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Design of Balancer Shaft In Single Cylinder Diesel Engine

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Abstract

High-speed engines and other machines have common phenomenon now days of Engine noise, vibration and harshness. It is, therefore, very essential that all the rotating and reciprocating parts should be balanced, as far as possible. If those parts are not properly balanced, the dynamic forces can occur. These forces not only increase the loads on bearings and stresses in the various members, but also produce unpleasant and even dangerous vibrations. This paper describes detailed procedure for the design of single balancer shaft to reduce the vibration. The force is measured on the engine at crankcase, running crankshaft at 2600rpm and unbalanced force on the crankshaft is measured after primary balancing. Balancer shaft is designed for the unbalanced forces exist after primary balancing. The force on crankcase measured after introducing balancer shaft. The effectiveness of primary balancing and balancer shaft is validated by comparing the force measured at crankcase. The effectiveness of integration of balancer shaft and the modal analysis is carried to validate the design of the balancer shaft in ANSYS.

Keywords: Balancer shaft; Primary inertia force; eccentric mass; balancing; counterweight; ANSYS,

1. Introduction

A major cause of NVH in a single cylinder engine is piston reciprocation. The piston is started and stopped twice during each rotation of the crankshaft, and reactions to the forces that accelerate and decelerate the piston are imposed upon the engine body as vibration in directions. This vibration is uncomfortable and could produce operator fatigue and tends to reduce the useful life of the machine.

To some extent such vibrations can be decreased by providing the engine with a counter Weight fixed on its crankshaft, and located at the side of the crankshaft axis directly opposite the crankpin by Which the piston, through the connecting rod, is connected to the crankshaft. More commonly, two counter weights may be used on the crank shaft, one located on each side of the piston axis. In either case, such a crankshaft counter Weight arrangement produces a net resultant centrifugal force vector that is diametrically opposite to the crankpin.[6]

Although such a crankshaft counter weight arrangement can be designed to cancel some or even all of the primary acceleration and deceleration forces on the piston assembly along the piston axis, the centrifugal force of the crankshaft counter weights also has a component transverse to the piston axis. This transverse force component produces lateral vibration, the amount of which increases in direction proportion to the degree to which the crankshaft counter weights

successfully cancel out the acceleration and deceleration forces on the piston assembly.

That's why most single cylinder engines introduce crankshaft counter weights having a mass that provides a condition of about 50% overbalanced, such that the centrifugal force due to the counter weights has a component along the piston axis that is equal to about 50% of the acceleration and deceleration forces on the piston assembly. This represents a compromise between the vibration in directions parallel to the piston axis that would result with the condition of no overbalance, and the severe vibration transverse to the piston axis that Would result with the condition of 100% overbalance. Because use of crankshaft counter weights having a 50% overbalance condition does not entirely eliminate the undesirable vibration occurring in single cylinder engines, additional techniques have been employed to further reduce such vibration. Balancer Shaft is one of them.

This vibration can be minimize 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. [1]

2. 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 Lancaster around a hundred years ago. This invention brought a revolution in the engine vibration and balancing.

David Meek and Martyn Roberts [2], 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 [3] 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 rotate at a speed twice that of the crankshaft and reduces the second order vibration present in the four-cylinder configuration.

Jonathan Saunders and Patrick Walker [4] 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 [5] provided with the strategy to minimize the elastic strain energy and kinematic energy of balancer shaft. This paper describes about the bending deformation due to balancer shaft and loss of power due to use of balancer shaft for specified engine target.

3. Objective

It is essential to balance the engines to reduce the vibration. There is method to balance the engine is to attach the bob weight to the crank shaft. However, it only balances the 50% of the reciprocating masses from primary unbalance force. Hence main goal of this research work is, "To balance the remaining 50% from the primary unbalanced force by using balancer shaft, hence design the balancer shaft for the same.

4. Methodology

1. The existing engine is already primary balance by means of the bob weight is attached to the crankshaft. After primary balancing, the unbalance force is measured on the engine crankcase, in simulation software.
2. Balancer shaft is designed for the unbalanced forces exist, after primary balancing. It includes the shaft diameter, gear design, bob weight of balancer shaft etc.
3. Balancer shaft is introduced into the simulation model and simulated the running at 2600rpm and unbalance force on the crankcase is measured
4. The effectiveness of primary balancing and balancer shaft is validated by comparing the unbalanced force measured at crankcase.

5. Technical Writing

The piston moves along a straight line, about axis of the cylinder. However, its velocity is continually changing throughout a cycle, it is stationary when at both TDC and BDC, achieving maximum velocity somewhere around the mid-stroke. An oscillating moment must be applied to the piston which may result in alternating accelerations. If these inertia forces are not balanced internally within an engine. They must pass through the connecting rod to the crankshaft then on to the main bearings and onto the crankcase, from the crankcase they are passed into the frame through the engine mountings. The rider feels these forces as annoying or incapacitating vibration, depending on their severity.

To determine the masses and forces, following procedure is to be followed.

We know that total bob weight on crankshaft =
(Wt of Crank + 2/3 wt of con Rod) +
Rotating Part
50% (1/3 wt of Con Rod + wt of Piston)
Reciprocating Part

a) For rotating balancing, two similar counterweights are fitted on the web extension. The center of gravity of cw is at a distance of μ from crankshaft axis.

$$2m_{cwRo} \mu \omega^2 = m_{Ro} R \omega^2$$

Where m_{cwRo} = Mass on bob weight for compensation of rotating imbalance.

m_{Ro} = Total rotating mass act on crank shaft

R = Crank radius

μ = Distance between CG of CW from crankshaft axis

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:

$$2m_{cwRc} \mu \omega^2 = 0.5 m_{Rc} R \omega^2$$

Where m_{cwRc} = Mass on bob weight for compensation of reciprocating imbalance.

m_{Rc} = Total reciprocating mass act on crank shaft

R = Crank radius

μ = Distance between CG of CW from crankshaft axis

c) Hence total mass on crank bob $m_{cw} = m_{cwRo} + m_{cwRc}$

Total mass of bob on crankshaft is 2.3kg

d) As the balancer shaft rotates at a same speed of crankshaft, hence number of teeth on balancer shaft is equal to the teeth on crankshaft. Hence, $Z_1 = Z_2$

We know that,

$D = (mz)$

$C = m [Z_1 + Z_2] / 2$

Where,

D = pitch circle diameter of gear; mm.

m = module on gear; mm

C = center distance between axes of two shaft; mm

We have the centre distance between the axis of rotation of crankshaft and balancer shaft as 100mm. 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. And hence number of teeth is 50.

e) Balancer shaft design

As per the driver and driven gear select, with number of teeth on gear as 50 and 100mm diameter, we take the value of shaft diameter as 20mm, as per standard availability.

6. Analysis

In ANSYS first, model of the single cylinder diesel engine was imported in rigid dynamics project. 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. The mass properties and the moment of inertia are assigned to all the 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.

6.1 Without balancer shaft

To get the unbalanced forces in engine, we have major the reaction force at crankcase to ground, in rigid dynamics solver of ANSYS. The result shown by solver as follows

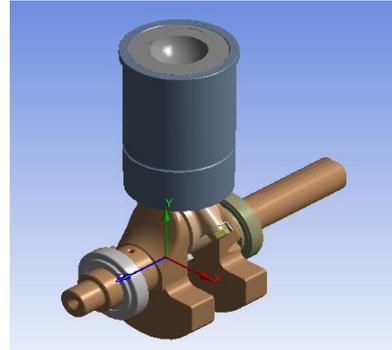


Fig 1. Model without balancer shaft

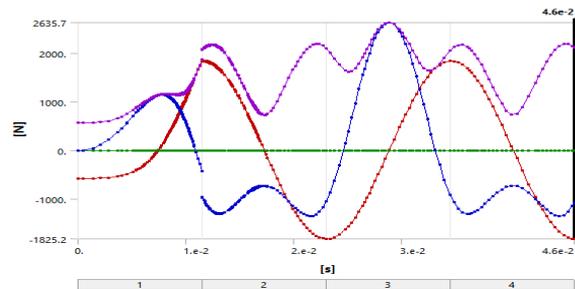


Fig 2. Total Resultant force without balancer shaft

From above result, we come to know that total resultant unbalance force is 2635N is remain in the engine.

6.2 With balancer shaft

To reduce the remaining unbalance force mention above, introduce the balancer shaft in the engine and simulate the same again with balancer shaft in rigid dynamics solver of ANSYS.



Fig3. Model with balancer shaft

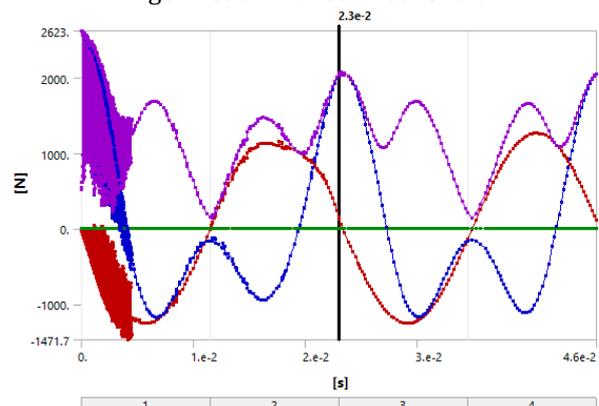


Fig 4. Total Resultant force with balancer shaft

In-Line Twin Cylinder Engines, SAE Paper No. 2003-32-0026.

7. Modal Analysis

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. Thus, it prevents the resonance in the system. Hence, it is validated that the balancer shaft is safe in modal analysis.

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

[6]<http://www.tonyfoale.com/Articles/EngineBalance/EngineBalance.pdf>

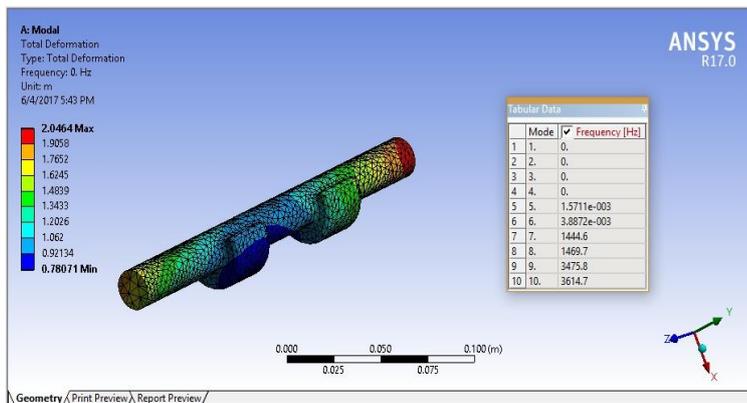


Fig 5. Modal Analysis

8. Conclusion

From above, find that unbalanced force at crankcase is reduced from 2635N to 2074N by introducing the balancer shaft in the engine, which ultimately help to reduce the noise and vibration in the engine and engine will be quieter and smoother.

9. References

[1]Terrence M. Rotter, Sheboygan Falls, WI (US); William D. Koenigs, Fond duLac, WI (US)(2003) *Balance System For Single Cylinder Engine*, US2003/0121489 A1

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

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

[4]J. Saunders and P. Walker,(2003), *Design and Development of a Novel Balancing System for Single and*