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Efficient strategy to increase natural frequencies in pods of the new Red Sea Airport

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Abstract

In March 2024, Lanik completed the construction of the spatial structure for the new Red Sea Airport terminal in Hanak, located on the north western coast of Saudi Arabia. Designed by Foster and Partners, the project comprises five interconnected buildings (pods) that follow an architectural freeform surface. The pods are linked by ornamental louvers. Each structure features double-curved vaults, supported by four points, primarily working as compression shells. The span between supports reaches 64m at its widest point, with a 40m cantilever. The geometry is defined with the external cladding and the ceiling, resulting in a space frame with a thickness of just 2m, making it a slender spatial mesh structure. It is found that there are no global buckling problems, however, the entire perimeter supports a high-mass GRC cladding bullnose. Coupled with the low bending stiffness of the structure, it leads to low-frequency fundamental modes. Wind tunnel studies indicate a minimum natural frequency of above 0.6Hz. Relying solely on strength-based optimization of the structure resulted in first vibration frequencies well below the target, so additional steps were required to improve the stiffness. This paper shows the different strategies followed to stiffen the structure, and which has proven to be most efficient (greatest increase in stiffness of the first vibration modes with the smallest possible increase in material).

Keywords: Red Sea Airport, vibration modes, space frame shell

1. Introduction

In 2022 Lanik was commissioned to engineer and construct the space frame structure for the terminal building of the new Red Sea Airport in Saudi Arabia.

Figure 1: General view of the terminal building of the new Red Sea Airport

The airport is designed by Foster + Partners, with AECOM serving as the project's supervising consultant and JACOBS as the lead consultant.

The main terminal building is divided into five "Pods" connected by four pergolas, ensuring visual continuity and eliminating the need for expansion joints. Although designed to minimize the load transfer between pods, the pergolas do influence the overall stiffness matrix and must be considered in the behavior of the structures. The resulting size of the complete structural system is 298m by 170m and should be modelled in a single model.

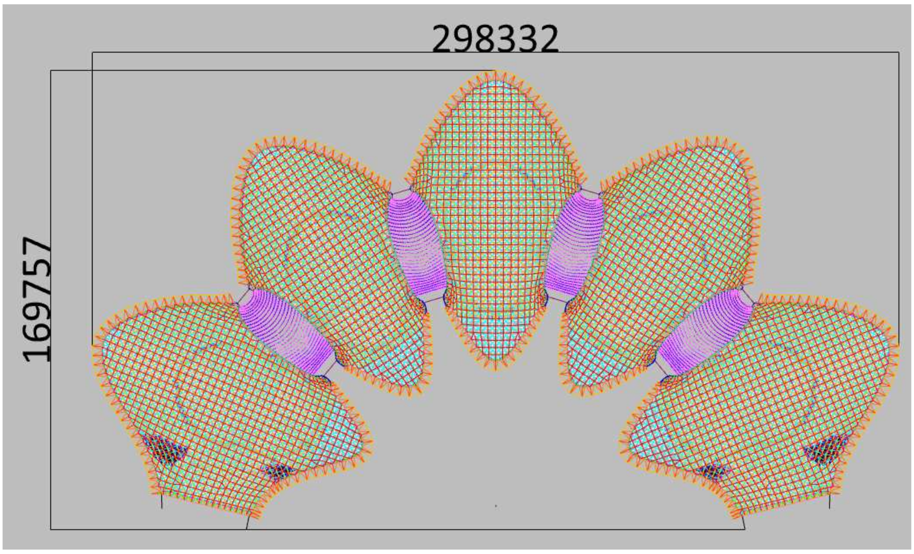

Figure 2: Plan view of the terminal building structure

As of march 2024, the erection of the space frame structure is completed. Substructure and cladding are still being installed.

Figure 3: Space frame structure assembly completed

Lanik's scope in the project includes the full engineering of the space frame structure. In addition to ensuring resistance, stability and stiffens, the assessment of the dynamic behavior to prevent resonance caused by alternating wind loads must be also verified.

The structure was optimized to achieve a significant reduction of the required amount of steel for the space frame compared to the original project. However, the dynamic behavior was not initially considered in early stages of the engineering. Upon completing the dynamic analysis, the low fundamental frequencies and the potential resonance problems with alternating wind loads were detected.

This paper shows the process followed to find a way to increase the natural frequencies of the first vibration modes with the minimum increase of structural weight.

2. Description of the dynamic behavior

Each of the five pods features a big cantilever of 44m in the front. Each cantilever has a height of 18m and works structurally as a shell. The thickness of the space frame is architecturally constrained to a maximum of 2m. With a span between support of 64m, the resulting width/span ratio is approximately 1/32. This ratio indicates a very slender space frame design in contrast to traditional pure bending space frames, which typically have a width/span ratio of about 1/15.

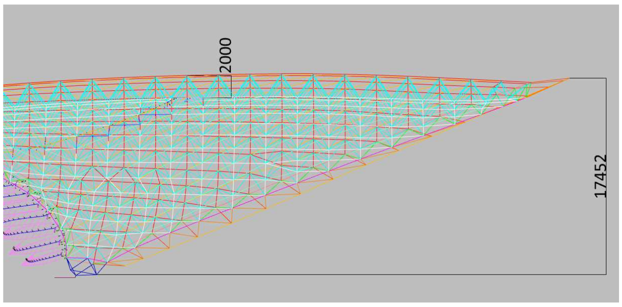

Figure 4: height and width of the cantilever

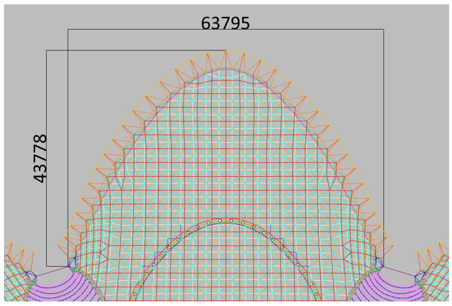

Figure 5: length and span of the cantilever

In addition to the small bending stiffness, the perimeter of the structure supports a high-mass GRC cladded bullnose as shown in the detail below.

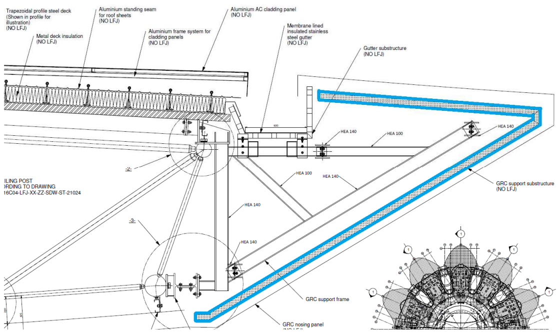

Figure 6: GRC cladding along the Edge of the cantilever

Combining the small bending stiffness of the shell and the significant suspended mass results in a predictably low natural vibration mode frequencies. This hypothesis was confirmed through dynamic calculation, revealing a natural frequency below 0.55Hz for the first vibration mode .

During the preliminary engineering phase, a CFD wind analysis was conducted showing the possibility of vortex shedding near a frequency of 0.6Hz along the perimeter of the cantilevers. Lanik was asked to increase the first natural vibration frequency to a value above 0.7 Hz.

2.1. Description of the first vibration modes

Table 1 shows the activated masses and frequencies of the first 5 vibration modes for a configuration that satisfies the requirement of minimum vibration frequency (f>0.7Hz).

Table 1: activated masses of the first 5 modes

Masa nodales - masa activada

No.	LC	frequency [Hertz]	effective modal Mass X[t]	Y[t]	Z[t]	factor X[%]	Y[%]	Z[%]	activated mass [%]*
1	600	0.703	131.7	11.4	1.6	14.98	1.30	0.18	8.74867
2	601	0.722	22.0	355.0	55.9	2.50	40.39	6.36	13.71311
3	602	0.799	443.9	5.2	1.1	50.50	0.59	0.12	12.86294
4	603	1.546	0.0	82.3	33.4	0.00	9.36	3.80	6.90289
5	604	1.696	63.4	0.1	0.8	7.21	0.01	0.09	4.53620

Only the first three vibration modes have frequencies below 1Hz. The 4th vibration mode well above with a natural frequency of 1.55 Hz. Therefore, only the first three vibration modes are critical. These three modes are discussed.

2.1.1. Mode A. Front cantilever waving

The activated mass is relatively small (only 15% in horizontal direction). It represents a waving mode rather than a translational mode. The shape of this is shown in the following figure.

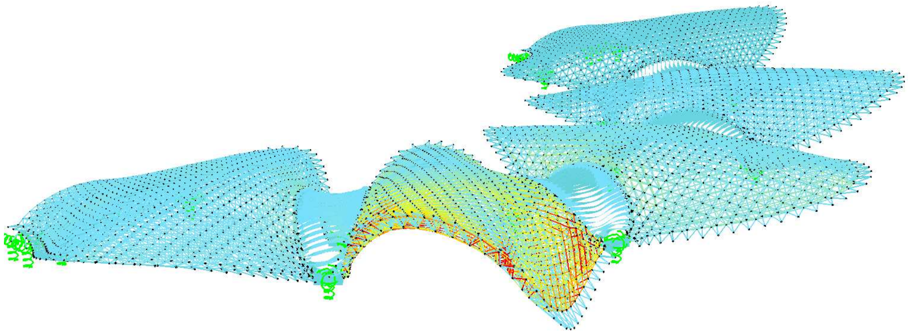

Figure 7: Vibration Mode A

2.1.2. Mode B. front-back movement

The activated mass is significant (40% in horizontal direction) due to its big translational component. This mode has the shape shown in the following figure.

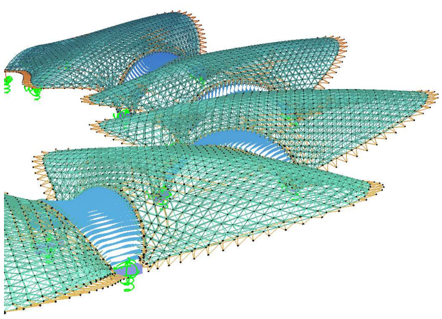

Figure 8: structural model undeformed

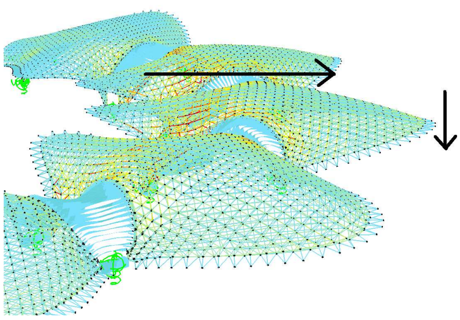

Figure 9: Vibration mode B

2.1.3. Mode C. lateral movement

The activated mass is significant (50% in horizontal direction) due to its big translational component. The shape of this mode is shown in the following figure.

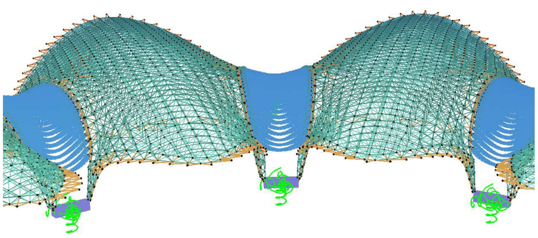

Figure 10: structural model undeformed

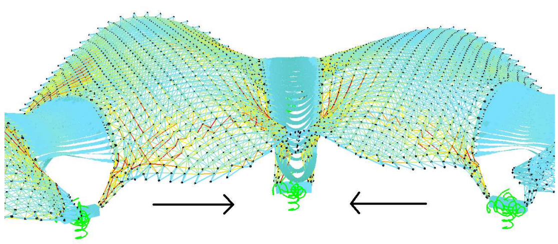

Figure 11: Vibration mode C

The obtained natural frequencies according to the first calculation models (C01A) were much lower than 0.7Hz. As explained before, driving factors for the first dimensioning attended to resistance, stability, and deformability.

Table 2: natural frequencies model C01A

	frequency [Hz]
MODE A	**0.544**
MODE B	**0.577**
MODE C	**0.664**

The total weight of the version C01A is 171743 kg. The goal is to achieve 0.7 Hz of minimum natural frequency with the minimum increase of total weight.

3. Sequence of variations

In the following points it is described the sequence of modifications done to the structure and the effect to the natural frequencies and total weight of each of them.

3.1. Model C01B

In model C01B, diagonals in the top layer were added to increase the torsional stiffness of the cantilever. It is automatically optimized with strength requirements only, without any additional manual reinforcement.

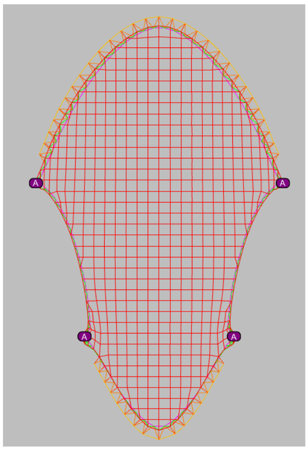

Figure 12: Model C01A top layer

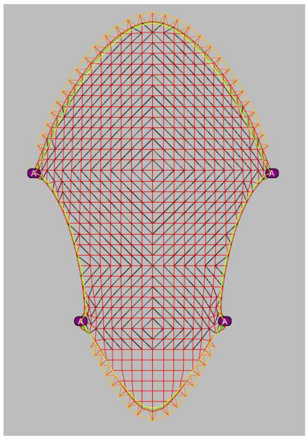

Figure 13: Model C01B top layer

The total weight of the version C01B is 180247 kg. The natural frequencies obtained with this modification are shown in the following table.

Table 3: natural frequencies models C01B

	frequency [Hz]
MODE A	**0.543**
MODE B	**0.621**
MODE C	**0.681**

There is almost no increase of natural frequency of mode A.

3.2. Model C01C

In model C01C, the diagonals were also added to the bottom layer to increase the torsional stiffness of the cantilever. It is automatically optimized with strength requirements only, without any additional manual reinforcement.

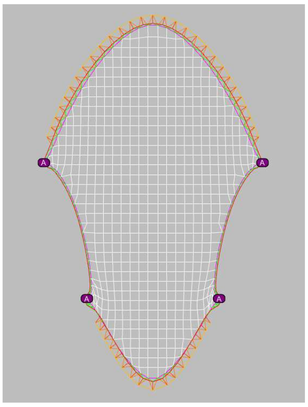

Figure 13: Model C01B bottom layer

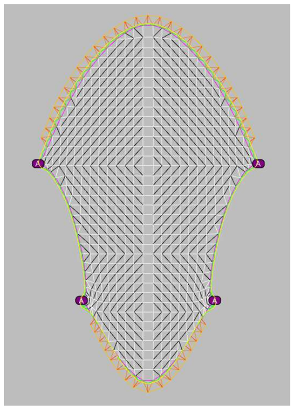

Figure 14: Model C01C bottom

The total weight of the version C01C is 178401 kg. The natural frequencies obtained with this modification are shown in the following table:

Table 4: natural frequencies model C01C

	frequency [Hz]
MODE B	**0.653**
MODE A	**0.707**
MODE C	**0.78**

In this case, the natural frequency corresponding to mode A is increased from 0.54 Hz to 0.71 Hz. This means that the diagonalization of both top and bottom layers increases a lot the torsional stiffness of the space frame structure. In this stage, Mode B becomes the first vibration mode, and the next variations are focused on increasing the stiffness of mode B.

3.3. Models C01D, C01E and C01F

In models C01D, C01E and C01F the cross-section of the elements were manually increased. The number of manual changes done to the sections is extensive. In this paper it is shown just an example to understand the kind of changes done. As an example: in model C01E, the elements in the arches between supports were stiffened. In this way, D76/2 sections were manually changed to D100/4; D114/5 sections were changed to D159/6; and so on.

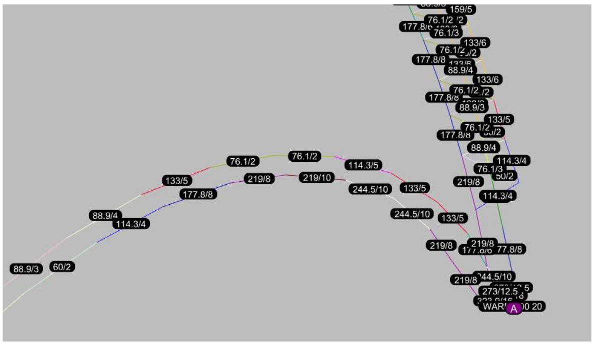

Figure 15: arch member sections in model C01C

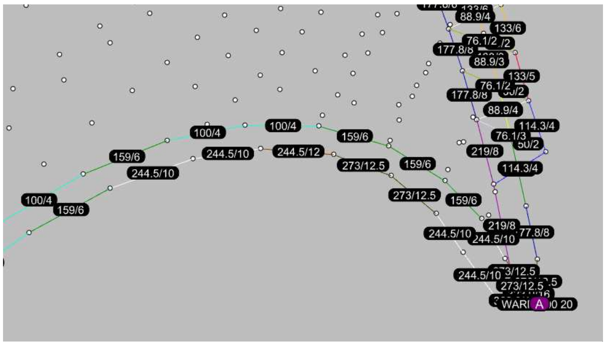

Figure 16: arch member sections in model C01E

The results of these arbitrary changes, as presented in the following table, indicate a positive increase of the natural vibration frequency of mode B. However, the increase of total structural steel is significant. The natural frequency is proportional to the square root of the stiffness divided by the mass. While, an increase of stiffness from section increments has a positive effect, the simultaneous increase in mass may render this manual "blind" exercise inefficient. Large amounts of added steel may not necessarily contribute to an increase mode B frequency but rather lead to an increase of mass.

Table 5: natural frequencies model C01D, C01E and C01F

	C01D	C01E	C01F
	frequency [Hz]	frequency [Hz]	frequency [Hz]
MODE B	**0.662**	**0.672**	**0.696**
MODE A	**0.754**	**0.709**	**0.71**
MODE C	**0.792**	**0.813**	**0.851**
Total Weight [kg]	193098	190740	195174

Option C01F satisfies the requirement of minimum natural frequency (0.697 Hz) but with a mass of 195tn. There appears to be a need for a more efficient approach to achieve an increase of the stiffness of mode B without increasing the weight so much.

3.4. Model C01G

The shape of vibration mode B is very similar to the deformation plot resulting for wind load suction Hypothesis H6 (figure 9 shows the opposite sense of the vibration mode).

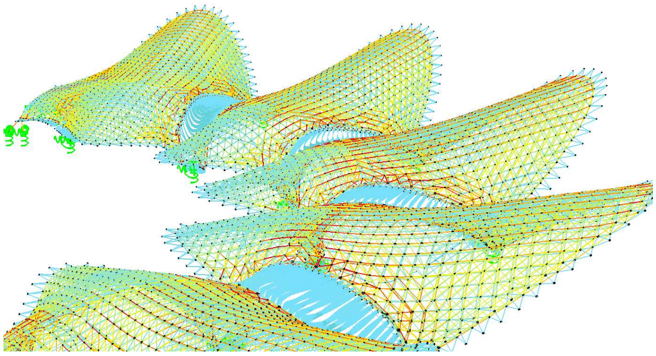

Figure 17: Deformation according to wind suction hypothesis H6

Given the similarities in the displacements, it is expected that the distribution of efforts across elements of the structure will closely resemble one another. The proposed strategy entails automatically optimizing the structure's response to the factored load case H6. This optimized reinforcement should primarily target the elements influencing the behavior of Mode B.

In an automatic dimensioning loop within GOOSE [1], both the element cross-section and connection bolts are typically increased to the minimum size to withstand the new loads. However, for this exercise, a more efficient approach is to only increase the cross-sections, as the bolts do not contribute to the increase of stiffness. Therefore, instead of applying a significant multiplication factor to load case H6 to increase the efforts in the elements, a modified approach is used. Using model C01C as the basis, load case H6 is rerun with increased material factors: the limit of the steel in this calculation is reduced to 50% of its original value. As a result, only the most heavily stressed elements are increased in size, without affecting the bolts size.

Table 6: natural frequencies model C01G

	frequency [Hz]
MODE B	**0.71**
MODE A	**0.714**
MODE C	**0.825**

Option C01G satisfies the requirement of minimum natural frequency (0.71 Hz) with a total weight of 187337 kg. This represents a clear improvement compared to model C01F, which achieves a lower frequency with an additional 8tn of material. Thus, the hypothesis regarding the advantage of using load case H6 as the dimensioning load appears to be validated.

3.5. Model C01H

While model C01G satisfies the requirement of minimum modal frequency, it is based in model C01C. This model feature the complete top and bottom layer with diagonals, which increases the total weight. It is expected that not all the diagonals have significant impact in the increase of the torsional stiffness.

For this reason, in model C01H, all the added top and bottom layer diagonals with the smallest connection bolts (M16) are removed.

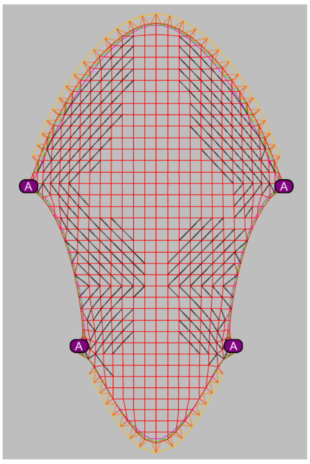

Figure 18: Remaining upper layer diagonals in C01H

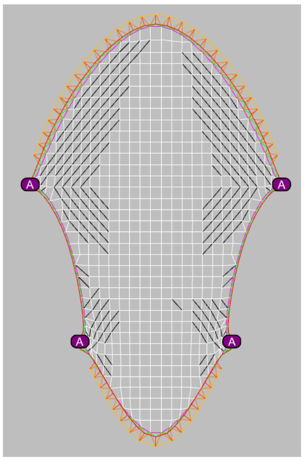

Figure 19: Remaining lower layer diagonals in C01H

The automatic optimization loop is rerun one additional time using the same criteria discussed in point 3.4 to the modified model C01G. Table 7 shows the obtained frequencies for model C01H.

Table 7: natural frequencies model C01H

	frequency [Hz]
MODE A	**0.668**
MODE B	**0.709**
MODE C	**0.779**

The weight decreases to just 177089 kg, but the natural frequency of mode A becomes again lower than 0.7Hz so this option C01H does not fit the requirement.

3.6. Model C01I

Model C01I follows the same approach discussed in section 3.5. However, in this iteration, only the elements of the top a bottom layer diagonal with an M16 bolt and a section equal or lower than D76 are removed.

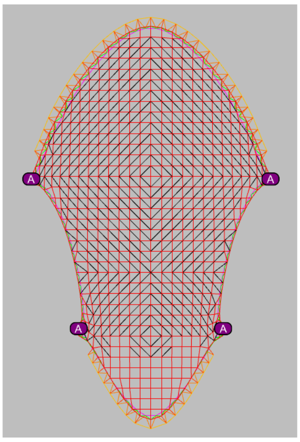

Figure 20: Remaining upper layer diagonals in C01I

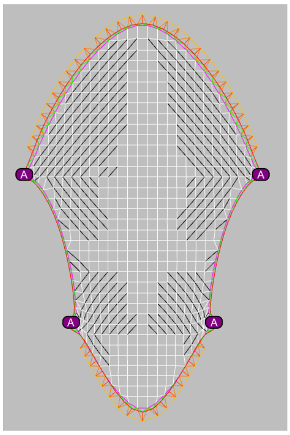

Figure 21: Remaining lower layer diagonals in C01I

The automatic optimization loop is rerun one additional time using the same criteria discussed in point 3.4. Table 8 shows the obtained frequencies for model C01I.

Table 8: natural frequencies model C01I

	frequency [Hz]
MODE A	**0.698**
MODE B	**0.71**
MODE C	**0.79**

In this case the total weight of model C01I decreases to 183519 kg in comparison with the 187337 kg of model C01G. The reduction is not so big as the obtained in model C01H, but in this case the requirement is satisfied as 0.698 Hz can be rounded to 0.7 Hz.

4. Result summary

Table 9 shows the summary of results of the different models analyzed.

Table 9: result summary

	C01A	**C01B**	**C01C**	**C01D**	**C01E**	**C01F**	**C01G**	**C01H**	**C01I**
f mode A [Hz]	0.544	0.543	0.707	0.754	0.709	0.71	0.71	0.668	**0.698**
f mode B [Hz]	0.577	0.621	0.653	0.662	0.672	0.696	0.714	0.709	**0.71**
f mode C [Hz]	0.664	0.681	0.78	0.792	0.813	0.851	0.825	0.779	**0.79**
Weight [kg]	171744	180247	178402	193098	190740	195174	187337	177089	**183520**
f min [Hz]	0.544	0.543	0.653	0.662	0.672	0.696	0.71	0.668	**0.698**
f min/Weight [% of C01A]	100%	95%	116%	108%	111%	113%	120%	119%	**120%**
Weight inc. [% of C01A]	100%	105%	104%	112%	111%	114%	109%	103%	**107%**
f inc. [% of C01A]	100%	100%	120%	122%	124%	128%	131%	123%	**128%**

The information of this table can be interpretated more clearly by means of a graphical representation.

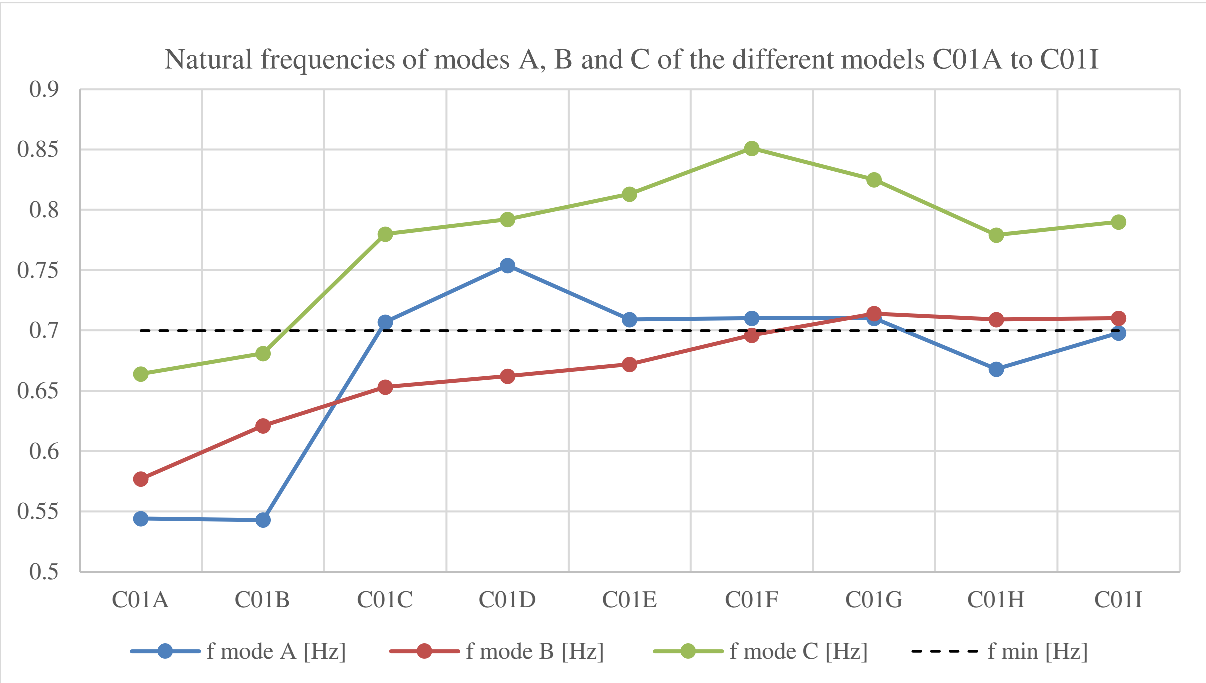

Figure 22: Natural frequencies of modes A, B and C of the different models C01A to C01I

Model C01C with upper- and lower-layer diagonals improves significantly mode A but it is not enough to improve mode B up to 0.7Hz. Model C01F achieves this goal but with a poor efficiency of usage of material. The strategy of model C01G improves the efficiency to a ratio of 120%, but model C01I achieves also the required limit with a total weight of 183tn instead of 187tn of model C01G.

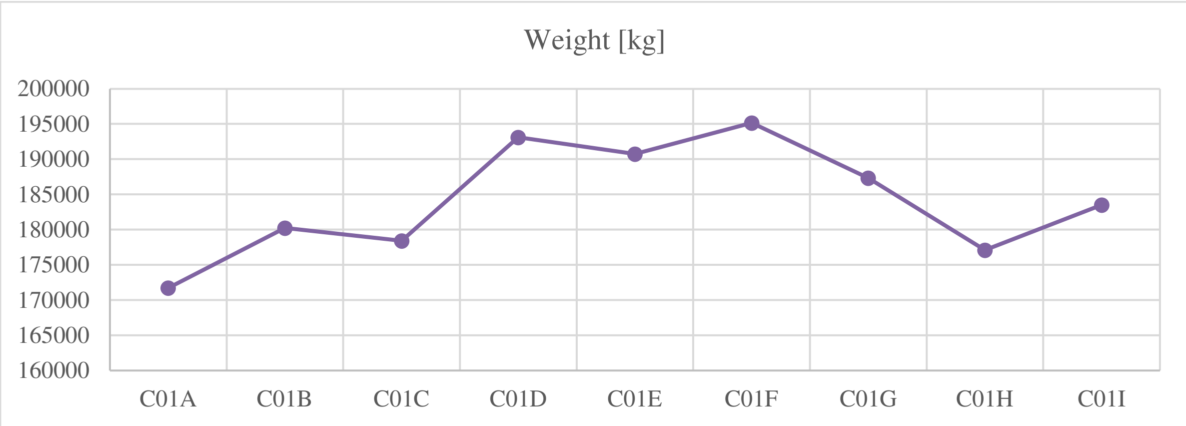

Figure 23: Weigth of the different models C01A to C01I

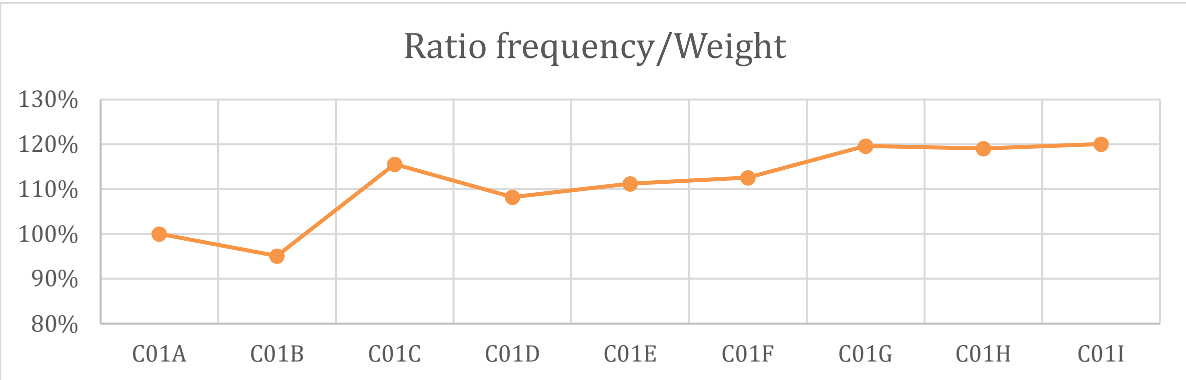

Figure 24: Ratio frequency/weight of the different models C01A to C01I

5. Conclusion

The final solution implemented to solve the problem of the minimum modal frequencies corresponds to model C01I. The design optimization results in an increase of 28% of the smallest natural frequency with a mere 7% increase of total weight.

References

[1] I. Montoya, A. Souto, B Ochoa, R Virto and J. Goñi , *GOOSE: Integrated non-commercial software for shell and spatial structures design, calculation, fabrication, and assembly engineering*, Melbourne, 2023.