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REVIEW OF TURNING VANES, CURVED DIFFUSERS AND THEIR ROLE IN COMPACT WIND AND WATER TUNNELS

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Abstract*: Turning vanes and curved diffusers are an integral part of many modern wind and water tunnels. They facilitate the turning of flow around tight bends, increasing flow uniformity while reducing energy losses and turbulence. Ample literature exists discussing these flow conditioners, but the information is scattered. This necessitates a novel overview of the developments since the first closed circuit wind tunnel was constructed in 1909. Particular attention is paid to modern expansion vanes, which allow for the construction of more compact research facilities. A quantitative meta-analysis of pressure loss coefficients will illustrate the real-world benefits of using specific turning vanes. The best performance is achieved by the vanes developed at KTH and MIT for standard and expanding corners, respectively.*

Keywords*: turning vanes, curved diffusers, turbulence, flow uniformity, pressure drop*

1. INTRODUCTION

Ever since Prandtl constructed the first closed circuit wind tunnel in 1909, engineers have been trying to optimise the flow around 90° corners [1]. The aim is to reduce pressure losses while increasing flow uniformity. This is usually achieved through the installation of turning vane cascades. In recent years, research has pivoted to investigating expanding corners. By simultaneously turning the flow and increasing channel width, more compact wind and water tunnels can be constructed. This can be achieved through the use of curved diffusers or expansion vanes. As the information on these flow conditioners is scattered, a novel overview is presented herein. A quantitative comparison between common turning vanes is made, before discussing the workings of standard diffusers, curved diffusers and expansion vanes.

2. STANDARD TURNING VANES

Of course, not all research facilities have the same requirements and therefore, there is not a one-size-fits-all vane. Flat-plate vanes offer a significant cost advantage over profiled vanes. Asymmetric profiled vanes minimise turbulence and maximise flow uniformity, thereby providing the best experimental environment. However, if bi-directional flow is required, symmetrical vanes might be more suitable, as they deliver the same performance in both directions. An overview of the different types of turning vanes can be seen in Fig. 1. The origin and performance of each vane type will be discussed in the subsequent sections.

2.1 ¼ Circle vanes

The most common type of turning vane is manufactured from a thin sheet of material that is subjected to a 90° bend with a large radius. This type of vane offers high manufacturability at a low cost. However, standard ¼ circle vanes also exhibit relatively high energy losses and only deliver a deflection angle of 81.4° in lieu of the desired 90°. The flow at the outlet is therefore not parallel to the flow channel, leading to an uneven velocity distribution. To achieve perpendicular deflections, vanes can be over-bent to 101°. Unfortunately, this increases energy losses even further [2].

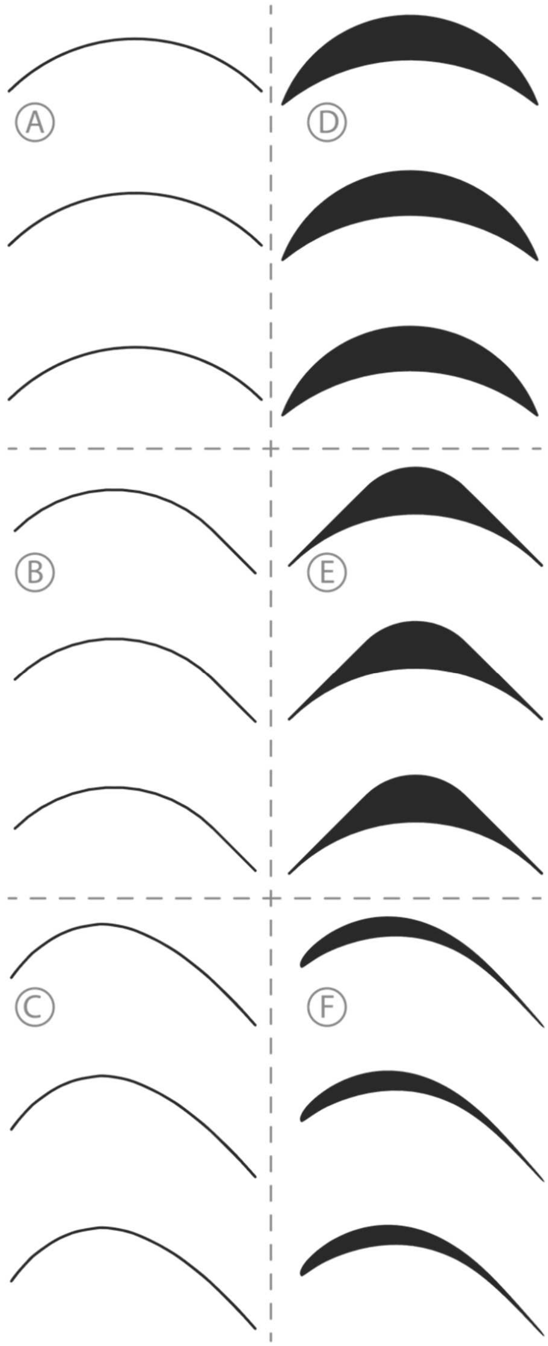

Fig. 1. Flat-plate (left) and profiled (right) vanes. These will be referred to as ¼ circle vanes (A), ¼ circle vanes with an extended trailing edge (B), Kröber vanes (C), crescent vanes (D), constant channel vanes (E) and KTH (Royal Institute of Stockholm) vanes (F). All vanes shown share a common chord length.

2.2 ¼ Circle vanes with longer trailing edges

An alternative to overbending the vanes is the extension of the trailing edge. According to de Vega et al., a straight extension measuring 30% of the chord length of the ¼ circle is optimal. Below this value, the flow is not turned 90° and longer extensions merely lead to additional energy losses [2]. A slightly different geometry was used for the optimisation of a wind tunnel at the University of Loughborough. Johl et al. tested vanes with an extension of 48%, but no justification for this particular parameter choice is given. However, it is shown that decreasing the gap-to-chord ratio results in more even flow downstream of the cascade, but also leads to higher energy losses [3].

2.3 Vanes derived from flat aerofoils

In 1925, Birnbaum derived a flat-plate aerofoil where lift and pitching moment can easily be estimated theoretically from the angle of attack [4]. Betz published a method of turning standard aerofoils into cascade vanes using conformal mapping in 1931 [5]. The following year, Kröber used the findings of these two papers to optimise flat-plate vane cascades. The loss factor was estimated to be approximately 0.13 [6]. It should be noted that these vanes are highly asymmetric, rendering them unsuitable for environments where high-quality flow is required in both directions.

2.4 Crescent vanes

In all ducts that undergo sharp 90° bends, the diagonal width across this bend is wider than the inlet and outlet channel. Flow is therefore forced to expand and contract, encouraging flow separation and energy losses. Flat-plate vanes do not solve this issue, since they too, cause a local change in channel width. This problem can be addressed by installing crescent-shaped vanes instead. These profiled vanes consist of two arcs with different radii, resembling a familiar moon-like crescent shape. In theory, this geometry can lessen the amount of channel width change and thus reduce energy losses. The optimisation of a crescent vane is the central theme of a publication by de Vega et al. A circular arc and a parabola are used respectively as the basis of the pressure and suction side of the vane. Using the MISES program, the researchers optimised the leading-edge radius and vane thickness. The rough basic shape is somewhat maintained throughout the process to ensure high manufacturability [2].

2.5 Constant channel vanes

Just like crescent vanes, constant channel vanes are based on the idea of maintaining a constant channel width throughout the turn. The pressure side of these vanes is formed by one large ¼ circle, while the suction side consists of a concentric smaller arc and a straight section on either side of it. This shape was first considered by Collar in 1936, who used it as a basis for developing an asymmetric aerofoil [7]. A symmetric version of the constant channel vane was again mentioned by Eck in 1966 [8]. It seems that for most wind tunnels, reversible flow is not a design requirement. Therefore, symmetric vanes are rarely employed, as much better performance can be achieved with an asymmetric design.

2.6 Vanes derived from profiled aerofoils

The vanes designed by Prandtl for the first recirculating wind tunnel were, in fact, profiled aerofoils with a relatively low loss factor of 0.110 [9]. With the increasing sophistication of computers and programs, new optimisation techniques have since become available. Computational algorithms, known as “panel methods”, can calculate the lift distribution across an aerofoil at any given angle of attack, providing a myriad of alternatives to the Birnbaum aerofoil. Sahlin et al. have derived a vane-geometry based on the Wortmann FX60-100 aerofoil. This particular profile had been chosen for its high lift-to-drag ratio. A vorticity distribution was used, instead of conformal transformations, to transfer the characteristic of the aerofoil to the equivalent vane. The result is an asymmetric profiled vane with a very low loss factor of 0.036 [10].

2.7 Channel dividing walls

Dividing walls are a simple alternative to turning vanes. While vanes are a repeated geometric entity, each dividing wall has a different shape. In pipe bends, the ratio between the channel width and the central turning radius has a strong impact on energy efficiency. A wider channel and/or a smaller turning radius will result in increased energy losses. Channel dividing walls can be used to decrease said ratio and increase efficiency. Care should be taken that the individual ratios of each resulting channel are the same across the bend, as shown in Fig. 2. Kröber notes that this design leads to higher losses than standard turning vane cascades, due to the increased surface area and resulting skin friction drag [6].

Fig. 2. Bend with seven dividing walls

2.8 Comparing turning vane performance

One of the primary performance indicators of any flow conditioner is the loss factor K, a dimensionless measure of how much pressure energy is lost. Here Δp is the change in total pressure, ρ is the fluid density and u is the flow velocity:

$$K = \frac{2\Delta p}{\rho u^2} \tag{1}$$

The loss factors of the various turning vanes described herein are summarised in Table 1:

Table 1. K-factors of various turning vanes

Vane Name	Vane Type	*K*
Dividing walls	not applicable	0.220 [6]
Standard ¼ circle	flat-plate	0.200 [6]
Extended ¼ circle	flat-plate	0.138 [9]
Optimised flat-plate	flat-plate	0.130 [6]
Constant channel	aerofoil	0.150 [10]
Early aerofoils	aerofoil	0.110 [9]
Crescent vanes	aerofoil	0.090 [2]
Johansson aerofoil	aerofoil	0.036 [10]

3. STRAIGHT DIFFUSERS

At their core, diffusers are simply an expansion in the flow channel. With this expansion, the flow velocity is decreased and some dynamic pressure (the kinetic energy per unit volume) is converted to static pressure. If the dynamic pressure in the boundary layer is lower than the static pressure rise, flow separation is inevitable. However, if diffusers are designed correctly, they allow for an energy efficient expansion of flow, which makes them an integral part of many flow systems. The most important attributes of diffusers are the expansion angle (θ) and the normalised diffuser size, which is the ratio of inlet channel width (D_1) to channel length (L). Care should be taken when evaluating the expansion angle. Conventionally, this is the angle from the centre line to one side of the diffuser, though some publications will use twice that value instead.

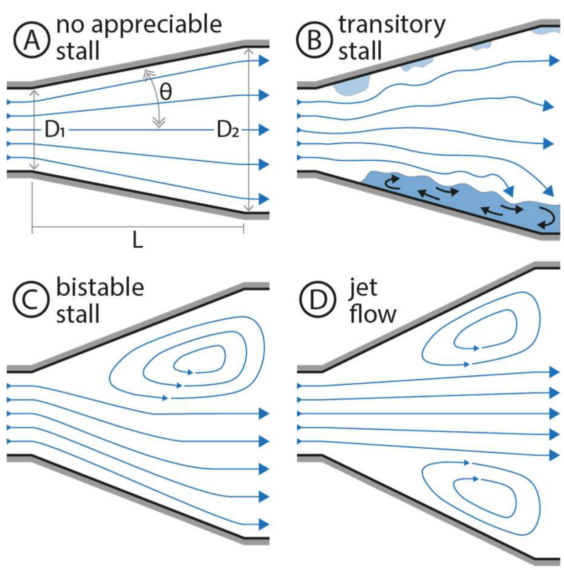

Fig. 3. Diffusers with increasing (from A to D) angle θ and resulting flow regimes. Adapted from [11, 12].

Of the four scenarios shown in Fig. 3, no appreciable stall (A) produces the most undisturbed flow. In steady bi-stable stall (C) the flow attaches to one side and will remain attached to that side unless physically prevented. If external intervention forces the flow to the opposite side, it will then attach itself to that side (hence the term "bi-stable"). In jet flow (D), the expansion is too great for the flow to attach to either side. Instead, it leaves the narrow section as a jet, which brings about significant energy losses. Highly unsteady flow is generated if a diffuser exhibits unsteady transitory stall (B). Within this flow regime, the diffuser will periodically go through a stall-wash-out cycle as shown in Fig. 4. Initially small local areas of stall build up (B1). On one side these localised areas grow into a larger stall (B2). This larger stall will keep growing towards the diffuser inlet (B3). Eventually, the stalled area is swept out (B4) and the cycle starts anew. Large fluctuations in pressure recovery can be encountered over a single cycle and the time taken to complete a cycle is known as the stall period.

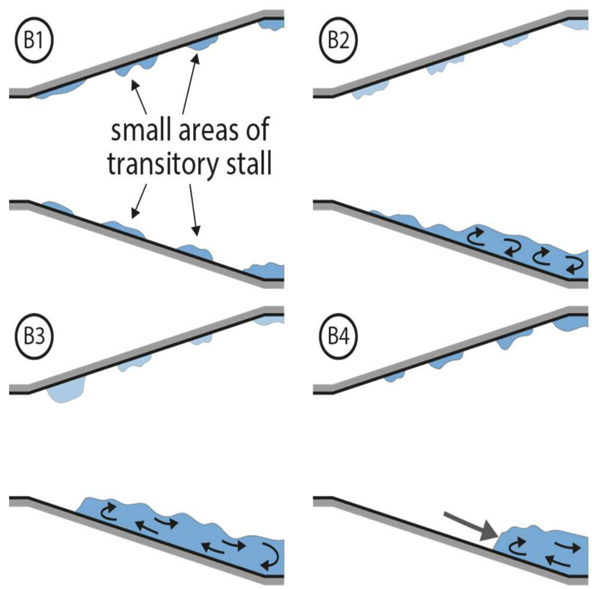

Fig. 4. Diffuser in various stages in the stall-wash-out cycle within transitory stall. Adapted from [13].

4. CURVED DIFFUSERS

As with their straight-walled counterparts, curved diffusers are usually intended to expand flow evenly and efficiently. Just as with expansion angle, an increase in turning angle can increase flow separation (Fig. 5). Fox et al. were the first to conduct extensive tests with a large variety of curved diffusers and discovered that for turning angles of 30° or less, straight-walled and curved diffusers behave almost identically. It was also shown that curved diffusers encounter the same flow regimes as found in straight diffusers. They investigated the lower three common stall regions (no appreciable stall, transitory stall, and fully developed stall), while the jet flow region was not investigated [14].

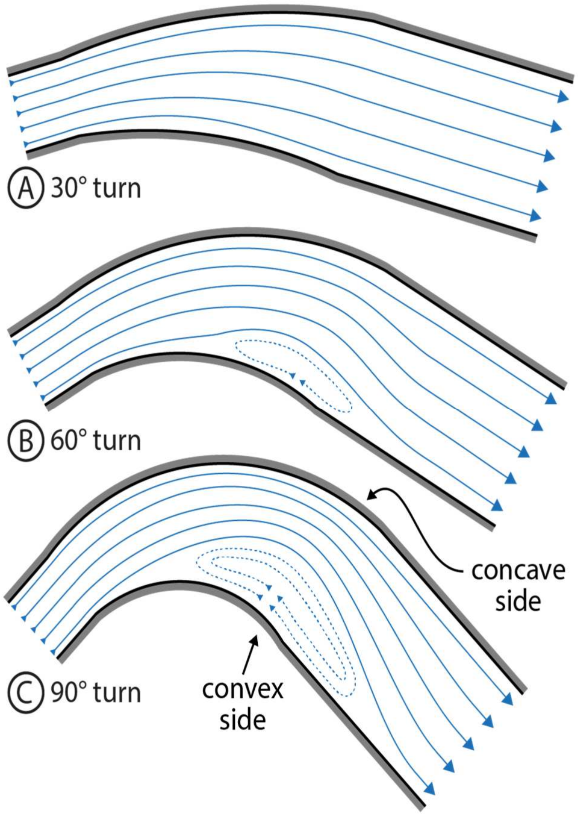

Fig. 5. Stall regime in curved diffusers. With an increasing expansion angle, the flow separation moves upstream and increases in magnitude. Adapted from [15].

In straight diffusers, the flow in a fully developed stall can attach itself to either side of the diffuser. In curved diffusers, however, the flow attaches itself to the concave pressure side. Furthermore, Fox and Kline highlight the necessity to artificially force the formation of a turbulent boundary layer in experiments. This is achieved by placing a wire upstream of the diffuser, which is commonly referred to as "tripping" the turbulent boundary layer [14]. In their 1983 publication, Sullery et al. confirm that increased inlet turbulence is another factor that helps to improve the performance of curved diffusers through an increase in pressure recovery [16].

Another important geometric property of curved diffusers is their aspect ratio, which is found by dividing the diffuser inlet height by the inlet width (D_1). Note that many diffusers are quasi two-dimensional, meaning that the height remains constant while the width is determined by the expansion. As expansion vanes are installed in a manner that divides channel width, without changing channel height, they tend to have large aspect ratios and relatively low expansion ratios. For this reason, a 1998 publication by Majumdar et al. is of particular interest to this investigation. The researchers studied a curved diffuser with a relatively high aspect ratio of 6, and a low expansion ratio of 2 [17]. The velocity profiles were mapped at seven discrete locations along the diffuser as shown in see Fig. 6.

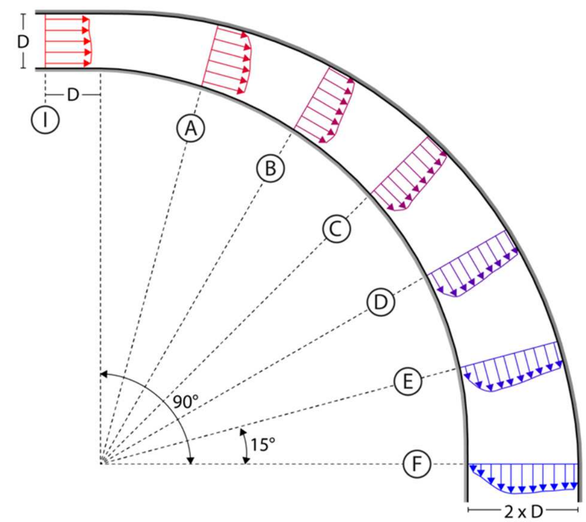

Fig. 6. Curved diffuser geometry and flow profiles based on real velocity data presented by Majumdar et al. [17]

The low expansion angle and expansion ratio cause very little separation despite the large 90° turning angle. This is in agreement with the findings of Fox and Kline [14], which are shown in Fig. 7. in the following section. Majumdar et al. report that flow separation is confined to the convex wall at the very end of the diffuser. They state that the main flow is accelerating on the convex side. Inversely, on the concave side, centrifugal forces lead to an increase in static pressure and a reduced flow velocity. Only towards the end of the diffuser does the fast flow start to migrate towards the concave wall.

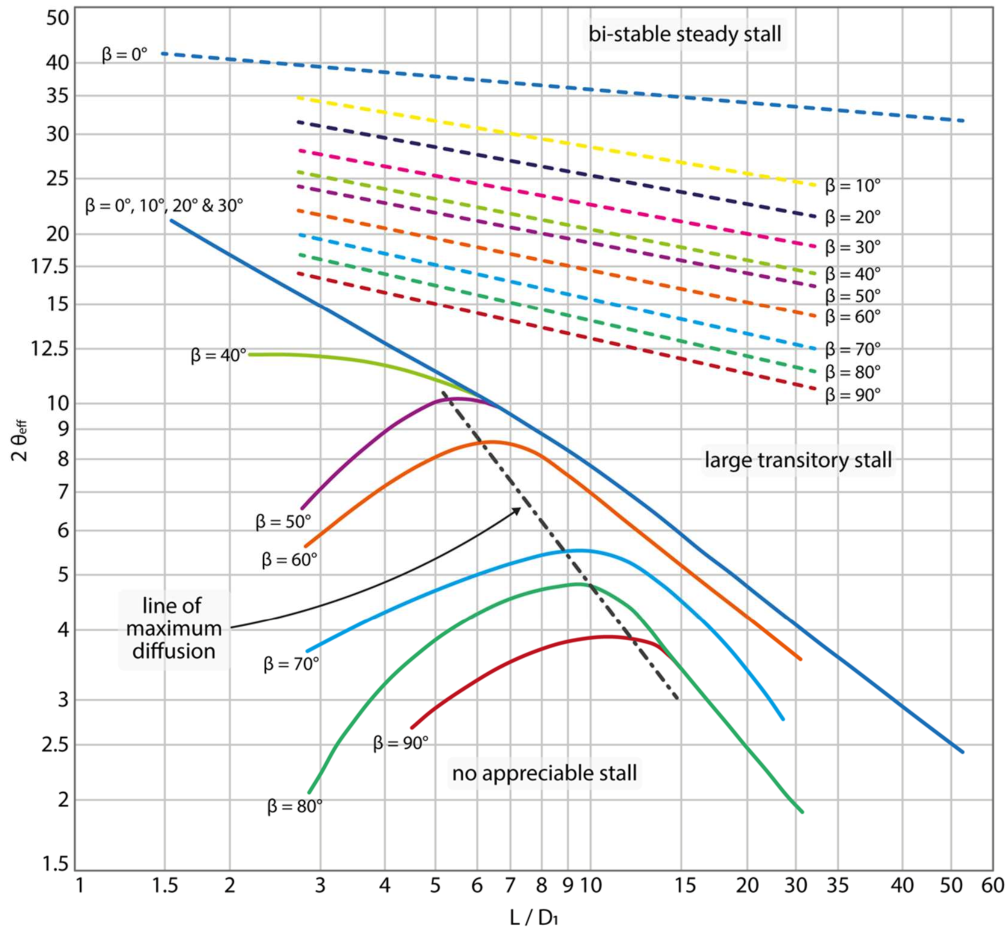

Fig. 7. Stall behaviour mapped over normalised diffuser size (L/D_1) and total effective expansion angle ($2\theta_{eff}$) for curved diffusers with varying turning angles β. Solid lines mark the lower boundary of the transitory stall regions, while dotted lines represent the upper boundary. Also included, is the line along which the lowest L/D_1 results in the maximum level of diffusion. Adapted from [14].

5. EXPANSION VANES

Expansion vanes somewhat combine the aforementioned theories of turning vanes and curved diffusers. One of the earliest investigations into expansion vanes was carried out by Friedman and Westphal in 1952 [18]. More recently, in 1998, Lindgren et al. conducted a series of tests on turning vanes and found very encouraging results for moderate expansion ratios [19]. In 2020, Drela et al. used a combination of wire screens and expansion vanes to achieve high flow quality even at larger expansion ratios [20]. Note that these vanes were designed to be able to work without a honeycomb, but using one further improves performance. The different expansion ratios and loss factors are summarised in Table 2; the geometry of some expansion vanes is compared in Fig. 8. Here the vanes developed by Sahlin et al. have not been included, as their shape is optimised for a non-expanding corner.

Table 2. Various expansion vanes and their associated loss factors at specified Reynolds Numbers

Vane Type	Expansion		Reynolds Number	*K*
	(%)	(ratio)		
Sahlin	0	1:1	1.54×10^5	0.036
Sahlin	33	4:3	2.00×10^5	0.054
Lindgren	33	4:3	2.00×10^5	0.047
Friedman	45	29:20	3.30×10^5	0.110
Sahlin	50	3:2	2.00×10^5	0.080
Sahlin	66	5:3	2.00×10^5	0.140
Drela	100	2:1	1.00×10^5	2.200

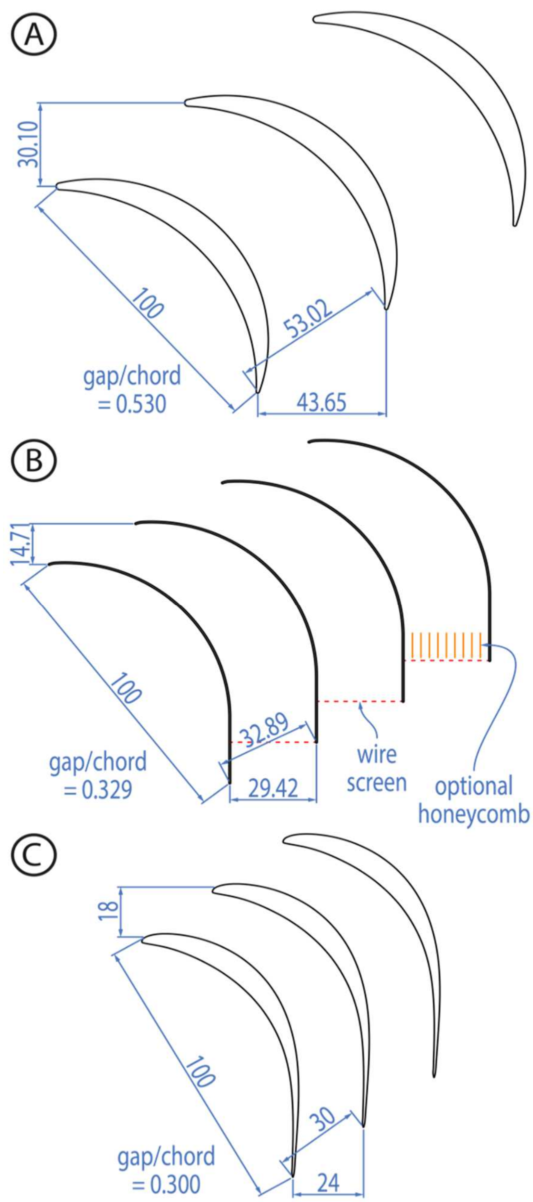

Fig. 8. Comparison of pitch, shape and setup of expansion vanes by Friedman et al. (A), Drela et al., (B) and Lindgren et al. (C). All vanes are scaled to have the same chord length for comparative purposes.

It seems that the vanes developed by Drela et al. are capable of reaching a much higher expansion ratio at the cost of a significantly increased loss factor. The concept was first used in a newly built closed-circuit wind tunnel at Brown University intended for animal flight testing. Breuer et al. incorporated the expansion vanes in the fourth corner to achieve a 1.875:1 expansion followed by an 8:1 contraction typical for wind tunnels [21]. The expansion in the final corner allowed for a substantial reduction in facility footprint at the cost of a small rise in the required operating power.

6. CONCLUSIONS

There exists comprehensive research on turning vanes and diffusers of numerous shapes and sizes. After reviewing the existing state of technology and the underlying theory, the following conclusions can be drawn:

- Some turbulent flow is desirable between turning vanes, as it will increase flow uniformity and decrease the size of the wakes downstream of the turn.
- To achieve minimum pressure loss and high flow quality, turning vanes need to be profiled.
- If bi-directional flow is required, some profiled vanes are unsuitable as they do not turn the flow sufficiently in reverse.
- Decreasing the vane gap increases flow uniformity at the cost of increased pressure losses.
- The diffuser geometry determines what stall regime the flow experiences while expanding, and the highest pressure recovery is achieved at the lower bounds of the transitory stall regime.
- With increased diffuser curvature, the bounds of the transitory stall regime are lowered, which limits the practical L/D_1 ratio of curved diffusers (and expansion vanes).
- All existing experiments are either limited to small expansion ratios or employ dense wire screens, leaving room for further research into larger expansions without screens.

7. CONTRIBUTIONS

Alexander Baron von Hohenhau carried out the investigation, data curation, visualisation, formal analysis, writing and editing. Vasile Năsui was paramount in project administration and funding acquisition.

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ANALIZA PALETELOR DE ROTIRE, A DIFUZOARELOR CURBATE ȘI A ROLULUI LOR ÎN TUNELURILE COMPACTE DE VÂNT ȘI APĂ

Paletele de rotire şi difuzoarele curbate fac parte integrantă din multe tuneluri moderne de vânt şi apă. Acestea facilitează rotirea fluxului în jurul curbelor strânse, crescând uniformitatea fluxului, reducând în acelaşi timp pierderile de energie şi turbulenţele. Există literatură amplă care discută despre acesti condiţionatori de flux, dar informaţia este dispersată. Acest lucru necesită o nouă imagine de ansamblu a evoluţiilor de la construirea primului tunel de vânt cu circuit închis în 1909. O atenţie deosebită este acordată paletelor moderne de expansiune, care permit construirea unor facilităţi de cercetare mai compacte. O meta-analiză cantitativă a coeficienţilor de pierdere a presiunii va ilustra beneficiile din lumea reală ale utilizării paletelor de expansiune specifice. Cea mai bună performanţă este obţinută de paletele dezvoltate la KTH şi MIT pentru colţuri standard şi, respectiv, în expansiune.

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