Effects of a Curved Surface of the Blade on the Performance Characteristics of Axial Flow Rotors

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ABSTRACT

The benefit of using a Computational Fluid Dynamics (CFD) technique is to predict the effects associated with non-radial stacking (NRS). Three-dimensional (3D) flow field for NRS blades of an axial turbomachine is studied. The non stacking line of the fan is provided by a curved surface by means of enlarging blade airfoil only at the base of the fan blade. The effectiveness of the NRS on the axial fan rotor of low-aspect-ratio was studied in two separate case studies a standard blade (SB) and a curved blade (CB).

In developing a complete structured hexahedral mesh for the entire computational domain, comparative studies of CB and SB were conducted at the design and off-design flow rates. The structured mesh technique minimizes cell counts, cell skewness, and enables cost-effective CFD investigation.

The results are presented in the form of local radial velocity, local ideal and total pressure rise of the outlet as well as the static pressure on the blade suction side. Specifically, it is pointed out that in the design point the CB rotor exhibits the highest efficiency for the most part of the entire span, whereas SB exhibits the lowest efficiency along the entire span. While in the off-design point, the CB rotor exhibits the highest efficiency at the blade hub only.

Keywords: Three-dimensional turbomachinery flow, axial fan, non-radial stacking, structure meshing.

1 Introduction

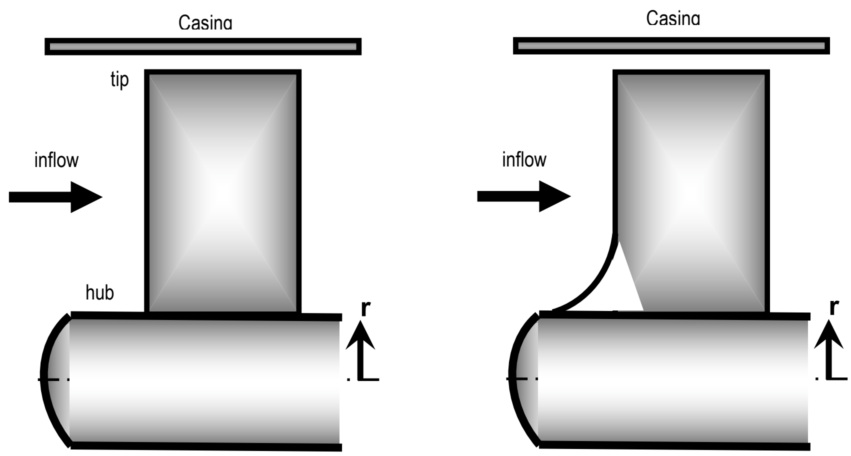
Improving the axial fan blade shape, by means of the stacking line technique, has become the main variable design. The blade shape designs, provided by stacking line performance, are often offered to increase efficiency and to reduce losses in axial turbomachines. Distortion of stacking line using sweep, dihedral, skew and/or any other forms have become nowadays a matter of requiring need in the design of turbomachinery blades. The form of the stacking line, which is only a radial line, is modified to take on new shapes by changing the blade profiles in definite directions. Some of these modifications are called a sweep, dihedral and skew (the combination of sweep and dihedral) depending on a certain movement direction of the blade profile.

The blade can be modified by sweep and/or dihedral if the blade sections of a datum blade of radial stacking line are displaced parallel to and/or normal to the chord, respectively [1].

There are four main types of sweep in the blade which are backward, forward, positive, and negative sweep. A blade is swept forward or backward at a given radius if the blade sections of a radially stacked datum blade are shifted parallel to their chord in such a way that a blade section under consideration is upstream or downstream of the neighbouring blade section at lower radius, respectively, sweep is said to be positive or negative near an endwall when a blade section under consideration is upstream or downstream of the adjacent inboard section, respectively [2].

In this work, two axial fan blades are designed and analyzed. The results are also discussed and interpreted. The first blade is called the Standard blade (SB) which is to be easy to manufacture, and the second one has the same characteristics of the first one, but it was modified with an additional curved surface at the base of the blade, this blade is called Curved blade (CB).

The curved blade is prepared by means of increasing the blade chord length and enlarging blade airfoil at the base the fan blade, this curved blade is defined to be as a part of positive backward sweep profile at the hub as shown in **Figure 1**.



a) Standrd blade rotor b) Curved blade rotor

Figure 1: Schematic drawing for CB rotor.

The use of properly curved surface attached to the base of the fan blade is considered to prevent a total pressure loss and to provide a smooth transition of the flow stream through the blades of the fan. As an example, compressors have generally benefited from the use of backward sweep at the hub and/or forward sweep at the blade tip [1].

In general, Govardhan et al. [3], mentioned that, backward swept blades show minor efficiency improvement relative to the unswept rotor but at a considerable loss in stall margin. The peak efficiency of the backward swept rotor is often found to be much closer to the stall line to be of practical use.

The aim of this study is to understand the nature of three-dimensional flow when the blade has a curved surface on the hub in a low speed axial fan. The performance of the CB blade type is compared to the SB blade type and analyzed as well.Geometry construction

1.1 Curved blade geometry

To improve the performance of the axial fan, we made a modification to the design of the blade itself similarly to the designs of the sweep. The procedure was in such a way that modifying the base of the fan blade. Simply by magnifying and enlarging the blade airfoil section and increasing the chord line at the blade hub, then reducing this amplification towards the blade tip, without displacing the blade sections from the trailing edge and without changing the type of blade sections. The blade section used is profile of C4 (10%) along the entire span [4] as shown in **Figure 2**. However, there is a lack of information and knowledge on the curved surface applied in the blade design and the reference [4] serves as a preliminary reference for the data.

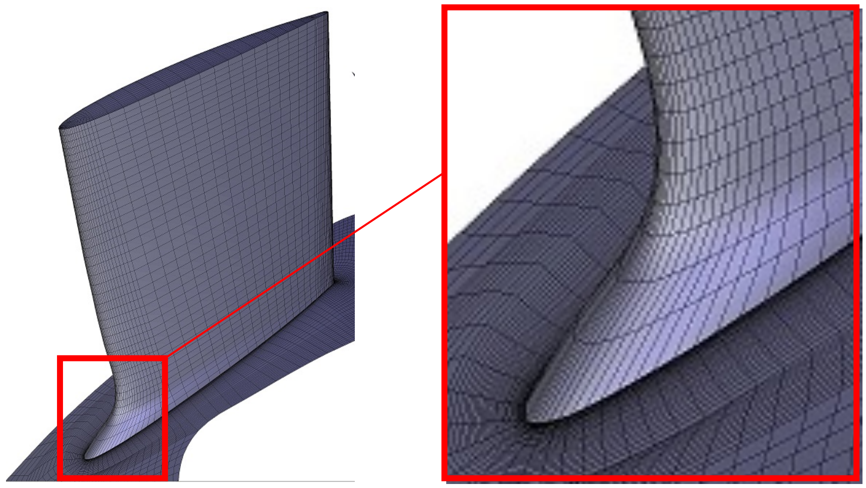


Figure 2: 3D view of the curved surface of the CB blade geometry.

The Reynolds number (Re) is approx. 1.07×10^6 based on the characteristic length as the chord line at the blade tip (0.372 m), the circumferential speed of the blade tip (43.56 m/s), and the kinematical viscosity of air at 20 °C (1.516×10^{-5} m^2/s).

1.2 Rotating cascades

The rotor blades are arranged so that the blades are assembled in an annular cascade, which is surrounded by a cylindrical casing, as shown in **Figure 3**. Another choice to use instead of the annular cascade is a fixed linear cascade of blade rotors in a linear cascade, which is an easy technique to design and it is not used in this study, because the radial effects such as the radial pressure gradients, the radial distribution and the secondary flows will not be developed as in the annular cascade. An important interest in the rotation cascades is their ability to simulate the effect of radial forces on the flow through the blade passages.

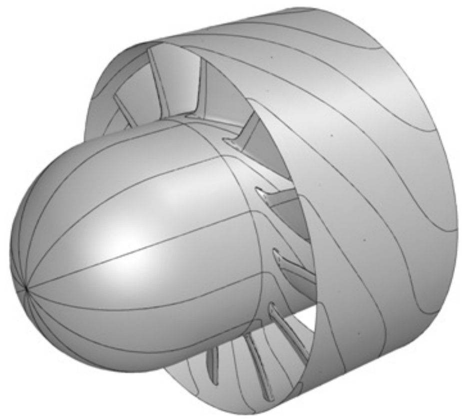


Figure 3: The CB rotor in cascade form.

Because the mesh periodicity for one domain is one of the most important constraints on the organized structured meshes, it has been taken into consideration more seriously. In cascade investigations this means that the flow periodicity adjustment becomes more difficult.

2 CFD technique

A commercially available ANSYS-17.2 finite volume code is used. The examined geometry in this work has 12 blades surrounded by a casing of diameter 2000 mm. Initially, a three-dimensional volumetric domain was constructed around one blade. The domain is divided into three parts. From the inlet to the exit of the domain, the first part begins with the inlet until the end of the hemisphere; the second part begins at the end of the first part until the front of the blade hub, and the last one contains the rest of the domain including the blade, hub and the outlet. The tip clearance (ν) is a 5% span with a Hub-to-tip ratio (τ) of 0.6. A typical computational domain for the CB rotor is shown in **Figure 4**.

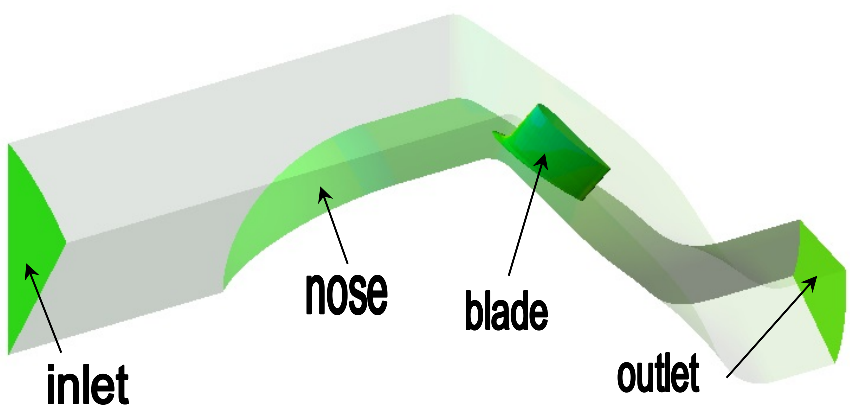


Figure 4: Computational domain for CB rotor.

A grid is said to be structured if all the vertices of the internal cell belong to the same number of cells of the control volume in which the domain is divided. The grid can be mapped to a rectangle (in 2D) or a block (in 3D), while unstructured grids divide the domain into simple elements, generally triangular (in 2D) or tetrahedral (in 3D) without implicit connectivity [5].

Because good mesh quality is crucial for good CFD analysis, it is important to construct a high-quality grid, especially where detailed flow are needed and at the interesting zones of the domain, such as in the vicinity of blade surfaces. The multi-block structure method allows the use of specific local structural grids in different places. Ali and Tucker [6] found that the multi-block structured meshes offer better computational efficiency than the unstructured meshes which, on the other hand, are more flexible for complex geometries.

At beginning the domain was divided into multiple separate volumes. By using an adequate hexahedral meshing each volume was meshed separately. Only H-Grid and C-Grid meshes were used for organized 2D meshes.

The majority of the number of cells in the domain is condensed around the vicinity of the blade. The multi-block structure method is used in this work; the domain was divided to 39 blocks and meshed to 304306 hexahedral cells. The equiangle skewness was 69.41% far from the blade, with restriction of periodic domain. A domain of structured grid is shown in **Figure 5**.

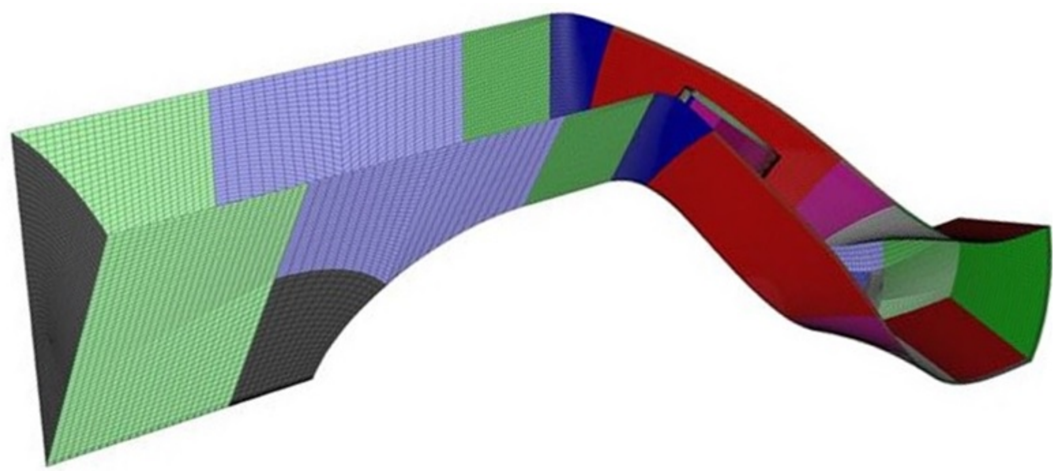


Figure 5: The splitted computational domain of structured grid.
(The casing is hidden for clarity)

3 Boundary conditions

For the design flow rate, the velocity of 9.2 [m/s] is used at the domain inlet, while 7.5 [m/s] is used for the off-design flow rate, the outflow is used as an outlet and the turbulence model used is k-ε model with enhanced wall treatment. The y+ values are mostly between 30 and 100 in the interesting regions such as at the vicinity of the blade. Some values of y+ are larger than 300 on grids where the areas are not interesting and located far from the blade. Typical computational run required about 6000 iterations.

4 Results and Discussion

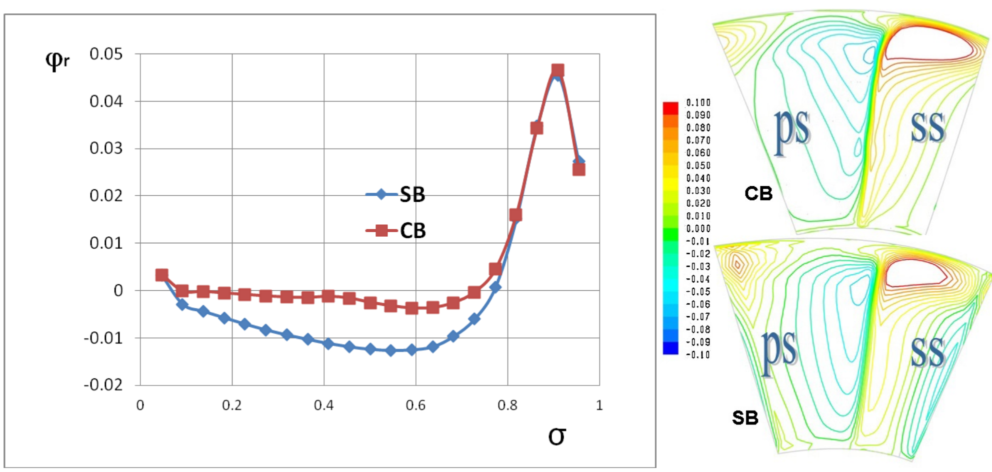
The results provided to be on outlet planes mounted at the axial direction very close to the blade. Also, other investigations as well, located at 20% and 90% measured from the hub in the spanwise direction. Some parameters and symbols must be mentioned here and clarified, such as (SS) and (PS). They represent the blade suction side and pressure side, respectively.

The fraction of span (σ) is the radial distance from the hub divided by blade span height, the reference velocity (u_{ref}) or (u_t) in [m/s] is represented as the blade tip speed = (d_t π n) where, d_t is the blade tip diameter and n is the rotor speed (in revolutions per second), accordingly the definition of the radial local flow coefficient (φ_r) *is* [4]

$\varphi_r = v_r \,/\, u_t$ ....(1)

Where, (v_r) is the radial velocity. **Figure 6** shows, in the range of the rotor hub up to 65% of span, the averaged radial velocity of CB is almost stable while, the averaged radial velocity of SB is reduced in this range, and then the radial velocity for rotors have increased to 90% of span, after that the averaged radial velocities are reduced at the blade tip. The velocities are quite similar for both rotors from 80 % of span to the casing. According to continuity, this yield to rearranging the values of radial velocity at the blade tip between SS and PS, so that pitch-wise averaged radial velocities has been the same for both blades.

Lower than the 80 % of span, some different reorganization in radial fluid for SB and CB blades are appearing with more radially inward flow for CB blade, as observed by Beiler [7].



a) outlet pitchwise averaged data b) distribution at outlet plan.

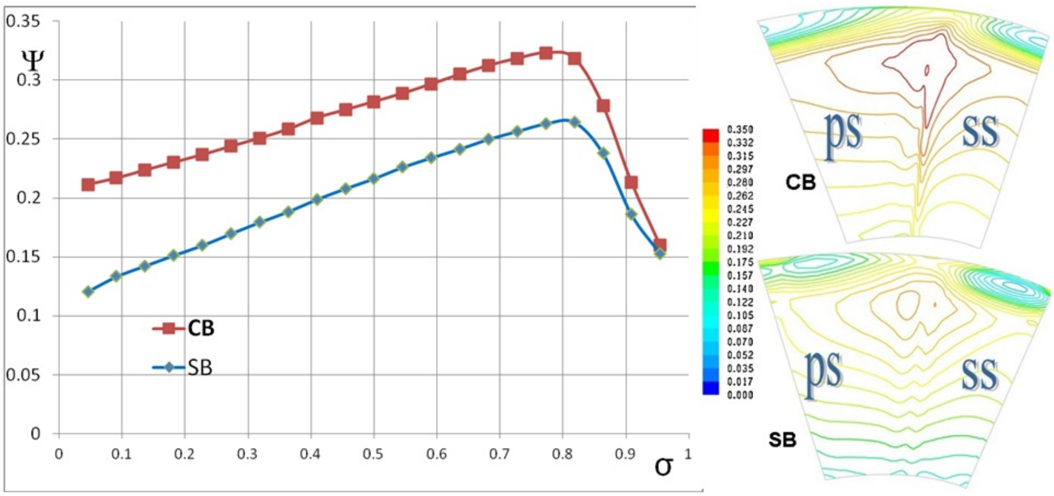
Figure 6: Outlet radial velocity.

The definition of the local total pressure rise coefficient (ψ) [4] is

$\psi = \Delta p_t \,/\, (\rho\, u_{ref}^2/2)$ ....(2)

Where, ρ is the air density and Δp_t is the pitchwise mass-averaged local total pressure rise. A mass averaged quantity is obtained by integrating the scalar time mass flow divided by total mass flow over the region. The outlet plan is divided to 21 slide regions.

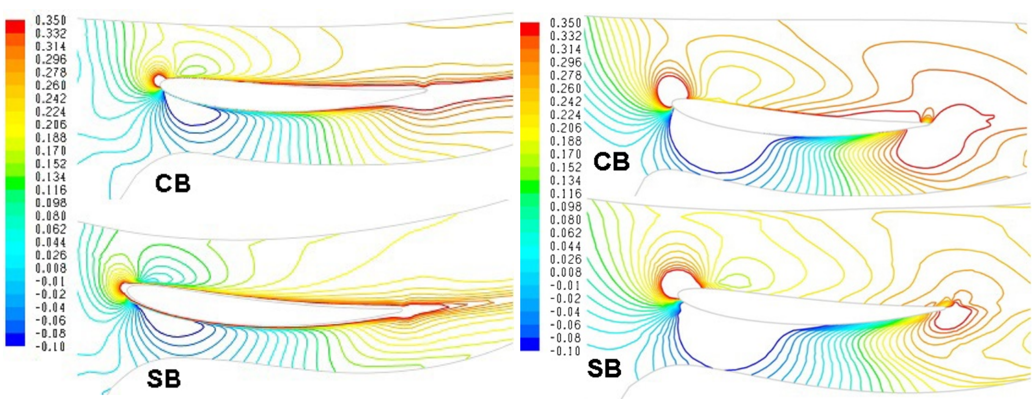
The CB rotor recognized increased total pressure rise compared to SB at the dominant blade spanwise. The contrary noticed by Beiler and Carolus [8]. CB rotor tends to increase the total pressure rise, especially in the vicinity of the trailing edge. Figure 7 indicates, SB and CB rotors perform lower total pressure rise at the hub and tip.



a) outlet pitchwise averaged data b) distribution at outlet plan.

Figure 7: Local total pressure rise.

In general for both rotors, there is less total pressure rise at SS close to blade tip compared to lower radii. Figure 8 shows that the blades having curved surfaces cause increasing the total pressure on PS's at lower and higher radii.



a) at 20% b) at 90%

Figure 8: Distribution of local total pressure rise at 20% and 90% of span.

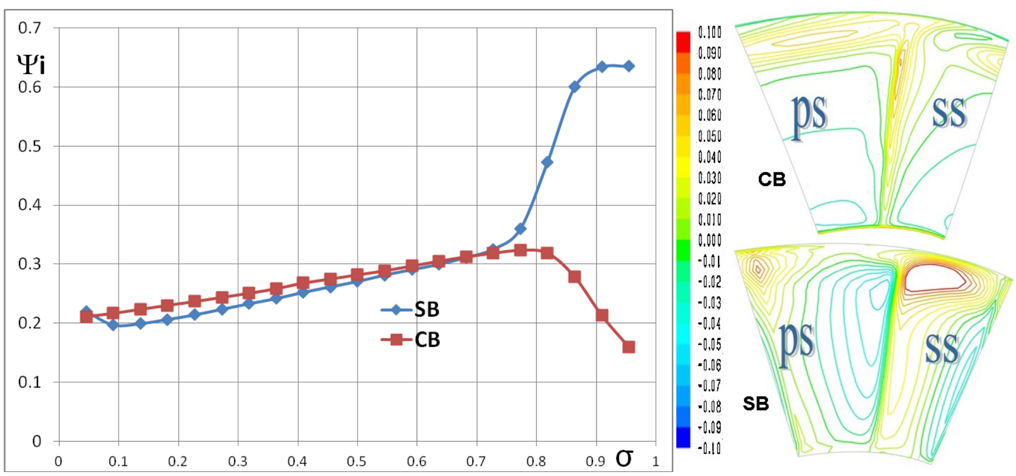
Dealing to the equation of the Euler, the total pressure rise of the inviscid flow is called the ideal total pressure rise. The definition of ideal total pressure rise coefficient (ψ_i) [4] is

$\psi_i = \Delta p_{ti} / (\rho \, u_{ref}{}^2/2)$ ....(3)

and, $\Delta p_{ti} = \rho \, r \, \omega \, v_{u2}$ ....(4)

Where, v_{u2} is the tangential pitchwise mass-averaged tangential velocity, r is radial coordinate and ω is rotor angular speed.

Figure 9 shows, CB rotor performs increased a little ideal total pressure rise compared to SB up to 75 % spanwise. The SB rotor increases the ideal total pressure rise after 75 % spanwise especially close the blade tip SS. However, due to the non-radial blade stacking, the Euler work at the tip is rapidly reduced; this is also being noticed by Clemen et al. [9]. Such an effect may cause a significant increasing local efficiency in this range of blade spanwise.



a) outlet pitchwise averaged data b) distribution at outlet plan.

Figure 9: Ideal local total pressure rise

In general, the fluid in the boundary layer of SS has a propensity to move radially outward. Due to CB blade, the isobars in the tip region are disorganized in a coarse tendency than for SB blade. In CB blade, the local radial outward flow is located in a higher location region on the blade trailing edge. Similar results have been illustrated in Yamaguchi et al. [10]. **Figure 10** illustrates the mechanism of CB blade to redirect the outer radial flow on the blade suction side.

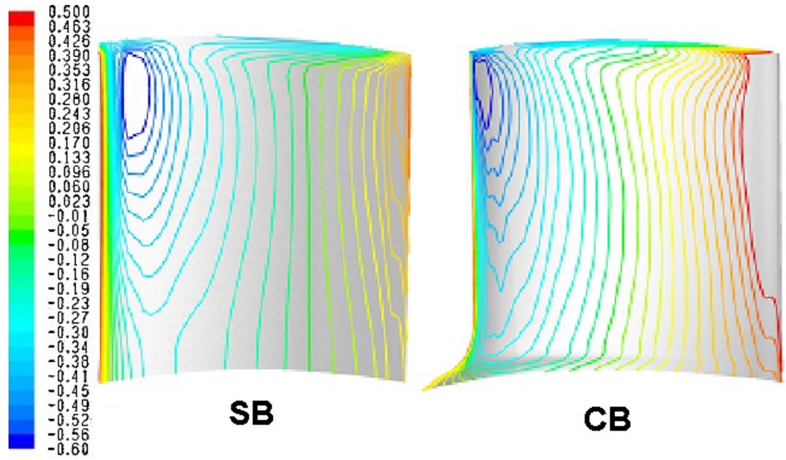
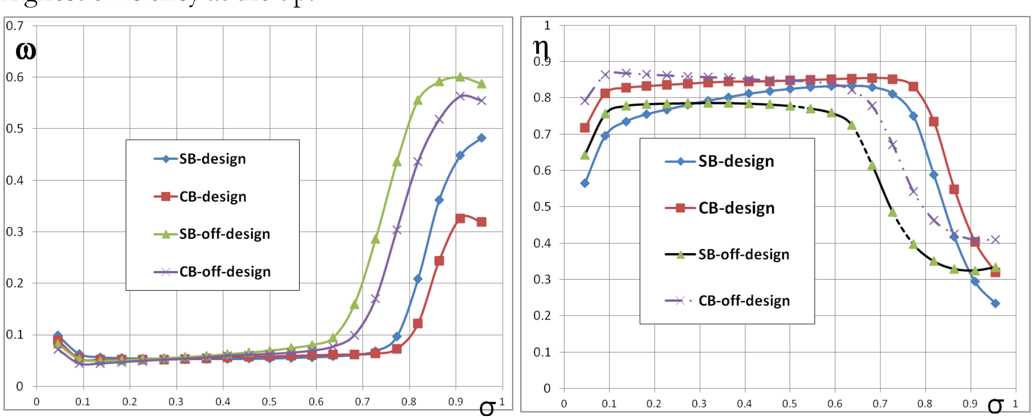


Figure 10: Distribution of static pressure coefficient on the SS.

The definition of the total pressure loss coefficient is expressed as (ω) = (ψ_i - ψ) and the local total efficiency is expressed as (η) = (ψ / ψ_i) [4]. **Figure 11a** shows the two rotors at the design flow rate having the same pressure loss coefficient at the hub and midspan until 75% of span, while the values of pressure loss at the blade tip are more increased for the SB rather than for CB rotor, as observed by Halder and Samad [12]. Whereas, at the off-design flow rate, the two rotors are having the same pressure losses up to 60% of blade spanwise. Toward the blade tip, a similar tendency of total pressure loss occurs, but with higher values for the off-design flow rate than for the design flow rate. In the axial fan blade shape design, Seo et al. [11] mentioned that, the efficiency parameter is the major objective for the design by using the stacking line technique. **Figure 11b** shows the local total efficiency profiles at the design and off-design flow rate along the span. The CB rotor at off-design flow rate exhibits the highest efficiency at the hub, whereas CB rotor at design flow rate exhibits the highest efficiency at the tip.



a) total pressure loss coefficient

b) total efficiency

Figure 11: Local spanwise distributions.

5 Conclusions

The purpose of this work is to examine the effects of the Non-radial stacking line by means of an applied curved surface to the rotor of an axial fan at the design and off-design flow rates. Comparative CFD studies have been carried out on the CB and SB rotors, without geometrical correction of the elemental blade cascades of the blade cross sections.

CB rotor obviously exhibits the highest local total efficiency in the design flow rate at the blade tip and in the off-design flow rate at the blade hub. While SB rotor exhibits the lowest local total efficiency in the design flow rate at the blade hub and in the off-design flow rate at the blade tip

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