Prediction and measurements of the pressure and velocity distributions in cylindrical and tapered rigid ceramic filters

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Abstract

A rigid ceramic filter is an effective gas filtration device to remove particles from gases at high temperature. It is important to study the gas flow dynamics in both the filtration and reverse flow mode in order to achieve better filter cleaning. Previous studies have shown that the axial pressure and velocity profiles are not usually uniformly distributed along the filter element. The non-uniformities may lead to insufficient filter cleaning. The uneven pressure difference across the wall of the filter element can be reduced by employing a tapered filter. A one-dimensional model is developed to predict the pressure and flow rate profiles, taking into account for the effects of both momentum changes and friction. The calculations show a good agreement with the experimental results. It is found that the friction factor term in the model has a strong influence in the reverse flow simulation.

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Keywords: Ceramic filters; Tapered filter; Pressure difference; Flow resistance; Friction factor

1. Introduction

A rigid ceramic filter is an effective gas filtration device that is designed to remove particles from gases at high temperature. Rigid ceramic filters were first utilised in both combustion and gasification of coal for environmentally clean power generation in the early 1980s[1,2]. Direct application of high-temperature particulate control ceramic filters is expected to be beneficial not only to the advanced fossil fuel processing technology, but also to selected high temperature industrial and waste incineration processes.

It is important to study the gas flow dynamics in both the filtration and reverse flow modes in order to achieve better filter cleaning. Several studies have been carried out on filtration and pulse gas cleaning on ceramic filters. All these studies have shown that the axial pressure is not usually uniformly distributed along the filter element.

Differential pressure measurements along the rigid ceramic filter candles have been mostly performed at room temperature[3–6]. Measurements at high temperatures have been described and reported by Berbner and Löffler [7], Hajek and Peukert [8] and Ito et al. [9]. Table 1 summarises the literature concerning the flow characteristics during cleaning of various ceramic filter candles.

Velocity measurements in ceramic filter elements during pulse cleaning have been carried out by Baik et al. [10], Christ and Renz [11], and Biffin et al. [12]. Baik et al. [10] found that a new filter candle will exhibit a certain degree of inhomogeneity and has the possibility of blinding (dust cake remaining accumulated on the filter surface). This dust will remain on the candle surface after cleaning. These factors will severely affect the flow within the candle, both under operation and cleaning. Christ and Renz [11] measured the axial velocity with a laser doppler anemometer in order to validate the numerical simulations of a single filter element during pulse cleaning. Biffin et al. [12] measured the axial velocity along the filter candle using a hot-wire probe. In their studies, a larger duration pulse showed no significant influence on the maximum velocity level inside the filter.
It is believed that one way in which the pressure difference generated across the wall of the filter element can be made more nearly uniform along its length is to vary the cross sectional area of the element, most simply by tapering from the open end to the closed end. A variation on the “traditional” filter candle geometry, the tapered filter is expected to show some promise of a more even pressure drop distribution on reverse pulse cleaning which should improve the cake detachment [13,14]. There is little published data describing the pressure drop distribution along the tapered filter axis. It is useful to investigate the pressure drop distribution as it may help to understand how the filter cake is formed on the filter surface and hence lead to improvements in the cleaning mechanism and novel design of filter geometry. Hence, experimental studies of tapered filters were undertaken in order to understand the effect of the filter geometry on the pressure difference distribution.

Table 1
Overview of publication concerning flow characteristics during recleaning of various candle types

<table>
<thead>
<tr>
<th>Candle type (manufacturer)</th>
<th>D_i (mm)</th>
<th>D_o (mm)</th>
<th>Length (mm)</th>
<th>Porosity, ε (%)</th>
<th>Resistance (kPa/(m s))</th>
<th>Data published by</th>
</tr>
</thead>
<tbody>
<tr>
<td>Ceramic filter (not indicated)</td>
<td>40</td>
<td>60</td>
<td>500</td>
<td>30-48</td>
<td>NI</td>
<td>[21]</td>
</tr>
<tr>
<td>Tube filter (not indicated)</td>
<td>40</td>
<td>70</td>
<td>1000</td>
<td>NI</td>
<td>NI</td>
<td>[9]</td>
</tr>
<tr>
<td>Tube filter (NGK)</td>
<td>40</td>
<td>70</td>
<td>1000</td>
<td>NI</td>
<td>NI</td>
<td>[9]</td>
</tr>
<tr>
<td>Cerafil S-1000 (FOSECO)</td>
<td>40</td>
<td>60</td>
<td>1000</td>
<td>86</td>
<td>NI</td>
<td>[4]</td>
</tr>
<tr>
<td>KEES550 (BWF)</td>
<td>42</td>
<td>60</td>
<td>975</td>
<td>83</td>
<td>1</td>
<td>[5,6]</td>
</tr>
<tr>
<td>KEES550 (BWF)</td>
<td>42</td>
<td>60</td>
<td>675</td>
<td>93</td>
<td>1</td>
<td>[5,6]</td>
</tr>
<tr>
<td>Schumalith 40 (SUT)</td>
<td>30</td>
<td>60</td>
<td>1000</td>
<td>37</td>
<td>8.6</td>
<td>[6]</td>
</tr>
<tr>
<td>Schumalith 20 (SUT)</td>
<td>40</td>
<td>60</td>
<td>1000</td>
<td>36</td>
<td>19.4</td>
<td>[6]</td>
</tr>
<tr>
<td>DIA-Schumalith 10-20 (SUT)</td>
<td>40</td>
<td>60</td>
<td>1000</td>
<td>37</td>
<td>29.6</td>
<td>[6]</td>
</tr>
</tbody>
</table>

NI: not indicated.

Table 2
Technical data for Cerafil XS-1000

<table>
<thead>
<tr>
<th>Property</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Candle length (mm)</td>
<td>1000 ± 2</td>
</tr>
<tr>
<td>Candle external diameter (mm)</td>
<td>60 ± 2</td>
</tr>
<tr>
<td>Candle internal diameter (mm)</td>
<td>40 ± 2</td>
</tr>
<tr>
<td>Filtration area (m²)</td>
<td>0.19</td>
</tr>
<tr>
<td>Density (kg m⁻³)</td>
<td>450 ± 5</td>
</tr>
<tr>
<td>Porosity (%)</td>
<td>86 ± 2</td>
</tr>
<tr>
<td>Chemical analysis (after firing) Al₂O₃ (wt.%)</td>
<td>43.7</td>
</tr>
<tr>
<td>SiO₂ (wt.%)</td>
<td>56.3</td>
</tr>
<tr>
<td>Filtration efficiency (wt.%)</td>
<td>&gt;99.9</td>
</tr>
<tr>
<td>Temperature limit (°C)</td>
<td>900</td>
</tr>
</tbody>
</table>

Fig. 1. Schematic diagram of the tapered filter.

2. Experimental methods

2.1. Filters materials

2.1.1. Cylindrical filter

The Cerafil XS-1000 filters were provided by the Scapa Filtration Europe Ltd. (Broadway Mill, UK). The technical specifications are given in Table 2. They are vacuum formed on the inside of a permeable mandrel from a slurry of Al₂O₃ and SiO₂ fibres (diameter, \( \phi \): 2–2.5 \( \mu \)m), water and a mixture of organic and inorganic binders. The water content is reduced by vacuum caking the solid components and the filter candle heated to dry it and solidify the binder phases.

2.1.2. Tapered filter

The filter element with its length tapering from the open end to the closed end is shown in Fig. 1. The material used to make the tapered elements is exactly the same as that used for the Cerafil XS-1000 filter elements. They are also...
Table 3

<table>
<thead>
<tr>
<th>Property</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Length (mm)</td>
<td>1250 ± 2</td>
</tr>
<tr>
<td>Filter external diameter</td>
<td></td>
</tr>
<tr>
<td>Open end (mm)</td>
<td>9.0 ± 2</td>
</tr>
<tr>
<td>Closed end (mm)</td>
<td>4.0 ± 2</td>
</tr>
<tr>
<td>Filter internal diameter</td>
<td></td>
</tr>
<tr>
<td>Open end (mm)</td>
<td>4.0 ± 2</td>
</tr>
<tr>
<td>Closed end (mm)</td>
<td>2.0 ± 2</td>
</tr>
<tr>
<td>Porosity (%)</td>
<td>88</td>
</tr>
<tr>
<td>Filtration area (m$^2$)</td>
<td>0.32</td>
</tr>
<tr>
<td>Temperature limit (°C)</td>
<td>900</td>
</tr>
</tbody>
</table>

vacuum formed on the inside of a permeable mandrel from a slurry of Al$_2$O$_3$ and SiO$_2$ fibres (diameter, φ: 2–2.5 μm), water and a mixture of organic and inorganic binders. This filter contains a proportion of ‘shot’ which is typically 100 μm diameter particles. Shot content is typically 20% by weight. It has an external surface area of 0.32 m$^2$, ≈73% greater than 1.0 m Cerafil filter element. The technical specifications of the tapered filter are given in Table 3. The determination of the porosity of the tapered filter is described by Chuah [15].

2.2. Filtration rig

The equipment test facility and set-up have been developed in the laboratory to investigate both filtration and the reverse flow process of low-density rigid ceramic filters at ambient conditions. All the filtration equipment designed and used in this work was equipped with flow control devices which were able to measure and control the face velocity and hence compensate for changes in flow resistance that take place after the cleaning event and during filtration. The experimental rig is shown schematically in Fig. 2. The single filter rig with the filter house is shown together with the fan, the controller and the measurement apparatus.

The filter vessel and its dimensions are shown in Fig. 3a. The dirty gas (air with lime dust) entry was 1.4 m above the top of the filter and 0.75 m from its open end. The element was mounted in place by forcing the open end of the filter candle against the header plate. This allowed an excellent seal to be made by using a rubber gasket and a rubber sealant compound. However, it required the clamping collar to be passed over the length of the candle which, although not damaging the filter itself, can cause some damage to any interesting filter cake formations when the filter was removed after an experiment had been completed. The filter is clamped horizontally, as mounting filter elements this way has some advantages, most importantly the use of the momentum of the inlet gas to enhance dust settling by forcing it downwards to the filter base.

2.3. Reverse flow experimental rig

The experimental rig is shown schematically in Figs. 4 and 5. The filter rig for reverse flow is shown together with the blower, the controller valve and measurement apparatus.

2.4. Average velocity and pressure difference measurements

Holes were drilled along the candle for insertion of pitot tube as illustrated in Fig. 6. Holes not in use were plugged. Air from the blower was injected into the open end of the
filter candle. Small steel pressure tappings with an outside diameter of 1.25 mm (o.d., \(\phi\)) and inside diameter of 1 mm (i.d., \(\phi\)) were utilised for the pressure difference and flow rate measurements. Measurements were taken using the procedure recommended by Perry and Green [16], i.e. at the centre of the filter and at three different radii. These radial positions were determined from the Eq. (1):

\[
\sqrt{\frac{2n - 1}{N}} = \frac{r_n}{R}
\]  

where \(N\) is the number of traverse points (for this case, \(N = 7\)), \(n\) the order of traverse radius (\(n = 1, 2, 3, \ldots, N/2\)), \(R\) the candle inner radius (20 mm) and \(r_n\) is traverse radius. The arithmetical mean of the velocities on each radius (exclusive of the centre) was taken as the average velocity for the cross-section. The assumption in this method is that flow is axisymmetric within the candle, so measurements were only taken on one point for each radius. In order to achieve a better accuracy, more measurements should be taken per radius, to get a mean velocity per radius as well [17]. Since the wall-roughness of the filter interior surface is estimated to be 1 mm [18], measurements close to the wall would not be expected to be very accurate.

For the determination of the volumetric flow rate and the velocity in the filter candle at a cross-section, Eqs. (2) and (3) were used:

\[
Q = \sqrt{\frac{2 \Delta P}{\rho A}}
\]  

\[
\frac{2 \Delta P}{\rho A} = \frac{2 \Delta P}{\rho A}
\]  

Fig. 3. (a) Dimensions of single candle rig, (b) candle mounted in single candle rig.

Fig. 6. Candle measurement positions.
where $\Delta P$ is the average of the dynamic pressure at the cross-section, determined by traversing, $\rho$ the density of air, and $A$ the cross-section area of the filter candle, $\pi D^2/4$.

$$\Delta P = \frac{P_1 + P_2 + P_3}{3}$$
with \( P_1 \), \( P_2 \) and \( P_3 \) as the local dynamic pressures at measurement point (traverse radius) of 1, 2 and 3.

Since the air flow is assumed to be in a straight parallel line with the filter candle, a stainless steel tapping tube can give satisfactory results for the measurements of the pressure difference, as long as it is properly aligned with the stream [16]. The tube has its opening and spacing as specified for a typical pitot tube. Static pressure tappings at different sections had been fixed. Pressure difference across the wall was measured at various points along the filter using the tapping tubes and micromanometer.

3. One-dimensional steady-state filter modelling

One-dimensional steady state models have been used to predict gas flow in a rigid ceramic filter. The models determine the axial pressure loss along the clean side of the filter element by accounting for variations in the gas momentum and the internal wall friction of the filter candle. The basis for the model used here was set out by Clift et al. [18]. A computer code, written in the computer programming language Fortran 90, for solving such a model has been developed and adapted to account for measured variations in the resistance to flow of the filter medium and geometry. Additionally, the adaptations enable it to be run in the reverse flow direction, i.e. when the axial gas flow is in the cleaning direction. Predictions for the pressure drop and flow distribution in the filtration and reverse flow directions for cylindrical candles of uniform geometry and conical or “tapered” filters will be presented. Experiments that have been conducted to determine the variation in the resistance to flow of filter candles along the length and to validate the model.

3.1. Description of the Model in the filtration direction

The model of Clift et al. [18] is based on the following key assumptions:

- The external gas pressure is constant.
- The local face velocity is proportional to the local pressure drop across the filter wall.
- The medium is of uniform resistance along the length of the filter candle.
- The axial pressure loss arises only from changes in momentum of the gas and the internal friction of the candle wall.
- No velocity head is recovered from the closed end of the filter.
- The gas is incompressible.

![Fig. 7. Schematic diagram of flow in the filtration direction.](image)

Fig. 7 shows schematically a single filter candle with flow in the filtration direction. The length of the “active” cylindrical section is \( L \). The volumetric gas flow along the core of the candle, \( Q \), will increase with \( z \) (distance from the end cup of the filter candle). Therefore, the pressure in the direction of the axial gas flow will decrease due to gas momentum changes arising from the “injection” of additional gas through the wall and the friction of the internal walls of the filter candle.

Clift et al. [18] considered the flow through an incremental section, such as that shown in Fig. 7.

3.1.1. Model assumptions

In analysing the overall pressure drop experienced by a gas passing through a filter candle, the following simplifying assumptions are made:

1. The gas pressure is constant at \( P_o \) everywhere on the external surface of the candle. This is valid when the velocities external to the candle are low. However, the model must be modified when this is not the case, for example when the candles form a closely spaced array.
2. The gas velocity through the candle at any section is proportional to \((P_o - P)\), where \( P \) is the local static pressure inside the candle. Seville et al. [1] have shown that this simple proportionality is valid.
3. The medium is uniform along the length of the candle, so that the gas face velocity varies as \((P_o - P)\) varies. This limits the analysis in its present form to clean candles, because the variation in face velocity will give rise to variations in the thickness of the deposited dust cake and hence to variations in flow resistance.
4. Pressure variations along the inside of the candle arise from the momentum change of the gas and from friction at the internal candle wall. The momentum contribution is evaluated, as in the analysis of Ushiki and Tien [19].
by a simple linear momentum balance, assuming that gas flowing through the walls has no axial momentum. The frictional contribution, which is neglected completely by Ushiki and Tien, is evaluated by treating the gas as a fully developed turbulent pipe flow. The friction factor is determined by direct measurement on gas flow along the candle with no flow through the walls. Because the gas Reynolds number is high and the walls are rough, the friction factor is constant. It is assumed implicitly that friction is unaffected by the flow entering through the walls. Given that the permeation flow will thicken the boundary layer at the walls, it may be anticipated that the friction factor should be lowered, but this is best investigated experimentally.

5. None of the "velocity head" of the gas issuing from the end of the candle in the filtration direction is recovered.

6. The gas is taken as incompressible, because pressure changes are normally negligible by comparison with the absolute pressure.

3.1.2. Derivation of the equations

Fig. 8 shows schematically an element of the candle, at distance \( z \) from the beginning of the cylindrical section. The local internal pressure is \( P_i \), so that the local gas flow entering through the differential length \( dz \) is

\[
dQ_j = \frac{(P_i - P_i) dz}{R_j} \tag{5}
\]

where \( R \) is the "flow resistance" which depends on the permeability of the medium, on the internal and external diameters of the candle, and on the gas viscosity \([1]\) and \( j \) denotes the position of candle (at the close end, \( j = 0 \); at the open end, \( j = z \)). In terms of the velocity through the external surface of the candle, \( U_o \),

\[
dQ_j = \pi D_{o,j}(U_o dz) \tag{6}
\]

so that

\[
R_j = \frac{P_i - P_i}{\pi D_{o,j}(U_o dz)} \tag{7}
\]

where \( D_o \) is the outside diameter of the candle.

The mean gas velocity, \( v \), inside the candle at this section is

\[
v_j = \frac{4Q_j}{\pi D_{o,j}^2} \tag{8}
\]

where \( D_i \) is the internal diameter of the candle. In terms of the Fanning friction factor, \( f \), the wall shear stress, \( \tau \), is

\[
\tau_j = \frac{\rho v_j^2}{2} \tag{9}
\]

where \( \rho \) is the gas density. Therefore, substituting for \( v \) using Eq. (8) gives the total shear force at the walls in the differential length \( dz \) as

\[
dF_j = \pi D_{o,j} \tau dz = \frac{\pi D_{o,j} \rho v_j^2 dz}{2} = \left( \frac{8 \rho Q_j^2 f}{\pi D_{o,j}} \right) dz \tag{10}
\]

The momentum flux entering the differential element is

\[
M_j = \rho Q_j v_j = \frac{4 \rho Q_j^2}{\pi D_{o,j}^2} \tag{11}
\]

so that the increase in momentum flux across the element is

\[
dM_j = \left( \frac{\pi D_{o,j}^2}{4} \right) \frac{dP_i}{dz} \frac{dF_j}{dz} + \frac{dM_j}{dz} = \frac{8 \rho Q_j^2 f dz}{\pi D_{o,j}^2} \tag{12}
\]

The linear momentum balance finally takes the form

\[
- \frac{\pi D_{o,j}^2}{4} \frac{dP_i}{dz} = \frac{dF_j}{dz} + \frac{dM_j}{dz}
\]

i.e.

\[
\frac{dP_i}{dz} = \pi D_{o,j}^2 \frac{dF_j}{dz} - \frac{8 \rho Q_j^2 f dz}{\pi D_{o,j}^2} \tag{13}
\]

Eq. (13) together with Eq. (14)

\[
\frac{dQ_j}{dz} = \frac{P_i - P_j}{R_j} \tag{14}
\]

which is rearranged from Eq. (5), constitute the model describing pressure variations along the candle. The pressure at \( z = L \), where \( L \) is the length of the candle, is taken as the pressure on the clean side of the candle following the assumption that there is no pressure recovery at the exit from the candle. Eq. (14) can be rewritten as (using Eq. (6)):

\[
R_j = \frac{P_i - P_j}{\pi D_{o,j}(U_o dz)} \tag{15}
\]

where \( R \) is the flow resistance of the candle walls in \( \text{N m}^{-1} \).
Table 4
Resistance of filters at open and closed ends

<table>
<thead>
<tr>
<th>Filter Type</th>
<th>Open end (N s/m^4)</th>
<th>Closed end (N s/m^4)</th>
<th>Mean resistance of five sectional filters (N s/m^4)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Cerel Cylindrical</td>
<td>22290</td>
<td>43120</td>
<td>31700</td>
</tr>
<tr>
<td>Tapered</td>
<td>14860</td>
<td>63900</td>
<td>39100</td>
</tr>
</tbody>
</table>

Of the terms on the right of Eq. (13), the former represents the pressure loss due to the momentum change and the latter represents pressure loss due to the internal wall friction.

The pressure variation along the candle is calculated by numerical integration of Eqs. (13) and (14). The overall pressure drop across the candle is calculated as \( (P_o - P(L)) \) where \( P(L) \) is the internal gas pressure at the open end of the candle assuming no “pressure recovery” in the issuing gas. This assumption is valid in the filtration direction when the total gas flow out of the filter is low compared with that during reverse flow.

A computer code was written in FORTRAN 90 for the numerical integration (using the Runge–Kutta method) of Eqs. (13) and (14). The program assumes that the hemispherical end of the filter is impermeable so that at \( z = 0 \), \( Q = 0 \). This assumption is valid for Cerel filter elements where it has been observed that the thickness of the end cap is more than twice that of the filter wall. In this case, the flow through the end cap will be much less than through the adjacent candle wall. Clift et al. [18], however, described the details of flow through the end cup of the filter candle in their work. For tapered filter, \( D_i \) at \( z = 0 \) (close end) and at \( z = 1.25 \) (open end) are set according to its diameters, respectively.

4. Results and discussion

4.1. Filtration simulation of cylindrical filters

Fig. 9 shows the results of numerical integration of Eqs. (13) and (14), calculated using constant values of friction factor. For a mean face velocity of 4 cm/s, the model underpredicted the pressure difference across the wall by approximately 17%. The theory of Clift et al. [18] also underpredicts the pressure difference across the filter wall in a 1.5 m filter candle for the lower mean face velocity (<8 cm/s). In the simulation of a long candle, the friction and momentum terms made the pressure difference across the wall a non-linear function of the total flow (or nominal mean face velocity).

Fig. 10 shows the calculated results of the variation in the local pressure drop over the wall as \( f \) is increased from 0.01 to 0.1 for a cylindrical filter, at face velocities of 4 and 5 cm/s. As \( f \) increases from 0.01 to 0.1, there is approximately a 12% increase in total pressure difference over the filter, from 278 to 311 Pa, at a face velocity of 4 cm/s. For a velocity of 5 cm/s, the total pressure difference over the filter has increased about 14%, by 360–410 Pa. This shows that, even though the friction factor increases from an order of magnitude, the effect on the total pressure difference is low, since volumetric flow, \( Q_1 \), in the filtration direction is low 0.008 m^3/s at 4 cm/s and 0.01 m^3/s at face velocity 5 cm/s.

\[
\frac{\pi^2 D_i^4}{32 \rho} \frac{dP}{dz} = -Q \frac{dQ}{dz} - fQ^2 D_i
\]
Eq. (13) indicates that the pressure within the candle decreases in the direction of axial gas flow, due to two effects: increase in gas momentum, and “friction” on the inside candle walls. Of the terms on the right of Eq. (13), the former term represents the momentum change and the latter term represents wall friction, which is ignored by Ushiki and Tien [19] analysis. The effects of both terms are plotted in Figs. 11 and 12. And $f$ is increased from 0.01 to 0.1 for a cylindrical filter at face velocities of 4 and 6 cm/s. As $f$ increases from 0.01 to 0.1, there is approximately a 12% increase in momentum term at a face velocity of 4 cm/s. For a velocity of 6 cm/s, the total pressure difference over the filter has increased about 16%. As $f$ increases from 0.01 to 0.1, frictional term is increased by the factor of 10 for both velocities, 4 and 6 cm/s.

Since $(dQ/dz)$ is assumed to be independent of $z$, Eq. (10) can be written as
\[ Q = \pi D_o U_0 z \]  
Eq. (13) can then simplify to
\[ \frac{dP}{dz} = -\frac{32 \rho U_0^2 D_o^2}{D_i^2} \left[ \frac{D_o}{D_i} \right]^2 \left[ 1 + \frac{f \rho U_0}{D_i} \right] z \]  
As in Eq. (13), the two terms in the last bracket of Eq. (18) represent momentum change and wall friction, respectively.
The ratio of friction to momentum change is then \( f/D \). The friction factor is typically of the order of \( 10^{-2} \). Therefore, momentum changes are the more significant contribution if \( z < 100D \).

Fig. 13 shows measurements made of the variation of overall pressure drop with nominal mean face velocity for two lengths of filter by Clift et al. [18]. The 1.5 m length filter candles showed a non-linear variation whilst the short candle showed a linear relationship between pressure drop and nominal face velocity, showing the strong effect of internal (axial) pressure loss on overall behaviour.

The Reynolds number for a filter candle with an inner diameter of 4 cm was in the range of \( 10^4 \) for the low mean face velocity (see Fig. 14). Therefore, the candle cavity cannot be considered as “fully rough” and the friction factor would in practice has been higher than the constant value used in the simulation [16,20]. Due to the limitation of the experimental rig, the controller cannot maintain the flow stability.
when the mean face velocity was over 6 cm/s. Therefore, in this work, experiment is not carried out with the face velocity higher than 6 cm/s. However, in industrial practice face velocities used are usually not greater than 0.05 m/s due to pressure drop limitations.

4.2. Reverse flow simulation of cylindrical filters

Fig. 15 shows the calculated results for the pressure difference distribution and compares them with the experiment data for reverse flow rates of 0.03 and 0.05 m$^3$/s. The model underpredicted the pressure difference in both cases. It seems that the actual observed friction factor is higher at the position closed to the open end of the filter than the value used in the calculation, $f = 0.01$. Higher friction will reduce the pressure difference across the wall. Fig. 16 shows a better agreement was noticed in the axial velocity profile.

4.3. Simulation of tapered filters

A 1D model was applied by Stephen [13] to simulate tapered filters. However, he did not carry out any comparison between the simulation results and experimental data. Due to the different geometry of tapered filters from the cylindrical filters, some modifications were made to the calculation. The resistance was no longer assumed to be a constant along the filter axis. Since the resistance to flow, $R$, as defined here is a function of the external diameter, the
The calculated pressure difference distribution agreed well with the experimental data for the three different face velocities, 3, 4 and 5 cm/s (Fig. 17). For the reverse flow simulation of the tapered filter, it can be seen from Figs. 18 and 19 that for approximately the first 0.6 m, the pressure difference distribution and axial velocity simulation agreed well with the experimental data. The internal wall friction only had significant effect for the first 0.6 m of the filter, whilst the axial velocity may be greatly influenced by the friction factor of the calculation as shown in Eq. (13).

The discrepancy between the model prediction and experimental results occurred at the open-end section of the filters. As the gas was pulsed from the tube a large quantity of pulse gas extends into the filter. Secondary entrainment gas enters from the throat of the filter (Fig. 20) in the opposite direction to the cleaning flow and will suck dusty gas through the round of throat of the filter. The model does not
Fig. 18. Comparison between the simulation results and the reverse flow experiment data of the pressure difference distribution in the tapered filters.

Fig. 19. Comparison between the simulation results and the reverse flow experiment data of the axial velocity profile in tapered filters with a reverse flow rate of 0.05 m³/s.

Fig. 20. Gas entrains from surrounding of the filter.
model the secondary entrainment and it is significant that the calculated results are different from the measured data at the open end of the filter. The similar observations are found in the studies of the cylindrical filters.

5. Conclusion

A simple one-dimensional model was quite adequate for predicting the pressure difference distribution along the filter axis. It gave reasonable agreement with the experimental data. However, the simulation in general underestimated the pressure difference distribution in filtration. This is because at low face velocities, the internal gas Reynolds number is relatively low ($Re \approx 10^4$). The filter candle internal should not therefore be considered as “fully rough” and the friction factor might be higher than that used here. The predicted results show good agreement with the experimental data for the tapered filter in the filtration direction. However, the friction factor term in the model has a strong influence in the reverse flow simulation. Current studies on ceramic filter have been carried on ambient condition. Further work should be done to study the operating conditions in high temperature.

Further reading


References