Advances in Mechanical Engineering and its Applications (AMEA)
Vol. 1, No. 4, 2012, ISSN 2167-6380

www.worldsciencepublisher.org

Vibration Analysis of the Friction clutch Disc Using Finite Element Method

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Abstract – The new direction of development of the automotive vehicle ride comfort and smooth driving is associated with the advancement of the machine parts design, e.g. the dry friction clutch, this part which consider as essential element to transfer power from engine to gearbox. In this research, a numerical technique (finite element method) is used to model a disc of friction clutch and compute the natural frequencies and mode shapes. Natural frequencies calculation has been made for frictional lining of different thicknesses and types of materials. Furthermore, the effect of dimensionless radius ratio R (R=inner disc radius/outer disc radius) on the vibration characteristics is investigated as well. The ANSYS/WORKBENCH 13 has been used to perform the numerical calculation in this paper.

Keywords – Friction clutch disc; free vibration; friction material; 3D FEM.

1. Introduction

Automobile friction clutch is an essential component in the process of power transmission, therefore all designers want to obtain the best possible performance with comfortable condition (reduce the noise and vibration as much as possible) for the friction clutches. The vibration and noise generated during the engagement is one of the biggest obstacles faced designers; this is because there are many variables that affect on this phenomenon such as pressure distribution, coefficient of friction, materials properties, and sliding velocity …etc. For that reason, it's very important to estimate the natural frequencies of clutch disc and the corresponding modal shapes within acceptable degree of accuracy at the design stage.

Plakhtienko et al. [1] used the iteration method (different mechanisms of vibration) to study the dynamic behavior of a two-flywheel system that contains a friction pair and is brought into motion by a motor. The torque for the system decreases linearly with an increase in rotor speed. It's found, that the energy dissipated during the slipping between contact surfaces is effective factor on the flywheel response.

Gaillard and Singh [2] use five lumped parameter linear or nonlinear models to simulate the torsional dynamic system of an automotive clutch. Each nonlinear model consists of visco-elastic and dry friction elements. It's clear from the result of dynamic stiffness and energy dissipation spectra, that the excitation amplitude and frequency dependent behavior. The dynamic history of the system is estimated and analyzed. The results of the proposed models show good agreement with the experimental results.

During the engagement of a dry friction clutch in a vehicle with manual transmission, the noise problem (Eek) arises, and this phenomena produced several disadvantages which effect on the performance of the vehicle. The measurements show that near the full engagement the pressure plate suddenly starts vibrating with a frequency close to the first natural frequency of the rotational sub-system. The dynamic stability during self-excited vibration exhibits when the coefficient of friction is constant. Due to the high noise levels occurs in the transient period, some of consumers change the clutches prematurely in an effort to eliminate this noise. For this reason it’s very important to reduce the noise during this period to increase the comfortable conditions for consumers [3].

Ahn et al. [4] developed a new model in which the spring used in the clutch damper is divided into a finite number of elements. The model takes many unique properties of arc-shaped springs into consideration and is anticipated to be more precise than conventional simple models. With the model, two meaningful results were presented which can be utilized afterwards. One is a simulation concerning the peak torque transmitted via the clutch damper; also the simulation shows the hysteretic characteristics of the clutch damper.

Skup [5 & 6] investigated the theoretical process of damping of non-linear vibrations when using a three-mass model of power transmission system with a friction clutch. Both effects of passive damping and active damping on the simultaneous structural friction phenomenon have been taken into account. This work covers the effect of external load, unit pressure, viscotic damping coefficient of friction and gain factor (geometric parameters) on the resonance behavior of the steady-state vibrations. The equations of motion examined the system when solved by means of the Van der Pol method and digital simulation.

Crowther et al. [7] investigated the effective factors on the judder phenomena during clutch engagement and stick–slip using analytical and numerical methods. Four degree-of freedom used for model the torsional system with slipping clutch and for a power train with automatic transmission system. Stability analysis shows that the clutch judder is dependent on values of friction coefficient. Numerical simulations used to develop the modeling of algorithm for stick–slip, and the results of this algorithm show the possibility of stick-slip is increased with clutch pressure fluctuations, judder engagement, and external torque fluctuations. Numerical simulations for second to third gear up shifts display that the possibility of stick-slip occurring for clutch engagement is increased with applied pressure , judder approaching engagement, and external torque and that the possibility of stick-slip occurring decreased dramatically when applied ramps pressure.

In this study a finite element method has been used to compute the natural frequencies and mode shapes of friction clutch discs, this investigation covers the effect of dimensionless radius ratio R, friction material type and friction material thickness on the vibration characteristics of the disc clutch.

2. Modal Analysis

The modal analysis is considered essential step in the design process to estimate the vibration characteristics of the designed structure. Hence, the goal of a modal analysis is determining the natural frequencies and mode shapes. Modal analysis can also be taken as a basis for other more detailed dynamic analyses such as a transient dynamic analysis, a harmonic analysis or even a spectrum analysis based on the modal superposition technique. The main assumption in the modal analysis is that the system is linear and ignored any nonlinearity in the system [8]. Fig. 1 illustrates the general procedure to find the natural frequencies and mode shapes for any structure. It's clear from this diagram, building model is considered the most important step in the modal analysis, because of this process is the first requirement for modal analysis, and then for solving more complicated dynamic problems. Fig. 2 demonstrates four models of dry friction clutch (commercially types) built by using ANSYS13.

3. Finite Element Formulation

The general equation of motion can be written as [9]: -

$$[\mathrm{M}]\{\ddot{\mathrm{U}}\}+[\mathrm{C}]\{\dot{\mathrm{U}}\}+[\mathrm{K}]\{\mathrm{U}\}=\{\mathrm{R}\} \qquad (1)$$

Where [M], [C] and [K] are mass, damping and stiffness matrices respectively. {R} is the external load vector, {U}, $\{\dot{U}\}$ and $\{\ddot{U}\}$ are the displacement, velocity and acceleration vectors respectively. Assume the excitation force and damping are neglected, then rewrite Eq. (1) yield,

$$[\mathrm{M}]\{\ddot{\mathrm{U}}\}+[\mathrm{K}]\{\mathrm{U}\}=0 \qquad (2)$$

Assume,

$$U_i=\Phi_i \sin(\omega_i t+\theta_i), \quad i=1,2.....DOF \qquad (3)$$

In this harmonic expression, Φ_i is a vector of nodal amplitudes (mode shape) for the ith mode of vibration. The symbol ω_i represents the angular frequency of mode i, and θ_i denotes the phase angle. By differentiating Eq. (3) twice with respect to time yield:

$$\ddot{U}_i=-\omega_i^2\Phi_i \sin(\omega_i t+\theta_i) \qquad (4)$$

Substitution of Eq. (4) and Eq. (3) into Eq. (2) allows cancellation of the term sin (ω_i t+ θ_i) and re-arranged yield,

$$([K]-\omega_i^2[M])\Phi_i=0 \qquad (5)$$

Eigenvalue problem [Eq. (5)] solved by subspace Block Lanczos eigenvalue extraction method. In all computations for the dry clutch disc, it has been assumed homogenous and isotropic materials, and all parameters and materials properties are listed in Table. 1. Fig. 3 & 4 show the three dimensional model and suitable mesh size for clutch disc.

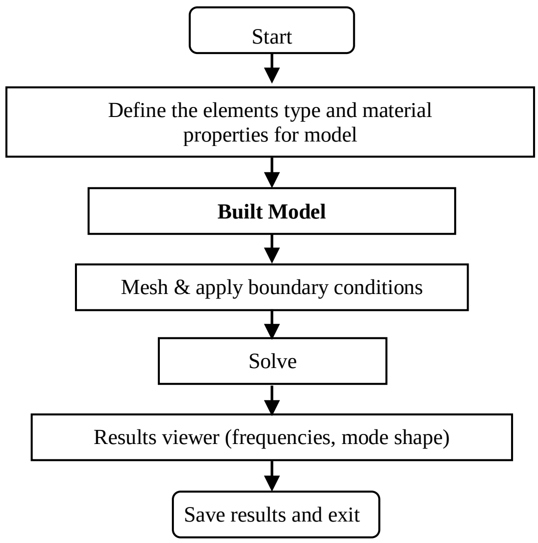
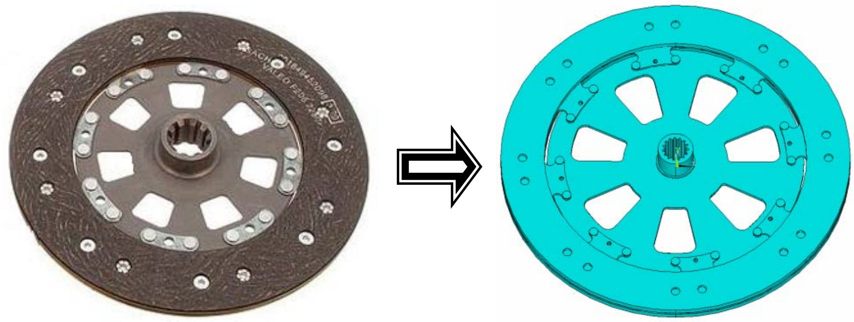
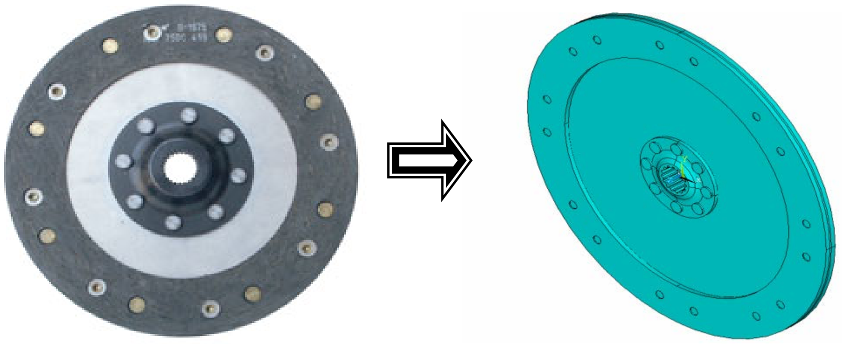


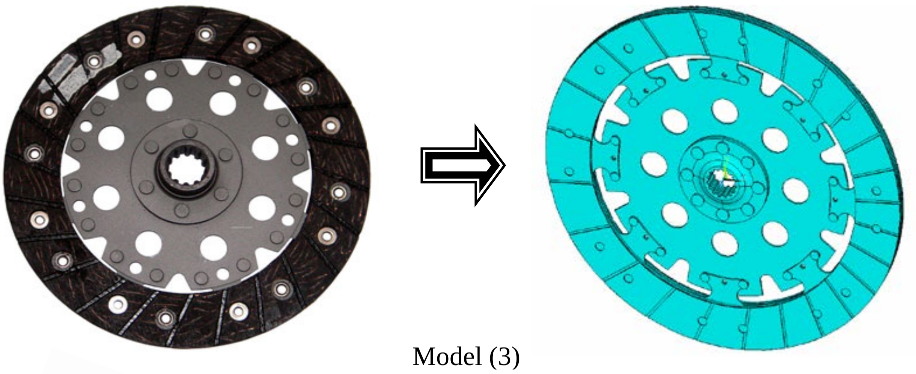
Fig.1 The typical block diagram for the modal analysis



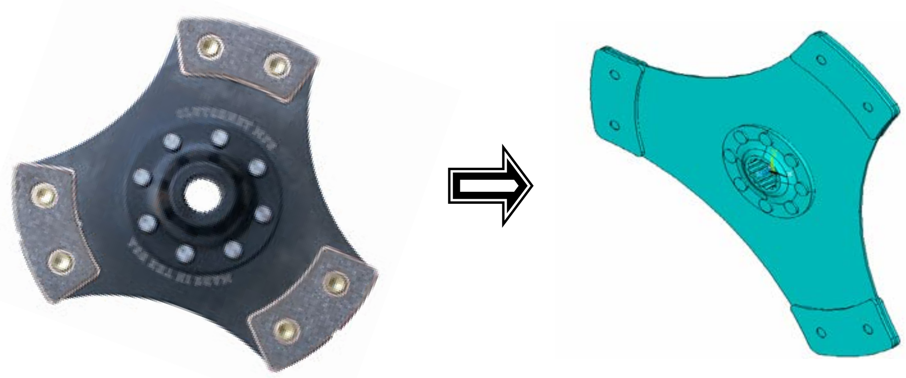
Model (1)



Model (2)



Model (3)



Model (4)

Fig. 2 Three dimensional models for clutch discs

Table. 1 The model parameters and material properties

Parameters	Values
Outer radius, r_o [m]	0.14
Thickness of plate disc, t_p [m]	0.003
Steel material	
Young's modules [Gpa]	125
Poisson's ratio	0.25
Density [kg/m^3]	7800
Friction material (A)	
Young's modules [Mpa]	300
Poisson's ratio	0.25
Density [kg/m^3]	2000
Friction material (B)	
Young's modules [Mpa]	75
Poisson's ratio	0.25
Density [kg/m^3]	733

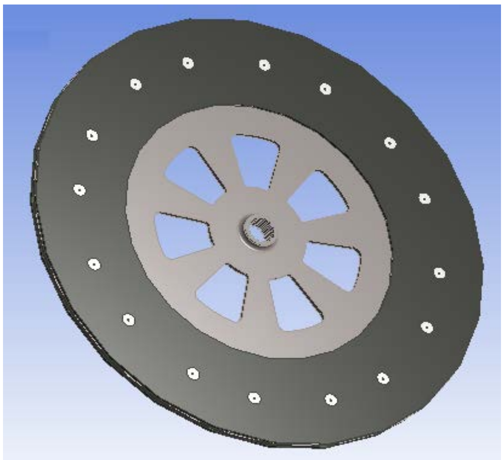


Fig. 3 Three dimensional model for disc of dry friction clutch

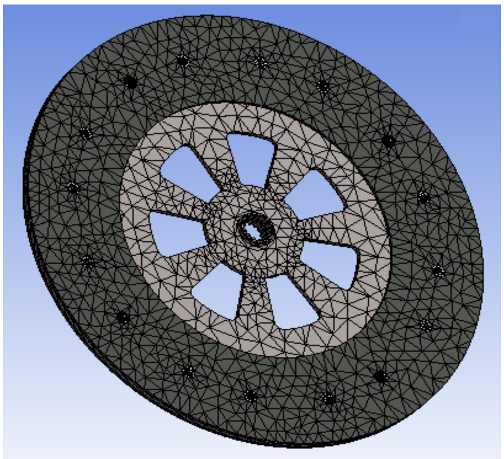


Fig. 4 Suitable mesh size for disc of dry friction clutch (element No.=21878 & node No. =48452)

4. Results and Discussions

In this paper the vibration characteristics for the clutch disc has been investigated, the natural frequencies and mode shapes are computed for different material type, friction lining thickness (t_f) and dimensionless radius ratio (R). This analysis has been done using ANSYS 13 software. Fig. 5 shows the mode shapes for the first six modes of disc clutch.

Table. 2 exhibits the values of natural frequencies (the first six modes) with dimensionless radius ratio (R). It can be seen from this table, that the values of natural frequencies increase when the dimensionless radius ratio (R) increase, the reason for this result is due to the change in the mass of frictional lining (when R increases the mass of frictional lining decreases and R decreases the mass of frictional lining increases). And when the mass of frictional lining change the natural frequency of disc clutch will change also, the basic principles of vibration theory states that the inverse relationship between the mass and natural frequency of the body [10].

$$f_n = \frac{1}{2\pi}\sqrt{\frac{k}{m}} \tag{6}$$

Where f_n, k and m are the natural frequency [Hz], stiffness and mass respectively.

Table. 3 presents the values of natural frequencies (the first six modes) with thickness of frictional lining. It can be noted, that the natural frequencies values decrease when the frictional lining thickness increases (the natural frequency decreases with increases of mass for disc clutch).

Fig. 6 shows the values of natural frequencies (first six modes) for two types of friction material (A & B). It can be seen from this figure, that the values of natural frequencies for material (A) are less than material (B) and the different between their is very small, these result because of two reasons the first one is the density for the material (A) is greater than material (B) and this make the values of natural frequencies for material (B) greater than material (A) and the second reason for small different between results is the modulus of elasticity of material (A) is greater than material (B).

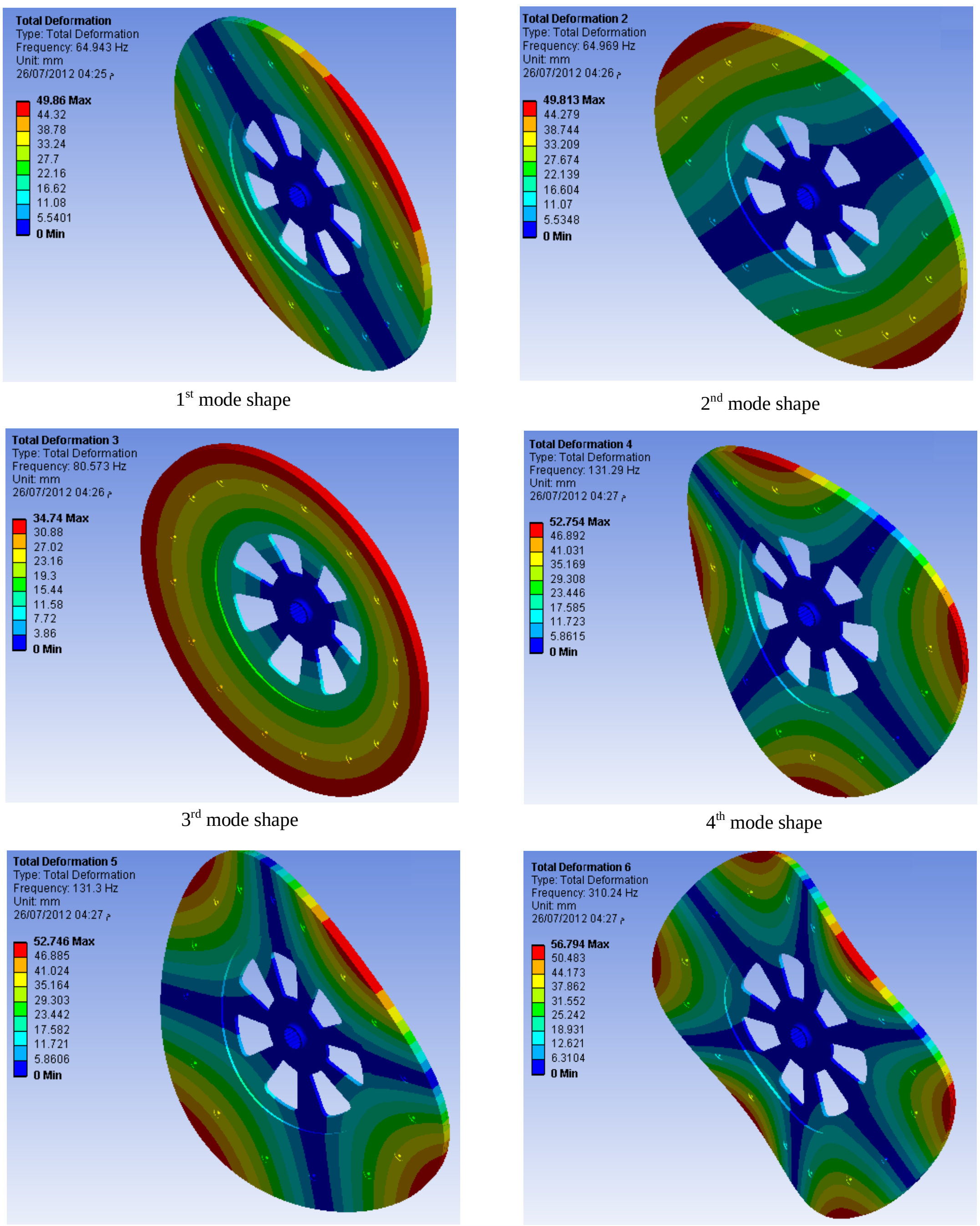
5. Concluding Remarks

In this paper the vibration analysis for the friction clutch disc was performed to study the influence of various parameters such as friction material thickness, type of friction material and dimensionless radius ratio R on the natural frequency for clutch disc. Three-dimensional model was built to obtain the vibration characteristics.

The conclusions obtained from the present work can be summarized as follows:

1. The natural frequency increases when R increases, and this increment in frequency value depend on the values of modulus of elasticity and density of friction material.
2. The natural frequency decreases when the thickness of the friction material increases because the total mass of clutch disc increases.
3. The natural frequencies for friction material (B) are greater than material (A) and the difference between results increases with increase number of modes.

The knowledge of vibration characteristic for the friction clutch disc is considered essential item for successful design, because of there are several disadvantages produces by vibration such as excessive and unpleasant stresses, rapid wear, large amplitude produces by resonance (this will lead to failure of the system when clutch disc work long time in this situation). Therefore it's necessary to investigate the vibration characteristics of friction clutches to increase the efficiency of performance and reduce the noise. The present work is a preliminary of subsequent investigation of the dynamic behavior for dry friction clutch, and effect of geometric parameters and type of materials on this behavior.



1st mode shape
2nd mode shape
3rd mode shape
4th mode shape
5th mode shape
6th mode shape

Fig.5 The first six mode shapes for the dry friction disc (R=0.55, t_f=2mm).

Table. 2 The values of natural frequencies of dry friction disc for different values of dimensionless radius ratio (R)

	Natural frequency [HZ]			
Mode No.	**R=0.5**	**R=0.55**	**R=0.6**	**R=0.6**
1	64.78	64.94	65.18	65.53
2	64.81	64.96	65.21	65.53
3	80.42	80.57	80.83	81.23
4	131.14	131.29	131.59	132.07
5	131.15	131.3	131.6	132.08
6	310.05	310.24	310.66	311.43

Table. 3 The values of natural frequencies of dry friction disc for different thicknesses

	Natural frequency [HZ]		
Mode No.	**t_f= 2mm**	**t_f =2.5mm**	**t_f =3 mm**
1	64.94	63.36	61.57
2	64.96	63.53	61.59
3	80.57	78.69	76.73
4	131.29	127.92	124.91
5	131.3	127.97	124.92
6	310.24	302.02	295.04

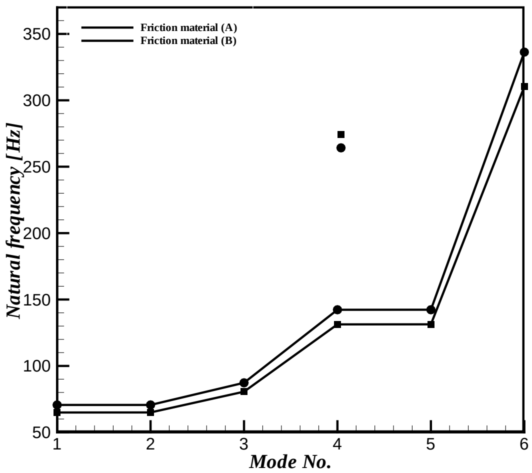


Fig. 6 The first six modes of dry friction disc for different friction material (A & B)

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