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Short Communication

# Automotive seating: the effect of foam physical properties on occupied vertical vibration transmissibility

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## 1. Introduction

There are two aspects of occupant discomfort as it pertains to driving. They are ride quality (sometimes referred to as a dynamic or long-term comfort) and showroom feel (sometimes referred to as static or short-term comfort). At the fundamental level, the principle difference between the two is that ride quality involves exposure to vibration transmitted from the road, through the tires, through the rigid body modes, and through the seat [1]. Managing this vibration, which has been linked to decreased hand–eye coordination, vision impairment, and back disorders [2–4], is an important issue in vehicle design.

Clearly, improvement of ride quality depends upon the control of vehicle vibration. This can be accomplished by manipulating the vehicle parameters. However, changing suspension characteristics, for example, involves a trade-off. By softening the suspension to improve ride quality, vehicle handling suffers. Conversely, a stiff suspension provides good handling but poor ride quality. The improvement of ride quality through the manipulation of seat parameters is, therefore, an active and worthwhile area of research.

The vibrational characteristics of the seat should compliment those of the vehicle; ride quality depends on it. A correctly tuned seat has a natural frequency that does not overlap other vehicle natural frequencies and provides attenuation in the frequency ranges that lead to human discomfort, which, according to Miwa et al. [5], is 5–10 Hz. In composite seats (i.e. those in which

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a foam pad rests on top of a spring system), this tuning is usually accomplished by manipulation of the spring stiffness [6].

A recent trend in the automotive seating industry is the implementation of full foam seating (i.e. a seat design where the foam is placed on a "dead pan" rigidly mounted to the vehicle floor pan). This change in seating design is driven by cost and weight reduction of the assembled seat and green considerations (disassembly for recycle). In a full foam seat there are no springs to adjust and the foam is the sole means of controlling the seat ride dynamics. Understanding the vibrational characteristics of the foam and how the foam dynamics relate to ride quality is crucial to tuning a full foam seat. Said another way, the foam must be engineered just as the springs are in composite seats.

Researchers have, for many years, used transmissibility as an indicator of ride quality [7]. Transmissibility is the non-dimensional ratio of the response amplitude of a seat system in steady-state forced vibration to the excitation amplitude expressed as a function of the vibration frequency [2]. The ratio may be one of forces, displacements, velocities, or accelerations. The most direct method of measuring the transmissibility is to compare the acceleration on the seat with that at the base of the seat. To accomplish this, standard practice is to acquire signals provided by accelerometers mounted at the base of a seat (e.g. floor attachment points) and at the interface between the seat surface and the human body.

The transmissibility of a seat can be measured on any axis (e.g. vertical, lateral, or horizontal) or at any point (e.g. beneath the ischial tuberosities or between the human back and the seatback). However, most of the published studies involve only the vertical transmissibility from the seat base to the ischial tuberosities, as this is thought to be the primary axis affecting ride quality [8,9].

Research endeavors in the automotive seating industry are focused on optimizing the physical properties of foam. These properties can be audited for quality assurance in manufacturing and are thought to relate to transmissibility performance, particularly when coupled with structure and trim characteristics. Two of the most commonly employed foam properties are density and firmness. Density is important because it corresponds to cost and mass. Firmness is important because it is critical to the end consumer. It affects (1) subjective perceptions of comfort, (2) the level of accommodation the seat provides to occupants with different anthropometric characteristics (foam, because it complies when loaded, is forgiving), and (3) the feasibility of attaining appearance/craftsmanship expectations. Firmness is typically determined from a force deflection curve [1]. The actual metric, referred to as indentation force deflection (IFD), represents the reaction force at a specific load point in the force deflection curve. Hilyard and Collier [10] suggest that it is most meaningful to assess IFD at 25% compression, although there is no universal standard. For more detail on the measurement of IFD, in terms of equipment and test conditions, the interested reader is referred to ISO 2439 [11].

The purpose of this paper was to determine the effect of foam density and firmness on ride quality as indicated by occupied vertical vibration transmissibility.

## 2. Method

To begin, a seat system with a full foam cushion was selected to act as the experimental platform. Production level tools from this seat system, which was from a popular North American compact car, were used to create four different cushion foam pads. The foam pads were poured at two levels of density (high and low) and two levels of IFD (high and low). The levels are described in Table 1. By using one seat system it was possible to control the entire seatback subassembly, the cushion A-surface contour, and the cushion thickness, thereby eliminating the potential confounding effect of these parameters. As an aside, it is most critical to control foam thickness in an investigation like this because a change in foam thickness will affect firmness. That is, the IFD measurement is meaningless without information on thickness (i.e. two pads with the same IFD but different thickness will feel/perform differently). The selected seat system had a cushion thickness of approximately 100 mm at the point where IFD was measured. Also, the investigation was performed without cushion trim covers.

The seat system was mounted to a six-axis, hydraulically actuated, human-rated shaker table in design position using a fixture and a carpeted heel plate. It is standard practice, in the automotive seating industry, to evaluate seats in their design position, which refers to the manufacturer-specified chair height, cushion angle, and seatback angle (as opposed to an occupant-selected seat position). The design position for this study included a chair height of 290 mm (measured from the floor to the H-Point, which is based on a manikin that represents how medium-sized men sit in, and interact with, different vehicle seats and vehicle environments [12]), a cushion angle of 20° (measured from horizontal to the cushion surface), and a seatback angle of 110° (measured from horizontal to the seatback surface).

A low-profile accelerometer was then placed on the top of the seat cushion (positioned in the ischial tuberosity region) with the sensitive axis at a right angle to the cushion surface. Another accelerometer was mounted to the top of the shaker table approximately beneath the first accelerometer with the sensitive axis in the $Z$ direction. The ICP piezoelectric accelerometers were manufactured according to SAE standard J1013 [13].

Nine occupants, three females and six males, participated in this repeated measures experimental design. Their anthropometric characteristics, obtained in a self-report fashion, are included in Table 2.

The occupants, after emptying their pockets, were allowed to adjust only the fore/aft position of the seat track. This provision was made so that all occupants could comfortably reach the carpeted heel plate. The test subjects were instructed to maintain a relaxed and alert position at all times. They were reminded to sit with both heels on the heel plate and with both hands in their lap. As an aside, subjects were not permitted to use the head restraint or the arm rest.

Finally, the accelerometer channels were connected to a data acquisition system, which was set to use a 256 Hz sampling rate. A random, "white noise" signal (one containing the full spectrum

Table 1
Foam physical properties used for experimental treatments

| Experimental treatment | Foam density (g) | Foam IFD (N at 25% compression) |
|---|---|---|
| A | 1210 (low) | 166 (high) |
| B | 1462 (high) | 145 (low) |
| C | 1378 (high) | 164 (high) |
| D | 1214 (low) | 135 (low) |

Table 2
Anthropometric characteristics of test occupants and corresponding classification

| Subject | Stature (cm) | Body mass (kg) | Gender |
|---|---|---|---|
| 1 | 178 | 102 | Male |
| 2 | 175 | 70 | Male |
| 3 | 178 | 68 | Male |
| 4 | 178 | 102 | Male |
| 5 | 173 | 66 | Male |
| 6 | 163 | 58 | Female |
| 7 | 188 | 100 | Male |
| 8 | 170 | 57 | Female |
| 9 | 158 | 59 | Female |

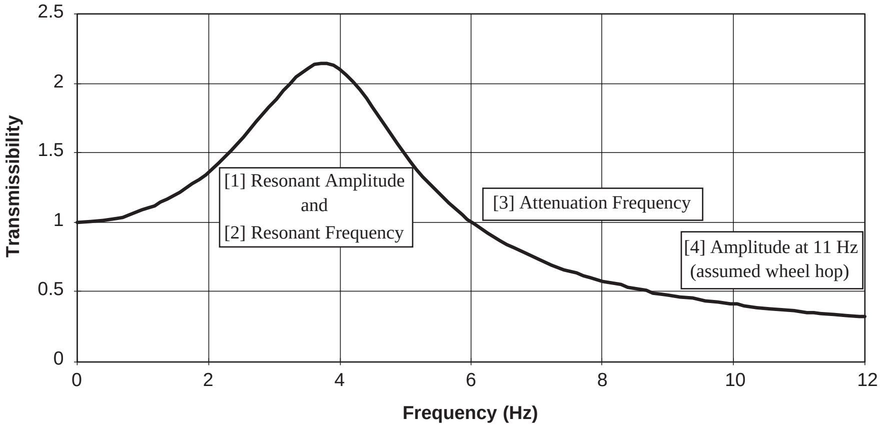


Fig. 1. Typical occupied vertical vibration transmissibility output.

of frequencies up to 50 Hz) was input into the shaker table with a constant input acceleration of $0.2\,m/s^2$ (rms). The signal, which spanned 1 min, was set to excite the table in four successive passes or loops.

Fig. 1 shows a typical transmissibility vs. frequency plot produced using the previously described method. For the purposes of this research, four transmissibility characteristics were defined. Two measures were related to the resonant peak region. In this region, the response vibration is greater than the input vibration. In other words, the input vibration is amplified. The two measures are resonant amplitude (related to seat bounce, which is the principle concern in the spillage of drinks [2]) and resonant frequency (the frequency at which the peak amplitude is manifested).

Designers strive to develop seats with an occupied resonant frequency that falls between the resonant frequency of the typical body-in-white (usually less than 2 Hz) and the range of human sensitivity (i.e. 5–10 Hz). From the perspective of ride quality, it is imperative that the seat system attenuates vibration in the 5–10 Hz range. As attenuation occurs only after the natural frequency, a lower resonant frequency is desired to expand the attenuating properties into lower frequencies (i.e. a lower natural frequency correlates with a lower attenuation frequency in the spring/dashpot model). Once a lower natural frequency is obtained, a lower transmissibility at the resonant frequency would be beneficial also, as road noise is present to some extent at all frequencies. The optimum balance of precedence between low resonant frequency and low peak transmissibility is currently a matter of debate [2]. Based on the preceding discussion, the third transmissibility characteristic defined for this study was the attenuation frequency (Fig. 1), or the frequency at which the transmissibility crosses below one (after the resonant frequency).

The fourth transmissibility characteristic defined for this study was the amplitude at 11 Hz (Fig. 1). This characteristic was included because it is assumed to correspond to wheel hop in a typical vehicle vibration spectrum. The amplitude at 11 Hz should be as low as possible.

## 3. Results and discussion

Fig. 2 and Table 3 provide a qualitative assessment of the effect of foam on occupied vertical vibration transmissibility. IFD appears to impact resonant amplitude (firmer foam produces a lower resonant amplitude), while density appears to impact attenuation frequency and transmissibility at 11 Hz (high-density foam tends to (1) attenuate more quickly and (2) produce a lower amplitude at the assumed wheel hop). The combination of high IFD and high density seems to produce the most optimal condition (i.e. the best ride quality).

A series of nonparametric statistical analyses were used to quantify the effect of IFD and density. A separate Kruskal–Wallis test was performed for each of the four measures (i.e. resonant

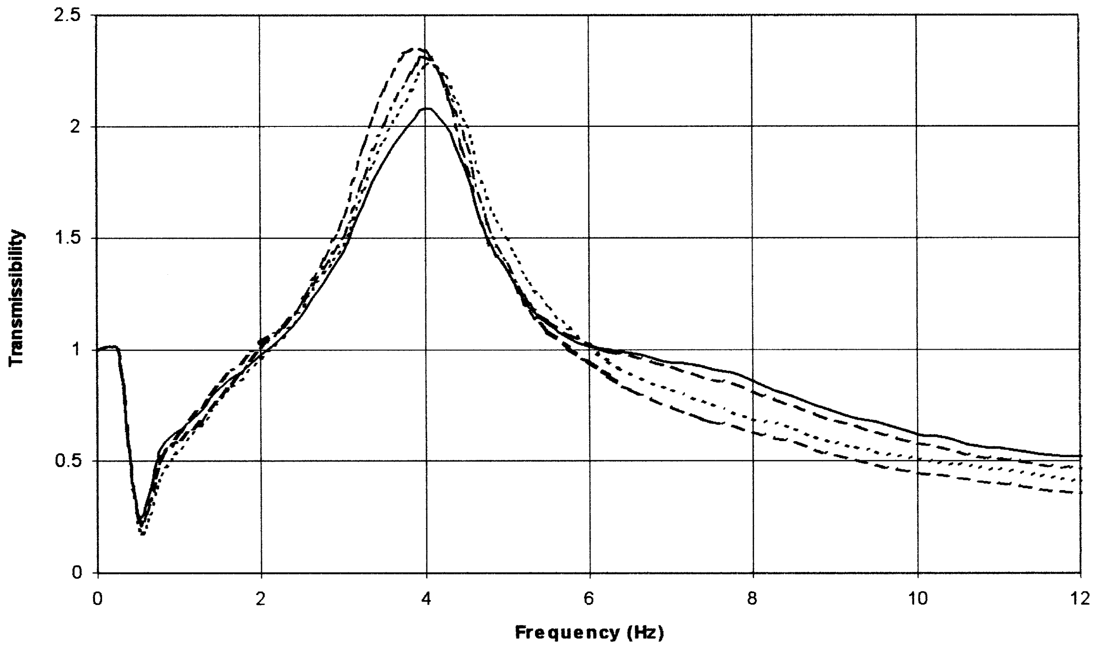


Fig. 2. Graphic representation of foam effects on occupied vertical vibration transmissibility: ———, treatment A; - - - - - - - - - -, treatment B; . . . . . . . . ., treatment C; -.-.-.-.-.-.-., treatment D.

Table 3
Qualitative assessment of foam effects on occupied vertical vibration transmissibility

| | Treatment A | Treatment B | Treatment C | Treatment D |
|---|---|---|---|---|
| Resonant amplitude | 2.08 | 2.34 | 2.28 | 2.31 |
| Resonant frequency | 4 | 4 | 4 | 4 |
| Attenuation frequency | 6.25 | 5.75 | 6.25 | 6.25 |
| Amplitude 11 Hz | 0.56 | 0.41 | 0.46 | 0.51 |

Table 4
Kruskal–Wallis test for effect of IFD

| | Resonant amplitude | Resonant frequency | Isolation frequency | Amplitude at 11 Hz |
|---|---|---|---|---|
| Chi-square | 5.336 | 0.413 | 0.953 | 2.707 |
| df | 1 | 1 | 1 | 1 |
| Asymp. sig. | 0.021* | 0.521 | 0.329 | 0.100 |

*Statistically significant difference between levels of IFD.

Table 5
Kruskal–Wallis test for effect of density

| | Resonant amplitude | Resonant frequency | Isolation frequency | Amplitude at 11 Hz |
|---|---|---|---|---|
| Chi-square | 1.484 | 0.083 | 1.049 | 11.036 |
| df | 1 | 1 | 1 | 1 |
| Asymp. sig. | 0.223 | 0.774 | 0.306 | 0.001* |

*Statistically significant difference between levels of density.

amplitude, resonant frequency, attenuation frequency, and amplitude at 11 Hz). The IFD results are captured in Table 4 and the density results are captured in Table 5.

Using a decision criterion of .05, IFD was found to significantly effect resonant amplitude. Fig. 3 reveals that the average resonant amplitude with a high IFD foam was 2.18 and the average resonant amplitude with a low IFD foam was 2.33. This was an expected result. Firmer foam produces less bounce at low frequencies.

Density significantly affected amplitude at 11 Hz (at the .05 level). The average amplitudes were 0.43 for the high-density foam and 0.54 for the low-density foam. These effects can be visualized in Fig. 4.

The transmissibility differences associated with the experimental treatments were assumed to be large enough to influence subjective perceptions of ride quality in any environment where there is

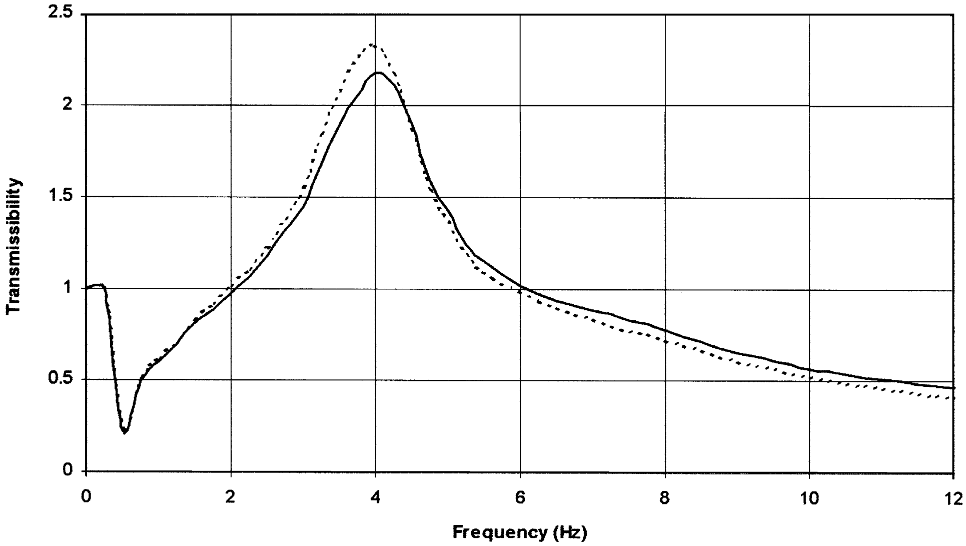


Fig. 3. Statistically significant occupied vertical vibration transmissibility effects attributable to IFD: ———, high IFD (resonance amplitude = 2.18); ........., low IFD (resonance amplitude = 2.33).

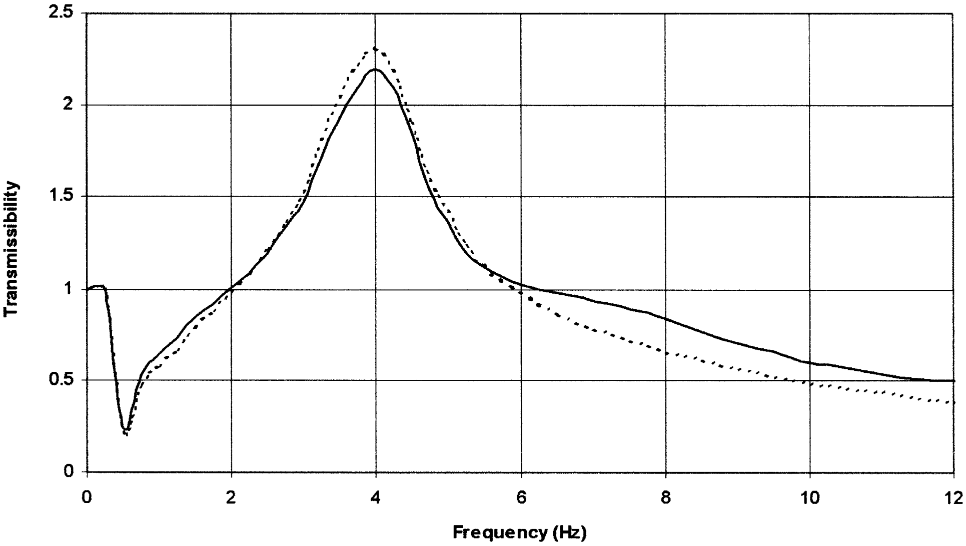


Fig. 4. Statistically significant occupied vertical vibration transmissibility effects attributable to density: ———, low density (amplitude at 11 Hz = 0.53); ........., high density (amplitude at 11 Hz = 0.43).

significant vertical vibration. It would be interesting, as part of future research, to verify the relationship between perceptions of ride quality and measures of transmissibility (as collected in this study).

## 4. Conclusion

In some market segments, the consumer may want or even expect to feel more or less of the road (compare compact SUV buyers to luxury car buyers). This investigation demonstrated that designers of full foam automobile seat systems could meet these expectations by manipulating IFD and density.

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