EXPERIMENTAL INVESTIGATION OF THE EFFECTS OF TORSIONAL EXCITATION OF VARIABLE INERTIA EFFECTS IN A MULTI-CYLINDER RECIPROCATING ENGINE

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ABSTRACT

The failures of crank shaft assemblies because of excessive torsional stresses have long been reported in the literature. Many different factors were attributed 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 non linear 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. This theory of variable inertia effects and secondary resonance in case of reciprocating engines has been explored, analyzed and confirmed by many earlier researchers, but theoretically. Experimental investigations were not attempted by earlier researchers, to verify the above phenomenon. One experimental investigation was carried out a decade ago on single cylinder engine. During that investigation the theory was verified. However this problem is more prominent in multi-cylinder engines and experimentation has been conducted to this effect. The objective of this paper is to report the results of experimental investigations using an electronic exciter to excite the system at various excitation frequencies and a ROTEC vibration analyzer. The results could confirm the theory. As predicted the non linear frequency coupling was observed experimentally. The reciprocal of the effect was also confirmed with the existence of secondary resonance.
Key words: Variable inertia, Torsional vibration analysis (TVA), Secondary Resonance, Mode Shapes, Eigen Values, Side Bands


1. INTRODUCTION

The effective inertia of an engine is a function of crank angle. For every next configuration the geometry of engine in terms of reciprocating parts and connecting rod masses would continuously change with respect to the axis of rotation of crank shaft. So for each new crank angle the effective inertia would be different. And the engine geometry follows the same path during every rotation of the crank shaft which results in smooth variation of inertia function. In one of the papers presented by Archer [1] about four crank shafts of four different ships had failed due to variable inertia effect. Majority of the researchers analyzing the torsional vibrations of the engines have neglected fluctuating inertia which would result in linear torsional vibration analysis. The guidelines for linear torsional vibration analysis are given in [2, 3, 4, 5] which are universally followed by all researchers. When non linear parameter like variable engine inertia is considered, they would result in non linear vibration analysis. When variable inertia effect is considered for analysis, this would result in non linear frequency modulation that takes place between the torsional natural frequencies and the engine rotational speed and their harmonic components. This kind of modulation is generally referred as Secondary Resonance because it appears itself as side bands. These side bands can also be called secondary resonances around the major torsional natural frequency. The major resonance occurs at average natural frequency and the secondary resonances are located on either side at plus or minus double the engine speed. When engine speed varies the side bands also would move. The major resonance and the side bands are coupled in such a fashion that as per theory excitation at one of the secondary resonance can excite the main resonance and vice versa. This would infer that an excitation frequency which can be taken to be considerably safer margin from the average torsional natural frequency could be capable of exciting resonance if it matches with one of the side resonances. This phenomenon of secondary resonance was first studied by Draminsky[6,7,8]. He reported on destructive failures of large marine diesel engines and concluded that they happened because of secondary resonance. A tremendous amount of work was carried out and an exhaustive literature review was published by Hesterman and Stone[9]. However all their work was theoretical involving no experimental evidence to confirm the theory. Perhaps one of the major factors for the fact that all earlier researchers have carried out either very little or no experimental investigation is due to the limitation of exciting the system at various torsional excitation frequencies and measurement of the same. These days the torsional laser vibrometers of non contact type are used to measure the angular velocity. But they are found to be expensive. They can be used as only analyzers. However commercially viable simple exciters are available. Hesterman and Stone [10] analyzed the broad band excitation of combustion forces to investigate the first evidence of side bands on torsional resonance of a small single cylinder engine. Lee and Stone [11] could carry out
analysis of the engine under operating conditions with gas force excitation. Although this work was carried out soon the results were found to be insufficient to investigate variable inertia effects in detail. This calls for a requirement of exciting the system at a known frequency. Drew, Hesterman and Stone [12] designed a torsional exciter that was used for both driving and exciting. The broad band excitation at required frequencies along with the sufficient power to run a small single cylinder engine without gas forces was achieved. The experimental investigation in to variable inertia effects in case of a small 120 cc single cylinder engine was reported. This was for a single cylinder engine.

This paper deals with the findings of first detailed experimental investigation in to variable inertia effects and secondary resonance in multi-cylinder engine.

2. EXPERIMENTAL SET UP

2.1 Static Set Up
As shown in the fig 1, this experimental set up consists of a hydraulic actuator, Engine model, two accelerometers, analog board and ROTEC torsional vibration analyzer. Two crank assemblies of two single cylinder engines were taken. The flywheel, cam shaft, cylinder head, and timing gear were removed as to enable to achieve simplified engine model and to maximize the effects of fluctuating inertia function of the assembly. These two crank assemblies were welded together with 180 degrees phase difference. This is how engine model to be tested was made.

The static exciter consists of hydraulic circuits, electronic circuits, actuator, servo valves, spool valves and a locator assembly. This was used to excite the system in static condition. The locator of the exciter has a flange with holes. One flange of the same dimensions with the same number of holes was manufactured. Engine model can be fitted in to this flange with a key. These holes are at an interval of 30 degrees enabling the engine model to be fitted at an interval of crank angle of 30 degrees i.e. 0 to 360 degrees. Two accelerometers were mounted one on the actuator flange on input side and the other on the crank web. The cables from these two accelerometers were taken to an analog board to give display of the system response in the time domain. By means of another cable which connects the analog board with the ROTEC vibration analyzer. ROTEC vibration analyzer further displays FRF (frequency response function) which is the response of the system in the frequency domain.

2.2 Dynamic Set Up
As shown the fig 2, this consists of an electric motor, engine model, generator, couplings (2 in number), rotary encoder, dynamic exciter, a halogen lamp, torsional vibration analyzer. A single phase a. c. motor one hp was connected to the engine model through a coupling. The engine model is further connected to a d. c. generator of half hp. This generator is connected to the dynamic torsional exciter and also to electrical load. This electrical load tries to draw current as a result of which a magnetic field is induced. This magnetic field set up would try to create hindrance to the armature of the generator there by slowing it down. By switching the load on and off once, within a fraction of a second one torsional vibration pulse is induced in the moving shaft.
By means of an electronic circuit which is used in dynamic exciter, if this action is exercised 500 times in one second, 500 such torsional pulses are induced in one second giving a torsional excitation frequency of 500 Hz. By changing loads different amplitudes of excitation frequency can be achieved. This is how the dynamic exciter can be used to apply a known torsional excitation frequency of any desired value in the range of 1 Hz to 15000 Hz. Halogen bulb is used for different loads. An encoder fitted at the end of shaft line measures angular velocity signal and transmits the signal to ROTEC torsional vibration analyzer. ROTEC torsional vibration analyzer processes this signal and gives the response in frequency domain.

3. EXPERIMENTAL PROCEDURE

3.1 Static Test

Engine model was fitted in to static exciter by adjusting to suitable hole as to give a crank angle of 0 degrees. For both the input and output accelerometers, the readings in the time domain were noted and also noted the frequency response given by the vibration analyzer. The engine model was fitted in 12 different positions to give us crank angles of 12 different values. This means that the model was mounted with the crank angle starting from 0 degree to 360 degrees at an interval of 30 degrees. Each time FRF plot was taken to measure the natural frequencies of the system and to note the variation of natural frequencies with respect to crank rotation. However the natural frequencies varied with crank angle, the average natural frequencies as given by all the 12 FRF plots were found to be 100 Hz, 300 Hz.
3.2 Dynamic Test

Fig 3 shows actual photograph of the experimental setup. With the help of single phase one hp motor the entire system was run at an average speed of 1500 rpm (25 Hz). Angular velocity spectrum was taken by vibration analyzer. Two types of dynamic tests were carried out. In the first one no additional weights were added on reciprocating parts. In the second an additional weight of 70 grams was added on the reciprocating parts of both the cylinders.

The purpose of adding additional weights on pistons of both the cylinders is to enhance the effect of variable inertia. The natural frequencies obtained from the mass elastic system model of lumped parameters in the case of dynamic test without additional weights were 100.745 Hz, 311.625 Hz, 376.87 Hz, 2813.85 Hz, 10021.5 Hz. The natural frequencies obtained from the mass elastic system model of lumped parameters in the case of dynamic test with additional weights were 100.0136943 Hz, 302.866242 Hz, 365.1273885 Hz, 2815.286624 Hz, 10026.59236 Hz. From all the above three sets of readings it is evident that out of all the natural frequencies the one which is dictated by the engine happens to be 100 Hz. Secondary resonance phenomenon is nothing but frequency coupling between variable natural frequencies and the rotational speed and the harmonics.

![Figure 3: Setup for dynamic experiment](image-url)

With the help of dynamic torsional exciter unit system was excited at 100 Hz and was varied in the increments of 25 Hz that is 25 Hz, 50 Hz, 75 Hz, 100 Hz, 125 Hz, 150 Hz, 175 Hz and 200 Hz. The system was also excited at higher natural frequencies and the harmonics of the engine speed. In addition system was also excited at a non resonant excitation frequency to see the effect. At all these excitation frequencies the side bands were investigated.
4. RESULTS AND DISCUSSION

The physical model of the entire system whose analysis is to be carried out is made and is shown in fig 4. The physical model is further converted to lumped mass elastic vibration model to carry out the torsional vibration analysis.

![Physical model and Lumped mass elastic system for dynamic setup](image)

Figure 4: Physical model and Lumped mass elastic system for dynamic setup

To obtain average natural frequencies and the modal vectors of the system HOLZER method was used and the HOLZER calculations for first two natural frequencies and mode shapes, residual torques are shown in table 1 & table 2

<table>
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<tr>
<th>Name Of The Rotors</th>
<th>$\omega$</th>
<th>$\omega^*\omega$</th>
<th>$I$</th>
<th>$I\omega^2/10^6$</th>
<th>$\beta$</th>
<th>$I\beta\omega^2/10^6$</th>
<th>$\sum I\beta\omega^2/10^6$</th>
<th>$K_T/10^6$</th>
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Table 1 HOLZER calculation table for first natural frequency – Dynamic

<table>
<thead>
<tr>
<th>Name Of The Rotors</th>
<th>$\omega$</th>
<th>$\omega^*\omega$</th>
<th>$I$</th>
<th>$I\omega^2/10^6$</th>
<th>$\beta$</th>
<th>$I\beta\omega^2/10^6$</th>
<th>$\sum I\beta\omega^2/10^6$</th>
<th>$K_T/10^6$</th>
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Experimental Investigation of The Effects of Torsional Excitation of Variable Inertia Effects In A Multi-Cylinder Reciprocating Engine

The variation of effective engine inertia and natural frequencies and corresponding modal vector as function of crank angle in every rotation of crank shaft are shown in fig 5, fig 6 & fig 7.
Fig 7: Axial Distances of Rotors Vs Modal shapes

Fig 8, Fig 9 & Fig 10 shows the measured frequencies given as per FRF plot at crank angles being 0°, 60° and 90° and the nature of variation with respect to crank angle.

Figure 8 FRF plot for 0 Degrees of crank angle

Fig 10: FRF plot for 90 Degrees of crank angle
Experimental Investigation of The Effects of Torsional Excitation of Variable Inertia Effects In A Multi-Cylinder Reciprocating Engine

Figure 10 FRF plot for 90 Degrees of crank angle

As shown in the fig 11, when the system was excited at 50 Hz, main resonance was observed to occur at 100 Hz with a magnitude of 580 rad/s² and secondary resonances at 50 Hz, 75 Hz, 125 Hz & 150 Hz of magnitudes of 425 rad/s², 120 rad/s², 50 rad/s² & 130 rad/s² respectively were observed.

Figure 11 Frequency spectrum when excited at 50 Hz without weights in dynamic test

Figure 12 Frequency spectrum when excited at 100 Hz without weights in dynamic test

As shown in the fig 12, when the system was excited at 100 Hz, main resonance was observed to occur at 100 Hz with a magnitude of 1260 rad/s² and secondary resonances at 50 Hz of magnitudes of 3805 rad/s² was observed.
Figure 13 Frequency spectrum when excited 125 Hz without weights in dynamic test

Figure 14 Frequency spectrum when excited at at non-resonant frequency of 135 Hz with weights in dynamic test

As shown in the fig 13, when the system was excited at 125 Hz, main resonance was observed to occur at 100 Hz with a magnitude of 33 rad/s and secondary resonances at 50 Hz, at 75 Hz, 125 Hz & 150 Hz of magnitudes of 14 rad/s, 10 rad/s, 6 rad/s & 8 rad/s respectively were observed.

As shown in the fig 14, when the system was excited at non resonant frequency of 135 Hz, insignificant couplings between the harmonics of rotational speed and natural frequencies were observed.

5. CONCLUSION
Torsional vibration analysis considering non linear parameter of variable engine inertia was carried out on the experimental test rig. The existence of secondary resonance was observed. When the system was excited at first natural frequency that is 100 Hz, the main resonance peak was observed at that frequency and the side bands were observed at 25 Hz, 50 Hz, 75 Hz, 125 Hz, 150 Hz. When the system was excited at any one of the side bands then also the main peak was still at 100 Hz and side bands were observed at 25 Hz, 50 Hz, 75 Hz, 125 Hz and 150 Hz, which were harmonics of rotational speed. No side bands were observed between the above said frequencies. When the rotational speed of the system was changed the side bands also could shift. When the system was excited at a non resonant frequency of 135 Hz, very insignificant frequency coupling was observed. This analysis can be further used to determine the regions of instability and secondary resonances for any marine engine.
Experimental Investigation of The Effects of Torsional Excitation of Variable Inertia Effects In A Multi-Cylinder Reciprocating Engine

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