Design and Flow Analysis of Double-suction Centrifugal Pump Impeller

NWE NI WIN

Ph.D. Researcher (Mechanical Engineering) Yangon Technological University, Yangon, Myanmar.

Email - shwekyikhin@gmail.com

Abstract: This paper describes the study of fluid flow in the backward-curved impeller by using SoildWorks Software. Flow Analysis is also based on the computational fluid dynamic (CFD) and the pressure distribution and velocity distribution of the centrifugal pump impeller. SoildWorks Software is used to build 3D geometry for using Flow Simulation. SolidWorks is widely used in engineering fields for product design, testing and manufacture. Computational fluid dynamics (CFD) analysis is being increasingly applied in the design of centrifugal pumps. CFD analysis is very useful for predicting pump performance at various mass flow rates. For designers, predicting of operating characteristics curve is most important. SolidWorks flow simulation was used to simulate the inner flow field under steady condition. Governing equations for SolidWorks flow simulation solves three- dimensional Navier-Stokes equations using IL turbulence model with the finite volume method. The design data of the model pump are the flow rate (0.13 m^3/sec), head (35 m) and operating speed (1470 rpm).The inlet and outlet diameter of impeller are 191 mm and 360 mm. The inlet and outlet passage width of impeller are 35 mm and 49 mm respectively. And the numbers of blade are 8.

Key Words: Centrifugal pump, Double-suction impeller, Flow analysis, SolidWorks Software.

1. INTRODUCTION:

A pump, in general may be defined as a machine, when drive from some external sources, lifts to a lower level to a higher level. In other words, a pump may also be defined as a machine, which converts mechanical energy into pressure energy. A centrifugal pump consists of a set of rotating vanes, called impellers, enclosed within a stationary housing called a casing. Water is forced into the centre (eye) of the impeller by atmospheric or other pressure and set into rotation by the impeller vanes. The resulting centrifugal force accelerates the fluid outwards between the vanes until it is thrown from the periphery of the impeller into the casing. The casing collects the liquid, converts a portion of its velocity energy into pressure energy and directs the fluid to the pump outlet.

The two main parts of the pump are the impeller and the diffuser. Impeller, which is the only moving parts, is attached to a shaft and driven by a motor. Impellers are generally made of bronze, cast iron, stainless steel as well as other materials. The diffuser (also called as volute) houses the impeller and directs the water off the impeller.

An impeller is a rotating disk with a set of vanes coupled to the engine (or) motor shaft that produces centrifugal force within the pump casing. The liquid is forced by atmospheric or other pressure into a set of rotating vanes. These vanes constitute an impeller that the liquid at its periphery at a higher velocity. The design of the impeller depends on the requirements for pressure, flow and application. The impeller is the primary component determining the pump performance.

The centrifugal pumps may be either single or multi-stage. In addition, the single-stage centrifugal pumps may be either single-suction or double-suction. In single-suction centrifugal pumps, the water enters on one side of the casing and impeller. Double-suction centrifugal pump is usually used in water service. A double-suction impeller is the same in effect as two single-suction impellers placed back to back on a horizontal shaft, supported by bearings on either side. This type allows liquid to enter the eye of impeller from both sides.

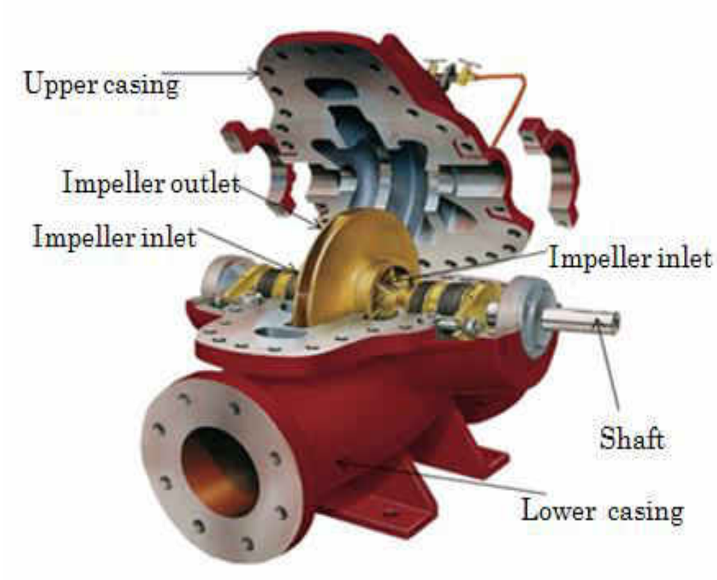

Figure1.Typical Double-suction Centrifugal pump

2. DESIGN PROCEDURE OF DOUBLE-SUCTION IMPELLER:

The centrifugal pump is analysed single stage double-suction backward-curved impeller. The design procedure of the double-suction impeller involves the following steps:

Table 1

Parameters Consideration for Design

Head (H)	35 m
Flow rate(Q)	0.13 m^3/s
Operating speed (n)	1480 rpm
Efficiency (η_o)	0.70

Table 1 shows the parameters considered for design of a centrifugal pump.

2.1 Design Calculation Procedure of Flow Chart

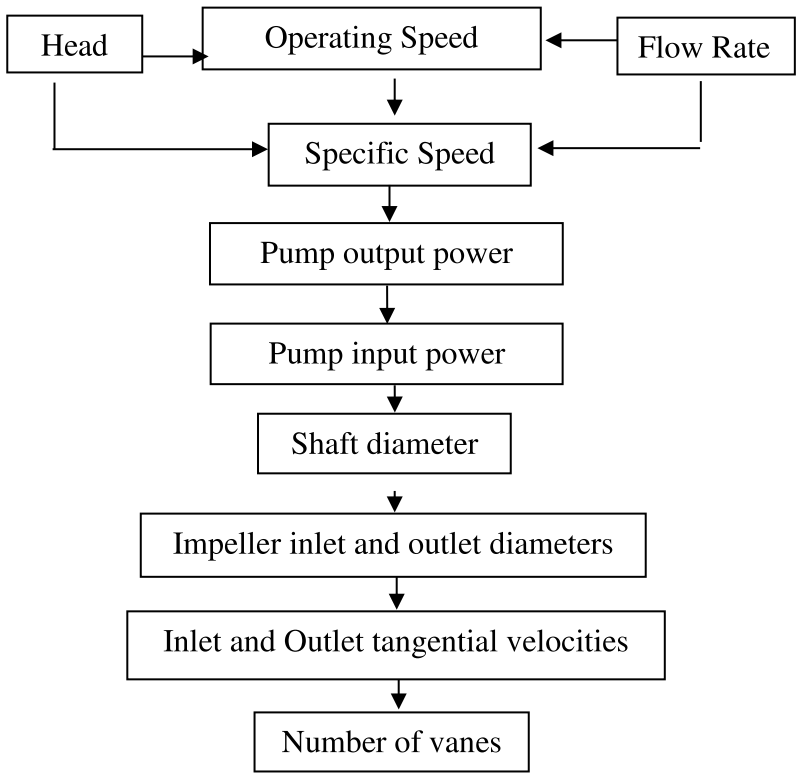

Figure1. Flow Chart of Impeller Dimensions

For radial type impeller, the specific speed is calculated by the following equation. The specific speed,

$$n_s = \frac{n\sqrt{Q}}{H^{\wedge\frac{3}{4}}} \qquad (1)$$

The input power,

$$L = \frac{\rho g Q H}{\eta} \qquad (2)$$

2.2 Rated Output Power of Electric Motor

The rated output power of electric motor, L_r is obtained by

$$L_r \geq \frac{(1+f_a)L}{\eta_{tr}} \qquad (3)$$

The allowable factor, $f_a = 0.4$ (for electric motor)

Transmission efficiency, $\eta_{tr} = 1$ (for direct coupling)

The volumetric efficiency is calculated by using the following equation.

$$\eta_v = \frac{1}{1+\frac{1.124}{n_s^{\frac{2}{3}}}} \qquad (4)$$

The diameter of end of the main shaft is obtained by

$$d_c \geq K_{sh}\sqrt[3]{\frac{L_r}{n}} \qquad (5)$$

The shaft diameter D_{sh} is obtained twenty percent greater than the diameter of the end of the main shaft. Therefore,

$$D_h = (1.5\sim2.0)\,D_{sh} \qquad (6)$$

The length of the hub, L_h at the impeller is calculated by the Equation:

$$L_h = (1.0\sim2.0)\,D_{sh} \qquad (7)$$

The diameter of impeller eye D_0 is calculated from the following equation:

$$D_0 = \sqrt{\frac{4Q_s'}{2\pi V_{m0}} + D_h^2} \qquad (8)$$

Where the velocity at the eye suction is

$$V_{m0} = K_{m0}\sqrt{2gH} \qquad (9)$$

$$K_{m0} = 0.07 + 0.00023 n_s \qquad (10)$$

Vane inlet velocity, V_{m1} is $V_{m1} = K_{m1}\sqrt{2gH}$ (11)

Vane outlet velocity, V_{m2} is

$$V_{m2} = K_{m2}\sqrt{2gH} \qquad (12)$$

Vane outlet tangential velocity, U_2 is

$$U_2 = K_u\sqrt{2gH} \qquad (13)$$

2.3 Dimensions of Impeller Inlet and Outlet Diameter

The impeller inlet diameter is obtained by

$$D_{1m} \sim D_1 = \frac{D_{1h} + D_{1s}}{2} \qquad (14)$$

The impeller outlet diameter is obtained from

$$D_2 = \frac{60\,U_2}{\pi\,n} \tag{15}$$

Inlet tangential velocity, U_1 is

$$U_1 = \frac{\pi\,D_1 n}{60} \tag{16}$$

2.4 Dimensions of Vane Inlet and Outlet Angle

The water is assumed to enter the vanes radially, so that the absolute velocity α_1 is 90˚. After V_{m1} and U_1 have been calculated, the vane inlet angle β_1 is obtained by the Equation.

$$\beta_1 = \tan^{-1}\left(\frac{V_{m1}}{U_1}\right) \tag{17}$$

In this design the vane outlet angle β_2, is assumed as 22.5˚ which is the average value for all specific speed.

The number of blades, Z is $Z = 6.5 \times \frac{D_2 + D_1}{D_2 - D_1} \sin\left(\frac{\beta_1 + \beta_2}{2}\right)$ (18)

The inlet passage width b_1 and b_2 are calculated by

$$b_1 = \left(\frac{Q_s{}'}{2\pi D_1 V_{m1}}\right) \times \left(\frac{\pi D_1}{\pi D_1 - S_1 Z}\right) \tag{19}$$

$$b_2 = \left(\frac{Q_s{}'}{\pi D_2 V_{m2}}\right) \times \left(\frac{\pi D_2}{\pi D_2 - S_2 Z}\right) \tag{20}$$

Table 2

Result Data of impeller Dimensions

No	Dimensions	Values	Units
1	Specific speed, n_s	206	-
2	Pump output power,	44.63	KW
3	Pump input power, L	63.77	KW
4	Diameter of shaft, D_s	60	mm
5	Impeller inlet diameter, D_1	191	mm
6	Impeller outlet diameter, D_2	360	mm
7	Impeller eye velocity,Vm_0	3.145	m/s
8	Impeller inlet tangential velocity,U_1	15.00	m/s
9	Impeller outlet tangential velocity,U_2	27.80	m/s
10	Impeller inlet width,b_1	35	mm
11	Impeller outlet width, b_2	49	mm
12	Numbers of blades, Z	8	

2.5 Required Parameters for Impeller Blade Shape

To draw the curvature of the blade curve equally spaced circles are drawn between impeller outside circle and impeller inside circle. Vane slope angles are also drawn. The angle between β_1 and β_2 are equally divided into three angles.

Impeller outside diameter,

$D_2 = D_A = 360$ mm

Impeller inside diameter,

$D_D = D_{1h} = 171$ mm

The required parameter to layout he impeller blade,

$$\rho = \frac{R_b^{\;2} - R_a^{\;2}}{2(R_b \cos\beta_b - R_a \cos\beta_a)} \tag{21}$$

Table 3

Result Data of Blade profile

Basic Circle Radii	mm	Blade parameter	mm	Angle	Degree
R_A	180	ρ_A	180	β_2	22.5
R_B	149	ρ_B	145	β_B	20
R_C	117	ρ_C	110	β_C	17.5
R_D	86	ρ_D	86	β_1	15

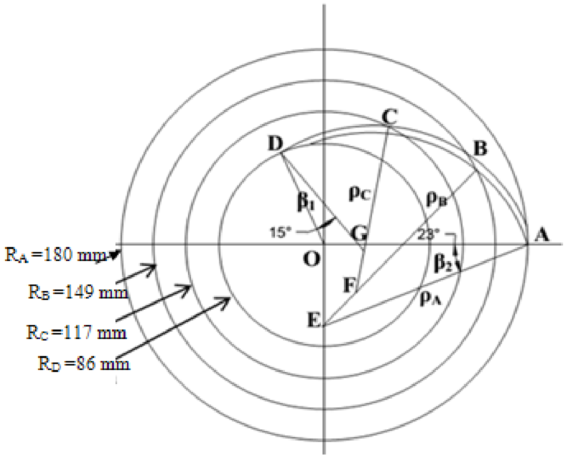

Figure.2.Two Dimensional of Impeller Blade Shape

Impeller blade curve is drawn by using radii of base circle, diameter of the shaft, diameter of the hub, blade height, and shroud thickness. The three dimensional view of the impeller are shown in the following Figure 3.

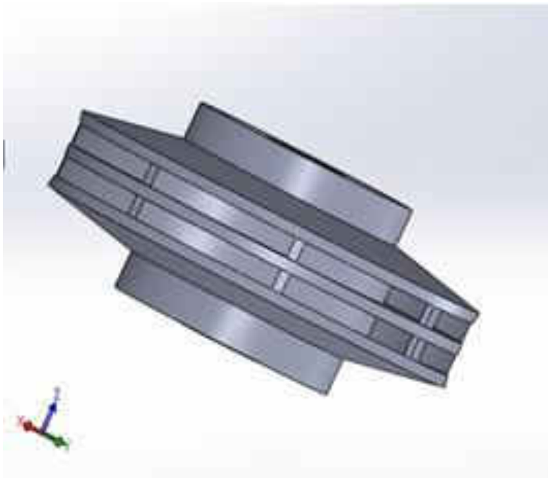

Figure3.Modeling of a double-suction impeller by SolidWorks

3. FLOW ANALYSIS OF DOUBLE-SUCTION IMPELLER:

SolidWorks flow simulation was used to simulate the inner flow field under steady condition. The simulation is steady, and turbulence intensity and length are 2% and 0.002m. Governing equations for SolidWorks flow simulation solves three- dimensional Navier-Stokes equations using IL turbulence model with the finite volume method.

The first step in CFD simulation is preprocessing. In preprocessing modeling and mesh is generated. The CAD model and mesh was carried out using SolidWorks. Automatic mesh generation with automatic detection of initial mesh settings resolves the governing equations. Automatic meshing tools allowed creating mesh for any arbitrary three-dimensional model. Meshing subdivided the model and the fluid volume into several tiny pieces called cells. The multi block multi grid approach was used to approximate the solid fluid boundary.

3.1 Input Data for flow analysis

Input data for flow analysis of the simulation is the assigning of computational domain, boundary conditions and rotating region. Computational domain in the shape of a rectangular prism enclosing the model. The computational domain’s boundary planes are orthogonal to the model’s global coordinate system axes. In this study, the SolidWorks simulation is used to simulate the double-suction pump impeller. This analysis states velocity distribution and pressure distribution of double-suction pump impeller.

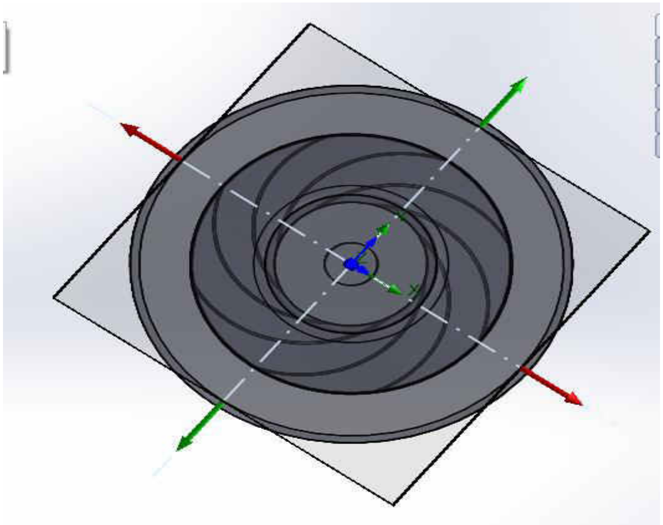

Figure 4.Computatonal Domain of Double-suction Impeller

Analysis type is chosen internal; exclude cavities without flow condition. Physical feature of the rotation is global rotating and Z axis of global coordinate system. And then, angular velocity of the impeller is 155 rad/s as shown in Figure 5 and water is selected for default fluid.

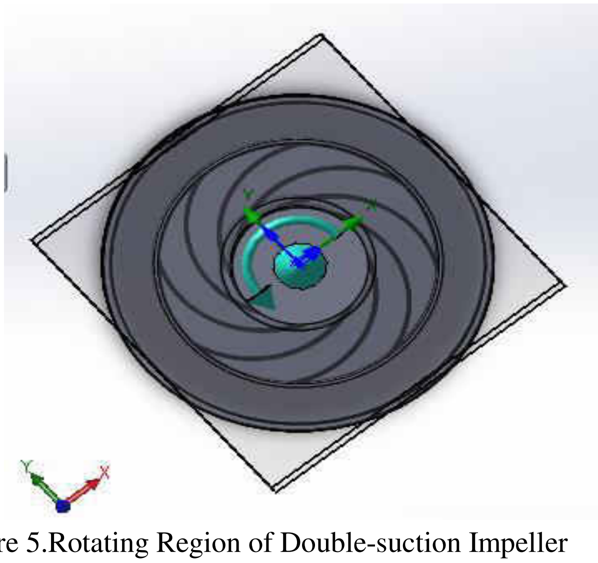

Figure 5.Rotating Region of Double-suction Impeller

3. 2. Boundary Conditions

After assigning the computational domain, the boundary conditions for the inlet and outlet of the double-suction impeller are assigned as shown in figure 6. The inlet volume flow at the entrance of the impeller was given as inlet, the outlet was set as static pressure equal to the environmental pressure.

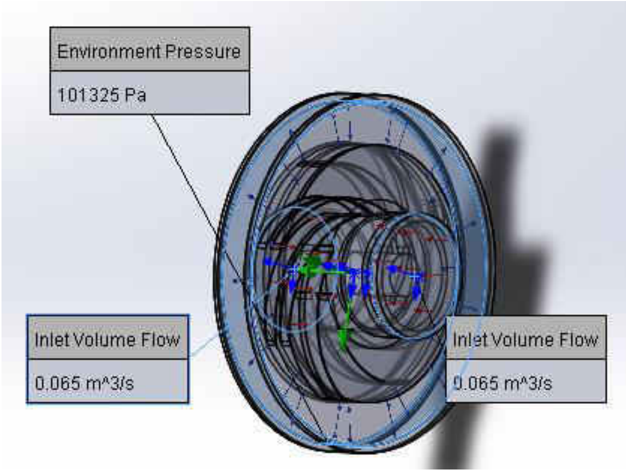

Figure 6.Inlet and Outlet Boundary Conditions

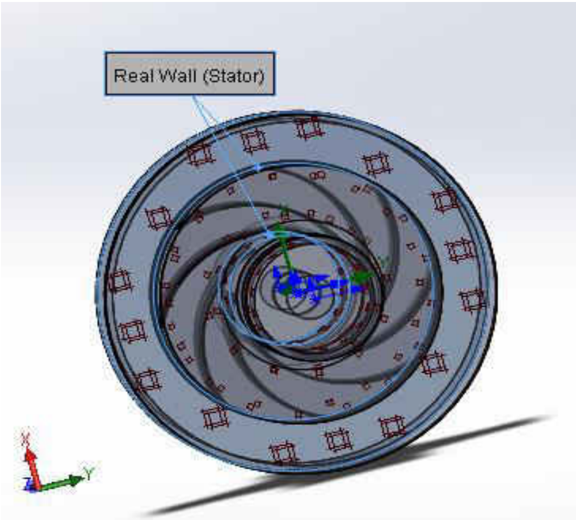

Figure 7. Stationary Wall Condition

3.3. Goal Type

The second step is to choose the simulation goal. In this simulation, the goal for simulation is chosen the surface goal. Since the pressure and volume flow rate boundary condition are specified, it makes sense to set the mass flow rate surface goal at the pump's inlet and outlet.

3.4. Results for Flow Simulation

The last step for the flow analysis in Solid Works is running the impeller design by using the boundary conditions. The simulation results for the velocity and pressure distribution are obtained in figures 8 and 9.

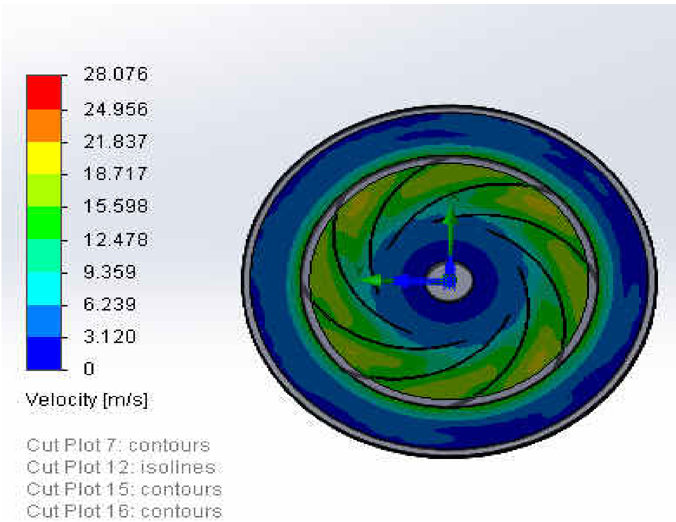

Figure 8. Velocity distribution in the impeller

Table 4

Comparison Between Theoretical and Simulation Result of Velocity Distribution

	Theoretical Result	Simulation Result	% error
Impeller eye velocity (m/s)	3.145	3.12	0.79
Impeller outlet velocity (m/s)	27.78	28.076	1.03

The theoretical result of the impeller eye velocity is 3.145 m /sec and simulation result is 3.12 m/sec. The percent error is 0.79%. The velocity increases from the impeller inlet to outlet and reaches a peak value at the impeller outlet. At discharge of the impeller, the velocity begins to fall down. The maximum value of the velocity is 28.076 m/sec. Figure 8 is the velocity distribution of the impeller. It is slightly decreased in the casing cover, where the kinetic energy of the fluid is transformed into pressure energy.

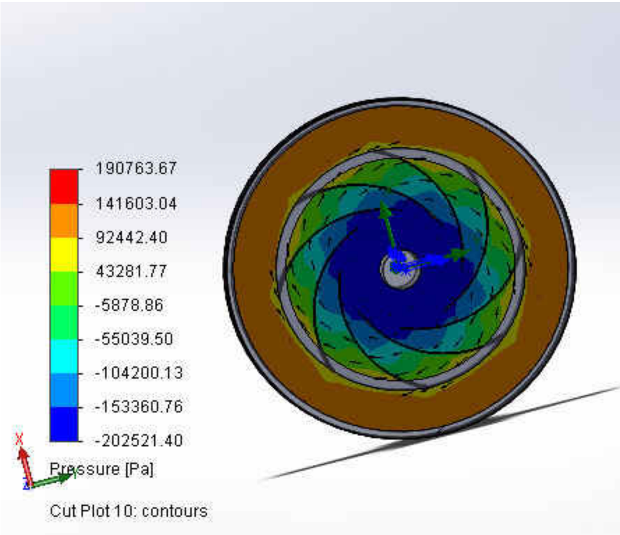

Figure 9. Pressure distribution in the impeller

It is found that the static pressure distribution in the impeller is increasing from impeller inlet to outlet as shown in Figure 9. Moreover, the static pressure on the discharge side is larger than that on suction side of the impeller radius.

The pressure head developed by the pump is calculated as follow.

$$H = \frac{P_{out} - P_{in}}{\rho g}$$

Table 5

Comparison Between Theoretical and Simulation Result of Pressure Head

	Theoretical Result	Simulation Result	% error
Pressure Head(m)	34.03	35.07	3.05

CONCLUSIONS:

The double-suction impeller blade 3D model was generated with SolidWorks Software and analyze in flow simulation of the impeller. In this study, a steady state CFD analysis of backward curved eight blades impeller is carried out. The contour and vector plot of velocity and pressure distributions in the flow passage are displayed. From the study, it was observed that there is a low pressure area at the suction side of inlet impeller.

The minimum pressure is -202521.4 Pa and the maximum pressure reached up to 190763.67 P_a. At the impeller eye, the value of the static pressure is -202521.40 Pa and at the impeller discharge, the value of the static pressure is 141603.04 Pa. The distribution of the pressure in the casing cover is rising.

In this paper, the comparison between design and analysis of the velocity and pressure distributions are found to the similar point. Therefore, it can be seen that the numerical results of this study is nearly the same with the theoretical concepts.

REFERENCES:

1. SolidWorks Corporation: *Driving Better Product Design with SolidWorks Simulation,* (2012), http://www.solidworks.com/sw/wp-driving- product-design-with-simulation.htm.
2. Larry Bachus and Angel Custodio: *Known and Understand of Centrifugal Pump,* Japan: Bachus Company, Tokyo 113, (2003).
3. Igor, J. Joseph, P. and Charles, C.: *Pump Hand Book,* McGraw-Hill Company, Third Edition, USA, (2001).
4. Tuzson, J.2000. "Centrifugal Pump Design". USA: John Wiley and Sons.Inc.
5. Frank M. White: *Fluid Mechanics,* McGraw-Hill Book Co, (1999).
6. Nelik, Lev: *Centrifugal and Rotary Pump, Fundamental with Application,* New York, (1999).
7. S.L.Dixon, B.Eng, Ph.D.: *Fluid Mechanics and Thermodynamic of Turbomachinery,* Fourth Edition, (1998).
8. Kyushu Institute of Technology, *Fluid Mechanics of Turbo Machinery Training Course,* (1996).
9. Val S. Lobanoff, Robert R. Ross et al,: *Centrifugal Pump: Design and Application,* Second Edition, (1992).
10. Kumar, Dr. Ds, Fluid Mechanical and Fluid Power Engineering, 1987.
11. Frank, A. 1953. Pumps. 2nd Edition. Mc Graw-Hill.
12. Daugherty, L. 1951. Centrifugal Pumps. McGraw-Hill Book Company.
13. http://www.mcnallyinstitute.com

BIOGRAPHY:

Mrs. Nwe Ni Win is a Ph.D Researcher from Department of Mechanical Engineering, Yangon Technological University, Yangon, Myanmar. She received Master Degree in Mechanical Engineering from Technological University (Monywa), in 2009. She has participated in the fifth International Conference on Science and Engineering (ICSE) 2014.