Design and Analyis of Balancer Shaft for a Four Stroke Single Cylinder Diesel Engine

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Abstract - **Present Automotive Engines are obligated to have acceptable level of characteristics such as low pollution, low specific fuel consumption, high thermal efficiency, energy saving and passenger comfort. Engine noise, vibration and harshness are key issues in any automotive application. One of the chief role is of the first order unbalanced vibration in single cylinder four stroke engine configuration.**

Lot of work has been done for introduction of two balancer shafts in engine but less work is done including single balancer shaft. This paper describes detailed procedure for the design of single balancer shaft as a solution for the chief source of vibration i.e. first order unbalanced force in stated engine configuration. The paper focuses upon the introduction of balancer shaft in single cylinder engine to improve balancing and determine the exact location, mass and geometry of balancer shaft. Multi body simulation procedure is established and results are validated through experimental measurements. Influence of addition of balancer shaft on crankcase vibration is studied. The effectiveness of integration of balancer shaft is simulated in dynamic analysis software ADAMS. Modal analysis is carried to validate the design of the balancer shaft in ANSYS.

Keywords—Balancer shaft; Primary intertia force; eccentric mass; balancing; counterweight, ANSYS,ADAMS.

I. INTRODUCTION

Each time when there is oscillation of an object, it is bound to vibrate. For preventing vibration of the object, there should be isolation of the source i.e. oscillating mass from other objects. Each time if engine speed multiplies its value by two, the resultant unbalanced force multiplies its value by four. If it is desired to reduce this engine vibration from transferring to the chassis and finally to the passenger; special elastic engine mounts have to be incorporated which will facilitate the absorption of engine vibration. From reliability of each component point of view, control of vibration is essential.

When two different vibration or waves interfere, they can either sum up or cancel each other. If the two waves of same magnitude and in same phase, they add up whereas if the two waves are out of phase, whey there will cancel out each other. This is so called as principle of superposition of waves. Minimizing unsolicited vibration can be achieved with equal and with out of phase vibration. This line of attack methodology can be demonstrated by application of balance shaft in an engine. A balancer shaft is a mechanical unit which limits rotating and reciprocating vibrations by excitation in same magnitude and opposite in phase of harmonic vibration. Balancer shaft are especially designed to eliminate vibration caused due to the excitation caused due to piston and crankshaft rotation. The vertical component of imbalance is reduced to a very low magnitude by the balance shaft providing smooth drive and vibration of wheels. The soft engine mounts provides low magnitude of rocking couple in horizontal direction in the rear wheel drive application for a comfort ride to passenger.

II. LITERATURE REVIEW

Engine balancing is a process of balancing undesirable rotary and reciprocating forces that are produced during normal engine operation in earlier days, lot of study was done in order to reduce vibration in railway and marine application. A harmonic mechanical rotating shaft with respect to the crankshaft, or a balancer shaft, was for the first time introduced by British engineer Lord Frederick Lancester. This invention brought a revolution in the engine vibration and balancing.

David Meek and *Martyn Roberts* [1], described the incorporation of two balancer shaft to four cylinder engine rotating at twice the speed of crankshaft. The resulting system attained balancing of secondary forces up to 92.5% along with packaging and oil drying restrictions. Also, the different favorable location arrangement of the shaft is provided in this paper.

Hirokazu Nakamura [2] calculated the value of vertical forces and rolling moments with the engine displacement of two liters four cylinder configuration. The paper described the reduced level of vibration with use of unique counter balance shafts. These shafts rotates at a speed twice that of the crankshaft and reduces the second order vibration present in the four cylinder configuration.

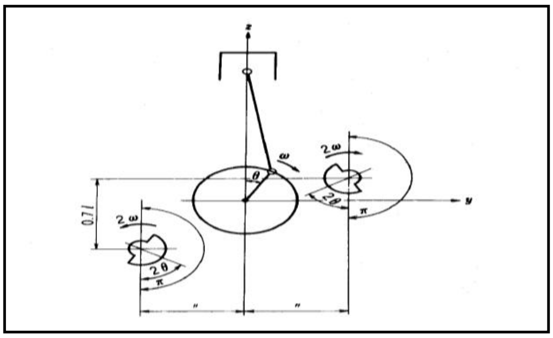

Fig. 01. Unique counter balance shafts

Jonathan Saunders and Patrick Walker [3] described about the novel balancing system providing a cut edge advantage of low cost engine production of single cylinder engine as well as refinement of multi-throw crank configuration. This paper outlines the theory of balancing system applied to single and twin cylinder arrangement. Also, analysis has quantified improvement in refinement of the single or twin cylinder arrangement over conventional outline with small size, light weight, reduced ambiguity and charming alternative to multi throw configuration.

Chan-Jung Kim, Yeon June Kang, Bong Hyun Lee and Hyeong Joon Ahn [4] provided with the strategy to minimize the elastic strain energy and kinematic energy of balance shaft. This paper describes about the bending deformation due to balance shaft and loss of power due to use of balance shaft for specified engine target.

III. DESIGN AND ANALYSIS METHODOLOGY

The engine specification for the selected single cylinder diesel engine are as follows:

Table I: Engine specification

Engine bore	86mm
Engine stroke	75mm
Crank throw	37.5mm
Connecting rod length	118mm
Rated speed	3600rpm

A. Calculating total primary inertial forces

The piston crank assembly can be modeled as slider crank mechanism. The primary inertia force can be calculated as:

$$F = m_{rec} r w^2 \cos\theta$$

m_{rec}: total reciprocating mass; kg
r: crank throw; mm
The maximum value of inertial forces acting at rated speed is 4118.704N.

B. Determination of unbalanced mass on each counterweight

a. In a single cylinder engine, there are no unbalanced moments. To balance the centrifugal inertia forces of rotating masses, two similar counterweights are fitted on the web extension. Their center of gravity are at a distance of γ from crankshaft axis. Hence we have,

$$2m_{cwR}\gamma\omega^2 = m_R R\omega^2$$

Where,
m_{cwR}: Mass required on crankshaft for compensation of rotating imbalance; kg
m_R : Rotating mass on crankshaft; kg.

b. Due to structural constraints, reciprocating unbalance cannot be completely balanced [usually 50% is taken for reciprocating balancing].Hence in order to balance the reciprocating unbalance forces, the mass required on crankshaft are:

$$2m_{cw,j} = 0.5m_j R/\gamma$$

Where,
$m_{cw,j}$: Mass required on crankshaft for compensation of reciprocating imbalance; kg
m_j : Reciprocating mass on crankshaft; kg

c. Thus total mass of each counterweight in a single cylinder engine will be:

$$m_{cw} = m_{cwR} + m_{cw,j} = \frac{R}{2\gamma}[\mathrm{m}_R + 0.5\,\mathrm{m}_j]$$

The value of theoretical total mass of each counterweight at rated speed of 3600rpm is calculated as 0.5793kg.

C. Compensation of this unbalance mass in balancer shaft

Unbalance mass vector is calculated by taking the difference between the theoretical counterweight and actual counterweight and multiplying the net by the center of gravity of crankshaft. Its value is calculated as 2.513kg-mm. This unbalanced value has to be balanced in form of eccentric mass in balancer shaft.

D. Counter shaft design

The following stress acts on the balance shaft:

a. Bending stress due to gear forces acting as well as self-weight.
b. Stress due to torsion and bending moment

The tangential load acting due to gear transmission and force acting due to the eccentric mass is calculated and applied in the horizontal load diagram. The self- weight of the balancer shaft is taken into consideration and the vertical and horizontal bending moments are calculated. The value for k_m and k_t which are factor for suddenly applied load with shocks in bending and twisting moment respectively are taken as three each. The factor of safety is taken as two. Hence the diameter of balancer shaft can be calculated as upon the value twisting moment as,

$$T_e = \sqrt{(\mathrm{k}_m.\mathrm{M})^2 + (\mathrm{k}_t.\mathrm{T})^2}$$

Also,
$$T_e = \frac{\pi}{16}.\frac{\tau}{f_s}.d^3$$

Where,
Te = Equivalent twisting moment; N-mm.
f_s = factor of safety.
d = diameter of balance shaft; mm.

The value of diameter of balance shaft is calculated as 17.6902mm. We take the value as 20mm.

E. Counter shaft gear design

We have the center distance between the axis of rotation of crankshaft and balancer shaft as 96mm. Next step is selecting of module for gear. If high value of module is selected, it increases the problem of backlash in gear. So also, low value of module requires high accuracy which in short increases the machining and quality cost. We take the value of module as 2. As the balancer shaft rotates at a harmonic speed of crankshaft, hence number of teeth on balancer shaft is equal to the teeth on crankshaft. Hence, $z_1 = z_2$

Where, z_1 = number of teeth on crankshaft.
z_2=number of teeth on balancer shaft.

We know that,

$D = (m.\ z)$

$C = m.\ [z_1+z_2] / 2$

Where,

D = pitch circle diameter of gear; mm.

m = module on gear; mm

C = center distance between axes of two shaft; mm

We calculate the number of teeth on gear and pinion as 48 and diameter of both gear as 96mm. Also, the face width[b] is nine times module. Hence, b is calculated as 18mm.Spur gear configuration is used for the gear geometry.

F. *Creation of CAD model of each componenets of engine.*

The CAD modelling of each engine components are done in CATIA V5 environment. The assembly consists of crankshaft, connecting rod, gudgeon pin, piston, balancer shaft and crankcase. All the components are modelled and checked for the mass properties. Fig. 02 shows modelled balancer shaft which is introduced in the single cylinder engine.

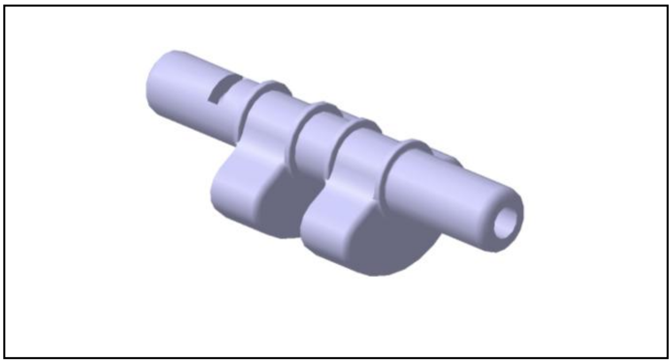

Fig 02: Balancer Shaft modelling

After the successful completion of the CAD modelling of each part, assembly is done as show below along including balance shaft. The weight of the balancer shaft is 0.605kg. Fig. 03 shows the simplified assembly of crankshaft including the balancer shaft.

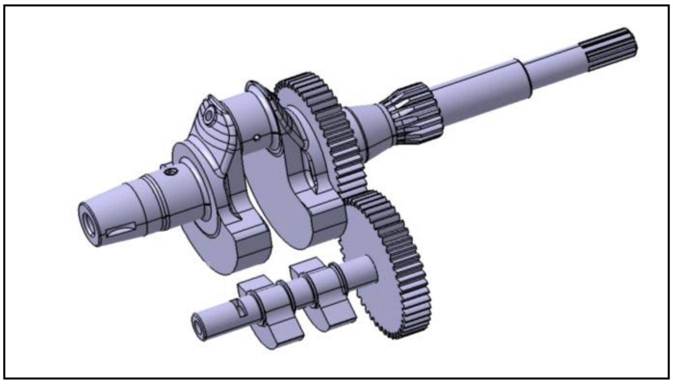

Fig 03: Assembly including Balancer Shaft

G. *Importing each module in dynamic analysis software.*

Each module is individually imported into MSC ADAMS software. The mechanism model is built using different joints at various modules. Two set of mechanism are made. One including no balancer shaft and one with incorporation of balancer shaft.

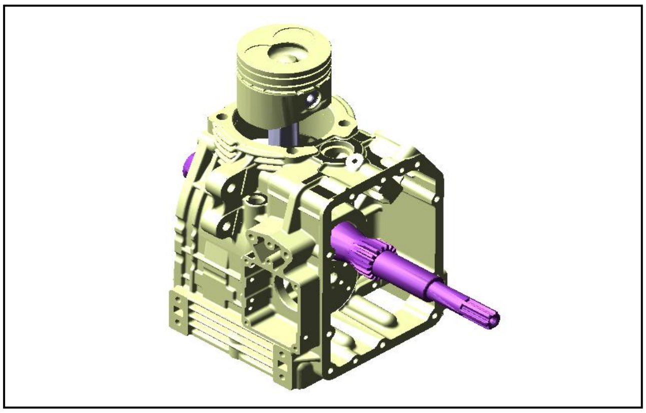

Fig 04: Mechanism including no balancer shaft.

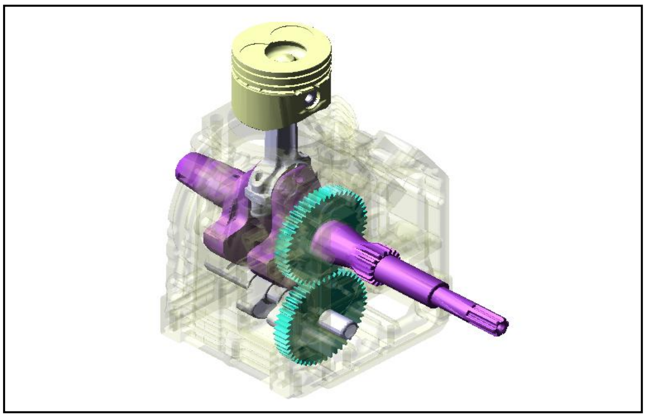

Fig 05: Mechanism with incorporation of balancer shaft

H. *Assignment of mass properties and joints to all components*

The mass properties and the moment of inertia are assigned to all the parts that are imported in ADAMS from CATIA V5 environment. Also, various joints are defined to different parts. The crankshaft and the connecting rod are assigned with the revolute joint. The pinion is fixed to the crankshaft whereas the gear is fixed to balancer shaft. The revolute joint is defined for the crankcase including balancer and the crankshaft to have free rotation motion. The gudgeon pin is defined with revolute joint with respect to the connecting rod whereas it is fixed with the piston. The translational motion is assigned to the piston. The rotational motion is assigned to the crankshaft at the different point of revolute joining between crankshaft and connecting rod. All the joints are defined and are hence checked for the constraint to avoid redundancy in the final simulation of the mechanism and simulated as per actual working conditions.

I. Experimental and simulated vibration check on crankcase including no balancer shaft

Structural vibration measurements were carried experimentally and acceleration values were obtained for different speed on the crankcase. On similar ground, the simulated values of vibration on crankcase were determined for different speed in ADAMS and the comparison between the experimental and simulated is as given below in Fig. 06

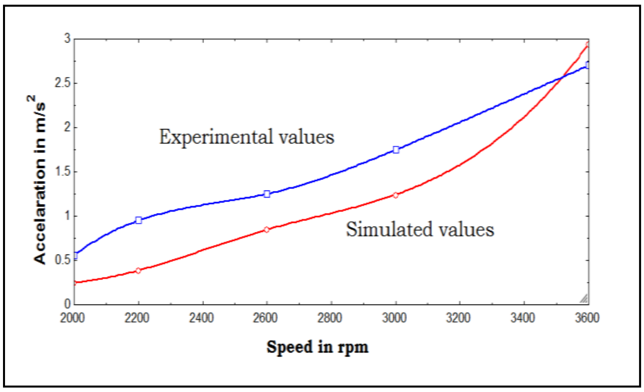

Fig 06: Comparison of experimental and simulated values of vibration on crankcase including no balancer shaft.

J. Simulated values of vibration check on crankcase with incorporation of balancer shaft

Vibration check was performed on the crankcase of engine with application of balancer shaft. The magnitude of vibration is checked at 3600rpm as follows:

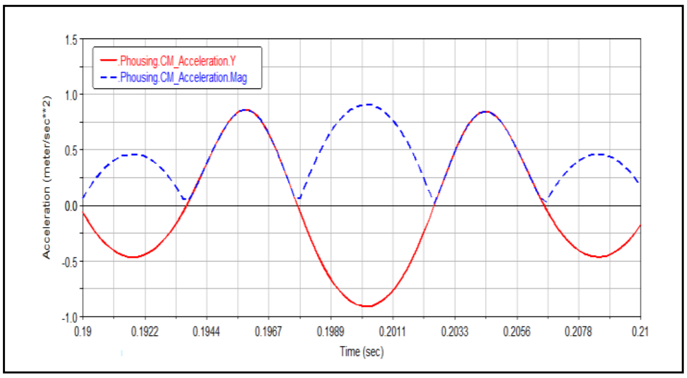

Fig 07: Simulation values of acceleration on crankcase including balancer shaft

IV. ANALYSIS RESULTS AND INFERENCES.

The CAD model after successful importing into the Multi-body dynamic software, is used for analysis, the value of acceleration on crankcase at different speeds were noted down and the maximum peak acceleration value was taken into consideration. Firstly, one set of vibration check was performed on the mechanism including no balancer shaft. On the same grounds, one more check was performed on the engine including balancer shaft in the system. The two different values were compared for different speed as given.

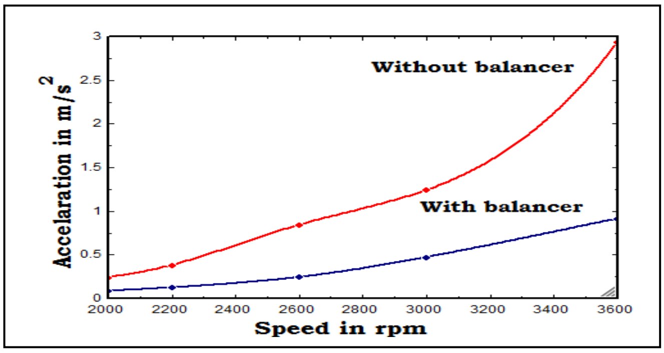

Fig 08: Comparison between simulated vibration values on crankcase including and excluding balancer shaft

Table III: Comparison between simulated vibrations on crankcase.

Speed under consideration	Simulated value of vibration on crankcase including no balancer shaft	Simulated value of vibration on crankcase including balancer shaft
2000 rpm	0.2466 m/s^2	0.0866m/s^2
2200 rpm	0.3871m/s^2	0.1268m/s^2
2600 rpm	0.8482 m/s^2	0.2471 m/s^2
3000 rpm	1.2439 m/s^2	0.4747 m/s^2
3600 rpm	2.9445 m/s^2	0.9194 m/s^2

The graph shows the comparisons between vibration including and excluding balancer shaft on the crankcase. It shows the effectiveness of balancer shaft in order to reduce the net vibration on crankcase and further to passenger via chassis. The vibration reduction at rated speed of 3600 rpm is calculated as 68.77% whereas at lower speed of 2000 rpm it is calculated as 64.88%.

V. FINITE ELEMENT ANALYSIS ON BALANCER SHAFT

The finite element analysis is carried on the balancer shaft using the ANSYS software for checking its modes of vibration i.e. Modal analysis. Modes are inherent properties of any assembly or a module which can be given by its material properties which are stiffness, damping and mass. A mode is defined by natural frequencies and modal parameter. If any one of the parameter is altered, there is change in the modal frequency of the module. For example, if certain amount of mass is added or subtracted from the module, then it will vibrate differently than before.

Modal analysis is used to determine the structure's vibrational characteristic i.e. its natural frequencies and mode shapes. In the following segment, we determine the modes of vibration of balancer shaft and its corresponding vibrating frequency.

The finite element analysis on balancer shaft shows that the frequency of vibration of balancer shaft is much higher than the actual vibration of engine as shown in the Fig. 09. Thus, it prevents the resonance in the system. Hence, it is validated that the balancer shaft is safe in modal analysis.

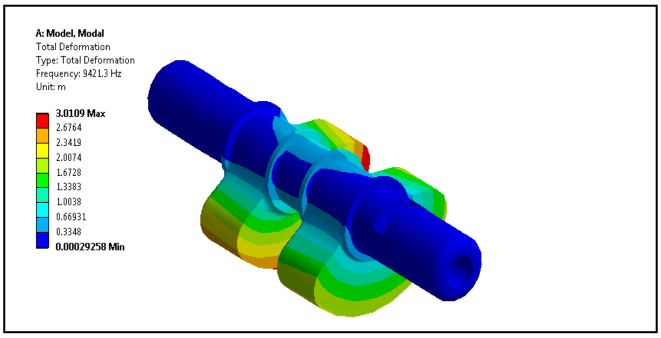

Fig 09: Modal analysis of balancer shaft

The mode of vibration for first to sixth mode is tabulated as follows.

Table II: Modes of vibration of balancer shaft and respective frequency.

Mode	Frequency [Hz]
01.	Rigid body mode
02.	9421.3
03.	11420
04.	11803
05.	16520
06.	18450

VI. CONCLUSION.

In this way, step by step design procedure of balancer shaft was carried out and its influence on engine performance, in particular to crankcase was studied. The conclusion can be given as follows:

At first, analytically major forces acting upon the engine i.e. primary inertia force is calculated. Then the unbalanced vector is calculated by taking the difference between the theoretical mass and actual mass and multiplying the difference by center of gravity of crankshaft. This vector is then compromised in the eccentric weights of balancer shafts.

Then, we calculated the forces that acts on balance shaft and designed the same considering the equivalent twisting moment and factor of safety. The balancer shaft gear is designed selecting the module as two.

Then the CAD modelling of each engine module is done and the assembly is imported into a multi-dynamic software ADAMS. Vibration is checked on the crankcase including and excluding balancer shaft and inference is given. Also, the modal analysis of balancer shaft validates the design of balancer shaft.

ACKNOWLEDGEMENT

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REFERANCES

[1] D. Meek and M. Roberts, "Balance Shaft Conversion of a Four Cylinder Engine – The Challenges of Packaging and Mechanical Integration" , SAE Paper No. 981084.

[2] H. Nakamura, "A Low Vibration Engine with Unique Counter-Balance Shafts", SAE Paper No. 760111.

[3] J. Saunders and P. Walker, "Design and Development of a Novel Balancing System for Single and In-Line Twin Cylinder Engines", SAE Paper No. 2003-32-0026.

[4] C. Kim, Y. Kang, B. Lee, H. Ahn, "Determination of optimal position for both support bearing and unbalance mass of balance shaft", SciVerse ScienceDirect Paper ,2011.

[5] A. Kolchin and V. Demidov - Design of Automotive Engines, Mir Publication, 1984.

[6] Heinz Heisler – Advanced Engine Technology, Butterworth-Heinermann Linacre House, 1995

[7] A. Ferreira, C. Calviti, F. Barbieri and C. Argachoy, "Balancer Shaft Development for an In-line 4-Cylinder High Speed Diesel Engine", SAE Paper No. 2008-36-0219.

[8] K. Suh, Y. Lee and H. Yoon,"A Study on the Balancing of the Three-Cylinder Engine with Balance Shaft", SAE Paper No. 2001-01-0601.

[9] T. Gilles – Automotive Engines: Diagnosis, Repair and Rebuilding, Cengage Learning Center, 6th Edition, 2011

[10] J. Erjave – Automotive Technology: A System Approach, Cengage Learning Center, 5th Edition, 2010.