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Slip factor evaluation for centrifugal pump impeller with and without splitter blades, using CFD technique

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***Abstract*— Slip factor is defined as the ratio of the actual and ideal values of the whirl velocity components at the exit of impeller [1]. The ideal values can be calculated using analytical approach while the actual values by the CFD (Computational Fluid Dynamics) results which is in the last years get high accurate results compared with experimental or analytical data which we can depend on, to save more of time and money. On the other hand take care of the boundary conditions, turbulence model, mesh type and number maintaining the integrity of the specifications are so urgent. There is no formula for calculating the slip factor of centrifugal pump impeller with adding short blades, so, the comparison should be between the case values with no short blades. The value of slip factor with no short blades using CFD results is 0.66, with good agreement with the standard.**

Keywords*— *centrifugal pump, pump impeller, blades, slip factor, whirl velocity, CFD technique.

I. INTRODUCTION

In a centrifugal pump, the liquid is forced by atmospheric or other pressure into a set of rotating vanes. A centrifugal pump consists of a set of rotation vanes enclosed within a housing or casing that is used to impart energy to a fluid through centrifugal force (impeller). In turbo-machinery, the slip factor is a measure of the fluid slip in the impeller of a centrifugal machine. Fluid slip is the deviation in the angle at which the fluid leaves the impeller from the impeller's blade/vane angle. Being quite small in axial impellers (inlet and outlet flow in same direction), slip is a very important phenomenon in radial impellers and is useful in determining the accurate estimation of work input or the energy transfer between the impeller and the fluid, rise in pressure and the velocity triangles at the impeller exit.

II. THEORETICAL HEAD DEVELOPED BY AN IMPELLER

Figure 1 shows impeller of a centrifugal pump and the direction of flow velocities outlet the impeller which represents variable by 2 and in inlet which represents variable with 1. Moreover, Figure 2 shows the free body that use for estimating the energy gained by impeller to the fluid [2].

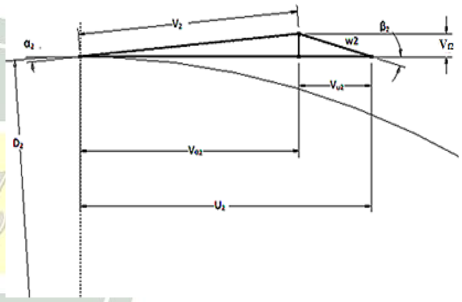


Fig. 1. Outlet Velocity Triangle.

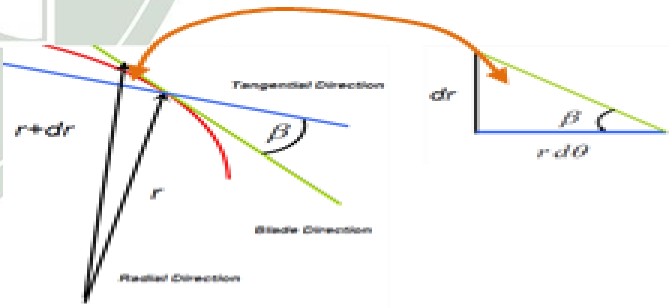


Fig. 2. Free Body of Fluid Leaves Impeller.

The mass of segment is shown in equation (1), where the centrifugal force acting in this elementary will be shown in equation (2) as follows;

$$dm = \rho \times dv = \frac{\gamma}{g} \times h \times r d\theta \times dr \quad (1)$$

$$dF = dm \times r \times \omega^2 = \frac{\gamma}{g} h \times r^2 \times \omega^2 \times d\theta \times dr \quad (2)$$

The pressure increases due to the action of centrifugal force is the force per unit area and it is shown in equation (3).

$$dP = \frac{dF}{dA} = \frac{\frac{\gamma}{g} h \times r^2 \times \omega^2 \times d\theta \times dr}{h \times r \times d\theta} = \frac{\gamma}{g} \times r \times \omega^2 \times dr \qquad (3)$$

Thus, by integrating equation (3) between R1 (intake) and R2 (discharge) when U=Rω, the result will be (P2-P1) as in equation (4).

$$P_2 - P_1 = \frac{\gamma}{2g} \times \omega^2 (R_1^2 - R_2^2) = \frac{\gamma}{2g} (U_2^2 - U_1^2) \qquad (4)$$

In term of head difference and adding the potential head duo to the centrifugal force and change in relative velocity of fluid at impeller with also adding the kinetic energy head duo to diffuser, the theoretical head difference will be shown in equation (5).

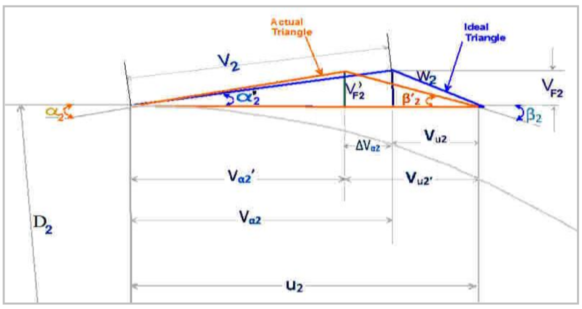
$$H_{th} = \frac{1}{2g} [(U_2^2 - U_1^2) + (W_1^2 - W_2^2) + (V_2^2 - V_1^2)] \qquad (5)$$

From the velocity triangles the values of W1 and W2 could be calculated. Then the theoretical head can be expressed by equation (6).

$$H_{th} = \frac{1}{g} (U_2 V_{\alpha 2} - U_1 V_{\alpha 1}) \qquad (6)$$

III. SLIP FACTOR DESCRIPTION

Under certain circumstances, the angle at which the fluid leaves the impeller may not be the same as the actual blade angle. This is due to a phenomenon known as fluid slip, which finally results in a reduction in Vα2 the whirl component of fluid velocity at impeller outlet. One possible explanation for slip is given as follows. In cause of flow through the impeller passage, there occurs a difference in pressure and velocity between the leading and trailing faces of the impeller blades. On the leading face of a blade there is relatively a high pressure and low velocity, while on the trailing face, the pressure is lower and hence the velocity is higher. This results in a circulation around the blade and a non-uniform velocity distribution at any radius. The mean direction of flow at outlet, under this situation, changes from the blade angle at outlet β2 to a different angle β2̀ as shown in Figure 3.



IV. ESTIMATING SLIP FACTOR

Therefore the whirl velocity component at outlet Vα2 is reduced to Vα2̀ , as shown by the velocity triangles in Figure 3 and the difference between ΔVα them is defined as the slip. The slip factor σ is defined as in equation (7).

$$\sigma = \frac{V_{\alpha 2}^{`}}{V_{\alpha 2}} \qquad (7)$$

Now, the ideal values can be calculated using analytical approach while in the last the actual values should be observed experimentally. Furthermore, there are three famous correlations for estimating slip factor as follow.

(A) Stodola's Equation:

According to Stodola equation (8), it is the relative eddy that fills the entire exit session of the impeller passage. For a given flow geometry, the slip factor increases with the increase in the number of impeller blades, thus, accounts for one of the important parameter for losses, [3].

$$\sigma = 1 - \frac{\pi}{z} \frac{\sin \beta_2}{(1 - \phi_2 \cot \beta_2)}$$

(B) Stanitz's Equation:

Stanitz found the slip velocity does not depend upon of the blade exit angle and hence, gave the equation (9) [1].

$$\sigma = 1 - \frac{1.98}{z(1 - \phi_2 \cot \beta_2)}$$

(C) Balje's Formula:

An approximate formula given by Balje equation (10) for radial-tipped (β2=90°) blade impellers [1]:

$$\sigma = [1 + \frac{6.2}{z.n^{2/3}}]^{-1}$$

Where; n : is the impeller tip diameter over impeller eye diameter.

In the present work the tangential velocity Vu2̀ in actual performance can be got from drawing actual triangle of velocity exit impeller that need to know the actual absolute velocity leaving the impeller which is known by the CFD results which is in the last years get high accurate results compared it to experimental or analytical data thus we can depend on this tool to save more of time and money but take care of the boundary condition and turbulence model and mesh type and number maintaining the integrity of the Specifications that will be determined from formula in equation (11), [3,4].

$$V_2^{`} = \frac{1}{2\pi} \int_0^{2\pi} V_{2\,CFD}\, d\theta$$

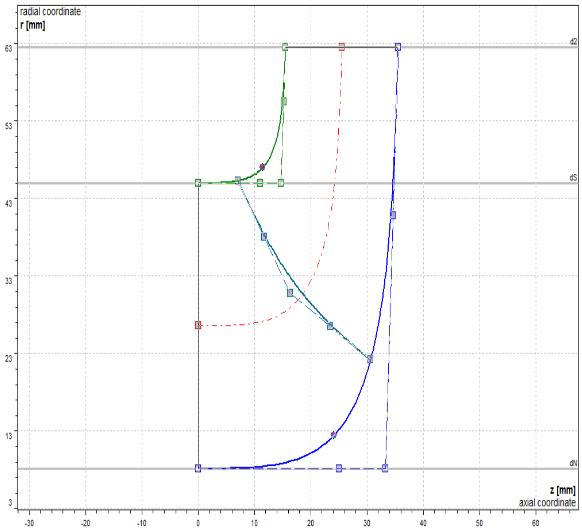
V. COMPLETE PUMP MODEL

(A) Construction Of Pump Model

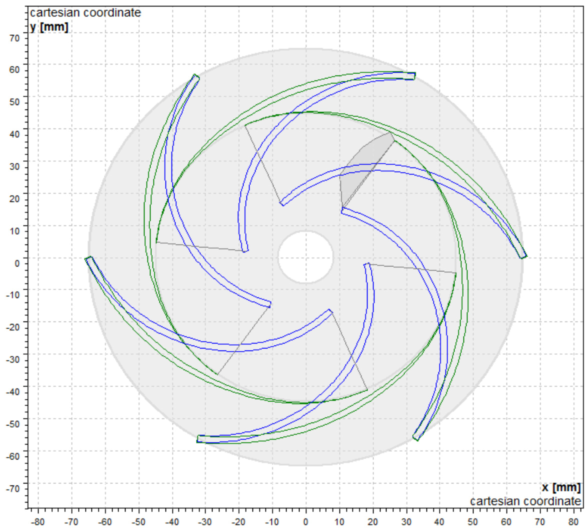
Pump design is complex and time consuming. Therefore modern high-quality software tools are required to enable the engineer to create and analyze several geometry variations and find quickly an optimal solution. CF-Turbo is interactive design software for turbo machinery components: impellers, stators with & without vane and volutes, [5].

1) Impeller Design:

Starting point is the definition of the design point data (flow rate, pressure ratio or difference or total head and rotational speed) in this case these values are (72 m3/hr. flow rate, 20 m total head and 2900 RPM rotational speed) as well as the fluid properties. The meridional contour and passage shape of the impeller are shown in Figure 4 a and b respectively.



(a)

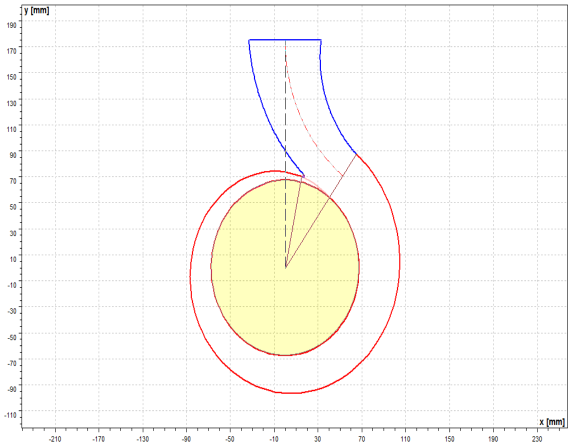


(b)

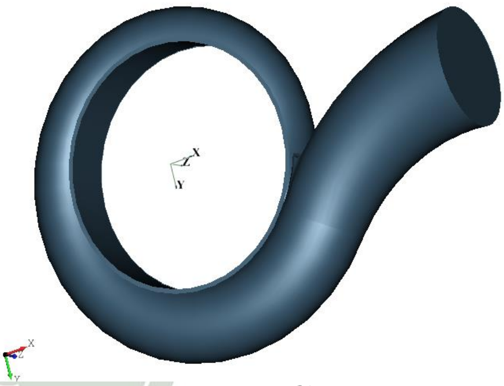
Fig. 4. Meridional Contour and Passage Shape of Impeller.

2) Volute Design

Volute cross section can be variously shaped. The spiral development areas are calculated by the theory of Pfleiderer or Stepanoff. The 2D and 3D volute shape are shown in Figure 5 a and b respectively.



(a)



(b)

Fig 5 Volute Shape

(B) Grid Generations

The mesh size required for the calculations were generated and checked by using the CFD preprocessing package, GAMBIT. Because the geometry of the pump is very complex, unstructured tetrahedral mesh is used. “Equi-Angle Skew” and “Equi-Size Skew” of the grid were all less than 0.95, so the grid quality is good. Figure 6 shows the 3D model and wall grid of the calculation region with 6 blades. The model has 800000 nodes with tetrahedral mesh.

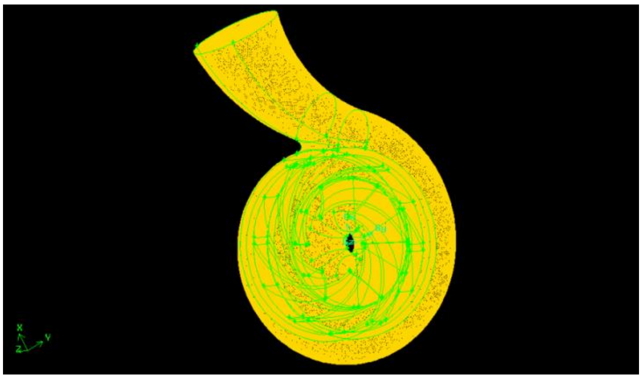


Fig 6 Model and Wall Grid.

(C) Numerical Simulation For Pump Model

FLUENT was used to simulate the inner flow field under non cavitation's' condition. The standard k-ε turbulence model and simple algorithm are applied to solve the equation of k-ε. The simulation is steady and moving reference frame is applied to take into account the impeller-volute interaction. Convergence precision of residuals is 10 –5, [6].

(D) Boundary Conditions

In the present study, mass flow inlet and outflow boundary conditions were used for the inlet and outlet, respectively. Outer walls were stationary but the inner walls were rotational. There were interfaces between the stationary and rotational regions.

VI. Pump Impeller With Short Or Mid Blades

This study explain calculating slip factor of the centrifugal pump impeller with and without short blades , the mean length factor Lb will be considered that is defined as the mean length of short blade Lms to the mean length of blade Lmb as shown in equation (12).

$$L_b = \frac{L_{ms}}{L_{mb}} \qquad (12)$$

The pump tested for impeller with different shorted blades size A to D as a following sequence.

Case A represents the pump with original impeller

Case B represents the pump with shorted blades impeller Lb = 0.33

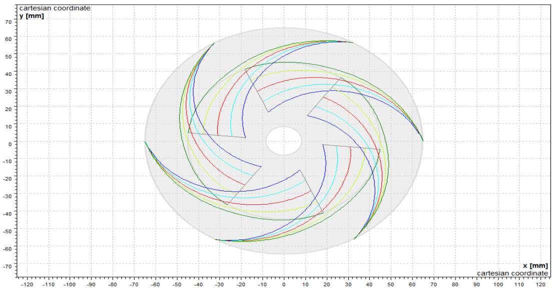
Case C represents the pump with shorted blades impeller Lb = 0.26

Case D represents the pump with shorted blades impeller Lb = 0.16

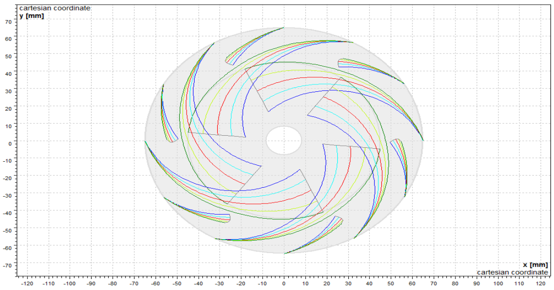
All of these impeller cases have the same basic geometry as the standard impeller (case A). Table 1 shows the main geometry dimensions for the standard impeller (case A) and the modified impeller with short blades (case B, C, and D). Impellers from B, C, and D are modified impellers with shorted blades. The shorted blades are partial blades with shorter length than the main full blades. The leading edges of the shorted blades are located downstream from the leading edges of the full blades at diameters as shown in table 1 and Figure 7.

TABLE I. Main Dimensions For Different Impeller Configurations

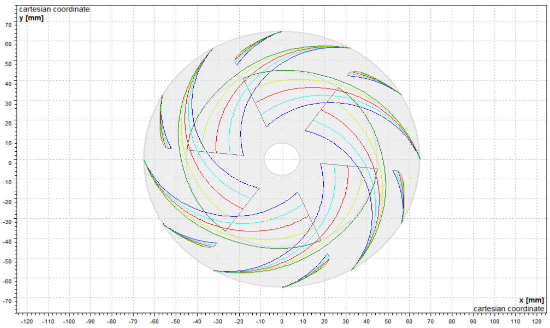
Impeller	**A**	**B**	**C**	**D**
Main blades inlet diam. (D_1) mm	36	36	36	36
Main blades outlet diam. (D_2) mm	125	125	125	125
Shorted blades inlet diam. (D_S) mm	-	100	105	110
(D_S / D_2) %	-	80	84	88
Number of full blades (Z)	6	6	6	6
Number of shorted blades	0	6	6	6
Position of shorted blades	-	Mid way	Mid way	Mid way
Mean length factor (L_b)	-	0.33	0.26	0.16



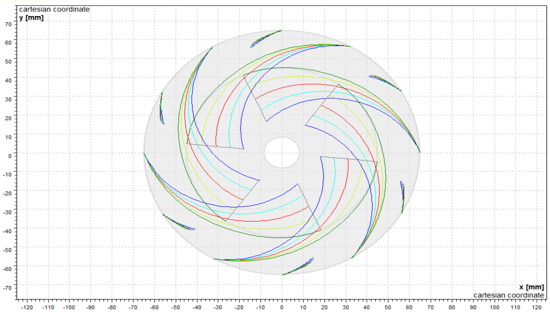
(a)



(b)



(c)



(d)

Fig. 7. Impeller with Different Size Short Blades.

VII. Results

This part presents a few selected results from the CFD analyses in order to make some comparisons with other sources of published data. Numerical results show distribution of static pressure and velocity through the midspan of all cases.

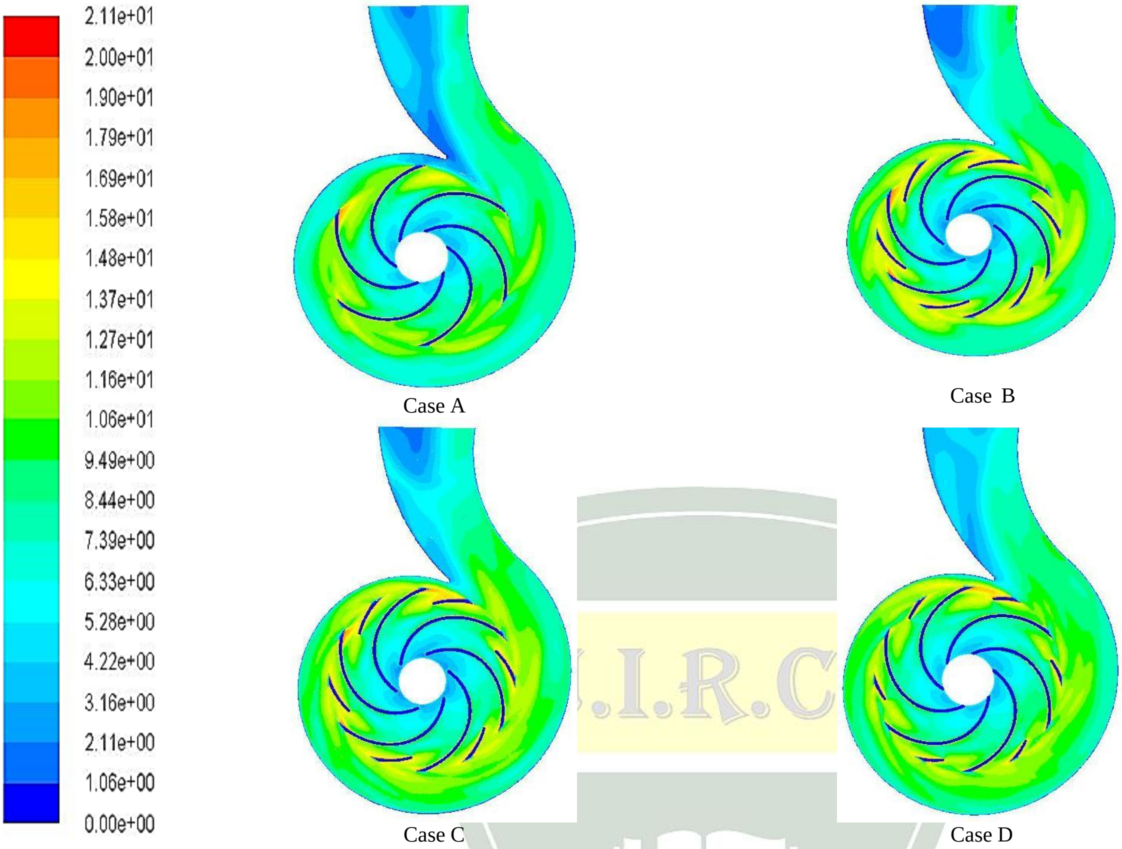
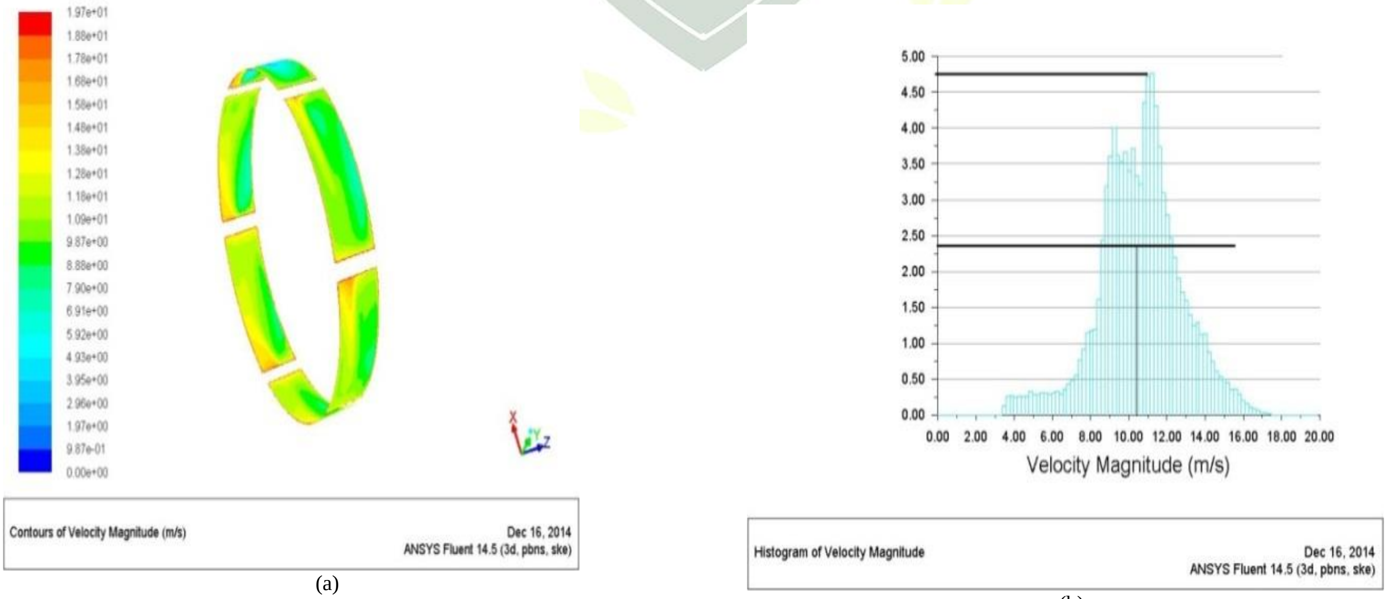


Fig. 8. Absolute velocity contour for all cases.

(A) CFD results of absolute velocity at impeller exit.



(a) (b)

Fig. 9. Absolute Velocity at Impeller Exit Surface Case A.

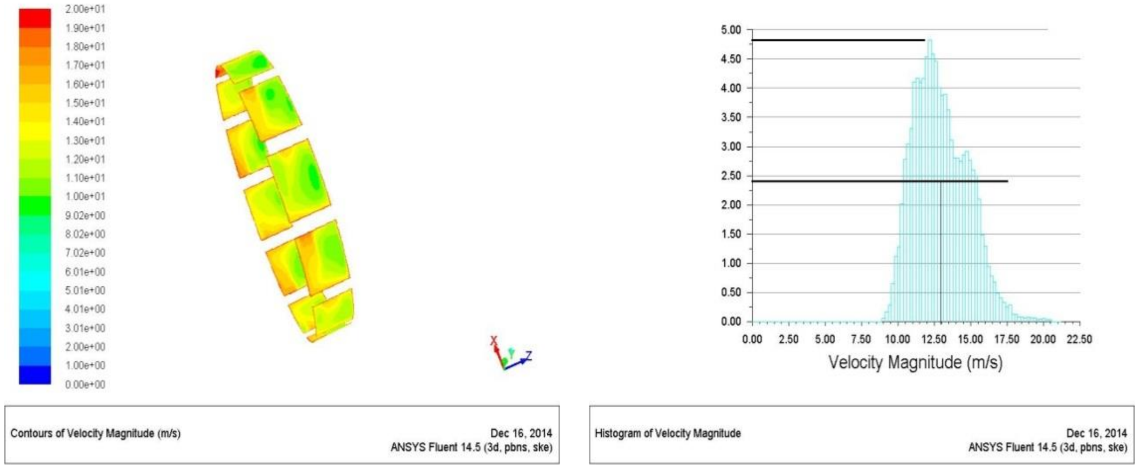


Fig. 10. Absolute Velocity at Impeller Exit Surface Case B.

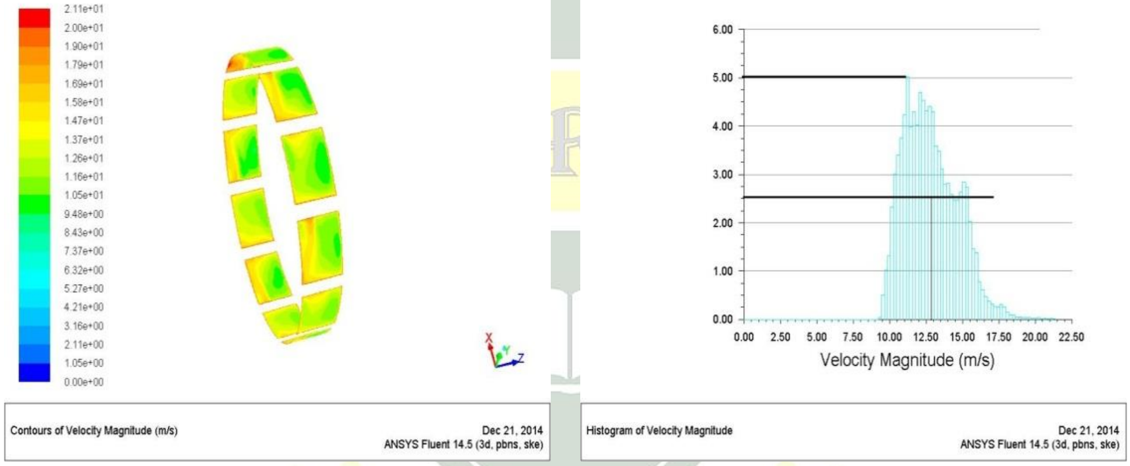


Fig. 11. Absolute Velocity at Impeller Exit Surface Case C.

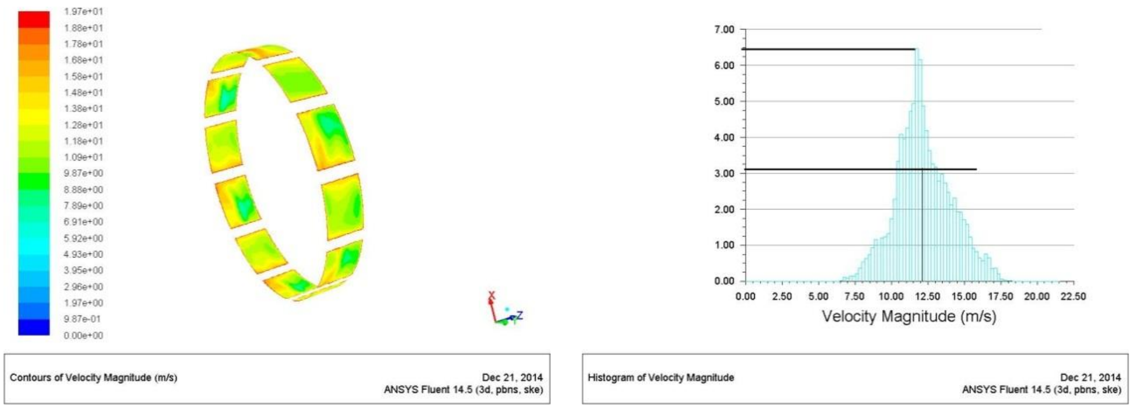


Fig. 12. Absolute Velocity at Impeller Exit Surface Case D.

(A) The results of velocities at impeller exit

Figures from 13 to 16 show the ideal and actual velocity triangle at impeller exit, and table 1 shows the value of actual absolute velocities, whirl velocities, ideal whirl velocities, and slip factor calculated from this results.

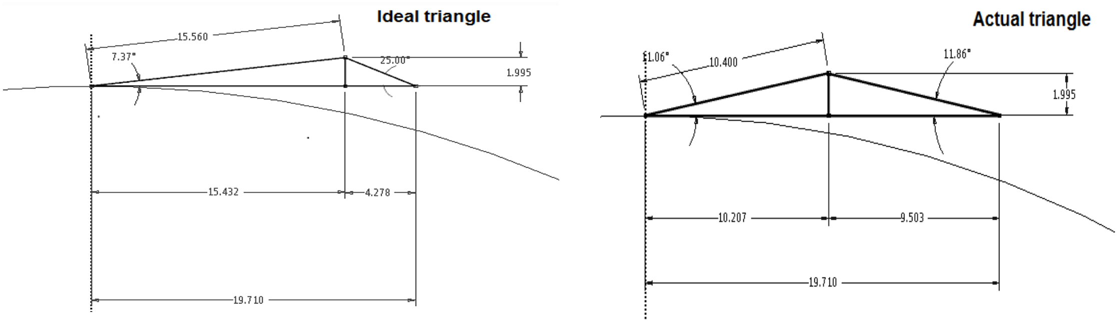


Fig. 13. Ideal and Actual Velocity Triangle at Impeller Exit Case A.

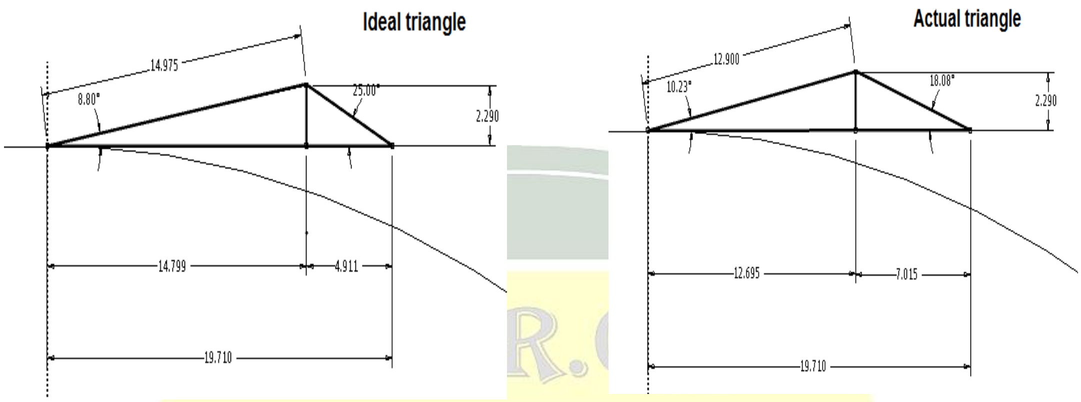


Fig. 14. Ideal and Actual Velocity Triangle at Impeller Exit Case B.

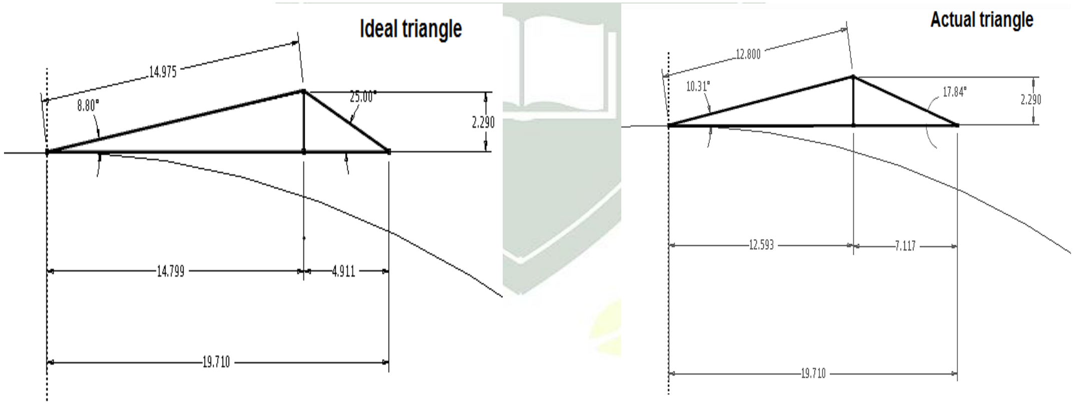


Fig. 15. Ideal and Actual Velocity Triangle at Impeller Exit Case C.

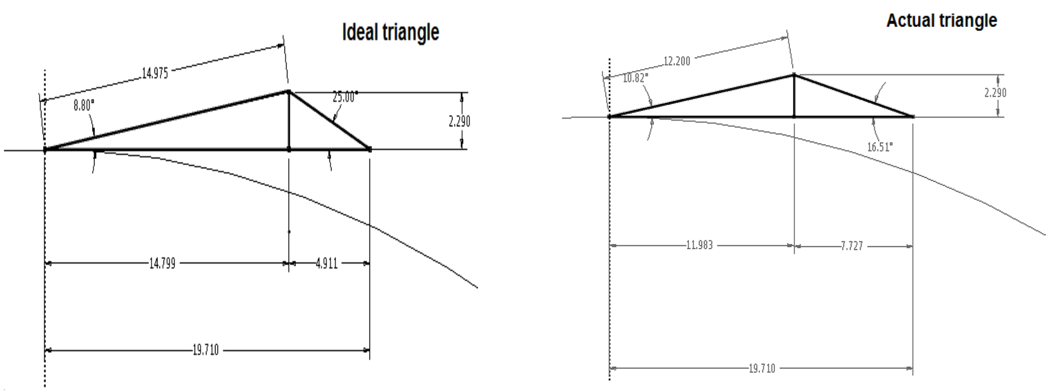


Fig. 16. Ideal and Actual Velocity Triangle at Impeller Exit Case D.

TABLE II. VELOCITIES AND SLIP FACTOR VALUES FOR ALL CASES

Case	V2 (average) m/s	Vα2 m/s	Vα2 m/s	σ	σ Stodla formula	σ Stanitz formula	σ Baljes formula
A	10.4	15.43	10.207	0.66	0.507	0.65	0.39
B	12.9	14.799	12.695	0.857	-	-	-
C	12.8	14.799	12.593	0.85	-	-	-
D	12.2	14.799	11.983	0.81	-	-	-

VIII. CONCLUSIONS

In short blades cases ‘B, C, D’ the flow leaves impeller with higher velocity and reaches the volute exit with higher velocity than the cases with no short blades. The value of the ideal whirl velocity at impeller exit is 15.43 for case A and 14.799 m/s for case B, C, and D. The value of the actual mean absolute velocity at impeller exit is 10.4, 12.9, 12.8, and 12.2 m/s for case A, B, C, and D respectively. The values of the actual whirl velocity at impeller exit are 10.207, 12.695, 12.593, and 11.983 m/s for case A, B, C, and D respectively. The slip factor is 0.66, 0.857,0,85, and 0.81 for case A, B, C, and D respectively. The value of slip factor with no short blades in this research has a good agreement with the value of stanitz formula.

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