Outlet Surface Area Influence in Spiral Casing Design on Centrifugal Fan Performance

Ardit Gjeta, and Lorenc Malka

Abstract—In this paper, the effect of the outlet surface area of the spiral casing on the performance of a centrifugal fan was investigated using open source CFD software OpenFOAM [1]. An automated loop with RANS and data post-processing is set up using Matlab, simulating and considering a large number of parameter variations. The effect was analyzed as a function of total pressure loss, static pressure recovery coefficient and on total efficiency as well.

Index Terms—Efficiency, Outlet Surface Area, Pressure Recovery, Spiral Casing.

I. INTRODUCTION

A centrifugal fan is widely used machinery in the industry which consists of an impeller in a spiral casing as a typical piece of turbo-machinery that converts external mechanical energy into pressure and kinetic energy of the working fluid. An impeller is a mechanical device constructed of a number of vanes that supplies mechanical energy to the fluid and is a key component of the fan. Therefore, many studies have focused intensively on impellers. Fluids obtain energy from the impeller and it is discharged through the spiral casing. Currently, the minimization of energy loss is dependent on the characteristics of the spiral casing. Research on the spiral casing has drawn relatively little attention, but in order to improve the performance of centrifugal fans to an acceptable level, a study of the characteristics of the spiral casing is absolutely needed.

II. SPIRAL CASING DESIGN METHOD

Constant circulation method [2], [3], is technically applied by drawing a spiral case based on the fact that velocity circulation is a constant \( r_c u = \text{constant} \). In practice, this rule is valid considering that one spiral must be so far displaced from the impeller that deflections conditioned by the consideration of a finite number of vanes can be ignored. This rule constitutes the basis for the dimensioning of a volute in the cases where friction has been ignored. The velocity \( c \) at an arbitrary place can be calculated from its components \( c_m \) and \( c_u \): \( r_c u = r_2 c_m \). From the condition that the same volume-flow must flow (the continuity equation) through all the streamline in volute it gives the correlation:

\[
Q = 2\pi r_2 b_2 c_m = 2\pi r B c_m
\]  

Mass flow rate balance gives: \( r_2 b_2 c_m = r B c_m \), the following inclination \( \alpha \) of the streamlines:

\[
tg(\alpha) = \frac{c_m}{c_u} = \frac{c_m b_2}{c_u B}
\]

Because we obtain the boundary of the volute from the streamline, again it yields, \( \frac{dr}{r} = d\varphi \ t g(\alpha) = d\varphi \ t g(\alpha_2) \frac{b_2}{B} \)

The solution states,

\[
ln \frac{r}{r_2} = \varphi \ t g(\alpha_2) = \varphi \ t g(\alpha_2) \frac{b_2}{B}
\]

Accordingly, the trajectory of fluid particles in the volute is as follows (Carolus 2013) [3],

\[
r(\varphi) = r_2 e^{\varphi tg(\alpha)} = r_2 e^{\varphi tg(\alpha_2) \frac{b_2}{B}}
\]

\( r(\varphi) \) is the radius of the volute at an angle \( \varphi \), \( r_2 \), is the outer radius of the impeller that is equal to 150mm in our case.

\( \alpha \), is the angle that the absolute velocity vector makes with the peripheral direction \( tg(\alpha) = c_m/c_u \), \( b_2 \), outlet width of impeller and \( B \), the width of volute.

The effect of the outlet surface area on the spiral casing is the main focus of this paper. In our previous study [4], [5], the influence of geometric parameters is given separately and as a function of clearance gap \( s_z \). Considering the design of the spiral casing (eq.5) outlet surface area is a function of independent parameters: logarithmic spiral, \( \alpha \), relative width of spiral casing \( B/b_2 \), and insignificantly from the radius \( r_z \) and angle of the spiral casing \( \varphi_z \)
III. OPTIMAL IMPELLER

Starting point are optimal impellers for the whole range of specific speeds. Following a current and nearly finished study on aerodynamic optimization of centrifugal fan impellers using CFD-trained meta-models [6], where a method for optimization of impellers of the whole class of centrifugal fans has been developed. For the first simulations it is accepted one optimized impeller (VAL-1) with the design flow coefficient $\phi = 0.12$, which correspond to flow rate of $Q = 0.4m^3/s$, since the diameter of the impeller is $D_2 = 0.3m$ and the rotational speed is $n = 3000$ rpm. The detailed flow field at the impeller’s outlet from preceding RANS simulations is used as boundary conditions for a RANS of the flow in the volutes.

IV. PERFORMANCE OF THE VOLUTE

The overall performance of the volute can be analyzed by using [7]:

Total pressure loss coefficient of volute:

$$K_p = \frac{p_{t2} - p_{t3}}{p_{t2} - p_2} = \frac{p_{t2} - p_{t3}}{\rho \frac{c_2^2}{2}}$$ (6)

$K_p$, is defined as the ratio between the total pressure losses in the volute to the dynamic pressure at the impeller exit.

Static pressure recovery coefficient of volute:

$$C_p = \frac{p_3 - p_2}{p_{t2} - p_2} = \frac{p_3 - p_2}{\rho \frac{c_2^2}{2}}$$ (7)

$C_p$, is defined as the ratio between the static pressure recovered in the volute to the dynamic pressure at the impeller exit.

Total efficiency of volute

$$\eta_v = \frac{p_{t3}}{p_{t2}}$$ (8)

From equation (6, 7) becomes:

$$C_p = 1 - K_p - \left(\frac{c_3}{c_2}\right)^2$$ (9)

$\left(\frac{c_3}{c_2}\right)^2$, is the ratio of volute outlet/inlet kinetic energy.

Maximizing performance of a complete fan requires, finding $c_{3, opt}$. $K_{p, opt} = K_{p, min}, C_{p, opt} = C_{p, max}$ from CFD as a function of outlet surface area of volute (function of alpha spiral angle and relative width scale $B/b_2$).

V. NUMERICAL ANALYSIS

Three-dimensional, incompressible, steady-state flow simulations were performed using the Open Source CFD software, OpenFOAM v3.0.x [1]. This solves discretized forms of the Reynolds-averaged Navier–Stokes equations for turbulent flow using the finite volume method (Ferziger, Perić 2002) [8].

The unstructured grid solution procedure is based on a variant of the SIMPLE pressure correction technique (Patankar 1980) [9]. The iterative solution was deemed to be converged when the normalized absolute error over the mesh had reduced to $10^{-5}$ for each variable. OpenFOAM supports the standard $k-\omega$ model by Wilcox (1998) [10], and Menter’s SST $k-\omega$ model (1994) [11]. The $k-\omega$ SST turbulence model was employed for these calculations, with near-wall conditions supplied by the ‘wall function’ conditions of Launder and Spalding, 1974 [12].

VI. BOUNDARY AND INITIAL CONDITIONS

The inflow boundary conditions were based on known flow rates and the flow direction [6]. Nonuniform velocity profiles were prescribed at the volute inlet (fan impeller outlet) by implementing radial and tangential velocity components, also axial velocity is included. The front and backside of the impeller as the rotating wall, the other parts wall with the no-slip condition and for the outlet ambient pressure is used. Turbulent kinetic energy is $k = 3m^2s^{-2}$, and the specific turbulence dissipation rate is $\omega = 4000s^{-1}$.

The geometry of volutes is generated from MATLAB as stereolithography (.stl file) than cfMesh v1.1.2 software is used to create a mesh. The grid resolution is made according to $y^+$ value $30 < y^+ < 200$.

VII. CFD SIMULATION RESULTS

The parameters to be modified in this study are alpha spiral angle, $\alpha = 12 + 20^\circ$ and the relative width of the spiral casing $B/b_2 = 1.4 + 3.0$, see Fig. 1. For the same alpha spiral angle as the width scale increases, the outlet surface area decreases at the exit of the spiral casing, so the values of the width scale are increasing from right to left.

![Outlet Surface Area vs. Total Efficiency](image)

Fig. 2 Total efficiency of the spiral casing as a function of outlet surface area

Total to total efficiency of volutes as a function of outlet surface area is shown in Figure 2 for each of the alpha spiral angles. As a result, maximum efficiency is obtained for smaller alpha values, corresponding to a compact volute. In the case of the larger one's efficiency is decreased, due to the increased surface area of the walls.

Referring to the angle of the logarithmic spiral, with increasing width scale values $B/b_2 = 1.4 + 2.6$, a slight
increase in efficiency is observed. This increase is observed for almost all alpha angles but with further increase for width scale \(B/b_2 = 2.8 \div 3.0\), there is a slightly decrease in efficiency.

Based on the numerical results for alpha spiral angle 12°, 14° and 16°, for width scale \(B/b_2 < 1.8\), with the further decrease of the outlet surface, a very small change in efficiency is observed.

Referring to the outlet surface area value between 0.029-0.033m², it is recommended to use alpha spiral angle 18° with higher values of width scale \(B/b_2 > 2.6\), rather than angle 16° with lower width scale \(B/b_2 < 1.8\), although for the same outlet surface value.

Static pressure recovery coefficient as a function of outlet surface area is shown in Figure 3 for each of alpha spiral angle.

![Fig. 3 Static pressure recovery coefficient as a function of outlet surface area](image)

By increasing the outlet surface see graph no.3 it is also clearly observed a significant increase in values of static pressure recovery coefficient as well. For alpha spiral angle 12° and partially for the alpha 14°, no positive value of static pressure recovery coefficient is observed. For these alpha angles, with increasing width scale \(B/b_2\), is observed a significant decrease in the static pressure recovery coefficient. While with the further increase of the alpha spiral angle, there is a continuous slightly decrease in the static pressure recovery coefficient. By comparing the results for alpha spiral angle 16° and 18°, for the same value of the outlet surface area, it is observed that the higher value of the static pressure recovery coefficient is provided for higher alpha spiral angle and higher values of width scale, a similar result for efficiency too.

![Fig. 4 Total pressure loss coefficient as a function of outlet surface area](image)

Since for the total pressure loss coefficient, a value as small as possible is required, the recommended alpha spiral angle values are 18° and 20°. While for small value of alpha spiral angle (12° + 16°) and for width scale \(B/b_2 > 2.6\), high value of total pressure loss coefficient is observed. Even for this parameter, it is recommended to use increased values of alpha spiral angle and high values of width scale \(B/b_2\), for the same value of the outlet surface area.

Another parameter called the ratio of the outlet/inlet kinetic energy ratio \((c_3/c_2)^2\), must be considered, where it must be ensured that this ratio is less than 1.

![Fig. 5 Outlet/Inlet kinetic energy ratio as a function of outlet surface area](image)

In the depicted graph above is shown that for the alpha spiral angle 12°, the ratio \((c_3/c_2)^2\) is around 1, and in this case, it is not recommended, even though it has the maximum value of efficiency. While for a higher value of alpha, the outlet/inlet kinetic ratio is below 1.
VIII. CONCLUSIONS

From the results of numerical simulations, the following conclusions are achieved:

a) Smaller volutes have higher efficiency, but there is no pressure recovery. The insufficient increase of the radius of the spiral casing cross-section causes a fluid acceleration and partially "destroys" the static pressure generated in the diffuser, therefore it is recommended that the values of the logarithmic spiral angle (alpha), to be greater than 10°, respectively in the range 14° ÷ 20°.

b) Positive static pressure recovery can be carried out considering alpha spiral angle over 14° and outlet/inlet ratio of the kinetic energy should be lower than 1 which states for alpha spiral angles over 12°.

c) The internal flow distribution could be improved by selecting, higher values of alpha spiral angle and width scale \( B/b_2 \) assuming constant outlet surface area. From 8, 9 it is shown the influence of outlet surface area on flow distribution. For small values of relative width scale \( B/b_2 \) the velocity profile distribution \( U \) varies from 3.8 ÷ 19 \( ms^{-1} \), while for higher values of width scale \( B/b_2 \) the velocity profile distribution \( U \) varies from 4.9 ÷ 14 \( ms^{-1} \).

So finally, the second outlet surface area shape performed better (see fig. 9).

**Fig. 6** Velocity magnitude streamlines inside spiral casing for \( \alpha = 16°, B/b_2 = 1.6 \)

**Fig. 7** Velocity magnitude streamlines inside spiral casing for \( \alpha = 18°, B/b_2 = 2.6 \)

**Fig. 8** Outlet velocity magnitude for \( \alpha = 16°, B/b_2 = 1.6 \)

**Fig. 9** Outlet velocity magnitude for \( \alpha = 18°, B/b_2 = 2.6 \)

**NOMENCLATURE**

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**ABBREVIATIONS**

- CFD: Computational Fluid Dynamics
- RANS: Reynolds Averaged Navier-Stokes
- SST: Shear stress transport
- FOAM: Field Operation And Manipulation
REFERENCES


Ardit GJETA was born in Peshkopi, Albania in 1988. He received his M.Sc. degrees in Mechanical Engineering from the Polytechnic University of Tirana in 2012.

Since 2013, he joined the Polytechnic University of Tirana as a lecturer in the energy department, where he teaches subjects like thermodynamics and fluid machines. He is currently following the Ph.D. studies.

He recently has published several articles in national and international conferences. He is continuing research on the mechanic fields, especially Turbo Machinery, CFD, CFTurbo and OpenFOAM.