INVESTIGATION OF EFFECT OF STRAIGHT SIDED & SCALLOPED SHIM PROFILE ON PERFORMANCE OF SHIM COUPLING BY FINITE ELEMENT ANALYSIS

Rahul D. Belekar[1], Prof.U. D. Gulhane[2]

[1,2]*Department of Mechanical Engineering, Finolex Academy of Management and Technology, Ratnagiri, Maharashtra, (India)*

ABSTRACT

In industries & various applications two shafts are connected each other by couplings. But due to various misalignment stresses are to be induced into the coupling. To control & reduced these stresses different types of shapes of disc are used. This study focuses on only two types of disc i.e. Straight sided & Scalloped disc. In FEA study, static structural analysis method is used to find deflection, shear stresses & equivalent stresses. Ansys 13 software is used for FEA study, & discusses which type of disc profile is best suitable for Disc coupling. To back up the results obtained by FEA study Mathematical modeling is done to find deflection, shear stresses & principle stresses.

I. INTRODUCTION

A coupling is a device is used to connect two shafts together at their ends for the purpose of transmitting power. The primary purpose of couplings is to join two pieces of rotating equipment while permitting some degree of misalignment or end movement or both. Misalignment reduces the life span, reliability of motors & their associated components. Poor choice and poor design of a flexible coupling results in unwanted mechanical vibration in rotating machinery. The effects of excessive vibration may lead to the premature failure of connected machinery, reduced efficiency and may pose threats to a worker's health and comfort.

Flexible couplings are used to transmit torque from one shaft to another when the two shafts are slightly misaligned. Flexible couplings can accommodate varying degrees of misalignment up to 3° and some parallel misalignment. In addition, they can also be used for vibration damping or noise reduction. The various applications of flexible coupling in industry are generator sets, pump sets, compressors, wind turbine, general purpose heavy duty applications.

A flexible coupling subjected to torque, misalignment and speed reacts on the connected equipment. These reactions not only affect the life of the coupling but the life of the equipment as well. Resulting moments and forces may cause unacceptable loads on the bearings, seals, the equipment shafts and even the equipment support structure that can cause them to wear, twist, deform and even fail prematurely. This can create unexpected downtime and increased maintenance, which increase the operating cost of the equipment.
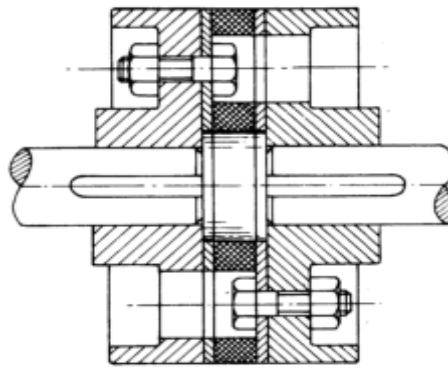
Fig.1 – Flexible Rubber Disc Coupling

There are three basic functions of flexible coupling -:

1)   Transmit power

2)   Accommodate misalignment

3)   Compensate for end movement

II. DISC COUPLING

In 1971, Zurns Mechanical drives division developed a multiple diaphragm design, with number of thin plates in parallel, rather than a single thick one. This type of design provides improve flexibility & lower stresses, as the stresses are proportional to the cube of the material thickness. The disc coupling is available in a number of forms, all have the driving & driven bolts on the same bolts on the same bolt circle. The flexibility of misalignment that each type can handle depends upon the length of the material between bolts. Disc coupling on the other hand, use a series of thin laminates to form one ring, or disc pack.

In our project we work on the two disc profile. First is the straight sided disc profile & second one is the scalloped disc profile. Straight-sided and scalloped flex discs are the two most commonly used blade types. Round discs are still commonly used in many older coupling styles. Straight-sided discs are stiffer torsionally. With these designs, less pre-stretch is required than with scalloped designs because there is less torsional windup under load. Because less pre-stretch is required, straight-sided discs are somewhat easier to assemble. Each is suited to particular applications. By FEA analysis we have to decide which type of shim is best suitable for coupling.
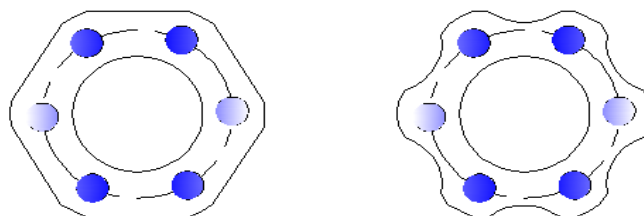

Fig. 2 - Common Blade Types

2.1 OBJECTIVES

Many researchers had done analysis to find out shim failure and however in the different applications disc failures are different also causes are different, therefore it need to study the misalignment problem to improve shim coupling life. Main objectives are

- Prepare a Finite element Analysis for Straight sided & Scalloped Shim profile.
- Check the deformation & Stresses of two different shim profiles for same loading & boundary condition.
- Prepare Mathematical Modeling for Axial Misalignment, Radial Misalignment, Angular Misalignment.
- Find out the optimum one for Shim Coupling.

2.2 DEFINITION OF PROBLEM

In shim coupling different types of shim profiles are used to transfer power from one shaft to the other by taking different misalignment. While transmitting power with misalignment the disc bend & various stresses are induced into the shim profile. The intensity of various stresses induced in shim is depending upon shim profile. In circular disc stresses are more as compare to straight sided disc. In case of straight sided shim the stresses are less at the edges between two bolts. And failure of disc is occurs at the portion between two bolts. If material is removed from the edges then it becomes scalloped shim. So the shim can bent very easily, so it can take more misalignment with less torque. The problem under consideration is to investigate the Deformation & Various Stresses in Shim profile due to

- Bolt pretension
- Axial Misalignment
- Peak Torque

2.3 SCOPE OF THE PROJECT

In shim coupling we used two types of shim profile i.e. straight sided & scalloped shim. Shim coupling subjected to axial misalignment, angular misalignment & radial misalignment.

The scope of the project focus -:

- Create a model of shim coupling assembly using Pro-E (wild fire 5.0).
- Analysis the Shim/Disc using FEA software (ANSYS 13).
- Mathematical Modeling for disc.
- Comparative study for both straight sided & scalloped shim/disc.

III. INTRODUCTION OF PRO-E MODELING

Total assembly of Shim coupling has done in the software Pro-engineer. Pro-engineer is a parametric, integrated 3D CAD/CAM/CAE solution created by Parametric Technology Corporation (PTC). It was the first successful, parametric, feature-based, associative solid modeling software in the market. The application run on

Microsoft Window, Linux and UNIX platforms, and provides solid modeling, assembly modeling and drafting, finite element analysis, and NC and too long functionality for mechanical engine.

Pro-E is modeling tool used for the modeling of the component and assembly for analysis purpose. This tool also used for analysis purpose. By using this software we can construct different models.

3.1 FINITE ELEMENT ANALYSIS

The Finite Element Method (FEM) is a numerical technique for finding approximate solutions of partial differential equations (PDE) as well as of integral equations. Its practical application often known as Finite Element Analysis (FEA). Finite Element Analysis is a simulation technique which evaluates the behavior of components, equipment and structures for various loading conditions including applied forces, pressures and temperatures. Thus, a complex engineering problem with non-standard shape and geometry can be solved using finite element analysis where a closed form solution is not available.

The finite element method has become a powerful tool for the numerical solutions of a wide range of engineering problems. It has been developed simultaneously with the increasing use of the high- speed electronic digital computers and with the growing emphasis on numerical methods for engineering analysis.

3.2 GENERAL DESCRIPTION OF THE METHOD

In engineering problems there are some basic unknowns. If they are found, the behavior of the entire structure can be predicted. The basic unknowns or the Field variables which are encountered in the engineering problems are displacements in solid mechanics, velocities in fluid mechanics, and electric and magnetic potentials in electrical engineering and temperatures in heat flow problems. In a continuum, these unknowns are infinite. The finite element procedure reduces such unknowns to a finite number by dividing the solution region into small parts called elements and by expressing the unknown field variables in terms of assumed approximating functions (Interpolating functions/Shape functions) within each element. The approximating functions are defined in terms of field variables of specified points called nodes or nodal points. Thus in the finite element analysis the unknowns are the field variables of the nodal points. Once these are found the field variables at any point can be found by using interpolation functions.

Thus the various steps involved in the finite element analysis are

1)   Select suitable field variables and the elements.

2)   Discritise the continua.

3)   Select interpolation functions.

4)   Find the element properties.

5)   Assemble element properties to get global properties.

6)   Impose the boundary conditions.

7)   Solve the system equations to get the nodal unknowns.

8)   Make the additional calculations to get the required value

3.3 BOUNDARY CONDITIONS

In this analysis pump end hub keep fixed & torque is applied at the spacer as shown in figure. Also gives the bolt pretension at six bolts. This coupling provides 0.5^o angular misalignment, 1.4 mm Radial misalignment & 1mm axial misalignment.
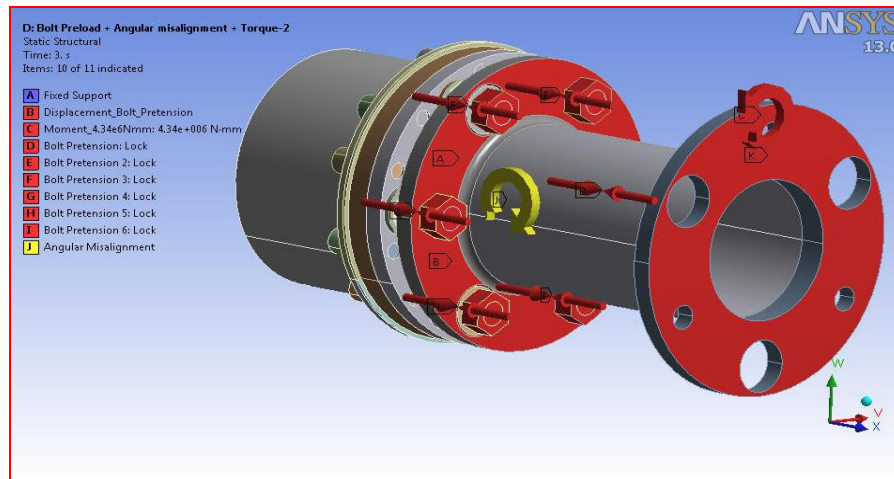

Fig. 2 - Boundary Conditions

3.4 STRESS ANALYSIS

Stress analysis is the determination of various stresses i.e. equivalent stress, Principal stress & Shear stress etc in the straight & scalloped disc coupling. After finding the stresses in disc we select the suitable disc for coupling. For that purpose we select the area between two bolts. Because that area just bent by taking actual Axial misalignment, Angular misalignment & Radial misalignment. At this area stress is minimum at outer edges. So we can reduce this area so the straight sided disc becomes scalloped disc.

3.5 FEA RESULT

The aim of this project is to find out the suitable Shim profile for Shim coupling. In our project we select two types of Shim profile i.e. straight sided & Scalloped Shim profile. Scalloped profile is the modification in the straight sided profile as. In this project we study the stress analysis of shim profile by software Ansys 13 & validate the result of Straight sided Shim profile with mathematical modeling.

3.6 FEA RESULT FOR THREE STEPS

FEA results are the get for both straight sided & scalloped Shim. In this result we find principle stress, shear stress, equivalent stress & total deformation. For this analysis there are three cases i.e. axial misalignment, radial misalignment & angular misalignment. And for each case we tabulate the result in three steps. In first step we find the result for bolt pretension. In second step we find result for bolt pretension + Axial misalignment. In third case we find the result for bolt pretension + Axial misalignment + torque applied.

IV. AXIAL MISALIGNMENT

In this project for analysis purpose, axial misalignment takes 1mm. In axial misalignment axis of one shaft is misalign with 1mm along axis.

FOR BOLT PRETENSION FOLLOWING RESULTS ARE OBTAINED -:

4.1 TOTAL DEFORMATION -:
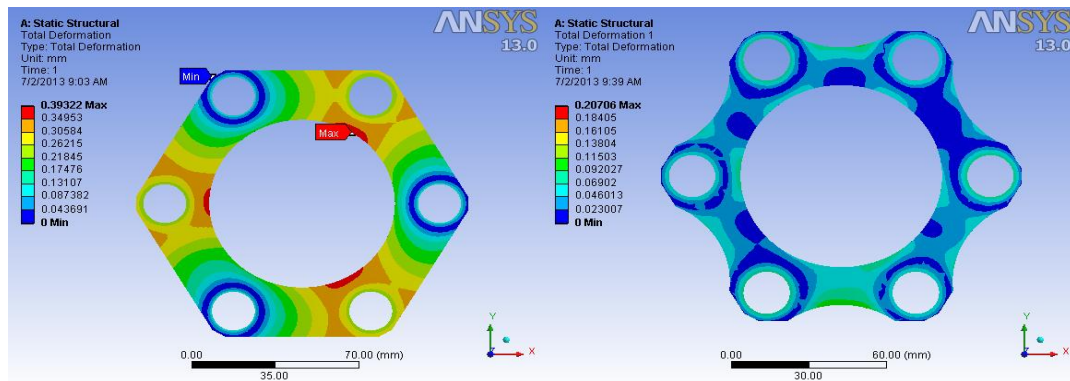
The maximum deformation is done in straight sided shim is 0.39322 mm & for scalloped shim it is 0.20706 mm. The deformation at area in between two bolts is 0.26215 mm & for scalloped shim is 0.046013mm. From above analysis we conclude that for same input, deformation is minimum in scalloped shim profile as compared to straight side shim profile.

4.2 MAXIMUM PRINCIPLE STRESS
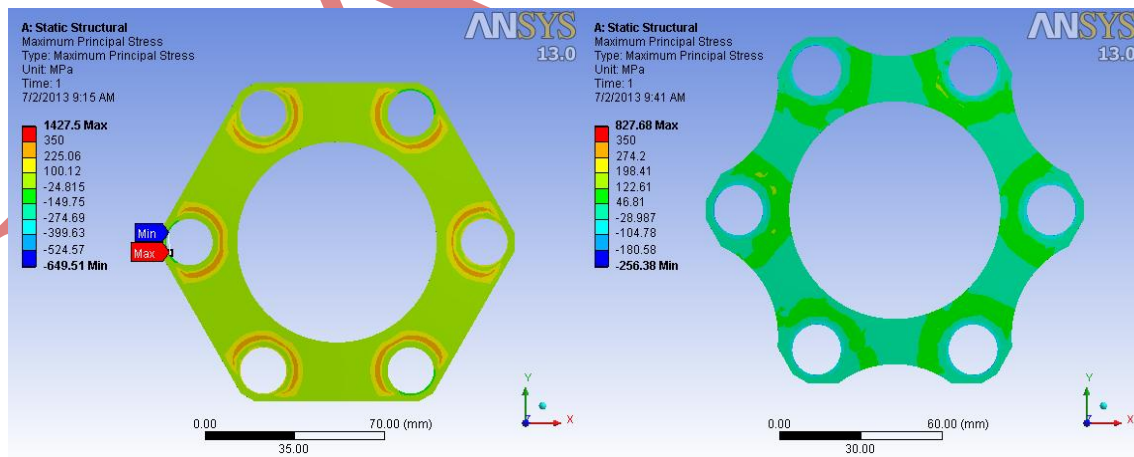
Maximum principle stress developed in straight shim is 1427.5 N/mm^2 & for scalloped shim is 827.60 N/mm^2. Principle stress developed at area between two bolts is 100.12 N/mm^2 for straight sided shim & for scalloped shim it is 46.81 N/mm^2. So we conclude that for same input, maximum principle stress developed in scalloped shim is less as compared to straight sided shim.

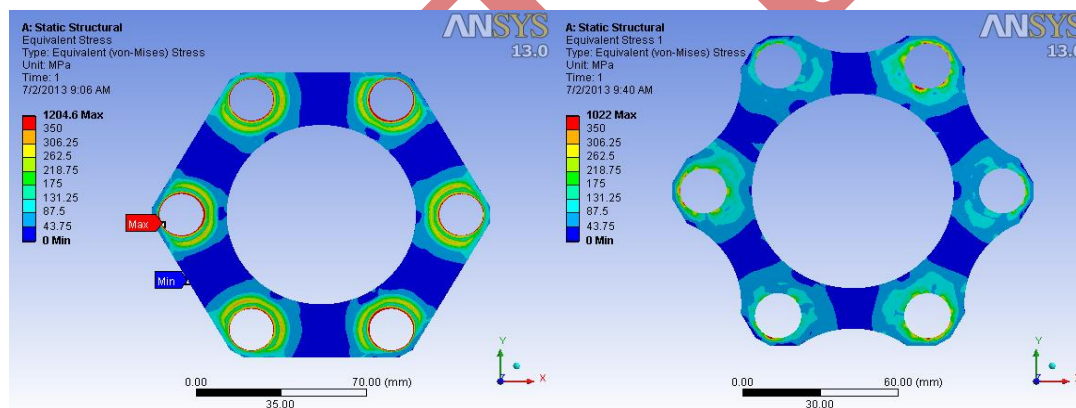### 4.3 SHEAR STRESSES
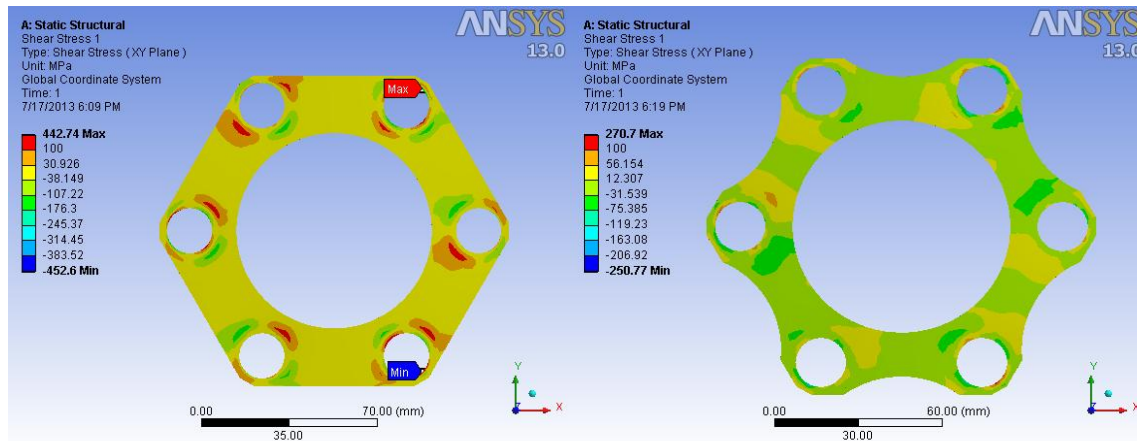
Shear stress developed in straight shim is 442.74 N/mm^2 & for scalloped shim is 270.7 N/mm^2. Shear stress developed at area between two bolts is 30.926 N/mm^2 for straight sided shim & for scalloped shim it is 12.307 N/mm^2. So we conclude that for same input, shear stress developed in scalloped shim is less as compared to straight sided shim.
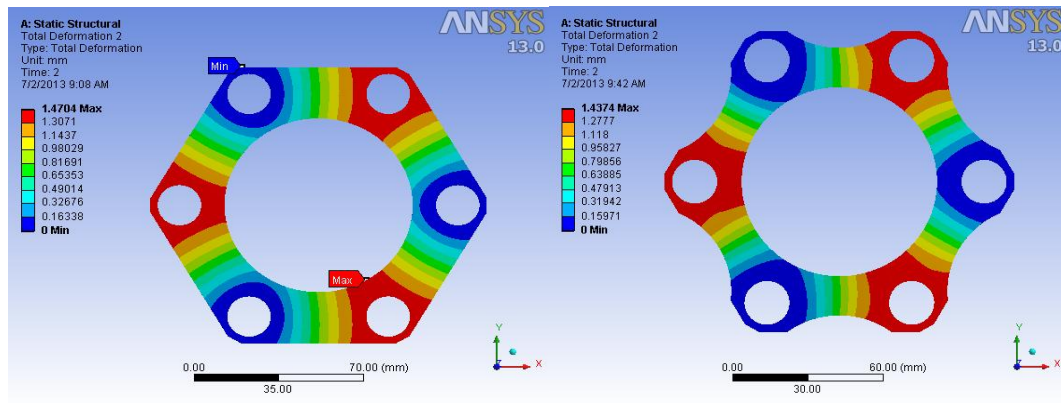
4.4 EQUIVALENT STRESSES (VON-MISES)

Equivalent stress developed in straight shim is 442.74 N/mm^2 & for scalloped shim is 270.7 N/mm^2. Equivalent stress developed at area between two bolts is 30.926 N/mm^2 for straight sided shim & for scalloped shim it is 12.307 N/mm^2. So we conclude that for same input, Equivalent stress developed in scalloped shim is less as compared to straight sided shim.

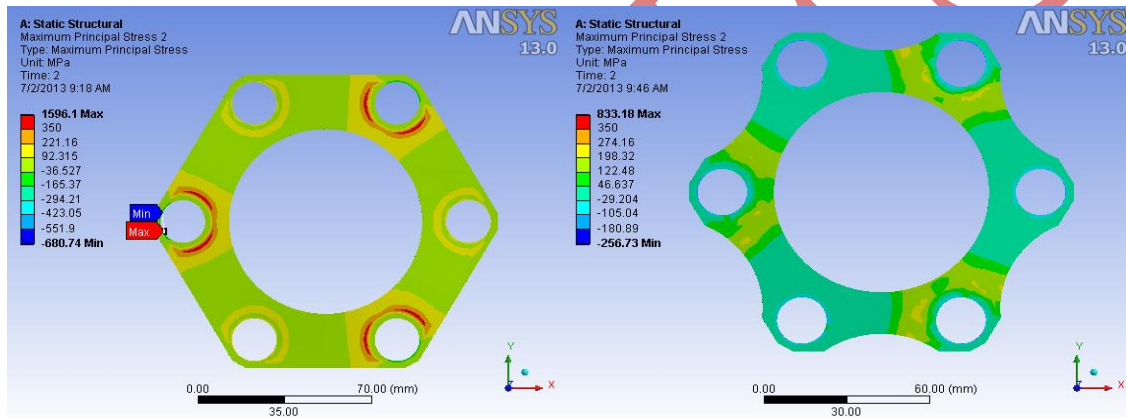## V. BOLT PRETENSION + AXIAL MISALIGNMENT

This is the second step of analysis. In this analysis bolt pretension & particular misalignment input given to coupling & results are tabulated. The results of second step are as follows-:
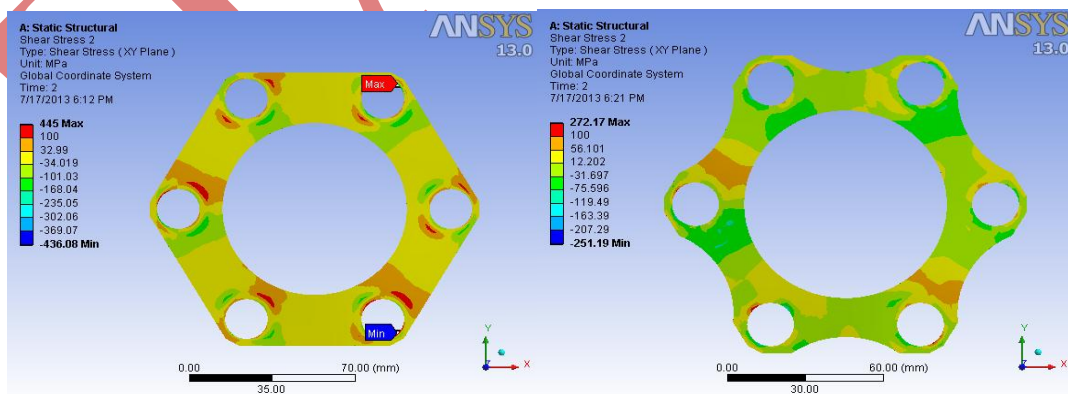
5.1 TOTAL DEFORMATION

The deformation at area in between two bolts is 1.1437 mm & for scalloped shim is 1.118mm.

5.2 MAXIMUM PRINCIPLE STRESS

Principle stress developed at area between two bolts is 92.315 N/mm^2 for straight sided shim & for scalloped shim it is 122.48 N/mm^2
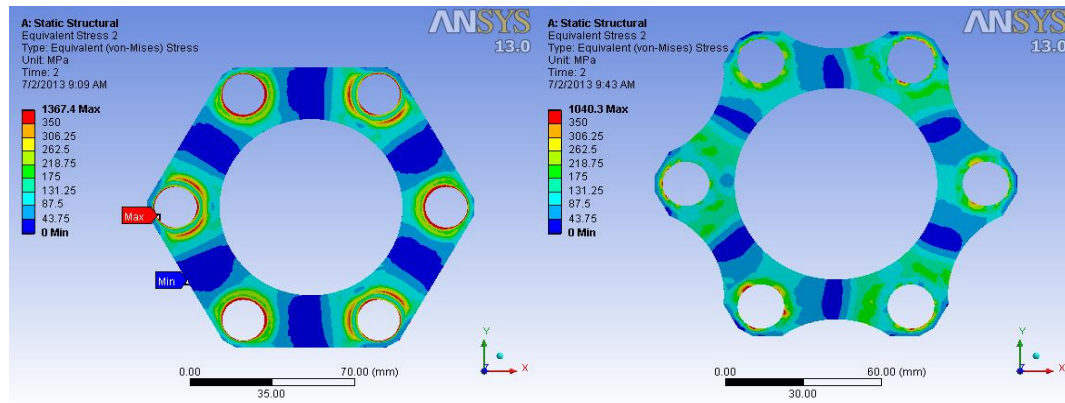
5.3 STRESS

Shear stress developed at area between two bolts is 32.99N/mm^2 for straight sided shim & for scalloped shim it is 12.202 N/mm^2.
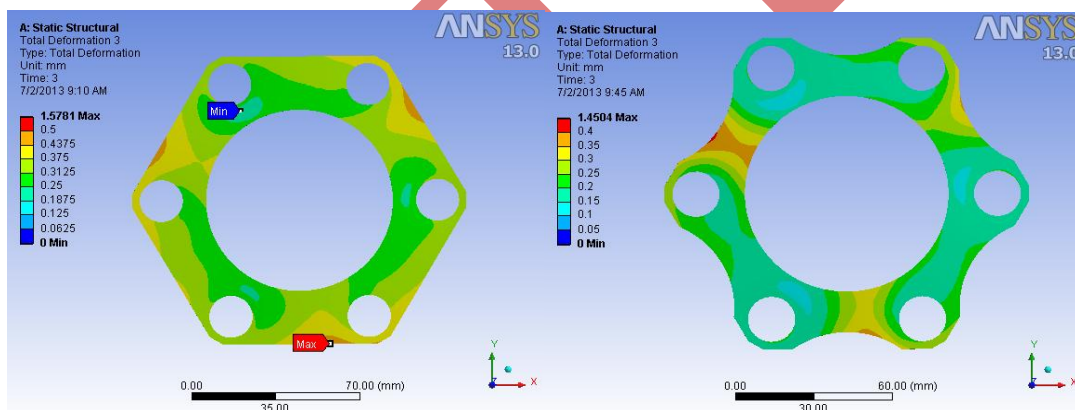
5.4 EQUIVALENT STRESSES (VON-MISES)

Equivalent stress developed at area between two bolts is 43.75 N/mm^2 for straight sided shim & for scalloped shim it is 43.75 N/mm^2.

## VI. BOLT PRETENSION + AXIAL MISALIGNMENT + TORQUE

This is the third step of analysis. In this step bolt pretension value, particular misalignment & torque inputs are given to coupling & results are tabulated. For all three cases bolt pretension & torque remains constant.
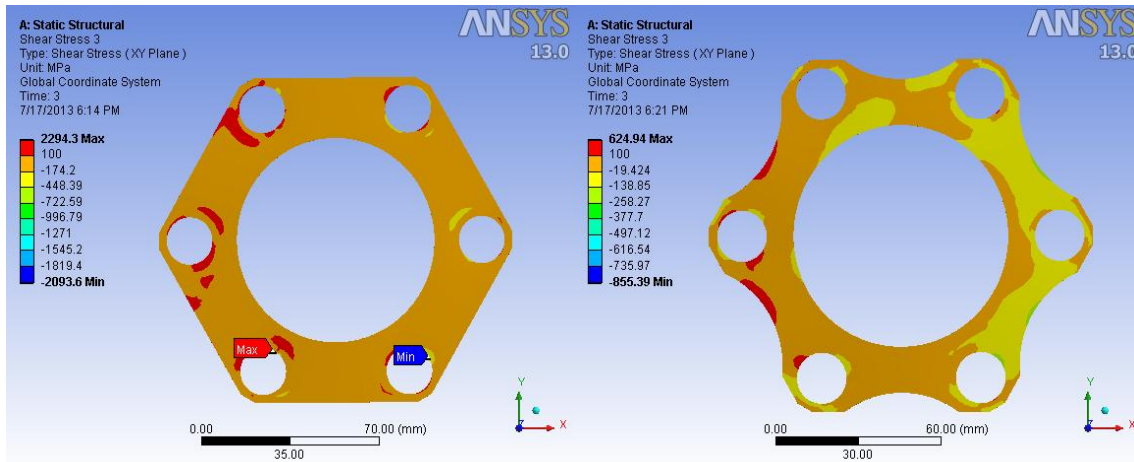
6.1 TOTAL DEFORMATION

The deformation at area in between two bolts is 0.3125 mm & for scalloped shim is 0.2 mm
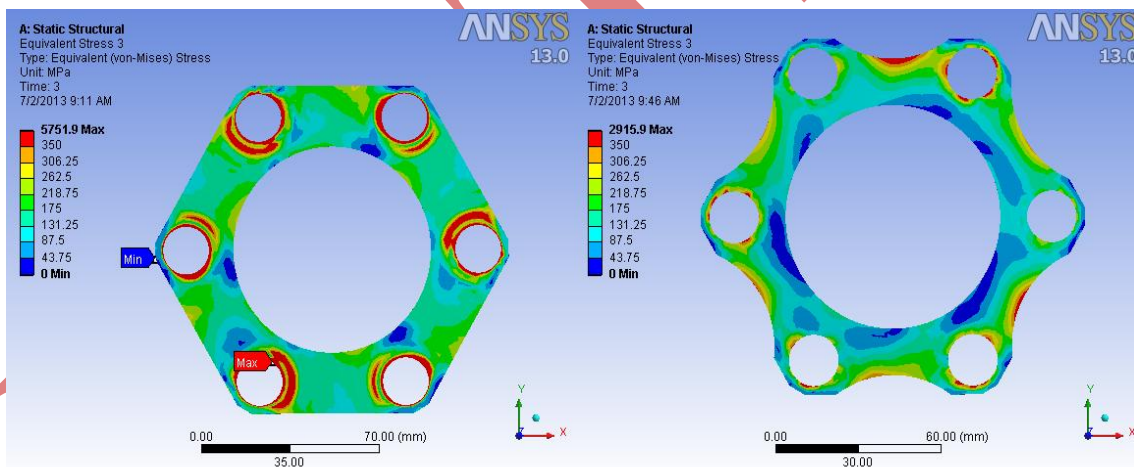
6.2 MAXIMUM PRINCIPLE STRESS
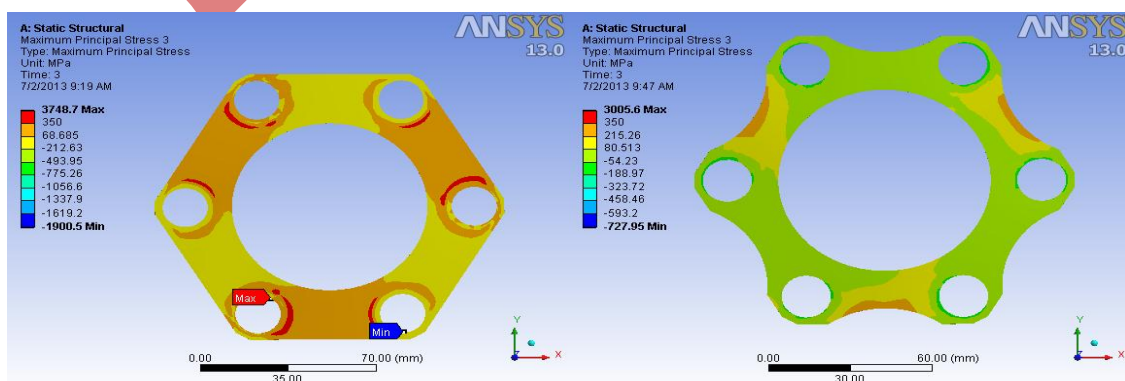
Principle stress developed at area between two bolts is 350 N/mm^2 for straight sided shim & for scalloped shim it is 215.26 N/mm^2.

6.3 SHEAR STRESS

Shear stress developed at area between two bolts is 100 N/mm^2 for straight sided shim & for scalloped shim it is 100 N/mm^2.
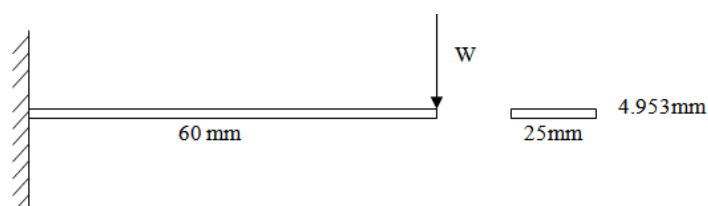
6.4 EQUIVALENT STRESSES (VON-MISES)

Equivalent stress developed at area between two bolts is 262.5 N/mm^2 for straight sided shim & for scalloped shim it is 262.5 N/mm^2.

VII. MATHEMATICAL RESULT-:

7.1 FOR BOLT PRETENSION (TIME-1)

Maximum deflection $\delta f = 0.34953$ mm.

We Know,

$$\delta f = \frac{wl^3}{3EI}$$

$$0.34953 = \frac{w \times 60^3}{3 \times 193257 \times \left(\frac{25 \cdot 4.953^3}{12}\right)}$$

$$= \frac{w \times 216000 \times 12}{1761170819}$$

W= 237.49 N

$M = w \times l = 237.49 \times 60 = 14249.584$ N.mm

We know that

$$\frac{M}{I} = \frac{\sigma b}{y} = \frac{E}{R}$$

$I = \frac{25 \times 4.953^3}{12} = 253.14$ mm^4 , $y = \frac{4.953}{2} = 2.4765$ mm

$\sigma b = \frac{14249.584}{253.14} \times 2.4765 = 139.405$ N/mm^2.

We have the formulae for Principle stress-:

$$Principle\ Stress = \frac{P}{2} \pm \sqrt{\left(\frac{\sigma b^2}{2}\right) + T^2}$$

Where $T = Shear\ Stress$ But $T = \frac{Say}{Ib}$

S= Shear force

For cantilever beam S = W

S0 S = W = 237.49 N

Shear stress is given by

$$T = \frac{237.49 \times (25 \times 4.953) \times 1.2382}{253.14 \times 25}$$
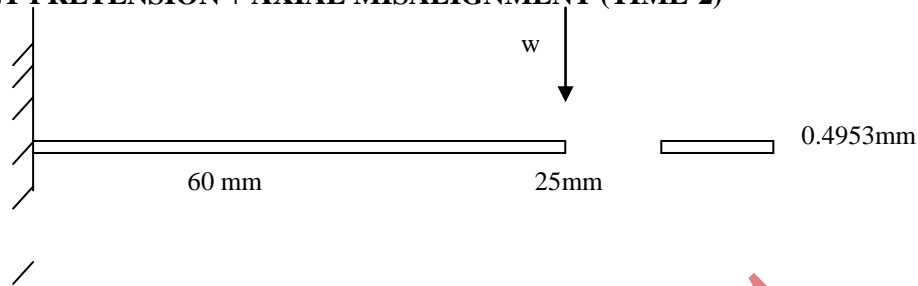
$T = 5.7536$ N/mm^2.

Principle stresses are given by-:

$$Principle\ Stress = \frac{\sigma b}{2} \pm \sqrt{\left(\frac{\sigma b}{2}\right)^2 + T^2}$$

$$= \frac{139.405}{2} \pm \sqrt{\left(\frac{139.405}{2}\right)^2 + (5.7536)^2}$$

$$= 69.702 \pm \sqrt{4858.36 + 33.0625}$$

Principle stresses = 139.6407 N/mm^2

7.2 FOR BOLT PRETENSION + AXIAL MISALIGNMENT (TIME-2)

Maximum deflection   δf = 0.8169 mm.

We Know, $\delta f = \dfrac{wl^3}{3EI}$

$$0.8169 = \frac{w \cdot 60^3}{3 \cdot 193257 \cdot \left(\frac{25 \cdot 4.953^3}{12}\right)}$$

$$= \frac{w \times 216000 \times 12}{1761170819}$$

W= 555.54 N

M = w × l = 555.54 × 60

M = 33303.25 N.mm

We know that

$$\frac{M}{I} = \frac{\sigma b}{y} = \frac{E}{R}$$

I= $\dfrac{25 \times 4.953^{\,3}}{12}$, I = 253.14 mm^4, y = $\dfrac{4.953}{2}$ = 2.4765 mm

$\sigma b = \dfrac{33303.25}{253.14} \times 2.4765 = 325.80$ N/mm^2.

We have the formulae for Principle stress-:

$$Principle\ Stress = \frac{\sigma b}{2} \pm \sqrt{\left(\frac{\sigma b^2}{2}\right) + \tau^2}$$

Where $\tau = Shear\ Stress$, But $\tau = \dfrac{Say}{Ib}$, S= Shear force, For cantilever beam S = W

S0 S = W = 555.54 N

Shear stress is given by

$$\tau = \frac{555.54 \times (25 \times 4.953) \times 1.2382}{253.14 \times 25}$$

τ = 13.459 N/mm^2.

Principle stresses are given by-:

$$Principle\ Stress = \frac{\sigma b}{2} \pm \sqrt{\left(\frac{\sigma b}{2}\right)^2 + \tau^2}$$
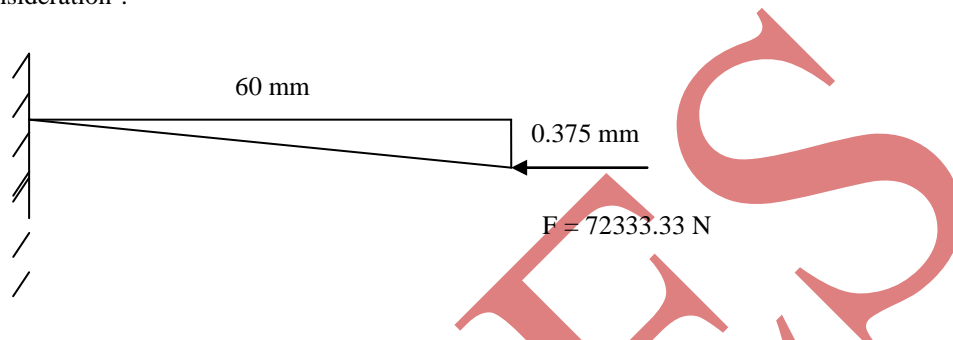
$$= \frac{325.80}{2} \pm \sqrt{\left(\frac{325.80}{2}\right)^2 + (13.459)^2}$$

$$= 162.9 \pm \sqrt{26536.41 + 181.145}$$

Principle stresses = 326.355 N/mm^2

7.3 FOR BOLT PRETENSION + AXIAL MISALIGNMENT + TORQUE (TIME-3)

For Torque Consideration-:

60 mm

0.375 mm

F = 72333.33 N

We have the formula for Torque required,

$T = Force * r$

Here, T = 4.34×10^6 N-mm, r = 60 mm.

Therefore $4.34 \times 10^6 = F \times 60$

F = 72333.33 N.

M = w × 1, M = 72333.33 × 0.375 = 27124.998 N.mm

$\sigma b = \frac{M}{I} \times y$

$$\sigma b = \frac{27124.998}{253.14} \times 2.4765 = 265.367 \text{ N/mm2}$$

W

Shear stress is given by-:

$T = \frac{Say}{Ib}$

S = 72333.33 N

a = 25 × 60
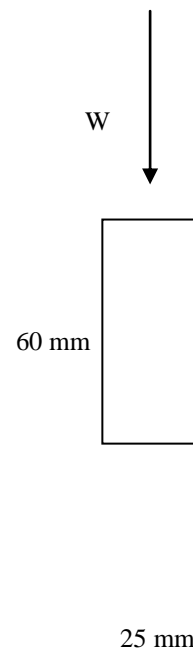
y = 15 mm

$I = \frac{bd^3}{12}$

$I = \frac{25 \times 60^3}{12} = 450000$ mm^4

60 mm

25 mm

$$T = \frac{72333.33 \times (25 \times 60) \times 15}{450000 \times 25}$$

$T = 144.66 \, N/mm^2$

Principle stresses are given by-:

$$Principle\ Stress = \frac{\sigma b}{2} \pm \sqrt{\left(\frac{\sigma b}{2}\right)^2 + \tau^2}$$

$$= \frac{265.367}{2} \pm \sqrt{\left(\frac{265.367}{2}\right)^2 + (144.66)^2}$$

Principle stresses = 328.977 N/mm^2

VIII. ANALYSIS FOR AXIAL MISALIGNMENT RESULT

Sr. No.	Load Cases		Mathematical Calculations	FEA Result	
				Straight Sided	Scalloped
1	Shear Stress	Bolt Pretension	5.7536	6.67	4.15
		Bolt Pretension + Axial Misalignment	24.2046	24.02	22.77
		Bolt Pretension + Axial Misalignment + Torque	144.66	148.94	145.82
2	Principle Stress	Bolt Pretension	139.6407	136.52	133.38
		Bolt Pretension + Radial Misalignment	326.355	332.64	204.63
		Bolt Pretension + Radial Misalignment + Torque	328.977	331.18	182.86
3	Equivalent Stress	Bolt Pretension	43.75 To 306.25	43.75 To 262.5	43.75 To 306.25
		Bolt Pretension + Angular Misalignment	43.75 To 306.25	43.75 To 262.5	43.75 To 306.25
		Bolt Pretension + Angular Misalignment + Torque	87.5 To 350	43.75 To 306.25	87.5 To 350

Table No. 7.1 - Shear Stress Result

IX. CONCLUSIONS

From all the above diagrams of straight sided disc, it is observed that the stress concentration is less at the middle portion between two bolts. Stress concentration is very less at the outer edge middle portion of the disc. If we compare the FEA results of both straight sided & scalloped disc, then it is observed that various induced stresses (Equivalent stresses, Shear stresses, Principal stresses) are less in scalloped disc as compare to straight sided disc. So vibrations induced & load acting on both driven & driving equipment are reduced. Therefore overall life, efficiency, cost & working ability of coupling will be improved. Scalloped discs are failing in the center section between the bolts and not at the flex point near the bolts. FEA analysis shows that higher working stresses are observed in the scalloped disc between and near the bolts. Therefore failure should typically occur at

the flexing points near the bolts. Failure in the center of the disc may be more related to stress concentrations resulting from the manufacturing process than to the application conditions.

Scalloped blades are lighter, less stiff and appropriate for custom, high speed applications such as turbine-driven compressors. FEA Analysis shows that centrifugal stresses are lower in the scalloped designs. Centrifugal stresses can become significant at speeds which are much higher than standard motor speeds, which is another reason why scalloped elements are preferred for custom or high speed applications.

Therefore the use of scalloped disc instead of straight sided disc is confirmed in the application of Disc coupling from FEA analysis.

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