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Parametric Analysis on Selection of Isolation Material for Vibration Isolation of the Engine Test Bed

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ABSTRACT: *A machine in operation without vibration is not possible in real life. Mechanical vibration is the term used for the dynamic behavior of a system under self-loading or external excitation. The response of unbalance forces measured in terms of displacement velocity and acceleration in the normal practice for maintenance engineers to monitor the structural health of the dynamic system. Each mechanical system under excitation has some finite life in terms of operating hours or operating cycles. The failure of a dynamic system is supposed to occur due to fatigue under the live loading condition. This research study was conducted with the motivation to measure the response of internal combustion engine test bed under excitation speed of low 1200rpm, medium 1800 rpm, and high level 2400 rpm. The results of dynamic response in terms of acceleration, velocity, and displacement were measured at the centre corner and near engine for each specified speed with and without isolation. The response of vibration in terms of displacement was mitigated upto 90% at a speed of 1200 rpm with wood as isolation material The response in terms of velocity and acceleration were also reduced at each excitation speed with all isolation methods the mitigation was maximum by 50% at1200 rpm, wood isolation and 64%at 2400 rpm with rubber and wood isolation for velocity and acceleration respectively. The stable operation of the engine test bed was improved. The structural health of the engine test bed was improved due to the attenuation of dynamic responses.*

Key Words*:* *Excitation, Dynamic behavior, Vibration Isolation, Structural Health, Noise Pollution*

I. INTRODUCTION

Mechanical vibration is the oscillating movement of dynamic systems, following the scientific definition. In engineering language, however, Mechanical Vibration deals with the relationship between the forces acting on the mechanical system and the oscillatory motion of the mechanical system on a point within the system [1]. The theory of vibration involves the oscillatory motion of the bodies and forces associated with them. Any fundamentally dynamic structure has a certain form of vibration in it [2]. Vibration in the device can cause some serious damage, such as a reduction in performance, a reduction in life or damage to parts or the whole unit, and by creating noise that is not good for workers, etc. The primary one, the mass is inherent in the body, and the elasticity is due to the relative motion of the parts of the body. The system is easy or complicated. It can take the form of a system, a machine, a group of machines or components. The motion for oscillation could be unacceptable or required to carry out a task [3]. The designer aims to regulate the vibration when rejected and to enhance the vibration when useful. Object vibration of the machine may lead to the loosening or failure of the parts. The vibration is required for shakers of foundries and vibrators of testing machines [4]. The working of a large number of instruments relies on proper vibration regulation. The main aim is to analyze the otoscopic system. A spring force exists if the spring is deformed. The work done in deforming a spring is transformed into potential energy; that is, the strain energy stored in the spring [2].

The damper has neither mass nor elasticity. Damping force exists only if there is a relative motion between the two ends of the damper. The work or energy input to a damper is converted into heat. Machines with repetitive disturbing forces such as engines, motors, turbines, etc. often have vibration problems [5]. Serious vibration problems may cause damage, malfunction, or even failure of the structure or machine itself or machine parts their selves. Vibration causes interruption of production, reduction of working lives of machines, loss of power, and energy. Vibration also causes an unpleasant feeling or noise that can permanently damage human ears [6]. Vibration can cause a system defect that leads to a complete system or a failure of a particular component. To achieve a reliable experimental result, the vibrations in the engine test bed caused the experimental results to vary and faulty results to be checked. Fig.1 indicates the effect of vibration on the health of machine components; the failure of dynamic systems occurs mainly due to continuous vibrations. The vibration generated from the unbalance forces within the dynamic system or some external excitation effects is transmitted to all the components of a dynamic system and initiates cracks within the structure the cracks are then propagated to surface of the structures which are visible as in the fig below, these crack lead to complete failure of the components [7].

Fig1. Failure of Machine Components Due to Vibration

Vibrations cause fatigue failure or damage to the supporting engine structures. The impact entails excessive forces and, therefore, excessive stresses, which can cause the mechanisms to vibrate and fail early. The purpose of this experimental study was the measurement of vibration of the diesel-powered internal combustion engine test bed and supported, including bolted joints, holding clamps. The health and life of operating machines can be increased by the proper assembly provided to reduce vibration. Dampers are sometimes available at the engine and base interface. The internal combustion engine is the concentrated mass, which results in high-frequency, low amplitude vibrations, transfer to the supporting structures when not properly fastened [8]. Figure 2 indicates two isolation materials: the wooden pad and the spring assembly. The system under excitations when directly is in contact with the rigid foundations receives back the reaction forces, which lead the system towards several modes of vibration, having different frequency ranges. The isolation materials such as wooden pad and flexible springs provide a medium of energy dissipation between the vibrating machines and the rigid foundation [9]. The reactive response from the foundation to the system is minimized, which in turn improves the stable operation and life of the machine component.

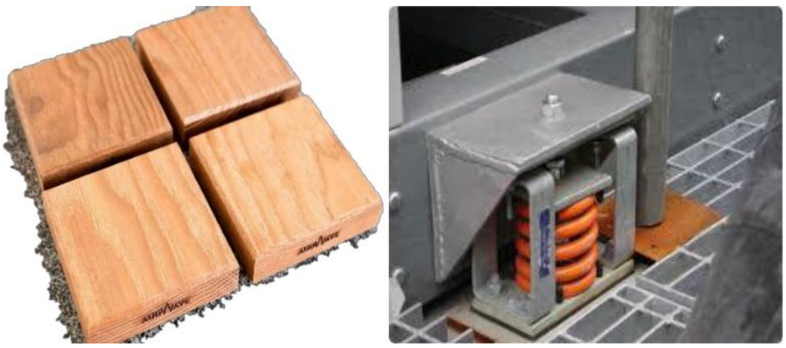

Fig2. Vibration Isolation Methods

Because of various working conditions, a nonlinear full-body vibration with a large frequency band may be generating complicated non-linear vibration [10]. Because conventional vibration control methods are usually developed for external excitement and based on linear models, their performance is not easy to improve. In this study, the longitudinal torsional vibration of the engine system decreases, through a new, semi-active vibratory absorption method that is independent of external stimulation and can handle the non-linear vibration problem [11]. It uses nonlinear connection and modal interaction sufficiently to transfer and dissipate vibration this study; a robust adaptive control method is employed for an active engine mount in a six-degree-of-freedom model of the engine on the mounts to improve the vibration behaviour of the engine. The vibration isolation performance and robustness of the employed robust adaptive controller are compared with a robust and adaptive control technique [12]. Besides, the effectiveness of robust adaptive control is evaluated in transient conditions (accelerating and gear change conditions). In this regard, a dynamic model for the engine supported by rubber and active mounts and its governing equations they're introduced. Good adaptive control is then developed by choosing a proper reference model, namely the Model Reference Adaptive Control (RRAC) technology, based on the gradient method with adjustment. Besides, for the active mount, robust control is implemented, namely the H1control device and adaptive control, MRAC. The simulation results show that the powerful MRAC has enhanced control performance in comparison with the H1-scheme in that the transmitted force to the chassis [13].

Furthermore, MRAC may be divergent and inconsistent in the face of significant uncertainties. But in the face of major uncertainties, the robust MRAC is robust. Besides, robust MRAC does not work through many computationally expensive internal combustions engine vibration models are available, simple and computationally efficient tools are required for preliminary design work [14]. The unbalanced strengths of rotating and reciprocal elements are the main sources of motor vibration, which in turn reduces consumer and market motives ' durability and reliability. For the engine and the interface between the mounting mounts, therefore, considerable vibration isolation is needed, and this can be achieved using rotating balance disks at both ends of the shaft. Based on the engine vibration, the masses of the balancing disks and their led angles affect the effectiveness of vibration insulation measured by the displacement of the motor mount. The balanced masses and lead angles of the balancing disks reduced the mounting of the motor [15]. The movement equations in an internal combustion engine for the main components are as a result of this developed using a recursive formulation. The (rigid) motor block, pistons, connection bars, (flexible) cranking shaft, balance shafts, main bearings, and mountings for the engine are part of the components [16]. Relative coordinates are used that satisfy all constraints automatically, leading to the minimum number of ordinary varying motion equations. The derivation of motion equations is automated using computer algebra to generate the subroutines (C or Fortran) for numeric integration automatically. The whole automated process forms the foundation. The transmission of vibroacoustic energy from an internal combustion engine (ICE) to its environment depends largely on the mounting process, the available transport pathways, and the design of the body and the surrounding structures [17]. This applies particularly to low-speed motors in adjoining zones, which create noticeably weak noise but strong low-frequency waves where energy damages the health, comfort, and safety of humans, especially when exposed to source over time.

This research study was conducted with the motivation to measure the mechanical vibration resulted due to the rotation of output shaft and couple with eddy current dynamometer, mitigate these mechanical vibrations using isolation materials like wood, rubber, spring and combinations of these isolation materials, and predict the health, stable operation and minimized noise pollution. Improvement in the stability and operating noise level of the engine test bed were also the expected outcomes of this research

II. BACKGROUND

From a single-degree-of-freedom model used to illustrate the concept of vibration isolation, a method to transform the design for suspension into a design for the robust controller is presented. Fractional differentiation is used to model the viscoelastic behaviour of suspension[16]. The use of fractional differentiation not only permits optimization of just four suspension parameters, showing the ‘compactness’ of the fractional derivative operator, but also leads to the robustness of suspension performance to the uncertainty of the spring mass [18]. As an example, engine suspension is studies based on the Halbach magnetic array, the active and passive hybrid vibration isolation technology for the WD618 diesel engine is studied [2].

Firstly, the active and passive hybrid vibration isolators based on Halbach magnetic array is designed. The content includes design requirements, scheme design, and design method; design suitable power amplifier for isolators and verify whether the performance index requirements are met and test the resistance, inductance, stiffness, and static thrust of isolators; the vibration isolation effect of the vibration isolator was tested through a single-degree-of-freedom vibration isolation experiment[8]. The vibrations generated in the IC engine in the automotive industry affect vehicle performance and comfort while driving. Conductivity, stability, and comfort are the factors affecting the continuous vibration of the IC engine. All these considerations are reduced as the vibration of the engine increases. Both the alternating and revolving parts of the motor produce constant vibration in the internal combustion engine. Both reciprocal and rotating sections generate the inertial forces. The compressive and combustion properties in the engine change these inertial forces [6]. Leandro M Campeiro [et al.] in the year 2017 studied the effect of vibration and noise on the health of machine components and developed an impedance-based system for prediction of damage in an aluminum bar [19]. Fluid solid interaction is a common phenomenon in fluid transmission and flow systems. C wang[et al.] investigated the relationship between flow, pressure, friction, and flow-induced vibrations in a multistage pumping system. The results of the study indicated a strong dependence on the stator and rotor interaction unbalances [20]. Piezoelectric shunt damping is the passive mechanism of vibration abatement Widely used for low-frequency vibrations. J.A. Gripp in 2018 investigated the location and effect of passive shunt piezoelectric resonant circuit to minimize the vibration [21]. N.S. Ahirrao presented the effect of vibration of structures and fatigue analysis of test bench under vibration at an international conference of materials, manufacturing, and design [22]. Hiroyuki Kayaba Nikon Corporation, Japan, presented a fast Fourier transform-based noncontract vibration measurement mechanism at the IEE conference [23]. D. Goyal in 2016 presented some noncontact vibration metering methods like ANN, ARMA, HMM, and spatial domain features [24]. In the year 2018, Christophe Marchetto investigated the vibroacoustic behavior of the tunnel-like simulated model under turbulent flow conditions [25]. S.J. Rothberg [et al.] proposed and investigated a laser doppler velocity measurement device for structural health monitoring of mechanical systems having multi-input and multi output [26]. Timothy J. Beberniss developed and presented an advanced high-speed image correlation vibration measurement system [27].

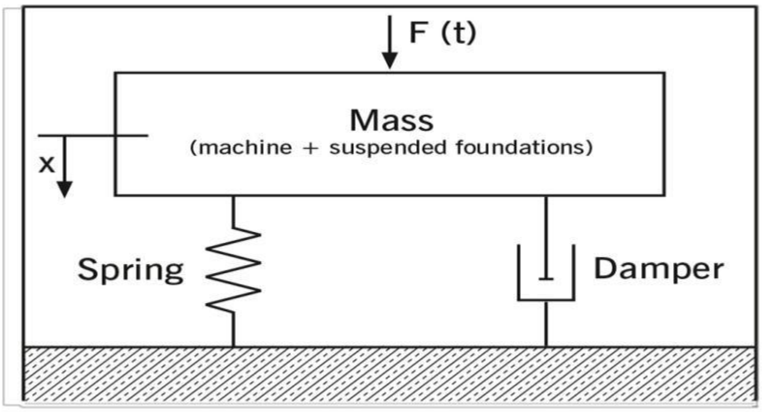

Fig.3 Basic SDOF vibration system approximation

The schematic diagram above indicates a single degree of freedom system attached to a rigid foundation. The mathematical model may determine the forced damped vibration approximation response of the system as in equation1[28

$$m\ddot{x} + cx + k = F_0 cos\omega t \qquad (1)$$

$$X = \frac{F_0}{\sqrt{(K - m\omega^2)^2 + C\omega^2}}$$

F_0=Initial force N

X= amplitude od vibration mm

K=stiffness N/mm

C=damping coefficient

M=Lumped mass kg

$$\omega n = 2 \times \pi \times f \qquad (2)$$

$$\omega n = \alpha^2 \sqrt{\frac{EI}{A\rho L4}} \text{ cyc/s} \qquad (3)$$

The natural frequency of the structure with length L, Elasticity E, and cross-section area A can be evaluated from equation 3. Equation 2 can be used to determine the natural frequency of the system [29]

III. MATERIAL AND METHODS

The main objective of this research study was to experimentally evaluate the effect of excitation speed and passive vibration isolation method on the vibration parameters of an internal combustion engine test bed. The experimental evaluation of reduction in the displacement velocity and acceleration of vibration in the test bed with wood rubber spring and combination of wood &rubber at excitation speeds of 1200, 1800, and 2400 rpm was performed under steady-state condition. This section of the research article contains materials used for vibration isolation, the methodology of the experimental study, and the design of experiments. Pareto chart indicating the significance of the various control variables on the output results is also presented. The factorial regression models for the response of the system with and without vibration isolation are presented in this section.

3.1 Materials

The experiments were conducted on the internal combustion engine test bed having a frame made of a galvanized iron solid sheet of gauge 14. The internal combustion engine with specifications as in table1 and eddy current dynamometer of ratted load capacity of 746 W was mounted on the test bed with threaded fasteners. The parameters of vibration like velocity, displacement, and acceleration were measured with vibrometer model UNI-T, UT-315 with specifications as in table 2

Table 1 Specification of Engine

Rated Power	2.8 KW	3.1 KW
Continuous Power	2.5 KW	2.8 KW
Output Engine Speed	3000 rpm	3600 rpm

Table 2 Specification of Vibrometer

Specifications	Range	Best Accuracy
Acceleration	0.1~199.9 m/s^2 (10Hz -10KHz)	(5%+2)
Velocity	0.01~19.99cm/s (10Hz - 1KHz)	(5%+2)
Displacement	0.001~1.999mm (10Hz - 500Hz)	(5%+2)

The vibration parameters were measured at specified the excitation speed of 1200 rpm, 1800rpm, and 2400rpm low medium and high speeds relatively. The speed in rotations per minute was measured with a digital noncontact photo type tachometer model DT2234A+ digital tachometer with specifications as in table3.

Table 3 Specification of Digital Tachometer

Model	DT-2234A+
Rotating speed	2.5 to 99,999 RPM
Resolution	(More than 100RPM, 0.1RPM(<1,000RPM)
Accuracy	± (0.05%+ 1 degree)

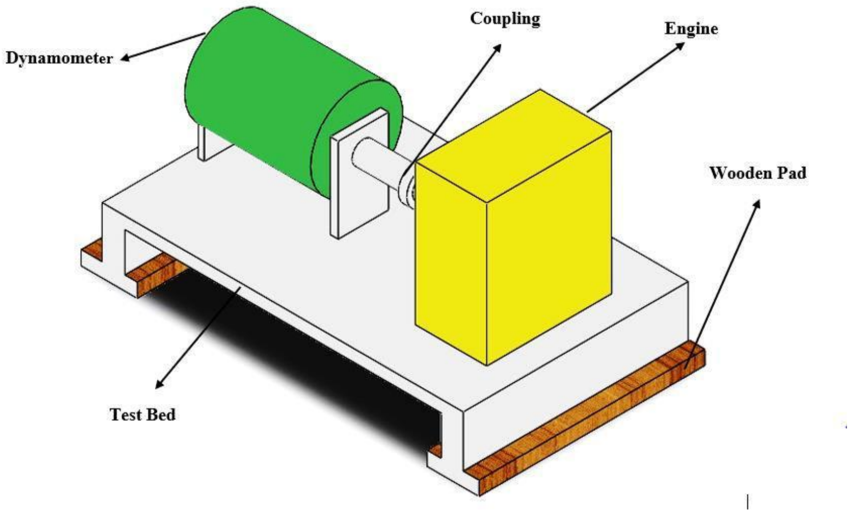

Fig.4 IC Engine Test bed Model

Figure4 indicates the simulated model of the engine test bed used for the experimental investigation. Modal analysis was performed for the test bed using solid works 16 with specified boundary conditions and using coarse mesh. The modal analysis contour plot with resultant deflection results are presented in the plot below. Three modes of vibration with a maximum deflection of 0.59 mm, 0.43mm and 0.56mm are presented in the plot. The trends of mode shapes predict the bending failure of the test bed at the center was the vibration effect is maximum due to lumped mass (Engine and Dynamometer). The objective and motivation for the research study inducted from the modal analysis were to monitor and attenuate the vibration at various excitation speed using various isolation Materials.

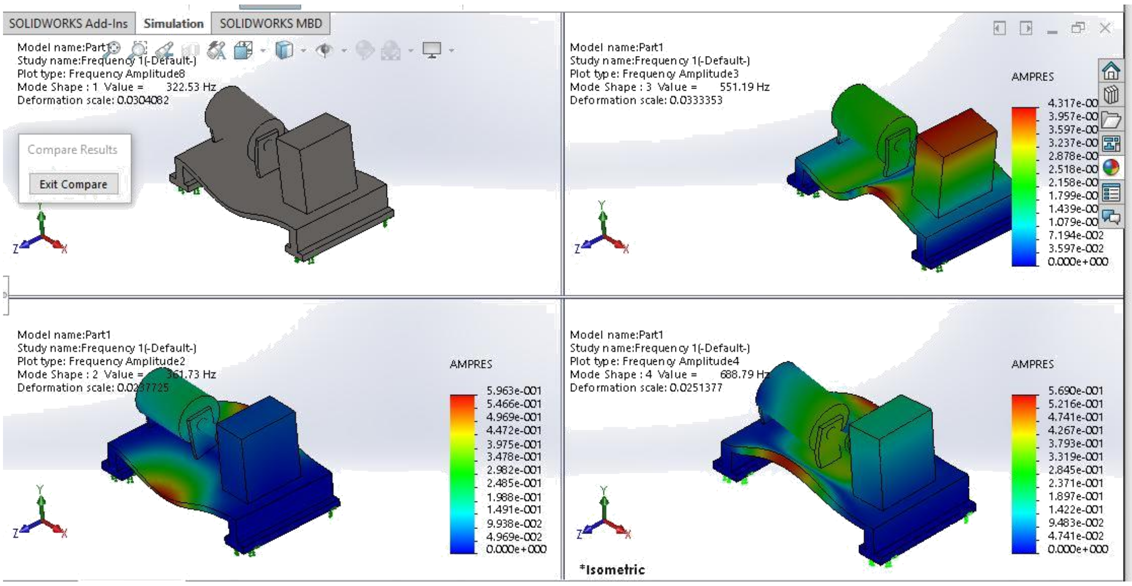

Fig5. Modal Analysis of the Test Bed

Vibration isolation methods widely used are categorized as an active and passive method of vibration control. The passive method is mainly used to mitigate acoustic low-frequency amplitudes by electrical impedance counter resonant loads. Mechanical vibrations of high frequency are mainly controlled by passive isolation material by constructing a flexible foundation to isolate the vibrating system from the rigid base [30]. The isolation materials used in this experimental investigation are wooden pad with size following the engine test bed bottom side. Wooden pads were installed between the concrete ground and metallic test bed. The bed was then tightly fastened, using threaded nuts and bolts [31]. The vibration parameters like velocity, acceleration, and displacement were measured compared with baseline measurements without isolation.

3.2 Design and Analysis of Experiment (DOE)

The hierarchy of experiments to be conducted the investigation was predicted using Minitab 17. The experiment design was predicted at 95% confidence and using speed, Material of Isolation, and corresponding vibration parameters as control variables. The Pareto chart of standardized effect shows the effect of significant variable coded as A B C and D for isolation, speed, velocity, and acceleration upto 2.08 mm response [32].

Table4. Factors of Experimental Design

Control variables	**Codes**	**Level**		
		1	2	3
Material of Isolation	A	Wood	Rubber	Wood and Rubber
Speed (RPM)	B	1200	1800	2400
vibration parameters	C	displacement	velocity	Acceleration

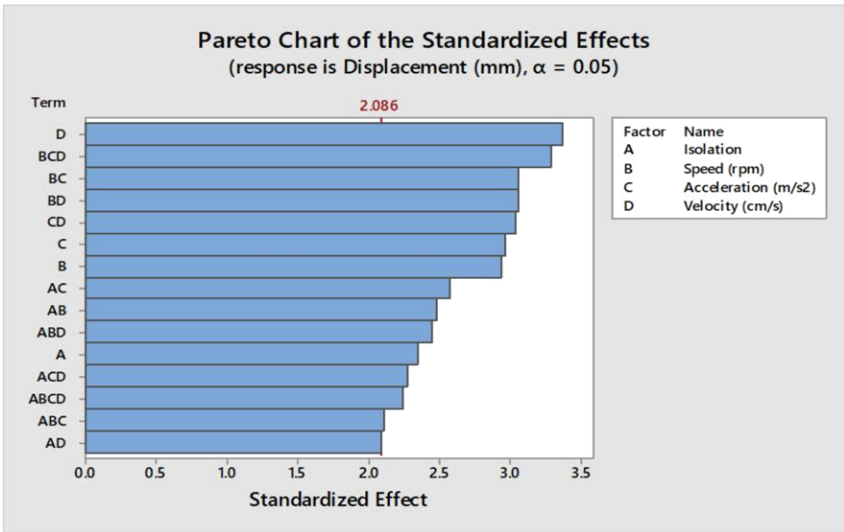

Fig.6 Standardized effect of A B C D

The chart indicates that each combination of factors is significantly affecting the response upto some extent. The velocity of vibration coded as D has the most significant effect on the response of the vibration, while its effect was minimized when combined with isolation A.

3.2.1 Factorial Regression: Displacement (mm) versus Isolation Material, Speed (rpm), Acceleration m/s^2, Velocity (cm)

The displacement of vibration at various excitation speeds and isolation materials was further statistically evaluated and analyzed using the factorial regression model, as in table 5. It is indicated that in a linear model, speed acceleration and velocity are significant factors, while in a two-way interaction, all the factors are significant [33]. The variance and regression coefficients in the model summary indicate the convergence of the response upto 80.60%

Table5. Analysis of Variance

Source	**DF**	**Adj SS**	**Adj MS**	**F-Value**	**P-Value**
Model	15	2.8870	0.19246	5.54	0.00
Linear	04	0.4693	0.11732	3.38	0.29
Isolation	1	0.1911	0.19112	5.50	0.29
Speed (rpm)	1	0.3008	0.30075	8.65	0.008
Acceleration (m/s2)	1	0.3066	0.30664	8.82	0.008
Velocity (cm/s)	1	0.3968	0.39684	11.42	0.003
2-Way Interactions	6	0.6009	0.10015	2.88	0.034
Isolation*Speed (rpm)	1	0.2147	0.21467	6.18	0.022
Isolation*Acceleration (m/s2)	1	0.2308	0.23083	6.64	0.018
Isolation*Velocity (cm/s)	1	0.1516	0.15159	4.36	0.05
Speed (rpm)*Acceleration (m/s2)	1	0.3270	0.32701	9.41	0.006
Speed (rpm)*Velocity (cm/s)	1	0.3270	0.32697	9.41	0.006
Acc. (m/s^2) *Velocity (cm/s)	1	0.3215	0.32145	9.25	0.006

Model Summary

S	R-sq	R-sq(adj)
0.186420	80.60%	66.04%

3.2.2 Factorial Regression: Velocity (cm) versus Isolation, Speed (rpm), Acceleration, Displacement

The factorial regression for velocity as the response of the system indicates that acceleration, along with speed and displacement, are the significant factors for the velocity of the vibration system; as a result, the response values are 95.91% fitted towards the ideal line.

Table 6. Analysis of Variance

Source	DF	Adj SS	Adj MS	F-Value	P-Value
Model	15	639808	42.6539	31.23	0.000
Linear	4	104.561	26.1402	19.14	0.000
Isolation	1	0.730	0.7301	0.53	0.473
Speed (rpm)	1	0.910	0.9096	0.67	0.424
Acceleration (m/s^2)	1	5.325	5.3250	3.90	0.062
Displacement (mm)	1	15.356	15.3556	11.24	0.003
2-Way Interactions	6	56.303	9.3839	6.87	0.000
Isolation*Speed (rpm)	1	1.134	1.1339	0.83	0.373
Isolation*Acceleration (m/s^2)	1	1.201	1.2007	0.88	0.360
Isolation*Displacement (mm)	1	0.000	0.0000	0.00	0.998
Speed (rpm)*Acc.(m/s^2)	1	18.888	18.8880	13.83	0.001
Speed (rpm)*Disp. (mm)	1	6.027	6.0272	4.41	0.049
Acc. (m/s^2) *Disp. (mm)	1	2.356	2.3563	1.73	0.204

Model Summary

S	R-sq	R-sq(adj)	R-sq(pred)
1.16859	95.91%	92.84%	48.40%

3.2.3 Factorial Regression: Acceleration versus Isolation, Speed (rpm), Velocity (cm), Displacement

In this factorial regression model, the acceleration of vibration was selected as the response of the system, and the regression model predicted the significant factors for this response. The summary of results indicates that displacement is the most significant factor for the prediction of acceleration of the vibration system. The 95.6% variance and 8.01 deviation value indicate the convergence of results towards the baseline.

Table7. Analysis of Variance

Source	DF	Adj SS	Adj MS	F-Value	P-Value
Model	15	25912.2	1727.48	26.83	0.000
Linear	4	8467.6	2116.89	32.88	0.000
Isolation	1	43.9	43.89	0.68	0.419
Speed (rpm)	1	199.1	199.14	3.09	0.982
Velocity (cm/s)	1	0.0	0.03	0.00	0.213
Displacement (mm)	1	106.5	106.54	1.65	0.006
2-Way Interactions	6	1686.7	281.12	4.37	0.823
Isolation*Speed (rpm)	1	3.3	3.32	0.05	0.693
Isolation*Velocity (cm/s)	1	10.3	10.31	0.16	0.785
Isolation*Displacement (mm)	1	4.9	4.94	0.08	0.569

Speed (rpm)*Velocity (cm/s)	1	21.6	21.56	0.33	0.569
Speed (rpm)*Displacement (mm)	1	56.7	56.69	0.88	0.359
Velocity (cm/s)*Displacement (mm)	1	149.9	149.90	2.33	0.143

Model Summary

S	R-sq	R-sq(adj)	R-sq(pred)
8.02379	95.27%	91.72%	0.00%

3.3 Experimental Methodology

The flow chart of the experiment is presented for the elaboration of experimental investigation. After performing a static model analysis, the engine was started and set to run at a speed of 1200 rpm without any isolation. The vibration parameters were recorded at the specified speed at the centre corner and endpoint of the test bench. The experiments were repeated three times at each speed, and the average of results was then analyzed. Similar experiments were repeated for speeds of 1800 and 2400 rpm with and without isolation materials, and the health of engine test bed was continuously monitored, so was the noise level monitored. Table 8 indicates a summary of the results with significant factors.

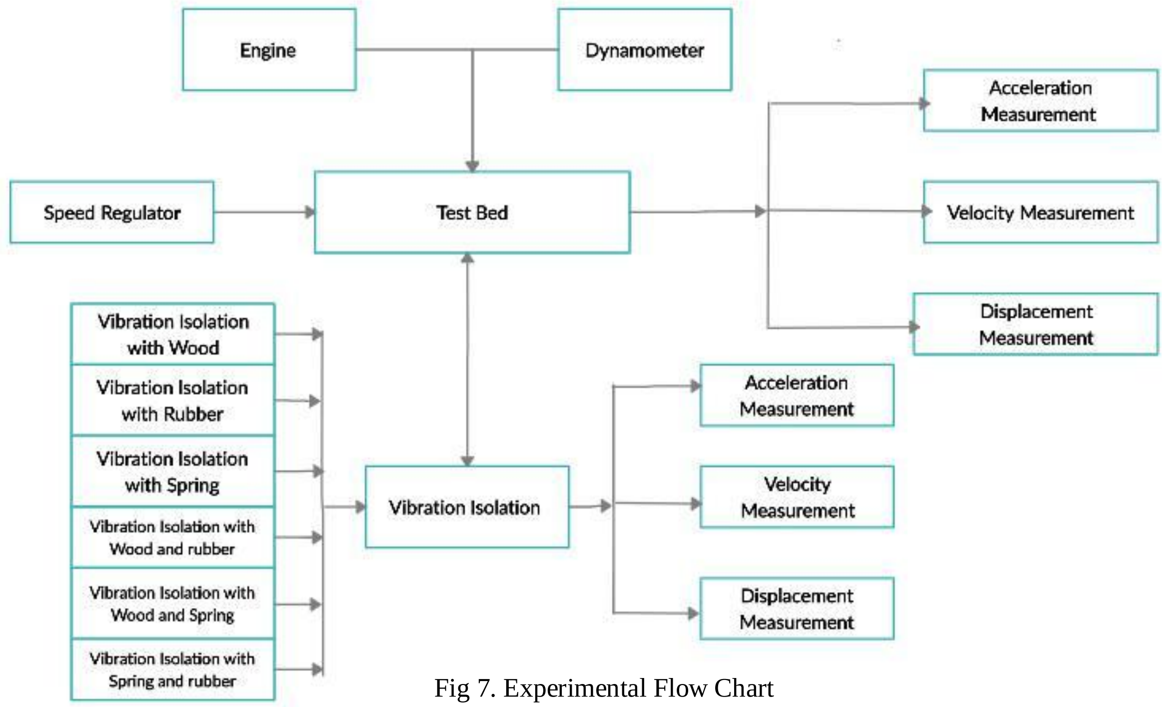

Fig 7. Experimental Flow Chart

Table 8. Results

Orthogonal Array			Responses		
Run NO	A	B	Displacement (mm)	Velocity (cm/s)	Acceleration m/s^2
1	Without Isolation	1200	1.30	15.7	31.67
2	Without Isolation	1800	0.15	5.7	36.30
3	Without Isolation	2400	1.54	30.9	95.43
4	Wood Isolation	1200	0.13	7.6	27.37
5	Wood Isolation	1800	0.10	7.3	28.00

6	Wood Isolation	2400	0.52	26.0	102.53
7	Rubber Isolation	1200	0.90	5.9	22.57
8	Rubber Isolation	1800	0.40	13.5	28.27
9	Rubber Isolation	2400	0.98	28.8	67.20
10	Wood and Rubber Isolation	1200	0.76	9.8	22.77
11	Wood and Rubber Isolation	1800	0.16	5.0	23.83
12	Wood and Rubber Isolation	2400	0.79	27.7	54.33

IV. RESULTS AND DISCUSSIONS

4.1 Response/Displacement (mm) of Vibration with and Without Isolation

The vibration motion was generated in the engine from the motion of the piston, connecting rod, crankshaft, and brake shaft connected with the eddy current dynamometer. The experiments were conducted on the engine test bed at speeds of 1200, 1800, 2400 rpm. The displacements of vibration were measured at the centre, corner, and near the engine mounting. The average of these results is plotted below with and without isolation material. The response of vibration was minimum with wood isolation at excitation speed of 1200 rpm. The trends of results indicated higher response at low speed, which decreased at medium speed and increased to the maximum level. Maximum excitation speed. At the low speed of the engine, the test bed was vibrating at higher amplitudes upto 1.30mm by application wooden pad isolation the response decreased upto 90%. The response was decreased by 65 % speed of 2400. The chart indicates the mitigated response at each Isolation method and corresponding excitation speed. The results indicated that wooden pad isolation is best-suited isolation method to control the displacement of vibration.

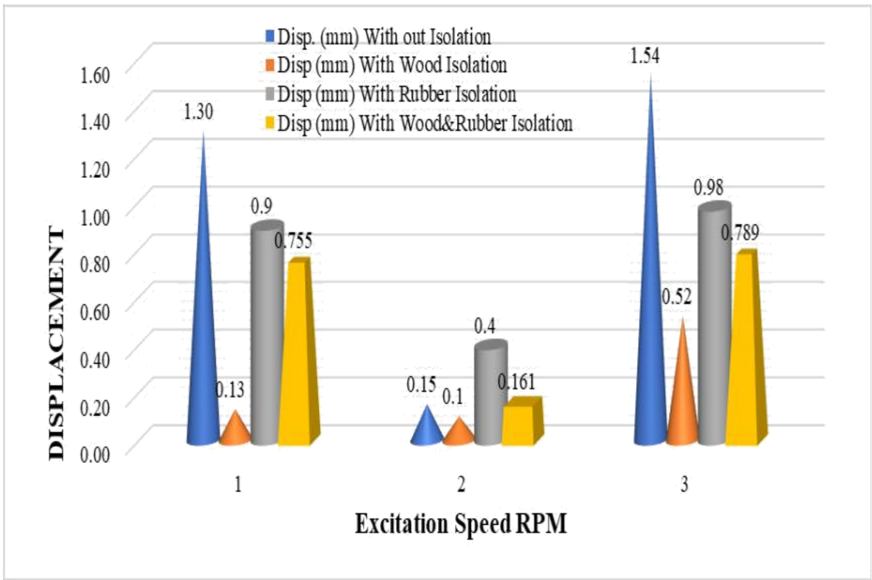

Fig.7 Response (mm) at excitation speeds and isolation Method **4.2**

Response/Velocity (cm/s) of Vibration with and without Isolation

The vibration of the engine test bed was also analyzed for the velocity of vibration as a response parameter and speed as an excitation source. The chart below indicates the velocity of vibration at low medium and high speed with and without isolation. The full conical solid shapes indicate the velocity of vibration without isolation, whereas truncated pyramids indicate the velocity of vibration with wood isolation. Full pyramid indicates the velocity with Rubber isolation. Cone at last position indicates Rubber and Wood isolation in combine form. The velocity of Vibration was lower at each excitation speed with all isolation mechanism as compared to velocity without vibration. The velocity of vibration was abated upto 50% with wood isolation at 1200 rpm engine speed.

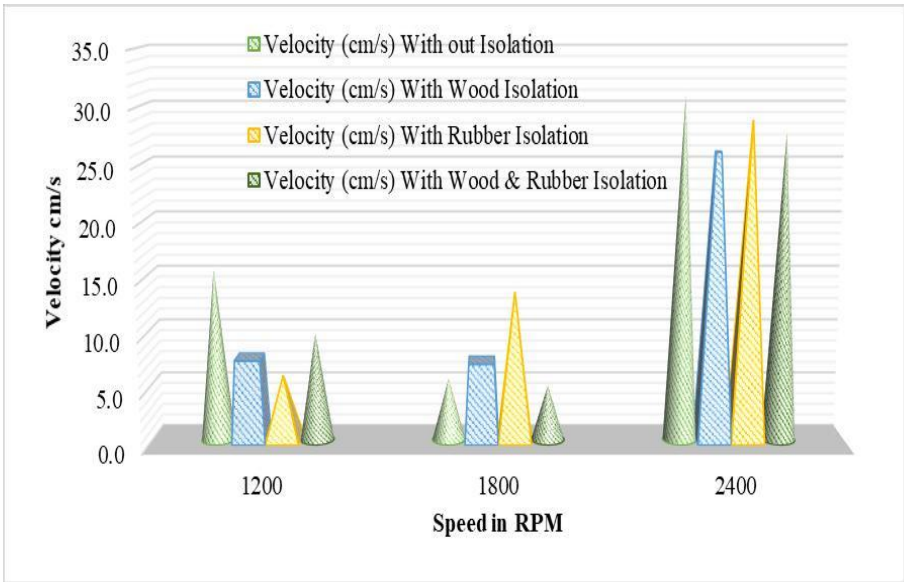

Fig8. The velocity of vibration with and without Isolation

4.3 Response/Acceleration (cm/s) of Vibration with and without Isolation

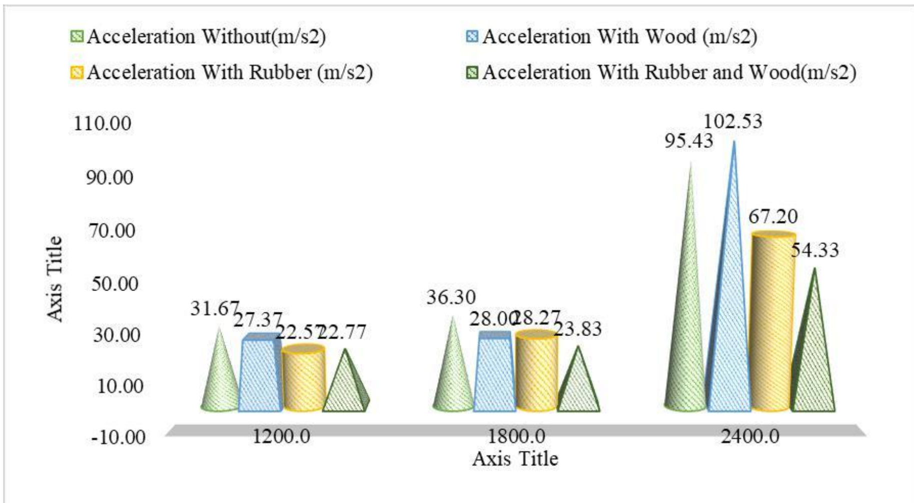

Fig.9 Acceleration of Vibration with and without Isolation

The acceleration of vibration was also measured at speeds of 1200, 1800, and 2400 rpm with and without isolation materials. The trends as an increase in acceleration with the speed of engine test were observed at each speed and isolation mechanism. The extreme value of acceleration n was at 2400 rpm with as isolation material. The level of acceleration was minimum upto 22.5 m/s^2 at1200 rpm and rubber pad as isolation material. All of the isolation materials resulted in mitigation of acceleration, but the maximum mitigation upto 43.15 % was evaluated at a speed of 2400 rpm with rubber and wood as isolation material.

4.4 Residual Plots

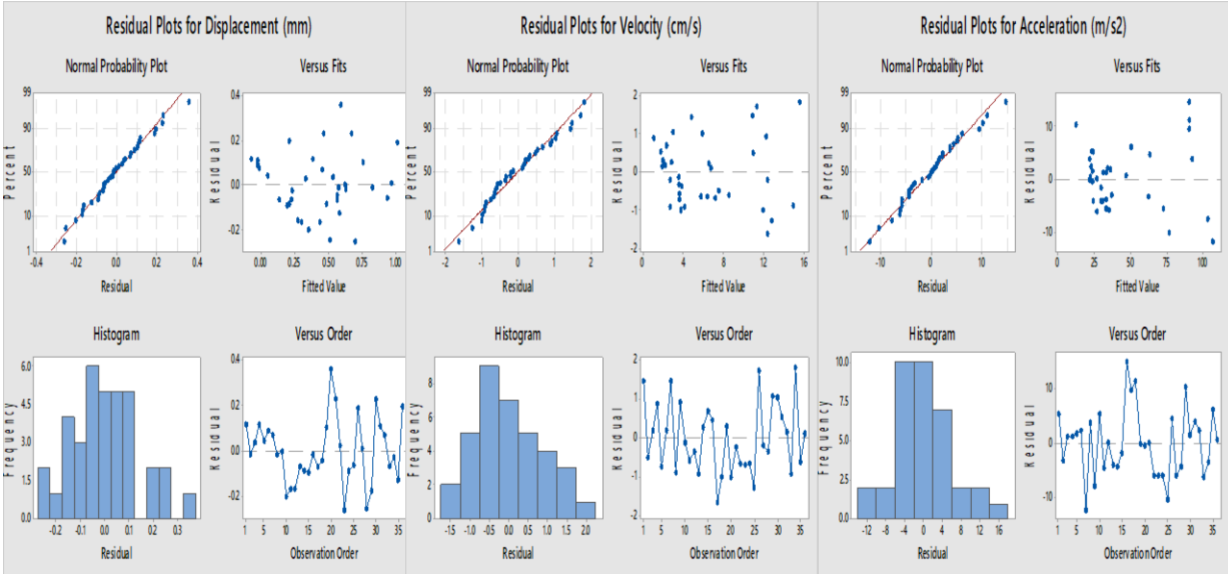

Fig.10 Residual Plots of Response

The operational health of any machine under dynamic excitation condition can be predicted by the response of vibration of that system. In this experimental study, the health and stable operation along with noise level were measured at the specified speeds with and without isolation materials for an internal combustion engine test bed as discussed in section 3 of the paper. The responses of vibration were measured at three positions on the test bed at specified speeds with and without isolation, and the average response was further statistically analyzed. Residual charts below indicated the frequency, normal probability, fitted value and ordered scatter of the measured results. The displacements of the vibrating system are negatively skewed, whereas velocity and acceleration of vibration are normally distributed along the mean fitted line.

4.5 Contour plot for the vibration response

The contour plot in figure11 indicates the significant effect of control parameters like isolation, displacement, speed and acceleration on the resultant response of the vibration system. At lower speeds, the isolation works more efficiently to control the displacement of vibration, whereas the controlling effect becomes insignificant at higher speeds. The contour plot was obtained from the analysis of factorial design of experiment taking displacement as output response and speed, Isolation and acceleration and velocity as control variables for the response. The plot obtained was

at hold value of 62.8 for acceleration and 9.615 for velocity. The horizontal axis indicates the significance of isolation method upto level 3. The levels 0, 1, 2, and 3 were provided to the isolation method as 0 for without isolation, one wood isolation, 2 for rubber isolation and 3 for wood & rubber isolation. Similarly, the effect of corresponding control factors can be visualised for the

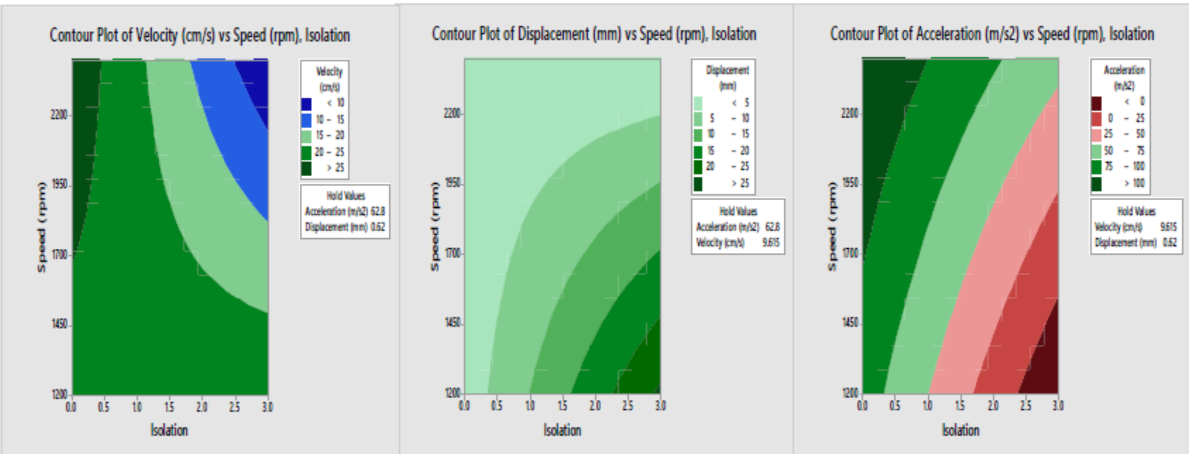

Fig.11 Contour plot for responses of vibration

analysis of speed and acceleration. The contour plots could understand the dependency of responses on controll parameters for displacement velocity and acceleration of vibration. The plots indicate the minimized vibration responses which predict the improved structural health and reduced the noise level.

4.6 Interaction plots

The line plots below indicate the interactions between the control factors for the response using fitted means. Lines with circular dots indicate the minimum and square dot lines indicate maximum values of the parameters. Figure 9, 10 and 11indicate the interactions between al the factorial variable. The effect of displacement, velocity, acceleration speed and isolation method can be visualized from the interaction plots. The behaviour of vibration parameters may be analyzed by the parameters varying from minimum to maximum values.

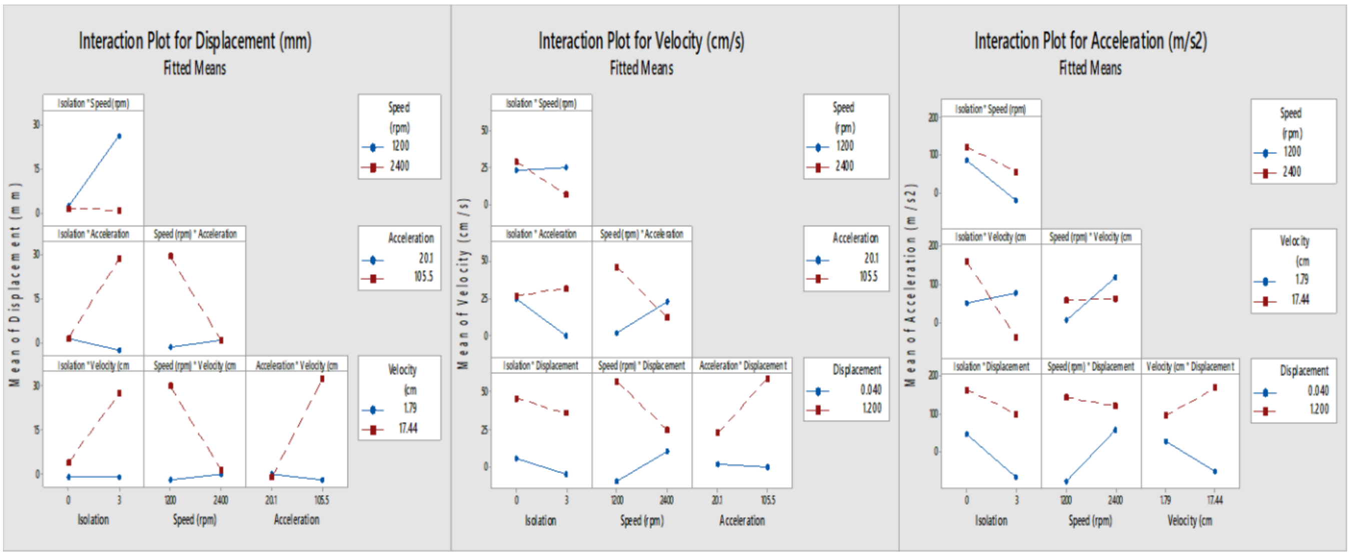

Fig.13 Interaction plots for responses of the vibrating system

V. CONCLUSION

A machine in operation without vibration is not possible in real life. Mechanical vibration is the term used for the dynamic behaviour of a system under self-loading or external excitation. The response of unbalance forces measured in terms of displacement velocity and acceleration in the normal practice for maintenance engineers to monitor the structural health of the dynamic system. Each mechanical system under excitation has some finite life in terms of operating ours or operating cycles.

The failure of a dynamic system is supposed to occur due to fatigue under the live loading condition. This research study was conducted with the motivation to measure the response of the internal combustion engine test bed under excitation speed of low medium and high level. The results of dynamic response in terms of acceleration, velocity and displacement were measured at centre corner and near engine for each specified speed. Each experiment was repeated three times to ensure repeatability. The internal combustion engine test was isolated from the rigid foundation using the wooden pad, rubber pad and wood and rubber combined. The responses were again measured under similar excitation and precision conditions. The results were then statistically analyzed using Minitab 17 package. The responses of dynamic behaviour, the significant factors and residual error were analyzed and presented in the form of bar charts, contour plots, residual plot, Pareto chart and interaction plots.

- The response of vibration in terms of displacement was mitigated upto 90% at a speed of 1200 rpm with wood as isolation material
- The response in terms of velocity and acceleration were also reduced at each excitation speed with all isolation methods the mitigation was maximum by 50% @ 1200 rpm, wood isolation and 64% @ 2400 rpm with rubber and wood isolation for velocity and acceleration respectively.
- The stable operation of the engine test bed was improved

- The implementation of isolation materials decreased the noise pollution level in the specified research area.
- The structural health of the engine test bed was improved due to the attenuation of dynamic responses.

ETHICAL STATEMENT

It is declared that the none of the authors has any conflict of interest for this research study and no funding was claimed from national or international source.

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