flow velocity profiles with a non-uniform flow resistance

Flow velocity gets changed after passing through a zone of non-uniform flow resistance-reduced after a high-resistance zone and increased after a low-resistance zone, in general. If the flow resistance is produced by a perforated plate with holes, the local flow field around each hole can be simply modeled by a flow through an orifice. When a single orifice is installed in a flow channel with an incompressible turbulent flow of density ρ and mean velocity V_0, the pressure drop (ΔP) across the orifice changes with the orifice opening or porosity (β_0) (Perry et al., 1997):

$$\Delta P = \frac{1}{2}\rho V_0{}^2\left[\frac{1}{\beta_0^2}-1\right]\frac{1}{C_d{}^2}. \quad (1)$$

Abbreviations: B, channel breadth [m]; C_d, discharge coefficient [dimensionless]; D, hole diameter [m]; $f(y)$, normalized velocity profile at the inlet ($=V_1/V_0$); $g(y)$, normalized velocity profile at the outlet ($=V_2/V_0$); H, channel height [m]; N, number of holes [EA]; ΔP, pressure drop through the distribution panel [Pa]; P_0, pressure at the inlet [Pa]; P_1, pressure just upstream of the distribution panel [Pa]; P_2, pressure after the distribution panel [Pa]; P_b, pressure at the outlet [Pa]; t, panel thickness [m]; V_1 (y), velocity profile across the flow cross-section at the inlet [m/s]; V_2 (y), velocity profile across the flow cross-section at the outlet [m/s]; V_0, average velocity [m/s]; W, length of the separating plate [m]; x, coordinate along the flow direction; y, coordinate normal to the flow direction; α, parameter representing the change of velocity profile per hole; β, local porosity ($=D/(D+W_{(y)})$) [dimensionless]; β_0, average porosity [dimensionless]; ε, maximum % error between the predicted and the real results; ρ, density [kg/m3]

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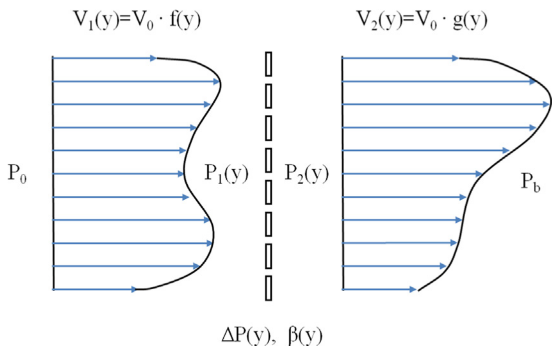

Fig. 1. Schematic diagram of the system and conditions used for the analysis.

Here, C_d is the discharge coefficient, which is close to 0.6 for sharp-edged orifices and 1.0 for long orifices with rounded entrances over a wide range of Reynolds numbers.

Consider a simple flow in a straight channel of uniform cross section, where a turbulent flow enters the channel with a non-uniform velocity profile, $V_1(y)/V_0=f(y)$, and exits with another non-uniform velocity profile, $V_2(y)/V_0=g(y)$ (Fig. 1). Pressure is uniform both at the inlet and at the outlet at P_0 and P_b, respectively. A perforated baffle of non-uniform porosity $\beta(y)$ is installed in the middle (Fig. 1).

If it is assumed that the pressure drop occurring in the open channel space is negligible compared to that through the perforated baffle, the pressure and velocity at any pair of cross sections can be correlated by the simple modified 1-D Bernoulli equation. Eq. (2). Considering the continuity condition, flow velocity profile just upstream and just downstream of the porous zone should be equal to that at the exit. After inserting Eq. (1) for ΔP in Eq. (2) and rearranging, Eq. (3) is obtained, which is the equation for $\beta(y)$ required to change the velocity profile from $f(y)$ at the inlet to $g(y)$ at the outlet. β_0 is the average porosity.

$$\begin{aligned} P_1+\frac{1}{2}\rho V_2(y)^2-\Delta P(y) \\ =P_0+\frac{1}{2}\rho V_1(y)^2-\Delta P(y) \\ =P_2(y)+\frac{1}{2}\rho V_2(y)^2=P_b(y)+\frac{1}{2}\rho V_2(y)^2 \end{aligned} \tag{2}$$

$$\left[\frac{1}{\beta(y)^2}-1\right]g(y)^2=\left[\frac{1}{\beta_0^2}-1\right]+\left[f(y)^2-g(y)^2\right]C_d^2 \tag{3}$$

Since the flow transformation is dominated by the pressure drop in the porous zone, all the above equations can be extended to flows with non-uniformities in two dimensions: $f(y)$, $g(y)$ and $\beta(y)$ can be extended to $f(y,z)$, $g(y,z)$ and $\beta(y,z)$.

The average porosity β_0 is arbitrary, but usually $0.4<\beta_0<0.6$ is considered a practical optimum (Ma et al., 2013), since a small opening (β_0) results in a large pressure loss and a large opening requires a long developing distance after the orifice.

3. Validation through numerical simulation

3.1. Model geometry for analysis

Validity of the proposed simple equation, Eq. (3), was first checked by numerical simulation. The system used for numerical analysis was a straight 2-D channel of height H (Fig. 2). The perforated panel was made of a linear array of N units of hole–plate pairs of uniform thickness t. Each unit was a slit of gap D with two platelets of length $W/2$ on both sides. The local porosity can be varied by either D or W: $\beta=D/(D+W)$.

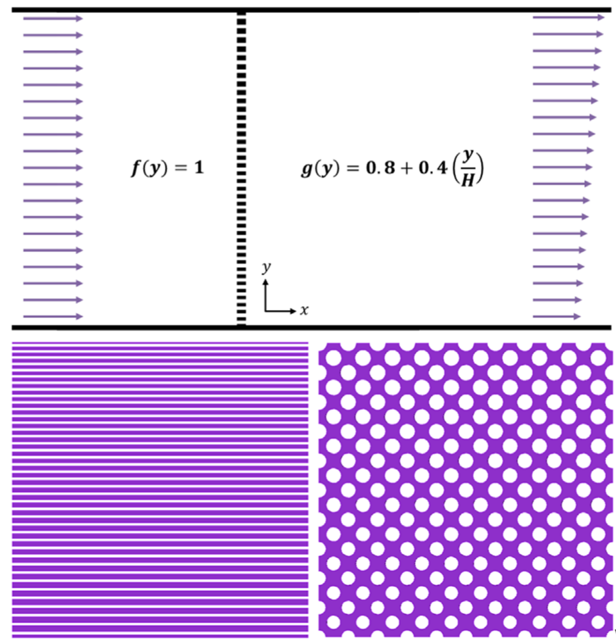

Fig. 2. Schematic of the panel for transforming the uniform velocity at the inlet to a linear profile at the outlet: (left) 2-D panel made of uniform slits (clear) and varying hole-to-hole gaps, and (right) 3-D plate made of holes instead of slits to the same porosity distribution, with variable hole size and uniform hole-to-hole distance. H is the channel height.

Though the design equation specifies a continuous distribution of porosity, porosity values were satisfied only at the pore centers in real situation. The panel designed using Eq. (3) for transforming a uniform flow, $f(y)=1.0$, into a linear profile, $g(y)=0.8+0.4(y/H)$, is shown in Fig. 2 as an example. The left panel is a 2-D panel made of slits with $\beta_0=0.4$, where the slit-to-slit distance (W) varied with the slit size (D) uniform, and the right panel is a 3-D plate made of holes of varied size and uniform hole-to-hole distance. Long slits with $C_d\sim1.0$ were used in this study to facilitate stable computation and better agreement with 1-D modeling, but the design equation is valid equally well for short slits (Choi et al., 2014). The distance between the inlet and the panel was 1.0H, and the transformed velocity profile was checked at $x/H=1.0$.

3.2. Numerical technique

Steady turbulent flow field for the system shown above was obtained numerically from the 2-D Reynolds-averaged Navier–Stokes (RANS) equations using the commercial software package Fluent 13.0. An improved k–ε turbulence model and an enhanced wall function, both installed in the software, were used for better speed and convergence (ANSYS® Inc., 2010). Air at 300 K and 1 bar was used as the working fluid.

Rectangular grids were used, and the convergence criterion was set such that the sum of the absolute residuals of the sources for the velocities, turbulence kinetic energy, and turbulence dissipation rate were all less than 10^{-5}. Total number of grids was about 640,000, which gave a good enough resolution of the flow field, and the maximum difference in velocity obtained with finer grids of about 1,100,000 points was less than 1%.

3.3. Results

The validity of Eq. (3) was first checked for two simple cases of: (a) geneating a linear velocity profile, $g(y)=0.8+0.4(y/H)$, from a uniform flow, $f(y)=1.0$, and (b) generating a skewed velocity profile, $g(y)=-1.5(y/H)^4+1.5(y/H)^2+0.8$, from another skewed