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INFLUENCE OF NUMBER OF IMPELLER BLADES ON THE PERFORMANCE OF CENTRIFUGAL PUMPS BY USING ANSYS SOFTWARE

N.Govendan[1]

[1]*PG Scholar -M.E(CAD/CAM),Department of Mechanical Engineering,S.R.V.Engineering College,Sembodai-Nagapattinam district,Tamil nadu,India -614809.*

P.Siva kumar [2]

[2] *Assistant professor ,Department of Mechanical Engineering ,S.R.V.Engineering College ,Sembodai-Nagapattinam district -Tamil nadu,India-614809.*

T.N.Anbarasu [3]

[3] *Assistant professor ,Department of Mechanical Engineering ,S.R.V.Engineering College ,Sembodai-Nagapattinam district -Tamil nadu,India-614809.*

I.Justin antony raj [4]

[4] *Assistant professor ,Department of Mechanical Engineering ,S.R.V.Engineering College ,Sembodai-Nagapattinam district -Tamil nadu,India-614809.*

ABSTRACT

Purpose:*The purpose of this work is to investigate the influence of number of blades on the performance of the centrifugal pump.*

Design/Methodology/Approach :*In this study ,the centrifugal pumps were developed by varying the number of blades without varying other parts.The centrifugal pumps were developed as per the industrial procedure.The characteristics of the centrifugal pumps were investigated by using the ANSYS fluent software .The space between the blades and flow stream were analysed by ANSYS.Hydraulic efficiency was analysed by ANSYS.Impeller with efficiency was analysed by ANSYS software.*

Findings:*The experimental results indicate that the 8 blades impeller produced optimum efficiency compare with other blades.*

Originality/Value: *This paper explains the effect of number of impeller blades in centrifugal pumps for enhancing the efficiency of the pump.*

KEYWORDS:*Centrifugal pump,ANSYS,Number of blades,Efficiency,Hydraulic efficiency.*

1. INTRODUCTION

Impeller blades play a crucial role in centrifugal action ,accelerates the liquid to a high velocity ,transferring mechanical (rotational) energy to liquid and that converts driver energy into the kinetic energy .The low number of impeller blades are used commonly in centrifugal pump,but it has some disadvantages that tends to equivalent stresses increases ,total deformation increases and efficiency reduces .To provide optimum efficiency ,the centrifugal pumps are used with higher number of blades in impeller .The number of blades in impeller increases the efficiency, head is increasing ,equivalent the stress ,deformation reduces and the diffuser loss is decreases.E.C.Bacharoudis et.al assessed the various parameters which affect the pump performance and energy consumption like the impeller outlet diameter,the blade angle ,the blade number and evaluated the performance of impellers with the same outlet diameter having different outlet blades angles.Andrzej wilk et.al studied the measurements of parameters of a high speed impeller pump with open flow impeller having radial blades.They found that at high rotational speed pump obtained a large delivery head because the blade angle at outlet from the impeller was wide,liquid flowing out the impeller was large obsolute velocity and dynamic delivery head of the impeller was large.The kinetic energy of the liquid was converted to pressure in spiral case and in the diffuser.R.R.Singh et.al studied the characteristics of low specific speed centrifugal water pump by studying the relationships among impeller eye diameter ,vane exit angle and width of the blade at exit.Ashok Thummar et.al studied the various performance parameters of centrifugal pump such as over all efficiency ,cavitation ,slip factor ,losses calculated .Ling Zhou et.al studied the impeller design parameters and original pump were acquired and compared with the data predicted from numerical simulation ,which presents a good agreement under all operating conditions.Although few literature focuses on the impeller blades ,the effect of blades as varying ingredients is unexplored .Thus the current study dealt with the development of increasing number of blades (4,5,6,7,8,9,10) based centrifugal pump and to characterize them as per the standard industrial practice.

2. MATERIALS AND METHODS

2.1. Materials

Cast iron ,plastics,steel and stainless steel alloys ,aluminium ,brass ,bronze,ceramics,nickel alloy,grey cast iron ,copper base alloys,nickel copper alloys,stainless steels SS304,SS316,ferritic grade 410(12%cr) ,430(17% cr),duplex stainless steels UNS31803.

2.2.Formation and designation of the centrifugal pump.The developed centrifugal pump possessed 4,5,6,7,8,9,10 impeller blades .The broad categories of impeller blades in centrifugal pump is given in table 2.The developed pumps were designed to suit for Indian pump scenario .

2.3. Development of centrifugal pumps.

The methodology involved in the development of centrifugal pumps are given below table.1.The preparation of the centrifugal pumps with different impeller blades (4,5,6,7,8,9,10) designed through by ANSYS software.

2.4. Characterizations

For Compressor impeller 3 materials investigation was done using structural analysis and modal analysis ,for turbine impeller 3 materials investigation was done using structural analysis, modal analysis and thermal analysis. The variation of von misses stress, Von misses strain and deformation for three different materials of compressor impellers, using structural analysis.

In order to determine the best shape for yields the best performance in a centrifugal pump, about it was necessary to have basic shape and blade-angles of the impeller. The classical method for obtaining such data was based on the basic pump-theory, in which the sought parameters were derived from specifications of performance. Inlet blade-angle and exit blade-angle were usually determined by the fluid angle of the inlet and outlet, according to parameters. The sweep angle, which was related to the blade-length, was a dependent variable and not an important design parameter of impeller, if a smooth blade-angle distribution was employed.

From the combination of the shape of impeller can be described by a vane plane, and the length and the angle of the blade at the impeller inlet and exit can readily be derived from the vane plane. Because the vane plane determines the shape of impeller and affects the performance of pump, it was essential to understand how the performance of the pump was affected by several design parameters and also to compute the best values for each parameter.

In our study, we defined the parameters of the vane plane were allowed to vary, but the shape of meridional geometry of impeller was fixed, thus allowing for a systematic analysis of the effect of the parameters on performance. The analysis was based on 2k factorial designs, with numerical analysis done through ANSYS instead of actual experiment, and consisted of examining changes in performance for each impeller shape. The parameters for best performance were computed by using the response surface method.

2.5. PERFORMANCE OF CENTRIFUGAL PUMP

Inlet diameter (D_1) = 100 mm

Outlet diameter (D_2) = 290 mm

Speed of the impeller (N) = 1200 rpm

Width of blade at inlet (B_1) = 25 mm

Width of blade at outlet (B_2) = 23 mm

No of blades (n) = 8 Numbers

Manometric head (H_{mano}) = $h_s + h_d = 36$ m

Inlet angle (θ) = 30^0

Outlet angle (φ) = 27^0

Tangential velocity at inlet (U_1) = $\frac{\pi D_1 N}{60}$

= $\frac{\pi x 0.1 x 1200}{60}$

= 6.283 m/sec

Tangential velocity at outlet (U_2) = $\frac{\pi D_2 N}{60}$

= $\frac{\pi x 0.29 x 1200}{60}$

= 18.22 m/sec

Whirl velocity at inlet (V_{w1}) = 0

Whirl velocity at outlet (V_{w2}) = $U_2 - \frac{V_{f2}}{tan\emptyset}$

Where Flow velocity at inlet = Tanθ $= \frac{V_{f1}}{U_1}$

➔ V_{f1} = $Tan 30^0 x 6.283$

= 40.244 m/sec

➔ V_{f2} = V_{f1}

= 40.244 m/sec

Whirl velocity at outlet V_{w2} = $U_2 - \frac{V_{f2}}{tan\emptyset}$

= $18.22 - \frac{40.244}{tan 27^0}$

= 5.926 m/sec

Discharge water at outlet Q = $\pi V_{f2} D_2 B_2 n$

= π x40.244x0.29x23x8

= 6746.32 m^3/sec

Manometricefficiency(η_{mano})

η_{mano} = $\frac{manometric\ head}{head\ imparted\ by\ the\ impeller\ to\ liquid}$

η_{mano} = $\frac{gH_{mano}}{V_{w2}u_2}$

= $\frac{9.81x36}{5.926x18.22}$

= 81.77 %

Volumetric efficiency

η_v = $\frac{liquid\ discharged\ per\ second\ from\ the\ pump}{quantity\ of\ liquid\ passing\ per\ second\ through\ the\ impeller}$

η_v = $\frac{Q}{Q+q}$

= $\frac{6746.32}{6746.32+30}$

= 97.42 %

Overall efficiency

The ratio of power output of the pump to the power input the pump is known as overall efficiency.

η_o = $\frac{power\ output\ of\ the\ pump}{power\ input\ to\ tne\ pump/shaft}$

η_o = $\frac{wQH_{mano}}{P}$

= $\frac{1x6746.32x36}{84.823}$

= 81.52 %

CALCULATION OF STRESS AND DEFLECTION

Impeller Weight = lbtρ

= 0.025x0.8x0.003x 7.845

= 0.4491 Kg

I_{tw} = 0.4491x8

= 3.59 Kg

Shaft Weight = $\frac{\pi}{4}$ x d^2 x l x ρ

= 1.469 kg

Shaft Power = 84.823 KW,

M.I of Shaft = 1.5558 x 10-5 m^4

Torque developed due to power transmission

T = $fxd/2$

= 54x12.5

= 675 N-m

Power transmitted by the shaft

P = $\frac{2\pi NT}{60}$ watts

= $\frac{2\pi x1200x675}{60}$

= 84.823 KW

Shear stress induced from the shaft

τ = $\frac{Tx16}{\pi xd\ \hat{}3}$ N/m^2

= $\frac{675x16}{\pi x0.25\hat{}3}$

= 11140000 N/m^2

= 11.14X10^6 N/m^2

= 11.14 MPa

DETERMINATION OF THE DEFLECTION IN THE SHAFT

Deflection of Shaft due to self-weight (δ1):

δ1 = $5WL^4/384EI$

= $5x1.469x1^4/384x210x10^3x1.5558x10^{-5}$

δ1 = 0.00587 m

Deflection of Shaft due to Impeller weight (δ2):

δ2 = $WL^3/48EI$

= $8x0.449x1^3/48x210x10^3x1.5558x10^{-5}$

δ2 = 0.00419 m

Total deflection (δ)

δ = δ1+ δ2

= 0.00587+0.00419

= 0.001006m

= 10.06 mm

3. RESULTS AND DISCUSSIONS

3.1. IMPELLER GEOMETRY

The shape of the impeller may be represented by a meridional view and a frontal view. Fig. 1(a) is a meridional view and represents the direction of axis and radius of the impeller. Fig. 1(b) is a frontal view showing the angle of impeller and represents the radius and rotational direction; and Fig. 1(c) is a vane plane development that is made by combination of the meridional view and front view of impeller. The angles of inlet and exit as well as the length of impeller are readily included in Fig. 1(c)

In such design, if the sweep angle of the blade is fixed for given inlet and exit angles, it is not possible to smoothly connect the inlet and exit blade angles of the impeller. In order to get a smooth connection for given sweep-angle and inlet and exit blade-angles, we proposed a modification of the traditional design method. We get a new design method for a fixed by using a Bezier curve to maintain a smooth curve in the vane plane.The Bezier curve, shown as the line in the middle, is a curved line that smoothly connects the start and end points by using control points,the shape of the curved line can be changed with the position of each control point. If the control points are determined as shown in Fig. 2, then it is possible to get a smooth curved line of blade for given inlet and exit angles.The fixed meridional shape in the present study is a centrifugal pump with six blades, and the specific speed (rpm, m3/min, and m)is 280. The rotational speed of the pump is 593rpm, the operating flow rate is 8145CMH, and the total head is 70m.

The number of blades for radial impeller of centrifugal pump is taken up to 10. In this paper the numerical analysis has been carried out for a number of impeller using different number of blades, but the impeller size, speed and blade angle being identical. From the table x- it is easily visible that with the increase of blade number the head is increasing and equivalent stress, total deformation should be reduced. With the increases of blade number, the head grows all the time, and the static pressure too. If the number of blades of impeller is infinite, then only the ideal head is developed by the impeller.

If blade number is too more, the crowding out effect phenomenon at the impeller is serious and the velocity of flow increases, also the increases of interface between fluid stream and blade will cause the increment of hydraulic loss; because the greater the number of blades, the more will be the area of obstruction which means the frictional losses will be greater and the passage between the blades will be chocked by undesirable material passing through the impeller. If the blade number is too few, the diffuser loss will increase with the grow of diffuse extent of flow passage.

The equivalent stresses and total deformations are analysed through figures 3-16.To study the distribution of equivalent stress in the impeller blades mapping for the stresses of 4,5,6,7,8,9,10 blades were done and given in figure 17 .In figure 17 the value 11.14 Mpa denote the minimum equivalent stress in Mpa about 8 blades comparing with other number of blades.To study the distribution of total deformation in the impeller blades of centrifugal pump mapping for the deformation of 4,5,6,7,8,9,10 blades were done and given in figure 18 .In figure 18 the value 10.06mm denote the minimum total deformation in mm about 8 blades comparing with other number of blades.

We can see that at 8 numbers of blades the efficiency is more than the other numbers of blades for the pump. The efficiency is dropping at 10 numbers of blades,that with the increase of blade number the head is increasing and equivalent stress, total deformation should be reduced. So the optimum efficiency for 8 blade number is showing better results compared to other blade numbers.

4. CONCLUSIONS

The centrifugal pumps were developed using different number of blades and tested for various properties .Based on the analysis results the following inferences were drawn .

- The limitation of space between blade and flow stream got increased with 8 blades centrifugal pump.
- The area of low pressure region at the suction of the blade and static pressure was increased gradually.
- The uniformity of static pressure distribution at diffuser section better and better.
- The efficiency for 8 blades showed optimum desired results compared to other number of blades.
- The total pressure at suction side got increased with 8 blades impeller in centrifugal pump.
- The head of centrifugal pump got increased with 8 blades impeller in centrifugal pump.

Thus ,The 8 blades impeller would be positive solution for meeting out the present scenario trend in centrifugal pump applications.

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FIGURES AND TABLES

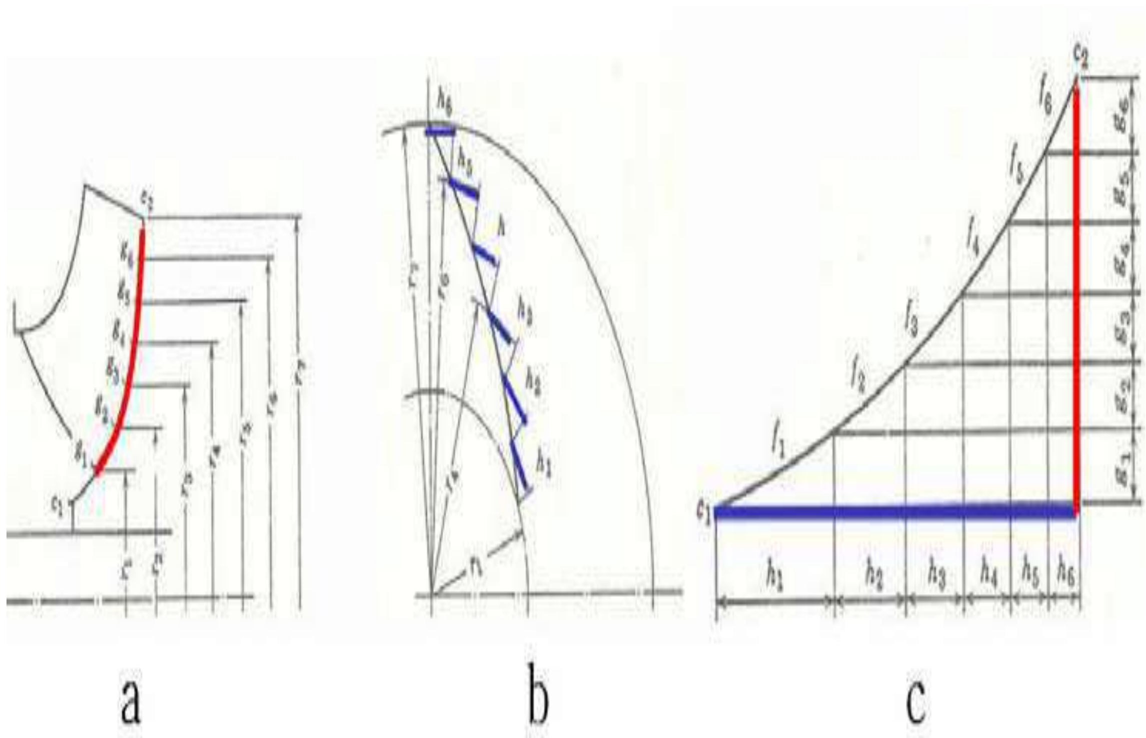

Fig 1. Traditional impeller design method a) meridional view, b) front view, c) vane plane development

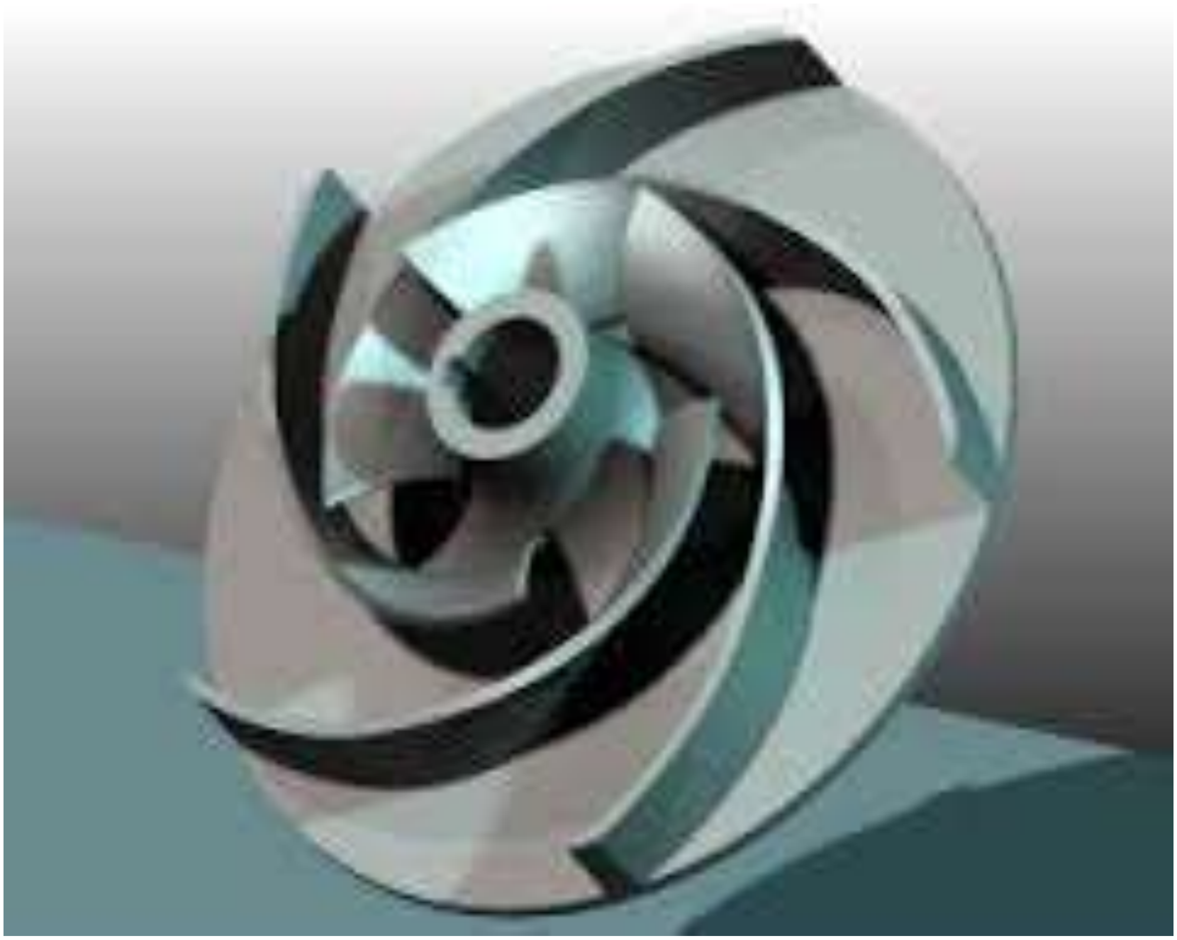

Fig 2. Simple solid model of impeller

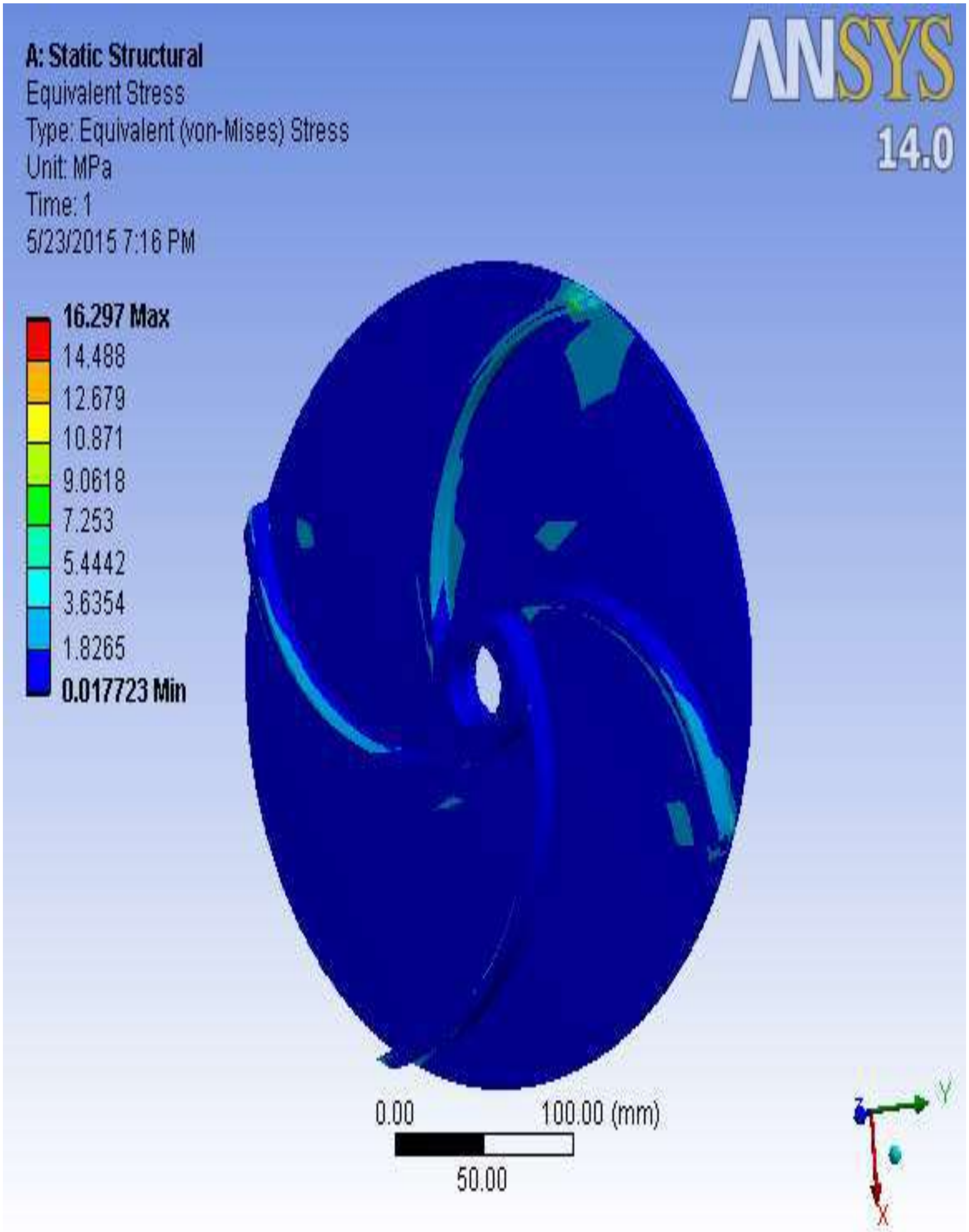

Fig 3. Equivalent stress of four blade impeller

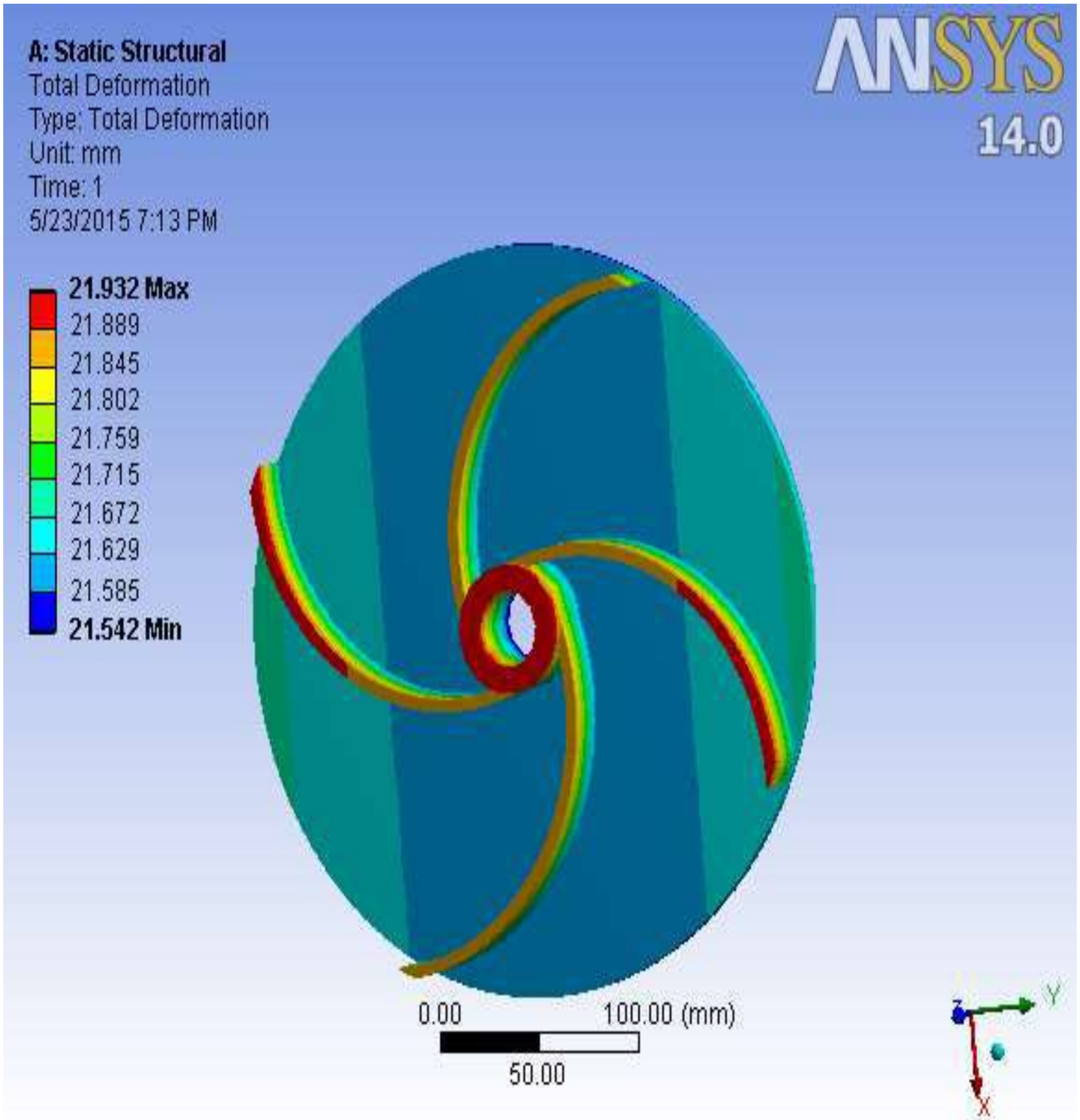

Fig 4. Total deformation of four blade impeller

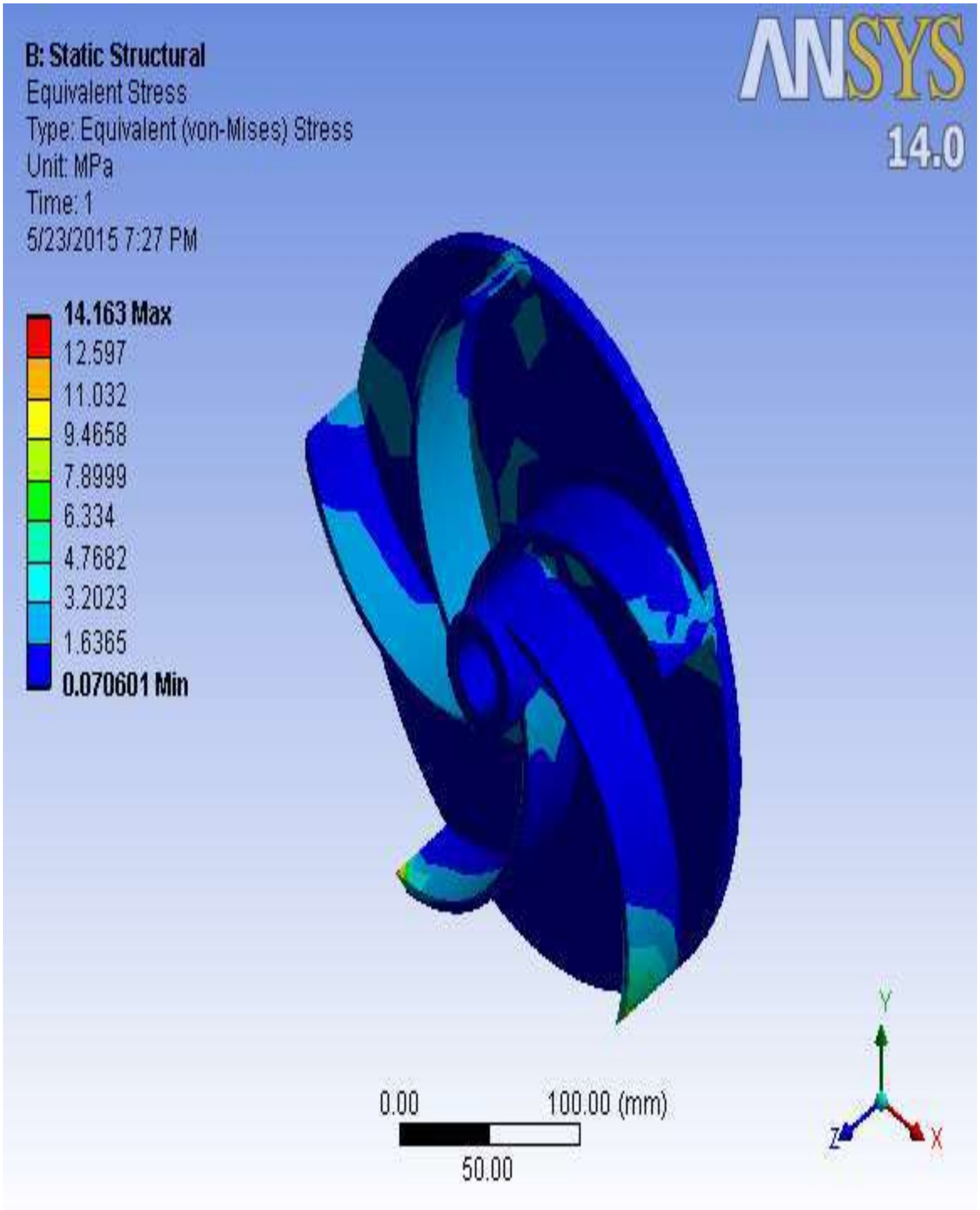

Fig 5 . Equivalent stress of five blade impeller

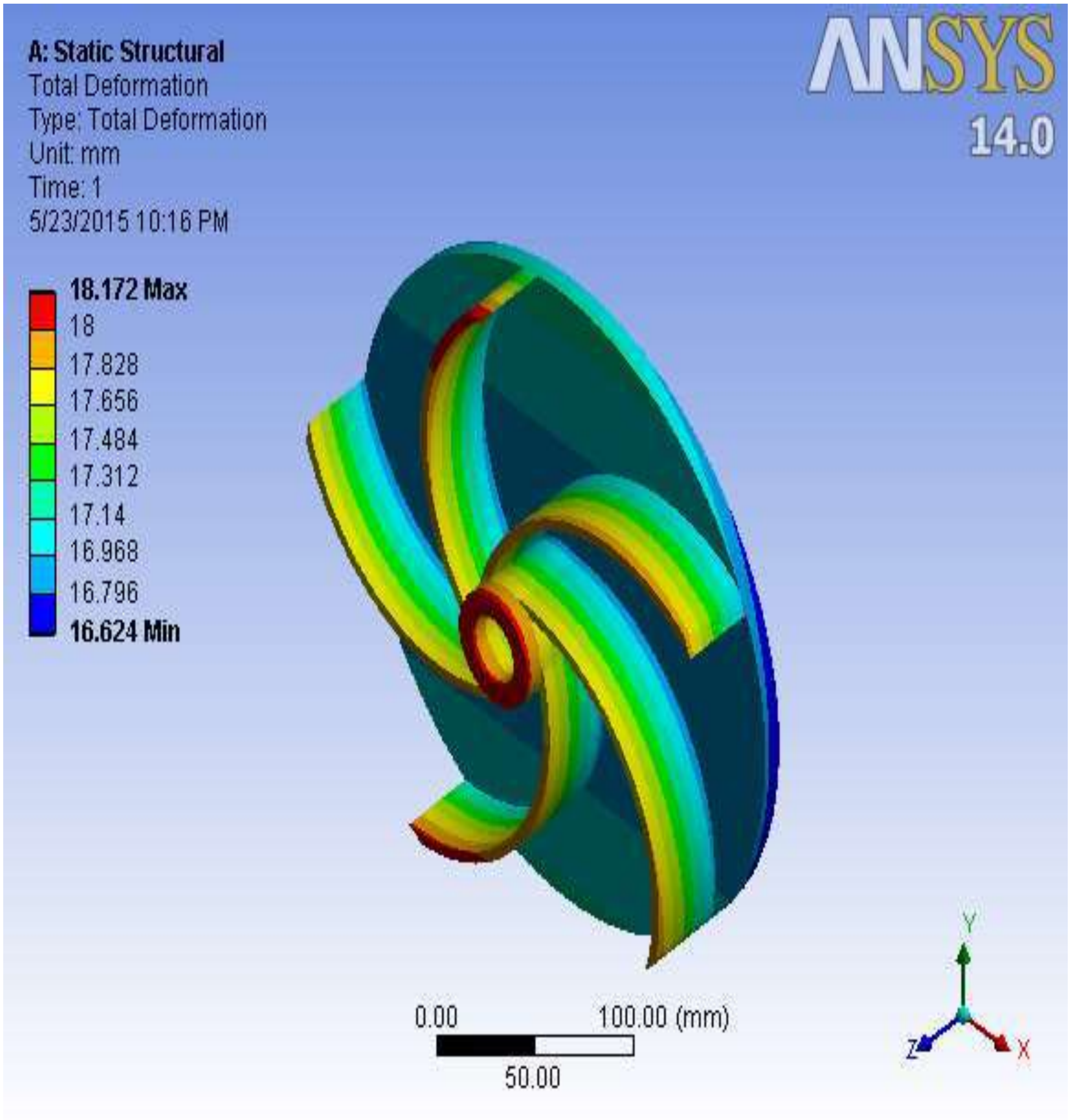

Fig 6. Total deformation of five blade impeller

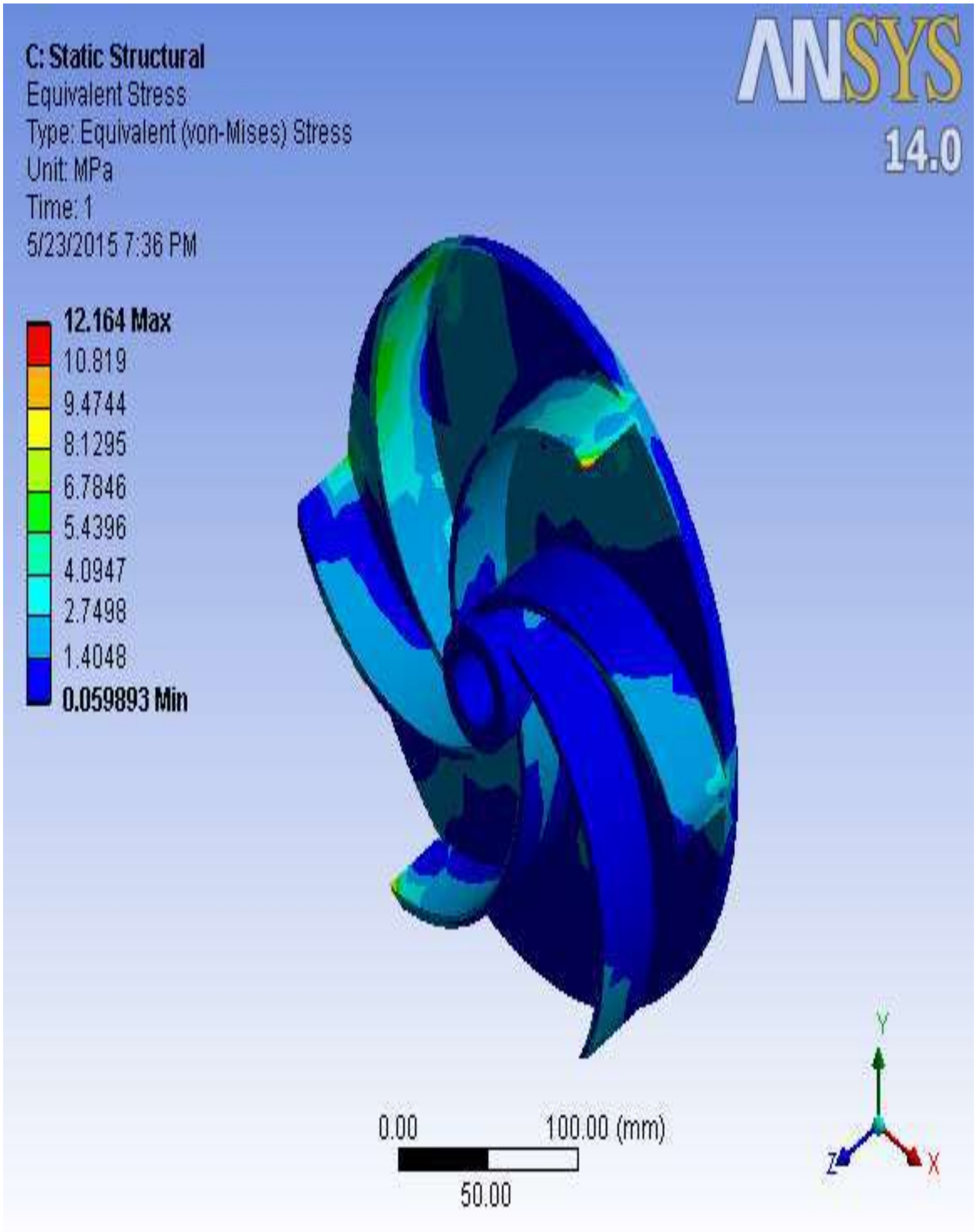

Fig 7. Equivalent stress of six blade impeller

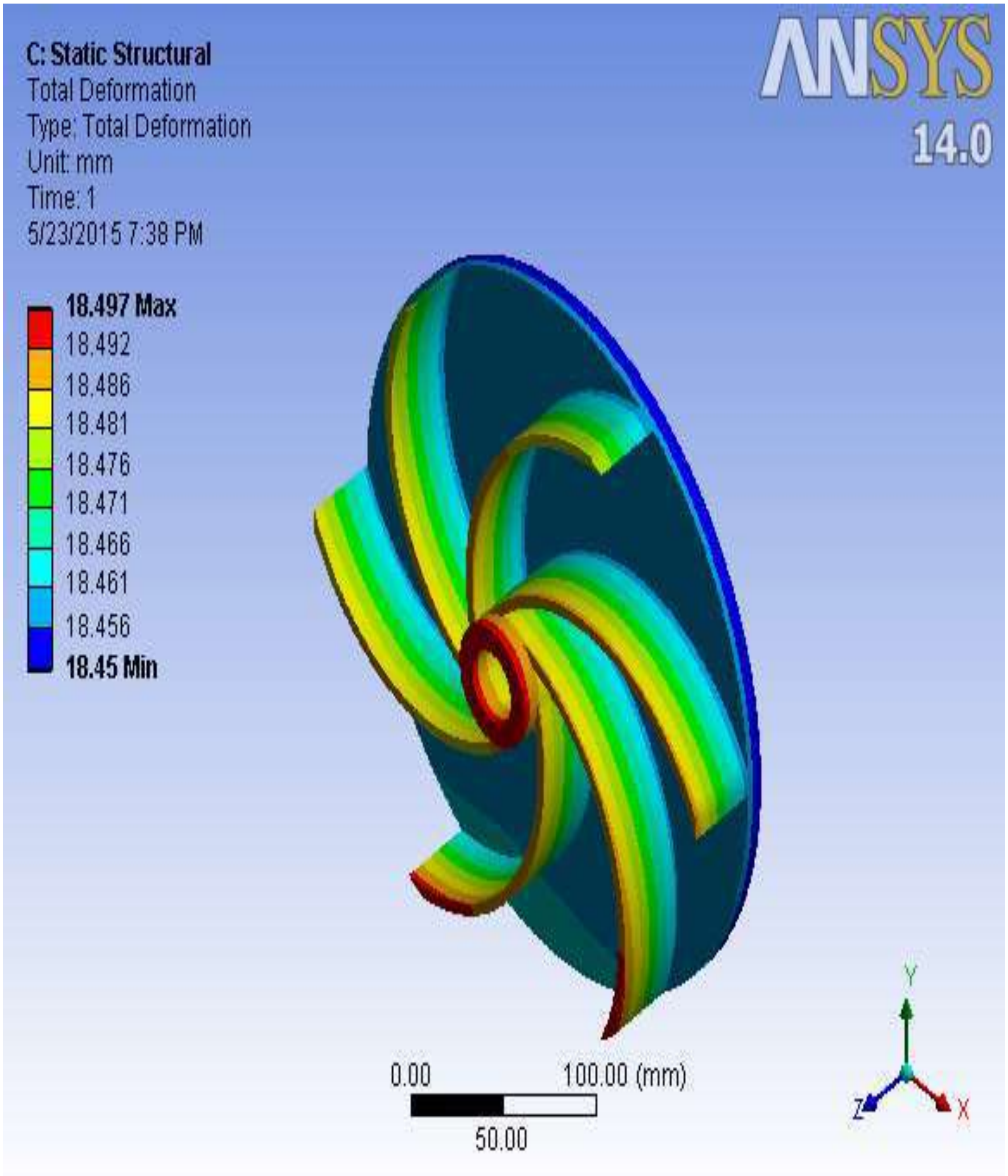

Fig 8.Total deformation of six blade impeller

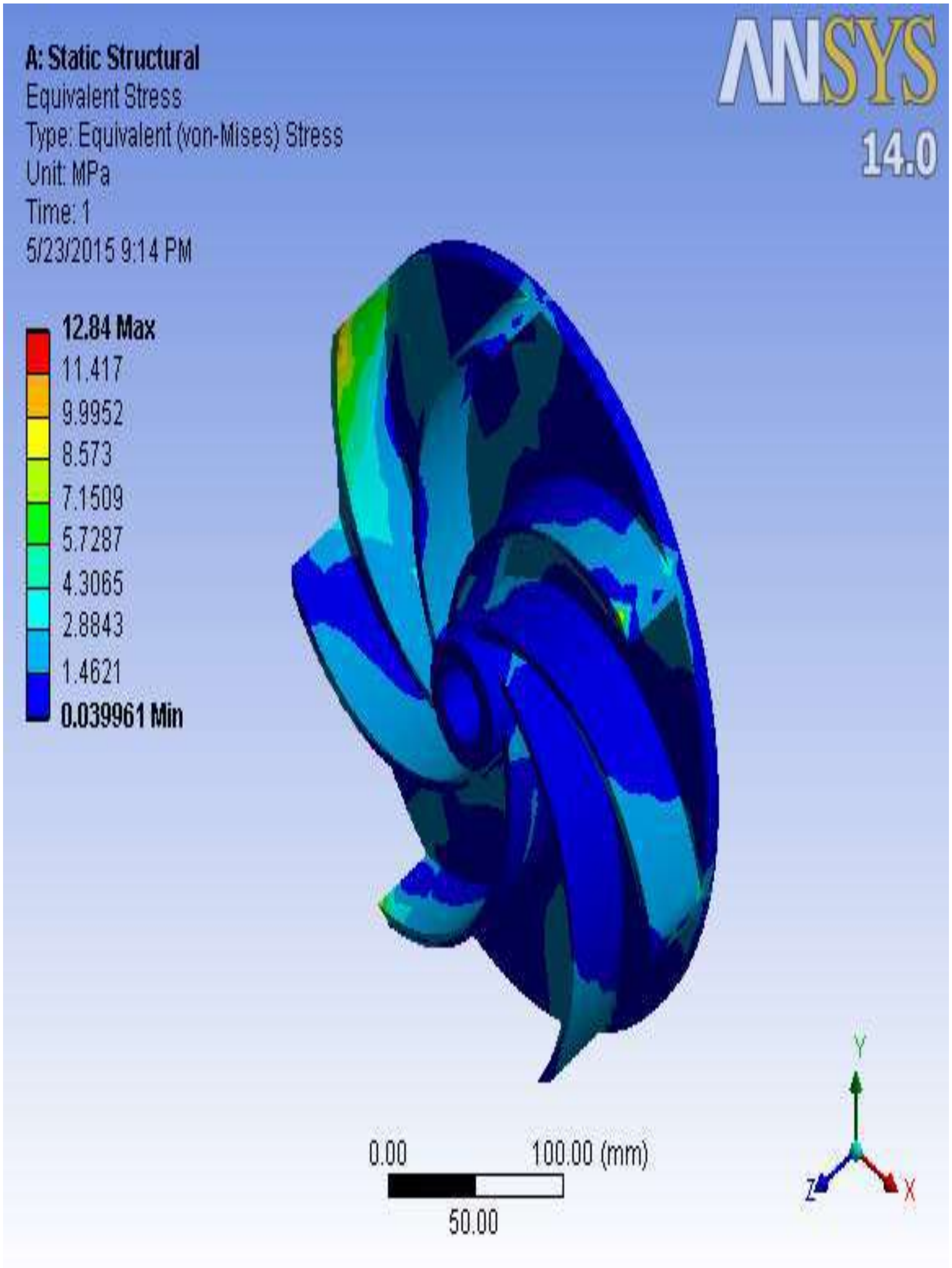

Fig 9.Equivalent stress of seven blade impeller

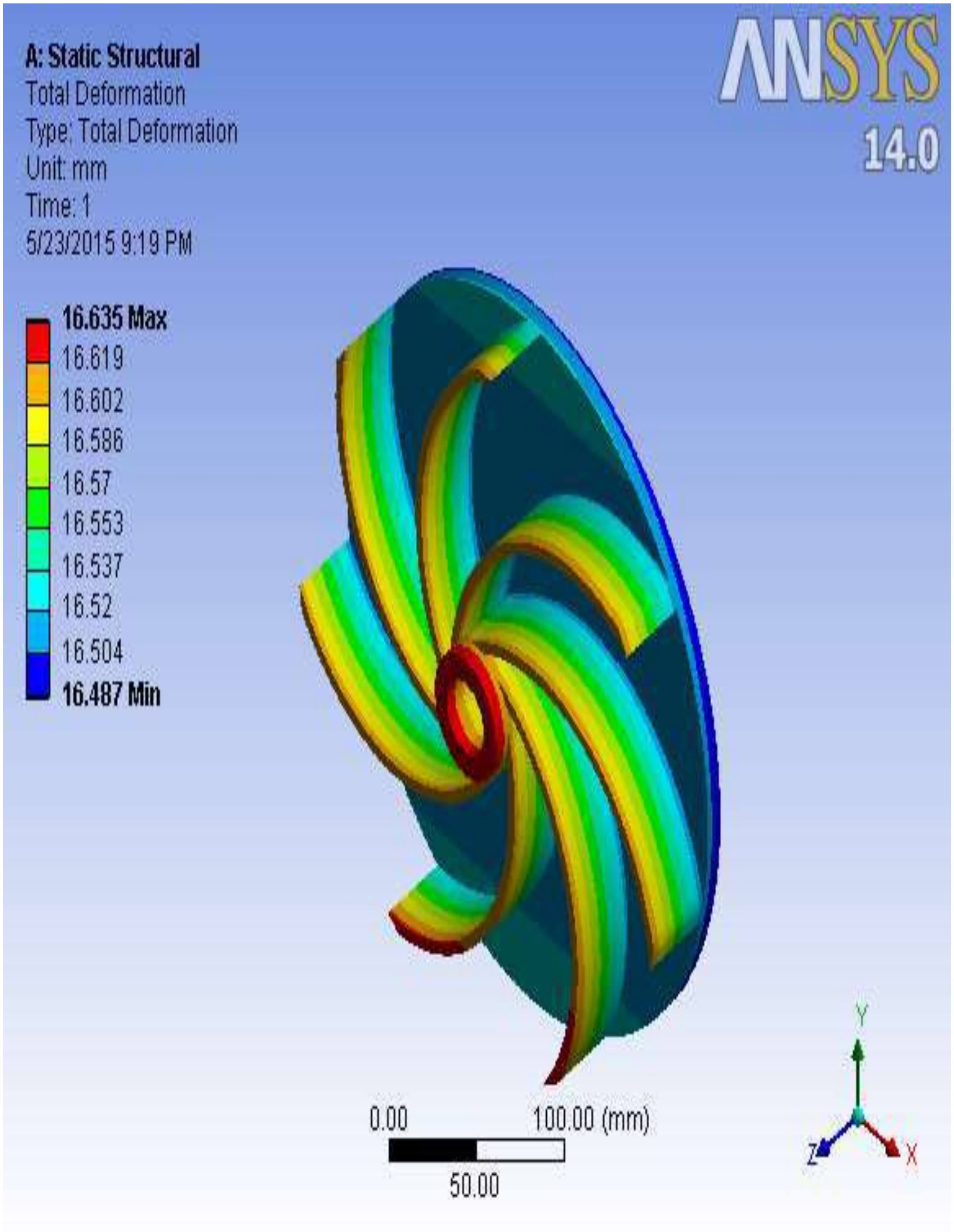

Fig 10. Total deformation of seven blade impeller

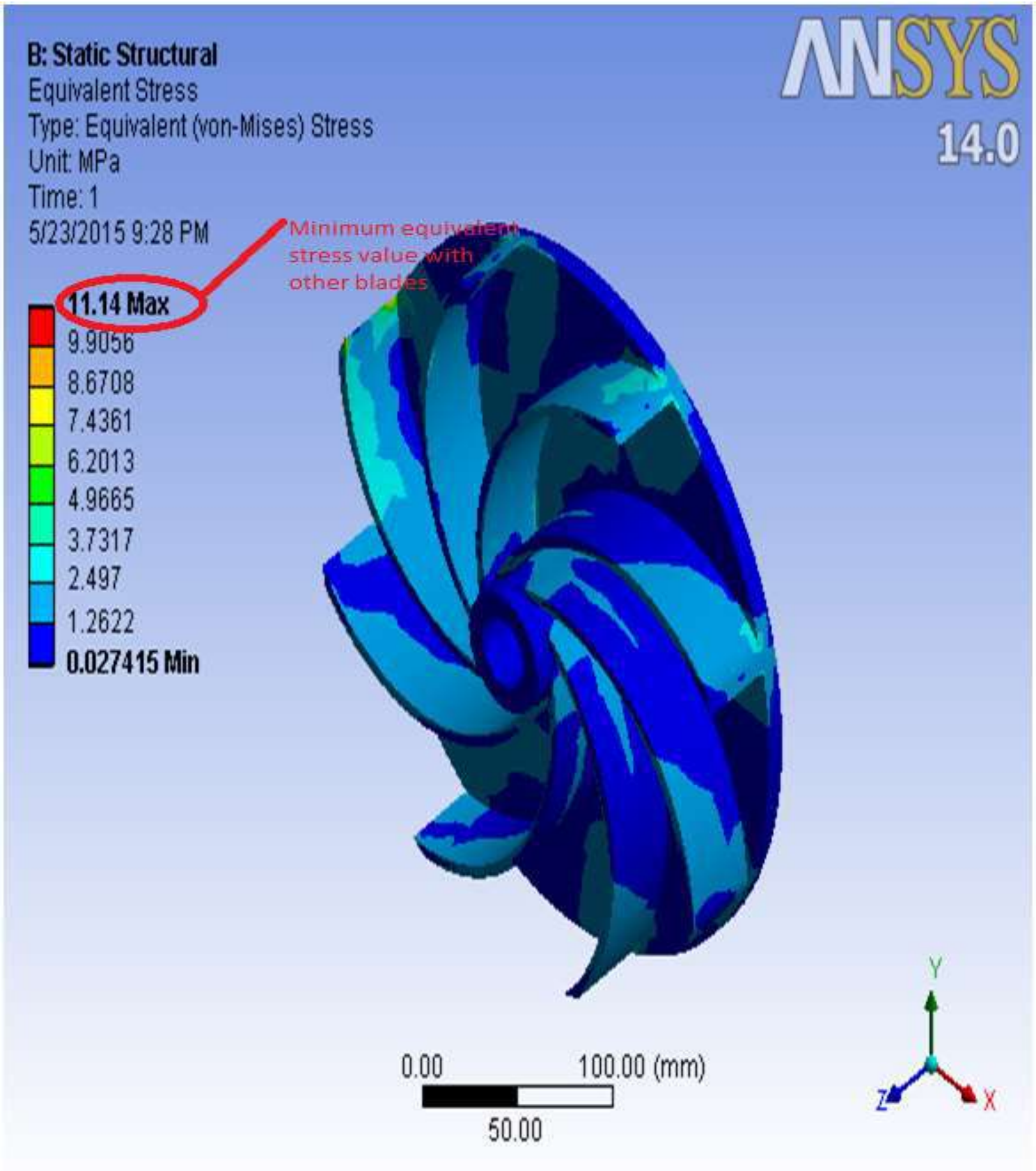

Fig 11.Equivalent stress of eight blade impeller

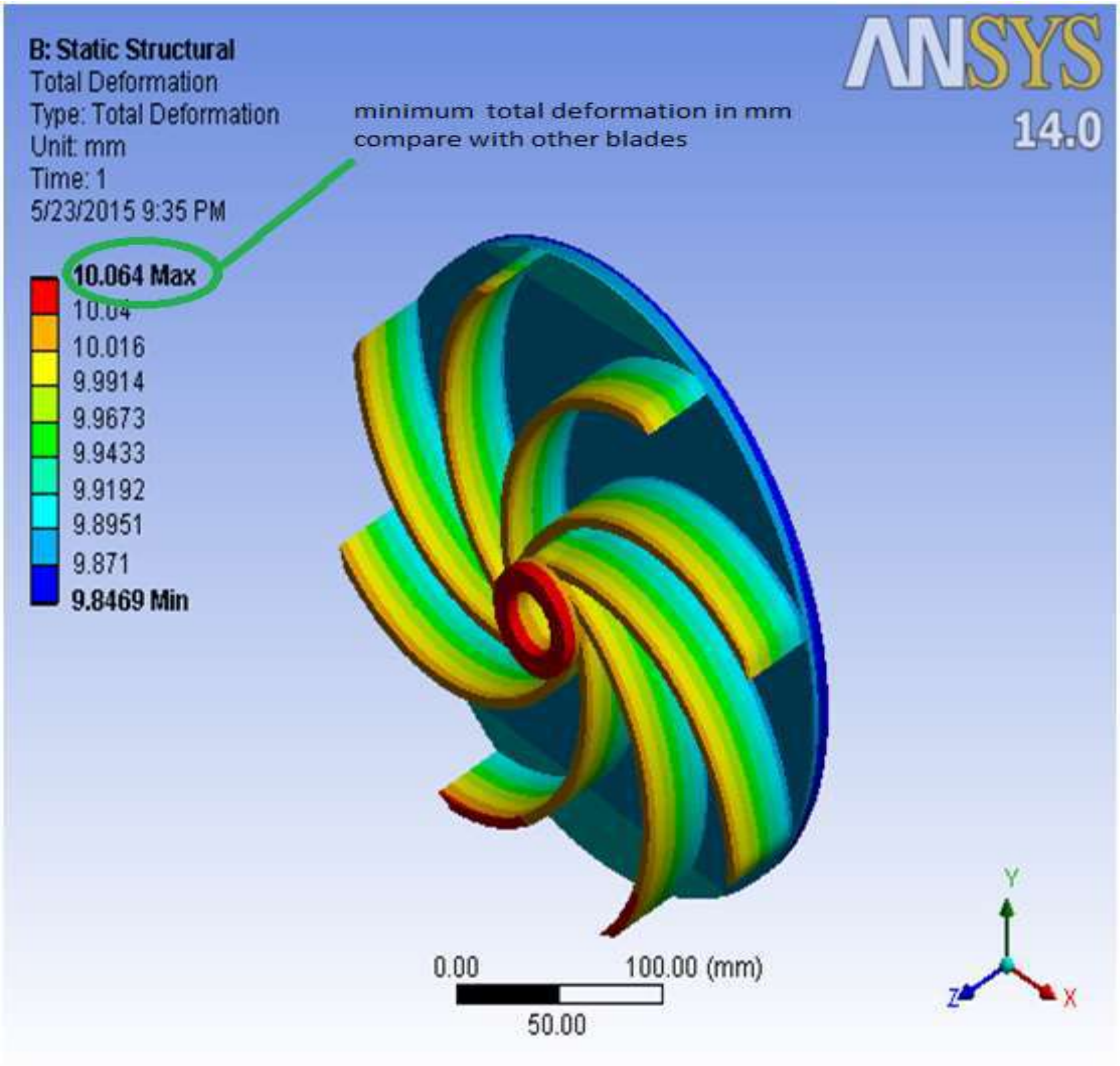

Fig 12. Total deformation of eight blade impeller

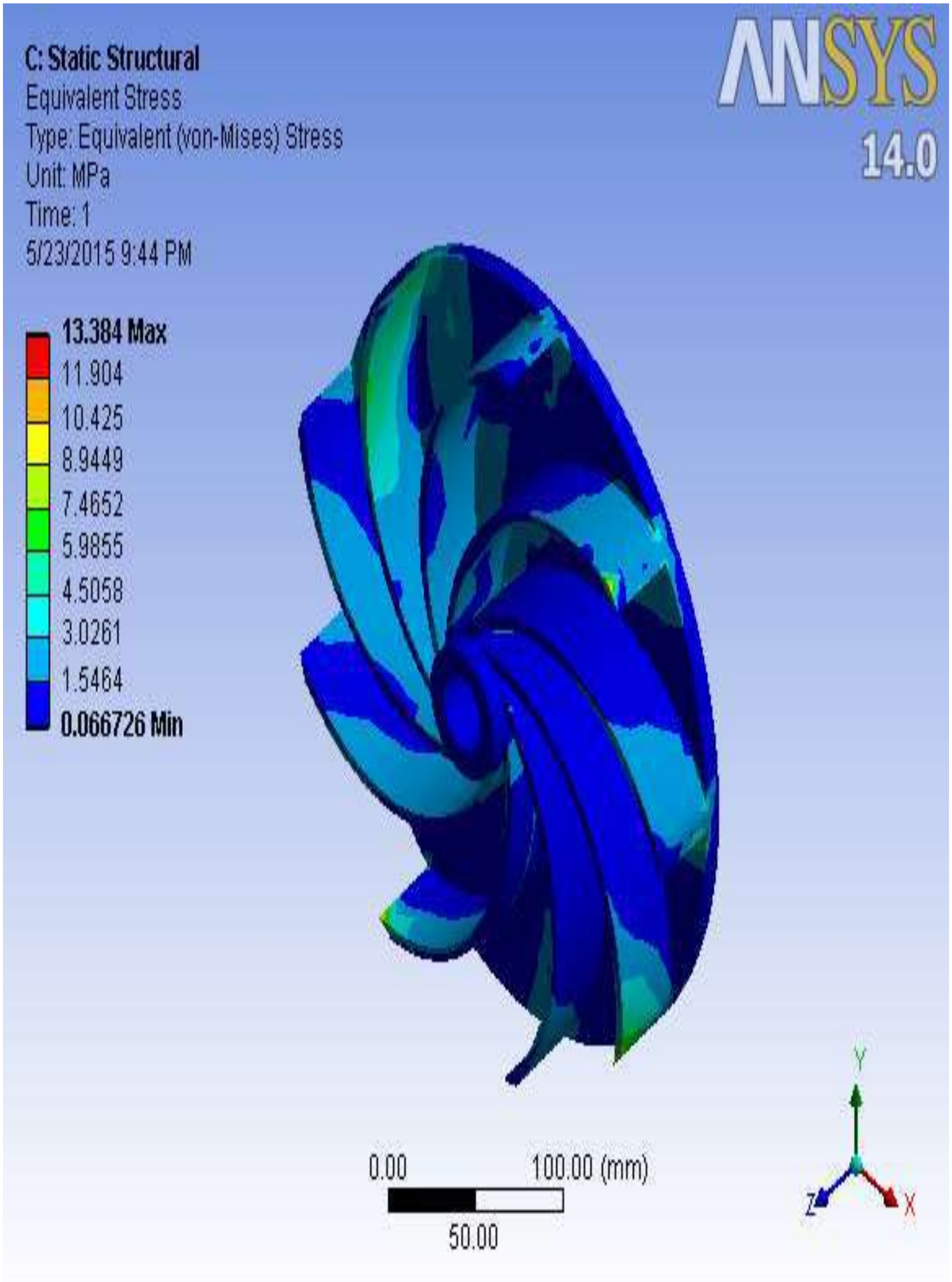

Fig 13.Equivalent stress of nine blade impeller

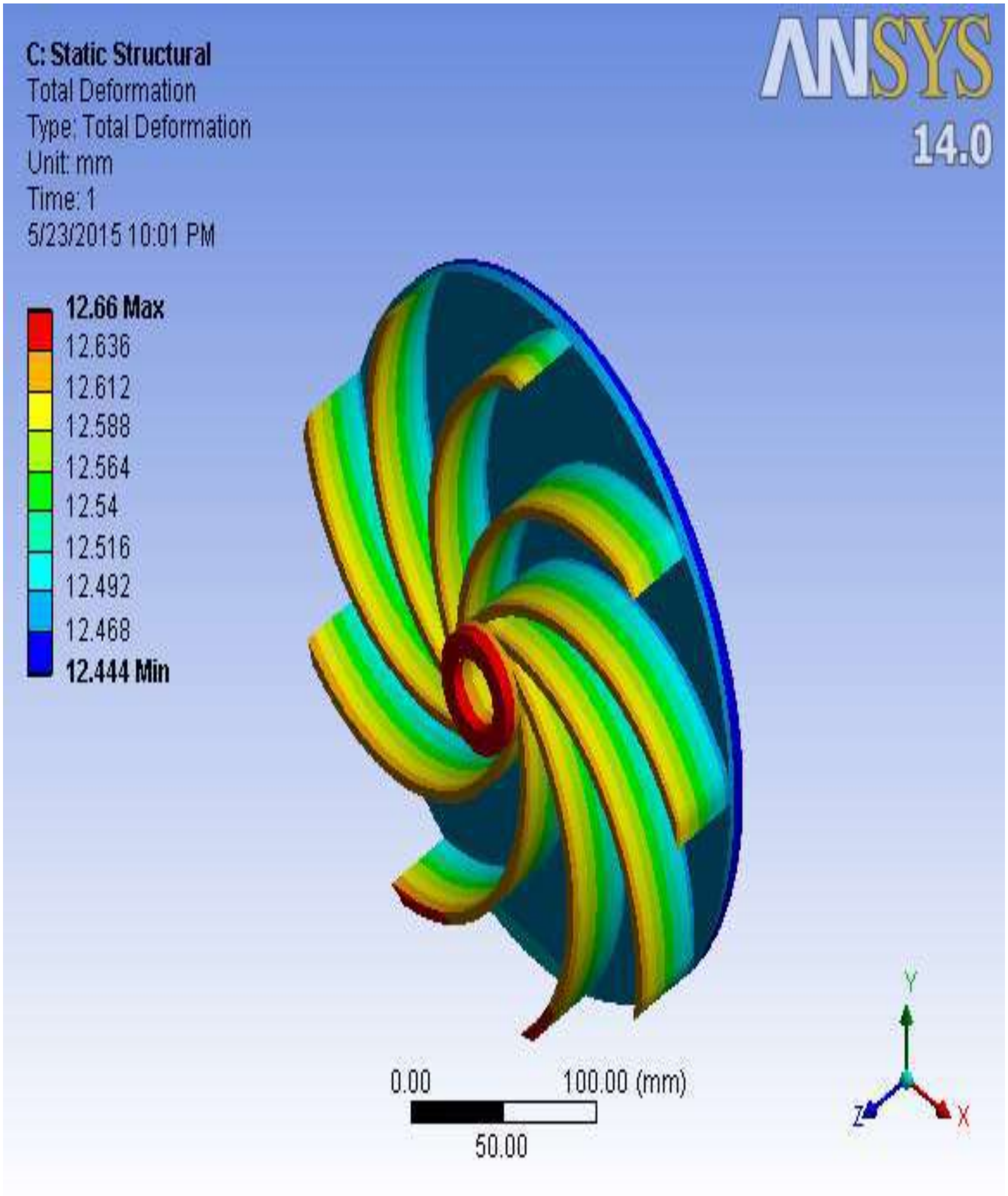

Fig 14.Total deformation of nine blade impeller

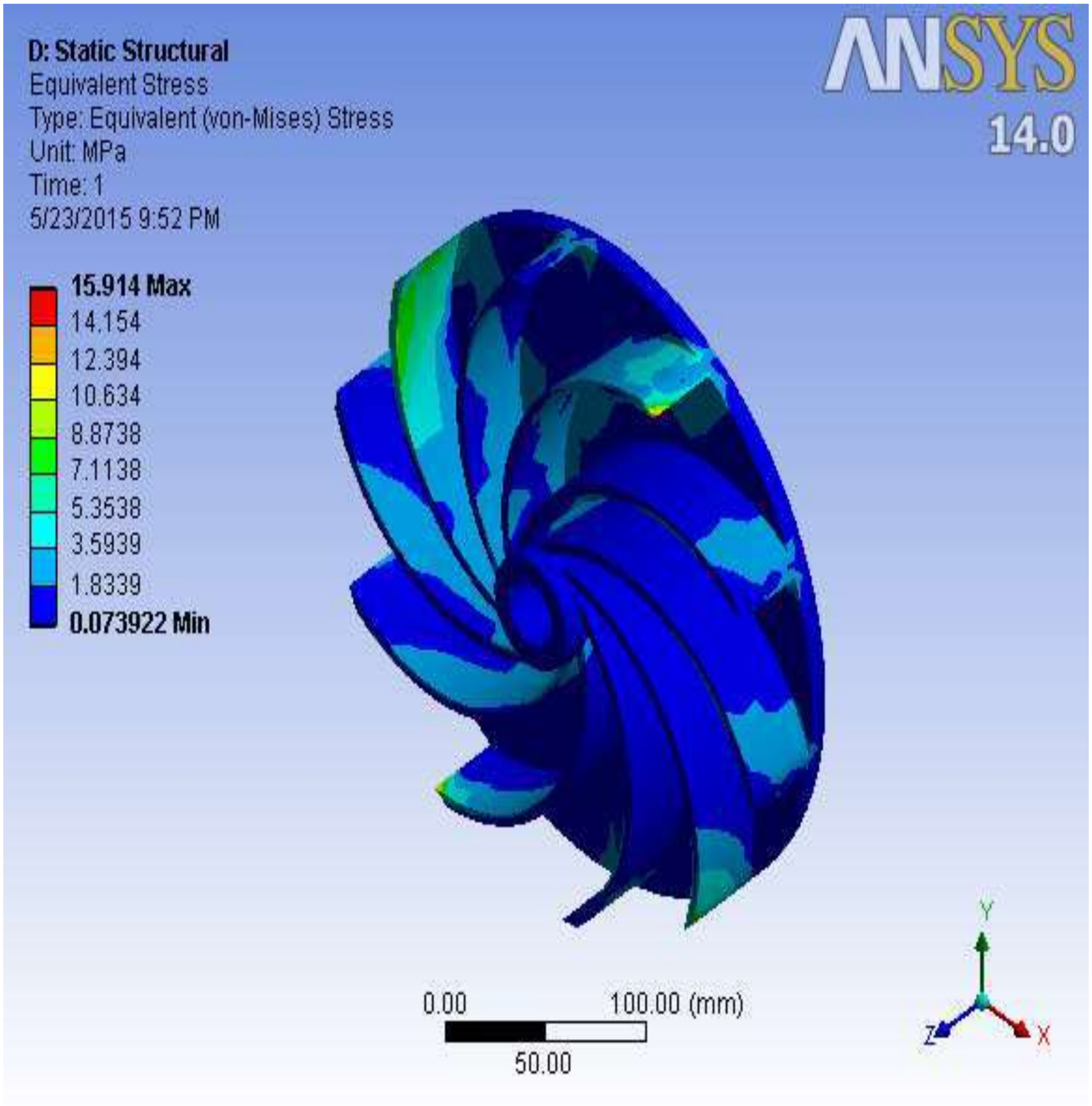

Fig 15.Equivalent stress of ten blade impeller

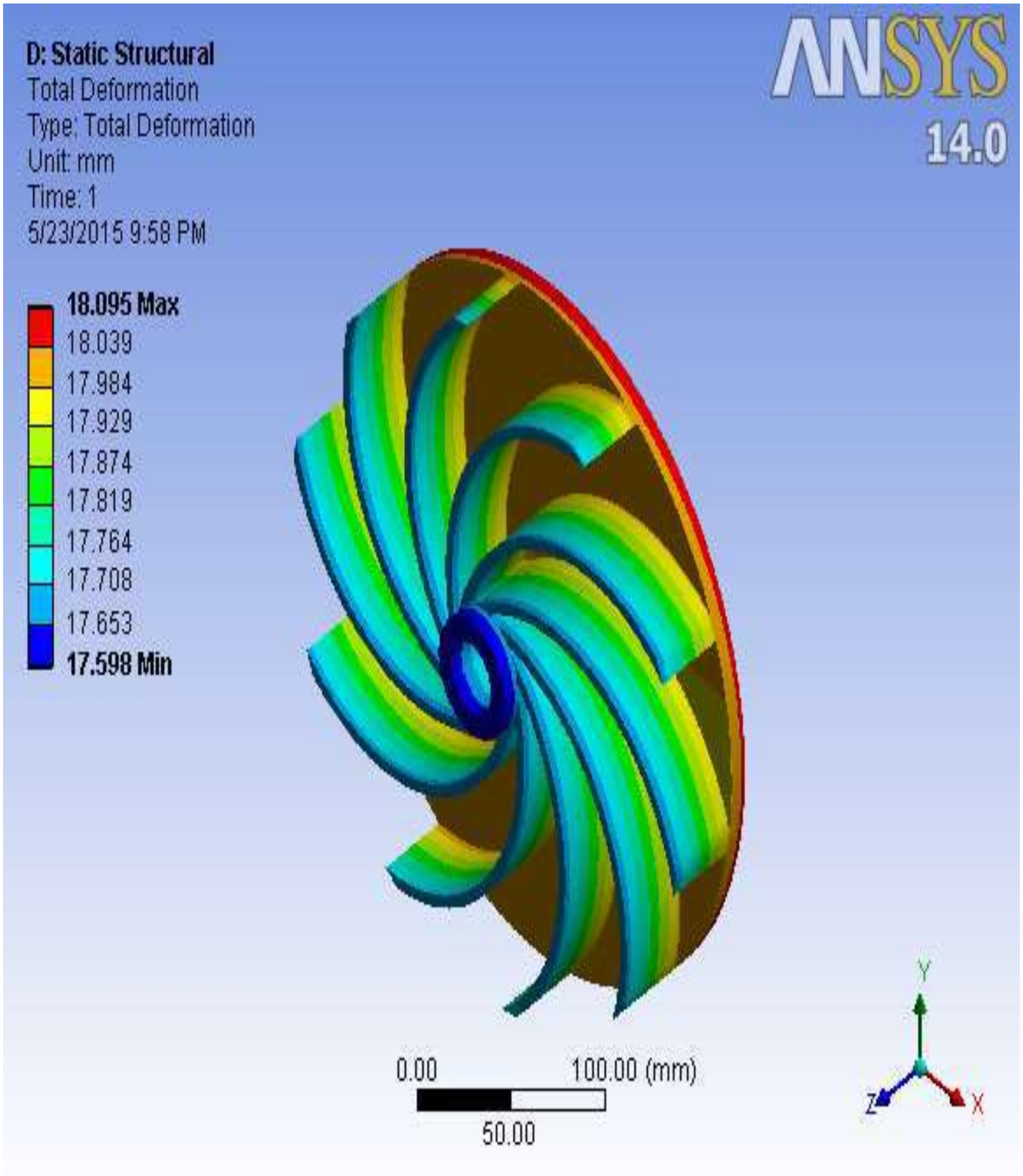

Fig 16.Total deformation of ten blade impeller

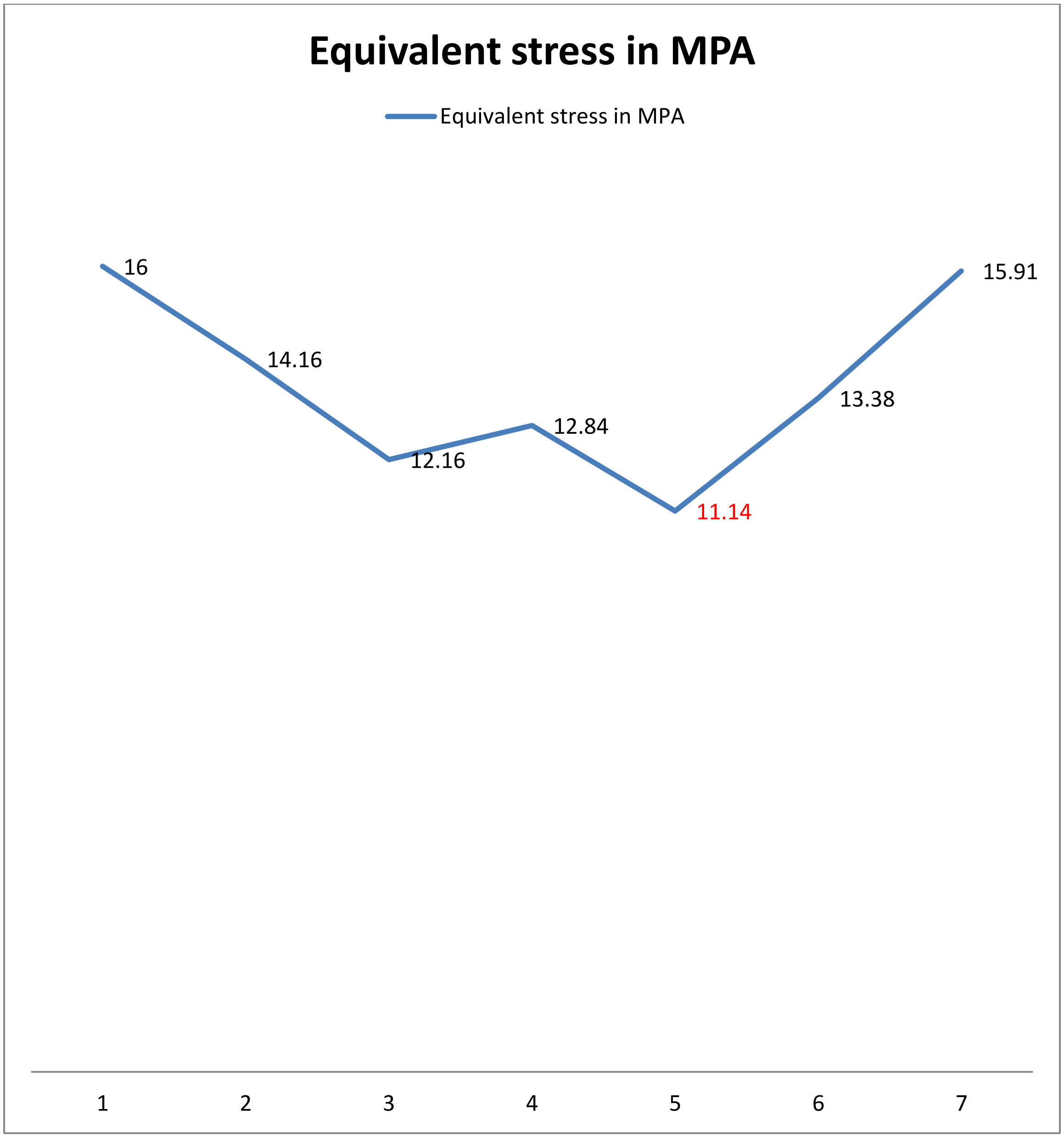

Fig 17.Number blades Vs Equivalent stress in MPA

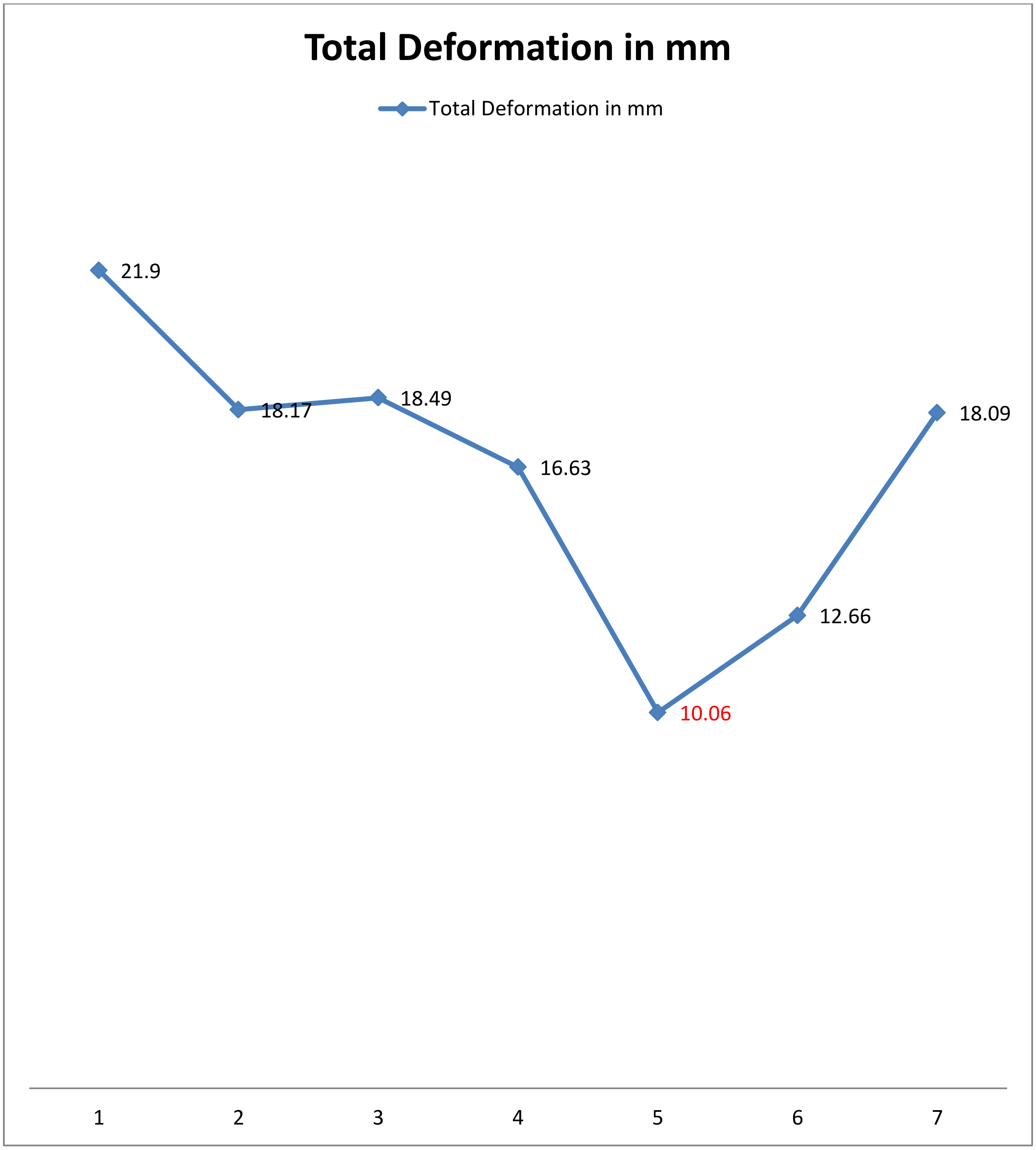

Fig 18.Number of blades Vs Total deformation in mm

Table.1.Methodology.

S.NO	METHODOLOGY
1.	Concept of design
2.	Experimental procedure
3.	Selection profile of impeller blade
4.	Impeller blade modeling
5.	Analysis of modeling
6.	Optimization of blade number and angle
7.	Interpretation of result and conclusion

Table.2.ANSYS result of stress and deformation

NUMBER OF BLADES	EQUIVALENT STRESS in Mpa		TOTAL DEFORMATION in mm	
	Min	**Max**	**Min**	**Max**
4	0.017	16	21.5	21.9
5	0.076	14.16	16.62	18.17
6	0.059	12.16	18.45	18.49
7	0.039	12.84	16.48	16.63
8	0.02	11.14	9.84	10.06
9	0.06	13.38	12.44	12.66
10	0.07	15.91	17.59	18.09