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DESIGN AND ANALYSIS OF AN AUTOMOTIVE SINGLE PLATE CLUTCH

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RESEARCH ARTICLE

ABSTRACT: This paper presents the stresses and deformations of the assembly of the automotive single plate clutch depending on the applied materials. Structural analysis of the clutch was performed by using the finite element method for a repressive vehicle example - Toyota KUN 25. First, the input data for the numerical analysis was calculated. Numerical analysis was performed in the ANSYS software package. As a result, the values of stresses and deformations that occur on the clutch during the vehicle exploitation are obtained.

KEY WORDS: stress, deformation, material, ANSYS, vehicle exploitation

DIZAJN I ANALIZA AUTOMOBILSKE JEDNODISKOSNE SPOJNICE

REZIME: Ovaj rad predstavlja prikaz napona i deformacije sklopa automobilske jednodiskosne spojnice u zavisnosti od primenjenog materijala. Strukturna analiza spojnice izvršena je metodom konačnih elemenata za reperezetni primer vozila – Toyota KUN 25. Najpre su proračunati polazni podaci za numeričku analizu. Numerička analiza izvršena je u softverskom paketu ANSYS. Kao rezultat dobijene su vrednosti napona i deformacija koje se javljaju na spojnici u toku eksploatacije vozila.

KLJUČNE REČI: motori sa sagorevanjem, obrada signala

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1. INTRODUCTION

A clutch is a machine member used on the transmission shafts. Some friction plates, sometimes known as clutch plates are kept between these two members. This whole assembly is known as clutch as seen in Figure 1.
The clutch is based on the friction. When two friction surfaces brought in contact and pressed, then they are united due to friction force between them. This is the basic principle of clutch Narayan S. [1]. The friction between these two surfaces depends on the area of surface, pressure applied upon them and the friction material between them.

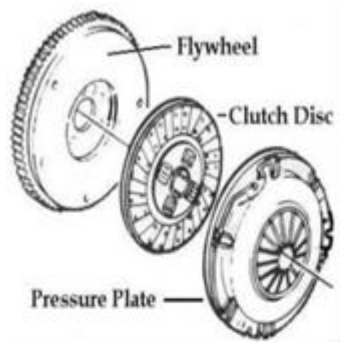

Figure 1. Clutch and flywheel [2]

The driving member of a clutch is the flywheel mounted on the engine crankshaft and the driven member is pressure plate mounted driving shaft to the driven shaft so that the driven shaft may be started or stopped at will, without stopping the driving as shown by Narayan S. [3, 4]. There are three states in which clutch can be found. These states are disengaged, clutch slipping and engaged. In automobile, a gearbox is required to change the speed and torque of the vehicle as analysed by Buzzoni [5] and Fang [6]. If we change a gear, when the engine is engaged with gearbox or when the gears are in running position then it can cause of wear and tear of gears. To overcome this problem a clutch is used between gearbox and engine. Clutch is the first element of power train. The main function of clutch is to engage and disengage the engine to transmission, when the driver need or during shifting of gear. When the clutch is in engaged position, the power flows from the engine to the wheel and when it is in disengage position, the power is not transmitted as seen by Zhao et al. [7].

The two main types of clutch are: positive clutch and friction clutch. Positive clutches are used when positive drive is required. The simplest type of a positive clutch is a jaw or claw clutch. A friction clutch has its principal application in the transmission of power of shafts and machines which must be started and stopped frequently. The force of friction is used to start the driven shaft from rest and gradually brings it up to the proper speed without excessive slipping of the friction surfaces. In automobiles, friction clutch is used to connect the engine to the drive shaft. The primary aim of this work is to design a rigid drive clutch system that meets multiple objectives such as Structural strength.

Gradual engagement clutches like the friction clutches are widely used in automotive applications for the transmission of torque from the flywheel to the transmission. The three major components of a clutch system are the clutch disc, the flywheel and the pressure plate. Flywheel is directly connected to the engine's crankshaft and hence rotates at the engine rpm. Bolted to the clutch flywheel is the second major component: the clutch pressure plate. The spring-loaded pressure plate has two jobs: to hold the clutch assembly together and to release tension that allows the assembly to rotate freely. Between the flywheel and the pressure plate is the clutch disc. The clutch disc has friction surfaces similar to a brake pad on both sides that make or break contact with the metal flywheel and pressure plate surfaces, allowing for smooth engagement and disengagement.

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When the clutch begins to engage, the contact pressure between the contact surfaces will increase to a maximum value at the end of the slipping period and will continue to stay steady during the full engagement period. During the slipping period, large amount of heat energy is generated at the contact surfaces, which gets converted to thermal energy by first law of thermodynamics. The heat generated is dissipated by conduction between the clutch components and convection to the environment. Another loading condition is the pressure contact between the contact surfaces that occurs due to the axial force applied the diaphragm spring. In addition to the above output responses, this work also considers the Vibrationa l characteristics of the clutch plate during the full engagement period. The engine and the transmission components experience dynamically varying loads during normal operation. This will cause vibrations and hence, one must design the clutch system so as to avoid resonance with the transmission and engine components as shown by Fang [6].

2. METHODOLOGY

Clutch friction linings are subjected to severe rubbing so that generation of heat in relatively short periods takes place. Therefore, the lining material should have a combination of the following properties to withstand the operating conditions [8]:

- Relatively high coefficient of friction under entire operating conditions.
- Maintenance of friction properties during entire working life.
- Relatively high energy absorption for short periods.
- Withstanding high pressure plate compressive loads.
- Withstanding high impacts of centrifugal force during gear changing.
- Adequate shear strength to transmit engine torque.
- High level of endurance in cyclic working without effecting friction properties.
- Good compatibility with cast iron facings over the entire range of operating temperature.
- A high degree of tolerance against interface contamination without affecting its friction take up and grip characteristics.

Considerations in Designing of a Friction Clutch:

- The following considerations must be kept in mind while designing a friction clutch. The suitable material forming the contact surfaces should be selected.
- The moving parts of the clutch should have low weight in order to minimize the inertia load, especially in high speed service.
- The clutch should not require any external force to maintain contact of the friction surfaces.
- The provision for taking up wear of the contact surfaces must be provided.
- The clutch should have provision for facilitating repairs.
- The clutch should have provision for carrying away the heat generated at the contact surfaces.
- The projecting parts of the clutch should be covered by guard.

Properties of some of commonly used materials for lining of friction surfaces are listed in Table 1.

Table 1: Properties of different materials [9]

Material	Friction Coefficient, f	Maximum Pressure, p_{max} [psi]
Cermet	0.32	*150*
Sintered metal (dry)	*0.29-0.33*	*300-400*
Sintered metal (wet)	*0.06-0.08*	*500*
Rigid molded asbestos (dry)	0.35-0.41	*100*
Rigid molded asbestos (wet)	0.06	*300*
Rigid molded asbestos pads	0.31-0.49	*750*

Rigid molded nonasbestos	0.33-0.63	*100-150*
Semirigid molded asbestos	0.37-0.41	*100*
Flexible molded asbestos	0.39-0.45	*100*
Wound asbestos yarn and wire	0.38	*100*
Wound asbestos yarn and wire	0.38	*100*
Woven cotton	0.47	*100*
Resilien paper (wet)	0.09-0.15	*400*

3. CALCULATIONS

Clutch plate of a Toyota Hilux KUN 25 was selected for sake of analysis. The vehicle has the following specifications:

- Maximum torque Nm
- Outer diameter mm
- Hub diameter with 21 splines.

3.1 Torque transmission under uniform pressure

This theory is applicable to new clutches. In new clutches employing a number of springs, the pressure can be assumed as uniformly distributed over the entire surface area of the friction disk. With this assumption, the intensity of pressure between disks, is regarded as constant.

3.2 Design Calculations

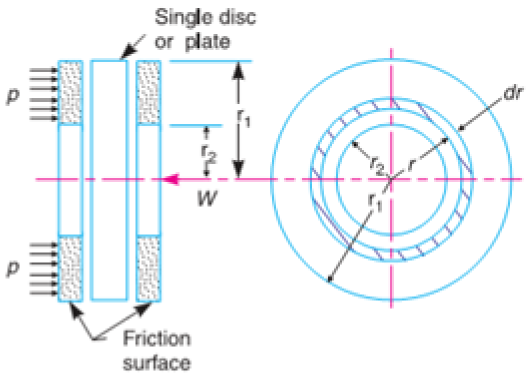

Figure 2. Clutch lining terminology [10]

Various Parameters used for calculations are as follows: (Total axial load of spring), (External radius of clutch plate), (internal radius of the clutch plate), (intensity of pressure between contacting surfaces), (coefficient of friction between contacting surfaces)

Frictional resistance force offered by ring is given by:

$$F_r = \mu \cdot (2 \cdot \pi \cdot p \cdot r \cdot \delta r) = 2 \cdot \pi \cdot \mu \cdot p \cdot r \cdot \delta r \tag{1}$$

Frictional torque acting on the ring is given as:

$$T_r = F_r \cdot r \tag{2}$$

$$T_r = 2 \cdot \pi \cdot \mu \cdot p \cdot r \cdot \delta r \cdot r = 2 \cdot \pi \cdot \mu \cdot p \cdot r^2 \cdot \delta r \tag{3}$$

This Torque transmitted by single plate clutch is obtained by considering following two assumptions theory.

3.3 Torque transmission under uniform wear

This theory is based on the fact that wear is uniformly distributed over the entire surface area of friction disk. This assumption can be used for worn out clutches/old clutches. The axial wear of the friction disk is proportional to frictional work. The work done by the friction is proportional to the frictional force ($\mu \cdot p$) and the rubbing velocity ($2 \cdot \pi \cdot r \cdot n$) where n is the speed of the disk in revolution per minute.

When the speed n and the coefficient of friction μ are constant for a given configuration, then $Wear \ \alpha \ pr$ [11].

According to this assumption, $p \cdot r = const$.

When the clutch plate is new and rigid, the wear at the outer radius will be more, which will reduce pressure at the outer edge due to rigid pressure plate. This will change pressure distribution. During running condition, the pressure distribution is adjusted such that the product (pr) is constant. Therefore,

$$p \cdot r = p_a \cdot r \tag{4}$$

Where p_a is the pressure at the inner edge of plate, which is also the maximum pressure.

The uniform-pressure theory is applicable only when the friction lining is new. When the friction lining is used over a period of time, wear occurs. Therefore, the major portion of the life of friction lining comes under uniform-wear criterion. Hence, in the design of clutches, the uniform wear theory is used.

The torque transmitting capacity can be increased by three methods:

- Using the friction material with a higher coefficient of friction (μ);
- Increasing the intensity of pressure (p) between disks; and
- Increasing the mean radius of friction disc $\dfrac{R+r}{2}$.

  There are five different materials utilized in modern clutch design [12]:
- “Organic” clutch material, which is a mix of fiberglass and other materials (including brass in some cases) molded or woven into a friction pad.
- Kevlar (and its cousin Twaron), which are synthetic fibers that make for extremely long- lasting (and very forgiving) clutch friction pads.
- Ceramic clutch material, which is mostly a mix of silicon dioxide and various metals and additives, sintered or brazed onto the clutch disc.
- Feramic clutch material, which is fairly similar to ceramic material, except containing a much larger percentage of metal.
- FeramAlloy, which is a new and superior alternative to feramic and ceramic clutch material.

The ratio of inner to outer diameter for maximum torque transmission is given by following equations:

- $X = \dfrac{D_i}{D_o} = 0.48$ for uniform pressure (5)
- $X = \dfrac{D_i}{D_o} = 0.577$ for uniform wear (6)

3.4 Calculation for the friction lining based on uniform wear

Baseline data for the calculation of the

$$X = \frac{D_i}{D_o} = 0.577 \text{ for uniform wear}$$

$$D_i = 0.577 \cdot D_o = 0.577 \cdot 260$$

$$D_i = 150 \text{ mm}$$

Outer radius $r_o = 130$ mm

Inner radius $r_i = 75$ mm

Torque transmission capacity $T = 343$ Nm

Effective mean radius $R = \frac{r_o + r_i}{2} = \frac{130 + 75}{2} = 102.5$ mm

The necessary relationships for the calculation are:

$$T = 2 \cdot \mu \cdot W \cdot R$$

$$Normal\ force = 2\pi C\left(r_o - r_i\right)$$

$$W = \pi \cdot p \cdot D_i \cdot \left(D_o - D_i\right)$$

a) Lining mate rial: Asbestos

Coefficient of friction $\mu = 0.35$

$W = 4780.5$ N

$p = 0.184$ N/mm^2

b) Lining mate rial: Sintered metal

Coefficient of friction $\mu = 0.29$

$W = 5769.5$ N

$p = 0.2245$ N/mm^2

c) Lining mate rial: Cermet

Coefficient of friction $\mu = 0.4$

$W = 4183$ N

$p = 0.1614$ N/mm^2

3.5 Calculation for the friction lining based on Uniform pressure

The ratio of inner to outer diameter for maximum torque transmission is:

- $X = \frac{D_i}{D_o} = 0.48$ for uniform pressure (7)
- $X = \frac{D_i}{D_o} = 0.577$ for uniform wear

$$D_i = 0.48 \cdot D_o = 0.48 \cdot 260 \tag{8}$$

$$D_i = 124.8 \text{ mm}$$

Baseline data for the calculation of the:

Outer radius $r_o = 130$ mm

Inner radius $r_i = 62.4$ mm

Torque transmission capacity $T = 343$ Nm

$$\text{Effective mean radius } R = \frac{2}{3}\frac{r_o^3 - r_i^3}{r_o^2 - r_i^2} = \frac{2}{3}\frac{130^3 - 62.4^3}{130^2 - 62.4^2} = 100.16 \text{ mm}$$

The necessary relationships for the calculation are:

$$\text{Torque transmission capacity (T): } Torque = n \cdot \mu \cdot W \cdot R$$

$$W = \pi \cdot p \cdot \left(r_o^2 - r_i^2\right)$$

a) Lining mate rial: Asbestos

$$\text{Coefficient of friction } \mu = 0.35$$
$$W = 9784.3 \text{ N}$$
$$p = 0.1145 \text{ N/mm}^2$$

b) Lining mate rial: Sintered metal

$$\text{Coefficient of friction } \mu = 0.29$$
$$W = 5904.43 \text{ N}$$
$$p = 0.1445 \text{ N/mm}^2$$

c) Lining mate rial: Cermet

$$\text{Coefficient of friction } \mu = 0.32$$
$$W = 5350.815 \text{ N}$$
$$p = 0.131 \text{ N/mm}^2$$

d) Lining mate rial: Ceramic

$$\text{Coefficient of friction } \mu = 0.4$$
$$W = 4280.65 \text{ N}$$
$$p = 0.1045 \text{ N/mm}^2$$

Table 2: Propert Comparison of lining materials with uniform wear and uniform pressure

Materials	Uniform wear		Uniform pressure	
	Axial force, [N]	Pressure, [N/mm^2]	Axial force, [N]	Pressure, [N/mm^2]
Asbestos	4780.5	0.184	4892.5	0.1145
Sintered	5769.5	0.2245	5904.43	0.1445
Cermet	5228.5	0.2015	5350.81	0.131
Ceramic	4183	0.1614	4280.65	0.1045

4. FEA ANALYSIS

The finite element analysis is the most widely accepted computational tool in engineering analysis [13]. Through solid modelling, the component is described to the computer and this description affords sufficient geometric data for construction of mesh for finite element modelling. In this project the clutch plate assembly using different lining materials was designed in the previous chapter. In this chapter finite element analysis is going to be done using Ansys software. In the following subchapters structural and thermal analysis is going to be done for each clutch assembly parts and compare different lining to choose the best lining, which were designed in the previous part. 3D modelling was done using Catia V5 for each plate assembly parts and now we are going to import those models to Ansys to do static structural analysis.

Table 3: Propert Comparison of lining materials with uniform wear and uniform pressure

Materials	Tensile yield strength [MPa]	Calculated *p* [N/mm^2]	Poisson ratio [-]	Modulus of elasticity [GPa]	Density [kg/m^3]
Asbestos	800	0.184	0.28	165	2800
Sintered metal	140	0.2245	0.24	115	6400
Cermet	1039	0.2015	0.23	380	5000
Ceramic	1138	0.1614	0.22	325	2130
Kevlar	3240	0.184	0.36	71	1470
Spring steel	1000		0.273	210	7861
Cast ion alloy ASTM A220	130		0.28	110	7200

4.1 Clutch plate assembly with asbestos lining

Clutch disk shown in Figure 3, in the form of finite elements mash.

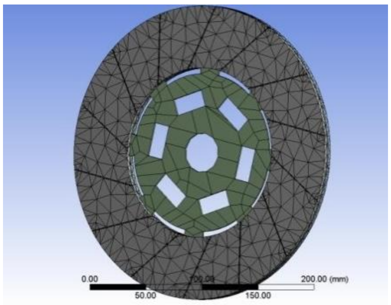

Figure 3. Meshed clutch disc

Material property and geometry da ta were defined (as per Table 3). The Environment (a combination of loads and supports) was defined as follows:

- Moment: 171.5 Nm (each side)
- Pressure: 0.184 Mpa.

The Model was submitted to the ANSYS solver and the solutions for the Equivalent von-Mises stress, Total Deformation and Stress Tool were obtained. Figure 4 shows the distribution of equivalent von-Mises stress over the clutch plate. Figure 5 shows the distribution of total deformation over the clutch plate. Figure 6 shows the distribution of Factor of Safety (Stress Tool) over the clutch plate.

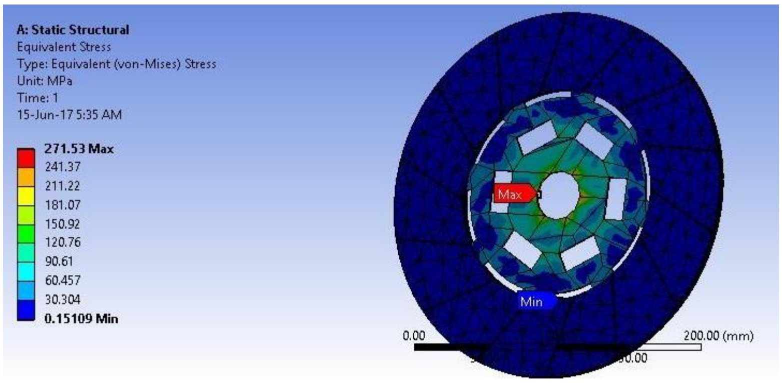

Figure 4. The equivalent von-Mises stress plot for the clutch plate

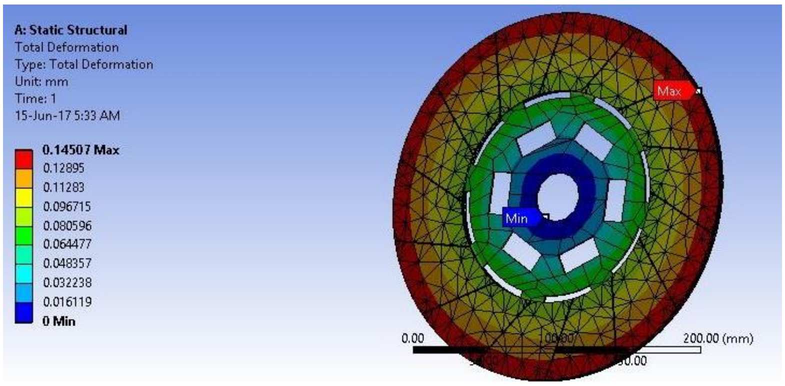

Figure 5. The Total Deformation plot for the clutch plate

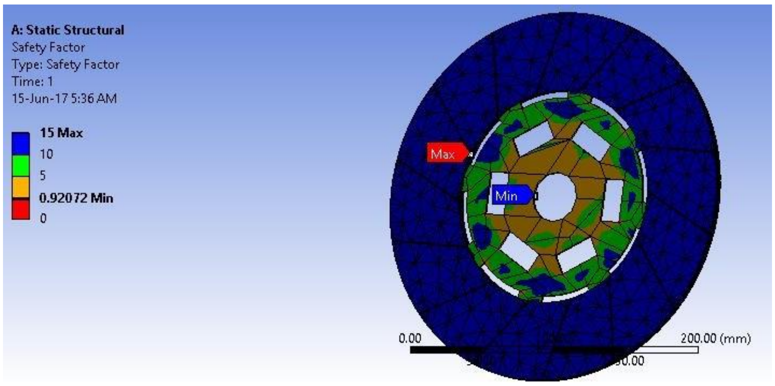

Figure 6. The Factor of Safety (Stress Tool) plot for the clutch plate

4.2 Clutch plate assembly with sintered metal

A Mesh was created (Dividing the model into small elements). Material property and geometry data were defined (as per Table 3). The Environment (a combination of loads and supports) was defined as follows:

- Moment: 171.5 Nm (each side)
- Pressure: 0.2245 Mpa.

The Model was submitted to the ANSYS solver and the solutions for the Equivalent von-Mises stress, total Deformation and Stress Tool were obtained as seen in Figure 7–9.

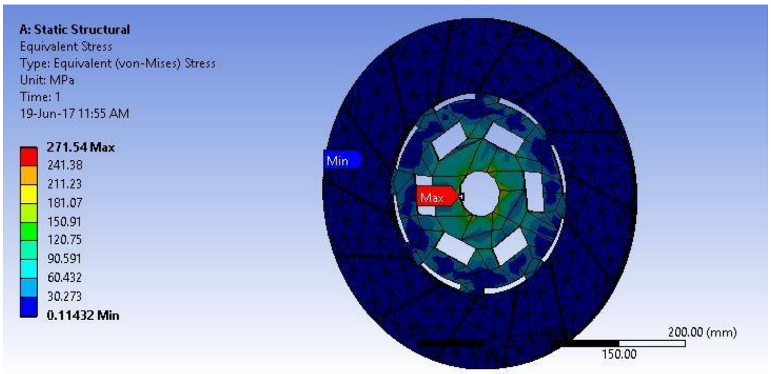

Figure 7. The equivalent von-Mises stress plot for the clutch plate

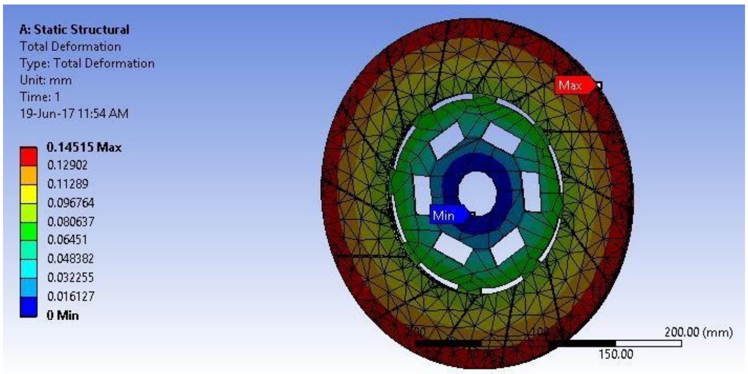

Figure 8. The Total Deformation plot for the clutch plate

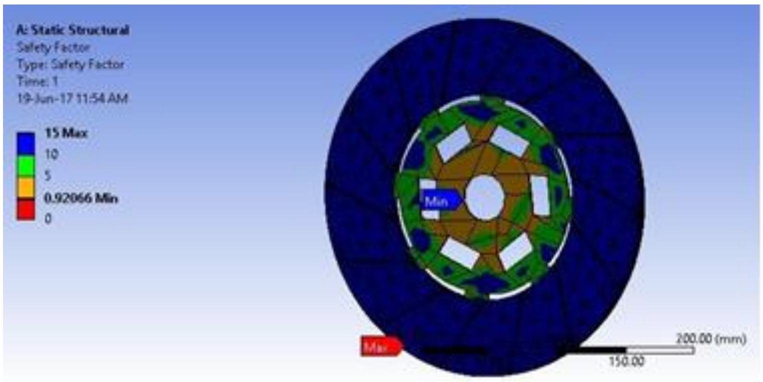

Figure 9. The Factor of Safety (Stress Tool) plot for the clutch plate

4.3 Clutch plate assembly with cermet

A Mesh was created (Dividing the model into small elements). Material property and geometry data were defined. The Environment (a combination of loads and supports) was defined as follows:

- Moment: 171.5 Nm (each side)
- Pressure: 0.2015 Mpa.

Obtained results are shown on Figures 10-12.

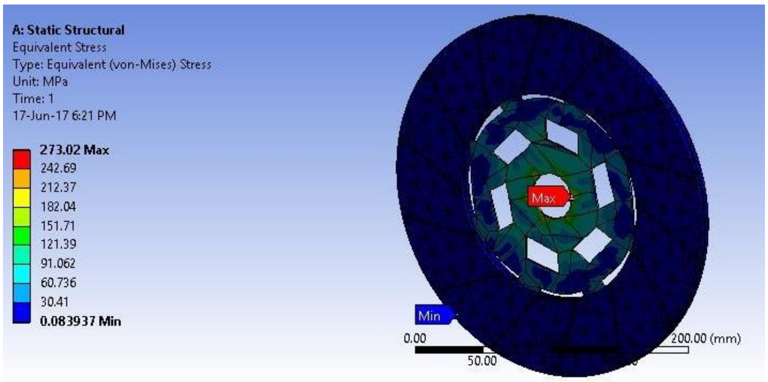

Figure 10. The equivalent von-Mises stress plot for the clutch plate

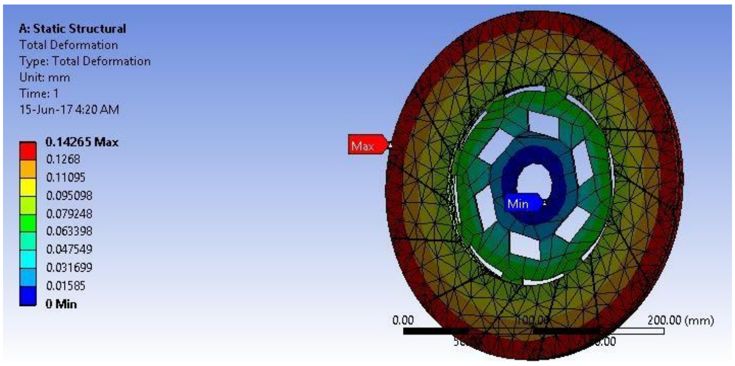

Figure 11. The Total Deformation plot for the clutch plate

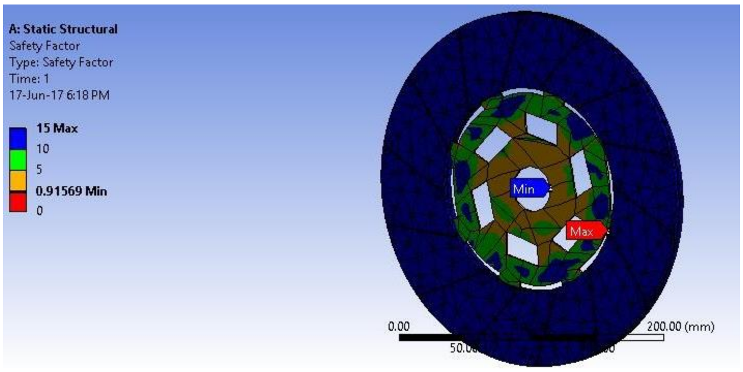

Figure 12. The Factor of Safety (Stress Tool) plot for the clutch plate

4.4 Clutch plate assembly with ceramic

A Mesh was created (Dividing the model into small elements). Material property and geometry data were defined. The Environment (a combination of loads and supports) was defined as follows:

- Moment: 171.5 Nm (each side)
- Pressure: 0.1614 Mpa.

Obtained results are shown on Figures 13-15.

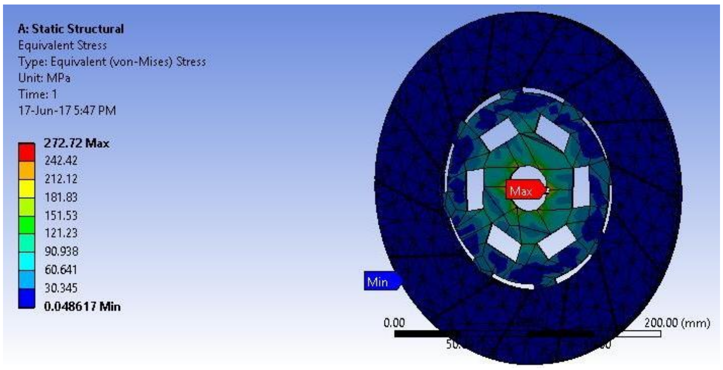

Figure 13. The equivalent von-Mises stress plot for the clutch plate

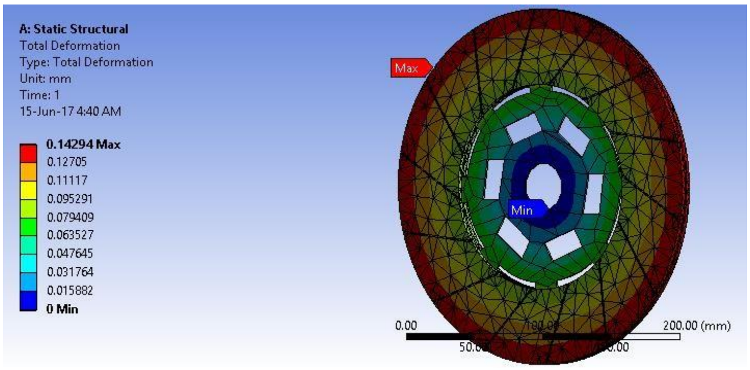

Figure 14. The Total Deformation plot for the clutch plate

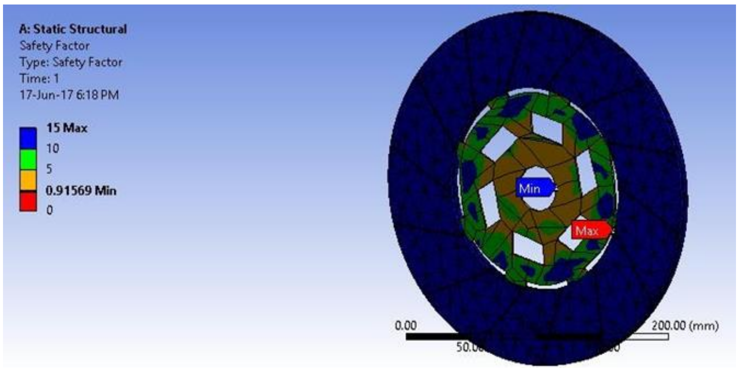

Figure 15. The Factor of Safety (Stress Tool) plot for the clutch plate

5. CONCLUSIONS

In this work, clutch plate of an automotive clutch assembly has been designed using different materials and simulated using ANSYS software for comparison. Among those different lining materials cermet friction material was selected. Generated Heat between friction disk and flywheel, which is the main reason for clutch wear can be minimized by selecting suitable material. A good contact pressure also reduces wear during slippage time.

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