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Analytical solution of flow coefficients for a uniformly distributed porous channel

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Abstract

A general theoretical model is introduced to calculate flow distribution and pressure drop in a channel with porous wall. Analytical solution of nonlinear ordinary differential equations, based on the varying flow coefficients, was obtained, and comparison was made with the solution with flow coefficients. Predicted flow distribution agrees well with experimental data.

Keywords: Porous pipe; Dividing flow; Manifolds; Gas distributor; Flow distribution; Radial flow reactor

1. Introduction

Uniform mixing between species in a reactor depends on flow distribution. Hence, flow distributors are extensively encountered in many chemical processes. One of the most commonly used devices employs a porous pipe with constant cross-sectional area, as shown in Fig. 1. In this pipe, some fluid flows through the holes in the pipe wall and the remainder flows downstream at decreasing flow rates. The static pressure in such a pipe varies along the length due to two causes: (a) the friction of the fluid against the internal surface of the main channel makes the pressure fall in the flow direction; (b) the momentum of the main fluid stream flowing into a manifold tends to carry the fluid toward the closed end, where an excess pressure is produced. Therefore, it is possible to obtain a uniform pressure along the pipe axis by suitable adjustment of the flow parameters or good design so that the pressure recovery due to flow branching balances the pressure losses due to friction. However, the flow, even in this simple pipe distributor, is also complicated; the flow strongly depends on the geometry of the device, such as opening, spacing, dimension, etc. It would be too much effort to test the effect of all the geometric shapes of distributor, so theoretical simulation is a better approach.

The analysis of the performance of a porous pipe is traditionally based on the Bernoulli theorem, the energy theorem, or the conservation of momentum theorem. The difficulty with applying a Bernoulli equation to the varying mass process is identifying a relevant streamline to conserve energy and estimate frictional losses. In addition, because the lower energy fluid in the boundary layer branches through the holes the higher energy fluid in the pipe center stays in the pipe. So the average specific energies in a cross-section will be higher in the downstream than in the upstream. If an energy balance is based on the average value of the cross-section, these higher specific energies cannot be corrected and lead to an error. Hence, according to the First Law of Thermodynamics, when the specific mechanical energies are multiplied by the relevant mass flow rate terms, the mechanical energy after branching for the manifold can apparently be greater than the approaching energy. Alternatively, if the specific energy equation is used on the flow streamlines, there will be an equation corresponding to every dividing flow streamline. Thus, there will be many equations for a manifold system and the energy theorem becomes rather complicated. Recent researchers have avoided this problem by applying a momentum equation along the porous pipe [1,2]. The advantage of applying the momentum balance is that one does not need to know detailed flow patterns and the flow process can be simplified. Any errors due to simplification can be corrected with a pressure recovery factor and friction factor. However, for the past several decades an analysis has been based on the assumption of

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Nomenclature

d	lateral outlet diameter
D	the diameter of pipe distributor
E	the ratio of distributor length to diameter
Eu_{x0}	Euler number $(= (p_x - p_0)/\rho w_0^2)$
F_0	cross-sectional area of pipe distributor
F_1	cross-sectional area of a hole
H	lateral head loss coefficient
k	pressure recovery factor
L	active length of pipe distributor
n	the number of lateral outlets
p	static pressure in the distributor
p_c	ambient pressure
$\bar{P}$	dimensionless pressure $(p/\rho w_0^2)$
s	spacing distance
u	hole velocity
w	axial velocity in the distributor
w_c	axial component of the pipe distributor velocity
w_x	axial velocity at point x
x	axial coordinate
$\bar{x}$	dimensionless axial coordinate (x/L)

Greek symbols

α, β	coefficients
ρ	fluid density
λ	friction factor
τ_w	wall shear stress
ζ	correction factor

Subscripts and superscripts

0	beginning of distributor
$'$	differentiation

the constant friction and pressure recovery factors. In practice, the two factors would vary along the manifold because of varying flow velocity. Therefore there are limitations to apply the constant factors to the momentum equation. Analysis using the constant factors would result in two problems: (a) a large error, and (b) difficulty in analyzing flow mechanisms and model behavior, because the solution includes the pressure recovery factor and friction factor, both being dependent on the flow rate and manifold structure. Most researchers assumed the pressure recovery factor is constant when solving the momentum equations. Some researchers have recognized the problems. Haerter [3] assumed that the pressure recovery factor decreases linearly from the entrance value to an assumed value, 0.2, at the closed end. However, the assumption is unreasonable because the pressure profile is nonlinear along the manifold and the value at the closed end is not equal to 0.2. Bajora [1] recognized variation of friction factor with Reynolds number according to a power law, but he did not obtain the solution based on the power law. In practice, the effects of friction were considered negligible for a short manifold. Also, Bassiouny and Martin [2] neglected friction loss but they also used a constant friction factor and a constant pressure recovery factor. Their models were applied only for a short manifold. A study by Shen [4] showed that even for short manifolds the friction effects on the flow distribution are not negligible. But Shen also used a constant friction factor and a constant pressure factor.

For these reasons, it is desirable to investigate the possibility of obtaining a solution to the momentum equation for manifold flow under the condition where the factors vary along the manifold axis. In this paper the widely scattered values of flow coefficients for different system geometries have been systematically analyzed. A generalized analytical method for varying flow coefficients is given for both a short and a long manifold.

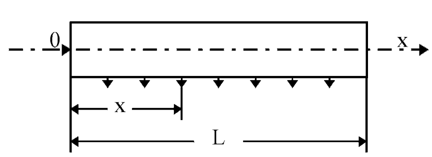

Fig. 1. Flow schematics for diving flow.

2. Analytical model

Fig. 1 shows the schematic of a manifold pipe. The development of our theoretical flow model is based on the following assumptions:

1. the ambient pressure does not vary;
2. the manifold pipe is of constant cross-sectional area and has equally spaced holes of uniform size at a right angle;
3. based on (a) and (b), a uniform discharge per unit length depends on a uniform static pressure along the length of the distributor.

Consider a section of the pipe distributor near one of the branching outlets as shown in Fig. 2. The hole part can be regarded in all practical cases as a quasi-continuous system. Therefore, equations for the dividing flow configuration are given as follows:

Momentum balance

$$\frac{1}{\rho}\frac{\mathrm{d}p}{\mathrm{d}x} + \frac{\lambda}{2D}w^2 + 2w\frac{\mathrm{d}w}{\mathrm{d}x} + \frac{F_1 n}{F_0 L}uw_c = 0 \quad (1)$$

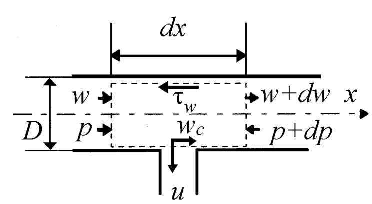

Fig. 2. Control volume for branch point.

Mass balance

$$u = -\frac{F_0 L}{F_1 n} \frac{dw}{dx} \quad (2)$$

Lateral discharge

$$p - p_c = H\rho \frac{u^2}{2} \quad (3)$$

A detailed derivation process is given in [1,2]. w_c stands for the axial velocity component of the fluid in the distributor, which will be branched off through the holes. It can be smaller or larger than w, depending on the dimension of the distributor and the location of the holes. If w_c is expressed as a fraction of w, namely $w_c = (2-2k)w$, and Eq. (2) and w_c are substituted into Eq. (1), one obtains

$$\frac{1}{\rho}\frac{dp}{dx} + \frac{\lambda}{2D} w^2 + 2kw\frac{dw}{dx} = 0 \quad (4)$$

Generally, there are two approaches to solve this flow distribution problem.

The first approach requires the design or adjustment of the side openings to establish prescribed flow conditions along the distributor, usually a uniform outflow along the pipe length. For a given pipe area and the desired distribution, the mean velocity can be calculated at each station x or a linear velocity distribution along the length of pipe. Eq. (4) can then be integrated with constant k and λ to give the pressure distribution along the pipe.

The second approach is to establish the flow and pressure distributions for a given distributor configuration. This involves the simultaneous solution of Eqs. (4) and (3). In general it requires a computer. For a short pipe, where friction losses can be neglected, analytical solutions are possible.

Obviously, either approach involves the variation of k and λ with the main velocity. No existing models consider this problem. In this paper, solutions are given for varying factors.

3. Flow coefficients

It has been shown that many of the existing experimental flow factors are empirical and inconsistent [5]. Such failure results from a lack of understanding of the fundamental phenomena. This study attempts to systematically analyze the existing experiment data, and to obtain results that can be used for calculation purposes.

3.1. Friction factors

The effect of the manifold structure on friction coefficient is well known. However, the degree of such an effect has not been fully studied. Unlike the data for a smooth pipe, the value of friction factor for porous pipes or manifolds varies widely. Generally, there are three explanations for the friction coefficient of porous pipes. Firstly, some researchers [6] recognized that rough peaks caused by small holes result in an increase in wall friction factor. Secondly, Haerter [3] assumed that the friction of a porous pipe is one third of the friction loss for a smooth pipe. Davis's experiments [7] show that the flow friction factor decreases through a T-junction. A series of T-junctions such as manifolds, decreases the flow friction coefficient. Finally, Acrivos et al. [8] indicated that friction factors computed from the pressure gradients were in agreement with the well established relations for smooth pipes; the effect of the branching flow on the wall friction was not apparent, at least in the region within an inch or two of the side port (note: $s/d > 4 \sim 8$). Based on more experimental data, Wang [9,10] indicated that there are three ratios which affect the friction factors of porous pipes, namely, the ratio of hole diameter to pipe diameter (d/D), spacing length to hole diameter (s/d), and sum of the areas of all holes to the cross-sectional area of pipe ($nd^2/4DL$). Wang [5] also summarized the friction factors for various pipes as follows:

1. When d/D and $nd^2/4DL$ are small, and s/d is large, the wall friction factor increases smaller since rough peaks of small hole jets is small compared with the whole wall friction factor. The wall shear stress in the nonporous sections of a manifold is nearly the same as in a long, straight pipe and so it can be predicted from existing data on pipe friction.
2. When d/D and s/d are small, and $nd^2/4DL$ is large, the effect of rough peaks of small hole jets becomes significant. This increases the wall friction. On the other hand, because of small d/D the effect of the sudden expansion of flow passage due to branching on the flow boundary layer is not obvious. Hence, the wall friction factor will increase.
3. When d/D and $nd^2/4DL$ are large, and s/d is small, the rough peaks do not occur, and branching flow results in an sudden expanding flow passage. The flow boundary layer could not be fully developed. The wall friction factor then decreases.

According to the above analyses, a general expression can be introduced for the friction factor:

$$\lambda = \zeta\lambda' = \zeta f(Re) \quad (5)$$

where λ' is the smooth pipe coefficient, which is given by Nikuradse's experiments; ζ a correction factor. In the above three cases ζ is greater than 1 for case 2, equal to 1 for case 1, and smaller than 1 for case 3.

The value of friction factor is affected by the pipe material or surface finish. For pipes commonly used for the chemical reactors ζ equals 1 corresponding to case 1. Hence, the well-known expression [9] for the friction coefficient can be introduced for porous pipes,

$$\lambda = \frac{64}{Re} \quad \text{for } Re < 2200 \quad (6)$$

$$\lambda = \frac{0.3164}{Re^{0.25}} \quad \text{for } 2200 < Re < 10^5 \tag{7}$$

$$\lambda = 0.0032 + \left(\frac{0.221}{Re^{0.237}}\right) \quad \text{for } Re > 10^5 \tag{8}$$

3.2. The pressure recovery factor k

k in the theoretical model derivation is defined as the correction factor for the loss of some axial momentum when fluid is branched off through holes. A value of 1 for k implies that the flow leaves the manifold at a right angle and represents the maximum possible static pressure recovery. Under this interpretation it is possible to assemble the appropriate equations and basic data into a program to compute k. k is a function of the fluid velocity before and after the hole, namely $k = f(w_i, w_{i+1})$. Analytically k represents a fraction of the relative momentum difference ($\beta\Delta w^2/w^2$), i.e.

$$k = \alpha + \beta\frac{\Delta w^2}{w^2} \tag{9}$$

α is the pressure recovery factor through the first hole. α and β depend on the geometry of the manifold and are independent of the properties of the working fluid. The main geometrical dimension is the ratio of manifold length to diameter (*L*/*D*). β is also dependent on the ratio of the sum of all areas of holes to the pipe area. However, this representation is semi-theoretical. The mathematical expression is required for calculation.

3.2.1. The relative momentum difference

Because k varies linearly with the relative momentum difference, a mathematical expression can be derived as follows:

$$\frac{w_i^2 - w_{i+1}^2}{w_i^2} = \frac{(w_i + w_{i+1})(w_i - w_{i+1})}{w_i^2} = \frac{2w_\xi\,\Delta w}{w_i^2} \tag{10}$$

where $w_i + w_{i+1} = 2w_\xi$ is from the mean-value theorem.

Dividing both sides of Eq. (10) by Δx, and taking the limit when $\Delta x \to 0$ (note: both w_i and $w_{i+1} \to w$):

$$\lim_{\Delta x\to 0}\frac{w_i^2 - w_{i+1}^2}{w_i^2\,\Delta x} = \lim_{\Delta x\to 0}\frac{2w_\xi}{w_i^2}\left(\frac{\Delta w}{\Delta x}\right) = -\frac{2w'}{w} \tag{11}$$

or

$$\left(\frac{w_i^2 - w_{i+1}^2}{w_i^2}\right) = -2\frac{w'}{w} \tag{12}$$

Integrating (12) from 0 to x, one obtains

$$\frac{w_i^2 - w_{i+1}^2}{w_i^2} = \int_0^x -\frac{2w'}{w}\,\mathrm{d}x = -2\ln\frac{w}{w_0} \tag{13}$$

Finally, substituting Eq. (13) into (9), one obtains

$$k = \alpha + 2\beta\ln\frac{w}{w_0} \tag{14}$$

At the inlet end, $x = 0$ and $k = \alpha$. That is to say, α is the pressure recovery factor k_0 of the first branch. At the closed end, $x = L$ and $\Delta w^2/w^2 = -1$. Applying these boundary conditions to Eq. (9), $k = \alpha - \beta$. $\alpha - \beta$ is the increment of the pressure recovery coefficient per unit length and β the pressure recovery coefficient of the last branch. Obviously, this representation is different from the rule-of-thumb formulas. The present k representation gives a physical meaning for the two constant factors, α and β. This shows the advantage of the present analyses. Hence, using data of Haerter's, we can obtain the pressure recovery factor for the first branch, α, for $L/D = 1 \sim 1000$. Wang [8] give $\alpha \cong 0.5$, $\beta \cong 0.146$ for $L/D = 20 \sim 30$, and $\alpha \cong 0.6$, $\beta \cong 0.15$ for $L/D = 30 \sim 40$. Both are similar.

3.2.2. The main flow velocity

If the holes are of the same size and their distribution along the length of the pipe is uniform, uniform fluid flow through the holes requires that the rates of flow through the pipe must vary linearly from a maximum at the inlet to zero at the dead end, i.e.

$$w = a(x + b)$$

with the following boundary conditions:

$$x = 0, \quad w = w_0, \qquad x = L, \quad w = 0$$

Substituting boundary conditions into the above equation, one obtains

$$a = -\frac{w_0}{L}, \quad b = -L \tag{15}$$

Hence, $w = w_0\,(1 - x/L)$.

4. Solution of equation

Now, we need to solve the varying mass momentum equation (4). In the derivation of Eq. (4), k and λ have not been restricted to be constant quantities. Hence, Eq. (4) is also suitable for varying k and λ along a pipe manifold. The flow coefficient expressions have been derived above. Fortunately, the design or adjustment of the distributor in chemical engineering requires usually a uniform outflow along the pipe length. Hence, the linear distribution of velocity can be been assumed, and the set of equations is closed.

Inserting Eqs. (5) and (14) into Eq. (4) gives

$$\frac{1}{\rho}\frac{\mathrm{d}p}{\mathrm{d}x} + \frac{\zeta f(Re)}{2D}w^2 + \left(\alpha + 2\beta\ln\frac{w}{w_0}\right)\frac{\mathrm{d}w^2}{\mathrm{d}x} = 0 \tag{16}$$

Inserting Eq. (15), one obtains

$$\frac{1}{\rho}\frac{\mathrm{d}p}{\mathrm{d}x} + \frac{\zeta f(Re)}{2D}w_0^2\left(1 - \frac{x}{L}\right)^2 + w_0^2\left[\alpha + 2\beta\ln\left(1 - \frac{x}{L}\right)\right] \times \frac{\mathrm{d}(1 - x/L)^2}{\mathrm{d}x} = 0 \tag{17}$$

The friction factor (λ) is dependent on the flow region and this is represented by Eqs. (6)–(8). Integrating Eq. (17) from 0 to x, we have the following solution for the three flow regions.

1. for $Re \leq 2200$

$$Eu_{x0} = \alpha[1-(1-\bar{x})^2] - \frac{16\upsilon L}{w_0 D^2}[1-(1-\bar{x})^2] - 2\beta\left[(1-\bar{x})^2\ln(1-\bar{x}) - \frac{1}{2}\bar{x}(\bar{x}-2)\right] \quad (18)$$

or

$$Eu_{x0} = \left(\alpha - \frac{16E}{Re_0}\right)[1-(1-\bar{x})^2] - 2\beta\left[(1-\bar{x})^2\ln(1-\bar{x}) + \frac{1}{2}\bar{x}(2-\bar{x})\right] \quad (19)$$

$$Eu_{x0}|_{x\to 0} = \alpha - \frac{16E}{Re_0} - \beta$$

where $Eu_{x0} = (p_x - p_0)/\rho w_0^2$, $\bar{x} = x/L$, and $E = L/D$.

2. for $2200 < Re < 10^5$

$$Eu_{x0} = \alpha[1-(1-\bar{x})^2] - \frac{0.058E}{Re_0^{0.25}}[1-(1-\bar{x})^{2.75}] - 2\beta\left[(1-\bar{x})^2\ln(1-\bar{x}) - \frac{1}{2}\bar{x}(\bar{x}-2)\right] \quad (20)$$

$$Eu_{x0}|_{x\to 0} = \alpha - \frac{0.058E}{Re_0^{0.25}} - \beta$$

3. for $Re \geq 10^5$

$$Eu_{x0} = \alpha[1-(1-\bar{x})^2] - \frac{0.0032E}{6}[1-(1-\bar{x})^3] - \frac{0.04E}{Re_0^{0.237}}[1-(1-\bar{x})^{2.763}] - 2\beta\left[(1-\bar{x})^2\ln(1-\bar{x}) - \frac{1}{2}\bar{x}(\bar{x}-2)\right] \quad (21)$$

$$Eu_{x0}|_{x\to 0} = \alpha - \frac{0.0032E}{6} - \frac{0.04E}{Re_0^{0.237}} - \beta$$

The above solutions are applicable to a variable friction factor for different Re regions. To compare with constant factor solution Eq. (19) for Re less than 2200 is written as follows:

$$Eu_{x0} = \left\{\alpha[1-(1-\bar{x})^2] - 2\beta \times \left[(1-\bar{x})^2\ln(1-\bar{x}) - \frac{1}{2}\bar{x}(\bar{x}-2)\right]\right\} - \frac{\lambda_0 E}{6}[1-(1-\bar{x})^3]\frac{3(2-\bar{x})}{2(1+\bar{x}^2)} \quad (22)$$

where $\lambda_0 = 64/Re_0$.

On the other hand, the constant factor solution of Eq. (4) is from [11]

$$Eu_{x0} = k[1-(1-\bar{x})^2] - \frac{E\lambda}{6}[1-(1-\bar{x})^3] \quad (23)$$

If we compare the two terms on the right-hand side of Eqs. (22) and (23) the first term represents momentum effect, and the second term measures friction loss. In Eq. (22) the first term has a corrective part, $2\beta[(1-\bar{x})^2\ln(1-\bar{x}) - (\frac{1}{2})\bar{x}(\bar{x}-2)]$, which varies as $\bar{x}$. At $\bar{x}=0$, it equals 2β, and at $\bar{x}=1$, it is β. When the corrective part vanishes, the first term becomes the constant factor solution of Eq. (23). The second term also has a corrective part, $3(2-\bar{x})/2(1+\bar{x}^2)$, which also varies with $\bar{x}$. When the corrective part equals 1, λ becomes the constant form of Eq. (23). Obviously, when $\bar{x}=0$, $3(2-\bar{x})/2(1+\bar{x}^2)=3$, that is, if Eq. (4) is solved with the inlet end friction factor, λ_0, the friction loss will be $\frac{2}{3}$ lower. When $\bar{x}\approx\frac{1}{2}$, $3(2-\bar{x})/2(1+\bar{x}^2)\approx 1$ the calculated value with the present λ will be similar. Furthermore, when $\bar{x} < \frac{1}{2}$, $3(2-\bar{x})/2(1+\bar{x}^2) > 1$, the calculated friction loss with constant friction factor will be less; when $\bar{x} > \frac{1}{2}$, $3(2-\bar{x})/2(1+\bar{x}^2) < 1$, the loss will be larger. Therefore, because of varying corrective terms, the calculation with the constant factor will give rise to errors. Similar analysis can be made for $2200 < Re < 10^5$ and $Re > 10^5$.

5. Results and discussion

Fig. 3 shows the effects of E/Re_0 on Eu_{x0} for the case of $Re \leq 2200$. The parameter, E/Re_0, combines three important characteristics of the manifold: namely, ratio of length to diameter (L/D), the entrance dynamic energy, and friction force. As E/Re_0 increases, i.e. E increases or Re_0 decreases, Euler number, Eu_{x0}, decreases. The negative values indicate that friction effects dominate. Obviously, it is possible to alter the ratio of length to diameter or the entrance dynamic energy to improve system performance. A uniform

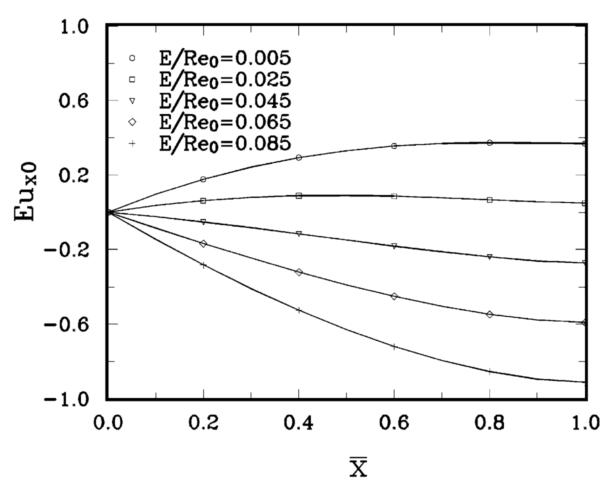

Fig. 3. Pressure profiles for distributor as varying E/Re_0.

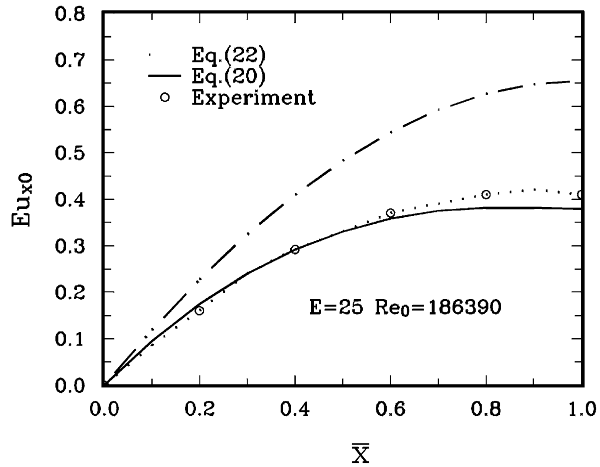

Fig. 4. Comparison of analytical model with experimental data for $Re_0 = 186\,390$.

pressure along the dividing flow manifold can be achieved through appropriate adjustment of the flow parameters so that the pressure recovery due to flow branching balances the pressure losses due to friction.

To validate the analytical solutions, experiments were performed using a pipe of 21 mm ID and 525 mm active length with 21 side holes bored at intervals of 3 mm. The static pressure in the main channel is observed by means of multiple tube pressure taps. A detailed description of experimental facility and test conditions is given by Wang [9]. Figs. 4 and 5 present a comparison between the theoretical results and experimental data.

Figs. 4 and 5 show that the solution using the constant factors is larger than the experimental data at the closed end. For a short pipe ($E = 25$), the effect of momentum is dominant. In Eq. (22) there is a corrective part for the momentum term, $2\beta[(1-\bar{x})^2\ln(1-\bar{x}) - (\frac{1}{2})\bar{x}(\bar{x}-2)]$, which varies with $\bar{x}$. That is to say, the momentum term in the solution with varying factors is smaller than that using the constant factors. Hence, Eqs. (19) and (20) agree well with experimental data as a result of a friction corrective term and a pressure recovery corrective term.

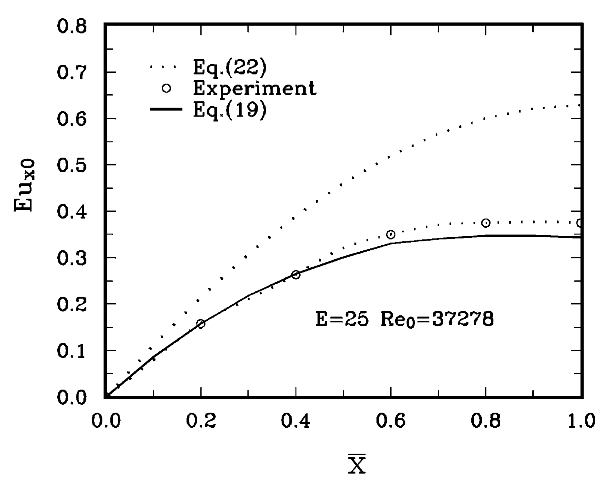

Fig. 5. Comparison of analytical model with experimental data for $Re_0 = 37\,278$.

6. Conclusions

The present model can be used to describe the performance of flow distributor or manifold systems in terms of geometrical and varying parameters. A general approach is taken in formulating the governing equations, and hence the model is also applicable to other geometrical configurations. The performance parameter E/Re_0 enables a simple assessment of the flow distribution in a given manifold system.

The pressure recovery coefficient has been expressed as a function of flow velocity. An analytical solution of the momentum equation with varying mass and varying coefficients was obtained. The solution of varying coefficients agrees with experimental results. The results can be used to optimize the design of new flow distributors or to achieve optimum operating conditions for flow systems. In addition, the solution can be used for modeling and performance assessment of operating processes. The solution with constant coefficients is a special case of varying coefficients when momentum correction is zero and friction correction is one. In the solution with varying coefficients the static pressure is dependent on geometrical parameters (the ratio of length and diameter) and flow parameter Re_0. In the design of a new distributor the structure of the distributor can be optimized.

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