A Numerical Investigation into the Effect of Blade Trailing-Edge Thickness on the Performance of Axial Fans

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Abstract. This paper presents the results of a numerical investigation into the effect of blade trailing-edge thickness and shape on the performance of a rotor ring model of axial fan. The numerical simulations carried out under this investigation provided the performance characteristics of efficiency, working medium power, and total pressure increase in the function of the volumetric flow rate of the rotor ring. The investigated blade trailing-edge thickness values were 1 mm, 2 mm, and 3 mm. The models for the simulation series were developed with rounded and sharp blade trailing edges, for all thickness values thereof. The rounded trailing blade edges were modelled in the form of an arc over which the conditions of tangency with the upper and lower contours of the airfoil were imposed. The blades of the modelled blade fan were designed with the NACA 65-810 airfoil. To verify the applied turbulence model and mesh settings, experimental tests of the model rotor ring were performed on an axial fan test bed.
The obtained experimental data was compared with numerical results. The results showed a significant impact of the thickness and shape of the blade trailing edge on the performance characteristics of axial fans.

**Keywords:** CFD, fluid-flow machines, axial fan characteristics

1. BACKGROUND

The design engineering of fluid-flow machines is extremely complex and requires the determination of key geometric parameters of the rotor rings, which are essential components of fluid-flow machines. The basic parameters which define the geometry of the blades include: blade height, aerodynamic and geometric twist, blade airfoil type, and hub engagement angle. The manufacturing process of the blades has a major effect on the performance parameters of the fluid-flow machines, given the quality of trailing-edge reproduction.

The trailing edge of a blade is where the working medium’s flow detaches from the blade, resulting in loss of flow. Naturally, the lowest loss of flow occurs in a blade where the trailing edge has a thickness of zero. Theoretically speaking, Żukowski’s airfoil features a zero-thickness trailing edge [11]. However, no manufacturing process is capable of reproducing blades with a zero-thickness trailing edge. When trailing edges are suitably machined (ground and milled), the ratio of trailing-edge thickness, $g$, to airfoil chord, $l$, is approximately 0.01, the loss of flow from a finite thickness of the trailing edge is relatively low [15]. Cast blades and blades made of metal sheets of a constant thickness have much higher values of $g/l$.

The effect of blade trailing-edge thickness on the loss of flow can be estimated by an analysis in which the flow of the working medium in a blade passage is investigated [15]. However, the accuracy of the method is rather poor, especially when the airfoil curvature is relatively high.

Figure 1. shows the velocity triangles of the working medium flow through a blade passage in a flat airfoil palisade, formed by airfoils with chord $l$ and a finite trailing-edge thickness $g$. In Fig. 1, velocities $C_2$, $C_u$ and $C_z$ were the result of a complete blending of the working medium downstream of the flat airfoil palisade [15]. Superscript $m$ denotes the velocity values within the blade passage.
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Fig. 1. Changes in velocity of the working medium flow within the blade passage where the blade trailing-edge thickness was finite. $C_2$ – mean velocity downstream of the trailing edge; $C_u$ – tangential velocity; $C_z$ – axial velocity.

According to [15], the loss factor from the finite trailing-edge thickness could be expressed as follows:

$$
\zeta_m = B \left( \frac{g}{t \sin \beta} \right)^2 \tan^2 \beta \left[ 1 + \left( 1 - \frac{g}{t \sin \beta} \right)^2 \tan^2 \beta \right]^{-1}
$$

with:

- $B$ – blending loss constant [15];
- $g$ – trailing-edge thickness;
- $t$ – tangential spacing between airfoils;
- $\beta$ – exhaust cross-sectional angle of the blade.

Note that the theoretical considerations of an axial fluid-flow machine flow apply to a straight-line airfoil palisade only. The actual flow through a rotor ring of a fluid-flow machine is a three-dimensional non-stationary phenomenon.

As already said, the method of trailing-edge reproduction is a major contributor to the loss of flow, and to the improvement of performance of fluid-flow machines. Given certain difficulties encountered with the analytical studies of the effect of blade trailing-edge thickness on the performance of fluid-flow machines [15], the investigation discussed herein was based on experimental tests [12].
Concerning the requirements for improvement of fluid-flow machine performance [13], it is numerical methods only which enable a cost-efficient assessment of the effect of blade trailing-edge reproduction methods on the loss of flow [3, 9].

2. METHODOLOGY OF NUMERICAL ANALYSIS

The effect of rotor blade trailing-edge thickness on the performance of axial fans was analysed with computerised methods of fluid mechanics, implemented in the Ansys Fluent software tool. Given that the gas flow was non-stationary and determined by a finite number of blades, the flow was assumed to be turbulent.

A turbulent flow is often characterised by the flow in a passage and the force applied to the flow passage being invariable in time [6]. Given this, and despite a chaotic and pulsating pattern of motion of flow, the mean flow parameters were constant in time. This motion is called a mean-stabilised motion [6]. In this case, the motion parameters could be expressed as the sum of mean (stabilised) values and time-transient pulses. The flow parameters of this mean-stabilised turbulent motion could be expressed as follows:

– pressure:

\[ p = \bar{p} + p' \] (2)

– velocity components:

\[ u_x = \bar{u}_x + u_x' \]
\[ u_y = \bar{u}_y + u_y' \]
\[ u_z = \bar{u}_z + u_z' \] (3)

The notation of the flow parameters (2) and (3) was used to derive the so-called averaged Navier-Stokes equation, an expression applied in computer fluid mechanics.

By applying suitable transformations [6], the set of averaged Navier-Stokes equations, which are termed Reynolds Averaged Navier-Stokes (RANS), was:

\[ \frac{\partial \bar{u}_i}{\partial x_i} = 0 \]

\[ \frac{\partial}{\partial x_j} (\rho \bar{u}_i \bar{u}_j) = -\frac{\partial \bar{p}}{\partial x_i} + \] (4)

\[ + \frac{\partial}{\partial x_j} \left[ \mu \left( \frac{\partial \bar{u}_i}{\partial x_j} + \frac{\partial \bar{u}_j}{\partial x_i} \right) - \frac{\partial}{\partial x_j} (\rho \bar{u}_j \bar{u}_i') \right] \]
with:

- $i, j$ – indicators $i, j = 1, 2, 3$;
- $\overline{u}$ – averaged velocity;
- $\overline{u}'$ – averaged pulse rate;
- $\rho$ – density;
- $\mu$ – dynamic coefficient of viscosity.

The equation (4) member $- \rho \overline{u}_j \overline{u}_i'$ represented the stresses related to the average rate of momentum change caused by turbulent stresses. The stresses could be expressed as a square table with the following form, given the traditional Cartesian denotation [6]:

$$
\Pi_T = \begin{bmatrix}
\rho \left( \overline{u}'_x \right)^2 & \rho \overline{u}'_x \overline{u}'_y & \rho \overline{u}'_x \overline{u}'_z \\
\rho \overline{u}'_y \overline{u}'_x & \rho \left( \overline{u}'_y \right)^2 & \rho \overline{u}'_y \overline{u}'_z \\
\rho \overline{u}'_z \overline{u}'_x & \rho \overline{u}'_z \overline{u}'_y & \rho \left( \overline{u}'_z \right)^2
\end{bmatrix}
$$

(5)

Tensor (5) is termed a “turbulent stress tensor” or a “Reynolds stress tensor”. Note that the system of equations (4) was not closed. Its closure would require the addition of six equations which determine the components of turbulent stress tensor (5). In computer fluid mechanics, the equations resulted from the applied turbulence model [7]. The closure of the system also required determination of the following dependencies:

$$
p = p(\rho, T)
$$

(6)

$$
\mu = \mu(T)
$$

(7)

In this paper, the numerical calculations were performed by the application of an SRF (Single Reference Frame) model, for which a set of recommended turbulence models was given [1]:

- Spalart-Allmaras;
- realizable $k$-$\varepsilon$;
- SST $k$-$\omega$.

Based on the results from [7], it was demonstrated that the most appropriate turbulence model for this numerical investigation of the working medium flow through the blade passage of an axial fluid-flow machine was a realisable $k$-$\varepsilon$ model. The following characteristics were determined with the series of numerical analyses:

- total pressure of the axial fan rotor ring as a function of the volumetric flow rate, $\Delta p = f(Q)$;
- total efficiency of the axial fan rotor ring as a function of the volumetric flow rate, $\eta = f(Q)$;
- working medium power of the axial fan rotor ring as a function of the volumetric flow rate, \( P = f(Q) \).

3. GEOMETRIC MODELS OF THE BLADES

To complete the verification experimental tests, a virtual model of the rotor blade was developed from the NACA 65-810 airfoil. This airfoil type was selected from a study of available catalogue data [5] and the aerodynamic characteristics of the airfoil in its palisade. Note that the airfoil characteristics in a palisade are quite different from the characteristics of an isolated airfoil.

The virtual model of the rotor blade was built in a dedicated CAD software package, Siemens NX (Fig. 2).

Fig. 2. Virtual model of the blade

Two sets of geometric models of the blades which comprised the rotor ring were prepared for the numerical analysis of the effect of blade trailing-edge thickness on the performance characteristics of axial fans. Each set included three models, each with a different blade trailing-edge thickness: 1 mm, 2 mm, and 3 mm. The models in the first set featured a rounded trailing edge. The models in the second set had no rounding at the trailing edge. All blades were developed with three airfoils, equally spaced along the blade span. The blades featured identical airfoils (NACA 65-810), airfoil orientation angle, and chord orientation angle at the corresponding cross-sections.

The blade trailing-edge thickness was increased by adding a triangle confined within the tangent which originated from the corresponding point along the airfoil centre line, the part of the profile line from the tangent point to the trailing-edge point, and the closing section perpendicular to the airfoil centre line at the trailing-edge point. While the modelling method proposed herein changed the flow trailing angle of the blade, it was consistently applied to avoid any intervention in the form of the blade’s inner surface.
A similar method of trailing-edge formation for a numerical analysis of the effect of blade trailing-edge thickness on the performance of axial fans was applied in [3]. Figure 3 shows a modified NACA 65-810 airfoil of the blade trailing edge thickened to 1 mm, 2 mm, and 3 mm. This approach produced the blades without rounded trailing edges.

Fig. 3. Method of modifying the NACA 65-810 airfoil trailing edge

Figure 4 shows an example of the blade with a rounded trailing edge that was 1 mm thick. The rounded trailing edge in all models was produced by inscribing a circle with a suitable diameter in the edge part of the airfoil. Conditions of tangency with the upper and lower contours of the airfoil were imposed onto this circle.

Fig. 4. Example of the blade with a 1 mm thick rounded trailing edge

4. NUMERICAL MESHES

To complete the numerical analysis series for the air flow through the rotor rings developed from the modelled blades, hybrid numerical meshes were developed. The meshes featured identical parameters for all the investigated cases (with the minimum number of mesh elements equal to 223,000 at the boundary layer). A triangular mesh was developed on the surfaces of the defined calculation area. A boundary layer mesh in the form of 5 layers of prismatic mesh elements was modelled in the boundary layer area. The remaining volume of the calculation area was discretized with tetrahedral mesh elements.

These hybrid meshes were typical mesh types applied in the calculations of fluid-flow machine elements with complex geometric shapes. With the hybrid meshes developed, their quality was checked; the quality indicators had to be confined to certain limits [2]. The quality check included the obliquity and orthogonality of each numerical mesh.
Figure 5 shows an impression of the finished numerical mesh with the prismatic elements shown within the boundary layer area.

Fig. 5. Digitisation of the calculation area with prismatic elements shown

5. BOUNDARY CONDITIONS

The performance characteristics of axial fans are usually determined by experimental testing on standard test beds [10]. The experimental test procedure is to gradually throttle down the working medium flow at a thurling operated with a damper.

The dynamic gas parameters are measured at predetermined throttling ratios and at a suitable test bed cross-section; the electrical performance parameters of the fan driving motor are also measured.

A similar approach was taken for the numerical analysis; the throttling of the test axial fan was simulated by modifying the mass flow of air at the axial fan intake side. It was a standard procedure applied in numerical simulation of working medium flow through axial fans [1].

The following boundary conditions were used, as defined at the boundaries of the calculation areas:

- axial fan intake: mass flow;
- axial fan exhaust: pressure at 0 Pa (relative to a reference pressure of 101 325 Pa);
- axial fan hub: rotational speed at 1450 rpm;
- axial fan casing: rotational speed at 0 rpm.

These boundary conditions were typical in similar investigative cases [4], [8], [14] and [16].
The series of numerical analyses was carried out with an SFR (Single Reference Frame) model. SFR models facilitate the application of periodic boundary conditions input to the lateral boundaries of the calculation domain. These conditions suffice for the modelling of just a sector of a rotor ring, inclusive of the calculation area developed for it. During the calculations, the SRF model applied automatically included interference between the blades around the whole rotor ring.

Modelling just a sector of a complex geometrical form reduces the rendering time of the meshes and the numerical calculation time.

6. ANALYSIS OF NUMERICAL SIMULATION RESULTS

The following is a discussion of the results from the series of numerical simulations of the air flow through rotor rings designed with the blade sets developed as explained before.

Figure 6 shows the performance characteristics of total pressure vs. volumetric flow rate for the blades with a preset thickness of the rounded trailing edge.

![Graph showing performance characteristics of total pressure vs. volumetric flow rate.](image)

Fig. 6. Characteristics of the increase of total pressure vs. volumetric flow rate for the three values of rounded blade trailing-edge thickness

The plot revealed that the highest increase of total pressure throughout the performance range was produced by the ring rotor that featured the blades with 1 mm thick trailing edges. The smallest increase of total pressure was found with the blades that featured the rounded 3 mm thick trailing edges. Considering the range of high volumetric flow rates, the reduction of total pressure from the rotor rings with 1 mm thick blade trailing edges was as follows for the 2 and 3 mm thick blade trailing edges: 5.1% and 9.3%.
The difference of the total pressure values between the blade trailing-edge thickness values was reduced with volumetric flow rate. This effect came from a reduced loss of flow, caused by the reduction of flow velocity.

Figure 7 shows the performance characteristics of total pressure vs. volumetric flow rate for the blades with a preset thickness of the sharp trailing edge. Here too, the highest increase of total pressure was found for the rotor ring that featured the blade trailing-edge thickness of 1 mm, while the lowest increase of total pressure was found for the rotor ring that featured the blade trailing-edge thickness of 3 mm; the blade trailing edges of all three thickness values were sharp (not rounded). Considering the range of high volumetric flow rates, the reduction of total pressure from the rotor rings with the blade trailing-edge thickness of 1 mm was as follows for the 2 and 3 mm thick blade trailing-edge thicknesses: 2.1% and 6.5%. The convergence of the characteristics in the range of low volumetric flow rates was caused by the reduction of flow velocity, which in turn caused a reduced loss of flow.

Figure 8 shows the performance characteristics of total pressure vs. volumetric flow rate for all three thickness values of blade trailing edges. The broken lines in the plot represent the blade versions with the sharp trailing edges; the solid lines represent the blade versions with the rounded trailing edges. A comparison of $\Delta p_t$ plots developed for the rotor rings with the rounded blade trailing edges to the plots of the same parameter for the sharp blade trailing edges revealed that for the same blade trailing-edge thickness, the sharp version generated a higher pressure than the rounded version.
Figure 9 shows the performance characteristics of efficiency vs. volumetric flow rate for all three thickness values of blade trailing edges. The plot shows that the highest efficiency was obtained for 3 mm thick blade trailing edges and it reached approx. 93.4% at a volumetric flow rate of 8 m$^3$/s.

The maximum efficiency values achieved for all three blade trailing-edge thickness values ranged from 93% to 93.4%.
From the plots, it was established that the sharp blade trailing edges generated a higher efficiency than the rounded blade trailing edges.

Figure 10 shows the performance characteristics of working medium power vs. volumetric flow rate for all three thickness values of blade trailing edges. The plot shows that the highest working medium flow rate was generated with the rotor rings that featured 1 mm thick blade trailing edges and equal to 34.4 kW. This was an effect of the higher total pressure increase generated by the rotor rings with the same blade trailing-edge thickness. The maximum working medium power range in all cases ranged from 32.8 kW to 34.4 kW. The authors found from these results that the rotor rings with the sharp blade trailing edges provided higher working medium power values.

![Fig. 10. Performance characteristics of working medium power vs. volumetric flow rate for all the three thickness values of blade trailing edges](image)

7. CONCLUSION

This paper presents the results of a numerical investigation into the effect of blade trailing-edge thickness on the performance of axial fans by application of a numerical method. The series of numerical simulations was preceded by experimental tests which demonstrated that the applied numerical mesh and turbulence model were valid. The results confirmed the feasibility of the applied numerical method in the investigative cases presented herein.

The design engineering process of fluid-flow machines should consider the losses from the finite trailing-edge thickness and the method of reproduction of blade trailing edges. It was demonstrated herein that the thickness and shape of the blade trailing edge have a significant effect on the performance characteristics of axial fans.
REFERENCES

Streszczenie. W niniejszej pracy przedstawiono wyniki numerycznego badania wpływu grubości i kształtu krawędzi spływu łopatki wirnikowej na charakterystyki pracy modelowego wieńca wirnikowego wentylatora osiowego. W toku symulacji numerycznych uzyskiwano charakterystykę sprawności, mocy oraz przyrostu ciśnienia całkowitego w funkcji objętościowego natężenia przepływu przez wieniec wirnikowy. Przyjęte do analizy grubości krawędzi spływu łopatki wynosiły 1 mm, 2 mm, 3 mm. Modele do serii symulacji numerycznych wykonano dla zaokrąglonych i niezaokrąglonych krawędzi spływu o podanych wyżej grubościach. Zaokrąglone krawędzie spływu odwzorowywano w postaci łuku, na który narzucono warunki styczności do górnego i dolnego obrysu profilu definującego kształt łopatki. Łopatki modelowego wieńca wirnikowego zbudowano w oparciu o profil NACA 65-810. Na potrzeby sprawdzenia poprawności przyjętego modelu turbulencji oraz ustawień siatki przeprowadzono badania doświadczalne modelowego wieńca wirnikowego na stanowisku do badań wentylatorów osiowych. Uzyskane dane eksperymentalne zestawiono z wynikami numerycznymi. Przedstawione wyniki wykazały istotny wpływ grubości i kształt krawędzi spływu łopatki wirnikowej na charakterystyki pracy wentylatora osiowego.

Słowa kluczowe: CFD, maszyny przepływowe, charakterystyki wentylatorów osiowych