AN ENHANCED HEURISTIC TECHNIQUE FOR OPTIMIZATION OF GANTRY OF A CNC MACHINE

Abhay Singh Rana[1*], Dr. Manu Sharma[2] ,Dr. Prashant Jindal[3]

Abstract

Industrial demand for manufacturing large sized work pieces with intricate designs has been on the rise. In order to improve the machining processes, CNC machines are adopted due to their high speed and precise manufacturing capabilities. In these machines, optimization of certain parameters plays a significant in their dynamic and static characteristics. Numerous existing algorithms are used for designing an efficient CNC, with each one dedicated to optimize a specific parameter of the machine tool. This work proposes a novel design algorithm for designing a CNC machine tool gantry with columns which is mounted on a well-designed feed drive system. The objective of the current research, is to design a gantry type machine tool with high rigidity and optimal weight in order to reduce vibrations of the machine and improve the surface finish of the finished product. An algorithm is devised in the present study to achieve the optimal gantry design. Algorithm devised in the present study reduces the mass and stress level with application of stiffeners, strategic removal of weight, minimizing computational complexities, and maintaining considerable rigidity. The design is analysed by FE method for calculating rigidity, stresses and deformation according to the applied boundary conditions. The shape optimization provides the optimum shape of the gantry required for its specific design purpose. In the present research that design purpose is to fabricate a gantry integrated with columns for an 8 feet by 4 feet CNC Router. The topology optimization technique is applied on the optimal shaped gantry to achieve further weight reduction. Strategic reduction of weight enhances the stiffness and the natural frequency. The modal analysis results of new gantry assembly are compared with the natural frequencies of existing models to communicate the effectiveness of the proposed optimization methodology. The proposed design of the gantry is also tested with the help of laser vibrometer to estimate the natural frequency at which the machine can function without attaining the resonance condition.

Keywords: CNC machine, gantry, topology optimization, shape optimization, mass, stiffness, natural frequency

---

[1*]Research Scholar, University Institute of Engineering and Technology, Panjab University, Chandigarh
[2]Professor, University Institute of Engineering and Technology, Panjab University, Chandigarh
[3]Associate Professor, University Institute of Engineering and Technology, Panjab University, Chandigarh

[*]**Corresponding Author:** Abhay Singh Rana
[*]Research Scholar, University Institute of Engineering and Technology, Panjab University, Chandigarh

DOI: 10.53555/ecb/2022.11.11.105

1. Introduction

The rapid increase in the development of numerous industries in transportation, machineries and other areas lead to the growth of demand in higher speed and precise level of manufacturing of larger and heavier parts [1]. CNC (Computer Numeric Control) machining is the extensively utilized manufacturing component and the emerging development of applications with larger and complex designing of components compressed with materials found to be difficult in processing [2]. The CNC tool along with the gantry and feed drives with the greater distance within slides have been considered to the significant tool for machining larger parts. Hence, the necessities of the CNC performance level included with the size of the operating area, stiffness and torque becomes rigorous. For processing of similar components, five axis type of CNC gantry tool has been utilized because of its flexible space, extensive cutting ability and the stiffness along with accuracy. The entire quality of the work piece lies on the accuracy level of the machine tool. When the operating time and life of the tool has increased, the wear and tear of components have also increased. The anomalous positioning of the irregular deflections has occurred in the rotary axis which has endangered the quality. The adjustments on mechanical equipment have been found to be the general solution. Though, the considered system [3] has to dismantle on B-axis structure and the cost of maintenance has also been found to be expensive, the hydraulic-analysis has been referred as the efficient method.

The impact of limitations in dimensions and material have paved designing the structure of work piece processing through AB- rotary five axis gantry tool with the larger torque. But, the B-axis structure of machining tool has been analyzed to be bulky and complex which have provided hydraulic machines for balancing the gravity [4]. After continuous and longer production of the tool, the hydraulic force of B-axis has been lost and spindle has brought down of control which have seriously affected the quality of the work piece. The suggested research [5] has been analyzed on the point of hydraulic system leaving out the adjustment made on the mechanical structure. It also has provided the mathematical formulation model for the system. Simulations have been performed with the MATLAB for the optimization of the hydraulic –control system. Based on enhancing the conditional stability of system, duration of hydraulic force has been extended and time of deflection has been delayed to 2 seconds which have effectively reduced the computation risk of the work piece [6].

The CNC machines included with the mobile-element (mechanical-coupling in between the feed-chains) has important effects on the static and dynamic nature of the machine. The accuracy in machines have been affected by the deformation occurrence in mobile-element [7]. The static level properties has been studied and summarized that, deformation has greater value in efficiency and accuracy of the machine. Along with that, the weight of the machine has also been recognized as important which have greater effects on the dimension of machines. With the deployment of FEA exploration on 3D analysis with CAD system, the suggested approach has focused on the mesh and topological optimization of the mobile element. It also has shown that the optimizations have reduced the inertial forces for 10 % -30 % and has maintained the rigidity considerably [8]. The deformational errors have been caused mainly due to gravity and has required to be focused in the design of precision. Following that, the recommended system [9] has utilized the precision design-model for the vertical machining system based on gravity. Through abstraction of the machining tool into the multi-body topological structure, the statical error of the tool has been established based on multi body theory along with the homogenous transformation. Every motion axis has been controlled with two significant parts. The geometrical error during manufacturing along with the gravity-based deformation-error and the other one has been the stiffness structure of the tool which have provided relationship in between the deformation and gravity with the utilization of the beam-elements. During the process of modelling, the coefficients on stiffness and volume has been recognized as the influencing parameters of the precision in machine-tool [10]. On considering both coefficients as the aviation variables depending on the analysis of sensitivity error and global-optimization technique, the allocation and requirements on design has been determined and has provided the structural designing and processing in manufacture of machinery components [11]. Outcomes have exposed that, the geometric error in manufacturing has been controlled and has met the precision requirements effectively. It has been clear that, numerous researches have undertaken various studies for optimization in gantry. This study affords a research taken for expounding the stiffness characteristics of the machine and enhancing the frequency with the feed drive assembly.

The main contributions of the present study are,

- To perform stiffening of the feed drive system for enhancing the natural frequency of the machine tool through modal analysis.
- To accomplish the optimization of the gantry with the utilization of mode shifting based on modal analysis.
- To reduce the mass added by stiffeners for cantilever type plate using topological and shape optimization through analysis using the Newark method.
- To assess the performance of the system after stiffening with the implementation of the ansys software and measures the level of natural frequency.

1.1. Paper Organization

The organization of the paper is given with Section 2 discussing the various existing literature works focused on optimizations in the gantry of machine during manufacturing. Following this, Section 3 precisely discusses the major areas of present research, Section 4 discusses the modelling and simulation phases of the proposed system. Lastly, the overall review is concluded in Section 5.

2. Literature Review

The following section discusses on various traditional literatures related to optimization of static and dynamic characteristics of the modes of gantry in machine tools. Many studies have focused particularly on the analysis of both the characteristics in a single machine positions and working conditions in an individual section and has failed in concentrating entire structure [12]. The suggested strategy [13] has been designed on the multi-objective based optimization approach depending on NDGA (Non-Dominated sorting Genetic Algorithm) and BP (Back Propagation) Neural Network which has ensured the accuracy of the turntable which has reduced the entire mass and has improved the characteristics considerably. Both the static analysis and modal-analysis has determined the maximum level of deformation and parts prone to vulnerable. In addition to that, the analysis based on topological optimization has been used in the identification of redundant-mass segments. It has also performed explorations on the optimization-objectives along with parameters related to dimension based on optimization oriented outcomes. With the utilization of composite design, the suggested approach has acquired test-points using neural network [14]. Optimal outcomes have been retrieved from the optimization and has reduced the mass with the deployment of the light-weighted design.

The process of topological optimization is a significant strategy in designing the light-weight components of machine-tool. Though the results achieved have not been found to be suitable on processing with the greater speed and light-loaded and lower cost tools. The recommended approach [15] has solved the aviation complication of gantry specifically in slicing machines. It has used the aluminum model as the structural components. Initially, the gantry has been verified with the finite constituent strategy. The output have shown that after the structural light weighted design, the mass has been lowered. Moreover, the metrics like stress and maximum level of deformation have been estimated and the modal factors have satisfied the demands in design. The traditional machining tool with iron and steel models have experienced vibrations resulted with the poor level of surface components [16]. The characteristics of the machine has been enhanced with the steel type reinforcement by epoxy-granite. In order to optimize the designing space for a particular set loads, constraints and conditions for the maximization of the system-level performance. The optimization of the steel strengthening configuration of vertical location of machine-centre. It has carried out the maximization of the stiffness and along with that the natural-frequencies while on the way of minimizing the mass of the model [17]. Multiple level of aviation configuration has been carried out for reinforcement of steel based on optimizations in topology and investigations has been carried out with the FEA. The characteristics of EG-columns have been compared with existing SREG and CI columns. Outcomes have exposed significant improvement in reduction of mass and enhanced natural frequencies. The suggested model [18] have designed internal-stiffeners structure of the auxiliary parts in machine. The strategy has skillfully adopted the scientific growth of veins. Initially, the inter-relationship between the both has been explored through the analysis on mechanical-properties of various growth phases with the finite segment simulation. The parameter based optimization on stiffener structure based on variable structure have been used. Different from conventional design on stiffeners, the considered approach has not only adopted the method of variable cross-sectional framework but also has simulated the adaptive development law of veins in leaf with the parametric strategy [19]. Moreover, three different modelling techniques have been compared which have revealed the efficiency of the GABPNN (Genetic Algorithm and BP Neural Network). Along with that, the marine-predator technique has optimized the meta-model.

Due to the significant load balancing structure of heavier machinery tools and other larger scale structures bear moving-loads and outcomes of direct topological optimization have complications like difficult identification of the load transfer skeleton and difficulties with the working conditions have been comprehensively considered. To this purpose, the considered approach [20] has prepared a layout on the stiffened plates for larger scale-box model under moving loads depending on multiple working condition on topological optimization. From the principles on force, the box-models have been simplified into the bending functional segment, auxiliary functional part and the additional torsional functions which have reduced the complexities and structural-dimensions in the topological optimization [21]. Further, moving part of loads have been streamlined to various positions and evaluations have been accomplished with the programming. The mathematical formulations have been performed for the optimization of the functional portions. Based on considerations with the exposure on crossbeam- heavy-turning along with milling-machine center, the optimization outcome have revealed the enhancements in strength and stiffness of the crossbeam with the reduction in weight [22]. Since the cross beam has been considered as the significant moving-element, it has determined the dynamical characteristics of machine in either X axis or Y axis which have affected the quality and efficiency of the machine directly. The acceleration attribute has been found to be the important characteristics which is inversely proportional to mass of moving-parts in order to reduce the mass. It also has ensured stiffness as the vital part in the optimization of the machine. The outcome have revealed that improvements have been identified in the deformation and lower order of amplitude in crossbeam with the minimization of mass. It has provided the theoretical base value for the optimization related structure. From the analysis on the suggested approach [23], it has been found that the cost of manufacturing been reduced considerably along with the retrieval of satisfied efficiency in the machinery tool.

2.1. Research Gap

The review made on existing studies have taken many optimization of gantry methods in manufacturing machineries for achieving accuracy along with reduction of mass and stress. The common limitations with respect to validation of appropriate topological optimization patterns have been analyzed from the existing literature and are listed as follows,

- Model of the gantry should have been modelled as an elastic support as fixed support simulations of the gantry does not match the practical results. Even machines with base fixed to the ground are precisely modeled by including stiffness of the soil or the reinforced concrete [24].
- There are few methods in the literature to find the expressions for elastic support. Expressions are derived from effect of stiffness and damping linear guide assembly also called as feed drive characteristics [25]

The primary objective of this study is to create an algorithm that effectively optimizes all relevant parameters involved in the design and fabrication of a gantry for a machine tool structure. The algorithm addresses the specific challenges related to gantry design, including any discrepancies observed between expected and actual outcomes, as outlined in the research methodology. Furthermore, the algorithm's applicability extends beyond gantry optimization, as it can be adapted to optimize various other structural optimization problems.

Algorithm Methodology

- This research proposes the development of an algorithmic methodology for designing a gantry system with optimized geometry and mass. The algorithm employed is heuristic in nature and demonstrates the potential to consistently yield reliable outcomes. The gantry system is integrated with a feed drive system, comprising linear motion mechanisms and their corresponding mounting structure. In order to assess the structural behavior at various stages of optimization, finite element analysis (FEA) is utilized to determine the natural frequencies of the structure. A modal analysis is conducted on the machine and gantry structure, comparing the system with a reinforced feed drive mechanism to the system with the existing feed drive. The objective is to estimate the resulting increase in frequency. By implementing techniques such as Topology and Shape Optimization, the gantry is designed to possess an optimal shape, size, and stiffness. In the FEM and meshing section, the 3D model of the gantry structure was created using SolidWorks and exported to ANSYS in a binary Parasolid file format. The program-controlled mesher with adaptive sizing and varying refinement, mainly using tetrahedron elements, was employed. Different refinement levels were tested, but the results showed a maximum deviation of 2 Hz, indicating consistent outcomes. This approach ensured accurate representation of structural behavior while minimizing computational costs.

3.1) Gantry mode shifting and stiffening feed drive system

The initial gantry model used for optimization was derived from a previous study conducted by the author. In that study, it was assumed that both columns of the gantry were fixed in place, but the mounting cantilever had a high elasticity. The simulated natural frequency of the gantry in that study was found to be 226Hz. However, when the gantry structure was experimentally tested using a DIC correlation camera, the measured first natural frequency was determined to be 127Hz. This discrepancy between simulation and experimental results was attributed to suboptimal design of the mounting structure and feed drive components. The fabricated gantry is shown in Fig.1.

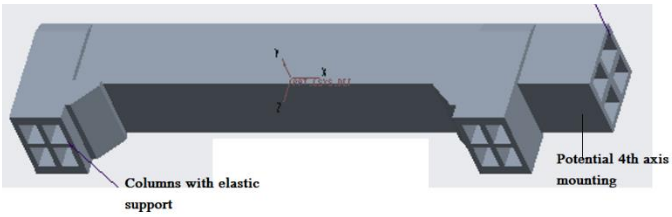

Fig.1 (i) 3D Design of the gantry

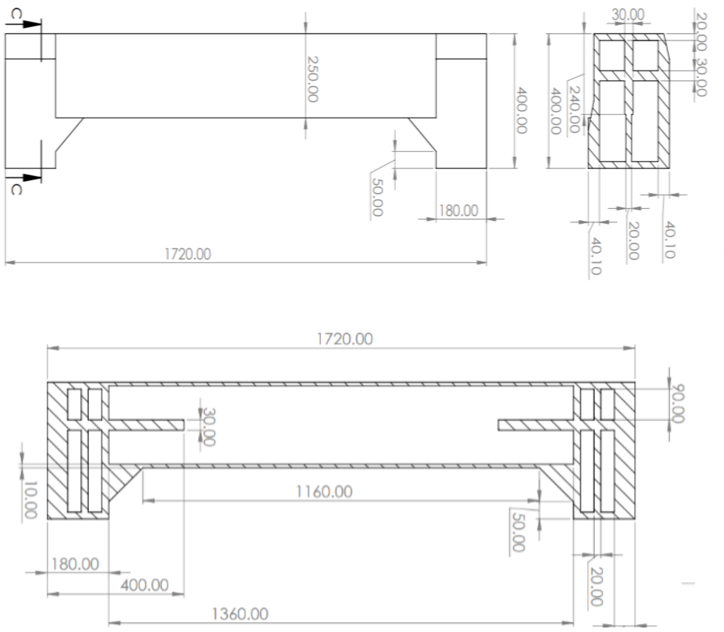

Fig.1 (ii) Drawings highlighting dimensions and internal ribs

To enhance the rigidity of the gantry in the previous machine, an extra row of square pipe columns was incorporated into the structure. These additional columns, totaling five in number, were designed with a heavy wall thickness. Unlike the previous setup where the mounting cantilever functioned as a full cantilever without any perpendicular ground force, this modification aimed to increase the overall stiffness of the gantry. The inclusion of the extra columns with a heavier wall thickness led to a substantial improvement in the rigidity of the gantry structure.

To facilitate the Finite Element Analysis (FEA), the gantry structure, along with the five additional supports illustrated in Figure 2 (ii), was treated as an elastic support. The cantilever's elastic support was extended following transformed supports and modeled as a bonded contact in the Ansys™ software. The described process of mass addition was applied to modify the geometry, allowing for the assumption of a fixed support boundary condition at the gantry columns. These adjustments validate the assumption of a fixed support at the gantry base.

It is important to note that the mass addition in the feed drive mechanism does not alter the mode shape of the gantry. Instead, it primarily affects the natural frequencies of all the modes, causing them to be extended or increased. The purpose of the mass addition is to enhance the overall rigidity and stability of the gantry structure, rather than influencing the specific mode shapes exhibited by the system.

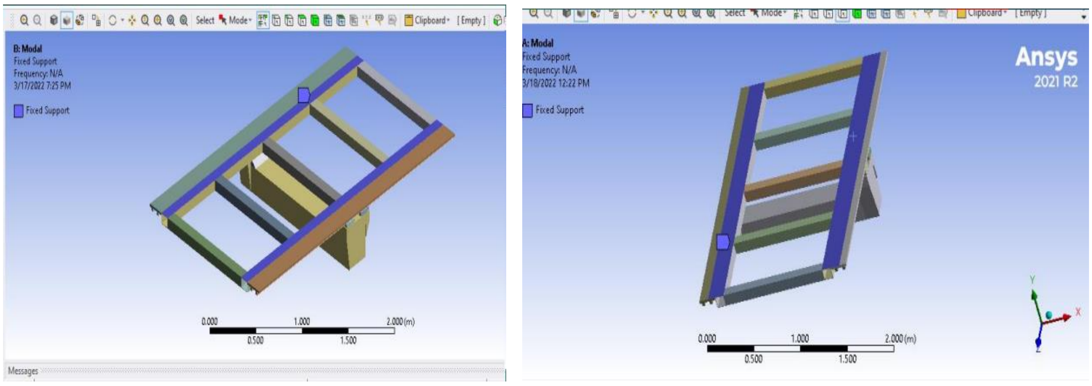

Figure 2. Modal Analysis of Gantry without Feed Drive System

The cantilever and complete feed drive system was not stiffened in this step as the mode shape does not change with the stiffening of feed drive. Chamfer type ribs were added to shift the mode. This rib shown in figure 3 increased the axial stiffness and shifted the first mode.

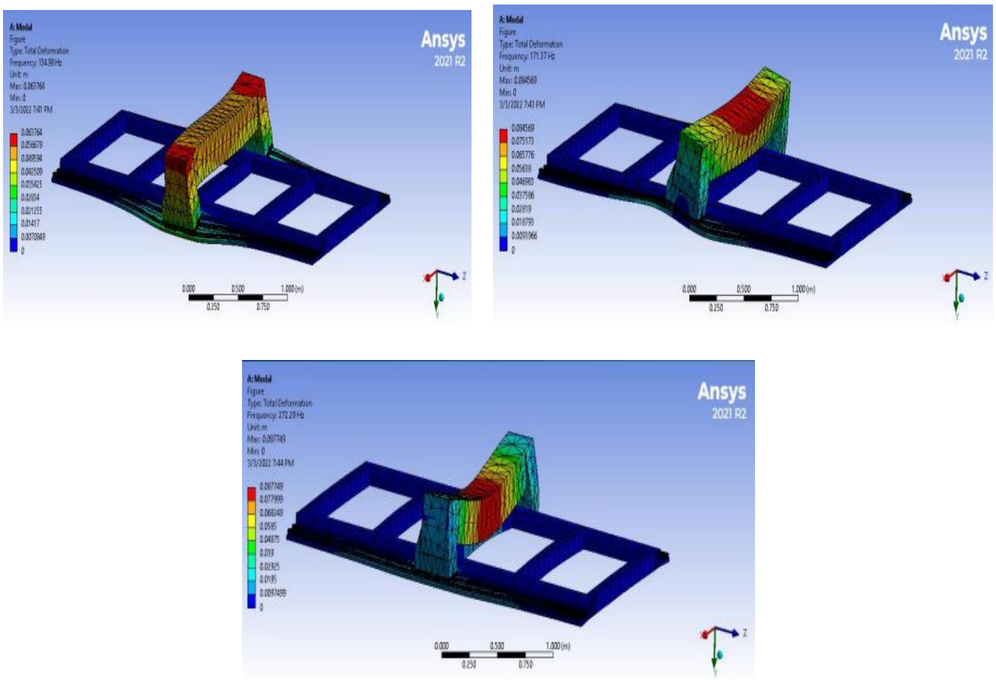

Figure 3. Boundary Conditions

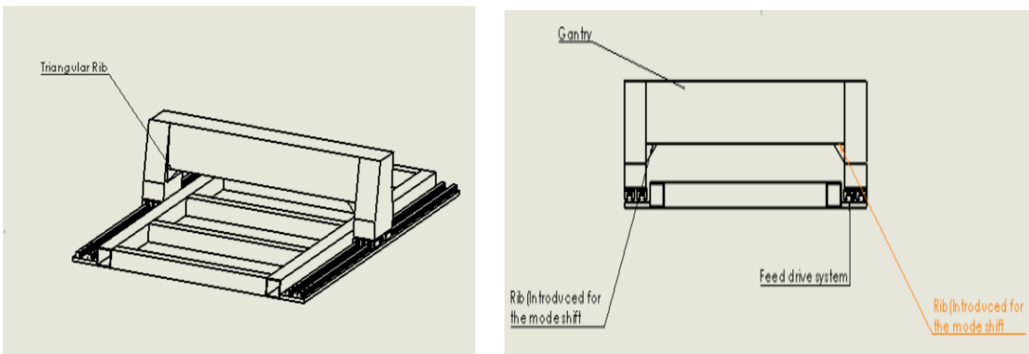

Figure 4. Gantry with Ribs

The inclusion of ribs in the design provides an increased level of strength, enabling the structure to endure heavy loads without experiencing substantial deflection and increased rigidity. This surpasses the performance of a design that lacks ribs, as the additional structural elements enhance its ability to withstand applied forces and resist deformation.

3.2 Shape Optimization

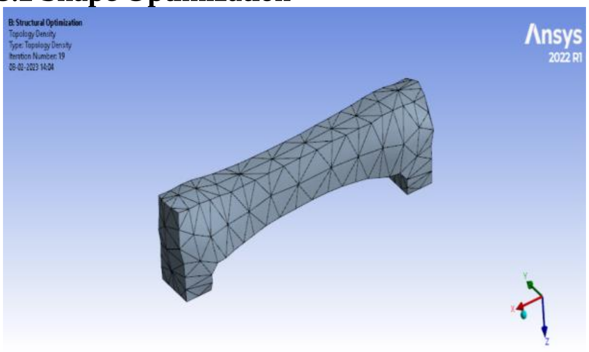

Figure 5. Shape Optimization of gantry

The shape optimization is done to minimize the shape of the system within the given constraints in order to reduce cost. The strength of the system and its rigidity should not get affected and the gantry should be functionally stable at dynamic conditions for the optimal shape. The topology optimization defines the boundary and space within which the gantry should be designed.

3.3) Topology Optimization

The ribs are introduced for obtaining higher values of natural frequencies during dynamic conditions. In order to improve the efficiency of the gantry, it is also necessary to consider the material usage for fabricating the structure. Since the ribs are acting as a rigid support for the gantry at its base, now the mass of the gantry can be reduced, thereby resulting in less usage of materials. To determine the suitable mass of material to be used for fabricating the gantry, we apply the Topological optimization technique.

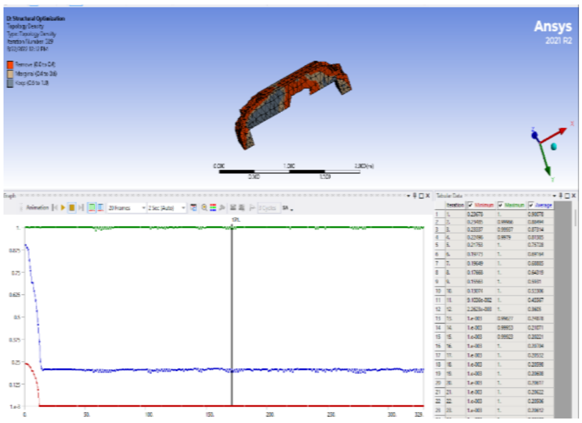

Figure 6. Density Vs Volume Fraction

The optimum mass of the gantry is obtained by reducing the volume of structure without affecting the size ratio. Hence, even if the mass is reduced, the gantry can still carry equal amount of load and work efficiently as earlier, because of the presence of ribs.

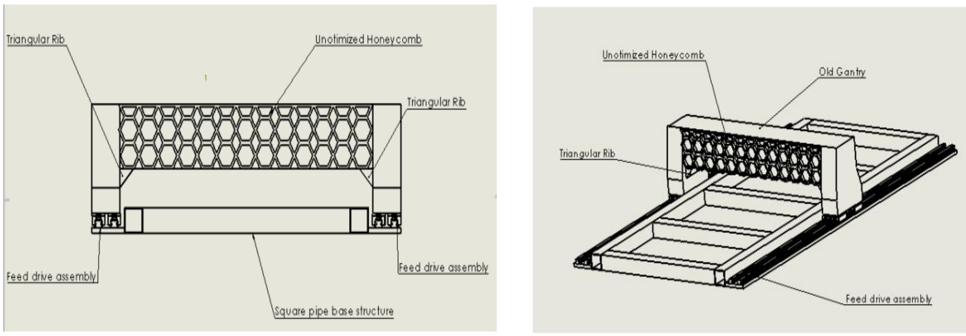

Figure 7. Gantry with Rib and Honeycomb Structure

The gantry, when introduced with a honeycomb structure on its vertical face or the top face, tends to have improved dynamic stability. But, the honeycomb structure enhances the natural frequency of the gantry structure only marginally, and thus the optimization of result of that structure is not discussed in this research.

3.4) Heuristic stiffness sensitivity coefficient (μ_{hs})

The proposed metric for assessing the effectiveness of a heuristic rigidity optimization step is defined as the 'heuristic stiffness sensitivity coefficient (μ_{hs})'. $\mu_{hs} = \left({}^{\Delta f}/_{\Delta m}\right), where\ \Delta f > 0$, where f denotes the first natural frequency of the system and m denotes the mass of the system. The heuristic stiffener efficiency (μ_{hs}) is evaluated for each of first 6 frequencies. The primary goal of the optimization step is to eliminate the first mode shape and enhance the first natural frequency of the system. However, since this optimization step affects all the natural frequencies of the system, it is important to identify any potential undesired complications, such as mode coupling. (μ_{hs}) Is calculated for each optimization step and is verified if (μ_{hs}) it is consistent with the each unintended frequency change due to shifting of first mode shape.

3. Results and Discussions

4.1 Optimization Analysis and Outcomes

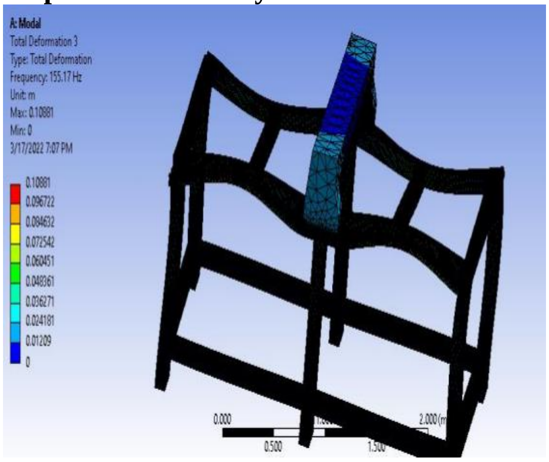

Figure 8. Mode Shape of Complete Assembly

The base model utilized for further optimization is mentioned in section 3.1. The design is taken from a previous machine, which was built with same purpose. In that machine experimental frequency was found to be 127 Hz in contrast with designed natural frequency of 260 Hz. The result was found to be non-optimal and flexible feed drive with fixed support assumption was the primary reason for simulation result and experimental result mismatch. The image of the machine and the gantry is shown in Figure 9. In light of previous attempt following optimization algorithm was employed to increase the first natural frequency of the system.

4.1.1) Cantilever Supports/Feed drive stiffening (Step 1): The frequency of the assembly increased from an initial value of 155 to a final value of 189.5. This indicates an improvement in rigidity, as the assembly became less prone to vibrations.
In order to attain optimal dynamic performance, the base of the gantry columns was subjected to a fixed support boundary condition. To simulate the effects of fixed boundary conditions, the rigidity of the cantilever was increased by adding additional support at its base. This was achieved by affixing extra linear blocks to the column's base.

The mass of the assembly also increased by 104 units. The negative value of (μ_{hs}) = -0.331730769 suggests an increase in rigidity and a large negative value of μ_{hs} corresponds to an efficient mode shifting step.

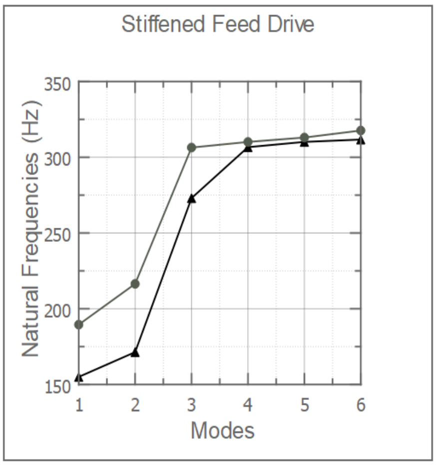

Figure 9. Impact of Feed Drive System stiffening on the Natural Frequency

4.1.2) Fillet Stiffener (Step 2):
By introducing a large fillet stiffener, the frequency of the assembly increased from 205.14 to 246.47. This significant change indicates a notable improvement in rigidity. The mass of the assembly increased by 22 units. The negative value of (μ_{hs}) =-1.878636364 suggests a considerable increase in rigidity and an efficient mode shifting optimization step.

4.1.3) Honeycomb Structure (Step 3):
The optimization of the honeycomb structure resulted in a slight decrease in the frequency of the assembly from 246.47 to 238 and the mass of the assembly increased by 28 units. However, this change is relatively small compared to the other steps. The mass of the assembly increased by 28 units. The positive value of (μ_{hs}) = 0.3025 suggests a slight increase in flexibility and questions the viability of this optimization step.

The placement of a honeycomb stiffener on the top face/vertical face of the gantry has adversely affected its stability. In the modal analysis, it was observed that the natural frequency decreased by 8 Hz compared to the gantry without the honeycomb stiffener. This decrease in natural frequency is attributed to the additional load exerted on the middle area of the gantry's top face. This load leads to bending of the gantry, causing instability during dynamic conditions. Value of (μ_{hs}) coefficient suggests that honeycomb stiffener step is not very useful when it is used for mode shifting (mass addition).

It is important to note that this study did not involve optimizing the honeycomb structure, as the observed increase in natural frequency was marginal and had minimal impact on gantry stability. Although a honeycomb structure could potentially serve as an ideal stiffener in many situations, designing an optimal honeycomb structure for this specific scenario would incur high computational and manufacturing costs, rendering it inefficient.

In summary, the honeycomb stiffener placement had a negative impact on gantry stability due to reduced natural frequency. However, optimizing the structure with a honeycomb design was deemed inefficient for this particular case, considering the marginal increase in natural frequency and associated costs.

4.1.4) Shape Optimization:

Each optimization step yields different pattern of natural frequency increase, natural frequency of first six modes, before and after the first 3 optimization steps is shown in graphs above.

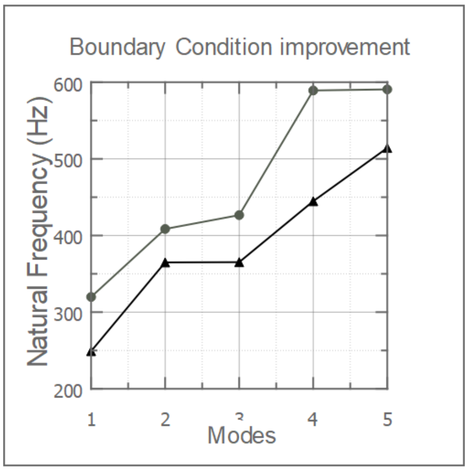

Figure 10. Natural frequency of Gantry before and after realizing optimum boundary conditions

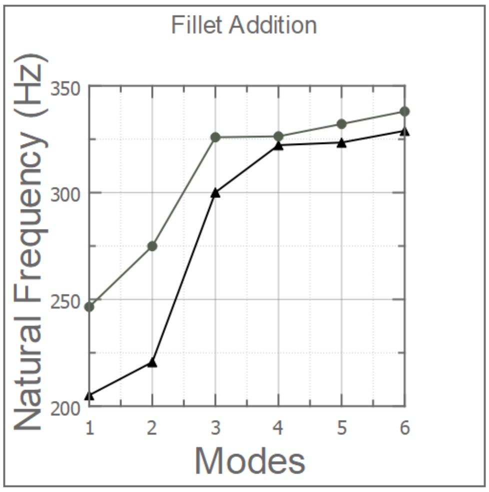

Figure 11. Natural frequency of Gantry before and after adding Stiffener

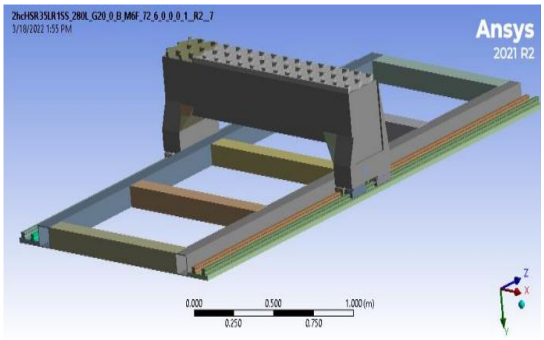

Figure 12. Modal Analysis of gantry with Honeycomb structure on top face

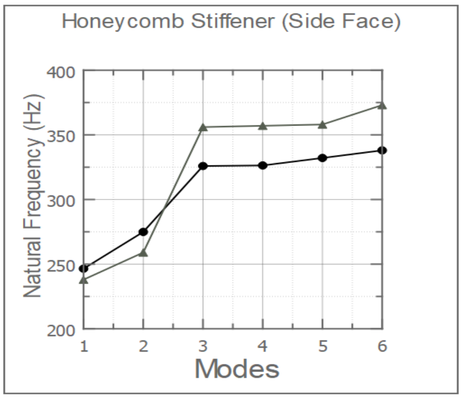

Figure 13. Impact of honeycomb stiffener on the natural frequency

The stability of the gantry has been affected by positioning the honeycomb stiffener on the top face of gantry. The natural frequency on the first modal analysis has reduced by 8 Hz, compared to modal analysis of gantry without honeycomb stiffener. The natural frequency is affected by the additional load exerted on the middle area of the gantry top face. This load tends to bend the gantry and thereby, causes instability of the system during dynamic conditions. This study does not include the optimization of this structure because, only a marginal increase in the natural frequency is observed, causing very less impact on the stability of the gantry. An optimal design of honeycomb structure which often is perfect stiffener for most of the situation can be made which increases the natural frequencies, but associated computational and manufacturing costs make it inefficient in this specific scenario.

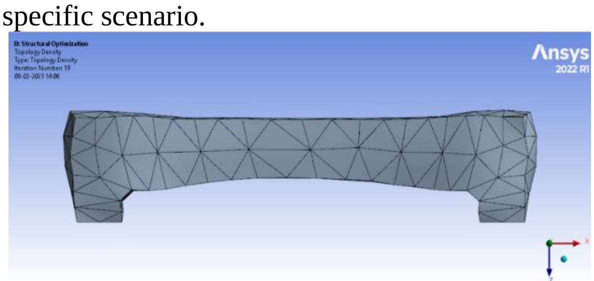

Figure 14. Shape Optimization of gantry after adding stiffener

The addition of stiffener provided sufficient strength and stability to the gantry, thereby enabling to conduct topology and shape optimization. The optimal range of boundary and volume has gained reduction in loss of material and cost.

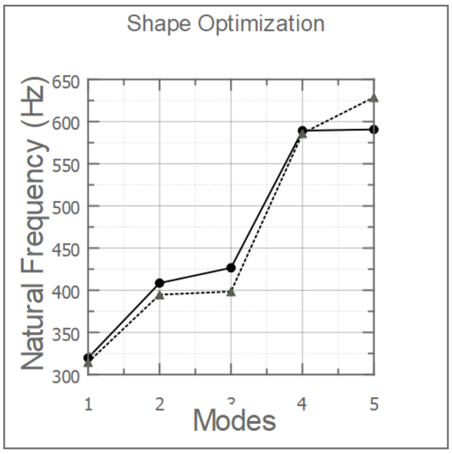

Figure 15. Impact of Shape Optimization on natural frequencies

4.1.5) Topology Optimization:
Topology optimization does not increase the frequency in large quantity usually, but it removes a lot of mass. In this case if $\Delta f > 0$, low (μ_{hs}) will suggest the efficiency of topology optimization steps.
Final design of the gantry is shown in Fig.16

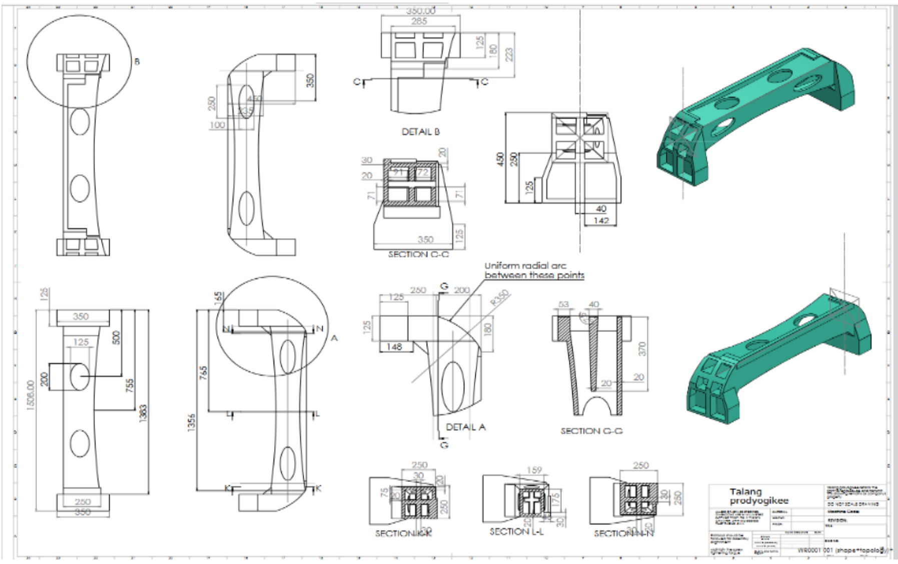

Figure 16. Final Design

Testing and Validation
The gantry was tested using an impact hammer and a Laser Doppler Vibrometer to measure its natural frequency. The impact hammer was used to provide excitation at different locations on the surface of the gantry. The response of the gantry was then analyzed using FFT (Fast Fourier Transform) to determine the natural frequency. The test setup can be seen in the picture shown in Fig. 17.

Experimental Setup

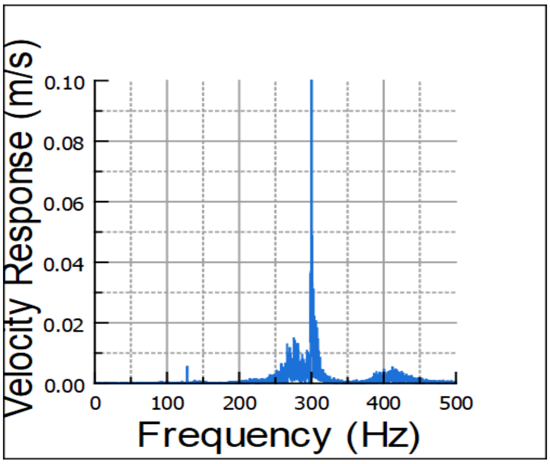

Figure 17. FFT at center of gantry structure (Impact hammer excitation)

Based on the results obtained, when cutting MDF (Medium-Density Fiberboard), the first peak was observed at approximately 300Hz. On the other hand, it was found that when cutting aluminum, the first resonance peak occurred at around 167Hz. FFT of both responses are depicted in Fig.17 and Fig. 18

To optimize the performance of the machine on both materials, it is important to avoid the super harmonics of the frequency results. Super harmonics are multiples of the natural frequency and can lead to unwanted vibrations and decreased performance. By avoiding these super harmonics, the machine can operate more efficiently and effectively on both aluminum and MDF. In practical terms, this means that when setting the RPM (Revolutions Per Minute) for cutting operations, it is necessary to select values that do not coincide with the multiples of the natural frequencies of the gantry on both materials. By doing so, the machine can avoid resonance and achieve optimal performance while cutting aluminum and MDF.

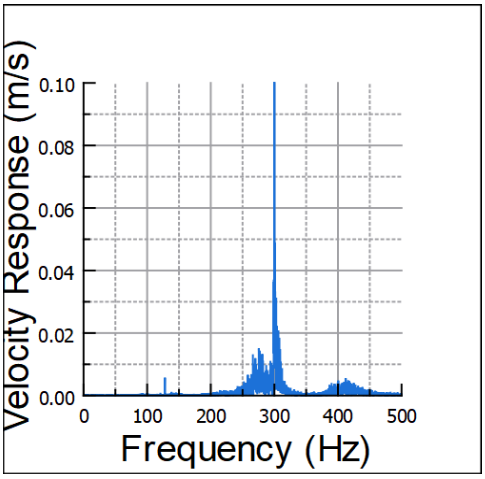

Figure 16. FFT at center of gantry structure (Feed 2500mm/min, Depth of cut: 7mm, Material: MDF)

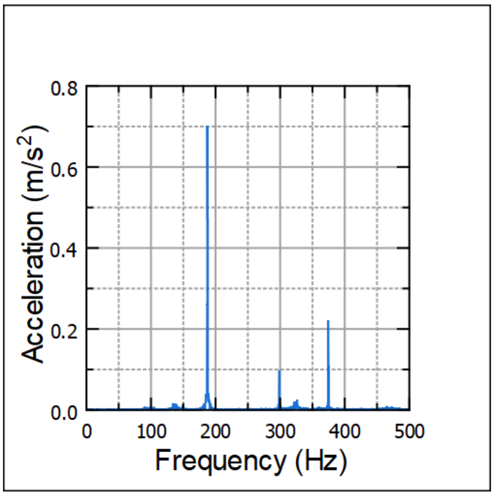

Figure 17. FFT at center of gantry structure (Feed 2500mm/min, Depth of cut: 0.9mm, Material: Aluminum)

The highest natural frequency obtained in literature for a machine tool gantry with comparable dimensions is obtained in [26], the first natural frequency of the gantry in the said study is 202Hz. In our study the complete algorithm increased the frequency from 210 Hz to 301 Hz, which is an improvement of almost 38%.

Conclusion

The following conclusions are derived from the results of the present research work:

- In order to design the gantry more accurately with more rigidity, it is essential to stiffen base feed drive structure. Most importantly, the gantry should be simulated assuming feed drive as elastic support. Approximation of boundary condition of gantry column base as fixed support can be done if the rigidity of feed drive is comparable to a ground bolted mechanical system.
- Optimization of feed drive system in the structure tends to provide more stability and yields better results when its natural frequency is measured.
- Providing fixed supports to the cantilever portion of the feed drive along the base of linear guide mounting plate will improve the dynamic stability and gives higher values of natural frequency.
- The modified design which included stiffener showed more rigidity and stiffness of the gantry system. Topology optimization was conducted for this design and the mass of the structure was considerably reduced for effective usage of material. The modal analysis results were remarkable, and proved that the gantry, though reduced in mass and volume, can withstand equal static and dynamic loads with the support of

stiffener compared to the previous design, which was found to be non-optimal.

- The introduction of honeycomb structure to the top face of the gantry provides only marginal improvement in the natural frequency. This marginal frequency does not impact much on the dynamic condition of the gantry system.
- After obtaining optimal mass distribution throughout the domain using mode shifting, shape optimization and topology optimization respectively.
- The natural frequency of the gantry can be investigated by conducting the laser vibrometer test on three different locations on the gantry surface to identify the resonance frequency which can serve the best operating RPM guide.
- The stiffener efficiency proposed in section 4.1, can be universally used to find the applicability of any rigidity enhancing stiffener. Low negative value of proposed efficiency (η_r) can be considered as indicator of mode shifting optimization step efficiency. A positive value indicates the efficiency of topology/shape optimization.

Acknowledgments

- I would like to acknowledge and thank Talang Praudyogikee LLC, for their financial support, which made this research possible.

I am grateful to Dr. Navin Kumar and his team from IIT Ropar for their invaluable assistance in vibration testing and validation, providing necessary equipment and guidance.

References

1. R. Shindo and S. Nishiwaki, "Latest machine tool structural design technology for ultra-precision machining," *International Journal of Automation Technology,* vol. 14, pp. 304-310, 2020.
2. Y. Fei, X. Chen, Y. Li, G. Liao, W. Zhang, and Y. Zhang, "Modeling and optimization of B-axis hydraulic delay for gantry-type CNC machine tool," in *Journal of Physics: Conference Series*, 2021, p. 012129.
3. H. Lu, Q. Cheng, X. Zhang, Q. Liu, Y. Qiao, and Y. Zhang, "A novel geometric error compensation method for gantry-moving CNC machine regarding dominant errors," *Processes,* vol. 8, p. 906, 2020.
4. K. T. Mohamed, M. H. Abdel-razak, E. H. Haraz, and A. A. Ata, "Fine tuning of a PID controller with inlet derivative filter using Pareto solution for gantry crane systems," *Alexandria Engineering Journal,* vol. 61, pp. 6659-6673, 2022.
5. L. Petrea and G. Stan, "Topological optimization of the mobile element structure of Gantry type CNC machine using FEA method," in *IOP Conference Series: Materials Science and Engineering,* 2021, p. 012061.
6. K. Nesteruk, C. Calzolaio, D. Meer, V. Rizzoglio, M. Seidel, and J. Schippers, "Large energy acceptance gantry for proton therapy utilizing superconducting technology," *Physics in Medicine & Biology,* vol. 64, p. 175007, 2019.
7. X. Li, C. Li, P. Li, H. Hu, and X. Sui, "Structural design and optimization of the crossbeam of a computer numerical controlled milling-machine tool using sensitivity theory and NSGA-II algorithm," *International Journal of Precision Engineering and Manufacturing,* vol. 22, pp. 287-300, 2021.
8. L. Brouwer, A. Huggins, and W. Wan, "An achromatic gantry for proton therapy with fixed-field superconducting magnets," *International Journal of Modern Physics A,* vol. 34, p. 1942023, 2019.
9. H. Wang, T. Li, X. Sun, D. Mynors, and T. Wu, "Optimal Design Method for Static Precision of Heavy-Duty Vertical Machining Center Based on Gravity Deformation Error Modelling," *Processes,* vol. 10, p. 1930, 2022.
10. Y. H. Sahirov, K. Tkachuk, and V. Suglobov, "Impact Of Stress Concentration On Reliability Of Metal Structure Elements Of Gantry Cranes," *Scientific Bulletin of National Mining University,* 2022.
11. H. Patel, D. Upadhyay, and D. Patel, "Design optimization of box girder in gantry crane using finite element analysis software," *International Research Journal of Engineering and Technology,* vol. 7, pp. 1906-1917, 2020.
12. J. Carvalho, P. J. Cruz, and B. Figueiredo, "Ceramic AM Gantry Structures-Discretisation and connections between beams and columns," 2021.
13. B. Huang, J. Wang, B. Tan, J. Zhao, K. Liu, and J. Wang, "Analysis and Optimization of Dynamic and Static Characteristics of Machining Center Direct-Drive Turntable," *Applied Sciences*, vol. 12, p. 9481, 2022.
14. M. I. Solihin, C. Y. Chuan, and W. Astuti, "Optimization of fuzzy logic controller parameters using modern meta-heuristic algorithm for gantry crane system (GCS)," *Materials Today: Proceedings,* vol. 29, pp. 168-172, 2020.
15. Y. Lv and W. Xiong, "Aluminum Profile-Based Lightweight Design of the Gantry Frame on Slicing Machines," in *Frontier Computing: Proceedings of FC 2020*, 2021, pp. 1787-1796.

16. S. Liu and W. Zheng, "Priority order recognition method of module redesign for the CNC machine tool product family to improve green performance," *Transactions of the Canadian Society for Mechanical Engineering,* vol. 45, pp. 513-531, 2021.
17. V. Arjun, A. Arun Thiyagarajan, S. Mohanraj, M. Kalayarasan, V. P. Raja, and M. Thivagar, "Multi-objective Topology Optimization of Steel Reinforcement for Epoxy Granite VMC Column," in *Advances in Simulation, Product Design and Development: Proceedings of AIMTDR 2021*, ed: Springer, 2022, pp. 343-355.
18. Y. Hao, L. Zhu, B. Yan, T. Ren, J. Zhao, X. Ning, *et al.*, "Stiffness design and multi-objective optimization of machine tool structure based on biological inspiration," *Journal of Vibration and Control,* p. 10775463221085858, 2022.
19. X. Li, Y. Huang, and Z. Zhou, "Structural Design and Optimization in the Beam of a Five-axis Gantry Machining Center," *Journal of Engineering Science & Technology Review,* vol. 13, 2020.
20. Z. Wang, C. Yang, X. Xu, D. Song, and F. Wu, "Layout design of stiffened plates for large-scale box structure under moving loads based on topology optimization," *Mathematical Problems in Engineering,* vol. 2020, pp. 1-11, 2020.
21. X. Lu, Y. Chen, and X. Sheng, "Analysis of the Influence of Structural Parameters on the Modes of a Gantry Machine Tool Crossbeam Based on Analytical Method," *Applied Sciences,* vol. 12, p. 11083, 2022.
22. H. Zhang, Y. Xu, J. Tao, M. Yang, and T. Chen, "Analysis of Static Dynamic Characteristics and Structural Optimization of New Structural Vertical Plus Beams," *Academic Journal of Science and Technology,* vol. 4, pp. 92-97, 2022.
23. E. Yuksel, E. Özlü, A. Oral, F. Tosun, O. F. İğrek, and E. Budak, "Design and analysis of a 5-axis gantry CNC machine tool," in *MATEC Web of Conferences*, 2020, p. 01019.
24. Y. Tian, Q. Shu, Z. Liu, and Y. Ji, "Vibration characteristics of heavy-duty CNC machine tool-foundation systems," *Shock and Vibration,* vol. 2018, 2018.
25. Z. Guo, S. Liu, and H. Wang, "Dynamic modeling and experimental verification for the feeding system of a gantry machine tool based on thermal-mechanical coupling," *The International Journal of Advanced Manufacturing Technology,* vol. 85, pp. 227-236, 2016.
26. Besharati, Seyed Reza, et al. "Multi-objective selection and structural optimization of the gantry in a gantry machine tool for improving static, dynamic, and weight and cost performance." *Concurrent Engineering*, vol. 24, pp. 83-93, 2016.