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Computerized Analysis of Automobile Crankshaft with novel aspects of Inertial analysis

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Abstract: *Torsional vibration analysis of crankshafts of internal combustion engines such as automobile and especially marine applications is of tremendous importance. A lot of failures of crankshafts are found to be due to torsional vibrations. It is extremely important to predict the natural frequencies of crankshafts as the resonance phenomenon can be destructive. In literature the determination of natural frequencies for multi-cylinder engines has been done by discrete methods. These methods typically involve a multi rotor approach on a shaft and commonly used methods are by Holzer and Stodola. Other methods like influence coefficient matrix are also used. With the development of finite element method full 3-D development of solid model has been very effective way to consider the actual geometry of the system and predict the natural frequencies. In theory some analytical methods for prediction of natural frequencies considering the well-known variable moment of inertia effect have been developed and secondary resonance frequencies have also been noted. Here in this paper, the main aim is to predict the natural frequencies for crankshaft of a multi-cylinder (3 Cylinder) automobile engine by using finite element method and to experimentally verify the natural frequencies. Value from Finite Element Analysis deviates from experimental value and theoretical value by 4%.*

Keywords*: Finite element analysis, torsional vibrations, crankshaft, engine.*

I. INTRODUCTION

The traditional and convenient way to analyze the mechanics of crankshaft is to treat the object as having a fixed moment of inertia and from that the expressions of velocity and acceleration are defined or analyzed by either analytical methods such as vector mechanics or complex algebra or graphical methods such as Klien's construction etc. However we know that a vertical engine has gravity assisted motion for the downward stroke and antigravity movement in the upward stroke. This means that the rotational motion has two distinct speeds or frequencies and further the moment of inertia changes due to the motion not being exactly reciprocating or rotary but rather a combination of the two .

The present work is aimed at determining the natural frequencies of an automobile crankshaft using various analytical and experimental methods and then considerations to variable inertia will be given in next phase of the work. The failures of crank shaft assemblies because of excessive torsional stresses have long been reported in the literature. Many different factors were attributed to these failures. One of the important attributes in case of few marine engines as reported by authors was variable inertia effect. All reciprocating engines have periodically fluctuating inertia function which leads to variation in natural frequencies. This would further lead to a nonlinear frequency coupling between the torsional natural frequencies of the system and the rotational speed and their harmonics. This is generally referred to as Secondary Resonance. Majority of the researchers analyzing the torsional vibrations of the engines have neglected fluctuating inertia which would result in linear torsional vibration analysis. When nonlinear parameter like variable engine inertia is considered, they would result in nonlinear vibration analysis. When variable inertia effect is considered for analysis, this would result in nonlinear frequency modulation that takes place between the torsional natural frequencies and the engine rotational speed and their harmonic components. The increased power density of vehicle engines results in the enhancement of the excitation torque within the broad engine speed range and causes serious torsional vibration problems for the crankshaft assembly. Assuming the dynamic parameters are constant is one of the main causes which lead to errors in the experimental values with the theoretical considerations. The displacement and angular velocity of reciprocating components (the piston and small end of the connecting rod) vary with crankshaft motion, and maintain an intricate functional relationship with the crank angle, which engenders non constant inertia for the crankshaft assembly. Material nonlinearity, nonlinear friction of the piston liner, and oil film within bearings result in nonlinear stiffness and damping. Furthermore, the torsional vibration absorber and elastic coupling show stronger nonlinearities. Normally, nonlinear parameters are linearized in practical modeling, which can inevitably induce model errors. Therefore, establishing accurate expressions for nonlinear dynamic parameters is critical. Dynamic modeling methods include the system matrix method, finite element method (FEM), and multi-body dynamic method (MBD). Traditional lumped mass modeling adopting the system matrix method has been widely used. Comparison between theoretical calculations and experimental evaluation indicates that high errors exist in the theoretical calculations, which can reach 15 % at low speed.

II. LITERATURE SURVEY

A lot of work has been done in this field and here we mention some of the work here. It would be impossible to give a full coverage of the crankshaft vibration analysis due to limitations of space and time.

Characteristics analysis of non-linear torsional vibration in engine and generator shafting system presented by Wei Zhang, Weinming Zhang, Xuan Zhao, Miaomiao Guo [3] was based on body structural vibration and noise in motorized wheel vehicle, where the engine and the generator are connected directly. A mathematical model was used to solve the problem caused non-linear torsional vibration. The mathematical model reflects the effects of the dynamic performances, structural parameters and electromagnetic parameters of the shafting system on shafting torsional vibration. Meanwhile electromechanical coupling term reflects the intrinsic link between the three. The model reflects the shafting dynamic performances and vibration law.

Non-linear torsional vibration characteristics of an internal combustion engine crankshaft assembly presented by Huang Ying, Yang Shouping, Zhang fujun, Zhao Changlu, Ling Qiang and Wang Haiyan [2] in which they concluded that the piston pin has little effect on the instantaneous inertia of the crankshaft assembly, the phase difference between the instantaneous inertia and additional damping results in a non-harmonic forced response, the angular displacement amplitude of the non-constant inertia model is much higher at high orders and with increased reciprocating inertia torque, the rolling vibration decreases for low harmonic orders, while the trends of torsional vibration increase. Non-linear torsional vibration analysis of variable inertia reciprocating engines presented by Hameed D. Lafta, Akram H. Shather [1] used fourier analysis combined with non-linear torsional vibration governing differential equation obtained by Lagrange's equation examined by developing a computer program using MATLAB/Simulink concluded that with increasing the inertia ratio of connecting rod, the system torsional vibration is no longer within expectable values, and an attention should be given for the connecting rod part of reciprocating masses than other parts.

Finite element analysis of 4-Cylinder diesel engine crankshaft presented by Jian Meng, Yongqi Liu, Ruixiang Liu [5] used the FEA tools to carry out the stress and modal analysis of the 4-Cylinder crankshaft. They concluded that the maximum deformation appears at the center of crankpin neck surface. The crankshaft deformation was mainly bending deformation under the lower frequency. And the maximum deformation was located at the link between main bearing journal and crankpin and crank cheeks. The resonance vibration of system can be avoided effectively by appropriate structure design. The results provide a theoretical basis to optimize the design and fatigue life calculation.

Crankshaft strength analysis using finite element method presented by Momin Muhammad Zia Muhammad Idris [6] concluded that strength Analysis is a powerful tool to check adequacy of crankshaft dimensions and find scope for design modification. . Based on Result Analysis, a design modification is proposed. The torsion stress was also included in the analysis. It is found that weakest areas in crankshaft are crankpin fillet and journal fillet. The reduction in mass obtained by design modification is 38%. A dynamic analysis is required to be done for the modified design to study its vibration characteristics.

III. OBJECTIVES

The project concerns with the simulation of nonlinear analysis of crankshaft vibration with variable inertia. It involves the study of variable inertia which comes into picture due to moving parts in IC engine and its impact on estimation of natural frequencies. The main aim is to build an analytical model of the phenomenon and study its impact on frequencies.

IV. FINITE ELEMENT MODEL

The Finite element model was constructed from 3D solid model and the software used was HYPERMESH.
The details of the finite element model are: - No. of nodes: 95678, No. of elements: 70435. Automatic option was used with second order tetra elements for the solid meshing and all quality checks were performed to ensure that there are no bad elements in the model.
A common check in finite element analysis is to make a free natural frequency check to ensure that there are no loose elements in the model and the model passed this test.
The details of the first six natural frequencies are as under:

Sr. No	Frequency Hz	Mode Shape description	Comment
1	0.006841	Translation in Y-direction	Close to zero
2	0.006889	Translation in Z-direction	Close to zero
3	0.006906	Translation in X-direction	Close to zero
4	0.007043	Rotation about Y-axis	Close to zero
5	0.007103	Rotation about Z-axis	Close to zero
6	0.007266	Rotation about X-axis	Close to zero

Thus, the model is a perfect model for further analysis

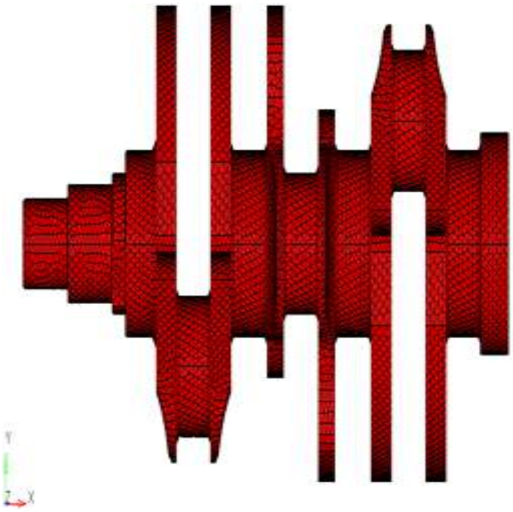

Fig.1 Meshing

Fig. 2 Meshing

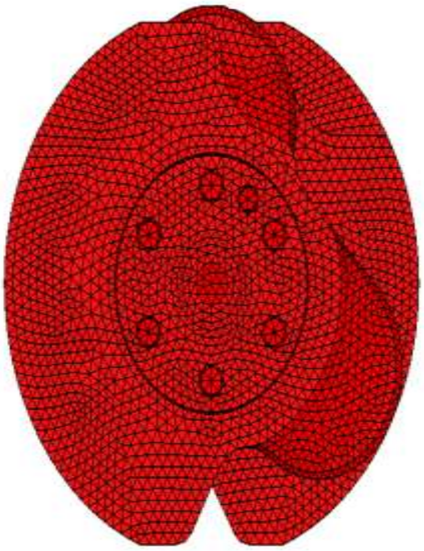

Fig. 3 Meshing

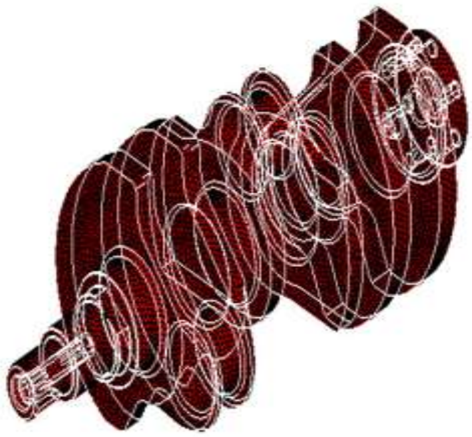

Fig. 4 Details of surfaces and Meshing superimposed

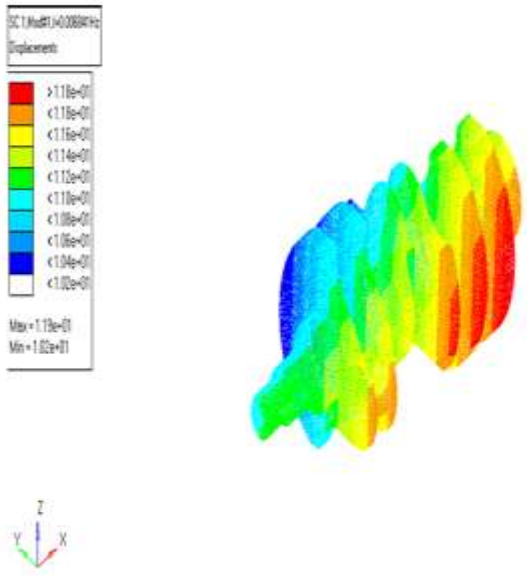

Fig. 5 Mode 1:- Translation in Y direction

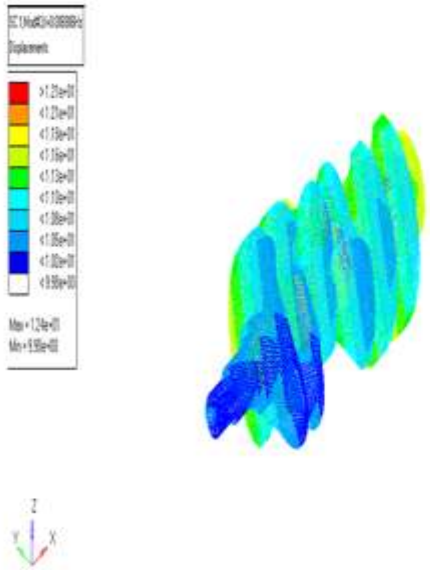

Fig. 7 Mode 3:- Translation in X direction

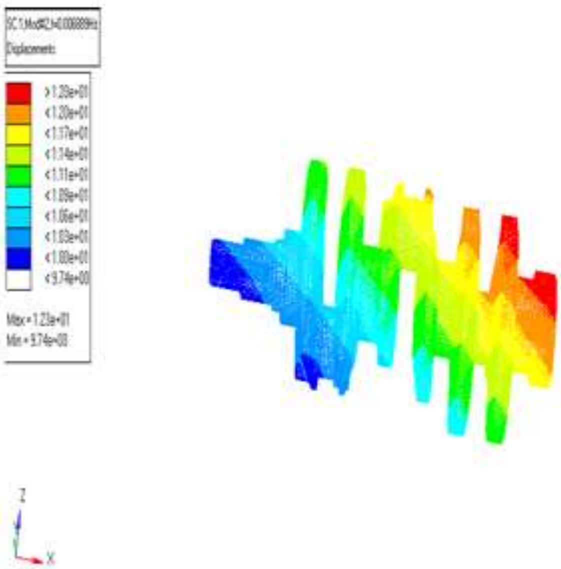

Fig. 6 Mode 2:- Translation in Z direction

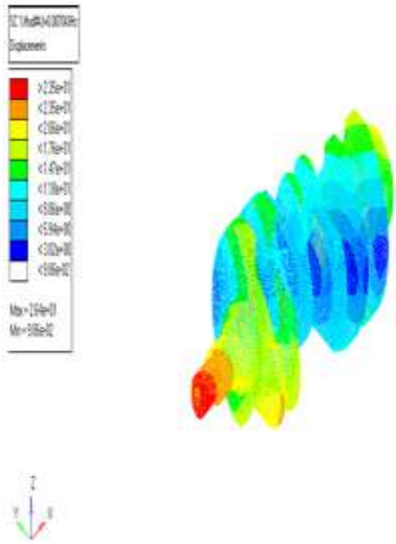

Fig. 8 Mode 4:-Rotation about Y-axis

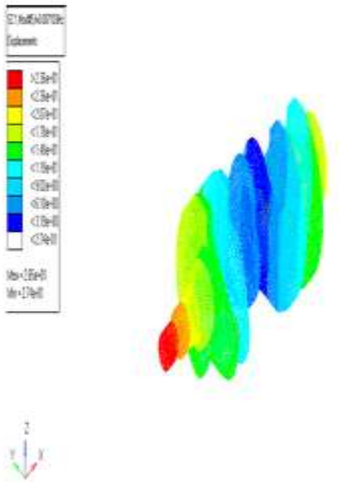

Fig . 9 Mode 5:- Rotation about Z-axis

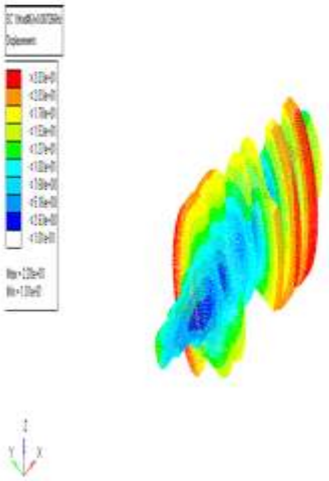

Fig. 10 Mode 6:- Rotation about X-axis

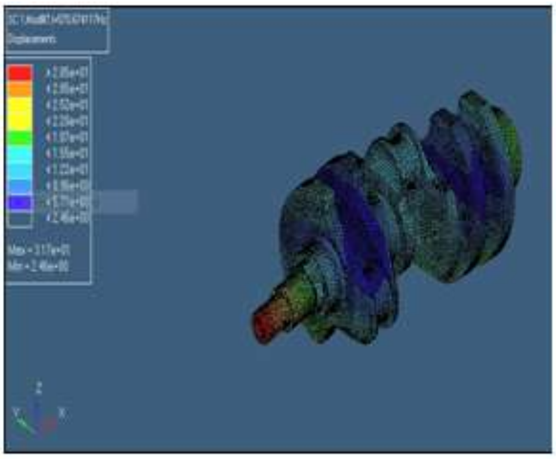

Fig. 11 Mode 7:- Bending in ZX plane

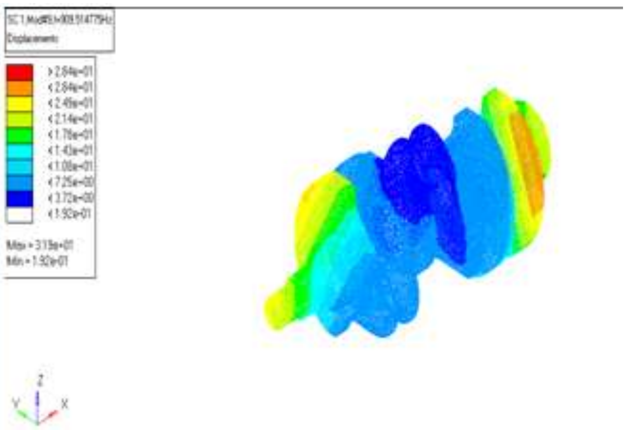

Fig. 12 Mode 8:- Bending in ZX plane

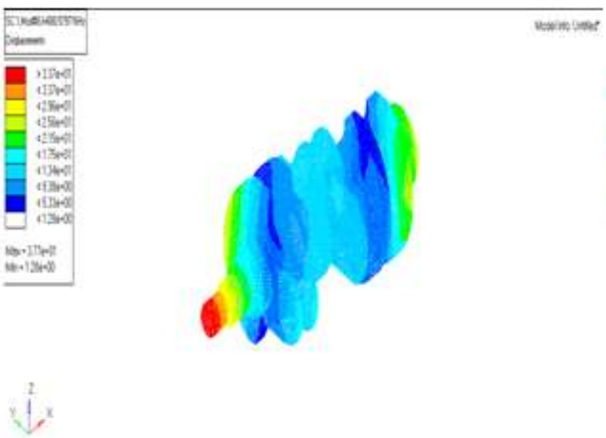

Fig. 12 Mode 8:- Bending in ZX plane

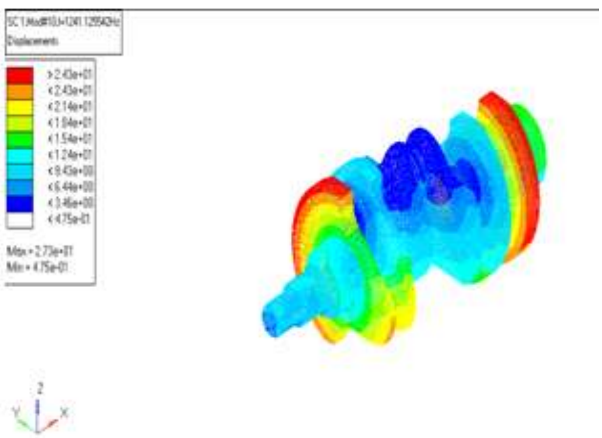

Fig.14 Mode 10:- Torsional deformation about X- axis

Now, we analyze the further deformable modes of the model and then we find that these start from mode shape no. 7 and above. These are classified as bending or torsion or a mix of bending and torsion depending on how the motion animation appears to the eyes.

Since, the interest is in torsional analysis and the first frequency of motion is important, we stop analyzing further modes and note down the natural frequency of torsional vibration as 1241.12 Hz.

V. Concluding Remarks

In this paper we have analyzed an automobile crankshaft with three different approaches and the comparison is as follows:

Sr. No	Method of Analysis	Torsional Vibration frequency	Remark
1	Finite Element Model	1241.12 Hz	Lot of checks have to be done to make sure the model behaves realistic. Requires resources such as computer and software.
2	Discrete Model / Holzer Method	1195 Hz	Simple approach, but difficult to determine the moment of inertia and stiffnesses exactly.
3	Experiment	1290 Hz	Requires a lot of patience as it is very difficult to correlate the modes and compare them with FEA.

From the table, we find that the discrete model under predicts the natural frequency and the experimental approach gives a slight over prediction.

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