A Numerical Procedure for Flow Distribution and Pressure Drops for U and Z Type Configurations Plate Heat Exchangers with Variable Coefficients

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Abstract. In Plate Heat Exchangers it is important to determine the flow distribution and pressure drops, because they affect directly the performance of a heat exchanger [1]. This work proposes an incompressible, one-dimensional, steady state, discrete model allowing for variable overall momentum coefficients to determine these magnitudes. The model consists on a modified version of the Bajura and Jones [2] model for dividing and combining flow manifolds. The numerical procedure is based on the finite differences approximation approach proposed by Datta and Majumdar [3]. A linear overall momentum coefficient distribution is used in the dividing manifold, but the model is not limited to linear distributions. Comparisons are made with experimental, numerical and analytical data, yielding good results.

1. Introduction
Manifold flow distributed systems are extensively used in a wide range of engineering applications. Among them we can mention: Plate heat exchangers, micro channel heat exchangers, fuel cells, solar collectors, etc. In these devices, where two manifolds are interconnected with several channels, it is important to determine the flow distribution and pressure drops through each channel between both manifolds, because they affect the efficiency and performance of the system [1] and are a basic framework for further thermal analysis.

The problem of exactly determining these magnitudes is a difficult task since the flow is 3-dimensional, of elliptic character and, in some cases, can be transient depending on the operating conditions. The system has to be modelled in such a way that the information obtained represents with accuracy the real flow in a reasonable computing time. In the past fifty years the problem has been studied using three different approaches [4]: (i) computational fluid dynamics (CFD), (ii) analytical models and (iii) discrete models. The CFD approach consists in solving 2-D or 3-D momentum, energy, mass and state equations, with turbulence models for closing the problem. This model provides the most detailed flow and offers the advantage of not requiring knowing in advance the loss associated to the flow turning between manifold and channel neither the overall momentum nor friction coefficients that consider the irreversibilities associated with separated flow. However, the computational cost and empiricism of the turbulence models are limiting factors of this kind of approach.
On the other hand, analytical and discrete approaches rely on 1-D formulations considering mean values of velocities and pressure at each section, taking advantage of the slenderness of the manifold and channel tubing. Obviously, the flow is far away from being 1-D, so it is necessary to introduce well founded coefficients to ensure that the 1-D flow approaches the real flow. The analytical models are particular cases of the governing equations of some discrete models. These analytical models give the designer a simplified view on how the fluid is distributed among channels and to estimate the pressure losses, but no analytical solution has been found for the case when the coefficients are correlated with the flow variables. In this scenario, discrete models incorporate the capability to solve problems with variable coefficients of any kind. This, jointly with the development of numerical algorithms and the nowadays computational power available make this kind of approach a very promising tool for analyzing and optimizing manifold systems.

Another factor influencing the velocity and pressure distributions is the arrangement type; the two basic ones are the “U” and “Z” type. The difference between them is how the fluid exits the manifold; as Figure 1 shows.

![Figure 1. Manifold distribution systems arrangement of conduits, with seven identical channels between two manifolds. The header (divider) in the lower part a) Type U, b) Type Z.](image)

2. Previous work

In the open literature there is ample experimental, theoretical and numerical work done to determine the flow distribution and pressure drops in manifolds. Kubo and Ueda [5], Zeisser [6], and McNown [7] are examples of experimental on combining and/or dividing manifolds. Kubo and Ueda [5] studied the pressure drop in dividing manifolds and developed some formulae to calculate the flow rate through each channel and the loss coefficients ($\sigma_D$ and $\sigma_C$) associated with lateral turning. They also observed that the mass flow distribution in each channel is almost independent of the Reynolds number at the inlet. Zeisser [6] and McNown [7] made experimental measurements to determine the pressure change coefficient ($\beta$), (see Eq. 1 for details). They found that this coefficient is almost constant in a dividing manifold and it is insensitive to channel/manifold velocities and diameters ratio, but it can increase about 20% when the spacing between channels decreases. They also measured this coefficient for a combining manifold, but in this case, the interaction between the fluid streams is different and more complex; as a result, the pressure change coefficient is highly dependent on the channel/conduit hydraulic diameter ratio.

Considering the flow as 1-D, a straightforward use of the Bernoulli theorem is generally not suitable because it is not possible to establish steady stream lines where the energy is conserved. To avoid this difficulty, some researchers have employed differential equations formulations that arise from applying momentum conservation in a control volume. Some of these approaches can be found in Markland [8] and Acrivos et al. [9] just to mention a few. In some cases these formulations have analytical solutions, corresponding to the analytical models commented in the introduction section. If these solutions are not possible, it is necessary to obtain approximate solutions under numerical schemes, and these are called “discrete models”.

Among the analytical models that have been built for manifold flows, one of the more extended models is the one proposed by Bassiouny and Martin [10, 11] who found an analytical solution to the non-friction flow in the manifolds. This work introduced the characteristic parameter $m^2$ that quantifies the flow behaviour; the solutions found were for both, U and Z type configurations. Later on, Wang [12, 13] also found analytical solutions for both configurations, but considering the manifold friction term. In the two analytical methods mentioned above, a common hypothesis is used:
the flow distribution system is considered as an infinite succession of channels in a continuum fashion. This over-predicts the pressure drop, due to the lateral turning loss coefficient ($C_d$). More recently, Wang [4] developed a numerical procedure with the analytical solution for friction in the manifold [12,13] using a predictor-corrector strategy to obtain distributions allowing variable friction factors and pressure change coefficient ($\beta$), see Eq. 1. In addition, Bajura and Jones [14] first proposed models considering discrete channels for both combining and dividing lateral inlets. They also offered an extensive compilation of experimental work, introducing the pressure change coefficient in the models, which take into account the non-conservation of axial linear momentum in the manifolds. Subsequently, they developed a model for U and Z type configurations [2]. They also performed an experimental research on combining and dividing manifold systems. One of the most useful results in this work was the measurement of the overall momentum coefficient ($C_T$) in Eq. 1, which quantifies the axial linear momentum transported through the channel, establishing this parameter as 1.05±0.05 and 2.66±0.05, for dividing and combining manifold respectively. In a further study, Datta and Majumdar [3] proposed a discrete model via the finite differences method based on the non-variable overall momentum coefficient version of the Bajura and Jones model [2]. Their results show good agreement with experimental data.

3. Present work

The present work proposes a 1-D, steady-state and parabolic discrete model introducing the novelty of allowing for variable overall momentum coefficient and friction factors in the manifold. The discretization is made via finite differences method. The coupling of velocities and pressure is implemented with a pressure correction (SIMPLE) algorithm, Patankar [15]. Another feature is that the equations and boundary conditions are expressed with the use of the auxiliary functions $\epsilon$, $\delta$ and $\kappa$. They allow a compact formulation for “Z” and “U” distribution system types.

3.1. Overall momentum coefficient and friction factor

Inside a manifold, the flow is not uniform in every section, as assumed by the 1-D simplification. In fact, it has a certain transversal profile. The lower velocity near the walls due to viscous effects implies less energy, so when the fluid passes a channel inlet, the fluid with less axial velocity will enter the channel. That entrance is not instantly parallel to the channel axis because of inertia, so part of the upstream axial energy is transferred to the mass that enters the channels. In addition, an error can be made in quantifying the upstream axial momentum because of the uniform velocity profile simplification. So in order to correct this, a coefficient is introduced for the header flow; the so-called header momentum coefficient ($\alpha$) or simply the Boussinesq coefficient. Mathematically, the header momentum, pressure change ($\beta$) and overall momentum coefficient ($C_T$) are defined as:

$$\alpha = \frac{1}{W^2} \cdot \int \frac{W^2 \cdot dA}{A} \quad \beta = \frac{1}{W \cdot U_e \cdot A} \cdot \int W \left(A_e^2 - U_e^2\right) \cdot dA_e$$

$$C_T = \frac{1}{2} (2\alpha - \beta)$$

(1)

It is important to mention that $\beta=0$ implies that the manifold flow reaches the maximum static pressure recovery downstream the channel inlet.

By observations made by Bajura and Jones [2], the overall momentum coefficient $C_T$ is approximately fixed when the channel manifold diameter ratio is greater than 0.5. Those values in $C_T$ are 1.05 and 2.66 for dividing and combining manifolds respectively, but in order to explore a possible enhancement, in the present work we used a linear variation of $C_T$ from 1.3 to 1.05 between $z=0$ and $z=0.2$ to take into account the readjustment of the velocity profile after the first few laterals channel inlets [14]. The values of $d\alpha/dz$ are obtained differentiating Eq. 1 assuming that $\beta$ is constant and equal to 0.8 [6,7]. In the combining manifold, the value of $C_T$ is fixed to 2.66 [14] and the values of $\beta$ are calculated from the experimental values given by McNown [7] and Zeisser [6]. It is relevant to notice that the present formulation is not restricted to a linear distribution of $C_T$; and other kind of functions
can be introduced. The friction factors in the manifolds and the channels have been determined with the Blasius formula for the manifold and the Rao and Sunden [1] correlation for the channels:

\[ f = 0.3164 \cdot Re^{-0.25} \]  
\[ f_c = 21.41 \cdot Re_c^{0.301} \]

Where \( Re \) is the Channel Reynolds number based on twice the plate spacing.

### 4. Mathematical formulation

For the mathematical formulation here proposed, steady-state, 1-D and incompressible flow has been assumed. The analysis uses the control volume shown in Fig. 2 where, the total mass enters through section A, some fluid goes through section B and the remaining enters the channel through section C with a certain angle implying components of velocity \( W_c \) and \( U_c \). It has been considered that the shear stress at the walls can be approximated by the Darcy-Weisbach formulation \( \tau = f \cdot \rho \cdot W^2 / 8 \). With these assumptions, applying conservation of mass and linear momentum, the governing equations in dimensionless form are:

\[ \frac{d}{dz} \left( C_f \cdot w^2 \right) - w^2 \cdot \frac{dc_f}{dz} = - \frac{dp}{dz} \left( \frac{L \cdot f \cdot S}{8 \cdot A} + \frac{d\alpha}{dz} \right) \cdot w^2 \]

\[ \frac{dw}{dz} = \frac{\varepsilon \cdot u_c}{\Delta \varepsilon} \left( \frac{A_c}{A} \right) \]

Equation 4 is a modified version of the Bajura and Jones model, for further details about the derivation of the original version of Eqs. 4 and 5, see Bajura and Jones [2]. The modification consists in introducing inside the convective term the overall momentum coefficient \( C_T \). To do so, the chain rule is applied to the original convective term. The introduction of this factor inside the convective term allows an easy integration over the control volume when it is variable.

![Figure 2. Combining and dividing flows control volumes for: a) Type U, b) Type Z](image)

### 4.1. Boundary conditions

The velocities can be specified at both boundaries, for either a dividing or a combining manifold. In particular, the pressure can be specified only in \( z = 0 \) for the dividing manifold and in \( z = 1 \) for the combining manifold for type U or \( z = 0 \) for type Z configuration in the combining manifold. The generalized boundary conditions for velocity and pressure in dimensionless form are:

\[ w(z = 0, \delta, \varepsilon) = \frac{1}{2} \left( 1 + \varepsilon \right) + \frac{1}{2} \left( 1 - \delta \right) \cdot \left( \frac{A}{A^*} \right) \]

\[ w(z = 1, \delta, \varepsilon) = \frac{1}{4} \left( 1 + \delta \right) \cdot \left( 1 - \varepsilon \right) \cdot \left( \frac{A}{A^*} \right) \]

\[ p(z = 0, dividing) = 0 \]

\[ p^*(z = 0, combining) = 0 \quad \text{if} \quad \text{Z type} \]

\[ p^*(z = 1, combining) = 0 \quad \text{if} \quad \text{U type} \]
4.2. Auxiliary integer functions \( \delta \) and \( \epsilon \)

The \( \delta \) function defines the direction of the flow respect to the \( z \) axis and the \( \epsilon \) function determines if mass enters or leaves the control volume for the mass conservation equation. Those functions take values of 1 or -1 and they are defined as:

\[
\delta = (-1)^{\gamma} \quad \epsilon = (-1)^{\kappa}
\]

\[
\kappa = \begin{cases} 
2 & \text{if Dividing Manifold} \\
1 & \text{if Combining Manifold}
\end{cases} \quad \gamma = \begin{cases} 
2 & \text{if Dividing Manifold} \\
1 & \text{if Combining Manifold}
\end{cases} \quad \eta = \begin{cases} 
2 & \text{if U Type} \\
1 & \text{if Z Type}
\end{cases}
\]

4.3. Velocity in the channels

To solve Eqs. 4 and 5, a first assumption on the magnitude of channel velocity was made, as commented on sub-section 5.3. Therefore, in every iteration level, the velocity in channels must be recalculated. For that purpose, applying conservation of linear momentum in the channels, the channel velocity can be calculated as:

\[
(u_e)_i = C_d \cdot \left\{ 2 \cdot (p_i - p^*) \right\}^{1/2}
\]

Where \( C_d \) is the channel discharge coefficient, defined as: 
\[
C_d = \left[ \left( \frac{f_c \cdot L_c}{d_c} \right)_i + \sum_{j \neq i} \xi_i \right]^{-1/2}
\]

5. Discrete equations and numerical procedure

The finite differences approximation of the momentum equation is obtained by integrating Eqs. 3 and 4 over the control volume shown in Fig. 3. The source terms (friction and overall momentum coefficient correction terms) are approximated using a Taylor series expansion around the iteration “\( k \)”. The other details can be found in Datta and Majumdar [3]. The discrete form of the equation is as follows:

\[
A_e \cdot w_e^k = A_n \cdot w_n^k + A_W \cdot w_W^k + A \cdot (p_p - p_E) + S_e
\]

Where

\[
A_n = (C_f)_e \cdot A \cdot w_e^{k-1} \quad A_W = -\left( (C_f)_{e+1} \cdot A \cdot w_e^{k-1} \right) \quad A = A_n + A_E + S_p
\]

\[
S_e = \left( \frac{L \cdot f \cdot S}{8A} \right) \left( \frac{d \alpha}{d \xi} \right) (w_e^{k-1})^2 \quad S_p = \left( \frac{L \cdot f \cdot S}{4A} \right) \left( \frac{d \alpha}{d \xi} \right) w_e^{k-1}
\]

5.1. Velocity-pressure coupling and pressure correction equation

A straightforward use of Eq. 5 is not suitable because the pressure does not appear explicitly, so another equation must be employed to guarantee the conservation of mass. To do so, the semi-implicit method for pressure linked equations (SIMPLE) algorithm is used, Patankar [15]. The resulting form of the pressure correction is as follows:

\[
(D_e + D_n) \cdot \Delta p_p = D_e \cdot \Delta p_E + D_n \cdot \Delta p_w - \epsilon \left( \frac{A}{A_e} \right) (u_e)_i + (w_e^* - w_e^k)
\]

Where

\[
D_e = \frac{A_E}{A_e} \quad D_n = \frac{A_n}{A_e}
\]
5.2. Pressure and velocity correction equation

\[ w' = w_c + D_c \cdot (\Delta p_p - \Delta p_E) \]

\[ p_p = p'_p + \Delta p_p \]  \hspace{1cm} (9)

5.3. Numerical procedure

The numerical procedure is based on a predictor-corrector strategy. The idea is to obtain solutions for velocities and pressure for fixed coefficients. Then, \( C_T \) and \( \beta \) are recalculated and the process is repeated until the flow coefficients converge. Briefly, the algorithm is as follows:

a) Start with an initial guess for \( w', u_c, \) and \( \beta \).

b) Solve Eq. 8 to get \( w' \) and with this value solve the pressure correction equation, Eq. 9. In order to accelerate the algorithm, the linear systems of Eqs. 8 and 9 are solved with the TDMA algorithm.

c) Correct axial velocities and pressures, Eq. 10.

d) Correct channel velocities with Eq. 7. Check if conservation of mass is fulfilled; if not, correct channel velocity, [13].

e) Recalculate \( C_T \); if \( |C_T^{k+1} - C_T^k| \leq tol \) finish, else go to step b) and repeat the process.

6. Numerical experiments

This section offers a comparison of the present model with experimental and analytical results. There is lack of experimental data for channel velocity and pressure distributions in plate heat exchangers. These devices can be seen as a manifold system with high lateral resistance, since; the friction in the channels is in the order of 50 times the friction in the manifolds. Because of that, the first case of our study consists in the numerical replication of the high lateral resistance experiment made by Bajura and Jones [14]. For this numerical experiment we present the dimensionless pressure distributions (\( P_c, P_d \) and \( \Delta P \), see Eq. 11). The comparison is also made with the numerical results with constant coefficients of Datta and Majumdar [3]. Later on, the dimensionless channel velocity (\( u_c \)) distribution is compared with the analytical solution of Bassiouny and Martin [10, 11], obtaining the characteristic flow parameter \( m^2 \) as the solutions of Eq. (12) or (13). The characteristics of this case are given in Table 1.

| \( W_0 \) | \( D \) | \( L \) | \( d_c/D \) | \( l/D \) | \( Exp. H \) | \( C_d \) |
|---|---|---|---|---|---|---|
| 8.98 m\( \cdot \)s\( ^{-1} \) | 0.106 m | 1.56 m | 0.375 | 2.55 | 12.2 | 0.4048 |

The second and third case of study in this paper consist in the numerical replication of the experiments made by Rao and Sunden [1], in the second case the influence of the number of channels and mean channel Reynolds number is studied. Finally, the third case is the study of the dimensionless channel velocity (in respect to the mean channel velocity) distribution for port diameters of 70 mm and 35 mm and inflow conditions of 3.6 L\( \cdot \)s\( ^{-1} \) and 0.13 L\( \cdot \)s\( ^{-1} \). The characteristics of these two cases are presented in Table 2. In all cases the study was made for both U and Z type configurations.

| \( p_c = \frac{P^*-P_{exit}}{P_{inlet} - P_{exit}} \) | \( p_d = \frac{P - P_{exit}}{P_{inlet} - P_{exit}} \) | \( \Delta P = \max \left[ \frac{P - P^*}{P_{inlet} - P_{exit}} \right] \) |
|---|---|---|

Table 2. Geometric characteristics of the plate for cases 2 and 3.

| Port diameter (\( D \)) | 70 - 35 mm |
| Vertical distance between ports (\( L \)) | 0.6 m |
| Chevron angle | 60° |
| Spacing between plates (\( b \)) | 2.9 mm |
| Hydraulic diameter of the channels | 5.8 mm |
| Plate width (\( w \)) | 0.4 mm |
| Number of plates (\( N \)) | 10, 15 or 18 |
\[ p_{\text{inlet}} - p^*_{\text{exit}} = \frac{\zeta_c}{2} \left( \frac{A}{N \cdot A_c} \right)^2 \cdot \left[ \frac{m}{\tanh(m)} \right]^2 \text{ for U type} \quad (11) \]

\[ p_{\text{inlet}} - p^*_{\text{exit}} = \frac{\zeta_c}{2} \left( \frac{A}{N \cdot A_c} \right)^2 \cdot \left[ \frac{m}{\tanh(m)} \right]^2 + \lambda \cdot \left( 1 + \frac{\lambda}{m^2} \right) \cdot \left[ \frac{1 - \sech(m)}{\tanh(m)} \right]^2 \text{ for Z type} \quad (12) \]

Where:
\[ \lambda = \frac{2 - \beta}{\zeta_c} \left( \frac{A_c \cdot N}{A} \right)^2 \]

7. Discussion of results

Figs. 4.a and 4.b show the dimensionless pressure for U and Z type for case 1. For both of them, the predictions are in good agreement with experimental and numerical data, but, in figure 4b the numerical results with constant coefficients (Datta and Majumdar) in the dividing manifold, pressure distribution are slightly more in accordance with experimental data. One reason for this discrepancy is the linear overall distribution function assumed in this work, the real form of this function is of different shape and therefore must be established empirically. Fig. 5 shows the dimensionless channel velocity distribution, calculated with the present model, and the analytical solution of the Bassiouny and Martin model. The analytical velocity profiles are calculated with the value of the flow parameter \( m^2 \) determined with the numerical pressure distribution (Figs. 4.a and 4.b). In these figures a greater deviation in the Z type configuration is observed. The deviation can be in part because the analytical model does not take into account the friction losses in the manifolds. However, the behaviour of the present numerical model corresponds to the one predicted by the analytical one.

![Figure 4](image1.png)

Figure 4 Pressure profiles for case 1, a) U type, b) Z type.

![Figure 5](image2.png)

Figure 5. Channel velocity distribution comparison with analytical models a) U type, b) Z type.

Fig.6 presents the total pressure drop in the modelled heat exchanger and again the deviation from experimental results is larger for the Z type configuration but it is also shown that this difference is not
only affected by the configuration type, but also by the number of channels and the channel Reynolds number (as a consequence of the inflow Reynolds number). Finally, Figs. 7 and 8, show that for port diameters of 70 mm the model reproduces almost perfectly the experiments but for diameters of 35 mm, predictions of the channel velocity distribution slightly differ from the experimental measurements. This occurs because in the calculation of the lateral discharge coefficient, the pressure loss due to sudden expansion or contraction in the channels was neglected; and for lower port diameter this expansion or contraction is greater, so the velocity in the first channels and the last ones for U and Z type configurations respectively are greater than the prediction made. From the results obtained it can be said that the present model reproduces with a good degree of accuracy the experiments. Moreover a model with variable coefficients has been obtained, but in order to improve the model it is necessary to find either analytically or experimentally the form of the overall momentum coefficient distribution along the dividing manifold.

Figure 6. Dimensionless total pressure drops with a port diameter of 70 mm: a) U type, b) Z type.

Figure 7. Dimensionless channel velocity distribution for Type U with 15 channels: a) $Q=3.6 \text{ L/s}$, b) $Q=0.13 \text{ L/s}$

Figure 8. Dimensionless channel velocity distribution for Type Z with 15 channels: a) $Q=3.6 \text{ L/s}$, b) $Q=0.13 \text{ L/s}$

8. Conclusions
On the grounds of the detailed comparison shown, it can be concluded that with the model developed:
a) The predictions are in accordance with experimental data and are especially accurate for U type configurations.
b) The velocity distributions are in accordance with the velocity predicted by the Bassiouny models.
c) To further improve the model it is desirable to find an analytical or experimental model for the overall momentum coefficient distribution along the dividing manifold.

References
[1] Rao B, Sunden B, Das S 2005 An experimental and theoretical Investigation of the effect of flow maldistribution on the thermal performance of Plate Heat Exchangers J. Heat Transfer, 127(3), 332-45.
[2] Bajura R A, Jones E H 1976 Flow distribution manifolds J. Fluids Eng. 98(4) 654-665.
[3] Datta A B, Majumdar A K 1980 Flow distribution in parallel and reverse flow manifolds Int. J. Heat & Fluid Flow 2(4) 253-262.
[4] Wang J 2011 Theory of flow distribution manifolds Chemical engineering science 168 1331-45.
[5] Kubo T, Ueda T 1969 On the characteristics of divided flow and confluent flow in headers Bulletin of JSME, 12(52) 802-809.
[6] Zeisser M H Summary Report of Single-Tube Branch and Multi-Tube Branch Water Flow Tests conducted by the University of Connecticut Pratt and Whitney aircraft division, United aircraft corporation, Report No PWAC-231 USAEC Contract AT(11-1)-229, May 1963.
[7] McNown J S 1954 Mechanics of manifold flow Transactions ASCE 119 1103-42.
[8] Markland E 1959 Analysis of flow from pipe manifolds Engineering 187 150-151.
[9] Acrivos A, Babcock B D and Pigford R L 1959 Flow Distribution in manifolds Chemical engineering science 10 112-124.
[10] Bassiouny M K, Martin H, 1984 Flow distribution and pressure drop in plate heat exchangers-I: U type arrangement Chemical engineering science 39(4) 693-700.
[11] Bassiouny M K, Martin H, 1984 Flow distribution and pressure drop in plate heat exchangers-II: Z type arrangement Chemical engineering science 39(4) 701-704.
[12] Wang J, 2008 Pressure drop and flow distribution in parallel-channel fuel cell stacks: U-type arrangement. Int. J. of Hydrogen Energy, 33(21) 6339-6350.
[13] Wang J, 2010 Pressure drop and flow distribution in parallel-channel configurations of fuel cells: Z-type arrangement. Int. J. of Hydrogen Energy, 35(11) 5498-5509.
[14] Bajura R A, Jones E H 1971 A model for flow distribution in manifolds Journal of Engineering for Power Fluids Eng Trans ASME 93(1) 7-12.
[15] Patankar S 1980 Numerical Heat Transfer and Fluid Flow McGraw Hill.

Nomenclature

| Symbol | Description |
|--------|-------------|
| A      | Conduit area \((m^2)\) |
| A,D,S  | Discrete equation coefficients |
| c      | Channel velocity \((m\cdot s^{-1})\) |
| C      | Overall momentum coefficient. |
| D      | Conduit hydraulic diameter \((m)\) |
| d      | Channel hydraulic diameter \((m)\) |
| f      | Friction factor |
| Re     | Reynolds number \(= \rho \cdot W \cdot D / \mu\) |
| S      | Perimeter |
| \(\zeta\) | Head loss coefficient |
| \(\xi\) | Channel, combining manifold |
| \(c\) | |
| \(d\) | Dividing manifold |
| \(i, j\) | Index |
| \(W, E, P\) | Cardinal location for scalar variables |
| \(w, e, Ee\) | Cardinal location for velocity |
| Symbol | Description |
|--------|-------------|
| $H$    | Lateral resistance coefficient |
| $L$    | Length of manifold |
| $L_c$  | Length of channel |
| $m^2$  | Flow characteristic parameter |
| $N$    | Number of channels |
| $P$    | Dimensional pressure (Pa) |
| $P$    | Dimensionless pressure ($= (P - P_{ref}) / \rho \cdot W^2$) |
| $Z$    | Axial coordinate ($m$) |
| $Z/L$  | Superscripts |
| $k$    | Iteration level index |
| $\alpha$ | Header momentum coefficient |
| $\beta$ | Pressure change coefficient. |
| $\varepsilon$ | Combining and dividing functions |
| $\delta$ | Finite difference |

Greek symbols:
- $\kappa$ - Mean value
- $(*)$ - Approximate solution, comb. man.
- $(-)$ - Correction

Arrangement functions:
- $'$