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Original Research Article

Modeling and Predicting of Tribological Behaviour of the Automotive Brake Pad Using Response Surface Methodology

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Abstract

In the present study, wear and frictional behaviours of automotive brake pad sliding against GG 20 cast iron counter face have been investigated experimentally using a pin on disc type friction tester. Wear tests were carried out under dry conditions. A central composite design was used to describe response and to estimate the process parameters in the model. Empirical models have been developed to predict wear loss and friction coefficient as a function of braking pressure and sliding velocity using multiple regression. The statistical analysis show that the braking pressure is identified as the most dominant factor affecting wear loss, sliding velocity is identified as the most dominant factor affecting friction coefficients.

Key Words: wear, friction coefficient, central composite design, regression model, friction

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1. Introduction

Automotive brake system plays a key role for effective and safe brake performance related issues such as friction force, noise, wear resistance, and brake-induced vibration [1-3]. Automotive brake pads request the friction materials with higher and stable friction coefficient, low wear and noise, low cost and composed of environmentally friendly components. The brake system is very important component for vehicles and machinery equipment in industries [4]. The researchers investigate wear and frictional behaviours of brake system because of the adverse effect observed in the performance and life of brake components. A considerable number of papers dealing with the wear and frictional behaviours of automotive friction materials have been published [5-13]. Several researchers have developed mathematical models by using response surface methodology, Taguchi techniques and artificial neural networks to predict the wear loss and the friction coefficient in terms of various process parameters of different materials [14-20].

Automotive friction materials have been formulated for about 100 years [6]. In the early 1920s, asbestos fiber was chosen as a friction material to use in all kind vehicles. Nowadays, non-asbestos organic formula becomes very important to overcome the negative effect of asbestos on human respiratory system. The abrasives in the brake friction materials play important roles in determining the stopping distance, counter disc wear and noise propensity [7]. The selection of the abrasives used in commercial brake friction materials depends on their hardness, size, shape, fracture toughness, wear resistance and aggressiveness against the counter discs [9].

In this study, the tribological performance of the non-asbestos organic type non-commercial brake pad sliding against GG 20 cast iron counter face has been investigated experimentally under dry sliding conditions. The Design of Experiments was done based on response surface methodology (RSM). Braking pressure and sliding velocity were selected as factors, wear loss and friction coefficient were selected as response variables for the statistical analysis.

2. Experimental Procedures

The brake pad material studied in this study was based on a typical non-asbestos organic type and contained a binder, reinforcements, fillers, and friction modifiers. The composition of the investigated non-asbestos friction material is summarized in Table 1.

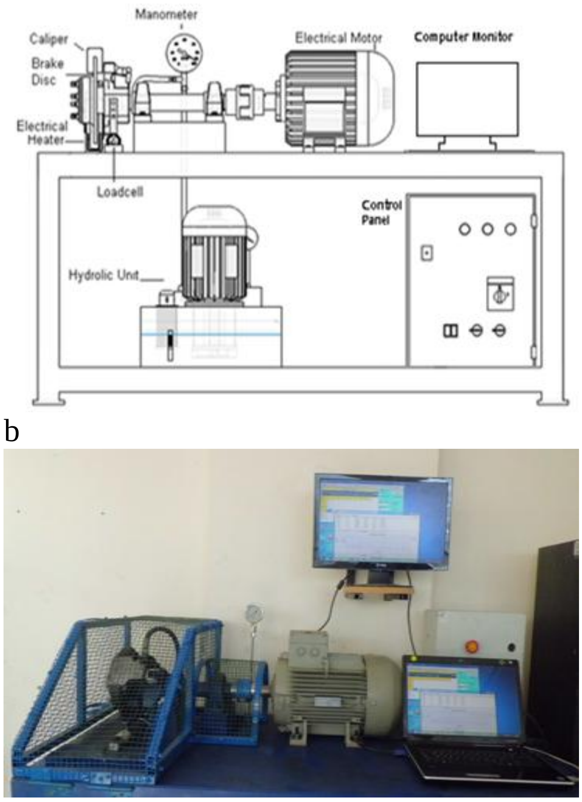
Table 1. The ingredients of the non-commercial brake pad materials (weight %)

The ingredients	%
Phenolic resin	15
Steel Fibers	10
Brass Particle	4
Graphite	10
Cu particles	15
Cashew	10
Barite	29
Al_2O_3	7

Brake pad specimens were produced by a conventional procedure for a dry formulation following dry-mixing, pre-forming, hot pressing, post-curing, scratching, and grinding. All constituents were weighed with a sensitivity of 0.1 mg and mixed in a blender for 3–4 min until a uniform dispersion was obtained. After the mixing operation, the mixture was charged in a mold. The mixture was hot pressed at 150°C and 15 MPa braking pressure for 10 min and subsequently post cured. The brake pad was produced approximately 25.4 mm in diameter. A pin on disc test equipment used in this study has been shown in Figure 1. Tests with GG 20 as the counter face material were carried out under dry sliding condition at room temperature. The test sample was mounted on the hydraulic holder and pressed against the flat surface of the rotating disc. Disc samples of 227 mm in diameter and 9.75 mm in thickness were obtained from a domestic company in Turkey. Before performing the tribological test, the surfaces of the test samples and the GG 20 cast iron discs were ground with 320-grid emery paper. The average roughness (Ra) of gray cast iron disc was measured as 1.40 μm. The hardness of the disc samples was also measured as 191

HB using 5 mm ball and 150 kgf load. By weighing the test samples to determine loss of mass in braking pads, wear loss amount was calculated for each test conditions.

a



b

Figure 1. Test set up a) Schematic illustration pin on disc wear tester, b) Wear test machine

Table 1. Experimental factors and levels for CCD

Factors/Levels	-1.41	-1	0	1	+1.41
Braking pressure (kPa)	438	500	650	800	862
Sliding velocity (m/s)	5.4	6	7.5	9	9.6

The objective of the study is to find the influence of factors affecting the tribological performance of brake pad-GG 20 disc system. Braking pressure and sliding velocity were considered as model variables and wear loss and friction coefficient as response variables. The Design Expert 7 software was used for designing and analysing experiment. Central composite design (CCD) was adopted to obtain an empirical model of wear loss and friction coefficient as a function of the braking pressure (kPa) and sliding velocity (m/s). The range of each parameter was coded in five levels (−1.41, −1, 0, 1, +1.41). The levels of model variables were shown in Table 2. The plan of experiments was made by randomising the experiments to avoid accumulation of errors. The experiments were conducted based on randomised run number.

Three replications of each factor level combinations were conducted resulting in a total of 39 tests and average values were reported. To apply different abrasive conditions during each test, on the rotating disc surface and the sample is fixed in a holder. The samples were loaded against the abrasive medium. Samples were weighed by analytical scales with 0.01 g sensitiveness. After each test, samples were weighed again. The wear loss was computed from the mass loss of the sample. The friction coefficient values were obtained from databank in wear tester machine.

3. Results and Discussion

CCD is an experimental design in RSM for building an empirical model for the response variable. This design consists of a factorial portion and axial portion and a central point [21]. Experimental levels for process variables were selected according to a CCD. This design has 9 different design points for all combinations of process variables. The arrangement and the experimental (actual) and predicted wear loss values of based on the CCD rotatable design are shown in Table 3. The experimental results have been used to build the mathematical models. Regression analysis indicate second order model adequately represents the wear loss and friction coefficient. In order to predict, wear loss and friction coefficient second order regression equations were expressed in Equations 1 and 2 in terms of coded factors.

$$y = 0.094 + 0.067x_1 + 0.022x_2 + 0.038x_1^2 + 0.016x_2^2 \quad (1)$$

$$\mu = 0.30 - 0.037x_1 - 0.10x_2 - 0.026x_1^2 - 0.012x_2^2 - 0.012x_1x_2 \quad (2)$$

where; y is the estimated wear loss, μ is the friction coefficient, x_1 is the coded factor that represents the braking pressure x_2 is coded factor represents sliding velocity.

Analysis of Variance (ANOVA) tables for the second order model for the wear loss and friction coefficient have been given in Table 4 and Table 5, respectively. ANOVA was used to identify the relationships between the output and input parameters. It was also employed to find significance of the factor effects based on a significance level 5%. When the p value is less than 0.05, the results indicate statistical significance. The performance criterion such as mean absolute percentage error (MAPE) and absolute fraction of variance (R^2) were used to the developed models. The value of Adjusted R^2 between experimental results and predictive values is obtained 97% for the wear loss. The R^2 value indicates that the wear parameters explain 97% of variance in wear loss. This value showed that the empirical model fits well with experimental results. The comparisons of experimental results with the CCD predictions have been depicted in terms of MAPE. It was found to be as 3% for the wear loss. The value of Adjusted R^2 between experimental results and predictive values is obtained 96% and MAPE was found to be as 5.3% for the friction coefficient. Results in Table 4 suggest that the most significant effect on the wear loss was exhibited by the braking pressure, followed by sliding velocity. Results in Table 5 suggest that the most significant effect on the friction coefficient was exhibited by the sliding velocity, followed by the braking pressure. As seen Tables 4 and 5, higher F value indicates that the variation of the process parameter makes big changes and demonstrates greatest contribution on the wear loss and friction coefficient.

Table 3. Actual and predicted wear loss (*g*) and friction coefficient values for CCD

Std	**Run**	**Braking pressure (kPa)**	**Sliding velocity (m/s)**	**Actual wear loss (g)**	**Predicted wear loss (g)**	**Actual friction coefficient**	**Predicted friction coefficient**
13	1	650	7.5	0.093	0.094	0.292	0.30
10	2	650	7.5	0.095	0.094	0.300	0.30
5	3	438	7.5	0.075	0.076	0.325	0.30
6	4	862	7.5	0.253	0.260	0.187	0.20
12	5	650	7.5	0.094	0.094	0.280	0.30
4	6	800	9.0	0.238	0.240	0.115	0.11
1	7	500	6.0	0.070	0.060	0.378	0.39
11	8	650	7.5	0.093	0.094	0.237	0.30
7	9	650	5.4	0.080	0.097	0.416	0.42
3	10	500	9.0	0.098	0.100	0.192	0.21
2	11	800	6.0	0.211	0.190	0.350	0.34
9	12	650	7.5	0.097	0.094	0.350	0.30
8	13	650	9.6	0.163	0.160	0.150	0.14

Table 4. The ANOVA table for the wear loss

Source	**SS**	**DF**	**MS**	**F value**	**p- value**
Model	0.050	5	0.010	80.54	<0.0001 significant
A	0.035	1	0.035	285.70	0.0001*
B	3.71E-003	1	3.71E-003	29.91	0.0009*
A^2	9.87E-003	1	9.87E-003	79.52	0.0001*
B^2	1.87E-003	1	1.87E-003	15.11	0.0060*

Table 5. The ANOVA table for the friction coefficient

Source	**SS**	**DF**	**MS**	**F value**	**p- value**
Model	0.096	5	0.019	59.42	<0.0001 significant
A	0.011	1	0.011	34.35	0.0006*
B	0.079	1	0.079	244.60	<0.0001*
A^2	4.73E-003	1	4.73E-003	14.56	0.0066*
B^2	1.05E-003	1	1.05E-003	3.25	0.1142
AB	6.00E-003	1	6.00E-003	1.85	0.2161

4. Conclusions

From the above experiments and analysis following conclusions are derived.

- Regression analysis indicate that the second order regression model adequately represent wear loss and friction coefficient in terms of process variables.
- The correlation with experimental results and predicted values was good within the range of their investigation

and the individual operating parameters.

- According to the statistical analysis, Adjusted R^2 values were obtained as 0.97 and 0.96 for the wear loss and friction coefficient, respectively.
- Results show that the predicting values close to the actual values. Mean absolute percentage errors for CCD were found to be as 3% and 5.3% for wear loss and friction coefficient, respectively.
- ANOVA results show that braking pressure is the most dominant factor affecting wear loss, and sliding velocity is the most dominant factor affecting friction coefficient.

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