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Application of Grey-Taguchi Method Optimizes the Structural Friction Clutch Disc

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ABSTRACT

In which the clutch disc is the part that is heavily influenced by the pressure disc and engine flywheel. Therefore, the design of the clutch disc faces many difficulties. In this investigation, the model of the clutch disc with design variables and their level was built in SolidWorks. In which, variable A is the friction disc thickness with 3 levels of 3mm, 3.3mm, 3.5mm and B is the diameter of the friction disc groove with levels of 0.25mm, 0.5mm, 0.75mm and variable C is the diameter rivet hole with grades 2.75mm, 3mm, 3.25mm finally D is the young's modulus of the materials for all the part of the clutch are aluminum of 70GPa, copper of 128GPa and structural steel of 200GPa, respectively. The deformation and the equivalent stress were estimated by finite element analysis (FEM) in ANSYS. The simulation data of the study are used to minimize the deformation and stress of the clutch disc by grey relation analysis based on Taguchi method. The results of the FEM indicated that the input variables had a significant influence on the deformation and stress of the clutch disc. Then, the above results were verified by signal to noise (S/N) analysis, analysis of means, analysis of variance, regression analysis, and plot surface. All are in agreement with the error of the predicted value and the optimal value of the grey relational grade is 1.69%. The optimal value of the deformation and stress are 0.033115mm and 66.889MPa, respectively. This proves that these results are very reliable. Therefore, the proposed method is very effective in optimizing the structure of mechanical products.

1. INTRODUCTION

The clutch disc in an automobile has a structure consisting of a hub with a key sprocket that aligns with the primary shaft and a round metal plate with friction material joined by rivets. The keyway is engaged and rotates along the primary axis and helps the disc move along the axis when the clutch is switched off. The friction disc is made of high-temperature resistant asbestos, carbon fiber and copper combined to help the coefficient of friction is high and stable when working. The springs on the friction disc have a damping effect. When the clutch is closed, the presser plate squeezes the friction disc against the rotating flywheel, and the spring damps when the friction disc rotates with the flywheel. Between the two surfaces of the friction disc, there is also a buffer spring in the form of a corrugated or bent trough. This spring weakens when the clutch closes and allows the friction disc to bend inward to help the clutch close smoothly. The clutch disc has the advantage of high durability when working without affecting the frictional properties. Withstands great centrifugal force. Withstands high pressure plate compression loads. However, the clutch disc has the disadvantage that due to operating in high pressure and temperature conditions, the clutch disc is prone to wear and damage, besides the selection of materials to make the clutch disc is difficult. It is a difficult problem to meet the technical requirements while ensuring the factors of operation, durability, and volume. Therefore, to support the selection of clutch disc design, an optimal analysis of the clutch disc structure design on passenger cars is carried out. The finite element ANSYS pointed out that kevlar 49 material has less deformation than gray cast iron, so kevlar 49 has more advantages in making clutches [1]. The deformation and stress of single disc clutch model [2] were obtained by static analysis in ANSYS. The design of single clutch disc by theoretical calculation method using 2D drawings [3]. Suitable materials for performing analysis and comparison are cast iron, alloy steel and copper. The analysis process is used on SolidWorks to analyze the stress, deformation, and displacement of the clutch. As a result, cast iron has a lower stress. So cast iron is most suitable for making clutch discs. The deformation and stress analysis of clutch [4] was determined by ANSYS, with gray cast iron, sintered iron, kevlar and aluminum Metal Matrix Composite materials to find suitable materials. The replacing traditional manufacturing materials are cast iron and aluminum alloy. The analysis results show that Metal Matrix Composite has less deformation than cast iron and has a higher strength-to-material ratio than cast iron, so Metal Matrix Composite is better than other materials. In order to select the best friction material for clutch design [5]. The models were surveyed on asbestos, epoxy glass, porcelain, kevalr, cast iron. The deformation and stress were estimated by ANSYS. The obtained results show that porcelain is the most suitable material. The stress and deformation of the single-disc clutch model of the Tata Sumo be analyzed on ANSYS to find the best material for the clutch [6]. The analysis surveyed on 6 materials of steel, stainless steel, ceramic, Kevlar, aluminum alloy and gray cast iron. The analysis results identified that ceramic has less deformation. So ceramic is more suitable for clutch design. The effect of radius on the clutch disc designed and analyzed by ANSYS to find the maximum stress in the deformation area by theoretical analysis and calculation [7]. The result of the analysis obtained, stress and deformation are slowed down at a radius of 2.5mm. Finally, the 2.5mm radius of the drive plate is suitable for the clutch design and the life of the clutch is increased. The single clutch disc using a purpose steel material [8]. Study on stress and deformation analysis of 3D clutch disc model. The result that the new clutch disc produces less stress than the cast iron clutch disc shown that this design is more optimal. The clutch disc designed using a material (En-Gjs-400-15steel) to replace the traditional material, which is gray cast iron [9]. Through the analysis and comparison of 3D model analysis results on ANSYS, it is concluded that steel materials (En-Gjs-400-15steel) reduce the stress of friction discs compared to gray cast iron materials. The proposed 3D models of the friction clutch disc designed to reduce stress, deformation and vibration in steady state [10]. The analysis results from the software ANSYS identified models 3 and 4 respectively with the weight change from 205g to 225.1g and 204.8g to 312.9g reducing the maximum stress of the clutch disc. The clutch disc model with different radiator groove shapes was analyzed by ANSYS for inorganic and organic materials [11]. The results of improving heat transfer efficiency during the clutch disc design, used for the friction lining. Thereby pointed out that the inorganic-based lining has a temperature of about 30% higher than that of the organic-based lining. The three-dimensional friction clutch model was built [12]. The heat in the slip phase of the clutch was calculated by finite element analysis and numerical methods. The results are for the numerical method is very accurate and the difference with the finite element analysis method is not more than 1%. The temperature of the clutch friction disc determined through four stages of temperature change by a two-dimensional model of the clutch disc, based on the hypothesis of uniform pressure and the theory of heat transfer to finite element analysis and data collection test [13]. The results identified that the proposed model has high accuracy and the error at the highest temperature is not more than 5%. The main factors affecting the thermal characteristics of the friction clutch such as material, pressure, sliding speed was determined [14]. The studied using different materials such as Asbestos, S2 Glass Fiber, Carbon-Carbon Composite, Aluminum Metal Matrix Composite, applying the lining of dry friction clutch disc to increase the durability of the clutch disc [15]. The results of finite element analysis using ANSYS indicated that the temperature causing wear is reduced when the aluminum metal matrix composite material is used for the friction disc lining. The heat generated and transmitted of the clutch friction disc lining during operation to limit wear and increase the durability of the clutch disc. The principle of conservation energy was utilized together with measurement of the dry clutch disc friction liner properties under surface slip-related operating conditions. The results of the analysis identified that the new lining material with a high coefficient of friction will reduce slip, power loss and surface heat generation [16]. Some new Hino MB clutch discs has designed [17]. Then the finite element method was applied to determine the stress of the above models. The results indicated that the best clutch disc has design model with 35mm groove length of 35mm and stress of 12.1MPa. The friction disc with inner diameter of 30 mm, outer diameter of 38mm, thickness of 3.5mm tested [18]. G95 woven frictional material, studied using XTM 500 test rig with conditions ambient temperature, rotational speed and applied load, the results show that the coefficient of friction is influenced by the ambient temperature, and the applied load affects the friction force and the coefficient of friction. The thermal characteristics of the clutch disc in order to limit the damage of the clutch disc during operation [19] used SOLID90 geometry and ANSYS to analyze the FEA of the clutch disc in 4 tests, which shows that it is important to increase the contact area of the clutch disc.

Most previous studies have shown the influence of design variables and materials on deformation, stress as well as the working ability of friction clutch discs using the finite element method. However, no work has been published on minimizing these effects. It is important to achieve a suitable friction disc model with reasonable dimensions and suitable materials to ensure the working ability of the friction clutch disc with high durability, low deformation and stress.

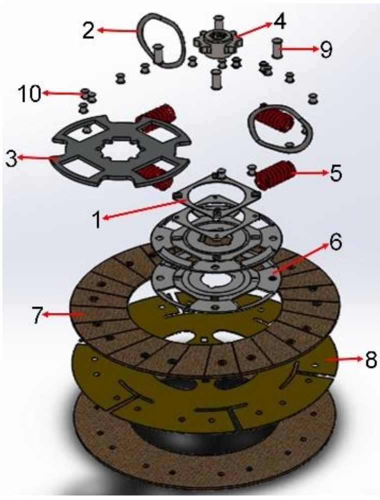
Recent research has aimed at selecting suitable materials for friction clutch discs based on the ANSYS numerical method without considering the influence of the structure as well as the dimension of each part in the model.

A novel in this study is the application of grey relation analysis based on Taguchi's method to minimized the influence of the design variables on the deformation and stress of the clutch structure in the car. The Taguchi method, analysis of variance, interaction analysis, regression analysis, means analysis, surface analysis and grey relational analysis were used to verified the results of FEM. The rest of this work such as model design and finite element methods in ANSYS are presented in Section 2, grey relation analysis is written in Section 3, and the results and discussions are analyzed in Section 4 and conclusions are presented in Section 5, and finally references.

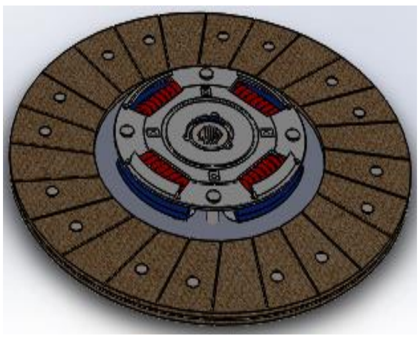
2. METHODOLOGY FOR DESIGN THE CLUTCH MODEL

2.1 Design modeling of the clutch

The clutch disc model used in this study was implemented in SolidWorks and is shown in Figure 1. In which, part 1^{rst} is stop ring, part 2nd is shock absorber cage, part 3rd is center disc, part 4th is hub, part 5th is anti-twist spring, part 6th is spacer, part 7th is friction plate, part 8th is elastic disc, part 9th is rivet, part 10th is pin. In this study, the input variables affecting the stress and deformation of the clutch disc will be studied, thereby optimizing the clutch disc design. The clutch friction liners are subjected to intense rubbing, thus generating heat in a relatively short time. The backing material therefore requires a combination of the following properties to withstand operating conditions such as a relatively high coefficient of friction over the entire operating condition. The maintenance of frictional properties over a lifetime working time, relatively high ability to absorb energy in short time. The withstand high pressure plate compression load. The withstand large impact of centrifugal force during gear shifting, sufficient shear strength for tissue transmission engine torque, high degree of endurance in cyclic operation without affecting friction, high tolerance against surface contamination without affecting its friction and properties its grip. When designing a friction clutch where the appropriate material forming the contact surface should be selected. The moving parts of the clutch should be of low weight to minimize inertial loads, especially at high speeds. The clutch shall not require any external force to maintain contact of the friction surfaces. The provision must be made to take account of the wear of the contact surfaces. The clutch should be conditioned to facilitate repair. The clutch must be able to carry away the heat generated at the contact surfaces, and the protruding parts of the clutch must be protected.



(a) Disassemble



(b) Assembly

Figure 1. The clutch model

2.2 Finite element method

The finite element model is set up for clutch disc in ANSYS as shown in Figure 2. It uses the automatic meshing method and set the constraints.

The automatic meshing tool in ANSYS is selected to mesh the model with mesh size of 0.5mm. The mesh result with 619047 triangular elements and 1103647 nodes was shown in Figure 2. Set the boundary condition for the model as fixed support placed on surface B and set load to the model to perform simulation as the pressure of 0.134 MPa applied to (A) as shown in Figure 3. The materials were selected for all the part of the clutch are aluminum of 70GPa, copper of 128GPa and structural steel of 200GPa, respectively. The model is first made up of an elastic disc (8) with a shock absorber cage (2), followed by a connection between the shock absorber cage (2) and a central disc (3). Center disc linkage to shock absorber cage (2), shock absorber cage (2) to stop ring (1), shock absorber cage linked to spacer (6), gasket linkage (6) and center disc (3), spacer (6) connects to elastic disc (8), anti-twist spring (5) connects to center disc (3), pin (10) connects to pad (6). The friction plate (7) is connected to the elastic disc (8), the rivet (9) is connected to the elastic disc (8) and the friction plate (1), the rivet (9) is connected to the friction plate (7). The simulation analysis includes the deformation and equivalent stress in the friction plate.

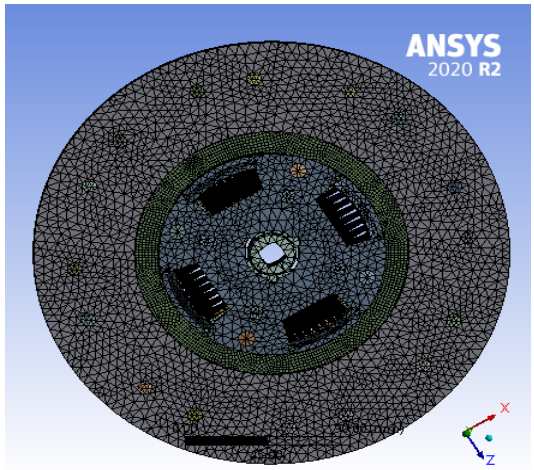


Figure 2. Divide mesh result for the model

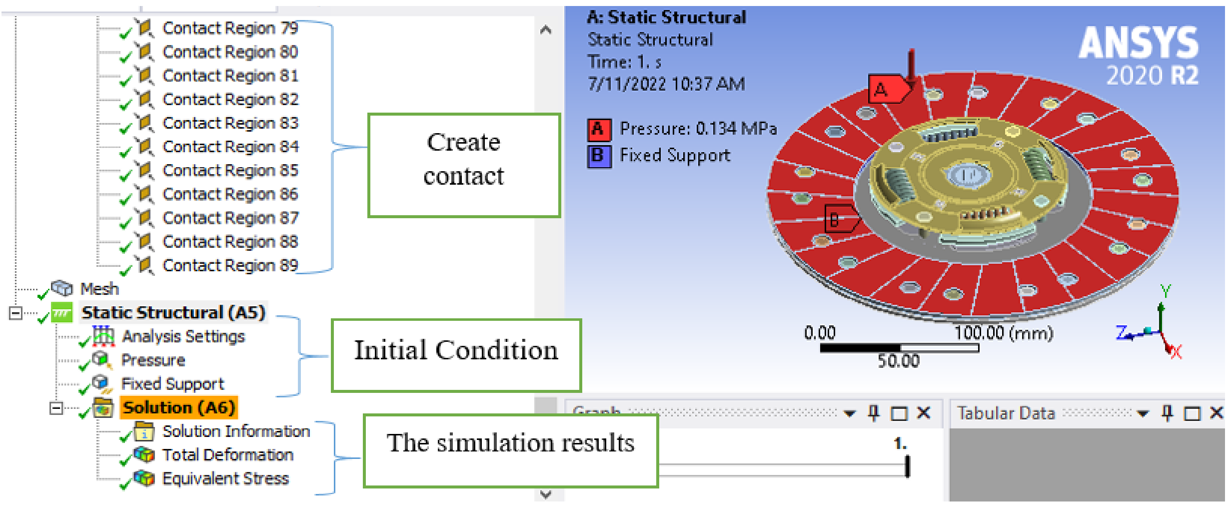


Figure 3. The finite element model

3. GREY RELATIONAL ANALYSIS

In order to select the optimal model among many models to ensure many goals. While the Taguchi Method only optimizes the single target. However, the Taguchi Method allows performing few experiments while still being able to select the optimal model. Therefore, in this study, Taguchi applied this advantage to design 27 cases and then apply the gray relationship analysis method [20-30] to select the optimal model according to two objective stress and deformation.

Firstly, dimension design and dimension levels were selected for the fiction clutch disc. Next step, Minitab software was utilized to design 27 cases with different dimension. SolidWorks was used to design 76 models. Finite element method in ANSYS was used to analyze deformation and stress of 27 models. Finally, in or der to determine an optimal model, the grey relational analysis (GRA) was selected. This method was performed as following:

The objective the larger is the better:

$$D_i^* = \frac{D_i^{(0)}(k) - \min D_i^0(k)}{\max D_i^{(0)}(k) - \min D_i^{(0)}(k)} \tag{1}$$

The objective the lower is the better:

$$D_i^* = \frac{\max D_i^{(0)}(k) - D_i^0(k)}{\max D_i^{(0)}(k) - \min D_i^{(0)}(k)} \tag{2}$$

The deviation is determined as follows:

$$\Delta_{0i} = \| D_0^*(k) - D_i^*(k) \| \tag{3}$$

$$\Delta_{\min} = \max_{\forall j \in i} \min_{\forall k} \left\| D_0^*(k) - D_j^{**}(k) \right\| \tag{4}$$

$$\Delta_{\max} = \max_{\forall j \in i} \max_{\forall k} \left\| D_0^*(k) - D_j^*(k) \right\| \tag{5}$$

The optimal model is a model with stress and deformation are "the lower is the better".

Compute the grey relational coefficient (GRC) (γ):

$$\gamma_i(k) = \frac{\Delta_{\min} + \xi \Delta_{\max}}{\Delta_{0i} + \xi \Delta_{\max}}, \xi = 0.5 \tag{6}$$

Entropy measurement technique was used to determine the weights for the two variables displacement and stress as presented in Eq. (7):

$$\omega_e(x) = x \cdot e^{(1-x)} + (1-x)e^x - 1 \tag{7}$$

$$w \equiv \frac{1}{\left(e^{0.5} - 1\right)} \sum_{i=1}^{m} \omega_e(x) \tag{8}$$

$\epsilon = \{\gamma_i(1), \gamma_i(2), \ldots, \gamma_i(n)\}$. Note that $i = 1,2, \ldots, n$.

Total of grey relational coefficient D_j obtained by Eq. (9):

$$D_j \equiv \sum_{i=1}^{m} \gamma_i(j), j = 1, 2, \ldots, n \tag{9}$$

The coefficient (k) was computed by Eq. (10):

$$k = \frac{1}{\left(e^{0.5} - 1\right) \times m} = \frac{1}{0.6487 \times m} \tag{10}$$

Eq. (11) was used to calculate Entropy e_j:

$$e_j = k \sum_{i=1}^{m} \omega_e \left(\frac{\gamma_i(j)}{D_j} \right), j = 1, 2, \ldots, n \tag{11}$$

Here, $\underset{e}{\omega(x)}$ uses Eq. (11). Entropy total was computed by Eq. (12):

$$E = \sum_{j=1}^{n} e_j \tag{12}$$

The weight was computed by Eq. (13):

$$\omega_j = \frac{1}{n-E} \cdot \frac{\left[1 - e_j\right]}{\sum_{j=1}^{n} \frac{1}{n-E} \cdot \left[1 - e_j\right]} \text{ here }, j = 1, 2, \ldots, n. \tag{13}$$

GRG (ψ_i) was computed by Eq. (14):

$$\psi_i = \sum_{k=1}^{n} \omega_k \gamma_i(k), n = 27 \tag{14}$$

4. RESULTS AND DISCUSSIONS

4.1 Set up experiment

To optimize clutch disc structure. Input boundaries are used with their levels for analysis. The input variables are divided into A, B, C, D. In which A is the friction plate thickness (the thickness of part 7th) with levels of 3mm, 3.3mm, and 3.5mm. B is the friction plate groove dimension (groove hole dimension on the part 7th) with levels of 0.25mm, 0.5mm and 0.75mm. C is the diameter rivet hole diameter on friction plate of 2.75mm, 3mm and 3.25mm, respectively and finally the most important input variable D is the young's modulus of the materials for all the part of the clutch are aluminum of 70GPa, copper of 128GPa and structural steel of 200GPa, respectively as listed in Table 1.

Table 1. The design variables and their level

Factor	Symbol	Unit	Level		
			1	2	3
Friction plate thickness	A	mm	3.0	3.3	3.5
The friction plate groove dimension	B	mm	0.25	0.50	0.75
The diameter rivet hole diameter on friction plate	C	mm	2.75	3.00	3.25
Elastic modulus of friction plate lining material	D	GPa	70	135	200

These design variables are randomly selected with the objective of the stress and deformation are the lower is the best. In order to this objective. The Taguchi method is single objective optimization. This problem is not suitable for multiple objectives optimization. However, the advantage of this method is that it designs few experiments to achieve optimal and reliable results. In order to take advantage of these advantages, the Taguchi method applied to design 27 models different. This work has performed 27 completely different simulation cases to serve the optimization and the results of simulation of stress and deformation of the clutch disc are performed as shown in Table 2. Next to the data in Table 1 were used to optimize multiple objectives with application grey relational analysis. These optimal results were verified by Taguchi method (signals to noise analysis, mean analysis, interaction analysis) and analysis of variances for grey relational grade. This will increase the efficiency of the grey relational analysis method making it more reliable in multi-objective optimization. The results of FEM proved that the design variables significantly affected on the deformation and stress. Because within 27 different cases, the values of the deformation and stress obtained also different.

Table 2. Orthogonal array L27 and analysis outcomes

Trial No.	A (mm)	B (mm)	C (mm)	D (MPa)	Deformation (mm)	Stress (MPa)
1	3.0	0.25	2.75	70	0.15803	131.530
2	3.0	0.25	3.00	128	0.06982	127.190
3	3.0	0.25	3.25	200	0.04416	82.311
4	3.0	0.50	2.75	128	0.06849	121.950
5	3.0	0.50	3.00	200	0.04578	90.199
6	3.0	0.50	3.25	70	0.16074	102.110
7	3.0	0.75	2.75	200	0.04225	93.146
8	3.0	0.75	3.00	70	0.16640	121.460
9	3.0	0.75	3.25	128	0.06900	82.897
10	3.3	0.25	2.75	70	0.13382	114.740
11	3.3	0.25	3.00	128	0.05821	83.682
12	3.3	0.25	3.25	200	0.03765	71.674
13	3.3	0.50	2.75	128	0.05625	95.690
14	3.3	0.50	3.00	200	0.03879	92.527
15	3.3	0.50	3.25	70	0.1397	109.300
16	3.3	0.75	2.75	200	0.03498	77.921
17	3.3	0.75	3.00	70	0.14413	114.810
18	3.3	0.75	3.25	128	0.05865	111.280
19	3.5	0.25	2.75	70	0.10523	90.587
20	3.5	0.25	3.00	128	0.05113	82.913
21	3.5	0.25	3.25	200	0.03440	68.227
22	3.5	0.50	2.75	128	0.04520	83.163
23	3.5	0.50	3.00	200	0.03634	78.071
24	3.5	0.50	3.25	70	0.12012	98.934
25	3.5	0.75	2.75	200	0.03306	80.645
26	3.5	0.75	3.00	70	0.11976	101.98
27	3.5	0.75	3.25	128	0.05045	72.694

Table 3. Objective function, deviation, GRC, GRG, rank of GRG

Trial No.	Di (1)	Di (2)	Δoi (1)	Δoi (2)	γi (1)	γi (2)	(ψi)	rank
1	0.0650	0.0000	0.935	1.0000	0.3484	0.3333	0.3409	27
2	0.7530	0.1409	0.247	0.8591	0.6693	0.3679	0.5191	21
3	0.9010	0.7631	0.099	0.2369	0.8347	0.6785	0.7568	10
4	0.7466	0.6731	0.2534	0.3269	0.6637	0.6047	0.6343	15
5	0.9210	0.7383	0.079	0.2617	0.8636	0.6564	0.7603	9
6	0.0869	0.5669	0.9131	0.4331	0.3538	0.5358	0.4445	24
7	0.9096	0.4501	0.0904	0.5499	0.8469	0.4762	0.6621	14
8	0.0000	0.3480	1.0000	0.652	0.3333	0.434	0.3835	25
9	0.7238	0.6504	0.2762	0.3496	0.6442	0.5885	0.6164	16
10	0.2620	0.0414	0.738	0.9586	0.4039	0.3428	0.3734	26
11	0.8691	0.8634	0.1309	0.1366	0.7925	0.7854	0.789	7
12	0.9831	0.9582	0.0169	0.0418	0.9673	0.9228	0.9451	3
13	0.8415	0.2608	0.1585	0.7392	0.7593	0.4035	0.5819	17
14	0.9667	0.8765	0.0333	0.1235	0.9376	0.8019	0.8700	4
15	0.2509	0.6532	0.7491	0.3468	0.4003	0.5905	0.4951	22
16	0.9725	0.4743	0.0275	0.5257	0.9479	0.4875	0.7184	13
17	0.2675	0.6130	0.7325	0.387	0.4057	0.5637	0.4845	23
18	0.8146	0.8254	0.1854	0.1746	0.7295	0.7412	0.7353	12
19	0.4249	0.7235	0.5751	0.2765	0.4651	0.6439	0.5542	19
20	0.8069	0.8382	0.1931	0.1618	0.7214	0.7555	0.7384	11
21	1.0000	1.0000	0.0000	0.0000	1.0000	1.0000	1.0000	1
22	0.8698	0.9237	0.1302	0.0763	0.7934	0.8676	0.8304	6
23	0.9987	0.9991	0.0013	0.0009	0.9974	0.9982	0.9978	2
24	0.4067	0.7456	0.5933	0.2544	0.4573	0.6628	0.5597	18
25	0.9963	0.6371	0.0037	0.3629	0.9927	0.5794	0.7867	8
26	0.3445	0.7317	0.6555	0.2683	0.4327	0.6508	0.5414	20
27	0.8789	0.9385	0.1211	0.0615	0.805	0.8905	0.8476	5

4.2 Grey relational analysis

Di (1) and Di (2) are optimal criterion of the deformation and stress of the clutch disc. For stable clutch operation, the stress of the clutch disc must be the smaller the better. Therefore, Eq. (2) is applied to minimize the deformation and stress. The result of Di (1) and Di (2) are shown in Table 3. Δoi (1), Δoi (2) are the deviation values of the deformation and equivalent stress of clutch disc determined by Eq. (3). γi (1) and γi (2) are GRC values of the clutch disc stress, respectively obtained by Eq. (6), GRG (ψi) is calculated by Eq. (14). In Table 3, the minimum value of Δi is 0 and the maximum of Δi is 1. In this Table, the maximum and minimum values of GRG are the 21st and 1st experiments, respectively.

The analytical outcomes of 27 cases were listed in Table 2 we once again confirmed that the input variables greatly affect the stress and deformation from which we can find the most optimal case in the 27 experiments which is the experiment 21 with elastic modulus of friction plate lining of 200GPa, friction plate thickness 3.5mm rivet hole diameter 3.25mm friction plate groove is 0.25mm while the least optimal case is experiment 1 with elastic modulus of 70GPa friction plate thickness 3 rivet hole diameter is 2.75mm friction plate groove is 0.25mm. We infer that the most optimal design is achieved when the variables A, C, D reach the maximum and B reaches 0.25mm with the stress of 68,227MPa and the deformation of 0.034067mm. This result is similar to the results of the partial analysis. The finite element is presented in 2 with 0.525.

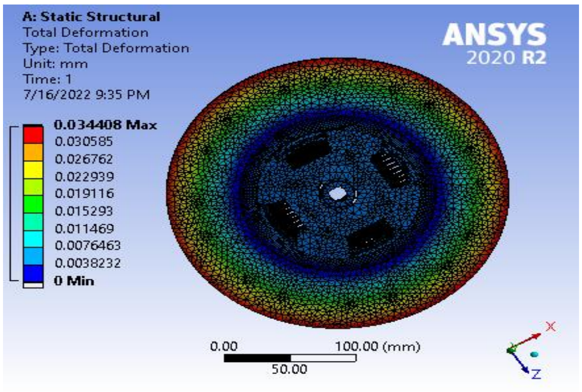
4.3 Analytical signal to noise (S/N)

Based on Table 4, we have the values in columns A, B, C and D that represent the average value of each variable at each level of the S/N analysis method from which it is possible to deduce the influence of the variables. The input variable is based on the variable's delta index. We see that the influence of elastic modulus is greatest when the delta index of this axis is the largest, which means that the S/N index of column D has the largest difference of 5.123 followed by column A with a thickness of 2.629 after that. is C with 1.381 finally B with 0.525.

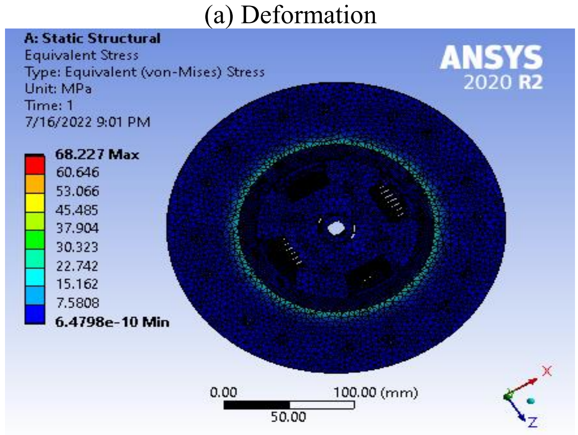
Table 4. Mean values of signal to noise

Level	A	B	C	D
1	-5.217	-4.037	-4.649	-6.797
2	-3.885	-3.563	-3.772	-3.219
3	-2.587	-4.088	-3.268	-1.673
Delta	2.629	0.525	1.381	5.123
Rank	2	4	3	1

The results of the optimal case as presented in Figure 4, which indicates that the optimal values of the deformation and stress of the clutch disc are 0.034 mm and 68.2727 MPa. The S/N chart is plotted against the table and the Delta index is calculated by subtracting the minimum from the maximum value in the same column. When analyzing the S/N chart in Figure 5. The larger the slope of the graph, the greater the design variables affecting the stress and strain of the clutch disc. Again, the input variables are the friction plate thickness (A) the friction plate groove hole (B) the friction plate rivet hole (C) and the elastic modulus of the friction plate lining material (D). has a strong influence on the main stress of the clutch disc, where the elastic modulus of the friction plate lining (D) is the variable that has the most influence on the stress of the clutch disc and (B) has the least effect on the clutch disc stress. stress. We see that the analysis results are similar to the finite element analysis when the analysis also gives the conclusion that the initial boundaries and both affect the stress and deformation of the disc. After analyzing Table 4 and the chart in Figure 5, we can answer the question of the influence of lining material on the deformation of the clutch. As shown in the diagram, we see that the elastic modulus of the lining material has the strongest influence on the main stress and deformation of the clutch. So the optimal level of the combined variable here is A3B2C3D3. This result is similar to the analytical results in the finite element method when the elastic modulus of all parts of the clutch is 200GPa and the thickness is 3.5mm, the friction plate groove is 0.5 and the rivet hole diameter is 3.25mm. then the most optimal model. This results not only clarifies the optimal value, but also clarifies that the variables have strongly affected on the deformation and stress of the friction clutch disc.



(a) Deformation



(b) Equivalent stress

Figure 4. Analysis results of the 21st case

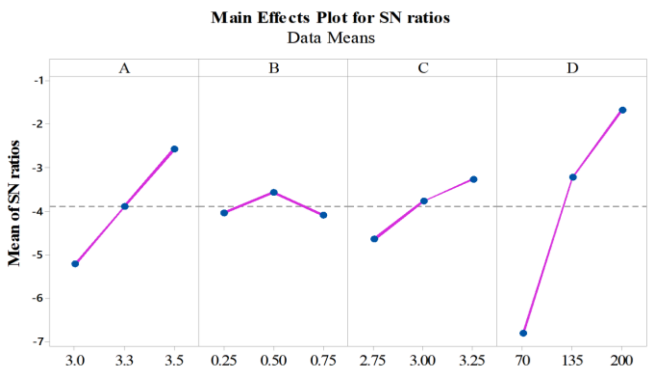


Figure 5. Plot of S/N of GRG

4.4 Analytical means

After analytical means, all the outcomes were presented in Table 5, we once again confirmed the stated point of finite element analysis as input variables A, B, C, D that greatly affect the stress and deformation of the clutch disc at the same time. Reconfirming the deformation and stress of the clutch disc were affected by the variables, namely the variable D that has strongly influenced on the stress and deformation of the friction clutch disc, then to A and C at the end of the arc. B. Specifically, the table shows the level of influence based on the Delta index of each column such as column D being the highest with 0.3689 indicating that factor D is the factor that has the strongest influence on stress. Meanwhile, with column B, the delta index is the lowest when reaching 0.0442, thereby proving that B has little influence on the stress of the clutch disc. The graph of the average value of GRG by the level of each variable is shown in Figure 6. In which, the x-axis represents the level of each variable and the y-axis is the average value of the GRG of each variable by each level It continues to confirm determined that the three variables A, B, C and D have a stronger influence on the stress and strain of the clutch. In measurement D is the most influential factor because the higher the slope of the graph, the greater the influence of that boundary on the stress. Accordingly, the graph D has the highest slope, so the variable (D) elastic modulus of the friction plate lining material is the variable that has the greatest influence, then on A thickness, then C rivet hole diameter and finally is B the friction plate groove diameter. Conclusions after analyzing Table 5 and the chart in Figure 6, we once again confirm that the input variable elastic modulus of the friction plate lining material D has the strongest influence on the main stress and deformation of the clutch disc. then A and C. Finally, B has the least influence. From that, it is inferred that the choice of lining material is important in the clutch design and the most optimal model is A3B2C3D3. This result is similar to the analytical results in the finite element method when the elastic modulus of the lining material is 200GPa and the thickness is 3.5mm, the friction plate groove is 0.5 and the rivet hole diameter is 3.25mm. then the most optimal model.

Table 5. The average values of GRG by each level of each variable

Level	A	B	C	D
1	0.5687	0.6685	0.6091	0.4641
2	0.6659	0.6860	0.6760	0.6992
3	0.7618	0.6418	0.7112	0.8330
Delta	0.1931	0.0442	0.1020	0.3689
Rank	2	4	3	1

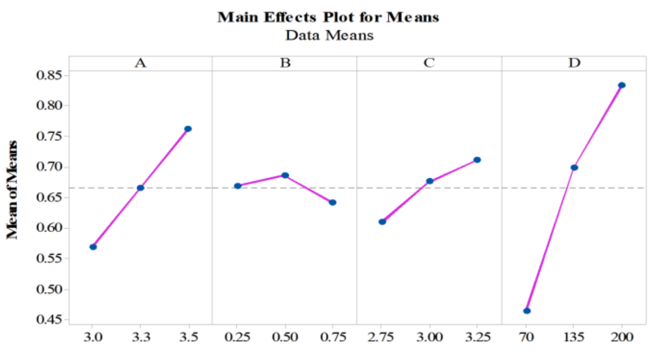


Figure 6. Plot of means of GRG

Figure 7. The plot of interaction of S/N

4.5 Analytical interaction

The interaction analysis result was presented in Figure 7 and Figure 8. If the relationship is represented in the graph by non-parallel lines, then the interaction is said to exist and the parallel line relationship indicates that the input parameters have no interaction relationship with each other. From figures pointed out the input parameters have no interaction relationship with each other. This problem confirmed that input variables have strongly influenced on deformation and stress. These results are consistent with the results of finite element analysis, gray relational analysis and Taguchi analysis results.

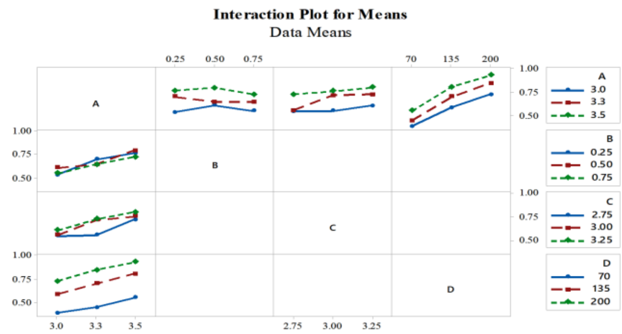


Figure 8. The interaction plot of mean

4.6 Analysis of variance

The results as presented in Table 6 were obtained by Minitab software, the influenced percentage contribution of the regression equation is 98.24%, the influenced percentage contribution of A is 18.13%, B is 4.35%, C is 7.12%, D is 66.96%, D*D is 1.68%, error is 1.76%. The mean square of the regression equation is 0.168715, A is 0.165806, B is 0.003227, C is 0.046838, D is 0.612356, D*D is 0.015349 and the error is 0.00338. The P-value reveals that the input parameters are insignificant, since the P-value is less than 0.05. And the F value is greater than 2 The results of ANOVA analysis are in good agreement with the results of signal-to-noise analysis**.** This result confirms that design variables have strongly influenced deformation and stress. And also confirmed good agreement with the FEM results, GRA method, Taguchi Method and interaction analysis.

The conclusion after analyzing the variance obtained by Minitab software. From results in Table 6, again confirmed that the input variables have strongly affected on the deformation and stress of the clutch disc, in which the input variable D elastic modulus of the friction plate lining material is having the most influence, specifically, the influence of D is 66.69% of A is 18.13% and the influence of B is the least with 4.35% and C is 7.12%. The experimental statistic value F is 49.91 greater than 2 and finally the regression value is 0.000 less than 0.05, which proves that the variables have strongly influenced on stress and deformation of the friction clutch disc. The results are better than the previous published results [8, 9].

The values of R-square obtained over 94% as presented in Table 7. Thereby, the values indicated that the designed variables have impacted strongly on the deformation and stress of the clutch disc and cannot ignore while design the clutch disc.

Table 6. ANOVA results

Source	DF	Seq SS	Contribution	Adj SS	Seq MS	F-Value	P-Value
Regression	5	0.843576	98.24%	0.84357	0.168715	49.91	0.000
A	1	0.165806	18.13%	0.16580	0.165806	49.05	0.000
B	1	0.003227	4.35%	0.00322	0.003227	2.95	0.034
C	1	0.046838	7.12%	0.04683	0.046838	13.86	0.001
D	1	0.612356	66.96%	0.05310	0.612356	181.16	0.000
D*D	1	0.015349	1.68%	0.01534	0.015349	4.54	0.045
Error	21	0.070986	1.76%	0.07098	0.003380		
Total	26	0.914561	100.00%				

Table 7. Summary of feedback from analysis

S	R-Sq	R-Sq (adj)	PRESS	R-Sq (pred)
0.0581400	98.24%	96.39%	0.115454	94.38%

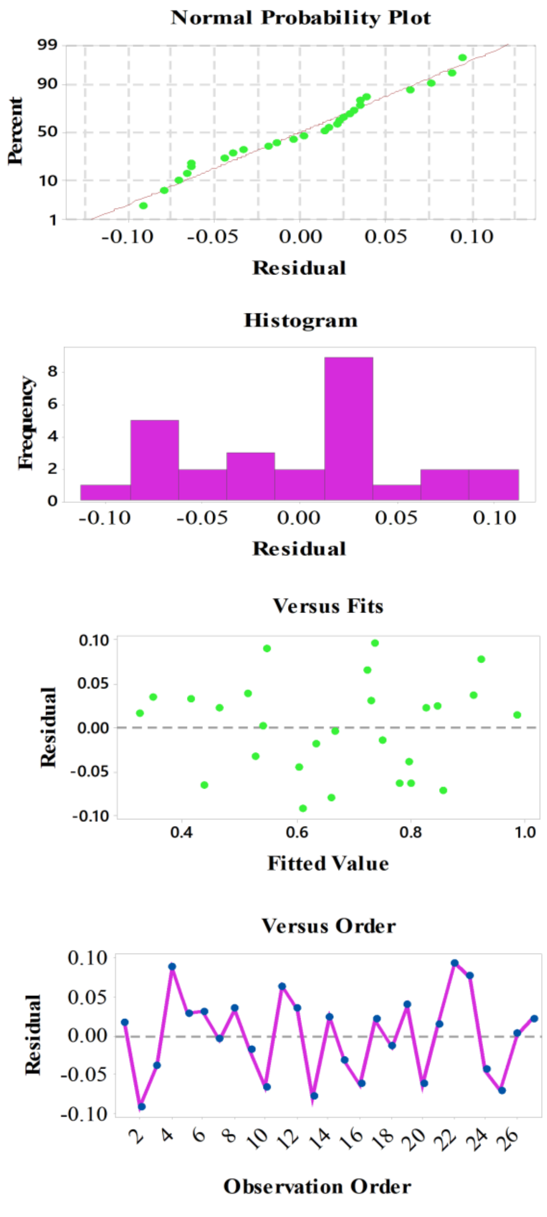
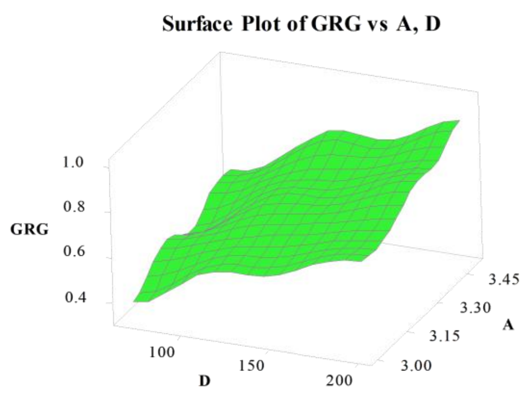


Figure 9. The residual plot of GRG

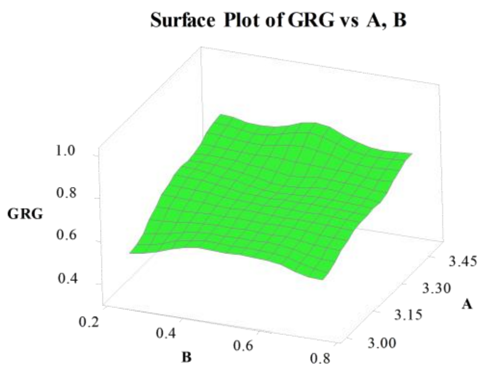
4.7 Regression analysis

$$GRG = -1.733 + 0.3814\ A - 0.0536\ B + 0.2040\ C + 0.00607\ D - 0.000012\ D^*D \tag{15}$$

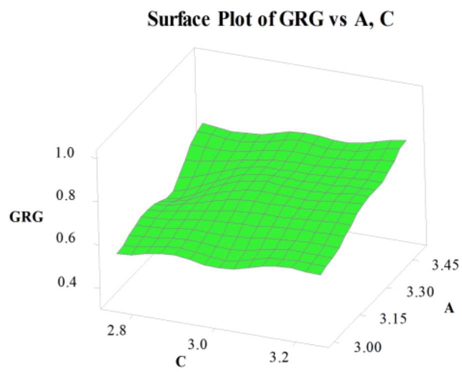
The outcome of the regression analysis as presented in Eq. (15) obtained by Minitab software, and the forecasted value of GRG achieved at case A3B2C3D3 is 0.9721. The GRG is 0.979 when simulation with combined variables is A3B2C3D3, the difference between the two values is 0.215%. The regression chart illustrated in Figure 9 is obtained from the results of regression analysis. The normal probability chart presented the relationship between the probability percentage and the residual. When the residual increases from -0.1 to 0.1, the probability squared percentage increases from 5% to 95%. Nest, the histogram pointed out relationship between frequency and the residual. When the residual increases from -0.1 to 0.1, frequency increases from 0 to 8. The versus fits plot described relationship between fitted value and residual. The fitted value increases from 0 to 1, the residual increase from -0.1 to 0.1. The versus order plot pointed out relationship between 27 simulation cases and the residual. Each simulation case receives a residual value. 27 simulation cases get 27 residual values. The low residual values from -0.1 to 0.1 indicated that the regression model is meaningful and reliable. The residual interval corresponds to the error of the regression model.



(a)



(b)



(c)

Figure 10. Surface graph results

From the regression chart we can draw the conclusion that there are no outliers appearing in the graph, thereby inferring that all cases follow the conclusions of the finite element analysis as the input variable D. has the strongest effect on stress and deformation, followed by A and finally B with the least influence. From there, we can verify the results of the finite element analysis and the above analysis.

4.8 Surface analysis

Figure 10 shows that the increased thickness causes the stress and deformation of the clutch disc to increase and variable B does not have too much impact on variable A. The surface plot of GRG as depicted in Figure 10(a) did not change significantly as B gradually increased. In Figure 10(b), the GRG value changes significantly when D decreases and at the same time decreases, strong when increasing A. It shows that the elastic modulus of the lining material increases, causing deformation and tensile stress to change. Variable D has a large impact on boundary A, causing strong changes in stress and strain. In Figure 10(c), the GRG value changes significantly as A increases gradually. The GRG value does not change significantly as C increases gradually. It shows that the friction plate groove hole (B) the increased friction plate rivet hole does not cause the equivalent tensile stress and the clutch disc deformation to change significantly. Variable C does not have a large effect on variable A. After analyzing the surface, we can see that the A and D boundaries have the strongest influence, thereby inferring the cases where the elastic modulus of the friction plate lining material is 200MPa and the thickness is 3.5mm will give the results. stress and strain are smaller than the other cases. When compared with the finite element analysis, we see the similarity when this result is similar to the analyzed result in the finite element method when the elastic modulus of the lining material is 200GPa and the thickness is 3.5mm friction plate groove is 0.5mm and rivet hole diameter is 3.25mm, the most optimal model. Thereby we can confirm that the results of FEM are optimal. This result is similar to the results of finite element analysis and is more optimal than the results in the study [8] with a stress of 271.97MPa and [9] with a stress of 167.91MPa.

At 95% confidence intervals (CI) were obtained using Eq. (16). The optimum value and predicted value of GRG are 0.979 and 0.9958, So the percentage error between these two values is 1.69%.

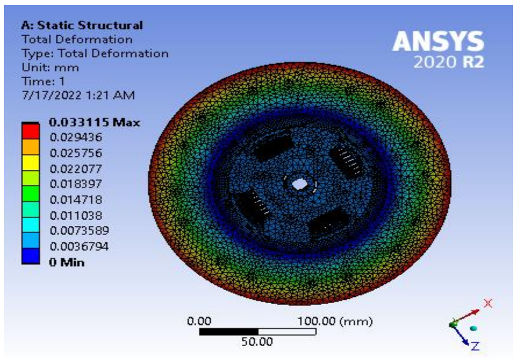
$$\mu_G = G_m + \sum_{i=1}^{q}(G_0 - G_m) = A3 + B2 + C3 + D3 - 3 \times G_m = 0.7618 + 0.6860 + 0.7112 + 0.8330 - 3 \times 0.6654 = 0.9958 \quad (16)$$

For GRG, at a=0.05, fe=20, $F_{0.05}(1,21)=4.3513$, Ve=0.00338 [31], R=4, Re=1, n=27.

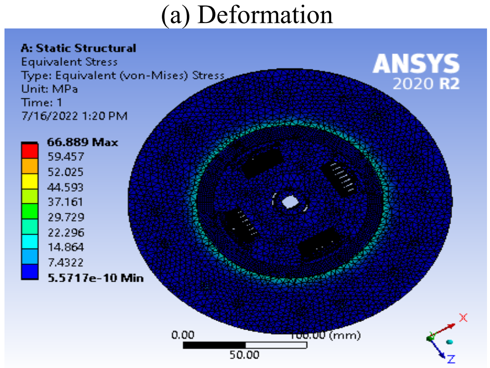
$$CI_{CE} = \pm\sqrt{4.3248 \times 0.00338 \times \left(\frac{1}{\frac{27}{1+6}} + 1\right)} = \pm 0.1357,$$

$$0.8643 < \mu_{\text{corfirmation}} < 1.1357$$

The optimal model after analysis is the clutch disc model of Figure 11 with the friction plate lining material Steel(30Cr13) with the elastic modulus of 200GPa with a thickness of 3.5 clutch plate groove diameter of 0.5mm. rivet hole diameter is 3.25mm with a stress of 66.889MPa and a strain of 0.033115mm. This result is similar to the results of finite element analysis and is more optimal than the results in the study [8] with a stress of 271.97MPa [9] with a stress of 167.91MPa.



(a) Deformation



(b) Stress

Figure 11. The optimal results

Table 8. Comparing results between GRA and TM

Method	GRA	TM	Error (%)
Optimal model	A3B1C3D3	A3B2C3D3	
Predicted value of De	0.03408	0.02743	19.51
Optimal value of De	0.03408	0.033115	2.83
Error (%)	0.00	17.25	17.25
Predicted value of St	68.227	64.6325	5.27
Optimal value of St	68.227	66.889	1.96
Error	0.00	3.37%	3.37

From the results of Table 8, it is shown that the error of the Taguchi method is much larger than that of the GRA method. The error of predicted values of deformation and stress are 19.5% and 5.27% respectively. While the error between the predicted value and the optimum value of deformation the two methods GRA and TM are 0% and 17.25% respectively. The error between the predicted value and the optimum value of stress the two methods GRA and TM are 0% and 3.37% respectively. This problem proved that the Taguchi method is not only single-objective optimization but also large prediction and optimization errors. Therefore, using the GRA method for multi-objective problems is necessary. However, the percentage error of the optimal results of the two methods is not significant. The percentage error for displacement and stress are 2.83% and 1.96%, respectively. This shows that the optimal results of the two methods are reliable. Although the optimal results achieved are comparable with previous studies. However, this result has not yet been confirmed by experiments and algorithms. This is a limitation in this study.

5. CONCLUSIONS

This paper presented and discussed how to optimize the stress and deformation of the friction clutch disc. After analysis, it was concluded that of the input variables, the friction plate lining material affects the stresses and deformation the most, then the thickness. But the thickness when reducing the clutch disc stress inadvertently increases the disc mass, hindering the manufacturing and operating process, and the lining material when reducing the stress does not affect the mass so when the design of the clutch disc, the top priority is still the material. The analysis was verified by grey relation analysis based on Taguchi method, ANOVA, signal-to-noise analysis. Nonlinear regression analysis, facet analysis and interaction analysis were applied to optimize the design deformation and stress of the fiction clutch disc. Variables include: friction plate thickness, hole diameter on friction plate, friction plate groove diameter and the elastic modulus of the friction plate lining material. Finite element analysis was performed by ANSYS software. And the research model is done on CATIA. The finite element analysis results will be verified by grey relation analysis. The result after optimization is that the optimum stress of the clutch is achieved with the stress of 66.889MPa and the deformation of 0.033115mm respectively. The Taguchi method is limited for multi-objective problems. However, the great advantage of this method is that it only requires a few experiments but the results are reliable. This can be effectively combined with multi-objective optimization methods such as grey relational analysis. In the future, it is possible to apply the Taguchi method in combination with other optimization methods such as TOPSIS method, SAW method, artificial neural network, ANSFIS, EDAS method, MOORA method.... to optimize the structure design or any other area that needs optimization. In addition, to achieve better results, it is necessary to perform experiments to confirm the obtained results.

REFERENCES

[1] Gouse Seema Begum, S., Balaraju, A. (2015). Design and analysis of friction clutch plate using ANSYS. International Journal of Advanced Engineering Research and Science, 2(5): 1-5.

[2] Sreevani, B., Mohan, M.M. (2015). Static and dynamic analysis of single plate clutch. International Journal of Innovative Research in Science, Engineering and Technology, 4(9): 8408-8418.

[3] Deshbhratar, V.J., Kakde, N.U. (2013). Design and structural analysis of single plate friction clutch. International Journal of Engineering Research and Technology, 2(10): 3726-3732.

[4] Sahu, M. (2018). Finite element analysis of single plate clutch by using ANSYS. International Journal for Research in Applied Science & Engineering Technology, 6(6): 1337-1346. http://doi.org/10.22214/ijraset.2018.6195

[5] Virmani, K., Madhogaria, T., Baskar, P. (2021). Design optimization of friction lining of a clutch plate. Materials Today: Proceedings, 46: 8009-8024. https://doi.org/10.1016/j.matpr.2021.02.775

[6] Patil, K.K., Randive, V., Mulla, S., Parit, R., Mane, S., Kadam, S. (2020). Design and analysis of single plate clutch by mathematical modelling and simulation. International Journal of Advance Research and Innovation, 8(3): 248-252.

[7] Nutalapati, S., Azad, D., Swami Naidu, G. (2017). Structural analysis of friction clutch plate by changing fillet radius. International Journal of Engineering Research & Technology, 5(1): 79-86.

[8] Narayan, S., Grujic, I., Stojanovic, N., Usman, K.M., Shitu, A., Mahroogi, F.O. (2018). Design and analysis of an automotive single plate clutch. Mobility & Vehicle Mechanics, 44(1): 13-26. https://doi.org/10.24874/mvm.2018.44.01.02

[9] Nivas, B., Nithiyanandam, M., Tharaknath, S., AvinashKumar, A. (2014). Design and analysis of clutch plate using steel material [En–Gjs-400-15steel]. IOSR Journal of Dental and Medical Sciences (IOSR-JDMS), 13(5): 76-78.

[10] Abdullah, O.I., Schlattmann, J., Pireci, E. (2013). Optimization of shape and design parameters of the rigid clutch disc using FEM. FME Transactions, 41(4): 317-324. https://doi.org/10.15480/882.2997

[11] Ramesh, M.R., Ravindra, K.A., Ashok, B., Kannan, C. (2021). Optimizing thermal performance of a dry rigid clutch by varying groove pattern and friction material. Materials Today: Proceedings, 46: 7459-7467. https://doi.org/10.1016/j.matpr.2021.01.130

[12] Sabri, L.A., Topczewska, K., Jweeg, M.J., Abdullah, O.I., Abed, A.M. (2021). Analytical and numerical solutions for the thermal problem in a friction clutch system. Computation, 9(11): 122. https://doi.org/10.3390/computation9110122

[13] Meng, F., Xi, J. (2021). Numerical and experimental investigation of temperature distribution for dry-clutches. Machines, 9(9): 185.

https://doi.org/10.3390/machines9090185
[14] Jabbar, N.A., Hussain, I.Y., Abdullah, O.I. (2021). Thermal and thermoelastic problems in dry friction clutch: A comprehensive review. Heat Transfer, 50(8): 7855-7878. https://doi.org/10.1002/htj.22257
[15] Ali, A., Ali, L., Shah, S.R., Khan, M., Imran, S.H., Butt, S.I. (2017). Dry friction clutch disc of an automobile under transient thermal load: A comparison of friction lining materials. In MATEC Web of Conferences. EDP Sciences, 124: 07003. https://doi.org/10.1051/matecconf/201712407003
[16] Gkinis, T., Rahmani, R., Rahnejat, H., O'Mahony, M. (2018). Heat generation and transfer in automotive dry clutch engagement. Journal of Zhejiang University Science A, 19(3): 175-188. https://doi.org/10.1631/jzus.A1700481
[17] Jonoadji, N., Siahaan, I.H., Simanjuntak, M.E., Sitorus, M.B.H., Suryajaya, M. (2021). Dimensional optimization of clutch disc with simulation of stress analysis (Study cases: Clutch disc Hino FM 260Ti). In IOP Conference Series: Materials Science and Engineering. IOP Publishing, 1034(1): 012005. https://doi.org/10.1088/1757-899X/1034/1/012005
[18] Al-Zubaidi, S., Abdullah, O.I. (2020). Investigation of thermal influence on the frictional characteristics of friction materials. Journal of the Balkan Tribological Association, 26(3): 14-29.
[19] Abdullah, O.I., Sabri, L.A., Al-Sahb, W.A. (2016). Finite element analysis of the thermal behaviour of single-disc clutches during repeated engagements. Tribologia, (2): 9-24.
[20] Huynh, N.T., Huang, S.C., Dao, T.P. (2021). Optimal displacement amplification ratio of bridge-type compliant mechanism flexure hinge using the Taguchi method with grey relational analysis. Microsystem Technologies, 27: 1251-1265. https://doi.org/10.1007/s00542-018-4202-x
[21] Huynh, N.T., Huang, S.C., Dao, T.P. (2020). Design variables optimization effects on acceleration and contact force of the double sliders-crank mechanism having multiple revolute clearance joints by use of the Taguchi method based on a grey relational analysis. Sādhanā, 45: 1-22. https://doi.org/10.1007/s12046-020-01346-w
[22] Huynh, N.T., Nguyen, Q.M., Vo, L.K.T. (2021). Computed methodology for design and optimization on parameters of a tensural displacement amplifier employing flexible hinges. Journal of Mechanical Engineering Research and Development, 44(10): 66-79.
[23] Thai, H.N., Manh, N.Q. (2021). Application of grey relational approach and artificial neural network to optimise design parameters of bridge-type compliant mechanism flexure hinge. International Journal of Automotive and Mechanical Engineering, 18(1): 8505-8522. https://doi.org/10.15282/ijame.18.1.2021.10.0645
[24] Wang, C.N., Truong, K.P., Huynh, N.T. (2019). Optimization effects of design parameter on the first frequency modal of a Bridge-type compliant mechanism flexure hinge by using the Taguchi method. In Journal of Physics: Conference Series. IOP Publishing, 1303(1): 012063. https://doi.org/10.1088/1742-6596/1303/1/012063
[25] Wang, C.N., Yang, F.C., Nguyen, V.T.T., Nguyen, Q.M., Huynh, N.T., Huynh, T.T. (2021). Optimal design for compliant mechanism flexure hinges: Bridge-type. Micromachines, 12(11): 1304. https://doi.org/10.3390/mi12111304
[26] Tran, Q.P., Le, T.D.M., Huang, S.C. (2021). Multi-objective optimization of carbon fiber-reinforced polymer drilling process based on grey fuzzy reasoning grade analysis. The International Journal of Advanced Manufacturing Technology, 115: 503-513. https://doi.org/10.1007/s00170-021-07224-x
[27] Huynh, N.T., Nguyen, T.V., Tam, N.T., Nguyen, Q.M. (2021). Optimizing magnification ratio for the flexible hinge displacement amplifier mechanism design. In Proceedings of the 2nd Annual International Conference on Material, Machines and Methods for Sustainable Development (MMMS2020). Springer International Publishing. Springer, Cham, pp. 769-778. https://doi.org/10.1007/978-3-030-69610-8_102
[28] Jung, J.H., Kwon, W.T. (2010). Optimization of EDM process for multiple performance characteristics using Taguchi method and Grey relational analysis. Journal of Mechanical Science and Technology, 24: 1083-1090. https://doi.org/10.1007/s12206-010-0305-8
[29] Dhuria, G.K., Singh, R., Batish, A. (2017). Application of a hybrid Taguchi-entropy weight-based GRA method to optimize and neural network approach to predict the machining responses in ultrasonic machining of Ti-6Al-4V. Journal of the Brazilian Society of Mechanical Sciences and Engineering, 39: 2619-2634. https://doi.org/10.1007/s40430-016-0627-2
[30] Balaji, V., Ravi, S., Chandran, P.N. (2018). Optimization on cryogenic CO_2 machining parameters of AISI D2 steel using Taguchi based grey relational approach and TOPSIS. International Journal of Engineering &Technology, 7(3.12): 885-893.
[31] Roy, R.K. (2010). A primer on the Taguchi method. Society of Manufacturing Engineers, p. 329.