Optimization of Shape and Design Parameters of the Rigid Clutch Disc Using FEM

Oday I. Abdullah
Research associate
Dept. of System Technology and
Mechanical Design Methodology
Hamburg University of Technology
Germany

Josef Schlattmann
Professor
Head of Dept. of System Technology and
Mechanical Design Methodology
Hamburg University of Technology
Germany

Emir Pireci
Diploma student
Dept. of System Technology and
Mechanical Design Methodology
Hamburg University of Technology
Germany

A friction clutch is an essential component in the process of power transmission. Due to this importance, it's necessary to investigate the stresses and vibration characteristics of the rigid clutch disc to avoid failure and obtain optimal weight and cost. This work presents the numerical solution of computing the stresses and deformations during the steady-state period, as well as the vibration characteristics of the rigid clutch disc. Furthermore, new models for rigid clutch disc have been suggested. The response of the new suggested models have been compared to the reference model. The numerical results show that the stresses and vibration characteristics of rigid clutch disc can be controlled by adjusting design parameters. They show as well that the suggested models improve the response of the friction clutch considerably. The ANSYS/WORKBENCH14 and SolidWorks 2012 have been used to perform the numerical calculation in this paper.

Keywords: *rigid clutch disc, FEM, stresses and deformation, modal analysis.*

1. INTRODUCTION

A common application of the clutch is in automotive vehicles where it is used to connect the engine to the gearbox. The new direction of development of the automotive vehicle ride comfort and driving smoothness is associated with the advancement of the machine parts design, e.g., rigid clutch disc is considered as essential part to transfer torque from the driving to driven shaft in a way that is consistent with the comfort and robustness purposes [1-3].

Naunheimer and Bertha [4] presented the design procedures for the friction clutch, and the parameters that affect the clutch performance. Also, they present different types of design for the clutch plate for passenger cars (Fig. 1 a–c) and commercial vehicles (Fig. 1. d–e).

There are many companies that produce the rigid clutch plate passenger cars (Fig. 1.c) such as LUK Limited Liability Corporation [5] and ZF SACHS Race Engineering Limited Liability Corporation [6].

The failures of the rigid drive disc of clutch due to the stress concentration have been investigated using finite element method. Three-dimensional model has been used to obtain the stress and deformations [7 & 8].

In the present research, a numerical technique is used to simulate new models for rigid clutch disc. Shape optimizations of the proposed models are performed to improve the vibrational characteristics of the system and hence the absorption of shocks that occur during clutch engagement. The finite element method has been used to compute the stresses and deformations during the steady-state period, and the vibration characteristics of the rigid clutch disc.

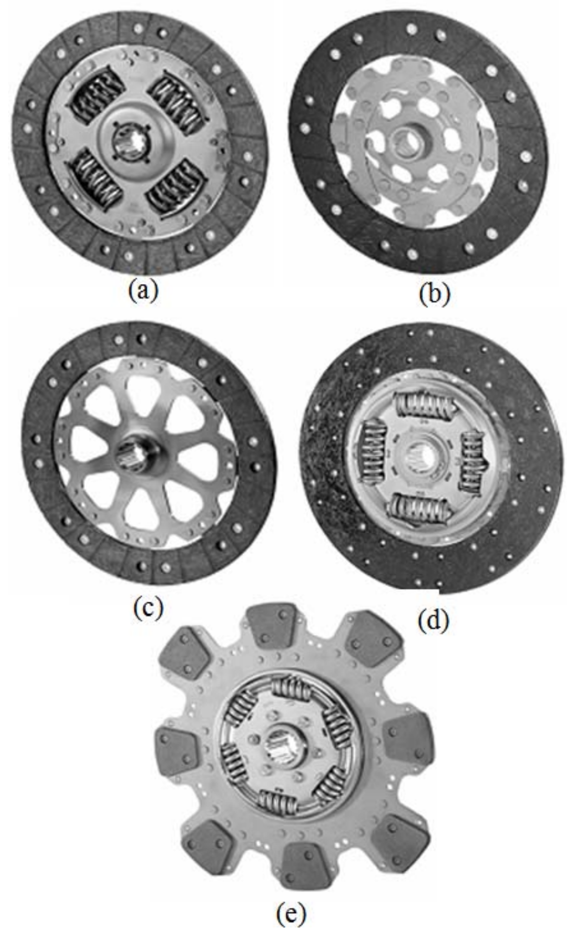


Figure 1. Clutch plates (ZF Sachs). a Clutch plate with torsion damper (passenger car); b flexible clutch plate (passenger car); c rigid clutch plate (passenger car); d clutch plate with torsion dampers (commercial vehicle); e clutch plate with cerametallic lining pads (commercial vehicle) [4]

Received: July 2013, Accepted: November 2013
Correspondence to: Oday I. Abdullah
Hamburg University of Technology,
Denickestraße 17, D- 21073 Hamburg
E-mail: oday.abdullah@tu-harburg.de

2. FUNDAMENTAL PRINCIPLES

The main system of the friction clutch consists of pressure plate, clutch disc and flywheel as shown in figure 2. When the clutch starts to engage, slipping will occur between contact surfaces due to the difference in the velocities between them (slipping period). After this period all contact part are rotating at the same velocity without slipping (full engagement period). During the first period the transmitted torque starts from zero and increases to maximum value T at the end of this period and in the second period the transmitted torque will stay steady. Figure 3 shows the transmitted torque during the engagement cycle of the clutch, where ts is the slipping time and T is the transmitted torque by clutch.

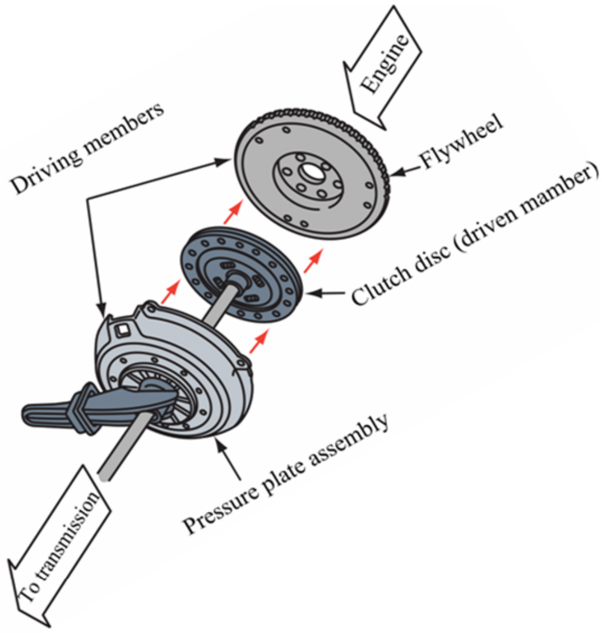


Figure 2. The main parts of clutch system

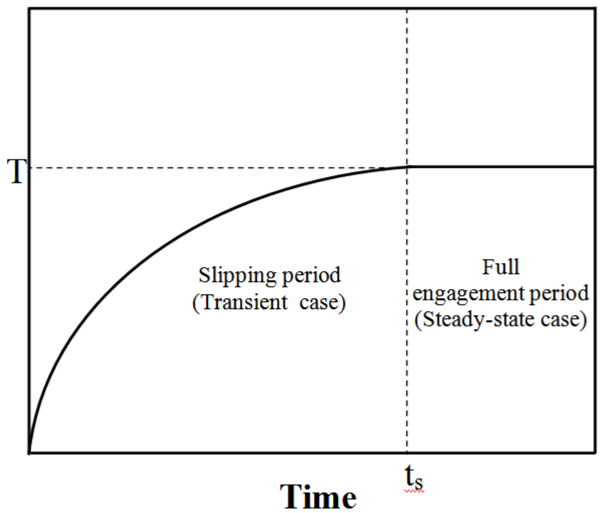


Figure 3. The transmitted torque during the engagement cycle of the clutch

3. MATHEMATICAL MODELS

This section presents the load and boundary conditions of the rigid clutch disc. The total frictional torque will be transmitted by frictional lining assuming the uniform wear between the contact surfaces is [9],

$$T = \mu\, n\, F \left(\frac{r_o + r_i}{2} \right) = \mu\, n\, F\, r_{eff} \tag{1}$$

where *μ, n, F, ri, ro, reff* are the coefficients of friction, the number of friction surfaces, axial force, inner disc radius, outer disc radius and the effective disc radius, respectively. The same torque will be transmitted by the rivets which connect the axial cushion and the drive disc of clutch. The total force of rivets is

$$f_{tr} = \frac{T}{r_R} \tag{2}$$

where r_R is the distance from the center of drive disc to the center of the rivet. The force for each rivet is

$$f_r = \frac{f_{tr}}{n_r} \tag{3}$$

where n_r is the total number of rivets. Figure 4 shows the load conditions on the clutch disc. Figure 5 shows the reference model and the suggested models (2-4,5) for the drive disc of clutch disc. All suggested model have approximately the same mass (≈ 205g). The factor of safety for the structural applications is

$$f_s = \frac{\sigma_y}{\sigma_{al.}} \tag{4}$$

where σ_y and $\sigma_{al.}$ are the yield stress and the allowable working stress. The value of the static safety factor is approximately 3.0 in many fields of mechanical engineering [10]. Figure 6 shows the suggested models manufactured by the company LZN Laser Zentrum Nord GmbH using modern production process of laser manufacturing (polyamide powder).

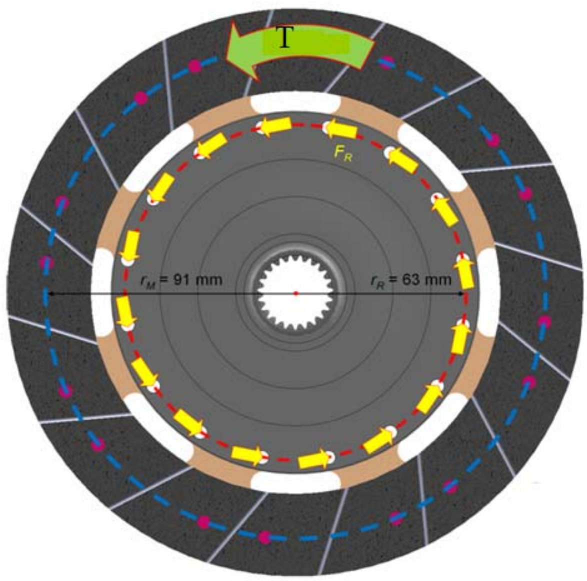
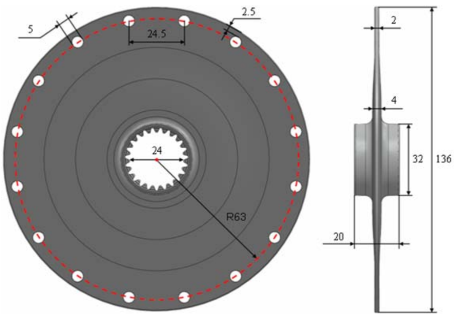
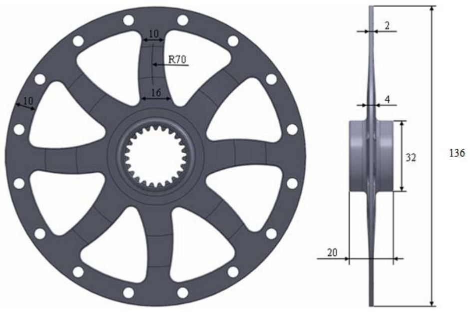


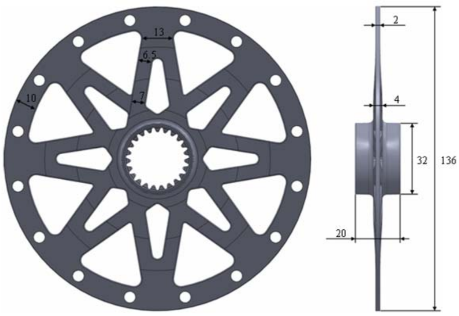
Figure 4. The friction clutch with load condition



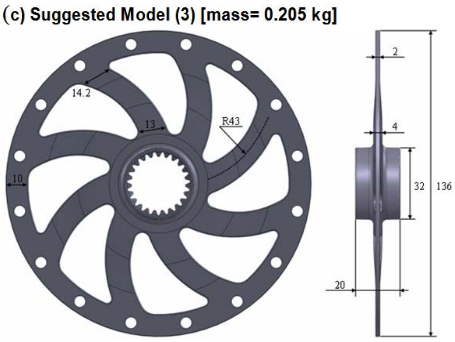
(a) The reference model (1) [mass=0.3056 kg]



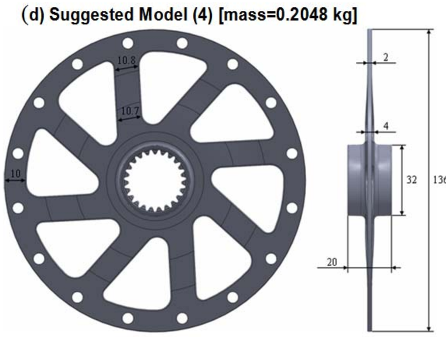
(b) Suggested Model (2) [mass=0.2046 kg]



(c) Suggested Model (3) [mass= 0.205 kg]



(d) Suggested Model (4) [mass=0.2048 kg]



(e) Suggested Model (5) [mass= 0.2048 kg]

Figure 5. The reference and suggested models for the rigid drive disc [all dimensions in mm]

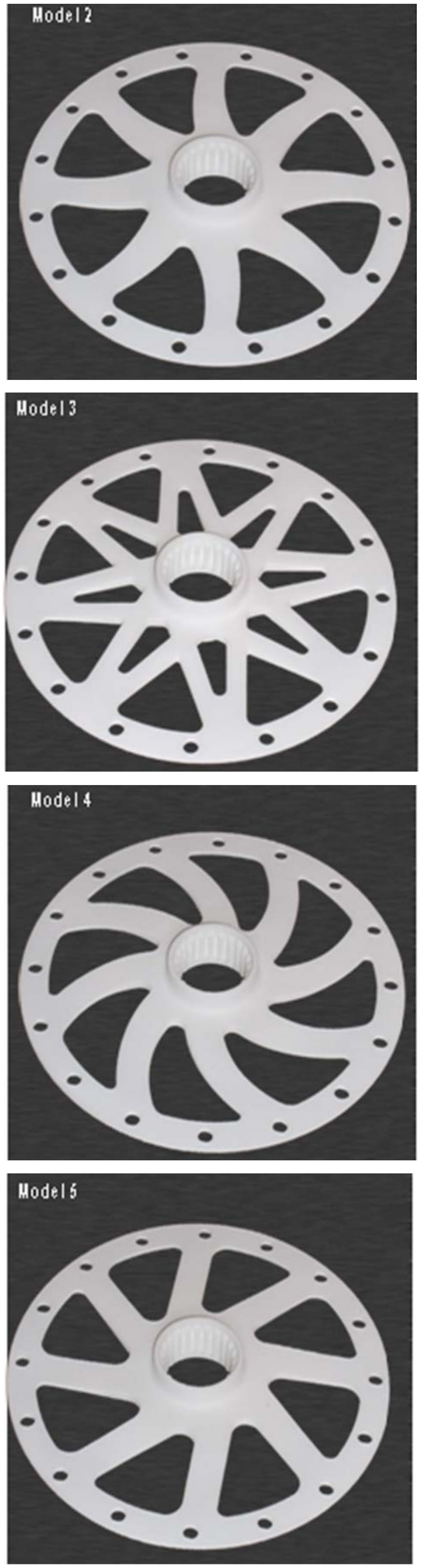


Figure 6. The suggested models of the rigid drive disc manufactured from polyamide material

4. FINITE ELEMENT FORMULATIONS

The equation of motion can be written as [11],

$$[M]\{\ddot{U}\}+[C]\{\dot{U}\}+[K]\{U\}=\{R\} \tag{5}$$

where [M], [C] and [K] are the mass, damping and stiffness matrices, {R} is the external load vector, and {U}, $\{\dot{U}\}$ and $\{\ddot{U}\}$ are the displacement, velocity and acceleration vectors of the finite element assemblage, assuming all of them are time-dependent. Therefore, in dynamic analysis, in principle, static equilibrium at time (t), which includes the effect of acceleration-dependent inertia forces and velocity-dependent damping forces, is considered. Vice versa, in static analysis eq. (5) is considered with inertia and damping effects are neglected. The equation for the static analysis can be written as

$$[K]\{U\}=\{R\} \tag{6}$$

The solution of eq. (6) is obtained by means of Gaussian elimination and back substitution method. If any elastic structures are disturbed in an appropriate manner initially at t=0, the structure can be made to oscillate harmonically. This oscillatory motion is a characteristic property of the structure and depends on the distribution of mass and stiffness in the structure Rao [12],

$$[M]\{\ddot{U}\}+[K]\{U\}=0 \tag{7}$$

To investigate the steady-state and the modal analysis behaviour of drive disc of clutch disc, SolidWorks 2012 and ANSYS/workbench14 software were used. Figure 7 shows the three dimensional FE model of rigid clutch disc and boundary conditions (model 4). The ten-noded element (SOLID187) was used in this analysis and the element has three degrees of freedom at each node. The element is defined by 10 nodes having three degrees of freedom at each node: translations in the nodal x, y, and z directions localized near the contact interfaces at the small values of time, the mesh refinement is needed in this region. A mesh sensitivity study was done to choose the optimum mesh from computational accuracy point of view. In all computations for the rigid clutch disc models, it has been assumed a homogeneous and isotropic material and all parameters and materials' properties are listed in Table. 1.

Table 1 The properties of materials and operations

Parameters	Values
Yield strength, [N/mm^2]	580
Elastic modulus [N/mm^2]	210000
Poisson's ratio	0.28
Density [g/cm^3]	7.85
r_R [mm]	63
Torque [Nm]	500

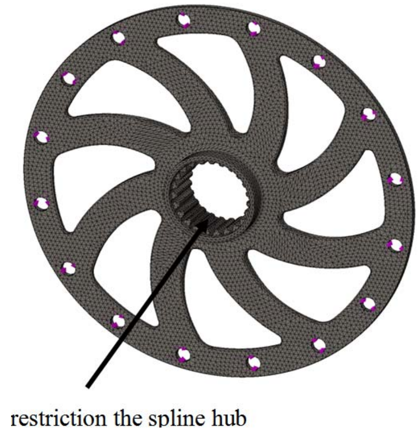


Figure7. FE model with boundary condition for the suggested drive disc (model-4)

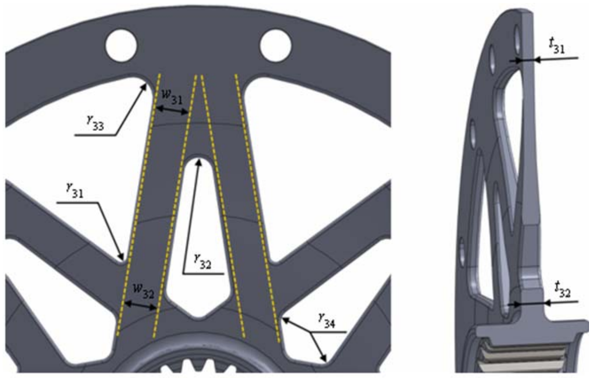
5. RESULTS AND DISCUSSIONS

The first step in this work is to find the stresses and displacements for the suggested models for the rigid clutch disc. SolidWorks 2012 has been used to obtain the numerical simulation for the suggested models in this step.

Table 2 shows the results of the Von-Mises stresses and the total displacements $(U_R=\sqrt{U_x^2+U_y^2+U_z^2})$ for the reference model (1) and the suggested models (2-5) of the rigid clutch disc. It can be seen that the values of the stresses for the models (3,4) are lower than the values for models (2-5). Due to these results in Table 2, the models (3,4) have been selected for the design optimization process to obtain the optimal design parameters of the rigid clutch disc. Figures 8 and 9 show the design parameters for the suggested models (3,4).

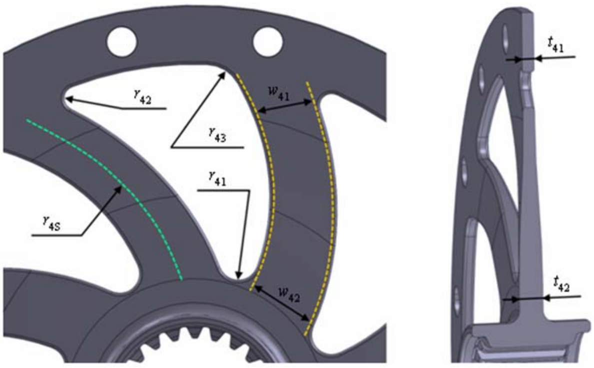
Table 2 The values of the Von-Mises stresses and the total displacement for the rigid clutch disc models (1-5)

Model number	Von-Mises stresses [N/mm^2]	u_R [mm]
1	163.6	0.040
2	419.6	0.175
3	395.3	0.112
4	416.6	0.188
5	421.6	0.187



The initial design parameters for model 3 [mm]							
r_{31}	r_{32}	r_{33}	r_{34}	w_{31}	w_{32}	t_{31}	t_{32}
1	2.3	3.2	2	6.5	7	2	4

Figure 8. The design parameters for the suggested model (3)



The initial design parameters for model 4 [mm]							
r_{41}	r_{42}	r_{43}	r_{4S}	w_{41}	w_{42}	t_{41}	t_{42}
3.6	2	6	43	14.2	13	2	4

Figure 9. The design parameters for the suggested model (4)

A series of computations have been carried out to find the optimal values of the suggested design parameters for models (3,4). Figure 10 shows the variation of the Von-Mises stresses, displacements and mass with the design parameters for the suggested model (3). It can be seen that the parameters r_{32}, t_{31} and t_{32} have significant effects on the results of stresses and displacements, small change of one of these parameters cause considerable effect on values of the stresses and mass of the model whereas the other design parameters have less effect on the results of stresses and displacements. Due to these results, the design parameters can be classified into two types, the main design parameters and the secondary design parameters. The main design parameters for this model are r_{32}, t_{31} and t_{32}, while the secondary design parameters are r_{31}, r_{33}, r_{34}, w_{31} and w_{32}.

The variation of the Von-Mises stresses, displacements and the mass with the design parameters for the model (4) are shown in Fig. 11. It can be observed that the main design parameters for this model are r_{41}, t_{41} and t_{42}, and the secondary design parameters are r_{42}, r_{43}, r_{4s}, w_{41} and w_{42}.

The values of safety factor considered as a criterion to select the final design parameters of the rigid clutch disc, and, in this work, the accepted values of safety factor are equal or greater than 2.8.

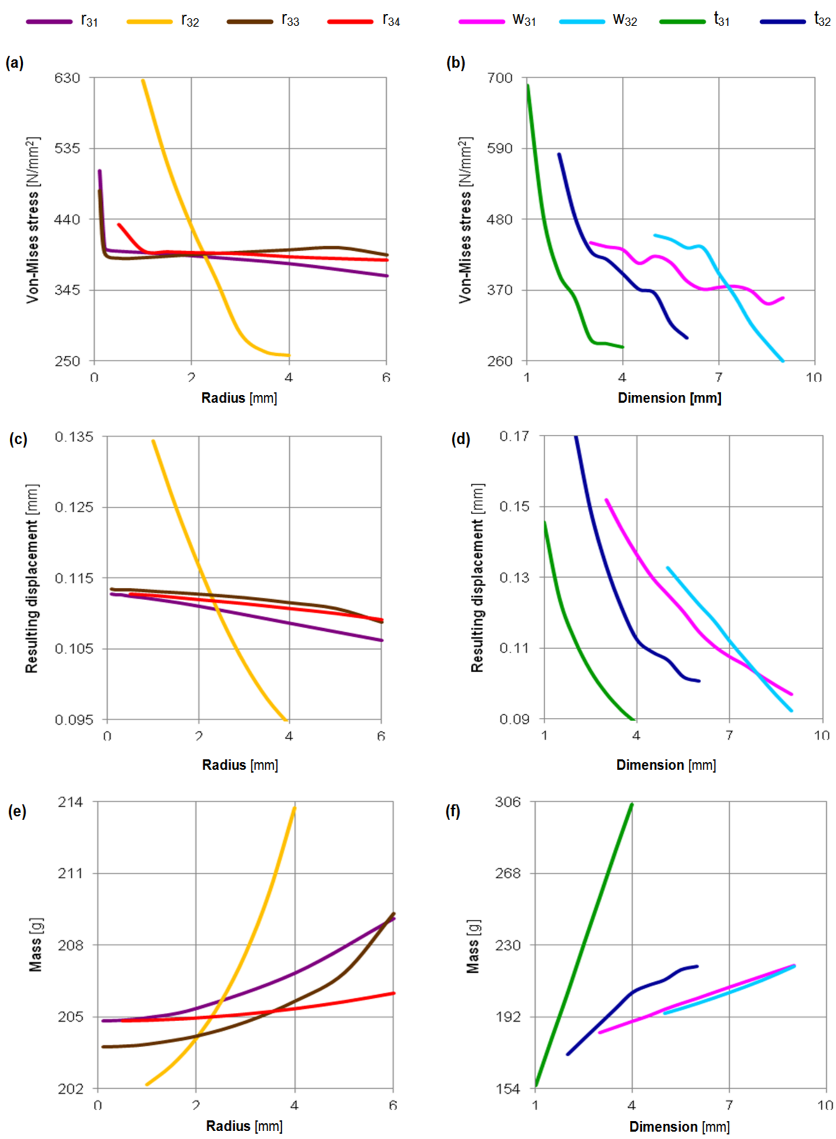


Figure 10. The variation of the Von-Mises stresses, displacements and mass with the design parameters for the suggested model (3)

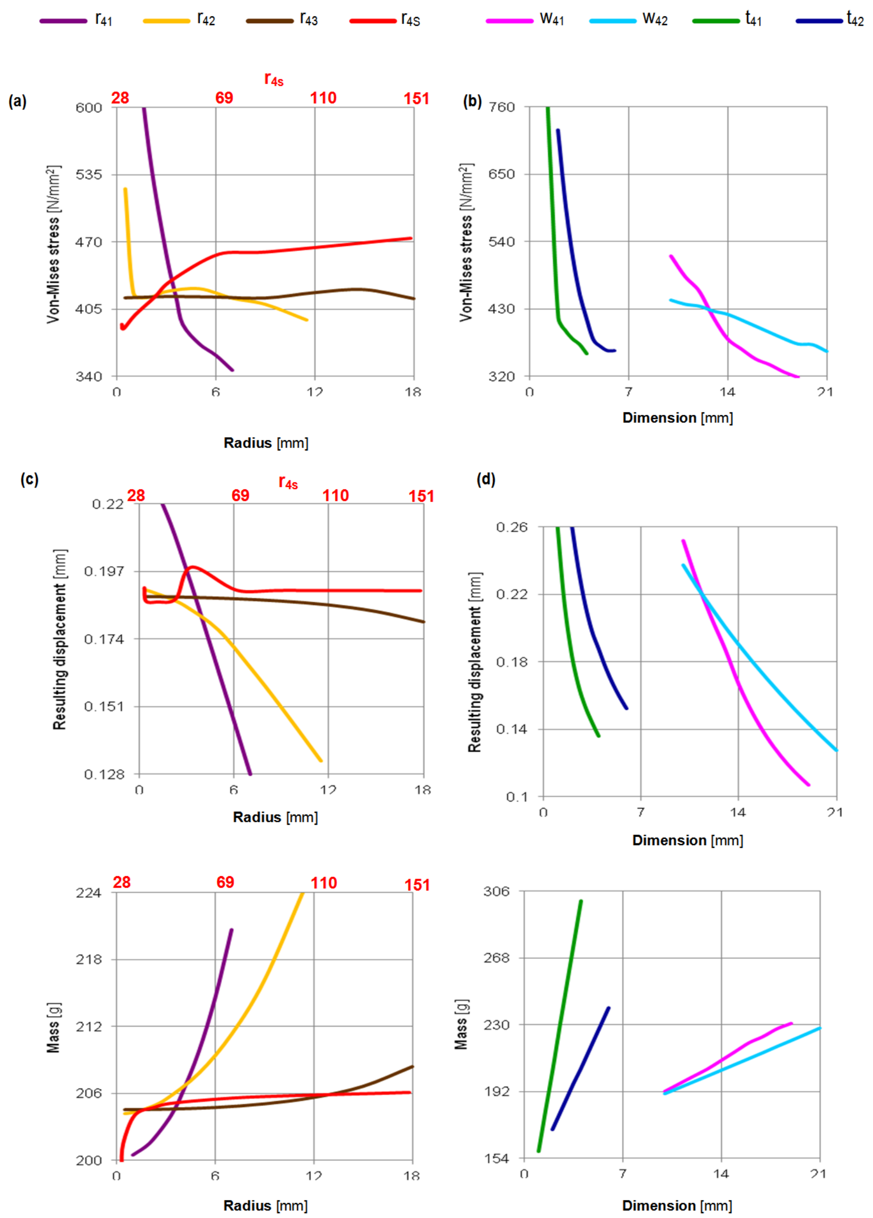


Figure 11. The variation of the Von-mises stresses, displacements and mass with the design parameters for the suggested model (4)

Table 3 lists the final values of the design parameters and masses for the suggested models (3,4) after the optimization process. It can be realized that the value of the mass of the model (3) is less than of the model (4).

Figures (12-15) illustrate the Von-Misses stresses and the displacements for the models (3,4) using the final design parameters (Table 3). It can be noted, that the maximum stresses occur at the fillets near the hub, therefore it is important to increase the fillet radius as much as possible to reduce the values of maximum stresses in this zone.

Table 3 The final values of the design parameters and masses for the suggested models 3 and 4

The design parameters for model 3 [mm]							
r_{31}	r_{32}	r_{33}	r_{34}	w_{31}	w_{32}	t_{31}	t_{32}
1.4	3.7	5	5	6	6.5	2	5
mass = 225.1 g							
The design parameters for model 4 [mm]							
r_{41}	r_{42}	r_{43}	r_{4S}	w_{41}	w_{42}	t_{41}	t_{42}
4.4	3.3	5	40	15.3	16	3	6
mass = 312.9 g							

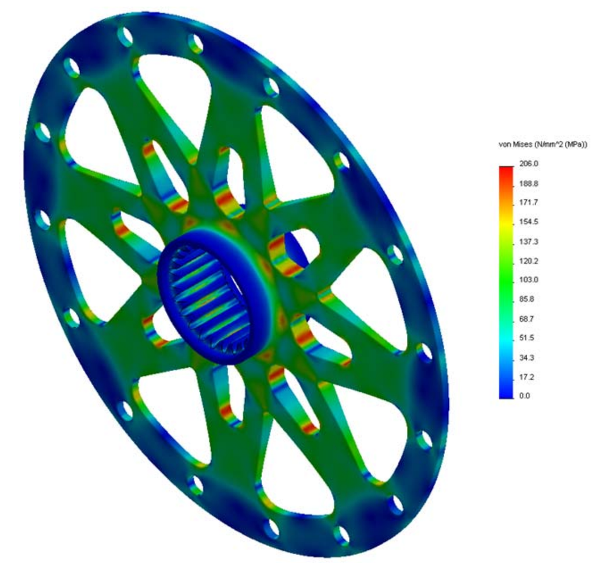


Figure 12 Von-mises stresses for the model (3) [σmax.= 204.1 N/mm2]

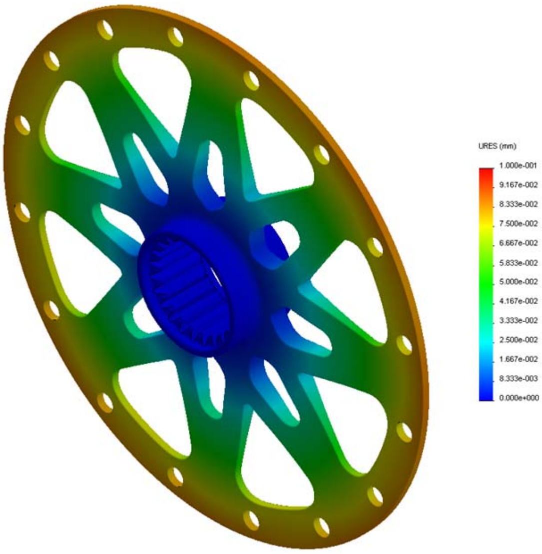


Figure 13 The total displacement for the model (3) [URmax.= 0.0856 mm]

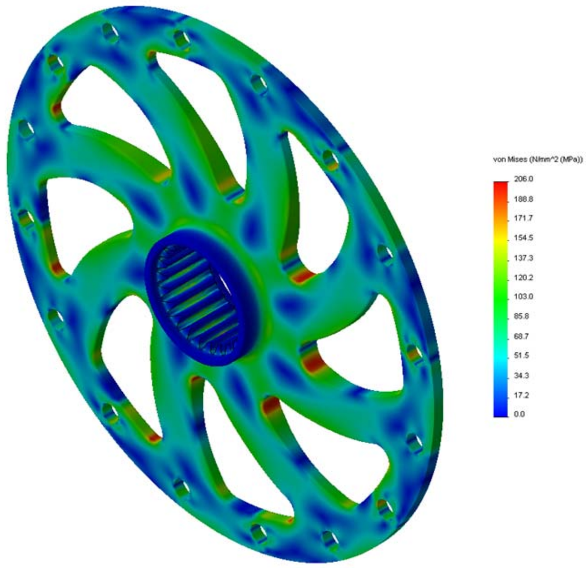


Figure 14 Von-mises stresses for the model (4) [σmax.= 205 N/mm2]

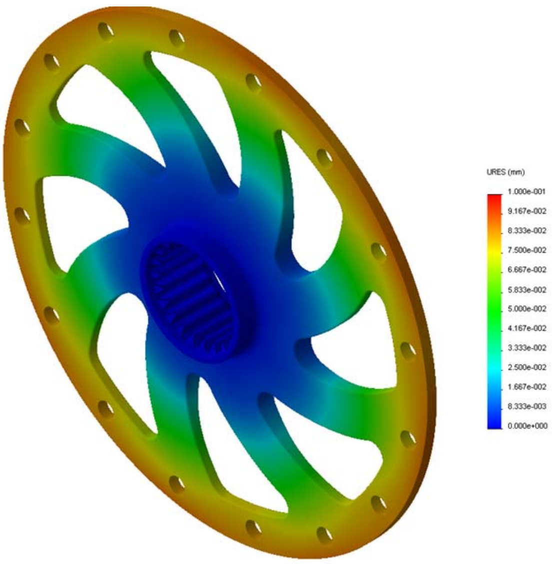


Figure 15 The total displacement for the model (4) [URmax.= 0.0873 mm

Table 4 lists the values of the first fourth natural frequencies for the reference and suggested models (1, 3 & 4). Figures (16-18) demonstrate the fundamental mode shapes for the reference and suggested models (1, 3 & 4). Although the mass of the model 4 is greater than the mass of the model 3, the values of the frequencies of the model 4 are greater than of the model 3. These results can be attributed to difference in the geometric stiffness between these models.

Table 4 The values of the natural frequencies for the reference and suggested models (1,3,4)

Mode No.	Natural frequency [HZ]		
	Model (1)	Model (3)	Model (4)
1	1274.1	1058.5	1325.2
2	1274.5	1086.6	1351.3
3	1280.4	1086.7	1351.5
4	1391.1	1236.7	1545.6

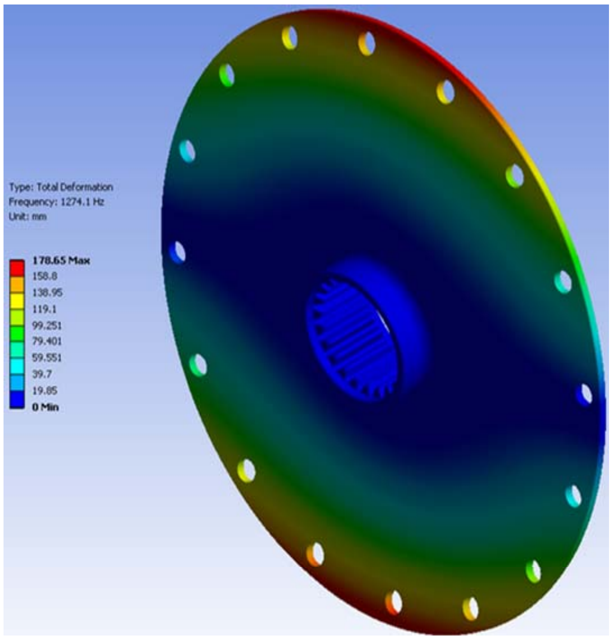


Figure 16 The fundemantal mode shape for the model (1

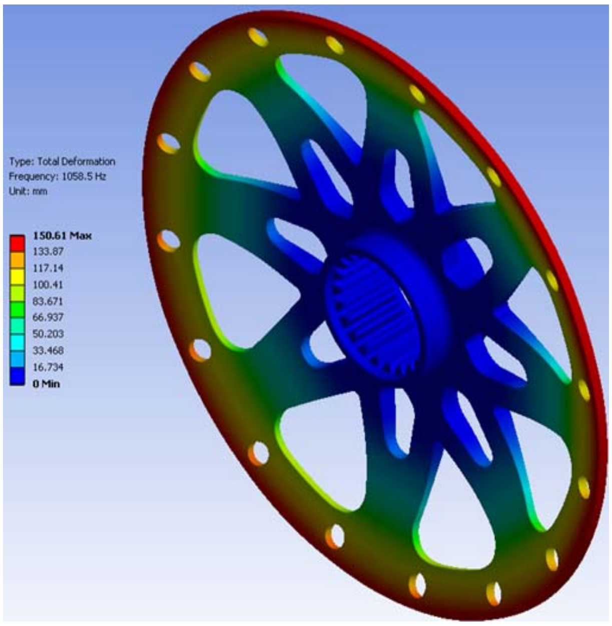


Figure 17 The fundemantal mode shape for the model (3)

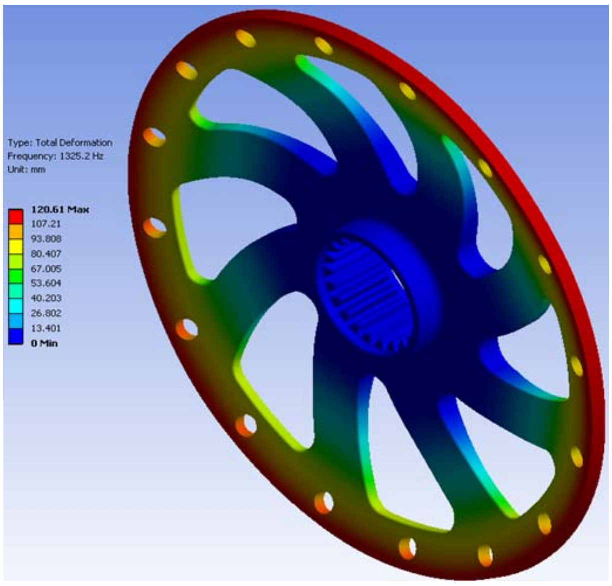


Figure 18 The fundemantal mode shape for the model (4)

6. CONCLUSIONS AND REMARKS

In this paper, the steady-state and modal analyses of rigid clutch disc were performed. Three-dimensional models were built to obtain the optimal design parameters for the suggested models of the rigid clutch disc. Two types of rigid clutch discs have been investigated the reference model and new suggested, models.

The results show that the values of maximum Von-Mises stresses for the suggested models (3,4) are lower than forthe other suggested models (2-5) under the same conditions (note all these models have approximately the same mass ≈ 205g).

From the comparison between the results of the Von- Mises for the suggested models (3,4) and the reference model (1), it can be concluded that the model (3) improves the trend to reduce the maximum stresses with minimum value of mass compared with the other suggested model. When the mass of model (3) changes from (205g to 225.1g), the percentage of reduction in the maximum stress is found to be 93%, whereas if the mass of model (4) changes from (204.8g to 312.9g) the percentage of reduction in the maximum stress is found to be 103%.

The present paper is a preliminary of subsequent investigation to study the suggested models experimentally for the rigid clutch disc.

REFERENCES

[1] VDI guideline 2241, Part 1: Friction Clutches and Brakes, Vocabulary, Types, Characteristic Data, Calculations, Verein Deutscher Ingenieure (VDI), Düsseldorf, 1982.

[2] Dittrich, O. And Schumann, R.: *Anwendungen der Antriebstechnik, Band 2*: Kupplungen, Krausskopf-Verlag, Mainz, 1974.

[3] Sfarni, S. and Malley, M.: Numerical modeling of automotive riveted clutch disc for contact pressure verification, scandinavian simulation society, in: *49th Scandinavian Conference on Simulation and Modelling*, Oslo, 2008.

[4] Naunheimer, H. And Bertche, B.: *Fahrzeuggetriebe– Grundlagen, Auswahl, Auslegung und Konstruktion*, Springer-Verlag, Berlin, 2007.

[5] Kupplungen, L.: *Einführung in die Kupplungstechnik für Personen-Kraftwagen, Veröffentlichte Broschüre, Schaeffler Automotive Aftermarket GmbH*, Bühl (Germany), 2012.

[6] Kupplungen, S.: *Performance-Kupplungssysteme, Lieferprogramm 2008*, Veröffentlichte Broschüre, ZF SACHS Race Engineering GmbH, Schweinfurt (Germany), 2008.

[7] Hazwan, M.: *Design and Analysis of a Lightweight Clutch Disc for a Single Seated Race Car*, Bachelor thesis, Faculty of Mechanical Engineering, Melaka University of Technology, Malaysia, 2011.

[8] Ibnor, M.: *Design and Analysis of a Lightweight Clutch Disc for a Mini Farming Tractor*, Bachelor thesis, Faculty of Mechanical Engineering, Melaka University of Technology, Malaysia, 2011.

[9] Norton, R.L.: *Machine Design: An Integrated Approach*, 4th Edition, Prentice Hall, 2010.

[10] Grote, H. and Feldhusen, J.: *Dubbel – Taschenbuch für den Maschinenbau*, Springer-Verlag, Berlin, 2011.

[11] Batha, K.J and Wilson, E.L.: *Numerical Method in Finite Element Analysis*, Prentice-Hall, Englewood Cliffs, 1976.

[12] Rao, S.S.: *Finite Element Method in Engineering*, Pergamon Press, Oxford, 1982.

ОПТИМИЗАЦИЈА ПАРАМЕТАРА ОБЛИКА И КОНСТРУКЦИЈЕ КРУТОГ ДИСКА КВАЧИЛА ПРИМЕНОМ ФЕМ

Oday I. Abdullah, Josef Schlattmann, Emir Pireci

Фрикционо квачило је основна компонента процеса преноса снаге. Због значаја фрикционог квачила, потребно је истражити карактеристике напона и вибрација крутог диска квачила да би се избегао лом и постигла оптимална тежина и цена. Овај рад приказује нумеричко решење које је добијено прорачунавањем напона и деформације у периоду стационарног стања као и карактеристика вибрација крутог диска квачила. Такодје су предложени нови модели крутог диска квачила. Понашање нових, предложених модела је упоређено са референтним моделом. Резултати прорачунавања показују да управљање карактеристикама напона и вибрацијама крутог диска квачила може да се врши прилагођавањем конструкцијских параметара. Резултати такодје показују да предложени модели значајно побољшавају понашање фрикционог квачила. Прорачунавања су вршена коришћењем софтвера Ansys 14 и Solid Works 2012.