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An Attempt to Evaluate the Cycloidal Rotor Fan Performance

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Abstract

In this work, the cycloidal rotor fan (CRF) performance was estimated by means of a numerical method based on Unsteady Reynolds Averaged Navier-Stokes equations (URANS). The fan with a cycloidal rotor belongs to the positive displacement machines of the rotary type. The numerical algorithm for simulation of the flow in the cycloidal rotor as well as postprocessing of the CFD results was prepared using Ansys CFX CEL. The methodology for the assessment of the CRF performance was proposed and verified. It was found out that the CRF performance strongly depends on the shape of the profile of the applied rotor blade. The NACA 0012 and CLARK Y profiles were tested for the same CRF structure and flow conditions. Also, the crucial importance for the CRF performance has the range of the blade pitch angle change.

Keywords

Cycloidal Rotor, Computational Fluid Dynamics, Fan, Performance

1. Introduction

The application of the cycloidal rotor idea to the fans is not known so far. For the time, being the most widespread application of cycloidal rotors is for ship propulsion and control but the origin of the cycloidal rotor relates to the aeronautic domain [1] [2] [3]. Cycloidal rotor is an attractive propulsion system for a wide range of aircraft applications. This type of propulsion system has very good maneuverability for various flight conditions, including hovering, and is mainly dedicated to vertical take-off and landing (VTOL) aircrafts [1] [4]. It was experimentally proved that besides good aerodynamic performance the cycloidal rotor produces little aerodynamic noise, what may be promising in terms to use it for fans. So far several analytical models were developed [4] [5] [6] which together with experimental investigations were helpful in design process of cycloidal rotors. However, currently, the Computational Fluid Dynamics (CFD) allows to take into account unsteady phenomena and it is willingly used in design and analysis of the operation of the cycloidal rotor of various configurations [7] [8] [9] [10].

The cycloidal rotor fan (CRF), like other cycloidal rotor applications, consists of several blades that describe the cycloidal path rotating about an axis perpendicular to the flow direction. The implementation of the cycloidal rotor for fans was numerically and experimentally investigated by authors in former works [7] [8]. The effectiveness of the elaborated CFD model was positively verified by experimental research using LDA measurements [8]. From the point of view of a positive displacement machine of the rotary type, the most effective design is the CRF with four blades. It presents a compromise between preserving a low oscillation of the flow stream, the complexity of the design and required rotational speed.

In the presented work the elaborated CFD algorithm for modelling the flow in the cycloidal rotor fan was used in order to estimate the performance characteristics of this type of machine. Also, it was investigated the influence of the blade profile shape on the CRF performance, the NACA 0012 and CLARK Y profiles were tested. Additionally, it was attempted to find the optimal blade pitch angle for the given rotational speed of the rotor.

2. Physical and Numerical Model

The cycloidal rotor fan (CRF), in contradiction to typical cross-flow fan, has no diffuser and its rotor generates the flow in any specified direction. It is possible by changing the blade pitch angle, α, of each blade individually during the rotor rotation with constant angular velocity, ω (**Figure 1**). The change of the blade pitch angle from the positive value to the negative one during rotation makes possible to work each blade both as a puller or pusher, depending on the blade position angle, γ. For considered here cycloidal rotor, the top blade has initial position with a maximum value of blade pitch angle. During the anticlockwise rotation with 90˚, the blade pitch angle decreases to the zero value. After further 90˚, the blade pitch angle takes a negative value of maximal blade pitch angle. After the next 90˚, the blade pitch angle has again zero value and during the last quarter of rotation, the blade returns to the initial position. The flow direction in such fan can be easily changed by setting the arbitrary initial position of the blades pitch angles. The number of the blades in the proposed design is dependent on the ratio of the chord of the rotor blade, c, and rotor radius, R. In the presented design, the blade chord was 50 mm and the radius was 70 mm. For the value c/R in the range of ~0.25 - 1.00, the maximum number of blades is four [11] (see **Figure 1**).

The volume flow rate of the cycloidal rotor fan (CRF) is affected by rotor swept area (*i.e.* rotor span and radius) and airflow velocity, which depends on the angular velocity (ω) and the range of change of the blade pitch angle (2α).

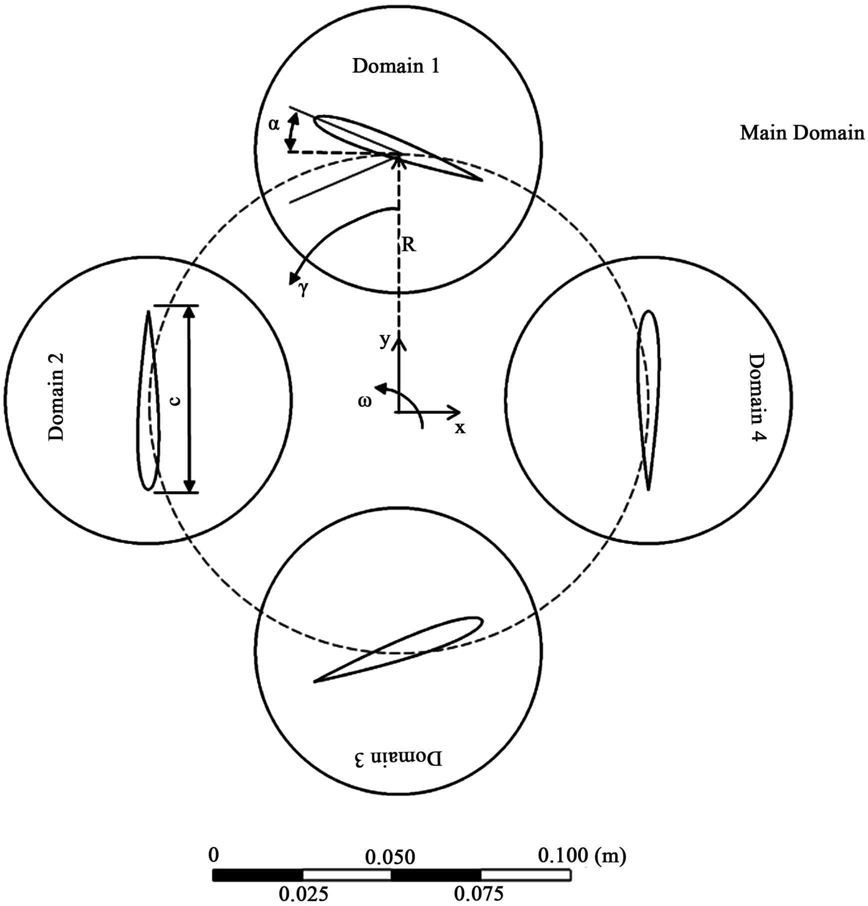

Figure 1. Scheme of the cycloidal rotor fan with four blades of the NACA0012 profiles [8].

The CFD simulations of the flow in the cycloidal rotor fan (CRF) with blades of NACA 0012 and CLARK Y profiles were made and herein presented. Numerical calculations of unsteady flow field in CRF were performed by means of Ansys CFX software. To this end the URANS solver with SST k-ω turbulence model was employed. The computational mesh consisted of about 0.7 M nodes in total, for five domains, four domains for rotors and one main circular domain of the radius 10·R (**Figure 2**). The grid was three dimensional, but only three layers in third (Z) direction were used. The slice of 5 mm thick was assumed with "Symmetry" boundary conditions. Therefore, obtained results of the flow field should be treated as a two dimensional only. The "Opening" boundary condition with opening pressure of 0.1 MPa and temperature 15˚C was set in main circular domain on the outer surface [2] [3]. Whereas, for the connections between rotors domains and main domain, the "General connection" boundary condition was used. For swinging the rotor domains during CRF rotation, *i.e.* modeling the change of the blade pitch angle during the rotor rotation, a special script in Ansys CFX CEL was written, because in the standard available settings it was not possible to rotate rotor simultaneously around several different centers of rotation (for rotor blades and entire CRF).

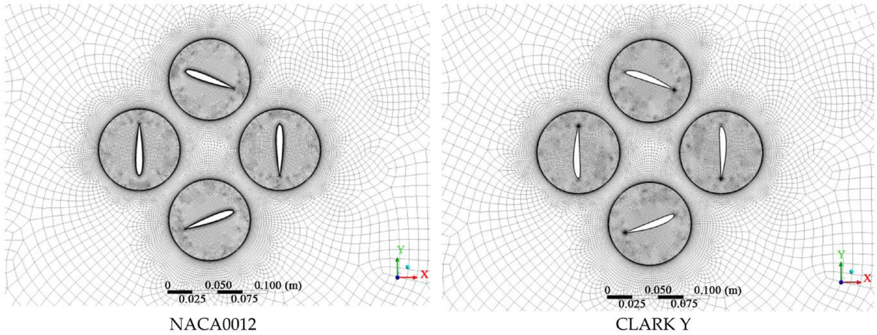

Figure 2. The part of the numerical meshes for CRF with NACA 0012 and CLARK Y blade profiles.

The transient calculations were performed with constant time step calculated on the basis of angular velocity:

$$\Delta t = \frac{2\pi/\omega}{n} \tag{1}$$

where n is the constant used for keeping the similar time step value for numerical simulations with various angular speed, ω.

An increment of the blade pitch angle in certain time step was calculated according to the relation [7]:

$$\Delta\alpha = \Delta t \cdot \omega \cdot \frac{2\alpha_0}{2\pi} \cdot \frac{\sin(\omega \cdot t + \gamma)}{|\sin(\omega \cdot t + \gamma)|} \tag{2}$$

It can be seen from relation (2) that $\Delta\alpha$ may have a positive or negative value what ensures the return of the blade pitch angle to the initial position after full rotor rotation (2π).

3. Cycloidal Rotor Fan Performance Estimation

Let us assume that useful power (P_u) is the power transmitted by the machine (fan) to the working medium (air), and it may be calculated from the relation:

$$P_u = \dot{V} \cdot \Delta p_0 \tag{3}$$

where $\dot{V}$ is a volume flow rate of the air and Δp_0 represents an increase of the total pressure between the fan outlet and inlet. For calculations of the volume flow rate for the open rotor fan, it was necessary to assume some outlet control surface, A_{out} (**Figure 3**). On this surface, the flow velocity, as well as total pressure, were averaged. It has to be emphasized that such an assumption may lead to small inaccuracy in the power determination of the analysed herein cycloidal rotor fan. Therefore, it was decided to find another way for power calculation.

Another available method for determining fan power is to calculate its mechanical power (P_m) which results from the product of torque (M) and angular velocity (ω):

$$P_m = M \cdot \omega \tag{4}$$

The torque of the cycloidal rotor fan is calculated as a sum of the torques of

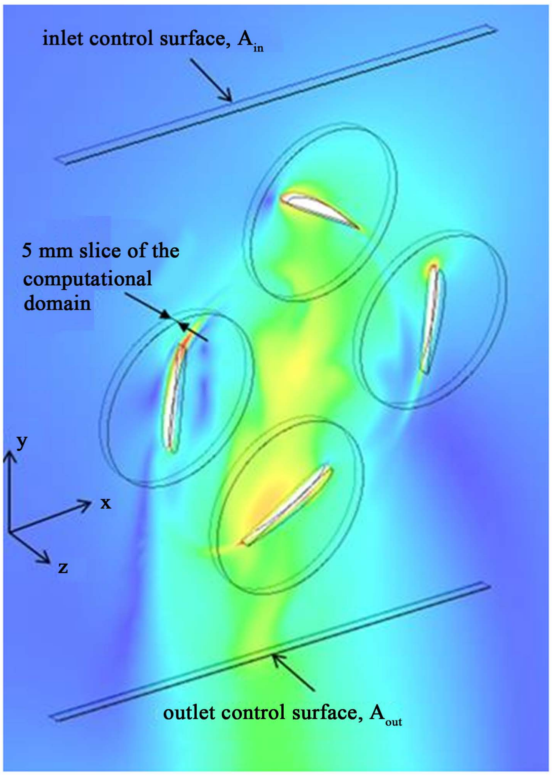

Figure 3. A part of the computational domain with marked surface assumed for determination of the volume flow rate.

the individual blades whereas, the torque of the individual blade is determined as a mean value from the full one rotation.

Because the available value of the torque in Ansys CFX CEL led to wrong results it was decided to write an own expression. As we know the torque is the cross product of a force and a length

$$\boldsymbol{M} = \boldsymbol{F} \times \boldsymbol{r} \tag{5}$$

where, $\boldsymbol{F}$ is a force vector and $\boldsymbol{r}$ is a length vector, directed from the centre of rotation to the force point. According to the assumed Cartesian coordinate system, the considered fan rotates due to the positive torque acting in the z-direction:

$$\boldsymbol{M}_z = \left(F_x \cdot y - F_y \cdot x\right) \cdot \boldsymbol{k} \tag{6}$$

where $F_x = \int \left(\tau_x + N_x p\right) \mathrm{d}A$ and accordingly $F_y = \int \left(\tau_y + N_y p\right) \mathrm{d}A$. N stands for a normal vector to the corresponding surface.

Finally, mechanical power was estimated from the quotient of the torque and required time for one full rotation of the fanmultiplied by the constant factor 2π. It had to be done in this way because all CFD simulations were carried out as a transient calculation and any instantaneous values gave wrong results.

In **Figure 4** the comparison of the useful and mechanical power determined from CFD results for CRF with NACA 0012 and CLARK Y profiles was depicted. The presented hereafter values of the power deal with the CRF of a 5 mm

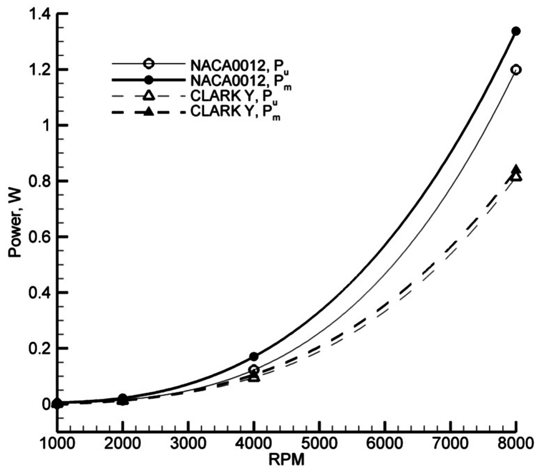

Figure 4. Comparison of the cycloidal rotor fan power for NACA0012 and CLARK Y profiles for different rotational speed, $\alpha_{max} = \pm 20^\circ$.

blade span. It means that e.g. for the CRF of 50 cm blade span its power will be approximately 100 times higher. The CFD calculations were performed for the blade pitch angle range of change ±20˚ and for the rotational speed of 1000, 2000, 4000 and 8000 RMP. The obtained results were extrapolated by using polynomials of the third order. One can see that the power generated by CRF with NACA 0012 profile is higher than the power of CRF with CLARK Y profile, and for 8000 RPM the power is even two times higher. The CRF with NACA 0012 airfoil generates the volume flow rate (power) by means of both lift and drag forces. Whereas, CRF with CLARK Y airfoil is almost purely lift-type rotary positive displacement machine. The value of mechanical power is higher than the useful power for all presented cases. However, the differences between the useful and mechanical power for CRF with NACA 0012 blade profile are much higher than for the case with CALRK Y profile. It may suggest that the CRF with NACA 0012 is less efficient.

Figure 5 shows the velocity distributions for the CRF with NACA 0012 and CLARK Y profiles for 4000 RPM. One can observe a significant flow separation around the NACA 0012 profiles, what was already observed in previous works [7] [8]. Whereas, the velocity distribution for CRF with CLARK Y profile does not repeat such behaviour, no significant separations around blades are observed. Another important difference in flow field for CRF with both blade profiles is the direction and width of the outlet stream. For CRF with CLARK Y blade profile, the flow is more vertical whereas with NACA 0012 the flow stream leans according to the direction of rotation.

The best measure of the quality of a process or a machine is an efficiency, which is often calculated as the ratio of useful output to total input. For fans, the most commonly used is the polytropic (or isentropic for compressors) efficiency. However, in the case of open rotor fans, these definitions cannot by directly applied. In the considered herein case, one can use the ratio of the calculated useful and mechanical power, what may tell us about how many energy from fan blades

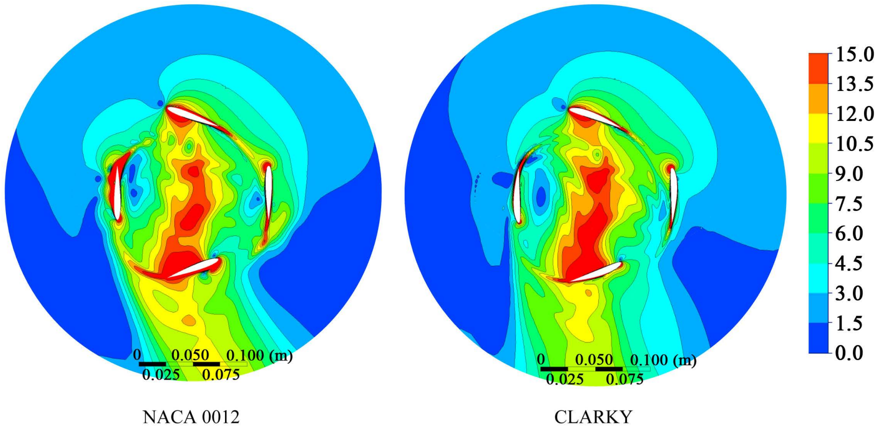

Figure 5. Comparison of the velocity magnitude field for the cycloidal rotor fan for 4000 RPM ($v = 0 \div 15$ m/s, $\Delta v = 1.5$ m/s), $\alpha_{max} = \pm 20^{\circ}$.

is given to the fluid. **Figure 6** presents the comparison of the useful and mechanical power ratio for CRF with NACA 0012 and CLARK Y blade profiles. One has to bear in mind that the determination of the useful and mechanical power on the basis of CFD results is very sensitive to the many parameters, especially for transient simulation. It can be seen that the CRF with CLARK Y blade profile is much efficient than this with NACA 0012 blade profile, even though it generates less power.

Another possibility for estimation of the CRF performance is to apply the definition of the propulsive efficiency, which is defined as follows:

$$\eta_p = \frac{2}{1 + \frac{v_{out}}{v_{in}}} \tag{7}$$

where v_{out} and v_{in} are mean velocities at the outlet and inlet, respectively, calculated on the control surfaces (**Figure 3**). The results of the calculated propulsive efficiency are included in **Figure 7**. One can conclude that also in this case the CRF with CLARK Y profile has higher efficiency for entire range of rotational speed and we may see that both characteristic curves are almost flat.

However, whereas the CRF was applied for the work mode in a duct, the propulsive efficiency rises rapidly. For the 4000 RPM for CRF with CLARK Y profile, this value rises to the 0.94, from 0.63 for the open rotor mode (**Figure 7**).

In **Figure 8** the velocity field for the CRF built-up in the straight duct is presented. It can be easily concluded, that this type of fan is preferred rather to work as a puller than as a pusher. The velocity field at the CRF outlet is much less regular than in the inlet part. Application of the CRF to the ductwork is very promising, especially for the ducts of the "very" rectangular shape, where the transition from a rectangular to circular shape is very difficult to manufacture.

CRF performance also depends on the value of the maximal blade pitch angle, α_{max}. It means that the air volume flow rate can be regulated by both rotational speed RPM and α_{max}. **Figure 9** shows the velocity field comparison for CRF with

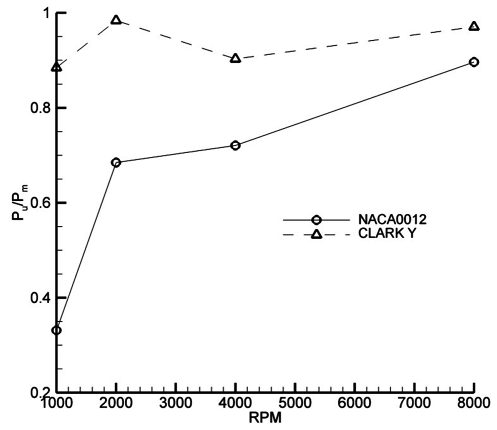

Figure 6. Comparison of the useful and mechanical power ratio for CRF with NACA0012 and CLARK Y blade profiles, $\alpha_{max} = \pm 20^{\circ}$.

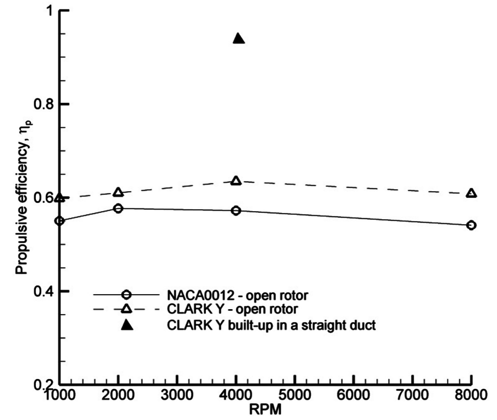

Figure 7. Comparison of the propulsive efficiency of the CRF with NACA0012 and CLARK Y blade profiles, $\alpha_{max} = \pm 20^{\circ}$.

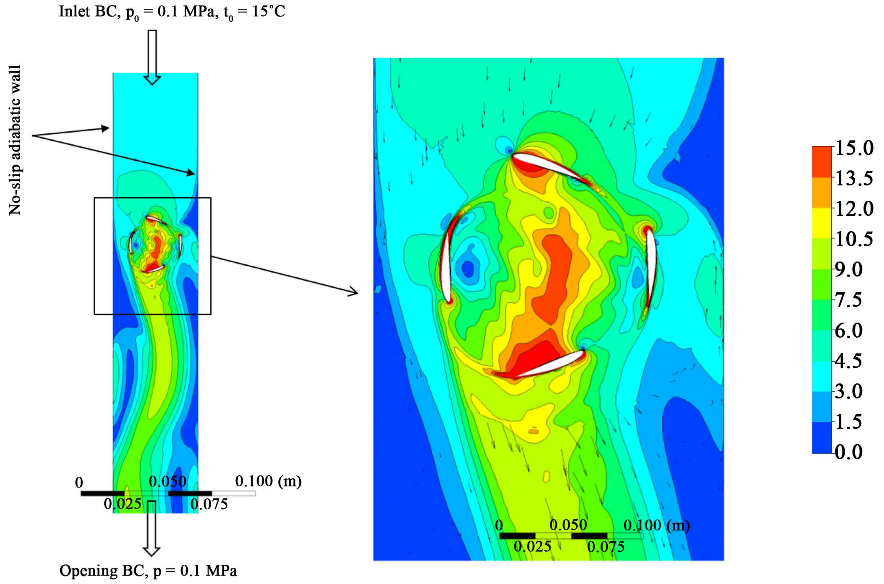

Figure 8. Velocity magnitude field for the cycloidal rotor fan with CLARK Y profile for 4000 RPM working in the straight duct ($v = 0 \div 15$ m/s, $\Delta v = 1.5$ m/s), $\alpha_{max} = \pm 20^{\circ}$.

CLARK Y profile for rotational speed 4000 RPM with the maximal blade pitch angle 15˚, 20˚, 25˚ and 30˚. The outlet velocity from CRF significantly increases together with the increase of maximal blade pitch angle.

In **Figure 10** the CRF with CLARK Y profile performance parameters for different values of maximal blade pitch angle was presented. We may see that by

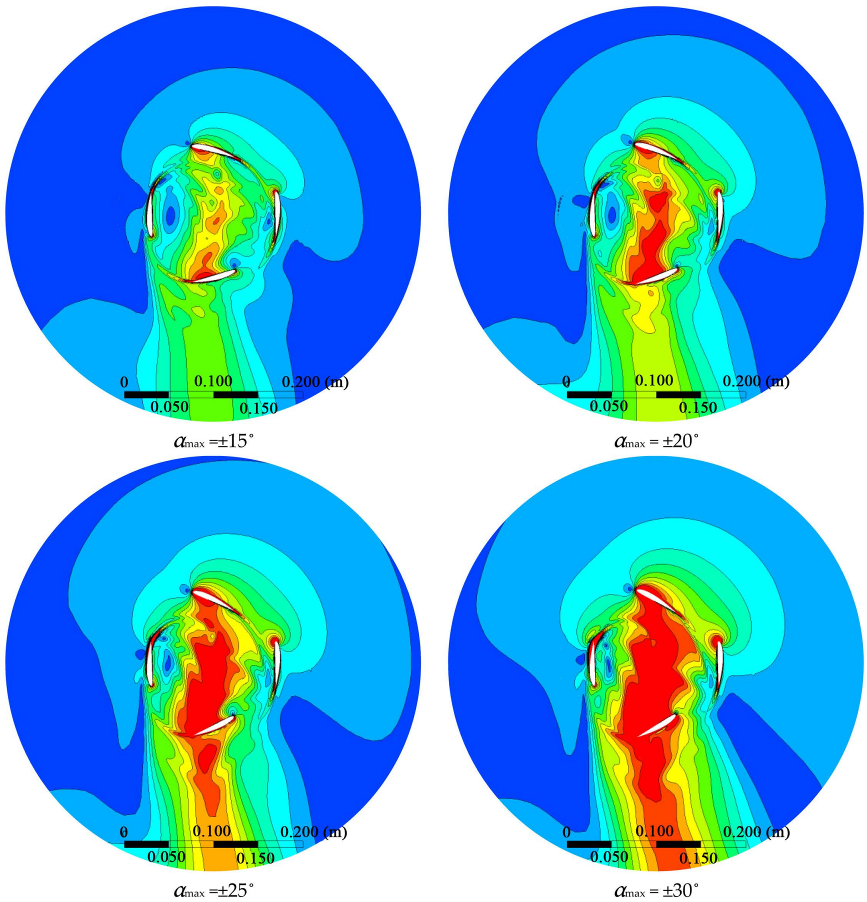

Figure 9. Velocity magnitude field for the cycloidal rotor fan with CLARK Y profile for 4000 RPM and different α_{max} ($v = 0 \div 15$ m/s, $\Delta v = 1.5$ m/s).

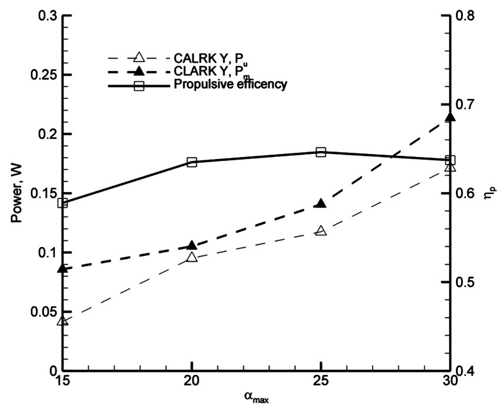

Figure 10. Comparison of the useful, mechanical power and propulsive efficiency for CRF with CLARK Y blade profile for 4000 RPM and different α_{max}.

doubling the value of blade pitch angle we may increase the power of the CRF about two times.

The characteristic of the propulsive efficiency vs. maximal blade pitch angle is in this case almost flat, similarly, like for the variable value of rotational speed.

4. Conclusions

The numerical analysis of the flow in cycloidal rotor fan (CRF) for two shapes of the blades, with NACA0012 and CLARK Y profiles, was performed by means of Ansys CFX CFD software. The transient analysis was made for four rotational speeds and values of maximal blade pitch angle in order to determine CRF performance characteristic. On the basis of the carried out CFD simulations, the following conclusion may be drawn:

- By estimation of the torques and forces in Ansys CFX, the implementation of the own expressions is recommended.
- It was extremely difficult to obtain quantitative information from CFD calculations of the flow in cycloidal rotor fan.
- Definition of the useful and mechanical power ratio may be used for estimation of the CRF performance as well as the propulsive efficiency, and be an alternative for polytropic and isentropic efficiency.
- The shape of the blade profile affects significantly the performance characteristic of a cycloidal rotor fan. The CRF with CLARK Y blade profile works more as a lift-type rotary positive displacement machine, which provides it with higher efficiency.
- The useful power calculated for the CRF with open rotor and for CRF build-up in straight ducts differs from each other less than 3%, for the same working conditions.
- The CRF may efficiently work as a puller fan for ducts of the rectangular shape of the cross-section. The proposed design of CRF is well scalable to arbitrary duct cross-section, especially for the cases of large ratio of width to height of the cross-section.
- The higher value of the maximal blade pitch angle the higher power, but the choice of α_{max} should take into account the maximum efficiency of CRF.
- For the cycloidal rotor fan (CRF), all three laws valid for dynamic fans are preserved, *i.e.*

$$\frac{\dot{V}_2}{\dot{V}_1} \cong \frac{RPM_2}{RPM_1},\ \frac{\Delta p_2}{\Delta p_1} \cong \left(\frac{RPM_2}{RPM_1}\right)^2,\ \frac{P_2}{P_1} \cong \left(\frac{RPM_2}{RPM_1}\right)^3,$$

but one can constitute an additional ones relating to CRF radius and a blade pitch angle

$$\frac{\dot{V}_2}{\dot{V}_1} \cong \left(\frac{R_2}{R_1}\right)^2,\ \frac{P_2}{P_1} \cong \frac{\alpha_{\max,2}}{\alpha_{\max,1}}.$$

Presented research in this paper, as well as the previous numerical and experimental investigations [2] [3], allow drawing a conclusion that proposed CRF design with four blades of CLARK Y profile and the blade chord to rotor radius ratio of 5/7 (what provides a compact and rigid structure) has a good performance in the application as an open rotor and as the rotor in a ductwork.

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Conflicts of Interest

The authors declare no conflicts of interest regarding the publication of this paper.

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