IOSR Journal of Mechanical and Civil Engineering (IOSR-JMCE)
e-ISSN: 2278-1684,p-ISSN: 2320-334X, Volume 17, Issue 6 Ser. II (Nov. – Dec. 2020), PP 62-69
www.iosrjournals.org

A Review Simulation s-DOF Double Acting Force Suspension Seat Adopted Sarrus Linkage

R. Agam A.[1], Irsyadi Yani[1], AmirArifin[1]
[1]*(Mechanical Department, University Sriwijaya, Indonesia)*

Abstract:*Suspension seats are used to isolate passengers from off-road vehicles, on-road vehicles, trains, ships and aircraft from vibrations. Transmissibility factor can be calculated by the percentage ratio of the frequency-weighted accelerationon the chair surface to the acceleration to the frequency on the floor. Two standards apply slightly different frequency weights to the vertical vibration of the seat. For most vehicles, the estimated magnitude of the vibration is higher than 0.47* m/s^2 *RMS, which corresponds to the lower limit of the health guidance warning zone of 8 hours exposure in the period. Twenty-four hours, according to ISO 2631-1 standard.Conducting a simulation review study is necessary to determine the appropriate calculation method so that the results are not far from the actual real world.*

Background*: The use of simulations can help reduce research costs and allow it to be repeated under similar conditions. Another advantage of simulation is that it can be conditioned to reach the worst point and with the condition of passengers and vehicles according to emergency conditions. The simulation model is an imitation of a real world system and never actually mimics a real world system. Therefore, the model must be verified and validated to the extent necessary for the intended purpose or application of the model.*

Materials and Methods*: The simulation will use SolidWorks and Matlab software. Motion study on solidworks will be used to determine the motion geometry of the suspension frame and the dynamic response. Matlab programming will be used as a validator using mathematical modeling methods in the simulation produced by SolidWorks.*

Results*: The precise and easy method for verification and validation of the kinematic and kinetic simulation results of the suspension seat frame that adopts the sarrus mechanism is to use the slider-crank and passive suspension mathematical modeling and under condition following acceleration of ISO 2631-1.*

Key Word*:Passive suspension; Suspension seat; Dynamic vibration;Sarrus mechanism; Kinematic and Kinetic Slider-Crank.*

Date of Submission: 27-11-2020 Date of Acceptance: 11-12-2020

I. Introduction

Environmental factors such as vibration, noise, temperatureplay an essential role in the design or development of the transportation system. Factors like these can affect passenger comfort and hence the public acceptance of the transportation system. High-speed transport has the potential to experience more severe vibrations than those found in most operating systems today. Consequently, passenger acceptance is expected to become a more critical design factor and a compromise between "good travel" and the complexity of the transportation system and costs will take a more critical role in the design process.

Then vibration isolation can be realized using seat suspension. Suspension seats are used to isolate passengers from off-road vehicles, on-road vehicles, trains, ships and aircraft from vibrations.Transmissibility factor can be calculated as the percentage ratio of the frequency-weighted acceleration on the chair surface to the acceleration to the frequency on the floor to control chair vibration.Theinvestigations can be carried out in the fieldcrank, in laboratories, on real transport vehicles, and recently motion simulators mimicking real operating conditions have come into use. Implementing simulators helps reduce the cost of test procedures, enabling them to be repeated under the same conditions. Another advantage of the simulator is that the operating conditions can be modelled in the worst situations and the combination of loads acting on the means of transportation and passengers and emergency conditions can also be carried out.

Modeling simulation is an approach method with similarity of the actual event and the simulation never produces the actual event. therefore, a model should be validated andverified according to boundary conditions that are close to reality[1].This study focuses on review validated simulation method by mathematical modelling of the suspension seat adopting the Sarrus Linkage design and passive suspension with modifications that allow the shape of the force to be accepted in two directions, namely vertical and lateral at a non-linear maximum acceptable point.

II. Literature Review

There are four types of suspension seats: passive, semi-active[2], adaptive, and active[3]–[5]. The four types of suspension are variations of the mass-spring damping system that are usually ineffective for under acceleration1.0 m/s^2[6], [7]. Research on suspension seats has been done a lot with various methods and designs, one of which is using Vibro-isolation [8], [9], air suspension [10], magneto-rheological [11]–[13], and pneumatic muscle [8], [14].The horizontal seat had also been researched with lateral, fort and alt[15], [16] to make sure passenger is safe in multi-axis forces[17]. The measurement of the whole body of passenger during on vehicle was conducted to get valid data to improve suspension seat [18], measurement by project vibseat[19], and measurement transmissibility of an agricultural tractor seat[20].Many issues with passenger health have happened during on high-speed vehicle, for the example when a passenger on high speed craft or on speed-boat, this is already concerned by many researcher [21], [22]. The aircraft acceleration was the most effecting on mental health body[23], such as G-Force[24] during take-off, landing, and maneuver [25], [26].To produce suspension seat in effective way, we need to simplification the engineering design for isolated vibration seat from vertical and horizontal force, and the simulator is the lower cost before real testing vibration[27], by matlab dynamic simulation on single degree of freedom [28], and also for comparison with three different type of suspension of passive, semi active [29], and active suspension seat [30].

III. Methodology andDiscussion

To predict the response of a moving saddle and a passenger to the input of wave motion or windshear or road conditions, the passenger-seat model, as described,will be simulated using Solidworks and Matlab programs. The suspension seat frame design will be carried out first using Solidworks, then proper geometry, followed by using MatLab for dynamic response simulation. The data used by the author in this study were obtained from previous research and relevant literature, such as scientific articles and books.

Motion simulation is capable of importing time history data from tests. In this way, the movements of the existing mechanisms can be easily reproduced and thoroughly analyzed, including all combined reactions, inertia effects, power consumption and much more. Use an inexpensive computer model rather than a time-consuming and expensive test.

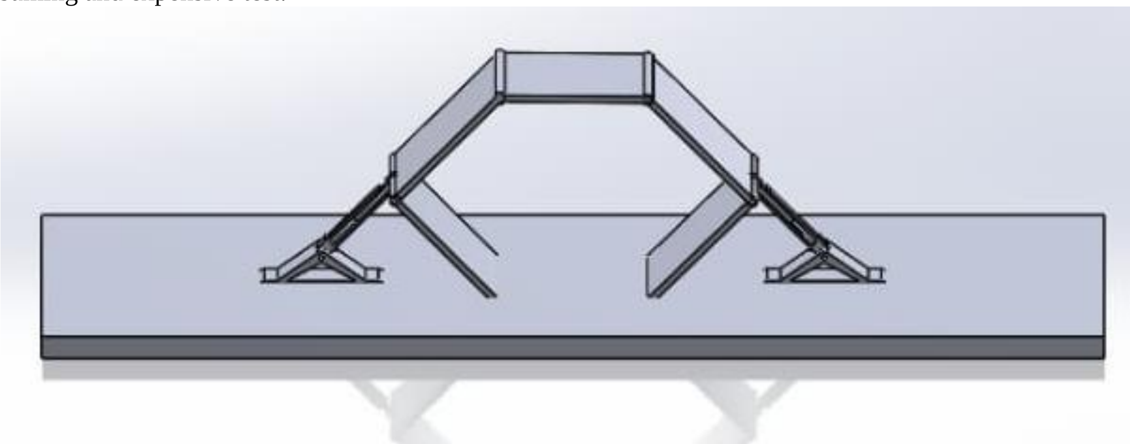

Fig 1. Suspension seat adopted sarrus linkage

Fig 1 is shown as a link for suspension seat, which was expected to absorb double-acting force vertical and lateral. The simulation will be compared with the mathematical method to make sure the result of the simulation was accurate and can be implemented into a prototype test.

Passive suspension

There are physical dampers themselves and equations covering motion and damper control. Hydraulic dampers are the most common type used in private cars. An attractive and relatively inexpensive design variant of the vibration shock absorber (damper) with the self-pumping levelling feature is featured in fig 1[31], [32].

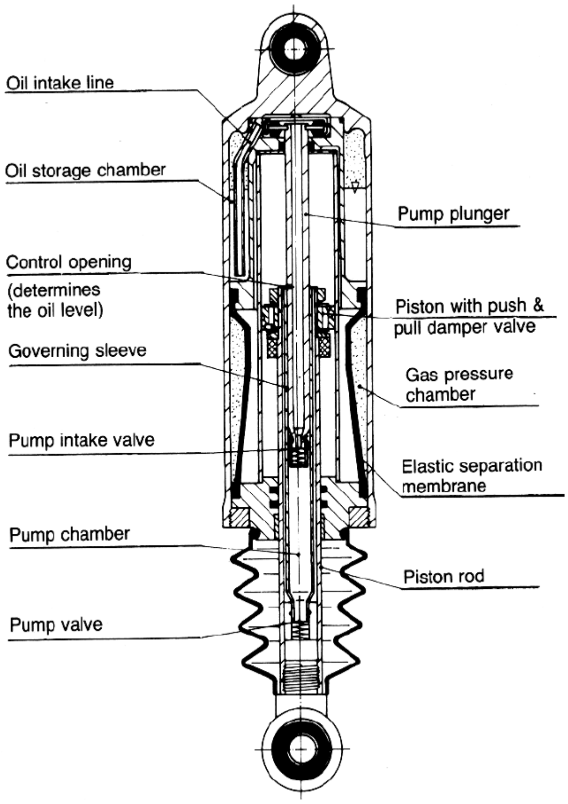

Fig 2. Passive suspension with self-levelling

This damper action exerts a force generated only at the relative speed of the damping element. However, modern research is exploring whether a damper can be given a non-linear response or a response determined by other factors.Concerning light vehicles, the acceleration and road hardness experienced by passengers increases along with decreasing weight, for the classic single-degree system using a passive damper, as illustrated in fig 2.

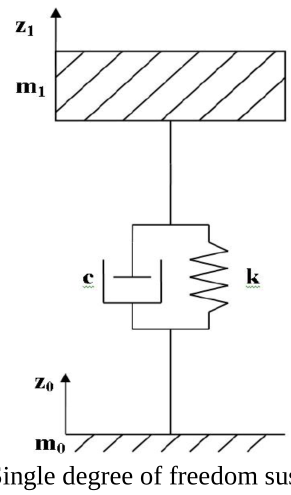

Fig 3. Single degree of freedom suspension

$$m\ddot{z} + c(\dot{z}_2 - \dot{z}_1) + k(z_2 - z_2) = 0 \quad (1)$$

Where m is the mass of the spring, $\ddot{z}$is the acceleration of the mass of the spring, c is the damping coefficient, $\dot{z}_2 - \dot{z}_1$ is the relative velocity of the mass of the spring and the ground surface, k is the spring constant and the relative displacement of the mass of the spring and the surface$(z_2 - z_2)$. Take the relationship between the damping coefficients:

$$c = 2\zeta\sqrt{mk} \quad (2)$$

Where C is the damping ratio, which is the ratio of the damper damping coefficient to the critical damping coefficient of $2m\sqrt{k/m}$. Then (2) can be substituted into equation (1) and then equation (1) can be rearranged into equation (3):

$$\ddot{z} = \frac{2C\sqrt{mk}(\dot{z}_2 - \dot{z}_1) + k(z_2 - z_2)}{m} \tag{3}$$

Looking at the relationship between damping and mass, this can be seen in (4):

$$\ddot{z}\ \alpha\ \frac{1}{\sqrt{m}} \tag{4}$$

From this relationship, it can be seen that the less the mass of the vehicle, the acceleration will increase so that the violence felt by passengers will increase.

The relation between Sarrus linkage and slider-crank

Since Sarrus "platform", link 3, is translated as 1DOF, each "leg" movement can be viewed as a slider-crank movement[33], [34].

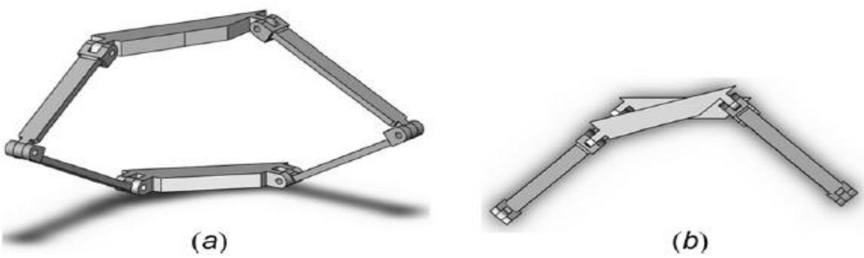

Fig 4. (a) Seat adopted sarrus linkage and (b) from the above overview

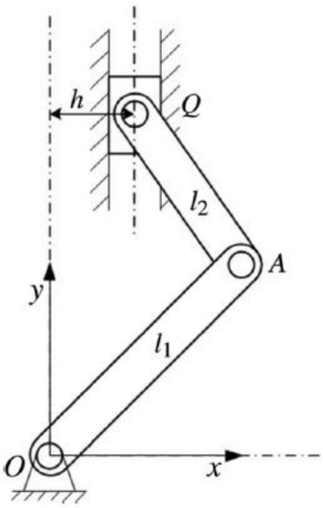

Fig 5. Slider crank mechanism

To get a precise picture in the kinematic and kinetic analysis of the slider-crank, the following is an example of another kinematic and kinetic pad application that can be used as shown in Figure 6, θ (crank angle to the X-axis) is chosen as the only degree of freedom. Hence, the position r_B, the linear velocity v_B, and the linear acceleration of the vector a_B point B in the X-Y coordinate system[35]–[37].

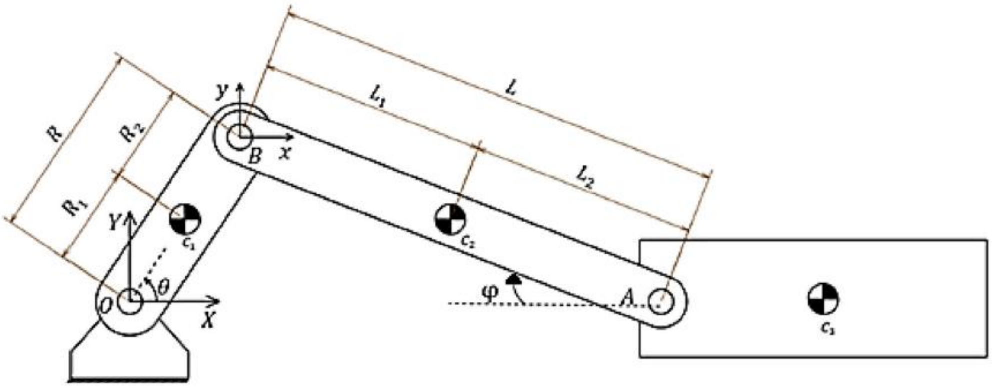

Fig 6. Kinematic slider-crank

The mobile coordinate system is used at point B to determine the angular velocity and angular acceleration of the connecting rod, since the position vectors, linear velocity, and linear acceleration are related to point A. A is a point on the slider; its velocity and acceleration have the only components. Nonzero in the X direction which can be obtained. The velocity and acceleration of point A can be calculated for any θ, the acceleration of the crankmass centre, and the connecting rod. The$C_1$and $C_2$are the centres of mass of the crank and connecting rods. It should be noted that it can have nonzero components in both the X and Y directions.

The free-body diagram is depicted in Figure 7. It should be noted that the sliders are assumed to be concentrated masses. It is displayed as a distributed mass for easy drawing of its free body diagram.Force P also contains two parts: P_1 due to Coulomb friction created by N_1 and P_2 developed by viscous damping on bearings.

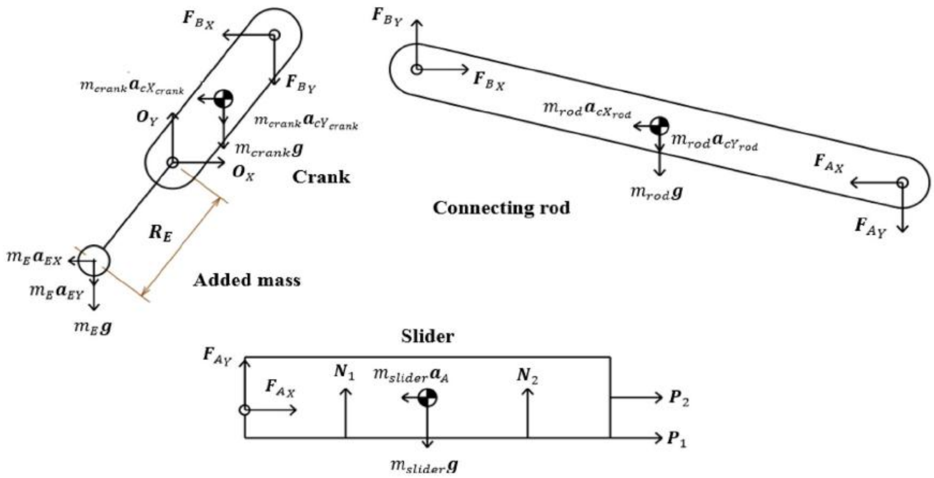

Fig 7. Free body diagram slider-crank

$$O_X - F_{B_X} = m_{crank} a_{cX_{crank}} \tag{5}$$
$$O_Y - F_{B_Y} - m_{crank} g = m_{crank} a_{cY_{crank}} \tag{6}$$
$$M_{B1} + M_O + T = I_{crank} \ddot{\theta} \tag{7}$$
$$F_{B_X} - F_{A_X} = m_{rod} a_{cX_{rod}} \tag{8}$$
$$F_{B_X} - F_{A_Y} - m_{rod} g = m_{rod} a_{cY_{rod}} \tag{9}$$
$$M_{B2} + M_A = I_{rod} \ddot{\varphi} \tag{10}$$
$$F_{A_X} + P = m_{slider} a_A \tag{11}$$
$$F_{A_X} + N - m_{slider} g = 0 \tag{12}$$

The magnitude of some of the forces appearing in the above equation is defined as follows: O_X and O_Y are the reaction forces of the O connection applied to the crank, F_{B_X} and F_{B_Y} the reaction force of the B connection on the crank and connecting rod, and F_{A_X} and F_{A_Y}Connection reaction forces A on the connecting rod and slider. The subscripts X and Y represent X and Y components, respectively. Then M_O is the moment O around c_1, M_{B1} moment F_B around c_1, M_{B2} moment F_B around c_2, and M_A moment F_A around c_2. M is the mass, and I is the moment of inertia of the mass for the centre of mass and g the acceleration due to gravity. Equations (5 - 12) form a set of 8 equations and nine unknowns. The magnitude and orientation of P or T must be determined to complete the set.

Discussion

Currently, there are two primary standards for evaluating vibrations to the human response to whole-body vibrations. There are some differences in measurement, evaluation and assessment procedures defined in the two standards. For example, BS 6841, recommends measuring four axes of vibration in a seat (front and rear, lateral and vertical vibrations). on the seat surface and front and rear vibrations in the backrest) Furthermore, incorporate this into the evaluation procedure before assessing the severity of the vibration. In ISO 2631, it is recommended that vibrations in multiple axes be measured, but the rating is based only on the most severe axis. The two standards apply slightly different frequency weights to the vertical vibration of the seat.For most vehicles, the estimated magnitude of the vibration is higher than 0.47 m / s¬2 RMS, which corresponds to the lower limit of the health guidance warning zone of 8 hours exposure in the period. Twenty-four hours, according to ISO 2631-1 standard.

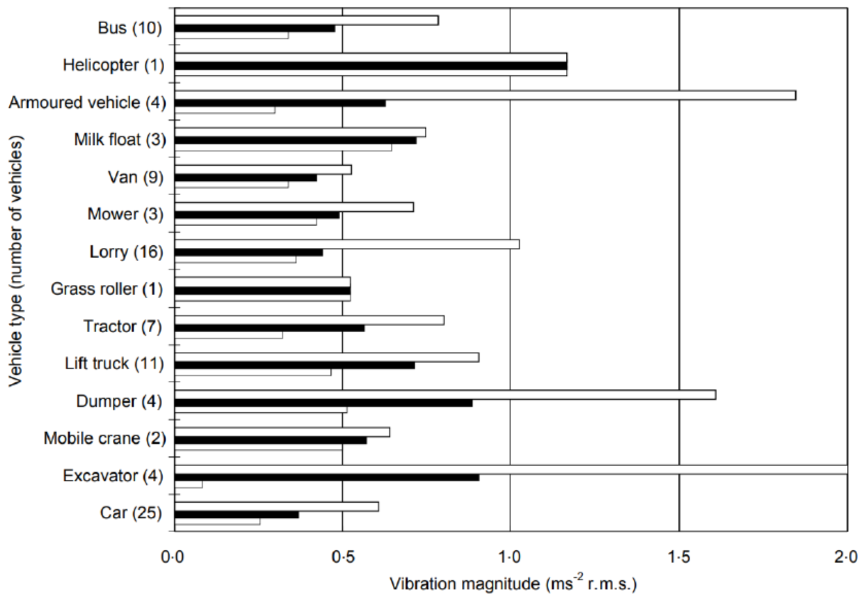

Fig 8. The median RMS acceleration value uses ISO 2631-1 and BS 6841 for different vehicles[4], [18]

Acute health effects refer to brief exposure to whole-body vibrations, which usually do not cause significant physiological changes in the human body, and most of the impacts can be eliminated by eliminating the vibration source.

Table no1: Uncomfortable reaction rate to vibration magnitudes (ISO 2631-1).

Comfortable level	The RMS acceleration range is frequency weighted$a$¬v (m/s2)
Extreme uncomfortable	> 2.0
Very uncomfortable	1.25 – 2.5
uncomfortable	0.8 – 1.6
Fairly uncomfortable	0.5 – 1.0
A little uncomfortable	0.315 – 0.63
Not uncomfortable	< 0.315

To determine the response of the suspension and passenger seats to given motion input, it requires specific physical characteristics of the seat, the start of the simulation, the location and name of the parameter configuration input file and the desired input motion file for the output file to be determined. Here is an accelerometer measurement of the floor surface of several vehicles based on measurements.

Table no2: measurement of vibration on a chair at constant speed.

Kendaraan	Velocity	Distance	Vertical(z) (m/s^2)	Lateral(y) (m/s^2)
Semi Trailer	70km/jam	100 m	0.35	0.12
Tractor	24 km/jam	100 m	5.14	1.31
Industrial loader	14 km/jam	100 m	3.08	1.06
Locomotive	90 km/jam	100 m	0.15	0.23

Table no3: The shock vibration of the fast boat, 35 knots[38].

Description	Peak Accelaration			Shock Pulse Duration		
	Longitudinal (g's)	Lateral (g's)	Vertical (g's)	Longitudinal (sec)	Lateral (sec)	Vertical (sec)
Max	10.4	2.84	7.13	0.037	0.201	0.346
Min	0.22	0.17	0.36	0.004	0.002	0.002
Average	1.43	0.86	2.99	0.012	0.018	0.033

While many measurements can be used to evaluate ride comfort, such as historical speed, acceleration and jerk time or in terms of RMS value, ISO 2631-1, much of the literature suggests acceleration magnitude with a suitable frequency-weighted filter to assess ride quality. Hence, the frequency-weighted RMS acceleration value is considered here as an essential quantity for analyzing the effects of whole-body vibrations in terms of comfort, health and perception, as suggested by the ISO 2631-1 standard.Equations (5 - 12) form a set of 8 equations and nine unknowns. The magnitude and orientation of P or T must be determined to complete the set.

IV. Conclusion

This review was intended to get a detailed understanding simulation study for simplification of double-acting movement with a single degree of freedom dynamic response of suspension seat adopted sarrus linkage.Result simulation of passive suspension seat was compared with mathematic modelling to ensure simulation on track before heading up into the prototype test. Kinematic and kinetic of the slider-crank method were used to get more easily calculation between geometry and dynamics of suspension seat. Standard measurement ISO 2631-1 as references to evaluate vibration passenger were enough for passenger vehicle afterthe simulation result.

References

[1] R. G. Sargent, "Verification and Validation of Simulation Models," *Proceedings - Winter Simulation Conference*, no. January 2011, pp. 166–183, 2010, doi: 10.1109/WSC.2010.5679166.

[2] k. S. R. & a. P. T. Ram Mohan Rao, G. Venkata Rao, "Analysis of Passive and Semi Active Controlled Suspension Systems for Ride Comfort in an Omnibus Passing Over a Speed Bump," *International Journal of Research and Reviews in Applied Sciences*, vol. 5, no. October, pp. 7–17, 2010.

[3] M. Gohari and M. Tahmasebi, "Active Off-Road Seat Suspension System Using Intelligent Active Force Control," *Journal of Low Frequency Noise Vibration and Active Control*, vol. 34, no. 4, pp. 475–490, 2015, doi: 10.1260/0263-0923.34.4.475.

[4] A. Alfadhli, "Active Seat Suspensions for Automotive Applications," p. 1, 2018.

[5] F. Braghin, F. Cheli, A. Facchinetti, and E. Sabbioni, "Design of an Active Seat Suspension for Agricultural Vehicles," *Conference Proceedings of the Society for Experimental Mechanics Series*, vol. 3, no. PART 2, pp. 1365–1374, 2011, doi: 10.1007/978-1-4419-9834-7_120.

[6] A. Wice, "Spatial Dynamic Modelling of High Speed Craft Suspension Seating," no. April, 2015.

[7] SAMY ALY HASSAN, "FUNDAMENTAL STUDIES OF PASSIVE, ACTIVE AND SEMI-ACTIVE AUTOMOTIVE SUSPENSION SYSTEMS," 1986.

[8] I. Maciejewski, T. Krzyzynski, and H. Meyer, "Modeling and Vibration Control of an Active Horizontal Seat Suspension with Pneumatic Muscles," *JVC/Journal of Vibration and Control*, vol. 24, no. 24, pp. 5938–5950, 2018, doi: 10.1177/1077546318763435.

[9] I. Maciejewski, T. Krzyzynski, L. Meyer, and H. Meyer, "Shaping the Vibro-Isolation Properties of Horizontal Seat Suspension," vol. 36, no. 3, pp. 203–213, 2017, doi: 10.1177/0263092317717586.

[10] I. Hostens, K. Deprez, and H. Ramon, "An Improved Design of Air Suspension for Seats of Mobile Agricultural Machines," *Journal of Sound and Vibration*, vol. 276, no. 1–2, pp. 141–156, 2004, doi: 10.1016/j.jsv.2003.07.018.

[11] H. J. Yao, J. Fu, M. Yu, and Y. X. Peng, "Semi-Active H∞ Control of Seat Suspension with MR Damper," *Journal of Physics: Conference Series*, vol. 412, no. 1, 2013, doi: 10.1088/1742-6596/412/1/012054.

[12] S. S. Sun *et al.*, "Horizontal Vibration Reduction of a Seat Suspension Using Negative Changing Stiffness Magnetorheological Elastomer Isolators," *International Journal of Vehicle Design*, vol. 68, no. 1–3, pp. 104–118, 2015, doi: 10.1504/IJVD.2015.071076.

[13] S. Eshkabilov, H. Jumaniyazov, and D. Riskaliev, "Simulation and Analysis of Passive vs. Magneto-Rheological Suspension and Seat Dampers," *Lecture Notes in Mechanical Engineering*, pp. 269–279, 2019, doi: 10.1007/978-3-319-93587-4_28.

[14] D. Sun, B. Yan, B. Han, Y. Song, and X. Zhang, "Vibration Characteristic Simulation of a Pneumatic Artificial Muscle Damping Seat," *Journal of Low Frequency Noise Vibration and Active Control*, vol. 35, no. 1, pp. 39–51, 2016, doi: 10.1177/0263092316628264.

[15] G. J. Stein, R. Zahoranský, T. P. Gunston, L. Burström, and L. Meyer, “Modelling and Simulation of a Fore-and-Aft Driver’s Seat Suspension System with Road Excitation,” *International Journal of Industrial Ergonomics*, vol. 38, no. 5–6, pp. 396–409, 2008, doi: 10.1016/j.ergon.2007.10.016.
[16] G. J. Stein and P. Múčka, “Study of Simultaneous Shock and Vibration Control by a Fore-and-Aft Suspension System of a Driver’s Seat,” *International Journal of Industrial Ergonomics*, vol. 41, no. 5, pp. 520–529, 2011, doi: 10.1016/j.ergon.2011.03.003.
[17] S. D. Smith, J. A. Smith, and D. R. Bowden, “Transmission Characteristics of Suspension Seats in Multi-Axis Vibration Environments,” *International Journal of Industrial Ergonomics*, vol. 38, no. 5–6, pp. 434–446, 2008, doi: 10.1016/j.ergon.2007.10.028.
[18] G. S. Paddan and M. J. Griffin, “Evaluation of Whole-Body Vibration in Vehicles,” *Journal of Sound and Vibration*, vol. 253, no. 1, pp. 195–213, 2002, doi: 10.1006/jsvi.2001.4256.
[19] E. Contract *et al.*, “September 2002 to 30,” 2005.
[20] R. Deboli, A. Calvo, and C. Preti, “Whole-Body Vibration: Measurement of Horizontal and Vertical Transmissibility of an Agricultural Tractor Seat,” *International Journal of Industrial Ergonomics*, vol. 58, pp. 69–78, 2017, doi: 10.1016/j.ergon.2017.02.002.
[21] L. Gannon, “Single Impact Testing of Suspension Seats for High-Speed Craft,” *Ocean Engineering*, vol. 141, no. September 2016, pp. 116–124, 2017, doi: 10.1016/j.oceaneng.2017.06.017.
[22] Y. Fouad and P. Affairs, “AN EXPERIMENTAL METHODOLOGY FOR CHARACTERIZING HIGH SPEED CRAFT SEAT SUSPENSION COMPONENTS,” no. December, 2014.
[23] H. Ciloglu, M. Alziadeh, A. Mohany, and H. Kishawy, “International Journal of Industrial Ergonomics Assessment of the Whole Body Vibration Exposure and the Dynamic Seat Comfort in Passenger Aircraft,” *International Journal of Industrial Ergonomics*, vol. 45, pp. 116–123, 2015, doi: 10.1016/j.ergon.2014.12.011.
[24] J. D. Leatherwood, “Vibrations Transmitted to Human Subjects through Passenger Seats and Considerations of Passenger Comfort.,” *Applied Ergonomics*, vol. 7, no. 2, p. 114, 1976, doi: 10.1016/0003-6870(76)90195-2.
[25] H. Wang, J. T. Xing, W. G. Price, and W. Li, “ARTICLE IN PRESS An Investigation of an Active Landing Gear System to Reduce Aircraft Vibrations Caused by Landing Impacts and Runway Excitations,” vol. 317, pp. 50–66, 2008, doi: 10.1016/j.jsv.2008.03.016.
[26] Z. Ilic, B. Rasuo, M. Jovanovic, S. Pekmezovic, A. Bengin, and M. Dinulovic, “Potential Connections of Cockpit Floor-Seat on Passive Vibration Reduction at a Piston Propelled Airplane,” *Tehnicki Vjesnik*, vol. 21, no. 3, pp. 471–478, 2014.
[27] G. Tora and W. Trzaska, “MECHANISM OF A MOTION PLATFORM INDUCING THE MOVEMENT OF THE OPERATOR ’ S SEAT IN A HEAVY MACHINE,” vol. 21, no. 4, pp. 1007–1015, 2016, doi: 10.1515/ijame-2016-0062.
[28] Prof. Amol P. Kokare, Akshay Kamane, Vardhan Patil, and Vikrant Pakhide, “Performance Evaluation of Shock Absorber Acting as a Single Degree of Freedom Spring-Mass-Damper System Using MATLAB,” *International Journal of Engineering Research and*, vol. V4, no. 09, pp. 730–734, 2015, doi: 10.17577/ijertv4is090621.
[29] A. R. Bhise, R. G. Desai, R. N. Yerrawar, and A. C. R. R. Mitra, “Comparison Between Passive And Semi-Active Suspension System Using Matlab / Simulink,” vol. 13, no. 4, pp. 1–6, 2016, doi: 10.9790/1684-1304010106.
[30] B. K. K. S. Patil, Vaibhav Jagtap, Shrikant Jadhav, Amit Bhosale, “Performance Evaluation of Active Suspension for Passenger Cars Using MATLAB Using MATLAB,” no. January, 2017.
[31] B. T. Fijalkowski, “Automotive Mechatronics: Operational and Practical Issues,” *Automotive Mechatronics: Operational and Practical Issues*. 2011, doi: 10.1007/978-94-007-1183-9.
[32] A. Brandt, *Noise and Vibration Analysis*. 2011.
[33] G. Chen, S. Zhang, and G. Li, “Multistable Behaviors of Compliant Sarrus Mechanisms,” *Journal of Mechanisms and Robotics*, vol. 5, no. 2, 2013, doi: 10.1115/1.4023557.
[34] S. Lu, D. Zlatanov, X. Ding, R. Molfino, and M. Zoppi, “Novel Deployable Mechanisms with Decoupled Degrees-of-Freedom,” *Journal of Mechanisms and Robotics*, vol. 8, no. 2, 2016, doi: 10.1115/1.4031639.
[35]. L. G. Kraige J. L. Meriam, *engineering mechanics dynamics*. John Wiley & Sons, Inc., 2012.
[36] N. Sivakumar, K. Thamaraikannan, R. Kalaiyarasan, S. Veerakumar, and A. Vijay, “Design and Fabrication of Industrial Conveyor Using Crank Mechanism,” pp. 2868–2879, 2016.
[37] J. Thaddaeus, “Synthesis and Simulation of an Offset Slider-Crank Mechanism,” vol. 7, no. 10, pp. 1842–1852, 2016.
[38] S. D. Kearns, “Analysis and Mitigation of Mechanical Shock Effects on High Speed Planing Boats,” pp. 1–149, 2001.
[39] I. Maciejewski, T. Krzyzynski, L. Meyer, and H. Meyer, “Shaping the Vibro-Isolation Properties of Horizontal Seat Suspension,” *Journal of Low Frequency Noise Vibration and Active Control*, vol. 36, no. 3, pp. 203–213, 2017, doi: 10.1177/0263092317717586.