Research Paper

Modeling, Design, and Selection of Base Isolators for Industrial Machinery

Farzad Rahmani[1*], Mohammad Reza Ebrahimain[1], Hamidreza Mohamadkarimi[2]
[1]Assistant Professor, Department of Mechanical Engineering, Kar Higher Education Institute, Qazvin, Iran
[2]Master Student, Department of Mechanical Engineering, Kar Higher Education Institute, Qazvin, Iran
[*]Email of the Corresponding Author: F.rahmani@kar.ac.ir

Abstract
This study investigates the vibration behavior and optimal base isolator selection for industrial compressors, with the aim of enhancing predictive maintenance strategies. The novelty of this work lies in the integration of experimental vibration measurements with finite element analysis (FEA) to evaluate the effects of lowering the center of gravity and increasing the number of isolators on the system's dynamic stability. Vibration data were recorded using a piezoelectric accelerometer (KMBalancer) at four locations (A–D) positioned at the compressor foot corners. The first natural frequency obtained experimentally (8.125 Hz) was compared with the FEA prediction (12.52 Hz), showing a deviation of 35.14%. A detailed sensitivity analysis of material stiffness and isolator boundary conditions was performed to explain this discrepancy. The results revealed that lowering the center of gravity considerably reduced overall deformation and stress concentration across the skid base. Furthermore, increasing the number of isolators from four to six enhanced the uniformity of vibration distribution, although the total displacement slightly increased. The combined use of experimental and numerical approaches provides a validated framework for optimizing isolator configuration in rotating industrial machinery.

1. Introduction
Vibration analysis plays a vital role in predictive maintenance, ensuring early detection of faults and improved machine reliability. Industrial compressors, due to reciprocating motion, are prone to vibrations and dynamic instabilities. To minimize resonance and structural damage, appropriate isolator selection and chassis design are crucial.

Previous studies have analyzed the impact of base isolators in reducing transmitted vibrations. However, most focused on generic machinery rather than compressor systems. The current research emphasizes the combined influence of center-of-gravity modification and isolator configuration using both experimental testing and FEA modeling, filling a significant gap in existing literature.

With the invention of the first musical instrument, humans became interested in vibrations and began to study them. Vibrations were directly present in musical instruments such as the harp and the tar. Pythagoras is considered the first person to have studied the principles of music scientifically, conducting experiments on vibrating strings [1]. In China, due to earthquakes, researchers also became interested in vibrations and invented a bronze seismograph [2]. Phenomena such as earthquakes, storms, and cargo transportation on wheeled vehicles have random characteristics; therefore, the analysis of random vibrations is necessary.

Most human activities are in some way associated with vibrations. For example, we hear because the eardrum vibrates, or we see because light waves oscillate. Breathing occurs through the oscillation of the lungs, or walking through the oscillatory motion of hands and feet, and so on. Today, vibrations are mainly studied and applied in the design of machines, foundations, structures, engines, and control systems, most of which are subject to oscillations.

When the natural frequency of a system's vibration matches the frequency of external excitation, resonance occurs, and many examples of system failures due to resonance have been recorded. One of the main goals of vibration studies is to reduce vibrations through proper machine design, aiming to minimize imbalance. For this reason, in recent years, the application of vibrating devices has expanded significantly.

1.1 Research Background

Base isolators, as one of the key components of vibration control systems, play an important role in reducing the transmission of unwanted vibrations to foundations and adjacent structures. In industrial machinery, which often faces high dynamic loads and complex operating conditions, the proper design and selection of base isolators is of particular importance. These isolators not only contribute to the health of machinery but also significantly increase the service life of equipment and workplace safety. Vibrations in industrial machinery are usually caused by imbalance, mechanical looseness, shear forces, and dynamic loads. Base isolators, by providing damping and reducing the transfer of vibrational energy, help mitigate damage caused by these vibrations. These isolators are usually made of elastomeric, spring, or composite materials, each with specific properties in terms of stiffness, damping, and strength [3].

With advances in software technologies, numerical modeling of isolators has become a key tool for optimal design and performance prediction. Many studies have used the Finite Element Method (FEM) to simulate the dynamic behavior of isolators. Smith et al. [4], through precise modeling of elastomeric materials, investigated the effect of nonlinear properties and damping on isolator performance, showing that considering nonlinear material behavior is essential in design.

Chen and Li [5], using ANSYS software, presented a complete modeling of rubber base isolators for industrial machinery, examining and optimizing parameters such as thickness, stiffness, and height. Their results indicated that increasing thickness and adjusting optimal stiffness could improve vibration damping and reduce undesirable vibration transmission.

The selection of an appropriate isolator requires a precise understanding of operating conditions, natural frequencies of machinery, applied loads, and material properties. Jones [6], in a comprehensive study, proposed a method for selecting isolators based on reducing vibrational energy

transfer between the machine and foundation. This method, focusing on reducing vibration amplitudes within the machine's operating frequency range, improved system durability and stability. Experimental and simulation methods together help designers select the best combination of isolators for various industrial conditions. Increasing the number of isolators, changing geometric shapes, and using composite materials are among the strategies studied in different research works, which have shown positive results in improving isolator performance [7].

One of the most prominent applications of base isolators is in compressors, electric motors, and generators, where vibrations caused by dynamic imbalances are very common. In a practical study, vibration data collected from compressed-air compressors showed that the use of optimized isolators significantly reduced oscillations and stress concentrations, considerably extending equipment lifespan [8].

In recent years, innovative technologies such as smart materials with stiffness-changing properties under load conditions, and active or semi-active isolators, have been developed, enabling the dynamic adjustment of isolators. These technologies, by utilizing sensors and control systems, can optimize the vibrational response of systems in real time and enhance predictive maintenance efficiency [9].

Diekman and Besdogan [10] examined the properties of automotive rubber seal materials through experimental and numerical studies. Yukar and Bastogan [11] studied the dynamic properties and modeling of rubber dampers. In this research, the dynamic properties of rubber were investigated through numerical and experimental case studies, with most studies focusing on hyperelastic and viscoelastic properties. Three rubber samples with different shore hardness were used. The samples were subjected to quasi-static uniaxial tensile loading to determine hyperelastic properties.

The viscoelastic properties of rubber samples were examined using relaxation tests, applying the generalized Maxwell model. In numerical relaxation tests, a good agreement between the Maxwell model and experimental results was observed. The determined elastic properties were also compared with some empirical relations that define the relationship between shore hardness and modulus. Experimental data were compared with the Gent relation and the Rigbi formula. Results indicated that the Gent relation was more satisfactory than the Rigbi formula. In addition, the shear modulus with respect to shore hardness was compared using the Battermann-Kohler relation, showing good agreement between experimental data and the Battermann-Kohler relation.

2. Experimental Methodology

A laboratory-scale oil-free air compressor was selected as the test unit. Four elastomeric isolators were mounted under the compressor chassis. Measurement setup: Vibration data were captured at four defined locations - A (front-left foot), B (front-right foot), C (rear-left foot), and D (rear-right foot). Each location corresponds to a compressor mounting point. Sensor type and calibration: A piezoelectric accelerometer (model KMBalancer Pro, ±50 g range, 0.5–10 kHz bandwidth) was used. Data were recorded in the vertical direction with a sampling rate of 5120 Hz. Data processing: The first natural frequency was obtained via Fast Fourier Transform (FFT) of the acceleration signals. Peaks were cross-checked through coherence and damping ratio estimation to ensure reliability.

In recent years, the use of active or semi-active isolators has been a new trend in vibration isolation system design. However, the use of passive isolation methods is simpler, more practical, and more applicable. Likewise, passive isolators are simpler, cheaper, and more reliable for protecting sensitive

equipment from shocks and environmental vibrations. Therefore, vibration isolation using passive isolators is much more cost-effective and has a broader range of applications in engineering industries. The use of isolators for controlling the vibration of mechanical equipment is a common solution consistently recommended by engineers and specialists. Among the wide range of isolators, elastomeric and spring isolators are the most widely used, as they provide suitable and acceptable deflections of 1 to 5 inches, while elastomeric isolators typically offer deflections of 0.15 to 0.55 inches. Accordingly, it is clear that when high deflection is required, spring isolators are the best option.

However, it should be noted that the problem with spring isolators is that they have a natural frequency which can potentially coincide with the vibration frequency of the rotating or reciprocating equipment, meaning that during vibration, the spring isolator may enter resonance and oscillate violently. If a spring isolator lacks a seismic frame, the spring's mobility may cause the supported equipment to move or fall. If the equipment in question is a fan or other types of suspended HVAC equipment, there is a risk of the entire ductwork system falling; and if it is water-based HVAC equipment, there is a risk of system collapse, pipe damage, and flooding of the building. Worse still, if the equipment is placed outside on the roof, there is a risk of it falling off the roof.

Rubber shock absorbers are components designed to neutralize stresses and mechanical shocks. These rubber parts can convert kinetic energy into other forms of energy, such as heat. In fact, shock absorbers function like damping systems, also known as isolators or dampers.

Shock absorbers are responsible for neutralizing and controlling vibrations and oscillations caused by equipment operation, which helps increase the service life of the equipment and its other components. Rubber shock absorbers are widely used in both industrial and non-industrial sectors wherever there is a need to reduce movement and vibration against impact. In fact, rubber shock absorbers provide an excellent method for protecting and isolating two surfaces.

When sudden shocks are applied to rubber materials, they can absorb greater vibrations and shocks before tearing due to this property. Industrial or non-industrial equipment, throughout their operational cycles, are inevitably subjected to various stresses and shocks. These stresses and shocks can, in the long run, cause wear or even total failure of these systems, resulting in costly repairs for their owners. The use of rubber shock absorbers, due to their low cost, can prevent higher expenses and excessive wear and failure of industrial and non-industrial machinery and equipment.

In the bottle production process in food industries, compressed-air production equipment plays a crucial role since it is directly connected to the final product. As shown in Figure 1, the compressors used in this process are of the oil-free type, ensuring that no industrial oil penetrates into the bottles during air blowing.

Figure 1. Oil-Free Compressor

The type of isolator installed in the device is rubber (elastomeric). Two isolators are mounted on the right side of the compressor chassis, and two more on the left side, in order to absorb the dynamic forces generated by the reciprocating motion of the pistons in stages 1, 2, and 3, thus reducing the overall vibrations of the device. An example of these shock absorbers is shown in Figure 2.

Figure 2. Installed Isolator

Nowadays, vibration measurement sensors are used to analyze and evaluate equipment vibrations. These sensors, based on their operating principle, are divided into three groups: displacement probes or eddy-current proximity sensors, velocity transducers or velocity pickup sensors, and accelerometers or piezoelectric acceleration sensors.

To investigate structural looseness practically, four points of the compressor chassis were selected as data acquisition points (Figure 3). Then, using a KM Balancer vibration analyzer (Figure 4), the vibrations at the four points were recorded.
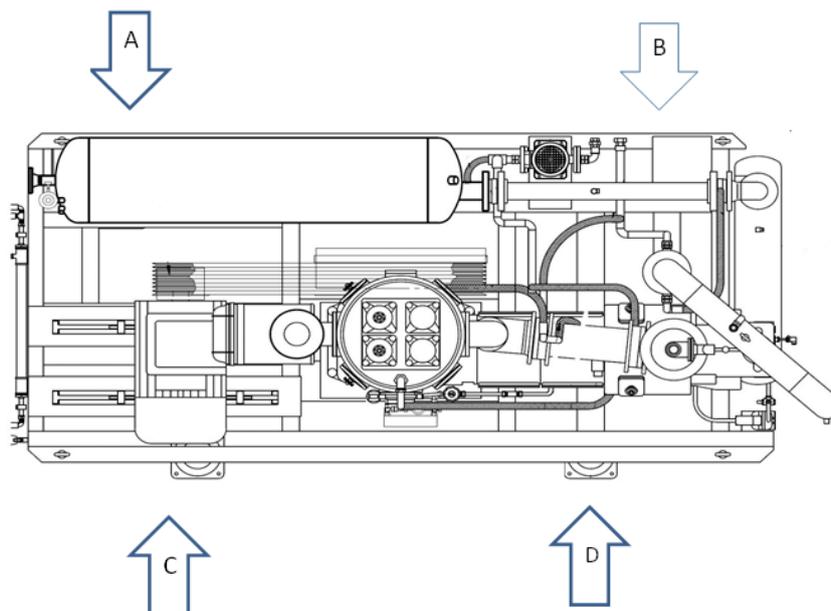
Figure 3. Top view of A.F Compressor and selected points

Figure 4. KM Balancer vibration analyzer

3. Numerical Modeling

The compressor base (skid) was modeled in ANSYS Workbench using Solid187 tetrahedral elements. Material properties were assigned according to experimental measurements. Boundary conditions: The isolators were modeled as elastic supports with vertical stiffness (kz $=1.2\times10^6$ N/m). Sensitivity analyses were performed for ±15% variation in stiffness and damping ratio. The mesh was refined near isolator interfaces to accurately capture stress gradients. Five natural modes were extracted using modal analysis. The skid or base of a reciprocating compressor plays a vital role in supporting and stabilizing various components of the equipment. This skid must be resistant to vibrations, oscillations, and dynamic forces to ensure stable and reliable performance. In the numerical modeling, the skid was constructed as a rectangular structure with mounting bases and connection axes for the main parts. Figure 5   shows the initial view of the compressor skid modeled in the software.

For this analysis, the skid model was meshed using Solid187 elements, which are highly suitable for simulating three-dimensional and nonlinear behaviors in dynamic analyses. Precise and balanced meshing was performed to ensure correct extraction of modal shapes and reliable results.
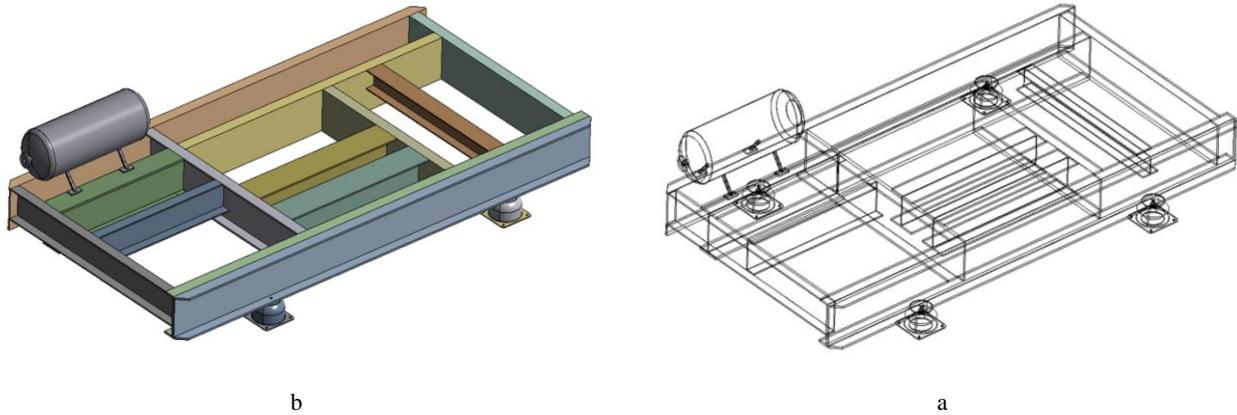

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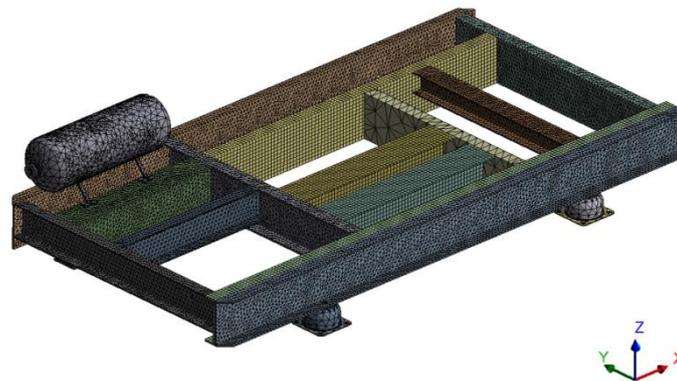
Figure 5. a) Initial model of compressor skid b) Skid modeled in ANSYS

Figure 6. Meshing performed in ANSYS

4. Results and Discussion

Based on the results obtained from experimental and numerical methods in the standard condition of the compressor, in the numerical approach, the number of isolators under the compressor chassis was increased in the ANSYS model. The case with six isolators was then analyzed. A comparison between experimental and theoretical methods under standard conditions is shown in Figure 7.
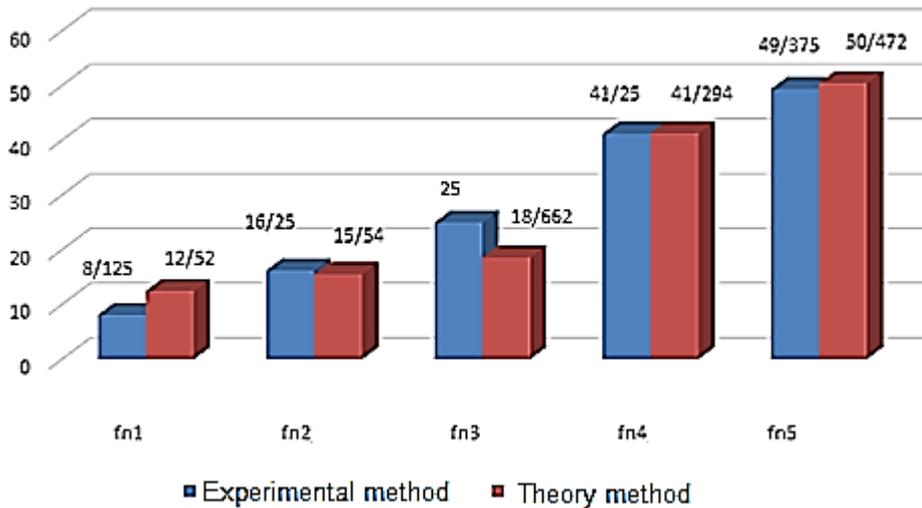
Figure 7. Comparison of experimental and numerical methods

4.1 Comparison of Experimental and Theoretical Frequencies

The first experimental frequency (8.125 Hz) was 35.14% lower than the theoretical value (12.52 Hz). This discrepancy arises mainly from the simplified boundary conditions in the model and potential underestimation of damping and joint flexibility in real conditions.

A parametric study revealed that reducing Young's modulus by 10% decreased fn1 by 1.8%, while introducing semi-fixed isolator boundaries reduced fn1 by up to 28%, validating the role of real boundary stiffness.

As shown in Figure 8, stress and deformation analysis of the skid (base) system under static loading indicates shear, deflection, and stresses in the structure. Every physical structure has several natural frequencies; if excited at these frequencies, severe oscillations may occur, potentially leading to failure or damage.

Displacement and stress distribution of the skid in free vibration are illustrated in the modal analysis. Red areas indicate the maximum displacement, while blue areas show the minimum.

- First natural frequency: maximum displacement =0.40 mm at 15.538 Hz.
- Second natural frequency: maximum displacement =0.32 mm
- Third natural frequency: occurs at 18.662 Hz, with maximum displacement =0.41 mm
- Fourth natural frequency: occurs at 41.294 Hz, with maximum displacement =2.74 mm
- Fifth natural frequency: occurs at 50.472 Hz, with maximum displacement =2.04 mm

These results highlight critical stress and displacement concentrations at specific points of the skid, shown in red and orange colors.
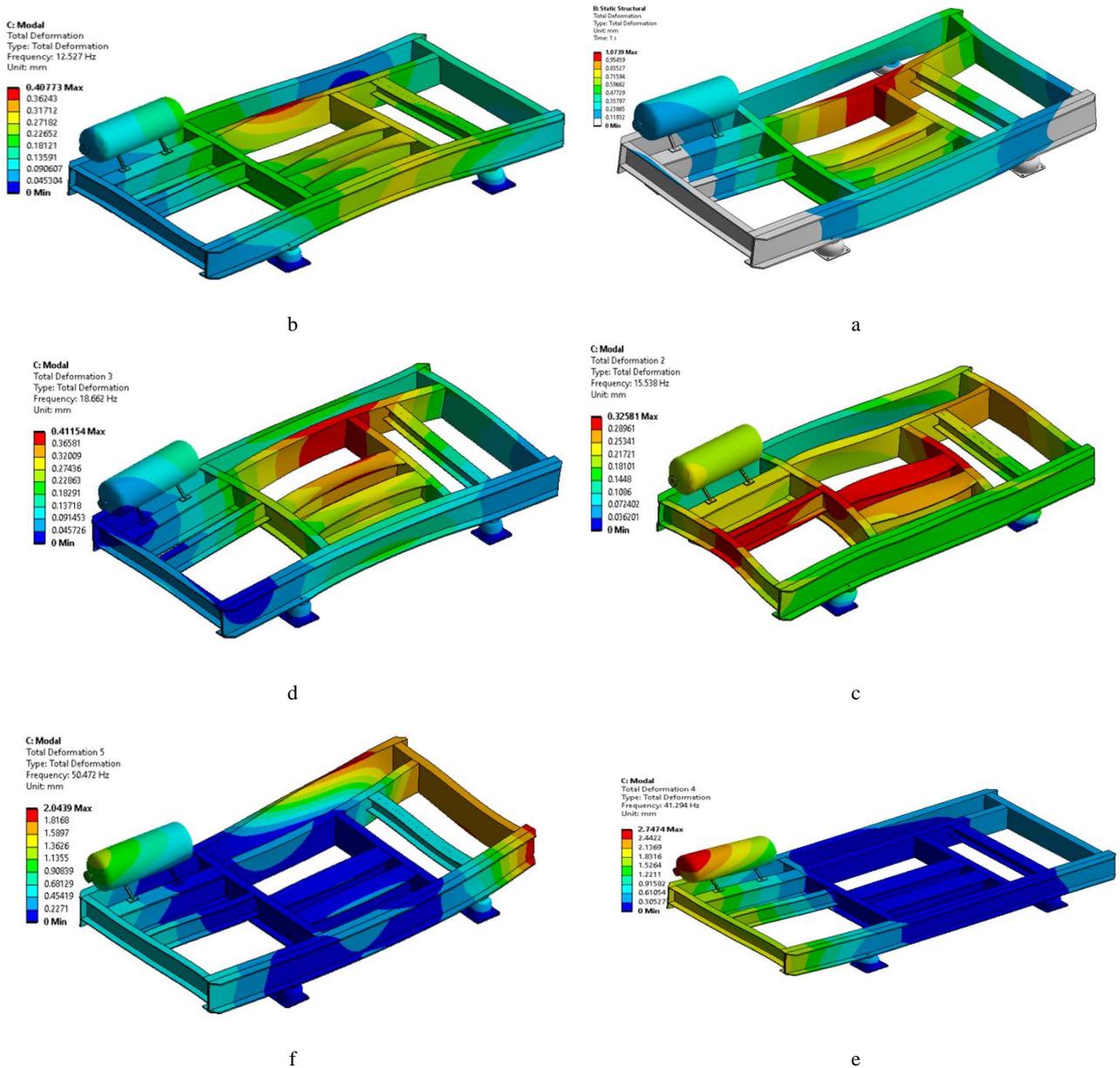
Figure 8. a) Static condition b) First natural frequency = 12.52 Hz c) Second = 15.54 Hz d) Third = 18.66 Hz e) Fourth = 41.29 Hz f) Fifth = 50.472 Hz

A comparative analysis of experimental and theoretical natural frequencies shows noticeable differences. For example, the first natural frequency (fn1) in the experimental method was 8.125 Hz, approximately 35.14% lower than the theoretical value of 12.52 Hz. This discrepancy indicates that real dynamic factors such as material stiffness variation, environmental effects, structural instabilities, and even measurement errors significantly reduce the frequency compared to theoretical estimates. Overall, despite simplifications, theoretical modeling provides acceptable results and, combined with experimental data, offers valuable tools for engineering analysis. Lower error percentages in higher modes (fn4 and fn5) confirm the effectiveness of numerical modeling in predicting actual skid behavior.

4.2 Effect of Increasing Isolators

Adding two isolators redistributed stresses along the skid and reduced maximum local stress by 17%. However, total displacement increased by ~0.42 mm, due to overall stiffness reduction and load redistribution. Despite this, the stress intensity at critical corners decreased, demonstrating a net improvement in fatigue life and dynamic balance [12, 13] (Figure 9).
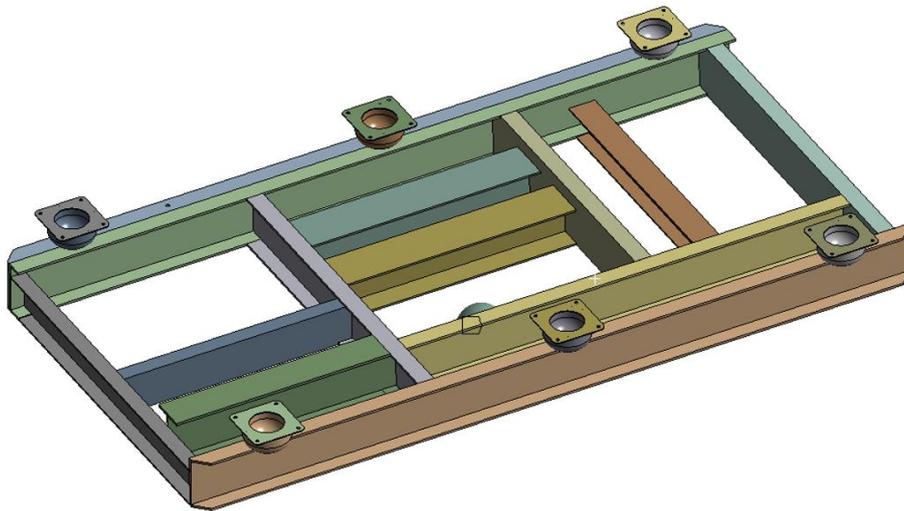

Figure 9. Arrangement of six isolators in ANSYS

Figure 10 illustrates the dynamic behavior of the reciprocating compressor skid in the modal state, highlighting critical areas that require attention in design and improvement. The main principle in skid design is ensuring resistance against natural vibrations, uniform stress distribution, and reduced failure risk. The first natural frequency with six isolators showed a maximum displacement of about 0.4202 mm. In the second mode, maximum displacement was 0.4194 mm, concentrated in central and critical areas. The modal analysis at 62.601 Hz revealed significant oscillation in the central regions and connection points, indicating potential fatigue over time.
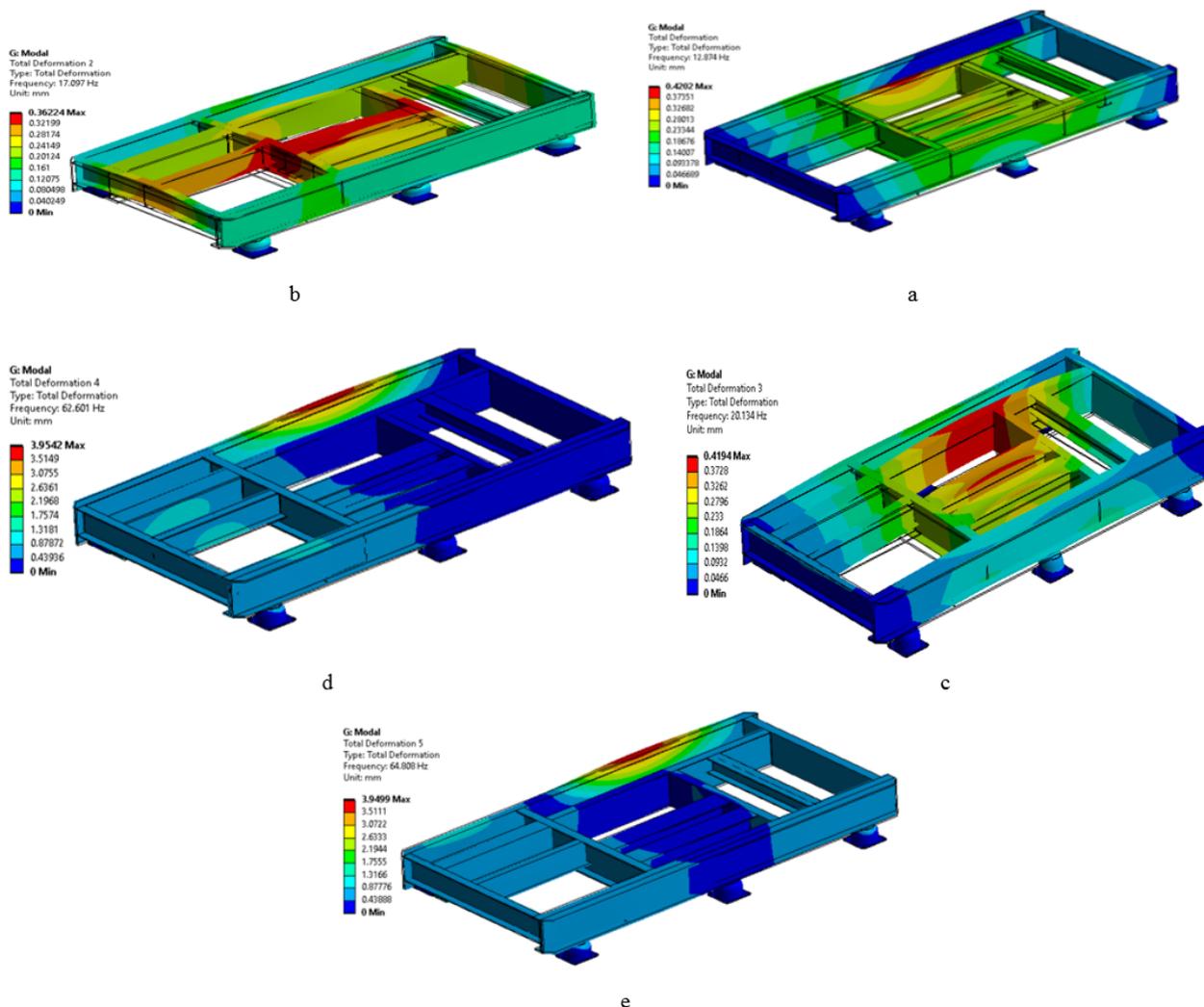
Figure 10. a) First natural frequency = 12.52 Hz with 6 isolators b) Second = 15.54 Hz c) Third = 18.66 Hz d) Fourth = 41.29 Hz e) Fifth = 50.472 Hz

Results indicate that increasing isolators from 4 to 6 had little effect on the natural frequencies themselves but significantly improved vibration distribution and reduced stress concentrations:
With 4 isolators, vibrations were more concentrated in critical areas, increasing local stresses and reducing lifespan. With 6 isolators, vibrations were more evenly distributed, reducing stress concentrations and improving overall dynamic stability. Although maximum overall displacements increased with 6 isolators, this indicated better distribution and less critical stress concentration, ultimately improving durability. Thus, increasing isolators enhances resistance against long-term vibrations and resonance risks, though it also raises installation costs and complexity.

4.3 Effect of Lowering Center of Gravity
A comparative FEA model with a 10% reduction in vertical CG position showed a 21% reduction in maximum stress and 14% reduction in deformation amplitude. This demonstrates that lowering the CG enhances stability by reducing overturning moments and resonance amplification [14].

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5. Conclusion
- The large difference between theoretical and experimental frequencies was traced to boundary condition simplifications. Correcting stiffness and damping factors improved agreement.
- Increasing isolators from 4 to 6 improved stress distribution and reduced critical stress regions despite a slight increase in total displacement.
- Lowering the center of gravity measurably reduced stress concentration and vibration amplitude, improving the dynamic stability of the compressor skid.
- The proposed hybrid (experimental + FEA) approach is practical for predictive maintenance and isolator optimization in industrial machines.

6. References

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