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Torsional Vibration Analysis Of Meshed Gear System

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Abstract

Vibrations are generally produced when the object is provided a motion and there is an unbalance force generation and because of that the machine or Vehicle starts shaking. Vibrations are generally classified into two types Longitudinal and Transverse vibrations according to the axis or direction of the rotation. Torsional Vibrations are categorised in transverse Vibrations. Torsional Vibrations are generally defined as the Vibrations along its axis of rotation. It is generally linked with the shafts having large masses like crankshaft and needed to be analysed as it also causes Vibrations to certain extent in addition with longitudinal vibrations. Gearbox is one of the major units of the Transmission system of a vehicle, so needed to be analysed on the basis of production of Vibrations during different loading and working conditions which may lead to severe damage and reduction in the lifetime and efficiency of the gearbox. This paper mainly focuses on the Torsional Vibration analysis of the Gearbox of a Vehicle by modal analysis, using this methodology this paper aims to study the vibrations during changing and working of gears in their different orientations and generating characteristic curves of amplitude ratios to the gear ratios and angular frequencies. Vibration at different RPM of engine and by simplification of the gearbox will be calculated at different natural frequencies. Solution for the reduction of the Vibrations in the Gearbox will be also discussed in brief manner. Damping solution may be also searched with proper material and design procedure selection. Optimization of the parameters should be conducted to find out the design natural frequency.

Index Terms: **-** Amplitude ratios, Mode shapes, Natural frequencies, Torsional vibrations.

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1. INTRODUCTION

1.1 Torsional Vibrations

The concept of Torsional Vibrations is derived from the word Torque, which is the tendency of the force to rotate the object around the axis or pivot. As the torque is created the object starts rotating with a velocity and the centrifugal force starts acting on the surface which leads to the object centre of mass to disturbed and starts vibrating along direction of the axis of the shaft or gear. Torsional Vibrations also leads to the deviation in the desired results. Torsional Vibrations are generally studied for the design of the crank shaft of the internal combustion engines, if this type of vibration not studied and analysed properly may lead to the shear-off the flywheel or breakage of the gear and chain drives also. Xu X [1] et al. discussed about the elastic deformation of the meshed gear system and the safe conditions have been utilised in this study. Tian S [2] et al. briefed about the mini Vehicle transmission and developed a meshed gear system and conducted abstract studies on it. An Sun Lee [3] et al. explained in his publication about the coupled torsional vibrations characteristics of a speed increasing gear rotor bearing system, he generated a comprehensive model of coupled meshed gear system and generated a solution for the working conditions. Zhu C. [4] et al. worked on the small inclinations on the meshed gear systems in a gearbox. In this research a simple model is extracted from the previous models suitable for the designs that also within the safety limits.

The vehicles running on roads are having different gearbox arrangements and different gear ratios also, so analysis is required to be done for each and every set of arrangements and the optimum properties are needed to be installed, in this paper we will discuss the arrangement of the Chevrolet corvette 2005 model. By fixing the system we can analyse the torsional vibration properties, the most common way to measure torsional vibration is the approach of using equidistant pulses over one shaft revolution. Dedicated shaft encoders as well as gear tooth pick up transducers (induction, hall-effect, variable reluctance etc.) can generate these pulses. The resulting encoder pulse train is converted

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into either a digital rpm reading or a voltage proportional to the rpm.

The use of a dual-beam laser is another technique that is used to measure torsional vibrations. The operation of the dual-beam laser is based on the difference in reflection frequency of two perfectly aligned beams pointing at different points on a shaft. Despite its specific advantages, this method yields a limited frequency range, requires line-of-sight from the part to the laser, and represents multiple lasers in case several measurement points need to be measured in parallel. The impact of longitudinal vibrations is not considered in this study just to simply depict the sole impact of torsion vibration on the system under loading.

Here theoretical approach is used to solve the different specifications and arrangements by converting the gearbox system into simple gear mesh arrangement and by converting them to two rotor system and considering the transverse vibrations by converting the shaft to a spring of known stiffness.

By this study we will be able to find out the natural frequencies of the rotating members to analyse the mode shapes and the amplitude ratios of the system.

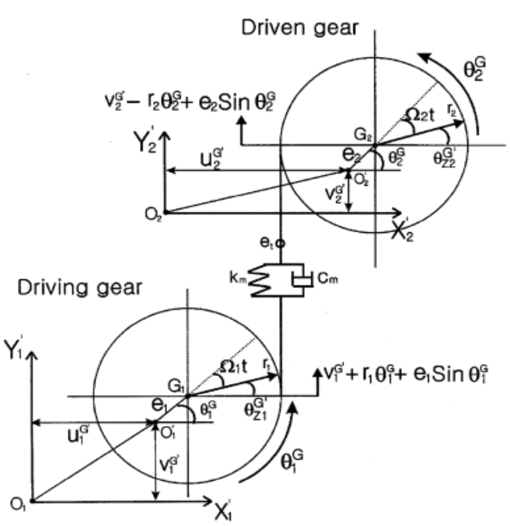

Fig.1 Co-ordinate systems of a gear pair at the pressure line.

1.2 AMPLITUDE RATIO

Amplitude ratio is defined as the ratio of the amplitudes at the two different natural frequencies. This is important to find out the behaviour of the meshed gear system to generate the characteristic curves at different values of the speed reduction ratios and at different shaft lengths for these set of readings particularly(in this paper). The lower it becomes in value the more vulnerable the system becomes to the resonance conditions, at resonance the maximum vibration and transmissibility ratio is observed so to avoid this scenario the spring stiffness and damping factor is chosen accordingly.

Methods and calculations

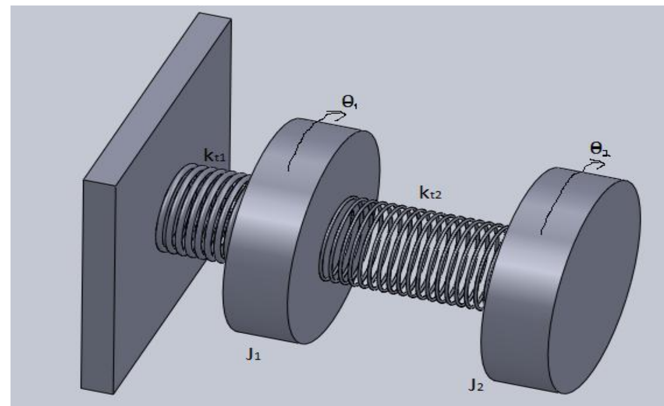

Fig. 2 Showing the torsion springs connected with fixed mass and rotating members.

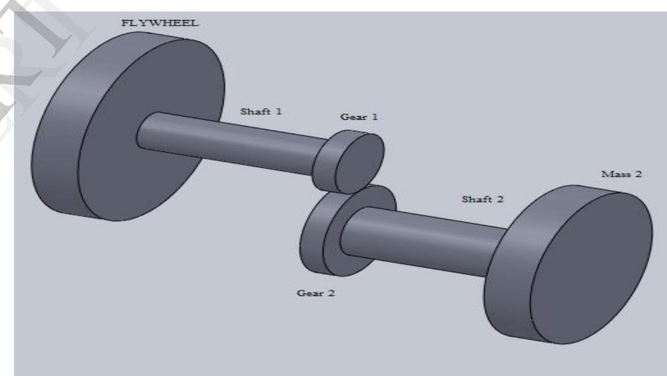

Fig. 3 Showing the meshed gear with the Flywheel and fixed mass at other end.

Calculations are conducted by combining the two individual gears with each other and calculating the equivalent mass moment of inertias according to the gear ratios, the natural frequencies are calculated hence the mode shapes are determined. Similar calculations may be conducted to achieve the desired results for the various working conditions and for different gearbox arrangements. In this paper the gear ratios are taken of Corvette 2005 model for the reference.

By considering $k_3=0$, $k_1 = k_{t1}$, $k_2 = k_{t2}$, $m_1= J_1$, $m_2 = J_2$.

ω_1, ω_2 Natural Frequencies

k_{t1}, k_{t2} Torsional stiffness

J_1, J_2 Rotational mass moment of Inertia

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Θ_1/Θ_2 Amplitude Ratio

The torsional equation of motion of the single stage system is

$$[J]\{\ddot{\Theta}\} + [C_T]\{\dot{\Theta}\} + [k_T]\{\Theta\} = \{F_T(t)\} + \{F_{GT}(t)\}$$

Deriving and reducing the matrix form to simple form we get the values for the natural frequencies, equivalent mass moment of inertias and amplitude ratios for the torsional vibrations is generated.

$$\omega_1^2, \omega_2^2 = 1/2\{(k_{t1} + k_{t2})/J_1 + k_{t2}/J_2\} \pm [[1/2\{(k_{t1} + k_{t2})/J_1 + k_{t2}/J_2\}]^2 - k_{t1}*k_{t2}/J_1J_2]$$

$$r_1 = \{-J_1\omega_1^2 + (k_{t1} + k_{t2})\}/k_{t2}$$

$$r_2 = \{-J_1\omega_2^2 + (k_{t1} + k_{t2})\}/k_{t2}$$

$$\{\Theta_1/\Theta_2\}_{(1)} = \{1/r_1\}$$

$$\{\Theta_1/\Theta_2\}_{(2)} = \{1/r_2\}$$

Results and Discussions :-

The calculations according to the specifications were performed and the results were drawn and plotted between the required parameters which in this case are the amplitude ratio and frequency.

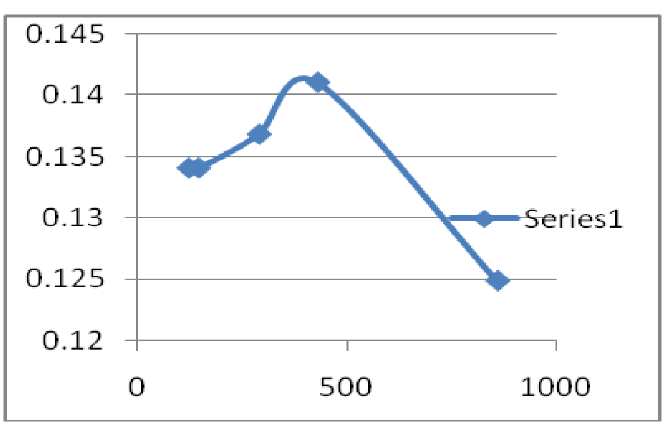

Fig.4 At shaft length 0.2m plot of ω_1 with amplitude ratio.

The above image depicts the relation of ω_1 to the ampiltude ratio when the shaft length is taken as 20 cms. with the diameter taken as 8 cms. for the shaft joining flywheel to the gear 1, the diameter of the shaft joining the gear 2 and fixed mass of the rest gears is taken as 6 cms. and the mass is adjusted accordingly with the balance of the moment and mass along the bearings the mass is found out. In this we observe that as the natural frequency increases the amplitude ratio increases and then again decreases.

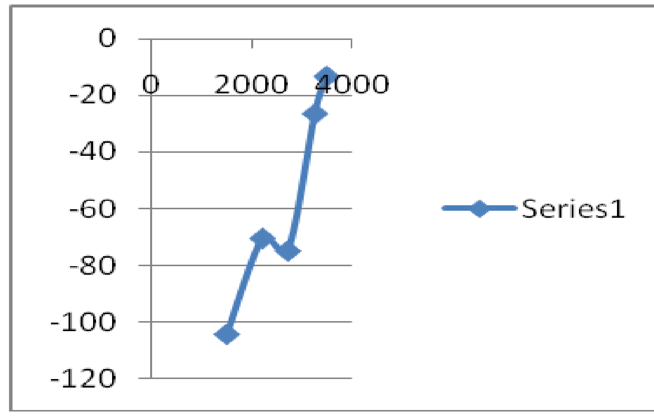

Fig.5 At shaft length 0.2m plot of ω_2 with amplitude ratio.

This graph shows the relation between the natural frequency(ω_2) and amplitude ratio at shaft length 20 cms. and diameter same as the previous case, it shows a decreement in the absolute value of the amplitude ratio, with a small fluctuation in between the middle of the curve.

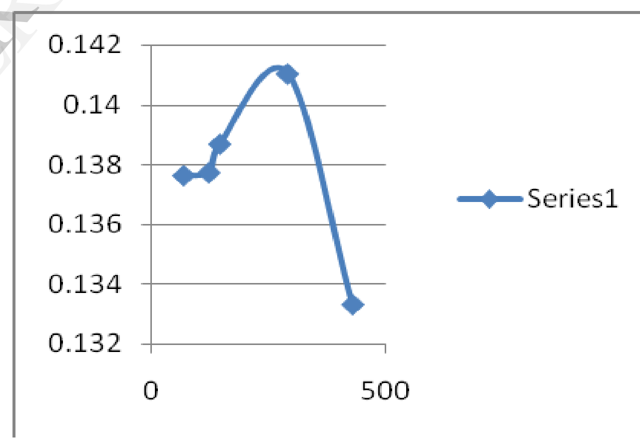

Fig.6 At shaft length 0.8m plot of ω_1 with amplitude ratio.

The above image shows the observations at shaft length 80 cms and the diameter of the shaft is kept constant as the previous case, this conditon is observed in heavy vehicles where the engine size is considerably larger than other engines. The pattern observed between ω_1 to the ampiltude ratio for this shaft length quite resembles that of at the shaft length 0.2m length but with a steep fall after the increment and peak of the graph, this represents the amplitude ratio at near the resonance conditions when the vehicle's engine will produce higher amount of torsional vibrations in comparison of other values of natural frequencies. This condition should be avoided for smooth performance of the engine.

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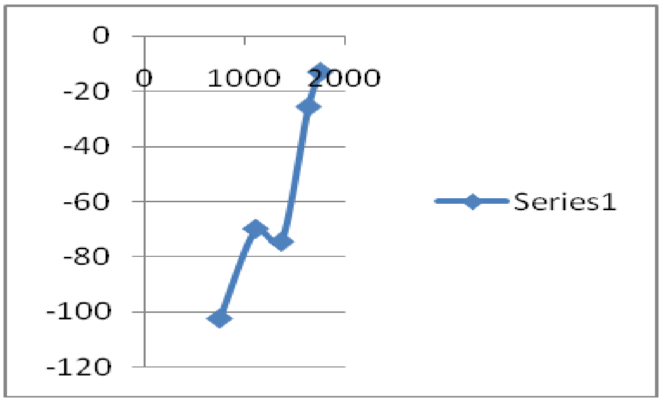

Fig.7 At shaft length 0.8m plot of ω_2 with amplitude ratio.

The above plot shows the relation between ω_2 and amplitude ratio at shaft length 80 cms, this is similar to the 20 cms shaft length observations with steep reduction in the absolute value of the amplitude ratio.

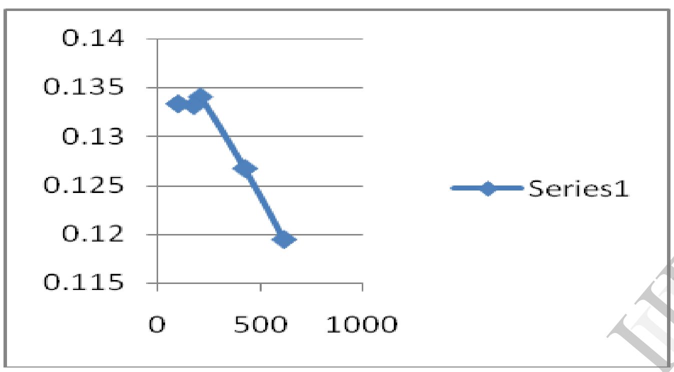

Fig.8 At shaft length 0.4m plot of ω_1 with amplitude ratio

The above plot shows the relation of ω_1 and amplitude ratios, a similar observations has been recorded for this but continous decreement in the amplitude ratio with increase in the natural frequency.

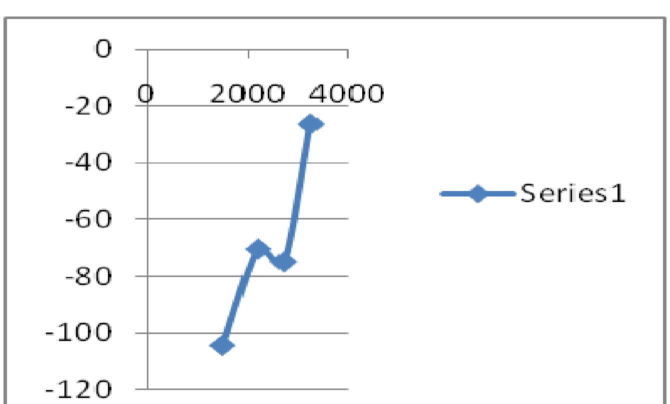

Fig.9 At shaft length 0.4m plot of ω_2 with amplitude ratio

The above plot shows the relation between ω_2 and amplitude ratio at the shaft length 40 cms, in this curve we observed the same pattern as the rest of the observation between ω_2 and amplitude ratios at different shaft lengths.

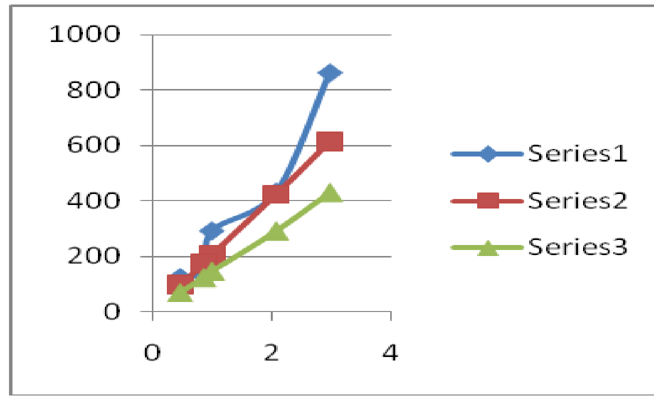

Fig.10 shows the relationship of the gear ratios to the natural frequency(ω_1) at different shaft lengths.

The above curve shows the relation between the Gear ratios and natural frequencies (ω_1) at differnent shaft lengths series 1 shows the relation at 80 cms shaft length, series 2 shows the realtion at 40 cms shaft length and series 3 shows the relation at 20 cms shaft length, this shows that the natural frequncy increases with increase in the shaft length and with the speed reduction ratio also the similar behaviour is observed.

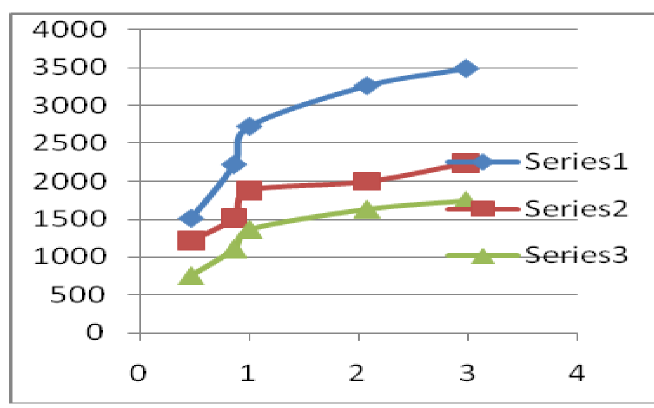

Fig.11 shows the relationship of the gear ratios to the natural frequency(ω_2) at different shaft lengths.

The above graph depicts the realtion between the Different gear ratios and the observed natural frequencies at different shaft lengths, series 1 represents the 0.2m shaft length series 2 showing the characteristics of shaft length 0.4m and series 3 represents 0.8m shaft length, observations from the above image shows that when the shaft length increases keeping the diameter constant, the natural frequency reduces and the chances of the resonance increases. From the above image it can also be shown that when the speed reduction ratio nears unity there is a sudden change in the natural frequency. Gao

[6] et al. discussed about the torsional energy transient model and presented results for the variations against time, but this paper maily aimed at the relation between the natural frequencies and the amplitude ratios showing that torsional vibration intensities at different frequencies.

Conclusion

By these calculations we calculated and plotted the results for the different values of the natural frequencies and amplitude ratios, a relation between gear ratios and amplitude ratio has been also generated for the specific engine configuration with an accuracy of ± 8 percent.

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