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State-of-the-art in recent developments concerning suspension seats

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PEER REVIEW
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SUMMARY

Vehicle drivers in various work sectors in Quebec are exposed to significant levels of whole-body vibration (WBV) and intermittent shocks, which have been associated with an increased risk of lower back pain and degeneration of the spine. Suspension seats are widely used to limit WBV exposure of operators. The vibration reduction performance of a suspension seat is strongly affected by the nature of vibration (magnitude, direction and frequency components) of the vehicle. An optimal performance of a suspension seat can thus be realized only when the suspension is tuned for the specific target vehicle.

Suspension seat manufacturers generally recommend a particular suspension seat design for a wide range of vehicles, which may not provide optimal suspension performance considering significant differences in vibration characteristics of vehicles. Although widely different designs of suspension seats have been commercially available for the past decades, only limited knowledge exists in the tuning and in the adaptation of suspension seats to specific vehicles.

The researchers have frequently encountered inquiries from vehicle operators regarding specific recommendations for an appropriate suspension seat for their vehicle. This study was motivated considering the need for a suspension seat advisor for vehicle operators. The overall objective of this activity was formulated to conduct a review of the state-of-the-art developments in suspension seats so as to gain knowledge towards an eventual development of a suspension seat advisor. The specific goal of the activity is twofold. The reported advances in suspension seats are firstly explored through a comprehensive review of scientific publications and patents, to identify desirable technical features and to gain knowledge of methods for designing vehicle-specific suspension seats. The technical features of commercially available suspension seats are subsequently reviewed to build knowledge of the performance characteristics of suspension seats for different vehicles in view of the attenuation of multi-axis vibration and of adequate ergonomic design factors.

This research activity involved comprehensive reviews of various technical features of commercially available suspension seats, and critical reviews of reported technical advances in passive, semi-active and active suspension seats, including the performance characteristics and the assessment methods. The technical features of commercially available suspension seats were mostly compiled from the data provided by the manufacturers' websites. This was supplemented by personal discussions with leading suspension seat manufacturers in the USA and in Europe, as well as with a leading manufacturer of agricultural machines in the USA. The review was particularly focused on vibration isolation performance of suspension seats and their adaptation to vehicles employed in different work sectors such as construction, forestry, mining, agriculture, material handling and public transportation. In addition, ergonomic design features of suspension seats, namely, lumbar support, cushion design, height/weight adjustment and other adjustments were gathered and examined. The scientific studies reporting advances in suspension seat design were reviewed with particular focus on: (i) performance assessment methods; (ii) concepts in passive, semi-active and active suspension seats and their practical implementations; (iii) developments in fore-aft and lateral vibration isolators; (iv) laboratory/field assessments of suspension seats; and (v) numerical modelling and assessments.

From the reviews of the available suspension seats, it was evident that the vast majority of them employed a cross-linkage platform, with either mechanical or air spring, one or two hydraulic dampers and elastic suspension travel limiters. All designs provided an adjustable seat height, which was generally coupled with an adjustment for the occupant weight. Some designs provided automatic ride height adjustment to ensure mid-ride suspension position and thereby a reduced risk of shocks induced by interactions with the suspension travel limiters. The seats also provided a fore-aft adjustment apart from adjustable cushion and backrest inclinations in order to provide more comfortable and controlled sitting posture. Some of the seats were equipped with an either fixed or adjustable lumbar support, although a quantitative assessment of such support could not be found. In addition, the same suspension designs were recommended for widely different vehicles with notably different WBV patterns, while very little to negligible information was available on shock/vibration isolation performance of the different seats. Moreover, data on the effectiveness of fore-aft and lateral seat isolators were not available.

In recent years, developments in semi-active and active suspension systems have been emphasized and a large number of controller designs have been proposed to achieve enhanced attenuation of WBV. These are mostly focused on vertical vibration isolation. The practical implementations of active suspension seats have been limited to only a few, while a number of manufacturers have developed semi-active suspension seats using controllable magnetorheological dampers. These devices have shown a superior performance in eliminating impacts with the suspension travel limiters, but with only minimal gain in the vibration isolation effectiveness of the suspension. Moreover, the assessments of active and semi-active suspension seats have been limited to only a few classes of vehicle excitation.

From the state-of-the-art reviews of the commercially available suspension seats and of the reported technical advances, it is concluded that only limited knowledge exists on vehicle-specific suspension seat designs. Furthermore, the manufacturers offer only limited designs of add-on horizontal isolators for the control of fore-aft and lateral WBV exposure. In addition, the vibration isolation effectiveness of such isolators has not been examined. Further analytical and experimental efforts are thus highly desirable in developing methods for assessing vehicle-specific vibration isolation performance of suspension seats, so as to develop a reliable suspension seat advisor in the future.

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1. INTRODUCTION

Drivers of vehicles in forestry, mining, agriculture, and freight and public transportation are exposed to comprehensive levels of low frequency whole-body vibration (WBV) and intermittent shocks. Apart from the discomfort, driver fatigue and poor performance rate, WBV is linked epidemiologically to greater risks of low back pain and degenerative changes in the spine among the exposed drivers (Bovenzi, 2017; Burström, Nilsson & Wahlström, 2015). It has been estimated that 4 to 7% of all employees in the USA, Canada and Europe are exposed to potentially harmful WBV with annual costs of health care, compensation and lost workdays of several billion dollars (Ekman, Johnell & Lidgren, 2005; Punnett et al., 2005), although this entire cost cannot be attributed only to WBV due to several confounders, especially the prolonged sitting within a confined space. While current epidemiological data are not sufficient to define dose-response relationships, control of WBV has been emphasized together with effective frequency weighting for assessing the risks (Lings & Leboeuf-Yde, 2000; Plewa, Eger, Oliver & Dickey, 2012).

Suspension seats are the most convenient and common means of limiting WBV exposure of operators, while providing a controlled sitting posture for an effective vehicle operation. A seat constitutes the primary contact point for the transmission of WBV to the vehicle driver through the seat cushion and suspension, directly beneath the ischial tuberosities. A seat is also the final suspension mechanism for isolating the driver from the terrain-induced WBV. Proper selection of the seat for a given vehicle is thus vital for limiting the human driver's WBV exposure. The vibration and shock isolation properties of seat-suspension systems have been widely studied experimentally and analytically (Blood, Dennerlein, Lewis, Rynell & Johnson, 2011; Blood, Ploger & Johnson, 2010; Blood, Ploger, Yost, Ching & Johnson, 2010; Boileau & Rakheja, 1997; Dong & Lu, 2012; Ma, Rakheja & Su, 2008a; McManus, Clair, Boileau, Boutin & Rakheja, 2002; Rakheja, Boileau & Wang, 2004; Rakheja, Boileau, Wang & Politis, 2003). Although the WBV environment of several work vehicles comprises significant vibration equally along the vertical, fore-aft and lateral directions (Rakheja, Mandapuram & Dong, 2008), reported studies have mostly focused on the vertical suspensions alone. This is partly due to the lack of effective suspension systems for attenuation of very low frequency horizontal vehicular vibration, which generally predominate around 1 Hz. These studies have shown that the vibration isolation effectiveness of a suspension seat is strongly influenced by a number of design and operating factors, including terrain roughness, vehicle tires and suspension properties, vehicle load, suspension seat characteristics, forward speed, body mass, and suspension seat ride height with respect to the motion limiters.

Although effective designs of various wheel suspensions have evolved to limit WBV exposure (Cao, Rakheja & Su, 2010; Pazooki, Rakheja & Cao, 2012; Uys, Els & Thoresson, 2007), vibration control in small to medium-size vehicles, which constitute the majority of the work vehicles in Quebec, is limited only to a suspension at the driver seat in addition to large and soft tires. This is partly due to the strong coupling between the ride and the roll/directional stability of suspended vehicles, which imposes conflicting design requirements for wheel suspensions. Wheel suspensions are designed with higher weighting on roll/slope stability than on ride vibration. Moreover, the ride and handling/stability performance characteristics of wheel suspensions lie on opposite ends of the design space (Els, Theron, Uys & Thoresson, 2007). Despite wide usage of suspension seats, the WBV exposure in many off-road vehicles is known to exceed the limits of the health guidance caution zone as defined in the ISO 2631-1:1997 standard (International Organization for Standardization [ISO], 1997) and the action limit of the European *Directive*

2002/44/EC. Moreover, the increasing demand for vehicle operations at relatively higher speeds in some sectors such as open-pit mining and construction is likely to contribute to even higher magnitudes of WBV.

Reported studies have shown that vibration isolation effectiveness of a suspension seat is strongly and nonlinearly dependent upon the nature of vibration encountered in a particular vehicle, namely, the magnitude and dominant frequencies of vibration. The performance of a vertical suspension seat can be described in three distinct categories depending on the nature of vehicle vibration: (i) suspension lock-up under low levels of vehicle vibration due to friction; (ii) attenuation or amplification of vibration under medium to higher levels of continuous vibration, leading to suspension travel within the permissible free travel; and (iii) amplification of vibration and shock motions when the suspension travel exceeds its free travel, leading to impacts against the elastic end-stops. The shock and vibration isolation performance of the suspension seat within the last two categories pose conflicting design requirements, particularly for the suspension damping. Moreover, the suspension performance within these two categories is of primary concern for most work vehicles, where the WBV comprise large magnitude vibration and intermittent shock motions. Light damping is desirable for attenuation of continuous vibration when the suspension motion is limited to its free travel, while the potential impact against the elastic end-stops under large magnitude vibration or shock excitation can be reduced via higher damping (Boileau, Rakheja & Wang, 2004; Rakheja et al., 2004).

The design of a suspension seat also involves additional challenges associated with variations in the operators' body mass and seat height. Variations in body mass may affect suspension natural frequency and thereby the vibration isolation performance (Hostens, Deprez & Ramon, 2004; Rakheja et al., 2003; Zhou, Zhao, Yu, Yang & Wang, 2018). A suspension seat yields its best vibration isolation performance when adjusted to its mid-ride position so as to permit maximum suspension travel in compression and rebound. However, the effective suspension stiffness, especially for air suspensions, and the permissible suspension travel are affected by the driver selected seated height, which may lead to reduced vibration isolation and impacts with the elastic end-stops (Boileau et al., 2004).

The vibration isolation characteristics of suspension seats are generally evaluated in the laboratory using methods described in standards such as ISO 7096:2000 (2000), which requires measurements with a seat adjusted to mid-ride position and loaded with human subjects of specific body mass, namely 52 to 55 kg and 98 to 103 kg. A few studies have also measured seat-suspension performance in the field and suggest that the field-measured vibration transmissibility magnitudes are generally higher than those obtained in the laboratory (Burdorf & Swuste, 1993). This has been attributed to suspension friction and end-stop impacts, which may not be encountered during the standardized laboratory tests. Studies reporting field measured vibration performance of vertical suspension seats employed in wheel loaders and forestry skidders have shown that they provide either a limited attenuation or even an amplification of vertical vehicle vibration transmitted to the driver (Cation, Jack, Oliver, Dickey & Lee-Shee, 2008; Wegscheid, 1994). This has been attributed to a lack of suspension tuning in order to adapt to the magnitude and frequency contents of the target vehicle vibration (Tiemessen, Hulshof & Frings-Dresen, 2007). Suspension seat manufacturers generally recommend an identical design for a broad range of vehicles, whose vibration may differ substantially. This approach cannot provide optimal vibration reduction by the seat. The design/tuning of a suspension seat to a target vehicle vibration is thus vital for limiting the WBV exposure of the drivers.

The designs of vehicle-specific suspension seats have not been adequately addressed thus far, which is partly due to a lack of appropriate guidance and partly due to various design complexities and constraints. Several studies have provided only general guidelines for the design of suspension seats, which are not quantitative for applications towards vehicle-specific seat designs. These are summarized below:

(a) A vertical suspension seat for wheeled vehicles employed in the construction, mining and forestry sectors must be designed with low natural frequency, in the order of 1.5 Hz or lower. This will ensure adequate vibration isolation and limit suspension motion within its free travel (Hostens et al., 2004). Even lower natural frequency is required for horizontal suspensions, since these dominate at very low frequencies in the order of 1 Hz (Rakheja et al., 2008).

(b) On-road vehicles with primary wheel suspensions such as city buses must use a suspension seat with a natural frequency well below 1.5 Hz or the vertical dominant frequency of the vehicle (Boileau et al., 2004).

(c) Low natural frequency designs cause large suspension motion and thereby require greater headroom, and yield sensation of discomfort among the drivers. Suspension motion is thus limited by the elastic end-stops, which transmit shocks to the occupant under large magnitude vibration or intermittent motions caused by wheels interactions with obstacles or terrain discontinuities (Boileau et al., 2004; Wu & Griffin, 1997).

(d) Light suspension damping is required to achieve a reduction of vibration in the absence of interactions with end-stops. High damping, however, is vital for limiting resonant vibration and shock motions, and for reducing impacts against the motion limiting end-stops (Ma et al., 2008a).

(e) Suspension natural frequency must not be very sensitive to variations in body mass (Boileau & Rakheja*,* 1997).

(f) A suspension seat should provide easy adjustment to achieve mid-ride height and comfortable posture for the driver. Inadequate height adjustment can lead to more frequent impacts with end-stops leading to higher magnitude of transmitted vibration (Boileau & Rakheja, 1997; Wu & Griffin, 1997). Automatic height adjustment is thus preferred.

In recent years, suspension seat manufacturers have emphasized vibration reduction performance of seats and their adaptability to different classes of vehicles together with various ergonomic aids to provide ease of adjustments and greater sensation of comfort by the drivers. These include controllable magnetorheological (MR) suspension dampers (McManus et al., 2002), and controllable semi-active and active suspensions with pneumatic or electric actuators to achieve improved vibration reduction performance under varying magnitudes of base vibration (Blood et al., 2011); active fore-aft and lateral isolation modules for limiting the transmission of horizontal vibration (Sun, S. et al., 2015); gel foam or air seat cushion designs with contours to minimize concentration of high and localized contact pressure and thus subcutaneous stresses (Lee, S.-H., Park, Jung & Lee, 2016); automated height adjustment; adjustable air or mechanical lumbar support; and more. Only limited data, however, are available to assess vibration isolation effectiveness of different seat designs as applied to specific vehicles.

This activity report presents a critical review of reported technical advances in passive, semi-active and active suspension seats including performance characteristics and assessment methods, and state-of-the-art developments in suspension seats compiled from a comprehensive review of the reported technical features of commercially available suspension seats. The design approaches used for adapting suspension seats to specific vehicles are particularly discussed.

2. OBJECTIVE

The overall objective of this activity is to conduct a review of the state-of-the-art developments in suspension seats so as to gain knowledge towards an eventual development of a suspension seat advisor. The specific goal of the activity is twofold:

To report the advances in suspension seats through a comprehensive review of scientific publications and patents in order to identify desirable technical features and to gain knowledge of the current methods for designing vehicle-specific suspension seats.

To review the technical features of commercially available suspension seats to build knowledge of performance characteristics of suspension seats for different vehicles in view of attenuation of multi-axis vibration and ergonomic design factors.

3. METHODOLOGY

3.1 Review of the relevant literature

In order to build knowledge regarding the reported advances in suspension seat designs, assessment and analysis methods, and performance characteristics, a thorough review of studies reporting suspension seats design concepts, analytical and numerical models, controller syntheses, assessment methods, and performance assessments via numerical simulations, laboratory and field measurements, was conducted. A systematic search of studies reporting developments in suspension seats and WBV control was undertaken using Google Scholar and Google. The following keywords were used: ‘suspension seat’, ‘active suspension seat’, ‘semi-active suspension seat’, ‘suspension seat performance’, ‘whole-body vibration exposure’, ‘horizontal seat suspension’, ‘fore-aft seat suspension’, ‘vehicle-specific seat suspension’, ‘seating ergonomics’, and ‘seating dynamics’. The search was limited to publications appearing since 1990 in the English language.

The articles were screened and those fitting the goals of the activity were retained for the review. Studies reporting suspension seat designs for automobile driver/passenger comfort were excluded. The search resulted in a number of articles reported in various vehicle and vibration journals (Journal of Vehicle Design, Journal of Heavy Vehicles Systems, Journal of Sound and Vibration, Applied Ergonomics, Journal of Automobile Engineering, SAE Journal of Commercial Vehicles, International Journal of Industrial Ergonomics, etc.) and proceedings of the relevant conferences such as those of the Society of Automotive Engineers (SAE).

The search also identified a few reports published by NIOSH and IRSST, apart from a number of patents. However, the discussion of the reviewed patents was omitted in the report, except for one patent, either because the topics were already addressed in one of the other reviewed items, or because they did not provide knowledge regarding the vibration isolation performance of the patented design or design concept. In recent years, the vast majority of the patents have focused on design concepts and syntheses of semi-active and active suspension seats. These patent reports, however, do not provide any information on their vibration performance. Moreover, some of them focused on locking the suspension to ensure driver safety in the event of extreme events such as excessive braking, or potential rollover or an accident.

Finally, the identified reports/articles were grouped so as to limit the focus on passive, semi-active and active-suspension designs and concepts, laboratory/field assessments, analytical modelling and vibration isolation analyses, performance analyses and applications in the context of specific classes of vehicles. The reported studies mostly focused on the vertical seat suspensions and only a few studies could be found on the fore-aft and lateral seat isolators.

3.2 Review of commercially available suspension seats

In this section, a review of the suspension seats available in the market was performed from the information available on the manufacturers’ websites. The review was conducted on the main features of the seats that can be considered for the selection of a commercial seat for a specific machine and application. Eleven seat manufacturers were considered in this review, namely Grammer, KAB, Sears Seating, Isringhausen, Bose, National Seating, Recaro, Baultar, Knoedler, Amobi Seats, and USSC Seats. The team also conducted personal interviews with engineering professionals of two suspension seat manufacturers in the USA (Sears Seating) and Europe (Grammer) with regard to guidance on selection of seats for particular machines, and to vibration performance of the available seats. The team also met with the engineering staff of a vehicle manufacturer (AGCO, USA) in order to seek user perspective of the suspension seats performance.

4. REVIEW OF RELEVANT LITERATURE

The vibration environment of heavy on-road and off-road vehicles employed in public transit, freight transportation, construction, industrial and resource sectors comprises high amplitudes of low frequency vibration and repeated mechanical shocks or transient vibration in the 1-20 Hz frequency range (Périsse & Jézéquel, 2000a). Mechanical shocks generally arise from travel over rough terrains or discontinuities in the roadways/terrains. Exposure to such vibration and shocks has been related to the occurrence of health disorders among the drivers. In an effort to control whole-body vibration exposure, the European Union has set forth a directive (*Directive 2002/44/EC*), which mandates the WBV exposure limit of 1.15 m/s^2 and an action level of 0.5 m/s^2 for an equivalent 8-hour exposure period measured in accordance with the ISO 2631-1:1997 (1997) standard. This standard also defines a health guidance caution zone for determining potential health risks due to WBV exposure. A number of vehicles in the construction, mining and forestry sectors, however, show exposure levels in excess of the health exposure limits and action limit value (Cation et al., 2008; Gunaselvam & Van Niekerk, 2005).

The transmission of WBV to the vehicle operators occurs through the seat cushion, directly beneath the ischial tuberosities (Griffin, 1990). A suspension of the vehicle seat is considered vital for limiting the WBV exposure. This is particularly important for the small to mid-size vehicles without primary suspensions (Jack et al., 2010; Rakheja, Kordestani & Marcotte, 2011). The developments in low frequency suspension seats have definitely contributed towards reductions in vibration exposure of the drivers and thereby reducing the associated health and safety risks. Such developments, however, have been mostly limited to vertical suspensions, although the vast number of off-road vehicles exhibits significant levels of vibration equally along the lateral (Y) as well as the longitudinal (X) directions (Cation et al., 2008; Eger, Kociolek & Dickey, 2013; Marin et al., 2017; Rakheja et al., 2008). The vertical suspension seats in many applications, however, have been shown to amplify the base vibration (Cation et al., 2008; Gunaselvam & Van Niekerk, 2005; Lines, Stiles & Whyte, 1995), which is likely due to a lack of tuning or design of suspension systems for particular vehicles (Shangguan, Shui & Rakheja, 2017). Vibration reduction performance of a suspension seat is related to many design and operating factors in a highly complex manner.

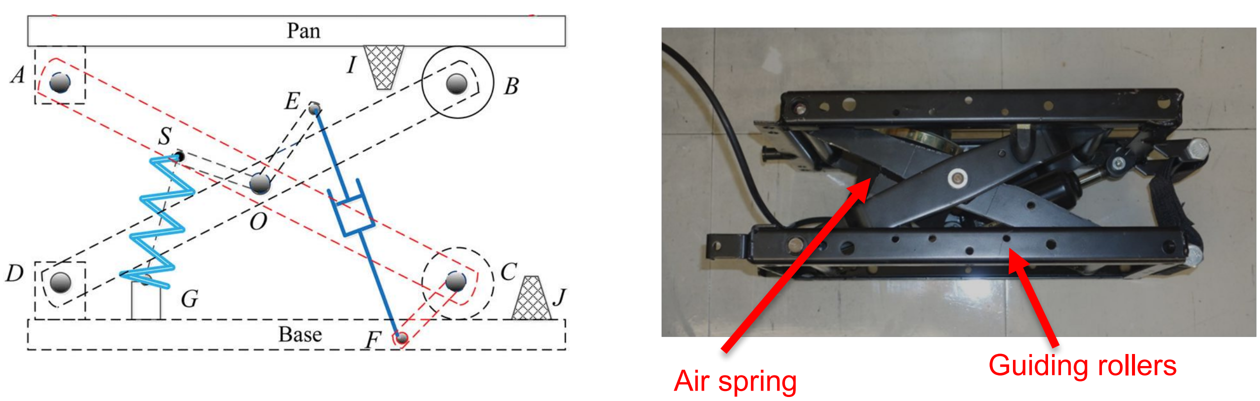
Different designs of passive, semi-active and active suspension seats have been extensively studied experimentally as well as analytically. The reported performance characteristics of various designs and methods of analyses are described in the following subsections together with the important factors influencing the performance.

4.1 Suspension design features and design challenges

Although the reported studies have investigated different designs of vertical seat suspension systems for on-road as well as off-road vehicles, the designs exhibit common essential components. These include: (a) a linkage mechanism that ensures nearly vertical motion of the sprung seat pan; (b) a mechanical or pneumatic spring; (c) a hydraulic or gas damper; (d) elastic bump-stops limiting the motion of the seat pan with respect to the seat base and the vehicle controls; (e) a seat cushion; (f) seat height and fore-aft adjustments; and (g) body weight adjustment. Contoured seat cushions are primarily used to provide comfortable sitting posture and reasonable body pressure distribution (Ebe & Griffin, 2001; Grujicic et al., 2009). A seat cushion interestingly allows the decoupling of static and dynamic comfort performances (Périsse

& Jézéquel, 2000a). The dynamic comfort performance of the seat is mostly dependent on the suspension design. The majority of the vertical suspension seats employed for on-road as well as off-road vehicles comprise a seat pan supported by a cross-linkage mechanism, which ensures nearly vertical movement of the suspended seat pan and a compact design. The suspension system consists of either a mechanical or an air spring together with a hydraulic damper, installed within the cross-linkage frame, as shown in Figure 4.1. Some of the highway vehicle seats employ parallel link platform, as shown in Figure 4.2(a). Compact behind-the-seat-suspension designs have also evolved for small size industrial vehicles with limited cabin space. In this design, the suspension spring and damper are integrated within a column support located behind the seat, as seen in Figure 4.2(b). Suspension designs offer essential adjustments for fore-aft positioning of the seat, seated height, body weight, and cushion and backrest inclinations.

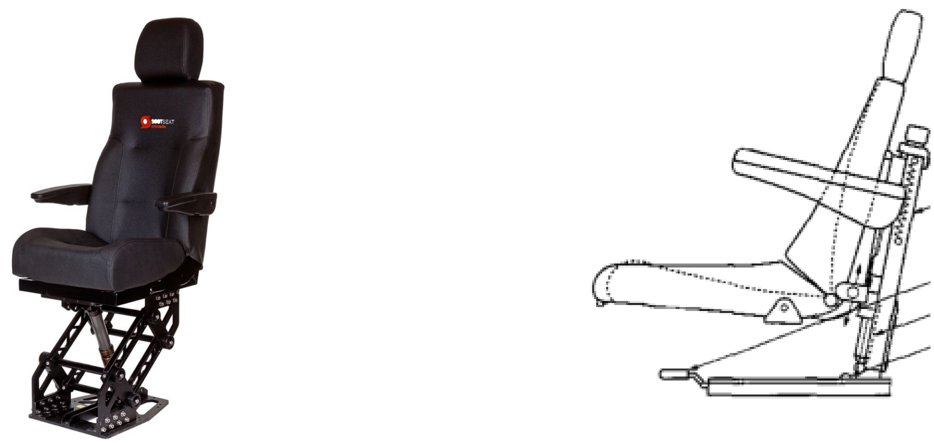
In the most commonly used cross-linkage design, the two ends of each link are supported via a hinge joint and a roller joint, which is permitted to move horizontally in a guiding track (Figure 4.1). The roller motion within the guiding track yields considerable friction force that would depend upon the instantaneous normal load on the guided supports. Owing to their low natural frequency design (1 to 1.5 Hz), seat suspensions exhibit considerable motion of the seat pan with respect to the seat base, which may interfere with the operator's control activities. Elastic end-stops are thus incorporated within the suspension to limit maximum seat travel. The vibration isolation properties of the seat suspension are related not only to the component characteristics but also many operating factors in a highly complex manner.



(a) (b)

Figure 4.1. (a) Illustration of a cross-linkage seat suspension system with mechanical and air springs. (b) pictorial view.

(a) From "The kineto-dynamic analysis and optimal design using suspension seat-human couple model", by Y. Shui, 2016. ©Y. Shui, 2016. Reprinted with permission.



(a) Parallel-link suspension (b) Behind-the-seat suspension

Figure 4.2. Illustrations of suspension designs: (a) parallelogram. (b) behind-the-seat.
(a) From "De « X-Craft C-FORCE suspension seat", n.d. ©NauticExpo, 2021. Retrieved from : https://www.scotseats.co.uk/s2h-seat. Reprinted with permission.
(b) Adapted from "Étude des paramètres affectant l'efficacité d'atténuation des vibrations par un siège suspendu", by P.-É. Boileau and S. Rakheja, 1995. ©IRSST, 1995.

Wu, Rakeja and Boileau (1999) have described the performance of a vertical suspension seat in distinct categories depending on the nature of vehicle vibration: (i) suspension lock-up under low levels of vehicle vibration due to friction; (ii) amplification of base vibration as the friction breaks away under increasing levels of continuous vibration, leading to suspension travel within the permissible free travel, depending upon the frequency components of vehicle vibration and suspension design; (iii) attenuation of vibration under increasing levels of vibration and suspension motion, as the friction becomes less dominant; and (iv) increase in transmitted vibration and shock motions with further increase in base vibration magnitude, when the suspension travel exceeds its free travel, leading to impacts against the elastic end-stops.

The shock and vibration isolation performance of a suspension seat within the last two categories pose conflicting design requirements, particularly for the suspension damping. Moreover, the suspension performance within these two categories is of primary concern for most work vehicles, where the WBV comprises large magnitude vibrations and intermittent shock motions. Light damping is desirable for the attenuation of continuous vibrations, when the suspension motion is limited to its free travel, while the potential impacts against elastic end-stops under large vibration or shock excitation can be reduced via higher damping (Boileau et al., 2004; Rakheja et al., 2004). Since the magnitude of vehicle vibration varies considerably during the operation, the design of a suspension seat poses complex compromises, which are summarized below:

- The suspension friction causes suspension lock up and stick-slip behaviour, while it contributes to damping (Rakheja, Afework & Sankar, 1994; Rakheja et al., 2003). It is shown that high friction is detrimental to the suspension performance (Gunston, 2000; Hostens et al., 2004). It is thus recommended to minimize suspension friction. Current designs of suspension seats, however, exhibit notable friction with coefficient ranging from 0.06 to 0.09 (Boileau & Rakheja, 1995).

- Suspension springs are selected to achieve low natural frequency. Reported studies have emphasized the benefits of low natural frequency designs in realizing improved vibration attenuation (Gad, Metered, Bassuiny & Abdel Ghany, 2015; Lee & Goverdovski, 2002). Such a design, however, causes great suspension travel under higher magnitude base vibration. This may cause potential impacts with bump-stops leading to high magnitude vibration and shock motions.

- Light suspension damping is preferred for superior vibration isolation performance when the suspension motion is limited to its free travel. This can be achieved only at the expense of great suspension travel and potential bump-stop impacts under higher base vibration. High suspension damping, on the other hand, is desirable for limiting the suspension travel (Hostens et al., 2004; Ma et al., 2008a; Rakheja et al., 2004).

- The kinematics of the suspension spring, damper and linkages cause considerable variations in the effective suspension stiffness and damping properties during an oscillation cycle. The contributions due to kinematics of the suspension system, however, have been mostly ignored. In recent studies by Shangguan et al. (2017) and Zhou et al. (2018), it is shown that effective stiffness during a vibration cycle may vary as much as 100%. These studies suggest that a generally applicable suspension design guideline may not be feasible due to considerable differences in the nature of the vibrations of different vehicles.

- The suspension natural frequency and thereby the vibration isolation performance vary with the seated body mass (Blood, Ploger, Yost, et al., 2010; Yu, Dong, Zhang & Chen, 2019; Zhao, Y., Zhao & Gao, 2010). Air springs can offer nearly constant natural frequency of the suspension. Suspension designs offer essential adjustment for body weight, which is generally not independent of the suspension ride height.

- A suspension seat yields best vibration isolation performance when adjusted to the mid-ride position so as permit maximum suspension travel in compression and rebound (Boileau et al., 2004). The effective suspension stiffness, especially for air suspensions, and the permissible suspension travel, however, are affected by the driver selected seated height, which may lead to impacts with the motion limiting stops. The vibration isolation properties of seats deteriorate when the suspension is adjusted to a non mid-ride position.

The above suggest that the design of a suspension system requires difficult compromises considering the uncertainties in the nature of vehicle vibration, body mass and individual sitting height preference. A general design guideline has thus not yet evolved. Boileau and Rakheja (1990) suggested a suspension seat natural frequency of 1.8 Hz or less for forestry skidders, while Hostens et al. (2004) reported that a suspension natural frequency of 1.5 Hz serves as a good trade-off between vibration isolation and suspension travel. Shangguan et al. (2017) recommended a suspension natural frequency of 1.3 Hz for earth-moving vehicles. A number of modern suspension seats exhibit a natural frequency near 1 Hz or lower, which results in excessive suspension travel. Such seats can also lead to a loss of contact between the operator and the seat during rebound motions (A. Kordestani, personal communication, 2017).

4.2 Experimental assessments of suspension seats

The performance of vehicle seats has been evaluated via subjective and objective methods. The comfort, postural supports and other ergonomic features are generally assessed via subjective methods, which invariably show large variations considering differences in anthropometric dimensions and individual preferences (Griffin, 1990; Mehta & Tewari, 2000). The experimental evaluations of suspension seats, however, are generally based on objective measures of vibration transmission, which is the primary design focus. The laboratory or field evaluations are generally conducted to identify a good seat for specific vehicle applications and to investigate relative properties of different seats. Such evaluations can not only provide a reliable assessment of seats when representative samples of subjects and test conditions are employed but also facilitate tuning of the suspension for specific vehicle applications. The reported laboratory and field evaluations have involved widely different designs of vertical suspension seats, objective measures, excitations and seat loads (rigid and human subjects of varying mass). The reported performance measures thus cannot be compared.

The performance of a suspension seat is strongly dependent upon static and dynamic properties of the suspension system, dynamic interactions of the seated human occupant, nature of excitation (frequency and magnitude), seated body mass and ride height. The laboratory/field evaluations of suspension seats thus need to consider the following:

<u>Seat load</u>: The dynamic responses of the body contribute considerably to the vibration performance of the seat in a highly complex manner (Lewis, C. H. & Griffin, 2002; Politis, Rakheja, Juras, Boileau & Boutin, 2003). A seat loaded with an equivalent rigid mass (body mass supported by the seat) may show higher transmissibility at resonance and yield a considerably higher natural frequency of the seat. The biodynamic response of a seated human body, however, resembles that of a rigid mass at excitation frequencies below 2 Hz (Griffin, 1990), where most suspension seats have their resonant frequency. Low natural frequency suspension seats for vehicles with dominant vibration at low frequencies such as urban buses and earth-moving machines have shown 10% lower vibration transmissibility when coupled with human subjects. Seats employed in vehicles with relatively higher frequency components such as forklift trucks excite the seated body vibration modes and exhibit significant effect on the human body dynamics (Politis et al., 2003). Wu and Griffin (1996) reported that the resonance frequency of a suspension seat loaded with a human subject and a bag of sand are similar when the input acceleration magnitude is high. The resonant transmissibility magnitude with the sand bag, however, is much higher than that with the human subject.

<u>Excitation</u>: The vibration transmission performance of a suspension seat is strongly dependent upon the nature of excitation. This is due to the nonlinear dependence of static and dynamic properties of the suspension components on the magnitude and frequency contents of the excitation (Boileau et al., 2004). The evaluations should therefore be performed under representative vehicle excitation.

<u>Body mass</u>: Variations in body mass can directly affect suspension natural frequency and thereby the vibration isolation performance of the suspension seat (Blüthner, Hinz, Menzel, Schust & Seidel, 2006; Boileau & Rakheja, 1997; Maciejewski, 2012b; Shangguan et al., 2017; Zhao, Y., Zhao, et al., 2010). The vibration performance of a suspension seat should thus be evaluated for different body masses, ranging from 5^{th} percentile female to 95^{th} percentile male population.

Suspension ride height: A suspension seat yields best vibration performance when adjusted to mid-ride. Adjusting the seat to a non mid-ride position can reduce effective suspension travel in compression or rebound, which may cause vibration amplifications due to impacts with motion limiters under high magnitude excitations (Ma et al., 2008a; Marcotte, Beaugrand, Boutin & Larue, 2010; Rakheja et al., 2003). The vibration performance of the seat without an automatic height adjustment mechanism should thus be evaluated under non mid-ride settings, especially under high magnitude excitations.

4.2.1 Measures of vibration performance

Reported studies have employed widely different measures for assessing vibration isolation performance of suspension seats. Field studies generally employ measures related to the WBV exposure, namely, frequency-weighted acceleration and VDV (vibration dose value), given by:

$$\bar{a}_s = \sqrt{\frac{1}{T}\int_0^T a_w^2 dt} \qquad VDV_s = \sqrt[4]{\int_0^T a_w^4 dt} \tag{4.1}$$

where $\bar{a}_s$ and VDV_s are the RMS value and VDV, respectively, of the frequency-weighted acceleration measured at the seat-occupant interface, and T is the integration period. The frequency-weighted acceleration a_w due to vibration is obtained using the weightings defined in the ISO 2631-1:1997 standard (1997), which accounts for human sensitivity to whole-body vibration at frequencies up to 80 Hz. The VDV measure is employed under high magnitude excitations with occasional shocks or when impacts with the elastic bump-stops are anticipated.

The vibration transmission characteristics of suspension seats are generally evaluated in terms of transmissibility ratio in the frequency range of interest. Under swept harmonic or broadband random vibration, the transmissibility ratio, $T(\omega)$, is computed from:

$$T(\omega) = \frac{S_{bs}(\omega)}{S_b(\omega)} \tag{4.2}$$

where $S_{bs}(\omega)$ is the cross-spectral density of the base acceleration and the acceleration at the seat corresponding to excitation frequency ω, and $S_b(\omega)$ is the auto spectral density of the base acceleration. The above measure is widely used to identify suspension natural frequency and resonant response.

The suspension performance under the base vibration, representative of a vehicle, is evaluated in terms of SEAT (Seat Effective Amplitude Transmissibility), a measure of the ratio of the frequency-weighted RMS acceleration of the seat, $\bar{a}_s$, to that at the base, $\bar{a}_b$, such that (Griffin, 1990):

$$SEAT = \frac{\bar{a}_s}{\bar{a}_b} \tag{4.3}$$

The above measure provides the overall vibration isolation effectiveness of a suspension seat. A lower SEAT value reflects good vibration attenuation performance of the suspension, while a value in excess of 1.0 implies an amplification of the base vibration by the suspension. The vibration performance of a seat under excitation containing repeated or occasional shocks has been evaluated in terms of VDV ratio, defined as the ratio of VDV due to acceleration at the seat to VDV due to acceleration at the base, VDV_b (Blood, Ploger & Johnson, 2010; Hostens et al., 2004):

$$VDV_{ratio} = \frac{VDV_s}{VDV_b} \tag{4.4}$$

The measures defined in (4.1) to (4.4) have also been used to evaluate the effects of variations in the body mass, seat height and excitation. These measures have also been employed for suspension tuning and design evaluations via numerical simulations (e.g. Maciejewski, Kiczkowiak & Krzyżyński, 2011; Nagarkar, Patil & Patil, 2016; Stein, Múčka, Gunston & Badura, 2008; Stein, Múčka & Gunston, 2009; Wang, C., Zhang, Guo, Lv & Yang, 2016). The laboratory and field evaluations do not consider the relative motion of the suspension or the seated driver with respect to the base and the controls, which is an important measure for the low frequency suspension seats. Some of the studies reporting suspension design optimization, however, consider the relative suspension travel (e.g. Alfadhli, Darling & Hillis, 2018; Gad et al., 2015; Maciejewski & Krzyżyński, 2011; Metered & Šika, 2014; Zhao, Y., Ou, Zhang & Gao, 2009; Zhao, Y., Zhao, et al., 2010).

4.2.2 Standardized laboratory evaluation methods and excitations

The international standard ISO 7096:2000 (2000) outlines a laboratory evaluation method for evaluating the effectiveness and acceptance of a seat in reducing vertical WBV transmitted to the operators of earth-moving machines within the 1 to 20 Hz frequency range. The standard also specifies vibration spectra of 9 classes of earth-moving vehicles on the basis of representative measured data under typical working conditions, denoted as EM1 to EM9, as shown in Figure 4.3. These are applicable to dumpers, wheel loaders, scrapers, graders, wheel dozers, soil compactors and skid-steer loaders. The spectral classes suggest dominant vibration near 2 Hz for large size dumpers, loaders and graders, and in the 3 to 4 Hz range for compact vehicles (≤ 4500 kg). Peak acceleration power spectral density varies from a low value of 0.34 to as high as 5.55 $(m/s^2)^2$/Hz for EM7 class of vehicles. The standard uses two different criteria for evaluating seats: (i) overall vibration transmission performance in terms of SEAT under a specified spectral class; and (ii) peak vibration transmissibility or the damping test in accordance with the ISO 10326-1:2016 standard (2016).

The peak transmissibility and damping of the suspension are measured under harmonic excitations with a seat carrying a 75 kg load. The peak-to-peak displacement of the harmonic excitation is chosen so as to achieve suspension motion equal to 75% of its free travel. The frequency of harmonic vibration is varied from 0.5 to 2 times the expected resonance frequency of the suspension. The evaluations of the SEAT values require measurements of the seat with human subjects with two specific body masses (52 to 55 kg and 98 to 103 kg). The standard requires that peak acceleration transmissibility must not exceed 1.5 under EM1, EM2, EM3, EM4 and EM6 classes of vibration, and must be below 2 for the other classes. The standard also defines the acceptance criterion for the suspension seats evaluated under the nine classes of vehicles (Table 4.1). These suggest that amplification or minimal attenuation of vibration by the seat is acceptable for classes EM1 to EM4, which exhibit dominant vibration in the vicinity of 2 Hz.

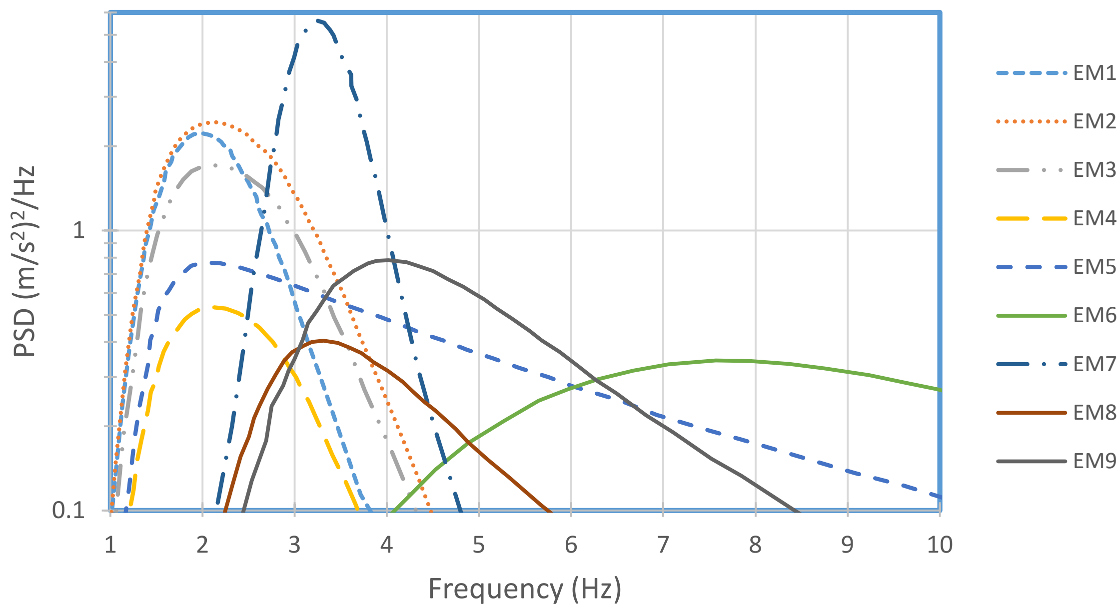


Figure 4.3. Spectra of acceleration at the seat base of earth-moving machines.
Adapted from data provided by the standard "Earth-moving machinery: Laboratory method for evaluating vehicle seat vibration: Part 1: Basic requirements", by the International Organization for Standardization, 2000. ISO standard 7096:2000. ©ISO, 2000.

Table 4.1. Acceptable values of SEAT for different spectral classes (ISO 7096:2000)

Vehicle spectral class	Acceptable value of SEAT
EM1	< 1.1
EM2	< 0.9
EM3	< 1.0
EM4	< 1.1
EM5	< 0.7
EM6	< 0.7
EM7	< 0.6
EM8	< 0.8
EM9	< 0.9

While the standardized test method has been widely used by the suspension seat manufacturers and in many reported studies, the standard has been subject to many critiques, which are summarized below:

- The recommended subject masses of 52 to 55 kg and 98 to 103 kg correspond to the 1st and 99th percentile of the user populations, respectively. Considering the strong dependence of the suspension performance on the body mass, the seat performance needs to be evaluated under more representative body masses (Hinz, Menzel, Blüthner & Seidel, 1998).

- The standardized method does not account for the notable inter-subject variability observed in human biodynamic responses. The variability in the SEAT factor thus needs to be considered through measurements with several subjects of comparable body mass (Hinz et al., 1998).

- The test method is limited only to neutral sitting posture, while the sitting posture can affect the SEAT factor considerably (Hinz, Seidel, Menzel & Blüthner, 2002).

- The standardized test method is limited to the seat adjusted to the mid-ride position. The contributions of potential impacts with the elastic stops are thus not considered. Burdorf and Swuste (1993) showed that the field-measured vibration transmissibility magnitudes are generally higher than those obtained in the laboratory using the standardized test method. This may be either due to non mid-ride positioning of the seat in the field or the presence of occasional shocks that may cause end-stop impacts.

- A suspension seat designed with low natural frequency can easily satisfy the SEAT criterion defined in the standard. Such a suspension, however, will yield large suspension travel, which is generally perceived as uncomfortable and annoying by the seated occupant. Recent laboratory evaluations of a modern air suspension seat, conducted by the researchers, revealed natural frequency in the order of 0.7 Hz and superior SEAT factor but excessive relative travel. The sensation of discomfort by the users of such seats was also confirmed (A. Kordestani, personal communication, 2017). The seated body vibration and suspension travel constitute two opposite design targets for a suspension. The standard thus needs to consider suspension travel as an additional performance measure.

- The standardized test method is limited only to vertical suspension systems. Blüthner et al. (2006) and Blüthner, Seidel and Hinz (2008) proposed a methodology for the testing of seats with a horizontal suspension.

- The ISO 10326-1:2016 standard (2016) also defines a similar test method but it acknowledges the significance of the ride height setting of the suspension. An informative test method is described to assess the ability of a seat to control the effects of impacts caused by suspension over-travel in industrial trucks, earth-moving machines, agricultural tractors and forestry forwarders. The test method requires measurements under a harmonic excitation of transient nature with frequency being the dominant frequency of the vehicle vibration class, as defined in the ISO 7096:2000 standard (2000). The seat is loaded with a 75 kg rigid load and the suspension performance is evaluated in terms of the VDV ratio.

- The ISO 5007:2003 standard (2003) also defines a test method for the evaluation of the vibration transmission of suspension seats for wheeled agricultural tractors. The standard defines spectral classes of vibration for tractors with unballasted mass of ≤ 3600 kg, 3600 to 6500 kg and > 6500 kg (denoted by classes 1 to 3 respectively), as shown in Figure 4.4. These suggest dominant vibration in the 2-2.5 Hz range for the heaviest tractors, and near 3.2 Hz for lighter vehicles. The peak vibration levels for all the classes, however, are substantially greater than those observed for the earth-moving machines, as defined in the ISO 7096:2000 standard (2000). The test method also requires measurements with two subjects (body mass 52 to 55 kg and 98 to 103 kg), similar to those defined in the ISO 7096:2000 standard (2000), while the suspension performance is assessed in terms of the SEAT factor.

The methods described in the above standards are quite similar, while they differ only in the excitation considered. The vibration spectra, however, have been mostly defined for vehicles without primary suspension. Since the vehicle manufacturers are increasingly implementing wheel as well as cabin suspensions in many off-road vehicles, the vibration spectra of modern vehicles may differ not only in the vibration magnitude but also the dominant frequencies. A few studies have also presented vibration spectra of different on-road as well as off-road vehicles, which show important differences among them. As an example, Figure 4.5 illustrates vertical vibration spectra defined for urban buses, forklift trucks and snowplows (Boileau & Rakheja, 2000). The figures show the mean and maximum spectra of vertical acceleration measured at the seat base of the vehicles. These show wide variations in the vibration magnitude and the dominant frequencies. Urban buses with primary air suspensions show dominant vibration near 1.5 Hz, while the unsuspended forklift trucks exhibit dominant vibration in the 2.75 to 5.1 Hz frequency range. The dominant vibration of the snowplows occurs near 1.9 Hz and 4.4 Hz.

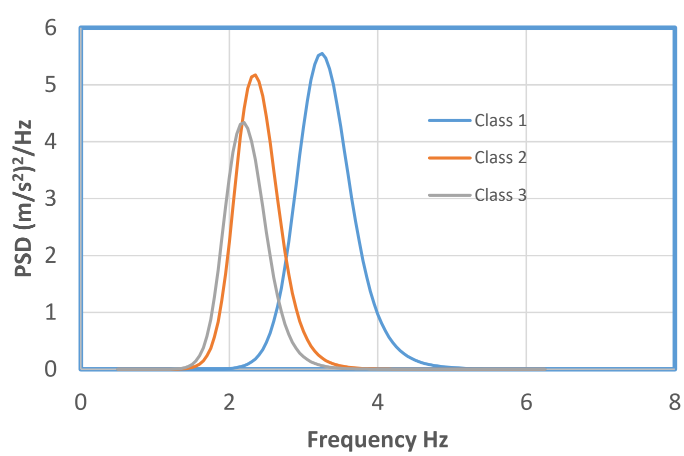


Figure 4.4. Vibration spectra of three classes of agricultural tractors, as defined in the ISO 5007:2003 standard.

Adapted from *Agricultural wheeled tractors: Operator's seat: Laboratory measurement of transmitted vibration*, by the International Organization for Standardization, 2000. ISO standard 5007:2003. ©ISO, 2003.

The assessments of seats with human subjects pose a major ethical dilemma that concerns safety risks associated with the shock and vibration exposure of human subjects. The laboratory methods thus require extremely safe or man-rated vibration simulators. Alternatively, considerable progress has been made in developing anthropodynamic manikins simulating the seated body biodynamic behaviour (Nélisse, Boileau, Rakheja, Patra & Boutin, 2006). The validity of such manikins, however, has been demonstrated for a limited number of seats and vibration excitations (Cullmann & Wölfel, 2001; Gu, 1999; Lewis, C. H. & Griffin, 2002). The variations in subject weight and built further yield considerable variations in the measured data. Studies on vibration biodynamic responses of the seated body have contributed to developments in human body models that may be implemented to suspension models to evaluate the response characteristics of the coupled seat-occupant system, and desirable design and tuning through computer simulations (Rakheja, Dewangan, Dong & Marcotte, 2020). The validity of the coupled seat-body models, however, has not been demonstrated.

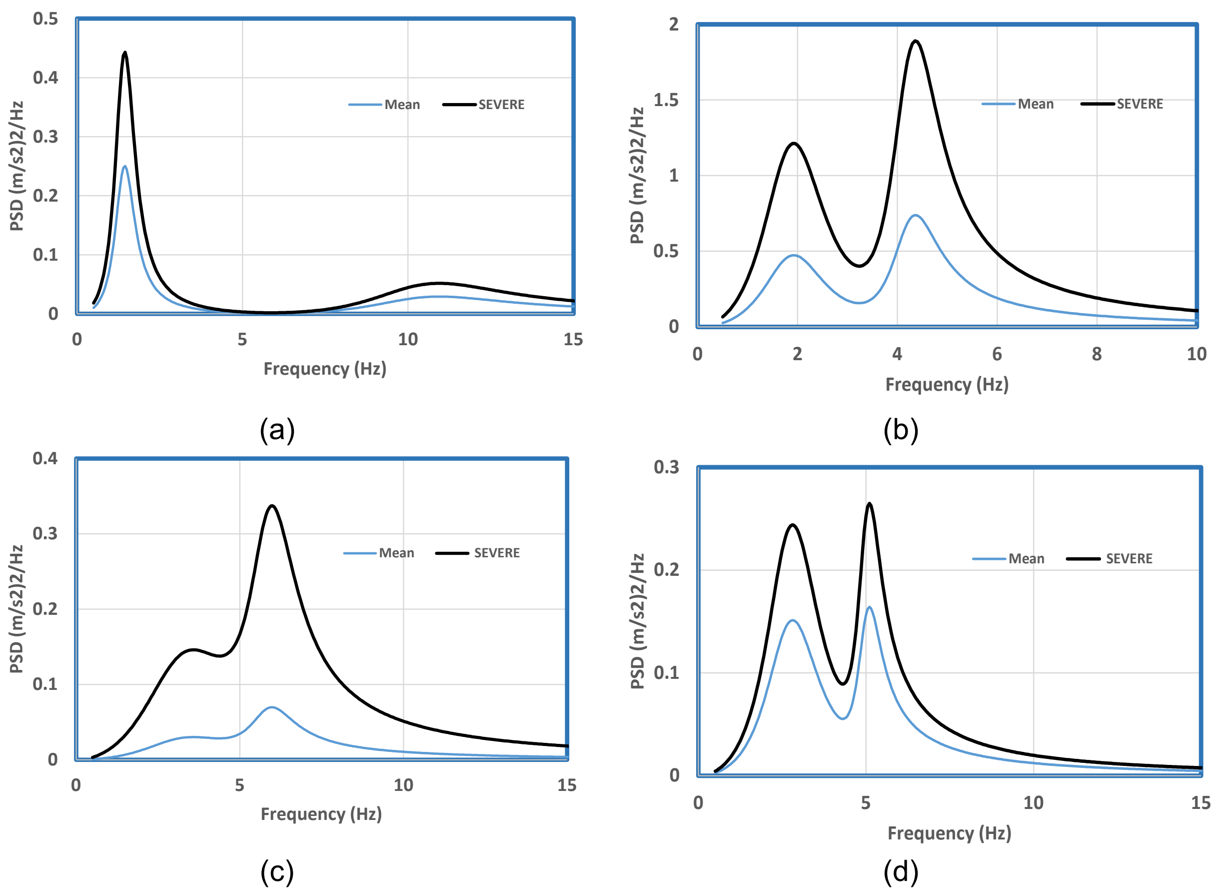


Figure 4.5. Mean and upper limits of spectra of vertical vibration measured at the seat base of (a) city buses; (b) sidewalk snowplows; (c) forklift truck type I; and (d) forklift truck type II.

From *Caractérisation de l’environnement vibratoire dans différentes catégories de véchicules : industriels, utilitaires et de transport urbain* (Report No. R-242), by P.-É. Boileau and S. Rakheja, 2000. ©IRSST, 2000. Reprinted with permission.

4.2.3 Field and laboratory evaluations

Field measurements of suspension seats have been performed for assessing the exposure and the effects of operating conditions, and identifying the appropriate seat for specific vehicles. The assessments are generally performed using the measures defined in section 4.2.1. Blood, Ploger and Johnson (2010) evaluated the relative vibration performance of a mechanical and an air suspension seat applied to a forklift truck. The field-measured vibration data were analyzed in terms of frequency-weighted RMS acceleration, VDV, crest factor and static compression dose of the spine in accordance with the ISO 2631-5:2018 standard (2018). Authors also evaluated the relative performance of two different air suspension seats in a city bus, which showed SEAT values around 0.9, while the VDV ratio was lower (Blood, Ploger, Yost, et al., 2010). The WBV exposure slightly exceeded the action limit defined in the *2002/44/EC Directive* . The use of a silicon seat pad instead of the conventional foam cushion resulted in lower SEAT value of 0.84. The WBV exposure was found to be lower with the air suspension than with the mechanical suspension. This finding was also supported by field evaluations of an air suspension seat for a forklift truck as reported by Motmans (2012), which showed that the seat could provide a 39% reduction in the vertical WBV exposure.

Jonsson, Rynell, Hagberg and Johnson (2015) conducted field evaluations of seats for a city bus and concluded that the air suspension seat did not provide any benefits compared to a pedestal foam seat (unsuspended). This is due to the fact that the vertical vibration of a city bus with a wheel suspension dominates at a very low frequency (≈1.5 Hz), which cannot be effectively attenuated by a seat suspension. This is also supported by an earlier study by Boileau and Rakheja (1997). Lewis, C. A. and Johnson (2012) performed similar measurements on WBV exposure of bus drivers and concluded that the bus floor vibration was amplified by the suspension seat. Another study reporting on the evaluations of a seat for low-floor, high-floor and articulated city buses, showed only marginal performance gains of an air suspension seat, with SEAT values ranging from 0.76 to 0.92 (Thamsuwan, Blood, Ching, Boyle & Johnson, 2013). Jin, Zhang, Wang, Yang and Zhang (2014) performed field measurements on an air suspension seat with an external air reservoir installed on a highway truck. The data obtained at different speeds were used to develop and verify a multi-body dynamic model of the suspension seat, using the ADAMS software.

The vibration isolation performance of an active suspension seat comprising a controlled linear electromagnetic force actuator was evaluated on a highway tractor-semitrailer (Blood et al., 2011). The vibration exposure with the active suspension seat was found to be 19-55% lower than that with a passive air suspension. Measurements revealed crest factors in excess of 9 suggesting the presence of repeated or occasional shocks. The laboratory evaluations of the same seats under simulated truck and bus excitations also showed superior performance of the active seat in view of the WBV exposure and acceleration transmissibility (Blood, Yost, Camp & Ching, 2015). These studies have also shown important differences between the seats, and strong effects of excitation magnitude (road roughness), vehicle speed, body mass and seat height setting. Moreover, vibration exposure measured with a mechanical suspension seat showed a greater dependency on the driver weight.

Boileau and Rakheja (1990) performed laboratory and field evaluations of four different suspension seats in order to identify an appropriate suspension seat for a small size log skidder. These included a compact behind-the-seat suspension, an air suspension, and two mechanical suspension seats. A good correlation was observed between the field and laboratory measured vibration transmissibility of the seats. The SEAT values of the behind-the-seat and air suspension seats ranged from 0.76 to 0.79 and from 0.62 to 0.80, respectively, depending on the driver mass and the terrain condition, while the mechanical suspension seats resulted in very little attenuation or even an amplification of the cabin vibration. The vibration transmissibility of a mechanical suspension seat was further evaluated in the laboratory under harmonic excitation and class 2 agricultural tractor vibrations as defined in the ISO 5007:2003 standard (2003), for the purpose of verification of a simulation model (Boileau, Rakheja & Liu, 1997).

A number of studies have reported on laboratory evaluations of commercially available seats and different prototype designs under harmonic or transient excitation or following the procedure defined in the ISO 7096:2000 standard (2000). The primary objectives of these studies included the assessments of passive, active and semi-active suspension design concepts, tuning of the control algorithm, and model verification. The vibration transmissibility characteristics of mechanical and air suspension seats were evaluated in the laboratory under harmonic excitations and representative vehicle vibrations for the purpose of the verification of simulation models (Boileau et al., 1997; Bouazara, Richard & Rakheja, 2006; Ma et al., 2008a; Ning, Sun, Li, Du & Li, 2016; Ning, Sun, Zhang, et al., 2016; Prasad, Tewari & Yadav, 1995). Smith (1997) measured the vibration transmissibility of a seat-occupant system in order to verify different occupant models ranging from 1 to 5-degrees-of-freedom (DOF). Gunston, Rebelle and Griffin (2004) measured the acceleration and VDV responses of a suspension seat under transient excitations predominant at 2.1, 2.25 and 3.25 Hz, in order to verify the seat suspension models. Holtz and van Niekerk (2010), Maciejewski, Meyer and Krzyżyński (2009) and Maciejewski et al. (2011) measured the acceleration transmissibility and the relative displacement of an air suspension seat with an external air reservoir under harmonic excitation as well as representative vehicle vibrations (bus, truck, earth-moving and agricultural vehicles) in order to illustrate the performance benefits of the external reservoir. The laboratory assessments of different design concepts in passive, semi-active, active electrohydraulic and active electropneumatic suspension seats under excitations representative of locomotives, agricultural (ISO 5007:2003 standard), earth-moving (EM3) and mining machines have been reported in a number of studies (Duke & Fow, 2012; Duke & Goss, 2007; Le & Ahn, 2011; Stein, 2014; Stein & Ballo, 1991; Stein, Múčka, Gunston & Badura, 2008; Stein, Múčka & Gunston, 2009; Valero, Amirouche, Mayton & Jobes, 2007). Similarly, Choi, S.-B., Choi, Nam, Cheong and Lee (1998) and Choi, S.-B., Nam and Lee (2000) performed laboratory measurements on semi-active suspension seats with controllable electrorheological (ER) and magnetorheological (MR) fluid dampers under harmonic excitation. The controller synthesis was realized via a hardware-in-the-loop test approach. McManus et al. (2002) performed laboratory measurements on a semi-active seat with MR damper under different magnitudes of harmonic and transient vibration in order to demonstrate the performance gains of the MR suspension. It was shown that the bump-stop impacts can be entirely eliminated via the controlled MR damper.

The majority of the above-stated studies have employed rigid loads strapped to the seat instead of human subjects for conducting the tests. The contribution due to human dynamics was thus not considered. Moreover, strapping of a rigid load to the seat cushion can alter its static and dynamic properties. In addition, the experiments were limited to controlled conditions, which did not permit studying the factors most likely to have affected the suspension performance, such as variations in seated body mass and suspension ride height. Developments in passive suspensions

The introduction of air springs was the most notable advancement, which permitted height adjustment with greater ease than with mechanical springs. The air springs also facilitate the implementation of the automatic ride height adjustment and the adjustment of the suspension for body mass variations. Higher body mass requires relatively higher air pressure in the spring leading to a higher suspension stiffness. A lower body mass, on the other hand, requires relatively lower pressure in the spring resulting in lower spring stiffness. Air springs could thus minimize changes in the suspension natural frequency with varying body mass. Moreover, a lower stiffness of the air spring could be easily realized by increasing its size, since the effective stiffness of an air spring is inversely related to the air volume (Hostens et al., 2004; Shangguan et al., 2017):

$$k_{eff} = \frac{p\gamma A_e^2}{v} \tag{4.5}$$

where k_{eff} is the effective spring rate, γ is the polytropic constant of air, A_e is the effective area, and p and v are instantaneous air pressure and volume in the air spring, respectively.

Hostens et al. (2004) reviewed some of the suspension design factors affecting vibration transmissibility and proposed an air spring with an auxiliary chamber to achieve lower natural frequency of the suspension. An additional air chamber permitted the use of small size air springs and compact suspension design. Although the integration of an additional air volume imposed some challenges considering the limited space available within the cross-linkage platform, innovative packaging methods have been used in modern suspension seats. Figure 4.6 illustrates two commercially available suspension seats integrating an additional air reservoir. The natural frequency of the suspension seat is directly related to the additional air volume. Too large volume could result in a very low natural frequency. Preliminary laboratory assessments conducted by the team have shown that such a design can yield a natural frequency as low as 0.7 Hz, as seen in Figure 4.7, which illustrates the acceleration transmissibility of the seat loaded with an inert mass of 52.6 kg and subjected to three different levels of white noise vibration in the 1 to 20 Hz frequency range (RMS acceleration: 0.25, 0.5 and 1.0 m/s^2). It can be seen that the frequency corresponding to peak response (resonance frequency) reduces to as low as 0.7 Hz, when overcoming the suspension friction under high magnitude excitations. Such a low frequency suspension design could cause excessive suspension travel and possible impacts with the bump-stops.

Figure 4.6. Pictorial views of suspension mechanisms with an additional air reservoir.

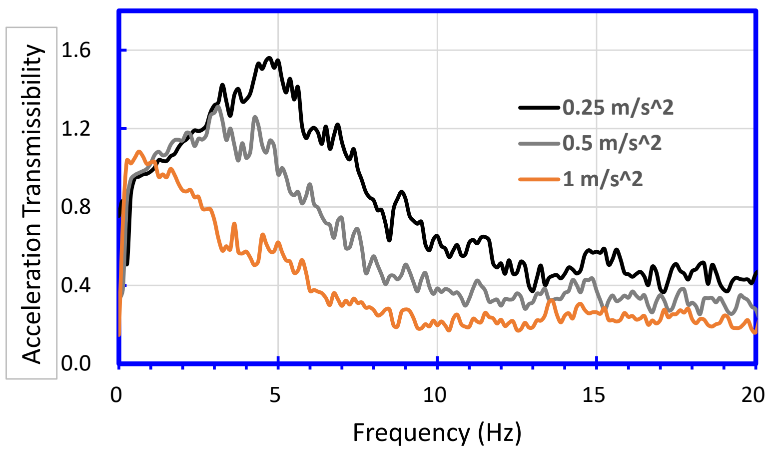


Figure 4.7. Acceleration transmissibility of an air suspension seat with an additional air reservoir measured under different levels of white noise vibration in the 1 to 20 Hz frequency range (seat load = 52.6 kg).

The use of a throttle valve between the air spring and the additional reservoir has also been proposed to achieve damping effect. The damping effect, however, could not be realized under excitations at relatively higher frequencies due to partial or total choking of the airflow through the valve. Holtz and van Niekerk (2010) have investigated the performance of an air suspension seat with similar external reservoir analytically and experimentally under different classes of earth-moving machinery excitations, as defined in the ISO 7096:2000 standard (2000). The seat was modelled as a single-DOF system with nonlinear stiffness and damping due to the air spring. The simulation results showed a 27% reduction in vibration transmissibility and a 21% reduction in the natural frequency with the added reservoir. A prototype seat was developed with the target of achieving a SEAT (seat effective amplitude transmissibility) of 1.1. The experimental results obtained with the prototype showed good correlations with the simulation results. A SEAT value of 1.1, however, implies the amplification of the base vibration. Moreover, the resulting suspension travel was not considered as part of the study. Considering the strong dependence of the suspension performance on the magnitude and the frequency of the target vehicle vibration, Maciejewski et al. (2011) identified optimal parameters of the auxiliary chamber and the flow opening under excitations due to different classes of earth-moving machines ranging from low frequency and high magnitude to high frequency and low magnitude base vibration.

Lee, C.-M. and Goverdovskiy (2002) proposed the concept of a suspension seat with elastic links, which permitted variable negative stiffness. The measurements performed on such a seat showed improved performance under high magnitude vibration excitations. The study proposed the use of non-uniform thin walled structures as elastic elements, which may incur fatigue failures in an off-road vehicle environment. Similarly, Le and Ahn (2011) proposed a negative stiffness structure for the attenuation of low frequency vertical vibration. A vertical suspension design was conceived, which comprised two symmetric stiffness structures and a primary vertical spring with a damper. Each structure comprised a horizontally constrained spring coupled to the suspension mass via a sliding block. Simulation and laboratory measurements showed that the proposed design could achieve a very low natural frequency and an improved attenuation of vertical vibration.

4.2.4 Suspension design optimization

The design factors affecting the vibration attenuation performance of suspension seats have been widely studied using linear and nonlinear simulation models together with parameter optimization. Appendix A summarizes the essential features and objectives of the reported simulation models. The suspension seat system is, generally, modelled by a two-DOF dynamic system with either linear or nonlinear properties, and with rigid body representation of the seated body, as shown in Figure 4.8. In this model, m_0 is the seated body mass supported by the viscoelastic cushion with constant stiffness K_c and damping coefficient C_c. m_s is the suspension or seat pan mass, K_s is the equivalent vertical suspension stiffness, F_D is the damping force developed by the damper oriented with an inclination within the suspension linkages, and F_F represents the lumped friction due to the suspension links and guiding rollers. Some models also consider the end-stop impacts by representing the end-stop buffers by either linear or nonlinear clearance springs (K_{ST}) with total clearance (suspension travel) of 2β. Different lumped-parameter occupant models have also been used in an attempt to account for the contributions of the human biodynamics to the suspension seat dynamics. The occupant models range from linear single-DOF (Abbas, Emam, Badran, Shebl & Abouelatta, 2013; Książek & Ziemiański, 2012; Rakheja et al., 1994; Smith, 1997; Tewari & Prasad, 1999) to many-DOF (Boileau et al., 1997; Choi, Y. T. & Wereley, 2005; Dong & Lu, 2012; Gohari, Rahman, Raja & Tahmasebi, 2012; Gohari & Tahmasebi, 2014; Hill & Dhingra, 2003; Nagarkar, Patil & Patil, 2016; Smith, 1997; Valero et al., 2007; Wan & Schimmels, 1997; Yan, Zhu, Li & Wang, 2015). The models exhibit many shortcomings, which are summarized below:

- The validity of the coupled occupant-seat models has not been demonstrated. This may be partly due to the lack of consideration of the effect of the body coupling with the viscoelastic seat cushion. The occupant model integrated with the suspension model is generally derived from the apparent mass response of the body seated on a rigid seat pan. It has been shown that sitting on a viscoelastic cushion affects the biodynamic responses of the body (Dewangan, Rakheja, Marcotte, Shahmir & Patra, 2013; Hinz et al., 2006; Rakheja, Dewangan, Marcotte, Shahmir & Patra, 2015).

- The seated body biodynamic responses to vibration are strongly influenced by the body mass, while the models are formulated for a specific body mass.

- The majority of the models consider linear component properties, although laboratory characterizations of components have shown highly nonlinear properties of the cushion and suspension components (Boileau et al., 2004; Wu & Griffin, 1998). Mechanical suspension springs show progressive hardening stiffness, while air suspension yields nearly linear stiffness for a given seated mass. A higher seated mass yields higher effective spring rate of the air spring and higher magnitude of friction.

- The hydraulic dampers used in suspension seats exhibit asymmetric damping in compression and rebound, and relatively lower damping coefficient at higher velocities. The vast majority of the models consider either linear or symmetric damping properties.

- The effective stiffness and damping properties are further affected by the suspension kinematics and by the ride height adjustment (Shangguan et al., 2017; Zhou et al., 2018), which are generally ignored in the reported models. Figure 4.9 illustrates the kineto-dynamic model structure of an air suspension seat and variations in the effective vertical suspension stiffness during an oscillation cycle. The figure also shows the effect of a suspension height adjustment, with variations of ±10 and ±20 mm from the mid-ride position.

Simulation models have been used to study the effects of component properties on vibration transmission performance and to identify optimal design parameters. These generally focus on realizing low natural frequency suspension to achieve low vibration transmissibility in the vicinity of the dominant vehicle vibration frequencies. A low natural frequency suspension, however, yields higher suspension travel under high magnitude vibration and shock motions. Maciejewski et al. (2009) stated that the design challenge is due to the existence of two opposite requirements: minimization of the vibration transmitted to the occupant and minimization of the relative travel of the suspension. The best compromise between these two opposite criteria creates a complex and nonlinear design optimization problem. A number of studies have attempted to optimize suspension components' properties to achieve improved vibration isolation performance, while limiting the suspension travel. These have employed widely different nonlinear optimization algorithms such as genetic algorithms (Abbas et al., 2013; Afkar, Javanshir, Taghi Ahmadian & Ahmadi, 2013; Gad et al., 2015; Gohari & Tahmasebi, 2014; Guglielmino, Stammers, Stancioiu, Sireteanu & Ghigliazza, 2005; Nagarkar et al., 2016; Orečný, Segľa, Huňady & Ferková, 2014; Shangguan et al., 2017; Shirahatti, Prasad, Panzade & Kulkarni, 2008; Wan & Schimmels, 1997; Zhao, L., Zhou & Yu, 2016) and nonlinear search algorithms (Bouazara et al., 2006; Shangguan et al., 2017; Wan & Schimmels, 2003). These have also employed widely different objective functions, vibration excitations and suspension seat models to identify optimal component properties. Optimal component properties identified in different studies thus differ considerably. Moreover, relatively fewer efforts have been made to study the robustness of the optimal suspension under conditions involving ride height and body mass variations as well as the occurrence of occasional shocks.

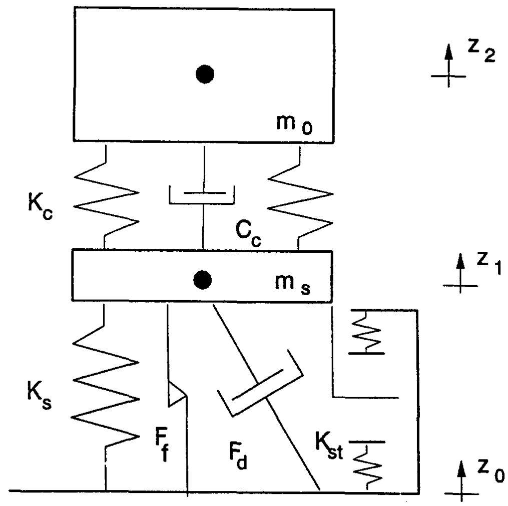
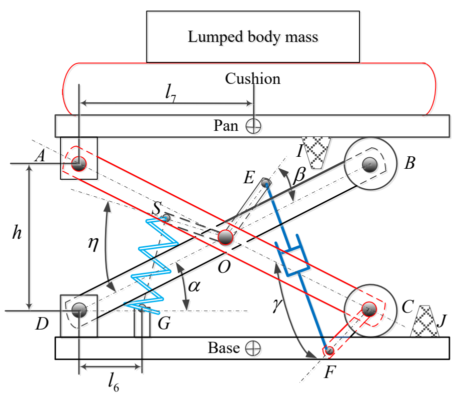
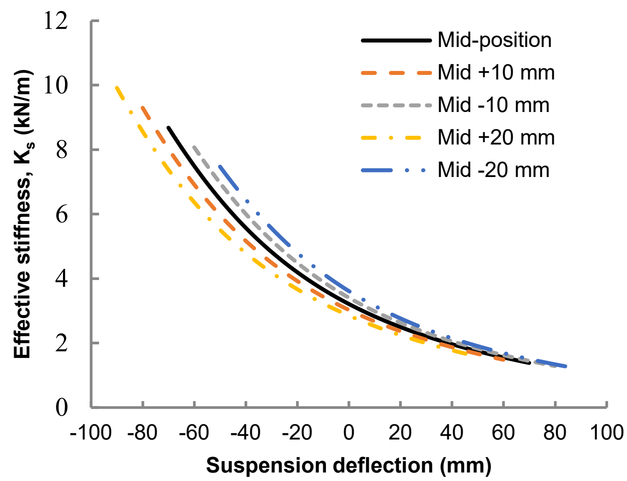


Figure 4.8. Two-degree-of-freedom (DOF) model of the suspension seat with a rigid mass representation of the seated body.

From *Étude des paramètres affectant l'efficacité d'atténuation des vibrations par un siège suspendu* (Report No. R-095)", by P.-É. Boileau and S. Rakheja, 1995. ©IRSST, 1995. Reprinted with permission.



(a)



(b)

Figure 4.9. (a) Kineto-dynamic model of the seat suspension; (b) variations in effective suspension stiffness during an oscillation cycle and effect of the ride height.
From *The kineto-dynamic analysis and optimal design using suspension seat-human coupled model*, by Y. Shui 2016. ©Y. Shui, 2016. Reprinted with permission.

Nonlinear suspension damping properties can help realize a better compromise between vibration attenuation and relative displacement response, although it has been attempted in fewer studies (Ma et al., 2008a; Wan & Schimmels, 2003). Reported studies show conflicting guidance for suspension damper design. The design optimization study by Wan and Schimmels (1997) suggested significantly higher damping compared to that used in the current designs. A higher damping, however, is known to be detrimental to the vibration isolation performance (Hill & Dhingra, 2003; Rakheja et al., 2003). Stein et al. (2009) showed marked differences between the optimal or near-optimal stiffness and damping coefficients established from the laboratory and field evaluations for a locomotive seat. Laboratory evaluations were conducted with a seat loaded with a rigid mass of 57.1 kg and adjusted to the mid-ride, while the field tests were conducted with one operator selecting the ride height. Asymmetric damping in compression and rebound, and higher rebound damping at higher speeds are shown to be beneficial in reducing the severity of end-stop impacts (Rakheja et al., 2004). Gunston (2000) showed negligible effect of non-frictional suspension damping on the suspension performance under low magnitude vibration. Reducing the damping had only marginal effect under moderate magnitudes of vibration (less than 5% improvement) but detrimental effect under high magnitude vibration, which is consistent with the results reported in Ma et al. (2008a) and Rakheja et al. (2003). It was further concluded that the suspension friction should be reduced as much as practically possible, and the non-frictional damping should be used to control the occurrence of end-stop impacts. Dong and Lu (2012) numerically investigated the performance of a suspension seat with three stages of elastic and two stages of damping characteristics. The initial stages with low stiffness and damping were proposed for vibration control, while the latter stages with progressively increasing stiffness and damping provided protection from blast-induced shocks. Despite the differences among the reported studies, it is evident that suspension damper design is quite complex due to its strong coupling with the nature of vibration, seated mass and the suspension stiffness. It can be further concluded that high suspension damping is vital for control of resonant response encountered under shock excitations and bump-stop impacts.

Low natural frequency suspension designs with limited permissible travel present a greater potential for interactions with elastic end-stops. Such interactions transmit high magnitude vibration and shocks to the seated body. These also cause resonant oscillations of the suspension and impose high damping demand to reduce the magnitude of transmitted vibration. Ma et al. (2008a) suggested higher suspension damping for improved attenuation of high intensity vibration and intermittent shock motions, particularly for vehicles with predominantly low frequency vibration such as on-road vehicles operating on relatively rough urban roads. A few studies have investigated designs of elastic motion limiting end-stops for minimizing shock motions of the driver. These suggest that elastic stops with progressively hardening stiffness are desirable for limiting the magnitude of transmitted shocks, which are generally modelled by a cubic function of the deformation. Relatively low stiffness corresponding to low deformation, however, is vital to minimize the peak body acceleration and the resulting VDV values (Boileau et al., 2004; Rakheja et al., 2004). Rebelle (2004) identified optimal stiffness parameters and height of end-stops, and computed the VDV and rate of deformation of the elastic end-stops. The study concluded that the height of the end-stop buffer was not a predominant parameter compared with the stiffness for limiting the VDV. Moreover, the height of the end-stop buffer and the VDV response of the suspension could be reduced by increasing the buffer damping. Shangguan et al. (2017) investigated the suspension seat performance under standardized earth-moving vehicle excitations (EM1, EM4, EM6, EM9) superimposed by filtered shock pulses near the dominant vehicle vibration frequency (2 Hz) as recommended in the ISO 10326-1:2016 standard (2016). The study proposed optimal suspension damping in order to limit the peak suspension relative displacement within the permissible suspension travel. The results of these studies suggest that the effects of end-stop impacts can be minimized by using relatively soft end-stop buffers and by increasing suspension damping. End-stop buffers with enhanced material damping can also reduce the VDV due to end-stop impacts.

Simulation models have been mostly used to analyze the vibration transmissibility characteristics under idealized harmonic as well as broad band random vibration excitations. Some studies have identified desirable suspension parameters for specific vehicular excitations. These include forestry vehicles (Boileau et al., 1997), agricultural tractors (Prasad et al., 1995; Hill & Dhingra, 2003; Hostens et al., 2004), urban buses (Duke & Goss, 2007; Rakheja et al., 2003, 2004; Ma et al., 2008a; Bouazara et al., 2006), earth-moving vehicles (Wan & Schimmels, 2003; Ma et al., 2008a; Maciejewski et al., 2009, 2011), industrial vehicles (Rebelle, 2004), mining vehicles (Valero et al., 2007), and locomotives (Stein et al., 2008a, 2009). These reported widely different optimal suspension design parameters suggesting a strong dependence on vehicle vibration spectra. Shangguan et al. (2017) proposed a methodology for identifying vehicle-specific suspension design parameters, although the model used in the study did not incorporate the elastic end-stops.

4.2.5 Horizontal suspension seats

Although the magnitudes of horizontal vibration along the lateral and longitudinal axes are known to be high, only limited efforts have been made to develop effective horizontal suspensions at the seat. This is mostly due to very low frequencies of horizontal vibration of the vast majority of the off-road vehicles (≈ 1 Hz) (Cation et al., 2008; Eger et al., 2013; Rakheja et al., 2008). This necessitates very low natural frequency design, leading to large relative motion of the seat with respect to the cabin and the vehicle controls, and thereby interference with the operators' tasks. Seat manufacturers offer optional longitudinal vibration isolators that can be added to a vertical suspension seat. Figure 4.10 illustrates a typical commercially available add-on horizontal suspension module. The large fore-aft travel of the isolator is often perceived as being annoying and uncomfortable by the operators, who frequently tend to lock the suspension (A, Kordestani, personal communication, 2017). Such designs are thus realized with very high friction that limit longitudinal seat motion at low excitation frequencies. Some of these suspensions have shown negligible to minimal attenuation of longitudinal vibration (Stein & Múčka, 2011; Stein, Zahoranský, Gunston, Burström & Meyer, 2008).

Figure 4.10. A horizontal suspension that may be installed on a vertical suspension seat.

From *K&M 8246 KM 1000 replacement lateral isolator, price/EA*, 2021. ©OpenTip, 2021. Retrieved from https://www.opentip.com/product.php?products_id=10393622. Reprinted with permission.

Sankar and Afonso (1993) developed a lateral vibration isolator with a mass absorber for a dump truck. The proposed design could be integrated to a vertical seat suspension. The effectiveness of the isolator was evaluated in the laboratory and in the field. The results showed good potential of the isolator for attenuating lateral vibration. The isolator, however, required large absorber mass (≈40% of the occupant and seat mass). Fleury and Mistrot (2006) developed a fore-aft vibration isolator model coupled with a human occupant model, derived from the apparent mass response. The isolator model comprised two pre-constrained steel springs, a hydraulic damper, two elastic bump-stops and a slide system with friction with total suspension travel of 720 mm. Optimal design parameters of the isolator were identified, which showed a fore-aft SEAT factor around 0.8. Stein, Zahoranský*,* et al. (2008b) proposed the design of a high-friction horizontal isolator, which could prevent suspension movement under low intensity vibration. Simulation results and laboratory measurements showed only minimal reduction in the fore-aft vibration. Reducing the friction resulted in an amplification of the fore-aft vibration with 10–17% increase in the SEAT factor (Stein & Múčka, 2011). An innovative design of a seat with omnidirectional vibration isolators was proposed by Ropp (2008). The design incorporates several links to suspend the isolator from the base in addition to the conventional platform with cross-linkage. The design permits the swivel and horizontal movements of the isolator relative to the base. The performance of the suspension design in limiting horizontal vibration is not reported, although it is claimed to reduce the transmission of horizontal forces to the seated occupant.

Kim et al. (2016) evaluated the relative performance of a vertical suspension seat with and without an add-on lateral isolator, developed by Grammer Seating (Germany). The laboratory experiments were performed to evaluate the vibration performance of the seat with and without a lateral isolator in terms of SEAT and acceleration dose value, in accordance with the ISO 2631-5:2018 standard (2018), as well as head acceleration and electromyographic activities of the major lower back and neck muscles. The study concluded that the two-axis suspension helped to reduce the WBV exposure and muscle activities compared to the vertical suspension alone, although the differences were not found to be statistically significant. The lateral suspension was judged to be effective in reducing the transmission of lateral vibration. Another study evaluated the vibration performance of the same vertical suspension seat equipped with either a longitudinal or a lateral vibration isolator under different mining vehicle excitations synthesized in the laboratory (Kim, Dennerlein & Johnson, 2018). The experiments were also performed with a vertical electrodynamic suspension seat, manufactured by Bose Corp. This seat comprised a controlled linear electromagnetic actuator in addition to the air spring. While the electrodynamic active suspension seat provided improved attenuation of the vertical vibration, the horizontal isolators did not provide attenuation but rather resulted in an amplification of the horizontal floor vibration in some cases.

4.2.6 Semi-active and active suspension seats

Owing to the high cost and high-power demand for actively controlled suspensions, considerable developments in semi-active suspensions have been reported. A semi-active suspension is realized by implementing a controllable damper to the conventional suspension to achieve a better compromise between the vibration isolation performance and suspension travel. Semi-active suspension seats with controllable ER (Choi, S.-B. & Han, 2007; Wu & Griffin, 1997)and MR (Choi, S.-B. et al., 2000; McManus et al., 2002) fluid dampers have been explored. ER and MR dampers exhibit variable damping properties due to rapid variations in their rheology under applications of electric and magnetic fields, respectively. Wu and Griffin (1997) experimentally investigated the vibration performance of a suspension seat with an ER fluid damper under excitations representative of a truck and a tractor. The natural frequency of the suspension system was 1.7 Hz, and on-off control scheme was used to modulate damping between the low and high states. The results obtained under the ‘on’ state showed superior performance in view of suspension travel and thereby the end-stop impact performance. Choi, S.-B. et al. (1998)), Choi, S.-B. and Han (2007), Han, Jung, Choi and Wereley (2005) and Han, Jung, Choi, Choi and Wereley (2006) evaluated the vibration performance of an ER fluid damper suspension in the laboratory in terms of SEAT, VDV and vibration transmitted to the seated subject’s head under a commercial truck vibration. The laboratory evaluations showed improved suspension performance. Han et al. (2006) and Choi, S.-B. and Han (2007) showed that the damping force of an ER damper can be increased from 67 N (in the absence of the electric field) to a maximum of 155 N under an electric field of 4 kV/mm. Apart from the electric field, the damping force developed by an ER fluid damper is strongly dependent on excitation frequency (Wu & Griffin, 1997). Among the shortcomings of the ER fluid dampers, are the requirement of a high electric field intensity and the degradation of the damping property with increasing temperature.

Unlike the ER fluid dampers, MR fluid dampers exhibit a rapid variation in damping property under an applied magnetic field. A MR damper with a controller has been commercially developed for the suspension seats by Lord Corp. (denoted as ‘Motion Master’), which requires only 12 V excitation with a maximum current of 1.5 A (McManus et al., 2002). The suspension relative displacement serves as the feedback to the controller. The design also permits ‘medium’ and ‘firm’ settings of the damper, which could be selected by the driver depending on the terrain condition. Suspension seats with this MR damper have been developed by a number of suspension seat manufacturers (Sears Seating, Isringhausen, Knoedler Manufacturers Canada Ltd.). Considerable differences have been reported on the performance characteristics of the MR suspension seats obtained from laboratory measurements and simulations. This is partly due to differences in the excitation conditions and damper controllers considered in the reported studies.

A laboratory evaluation of a MR suspension seat, conducted by the team, under an EM2 excitation showed frequency-weighted SEAT values of 1.21 and 0.94 for the low and high damper settings, respectively. Comparisons of RMS accelerations at the seat and on the floor revealed amplification of the vibration up to 2 Hz and only marginal gains were obtained at frequencies above 2.5 Hz. A comprehensive laboratory evaluation of a MR suspension seat has been reported by McManus et al. (2002) under transient as well as earth-moving vehicle vibration (EM1). The study also investigated suspension performance under an EM1 excitation amplified by 150%, with suspensions adjusted to the mid-ride height, and ±2,54 and ±5,08 cm relative to the mid-ride height. The results of the study showed that the end-stop impacts could be mostly eliminated by the semi-active MR damper suspension with only a small gain in the SEAT performance. Mayton, DuCarme, Jobes and Matty (2006) evaluated the vibration attenuation performance of a MR damper suspension under an agricultural tractor excitation, as defined in the ISO 5007:2003 standard (2003), and compared its performance with a conventional passive suspension seat. In that study, the MR suspension seat showed peak acceleration transmissibility of 0.95, which was substantially lower than the value of 1.3 obtained with the passive suspension. Choi, S.-B. et al. (2000) evaluated the performance of a MR suspension seat in the laboratory based on the skyhook control law under harmonic excitation. The study showed improved vibration isolation performance of the controlled suspension compared to the uncontrolled suspension (fixed damping under zero damper current). The peak acceleration transmissibility of the controlled suspension was about 1.25, while the vibration attenuation was obtained at frequencies above 2.2 Hz. These results suggest that marginal benefits of the MR suspension can be obtained compared to the conventional passive suspension.

The performance characteristics of a suspension seat with a MR fluid damper strongly rely on the damping control logic. Earlier studies used either ‘on-off’ or continuous control based on skyhook control policy, proposed by Karnopp, Crosby and Harwood (1974). Sapiński (2005b) experimentally evaluated the performance of a MR suspension seat with three different ‘on-off’ control schemes, which showed notable improvement in shock isolation but only marginal gain in vibration isolation. Switching discontinuities in an ‘on-off’ control tend to deteriorate the suspension performance. Controller synthesis is challenging due to the highly nonlinear and hysteretic nature of the damping force developed by a MR damper. It has been suggested that hybrid control combining different control algorithms such as the fuzzy logic, sliding-mode control and neural network control methods can improve the control performance and the robustness of the suspension system for variations in seated body mass, ride height and excitations (Phu, Choi, Lee & Han, 2016). Metered, Bonello and Oyadiji (2009) employed a sliding mode controller in conjunction with a neural network controller to achieve the desired damping variations. Ma, Rakheja and Su (2008b) proposed a skyhook based on-off control algorithm together with a modulation function to compensate for switching discontinuities. The hardware-in-the-loop experiments and simulations showed substantial improvements in the shock attenuation under transient excitations and in the SEAT values under urban bus and earth-moving vehicle excitations. A fuzzy controller was further developed, which showed improved vibration isolation performance (Sapiński, 2005a). Yu et al. (2019) proposed the design of a suspension seat with an adaptive rotary MR damper actuated by rotational motions of suspension links. Simulation and laboratory test results for the seat in the open-loop mode showed 21% and 60% reductions in seated mass acceleration response under 0.4 A and 0.8 A current, respectively, applied to the damper. Results obtained for the on/off control showed resonant transmissibility peak of about 1.8 near 2.6 Hz suggesting that such a suspension will not be feasible for most vehicles. Sun, S. et al. (2016) demonstrated an improved performance of the rotary MR damper suspension using a fuzzy control method.

A few studies have also explored semi-active horizontal seat suspensions. Bai, Jiang and Qian (2017) proposed an integrated vertical/longitudinal semi-active seat suspension with a rotary MR damper, similar to that reported by Sun et al. (2016). The simulation results obtained for the torque-controlled suspension showed variations in the fore-aft and vertical mode resonance frequencies in the 2.5 to 3.0 Hz range. The simulation and experimental results obtained under constant current excitations showed lower acceleration response with increasing current under sinusoidal vibration. The natural frequency of the suspension design, however, was considerably high for on-road and off-road vehicle applications. Sun et al. (2015) used the same concept to develop a horizontal seat isolator using actively controlled MR elastomers, which exhibit negatively changing stiffness. A skyhook control algorithm was used to modulate the control current and thereby the stiffness changes. The simulation and experimental results obtained under harmonic excitation at 5, 7 and 10 Hz showed 45%, 44% and 27% lower vibration, respectively, compared to those obtained with the uncontrolled suspension. The suspension natural frequency, however, was quite high for applications in vehicle seats.

Semi-active suspension seats employ controllable damping and are considered effective only during the energy dissipation stage. Despite their several advantages such as fast response time, controllable damping force and low energy consumption compared with the active suspension systems; such systems are only effective in a narrow frequency range (Li et al., 2012). Alternatively, active suspension seats comprise a controlled force generator, which can either add or dissipate energy to achieve superior control of transmitted vibration. In recent years, a large number of active control algorithms have been proposed, which offer superior vibration suppression. These also offer greater robustness against variations in seated mass, ride height and excitation conditions. The high-power demand of the force generator, however, is generally considered prohibitive of their implementations (Ning et al., 2017b).

Reported studies have employed various schemes for realizing active suspension control such as sliding mode control (Lathkar et al., 2016; Ning et al., 2017a; Oshinoya et al., 1998, 1999; Phu et al., 2016), robust H_∞ control (Zhao et al., 2009; Zhao et al., 2010b; Wang et al., 2015; Zhao et al., 2011a, 2011b; Ezzine & Tedesco, 2009; Zhang et al., 2011), optimal control (Cheok et al., 1989), and fuzzy control (Avdagic et al., 2013; Ning et al., 2017b; Guclu, 2005; Phu et al., 2015; Rajendiran & Lakshmi, 2016; Tanovic & Huseinbegovic, 2009). These have employed seat suspension models with the human body represented by a rigid mass or a biodynamic model. Sun et al. (2011) explored the robust H_∞ control for an active seat suspension system using dynamic output feedback control coupled with a human body model. The generalized Kalman–Yakubovich–Popov (KYP) lemma was employed for the H_∞ norm considering a disturbance rejection over a finite frequency band where the human body is more sensitive. Zhao et al. (2010a) investigated the vertical vibration of the human body coupled with an active seat suspension system using a robust state-feedback controller employing a delay-range-dependent Lyapunov function. The major focus of the controller design was the attenuation of the body acceleration considering saturation of the controller input, parametric uncertainty and the actuator delay. Du et al. (2012, 2013) proposed an integrated human body, seat suspension and vehicle chassis model for the vibration control analysis against the road disturbance in the presence of input saturation and varying driver body mass. Lyapunov function was employed in the static output-feedback controller for body acceleration attenuation, while an isolated bump was used to represent the road disturbance. A Terminal Sliding Mode Control (TSM) state and a disturbance observer were employed as a robust observer scheme for the estimation of the driver's body mass and the friction forces, integrated with a linear matrix inequality (LMI) approach for the robustness analysis of the closed-loop system (Ning et al., 2017a). Du et al. (2013) further assessed the potential of an optimized proportional-integral-derivative (PID) controller for the driver vibration reduction, where the parameter 'estimation' was performed using the parameter 'sensitivity analysis' under a random excitation frequency, considering the system parametric uncertainty.

The key features of the reported studies on seat suspension control problems may be deduced considering the important significant design issues. Firstly, an immense body of the studies have emphasized minimization of the seated body acceleration, while neglecting the limitations on permissible suspension-free travel. The relative suspension travel is a significant design issue in the context of a suspension seat, since it can contribute to transmission of shock motions to the seated body under high magnitudes of low frequency vibration, when the suspension travel exceeds its free travel. Secondly, the road excitations employed in most of the studies include an isolated single road bump, random excitations or excitations from a road surface which do not account for the dynamic tire-terrain interactions and the primary suspension dynamics, when present. Thirdly, relatively minimal efforts are evident on the effects of intermittent or repeated shocks arising from the tire-terrain interactions. Furthermore, a number of studies have considered head acceleration as the primary control objective by using multi-degree-of-freedom lumped-parameter biodynamic models of the seated operator (Du et al., 2013; Choi & Han, 2007; Zhao et al., 2010b). Although such an objective corresponds to the seat-to-head vibration transmissibility, the reduced head acceleration does not necessarily reflect the reduced vibration exposure of the operator. Moreover, the validity of the biodynamic models was not evident when coupled with the seat. Active suspension seat concepts employing combinations of different control schemes such as integrated fuzzy and sliding mode control showed improved control of vibration (Phu et al., 2016).

Studies on active suspensions have focused on sophisticated controller syntheses, as stated above, while the performance analyses have been limited to model simulations. Active suspension poses considerable challenges in view of hardware implementations, power demand, cost and reliability of the system in an off-road environment. The active suspension designs also require several parametric measurements such as acceleration, velocity and displacement to serve as feedbacks for the controller, which further adds to the cost and complexities. Alternatively, hybrid suspension designs integrating passive suspension together with an actively controlled actuator can be beneficial to achieve improved vibration suppression with lower power demand and hardware complexities. Stein (1997) developed a hybrid active suspension with a proportionally controlled electropneumatic actuator, and acceleration and relative displacement feedback. Laboratory evaluations performed under EM1 and EM2 excitations showed 60-70% lower vibration of the seated mass compared to the passive suspension. Frechin et al. (2004) evaluated the performance of an active pneumatic seat under harmonic vibration up to only 2 Hz, and showed 66% reduction in RMS acceleration compared to a passive suspension. The evaluations of a similar seat under EM3, EM5 and EM6 excitations showed a strong dependence of the SEAT factor on permissible suspension travel and a lower sensitivity to body mass variations. The SEAT factors with the active seat were substantially lower than those with the passive suspension. A pneumatic actuator is considered to be slow for suspension applications, while a hydraulic actuation system is too bulky due to the requirement of hydraulic power supply (Maciejewski, 2012a). A number of studies have explored active seat suspensions with electrodynamic actuators such as linear and rotational motors. Périsse and Jézéquel (2000a) employed a DC motor with a rack/pinion arrangement to develop an active seat suspension. A current feedback control was implemented for regulating the motor torque to achieve improved vibration control. Périsse and Jézéquel (2000b) proposed active and semi-active seat suspensions with an electrodynamic actuator together with a simple PI control based on current as a regulator. Both the active and semi-active suspensions revealed significant reductions in transmitted accelerations, and nearly 50% lower acceleration peaks compared to the conventional suspension. The gear reducers or mechanisms used to amplify the actuator's force/torque output, however, were found to deteriorate the performance due to high friction and chatter. Ning et al. (2016a) developed an active seat suspension with two controlled motors and a H_∞ controller for compensating friction. The controller was synthesized to minimize the low frequency transmitted vibration and maintaining passivity of the suspension at higher frequencies to achieve a compromise between the cost and performance. Experimental evaluations under harmonic (1-4.5 Hz range) and random vibration showed a substantial reduction in acceleration transmissibility and nearly 35% lower VDV and SEAT values compared to a passive suspension. Gohari and Tahmasebi (2015) employed an active force control method with artificial neural network (neuro-AFC control). Simulation results obtained for the suspension with neuro-AFC and PID control showed superior performance of the neuro-AFC control method. Simulations and experimental evaluations of a truck seat with a servo motor and an optical control law showed effective suppression of the resonance peak (Kawana & Shimogo, 1998).

Simulation results and laboratory assessments of hybrid active-passive suspension designs have shown promising performance potential in limiting the vibration transmission. The reported studies, however, do not describe the relative displacement performance of the suspension. A hybrid active suspension seat has been commercially developed by Bose Corp., while the information on its control algorithm could not be found. Field evaluations of this seat comprising a controlled linear electromagnetic force actuator for a highway tractor-semitrailer showed 19-55% lower vibration exposure compared to a passive air suspension (Blood et al., 2011). The laboratory evaluations of the same seat under simulated truck and bus excitations also showed superior performance of the active seat in view of the WBV exposure and acceleration transmissibility (Blood et al., 2015).

4.3 Summary

Reported studies have proposed an array of passive, semi-active and active suspension seat concepts, analytical models, controller syntheses and methods of analyses. Passive suspension seats are mostly designed with a cross-linkage mechanism to ensure vertical movement of the suspended seat pan. Seats with additional air reservoirs have been developed to achieve low natural frequency in a compact suspension design, which can provide enhanced vibration isolation. Low natural frequency designs, however, cause excessive relative motion of the seat with potential for end-stop impacts and thereby driver exposure to shock motions. Apart from the stiffness and damping properties, the performance of a suspension seat is affected by the seated mass, the user-selected ride height and the nature of vibration excitation in a highly complex manner. Although reported studies have provided considerable knowledge on the roles of various design and operating factors, a general guidance for designing or adapting suspension seats to a particular vehicle has not yet evolved. Design of a horizontal suspension seat is even more complex due to the dominance of horizontal vibration near very low frequencies (≈ 1 Hz). A standardized method has been developed to assess the vibration performance of vertical suspension seats, which presents many inherent limitations. The performance characteristics of suspension seats reported in different studies could not be compared due to broad differences in the objective measures, excitations and seat loads. The performance potential of semi-active MR suspension seats have been extensively evaluated via analytical and experimental methods. MR seat suspensions effectively eliminate end-stop impacts with only marginal gain on vibration isolation. Semi-active suspension seats, however, are considered effective only during the energy dissipation stage, since the control is limited only to suspension damping. Active suspension seats comprising controlled force generators can add or dissipate energy to achieve superior control of transmitted vibration. The vibration control performance of a semi-active or active suspension, however, is strongly dependent on the control logic. The studies have reported an array of controller syntheses for realizing improved vibration control such as sliding mode control, robust H_∞ control, fuzzy control, neural network control, and combinations of these. The hardware implementations of active suspensions pose considerable challenges due to the measurement requirements of many parameters such as acceleration, velocity and displacement, and high-power demand and cost. Hybrid suspension seat designs integrating an active force generator with a passive spring offer an attractive trade-off between cost/complexity and performance. A number of studies have reported designs of hybrid suspensions employing pneumatically and electrodynamically controlled actuators. A hybrid active suspension seat comprising a controlled electromagnetic actuator in parallel with an air spring has been commercially developed, and field and laboratory evaluations of this suspension suggested effective reductions in the vertical vibration exposure.

5. REVIEW OF COMMERCIALLY AVAILABLE SUSPENSION SEATS

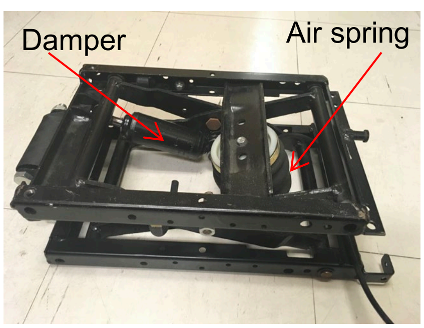
In this section, a review of the suspension seats available in the market is performed from the information available on the manufacturers' websites. The review is conducted on the main features of the seats that can be considered for the selection of a commercial seat for a specific machine and application.

5.1 Suspension features of commercially available seats

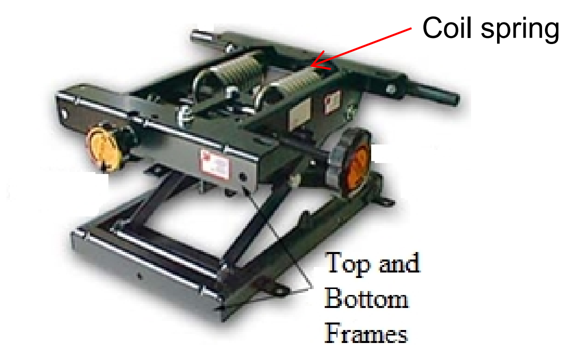
This section describes the main suspension features that are available on the seat manufacturer's website. The main suspension features of the reviewed seats, which consist of the target application or equipment (vehicle), the suspension type (passive, semi-active or active), the axis of vibration attenuation as well as the type of damping (fixed or adjustable) of the fore-aft isolator, when available, are listed in Appendix B, section B.I.

5.1.1 Suspension types

The two main categories of passive suspension are the air spring and coil spring suspensions. The suspensions based on an air spring are also called air suspensions (Figure 5.1a). They are generally used in most of the suspensions that have relatively large travel distance between the top and the bottom motion limiting end-stops. Suspensions with coil springs are also called mechanical suspensions (Figure 5.1b). They are generally found in low-profile suspensions which require smaller distance between the frames. Further information on suspension design is given in section 4.1.



(a)



(b)

Figure 5.1. Configurations of seat suspensions constructed with two different spring types: (a) Air suspension with a single air spring; (b) Mechanical suspension with a set of two coil springs.

From *Low profile mechanical suspension*, by Seats Canada, 2009. ©Seats Canada, 2009. Retrieved from http://seatscanada.com/catalogue/lowprofile.htm. Reprinted with permission.

5.1.2 Attenuation of vertical vibration and compliance with the ISO 7096:2000 standard

The ability of the seat to reduce the transmitted vibration from the cabin floor to the seated operator is evaluated following the general guidance of the ISO 10326-1:2016 standard (2016). More specific application standards are available for certain types of vehicles. As an example, for earth-moving machinery, the more specific ISO 7096:2000 standard (2000) defines some screening criteria. A suspension seat is considered acceptable to be used only if it fulfils two different vibration criteria defined in the ISO 7096:2000 standard (2000): a simulated input vibration test and a damping test. The simulated input test is conducted in the laboratory to evaluate the SEAT (Seat Effective Amplitude Transmissibility) factor based on nine different input spectral classes. Each spectral class represents the vibration generated on the cabin floor of a specific type of earth-moving machine (Figure 4.3). The ISO 7096:2000 standard further provides guidance on the selection of suspension seats for a class of vehicles on the basis of maximal acceptable values of SEAT factors (Table 4.1). Similarly, the damping test is used to evaluate the damping performance of the seat by identifying the transmissibility magnitude at the resonance frequency of the seat. For a seat to be acceptable, transmissibility magnitude should be less than 1.5 or 2.0, depending on the input spectral class (section 4.2.2 gives more detailed information on the standardized assessment of suspension seats).

The information available from the manufacturers' website does not provide any knowledge on the vibration isolation performance of the suspension seats. Only a few of the manufacturers mention compliance with the ISO 7096:2000 standard (2000), without stating the essential vibration performance measures such as the SEAT factor, when applied to a specific vehicle. There is also no declared information about the damping test and the expected natural frequency of the seat, as required in the standardized test. Moreover, the majority of the manufacturers recommend a particular suspension seat for widely different types of machines, whose vibration properties may differ considerably. Considering that the SEAT criterion in the standard is limited to specific classes of vehicles, the guidance provided by the manufacturers could, at the best, be considered as vague. This was also evident during personal meetings with the leading manufacturers, who expressed only limited knowledge of vibration properties of the suspension seats and their adaptability for different types of vehicles.

5.1.3 Damping control

The resonant response and vibration isolation performance of a suspension seat are strongly affected by its damping property, apart from many other factors. A seat suspension may comprise either one or two hydraulic dampers. Some suspension seats offer damping control, in the form of a switch to choose between low and high damping, in order to adapt the dynamic response of the seat to different ride conditions. However, there is a lack of information provided by the manufacturers on when to use the high or low damping mode. Thus, the operator has to decide upon the mode of suspension damping on the basis of his own perception of vibration. A number of suspension seats are also equipped with a semi-active magnetorheological (MR) fluid damper, which can automatically adjust the amount of damping depending on the suspension response to the vibration level at the seat base. However, the information about the vibration performance of the suspension seats with damping control, when applied to different vehicles, is not available.

5.1.4 Fore-aft and lateral vibration isolation

A large number of seats on the market are equipped with an optional fore-aft isolator that can be added to a vertical suspension seat to achieve attenuation of vibration along the x-axis (longitudinal). An add-on horizontal seat isolator has been presented in Figure 4.10. A few of the suspension seat manufacturers also offer a similar additional isolator for control of vibration along the y-axis (lateral). Discussions with the vehicle manufacturer revealed that the fore-aft and side-to-side motions, caused by the horizontal suspension, are generally perceived as annoying by the vehicle operators (A. Kordestani, personal communication, 2017). Such isolators are thus generally equipped with an on-off mechanism that permits the user to lock up the horizontal isolator if necessary. It is important to note that standardized test methods, similar to the ISO 7096:2000 and ISO 10326-1:2016 standards, do not yet exist for assessing vibration isolation performance of horizontal suspension seats. Moreover, the suspension seat manufacturers do not provide any quantitative knowledge of the vibration performance or suitability of such isolators for specific machines.

5.1.5 Height and weight adjustments and suspension stroke

The vast majority of the suspension seats offer a number of adjustments to enhance ergonomic performance of the seat and to provide a more controlled driving posture. Among the various adjustments available in a suspension seat, the adjustments for the ride height and body mass are most critical in view of the suspension performance and operators’ reach for the controls. All of the suspension seats thus offer these primary seat adjustment features. In most air suspensions, both the seat height and weight adjustments are achieved simultaneously by deflating or inflating the air spring. Some of the suspensions, however, offer height control independent of the weight adjustment. Air suspension seats mostly comprise a manual electric switch or pneumatic valve for changing the air volume or pressure in the air spring, together with a height indicator, which allows the operator to adjust the seat height properly. The indicator guarantees that the suspension operates in the designed operating range. For this type of suspension, the weight is adjusted automatically with the height, although it can alter the effective suspension stiffness with the body weight. Fully automatic height and weight adjustment seats are also available on the market. Depending on the operator’s weight, the seat adjusts its height to the suspension mid-ride position automatically, by controlling the flow of air to the air spring. Then, the operator can adjust the height with a switch from the mid-ride position to the desired height. On the other hand, mechanical suspensions generally possess two manual levers or knobs for weight and height adjustment. Generally, the weight is initially adjusted and is followed by the height adjustments. The mechanical suspensions incorporate a height indicator that permits the operator to adjust the seat height within the designed operating range, to ensure maximum free travel of the suspension between the two end-stops.

The suspension stroke is an important feature stated by all manufacturers and corresponds to the total available travel of the suspension between the end-stops. The stops are typically made from rubber material that helps to reduce the shocks caused by impacts with the end-stops, when the suspension travel exceeds its free travel.

Another important parameter is the adjustable height of the seat suspension, and it is defined as the allowable distance for the suspension to operate safely within the suspension stroke. The adjustable height is usually stated along with the suspension stroke. As an example, a suspension with a ±90 mm total stroke, having an adjustable height of ±40 mm from the suspension midposition will have a minimum stroke of 50 mm before reaching the end-stops. It is known that adjusting the seat height away from the suspension mid-ride position can degrade the suspension performance in attenuating the vibration and may cause end-stop impacts, leading to transmission of high magnitude vibration and shocks to the operator.

5.2 Ergonomic features of the seats

Suspension seats are designed to maximize the operator comfort during extensive working hours. Several ergonomic features are offered by the seat manufacturers, namely, lumbar support, adjustable cushion angle and depth, adjustable backrest angle and armrests, and seat rotation (swivel). The main ergonomic features of the reviewed seats are listed in Appendix B, section B.II.

5.2.1 Lumbar support

Most suspension seats are equipped with a mechanism that allows the operator to adjust the curve of the backrest cushion, depending on his perception of comfort. The mechanism is either pneumatic or mechanical. For pneumatic backrests, the operator can adjust the air pressure distribution inside the backrest cushion. For mechanical backrests, the lumbar support mechanism is controlled by a lever with more limited levels of adjustments as compared with the pneumatic backrest.

5.2.2 Seat cushion contour and angle/depth adjustments

Seat cushions are manufactured in different sizes to accommodate different machine operators. In North America, seat manufacturers offer wider and longer seat cushions than the ones used in the other markets. Seat cushions are typically flat or possess side wing supports, which provide lateral support for the seated driver. Seat manufacturers generally suggest seats with side-supported cushions for the off-highway, construction and mining trucks for enhanced lateral stability of the operator while driving. Figure 5.2 shows an example of two seats; one with a flat cushion and one with a side-supported cushion.

(a) (b)

Figure 5.2. Seats with two different cushion configurations: (a) Flat and (b) with side supports.

(a) From *ISRI NCS suspension seat*, 2012. ©Quarry, 2012. Retrieved from https://www.quarrymagazine.com/Article/2393/ISRI-NCS-suspension-seat. Reprinted with permission.

(b) Adapted from *Kingman standard seat*, 2021. ©Grammer, 2021. Retrieved from https://www.grammer.co.uk/kingman-standard-truck-seat

The seat cushions are fixed to the suspension top frame, also called the seat pan. Although seats incorporate a fore-aft adjustment mechanism, some seat cushions can also be adjusted with respect to the top frame in the horizontal direction. This adjustable distance is usually denoted by seat depth. Figure 5.3a illustrates the allowable motion of the seat cushion which can be activated using one of the controls below the cushion. The cushion angle, in addition, can be adjusted in some seats to maximize the operator's perception of comfort (Figure 5.3b).

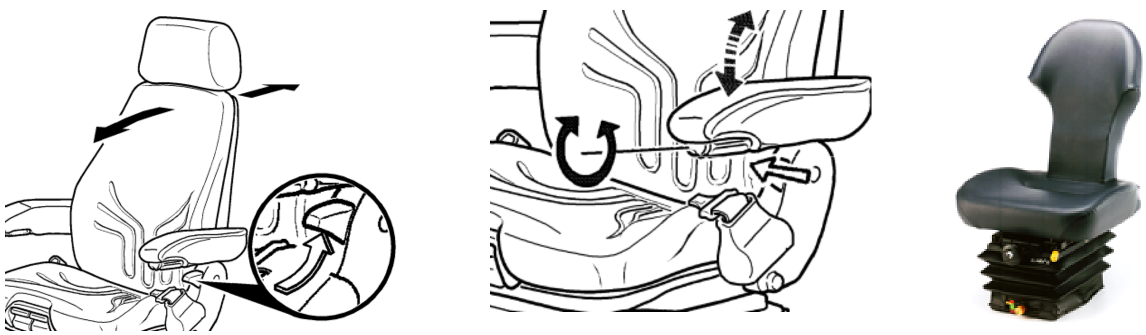
(a) (b)

Figure 5.3. Cushion adjustments with respect to the suspension top frame: (a) Cushion depth and (b) angle control.

From *Grammer MSG 97*, 2003. ©Grammer, 2003. Retrieved from https://usa.grammer.com/fileadmin/_migrated/content_uploads/MSG97_Series_Operating_Instructions_-_English_Only.pdf. Reprinted with permission.

5.2.3 Back and armrest adjustment

On most of the seats, the angle of the seat backrest can be adjusted to the operator's desired position using a mechanical locking lever, as shown in Figure 5.4a. The maximum inclination of the backrest is, however, limited by the available space behind the seat. Most seats also provide armrests that can be lifted to a vertical position to permit easy access to the seat. Some of the suspension seats comprise adjustable armrests, which permit variable angular position of the armrest. For example, turning the knobs located at the bottom of each armrest allows for variable angular position of the armrests (see Figure 5.4b). The space limitation in some underground mining machines can restrict the backrest adjustments as well as the existence of armrests. Figure 5.4c shows an example of a seat with no armrests and fixed backrest that is used in limited space environment cabins. Some of the seats offer essential provision to incorporate a fixed driving control panel on the right-side arm of the seat, especially for large size machines with hand-activated controls.



(a) (b) (c)

Figure 5.4. (a) Backrest, (b) armrest adjustments and (c) underground mining seat with no armrests and fixed backrest.

(a) and (b) From *Grammer MSG 97*, 2003. ©Grammer, 2003. Retrieved from https://usa.grammer.com/fileadmin/_migrated/content_uploads/MSG97_Series_Operating_Instructions_-_English_Only.pdf. Reprinted with permission.

(c) From *KAB 11-F1*, 2013. ©Kab Seating, 2013. Retrieved from http://www.kabseating.com.au/Product/kab-11-f1. Reprinted with permission.

5.2.4 Seat swivel

Some seats offer swivel feature that allows for adjustment of the seat orientation. Swivel adjustment is necessary for all machines that incorporate rear equipment similar to that of a backhoe loader. In order to operate the rear equipment, the operator unlocks the swivel mechanism and rotates the seat by 180°.

5.3 Summary

The vast majority of the suspension seats employ a cross-linkage platform, with mechanical or air spring, one or two hydraulic dampers and elastic travel limiters. All the designs provide adjustable seat height and adjustment for body weight. Some of the designs provide automatic ride height adjustment to ensure mid-ride suspension position. All the suspension designs also offer fore-aft adjustment, and adjustable cushion and backrest inclinations. The majority of the manufacturers recommend the same suspension designs for many different types of vehicles, which show notably different WBV patterns. Many manufacturers also offer semi-active suspension seats with controllable magnetorheological dampers. However, the performance in view of shock/vibration isolation performance of the different vertical and horizontal suspension seats could not be found.

The majority of the manufacturers offer, on most of their seats, a number of ergonomics design features, namely lumbar support, seat cushion angle and backrest angle. Adjustable armrests, cushion depth and seat swivel are also available on some of the seat models.

6. CONCLUSIONS

The transmission of whole-body vibration to the vehicle operators occurs through the seat cushion. A suspension of the seat is thus vital for limiting the WBV exposure, especially for the small- to mid-size vehicles without primary suspensions. A wide range of low frequency suspension seats have been commercially developed to reduce vibration exposure and thereby the associated health and safety risks among the exposed drivers. The vast majority of the suspensions employ a cross-linkage platform with either mechanical or air spring, one or two hydraulic dampers and elastic suspension travel limiters. All the designs permit seat height adjustment, which is generally coupled with adjustment for the occupants' weight in addition to fore-aft adjustment. Some of the designs also provide automatic ride height adjustment to ensure mid-ride suspension position and thereby reduced risk of shocks induced by interactions with the suspension travel limiters. The developments are mostly limited to vertical suspensions, although many off-road vehicles exhibit significant levels of vibration equally along the lateral as well as longitudinal directions. Horizontal suspensions have also been commercially developed, which can be added to a vertical seat suspension. Such suspensions, however, are not effective in limiting the horizontal WBV exposure due to their predominance at very low frequencies (≈ 1 Hz). Horizontal suspensions are thus designed with large friction to control the fore-aft or side-to-side motion of the seated driver. Large fore-aft travel of the horizontal isolator is also perceived annoying and uncomfortable by the operators, who frequently tend to lock the suspension.

Reported analytical and experimental studies have shown that suspension seats with air springs yield enhanced isolation of vertical vibration with relatively lower sensitivity to body mass variations, and permit height adjustment with greater ease. The introduction of an external air reservoir was perhaps the most notable design advancement, which facilitated compact designs of low frequency suspensions. Despite the advances, many field and laboratory evaluations have shown that suspension seats amplify the cabin vibration for many vehicles or yield only marginal reductions in transmitted vibration. This is partly due to lack of tuning or design of suspensions for particular vehicles. Moreover, the vibration reduction performance of a suspension seat is strongly affected by many design and operating factors in a highly complex manner. These include the static and dynamic properties of the suspension system, dynamic interactions of the seated human occupant, nature of excitation (frequency and magnitude), seated body mass and ride height.

Low frequency suspension seats can satisfy the SEAT criterion defined in the seat testing standards when adequately tuned for the target vehicle. The low frequency suspensions, however, yield large suspension travel, which is not only perceived uncomfortable and annoying by the seated occupant, but also presents greater risk of end-stop impacts, which cause shock motions at the seat. The seated body vibration and suspension travel constitute two opposite design targets for a suspension, which are not adequately addressed in the standardized test and evaluation method. The suspension design also involves conflicting damping requirements. Light damping is desirable for attenuation of continuous vibration, when the suspension motion is limited to its free travel, while the potential impacts against elastic end-stops under large vibration or shock excitations can be reduced via higher damping. An optimal performance of a suspension seat can thus be realized only when the suspension is tuned for a specific target vehicle. Suspension seat manufacturers, however, generally recommend an identical design for broad ranges of vehicles, whose vibration may differ substantially. This approach thus cannot provide optimal vibration reduction by the seat.

In recent years, considerable efforts have been made to design semi-active and active suspensions to achieve improved vibration isolation with limited suspension travel. Semi-active suspension seats with controllable magnetorheological (MR) fluid dampers have been commercially developed. Reported evaluations of MR seat suspensions show that such suspensions can effectively eliminate end-stop impacts with only marginal gain in the vibration isolation. The performance potential of semi-active suspension seats has been extensively evaluated via analytical and experimental methods, which suggest that the suspension performance is strongly affected by the damping control algorithm. Semi-active suspension seats, however, are considered effective only during the energy dissipation stage, since the control is limited only to suspension damping.

Many concepts in active suspension seats have evolved to achieve not only improved vibration isolation, but also enhanced robustness against uncertainties attributed to body mass and ride height variations, and differences in the vibration or shock excitations. Such suspensions comprise a controlled force generator that can add or dissipate energy to achieve superior control of transmitted vibration. The vibration control performance of an active suspension, however, is strongly dependent on the control logic, as reported for the semi-active suspensions. The reported studies have explored an array of controller syntheses, which invariably show superior vibration control performance. The hardware implementations of active suspensions, however, pose considerable challenges due to the requirements of parametric measurements to serve as feedback for the controller, and high power and cost. Alternatively, hybrid suspension seat designs integrating an active force generator with a passive spring offer attractive trade-off between cost/complexity and performance. A number of studies have reported designs of hybrid suspensions employing pneumatically or electrodynamically controlled actuators. A hybrid active suspension seat (Bose Ride® System II) comprising a controlled electromagnetic actuator in parallel with an air spring has now been commercially developed. Field and laboratory evaluations of this suspension suggested effective reductions in the vertical vibration exposure.

From the review of the currently available designs and reported analytical/experimental studies, it is evident that a generalized design/tuning methodology does not yet exist for adapting a suspension seat for specific target vehicles so as to achieve optimal vibration control. Further efforts are desirable for both design and assessments of suspension seats, which are summarized below.

- The current standardized method recommends measurements of the vibration isolation performance of suspension seats adjusted to mid-ride and loaded with subjects of 52 to 55 kg and 98 to 103 kg standing body mass. This approach does not permit the assessment of the suspension under different body masses and ride heights. The test method is also limited to the neutral sitting postures. The test/assessment method needs to be revised to account for the effects of body mass, seat height variations and sitting posture on the vibration isolation performance.

- The standardized method is limited only to vertical vibration. Owing to equally large magnitudes of horizontal vibration in many vehicles, it is important to develop a standardized methodology for assessing horizontal suspension seats.

- The relative displacement of the low natural frequency suspension is a critical design and performance factor, which is not addressed in the standardized methods or the vast majority of the reported studies. The probability of end-stop impacts is directly related to suspension travel and the user-selected ride height. Suspensions with automatic ride height adjustments can not only prevent end-stop impacts but also provide improved vibration isolation. Further efforts are also needed to develop threshold values of the seat displacement, which are perceived comfortable by the operators. These will serve as an essential guidance for optimal suspension seat design.

- A wide range of passive and semi-active suspension seats have been commercially developed. Manufacturers generally recommend a suspension design for a broad range of vehicles that exhibit considerably different vibration properties, while the vibration performance of these seats under the recommended vehicles' vibration is not known. Field studies, however, show that suspension seats can amplify cabin vibration in many vehicles. This suggests that the suspension designs are not adequately tuned for the target vehicle vibration. The development of a vehicle-specific suspension seat design methodology is thus vital for realizing optimal suspension performance for different classes of vehicles.

- The kinematics of the cross-linkage suspension platform offers considerable design flexibility for convenient tuning of the suspension for given vehicle vibration spectra and body mass. Only limited efforts have been made on the effect of suspension kinematics upon the effective stiffness and damping properties of the suspension. Further efforts on kineto-dynamic analyses of suspensions can provide important effects of the spring and damper mounting locations on the effective stiffness and damping properties. The resulting relations can provide guidance for tuning/adapting a given suspension for specific body masses and vehicle vibration spectra.

- Systematic laboratory and field assessments of semi-active MR suspension seats with different control logic are needed to identify their performance benefits for broad classes of vehicles. This will permit tuning of the controller for specific vehicle vibration spectra.

- A hybrid suspension integrating a controlled electromagnetic actuator such as a servo motor with the passive suspension components offers considerable performance potential with a good trade-off between performance and cost. Further efforts are desirable for developing robust controllers to account for variations in body mass, seat height and vibration excitations. This will allow their adaptation to different vehicles and drivers with varied body mass and stature. The laboratory/field assessments of such suspensions under different classes of vehicle vibration are most important.

REFERENCES

Abbas, W., Emam, A., Badran, S., Shebl, M. & Abouelatta, O. (2013). Optimal seat and suspension design for a half-car with driver model using genetic algorithm. *Intelligent Control and Automation*, *4*(2), 199-205. doi: 10.4236/ica.2013.42024

Afkar, A., Javanshir, I., Taghi Ahmadian, M. & Ahmadi, H. (2013). Optimization of a passenger occupied seat with suspension system exposed to vertical vibrations using genetic algorithms. *Journal of Vibroengineering*, *15*(2), 979-991.

Alfadhli, A., Darling, J. & Hillis, A. J. (2018). The control of an active seat with vehicle suspension preview information. *Journal of Vibration and Control*, *24*(8), 1412-1426.

Avdagic, Z., Besic, I., Buza, E. & Omanovic, S. (2013). Comparation of controllers based on Fuzzy Logic and Artificial Neural Networks for reducing vibration of the driver's seat. In *IECON 2013, 39th Annual Conference of the IEEE Industrial Electronics Society: 10-13 Nov. 2013, Vienna, Austria: Proceedings* (p. 3382-3387). Piscataway, NJ: IEEE. doi: 10.1109/IECON.2013.6699671

Bai, X. X., Jiang, P. & Qian, L. J. (2017). Integrated semi-active seat suspension for both longitudinal and vertical vibration isolation. *Journal of Intelligent Material Systems and Structures*, *28*(8), 1036-1049.

Blood, R. P., Dennerlein, J., Lewis, C., Rynell, P. & Johnson, P. W. (2011). Evaluating whole-body vibration reduction by comparison of active and passive suspension seats in semi-trucks. *Proceedings of the Human Factors and Ergonomics Society Annual Meeting*, *55*(1), 1750-1754. doi: 10.1177/1071181311551363

Blood, R. P., Ploger, J. D. & Johnson, P. W. (2010). Whole body vibration exposures in forklift operators: Comparison of a mechanical and air suspension seat. *Ergonomics*, *53*(11), 1385-1394.

Blood, R. P., Ploger, J. D., Yost, M. G., Ching, R. P. & Johnson, P. W. (2010). Whole body vibration exposures in metropolitan bus drivers: A comparison of three seats. *Journal of Sound and Vibration*, *329*(1), 109-120.

Blood, R. P., Yost, M. G., Camp, J. E. & Ching, R. P. (2015). Whole-body vibration exposure intervention among professional bus and truck drivers: A laboratory evaluation of seat-suspension designs. *Journal of Occupational and Environmental Hygiene*, *12*(6), 351-362.

Blüthner, R., Hinz, B., Menzel, G., Schust, M. & Seidel, H. (2006). On the significance of body mass and vibration magnitude for acceleration transmission of vibration through seats with horizontal suspensions. *Journal of Sound and Vibration*, *298*(3), 627-637.

Blüthner, R., Seidel, H. & Hinz, B. (2008). Laboratory study as basis of the development for a seat testing procedure in horizontal directions. *International Journal of Industrial Ergonomics*, *38*(5-6), 447-456. doi: 10.1016/j.ergon.2007.08.008

Boileau, P.-É. & Rakheja, S. (1990). Vibration attenuation performance of suspension seats for off-road forestry vehicles. *International Journal of Industrial Ergonomics*, *5*(3), 275-291.

Boileau, P.-É. & Rakheja, S. (1995) *Étude des paramètres affectant l'efficacité d'atténuation des vibrations par un siège suspendu* (Report No. R-095). Montreal, QC: IRSST.

Boileau P.-É. & Rakheja S. (1997) *Essais en vibration de sièges pour autobus urbains : phase 2 : évaluation de sièges candidats* (Report No. R-161). Montreal, QC: IRSST.

Boileau, P.-É. & Rakheja, S. (2000) Caractérisation de l'environnement vibratoire dans différentes catégories de véhicules: industriels, utilitaires et de transport urbain. (Report No. R-242). Montreal, QC: IRSST.

Boileau, P.-É., Rakheja, S. & Liu, P. J. (1997). A combined suspension seat–vehicle driver model for estimating the exposure to whole-body vehicular vibration and shock. *International Journal of Heavy Vehicle Systems*, *4*(2-4), 244-265.

Boileau, P.-É., Rakheja, S. & Wang, Z. (2004) *Étude des mécanismes de conception visant à limiter l'importance des impacts occasionnés par le talonnement des sièges à suspension* (Report No. R-368). Montreal, QC: IRSST.

Bouazara, M., Richard, M. J. & Rakheja, S. (2006). Safety and comfort analysis of a 3-D vehicle model with optimal non-linear active seat suspension. *Journal of Terramechanics*, *43*(2), 97-118.

Bovenzi, M. (2017). Low back pain and exposure to whole-body vibration and mechanical shocks.In M. Hagberg (Ed.), *WBV 2017: 6th International Conference on Whole-Body Vibration Injuries: Abstracts* (p. 53-55). Gothenburg, Sweden: AMM.

Burdorf, A. & Swuste, P. (1993). The effect of seat suspension on exposure to whole-body vibration of professional drivers. *The Annals of Occupational Hygiene*, *37*(1), 45-55.

Burström, L., Nilsson, T. & Wahlström, J. (2015). Whole-body vibration and the risk of low back pain and sciatica: A systematic review and meta-analysis. *International Archives of Occupational and Environmental Health*, *88*(4), 403-418.

Cao, D., Rakheja, S. & Su, C. Y. (2010). Roll- and pitch-plane coupled hydro-pneumatic suspension: Part 1: Feasibility analysis and suspension properties. *Vehicle System Dynamics*, *48*(3), 361-386.

Cation, S., Jack, R., Oliver, M., Dickey, J. P. & Lee-Shee, N. K. (2008). Six degree of freedom whole-body vibration during forestry skidder operations. *International Journal of Industrial Ergonomics*, *38*(9-10), 739-757.

Cheok, K. C., Hu, H. X. & Loh, N. K. (1989). Discrete-time frequency-shaping parametric LQ control with application to active seat suspension control. *IEEE Transactions on Industrial Electronics*, 36(3), 383-390.

Choi, S.-B., Choi, J. H., Nam, M. H., Cheong, C. C. & Lee, H. G. (1998). A semi-active suspension using ER fluids for a commercial vehicle seat. *Journal of Intelligent Material Systems and Structures*, *9*(8), 601-606.

Choi, S.-B. & Han, Y. M. (2007). Vibration control of electrorheological seat suspension with human-body model using sliding mode control. *Journal of Sound and Vibration*, *303*(1-2), 391-404.

Choi, S.-B., Nam, M. H. & Lee, B. K. (2000). Vibration control of a MR seat damper for commercial vehicles. *Journal of Intelligent Material Systems and Structures*, *11*(12), 936-944.

Choi, Y. T. & Wereley, N. M. (2005). Mitigation of biodynamic response to vibratory and blast-induced shock loads using magnetorheological seat suspensions. *Proceedings of the Institution of Mechanical Engineers, Part D: Journal of Automobile Engineering*, *219*(6), 741-753.

Cullmann, A. & Wölfel, H. P. (2001). Design of an active vibration dummy of sitting man. *Clinical Biomechanics, 16(1), S64-S72.* doi: 10.1016/S0268-0033(00)00107-8

Dewangan, K. N., Rakheja, S., Marcotte, P., Shahmir, A. & Patra, S. K. (2013). Comparisons of apparent mass responses of human subjects seated on rigid and elastic seats under vertical vibration. *Ergonomics*, *56*(12), 1806-1822.

Directive 2002/44/EC of the European Parliament and the Council of 25 June 2002 on the minimum health and safety requirements regarding the exposure of workers to the risks arising from physical agents (vibration) (sixteenth individual Directive within the meaning of Article 16 (1) of Directive 89/391/EEC). *Official Journal of the European Communities, L, 117*(13), July 6th, 2002 (EC).

Dong, Y. P. & Lu, Z. H. (2012). Analysis and evaluation of an anti-shock seat with a multi-stage non-linear suspension for a tactical vehicle under a blast load. *Proceedings of the Institution of Mechanical Engineers, Part D: Journal of Automobile Engineering*, *226*(8), 1037-1047.

Du, H., Li, W. & Zhang, N. (2012). Integrated seat and suspension control for a quarter car with driver model. *IEEE Transactions on Vehicular Technology*, *61*(9), 3893-3908.

Du, H., Li, W. & Zhang, N. (2013). Semi-active control of an integrated full-car suspension with seat suspension and driver body model using ER dampers. *International Journal of Vehicle Design*, *63*(2-3), 159-184.

Duke, M. & Fow, A. (2012). Comparison of simulation and experimental results of a tractor seat with nonlinear stiffness and dead-band damping. *Journal of Vibration and Acoustics*, *134*(5).

Duke, M. & Goss, G. (2007). Investigation of tractor driver seat performance with non-linear stiffness and on–off damper. *Biosystems Engineering*, *96*(4), 477-486.

Ebe, K. & Griffin, M. J. (2001). Factors affecting static seat cushion comfort. *Ergonomics*, *44*(10), 901-921.

Eger, T. R., Kociolek, A. M. & Dickey, J. P. (2013). Comparing health risks to load-haul-dump vehicle operators exposed to whole-body vibration using EU Directive 2002/44EC, ISO 2631-1 and ISO 2631-5. *Minerals*, *3*(1), 16-35.

Ekman, M., Johnell, O. & Lidgren, L. (2005). The economic cost of low back pain in Sweden. *Acta Orthopaedica, 76*(2), 278-284.

Els, P. S., Theron, N. J., Uys, P. E. & Thoresson, M. J. (2007). The ride comfort and handling compromise for off-road vehicles. *Journal of Terramechanics*, *44*(4), 303-317.

Ezzine, J. & Tedesco, F. (2009). Mixed sensitivity H∞ control approach for regulation of active suspension on half-car model with seat passengers. *Proceedings of the 7th EUROMECH Solid Mechanics Conference, Lisbon, Portugal* (p. 7-11). Retrived from https://tedescof.files.wordpress.com/2011/02/ezzine-2009-mixed-sensitivity-he2889e-control-approach-for-regulation-of-active-suspension-on-half-car-model-with-seat-passengers.pdf

Fleury, G. & Mistrot, P. (2006). Numerical assessment of fore-and-aft suspension performance to reduce whole-body vibration of wheel loader drivers. *Journal of Sound and Vibration*, *298*(3), 672-687.

Frechin, M. M., Arino, S. B. & Fontaine, J. (2004). ACTISEAT: Active vehicle seat for acceleration compensation. *Proceedings of the Institution of Mechanical Engineers, Part D: Journal of Automobile Engineering*, *218*(9), 925-933.

Gad, S., Metered, H., Bassuiny, A. & Abdel Ghany, A.M. (2015). Ride comfort enhancement of heavy vehicles using magnetorheological seat suspension. *International Journal of Heavy Vehicle Systems, 22*(2), 93–113

Gohari, M., Rahman, R. A., Raja, R. I. & Tahmasebi, M. (2012). Bus seat suspension modification for pregnant women. *In International Conference on Biomedical Engineering (ICoBE), 2012 27-28 Feb. 2012, Penang, Malaysia* (p. 404-407). Piscataway, NJ: IEEE. doi: 10.1109/ICoBE.2012.6179047

Gohari, M. & Tahmasebi, M. (2014). Off-road vehicle seat suspension optimisation: Part II: Comparative study between meta-heuristic optimisation algorithms. *Journal of Low Frequency Noise, Vibration and Active Control*, *33*(4), 443-454.

Gohari, M. & Tahmasebi, M. (2015). Active off-road seat suspension system using intelligent active force control. *Journal of Low Frequency Noise, Vibration and Active Control*, *34*(4), 475-489.

Gouw, G., Rakheja, S., Sankar, S. & Afework, Y. (1990). Increased comfort and safety of drivers of off-highway vehicles using optimal seat suspension. *SAE Transactions, 99*, 514-548. doi: 10.4271/901646

Griffin, M. J. (1990). *Handbook of human vibration*. Cambridge, MA: Academic Press. doi: 10.1016/C2009-0-02730-5

Grujicic, M., Pandurangan, B., Arakere, G., Bell, W. C., He, T. & Xie, X. (2009). Seat-cushion and soft-tissue material modelling and a finite element investigation of the seating comfort for passenger-vehicle occupants. *Materials Design*, *30*(10), 4273-4285.

Gu, Y. (1999). *A new dummy for vibration transmissibility measurement in improving ride comfort* (Technical Paper No. 01-0629). S.l.: SAE. doi: 10.4271/1999-01-0629

Guclu, R. (2005). Fuzzy logic control of seat vibrations of a non-linear full vehicle model. *Nonlinear Dynamics*, *40*(1), 21-34.

Guglielmino, E., Stammers, C. W., Stancioiu, D., Sireteanu, T. & Ghigliazza, R. (2005). Hybrid variable structure-fuzzy control of a magnetorheological damper for a seat suspension. *International Journal of Vehicle Autonomous Systems*, *3*(1), 34-46.

Gunaselvam J. J. & van Niekerk J. L. (2005). Seat selection guidelines to reduce whole-body vibration exposure levels in the SA mining industry. *Journal of the Southern African Institute of Mining and Metallurgy*, *105*(10), 675-686.

Gunston, T. (2000). *An investigation of suspension seat damping using a theoretical model*. Paper presented at the 35th UK Conference on Human Responses to Vibration, Southampton, England (p. 13-15).

Gunston, T. P., Rebelle, J. & Griffin, M. J. (2004). A comparison of two methods of simulating seat suspension dynamic performance. *Journal of Sound and Vibration*, *278*(1-2), 117-134

Han, Y. M., Jung, J. Y., Choi, S. B. & Wereley, N. M. (2005). Sliding mode control of ER seat suspension considering human vibration model. *International Journal of Modern Physics B*, 19(7-9), 1689-1695.

Han, Y. M., Jung, J. Y., Choi, S. B., Choi, Y. T. & Wereley, N. M. (2006). Ride quality investigation of an electrorheological seat suspension to minimize human body vibrations. *Proceedings of the Institution of Mechanical Engineers, Part D: Journal of Automobile Engineering*, *220*(2), 139-150.

Hill, K. & Dhingra, A. (2003). Modeling, analysis and optimization of a scissors linkage seat suspension. *Engineering Optimization*, *35*(4), 341-357.

Hinz, B., Menzel, G., Blüthner, R. & Seidel, H. (1998). Laboratory testing of operator seat vibration with 37 subjects: Critical comment on ISO/DIS 7096. *Journal of Sound and Vibration*, *215*(4), 977-988.

Hinz B., Rutzel S., Blüthner, R., Menzel G. & Wolfel H. P., Seidel H. (2006). Apparent mass of seated men: First determination with a soft seat and dynamic seat pressure distribution. *Journal of Sound and Vibration*, *298*(3), 704-724.

Hinz, B., Seidel, H., Menzel, G. & Blüthner, R. (2002). Effects related to random whole-body vibration and posture on a suspended seat with and without backrest. *Journal of Sound and Vibration*, *253*(1), 265-282.

Holtz, M. W. & Van Niekerk, J. L. (2010). Modelling and design of a novel air-spring for a suspension seat. *Journal of Sound and Vibration*, *329*(21), 4354-4366.

Hostens, I., Deprez, K. & Ramon, H. (2004). An improved design of air suspension for seats of mobile agricultural machines. *Journal of Sound and Vibration*, *276*(1-2), 141-156.

International Organization for Standardization. (1997*). Mechanical vibration and shock: Evaluation of human exposure to whole-body vibration: Part 1: General requirements*. Standard ISO 2631-1:1997. Geneva, Switzerland: ISO.

International Organization for Standardization. (2000). *Earth-moving machinery: Laboratory evaluation of operator seat vibration*. Standard ISO 7096:2000. Geneva, Switzerland: ISO.

International Organization for Standardization. (2003). *Agricultural wheeled tractors: Operator's seat: Laboratory measurement of transmitted vibration.* Standard ISO 5007:2003. Geneva, Switzerland: ISO.

International Organization for Standardization. (2016). *Mechanical vibration: Laboratory method for evaluating vehicle seat vibration:Part 1: Basic requirements*. Standard ISO 10326-1:2016. Geneva, Switzerland: ISO.

International Organization for Standardization (2018). *Mechanical vibration and shock: Evaluation of human exposure to whole-body vibration: Part 5: Method for evaluation of vibration containing multiple shocks*. Standard ISO 2631-5:2018. Geneva, Switzerland: ISO.

Jack, R. J., Oliver, M., Dickey, J. P., Cation, S., Hayward, G. & Lee-Shee, N. (2010). Six-degree-of-freedom whole-body vibration exposure levels during routine skidder operations. *Ergonomics*, *53*(5), 696-715.

Jin, A., Zhang, W., Wang, S., Yang, Y. & Zhang, Y. (2014). Modeling air-spring suspension system of the truck driver seat. *SAE International Journal of Commercial Vehicles*, *7*, 157-162.

Jonsson, P. M., Rynell, P. W., Hagberg, M. & Johnson, P. W. (2015). Comparison of whole-body vibration exposures in buses: Effects and interactions of bus and seat design. *Ergonomics*, *58*(7), 1133-1142.

Karnopp, D., Crosby, M. J. & Harwood, R. A. (1974). Vibration control using semi-active force generators. *Journal of Engineering for Industry*, *96*(2), 619-626.

Kawana, M. & Shimogo, T. (1998). Active suspension of truck seat. *Shock and Vibration*, *5*(1), 35-41.

Kim, J. H., Dennerlein, J. T. & Johnson, P. W. (2018). The effect of a multi-axis suspension on whole body vibration exposures and physical stress in the neck and low back in agricultural tractor applications. *Applied Ergonomics*, 68, 80-89. doi: 10.1016/j.apergo.2017.10.021

Kim, J. H., Zigman, M., Aulck, L. S., Ibbotson, J. A., Dennerlein, J. T. & Johnson, P. W. (2016). Whole body vibration exposures and health status among professional truck drivers: A cross-sectional analysis. *Annals of Occupational Hygiene*, *60*(8), 936-948.

Kitazaki, S. & Griffin, M.J. (1998). Resonance behaviour of the seated human body and effect of posture. *Journal of Biomechanics*, *31*(2), 143-149.

Książek, M. A. & Ziemiański, D. (2012). Optimal driver seat suspension for a hybrid model of sitting human body. *Journal of Terramechanics*, *49*(5), 255-261.

Lathkar, M., Tiwari, A., Shendge, P. D. & Phadke, S. B. (2016). Active seat suspension based on second order inertial delay control combined with sliding mode control. In *Proceedings of the 1st IEEE International Conference on Power Electronics, Intelligent Control and Energy Systems (ICPEICES): July 4-6, 2016*. Piscataway, NJ: IEEE. doi: 10.1109/ICPEICES.2016.7853309

Le, T. D. & Ahn, K. K. (2011). A vibration isolation system in low frequency excitation region using negative stiffness structure for vehicle seat. *Journal of Sound and Vibration*, *330*(26), 6311-6335.

Lee, C. M. & Goverdovskiy, V. N. (2002). Alternative vibration protecting systems for men-operators of transport machines: Modern level and prospects. *Journal of Sound and Vibration*, *249*(4), 635-647.

Lee, S. H., Park, J. S., Jung, B. K. & Lee, S. A. (2016). Effects of different seat cushions on interface pressure distribution: A pilot study. *Journal of Physical Therapy Science*, *28*(1), 227-230.

Lewis, C. A. & Johnson, P. W. (2012). Whole-body vibration exposure in metropolitan bus drivers. *Occupational Medicine*, *62*(7), 519-524.

Lewis, C. H. & Griffin, M. J. (2005). Evaluating the vibration isolation of soft seat cushions using an active anthropodynamic dummy. *Journal of Sound and Vibration*, 253(1), 2002, 295-311.

Li, H., Liu, H., Gao, H. & Shi, P. (2012). Reliable fuzzy control for active suspension systems with actuator delay and fault. *IEEE Transactions on Fuzzy Systems*, *20*(2), 342-357.

Lines, J., Stiles, M. & Whyte, R. (1995). Whole body vibration during tractor driving. *Journal of Low Frequency Noise, Vibration and Active Control*, *14*(2), 87-104.

Lings, S. & Leboeuf-Yde, C. (2000). Whole-body vibration and low back pain: A systematic critical review of the epidemiological literature 1992-1999. *International Archives of Occupational and Environmental Health*, *73*, 290-297. doi: 10.1007/s004200000118

Ma, X. Q., Rakheja, S. & Su, C. Y. (2008a). Damping requirement of a suspension seat subject to low frequency vehicle vibration and shock. *International Journal of Vehicle Design*, *47*(1-4), 133-156.

Ma, X. Q., Rakheja, S. & Su, C. Y. (2008b). Synthesis of a semi-active suspension seat for attenuation of whole-body vibration and shock. *International Journal of Vehicle Design*, *47*(1-4), 157-175.

Maciejewski, I. (2012a). Control system design of active seat suspensions. *Journal of Sound and Vibration*, *331*(6), 1291-1309.

Maciejewski, I. (2012b). Load-dependent controller of the active seat suspension with adaptive mass recognizing. *Acta Mechanica et Automatica*, *6*(2), 66-70.

Maciejewski, I., Kiczkowiak, T. & Krzyzynski, T. (2011). Application of the Pareto-optimal approach for selecting dynamic characteristics of seat suspension systems. *Vehicle System Dynamics*, *49*(12), 1929-1950.

Maciejewski, I. & Krzyżyński, T. (2011). Control design of semi-active seat suspension systems. *Journal of Theoretical and Applied Mechanics*, *49*(4), 1151-1168.

Maciejewski, I., Meyer, L. & Krzyzynski, T. (2009). Modelling and multi-criteria optimisation of passive seat suspension vibro-isolating properties. *Journal of Sound and Vibration*, *324*(3-5), 520-538.

Marcotte, P., Beaugrand, S., Boutin, J. & Larue, C. (2010). Design and evaluation of a suspension seat to reduce vibration exposure of subway operators: A case study. *Industrial Health*, *48*(5), 715-724.

Marin, L. S., Rodriguez, A. C., Rey-Becerra, E., Piedrahita, H., Barrero, L. H., Dennerlein, J. T. & Johnson, P. W. (2017). Assessment of whole-body vibration exposure in mining earth-moving equipment and other vehicles used in surface mining. *Annals of Work Exposure and Health, 61*(6), 669-680.

Mayton, A. G., DuCarme, J. P., Jobes, C. C. & Matty, T. J. (2006). Laboratory investigation of seat suspension performance during vibration testing. In *Proceedings of the ASME Design Engineering Division and the ASME Computers and Information in Engineering Division 2006: Presented at 2006 ASME International Mechanical Engineering Confress and Exposition, November 5-10, 2006, Chicago, Illinois, USA* (p. 177-183). New York, NY: ASME. doi: doi.org/10.1115/IMECE2006-14146

McManus, S. J., Clair, K. S., Boileau, P. E., Boutin, J. & Rakheja, S. (2002). Evaluation of vibration and shock attenuation performance of a suspension seat with a semi-active magnetorheological fluid damper. *Journal of Sound and Vibration*, *253*(1), 313-327.

Mehta, C. R. & Tewari, V. K. (2000). Seating discomfort for tractor operators: A critical review. *International Journal of Industrial Ergonomics*, *25*(6), 661-674.

Metered, H., Bonello, P. & Oyadiji, S. O. (2009). Vibration control of a seat suspension system using magnetorheological damper. In *ASME 2009 International Design Engineering Technical Conferences and Computers and Information in Engineering Conference (*p 1021-1029). New York, NY: ASME. doi: 10.1115/DETC2009-86081

Metered, H. & Šika, Z. (2014). Vibration control of a semi-active seat suspension system using magnetorheological damper. In *IEEE/ASME 10th International Conference on Mechatronic and Embedded Systems and Applications (MESA), 2014 10-12 Sept. 2014, Hotel Duchi della Rovere, Senigallia, Ancona, Italy.* Piscataway, NJ: IEEE. doi: 10.1109/MESA.2014.6935527

Motmans, R. (2012). Reducing whole body vibration in forklift drivers. *Work*, *41*(1), 2476-2481. doi: 10.3233/WOR-2012-0484-2476

Nagarkar, M. P., Patil, G. J. V. & Patil, R. N. Z. (2016). Optimization of nonlinear quarter car suspension–seat–driver model. *Journal of Advanced Research*, *7(*6), 991-1007.

Nélisse, H., Boileau, P.-É., Rakheja S., Patra, S. & Boutin, J. (2006). Évaluation de mannequins anthropodynamiques destinés aux essais de vibrations des sièges en laboratoires (Report No. R-478). Montreal, QC: IRSST.

Ning, D., Sun, S., Li, H., Du, H. & Li, W. (2016). Active control of an innovative seat suspension system with acceleration measurement based friction estimation. *Journal of Sound and Vibration*, *384*(8), 28-44.

Ning, D., Sun, S., Wei, L., Zhang, B., Du, H. & Li, W. (2017). Vibration reduction of seat suspension using observer based terminal sliding mode control with acceleration data fusion. *Mechatronics*, *44*, 71-83. doi: 10.1016/j.mechatronics.2017.04.012

Ning, D., Sun, S., Zhang, F., Du, H., Li, W. & Zhang, B. (2017). Disturbance observer based Takagi-Sugeno fuzzy control for an active seat suspension. *Mechanical Systems and Signal Processing*, *93*, 515-530. doi: 10.1016/j.ymssp.2017.02.029

Ning, D., Sun, S., Zhang, J., Du, H., Li, W. & Wang, X. (2016). An active seat suspension design for vibration control of heavy-duty vehicles. *Journal of Low Frequency Noise, Vibration and Active Control*, *35*(4), 264-278.

Orečný, M., Segľa, Š., Huňady, R. & Ferková, Ž. (2014). Application of a magneto-rheological damper and a dynamic absorber for a suspension of a working machine seat. *Procedia Engineering*, *96*, 338-344. doi: 10.1016/j.proeng.2014.12.127

Oshinoya, Y., Ooka, K. & Ishibashi, K. (1998). Simulation study on active suspension of truck seat using sliding mode control. *Proceedings of the School of Engineering of Tokai University*, *23*, 57-73.

Oshinoya, Y., Ooka, K. & Ishibashi, K. (1999). Experimental study on active suspension of truck seat using sliding mode control. *Proceedings of the School of Engineering of Tokai University*, *24*, 43-49.

Patil, M. K. & Palanichamy, M. S. (1988). A mathematical model of tractor-occupant system with a new seat suspension for minimization of vibration response. *Applied Mathematical Modelling*, *12*(1), 63–71

Pazooki, A., Rakheja, S. & Cao, D. (2012). Modeling and validation of off-road vehicle ride dynamics. *Mechanical Systems and Signal Processing*, *28*, 679-695. doi: 10.1016/j.ymssp.2011.11.006

Périsse, J. & Jézéquel, L. (2000a). An original feedback control with a reversible electromechanical actuator used as an active isolation system for a seat suspension: Part I: Theoretical study. *Vehicle System Dynamics*, *34*(5), 305-331.

Périsse, J. & Jézéquel, L. (2000b). An original feedback control with a reversible electromechanical actuator used as an active isolation system for a seat suspension: Part II: Experimental study. *Vehicle System Dynamics*, *34*(6), 381-399.

Phu, D. X., Choi, S. B., Lee, Y. S. & Han, M. S. (2016). Vibration control of a vehicle's seat suspension featuring a magnetorheological damper based on a new adaptive fuzzy sliding-mode controller. *Proceedings of the Institution of Mechanical Engineers, Part D: Journal of Automobile Engineering*, *230*(4), 437-458.

Phu, D. X., Shin, D. K. & Choi, S.-B. (2015). Design of a new adaptive fuzzy controller and its application to vibration control of a vehicle seat installed with an MR damper. *Smart Materials and Structures*, *24*(8).

Plewa, K. M., Eger, T. R., Oliver, M. L. & Dickey, J. P. (2012). Comparison between ISO 2631–1 comfort prediction equations and self-reported comfort values during occupational exposure to whole-body vehicular vibration. *Journal of Low Frequency Noise, Vibration and Active Control*, *31*(1), 43-53.

Politis, H., Rakheja, S., Boileau, P.-É., Juras, D. & Boutin, J (2003). Limits of application of human body dynamics in assessing vibration comfort of seats. *SAE Transactions, Journal of Passenger Cars: Mechanical Systems*, *112*(6), 973-979.

Prasad, N., Tewari, V. K. & Yadav, R. (1995). Tractor ride vibration: A review. *Journal of Terramechanics*, *32*(4), 205-219.

Punnett, L., Prüss-Ütün, A., Nelson, D. I., Fingerhut, M. A., Leigh, J., Tak, S. & Phillips, S. (2005). Estimating the global burden of low back pain attributable to combined occupational exposures. *American Journal of Industrial Medicine*, *48*(6), 459-469.

Rajendiran, S. & Lakshmi, P. (2016). Simulation of PID and fuzzy logic controller for integrated seat suspension of a quarter car with driver model for different road profiles. *Journal of Mechanical Science and Technology*, *30*(10), 4565-4570.

Rakheja, S., Afework, Y. & Sankar, S. (1994). An analytical and experimental investigation of the driver-seat-suspension system. *Vehicle System Dynamics*, *23*(1), 501-524.

Rakheja, S., Boileau, P. - É. & Wang, Z. (2004). Performance analysis of suspension seats under high magnitude vibration excitations: II: Design parameter study. *Journal of Low Frequency Noise, Vibration and Active Control*, *23*(1), 7-25.

Rakheja, S., Boileau, P. - É., Wang, Z. & Politis, H. (2003). Performance analysis of suspension seats under high magnitude vibration excitations: Part 1: Model development and validation. *Journal of Low Frequency Noise, Vibration and Active Control*, *22*(4), 225-252.

Rakheja, S., Dewangan, K., Marcotte, P., Shahmir, A. & Patra, S. (2015). An exploratory study for characterizing seated body apparent mass coupled with elastic seats under vertical vibration (Report No. R-884). Montreal, QC: IRSST.

Rakheja, S., Dewangan, K., Dong, R. G. & Marcotte, P. (2020). Whole-body vibration biodynamics: A critical review. I. Experimental biodynamics. *International Journal of Vehicle Performance, 6(*1), 1-51.

Rakheja, S., Kordestani, A. & Marcotte, P. (2011). Evaluation of whole-body vibration exposure of operators of soil compactors (Report No. R-709). Montreal, QC: IRSST.

Rakheja, S., Mandapuram, S. & Dong, R. G. (2008). Energy absorption of seated occupants exposed to horizontal vibration and role of back support condition. *Industrial Health*, *46*(6), 550-566.

Ranganathan, R. & Sriram, K. (1994) *Development of a PC-based software for analysis of off-road vehicle seat suspension* (Technical Paper No. 941677). S.I.: SAE.

Rebelle, J. (2004). Methodology to improve the performance of the end-stop buffers of suspension seats. *Vehicle System Dynamics*, *42*(4), 211-233.

Ropp, D. (2008). *U.S. Patent No. 7,338,126*. Washington, DC: U.S. Patent and Trademark Office.

Sankar, S. & Afonso, M. (1993). Design and testing of lateral seat suspension for off-road vehicles. *Journal of Terramechanics*, *30*(5), 371-393.

Sapiński, B. (2005a). Fuzzy control for MR damper in a driver's seat suspension. *Journal of Theoretical and Applied Mechanics*, *43*(1), 179-201.

Sapiński, B. (2005b). Real-time control for a magnetorheological shock absorber in a driver seat. *Journal of Theoretical and Applied Mechanics*, *43*(3), 631-653.

Segla, S. & Trišović, N. (2013). Modeling and optimization of passive seat suspension. *American Journal of Mechanical Engineering*, *1*(7), 407–411.

Shangguan, W. B., Shui, Y. & Rakheja, S. (2017). Kineto-dynamic design optimisation for vehicle-specific seat-suspension systems. *Vehicle System Dynamics*, *55*(11), 1643-1664.

Shirahatti, A., Prasad, P. S. S., Panzade, P. & Kulkarni, M. M. (2008). Optimal design of passenger car suspension for ride and road holding. *Journal of the Brazilian Society of Mechanical Sciences and Engineering*, *30*(1), 66-76.

Shui, Y. (2016). *The kineto-dynamic analysis and optimal design using suspension seat-human coupled model* (Master's thesis, South China University of Technology, Guangzhou, China).

Smith, S. D. (1997). Cushions and suspensions: Predicting their effects on the biodynamic responses of humans exposed to vertical vibration. *International Journal of Heavy Vehicle Systems*, *4*(2-4), 296-316.

Stein, G. J. (1997). A driver's seat with active suspension of electro-pneumatic type. *Journal of Vibration and Acoustics*, *119*(2), 230-235.

Stein, G. J. (2014). An electro-pneumatic active vibration control system for the driver's seat of agricultural tractors. *Archives of Acoustics*, *21*(3), 345-353.

Stein, G. J. & Ballo, I. (1991). Active vibration control system for the driver's seat for off-road vehicles. *Vehicle System Dynamics*, *20*(2), 57-78.

Stein, G. J. & Múčka, P. (2011). Study of simultaneous shock and vibration control by a fore-and-aft suspension system of a driver's seat. *International Journal of Industrial Ergonomics*, *41*(5), 520-529.

Stein, G. J., Múčka, P., Gunston, T. P. & Badura, S. (2008). Modelling and simulation of locomotive driver's seat vertical suspension vibration isolation system. *International Journal of Industrial Ergonomics*, *38*(5-6), 384-395.

Stein, G. J., Múčka, P. & Gunston, T. P. (2009). A study of locomotive driver's seat vertical suspension system with adjustable damper. *Vehicle System Dynamics*, *47*(3), 363-386.

Stein, G. J., Zahoranský, R., Gunston, T. P., Burström, L. & Meyer, L. (2008). Modelling and simulation of a fore-and-aft driver's seat suspension system with road excitation. *International Journal of Industrial Ergonomics*, *38*(5-6), 396-409.

Sun, S. S., Ning, D. H., Yang, J., Du, H., Zhang, S. W. & Li, W. H. (2016). A seat suspension with a rotary magnetorheological damper for heavy duty vehicles. *Smart Materials and Structures*, *25*(10).

Sun, S. S., Yang, J., Deng, H. X., Du, H., Li, W. H., Alici, G. & Nakano, M. (2015). Horizontal vibration reduction of a seat suspension using negative changing stiffness magnetorheological elastomer isolators. *International Journal of Vehicle Design*, *68*(1-3), 104-118.

Sun, W., Li, J., Zhao, Y. & Gao, H. (2011). Vibration control for active seat suspension systems via dynamic output feedback with limited frequency characteristic. *Mechatronics*, *21*(1), 250-260.

Tanovic, O. & Huseinbegovic, S. (2009). Hybrid fuzzy-neural network structure for vehicle seat vibration isolation. In *IEEE International Conference on Control and Automation, ICCA 2009: 9-11 Dec., Christchurch, New Zealand*, (p. 2354-2359). Piscataway, NJ: IEEE. doi: 10.1109/ICCA.2009.5410379

Tewari, V. K. & Prasad, N. (1999). Three-DOF modelling of tractor seat-operator system. *Journal of Terramechanics*, *36*(4), 207-219.

Thamsuwan, O., Blood, R. P., Ching, R. P., Boyle, L. & Johnson, P. W. (2013). Whole body vibration exposures in bus drivers: A comparison between a high-floor coach and a low-floor city bus. *International Journal of Industrial Ergonomics*, *43*(1), 9-17.

Tiemessen, I. J., Hulshof, C. T. & Frings-Dresen, M. H. (2007). An overview of strategies to reduce whole-body vibration exposure on drivers: A systematic review. *International Journal of Industrial Ergonomics*, *37*(3), 245-256.

Uys, P. E., Els, P. S. & Thoresson, M. (2007). Suspension settings for optimal ride comfort of off-road vehicles travelling on roads with different roughness and speeds. *Journal of Terramechanics*, *44*(2), 163-175.

Valero, B., Amirouche, F., Mayton, A. & Jobes, C. (2007). *Comparison of passive seat suspension with different configuration of seat pads and active seat suspension* (Technical Paper No. 2007-01-0350). S.I.: SAE.

Wan, Y. & Schimmels, J. M. (1997). Optimal seat suspension design based on minimum “simulated subjective response”. *Journal of Biomechanical Engineering*, *119*(4), 409-416.

Wan, Y. & Schimmels, J. M. (2003). Improved vibration isolating seat suspension designs based on position-dependent nonlinear stiffness and damping characteristics. *Journal of Dynamic Systems, Measurement, and Control*, *125*(3), 330-338. doi: 10.1115/1.1592189

Wang, C., Zhang, X., Guo, K., Lv, J. & Yang, Y. (2016). Hierarchical optimisation on scissor seat suspension characteristic and structure. *Vehicle System Dynamics*, *54*(11), 1538-1553.

Wang, R., Jing, H., Karimi, H. R. & Chen, N. (2015). Robust fault-tolerant H∞ control of active suspension systems with finite-frequency constraint. *Mechanical Systems and Signal Processing*, *62*, 341-355. doi: 10.1016/j.ymssp.2015.01.015

Wegscheid, E. (1994). Another look at skidder ride vibration. *Journal of Forest Engineering*, *5*(2), 21-32.

Wen, G., Zhang, Z., Yao, S., Yin, H., Chen, Z., Xu, H. & Ma, C. (2012). Vibration control for active seat suspension system based on projective chaos synchronisation. *International Journal of Vehicle Design*, *58*(1), 1-14.

Wu, X. & Griffin, M. J. (1996). Towards the standardization of a testing method for the end-stop impacts of suspension seats. *Journal of Sound and Vibration, 192*(1), 307-319.

Wu, X. & Griffin, M. J. (1997). A semi-active control policy to reduce the occurrence and severity of end-stop impacts in a suspension seat with an electrorheological fluid damper. *Journal of Sound and Vibration*, *203*(5), 781-793.

Wu, X. & Griffin, M. J. (1998). The influence of end-stop buffer characteristics on the severity of suspension seat end-stop impacts. *Journal of Sound and Vibration*, *215*(4), 989-996.

Wu, X., Rakheja, S. & Boileau, P.-É. (1999). *Dynamic performance of suspension seats under vehicular vibration and shock excitations* (Technical Paper No. 1999-01-1304). S.I.: SAE.

Yan, Z., Zhu, B., Li, X. & Wang, G. (2015). Modeling and analysis of static and dynamic characteristics of nonlinear seat suspension for off-road vehicles. *Shock and Vibration*, *2015,* 1-13. doi: 10.1155/2015/938205

Yu, J., Dong, X., Zhang, Z. & Chen, P. (2018). A novel scissor-type magnetorheological seat suspension system with self-sustainability. *Journal of Intelligent Material Systems and Structures*, *30*(5), 665-676. doi: 10.1177/1045389X17754256

Zhang, Z., Liu, X. & Huang, C. (2011). H∞ output feedback control for vehicle seat suspension subject to parameter perturbations. *Procedia Engineering*, *16*, 444-452.

Zhao, L., Zhou, C. & Yu, Y. (2016). Identification and optimisation of surrogate model parameters of the bus driver's seat suspension. *International Journal of Vehicle Systems Modelling and Testing*, *11*(4), 330-342.

Zhao, L., Zhou, C. & Yu, Y. (2017). ComfortiImprovement of a novel nonlinear suspension for a seat system based on field measurements. *Strojniski Vestnik/Journal of Mechanical Engineering*, *63*(2), 129-137.

Zhao, Q., Chen, Y. & Feng, H. (2011). Vehicle seat suspension vibration reduction based on CMAC and PID compound control. In *International Conference on Transportation, Mechanical, and*

Electrical Engineering (TMEE), 2011 Changchun, China, 16-18 December 2011 (p. 2230-2233). Piscataway, NJ: IEEE. doi: 10.1109/TMEE.2011.6199663

Zhao, Q., Yue, Y., Chen, Y. & Mi, L. (2011). LMI-Based H∞ control for the active seat suspension. In *International Conference on Transportation, Mechanical, and Electrical Engineering (TMEE), 2011 Changchun, China, 16-18 December 2011* (p. 2226-2229). Piscataway, NJ: IEEE. doi: 10.1109/TMEE.2011.6199662

Zhao, Y., Ou, Y., Zhang, L. & Gao, H. (2009). H∞ control of uncertain seat suspension systems subject to input delay and actuator saturation. In *Proceedings of the 48th IEEE Conference on Decision and Control, 2009, held jointly with the 2009 28th Chinese Control Conference: CDC/CCC 2009: 15-18 Dec., 2009, Shanghai, China* (p. 5164-5169). Piscataway, NJ: IEEE. doi: 10.1109/CDC.2009.5400116

Zhao, Y., Sun, W. & Gao, H. (2010). Robust control synthesis for seat suspension systems with actuator saturation and time-varying input delay. *Journal of Sound and Vibration*, *329*(21), 4335-4353.

Zhao, Y., Zhao, L. & Gao, H. (2010). Vibration control of seat suspension using H∞ reliable control. *Journal of Vibration and Control*, *16*(12), 1859-1879.

Zheng, E., Fan, Y., Zhu, R., Zhu, Y. & Xian, J. (2016). Prediction of the vibration characteristics for wheeled tractor with suspended driver seat including air spring and MR damper. *Journal of Mechanical Science and Technology*, *30*(9), 4143-4156.

Zhou, C., Zhao, L., Yu, Y., Yang, F. & Wang, S. (2018). Kineto-dynamic model and simplified model of x-type seat system with an integrated spring damper. *Shock and Vibration, 2018*. doi: doi.org/10.1155/2018/4962436

APPENDIX A:
SUMMARY OF SIMULATION MODELS OF SUSPENSION SEATS

Author (year)	Model description	Driver model	Parameter identification	Target vehicle(s)	Objective	Model verification (excitations)
Gouw et al. (1990)	Two-DOF with linear spring and nonlinear damper	Rigid mass	Laboratory characterization of components		Vibration performance assessment	Laboratory measured acceleration transmissibility (harmonic)
Rakheja et al. (1994)	Two-DOF with linear spring and inclined nonlinear damper	Rigid mass; single-DOF and two-DOF	Laboratory characterization of components		Vibration performance assessment	Laboratory measured acceleration transmissibility (harmonic)
Ranganathan and Sriram (1994)	Two-DOF with linear spring and inclined nonlinear damper	Rigid mass			Software for evaluating vibration performance	
Boileau et al. (1997)	Two-DOF with linear spring and nonlinear damper;	4-DOF	Human body model parameters identified from measured biodynamic response	Forestry skidder	Occupant-seat model development; parametric sensitivity analyses	Laboratory measured acceleration transmissibility (harmonic; random and shock)
Smith (1997)	Single-DOF linear	1-, 3- and 5-DOF	Human body model parameters identified from measured impedance		Prediction of vibration transmissibility of a seat with a human occupant	Comparisons of model predictions with measured data
Wan and Schimmels (1997)	Two-DOF	4-DOF	Human body model parameters from Patil and Palanichamy (1988)		Optimal suspension and cushion damping for enhanced comfort performance	
Prasad and Tewari (1995)	Linear single-DOF	Single-DOF	Laboratory characterization of components	Agricultural tractor	Optimal suspension parameters for 5th, 50th and 95th percentiles of body mass through minimizing the area under the acceleration transmissibility curve; parametric study	Laboratory measured acceleration transmissibility (harmonic)
Gunston et al. (2004)	Two-DOF with linear spring and inclined nonlinear damper and cushion	Rigid mass	Laboratory characterization of components		Bouc-Wen model for characterizing cushion hysteresis; SEAT and VDV	Laboratory measurements of acceleration response (transient excitations at 2.1, 2.25 and 3.25 Hz)

Author (year)	Model description	Driver model	Parameter identification	Target vehicle(s)	Objective	Model verification (excitations)
Hill and Dhingra (2003)	Linear single-DOF	3-DOF	Laboratory characterization of components	Small agricultural tractor	Design optimization to minimize RMS acceleration for different body mass	
Rakheja et al. (2003, 2004)	A generalized two-DOF nonlinear model	Rigid mass	Laboratory characterization of components of 3 different suspensions	Bus and earth-moving machinery	Parametric study considering bump-stop impacts and body-hop	Component model validations; seat mass acceleration measurements (Bus, EM1 and transient; high magnitude excitations)
Wan and Schimmels (2003)	Single-DOF with nonlinear stiffness and damping	4-DOF		Off-road haul trucks	Optimal stiffness and damping parameters for minimal SEAT and peak transmissibility (EM1)	
Hostens et al. (2004)	Nonlinear single-DOF with auxiliary air reservoir	Rigid mass		Agricultural Machines, bus, truck	Optimal suspension and elastic limiter parameters	Laboratory measured SEAT, acceleration transmissibility and relative displacement (harmonic; track measured excitations)
Rebelle (2004)	Nonlinear two-DOF model of behind-the-seat suspension	Rigid mass	Laboratory measured component properties	Forklift truck	Optimal parameters of elastic motion limiters	Laboratory measured VDV (transient excitations)
Choi and Wereley (2005)	Linear single-DOF with MR damper	4-DOF			Semi-active MR controller synthesis for harmonic, random and blast-induced shock	
Bouazara et al. (2006)	Nonlinear two-DOF with passive, semi-active and active dampers	Rigid mass		Urban bus	Optimal suspension damping for improved comfort	Laboratory measured acceleration, transmissibility and RMS acceleration
Duke and Goss (2007)	Nonlinear single-DOF	Rigid mass		Tractor	On-off suspension damping with dead-band (only marginal reduction was obtained)	Laboratory measured suspension relative motion (ramp input)
Valero et al. (2007)	Single-DOF with an active suspension	Multibody dynamic model		Mining vehicle	Absorbed power analyses with different seat padding and active suspension	Passive seat validation using reported data
Ma et al. (2008a)	Two-DOF with nonlinear suspension and cushion	Rigid mass	Component model parameters from reported studies	Urban bus, earth-moving machine, snowplow	Parametric study with end-stop impacts and body hop to identify desirable damping	Laboratory measured seat acceleration, SEAT, VDV (transient and vehicular excitations)

Author (year)	Model description	Driver model	Parameter identification	Target vehicle(s)	Objective	Model verification (excitations)
Stein, Mucka Gunston and Badura (2008)_and Stein Mucka and Gunston (2009)	Single- and two-DOF	Rigid mass		Locomotive	Effect of damping parameters on acceleration transmissibility, SEAT, RMS acceleration and displacement; and optimal damper parameters	Laboratory/field measured acceleration transmissibility, seat acceleration and displacement (harmonic, field)
Maciejewski et al. (2009)	Single-DOF nonlinear model with auxiliary air reservoir and air damping	Rigid mass		Earth-moving machine (EM1, EM3)	Design of auxiliary air chamber and damping restriction	Laboratory measured SEAT and peak relative displacement
Holtz and van Niekerk (2010)	Single-DOF nonlinear model with auxiliary air reservoir and air damping	Rigid mass			Effects of additional volume and flow area on vibration transmissibility and SEAT	Laboratory measured acceleration transmissibility (very poor agreement for a larger flow area)
Abbas et al. (2013)	Linear single-DOF model of the seat with 4-DOFpitch plane vehicle model	Four-DOF			Optimal seat and vehicle suspension design using Genetic algorithm	
Le and Ahn (2011)	Nonlinear single-DOF with softening stiffness effect of horizontal springs	Rigid mass	.		Negative stiffness can reduce effective stiffness and natural frequency under high magnitude excitations	Limited laboratory measurements of the negative stiffness concept under harmonic and broad band random vibration
Maciejewski et al. (2011)	Single-DOF nonlinear model of air suspension; additional air reservoir and air damping	Rigid mass	Laboratory measured component properties and curve fitting	Earth-moving machines	Optimal design parameters by minimizing SEAT and suspension travel under earth-moving vehicle excitations (EM1, EM5, EM6)	Laboratory measured acceleration transmissibility (very poor agreement for larger flow area)
Dong and Lu (2012)	Nonlinear two-DOF model with 3-stage stiffness and 2-stage damping properties	4-DOF		Wheeled tactical vehicles	Parametric study and optimization to minimize pelvis, chest and head acceleration under blast input	
Duke and Fow (2012)	Linear single-DOF model with on-off damper with a dead-band	Rigid mass			Effect of damping constant on RMS acceleration and suspension travel	Laboratory-measured suspension relative displacement

Author (year)	Model description	Driver model	Parameter identification	Target vehicle(s)	Objective	Model verification (excitations)
Gohari et al. (2012)	Linear suspension seat model with multi-DOF model of a pregnant woman	Biodynamic model		Bus	Optimal design modification of a seat suspension for minimizing RMS acceleration of the womb under 4 Hz harmonic excitation	
Ksiazek and Ziemianski (2012)	Single-DOF with passive suspension and active force generator	Single-DOF hybrid mechanical model based on apparent mass and comfort weighting			Synthesis of an optimal hybrid passive-active seat suspension under narrow band random excitation	
Wen et al. (2012)	Linear single-DOF with an active force generator				RMS acceleration	Field measured acceleration response with and without controller
Segla and Trišović (2013)	Single-DOF seat models with semi-active damper and vibration absorber	Rigid mass		Bucket wheel excavator	Optimal skyhook damping and absorber parameters for minimizing RMS acceleration and suspension travel	
Gohari and Tahmasebi (2014)	Single-DOF	3-DOF model reported by Kitazaki and Griffin (1998)			Optimal suspension parameters to minimize spine: acceleration (harmonic excitation)	
Jin et al. (2014)	Single-DOF nonlinear model with auxiliary air reservoir	Rigid mass		Highway truck	Develop and validate a model of the suspension seat in ADAMS	Field measured vibration response of the seat on a test track at speeds ranging from 35 to 70 km/h
Metered and Sika (2014)	Linear two-DOF with MR damper	Rigid mass		Truck	Seat acceleration and travel under bump and road excitations	
Yan et al. (2015)	Linear two-DOF	4-DOF			Influence of nonlinear suspension stiffness and damping on displacement transmissibility and hip displacement response of the model under 1 and 4 Hz harmonic excitations	

Author (year)	Model description	Driver model	Parameter identification	Target vehicle(s)	Objective	Model verification (excitations)
Zhang et al. (2011)	Single-DOF with suspension kinematics	Rigid mass			Validation of the simple single-DOF model with the ADAMS model	
Ning, Sung, Li et al. (2016)	Linear two-DOF with active force generated by a motor	Rigid mass			Seat acceleration, acceleration transmissibility (harmonic; random road profile)	Laboratory measured acceleration transmissibility (harmonic)
Nagarkar et al. (2016)	Single-DOF seat model with a two-DOF vehicle model	4-DOF			Optimal vehicle and seat suspension parameters to minimize RMS acceleration and VDV of the head and upper torso, and suspension travel	
Wang et al. (2016)	Single-DOF with suspension kinematics		Identified from measured response under sinusoidal excitation (20 mm amplitude)	Earth-moving machine (EM1)	Optimal suspension parameters of seat suspension and structure to minimize SEAT and suspension travel under white noise and EM1 excitation	
Zhao, Sun and Gao (2010)	Linear single-DOF model of seat with three-DOF cab model	Rigid mass		Truck	Optimal collaborative cab and seat suspension damping to minimize frequency-weighted RMS acceleration under measured vibration of a truck	Field measured cab and seat vibration
Zhao et al. (2016)	Two-DOF		Identified from field measured bus floor and seat acceleration	Bus	Optimal suspension damping to minimize RMS acceleration	Laboratory measured RMS acceleration
Zheng, Fan, Zhu, Zhu and Xian (2016)	Single-DOF model of seat suspension with MR damper and linear tractor model	Rigid mass	Laboratory measured properties of vehicle suspension and tire	Agricultural tractor	Effects of forward speed, damping and MR damper current on RMS acceleration of the seat and cab under agricultural test track vibration (ISO 5007:2003)	

Author (year)	Model description	Driver model	Parameter identification	Target vehicle(s)	Objective	Model verification (excitations)
Shangguan et al. (2017)	Kineto-dynamic model and nonlinear 2-DOF model	Rigid mass	Laboratory measured component properties	Earth-moving vehicles	Vehicle-specific optimal design to minimize RMS acceleration and VDV (EM1, EM4, EM6 and EM9 together with transient)	Laboratory-measured acceleration transmissibility of baseline suspension model
Zhao et al. (2017)	Nonlinear single-DOF	Rigid mass		Truck	Suspension performance in terms of RMS acceleration on highway and gravel roads	Field measured seat acceleration on highway and a gravel road

APPENDIX B: FEATURES OF COMMERCIAL SEATS

B.I Suspension characteristics

Manufacturer	Model	Application or equipment (if mentioned)	System type : P: passive SA: semi-active A: active	Axis of vibration attenuation	Fore-aft isolator: ***fixed*** or ***adjustable*** damping (*NA: not available*)
GRAMMER	MSG83 Series	agricultural, turf care	P	x- and z-	fixed
	MSG283 Series	agricultural, turf care, material handling	P	x- and z-	fixed
	MSG85 Series	agricultural, turf care, construction, material handling	P	x-, y- and z-	fixed
	MSG87 Series	agricultural, construction, material handling	P	x-, y- and z-	fixed
	MSG93 Series	agricultural, turf care	P	x- and z-	fixed
	MSG95 Series	agricultural, turf care, construction, material handling	P	x-, y- and z-	fixed
	MSG97 Series	agricultural, construction	P	x-, y- and z-	fixed
	B12	turf care, construction	P	z-	NA
	MSG65 Series	turf care, construction, material handling	P	x- and z-	fixed
	MSG75 Series	turf care, construction, material handling	P	x- and z-	fixed
	MSG20	construction, material handling	P	z-	NA
	Kingman	on-road truck	P	x- and z-	fixed
KAB	81/E1&E6	agricultural, construction/large size agricultural machines, rough terrain forklift, telehandler, small road sweeper, mining	P	x- and z-	adjustable
	Air 15/U4 (AIRMASTER)	agricultural/all size machines	P	z-	NA
	Air 85/E1/E6	agricultural/all size machines, rough terrain forklift, telehandler, mining	P	x- and z-	adjustable
	Invictus Air 85/K6	agricultural/all size machines	P	x- and z-	adjustable
	SCIOX Base & SCIOX Comfort	agricultural/all size machines	P	x- and z-	base: NA; comfort: adjustable
	SCIOX Premium+ / Super / Super High	agricultural/all size machines	P	x- and z-	adjustable
	XH2/U4 Fieldmaster	agricultural/all size machines, mini-excavator, mining	P	z-	NA
	Air 15/E1 & E6	agricultural/medium & small size machines, rough terrain forklift, telehandler, small road sweeper	P	E1: x- and z- ; E6: z-	fixed

Manufacturer	Model	Application or equipment (if mentioned)	System type : P: passive SA: semi-active A: active	Axis of vibration attenuation	Fore-aft isolator: ***fixed*** or ***adjustable*** damping (*NA: not available*)
KAB	XH2/P2* & XL2/U1 Kabmaster	agricultural/medium & small size machines	P	z-	NA
	115	construction/compact-wheel loader & dozer	P	z-	NA
	515/525	construction/compact-wheel loader, dozer, excavator, wheel loader, mining	P	z-	NA
	83/E1	construction/backhoe loader	P	z-	NA
	61/K1/K4	construction/compact-wheel loader, dozer, dump truck, excavator, crane, wheel loader, mining	P	x- and z-	adjustable
	811/1	construction/compact-wheel loader, dozer, excavator, telehandler, mining	P	x- and z-	adjustable
	Air 65/K1/K4	construction/compact-wheel loader, dozer, dump truck, excavator, wheel loader, mining	P	x- and z-	adjustable
	Sentinel Air & Sentinel Mechanical	construction/compact-wheel loader, dozer, mini-excavator, telehandler, wheel loader, electric forklift	P	z-	NA
	21/T1	construction/compactor, forklift	P	z-	NA
	411	construction/dozer, excavator, crane, wheel loader, small road sweeper, heavy truck, van, mining	P	z-	optional
	Air 714, 714B	construction/crane, heavy truck & bus	P	z-	NA
	834K	construction/dozer	P	z-	NA
	Air 555	construction/dozer, dump truck, wheel loader, mining	P	z-	NA
	514C & 524C	construction/dump truck, mining	P	z-	NA
	Air 554	construction/dump truck	P	z-	NA
	414 & 414B	construction/excavator, van, bus (double deck)	P	x- and z-	fixed
	116	construction/mini-excavator, rough terrain forklift, telehandler, mining	P	x- and z-	fixed
	Air 156	construction/mini-excavator	P	x- and z-	NA
	11/E6	construction/rough terrain forklift, telehandler	P	z-	NA
	816	construction/rough terrain forklift, telehandler	P	x- and z-	adjustable
	Air 856	construction/rough terrain forklift	P	x- and z-	adjustable
	Air 151	construction/telehandler, mining	P	z-	NA
	Air 25/E1/T1	forklift/electrical, small road sweeper	P	z-	NA

Manufacturer	Model	Application or equipment (if mentioned)	System type : P: passive SA: semi-active A: active	Axis of vibration attenuation	Fore-aft isolator: ***fixed*** or ***adjustable*** damping (*NA: not available*)
KAB	Air 554B	construction/dump truck	P	z-	NA
	Air 559 ACS	heavy truck & bus	P	z-	NA
	Air 712	heavy truck & bus	P	z-	NA
	GSX Base, GSX Comfort	heavy truck & bus	P	z-	NA
	Air 711	heavy truck & bus	P	x- and z-	fixed
	Air 71/E1	small road sweeper	P	z-	NA
	11/F1 & 11/P1	underground mining	P	z-	NA
SEARS SEATING	608/708 630/730	agriculture, turfcare	P	z-	NA
	908/1408 930/1430	agriculture, turfcare	P	908/1408: z- ; 930/1430:z-	NA
	3008	agriculture, turfcare	P	x-, y- and z-	optional
	3030	agriculture, turfcare	P	x-, y- and z-	optional
	3045	agriculture, turfcare	P	x- and z-	fixed
	D3055	agriculture, turfcare	P	x- and z-	fixed
	5510	agriculture, turfcare	P	z-	fixed
	5545	agriculture, turfcare	P	x- and z-	fixed
	D5575	agriculture, turfcare	P	x- and z-	fixed
	D5585	agriculture, turfcare	P	x-, y- and z-	fixed
	D5590	agriculture, turfcare	P	x-, y- and z-	fixed
	D5595	agriculture, turfcare	SA	x-, y- and z-	fixed
	D5580 VIS	agriculture, turfcare	P	x-, y- and z-	fixed
	8510	agriculture, turfcare	P	x- and z-	fixed
	D8575	agriculture, turfcare	P	x- and z-	fixed
	D8595	agriculture, turfcare	P	x- and z-	fixed
	5520	construction, mining, quarrying	P	z-	NA
	D5560 / 70	construction, mining, quarrying	P	x-, y- and z-	fixed
	7008/7030/7050	construction, mining, quarrying	P	7008/7050: x-, y- and z- ; 7030: y- and z-	fixed

Manufacturer	Model	Application or equipment (if mentioned)	System type : P: passive SA: semi-active A: active	Axis of vibration attenuation	Fore-aft isolator: ***fixed*** or ***adjustable*** damping (***NA: not available***)
SEARS SEATING	7010/7020/7050	construction, mining, quarrying	P	7010/7020: y- and z-; 7060: x-, y- and z-	7010/7020: NA; 7060: fixed
	8550	construction, mining, quarrying	P	x-, y- and z-	fixed
	8560	construction, mining, quarrying	P	x-, y- and z-	fixed
	8589 3P/ 8589/99 4P	construction, mining, quarrying	P, SA	y- and z-	NA
	1407	material handling, industrial	P	z-	NA
	1502/1602	material handling, industrial	P	z-	NA
	1715/1815	material handling, industrial	P	z-	NA
BOSE	Bose Ride® system II - English	construction delivery truck, heavy duty truck (on-highway truck), transit bus, terminal tractor	A	z-	fixed
NATIONAL SEATING	CAPTAIN; CORSAIR	heavy duty truck	P	z-	NA
	COMMODORE; ADMIRAL/CT	heavy duty truck	P	z-	NA
	ENSIGN; REFUSE SEAT	heavy & medium duty truck	P	z-	NA
	CAPTAIN LO; ENSIGN LO	medium duty truck	P	z-	NA
	ROUTEMASTER 350/310	school bus	P	z-	NA
	ROUTEMASTER 640	school bus	P	z-	NA
ISRINGHAUSEN	6000CN/SK	construction	P	z-	NA
	6500CN	construction	P	z-	NA
	ISRI® 6030/ 880 NTS	construction	P	z-	NA
	ISRI® 6830 KA/880 NTS; KM	construction	P	x- and z-	fixed

Manufacturer	Model	Application or equipment (if mentioned)	System type : P: passive SA: semi-active A: active	Axis of vibration attenuation	Fore-aft isolator: ***fixed*** or ***adjustable*** damping (*NA: not available*)
ISRINGHAUSEN	ISRI® 6830KM/870 NTS REGULAR; COMPACT	construction	P	z-	regular: NA; compact: NA - 17 cm
	ISRI® 6830KM/875	construction	P	x- and z-	fixed
	6800 Bus Seats	bus	P	z-	fixed
	ISRI® 6832/872 NTS	bus	P	z-	info NA
	Premium 6860/881	truck	P	z-	fixed
	Deluxe 6860/880	truck	P	z-	fixed
	Comfort 6860/880	truck	P	z-	NA
	6800 Premium LX Seat	truck	P	z-	fixed
RECARO	ERGO M; ERGO MC II	bus, off-road, rail	P	z-	NA
	ERGO B	metro	P	z-	NA
BAULTAR	3300 /3500	rail	P	info NA	info NA
	4000	rail, metro	P	info NA	info NA
	5000	urban bus	P	info NA	info NA
	Siti	rail (European market)	P	info NA	info NA
KNOEDLER	Air Chief	on-highway truck - class 8	P	x- and z-	fixed
	Falcon	on-highway truck - class 8	P	x- and z-	fixed
	Harrier	on-highway truck - class 8	P	x- and z-	fixed
	Ex Lo Static Power Seat	on-highway truck - classes 5-7	P	z-	fixed
	Extreme Lowrider	on-highway truck - classes 5-7	P	z-	fixed
	Super Ultra Compact	on-highway truck - classes 5-7, off-highway	P	z-	fixed
	Ultra Compact	on-highway truck - classes 5-7	P	z-	fixed
	Mechanical Chief	off-highway	P	z-	fixed
	Mechanical Seat 092	off-highway	P	z-	fixed
	ABTS Bus Chief	bus	P	x- and z-	fixed
	Bus Chief	bus	P	x- and z-	fixed

Manufacturer	Model	Application or equipment (if mentioned)	System type : P: passive SA: semi-active A: active	Axis of vibration attenuation	Fore-aft isolator: ***fixed*** or ***adjustable*** damping (*NA: not available*)
USSC	Q Series	public transit, rail	P	z-	fixed
	G2ELP-P1A/ Evolution	public transit, rail	A	z-	adjustable
	9000 Series	public transit, rail	P	z-	fixed
	LX SERIES	public transit, rail	P	z-	fixed
	EVOLUTION G2A	public transit, rail	P	z-	fixed
	9009/ 9010	rail	P	z-	fixed
	G2M	rail	P	z-	fixed
AMOBI	CREST-AIR	urban transport, passenger bus, harvester, excavator, road transport and work in a sitting position	P	z-	fixed
	HANDY	harvester, excavator, road transport and work in a sitting position	P	z-	fixed
	ROCK	underground loader	P	z-	fixed

B.II Ergonomic features of the seats

Manufacturer	Model	1. Lumbar support: M: mechanical P: pneumatic NA: not applicable	2. Seat cushion angle/ depth distance adjustment (if mentioned) NA: not available	3. Back rest (angle; if mentioned)/ arm rest (if mentioned) adjustment NA: not available	4. Swivel (angle) NA: not available
GRAMMER	MSG83 Series	M	fixed	adjustable/adjustable	fixed
	MSG283 Series	M	fixed	adjustable/adjustable	fixed
	MSG85 Series	M	fixed	adjustable/adjustable	available (360^0)
	MSG87 Series	M	adjustable	adjustable/adjustable	available (360^0)
	MSG93 Series	M	fixed	adjustable/adjustable	available (360^0)
	MSG95 Series	M or P	adjustable	adjustable/adjustable	available (360^0)
	MSG97 Series	M or P	adjustable	adjustable (-10^0 to $+34^0$)	available (360^0)
	B12	NA	fixed/adjustable	adjustable	fixed
	MSG65 Series	M	fixed	adjustable (-5^0 to $+30^0$)/adjustable	available (170° right to 0° left)
	MSG75 Series	M	fixed	adjustable (-5^0 to $+30^0$)/adjustable	available (170° right to 0° left)
	MSG20	M	fixed	fixed	Fixed
	Kingman	P	adjustable (9 settings)	adjustable/adjustable	available (180°)
KAB	81/E1&E6	M	fixed	adjustable/adjustable	available (20°)
	Air 15/U4 (AIRMASTER)	NA	fixed	info NA/adjustable	fixed
	Air 85/E1/E6	M	fixed	info NA/adjustable	available (0-10-20^0)
	Invictus Air 85/K6	M or P	adjustable	info NA/adjustable	available (0-10-20^0)
	SCIOX Base & SCIOX Comfort	NA	base: fixed; comfort: adjustable (0 to 8^0) – 3 positions & adjustable (5 cm)	adjustable/adjustable	base: fixed; comfort: available (±20^0)
	SCIOX Premium+ / Super / Super High	Premium: P; Super: M	adjustable/adjustable	adjustable/adjustable	available (±20^0)
	XH2/U4 Fieldmaster	NA	fixed	info NA/adjustable	fixed
	Air 15/E1 & E6	M	fixed	adjustable/adjustable	available (±20^0)

Manufacturer	Model	1. Lumbar support: M: mechanical P: pneumatic NA: not applicable	2. Seat cushion angle/ depth distance adjustment (if mentioned) NA: not available	3. Back rest (angle; if mentioned)/ arm rest (if mentioned) adjustment NA: not available	4. Swivel (angle) NA: not available
KAB	XH2/P2* & XL2/U1 Kabmaster	NA	fixed	fixed/info NA	fixed
	115	M	fixed	info NA/adjustable	available (180°)
	515/525	M	fixed	adjustable/adjustable	available (180°)
	83/E1	M	fixed	adjustable/adjustable	info NA
	61/K1/K4	M	adjustable (7.5^0)/adjustable (5 cm)	adjustable/adjustable	info NA
	811/1	M	fixed	adjustable (-72^0 to +66^0)/optional	info NA
	Air 65/K1/K4	M	adjustable (7.5^0)/adjustable	adjustable/adjustable	info NA
	Sentinel Air & Sentinel Mechanical	info not available	fixed	adjustable/optional	info NA
	21/T1	M	fixed	adjustable/optional	info NA
	411	M	adjustable (7.5^0)	adjustable (info NA)/adjustable	info NA
	Air 714, 714B	M	adjustable/fixed	adjustable (info NA)/adjustable	fixed
	834K	info not available	fixed	adjustable (-72^0 to +66^0)/optional	available
	Air 555	M	adjustable/fixed	adjustable/optional	available
	514C & 524C	P	adjustable/fixed	adjustable/optional	available
	Air 554	M	adjustable/fixed	adjustable/optional	available (180°)
	414 & 414B	P	adjustable/fixed	adjustable/optional	fixed
	116	M	fixed	adjustable/adjustable	fixed
	Air 156	M	fixed	adjustable/adjustable	fixed
	Air 151	info not available	fixed	adjustable (-72^0 to +66^0)/adjustable & foldable	fixed

Manufacturer	Model	1. Lumbar support: M: mechanical P: pneumatic NA: not applicable	2. Seat cushion angle/ depth distance adjustment (if mentioned) NA: not available	3. Back rest (angle; if mentioned)/ arm rest (if mentioned) adjustment NA: not available	4. Swivel (angle) NA: not available
KAB	Air 25/E1/T1	info not available	fixed	adjustable (-72^0 to +66^0)/foldable	fixed
	Air 554B	P	fixed	adjustable/foldable	fixed
	Air 559 ACS	P	fixed	adjustable/adjustable	available (180°)
	Air 712	P	fixed	adjustable (-70^0 to +65^0)/foldable & adjustable	fixed
	GSX Base, GSX Comfort	Base: M; Comfort: P	adjustable	adjustable	fixed
	Air 711	M	adjustable/fixed	adjustable	fixed
	Air 71/E1	M	fixed	adjustable	fixed
	11/F1 & 11/P1	NA	fixed	fixed/info NA	fixed
SEARS SEATING	608/708 630/730	NA	fixed	fixed/adjustable	fixed
	908/1408 930/1430	908/1408: available; 930/1430: NA	fixed	fixed/adjustable	fixed
	3008	M	fixed/adjustable	adjustable/adjustable	available (20°-20°)
	3030	M	fixed	adjustable/adjustable	Info NA
	3045	NA	fixed/adjustable	adjustable/adjustable	available
	D3055	NA	fixed/adjustable	adjustable/adjustable	available
	5510	M	fixed	adjustable/adjustable	Info NA
	5545	NA	fixed	adjustable/adjustable	available
	D5575	P	adjustable	adjustable/adjustable	available (24°-24°)
	D5585	P	adjustable	adjustable/adjustable	available (24°-24°)
	D5590	P	21° & 6 cm	adjustable/adjustable	available (24°-24°)
	D5595	P	adjustable	adjustable/adjustable	available (24°-24°)
	D5580 VIS	P	21° & 6 cm	adjustable (55^0)/adjustable	available (24°-24°)

Manufacturer	Model	1. Lumbar support: M: mechanical P: pneumatic NA: not applicable	2. Seat cushion angle/ depth distance adjustment (if mentioned) NA: not available	3. Back rest (angle; if mentioned)/ arm rest (if mentioned) adjustment NA: not available	4. Swivel (angle) NA: not available
SEARS SEATING	8510	M	fixed	adjustable	available (24°-24°)
	D8575	P	adjustable	adjustable	available (24°-24°)
	D8595	P	21° & 6 cm	adjustable	available (21°-21°)
	5520	M	fixed	adjustable	available (0°-180°)
	D5592/5587	info not available	info NA	info NA	info NA
	D5560 / 70	M	adjustable (21°)/ adjustable (6 cm)	adjustable/adjustable	fixed
	7008/7030/7050	M	fixed	adjustable/adjustable	fixed
	7010/7020/7060	M	adjustable/adjustable	adjustable/adjustable	fixed
	8550	M	fixed	adjustable/adjustable	fixed
	8560	P	21° & 6 cm	adjustable/adjustable	fixed
	8589 3P/ 8589/99 4P	P	21° & 6 cm	adjustable (55^0)/adjustable	fixed
	1407	NA	fixed	fixed/adjustable	fixed
	1502/1602	1502: NA; 1602: P or M	fixed	adjustable/adjustable	fixed
	1715/1815	1715: NA; 1815: M or P	fixed	adjustable (12.5^0 to -12.5^0)/adjustable	fixed
BOSE	Bose Ride® system II - English	P	3 positions/3 positions	adjustable (12.5^0 to -12.5^0)/adjustable	fixed
NATIONAL SEATING	CAPTAIN; CORSAIR	P	adjustable/adjustable	adjustable (17^0)/foldable	CAPTAIN: info NA; CORSAIR: optional
	COMMODORE; ADMIRAL/CT	P	adjustable/adjustable	adjustable (17^0)/foldable	COMMODORE: optional; ADMIRAL: available
	ENSIGN; REFUSE SEAT	ENSIGN: NA, REFUSE: P	info NA	adjustable (17^0)/foldable	info NA
	CAPTAIN LO; ENSIGN LO	P	CAPTAIN LO: adjustable; ENSIGN LO: fixed	adjustable (17^0)/foldable	info NA
	ROUTEMASTER 350/310	350: P; 310: M	350: adjustable; 310: fixed	adjustable (17^0)/foldable	info NA
	ROUTEMASTER 640	P	adjustable/adjustable	adjustable (17^0)/foldable	info NA

Manufacturer	Model	1. Lumbar support: M: mechanical P: pneumatic NA: not applicable	2. Seat cushion angle/ depth distance adjustment (if mentioned) NA: not available	3. Back rest (angle; if mentioned)/ arm rest (if mentioned) adjustment NA: not available	4. Swivel (angle) NA: not available
NATIONAL SEATING	6000CN/SK	M	adjustable/adjustable	adjustable/foldable	info NA
	6500CN	P	adjustable/adjustable	adjustable (12^0)/foldable	info NA
	ISRI® 6030/ 880 NTS	P	adjustable/adjustable	adjustable ($+12^0$ to -40^0)/foldable	available
	ISRI® 6830 KA/880 NTS; KM	P	adjustable/adjustable	adjustable ($+12^0$ to -40^0)/foldable	available
	ISRI® 6830KM/870 NTS	P	adjustable/adjustable	adjustable ($+12^0$ to -40^0)/ foldable	info NA
	ISRI® 6830KM/875	P	adjustable/adjustable	adjustable	info NA
	6800 Bus Seats	P	adjustable/adjustable	adjustable (40^0)/foldable	info NA
	ISRI® 6832/872 NTS	P	adjustable/adjustable	adjustable	info NA
	Premium 6860/881	P	adjustable/adjustable	adjustable	info NA
	Deluxe 6860/880	P	adjustable/adjustable	adjustable;optional	info NA
	Comfort 6860/880	P	fixed	adjustable/info NA	info NA
	6800 Premium LX Seat	P	adjustable/adjustable	adjustable (40^0)/foldable & adjustable	info NA
RECARO	ERGO M; ERGO MC II	P	adjustable	adjustable (25^0)/adjustable	info NA
	ERGO B	P	adjustable	adjustable/info NA	optional
BAULTAR	3300 /3500	info not available	info NA	info NA	info NA
	4000	info not available	info NA	info NA	info NA
	5000	info not available	info NA	info NA	info NA
	Siti	info not available	info NA	info NA	info NA

Manufacturer	Model	1. Lumbar support: M: mechanical P: pneumatic NA: not applicable	2. Seat cushion angle/ depth distance adjustment (if mentioned) NA: not available	3. Back rest (angle; if mentioned)/ arm rest (if mentioned) adjustment NA: not available	4. Swivel (angle) NA: not available
KNOEDLER	Air Chief	P	adjustable	adjustable/adjustable	optional
	Falcon	M	info NA	adjustable/adjustable	info NA
	Harrier	P	adjustable	adjustable/adjustable	info NA
	Ex Lo Static Power Seat	info not available	adjustable	adjustable/adjustable	info NA
	Extreme Lowrider	P	adjustable	adjustable/adjustable	info NA
	Super Ultra Compact	M	adjustable	adjustable/info NA	info NA
	Ultra Compact	M	adjustable	adjustable/info NA	info NA
	Mechanical Chief	M	adjustable	adjustable/info NA	info NA
	Mechanical Seat 092	info not available	fixed	adjustable/fixed	info NA
	ABTS Bus Chief	M or P	adjustable	adjustable/fixed	info NA
	Bus Chief	P	adjustable	adjustable/fixed	info NA
USSC	Q Series	info not available	info NA	adjustable/adjustable	info NA
	G2ELP-P1A	M	info NA	adjustable/info NA	info NA
	9000 Series	M	adjustable	adjustable/info NA	info NA
	LX SERIES	P	info NA	adjustable/info NA	info NA
	9010/9009	M	info NA	adjustable/adjustable	info NA
	G2M	M	adjustable	adjustable/adjustable	optional
AMOBI	CREST-AIR	P	info NA	adjustable/adjustable	info NA
	HANDY	P	info NA	adjustable/adjustable	info NA
	ROCK	P	info NA	adjustable/adjustable	info NA