

Article

Effect of Pleat Angle on Pressure Drop in H14 HEPA Filters: A Mathematical Analysis with Corrections for Real Filter Behaviour

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Abstract

The influence of pleat angle on the pressure drop of H14 HEPA filters was investigated through a mathematical model that represents the filter as a system of converging–diverging channels coupled with porous filtration media. The analysis was conducted for pleat angles ranging from 1° to 20° under a constant laminar airflow rate of 0.167 m³/s and 0.45 m/s velocity. The model combines Darcy–Forchheimer flow through the filtration media with laminar channel flow theory, enabling the total pressure drop to be expressed as a function of pleat geometry and subsequently optimised through analytical differentiation. The results show that the pressure drop contribution of the filtration media increases with the pleat angle, from 10.57 Pa at 1° to 213.49 Pa at 20°, whereas channel losses decrease sharply from 1121.71 Pa to 2.75 Pa over the same interval. The competing behaviour of these two mechanisms generates a minimum total pressure drop of 94.56 Pa at a pleat angle of approximately 6°, compared with 120 Pa for the current industrial configuration operating at 3.73°. This represents a pressure drop reduction of approximately 21.2%, implying a corresponding decrease in fan energy consumption without compromising filtration performance. The analysis further demonstrates that very small pleat angles (1–2°) are highly unfavourable, producing total pressure drops between 301 and 1132 Pa due to severe channel constriction, while for angles above 13–14°, the channel contribution becomes negligible, and the overall pressure drop is governed almost entirely by the filtration media. These findings provide quantitative design criteria for optimising HEPA, EPA, and ULPA filter geometries, highlighting pleat angle as a critical parameter for improving aerodynamic performance, flow uniformity, and energy efficiency in high-purity environments. The proposed model was further assessed using a commercially available H14 HEPA filter with 188 pleats, an effective filtration area of 10.618 m², and a nominal airflow rate of 600 m³/h, demonstrating its applicability to real industrial filter configurations.



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Keywords: pleat angle; pressure drop; hydraulic diameter; MPPS; minipleat; wedge; HEPA

1. Introduction

Air quality is a fundamental determinant of human health, well-being, and life expectancy, as consistently evidenced by extensive epidemiological studies [1–3]. In increasingly polluted atmospheric environments, characterised by the presence of particulate

matter, biological contaminants, and nanoscale aerosols, air filtration systems play a crucial role not only in protecting public health but also in ensuring process control in critical industrial applications. Despite the considerable development of the sector, most advancements have predominantly focused on enhancing filtration media, improving efficiency, reducing pressure drop, and increasing particle holding capacity, while comparatively little attention has been devoted to the design and manufacturing parameters of pleated filters, particularly in minipleat configurations. The present study addresses the optimisation of H14 HEPA filters through the coupled analysis of three interdependent variables: filtration efficiency, flow uniformity, and pressure drop, all of which are intrinsically governed by pleat geometry. Conventional two-dimensional models, such as those based on Falkner–Skan formulations [4], exhibit limitations due to their inability to capture critical three-dimensional effects, notably flow distribution. This limitation motivates a geometry-focused approach to pleated HEPA filters (H14 or higher) [5], widely employed in high-purity environments, including the pharmaceutical and microelectronics industries, as well as containment applications, where the stability of laminar flow is essential. In such contexts, even minor velocity variations may compromise process integrity. From a fluid dynamic perspective, it is well established that low flow velocities favour laminar regimes [6,7], with typical operating conditions around 0.45 m/s. Nevertheless, pressure drop (ΔP) remains a key parameter, as it directly determines the energy consumption of the system. Accordingly, the increase in effective filtration area through pleating, combined with prefiltration stages [8–12], enables enhanced performance and extended service life in clean room applications [13]. Structural integrity is ensured by thermoplastic adhesive spacers (*hot-melt*), while pleat geometry governs outlet velocity uniformity [14–17], pressure drop, and behaviour at the most penetrating particle size (MPPS) [18,19]. Since the efficiency of absolute filters (EPA, HEPA, and ULPA), as defined by EN 1822-ISO 29463 [5], strongly depends on these factors, minimising velocity variations between pleats is of paramount importance.

As illustrated in Figure 1, for a fixed flow rate of 3400 m³/h, a reduction in effective filtration area leads to an increase in local velocity within the pleats. This, in turn, significantly elevates penetration at the most penetrating particle size (MPPS), reduces overall filtration efficiency, and shifts the critical particle size towards smaller diameters [9,20]. This behaviour clearly highlights the necessity of rigorous geometric optimisation in pleat design. Within this framework, the present work proposes a strategy aimed at improving flow distribution in minipleat filters through optimisation of the pleat angle while preserving structural integrity. Although the triangular (V-shaped) configuration remains the most widely adopted in industrial practice [21], it is important to note that, in reality, the pleat vertices are typically rounded due to manufacturing constraints.

The current state of the art may be broadly classified into three principal approaches: the development of alternative pleating geometries (such as U-shaped or sinusoidal profiles); the use of low-resistance filtration media, including nanofibers, glass–nanofiber composites, and ePTFE membranes; and the aerodynamic optimisation of the plenum through the use of diffusers [22–26]. In contrast, the present study focuses on optimising the existing geometry by determining the optimal pleat angle that minimises pressure drop while promoting a more uniform velocity distribution throughout the filter.

In this context, the work of Falkner–Skan [4] is of particular relevance, as it addresses parameters such as the inlet angle (β), external flow velocity ($U_{airflow}$), wedge velocity (U_{wedge} , equal to zero in the present case), and the boundary layer velocity profile $U(x)$, whose influence is limited under low-flow conditions, as well as the transverse velocity through the porous media (U_w). Although this latter parameter does affect the system, it does not constitute the primary focus of the present analysis. All the foregoing considera-

tions indicate that the present case closely resembles a Poiseuille flow model, characterised by laminar flow between two stationary plates, provided that the Reynolds number does not exceed the experimentally observed critical range, typically between 2500 and 3000. However, the typical operating conditions of this class of filters involve very low flow velocities, which inherently ensure a strongly laminar regime. It is well established that the Reynolds number is highly dependent on the hydraulic diameter D_h [6,7]. As shown in subsequent sections, this expression may be further simplified due to its strong dependence on the pleat angle tangent, a key parameter in the present study. The actual geometry of the wedge-shaped channels considered in this work is illustrated in Figure 2.

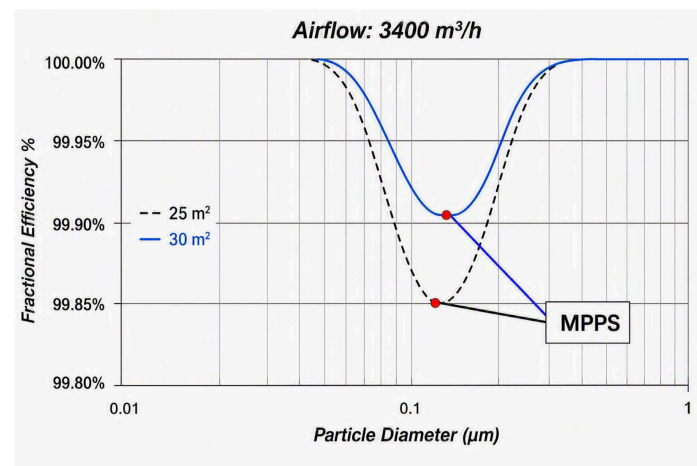


Figure 1. Fractional efficiency (%) (y-axis) of HEPA filter media as a function of particle diameter (μm) (x-axis) at two different velocities.

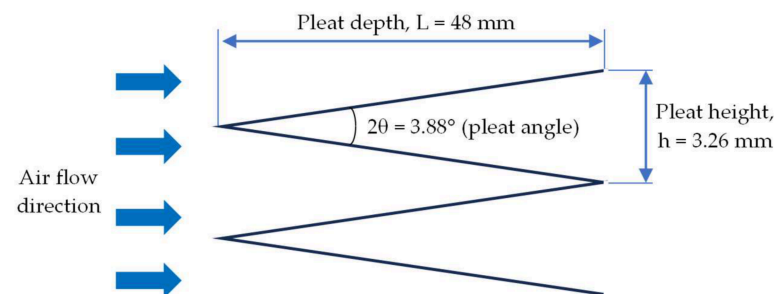


Figure 2. Actual H14 HEPA minipleat geometry values.

The geometric parameters considered correspond to an H14 HEPA filter with a nominal pleat angle of 3.88° ; a pleat depth of 48 mm; a 3.26 mm distance between pleats; and a pleat length of 588 mm perpendicular to the plane of analysis, which is assumed to be constant due to its negligible influence in the present study. From a functional perspective, the airflow fully traverses the filtration media from inlet to outlet, generating a characteristic pressure drop governed by the media thickness, permeability, and pleat geometry, in accordance with the fundamental operating principle of the filter. Flow field measurements [27] play a key role in understanding aerodynamic behaviour because they provide information on velocity distribution, flow non-uniformities, and recirculation phenomena that cannot be inferred from pressure drop measurements alone. In pleated filtration systems, the velocity field is particularly relevant for identifying the influence of pleat geometry on airflow redistribution and overall filter performance. Therefore, the combination of flow field measurements and analytical modelling provides a more comprehensive understanding of the aerodynamic mechanisms governing HEPA filter operation.

1.1. Flow in a Converging Channel: Sink Flow

The pleat geometry significantly restricts flow penetration at the inlet wedge vertex, where fibre deformation and breakage locally reduce the effective porosity, rendering this region nearly impermeable, as illustrated in Figure 3a. Consequently, the incoming flow that does not pass through the filtration media generates a pronounced vortex within the converging channel, as shown in Figure 3b.

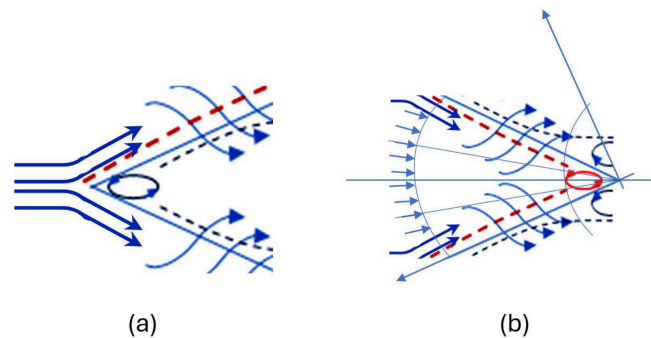


Figure 3. (a) Wedge apex detail with potential internal vortex formation; (b) converging channel with a partially blocked outlet and associated recirculation zone.

For this type of channel, local flow velocity may be expressed as follows:

$$v_{\infty}(x) = \sqrt{va} \frac{\eta}{x} = U \sqrt{\frac{v}{a}} \eta = U \frac{y}{x} \quad (1)$$

This velocity distribution $U(x)$ represents the radial flow field of an inviscid sink flow, characterised by straight streamlines converging towards a region where vortex formation may partially obstruct the channel. The formulation indicates that increasing the pleat depth reduces the local flow velocity, thereby enhancing the filtration area or allowing a slight increase in the pleat angle, ultimately improving pressure drop performance.

1.2. Flow in a Diverging Channel

The outlet flow profile, corresponding to the diverging channel, is even more critical to overall performance than the inlet, as it accumulates more pronounced adverse phenomena whose associated effects are described below.

1.2.1. Velocity Reduction and Pressure Increase

As in all diverging channels, the increase in cross-sectional area causes a reduction in flow velocity, which in turn leads to a rise in static pressure. This process introduces significant flow instability, typically distributed symmetrically about the central axis, although asymmetries may arise and further amplify instabilities at the outlet.

1.2.2. Boundary Layer Separation

This effect, driven by the adverse pressure gradient, may cause flow retardation near the channel walls. However, its influence is negligible under the low operating velocity considered in this study (0.45 m/s). Regarding diffuser design, several studies [6] have identified key criteria for optimal performance, foremost among which is maintaining an inlet divergence angle (2θ) between 5° and 15° . Below 5° , frictional losses increase due to an excessive channel length, whereas above 15° , flow separation may occur as a consequence of the aforementioned pressure rise. These findings are supported by experimental evidence indicating that a total divergence angle of approximately 10° provides optimal performance.

As illustrated in Figure 4, the minimum loss coefficient K occurs for any diameter ratio d_1/d_2 at angles close to 6° .

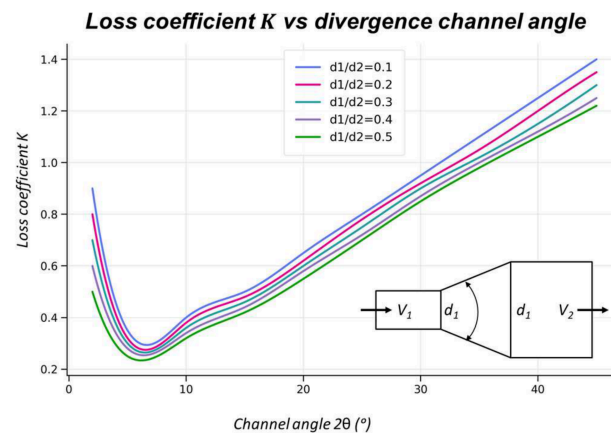


Figure 4. Loss coefficient K as a function of the total divergence angle (2θ) of the channel.

Another key parameter is the pressure recovery coefficient C_p , which may be defined as follows:

$$C_p = \frac{P_s - P_e}{P_{oe} - P_e} = 1 - \left(\frac{V_2}{V_1} \right)^2 = 1 - \left(\frac{A_1}{A_2} \right)^2 = 1 - AR^{-2} \quad (2)$$

This is achieved when C_p approaches unity (typically within a range of 0.90–0.98). Four fundamental flow regimes are identified, as illustrated in Figure 5, with the optimal operating region located below the line a–a. It should be noted that, even in very short diffusers, flow separation may occur if the total divergence angle $2\theta > 20^\circ$.

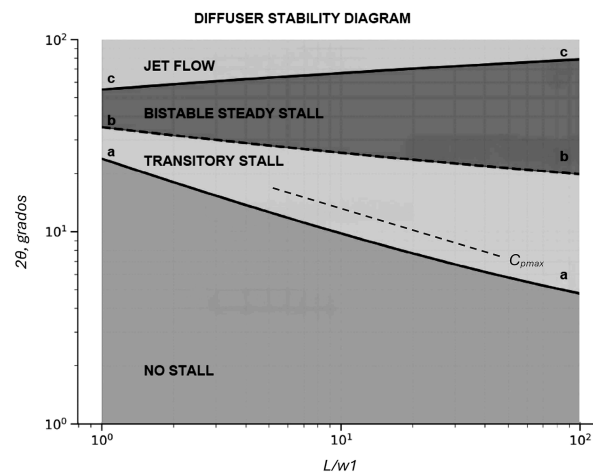


Figure 5. Stability diagram of a flat-walled diffuser.

Between lines a–a and b–b, boundary layer separation occurs, leading to transient stall and strongly unsteady flow. The highest performance (maximum C_p) is achieved within this regime, provided that operation remains close to the a–a limit. In the present case, a correction is required to accurately position the operating point within this framework.

1.2.3. Injection of Incoming Flow

The injection of air through the filtration media suggests behaviour analogous to boundary layer disturbance. The pressure drop increases whenever the outlet operates under negative pressure, as this condition promotes flow penetration from the inlet towards the outlet, a common and unfavourable scenario in negative-pressure cleanroom environments, where HEPA-filtered air is discharged. This additional pressure drop arises from an

increase in friction $f''(0)$. Nevertheless, it should be emphasised that, under these conditions, a well-defined boundary layer structure does not develop on the surface for injection ($f_0 < 0$), for an impermeable wall ($f_0 = 0$), or even in cases of very weak suction. This behaviour suggests that the boundary layer thickness may be inversely related to filtration efficiency, since increasing efficiency results in reduced suction due to the corresponding rise in pressure drop.

2. Materials and Methods

This section systematically presents the technical principles and experimental considerations underpinning the analysis performed. First, the geometric configuration of the filter is described, highlighting the significance of the pleat angle. This is followed by a mathematical assessment of the variation in L/w_1 with respect to 2θ , together with the development of a coupled theoretical model and the derivation of the pressure drop equation as a function of pleat angle, enabling the determination of the optimal value and associated wedge geometry effects. Subsequently, the procedure employed to calculate the minimum pressure drop and its corresponding angle is outlined. Finally, twenty geometric configurations are defined, designed to systematically evaluate the influence of pleat angle on the flow distribution and overall pressure drop within the system. To validate the theoretical model against experimental data, a commercially available H14 HEPA filter widely used in the industry for over two decades was employed, with its main specifications summarised in Table 1.

Table 1. Characteristics of the selected H14 HEPA filter.

Total airflow	0.167 m ³ /s
Frontal speed	0.450 m/s
Number of pleats	188
Frontal filtration surface	0.372 m ²
Effective filtration surface	10.618 m ²
Airflow/pleat	8.8883×10^{-4} m ³ /s
Pleat width	0.588 m
Pleat height	0.0031277 m
Pleat depth	0.0480000 m
Pleat hypotenuse	0.0480254 m
Actual pleat angle	3.73°

The actual pleat angle corresponding to the analysed H14 HEPA filter is highlighted in red, as optimisation of this parameter constitutes the primary objective of the present study. The effective filtration surface, once areas obstructed by adhesive and non-porous materials are excluded, is distributed across an array of 188 pleats. The pleat geometry transforms each channel into a miniature nozzle, as previously discussed, inducing two simultaneous effects: a marked reduction in flow velocity due to the increase in the cross-sectional area and a slight local acceleration resulting from the converging shape of the pleat. The balance between these effects defines an average filtration velocity of 0.0176809 m/s. Experimental measurements to characterise flow through the porous media were conducted by recording outlet velocities at multiple positions along the x- and z-axes, as well as at varying distances along the y axis, thereby generating a detailed flow distribution map.

2.1. Experimental Setup and Measurement Procedure

The experimental investigation was conducted using a full-scale H14 HEPA filter installed in a controlled airflow test bench designed to reproduce typical cleanroom operating conditions. The filter was operated at its nominal airflow rate of 600 m³/h (0.167 m³/s),

corresponding to a frontal velocity of approximately 0.45 m/s. The airflow was generated by a variable-speed centrifugal fan and conditioned through a sequence of prefilters, media efficiency filters, and flow-straightening diffusion plates to ensure a uniform and stable inlet velocity profile before reaching the test filter. The test bench layout is shown in Figure 6.

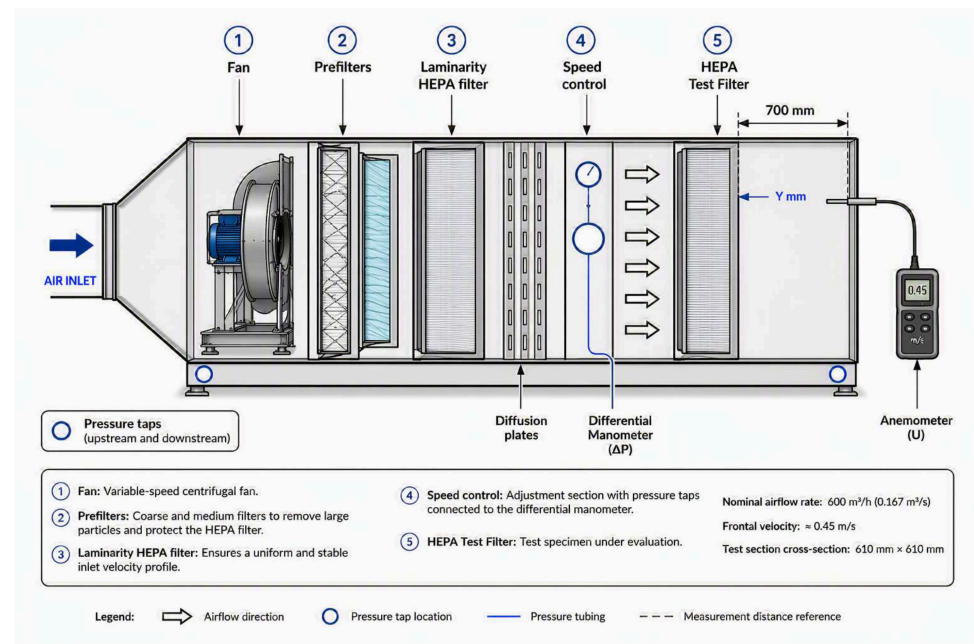


Figure 6. Schematic representation of the experimental setup.

2.2. Instrumentation

Air velocity measurements were performed using a calibrated hot-wire thermo-anemometer with a measurement range of 0–20 m/s, an accuracy of $\pm (3\% \text{ of reading} + 0.03 \text{ m/s})$, and a resolution of 0.01 m/s. The instrument was selected because of its suitability for low-velocity laminar airflow measurements typically encountered downstream of HEPA filters. The pressure drop across the filter was measured using a digital differential pressure manometer with a measurement range of 0–1000 Pa, an accuracy of $\pm 1\%$ of full scale, and a resolution of 1 Pa. The pressure taps were positioned upstream and downstream of the filter according to conventional cleanroom filter testing practices [5]. Ambient temperature and relative humidity were monitored throughout the experiments using a calibrated thermo-hygrometer with accuracies of $\pm 0.5^\circ\text{C}$ and $\pm 3\% \text{ RH}$, respectively, to verify stable environmental conditions during testing. Prior to testing, all instruments were calibrated according to the manufacturers' recommendations and verified against reference standards.

2.3. Measurement Locations

A three-dimensional representation of the experimental assembly is shown in Figure 7, with the origin located at (0,0,0). Outlet velocity measurements were taken at three positions along the y-axis (5, 100, and 200 mm), at the filter centre ($x = 305 \text{ mm}$), and across the entire z-axis, for several values of h , as detailed in the subsequent section.

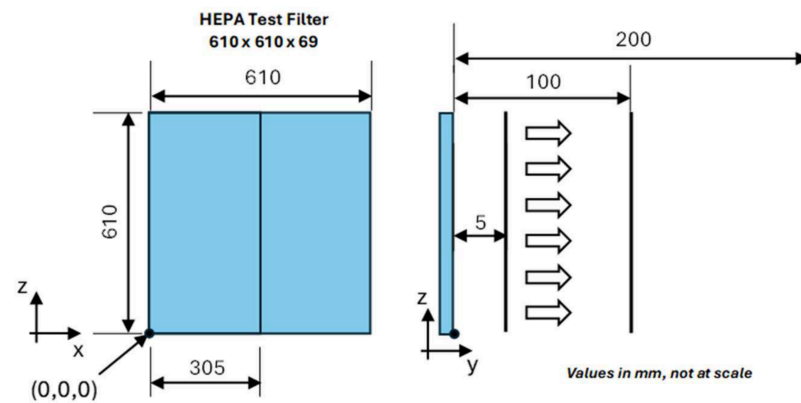


Figure 7. Measurement grid.

2.4. Experimental Procedure

The experimental methodology followed the sequence described below:

- Test filter was installed and checked for correct sealing within the test housing to ensure that no leakage was present.
- The fan speed was adjusted until a nominal airflow rate of $600 \text{ m}^3/\text{h}$ was achieved.
- The full system was operated for approximately 15 min to ensure steady-state conditions.
- Upstream and downstream static pressures were recorded continuously and used to determine the pressure drop across the filter.
- Velocity measurements were performed at all predefined locations within the measurement grid.
- Four independent measurement series were completed for each operating condition.
- To minimise the influence of random fluctuations, the highest and lowest values were discarded, and the remaining two measurements were averaged.
- The resulting values were used to construct velocity distribution profiles and were subsequently compared with those of the theoretical model developed in this work.

2.5. Uncertainty Analysis

The uncertainty associated with the measured pressure drop was estimated to be below $\pm 2\%$ under nominal operating conditions. For velocity measurements, the combined uncertainty resulting from instrument accuracy, positioning error, and flow fluctuations was estimated to remain below $\pm 5\%$. To reduce experimental variability, all tests were performed under stable environmental conditions, identical airflow settings, and using the same filter specimen throughout the experimental campaign. The repeatability of the measurements was verified by comparing the four independent experimental series, which showed deviations below 3% for pressure drop and below 5% for local velocity measurements. A summary of the measurement uncertainties is presented in Table 2.

Subsequently, and given the strong relationship between the hydraulic diameter (D_h), friction (f), and pressure drop (ΔP), nine additional scenarios were analysed to evaluate the influence of variations in D_h on pressure drop, with the aim of identifying the optimal configuration that minimises both frictional losses and total pressure drop. This variation in D_h was generated by varying the pleat angle through changes in the spacing parameter h . From the ten scenarios investigated, five were selected, as presented in Table 3, for the purposes of validating the model and examining the influence of pleat angle on pressure drop across the filtration media, within the channel, and on the overall pressure loss.

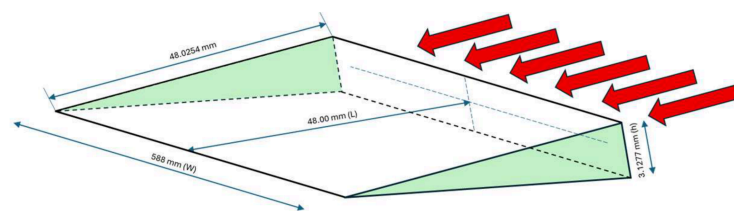
Table 2. Measurement equipment and uncertainty.

Instrument	Parameter	Range	Resolution	Accuracy
Hot-wire anemometer	Velocity	0–20 m/s	0.01 m/s	$\pm(3\% \text{ reading} + 0.03 \text{ m/s})$
Differential pressure manometer	ΔP	0–1000 Pa	1 Pa	$\pm 1\% \text{ FS}$
Thermo-hygrometer	T	−10 to 60 °C	0.1 °C	$\pm 0.5 \text{ °C}$
Thermo-hygrometer	RH	0–100% RH	0.1% RH	$\pm 3\% \text{ RH}$

Table 3. Input variables. Scenario 1: baseline (real) model.

Scenario	w (m)	h (m)	L (m)	U_c (m/s)
1	0.588	0.00313	0.0480	0.01768
3	0.588	0.00256	0.0480	0.01445
8	0.588	0.00368	0.0480	0.02683
9	0.588	0.00313	0.0480	0.02284
10	0.588	0.00256	0.0480	0.01867

Figure 8 illustrates the fundamental geometrical parameters of an individual pleat for improved conceptual understanding and visualisation. Here, w denotes the pleat width, which is assumed to be constant; h represents the pleat height, which varies with the pleat angle; and L corresponds to the pleat length or depth, which is considered to be constant, as its practical variation while maintaining a fixed total surface area is negligible. Finally, U_c denotes the inlet velocity into the pleat, which arises from the variation in the effective filtration surface defined by w , h , and L . Scenario 1 corresponds to the baseline configuration, representative of the selected filter model.

**Figure 8.** 3D Individual pleat schematic.

2.6. Use of AI Tools

Artificial intelligence (AI) tool Microsoft 365 Copilot (Microsoft Corporation, Redmond, WA, USA; accessed 4 June 2026), a generative artificial intelligence platform based on OpenAI GPT-5 family models was used to assist in generating the graphical abstract based on author-defined prompts. All AI-generated images were reviewed, verified, and, where necessary, modified by the authors, who take full responsibility for the final content.

3. Results

This section presents an analysis of the results obtained, with the aim of characterising the flow behaviour within the channel and benchmarking it against established theoretical models. The discussion enables a physical interpretation of the observed phenomena and serves to validate the methodology employed while identifying the principal deviations

from idealised models. Such discrepancies are primarily attributed to geometric effects, cross-sectional variations, and incomplete flow development. Furthermore, the influence of these factors on frictional losses and on the applicability of conventional friction coefficients is examined.

3.1. Mathematical Evaluation of Pressure Drop as a Function of Pleat Angle

The following section presents the mathematical framework developed to determine the total pressure drop, together with its individual components, namely, the pressure drop across the filtration media and within the channel, repeating the analysis for each defined scenario.

3.1.1. Boundary Conditions

- Inlet velocity within the channel (pleat): $U_c = 0.0176809$ m/s.
- Porous media thickness: $\delta = 2 \times 10^{-5}$ m.
- Permeability: $k = 1 \times 10^{-11}$ m².
- Air properties at 20 °C: $\rho = 1.204$ kg/m³, $\mu = 1.81 \times 10^{-5}$ Pa.s.
- Ambient pressure upstream of the porous media: $P_{\text{external}} = 0$.
- Closed lateral boundaries: the flow exits exclusively through the upper and lower surfaces along the wedge, with no discharge through the apex, except for negligible residual leakage.
- The pleat depth and its hypotenuse are assumed to be equal for simplification purposes, as their actual difference is negligible (0.0254 mm).

The boundary conditions adopted in this study were derived from the operating specifications of the commercial H14 HEPA filter selected as the reference case, including a nominal airflow rate of 0.167 m³/s, a frontal velocity of 0.45 m/s, pleat dimensions, and filtration media properties. These values were chosen because they represent realistic industrial operating conditions commonly encountered in cleanroom and contamination control applications. Nevertheless, the purpose of the model extends beyond reproducing the behaviour of the existing filter configuration. By keeping all operating parameters constant and modifying only the pleat angle, the analysis isolates the influence of geometry on pressure drop and flow distribution. Consequently, the imposed boundary conditions should be interpreted both as representative of the current filter operation and as a reference framework for evaluating potential future designs under identical filtration requirements. This approach enables the determination of an optimum pleat angle while ensuring that comparisons are not affected by variations in airflow rate, media characteristics, or external operating conditions. Furthermore, the selected boundary conditions correspond to EN 1822-ISO 29463 [5] operating conditions for H14 HEPA filters and therefore provide a realistic design basis for assessing future geometrical improvements without altering filtration efficiency, filter class, or nominal airflow capacity.

For the reference geometry, the characteristic Reynolds number is extremely low, more than two orders of magnitude below the onset of transition in internal flows. Therefore, turbulent effects can be safely neglected, and no turbulence intensity specification is required. The simplified governing equation adopted as a starting point for the mathematical analysis corresponds to the total pressure drop and its constituent components:

$$\Delta P_{\text{total}} = \Delta P_{\text{media}} + \Delta P_{\text{channel}} \quad (3)$$

Media Pressure Drop

$$\Delta P_{\text{media}} = \mu \frac{L_m}{k} U_m + \rho C_f U^2 \quad (4)$$

where:

C_f : inertial coefficient.

Channel Pressure Drop (due to geometry)

$$\Delta P_{channel} = f \frac{L}{D_h} \frac{\rho U^2}{2} \quad (5)$$

and

$$D_h = \frac{4A}{P} \quad (6)$$

So, the total pressure drop is

$$\Delta P_{total} = \mu \frac{L_m}{K} U + \rho C_f U^2 + f \frac{PL}{4A} \frac{\rho U^2}{2} \quad (7)$$

Pleat Geometry

The pleat angle (θ) defines the number of pleats, the packing density, and the channel width between adjacent pleats.

$$h = 2L \tan(\theta) \quad (8)$$

$$\theta = \arctan\left(\frac{h}{2L}\right) \quad (9)$$

The effect of pleat angle variation on the inlet frontal velocity at the cell is described by the following expressions, assuming a constant total flow rate:

$$U_c = \frac{Q}{A_{total \ channels}} \quad (10)$$

$$U_c = \frac{A_f \cdot U_f}{A_{total \ channels}} \quad (11)$$

So,

$$A_{total \ channels} \uparrow \Rightarrow U_c \downarrow \Rightarrow \Delta P \downarrow \quad (12)$$

Within the regime of very small pleat angles (hereafter referred to as the low-angle regime), the channel width decreases significantly, resulting in a high pleat density and a corresponding increase in the effective filtration area. Under these conditions, the Reynolds number is very low due to the reduced velocity through each pleat, resulting in a strongly viscous flow regime [28]. This promotes significant interaction with the channel walls and, consequently, a high pressure drop. The associated effects include elevated shear stress and a strong likelihood of partial blockage due to the limited channel width. For intermediate pleat angles (hereafter referred to as the nominal optimal range), a balance is established between the channel width and pleat density, enabling a more uniform flow distribution. The velocity gradient is reduced compared with that in the previous case, and the boundary layer develops in a stable manner (although it should be noted that, at such low velocities, its influence is negligible). The pressure drop is correspondingly reduced, resulting in lower frictional losses and a more effective utilisation of the filtration media.

In contrast, for larger pleat angles (referred to here as the high-angle regime), the resulting channels exhibit significant opening, leading to a reduction in the total filtration area. Local velocities increase accordingly, causing a substantial increase in the pressure drop. The associated consequences include higher resistance within the filtration media and the potential for non-uniform flow distribution.

3.1.2. Filtration Media Model (Darcy–Forchheimer Law) [29–31]

The pressure drop across the filtration media is first recalled, prior to analysing the inlet velocity into the pleat, which is subsequently distributed along the porous filtration media:

$$\Delta P_{media} = \mu \frac{L_m}{K} U_m \quad (13)$$

$$U_m = \frac{Q}{A_{effective}} \quad (14)$$

with the effective surface filtration area being

$$A_{effective} = N \cdot 2L \cdot W_f \quad (15)$$

The pleat number (N) depends on the angle:

$$N = \frac{L_f}{2L \tan(\theta)} \quad (16)$$

by substituting this into Equation (15), we obtain

$$A_{effective} = \frac{L_f}{\tan(\theta)} \cdot W_f \quad (17)$$

so, the medium speed is

$$U_m = \frac{Q}{A_{effective}} = \frac{Q \tan(\theta)}{L_f W_f} \quad (18)$$

by substituting this into Equation (13), we obtain

$$\Delta P_{media}(\theta) = \mu \frac{L_m}{K} \cdot \frac{Q \tan(\theta)}{L_f W_f} = A \tan(\theta) \quad (19)$$

3.1.3. Pleat Channel Model

The flow between adjacent pleats behaves as a rectangular channel, in which the hydraulic diameter is equivalent to the height of the pleat:

$$Dh = \frac{4A}{P} = \frac{4(wh)}{2(w+h)} \approx \frac{2wh}{w} = 2h = 2(2L \tan(\theta)) = 4L \tan(\theta) \quad (20)$$

This means that

$$Dh \propto \tan(\theta) \quad (21)$$

The pressure drop through the channel (laminar flow) is

$$\Delta P_{channel} = f \frac{L}{Dh} \frac{\rho U_c^2}{2} \quad (22)$$

We define the friction factor and Reynolds number as

$$f = \frac{64}{Re}; Re = \frac{\rho U_c Dh}{\mu} \quad (23)$$

So, by replacing the f and Re values in $\Delta P_{channel}$, we obtain

$$\Delta P_{channel} = \frac{32\mu L U_c}{Dh^2} \quad (24)$$

We also know that

$$U_c = \frac{U_f}{N} \quad (25)$$

in such a way that

$$U_c = \frac{U_f}{\frac{L_f}{2L \tan(\theta)}} \quad (26)$$

But let us keep in mind that our channel is not flat but convergent/divergent instead, so consider an infinitesimal control volume located inside one pleat channel. The local channel height is

$$h(x) = 2x \tan(\theta) \quad (27)$$

where $x = 0$ corresponds to the apex and $x = L$ to the pleat opening. The local channel cross-sectional area becomes

$$A(x) = W_f h(x) \quad (28)$$

or, equivalently,

$$A(x) = 2W_f x \tan(\theta) \quad (29)$$

Because airflow continuously penetrates through media, the flow rate inside the channel decreases progressively along the flow direction. If U_w denotes the filtration velocity through one porous wall, the conservation of mass gives

$$\frac{dQ(x)}{dx} = -2W_f U_w \quad (30)$$

integration over the channel length yields

$$Q(x) = Q_0 - 2W_f U_w x \quad (31)$$

where Q_0 is the flow rate entering the channel at the pleat opening. The local axial velocity is

$$U(x) = \frac{Q(x)}{A(x)} \quad (32)$$

substituting the previous expression gives

$$U(x) = \frac{Q_0 - 2W_f U_w x}{2W_f x \tan(\theta)} \quad (33)$$

which may be rearranged as (factoring $\tan(\theta)$)

$$U(x) = \frac{1}{\tan(\theta)} \left[\frac{Q_0}{2W_f x} - U_w \right] \quad (34)$$

consequently,

$$U(x) \propto \frac{1}{\tan(\theta)} \quad (35)$$

for every position inside the channel. This is the key result because it demonstrates that narrowing the pleat angle increases the axial velocity everywhere inside the channel due to the reduction in the available flow area. Having established this result, the pressure gradient for a laminar flow according to the local Darcy–Weisbach relation may be written as

$$\frac{dP}{dx} = f \frac{\rho U(x)^2}{2D_h(x)} \quad (36)$$

where the local hydraulic diameter is

$$D_h(x) = 4x \tan(\theta) \quad (37)$$

by substituting this into the pressure gradient equation, we obtain

$$\frac{dP}{dx} = f \frac{\rho}{8} \frac{U(x)^2}{x \tan(\theta)} \quad (38)$$

introducing

$$U(x) = \frac{G(x)}{\tan(\theta)} \quad (39)$$

where

$$G(x) = \left(\frac{Q_0}{2W_f x} - U_w \right) \quad (40)$$

gives

$$U(x)^2 = \frac{G(x)^2}{\tan^2(\theta)} \quad (41)$$

and therefore

$$\frac{dP}{dx} = f \frac{\rho}{8} \frac{G(x)^2}{x} \tan^{-3}(\theta) \quad (42)$$

which can be rewritten as

$$\frac{dP}{dx} = H(x) \tan^{-3}(\theta) \quad (43)$$

where

$$H(x) = f \frac{\rho G(x)^2}{8x} \quad (44)$$

Function $H(x)$ contains all terms independent of the pleat angle and can therefore be treated as a geometric constant for a given filter configuration. Integration between the channel entrance and the pleat apex yields the total channel pressure drop contribution:

$$\Delta P_{channel} = \int_0^L H(x) \tan^{-3}(\theta) dx \quad (45)$$

Since $\tan(\theta)$ is independent of x , it can be removed from the integral:

$$\Delta P_{channel} = \tan^{-3}(\theta) \int_0^L H(x) dx \quad (46)$$

defining

$$B = \int_0^L H(x) dx \quad (47)$$

all geometric and flow-related constants are grouped into a single coefficient B , yielding

$$\Delta P_{channel} = \frac{B}{\tan^3(\theta)} \quad (48)$$

This expression shows that channel losses decrease rapidly as the pleat angle increases. Physically, this behaviour results from the simultaneous increase in the hydraulic diameter and the reduction in the characteristic channel velocity, both of which contribute to lower frictional losses within the pleat channel.

3.1.4. Total Pressure Drop Model

By adding both pressure loss contributions, we obtain

$$\Delta P(\theta) = A \cdot \tan(\theta) + \frac{B}{\tan^3(\theta)} \quad (49)$$

where

$$A = \mu \frac{L_m}{K} \frac{Q}{W_f L_f} \quad (50)$$

and

$$B = \int_0^L H(x) dx = \int_0^L \frac{f \rho G(x)^2}{8x} dx = \int_0^L \frac{f \rho}{8x} \left(\frac{Q_0}{2w_f x} - U_w \right)^2 dx$$

both cases, A and B , are managed by geometric constants.

3.1.5. Angle Optimisation [32–34]

We find the minimum by differentiating Equation (49) and setting it equal to zero:

$$\frac{d(\Delta P)}{d\theta} = 0 \quad (51)$$

$$\frac{d}{d\theta} \left(A \tan(\theta) + B \tan^{-3}(\theta) \right) = A \sec^2(\theta) - 3B \tan^{-4}(\theta) \sec^2(\theta) \quad (52)$$

So,

$$\frac{A}{B} = \frac{3}{\tan^4(\theta)} \quad (53)$$

Thus, we obtain the optimal angle:

$$\theta = \arctan \left(\sqrt[4]{\frac{3B}{A}} \right) \quad (54)$$

3.1.6. Model Assumptions and Limitations

The mathematical model developed in this work is based on several simplifying assumptions that allow for the influence of pleat angle on pressure drop to be isolated and analysed analytically.

- The airflow is assumed to be steady, incompressible, and fully laminar throughout both the porous media and the pleat channels. This assumption is justified by the very low operating velocities and Reynolds numbers characteristic of H14 HEPA filters under standard operating conditions.
- The physical properties of the air, including density and viscosity, are assumed to be constant and correspond to standard atmospheric conditions (20 °C). Thermal effects, humidity variations, and compressibility effects are therefore neglected.
- The filtration medium is represented as a homogeneous porous material with constant permeability and uniform thickness across the entire filter surface. Local variations in fibre packing density, manufacturing imperfections, and material anisotropy are not explicitly considered.
- Pleat geometry is idealised using a regular and repetitive converging–diverging channel configuration. Although a correction factor is introduced to account for the partial blockage of the pleat apex observed in real filters, additional geometric irregularities arising from manufacturing tolerances are not included in the formulation.
- The model assumes a uniform flow distribution between pleats and does not explicitly resolve three-dimensional secondary flows, local recirculation structures, or small-scale

velocity fluctuations. These effects are incorporated indirectly through the correction factors included in the final pressure drop formulation.

Furthermore, the analysis corresponds exclusively to clean filter conditions. Dust loading, particle accumulation, clogging mechanisms, and permeability degradation during filter service life are outside the scope of the present study. Similarly, structural effects such as pleat deformation, media deflection, and the so-called accordion effect are not included in the current model. The objective of the model is not to reproduce every local flow phenomenon occurring inside a pleated HEPA filter but rather to identify the dominant physical mechanisms governing pressure drop and to establish quantitative design criteria for pleat angle optimisation.

3.1.7. Model Uncertainty and Sensitivity Analysis

The uncertainty analysis presented in Section 2.5 addresses experimental measurements. However, the analytical model itself also contains uncertainties associated with the estimation of its governing parameters. The most relevant sources of model uncertainty are the permeability of the filtration media (K), the effective pleat length (L), the airflow rate (Q), the air properties (μ and ρ), and the geometric correction factors introduced to represent real filter behaviour.

The optimum pleat angle is determined from Equation (54):

$$\theta_{opt} = \arctan\left(\sqrt[4]{\frac{3B}{A}}\right) \quad (55)$$

which shows that the optimum angle depends on the B/A ratio through a fourth-root relationship. Consequently, the prediction of the optimum angle is significantly less sensitive to parameter variations than the pressure drop prediction itself. Using a first-order uncertainty propagation approach, the relative uncertainty can be approximated as

$$\frac{\delta\theta_{opt}}{\theta_{opt}} \approx \frac{1}{4} \frac{\delta(B/A)}{(B/A)} \quad (56)$$

Thus, a $\pm 10\%$ uncertainty in the B/A ratio produces only an approximate $\pm 2.5\%$ uncertainty in the predicted optimum angle. Considering conservative engineering uncertainties in the dominant model parameters, the optimum angle of 6° is found to vary by approximately $\pm 0.3^\circ$, resulting in an estimated optimum interval between 5.7° and 6.3° . This relatively narrow interval demonstrates the robustness of the analytical optimisation procedure and confirms that the identified optimum remains essentially unchanged within the expected uncertainty range of the model. A sensitivity assessment further indicates that permeability and the geometric correction coefficient have the greatest influence on pressure drop estimation, whereas the influence of air density and viscosity is comparatively small under the low-Reynolds-number operating conditions considered, due to strong laminar flow.

3.2. Physical Interpretation of the Results

Based on the obtained expression, it can be observed that the optimal pleat angle depends on the ratio between channel losses and media losses, following a fourth-root relationship of this ratio.

$$\theta_{opt} \propto \left(\frac{\text{Channel pressure drop}}{\text{Media pressure drop}}\right)^{1/4} \quad (57)$$

Thus, if the filtration media is highly restrictive, that is, if its permeability is extremely low, which is equivalent to saying that the filter efficiency is very high (H14, U15, U16, or even U17), the total filtration area must be maximised. Consequently, the optimal pleat angle should be small or very small. So, when the permeability is higher and the overall filter geometry becomes the dominant factor, a larger pleat angle is recommended, as is typically the case for minipleat filters with efficiencies ranging from ePM1 50% to ePM1 75%, according to ISO 16890 [9].

3.3. Real H14 (acc. EN1822-ISO 29463 [5]) Filter Values

Based on the analysis presented thus far, the derived formulation is applied to the actual values of the filter under study, listed in Table 4. This enables the determination of the final expressions for the pressure within both the porous media and the channel as functions of the angle value.

Table 4. Typical values for the HEPA H14 test filter.

Parameter	Symbol	Value
Frontal velocity	U_f	0.45 m/s
Filter frontal width	W_f	0.610 m
Filter frontal height	H_f	0.610 m
Filter frame thickness	P_f	0.069 m
Flow rate	Q	0.167 m ³ /s
Flow rate per pleat	Q_p	8.8883×10^{-4} m ³ /s
Media thickness	L_m	2×10^{-5} m
Permeability	K	1×10^{-11} m ²
Air viscosity	μ	1.8×10^{-5} Pa·s
Air density	ρ	1.2 kg·m ⁻³
Pleat height	h	0.0031277 m
Pleat width	w	0.588 m
Pleat depth	L	0.048 m
Pleat angle	θ	3.73°

By applying the previously discussed equations, the following results are obtained:

$$A_{ef-pleat} = \frac{h w}{\tan(\theta)} = \frac{(0.0031277)(0.588)}{\tan(\theta)} = 0.00183 \quad (58)$$

$$U_{m-pleat} = \frac{Q_{pleat}}{A_{ef-pleat}} = \frac{8.8883 \times 10^{-4} \tan(\theta)}{0.00183} = 0.4857 \tan(\theta) \quad (59)$$

Thus, the pressure drop across the filtration media is given by

$$\mu \frac{L_m}{K} = 1.8 \times 10^{-5} \cdot \frac{2 \times 10^{-5}}{1 \times 10^{-11}} = 36 \quad (60)$$

$$\Delta P_{media} = 36 \cdot U_{media-pleat} \quad (61)$$

by substituting this, we obtain

$$\Delta P_{media} = 36 \cdot (0.4857 \tan(\theta)) = 17.48 \cdot \tan(\theta) \quad (62)$$

So, consequently, from (53),

$$\Delta P_{channel} = \frac{4.4 \times 10^{-5}}{\tan^3(\theta)} \quad (63)$$

Both pressures drop contributions across the filtration media and within the channel are evaluated on a per-pleat basis; consequently, the total value must be obtained by multiplying by the overall number of pleats, N . However, the resulting relationship is not strictly linear, as the airflow distribution is not uniform across all channels due to several effects:

Inlet/outlet effect (of particular significance) (f_{io})

This affects the pressure drop value, which may be expressed as follows:

$$\Delta P_{io} = (K_c + K_e) \frac{\rho V^2}{2}; f_{io} = \frac{1}{1 + K_c + K_e} \quad (64)$$

Flow non-uniformity (f_{nu})

Not all channels operate under identical conditions; both the flow rate and, consequently, the velocity may vary significantly between them. Since pressure drops are strongly dependent on the local velocity, this causes effective degradation of the theoretical flow utilisation.

$$CV_V = \frac{\sigma_V}{\bar{V}}; f_{nu} = \frac{1}{1 + CV_V} \quad (65)$$

Converging flow (wedge effect) (f_w)

Although relatively small in magnitude, this effect is inherently difficult to avoid, arising from the acceleration induced as the channel cross-section progressively decreases along the pleat, leading to local flow acceleration.

$$a = V \frac{dV}{dx}; f_w = \frac{1}{1 + \beta_w} \quad (66)$$

Three-dimensional losses and geometric deviations (f_{3D})

Finally, and of no lesser importance, geometric deviations resulting from variations in the volume of individual pleats introduce additional three-dimensional losses.

$$f_{3D} = \frac{A_{eff,3D}}{A_{ideal}} \quad (67)$$

All these factors were experimentally evaluated on the test filter, whose actual pressure drop was known to be 120 Pa. This allowed us to determine the differentiating factor (f_{cp}) in both the media and the channel, which directly affect the filter but to different degrees, with a three times greater pressure drop across the media than through the channel. Experiments were conducted with different angle filters but with the same media, and the results were similar. Based on experimental calibration against the reference H14 filter, a value of $f_{cp} = 0.412$ was obtained. This coefficient represents the cumulative influence of the above four independent effects not explicitly resolved by the analytical model. The correction coefficient is expressed as

$$f_{cp} = f_{io} \cdot f_{nu} \cdot f_w \cdot f_{3D} \quad (68)$$

where:

$$f_{io} = 0.75.$$

$$f_{nu} = 0.80.$$

$$f_w = 0.90.$$

$$f_{3D} = 0.76.$$

leading to

$$f_{cp} = 0.75 \times 0.80 \times 0.90 \times 0.76 \approx 0.412$$

A key aspect of the model is that the four correction mechanisms primarily affect the air penetration across the filtration media rather than the frictional losses inside the channel [35]. The media pressure drop model relies on ideal assumptions regarding:

- Effective filtration area;
- Uniform flow distribution;
- Local filtration velocity.

All four correction mechanisms directly modify these assumptions. Consequently, the complete correction factor is applied to the media contribution:

$$\Delta P_{real\ media} = f_{cp} \Delta P_{ideal\ media} \quad (69)$$

In contrast, the channel model already includes the dominant flow physics through:

- Hydraulic diameter variation;
- Reynolds number dependence;
- Darcy–Weisbach friction;
- Converging–diverging channel geometry.

Therefore, only a residual fraction of the global correction is required for the channel pressure drop. Experimental calibration showed that approximately one-third of the global correction should be assigned to the channel term:

$$\Delta P_{real\ channel} = \left[\frac{(f_{cp})}{3} \right] \Delta P_{ideal\ channel} \quad (70)$$

This weighting reflects the physical fact that inlet/outlet effects, flow maldistribution, effective area reduction, and three-dimensional deviations primarily affect the porous media, whereas the channel pressure drop model already captures most of the relevant flow physics and therefore requires only a modest residual correction.

3.4. Complete Final Model

$$\Delta P(\theta) = 17.48 \tan(\theta) N f_{cp} + \frac{4.4 \times 10^{-5} N \frac{f_{cp}}{3}}{\tan^3(\theta)} \quad (71)$$

A range of pleat angle values between 1° and 20° were introduced to evaluate the total pressure drop based on the preceding equation. The corresponding results are presented in Table 5.

The analysis indicates that the optimal pleat angle is approximately 6°, corresponding to a minimum pressure drop of 94.56 Pa. The most clearly unfavourable region lies between 1° and 2°, where the pressure drop reaches exceedingly high values, exceeding 300 Pa. Such narrow angles are therefore impractical, as they induce severe flow constriction; extremely high friction; and, from a practical standpoint, effective channel blockage. By contrast, the range between 3° and 5° may be regarded as a transition region, in which the pressure drop decreases significantly, from approximately 156 Pa to 98 Pa. This behaviour is fully consistent with the real case analysed herein, where the actual pleat angle is 3.77°, corresponding to a pressure drop of 120 Pa, thus providing a reasonable balance in system performance. Finally, the optimal region is identified between 6° and 7°, where the lowest pressure drops are achieved, approaching 95 Pa in both cases. It is noteworthy, however, that, although the pressure drop across the filtration media increases compared with that

at lower angles, the geometrical contribution to the pressure loss is substantially reduced, resulting in an overall optimisation of system performance.

$$\Delta P_{channel} \downarrow \text{ y } \Delta P_{media} \uparrow \quad (72)$$

$$\frac{d}{d\theta} \Delta P_{channel} \gg \frac{d}{d\theta} \Delta P_{media} \quad (73)$$

Table 5. Pressure drop for pleat angle values from 1° to 20°.

2θ	tan(θ)	ΔP Media (Pa)	ΔP Channel (Pa)	ΔP Total (Pa)
1	0.0087	10.57	1121.71	1132.27
2	0.0175	21.13	280.38	301.52
3	0.0262	31.70	124.58	156.29
4	0.0349	42.28	70.05	112.33
5	0.0437	52.86	44.81	97.68
6	0.0524	63.45	31.10	94.56
7	0.0612	74.05	22.84	96.89
8	0.0699	84.66	17.47	102.13
9	0.0787	95.29	13.79	109.08
10	0.0875	105.93	11.16	117.09
11	0.0963	116.58	9.21	125.80
12	0.1051	127.26	7.73	134.99
13	0.1139	137.95	6.58	144.53
14	0.1228	148.66	5.67	154.33
15	0.1317	159.40	4.93	164.33
16	0.1405	170.16	4.33	174.49
17	0.1495	180.95	3.82	184.77
18	0.1584	191.76	3.41	195.17
19	0.1673	202.61	3.05	205.66
20	0.1763	213.49	2.75	216.24

Finally, the high-angle regime, between 8° and 10°, shows the opposite behaviour. While the pressure drop across the filtration media continues to increase, the geometrical contribution from the channel is no longer able to compensate through a comparable or greater reduction. As a result, the total pressure drop rises once again. Beyond this range, from 10° to 20°, the same trend persists, albeit in a more pronounced manner.

$$\Delta P_{media} \propto \tan(\theta) \quad (74)$$

4. Discussion

The present results indicate that the pleat angle constitutes a governing geometric parameter controlling the total pressure drop [35,36], as it modulates the relative contributions of the porous media and channel flow in an opposing manner. As the pleat angle increases, the hydraulic diameter of the channel increases, leading to a reduction in viscous losses while simultaneously decreasing the effective filtration area, which enhances the resistance of the porous media. The results indicate a potential energy-saving improvement of over 21.2% in H14 HEPA filters (in accordance with EN 1822-ISO 29463 [5]), achieved through optimisation of the pleat angle. Although a strictly linear relationship between the total pressure drop and pleat angle is not observed, a quasi-linear behaviour is identified within the optimal range of 5–10°, where the pressure drop scales in proportion to approximately 1635·tan(θ). The mathematical optimisation yields an optimal pleat angle of approximately 6.00°, nearly double the current value. This behaviour is consistent with the analysis

presented in Section 1.2.2 (“Boundary Layer Separation”), where a minimum in the loss coefficient K was identified as a function of the total divergence angle (2θ), reflecting the underlying balance between viscous losses and flow separation effects.

The derived pressure drop equation enables the determination of the optimal pleat angle based on the specific characteristics of each filter, including geometrical parameters (height, width, and depth), as well as filtration efficiency, which is primarily governed by the permeability of the media. Additional operating parameters, such as flow rate and velocity, must also be considered, particularly in cases where the inlet air velocity deviates from the standard value of 0.45 m/s typically associated with H14 HEPA filter applications. Figure 9 presents the values reported in Table 5, clearly illustrating the existence of a minimum in the total pressure drop, resulting from the combined contributions of the media and channel pressure drop, in the vicinity of 7° . It should be noted that there is an almost linear relationship in the interval between 3.73° and 11° , which indicates a strong relationship between the pressure drop and pleat angle, as shown in Figure 10.

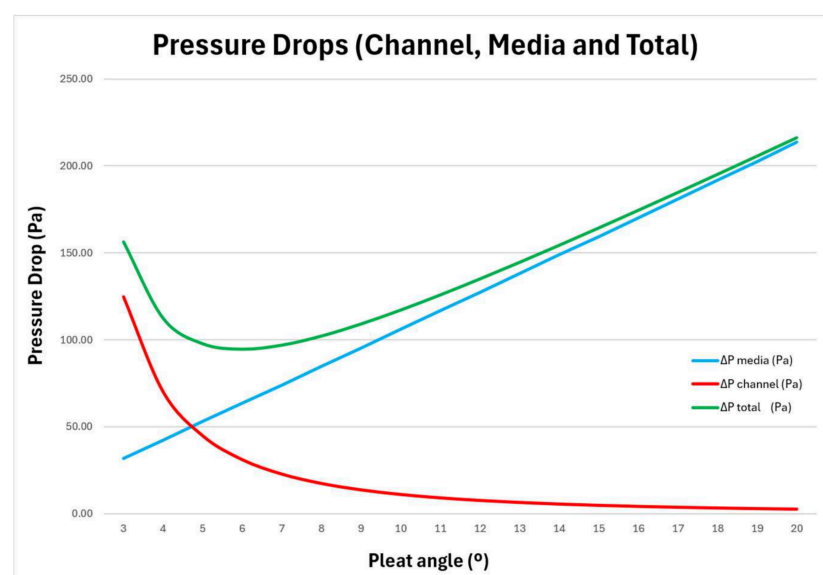


Figure 9. Pressure drop curves: across media, through the channel, and the total (results for 1° and 2° are not shown for greater clarity, and they are not significant).

2θ (°)	ΔP (Pa)	Relación
3	156.29	51.83%
4	112.33	71.88%
5	97.68	86.95%
6	94.56	96.81%
7	96.89	102.47%
8	102.13	105.41%
9	109.08	106.80%
10	117.09	107.34%
11	125.80	107.44%

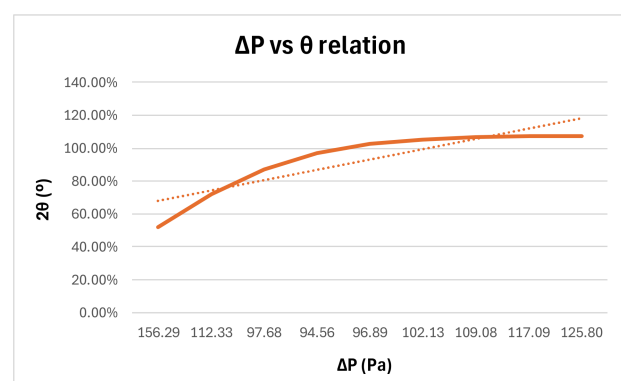


Figure 10. Relationship between pressure drop for different pleat angles and the corresponding percentage variation.

For pleat angles above $13\text{--}14^\circ$, the total pressure drop becomes effectively equal to the pressure drop across the filtration media, as the channel contribution becomes negligible. This observation carries significant practical implications, indicating that, for a channel

pleat height (h) equal to or greater than 22.16 mm, corresponding to approximately 26 pleats in a standard 592 mm $\times$ 592 mm filter, the overall pressure drop is governed solely by the filtration media. The optimal angle is highlighted in green in Table 5, with a calculated total pressure drop of 94.56 Pa, markedly lower than that associated with the actual pleat angle (3.73°), which yields 120 Pa. The scientific novelty of this work is the derivation of a closed-form analytical optimisation model that links the pleat angle, hydraulic diameter, filtration area, and total pressure drop in HEPA filters, allowing the optimal pleat angle to be calculated directly without CFD simulations or numerical optimisation procedures. Although the analytical model predicts a minimum pressure drop at a pleat angle of approximately 6° , this optimum has not yet been experimentally verified through the manufacture and testing of dedicated filter prototypes. Such validation would require a controlled industrial campaign involving multiple HEPA filter geometries manufactured with identical filtration media, pleat depths, frame dimensions, adhesive patterns, and operating conditions. Due to the complexity and costs associated with producing dedicated prototype series, this experimental verification is beyond the scope of the present work. Therefore, the predicted optimum angle should be interpreted as an analytical design recommendation derived from the governing flow equations rather than a fully validated industrial optimum.

5. Conclusions

This study demonstrates that pleat angle is the dominant geometric parameter governing the aerodynamic performance of H14 HEPA filters, as it directly controls channel width, hydraulic diameter, packing density, and airflow redistribution. Using a mathematical model that couples Darcy–Forchheimer flow through the filtration media with laminar flow in converging–diverging pleat channels, the total pressure drop was evaluated for pleat angles between 1° and 20° . The results reveal a strong trade-off between pressure losses in the filtration media and those generated by the pleat channels. As the pleat angle increased from 1° to 20° , the pressure drop across the filtration media increased from 10.57 Pa to 213.49 Pa, whereas channel losses decreased dramatically from 1121.71 Pa to 2.75 Pa. The interaction of these opposing mechanisms produced a minimum total pressure drop of 94.56 Pa at a pleat angle of approximately 6° . For the industrial H14 filter analysed in this work, the current pleat angle of 3.73° results in a pressure drop of approximately 120 Pa. Increasing the angle to the optimal value of 6° reduces the total pressure drop to 94.56 Pa, corresponding to an improvement of approximately 21.2%. This reduction can translate directly into lower fan energy requirements while maintaining the filtration efficiency and structural configuration of the filter.

Very small pleat angles (1 – 2°) were found to be highly inefficient, generating total pressure drops between 301 Pa and 1132 Pa because of severe channel constriction and high viscous losses. In contrast, angles above approximately 13 – 14° reduced channel losses to less than 6 Pa, causing the overall pressure drop to become almost entirely governed by the filtration media. The analysis identifies an optimal operating range between 6° and 7° , where the total pressure drop remains below 100 Pa and the balance between filtration area and channel resistance is maximised. These findings provide quantitative design criteria for the optimisation of HEPA, EPA, and ULPA filters and demonstrate that pleat angle optimisation represents an effective strategy for simultaneously improving energy efficiency, flow uniformity, and overall aerodynamic performance in high-purity applications. Finally, future work should focus on coupling the proposed aerodynamic model with structural analyses of pleat deformation and stability, particularly to assess the limits imposed by the accordion effect at larger pleat angles and to determine the optimum compromise between aerodynamic efficiency and mechanical robustness.

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Abbreviations

The following abbreviations are used in this manuscript:

ePTFE	Polytetrafluoroethylene membrane
HEPA	High Efficiency Particulate Air filter
ULPA	Ultra Low Particulate Air filter
MPPS	Most Penetrating Particle Size

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