OPTIMIZING SEAT OSCILLATION THROUGH THE DYNAMIC MODEL OF A PURPLE ONION HARVESTING SYSTEM

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ABSTRACT

This study constructs a 5-degree-of-freedom oscillation model for an onion harvesting machine system consisting of a tractor, harvester, and seat. The seat is modeled as a mass-spring-dampening system mounted on the harvester. The forces acting on the wheels are simulated as a complex combination of sine wave functions and random noise. A fourth-order Runge-Kutta method is used to solve the differential equations to analyze the oscillation response over time at the seat position. The ISO 2631-1:1997 standard was referenced to assess the impact of vibrations on workers through the RMS value of vibration acceleration. Based on this, the study proposes an optimization solution for parameters (mass, stiffness, damping coefficient, velocity) using a genetic algorithm to optimize vibration reduction at the chair. Through optimization, the result obtained was RMS = 0.2192m/s^2 compared to the pre-optimization value of RMS = 2.6349m/s^2. The results of the study can help increase the durability of the machine and ensure the health of the machine operators.

Keywords: *Optimization, oscillation model, vibration, purple onion harvesting system.*

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1. INTRODUCTION

The application of mechanization in agricultural production is receiving attention worldwide and in Vietnam. Currently, there are many studies on the mechanization of agricultural production in Vietnam in general and the Mekong Delta in particular [1, 2]. In onion production technology, onion harvesting machines are also being researched, manufactured, and applied to improve productivity and reduce manual labor costs [3, 4]. However, the increased use of machinery also leads to many negative consequences related to mechanical vibration, affecting the structural durability of equipment, harvesting productivity, and operator comfort [5, 6].

During the operation of shallot harvesting machines, vibrations are mainly generated by uneven field conditions and unstable soil-machine interaction forces [7]. If left uncontrolled, these vibrations can cause resonance, reducing the accuracy of digging and harvesting, and accelerating structural damage to the machinery [6, 8]. Studies on agricultural machinery frames have shown that vibration analysis and structural design optimization are effective solutions to ensure long-term durability and operational efficiency [9, 10].

To address these issues, many modern research approaches have been applied, including multi-degree-of-freedom dynamic modeling, the application of Bekker theory to simulate ground reactions, and the use of finite element analysis (FEM) to estimate stress and strain [10, 11]. Numerical simulation techniques, such as the Runge-Kutta method, are also widely used to solve the equations of motion of systems [12].

Simultaneously, to improve efficiency and optimize design parameters such as spring stiffness, damping coefficient, seat mass, and operating speed, optimization algorithms such as genetic algorithms (GA) and response surface methods (RSM) have been successfully applied [13]. Recent studies have also shown that adjusting the cutting blade angle and travel speed significantly affects the rate of mechanical damage and harvesting efficiency of shallots [13, 14].

In addition, frequency spectrum analysis (FFT) and resonance testing are also performed to accurately assess dangerous oscillation frequency ranges that need to be avoided. Optimizing the structure to push natural frequencies out of the motor excitation region has proven to be an effective solution in reducing vibration and increasing chassis durability [6, 15].

Based on the research above, this paper aims to develop a comprehensive kinematic model for a combined manual harvesting machine for shallots manufactured in Vietnam, simulating seat vibrations, and applying a genetic algorithm to optimize design parameters to reduce RMS acceleration values according to ISO 2631 standards. The research hopes to contribute to minimizing the impact of vibration on operators, extending equipment lifespan, and optimizing operational efficiency under real-world working conditions.

The main objective of this study is to analyze the impact of vibrations at the seat on the operator and to determine the optimal structural parameters to minimize vibration transmission to the seat. The research results will contribute to improving working conditions, reducing fatigue, and enhancing the operational efficiency of onion harvesting machines under real-world conditions in Vietnamese fields.

2. RESEARCH METHODOLOGY

2.1. Dynamic Model

The combined machine for harvesting shallots is a machine combination (LHM) consisting of a tractor and a harvester, designed and manufactured for the purpose of mechanizing and increasing productivity in the field of shallot harvesting.

These tractors are small, with a power output of 35 - 45HP, suitable for small and medium-sized farming conditions for households in the Mekong Delta region of Vietnam.

The shallot harvester has a body with mechanisms for harvesting and transporting onions, and wheels for mobility. During harvesting, it digs up and transports the onions to a packing area. The tractor and harvester are connected by a swivel joint.

Figure 1. Onion harvesting machine assembly

To study the dynamics, the machine assembly was modeled as a single-track model to analyze the forces acting on the LHM in the vertical plane during operation, as shown in Figure 2.

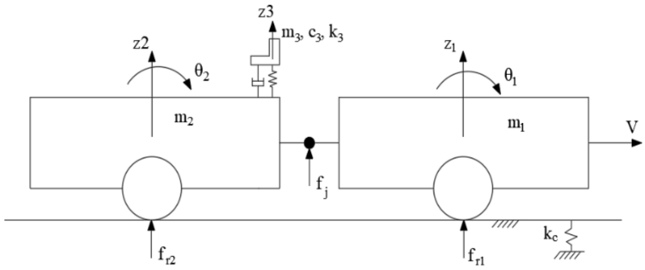


Figure 2. Model of a harvester's track

The onion harvesting system is modeled as a mechanical system with five degrees of freedom. The model structure includes:

- Tractor (m_1): consider the vertical motion z_1 and the longitudinal oscillation θ_1 around the center of gravity.

- Harvester (m_2): consider the vertical motion z_2 and the vertical oscillation θ_2 around the center of gravity.

- Driver's seat (m_3): considers vertical movement z_3 relative to the harvester body.

The two bodies are linked via a freely rotating joint, allowing for vertical power transmission without hindering the relative rotational movement between the tractor and harvester. The driver's seat is mounted on the harvester via an independent suspension system with its own stiffness and drag coefficient.

The system is actuated by ground forces applied to the wheels, simulating realistic terrain characteristics.

2.2. Assumptions

The main objective of this paper is to study the vibrations at the position of the person sitting while bagging, and the assumptions are as follows:

- The model consists of three rigid blocks: the tractor block (m_1) and the harvester block (m_2) which experience vibrations from the ground through the wheels, and the suspended seat (m_3) mounted on the machine body, which vibrates relative to m_2. The model only considers vertical (z-axis) and longitudinal oscillation (rotation around the y-axis) vibrations.

- Ground-based excitation force: The ground is modeled using a combination of sine wave signals and random noise, transmitted into the system through the wheel contact points.

- The LHM moves uniformly at a constant velocity of 5km/h; therefore, translational motion along the x-axis does not affect the overall oscillation and is therefore not considered in the model.

- The impact of rough terrain on the wheels is simulated as a random oscillation signal or wave function, reflecting the reality of the field.

The degrees of freedom and reference coordinates of the model are presented in Table 1.

Table 1. Degrees of freedom of the system

Ingredient	Degrees of freedom	Reference coordinate system
Tractor (m_1)	z_1, θ_1	O_1 (center of gravity of the tractor)
Harvesting machine (m_2)	z_2, θ_2	O_2 (harvesting machine center)
Seating capacity (m_3)	z_3	O_3 (attached to the harvesting machine body)

2.3. Constructing the equations of motion dynamics

Using Newton's second law for a system with 5 degrees of freedom, the oscillation equations of LHM take the form (1-5):

Equations for tractors (1), (2):

$$m_1\ddot{z}_1 = f_{r1} + f_j - m_1 g \tag{1}$$

$$I_1\ddot{\theta}_1 = l_1 f_{r1} - l_j f_j \tag{2}$$

Equations for the harvesting machine (3), (4):

$$m_2\ddot{z}_2 = f_{r2} - f_j - k_3(z_3 - z_2) + c_3(\dot{z}_3 - \dot{z}_2) - m_2 g \tag{3}$$

$$I_2\ddot{\theta}_2 = l_2 f_{r2} + l_3(k_3(z_3 - z_2) + c_3(\dot{z}_3 - \dot{z}_2)) \tag{4}$$

Equation for chair displacement:

$$m_3\ddot{z}_3 = k_3(z_2 - z_3) + c_3(\dot{z}_2 - \dot{z}_3) - m_3 g \tag{5}$$

where: l_1, l_2 are the distances from the center of gravity of the tractor and harvester to the point of force application f_{r1} and f_{r2}; l_3 It is the distance from the center of gravity of the harvester to the seat.

The forces at the joint are calculated as follows:

$$f_j = k_j(z_2 - z_1) + c_j(\dot{z}_2 - \dot{z}_1) \tag{6}$$

We rewrite the system of 5 oscillation equations (1-5) in the form:

$$M\ddot{q} + C\dot{q} + Kq = F(t) \tag{7}$$

In there:

q is the vector of generalized coordinates:

$$q = \begin{bmatrix} z_1 \\ \theta_1 \\ z_2 \\ \theta_2 \\ z_3 \end{bmatrix} \tag{8}$$

M is the mass matrix:

$$M = \begin{bmatrix} m_1 & 0 & 0 & 0 & 0 \\ 0 & J_1 & 0 & 0 & 0 \\ 0 & 0 & m_2 & 0 & 0 \\ 0 & 0 & 0 & J_2 & 0 \\ 0 & 0 & 0 & 0 & m_3 \end{bmatrix} \tag{9}$$

C is the barrier matrix:

$$C = \begin{bmatrix} c_c & 0 & -c_j & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 \\ -c_j & 0 & c_j + c_3 & 0 & -c_3 \\ 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & -c_3 & 0 & c_3 \end{bmatrix} \tag{10}$$

K is the stiffness matrix:

$$K = \begin{bmatrix} k_j & 0 & -k_j & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 \\ -k_j & 0 & k_j + k_3 & 0 & -k_3 \\ 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & -k_3 & 0 & k_3 \end{bmatrix} \tag{11}$$

F(t) is the external load vector:

$$F(t) = \begin{bmatrix} -f_{r1}(t) \\ -l_1 f_{r1}(t) \\ -f_{r2}(t) \\ -l_2 f_{r2}(t) \\ 0 \end{bmatrix} \tag{12}$$

2.4. Construct the equations for the forces acting from the ground on the wheels.

2.4.1. Uneven field surface

The driving force on the wheels comes from the uneven, sinusoidal and random field surface. The field surface irregularities have the following forms:

$$z_g(t) = z_{sin}(t) + z_{random}(t) \tag{13}$$

In there :

$z_{sin}(t) = A_{sin}\sin(\frac{2\pi V}{\lambda}t)$: Sine wave, where A_{sin} is the amplitude and λ is the wavelength.

$z_{random}(t)$ generated from noise according to ISO 8068 standard.

From formula (10), the uneven shape of the field surface is presented in formulas (14) and (15):

$$z_{g1}(t) = A_{sin}\sin(\frac{2\pi V}{\lambda}t) + z_{random}(t) \tag{14}$$

$$z_{g2}(t) = A_{sin}\sin(\frac{2\pi V}{\lambda}(t - \Delta t)) + z_{random}(t - \Delta t) \tag{15}$$

In this formula, Δt is the lag between the front and rear wheels; $\Delta t = d/V$, where d is the distance between the two wheels.

2.4.2. Ground contact pressure according to Bekker

The instantaneous wheel penetration is given in formula (16):

$$\varepsilon(t) = z_g(t) - z_{wheel}(t) \quad (16)$$

Because the wheels are rigidly attached to the car body, $z_{wheel}(t) = z(t)$

Earth pressure corresponding to settlement [11]:

$$p(t) = \left(\frac{k_c}{b} + k_\phi\right).(\varepsilon(t))^n \quad (17)$$

In there:

k_c: soil compressibility coefficient (N/m^2)

k_ϕ: internal friction coefficient (N/m^3)

b: wheel width (m)

n: nonlinear coefficient (0.6 - 0.8 depending on soil type).

2.4.3. Forces acting on the wheels

The earth force acting on the wheel is presented in formula (18)

$$f_r(t) = A.p(t) \quad (18)$$

In this formula, A is the contact area between the wheel and the field surface, calculated based on the wheel width and the contact length.

Combining the above equations, we have the force acting on the wheel according to equations (19), (20).

$$\boldsymbol{f_{r1}(t) = A_1.\left(\frac{k_c}{b_1} + k_\phi\right)((z_{g1}(t) - z_1(t))^n} \quad (19)$$

$$\boldsymbol{f_{r2}(t) = A_2.\left(\frac{k_c}{b_2} + k_\phi\right)((z_{g2}(t) - z_2(t))^n} \quad (20)$$

3. RESULTS

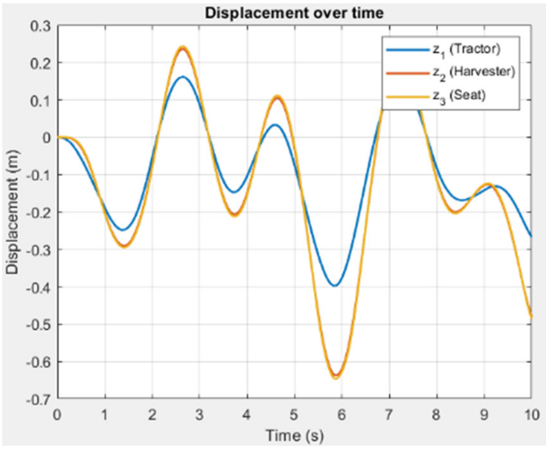
The oscillation model was solved to find the oscillations of the tractor, harvester, and on the seat. The ode45 method and the Runge-Kutta4 algorithm were used [12]. It is used to integrate systems of differential equations with respect to time.

Table 2. Simulation parameters

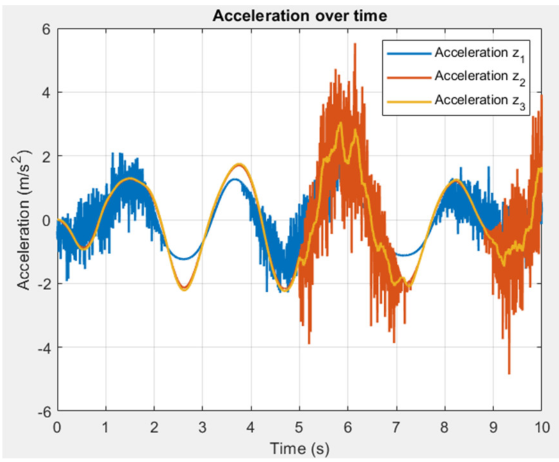
Parameter	Symbol	Value	Unit
Tractor weight	m_1	1400	kg
Tractor moment of inertia	J_1	520	kg.m^2
Harvester weight	m_2	613	kg
Moment of inertia of a harvester	J_2	320	kg.m^2
Seat weight + number of people seated	m_3	90	kg
Soil hardness	k_c	40000	N/m
Earth resistance coefficient	c_c	10000	Ns/m
Joint stiffness	k_j	20000	N/m
Joint resistance coefficient	c_j	500	Ns/m
Chair suspension stiffness	k_3	15000	N/m
Seat suspension damping coefficient	c_3	800	Ns/m
Distance from the tractor's center of gravity to the point of impact f_{r1}	l_1	1.2	m
Distance from the center of gravity of the harvester to the point of impact f_{r2}	l_2	1.5	m
Distance from the center of gravity of the harvester to the seat	l_3	0.5	m

Simulations were conducted using the actual parameters of the LHM shown in Table 2. The oscillation equations, consisting of two second-order differential equations, were transformed into a first-order system of equations and then solved using the RK4 method.

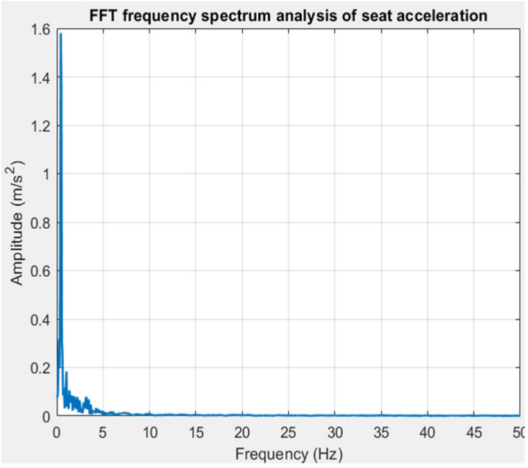
The results of the seat oscillations at the machine assembly are shown in Figure 3.



a) Displacement of IM



b) The oscillation acceleration of IM



c) Frequency spectrum of oscillations at the seat

Figure 3. Oscillation results of the machine assembly

Figure 3 shows the results of oscillation acceleration (3a) and oscillation frequency spectrum (3b) at the seat on the harvester. In addition, according to the simulation, the RMS value of oscillation acceleration at seat a is RMS = 2.6349m/s^2.

As shown in Figure 3a, the oscillation waveforms of the tractor, harvester, and seat all exhibit quite complex oscillation patterns, demonstrating that ground agitation is a combination of sinusoidal and random waves, consistent with the model under study. The maximum oscillation amplitude is approximately ±0.6m, which is a large amplitude given the unevenness of the paddy field. The seat's oscillations are amplified compared to the tractor and harvester. This is a common phenomenon due to the elastic joint and the seat having a lower natural frequency than the machine frame. However, the seat's oscillations do not exhibit dangerous resonance.

The acceleration curves in Figure 3b are quite complex, reflecting the nature of the ground as it contains both harmonic and random components; this is a common vibration signal in agricultural environments. The acceleration amplitude of the seat and harvester is greater than that of the tractor because the tractor is heavier and therefore absorbs vibrations better, and the seat experiences more motion, resulting in greater acceleration.

According to the oscillation frequency spectrum in Figure 3c, the main vibration frequency range is 1 - 5Hz. At 1 - 2Hz, there is resonance, resulting in a very high amplitude of approximately 1.6m/s^2, which then gradually decreases to 0. This is consistent with the reality that agricultural machinery oscillations mainly occur at low frequencies. Furthermore, the RMS acceleration value at seat a = 2.6349m/s^2, compared to the ISO 2631 standard, poses a health hazard to humans if exposed for a long period [16].

4. SOLUTIONS TO REDUCE SEAT VIBRATIONS

To find the lowest vibration at the seating position, we optimize parameters such as the seat suspension system, the weight of the seat and the occupant, and the machine's travel speed.

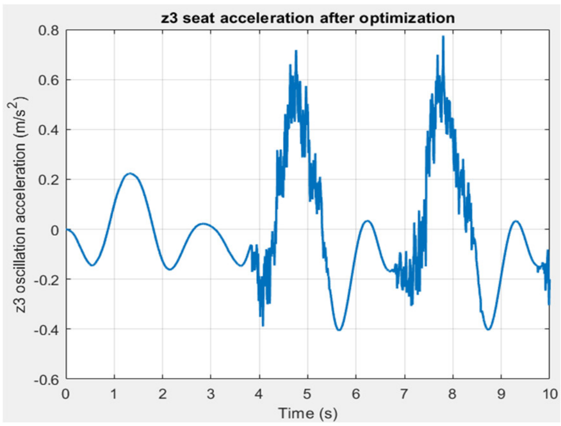
Use a genetic algorithm (GA) to optimize the above parameters. GA simulates the process of natural evolution based on genetic concepts [17]:

- Genes, chromosomes, crossing over, mutation
- Population, individual, fitness function

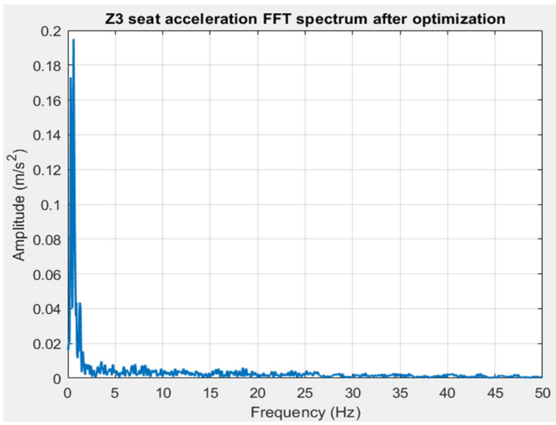
The design and operational variables used for optimization are shown in Table 3.

Table 3. Variables used in optimization

Variable	Limit	Note
m_3	50 - 90kg	Weight of the seating system (person + frame)
k_3	10,000 - 50,000N/m	Seat spring stiffness
c_3	1000 - 10000Ns/m	Seat damping coefficient
V	0.6 - 2.8m/s	Harvester speed



a) Acceleration of oscillation at the seat



b) Frequency spectrum at the seat

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Best parameters: k3 = 21000.69, c3 = 5506.42, m3 = 87.36, V = 0.65
Minimum RMS z3: 0.2192 m/s^2
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c) Optimal parameters output from Matlab

Figure 4. Results graph after optimization

Use the GA command in Matlab to find the combination of 4 variables that minimizes the RMS acceleration at the seat. The result is shown in Figure 4.

According to the oscillation acceleration diagram at the seat in Figure 4a, the oscillation pattern has two main groups of large amplitudes, concentrated in the time periods of 4 - 5 seconds and 7 - 8 seconds. The maximum oscillation amplitude is approximately $+0.7 m/s^2$ and the minimum is approximately $-0.4\ m/s^2$. The oscillations have a low frequency, with each large oscillation occurring approximately every 2 - 3 seconds, consistent with the cyclical stimulation from the large uneven ground. The oscillations are at a controllable level, with reasonable amplitudes and meeting expectations after optimization. The seat system handled small and medium vibrations very well, only experiencing strong impacts when encountering large uneven surfaces.

The frequency spectrum diagram (Figure 4b) shows a very low amplitude, indicating that the system has successfully eliminated resonance. The seating system mainly experiences low-frequency vibrations (1 - 4Hz), corresponding to the oscillations caused by the unevenness of the agricultural land surface. After optimization, the seating system has significantly reduced the amplitude of the main oscillations compared to before, and successfully eliminated higher-order oscillations.

The optimized RMS acceleration value (Figure 4c) is $0.2192\ m/s^2$, which, compared with ISO 2631 standards, does not affect human health and ensures operator comfort.

5. CONCLUSION

The study developed an IM oscillation model for harvesting shallots, with field-based stimuli modeled after the Bekker model, incorporating periodic and random oscillation components. The 5-degree-of-freedom oscillation model was solved using a 4th-order Runge-Kutta method in the MATLAB environment.

Through optimization using a Genetic Algorithm (GA), the design parameters of the seat suspension system, including seat mass, spring stiffness, damping coefficient, and operating speed, were adjusted to reduce the RMS acceleration value at the seat, a measure of whole-body vibration according to ISO 2631-1:1997.

The optimal results show: m_3 = 87.36kg, k_3 = 21000.69N/m, c_3 = 5506.42Ns/m, V = 0.65m/s for an RMS acceleration value RMS = $0.2192 m/s^2$, reaching the "non-discomfortable" threshold according to ISO 2631.

Based on these results, the study confirms the feasibility of optimizing the seat suspension system configuration using a genetic algorithm combined with dynamic simulation, contributing to improved smoothness and comfort for operators of the shallot harvesting machine. These results provide a solid basis for conducting experiments on real terrain. The research methods and results presented in this paper can serve as a reference for the design and manufacture of agricultural machinery in general, and shallot harvesting machinery in particular.

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