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SPIRAL CASING DESIGN PARAMETERS OF CENTRIFUGAL FAN PERFORMANCE WITH THE AIM OF INCREASING ENERGY EFFICIENCY AND REDUCTION OF CO2 EMISSIONS

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ABSTRACT: Industrial fans are subject to European Union energy labelling and Ecodesign requirements. The cost-efficient improvement potential through design is about 34 TWh per year, which corresponds to 16 Mt of CO2 emissions. Ecodesign requirements for industrial fans are mandatory for all manufacturers and suppliers wishing to sell their products in the EU.
In this paper, the effect of the diffusor angle and width-scale parameter of the spiral casing on the performance of the centrifugal fan was investigated using the CFD software OpenFOAM [1]. Considering a large number of parameter variations, the simulation results are performed by an automated loop in MATLAB which enables the processing of a considerable amount of data in a short time.
The performance characteristics studied were efficiency, static pressure recovery coefficient, total pressure loss coefficient, as well as graphical examples that show the pressure fields and fluid speed in different parts of the volute.
Keywords: *centrifugal fan, volute, diffuser, OpenFOAM, efficiency, CO2 emissions*

1.1. 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. Many articles related to centrifugal fans have studied and optimized only the fan impeller and some of them treat the fan as a whole unit, while the study of the spiral casing is less well-known. 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 needed. For this reason, this study of the effect of the diffuser angle and width-scale has been conducted, which should lead to better advanced recommendations for the shape of the spiral casing.

1.2. Volute Shape Design Method

Constant circulation method [2] is a method applied by drawing a spiral case based on the fact that velocity circulation is a constant $rc_u = constant$. In practice, this rule is valid with the restriction that one spiral must be so far displaced from the impeller that deflections conditioned by the consideration of a finite number of blades 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, $rc_u = r_2 c_{u2}$. From the condition that the same volume-flow must flow (the continuity equation) through all the streamline in volute it gives the correlation:

$$Q = 2\pi r_2 b_2 c_{m2} = 2\pi r B c_m$$

From which follows $r_2 b_2 c_{m2} = r B c_m$, by arranging terms in the equation, we obtain the following inclination α of the streamlines:

$$tg(\alpha) = \frac{c_m}{c_u} = \frac{c_{m2}}{c_{u2}}\frac{b_2}{B}$$

Because we obtain the boundary of the volute from the streamline, again it yields, $tg(\alpha) = \frac{dr}{rd\varphi}$;

$$\frac{dr}{r} = d\varphi\ tg(\alpha) = d\varphi\ tg(\alpha_2)\frac{b_2}{B}$$

Finally, the solution states,

$$ln\frac{r}{r_2} = \varphi\ tg(\alpha_2)\frac{b_2}{B} = \varphi\frac{c_{m2}}{c_{u2}}\frac{b_2}{B}$$

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

$$r_{(\varphi)} = r_2 e^{\varphi tg(\alpha)} = r_2 e^{\varphi tg(\alpha_2)\frac{b_2}{B}}$$

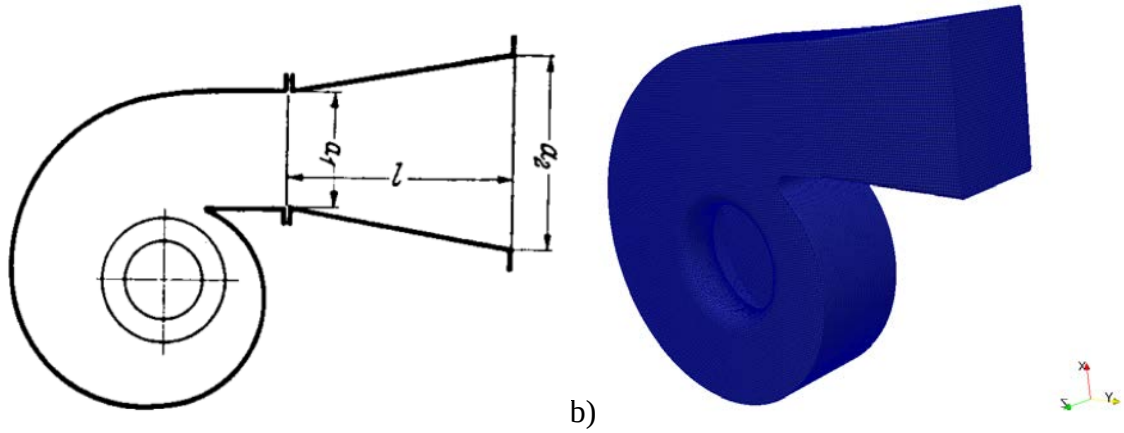
$r_{(\varphi)}$, is the radius of the volute at an angle φ,

r_2, is the outer radius of the impeller that is equal to 150mm in our case

α is the angle that the absolute velocity vector makes with the peripheral direction $tg(\alpha) = c_m/c_u$.

b_2, the width of outlet impeller; B, the width of volute

The spiral casing should be designed according to the rules discussed above, in order to avoid adverse effects on the rotor or undesirable effects on the spiral casing [4], [5], [6], [7]. However, to increase the performance of the volute, an additional diffuser can be installed as shown in Figure 1.



a) b)

Figure 1.a) Improving the spiral casing by adding diffuser to the end (Eck 1973) [2]
b) The geometry of spiral casing with diffuser after the meshing process.

This would continuously reduce the velocity of the air at the outlet as the cross-sectional area gradually increases. Similar but very large diffusers are widely used in mine ventilation system. Research on diffusers has focused on Reynolds small numbers. This indicates that, in certain circumstances, it is advisable to increase the area immediately after a short period of conical growth.

1.3. Matlab GUI for spiral casing design including diffuser

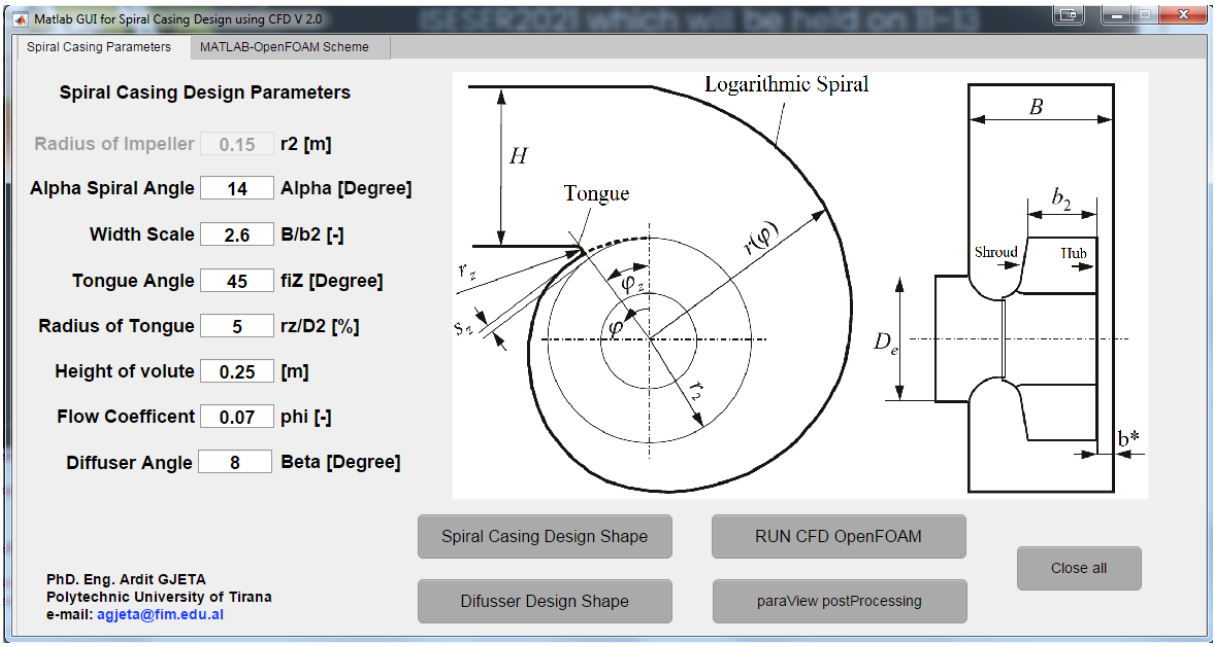


Figure 1. MATLAB graphical user interface for spiral casing design including diffuser [8]

1.4. Effect of diffuser angle

In this case study the parameters to modified is angle of diffuser. The geometrical parameters such as the angle of volute tongue $\varphi = 45^0$ and radius tongue $\frac{r_z}{D_z} = 5\%$ remains unchanged. The study takes into consideration the values of the diffusor angle from 2° up to 8°.

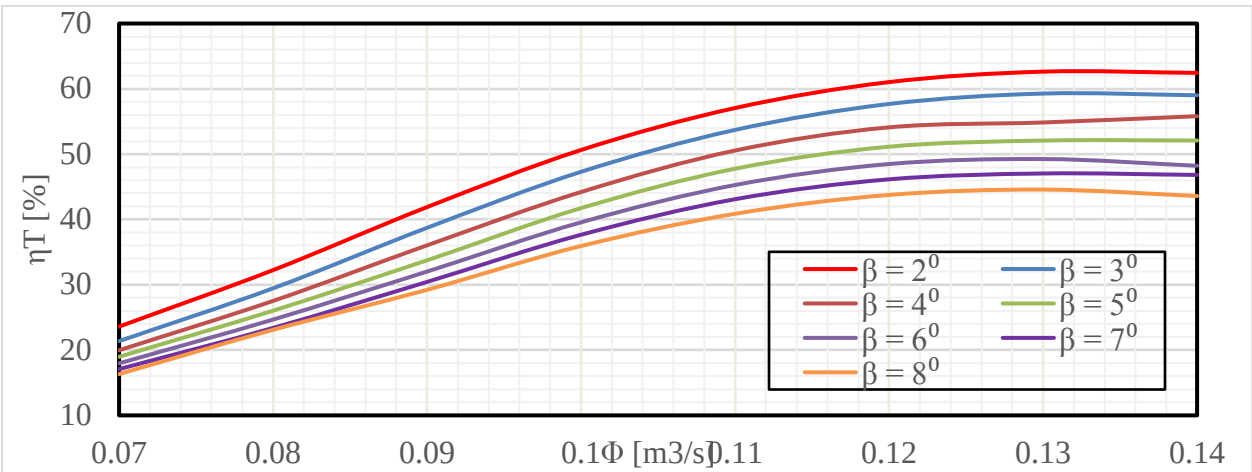


Figure 2. Total efficiency of the spiral casing as a function of flow coefficient

The simulation results shows that the maximum efficiency is obtained for a small number of diffuser angle and a high value of flow coefficient. Specifically, for a diffuser angel $\beta = 2^\circ$ the total efficiency reaches maximum value. Since the diffuser opening angle is small, this would result in a higher c_3 velocity. Also, for a small diffuser angle the average pressure value would result in an increased value. By increasing the diffuser angle, it is noticed that the total

efficiency decreases significantly. Specifically, for $\beta = 8^{\circ}$ the results shows that the total efficiency has the lowest value.

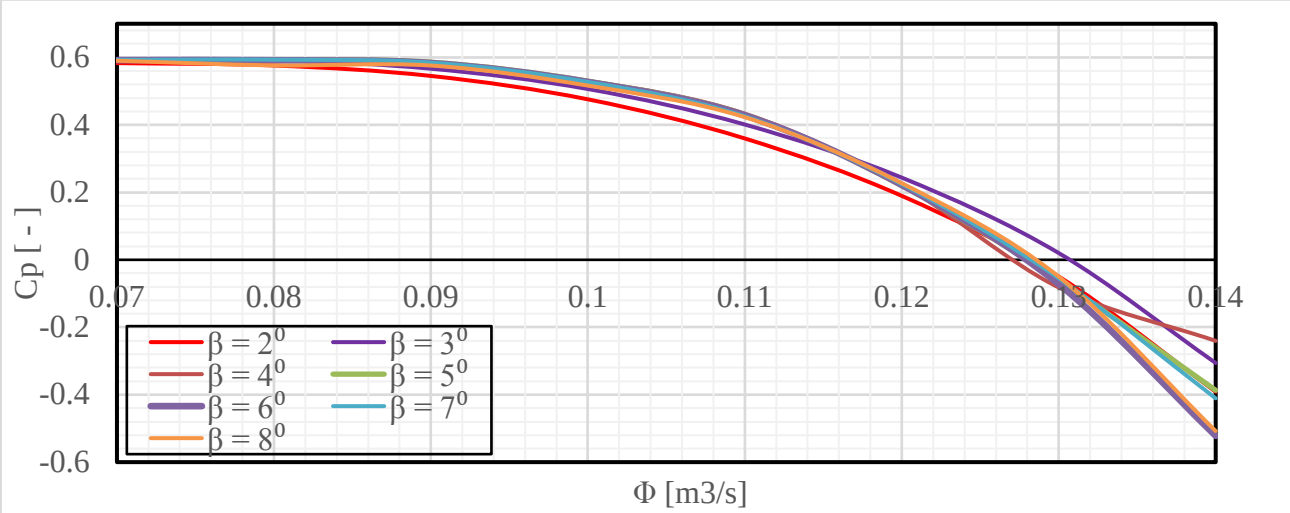


Figure 3. Static pressure recovery coefficient as a function of flow coefficient

The graph results show that for the all diffuser angles the static pressure recovery coefficient values have slight changes. For $\beta = 2^{\circ}$ it is noticed with increase of flow coefficient value the static pressure recovery coefficient significantly decreases. It should be noted that for negative values of the static pressure recovery coefficient results in incorrect volute design. As result we say that different diffuser angles do not have a significant impact on the static pressure recovery coefficient. As we notice in the graph between the values $\beta = 2^{\circ}$ and $\beta = 8^{\circ}$ with increasing the flow coefficient, the static pressure recovery values have a small difference.

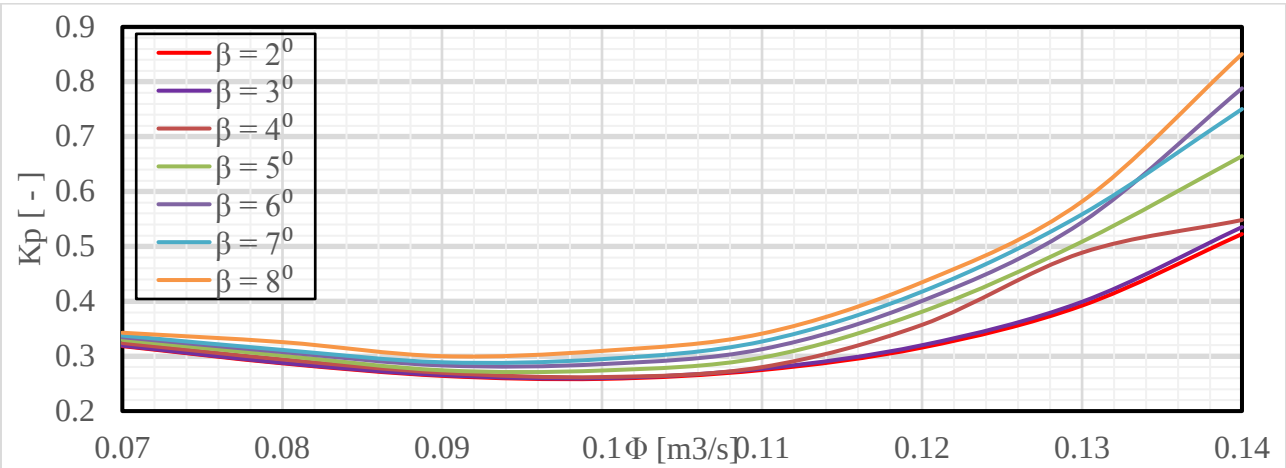
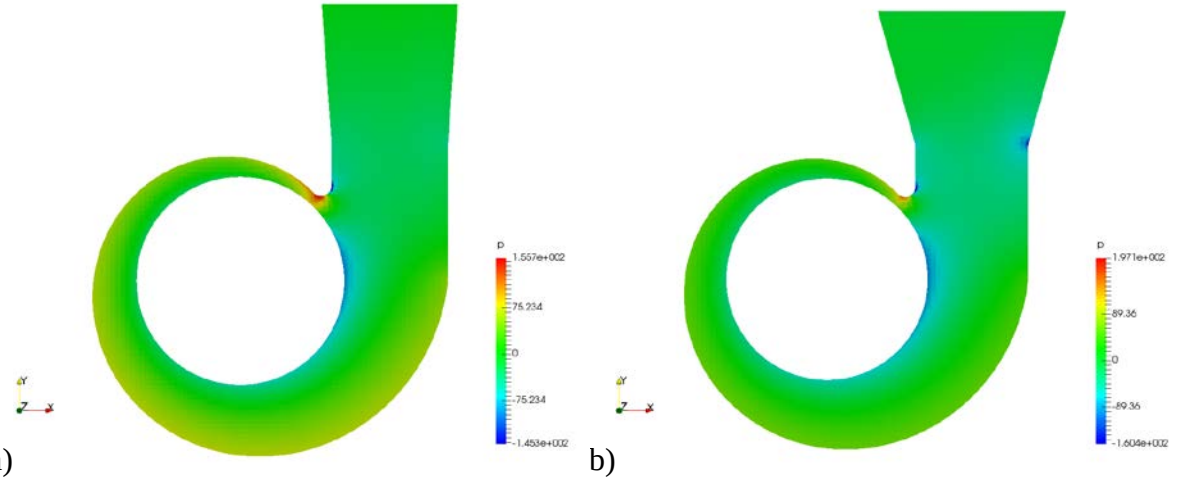


Figure 4. Total pressure loss coefficient as a function of flow coefficient

It is noticed that with the increase of the flow coefficient, the total pressure loss coefficient is increased. For a higher flow rate the total pressure loss coefficient will have a higher value. The graph shows that for $\beta = 2^{\circ}$ the total pressure coefficient has the smallest values compare to the other angles. The recommended working points for flow coefficients is from 0.09 to 0.11.



a) b)

Figure 5. Pressure magnitude field for diffuser angle a) $\beta = 2^{\circ}$and $\beta = 8^{\circ}$

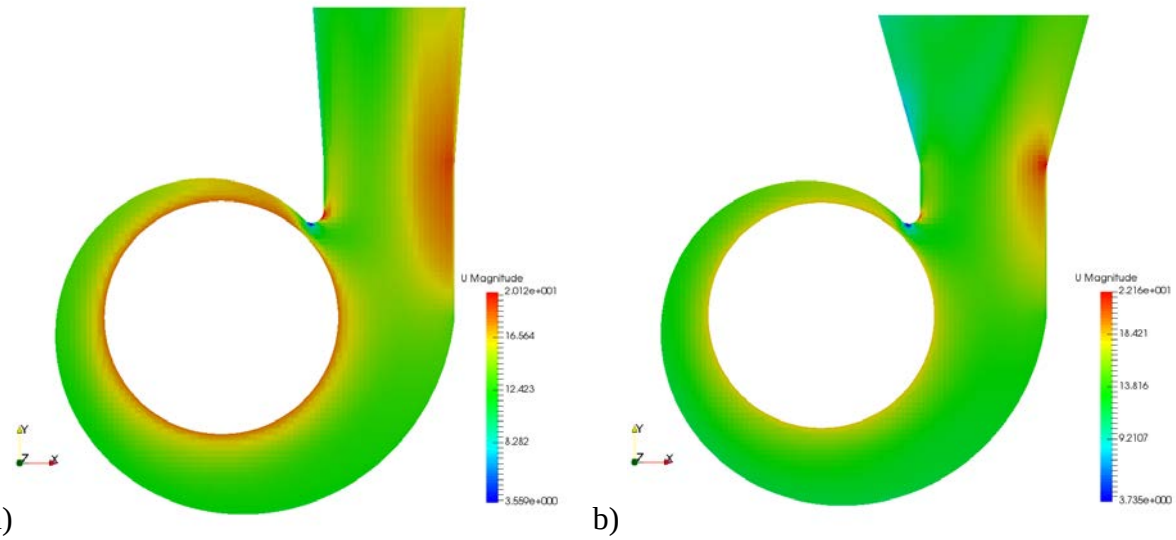


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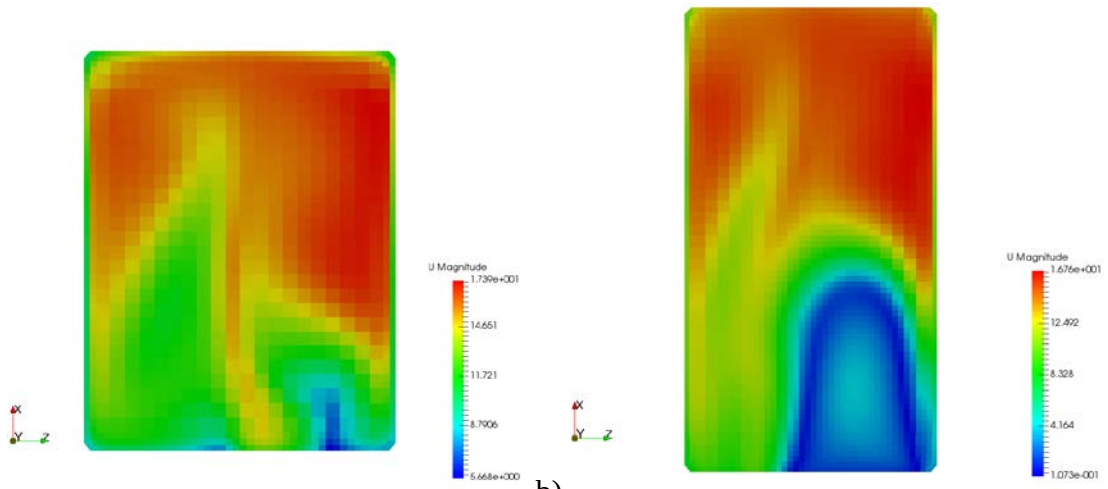
Figure 6. Velocity magnitude field for diffuser angle a) $\beta = 2^{\circ}$and $\beta = 8^{\circ}$



a) b)

Figure 7. Outlet velocity magnitude for diffuser angle a) $\beta = 2^{\circ}$and $\beta = 8^{\circ}$

1.5. Effect of width-scale parameter B/b2

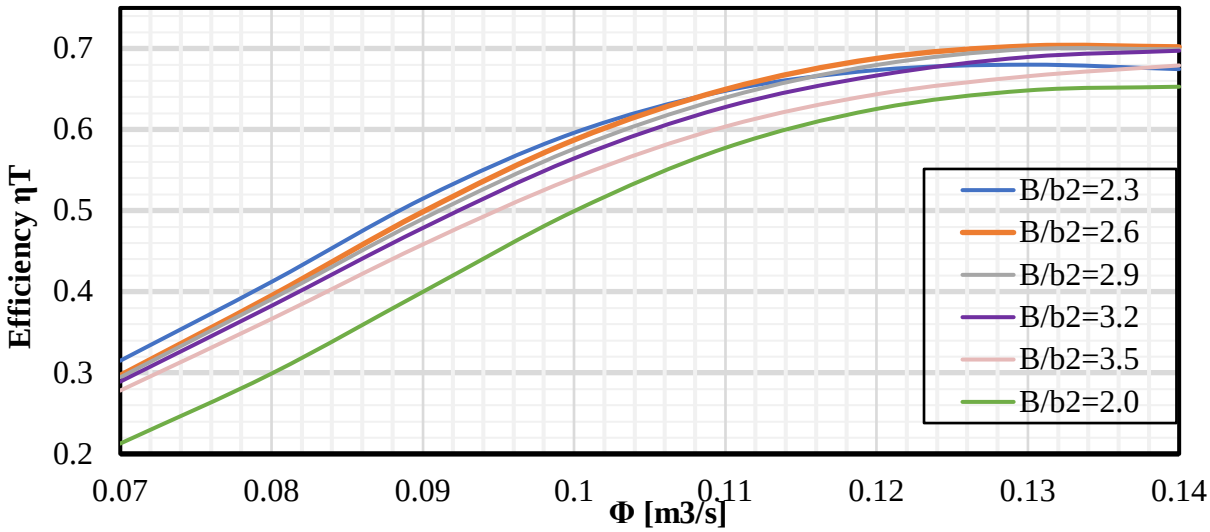


Figure 8. Total efficiency of the spiral casing as a function of B/b_2.

From this graph we can see that for the small values of B/b_2 the efficiency of the spiral casing is lowest compared to the other values. The line with the highest efficiency is for B/b_2=2.6 showing a maximal efficiency of barely past 70.2%. B/b_2=2.0 exhibits the lowest efficiency of all other values. Higher values of B/b_2 would cause an increase in size of the volute compared to the rotor thus lowering the pressure in the volute outlet.

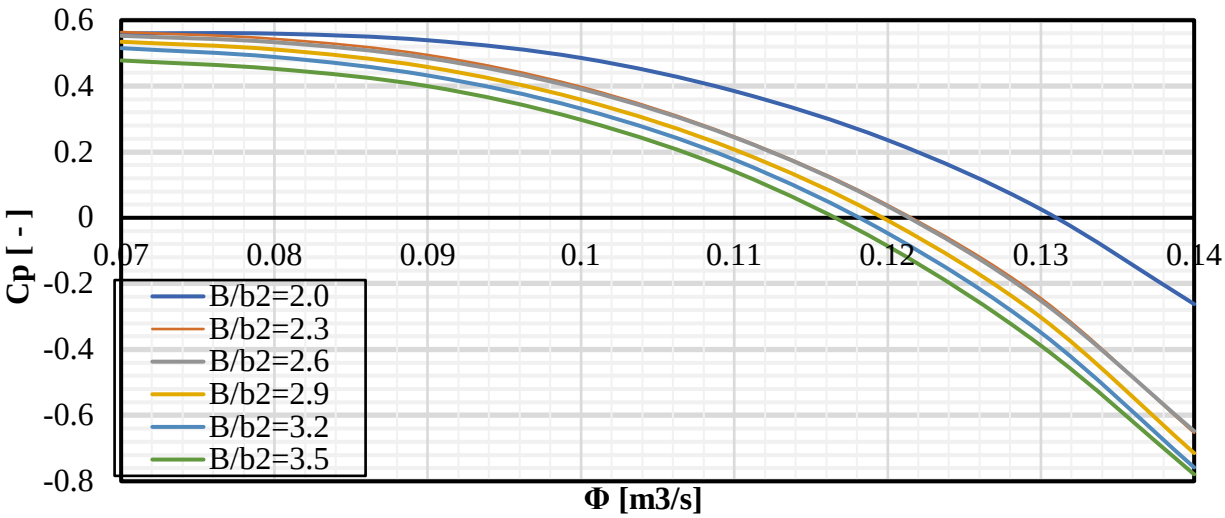


Figure 9. Static pressure recovery coefficient [$\mathbf{C_p}$] as a function of B/b_2

The second parameter that values the performance of the volute is the static pressure recovery coefficient $\mathbf{C_p}$. As we can see from the graph, the higher the B/b_2 the lower this coefficient gets, whilst for lower values of B/b_2 this coefficient takes its maximal values. Judging the graph, we could say that for values of B/b_2 above 2.0, the rate of static pressure drop gets small as B/b_2 values get higher thus having smaller and smaller effect in the decrease of the static pressure recovery coefficient. We can also notice that for values of $B/b_2 \geq 2.3$, the values of the static pressure recovery coefficient change very little.

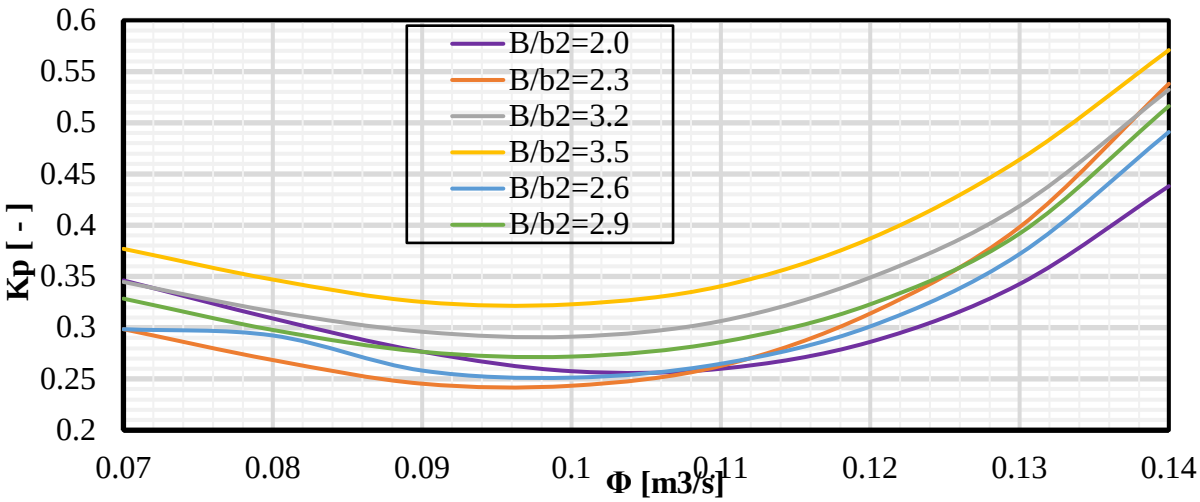


Figure 10. Total pressure loss coefficient [$\mathbf{K_p}$] as a function of B/b_2

From the graph we notice that the total pressure loss coefficient [$\mathbf{K_p}$], decreases in the average values of flow coefficient and then from the value of *0.1*, the values of the total pressure loss coefficient increase. The total pressure loss coefficient is function of a certain geometric parameters $K_{p_{opt}} = K_{p_{min}} = f(\alpha, \frac{B}{b_2}, \phi_z, \varphi_z, etc.)$ and its values should be as low as possible. For low to average values of **Φ**, the values of this coefficient are lowest for B/b_2=2.3. Past the average values of **Φ**, the lowest values of the total pressure loss coefficient are for B/b_2=2.0. From the graph we can see an increase of the volute outlet speed which increase almost linearly with the increase of **Φ**. Only for values of B/b_2=2.0 the speed is distinctly lower while for the other values of B/b_2, the values of speed are very close to each other.

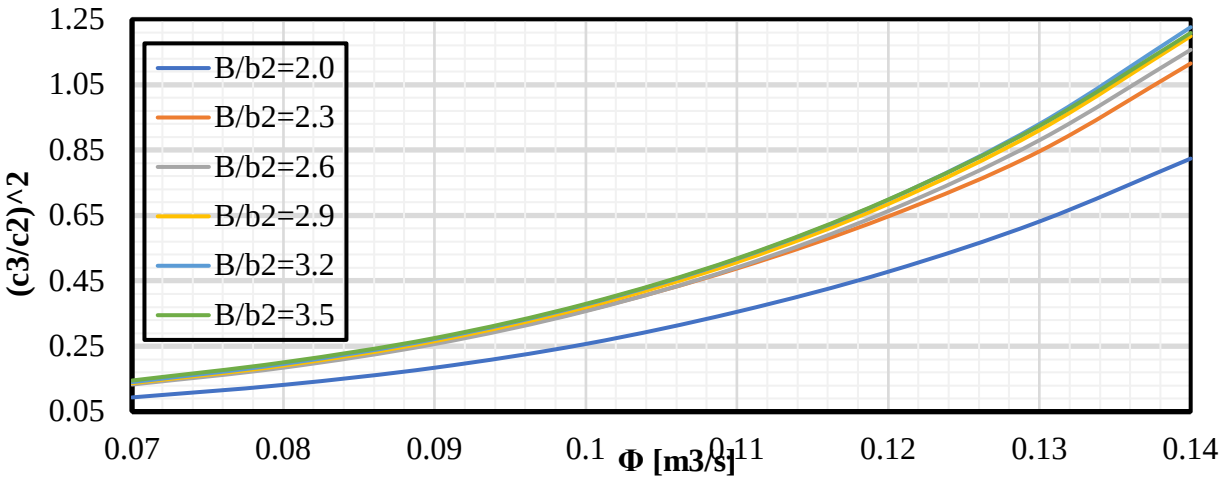


Figure 11. *K*inetic energy ratio in the outlet of the volute and outlet of rotor [$\mathbf{c_3/c_2^2}$] as a function of B/b_2

From this graph we notice that for higher values of B/b_2 we have an increase in the values of this coefficient which gets stronger in the higher values of **Φ**. Values of this coefficient should be less than 1.

1.6. Conclusions

From the results of the numerical simulation, we can obtain the following conclusions:

a) The internal flow distribution could be improved by decreasing the diffuser angle.
b) Smaller diffuser has higher efficiency and for this type of centrifugal fan, the maximum efficiency is reached for flow coefficient 0.13 ÷ 014, while for diffuser angles $\beta = 5^{\circ}, 6^{\circ}, 7^{\circ}, 8^{\circ}$ total pressure losses increase significantly.
c) For diffuser angel $\beta = 2^{\circ}$ it turns out that the centrifugal fan has higher performance. For the flow rate of $Q = 0.4\ {}^{m^3}/_{s}$ the efficiency results at $\eta_T = 61.031$ %.
d) Referring to static pressure recovery coefficient, no significant changes are observed.
e) Static pressure recovery coefficient is negative for flow coefficient 0.13 ÷ 0.14.
f) Optimal value of the total pressure loss coefficient is observed for flow coefficient 0.09 ÷ 0.11.
g) Maximum efficiency is achieved for values of B/b_2= 2,3 and 2,6 at around 70,02%.
h) The static pressure recovery coefficient is highest for smaller values of B/b_2 and decreases (also takes negative values) with the increase of B/b_2.
i) Best performance for low to average values of flow coefficient is shown by width-scale B/b_2=2.3, for average to high values of flow coefficient, lowest values of [$\mathbf{K_p}$] are shown by B/b_2=2.0.
j) Each data and graph taken from the numerical simulations should be verified experimentally in a laboratory in order to validate its results.
k) With Matlab-OpenFOAM coupling it is possible to perform a wide variety of repetitive tasks and cycles by changing certain parameters. Matlab then updated automatically the other values making it easier to get the expected results and in a shorter amount of time.

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