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Vibration Signature Analysis of One Cylinder Engine

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ABSTRACT
All parts of vehicle engine have a different frequency spectrum signature. The vibration signature collected by accelerometer is a time-based signal and may be used for further processes for its overall level, phase, peak, spike energy etc. The underlying principle is that, each component of the IC engine generates identifiable frequencies. So, any change in the vibration level at any given frequency can relate directly to the concerned engine parts. So, the change in the vibration level at any given frequency can relate directly to the concerned engine elements. The vibration spectrum shows the areas of stress and unnecessary energy. The trend of vibration measurements changes at different locations along the elements to guess problems. In this paper, an attempt has been made to analyze vibration signals for the IC engine to detect the dynamic characteristics utilizing the vibration signatures measured. This method depends on the variations in time and frequency to distinguish different operating conditions and to provide remedies to alleviate the engine from the increase of vibration levels which improve the engine performance.
KEY WORDS: *Vibration signature, spark ignition engine, frequency domain, time history, rotational speed.*

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I. INTRODUCTION

Vibration used as an effective method to diagnose and detect some of primary failures of the equipment and machine. The measured Vibration signature on the outer surface of the machine or a structure contains good amount of information, which may properly interpret, can expose the working condition of machine (Karanjikar and Supekar, 2015). It may be observed as the one of the languages through which the machine tells its ailments. Each element of the mechanical equipment has a different frequency signature spectrum. The vibration spectrum shows the areas of stress and unnecessary energy. The trend of Vibration measurements changes at different places along the element to predict problems. Misalignment, unbalance, eccentric shafts, mechanical looseness, gear wear, broken teeth, bearing wear can be determined using vibration. It is likely to get data using suitable accelerometer which is processes using various methods such as cestrum, spectrogram, FFT, cyclostationary analysis and wavelet transform.

An attempt has been done to analyze the vibration signals of IC engine to detect the existence of any fault utilizing Fast Fourier Transform (FFT) (Griffiths and Skorecki, 1964). The method depends on the variations in frequency to distinguish the different operating conditions of an internal combustion engine and providing remedies to ease the engine from vibrations excess which also improves the performance of the engine. A decrease in water temperature increases the average vibration in the frequency range between 500 to 2000 c/s while the opposite is true in the area of 5500 c / s. This effect was measured sufficient to require temperature control once investigating the effect of other engine variables on vibration. Removing certain sources of vibration and motoring engine were used to investigate piston slap. A solitary zone zero-dimensional model for any hydrocarbon fuel dependent on Wiebe heat discharge work has been executed in Simulink to test sparkle start motor execution. To show motor cycle, Annand's model for convective warmth misfortunes were utilized. Exploratory outcomes from writing have utilized for approval utilizing Simulink.

It was found that the peak pressure during the combustion process decrease with decreasing in intake pressure which was like the experimental trend. Speed up from 1000 rpm to 4000 rpm prompts decline of the brake warm productivity from 25% to 17% while the demonstrated effectiveness remains practically steady

(32%). Fluid dynamic two-dimensional computational model using smooth of experimental test engine is prepared for axisymmetric flow in the combustion chamber (Chaudhari el at., 2014).

The effects of kerosene-diesel mixtures on the operation of the diesel engine were studied when supplied with diesel and kerosene-diesel blends at different rates (Atole el at., 2017) (Azeem el at., 2016). A pilot study was conducted to evaluate the effect of diesel kerosene mixing on the performance characteristics of four-stroke single-cylinder diesel engine.

Various degrees of combination 10%, 20% and 30% of lamp oil mixing by volume with diesel fuel and they were picked as K10, K20 and K30 separately, while unadulterated diesel was considered as reference and named D.

The brake thermal efficiency, mechanical efficiency, exhaust temperature and fuel consumption performance have been studied. Pressure variation of gases in each cycle, reciprocating parts impact forces, rotating unbalance and mount structural characteristics are the main reasons for overall vibration behavior of IC engine (Jamadar and Pingle, 2017) (Patil and Raut, 2016).

During working of motor speed, load on the motor is changing relying on need. Because of which fuel supply and burning attributes additionally changes. Significant segments of an IC motor are constantly having relative movement in the middle of them. The inertial powers change regarding pressure and ignition variety inside the chamber. It is unequivocal that inertial powers bring about uneven powers and they are fluctuating with speed of the motor, fuel provided and burning attributes of the fuel. Thus for satisfying purchaser fulfillment and ascertain the vibration yield and its minimization, a numerically exact plan model alongside its reenactment is all things considered required.

The idleness powers change with veneration to pressure and ignition variety inside the chamber. It is unmistakable that dormancy powers cause ascend to uneven powers and they are fluctuating with speed of the motor, fuel provided and fuel ignition attributes. Subsequently, for satisfying buyer and figure the vibration yield, a numerical model alongside its reenactment is all things considered required. Traditional methods of reducing NOx such as water and steam injection cannot reach those extremely low levels required. This fact led to the development of dry lean premixed combustion systems. They operate with excess air to lower the combustion temperature which in turn decreases the NOx levels. However, the lean regime makes the combustor prone to thermo-acoustic instabilities. They may cause mechanical damage, increased heat transfer to walls, higher noise and pollutant emissions. This phenomenon is observed not only in gas turbines, but also in rocket motors, ramjets, afterburners, and domestic burners (Ajith el at., 2013). Thermo-acoustic insecurities emerge because of a criticism circle including changes in acoustic pressing factor, speed and heat release.

So, active control strategies are employed to alleviate these oscillations. Typically, the pressure signal is sensed and used by a suitable control algorithm to control an actuator, for instance a loudspeaker or a secondary fuel injector. For the design of more advanced active controllers, an accurate model of the process must be obtained. Moreover, an analytic model also provides insight into the intricacies of thermo-acoustic instabilities, and thus helps to improve combustors already in the design stage.

Neural network model for internal combustion engine was presented. Firstly, the data were collected from a 1.6L spark ignition engine coupled to a hydraulic dynamometer. Angle of injection, engine speed, and the amount of injected fuel were excited into a specific value and then emissions were measured **(**Martínez-Morales el at., 2012). The data were collected by a real time system regarding those variables. A local linear radial basis function network was used in addition to a training algorithm for online adaptation of neural network parameters. The results show that the proposed approach in modeling the studied gasoline engine is valuable.

The most expensive and complex part of any vehicle powertrain, which consists of the engine, intake, exhaust subsystems, transmission and drivetrain systems. It is has been counted as one of the major sources of vehicle sounds and vibrations (Singh el at., 2016). For the most part, vibration and sound sources comprise of the accompanying: 1) Engine, including ignition related sounds and vibrations, responding unbalance, turning unbalance, driving rod torsional motions, and so forth; 2) Valve Train framework, including valves, cam framework, crankshaft belt, or chain; 3) Accessories, including their unbalance and reverberation; 4) Intake framework and fumes framework vibrations; 5) Driveshaft first-request reverberation; 6) Universal joint second-request twisting vibrations and torsional vibrations; and 7) Axle vibrations because of stuff tooth formation blunder, transmission mistake, pinion standard whimsy, slip-stick among pinion and ring gear, and so on. In this work various sources of noise in an engine have been analyzed.

A nonlinear numerical model for single cylinder has been presented. The model includes all firm bodies inertial members, joints, support bearings, couplers, and connections between the numerous engine components. The model is parameterized, thus enabling virtual prototype testing of various engine designs, as well as letting engine designer to perform a inclusive noise, vibration, and harshness (NVH) study of engine performance. This method in engine design decreases the conceptual design-to-development cycle time and eliminates the need to extensive engine testing, this accounts for a large portion of the cost of engine design and production. The model solves massive displacement dynamics of engine parts, support bearing movements, and trapped air-fuel cylinder transient pressures all at the same time. The obtained results show the dynamic reaction of all inertial members, such as the flywheel, piston, and connecting rod, over time. (Boysal and Rahnejat, 1997). Frequency domain analysis of the results shows promise with generally known experimental spectra. Also, the numerical results come to an agreement with a closed-form analytical solution reported by several authors**.**

Fatigue failure which is harmful to engine supporting structures is a result of vibrations. The impact presents large forces and thus large stresses, which can cause both vibrations and early failure of the machines. The target of this work was to show the viability of motor help and impact of detachment on the primary vibrations (Ahirrao el at., 2018). To limit the potential issues related with motor vibration, a hearty and exact plan of help was required which incorporates shot joints. Giving legitimate mounting to decrease the motor vibrations. At times dampers at the interface of the motor and establishment are required. The interior ignition motor is the concentrated mass and if not appropriately upheld, it will prompt vibrations and move to supporting designs. Damping is difficult to display, so our attention was on the firmness as a boundary for this examination by direct estimation of vibration esteems close to the reverberation conditions. Aftereffects of this paper will give the architect a simple method to change the firmness boundary by fixing of bolts to the backings, which assists with diminishing vibration and commotion level.

The above review has shown that almost most of the efforts which had been done in this subject are directed towards the studying of vehicle vibration signature analysis. Their contributions are limited due to the difficult existed in measurements and the equipment. Furthermore, most of the studies which have been published are largely theoretical, and relatively little data is available concerning the practical or experimental implementation. However, in this paper an attempt has been made to utilize the IC engine vibration signatures signals in terms of time and frequency domains measured to detect its dynamic characteristics in different operating situations over different torque loading up to 90 Nm. Severity condition for the IC engine components before damages occur are computed in three directions (vertical, longitudinal and lateral). The engine rotational speeds used are 1000, 1500, 2000 and 2500 rpm. The time and frequency ranges are up to 22 second and 800 Hz respectively

II. ENGINE VIBRATION GENERATION CHARACTERISTICS

2.1 Background

The inner burning motor is the amassed mass in vehicle and if not appropriately planned, it will cause vibrations and move something very similar to the supporting designs. Ride solace, driving dependability and drivability are significant components in the exhibition of a vehicle and are influenced by the motor vibrations. As a result of the natural contemplations, just as changes in customer inclinations, vibrations actuated should be decreased. Vibration conduct of an IC motor relies upon uneven responding and turning parts, cyclic variety in gas pressure, shaking powers because of the responding parts and underlying qualities of the mounts. Motor vibrations are caused because of the responding and pivoting masses of the motor. The varieties of inertial powers are because of the ignition and the pressure contrasts of the cylinder chamber course of action during their activities. The motor inertial powers lead to the lopsided powers of the motor and they will in general shift concerning speed, fuel supply and burning qualities of the fuel.

2.2 Vibration signature

Vibration signature may be a combination of the various harmonics and they are obtained by processing the signal through a frequency is a potential indicator of engine condition. Vibration signature as obtained by accelerometer is a time-based signal which may be further processed for its overall level, peak, phase, spike energy etc. The underlying principle is that, each component of the IC engine generates identifiable frequencies. Thus, the change in the vibration level at any given frequency can relate directly to the concerned engine parts. The essential standards included are:

- Any failing or weakening in the activity of machine segment leads to increment in vibration level.
- Vibration radiating from the segment comprises of specific frequencies relying on the idea of the activity. This recurrence doesn't get changed or lost during the transmission of vibration. Notwithstanding, their vibration level might be weakened.
- Mixing of various vibrations (frequencies) doesn't bring about any deficiency of individual recurrence data.
- Every singular part or a framework has its own recurrence called normal recurrence, which changes just when framework boundary influenced.

2.3 Real time FFT

The fast Fourier transformer gives the direct representation of signal in frequency domain. The technique computes frequency spectrum and derived results like FRF (Frequency Response Function) from the time domain data acquired by block. Real-time can have two inferences; the first one is that one can see the frequency spectrum in live mode, continuously. This is equivalent to oscilloscope in frequency domain. The second inference is that the computation considers all the time domain data to compute the frequency spectrum. The consequence is that the all the computation rate must be faster than the acquisition time. Any loss in time domain data results in an error in the frequency spectrum. The real time band is then a critical parameter whenever precise measurement is required.

2.4 Vibration analysis

The frequency of a simple vibration can be calculated from its period. But the vibration signals from most rotating machinery contain harmonics of the fundamental rotational frequencies, so data must be analyzed by Fourier method which is established for periodic and random vibration using FFT algorithms. Most of the defects encountered in the rotating machinery give rise to a distinct vibration pattern (vibration signature analysis techniques). But it has been found that the defect developed may not show any indication in time domain unless it deteriorates to an advanced condition. But in frequency domain signal not only registers the problem early in its growth but actually conveys, with certain probability, the nature of impending faults like misalignment, bearing failure, gear faults etc. The principle objectives in analyzing the vibration data are:

- Simplify and reduce the vibration data into a more compact easily interpreted form.
- Associate characteristics of the vibration to specific features of the engine vibrating.
- Provide a consistent, repeatable measurement by which to characterize the vibration of an IC engine.
- Identify characteristics that change with time and operating condition, or both.

2.4.1 FFT analyzer

FFT analyzer also called spectrum analysis, which is defined as the transform of a signal from a time domain representation into a frequency domain representation. The FFT analyzer is a batch-processing device i.e., it samples the input signal for specific time interval collecting the sample in a buffer, after which it performs the FFT calculations on that batch and displays the resulting spectrum. By using FFT enabled researchers to look into the degree of chance that occurred at what frequency level and the frequencies produced from a specific place. Also minor deviations can be detected using frequency analysis (observing the waveform in the frequency domain).

Smart phone sensors

The smart phone the sensors (ICM-20608-G) are software-based and consist of three independent vibratory MEMS rate gyroscopes that detect rotation around the X, Y, and Z axes. Then, signal is amplified, and filtered to produce a voltage that is relative to the angular rate. Programming based sensors get their information from at least one of the equipment based sensors and are at times called virtual sensors or manufactured sensors. The direct speed increase sensor and the gravity sensor are instances of programming based sensors. As a rule, the sensor facilitate framework was appeared in Figure 1, where the sensor structure utilizes a standard 3-pivot organize framework to communicate information esteems. For most sensors, the arrange framework is characterized comparative with the gadget's screen when the gadget is held in its default direction. At the point when a gadget is held in its default direction, the X pivot is flat and focuses to one side, the Y hub is vertical and face up, and the Z hub highlights the outside of the screen face. In this framework, facilitates behind the screen have negative Z esteems. This organize framework is utilized by the accompanying sensor.

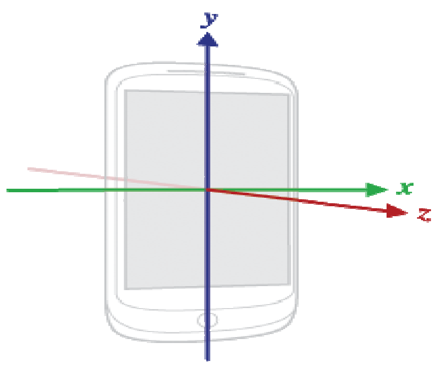

Fig.1 *Coordinate system (relative to a device)*

III. EXPERIMENTAL SETUP AND PROCEDURE

The experimental setup consists of the engine and hydraulic disc brake that were hard-mounted and aligned on a bed plate. The bed plate was mounted using isolation feet to avoid vibration transmission to the floor. The establishment of the experimental methodology and the smart phone sensors positions are presented in **Figure 2**, where the measuring of acceleration has been done at three directions (X, Y, Z) in three positions, namely, i) valve cover top, ii) liner-piston skirt and iii) crankcase - connected rod (CR) big end. Tests were conducted in order to calibrate the sensors configuration and insure the reparability of the recordings and the proper operation with minimum noise of the system as well as the various cables and connections. **Figure 3** shows Photograph of the experimental setup. The engine speeds are 750 rpm (no load only), 1000, 1500, 2000 and 2500 rpm. The load is provided by a hydraulic brake connected to engine and applied torque load up to 90 Nm. The results then fed into the computer for further processing. The speed is measured by a photo electric probe. Recordings were carried out at constant speeds. The engine specifications are tabulated in **Table 1**, while **Table 2** presents the physical properties of the IC engine structure components are considered in this work.

Fig.2 *Experimental setup*

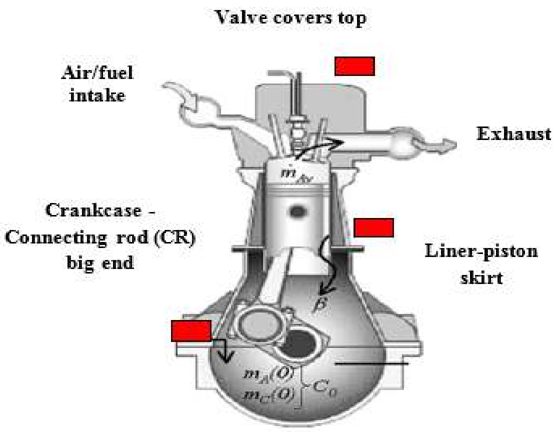

Fig. 3 *Engine components measuring points*

Table 1 *Engine Specification*

S/No.	Parameter	Character
1	Engine	Single cylinder, 4-stroke, air cooled
2	Model	VAZ 2103 1.5
3	Bore x Stroke, mm	62 x 49.5
4	Displacement, ml	149.4
5	Compression ratio	9:1
6	Starting Mode	Electric/kick-starter
7	Ignition system	CDI
8	Max. Power output, (kW) at rpm	10 / 8500
9	Rated Power output, (kW) at rpm	8.5 / 8500
10	Torque, Nm at rpm	10.0 / 7500
11	Engine oil type	SAE15W/40-SE
12	Engine oil capacity, L	1.1
13	Lubrication	Press/splash
14	Fuel	Unleaded gasoline with RQ-90 or higher

Table 2 *Physical properties of the IC engine structure components.*

S/No.	Part Description	Young's modulus (G), N/m^2	Density, (ρ_p) Kg/m^3	Poisson's ratio's (υ)	Remarks
1	Valve cover Top	7.1	2700	0.33	
2	Liner-piston Skirt	10.5	7700	0.28	
3	Crankcase - CR big end	19.5	7700	0.28	

3.1 Vibration system

The ICM-20608-G is the most recent 6-hub gadget Connected to past 6-hub gadgets, the ICM-20608-G offers lower power utilization, lower clamor, and a more slender bundle. Additionally, the ICM-20608-G offers an obligation cycled activity mode for the gyrator, that lessens spinner power utilization significantly or more relying upon office for debate goal (ODR) contrasted with prior 6-hub gadgets. It likewise offers about 20% lower commotion than past gadgets, and about 17% more slender bundle contrasted with past gadgets. The whirligig has a programmable full-scale scope of ±250, ±500, ±1000, and ±2000 degrees/sec. The accelerometer has a client programmable accelerometer full-scale scope of ±2g, ±4g, ±8g, and ±16g. Processing plant adjusted introductory affectability of the two sensors lessens creation line alignment prerequisites. The sensor connects with laptop with use wire cable and monitor the raw signal on it by using vibo-signal android program.

3.2 Test procedure

The experimental work was carried out on the IC engine components and the vibration acceleration responses in terms of time domain for the IC engine components were measured by three directions sensor (gyroscope sensor) which had been mounted on the center of the top of the valves cover as shown in **Figure 3**. The laptop with the software were handled for the analysis of vibration raw signal in time domain, while the conversion of the raw signal in time domain to frequency domain was done by using MATLAB software.

IV. RESULTS AND DISCUSSION

4.1 Results

The IC engine was tested by the above-described method. The results presented are an outcome of a series of experiments that were made on different IC engine components, where the vibration acceleration responses were measured for different engine components of valve cover top, liner-piston skirt and crankcase - CR big end. The modulus (M) was considered to be the measured and calculated values. **Figures 4 to 7** depict samples from the vibration acceleration measured in terms of time and frequency domains up to 20 s and 800 Hz respectively in narrow bands. The loading condition is 0.0 Nm and speed of 750 rpm in **Figures 4 and 5**. These tests are considered as reference (Base) values, while in **Figures 6 and 7** the load condition is 90 Nm and speed of 2000 rpm.

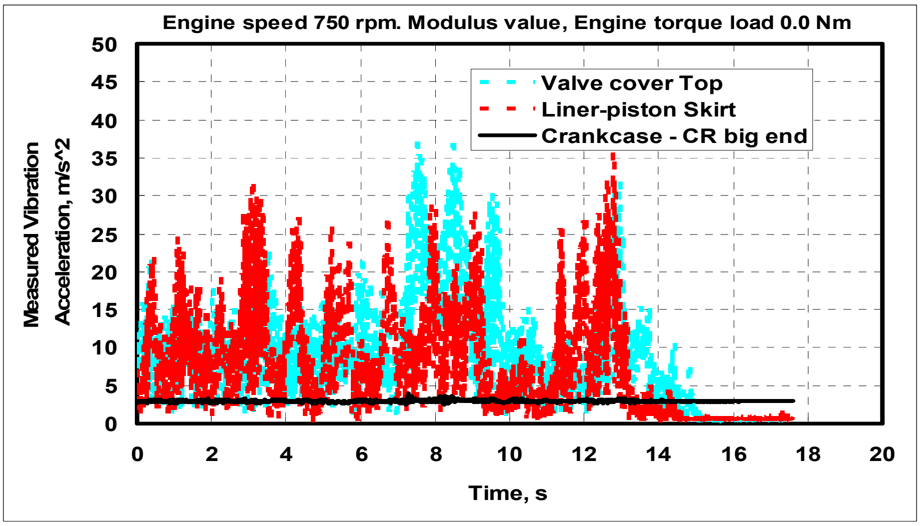

Fig. 4 Time history of the vibration level of IC engine components

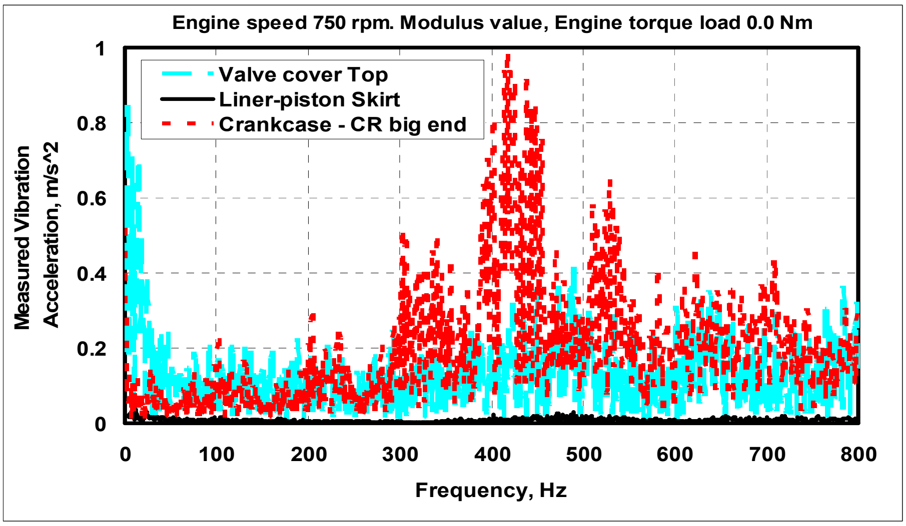

Fig. 5 Frequency domain of the vibration acceleration of IC engine components

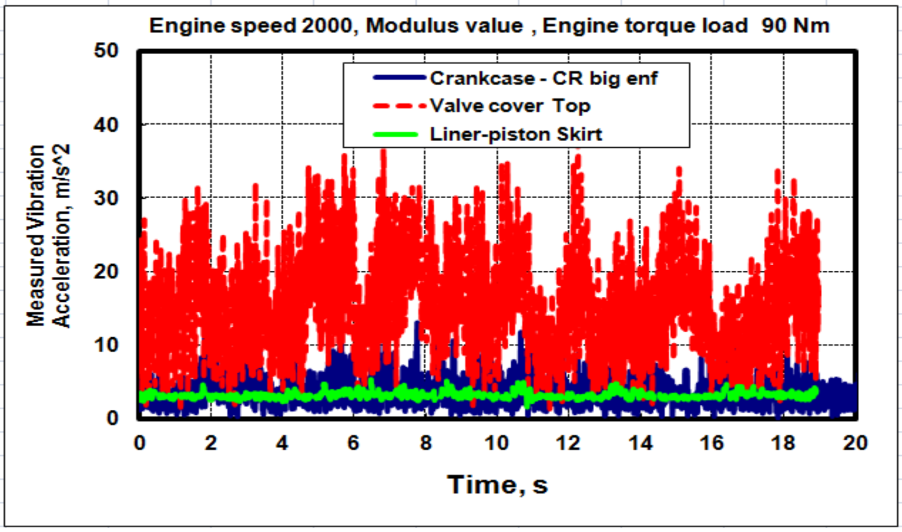

Fig. 6 Time history of the vibration acceleration of IC engine components at 750 rpm and 90.0 Nm

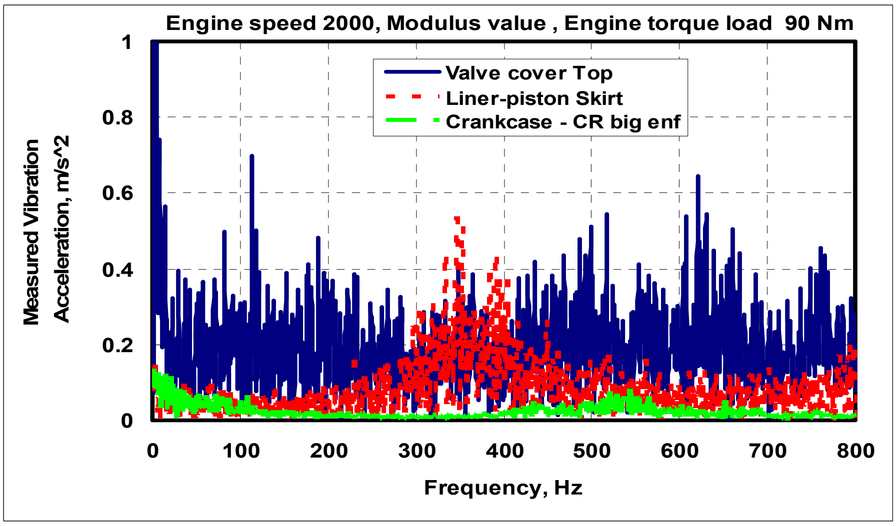

Fig. 7 Frequency domain of the vibration acceleration of IC engine components at 2000 rpm and 90.0 Nm

Figures 8 to 9 depict samples from the vibration acceleration measured in terms of time and frequency domains respectively. The results presented are an outcome of a series of experiments that were made on different IC engine components, where the vibration acceleration responses were measured based on three directions (X, Y, Z) in addition to the modulus (M) value in narrow bands. The torque loading condition is 90.0 Nm and speed of 2000 rpm and for crankcase - CR big end component. **Tables 3 and 4** tabulate the RMS value computed based on the relation (A!) presented in Appendix at different loading condition (0.0, 80.0 and 90 Nm) and at different IC engine rotational speeds of 750, 1000, 1500, 2000, 2500 rpm respectively.

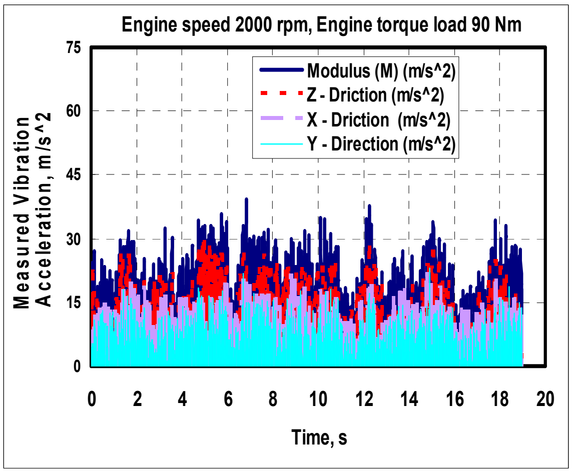

Fig. 8 Time history of the vibration acceleration at X, Y, Z, M for crankcase - CR big end component

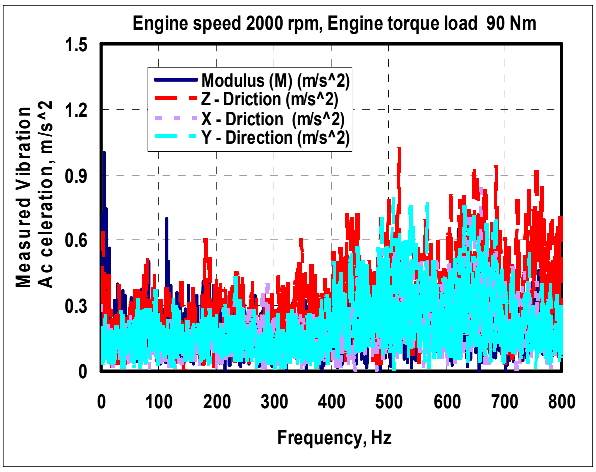

Fig. 9 Frequency domain of the vibration acceleration at X, Y, Z, M for crankcase - CR big end component

Table 3 RMS values for IC engine components in directions of X, Y, Z in addition to the modulus (M) at different loading conditions

S/No.	Measuring direction	Crankcase- CR big end 2000 rpm 0.0, 80, 90 Nm RMS, m/s^2			Liner-piston skirt 2000 rpm 0.0, 80, 90 Nm RMS, m/s^2			Valve cover top 2000 rpm 0.0, 80, 90 Nm RMS, m/s^2		
1	X	4.50	2.47	1.76	2.91	2.91	2.94	8.36	8.98	8.56
2	Y	2.16	2.26	1.85	0.87	0.84	1.02	9.66	9.06	7.66
3	Z	5.80	3.26	3.67	0.86	0.87	0.94	17.98	16.89	21.16
4	M	7.66	4.64	4.47	3.17	3.13	3.24	22.05	21.16	16.72

Table 4 RMS values for IC engine components in directions of X, Y, Z in addition to the modulus (M) at different rotational speeds

S/No.	Measuring direction	Crankcase- CR big end 90 Nm 750, 1000, 1500, 2000, 2500 rpm - RMS, m/s2					Liner-piston Skirt 80 Nm 750, 1000, 1500, 2000, 2500 rpm - RMS, m/s2					Valve cover top No load 750, 1000, 1500, 2000, 2500 rpm - RMS, m/s2				
1	X	- -	6.4	2.6	1.8	2.1	- -	2.8	2.8	2.9	2.9	6.8	7.3	7.2	8.4	11.
2	Y	- -	6.2	2.4	1.8	1.9	- -	0.7	0.8	0.8	0.9	4.8	6.0	7.9	9.7	10
3	Z	- -	6.9	4.6	3.7	3.9	- -	0.6	0.6	0.8	1.1	7.9	15	19	17	15
4	M	- -	11	5.8	4.5	4.8	- -	2.9	3.0	3.1	3.3	11	17	22	23	21

Figures 10 to 11 depict samples from the vibration acceleration measured in terms of time and frequency domains respectively. The results presented are an outcome of a series of experiments that were made on different IC engine components, where the vibration acceleration responses were measured based on the modulus (M) value in narrow bands. The torque loading condition is 90.0 Nm and for liner-piston skirt component. **Tables 5** tabulates the RMS value computed based on the relation (A!) presented in Appendix at different engine rotational speed (750, 1000, 1500, 2000 and 2500 rpm), at different torque loading and for all the IC engine components.

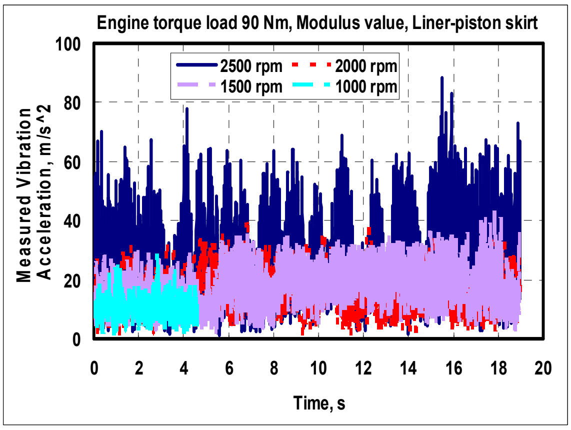

Fig. 10 Time history of the vibration acceleration for liner-piston skirt component

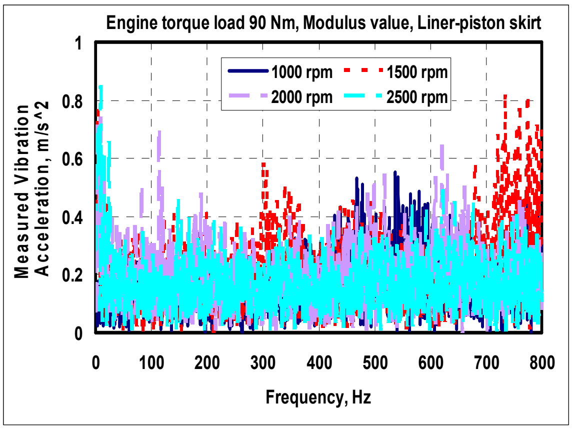

Fig. 11 Frequency domain of the vibration acceleration for liner-piston skirt component

Table 5 RMS values for IC engine components at different speeds for Modulus (M) values

S/No.	Speed, rpm	Crankcase-CR big end Modulus (M) 0.0, 80, 90 Nm RMS, m/s^2			Liner-piston skirt Modulus (M) 0.0, 80, 90 Nm RMS, m/s^2			Valve cover top Modulus (M) 0.0, 80, 90 Nm RMS, m/s^2		
1	750	3.40	--	--	2.94	--	--	11.49	--	--
2	1000	11.17	7.40	8.89	2.97	2.93	2.94	17.87	13.38	11.26
3	1500	0.84	5.46	5.82	2.97	3.01	3.05	22.60	17.60	19.48
4	2000	7.66	4.64	4.47	3.16	3.13	3.24	22.06	21.17	16.72
5	2500	6.60	5.16	4.85	3.30	3.26	3.76	21.77	19.58	30.51

Samples from the vibration acceleration measured in terms of time and frequency domains are depicted in **Figures 12 and 13** respectively. The results presented are an outcome of a series of experiments that were made on different IC engine components, where the vibration acceleration responses were measured based on the modulus (M) value in narrow bands. The rotational speed of 2000 rpm and for valve covers top component. **Tables 6** tabulates the RMS value computed based on the relation (A1) presented in Appendix at different torque loading conditions (0.0, 80.0 and 90 Nm) and at different IC engine rotational speeds and for all the IC engine components.

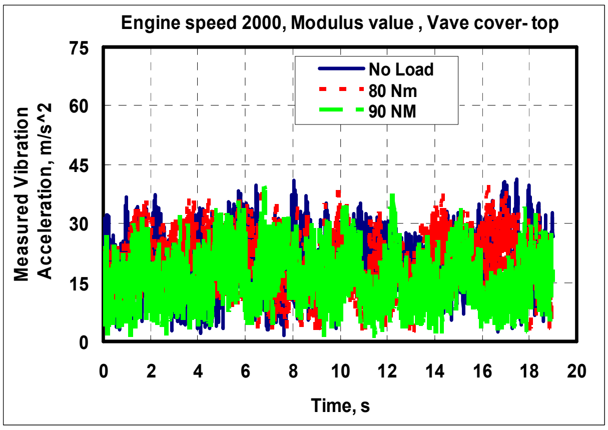

Fig. 12 Time history of the vibration acceleration for valve cover top component

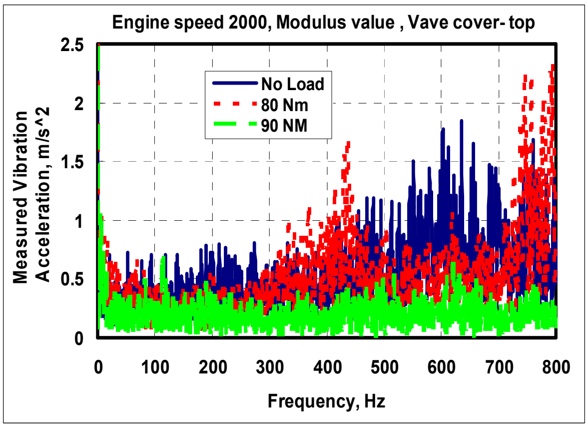

Fig. 13 Frequency domain of the vibration acceleration for valve cover top component

Table 6 RMS values for IC engine components at different torque loads for Modulus (M) values

S/No	Load Nm	Crankcase- CR big end Modulus (M) 750, 1000, 1500, 2000, 2500 rpm-RMS, m/s^2					Liner-piston skirt Modulus (M) 750, 1000, 1500, 2000, 2500 rpm-RMS, m/s^2					Valve cover top Modulus (M) 750, 1000, 1500, 2000, 2500 rpm-RMS, m/s^2				
1	0.0	43.6	11.2	0.8	7.7	6. 7	2.9	3.0	9.8	3.2	3.3	11.	17.	22.6	22.1	21.8
2	80.0	---	2.9	3.1	3.2	3.0	---	2.9	3.0	3.1	3. 3	--	13.	17.6	21.2	19.7
3	90.0	---	8.88	5.8	4.4	4.8	--	2.9	3.0	3.2	3. 8	----	11.3	19.5	16.7	30.5

In terms of RMS vibration acceleration, **Figures 14 and 15** depict the component vibration level (CVL) assessment which has been achieved by the developing the experimental technique at different engine rotational speeds and different torque loadings.

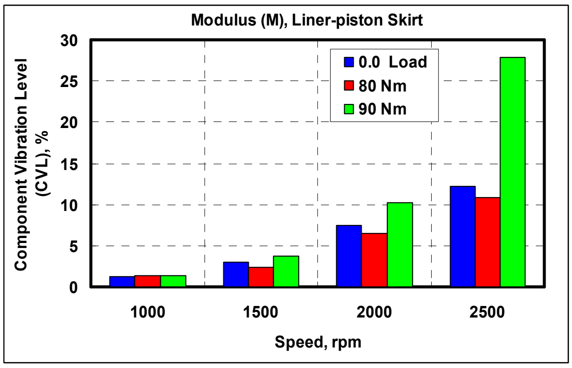

Fig. 14 Component vibration level (CVL) different engine rotational speeds

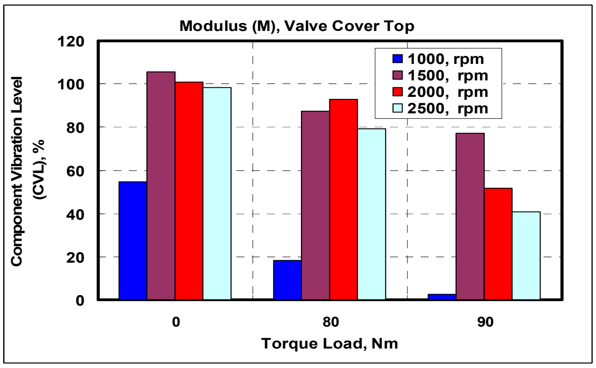

Fig. 15 Component vibration level (CVL) different torque loading

4.2 Discussion

Form the graphs of time and frequency domain of the (IC) engine it is clearly seen that the vibration acceleration responses at speed of 750 rpm without load (starting) of the engine are very high as shown in **the Figures 4 to 7** and the vibration reduces as engine starts running at higher speeds without any load. This can be concluded that the vibration acceleration responses will be very high at the time of ignition of the engine.

As the direction of the measurement is gradually changed like starts from X, Y, Z and M on the engine, the vibration levels are getting changed as shown in **Figures 8 to 9** and **Tables 3 and 4** clearly the trigger and spectrum graphs of all the three engine components on the engine.

As the speed is gradually increased like starts from 1000 rpm, 1500 rpm, 2000 rpm and 2500 rpm on the engine, the vibration levels are getting changed as shown in **Figures 10 to 11** and **Table 5** clearly the trigger and spectrum graphs of all the three engine components on the engine

As the loading is gradually increased like starts from 0.0 Nm, 80.0 Nm, and 90 Nm on the engine, the vibration levels are getting changed as shown in **Figures 12 to 13** and **Table 6** clearly the trigger and spectrum graphs of all the three engine components on the engine.

The figures indicate that CVL percentage is increased as the rotational speed is increased (**Figure 14**) and at all the torque loadings (**Figure 15**) considered. Due to the increase of the vibration which cause damage to structures and machine sub-assemblies, resulting in disoperation, excessive wear, or even fatigue failure. Vibration may have adverse effects on human beings. The primary effects are task performance interference; motion sickness; breathing and speech disturbance; and a hand-tool disease known as "white finger", where the nerves in the fingers are permanently damaged, resulting in loss of touch sensitivity

In this work the vibration levels of an IC engine are very high at the very start of an engine. That means at the ignition level of the engine there is more vibration occurs that shown in the graphs of time and frequency domain. The above Figures show the vibration levels of an IC engine.

V. CONCLUSIONS

1. The chosen measurement technique was based on vibration signature analysis. The sensors are attached to the engine by a magnet without an intrusive approach. The experiments that were done showed that there is a more vibration at the starting of engine and that transmitted in whole body of the vehicle and that may cause to discomfort of the passengers. Moreover, the results show that the system is capable of identifying fault engine operation and can direct the user to the source of the problem.

2. The results presented here for engine working without load in steady engine speed and also various load and speed conditions. But this technique works for a variety of engines speeds and loads, under steady operation conditions or under varying operation conditions. This method implemented as a feedback to the ignition system or as a test system for an engine.
3. The vibration of engine were measured and analyzed for the run engine. The amplitude of maximum vibration is found to be related to the rate of pressure rise and the maximum pressure in the cylinder during ignition. When the rate of pressure rise increases the vibration amplitude also increases.

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Appendix
High Order Statistics Parameters
A.1 Root Mean Square

In this paper only, high order statistics of root mean square (RMS) is used. This feature is usually called time-domain feature. RMS is a kind of average of signal, for discrete signals, However, the RMS value is defined as:

$$RMS = \sqrt{\frac{1}{N}\sum_{n=1}^{N}(x(n)-\overline{x})^2}$$

$$\overline{x} = \frac{1}{N}\sum_{n=1}^{N} x(n) \qquad \text{(A1)}$$

A.2 Vibration Severity Assessment

From the discussion of these results in this work, descriptive and higher order statistics (CVL) indices have been created intensive interest. The RMS values for the measured signal calculated based on relation (A1) was found to be a better indicator used to deal with CVL. However, the RMS values of the vibration accelerations measured were used to evaluate the vibration severity of the IC engine components. Component vibration level (CVL) can be calculated based on the following equation

$$CV \rightleftarrows L, (\%) = \frac{(RMS)_{Actual} - (RMS)_{Base}}{(RMS)_{Base}} \qquad \text{(A2)}$$

Where:

$CVL, (\%)$ = Component vibration level
$(RMS)_{Base}$ = RMS value for base condition
$(RMS)_{Actual}$ = RMS value for actual condition