Dynamic characterization of engine mount at different orientation using sine swept frequency test

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Abstract

Rubber engine mounts are commonly used to provide vibration attenuation. The loss factor and dynamic stiffness of engine mount provide fundamental information of the energy dissipation. The objectives of this work are to study the frequency-dependant stiffness and loss factor of engine rubber mount and to determine the effect of the orientation angle of the engine mount on the dynamic characteristics. The test consists of three engine mounts, shaker, an accelerometer , analyzer and force transducer with a preload mass of 2.72 kg steel plate mounted on the engine mounts. The plate is excited at the center and the input force and response is measured at the point of excitation which is called the driving point excitation. The electromagnetic shaker is used as an excitation source to excite the rubber engine mount system to obtain the frequency response function and dynamic stiffness of rubber mounts. The resulting signals captured from the force transducer and accelerometer are then analyzed using LMS spectral testing. The frequency response function (acceleration/ force) is estimated using the two channels FFT calculation where the dynamic stiffness is the reciprocal of frequency response function. The real and imaginary parts of dynamic stiffness can be obtained directly from the sine swept frequency test. The frequency response function and dynamic stiffness of engine are measured from sine swept frequency test for frequency range of)-200 Hz. The unity value in coherence function within the range above 10 Hz shows the linear dependency between force and displacement signals. The figure shows that the dynamic stiffness is frequency dependent. The dynamic stiffness is increasing when the frequency increased after the resonance at 40 Hz. For the excitation frequency before resonance, the dynamic stiffness decreased while the excitation frequency increased. Minimum dynamic stiffness is found at resonant frequency and the values of dynamic stiffness increased when the frequency increases above the natural frequency. The frequency which natural frequency of the system occurred becomes lower when the engine mount orientation is changed from 0 degree (axial) to 90 degrees (normal). At the same time, the values of dynamic stiffness decreases especially at the frequencies below the natural frequency. The measured stiffness and loss factor are validated by comparing the reconstructed system frequency response function of the single DOF system using the estimated frequency dependent stiffness and loss factor to that frequency response function which is obtained from direct sine swept test. Comparison between measured and estimated FRF showed that the correlation between the two FRFs is good at high frequencies especially away from the resonance.

Keywords: engine mounts, dynamic stiffness, loss factor, orientation angle

1. Introduction

Rubber is widely used in the vibration control by isolating the vibration source. It is a material which has both elastic and viscous properties. Mounts may have as many as three functions. First, it is an attachment point for a part or system to the chassis. Secondly, it often acts as an isolator – keeping noise and vibration from being transferred to the driver and passengers. In some cases, it may also be an adjustment point to keep a component in proper alignment. Engine mounts properly locate the engine in the chassis, and are an important factor in how smoothly a vehicle operates. The mounts are designed to allow a certain amount of rotation, plus dampening much of the engine vibration. Damping is an important parameter for engine mounts' performance.

Body and frame mounts are less important on automobiles since the introduction of uni-body construction. However, they are still used for mounting the cab to the frame on light trucks, and virtually all older automobiles use them. These mounts are basically rubber "doughnuts" between the body or cab and the frame. They help assure that road and driveline vibrations are further isolated from the driver and passengers. Like all other mounts, they deteriorate; drying out, cracking and eventually the rubber's resiliency disappears. In the most severe cases, the mount may break apart and fall out. Failing body-to-frame mounts will result in a harsh, unpleasant ride and poor interior noise quality. Additionally, they may begin to cause misalignment of critical control linkages for the throttle, clutch or transmission.

A survey of automotive engine mounting systems was presented by Yu et al.[1] where dynamic stiffness and damping are frequency and amplitude dependent and also the better performance of passive hydraulic mount when compared to elastomeric mount especially at low frequency range. Ashrafuion [2] carried out dynamic analysis of an airplane engine mount systems by assuming the linear vibration of the system. However, the tested frequency range is up to 80 Hz only. The analysis covers axial force transmissibility and amplitude of vibration for three fundamental models. The engine mounts are modeled as three dimensional spring with a significant amount of hysteresis damping.

Thompson et al.[3] described the indirect method to measure the dynamic stiffness of resilient elements and the possibility of extension for the measured frequency range. The study of Nader et al.[4] allows the dynamic measurement of rubber mount at frequency up to 5000 Hz using the high frequency test machine. A forced non-resonance method is applied by Nadeau to the engine mounts [5]. The ratio between imaginary and real part of complex stiffness is used to estimate loss factor of engine mounts. Lin et al.[6] has proposed to evaluate the frequency dependent stiffness and damping characteristic of rubber mounts by using impact test method. The advantage of the method is it can accurately predict the response at non-resonant frequency.

There are several methods to characterize the dynamic properties of viscoelastic materials[7, 8, 9]. Different measurement set up represents different definition of dynamic stiffness. For example, dynamic driving point stiffness is measured as the force and displacement at the input side while the output side is blocked and dynamic transfer stiffness is measured as the displacement at the input side with the blocked force at output side. Different definition and measurement of dynamic stiffness while produce different results of the loss factor.

The study of damping measurement at inclined angle is important because it can provide the information of dynamic characteristic under real operation condition. Since the need o improve the performance of vibration isolator is required, damping analysis of engine mount under its real operation condition helps to improve the accuracy of design and damping measurement. The concept of design optimization of engine mount is proposed by chang [10]. Inclined angle of the engine mount is one of the design parameters used for optimization but the results exclude the effect of orientation angle. The modeling of marine engine mounts system with design optimization was done by Tao [11]. The study is to select the stiffness and orientation of individual mount to produce minimum transmission of vertical force from engine to the floor and reduce noise. Euler transformation matrix is applied to include the orientation angle. However the effect of orientation angle on damping and dynamic stiffness is not included. No experimental work is done in the study. The work is studied detail in the minimization of dynamic force through the optimization of orientation angle and stiffness coefficient.

The study presented in this paper will measure the frequency dependent stiffness and loss factor of the engine mounts based on the non-resonant method. The engine mounts are first examined in the axial direction and extend to other orientation angles. The reason to measure frequency dependent stiffness and loss factor at various orientation angles is to study the effect of orientation angles of engine mounts on their damping characteristic. When the engine mounts are placed in an incline, the combined effect of axial force and shear force will influence the damping force at the particular angle. This effect should be study to determine the optimum orientation angle of the engine mounts.

The objectives of this work are as below:
1) To study the frequency-dependant stiffness and loss factor of engine rubber mount
2) To determine the effect of the orientation angle of the engine mount on the dynamic characteristics.

2. Experimental setup

Solid rubber engine mounts are used for mounting of 2.72 kg two stroke engine of a backpack type grass trimmer are selected in this study. The engine mounts with a dimension of diameter 15mm and 20mm long was used. Figure 1(a) and (b) illustrates the experimental set up for measuring the dynamic stiffness and loss factor of engine mount. The test consists of three engine mounts, shaker, an accelerometer (Kistler, type: 8776A50), analyzer (LMS spectral testing) and force transducer (B&K 2617). A preload mass of 2.72 kg is mounted on the engine mount to apply the preload similar to engine mass in the real operating conditions. The plate is excited at the center and the input force and response is measured at the point of excitation which is called the driving point excitation. The electromagnetic shaker is used to excite the rubber engine mount system and driven by LMS software (spectral analysis).

The resulting signals captured from the force transducer and accelerometer are then analyzed by using LMS spectral testing. The frequency response function (acceleration/ force) is estimated using the two channels FFT calculation. A frequency resolution of 0.5 Hz is selected in the analyzer. 5 averages are taken where the averages type is linear average. The unity value in coherence function for all frequency (except below 10 Hz) shows the linear dependency between force and displacement signals. Measurement is carried out in three conditions. The engine mounts are placed vertically to study the dynamic behavior under axial compression test. The engine mounts are subsequently placed horizontally to study the dynamics behavior under pure shear. The study are extended to other inclination angles (i.e 45 and 90 degrees).

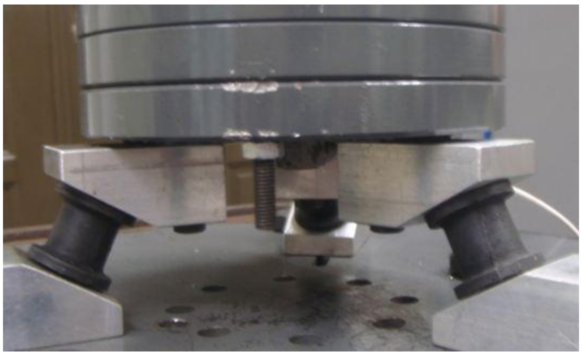

Figure 1(a)

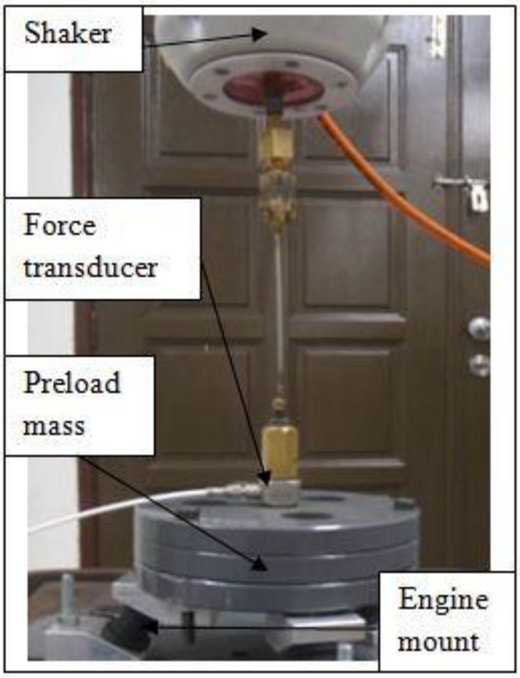

Figure 1(b)
Figure 1 (a) and (b) - Experimental set up for measuring the dynamic stiffness and loss factor of engine mount at an inclined angle

3. Theoretical background

The system used in this study is a single degree of freedom (SDOF) system. The equation of motion of rubber mass system with frequency dependent stiffness $(k(\omega))$ and loss factor $(\eta(\omega))$ can be written as:

$$m\ddot{x} + k(\omega)\big(1 + j\eta(\omega)\big)x = F \quad (1)$$

where m is the mass, x is displacement and $\ddot{x}$ is the acceleration of the system respectively. F is the applied force onto the system.

By assuming the applied force and the response are complex, then

$F = F_0\, e^{st}$ and $x = x_0 e^{st}$

where $s = j\omega$ and $\omega = 2\pi f$

The equation of motion become:

$$ms^2 + k(\omega)\big(1 + j\eta(\omega)\big)x_0 e^{st} = F_0 e^{st} \quad (2)$$

The receptance function which is used to characterize the system behavior can then be written as:

$$\frac{x_0}{F_0} = \frac{1}{k(\omega)\big(1+j\eta(\omega)\big) - m\omega^2} \quad (3)$$

This dynamic behavior of system can also be expressed in the form of dynamic stiffness at driving point where

$$k* = k(\omega) - m\omega^2 + j\eta(\omega)k(\omega) \quad (4)$$

The real and imaginary parts of dynamic stiffness can be obtained directly from the sine swept frequency test.

$$k* = k_{real} + k_{imaginary} \quad (5)$$

So by comparing the real and imaginary parts of the dynamic stiffness, the frequency dependent stiffness and loss factor are obtained. Then the frequency dependent stiffness from equation (4) can be written as below:

Frequency dependent stiffness,

$$k(\omega) = k_{real} + m\omega^2 \quad (6)$$

Loss factor,

$$\eta(\omega) = \frac{k_{imaginary}}{k(\omega)} \quad (7)$$

4. Results and discussion

The frequency response function and dynamic stiffness of engine are measured from sine swept frequency test for a range of frequency (0 Hz to 200 Hz) and shown in figure 2 and 3. The figure shows that the dynamic stiffness is frequency dependent. The dynamic stiffness is increasing when the frequency increased after the resonance of 40 Hz. For the excitation frequency before resonance, the dynamic stiffness decreased while the excitation frequency increased. This is similar to the observations made by Gent[12]. The minimum dynamic stiffness occurred at natural frequency of the system where small applied force resulted in large deformation. Similar trend was found[13] where the measurement on an isolator is done by resonant method. The test is done up to 60 Hz. Minimum dynamic stiffness is found at resonant frequency and the values of dynamic stiffness increased when the frequency increases above the natural frequency.

The results obtained are compared with those from axial compression tests to study the effect of orientation angle on the dynamic behavior of engine mounts. Result in figure 3 shows that changing of orientation of engine mount will change its dynamic behavior. This is turn lead to the change of natural frequency of the system. The changes of dynamic stiffness are significant at frequencies at and below the resonant frequency. The natural frequency of the SDOF system is determined from the same test. The frequency which natural frequency of the system occurred becomes lower when the engine mount is changed from 0 degree (axial) to 90 degrees (normal). At the same time, the values of dynamic stiffness decreases especially at the frequencies below the natural frequency.

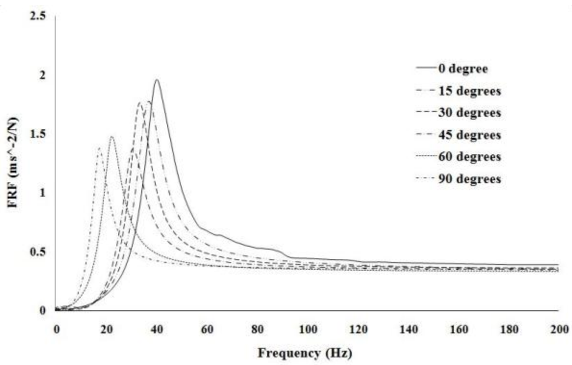

Figure 2: Comparison of frequency response function

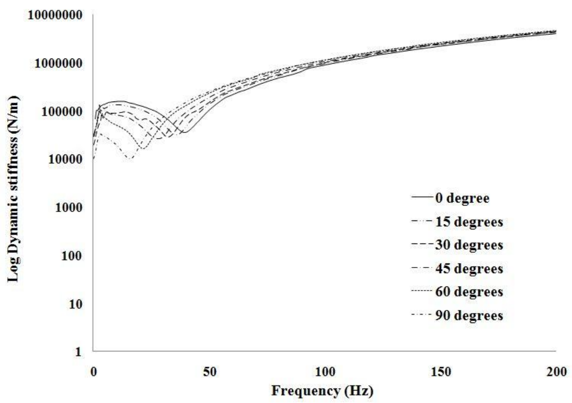

Figure 3: Comparison of dynamic stiffness

The comparison frequency dependent stiffness of the engine mount for a range of frequency is then estimated from equation (6) and shown in figure 4. Although the test is carried out from 0 Hz to 200 Hz, the stiffness is only estimated from 10 Hz due to low coherence below 10 Hz. The frequency dependent stiffness is between 170- 220 kN/m in the frequency range of 10-200 Hz. The unevenness of the values can be attributed to the differences of the elasticity for each engine mount measured. It is difficult to excite a pure heaving mode where geometrical coupling is inevitable.

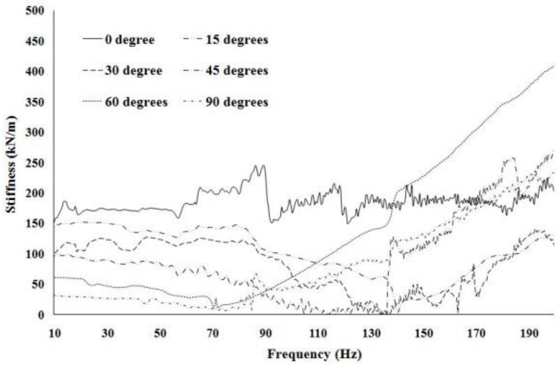

Figure 4: Comparison of frequency dependent stiffness

The loss factor of engine mount is calculated by equation (7) and shown in figure 5. To make the illustration clear, only the discreet frequencies (i.e 10 Hz, 20 Hz, 30 Hz…200Hz) are selected and plotted. The limitation of this approach is that equation (7) will cause the loss factor unrealistically when the frequency dependent stiffness approaches zero. The results of dynamic stiffness approach must be carefully evaluated to avoid the unrealistically high values of loss factor at frequencies where the frequency dependent stiffness $(k(\omega))$ is very low.

The benefit of this method is the suitability of determining loss factor at resonant frequency and also other frequencies concurrently. Most of the existing damping measurement methods are either resonance- based method or non-resonance based method.. For example, the work by Lin et al. [6] showed the measurement of stiffness and damping of rubber mounts by using impact test and their results showed that the evaluated values of loss factor approaches infinity at resonance frequency. It shows that the measurement of loss factor at resonance frequency using the impact test need to be carefully evaluated and checked by half power bandwidth method at resonance frequency.

The result from figure 4 shows that the frequency dependent stiffness of engine mounts decreases when the engine mount orientation is shifted from vertical to horizontal direction. It also decreases when the frequency increases. However, for 45 degrees orientation, the frequency dependent stiffness is 96 kN/m at 10 Hz and decreases until its minimum at 112 Hz. After this, the stiffness increases steadily to 272 kN/m at 200 Hz. Similar trend is also happened for 90 degrees orientation. According to equation (6), the effect of the mass of engine onto the stiffness of engine mount becomes significant at high frequency.

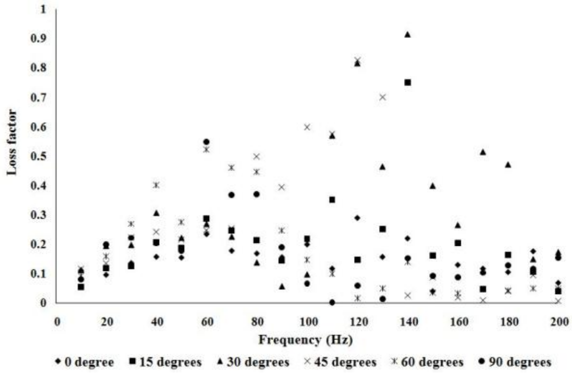

Figure 5: Comparison of loss factor

5. Reproduction of frequency response function

The measured stiffness and loss factor are validated by comparing the reconstructed system frequency response function of the DOF system (equation (3)) using the estimated frequency dependent stiffness and loss factor to that frequency response function which is obtained from direct sine swept test. Figure 6and 7 show the comparison between measured and estimated FRF. It can be noticed that the correlation between the two FRFs is good at high frequencies especially away from the resonance. For the frequencies nearest to the resonance, the response from estimated FRF under predict compared to the measured FRF. The same condition happened for all other orientation angles. The FRF values approaching constant when the frequency is far from resonance frequency. The correlation between the estimated and measured FRF is examined by finding the correlation index (R Square) between two curves for example in figure 8. Good correlations up to 0.98 are obtained between measured and estimated FRF.

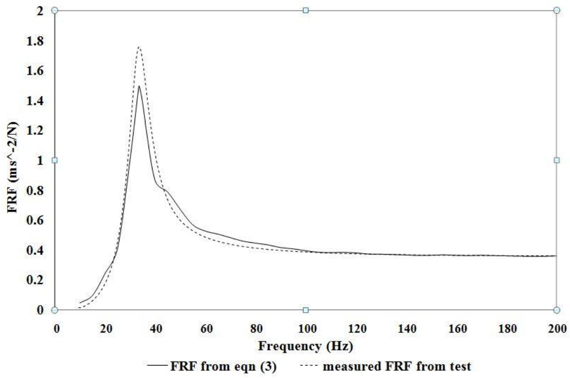

Figure 6: Comparison of FRF for 30 degrees

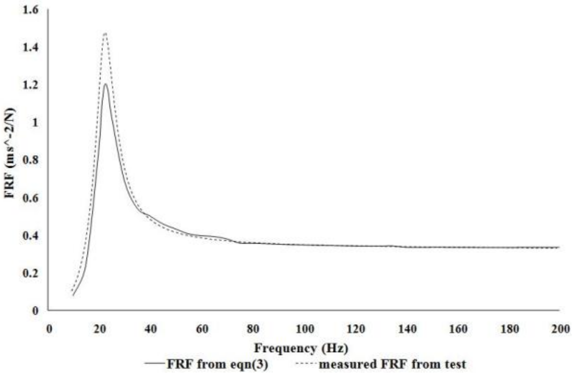

Figure 7: Comparison of FRF for 60 degrees

4. Conclusion

a) The frequency-dependant stiffness and loss factor of engine rubber mount are studied using non-resonance based technique and the values obtained are substituted in the equation of motion and the resulting response are compared. The correlation coefficient (R square) shows the correlation between two curves is up to above 98%.

b) The effect of the orientation angle of the engine mount on the damping characteristics are significant for frequencies below the first natural frequency. It is important to include the effect of the engine mount orientation effect in any optimization studies when the mount is not in its axial position as normally done.

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