Design of indenter rig for measuring seat cushion dynamic stiffness

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Abstract. Vibration induced on the car floor is transmitted to human body through seat structure, affecting both comfort and health. Seat structure components are mainly comprised of rigid frame and seat cushion. Previous studies have shown and suggested that the indenter is preferable option for replacing human subject in seat vibration experimental work when assessing the seat cushion dynamic characteristics. Indenter method offers more control, cheaper and less time intensive compare to other methods such as passive dummies, sandbag, and rigid mass. This paper proposes a design of indenter testing rig to measure the dynamic stiffness of seat cushion material. The rig was designed to operate at frequencies between 2 to 100 Hz and applied load of maximum 50 kg. The size of bed is proposed to be able to fit the size of standard sedan car seat. Modal analysis was conducted on indenter rig structure design to ensure the suitability of the design. Structural frame was designed to fit the mounting of the shaker used in this study. Six modes shapes were extracted from the modal analysis and its corresponding frequencies ranging from (121.63Hz – 507.3Hz). The lowest natural frequency for the proposed design of indenter testing rig occurs in the first mode of vibration with peak displacement of 174.04mm. From this result, it can be concluded that the proposed design met the criteria set and produces a fundamental frequency of greater than the operating range of the rig.

Keywords: seat cushion, dynamic stiffness, vibration, frequency

1. Introduction

Seat is one of the components in vehicle’s design of the transport system such as cars, trains, commercial vehicles, and aircraft. Under dynamic motion, the direct contact between seat material and human body plays a significant role in attenuating vibration transmission. This exposure can affect body comfort and health risks.

Many methods and experimental techniques have been developed to study the vibration transmitted from the vehicle’s structure to human body via seat system and for objective research, the evaluation of seating dynamics can involve several parameters such as dynamic stiffness, biomechanical response of whole body, seat transmissibility and SEAT value [1, 2]. Seat transmissibility has been the common method for determination of seat performance [2]. Measuring seat transmissibility requires subjects to be sitting on the seat. The interaction between human body and seat system acted as couple dynamic system where the vibration transmitted is dependent on the mechanical impedance of the seat and the sitting body [2]. It is preferable to measure vibration transmissibility using human subject although
larger vibrator is needed as it must be able to provide vibration with human subject actual weight. However, testing with human subject is time consuming, a slight change in human body impedance and sitting characteristics can affect the overall result [2]. In addition, working with human subjects requires medical and ethical approval which can be difficult to get for certain condition [3, 4]. Seat testing conducted on a small group of subjects can be inconsistent, non-repeatable of test result and objectivity [3, 4].

As an alternative to using human subjects, several researchers have proposed the use of passive or active dummies as replacement for human subjects [3, 4]. The passive dummy method is currently the common method to be used in seat vibration testing. Some passive dummies are constructed using single degree of freedom (SDOF) systems and as an improvement, some passive dummies have been developed with multi-degree of freedom (MDOF). A SDOF based passive dummy for seat vibration testing was previously developed but it was found that at higher resonance frequency, the results were not satisfying [3, 5]. By adding more degree of freedoms, it will improve the results but require extensive construction efforts [4, 6, 7]. It has also been reported that both single and multi DOF of passive dummies will exhibit friction and unwanted nonlinearities; the nonlinearities of human body dynamic response under vibration evinced great differences with nonlinearities of passive dummy subjects [2, 4, 6, 8].

Some researchers propose several other methods such as using sandbag, and rigid mass as replacement for human subjects. L. Wei & Griffin [1] found that using sandbag might not be appropriate especially for larger weight where it might consume greater contact area in comparison with normal human body. The excessive contact area reached beyond or to the edge of the seat will influence the measured seat transmissibility [1]. In the case of subject’s replacement using rigid mass, it will produce satisfying outputs and in good agreement with measured transmissibility using human subjects. However, rigid mass on the seat foam can easily be subjected to rotation and movement when exposed to lateral or front-aft vibration.

Another method of classifying seat performance is using indenter test. A rig with indenter is used to apply load before vibration is initiated vertically, and the measured output force and input acceleration are used to calculate seat dynamic performance. Indenter test method requires the measured seat dynamic stiffness to be used with a measured human biomechanical response to vibration, to predict seat transmissibility using mechanical impedance model. Indenter test offers more controllable options, and the variation of loading magnitudes can be applied in, easier and quicker than other seat testing methods. Hence, indenter has become preferred method to replace human subject for quantifying seat performance [1]. The use of indenter for assessing seat vibration testing has produced remarkably useful results since it will eliminate the drawbacks and limitations with human subjects and other types of mass elements [9-14].

The objective of this paper was to propose a design of indenter rig system for the assessment of seat dynamic stiffness. The design was based on the previous indenter test rig. An improvement was made on the structure, a smaller overall design for greater rigidness, to be able to connect on a different vibrator. With smaller sized, the natural frequencies of the rig were expected to be greater than the operating frequencies for the indenter test which is between 2 to 100 Hz.

2. Proposed Indenter Test Rig Design and SIT-PAD Indenter
The design of the indenter test system, components, and testing requirements were chosen and made according to the allowable size and capacity of the equipment available at Sound and Vibration Research (SVRG) lab, Universiti Putra Malaysia. The complete indenter testing rig is comprised of a vibration shaker machine, a structural frame, an indenter, an accelerometer, and a force sensor. Force sensor was suggested to be of a donut type (Futek LTH-350) with the maximum capacity of 2.2kN, which is a piezoresistive type, for the measurement of static pre-load as well as dynamic load during vibration exposure.

Vibration magnitude will be generated using ETS Solution L215M shaker machine which has maximum loading capacity of 70kg and supported frequency range of up to 4500Hz. A frame structure
was design and to be attached on the vibrator as shown in Figure 1. This setup can be operated using MATLAB and LMS Test Lab software with SCADAS data acquisition system. The type of material used to fabricate structural frame is mild steel with rectangular hollow section with dimension of 50mm x 500mm and a thickness of 3.2mm.

![Figure 1. Shaker machine and test frame assembly.](image)

Indenter test rig will be used to run with capacity and frequency range as specified in the British Standard BS2361-Part 1. For the SIT-PAD Indenter (see Figure 1), the material was chosen to be aluminium. Figure 2 shows the assembly of SIT-PAD indenter proposed of this work.

3. Modal Analysis on Structural Frame

Modal analysis on structural frame was conducted to determine the vibration characteristics by extracting the mode shapes and its corresponding natural frequencies. Both mode shapes and natural frequencies are important parameters as reference for designing structural system to be used in dynamic loading conditions [15].

3.1. Finite Element Analysis on Structural Frame

The structural frame was modelled, exported, and imported to ANSYS Workbench for simulation and analysis. The imported geometry was then meshed and applied with boundary conditions. For this simulation, fixed support boundary condition was applied with pre-stressed load on screw rod. Four fixed supports are applied at bolt locations and a pressure of 2.264 N/mm² (Force = 400N) was applied at both screw bar bottom surface in upward direction as shown in Figure 3.
The large displacement mode was switch on for the analysis and adaptive meshing was enable with resolution set to 4. This is based on checking by trial and error, the selected meshing resolution produced better stability for meshing and analysis output. The material used to fabricate the frame was mild steel and for the simulation, the material input is as displayed in Table 1.

**Table 1. Material property.**

| Parameter                  | Detail   |
|----------------------------|----------|
| Young Modulus, E (MPa)     | 200e3    |
| Poisson Ratio              | 0.3      |
| Shear Modulus, MPa         | 76923    |
| Bulk Modulus, MPa          | 1.6667e+005 |
Figure 4 displayed the flowchart of finite element analysis using ANSYS Workbench to obtain the natural frequencies and mode shapes of SIT-PAD indenter frame structure under dynamic condition.

![Finite element analysis flowchart](image)

**Figure 4.** Finite element analysis flowchart.

Table 2 displays the effect of number of elements on first mode natural frequency obtained from the modal analysis. The structural was meshed using Automatic Method which resulted the use of tetrahedron (10 nodes), hexahedrons (20 nodes) and wedge (15 nodes) elements.

| Element Size, mm | Number of elements | Number of Nodes | First mode Natural Frequency, Hz |
|------------------|--------------------|----------------|---------------------------------|
| 30               | 20879              | 65732          | 118.61                          |
| 28               | 22165              | 70062          | 117.44                          |
| 26               | 24437              | 76549          | 117.41                          |
| 24               | 29231              | 92915          | 116.88                          |
| 22               | 32439              | 106422         | 116.89                          |
| 20               | 34516              | 117405         | 116.96                          |
| 18               | 27212              | 107592         | 116.83                          |
| 16               | 32899              | 139052         | 116.38                          |
| 14               | 40176              | 174593         | 116.08                          |
| 12               | 48954              | 219348         | 115.95                          |
| 10               | 63858              | 292616         | 115.5                           |
| 8                | 100563             | 467683         | 115.56                          |
| 6                | 137177             | 522344         | 115.17                          |
| 4                | 307002             | 1179723        | 115.02                          |

The number of elements increased with the decreased of element size and at the same time, the natural frequency exhibited increasing trend as shown in Figure 5.

The use of finer size of element expected to produce better accuracy (not always true), however it can lead to several problems such as longer processing time or processing errors especially in multibody analysis.
3.2. Simulation Results

Six mode shapes were extracted from the analysis and the results using element size of 6mm was selected because skewness level of most elements falls in the acceptable range as proposed in ANSYS documentation. Skewness level lower than 0.75 produced acceptable quality and larger size of element will result more elements in poor quality range (0.75-1) [16] as shown in Figure 6 and Figure 7.

![Figure 6. Mesh metric at 30mm element size.](image)

![Figure 7. Mesh metric at 6mm element size.](image)

As shown in Figure 7, most elements fall in the skewness region lower than 0.75 which is satisfactory.
for this analysis. Further efforts to reduce element size beyond 6mm lead to higher processing time and resulted solver error at 2mm element size. Plus, natural frequency in the first mode for smaller size of 4mm produced small difference 6mm element size. The natural frequency for each mode using 6mm size of element is displayed in Table 3.

**Table 3.** Mode shapes and its corresponding frequencies.

| Mode Shape | Frequencies, Hz | Structural Deformation                     |
|------------|----------------|--------------------------------------------|
| 1          | 115.17         | Primary (X direction)                      |
| 2          | 225.91         | Primary (Y direction)                      |
| 3          | 398.06         | Primary (Z-torsion)                        |
| 4          | 418.38         | Secondary (Z direction)                     |
| 5          | 465.99         | Secondary (Y direction -Screw Bending)      |
| 6          | 498.8          | Secondary (X direction -Screw Bending)      |

As displayed in Table 3, there were different structural deformation at each mode shape. At first mode shape ($\omega = 115.17$ Hz), the structural frame translated in X direction and at second mode shape ($\omega = 225.91$ Hz), it resulted translation in Y direction. Third mode shape produced higher natural frequency of 298.06 Hz; the structural frame exhibited torsional deformation around Z axis. At fifth and sixth mode shapes, the deformation on testing rig only imposed on screw rod in bending mode through Y and X axis, respectively. All visual deformations are shown in Figure 8.

![Figure 8. Deformation at each mode shape of indenter testing rig.](image-url)
3.3. Discussion
Referring to simulation results of the modal analysis as shown in Table 3, the natural frequency should not be coinciding or close to, with the operating frequency, or ranges, of the indenter test rig. The lowest natural frequency out of all modes extracted, set as reference of guideline for structural design [17, 18].

The main criteria for indenter testing rig to operate properly under vibration condition is the natural frequency must not fall in the range of seated human body frequency. For seated human body under vibration condition, it can have significant effect for low frequency up to 50 Hz [19] and some other researchers reported that the fundamental frequency of seated human body under Whole-Body Vibration (WBV) is below 10Hz. In this case, the fundamental frequency for indenter testing rig is 121.63Hz, which way higher than the reported fundamental frequency for seated human body. This result satisfies the criteria of the proposed design for indenter testing rig and can be used in the experimental work.

4. Conclusions
The development of indenter testing rig requires several considerations such as the dimension of the test samples, machine’s mounting size or dimension, ease of fabrication, the force cell, and easy access to material quality. The test rig should be able to operate within the required frequency range and capacity as specified in the standardised method. Modal analysis conducted in this work showed that the first natural frequencies found to be greater than the operating range of the testing rig. This condition satisfies the requirement as specified in the standard for indenter testing rig.

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