PROJECT 1

Elastomeric mat to mitigate railway vibrations

Index:

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Elastomeric mat to mitigate railway vibrations

1. Introduction

Railway transportation is seen as an environmental friendly mode of transport. However, vibration and noise problems associated with the rolling of the wheel on the rail are concerning management authorities. The expansion of the rail transport and the increasing speed of the trains are aggravating vibrations problems, which require the implementation of mitigating measures.

Figure 1. 1 Railway transportation

The vibration propagates through the structure affecting the levels of user comfort. Historically, ballast mats have been used to reduce vibrations. They isolate the rail support from the surrounding structure providing effective attenuation of the vibrations. However, nowadays elastic materials are gradually applied in the construction of railway to reduce the track vibration. The degree of isolation achieved is far superior to almost all other available systems.

The main property to use elastomeric material to reduce the vibration is its resilience. Resilience is the ratio of energy released in deformation recovery to the energy that caused the deformation. They are used as damping material to attenuate disturbance in urban buildings located near the railway lines. The vibration can be significantly lowered by reducing the transmission from structure to the residential buildings. The elastomeric material would absorb the impact energy and minimize the impact over the surrounding buildings, what would reduce the nuisance for the residents.

Figure 1. 2 Elastomeric devices

2. Problem statement

The vibration problems are becoming more and more serious in the railway field. The solution to vibration isolation and damping is usually a combination of materials and design adjustments. The material must provide excellent resistance to compression and exhibit good vibration damping and impact absorption. The form of the device would also have relevance on the transmitted vibration. Therefore, the appropriate design should maximize the efficiency of the used material and decrease the vibrations.

- The analytical stiffness has been checked with the results obtained from the modelling.
- The efficiency of three different designs has been studied.

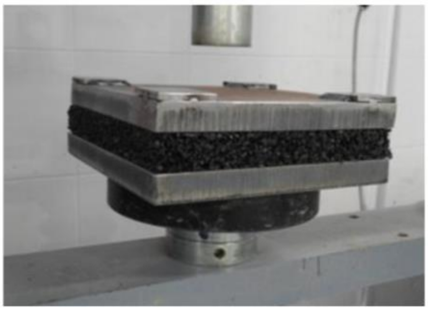
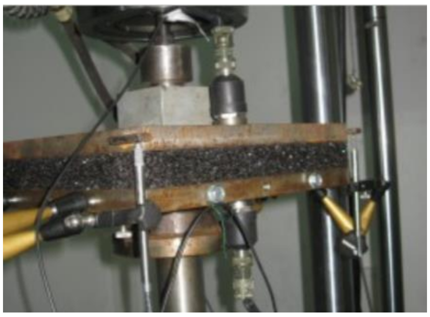



Figure 2. 1 Laboratory testing

3. Methodology

The inputs need to define the model are the material properties, geometry and loads. The properties are specified at the statement of the problem and they are introduced in the model without problems. However, the geometry and loads needs to be adjusted and completed to feed the FEM.

3.1. Geometry of the elastomeric material

Three different designs have been studied with the aim of optimizing material efficiency. The same volume of material is used in the three cases to simplify the comparison of the results and to define the thickness for the design A and B. The purposed designs and their dimensions are showed in the following figures:

- **<u>Original design:</u>** All the dimensions of the sheet are defined in the statement. These dimensions will define the reference volume:

$$V = 0.025 \cdot 0.3^2 = 0.00225 \text{ m}^3$$

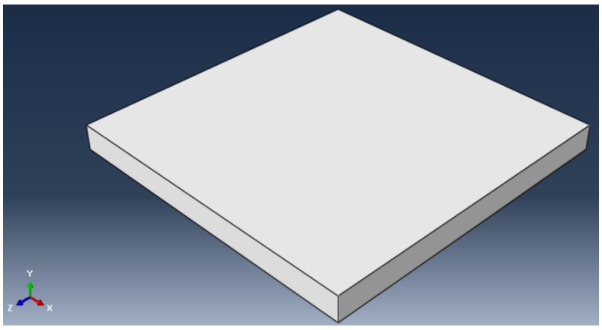


Figure 3. 1 Solid slab

Variable	L	H
Value (mm)	300	25

Table 3. 1 Solid slab dimensions

- **<u>Design A:</u>** The variable H_2 is defined imposing that the volume of the elastomeric material should be the same as the original design:

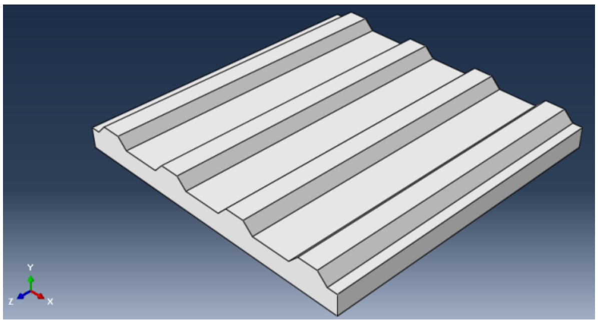


Figure 3. 2 Irregular slab

$$V = 0.3 \cdot [0.3 \cdot H_2 + 4(0.01 \cdot 0.03)] = 0.025 \cdot 0.3^2 \Rightarrow H_2 = 0.021 \text{ m}$$

Variable	L	H_2	H_1	a	b	c
Value (mm)	300	21	10	10	20	10

Table 3. 2 Irregular slab dimensions

- **Design B:** The variable H_2 is defined imposing that the volume of the elastomeric material should be the same as the Original design :

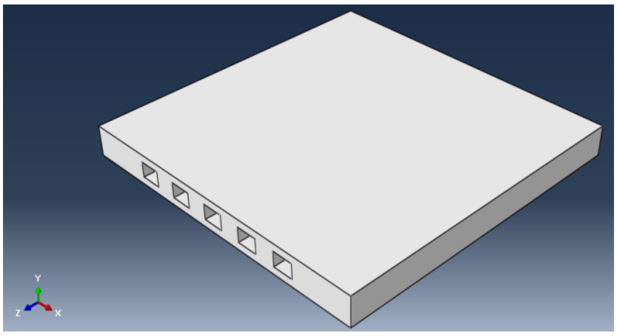


Figure 3. 3 Lightweight slabs

$$V = 0.3 \cdot [0.3 \cdot H_2 + 5(0.02 \cdot 0.015)] = 0.025 \cdot 0.3^2 \Rightarrow H_2 = 0.03 \text{ m}$$

Variable	L	H_2	H_1	a	b
Value (mm)	300	30	25	20	20

Table 3. 3 Irregular slab dimensions

3.2. Loads

Once the elastomeric material is defined, the next step will be defining the loads applied upon the element. On the one hand, there are gravity forces due to the metallic plate and the own weights of the elastomeric material. On the other hand, there are the applied pressures that they could be static or dynamic.

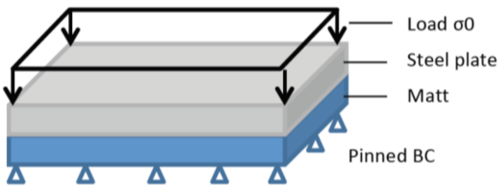


Table 3. 4 Loading

These loads have been applied at different steps:

- Step 0: The boundary condition are defined.
- Step 1: Gravity forces.

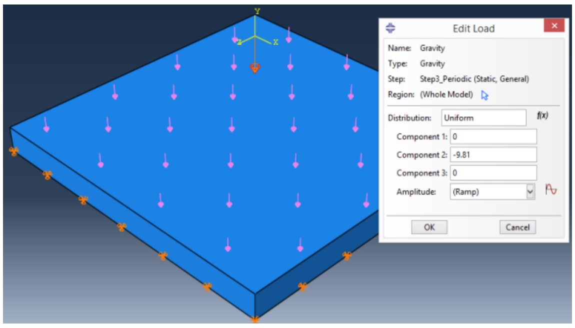


Table 3. 5 Gravity forces

- Step 2: Static pressure.

$$\sigma_s = 20000\ \frac{N}{m^2}$$

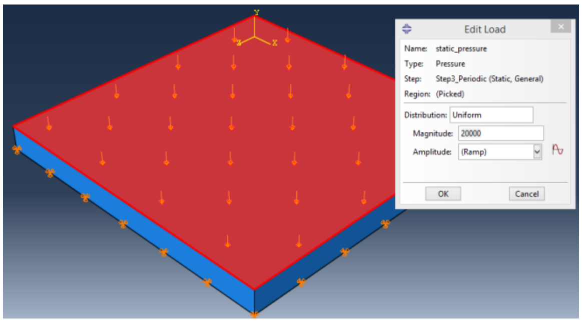


Table 3. 6 Static pressure

- Step 3: Dynamic pressure (periodic). The amplitude of the pressure varies harmonically with amplitude and frequency.

$$\sigma_d = A\ \sin(2\pi f t) = 1.4 \cdot 10000 \sin(2\pi 5 t) = 14000 \sin(31.416t) \frac{N}{m^2}$$

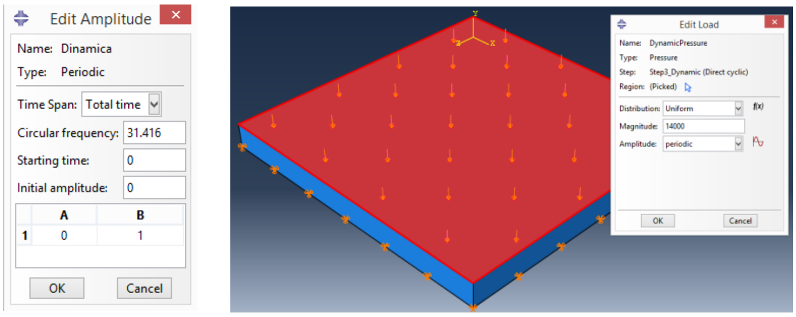


Table 3. 7 Dynamic pressure

3.3. Meshing

The size of the element used to mesh the elastomeric material and the elastic material are different. The size of the element of the elastomeric mats is smaller than the size of the steel plate. The purpose of this report is analyzing the displacements in the elastomeric mats, for this reason it needs a more detailed mesh. The metallic plate is only used to push uniformly the elastomeric material, what happens in the metallic plate left outside this study. As it is used as transmission element, it is not worth wasting computational time on it.

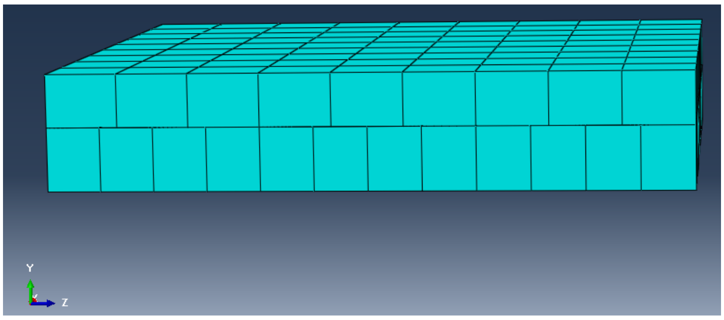


Table 3. 8 Meshing

4. Results and discussions

4.1. Tie constraint/Hard contact model

The elastomeric mat and the steel plate need to be connected to each other in some way, and this connection could be modeled in different ways.

- Tie constraint: This constraint tie the elastomeric surface to the surface of the steel plate, they are fully constrained. The constraint prevents slave nodes from separating or sliding to the master surface. Each node on the slave surface is constrained to have the same motion as the point on the master surface which it is closest. This means that the translational and rotational degrees of freedom are constrained. Slave nodes are adjusted so that they lie exactly on the master (=undeformed) surface.
  This constraint is steadier and would not generate convergence problems. However, there is no limit in the contact formulation on the magnitude of contact pressure that can be transmitted between the surfaces.

- Hard contact: The hard contact relationship minimizes the penetration of the slave surface into the master surface at the constraint locations and does not allow the transfer of tensile stresses. But hard contact model used in Abaqus allows some limited penetrations and some tensile stress across the interface. This parameter is use to model the cohesive aspects of the material.

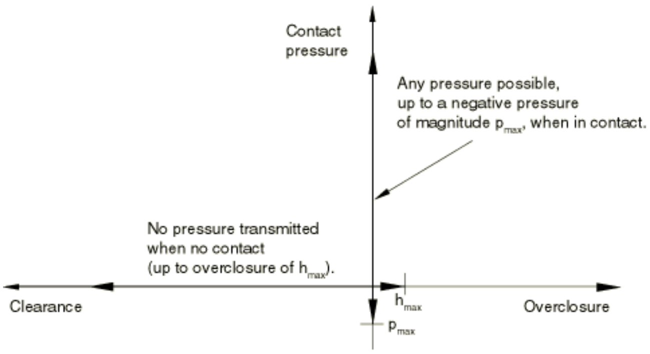


Figure 4. 1 Hard contact

4.2. Static case

At this point, it is going to apply a static load upon the metallic plate and study which design works better. The pressure is applied in two steps (1) gravity pressure (2) static pressure. Note that the volume of the elastomeric material is the same for the three cases.

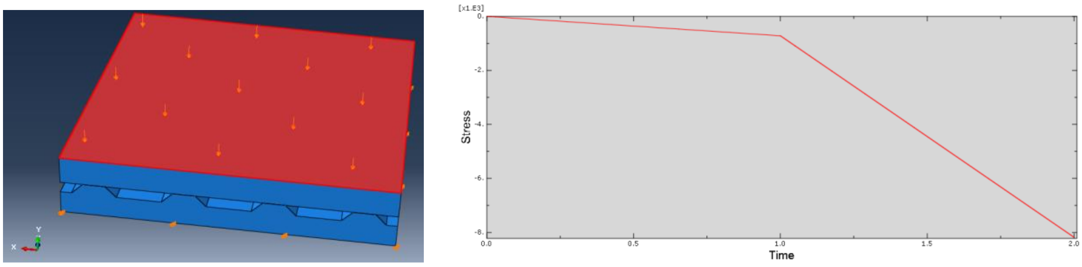


Figure 4. 2 Static analysis

4.2.1. Original design

The constitutive matrix for isotropic material has only two independent material parameters. They are Young´s modulus (E) and Poisson´s ratio (v) and takes the form,

$$D = \frac{E}{(1+\upsilon)(1-2\upsilon)}\begin{bmatrix} 1-\upsilon & \upsilon & \upsilon & 0 & 0 & 0 \\ \upsilon & 1-\upsilon & \upsilon & 0 & 0 & 0 \\ \upsilon & \upsilon & 1-\upsilon & 0 & 0 & 0 \\ 0 & 0 & 0 & \frac{1}{2}(1-2\upsilon) & 0 & 0 \\ 0 & 0 & 0 & 0 & \frac{1}{2}(1-2\upsilon) & 0 \\ 0 & 0 & 0 & 0 & 0 & \frac{1}{2}(1-2\upsilon) \end{bmatrix}$$

Substituting E and v into the above expression gives,

$$\mathrm{D} = 1.3 \cdot 10^9 \begin{bmatrix} 1 & 0 & 0 & 0 & 0 & 0 \\ 0 & 1 & 0 & 0 & 0 & 0 \\ 0 & 0 & 1 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0.5 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0.5 & 0 \\ 0 & 0 & 0 & 0 & 0 & 0.5 \end{bmatrix} = \begin{bmatrix} 1.3 & 0 & 0 & 0 & 0 & 0 \\ 0 & 1.3 & 0 & 0 & 0 & 0 \\ 0 & 0 & 1.3 & 0 & 0 & 0 \\ 0 & 0 & 0 & 0.65 & 0 & 0 \\ 0 & 0 & 0 & 0 & 0.65 & 0 \\ 0 & 0 & 0 & 0 & 0 & 0.65 \end{bmatrix} 10^9$$

The analytical stiffness matrix must be very similar to the one determined by the FEM program. It couldn´t be obtained directly from the informatic program, thus it is going to define by the relation between the stresses (σ) and displacements (ε) of any element of the elastomeric material at the end of the step, which could be obtained directly from the outputs given by Abaqus.

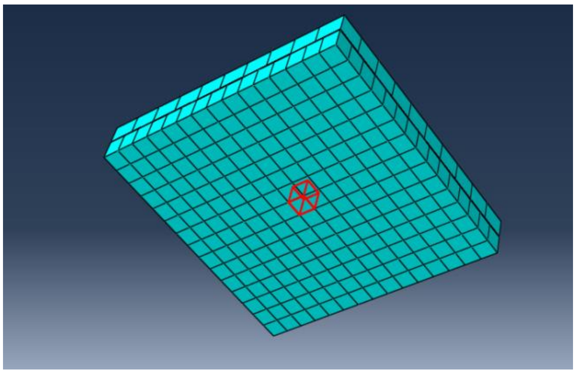


Figure 4. 3 Studied element

$$\{\sigma\} = [D]\{\varepsilon\}$$

$$\begin{Bmatrix} 18.559 \\ 18.559 \\ -21963.7 \\ 2.6184 \cdot 10^{-12} \\ 1.2835 \cdot 10^{-5} \\ 1.2835 \cdot 10^{-5} \end{Bmatrix} = \begin{bmatrix} D_{11} & 0 & 0 & 0 & 0 & 0 \\ 0 & D_{22} & 0 & 0 & 0 & 0 \\ 0 & 0 & D_{33} & 0 & 0 & 0 \\ 0 & 0 & 0 & D_{44} & 0 & 0 \\ 0 & 0 & 0 & 0 & D_{55} & 0 \\ 0 & 0 & 0 & 0 & 0 & D_{66} \end{bmatrix} \begin{Bmatrix} 1.4507 \cdot 10^{-8} \\ 1.4507 \cdot 10^{-8} \\ -1.6895 \cdot 10^{-5} \\ 4.0283 \cdot 10^{-21} \\ 1.9746 \cdot 10^{-14} \\ 1.9746 \cdot 10^{-14} \end{Bmatrix}$$

$$\begin{Bmatrix} D_{11} \\ D_{22} \\ D_{33} \\ D_{44} \\ D_{55} \\ D_{66} \end{Bmatrix} = \begin{Bmatrix} 1299993107 \\ 1299993107 \\ 1300004143 \\ 649999752 \\ 649999747 \\ 649999747 \end{Bmatrix}$$

The analytical values and the values obtained from modeling it in FEM are very close, which confirm that the inputs are well introduced in the program. The marginal difference is due to the analytical method provides the exact solution and the solutions obtained from the FEM are approximated.

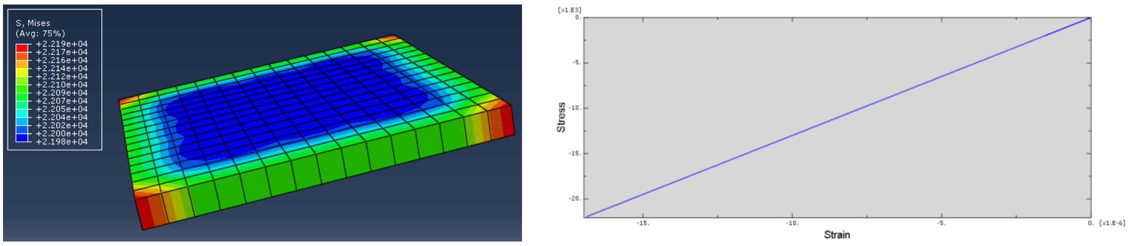


Figure 4. 4 Von Misses and stress-strain curve

4.2.2. Design A

The stiffness matrix is calculated using the strains and stresses of the element showed in the following figure and it is quite similar to the original section.

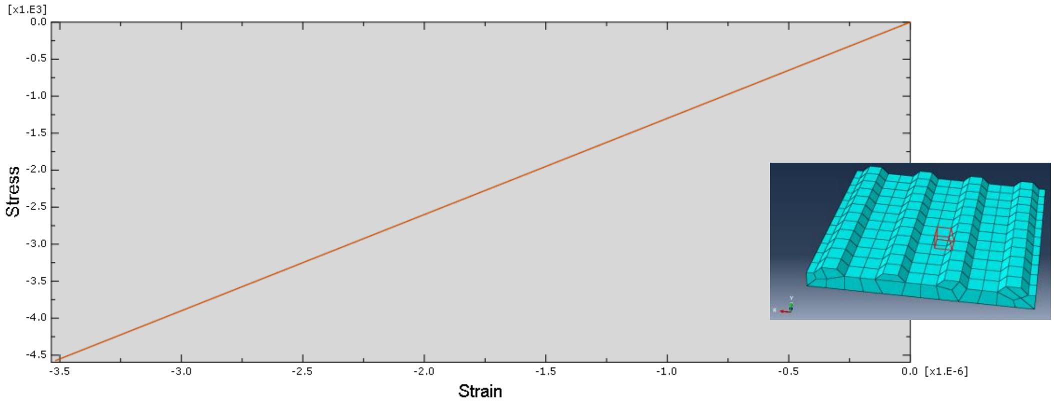


Figure 4. 5 Stress-strain curve

$$\begin{Bmatrix} 3241.26 \\ -4572.89 \\ 2.60848 \\ 7943.62 \\ 2.03447 \\ -1.73582 \end{Bmatrix} = \begin{bmatrix} D_{11} & 0 & 0 & 0 & 0 & 0 \\ 0 & D_{22} & 0 & 0 & 0 & 0 \\ 0 & 0 & D_{33} & 0 & 0 & 0 \\ 0 & 0 & 0 & D_{44} & 0 & 0 \\ 0 & 0 & 0 & 0 & D_{55} & 0 \\ 0 & 0 & 0 & 0 & 0 & D_{66} \end{bmatrix} \begin{Bmatrix} 2.4933 \cdot 10^{-6} \\ -3.51761 \cdot 10^{-6} \\ 2.0065 \cdot 10^{-9} \\ 1.2221 \cdot 10^{-5} \\ 3.13 \cdot 10^{-9} \\ -2.6705 \cdot 10^{-9} \end{Bmatrix}$$

$$\begin{Bmatrix} D_{11} \\ D_{22} \\ D_{33} \\ D_{44} \\ D_{55} \\ D_{66} \end{Bmatrix} = \begin{Bmatrix} 1300003610 \\ 1299999147 \\ 1300014951 \\ 650002864 \\ 650000799 \\ 650000562 \end{Bmatrix}$$

However, the stresses are lower in the major part of the body than in the original design but the worst aspect of this design is that there are some zones, as the flaps (red zone), in specially high demand. In terms of efficiency, the ideal situation is that the elastomer was stressed homogeneously (the entire piece in the same color) to make full use of capacities. For that reason, I think that the original design is better than this option.

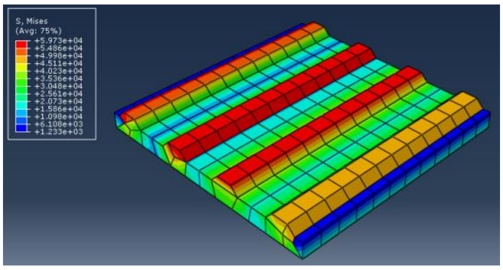


Figure 4. 6 Von Misses

In the following graph is represented the strain-stress curve of the most stressed element, which illustrate numerically the aspect explained in the previous paragraph. It can see that the stresses and strain are higher than in the original design.

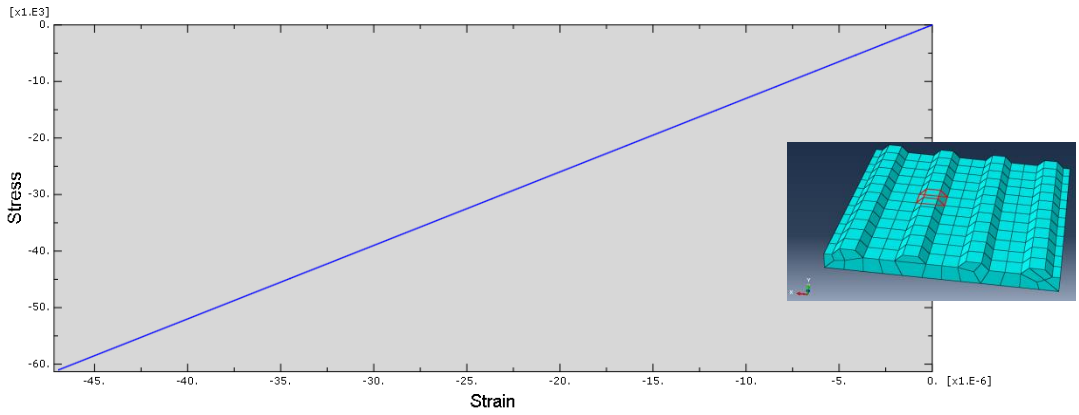


Figure 4. 7 Stress-strain curve

4.2.3. Design B

In this part it is going to do the same analysis as in the previous point and the results would be compared with the others designs.

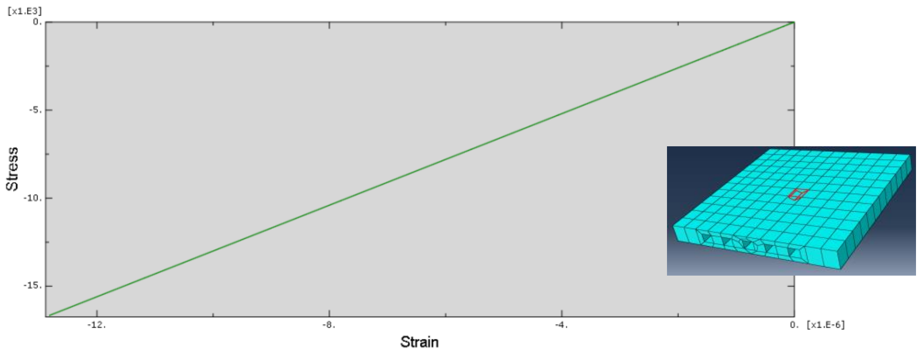


Figure 4. 8 Stress-strain curve

- The stiffness matrix: It is practically the same as in the previous cases.

$$\begin{Bmatrix} -563.818 \\ -45457.6 \\ 27.3785 \\ -15641.6 \\ 3.98237 \\ 11.211 \end{Bmatrix} = \begin{bmatrix} D_{11} & 0 & 0 & 0 & 0 & 0 \\ 0 & D_{22} & 0 & 0 & 0 & 0 \\ 0 & 0 & D_{33} & 0 & 0 & 0 \\ 0 & 0 & 0 & D_{44} & 0 & 0 \\ 0 & 0 & 0 & 0 & D_{55} & 0 \\ 0 & 0 & 0 & 0 & 0 & D_{66} \end{bmatrix} \begin{Bmatrix} -4.3371 \cdot 10^{-7} \\ -3.4967 \cdot 10^{-5} \\ 2.1060 \cdot 10^{-8} \\ -2.4064 \cdot 10^{-5} \\ 6.1267 \cdot 10^{-9} \\ 1.7249 \cdot 10^{-8} \end{Bmatrix}$$

$$\begin{Bmatrix} D_{11} \\ D_{22} \\ D_{33} \\ D_{44} \\ D_{55} \\ D_{66} \end{Bmatrix} = \begin{Bmatrix} 1300000461 \\ 1299999428 \\ 1300011396 \\ 650000000 \\ 650002448 \\ 649958258 \end{Bmatrix}$$

- Von Misses stresses: As it can be seen in the following figures, the stresses are distributed uniformly, except in the center part of the body. This design may be a possible alternative because it harnesses the capacities of the material.

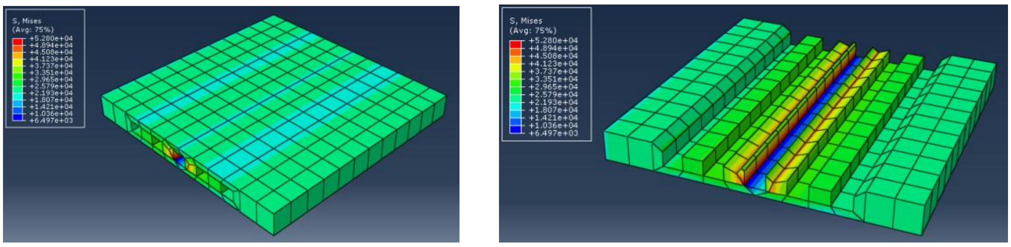


Figure 4. 9 Von Misses

The stress-strain curve confirm that the values are higher than the original design and lower thand the design A.

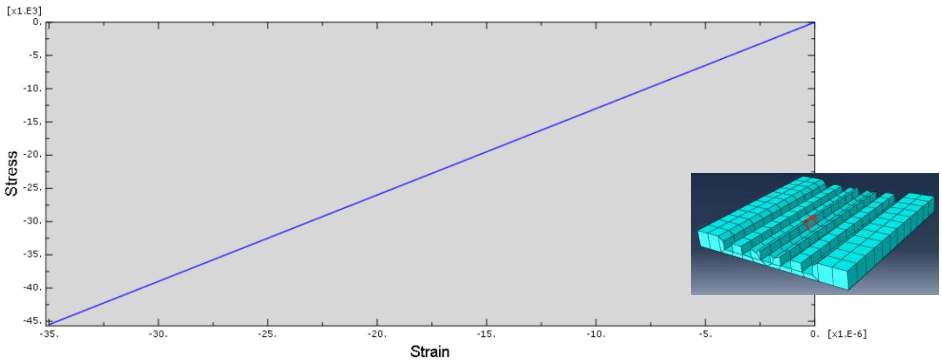
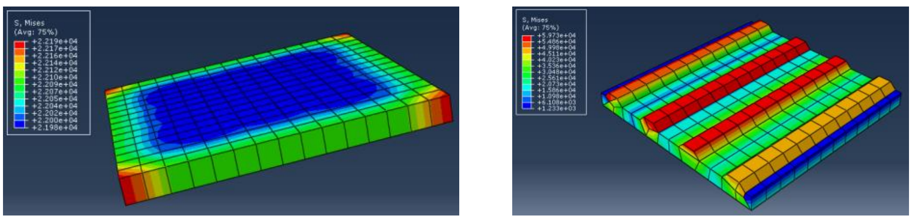


Figure 4. 10 Stress-strain curve

4.2.4. Conclusion

The displacements are different in all the cases. The displacements of the original design are smaller than in the design B, but this comparison is not very clear if we checked with the Design B. In the main body the displacements are smaller but on the flags, where the stresses are much higher, the displacements are the biggest ones.

The stresses are distributed equally throughout the original section and section B, thus the efficiency of the material is maximized. However, the design A has some areas very stressed (flags) and it is mostly under-used.



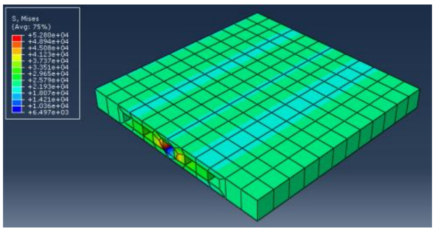


Figure 4. 11 Original design, Design A and Design B

4.3. Dynamic analysis

The dynamic analysis has been done neglecting the metallic plate and applying the load directly to the massive material. The loading operation has been done in three steps (1) Gravity forces (2) Static pressure (3) Dynamic pressure.

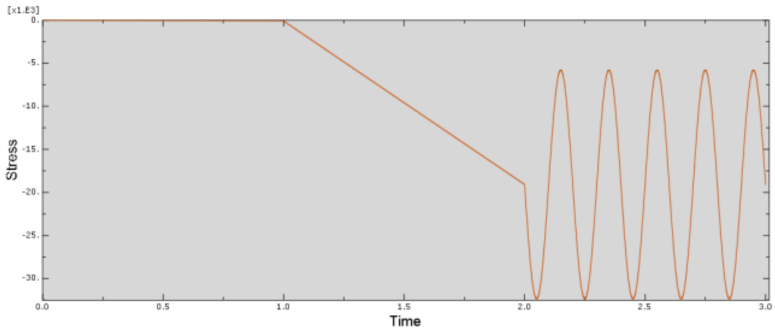


Figure 4. 12 Applied stress

4.3.1. Original design

The displacement of a node of the middle zone is shown in the following figure. It can be seen that the displacement increase in the same direction in the first two steps. In the last step, it starts oscillating because of the dynamic pressure. The displacements are between 0.1-0.65·10^{-6} meters during this period.

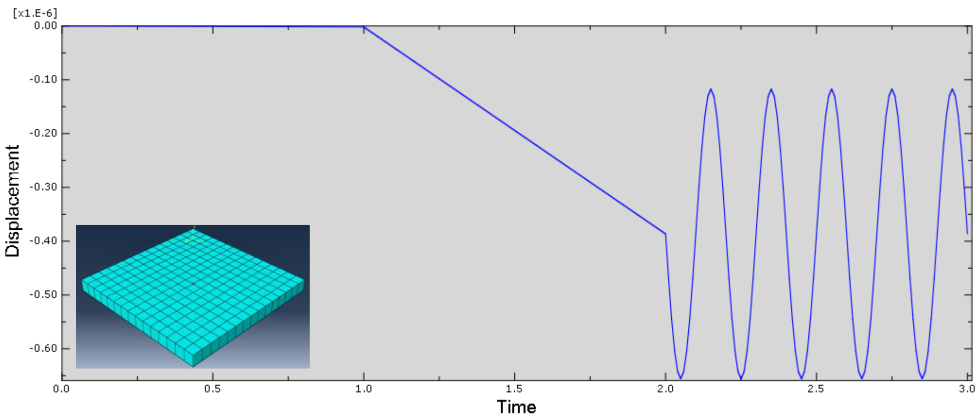


Figure 4. 13 Displacement

The stress-strain curve for any element is linear because they are in the elastic range. In this case the stress distribution is uniform and there are not going to be any difference among them.

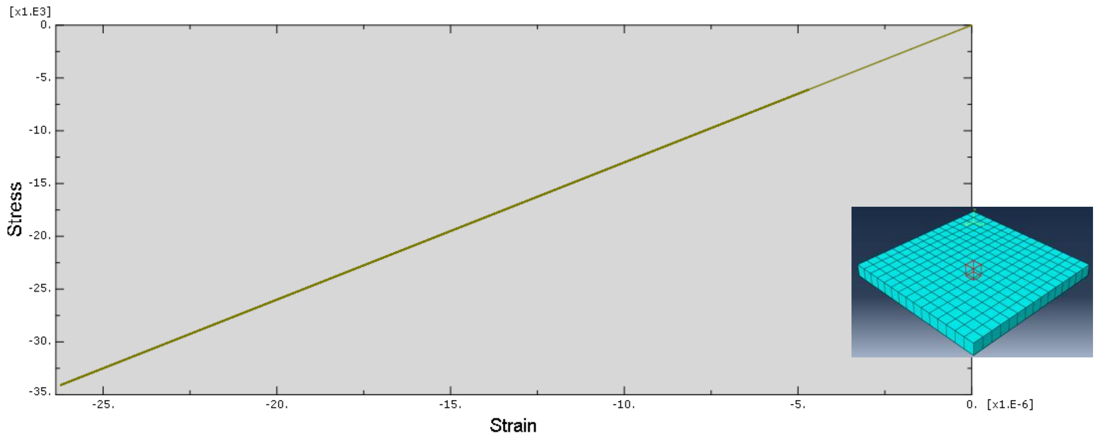


Figure 4. 14 Stress-strain curve

4.3.2. Design A

The differential aspect of this design is its irregular loading-surface. Despite the small size of the flaps, the pressure could be applied upon the flaps or the whole surface.

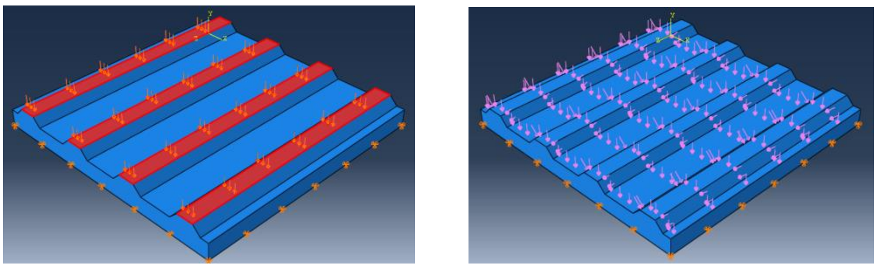


Figure 4. 15 Pressure application options

a) The pressure is only applied in the horizontal surface of the flaps. This surface is smaller than the original one, so the value of the pressure increases.

$$\sigma_{sta} = \frac{20000 \cdot 0.3^2}{4(0.02 \cdot 0.3)} = 75000 \frac{N}{m^2}$$
$$\sigma_{dyn} = \frac{14000 \cdot 0.3^2}{4(0.02 \cdot 0.3)} = 52500 \frac{N}{m^2}$$

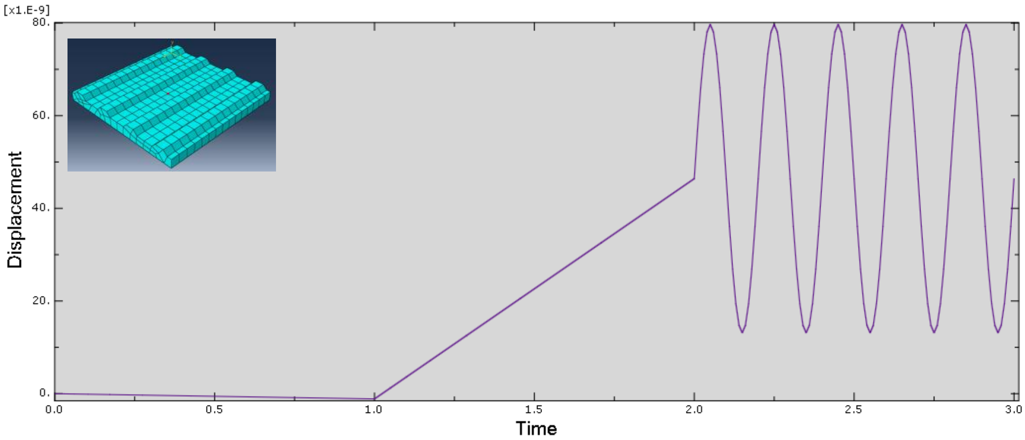


Figure 4. 16 The displacement when the pressure is applied upon the flags

The displacement of the middle point is totally different comparing with the other case. The displacement is in the opposite direction of the pressure because the flaps sinks on the body and push up the studied node. This performance does not seem very realistic, thus, it is dismissed.

b) The pressure is applied upon the whole surface and due to the irregularity of the surface two points have been studied, one in the flag and another one between the flags. The resultant displacement in the vertical direction are what they were expected, in the direction of the pressure. Their values are between 0.15-0.8 $\cdot 10^{-6}$ meters.

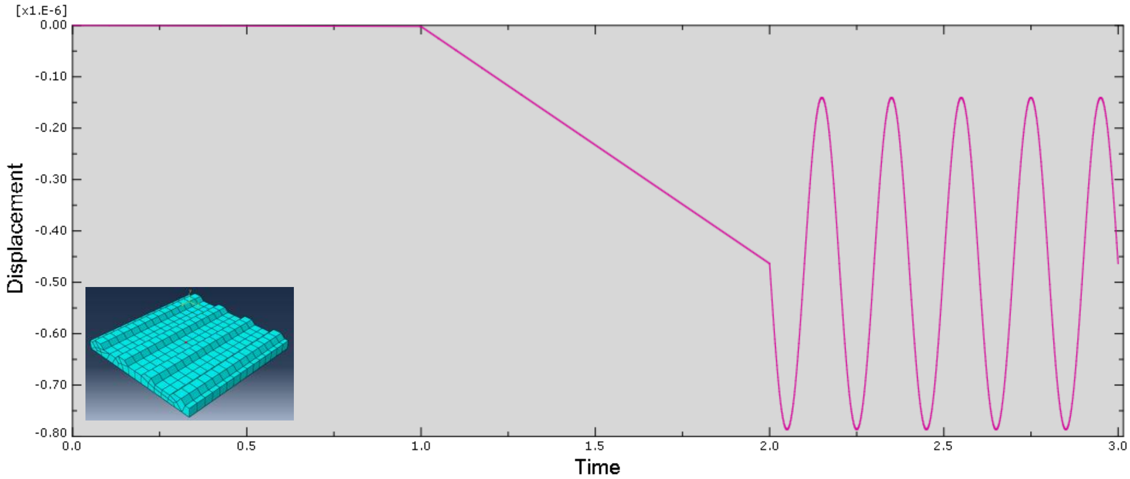


Figure 4. 17 Displacement on the middle point when the load is applied upon the whole surface

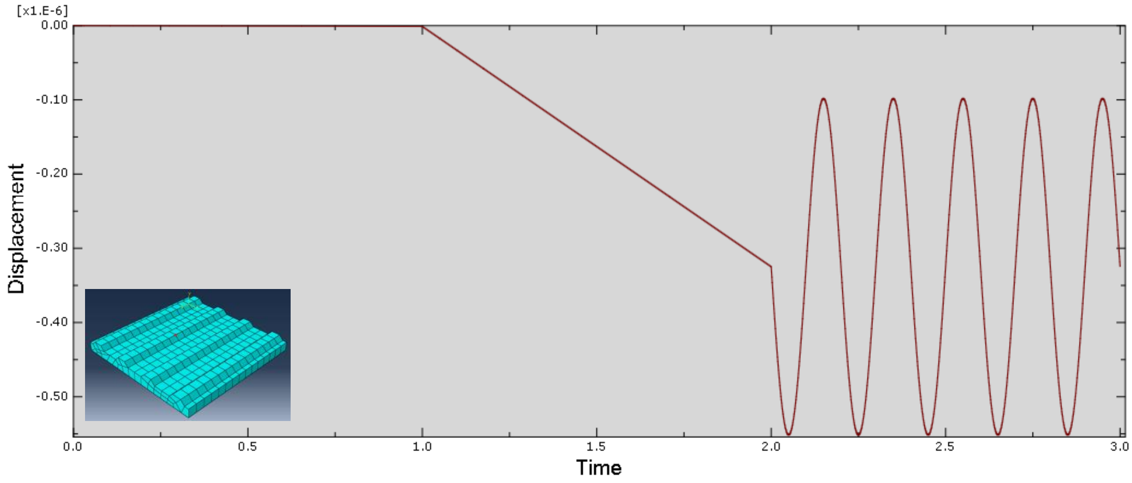


Figure 4. 18 Displacement on the flag when the load is applied upon the whole surface

At last, the strain-stress curve is drawn. This graph in the elastic range usually is linear and doesn´t give much information in this case.

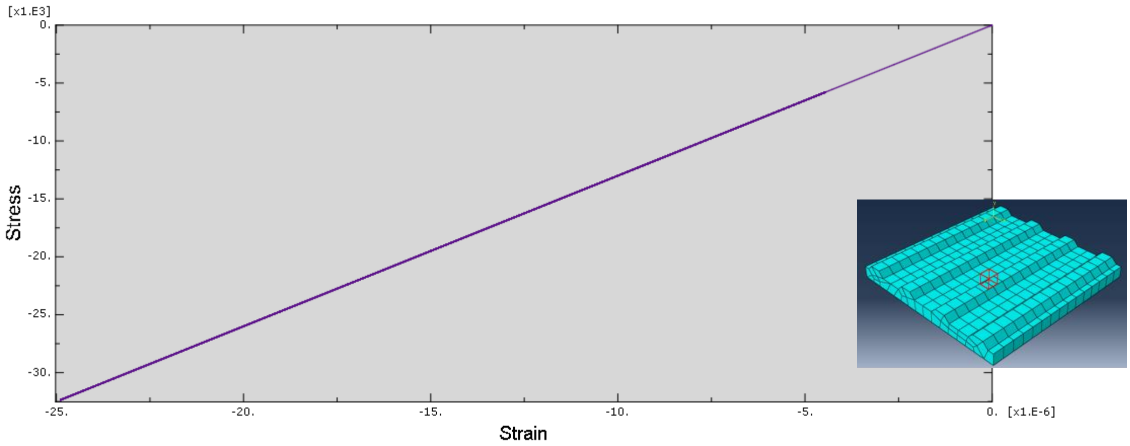


Figure 4. 19 Stress-strain curve of the element located in middle zone

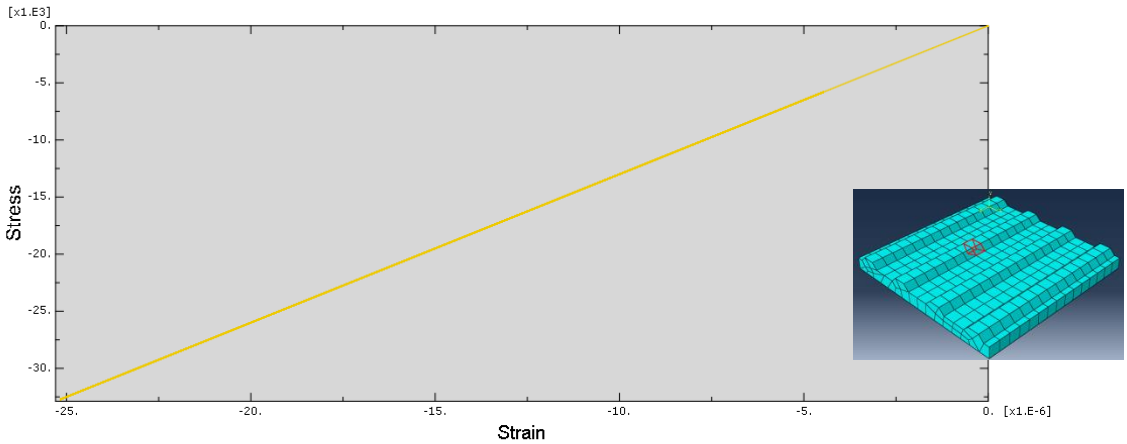


Figure 4. 20 Stress-strain curve of the element located in the flag

4.3.3. Design B

The uniformity of the piece allows that all the elements of the loading-surface works in the same way. Most points of it are going to suffer a similar displacement. The maximum displacement would be in the center of the plate and it is represented in the following figure. The displacement are ranged from 0.2 to $1.2 \cdot 10^{-6}$.

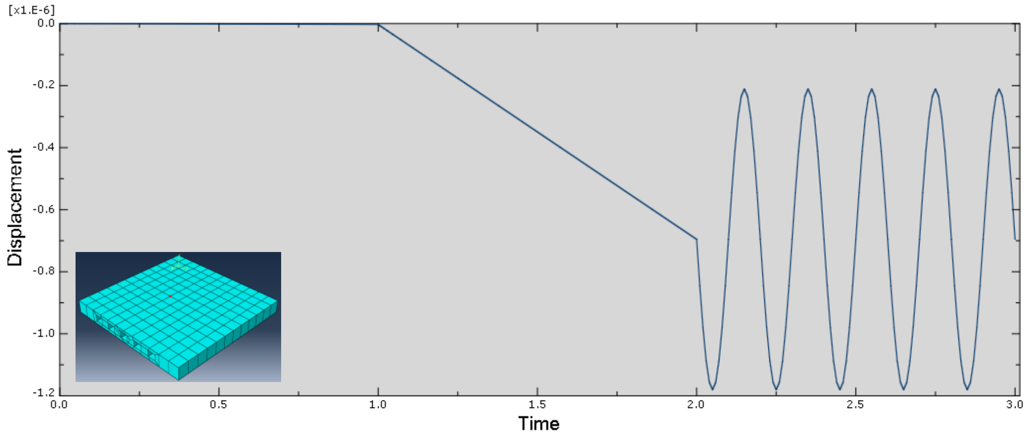


Figure 4. 21 Displacement

Its characteristic geometry generated overstressed elements in the inner zone. For this reason, it is interesting to analyze the stress distribution between the holes. The next graphs show clearly that the inner element withstands greater tensions and strains than element located in the surface.

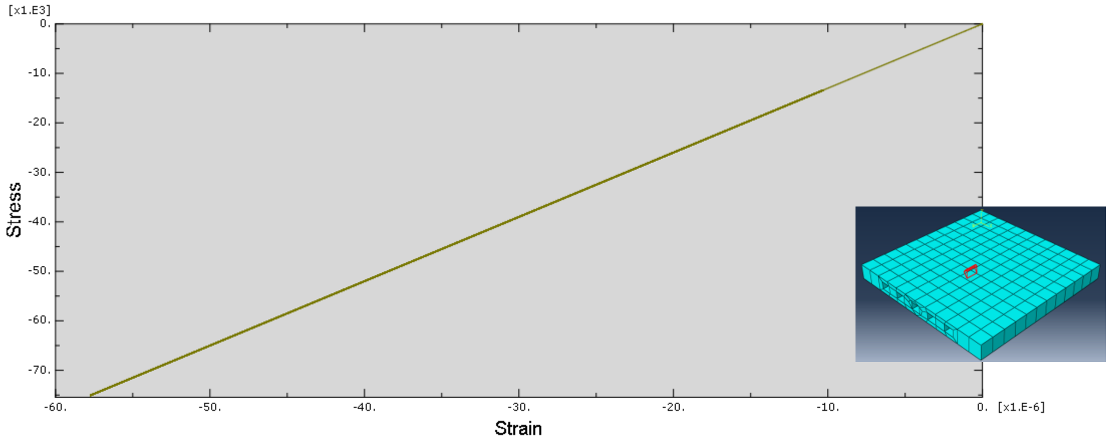


Figure 4. 22 Stress-strain curve

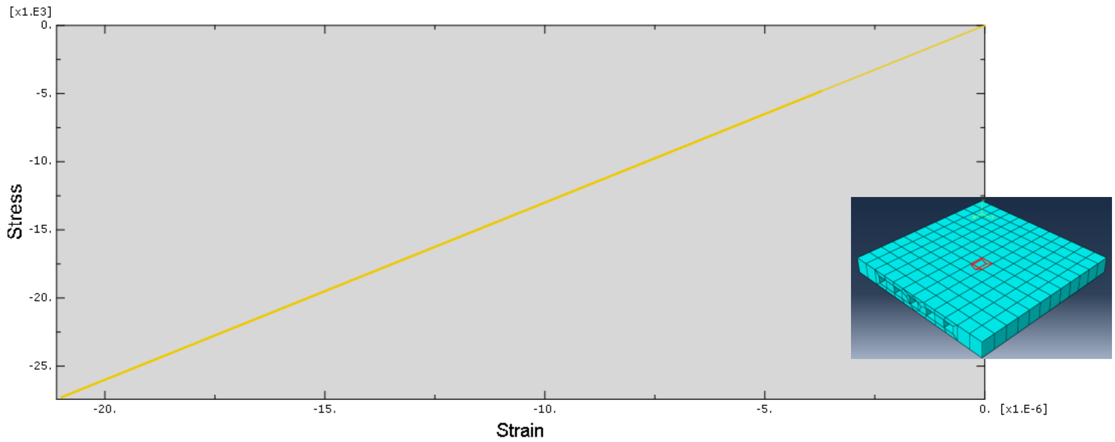


Figure 4. 23 Stress-strain curve

4.3.4. Conclusion

The dynamic analysis shows that the biggest displacements are in the Design B and in the other two options are similar between them. The stress pattern continues as in the static case. In the following table are the values of the displacement in the dynamic part:

Original design	Design A	Design B
$0.1\text{-}0.65\cdot10^{-6}$	$0.15\text{-}0.8\cdot10^{-6}$ $0.1\text{-}0.55\cdot10^{-6}$	$0.2\text{-}1.2\cdot10^{-6}$

Table 4. 1 Pressure amplitude

The Von Misses stress pattern shows where are located the most demanding elements. In which usually the biggest displacement would take place.

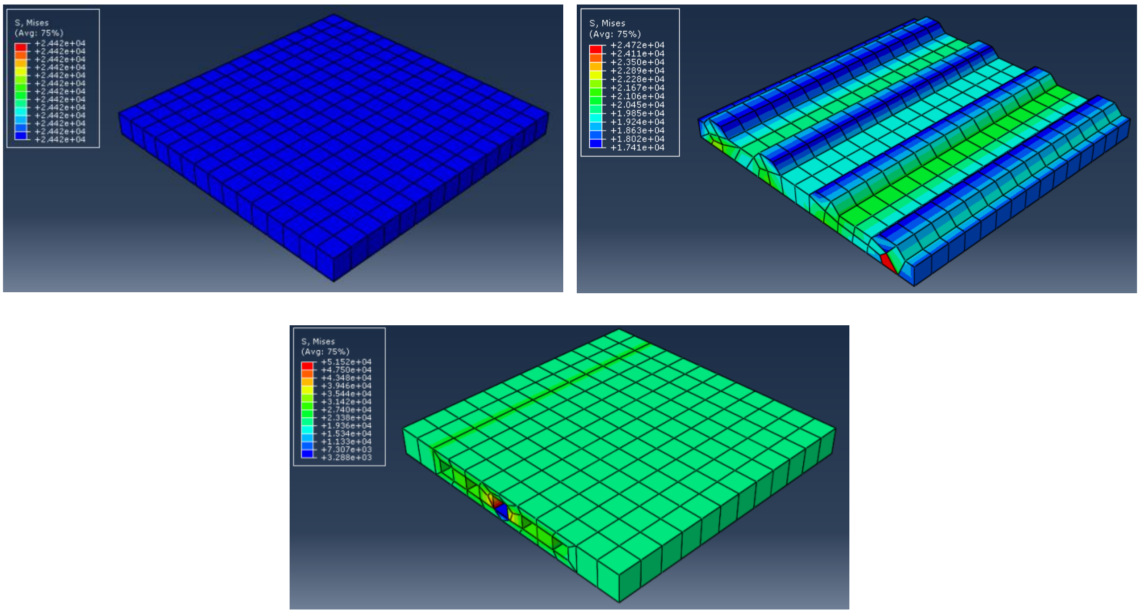


Table 4. 2 Von Misses

5. Conclusions and future work

The stress pattern showed in the static case it could be useful for the dynamic case. It is helpful to see how the stresses are distributed through the element. This aspect determines where are located the most displaced points. An interesting aspect to be considered in the analysis of the stress pattern is that most of the elements must work in the same range of pressure. Regular surfaces allow to harness and optimize the use the capacities of the material.

However, the dynamic displacements do not match with the static ones. The displacement in the loading surface of the option B are always the great ones and the other two options are closer. The difference between the displacement of the original design and the design A is greater in the static case than in the dynamic analysis. In this last analysis, the displacements are very similar for these two cases.

For reasons of simplicity, the best geometry is the solid plate. Using complex geometry could provide best results but it can complicate the installation and reduce its durability. It is interesting to use the maximum surface of contact to decrease the amplitude of the applied pressure.

Finally, the elastomeric materials must continue developing because the trains to be used on the rail system will continue increasing their speed and weight, which will generate dynamic stresses with higher amplitude. The new elastomeric material will be able to dissipate more vibration energy, thus, it is going to need materials with a higher resilience. The resilience or the capacity to dissipate energy is linked with the area closed by the strain-deformation curve. Therefore, the new material would need to develop a greater area while retains some rigidity to support the vertical loads. Another important property is the durability of the material. It is interesting extend the durability, while it maintains the properties. In many bridge it can see that the elastomeric slabs are severely damaged and they do not fulfil their initial function.