Konya Mühendislik Bilimleri Dergisi, c. 13, s. 2, 429-444, 2025
Konya Journal of Engineering Sciences, v. 13, n. 2, 429-444, 2025
ISSN: 2667-8055 (Electronic)
DOI: 10.36306/konjes.1613551

A DYNAMIC ANALYSIS OF AN INDUSTRIAL C-TYPE ECCENTRIC PRESS THROUGH MODELING, SIMULATION, AND EXPERIMENTAL TESTING

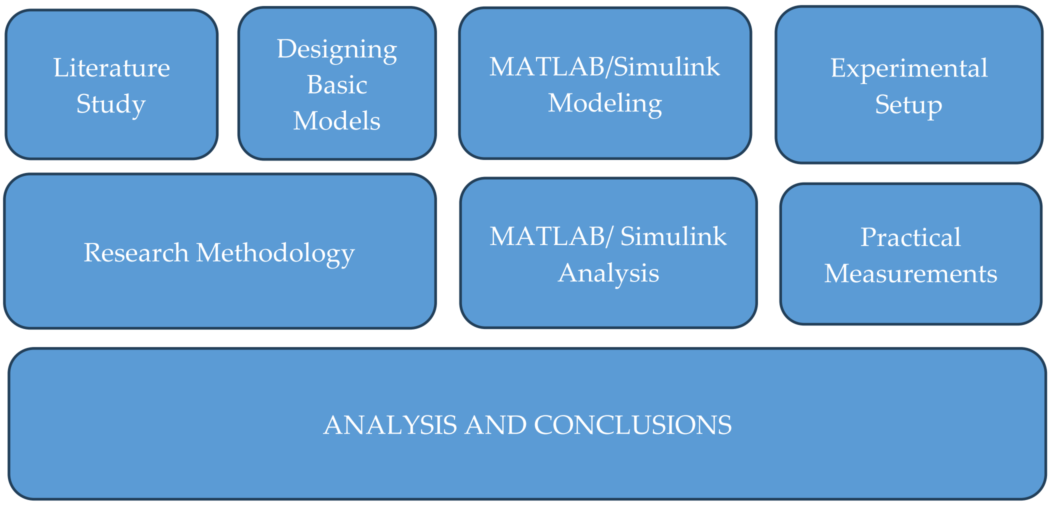
[1,*]**Ahmet KÖKEN**, [2]**Abdurrahman KARABULUT**

[1]*Kütahya Dumlupınar University, Kütahya Vocational School of Technical Sciences, Machinery and Metal Technologies Department, Kütahya, TÜRKİYE*
[2]*Afyon Kocatepe University, Technology Faculty, Mechanical Engineering Department, Afyonkarahisar, TÜRKİYE*
[1] ahmet.koken@dpu.edu.tr, [2] akarabulut@aku.edu.tr

Highlights

- MATLAB/Simulink modeling and analysis of the press-ground system
- A dynamic model was developed to analyze two mass impact load responses
- Stiffness and damping optimization of vibration isolator systems

Graphical Abstract



Flowchart of the proposed method

***Corresponding Author:** Ahmet KÖKEN, ahmet.koken@dpu.edu.tr

Konya Mühendislik Bilimleri Dergisi, c. 13, s. 2, 429-444, 2025
Konya Journal of Engineering Sciences, v. 13, n. 2, 429-444, 2025
ISSN: 2667-8055 (Electronic)
DOI: 10.36306/konjes.1613551

A DYNAMIC ANALYSIS OF AN INDUSTRIAL C-TYPE ECCENTRIC PRESS THROUGH MODELING, SIMULATION, AND EXPERIMENTAL TESTING

[1,*]**Ahmet KÖKEN**, [2]**Abdurrahman KARABULUT**

[1]*Kütahya Dumlupınar University, Kütahya Vocational School of Technical Sciences, Machinery and Metal Technologies Department, Kütahya, TÜRKİYE*
[2]*Afyon Kocatepe University, Technology Faculty, Mechanical Engineering Department, Afyonkarahisar, TÜRKİYE*
[1] ahmet.koken@dpu.edu.tr, [2] akarabulut@aku.edu.tr

(Received: 05.01.2025; Accepted in Revised Form: 06.03.2025)

ABSTRACT: This study provides a comprehensive analysis of the dynamic behavior, modeling, simulation, and experimental validation of industrial C-type eccentric presses, offering critical insights into the optimization of press ground vibration. Through detailed modeling, the forces and vibrations experienced during operation were mathematically characterized, while simulations effectively demonstrated the system's behavior under varying operational conditions, and experimental studies confirmed the reliability of these models. The investigation also examined the impact of dynamic loads on machine foundations, analyzing single and double mass-spring systems using MATLAB simulations and analytical solutions to assess the influence of ground-foundation characteristics on the press's dynamic response. Prior to vibration isolation, the average peak ground displacement (PGD) was measured at 5.075×10^{-2} mm, which decreased to 3.46×10^{-2} mm and 2.7×10^{-2} mm with the application of VI-1 and VI-3 isolators, respectively. The VI-3 isolator proved most effective, reducing transmitted dynamic forces to 2.57×10^{4} N. Parametric analyses highlighted the system's sensitivity to isolator stiffness and damping ratios, with a stiffness ratio of 0.01 between the isolator and ground reducing foundation vibrations by approximately 46.8%. This research emphasizes the importance of dynamic modeling in designing and optimizing vibration isolation systems, making a significant contribution to enhancing vibration control in industrial applications.

Keywords: *Eccentric Press, Vibration Isolator, 2-DOF Modeling, Dynamic Response*

1. INTRODUCTION

Press foundations undergo significant dynamic excitation due to short-duration impulsive loads experienced during normal operation. These dynamic effects, originating from the interface between the upper and lower press modules, often manifest as excessive vibrations that can compromise the press system's performance and disrupt the surrounding environment. The primary design goals for blocking foundations of press systems are to ensure stability and to mitigate vibration amplitudes and transmitted forces to minimize environmental disturbance. Non-linear dry friction mounting systems are commonly employed to support press systems to achieve these objectives. The presence of these systems significantly modifies the foundation system's dynamic response, particularly in terms of natural frequency and damping, when subjected to impulsive loads. Hence, developing a non-linear dry friction mounted block foundation model is crucial [1].

Press foundations can be categorized into two principal types: inertial block foundations (Figure 1(a)) and block foundations equipped with non-linear dry friction isolation systems (Figure 1(b) and 1(c)). When a press is rigidly coupled to an inertial block anchored in the ground, as depicted in Figure 1(a), the foundation system can be idealized as a single-degree-of-freedom system in the vertical axis. The foundation arrangements illustrated in Figures 1(b) and 1(c) can be represented as two-degree-of-freedom systems [1].

***Corresponding Author:** Ahmet KÖKEN, ahmet.koken@dpu.edu.tr

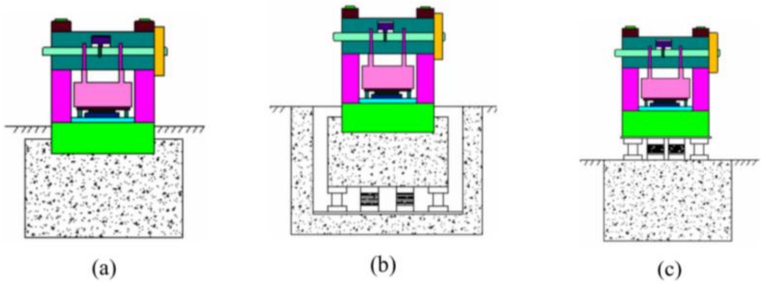


Figure 1. Schematic representations of press foundation configurations: (a) single-degree-of-freedom foundation system and (b, c) two-degree-of-freedom foundation systems with non-linear dry friction isolation [1].

The dynamic precision and reliability of high-speed, high-precision presses depend on effective vibration isolation. Minimizing press vibration is paramount for achieving high-precision stamping operations. Although commonly employed linear vibration isolators are effective in attenuating low-frequency vibrations, their performance is constrained at higher frequencies.

Vibration analysis has gained significant prominence in engineering disciplines, particularly in research and development endeavors, driven by the increasing demand for medium to high-frequency analysis [2]. Given the high dynamic loads and substantial vibration amplitudes generated by equipment such as presses, specialized isolation systems are imperative. The objective of these systems is to attenuate equipment-induced vibrations and minimize their environmental impact [3]. Drawing upon experimental investigations and computational modeling, various isolation systems tailored for presses have been developed [4], [5].

A comprehensive body of research has emerged in recent years focusing on vibration isolation techniques for impact machinery. Some of the studies conducted in the literature are as follows: Zheng et al. [4] presented a dynamic model for analyzing the dynamic performance of a closed-loop high-speed press system in their research. Jia et al. [6] studied the dynamic analysis of a closed-loop high-speed precision press using 2-DOF and 3-DOF models, including modeling, simulation, and experimental validation under periodic impact loads. Jancarczyk et al. [7] performed an in-depth investigation into vibration monitoring within hydraulic presses, employing cutting-edge measurement systems incorporating accelerometers. The suggested system, consisting of a three-axis accelerometer, a data acquisition device, and dedicated measurement software, was shown to streamline accurate vibration tracking and evaluation. They examined the impact of sensor positioning and sampling rate on the measurement outcomes. Temporal and spectral analyses were conducted on the collected data. The findings revealed a link between vibration magnitudes and diverse production variables, including the concurrent output of multiple parts and press force. These results emphasized the significance of vibration measurement as a vital factor in regulating production parameter configurations. Dal and Baklacı [8] placed vibration isolation rubber pads under the workbench legs in their study to reduce vibration-induced noise. After repeated measurements and analyses under the same conditions, they reported a significant decrease in noise and vibration levels of the polishing machine. In their study, Wang et al. [9] proposed a nonlinear vibration isolation platform with a nonlinear isolation system to mitigate ground vibrations from mechanical equipment. They investigated the effects of flexible foundations and isolator parameters on the platform's performance. Saberi et al. [10] carried out a comprehensive study involving both theoretical modeling and experimental testing to analyze the mechanical vibrations occurring during the hot forging process. Kekeç and Ghiloufi [11] investigated the propagation of ground vibrations induced by explosions within a medium of loose, dry sand, encompassing both surface and subsurface scenarios. The study involved the monitoring of particle velocities and the determination of dominant frequency components within the artificially generated vibration field. Chehab et al. [12] presented a rigorous investigation into the design of isolation foundations for forging equipment. Utilizing fundamental soil mechanics principles, the authors conducted a parametric study to evaluate the performance of various vibration reduction configurations and subsequently optimized the design parameters. Avcı and Yazgan [13] presented a comprehensive analysis of displacement rates in structures supported by sandy soils. The study employed nonlinear equivalent single-degree-of-freedom systems to model the dynamic behavior of the soil-structure system. Tekeci et al. [14] presented a comprehensive analysis of the fatigue life of the mounting connection in a shock absorber system. A finite element model, corroborated by modal testing, was utilized to predict the fatigue life. Vibration tests further substantiated the model's accuracy and facilitated the quantification of damping ratios. Köken [15] conducted a study to evaluate the efficacy of vibration isolators in reducing ground vibrations transmitted from press machines. The findings of the study demonstrated that vibration isolators can substantially attenuate ground vibrations. Wang et al. [16] presented a design optimization framework for low-impact transmission foundations tailored to forging hammer applications. The study demonstrated that a viscous spring isolator mounting system, when optimized, can effectively attenuate shock and vibration transmission, leading to a reduction in the overall foundation size. A case study involving a 3-ton forging hammer validated the superiority of the proposed optimization method over conventional design approaches in terms of impact and vibration isolation performance. Hızarcı and Kıral [17] investigated the vibration response of an engineering structure to harmonic ground excitation and explored the potential of air jet actuators for vibration control. Alhuhydan et al. [18] presented an investigation into the reduction of excessive vibration generated by a model. The study focused on the implementation of suitable vibration control strategies. Zhu et al. [19] conducted an analysis of a two-mass vibration system characterized by nonlinear stiffness and damping. Their study demonstrated that significant vibration suppression can be achieved by tuning system parameters and excitation frequency. Lang et al. [20] conducted an evaluation of the impact of nonlinear viscous damping on the performance of single-degree-of-freedom vibration isolators. They developed a frequency response function to characterize the relationship between damping and the transmissibility of vibrations. Ping et al. [21] formulated a mathematical model for a nonlinearly coupled isolator characterized by quadratic, viscous, and Coulomb damping, and a nonlinear spring force. The dynamic transmissibility of the system subjected to deterministic excitation was derived analytically. Kunadharaju and Borthakur [22] presented a study investigating the application of elastomeric dampers and spring vibration isolators beneath the anvil and foundation of forging hammers for vibration reduction. Through analytical and finite element modeling, they developed an isolation system that effectively meets current environmental vibration standards. Guo et al. [23] presented an investigation into the application of a vibration isolation system to reduce vibration transmission between a machine and its foundation. The results demonstrated the efficacy of the proposed system in mitigating unwanted shocks and vibrations. Kumar and Boora [24] experimentally analyzed the dynamic behavior of a machine foundation. They investigated the effects of two distinct combinations of a spring mounting base and a rubber pad interposed between the machine base and the concrete foundation block. In their studies, Köken et al. [25], [26] and Karabulut et al. [27] experimentally investigated the use of vibration isolators for the ground isolation of machine tools and emphasized that the use of vibration isolators has a significant effect on reducing ground vibrations. Abd-Elhamed et al. [28] analyzed the dynamic response of machine foundations under harmonic and impulsive loading conditions. They obtained closed-form solutions for the displacement, velocity, and acceleration time histories resulting from two distinct types of impulsive loads. Kam et al. [29], [30] it has been emphasized that controlling vibrations in systems is crucial, as excessive vibration amplitudes can lead to inefficiency and system failures.

To ensure satisfactory performance of the machine, the mounting system and/or foundation should be designed such that vibration amplitudes do not exceed 1.2 mm [31]. In the study, the dynamic behavior of an industrial C-type eccentric press was investigated using both MATLAB simulations and experimental analysis. Three vibration isolators (VI-1, VI-2, VI-3) with different stiffness and damping properties were comparatively analyzed, and their performances were examined in detail. It was found that the VI-3 isolator could reduce ground vibrations by up to 46.8% and was the most effective solution. The stiffness ratio (0.01) of the mounting system was determined to have a significant effect on vibrations and was found to considerably reduce vibration levels. The comparison of field measurements with simulation results demonstrated that the model accurately represents real operating conditions. The study provides a practical guide for the optimization of vibration isolation systems in industrial machinery. Unlike similar studies in the literature, the performances of different isolator types were compared, and the effect of a specific stiffness ratio was examined in detail.

2. MATERIAL AND METHODS

2.1. Mathematical Models

Press foundations are modeled as lumped mass systems. The foundation systems are commonly modeled as single or two-mass systems, contingent upon the foundation's configuration.

A press foundation devoid of a vibration isolator can be idealized as a single-mass system (Figure 2). In this idealized model, k_o and c_o denote the stiffness and damping coefficients of the soil, m_1 represents the mass of the press, m_2 represents the mass of the foundation block, and m_t denotes the total mass of the system. F_t is the impulsive force exerted on the foundation block, and $x(t)$ represents the resulting response of the foundation block.

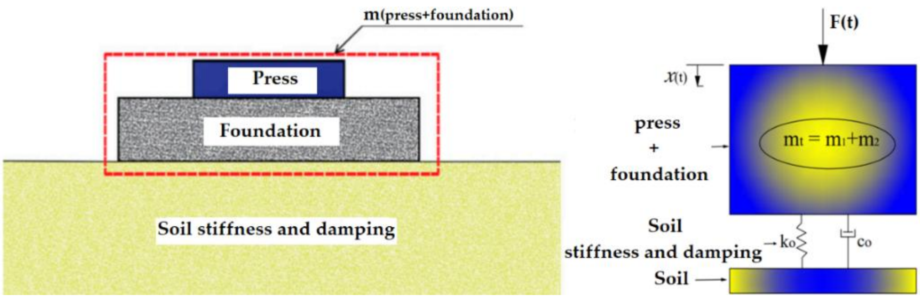


Figure 2. Actual system

A press foundation incorporating a vibration isolator can be idealized as a two-mass system (Figure 3). In this idealized model, k_1 and c_1 denote the stiffness and damping coefficients of the vibration isolator, and k_0 and c_0 denote the stiffness and damping coefficients of the soil. m_1 represents the mass of the press, and m_2 represents the mass of the foundation block. When subjected to dynamic excitation, the press and the foundation block exhibit a characteristic response. This response is contingent upon the magnitude, duration, and frequency of the excitation, as well as the system properties, such as mass, stiffness, and damping. $x_1(t)$ and $x_2(t)$ represent the displacements of the press and the foundation block, respectively.

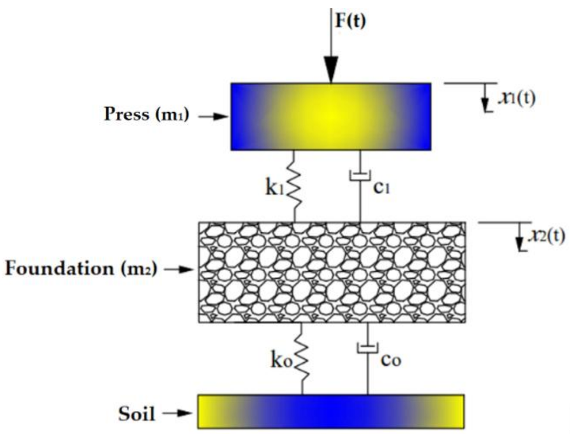


Figure 3. Idealized press- foundation- soil system

2.2. Impact Load

The impact loads resulting from strikes can take various forms. The differences between these loads arise from factors such as the size and shape of the material, its temperature and mechanical properties, the elastic properties of the materials constituting the press system components, the contact surface geometry, and the impact energy. The presence of numerous factors influencing the time-dependent variation of press forces makes it quite challenging to determine these forces theoretically. Therefore, these forces are determined through experimental methods. For theoretical investigations, the real and nonlinear time-dependent variation curves of impact loads are approximated using various methods, such as straight lines or trigonometric functions (Figure 4) [31].

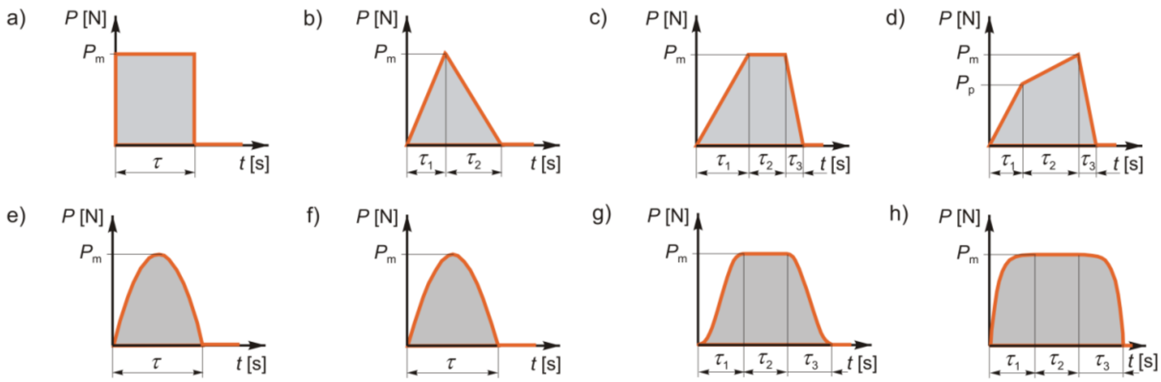


Figure 4. Shapes of impact loads: a) rectangular, b) triangular, c) trapezoidal, d) triangular-trapezoidal, e) parabolic, f) half-sine, g) inverted sine-rectangular, h) exponential-rectangular [31].

The time-varying nature of the impact was simplified by assuming a rectangular pulse shape, where the force is constant over the duration of the impact and then drops abruptly to zero.

$$f(t) = \begin{cases} p_m = \dfrac{S}{t_p} = \dfrac{(1+R)\sqrt{2\cdot U\cdot m_B}}{t_p} & 0 \le t \le t_p \\ 0 & t > 0 \end{cases} \tag{1}$$

Here, p_m maximum impact force, S magnitude of the impact load, t_p loading duration, R coefficient of restitution, U total energy of the press, m_B mass of the ram.

2.3. Analysis Method

To investigate the dynamic characteristics of single and dual-mass systems, equations of motion have been developed based on d'Alembert's principle. The derived equations serve as a mathematical model for describing the force-motion relationship within the system.

Equation (2) provides a mathematical representation of the single-mass system.

$$m_t\ddot{x} + c_0\,\dot{x} + k_0\,x = F(t) \tag{2}$$

Here, k_0 stiffness, c_0 damping coefficient, m_t total mass and $x(t)$ foundation displacement.

Equations (3) and (4) provide a mathematical representation of the two-mass system.

$$m_1\ddot{x}_1 + c_1(\dot{x}_1 - \dot{x}_2) + k_1(x_1 - x_2) = F(t) \tag{3}$$

$$m_2\ddot{x}_2 - c_1(\dot{x}_1 - \dot{x}_2) - k_1(x_1 - x_2) + c_0\dot{x}_2 + k_0x_2 = 0 \tag{4}$$

Matrix form,

$$[m]\{\ddot{x}\} + [c]\{\dot{x}\} + [k]\{x\} = \{F(t)\} \tag{5}$$

$$\begin{bmatrix} m_1 & 0 \\ 0 & m_2 \end{bmatrix} \begin{Bmatrix} \ddot{x}_1 \\ \ddot{x}_2 \end{Bmatrix} + \begin{bmatrix} c_1 & -c_1 \\ -c_1 & c_1 + c_0 \end{bmatrix} \begin{Bmatrix} \dot{x}_1 \\ \dot{x}_2 \end{Bmatrix} + \begin{bmatrix} k_1 & -k_1 \\ -k_1 & k_1 + k_0 \end{bmatrix} \begin{Bmatrix} x_1 \\ x_2 \end{Bmatrix} = \begin{Bmatrix} F(t) \\ 0 \end{Bmatrix} \tag{6}$$

The force passing through the assembly system can be calculated using equation (7).

$$f_a(t) = k_1(x_1 - x_2) + c_1(\dot{x}_1 - \dot{x}_2) \tag{7}$$

The force transmitted to the ground can be calculated using equation (8).

$$f_b(t) = k_0 x_2 + c_0 \dot{x}_2 \tag{8}$$

The response of a single-degree-of-freedom system to harmonic excitation has been mathematically investigated by Rao [32]. Figure 5 shows the variation of force transmission ratio (Tr) with the frequency ratio $r = \omega/\omega_n$ and damping ratio (ζ). For successful isolation, the transmitted force to the foundation must be lower than the applied excitation force. The graph shows that for frequency ratios up to ($\sqrt{2}$), increasing damping decreases the force transmission ratio, but for frequency ratios greater than ($\sqrt{2}$), the force transmission ratio increases again with increasing damping [33]. This finding indicates the necessity of an adjustable damping mechanism in the design of suspension systems.

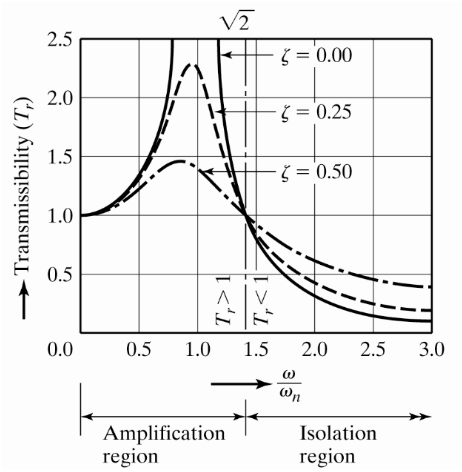


Figure 5. Variation of transmission ratio (T_r) with ω [32].

The transmissibility of a system subjected to harmonic motion is described by equation (9).

$$T_r = \sqrt{\left(\frac{k^2 + \omega^2 c^2}{[(k - m\omega^2)^2 + \omega^2 c^2]}\right)} \tag{9}$$

Equation (10) defines the relationship between transmissibility (T_r), damping ratio (ζ), and frequency ratio (r) of a system.

$$T_r = \sqrt{\frac{1 + (2\zeta r)^2}{(1 - r^2)^2 + (2\zeta r)^2}} \tag{10}$$

The frequency ratio (r) and the damping ratio (ζ) are determined by equations (11) and (12), respectively.

$$r = \frac{\omega}{\omega_n} \tag{11}$$

$$\zeta = \frac{c}{2m\omega_n} \tag{12}$$

2.4. MATLAB/Simulink Modeling

Simulink simulations are based on the numerical solution of sets of ordinary differential equations that describe the dynamic behavior of a system. When considering the steady-state conditions of the model, the variable-step, fourth-order Runge-Kutta method, which is Simulink's default solver, ensures accurate capture of all dynamic characteristics of the system. This allows for precise determination of the parameters necessary for the design of systems such as vibration isolators and presses. The numerical solution of the differential motion equations in the MATLAB program is illustrated in Figures 6, 7, and 8, which show the visualized simulation models of the press system created in MATLAB Simulink.

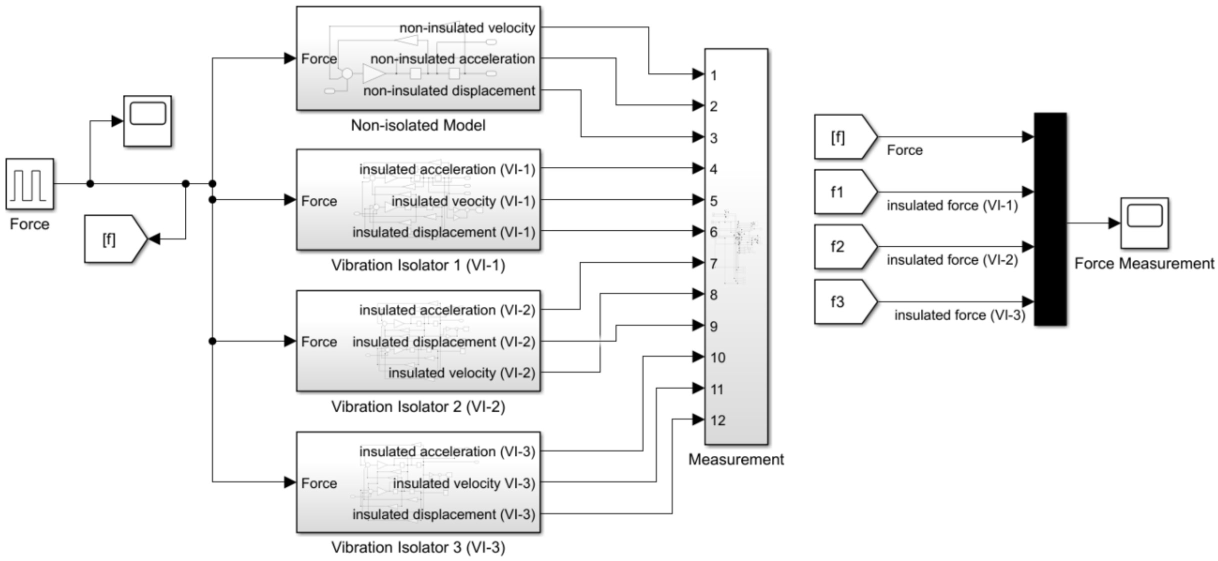


Figure 6. A Simulink-based simulation model of a press system

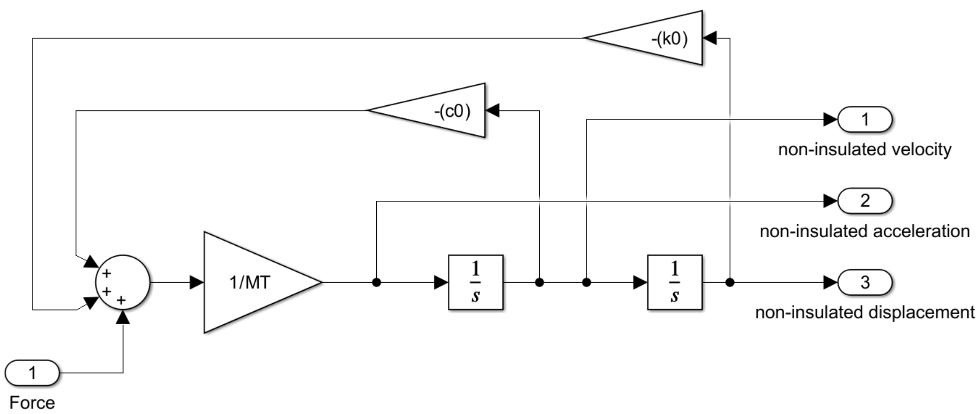


Figure 7. Simulink simulation model without isolation

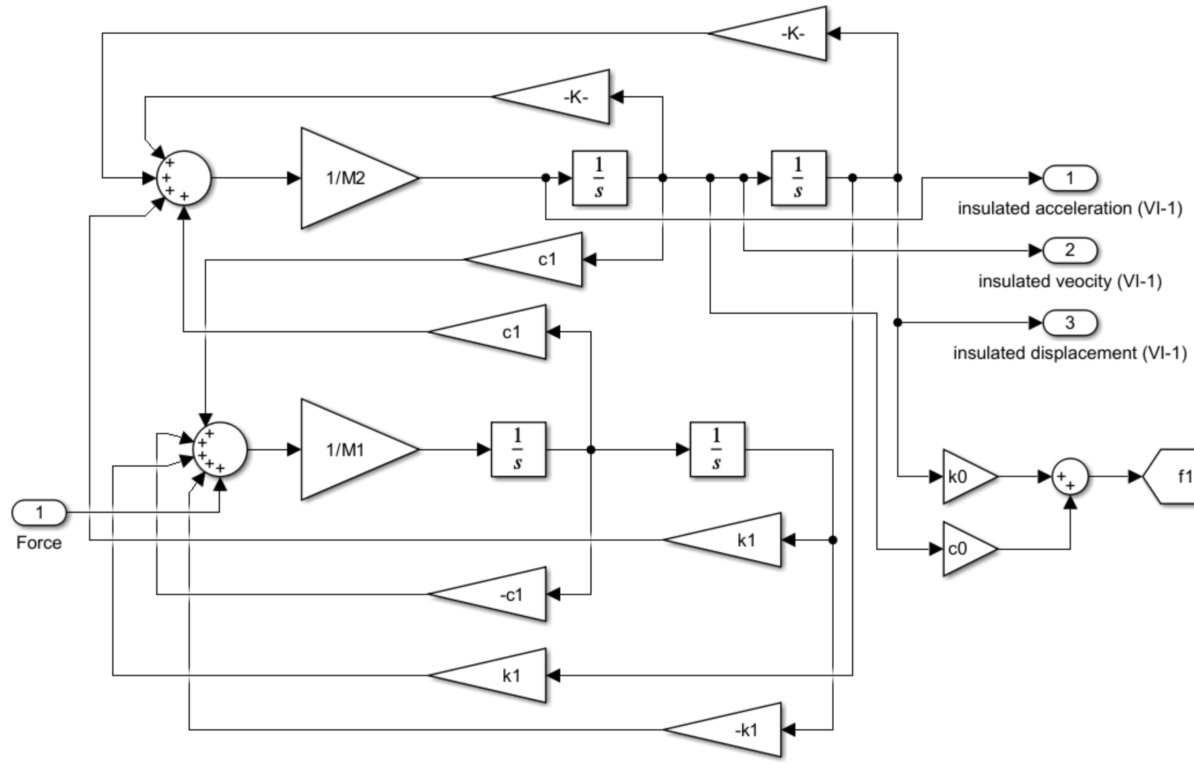


Figure 8. Simulink simulation model with isolation

The parameters used in the simulation model were determined based on the characteristic properties of the type C eccentric press and the actual foundation system. Based on the preliminary design results for stability and settlement, the foundation dimensions were determined to be 4m x 4m x 2m. The masses of the press and foundation block are presented in Table 1, while the stiffness and damping values of the soil and vibration isolators are given in Table 2. The stiffness ratio, defined as the ratio of the isolator stiffness to the soil stiffness, was selected as 0.01.

Table 1. Mass parameters of the press and foundation system

Parameters	Unit	Symbol	Value
Press	kg	m_1	6 000
Foundation	kg	m_2	40 000
Total mass	kg	m_t	46 000

Tablo 2. Stiffness and damping values

	Parameters	Unit	Symbol	Value
Soil	Stiffness	(N/m)	k_0	8.11×10^8
	Damping	(N.s/m)	c_0	1.13×10^7
Vibration Isolator 1 (VI-1)	Stiffness	(N/m)	k_1	8.11×10^6
	Damping	(N.s/m)	c_1	5.65×10^5
Vibration Isolator 1 (VI-2)	Stiffness	(N/m)	k_1	8.11×10^6
	Damping	(N.s/m)	c_1	3.36×10^5
Vibration Isolator 1 (VI-3)	Stiffness	(N/m)	k_1	8.11×10^6
	Damping	(N.s/m)	c_1	1.12×10^5

2.5. Experimental Study

An experimental study was undertaken to analyze the dynamic response of the system, a press machine, vibration isolator, and vibration test rig. In this study, the Solid 3D view (Figure 9) illustrating the details of the Type C eccentric press machine subjected to vibration isolation, along with its technical specifications (Table 3), are presented below. Vibration isolators with optimized stiffness and damping characteristics were chosen to mitigate vibration transmission.

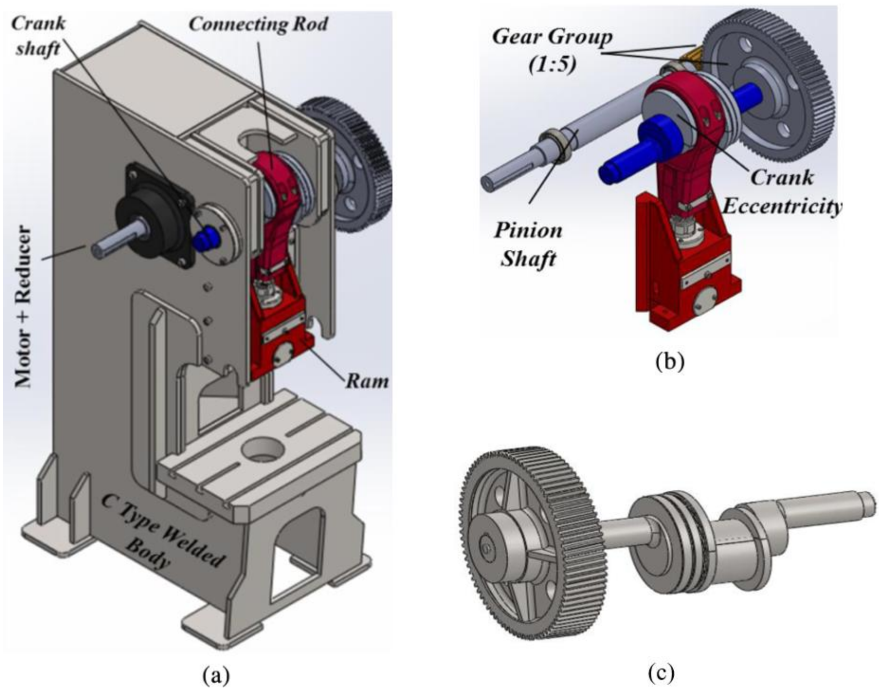


Figure 9. Solid model of the press: (a) complete assembly, (b) transmission mechanism, (c) solid CAD model of the crank assembly [34].

Tablo 3. Technical specifications of a press machine

Parameters	Unit	Value
Pressure	ton	100
Transfer	rpm	55
Stroke adjustment	mm	10-130
Connected max. mold height	mm	320
Table-ram distance	mm	450
Regulation setting	mm	80
Table size	mm	900x650
Engine power	kW	7,5
Engine speed	rpm	1450

2.5.1 Experimental setup and vibration analysis

To evaluate the performance of vibration isolators on a C-type eccentric press machine, an experimental setup, as shown in Figure 10, was established. Initially, the press was operated without isolators (Figure 10(a)) to measure baseline vibration values. Subsequently, vibration isolators were installed under the machine feet (Figure 10(b)), and the measurements were repeated. The vibrations generated during the cutting process of a 2 mm thick St 37 black sheet metal material in a cutting-bending compound die, as shown in Figure 10(c), were analyzed. Vibration measurements were conducted using the EXTECH VB500 – a four-channel vibration measurement and recording device, the technical specifications of which are provided in Table 4. In order to determine the effects of vibrations generated during the operation of the press machine on the operator and the surrounding environment, vibration data were measured from a reference point located 30 cm away from the press machine on the ground, representing the vibration level to which the operator is exposed. Throughout the experiments, care was taken to maintain a constant force on the machine. The experimental data were compared with the results of the theoretical model to gain a deeper understanding of the vibrations occurring during the cutting process.

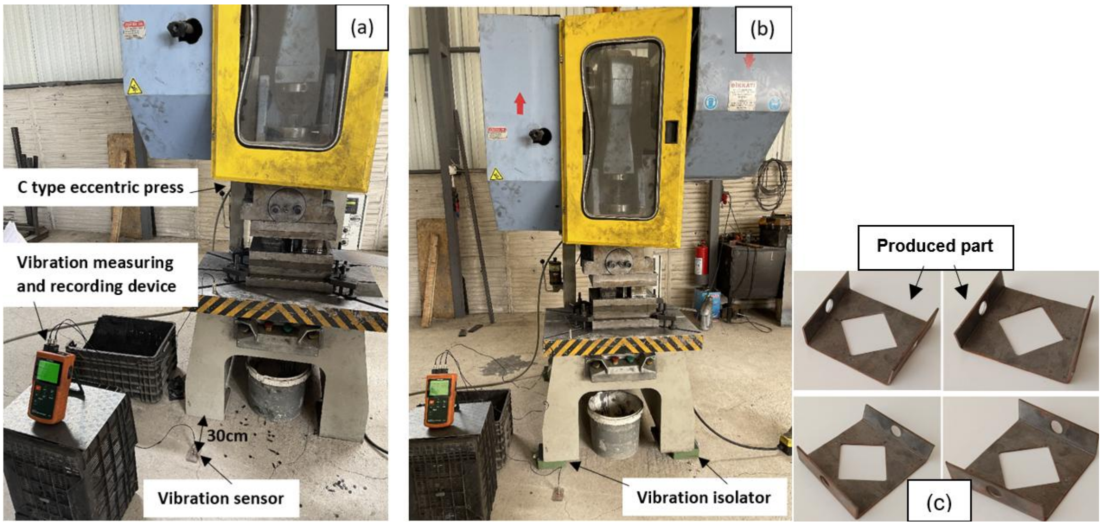


Figure 10. Vibration isolation of a type C eccentric press; (a) non-isolated, (b) isolated, (c) produced part

Table 4. Technical specifications of EXTECH VB500

Specifications	
Acceleration	656ft/s², 200m/s², 20.39g (peak)
Velocity	7.87in/s, 200mm/s, 19.99cm/s (peak)
Displacement	0.078in, 2mm (peak-to-peak)
Resolution	1ft/s2, 0.1m/s², 0.01g; 0.01in/s, 0.1mm/s, 0.01cm/s; 0.001in, 0.001mm
Basic accuracy	± (5%+2 digits)
Memory	20M data records using 2G SD card
Dimensions	8 x 3 x 1.5" (203 x 76 x 38mm)
Weight	1.13lbs (515g)

3. RESULTS AND DISCUSSION

3.1. Simulation of press using MATLAB/Simulink

The dynamic response of a press to a single impact load was investigated parametrically by varying the type of vibration isolator. The soil properties were assumed to be linear and characterized by stiffness and damping coefficients of $k_0 = 8.11 \times 10^8$ N/m and $c_0 = 1.13 \times 10^7$ N.s/m, respectively [1]. The impact duration, coefficient of restitution, and maximum force were defined based on. The effectiveness of isolation systems in mitigating ground vibrations was assessed by evaluating the maximum displacement, velocity, acceleration and force of the system.

Figure 11 illustrates the ground motion displacement values. The average peak ground displacement (PGD) in the uninsulated condition was measured to be 5.075×10^{-2} mm. When isolation was applied, the maximum PGD was reduced to 3.46×10^{-2} mm using isolator VI-1, and further reduced to 2.7×10^{-2} mm with isolator VI-3. The results indicate that VI-3 was most effective in mitigating ground vibrations. The parametric study revealed that the dynamic response of the system is highly sensitive to variations in the mass, stiffness, and damping ratio of the vibration isolator. This implies that vibration isolators provide a means to tailor the system response by adjusting critical parameters such as stiffness and damping to achieve the desired level of vibration isolation. These findings emphasize the significance of damping in enhancing the performance of vibration isolation systems.

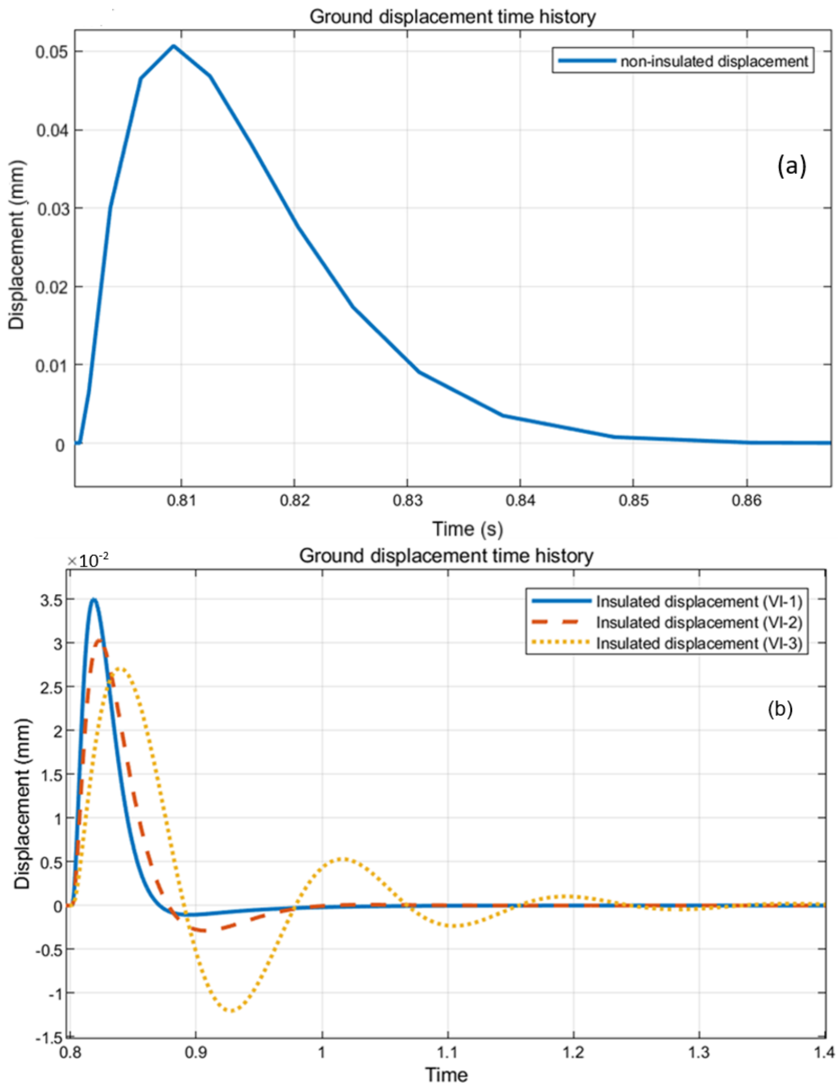


Figure 11. Time history of ground displacement: (a) without isolation, (b) with isolation

The velocity-time graph presented in Figure 12 demonstrates the effects of different vibration isolators (VI-1, VI-2, VI-3) on ground velocity variations. When the VI-1 vibration isolator was used, the maximum velocity was measured as 3.5 x 10^{-3} m/s. This indicates that the isolator reduces vibrations to a certain level but results in a higher velocity value compared to the other isolators. With the VI-2 vibration isolator, the maximum velocity was determined to be 2.45 x 10^{-3} m/s. This shows that it provides more effective vibration isolation than VI-1 and further reduces ground velocity. For the VI-3 vibration isolator, the maximum velocity was measured as 1.22 x 10^{-3} m/s. This demonstrates that it is the most effective among the three isolators, reducing ground velocity the most and controlling vibrations optimally. In conclusion, the graph reveals that the performance of vibration isolators varies depending on the type of isolator, with VI-3 being the most effective solution. This information can provide significant guidance in selecting the most suitable isolator for vibration control.

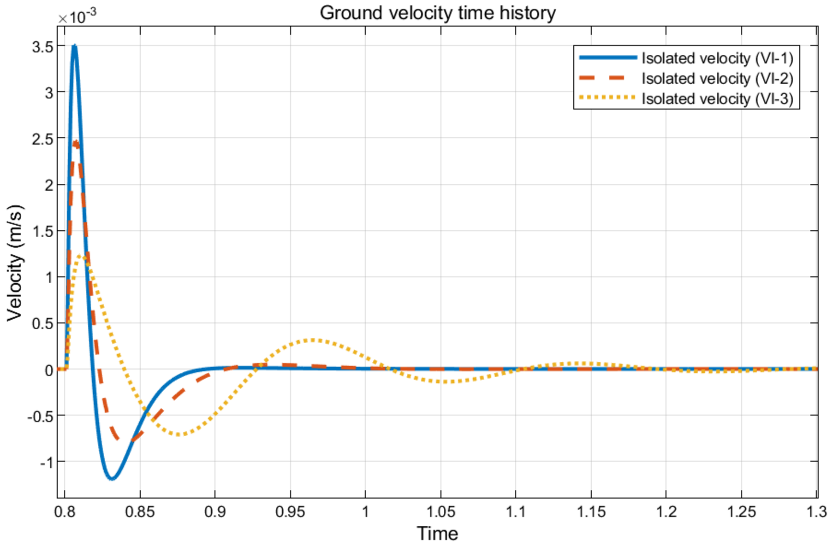


Figure 12. Time history of ground velocity

The graph presented in Figure 13 comparatively shows the ground vibration acceleration values of the press machine after the application of isolation. When examining the effects of isolation applications on vibration acceleration, the average peak value of ground vibration acceleration was measured as 1.62 m/s^2 when VI-1 was used. Although this value represents a significant improvement compared to the pre-isolation condition, even more effective results were achieved with VI-2 and VI-3. In the application using VI-2, the vibration acceleration decreased to 0.98 m/s^2, showing a noticeable reduction compared to VI-1. When VI-3 was used, the vibration acceleration was measured as 0.35 m/s^2, representing the lowest value among the vibration acceleration results. These results demonstrate that VI-3 is the most effective isolator for vibration control and can provide significant performance improvements in industrial applications. The trends in the graphs further support this improvement and highlight the success of isolation techniques in reducing vibration acceleration. Such isolators can both extend the lifespan of machinery and significantly reduce environmental vibration-related issues. In future studies, research on different operating conditions and isolator designs could be conducted to establish a broader database for vibration control. Such studies will contribute significantly to achieving more effective vibration control in industrial applications.

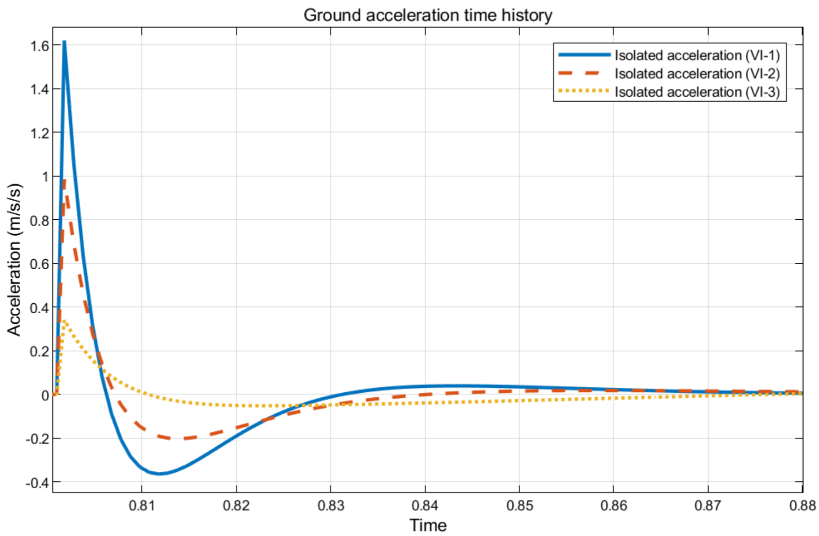


Figure 13. Time history of ground acceleration

Figure 14 illustrates the dynamic forces transmitted to the ground after the implementation of vibration isolation systems on a 100-ton press. The results indicate that the maximum average peak dynamic force transmitted to the ground was 5.14 x 10^4 N for VI-1, whereas the minimum was 2.57 x 10^4 N for VI-3. The lowest dynamic load on the ground was observed for the VI-3 type isolator. The parametric study demonstrated that selecting a stiffness ratio of 0.01 between the isolator and the soil can significantly reduce foundation vibrations. A decrease in the stiffness ratio results in a reduction in both the transmitted force and foundation response. However, this may lead to excessive vibration amplitudes of the press, compromising the performance. Conversely, an increase in the stiffness ratio will reduce the press vibration amplitude but may increase the foundation response and transmitted force.

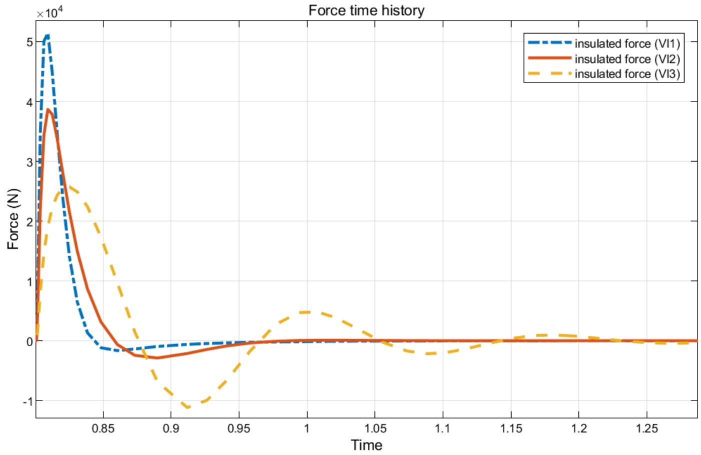


Figure 14. Time history of dynamic loads transmitted to the foundation

3.2. Comparison of simulation and experimental results

A comparative analysis of simulated and experimental dynamic responses of the press foundation is presented in Figure 15. For the isolated case with VI-3, the simulated average peak ground displacement was found to be 2.7 x 10^{-2} mm, which shows excellent agreement with the experimentally measured value of 2.5 x 10^{-2} mm. The high level of correlation between the numerical and experimental results provides strong evidence for the validity of the proposed dynamic model in capturing the actual behavior of the press under operating conditions. These findings emphasize the crucial role of dynamic modeling in the design and selection of appropriate vibration isolation systems.

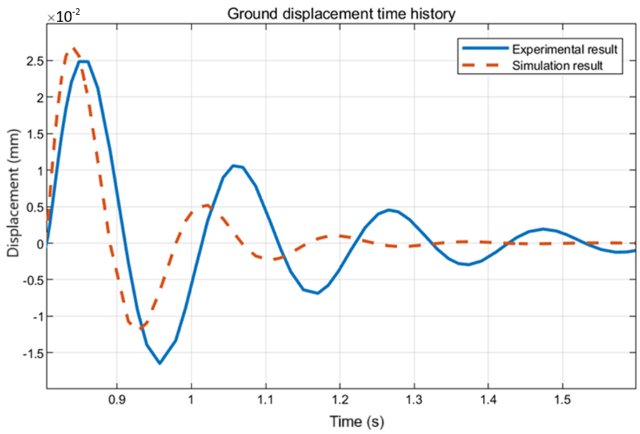


Figure 15. Comparison of simulated and experimental ground displacements

4. CONCLUSIONS

This study investigated the dynamic behavior of industrial C-type eccentric presses using MATLAB/Simulink modeling and simulation. The effectiveness of vibration isolators (VI-1, VI-2, VI-3) in reducing ground vibrations was evaluated under impact loading conditions. The results demonstrated that VI-3 was the most effective isolator, reducing peak ground displacement (PGD) from 5.075×10^{-2} mm (uninsulated) to 2.7×10^{-2} mm. Similarly, ground velocity and acceleration were minimized to 1.22×10^{-3} m/s and 0.35 m/s^2, respectively, using VI-3. The dynamic forces transmitted to the ground were also lowest with VI-3 (2.57×10^{4} N), highlighting its superior performance.

A parametric study revealed that the system's dynamic response is highly sensitive to isolator stiffness and damping. A stiffness ratio of 0.01 between the isolator and soil reduced foundation vibrations by approximately 46.8%. However, optimizing this ratio is critical to balance press vibration amplitudes and foundation response.

The simulation results showed excellent agreement with experimental data, validating the proposed dynamic model. For VI-3, the simulated PGD (2.7×10^{-2} mm) closely matched the experimental value (2.5×10^{-2} mm), confirming the model's accuracy.

These findings underscore the importance of dynamic modeling in designing effective vibration isolation systems. VI-3 emerged as the optimal solution for minimizing vibrations and enhancing press performance. Future studies should explore additional isolator designs and operating conditions to further improve vibration control in industrial applications.

Declaration of Ethical Standards

The authors of this article declare that the materials and methods used in this study do not require ethical committee permission and/or legal-special permission.

Credit Authorship Contribution Statement

Ahmet KÖKEN: Researched and supplied experimental materials, conducted the experiments, drafted the article, and wrote and reviewed the manuscript.

Abdurrahman KARABULUT: Conducted the experiments, reviewed the manuscript.

Declaration of Competing Interest

Declaration of Competing Interest The authors declare that they have no known competing financial interests or personal relationships that could have appeared to influence the work reported in this paper.

Funding / Acknowledgements

The authors have not disclosed any funding. The authors acknowledged that this study is derived from the PhD Thesis of Ahmet KÖKEN.

Data Availability

Data will be made available on request.

REFERENCES

[1] E. Zheng, X. Zhou, and S. Zhu, "Dynamic response analysis of block foundations with nonlinear dry friction mounting system to impact loads," *Journal of Mechanical Science and Technology*, vol. 28, no. 7, pp. 2535–2548, 2014, doi: 10.1007/s12206-014-0611-7.

[2] Y. Kara and H. Akbulut, "Mechanical behavior of helical springs made of carbon nanotube additive epoxy composite reinforced with carbon fiber," *Journal of the Faculty of Engineering and Architecture of Gazi University*, vol. 32, no. 2, pp. 417–427, 2017, doi: 10.17341/gazimmfd.322166.

[3] A. G. Chehab, M. Hesham, and E. Naggar, "Design of efficient base isolation for hammers and presses." *Soil Dynamics and Earthquake Engineering,* vol. 23, no. 2, pp. 127-141, 2003.

[4] E. Zheng, F. Jia, Z. Zhang, and J. Shi, "Dynamic modelling and response analysis of closed high-speed press system," *Proceedings of the Institution of Mechanical Engineers, Part K: Journal of Multi-body Dynamics*, vol. 226, no. 4, pp. 315–330, 2012, doi: 10.1177/1464419312445008.

[5] M. Heidari and M. H. El Naggar, "Using reinforced soil systems in hammer foundations," *Proceedings of the Institution of Civil Engineers: Ground Improvement*, vol. 163, no. 2, pp. 121–132, 2010, doi: 10.1680/grim.2010.163.2.121.

[6] F. Jia and F. Xu, "Dynamic analysis of closed high-speed precision press: Modeling, simulation and experiments," *Proc Inst Mech Eng C J Mech Eng Sci*, vol. 228, no. 13, pp. 2383–2401, 2014, doi: 10.1177/0954406213517093.

[7] D. Jancarczyk, I. Wróbel, P. Danielczyk, and M. Sidzina, "Enhancing vibration analysis in hydraulic presses: A case study evaluation," *Applied Sciences (Switzerland)*, vol. 14, no. 7, 2024, doi: 10.3390/app14073097.

[8] H. Dal and M. Baklacı, "Noise and vibration abatement study on a fabric polishing machine," *Gazi Journal of Engineering Sciences*, vol. 7, no. 2, pp. 121–133, 2021, doi: 10.30855/gmbd.2021.02.05.

[9] F. Wang *et al.*, "Research and application of vibration isolation platform based on nonlinear vibration isolation system," *J Sens.*, vol. 2023, 2023, doi: 10.1155/2023/9967142.

[10] S. Saberi, J. Fischer, M. Stockinger, R. Tikal, and R. Afsharnia, "Theoretical and experimental investigations of mechanical vibrations of hot hammer forging", *The International Journal of Advanced Manufacturing Technology*, vol. 114, pp. 3037-3045, 2021, doi: 10.1007/s00170-021-07061-y/Published.

[11] B. Kekeç and D. Ghiloufi, "Propagation characteristics of surface and in-depth vibrations in sand grounds: a comparative analysis," *Konya Journal of Engineering Sciences*, vol. 10, no. 1, pp. 1–17, 2022, doi: 10.36306/konjes.884110.

[12] A. G. Chehab and M. H. El Naggar, "Response of block foundations to impact loads," *J. Sound Vib.*, vol. 276, no. 1–2, pp. 293–310, 2004, doi: 10.1016/j.jsv.2003.07.028.

[13] Y. Avcı and U. Yazgan, "Maximum inelastic displacement ratio for systems with soil-structure interaction," *Journal of the Faculty of Engineering and Architecture of Gazi University*, vol. 34, no. 3, pp. 1527–1537, 2019, doi: 10.17341/gazimmfd.460502.

[14] U. Tekeci and B. Yıldırım, "Predicting fatigue life of a mount of a device with shock absorber," *Journal of Polytechnic*, vol. 27, no. 3, pp. 1005-1015, 2024, doi: 10.2339/politeknik.1210934.

[15] A. Köken, "A research on reduction of vibrations at foundation of press workbench using vibration isolator," *PhD thesis, Afyon Kocatepe University, Institute of Science and Technology*, 2023.

[16] G. Wang and Z. Dong, "Design optimization of low impact transmission foundation for forging hammers," *Engineering Computations (Swansea, Wales)*, vol. 23, no. 2, pp. 166–186, 2006, doi: 10.1108/02644400610644531.

[17] B. Hızarcı and Z. Kıral, "Active vibration control of engineering structures using air jet pulses," *Konya Journal of Engineering Sciences*, vol. 7, pp. 933–947, 2019, doi: 10.36306/konjes.624373.

[18] K. Alluhydan, A. T. El-Sayed, and F. T. El-Bahrawy, "The effect of proportional, proportional-integral, and proportional-integral-derivative controllers on improving the performance of torsional vibrations on a dynamical system," *Computation*, vol. 12, no. 8, 2024, doi: 10.3390/computation12080157.

[19] S. J. Zhu, Y. F. Zheng, and Y. M. Fu, "Analysis of non-linear dynamics of a two-degree-of-freedom vibration system with non-linear damping and non-linear spring," *J. Sound Vib.*, vol. 271, no. 1–2, pp. 15–24, 2004, doi: 10.1016/S0022-460X (03)00249-9.

[20] Z. Q. Lang, X. J. Jing, S. A. Billings, G. R. Tomlinson, and Z. K. Peng, "Theoretical study of the effects of nonlinear viscous damping on vibration isolation of sdof systems," *J. Sound Vib.*, vol. 323, no. 1–2, pp. 352–365, 2009, doi: 10.1016/j.jsv.2009.01.001.

[21] Y. Ping, Y. Jianming, and D. Jianning, "Dynamic transmissibility of a complex nonlinear coupling isolator," *Tsinghua Science and Technology,* vol. 11, no. 5, pp. 538-542, 2006.

[22] R. R. Kunadharaju and A. Borthakur, "Analysis and design of foundation systems to control the vibrations due to forging impact hammer," *Journal of Structural Engineering*, vol. 44, no. 5, pp. 404-413, 2017.

[23] M. Guo, B. Li, J. Yang, W. Li, and S. Y. Liang, "Active piezoelectric vibration isolation system of machine tools," in *Proc. 2015 Int. Conf. Electrical, Electronics and Mechatronics*, Atlantis Press, 2015, pp. 169–172. doi: 10.2991/iceem-15.2015.42.

[24] J. Kumar and V. Boora, "Dynamic response of a machine foundation in combination with spring mounting base and rubber pad," *Geotechnical and Geological Engineering,* vol. 27, no. 3, pp. 379–389, 2009, doi: 10.1007/s10706-008-9239-7.

[25] A. Köken and A. Karabulut, "The effect of vibration isolator on the dynamic response of machine tools," *Pamukkale University Journal of Engineering Sciences*, vol. 29, no. 1, pp. 104–109, 2023, doi: 10.5505/pajes.2022.68327.

[26] A. Köken and A. Karabulut, "Reducing the force transmitted to the ground by using vibration damper in machine tools", *Journal of Polytechnic*, vol. 25, no. 1, pp. 399-404, 2022. doi: 10.2339/politeknik.881839.

[27] A. Karabulut and A. Köken, "Investigation of the effect of the vibration wedge on the vibration isolation of the guillotine shears machine," *Gazi Journal of Engineering Sciences*, vol. 6, no. 3, pp. 210–216, 2020, doi: 10.30855/gmbd.2020.03.04.

[28] A. Abd-Elhamed, S. Alkhatib, and M. A. Dagher, "Closed-form solutions to investigate the nonlinear response of foundations supporting operating machines under blast loads," *Journal of Low Frequency Noise Vibration and Active Control*, vol. 42, no. 3, pp. 1162–1187, 2023, doi: 10.1177/14613484231174856.

[29] M. Kam and H. Saruhan, "Vibration damping capacity of deep cryogenic treated AISI 4140 steel shaft supported by rolling element bearings," *Materials Testing,* vol. 63, no. 8, pp. 742–747, 2021, doi: 10.1515/mt-2020-0118.

[30] M. Kam, H. Saruhan, U. Kabasakaloglu, and T. Guney, "Vibration damping capacity of a rotating shaft heat treated by various procedures," *Materials Testing,* vol. 63, no. 10, pp. 966–969, 2021, doi: 10.1515/mt-2021-0026.

[31] A. Trabka, "Effect of pulse shape and duration on dynamic response of a forging system," *Acta Mechanica et Automatica,* vol. 13, no. 4, pp. 226–232, 2020, doi: 10.2478/ama-2019-0030.

[32] S. S. Rao, *Mechanical Vibrations*, 5th ed., Prentice Hall, 2011.

[33] E. Kılıç and B. Kuşcu, "Vehicle suspension control with magnetic force," *Journal of the Faculty of Engineering and Architecture of Gazi University*, vol. 39, no. 1, pp. 649–664, 2023, doi: 10.17341/gazimmfd.1173153.

[34] R. Halicioglu, L. C. Dulger, A. T. Bozdana, "Structural design and analysis of a servo crank press," *Engineering Science and Technology, an International Journal* vol. 19, no. 4, pp. 2060–2072, 2016, http://dx.doi.org/10.1016/j.jestch.2016.08.008.