

Comparative analysis of three different models of navigational bridge wings

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Abstract. This paper focuses on the local vibration problems that occur at the extensions of the navigation bridges. These structures are modeled as systems composed of beams and analysed both from the analytical point of view and the FEM method, thus trying to determine the pulsations and modal forms of the structure. In critical moments, resonance is high, which has led to the modification of the shape of the wings. Thus, the wings models fixed by the structure of the laundry at one point and the other on the main deck were adopted. Methods and Research: Three wing models with fixed lengths, formed by beam systems fixed at certain points, under 90° and different materials are analysed. FEA was done using the FEMAP v11.2 software and the analytical study was done with the Matlab software. Results: The analysis shows that the beams' natural frequencies [00-90] are noticeably lower than the frequencies obtained in the other models. Also, the values of their natural frequencies are higher for FEM due to lack of damping. And AISI 1025 Steel proves to be more appropriate to the construction of the beams because it has lower resonance frequencies.

1. Introduction

Compliance with the requirements of the foreign market by the shipbuilding industry in Romania required the introduction of some methods for estimating the vibration level at the design stage. Developing appropriate modeling supported by a powerful computing technique, based on empirical formulas and finite element method, helps to avoid significant vibration problems. By using existing technologies, these possible problems can be discovered from the initial design stages. The first phase in the design of the ship was called the Conceptual Design phase and represents the starting point in the vibration prevention process.

To limit the level of vibration on board, classification societies have developed international standards for this purpose. [1] They also made some recommendations on calculating and measuring vibrations. [2]

This paper focuses on the study of local vibrations occurring at the wing structure of the ship's navigating bridge. These vibrations adversely affect all electronic navigation systems on the navigation deck, sometimes leading to even their failure. Resonances occurring at the level of the upper deck have a negative influence on its extensions, called wing of the navigation deck.[3]

The purpose of these structures is to provide sailors with good visibility in both ship's edges during maneuvers, as well as the possibility of installing electronic navigation devices.



Long wings are the most critical areas of the ship in terms of resonance. The elasticity of the wing is influenced to a large extent by its length. The two features are directly proportional. To limit the negative effect of resonance, it is recommended to support the main beam on a vertical one fixed to the main deck. [4]

The paper presents the finite element method, but also an analytical method, of three wing models for two types of building materials: SAE-AISI 4340 Ni-Cr-Mo Steel and SAE-AISI 1025 Carbon Steel.

1.1. Comparison of SAE AISI 4340 Ni-Cr-Mo Steel with SAE-AISI 1025 Carbon Steel

Both SAE-AISI 4340 and SAE-AISI 1025 are iron alloys. They have a very high 96% of their average alloy composition in common. [5]

Table 1. Comparison of the mechanical properties of the two iron alloys [5].

Properties	SAE-AISI 4340 Ni-Cr-Mo Steel	SAE-AISI 1025 Carbon Steel
Brinell hardness	220 to 360	130 to 140
Elastic modulus	190 GPa	190 GPa
Elongation at break	12 to 22 %	17 to 28 %
Fatigue strength	330 to 740 MPa	190 to 280 MPa
Poisson's ratio	0.29	0.29
Shear modulus	73 GPa	73 GPa
Shear strength	470 to 770 MPa	290 to 310 MPa
UTS	750 to 1280 MPa	450 to 500 MPa
Proof	470 to 1150 MPa	250 to 420 MPa

Table 2. Comparison of the alloy composition of the two iron alloys [5].

Properties	SAE-AISI 4340 Ni-Cr-Mo Steel	SAE-AISI 1025 Carbon Steel
Carbon, C	0.38 to 0.43 %	0.22 to 0.28 %
Chromium, Cr	0.7 to 0.9 %	0 %
Iron, Fe	95.1 to 96.3 %	99.03 to 99.48 %
Manganese, Mn	0.6 to 0.8 %	0.3 to 0.6 %
Molybdenum, Mo	0.2 to 0.3 %	0%
Nickel, Ni	1.7 to 2.0 %	0 %
Phosphorus, P	0 to 0.035 %	0 to 0.040 %
Silicon, Si	0.15 to 0.35 %	0 %
Sulfur, S	0 to 0.040 %	0 to 0.050 %

2. Research and methods: Analysis of bridge's wings by analytical and FEA method

In this paper, the structure of the navigation deck wing and the support beam on the main deck were simplified and represented by beams with rectangular sections of 50 x 30 mm, with a wall thickness of 2 mm. The length of the wing depends on the ship's constructive characteristics and therefore this structure has been mounted on support members at a right angle. The rectangles used are plain carbon steel profiles with an empty interior.[3] The systems analyzed by both the finite element method and the analytical method were made up of horizontal beams with a fixed length and support beams joined together at an angle of 90°. The springs beams were mounted at 3 different distances from the front at the end of the horizontal beams: (00-90), (20-90) and (40-90).

2.1. The analytical method of navigation deck wings

The frequencies of the wing models were calculated from the analytical point of view. The material from which the beams are made was considered to be linear, homogeneous and isotropic. It has also been taken into account that beam profiles have very small dimensions.

It is considered a finite element of the beam type, of any section but constant in length dx . It is defined by the two ends where the six movement components (displacements and rotations) are introduced as degrees of freedom. The six degrees of freedom in each node correspond to the forces and moments F_x, F_y, F_z, M_x, M_y and M_z . In the local reference system xyz , the N, T_y, T_z, M_t, M_y and M_z elements are defined as positive when compliant with the meanings in figure 1. The finite beam element has contributions to the motion equation of the structure in the stiffness and mass matrix. The general wording of the element for calculating the stiffness matrix corresponds to the beam Timoshenko, but it can also be reduced to Euler-Bernoulli's formulation. [8]

The Cartesian reference system from figure 1 is composed by the x -axis, representing the normal to the beam cross-section and the y and z -axes that are perpendicular to the cross-section. [3]

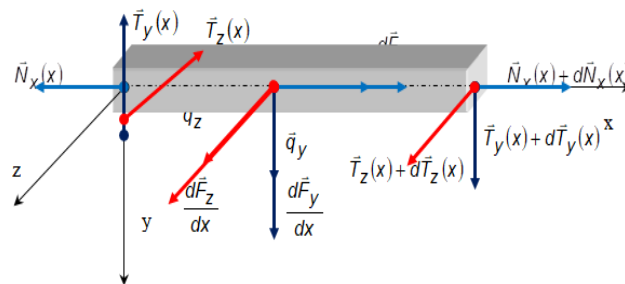


Figure 1. The elastic beams model.

A homogeneous straight section beam and an element of this beam are considered to be of a . The distributed load that deforms the beam is made up of distributed disturbing forces, q_x, q_y, q_z , and disturbing moments, m_x, m_y, m_z .

The equations of the free vibrations of a conservative system with N degrees of freedom can be written as:

$$[K]\{\ddot{q}\} + [K]\{q\} = \{0\} \quad (1)$$

The matrix of the masses $[M]$ and the matrix of the elastic constants $[K]$ are defined positively, non-singularly and usually symmetric, and $\{q\}$ is the column of the generalized coordinates.

$$([K] - \omega_i^2 [M]) \cdot \{\psi_i\} = \{0\} \quad (2)$$

The system has solutions only if:

$$\det([K] - \omega_i^2 [M]) = 0 \quad (3)$$

It only deals with linear systems with a finite number of degrees of freedom, because most methods of identifying elastic structures are based on discrete models of continuous real structures.[7]

For displacement, the boundary conditions for the beam fixed at $x=0$ and free at $x=l$ are:

$$\begin{aligned} u(0, t) &= 0 \\ N(l, t) &= EA \frac{\partial u}{\partial x} \Big|_{x=l} = 0 \end{aligned} \quad (4)$$

where $u(x, t)$ was the beam displacement relative to the equilibrium position, and EA is the stiffness modulus of the beam.

Results the dynamic equilibrium equation of the forces acting on the length element dx along the Oy axis:

$$\frac{dF_y}{dz} + q_y = 0 \quad (5)$$

and Oz axis:

$$T_z + dT_z - T_z + \frac{dF_z}{dx} + q_z = 0 \Rightarrow \frac{\partial T_z}{\partial x} dx + \frac{dF_z}{dx} + q_z = 0 \quad (6)$$

The dynamic equilibrium equations of disturbing moments distribute over the three axes is:

$$\frac{dM_x}{dx} + m_x = 0, \quad \frac{dM_y}{dx} + F_x + m_y = 0, \quad \frac{dM_z}{dx} - F_y + m_z = 0 \quad (7)$$

where $\frac{dM_x}{dx}$, $\frac{dM_y}{dx}$, $\frac{dM_z}{dx}$ are the moment of the inertial forces corresponding to the axes per unit length of the beam, and $\frac{dF_x}{dx}$, $\frac{dF_y}{dx}$, $\frac{dF_z}{dx}$ are forces of inertia per unit of length.

Based on the equations of forces and moments of inertia, the derived expressions of the displacements and rotations can be written [6]. Thus, for 3 translations and 3 rotations, the first 6 differential equations are obtained:

$$\frac{du}{dx} - \frac{F_x}{EA} = 0 \quad (8)$$

$$\frac{dv}{dx} + \Omega_z - \frac{F_y}{GA/k_y} = 0 \quad (9)$$

$$\frac{du}{dx} - \Omega_y - \frac{F_z}{GA/k_z} = 0 \quad (10)$$

$$\frac{d\Omega_x}{dx} = \frac{M_x}{GI_p} \quad (11)$$

$$\frac{d\Omega_y}{dx} = \frac{M_y}{EI_y} \quad (12)$$

$$\frac{d\Omega_z}{dx} = \frac{M_z}{EI_z} \quad (13)$$

I_p - the inertia polar moment on the length unit along Ox axis; I_y, I_z - the axial inertia moments corresponding to the Oy and Oz axes; $k_x = \frac{EA}{l}$ elastic twist constant; $k_y = k_z = \frac{GA}{l}$ elastic bending constants.

Assuming the vibrations are harmonic, then the relations of displacements, rotations, inertia forces and moments depending on time can be written as Fourier series:

$$u(x, t) = u(x) \cdot e^{i\omega t} = u(x) (\cos \omega t + i \sin \omega t) \quad (14)$$

$$v(x, t) = v(x) \cdot e^{i\omega t} = v(x) \cdot (\cos \omega t + i \sin \omega t) \quad (15)$$

$$w(x, t) = w(x) \cdot e^{i\omega t} = w(x) \cdot (\cos \omega t + i \sin \omega t) \quad (16)$$

$$\Omega_x(x, t) = \Omega_x(x) \cdot e^{i\omega t} = \Omega_x(x) \cdot (\cos \omega t + i \sin \omega t) \quad (17)$$

$$\Omega_y(x, t) = \Omega_y(x) \cdot e^{i\omega t} = \Omega_y(x) \cdot (\cos \omega t + i \sin \omega t) \quad (18)$$

$$\Omega_z(x, t) = \Omega_z(x) \cdot e^{i\omega t} = \Omega_z(x) \cdot (\cos \omega t + i \sin \omega t) \quad (19)$$

$$F_x(x, t) = F_x(x) \cdot e^{i\omega t} = F_x(x) \cdot (\cos \omega t + i \sin \omega t) \quad (20)$$

$$F_y(x, t) = F_y(x) \cdot e^{i\omega t} = F_y(x) \cdot (\cos \omega t + i \sin \omega t) \quad (21)$$

$$F_z(x, t) = F_z(x) \cdot e^{i\omega t} = F_z(x) \cdot (\cos \omega t + i \sin \omega t) \quad (22)$$

$$M_x(x, t) = M_x(x) \cdot e^{i\omega t} = M_x(x) \cdot (\cos \omega t + i \sin \omega t) \quad (23)$$

$$M_y(x, t) = M_y(x) \cdot e^{i\omega t} = M_y(x) \cdot (\cos \omega t + i \sin \omega t) \quad (24)$$

$$M_z(x, t) = M_z(x) \cdot e^{i\omega t} = M_z(x) \cdot (\cos \omega t + i \sin \omega t) \quad (25)$$

$$\frac{d}{dz} \begin{bmatrix} u \\ v \\ w \\ \Omega_x \\ \Omega_y \\ \Omega_z \\ F_x \\ F_y \\ F_z \\ M_x \\ M_y \\ M_z \end{bmatrix} = \begin{bmatrix} \vec{r}_{ii} & \vec{r}_{ij} \\ \vec{r}_{ji} & \vec{r}_{jj} \end{bmatrix} \begin{bmatrix} u \\ v \\ w \\ \Omega_x \\ \Omega_y \\ \Omega_z \\ F_x \\ F_y \\ F_z \\ M_x \\ M_y \\ M_z \end{bmatrix} \quad (26)$$

By using the differential equations of the rotations and those of the disturbing forces and disturbing moments, in the above 12 equations the following matrix is obtained [6]. This set of equations is written as a vector (equation 23).

Thus the matrix becomes:

$$\frac{d}{dz} \begin{bmatrix} u \\ v \\ w \\ \Omega_x \\ \Omega_y \\ \Omega_z \\ F_x \\ F_y \\ F_z \\ M_x \\ M_y \\ M_z \end{bmatrix} = \begin{bmatrix} 0 & 0 & 0 & 0 & 1 & 0 & \frac{k_x}{GA} & 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & -1 & 0 & 0 & 0 & \frac{k_y}{GA} & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & \frac{1}{EA} & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & \frac{1}{EI_x} & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & \frac{1}{EI_y} & 0 \\ 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & \frac{1}{GI_b} \\ \mu\omega^2 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 \\ 0 & \mu\omega^2 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & \mu\omega^2 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & \mu \frac{I_x}{A} \omega^2 & 0 & 0 & 0 & 1 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & \mu \frac{I_y}{A} \omega^2 & 0 & -1 & 0 & 0 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0 & \mu \frac{I_z}{A} \omega^2 & 0 & 0 & 0 & 0 & 0 & 0 \end{bmatrix} \begin{bmatrix} u \\ v \\ w \\ \Omega_x \\ \Omega_y \\ \Omega_z \\ F_x \\ F_y \\ F_z \\ M_x \\ M_y \\ M_z \end{bmatrix} \quad (27)$$

The coefficient matrix of system [A] can be defined as follows:

$$\frac{d\vec{y}(x)}{dx} = [A] \cdot \vec{y}(x) \quad (28)$$

This equation $\vec{y}(x_0) = \vec{y}_0$ is solved by knowing the initial values of the vectors:

$$\vec{y}(x) = e^{[A]x} \cdot \vec{y}_0 \quad (29)$$

By solving the equation (26) we obtain:

$$[f(\omega)]_{12 \times 12} \cdot [y_0(x)]_{12 \times 1} = [0]_{12 \times 1} \quad (30)$$

For a result different from 0, the determinant must be zero, $\det[f(\omega)] = 0$. By equating the determinant with 0 we obtain an equation with unknowns. Solving this equation gives us the values of our own frequencies. Wing patterns to be studied from the analytical point of view are made up of 3 segments. We consider the main beam made up of 2 segments (AC and CD) and one segment (BC) being the support beam.

In figure 2 we can see the analytical model of the wing. In this system the ends A and B are considered fixed, the end D is free, and the point C is the connecting area between the two beams.

There are 3 translations and 3 rotations in points A and B. So, there are 6 equations in each point, and a total of 12 equations is obtained. Since the end D is free and the external loads do not influence the cross-section, 6 equations (3 for force and 3 for the moment) will be written at this point. Point C has 18 continuity equations. Thus we obtain a total of 36 equations, which form the primary matrix of the order of 36.

This matrix was then solved using the MATLAB R2011b program for all winged models proposed for study.

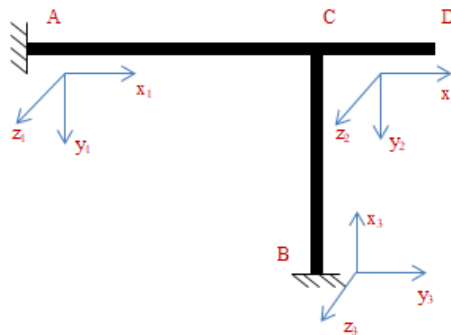


Figure 2. The analytical model of the wing.

The determination of its own frequencies is limited to the determination of the quadratic matrix's own values. To solve this matrix the eig (eigenvalue) function is used, with 2 output arguments, thus obtaining its own vectors and its own values. The results obtained are presented in the following table. (Sees table 3)

Table 3. Own frequencies of wing models calculated using the AM.

Mod	1	2	3	4	5
(00-90)	65.45	143.75	221.29	275.76	406.52
(20-90)	72.73	148.71	213.74	363.22	475.84
(40-90)	79.63	136.02	188.29	195.03	452.35

2.2. FEM analysis of deck wings

Models previously analyzed by the analytical method (AM) are now studied by the Finite Element Analysis (FEM) method, thus obtaining modal vibrations and their own frequencies.

The patterns were originally created in Nastran NX v.8 under the .prt extension, then exported under the .stp extension in FEMAP v.10 where they were defined: the structure material, its properties, the constraints in points A and B, then it was performed model meshing, and modal analysis was run.

Following the analysis, the model (00-90) has 11484 elements and 23070 nodes, the model (20-90) has 11443 elements and 22950 nodes, and the model (40-90) has 11407 elements and 22882 nodes. (See figure 3)

Finally, the results obtained by the finite element analysis method were compared with the results obtained by the analytical method.



Figure 3. Models: (00-90), (20-90) and (40-90) in Nx v.8.

3. Results

In the following pictures only the first 5 modes of vibration will be presented, and in the centralizing table the frequencies corresponding to the studied models, but made of SAE-AISI 4340 Ni-Cr-Mo Steel and SAE-AISI 1025 Carbon Steel will be passed.

3.1. FEA analysis results of (00-90) model

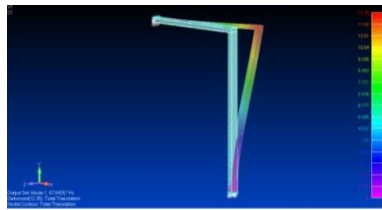


Figure 4. Mode 1 of vibration.

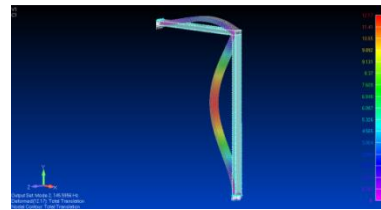


Figure 5. Mode 2 of vibration.

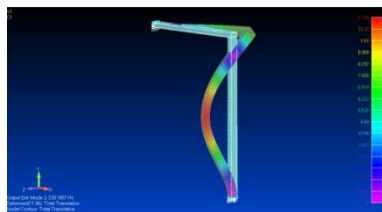


Figure 6. Mode 3 of vibration.

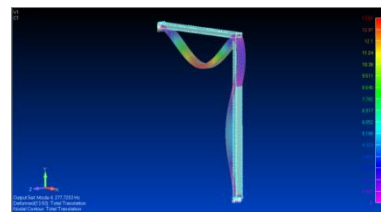


Figure 7. Mode 4 of vibration.

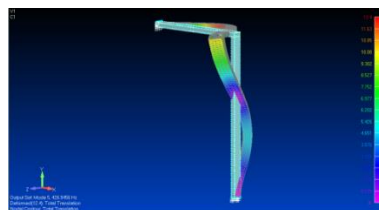


Figure 8. Mode 5 of vibration.

3.2. FEA analysis results of (20-90) model

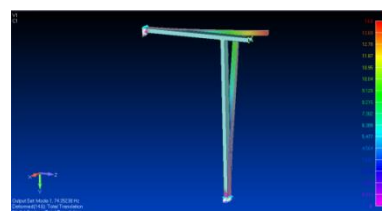


Figure 9. Mode 1 of vibration.

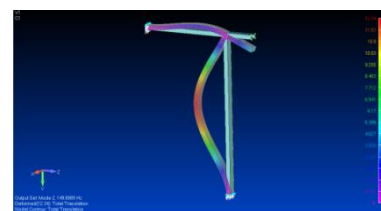


Figure 10. Mode 2 of vibration.



Figure 11. Mode 3 of vibration.

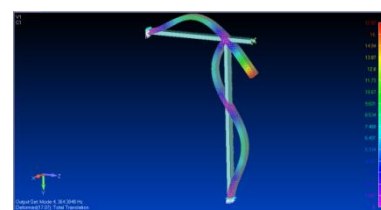


Figure 12. Mode 4 of vibration.



Figure 13. Mode 5 of vibration.

3.3. FEA analysis results of (40-90) model



Figure 14. Mode 1 of vibration.

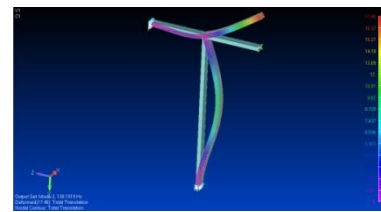


Figure 15. Mode 2 of vibration.

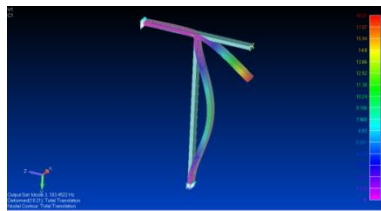


Figure 16. Mode 3 of vibration.

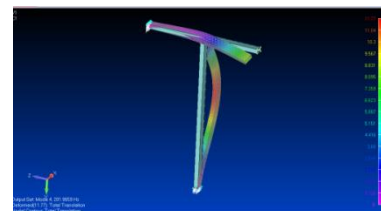


Figure 17. Mode 4 of vibration.



Figure 18. Mode 5 of vibration.

Due to the structural similarities, the model (40-90) has a dynamic behavior similar to the models (090) and (20-90), namely: at high frequencies, at the main beam, and at the support beam simultaneously occur bending stresses, torsion and stretching. At some frequencies, it can be observed that the vibrations do not act strongly on the support beam, and in another modeling, the support beam exhibits only axial behavior, the free end of the main beam being deeply affected by the vibrations.

3.4. Comparing the results obtained through the MEF and the analytical method

Table 4. Results.

Mod	1	2	3	4	5
	(00 – 90)				
FEM 1 [Hz] AISI 4340	67.96	145.85	230.57	278.21	427.70
FEM 2 [Hz] AISI 1025	67.84	145.59	230.17	277.72	426.94
AM [Hz]	65.45	143.75	221.29	275.76	406.52

FEM 1– AM [%]	3.693	1.439	4.025	0.881	4.952
FEM 2- AM [%]	3.523	1.264	3.858	0.706	4.783
FEM 1 – FEM 2 [%]	0.176	0.178	0.173	0.176	0.178
(20 – 90)					
FEM 1[Hz] AISI 4340	74.48	150.15	222.41	365.03	487.35
FEM 2 [Hz] AISI 1025	74.35	149.89	222.01	364.38	486.49
AM [Hz]	72.73	148.71	213.74	363.22	475.84
FEM 1– AM [%]	2.350	0.959	3.898	0.496	2.361
FEM 2- AM [%]	2.179	0.787	3.725	0.318	2.189
FEM 1 – FEM 2 [%]	0.175	0.173	0.180	0.178	0.176
(40 – 90)					
FEM 1[Hz] AISI 4340	80.84	138.36	183.71	202.33	429.13
FEM 2 [Hz] AISI 1025	80.70	138.10	183.45	201.97	428.41
AM [Hz]	79.63	136.02	188.29	195.03	452.35
FEM 1– AM [%]	1.497	1.691	-2.493	3.608	-5.411
FEM 2- AM [%]	1.326	1.506	-2.638	3.436	-5.588
FEM 1 – FEM 2 [%]	0.173	0.188	0.141	0.178	0.168

4. Conclusions

Generally, it can be observed that the values of the own frequencies obtained by the analytical method are lower than the values obtained by the finite element method. The reason for obtaining higher values using the finite element method is that no damping is used for this method. Due to lack of damping, the values of the own frequencies obtained by the finite element method are superior to the other methods.

For the model (00-90) made of SAE-AISI 4340 Ni-Cr-Mo Steel, the frequencies obtained by FEM are 3% higher than the AM and when the model is made of SAE-AISI 1025 Carbon Steel the difference is 2.8%.

For the model (20-90) made of SAE-AISI 4340 Ni-Cr-Mo Steel, the FEM's own frequencies are about 2% higher than AM, and for SAE-AISI 1025 Carbon Steel the difference is 1.8%.

And the (40-90) model made of SAE-AISI 4340 Ni-Cr-Mo Steel, the FEM's own frequencies are about 2% higher than AM for the first variation modes. For the other modes of variation, the values of own frequencies obtained by AM are 4% higher than FEM. The situation is similar to SAE-AISI 1025 Carbon Steel. (See table 4)

It can also be observed that the materials are chosen for the study behave similarly. Values obtained vary very little.

It can be concluded that the SAE-AISI 1025 Carbon Steel proves to be more appropriate to the construction of the beams because it has lower resonance frequencies.

5. References

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