# INVESTIGATION OF FLOW CHARACTERISTICS INSIDE THE IMPELLER PASSAGES FOR BACKWARD CURVED CENTRIFUGAL FAN

**[1]KHEMA THEINT, [2]WAR WAR MIN SWE, [3]AUNG KYAW SOE, [4]AUNG KO LATT**

Mechanical Engineering Department Mandalay Technological University
E-mail: [1]khematheint@gmail.com, [2]warwarminswe288@gmail.com, [3]aungkyawsoe.tu@gmail.com, [4]dr.aungkolat@gmail.com

**Abstract** - This paper presents the investigation of flow characteristics inside the impeller passages with free inlet to free outlet (without casing). The air flow in centrifugal fan is complex with the interacting flow within the impeller passages. Numerical analysis was considered at the opening condition by using ANSYS 21.0 (CFD-CFX). The designed inlet impeller diameter is 0.128m and outlet diameter is 0.178m. The inlet and outlet span height are 0.03m. The inlet blade angle is 23° and the outlet blade angle is 30° respectively. The design flow rate is 3m$^3$/min and design head is 4m. The rotating speed of impeller is 1500rpm.The working fluid is considered to be an ideal gas (air at 25°C). The flow velocity and differential pressure were analyzed in the upstream and downstream of the impeller passages.

**Keywords** - ANSYS 21.0 (CFD-CFX), Backward Curved Centrifugal Impeller, Flow Analysis, Without Casing

## I. INTRODUCTION

Centrifugal fans of various descriptions constitute one of several types of turbomachinery, which are used to transfer energy to a flowing fluid.The centrifugal fan uses a rotating impeller to move air fast radially outwards by centrifugal action, and then tangentially away from the blade tips. In backward curved centrifugal fan, the blades are inclined away from the direction of rotation. Flow pattern in centrifugal fans is uneven because of flow separation and recirculation zones in the flow passages. Many researchers contributed to increasing the efficiency of the fan by novel approaches both numerically and experimentally.

Swe, et al. (2017) had observed the flow distribution in the circumferential direction varies remarkably and the flow separation on the blade occurs only at the back region of the fan [1]. Qi, et al. (2006) and Yu, et al. (2005) had studied the flow characteristics are researched to improve the fan performance [2]-[3]. Yamazaki, etal. (1999) had studied the velocity and pressure distribution within the casing and that is mainly concerned to the interaction between the tongue and the impeller [4].

Total pressure and efficiency are important parameters of fan performance. This research focus on the investigation of flow characteristics inside the impeller passages for backward curved centrifugal fan to improve the fan performance.

## II. THE ORETICAL APPROACH

### 2.1. Backward Curved Centrifugal Impeller Design

Impeller design is the heart of any turbomachine and it must be designed to get optimum efficiency. In this research, the impeller design is based on the theory of Centrifugal Pumps and Blowers by Austin H. Church and the formal set of ideas from the National Imperial College of Engineering [5].

The backward curved centrifugal impeller is designed for the head H = 4 m, the volume flow rate Q = 3 m$^3$/min, the motor speed is 1500 rpm and the blade outlet angle is 30 degree.The impeller design parameters are calculated with equation (1) through (19).

$$Q_{real} = 1.05Q \quad (1)$$

Input power of the fan, $$W = \frac{\rho_{air} g Q H}{\eta_0} \quad (2)$$

Torque, $$T = \frac{W}{\omega} \quad (3)$$

Specific speed, $$N_s = \frac{N\sqrt{Q}}{H^{3/4}} \quad (4)$$

Shaft Diameter, $$D_S = \sqrt[3]{\frac{16T}{\pi S_S}} \quad (5)$$

Diameter of hub, $D_H = Ds + (19.05 \text{ to } 50.88)$ (6)

Impeller Eye Diameter, $$D_0 = \sqrt{\frac{4}{\pi} \times \frac{Q_0}{V_0} + D_H^{\,2}} \quad (7)$$

Impeller Inlet Diameter, $D_1 = 1.1\, D_0$ (8)

Impeller outlet diameter, $$D_2 = 19.2\left(\frac{N}{100}\right)^{\frac{1}{6}} \frac{\sqrt{2gH}}{N} \quad (9)$$

Inlet peripheral velocity, $$U_1 = \frac{\pi D_1 N}{60} \quad (10)$$

Outlet peripheral velocity, $$U_2 = \frac{\pi D_2 N}{60} \quad (11)$$

Continuity equation, $Q_{re} = A_1 V_{f1}$ (12)

Relative velocity at inlet, $$V_{r1} = \sqrt{U_1^2 + V_1^2} \quad (13)$$

Impeller Inlet angle, $$\tan\beta_1 = \frac{V_{f1}}{U_1} \quad (14)$$

Flow velocity at outlet, $$V_{f_2} = \sin\beta_2 V_{r_2} \quad (15)$$

Whirl velocity at outlet, $$V_{w_2} = \sqrt{V_{r_2}^{\,2} - V_{f_2}^{\,2}} \quad (16)$$

Absolute velocity at outlet, $V_2 = \sqrt{(U_2 - V_{w_2})^2 - V_{f_2}^2}$ (17)

Number of blades, $Z = \frac{l}{t} \times \frac{D_2 + D_1}{D_2 - D_1} \sin\frac{\beta_1 + \beta_2}{2}$ (18)

Blade chord length, $l = \frac{D_2 - D_1}{2\sin(\frac{\beta_1 + \beta_2}{2})}$ (19)

The result data of the backward curved impeller parameters in design conditions are listed in Table 1.In this research, the impeller blades are manufactured by CNC machine based on the theoretical result. Fig.1 shows the prototype of impeller blade. The impeller is cast in aluminum.

| Parameter | Symbol | Dimension | Units |
|---|---|---|---|
| Specific speed | Ns | 918 | rpm |
| Inlet diameter | $D_1$ | 128 | mm |
| Outlet diameter | $D_2$ | 178 | mm |
| Eye diameter | $D_0$ | 117 | mm |
| Shaft Diameter | Ds | 10 | mm |
| Hub diameter | $D_H$ | 60 | mm |
| Blade angle at inlet | $\beta_1$ | 23 | degree |
| Blade number | Z | 14 | - |
| Blade chord length | l | 55 | mm |
| Blade thickness | t | 2 | mm |

**Table1: Result Dataof Backward Curved Centrifugal Impeller Parameters**



**Fig.1. Backward Curved Centrifugal Impeller Blade**

### 2.2. Velocity Diagram of Backward Curved Blade

In a centrifugal fan, the air enters the impellerin an axial direction and is discharged at the periphery, the impeller rotation being toward the casing outlet. The theoretical result of flow velocity, relative velocity and absolute velocity of the impeller were tabulated in Table 2andthe inlet and outlet velocity diagram as shown in Fig.2.

| Parameter | Symbol | Dimension | Units |
|---|---|---|---|
| Flow velocity at inlet | $V_{f1}$ | 4.35 | $ms^{-1}$ |
| Flow velocity at outlet | $V_{f2}$ | 3.12 | $ms^{-1}$ |
| Relative velocity at inlet | $V_{r1}$ | 10.95 | $ms^{-1}$ |
| Relative velocity at outlet | $V_{r2}$ | 6.28 | $ms^{-1}$ |
| Absolute velocity at inlet | $V_1$ | 4.35 | $ms^{-1}$ |
| Absolute velocity at outlet | $V_2$ | 7.92 | $ms^{-1}$ |
| Whirl velocity at outlet | $Vw_2$ | 5.45 | $ms^{-1}$ |
| Tangential velocity at inlet | $U_1$ | 10.05 | $ms^{-1}$ |
| Tangential velocity at outlet | $U_2$ | 13.98 | $ms^{-1}$ |

**Table 2: Theoretical Result of Inlet and Outlet Velocities**

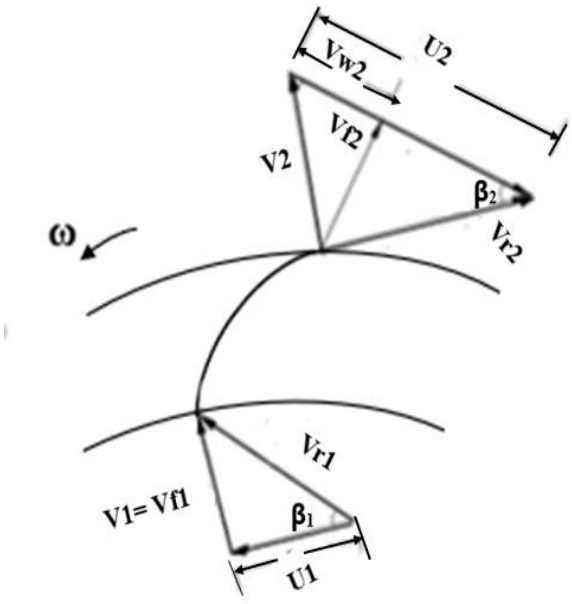



**Fig.2. Inlet and Outlet Velocity Diagram of Backward Curved Impeller**

## III. NUMERICAL APPROACH

### 3.1 Geometry Generation

In present work, the geometry is constructed in ANSYS-CFX workbench. After creating the model, construct the backward curved centrifugal fan impeller by CNC machine. To analyze the flow through the numerical simulation, the impeller is run

by a motor having rotational speed of 1500 rpm. Theimpeller consists of fourteen backward curved blades with an exit angle of 30° relative to the tangential direction.

The simplified physical model of impeller blade is created in ANSYS CFX as shown in Fig.3.The creating of fluid domain in each geometry as shown inFig.4. The three domains: the inlet, the rotating and opening domains are created. In that geometry, the impeller is rotation with 1500 rpm, and it is created as rotating domain.

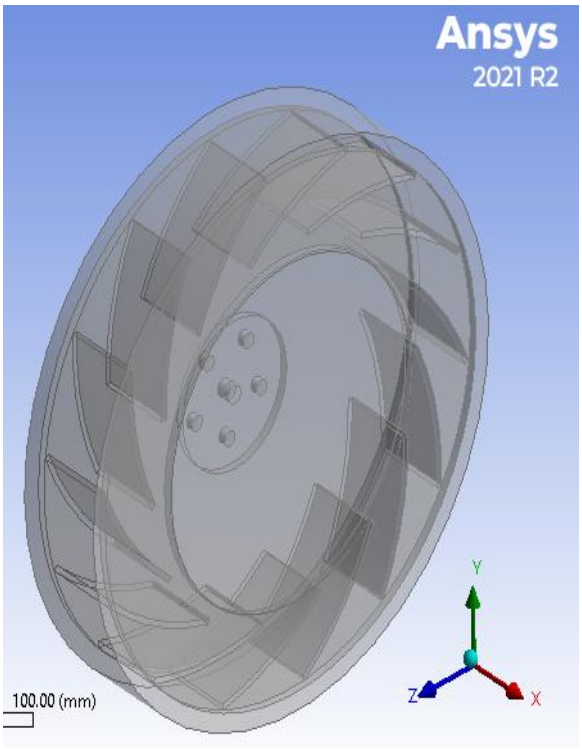



**Fig.3.Geometry of Impeller**

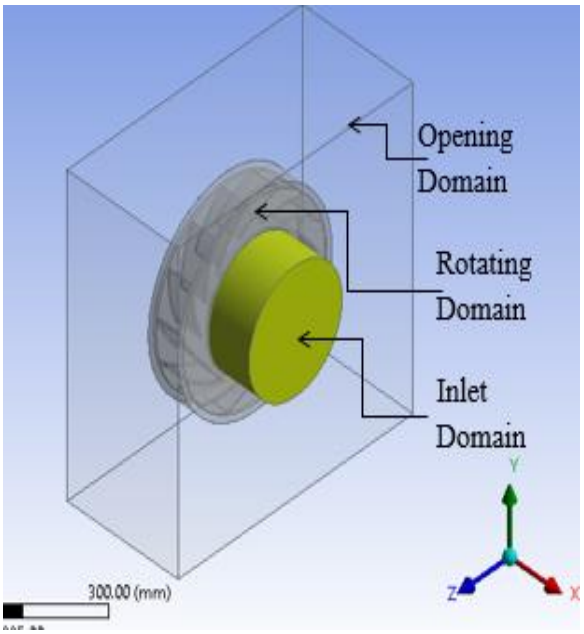
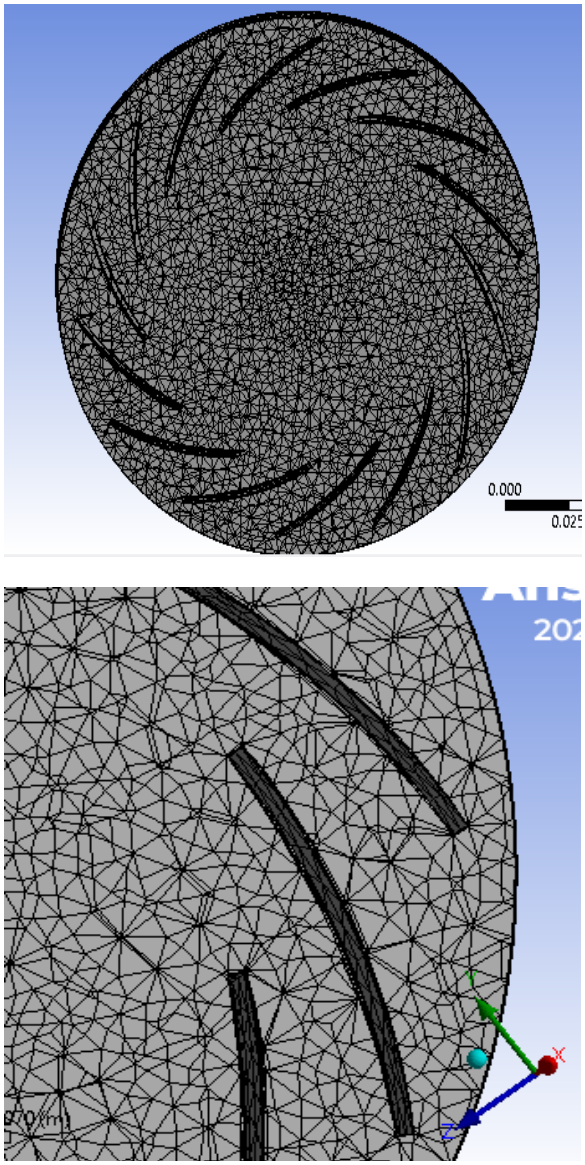



**Fig. 4. Creating of Fluid Domains**

### 3.2 Mesh Generation

The mesh in rotating impeller domain is shown in Fig.5. The total number of mesh for all domain is about two million with tetra mesh shown in Fig. 5(a). The minimum size of tetra mesh is 0.7 mm. The mesh of blades is created as inflection as shown in Fig. 5(b). The mesh inflection is set 5 layers.




**(a)Meshing Model of Impeller (b)Inflection Meshing**
**Fig.5. Meshed Model of Impeller**

### 3.3. Setup Boundary Conditions

Simulation of impeller blade without scroll casing is shown in Fig. 6.The fluid entersthe impeller passages, and passthrough the impeller blades and then discharged by blades.The mass flow rate (0.064 $kgs^{-1}$) is set at the inlet boundary.The impeller is exposed to the surrounding air (1 atm)andthe boundary is opening pressure and Drin (1 atm). The working fluid is considered to be an ideal gas (air at 25°C). The solid surfaces in the computational domain are considered to be no-slip wall and smooth wall conditions.

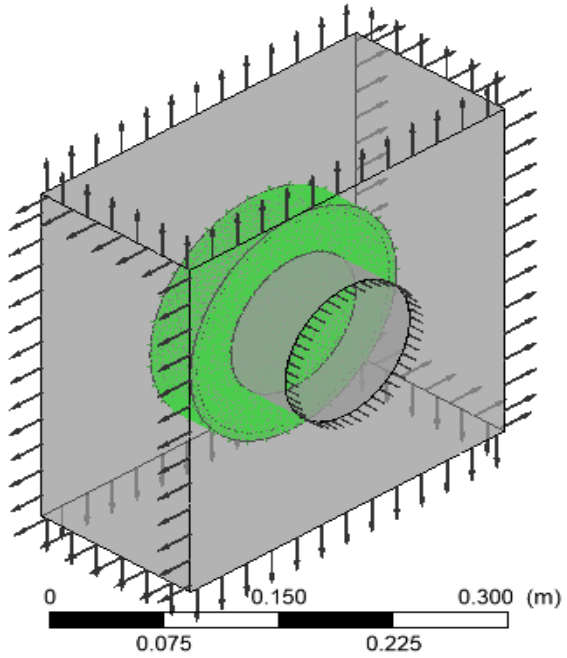



**Fig.6. Setting up the Boundary Conditions**

The general parameters and boundary conditions used for 3D flow simulation of the centrifugal impeller are summarized in the Table 3. For all simulations the boundary conditions are as follows:

| Parameters | Boundary conditions |
|---|---|
| Analysis | Steady State |
| Working Fluid | Air at 25° C |
| Impeller | Rotating 1500 rpm |
| Inlet | Mass Flow rate |
| Outlet | Opening Pressure Drin |
| Wall | No-slip wall |
| Turbulence model | Shear Stress Transport(SST) |

**Table3: Boundary Conditions Used for Simulation**

## IV. NUMERICAL ANALYSIS RESULTS

There were two parts of analysis, i.e., the first one was the flow pattern in the impeller inflow and outflow. The other was the result of the impeller efficiency. The impellerhydraulic efficiency can be calculated by using equation (20). In equation (20), the input power is blade load and the output power is the total pressure rise from impeller inlet tooutlet.

$$\text{Hydraulic efficiency of impeller, } \eta_0 = \frac{\Delta PQ}{T\omega} \quad (20)$$

### 4.1 Flow Pattern in Blade Passages

The flow characteristics and pressure distribution of impeller blades were demonstrated in Fig. 7to Fig.14.As the impeller rotate, it throwaway the fluid particles radially towards the volute casing of the fan. Therefore, high fluid particles velocity is observed around the impeller tip(seeFig.7toFig.10).

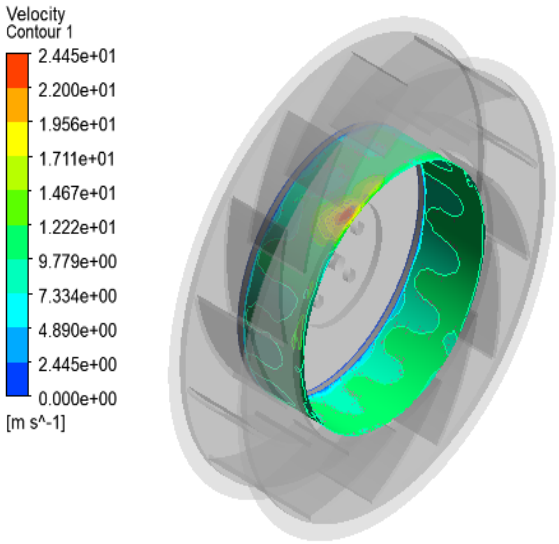



**Fig. 7. Relative Velocity of ImpellerInlet**

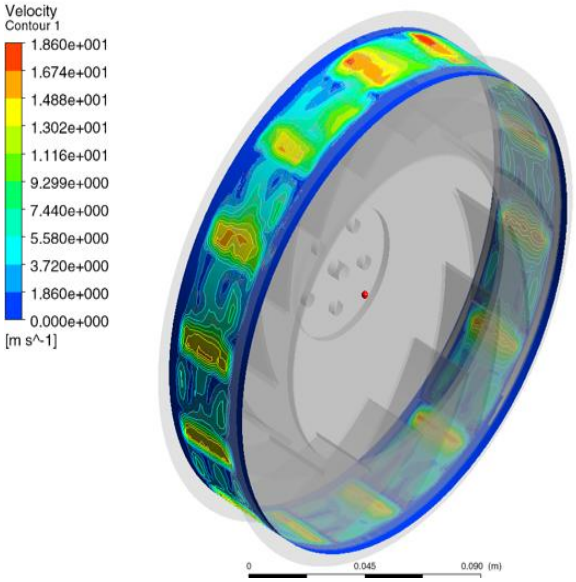



**Fig .8. Relative Velocity of Impeller Outlet**

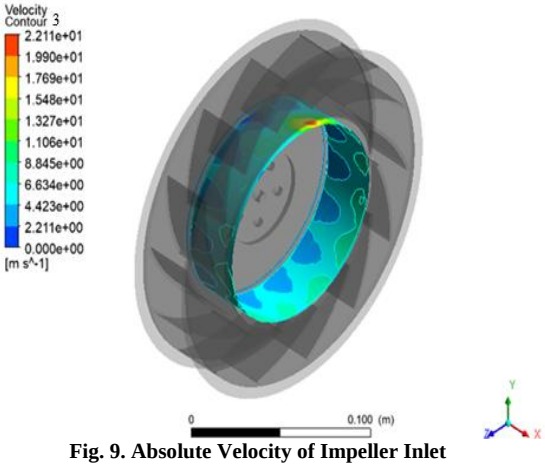



**Fig. 9. Absolute Velocity of Impeller Inlet**

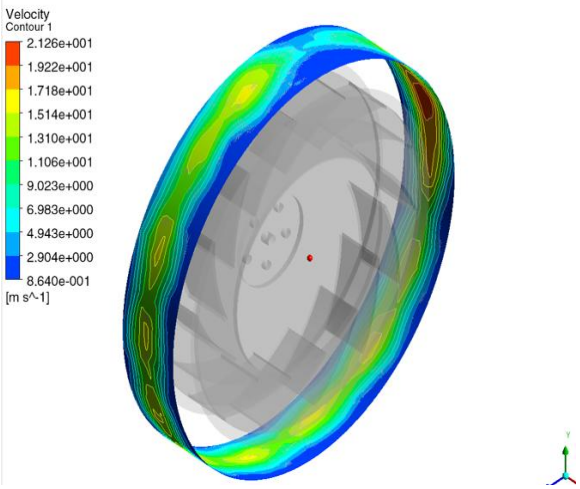



**Fig. 10. Absolute Velocity of Impeller outlet**

Fig.11through Fig.14were the static pressure and total pressure distributions in impeller inlet and impeller outlet.The maximum static pressure occurs at inlet circumference and the minimum static pressure occurs at outlet circumference.Then, the total pressure distribution gradually decreases from the impeller's inlet to outlet when it approaches to the free outlet condition (without casing).

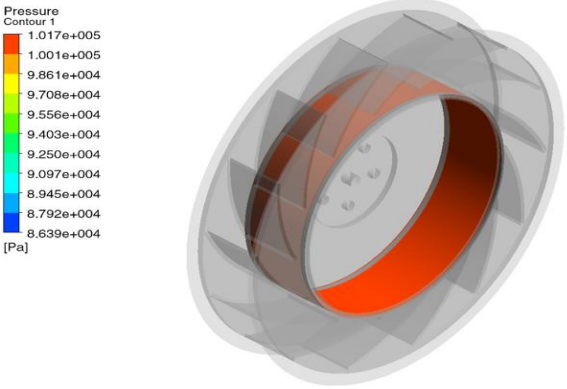



**Fig.11. Static Pressure at Impeller Inlet**

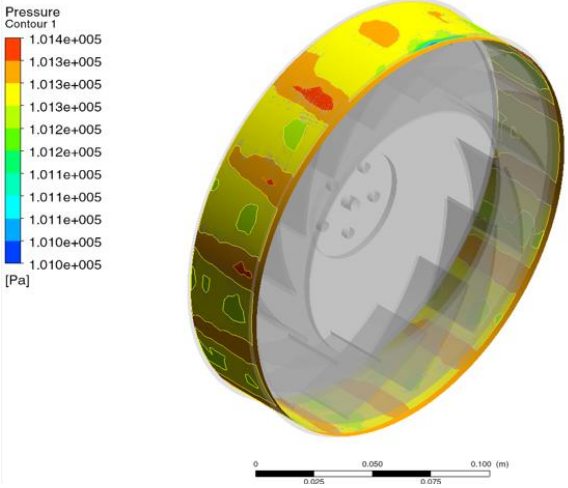



**Fig. 12. Static Pressure at Impeller Outlet**

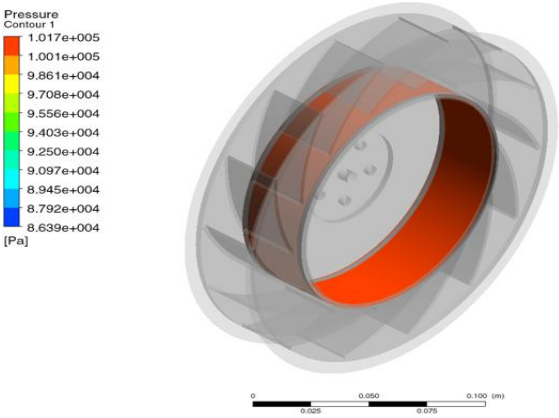



**Fig. 13. Total Pressure at Impeller Inlet**

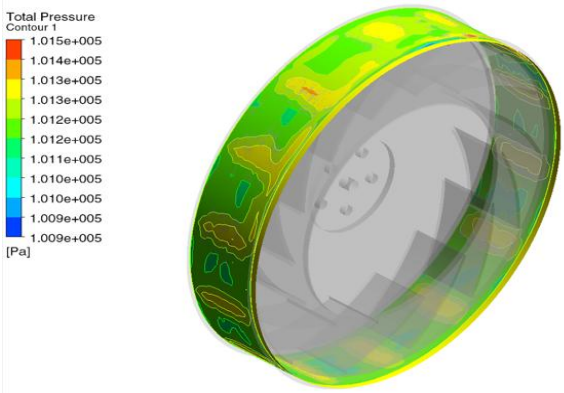



**Fig.14. Total Pressure at Impeller Outlet**

### 4.2. Pressure and Velocity Distribution Results

Table 4 shows the numerical result data of pressure and velocity distributionfor the impeller inlet and outlet were collectedfrom CFD post.The comparison of velocities result on the theoretical and numerical are described as shown in Table 5. The purpose of this comparison is to verify the accuracy of the numerical design of backward curved centrifugal impeller. Thesevalues of velocity at impeller inlet and outlet are plotted in Fig.15. According to the Fig.15, it was found that, the velocity distribution is nearly equal to the theoretical and numerical approach.

| Parameter | Symbol | Inlet | Outlet | Unit |
|---|---|---|---|---|
| Avg Static pressure | $p_s$ | 101.508 | 101.293 | kPa |
| Avg Total Pressure | $p_t$ | 101.544 | 101.233 | kPa |
| Avg Relative velocity | $V_r$ | 10.64 | 7.41 | $ms^{-1}$ |
| Avg Absolute Velocity | $V_{abs}$ | 6.00 | 8.19 | $ms^{-1}$ |

**Table 4: Numerical Result of Pressure and Velocities**

| Parameter | Symbol | Theoretical | Numerical | Unit |
|---|---|---|---|---|
| Relative velocity at inlet | $Vr_1$ | 10.95 | 10.64 | $ms^{-1}$ |
| Relative velocity at outlet | $Vr_2$ | 6.28 | 7.41 | $ms^{-1}$ |
| Absolute Velocityat outlet | $V_2$ | 7.92 | 8.19 | $ms^{-1}$ |

**Table 5: Comparisonof Velocities on Theoretical and Numerical Results**

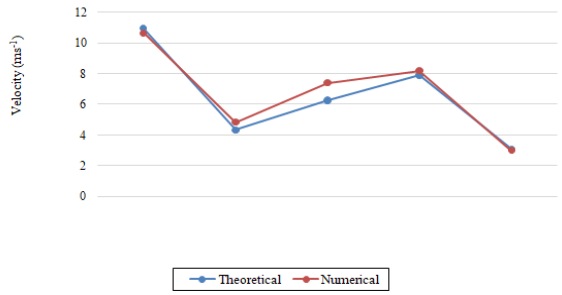



**Fig.15.Comparison of Velocities on Theoretical and Numerical Results**

### 4.3. Impeller Hydraulic Efficiency from Numerical Results

In hydraulic efficiency equation (20), total pressure difference at impeller inlet and outlet circumference are substituted by 311 Pafrom numerical result. The impeller efficiency is that computed by using equation(20) is 65 percent.

## V. CONCLUSION

In this research, investigation of flow characteristics inside the impeller passages for backward centrifugal fan without casing have been presented through theoretical and CFD simulation. The total pressure values in the circumference of the inlet and outlet are 101.544 kPa and 101.233 kPa. The static pressure at entry is 101.508 kPa and at exit of impeller is 101.293 kPa.The relative velocity ($V_r$), the absolute velocity ($V_2$) in the impeller outlet is almost the same in theoretical and numerical simulation. The impeller hydraulic efficiency is 0.65 withoutcasing design.

## ACKNOWLEDGEMENTS

First and foremost, the author would like to express her deep gratitude to her parents for their supports and encouragements. The author is also grateful all the thankworthy persons and friends for their kind help while working this paper.The author would like to thank respective reviewers.

## REFERENCES

[1] Swe, W. W. M., Morimatsu, H., Hayashi, H., Okumura, T., & Oda, I. (2017). Study of unsteady flow simulation of backward impeller with non-uniform casing. Journal of Thermal Science, 26(3), 208–213. doi:10.1007/s11630-017-0931-x

[2] Qi, D., Zhang, Y., Wen, S., and Liu, Q., “Measurement and Analysis of Three-Dimensional flow in a Centrifugal Fan Volute with Large Volute Width and Rectangular Cross Section,” IMechE, Vol.220, pp.133-153, 2006.

[3] Yu, Li, S., He, W., Wang, W., Huang, D., and Zhu, Z., “Numerical Simulation of Flow Field for a Whole Centrifugal Fan and Analysis of the effect of Blade Inlet Angle and Impeller Gap,” HVAC & R Research, Vol.11-2, pp.263-283, 2005.

[4] Yamazaki, Komatsu, S., Obata, T., and Yokoyama, S., “Investigating of Flow Pattern in a backward-Curved Fan and Influence of Its Fan Shape on performance and Noise,” JSME, Vol.65-636, pp.2770-2776, 1999.

[5] Church Austin H., “Centrifugal Pumps and Blowers”, John Wiley & Sons, New York, 1962.

★★★