PERFORMANCE ANALYSIS OF TWO CYLINDER DIESEL ENGINE VIBRATIONS USING DIFFERENT ENGINE MOUNTINGS

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ABSTRACT

The effective approach of reducing the vibration transmitted from the engine to the supporting structures is by the employment of Engine mounts. A 2 cylinder 4-stroke cool direct injection ICE with a displacement volume of 998cc, compression quantitative relation eighteen.5:1, developing seventeen power unit at 2000 rev was used for this analysis work. The engine is fitted with typical fuel injection system, that includes a five holed nozzle of zero.262mm separated at 1460, inclined at associate degree angle of 600 to the cylinder axis. The widget gap pressure counselled by the manufacturer was 250 bar. 2 forms of mountings that area unit utilized in this project area unit rubber mounting and hydraulic mounting. At totally different masses (third, 25%, 50%, seventy fifth and 100%) of the utmost load (90NM@3600RPM) and rpm(1200) vibrations area unit measured within the Dewe soft package victimisation varied sensors. currently the rubber mounting is replaced with the Hydraulic Mounting and therefore the readings area unit noted within the package. The vibration sensors used for measurement the vibrations in FFT instrument and measuring instrument. These sensors area unit connected to the Dewe package victimisation analog cables and adapter the readings obtained area unit enforced and compared in an exceedingly Graph victimisation stand out package. So the appropriate mounting that may be used for reducing the vibrations in associate degree engine is analysed.

Keyword: *Diesel engine, Unbalanced forces, Vibration, Engine mount.*

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1. INTRODUCTION

The internal combustion (IC) engine is the concentrated mass in vehicle and if not properly designed it will cause vibrations and transfer to the supporting structures ride comfort, driving stability and drivability are important factors for the performance of a vehicle and are affected by the engine vibrations. Because of the environmental considerations, as well as changes in consumer preferences regarding vibration induced must be reduced. Vibration behaviour of an IC engine depends on unbalanced reciprocating and rotating parts, cyclic variation in gas pressure, shaking forces due to the reciprocating parts and structural characteristics of the mounts. Engine vibrations are caused due to the reciprocating and rotating masses of the engine. The variations of inertial forces are due to the combustion and the compression differences of the piston cylinder arrangement during their operation. The engine inertial forces leads to the unbalanced forces of the engine and they are quiet varying with respect to speed, fuel supply and combustion characteristics of the fuel. To predict the vibration output of an engine and to minimize the possible durability and consumer perceived quality problems associated with engine vibration, a robust and accurate design and simulation model is needed. To reduce the engine vibration proper mounting must be provided as dampers at the interface of the engine and chassis.

The vibrations caused at the engine are two types they are torsional and longitudinal vibrations. Engines always have some degree of torsional vibration during operation due to their reciprocating nature. The rotation of crankshaft of an engine increases the cylinder pressure as the piston approaches top dead centre (TDC) on the compression stroke. Ignition and combustion increases the pressure just after TDC and the pressure starts to decrease when the piston moves down to bottom dead center (BDC). The pressure on the piston generates the tangential 3 force that does useful work and increases the rotational speed of the crankshaft during this combustion stroke, whereas the compression stroke decreases the engine's angular velocity. TO identify the various methods used and assumptions followed to find and reduce the engine vibrations a survey was made on the Engine.

1. Rigid body modelling
2. Vibrations and Mountings

1.1 ENGINE RIGID BODY MODELING:

IC engine consists of different components such as engine block, engine head, piston, connecting rod, crank shaft, flywheel and cam shaft, valves, manifolds, pulleys etc. Some of parts are identified by the researchers and engine manufacturers as the vibration producing parts. The piston connecting rod, crank shaft and engine block are the major components which produces unbalanced forces during the operation. As they are interconnected together the forces are transferred to the engine block and hence to the supporting structures. Many mathematical rigid body models were proposed by the researchers using multibody modeling of the engine structure to calculate the unbalanced forces from the engine. Sudhir Koul et. al [4], developed a mathematical model based on the structural dynamics that includes frame, power train assembly, swing arm assembly and engine mounting system. The authors developed two models with six degrees of freedom rigid body modeling of the flexible frame. In the first model, it consists of finite element modeled stiffness matrix such that the nodes of the frame connect the other sub-system. The other is the respective dynamic model of the frame and the swing arm.

1.2 VIBRATIONS AND MOUNTINGS

Engine produces the vibratory forces due to the unbalanced forces from the engine parts during the operation. The vibration caused by the engine at the supports is torsional vibration and the longitudinal vibration. The torsional vibration is caused at the crankshaft due to the fluctuating engine combustion pressures and engine loads. The longitudinal vibrations are caused at the block and the mounts by the reciprocating and rotating parts of the engine. Nyman et.al concerned with minimization of engine vibration in the mounted 4-cylinder internal combustion engine. They analyzed the mathematical model and the balancing mass and the lead angles were taken as the design variables. The objective function used in their research is the vibratory forces from the engine, transmitted to the engine mounts. The objective function is to be minimized to minimize the vibratory output of the 7 engine and the Leap frog optimization algorithm is employed to minimize the objective function. Conti and Bretl presented a new method for determining an analytical model of rigid body on it mounts and the method is based on data acquired. They used modal experimental data from an artificial excitation of vibration of test of the article suspended to ground through mounts as input to the model and the output is rigid body mass properties and stiffness of the mounts. The extracted modal data for these six modes is input to a least square algorithm, which was used to compute mass and Centre of gravity of location, mass moments and principal axes of inertia and tri-axial stiffness of mount. Chung–Ha and Clifford.G.Smith Shu, presented simplified method to determine the vibrational amplitude developed by a 4-cylinder engine when supported on viscoelastic mounts. The simulations were created to seek out mount displacements throughout taxi, climp, cruise and tight conditions of the air craft. The authors declared that the warmth transfer rate between recess and outlet air was excessive and can be thought of in future study. J.Christopherson and G.Nakhgie Jazar[9], investigated the linear and non-linear aspects of 2 distinct passive hydraulic engine mounts with floating decoupler and with direct decoupler. each the decoupler mechanisms were wont to management the amplitude dependent behavior of the mount. within the 1st kind, the fluid is forced through the inertia track attributable to relative motion between the engine and therefore the chassis and this style depends upon the acceptable combination of inertia track and decoupler gap size. within the second kind, the decoupler directly connected to the engine mount and its motion is directly controlled by the engine motion of the engine and therefore the chassis. The linear and non-linear mathematical models of the each kind were simulated and from the results it's been known that the direct decoupler mount exhibits rock bottom transmissibility in low frequency domains whereveras the floating decoupler shown higher performance because the excitation frequency will increase. The non-linear response solutions of the each mounts were valid by direct comparison with the linear counter components and it's been known that the similarity between the answer regions sufficiently removed the resonance within the non-linear modeling They additionally toughened difficulties within the mathematical modeling of the direct decoupler J.S.Lee and S.C Kim[11], centered on the performance sweetening of engine mount rubber(EMR) by adopting a type a style optimisation approach. The optimum style was arrived by considering the fabric stiffness and fatigue strength of a rubber.

2: ENGINE SPECIFICATIONS:

A single cylinder 4-stroke water-cooled direct injection diesel engine with a displacement volume of 1670cc, compression ratio 18.5:1, developing 21 kW at 2000 rpm with a dynamometer was used for the present research work. The specifications of the engine are listed in Table 4.1 The engine is fitted with conventional fuel injection system, which has a 5 hole nozzle of 0.262mm separated at 146 degrees, inclined at an angle of 60degrees to the cylinder axis.. The injector opening pressure recommended by the manufacturer was 250 bar.

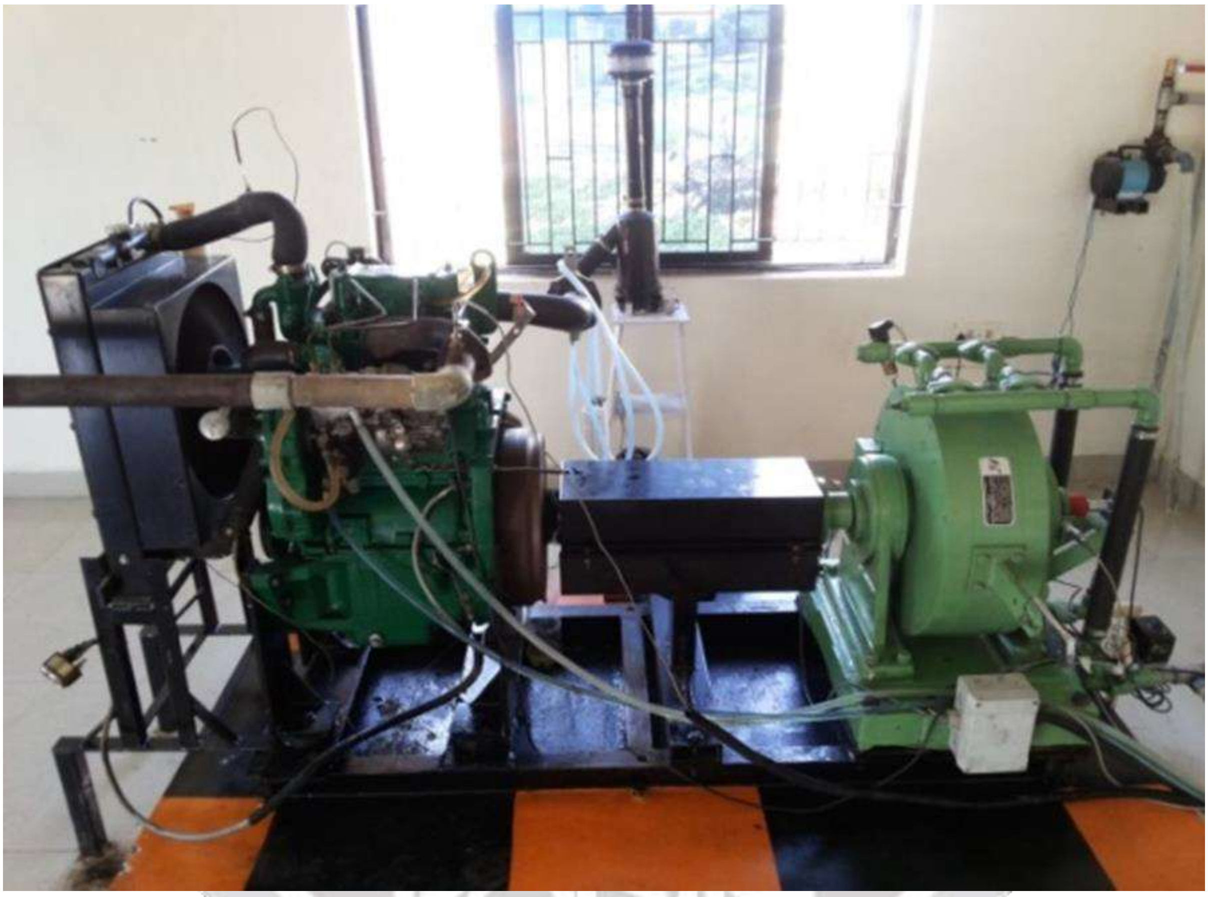

Fig -1: Experimental setup

The Bosch fuel pump which is fitted on the engine enables the automatic regulation of the engine speed. The combustion chamber is hemispherical in shape with the overhead valve arrangement operated by push rods. A provision was made to mount a piezoelectric pressure transducer flush with the cylinder head surface in order to measure cylinder pressure. The injection system of the engine was periodically cleaned and calibrated as recommended by the manufacturer. The specifications of the test engine are given in Table 1.

MODEL	S 217
Capacity	21 kW (28 bhp @ 2000 rpm)
Type / Configuration	Vertical in-line Diesel Engine
Bore	91.44 mm
Stroke	127 mm
No. of Cylinders	2
Displacement	1670 cc

Compression ratio	18.5:1
Cycle	4 Stroke
Rotation	Clockwise (viewed from front)
Aspiration	Natural
Combustion System	Direct Injection
Fuel Pump	MICO Bosch In-line Pump
Governing	Mechanical
Engine Starting System	Electrical
Cooling System	Water
Electrical System	12 Volts (Dynamo/Alternator)
Flywheel Housing	SAE 1 or SAE 3
Flywheel	Can be made to suit application
Weight (Bare Engine)	200 kg

Table -1: ENGINE SPECIFICATIONS

3. MOUNTINGS

The unbalanced forces produced from the engine are transferred to the engine supporting members and causes the structure borne vibrations. To reduce the vibratory forces from the engine to the structures, the engine is supported by the damping members called vibration isolators (engine mountings). The mountings are the final most sources to reduce the vibratory forces by its damping property.

3.1 Engine Mounts:

To achieve the best vibration isolation for the powertrain, a mounting system is used to mount the powertrain in place. The mounting system will provide isolation That will in turn minimize the transmitted forces to/from the engine to the frame. On the other hand, it will also prevent engine bounce caused from shock excitation. transmitted forces to/from the engine to the frame. On the other hand, it will also prevent engine bounce .

Fig -2: Engine mounts

3.2 ELASTOMERIC MOUNTS:

Elastomeric mounts, which are made of rubber, have been used to isolate engines since 1930s. A lot of changes have been made over the years to improve the performance of the elastomeric mounts for proper vibration isolation, elastomeric mounts are designed for the necessary elastic stiffness rate characteristics in all directions. They are maintenance free, cost effective and compact the elastomeric mounts can be represented by a Voigt model which consists of a spring and Viscous damping as shown in Fig. 1.1. It is difficult to design a mounting system that satisfies a broad array of design requirements. A mount with high stiffness or high damping rates can yield low vibration transmission at low frequency, but its performance at high frequency might be poor. On the other hand, low stiffness and low damping will yield low noise levels but it will induce high vibration transmission. A compromise is needed to obtain balance between engine isolation and engine bounce. In order to achieve low vibration transmissibility, the mount stiffness must be as low as possible. However, this causes increased static deflection. Lower damping is also desirable for lower transmissibility at higher frequency range.

Fig -3: Elastomeric mounting

3.3 Passive Hydraulic Mounts:

Hydraulic mounts were first introduced in 1962 for use as vehicle mounting systems. Since then, their popularity has improved for two reasons. The first one is that the current vehicles tend to be small, lightweight and front wheel drive with low idle speeds. The second one is that the hydraulic mounts have developed in to highly tuneable devices. Three types of hydraulic mounts are in use these days and. these are: hydraulic mount with simple orifice, hydraulic mount with inertia track, and hydraulic mount with inertia track and decouple. A general schematic diagram of the hydraulic mount is 5 Liquid Rubber Orifice Secondary Rubber Air shown in Fig although there are differences between orifice and inertia track mounts, all of them cause damping at low frequency ranges. These mounts can be tuned to have high damping at the shock excitation frequency which is used to reduce the vibration levels. The dynamic stiffness of these mounts is usually higher than that of the elastomeric mounts.

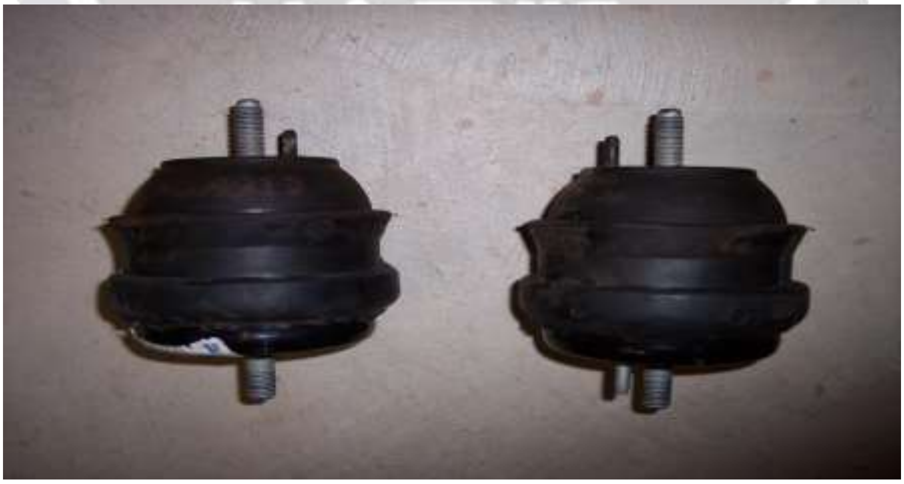

Fig -4: Passive Hydraulic Mount

4. VIBRATION SENSORS:

4.1 FFT ANALYZER:

FFT analyser 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 analyser 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. The name of the FFT analyser used for the experiment is NI 9234. By using FFT enabled to investigate the degree of chance that occurred at what level of frequency and the frequencies generated from a particular position. Frequency analysis (observing the waveform in the frequency domain) makes detection of even slight anomalies possible.
NI 9234 ±5 V, IEPE and AC/DC Analog Input, 51.2 kS/s/ch, 4 Ch Module

- 51.2 KS/s per channel maximum sampling rate; ±5 V input
- 24-bit resolution; 102 dB dynamic range; anti-aliasing filters
- Software-selectable AC/DC coupling; AC-coupled (0.5 Hz)
- Software-selectable IEPE signal conditioning (0 mA or 2 mA)
- Transducer Electronic Data Sheet smart sensor compatibility
- -40 °C to 70 °C operating, 5 g vibration, 50 g shock

Overview The NI 9234 is a 4-channel C Series dynamic signal acquisition module for making high-accuracy audio frequency measurements from integrated electronic piezoelectric (IEPE) and non-IEPE sensors with NI Compact DAQ or Compact RIO systems. The NI 9234 delivers 102 dB of dynamic range and incorporates software-selectable AC/DC coupling and IEPE signal conditioning for accelerometers and microphones. The four input channels simultaneously digitize signals at rates up to 51.2 kHz per channel with built-in anti-aliasing filters that automatically adjust to your sampling rate.
Recommended Software NI sound and vibration analysis software, including the NI Sound and Vibration Measurement Suite and the NI Sound and Vibration Toolkit, provides signal processing functionality for performing audio measurements, fractional-octave analysis, frequency analysis, transient analysis, and order tracking. NI analysis software features NI Sound and Vibration Assistant interactive software for quickly acquiring, analysing, and logging acoustic, noise, and vibration data. With a configuration-based, flexible measurement library and open-analysis capability, the Sound and Vibration Assistant is designed for quick data capture through a unique software-based measurement approach to create customized applications.

4.2. Accelerometer:

Accelerometer is a Piezo-electric accelerometer and it is considered as the standard vibration transducer for machine vibration measurement. Data capture regarding the vibration emitted by a machine, or other body, begins with the sensor. The accelerometers shown in Fig consist of a piezoelectric crystal which has a mass attached to one of its surfaces. When the mass is subjected to a vibration signal, the mass converts the vibration (acceleration) to a force, this then being converted to an electrical signal. This is the basis of the "accelerometer". The accelerometer output may then be processed to provide the instantaneous vibration and displacement signals.

TYPE OF MEASURE	RANGE / LIMIT in g
1. Acceleration range g	±50 g
2. Acceleration limit	±500 g
3. Transverse limit	±500 g

4. Threshold grams 0.0025 Sensitivity	100 mV/g
5. Resonant frequency (nom.) kHz	44 40 kHz
6. Frequency response, ±5% Hz	1 to 10k 1 to 7k Hz
7. Time constant (nom.)	0.5 s
8. Shock (1 ms pulse), max.	5,000 g
9. Transverse sensitivity (nom.) %	≤1.5
10. Linearity %	±1
11. Operating temperature range°F	–65 to 250 (@ 4mA excitation)
12. Temperature coefficient of sensitivity	–0.1
13. Output impedance Ω	<100
14. Power supply mA & VDC 2 to 20(constant current)	18 to 30
15. Weight	4g

4. CONCLUSION:

Based on the literatures from varied analysis articles for the engine varied multibody modelling of engines, engine vibration testing and measure techniques used and therefore the varied mountings accustomed scale back the vibration and up to date technologies fictitious and adopted. The subsequent observations are created on the on top of literature survey:

• Many of the researchers thought of the engine as rigid body model not the versatile one.

• Meagre range of articles solely took the forces from the engine to the mount for the mathematical modelling as 2 manner coupling.

• Less analysis were created by combining the each the torsional and longitudinal vibrations.

• For the determination of the engine vibration calculations most of them weren't thought of the engine combustion force variations.

• In the vibration modelling of the vehicles the engine vibration modelling and road vibration modelling were thought of severally not as single model.

• The recent developments of the mounts on the hydraulic and magnetic attraction mounts were used because the isolator not the standard rubber mounts.

• A straightforward and versatile model needed to be developed which incorporates the multibody dynamics of engine and dynamics characteristics of the rubber or hydraulic mounts to analyse the vibrating characteristics of the ability plant.

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