Anisotropy effects on convective heat transfer and pressure drop in Kelvin’s open-cell foams

M Iasiello 1*, N Bianco*, W K S Chiu4 and V Naso

* Dipartimento di Ingegneria Industriale, Università degli studi di Napoli Federico II, P.le Tecchio 80, 80125, Napoli, Italy

# Department of Mechanical Engineering - University of Connecticut, 191 Auditorium Road, 06269, Storrs (CT), USA

1 E-mail: marcello.iasiello@unina.it

Abstract. Open-cell foams have the potential to offer heat transfer enhancement in many applications, such as heat exchangers, solar air receivers, porous burners, thanks to their high heat transfer surface to volume ratio and to tortuosity, that promotes the internal flow mixing. Their microstructural transport features are affected by foam anisotropy, and could play an important role not only because manufacturing processes can stretch the cells along a preferential direction, but also because modern techniques allow for the customization of the foam cell shape. However, structural anisotropy in open cell foams has been scarcely investigated. In this paper, anisotropy effects on convective heat transfer and pressure drop in Kelvin’s open-cell foams are analyzed. Since Kelvin’s model is geometrically regular, anisotropy has been investigated by stretching the foam along three orthogonal directions, at equal cell volume. Governing mass, momentum and energy equations have been solved using a finite element method. Results are presented for different fluid inlet velocities and cell sizes along three orthogonal stretching directions. They show that anisotropy affects the velocity and temperature fields, and, consequently, the permeability, inertial factor and volumetric heat transfer coefficients.

Nomenclature

| Symbol | Description |
|--------|-------------|
| a, b | Ellipsoid axes half lengths (m) |
| C_p | Heat capacity at constant pressure (J/kg K) |
| d | Diameter (m) |
| h | Heat transfer coefficient (W/m² K) |
| H_g | Hagen number |
| k | Thermal conductivity (W/m K) |
| L | Foam length (m) |
| Nu | Nusselt number |
| p | Pressure (Pa) |
| q | Heat flux (W/m²) |
| Re | Reynolds number |
| S | Heat exchange surface (m²) |
| T | Temperature (K) |
| u | Velocity vector (m/s) |
| V | Volume (m³) |
| x, y, z | Rectangular coordinates (m) |

Greek symbols:

- ε: Porosity
- μ: Dynamic viscosity (kg/m²·s)
- ρ: Density (kg/m³)

Subscript:

- c: Cell
- f: Fluid
- h: Hydraulic
- in: Inlet
- m: Maximum
- s: Solid
- v: Volumetric

Other symbols:

- <>: Volume average

Content from this work may be used under the terms of the Creative Commons Attribution 3.0 licence. Any further distribution of this work must maintain attribution to the author(s) and the title of the work, journal citation and DOI.

Published under licence by IOP Publishing Ltd
1. Introduction
For applications in which enhancing heat transfer is a primary aim, open-cell foams are a promising class of materials because of their weight, high effective thermal conductivity, mixing-flow characteristic and high heat exchange surface to volume ratio. They are used in various applications, such as volumetric solar receivers [1, 2], heat sinks [3-5], phase change materials [6, 7] and porous burners [8].

Flow and temperature fields in open-cell foams can be obtained experimentally, numerically or analytically. Since the scales of the problem are very small, the last two techniques allow one to carry out analysis at these scales. Due to the complex geometry, thermal behavior of open-cell foams is analyzed by employing the Volume Averaging Technique (VAT) [9], in which the whole foam is treated as an equivalent medium with variables averaged over a Representative Elementary Volume (REV). The local temperature difference between the two phases can be taken into account with a Local Thermal Non-Equilibrium (LTNE) model, with energy equations written for each phase and coupled with the so-called volumetric heat transfer coefficient.

The volumetric heat transfer coefficient, together with permeability and inertial factor for the momentum equation, can be obtained from experiments [10] or predictions. The latter can be performed by simulating the single-phase flow that passes through the foam, whose geometry needs to be modeled. The most accurate approach uses tomographic scans of real foam samples. However, a less costly approach models the geometry by making reference to geometrical models that represent the foam structure as accurately as possible.

The first models were proposed by Du Plessis and Masliyah [11] and Lu et al. [12]. In the former, the foam geometry was treated as three mutually perpendicular struts and the pressure drop was predicted. In the latter, the geometry was modeled as various orthogonal cylinders. During the years, the aim has been to describe the foam geometry as accurately as possible. To the authors’ knowledge, the most accurate foam cell model was proposed by Weaire and Phelan [13], that reduces the surface energy by a 0.3% when compared to the Kelvin’s foam model (that was the most accurate model until then). The Weaire and Phelan’s model was used by Boomsma et al. [14], Kopanidis et al. [15] and by Cunsolo et al. [16]. The Kelvin foam model was used by [17-19]. Iasiello et al. [18] compared the Kelvin’s foam model with a tomography-based model, while Lucci et al. [19] showed that a better accuracy is obtained if the Kelvin’s model is built up at equal surface to volume ratio. The comparison between Weaire and Phelan’s and Kelvin’s models carried out by Cunsolo et al. [16] exhibited similar Nusselt numbers and friction factors, especially for high porosities. The Kelvin’s foam model is computationally lighter since the unit cell consists of one polyhedron, whereas the Weaire and Phelan’s model contains eight polyhedrons.

Some of the studies carried out using Kelvin’s foam model took into account the effects of microstructural parameters, since they can strongly affect transport parameters. The effects of structural parameters, such as angulation and tapering, were analyzed by Pusterla et al. [20], in order to optimize heat transfer features. Ambrosio et al. [21] showed that the accuracy of the model is improved when the strut shape is accounted for. It is also widely known that, because of viscosity and gravity effects, a slight anisotropy stretches a foam in a certain direction during the foaming process. The foam cell anisotropy can be also induced with modern manufacturing techniques, like 3D printing [22], that allow for the customization of foam cells. Ranut et al. [23] investigated anisotropy in real foams and showed that the anisotropy effect is inversely proportional to PPIs, since increasing PPI makes the foam cells more regular.

Anisotropy effects in a Kelvin-based foam are analyzed in the present study. A 0.94 porosity foam is reconstructed by using Surface Evolver [24] and simulations of a stream of air under uniform solid/fluid boundary heat flux are carried out by means of a finite element scheme. Various fluid velocities, cell sizes and stretching factors are analyzed in order to evaluate anisotropy effects caused by the foam cell stretching along three orthogonal axes.

2. Geometrical model
The foam generation process is described in the following. An open-cell foam with 0.94 porosity is generated with the free-to-use software Surface Evolver [24]. The procedure is the same as that used
in [18] and [25]. After importing the Kelvin’s foam in the COMSOL Multiphysics software, the open-cell foam is modified in order to analyze anisotropy effects. Since the starting structure is perfectly isotropic, a sphere of volume \( V_c = \frac{4}{3} \pi (d_c/2)^3 \) and diameter \( d_c \) can be inscribed through the cell. At equal volume, the cell is stretched along one of the principal (\( x \), \( y \) or \( z \)) directions; thus, the cell is better represented as an ellipsoid rather than as a sphere. Under the hypothesis that the two minor axes of the stretched ellipsoid are equal, the cell volume is expressed as follows:

\[
V_c = \frac{4}{3} \pi \left(\frac{d_c}{2}\right)^3 = \frac{4}{3} \pi ab^2
\]  

where \( a \) is the half length of the axis along the stretching direction and \( b \) is the half length of the other two axes. It is worth noting that the elliptical nature of foam cells has been already discussed in the literature by defining two cell sizes that are the axes of the ellipse that approximates the foam cell structure [26]. The parameter \( a \) in equation (1) is varied in order to appreciate the foam anisotropy; on the other hand, the length of minor axes is obtained from the following equation:

\[
b = \left[ \left(\frac{d_c}{2}\right)^3 \right]^{1/2}/a
\]  

A resume of the stretching process is presented in figure 1. After stretching the foam cell, a boundary box is built up in order to distinguish the fluid domain from the solid domain. The latter is replaced by a boundary condition. Fictitious inlet and outlet regions are built up in order to make lead effects negligible, with the length prescribed depending on how much the foam is stretched.

3. Mathematical model

Mass, momentum and energy equations for the fluid phases (air) are solved under the hypothesis of laminar, steady and incompressible flow, negligible buoyancy effects and uniform thermophysical properties:

| Stretching direction | 0 direction | x direction | z direction |
|----------------------|-------------|-------------|-------------|
| Inscribed ellipsoid  | ![Inscribed ellipsoid](image) | ![x direction](image) | ![z direction](image) |
| Kelvin’s             | ![Kelvin’s x](image) | ![Kelvin’s z](image) |

**Figure 1.** Stretching process in the \( x \) direction and \( z \) direction (\( \theta \) direction refers to an isotropic cell).
\[ \nabla \cdot \mathbf{u} = 0 \]  
\[ \rho \mathbf{u} \cdot \nabla \mathbf{u} = -\nabla p + \mu \nabla^2 \mathbf{u} \]  
\[ \rho C_p \mathbf{u} \cdot \nabla T = k \nabla^2 T \]

where \( \mathbf{u} \) is the air velocity vector, \( \rho \) the density, \( p \) the pressure, \( \mu \) the dynamic viscosity, \( C_p \) the heat capacity at constant pressure, \( T \) the temperature, and \( k \) the thermal conductivity.

Boundary conditions of the problem are the same as those in [18] and [25]. At the inlet section, an air uniform temperature \( (T = 293.15 \text{ K}) \) plug flow is assumed. At the walls of the computational domain, symmetry conditions are employed, i.e., slip condition with no shear stresses and a zero temperature gradient condition. At the solid/fluid boundary, a no-slip condition with uniform entering heat flux is assumed. Finally, atmospheric pressure is assumed at the outlet section, with an outflow condition for heat transfer. Since both geometry and boundary conditions have symmetrical features, only a quarter of the cell is analyzed.

Governing equations with boundary conditions are solved with the Galerkin-based finite element method by means of the commercial code COMSOL Multiphysics. A tetrahedral mesh is employed in the computations, with a number of elements proportional to the cell size. Grid convergence has been verified in terms of heat transfer coefficients and pressure drop, by employing the numerical grid with deviations less than \( 10^{-3} \) when compared to two finer grids. A convergence criterion of \( 10^{-6} \) has been adopted.

It is worth noticing that few experimental data on convective heat transfer in open-cell foams are available in the literature, and, to the authors’ knowledge, none of them refers to anisotropic foams. Comparisons among various foam morphological parameters, presented by Iasiello et al. [25], exhibited deviations lower than about 30% in the worst case, with velocities far larger than those in the present paper. One can, therefore, argue a better accuracy of the model at the velocities investigated in the present study. The model had been already validated for an isotropic cell in previous studies by the same authors [18, 25].

4. Results and Discussion

Simulations are carried out for cell diameters equal to 1.0, 2.0, 3.0 mm; air inlet velocities equal to 0.5, 1.0, 1.5, 2.0 m/s, and stretching factor in the range \( 1a - 1.5a \). The imposed heat flux is equal to 1000 W/m², so that temperature variations are small enough to ensure the reliability of the uniform properties assumption. Scaled velocity fields for stretching along \( x \) and \( y \) directions, with stretching factors equal to \( 1a, 1.2a \) and \( 1.4a \), are presented in figure 2. Five slices are considered, at \( x/L = 0, 0.25, 0.50 \), and along the \( y \) direction: \( 1a \); \( 1.2a \); \( 1.4a \).

![Figure 2](image-url). Scaled velocity slices for stretching along the \( x \) direction: a) \( 1a \); b) \( 1.2a \); c) \( 1.4a \) and along the \( y \) direction: d) \( 1a \); e) \( 1.2a \); f) \( 1.4a \).
0.50, 0.75 and 1.0 in order to make the comparisons reliable. As to the \( x \) direction stretching (\( x \leftrightarrow y \)), figures 2a, 2b and 2c exhibit slight differences in the shape of the velocity profile along the flow direction. One can also notice that, at the same \( x/L \), the larger the stretching factor the more uniform the velocity profile in the cross section, since the decrease in the area available for fluid flow increases the fluid velocity in the core region. For the stretching along the \( y \) direction (\( y \leftrightarrow y \)) figures 2d, 2e, 2f show that velocity profiles are almost unaffected by the increase in the scaled velocity. It is likely due to the opposite effects of the stretching: the fluid velocity is decreased by the increase in the cross section available to the fluid whereas it is increased by the decrease in the domain length along \( x \) axis.

Iasiello et al. [27] compared various characteristic lengths for forced convection, and showed that the best scaling parameters for open-cell foams are the ratio of the cell size to the porosity and the hydraulic diameter. The predicted dimensionless pressure drop, convective heat transfer coefficient and volumetric convective heat transfer coefficient are presented in the following, using the hydraulic diameter as the scaling parameter. With reference to the hydraulic diameter

\[
d_h = 4\varepsilon S/V
\]

with \( \varepsilon \) the porosity, \( S/V \) the ratio of the heat transfer surface to the total foam volume [28], the Reynolds number, Re, is defined as

\[
Re = \frac{\rho |u|_m d_h}{\mu}
\]

where \( |u|_m \) is the air inlet velocity, that is equal to the ratio of the volumetric flow rate to the inlet section of the domain. In order to scale pressure drop and heat transfer coefficients, the Hagen number, Hg, and Nusselt number, Nu, are employed:

\[
Hg = \frac{\Delta p \rho d_h^3}{d_c \mu^2}
\]

\[
Nu = \frac{h d_h}{k} = \frac{|q|}{\langle T \rangle_s - \langle T \rangle_f} d_h
\]

where \( \Delta p \) is the pressure drop between the entrance and exit sections, \( |q| \) is the imposed heat flux, \( \langle T \rangle_s \) is the solid/fluid boundary average temperature and \( \langle T \rangle_f \) is the volumetric average fluid temperature. Finally, reference is also made to a volumetric heat transfer coefficient, \( h_v \), and to its dimensionless form, the volumetric Nusselt number, \( Nu_v \):

\[
h_v = \frac{h}{V} = Nu_v \frac{d_h^2}{k}
\]

It is worth noticing that the hydraulic diameter depends on both porosity and specific surface area. This means that scaling variables into their dimensionless form is meaningful, since scaled variables aren’t proportional to their dimensional form.

We can also notice that \( S/V \) slightly increases with the stretching factor (\( S/V \) at 1.5a is 1.05 times \( S/V \) at 1a); thus the hydraulic diameter, as well as the dimensionless numbers based on it, vary. On the contrary, they don’t vary when the stretching direction changes, the same stretching factor remaining the same. The pressure drop per foam length and the heat transfer coefficient, for a stretching factor equal to 1a, as a function of the cell diameter, at different inlet velocities, as well as a function of the inlet velocity, at different cell diameters, are presented in figures 3a and 3b, respectively. Both figures exhibit an increase in pressure drop and heat transfer coefficient at decreasing cell size and increasing inlet velocity. One can also remark that pressure drop and heat transfer coefficient are more affected by the cell size than by the inlet velocity.

The Hagen number as a function of the Reynolds number, for all the investigated values of inlet velocity and cell size and for stretching along the \( x \) direction and along the \( y \) direction, is presented in
Figure 3. Pressure drop per foam length and heat transfer coefficient vs.:
a) cell size, at different inlet velocities; b) inlet velocity, at different cell sizes.

Figs. 4a and 4b, respectively. Data for the stretching along \( x \) are more scattered than those along \( y \). They are correlated by the power laws reported in the figures, whose regression coefficients are rather different. Similar differences were also found between results for the stretching along the \( y \) direction and for the stretching along the \( z \) direction, not reported in this paper.

The pressure drop per foam length as a function of the stretching factor, for stretching along the three directions as well as for different cell diameters and inlet velocities, is presented in figures 5a and 5b, respectively. Both figures point out that the pressure drop increases at smaller cell diameter and larger inlet velocity. One can also remark a higher dependence of the pressure drop on the stretching factor when the foam is stretched along the \( x \) direction as well as a stronger effect when it is stretched along \( y \) and \( z \) directions. Since it is well known that the flow can hydrodynamically and thermodynamically develop in a foam [25, 29] and taking into account that stretching a foam along the \( x \) direction makes the domain longer, as depicted in figure 2, the flow is more prone to hydrodynamically develop, with a reduction in the pressure drop. On the other hand, stretching along the \( y \) and \( z \) directions makes the foam domain shorter along the flow direction, thus making the hydrodynamic development less likely and enhancing the pressure drop.

The Hagen number as a function of the stretching factor, for stretching along the three directions as well as for different cell diameters and inlet velocities, is presented in figures 6a and 6b, respectively. A slight increase with the stretching factor is remarked for \( y \) and \( z \) directions, while a strong decrease is found for higher stretching factors when the foam is stretched along the \( x \) direction. This likely occurs because the non-dimensionalization involves the hydraulic diameter, that depends on the
Figure 5. Pressure drop per foam length vs. stretching factor, for stretching along the three directions: a) $|u|_{in} = 1.0 \text{ m/s}$ and $d_c = 2.0 \text{ mm}, 3.0 \text{ mm}$; b) $d_c = 2.0 \text{ mm}$ and $|u|_{in} = 1.0 \text{ m/s}, 2.0 \text{ m/s}$.

specific surface area, which, in turn, slightly increases with the stretching factor.

The heat transfer coefficient as a function of the stretching factor, for stretching along the three directions as well as for different cell diameters and inlet velocities, is presented in figures 7a and 7b, respectively. Both figures show that the heat transfer coefficient increases when the cell diameter is smaller and when the velocity is larger. As far as the stretching effect is concerned, we can notice that the heat transfer coefficient is practically unaffected by the stretching factor for stretching along the $y$ and $z$ directions, whereas it decreases at increasing stretching factors along the $x$ direction. The above different behaviors can be caused by variations in the velocity slices due to developing effects for the stretching along the $x$ direction (see figure 2) that, on the contrary, do not occur when the foam is stretched along the $y$ and $z$ directions, thus making the heat transfer coefficient independent of the stretching factor.

The Nusselt number as a function of the stretching factor, for stretching along the three directions as well as for different cell diameters and inlet velocities, is presented in figures 8a and 8b, respectively. Since the heat transfer coefficient is made dimensionless through the hydraulic diameter, depending on $S/V$, the Nusselt number decreases at increasing stretching factor, which is larger when the foam is stretched along the $x$ direction.

The volumetric heat transfer coefficient as a function of the stretching factor, for stretching along the three directions as well as for different cell diameters and inlet velocities, is presented in figures 9a and 9b, respectively. Both figures show that the volumetric heat transfer coefficient is almost

Figure 6. Hagen number vs. stretching factor, for stretching along the three directions: a) $|u|_{in} = 1.0 \text{ m/s}$ and $d_c = 2.0 \text{ mm}, 3.0 \text{ mm}$; b) $d_c = 2.0 \text{ mm}$ and $|u|_{in} = 1.0 \text{ m/s}, 2.0 \text{ m/s}$.
independent of the stretching factor along the $x$ direction, whereas it increases when the foam is stretched along $y$ or $z$ directions. This can be physically explained by the following considerations. In the volumetric heat transfer coefficient, that can be viewed as the heat rate per foam volume, the imposed heat flux doesn’t vary, whereas the specific surface area increases with the stretching factor. When the foam is stretched along the $y$ or $z$ direction, the temperature difference $<T_s> - <T_f>$ is nearly independent of the stretching factor because of the negligible developing effects (see figure 2), and the heat rate per foam volume increases. Consequently the volumetric heat transfer coefficient increases as well. On the contrary, when the foam is stretched along $x$ direction the increase in the heat rate per foam volume is balanced by the increase in $<T_s> - <T_f>$ caused by developing effects (see figure 2).

The volumetric Nusselt number as a function of the stretching factor, for stretching along the three directions as well as for different cell diameters and inlet velocities, is presented in figures 10a and 10b, respectively. Similarly to the Nusselt number, also the volumetric Nusselt number decreases with the stretching factor, since it is inversely proportional to $S/V$.

5. Conclusions

Anisotropic effects on convective heat transfer and pressure drop in Kelvin’s foams have been investigated numerically by stretching a foam along three orthogonal directions. Pressure drop per unit length, heat transfer coefficients, volumetric heat transfer coefficients, and their dimensionless forms

![Figure 7](image7.png)  ![Figure 8](image8.png)
are presented, for different cell diameters, inlet velocities and stretching factors.

Results show that anisotropy of the foam affects the coefficients depending on the stretching direction, in particular hydrodynamic and thermal development. Pressure drop and heat transfer coefficient decrease with the stretching factor when the foam is stretched along the flow direction. In the orthogonal directions, pressure drop increases with the stretching factor whereas heat transfer coefficients are unaffected by the stretching factor. The increase of the surface to volume ratio at increasing stretching factors makes the dimensionless numbers dependent on the stretching factor. They generally decrease when the stretching factor increases, more markedly when the foam is stretched along the flow direction than in the orthogonal directions.

Figure 10. Volumetric Nusselt number vs. stretching factor, for stretching along the three directions: a) $|u|_{in} = 1.0 \text{ m/s and } d_c = 2.0 \text{ mm, } 3.0 \text{ mm}$; b) $d_c = 2.0 \text{ mm and } |u|_{in} = 1.0 \text{ m/s, } 2.0 \text{ m/s}$.

6. References
[1] Avila-Marín A L 2011 Solar energy 85 891-910.
[2] Wu Z, Caliot C, Flamant G and Wang Z 2011 Solar Energy 85 2374-85.
[3] Bhattacharya A and Mahajan R L 2002 ASME J Electronic Packaging 124 155-63.
[4] Hsieh W H, Wu J Y, Shih W H and Chiu W C 2004 Int. J. Heat Mass Transfer 47 5149-57.
[5] Andreozzi A, Bianco N, Iasiello M and Naso V 2017 J. Phys.: Conf. Series 796 012002
[6] Mancin S, Diani A, Doretti L, Hooman K and Rossetto L Int. J. Thermal Sciences 90 79-89.
[7] Di Giorgio P, Iasiello M, Viglione A, Mameli M, Filippeschi S, Di Marco P, Andreozzi A and Bianco N 2017 J. Phys.: Conf. Series 796 012032.
[8] Bouma P H and De Goey L P H 1999 Combustion and Flame 119 133-43.
[9] Vafai K 2015 Handbook of porous media (Boca Raton: CRC Press).
[10] Kamiuto K 2005 *Int. Comm. Heat Mass Transfer* **32** 947-53.
[11] Du Plessis J P and Masliyah J H 1988 *Transport in Porous Media* **3** 145-61.
[12] Lu T J, Stone H A and Ashby M F 1998 *Acta Materialia* **46** 3619-35.
[13] Weaire D and Phelan R 1994 *Phil. Mag. Letters* **69** 107-10.
[14] Boomsma K, Poulikakos D and Ventikos Y 2003 *Int. J. Heat Fluid Flow* **24** 825-34.
[15] Kopanidis A, Theodorakakos A, Gavaises E and Bouris D 2010 *Int. J. Heat Mass Transfer* **53** 2539-50.
[16] Cunsolo S, Iasiello M, Oliviero M, Bianco N, Chiu W K S and Naso V 2016 *ASME J. Heat Transfer* **138** 022601.
[17] Wu Z, Cailot C, Flamant G and Wang, Z 2011 *Int. J. Heat Mass Transfer* **54** 1527-37.
[18] Iasiello M, Cunsolo S, Oliviero M, Harris W M, Bianco N, Chiu W K S and Naso V 2014 *ASME J. Heat Transfer* **136** 112601.
[19] Lucci F, Della Torre A, Montenegro G, Kaufmann R and Eggenschwiler P D 2017 *Int. J. Heat Mass Transfer* **108** 341-50.
[20] Pusterla S, Barbato M, Ortona A and D’Angelo C 2012 *Int. J. Heat Mass Transfer* **55** 7902-10.
[21] Ambrosio G, Bianco N, Chiu W K S, Iasiello M, Naso V and Oliviero M 2016 *Appl. Thermal Engineering* **103** 333-43.
[22] Ortona A, D’Angelo C, Gianella S, and Gaia D 2012 *Materials Letters* **80** 95-8.
[23] Ranut P, Nobile E and Mancini L 2014 *App. Thermal Engineering* **69** 230-40.
[24] Brakke K A 1992 The surface evolver *Exp. Math.* **1** 141-65.
[25] Iasiello M, Cunsolo S, Bianco N, Chiu W K S, Naso V 2017 *Int. J. Thermal Sciences* **111** 129-37.
[26] De Jaeger P, T’Joen C, Huissene H, Ameel B and De Paepe M 2011 *J. Appl. Phys.* **109** 103519.
[27] Iasiello M, Cunsolo S, Bianco N, Naso V, Oliviero M, Harris W M and Chiu W K S 2014 *Proc. 3d Int. Conf. Computational Methods for Thermal Problems (Bled, SL)* Napoli: Massimo Giannini Editore) 117-20.
[28] Dietrich B, Schabel W, Kind M and Martin H 2009 *Chem. Eng. Science* **64** 3633-40.
[29] Dukhan N and Suleiman A S 2014 *Transport in Porous Media* **101** 229-46.