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Vibration Analysis of Dry Friction Clutch Disc by 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 the various parameters by changing and is investigated on the vibration characteristics as well. Also the numerical approach is applied for the verification. The software CATIA-V5R18 is used for modeling. The ANSYS/WORKBENCH 14.5 has been used to perform the numerical calculation in this paper.

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.

VinayakRanjanand M.K. Ghosh[1] is studiedthe in-plane free vibration of an elastic and isotropic disk on the basis of the two-dimensional linear plane stress theory of elasticity. The boundary characteristic orthogonal polynomials are employed in the Rayleigh–Ritz method to obtain the natural frequencies and associated mode shapes. In the work, free and forced transverse vibration behavior of a spinning disc with a rigid core having discrete patches and discrete masses attached at its periphery have been analyzed using finite element method.

PROBLEM STATEMENT

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.

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.
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 and thickness of disc (t_p) on the vibration characteristics of the disc clutch.

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 an nonlinearity in the system. Fig. 2 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 ANSYS14.5.

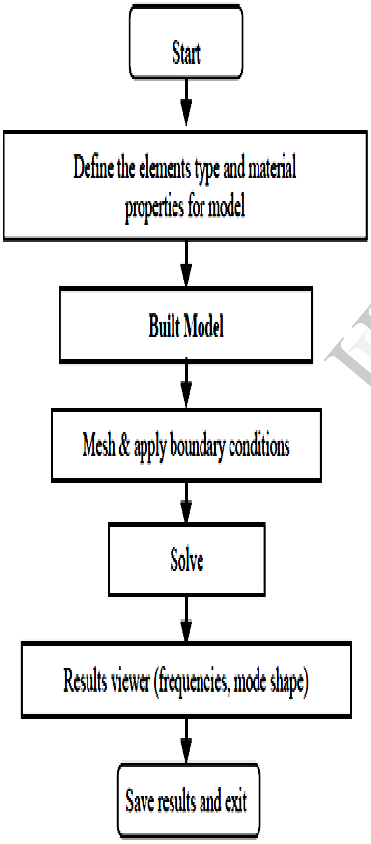

Fig.2 The typical block diagram for the modal analysis

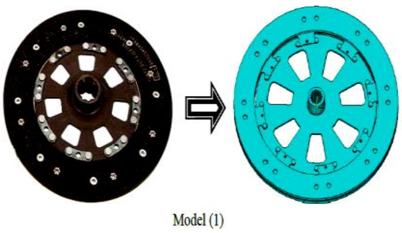

Fig. 3 Three dimensional model for clutch disc

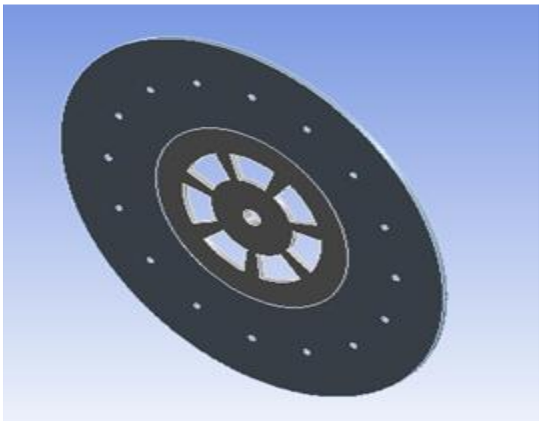

Fig. 4 Three dimensional model disc of dry friction clutch

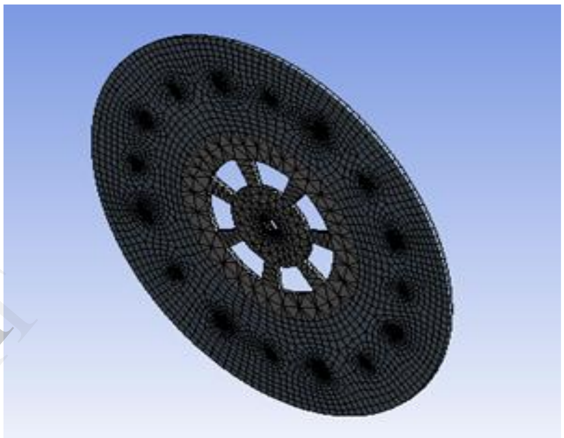

Fig. 5Suitable mesh size for dry for disc of dry friction clutch

Table. 1 The model parameters and material properties

Parameters	Values
Outer radius, r_o [m]	0.14
Thickness of plate disc, T_p[m]	0.03
Steel material	
Young's modules [Gpa]	125
Poisson's ratio	0.25
Density [kg/m^3]	7800
Friction material	
Young's modules [Mpa]	300
Poisson's ratio	0.25
Density [kg/m^3]	2000

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.

Also in actual, the natural frequency of dry friction clutch disc should be always be greater than forced frequency[4]. As when the forced frequency is became same as that of natural frequency then the resonance will occur. This is more dangerous condition for any working component. So to avoid resonance, natural frequency should be greater than forced frequency. Now we have equation for natural frequency as,

$$f_n = \frac{1}{2\pi}\sqrt{\frac{k}{m}} \qquad (1)$$

Where f_n, k and m are the natural frequency [Hz], stiffness and mass respectively.
The frequency can be altered by following two ways:-
1) Changing the mass of the component
2) Changing the stiffness of the component

Again there is one thing to describe i. e. on the basis of rpm of particular disc, dimensionless radius ratio(R)should be designed.

A numerical approach is used for the calcuiation of natural frequency of clutch disc as follows:

The natural frequency is given as,

$$\omega_n = B\sqrt{Et^3 / \rho a^4 (1-\upsilon^2)}$$

and

$$\omega_n = 2\pi N$$

Where,
B – Frequency parameter
E - Young's modulus
a - diameter of plate
u - Poisson's ratio
□- density
N – Natural frequency in Hz
The various mode parameter value of 'B' is given as follows:

Table. 2 The values of frequency parameter 'B'[3]

m \ n	0	1	2	3	4
0	-	-	2.009524802	3.115921966	4.176852520
1	3.000522846	4.524881227	5.892050377	7.189832951	8.444916203
2	6.200257918	7.733795398	9.166760558	10.53907278	11.86939309
3	9.367509371	10.90675641	12.37183066	13.78540518	15.16047485

n- no. of nodal diameter
m - no. of nodal circle

RESULTS AND DISCUSSIONS

In this paper the vibration characteristic for the clutch disc has been investigated, the natural frequencies and mode shapes are computed for dimensionless radius ratio (R) and also for thickness (t_p). This analysis has been done using ANSYS 14.5 software. Fig.6 and 7 shows the mode shapes for the first four modes of disc clutch for dimensionless radius ratio (R) as 0.5and 0.55, respectively. Fig. 8 shows the mode shapes for the first four modes of disc clutch for dimensionless radius ratio 0.5 and thickness (t_p) of disc 0.04.

Table. 3exhibit the values of natural frequencies (the first four 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 willalso change, the basic principles of vibration theory states that the inverse relationship between the mass and natural frequency of the body.

Table. 4 exhibit the values of natural frequency (the first four modes) with changing thickness (t_p). It can be seen from this table that the values of natural frequencies increase when thickness (t_p) of disc increases. The reason for this is increase in thickness increases the stiffness of disc. Therefore the natural frequency of disc increases.
Table. 5 give the limit of RPM of particular disc by which it has to be rotate and having dimensionless radius ratio (R) as a limit factor. If number of RPM is increased, then the required dimensionless radius ratio (R) will also increase. So according to requirements the design will also change.

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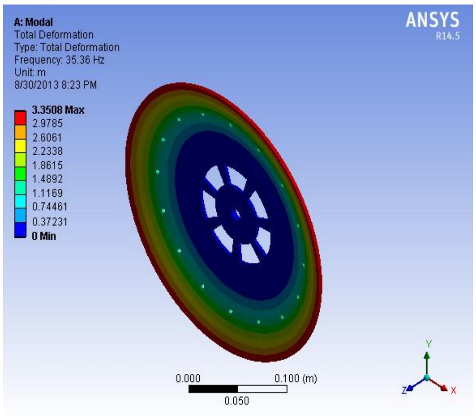
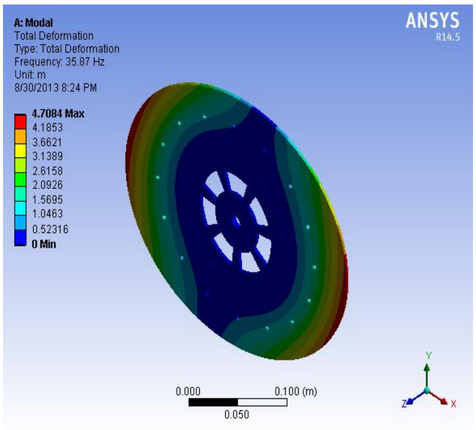
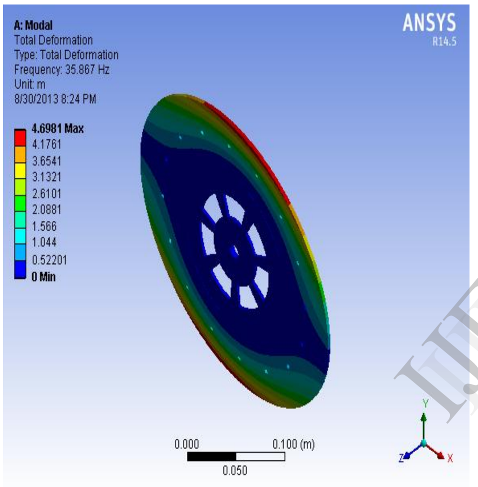


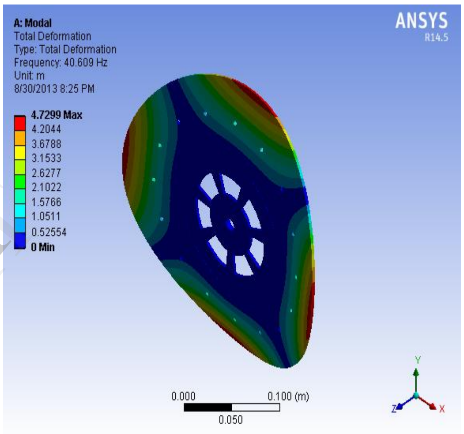
1st mode shape



3rd mode shape

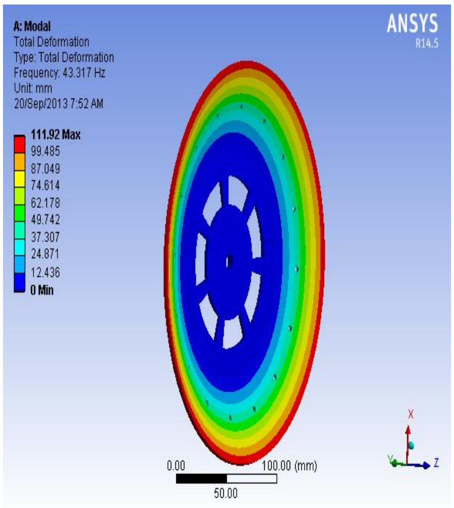


2nd mode shape

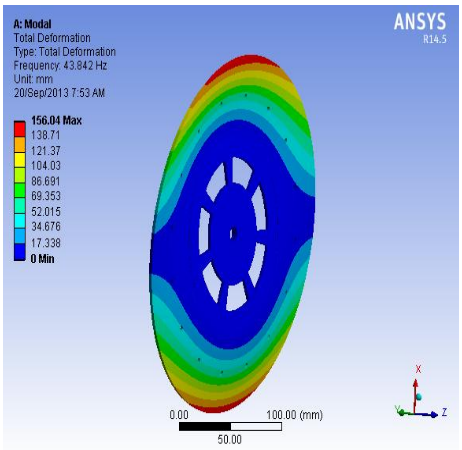


4th mode shape

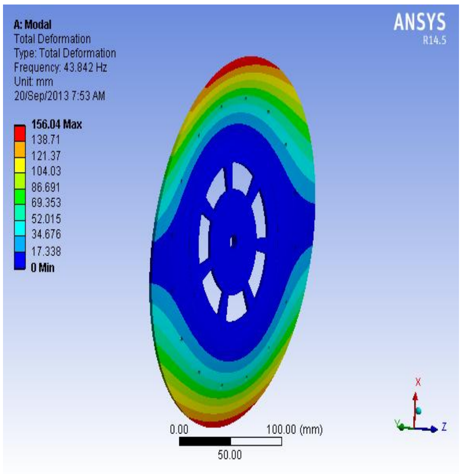
Fig.4 The first four mode shapes for the dry friction disc(R=0.5)



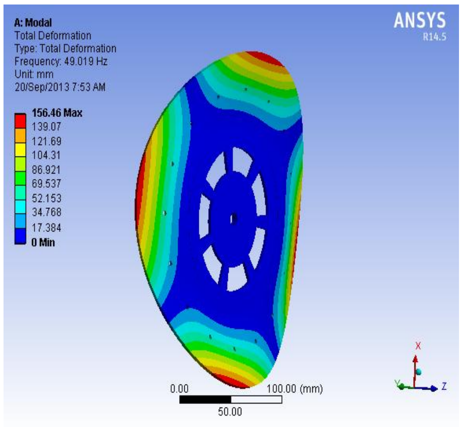
1st mode shape



3rd mode

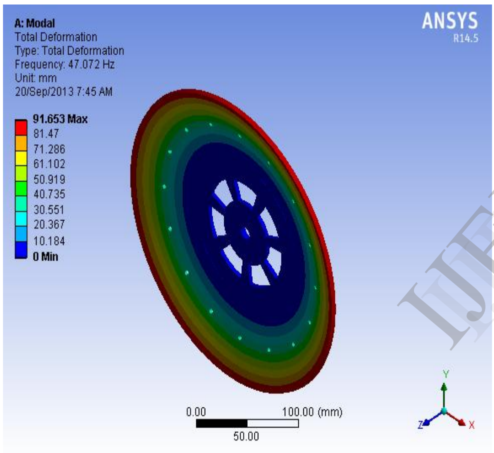


2nd mode shape

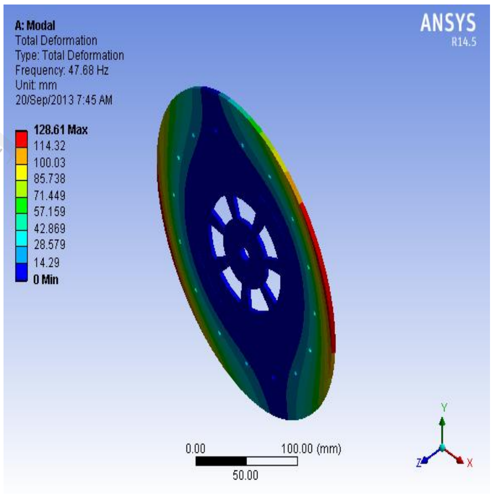


4th mode shape

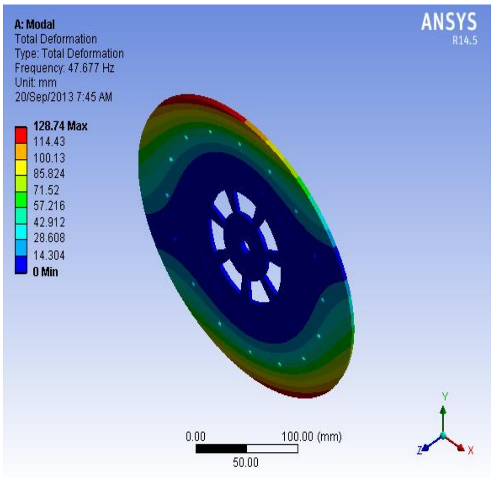
Fig.5 The first four mode shapes for the dry friction disc(R=0.55)



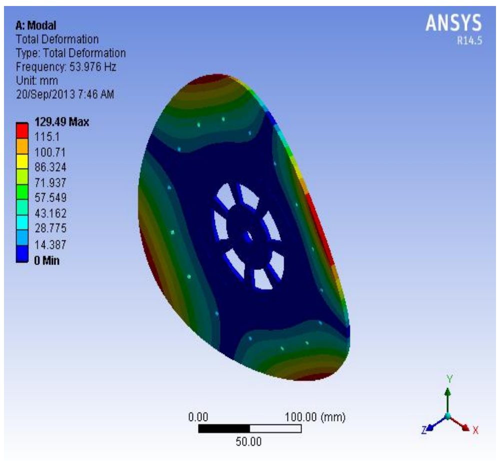
1st mode shape



2nd mode shape



3rd mode shape



4th mode shape

Fig.8 The first four mode shapes for the dry friction disc(R=0.5&t_p=0.04m)

Table. 3 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	
	Ansys	Calculated	Ansys	Calculated
1	35.36	31.60	43.317	33.74
2	35.867	41.37	43.837	43.16
3	35.87	50.19	43.842	53.59
4	40.607	59.01	49.019	63

Table.4 Comparison between natural frequency of dry friction disc by changing thickness T_p(m) of disc for dimensionless radius ratio (R=0.5)

	Natural frequency [HZ]	
Mode no.	T_p= 0.03m	T_p=0.04m
1	35.36	47.072
2	35.867	47.667
3	35.87	47.68
4	40.607	53.976

Table. 5 Allowable RPM by which the disc to be rotated

Dimensionless radius ratio(R)	0.5	0.55
Fundamental frequency	35.36	43.317
RPM	<2100	<2580

8. CONCLUDING REMARKS

In this paper the vibration analysis for the friction clutch disc was performed to study the influence of various parameters such as thickness of disc (t_p) 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 increases when the thickness of the clutch disc increases because the total stiffness of clutch disc increases.

[3] The best result is getting by increasing thickness than by increasing the dimensionless radius ratio. So it is better to go with the increase in thickness of disc (t_p), but it has a certain limit. Because the mass of disc will also increase and with this the working of a component is so difficult. That's why dimensionless radius ratio (R) is to be preferred for increasing after some limit.

[4] With the certain dimensionless radius ratio (R), the disc should be used for its designed limiting RPM.

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