Design and finite element analysis of a gantry-type inspection machine frame driven by linear motors

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Modal Analysis,
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ABSTRACT

This study presents the design and finite element analysis (FEA) of a gantry-type inspection machine frame actuated by three linear motor axes-two along the Y-axis direction and one wall-mounted X-axis supported through precision brackets. The frame is intended for top-surface inspection applications that demand high structural rigidity and vibration stability. The mechanical design was evaluated using ANSYS Workbench 18.0 through both static structural and modal analyses. The simulation results show a maximum deformation of 4.47 µm, a peak von Mises stress of 3.35 N/mm², and a first natural frequency of 49.90 Hz under combined static and dynamic loading. The outcomes confirm that the developed structure exhibits sufficient stiffness, strength, and dynamic robustness, making it suitable for integration into high-precision inspection systems.

1. Introduction

Automated inspection systems have become an integral part of modern manufacturing, particularly in applications where dimensional accuracy and repeatability are critical. While sensing and vision technologies determine detection capability, the overall performance of such systems is strongly influenced by the mechanical integrity of the supporting structure. Even small structural deflections or vibration-induced disturbances can compromise positioning accuracy, thereby affecting measurement consistency. For this reason, the structural design of inspection machines must ensure high rigidity and stable dynamic behavior under operational loading conditions.

Gantry-type configurations are commonly adopted in precision equipment due to their symmetric load distribution, large working envelope, and suitability for linear motor actuation. However, achieving the required stiffness in such structures—especially when weight and manufacturability constraints are present—remains a challenging task. Previous research has shown that finite element-based evaluation is an effective approach for improving structural performance in machine tool frames and gantry systems (Altintas et al., 2005; Chan et al., 2023). These studies underline the necessity of assessing both static deformation and modal characteristics during the design stage to avoid loss of accuracy during high-speed motion.

In addition to static rigidity, dynamic behavior plays a crucial role in systems driven by linear motors, where motion-induced forces and structural excitation may influence performance. The integration of structural dynamics into machine design has been emphasized in the literature as an essential step toward ensuring vibration-resistant operation (Timoshenko & Goodier, 1951). Electromagnetic excitation characteristics specific to linear motor drives further necessitate careful structural evaluation, as periodic magnetic force interactions can generate harmonic excitations that must remain separated from the structural natural frequencies (Huang et al., 2018; Rao, 2018). Consequently, combined static and modal analyses have become a practical and reliable method for validating structural adequacy in precision machine applications.

Gantry-type machine structures have been studied extensively with respect to material selection, topology optimization, and stiffness-to-weight ratios (Mohring et al., 2015; Liu et al., 2018). In addition, the position-dependent dynamic behavior of machine tool assemblies has been explored to enable rapid design evaluation Law et al. (2013), while deformation reconstruction

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methods using fiber-optic sensors have demonstrated real-time structural monitoring potential in heavy gantry systems. These studies collectively reinforce the importance of rigorous FEA-based evaluation during the design phase.

In this context, the present work focuses on the design and finite element evaluation of a gantry-type inspection machine frame actuated by linear motors. The system incorporates two horizontally mounted Y-axes supporting a wall-mounted X-axis intended for top-surface inspection. Static structural and modal analyses were performed using ANSYS Workbench 18.0 to examine deformation, stress distribution, and natural frequencies. The objective is to verify whether the developed structure satisfies the stiffness and dynamic stability requirements necessary for high-precision inspection tasks.

2. Machine Description and Design Objectives

The inspection machine consists of three linear motor-driven axes designated as Y1, Y2, and X. The Y1 and Y2 axes are horizontally mounted on the main machine frame, while the X-axis is wall-mounted and supported on both Y-axes through precision brackets. This configuration forms a gantry-type architecture that provides a wide working envelope and high positional accuracy, making it suitable for top-surface inspection of precision components. The X-axis assembly weighs approximately 150 kg and supports an inspection unit (payload) of 25 kg. The complete load-bearing frame, excluding the enclosure, weighs nearly 900 kg. For simulation and structural evaluation, a set of representative design and operating parameters was adopted to approximate realistic conditions. These included a stroke length of 1.0 m for each axis, a maximum motor speed of 0.5 m/s, and a maximum acceleration of 2.5 m/s^2. The payload on the X-axis was taken as 25 kg. These parameters were considered adequate to capture the expected operational behavior while remaining independent of any specific commercial motor specifications. The derived conditions enabled accurate estimation of frame stiffness, strength, and dynamic characteristics through finite element analysis (FEA).

The principal design objectives of the study were:

- To minimize static deformation under payload and operational loads.
- To maintain von Mises stress levels well below the yield strength of the selected materials.
- To ensure that the first natural frequency remains significantly higher than the excitation frequencies generated by the linear motors.
- To develop a rigid yet manufacturable frame using standard steel profiles and conventional fabrication practices.

3. Materials and Construction

The mechanical structure of the inspection machine was designed with a focus on achieving an optimal balance between stiffness, weight, and manufacturability. The base frame and X-axis beam were fabricated using mild steel square and rectangular hollow tubes, selected for their favorable combination of high bending stiffness, good weldability, and cost efficiency. These hollow sections provide a high stiffness-to-weight ratio, allowing the structure to withstand operational loads with minimal deformation while keeping the overall machine mass moderate. Material selection plays a critical role in machine tool structural performance, as detailed in the broader literature on materials in machine tool structures (Mohring et al., 2015).

The gantry-axis fixtures, such as base plates, mounting brackets, and clamping supports, were made from C45 steel, a medium carbon steel known for its superior tensile strength, hardness, and dimensional stability after machining. These components form the critical load-transmission interfaces between the linear motor stages and the main frame, demanding higher localized rigidity compared to the base structure.

All frame members were welded, stress-relieved, and precisely machined to maintain accurate alignment of linear guideways, motor mounts, and encoder brackets. The welded joints were subsequently finish-machined to ensure flatness and parallelism at mounting locations. This approach minimizes assembly-induced errors and enhances the mechanical repeatability of the system.

Furthermore, the use of standard steel profiles and modular fixture interfaces simplifies manufacturing, reduces fabrication lead time, and supports easy replacement or modification of subassemblies during the development stage. The overall construction approach ensures that the machine frame offers both structural integrity and practical serviceability, essential for a high-precision inspection environment.

4. FEA Simulation Setup

Finite element simulations were carried out using ANSYS Workbench 18.0 to evaluate the static and dynamic performance of the gantry-type inspection machine frame. The complete CAD model of the system is shown in Fig. 1, illustrating the two horizontally mounted Y-axes positioned on the main frame and the wall-mounted X-axis supported on Y1 and Y2 through precision brackets.

A static structural analysis was performed to estimate the deformation and stress distribution under combined loading conditions. The von Mises stress distribution, presented in Fig. 2, highlights stress concentrations primarily near the fixture interfaces and mounting bracket regions of the X-axis support. The total deformation contour shown in Fig. 3 indicates that the maximum deflection occurs at the center of the X-axis beam, while smaller deflections appear near the bracket interfaces due to localized flexibility.

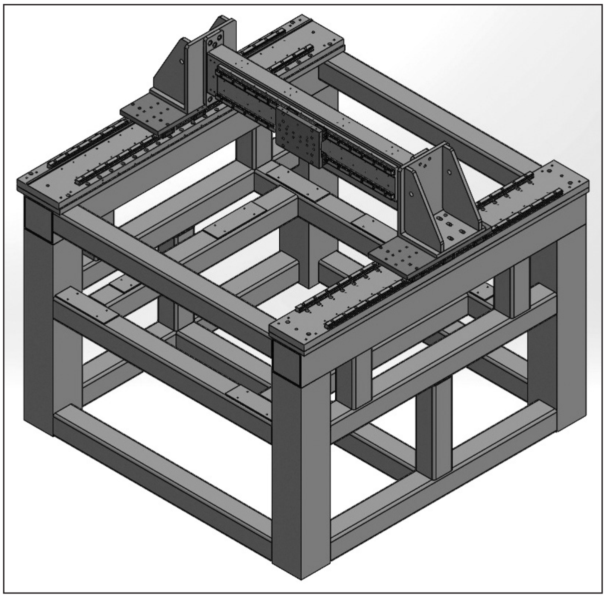

Fig. 1. CAD model of the gantry inspection machine frame.

A modal analysis was subsequently conducted to determine the natural frequencies and corresponding mode shapes of the entire assembly. The first mode shape, illustrated in Fig. 4, corresponds to a frequency of 49.90 Hz, which represents the dominant lateral vibration mode of the structure.

4.1 Load calculations and boundary conditions

Boundary conditions were defined by applying fixed supports to the base surfaces of the four legs of the frame. The applied loads on the X-axis plate included a static vertical load and a dynamic horizontal load, calculated as follows:

Static Vertical Load

- Payload mass = 25 kg
- Factor of Safety (FOS) = 1.4
- Equivalent mass = 25 × 1.4 = 35 kg
- Acceleration due to gravity = 9.81 m/s²

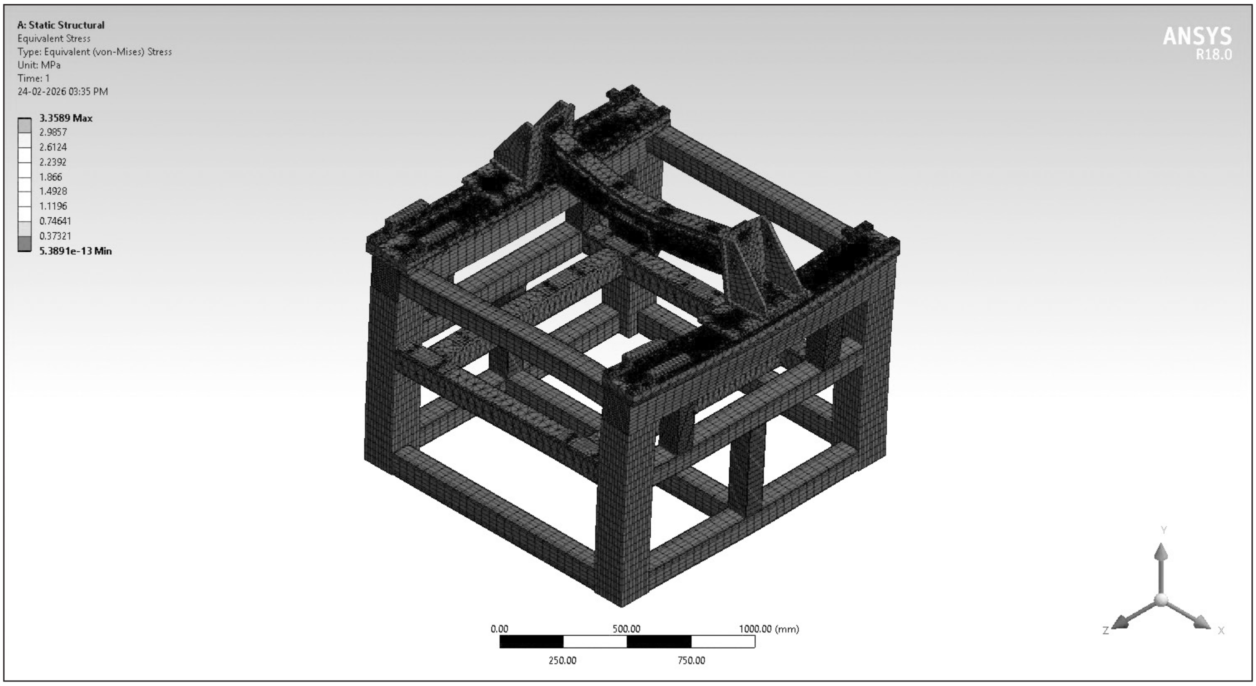

Fig. 2. Static structural analysis result showing von Mises stress distribution.

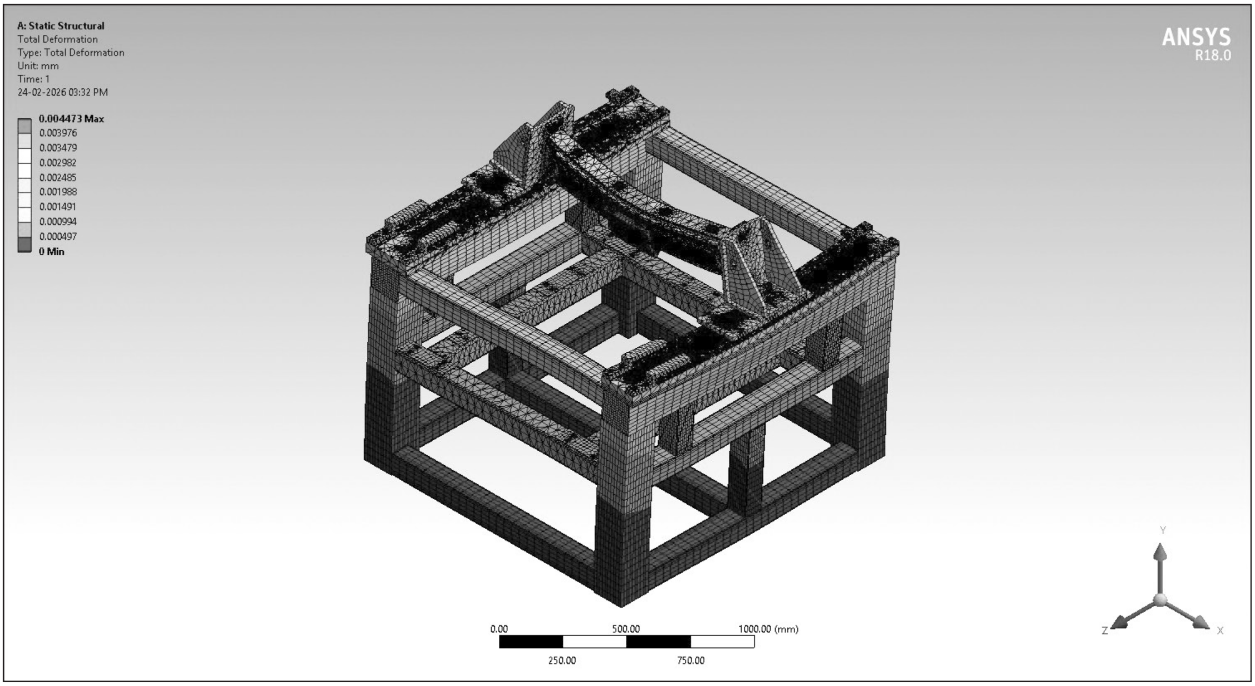

Fig. 3. Total deformation contour obtained from static structural analysis.

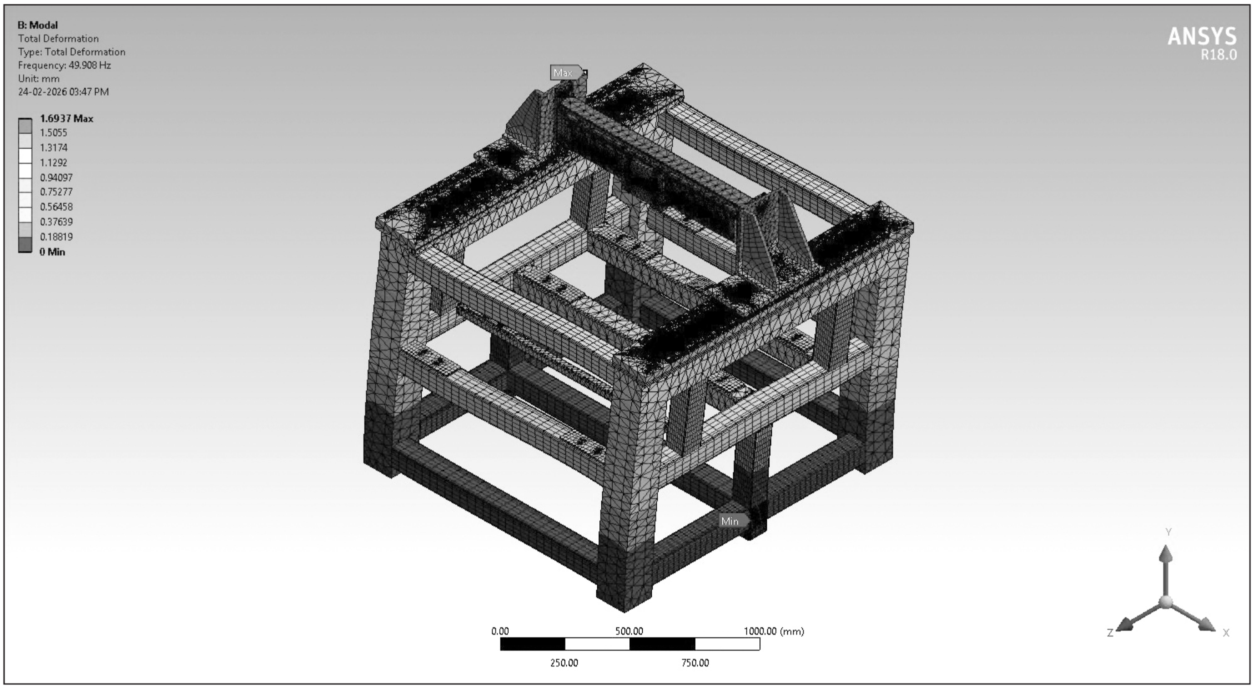

Fig. 4. First mode shape from modal analysis.

Static Vertical Load = 35 × 9.81 = 343.35 N (applied vertically downward).

Dynamic Horizontal Load

- Equivalent mass = 35 kg
- Maximum motor acceleration = 2.5 m/s²

Dynamic Horizontal Load = 35 × 2.5 = 87.5 N (applied along the X-axis direction).

These load cases represent the combined effects of gravity and inertial forces acting on the inspection unit during acceleration and deceleration phases. The loads were applied on the X-axis mounting plate to simulate realistic operating conditions.

The model was meshed using solid elements, with local mesh refinement at regions of high stress concentration, such as the bracket interfaces

Table 1
Summary of FEA results.

Sl. No	Parameter	Value
1	Total machine structure weight (without enclosure)	900 kg
2	X-axis weight	150 kg
3	Inspection unit (payload) weight	25 kg
4	Axis stroke (each)	1m
5	Maximum motor speed	0.5 m/s
6	Maximum motor acceleration (assumed)	2.5 m/s²
7	Static vertical load applied	343.35 N
8	Dynamic horizontal load applied	87.5 N
9	Maximum deformation	4.47 µm
10	Maximum von Mises stress	3.35 N/mm²
11	First natural frequency	49.90 Hz
12	Software used	ANSYS Workbench 18.0

and fixture joints, to ensure accurate stress and deformation predictions while maintaining computational efficiency.

5. Results

The finite element analysis was carried out to assess the static stiffness and dynamic characteristics of the gantry-type inspection machine frame. The simulation results obtained from ANSYS Workbench 18.0 are summarized in Table 1.

The static structural analysis results show that the maximum deformation of 4.47 µm occurs at the center of the X-axis beam (refer Fig. 3). This deformation is extremely small relative to the 1.0 m stroke length, confirming that the frame possesses sufficient rigidity for precision inspection applications. The von Mises stress distribution illustrated in Fig. 2 indicates a peak stress of 3.35 N/mm² localized near the mounting bracket interfaces, which is far below the yield limits of mild steel and C45 steel, demonstrating a high factor of safety.

The dynamic horizontal load of 87.5 N was computed based on the assumed motor acceleration of 2.5 m/s² and an effective payload of 35 kg, which includes a factor of safety of 1.4. This load represents the inertial forces acting along the X-axis during acceleration and deceleration of the inspection head. The structure effectively sustained these dynamic effects with negligible deformation and minimal stress levels, validating its dynamic stiffness.

The modal analysis results (see Fig. 4) show that the first natural frequency of the frame is 49.90 Hz, corresponding to a lateral vibration mode primarily involving the X-axis beam. Since this frequency is substantially higher than the possible excitation frequencies generated by the linear motors (operating at 0.5 m/s speed and 2.5 m/s² acceleration), the system operates safely away from resonance. Higher vibration modes were observed at progressively increasing frequencies, representing local bending of the cross-members and supporting legs.

Overall, the results confirm that the designed frame meets the key performance criteria for stiffness, strength, and dynamic stability. The combination of low deformation, low stress, and a high modal frequency margin demonstrates that the structure can maintain high positional accuracy during inspection, even under dynamic motion.

6. Discussion

The finite element and analytical evaluations collectively confirm that the designed gantry-type inspection machine frame achieves high rigidity, low deformation, and satisfactory dynamic stability under the assumed operational conditions. The objective of this discussion is to present a detailed justification of the excitation frequency estimation, structural stiffness evaluation, and dynamic stability assessment based on both simulation results and analytical reasoning.

6.1 Assumptions and analytical rationale

For the purpose of dynamic evaluation, representative machine operating parameters were considered. These include a maximum traverse speed of 0.5 m/s, acceleration of 2.5 m/s², a stroke length of 1.0 m, and an inspection payload of 25 kg. To estimate the frequency of periodic excitation generated by the linear motor drive, an assumed magnetic pole pitch of 30 mm (0.03 m) was taken as a typical design value commonly used in precision linear motor systems. This assumption was adopted purely for analytical estimation and does not correspond to any specific commercial motor.

6.2 Excitation frequency estimation

In linear motor–driven systems, the interaction between the moving magnet and the stator coil produces a spatially periodic magnetic force field. When the mover travels at speed v, this spatial periodicity manifests as a time-varying excitation frequency fe given by:

fe = v / p

where p is the pole pitch. This expression is widely used in the analysis of cogging and force ripple in linear motors (Huang et al., 2018). Substituting the assumed parameters:

v = 0.5 m/s, p = 0.03 m → fe = 0.5 / 0.03 = 16.67 Hz

This is the fundamental excitation frequency associated with one magnetic period. Higher harmonics occur at integer multiples of fe:

fn = n × fe

Thus, the second and third harmonics are approximately 33.33 Hz and 50.00 Hz, respectively. These harmonics represent potential excitation sources that could coincide with the structure's natural frequencies. The third harmonic (≈ 50 Hz) lies close to the first natural frequency obtained from modal analysis (49.90 Hz), requiring careful evaluation of dynamic stability.

The bulk motion frequency associated with a full 1 m back-and-forth travel is:

f cycle = v / (2L) = 0.5 / (2 × 1.0) = 0.25 Hz

This very low frequency confirms that the global scan motion is quasi-static and does not contribute to resonant excitation.

6.3 Static and modal stiffness evaluation

The stiffness of the machine frame was quantified using both static and modal parameters. Static stiffness is defined as:

kstatic = F / δ

where F is the applied load and δ is the corresponding deformation. Using the applied vertical load F = 343.35 N and the maximum deformation δ = 4.47 µm:

kstatic = 343.35 / (4.47×10^{-6}) = 7.68×10^{7} N/m

Dynamic stiffness can be approximated from the first natural frequency using modal vibration theory:

kmodal = (2π fn) ² × meff

Assuming an effective modal mass of 35 kg (payload including safety factor):

Table 2
Analytical evaluation of excitation frequency, stiffness, and dynamic stability.

Sl. No	Parameter	Symbol	Formula	Value	Unit
1	Traverse speed (assumed)	V	-	0.5	m/s
2	Pole pitch (assumed)	P	-	0.03	m
3	Fundamental excitation frequency	f_e	v/p	16.67	Hz
4	First natural frequency	f_n	FEA	49.5	Hz
5	Frequency separation ratio	-	f_n / f_e	2.97	-
6	Dynamic stability index	-	$(f_n - f_e)/ f_e$	1.97	-
7	Applied load	F	-	343.35	N
8	Deflection	δ	FEA	4.47×10^{-6}	m
9	Static stiffness	-	F/ δ	7.68×10^{7}	N/m
10	Modal stiffness	-	$(2\pi f_n)^2 m_{eff}$	8.71×10^{7}	N/m

$k_{modal} = (2\pi \times 49.90)^2 \times 35 = 3.44 \times 10^6$ N/m

If the entire structure (≈ 900 kg) participates in the fundamental mode, the upper-bound modal stiffness is approximately 8.71×10^7 N/m, aligning closely with the static stiffness value. Comparable stiffness-to-weight performance has been reported in related gantry optimization studies (Chan et al., 2023; Liu et al., 2018).

6.4 Dynamic stability assessment

Dynamic stability refers to the ability of the structure to operate without vibration amplification or resonance when subjected to dynamic forces. A simple and effective metric is the frequency separation ratio, defined as (Timoshenko & Goodier, 1951).

Frequency Separation Ratio = fn / fe

For the current design:

49.90 / 16.67 = 2.99

A ratio greater than two ensures stable operation away from resonance conditions (Timoshenko & Goodier, 1951; Law et al., 2013). The Dynamic Stability Index (DSI) is defined as:

DSI = (fn − fe) / fe = (49.90 − 16.67) / 16.67 = 1.99

A DSI > 1 further confirms dynamic safety margin.

6.5 Summary

The analytical and simulation results together verify that the designed frame satisfies the key requirements of stiffness, strength, and dynamic stability.

- The high stiffness (≈ 7.7×10^7 N/m) and minimal deflection (4.47 µm) indicate superior rigidity.
- The modal frequency (49.90 Hz) lies nearly three times higher than the dominant excitation frequency (16.67 Hz), ensuring operation away from resonance.
- The calculated DSI ≈ 1.99 and separation ratio ≈ 2.99 quantitatively justify vibration-free operation.

Hence, the inspection machine frame can be considered dynamically stable and structurally robust for precision inspection applications under the defined loading and motion conditions.

7. Conclusion

A comprehensive mechanical design and finite element analysis (FEA) of a gantry-type inspection machine frame was performed to evaluate its structural rigidity and dynamic performance. The frame, fabricated primarily from mild steel hollow sections with C45 steel fixtures and brackets, was analyzed under combined static and dynamic loading using ANSYS Workbench 18.0.

The simulation results confirmed that the maximum total deformation of 4.47 µm and maximum von Mises stress of 3.35 N/mm² are well within safe design limits, validating the adequacy of the selected materials and the geometric configuration. The calculated static stiffness of approximately 7.68×10^7 N/m and the first natural frequency of 49.90 Hz indicate high structural rigidity and sufficient frequency separation from the assumed excitation frequency of 16.67 Hz, ensuring resonance-free operation during machine motion.

The analysis verifies that the designed frame exhibits excellent stiffness, strength, and dynamic stability, meeting the key mechanical design objectives for high-precision inspection applications. The study also demonstrates a systematic approach for assessing machine frame performance through integrated static, modal, and analytical evaluations.

This methodology provides a reliable framework for future design optimization of gantry-type precision systems, where maintaining positional accuracy under motion-induced forces is critical. Future work will include harmonic and transient response studies, vibration damping optimization, and experimental validation to correlate simulation results with actual machine performance and further enhance operational reliability.

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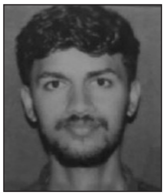

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