

# Reliability and structural integrity evaluation of rotating machinery: a case study on turbo-compressors with ANSYS workbench

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**Abstract.** Testbeds are essential structures in industrial labs for conducting machine testing, where the geometry and material properties play a critical role. These tests often use rotating equipment, such as turbines and turbo-compressors (TC). In order to achieve the best possible conditions, testbeds are created using ANSYS Workbench, which incorporates four mechanical procedures: modal, static, harmonic, and transient analyses. Simulations are used to reproduce attributes such as natural frequency, safety factor, and the highest and lowest levels of mechanical stress. This paper describes a quick way to check if the structure of a turbo-compressor is vibrated, by looking at things like its natural frequency, safety factor, and maximum mechanical stress. This study covers unknown factors and uncertainties by analyzing the operational states of the compressor. To achieve this, a reliability structure model and engineering methods like modal analysis for controlling vibrations, structural analysis to ensure the rotor rotates steadily, transient structural analysis to determine the appropriate startup conditions, and harmonic response analysis to determine how speeds change over time, are used to prevent to natural frequencies from interacting with operational frequencies, finally, transient analysis demonstrates initial shock and vibrations that result maximum stress.

**Keywords:** stand test, base plate mount, rotary machine, turbines, test performance, ANSYS.

## Nomenclature

TC	Turbo-compressor
TB	Testbed
SS-beam	Simply-supported beam
FEM	Finite Element Method
SS	Static Stress
DS	Dynamic Stress
TS	Transient Stress
SF	Safety Factor
SF <sub>u</sub>	Ultimate Safety Factor
SF <sub>y</sub>	Yield Safety Factor
ANSYS	ANalysis SYStems

## 1. Introduction

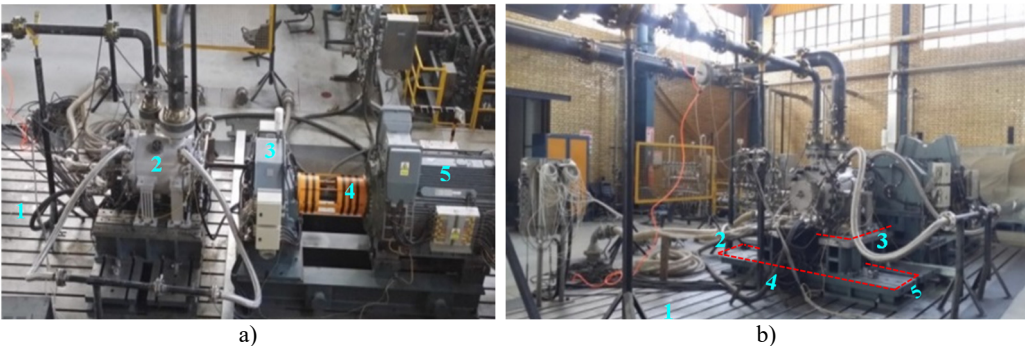
Testbeds (TB) handles environments in company labs for analyzing mechanical performance. Operation safety is paramount in design and testing, with factors like stiffness, mass, damping, and strength influencing structural design, stress, strain, and dependability in different subjects such as modal analysis [1] and crashworthiness [2, 3]. A thorough look at the changes that were made to a three-stage synthesis gas radial turbo-compressor so that it could work in new situations. It examines the scope of the upgrades, the methodology used, the results of the tests conducted,

and the continued performance of the compressor since 2015 [4]. The high-pressure super turbo, a centrifugal compressor equipped with an integrated gear system, has undergone modifications to improve rotor stability and minimize vibrations and temperature rise for high-pressure applications [5]. Similarly, a review examined the progress of high-pressure carbon capture technology under different operating situations [6]. For example, the compressor prototype presented exceptional efficiency, no noise and minimum resonance by use of aerodynamic and structural optimization techniques including a combustion chamber designed within the gears [7, 8] or optimized geometry of space, location, or safety concerns with compressor [9]. In the same way, a gas turbine optimized its efficiency, fuel economy, and operational flexibility through the use of GasTurb and CMap software [10]. In addition, by utilizing artificial intelligence methods, the system monitors and maintains the rotating equipment to prevent industrial damage in induction motors, roller bearings [11], misalignment, magnetic bearing loading [12], specifically, examine humidity in an air energy storage system [13-15], or just performance in high pressure [16], to illustrate in a multistage compressor [17], to solve this problem with an active vibration damper improving efficiency [18]. Another condition monitoring focused on mistuning of impeller blades in real-life stages to avoid resonance [19], highlighting the relationship between IBC flow and AVDs by assessing static pressure and temperature by controlling vanned diffusers[20] these vanes impact on flow distortions [21], in surge in unstable flow conditions by monitoring pressure output [22] and stall margin is influenced of surface roughness [23], simultaneously surge and stall [24], noise level in centrifugal compressor [25], vibrations in bearing turbocharger of internal combustion engines [26], as an example, in variable such as speed drive for centrifugal compressor suggested to control by the quadratic V/F [27], utilized a micro-perforated panel absorber in output pipe in which a two-stage compressor to mitigate noise [28], pressure gradient and frequency of pulse indicate instability [29], and leakage flows in hub and shroud lead to less efficiency [30]. Statistically, used the Bayesian models tested for a rig was pulsated on blade tip [31], correlation between rotating stall pressure and blade vibration to reduce stall resonance [32], numerically optimized vibrational noise in centrifugal fan [33] and turbine volutes [34]. In specified model, assessed the performance of 280-mm-diameter tilting pad journal bearings used in turbo-machines in order to examine their static and dynamic characteristics, power loss, and dynamic coefficients under different operating situations [35]. In rotary system and fault diagnosis, there has been researched to control multivariable in gas turbine like fuel inlet and IGV parameters through optimal PID [36], other cases experimented unbalanced shaft and clearance of bearing by driven data in ANFIS and ANN-PSO, respectively [37, 38]. In current study analyze TB as a beam needs to investigate theoretically, early studies in vibrations of beam, presented the non-linear stiffness calculated cantilever in large amplitude vibrations, although linear theory of vibrations may not be valid for high amplitude vibrations [39], another case studied the spring-hinged beam [40], take an experiment by using pouch cells designed with finite element method (FEM) [41]. A study compared strain and stress in cylindrical shell membrane regions during pressure tests using HSLA 15CDV6 pressure vessels. Non-linear FEM results show similar strain as that of test along the hoop direction, and the values of strain and stress are increased due to the presence of profile variation [42] in the cylindrical shell. An article analyzed prismatic and non-prismatic tapered cantilever beams using large amplitude free vibrations. It presents massive deflection data and uses a polynomial function viewpoint to construct a nonlinear solution. The study reveals nonlinearity in the hardening type of prismatic and non-prismatic cantilever beams [43]. By reviewing optimal vibration control methods [44] by using piezoelectric sensor such as LQR-MOPSO [45, 46], Fuzzy-PID [47], and MOPSO-PID controllers [48], and in passive control, applying band-gap approach by use of quasi-zero stiffness as unit cells in different location [49]. This study uses ANSYS and SOLIDWORKS software to look into the best places to put augmented supports for testing turbo-compressor (TC) machines. The study focuses on the shape and material qualities of TB to ensure accurate test conditions. It goes into detail about four mechanical processes in the ANSYS Workbench, with a focus on safety, natural frequencies, and mechanical stress to make sure that TC models are reliable. The

study's goals are to look at and understand the main parts of an industrial TC, make real models of different types of industrial compressors, and find the best places to put support beams under the TC to test how well it works. The best algorithm takes into account experimental knowledge, the TBs of critical areas, and the TC's impact on work time. It is based on material strength, numerical methods, and trial and error methods.

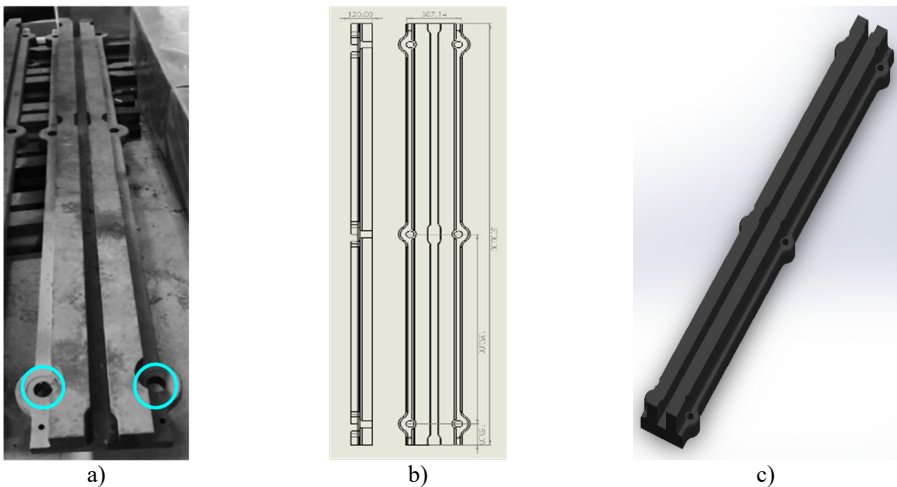
2. Methodology

The TCs vary in size, shape, capability, and performance, but all of them consist of key components such as the shaft, gearbox, impeller, bearing, and motor, as shown in Fig. 1(a). TBs are used in various directions to secure the TC and prevent unwanted vibrations while the TC is fixed, as illustrated in Fig. 1(b).



**Fig. 1.** Applications of testbeds in rotary machines: a) main parts of TC (1 – testbed, 2 – turbo-compressor, 3 – gearbox, 4 – shaft, 5 – motor), b) different TB under TC (1 – TB1, 2 – TB2, 3 – TB3, 4 – length, 5 – width)

In Fig. 2, the TB is shown below as real, drawn, and 3D models. The size of the model was real-sized.

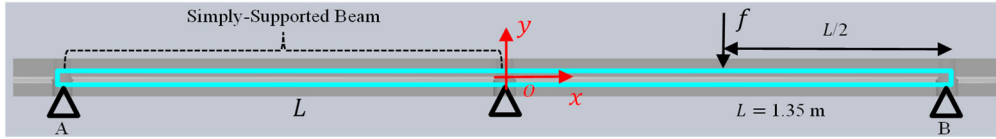


**Fig. 2.** Testbed structure: a) manufactured models, b) dimensions, c) 3D design

2.1. Euler-Bernoulli theory

Primarily, in actual circumstances, the TB is affixed to the ground using six holes and supported by bolts (cyan circles in Fig. 2(a)). Assuming a two-dimensional diagram, TB has

combined two simply supported beams (ss-beams). Fig. 3 depicts a double-ss-beam with a point load positioned at its center.



**Fig. 3.** Analogy model of the testbed to simply supported beam

The Euler-Bernoulli theory is a key theorem used to analyze the mechanical behavior of beam structures. Therefore, in the Euler-Bernoulli beam theory, it is assumed that there is no shear stress in beam deflection. As a result, the displacement is denoted as  $w(x)$  and the bending moment is denoted as  $M(x)$ . The displacement of a simply supported beam,  $w(x)$ , is described as follows:

$$M(x) = \begin{cases} \frac{fx}{2}, & 0 \leq x \leq \frac{L}{2}, \\ \frac{f(L-x)}{2}, & \frac{L}{2} < x \leq L, \end{cases} \quad (1)$$

$$w(x) = \begin{cases} -\frac{fx(4x^2 - 3L^2)}{48EI}, & 0 \leq x \leq \frac{L}{2}, \\ \frac{f(x-L)(L^2 - 8Lx + 4x^2)}{48EI}, & \frac{L}{2} < x \leq L. \end{cases} \quad (2)$$

By putting the middle load of the beam (as shown in Fig. 3) into Eq. (2), that calculates the maximum deflection of the simply supported beam at the midpoint of the beam:

$$w_{max} = \frac{fL^3}{48EI'} \quad (3)$$

$$\sigma_{max} = \frac{3fL}{2bh^2}. \quad (4)$$

In order to assess Eqs. (1) and (2), it is necessary to consider the parameters listed in Table 2, which may be determined during the procedure. Fig. 4 depicts the cross-section of the TB structure, where the black region represents the actual cross-section, while the red rectangle area represents a simplified version used for calculating Eq. (3).

**Table 1.** Material properties of the testbed

Parameters	Values	Parameters	Values
$E$	1.2e11 Pa	Poisson ratio	0.25
$I_x$	446.91 kg.m <sup>2</sup>	$f_1$	675 KN
$I_y$	450.91 kg.m <sup>2</sup>	$f_2$	108 KN
$I_z$	450.49 kg.m <sup>2</sup>	$f_3$	135 KN
$\rho$	7250 kg/m <sup>3</sup>	$f_4$	2025 KN

It is important to know the safety factor (SF) when judging how reliable a planned model is. The SF is found by comparing the planned values to the experimental values as a ratio. An SF greater than 1 is considered acceptable by engineers, while an SF less than 1 means the model failed. The SF is shown in two formats: ultimate strength and yield strength:

$$SF_u = \frac{S_u}{\sigma_u}, \quad (5)$$

$$SF_y = \frac{S_y}{\sigma_y} \tag{6}$$

Indices u and y define the ultimate and yield statements. They represent the tensile ultimate and/or yield strengths.  $S_u$  and  $S_y$  are the maximum values of the  $\sigma_u$  and  $\sigma_y$  strengths, respectively. Furthermore, the quantity of  $S_u$  or  $S_y$  is derived using empirical standard testing. Additionally, based on the mechanical characteristics of the material considered in the test bed construction, the tensile ultimate strength is 600 MPa and the yield strength is 370 MPa.

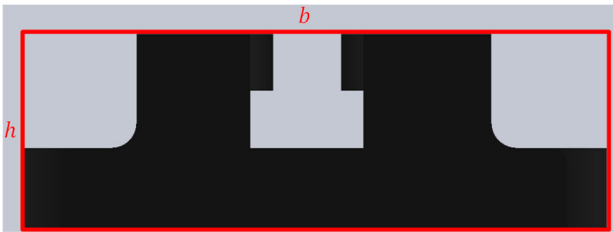
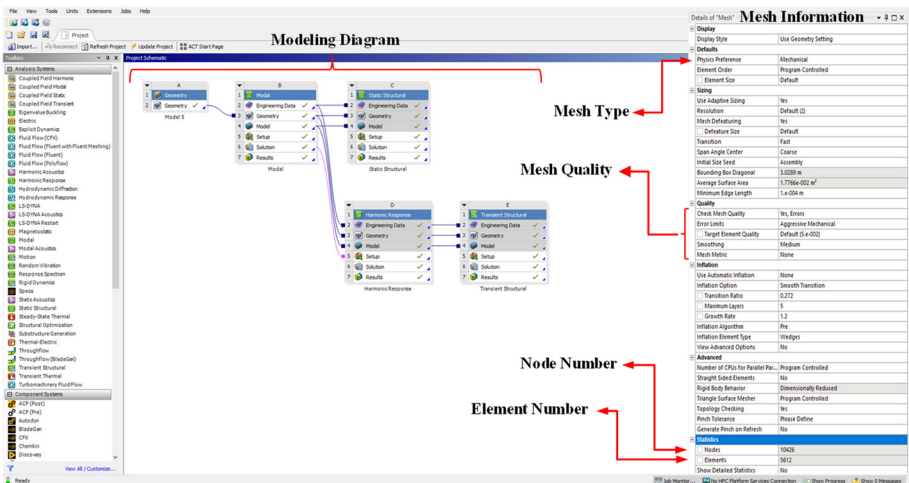


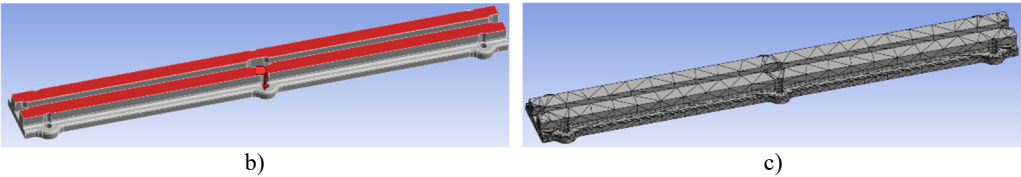
Fig. 4. Simplification cross-section of the testbed

2.2. Methodology

The FEM is one of the most practical numerical methods used for modeling and analyzing structures in various engineering research fields. ANSYS, a commercial FEM software, efficiently analyzes mechanical behaviors under different physical conditions, geometries, and materials. Based on the three-dimensional model (Fig. 2(c)), which is imported into ANSYS Workbench in .stp, .x\_t, or .igs format, the analysis follows a step-by-step approach across multiple studies, including modal, static, harmonic, and transient modeling to evaluate the mechanical properties of the TB. Considering the boundary conditions, such as supports at different locations and automatic meshing of the structure, the modal analysis is first performed to extract the natural frequencies of the structure. Subsequently, distributed loads are applied to the top face of the structure, preparing the TB for further analyses. For instance, the static study determines the Von Mises Stress (VMS) to evaluate the ultimate safety factor. Based on the extracted natural frequencies, the harmonic analysis is then conducted to examine VMS under cyclic loads, helping to compute the yield safety factor. Finally, the transient analysis monitors the VMS of the TB at the initial loading phase, where the operational load is applied as a step force (see Fig. 5).



a)

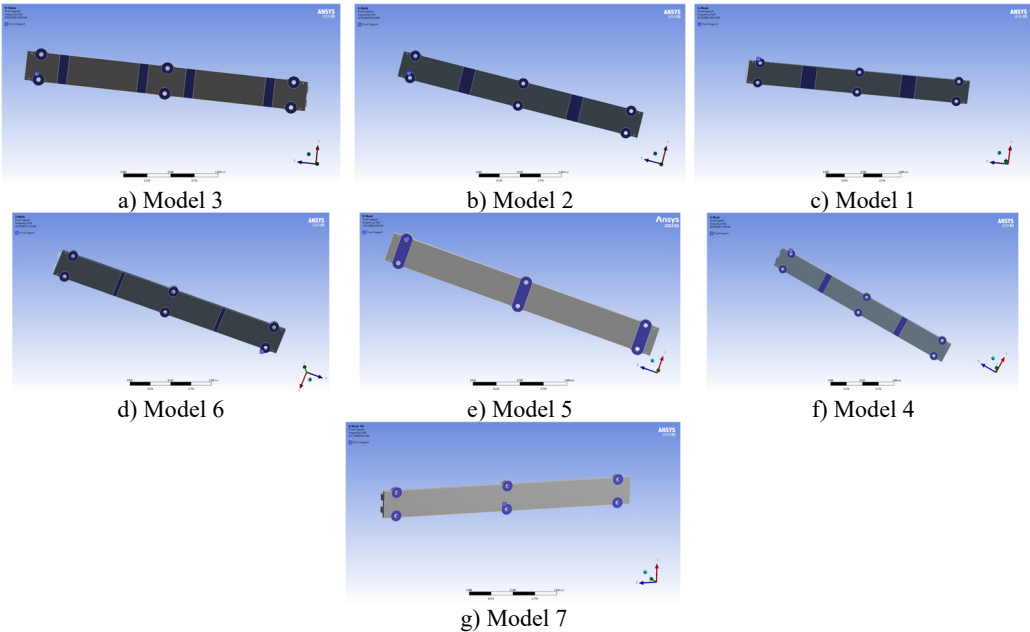


**Fig. 5.** FE procedures in ANSYS workbench: a) configuration model, b) loading, c) meshing

3. Result and discussion

This design and simulation will demonstrate three mechanical processes of the test bed in the ANSYS-Workbench area: modal analysis, static analysis, and harmonic analysis. Ultimately, the safety factor of the component will be presented in a tabular format. During the simulation process, seven models were implemented with supports of varying sizes under the test bed structure. These models were used to simulate the analyses mentioned in the advanced comparison. The goal was to select the best model based on the optimal mechanical strength of the material during compressor operation and its performance in the lab tests. The primary factor in design is the parameter d, which represents the width of rectangular sections situated between holes. This number varies across all models. The following outcomes derived using ANSYS-Workbench are displayed: The third and fourth variants change just slightly in the quantity of rectangular supports. The determination of support zones is based on the highest deflection seen in Fig. 6. They transition between openings of varying dimensions.

At first, in the ANSYS, mechanical properties are used for structure are followed as Table 2.



**Fig. 6.** Assumed supports in proposal models

**Table 2.** FEM’s properties in ANSYS

Parameter	Value
Young modulus (Pa)	2e11
Density (Kg/m <sup>3</sup> )	7850
Possion’s ratio	0.3
Shear modulus (Pa)	7.6923e10

3.1. Simulation process

The table below presents numerical data obtained from ANSYS Workbench 2019-R3, including static stress (SS), dynamic stress (or harmonic analysis) (DS), and transient stress (TS). These results are used to improve the reliability of the design. The selection models operate according to the following methodology:

1) Static Analysis: This analysis evaluates the long-term effects of the continuous application of force by the rotor on the structure of the TC.

2) Dynamic Analysis: This type of analysis looks at how the system responds to harmonics, which means that the angular speed of the TC changes over time while it is working.

Analysis Steps:

**Step 1:** Modal Analysis. To find resonance events, modal analysis simulates all the proposed models in ANSYS and compares them to a reference model that doesn't have any support. The natural frequencies of the proposed models show significant differences from those of the reference model, as illustrated in Fig. 7 and Table 3.

**Step 2:** Static Analysis. Static analysis checks to see if the proposed models' material strengths are realistic and stay below standard values, like ultimate tensile and yield strengths. This information is provided in the table.

**Step 3:** Safety Factor for Static Analysis ( $SF_u$ ). Eq. (5) is used to find models that can be used for further analysis in ANSYS by calculating the  $SF_u$  values of the static analysis. Proposed models with an  $SF_u$  value below 1, such as models 4 and 7, are rejected.

**Step 4:** Harmonic Analysis. Harmonic analysis looks at how the TC reacts to the forces that the rotor puts on its structure at different angular speeds.

**Step 5:** Safety Factor for Harmonic Analysis ( $SF_y$ ). Eq. (6) is used to figure out which proposed models should be left out of the next steps by finding their  $SF_y$  values for harmonic analysis. Models 7 is rejected based on these calculations.

**Step 6:** Transient Analysis. In ANSYS, transient modeling looks at the system's starting conditions, which could be changed by unknowns like electrical or mechanical shocks or impulses.

**Step 7:** Final Model Selection. The ultimate selection narrows down to models 1 to 6. Among these, model 5 demonstrates superior performance and is strongly recommended for implementation under real operating conditions.

In order to avoid resonance occurrences, it is crucial to take into account the natural frequencies of all models. The Fig. 7 and Table 3 show that when models 1 through 7 are looked at as a whole, their natural frequencies are very different from the test bed's natural frequency structure, in Table 3.

Table 3. Natural frequencies of the 7 models

Model (Hz)							
Pure	1	2	3	4	5	6	7
0	289.63	1306.1	1214.9	1124.1	1036.1	838.4	333.2
0	317.23	1340.5	1244.9	1158.4	1073.8	844.06	350.43
3.38E-04	553.96	1347.8	1261.9	1184.4	1117.2	1136.8	567.47
3.76E-04	557.09	1432	1346.2	1260.6	1189.1	1140	570.88
7.98E-04	643.82	1682.2	1587.6	1503.8	1419.8	1340.6	675.98
1.10E-03	673.73	1712.9	1603.2	1521.8	1449	1342.1	698.54
68.84	741.6	1726.4	1627.4	1548.3	1478.2	1995.4	852.44
156.59	819.46	1736.2	1645.5	1566.3	1490.9	2006.8	899.85
188.58	1208	1919.6	1820.6	1721.9	1624.8	2495.2	1237.4
220.02	1217.8	1930.6	1828.2	1735.1	1644.5	2500.6	1248.8

According to the Table 3, the natural frequencies of all the models are significantly shown different from those of the original model (Fig. 7), which is an outdated form of the structure. Therefore, models one to seven are eligible for usage under TC in the lab. However, it must need

to choose the most optimal model using the trial-and-error approach. Consequently, to suggest using supplementary simulation methods to assess the material durability of all models, each with distinct support positions. To clarify, this approach involves using basic analytical techniques such as static and harmonic approaches to represent many proposals and determine the most optimal choice. The forces applied to the structure are dispersed according on the lowest and maximum angular velocity of TC’s rotor in our lab. In Table 4, these forces are 50 to 150 tons per meter and are spread over the surface of the structure in a normal distribution, based on the weighted force formula  $F = mg$ .

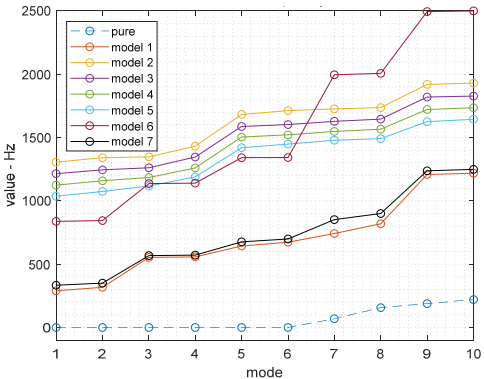


Fig. 7. Natural frequencies of the 7 models

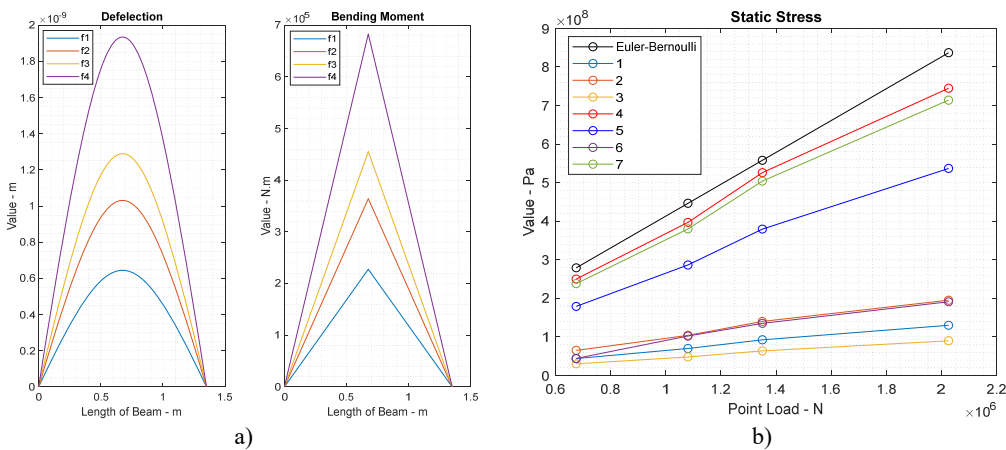
Table 4. ANSYS results of seven models alongside model validation checks

No	d (cm)	ton/m	SS (Pa)	$SF_u$	Check 1	DS (Pa)	$SF_y$	Check 2	TS (MPa)
1	20	50	21685	20.48	Pass	106230	20.47	Pass	29.24
		80	34696	12.8		169970	12.79		46.785
		100	43370	10.24		212460	10.23		58.48
		150	65055	6.83		318690	6.82		87.72
2	15	50	53864	19.73	Pass	177280	19.7291	Pass	30.34
		80	86183	12.33		283650	12.33		48.56
		100	107730	9.86		354560	9.86		60.71
		150	161590	6.576		531840	6.57		91.09
3	10	50	57500	19.54	Pass	22325	19.5	Pass	2.03
		80	92000	12.2		35721	12.21		48.81
		100	115000	9.77		44651	9.77		61.02
		150	172500	6.5135		66976	6.51		91.57
4	10	50	54977	18.5	Pass	190340	18.5	Pass	32.34
		80	294820	2.53		18354	2.53		230.21
		100	109950	9.25		380670	9.25		64.69
		150	164930	6.16		571010	6.17		97.07
5	13	50	184260	4.0516	Pass	11471	4	Pass	143.67
		80	67369	11.26		397890	11.26		53.15
		100	368530	2.02		22942	2.02		288.06
		150	552790	1.3		34414	1.35		433.16
6	5	50	42106	18	Pass	248680	18	Pass	33.21
		80	267510	1.82		22213	1.8		318.07
		100	84212	9		49737	9		66.44
		150	126320	6.1		41469	6		99.69
7	0	50	167200	2.91	Failed				
		80	267510	1.82					
		100	334390	1.45					
		150	501590	0.97					

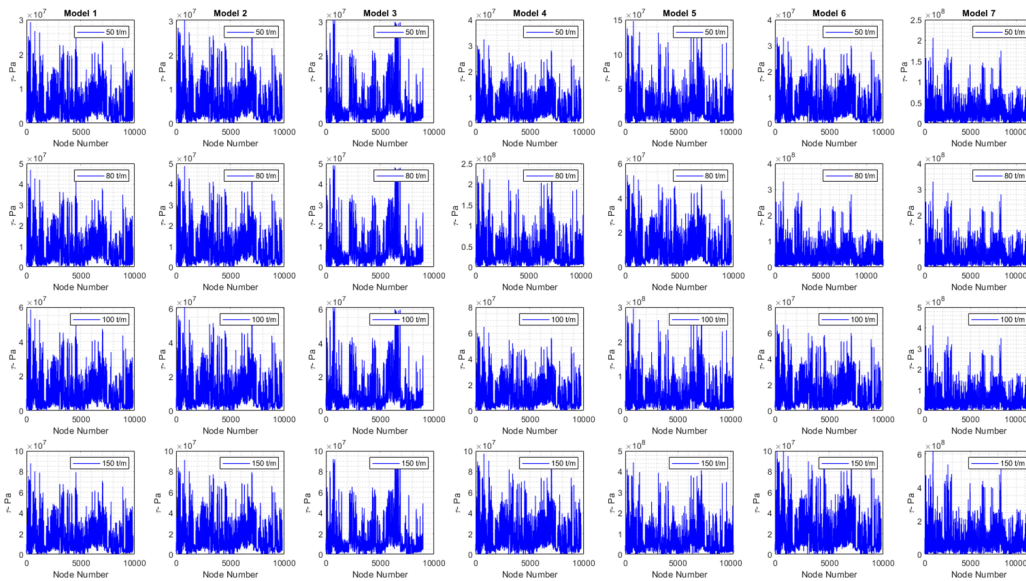


Fig. 8 displays the bending moments and deflections of ss-beam with different inputs, namely point loads specified in Table 4 (third column), as determined by Eqs. (1) and (2).

Above all the mentioned steps, the FEM's results derived from Table 4 (Fig. 5) include seven models, as shown in Figs. 9-14, which illustrate the VMSs in different analyses: static (Fig. 9), harmonic (Fig. 10), and transient (Fig. 11) compare the maximum mechanical stress in different models at each node. Similarly, based on the accepted six models in Table 4, Figs. 12 and 13 clearly compares the ultimate and yield safety factors, determining which model provides the best performance. Finally, model 5 stands out as one of the best designs, demonstrating high material and structural strength across different analyses and operating conditions.



**Fig. 8.** Comparison results: a) maximum torque and displacement of ss-beam based on TB's properties, b) mechanical stress in static forces



**Fig. 9.** Maximum stress of nodes in static analysis

In detail, Fig. 14 the numerical results (FEM models in ANSYS) for six acceptable models based on Table 4. These results include static, harmonic, and transient analyses. Models 1 to 6 demonstrate reasonable strength under various load conditions. Hence, the safety factors ( $SF_u > 1$  and  $SF_y > 1$ ) indicate that the maximum mechanical stresses in models 1 to 6 exceed

their ultimate and yield strengths under different loads. Therefore, these models are analyzed with different support configurations based on the boundary conditions of the TBs in various directions. Elaborately, Fig. 14 compares various maximum stress, safety factors from previously accepted models and highlighting the most notable ones.

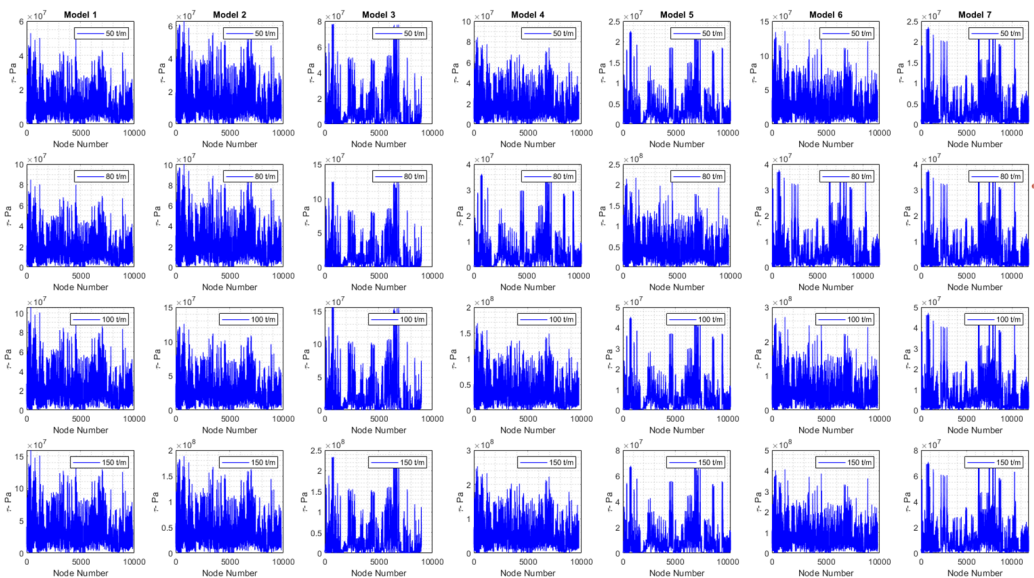


Fig. 10. Maximum stress of nodes in harmonic analysis

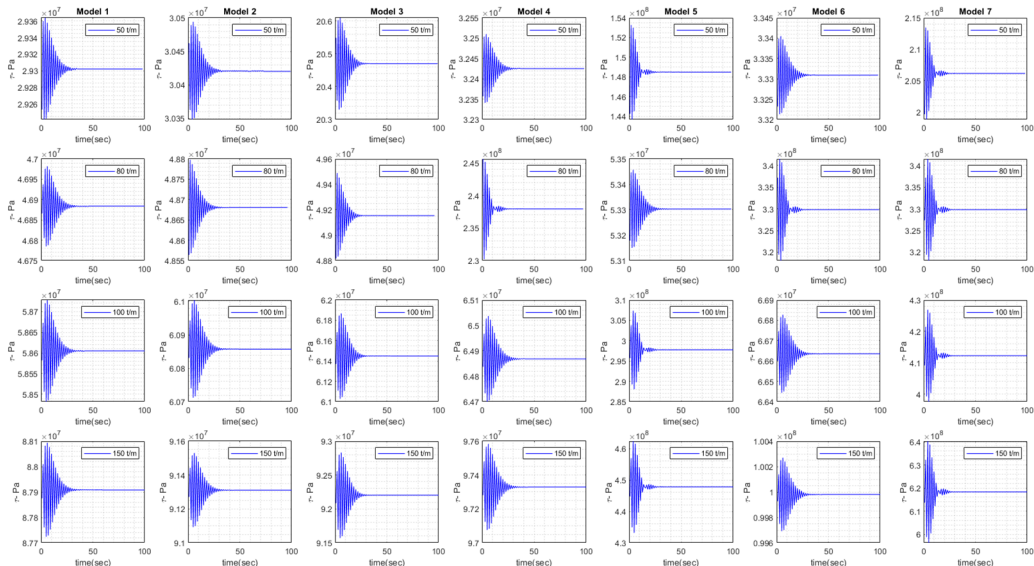


Fig. 11. Maximal stress in transient analysis

Currently, it is deemed acceptable to thoroughly study all seven models in order to proceed with the next phases. Utilizing ANSYS, the FEM of the fifth recommended model was analyzed using four methods: modal, static, harmonic, and transient studies. The contour spectrum in the following illustrations represents three-dimensional displacements and corresponding stress (see Figs. 8-11). Based on Fig. 5, four types of analyses are conducted in the tree design.

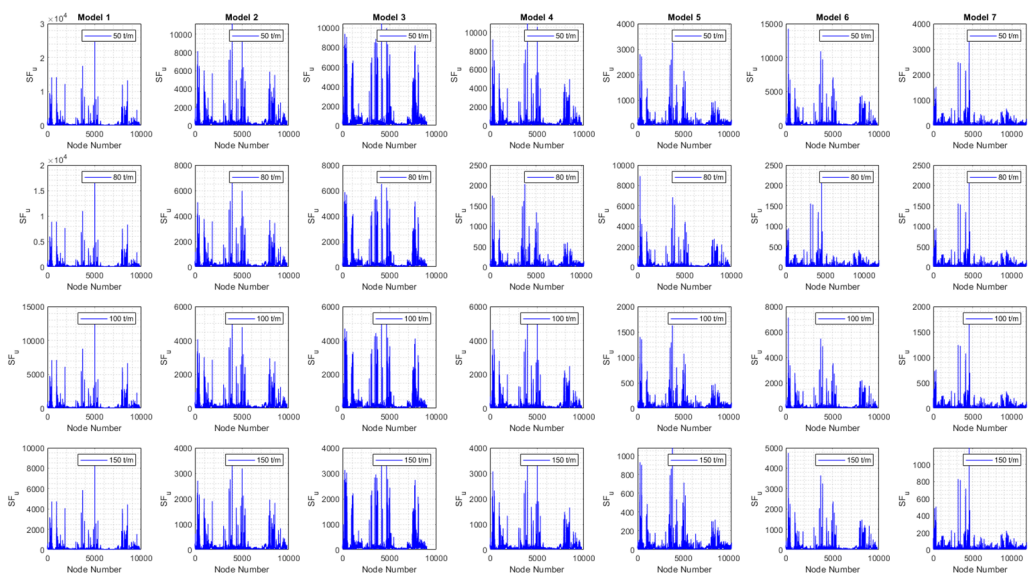


Fig. 12. Ultimate safety factor of the 7 models

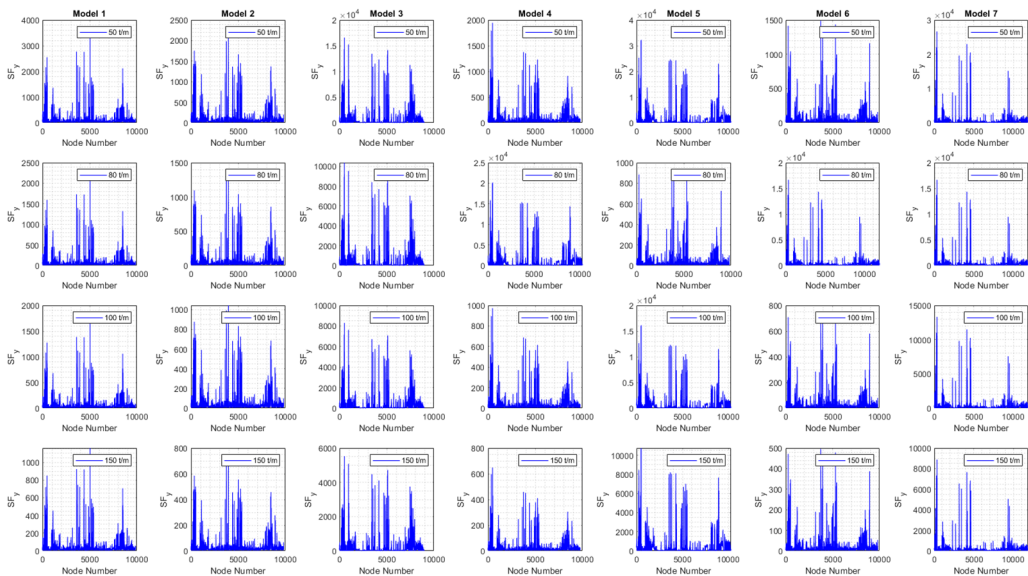


Fig. 13. Yield safety factor of the 7 models

In the modeling phase, a triangular mesh (mechanical type) is automatically and adaptively generated, as shown in the mesh information section of Fig. 5. For example, the number of nodes and elements are 10,426 and 5,612, respectively. At this stage, the model is ready for modal analysis. Secondly, fixed support is assumed at three positions at the bottom of the holes. Thirdly, the load distribution is applied to the left and right top faces of the structure with varying values of 50, 80, 100, and 150 tons per meter. The model is then utilized for static, harmonic, and transient analyses. In the harmonic analysis, the lower and upper frequency bands are determined based on the minimum and maximum natural frequencies extracted from the modal analysis. In the transient analysis, the applied load is assumed to be a step force with a time step of 0.01 seconds. As a consequence, the corresponding results of the shape modes, three-dimensional displacements, and mechanical stress for different loads in model 5 are shown in Fig. 15, Tables 5, 6, and 7 illustrate

the spectrum, where the critical zones are indicated by maximum values in red and minimum values in blue.

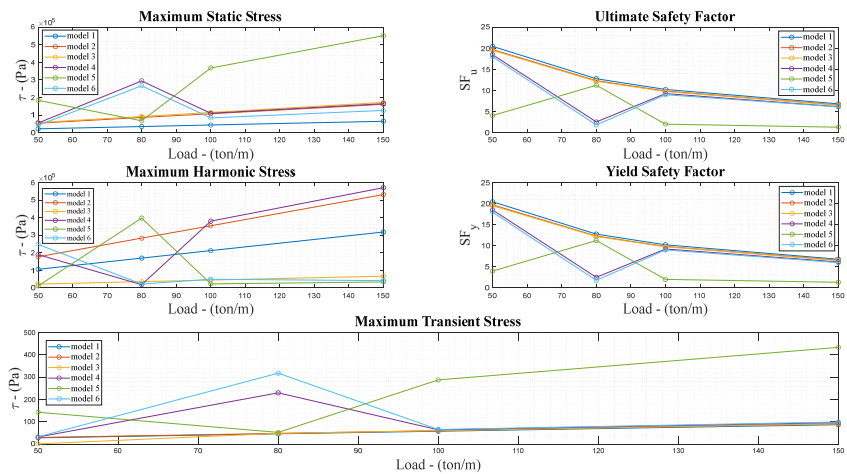


Fig. 14. Comparison maximum stress and safety factors of the 6 models

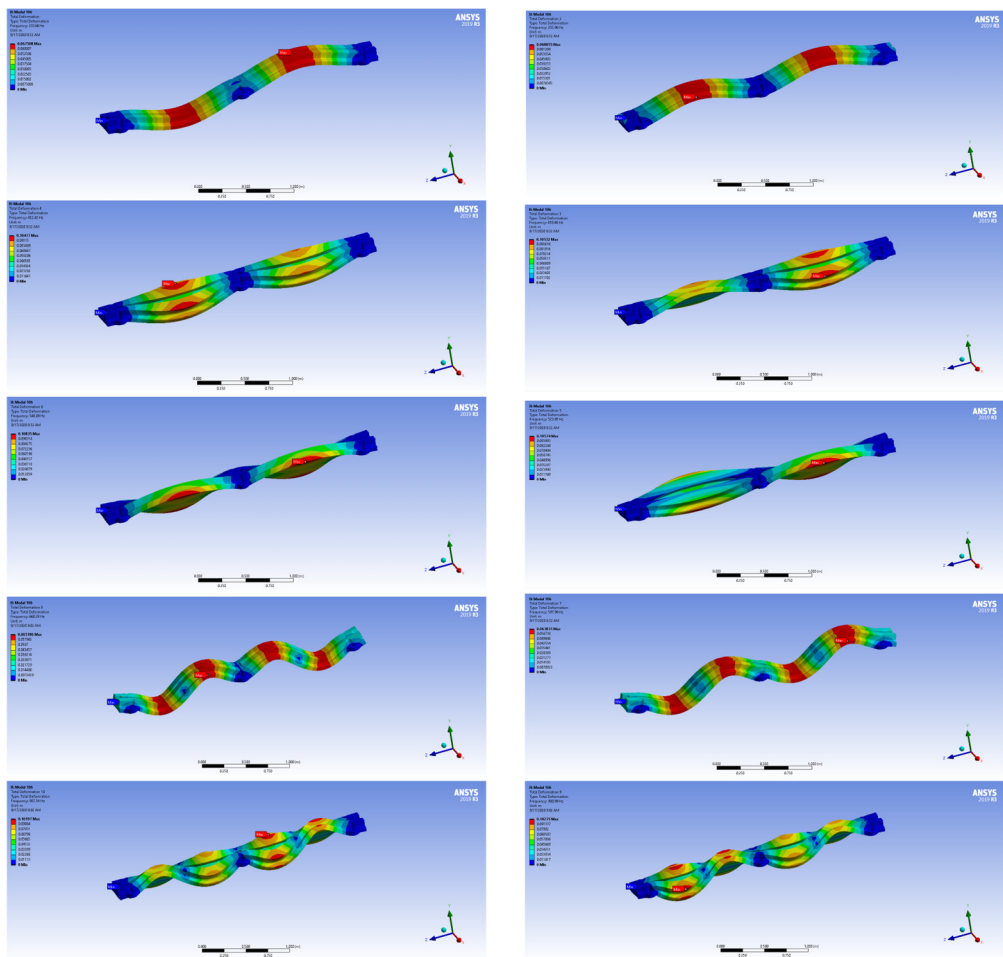
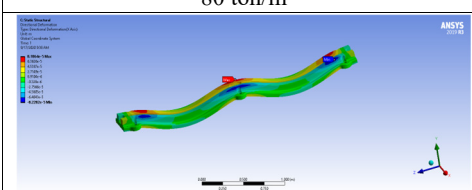
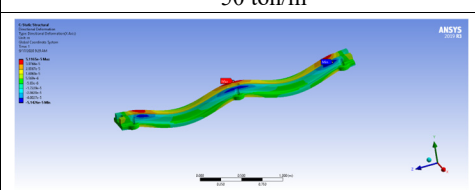
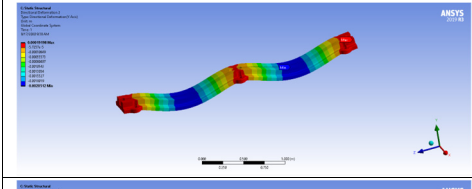
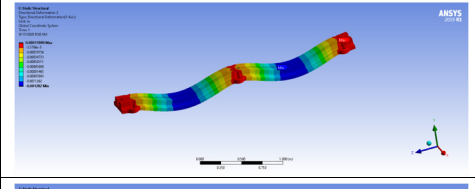
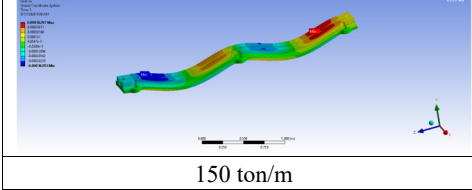
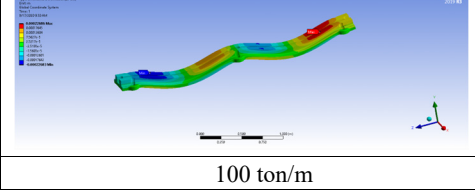
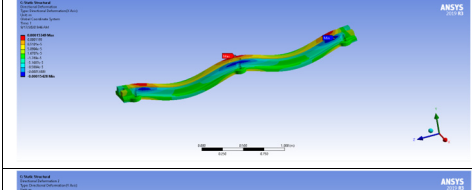
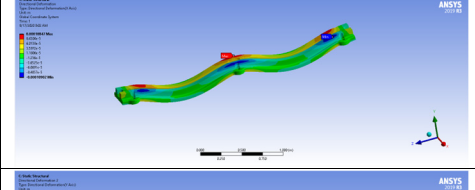
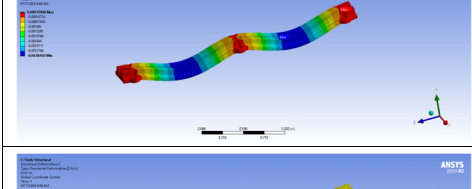
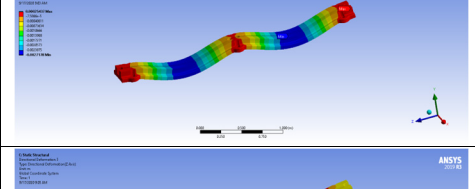
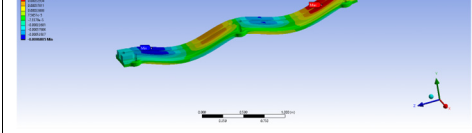
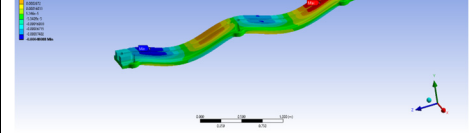
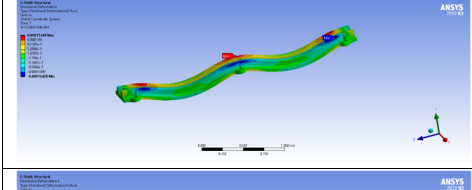
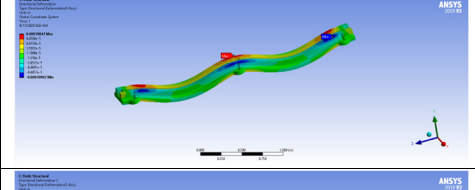
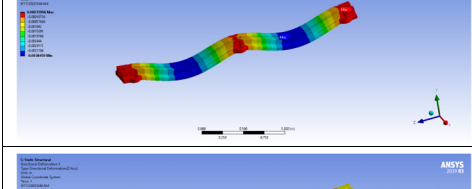
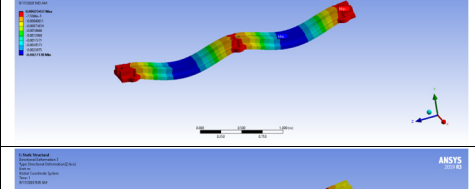
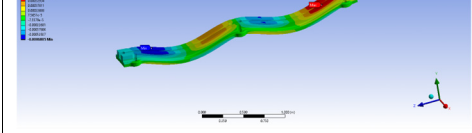
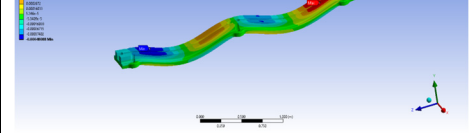


Fig. 15. Ten mode shapes of the fifth model

Table 5. Three-dimensional displacements of distributed forces for the best model

80 ton/m		50 ton/m		X
				
				
				Y
				
				
				Z
150 ton/m		100 ton/m		
				X
				
				

In short, Table 8 provides a brief description of each procedure in SOLIDWORKS and ANSYS simulations for understanding why the optimized model is considered the best model in this research. Finally, the fifth model is nominated as the optimal design, as shown with the size of supports in Fig. 16.

In short, the entire process of this research is summarized in Table 7.

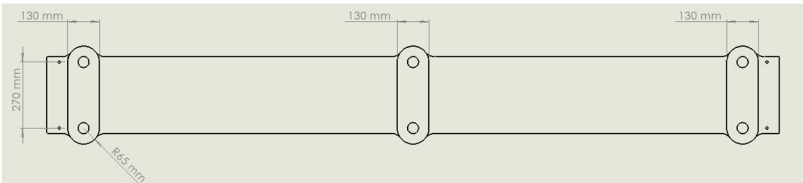
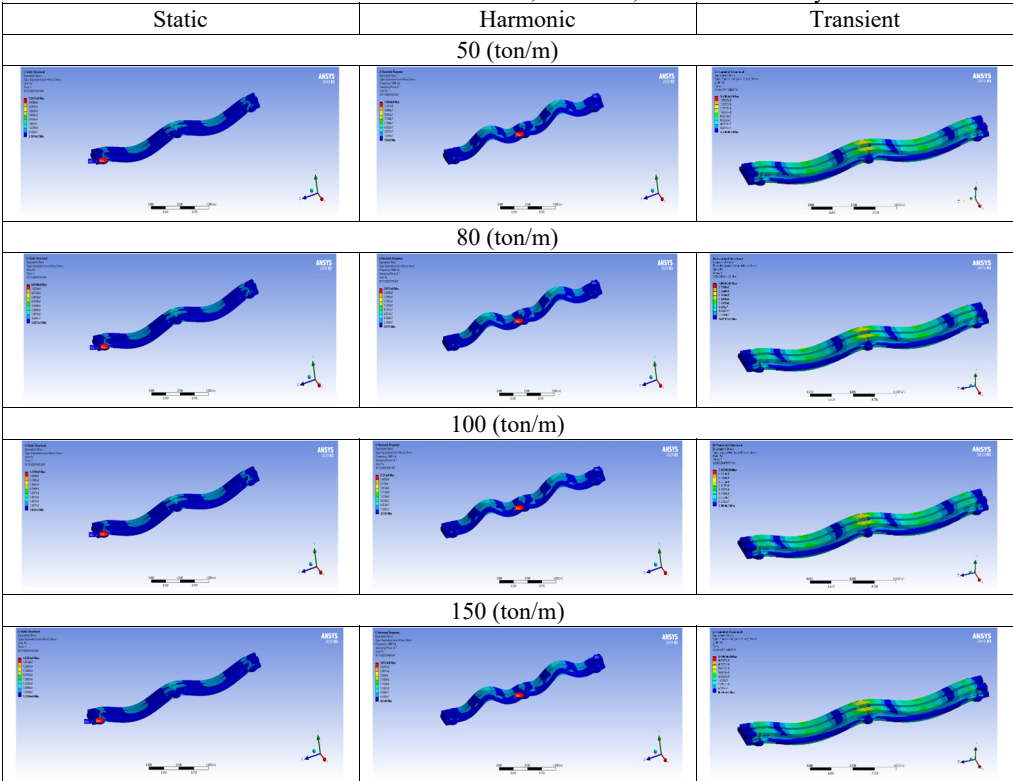


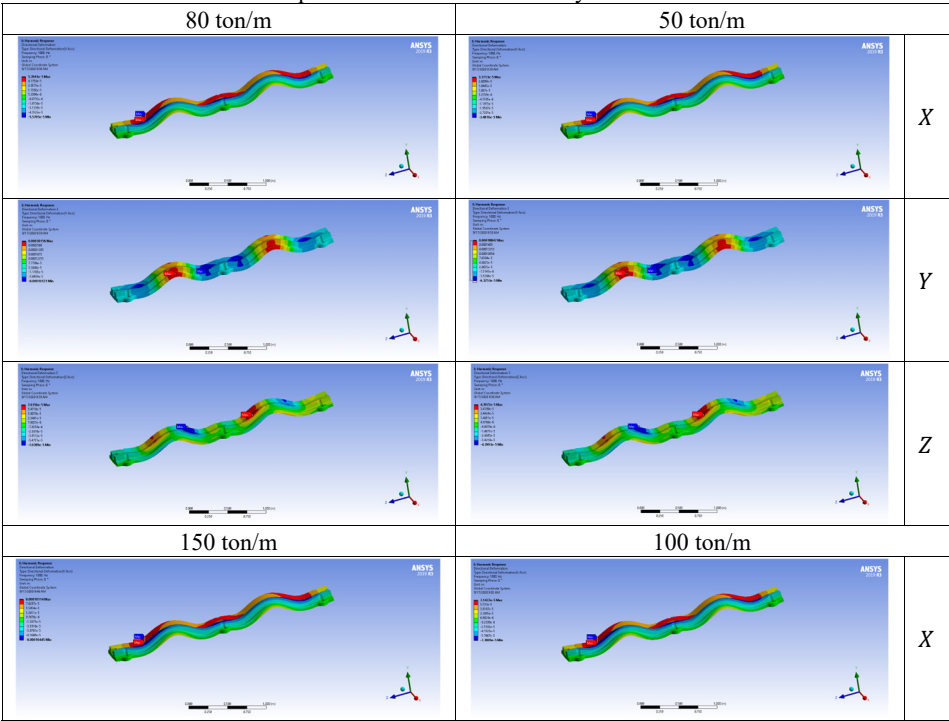
Fig. 16. Support size in model 5

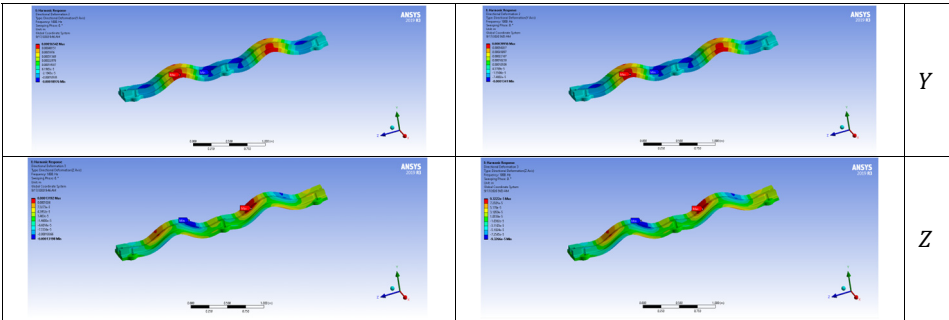


**Table 6.** Stress contours of the best model in static, harmonic, and transient analyses for model 5



**Table 7.** Three-dimensional displacements of harmonic analyses with distributed forces for model 5





**Table 8.** Simulation process for the optimal model

Step	Task
Drawing	Design feasible and different supports in SOLIDWORKS (Figs. 1(c), and 6)
Modal analysis	Calculate deflections and natural frequencies of purposed models (Eq. (2), Figs. 7-8, 15-16, and Table 3)
Static analysis	(1) Simulation in ANSYS (Figs. 9 and 18) (2) Calculate $SF_u$ of selected models (Fig. 12) (3) Select new models with $SF_u > 1$ (Fig. 14)
Harmonic analysis	(1) Simulation in ANSYS (2) Calculate $SF_y$ of selected model from static analysis (Fig. 13) (3) Select new models with $SF_y > 1$ (Fig. 14)
Transient analysis	(1) Simulation in ANSYS (2) Get and select maximum mechanical stress
Best model	Presentation of the structure with the highest strength (Fig. 18)

4. Conclusions

This research examined the most favorable position for turbo-compressors under the test bed construction in several scenarios, including initial operation, shock, impulse, long-term operation, and speed switching in the test laboratory. This work aims to address unknown and uncertain characteristics and variables by using strength material methodologies, such as free vibration and forced vibration methods. Turbo-compressor problems like misalignment, not enough lubrication, bearing clearance, imbalance, wear, erosion, corrosion, impeller cracks, worn seals, contamination, bent shafts from cracks, bending, and deflection, uncoupling problems, vibration and resonance, and not enough cooling due to blockages can make the data acquisition go wrong. Because these flaws in the manufacturing process are hard to see and can’t always be predicted, it’s important to include certain studies during simulation and design. Modal, static, harmonic, and transient studies enhance the structural integrity in the presence of uncertainty. Systematically, to conduct several studies to determine the most suitable option is selected one of the three available models as the optimal design for an actual turbo-compressor test.

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Data availability

The datasets generated during and/or analyzed during the current study are available from the corresponding author on reasonable request.

## Conflict of interest

The authors declare that they have no conflict of interest.

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