Vibration Analysis at the Operator's Seat of a Purple Onion Harvester

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ABSTRACT

Mechanical vibrations from uneven field operations are transmitted through the chassis of the harvester to the operator's seat, potentially leading to adverse health effects, especially under prolonged exposure. This study develops a three-degree-of-freedom dynamic model representing the harvester body and the seat structure. The seat is modeled as a mass-spring-damper system mounted on the harvester frame. The terrain excitation is simulated as a combination of sinusoidal and stochastic signals. The system's differential equations are solved using the fourth order Runge–Kutta (RK4) method to analyze time-domain vibration responses at the seat location. The vibration level at the seat is assessed in accordance with ISO 2631-1:1997, using Root Mean Square (RMS) acceleration. The results show that seat vibrations can exceed the acceptable threshold in the absence of an appropriate suspension system. Based on these findings, the study proposes the use of the Genetic Algorithm (GA) to optimize the design and operational parameters - including seat mass, stiffness, damping coefficient, and vehicle speed - to minimize the vibration transmitted to the operator. This contributes to improving the working conditions for the agricultural operators, enhancing the machine durability, and increasing the operational efficiency in agricultural environments.

Keywords-whole-body vibration; onion harvester; dynamic modeling; genetic algorithm optimization; Iso 2631-1

I. INTRODUCTION

Agricultural mechanization is developing rapidly in sectors, such as planting and harvesting. Specifically, in the field of onion harvesting, many studies aim to create harvesters to optimize the productivity and reduce the labor costs [1]. Most of these studies focus on the mechanical efficiency, disregarding the dynamic vibration and machine frame oscillations during the field operations [2, 3]. The impact of vibration and noise creates direct oscillations on the vehicle frame and the operator's body leading to long-term health consequences [4]. To determine the hazard levels, and thus ensure safe and comfortable working conditions, the vibrations at the working position of the operator must be analyzed and evaluated [5].

When evaluating Whole-Body Vibration (WBV) at the operator's seat of common agricultural machines, such as combined harvesters and tractors, studies show that the measured RMS vibration levels at the seat can exceed the safety limits prescribed by ISO 2631-1:1997, if the operator is exposed continuously for 8 h per day [6, 7]. Especially, vibrations between 0.4 Hz and 100 Hz — coinciding with the physiological resonance frequency of the spine and internal organs — can cause chronic back pain, spinal degeneration, fatigue, and reduced concentration.

Tractor drivers in tasks, such as plowing, fertilizing, and harvesting, are exposed daily to random and impact-type vibrations [8, 9]. Although some machines are equipped with suspended or damp seats, experiments reveal that this system reduces the vibrations partially [4].

Finite Element Modeling (FEM), natural frequency analysis, and structural optimization employing GA are utilized to optimize the vibrations for precision planter frames, improve the frame performance, reduce the vibrations, and increase the operational quality [10, 11]. GA is a search tool simulating natural selection processes to identify the optimal parameter sets in large search spaces, suitable for nonlinear and multi-objective mechanical design problems [12]. The use of GA to optimize the steel truss systems aims to reduce the structural weight, thereby saving costs and materials, while still satisfying the stress, displacement, and buckling constraints according to AISC standards [13].

Most studies focus only on the vibration of the machine frame or working components, while vibrations at the seat - where vibration is transmitted directly to the human body - have not been studied in detail [14]. The objective of the current study is to develop a vibration model between the machine body and the operator's seat, simulating the effects of uneven terrain on seat vibrations. Using the simulation results, the study evaluates the vibration levels according to ISO 2631-1 to assess the potential health impacts. Subsequently, it proposes optimizing the design parameters using algorithms, such as GA, to minimize the vibrations transmitted to the operator under real working conditions.

II. DYNAMIC MODELING

A. *Harvester Structure*

The purple onion harvester consists of a frame, a digging tooth mechanism used for extracting onions, a bulb transport system with three conveyors, and a set of wheels for mobility. During operation, the machine performs simultaneous tasks, such as digging, soil separation, and transporting onions to the operator's packing station. The harvester is powered by a tractor.

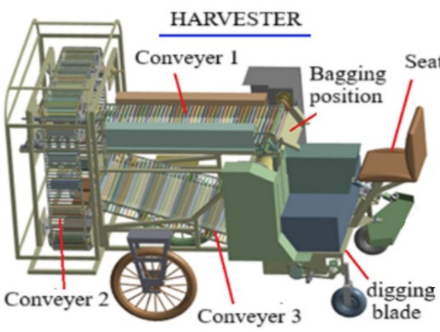

Fig. 1. Structure of the harvesting machine.

To analyze the loads acting on the purple onion harvester, a single-track dynamic model is developed to represent the machine's vertical plane behavior during the field operation, as illustrated in Figure 2.

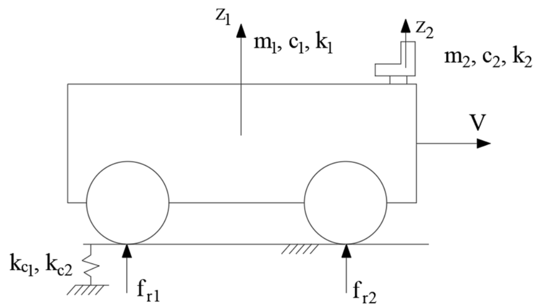

Fig. 2. Dynamic model of a single-track harvesting machine.

B. *Assumptions and Coordinate Systems*

This study aims to simulate and analyze the vibrations at the operator's seat, where vibration is indirectly transmitted from the harvester frame, ground surface, and seat suspension characteristics. The following assumptions and coordinate conventions are used:

- The model is divided into two main bodies: the harvester frame (m_1), which receives vibrations from the ground through the wheels; and the suspended seat (m_2), mounted on the frame, vibrating relative to m_1.
- Both the frame and seat are modeled as rigid bodies. Vibrations are considered in the vertical direction (z-axis) and pitching motion (rotation about the y-axis). Lateral or yaw motions are considered negligible.
- Ground excitation is modeled as a combination of sinusoidal signals and random noise, applied at the wheel-ground contact points.
- The seat is equipped with an independent suspension system: a spring–damper mounted between the seat and the harvester frame to reduce the vibrations transmitted to the operator.

Two coordinate systems are utilized to accurately describe the vibration transmission to the seat:

- Inertial coordinate system Oxyz: x – direction of forward motion; y – lateral direction; z – vertical direction.
- Frame-attached coordinate system: origin located at the center of mass of the harvester body, used to compute the relative motion between the frame and seat.

The generalized coordinates used in the dynamic model are listed in Table I.

TABLE I. GENERALIZED COORDINATES OF THE SYSTEM

Symbol	Description	Unit
$z_1(t)$	Vertical displacement of the harvester frame	m
$\theta_1(t)$	Pitch angle of the frame about the y-axis	rad
$z_2(t)$	Vertical displacement of the seat (relative to ground)	m
$F_r(t)$	Excitation force from ground surface	N

C. Derivation of Dynamic Equations

Applying the second-order Newton method to a system with three degrees of freedom, where z_1 and z_2 represent the vertical displacements of the machine body and seat, respectively, the governing equations employed refer to the harvester's body [15]:

$$F_{r1}(t) + F_{r2}(t) = \ddot{z}_1 + c_1(\dot{z}_1 - \dot{z}_2) + k_1(z_1 - z_2) \quad (1)$$

$$I_1\ddot{\theta}_1 = l_1F_{r1}(t) - l_2F_{r2}(t) \quad (2)$$

where l_1 and l_2 are the distances from the machine center to the front and rear wheels, respectively. Equation (3) refers to the seat:

$$m_2\ddot{z}_2 + c_2\dot{z}_2 + k_2z_2 + c_1(\dot{z}_2 - \dot{z}_1) + k_1(z_2 - z_1) = 0 \quad (3)$$

Rewriting the system in matrix form provides:

$$M\ddot{q} + C\dot{q} + Kq = F(t) \quad (4)$$

where q is the vector of generalized coordinates, M is the mass matrix, C is the damping matrix, K is the stiffness matrix, and $F(t)$ is the external force vector. These variables are described by:

$$q = \begin{bmatrix} z_1 \\ \theta \\ z_2 \end{bmatrix} \quad (5)$$

$$M = \begin{bmatrix} m_1 & 0 & 0 \\ 0 & I_1 & 0 \\ 0 & 0 & m_2 \end{bmatrix} \quad (6)$$

$$C = \begin{bmatrix} c_1 & 0 & -c_1 \\ 0 & 0 & 0 \\ -c_1 & 0 & c_1 + c_2 \end{bmatrix} \quad (7)$$

$$K = \begin{bmatrix} k_1 & 0 & -k_1 \\ 0 & 0 & 0 \\ -k_1 & 0 & k_1 + k_2 \end{bmatrix} \quad (8)$$

$$F(t) = \begin{bmatrix} f_{r1} + f_{r2} \\ l_1F_{r1} - l_2F_{r2} \\ 0 \end{bmatrix} \quad (9)$$

D. Excitation Forces from Terrain

Excitation forces from the uneven terrain, modeled as sinusoidal and random function and are described by [16]:

$$F_{r1}(t) = A_s\sin(\omega t) + A_r(2\mathrm{random}(t) - 1) \quad (10)$$

$$F_{r2}(t) = A_s\sin(\omega t + \varphi) + A_r(2\mathrm{random}(t + \tau) - 1) \quad (11)$$

where A_s and A_r are the amplitudes of the sinusoidal and random components respectively, ω is the angular frequency of terrain excitation, φ is the phase shift between front and rear wheels, and τ is the time delay between them.

III. SIMULATION AND RESULT ANALYSIS

The vibration model is solved numerically to evaluate the vibration levels transmitted to the seat during realistic field operation. The RK4 method is used for time-domain integration.

A. Model Parameters

The parameters used in the simulation were selected based on the actual configuration of the purple onion harvester and the operator seat:

TABLE II. SIMULATION PARAMETERS

Parameter	Symbol	Value	Unit
Mass of the harvester body	m_1	625	kg
Mass of operator seat (including operator)	m_2	80	kg
Stiffness of harvester frame suspension	k_1	20,000	N/m
Damping coefficient of the harvester frame	c_1	800	Ns/m
Stiffness of seat suspension	k_2	10000	N/m
Damping coefficient of seat suspension	c_2	400	Ns/m
Amplitude of periodic excitation force (determined by the Bekker model [16])	A_s	2036	N
Amplitude of random excitation force (determined by the Bekker model)	A_r	7.2	N
Simulation duration	T	10	s
Simulation time step	Δt	0.002	s
Harvester traveling speed	V	5	km/h

B. Solution Method

The vibration system consists of three second-order differential equations, which are converted into a set of first-order differential equations and solved using the RK4 method [17, 18].

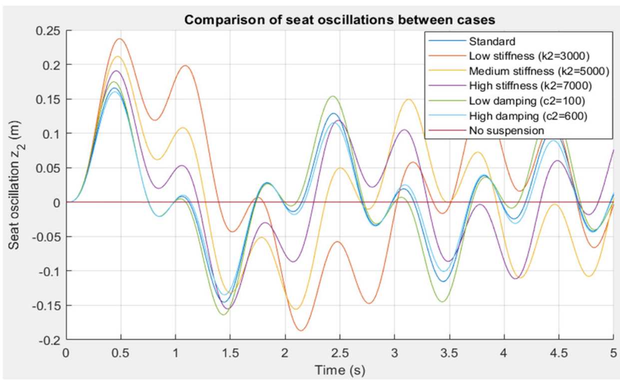

Fig. 3. Oscillation results at the seat.

The simulations are conducted under the following scenarios: the equations are solved to determine the vertical vibration and pitch angle of the harvester frame, and the vibration of the operator seat. The seat vibration is investigated under the original suspension stiffness and damping parameters. The seat suspension stiffness (k_2) is varied, evaluating values of 3000 N/m, 5000 N/m, and 7000 N/m. The seat damping coefficient (c_2) is varied, evaluating values of 100 Ns/m, 400 Ns/m, and 600 Ns/m. The case without a seat suspension system (c_2=0, k_2=∞) is analyzed. The vibration results at the seat are recorded in terms of time–displacement curves and analyzed by calculating the RMS acceleration for each set of parameters. The seat vibration results for each set of parameters are illustrated in Figure 3.

C. Result Analysis

Based on the results shown in Figure 3, several observations can be made:

- Low stiffness curve: This configuration exhibits the largest vibrations, with amplitudes reaching nearly ±0.25 m. This indicates a soft suspension system, resulting in strong and prolonged oscillations. The low stiffness leads to poor energy absorption, causing resonance and significant vibration. Therefore, a seat suspension that is too soft can cause instability and increase the risk of physiological resonance, making it unsuitable for the operator.
- No suspension curve: The amplitude appears very small, with almost no visible relative motion. However, this does not mean that the seat is not vibrating; rather, it is rigidly attached to the frame, resulting in direct transmission of the frame vibrations to the operator without any absorption. This condition is extremely dangerous from a physiological standpoint, as the RMS acceleration at the seat can be very high.
- Standard curve: The vibration is relatively stable and falls between the other configurations. However, the vibration level remains high enough to significantly affect the operator comfort.
- Other configurations: Generally, the vibration amplitudes remain high, making them unsuitable for ensuring the operator's health and comfort.

D. Vibration Reduction Strategy

To minimize vibration at the seat, key design and operational parameters are optimized, including seat mass, suspension stiffness and damping, and harvester speed. GA is used to search for the optimal configurations.

TABLE III. OPTIMIZATION VARIABLES

Variable	Range	Description
m_2	40 – 80 kg	Combined mass of seat and operator
k_2	2000 – 20000 N/m	Seat spring stiffness
c_2	500 – 20000 Ns/m	Seat damping coefficient
V	0.5 – 2.5 m/s	Harvester velocity

Using MATLAB's GA function, the algorithm searches for values minimizing the seat RMS acceleration [19]. The results are shown in Figures 4 and 5. A summary of the optimized performance is:

- Harvester body oscillation amplitude*:* ±0.05 m, stable damping after 2 s.
- Pitch motion: cumulative but manageable (approximately 2 rad after 10 s).
- Seat vibration: reduced from 0.01 mm to 0.002 mm after 2-3 s, suggesting strong damping effect.
- The final RMS acceleration (a_{RMS}) value, measured at the operator's seat, was 0.2726 m/s², which is lower than the threshold value of 0.315 m/s² for WBV exposure. According to ISO 2631-1:1997, the RMS values below this threshold fall within the no discomfort zone, indicating that the optimized seat suspension system effectively mitigates the transmitted vibrations and ensures a comfortable and safe environment for the operator during prolonged operation.

Fig. 4. Oscillation of the machine and seat after optimization.

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✅ OPTIMIZED RESULTS:
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Seat mass       m2 = 40.05 kg
Seat stiffness          k2 = 18726.12 N/m
Seat damping       c2 = 19940.50 Ns/m
Harvester velocity V = 2.48 m/s (8.94 km/h)
→ RMS acceleration at seat: 0.2726 m/s²
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Fig. 5. Parameters after optimization.

IV. EXPERIMENTAL SETUP AND IMPLEMENTATION ORIENTATION

A. Experimental Setup

The results were developed based on a dynamic model utilizing technical parameters collected from the actual configuration of the purple onion harvester. Tri-axial accelerometers (AKG 392) are installed at key locations, including the operator's seat, the machine frame, and the digging unit. These sensors have a sensitivity of 100 ± 15% mV/g, a measurement range of ±40 m/s², and a frequency bandwidth of up to 1000 Hz.

Fig. 6. AKG 392 tri-axial accelerometers.

Vibration data are collected under various operating conditions with traveling speeds between 4 and 10 km/h, across typical terrain types found in actual onion fields. The experimental results will be used to calibrate the simulation model and verify the correlation between the theoretical predictions and real-world measurements, thereby improving the model's accuracy and the overall practical value of the research.

B. Data Analysis

Vibration data at the seat will be processed using the RMS method in the time domain and the Fast Fourier Transform (FFT) in the frequency domain. The signal will be filtered using a fourth-order Butterworth low-pass filter to eliminate the high-frequency noise. The resulting frequency spectrum helps identify the hazardous resonance regions, particularly in the 1–4 Hz range, which corresponds to the physiological resonance frequencies of the human spine and internal organs. This information is significant for both the evaluating operator comfort and planning maintenance for the structural components subjected to repeated vibrations.

V. DISCUSSION AND RECOMMENDATIONS

The analysis results demonstrate the effectiveness of the proposed research method, confirming its potential for practical application. However, a limitation of this study is the lack of experimental data for model calibration. Although the technical parameters used closely reflect the actual configuration of the purple onion harvester, field measurements are necessary to compare, validate, and refine the simulation model. In addition, the current model considers only vertical vibration and pitching motion, and does not fully capture three-dimensional dynamics, such as rolling, yawing, or lateral sway.

The WBV transmitted through the operator's seat can have adverse long-term effects, particularly in the frequency range of 1–4 Hz, which coincides with the physiological resonance of the human spine and internal organs. A prolonged exposure to vibrations in this range may lead to chronic lower back pain, spinal degeneration, reduced concentration, and decreased work efficiency. Therefore, evaluating vibrations at the operator's seat is not only a technical necessity, but also a critical step in protecting the health of agricultural workers.

Based on the simulation and analysis results, the research team proposes several practical recommendations:

- Design: Seat suspension system improvement towards semi-active or pneumatic types and machine frame reinforcement at locations prone to resonance.
- Maintenance: Performance of regular inspections of rubber dampers, hydraulic shock absorbers, and ensure proper wheel balancing.
- Operation: Provision of guidance to operators on selecting the appropriate driving speeds for different terrain conditions to avoid operating within resonance-prone frequency ranges.
- Health protection: Equip operators with back-support belts, implement work shift rotation schedules, organize periodic medical check-ups focusing on musculoskeletal health, and integrate over-vibration warning systems using sensors and support software.

These directions can improve the working conditions and reduce the health risks for the operators, and enhance the durability and operational efficiency of the purple onion harvester under actual field conditions.

VI. CONCLUSION

This study develops a dynamic model of the harvester body and suspended operator seat under terrain-induced vibration, modeled using Bekker's formulation with combined periodic and stochastic excitations. The three-degree-of-freedom system is solved using fourth-order Runge–Kutta (RK4) in MATLAB. A Genetic Algorithm (GA) is employed to optimize the seat suspension parameters (mass, stiffness, damping, and speed) to minimize the Root Mean Square (RMS) acceleration. The optimal configuration achieved is: suspended seat mass (m_2) of 40.05 kg, seat suspension stiffness (k_2) of 18726.12 N/m, seat damping coefficient (c_2) of 19440.5 Ns/m, harvester traveling speed (V) of 2.48 m/s, and RMS acceleration (a_{RMS}) of 0.2726 m/s². This confirms the feasibility of using GA-based optimization combined with dynamic simulation to improve the ride comfort for onion harvester operators. The findings provide a basis for further field validation and can serve as a reference for agricultural machine design.

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REFERENCE

[1] L. Kumawat and H. Raheman, "Mechanization in Onion Harvesting and its Performance: A Review and a Conceptual Design of Onion Harvester from Indian Perspective," *Journal of The Institution of Engineers (India): Series A*, vol. 103, no. 1, pp. 295–304, Jan. 2022, https://doi.org/10.1007/s40030-021-00611-3.

[2] H. Gupta, H. Dheeraj, S. K. Yadav, D. Chaudhary, and A. Chandra, "Modification In The Design And Development Of Tractor Trails For Onion Harvester," in *Procedings of 1st International Conference on Sustainable Materials, Manufacturing & Energy Technologies*, Uttar Pradesh India, June 2022, pp. 820–825.

[3] H. E. Abdalla, M. M. Ibrahim, M. M. Abdel-Galeil, and M. M. Sh. Refaey, " Developing a Combine Machine for Harvesting and Collecting of Onion Crop," *Journal of Soil Sciences and Agricultural Engineering*, vol. 1, no. 4, pp. 349–361, Apr. 2010, https://doi.org/10.21608/jssae.2010.74189.

[4] J. Brunetti, W. D'Ambrogio, and A. Fregolent, "Analysis of the Vibrations of Operators' Seats in Agricultural Machinery Using Dynamic Substructuring," *Applied Sciences*, vol. 11, no. 11, May 2021, Art. no. 4749, https://doi.org/10.3390/app11114749.

[5] Z. W. Engel and P. Kowalski, "Investigation of the influence of simultaneous vibroacoustic exposures on the operator," *Journal of Theoretical and Applied Mechanics*, vol. 46, no. 4, pp. 799–811, 2008.

[6] M. Futatsuka, S. Maeda, T. Inaoka, M. Nagano, M. Shono, and T. Miyakita, "Whole-Body Vibration and Health Effects in the Agricultural Machinery Drivers," *Industrial health*, vol. 36, no. 2, pp. 127–132, 1998, https://doi.org/10.2486/indhealth.36.127.

[7] *Mechanical vibration and shock-Evanluation of human exposure to whole-body vibration, Part I: General requirements*, ISO-2631-1-1997, The International Organization for Standardization, London, United Kingdom, 1997.

[8] *Mechanical vibration and shock — Range of idealized values to characterize seated-body biodynamic response under vertical vibration*, ISO 5982:2001, The International Organization for Standardization, London, United Kingdom, 2001.

[9] H. Seidel and M. J. Griffin, "Modelling the response of the spinal system to whole-body vibration and repeated shock," *Clinical Biomechanics*, vol. 16, Jan. 2001, Art. no. S3–S7, https://doi.org/10.1016/S0268-0033(00)00095-4.

[10] V. A. Ovtov, N. S. Chirkova, K. A. Gorshkov, and D. A. Frolov, "Justification of the traction resistance of the coulter of the onion planter for ecological production of products," *IOP Conference Series: Earth and Environmental Science*, vol. 1138, no. 1, Oct. 2023, Art. no. 012046, https://doi.org/10.1088/1755-1315/1138/1/012046.

[11] Y. Gao *et al.*, "Optimization of Operating Parameters for Straw Returning Machine Based on Vibration Characteristic Analysis," *Agronomy*, vol. 14, no. 10, Oct. 2024, Art. no. 2388, https://doi.org/10.3390/agronomy14102388.

[12] M. Z. R. Khan and A. K. Bajpai, "Genetic Algorithm And Its Application In Mechanical Engineering," *International Journal of Engineering Research & Technology*, vol. 2, no. 5, pp. 677–683, May 2013.

[13] I. M. Mahmood, S. T. Yousif, and H. K. Issa, "Structural Efficiency Enhancement in Steel Trusses utilizing Genetic Algorithm," *Engineering, Technology & Applied Science Research*, vol. 15, no. 3, pp. 23699–23706, June 2025, https://doi.org/10.48084/etasr.9540.

[14] R. G. Dong, X. S. Xu, D. E. Welcome, and T. W. McDowell, "A Method for Analyzing the Effectiveness of Vibration-Reducing Gloves Based on Vibration Power Absorption," *Vibration*, vol. 4, no. 1, pp. 16–29, Dec. 2020, https://doi.org/10.3390/vibration4010002.

[15] E. Hairer and G. Wanner, *Solving Ordinary Differential Equations II*. New York, NY, USA: Springer, 1996.

[16] J. Y. Wong, *Theory of Ground Vehicles*. Hoboken, NJ, USA: John Wiley & Sons, 2001.

[17] L. F. Shampine and M. W. Reichelt, "The MATLAB ODE Suite," *SIAM Journal on Scientific Computing*, vol. 18, no. 1, pp. 1–22, Jan. 1997, https://doi.org/10.1137/S1064827594276424.

[18] M. Agostinacchio, D. Ciampa, and S. Olita, "The vibrations induced by surface irregularities in road pavements – a Matlab® approach," *European Transport Research Review*, vol. 6, no. 3, pp. 267–275, Dec. 2013, https://doi.org/10.1007/s12544-013-0127-8.

[19] J. G. Proakis and D. G. Manolakis, *Digital Signal Processing*. Hoboken, NJ, USA: Prentice-Hall International Inc., 1996.