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Design Verification of Tractor Clutch Cover under High Centrifugal Effect

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Abstract

Clutch components rotating at high speed remains under the influence of high centrifugal forces. Therefore; when the damage starts, the rotation speed should be known. For this purpose, the burst test is performed. Moreover, finite element analysis method is a powerful tool for the simulation of test environment and design verification process. Thus, time and cost can be saved by using finite element method. This paper presents an approach for validation process of tractor clutch cover under the high centrifugal forces. Firstly the real burst test setup is explained and the test conditions are described. After that, developed finite element model which simulate the real test conditions is defined. As a result of the study, the tractor clutch cover is durable at 6560 rpm. When it is compared real working conditions the safety factor of 3.5 is calculated for the clutch cover.

Keywords: Burst Test, Centrifuge Effect, Finite Element Analysis, Tractor Clutch Cover

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1. Introduction

Tractor clutch is one of the main components of the tractor transmission system. The tractor clutch transmits power from engine to both gearbox and Power Take-Off (PTO) shaft. During this transmission, it rotates its own axis between 0 and 2500 rpm. Thus the tractor clutch is exposed to high centrifugal forces. This force directly affects the clutch cover. Because the cover is the biggest component of the clutch and other components are assembled on it. Due to the importance of the clutch stability, the durability of the cover must be investigated under high centrifugal force.

In the literature, it is hard to see any investigation about the centrifugal effect of tractor clutch cover and other components. Generally, researchers are interested in automotive clutch systems instead of heavy vehicles. Abdullah et al. [1] create a finite element model to define contact pressure and centrifugal force, which are occurred diaphragm spring, the effect on the pressure plate, disk and flywheel. This model is solved by different solving methods which are located in ANSYS and results are compared each other. Also in this study, the effect of contact stiffness factor on the pressure between the contact surfaces, stress and deformations are investigated.

Karpat et al. [2] are investigated tractor finger mechanism by using finite element method in ANSYS. PTO finger working conditions are modeled in this study. The stress and deformation distribution are defined for different sheet metal thickness of the PTO finger. As a result of the study for each 0.5 mm sheet metal increment cause 13% stress decrement. Dogan et al. [3] are re-design failed tractor clutch finger mechanism by using topology and shape optimization. They defined the optimum finger geometry by topology optimization and then by using shape optimization he precise dimensions of the fingers are designed. As a result of this study, the stress reduction of 25.3% is achieved. The rigidity is improved up to 27.9% and the finger fatigue life is %50 extended. This process also gives as a design procedure in another study by Dogan et al. [4] The authors have developed new design procedure for tractor clutch PTO finger mechanism by using finite element analysis (FEA), experimental tests and response surface methods.

Özbakış [5] has investigated characteristics of diaphragm spring used in the clutch systems and optimized diaphragm springs depending on working conditions. Firstly, standard diaphragm spring characteristics are determined by using finite element method and theoretical calculations. Then, manufactured two different spring characteristics are simulated in the computer environment and also the characteristics of the springs are defined experimentally and the results are compared. Li-jun et al. [6] optimized basic parameters to improve the clutch quality and to design the clutch with smallest diameters. The optimization problem is solved Matlab optimization toolbox. Optimized clutch components redesigned by using Pro/E software and these components are validated by using finite element method.

Purohit et al. [7] designed a frictional clutch. Clutch plate, pressure plate and diaphragm spring is modeled. These components are analyzed in static structural module and stress and displacement values obtained from the FEA. Analysis results are evaluated in terms of three different materials and three different safety factors are determined for each component. Danev et al. [8] investigated the effect of diaphragm spring shape on the characteristics of diaphragm spring. The authors changed the diaphragm spring shape. In order to increase the moment of inertia of each spring finger, instead of designing the spring fingers straight, the fingers are designed in different forms and the moment of inertia is increased. Thus, higher pressure forces are obtained in the same volume and higher torque values can be transmitted. Nam et al. [9] optimized diaphragm spring design parameters by using finite element method and sequential linear programming methods. At first the design parameters is defined and the effects of design parameters on the diaphragm spring is evaluated by using FEA. Then the optimization procedure is follow and optimum diaphragm spring dimensions are defined.

Abdullah et al. [10] investigated effect of grove geometry on the clutch surface temperature by using finite element method. The ratio of total grove area and total contact surface area effect is investigated. Also new grove geometry is proposed to decreased clutch surface temperature. Li et al. [11] investigated the effects of clutch pedal force and stroke on driving comfort. Theoretical and practical calculations are done for clutch pedal force and stroke. Also real test are done for the definition of the optimum pedal force and stroke. As a result of this study, the optimum clutch force is defined between 100-125 mm and the optimum clutch store is defined between 85-100 N. Kim et al. [12] conduct a series of experiments under different land and speed conditions. The aim of the study is to develop a model for the fatigue life of the tractor clutch. The real time torque values for the drive shaft are acquired by using radio telemetry system. Load spectra obtained from different activities are compared. Cycle counting algorithms are used to calculate life expectancy data.

The objective of this study is to describe the design verification of the tractor clutch cover. Verification process of the clutch cover is explained. As a result of the study developed finite element model is matched well with the experimental results.

2. Material and Method

2.1 Experimental Test

To define the behavior of the clutch cover under high centrifugal effect, burst test, in other words, the centrifugal test is done (Fig. 1). The centrifugal test rig activation principle is designed based on the hydraulic turbine and it enables to test wide range of clutch diameter capacity from 150 mm to 450 mm. Maximum rotational speed of the test rig reaches up to 20000 rpm that gives opportunity to study each rpm range for vehicles such as passenger cars, tractors and trucks. Average temperature during centrifugal test rig is supposed to reach 250 °C under high rotational conditions.

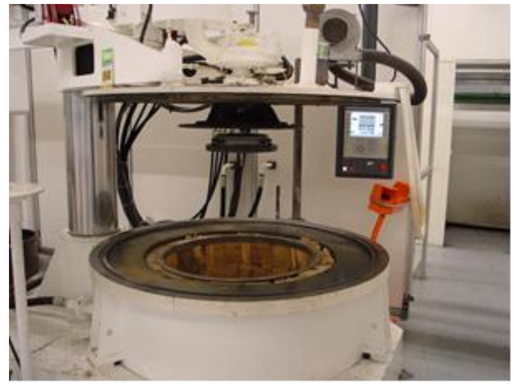

Fig 1. Centrifugal Test Rig with Hydraulic Turbine

In this test, the clutch cover is rotated on its axis between 0 – 6000 rpm. When the speed is reached 6000 rpm the test is continued 60 seconds. The validation criterion is no breakage at 6000 rpm for 60 seconds. If there is a breakage on the clutch cover, it means that the stress values are higher than the yield stress of the clutch cover material. In real conditions, the tractor clutch rotates between 0- 2500 rpm on the tractor. According to customer expectations, this test is done 6000 rpm. The safety factor of this test is nearly 2.5. In this test the clutch house temperature is adjustable thus; the temperature effect on the clutch cover could be seen. Moreover, tractor clutch cover with different diameters can be validated.

2.2 Finite Element Method

Tractor clutch cover design validation processes could be done experimentally as it is stated Chapter 2.1. These processes can be done with experimental tests but it is so costly and time-consuming. Computer-aided design (CAD) and finite element analysis (FEA) is accepted across a wide range of industries as a crucial tool for product design and optimization and validation phases. They are essential in order to predict accurately for the safety performance of automotive parts. FEA techniques allow the design engineers to predict the maximum stress locations and life of the parts before making the prototype. So these techniques can reduce the conceptual design costs.

To carry out FEA analysis, CAD geometry is needed. Thus, the first step of finite element analysis is CAD data creation of the tractor clutch cover. Solid model of clutch cover is designed in three-dimensional views with the help of the CATIA V5 Software for simulation purposes (Figure 2). The clutch cover is made of cast iron FGL 250 and the cover is manufactured by casting.

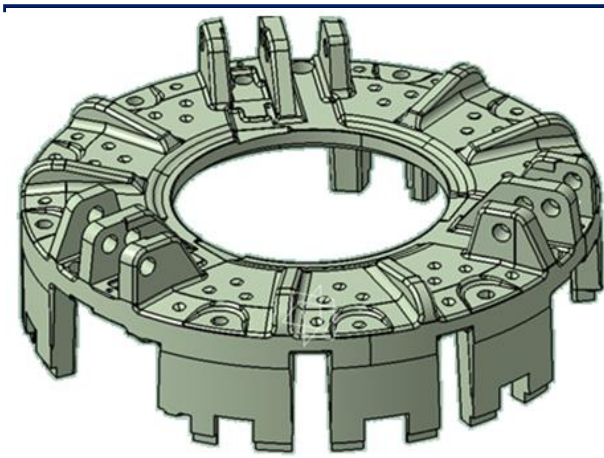

Fig 2. Solid model of the tractor clutch cover

After creation of the solid CAD model of the tractor clutch cover, it is imported ANSYS Workbench 16.0 for the creation of the finite element model. Because of the clutch cover cyclic symmetry conditions. 1/3 part of the clutch cover is enough to simulate the real conditions. Thus in the finite element analysis, 1/3 part of the cover is used. In this way, the number of nodes and elements are reduced 3 times so the analysis time reduced 3 times. In the mesh model, because of the complexity of the tractor clutch cover tetrahedral elements are used to create mesh construction which is consisting of nearly 350000 elements and 800000 nodes (Figure 3).

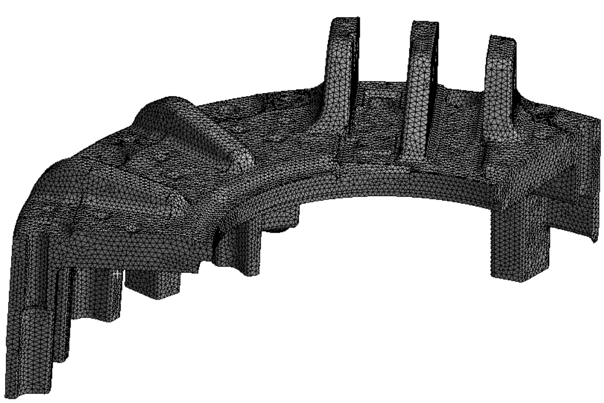

Fig 3. Mesh Model of the Clutch Cover

Loads and boundary conditions which are applied to the finger are defined in ANSYS. Because of the different rotation speed, the analysis is divided several steps. Each step the rotational velocity and pressure force depending on rotational velocity defined separately (Table 1).

Frictionless supports are defined in the cyclic symmetry regions. Pin and bottom region displacement values defined ‘’0 mm’’ due to structural constraints. Additionally, screw region of the cover defined fixed support in the analysis. All boundary conditions are shown in the (Figure 4).

Table 1. Time-Varying Rotation Speed and Pressure Force

Steps	Time (s)	Rotation Speed (rpm)	Pressure Force (N)
1	1	1200	586
2	2	2400	1173
3	3	3600	1760
4	4	4800	2347
5	5	6000	2934
6	6	6800	2934
7	7	7600	2934
8	8	8400	2934
9	9	9200	2934
10	10	10000	2934

In deciding whether a tractor clutch cover design is acceptable, Von-Mises stress distribution is investigated. In the linear static analysis, the examination of stress data is the responsibility of the user of FEA software. The software does not produce results that indicate failure.

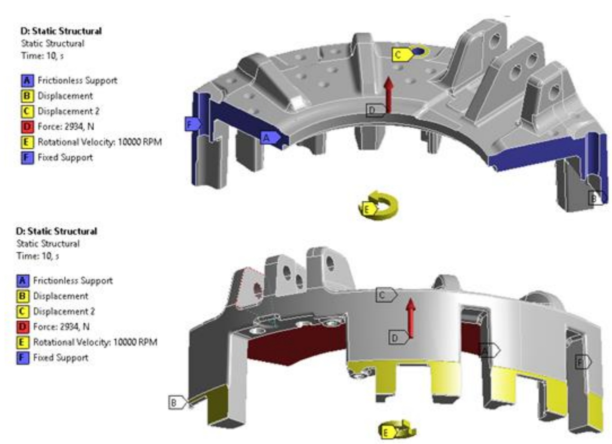

Fig 4. Boundary conditions of the tractor clutch cover

The failure criterion is Max. $\boldsymbol{\sigma}_{vm}=\boldsymbol{\sigma}_{y}$

Here $\boldsymbol{\sigma}_{y}$ is the yield stress of the material and $\boldsymbol{\sigma}_{vm}$ is the stress gained from FEA.

3. Results and Discussion

The linear static FE analysis is conducted by using ANSYS static structural module, FEA revealed locations of high stresses and all stress distribution of the tractor clutch cover. The highest stresses are observed at the screw region whose value is 263.97 MPa (Figure 5) at 10000 rpm. The maximum displacement is occurred as 1.7404 mm

This speed value is too much for the real conditions. In the experimental test maximum speed of the tractor clutch cover is 6000 rpm thus the results in 6000 rpm is more meaningful. In 6000 rpm, stresses are also observed at the screw region, the value is 238.83 MPa (Figure 6).

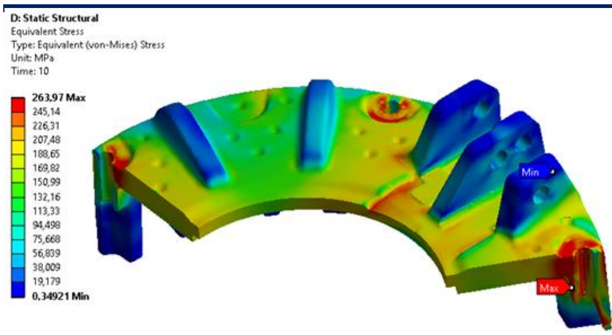

Fig 5. Stress distribution of the cover at 10000 rpm

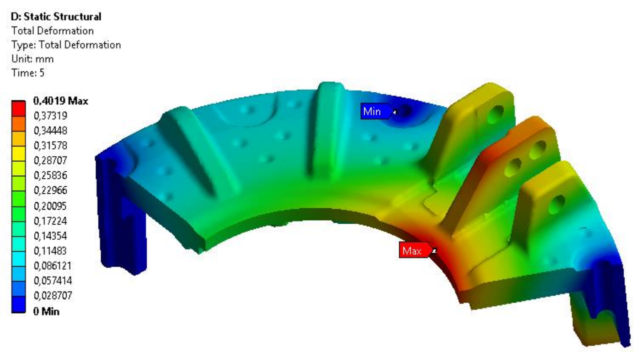

Fig 6. Stress distribution of the cover at 6000 rpm

The maximum displacement occurs as 0.4019 mm (Figure 7).

Table 2. Results of different speeds

Test Speed (rpm)	Eq. Stress (MPa)	Stress Criteria (MPa)	Dis-place-ment (mm)	Ma-terial
6000	238.83	250	0.4019	FGL 250
10000	263.97		1.7404	

As it is stated in Table 2 maximum equivalent stress are higher than the material stress criteria of the material thus in this speed some deformations could be seen, however, in real conditions tractor clutch cover do not reach this speed value.

Since the maximum equivalent stress is below the criteria 250 MPa for FGL 250) for burst test conditions, the clutch cover is validated for the speed of 6000 rpm. But it cannot withstand up to the speed of more than 6560 rpm. (Fur thermore; in real conditions, at 2500 rpm the clutch cover is very durable. The maximum stress occurs at 2500 rpm

nearly 70 MPa. Thus; in the normal working conditions, the safety factor of the clutch cover is 3.5 (Figure 8). In this clutch cover, also the experimental results are the same the validation criterion is no breakage at 6000 rpm for 60 seconds. At the end of the experimental test, there is no crack or breakage occurred on the cover.

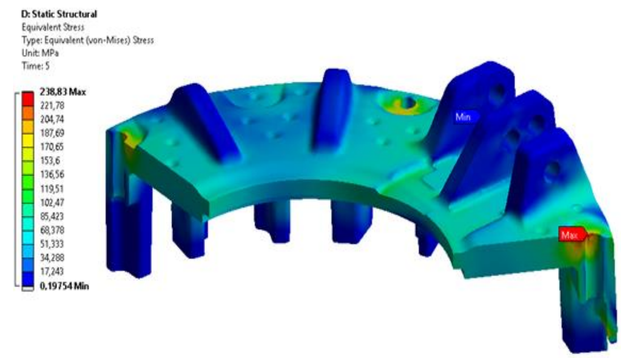

Fig 7. Displacement distribution of the cover at 6000 rpm

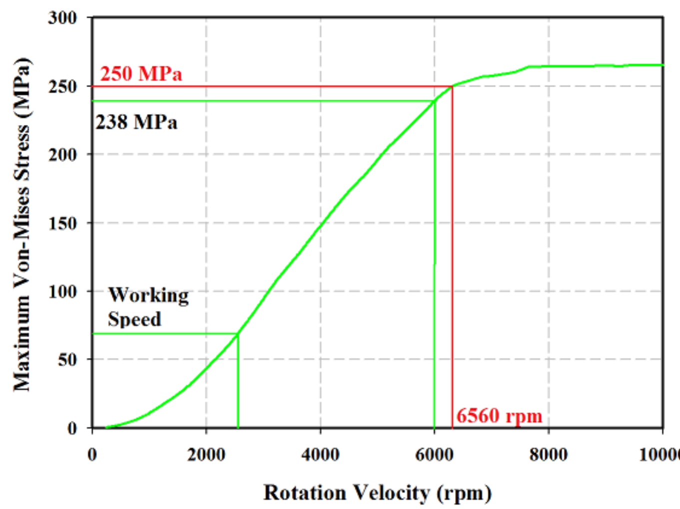

Fig 8. Maximum stress values depending on rotation velocity

4. Conclusion

In this study, the centrifugal effect on tractor clutch cover is investigated both experimentally and by using finite element method. In the experiment, the centrifugal test rig used to determine clutch cover durability. At 6000 rpm for 60 seconds the test is done and there are no cracks or breakage occurred. In the second part of the study, a finite element model is created to simulate the experimental test. For different rotational velocities which are a change from 0 – 1000 rpm applied on the clutch cover. It is seen that at 6000 rpm the maximum stress occurred 238 MPa which is below the material strength. Thus the cover is validated. The speed which more than 6560 rpm the maximum stress values reach the material strength.

Acknowledgment

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