Contents lists available at SciVerse ScienceDirect

Chemical Engineering Research and Design

journal homepage: www.elsevier.com/locate/cherd

Modeling pressure losses for Newtonian and non-Newtonian laminar and turbulent flow in long square edged orifices

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ABSTRACT

A lack of experimental data and predictive models prompted the determination of loss characteristics of four sharp square-edged orifices for laminar to turbulent flow regime ($1 \leq Re \leq 100{,}000$). Novel experimental data for β ratios of 0.36, 0.4, 0.5 and 0.7 obtained with Newtonian and non-Newtonian fluids and an empirical correlation for predicting pressure losses through long square edged orifice plates is presented. For turbulent flow, new experimental results compared well with existing predictive models, thus validating the experimental results. Comparison of existing correlations as well as the new correlation shows that, although with some shortcomings, good progress is made toward a design correlation that spans a wide range of laminar to turbulent flow conditions for long orifices.

Keywords: Long orifice; Loss coefficient; Laminar; Turbulent

1. Introduction

An orifice in which, the jet formed downstream of the orifice entry tends to re-attach to the orifice wall is defined as a long orifice. The re-attachment normally occurs at a *t/d* ratio greater than 0.75 (Ward-Smith, 1979). A schematic diagram of a long square edged orifice plate is presented in Fig. 1. In traditional hydraulics, orifice plates have been used predominantly for flow-metering purposes. Long orifices are used as fluid flow controller, as choked flow meter, in gas turbine-aircraft engine, food industry, mineral processing and petroleum industry, etc. However, recently they have also been applied on a large scale for the economic dissipation of energy in flood conduits (Zhang and Chai, 1999). Orifices (chokes) have also been used for this purpose in slurry hydro transport, a first-reported use being the Samarco pipeline in 1977 (Derammelaere and Shou, 2002). Orifice plates possess a number of features, i.e. reliable performance, low installation cost, simplicity, well documented performance, etc. (Morrison et al., 1990). Knowledge of the excess pressure loss due to local disturbances such as orifices, are of considerable engineering interest aside from the flow-metering characteristics (Lakshmana Rao and Sridharan, 1972). In 1930, Johansen initiated the first detailed studies on sharp edged concentric orifices. The experimental results indicated that a variation in β ratio in turbulent flow does not vary the discharge coefficient in high range. In 1972, Lakshmana Rao and co-workers obtained the pressure loss coefficients for five sharp square-edged long orifices with a constant β ratio of 0.2, varying the thickness to diameter ratio from 0.48 to 10.11 in laminar flow. A review published by ESDU (2007) identified the lack of pressure loss coefficient and discharge coefficient data in the open literature. It established that determination of experimental pressure loss and discharge coefficient for long square-edged orifices for β ratio of 0.3, 0.4, 0.5 and 0.7 has not been attempted at that time. Data and correlations found in open literature are generally for turbulent flow regimes for which the loss coefficients mainly depends on the geometry of the orifice plate, practically independent of Reynolds number. No suitable correlation was found in the literature to predict pressure losses through long square edged orifices from laminar to turbulent flow regimes.

To address the issue of scarcity of data and correlations in the literature, a study was undertaken to experimentally determine pressure loss coefficient data for different orifice geometries and to derive a correlation to predict pressure loss coefficients in both laminar and turbulent flow regimes.

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Received 19 November 2010; Received in revised form 25 October 2011; Accepted 4 November 2011

doi:10.1016/j.cherd.2011.11.001

Nomenclature

β	Diameter ratio
$\dot{\gamma}$	Shear rate (1/s)
ΔP_{or}	Orifice pressure drop (Pa)
μ_{ge}	Generalized viscosity (Pa s)
μ_{re}	Relative viscosity
μ_r	Ratio of actual to reference viscosity
Π	Constant
ρ	Fluid density (kg/m^3)
τ	Shear stress (Pa)
τ'	Empirical parameter used in the Idel'chik et al. (1994) equation
τ_0	Wall shear stress (Pa)
τ_y	Yield stress (Pa)
C_1	Correlation constant
C_2	Correlation constant
C_d	Discharge coefficient
D	Pipe diameter (m)
D_{shear}	Shear diameter (m)
d	Orifice bore diameter (m)
E_u	Euler function
g	Gravitational constant (m/s^2)
g_e	Geometry factor
H_{or}	Orifice head loss (m)
K_{or}	Orifice loss coefficient
K	Fluid consistency index (Pa s^n)
K'	Apparent fluid consistency index (Pa s^n)
K''	Equation constant (Hasegawa et al., 1997)
n	Flow behaviour index
n'	Apparent flow behaviour index
Q	Volumetric flow rate (m^3/s)
Re	Reynolds number
Re_D	Pipe Reynolds number
Re_{ge}	Generalized Reynolds number
Re_{Mod}	Modified Reynolds number
Re_{MR}	Metzner and Reeds generalized Reynolds number
t	Orifice plate bore thickness (m)
t/d	Aspect ratio
V	Fluid mean velocity (m/s)
V_{ann}	Average velocity in sheared annulus where shearing of a yield stress fluid takes place in a pipe (m/s)

1.1. *Definition and determination of the loss coefficient*

The loss coefficient is defined as the non-dimensionalized difference in the overall pressure between the ends of two long straight pipes when there is a fitting installed, and when there is no fitting installed (Miller, 1990). It requires knowledge of the frictional losses in the straight pipes as well as the losses encountered in orifices for accurate estimation of pressure losses in a hydraulic circuit. The loss due to the orifice is expressed as

$$K_{or} = \frac{\Delta P_{or}}{(1/2)\rho V^2} \tag{1}$$

where K_{or} is the pressure loss coefficient and ΔP_{or} is the pressure drop across the orifice.

In laminar flow a hyperbolic relationship exists between the loss coefficient and Reynolds number (Edwards et al., 1985). The loss coefficient is independent of Reynolds number in turbulent flow. In many cases it was shown that a Reynolds number should be used that accounts for the viscous characteristic of the fluids (Edwards et al., 1985; Polizelli et al., 2003; Fester et al., 2007). A Reynolds number (Re_{mod}) that can be used for Newtonian fluids, power-law and viscoplastic fluids is given by Slatter (1996):

$$Re_{mod} = \frac{8\rho V_{ann}^2}{\tau_y + K(8V_{ann}/D_{shear})^n} \tag{2}$$

where ρ is the density of the fluid, V_{ann} is the average velocity in annulus, τ_y is the yield stress, K is the fluid consistency index and n is the flow behaviour index. The rheological behaviour of these viscous fluids is often described by the Herschel–Bulkley equation (Chhabra and Richardson, 2008):

$$\tau_o = \tau_y + K(\dot{\gamma})^n \tag{3}$$

where τ_o is the shear stress and $\dot{\gamma}$ is the shear rate.

1.2. *Correlations for pressure loss coefficients*

A number of correlations have been published to predict pressure losses in long square edged orifices. Table 1 presents a summary of the available correlations found in the literature. The correlation developed by Ward-Smith (1971) had a root mean square error of 3.8% compared with their experimental data. The term τ' in the semi empirical correlation derived by Idel'chik et al. (1994) is an empirical parameter that depends on t/d and ranges from 0 to 1.35. The laminar flow equation published by Hasegawa et al. (1997) is only applicable for $\beta = 0.1$. The K'' values in the correlation were determined experimentally as 37.7. The only correlation, applicable for both Newtonian and non-Newtonian fluids flowing through long square edged orifices was derived by Bohra (2004). However, the equation is only applicable to $\beta = 0.023–0.14$.

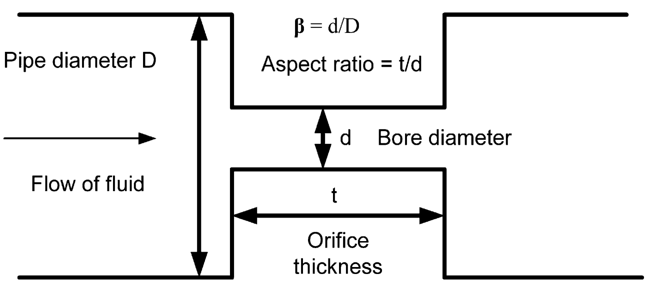

Fig. 1 – Schematic diagram of a long square edged orifice plate.

2. Experimental investigation

Experiments were conducted on a test rig (Fig. 2) in the slurry laboratory at the Cape Peninsula University of Technology. The test rig consists of five lines of PVC pipes with diameters ranging from 24 mm to 100 mm ID. Each line is 25 m long, to allow a fully developed flow before and after each test orifice. Test fluids were mixed in a 1.7 m^3 storage tank. The tank is rubber-lined to avoid chemical reactions of fluid with metal. The fluids were circulated from the mixing tank in a continuous loop using a progressive cavity positive displacement pump. The pump was driven by a 15 kW electric motor and variable speed