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Optimum Design and Structural Analysis of Centrifugal Pump by Varying Outlet Vane Angle

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Abstract: **There are so many parameters effect on pump performance like impeller diameters, width, inlet outlet angles, and blades numbers, many conventional studies of centrifugal pump are largely used of empirical application of empirical rules along with the use of the available information in the form of different types of charts and graphs as proposed by successful designers. Generally all parameters effect on each other to improve the pump performance should study the relationship between the main parameters, we study the effect of outlet blade angle on the pump performance by depended on the analytical methods, we calculate the major parameters for the require design (Q=1000 m3/h, H=40 m) like output, input diameters , width, blade radius, inlet angle, slip factor ,theoretical head throw the range of outlet angles from 22 to 40 degrees after that calculate the efficiency for each angle and change blades numbers "2 to7" increase outlet angle mean the head, discharge, power consumption , increase the blades number mean increase the head and reduce the discharge area lead to increase the discharge . Therefore from all these parameters we find the best efficiency at angle 25o with six blades. Also we design a closed type impeller model by CATIA V5-6R2013 program depended on the calculation data and analysis the model by ANSYS Workbench 14.5 structural and fluent analysis by use stainless steel metal, in the structural analysis applied the calculation torque 837 N.m on inside and the requirement pressure 4 bar on the blades area the maximum stress appear on greave between the shaft and impeller, in the fluent applied on the impeller inlet the fluid water speed 2 m/s and atmosphere pressure then we calculated the distribution of pressure , velocity, velocity vector, wall shear and forces on impeller area.**

Keywords: **Impeller diameters, width, inlet outlet angles, and blades numbers, Optimum Design and Structural Analysis.**

1. INTRODUCTION TO CENTRIFUGAL PUMPS

The hydraulic machines which convert the mechanical energy into hydraulic energy are called pumps. The hydraulic energy is in the form of pressure energy. If the mechanical energy is converted in to pressure energy +by means of centrifugal force acting on the fluid, the hydraulic machine is called centrifugal pump.

The centrifugal pump acts as a reverse of an inward radial flow reacting turbine. This means that the flow in centrifugal pumps is in the radial outward directions. The centrifugal pump works on the principle of forced vortex flow which means that when a certain mass of liquid is rotated by an external torque, the rise in pressure head of the rotating liquid takes place. The rise in pressure head at any point of the rotating liquid is proportional to the square of tangential velocity of the liquid at that point. (Rise in pressure head = $\frac{V^2}{2g} = \frac{w^2 r^2}{2g}$). Thus at the outlet of the impeller, where radius is more, the rise in the pressure head will be more and the liquid will be discharged at the outlet with a high pressure head. Due to this high pressure head, the liquid can be lifted to a high level. [3]

2. ENERGY EQUATION OF IMPELLER PUMPS

$$T = Q\rho(V_{t2}r_2 - V_{t1}r_1)$$

$$P = Tw = \rho g H_{th} Q$$

$$H_{th} = \frac{(V_{t2}r_2 - V_{t1}r_1)w}{g}$$

Applied to pumps, this expression is known Euler's equation. Pumps are usually designed for no angular momentum at the inlet, $V_{t1}r_1 = 0$ .Consequently, the equation can be simplified. [6]

$$H_{th} = \frac{UV_{t2}}{g}$$

Let the points of the liquids path at suction inlet, impeller inlet, impeller outlet and casing outlet be denoted by 0,1,2 and d, respectively.

By applying b Bernoulli's theorem for the fluid flow between two consecutive points, the energy equation can be formed as below. [3]

i)- for flow from point 0 to point 1 ,i.e., through the stationary suction pipe,

$$\frac{V_1^2}{2g} + \frac{P_1}{\rho g} + Z_1 = \frac{V_O^2}{2g} + \frac{P_O}{\rho g} + Z_O - h_{f(0-1)}$$

ii)-For flow from point 1 to point 2, i.e., through the movable impeller, where w_1 and w_2 represent the relative velocities of liquid ,the equation of energy is,

$$\left[\frac{W_2^2}{2g} - \frac{U_2^2}{2g}\right] + \frac{P_2}{\rho g} + Z_2 = \left[\frac{W_1^2}{2g} - \frac{U_1^2}{2g}\right] + \frac{P_1}{\rho g} + Z_1 - h_{f(1-2)}$$

iii)-for flow from point 2 to point d ,i.e., through the stationary casing,

$$\frac{V_d^2}{2g} + \frac{P_d}{\rho g} + Z_d = \frac{V_2^2}{2g} + \frac{P_2}{\rho g} + Z_2 - h_{f(0-d)}$$

Now adding the above three equations and rearranging, we get,

$$\frac{V_2^2 - V_1^2}{2g} + \frac{W_2^2 - W_1^2}{2g} + \frac{U_2^2 - U_1^2}{2g} = \left\{\left(\frac{V_d^2}{2g} + \frac{P_d}{\rho g} + Z_d\right) - \left(\frac{V_o^2}{2g} + \frac{P_O}{\rho g} + Z_o\right)\right\} + h_{f(0-d)}$$

The first term on the right hand side is the gross manometric head (actual head) of the pump, and the second term is the total pump losses due to fluid resistance (or friction losses) inside the pump. Therefore, the above equation can be written as

$$\frac{V_2^2 - V_1^2}{2g} + \frac{W_2^2 - W_1^2}{2g} + \frac{U_2^2 - U_1^2}{2g} = H_{mano} + \Delta H_{mano}$$

This is known as fundamental equation of centrifugal pump. The monomeric efficiency can be calculated as

$$\eta_{mano} = \frac{H_{mano}}{H_{mano} + \Delta H_{mano}}$$

$$\frac{V_2^2 - V_1^2}{2g} + \frac{W_2^2 - W_1^2}{2g} + \frac{U_2^2 - U_1^2}{2g} = \frac{H_{mano}}{\eta_{mano}} = H_{th}$$

The impeller, the above equation can be simplified as below.

$$W_1^2 = U_1^2 + V_1^2 - 2U_1V_1\cos\alpha 1$$

$$W_2^2 = U_2^2 + V_2^2 - 2U_1V_1\cos\alpha 2$$

Therefore,

$$W_1^2 - W_2^2 = U_1^2 - U_2^2 + V_1^2 - V_2^2 - 2U_1V_1\cos\alpha 1 + 2U_1V_1\cos\alpha 2$$

The simplified theoretical head is now,

$$H_{th} = \frac{H_{mano}}{\eta_{mano}} = \frac{U_2V_2\cos\alpha 2 - U_1V_1\cos\alpha 1}{g}$$

Generally, $\alpha 1 = 90$ then The theoretical head derived from above equation is known as Euler equation. This equation clearly indicates that the pump theoretical head is mainly determined by the products of $U_2V_2\cos\alpha 2$. The greater are the products, the higher is the theoretical head. [1]

3. SYSTEM MODELING

$$H = H(Q, N, \eta)$$

$$Q = Q(H, T, N, \eta)$$

$$H = \eta\left(\frac{\pi D}{60}\right)^2 \frac{\sigma}{g} N^2 - \eta\left(\frac{\pi D}{60}\right)\frac{\tan\beta_2}{\pi D B g} QN$$

$$Q = \frac{\eta}{\rho g}\frac{NT}{H}$$

$$H_{th} = \frac{H}{\eta} = \frac{U(U\sigma - W_r\tan\beta_2)}{g}$$

These equations are nonlinear and contain implicitly the head (H) and flow rate Q, as well as the efficiency, which depends on the flow rate. Unlike many electrical components, the practical relationship, characterizing hydraulic components, are mostly nonlinear, and therefore the variables cannot be calculation explicitly. They would have to be solved by trial and error, by progressive approximations. A further complication arises when the equations are time dependent, for example, when the time derivatives or integrals of the variable appear in the characteristic equations. Such cases occur when inertia or compressibility forces appear.

4. BLADE COORDINATES

A list of coordinates the radius r, axial distance z, and circumferential angle θ on several streamlines determines the complex surface of the blade shape. These coordinates are defined on drawings by showing the intersection curves with the hub, the shroud, and also a curve along the mid-streamline. In impellers with two-dimensional blades, the hub and shroud lines coincide, the axial coordinate z is constant, and the other two coordinates suffice to define the blade shape. One the variables, usually r or z is selected as the independent variable. Its extend, from inlet to exit, is divided into 10 or 20 equal increments. Before the value of the other variables can be calculated at each station, the desired stepwise increases in the angular momentum rV_t along the streamlines need to be specified. The incremental change in the angular momentum corresponds to the pressure difference exerted on the blade, the blade loading, it ideally should increase gradually from the inlet, reach a maximum at the middle of the blade length, and taper off at exit (Tuzon 1993). Heavy blade loading at the inlet may increase inlet losses. Heavy blade loading at the exit may create exit losses, pressure fluctuations, and noise .therefore, the angular momentum is gradually incremented from the inlet to its value, R_2V_{t2} . the local blade angle, measured from the radial direction or from the meridional direction in mixed flow impellers, can be calculated from the following geometric relationship,

$$\tan\beta = \frac{r^2 w\sigma - rV_t}{rw_m} \qquad 3\text{-}69$$

Where the angular momentum rV_t is a stepwise-increasing fraction of its value at the impeller exit.

$$R_2V_{t2} = \frac{gH}{\eta w}$$

The slip σ appears only toward the exit, and gradually decreases from 1, at a distance from the periphery along the blade equal to the blade spacing to its value at the exit. If one were to assume an eddy relative to the impeller between the blades, which causes the slip, its effect would extend over a distance on the order of the blade spacing. The value of the meridional or radial relative velocity W_m varies at inlet from hub to shroud, and has been calculated when the inlet angle were determined. After the inlet bend, after the flow has turned radial-or approximately radial in mixed flow impellers-the relative meridional flow velocity can be assumed to be uniform across the flow passage and can be calculated from the flow rate and the passage cross-sectional area, corrected for blade and boundary layer blockage. It should be apparent that a computer program, calculating the meridional velocity in the impeller, would facilitate the design. However, since several iterations and adjustments are needed to arrive at a satisfactory blade shape, the informed judgment of the designer remains in-dispensable and may override theoretical considerations. Having the blade angle β distribution along the blade, the increment of the variables Δr, Δz and $\Delta\theta$ can be calculated from. [6]

$$\tan\beta = r\frac{\Delta\theta}{\Delta m}$$

Where

$$\Delta m = \left(\Delta r^2 + \Delta z^2\right)^{1/2}$$

5. MATHEMATICAL CALCULATION FOR THE CASE STUDY

1. Calculate pump specific speed

$$N_s = \frac{1450 * \sqrt{1000/3600}}{40^{3/4}} = 48.07460 RPM$$

2. Calculate the input and output power

$$P_o = 1000 * 9.806 * 40 * 0.2778 = 109 kW$$

$$P_s = 109/0.86 = 126.7 kW \text{ say 127kW}$$

3- Calculate input torque to pump

T=127000*60/2*3.14*1450 =837 N.m

4-Calculate the shaft diameter

$$T = 71620\frac{127*1000}{746*1450}746*1450 = 8343 kg.cm$$

$$d = \frac{16*818.16}{\pi 40000000} = 47.05 mm \text{ say 50mm}$$

5- Calculate minimum speed

$$RPM = 10\frac{40^{3/4}}{\sqrt{1000/3600}} = 300RPM$$

6-Calculate the inlet impeller diameter

$$D_1 = 1.533\left(\frac{1000*60}{3600*2*1450}\right)^{1/3} = 187.488mm \text{ say } 187.5\text{mm}$$

6-Calculate inlet impeller width

$$b_1 = \frac{(1000/3600)}{\pi*0.15*0.1875*14.23} = 220.9mm \text{ say } 221\text{mm}$$

7-Calculate inlet velocity

$$U_1 = \frac{\pi*0.1875*1450}{60} = 14.23m/s$$

8-Calculate outlet velocity

$$U_2 = \sqrt{\frac{9.81*40}{0.6*0.86}} = 27.5m/s$$

9-Calculate the outlet impeller diameter

$$D_2 = \frac{60*27.5}{\pi*1450} = 363.2mm \text{ say } 365\text{mm}$$

10-Calculate outlet impeller width

$$b_2 = \frac{(1000/3600)}{\pi*0.15*0.365*27.5} = 58.56mm \text{ say } 59\text{mm}$$

11-Calculate blade number

$$r_m = 182.5 + \frac{365-187.5}{4} = 0.226875m$$

$$Z = 13\frac{226.875}{88.75}\sin 9 = 6.3 \text{ say } 6$$

12-Calculate bald radius

$$R = \frac{182.5^2 - 93.75^2}{2(182.5\cos 22 - 93.75\cos 35)} = 132.6mm$$

13-Calculate pipe diameter

$$p_d = \sqrt{\frac{4*1060}{3600*\pi*5.926}} = 251\text{mm say } 250\text{mm}$$

14-Calculate tongue angle

$$\alpha = \frac{0.031838}{\pi*0.376*1.75*0.0511} = 17.24^o$$

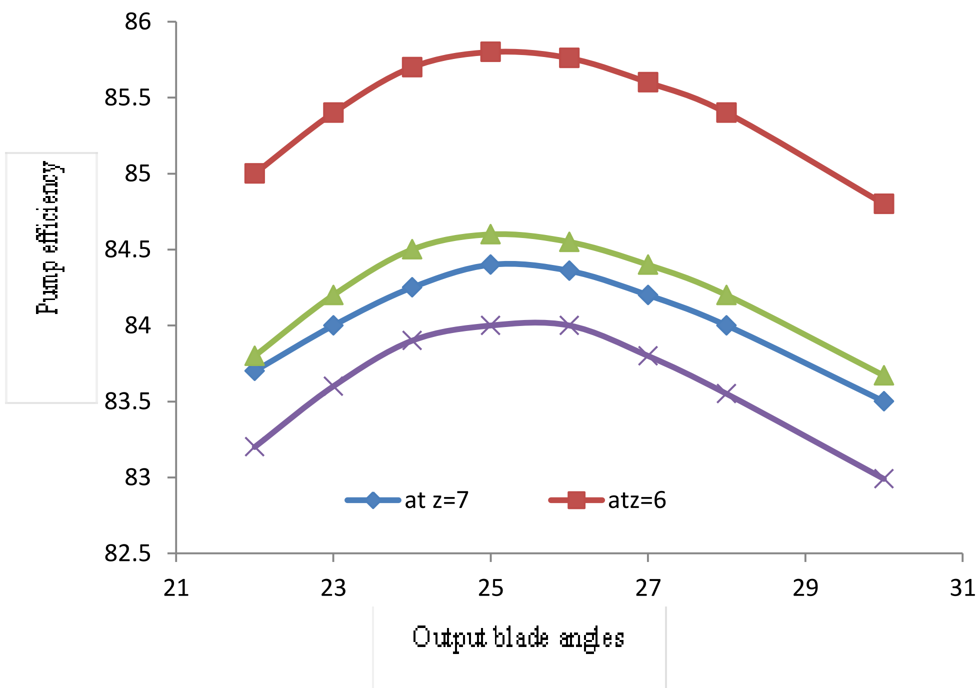
15-Calculate inlet blade ingle

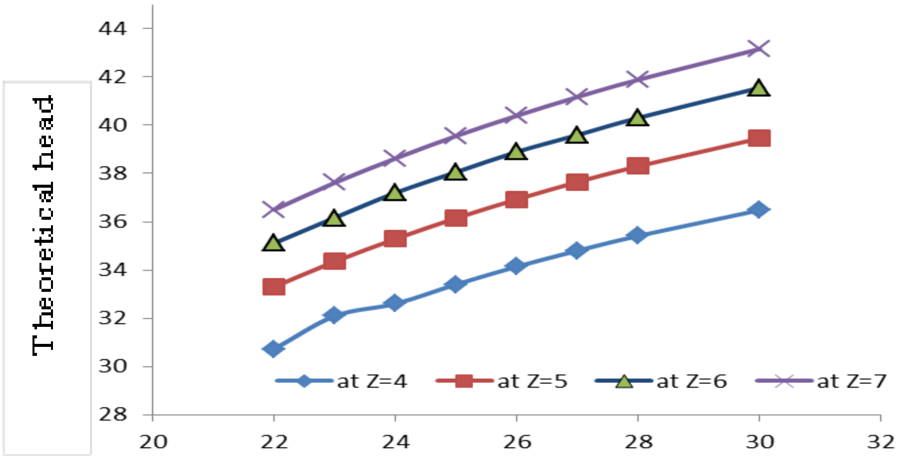
$$\beta_1 = \tan^{-1}\frac{1000/3600}{\pi * 0.1875 * 0.221 * 14.23} = 8.5^0 \text{ Say } 9^0 \text{ degrees}$$

Q=1000m³/h, N=1450 r.p.m , Z=6										
	β_2	22°	23°	24°	25°	26°	27°	28°	30°	40 °
σ		0.825	0.821	0.818	0.814	0.811	0.807	0.804	0.798	0.771
H_{th}		35.11	36.17	37.20	38.06	38.90	39.60	40.31	41.56	45.68
η		86.1	86.2	86.3	86.4	86.3	86.3	86.2	86.1	83.1

Q=1000m³/h, N=1450 r.p.m , Z=7										
	β_2	22°	23°	24°	25°	26°	27°	28°	30°	40 °
σ		0.84	0.8399	0.8366	0.8335	0.8304	0.8274	0.8245	0.8189	0.794
H_{th}		36.5	37.62	38.63	39.56	40.40	41.18	41.89	43.17	47.45
η%		84.8	84.9	85.0	85.1	85	85	84.9	84.8	81.1

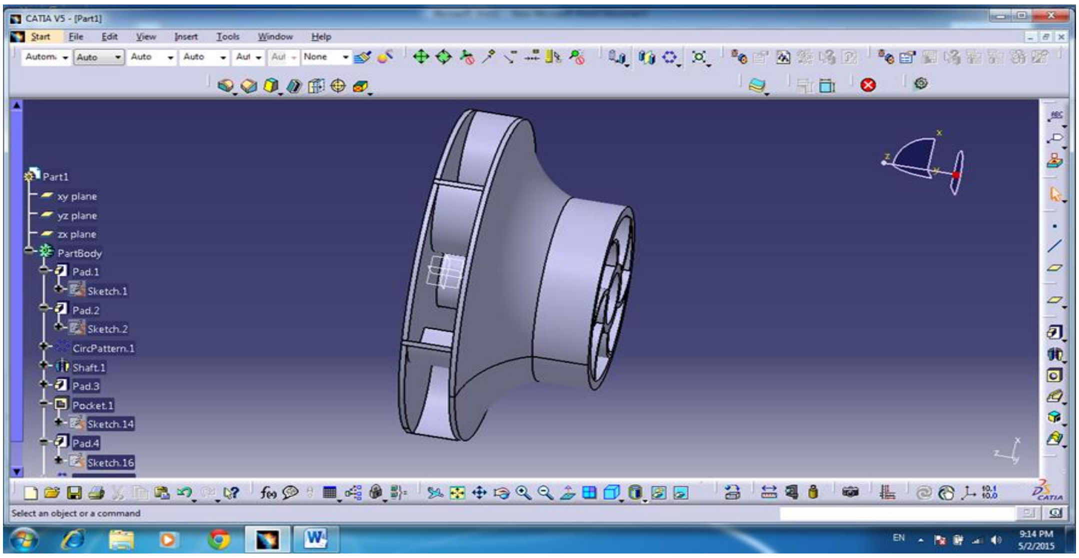
Q=1000m³/h, N=1450 r.p.m , Z=5										
	β_2	22°	23°	24°	25°	26°	27°	28°	30°	40 °
σ		0.8016	0.7974	0.7932	0.7892	0.7854	0.7816	0.7779	0.7708	0.740
H_{th}		33.30	34.35	35.29	36.15	36.93	37.65	38.30	39.46	43.29
η		84.3	84.4	85	84.6	84.5	84.4	84.3	84.2	82.6

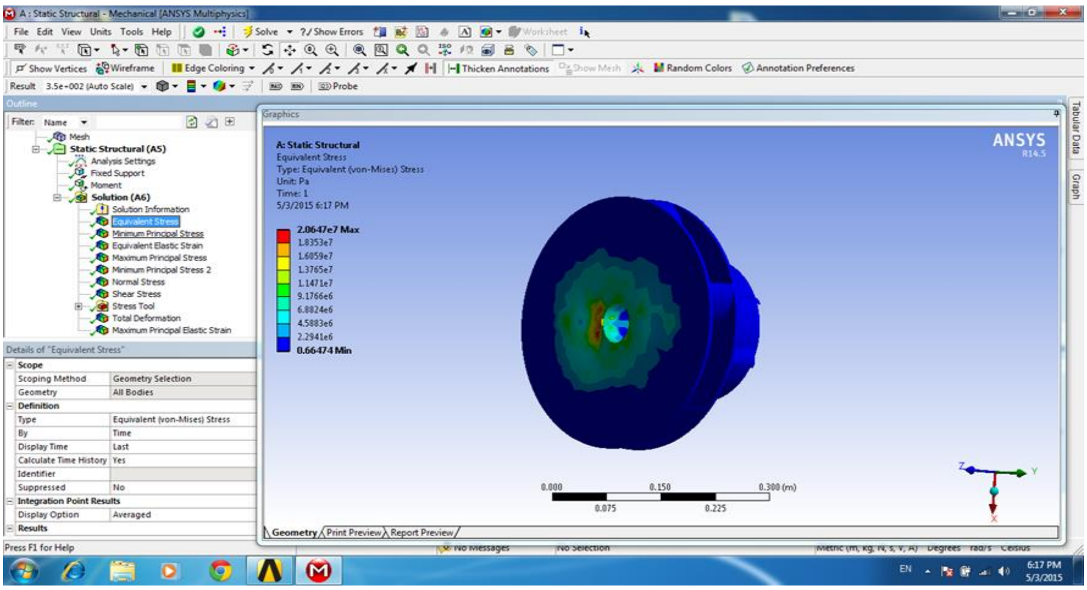




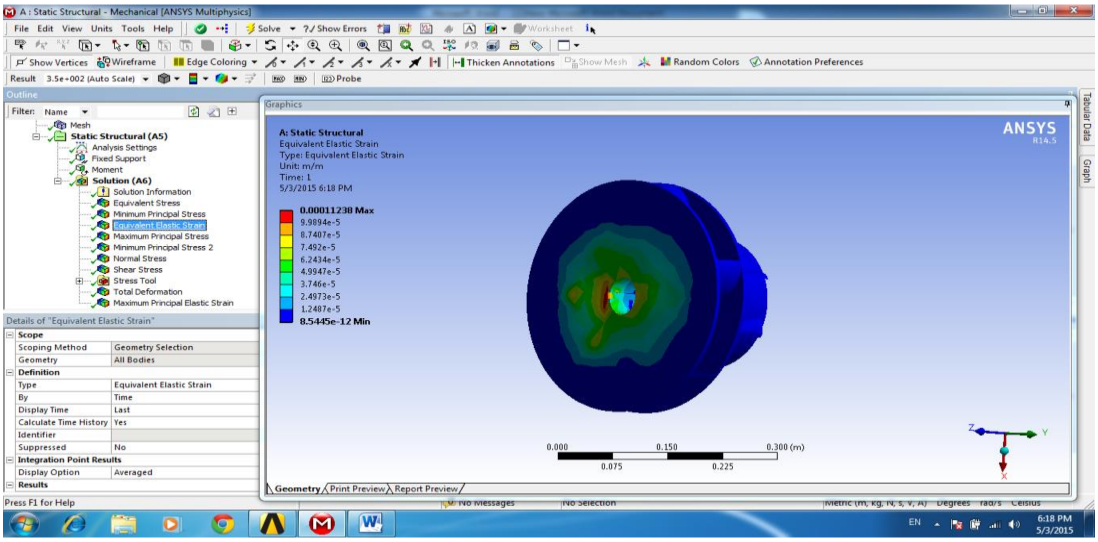
Output blade angles

6. STRUCTURAL ANALYSIS

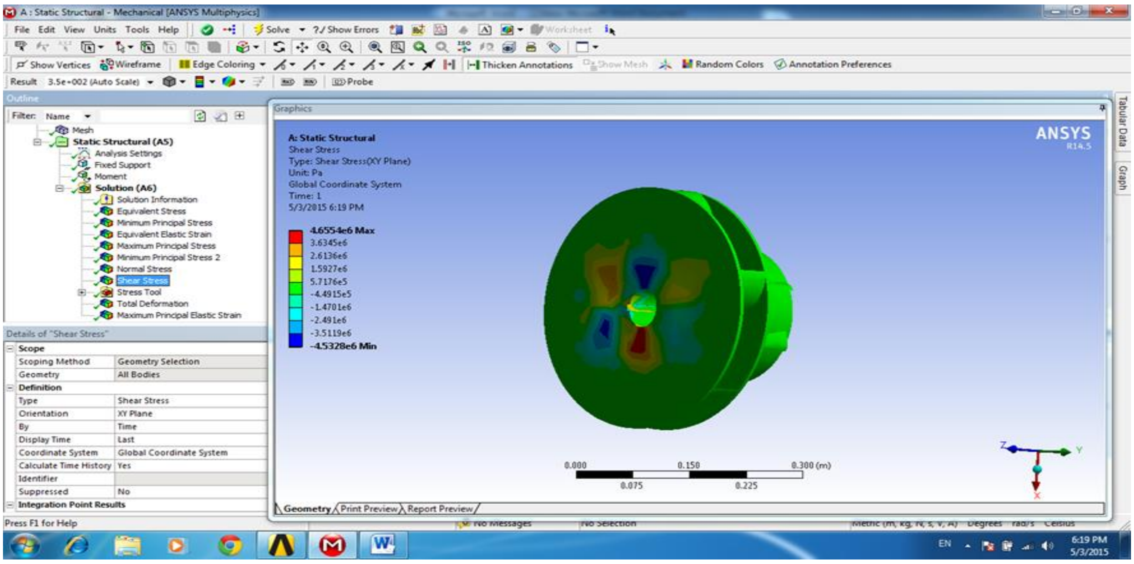




Von-Mises stress on impeller

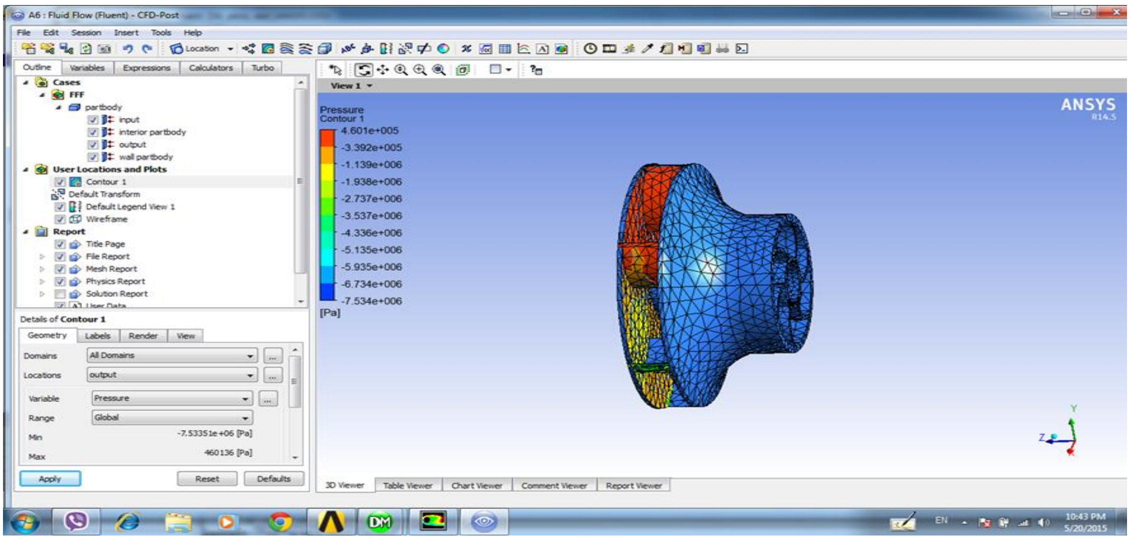


Equivalent stress on impeller

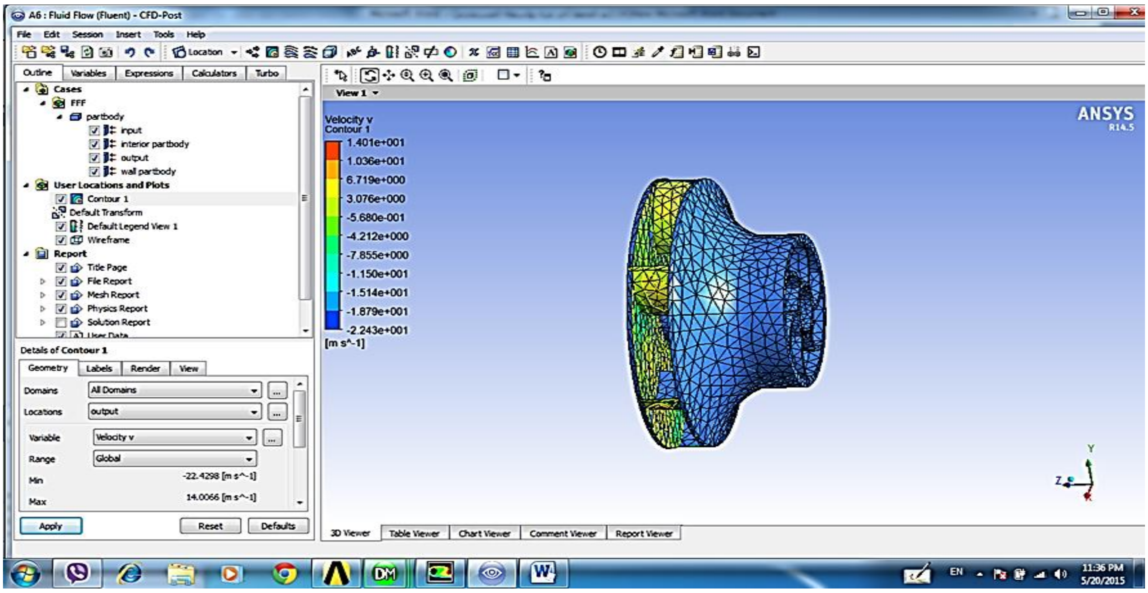


The shear stress on XY plain for the impeller

7. COMPUTATIONAL FLUID DYNAMIC (CFD) FLUENT ANALYSIS



Output result (pressure) by ANSYS (fluent)



Output result (fluid velocity) v component by ANSYS (fluent)

8. RESULTS DISCUSSION

There are four major parts represent impeller inlet, outlet blades and two side shroud show in fig (6-1,6-2) by CATIA, the outlet blades attach to the two shroud with 25 degree angle at outlet.

the torque applied by the shaft therefor the maximum stress appear on the impeller slit between the impeller and shaft, the slit consist from three sides the load applied only to one side the area of it (1540) mm^2 fig (6-5,6-6), the load applied suddenly on this area from the motor torque, therefor this area of the side for slit carry the maximum load and temporary sudden torque, this torque directly proportional with the output pressure and any sudden disturbance led to increase the pressure, this mean should design the slit with sufficient effective area, thickness and corners, any fail in this small area means loss the impeller.

The structural analysis appear the maximum Von-Mises stress , Equivalent stress , on the side the width is 7mm and length 220 mm the maximum stresses is suitable compare with metal yield stress, the shear stress on XY plain for the impeller for the shroud it has the maximum stress around the slit and hub diameter that it came from the source of the force, generally the yield stress for the impeller metal divided by the load (principle stress) is more than one, the above data mean correct design.

In the fluent analysis this analysis mean study the fluid properties between input and output impeller inside the impeller and on the surface of exit blade many properties effect on the impeller performance we study the (pressure, discharge ,velocity vector, wall shear),these parameters has the major effect on the pump efficiency, the pressure distribution in fig 6-18 show that the maximum pressure on the end of outlet blades is 0.46 MPa more than the required pressure with 0.15% it is permissible rang because there are many empirical equation it has some error in design like friction, roughness, viscosity , all these parameter we neglected it in analytical calculation, the velocity vector distribution appear that the maximum with the direction tangent to exit blade (u direction) with the x axes because this blade represent the source of the motion it push the fluid with the tangent vector to reduce the losses to the minimum by design the outlet fluid with this direction.

9. CONCLUSION

Generally the outlet blade angle has the major effect on pump performance and other parameters increase the output blade angle mean increase theoretical head and it has direct effect on the efficiency, number of blades effect on head, discharge, balancing finally it has major effect on the centrifugal pump efficiency ,the impeller speed has direct correlation on pump power, head, discharge, vibration and performance the ANSYS program appear von miss stress distribution on the impeller surface, blades, shaft and volute also calculated pressure distribution in the impeller and volute , the maximum stress appear on the grave between the shaft and impeller because the concentration moment on this small area , and at the end of outlet blades this point carry the maximum load because the pressure in this point represent the pump pressure , the load increase correlation with the shroud the two sides from the internal radius to the external radius at the second one represent the maximum load. The calculation show the maximum performance ,efficiency at the output blade angle (24-25)^{o} ,with six blades only , increase the blade number mean increase the head of discharge and in same time lead to reduce the discharge because reduce the area of flow.

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