Torsional Vibration Analysis of Crank Train and Design of Damper for High Power Diesel Engines Used in AFV

N. Venkateswaran, K. Balasubramaniyan, R. Murugesan, and S. Ramesh

Abstract This paper analysed the torsional vibration of a 12-cylinder V-engine used for tracked vehicles and design of a suitable damper to reduce the amplitude of vibration. The entire engine crank train was modelled analytically, and mass properties are calculated using CREO software. The crank train was analysed using ABAQUS software for calculating the natural frequency of the engine. The results of the analysis were compared and validated using Holzer's method. Suitable damper characteristics were designed using ABAQUS and MATLAB software to reduce the amplitude of vibration. Physical dimensions such as damper inertia, damper size, damping ratio and damping coefficient of the damping medium were determined to achieve the designed damper characteristics.

Keywords Torsional vibration · Damper · Mass properties · ABAQUS · Holzer's method

1 Introduction

The torsional natural frequencies (torsional critical speeds) of any rotating system depend on the torsional dynamics of all other rotating systems coupled with it, i.e., it depends on the torsional stiffness and mass moment of inertias of all the rotating components coupled with the rotating system under consideration, which is the crankshaft. It is very hard to detect the torsional vibration without special

N. Venkateswaran (✉) · R. Murugesan · S. Ramesh
Centre for Engineering Analysis and Design, Combat Vehicles R&D Estt., DRDO, Chennai 600054, India
e-mail: kasven1985@gmail.com

R. Murugesan
e-mail: murugesan.r@cvrde.drdo.in

S. Ramesh
e-mail: ramesh.s@cvrde.drdo.in

K. Balasubramaniyan
Anna University CEG, Chennai 600025, India

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equipment. However, the higher amplitude of vibration can be destructive without adequate damping. Therefore, the prediction of natural frequency is very important in order to reduce or minimize the effect of resonance. Since at the resonance, minimum energy is sufficient to cause the large amplitude of the vibration. Torsional vibration analysis is very important in the design of crankshaft of internal combustion engines because it can cause the failure of crankshaft when the frequency of the vibration matches the torsional resonant frequency of the crank. In this article, the necessary calculations were performed using ABAQUS to determine the natural frequency of the system Initially, the equivalent mass moment of inertia and equivalent torsional stiffness were calculated in order to conduct vibration analysis.

2 Literature Survey

Wu and Chen [1] have presented the natural frequency calculation using Holzer's method for undamped and damped systems. Guangming and Zhengfeng [2] have presented the empirical formula for calculating the torsional stiffness of the crankshaft. Buczek [3] has presented the empirical modelling of multi-cylinder engine. Nestorides [4] has explained about the phase vector summation. Ker Wilson [5] has presented the methodology for carrying out the harmonic analysis. Cagri Cevik et al. [6] has presented the approach to find the stiffness calculation of a crank with static finite element methodology. Xingyu et al. [7] has presented the literature on torsional vibration issue published in recent years, which summarizes on the modelling of torsional vibration, corresponding analysis methods, appropriate measures and torsional vibration control pointing out the problems to be solved in the study and some new research directions. ABAQUS Tutorial guide—Release 6.142, August 2014 [8] presented the methodology for carrying out the modal and torsional vibrational analysis.

3 Analytical Model of Crank Train

The crankshaft to be analysed for studying the requirement of a damper is shown in Fig. 1. Analytical model is preferred for the vibration analysis of crank train since it involves the variation in natural frequency during the revolution of crank train. Therefore, it is computationally very expensive to conduct the quasi-static modal analysis for such a huge crank train mechanism using the FEA method. Hence, it is desirable to simplify the system into an equivalent torsional rotor system with associated stiffness. Each piston assembly and flywheel was considered as an equivalent rotor. Therefore, the crank train was modelled as thirteen degrees of freedom equivalent spring rotor system.

The equivalent piston and connecting rod mass are assumed concentrated at the crank pin centres. Figure 2 shows the division of crank train for equivalent inertia and

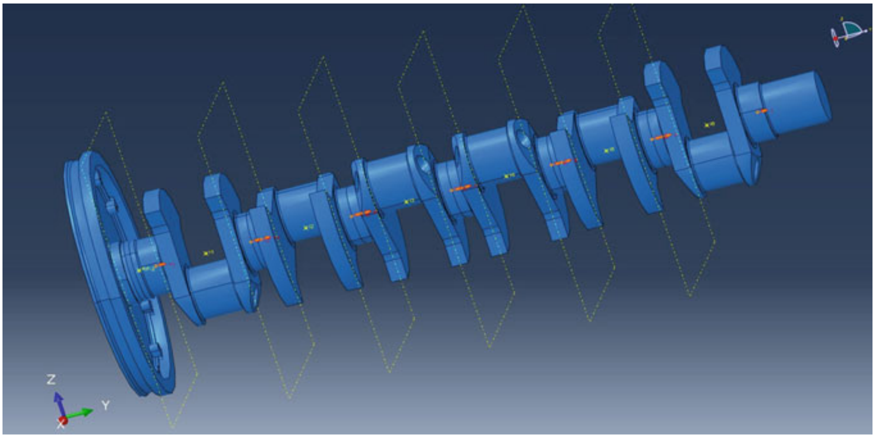


Fig. 1 Solid model of crank train assembly

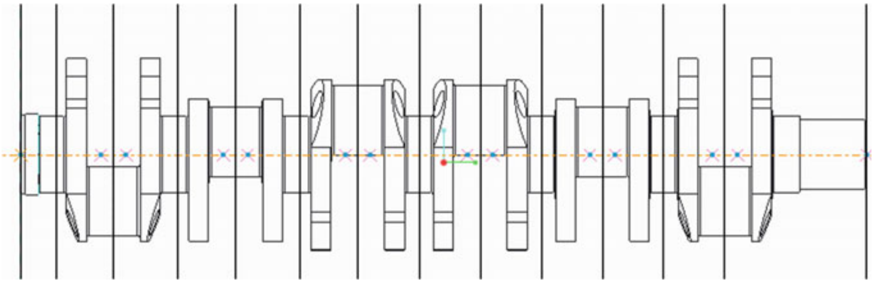


Fig. 2 Division of crank train for analytical modelling

equivalent stiffness calculation. Each piston assembly has half journal shaft, a crank web, half crank pin, connecting rod and piston assembly. The crankshaft portion between straight lines considered for equivalent inertia calculation and the length of each crankshaft portion between cylinder centers is considered as an equivalent length for the calculation of equivalent stiffness of the spring, which is connecting the equivalent rotors.

4 Natural Frequency and Critical Speeds

The equivalent inertia of each rotor was lumped at 13-points corresponding to their position. The points were connected by torsional springs, and the stiffness of each spring was assigned according to the calculated value of equivalent stiffness. Then the modal analysis was performed, and the natural frequency and corresponding mode shapes were extracted. The equation of motion was solved using Holzer's method in

Table 1 Natural frequency

S. No	Natural frequency using ABAQUS (Hz)	Natural frequency by Holzer method (Hz)
1	0	0
2	201.48	201
3	502.42	502

order to verify the ABAQUS results. Table 1 shows the first three natural frequencies. The first natural frequency of crank train (201.48 Hz) in rpm is 12060 rpm and the maximum speed of the engine is 3200 rpm.

In any torsional vibration, analysis of an engine system, order number and multiples of order number of the engine is an important factor to be considered. The Order number is defined as the number of disturbing pulses per revolution of the crankshaft. The order number is also defined as a harmonic number divided by two in case of four-stroke engine and order number equal to the harmonic number in case of two-stroke engine. A four-stroke engine has power stroke once in every two revolutions, which means the order numbers are ½ and multiples of ½. Critical speed is the speed at which the excitation frequency coincides with the natural frequency of the system.

Order number and the critical speed is related by

Critical speed = fn/order numbers

where f_n = natural frequency of *n*th mode.

In order to conduct the forced vibration analysis, the determination of excitation torque is very important and it is explained subsequently.

5 Generation of Torque Versus Theta Curve

Simulation of crank train arrangement with a single cylinder is required to generate the torque. It is done using ADAMS/VIEW software. The solid model of engine parts was assembled in order to study the dynamics of crankshaft due to inertia and gas pressure. Single cylinder arrangement is sufficient for analysis since the behaviour of all cylinders is the same. Figure 3 shows the solid model assembly of single cylinder and piston arrangement.

After importing the solid model assembly to ADAMS, the pressure vs crank angle data was given as input to find out the torque as output. The output torque was extended for 12 cylinders and it is shown in Fig. 4. Forced vibration analysis was carried out using the Finite Element Analysis (FEA) method. The amplitude of vibration was found to be 0.5°, which is higher than the allowable limit of 0.3°. In order to reduce the amplitude of vibration, a suitable damper has to be designed. The vibratory work at the resonance solely depends on a specific harmonic component which is exciting the natural frequency. Not all harmonic components are critical

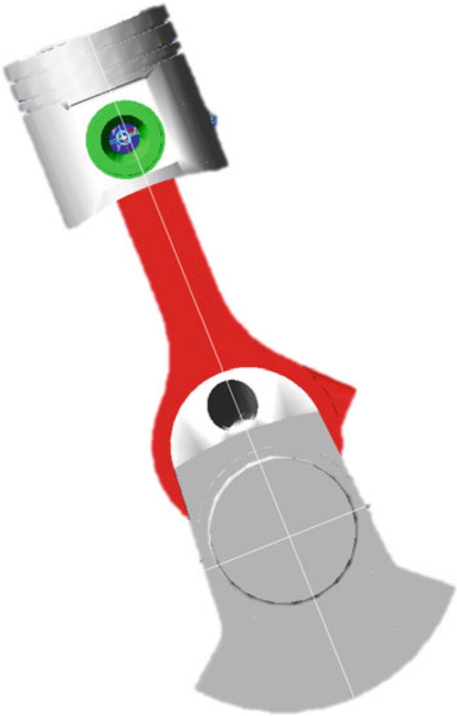


Fig. 3 Single cylinder and piston assembly

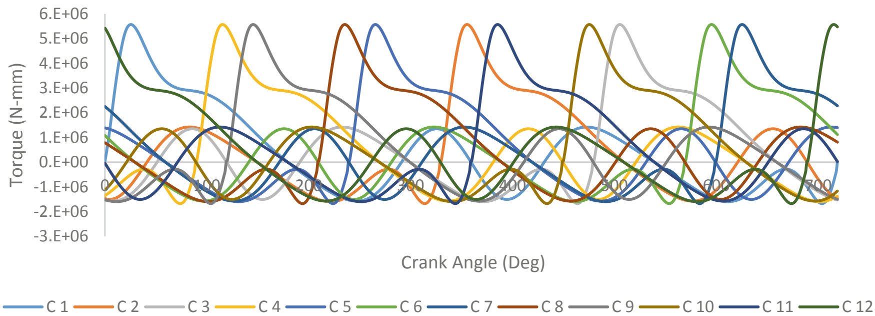


Fig. 4 Excitation torque for 12 cylinder

to the engine. Depending on the firing order and phase between the harmonics of individual cylinder, the torque is getting multiplied or cancelled with each other. Hence, it is required to carry out the signal processing to find out the harmonics of excitation torque.

6 Fast Fourier Transformation

In order to carry out the forced vibration analysis, the harmonic components of excitation torque corresponding to various orders are required. A harmonic analysis converts a non-harmonic tangential pressure curve of an engine cylinder into harmonic components of tangential pressure by Fourier analysis from which the harmonic components of excitation torque corresponding to various orders can be determined.

Fast Fourier Transformation is carried out using ADAMS/POST PROCESSING software. Table 2 shows the harmonic component of torque, and Fig. 5 shows the graphical representation of harmonic torque with respect to the various orders.

Table 2 Harmonic of torque

Harmonic number	Order number	Harmonic excitation torque (N-m)	Harmonic number	Order number	Harmonic excitation torque (N-m)
1	0.5	1841	11	5.5	288
2	1	2119	12	6	247
3	1.5	1290	13	6.5	213
4	2	1056	14	7	185
5	2.5	325	15	7.5	160
6	3	705	16	8	138
7	3.5	500	17	8.5	123
8	4	465	18	9	105
9	4.5	400	19	9.5	94
10	5	337	20	10	82

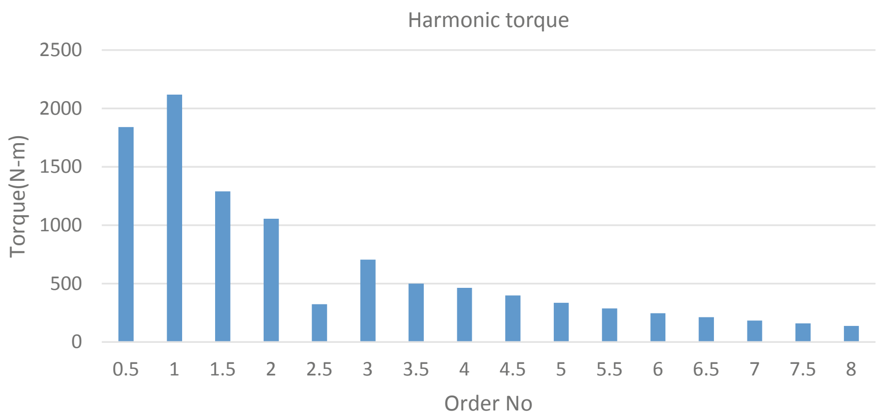


Fig. 5 Graphical representation of harmonic torque with respect to the various orders

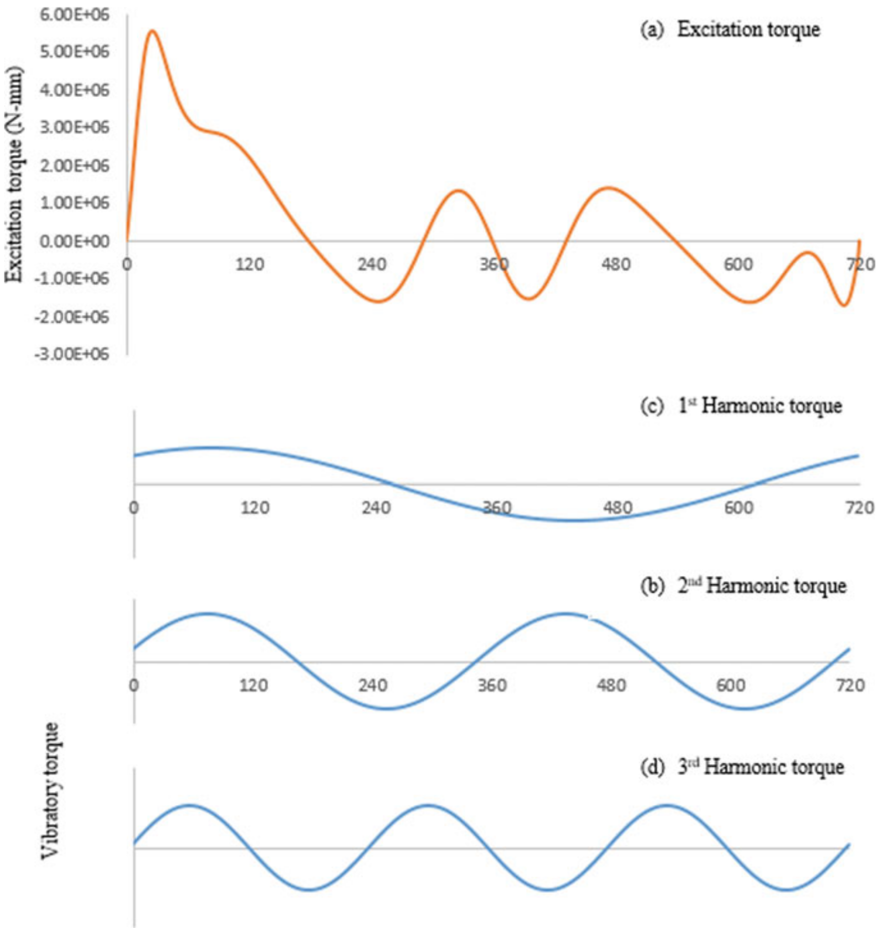


Fig. 6 Excitation torque and harmonics of excitation torque vs crank angle

Figure 6 shows the graphs of excitation torque and some of its harmonic components (up to 3rd harmonic) of excitation torque vs crank angle.

7 Critical Order

Fourier analysis finds out the harmonic components of excitation torque corresponding to various orders. Not all the orders are critical to engine, since some of the torque vectors will be cancelled and some of the torque vectors will be added. These orders in which the vectorial torque addition is critical to the engine, the excitation source for torsional vibration is mainly due to harmonics of gas pressure and inertia torque. Hence, the phasing between the harmonic torque components of individual cylinder and firing order is very important. The phase angle of the torque vectors depends on the phase of the harmonics relative to the firing positions of their respective cranks. Phase vector sum depends on two factors, namely firing sequence and angles between the various cranks. Figure 7 shows the phasing diagrams for various orders. It is found that both 4.5 and 12th order are very critical to the engine. Then the resultant torque to be calculated. Figure 8 shows the graphical representation of resultant torque with respect to various order. It is understood that the resultant torque obtained for the 12th order is negligible for this engine configuration.

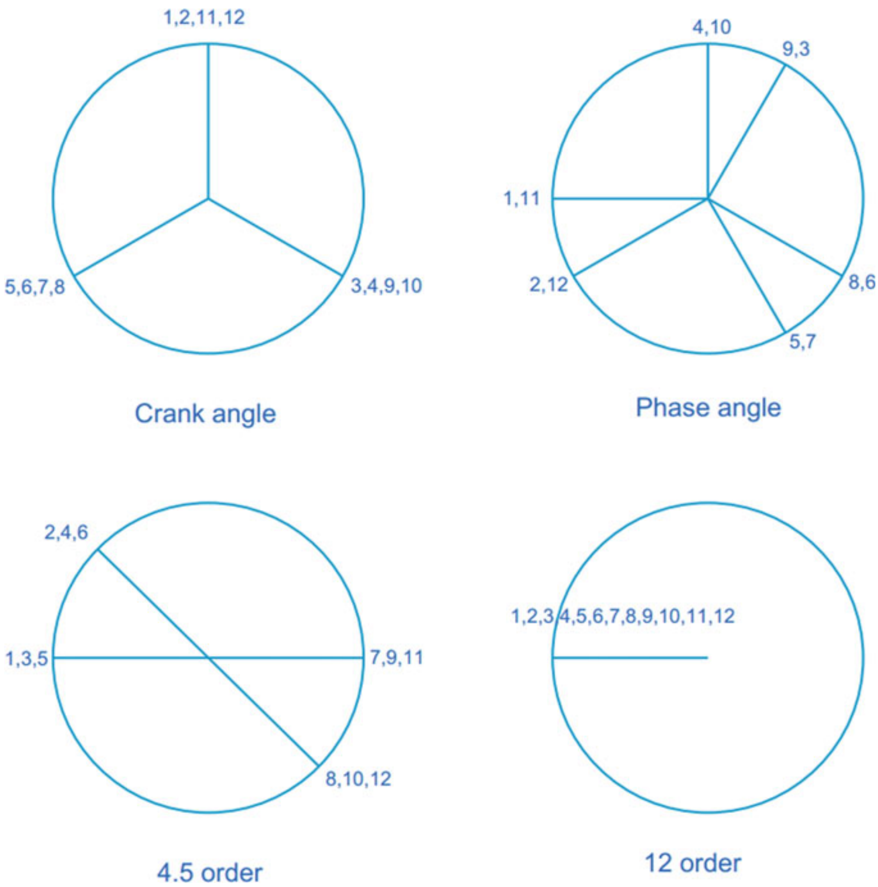


Fig. 7 Phasing diagrams for various orders

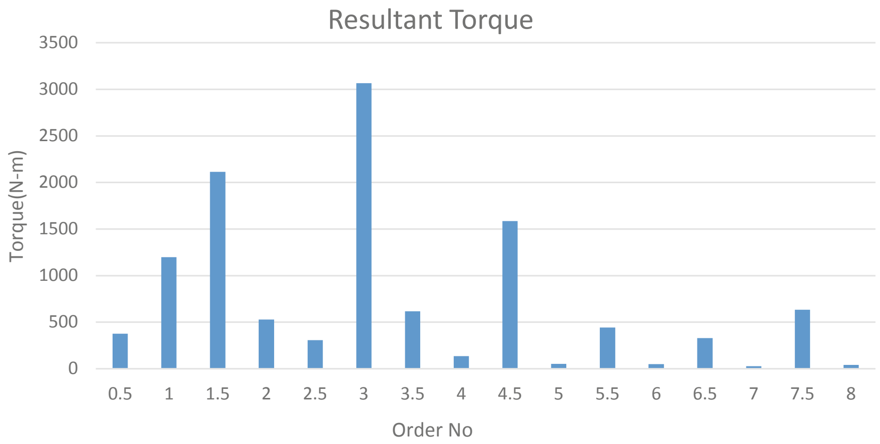
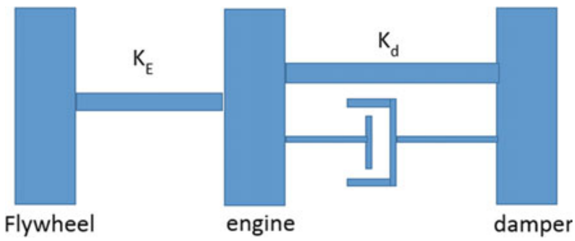


Fig. 8 Graphical representation of resultant torque with respect to various order

8 Design of Damper

It is essential to control the amplitude of vibration in the crankshaft in order to ensure the safe operation of the engine near the critical speeds of the engine. Theoretically, the vibration amplitude reaches infinity and practically to a very high value in resonance condition. This may lead to the failure of the crankshaft. Dampers are the devices used to control the effects of the torsional vibration. Dampers of this selected type consist of an annular seismic mass enclosed in a casing. The peripheral and lateral gaps between these two members are filled with a viscous fluid. The damper is untuned since there is no elastic member between seismic mass and the casing; only the viscous torque transmitted by the fluid acts upon the seismic mass. Figure 9 shows the 3-rotor system consisting of Flywheel, engine and damper. The damper consists of free mass, which is keyed to the shaft. Normally, the disc rotates at the shaft speed owning to the viscous drag of oil between the disc and the case. However, if the shaft vibrates torsionally, the viscous action of the oil between the disc and casing gives a damping action. In order to calculate the optimum-damping ratio, the crank train was modelled as a 3 DOF spring rotor system. The damper is to be designed for the more critical order, which is nothing but 4.5 order.

Fig. 9 Three rotor system consisting of Flywheel, Engine and Damper



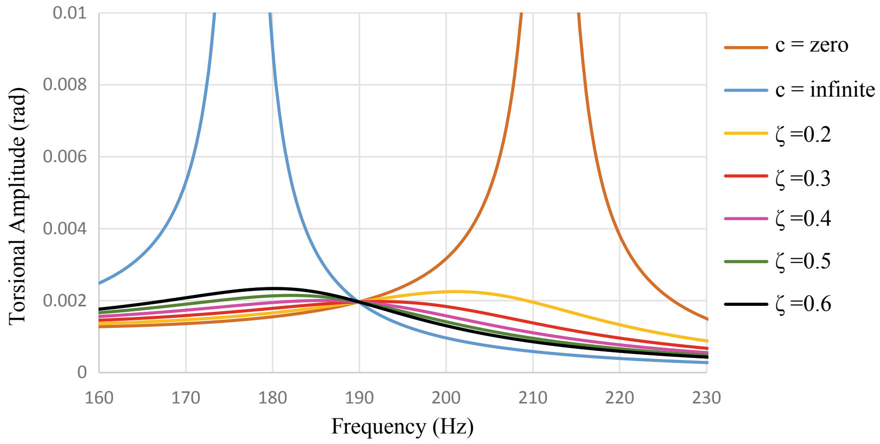


Fig. 10 Optimum damper characteristics

Table 3 Technical parameters of damper

Damper type	Torsional untuned viscous damper
Damper stub shaft stiffness	4.177 + 9 N-mm/rad
Inertia of damper	Jd = 552 N mm s2
Optimum-damping coefficient	430503.75 N-mm-s/rad

To design a suitable damper, the forced vibration analysis was carried out for different damping ratios. Figure 10 shows the optimum damper characteristics. In this, the intersection point of zero and infinite damping curve is called as optimum point. For all value of the damping coefficient, the curve will pass through this point. The damping coefficient will be called as optimum when maxima of the Frequency response curve lies on the optimum point. In order to find optimum-damping coefficient for viscous damper, the system was modelled as a 4 DOF spring rotor system. Table 3 shows the optimum damper characteristics viz., Inertia, stiffness and the damping coefficient.

9 Conclusion

In the present study, the torsional vibration analysis was carried out for a 12-cylinder V-engine of an armoured fighting vehicle (AFV). The critical speeds were estimated by modal analysis using ABAQUS software. From the analysis, it has been concluded that there are many critical speeds within the operating speed range of the engine. The harmonic analysis of excitation torque was carried out, and harmonic components of torque were found. From the forced vibration analysis, it is found out that the amplitude of vibrations in the crank train exceeds the permissible limits within the operating speed range. In order to control the effects of critical orders and to minimize the amplitude of vibration, a suitable damper has been selected and introduced in the crank train. Subsequently damper characteristics were designed.

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